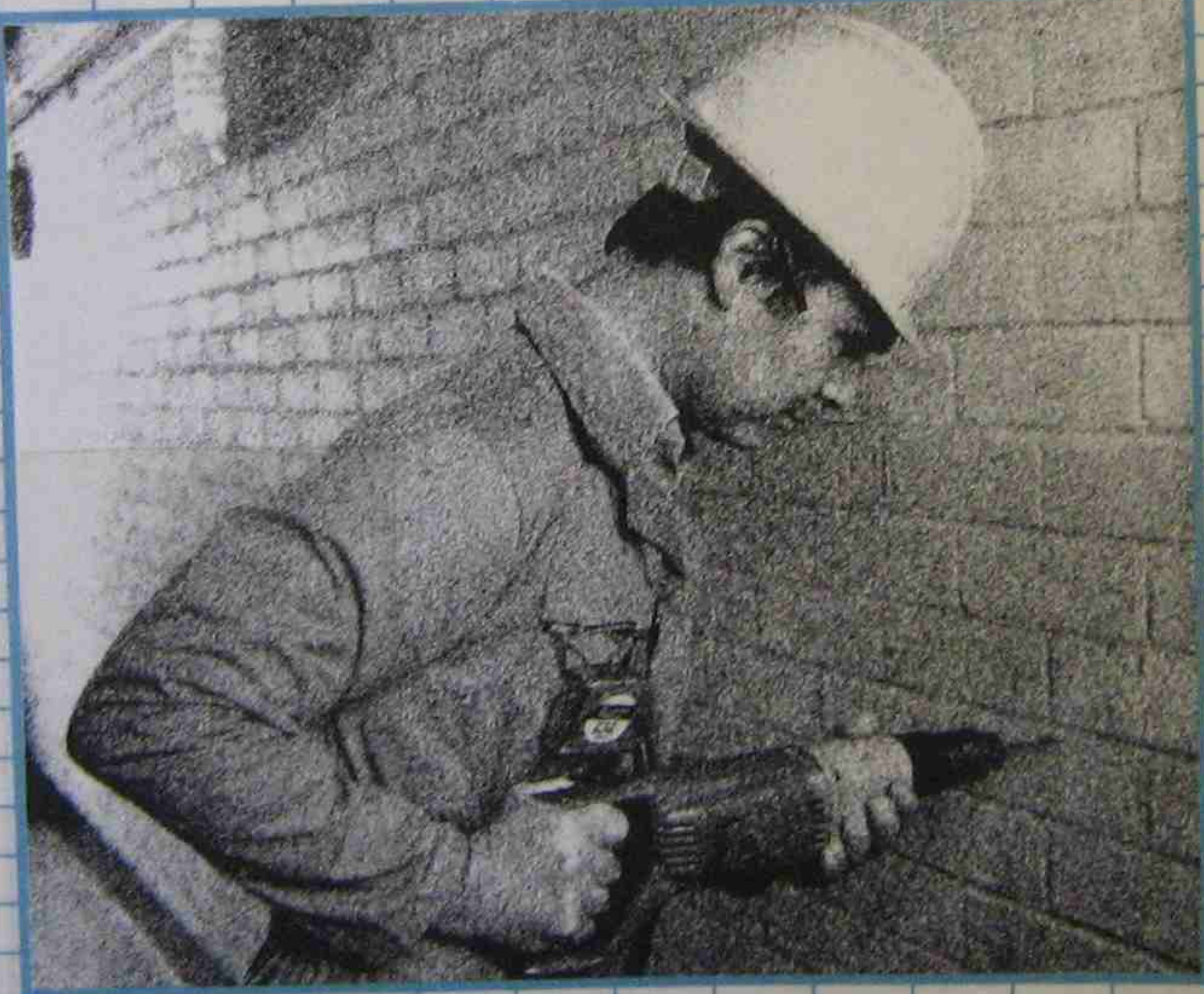
Site Practice Series

Fixings, Fasteners and Adhesives



P Marsh

SITE PRACTICE SERIES

General editors: Harold Lansdell, FCIOB, FCIArb, and Win Lansdell, BA

Timber frame housing – *Jim Burchell*
Security on site – *Len Earnshaw*
Site safety – *Jim Laney*
Fixings, fasteners and adhesives – *Paul Marsh*
Site engineering – *Roy Murphy*
Glazing – *Stanley Thompson*
Steel reinforcement – *Tony Trevorrow*

Books to be published in the Series
Making and placing concrete – *Edwin Martin Baker*
Site relations – *Tom Gallagher*
Careers in the building industry – *Chris and Lynne March*

Fixings, fasteners and adhesives

PAUL MARSH

Construction Press
LONDON AND NEW YORK

Construction Press
An imprint of:
Longman Group Limited
Longman House, Burnt Mill, Harlow
Essex CM20 2JE, England
Associated companies throughout the world

Published in the United States of America
by Longman Inc., New York

First published 1984

British Library Cataloguing in Publication Data
Marsh, Paul, 1931–
Fixings, fasteners and adhesives. — (Site practice series)
1. Fasteners
I. Title II. Series
693 TH153

ISBN 0-86095-035-2

Library of Congress Cataloging in Publication Data
Marsh, Paul Hugh.
Fixings, fasteners, and adhesives.

(Site practice series)
Includes index.
1. Joints (Engineering) 2. Fasteners.
3. Adhesives. I. Title. II. Series
TA660.J64M37 1984 621.8'8 83-7465
ISBN 0-80095-035-2

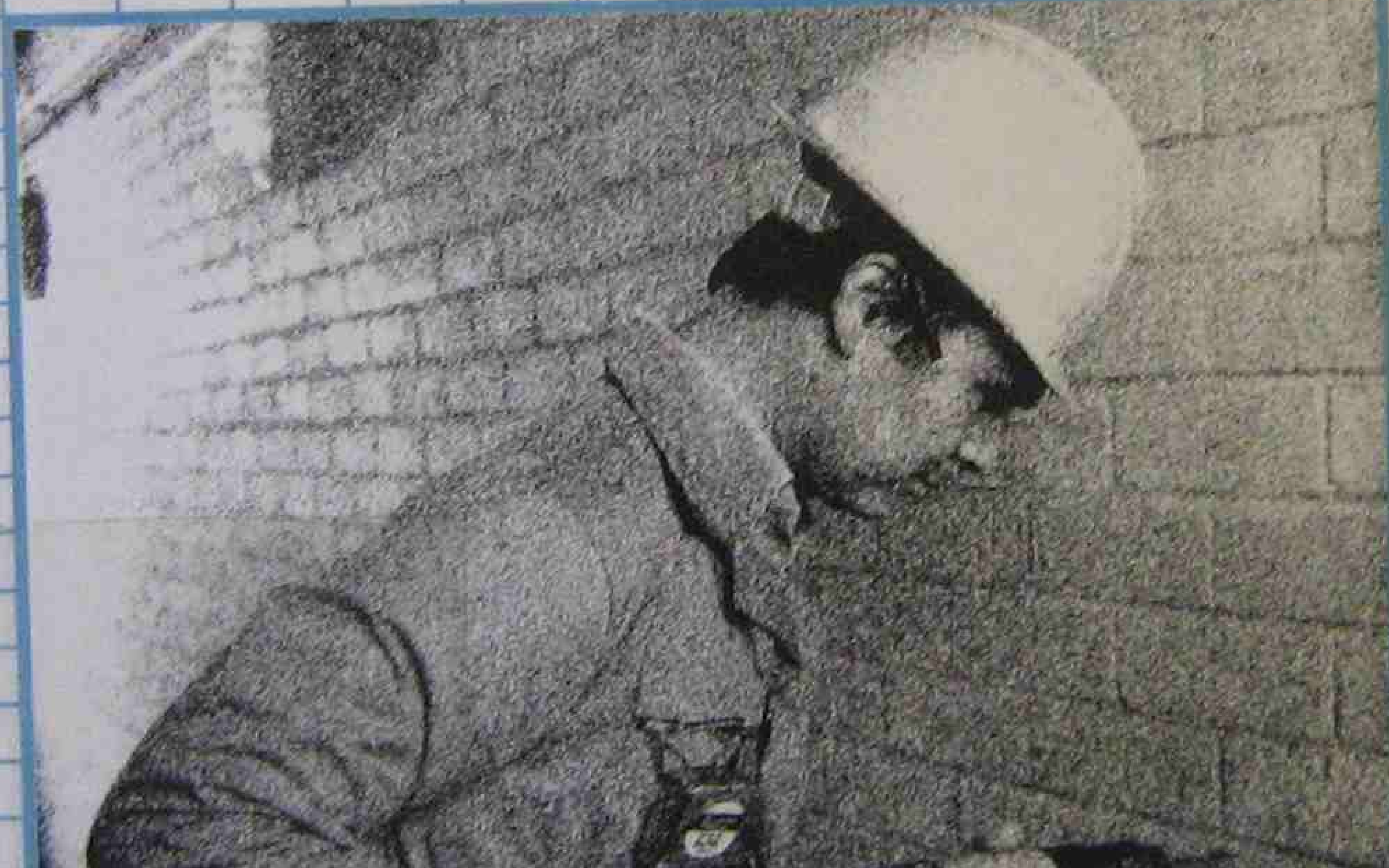
Set in 10/12 pt Lintron 202 Bembo
Printed in Hong Kong by
Commonwealth Printing Press Ltd

Contents

	Preface	vii
1	The importance of fixings	1
2	Types of fixings	3
3	Mechanical fixings in mass walling bases	12
	Section A Drilled-for fixings	13
	Section B Base-deforming fixings	44
	Section C In situ fixings	52
	Section D Fixing accessories	61
	Section E Background to fixing performance in mass walling	63
4	Mechanical fixings in timber bases	70
	Section A Base-deforming fixings	70
	Section B Through-fixings	92
	Section C Fixing accessories	100
	Section D Background	104
5	Mechanical fixings in metal bases	112
	Section A Heavy fixings	112
	Section B Light fixings	122
	Section C Background	141
6	Mechanical fixings to thin-walled cellular bases (and thin sheet materials)	146
	Section A Expanding cavity fixings	147
	Section B Rivet cavity fixings	153
	Section C Toggle fixings	154
	Section D Umbrella cavity fixings	158
7	Adhesives	161
	Appendix 1 List of proprietary fixings in base material classifications	182
	Appendix 2 Glossary	188
	Appendix 3 Comparative diameters of various devices	193
	Index	197

Site Practice Series

Fixings, Fasteners and Adhesives



P Marsh

SITE PRACTICE SERIES

General editors: Harold Lansdell, FCIOB, FCIArb, and Win Lansdell, BA

Timber frame housing – *Jim Burchell*
Security on site – *Len Earnshaw*
Site safety – *Jim Laney*
Fixings, fasteners and adhesives – *Paul Marsh*
Site engineering – *Roy Murphy*
Glazing – *Stanley Thompson*
Steel reinforcement – *Tony Trevorrow*

Books to be published in the Series
Making and placing concrete – *Edwin Martin Baker*
Site relations – *Tom Gallagher*
Careers in the building industry – *Chris and Lynne March*

Fixings, fasteners and adhesives

PAUL MARSH

Construction Press
LONDON AND NEW YORK

Construction Press
An imprint of:
Longman Group Limited
Longman House, Burnt Mill, Harlow
Essex CM20 2JE, England
Associated companies throughout the world

Published in the United States of America
by Longman Inc., New York

First published 1984

British Library Cataloguing in Publication Data
Marsh, Paul, 1931–
Fixings, fasteners and adhesives. — (Site practice series)
1. Fasteners
I. Title II. Series
693 TH153

ISBN 0-86095-035-2

Library of Congress Cataloging in Publication Data
Marsh, Paul Hugh.
Fixings, fasteners, and adhesives.

(Site practice series)
Includes index.
1. Joints (Engineering) 2. Fasteners.
3. Adhesives. I. Title. II. Series
TA660.J64M37 1984 621.8'8 83-7465
ISBN 0-80095-035-2

Set in 10/12 pt Lintron 202 Bembo
Printed in Hong Kong by
Commonwealth Printing Press Ltd

Contents

	Preface	vii
1	The importance of fixings	1
2	Types of fixings	3
3	Mechanical fixings in mass walling bases	12
	Section A Drilled-for fixings	13
	Section B Base-deforming fixings	44
	Section C In situ fixings	52
	Section D Fixing accessories	61
	Section E Background to fixing performance in mass walling	63
4	Mechanical fixings in timber bases	70
	Section A Base-deforming fixings	70
	Section B Through-fixings	92
	Section C Fixing accessories	100
	Section D Background	104
5	Mechanical fixings in metal bases	112
	Section A Heavy fixings	112
	Section B Light fixings	122
	Section C Background	141
6	Mechanical fixings to thin-walled cellular bases (and thin sheet materials)	146
	Section A Expanding cavity fixings	147
	Section B Rivet cavity fixings	153
	Section C Toggle fixings	154
	Section D Umbrella cavity fixings	158
7	Adhesives	161
	Appendix 1 List of proprietary fixings in base material classifications	182
	Appendix 2 Glossary	188
	Appendix 3 Comparative diameters of various devices	193
	Index	197

Preface

As the number and importance of fixings and fasteners have increased in recent years, so has the need to ensure their correct use on site. Faced with a bewildering multitude of devices and adhesives, the first problem is, which fixing to use; the second, how to use it so that it achieves its maximum effect.

This book is intended to help the practical man with both of these problems. It will not attempt to advise on the design of heavy duty structural connections – this work is the domain of the engineer. It will, however, give advice on the installation of the device specified by the engineer and, where a choice exists, explain the different types of fixings and fasteners available, and which should be used where.

Fixings will be considered within five major categories – the first four consisting of *mechanical fixings* (all fixings except adhesives), the fifth consisting of adhesives (*chemical fixings*).

Mechanical fixings are divided into four categories depending on the type of base material into which they are making a fixing (mass walling, timber, metal or cellular materials). A more detailed explanation of these divisions will appear in the next chapter when we discuss types of fixings. It is sufficient here to say that the reason for this choice of categories is because the character of the base material exerts considerable influence over the type of fixing which can be used, and therefore a classification by base material makes good sense.

Each category of fixings will be dealt with in a similar way. The chapter will start with a brief description of the problem of making fixings to the particular base material; then each fixing device or fastener will be dealt with individually. It will be described, its uses or applications discussed, and the method of its setting or installation given. These *setting instructions* represent the minimum amount of information necessary in order to use the fixing correctly. For quick, practical assistance, this is the only part of the chapter that need be read, along with the general advice contained at the beginning of each chapter and a glance at the appropriate illustrations. For those who are interested in finding out more, there is a section at the end of each chapter giving additional background information concerning each category of fixing. This will help to give a more broad understanding of the fixing being considered and may cast extra light on some of the advice given in the previous parts of the chapter.

In Appendix 1 appears a schedule of proprietary devices divided into the same categories, along with manufacturers' names and addresses. Manufacturers will always be pleased to give advice on the use of their products.

One word of warning: as the complexity of fixing devices grows day by day, more sophisticated devices are being added to the product ranges. The advice given in this book is applicable to the type of device concerned, but may not be wholly correct for newer or more specialized devices. It is always advisable to read and follow the manufacturer's instructions when using a new mechanical fixing device or adhesive of which you have no previous experience.

1

The importance of fixings

because construction materials have increased in efficiency + strength

The importance of fixings in the building industry has grown immeasurably over the last fifty years. As construction materials have generally increased in efficiency and strength, so structures have tended to become lighter, made up of materials with a higher strength/weight ratio. However, as a direct result of their lightness, structures have become more vulnerable to the effects of wind pressure, accidental impact damage and similar misfortunes. No longer can the sheer weight of a building, combined with the low-grade adhesion that mortar provided, be considered sufficient to hold the parts of the structure together. High-strength mechanical and adhesive fixings have had to come to the rescue, making good the structural shortfall that reduced weight brought in its wake.

What is more, as structural components have become lighter, they have also tended to become smaller, presenting less area of contact between the parts of the joint for whatever fixings may be necessary to hold them together. So twelve nails in a joint for safety's sake where practically four would do, is no longer physically possible – even if it were economically permissible today to use more fixings than are really necessary. For this reason alone fixings have often to be fewer in number, and therefore each has to be of greater strength and of thoroughly predictable performance.

All this has resulted in the growth of a new breed of high-performance fixings with assured engineering achievement. Also, as structural elements have become thinner, the character of fixings has had to change, leading to ever more sophisticated fasteners and adhesives which can make fixings to thin or cellular materials, like honeycomb core partitions and doors – fixings which would not have been possible a few years ago.

But it is not only in the design of fasteners and adhesives where the development pressure is on. The strength of the device or glue alone is not enough to determine the strength of the joint. The method and skill of its application or use, too, is a vital factor in the efficiency of the fixing.

It is for this reason that this book will concentrate on the practical techniques to be used in the installation of devices or the use of adhesives so that they can achieve their maximum efficiency. A clumsily-placed fastener, or an adhesive joint closed after the glue has exceeded its open time, can undermine the effectiveness of the joint and (in the case of medium- or heavy-duty fixings) make nonsense of the engineering design of the structure, which will have been calculated in many cases on the expected performance of the fixing or adhesive used with a particular material.

Finally, it is important to remember that even the most expensive fixing forms a very small part of the total cost of a building. A failure of one of these apparently minor devices, however, can cause damage (to life and property) out of all proportion to its cost. Fixings, their design and application, are vital to the performance of the building.

2

Types of fixings

Fixings in the building industry tend to be made between a *base* (usually the most substantial of two elements jointed together and often the first one to be constructed) and a *subsidiary component* (the less substantial of the two elements). Typical examples are:

(a) a fixing between the structural frame of a building (the base) and its claddings (the subsidiary component);
(b) or between a loadbearing wall (the base) and loadbearing components, such as floor joists, wall plates, roof trusses, bearers, etc. or special facing material (the component);
(c) or between a non-loadbearing cellular partition (the base) and a shelf bracket (the component);
(d) or between a wall (the base) and a skirting board (the component);
(e) or between a screeded subfloor (the base) and the floor finish (the component).

Clearly the strength required of the fixings in the above examples varies considerably; from the heavy-duty fixing with a reliable structural performance, through to the light-duty fixing for a minor component, such as a skirting board or a floor finish, which carries little or no load, except possibly the weight of the component. The strength of the fixing will always be affected by the strength of the base and of the component.

It is sometimes difficult to decide which of the two parts connected by a fastener is the base and which is the component. Usually the 'heavier' of the two materials is the base (the structural frame, rather than the cladding); but this is not always the case. Sometimes the two materials are identical in 'weight' (e.g. two similar-sized pieces of timber). This is usually the case where both elements being connected are parts of the same structural unit – members of a timber truss, or leaves of a cavity wall. In this case, the question of which of the two elements is the base is unimportant; but this situation is relatively uncommon and so throughout this book we shall refer to the *base* and the *subsidiary component*.

Fixings fall into two broad categories – *mechanical fixings* which involve a metal (or sometimes plastic) nail, bolt, screw, dowel or socket to connect the component to the base, and *chemical fixings* which consist primarily of adhesives. Two types of chemical fixing – the chemical anchor and the injection anchor – although they both rely on the setting of an adhesive, are more like mechanical fixings and therefore will be included in the appropriate mechanical fixing section of this book. All other adhesives will be examined together in Chapter 7.

Mechanical fixings, because the base usually dictates the design of the fixing, will be grouped according to the type of base material in which they are used:

(a) mass walling (brick, stone, concrete blocks and concrete structure);
(b) timber and timber-based products;
(c) metal – heavy or lightweight;
(d) cellular materials (hollow core doors, panels and partitions) and thin wall materials (plasterboard, plastic sheets, etc.).

The purpose of fixings

Fixings are necessary to hold the parts of a joint together. They usually have to be strong enough to carry the weight of the subsidiary component, together with any load that it might have to carry (i.e. its self-weight and its imposed loadings). These are said to be *loadbearing* fixings.

Sometimes a fixing is merely intended to locate the subsidiary component, while its self-weight and its imposed loadings are carried on other fixings, or on a ledge formed in the base material. These fixings are called *restraint fixings*.

Usually a fixing is subjected to forces which try to tear the joint apart. These can operate in *tension*, resulting from a force acting parallel to the axis of a mechanical fixing and in a direction away from the base material. This type of force tries to break the fixing (whether it be the shank of a screw or the film of adhesive), or to pull the fixing out of, or through, the base or the component. In connection with tension forces we therefore refer to *pull-out failure* or *pull-through failure* (Fig. 2.1).

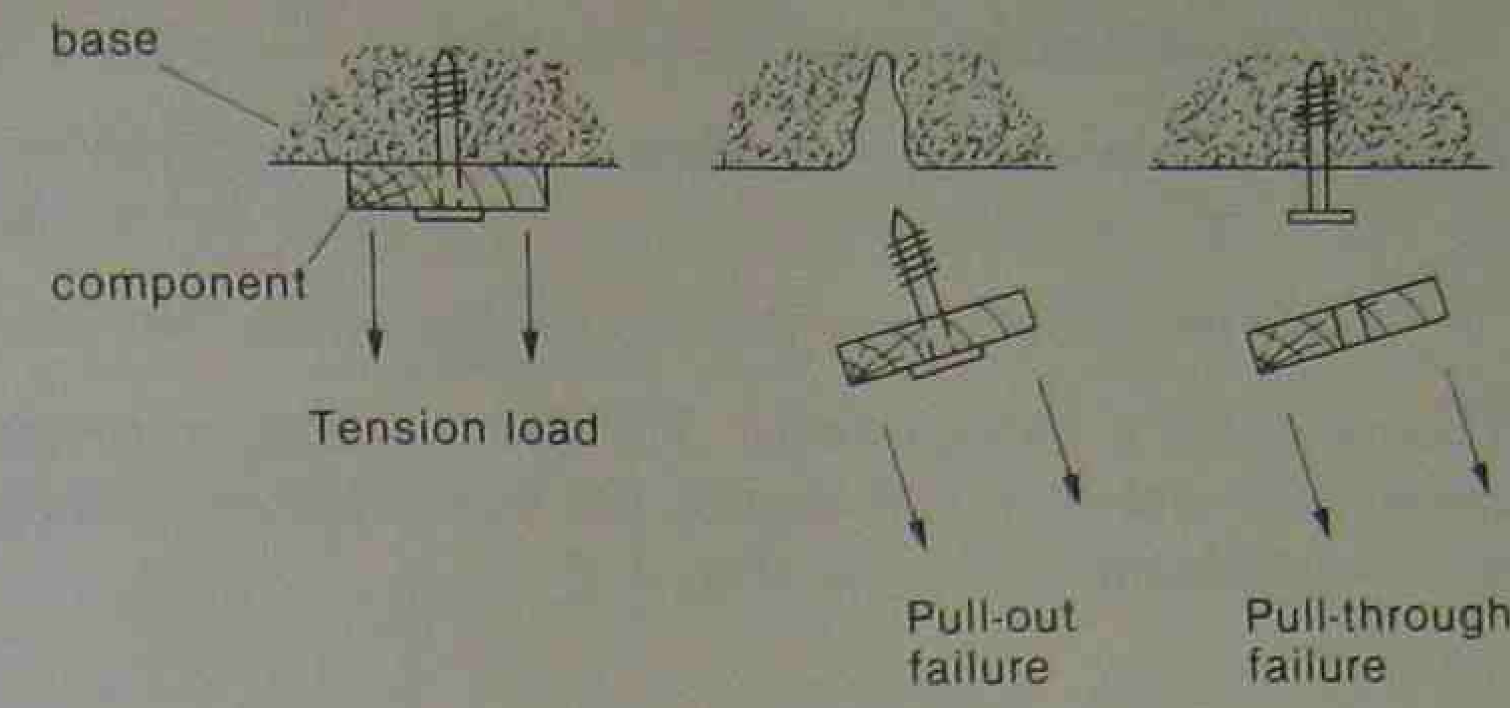


Fig. 2.1 Failures due to tension loading

Joints are often subjected to *shear* forces. These forces act at right-angles to the axis of the fixing. This type of force tries to distort or break the shank of the mechanical fixing, or tear the fixing across the component. This failure is referred to as *shear failure* (Fig. 2.2).

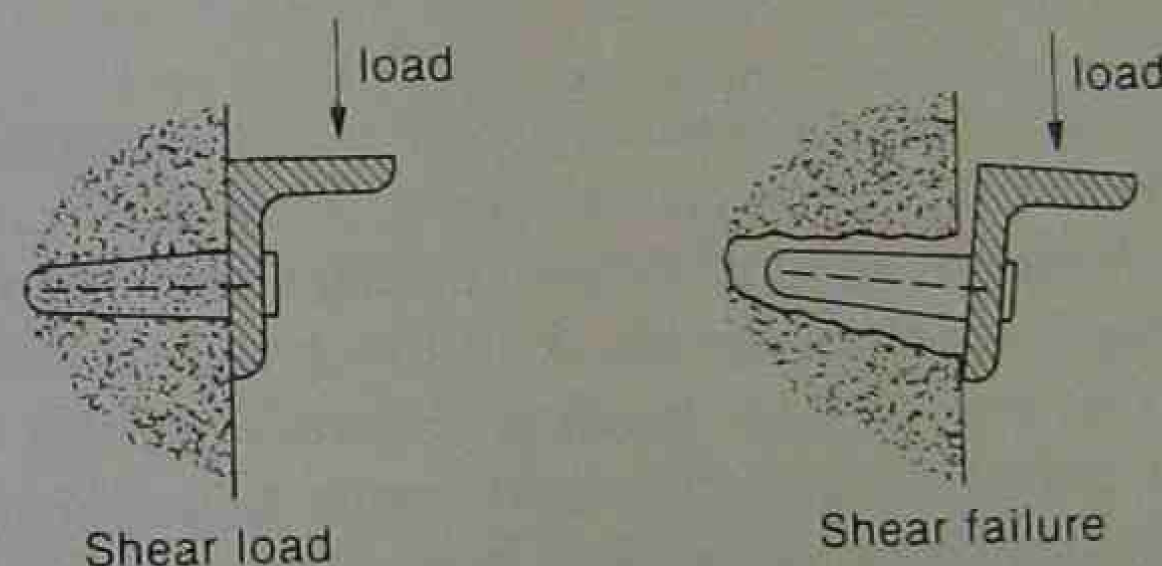


Fig. 2.2 Failure due to shear loading

Joints can be subjected to *compression* forces – forces acting parallel to the axis of a mechanical fixing and towards the base material. These tend to push the component into position against the base and hold it there. Compression is rarely the cause of fixing failure. It usually assists the fixing to carry out its task (Fig. 2.3).

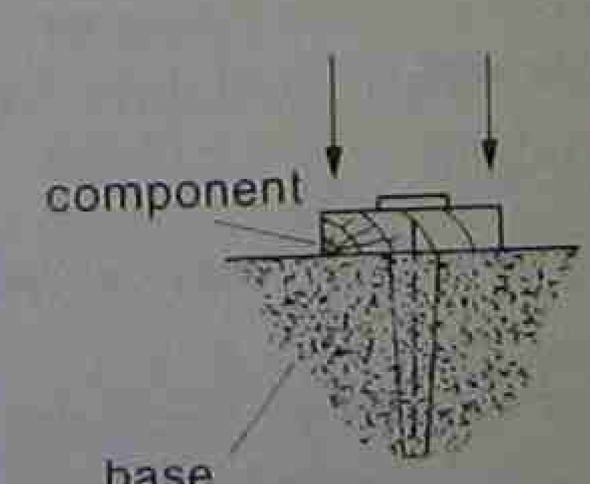


Fig. 2.3 Compression on a joint

How fixings work

Fixings work in one of three ways.

Type 1. Squeezing the component to the base

This is called a *through-fixing*. The device passes through both component and base and clamps the two together, usually as a result of tightening a screw (Fig. 2.4).

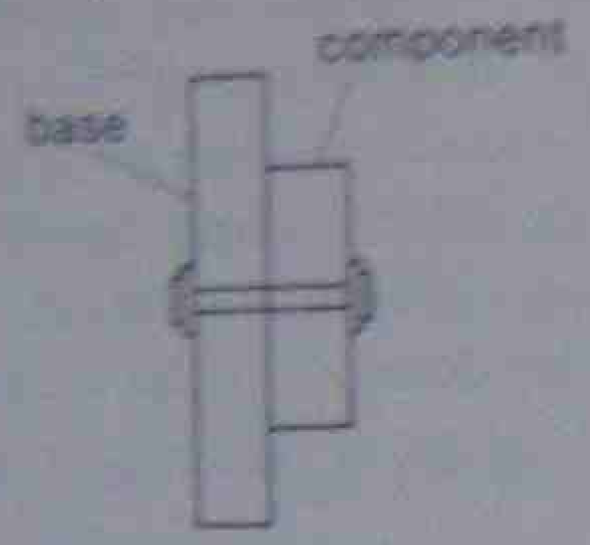


Fig. 2.4 Through-fixing

A through-fixing's strength depends firstly on the materials being secured together being strong enough not to allow the head of the fixing (or its washers or nut) to pull through the material and secondly on the strength of the shank of the device to resist fracture or distortion due to shear loading. Examples of this type of device include bolts, rivets, lightweight toggle fasteners and timber connectors (split-ring or toothed connectors). Generally this type of fixing generates considerable friction between the contact faces of the base and component and makes particularly strong shear joints.

Type 2. Making an anchorage in the base

The base material is penetrated by the anchorage fixing, but only enough to allow it to hold fast and make a secure fixing to support the load of the subsidiary component. There are three kinds of anchorage fixing. Those which depend on:

(a) making an anchorage in a pre-drilled hole (a *drilled-for fixing*);

(b) deforming the base material by driving or firing the fixing into the material, forcing it to be compressed into itself (a *base-deforming fixing*);

(c) building-in or casting-in a fixing into the base material.

All these devices achieve their pull-out strength by the friction of the body of the fastener against the base material (Fig. 2.5).

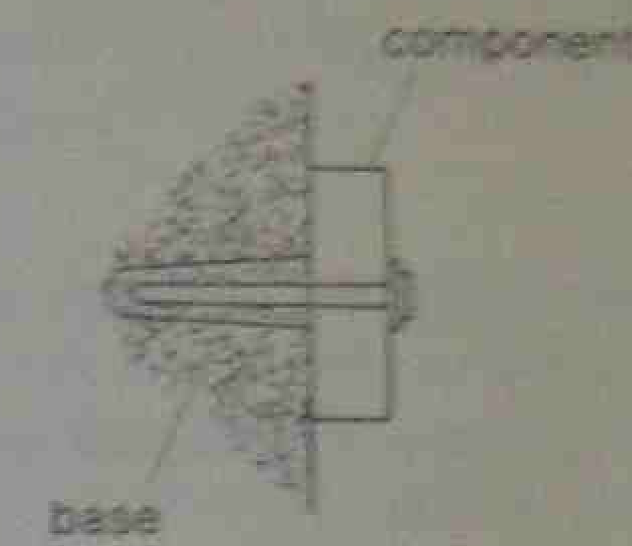


Fig. 2.5 Anchorage fixing

Type 2(a) devices include expanding anchors, plugs, chemical anchors and split-ring connectors.
Type 2(b) devices include nails, nail plates, screws and fired fixings actuated either by an explosive charge or compressed air.
Type 2(c) devices include joist hangers, sockets, corbels, wall ties and cast-in channels.

Anchorage devices are effective when pull-out or shear strength is required.

Type 3. Adhesion

The component is stuck to the face of the base material with no prior treatment to either material, other than smoothing their surfaces and spreading the adhesive. Glues tend to perform most effectively in shear rather than in tension (Fig. 2.6).

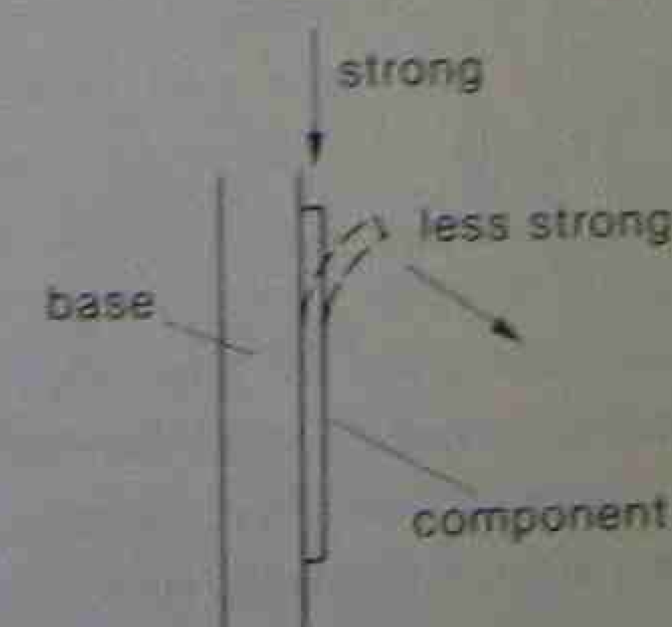


Fig. 2.6 Adhesive joint

In addition to the categories above there is a range of accessories intended to be used with mainly type (2) anchorage devices to provide loadbearing or restraint fixings for subsidiary components. These will be dealt with as accessories in the appropriate sections of this book.

How fixings are chosen

The choice of fixing type will depend firstly on the character and size of the base material, secondly on the character and size of the subsidiary component and lastly on the type, size and direction of the load which it is going to have to withstand.

The size and character of the base material dictates the type of fixing to be chosen, hence the division of mechanical fixings into base material groups in this book. The base's character alone will decide whether an expanding anchor can be used or if a particular type of nail or screw fixing is feasible.

The strength of the subsidiary component, too, needs to be considered if pull-through failure is to be avoided with certain types of fixing. It should be remembered that a fixing's strength is directly related to the strength of the base and of the component; and a joint can only be as strong as the materials joined.

The aim of a fixing is to achieve a secure joint which, in the case of a structural or high-performance fixing, must perform to a known standard. To this end fixings manufacturers produce load tables for their heavy-duty fixings used with base materials of specified known strength.

There is a number of secondary factors that need to be considered when choosing a fastener. Will the joint be subjected to thermal, moisture or other structural movement which may occur in the building – and if so, can the fixing accommodate the movement? Is the fixing made of a material that is compatible with the base and component materials, and are the base and component materials themselves compatible? This is an important point. If there is incompatibility of material, corrosion could result, leading to early breakdown of the joint. A typical case where care is needed is where a preservative has been used to treat the timber members of a truss. Some preservatives could cause the nail plates used in the truss manufacture to corrode. If there is any doubt about the side-effects of such treatments, the manufacturer should be consulted.

Corrosion

The problems of corrosion are particularly important when dealing with metal fixing devices and these problems will be referred to throughout the book when appropriate. Generally, corrosion falls into two categories: galvanic corrosion and oxidation (or rusting).

Galvanic corrosion

This is an effect which comes about when two metals are in contact and can result in one metal being eaten away. It is often called contact corrosion, or bi-metallic, or electrolytic attack. Some metals can be used together; others cannot. It all depends on the difference in voltage potential between the two metals. Table 2.1 (taken from information supplied by Harris and Edgar Ltd) indicates which metals can and cannot be used in contact. Where it is essential that incompatible metals are used together, the one must be shielded from direct contact with the other by a neoprene sleeve or gasket.

Oxidation

Steel or malleable iron fixings, because they are subject to oxidation in damp conditions, should not be used unless they have been zinc coated. There are three processes available:

1. *Hot-dip galvanizing*. The dipping of the element in molten zinc gives a good, but uneven, protection and may entail re-drilling or re-threading small holes and threads, with consequent loss of protection.
2. *Zinc plating*. This is an electro-plating process which gives a uniform and ductile protective coating. Re-drilling and re-threading is not usually necessary.
3. *Sherardizing*. This is a process involving the diffusion of hot zinc dust on to the element, giving a thin, uniform coating. It is often not recommended for threaded elements as there is a tendency for the coating to build up on the threads.

No zinc-coating process is completely without the danger of minor defects or damage to the coating which could be the starting point for corrosion. It is frequently recommended that any heavy-duty fixing in locations with little or no access (where

Table 2.1 Degree of galvanic corrosion hazard in bi-metallic contacts (derived from information provided by Harris and Edgar Ltd)

	Copper	*Phosphor bronze*	*Aluminium bronze*	*Stainless steel*	*Mild steel*	*Manganese bronze*	*Aluminium*	*Cast iron*
Copper	S	S	S	D	X	D	X	X
Phosphor bronze	S	S	S	D	X	D	X	X
Aluminium bronze	S	S	S	D	X	D	X	X
Stainless steel	D	D	D	S	X	D	X	X
Mild steel	X	X	X	X	S	X	X	S
Manganese bronze	D	D	D	D	X	S	X	X
Aluminium	X	X	X	X	X	X	S	X
Cast iron	X	X	X	X	S	X	X	S

S = safe combination
D = safe combination in dry conditions only
X = combination never to be used

corrosion could take place without detection) should be made of non-ferrous metal.

Certainly stainless steel or other non-ferrous devices should always be used in positions of extreme hazard (chemical plants, sewage works or marine locations) or where conditions of prolonged dampness are expected.

3

Mechanical fixings in mass walling bases

All fixings made into mass walling bases have their performance influenced by the strength of the base (Table 3.1).

Density and compressive strength are the two characteristics that most affect fixing performance. Dense masonry generally gives the more effective support for fixings, but it is usually more difficult to fix to a dense masonry base because of the difficulty of drilling or firing into it. Also the accuracy of holes drilled into dense masonry is not always as good as it should be, dramatically affecting the performance of the fixing. Low-density masonry, on the other hand, gives little support to fixings like expanding anchors and, once more, accurate drilling can be a problem. In the case of aerated concrete – the masonry with the lowest density – expanding anchors should not be used.

A few guide figures to masonry materials' vital statistics are given in Table 3.2.

Manufacturers produce performance data for their fixings when applied to base materials of particular compressive strength. These are usually the result of tests performed in accordance with BS 5080: Part 1: *Method of Test for Structural Fixings in Concrete and Masonry* and the recommendations of the Construction Fixing Association. If the base material does not compare with that specified in the manufacturer's data, the company should be consulted and, maybe, further site tests should be carried out, if the performance of the fixing is structurally important.

It must be remembered that no fixing into masonry can be stronger than the masonry itself and generally the deeper the fixing, the greater the volume of masonry resisting its withdrawal and therefore the stronger the fixing (Fig. 3.1).

Safety factors Because, even under controlled test conditions, fixing performance can vary, it is normal to determine working loads by applying a safety factor to a fixing's *ultimate load* (the

maximum load it can carry before failure) or alternatively to what is called its *first movement load* (the point at which the fixing is first observed to move 0.2 mm).

For tensile loading, the working load is usually taken as either $\frac{\text{first movement load}}{2}$ or $\frac{\text{ultimate load}}{5}$.

Shear loads are taken as 75 per cent of safe tensile load and tensile shock loading as half the safe sustained tensile load.

Section A: Drilled-for fixings – type 2(a)

The technique of drilling

Before dealing with the fasteners in this category, it would seem wise, as they all depend on the effective and accurate drilling of holes in the mass walling base, to examine drilling techniques and the type of equipment used.

Drilling in masonry is not a cutting operation, as it is in timber or metal; it is a pulverizing and scraping process, involving abrasion of the drill bit. Masonry bits, therefore, have hard carbide tips to give maximum wear.

There are six methods of forming holes in mass walling:

1. By hand, using a percussion tool — This is a slow and laborious method, particularly in hard walling. It can, however, result in accurate holes, particularly for the smaller plug-type fixing.
2. Hand rotary drill — Once more a slow process in hard walling, particularly when drilling larger holes. In softer walling, like common brickwork or lightweight concrete, it produces accurate holes.
3. Electric rotary drill — This is an accurate method of drilling holes, but it can be very slow when the hole diameter exceeds 16 mm in hard walling. It is recommended for softer mass walling bases, such as common brickwork or lightweight concrete.
4. Electric rotary/impact tool — An accurate method of drilling holes in hard walling materials, but its

Table 3.1 Mechanical fixings in mass walling bases: which fixing to use; performance, base requirements

	Duty			*Base*												*Services needed*	
Fixing type	*Light*	*Medium*	*Heavy*	*Dense concrete*	*L/w concrete*	*L/w concrete block*	*Aerated concrete*	*Aerated concrete block*	*Non-fines concrete*	*Brick-work: solid*	*Brick-work: cellular*	*Clay block: cellular*	*Stone: hard*	*Stone: soft*	*Electricity*	*Compressed air*
Plugs; fibrous/ plastic	√			√	√	√	√	√		√	√		√	√	√	
Hammer-set plug	√			√	√	√				√	√		√	√	√	
Plugging compound	√			√	√	√	√	√	√	√	√				√	
Expanding anchor		√	√	√	√	√			√	√			√	√	√	
Self-drill anchor		√	√	√	√	√			√				√	√	√	
Ceiling suspension anchor		√		√											√	
Insulation fastener	√			√	√	√	√	√		√			√	√	√	
Cavity wall repair (mech.)		√	√			√				√			√	√	√	
Chemical anchor		√	√	√	√	√			√	√			√	√	√	
Cavity wall repair (chem.)						√				√			√	√	√	
Injection anchor	√	√					√	√			√	√			√	
Screw-in anchor	√	√		√	√	√				√				√	√	
Masonry nail	√			√	√	√				√				√		
Powder-actuated pin		√	√	√	√	√				√				√		
Pneumatic-actuated pin		√	√	√	√	√				√				√		√
Hammer-in anchor	√						√	√								
Joist hanger		√				√		√		√	√					
Anchor strap		√				√		√		√	√					
Cavity wall tie		√				√		√		√	√	√	√	√		
Cramp/dowel		√		√		√		√					√	√		
Cast-in plug	√	√		√	√											
Cast-in socket		√	√	√	√											
Cast-in channel		√	√	√	√											
Cast-in corbel		√	√	√	√											

Table 3.2

Material	*Density* (kg/m^3)	*Compressive strength* (N/mm^2)
Structural concrete	2400	25–40
Staffordshire blue brick or similar	2000	48.4
Lightweight concrete	2000	15–30
Common brickwork	1760	20.0
Aerated concrete	700	2.8–5.5

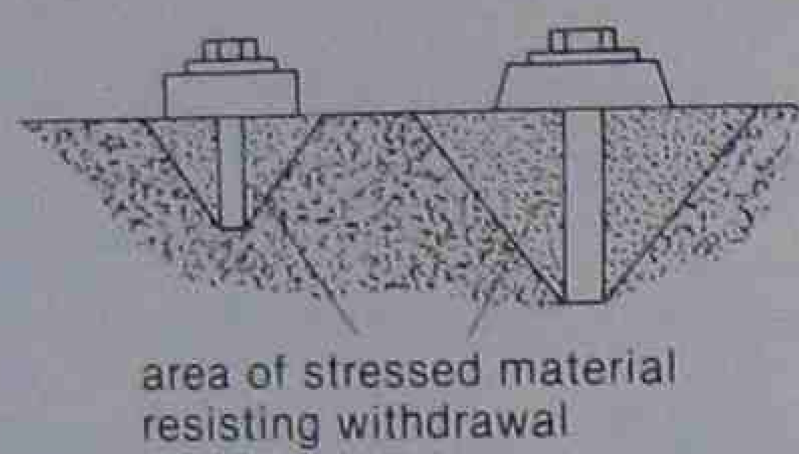


Fig. 3.1 Effect of the depth of a fixing

effectiveness falls off when the hole diameter exceeds 16 mm. It is recommended for most mass walling bases, except those which are extremely hard.

5. Electric hammer drill — This is a quick method of forming holes up to 24 mm diameter with reasonable accuracy in dense concrete, hard brickwork and hard stone.

6. Pneumatic hammer drill — This is the fastest method, but hole accuracy and shape are not always as good as they might be. This method is recommended for large-diameter holes for heavy-duty civil engineering fixings.

Drilling speed Masonry drills produce their best performance when used at medium or slow drilling speeds (400 to 600 rpm, fully loaded). High speeds result in overheating and consequent premature drill wear. Lower speeds, on the other hand,

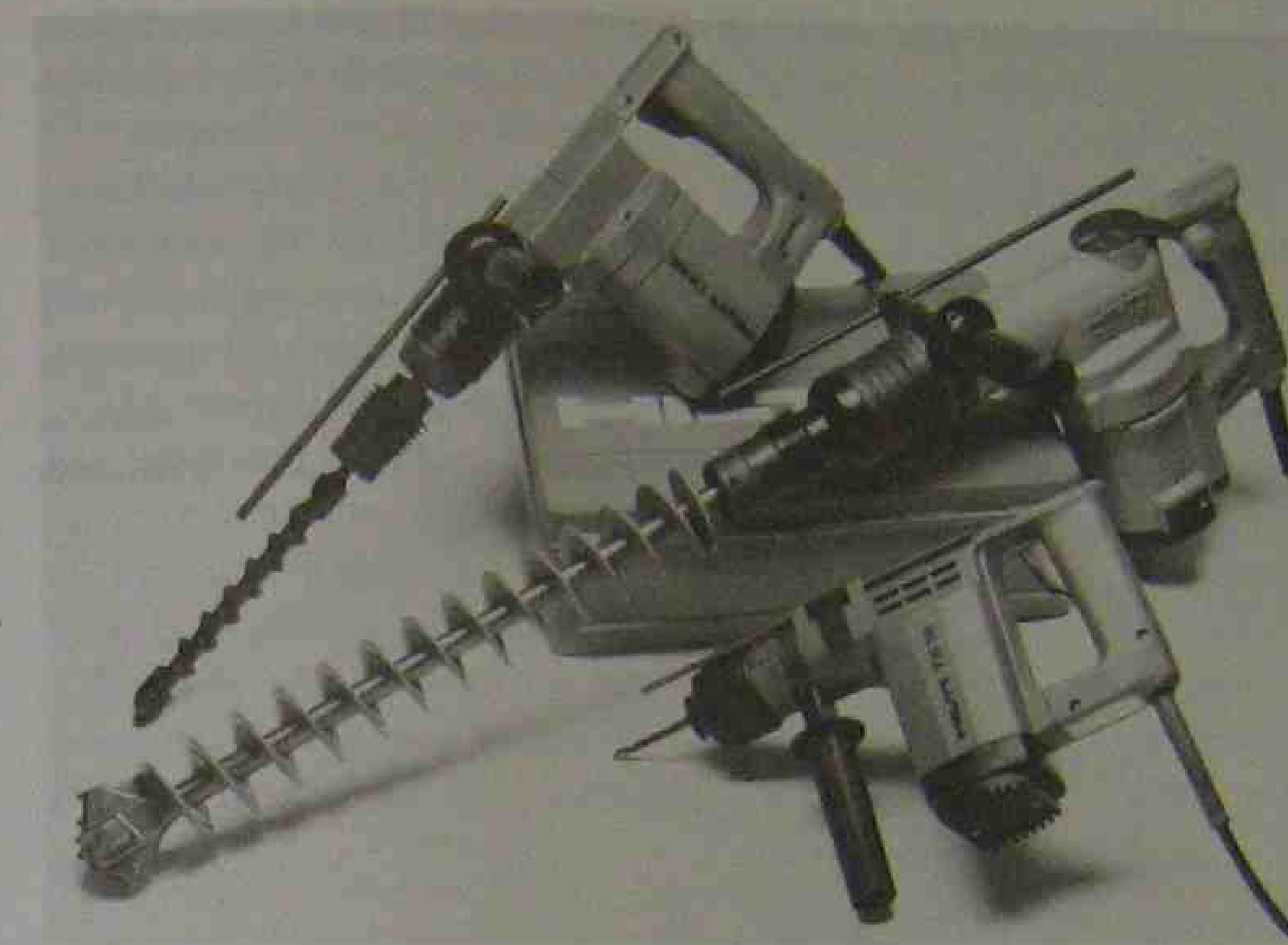


Fig. 3.2 The Hilti range of electro-pneumatic drilling machines

generate less heat which can be dissipated up the drill shank (Fig. 3.2).

De-speed adaptors are available to enable high-speed drills to operate at lower speeds. They also permit high-speed drills to make much larger holes.

Where a walling base is faced with a delicate material, such as glass or glazed tile, it is wise to use a Durium-tip drill at slow speeds.

Impact adaptor There are adaptors available which convert rotary drills into impact tools. These usually only permit the use of the smaller sizes of drill bit.

Points to remember

Whatever method of drilling is used, certain points should be observed:

1. The correct combination of rotary drill or hammer, drill bit or tool should always be used. The size of drill bit must always be that recommended for the particular size of fixing. This is essential if the fixing is to achieve its full performance.
2. Every rotary drill or hammer is limited in its performance

by the largest drill bit diameter that is recommended by the manufacturer for use in a specified base material. These recommendations should always be followed.

3. The procedures given in the manufacturer's operating instructions for the tool should always be observed, including work method, cleaning and servicing.
4. After each hole has been drilled (and before the fixing is placed in position) the hole should be cleared of cutting dust, preferably by using a blow-out bulb. The efficiency of the fixing can depend on this (Fig. 3.3).

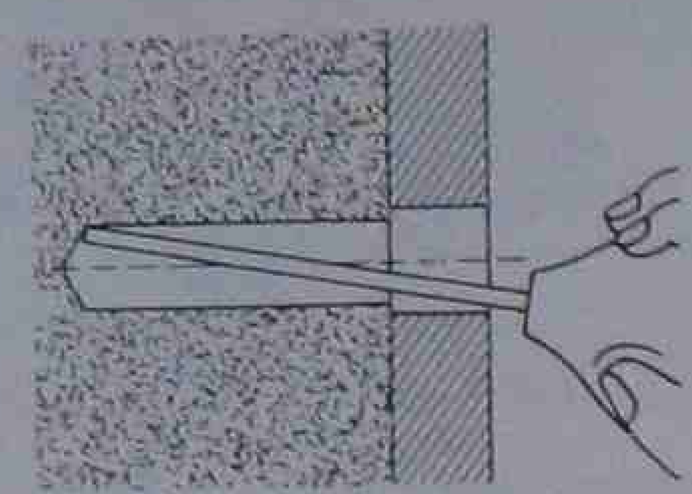


Fig. 3.3 Using a blow-out bulb

5. Finally (and most importantly) safety glasses should always be worn when drilling to prevent dust and chippings flying into the eyes. These can usually be worn over any normal glasses and are made of shatter-resistant polycarbonate.

A1 Expanding anchors and plugs

Fixing devices contained in this section include:

(a) plugs (fibrous or plastic);
(b) hammer-set plastic plugs;
(c) asbestos-based plugging compound;
(d) expanding anchors of three types: projecting bolt type – loose bolt type – socket type;
(e) self-drilling expanding anchors;
(f) expanding anchor: (non-friction type).
(g) specialized anchor devices: including ceiling suspension anchors – insulation fasteners – cavity wall repair anchors;

The common factor between all fixings in this category is that they make an anchorage in the base material by their expansion

within a drilled hole in the base. (The self-drilling expanding anchor is an exception to this rule, because it drills its own hole.) The expansion of the fixing presses on the sides of the drilling, building up friction which gives the device its holding power.

Advice on drilling holes in masonry bases is given in the previous section. The drilling of accurate holes of the correct diameter and depth is essential to the performance of the device.

The strength of these fixings varies greatly from the lightweight fixings made by plugs and insulation fasteners to the heavyweight structural performance demanded of larger expanding anchors. In all types of fixing, however, the care with which it is set in the base is critical to its performance.

Further information on the use of anchors in various types of mass walling base, their positioning and other matters affecting their performance are dealt with in the background section on page 63.

Plugs

Description These are light-duty, screwable inserts, placed in mass walling to receive woodscrews or coachbolts used to secure the subsidiary component. In the past they have been made of wood; today they can be made of a natural fibre (as in the Fibre Rawlplug) or plastic (polypropylene or nylon). In each case the action of driving the screw or coachbolt expands and distorts the plug against the sides of the drilled hole, filling up any surface irregularities of the drilling. Once a plug is placed, it cannot usually be easily removed, but plugs to which fixings have already been made can often receive a later screw fixing.

Applications Plugs are used for making relatively light-duty fixings to any mass walling material and are particularly applicable for use in soft building materials, such as aerated or lightweight concrete, for which some specially profiled plugs have been developed. Some of these are of the hammer-home variety: some like the Fischer Twist Lock anchor are recommended for aerated concrete; others like the Rawlplug Hammer Screw are not (Fig. 3.4).

The performance of all plug fixings depends on the drilled hole being of the correct size for the plug and the plug being of the correct size for the screw. In the case of irregular or ragged-shaped

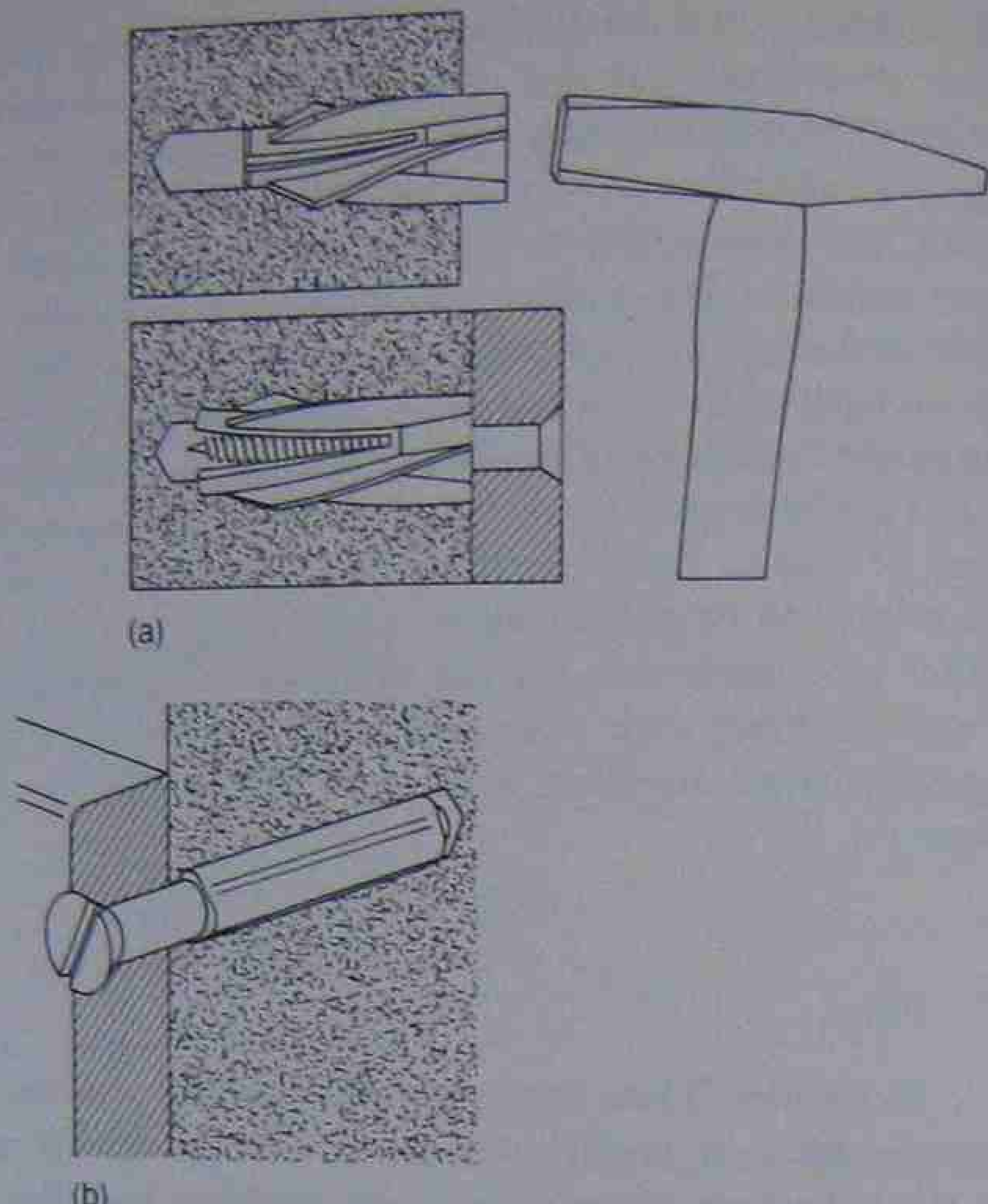


Fig. 3.4 (a) Fischer Twist Lock anchor (b) Rawlplug Hammer Screw

holes (or holes in poor-quality masonry) an asbestos-based compound (Rawlplastic) can be used instead of a preformed plug.

Types of plug

Fibrous plugs:	To receive woodscrews sizes from 6 to 20 or coachscrews sizes from 6 to 8 mm.
Plastic plugs:	To receive woodscrews sizes 4 to 20. In the case of plastic plugs one size of plug is recommended for several screw sizes, so that some manufacturers are able to cover the range of screw sizes in 6 or 7 plug diameters.
Hammer-set plastic plugs	
Asbestos-based compound:	To receive woodscrews sizes from 4 to 20.

Note: Some special plastic plugs are manufactured for use in lightweight concrete bases only. These are usually only available in a few sizes in the middle of the upper range of screw sizes (see Fig. 3.4(a)).

Setting instructions

Fibrous plugs

1. Drill hole of the recommended diameter and to a depth equal to the length of the plug.
2. Insert the screw one or two turns into the plug and use the screw to push the plug into the hole.
3. Turn the screw into the plug to the extent of the thread only.
4. Withdraw the screw leaving the plug set in the base.
5. Place the subsidiary component in position and replace the screw, driving until tight.

Note: The plug should always be the same length as the thread on the screw and the shank of the screw should preferably not enter the plug if damage is to be avoided. The plug should be set completely into the walling, below the surface of the plaster.

Plastic plugs

1. Drill hole of the recommended diameter.
2. Insert the plug so that its flange is on the surface of the mass walling base (or its plaster).
3. Pass the screw through the subsidiary component into the plug and drive the screw home.

Note: This is the procedure for plastic plugs with flanged tops. Other manufacturers produce non-flanged plugs which can be inserted on the end of the screw, through the subsidiary component, into the drilled hole, and then the screw can be tightened.

Hammer-set plastic plugs

(These are push-through fixings in which the plastic plug is already fixed on the point of the screw or nail; see (b) in Fig. 3.4.)

1. Select a device longer than the thickness of the subsidiary component plus the plaster. This is the length of the plug

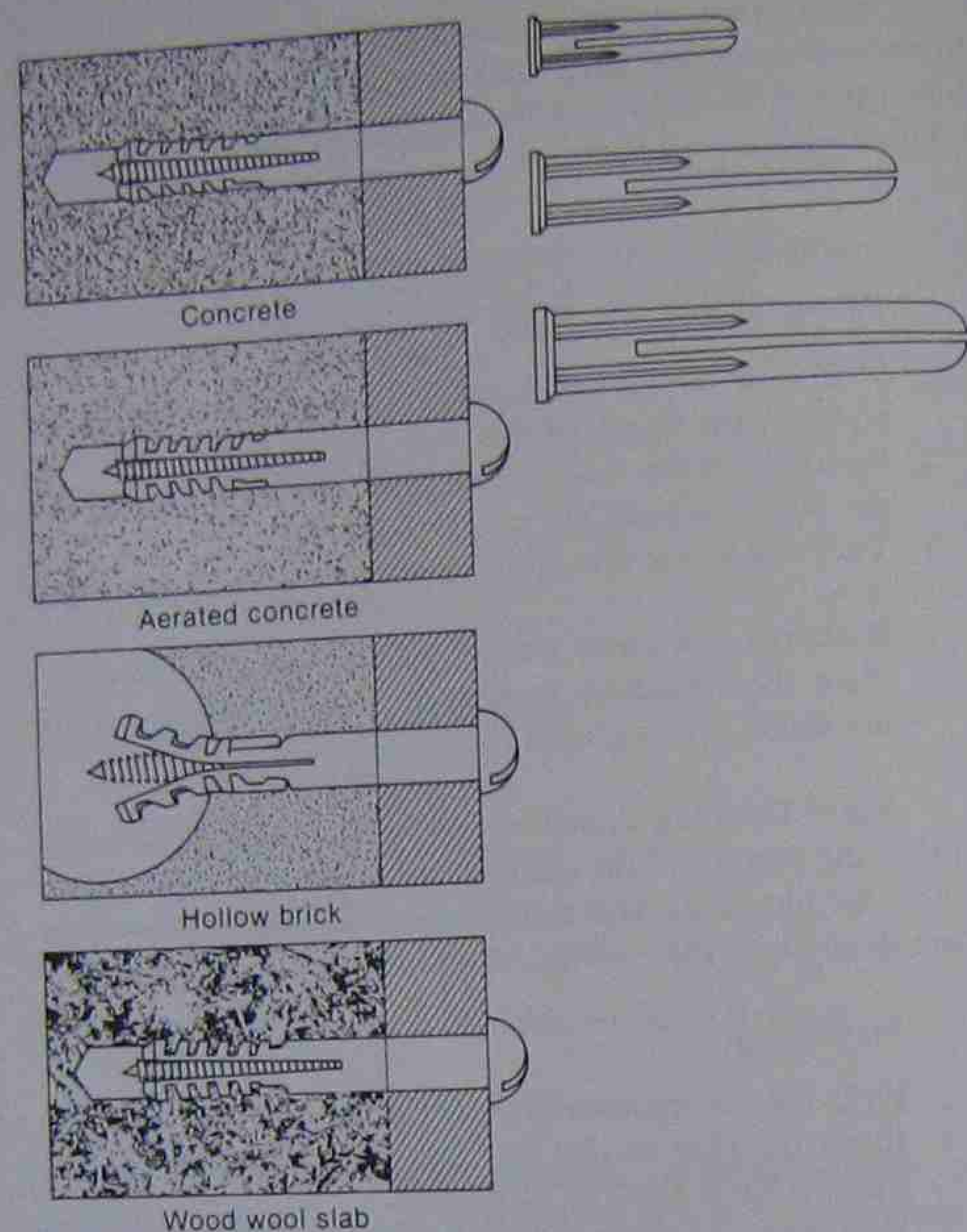


Fig. 3.5 Fischer Wallplug Type S and Split plastic plug range

that will be set in the walling. In heavier fixings this measurement should be at least 25 mm.

2. Drill hole of the recommended diameter and 5 mm deeper than the fixing.
3. Position secondary component over the hole and insert fixing and hammer the screw (or nail) home.
4. A screwdriver can be used as a setting tool where the hammer cannot reach the head of the fixing. In the case of screw fixings, a final tightening can be made by screwing.

Note: Those devices using a drive screw, rather than a nail,

can be easily unscrewed at some future date if the component needs to be removed. Refixing is possible by replacing the screw.

Asbestos-based compound

1. Drill a hole of a diameter no less than that of the screw.
2. Take a small quantity of the compound, immerse it quickly in water and squeeze and roll it into the form of a plug.
3. Ram the plug into the hole with a tool provided with the compound and, using the same tool, make a small lead hole in the surface of the plug when the drilling is completely filled.
4. Pass the screw through the subsidiary component and drive into the plug, being careful not to over-drive.

Note: Compounds contain asbestos and therefore should be handled with care. Always keep the container closed. Avoid inhaling the fibres. Dampen the product immediately it is removed from the container and always thoroughly wash your hands after using the compound.

Asbestos-based plugging compound can be used when the hole to receive the screw fixing is irregular or over-sized for the correct size of pre-formed plug. Thus drilling mistakes can be rectified.

Expanding anchors

Description Generally these represent the more heavy-duty drilled-for-fixings in mass walling. They are normally manufactured of steel (galvanized or zinc-plated), stainless steel or aluminium bronze, although there are heavy glass-reinforced nylon products on the market. They all depend for their holding power on the expansion of a part of the device within the drilled hole. This compresses the surrounding mass walling and produces a high frictional resistance to pull-out. The expanding section of the device can consist of metal or PVC sleeves, or a metal shell, depending on the pattern of the device (Fig. 3.6).

Applications Expanding anchors are used to make medium- to heavy-duty fixings with a predictable performance into a variety of walling materials, including dense concrete, no-fines concrete, lightweight concrete with a dense texture, stone

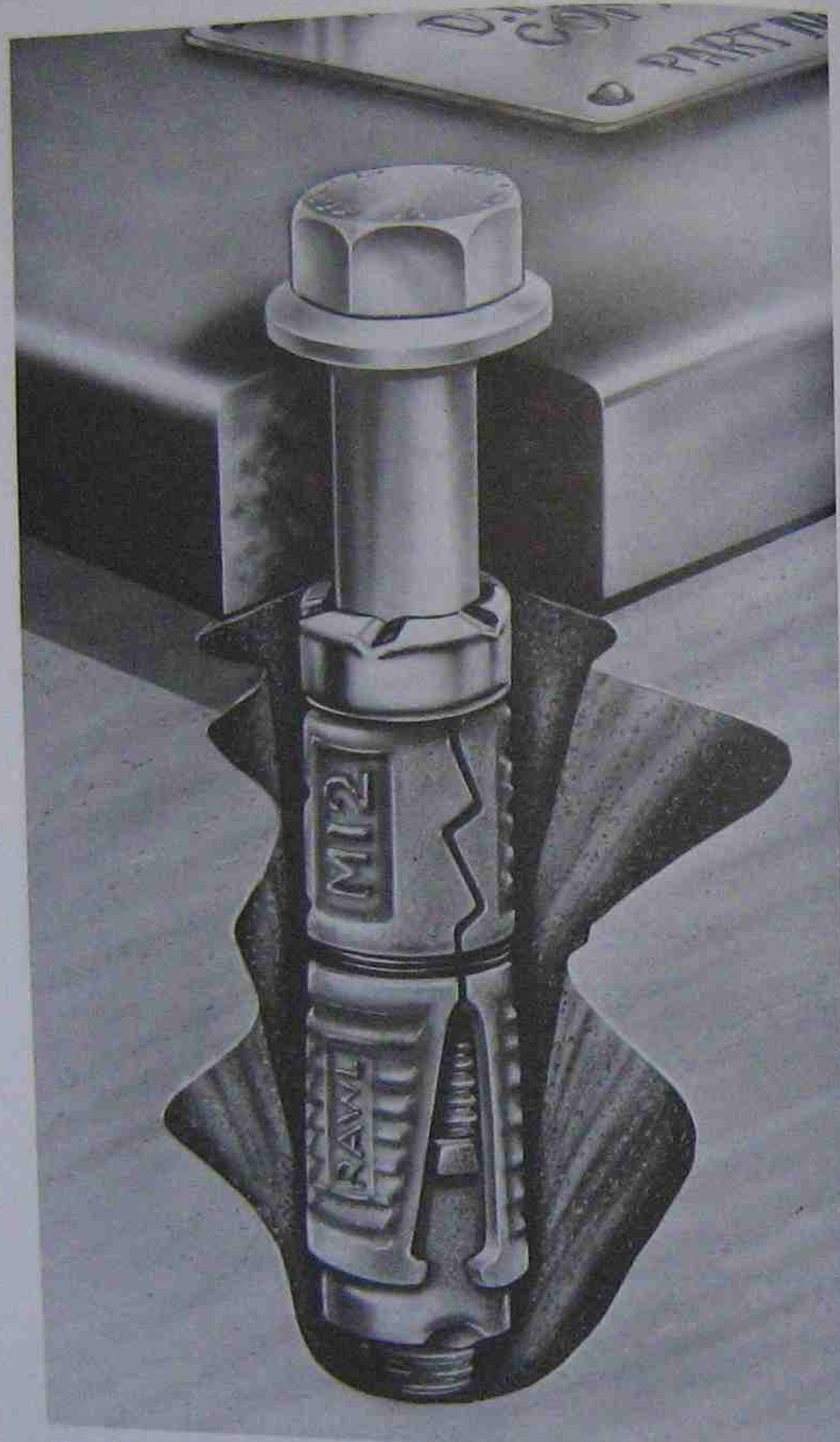


Fig. 3.6 Rawlplug expanding anchor (artist's impression)

and brickwork made up of solid bricks (without large internal voids). In the case of brickwork or blockwork, anchors should be set as centrally as possible in the bricks or blocks. In the case of bricks which are exceedingly hard and brittle, anchors can be placed in the mortar joints (but their pull-out strength may be affected).

Expanding anchors should not be used in aerated concrete – only lightweight fixings can be made into this material using injection anchors (see sect. A3, p. 39) or specially developed plastic plugs (Hilti H6 and FD anchors) or the Rawlnut (see Ch. 6).

Self-drilling anchors should not be used with any brick or clay block base.

An anchor's performance will vary with the strength of the base into which it is set and its position in the base in relation to its edges. Because of the expanding action of these devices, they can cause cracking or spalling if placed too close to the edges of the base. Simple rules of thumb that can safely be applied are: when the load is at right angles to the edge of the base, the minimum distance between the centre of the anchor and the edge of the base should be 2.5 to 4 times the anchor's length; when the load is parallel to the edge, 1.5 to 3 times the anchor's length. If anchors are placed too close together (less than 2 to 3.5 times the length of the anchor) their performance can also suffer. These and other matters affecting the performance of expanding anchors will be dealt with in the background part of this chapter.

As the outside diameter and length of an anchor increases (all other factors being equal) so its load capability increases. This, however, is dependent upon the correct drilling of the hole.

Over-deep holes can reduce fixing performance unless the device is the type which can be sleeved adequately. Holes of too large a diameter reduce the pull-out strength of the anchor because its expansion is not able to compress the base sufficiently. The anchor needs to be fully expanded in the right-sized hole to achieve its best performance.

Types of anchor

Projecting bolt type: These devices have a captive bolt or stud. Some are push-through fasteners (in other words the subsidiary component can be positioned before the anchor is placed in the drilling); some need to be set and the subsidiary component positioned over its projecting stud. Sizes vary from bolt diameters of 5 to 24 mm. Thicknesses of subsidiary components fixed can be up to 115 mm (Fig. 3.7).

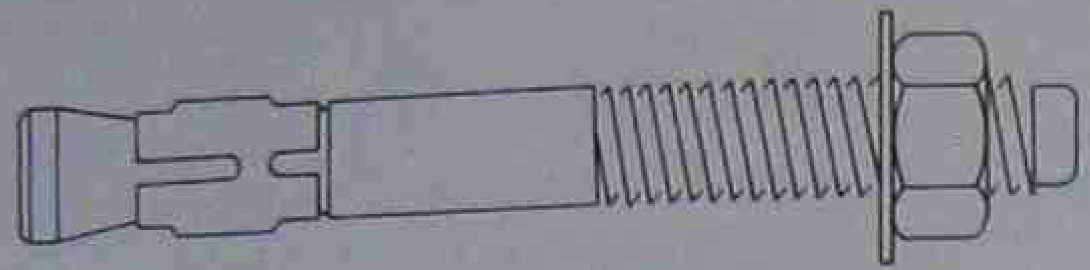


Fig. 3.7 Projecting bolt type of anchor

Loose bolt type: This anchor is set before the subsidiary component is positioned. The size range is approximately the same as the projecting bolt type of anchor (Fig. 3.8).

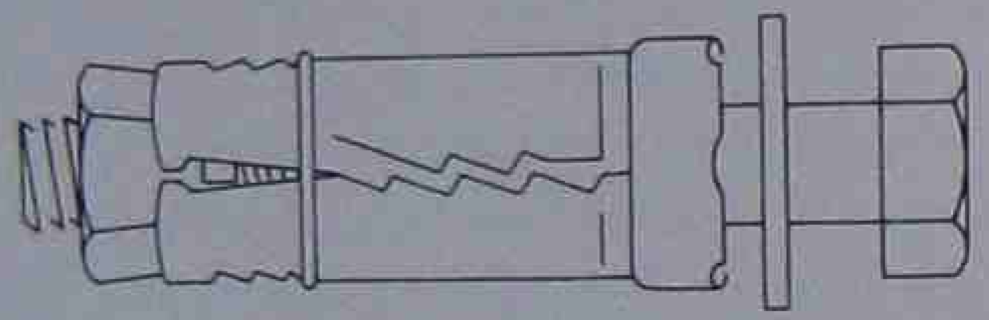


Fig. 3.8 Loose bolt type of anchor

Socket type: This is a threaded socket anchored in the base (much like a heavy-duty plug) prepared to receive a bolt fixing. Sizes suit bolts with diameters of 6 to 24 mm (Fig. 3.9).

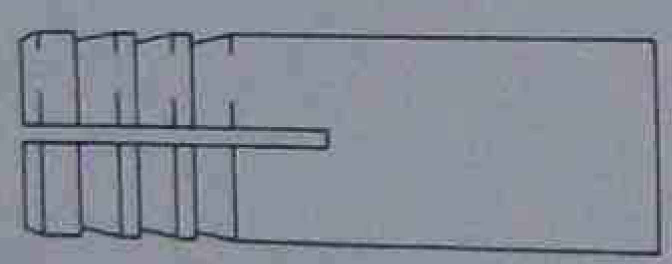


Fig. 3.9 Socket type of anchor

There is a variety of adaptors, particularly for use with loose bolt or socket types of anchor which vary the head style of the fixing (such as the hook-and-eye adaptor) or allow the anchor to be used in alternative situations (such as extended studding and metal or plastic collar adaptors or sleeves), (Fig. 3.10).

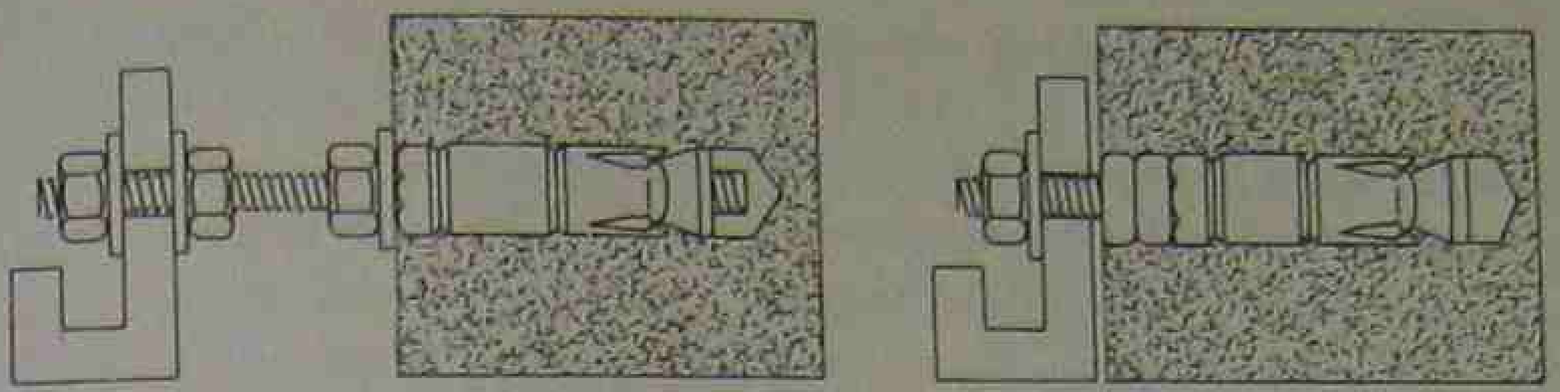


Fig. 3.10 (a) Extended studding (b) plastic collar adaptors

Setting instructions

Projecting bolt type (push-through version)

1. Drill a hole of the recommended depth and diameter. Often in this type of anchor the diameter of the drilling is the same as that of the device.
2. Insert the anchor through the subsidiary component with the nut and washer on the end of the stud.
3. Tighten the nut to the recommended torque. This expands the anchor and secures the component.

Note: The fact that the subsidiary component can be in place during the whole fixing operation is particularly useful when the component is heavy or awkward to move. In the case of some lighter devices used to fix door linings and window frames, the subsidiary component can be positioned and drilled through at the same time as the base is drilled. Some of these devices, known as *frame anchors*, have screw (rather than bolt) heads and are specially designed for stand-off fastenings where there is a gap between the back of the subsidiary component and the base.

Projecting-bolt type (non-push-through version)

1. Drill a hole of the recommended diameter and depth.
2. Insert the anchor.
3. Position the subsidiary component over the threaded studding of the anchor.
4. Apply the nut and washer and tighten to the recommended torque to expand the anchor and secure the component (Fig. 3.11).

Loose bolt type

1. Drill a hole of the recommended diameter and depth.

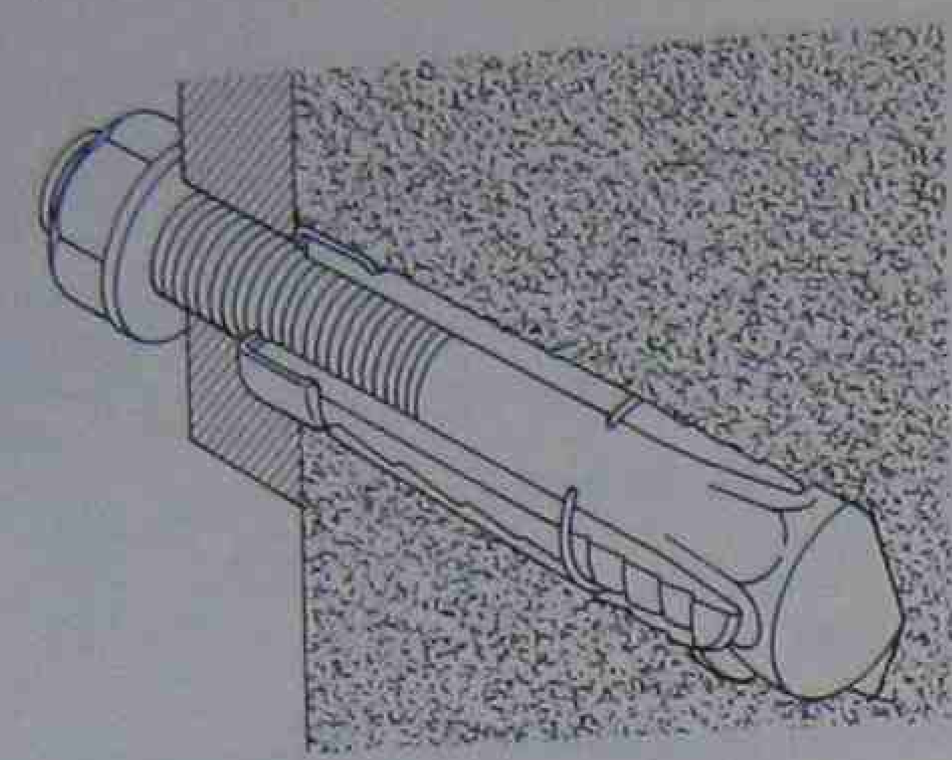


Fig. 3.11 Projecting bolt anchor in place

2. Insert the body of the anchor.
3. Position the subsidiary component over the anchor.
4. Pass the bolt through the subsidiary component into the body of the anchor and tighten to the recommended torque to expand the anchor and secure the component (Fig. 3.12).

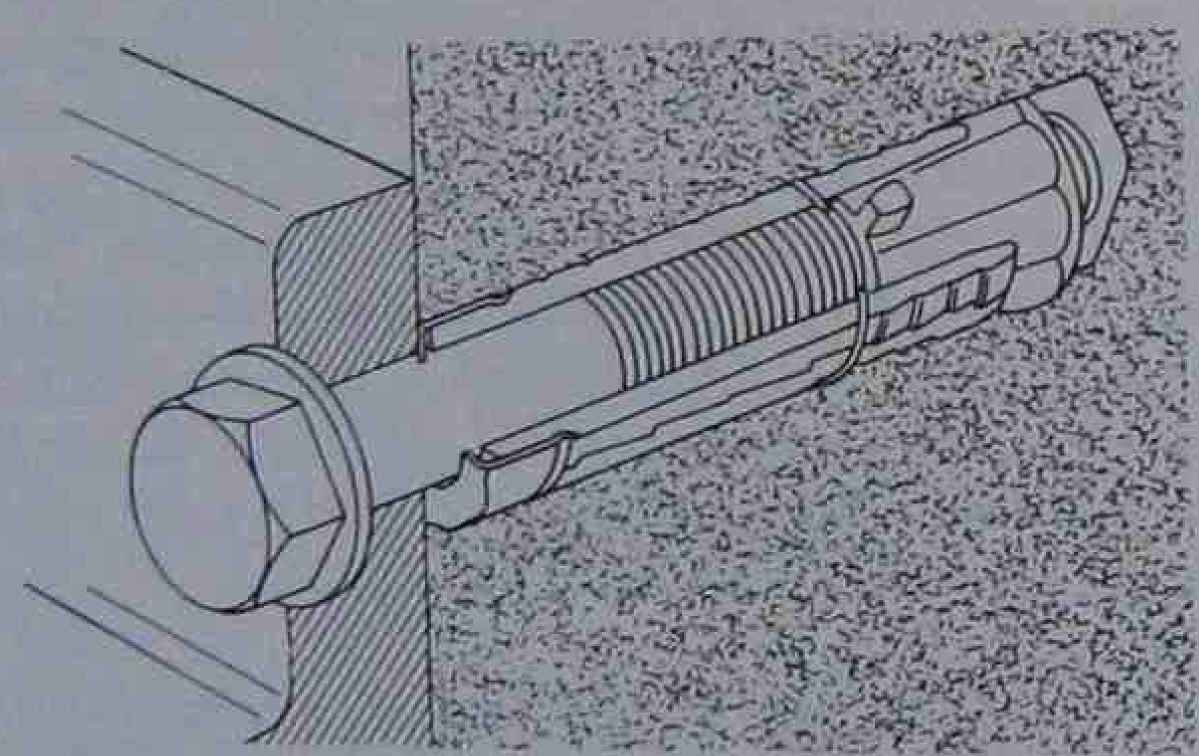


Fig. 3.12 Loose bolt anchor in place

Notes: (a) These devices can be converted to stand-off anchors by using a length of threaded studding in place of the bolt (see Fig. 3.10(a)). The procedure is as follows:

1. Drill a hole of the recommended diameter and depth.
2. Screw the studding into the body of the anchor in accordance with the manufacturer's instructions. At this point a nut and washer should be run on to the studding down to the body of the anchor.

3. Push the body of the anchor into the hole.
4. Tighten the nut to set the anchor.
5. Run another nut and washer on to the studding and position in line with the back surface of the subsidiary component (i.e. the stand-off position).
6. Offer up the subsidiary component and secure it with a third nut and washer.

Note: (b) This type of device (and other socket devices) can be set deeper into the base by using another adaptor – a solid collar or sleeve. This takes up the distance between the body of the anchor and the subsidiary component (see Fig. 3.10(b)).

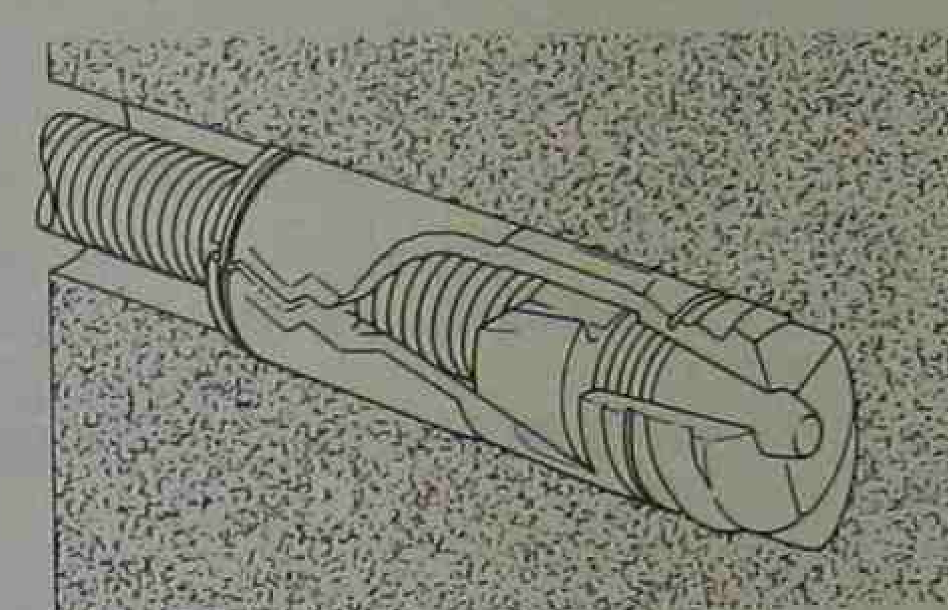


Fig. 3.13 Socket anchor in place

Socket type

1. Drill a hole of the recommended diameter and depth.
2. Screw the studding into the body of the anchor until expansion starts.
3. Push the body of the anchor into the hole using the studding.
4. Tighten the studding to the recommended torque to expand the anchor.
5. Offer up the subsidiary component and secure by running-on a nut and washer.

Note: There is a commonly used alternative device in which the socket is placed in the drilling and then expanded by driving a setting tool into the body of the anchor. This firmly fixes the socket, making it totally independent of any further fixing of the subsidiary component. This is an example in which the expansion anchor is directly comparable with a plug fixing, but with a greater load-carrying potential (Fig. 3.13).

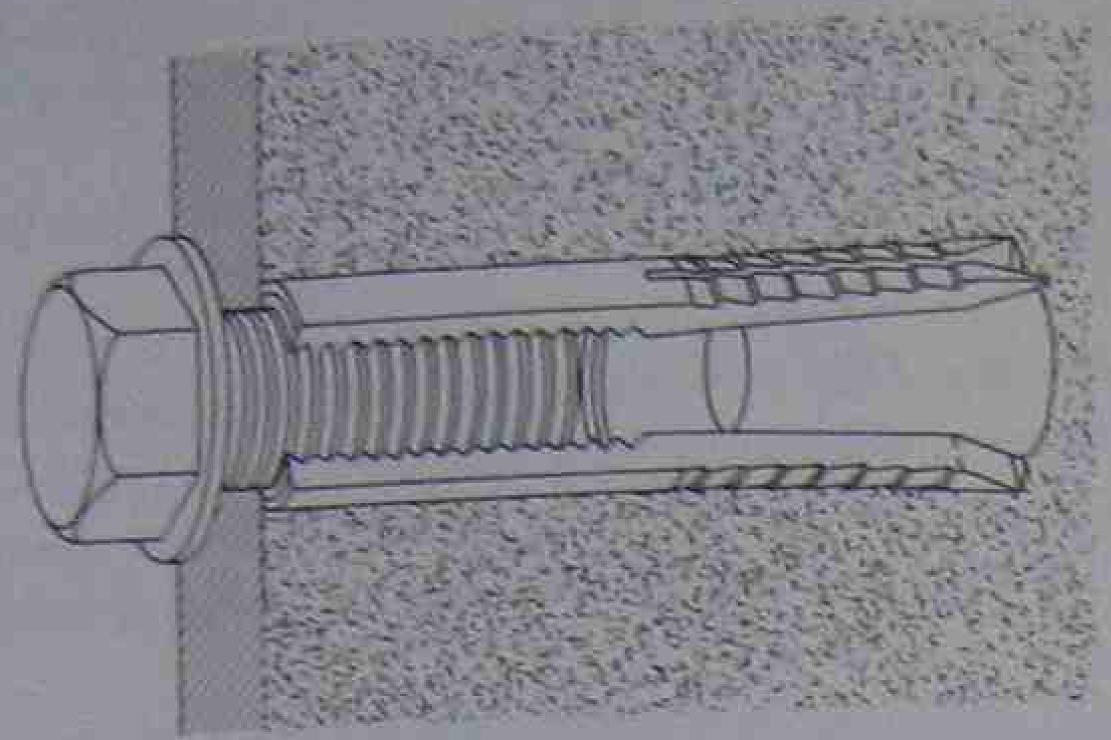


Fig. 3.14 Self-drilling anchor in place

Self-drilling type

This is a version of the socket type of anchor (Fig. 3.14).

1. Fit the anchor in an appropriate adaptor into the chuck of a rotary/impact hammer.
2. Commence drilling.
3. When the chuck almost hits the surface of the base material, withdraw the anchor and clear both anchor and hole of drilling debris and dust.
4. Insert the conical expander plug or wedge into the cutting end of the anchor and hammer the assembly back into the hole without rotating the hammer chuck. It is important only a hammer action is used.
5. Snap off the driving head of the anchor. A hand hammer may be necessary to do this in the case of larger anchors.
6. The anchor is now ready to receive a bolt or stud fixing just as if it were a normal socket anchor.

Expanding anchor (non-friction type)

A heavy-duty expanding anchor which is based on a rather different holding principle is the Leibig Ultra Plus. Instead of the expanding part of the anchor pressing against the sides of the drilled hole and inducing friction between the anchor and the mass walling base – this friction providing the holding power for the anchor – the Ultra Plus expands into an undercut in the concrete, com-

pressing the base upwards against the integral flange of the device set on the surface of the base. No expansion forces are directed into the concrete during setting or the application of load. This gives reliable load-carrying performance, and minimum spacing and edge distances of anchors are possible.

Description The Ultra Plus bolt is made of high-tensile steel with an integral stud (M8, M12 or M16 diameter). At the end of the stud is a round nut which supports the clamping segments. These are opened by a conical sleeve during setting – the segments opening into an undercut in the concrete. Variations in hole depth can be accommodated and embedment depths range from 95 to 190 mm.

Applications Ultra Plus bolts make heavy-duty fixings into a pre-drilled hole on a concrete base. This anchor provides strong, reliable fixings without exerting expansion forces on the concrete. It can withstand shock and dynamic loads, and reduced spacing and edge distances are feasible. Stand-off fixings can also be made using the Ultra Plus (Fig. 3.15).

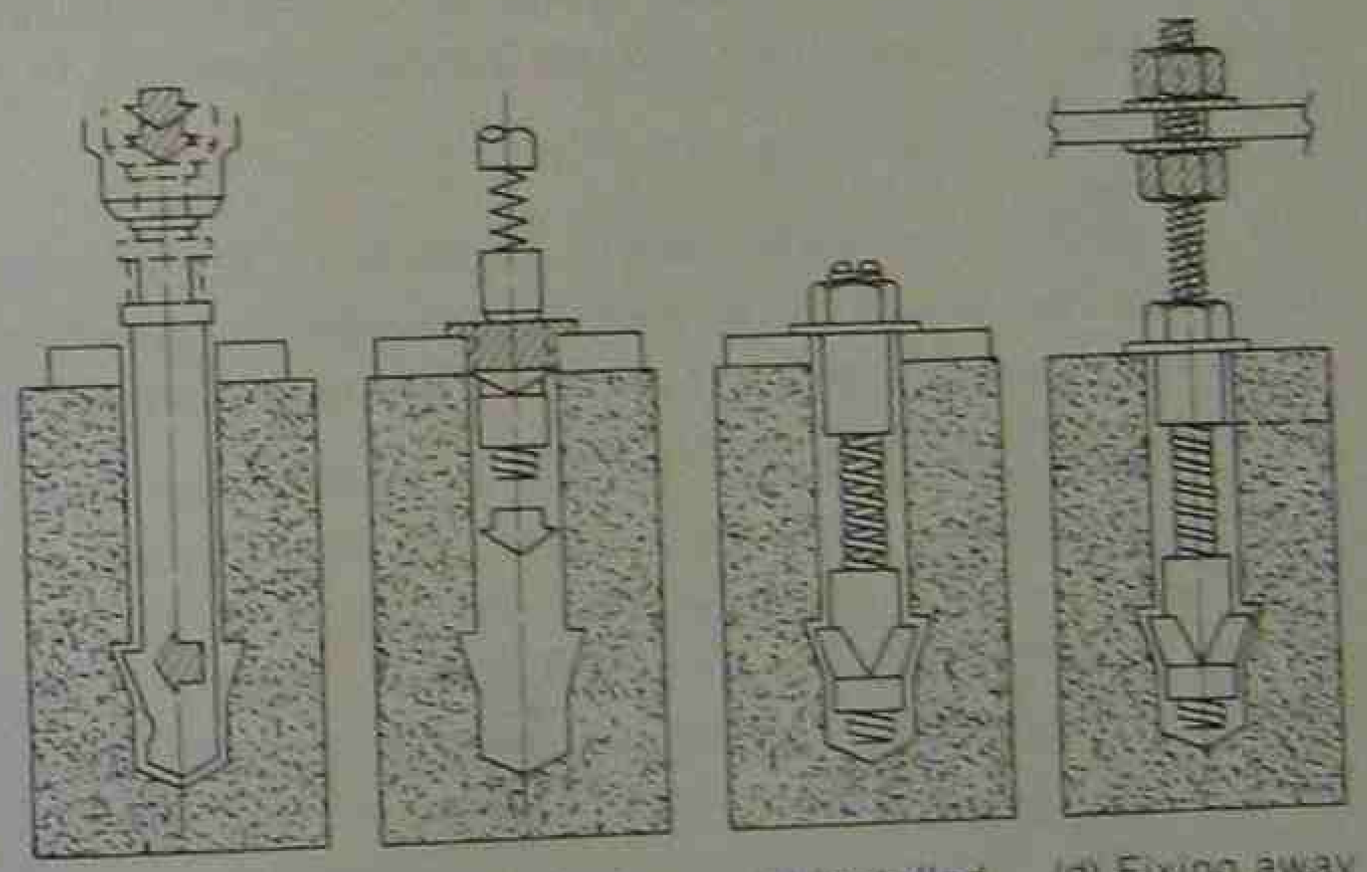


Fig. 3.15 Leibig Ultra-Plus anchor setting diagrams

Setting instructions

1. Drill a hole of the recommended diameter and depth. The

hole is to have an undercut rebate in the side of the drilling formed by a carbide undercutter supplied by the manufacturer.
2. Drill a clearance hole through the subsidiary component.
3. Pass the Ultra Plus through the component and into the drilling.
4. Run a nut onto the stud and tighten. This opens the clamping segments and sets the bolt.
5. Apply the specified torque.

Note: Stand-off fixings are placed in the same way, inserting the device in the hole so that its flange rests on the top of the concrete. Tighten the first nut to open the clamping segments and secure the device. Then run-on a second nut with a washer on top and set this at the required stand-off level on the threaded studding. Place the subsidiary component on the washer and fix with another washer and a third nut.

Specialized anchor devices

There are a number of other devices, developed for specialized applications, but which operate on a similar principle to the normal expanding anchors.

Ceiling suspension anchors These have been designed for use below concrete floors or roofs to provide a ring bolt or similar fixing for suspended ceiling systems. They make permanent, non-removable fixings (Fig. 3.16).

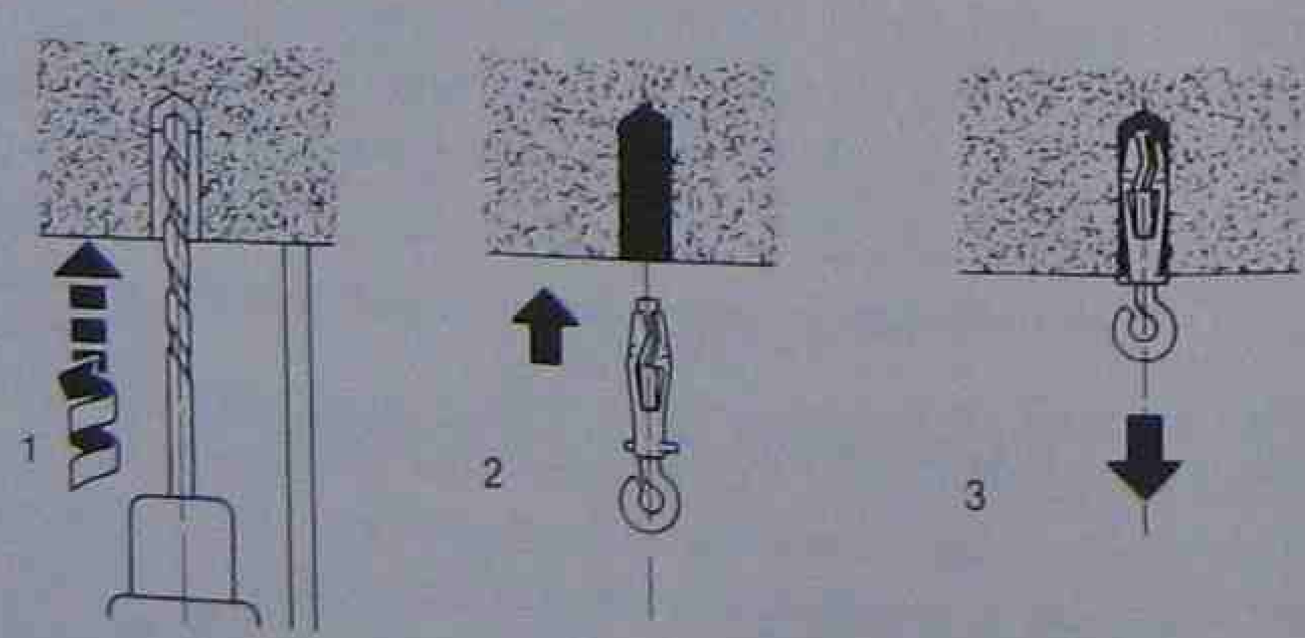


Fig. 3.16 Hilti HA8 R1 suspension anchor setting diagrams

Setting instructions

1. Drill a hole of the recommended diameter and depth.
2. Insert the anchor.
3. Sharply pull down the protruding ringbolt. This expands the anchor and sets it. A claw hammer or screwdriver may be used to pull the ringbolt downwards.

Another proprietary suspension anchor for ceiling duct or pipework fixing is the Fischer L8. Here an 8 mm diameter, 30 mm deep hole is drilled in the concrete soffit, the corrugated section of the anchor is inserted and the sleeve is hammered home to within 1 mm of the anchor plate (Fig. 3.17).

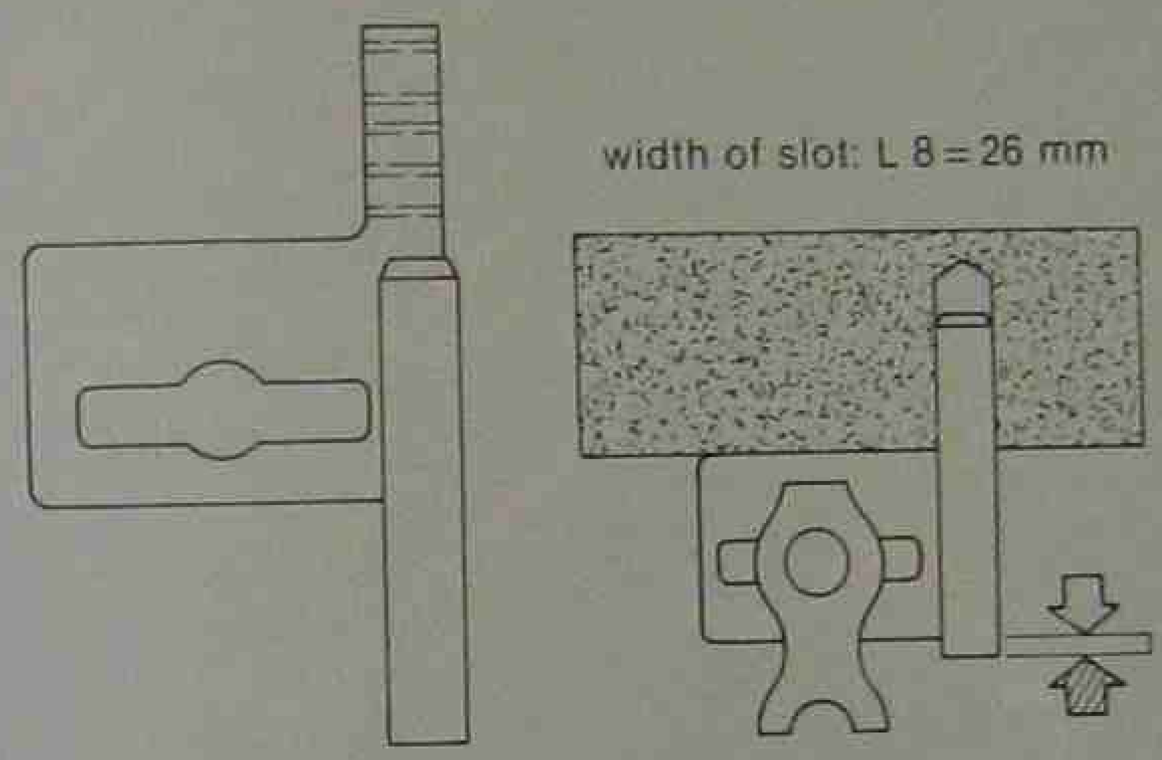


Fig. 3.17 Fischer L8 ceiling fixing

Insulation fasteners Used to secure non-self-supporting insulation, such as mineral wool, glass wool or expanded polystyrene, to mass walling, these lightweight devices have wide circular or star-shaped heads (Fig. 3.18). Usually they are manufactured from impact-resistant polypropylene and have a finned shank, which grips the sides of the pre-drilled hole in the base when the fixing is tapped home. Its holding power derives from the distortion of the fins.

Cavity wall repair anchors These special devices have been developed to replace corroded cavity wall ties without resorting to pulling down and rebuilding any part of the wall. There are several patterns which are based on the principle of the expanding anchor. Usually they consist of an expanding sleeve placed at both

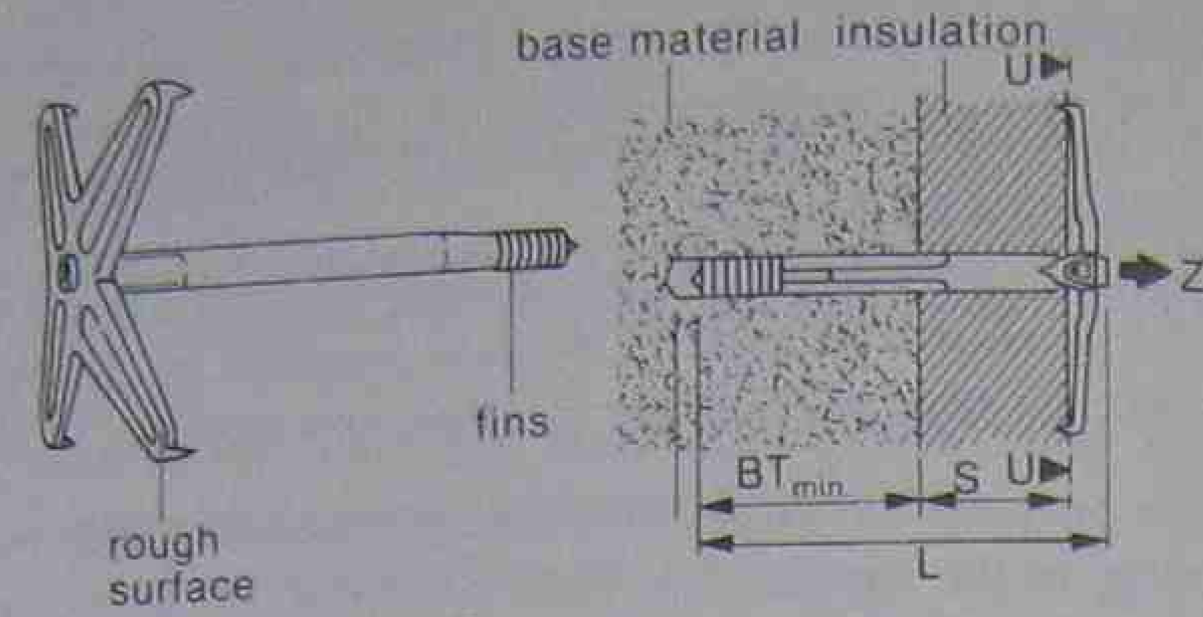


Fig. 3.18 Hilti IN insulation fastener

ends of a steel bar, sufficiently long to span the cavity. Cavity wall repair anchors are usually placed from outside the building, but it is possible to install them from inside if this is more convenient (Figs. 3.19, 3.20).

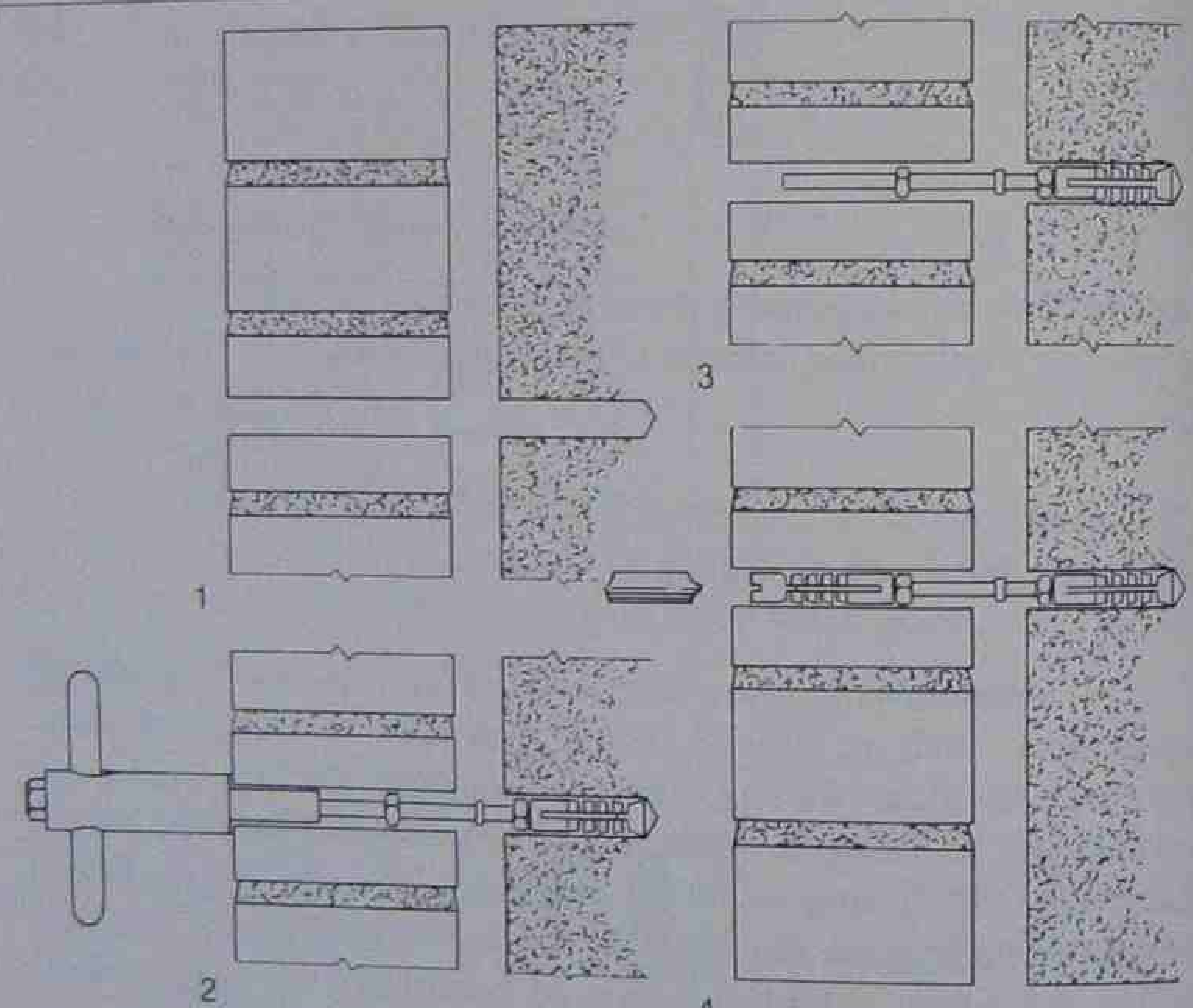


Fig. 3.19 Cavity wall repair anchor (Harris and Edgar);

Setting instructions

1. Drill the recommended diameter hole through the outer leaf of the wall and to a predetermined depth in the inner leaf.

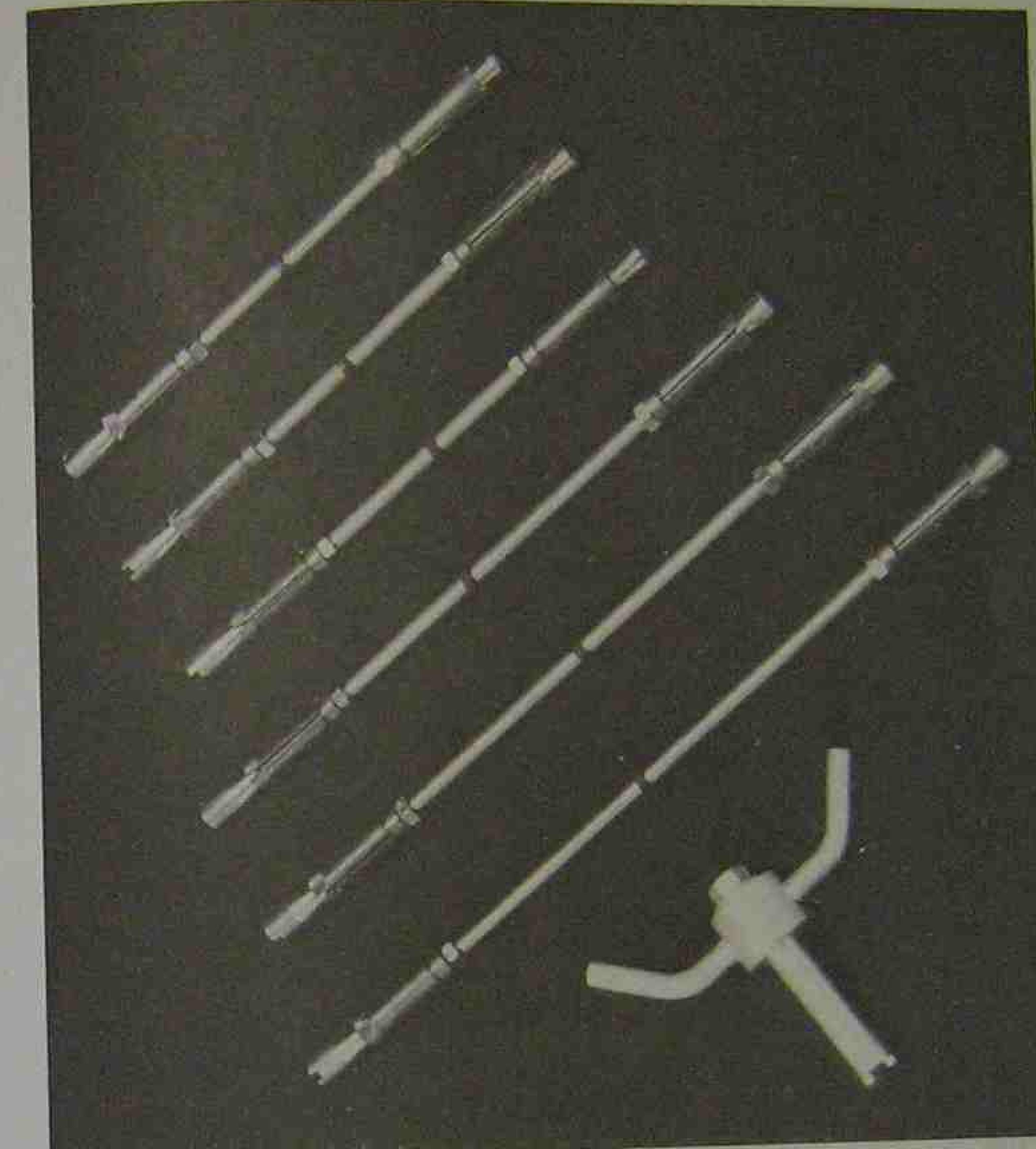


Fig. 3.20 Photograph

2. Insert the steel bar, complete with the inner expansion sleeve, through the outer leaf and to the correct depth in the inner leaf. Hand-tighten using the key provided. This expands the sleeve in the inner leaf. Complete the expansion by applying the correct amount of torque (usually 3 to 7 Nm) to the bolt head with a controlled torque device.
3. Remove the key and insert the outer leaf expanding sleeve over the end of the steel bar and tighten to the required torque using a screwdriver socket attachment, fixed to a torque wrench.
4. Make good the hole in the outer skin.

Note: Where the device does not have a drip collar in the centre of the cavity, the whole device should be set with a slight fall towards the outer leaf of the wall.

A2 Chemical anchors

These fixings are set in a pre-drilled hole, similar to expanding anchors, but they rely on an adhesive to provide their pull-out strength, not the expansion of the sides of the anchor against the face of the drilling.

Chemical anchors consist of two major parts: a resin cartridge (containing the resin and the hardener sealed within separate containers) and the anchor rod (a threaded stud) with its nut and washer. The anchor rod can be replaced by an internally threaded socket if a socket connection, rather than a threaded stud connection, is required.

A chemical anchor can be applied successfully in all locations where an expanding anchor could be used. In addition, because there are no expansion forces involved, it can make strong fixings closer to the edge of the base material than can an expanding anchor.

There is no present evidence to suggest the performance of this type of anchor deteriorates with age.

Further information on the use of anchors in various types of mass walling base, their positioning and other matters affecting their performance, are dealt with in the background section on page 63.

Description Chemical anchors are medium- to heavy-duty fixings which can be applied in dense and strong masonry bases to produce sound structural fixings. They usually consist of a glass phial of resin with a capsule of hardener contained within it. This is placed into the drilling and then a threaded stud, or internally threaded socket, of mild steel or stainless steel is forced into the hole, breaking the phial and mixing the resin and hardener. The fixing exerts no expansion stresses on the base and therefore can also be used for light-duty fixings close to the edge of even low-density concrete blocks.

Anchors are produced by various manufacturers to set studding from 8 to 30 mm diameter and of lengths varying from 110 to 240 mm.

Applications Chemical anchors can be used in all locations in which an expanding anchor can be used (i.e. in dense concrete, no-fines concrete, lightweight concrete with a dense texture, stone and brickwork made up of solid bricks without large internal voids). They can also be used in low-density concrete blocks, but high structural performance is only obtained in strong bases. The use of chemical anchors is not recommended in very porous or perforated masonry because the resin tends to become lost in the pores of the base. Fixings can be positioned more close to the edge of the base and generally these anchors can be used in positions subject to vibration or shock. They are unaffected by frost or weather and can be set in damp holes. Some can be installed under water. The drilling of a hole of precisely the correct diameter is essential for the fixing to achieve its full loading capability (Fig. 3.21).

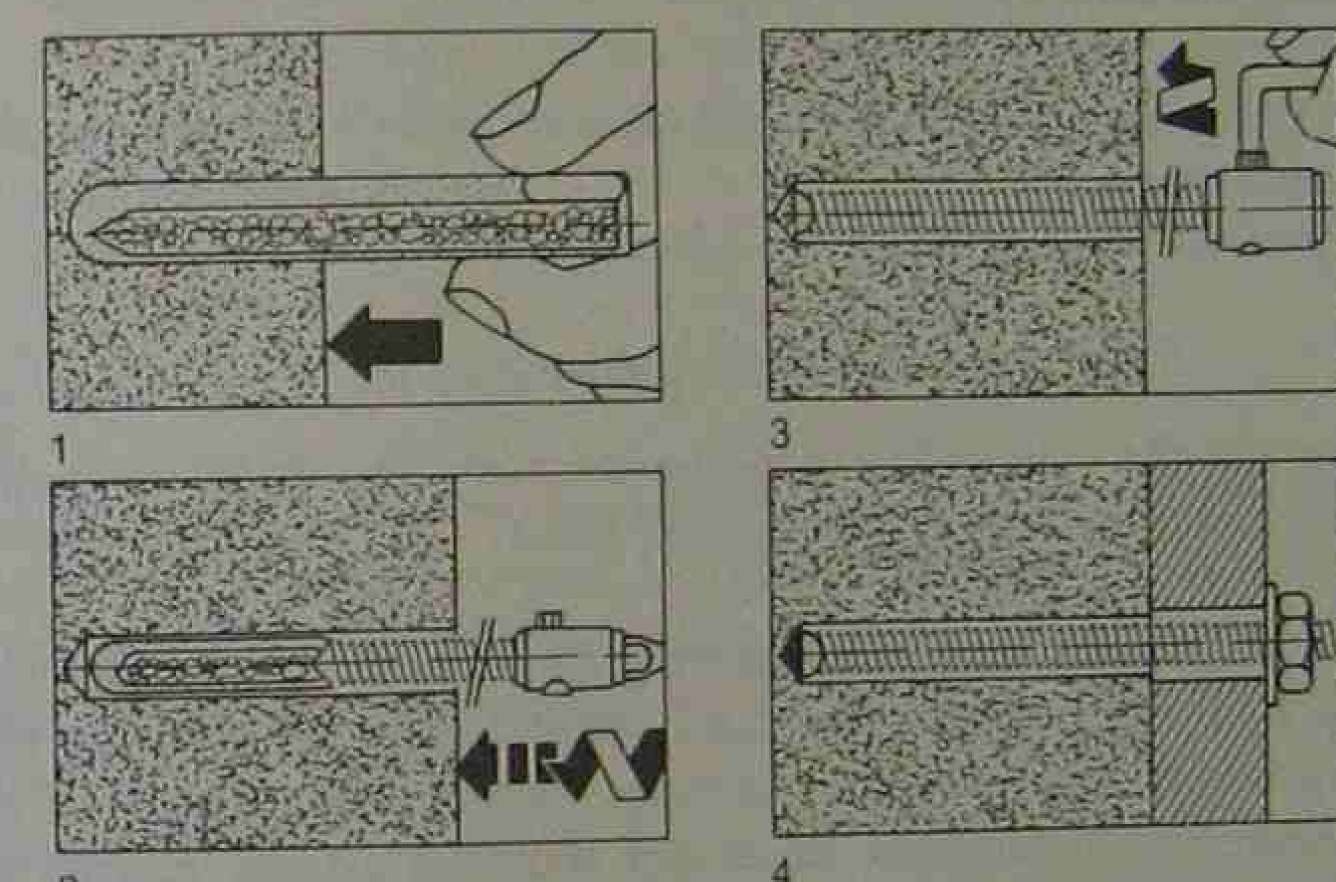


Fig. 3.21 Chemical anchor setting diagrams

Setting instructions

1. Drill a hole of the correct diameter and depth.
2. Insert the resin phial after cleaning the hole of debris and drilling dust.
3. Connect the threaded stud to the chuck of a drill by inserting the hexagonal drive bar in the end of the studding and fitting it into the chuck (in the case of a threaded socket, fit the cap screw and washer provided to the socket and insert the drive bar). Fit the drive bar into the

chuck of the drill. In some types of chemical anchor an adaptor is provided for the drill chuck.

4. Offer the stud (or socket) up to the hole, turn on the machine and force the stud (or socket) through the phial to the bottom of the hole.
5. Stop drilling as soon as the bottom of the hole is reached to prevent over-mixing. (Some studs have a setting depth mark on the shank.)
6. Leave the studding (or socket) in position until the adhesive has set. Then (in the case of the socket) remove the cap screw with a socket wrench.

Note: It is important that the drilling machine has sufficient power to drive the studding to its full depth. Rotary hammer machines are recommended. Do not use the hammer action without rotation.

Only use resin phials which are cool and do not expose phials to direct sunlight.

Specialized chemical anchors

As was the case with expanding anchors, the chemical anchor has been adapted for certain specific tasks.

Cavity wall repair anchors These devices have been developed to replace corroded cavity wall ties without resort to pulling down and rebuilding any part of the wall. There are several patterns available. The inner leaf fixing is usually carried out by chemical anchor, as described above, with an extended threaded stud which bridges the cavity and is gun-grouted into the outer leaf with thixotropic resin. Alternatively, both inner and outer leaf fixings can be made using gun-grouting. It is usual for these devices to be fitted from outside the building, but it is possible to carry out the work from inside if so desired (Fig. 3.22).

Setting instructions

1. Drill the recommended diameter hole through the outer leaf of the wall and to a predetermined depth in the inner leaf. Arrange the drilling so that it passes through a brick and not through a joint.

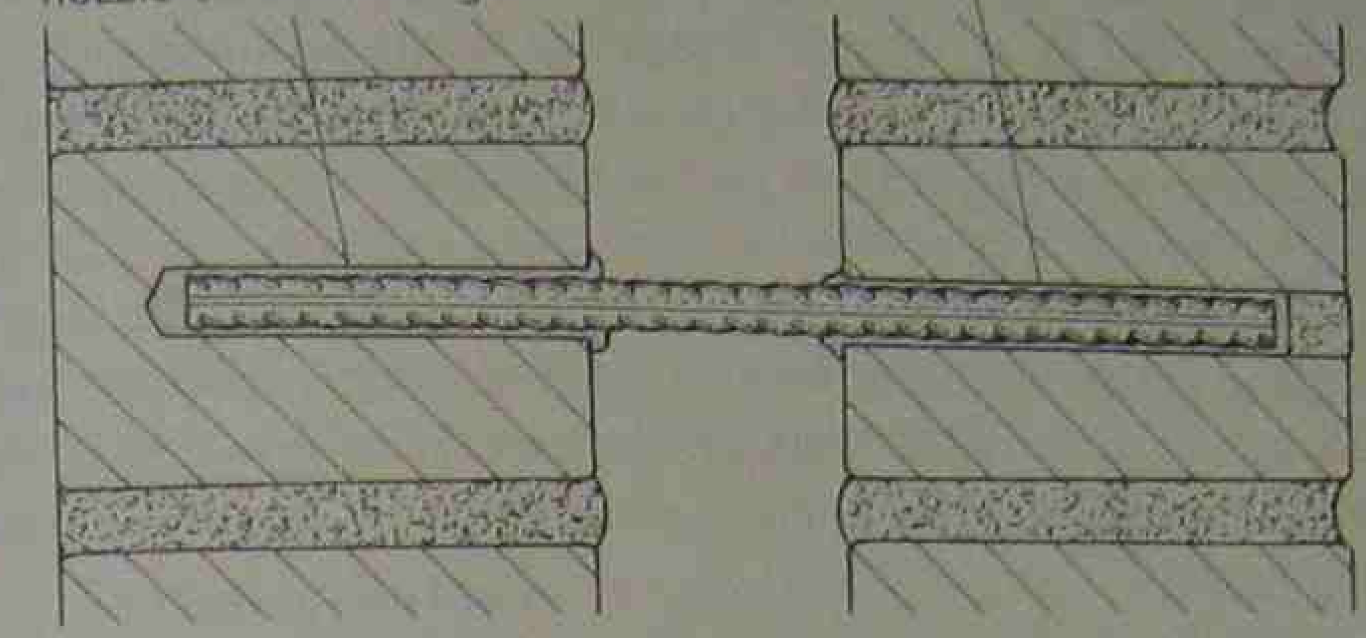


Fig. 3.22 Chemical cavity wall repair anchor

2. Insert the resin phial into the inner drilling and drive the extended studding into the inner leaf hole using a drill in the same way as the normal chemical anchor. Avoid over-mixing the adhesive.
3. Gun grout the studding in the outer leaf using a thixotropic resin.
4. Make good the hole in the outer brickwork.

Note: When the stud has no drip collar in the centre of the cavity, the whole device should be set with a slight fall towards the outer leaf of the wall. Resin systems can be used with success in damp brickwork, but should not be placed in rain-saturated brickwork.

A3 Injection anchors

The injection anchor is a fixing device which bears a resemblance to the chemical anchors described above. It, however, uses a quick-setting cement which is injected into the drilling through the device itself. It has been developed particularly for use in less dense walling bases.

Description Injection anchors are medium- to light-duty fixings for use in soft mass walling bases. They consist of a hollow metal body through which a quick-setting cement is injected. This seeps out through holes in the body of the anchor to fill an over-large (often dovetailed) drilling in which the anchor is set.

Injection anchors provide a threaded socket to receive bolt fixings from 8 to 12 mm diameter with insertion depths of from 9 to 25 mm; other types are made of nylon to accept woodscrews and coachscrews (4 to 6 mm diameters with anchorage depths of 50 mm). Some anchors, designed for use in bases with large cavities, have retaining nets to prevent loss of cement in the cavities.

Applications Injection anchors can be used in soft mass walling, such as aerated or pumice concrete, hollow brick or blockwork and friable brick or blockwork, to give fixings of predictable strength. They place no expansion forces on the base (Fig. 3.23).

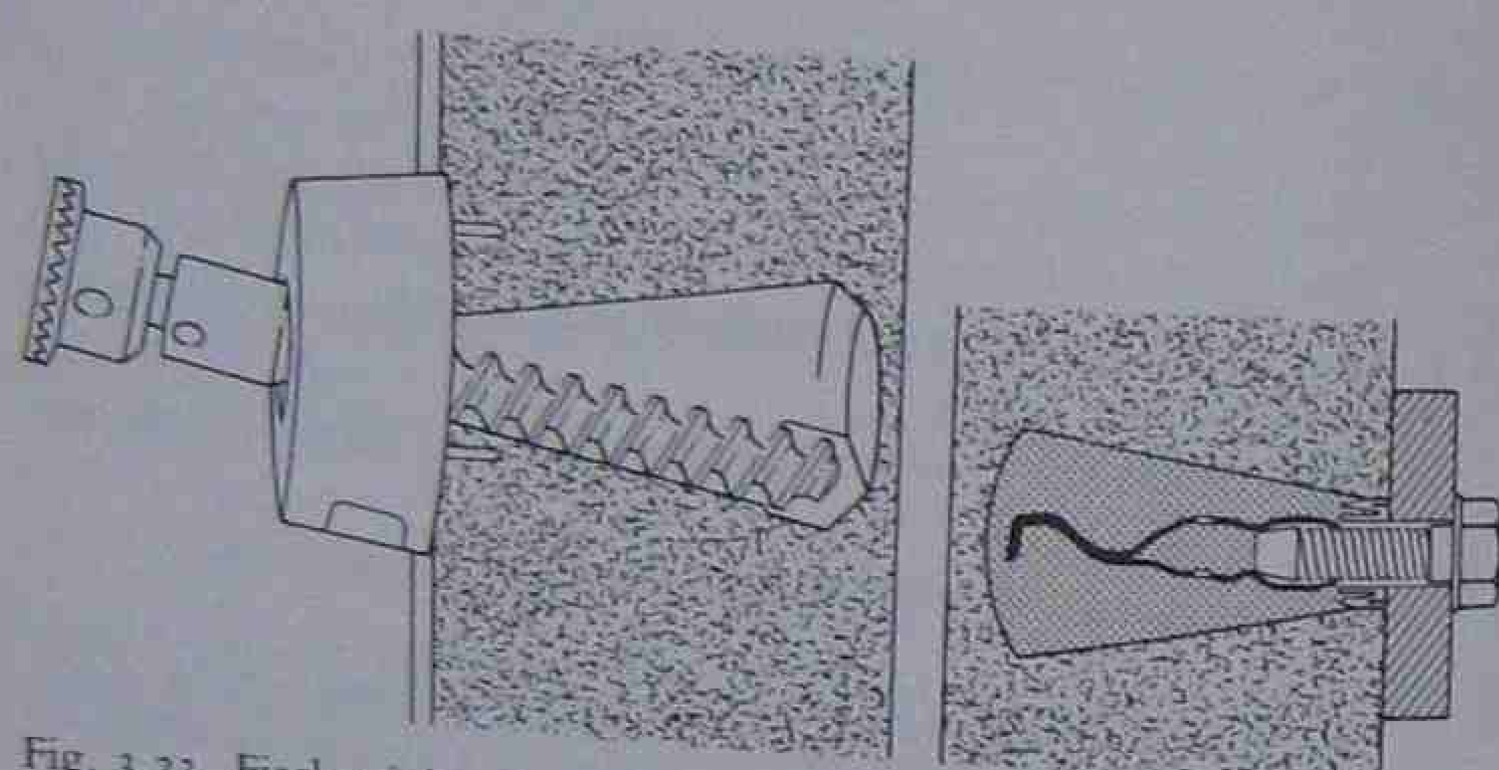


Fig. 3.23 Fischer injection anchor; drilling the hole and finished fixing

Setting instructions

1. Using a special drilling jig, drill a reverse-tapered (dovetailed) hole in the soft base. At first drill horizontally at slow speed until the four centering pins of the drill jig grip the base material. Then enlarge the hole into a reverse taper shape by making circular movements of the drill.
2. Switch off the drill before withdrawing the bit.
3. Insert the anchor until the sealing flange is flush with the wall surface. Insert the protective sleeve into the anchor. This avoids fouling the threaded socket during cement injection.
4. Stir the quick-setting injection cement using a whisk on the drill. The mixing proportions recommended by the manufacturer should be strictly followed.
5. Draw cement into the gun and inject it through the protective sleeve into the anchor body – and through the body into the wall cavities or pores.
6. Allow the cement to set.
7. Pull out the protective sleeve.
8. Fix the subsidiary component with a bolt (or screw).

Note: It is not necessary to drill a dovetailed hole in hollow bricks or blocks, or in friable material.

A4 Screw-in anchors

A fastener that is halfway between a drilled-for and a base-deforming fixing is the screw-in anchor. This requires the drilling of a highly accurate pilot hole, slightly smaller in diameter than the thread diameter of the screw, which is made from hardened carbon steel. This thread-forming screw is then driven into the pilot hole, using the same drilling tool, fitted with an adaptor.

Screw-in anchors have the advantage of making medium- to light-duty fixings without the need for plugs or expensive expanding anchors. The drilling can be performed through the subsidiary component to make a one-operation fixing. The subsidiary component can be removed, if required, after placing.

Description Screw-in anchor screws are made of hardened carbon steel with notched high threads which cut their own groove in the mass walling base. The fixing system comprises the anchor screws, a tool which is fitted to the drill to drive the screw and special highly engineered carbide-tipped drill bits to give consistent and accurate holes, (a most important factor with this fixing). These are supplied free by the manufacturers with purchases of a minimum quantity of screw-in anchors.

Screws have Phillips flat countersunk or hexagonal washer heads and range from 25 to 125 mm lengths in two diameters, 5 and 6 mm.

Applications These devices can be used in most solid mass walling bases where quick fixings for metal or timber subsidiary components (maximum thickness 100 mm) are to be made. Both components and base can be drilled in the same operation. Embedment depth should be between 25 and 37 mm. The longer

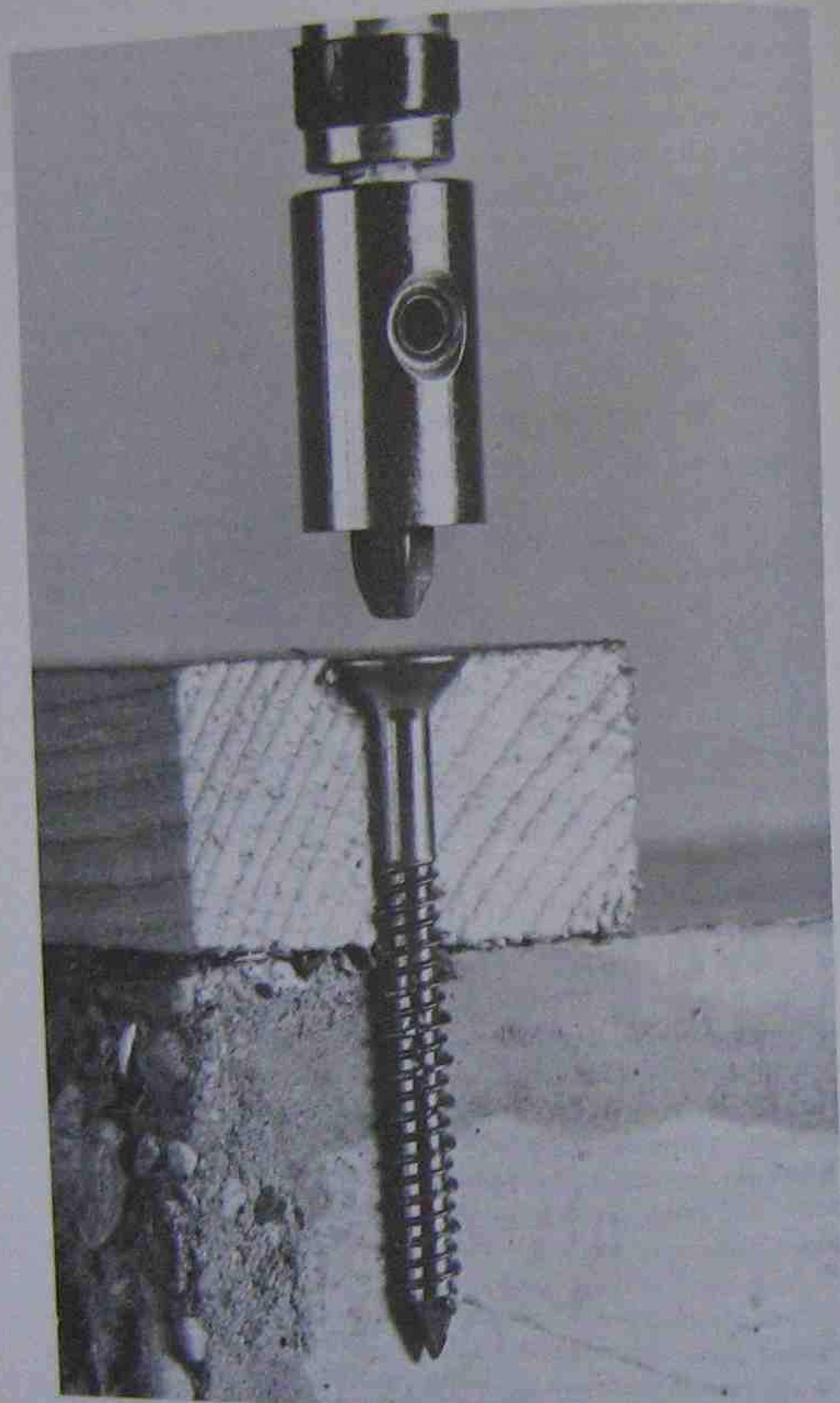


Fig. 3.24 Buildex Tapcon

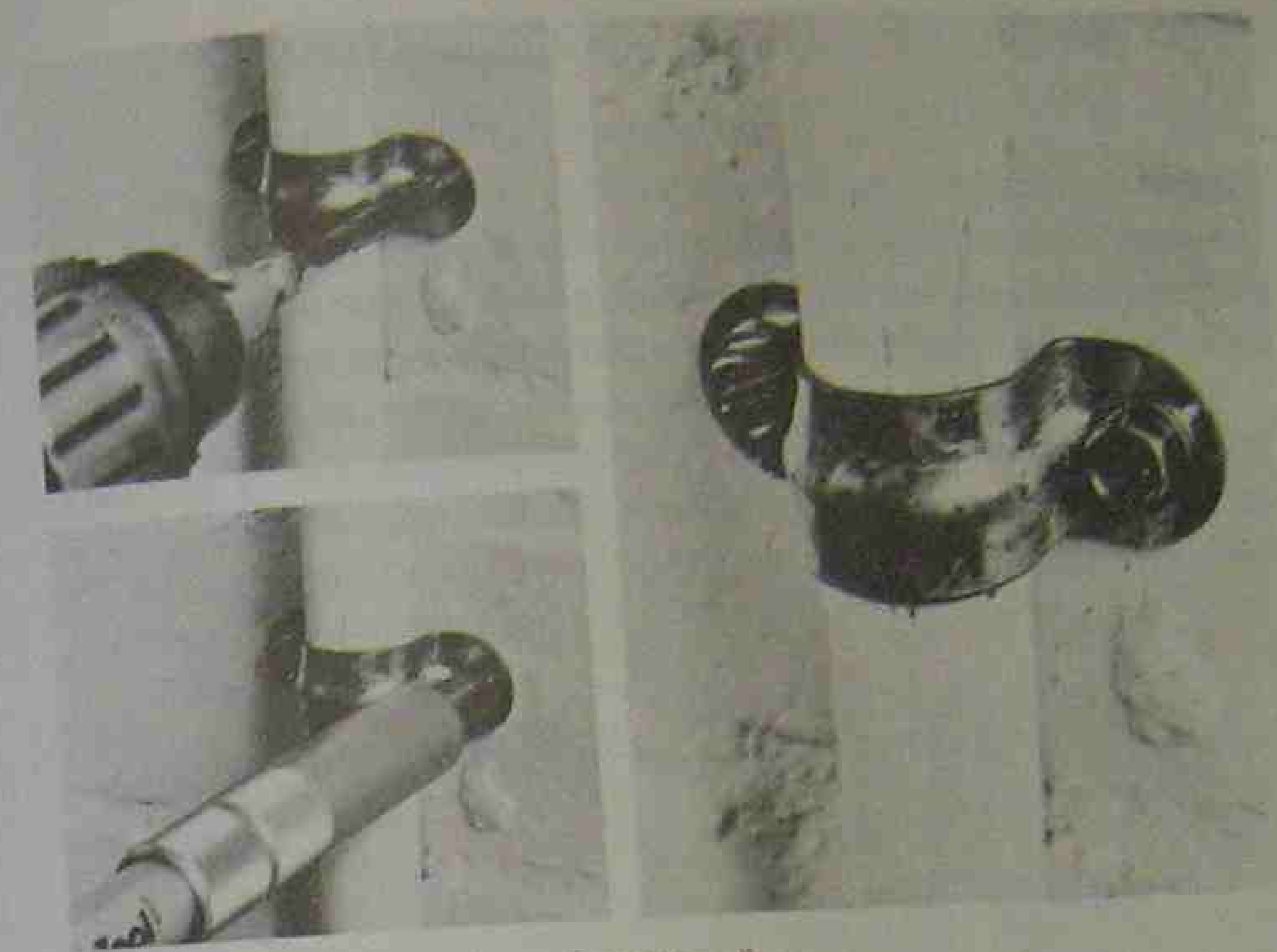


Fig. 3.25 One application of BIF Confas

screws in the range are intended for use with the thicker components (Figs 3.24, 3.25).

Setting instructions

1. Drill hole, using the drill bit provided in a hammer drill, through the subsidiary component and into the mass walling base. The depth in the walling should be at least 6 mm deeper than the embedment depth of the screw (embedment depth minimum 25 mm; maximum 37 mm).
2. Snap onto the drill the driving tool with the correct head style and drive the anchor into the hole, while holding the subsidiary component in position. An adjustable depth-sensing nosepiece avoids over-driving the screw anchor. This operates much like a clutch and throws the drill out of gear on contact with the component.

Note: The accuracy of the pre-drilled hole is essential to the good performance of this type of fixing. It is advisable to use the drill bit provided by the manufacturer and replace it regularly as it becomes worn.

Section B: Base-deforming fixings – type 2(b)

Base-deforming fixings in mass walling consist of various types of masonry nail, either driven manually or by means of an explosive or compressed-air tool, and a small group of hammer-in anchors which are used exclusively in aerated concrete bases.

The base-deforming fixing compresses the material of the base into itself as the nail is driven. The compression forces so generated produce friction on the shank of the nail, which resists its withdrawal (Fig. 3.26).

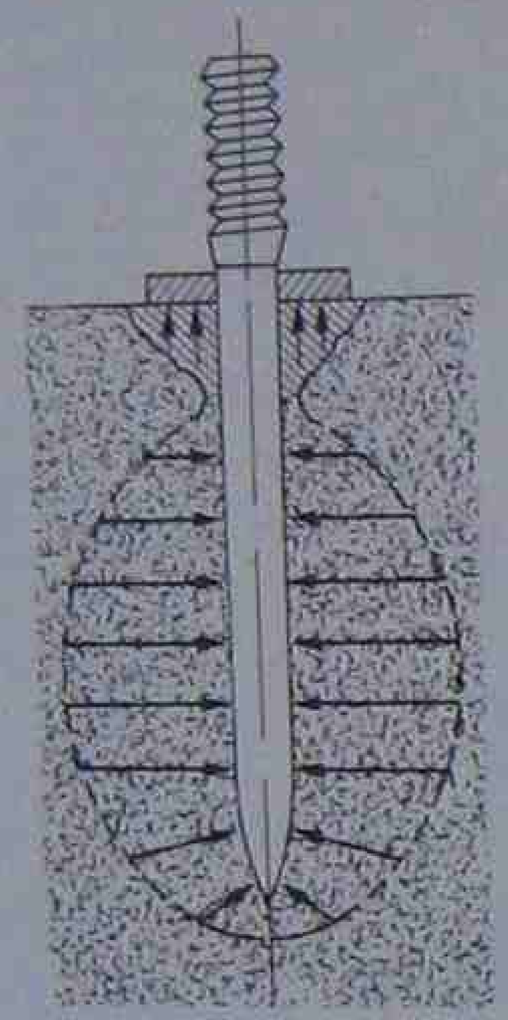


Fig. 3.26 Masonry nail in mass walling; forces generated

For a long time it was considered that, because mass walling bases (unlike timber) were not fibrous, nail fixings into mass walling could never be more than lightweight and relatively unreliable. With the introduction of first the explosive cartridge tool (which we will refer to as the powder-actuated tool) and then the pneumatic tool, the limited performance and application of the masonry nail was immensely increased.

As in the case of drilled-for fixings, there are a number of factors affecting the performance of base-deforming fixings; factors such as the strength of the base material, the way the fixing is driven and the strength of the fixing itself.

B1 Masonry nails

Description These nails require no prior preparation to the base. They are high-quality zinc-plated steel nails which are hand-driven through a subsidiary component (usually wood or light-gauge metal) into relatively soft mass walling bases to make light-duty fastenings. Diameters are usually within the range from 2 to 3.7 mm diameter and up to 100 mm long.

Applications Masonry nails are used to make light-duty fixings into soft brickwork, lightweight concrete with a dense texture and soft stone (Fig. 3.27).

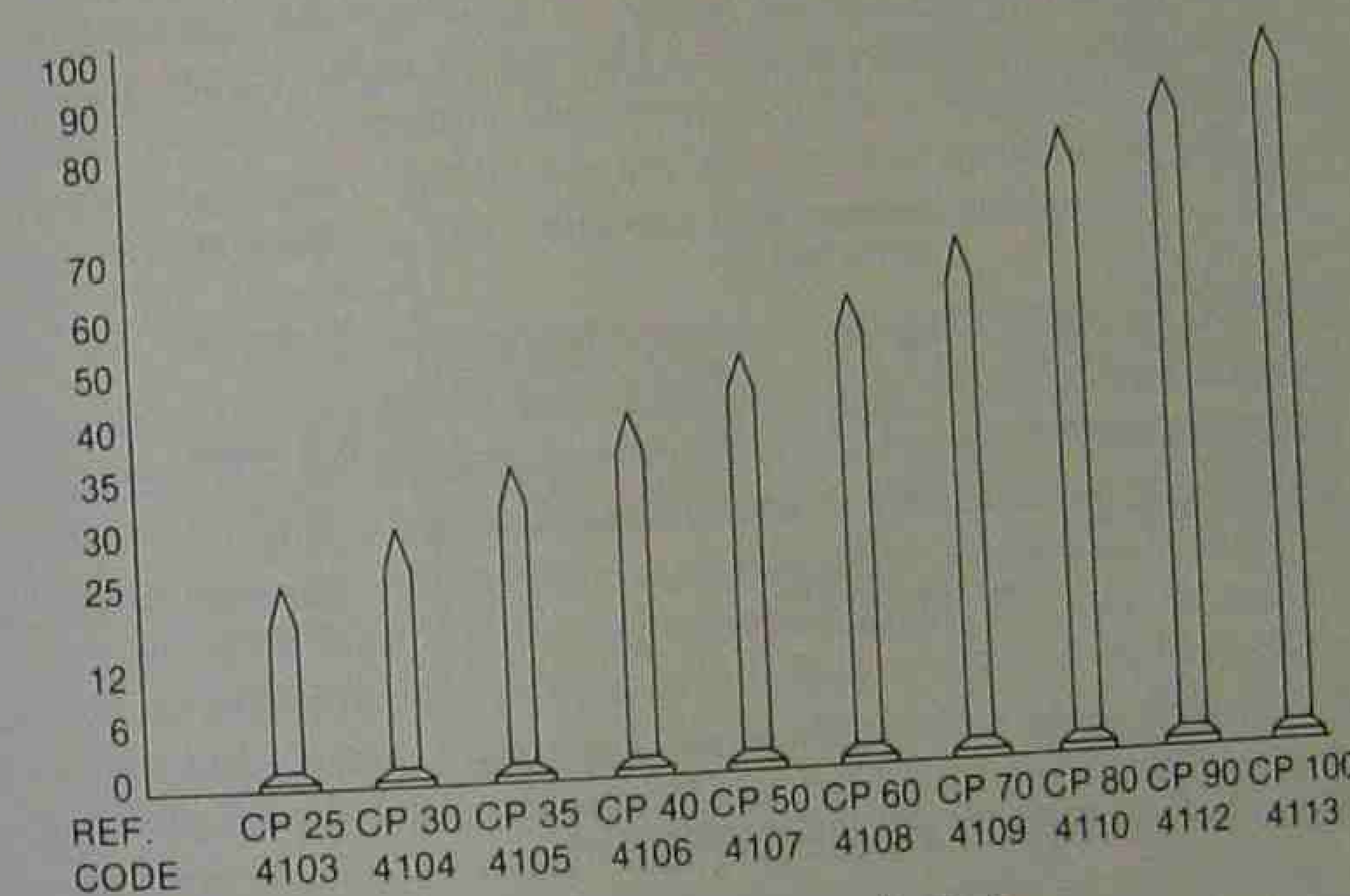


Fig. 3.27 Masonry nails; heavy-duty range by Spit

Setting instructions

1. Drive the nail with short positive hammer blows through the subsidiary component (maximum thickness of a wood component is 50 to 60 mm) to a maximum depth in the base of 20 mm.
2. Nails should be placed between 200 and 300 mm apart.
3. Avoid nailing into mortar joints.
4. Only place one nail in each brick if possible.

Note: It is wise to wear safety glasses while hammering.

B2 Powder-actuated fasteners

These fixings require no prior preparation of the base, as do expanding anchors or plugs. There is no pre-drilling and the fixing is merely driven into the base by an explosive charge from a cartridge in the fixing tool or gun. Fixings take two forms (Fig. 3.28):

(a) galvanized steel drive pins for direct firing through the subsidiary component to create a permanent fixing to the base in one operation;
(b) galvanized steel threaded studs for firing into the base to give detachable fixings for the subsidiary component. There are accessories such as ring or eye couplings, clamps and hooks which can be used with threaded studs.

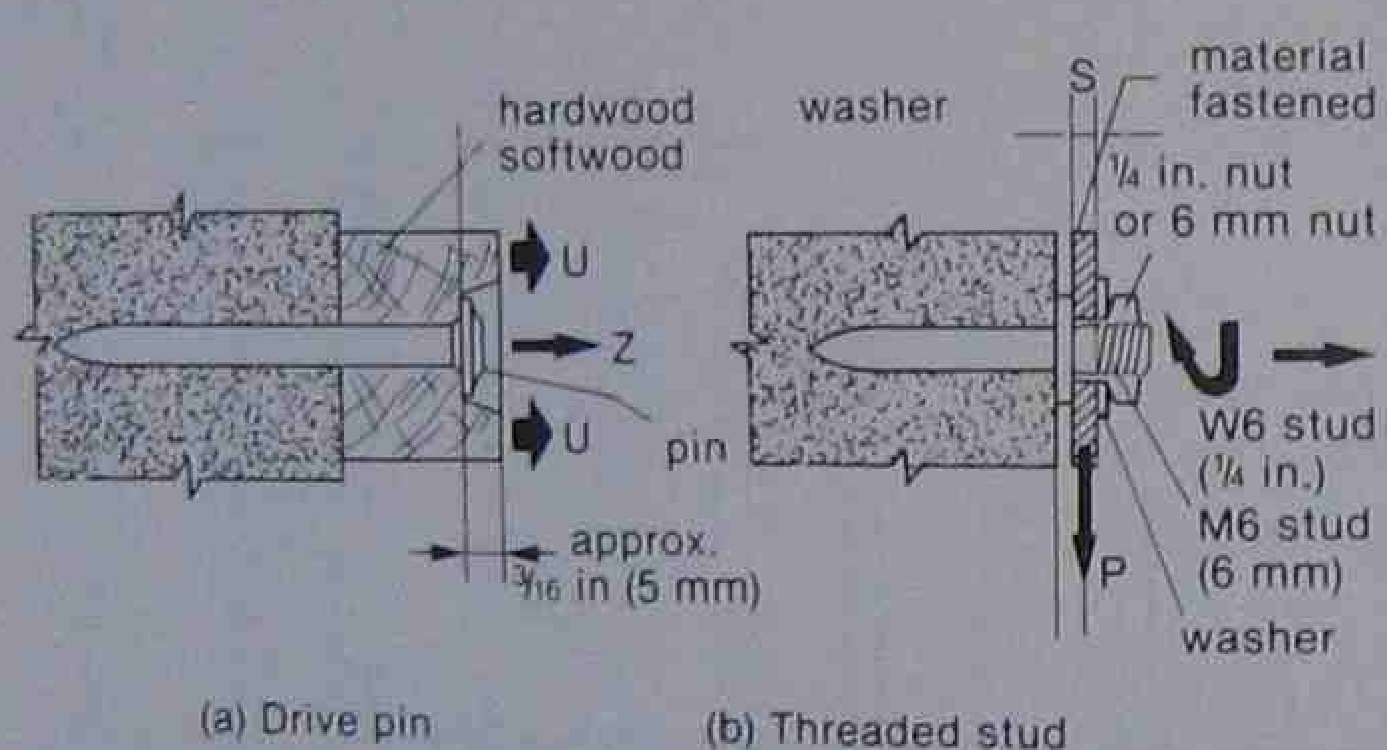


Fig. 3.28 Fired masonry nail and threaded stud

Both types of fixing achieve their holding power by friction against the shank of the pin.

Typical subsidiary components which can be fired through to make a direct, one-operation fixing to mass walling include:

(a) wood (softwood or hardwood);
(b) steel or aluminium sheeting or brackets;
(c) soft sheet materials (insulation board, wood wool, cork, etc.) usually employing an additional washer to avoid punching through.

Different sizes and shapes of drive nail are produced by the manufacturers to match the character and thickness of the subsidiary component material.

Further information on the use of powder-actuated fixings in various types of mass walling base, their positioning and other matters affecting their performance, are dealt with in the background section on page 63.

Description Powder-actuated fixings are drive pins or nails of hardened (austempered) steel fired from a tool by an explosive cartridge to make a permanent fixing into a mass walling base. Pins are produced with various head and shank styles to suit the material with which they are used. Shank diameters are 3.5 or 4 mm and lengths are up to 100 mm. Plain headed pins are intended to fix the subsidiary component direct to the base. Threaded stud-headed pins are in diameters from 6 to 10 mm. These are intended to provide a firm anchorage to make removable connection points for the subsidiary component. There are various colour-coded strengths of cartridge available to give the necessary force to drive the pin into various bases.

Applications Powder-actuated pins can make medium- to heavy-duty fixings in dense concrete, lightweight concrete with a dense texture, brickwork and stone, such as sandstone, soft dolomite and shaly bedded rock. Fired fixings are not suitable for dense concrete of extreme strength (over 60 N/mm^2), no-fines concrete, aerated concrete, perforated or porous clay or calcium silicate bricks (below 10 N/mm^2), hollow clay blocks or hard calcite stone and granite. Care needs to be taken when using these fixings with hollow bricks, calcium silicate bricks and hollow lightweight concrete blocks. In these cases preliminary trial fixings should be made if the performance of the fixing is critical.

The success of these fixings depends on the type of fixing tool, positioning of the pins and the thickness and type of the subsidiary component.

Fixing tools

Because the drive pins are considerably harder than the base material into which they are driven, the powder-actuated tool has to ensure that the pin is aligned and guided throughout driving to prevent sideways deflection.

There are two types of driving tool: one direct-acting, the other indirect- (or piston-) acting.

The direct-acting tool gives high velocities with entry speeds up to 500 m/sec and the pin is driven by the expansion of gas as a result of the firing of the cartridge. The fastener propulsion and depth of penetration is not controlled.

The indirect-acting tool operates by released gas acting on a piston in the barrel of the tool. This drives the pin into the base under controlled conditions at velocities up to a maximum of 100 m/sec. The piston is retained within the tool, being stopped at an appropriate point to give the required depth of penetration.

Both types of tool have safety devices which do not allow firing except when the barrel is in firm contact with the base. If the tool is 6° or more out of perpendicular to the base, one type of tool will not fire.

The depth of penetration into the base should be within 22 and 32 mm and this can be achieved by adjusting the strength of the cartridge until the penetration is correct.

Powder-actuated tools are sophisticated pieces of equipment and should be used with respect. Tool, pin and cartridge should all be elements of the same fixing system, their quality and sizing being designed to guarantee safety for the operator and successful fixing performance. There is no room for operator improvisation.

Some manufacturers offer training courses in the use of their equipment and the Powder Actuated Systems Association produces basic safety information. Operating instructions should be followed meticulously: only fasteners and cartridges compatible with the tool should be used and only skilled operators should have charge of the equipment.

Positioning of pins

The minimum distance between the pin and the edge of the component should be 50 mm. The minimum spacing of pins should be two times the penetration depth of the pin.

Thickness and type of subsidiary component

When the component is fixed direct to the mass walling, its thickness and type is critical to the success of the fixing. If the material is too thin, the pin may punch through (as in the case of thin metal) or splinter (as in the case of wood). If the material is too thick, the fastener could bend, or not penetrate the base sufficiently, even using a pin with the longest shank.

Manufacturer's data sheets give full details of the thicknesses of subsidiary components which can be fixed using various lengths of fastener.

When soft materials are being fixed, such as insulation board or

thin metal sheets, with the risk of pull-over failure or firing through, additional washers may be required.

Setting instructions

Always follow the manufacturer's operating instructions minutely.

1. Select the length and type of pin required from the manufacturer's data, bearing in mind the material of the base and the type and thickness of the subsidiary component.
2. To ensure the correct cartridge is used to give correct penetration depth, commence with a test firing, using a low-strength cartridge. If the pin does not penetrate the base sufficiently (22 to 32 mm) change to a higher-strength cartridge. The various strengths of cartridge are colour coded. (As a general guide, the harder the concrete, the less penetration of pin is required; the softer the concrete, the greater the penetration and the longer the shank length required.)
3. Place the tool firmly against the base (or, if firing through the subsidiary component, the component), ensure the tool is at right angles to the base, and fire.
4. When firing into concrete or stone, the use of a spall stop is advisable (Fig. 3.29). This is a steel ring on the end of

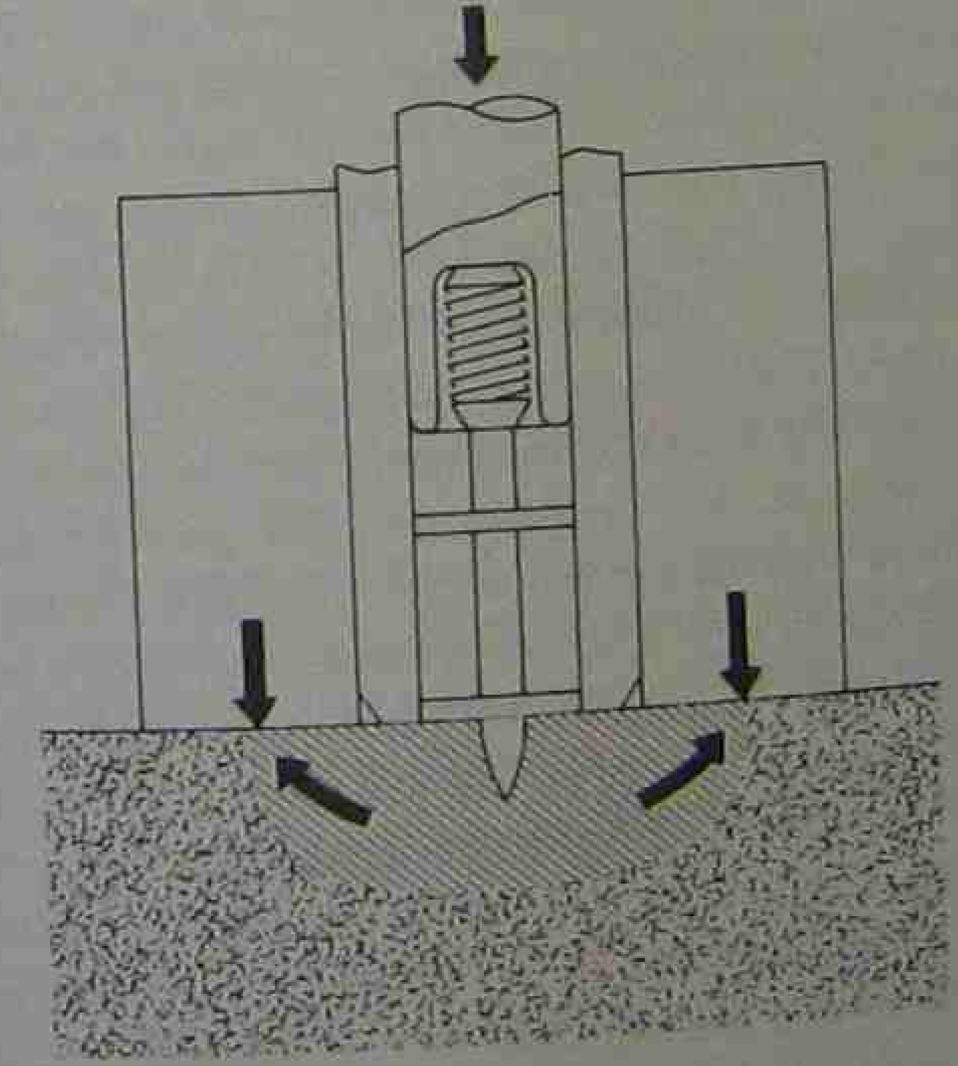


Fig. 3.29 Spall stop

the tool which exerts a supporting downward pressure on the surface of the base during firing. This encourages the compression of the base (and improves holding power) and reduces the danger of spalling.

Note: It is wise to wear safety glasses while making powder-actuated fixings and, in enclosed surroundings, ear protection too. When working on thin partitions, it is recommended that the space on the other side of the partition is fenced off in case of firing through.

Further safety advice is contained in the Health and Safety Executive guidance note PM14 *Safety in the Use of Cartridge Operated Fixing Tools*, the Powder Actuated Systems Association's *Guide to Basic Training* and in *Site Safety* by J. C. Laney in this Site Practice series.

B3 Pneumatic-actuated fixings

These fixings are identical to powder-actuated fasteners; the only difference is in the method of driving the pin. All parts of the preceding section can therefore be considered as applicable to pneumatic-actuated fixings, with the exception of that part dealing with the fixing tools.

In the case of pneumatic-actuated fixings, the pin is propelled by a piston driven by the release of compressed air through the tool. This gives a low-velocity, controlled impetus to the pin, unlike the direct-acting powder-actuated tool. Clearly the disadvantage of a pneumatic fixing tool is that it needs a compressor on site with a long air line connected to the tool. If this facility is available, however, the speed of placing pins can be greater than with a powder-actuated tool. The pins in the pneumatic tool are collated into strips of ten and fed into the magazine two strips at a time.

For the larger project, the pneumatic tool has productivity advantages. To these can be added other benefits, such as less recoil and less noise on firing. The air pressure can easily be regulated through a range from 6.2 to 10.5 bar to give minute adjustment to the penetration of the pins – a less clumsy process than the test firings with a variety of different-strength cartridges often needed to achieve the correct penetration of powder-actuated pins.

Although both types of tool have safety devices to prevent accidental firing, there is a feeling that the pneumatic tool is safer to use largely because of the absence of explosive cartridges. It

does, however, tend to have mobility problems due to the need for an air line. This has led to the use of compressed air cylinders for smaller fixing jobs. Pressure cylinders, similar to those used in the aqualung, can be carried on the operative's back allowing approximately thirty fixings per cylinder to be made.

Note: The wearing of safety glasses when using pneumatic-actuated tools is advisable.

B4 Hammer-in anchors

This small group of mass walling fixings has been introduced to make speedy and relatively light-duty fixings to low-density walling materials without the need for drilling.

Description Hammer-in anchors are made of zinc-plated steel with standard, looped, countersunk screw or hexagonal nut styles of head. Diameters of 5 and 6.5 mm are available and lengths from 50 to 150 mm.

Applications These anchors are designed specifically for making quick, low-cost and relatively light-duty fixings in aerated concrete without the need for drilling the base. The holding power of the fixing will be affected by the composition, density and compressive strength of the aerated concrete. Fixings can be made straight through softwood subsidiary components up to 75 mm thick, otherwise the component should be drilled (Fig. 3.30).

Setting instructions

1. Pass the anchor through the subsidiary component (if pre-drilled).
2. Insert the pin of the setting gauge into the hole in the head of the anchor.
3. Hammer the head of the gauge until the stop of the gauge comes into contact with the component.
4. Remove the gauge and drive the anchor the rest of the way home with direct blows of the hammer. This action will splay out the anchor sleeve over the expander in the base.
5. In congested situations, where positive hammer blows are difficult to deliver, a special setting tool, supplied by the manufacturer, can be used.

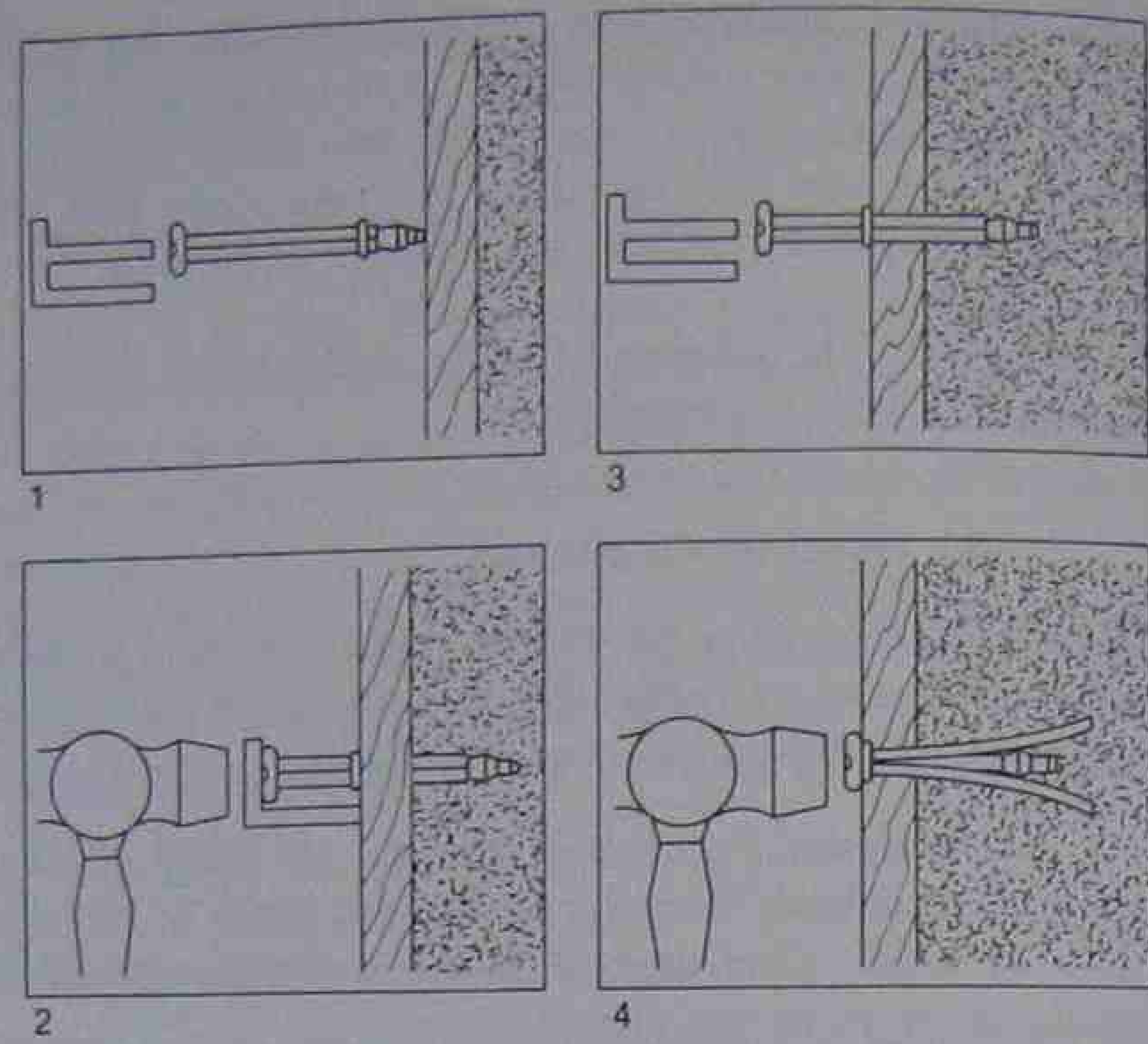


Fig. 3.30 Loden hammer-in anchor setting diagrams

Note: It is possible to remove standard and looped-head types of anchor by inserting a long, loose-fitting nail (supplied by the manufacturer) into the hole in the head of the anchor and driving the expander, within the anchor, free of the anchor sleeve. The sleeve can then be withdrawn, using pliers. Some damage may occur to the component and the surface of the base material.

Section C: In situ fixings – type 2(c)

Unlike the two preceding groups of fixings (types 2(a), 2(b)) in situ, devices are placed at the time of constructing the mass walling base. Their performance depends, as did the previous categories, on the bond between the fixing and the base, as well as the character of the base itself. In addition, because the in situ fixing is placed at the time of constructing the mass walling base, its success also depends on the accuracy of its positioning in the base and its maintenance in that position until the base materials have hardened. This requires considerable foresight in the construction process and care to protect the fixing during and after construction.

C1 Built-in fixings

Built-in fixings include such items as joist hangers and anchorage straps, designed to connect a subsidiary component to a mass walling base, and those used, like cavity wall ties, cramps and dowels, to connect individual parts of a mass walling base together. The former type replaces the older practice of building in subsidiary components, or ledging them on a corbel formed in the walling. The latter are present-day examples of a long line of historic lead cramps and iron dowels in masonry. All these achieve their holding power by adhesion between the fixings and the mortar in which they are bedded and the weight of the walling above the fixing.

Because of the need to bed these devices, their positioning is limited by the coursing discipline of the walling.

In the case of some fixings of this type, manufacturers produce tables of safe working loads for the device when used in specific types of walling material.

Further information on the use of in situ devices in various types of mass walling base and matters affecting their performance are dealt with in the background section on page 63.

Joist hangers

Description These are saddles, usually made of 2.7 mm thick galvanized steel with a build-in flange (sometimes fish-tailed). They are shaped to form a seating for either one timber floor or roof joist, or two similar joists on opposite sides of a partition wall (Fig. 3.31). Various sizes of hanger are available to suit joist sizes, from 100 to 250 mm deep and from 38 to 150 mm wide.

Applications Joist hangers are used to make connections of predictable engineering performance between timber joists or beams (installed after the building of the wall) and walling made up of coursed elements, such as bricks or blocks. These should not be confused with joist hangers without the build-in flange and designed to be connected to the base by other anchorage devices, such as expanding anchors. Such hangers are accessories to the main fixing device (see section D, p. 61). Also there is a range of similar hangers used to make timber-to-timber connections (see Ch. 4).

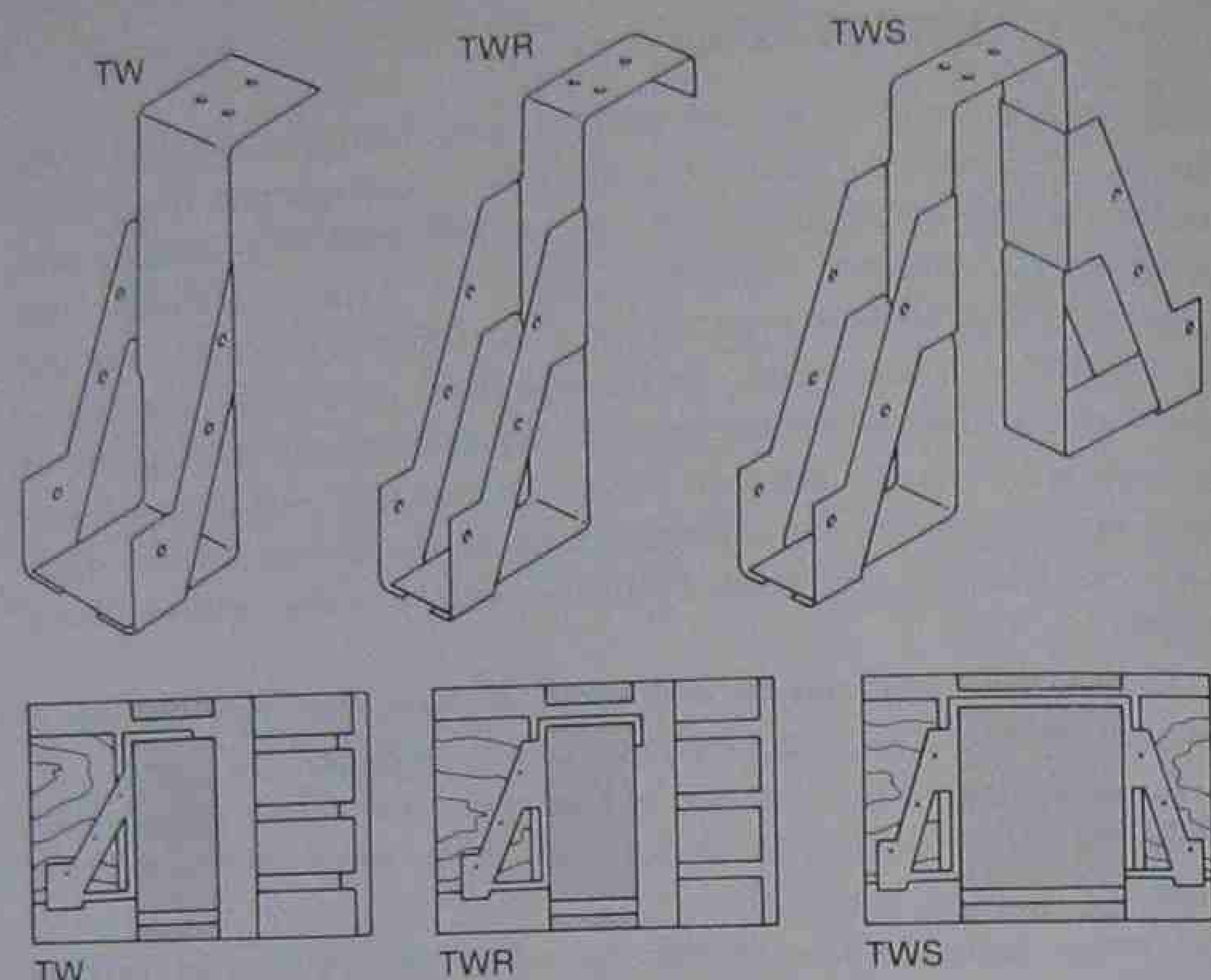


Fig. 3.31 Three versions of the Catnic TW joist hanger

Setting instructions

1. Joist hangers should be accurately positioned along the appropriate walling course, their flanges being placed on top of the course. Ensure the flanges are adequately surrounded by the mortar of the bedding joint.
2. Do not load the hangers (particularly single hangers) until their flanges are secured by higher lifts of walling and the mortar of the joint has completely hardened.
3. Later set the joist in the saddle and fix to the hanger with 30 mm × 375 mm square twisted sherardized nails through the pre-punched holes in the hanger.

Anchorage straps

Description There is a variety of patterns for these straps which are designed to withstand the effects of wind pressure on the structure and perform general bracing duties. They are mostly manufactured of 5 mm galvanized mild steel for horizontal restraint applications and 2.5 mm for vertical restraint. They are pre-punched (usually at 12.5 mm centres) to receive nail fixings to the

subsidiary timber components and there is a variety of shaped ends for building into the walling (Fig. 3.32).

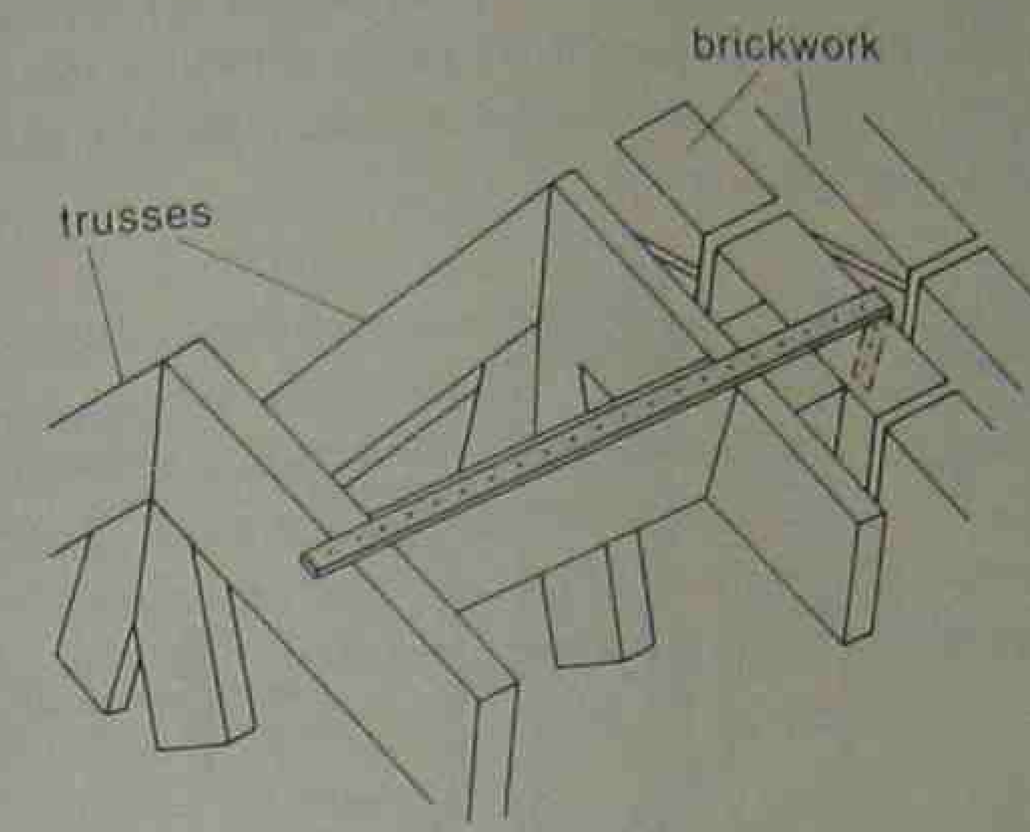


Fig. 3.32 Bev strap bracing roof trusses

Applications Anchorage straps are used to provide stiffening and lateral restraint to roof or floor joists running at right angles or parallel to the wall (horizontal restraint) or holding down wall plates and trusses (vertical restraint). Lateral restraint straps are sometimes used in combination with joist hangers.

Some vertical restraint straps are not built into the wall but are connected to it by masonry nails or screws into plugs. These straps are accessories to the fixing device (see section D, p. 61).

Setting instructions

1. Straps should be accurately set out as required by the building design, ensuring that their flanges are adequately surrounded by the mortar of the bedding joint.
2. Only attach the strap to the timber members when the mortar has had sufficient time to harden.
3. Nail connections to the joists should be made using 30 mm × 375 mm square twist sherardized nails.

Cavity wall ties (and other wall ties)

Description Galvanized steel, stainless steel or plastic wall ties are produced in a variety of patterns and sizes for tying the two leaves of a cavity wall together, or tying a thin stone or precast concrete facing to a backing wall. Cavity wall ties are most commonly produced in two forms: the galvanized wire butterfly tie, and the twisted fish-tail flat tie made from galvanized steel strip. Today these have been joined by polypropylene ties and ties with retention devices to hold insulating batts within the cavity against the inside leaf (Fig. 3.33).

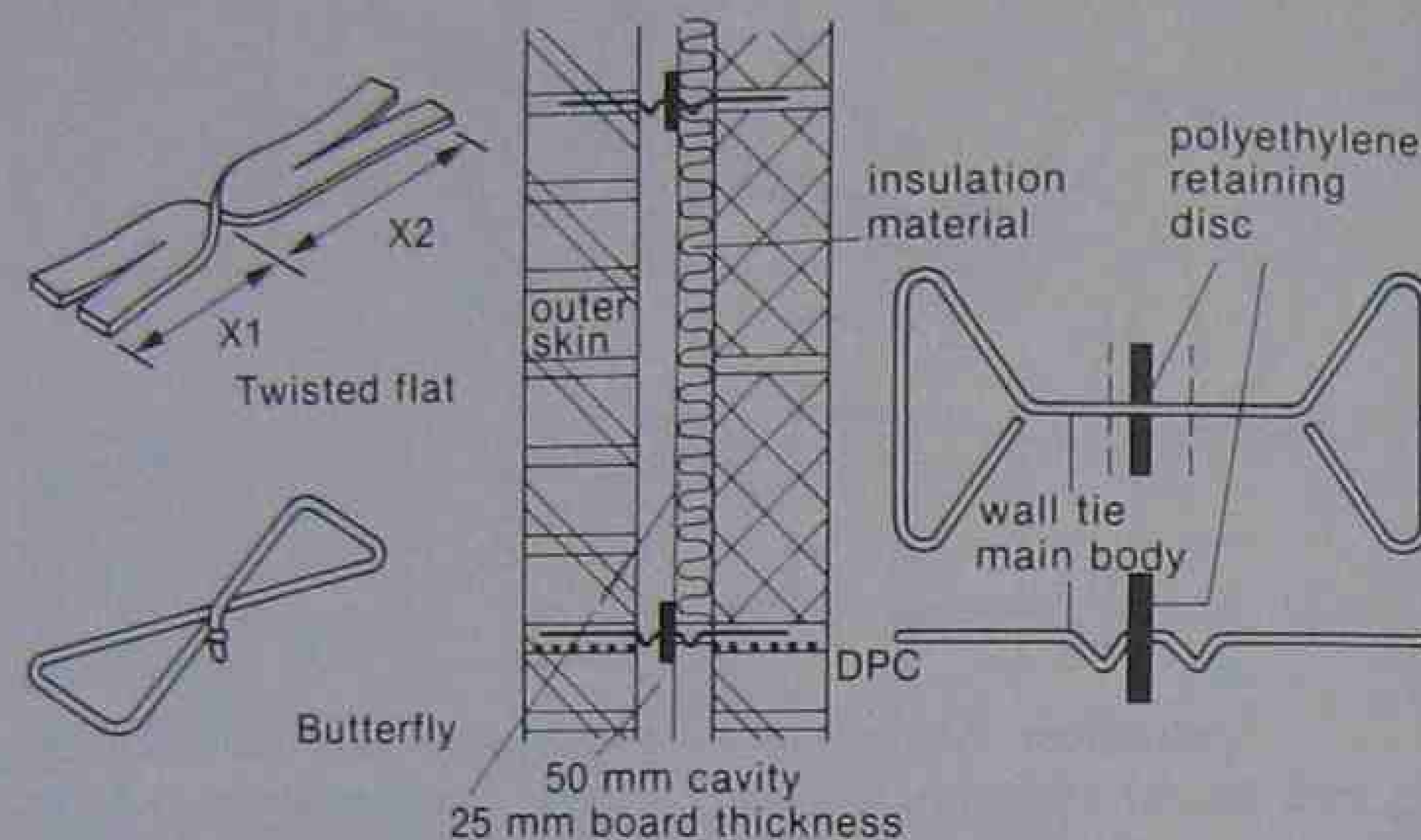


Fig. 3.33 Butterfly, twisted and insulation-retaining wall ties

It is worth noting that the condition of these ties can never be easily monitored, hence corrosion could occur and remain undetected for a long time. The thickness of galvanizing required by BS1243 has recently been increased due to the extensive corrosion suffered by wall ties galvanized in accordance with earlier requirements.

Applications Wall ties are used to tie parts of a mass walling base together, such as the two leaves of cavity walls, or thin claddings to brick or block backing walls. In the former case the ties are spaced at 900 mm centres horizontally and 450 mm centres vertically in a staggered pattern with greater frequency at wall openings; in the latter case ties will be positioned according to the pattern of the cladding joints and will have a dowelled or lipped end for fitting into holes or slots in the edge of the cladding slabs (see Fig. 3.34)

Setting instructions

1. Set the wall ties in the mortar bedding joint as the wall is

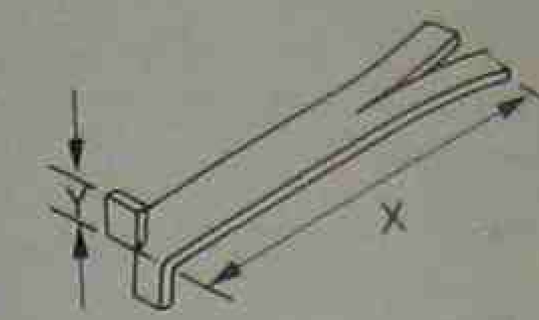


Fig. 3.34 Lipped tie

built, ensuring that each tie does not fall towards the inner leaf. An outward fall is preferable.
2. In the case of insulation retaining ties; build the inner leaf 450 mm higher than the first row of wall ties; set the insulation batt resting on the ties and between the retainer and the inner leaf; build up the outer leaf and set the next row of wall ties.

Cramps and dowels

Description A variety of galvanized and stainless steel cramps and dowels is available of many sizes and shapes. Cramps are usually made from steel flats, often bent to fit into grooves in the masonry elements; dowels are usually made from steel rod (Fig. 3.35).

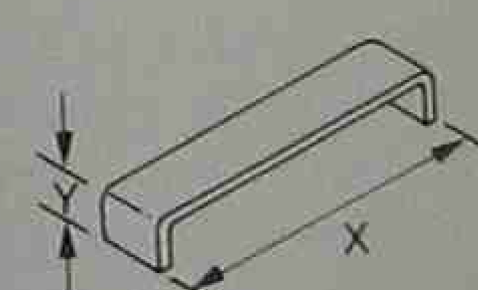


Fig. 3.35 Coping cramp

Applications These are used for reinforcing the joint between two adjoining stones or precast concrete elements, such as coping stones.

Setting instructions

1. Bed cramp (dowel) in slot or mortise cut or cast in the stone or concrete element.
2. Point with mortar.

C2 Cast-in fixings

Many of these anchorage devices (such as plugs and sockets) are similar to the drilled-for fixings (section A1 of this chapter) in concrete bases, except that cast-in fixings are placed at the time of pouring the concrete. Their precise position, therefore, has to be established early in the construction process. They have to be positioned either by wiring them firmly to the reinforcement, or by nailing them to the inside of the formwork with light, outward-pointing nails which will pull off when the formwork is struck, leaving the fixing device in the face of the concrete.

An alternative method of installation, and one which allows for minor positional adjustment at a later stage, is to form a pocket or mortise in the concrete by casting in a polystyrene or other soft material block. After the concrete has hardened, the block can be picked out leaving a recess in the concrete. The fixing can then be accurately positioned in the mortice and bedded in place using concrete or one of the many proprietary concrete grouting compounds. The alternative method allows for greater accuracy of positioning and avoids the fixing being displaced during concrete pouring; but care is needed to ensure that all traces of the temporary block are removed from the pocket and that the grouting bonds properly to the surrounding concrete.

Plugs and sockets

Description Traditionally, cast-in plugs were of wood – a small timber block lightly nailed to the inside of the formwork, so that when the formwork was struck, the block would remain set in the concrete. Today these lightweight fixings are usually made of rotproof plastic. Unlike timber, they will not shrink or swell and will receive a nail or screw fixing without the danger of splitting. They also protect the nail or screw from corrosion. Depths of proprietary plugs vary from 25 to 38 mm and they are usually wedge-shaped to afford greater pull-out resistance.

More heavyweight fixings are made by casting in a ferrous or non-ferrous socket to receive a bolt (size from M10 to M24) (Fig. 3.36).

Applications Cast-in plugs give light-duty fixing points for nails or screws, while cast-in sockets attend to the need for heavier fixings in precast or in situ concrete. They receive a bolt fixing for the subsidiary component or accessory.

Setting instructions
These will vary with the type of plug or socket used and how it is

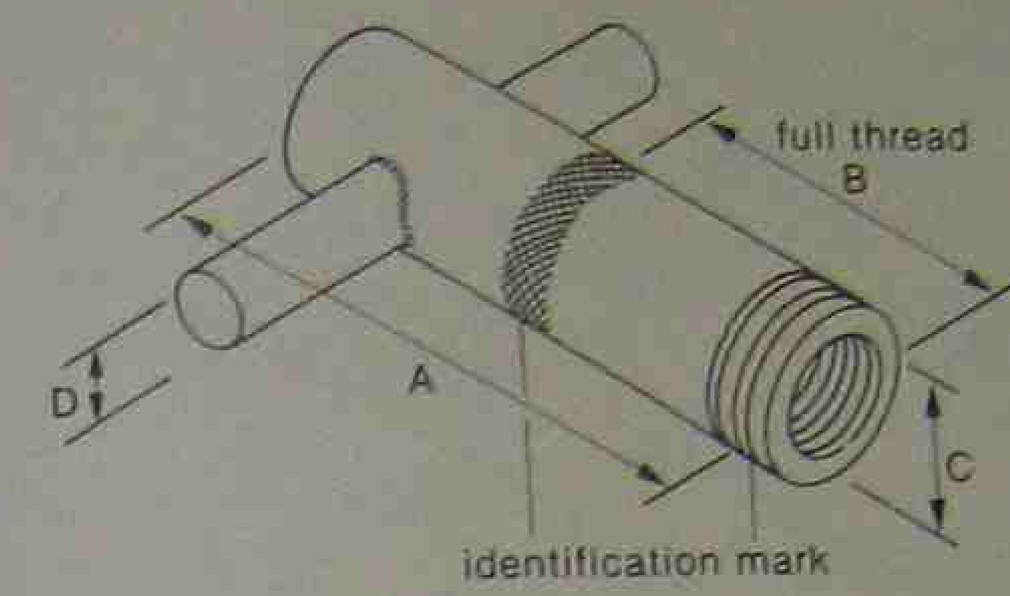


Fig. 3.36 Cast-in socket

to be secured within the base before pouring the concrete; i.e. whether wired to the reinforcement, etc.

Cast-in channels

Description There is a number of proprietary cast-in channel systems available, which are profiled to receive a variety of T-headed bolts or ties. The position of these accessories is clearly adjustable in one direction at least – parallel to the length of the channel. Channels and accessories are made of mild steel or stainless steel and in a wide variety of sizes and loading capacities. They are supplied with polystyrene filling inside the channel to avoid fouling during the pouring of concrete, and have several patterns of rear anchorage straps to improve their holding power (Fig. 3.37).

Applications Cast-in channels provide reliable restraint and loadbearing anchorage for a variety of matching accessories, such as T-headed bolts and ties, which are used to connect thin cladding panels direct to the concrete base, or loadbearing accessories, such as corbels.

Setting instructions
Once more, detailed procedure will vary with the particular system used.

1. Carefully mark out the position of the channel on the back of the formwork.
2. Depending on the type of channel and the material of the formwork, lightly nail or staple (supplied by the

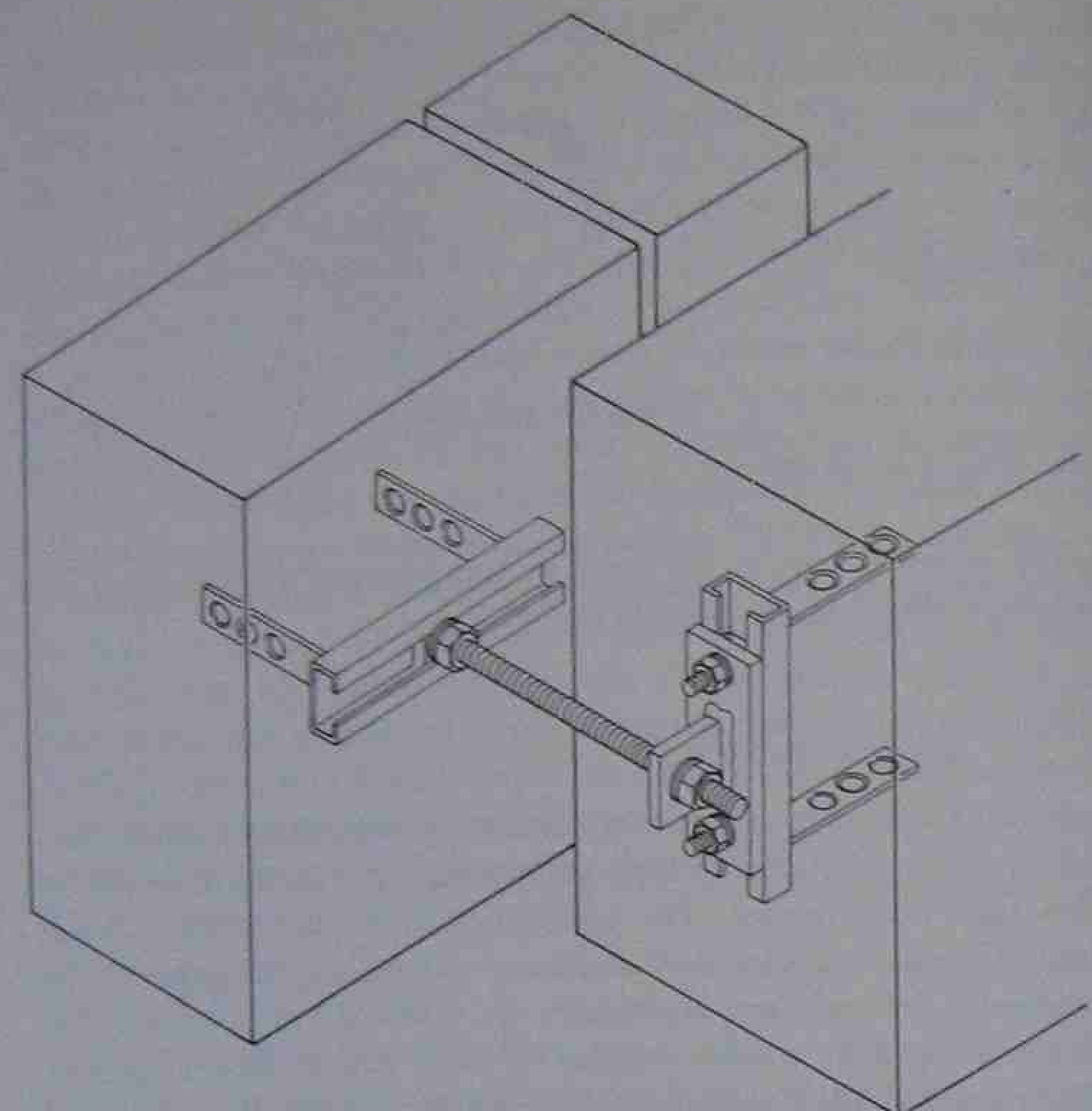


Fig. 3.37 One of the Halfen range of cast-in channel fixings

manufacturer) or bolt the channel in position.
3. Before the formwork is struck, unscrew the bolt nut.
4. Clean out the polystyrene filling from the channel. It is now ready to receive its matching accessory.

Cast-in corbels

Description Manufactured in a variety of ferrous and non-ferrous metals, these heavy-duty fixings come in a variety of shapes – flat, cranked, flat fish-tailed, cranked fish-tailed, or hoop-shaped (frame corbel) (Fig. 3.38).

Applications Corbels are loadbearing fixings used primarily to support heavyweight cladding panels.

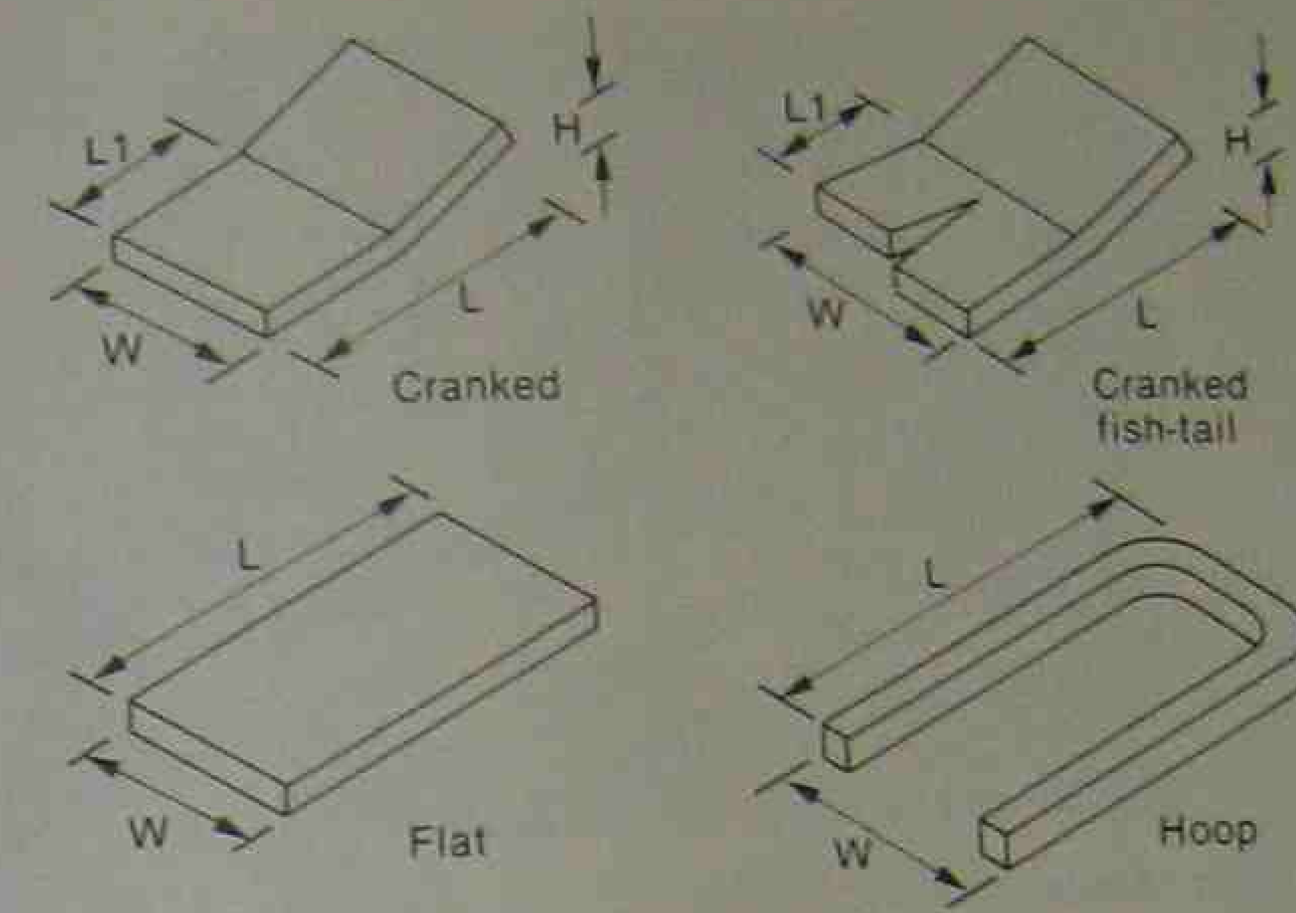


Fig. 3.38 Types of cast-in corbel

Setting instructions
Cast-in corbels are usually grouted into pre-formed pockets in the concrete, using either a concrete grout or a proprietary grouting compound.

Section D: Fixing accessories

A wide variety of accessories is used in combination with mass walling anchorage devices, such as expanding anchors or cast-in sockets. These accessories form an intermediate fixing between the device, which makes the anchorage in the base, and the subsidiary component.

An example of this is a range of loadbearing angle corbels or continuous angles used to carry heavyweight claddings (Fig. 3.39). These depend for their performance on anchorage supplied by expanding or chemical anchors, cast-in sockets, cast-in channels or similar fixings. A joist hanger without a build-in flange is a similar accessory, depending as it does on the anchorage provided by screws and plugs or masonry nails (Fig. 3.40). In addition, many anchorage straps are not built into the brickwork or blockwork, but rely on masonry nailing or similar fixings. These, too, are accessories.

There is a number of specialist accessories that concern the tying back of brick facing to concrete structure. Some, like the Harris

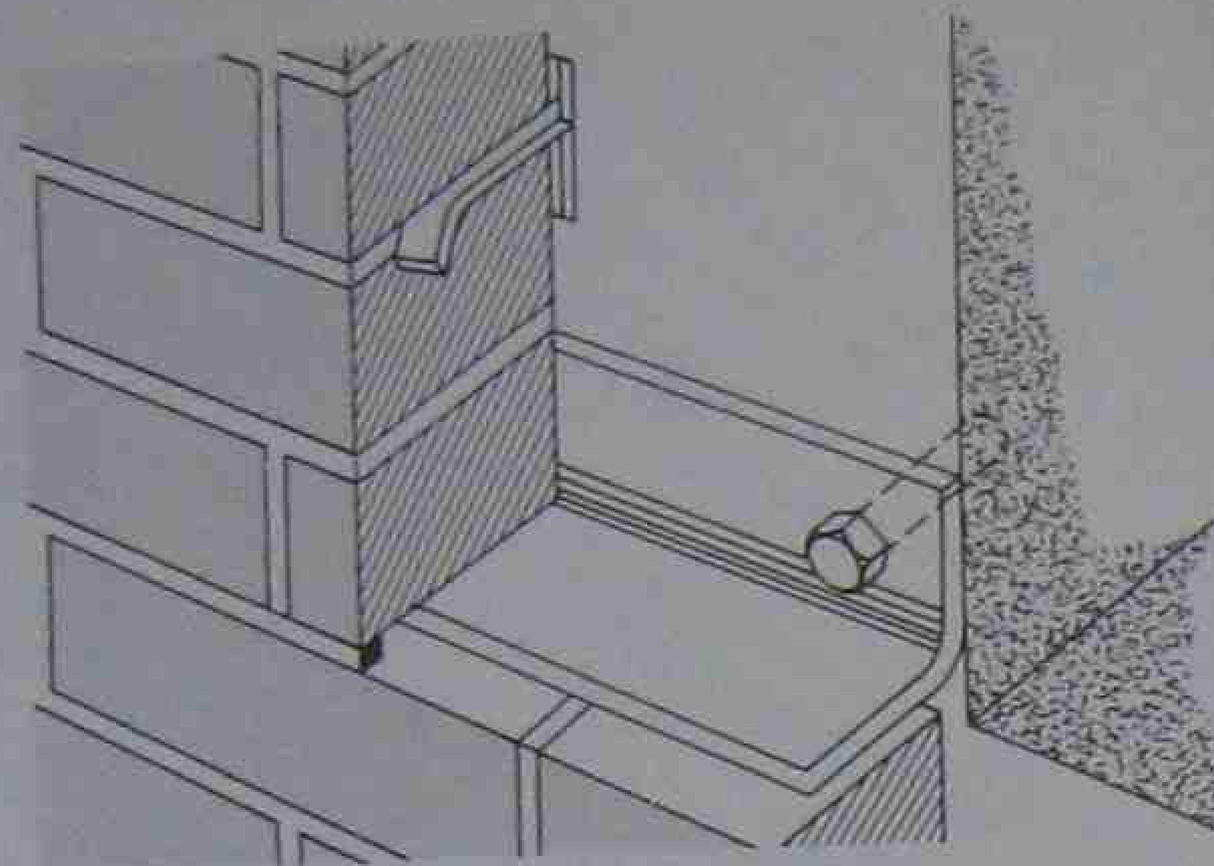


Fig. 3.39 Angle corbel application

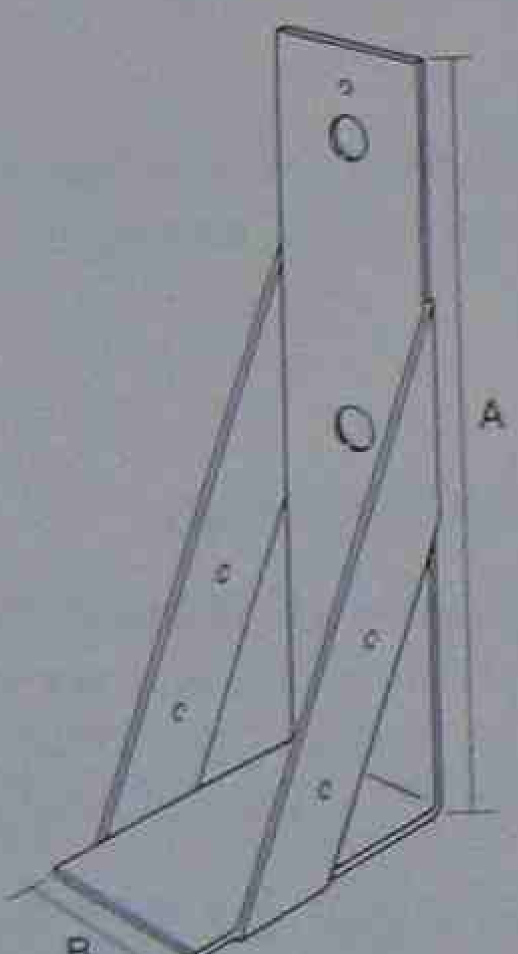


Fig. 3.40 Joist hanger without build-in flange

and Edgar sliding brick anchor, are designed to allow the differential movements of brick and concrete to take place without imposing stress on either material (Fig. 3.41), but the fixing is connected to the structure by expanding anchors, making the device an accessory. Similar accessories include bolted fish-tail ties and dowel ties, a few examples of which are illustrated (Fig. 3.42).

Also within this category are the matching accessories for use with cast-in channels and the innumerable pipe clamps and pipe

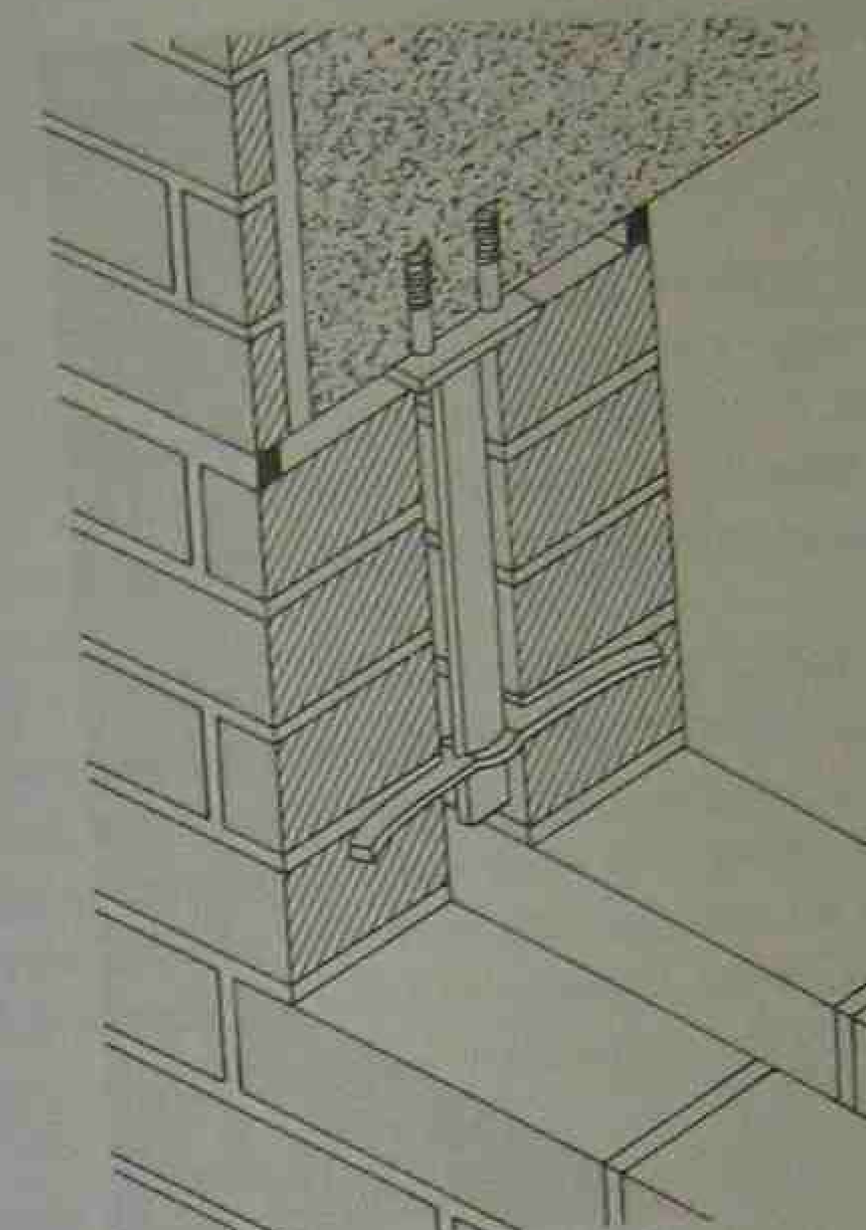


Fig. 3.41 Harris and Edgar sliding brick anchor

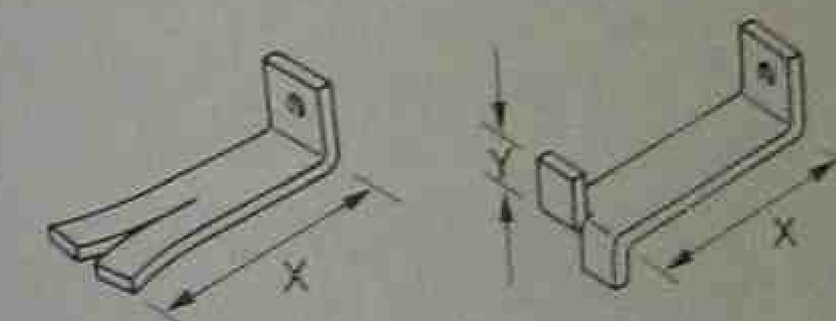


Fig. 3.42 Bolted fish-tail ties etc.

support systems, which once more form a bridge between the fixing device and the subsidiary component.

Section E: Background to fixing performance in mass walling

There is a number of factors which affect most medium- and heavy-duty fixings in mass walling bases. These have been referred to briefly and in a general way in the preceding sections of this chapter. For those who require more detailed background information, this section has been included.

Fixings in dense concrete bases

The following factors will influence the fastener's performance in dense concrete bases:

(a) compressive strength of the concrete;
(b) size of the concrete's aggregate;
(c) pH value of the concrete;
(d) reinforcement pattern in the concrete;
(e) fastener spacing in relation to base dimensions.

Compressive strength of the concrete

The higher the compressive strength of the concrete base, the greater the holding power achieved by all types of anchor fixings. The same is true of fired fixings. In their case, however, there are other factors involved. The stronger the concrete, the heavier the power load required to drive the fastener and the higher the failure rate due to spalling. Usually the compressive strength of structural concrete is around 25 to 40 N/mm^2. This range is suitable for both types of fixing. A fired fixing, though, should not be used in extremely strong concrete (higher than 60 N/mm^2). It is interesting to note that in the case of the chemical anchor, if the concrete strength is below 35 N/mm^2, a core of concrete is likely to break away with the chemical anchor, rather than the adhesive fail. Above that strength, failure will probably occur in the anchor itself rather than in the adhesive.

It is particularly important that the strength of the concrete is established where heavyweight anchors are being used which are expected to produce reliable structural performance. Manufacturer's performance data are compiled for fixings in stated grades of concrete. If the concrete involved is of a lower grade, the expected performance of the fixing must be reduced accordingly. It may even be necessary to carry out tests to establish the true strength of the fixing in a particular concrete.

Always bear in mind that the deeper the anchor, the greater the masonry support it receives and the greater its pull-out strength. The greater the depth of penetration of a fired fixing on the other hand, the greater the failure rate and the danger of spalling. The optimum depth of fired fixings is 22 to 32 mm.

Also it should be remembered not to set drilled-for anchors in (or fired fixings into) fresh concrete less than 7 days old. If anchors are set in fresh concrete 7 days old or slightly older, they should not be fully loaded until the concrete has been allowed to

reach its full compressive strength – usually 28 days for normal Portland cement concrete.

Size of aggregate

As the size of aggregate increases, so the minimum spacing and edge distance of expanding anchors must be increased to avoid spalling and failure in holding power. The guide dimensions given under 'Fastener spacing in relation to base dimensions' below are based on concrete with average-sized aggregate for a structural concrete.

If a fired pin strikes a large, hard piece of aggregate during driving, it may be deflected or bent. There is also a risk of spalling. The use of a spall stop – a metal disc on the nose of the gun or tool – is advisable.

pH value of the concrete

Concrete with a pH value below 7 or above 11 is extremely corrosive to zinc-plated steel anchors and masonry nails or pins. Normal Portland cement concrete without additives has a pH value between 9 and 11 and presents no corrosive hazard, but the presence of some additives can alter this situation.

Concrete in the neutral range between 7 and 9 can, under certain atmospheric and climatic conditions, be corrosive.

Any zinc-plated fixing set in a corrosive concrete should be considered as having a limited life. However, there are stainless steel and aluminium-bronze fixings available for use in corrosive conditions originating either in the base or from aggressive environments such as chemical plants, sewage works, etc.

Reinforcement pattern in the concrete

No drilling of holes, placing of anchors or firing of pins should be carried out near the reinforcing wires of prestressed structural concrete. In other types of concrete, expanding anchors should, as far as possible, be placed between the reinforcing bars. In heavily reinforced concrete it is possible to reduce the minimum spacing of anchors (see p. 66), but in coarse (large aggregate) concrete it would be increased.

Fired fixings in normal reinforced concrete do not generally impair its strength. However, fasteners should always be placed so as to avoid striking reinforcement, particularly in narrow structural members where the cross-section is likely to be heavily reinforced.

Fastener spacing in relation to base dimensions

Both the properties of the concrete and the anchor design govern the minimum anchor spacing, the minimum distance from the edge of the base to the anchor and the thickness of the base into which the anchor can be set.

Anchors which do not expand mechanically and exert a force on the base material (cast-in sockets, plugs and lightweight fixings generally) can be placed at closer minimum spacing than is recommended here, assuming their load is not large. Expanding anchors should never be placed too close to the edge of the base, otherwise cracking, spalling or breakaway could occur. It is interesting to note that the performance of two expanding anchors placed too close together can be dramatically reduced.

To achieve the best reliable fixing in average concrete using expanding anchors the following spacings are recommended by one manufacturer based on the length of embedment of the anchor (the dimensions in brackets represent the minimum spacing recommended with *reduced* performance):

1. Distance centre-to-centre of anchors: 2 to 3.5 times anchor length (1 to 2 times anchor length).
2. Distance between centre of fastener and edge of base:
   - (i) with load at right angles to edge: 2.5 to 4 times anchor length (1 to 3 times anchor length);
   - (ii) with load parallel to edge: 1.5 to 3 times anchor length (1 to 3 times anchor length).
3. Thickness of concrete base in relation to anchor length: 2 times anchor length.

In the case of chemical anchors, because there is no expansion involved in their placing, distance 1 can be reduced to 1 times the anchor length (with 0.5 times the anchor length as the minimum with reduced performance) and both distance 2(i) and 2(ii) are 1.5 times the anchor length (0.5 times the anchor length as the minimum with reduced performance).

Other manufacturers give alternative advice based on the diameter of the fixing. The range of dimensions given above results from the different types of expanding anchor available. It is, therefore, always advisable to follow the recommendations of the particular manufacturer concerning the actual device being used.

In the case of fired fasteners, the distance between the pin and the edge of the concrete should be 50 to 100 mm (depending on the grade of concrete and the shape of the base); the centre-to-

centre distance of pins should be 2 times the depth of penetration of the pins; and the depth of penetration of the pin should never exceed half the thickness of the base.

There are two further influences on the performance of chemical anchors – the effects of temperature and chemicals.

Effect of temperature

If chemical anchors are exposed to consistently high temperatures (in excess of 60 °C) they will show a decline in holding power. This, however, is a rare environment and one that should not often be encountered. Temperature, too, will affect curing times. Polyester-based resins can vary in hardening time from 10 minutes at 20 °C and over, to 1 hour at 0 to 10 °C, or 5 hours at minus 5 °C.

Influence of chemicals

Once more this will vary depending on the resin used in the device. Generally chemical anchors are resistant to most common acids and alkalis in normal concentrations. If there is a particular hazard in the location of the fixing, it would be prudent to check with the manufacturer.

Fixings in lightweight concrete bases

Generally, as far as lightweight concrete with a dense structure is concerned (i.e. concrete with lightweight aggregate as opposed to aerated or foam concrete), the considerations applying to dense concrete hold good. The proviso is, of course, that because the concrete has a lower compressive strength (up to 30 N/mm^2), the fixings will have proportionately reduced performance. In concrete of 1800 to 2000 kg/m^3 density, it is not advisable to use anchors with a bolt diameter over 16 mm. It is also unwise to use self-drilling anchors in any lightweight concrete.

Fired fixings make only the lightest duty fixing in concrete of a compressive strength less than 10 N/mm^2 and should not be used at all in aerated concrete.

Aerated concrete does produce an additional problem. Because of its large voids, it tends to assist the corrosive attack of some of its additives on zinc-plated anchors. As the strength of this type of concrete is only up to 8 N/mm^2, only lightweight anchorage fixings are usually considered, and therefore plastic plugs (such as the Hilti H6 and FD anchor, or the Rawlnut) are usually adequate.

Expanding anchors are not recommended for use in aerated concrete.

Fixings in brick and clay block bases

Because bricks and clay blocks are usually brittle, only fixings exerting relatively low expansion forces should be used. Fired fixings are not suitable for use in perforated bricks or hollow blocks.

The following factors influence the fastener's performance:

(a) brittleness and character of the bricks or blocks;
(b) position and size of the cavities in the walling units;
(c) character of the walling.

Brittleness and character of the bricks or blocks

If the brick is extremely hard and brittle, anchors can be set in the mortar joints, but widely varying pull-out performance may result, depending on the quality of the mortar. Tests should be carried out to determine the achievable holding power of such a fixing.

Otherwise anchors should be placed in the centre of the brick or block; but even then, if the fixing is expected to carry a heavy load, loading tests are advisable.

Position and size of cavities in the base

Cavities can considerably reduce the bond surface between anchor and base. They, therefore, must reduce the holding power of the fixing. If possible, the anchor should pass through as many walls between internal cavities as is practicable in order to achieve the best pull-out performance. Nevertheless, however good the performance is, it will never match that of a similar fixing in a solid brick or block. Injection anchors (section A3 of this chapter) have been developed specially to overcome the difficulties of fixing to this type of base. Fixings to some hollow clay blocks tend to fall more naturally within the scope of Chapter 6, 'Mechanical fixings in cellular bases'.

Character of the walling

The width and strength of the mortar joints affect the strength of the wall, as well as the holding power of any anchorage fixing placed in it. If the joint is of hard mortar, good anchoring is

achieved even if the brick cracks during expansion of the anchor. If, on the other hand, the joint is weak, the performance of the anchor is lost if the brick cracks.

It is unwise to use any self-drilling anchors in brick or clay block bases.

4

Mechanical fixings in timber bases

Most fixings in timber bases fall into two of the fixing types listed in Chapter 2 – base-deforming fixings – type 2(b) – and through-fixings – type 2(a) (p. 92). The base-deforming fixings include woodscrews, in spite of the fact that they need a pre-drilled pilot hole. This is because a screw's primary holding power is achieved by its thread cutting a matching groove in the timber, i.e. deforming the base.

Section A: Base-deforming fixings – type 2(b)

Because of timber's relatively soft, non-brittle and fibrous nature, most fixings which have evolved for use with timber bases are those which are driven or otherwise forced into the timber, deforming its fibres in the process. It is the grip which develops between the timber fibres and the shank of the fixing which gives the device its holding power and allows it to make an anchorage in the wood.

This group of timber fixings includes nails, woodscrews and toothed metal plates.

The pull-out strength of such fixings as plain shanked nails is considerably affected by the type of wood and its moisture content; while the performance of other fixings, such as annular ring shanked nails and screws, is less influenced by the moisture content of the wood. Screws generally produce a greater withdrawal resistance than nails; and annular nails than plain shanked nails.

A1 Nails (including staples)

These fixings generally require no prior preparation work to the

subsidiary component or the base. Some thin timber components, however, may have to be pre-drilled to avoid them being split when the nail is driven; and metal or brittle components (such as roof tiles) have to be punched or fabricated with pre-formed holes to receive the nails. It is sometimes said to be advisable to pre-drill pilot holes up to 80 per cent of the nail shank diameter when nailing some hardwoods (afrormosia, ash, greenheart, beech, gurjun/keruing, iroko, opepe, sapele, teak, jarrah and karri) but strong structural connections can be made with these timbers using plywood gussets without drilling holes.

Nails can be driven by hand or by pneumatic nailing machine. These machines are becoming increasingly common on site where compressors are in use. They either drive nails specially manufactured for machine nailing and collated into strips or coils, or similarly arranged batches of staples. Nails usually have a round, semi-round or tee head. Staples can either be equal- or unequal-legged. The air pressure can be adjusted to countersink the nail, or not, as required. There are smaller manually-operated staple machines for lighter stapling jobs and a mallet-actuated machine for nailing floor boards. Nailing machines clearly increase productivity. Up

Fig. 4.1 Coil nailers with magazine capacities of up to 400 nails from BeA

to 180 nails per minute can be placed using some pneumatic machines (Fig. 4.1).

The nail is one of man's oldest forms of mechanical fixing device, being a metal equivalent of the medieval carpenter's peg or dowel. Once nails were forged and then they were cut from metal plate (cut nails). Eventually, however, with the introduction of wire into nailmaking, the industry was revolutionized. Today, most nails are made from mild steel wire (wire nails), although copper and aluminium alloy wires are also used. There are still a few surviving cut nails, made from black rolled steel to produce square-sectioned nails, such as floor brads.

There is a considerable number of different types, sizes and head styles in nails, largely developed for the particular type of subsidiary component to be fixed. More detail of the structural use of nails will be contained in the background section of this chapter on page 104.

Description Mostly nails are driven through the subsidiary component and into the base in one operation. There are a few exceptions, such as the glazing sprig. Here the nail does not pass through the subsidiary component, but is driven into the base alongside the component, trapping it against a projection in the base.

Nails for use in timber bases come in a wide range of sizes and types, capable of making all strengths of joint from light- to heavy-duty. The major types of wood nail are set out and described in Table 4.1. The shape of head and shank are largely determined by the type of subsidiary component to be fixed; the size and shape of the shank is primarily dictated by the structural performance required. While most nails are made from mild steel or plate, others are made from non-ferrous metals (copper or aluminium alloys) for use where corrosion could be a problem. Also some nails, such as escutcheon pins, are manufactured in non-ferrous metals because they are intended to be seen with decorative ironmongery.

It will be observed that nail diameters are given in millimetres. This practice has now largely superseded the habit of quoting nail diameters in standard wire gauge, although s.w.g. sizes may still be encountered from time to time. A table of equivalent diameters is given in Appendix 3.

Applications Nails are used to make joints of varying

structural performance between pieces of wood, or between a variety of subsidiary components (such as roof tiles, bituminous felt, plasterboard, metal fixing straps and light metal accessories) and a timber base. The selection of the right nail for the job will be influenced by: (a) the type of subsidiary component; (b) the performance required from the fixing; (c) the appearance required; (d) the likelihood of corrosion. Brief advice on the correct use of various nails is given in Table 4.1.

Type of subsidiary component

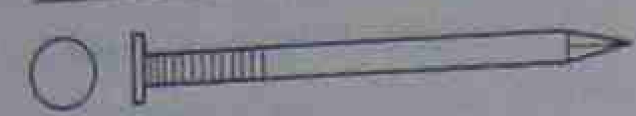
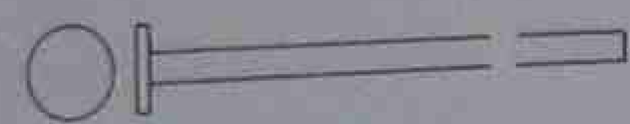
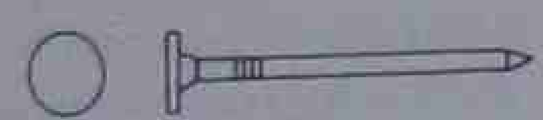
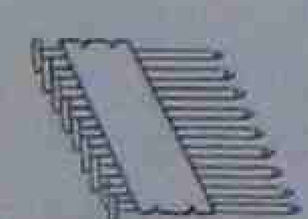
The avoidance of pull-over failure, in which the subsidiary component is torn over the nail head, often dictates the size and style of the nail head; for instance, soft, thin or flexible sheet materials, such as roofing felt, need to be fixed with large, flat circular headed nails. In the same way, when a nail is driven through a thin timber component, there is a danger that it will be split, unless the shape of the shank of the nail is selected carefully. An oval-shanked, lost-head nail could be used in this case, placed so that the oval shank aligns with the wood grain of the component. Specialist nails like corrugated sheeting nails may have round-dome heads with washers to discourage rainwater leakage round the nail shank.

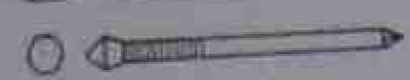
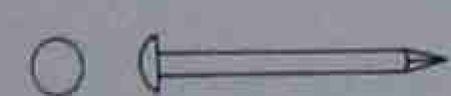
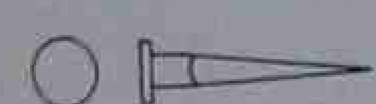
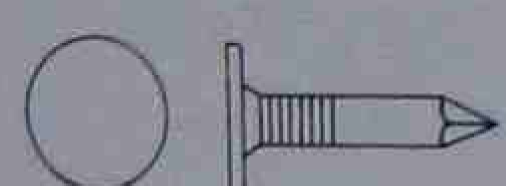
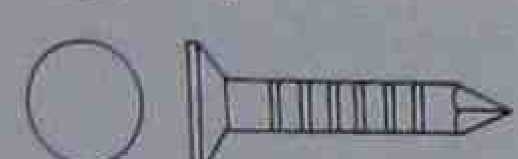
Performance required of the fixing

The engineering performance of the nail fixing will clearly be influenced by the nail type and size. Generally, round-headed plain wire nails are used for carpentry, timber-to-timber joints. Thicker-gauge nails withstand lateral loads better than thinner nails. The longer the shank of the nail (within reason), the greater is its pull-out resistance, and the more likely it is to split the wood (particularly in the case of a thick-gauge nail). When joining two pieces of timber together, the length of nail needs to be about two-and-a-half times the thickness of the attached timber. When plywood or other wood-based sheet material is being nailed to timber, the nail length should be four times the thickness of the plywood.

So called 'improved nails' (CP 112: Part 2: 1971), such as annular ring shank or square twisted nails, have greater resistance to withdrawal and lateral loading than ordinary plain wire nails. Cement-coated or resin-coated nails also have greater pull-out strength. In the case of the latter, the friction heat of driving the nails softens the resin, which later sets to adhere the nail shank to the base. Improved and coated nails discourage the tendency of

Table 4.1 Types and uses of wood nails

	Nail type	*Head style*	*Length range (mm)*	*Thickness range (mm)*	*Material*	*Applications and notes*
	Round plain wire (French nail)	Flat circular	15–200	1.4–80	Mild steel	Structural timber and general carpentry applications
	Purlin	Flat circular	150–250	3.7	Copper	A round plain wire nail used where ferrous nail might corrode
	Clout (slate nail)	Flat circular	10–100	2–4.5	Mild steel copper aluminium	A large-headed version of a round plain wire nail for fixing roofing felt, slates, plaster-board, etc.
	Plasterboard	Circular countersunk	31–38	2.6	Mild steel	A nail with a serrated shank for fixing plasterboard
	Square twisted (drive screw)	Flat circular	20–200	2–8	Mild steel aluminium silicon bronze	Improved nail (CP 112: Part 2: 1971) for structural timber joints
	Annular ring-shanked	Flat circular	20–200	2–8	Mild steel aluminium silicon bronze	Improved nail (CP 112: Part 2: 1971) for structural timber joints
	Machine driven	Flat circular semi-circular T-shaped	30–100 in strips or coils	2.2–3.1	High tensile steel aluminium	Mostly operated by pneumatic machines to fabricate structural timber assemblies
	Lost head	Small oval	15–75	1–3.75	Mild steel copper aluminium	Wire nail, often with an oval shank, for use in visible locations, punched home and the hole filled
	Panel pin Veneers pin	Small round	15–25	1–2.6	Mild steel copper aluminium	Wire nail for fixing light timber components; easily punched home and hole filled. A finer version of this nail is called a veneer pin

	Nail type	Head style	Length range (mm)	Thickness range (mm)	Material	Applications and notes
	Hardboard pin	Diamond profile	20–25	1.6–1.4	Mild steel	Wire nail for use with hardboard; no punching home required, merely filling of the hole
	Floor brad	Rectangular	40–75	2.3–3.3	Steel aluminium	A cut nail for fixing softwood flooring
	Serrated flooring nail	T-shaped	50	–	Steel	For use in a manual nailing machine for fixing softwood flooring
	Sprig	Headless	10–25	1.4–1.6	Steel copper	A rectangular section cut nail for fixing glazing
	Escutcheon pin	Dome	5–40	0.9–2.0	Steel, brass	For fixing minor ironmongery
	Tack	Flat circular	–	–	Blued steel	For fixing fabrics or carpets
	Large-headed clout nail	Flat circular	12.7–38.0	2.9–3.3	Steel copper aluminium	For fixing roofing felt
	Tile peg	Circular countersunk	–	–	–	For fixing roof tiles
	Corrugated sheeting nail	Dome or spring	25.4–76.2	3.3–5.9		For fixing corrugated roof sheeting
	Duplex headed (shutter nail)	Double heads	45–100	3.0–5.6	Mild steel	A wire nail for use in situations where withdrawal is anticipated.
	Staple (equal or unequal legs)	–	3.0–75	Wide range	Steel	Hand- or machine-driven for many strengths of joint from constructional to tacking insulation in place

nails to work out during loading movement. This characteristic is known as 'popping'. Indented shank nails (serrated-edge floor nails or plasterboard nails) also have increased resistance to 'popping'.

More information on the structural performance of nails is given in the background section of this chapter.

Appearance required

Generally a nail joint is not chosen for its appearance. Therefore nails in visible positions are usually lost-head wire nails (or panel pins in lighter applications), punched below the surface and the nail hole filled. In damp locations, corrosion staining can result from the use of mild steel nails. Even nails with a corrosion-resistant coating may be subject to this defect if hammering damages the coating. Non-ferrous nails should be used in these situations. Some specialist nails, like escutcheon pins, are designed with dome heads and are intended to be seen.

Corrosion hazard

Rusting of ferrous nails can occur in damp locations, as explained above, even when they are galvanized if the coating is damaged. Copper or aluminium alloy nails should be used in these applications. Comments made in Chapter 2 concerning contact corrosion should be borne in mind if two metals are to be used together.

Copper nails should not be used with cast iron, mild steel, galvanized steel, zinc or aluminium. Mild steel nails should not be used with aluminium, zinc or galvanized steel.

Setting instructions

Nails are one of the simplest forms of mechanical fixing device and a step-by-step description of how to drive a nail is unnecessary. However, the following points should be borne in mind:

1. If a thin timber component is to be fixed, an oval nail of small shank size is less likely to split the wood so long as the axis of the oval is aligned with the grain of the wood. Blunting the nail point can help to avoid splitting very small timber components.
2. Pilot holes, 80 per cent of the nail shank diameter, should be used when nailing hard hardwoods.
3. Firm blows of the hammer should be applied to the nail head so that the impact is precisely aligned to the shank of

the nail. Do not hit the nail too hard until it has penetrated a reasonable distance into the base this avoids initially skewing the nail and later bending its shank.
4. Avoid defacing the surface of the component, particularly if it is to remain visible, with blows of the hammer head. Use a nail punch to set lost-head nails below the surface of wooden subsidiary components and fill the nail hole.

Note: The techniques for driving a nail were set out about 250 years ago in the 1736 edition of the *City and County Purchaser and Builder's Dictionary* in the following words:

There is requir'd a pretty Skill in driving a Nail, for if, when you set the Point of a Nail, you be not curious in observing to strike the flat Face of the Hammer perpendicularly down upon the Perpendicular of the Shank, the Nail, unless it have good Entrance, will start aside, or bow, or break, and then you will be forced to draw it out again; therefore, when you buy a Hammer, chuse one with a true flat Face.

In this respect, nothing very much has changed in the last 250 years!

A2 Woodscrews

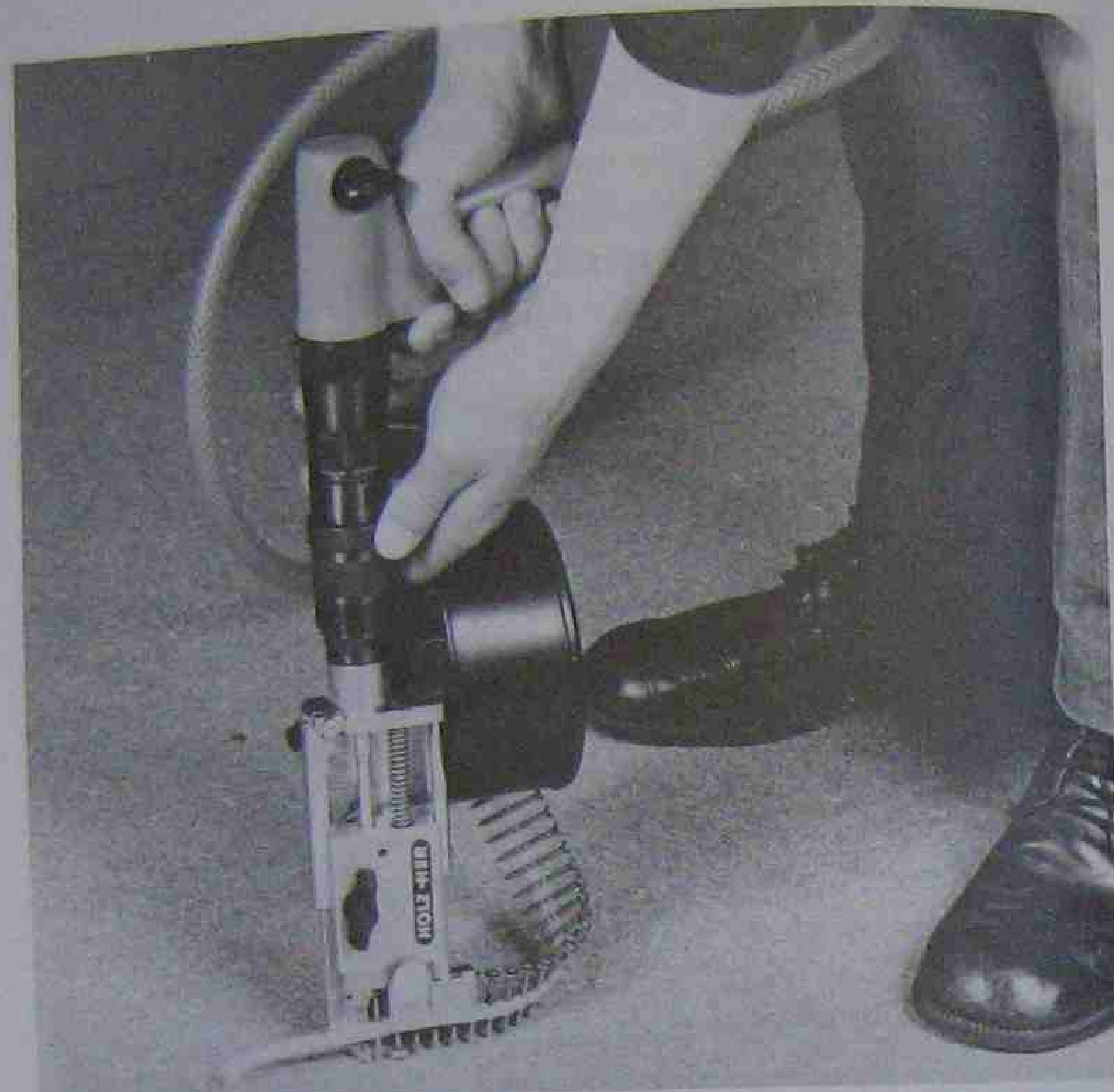
The driving action of woodscrews tends to clamp the subsidiary component to the timber base more tightly than the driving of nails. As a result a screw joint has greater strength than a nailed or stapled joint. Screws have superior withdrawal resistance and strength in lateral load (shear). Diameter for diameter, a nail needs a greater shank length to equal the shear performance of a woodscrew.

Screws also have the additional advantage of being capable of removal and replacement if subsequent adjustment of the assembly is necessary.

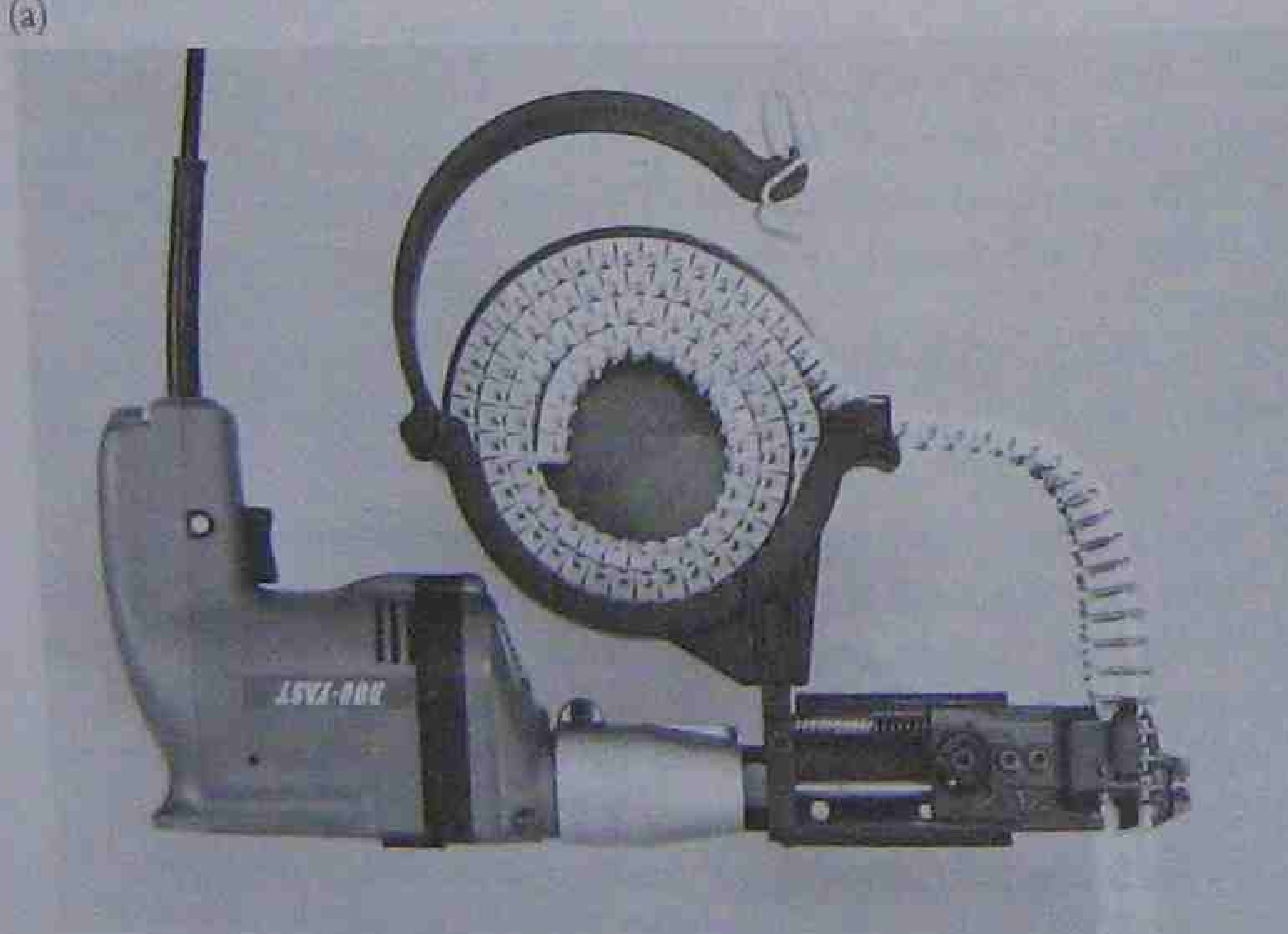
Screws, however, are more expensive in themselves and are more costly to install than a nail. They usually require the drilling of a pilot hole before being driven and, in certain circumstances, a clearance hole through the subsidiary component.

Driving can be undertaken manually or using an electric or pneumatic screwdriver (Figs. 4.2, 4.3).

Screws of a type have been used since Roman times, but the first recorded patent for machine-made screws was granted in 1760 and screws (albeit crude by today's standards) began to be



(a)



(b)

Fig. 4.2 (a)BIF Holz-Her power screwdriver (b) Duo-fast circular magazine screwdriver

mass-produced shortly after this date. There is now a number of types of woodscrew available with varying head styles, drive profiles, thread designs and material of manufacture.

Description A woodscrew consists of a head, containing a driving profile to engage with the screwdriver, and a pointed shank with a raised spiral thread running from the point throughout the majority of the shank (Fig. 4.3). The screw thread cuts a matching groove in the timber to make a firm anchorage for itself. It needs a pre-drilled pilot hole for effective placing and is driven through the subsidiary component (with or without a clearance hole) into the base in one fixing operation.

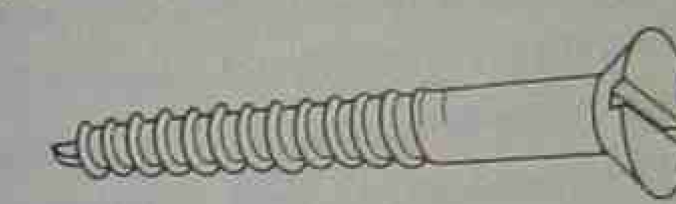


Fig. 4.3 GKN woodscrew

Head styles include countersunk, raised countersunk and round, containing a slotted, recessed or clutch driving profile (Fig. 4.4). The thread is usually a single spiral and is formed by recessing a groove in the shank of the screw. In this case the diameter of the thread is the same as the diameter of the shank: the shank is said to be *unrelieved*. Recently double-spiral threads have been introduced in which the thread is set above the shank diameter. The shank, in this case, is said to be *relieved* (Fig. 4.5).

Woodscrew diameters are still given in the traditional screw gauges from 0 to 20. (Unlike the standard wire gauge, the larger the number, the thicker the screw). Lengths are now quoted in millimetres.

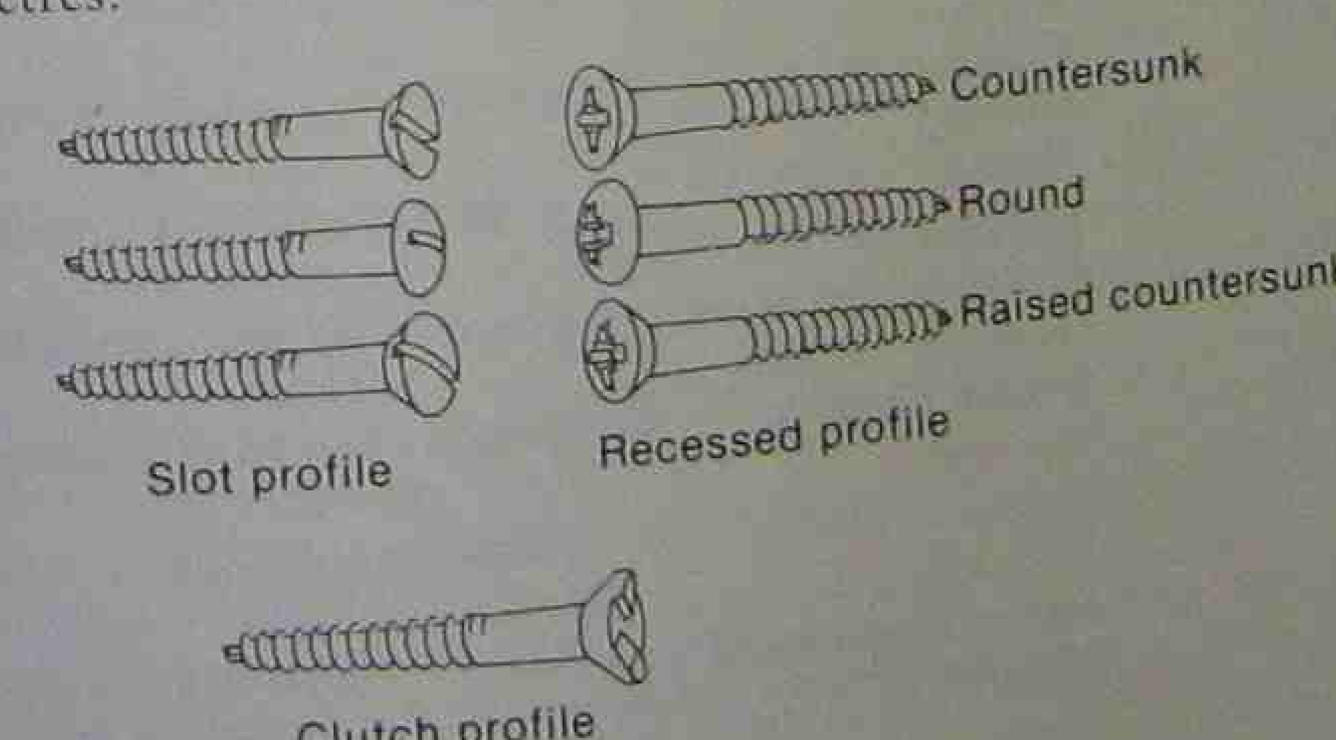


Fig. 4.4 Head styles and drive profiles in woodscrews

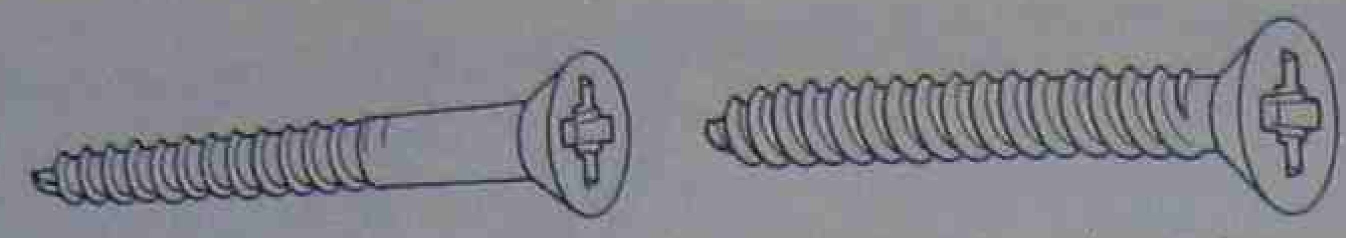


Fig. 4.5 Twinfast screw from GKN with double spiral thread

Screws are supplied in a number of different metals and finishes. Table 4.2 gives an example of GKN's range of screw types and sizes.

Applications Woodscrews are used to make joints of varying structural performance between pieces of wood, or between a variety of subsidiary components (such as sheet material, metal fixing straps and light metal accessories) and a timber base. The selection of the right screw for the job will be influenced by: (a) the performance required of the fixing; (b) the ease of driving the screw; (c) the appearance required; (d) the likelihood of corrosion.

Table 4.2 One manufacturer's range of woodscrew sizes and types

Material	*Head/ drive profile*	*Gauge no.*	*Lengths (mm)*	*Thread type*
Steel	Countersunk slot	0–20	6.4–152.4	Single
Steel	Round slot	0–14	6.4–76.2	Single
Steel	Raised countersunk slot	4–10	9.5–50.8	Single
Steel	Countersunk recessed	3–14	9.5–76.2	Single
Steel	Round recessed	3–12	9.5–50.8	Single
Steel	Raised countersunk recessed	4–10	12.7–50.8	Single
Steel	Countersunk recessed	3–12	12.7–63.5	Double
Steel	Round recessed	4–10	9.5–50.8	Double
Steel	Raised countersunk recessed	4–10	9.5–50.8	Double

Material	*Head/ drive profile*	*Gauge no.*	*Lengths (mm)*	*Thread type*
Steel	Countersunk clutch	6–12	19.05–50.8	Single
Brass	Countersunk slot	0–20	6.4–101.6	Single
Brass	Round slot	1–14	6.4–50.8	Single
Brass	Raised countersunk slot	3–12	9.5–50.8	Single
Brass	Countersunk recessed	4–12	9.5–44.5	Single
Brass	Round recessed	4–12	9.5–44.5	Single
Brass	Raised countersunk recessed	4–12	9.5–44.5	Single
Stainless steel	Countersunk slot	4–14	12.7–76.2	Single
Stainless steel	Round slot	4–14	12.7–76.2	Single
Stainless steel	Raised countersunk slot	4–14	12.7–76.2	Single
Stainless steel	Countersunk recessed	4–10	15.8–50.8	Single
Stainless steel	Round recessed	4–10	15.8–50.8	Single
Stainless steel	Countersunk recessed	4–10	15.8–50.8	Double
Stainless steel	Round recessed	4–10	15.8–50.8	Double
Stainless steel	Raised countersunk recessed	4–10	15.8–50.8	Double
Aluminium alloy	Countersunk slot	4–12	12.7–50.8	Single
Aluminium alloy	Round slot	4–12	12.7–50.8	Single
Aluminium alloy	Raised countersunk slot	4–12	12.7–50.8	Single
Aluminium alloy	Countersunk recessed	4–10	12.7–31.7	Single
Aluminium alloy	Round recessed	4–10	12.7–31.7	Single
Aluminium alloy	Raised countersunk recessed	4–10	12.7–31.7	Single
Silicon bronze	Countersunk slot	8–14	25.4–76.2	Single

Note: Steel and brass screws are available in a variety of finishes

Performance required of the fixing

A screw's holding power depends on the length of penetration and the gauge of the screw. Double-spiral threaded screws, such as Twinfast screws from GKN, offer a 3 per cent greater grip than conventional single-spiral threaded screws, assuming an equivalent length of thread penetration.

Unlike ordinary screws, 60 per cent of whose shanks are threaded, the Twinfast shank is more completely threaded. In fact, shorter Twinfast screws (up to 19 mm long) are threaded close to the underside of the head, offering a 25 per cent increase in holding power and making them particularly suitable for fixing hinges to chipboard bases and similar applications. Double-spiral threaded screws longer than 32 mm are threaded for three-quarters of their length. Double-threaded screws have enhanced performance when fixing to low-density chipboards, blockboards or softwood. What is more, the fact that the thread diameter is greater than the unthreaded shank (i.e. the shank is 'relieved') avoids the wedging action of an ordinary screw and minimizes the danger of the wood splitting.

Ease of driving

The ease with which a screw can be driven can be improved, particularly in positions of difficult access, by selecting a recessed driving profile (Phillips or Supadriv) rather than a slot. Supadriv, particularly, has been developed by GKN to improve the driving of screws at an angle. With slot-drive profiles and other recessed patterns, the driver may ride out of the recess (cam out) if the screw and the driver are not in perfect alignment. Supadriv, incidentally, is an improved version of this company's Pozidriv recessed head.

Generally, recessed driving profiles assist driving to higher torques, they aid removal and reduce recess damage from badly adjusted clutches on power drivers.

Clutch-headed screws are only used where later removal of the screw needs to be positively prohibited (e.g. in places where vandalism can be anticipated). Clutch driving profiles are normal slot profiles from which the anti-clockwise face of the slot has been chamfered off.

Double-spiral threaded screws tend to drive more quickly than single-thread screws. Their gimlet points also function rather like drill points, resulting in balanced driving – another aid to ease of placing.

Appearance required

Head styles can be chosen for the appearance required – countersunk, raised countersunk or round. Also the material of manufacture or finish can be chosen to match the metal of the component being fixed (dark Florentine bronze, barrel chromium plated, light bronze metal antique, etc.). Plastic snap-on caps or push-fit or screw-in dome caps are available to cover the screw head for certain special applications.

Countersunk headed screws are used for general joinery work or when the component has been prepared with sinkings to receive them. In timber-to-timber applications, where removal is anticipated, countersunk screws are often used with screw cups (Fig. 4.6). Raised countersunk-headed screws are normally used with high-standard ironmongery, the driver blade being less likely to damage the component. Round-headed screws are used especially (but not exclusively) for fixing general ironmongery or metal plates which are too thin to receive countersunk screws.

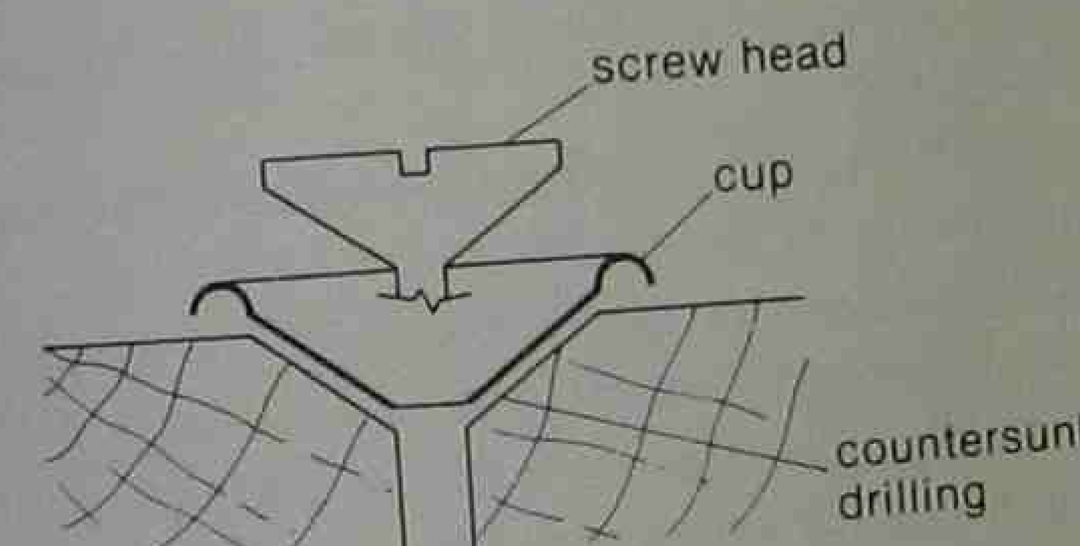


Fig. 4.6 Screw cups

Corrosion

There is a number of different types of woodscrew available for use in corrosive environments. Finishes applied to ferrous screws are applied usually for reasons of appearance, rather than corrosive protection. For conditions where rusting could take place, non-ferrous and corrosive-resistant screws should be used – aluminium alloy, brass, silicon bronze or stainless steel. Plated or coated-steel screws, such as bright-zinc plated, sherardized and black-japanned screws, should only be used externally when painted. Other plating, such as nickel, cadmium, barrel chromium and the various shades of bronze coating, should only be used internally.

Comments in Chapter 2 regarding contact corrosion should be borne in mind if two different metals are being used in close contact.

Setting instructions

1. Drill a pilot hole through the component and into the base. This is required for all but the smallest gauges of screw. In hardwood and dense chipboard the diameter of the pilot hole should be 90 per cent of the screw diameter; in softwoods and low-density chipboards, 70 per cent of the screw diameter.
2. The pilot hole should be shorter than the penetration depth of the screw from 3 mm for screw gauges 3 and 4, up to 9 mm for gauge 14.
3. Insert the screw into the pilot hole in the component and drive into the base.

Note: Where double-spiral thread screws are concerned, the component should be drilled with a clearance hole, the same diameter as the screw. This allows the component to be pulled tightly down on to the base by the tightening of the screw.

When power tools are used to drive screws, pre-drilling pilot holes is often unnecessary.

In order to avoid splitting the wood, screws should be no nearer than 10 times the diameter of the screw to the end of the component along the grain and 5 times the diameter of the screw across the grain. The first dimension can be reduced if the pilot hole is correctly drilled and both dimensions can be reduced if double-thread screws are used. Centre-to-centre spacing of screws should be 10 diameters along the grain and 3 diameters across.

Specialist screws

There is a number of other devices, developed from the wood screw, which are used for special applications.

Dowel screws These are steel dowels, threaded at both ends, which are used to make end-to-end secret connections between wood components (Fig. 4.7). The size range is limited; usually from screw gauge 6 to 12 and in lengths 25 to 50 mm.

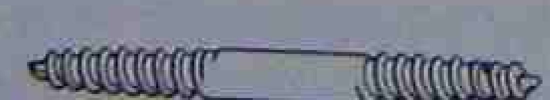


Fig. 4.7 Dowel screw

Headless woodscrews These steel devices are in effect dowels, threaded at one end and cut square at the other. The

square end is given a slot driving profile. The range is extremely small; usually restricted to 12 gauge diameter and lengths of 25 and 37 mm (Fig. 4.8).

Fig. 4.8 Headless screw

Chipboard joinery screws These are one-piece fasteners for making strong joints in chipboard or timber boards meeting at right angles. They are available in 5 and 7 mm diameters and lengths of 40 and 50 mm respectively. They have a double spiral thread.

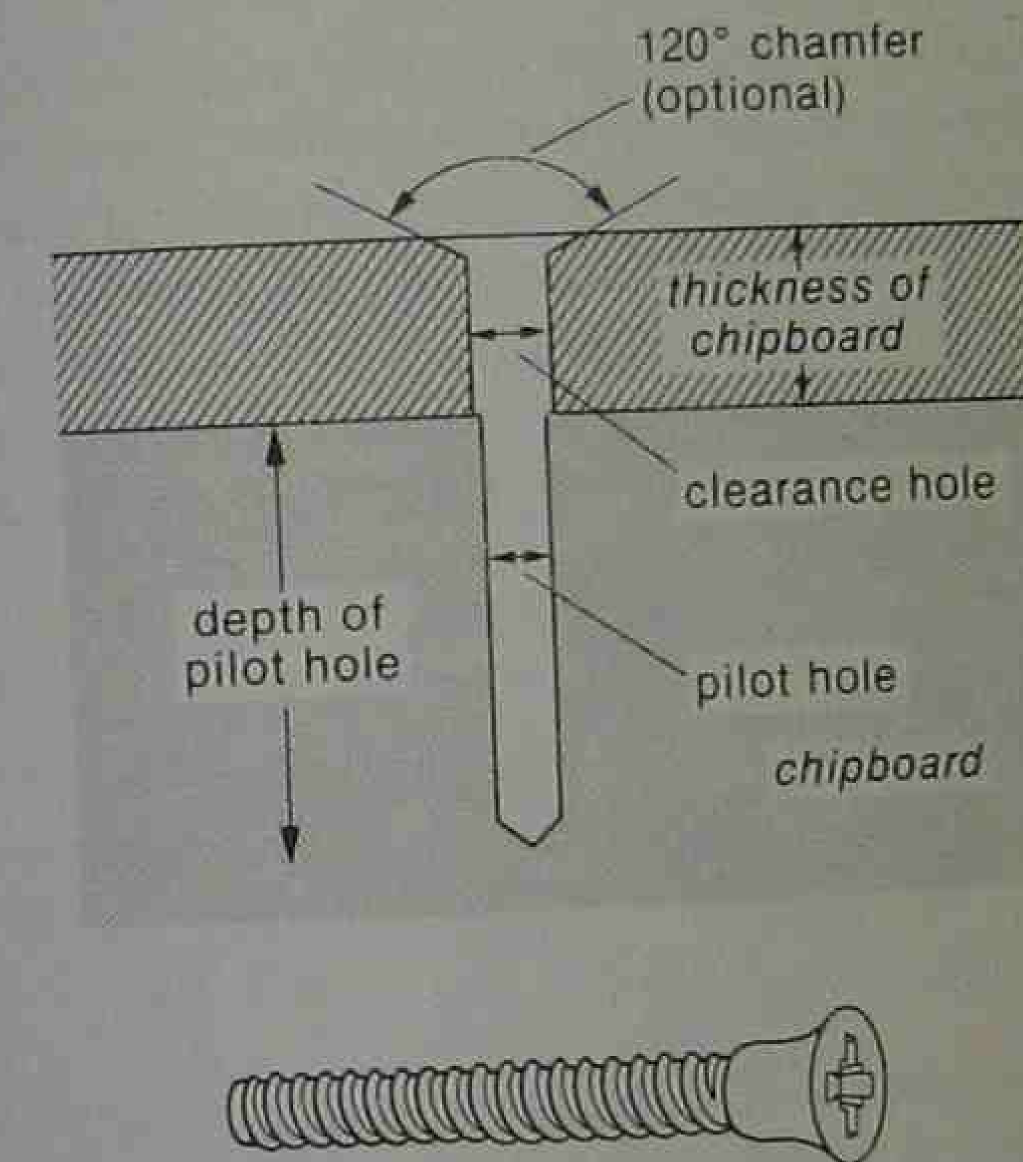


Fig. 4.9 Chipboard joinery screw and drilling data

Setting instructions

1. Select the correct size of screw for the thickness of board being joined – 5 mm screw for 12 mm chipboard and 7 mm screw for 16 mm chipboard.
2. Drill a clearance hole, the diameter of the screw, through the component.
3. Drill a pilot hole in the base, 80 per cent of the diameter of the screw and slightly deeper than the penetration of the screw.
4. Insert the screw and drive home.

A3 Coach screws

Unlike woodscrews, coach screws are not driven by a screw-driver. In addition they do not need a pilot hole. They are hammered into the wood and then tightened with a spanner.

Description Coach screws are medium- to heavy-duty threaded devices manufactured in mild steel in a limited range of sizes from 6 to 25 mm diameter and in lengths up to 400 mm (in the UK the maximum diameter obtainable can be as little as 12.7 mm, and length 150 mm). Each has a square head with which a spanner can engage.

Applications Coach screws are intended to make medium- to heavy-duty timber-to-timber, or steel-to-timber connections in which the structural performance is more important than the appearance. They are quick to install, not always requiring a pre-drilled, matching hole in the component and the base. Coach screws are removable by means of a spanner. They can be re-placed in the same hole, but repeated replacement invariably means loss of holding power.

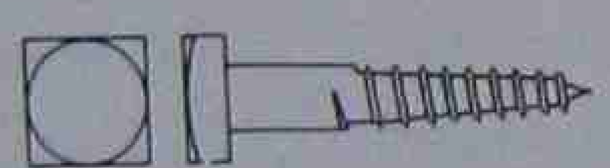


Fig. 4.10 Coach screw

Setting instructions

1. Hammer the coach screw into the timber. If the subsidiary component is metal, this will have to be pre-drilled. It is sometimes more convenient to drill a starter hole through the timber subsidiary component. This makes driving more controlled when the component is substantial; and if less robust, avoids splitting.
2. Finally tighten the coach screw using a spanner.

Note: To obtain the best performance from a coach screw, the plain (unthreaded) shank should penetrate the base by at least 25 per cent of its length.

A4 Nail plates

These devices are used to make edge-to-edge joints between pieces

of timber, usually of the same thickness, lying in the same plane. There are two basic types: toothed metal nail plates and hand-nail plates.

The former consists of metal plates with teeth punched out of their surface in a regular pattern (Fig. 4.11). They cannot be installed by hand, but are intended to be placed by special machinery in a factory, the plates being positioned over the junction between two or more timber components and then pressed home. They are used particularly in the manufacture of trussed rafters and similar prefabricated timber assemblies.

Fig. 4.11 Punched nail plate (Hydro-Air)

Because they are used by fabricators holding licences from the plate manufacturers, these devices are beyond the scope of this book.

Hand-nail plates, on the other hand, are not subject to the same restrictions. They are metal plates punched out with a pattern of nail holes which are intended to receive normal nails, driven either by hammer, or by a nailing machine (Fig. 4.12). They are used to make similar edge-to-edge joints as the punched nail plate.

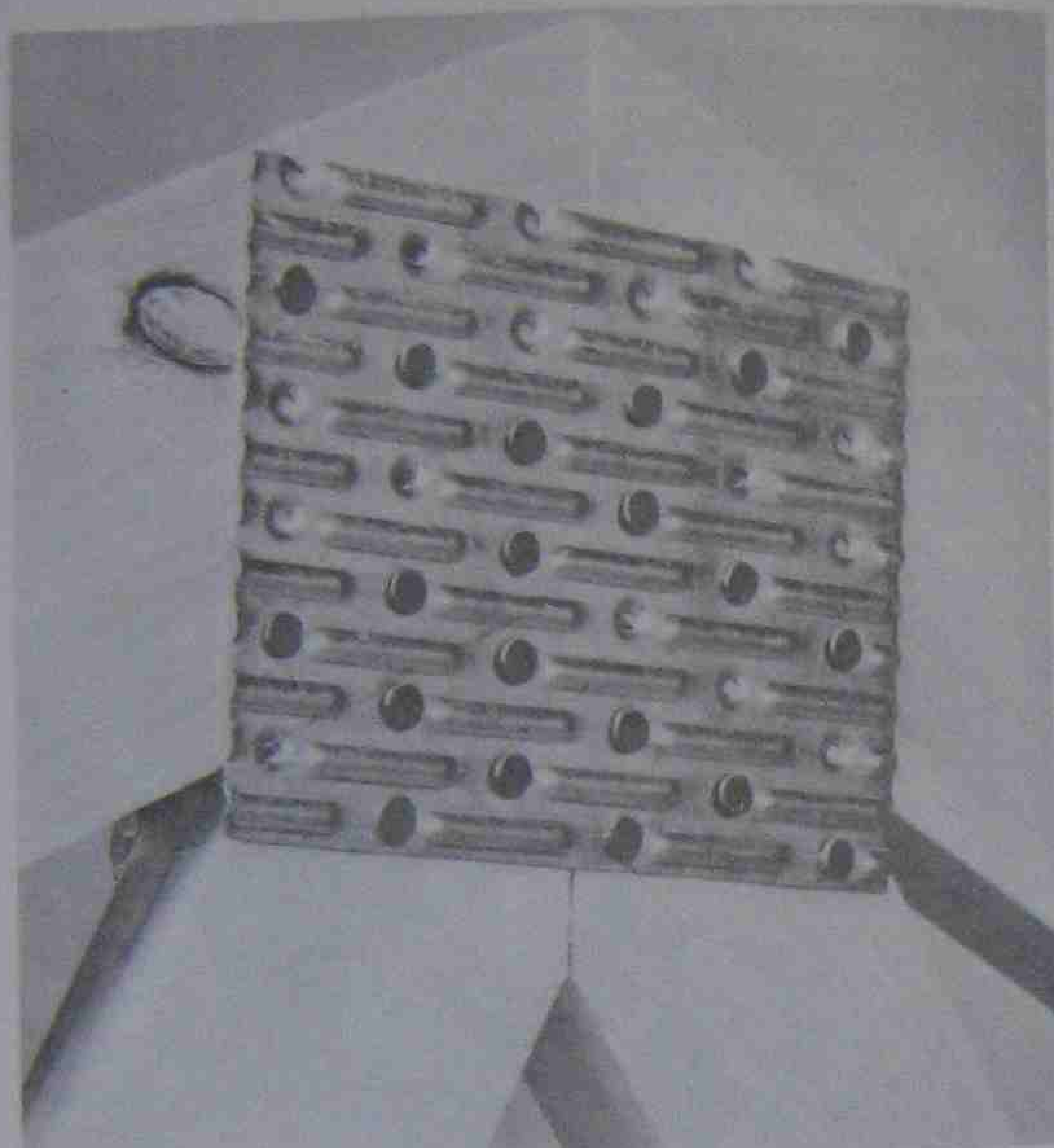


Fig. 4.12 Nail plate (Bat-U-Nail truss plate)

The most commonly specified nails used with hand-nail plates are 30 × 3.75 mm sherardized twist nails, hand-driven, or 30 × 3.35 mm round-shank T-nails driven by a pneumatic nailing machine.

It should be noted that hand-nail plates are similar to the metal accessories examined in section C of this chapter and could equally well have been considered in that section.

Description Hand-nail plates are usually made of 1 mm galvanized mild steel, perforated with rows of holes through which the specified size of nails is to be driven. The range of plate sizes available from different manufacturers varies considerably. The most commonly-used sizes are in a range from approximately 100 to 300 mm. Exact sizes are selected according to the size and

type of joint to be made and the number of nails that must be driven into each piece of timber.

Applications Nail plates can be used to construct edge-to-edge joints between two or more pieces of timber of the same thickness and lying in the same plane. They are particularly useful in the fabrication of such assemblies as trussed rafters, where their use allows smaller sections of timber to be joined together without resort to large plywood gusset plates. Nail plates are applied to both sides of the joint. The nail plate manufacturers' literature gives details of the size of plate and the number and size of nails to be used in various applications.

Setting instructions

1. Ensure that the pieces of timber to be joined have accurately been cut to fit and are firmly clamped together with no gaps between their contact surfaces.
2. Select a nail plate of the correct size and place it over the junction of the timber components in such a way as to allow the required number of nails (specified by the manufacturer or the engineer) to be driven into each piece of timber.
3. Distribute the nails evenly over the surface of the plate avoiding driving nails too close to the edge of the timber (usually minimum edge distance can be taken as 27 mm and end distance 38 mm).

A5 Corrugated fasteners

These fasteners, often referred to as 'dogs', are the simplest and most lightweight of the fasteners which are used to make edge-to-edge connections between pieces of timber in the same plane – butt joints and mitres (Fig. 4.13).

Description A corrugated fastener, as its name suggests, comprises a small corrugated plate with one edge sharpened for driving into the timber. The corrugations are skewed towards one another as they approach the sharpened edge so that the action of driving the fastener draws the parts of the joint together. Various widths of fastener are obtainable and depths range from 6 to 25 mm.

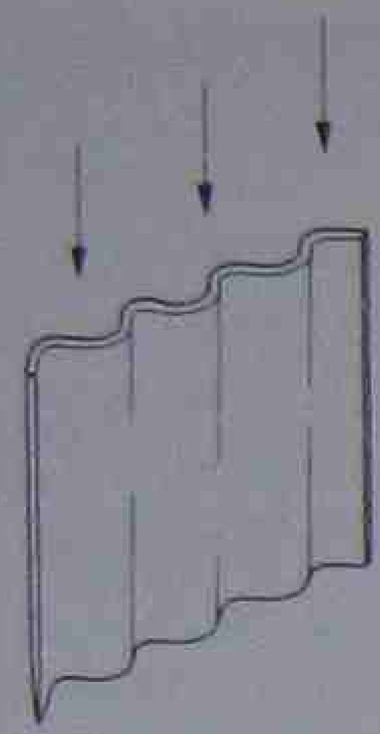


Fig. 4.13 Corrugated fastener

Applications Corrugated fasteners are used to make lightweight connections between timber components in the same plane, the device being driven across the line of the joint so that one half of the fastener is in one timber component and one half is in the other. They are only used where the appearance of the joint is not important. The slight taper on the fastener tends to draw the joint components together.

Section B: Through-fixings – type 2 (a)

Through-fixings, used to make overlap joints between timber components or between metal components and timber bases, depend for their effectiveness on some form of bolt to clamp the parts of the joint together. This clearly, therefore, involves the pre-drilling of an accurate hole through both component and base before the assembly of the joint.

In addition to the simple bolt (and its variants, the carriage bolt and the coach bolt), this family of fixings includes toothed connectors, split-ring connectors and shear plate connectors.

B1 Bolts

Hexagonal headed bolts, used alone to make structural connections between substantial pieces of timber, or between steel plates and timber bases, do have disadvantages if heavy loading is anticipated and good structural performance is needed.

Because the bolt exerts stress only on the relatively small area of timber through which it passes, the structural capacity of the

whole timber area is never fully used (maybe only 40 to 60 per cent of it is stressed). In addition, the clamping effect of tightening the bolt does not result in sufficient friction between adjoining faces of timber to enhance the strength of the joint. Seasonal moisture content changes in the wood and thermal movement in the bolt in time destroy even this limited effectiveness. As a result, a number of connections has been developed for use with bolts which improve their performance by spreading the load more widely and making component contact more positive (see sections B2, B3 and B4).

Two points should be remembered when obtaining bolts for use in timber construction:

1. Make sure the threaded portion of the shank is long enough to draw the joint together fully. This is particularly important when a toothed connector is being used (see section B2). Usually bolts, because they are designed primarily for metal construction, are only threaded up the shank for a distance of two-and-a-half times the bolt diameter, but bolts with longer threads can be obtained by special order.
2. Over-sized washers are usually essential under both the bolt head and the nut. This is to avoid the washer digging into the wood if loading tends to cause angular displacement of the bolt (Fig. 4.14). Once more the standard sized washers supplied with the bolt tend to be satisfactory for steel joints, but not for wood.

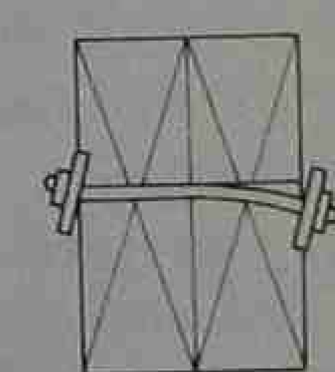


Fig. 4.14 Angular displacement of a bolt joint

Carriage bolts have practically disappeared from the British market. They have square heads and nuts, coarse buttress threads and are now only met in alteration work. Coach bolts, on the other hand, are still used to some extent in the building industry. Carriage bolts are sometimes confused with coach bolts, maybe because the Americans refer to coach bolts as carriage bolts.

The coach bolt has a cup head with a square underhead. This is

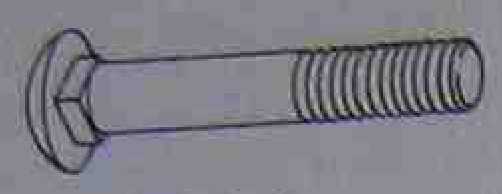


Fig. 4.15 Coach bolt

inset into the wood and resists the tendency of the bolt to revolve when the nut is being tightened. Coach bolts are ideal where access for a spanner is only available on one side of a joint.

Description Most bolts used in timber connections are ordinary mild-steel bolts (black bolts) with hexagonal heads and nuts. They are available in a wide range of lengths and diameters which are detailed more fully in Chapter 5.

Applications Bolts are used without the aid of an additional connector, to make medium-duty timber-to-timber or metal-to-timber connections. They normally have to be fitted with over-sized washers to spread the load under the bolt head and the nut in order to avoid timber deformation.

Setting instructions

1. Drill a hole through the component and the base, 1.5 mm greater than the bolt diameter for bolts up to 25 mm diameter. (A greater allowance will be needed for larger bolts.) Drilling can be carried out through component and base in one operation by spiking the members of the joint together, if this is convenient: otherwise careful marking out and the possible use of a template is recommended.
2. Assemble the elements of the joint. Place the over-sized washer on the bolt and insert in the drilling.
3. Place a second over-sized washer on the threaded end of the bolt and run-on the nut. Tighten with two spanners; one applied to the head to avoid it revolving, the other used to turn the nut.

B2 Toothed connectors

Toothed connectors can be either single- or double-sided: the single-sided version is used to make connections between metal components and timber bases (or between a timber component and a concrete base); the double-sided connector makes a permanent overlapping timber-to-timber joint. A demountable timber-

to-timber joint can be made by using two single-sided toothed connectors, back to back (Fig. 4.16).

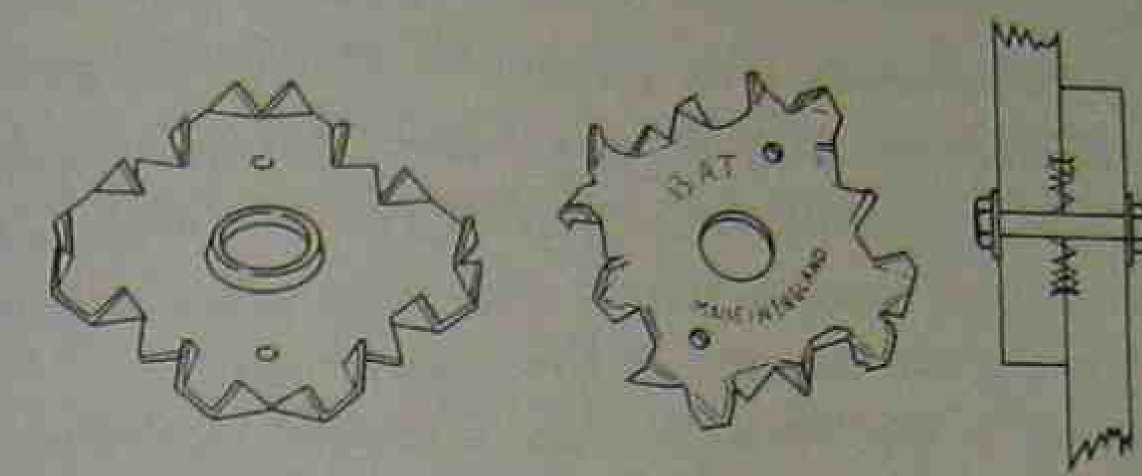


Fig. 4.16 Toothed connectors, single and double

Description Toothed connector plates are made of galvanized steel plate and are usually available in four sizes, either round or square; 38, 51, 63 and 76 mm. There is a less common larger-sized connector (89 mm), but this is not referred to in CP 112: Part 2: 1971. All connectors but the largest and the smallest, are designed for use with an M12 hexagonal-headed black bolt; the 38 mm connector is sometimes used with an M9 bolt and the 89 mm with an M20 bolt.

Applications Connectors are designed to make reliable heavy-duty timber-to-timber connections, or joints between a steel component and a timber base. In the former case a double-sided connector is used (unless the joint is to be demountable, when two single-sided connectors, back-to-back, are installed). In the latter case a single-sided connector is used with its teeth embedded in the timber.

Setting instructions

Double-sided toothed connector (single-sided connector similar)

1. Place the timber members in the position they will assume when the joint is assembled and clamp or spike them together.
2. Drill through all timbers at the point where their centre-lines intersect. Ensure that the bolt hole (no larger than 1.6 mm greater than the bolt diameter) is drilled at right angles to the surface of the joint. (Alternatively, each member can be drilled separately using a template. Complete accuracy is essential.)

3. Position the connector between the members of the joint and centrally over the drilling. Reposition the joint components.
4. Embed the connector by either:
   (a) threading the bolt through the joint and tightening the nut to compress the gaps between components (in the case of hard timbers, this may place too great a strain on the threads of the bolt or nut);
   (b) passing a high-tensile steel screwed rod with large plate washers through the joint and tightening the nuts (compression tools like this are obtainable from the fixing manufacturer);
   (c) using screw cramps or hydraulic pressure.
5. Once the teeth of the connector are fully embedded, remove the compression tool (if one were used) and replace it with the black bolt with square washers, and tighten.

Note: In positioning the connector, specified minimum edge and end distances should be observed carefully. The teeth of the connector should be completely embedded, but no effort should be made to squash the plate of the connector into the wood. The components of a correctly-made toothed connector joint will stand apart by the thickness of the connector plate.

Where more than three connectors are used in any one joint, a threaded rod compression tool must be used to achieve proper embedment.

B3 Split-ring connectors

Split-ring connectors are used to make overlapping timber-to-timber joints which will carry greater loads than toothed connector joints.

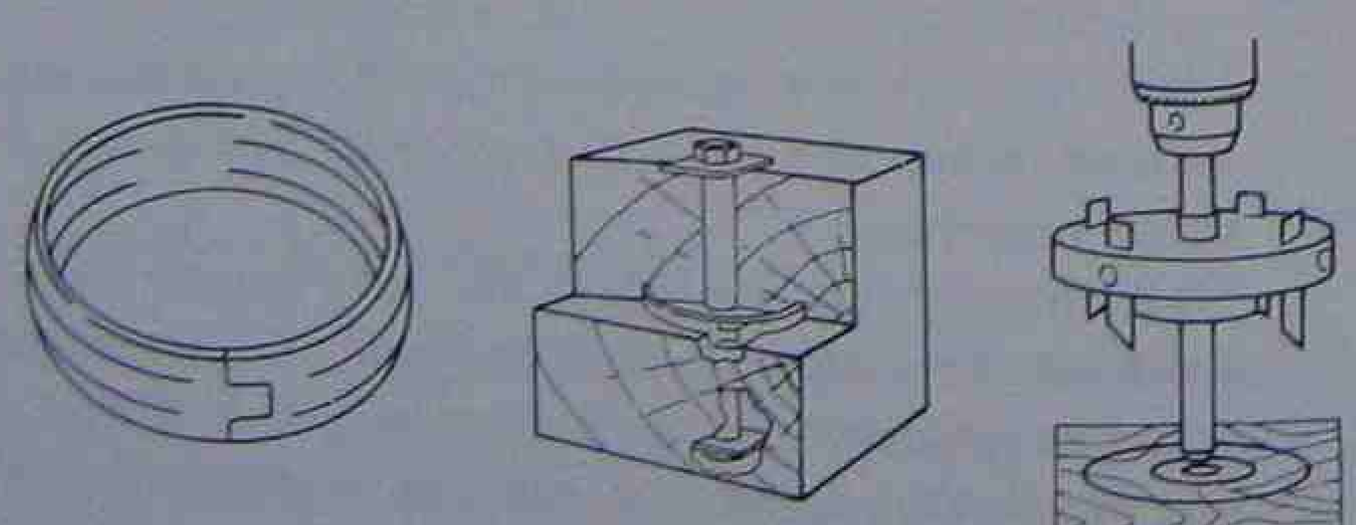


Fig. 4.17 Split-ring connector and grooving tool (also used with shear plate connectors)

Description Split-ring connectors consist of a galvanized hot-rolled low-carbon steel split ring, either bevelled or straight-sided, which is used in conjunction with a hexagonal-headed black bolt. Ring diameters are 64 and 102 mm (for use with M12 and M20 bolts respectively). At least one manufacturer produces a 50 mm diameter split ring, but this size is not included in CP 112: Part 2: 1971.

Applications The split ring is used to make heavy-duty timber-to-timber fixings of greater strength than a double-sided toothed connector. The bevelled ring is easier to place and has less slip under load than the straight-sided ring. Both types are set in a circular groove, cut in the contact faces of both pieces of timber, at the centre of which groove is the drilling for the bolt which holds the joint together. The adjoining faces of the timber components are in contact in a correctly assembled split-ring joint, not standing slightly apart, as in the case of the toothed connector joint.

Setting instructions

1. Place the timber members in the position they will assume when the joint is assembled and clamp or spike them together.
2. Drill through all timbers at the point where their centre-lines intersect. Ensure that the bolt hole (no larger than 1.6 mm greater than the bolt diameter) is drilled at right angles to the surface of the joint. (Alternatively, each member can be drilled separately using a template. Complete accuracy is essential.)
3. The contact surfaces should be grooved with a grooving tool to a depth of half the ring concentrically about the bolt hole. There are various tools available to carry out this work, some obtainable from the fixings manufacturers. A cutter-head with a 'pilot' which can be located in the bolt hole is one option; or the bolt hole and the groove can be cut in one operation using a power tool. There is, however, a limit to the diameter of ring groove that can be cut by this type of tool. Grooves should be cut slightly larger than the diameter of the split ring to ensure its best performance, even should the timber dry and shrink during use (see the note below).

4. Remove all chips and shavings from the groove.
5. Expand the ring and insert it in the grooves in both timber members.
6. With the joint reassembled, insert the bolt, complete with washers, run-on the nut and tighten.

Note: The dimensions of the groove to receive the split ring should always be those recommended by the manufacturer. In the case of 63 and 100 mm diameter split rings one manufacturer recommends the dimensions given in Table 4.3. The observance of edge and end minimum distances is essential.

Table 4.3

Split ring diameter	63 mm	100 mm
Inside diameter of groove (mm)	65.0	104.0
Width of groove (mm)	4.6	5.3
Depth of groove (mm)	9.5	12.7

B4 Shear plate connectors

When heavy-duty timber-to-steel joints are required, the shear plate connector has an equivalent performance to a split-ring connector (i.e. better than would be obtained by using a single-sided toothed connector).

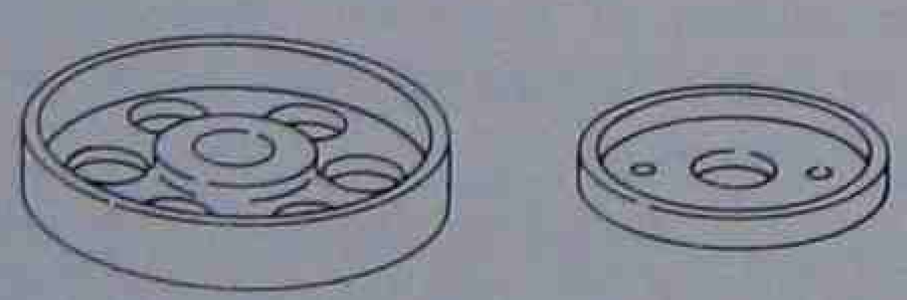


Fig. 4.18 Shear plate connector

Description Shear plate connectors are similar to single-sided toothed connectors, except that the circular plate has a flange on its circumference rather than a series of teeth. This is inset into a groove in the timber like a split-ring connector. The circular plate is also inset into the wood. Shear plate connectors are made from pressed steel or malleable iron and are used in conjunction with hexagonal-headed black bolts. Two sizes of shear plate are

commonly available: 67 and 102 mm diameter; both are used with M20 bolts.

Application Shear plate connectors make reliable heavy-duty connections between timber bases and metal components (they can also be used in pairs, back-to-back, to produce demountable timber-to-timber connections). Shear plate joints are stronger than those produced by toothed connectors. Because the shear plates are completely set in sinkings in the timber, the adjoining faces of the joint members are in contact when a correct shear plate connector joint has been made.

Setting instructions

1. Drill the timber member to receive the bolt. The drilling diameter should be no larger than 1.6 mm greater than the bolt diameter. Ensure the bolt hole is drilled at right angles to the surface of the joint. Complete accuracy is essential.
2. The timber contact surface should be grooved and rebated to receive the shear plate, concentrically about the bolt hole, using a 'dapping tool' (see note below). These are often obtainable from the fixings manufacturer.
3. Remove all chips and shavings from the groove and sinking.
4. Insert the shear plate. Assemble the joint and insert the bolt, complete with washers, and tighten the nut.

Note: The dimensions for the groove and sinking recommended by the manufacturer should be followed. In the case of 67 and 100 mm diameter shear plate connectors, one manufacturer recommends the dimensions given in Table 4.4. The observance of edge and end minimum distances is essential.

Table 4.4

Shear plate connector diameter	67 mm	100 mm
Sinking diameter (mm)	67.0	100.0
Sinking depth, maximum (mm)	11.5	16.5
Sinking depth, minimum (mm)	6.5	7.0
Width of outer groove (mm)	5.0	6.5

Section C: Fixing accessories

A number of metal fixing accessories has been developed specifically to eliminate the need to construct on site traditional joinery joints which require considerable skill and time to form. When used in conjunction with the specified nails, these metal accessories make heavy-duty structural connections between timber members meeting at right angles or in the same plane.

Joist hangers (similar to the joist hangers discussed in section D of Chapter 3, but which are nailed to timber members rather than bolted or nailed to, or built into, mass walling) remove the need to form a tusk tenon joint between a trimming joist and a trimmer joist or a notched joint between a trimmed joist and a trimmer (Fig. 4.19). Similarly, framing anchors can be used to make simple end-to-edge joints between timber framing members and cantilever brackets to extend a timber member rather than forming a spliced joint. Some of these devices are illustrated in Fig. 4.20.

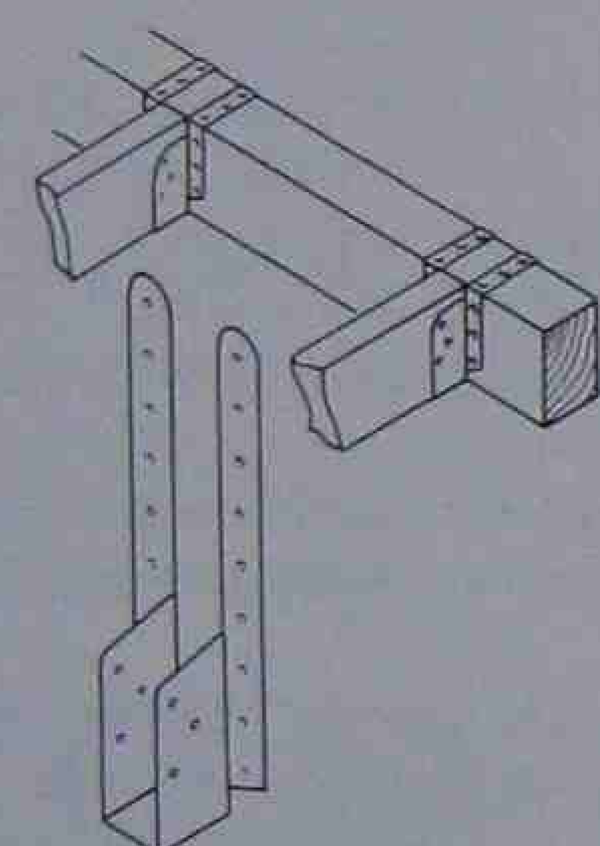


Fig. 4.19 Joist hanger to make timber-to-timber connections

The effectiveness of all these accessories depends on the thickness and length of nails used to fix them to the timber members. The manufacturer's recommendations should always be followed.

Joist hangers

Description Joist hangers are produced in a variety of sizes to suit various cross-sectional sizes of joist. They are usually manufactured of hot-dipped galvanized mild steel, from 2.7 to

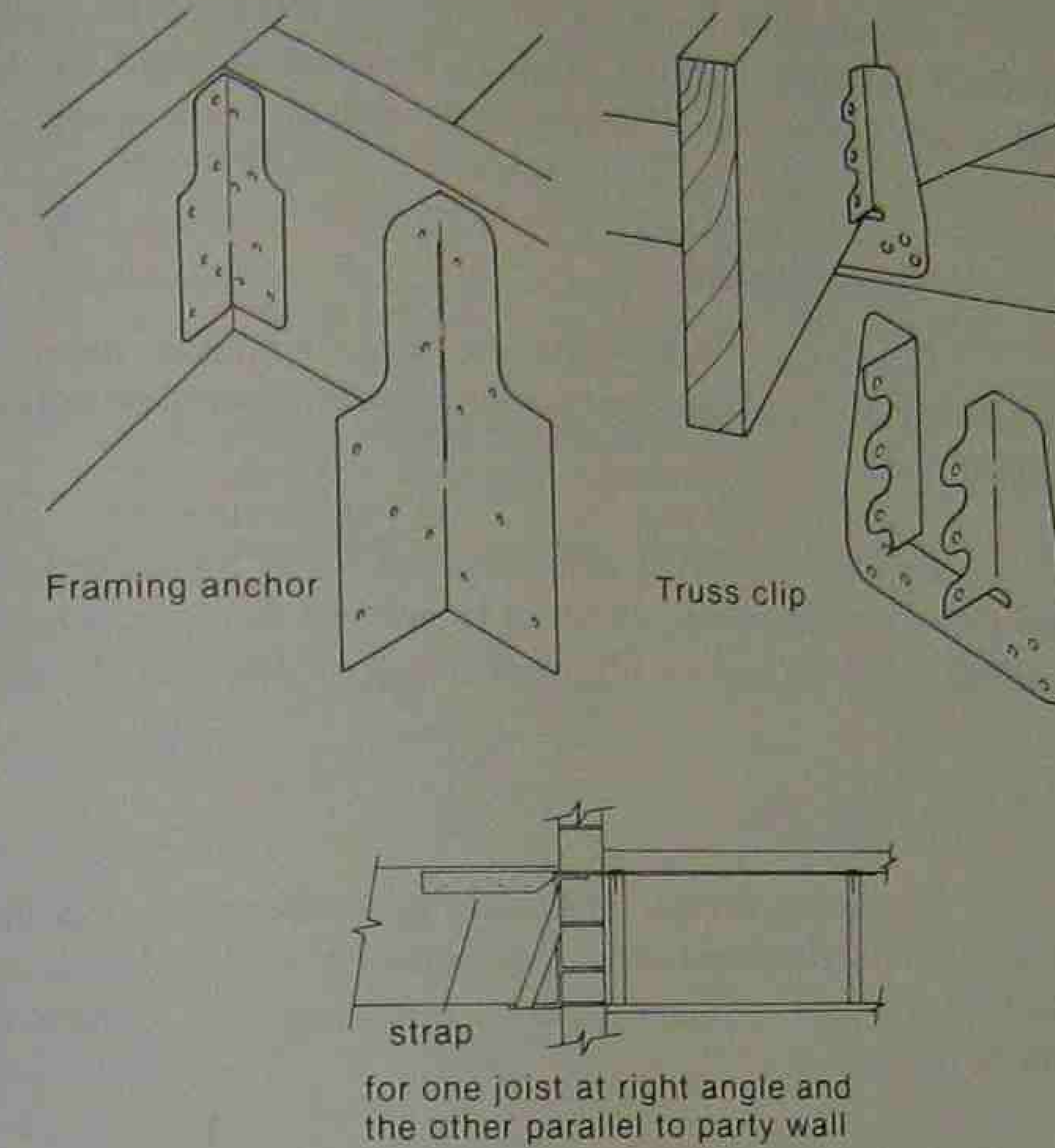


Fig. 4.20 Framing anchors, truss clips and straps

1 mm thick, depending on the strength required of the connection. These devices are usually pre-holed for fixing with 32 mm long × 3.75 mm square twisted nails, although some of the more heavy-duty hangers are intended to be used with 12 mm diameter bolts.

Applications These devices are intended to make connections between timber joists meeting at right angles. Nails should be driven through all holes in the hanger.

Framing anchors

Description Framing anchors are usually made from 1.2 mm galvanized steel and are intended to make medium-duty connections when fixed with 32 mm long × 3.75 mm square twisted nails.

Applications Framing anchors are intended to make connections between framing timbers meeting at right angles (see Fig. 4.20). Nails should be driven through all holes in the anchor.

Truss clips

Description Truss clips are usually made from 1 mm galvanized steel and are fixed with 32 mm long × 3.75 mm square twisted nails.

Applications Truss clips are used to fix a roof truss to the wall plate (see Fig. 4.20). In order to achieve the correct resistance to uplift, nails should be driven through all holes in the clip.

Straps

Description Straps are usually made from 2.5 mm thick × 35 mm wide galvanized steel, pre-punched with holes at 25 mm offset centres and are fixed with 32 mm long × 3.75 mm square twisted nails.

Applications Straps are used for a number of bracing applications, often in conjunction with built-in double joist-hangers (see Fig. 4.20). Nails should be driven through all holes backed by the timber members.

Cantilever brackets

Description Cantilever brackets are usually made from 1.5 mm thick zinc-coated mild steel to suit a range of joist widths from 38 to 75 mm and joist depths from 150 to 255 mm. Nails to be used are 32 mm long × 3.75 mm square twisted nails.

Applications These brackets can be used to form a lengthening joint between joists, provided the joint is made not too far away from a support. Manufacturers lay down precise instructions as to how the permitted dimension between joint and support can be determined. Cantilever brackets *do not* allow the indiscriminate use of random joist lengths, but they do avoid the overlapping of end-on joists over supporting walls with a consequent saving in

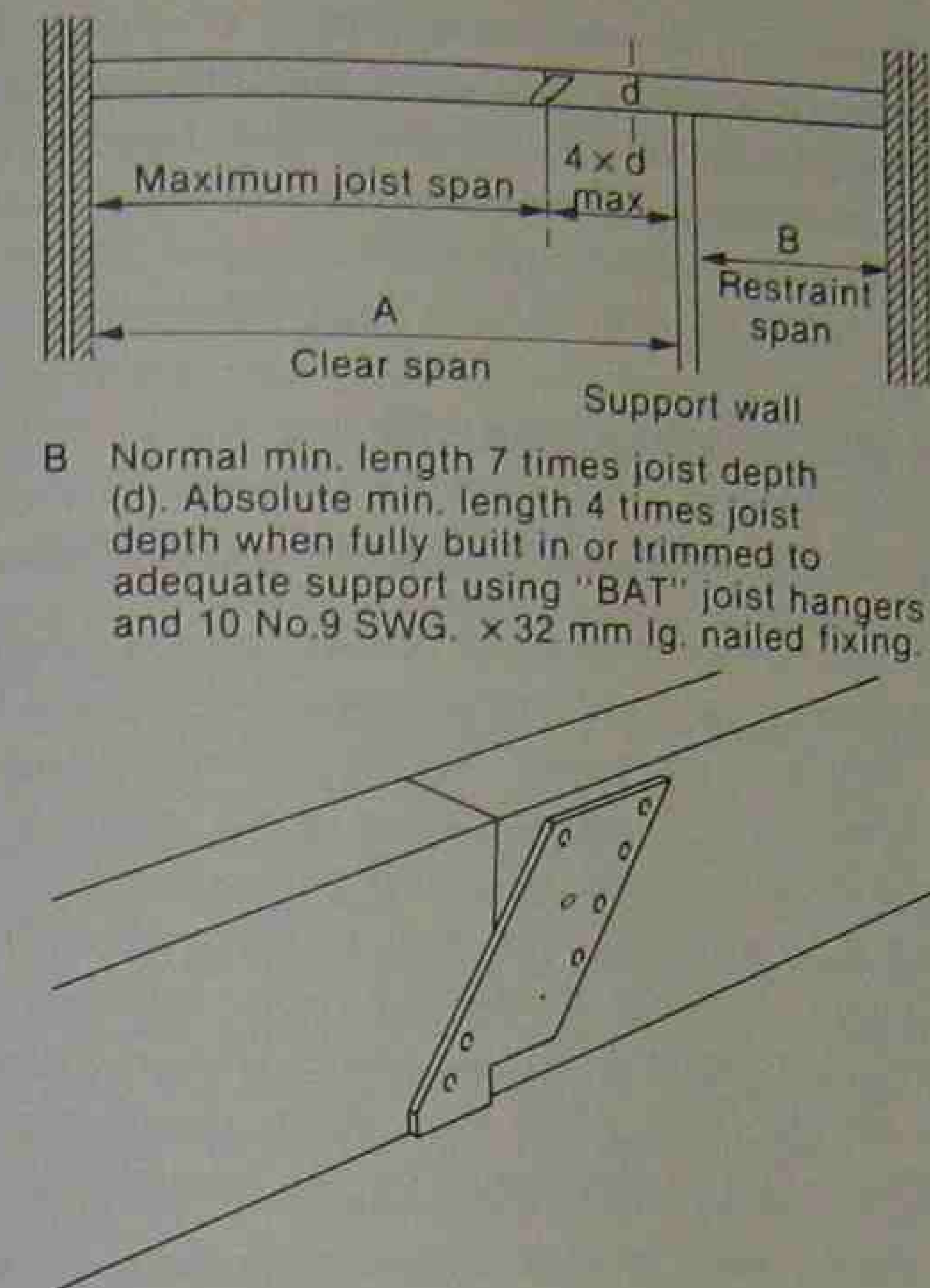


Fig. 4.21 Cantilever bracket

timber. They also result in a reduction in the cross-sectional area of timber required. (Fig. 4.21)

Setting instructions

1. Nail the cantilever bracket to the joist to be cantilevered over the supporting wall. This is best done on the ground. The joist must be located squarely against the upward projecting lug in the base of the bracket and with the underside of the joist and bracket in line.
2. Notch out the underside of the other joist (the meeting joist) to receive the bracket.
3. Place the cantilever joist in position.
4. Place the meeting joist so that its end fits squarely against the upward projecting lug in the base of the bracket and nail in position.
5. All nail holes should be filled.

Note: Since the increased popularity of timber-framed forms of house construction, a number of specialized metal accessories have been introduced (see *Timber Frame Housing* by Jim Buchell in this Site Practice series).

Section D: Background

Strength of mechanical fixings in timber bases

The strength of mechanical fixings to timber bases is affected by the species of timber in question, some species being considerably stronger than others. The terms 'softwood' and 'hardwood' should not be confused with an indication of the strength of the wood. The distinction is purely botanical and many hardwoods are not as strong as some softwoods. Softwoods are broadly timber from coniferous trees and hardwood from deciduous trees. Most timber used for structural framing comes from the softwood category. Generally the more dense the timber, the higher are its strength properties. This is illustrated by the statistics in Table 4.5.

Table 4.5

Species	*Density* (kg/m^3)	*Maximum bending strength* (N/mm^2)	*Maximum compression strength parallel to grain* (N/mm^2)
Pitch pine	769	107	56.1
Douglas fir	545	93	52.1
European larch	545	92	46.7
Baltic redwood	481	83	45.0
Western hemlock	465	83	47.4
European spruce	417	72	36.5
Western red cedar	368	65	35.0

The stronger the wood, generally the better the fixing, provided of course that the quality of the pieces of timber being compared is equivalent (same disposition of knots and slope of grain, etc.) and their moisture contents are the same. All the

figures in Table 4.5 relate to samples of the species with a 12 per cent moisture content.

The direction of the grain also affects the strength of the timber. For instance, because of the layout of the fibrous cells which make up the wood, compressive forces parallel to the grain can be sustained more readily than similar forces applied perpendicular to the grain. In tension, wood's strength perpendicular to the grain is lower, but parallel to the grain very high – higher, in fact, than compressive strength. All this has a marked effect on the way fixings are placed in timber joints, both in relation to the edges of the pieces of timber being joined and in relation to other fixings in multiple fixing joints.

The structural performance of mechanical fixings in timber bases, therefore, depends on the size and thickness of the timber base (and, if appropriate, the timber component) as well as the size and character of the nail, screw, bolt or connector used to make the fixing. Often the strength of the joint is determined by a combination of the characteristics of the timber member(s) and the fixing. Thus the size of the fixing is critical to the strength of the joint, but its strength can be undermined by its use in an inappropriately-sized piece of timber, or if it is placed in such a way as to cause the wood to split or form planes of weakness.

The other vital factor to be considered is the possibility of the corrosion of the metal fixing. This will be considered later in this section.

The design of joints made by timber connectors is a very specialized business and so joints containing these devices are usually provided with detailed engineering data in the form of drawings and specifications, giving precise instructions concerning the positioning of connectors. This aspect is somewhat comparable to bolted structural steelwork connections and is therefore, like them, beyond the scope of this book. Extensive recommendations are given in CP 112: Part 2: 1971 which should be consulted if specific advice is required concerning the design of joints using connectors.

There are a few pieces of basic advice which can be given concerning other structural connections using nails, staples, screws or bolts and which will help to ensure fixings of adequate performance.

Timber-to-timber joints can be loaded axially or laterally (Fig. 4.22).

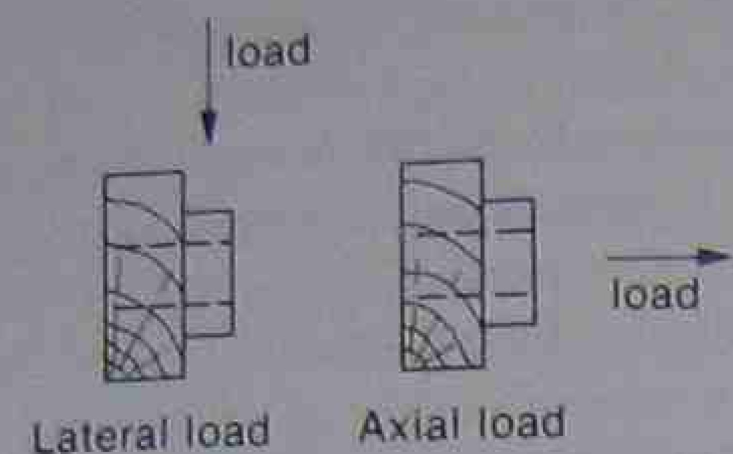


Fig. 4.22 Lateral and axial loading

Lateral loading

All mechanical fixings achieve their greatest strength when driven into the side grain of the wood. In the case of a nail, its optimum effect is realized when the length of the nail is two-and-a-half times the thickness of the timber component. When driven into the end grain, the nail's (or screw's) performance is reduced by about a third. Also nailing or screwing into green timber or into timber that will be exposed to the weather has a reduced performance. The same applies to bolted connections.

Staples have a performance under lateral load equal to a nail one-and-a-half times the staples diameter.

In making lapped joints between pieces of timber, the nailing area is clearly limited (Fig. 4.23) and therefore the number of nails is severely restricted. To achieve the best performance it is therefore necessary to use the largest-diameter nail possible with the deepest penetration that is consistent with the sizes of the component and the base without causing the timber to split. Spacing of nails is therefore critical. This will be discussed in more detail later in this section.

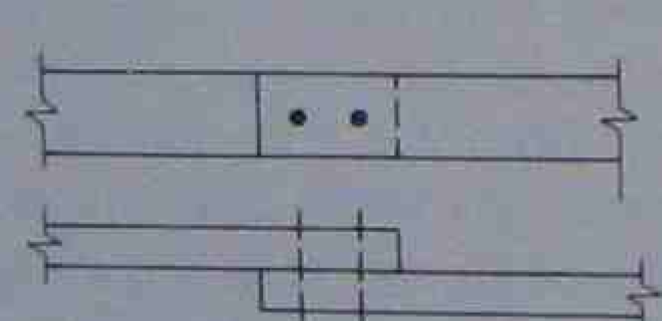


Fig. 4.23 Lap joint

If the members of the joint are in line, strong joints can be made using plywood gussets nailed with generally smaller nails (Fig. 4.24). A useful rule of thumb is that the nail length should be 4 times the thickness of the plywood – usually not less than 50 mm long. The following nail diameters (Table 4.6) are sug-

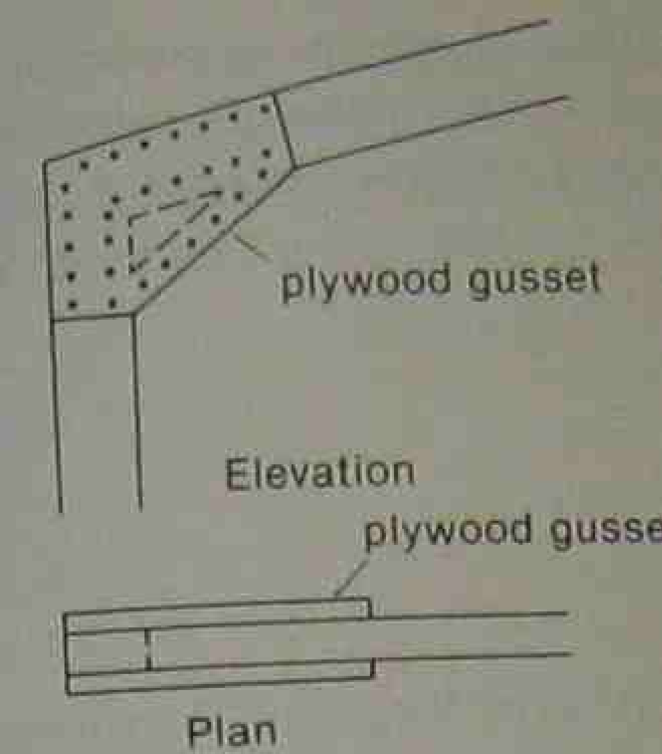


Fig. 4.24 Gussetted joint

gested by TRADA (Timber Research and Development Association) Mechanical fasteners for structural timberwork, Nov 78, *TRADA Wood Information*), based on the thickness of the plywood gusset:

Table 4.6

Diameter of nail (mm)	Thickness of ply (mm)
2.64	8.0
3.25	9.5
3.66	12.7
4.47	15.8

Axial loading

This form of loading is resisted by the holding power of the fixing, usually a nail or screw (a bolt is rarely used to withstand axial loading). Normal nails are not strong withdrawal fixings, but improved nails (square twisted or annularly threaded nails) have a better performance, allowing the penetration depth of the nail to be decreased by a third without reducing holding power. A screw has considerably better performance than a nail regarding withdrawal resistance.

No type of nail or screw produces good withdrawal performance when driven into end grain, but the performance of a nail is unaffected by use in green timber or timber exposed to cyclic damp conditions. A screw, however, has reduced performance in green timber. Double skew-nailing increases resistance of nailed joints to axial failure (Fig. 4.25).

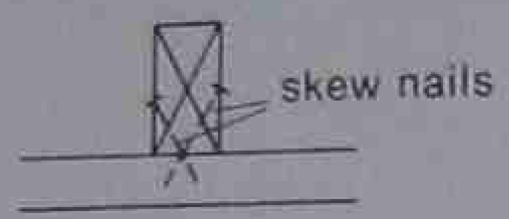


Fig. 4.25 Skew nailed joint

It should be remembered that it is the length of the thread of a screw embedded in the timber base which gives it its holding power. Threaded shank unembedded in the base has no beneficial effect on the joint.

Spacing of nails, screws and bolts

The recommended spacing of nails, screws and bolts to achieve their best performance in timber-to timber joints is set out in Table 4.7.

Corrosion

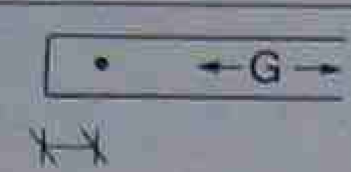
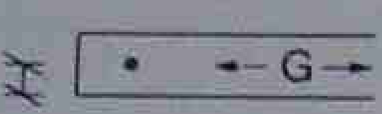
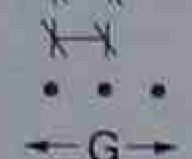
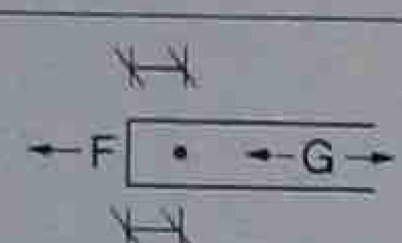
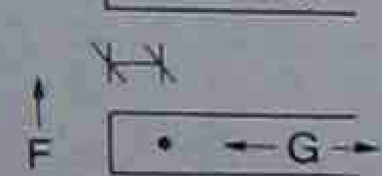
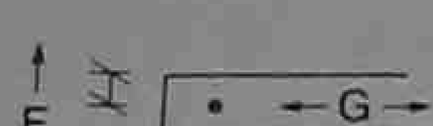
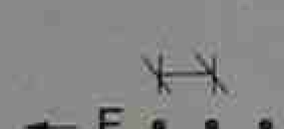
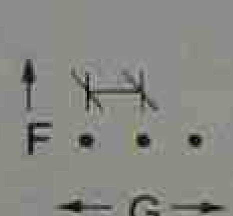
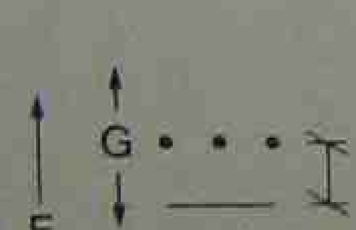
Dampness can be a cause of corrosion in ferrous timber fixings, as has already been noted in respect of other categories of ferrous fixing. However, non-ferrous alternatives are more expensive and rarely warranted except in the most critical circumstances. In positions where the fixing is to suffer damp conditions, the need for long-term, high-strength performance needs to be weighed against the additional cost of the non-ferrous alternative. Often the use of a zinc-coated fixing is sufficient defence. In the form of galvanizing it will stand a fair amount of rough treatment during installation before it is in danger of suffering premature failure.

Timber treated with water-borne preservative salts needs a stabilization period of about 14 days between treatment and the introduction of metal fixings. Otherwise fasteners *must* be non-ferrous (but *not* aluminium) after checking with the manufacturers of both the fixing and the preservative that their products are compatible. This does not apply to solvent-based preservatives.

When metal fasteners are used in timber impregnated with flame retardants, the manufacturer's advice should always be

sought. It is often necessary to use non-corroding fasteners made of materials like phosphor or silicon bronze when there is the slightest danger of damp conditions when timber treated with flame retardants is being fixed.

Table 4.7 Spacing of nails, screws and bolts

Type of fixing	*Illustration*	*Position/spacing*	*Relation to grain*	*Distance*	
				predrilled hole	*no hole*
Nails and screws		End distance	Along	10D	20D
		Edge distance	Across	5D	5D
		Spacing	Across	3D	10D
		Spacing	Along	10D	20D
Bolts		End distance	Tension; along	7D	
		End distance	Compression; along	4D	
		Edge distance	Any; across	4D	
		End distance	Any; across	4D	
		Spacing in direction of load		4D	
		Spacing at right angles to load	Load across	2.5D if $\frac{t}{D} = 1$ 5D if $\frac{t}{D} = 3$ or more (interpolate between)	
		Spacing at right angles to load (bolt to edge)	Load along	1.5D	

D = diameter of fixing; t = thickness of timber component; F → direction of load
All distances measure from centre of fixing. ← G → direction of grain

5

Mechanical fixings in metal bases

Fixings to metal bases divide broadly into two categories: heavy-duty fixings, being primarily fixings from which high and predictable performance is demanded (similar to those used to connect members of a structural steel framework together) and lightweight fixings, used generally to make connections to lighter metal bases or fixings with a less severe performance function. The first category consists of type 1 through-fixings (bolts and machine screws of various types); the second category of mainly type 2(b) base-deforming fixings.

There are two points which should be remembered in relation to both categories. Metal, as a generic description, covers a wide range of materials, some of which are more resistant to corrosion and atmospheric degrading than others. This means that the metal must either be protected from the conditions which may cause its decay, or the type of metal must be selected to be unaffected by the conditions.

Secondly, as the fixing devices are themselves made of metal and the subsidiary component may also be of metal, the problem of galvanic or bimetallic corrosion has to be considered and avoided. This problem and its avoidance has been discussed in Chapter 2 on page 9.

Section A: Heavy fixings

Heavy-duty fixings to metal bases comprise bolts and machine screws. These parallel-sided fixings are generally drilled-for, through-fixings in the category type 1 and are used in conjunction with a nut and (possibly) washers. The difference between a bolt and a machine screw is largely based on the length of its thread

and there is a fair amount of size duplication between the two ranges of fixings.

A bolt is a threaded fastener with a diameter greater than 6 mm (M6) and provided with a matching nut and washers. Below that diameter, threaded devices are considered as being machine screws. But this is not a complete definition, because machine screws can be obtained up to M12 diameter (12 mm) – well within the range of bolt diameters. A machine screw is, however, threaded for the whole of its length, whereas a bolt is only threaded for a part of its shank – usually about two times its shank diameter. This again becomes confusing in the case of shorter bolts, where this formula implies an almost complete threading of the bolt shank. In practice bolts below the lengths shown in Table 5.1 are considered as being machine screws.

Table 5.1

Shank diameter (mm)	*Black bolt lengths (mm)*	*Precision bolt lengths (mm)*
M5	25	20
M6	25	25
M8	30	30
M10	35	35
M12	40	40
M16	35	45
M20	45	55
M24	55	65
M30	85	80
M36	100	90
M42	120	110

A1 Bolts

The introduction of the metric bolt has rationalized a chaotic situation of varying threads used in this country, including British Standard Whitworth, British Standard Fine, British Association, Unified Coarse and Unified Fine threads. These still may be encountered on occasions. Now the ISO metric thread has been introduced, eventually to supersede not only previous British threads, but some continental threads as well (DIN, CNM and SI metric threads). There is also an ISO inch (unified) thread, but that is also likely to fall out of use.

ISO metric bolts are always preceded by the prefix M, followed by the diameter in millimetres (M10, M12 etc.). Coarse and fine ISO metric threads are available.

Various head styles of bolts are manufactured (Fig. 5.1) but the most common is the hexagonal all head, matched by a hexagonal nut.

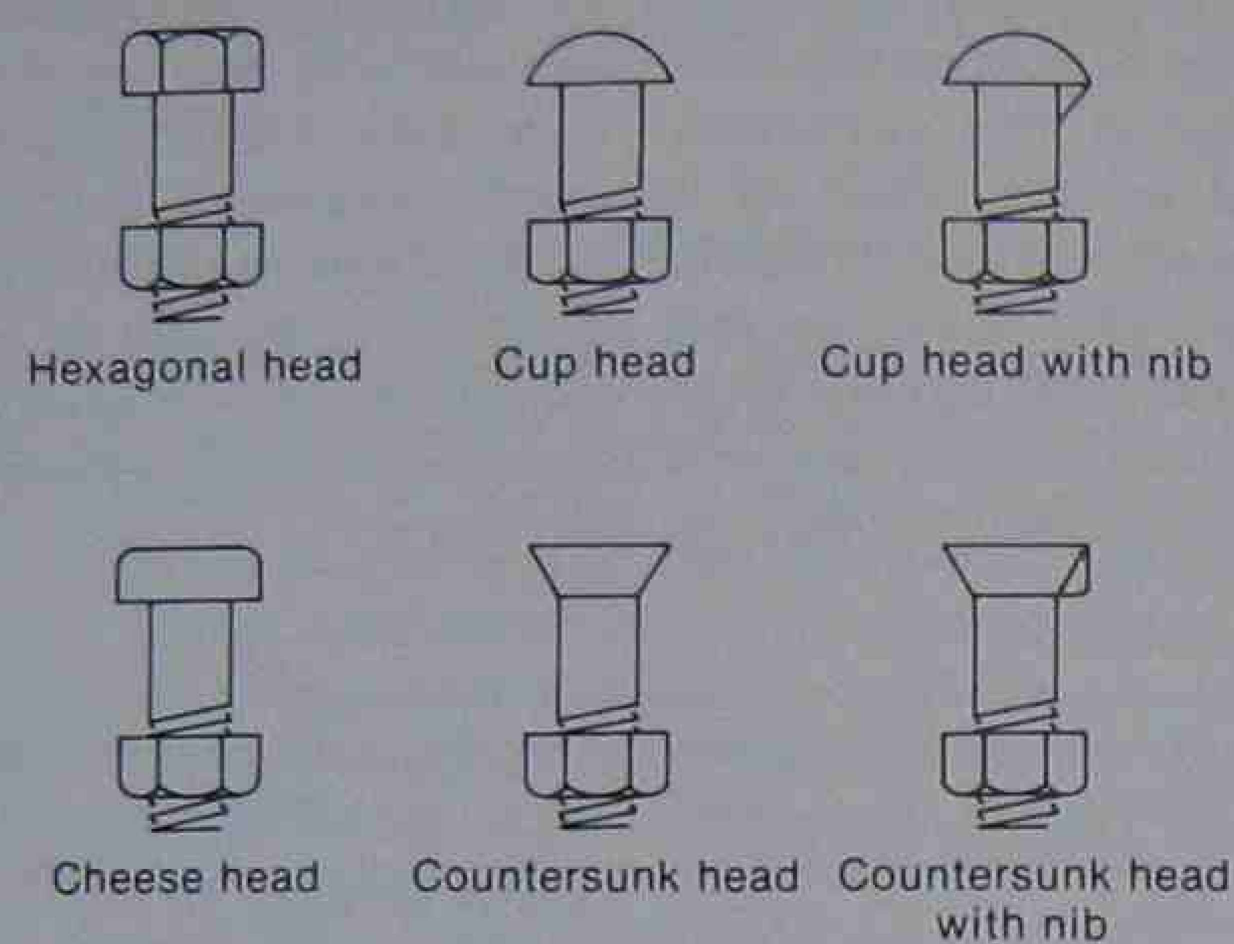


Fig. 5.1 Bolt head styles

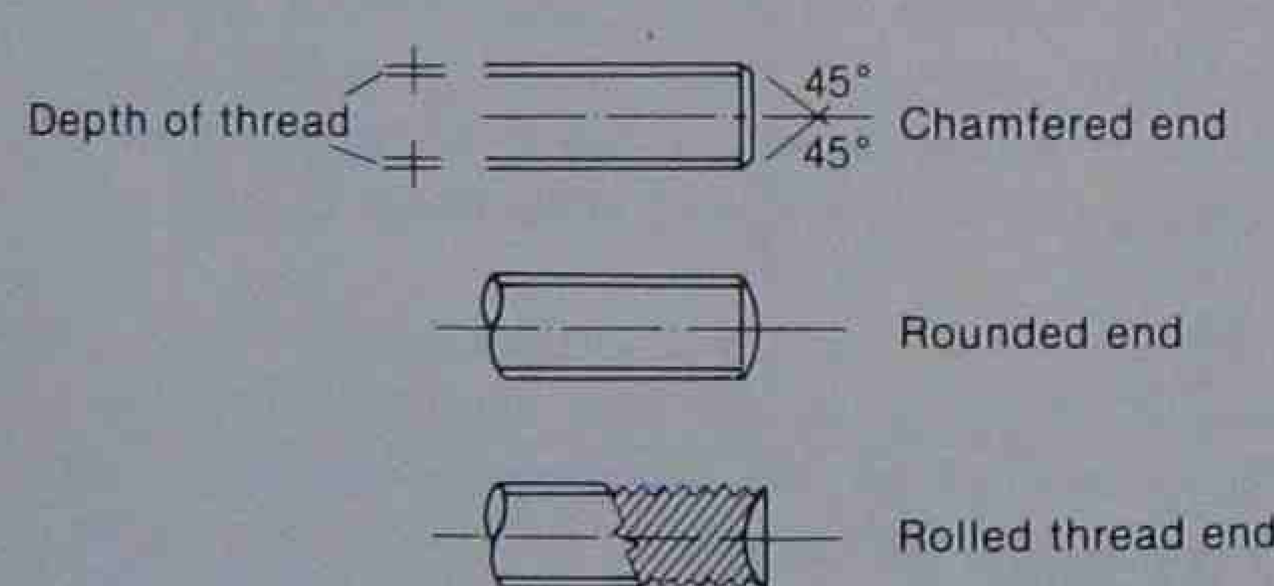


Fig. 5.2 Bolt end finishes

There are also a number of bolt end finishes (Fig. 5.2).

The effectiveness of a bolted connection is clearly dependent on the size, type and number of bolts, as well as their spacing and location. These are matters which will be detailed on the engineer's drawings or in his specification, assuming that they have structural significance, and therefore need not be covered in this book. Some guidelines on the spacing of bolts is included in section C of this chapter.

Black bolts

Description The name 'black bolt' does not refer to the appearance of the bolt, but implies an unfinished bolt with a comparatively wide range of tolerances – considerably wider than would be permissible in the case of a precision bolt. The current British Standard covers a range of diameters from M5 to M68, although manufacturers tend to stock a considerably reduced number of sizes. They are made from mild steel of a limited strength grade (see section C, p. 143) and are identifiable by an 'M' embossed on their heads, without any strength grade identification. Matching nuts and washers are supplied with the bolts.

Applications Black bolts are used for general structural steel applications where high performance is not demanded of the bolt and where slip and vibration are not critical considerations. Access is needed to both sides of the joint to make the fixing. The threaded portion of the bolt should not enter the shear plane (Fig. 5.3). To this end, bolts with shorter thread lengths, only one-and-a-half times shank diameter in length, are available.

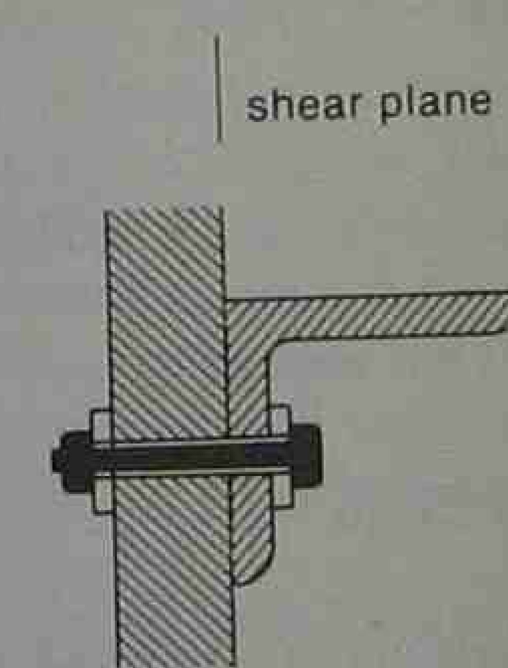


Fig. 5.3 Shear plane

Precision bolts

Description These bolts are usually made of carbon or alloy steel, supplied heat-treated after manufacture (dull black) or bright finished. They can be supplied in stainless steel. Generally, precision bolts have a greater degree of accuracy than black bolts and their strength grade is identified on their heads (8.8, 10.9 etc. – see section C of this chapter for a description of this designation). Matching nuts and washers are supplied with the bolts.

Applications Precision bolts are used where higher performance is required of the bolted connection. Access is required to both sides of the joint and the same proviso applies regarding the intrusion of the thread into the shear plane, as was made in respect of black bolts.

High-strength friction grip (HSFG) bolts

Description High-strength friction grip bolts are usually made of carbon steel and, when tightened to a predetermined shank tension, they clamp the component to the base in such a way that the loads are transferred by friction between the parts, not by shear in, or bearing on, the bolts or parts of the connected members of the joint. Part 1 general grade bolts (BS 4395) come in a size range from M12 to M36; Part 2 bolts (higher-strength grade) from M16 to M33. Thread lengths on Part 2 bolts are slightly longer than the normal thread length. High-strength friction grip bolts are used with various types of washer, flat round, flat square and square tapered (Fig. 5.4).

Fig. 5.4 Nuts and washers for use with HSFG bolts

In the case of Part 2 bolts, the strength grade is marked on the bolt; Part 1 bolts have three radial lines 120° apart on their heads, but no strength grade.

Applications High-strength friction grip bolts are used in high-strength applications where slip must be eliminated. Once more access is required to both sides of the joint to make the fixing. Part 2 bolts should not be used where the loads operate along the axis of the bolt. Instead use a waisted shank bolt.

As it is the action of tightening the nut on the bolt which induces the tension in the shank and causes the friction between the joint plies, the procedure of tightening must be carefully con-

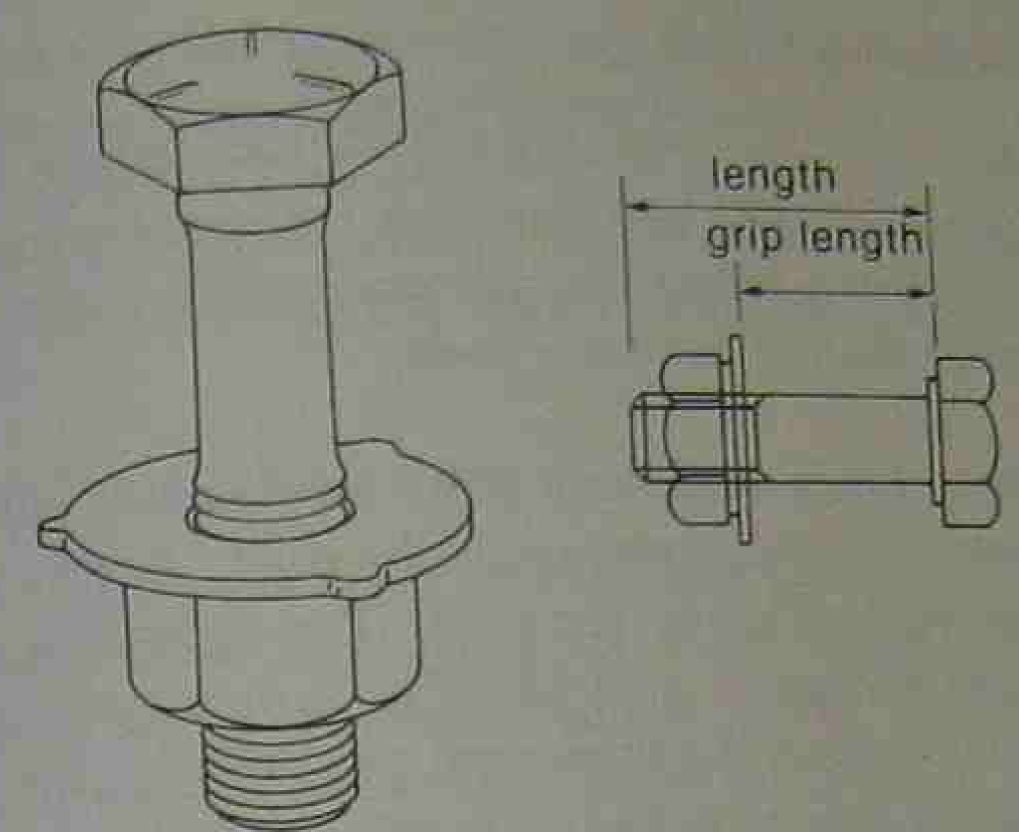


Fig. 5.5 HSFG bolt showing head markings

trolled. There are two methods: the part-turn method and the torque control method.

Setting instructions

Part-turn method (Part 1 bolts only)

1. Tighten the bolt using a podger spanner in the normal way to draw the joint surfaces into close contact.
2. Record the relative positions of the nut and protruding bolt end with a paint line or a cut from a cold chisel.
3. Finally tighten the nut using an impact wrench so that it turns relative to the bolt by the amount specified – usually half or three quarters of a turn.

Torque control method (Part 1 or Part 2 bolts)

1. Using a calibrated tightening device, tighten the nut to the torque necessary to induce the minimum bolt tension set out in the manufacturer's data. This is the preferred method.

Note: Bolts should be tightened in a staggered pattern and, in large groups of bolts, work should start from the centre of the group and move outwards. The use of a split-ring washer under the bolt head can help to maintain the pre-load of tightening. A matching plain washer should always be used under the nut.

Load-indicating bolts (LIB)

Description This is a special HSFG bolt which gives a physical indication when the bolt has been tightened correctly. It has a square head with a triangular pad under each corner of the head and a recessed section between pads.

Applications Load-indicating bolts are used in all applications where normal HSFG bolts could be used. Access is once more required on both sides of the joint. The tightening of the bolt compresses the triangular pads (A) (Fig. 5.6) and reduces the gap at C. When this reaches the minimum set down by the manufacturer, the required bolt tension has been achieved.

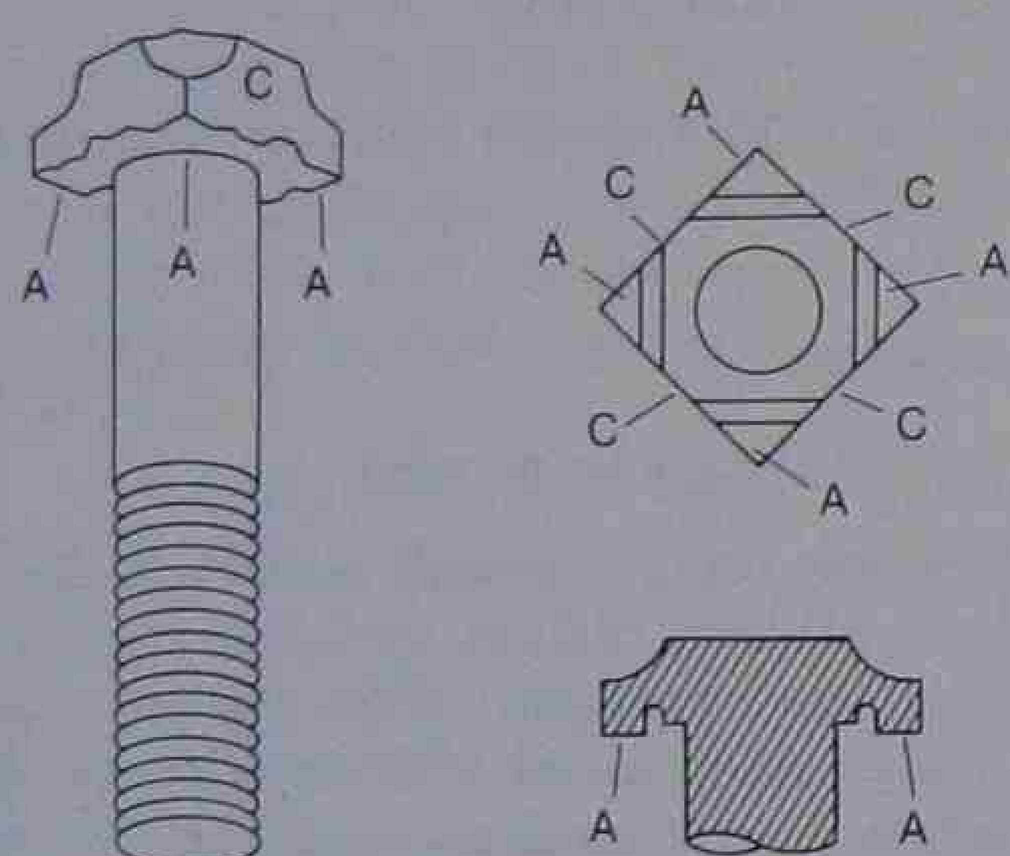


Fig. 5.6 Load indicating bolt

This is a more reliable indication of shank tension than the torque control method applied to normal HSFG bolts which tends to measure the friction between the mating surfaces of the thread, rather than shank tension.

Setting instructions

1. Tighten the bolt connection in the usual way.
2. With a feeler gauge, ascertain when the gap between the underside of the bolt head and the washer (or steel component) at C has been reduced to 1 mm (or the dimension laid down by the manufacturer). The bolt has then achieved the necessary bolt tension for the friction between the plies of the joint to have been achieved.

Note: Manufacturers usually recommend the size of no-go gauge should be 0.75 mm thick.

Load-indicating washers, which operate on similar principles, are also available for use with conventional HSFG bolts.

Waisted high-strength bolts

Description This special HSFG bolt is usually made of high-duty alloy steel, and its shank diameter is reduced (or 'waisted') below the root diameter of the thread. This allows an increased tension to be achieved in the bolt due to its high degree of elastic tension. Failure, therefore, occurs in the shank and not in the thread as is the case of standard bolts.

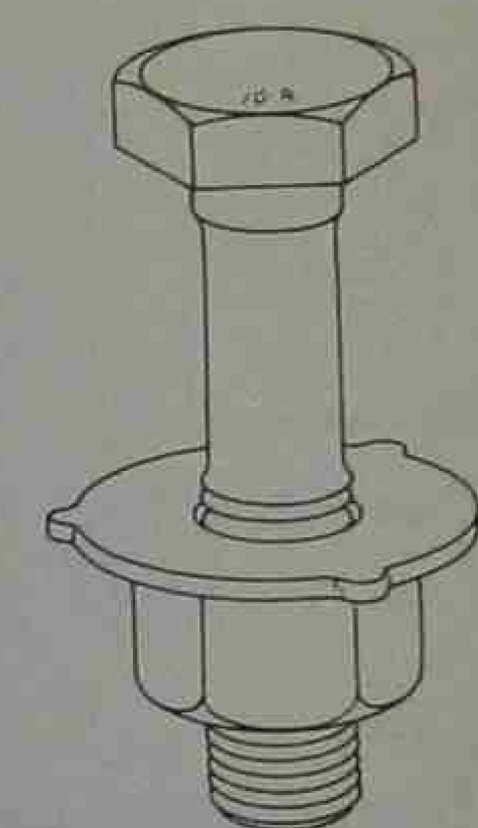


Fig. 5.7 Waisted high-strength bolt

Applications Waisted high-strength bolts are used in positions of severe fatigue or where bolts are subjected to tension as well as shear. Access is required to both sides of the joint to make connection.

Accessories

There is a number of types of locking nut available to withstand the effects of vibration on bolted connections (Fig. 5.8). These include slotted and 'castle' nuts, for use with a split pin or similar device, and self-locking turret nuts which include a turret section above the hexagonal nut which is cut twice and deformed to depitch the thread above the slot.

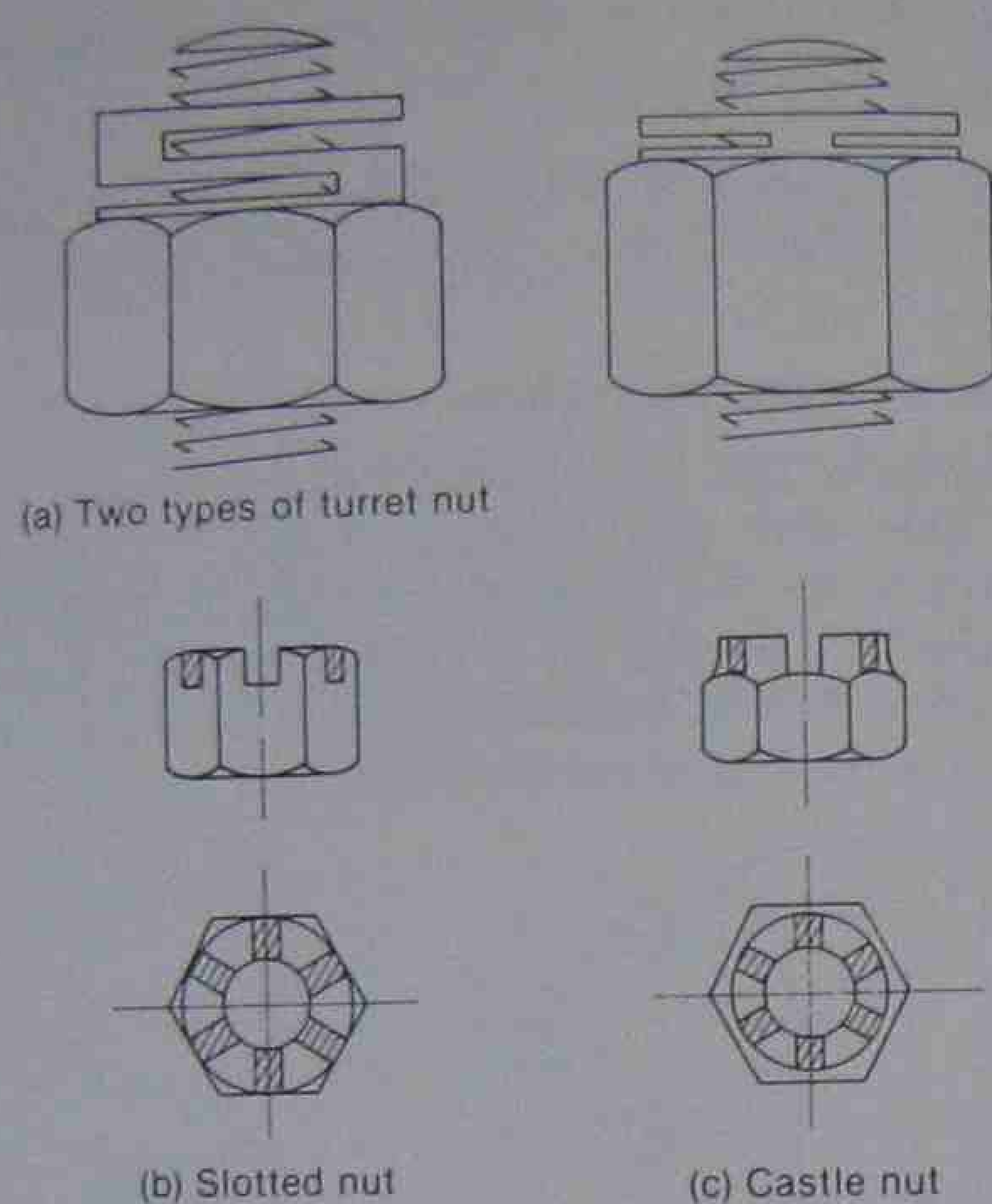


Fig. 5.8 Locking nuts (a) Two types of turret nut (b) Slotted nut (c) Castle nut

A2 Machine screws

Machine screws form an uneasy bridge between heavy-duty and light-duty metal base fixings. The difference between machine screws and bolts has been explained on page 112. Generally, machine screws are used to make medium-duty fixings and never move into the upper level of heavy-duty structural fixings, as do bolts. British Standard 4183: 1967 gives a preferred range of machine screw sizes from M2 to M12 diameters with lengths from 5 to 90 mm, but the majority of machine screws in common use are in the lower, light-duty end of the range.

Description Machine screws are made in a variety of metals, both ferrous and non-ferrous, and in a number of finishes. Head styles can be pan, countersunk, raised countersunk or cheese head (Fig. 5.9) with slot or recess driving profiles. There is also a range of hexagonal socket-headed screws, including cup, button and countersunk head styles (Fig. 5.10) as well as (at the larger

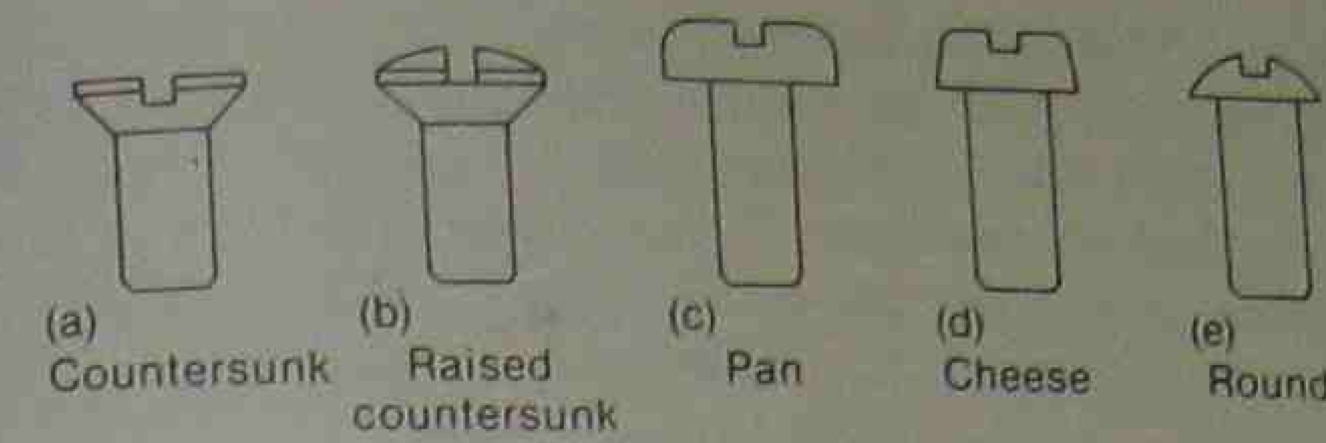


Fig. 5.9 Machine screw head styles (a) countersunk (b) Raised countersunk (c) Pan (d) Cheese (e) Round

end of the range) the hexagonal bolt head styles.

Machine screw points can vary from the normal flat end, to special lead-in ends like the cone point or dog point.
Matching nuts and washers are supplied with machine screws.

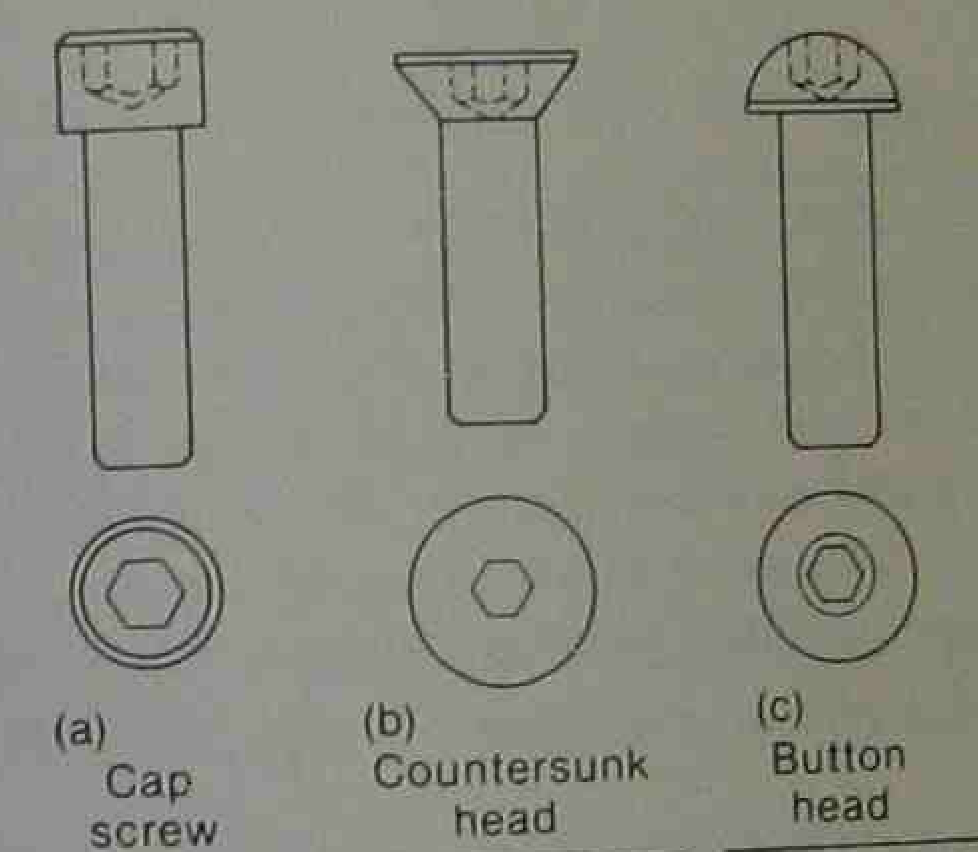


Fig. 5.10 Machine screw socket head styles (a) Cap screw (b) Countersunk head (c) Button head

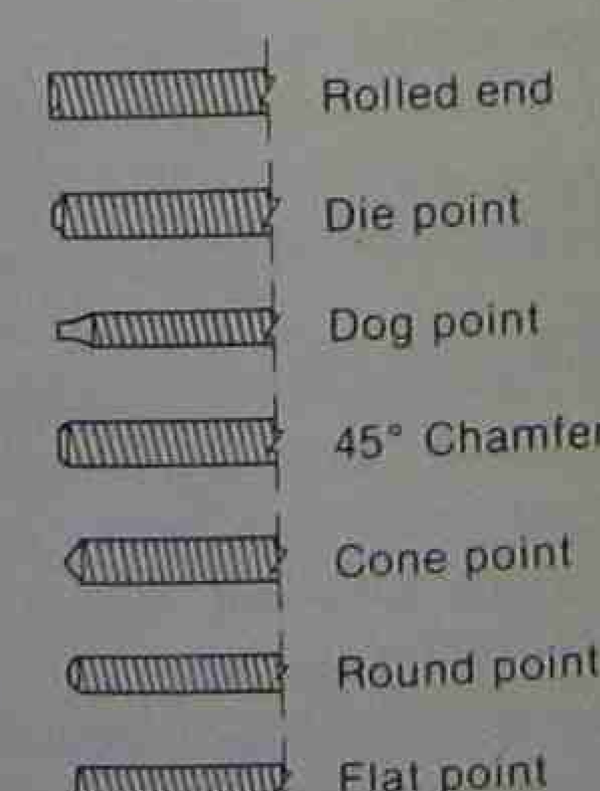


Fig. 5.11 Machine screw points

Applications Machine screws are used extensively to connect relatively thin materials to metal bases, when access is available to both sides of the joint. Locking nuts in a variety of forms are available to counteract the possibility of loosening, caused by vibration. There are also special nylon-lined nuts and screws with nylon inserts, which achieve the same locking effect.

Note: Although machine screws are generally used in conjunction with a nut, they can sometimes be used as blind fixings into a threaded hole in the metal base. Where a threaded hole is of a poor quality, or a higher standard of mating between the thread and the tapping is required, a thread insert may be used in the tapped hole. This reduces the size of the machine screw, but can accommodate variations in pitch and diameter.

A3 Set screws

These are in effect headless machine screws and are used for the same general applications as headed machine screws. Usually the smallest diameters of set screw are more common, although the range of available diameters extends from M3 to M24. They can have slotted or hexagonal socket-driving profiles and square, cone or dog points. Set screws tend more frequently to make blind connections into tapped drillings in the metal base, than to be used with a matching nut.

Section B: Light fixings

This group of fixings covers devices which make a connection between a thin component (often a sheet of metal or other material, e.g. industrial claddings) and a heavier metal base. It also includes devices which make a connection between two thin metal elements.

In spite of the name of the group, the engineering performance of many of these devices should not be underrated. This is particularly true of some of the extensive range of devices used to attach cladding to steel framework. These have not only to resist considerable uplift forces caused by wind pressure, they often, too, are required to fix the cladding with sufficient firmness to enlist the strength of the cladding to brace the structural frame.

Most fixings in this group are base-deforming type 2(b) devices – thread-forming, thread-cutting, self-drilling screws or powder- or pneumatic-actuated fixings. There is also a small group of type 2(a) drilled-for fixings which are associated with making single-sided, blind connections between a thin metal base and a thin component. These include light rivets, rivet bushes and self-locking tapped holes.

B1 Base-deforming devices (type 2(b))

Fixing devices in this category include: self-tapping screws; self-drilling screws; powder-actuated fasteners; pneumatic-actuated fasteners.

Self-tapping screws

These screws all require a pilot hole of the correct diameter to be drilled in the base into which the screw is driven. During driving, the screw produces its own mating thread in the base material. Clearly, self-tapping screws are only effective if there is sufficient thickness of base material in which to form a thread long enough to give a firm fixing (Fig. 5.12).

Description Self-tapping screws are usually made of hardened or stainless steel and fall into two categories: thread-forming and thread-cutting screws. The latter type are not greatly

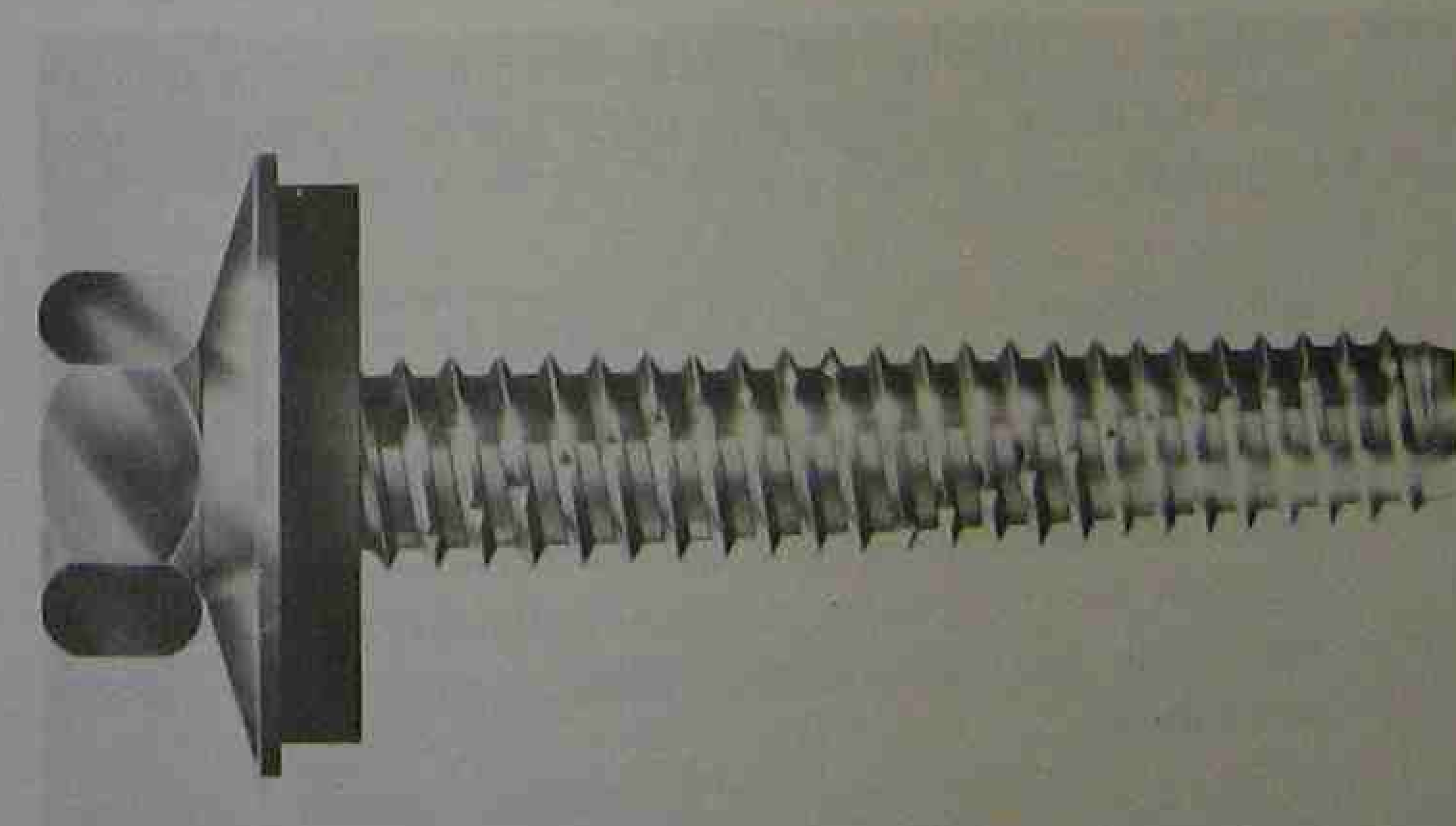


Fig. 5.12 Buildex ST and self tapping fasteners for roofing and cladding fixings

used in building. They form their mating thread by cutting and there are various point profiles, usually slotted or chipped, to aid the cutting action and to avoid the resulting metal swarf from clogging the screw.

Thread-forming screws, on the other hand, form their mating thread by deforming the base and not by removing any of its material. The two most commonly-used types of thread-forming screws are the type AB, with a widely-spaced thread and a gimlet point, and the type B (formerly Z) with a similar thread and a slightly tapered point. Head styles include countersunk, raised countersunk, mushroom, flange, pan and hexagonal; driving profiles are slotted or recessed. Sizes range from self-tapping screw gauge no. 2 (2.18 mm) to 14 (6.15 mm) and lengths from 5 to 50 mm.

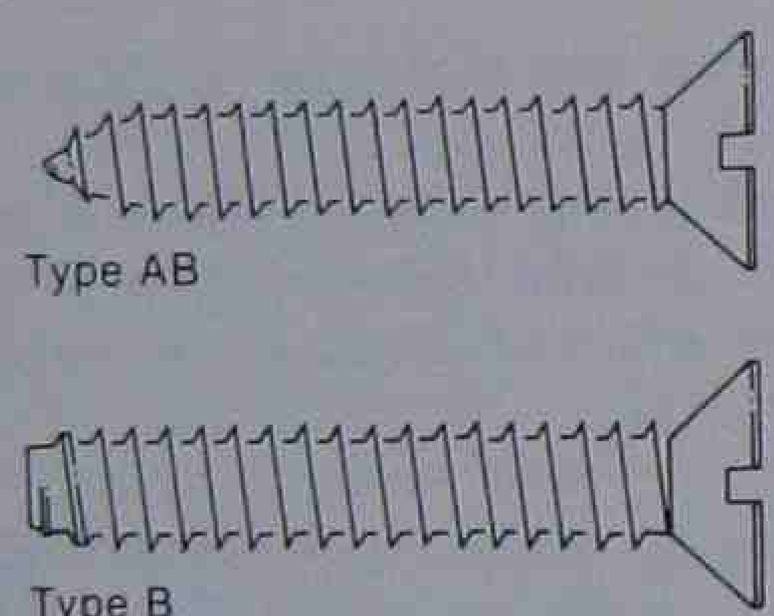


Fig. 5.13 Self-tapping screws, types AB and B

Applications The type AB thread-forming screw is used in light sheet-metal bases; the type B in light and heavy sheet-metal and non-ferrous castings.

Setting instructions

1. Drill a pilot hole in the base material of the diameter recommended for the particular screw diameter. A clearance hole should be drilled in the subsidiary component.
2. Position the component and place the screw point in the pilot hole and drive home using a manual or electric screwdriver.

Note: A unique type of thread-forming screw is produced by GKN. Called the *Taptite*, this device, made of cold-forging steel, has a trilobular shank (in section rather like an equilateral

triangle with its corners rounded off). This, when driven into a circular pilot hole, forms its mating thread by the action of the short forward face of each lobe (Fig. 5.14). This gives a high degree of thread engagement, reduced friction during driving and high clamping action of the subsidiary component to the base. It is also claimed that these screws have a tensile strength greater than a conventional self-tapping screw or machine screw.

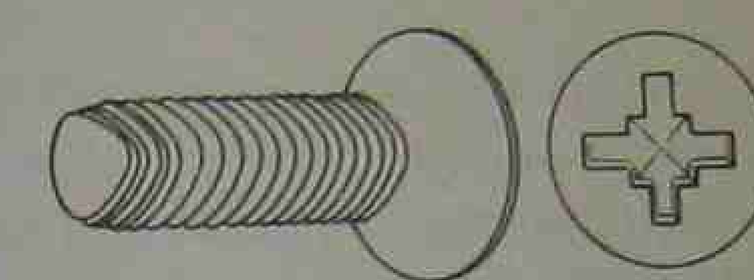


Fig. 5.14 Taptite self-tapping screws

The Taptite screw is available with various head styles and driving profiles, in diameters from M3 to M8 and lengths from 5 to 40 mm.

It can be used in thick steel sections, cored die-castings and thin sheet steel, provided that the pilot hole has an extruded rim. In the latter case a pilot hole is punched in the sheet steel so as to leave a protruding rim around the hole on one side of the sheet (Fig. 5.15). This clearly increases the 'thickness' of the sheet to receive the mating thread. Taptite screws can normally be used successfully with a drilled or clean (non-rimmed) punched hole when the thickness of the sheet is greater than three-quarters of the diameter of the fastener.

Another special type of thread-forming screw driven into a pilot hole in a metal base by a hammer, rather than a screwdriver, is the 'U' type of drive fastener. This is referred to in BS 4174:

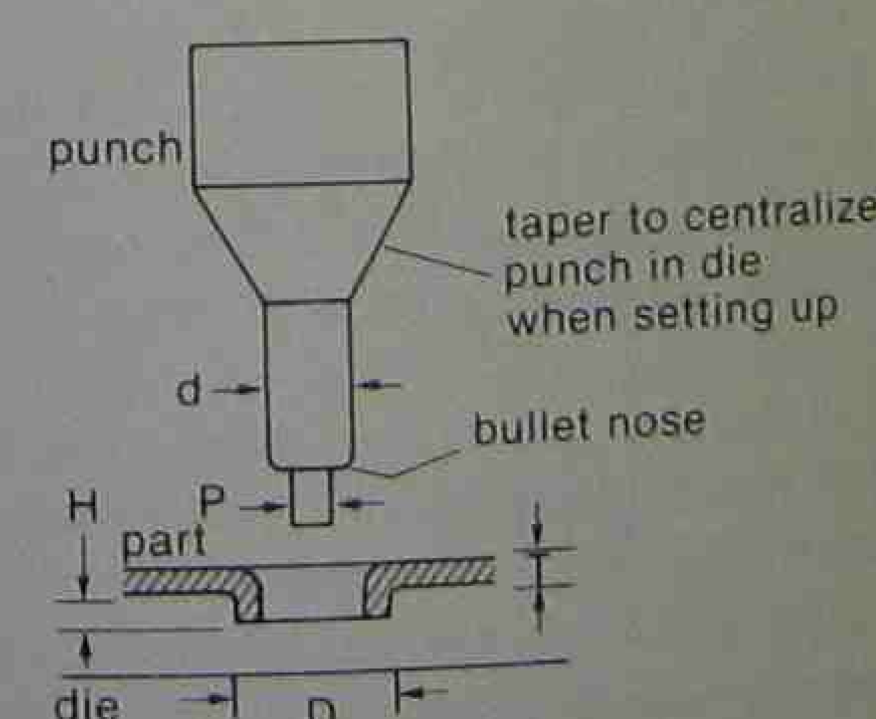


Fig. 5.15 Extruded hole to receive a Taptite screw

1972 as a 'metallic drive screw' and it usually has a round dome head. It is available in a range of diameters from self-tapping screw gauge no. 00 (1.5 mm) to 14 (6.15 mm) and lengths from 3 to 19 mm. This fastener, unlike the other self-tapping devices, is almost impossible to remove, once placed.

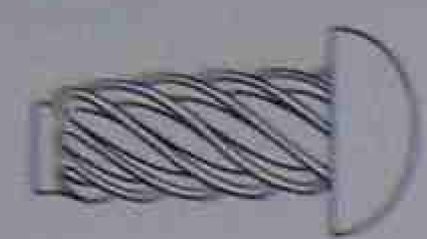


Fig. 5.16 U-type drive fastener

Self-drilling screws

Self-drilling screws are self-tapping screws which eliminate the need to drill a pilot hole. In fact three operations (drilling, tapping and placing) are included in one speedy fixing operation which results in high-performance connections which are particularly appropriate for firm cladding fixings. Self-drilling screws consist of a drill section whose length varies according to the thickness of steel base to which the fixing is to be attached. The drill section contains a drill flute, longer than the thickness of the steel to be drilled, and which provides top rake for the efficient cutting action and an exhaust passage for the drilling swarf (Fig. 5.17). Above the drilling section is the threaded part of the shank which taps its own mating groove in the base. Because the screw threads advance the fastener approximately ten times faster than the drill point can remove the metal, drilling must be completed before the

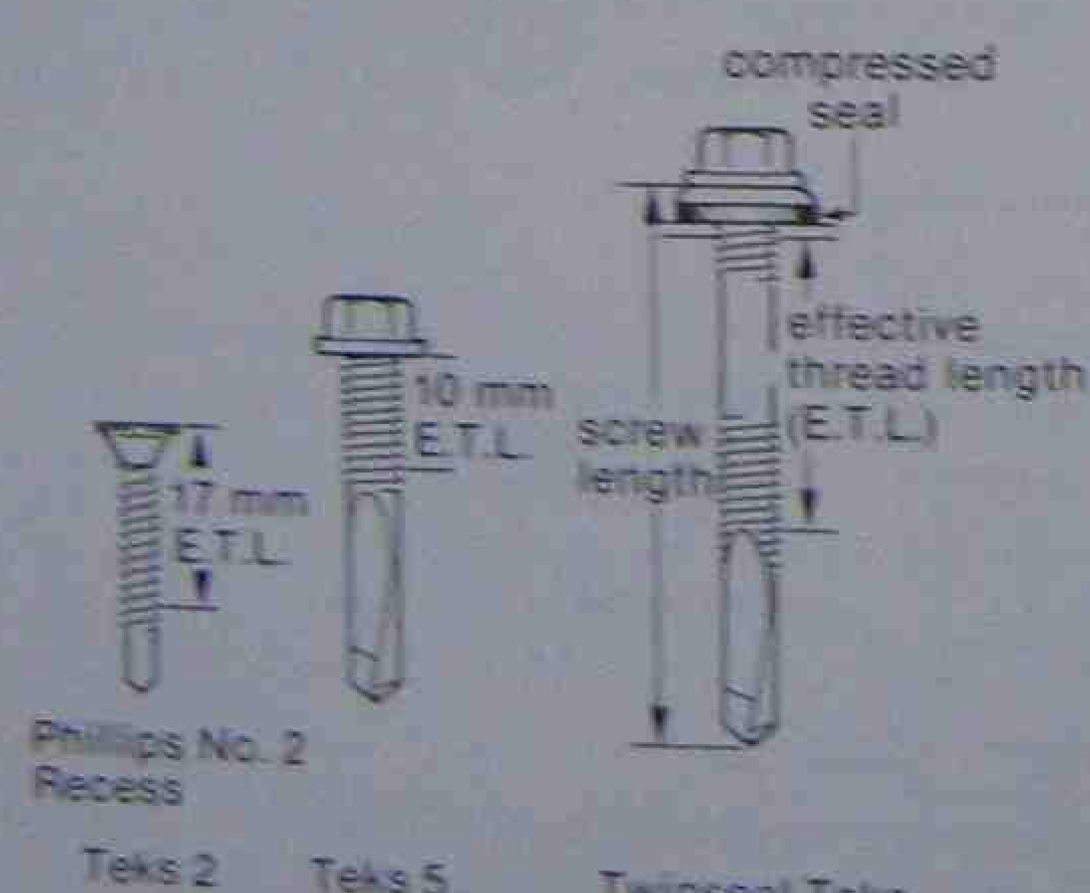


Fig. 5.17 Self-drilling screws (Twinseal, Teks 5 and Teks 2)

threaded section commences forming its mating thread. This makes the choice of the correct screw, which combines the necessary drilling and threading lengths, essential. Also the threaded portion of the shank has to penetrate the full depth of the drilling to realize the full holding power of the fixing.

Self-drilling screws are usually placed using an electric screwdriver.

Description Teks self-drilling screws are made of medium carbon cold forming steel, and have been heat treated. They are supplied in a variety of permutations of drilling and threaded section lengths to suit various thicknesses of steel base and component. They also have head/washer styles to match waterproofness need. The Sela range of British Screw and the Teks range from Buildex are the two most commonly-used families of self-drilling screws.

Twinseal cladding screws (Buildex) are available in two general ranges of self-tapping screw gauge no. 12 and lengths from 25 to 100 mm. There are some other gauges available for special purposes. The Twinseal screw has a metal/EPDM hexagonal head and sealing washer. Another Teks Screw – the Scots Teks – has a stainless steel head and is available in 25, 32 and 38 mm lengths. In addition there are similar ranges of deck-fixing self-drilling screws, including hexagonal heads or round heads with recessed driving profiles.

Applications Teks self-drilling screws are used primarily for fixing cladding or decking to steel framework in one operation, without prior preparation, and from the outside of the cladding or the decking. One range of Twinseal cladding screws is intended for fixing to cold-rolled purlins up to 3.5 mm thick; another to hot-rolled purlins from 5 to 12.5 mm thick. There is also a range of Standoff Teks intended to fix cladding and insulation in one operation without compressing the insulation (Fig. 5.18).

It is important to use the correct power screwdriver when fixing self-drilling screws. Buildex and British Screw recommend the Black and Decker HV25T because it has been designed to provide correct speed, power and a depth-sensing clutch to facilitate correct driving. A full range of accessories for use with this screwdriver and to suit both manufacturers' screws is available.

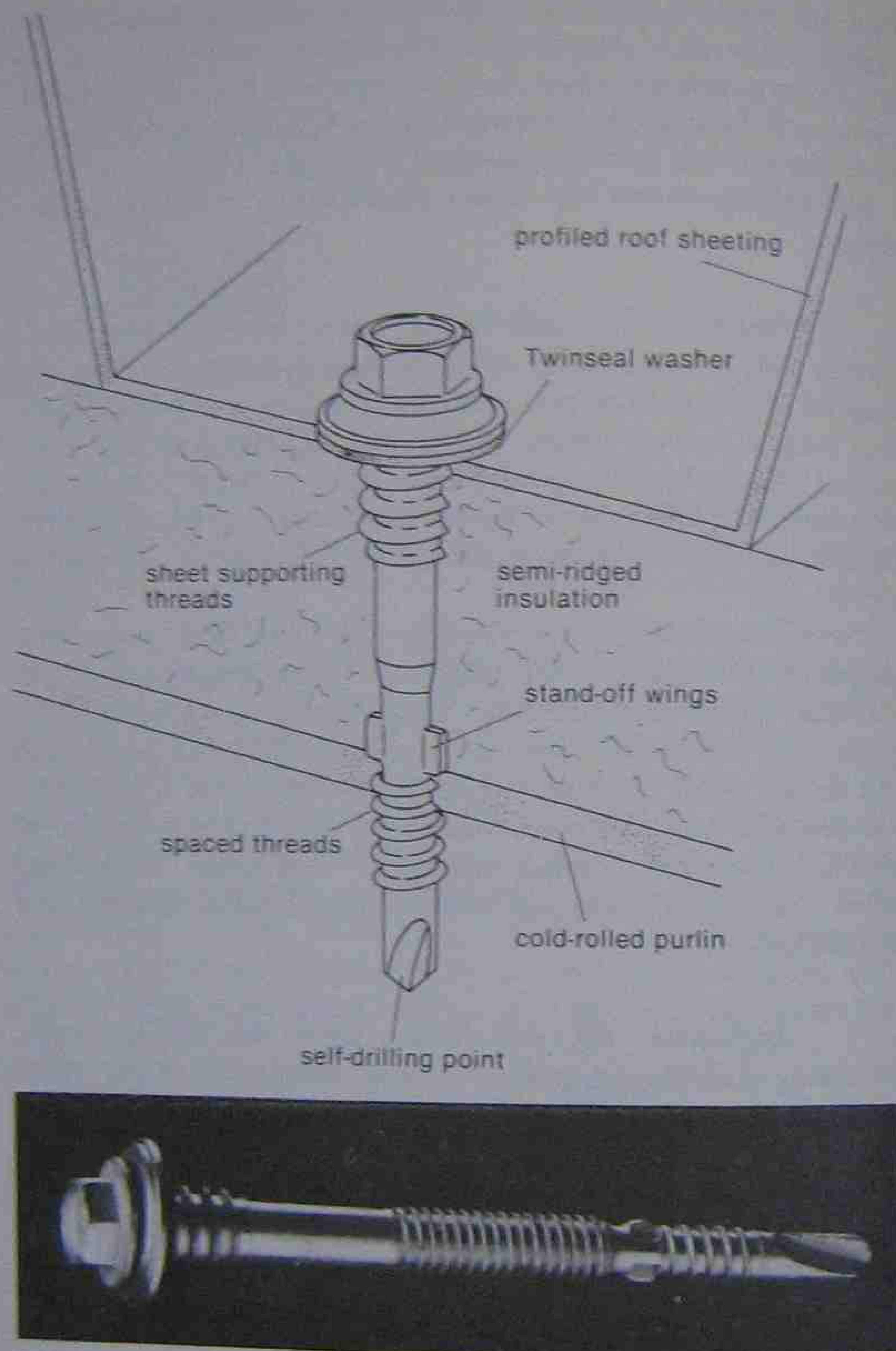


Fig. 5.18 The Standoff Teks

Setting instructions

1. It is particularly important when fixing cladding fasteners that consistent pressure on the sealing washer is main-

tained. In order to achieve this, always use an electric screwdriver with a depth-locating nose.
2. Adjust the depth-locating nose in accordance with the manufacturer's instructions for the particular device (Fig. 5.19).

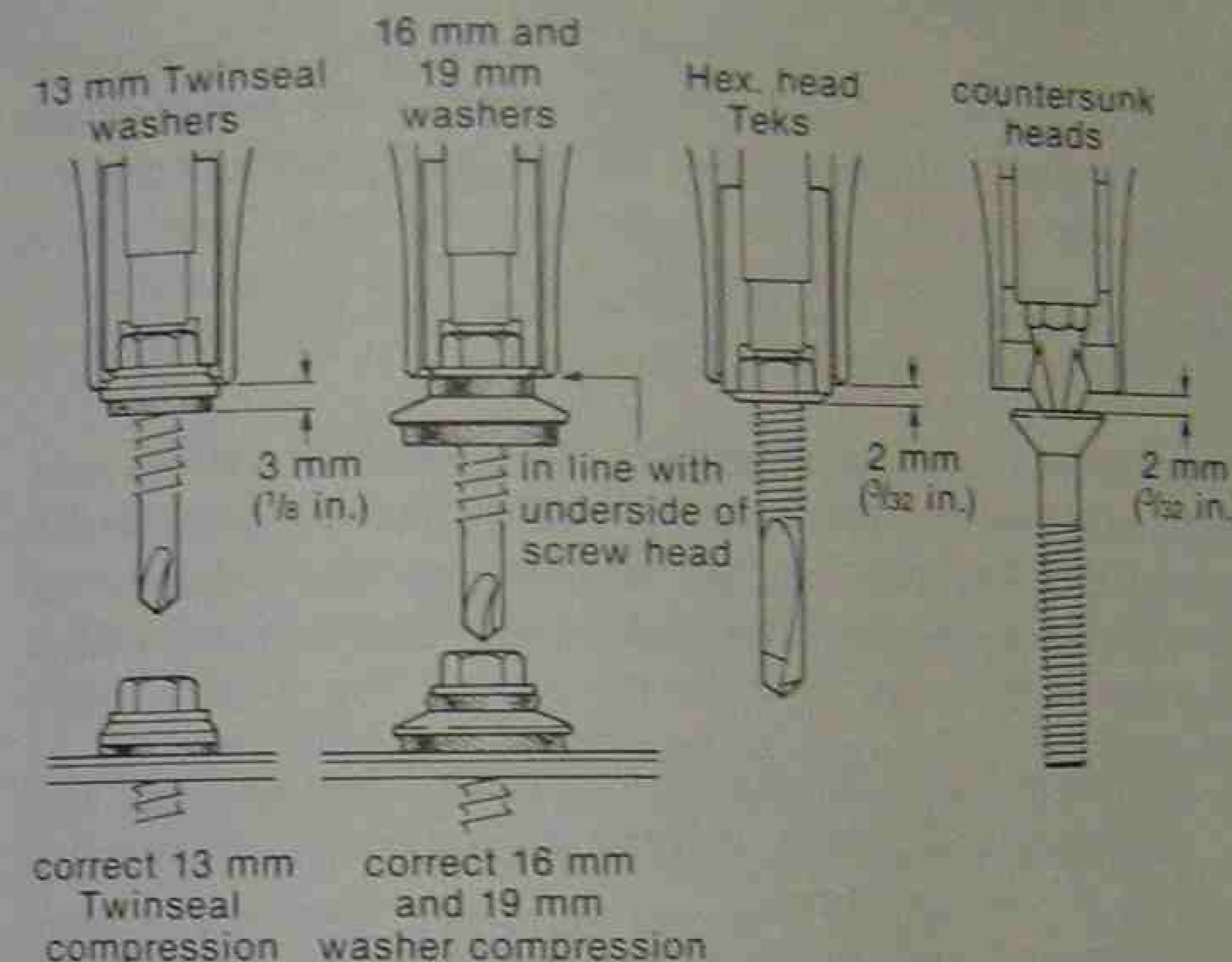


Fig. 5.19 Adjustment of depth-sensing nose on electric screwdriver

3. Holding the screwdriver perpendicular to the component, drill through the cladding and metal base until the screwdriver's nose makes contact with the component. The drive will then automatically disengage.
4. Check that the washer under the screw head is firmly bedding down. If not, make any necessary adjustment to the depth-locating nose.

Note: There is a number of special washers available for use with all self-drilling screws, either for weathersealing or spreading the fixing load over soft insulation boards (Fig. 5.20). There is also a number of snap-on colour caps for use with cladding fasteners.

Powder-actuated fasteners

These fixings require no prior preparation of the subsidiary com-

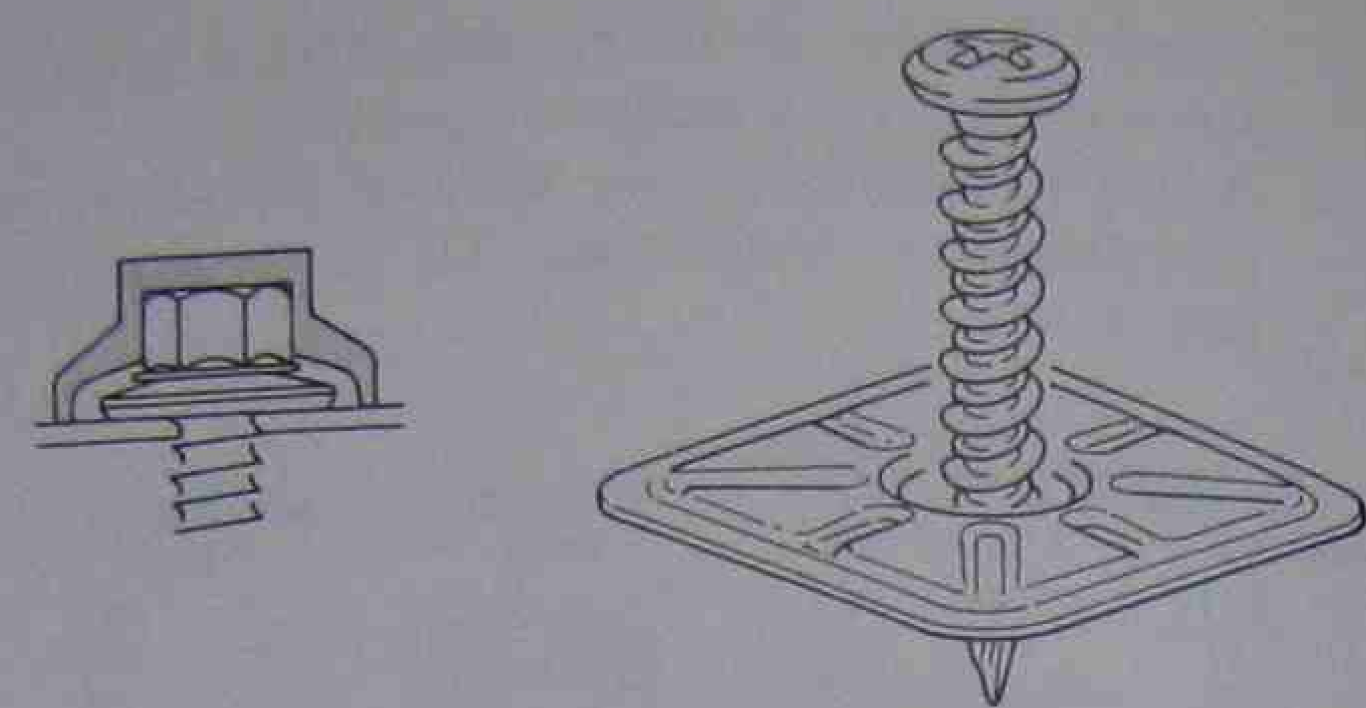


Fig. 5.20 Dec-Loc washer with Hilo self-drilling screw and British Screw head protective cover

ponent or the base, such as the drilling of pilot holes, as in the case of self-tapping screws. The fixing is merely driven through the component and into the base (or merely into the base) by an explosive charge from a cartridge in the fixing tool or gun. Fixings take two forms:

1. Galvanized steel drive pins for direct firing through the subsidiary component to create a permanent fixing to the base in one operation.
2. Galvanized steel threaded studs for firing into the base to give firm fixing points to which to make detachable fixings to the subsidiary component. There are accessories, such as ring or eye couplers, clamps and hooks which can be used with threaded studs.

Both types of fixing (Fig. 5.21) achieve their holding power by the plastic deformation of the steel, whose elasticity causes the shank of the pin to be gripped immediately after it is driven.

Typical subsidiary components which can be fired through to make a direct, one-operation fixing to a metal base include:

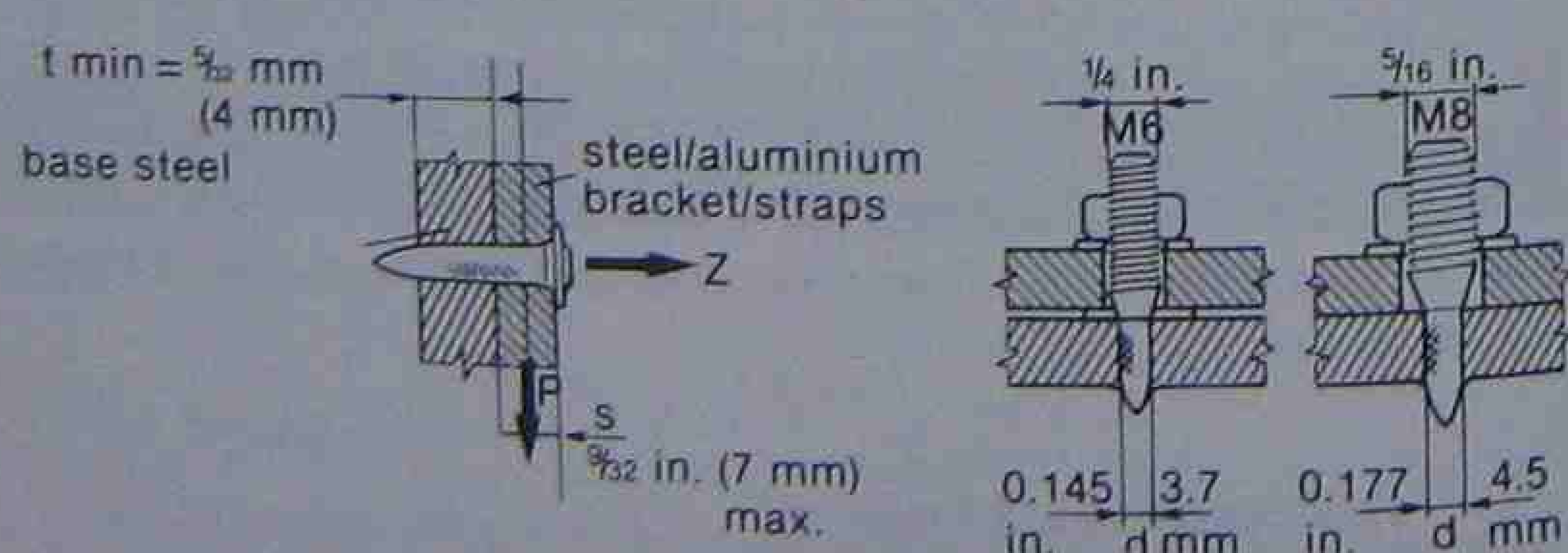


Fig. 5.21 Fired fixing types (a) drive pin (b) threaded studs

(a) wood (softwood or hardwood);
(b) metal sheeting or brackets;
(c) soft sheet materials (insulation board, wood wool, cork, etc.) usually employing an additional washer to avoid punching through.

Different sizes and shapes of drive nail are produced to match the character and thickness of the subsidiary component material.

Further information on the use of powder-actuated fixing in metal bases, their positioning and other matters affecting their performance are dealt with in the background section of this chapter on page 141.

Description Powder-actuated fixings are drive pins or nails of hardened (austempered) steel, fired from a tool by an explosive cartridge to make a permanent fixing into a metal base.

Pins are produced with various head and shank styles to suit the material with which they are used. Shank diameters are 3.7 or 4.5 mm and lengths are obtainable up to 65 mm. Plain-headed pins are intended to fix the subsidiary component direct to the base. Threaded stud-headed pins are available in stud diameters from M4 to M8. These are intended to provide a firm fixing to which to make removable connections for the subsidiary component. There are various colour-coded strengths of cartridge available to give the necessary force to drive the length of pin needed through component and base satisfactorily.

Applications Powder-actuated pins can make medium-duty fixings (limited to 15 kN pull-out strength) to metal bases over 4 mm thick, provided the metal in question is not brittle (e.g. cast iron) and has an ultimate tensile strength greater than 10 kgf/mm^2 (kilogram force per mm^2). The strength of the fixing will be dictated by the thickness and the strength of the metal base and the resistance of the subsidiary component to pull-through (in the case of direct fixings), or the strength of the nut connection to the threaded stud in the case of a removable fixing.

To obtain the most effective pull-out resistance, the point of the pin should pass right through the steel base, otherwise the compressive forces in the steel, acting on the point area, will tend to force the pin back out. When fixing to very thick steel bases, this proviso may not be valid, provided that the depth of penetration of the pin is sufficient to overcome the negative force on the embedment point. It is important that the diameter of the shank of the pin should not exceed the thickness of the steel base to obtain the best holding power.

If fixings are to be driven into metals other than structural steel, it is wise to carry out a suitability test on hardness, and if in doubt, consult the supplier of the metal.

The success of these fixings also depends on the type of fixing tool and the positioning of the pins and the thickness and type of subsidiary component.

Fixing tools

These have been described in some detail in Chapter 3, page 47, together with advice on their use. These pages should be referred to before attempting to make powder-actuated fixings.

It should be remembered that the Powder Actuated System Association (PASA) produces basic safety information and training data. Operating instructions should be followed meticulously and only fasteners and cartridges compatible with the tool should be used. Also only skilled operatives should be in charge of powder-actuated tools.

Positioning of pins

The minimum distance between the pin and the edge of the base should be two-and-a-half times the diameter of the shank. The minimum spacing of pins should be six times the diameter of the shank.

Thickness and type of subsidiary component

When the component is fixed direct to the mass walling, its thickness and type is critical to the success of the fixing. If the material is too thin, the pin may punch through (as in the case of very thin metal) or splinter (as in the case of wood). If the material is too thick, the fastener could bend, or not penetrate the base sufficiently, even using a pin with the longest shank.

Manufacturers' data sheets give full details of the thickness of subsidiary components which can be fixed using various lengths of fastener.

When soft materials are being fixed, such as insulation board, with the risk of pull-over failure, or firing through, additional washers may be required (Fig. 5.22).

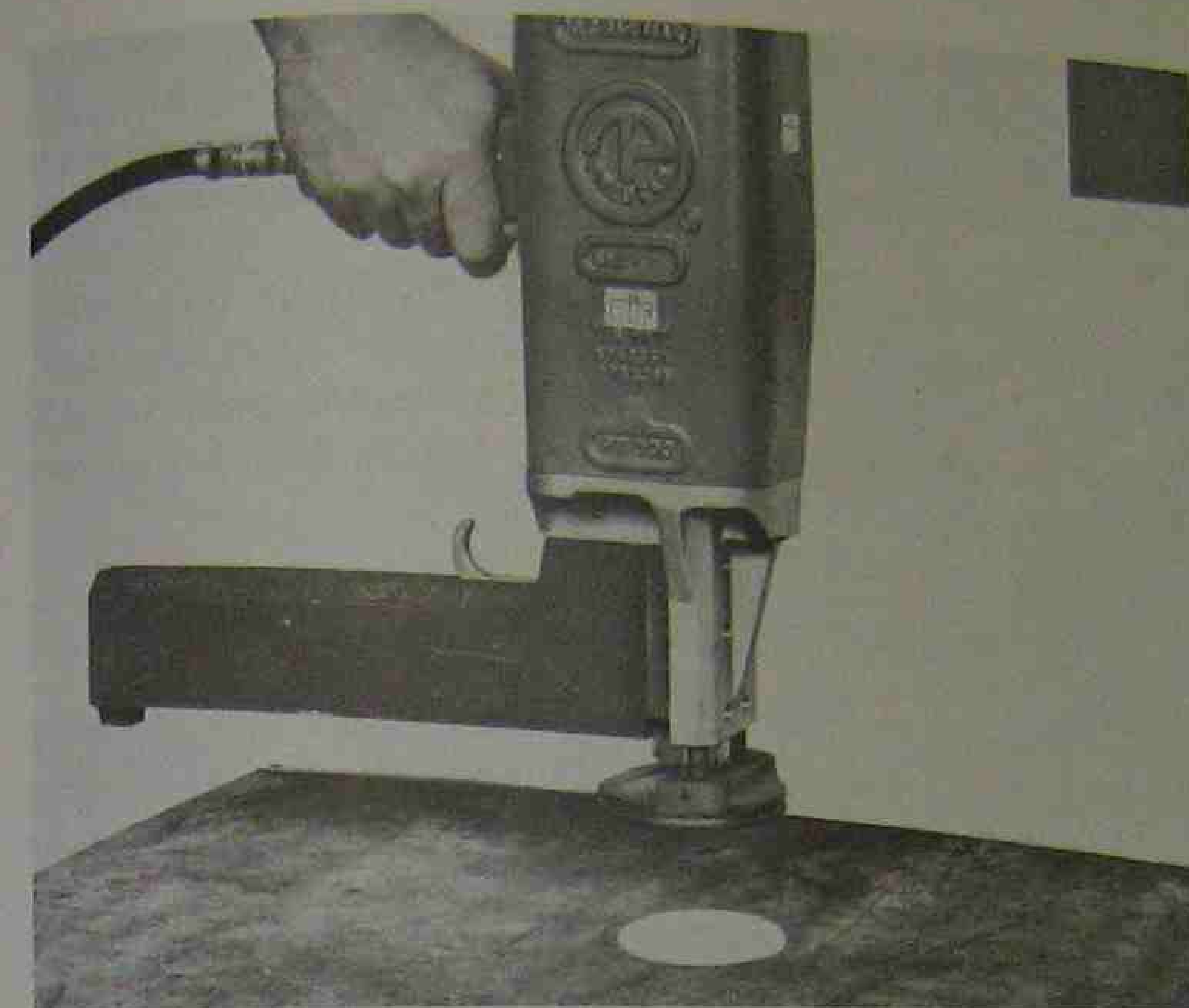


Fig. 5.22 A pneumatic-actuated fixing for insulation board

Setting instructions

Always follow the manufacturer's operating instructions minutely.

1. Select the length and type of pin required using the manufacturer's data, bearing in mind the type and thickness of subsidiary component.
2. To ensure the correct cartridge is used to give the right penetration depth, commence with a test-firing, using a low-strength cartridge. If the pin does not penetrate so that its point passes through the base (or penetrate the base sufficiently, if the base is thick), change to a higher strength of cartridge. The various strengths of cartridge are colour coded.
3. Place the tool firmly against the base (or, if firing through the subsidiary component, the component), ensure the tool is at right angles to the base, and fire.

Note: It is wise to wear safety glasses while making powder-actuated fixings and, in enclosed surroundings, ear protection too. Do not attempt to drive into areas of metal that have been welded or torch-cut – the material here may prove too hard. Do not fix, to a steel base, material which is thinner than the diameter of the shank of the pin. Do not use pins with a shank longer than is required.

Further safety advice is contained in the Health and Safety Executive Guidance Note PM 14, *Safety in the Use of Cartridge Operated Fixing Tools*, the Powder Actuated Systems Association's *Guide to Basic Training* and *Site Safety* by Jim Laney in Longman's Site Practice series.

Pneumatic-actuated fasteners

These fixings are identical to powder-actuated fasteners; the only difference is the method of driving the pin. All parts of the preceding section can, therefore, be considered as applicable to pneumatic-actuated fixings, with the exception of that part dealing with the firing tools (Fig. 5.23). Details of these are included in Chapter 3, page 50.

Sheeting fasteners

As a major application of the previous group of fixings is to connect cladding (roof and wall) to steel framework, it is appropriate to mention a group of fasteners here which fall into neither base-deforming nor drilled-for categories.

These fasteners were developed largely for the fixing of corrugated asbestos-cement sheeting and are all based on a simple hook bolt which hooks round one leg of the sheeting rails or purlins. The bolt is threaded on its non-hooked end and this end passes through a drilling in the sheeting to receive a washer and nut, which holds the sheeting in place. The waterproofing of the fixing is then completed by a plastic cap snapping ovet the nut (Fig. 5.24).

B2 Drilled-for fixings

These fasteners are mainly used to secure two thin sheets of metal where a self-tapping screw would not be able to make a sufficient length of mating thread in the base to obtain a firm anchorage, or where the use of a machine screw and nut is prohibited because

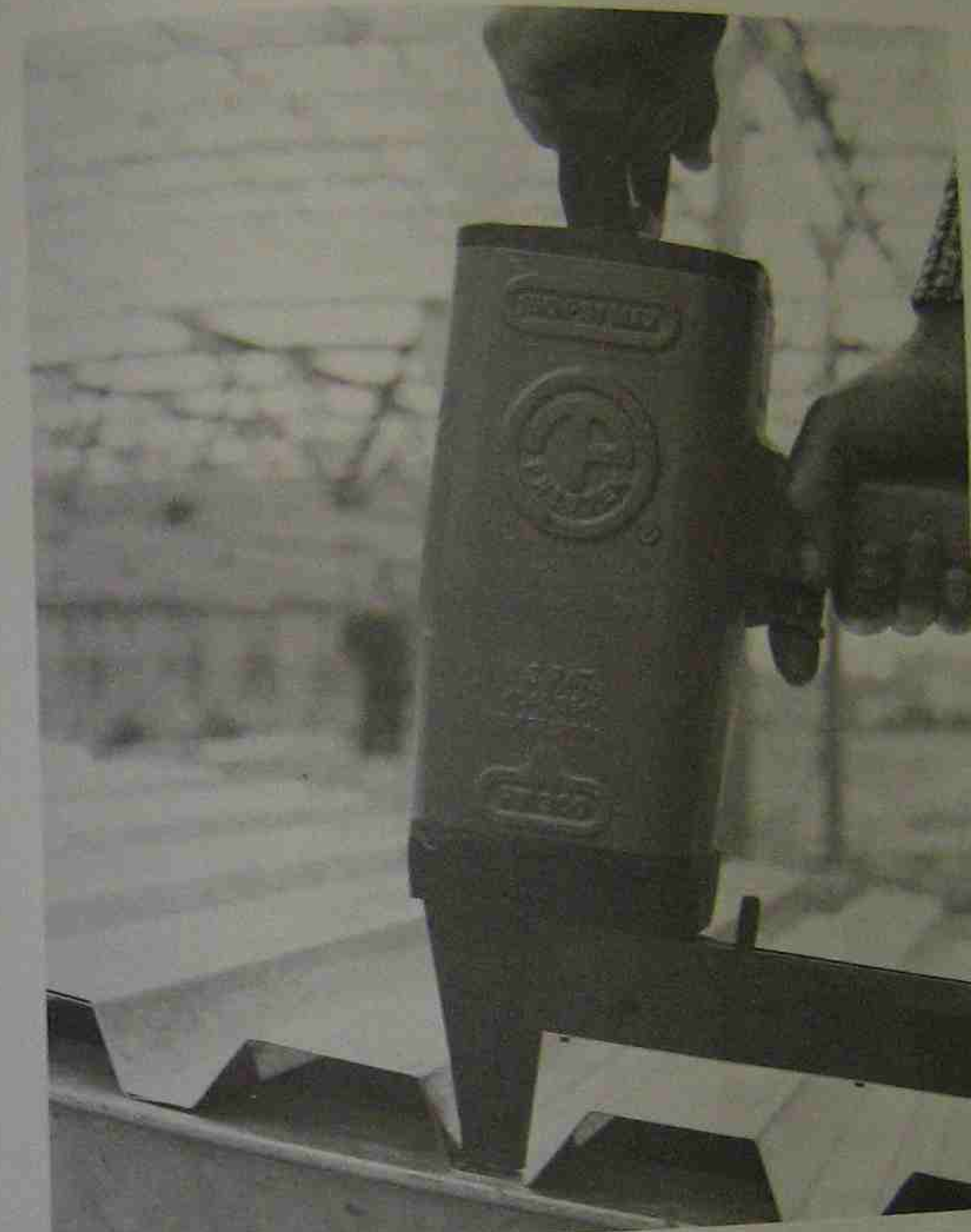


Fig. 5.23 BIF pneumatic fixing tool at work

the manipulation of the fasteners is restricted to one side of the base.

Many fasteners to light-gauge metal bases are of the rivet type, most of which are not greatly used on site in the building industry. Some other devices need sophisticated equipment, working from both sides of the base sheet, to prepare the base for a later fixing. An example of this is a rivet bush which is forced into a pre-drilled hole and is clinched there by a press (Fig. 5.25). It forms a reusable threaded bush in the metal base that can receive repeated machine screw fixings. Usually, though, these devices

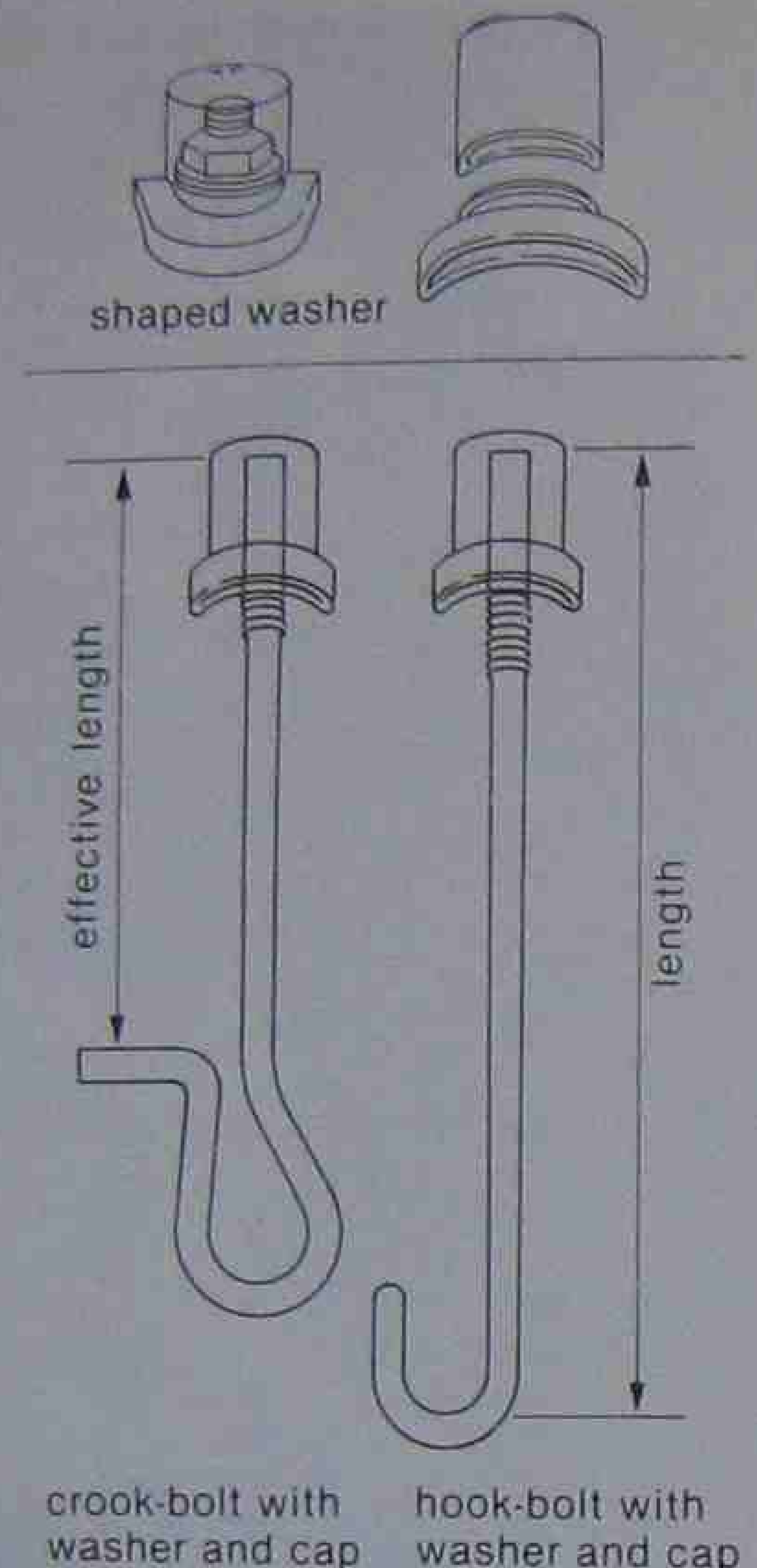


Fig. 5.24 Sheeting hook bolt and waterproofing cap

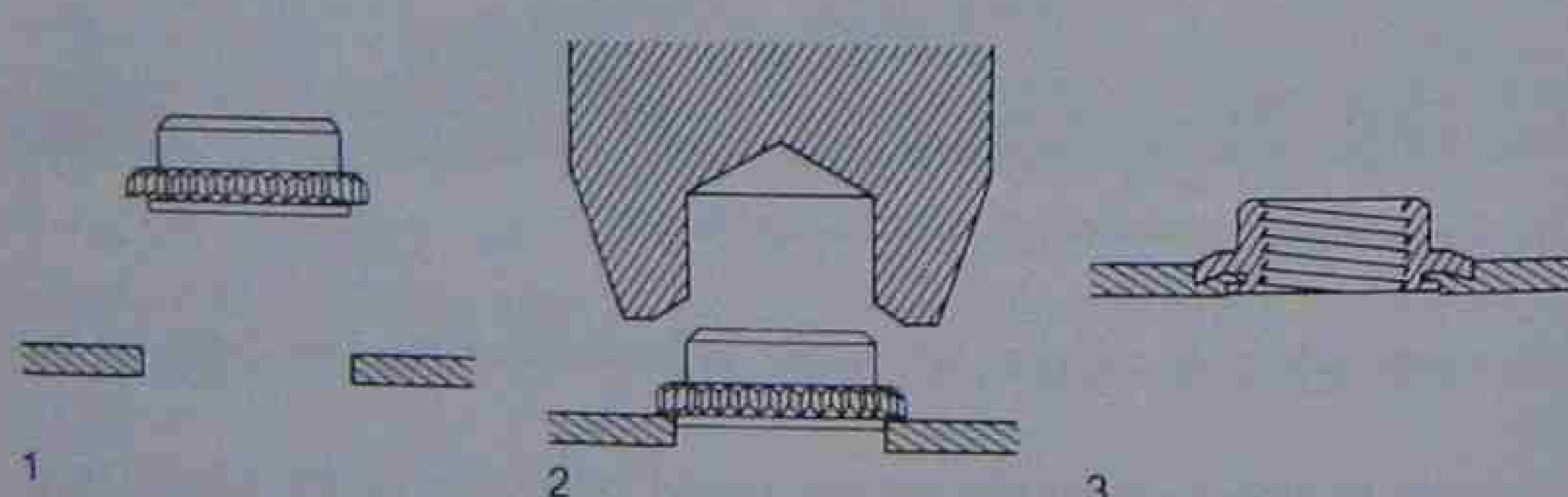


Fig. 5.25 Setting diagram for a rivet bush (ISC Press Nut)

are installed in the base sheet before it is delivered to site and therefore these are not applicable to this book.

Other devices which would be suitable for making connections to thin metal bases depend for their effect on the bunching of a flexible part of the device behind the base. These are in many

respects indistinguishable from some of the devices discussed in Chapter 6. In making a selection of this type of fixing, therefore, it is advisable to examine the relevant parts of Chapter 6 as well as this section. Only those fasteners specifically developed to make 'blind' connections to thin metal bases (tapped holes and blind rivets) are considered here.

Self-locking tapped holes

These devices have a similar purpose to that of rivet bushes in that they provide a permanent, reusable, threaded fixing point to receive a machine screw. Unlike rivet bushes, however, self-locking tapped holes can be placed from one side of the base and without using a press. They fall into two basic types: screw actuated and independently placed tapped holes.

Screw-actuated tapped holes

Description There are various proprietary forms of these fasteners. Generally they consist of a brass, aluminium or plastic internally threaded cylinder with a flange at one end. The cylinder is shaped externally and split to encourage the opening of the fixing at the rear of the base when the screw is inserted.

Applications These fasteners are used to provide tapped holes in thin metal bases from 0.5 to 3.0 mm thick to receive machine screws of about M5 diameter. Some plastic tapped holes are used with 3.10 mm woodscrews (Fig. 5.26).

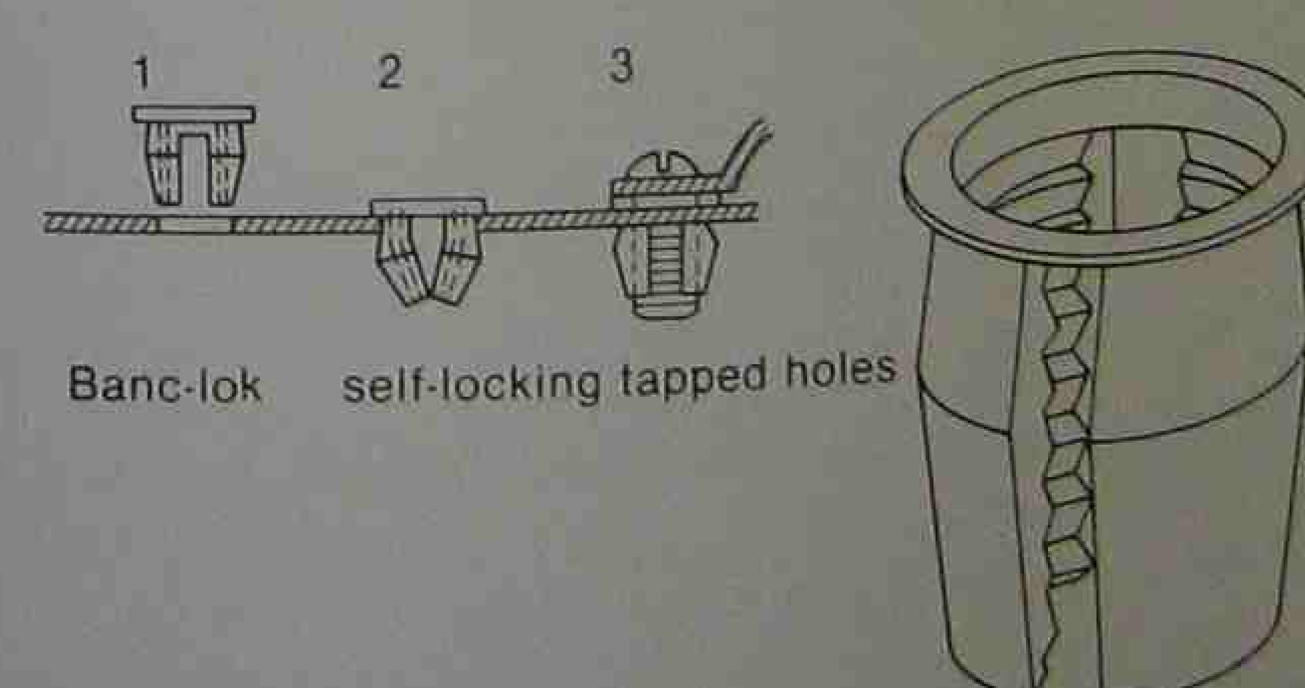


Fig. 5.26 Screw-actuated tapped hole

Setting instructions

1. In the base, drill a hole of the diameter specified by the fixing manufacturer.
2. Insert the fastener so that its flange rests on the surface of the base.
3. Pass the machine screw through the component and screw into the tapped hole until tight.

Independently placed tapped holes

Description These devices are usually made of alloy steel or stainless steel and consist of an internally threaded tube with a profiled flange on one end. This flange appears on the base as being a flat or countersunk head.

Applications These fasteners make a fixed, anti-rotational tapped hole in this bases which will receive machine screws from M2.5 to M16. They are usually placed using a special handtool (Fig. 5.27).

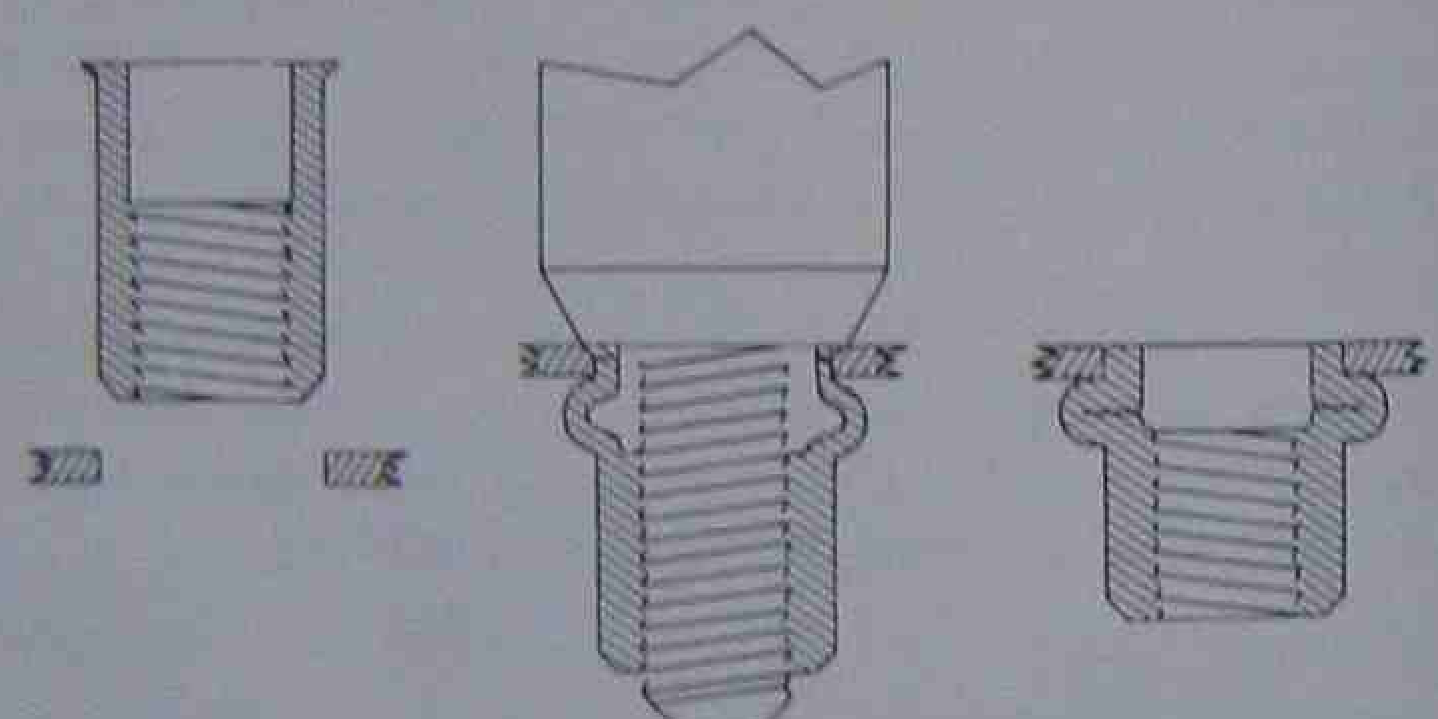


Fig. 5.27 Independently placed tapped hole (Clufix; ISC)

Setting instructions

1. In the base, drill a hole of the diameter specified by the fixing manufacturer.
2. Insert the fastener so that its flange rests on the surface of the base.
3. Using the handtool provided, drive the fastener into the base. This forces the profile under the flange into the base

material and avoids its rotation when the fastener receives the machine screw. It also bunches the fastener against the rear of the base.

4. Pass the machine screw through the component and screw into the fastener.

Note: A similar device sets a threaded stud in sheet metal, 0.5 mm and more thick, by buckling its shank at the rear of the base with a handtool (Fig. 5.28).

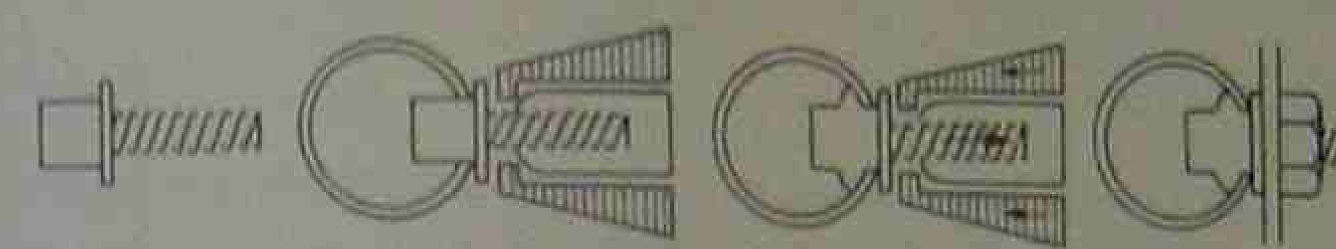


Fig. 5.28 Independently placed threaded stud (Rifbolt)

Blind rivets

Description All rivets depend on the enlargement of a part of the rivet on the further side of the base from the insertion side. There is a variety of lightweight rivets used in the manufacture of many factory-made products. Few of these are normally used on site. Those that are, are usually made of a variety of soft metals or plastics and either consist of a plastic insert which is hammer-set (Fig. 5.29) or a metal pop rivet which is bunched behind the base by the withdrawal of an integral steel pin using a special hand machine (Fig. 5.30).

Applications Most light rivets are used to connect thin sheets of metal or plastic of a combined thickness from 1 to 8.5 mm (hammer-set variety) or up to 12.5 mm (pop rivet variety).

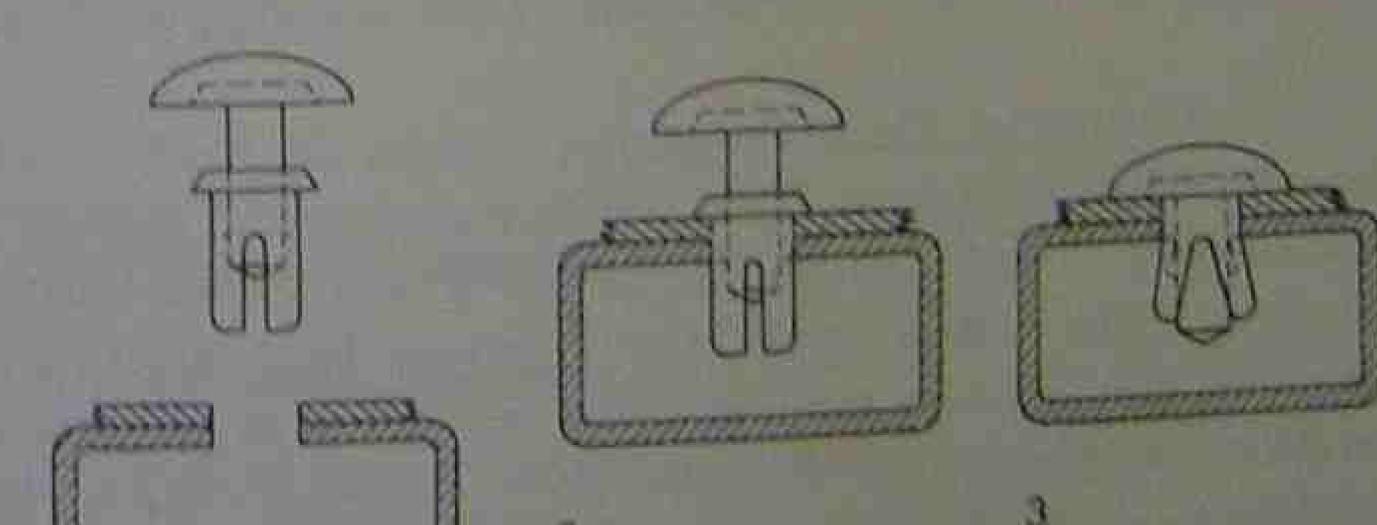


Fig. 5.29 Blind rivet (hammer-set variety) (Clic rivet)

Fig. 5.30 Blind rivet (pop rivet type)

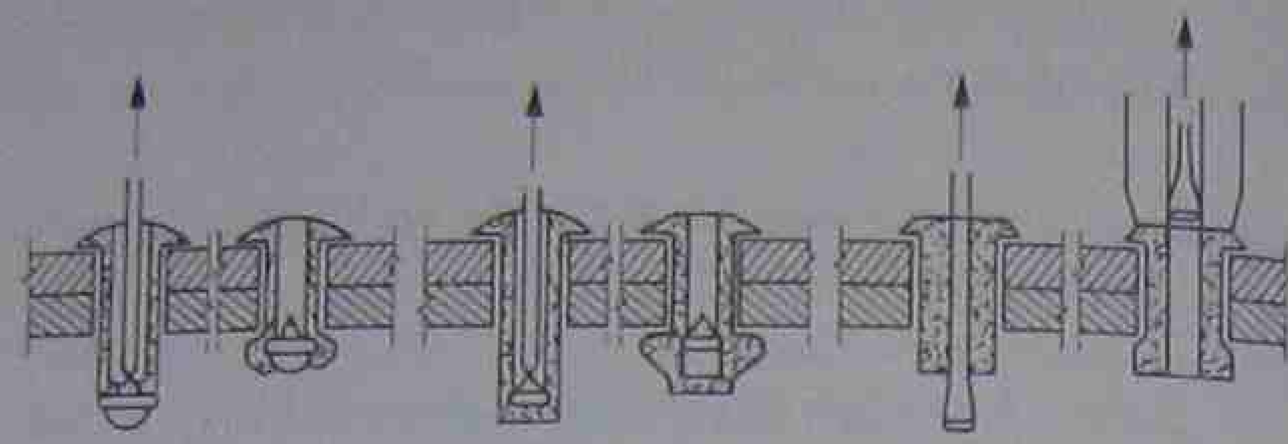


Fig. 5.30 Blind rivet (pop rivet type)

Setting instructions

1. Drill a hole of the diameter specified by the manufacturer through both component and base.
2. Align holes and insert the rivet until its shoulder lies on the surface of the component.
3. In the case of hammer-set rivets, drive the head of the plunger until it is flush with the surface of the component; in the case of pop rivets, draw the steel pin upwards, using the rivetting tool, until the pin breaks.

Note: Pop rivets come in a variety of types, including ones with a sealed end (which are air- and watertight) and the automatic feed rivet. This latter is set by the withdrawal of a setting mandrel which is retracted, unbroken, into the fixing machine, unlike the normal pop rivet in which the pin is integral with the fixing and breaks-off on setting.

Self-tapping inserts

There is one further variety of device which is similar to a self-locking tapped hole, but which is used to form a threaded socket in thicker soft-metal bases. It is called the self-tapping insert.

Description These cylindrical devices are made of brass or case-hardened mild steel. They are threaded on the outside with a self-tapping thread, and on the inside with a thread to receive a machine screw. The inside thread is also cut by a series of vertical grooves which receive the edges of a hexagonal driving piece used to drive the socket into the base.

Applications The mild steel version of this device is used

to create a tapped hole in a light alloy, the brass version in a plastic base. They vary in length from 9.6 to 15.7 mm and are intended to be used in bases which are thicker than these lengths. Self-tapping inserts are supplied in a range of sizes to receive machine screws from M4 to M10 (Fig. 5.31).

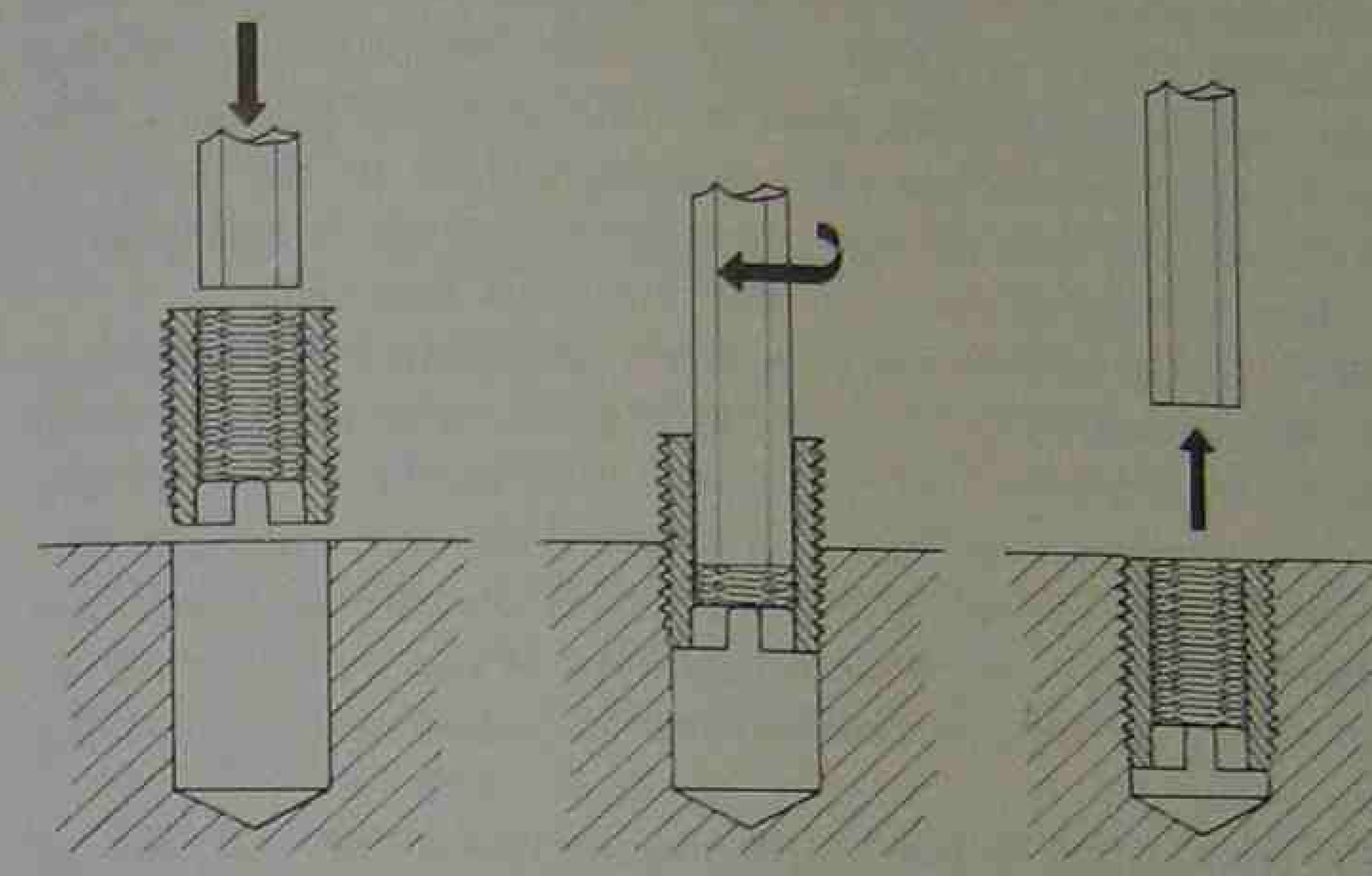


Fig. 5.31 Self-tapping insert (Kwik-sert; ISC)

Setting instructions

1. Drill a hole in the base of the specified diameter. The length of the drilling should be 1.5 mm longer than the insert.
2. Screw the self-tapping insert into the hole using a hexagonal wrench or an adaptor on a mechanical screw driver
3. When the top of the insert is flush with the base, stop screwing and remove the drivepiece.

Section C: Background

Bolted connections

Heavy-duty bolted connections, being of major structural importance, are always described by the engineer in detail on drawings and specifications. These instructions should be followed scrupulously. The design of bolted connections of this type is beyond the scope of this book. It might, however, be useful to learn a little of the background to these fixings.

Heavy metal-to-metal fixings derive from the peg, used in ancient timber construction. This later developed into the rivet in metalwork – a peg whose end was enlarged by beating on either side of the pieces of metal being joined. Nevertheless rivetted joints took a considerable time to form and did present a fire hazard, as a furnace was always needed close to the point of installation. The bolt, therefore, was developed which had most of the characteristics of the rivet, but few of its hazards.

The rivet has now almost completely disappeared from present-day construction, except for very light versions of the form (such as pop rivets, etc.), and even these tend to be used more frequently in equipment manufacture rather than on the construction site.

A bolted connection's strength depends on the size, type and number of the bolts, the thickness and type of steel (or other metal) from which the base and component are made and the spacing of the bolts.

The latter aspect is particularly important as the majority of steel-to-steel bolted connections put the bolts into single or double shear and their failure can result not only from the fracture of the shank of the bolts, but (more likely) from the deformation, splitting or eventual tearing through the base or component of the bolt. This is known as bearing failure.

As a result, the disposition of the bolts in relation to the thickness of the thinnest outer plate of the connection (for this purpose referred to as the component) is vital, and the assessment of minimum and maximum end and edge distances and centre-to-centre spacing of bolt becomes a balance between bolt diameter and thickness of the component. This is all laid down in the relevant British Standards and is too complicated to consider in this book. It is also unnecessary, as all bolted connections should be carefully designed by the engineer and thoroughly described, so that the work on site can be undertaken adequately.

The important factors to remember are:

(a) the bolt holes should be accurately drilled to the correct diameter;
(b) the bolt should fill the hole with practically no tolerance;
(c) bolt holes must not be misaligned;
(d) the nuts should be fully tightened – if high-strength bolts are used, all the relevant advice given earlier in this chapter should be minutely followed.

Note: The method of designating the strength grade of bolts is worth a brief note. Grade designations are laid down in two digit numbers. The first digit is one-tenth of the minimum-strength grade of the steel in kilogram force per mm^2 (kgf/mm^2). The second digit is arrived at by the formula:

$$\frac{1}{10} \times \frac{\text{yield stress}}{\text{minimum tensile stress}} \%$$

For instance, strength grade 4.6 (the normal grade for black bolts) indicates that the steel has a tensile strength of 40 kgf/mm^2 (first digit $= 1/10 \times 40 = 4$) and a yield stress of 24 kgf/mm^2 (second digit $= 1/10 \times 24/40 \times 100/1 = 6$).

Strength grades of nuts are designated by one digit which represents one-tenth of the proof stress in kgf/mm^2. This corresponds with the minimum tensile strength of the highest grade of bolt with which the nut can be used.

The use of the kgf/mm^2 unit is in accordance with continental practice, but it is expected that the correct SI unit (N/mm^2) will eventually be substituted (1 kg = 9.81 N).

Powder- or pneumatic-actuated fixings

There is a number of factors which affect the performance of fired fixings in metal bases. These have been referred to briefly and in a general way in the preceding section of this chapter. For those who require more detailed background information, this section has been included.

The following factors will influence the fastener's performance:

(a) strength and thickness of the metal base;
(b) the brittleness of the base;
(c) its yield point;
(d) the temperature at which the fixing is to be loaded;
(e) fastener spacing and positioning.

Strength and thickness of the metal base

Generally, fired fixings can be made to metal bases of 4 mm thickness or above, but the tensile strength of the metal has a great influence on the strength of the fixing and, generally, the lower the ultimate tensile strength of the metal, the greater the thickness of material needed to make a satisfactory fixing. Fired fixings should not be attempted to metals with an ultimate tensile strength less than 10 kgf/mm^2.

With regard to firing into steel, manufacturers produce data sheets which relate the grade of steel to the thickness to which a fastener can be driven. These establish application limits that should be observed. Particular attention should be paid to these when fastening to steels with an ultimate tensile strength higher than 37 kgf/mm^2.

If a fired fixing is attempted to steel less than 4 mm thick (or to greater thickness of other metals of a lower strength) holding power will be diminished and denting, or bulging of the base material, may result. Also unacceptable crushing or deformation of the steel may occur when side loads are applied. Using an over-powerful cartridge may also bring the risk of firing through when thin components and bases are being fixed.

The best pull-out loads are obtained when the point of the fastener just breaks the underside of the steel base. When the metal base has unknown strength and hardness characteristics, a suitability test based on trial fixings should be made.

Brittleness of the base

Fired fixings should not be attempted to brittle metals. Good fixings are difficult to obtain and accident hazard from nail breaking is considerably increased.

Yield point

Steel with a low yield point can be subject to deformation when fasteners are loaded laterally (in shear or in bending). In abnormal conditions, this can result in the loosening of the fastener.

Use temperature

There is a sudden and noticeable drop in the holding power of fired fixings in steel bases when the temperature exceeds 250 °C. Generally, use temperatures between minus 10 °C and + 250 °C. do not affect either the penetration of the fixing or its holding power. Below minus 10 °C the toughness/ductility of steel decreases and fired fixings should not be used.

Fastener spacing and positioning

The hole produced by a fired fixing will have little or no effect on

the structural performance of the base. Maximum effect will be in the order of less than 10 per cent reduction in strength. What is more, the use of fired fixings produces a better stress distribution in the steel than when a nut-and-bolt assembly is used. This is because the base suffers minimal loss of cross-section and, at the point where the fastener is driven, there is an increase in strength of the steel due to what is, in effect, the steel being worked cold by the action of driving the pin.

To obtain the best results and to avoid loosening already placed fasteners, the following spacing and positioning dimensions should be observed:

(a) distance between fastener and edge of steel base: 25 × shank diameter;
(b) distance centre-to-centre of fasteners: 6 × shank diameter.

Galvanic corrosion

When the fixing is made of steel and the base is also steel, clearly no question of galvanic corrosion exists. However, if the metal base is not steel, or the subsidiary component is of another metal, this danger should be checked out. Brief details of galvanic or bi-metal corrosion are contained in Chapter 2, page 9.

Table 6.1 Cavity fixings: which fixing to use

Description of base	*Expanding sleeve*	*Split sleeve*	*Plastic plug*	*Legged plug*	*Rivet*	*Gravity toggle*	*Spring toggle*	*Cord retention toggle*	*Plastic toggle*	*Umbrella fixing*
Plasterboard	√	√	√	√		√	√	√	√	√
Cellular partition	√	√	√	√		√	√	√	√	√
Lath and plaster						√	√	√	√	
Insulation board	√	√	√	√		√	√	√	√	√
Hardboard	√	√	√	√		√	√	√	√	√
Asbestos/silicate board	√	√	√	√		√	√	√	√	√
Plywood	√	√	√	√	√	√	√	√	√	√
Laminated materials	√	√	√	√		√	√	√	√	
Plastic sheet	√	√	√	√	√	√	√	√	√	√
Glass	√	√								
Sheet metal	√	√		√	√				√	
Hollow core doors					√					
Retrievable	√				√				√	
Unretrievable		√	√	√		√	√	√		
Permanent fixing even when unscrewed	√	√	√		√					√

against the rear surface of the lining material. Rawlnuts are available in diameters from 8 to 24 mm and with sleeve lengths from 11 to 27 mm. A neoprene sleeve version is also available for locations where oil, high ozone content or high temperatures could present a risk to a rubber sleeve.

Applications Rawlnuts make demountable fixings (i.e. the expanding sleeve can be left behind in the wall to receive further fixings, or the whole device can be removed and reused elsewhere) in all types of thin materials or cellular bases. They are also vibration resistant, electrically insulating, waterproof and corrosion resistant (Fig. 6.1).

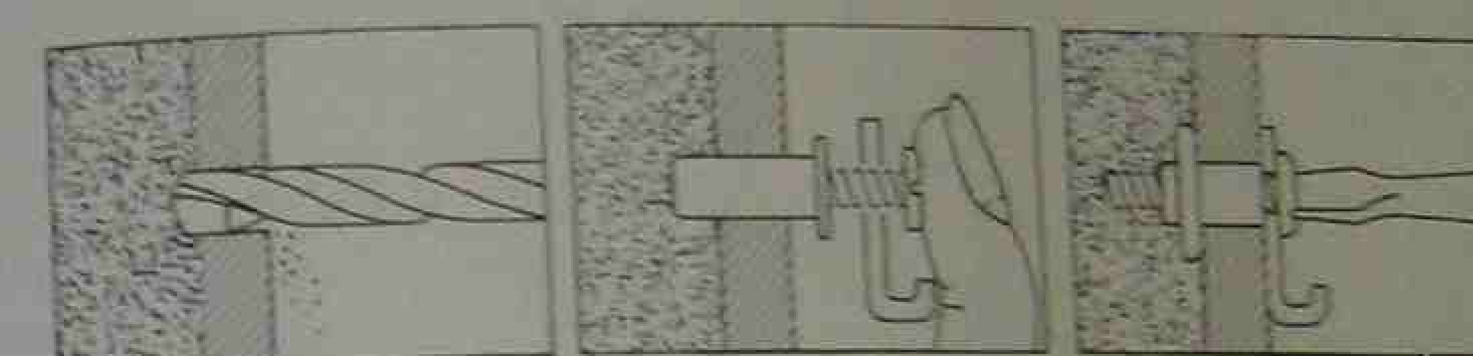


Fig. 6.1 Rawlnut setting diagrams

Setting instructions

1. Drill a hole of the diameter recommended by the manufacturer.
2. Push in the sleeve so that its flange rests against the surface of the base. Position the subsidiary component and pass the fixing screw through it and into the sleeve.
3. Tighten the screw.

Note: Rubber-sleeved Rawlnuts can be used with safety in areas where the temperature rises to 70 °C. They are not, however, oilproof. Neoprene-sleeved devices are unaffected by the air temperature up to 90 °C and are resistant to all types of oil up to temperatures of 110 °C.

A2 Split-sleeve cavity fixing (Thuscan Rosegrip)

Description This type of device, like the previous one, consists of a plastic sleeve with an integral nut cast into one end and a flange formed in the other. In this case, however, the sleeve has a series of longitudinal splits. A machine screw is inserted into the sleeve and, as it is screwed into the nut, the sleeve is drawn towards the surface material and the plastic strips are bunched behind the facing in the form of a rosette. These devices are available in a single screw size (5 mm diameter) and in 3 lengths, selected according to the thickness of the facing material – 3 to 8 mm; 8 to 16 mm and 16 to 26 mm.

Applications Split-sleeve devices make demountable fixings (i.e the sleeve remains in the wall and can receive further fixings, if required) in all types of thin material or cellular bases within the size limitations set out above (Fig. 6.2).

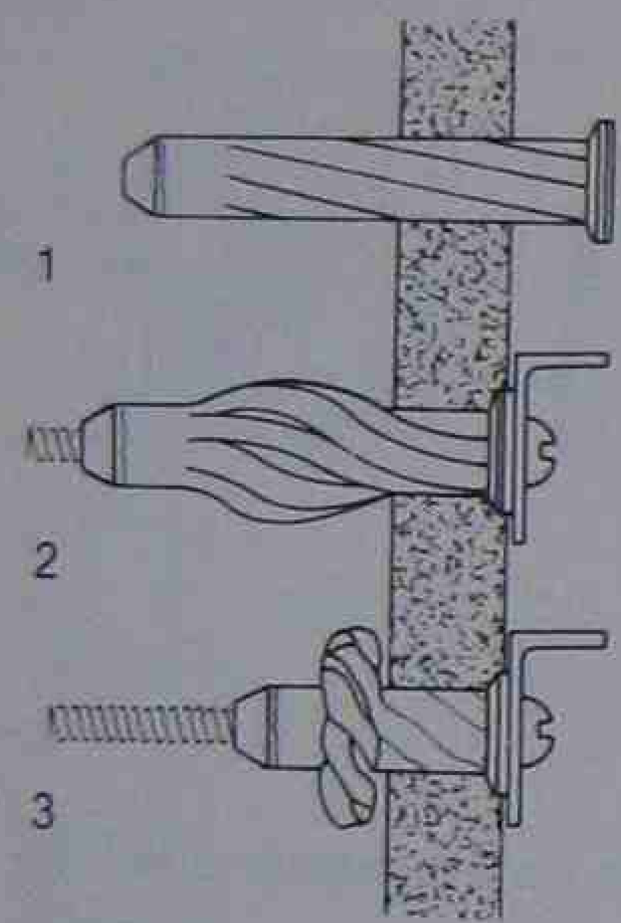


Fig. 6.2 Thuscan Rosegrip setting diagrams

Setting instructions

1. Drill a 10 mm diameter hole (or as recommended by the manufacturer). Push in the sleeve so that its flange rests against the surface of the base.
2. Position the subsidiary component and pass the fixing screw through it and into the sleeve.
3. Tighten the screw.

A3 Plastic plugs (Fischer NA Nylon rivet, Rawlanchor)

Description This is a nylon plug, internally threaded to accept a standard woodscrew. One end of the plug is flanged to avoid it falling into the cavity. Three sizes of woodscrew (3, 4 and

5 mm diameter) can be accommodated in a range of 7 plug sizes and 4 lengths. The length is selected according to the thickness of the facing sheet from 3 to 26 mm. As the woodscrew is driven into the plug, so the plug is drawn towards the facing material and folded against its rear surface.

Applications NA Nylon rivet anchors make demountable fixings (i.e. the expanded plug remains in the wall and can receive further fixings if required) in most types of thin material or cellular bases. The nylon is strong, durable and will not corrode (Fig. 6.3).

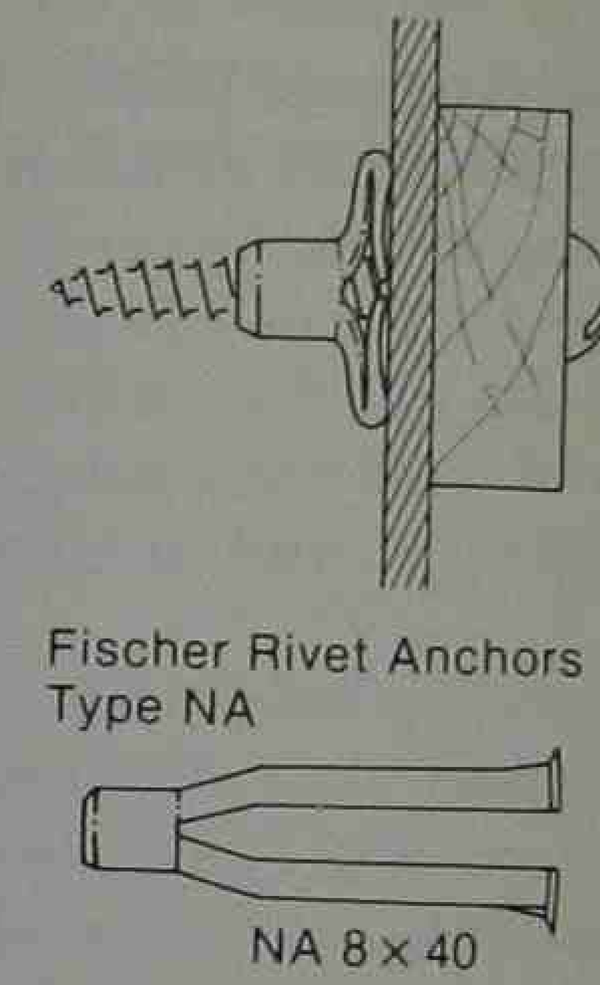


Fig. 6.3 NA Nylon rivet (Fischer)

Setting instructions

1. Drill a hole of the diameter recommended by the manufacturer for the particular size of anchor.
2. Push in the sleeve so that its flange rests against the surface of the base.
3. Position the subsidiary component and pass the woodscrew through it into the plug.
4. Tighten the screw. Do not overtighten.

Note: The anchors have a moulded thread which accepts the woodscrew. It is important that the correct size of screw is used and that it is not over-tightened. Also the plain (unthreaded) length of the shank should not exceed the combined thicknesses of the subsidiary component and the facing layer.

A4 Legged plugs (Fischer Type A hollow door fixing)

Description This nylon device is internally threaded to accept woodscrews and is available in 4 sizes for use in small cavities with screw sizes from 3 to 5 mm diameters. It has a series of outward-facing legs which are drawn back against the inside surface of the facing material when the screw is tightened.

Applications This type of anchor was developed specially for use in narrow cavities (between 15 and 20 mm wide) – the sort of size normally encountered in hollow-core flush doors. When the screw is unscrewed, the nylon body falls inside the cavity, making this device non-reusable. The thickness of the facing layer, however, is not critical to this fixing, provided that the length of screw will accommodate the thickness (Fig. 6.4).

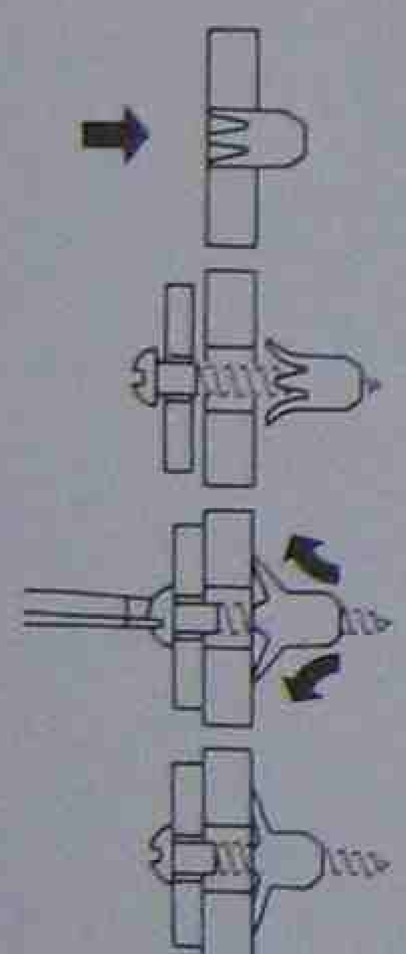


Fig. 6.4 Fischer Type A hollow door fixing

Setting instructions

1. Select a screw of the diameter appropriate to the nylon body and of a length equal to the thickness of the surface layer, plus the thickness of the subsidiary component, plus the length of the anchor.
2. Drill a hole of the recommended diameter.
3. Thread the screw through the subsidiary component and screw the anchor body on to the point of the screw until it is firmly gripped.

4. Position the subsidiary component, pushing the anchor through into the cavity.
5. Pull the screw outwards, drawing the anchor against the rear of the facing layer.
6. Tighten the screw.

Note: It is important that the recommended diameter of woodscrew is used.

Section B: Rivet cavity fixings

This group of fixings is designed specifically for connecting a thin component to a thin base, the combined thicknesses of the two being in some cases less than 4 mm. The action of rivet fixings is not dissimilar to that of the expanding cavity fixings above, the main difference being that the action of inserting a woodscrew or self-tapping screw into the device splays out the legs of the rivet in the cavity to form a wedge shape, which inhibits withdrawal.

Description Generally these devices are of nylon with a thin flange on their external end, the other end being split into a series of legs. These are splayed outwards by the insertion of a woodscrew or self-tapping screw (depending on the pattern of rivet). Sizes range from rivets to fit 4 to 8 mm diameter holes, and rivet lengths from 9.5 to 24.5 mm.

Applications These fixings are used to connect a thin (usually metal) subsidiary component to a thin base. Often both component and base are thin-gauge metal, in some cases the device can be used with thicker bases such as plywood or glass fibre sheet (Fig. 6.5).

Setting instructions

1. Drill a hole of the recommended size.
2. Insert the rivet so that its flange is set against the outer surface of the base.
3. Position the subsidiary component over the rivet and drive the screw into the device.

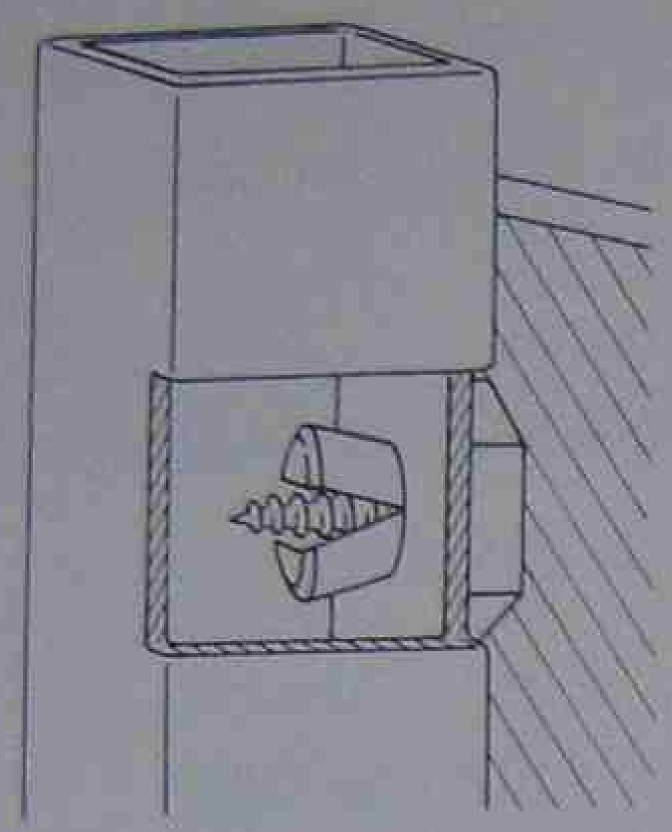


Fig. 6.5 Rivet cavity fixing

Section C: Toggle fixings

This group of mainly metal fixings includes those which open within the cavity automatically, either by the action of gravity or because of a spring in the device. They are suitable for most cellular materials provided the cavity is wide enough to allow the device to open. Some are not recommended for use with glass or metal bases. These fixings are generally not reusable, the toggle section remaining in the cavity if ever the assembly is dismantled.

C1 Gravity toggles

Description As the name suggests, these toggles open automatically as a result of the action of gravity. They consist of a plated steel toggle bar, pivoted off-centre on a swivel nut. This nut receives a machine screw, ranging in diameter from M3 to M6 and in lengths from 50 to 80 mm.

Applications Gravity toggles are applicable to most vertical cellular bases with soft facing materials like plasterboard, fibreboard, etc. provided that the width of the cavity will allow the bar to be inserted horizontally and then fall into a vertical position behind the facing sheet. The bar cannot be retrieved, if ever the fixing is disassembled, but will remain in the cavity (Fig. 6.6).

Setting instructions

1. Drill a hole in the base of sufficient diameter to take the

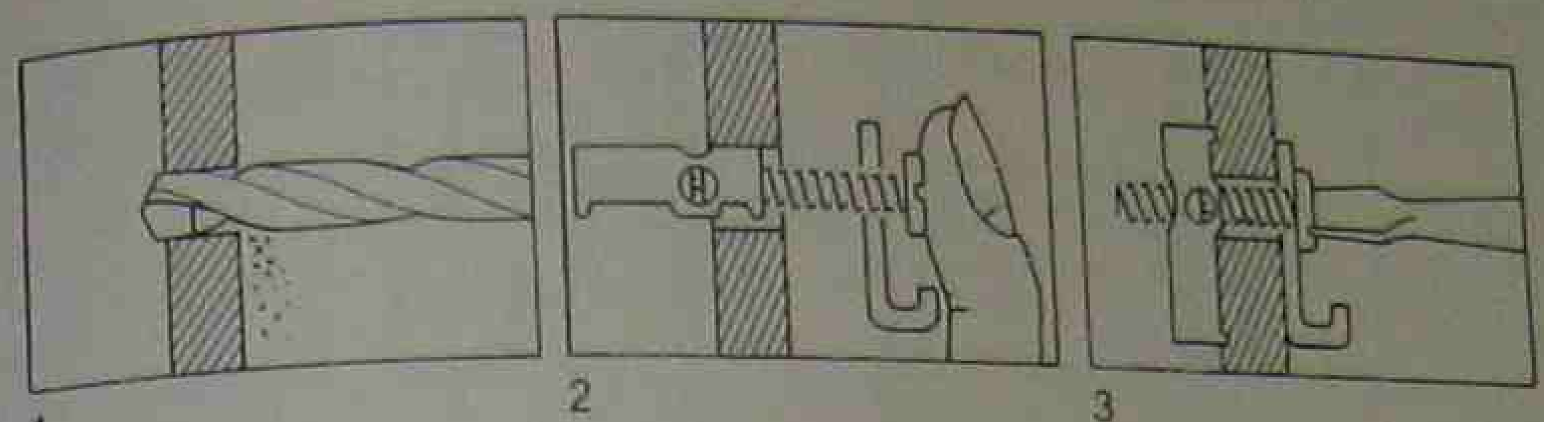


Fig. 6.6 Gravity toggle fixing

bar of the device.
2. Pass the machine screw through the subsidiary component and run the nut of the device on to the screw. Holding the bar horizontally, insert it through the fixing hole and allow it to fall freely within the cavity.
3. Draw the bar back against the inside face of the cavity and tighten the screw.

Note: This device will not operate when the base is lying in the horizontal plane. In that case a spring toggle should be used.

C2 Spring toggles

Description The spring toggle is often referred to as a 'butterfly' toggle and consists of a plated steel, hinged bar which is maintained in the open position by the operation of a spring. The toggle bar is pivoted on a swivel nut which receives a machine screw. Screw diameters range from M3 to M6 and lengths from 50 to 80 mm.

Applications This type is applicable to most vertical or horizontal cellular bases with soft facing materials like plasterboard, fibreboard etc, provided that the width of the cavity will allow the bar, which is folded for insertion, to open inside. The bar cannot be retrieved and will remain in the cavity whenever the machine screw is unscrewed (Fig. 6.7).

Setting instructions

1. Drill a hole in the base of sufficient diameter to take the bar in a folded condition.
2. Pass the machine screw through the subsidiary component and run the nut of the device on to the screw. Fold

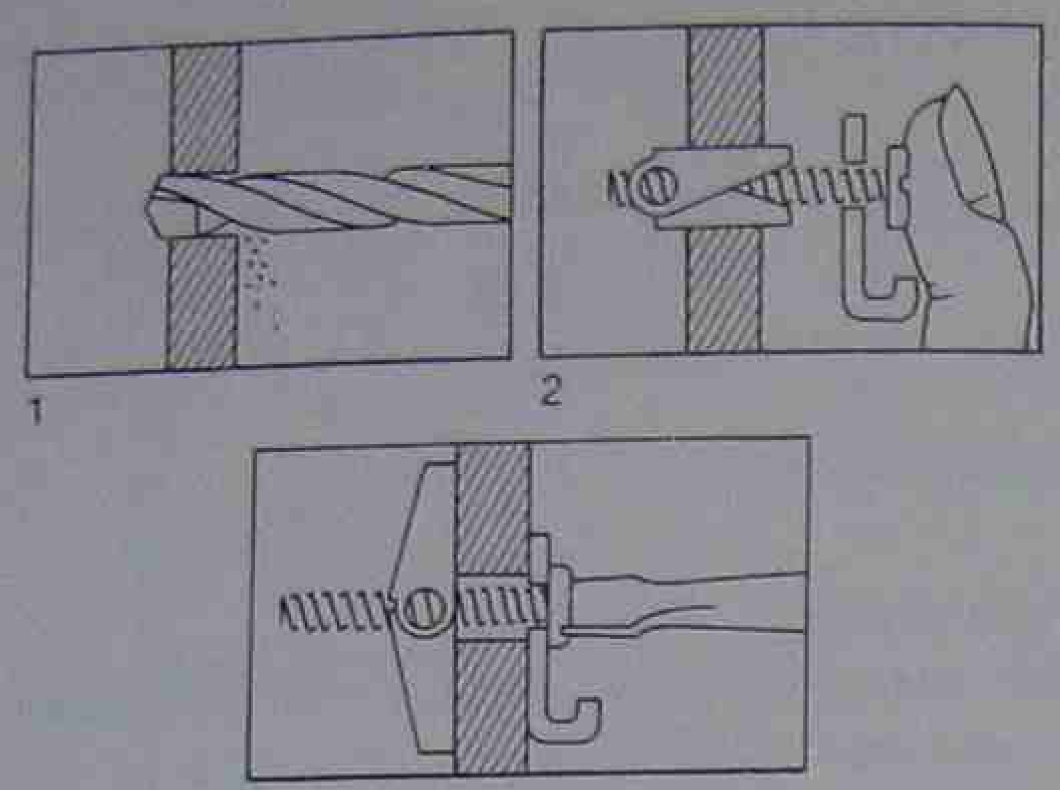


Fig. 6.7 Spring toggle

the hinged bar against the spring and insert it through the fixing hole until the spring is free to open the bar.

3. Draw the bar back against the inside face of the cavity and tighten the screw.

C3 Cord-retention toggles

Description Unlike the toggle fixings described above, this type of fixing has a notched nylon bar connected to a nylon cord running through a movable nylon guide sleeve. Toggle bars are either 31.8 or 57.2 mm long. This device makes a fixing point to receive a woodscrew.

Applications This type of device can be used in similar applications as the previous toggle fixings. It is particularly useful for making fixings in plasterboard ceilings. The fixing is non-retrievable (Fig. 6.8).

Setting instructions

1. Drill a hole in the base of sufficient diameter to take the toggle bar.
2. Manoeuvre the toggle bar into position using the attached nylon cord.
3. Slide the guide sleeve up the cord to close the fixing hole.
4. Pass the screw through the subsidiary component and

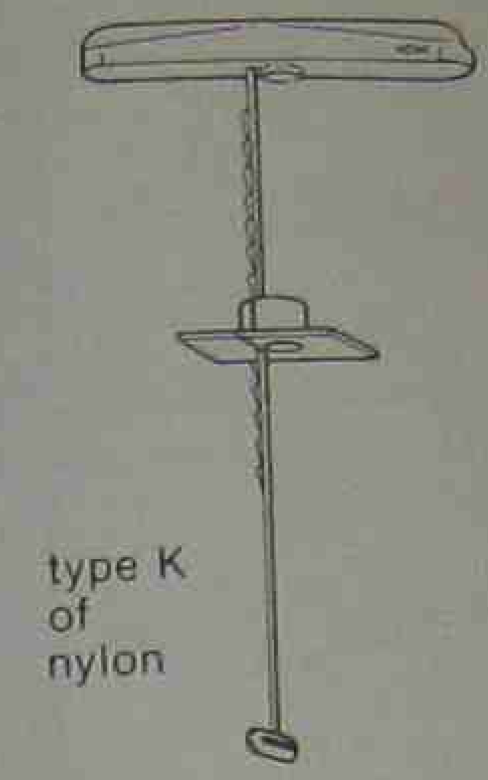


Fig. 6.8 Cord-retention toggle

centre the screw accurately through the guide sleeve and into the threaded hole in the toggle bar.

5. Cut off the surplus length of nylon cord protruding from the guide sleeve before finally fixing the subsidiary component.

C4 Light plastic toggle plugs (Thunder toggle plug)

Description These are similar to the gravity toggle, but the toggle bar in this case is housed in the body of the plug and is made of plastic. A screw is supplied with the device and a plastic strap is attached to the head of the toggle bar. The body of the plug has a small flange to prevent the plug falling into the cavity.

Applications These toggle fasteners can be used in most cellular bases where the cavity width is at least 31 mm and the facing board is between 8 and 9.6 mm thick. Plastic toggles are retrievable (Fig. 6.9).

Setting instructions

1. Drill a hole of the recommended diameter. Insert the plug.
2. Push the screw into the plug forcing the toggle into a position parallel to the base surfacing layer. This will align the screw with a threaded hole in the toggle.
3. Thread the screw through the subsidiary component and insert the screw into the toggle bar.

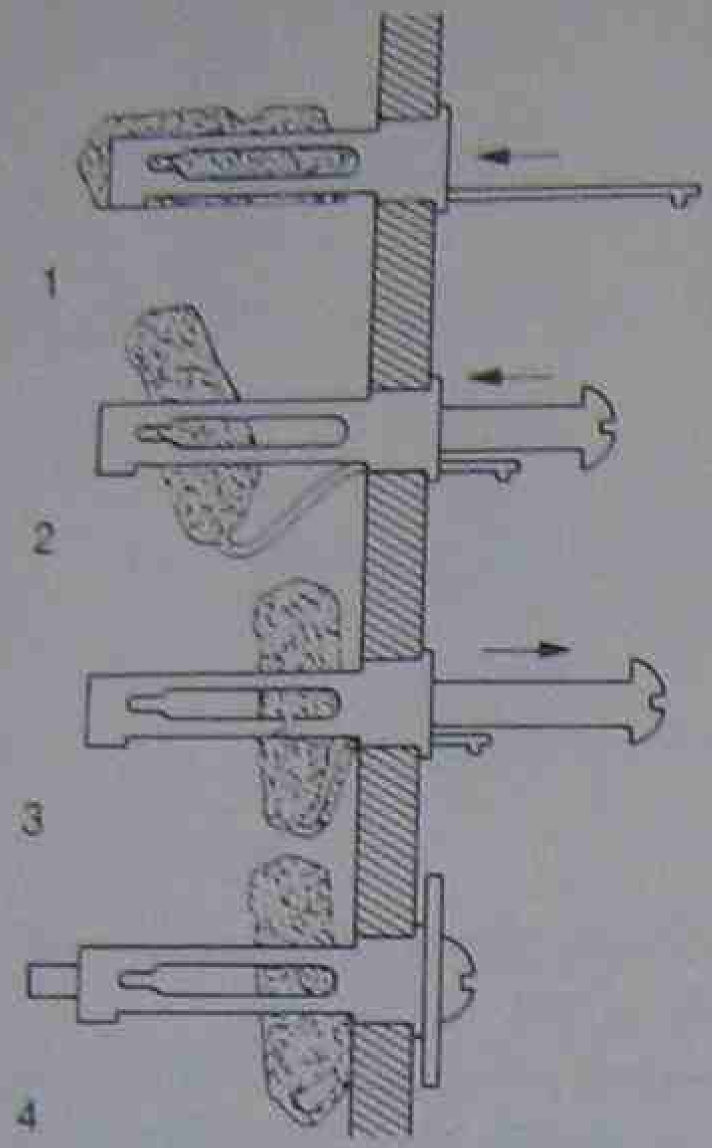


Fig. 6.9 Thunder toggle plug

4. Draw the toggle bar against the inside of the cavity using the screw and tighten the screw.

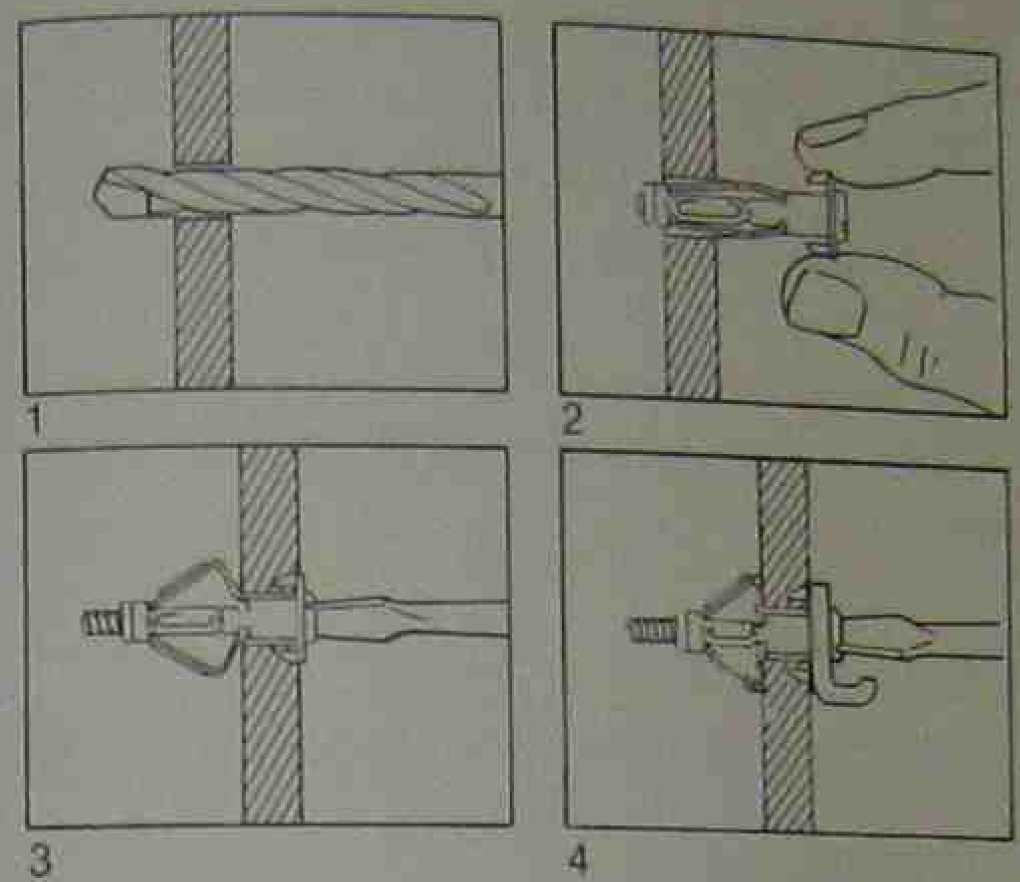
Note: Ensure that the plastic strap remains projecting from the plug during the whole of this operation and then conceal it behind the subsidiary component.

Removal instructions

1. Unscrew the subsidiary component.
2. Using the screw, push the toggle back into its original position parallel to the base surfacing layer and at the back of the plug body.
3. Using the integral strap, pull the bar back into alignment with the body of the plug.
4. Withdraw the plug.

Section D: Umbrella cavity fixings (Molly Nut, Rawlplug Interset, Fischer Fixoplac-metal anchor)

The fundamental difference between these fixings and expanding



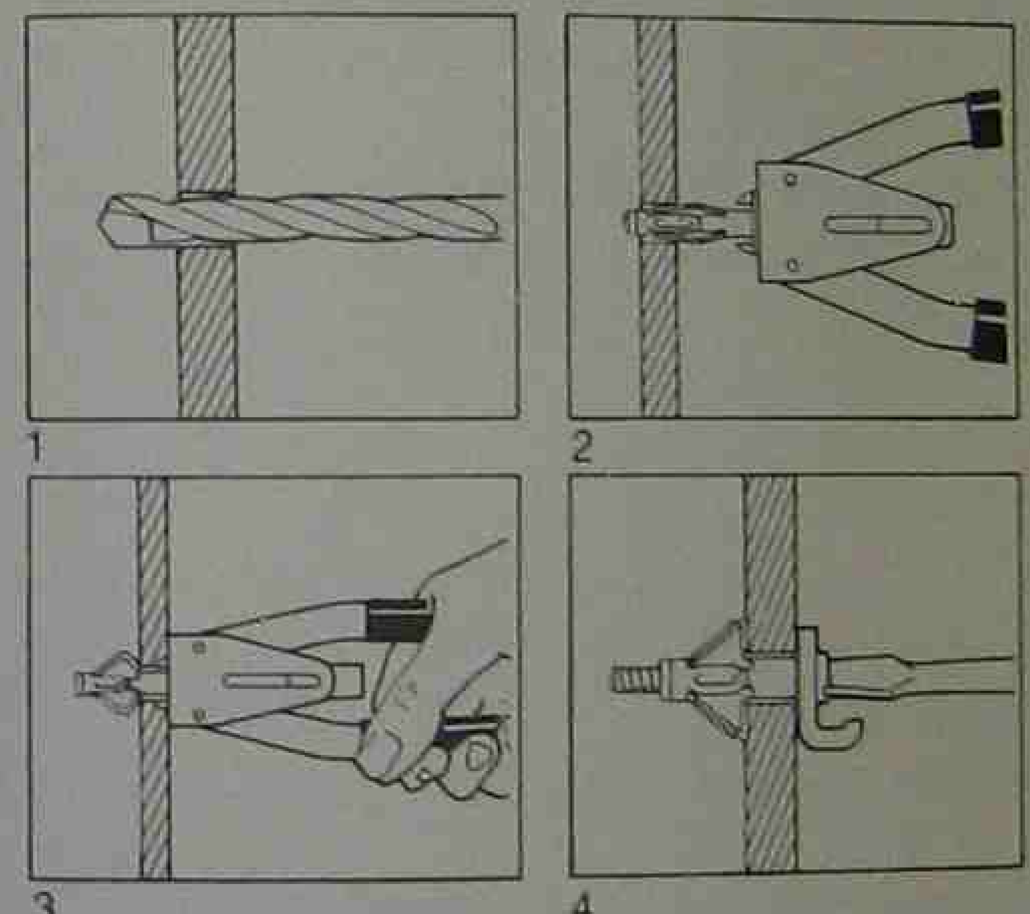


Fig. 6.10 Umbrella cavity fixing (a) Setting procedure using screwdriver (b) Setting procedure using setting tool.

cavity fixings is that the umbrella fixing provides a permanent threaded hole in the face of the base, supported within the cavity by a series of metal legs and from which a screw can be removed and replaced as frequently as is required. Both types of fixing do, however, have one characteristic in common; they both rely on the expansion of the device within the cavity.

Description These plated steel fixings consist of four folding legs attached at their near end to the flanged body of the device and at the further end to a nut which receives a machine screw of M4, M5 or M6 diameter. By screwing the machine screw into the nut, the legs are folded outwards and drawn back against the rear surface of the facing material. A setting tool is provided by some manufacturers to speed up setting operations.

Applications Umbrella cavity fixings can be used to make quick permanent fixings to plasterboard, plywood, chipboard and similar sheet materials (Fig. 6.10).

Setting instructions

1. Drill a hole of the diameter recommended by the manufacturer.
2. Insert the umbrella fixing.
3. Screw the machine screw into the device to fold out the legs. Alternatively a setting tool supplied by the fixing manufacturer can be used.
4. Remove the screw, thread it through the subsidiary component and screw it back again into the fixing to secure the component.

7

Adhesives

The range of available adhesives for use in building has considerably increased in recent years. Also the complexity of the chemical formulations involved in modern adhesives is so daunting that the subject can only be dealt with in general terms ~~in a book of this type~~. It is sometimes difficult to allocate with certainty a proprietary adhesive to a general category, or even to be sure of its suitability for a particular use. All recommendations given here should be checked against the manufacturer's specific instructions for the adhesive concerned.

Some of the more sophisticated modern adhesives, such as some of the thermosets (adhesives whose chemical set is encouraged by heat), are difficult, or even impossible, to use on site. Most of them require careful mixing and application which could prove impractical in site conditions. A few give off highly flammable and toxic gases which can introduce problems of safety in their use and storage.

Nevertheless, adhesives have distinct advantages to offer when compared with mechanical fixings in certain applications, particularly where modern plastic materials are being fixed, or where visible fixings are unacceptable, or in cases where high-strength repairs are necessary to concrete structures. Adhesives also eliminate much of the preparatory work to making the fixing, such operations as drilling and plugging, merely requiring the surfaces which are to be joined to be scrupulously clean and possibly sanded and primed.

With the introduction of synthetic resin adhesives, reliable high-strength structural joints could be achieved with glue for the first time. This structural use of adhesives originated in the fabrication of timber components, such as laminated beams and stressed-skin decking panels which were mass-produced in factories. This use of adhesives to produce structural components has remained largely a factory, and not a site, practice. However, more recently there has been a marked increase in the use of adhesives on site in connection with structural concrete. We have already discussed one aspect of resin adhesives applied in this way when we considered chemical anchors in section A2 of Chapter 3. These structural adhesives are primarily intended for site use and will be dealt with later in this chapter.

Generally, though, adhesives tend to be used on site for making non-structural joints, often between the structure and an applied finish, like a flooring or a ceiling tile or a decorative plastic wall lining. Some adhesives are manufactured to fulfil one specific task; others may have a number of different applications.

As in mechanical fixings, there tends to be a heavy, immobile part of the joint – the base – and a lighter, more mobile component. Both can theoretically be referred to as *adherends*, but the heavier part is sometimes in adhesive jargon called the *substrate*. Generally, for consistency, we shall continue to refer to the substrate, where appropriate, as the base.

How to use adhesives

The most important piece of advice that can be given to anyone using adhesives is to follow the manufacturer's instructions. These have been arrived at not to make the use of adhesives unnecessarily complicated, but for very good chemical and physical reasons in order to achieve the most efficient bond between adherends.

The way an adhesive is used depends on the type of adhesive, whether it is a *contact adhesive*, a *wet-stick adhesive*, or one of the other application types. There are other ways of classifying glues – by their ingredients, or their method of setting – but these aspects can more usefully be examined in the background part of this chapter.

Contact adhesive

This type of adhesive, sometimes referred to as a *two-way dry-stick adhesive*, is used when both the base and the component are non-porous. Once a joint made with a contact adhesive has been closed, there is no chance of adjusting the position of one part of the joint relative to the other. In other words, contact adhesives have high grab. This is very useful for some types of work where

clamping the parts of the joint together would be difficult; but it makes the correct positioning of the parts of the joint at the first attempt vital. Contact adhesives are usually neoprene rubber solutions in organic solvents and they give off flammable and toxic gases (see p. 172). Today a new breed of water-based contact adhesives is beginning to reach the market, but their use is limited to warm, indoor applications at present. Contact adhesives are used to fix plastic laminates, cork tiles, etc. in other applications where high grab is essential.

Method

1. Ensure the surfaces to be bonded are clean, free from dust, oil, grease or loose material.
2. Apply the adhesive to *both* surfaces to be joined in a thin, even coating to manufacturer's instructions. (This will often be done by means of a metal or plastic comb or notched trowel with notches of a specified size.)
3. Allow the adhesive to dry (its solvent to evaporate) for the period recommended by the manufacturer. This 'open time' is important: too short an open time will mean 'wet' adhesive being trapped in the joint, resulting in a poor bond and 'springing' of the component; too long an open time will mean that the two layers of dried adhesive will not fuse when brought together.
4. Bring the parts of the joint together and apply an even pressure over the whole area of the joint. It is important when sticking down large, thin sheet materials to place the sheet down starting from one side and working steadily across the length of the sheet, making sure no air bubbles are trapped in the joint.

Wet-stick adhesive

This adhesive, sometimes referred to as a *one-way wet adhesive*, is used when either the base or the component, or both of them, are porous, thus allowing the solvent in the adhesive to soak into the material. These adhesives have varying degrees of *grab* and usually allow minor adjustment to be made to the parts of the joint after closing. They also have varying degrees of 'thickness', the selection of this quality depending on the roughness of the adherends and whether the adhesive needs to fill gaps at the interface of the two materials (if the adhesive needs to be *gap-filling*).

Method

1. Ensure the surfaces to be bonded are clean, free from dust, oil, grease or loose material.
2. Apply adhesive to *one* surface to be joined in an even layer and to the thickness recommended by the manufacturer. (This will often be done by means of a metal or plastic comb or notched trowel with notches of a specified size.)
3. On highly porous surfaces it may be necessary to apply two coats of adhesive, the first being used to prime the surface.
4. Once more the manufacturer's instructions concerning open time should be followed. Emulsion adhesives on non-porous bases should be allowed to become tacky before closing; while epoxide and polyester-based adhesives should be closed immediately.
5. Apply an even pressure over the whole area of the joint after closing until the adhesive has hardened. This may involve the use of a roller. Depending on the degree of grab of the adhesive, position of the joint, weight of the component and the adhesive drying time, the joint may have to be clamped in position for some time.

Note: Gum spirit adhesives (used mainly for sticking-down linoleum) and some latex-based adhesives form surface skins after application. These must be broken on closing the joint, by sliding one surface against the other when placing the adherends together. If this is not done a poor bond could result.

Gun-grade adhesive

Unlike the two previous types of adhesive, this is not applied as a continuous coating, but in beads or blobs. It is used mainly to stick board materials, such as plasterboard or chipboard, to solid backings and has moderate gap-filling qualities. It is often neoprene rubber-based and has good wet-grab characteristics. This type of adhesive is often used for fixing skirtings and architraves, stair nosings, or refixing loose wood block flooring.

Method

1. Ensure the surfaces to be bonded are clean, free from dust, oil, grease or loose material.
2. Apply adhesive direct from the cartridge in a series of blobs to the back of the component and at spacing recommended by the manufacturer for the type of component. In some cases the recommendation may be for a continuous bead of adhesive round the edges of the component, with intermediate beads across the component. The bond develops more quickly in this case.
3. Slide the component into position, placing it about 25mm away from its final position and pushing it home. This movement breaks the surface skin of the adhesive. The manufacturer's guidance on open time should be observed.
4. Press into place with an even, overall pressure.

Note: Provided the component is not warped, bond will be sufficient to hold most components in place and work can proceed to the next component. If the component, however, is warped and there is a danger of it springing from the base, it will be necessary to pin it into position in the warped area, or the following procedure would be followed.

Apply the adhesive as above. Offer the component up to its final position so that the adhesive blobs or beads wet the base. Pull away and allow the solvent to evaporate for about 20 minutes. Then replace the component, at which time the grab should be sufficient to retain the component.

Foam pads

These are adhesive-impregnated foam pads used to provide an alternative to blob-applied gun-grade adhesive.

Method

1. Ensure the surfaces to be bonded are clean, free from dust, oil, grease or loose material.
2. Bond pieces of foam, at the intervals recommended by the manufacturer depending on the weight of the component to be fixed, to the base, using contact adhesive.

3. Apply a coat of contact adhesive to the outer face of the foam pads.
4. Place the component to be bonded briefly in position against the pads in order to transfer some of the adhesive to the component.
5. Apply further adhesive, if necessary, to the wetted areas on the component and also on the pads.
6. When tacky dry (or at a time recommended by the manufacturer), replace the board and apply pressure.

Note: This method can be used, when the surface of the base is uneven, by doubling-up the thickness of the pads in low-lying areas. A double-sided self-adhesive tape is also available as an alternative to the method given above. In this case merely peel off the protective paper from the adhesive surface before placing. The method is easy to use, but should only be applied to smooth surfaces and when the component to be fixed is reasonably light.

Adhesives used in structural repairs

Some adhesives can be added to sand/cement mixes to give a mortar of enhanced adhesion and one which can be used to effect thin repairs to elements like concrete structures and floor surfaces without the later danger of the repair cracking or curling. Polyvinyl acetate (PVA), epoxy resin, styrene butadiene (SBR) latex and styrene acrylic emulsion adhesives can be used in this context, but the former must not be used externally or in damp conditions.

There are many pre-mixed proprietary concrete repair formulations on the market. In all cases the manufacturer's instructions will need to be followed carefully and the surface to be treated should be clean and free from all loose material. Also, if reinforcement has been exposed, all rust will need to be cleaned off and the steel treated with a suitable rust inhibitor prior to the repair being carried out.

Adhesives as bonding agents

Adhesives are often used to form a bond between thin sand/-cement screeds, cement renderings and gypsum plaster coatings and smooth, dense backing surfaces. This is particularly necessary when a screed is laid on a concrete subfloor which is no longer

'green' to avoid the screed cracking and lifting. PVA emulsion adhesives are often used as bonding aids, applied as follows.

Method

1. Ensure the surface to be treated is clean, free from dust, loose material and contamination.
2. Prime the surface with a little of the polyvinyl acetate adhesive diluted with water. Allow to dry.
3. Apply a concentrated solution of polyvinyl acetate adhesive to the surface using a brush or spray.
4. Trowel-on the rendering, screed or plaster. In the case of most popular brands of polyvinyl acetate adhesive, the finish should be applied onto the wet and tacky coat. A few manufacturers, however, specify that their products must be allowed to dry before applying the rendering, screed or plaster.

Note: In more extreme cases where a bond is considered difficult to achieve a key can be provided by mixing a slurry of polyvinyl acetate bonding aid and ordinary Portland cement. This is first brushed onto the base, before the finish is trowelled-on, either before or after the slurry has fully set, depending on the recommendations of the manufacturer. Polyvinyl acetate should not be used in wet locations. Here epoxy, styrene butadiene latex or styrene acrylic alternatives should be used.

Hot-melt adhesives

There is a growing use of hot-melt adhesives (ones which are applied hot and set on cooling) on site. This is the result of the introduction of a number of light, highly mobile electric-powered pistol applicators, like the BIF Hipermatic range (Fig. 7.1).

This technique was originally developed for factory use, but now is becoming more familiar on site, provided electric power is available. Pistols have various interchangeable application nozzles and BIF, as an example, supplies a range of eight different hot-melt adhesives for use with different adherends, such as wood, plastics, glass and metal. The adhesive is supplied in a solid cartridge, 44.5 mm diameter by 50 mm in length. This is inserted into the applicator and, after an initial warm-up period of about 3 minutes, is extruded on the pressing of the applicator's trigger.

Fig. 7.1 Pneumatic BIF-Bond T 718 hot-melt applicator

Hot-melt adhesives have the benefit of being solvent-free, making their use and storage more safe, simple and practically odourless. It is also a less wasteful method of using adhesive, all the material being capable of being precisely placed where it is needed and without loss on tools, or in cans or mixing kettles.

General advice

In all cases where adhesives are to be used on site there are a few rules which should be observed:

1. Ensure that the adhesive you use is recommended for that particular application by the manufacturer. Is it compatible with the materials to be adhered to each other? Will it allow the necessary amount of adjustment to be made to the joint after it is closed? Has it the necessary gap-filling properties to suit the roughness of the adherends?
2. Always precisely follow the adhesive manufacturer's instructions in the use and application of the product.
3. All surfaces to be bonded should always be clean and free from dust, grease or loose, flaky material. If the surface has been gloss-painted, make sure the adhesive is compatible with the paint, ensure that the paint is not loose or flaking and then roughen its surface with sandpaper and wipe off the dust.
4. Make sure all surfaces are dry and protect the joint from moisture after closing until the adhesive has cured.
5. Use the correct tools to apply the adhesive (notched trowel with the right-sized notches) and make sure tools are cleaned after use, employing materials recommended by the adhesive manufacturer. Do not use solvents to clean adhesive from your skin.
6. Always observe the correct open time of a joint.
7. Do not attempt to make a glued joint when the temperature is outside the temperature range recommended by the adhesive manufacturer.
8. Do not store an adhesive beyond its recommended shelf life and always keep it within the storage temperatures recommended by the manufacturer.

Specific applications of adhesives

A number of specific applications of adhesives on site is listed in Table 7.1 together with the type of adhesive recommended for use.

Table 7.1 Adhesives recommended for particular applications

Application	*Type of adhesive*
Internal joinery	Casein, PVA (or the range of formaldehyde adhesives – not usually used on site)
Internal wall finishes	
Plasterboard	Gypsum plaster, PVA, gun-grade synthetic rubber
Decorative timber wallboard	Synthetic rubber (solvent based), synthetic rubber (emulsion)
Vinyl (rigid)	Synthetic rubber (emulsion), acrylic
Vinyl (flexible)	Acrylic
Textile	Synthetic rubber (solvent-based), PVA
Cork	Synthetic rubber (solvent-based), PVA
Ceramic wall tiles:	
to masonry base	Natural rubber latex cement, cement, PVA, synthetic rubber latex, epoxide, acrylic
to gypsum plaster	Natural rubber latex cement, PVA, synthetic rubber latex, acrylic
to timber-based boarding	Natural rubber latex cement, PVA, synthetic rubber latex, epoxide, acrylic
to paintwork	Natural rubber latex cement, PVA, synthetic rubber latex, acrylic
to metal	Natural rubber latex cement, PVA, synthetic rubber latex, epoxide, acrylic
Architraves/skirtings	Gun-grade rubber
Ceiling tiles	
Acoustic tiles	Gum spirit, synthetic rubber (solvent-based)
Polystyrene tiles	PVA
Gypsum plaster coving	Gypsum plaster
Insulating panels (glass fibre backed)	Synthetic rubber (solvent-based)
Floor finishes	
Thermoplastic tiles	Bitumen, synthetic rubber (emulsion)
Vinyl asbestos tiles	Bitumen (not on mastic asphalt screed)
Flexible vinyl tiles (BS 3261 Type B)	Bitumen, epoxide
Flexible vinyl tiles (BS 3261 Type A)	Synthetic rubber (emulsion), acrylic, epoxide
Cushion-backed vinyl	Synthetic rubber (emulsion), acrylic, epoxide
Linoleum	Gum spirit, lignin
Vinyl-backed linoleum	Synthetic rubber (emulsion)
Rubber	Rubber solution, epoxide
Textile	Synthetic rubber (emulsion), gum spirit
PVC-backed textile	Synthetic rubber (emulsion), acrylic
Cork (unbacked)	Elastomer (solvent-based), synthetic rubber latex, gum spirit, lignin
Ceramic tiles:	
to cementitious backings and boards	Natural rubber latex cement, cement, synthetic resin emulsion, epoxide
to asphalt	Natural rubber latex cement
to vinyl	Cement
to steel	Natural rubber latex cement, epoxide
Wood blocks	Bitumen
External wall finishes	
Ceramic wall tiles, mosaics, brick slips, etc.	Cement-based adhesives, synthetic rubber latex emulsion with cement/sand, epoxide
Decorative laminates to timber	Rubber solution (protecting joint from water penetration)
Joinery	Phenol formaldehyde, resorcinol formaldehyde (not usually site applied)

Additional explanatory detail can be found in the background section of this chapter.

Safety

All adhesive manufacturers now supply information on the safe handling of their products as required by the Health and Safety at Work Act, 1974. Particularly petroleum-based and synthetic-resin adhesives need careful handling and storage, not only on account of their flammability, but also because of the toxic fumes they give off. Adequate ventilation should always be ensured in areas where these adhesives are being used.

For further information see *Site Safety* by J. C. Laney in this Site Practice series.

Background

Principles of adhesion

If it were possible to produce two perfectly flat surfaces and bring them into contact, face-to-face, spontaneous adhesion would occur due to molecular forces of attraction at their interface. The nearest example of this phenomenon in everyday experience is the resistance that can be felt when two sheets of glass in face-to-face contact are separated.

But even the surface of glass does not have the degree of perfection required to produce spontaneous adhesion. The smoothest surface is rough by comparison with the molecular scale of smoothness required. As a result, a third component has to be introduced to overcome the surface irregularities of the parts to be joined together (the adherends) and form a rigid connection by solidifying or setting. This third component is a paste or liquid adhesive.

The degree of 'thickness' (or viscosity) of the adhesive will affect its gap-filling properties – its ability to fill the more major irregularities in the surface of the adherends. The adhesive must be compatible with both adherends; i.e. it must have molecular attraction for the materials of which they are made. For example, urea formaldehyde synthetic resin adhesive is not compatible with a rubber or polystyrene adherend. Urea formaldehyde is, however, compatible with wood and wood-based products. Table 7.2

Table 7.2 Material compatability groupings

Group A *Polar group*	*Group B* *Non-polar group*
Cellulose (wood, cotton, paper etc)	Rubber
Urea formaldehyde (UF)	Polystyrene
Phenol formaldehyde (PF)	Polyethylene
Water	Teflon
Alcohol	Benzene
Metal oxides	Mineral oils

All materials in each group are compatible with other members of the same group and incompatible with all members of the opposite group.

shows the two major compatibility groupings of adhesives and adherends.

Adhesives need to solidify in order to produce a firm joint. This setting process takes place as a result of one (or a mixture) of the following processes.

1. Through loss of solvent (water or organic solvent) due to evaporation. Non-aqueous solvent-based adhesives are often flammable. The setting of solvent-based adhesives can be reversed by reintroducing the solvent. This makes water-based adhesives, like animal and cellulose glues, vulnerable in damp conditions. Today there is a movement away from organic solvent-based adhesives towards water-based adhesives because of their greater ease of use and greater safety. Although organic solvent-based adhesives produce strong joints, they do release flammable and toxic vapours.
2. By the coalescing of binders in emulsion and latex adhesives as the water is absorbed in the adherends and evaporates.
3. By the cooling of an adhesive applied in a hot state. These hot-melt thermoplastic glues are usually supplied in a solid state and heated to 160–200 °C for application. They should not be confused with thermosetting adhesives which require heat to cure.
4. By a chemical cross-linking process. Synthetic resin adhesives, like resorcinol formaldehyde, melamine formaldehyde and the epoxide range belong to this chemically setting group. The process is irreversible and is

initiated by the introduction of a chemical called a hardener into the resin and/or the application of heat. Adhesives which set as a result of the application of heat are called thermosetting adhesives. They have little application on site due to the hot presses required for their use, but they are extensively used in the factory for the manufacture of timber components, plywood and other timber-board products.

What adhesive to use

For an adhesive to produce a permanent union between adherends it should:

1. Be compatible with both adherends.
2. As nearly as possible match the strength of the adherends.
3. Resist the conditions which it is likely to meet in use (dampness, heat, mould growth, chemical pollution, etc.), (see Table 7.3).
4. Fill the imperfections in the surfaces of the adherends to produce a consistent film; if necessary inert fillers such as wood flour may be used to produce gap-filling characteristics.
5. Remove minor surface impurities from the surface of the adherends.
6. In setting should not produce unacceptable stress due to shrinkage.
7. Have a setting time consistent with the nature of the work; i.e. sufficient pot life, sufficient open time of joint to allow correct assembly of its parts, and sufficient speed of setting to harden before the joint is likely to be disturbed.

Table 7.3 gives an indication of the resistance of various adhesives to degrading by moisture and their vulnerability to attack by bacteria. Table 7.4 lists many of the adhesives used in building in basic ingredient categories and explains their general characteristics and applications.

Table 7.3 Resistance classification of adhesives

Type	*Category*	*Durability*				*Is attacked by bacteria*
		WBP	*BR*	*MR*	*INT*	
Starch	Natural				X	X
Plant-protein	Natural				X	X
Casein	Natural				X	X
Cellulose	Natural				X	X
Bitumen	Natural				X	
PVA	Thermoplastic				X	
Epoxide	Thermoset	X				
Urea formaldehyde	Thermoset			X		
Melamine formaldehyde	Thermoset		X			
Resorcinol formaldehyde	Thermoset	X				
Phenol formaldehyde	Thermoset	X				

WBP = weatherproof and boilproof. BR = boil-resistant (good weather resistance) MR = moisture resistant. INT = interior use only.

Table 7.4 Chief adhesives used in building-in ingredient classification

Group	*Type*	*Constituents*	*Description*	*Uses*	*Suitable bases made from:*
Natural	Animal	Hides & bones	Set is reversible with moist conditions. Little used in building today	Internal joinery, furniture	Wood
	Starch	Maize, cassova or wheat		Wallpaper	Portland cement, plaster, wood
	Plant-protein	Soya bean		Internal joinery	Wood
	Cellulose			Expanded polystyrene, wallpaper	Portland cement, plaster, wood
	Casein	Milk curd	Designated INT – interior glue. Sets part by evaporation; part by chemical action. Little resistance to moisture and micro-organisms	Internal joinery	Wood
	Gum spirit	Rosins in alcohol with fillers	Insoluble in water, but degrades under wet alkaline conditions. Careful priming needed when used with porous friable adherends	*Floors*: cork, lino, most PVCs, felt *Walls/ceilings*: wood fibre, expanded polystyrene, cork, polyurethane (unbacked)	Portland cement, wood, plaster, mastic asphalt
	Lignin paste	Sulphite lye in water with fillers	Softened by water. Used as a flooring adhesive in dry areas	*Floors*: Cork, lino, cork-backed PVC *Walls/ceilings*: lino, cork	Portland cement, wood
	Bitumen	Bitumen	Applied hot; thermoplastic. No drying time needed	*Floors*: Wood block Bitumen roofing felt. Foam glass	Portland cement, wood
	Bitumen solution	Bitumen blends in solvents	Solvent adhesive	*Floors*: Flexible PVC, vinyl asbestos, thermoplastic tiles	Portland cement, wood
	Bitumen/rubber	Aqueous emulsion of bitumen with natural or synthetic rubber latex	Slow-setting	*Floors*: flexible PVC, vinyl asbestos, thermoplastic tiles, wood block, wood mosaic *Walls/ceilings*: cork, expanded polystyrene, polyurethane (unbacked), foamglass	Portland cement, wood
Elastomeric/ rubber	Rubber solution	Solution of natural or synthetic rubber in organic solvents, probably with resins and other modifiers	One-way wet-stick if one or both adherends is porous; two-way dry-stick contact adhesive with non-porous adherends. No positional adjustment possible after placing unless thixotropic grade used	*Floors*: cork, flexible unbacked PVC, rubber *Walls/ceilings*: wood fibre, ceramic, rubber, felt-backed PVC, chipboard, hardboard, decorative laminates, plasterboard, plywood, cork, felts, PVC or rubber-backed textiles,	Portland cement, wood, plaster

Table 7.4 Cont'd

Group	*Type*	*Constituents*	*Description*	*Uses*	*Suitable bases made from:*
				expanded polystyrene, paper-faced polyurethane, PVC skirtings & nosings	
	Filled-rubber solution	Solution of synthetic or reclaim rubber containing fillers, resins and modifiers	Mastic-like consistency, applied with notched trowel or cartridge gun. Gap-filling ability up to 6 mm	*Walls/ceilings*: wood fibre, expanded polystyrene, rubber, chipboard, decorative laminates, plasterboard, plywood, plasticized PVC, wood wool slabs, backed and unbacked PVC, PVC or timber skirtings	Portland cement, wood, plaster
	SBR emulsion	Aqueous dispersion of blends of styrene butadiene rubbers with synthetic resin modifiers and/or fillers	Good bond of moderate strength. Resistance to damp and alkaline conditions	*Floors*: cork, lino tiles, most PVCs, most textiles, thermoplastic tiles *Walls*: cork	Portland cement, wood, plaster; limited suitability for mastic asphalt
Thermoplastics	Acrylic emulsion	Emulsion of acrylate ester copolymers which may contain modifiers and/or fillers	Resistant to damp conditions. Good bond under warm conditions	*Floors*: felt-backed lino tiles, most PVCs, most textiles *Walls/ceilings*: ceramic, lino, plasterboard, PVC paper-backed polyurethane	Portland cement, wood, limited suitability for mastic asphalt
	Polyvinyl acetate (PVA)	Aqueous emulsion of PVA with fillers	*One-part* adhesive used primarily as a wood glue, but due to its elastic glue line, not used for structural joints. Slowly degraded by moist conditions. Designated INT-interior glue. *Two-part* modified adhesive can be used for external joinery; but not for structural joints	*Floors*: needle loom felt-backed PVC, felts *Walls/ceilings*: wood fibre, mineral fibre, expanded plastertiles, ceramic, expanded polystyrene, plasterboard, heavy wallpapers, paper-faced polyurethane	Portland cement, wood, plaster
Thermosets	Epoxides (epoxy adhesives)	Wide range of chemically-setting synthetic adhesives	Two or three-part adhesive in which liquid resin is mixed with hardener immediately before placing. High bond strength, good water and alkali resistance. Systems with fillers are used as adhesive mortar with negligable shrinkage but little flexibility	*Floors*: ceramic, flexible unbacked PVC *Walls/ceilings*: ceramic, metallic, chipboard, hardboard, plywood, rigid PVC *External*: brick slips, concrete repair, steel to concrete, stair nosings	Portland cement, wood, brick

Table 7.4 Cont'd

Group	*Type*	*Constituents*	*Description*	*Uses*	*Suitable bases made from:*
	Formaldehyde	Range of chemically-setting synthetic resin adhesives	Two-part adhesives which can be of cold-setting type; but which set more quickly with heat	Use generally restricted to timber-to-timber joints – mostly factory applied	
		Urea-formaldehyde (UF)	Designated MR adhesive – moisture resistant moderately weather resistant – only limited life used externally.		
		Melamine/urea-formaldehyde (MF/UF) Melamine-formaldehyde (MF)	Designated BR adhesive – boil resistant – longer life externally, but not prolonged use	Use generally restricted to timber-to-timber joints – mostly factory applied	
		Phenol-formaldehyde (PF) Resorcinol-formaldehyde (RF)	Designated WBP adhesive – weather and boil proof – prolonged external use		
	Unsaturated polyester	Chemically-setting synthetic resin adhesive	Two- or three-part with or without fillers, mixed immediately prior to application. High strength bonds, slight setting shrinkage. Good resistance to moisture and weathering. Systems incorporating fillers used as adhesive mortars	*Walls*: timber skirtings *External*: brick slips, concrete repair *Miscellaneous*: steel to concrete, stair nosings	Portland cement, brick
Cement based	Polymer modified cement	Synthetic resin modified cement	Grades available for wet and dry conditions	*Floors*: ceramic *Walls/ceilings*: mineral fibre, expanded plaster, ceramic, wood wool	Portland cement, brick *NOT* suitable for plaster and timber substrates
	Sand/cement plus SBR	Sand/cement mortar modified with SBR rubber	For external use, even to damp surfaces. Cheaper and easier to use than filled polyester and epoxide eqivalents. Has some initial grab	*External*: brick slips, concrete repairs	Portland cement, brick

Appendix 1

List of proprietary fixings in base material classifications

This list of mechanical fixing devices and manufacturers cannot aim to be comprehensive. It does, however, contain mention of the major manufacturers of proprietary fixing, all of whom have assisted in the preparation of this book. It does not endeavour to list manufacturers of standard non-proprietary items like nails or bolts.

Mass walling base

Plugs

Citmart: Syba anchor
Fischer: Nylon wallplugs; Wallplug Type S; Wallplug Type S-R; Wallplug S 10 RL; Wallplug S 10 H-R; Facade fixing; Frame fixing; Twist Lock Anchor Type GB
Hilti: HLD nylon anchor; FD nylon plug; HG anchor; HT frame anchor; Celanail
Mungo: Various patterns of nylon anchors
Rawlplug: Fibre plugs; Plastic plugs; Rawplastic compound; Rawlnut; Nailins
Spit: Plastic plug; Nylon anchor; Nail-in expansion anchor
Stainlessfix: UPAT Nylon wall plug; UPAT Nail plug UN; UPAT Nylon frame plug UR-S; UPAT Turbo low density anchor
Thorsman: Hiden anchor; Loden anchor; Nail plug TCP; Plug TP
Thunder Screw Anchors: Thuscan wallplug
Tucker: Molly anchor plugs; Molly brick fixing; Molly aerated concrete fixing; Molly collar plug; Molly frame fixing

Expanding anchors

Feb: Febolt
Fischer: Fischerbolts Type FA; Hammerset anchor Type EA; Heavy-duty nylon anchor; Heavy-duty steel anchor Type SL; Steel anchors Type MR; Suspension anchor Type L

Harris and Edgar: Ferrous and non-ferrous expanding anchors; Hemax 200 restraint fixing; Hemax 820 parallel expansion bolt; Hemax 2000 restraint fixing for thin claddings
Hilti: HA 8 R1 suspension anchor; HKD anchor; HSA anchor; HSL heavy duty anchor
Liebig: Ultra Plus expanding anchor; Safety bolt
Mungo: Various heavy duty anchors
Rawlplug: Rawlbolt loose bolt; Rawlbolt projecting bolt; Rawlbolt hook bolt; Rawlbolt eye bolt; Rawlok; Self-drill anchor; Duplex stud anchor
Spit: Heavy-duty double expansion anchor; Self-expanding stud anchor; Roc self-drilling anchor; Spit-fix anchor; Spit-grip Drop-in anchor; SDI Drop-in anchor; Frame anchor
Stainlessfix: UPAT Express anchor; UPAT expanding socket; UPAT PS anchor; UPAT wedge anchor
Thorsman: Torgrip
Tucker: Parabolt; Parabolt sleeve anchor

Insulation fasteners

Hilti: IN insulation nail; ID insulation nail
Stainlessfix: IPD insulation fastener; IPS insulation fastener

Cavity wall repair anchors

Harris and Edgar: Hemax 63

Chemical anchors

Cementation: Keyston chemical fixing
Fischer: Resin bonded anchor
Harris and Edgar: Resin anchor
Hilti: Resin anchor
Mungo: Chemical anchor
Rawlplug: Kenfix masonry anchor
Stainlessfix: UPAT chemical anchor; UPAT resin bonded Rockbolt
Tucker: Parabolt capsule anchor

Injection anchors

Fischer: Fischer injection system

Screw-in anchors

British Industrial Fasteners: Confas
Buildex: Tapcon

Masonry nails

Spit: Masonry nails
Rawlplug: [illegible]

Powder-actuated fasteners

Hilti: Various cartridge tools and pins
Rawlplug: Various cartridge tools and pins
Spit: Various cartridge tools and pins

Pneumatic-actuated fasteners

British Industrial Fastenings: Various tools and pins

Built-in fixings

[illegible]

Cast-in fixings

[illegible]

Timber base

[illegible]

Timber connectors

[illegible]

Other accessories

[illegible]

Metal base

[illegible]

Base-deforming devices

[illegible]

Powder- and pneumatic-actuated fasteners

Tools and pins manufactured by the following:
British Industrial Fastenings; Hilti; Rawlplug; Spit

Drilled-in fixings, rivets and self-tapping inserts

[illegible]

Thin-walled cellular bases

[illegible]

Schedule of manufacturers' names and addresses

Avon Manufacturing (Warwick) Ltd, P.O. Box 42, Montague Road, Warwick CV34 5LS
BAT Building and Engineering Products, Haybrook, Halesfield Industrial Estate, Telford, Shropshire TF7 4LD
Bevplate Ltd, Rectory Farm Road, Sompting, Lancing, W. Sussex BN15 0DP
British Industrial Fastenings Ltd, BIF House, Gatehouse Road, Aylesbury, Bucks HP19 3DS
The British Screw Co Ltd, 153 Kirkstall Ropad, Leeds LS4 2AT
Buildex, ITW Ltd, Darville House, 4 Oxford Road East, Windsor, Berks SL4 5DR
Catnic Components Ltd, Pontygwindy Estate, Caerphilly, Mid Glamorgan CF8 2WI
Cementation Chemicals Ltd, Denham Way, Maple Cross, Rickmansworth, Herts WD3 2SW
Citmart, Ashford Airport, Hythe, Kent CT21 4LT
Geo. Clark (Sheffield) Ltd, Greasebrough Street, Rotherham, S. Yorks S60 1RG
FEB (Great Britain) Ltd, Albany House, Swinton Hall Road, Swinton, Manchester M27 1DT
Artur Fischer Ltd, Fischer House, 25 Newtown Road, Marlow, Bucks SL7 1JY
Gangnail Ltd, The Trading Estate, Farnham, Surrey GU9 9PQ
GKN Screws and Fasteners Ltd, P.O. Box 60, Heath Street, Smethick, Warley, W. Midlands B66 2SA
Halfen Ltd, Griffin Lane, Aylesbury, Bucks HP19 3AS
Harris and Edgar Ltd, Progree Works, 222 Purley Way, Croydon CR9 4JH
Hilti (Gt Britain) Ltd, Hilti House, Chester Road, Manchester M16 0GW
Hydro-Air International (UK) Ltd, Midland House, New Road, Halesowen, W. Midlands B63 3HY
ISC Fasteners, Datim Screw Co. Ltd, 180 Brooker Road, Waltham Abbey, Essex EN9 1JJ
Liebig Bolts Ltd, Silica Road, Amington Industrial Estate, Tamworth B77 4DT
Linread-Fabco Ltd, Swindon Road, Kingsditch Industrial Estate, Cheltenham, Glos GL51 9LT
Mungo Fasteners (UK) Ltd, Kirk-Sandall Industrial Estate, Doncaster, S. Yorks DN3 1QR
The Rawlplug Co Ltd, Rawlplug House, 147 London Road, Kingston-upon-Thames, Surrey KT2 6NR
Spit Fixing Ltd, Unit D2, Old Brighton Road, Lowfield Heath, Crawley, W. Sussex RH11 0QN
Stainlessfix Ltd, Unit 2A, Hawley Trading Estate, Hawley Lane, Farnborough, Hants GU14 8EH

Thorsman and Co (UK) Ltd, Thor House, Yarrow Mill, Chorley, Lancs PR6 0LP
Thunder Screw Anchors Ltd, Unit 11, Industrial Estate, Southwater, Horsham, W. Sussex RH13 7HQ
Tucker Fasteners Ltd, Construction Industry Division, 177 Walsall Road, Birmingham B42 1BP

Appendix 2

Glossary

adherend
One part of a glued joint. A component which is connected to one or more components with an adhesive.

anchor
A device used to make an anchorage fixing.

anchorage fixing
A fixing device which makes a fixing to the base material by embedding in the material and not passing through it.

austempered steel
A specially hardened steel involving treatment with heat and quenching.

base
The major element of several parts, between which a fixing is made.

base-deforming fixing
A fixing which is driven or otherwise forced into the base material and achieves its holding power by friction with the base, or by cutting matching threads in it.

bi-metallic corrosion
Galvanic corrosion

blind fixing
A device which is capable of making a connection between two or more elements of a joint without needing access to the further side of the joint.

cam out
The action of a screwdriver, not completely perpendicular to a screw, flying out of the driving profile during screwing.

cavity fixing
A device to make a fixing to a cellular base, utilizing the cavity inside the base and not requiring access to that cavity to set the device.

chemical fixing
A fixing device which depends on the setting of an adhesive to achieve its holding power (cf. mechanical fixing p. 190).

clearance hole
A pre-drilled hole which is of greater diameter than a mechanical fixing which passes through it; often formed in the subsidiary component.

component
The smaller part of a joint – the subsidiary component (cf. base p. 188).

compression
A force which tends to compress a material or fixing; for instance, a force parallel to the shank of a nail which presses it into the base.

contact adhesive
An adhesive with high grab which is applied to both adherends and allowed to dry before the joint is closed. After closing, no adjustment of the parts of the joint is possible.

contact corrosion
Galvanic corrosion.

drilled-for fixing
A fixing which requires the prior drilling of a hole before it can be set.

electrolytic attack
Galvanic corrosion.

embedment
In an anchorage fixing, the depth the fixing is set into the base material to achieve the required pull-out strength.

fired fixing
A fixing which is driven into the base material by the explosion of a cartridge or the release of compressed air within the fixing tool (see powder-actuated and pneumatic fasteners pp. 46 and 50).

first movement load
The point at which a fixing under load is first observed to move 0.2 mm.

fixing accessories
Pieces of metalwork which form an integral part of a fixing device but require connecting to the base by an anchorage device; e.g. an angle corbel (the accessory) connected by an expanding anchor to the mass walling base.

galvanic corrosion
Corrosion resulting from one metal having electrical contact with another metal.

galvanizing
The coating of a ferrous element with zinc for corrosion protection by dipping the element in molten zinc.

gap-filling adhesive
An adhesive specially formulated with a proportion of inert fillers to join together adherends with slightly rough surfaces.

grab
The initial holding power of an adhesive when the joint is first closed.

gun-grouting
The process of injecting grout (often resin-based grout) into a mortice to bed and set a cast-in fixing.

holding power
The resistance of a fixing to pull-out failure.

imposed load
All loads placed on a structure, other than those imposed by the weight of the parts of the structure; e.g. the wind pressure on a wall panel, but not the weight of the panel itself.

lead hole
A hole of less diameter than the diameter of a screw (wood or self-tapping) into which it is driven (cf. clearance hole p. 189).

loadbearing fixing
A fixing which carries the weight of the component, as well as maintains its correct location.

mechanical fixing
A fixing that does not depend on the setting of an adhesive and is usually in the form of a metal (or occasionally plastic) device.

open time
The length of time after the spreading of an adhesive before the joint is closed; usually stated by manufacturers as maximum and minimum periods between which the glue will have effective adhesion.

plug
A screwable or nailable insert in mass walling.

popping
The phenomenon of nails working out under conditions of use, particularly when intermittent loading causes timber structure to deflect.

pull-out failure
Failure caused by an anchorage fixing, under load, pulling out of the base.

pull-out strength
The resistance of a fixing to pull-out failure.

pull-through failure
Failure caused by the subsidiary component, under load, being forced over the head of the fixing device.

relieved shank
Where the diameter of the shank of a bolt or screw is less than the crest of the thread.

restraint fixing
A fixing which maintains a component in its correct location, but is not required to carry its weight.

safety factor
The amount by which the ultimate load of a fixing is reduced to establish a safe working load (cf. ultimate load, p. 192 first movement load p. 189).

self-weight
The weight of the component without including any imposed load.

shank
The cylindrical part of a bolt or screw.

shear
A force acting at right angles to the axis of the fixing.

shelf life
The period during which an adhesive can be stored in its sealed container without danger of deterioration.

sherardizing
The diffusion of hot zinc dust on to a ferrous element to give it added corrosion resistance.

spalling
The fracture of mass walling, resulting in flaking and breaking away, around the point of entry of the fixing device.

subsidiary component
The minor part of an assembly, connected to the base by mechanical or chemical fixing.

substrate
In adhesive terminology, the major part of a connection – the base.

s.w.g.
Standard wire gauge. A method of measuring wire diameter, used to describe nail thickness; now rapidly being superseded by metric dimensions.

tension
A force applied to a fixing parallel to its axis and acting to pull the fixing away from the base.

thermoplastic adhesive
: An adhesive which softens with heat.

thermoset adhesive
: An adhesive which sets due to a chemical reaction brought about by heat.

thixotropic
: A thixotropic material is one whose normal consistency is that of a thick paste, which thins on stirring.

thread
: A spiral groove formed in or on the shank of a screw or bolt. The root of the thread is at the base of the groove; the crest at the external angle or apex between adjacent grooves.

through-fixing
: A fixing that is made by a mechanical device, such as a bolt, passing right through the joint, both subsidiary component and base.

ultimate load
: The load at which a fixing fails.

unrelieved shank
: The shank of a screw or bolt in which the diameter of the shank is the same as the diameter of the crest of the thread.

waisted bolt
: A bolt having its shank diameter reduced below the root diameter of the thread.

wet-stick adhesive
: An adhesive which is closed while the adhesive is wet, as opposed to a contact adhesive which is closed after the adhesive has dried to a degree.

working load
: The ultimate load of a fixing divided by a safety factor.

zinc plating
: The electroplating of a ferrous metal device with zinc for corrosion protection.

Appendix 3

Comparative diameters of various devices

(reprinted from *Mechanical Fixing Devices in the Building Industry* by Paul Marsh and Derrick Beckett, Construction Press 1975)

	Device							
	Bolts and Machine Screws					*Wood screws gauge*	*Nails (s.w.g.)*	*Self-tapping screws (A & B gauge)*
Diameter (mm)	*ISO Metric*	*BA*	*BSW*	*BSF*	*UNC/UNF*			
1.0							20 . 0.90	
							19 . 1.00	
							18 . 1.20	
							17 . 1.40	
	. M1.6					0 . 1.52	16 . 1.60	
						1 . 1.78	15 . 1.80	
2.0	. M2					2 . 2.08	14 . 2.00	
		8 . 2.18						2 . 2.18
						3 . 2.39	13 . 2.30	
		7 . 2.50					12 . 2.60	
		6 . 2.74			4 . 2.84	4 . 2.74	11 . 2.90	4 . 2.84
3.0	. M3					5 . 3.10		
		5 . 3.20	1/8 . 3.20				10 . 3.30	
		4 . 3.51			6 . 3.50	6 . 3.48		6 . 3.50
						7 . 3.70	9 . 3.70	7 . 3.83
4.0	. M4	3 . 4.08	5/32 . 4.00		8 . 4.16	8 . 4.17	8 . 4.10	8 . 4.16
						9 . 4.50	7 . 4.50	
		2 . 4.75						
5.0	. M5		3/16 . 4.80	3/16 . 4.80	10 . 4.83	10 . 4.88	6 . 4.88	10 . 4.83
		1 . 5.31					5 . 5.40	12 . 5.49
						12 . 5.59	3 . 5.60	
							3 . 5.90	
6.0	. M6	0 . 6.00	1/4 . 6.40	1/4 . 6.40	1/4 . 6.40	1/4. 6.30	2 . 6.30	14 . 6.15
							1 . 6.70	
						16 . 6.94	0 . 7.00	
7.0						18 . 7.72		
8.0	. M8		5/16 . 7.90	5/16 . 7.90	5/16 . 7.90			
						20 . 8.43		
9.0								
			3/8 . 9.50	3/8 . 9.50	3/8 . 9.50			
10.0	. M10							
11.0			7/16 . 11.1	7/16 . 11.1	7/16 . 11.1			
12.0	. M12							
			1/2 . 12.7	1/2 . 12.7	1/2 . 12.7			
13.0								
14.0								
			9/16 . 14.3	9/16 . 14.3	9/16 . 14.3			
15.0								
16.0	. M16		5/8 . 15.9	5/8 . 15.9	5/8 . 15.9			
17.0								
18.0								
19.0			3/4 . 19.1	3/4 . 19.1	3/4 . 19.1			
20.0	. M20							
21.0								
22.0								
			7/8 . 22.3	7/8 . 22.3	7/8 . 22.3			
		Gauge . Metric dia.	Imperial dia. . Metric dia.	Imperial dia. . Metric dia.	Imperial dia . Metric dia.	Gauge – . dia. in mm	SWG– . dia. in mm	Gauge – . dia. in mm

Index

adherend 162
adhesion 7
adhesive 7
aggregate size 65
anchorage fixing 6
anchorage strap 54
angle corbel 61

base 3
base-deforming fixing 6, 44, 70, 112
bi-metallic corrosion 9, 112
black bolt 115
blind fixing 122, 137, 146
blind rivet 139
bolt 6, 92, 113, 141
bonding agent 166
built-in fixing 7, 53
butterfly toggle 155

cantilever bracket 102
carriage bolt 93
cast-in channel 7, 59
cast-in fixing 7, 57
cavity fixing 146
cavity wall repair anchor 18, 33
ceiling suspension anchor 18, 32
chemical anchor 4, 7, 36
chemical fixing 4
chipboard screw 87
clutch-headed screw 85
coach bolt 93
coach screw 88
compression 5
concrete strength 64, 67
contact adhesive 162
contact corrosion 9, 78, 85
corbel 7, 60
corrugated fastener 91
countersunk head 85
cramp 57

dog 91
dowel 57
dowel screw 86
double-spiral thread 81, 84
drill 13
drilled-for fixing 6, 13, 123, 134, 146
drilling 13
driving profile 81, 84, 120

electric hammer drill 16
electric rotary drill 13
electric rotary impact drill 13
electric screwdriver 79, 124, 127, 129
electrolytic attack 9
expanding anchor 7, 18, 23, 61
expanding sleeve cavity fixing 147

fired fixing 7
first movement load 13
fixing accesories 61, 100
foam pads 165
frame anchor 27
framing anchor 101
friction grip bolt 116

galvanic corrosion 9, 10, 112, 145
galvanizing 9
gun-grade adhesive 164

hammer drill 13, 16
hammer-in anchor 51
hammer-set plug 21
hand rotary drill 13
headless screw 86
head style 81, 114, 120, 124
hot-dip galvanizing 9
hot-melt adhesive 167
HSFG bolt 116

impact adaptor 17
improved nail 73, 107

injection anchor 4, 39
in situ fixing 52
insulation fastener 18, 33

joist hanger 7, 53, 61, 100

LIB bolt 118
loadbearing fixing 4
load-indicating bolt 118
locking nut 119, 122

machine screw 112, 120
masonry nail 45
mechanical fixing 4
metric bolt 113
moisture movement 8, 93

nail 7, 70, 106
nailing machine 71
nail plate 7, 8, 88

oxidation 9, 78, 108, 112

Phillips head 4, 84
pH value of concrete 65
plug 7, 18, 19, 58, 150
plugging compound 18, 20, 23
pneumatic-actuated fixing 50, 134, 143
pneumatic hammer drill 16
pneumatic screwdriver 79
pneumatic tool 44, 50
powder-actuated fastener 46, 129, 143
powder-actuated tool 44, 47, 132
precision bolt 115
pull-out failure 4
pull-through failure 4

raised countersunk head 85
relieved shank 81
restraint fixing 4
rivet 6, 135, 139, 142, 153
rivet bush 135
rivet cavity fixing 135

safety factor 12
screw 7, 79, 106
screw-in anchor 41
Sela self-drilling screw 127
self-drilling expanding anchor 18, 30
self-drilling screw 126
self-locking tapped hole 137
self-tapping insert 140
self-tapping screw 123
set screw 122
shear 5, 142
shear failure 5
shear load 5, 13
shear plate 98
sheeting fastener 134
sherardizing 9
skew nailing 108
sliding brick anchor 62
socket 7, 58, 61
spall stop 49, 65
split ring connector 6, 7, 96
split-sleeve cavity fixing 149
staple 71, 106
strap 102
strength of timber 104
structural movement 8
subsidiary component 3
substrate 162
Supadriv 84
synthetic resin adhesive 161

Taptite 124
Teks self-drilling screw 127
tensile loading 4, 13
tension 4
thermal movement 8, 93
thermoset 161
thread 81
thread-forming screw 41
through fixing 6, 92, 112
timber connector 6, 94, 105
timber preservative 8, 108
toggle fixing 6, 154
toothed connector 6, 94
truss clip 102
Twinfast screw 84

ultimate load 12
umbrella cavity fixing 158
unrelieved shank 81
U-type drive fastener 125

waisted high strength bolt 119
wall tie 7, 55
wet-stick adhesive 163
woodscrew 79, 106
working load 12

zinc plating 9

As technological developments in the building industry have gathered pace, construction materials have become considerably stronger and lighter. Weight alone can no longer be relied upon to hold the parts of a building together. This trend has resulted in a new breed of fixings, fasteners and adhesives, a bewildering array of very strong, high performance fittings. It is absolutely vital that these are used with the same degree of care and precision that went into their design.
This book has been written for the site operative, the supervisor and the building apprentice. It is an up-to-date, practical and comprehensive guide to the safe and efficient application of both mechanical and chemical fittings. The author gives the reader the information needed to decide which fixing to use and how to use it to achieve its maximum effect. He also gives a list of proprietary fixings in base material classifications.

About the author
Paul Marsh is a qualified architect who spent many years working in the industry as an architect and director of a building company. Now a full-time writer on building subjects, his previous works include *Mechanical Fixing Devices in the Building Industry* and the *Illustrated Dictionary of Building.*

Site Practice Series
This series of books is aimed at all members of the construction team involved at site level. Written in straightforward style and designed to give quick access to information, each volume derives its authority from the practical up-to-date experience of its author.

General Editors: *Harold and Win Landsell*
A full list of titles in the series appears at the front of the book.

ISBN 0 86095 035 2

Construction Press

PRINCIPLES OF MACHINE OPERATION AND MAINTENANCE

DICK JEFFREY

PRINCIPLES OF MACHINE OPERATION AND MAINTENANCE

Dick Jeffrey

BUTTERWORTH HEINEMANN

OXFORD AMSTERDAM BOSTON LONDON NEW YORK PARIS

Butterworth-Heinemann
An imprint of Elsevier Science
Linacre House, Jordan Hill, Oxford OX2 8DP
200 Wheeler Road, Burlington, MA 01803

First published by Thomas Nelson Australia
Reprinted with amendments 1991
First published in Great Britain 1991
Reprinted 1993, 1997, 1998, 1999, 2001, 2003

British Library Cataloguing in Publication Data
A catalogue record for this book is available from the British Library

ISBN 0 7506 0293 7

For information on all Butterworth Heinemann publications visit our website at www.bh.com

Printed and bound in Great Britain by Martins the Printers, Berwick upon Tweed

C O N T E N T S

Chapter 1 **The maintenance function** 1
- Performing the maintenance function 1
- Trade skills 3
- Resources 4
- Organisation 4

Chapter 2 **Rotating machinery** 5
- Definitions 5
- Operating principles 6

Chapter 3 **Machinery mounting** 8
- Setting and levelling 8
- Grouting 10
- Special mountings 10

Chapter 4 **Balancing** 12
- Principles of balancing 12
- Static balancing 12
- Dynamic balancing 13

Chapter 5 **Lubrication** 15
- 5.1 Principles of lubrication 15
  - Friction 15
  - Wear 17
  - Theory of lubrication 17
  - Lubricant selection 18
  - Methods of application 19
- 5.2 Lubricants 21
  - Characteristics and properties 21
  - Liquid lubricants 22
  - Greases 22
  - Solid lubricants 23
  - Additives 24

Chapter 6 **Bearings** 25
- 6.1 Plain bearings 26
  - Principles of operation 26
  - Types and materials 26
  - Maintenance practices 28
  - Failure patterns 32
- 6.2 Rolling element bearings 37
  - Principles of operation 37
  - Types of bearings 38
  - Bearing assembly design 40
  - Maintenance practices 41
  - Failure patterns 48

Chapter 7 **Power transmission** 54
- 7.1 V-belt drives 54
  - Principles of operation 54
  - Types and arrangements 54
  - Maintenance practices 56
  - Failure patterns 58
- 7.2 Chain drives 61
  - Principles of operation 61
  - Types and arrangements 62
  - Maintenance practices 63
  - Failure patterns 65
- 7.3 Gear drives 68
  - Principles of operation 68
  - Types and arrangements 69
  - Maintenance practices 70
  - Failure patterns 71
- 7.4 Shaft couplings 77
  - Principles of operation 77
  - Types 77
  - Maintenance practices 79
  - Failure patterns 80
- 7.5 Clutches 82
  - Principles of operation 82
  - Types and applications 83
  - Maintenance practices 85
  - Failure patterns 86

Chapter 8 **Shaft alignment** 88

Principles of alignment 88
Methods of alignment 90

Chapter 9 **Seals** 96

9.1 **Gaskets** 97
Sealing principles 97
Gasket materials 98
Maintenance practices 100
9.2 **O-rings** 102
Sealing principles 102
Maintenance practices 102
Failure patterns 103
9.3 **Sealants** 104
Sealing principles 104
Materials 104
Maintenance practices 104
9.4 **Mechanical end face seals** 105
Principles of operation 105
Materials 106
Types and arrangements 106
Environmental control 109
Maintenance practices 109
Failure patterns 111
9.5 **Packings** 115
Sealing principles 115
Materials 116
Maintenance practices 116
9.6 **Lip seals** 118
Sealing principles 118
Types and materials 118
Maintenance practices 119
Failure patterns 119
9.7 **Circumferential split-ring seals** 12[illegible]
Sealing principles 12[illegible]
Types and arrangements 12[illegible]
Maintenance practices 12[illegible]
9.8 **Felt seals** 12[illegible]
Sealing principles 12[illegible]
Maintenance practices 12[illegible]
9.9 **Clearance seals** 12[illegible]
Bushings 12[illegible]
Labyrinth seals 12[illegible]

Chapter 10 **Condition monitoring** 12[illegible]

Methods 12[illegible]

Chapter 11 **Troubleshooting** 12[illegible]

Approach to troubleshooting 12[illegible]
Aids to troubleshooting 13[illegible]
Investigation guidelines 13[illegible]
Communication 13[illegible]
Systematic diagnosis 13[illegible]

Chapter 12 **Attitudes towards maintenance** 13[illegible]

Chapter 13 **Safety** 13[illegible]

Safety-related incidents 13[illegible]
Safe working practice 13[illegible]
Hazardous materials 14[illegible]

Index 14[illegible]

Preface

Principles of Machine Operation and Maintenance has been designed both as a training manual for maintenance tradespeople and technicians and as a reference handbook for engineers. It sets out to achieve three things:

- to establish and describe the basic principles of operation and maintenance of rotating machinery.
- to create a clear understanding of the maintenance function and the role of the technician.
- to emphasise the importance of a positive mental approach to maintenance and the need to develop troubleshooting skills as well as practical skills.

The book assumes that the reader will have already acquired a basic understanding of engineering terms and practices such as would be gained during the first stages of a fitting and machining apprenticeship.

The illustrations provided have been kept deliberately simple and schematic in nature so that basic principles rather than the precise details of machine element construction can be emphasised.

The underlying aim of the book has been to promote the principles of good engineering practice and it is hoped that it will provide a basis of understanding on which experience can be assimilated.

Thanks are extended to Mobil Oil Australia Ltd, SKF Australia Ltd, Glacier Metal Company Ltd, the Gates Rubber Company, Denver, Colorado, Rex Chainbelt Inc, Milwaukee, Caterpillar of Australia Limited, Renold Australia P/L, Wetherby Engineering Co. Ltd (Broadbent Drives), Bradford, UK, Stieber Ltd, Letchworth, UK, and Flexibox P/L for the use of illustrative material. I should also like to thank Ruth Siems for her expert editing and her sympathetic and patient treatment of a novice writer and Sandra McComb for her encouragement and faith in the project.

Dick Jeffrey
October 1984

CHAPTER 1

THE MAINTENANCE FUNCTION

The function of maintenance is to ensure that plant and equipment are available in a satisfactory condition for operation when required. The determination of what constitutes a 'satisfactory condition' for rotating machinery will depend largely on the operating situation and the following considerations:

Type of industry The nature of the industry concerned will determine, to some extent, the level of machine performance required.

Process requirements The function of the machine in the process will also determine performance requirements. The requirements of a machine that is critical to the process will be greater than for one that is 'non-critical'.

Business objectives Product demand and profit levels may give rise to fluctuating demands on equipment and performance requirements.

In all cases, however, the performance of the maintenance function can be judged by the condition of machinery as indicated by the following factors:

Performance Machines must be capable of performing the function for which they are intended.

Downtime Machines must operate with an acceptable level of downtime.

Service life Machines must provide a satisfactory return on investment before replacement becomes necessary.

Efficiency Machines must operate at an acceptable level of efficiency.

Safety Machines must operate safely and not be dangerous to personnel.

Environmental impact Machines must operate in a manner that is not detrimental to the environment or to adjacent plant and equipment.

Cost The cost of maintenance must be acceptable.

The goal of maintenance is to ensure that machinery performance is 'satisfactory' according to these seven factors and, although the specific requirements of an individual machine are rarely quantified, it is important that the criteria by which performance can be assessed are understood and monitored. Despite the fact that definite levels of acceptibility are hard to establish, trends in machine conditions can be observed and should be used as indicators of maintenance requirements.

PERFORMING THE MAINTENANCE FUNCTION

The maintenance function is performed in two ways:

- by the prevention of breakdown, and
- by the repair of breakdown.

The term, breakdown, is used here to indicate any machine condition that is considered to be less than satisfactory according to the seven factors listed above. This covers all situations from the need for minor adjustment to total machine collapse.

Although performance of the maintenance function has been separated here into 'prevention' and 'repair', the element of prevention is the key to all successful maintenance work, including repair. Preventative maintenance need not be treated as a special program because prevention should be the underlying theme of all maintenance work.

Rather than investigating repair and prevention as two different aspects, it is more useful to consider the various ways in which maintenance work arises and to recognise the preventative element in each.

Machine breakdown

Despite all attempts at prevention, machine breakdowns of various kinds do occur and often need to be fixed on an urgent or emergency basis. Although maintenance personnel are usually under great pressure to return equipment to service in the shortest possible time, there are two important considerations, vital to the 'preventative' element of the work, that should not be sacrificed.

Treatment of cause — not effect

It is important to make sure that the real cause of the breakdown is found and remedied and not just the effect patched up. Troubleshooting should be rigorous in finding the inherent cause of the problem. If the real cause of the problem is not corrected then further breakdown is likely to occur.

Preventative solutions

Restoration of equipment to its original condition is not necessarily the best solution to a breakdown problem. Consideration should be given to avoiding repetition of the problem by making minor changes in design or materials. Equipment performance can often be improved with little expense by the development of innovative solutions that are specially suited to local conditions and process requirements.

Routine work

Some basic preventative work can be easily identified for execution on a routine basis. This work will typically include such tasks as:

- lubrication
- adjustment
- cleaning

and the need for such work will usually be determined on the basis of:

- field experience
- manufacturer's recommendations
- production requirements

Routine work should, where possible, be carried out during machine operation. If this is not possible it may be scheduled to coincide with regular production shutdowns. In any event, this type of work should not normally interfere with production schedules.

Planned work

This category includes any work that requires an extended equipment shutdown or that needs to be planned and scheduled in advance.

All planned work should be considered as preventative and is essentially aimed at the avoidance of unscheduled breakdowns. The type of work involved will include:

- major overhaul
- replacement of specific components
- machine modification

It may originate from inspection, condition monitoring, manufacturer's recommendations, plant experience, statutory regulations or design modification.

Inspection

The condition of all machinery should be under continual surveillance by both operating and maintenance personnel. The casual and routine monitoring of equipment will yield information regarding operating condition on which maintenance requirements can be planned.

It is vital that maintenance personnel realise the importance of being critically aware of the operating condition of machinery and ensure that the observations are accurately reported. Inspection requires the use of the senses and maintenance personnel should develop an **eye** and an **ear**, and even a **nose**, for machine condition. Recognition of normal running characteristics are the basis from which deviations can be observed and trends in machine condition can be predicted.

Condition monitoring

In recent years a variety of techniques have been developed by which the operating condition of machinery can be either intermittently or continuously monitored. These techniques are a mechanised version of inspection by personnel and all operate on the same principle of observed deviation from normal condition.

The most important of these techniques is **vibration monitoring** and this and other methods are outlined in Chapter 10. Condition monitoring like personal inspection, yields information on which maintenance requirements can be based.

Manufacturers' recommendations

Most equipment manufacturers provide details of recommended maintenance requirements, from basic lubrication schedules to major overhaul information. Until plant experience indicates otherwise, it is wise to follow these recommendations in the early stages of operation. This information provides an initial basis on which to determine preventative work, such as overhaul and routine replacement of components, to be carried out during annual or other planned shutdowns.

Plant experience

Local knowledge of machine performance under specific plant conditions may well, in the longer term, prove to be a more appropriate method of establishing the frequency of major overhaul and other maintenance requirements. For such experience to be used effectively it is important that accurate maintenance records are kept so that performance patterns and characteristics can be clearly established.

Component life expectancy and wear rates can only be assessed on the basis of recorded information that represents a true reflection of operating conditions. The use of condition monitoring techniques provides the kind of detailed information on which maintenance records can be based.

Statutory regulations

Inspection and testing of certain types of equipment, especially pressure vessels, is normally carried out on an annual basis according to statutory regulations. In most cases this involves a total plant shutdown and is an opportunity to undertake an annual 'turn around'. A certain amount of work is generated during this period due to the inspection program itself as determined by the regulations, and the remainder originates in the ways previously described. Compliance with statutory regulations constitutes, in effect, an externally imposed prevention program.

Design modifications

When the performance of a machine is unsatisfactory according to any of the seven factors listed on page 1, then design modification may be undertaken in order to improve performance.

Modifications will normally be undertaken by engineering staff who support maintenance personnel. Unless the machine has already suffered major breakdown, in which case modification may be undertaken on an emergency basis, the work involved would normally be scheduled for a planned shutdown period.

No matter what the source of the work, it should be clear from the above descriptions that the common element is the prevention of future breakdown. Although no maintenance department can ever be one hundred per cent successful in this, it should be recognised that the general objective of **breakdown prevention** helps to encourage an attitude towards maintenance work that is most likely to give good results.

TRADE SKILLS

Whatever the philosophical approach and organisational methods adopted, it is the maintenance technician, the person who actually performs the work, who is the focal point of the maintenance operation. The technician constitutes the key element in the operation around whom other resources must be organised. In order to fulfil this critical function technicians must possess certain skills, both practical and theoretical.

Practical skills

A maintenance technician should have completed an apprenticeship in fitting and machining (or the local equivalent). Where elective components are available in the apprenticeship program, options that concentrate on fitting, and particularly maintenance fitting, should obviously be selected where possible. This training should provide all the basic practical skills needed to undertake maintenance work. Such skills will include all basic manipulative tasks associated with fitting and machining including the use of appropriate tools and equipment. The level of competence and the range of skills involved will normally be determined by the relevant regulatory body (Industrial Training Commission or otherwise).

The skills taught in an apprenticeship program represent the minimum requirement and the practical skills of maintenance technicians should be further developed on the job to meet specific plant requirements and to keep pace with technological development.

Theoretical skills

In addition to the theoretical background provided to complement the practical skills acquired during an apprenticeship, technicians should possess an understanding of the operating principles of basic machine elements and common types of machinery as well as the principles and practices of maintenance associated with them. The theory component of an apprenticeship is normally too basic and general in nature to provide an adequate background for maintenance work. In order to be fully equipped to troubleshoot and to make effective repairs, the technician must have a clear understanding of the function and principles of operation of the machinery involved.

These theoretical skills, specifically related to rotating machinery, are the primary focus of this book.

Mental attitude

Although it is often neglected, one of the most important skills that a technician can possess is a good mental attitude towards the maintenance task. The most obvious example of this is the way in which a careless and casual approach is likely to lead to error and bad workmanship. The adoption of a positive mental attitude toward the task not only has the potential to improve performance but also leads to increased personal satisfaction. Some of the ways in which mental attitude can be improved are discussed in Chapter 12.

Troubleshooting skills

A vital skill in all aspects of maintenance work that assumes even greater importance when the technician takes on a supervisory role is the ability to troubleshoot and diagnose faults. Like most skills, troubleshooting ability can be learned. It is, in a sense, the key element in the technician's resources drawing together practical and theoretical skills to produce the most effective performance.

Troubleshooting skills are largely an extension of certain aspects of mental attitude and are discussed in detail in Chapter 11.

RESOURCES

To complement the technician's personal skills, there is a range of additional resources that should be available within the maintenance department. The responsibility for organising these resources so that they are available when and where required lies with management, but it is the responsibility of the technician to appreciate their nature and to understand how they can be most effectively utilised.

The exact nature of these resources will vary from one organisation to another but the following list identifies the principal resources that should be available.

Tools	Standard hand tools
	Specialised tools
	Workshop facilities
Materials	Spare parts
	Stock materials
	Standard consumables
	Salvaged stock
Expertise	Line supervision
	Staff engineers
	Consultants
Systems data	Plant records
	Engineering drawings
	Process flow diagrams
	Manufacturers' information
	Technical publications
Planning	Job-recording systems
	Scheduling services
	Plant-delivery systems
	Communications systems

ORGANISATION

The maintenance department should be organi in such a way that all resources, including th skills, are available when required at the workfa that the maintenance function can be effectively efficiently performed. Various organisational mo can be adopted, depending on the size and na of the production activity. It is strongly recommen that technicians clearly understand how their organisation functions so that they can use information to maximum advantage. In order to things done it is important that the various lines authority and responsibility are fully understood.

C H A P T E R 2

ROTATING MACHINERY

DEFINITIONS

A rotating machine is one in which the main working components rotate about a fixed centre in a regular manner. Most such machines incorporate additional subsidiary mechanisms such as linkages, slides, gears and reciprocating components, and many of the operating principles that apply to the rotating assembly also apply to these other elements.

Although there are many different types of rotating machines, they can all be classified into three basic groups in terms of their function.

Driving machines (engines or prime movers)

This group includes all machines whose purpose is to drive other machines. Examples include:

- electric motors
- steam turbines
- diesel engines
- petrol engines
- air motors

The common characteristic of these machines is that they convert an energy input of varying kinds into a mechanical output in the form of a rotating drive shaft.

Transmission machines

These are machines whose purpose is to transmit mechanical energy from a driving to a driven machine. Examples include:

- gearboxes
- differentials
- variable speed drives

The mechanical energy transmitted often undergoes a speed transformation and these machines often incorporate some means of drive disengagement such as a clutch.

Driven machines

These machines cannot operate independently and need to be coupled to a driving machine. Examples include:

- pumps
- compressors
- fans
- generators
- blenders
- machine tools

This group is by far the largest and includes a large number of different types of machines. The common characteristic is that the energy input is normally in the form of a rotating drive shaft while output may be in a variety of forms including kinetic or pressure energy of a fluid, electrical energy, kinetic or potential energy of solid materials, etc.

Because of the ways in which different types of machines are sometimes combined into sets, the distinction between the separate elements is not always clear. For example, some pumping units are directly coupled through a reduction gear to an electric motor with all three elements contained in a single housing. Although such an item may be treated as a single machine it does, in fact, contain all three elements of driving, transmission and driven.

From a maintenance viewpoint, it is important that the function of a machine, or the elements of a machine set, is clearly understood. Troubleshooting and fault correction depend on an ability to detect deviations from normal operation, and the assessment of operating conditions demands a knowledge of the function of a machine as well as the principles on which it operates. It is recommended that technicians make sure that they are fully familiar with the function (i.e. what the machine 'does') of the particular machinery with which they are concerned.

OPERATING PRINCIPLES

A prerequisite of good maintenance practice is a critical understanding of the principles on which the satisfactory operation of rotating machinery is based. This understanding provides the foundation of the maintenance technician's ability to diagnose and correct faults.

Although each machine is different in some way, and each engineering situation gives rise to special requirements, there are a number of conditions that are critical to the operation of all rotating machines. In Chapter 1, seven criteria were identified that can be used to determine the operating condition of a machine. They were:

Performance
Downtime
Service life
Efficiency
Safety
Environmental impact
Maintenance cost

If a machine is to perform 'satisfactorily' according to these criteria, then the following conditions must be satisfied.

Mounting

A rotating machine must be correctly mounted on a suitable foundation. If this condition is not met then operation of the machine may cause damage both to itself and to adjacent equipment. The methods by which a machine should be mounted on a foundation are covered in Chapter 3.

Mechanism

Every rotating machine must have an internal mechanism that is operable, in good repair and capable of achieving the performance required. There are many excellent books that describe the operation and maintenance of specific types of machines and the technician should consult these for information when necessary.

Balancing

The rotating components or assemblies of a rotating machine must be correctly balanced. Assemblies that are not balanced cause excessive vibration and high stresses. Not only does this condition cause rapid wear and frequent breakdown but sudden and dramatic failure may occur with dangerous consequences.

Correct balancing is essential to the safe, reliable operation of any rotating machine and is covered in Chapter 4.

Lubrication

A rotating machine, like any mechanism where relative motion of contacting parts is involved, cannot operate satisfactorily unless it is lubricated to reduce friction and wear. The principles of lubrication and the characteristics of lubricants are explained in Chapter 5.

Bearings

Every item of rotating machinery requires a set of properly maintained bearings that support the mechanism and restrain its motion with minimum resistance. Bearings are an essential component and a common element in all rotating machines. The principles of operation and maintenance of bearings are covered in Chapter 6.

Transmission

Because machines normally operate in 'sets', i.e. driving-driven or driving-transmission-driven, some means of connecting the input and output shafts of the separate elements is required. This may be accomplished in several ways and the principles of operation and maintenance of the most common of these are described in Chapter 7.

Alignment

When machines are assembled in sets as described above, it is vital that the inter-connecting shafts are properly aligned to each other. Poor alignment will cause vibration and lead to rapid wear of couplings, bearings, seals and other rotating elements. The principles of alignment and methods by which good alignment can be achieved are explained in Chapter 8.

Seals

Rotating machines usually contain a number of fluids such as process fluids, fuels, cooling water and lubricants. At the very least there will always be a lubricant present. The escape of any of these fluids must be prevented to avoid waste and the creation of hazards to personnel and the environment. Hence all machine joints and connections must be properly sealed.

As well as containing fluids within the machine, sealing also serves to prevent contamination from external sources such as dirt and moisture.

The principles of operation and maintenance of common types of seals are explained in Chapter 9.

Guards

In order to prevent injury to personnel, the minimum safety requirement on any rotating machine is that all exposed rotating elements should be guarded during operation. Guarding should be designed and installed in such a way that accidental interference with rotating elements is impossible. The construction of machine guards is usually straightforward and the maintenance technician should ensure that they are always secure and in good repair.

No matter what specific type of rotating machine is considered, these nine conditions represent the minimum requirement for satisfactory operation according to the criteria listed in Chapter 1. If proper attention is given to these conditions then there is every chance that satisfactory operation will be achieved.

CHAPTER

MACHINERY MOUNTING

Rotating machines are frequently required to operate in 'sets' e.g. driver-driven, and therefore require a common bedplate on which the separate units can be mounted. This common bedplate provides a rigid, level structure which enables the machines to be maintained in alignment during operation. Independent machines that are sufficiently rigid and are not required to align with other equipment may be mounted directly onto a foundation.

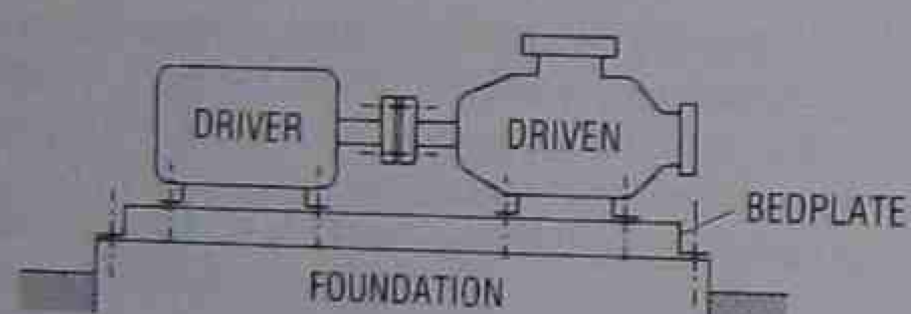


Fig. 3-1 A typical mounting arrangement for rotating machines.

It is important to realise that the mounting of the bedplate onto the foundation is critical to the operation of the machinery. If the bedplate is to provide a suitable base on which satisfactory machine operation can be achieved, then it must be level and firmly secured to the foundation without distortion. One of the functions of the foundation is to absorb machine vibrations caused by any unbalanced or inertia forces that may be present. If there is any relative movement between the bedplate and the foundation then any vibrations present, instead of being absorbed by the foundation, will tend to further loosen the bedplate mounting and cause damage to the machine. If the bedplate is not pulled down evenly and without distortion onto the foundation, then alignment of the individual machines will be difficult and casing distortion will occur when they, in turn, are pulled down onto the bedplate. This will eventually lead to rapid wear and damage to machine elements during operation.

SETTING AND LEVELLING

The setting and levelling process ensures that the mounting points for the individual machines are true and level. The bedplate should be provided with machined flats on which the machines are mounted and these should be taken as reference points in the levelling process. If level itself is not highly critical then any flat horizontal surface on the bedplate can be used. For most machinery, level can be adequately determined by using an engineers' spirit level, but for large equipment, or where special accuracy is required, then a surveyors' level may be used.

The bedplate should be mounted on shims that can be adjusted during the levelling process. Shim material should preferably be corrosion resistant and of sufficient proportions to support the weight of the machine. If the shims are crushed under the load of the machine then level and alignment will be destroyed. Shims may be either flat material or wedge shaped and should always be placed at the anchor bolt locations where the weight of the machinery is concentrated. When flat shim material is used it is recommended that it is shaped to fit around the anchor bolts.

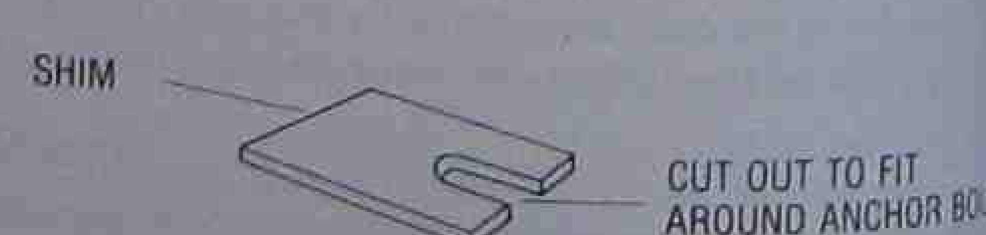


Fig. 3-2 A flat shim shaped to fit around the anchor bolt.

Wedge-shaped shims provide an easy method of adjustment but care must be taken to ensure that they are installed correctly so that they are aligned with the bedplate and fully support the load. Fig. 3-3 shows the right and wrong ways of using wedge shaped shims.

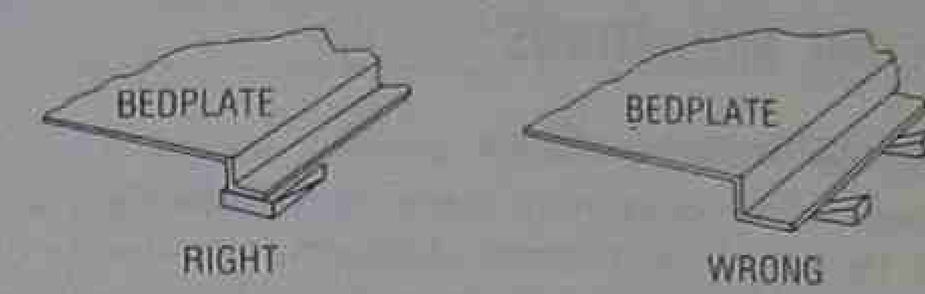


Fig. 3-3 Using wedge-shaped shims.

In order to provide some means of determining whether or not the bedplate is pulled down evenly and without distortion, it is recommended that a dial indicator is set up across the coupling, in the same way as for coupling alignment.

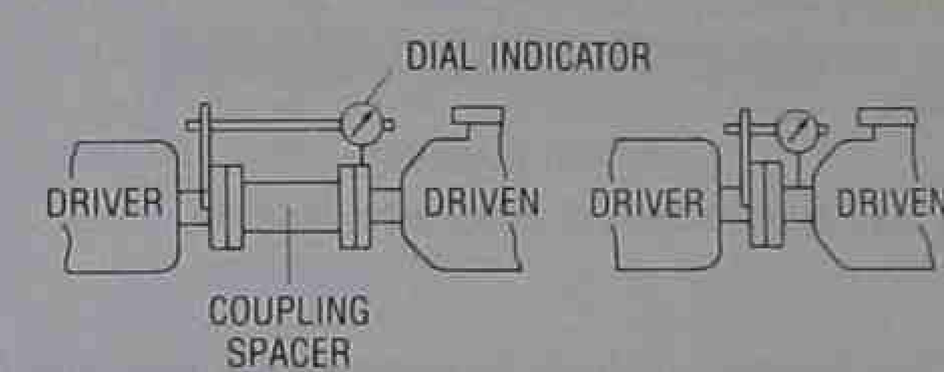


Fig. 3-4 A dial indicator set up across a coupling.

The dial indicator will allow the relative position of the machine shafts to be compared before and after the bedplate anchor bolts are pulled down. If there is a significant change in that relative position then this is an indication that the bedplate was distorted when the bolts were tightened. To overcome this problem it will be necessary to slacken the anchor bolts one at a time and use feeler gauges to determine which mounting requires extra shims under the bedplate to prevent the distortion.

Procedure for setting and levelling

1 Ensure that the foundation is clean and free of debris and that anchor bolts are correctly positioned.

2 Lower the bedplate, with the machines installed, onto the foundation and install preliminary shims under the bedplate and adjacent to each anchor bolt. The thickness of the shimpack used will vary depending on the size and weight of the machine but should range from a minimum of about 2cm (¾") up to 5cm (2") for large machines. Remember the shims have to carry the full weight of the loaded machine.

3 If the baseplate has machined flats to assist levelling, then mount a spirit level on the flats and check for level in both directions.

If no such levelling points are provided then select a clean, flat surface on the bedplate on which to mount the level. If levelling is critical and no machined flats are provided then it may be necessary to remove one of the machines and use the machined mounting points as a reference for levelling.

4 Adjust the shims under the anchor points until the bedplate is level in both directions. To save time and to avoid trial and error, the adjustment required can be calculated as follows:

(i) Use feeler gauges to determine the amount of adjustment required to bring the spirit level into the true horizontal position.

Fig. 3-5 Using a feeler gauge to determine adjustments.

(ii) Determine the ratio of the length of the bedplate to the length of the spirit level as shown in Fig. 3-6.

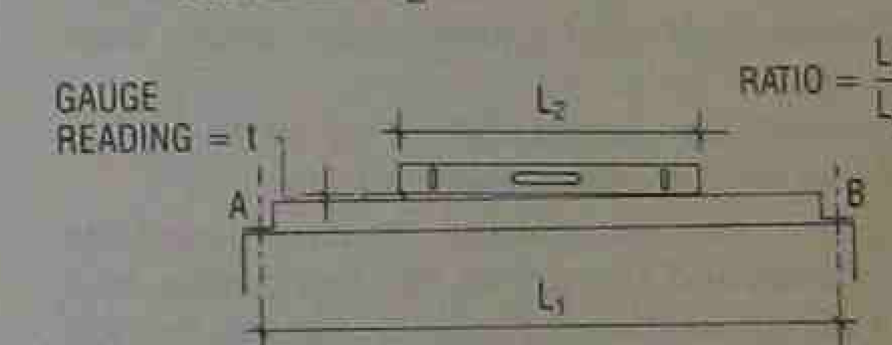


FIG. 3-6 Determining the ratio.

(iii) Multiply the ratio by the feeler gauge measurement. This will give a figure for the adjustment required at the mounting point.

$$\text{adjustment at A} = t \times \frac{L_1}{L_2}$$

5 Tighten the anchor bolts and recheck the level. Further adjust the shims if necessary.

6 Set up a dial indicator across the machine

coupling as shown previously in Fig. 3-4. With the anchor bolts loose, set the indicator to zero at top dead centre, rotate the machine and record the indicator readings at 90° intervals.

7 Tighten the anchor bolts and again, with the indicator zeroed at top dead centre, take readings at 90° intervals by rotating the machine.

8 Compare the two sets of readings. If a deflection of more than 0.050mm (0.002") in shaft position is measured, then some distortion of the bedplate is indicated.

9 Loosen the anchor bolts one at a time and use a feeler gauge to establish whether clearance exists between the shims and the bedplate.

10 Adjust the shims and recheck as before until the deflection recorded at the coupling is acceptable.

11 When level and deflection are satisfactory, tighten the anchor bolts making sure that all shims are correctly positioned.

The use of dial indicators is explained in Chapter 8. The arrangement used to support the dial indicator during the levelling process will also be useful when shaft alignment is carried out.

GROUTING

Once the setting and levelling procedure is complete and the anchor bolts have been tightened, the space between the bedplate and the foundation should be filled with grout, a fluid mixture of mortar-like concrete which expands when it sets. The function of grout is as follows:

- To secure the shims and ensure that they cannot be dislodged. It also protects the shims from corrosion.
- To prevent the accumulation of corrosive or hazardous liquids inside the bedplate.

Before grout can be poured, a dam must be constructed around the top of the foundation to a height of about 10-12mm (⅜" - ½") above the top surface of the shims.

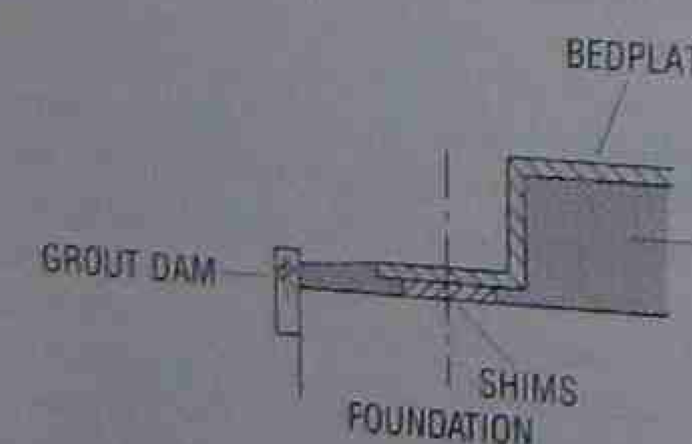


Fig. 3-7 Constructing a grout dam.

The grout is then poured inside the bedplate, thoroughly puddled to ensure that it is properly distributed and that there are no voids. Grout should be allowed to completely fill the bedplate so that there are no spaces left where liquid may accumulate.

SPECIAL MOUNTINGS

Adjustable mountings

In cases where machinery items are free-standing or may be subject to frequent changeover or change of position, anchor bolts may be replaced by adjustable mountings that provide a quick and easy means of levelling.

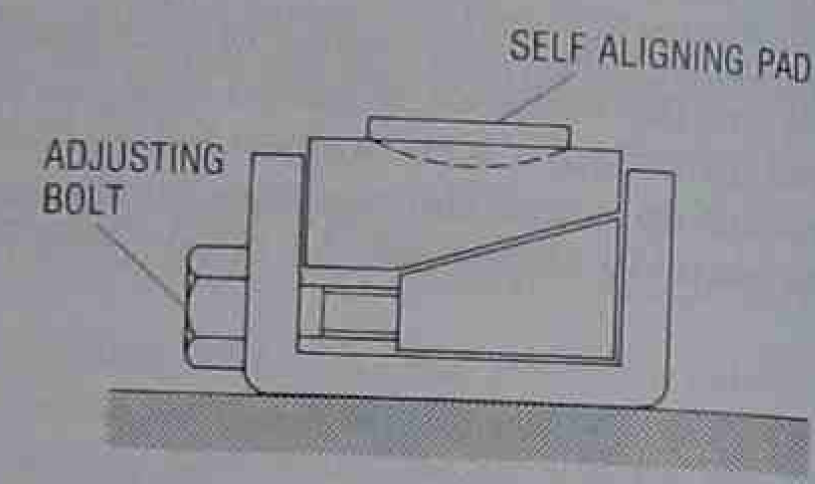


Fig. 3-8 A typical example of an adjustable mounting.

The levelling features incorporated in this type of mounting enable a machine to be levelled to precise limits in a minimum amount of time. Machines supported in this way would normally need to be equipped with flexible service lines for power, air, water, etc.

Before attempting to use this type of mounting it is important to ensure that it is clean and greased so that the necessary adjustments can be made and so that the mounting doesn't jam. It is also important to use mountings that are designed to carry the load of the machine.

Vibration isolators

An important consideration in many machinery installations are the problems of vibration, noise and shock. To overcome these problems various types of special mountings are available. The simplest solution, if these problems are not too severe, is to insert pads of absorbent material, such as felt or rubber, between the bedplate and the foundation. In order to distribute the load the material should be sandwiched between two steel plates as shown in Fig. 3-9.

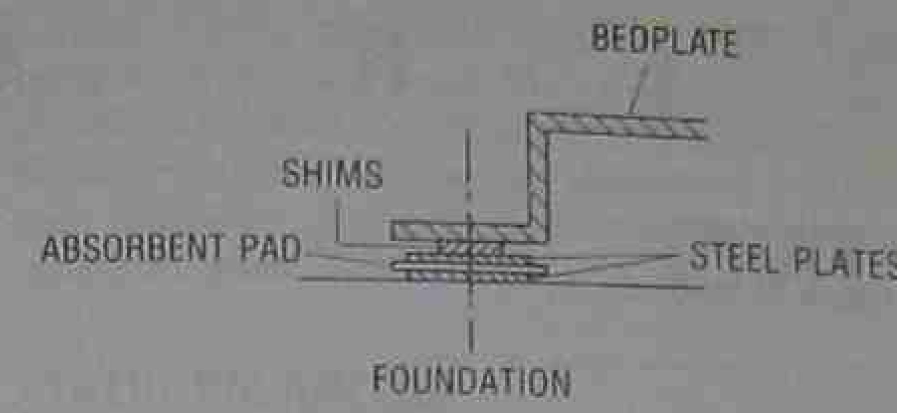


Fig. 3-9 Inserting absorbent material between bedplate and foundation

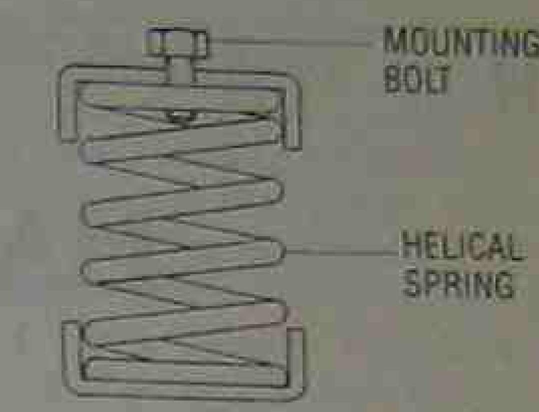


Fig. 3-10 A simple spring mounting

Felt and other materials are available in different grades and thicknesses for different loading pressures. Where the weight of a machine is not evenly distributed over the bedplate it may be necessary to use different grades of material at different mounting points.

Where vibrations are more severe, a significant degree of deflection may be required if they are to be absorbed by the mountings. In such cases a spring type of mounting is preferred.

Various devices of this type are available ranging from a simple spring mounting of the type shown in Fig. 3-10 to a more sophisticated type which incorporates a self-damping feature.

Spring mountings normally provide a means of level adjustment which eliminates the need for shims and may also include padding to reduce noise transmission.

Vibration isolators of this type serve the dual function of preventing machine vibrations from being transmitted to other adjacent equipment and preventing the amplification of internal stresses which cause misalignment and wear of machine elements.

C H A P T E R

BALANCING

PRINCIPLES OF BALANCING

If the centre of gravity of a rotating machine element does not coincide with its centre of rotation, then the machine is said to be unbalanced. When the machine is stationary, the off-centre mass causes the machine element to settle in a fixed position. (Fig. 4-1)

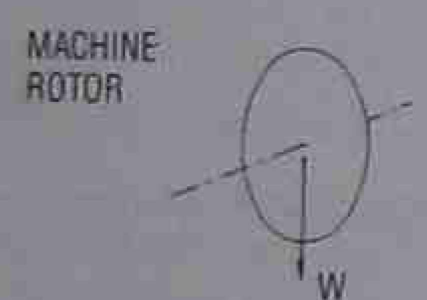


Fig. 4-1 Stationary machine with an off-centre mass.

As the machine element rotates, a centrifugal force associated with the off-centre mass develops and imposes a fluctuating load on the shaft support bearings as shown in Fig. 4-2.

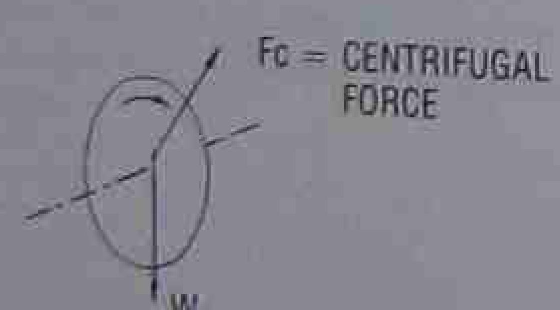


Fig. 4-2 Rotating machine with an off-centre mass.

The size of this force depends not only on the mass of the rotating element, but also on the extent to which the mass is off-centre and the speed of rotation. The resulting load imposed on the bearings cycles continuously through 360° with every rotation of the shaft. In addition to imposing high loads and fa stresses on the bearings, a fluctuating load of type will set up vibrations that will be transm through the machine and the surrounding struc

Balancing is the term used to refer to the proce improving the mass distribution of a rota machine element, so that it rotates in its bear without giving rise to unbalanced centrifugal for A machine element in which the centre of grav and the centre of rotation coincide will settle in a position when stationary, and when rotating will impose any additional loads on the bearings du unbalanced centrifugal forces. The only (radial) l imposed on the bearings should be due to the wei of the rotating element.

In practice, it is impossible to achieve perf balance and, even after sophisticated balanc techniques have been used, a rotating element w always possess some residual imbalance. The ne for accurate balancing increases with the mass a size of the rotor and the speed of rotation.

To ensure safe and reliable operation of a rotating machine it is desirable that the rotat elements be balanced within defined lim according to the size and nature of the machine. I common practice these days to monitor machi vibrations as a means of determining standard machine balance and the technique involved discussed in Chapter 10.

The mass distribution of a machine element can changed by either adding or subtracting mass at a position on the rotor. Mass may be added welding, bolting or otherwise attaching additio material to the rotor, usually in measured quantiti Mass may be removed either by drilling a hole in t rotor or by grinding or filing material from t surface. The key to successful balancing lies in t method used to determine the position at which m is to be added or removed. In most cases it preferable to remove material from the heavy side a rotor, rather than add material to the light side because this overcomes the problem of having to provide a means of attaching the additional material. However, there may be circumstances where this is not possible because of strength or visual considerations.

STATIC BALANCING

The simplest and easiest method of balancing is one that uses static conditions to determine the relative position of the centre of gravity and centre of rotation. This method is relatively straightforward but it is strictly limited to machine elements with only one plane of correction, such as circular saw blades and other 'thin' rotors.

The procedure involves setting up the rotor so that it can rotate freely and settle in its equilibrium position. This usually requires the rotor to be removed from the machine and balanced on knife edges as shown in Fig. 4-3.

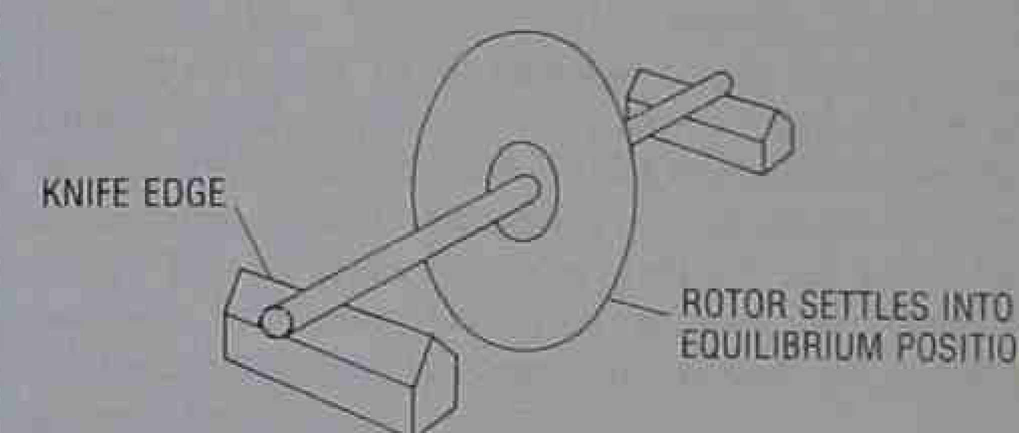


Fig. 4-3 Preparing the rotor for static balancing.

It is vital that, whatever means are used to support the rotor, there is no resistance to the rotor finding its true equilibrium position. If there is any friction present this may cause the rotor to stick and lead to mass being added or removed in the wrong position. The amount of material to be added or removed is largely determined by trial and error, although observation of the rotor as it settles to its equilibrium position will help to determine the degree of imbalance. It is necessary to check after each adjustment until satisfactory balance is achieved.

It is recommended that the rotor is first balanced by adding a lump of plasticene or similar substance at the required location. When balance is achieved this location is marked and the plasticene can be weighed. An equal amount of metal can then be added in the same position, or removed from a position 180° opposite.

DYNAMIC BALANCING

Where a rotor has more than one plane of correction, i.e., anything other than a 'thin' rotor, balancing must be carried out dynamically. It is normal practice for the machine rotor to be removed and temporarily installed on a special balancing machine. The machine is then run up to a suitable speed and the out-of-balance forces measured.

The principle of dynamic balancing is based on the measurement of the rotating couples that are set up as a result of the out-of-balance forces. Because most rotating machine elements have their mass distributed over some axial distance, the problem of determining the correct position for mass compensation becomes more complicated. Fig. 4-4 shows the twisting effect on the support bearings of the centrifugal force associated with the off-centre mass distribution.

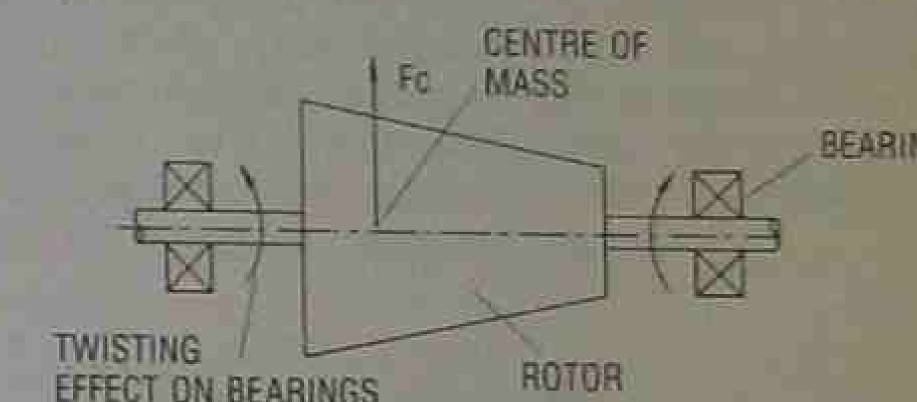


Fig. 4-4 Twisting effect associated with off-centre mass distribution in a machine rotor.

It would be a simple matter to determine the magnitude of the necessary compensatory mass by the static balancing method described above. However, the difficulty lies in determining the axial position for the mass to be located. If this position is chosen incorrectly then out-of-balance forces will continue to exist, as shown in Fig. 4-5.

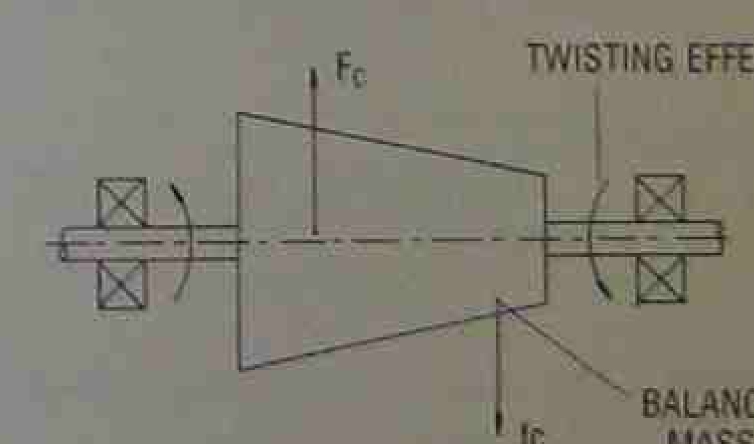


Fig. 4-5 Compensatory mass in the incorrect axial position.

Although the rotor may now be statically balanced, when it is rotated the centrifugal forces associated with the off-centre mass distribution continue to create a twisting effect on the support bearings. Because the rotor is rotating, this twisting effect fluctuates just as the original out-of-balance forces did, and proper balance is not achieved.

In a dynamic balancing machine the rotor is set up in bearings fitted with sensing devices which measure the effect of the rotating couple due to the out-of-balance mass. The information is then computed, along with other information regarding the rotor dimensions, to determine the size correct location of the required mass compensa As with static balancing, material is then added removed and balance rechecked until a satisfac standard for the particular machine is reached.

It should be recognised that, while static balan is a relatively simple procedure that any techn can perform given the right equipment, dyna balancing is a specialised activity normally car out in a specially equipped workshop by tra personnel.

CHAPTER 5

LUBRICATION

5.1 PRINCIPLES OF LUBRICATION

Whenever the surfaces of two bodies are in contact, the force of friction will resist relative motion between them. The operation of almost all industrial equipment relies on the relative motion of separate machine elements and lubrication is necessary to overcome the effects of the friction forces. To lubricate means 'to make smooth and slippery', and thus the application of a lubricant helps to reduce the effect of friction. Friction causes energy to be wasted in the form of heat and causes the rubbing surfaces to wear. The introduction of a lubricant separates the surfaces in contact and thus reduces the effects of friction although friction can never be entirely eliminated.

The basic purposes of lubrication are to:

- reduce friction
- reduce wear
- dampen shock
- cool moving elements
- prevent corrosion
- seal out dirt

FRICTION

The force of friction always opposes relative motion between two bodies regardless of the shape, size or nature of the bodies. Friction not only exists between solids but also between liquids and solids and between gases and solids. However, we are only concerned here with friction between solids.

Causes of friction

Molecular attraction

The molecules of a substance are held together by electromagnetic forces between positively and negatively charged atoms as shown in Fig. 5-1. These forces can also act across the boundary of two substances in contact. Hence the negatively charged atoms of one substance attract the positively charged atoms of the other and vice versa. This molecular attraction increases as the surfaces in contact become smoother and theoretically two perfectly flat surfaces can not be separated once they come into contact except by mechanical means.

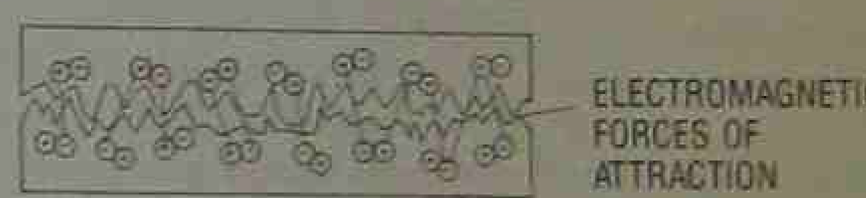


Fig. 5-1 Electromagnetic forces between positively and negatively charged atoms.

Interlocking of asperities

When viewed under a microscope even the smoothest surface is seen to contain a series of peaks and valleys (asperities). When two surfaces are in contact with each other, these asperities interlock and cause resistance to relative motion.

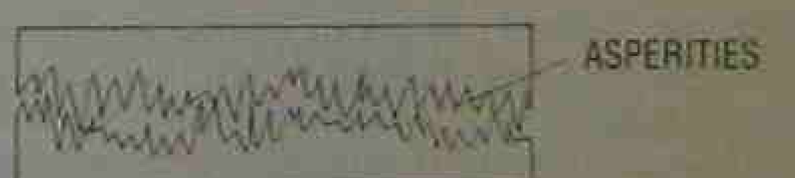


Fig. 5-2 Interlocking of asperities of two surfaces in contact.

Surface waviness

No surface is, in fact, perfectly flat but contains an element of waviness on which the irregularities

referred to above are superimposed, as shown in Fig. 5-3.
This waviness also creates resistance to relative motion in a similar way to the interlocking of the asperities.

Fig. 5-3 Surface waviness

Local welding

Because it is only the tips of the asperities that come into contact, the actual area of contact is much less than the total surface area. This means that the force between the surfaces, instead of being carried by the whole surface area, is only carried by that part in actual contact. Hence the pressure developed on these areas is extremely high and often can be sufficient to generate enough heat to cause the surfaces to melt and stick together as shown in Fig. 5-4.

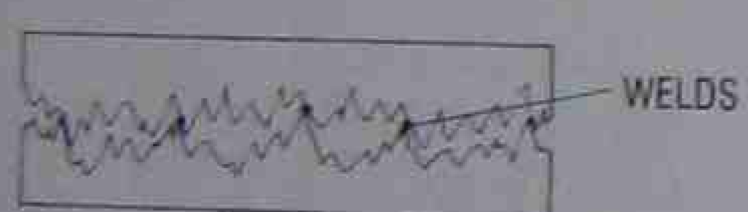


Fig. 5-4 Local welding of two surfaces in contact.

In order for the surfaces to move relative to each other, it is necessary for these welds to be broken.

Friction force

The amount of friction developed, that is, the size of the friction force, depends on:

- The type of frictional system.
- The nature and condition of the surfaces.
- The normal force between the surfaces.

The first two factors are given by the coefficient of friction and the relationship can be expressed mathematically as

$F = \mu N$ where F = force of friction
μ = coefficient of friction
N = normal reaction between the surfaces

Types of friction

There are three types of friction that can exist between solids.

Static friction

When a force is applied in order to move one surface with respect to another there is a certain minimum force required before movement can be achieved. The friction force resisting motion can be calculated from the relationship $F = \mu_s N$ where μ_s is the coefficient of static friction.

Dynamic friction

Once motion occurs, the force required to maintain it reduces and is less than the force required to initiate motion. This force can also be calculated from the relationship $F = \mu_d N$ where μ_d is the coefficient of dynamic friction.

The coefficient of dynamic friction is always less than the coefficient of static friction for any two surfaces because of the increased adhesion due to local welding that occurs when the surfaces are stationary. These coefficients are determined by experiment and common values can be found from Engineering Tables.

Dynamic friction is the main concern in the case of machine elements and there are two types to consider.

Sliding friction This is the commonly understood situation giving rise to friction involving one surface sliding over another.

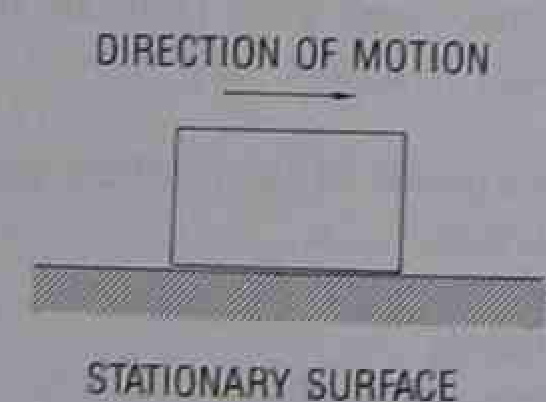


Fig. 5-5 Sliding friction.

This is the type of friction which occurs in plain bearings, cylinders, slides and cross heads, etc.

Rolling friction This is a rather different form of friction and to understand how it arises we must consider what happens when a ball, roller or wheel rolls across a surface. Although it is not usually visible to the naked eye some deformation of the opposing elements occurs with the result that a 'swell' in the surface occurs ahead of the rolling element as shown in Fig. 5-6.

The swell causes resistance to motion and this can be reduced, as with rolling element bearings, by making the surfaces as hard as possible in order to resist deformation.

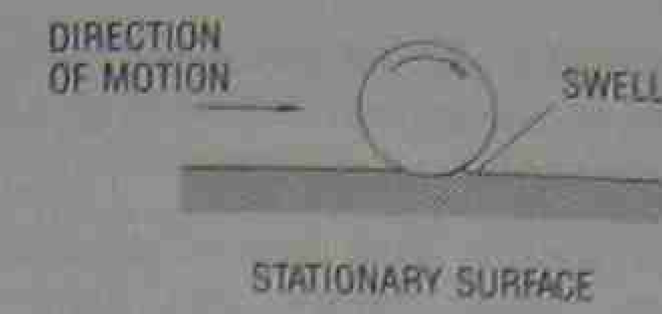


Fig. 5-6 Rolling friction.

Effects of friction

There are two major effects of friction which have consequences for machine elements and which lubrication is intended to overcome. Much of the energy that is used to overcome the force of friction is converted into heat. If this heat is not dissipated, but is allowed to build up, the moving elements may fuse together or 'seize up'.

The other effect of friction, which also uses up energy, is **wear**. The constant rubbing of one surface on another causes a physical change to occur to the topography of the surfaces and this may ultimately lead to a stage where the elements concerned can no longer perform their function effectively. The analysis of wear patterns is an important step in the troubleshooting process and in the assessment of lubricant effectiveness.

WEAR

Wear can be defined as 'the transformation of matter by use' and it results in a diminishing of the dimensions of machinery parts. Mechanical wear is the most important of four types of wear which convert useful materials into useless debris. The others are chemical (corrosion), bacteriological (decay) and electrolytic (metal removal). Mechanical wear is primarily caused by the relative motion of surfaces in physical contact and is a complex process often occurring as a result of some combination of the following:

Adhesive wear

Two surfaces in contact bear on the tips of the asperities which results in local welding and causes surface wear. When relative motion takes place a shearing process occurs. Sometimes the local welding is so strong that instead of shearing taking place at the surface, grain displacement occurs, creating pits or craters. This is known as **galling**. Adhesion is the most fundamental type of wear and occurs according to the nature of the opposing surfaces.

Abrasive wear

Abrasion occurs when particles of hard material become trapped between the surfaces or when hard particles are embedded in or attached to an opposing surface. The hard particles remove metal from the softer material rather like a grinding operation. The hard particles may be present in the materials, or may accumulate due to the shearing action between the surfaces. Abrasive particles may also enter the space between the surfaces from external sources.

Fatigue

Fatigue failure is caused by repetitive or cyclic stresses which weaken a material and cause failure to occur at a level of stress well below the normal strength of the material. This can happen in rolling element bearings due to the cycling effect of the rolling elements, and in plain bearings due to fluctuating load patterns. Fatigue first affects the sub-surface region and ultimately causes cracks to appear on the surface of the material. As the process progresses the surface begins to flake off. When this condition becomes severe it is known as **spalling**.

Fretting

Fretting, often known as false brinelling, is a type of corrosion which occurs when there is a slight, imperceptible motion between two surfaces in contact. The source of such motion is usually vibration. The surfaces at the area of contact deteriorate and break down to form oxides. This problem is sometimes encountered when machinery is in transit and the vibration of the carrier is transmitted to rolling element bearings. The slight motion between the rolling elements and the raceway can be sufficient to cause fretting to occur. The problem of fretting can be solved by eliminating the relative motion or plating the surface with non-oxidising materials, rather than by lubrication. In the example given above, barring the machine over from time to time may avoid the problem.

THEORY OF LUBRICATION

The purpose of lubrication is to reduce the friction between two contacting surfaces in relative motion by introducing a lubricant between the surfaces. In theory, any substance, whether it be liquid, solid or gaseous, may act as a lubricant, although in practice only a limited number of materials have the properties necessary to act as effective lubricants (see Section 5-2).

Lubricants work by creating special conditions between the surfaces in contact. There are four main types of lubricant film conditions that can exist:

Dry friction

When surfaces are clean and dry and no lubricant exists, the condition is referred to as dry friction. This condition gives rise to the greatest frictional resistance to motion.

Boundary lubrication

In this condition a thin layer of lubricant is present but significant metal-to-metal contact still exists. Hence part of the load is taken by the lubricant but most is still taken by the surface high spots. This condition, when used to combat heavy loading, is known as extreme pressure lubrication.

Mixed-film lubrication

This is an intermediate condition between boundary and full-film lubrication in which the lubricant layer is thicker than in boundary lubrication but some metal-to-metal contact still exists.

Full-film lubrication (also called thick-film)

This is the condition in which the moving surfaces are completely separated by the lubricant film. This can occur in three different ways:

Hydrostatic When the lubricant is supplied under pressure from an outside source e.g. a pump or gravity feed.

Hydrodynamic In which the pressure develops due to the resistance of the lubricant itself. This is the type of lubrication that occurs in a plain bearing and can best be understood by examining how the lubricant film forms in that situation. Fig. 5-7 shows the stages involved.

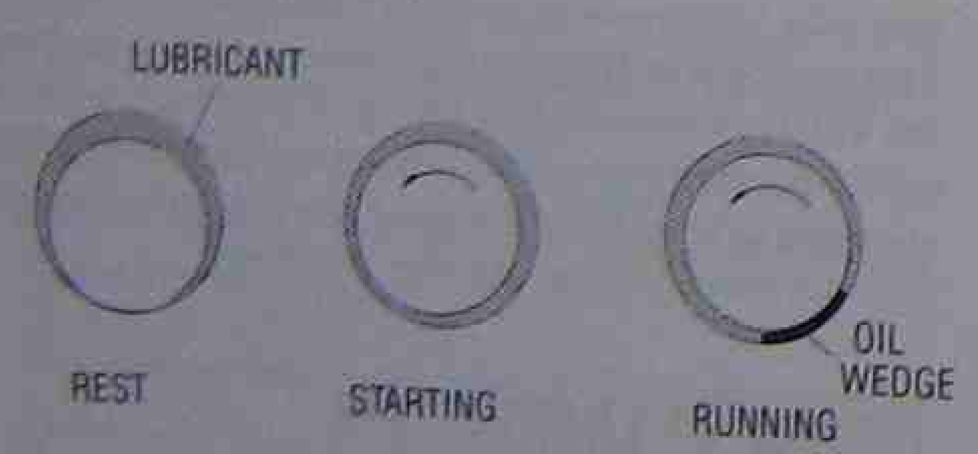


Fig. 5-7 Stages in the formation of lubricant film in a plain bearing.

While at rest the journal sits on the bottom of the bearing as shown. On start-up it climbs up the side of the bearing and establishes a lubricant film which, as it develops, forces the journal over to the other side of the bearing where it rides on a wedge of lubricant. As the lubricant is drawn into the wedge by the action of the journal it is compressed, and the pressure developed keeps the metal surfaces apart.

As the velocity of the journal increases, the w builds up and forces the journal and bearing clo a concentric relationship.

Elastohydrodynamic This form of lubrica associated with rolling friction and involv combination of the effects of the elasticity of surfaces in contact and the lubricant characteris When a ball or a roller contacts a raceway, surfaces deform slightly but return to their or shape when the element moves on, due to elasticity of the materials. As the rolling ele climbs out of the depression lubricant is drawn the space between the element and the raceway a hydrodynamic film is formed similar to that plain bearing.

In engineering applications such as plain rolling element bearings it is desirable to ach full-film lubrication to provide maximum separa of the surfaces. Where loads are very high this not be possible and the use of extreme pres lubricants may be needed in order to main boundary lubrication.

For components where positional accura essential, such as ways and slides, full-f lubrication is undesirable and boundary lubricati preferred because the metal-to-metal con involved ensures more precise positioning of components.

LUBRICANT SELECTION

The selection of a lubricant is determined by following factors:

Load The load on the bearing will determine pressure that the lubricant will have to work agai

Speed As operating speeds increase the lubrica surfaces will tend to wear faster.

Temperature The operating temperature m affect the properties of the lubricant.

Environment The lubricant may be required cope with the presence of water or corro materials.

Lubricant selection should normally be left those who are expert in the field. However, a general rule it is worth remembering that for pl journal bearings:

- For light loads and high speeds – use a lubricant low viscosity; and
- For high loads and low speeds – use a lubricant high viscosity.

The decision of whether to use oil or grease as lubricant will depend on the operating conditi

The following comparative advantages should be taken into account:

Oil: provides cooling
feeds more easily and can be fed from a central supply
washes away dirt
can also lubricate other elements such as gears
absorbs less torque

Grease: allows simpler bearing designs
provides better sealing against dirt
is easier to contain and seal
allows longer periods without attention

METHODS OF APPLICATION

The golden rule of lubrication is said to be: 'Good lubrication depends on the right lubricant being available in the right quantity at the right time.' For this to be achieved the technician must be aware of a number of basic principles governing the application of lubricants.

- Cleanliness is vital. Lubricating equipment must be kept free of dirt and other contaminants.
- Lubricants are not necessarily interchangeable and as a general rule should not be mixed. Before changing lubricant the equipment should be cleaned out.
- An excess of lubricant, especially grease, will cause excessive heat to build up and eventual breakdown of the lubricant.
- Lubricant filters or strainers should always be changed at the recommended time.
- The selection of lubricant for a particular application should be left to qualified personnel if possible.
- Inadequate lubrication can often be identified by the operating condition of a bearing, especially its temperature. As a general rule, if a bearing is too hot to hold a hand on it, then lubrication may be inadequate and should be investigated.
- Lubricants are potentially hazardous materials and should be stored with regard to safety and effect on the environment.

There are four basic methods by which lubricants can be applied and these are selected according to design criteria and the particular demands of the equipment.

Manual application

Whether the lubricant is liquid, semi-solid or solid, the simplest method of application is by hand. An oil can may be used for liquid lubricant, a grease gun for grease and a brush or spray gun for solid lubricant.

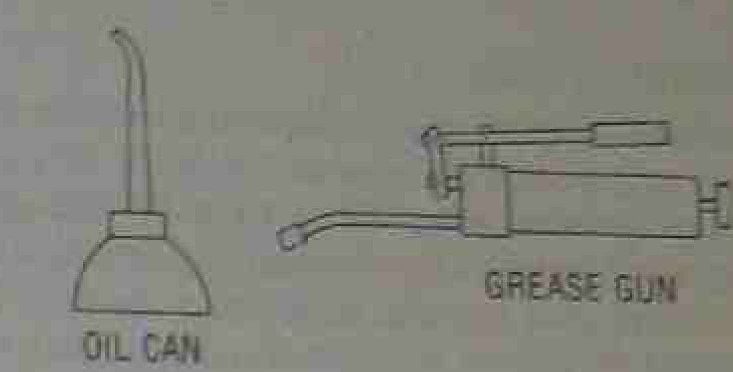


Fig. 5-8 Equipment for manual application of lubricant.

Gravity

This method is only suitable for liquid lubricants and is sometimes referred to as drip-feed oiling. There are various types of drip-feed oilers and they usually include some method of feed regulation.

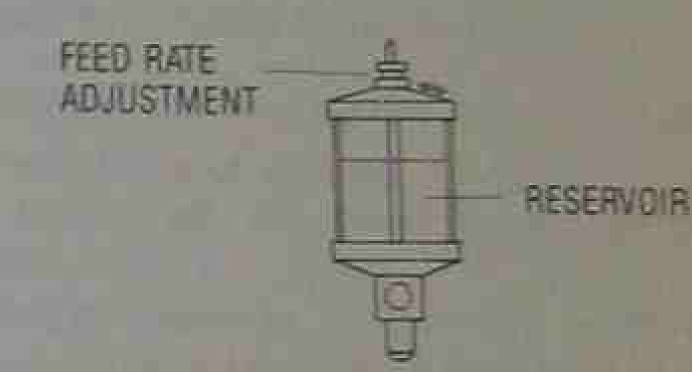


Fig. 5-9 Drip-feed oiler.

Splash lubrication

Splash lubrication relies on the components requiring lubrication being partially immersed in an oil sump so that they pick up oil as they rotate. The oil picked up in the process may also be deposited on the shaft bearings and other components. A variation on this method is the ring-type oiler which uses a steel or brass ring which rotates with the shaft and picks up oil which it deposits on the upper surface of the shaft. Examples of these methods are shown in Fig. 5-10.

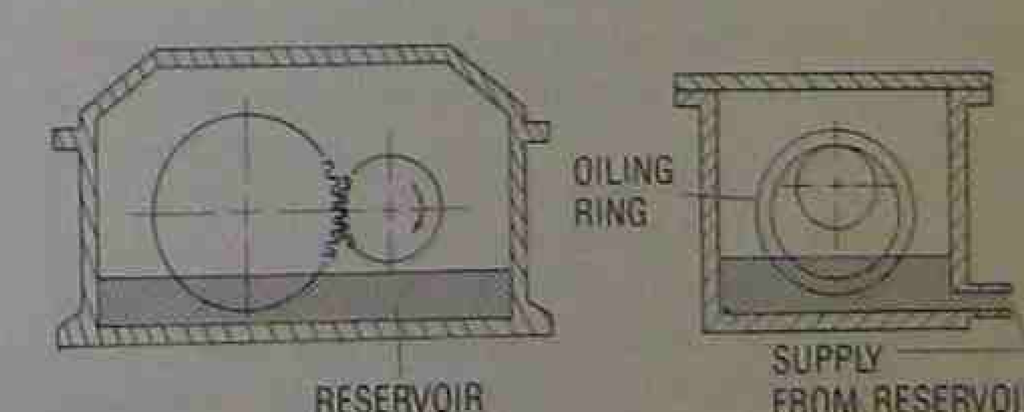


Fig. 5-10 Splash lubrication methods.

Pressure lubrication

Many industrial applications, especially where loads are heavy and operating speeds are high, require a pressurised system to ensure that an adequate supply of lubricant can be maintained. This usually takes the form of a circulating system such as that shown in Fig. 5-11.

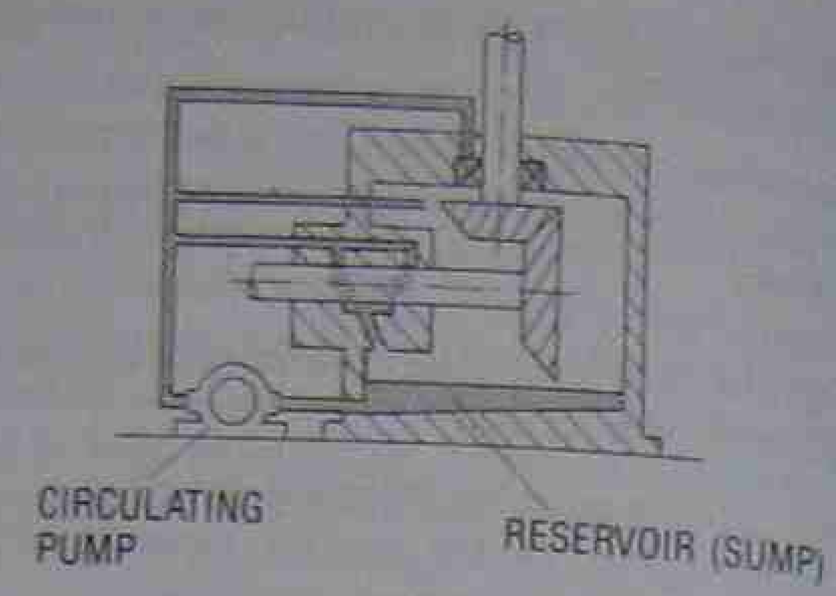


Fig. 5-11 Circulating system of lubrication.

5.2 LUBRICANTS

CHARACTERISTICS AND PROPERTIES

There are four basic types of lubricant:
- liquid
- semi-solid or plastic
- solid
- gaseous

and they can be classified according to their source:
- animal
- vegetable
- mineral

Most lubricants are mineral based and are obtained from petroleum by refining processes and further purification and blending. Petroleum and petroleum products belong to the group of chemicals known as 'hydrocarbons' because they are compounds of the elements hydrogen and carbon in varying combinations.

Animal fats come from common animals such as cattle, sheep, pigs and fish and are melted down to remove impurities. Vegetable oils are produced by squeezing vegetables and seeds (e.g. soyabean and rapeseed) to produce pulp and juice and then further refined to remove impurities. Animal and vegetable oils are less stable than mineral oils and break down more easily.

Lubricants are classified according to their properties and a number of standard tests are applied to common types of lubricants, such as oils and greases, in order to determine their properties.

Criteria for measuring the properties of oils

The properties of lubricating oils are measured by the following criteria:

Viscosity This is the single most important characteristic and refers to the 'thickness' of a fluid and is also described as resistance to flow. Viscosity is affected by temperature and decreases with increasing temperature. There are various ways of measuring viscosity, all of which are based on the time taken for a fixed volume of oil to pass through a standard orifice under standard conditions. The SI unit is the centistoke (cSt).

Viscosity index The rate of change of viscosity with temperature is known as the viscosity index. It is normally desirable for the viscosity of a lubricant to remain the same over a wide range of temperatures. A high viscosity index indicates such a property whereas a low index indicates that the oil tends to thin out rapidly with increasing temperature.

Flash point This is the temperature at which the vapour of a lubricant will ignite.

Fire point This is the temperature, higher than the flash point, required to form sufficient vapour from the lubricant to cause it to burn steadily.

Pour point This is the low temperature at which the lubricant becomes so thick that it ceases to flow.

Oxidation resistance When hydrocarbons are exposed to the atmosphere, especially at increased temperatures, they tend to absorb oxygen. This causes a chemical change in the oil that makes it useless for lubricating purposes.

Emulsification An emulsion, e.g. a mixture of water and oil, is undesirable because it has poor lubricating properties. Emulsification is a measure of the tendency of an oil to mix intimately with water. Demulsibility measures the readiness of an oil to separate from water in an emulsion.

Criteria for measuring the properties of greases

The properties of greases, being semi-solid rather than liquid, are measured by a separate set of criteria:

Hardness Because greases are semi-solid they can be considered as ranging from hard to soft. These ratings are based on the results of a penetration test and the standard gradings used by the National Lubricating Grease Institute (US) are as follows:

NLGI No.	*Consistency*	*ASTM worked penetration at 25°C (77°F) 10^{-1} mm*
000	very fluid	445–475
00	fluid	400–430
0	semi fluid	355–385
1	very soft	310–340
2	soft	265–295
3	semi firm	220–250
4	firm	175–205
5	very firm	130–160
6	hard	85–115

Dropping point This is the temperature at which the grease will change from semi-solid to liquid i.e. the melting point.

Pumpability This is a measure of the ease with which the grease will flow through a system.

Water resistance This determines whether or not a grease will dissolve in water. This property is important where there is a likelihood of water coming into contact with the lubricant.

Stability This property determines the ability of a grease to retain its characteristics with time. Some greases become soft and thin after being in use for a while.

LIQUID LUBRICANTS

Rather than classifying lubricants according to their composition, it is more useful for industrial purposes to classify them according to their application. There are several commonly recognised categories.

Circulating oils

These oils are designed to circulate through a closed system such as a crankcase or hydraulic circuit. Because they remain in the system for some time they normally include a range of additives such as anti-oxidants, anti-foam agents, rust inhibitors, etc.

Gear oils

Gear oils are required to handle the relatively high pressures that develop between gear teeth and also to dampen the shock of impact. They are generally required to have high viscosity although they are often required to lubricate bearings as well and must be able to transfer heat. They are normally classified as follows:

Grade	Viscosity (cSt)
Light	140–160
Medium	200–240
Heavy	420–500
Light E.P.	60– 75
Heavy E.P.	300–360
Cling type gear shield	200–240

The oils in the first group are designed for gears running in enclosed spaces and lubricated by splash or pressure systems. The extreme pressure type of oils are particularly used where tooth loads are high, such as hypoid gears used in automotive transmissions. Cling type oils are used for open gears and are specially compounded to resist being thrown off as the gears move. Gear oils often contain anti-wear additives where tooth loads are expected to be heavy.

Machine and engine oils

These are general purpose oils suitable for once-through systems and were originally designed for the external operating parts of machinery that could be oiled with cans or cups. They are commonly used on plain bearings and for slides and ways. Viscosities range from 35–200 cSt. They should not be used in situations where sludge may form.

Spindle oils

These are very carefully refined, high quality oils designed to lubricate the spindles of the textile industry and also to lubricate delicate instrum[ents] and other sensitive equipment. They are us[ed] straight mineral oils with low viscosities ranging f[rom] 1–25 cSt.

Refrigeration oils

Oils used for lubricating refrigeration equipment [are] special straight mineral oils with low pour point [and] free of wax and moisture. They range in visco[sity] from 15–120 cSt and are usually non-foaming.

Steam cylinder oils

These are special purpose oils, formulated [by] compounding mineral oils with animal oils, [and] sprayed into the steam or applied directly to [the] walls of steam cylinders. The animal oils help [to] ensure that they adhere to the surfaces they [are] supposed to lubricate. Viscosities are high and ra[nge] from 20–35 cSt at 100°C (210°F). [500–1300 cSt [at] 38°C (100°F)].

Special purpose oils

A number of special purpose oils are available [for] particular applications, not necessarily associat[ed] with lubrication. These include:

fire resistant oils
cutting oils
heat treatment oils

General purpose oils

In recent years manufacturers have developed [a] range of multipurpose oils which meet multiviscos[ity] or multigrade specifications. These are primari[ly] used for lubricating automotive engines a[nd] transmissions and can withstand the variati[ons] between summer and winter conditions.

The above categories are only general and ma[ny] oils are suited to more than one particul[ar] application. The categories are listed merely as [a] guide and the selection of lubricants should b[e] referred to the supplier.

GREASES

Categories of greases

Soap-thickened mineral oils

These are the most commonly used greases and ar[e] classified according to the type of soap base used t[o] thicken the oil.

Calcium-based The most important gener[al] purpose greases. They do not dissolve in water an[d] are limited to applications below 70°C (160°F).

Sodium-based General purpose greases with a high dropping point of 150°C (300°F). They can be used for parts near a heat source.

Barium-based General purpose greases suitable for temperatures up to 135°C (275°F) but less suitable for low temperature applications due to the high soap content.

Lithium-based Have good water resistance and stability and are suitable for both low temperature −50°C (−60°F) and high temperature 150°C (300°F) applications.

Strontium-based Have good resistance to water washing and to corrosion from substances such as salt water. They can also operate up to 175°C (350°F).

Aluminium-based Have the particular advantage that they adhere well to the surfaces they are supposed to lubricate.

Complex Greases whose basic ingredients have been fortified, modified or treated in some way in order to achieve high performance in a particular application.

Mixed-base Combine the advantages of two or more different soap bases.

Multipurpose The advantage of these greases is that they reduce the number of types of grease required in a particular industrial situation and also reduce the possibility of using an inappropriate grease. They are usually either barium or lithium-based and combine the properties of specialised greases.

Mineral oils mixed with solids

These are heavy greases used to lubricate rough-fitting components which operate under high pressure or at low speed such as conveyor systems.

Heavy asphaltic-type oils

Although these are really thick oils they are classified as greases and are used to lubricate open gearing and wire ropes. They are painted on when hot, and cool to form a protective coating.

Extreme pressure greases

These are designed to give improved load carrying capacity in rolling element bearings and gears. Various types of additives are used to improve the film-strength (resistance to rupture) of the grease. They are often lithium-based and also contain further additives to improve lubricity (oiliness).

Roll-neck greases

These are specialised greases used for lubricating plain bearings in rolling mill equipment and are usually used in block form which is cut to shape to fit the bearing.

Synthetic oil-based greases

Various synthetic fluids are used to produce greases with special characteristics not easily attainable with mineral oils. The fluids used include: chlorofluorocarbon polymers, dibasic acid esters, polyglycols, silicones and polyphenyl ethers.

The silicone-based greases are particularly useful because they are suitable for a wide temperature range and are resistant to oxidation. For this reason they are often used in sealed-for-life bearings.

Applications of greases

Ways and slides

The greases used for ways and slides are usually sodium-based of consistency NLGI No. 1 or 2.

Plain bearing greases

Greases can be used for the lubrication of plain bearings operating at low speeds up to about 400 RPM. Grease has a tendency to work its way out of the ends of the bearing so it must be regularly replaced, especially as speeds increase. Most of the soap-based greases can be used for lubricating plain bearings. Lithium-based grease is one of the most versatile.

Rolling element bearings

Greases used for rolling element bearings must be free of contaminants and chemically active elements that may attack the bearing or the seal, and they must also maintain consistency in operation. Most general purpose greases are suitable depending on the particular conditions.

SOLID LUBRICANTS

Solid lubricants, such as graphite and molybdenum disulphide, may be used as additives for oil and grease or may be used alone in their dry state where oils and greases do not perform satisfactorily. Solid lubricants may have advantages over oils and greases where:

- Bearings are inaccessible or likely to be missed by routine maintenance;
- The use of liquid lubricants may cause product contamination;
- There is a tendency for galling or seizing to occur;
- Operating temperatures are either too low or too high for oils and greases;
- Fretting corrosion is a possibility;
- Vacuum conditions exist which tend to cause evaporation of most substances;
- The lubricant is exposed to nuclear radiation

which causes ionisation of organic compounds and change in viscosity.

The most satisfactory method of application for solid lubricants in the dry state is spraying or dipping. They may also be brushed on but care must be taken to ensure that film thickness is even. When applied in this way they are usually used in a bonded form using a 'binder' which helps the lubricant to adhere to the surface. Phenolic-resins are often used for this purpose.

Molybdenum disulphide in its powdered form is often used on metal forming dies and on threaded parts. Bolts that have been lubricated can be tightened closer to strength limits and undone more easily.

ADDITIVES

The wide variation in the requirements of modern machinery is beyond the ability of straight oils and greases to handle and a range of additives has been developed to improve lubricant properties. The most common types of additives are:

Oxidation inhibitors These are designed to prevent the chemical breakdown of lubricants and the formation of acids.

Detergents and dispersants Detergents help to keep surfaces clean by preventing the formation of dirt particles. Dispersants keep the dirt particles in suspension by enveloping the dirt particles and preventing them from adhering to the metal surfaces.

Rust and corrosion inhibitors These preven retard the formation of rust and also protect m parts against corrosion by contaminants.

Pour point depressants These are used ensure that the lubricant will maintain its ability flow at low temperatures.

Viscosity index improvers These are added reduce the effect of changes in temperature viscosity.

Anti-foam agents These help to break up the bubbles that tend to form in a circulating or hydraulic system.

Anti-friction compounds Oiliness and lubric of the lubricant are increased in order to reduce t coefficient of friction between the rubbing surfac

Anti-wear agents These also reduce friction a wear due to scoring, seizing and rubbing.

Extreme pressure agents These are u primarily in gear oils and help to cushion the sho between gear teeth at high loads.

Emulsifiers These keep water away from conta surfaces by forming an oil film around wa particles.

Emulsion breakers When an emulsion undesirable an emulsion breaker is used to help t oil and water separate more easily.

Adhesive compounds These ensure th lubricants adhere to surfaces and prevent the being thrown off by centrifugal force or turbulenc

A range of organic and inorganic substances used to achieve these various effects and individu manufacturers of lubricants have their ov particular formulae.

CHAPTER 6

BEARINGS

A bearing is a device which supports a rotating shaft or spindle or guides one component which slides over another. In addition to its supporting function, a bearing is designed to allow relative movement between two separate components to occur with the least possible frictional resistance. Almost all industrial mechanisms contain elements which require relative movement between contacting surfaces and therefore include some sort of bearing.

In principle, bearings fit into two main categories:

Plain bearings in which the surface of one component slides over the surface of another and where the surfaces in contact are specially prepared in order to minimise friction and wear.

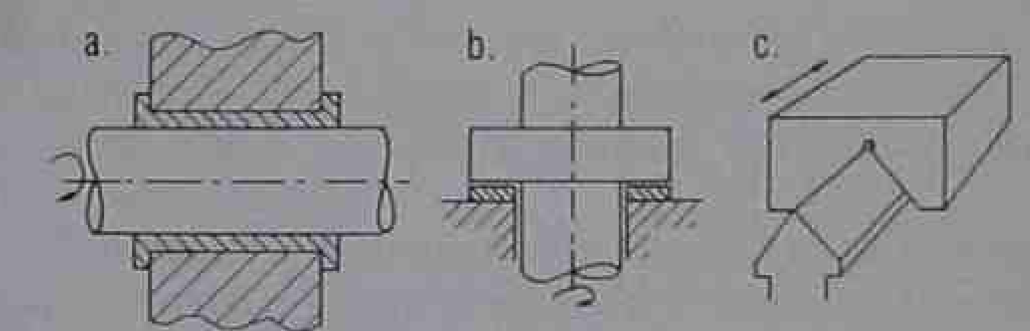


Fig. 6-1 Examples of plain bearings.

Rolling element bearings in which a series of rolling elements, i.e. either balls or rollers of various shapes, are interposed between the two surfaces in order to facilitate movement of one with respect to the other. Rolling element bearings are sometimes referred to as anti-friction bearings because the relatively small contact area of the rolling elements helps to reduce, though not eliminate, the resistance to relative motion.

Bearings can also be classified according to the type of function they perform:

Journal bearings which support a rotating shaft or spindle and confine radial motion as in Figs. 6-1a and 6-2a.

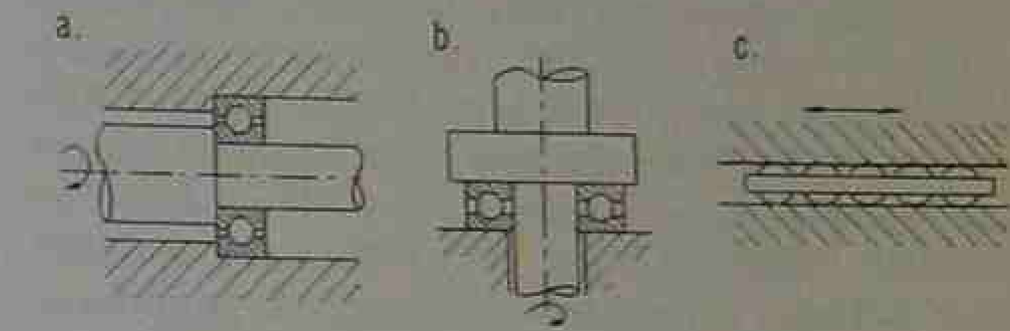


Fig. 6-2 Examples of rolling element bearings.

Thrust bearings which prevent axial motion of a shaft as in Figs. 6-1b and 6-2b.

Linear bearings which guide or support relative motion between components in a straight line as in Figs. 6-1c and 6-2c.

The function a bearing performs is largely determined by the type of load that it has to carry. The type of load also determines the particular type of bearing selected. Loads can be one of three kinds:

Radial
Axial
Combination

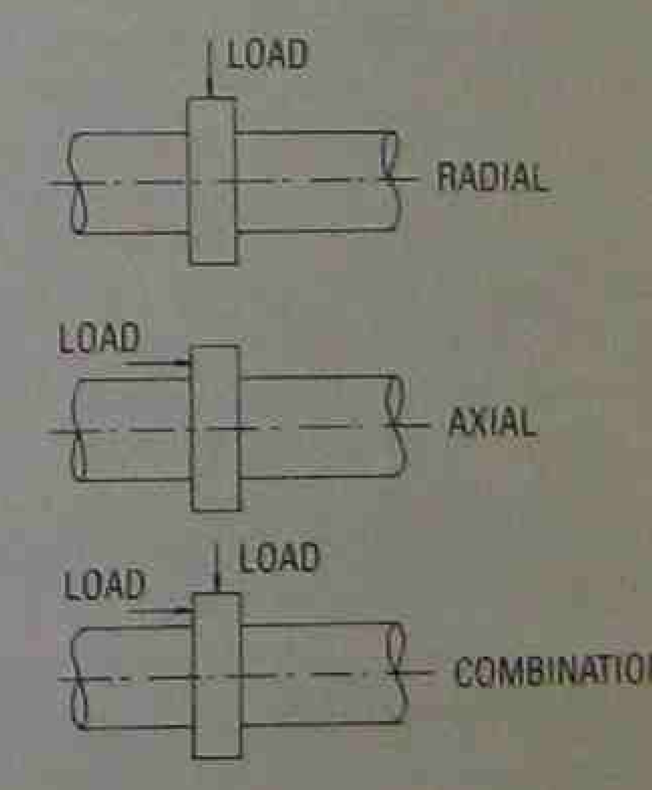


Fig. 6-3 Bearing loads.

6.1 PLAIN BEARINGS

PRINCIPLES OF OPERATION

The operation of a plain bearing relies on relative motion between plain surfaces which are either cylindrical, in the case of journal bearings, or flat, in the case of thrust or slider bearings. In order that motion may occur with minimum resistance the effect of friction between the two surfaces must be reduced as much as possible. This is achieved by ensuring that the two surfaces are smooth and by introducing a film of lubricant between them. In some cases a bearing material which has self-lubricating properties may be used, otherwise a separate supply of oil or grease must be provided. For the best operation it is preferable that full-film lubrication is achieved so that metal-to-metal contact is eliminated. In some cases however, this may not be possible. When loads are very high it is impossible for a lubricant to maintain separation of the surfaces and even with extreme pressure lubricants, only boundary lubrication can be achieved. In the case of slider bearings, where positional accuracy of the sliding element is required such as for machine tool carriages, full-film lubrication is undesirable because the lubricant film tends to vary in thickness. Hence boundary lubrication, which allows some metal-to-metal contact and thus more accurately locates the sliding element, is preferable.

The material of a plain bearing is always softer than the material of the shaft or slider it supports. This allows the bearing to wear out rather than the shaft or slider. The design of plain bearing assemblies is usually such that the bearing material itself can be relatively easily replaced.

The key to satisfactory operation of a plain bearing is the maintenance of the correct lubricant film. This is often assisted by machined grooves in the bearing surface which act as reservoirs and ensure even distribution of lubricant across the bearing surface. The grooves are usually connected to the lubricant inlet which is often located just ahead of the most highly loaded area of the bearing.

One of the most important aspects of bearing operation is the control of temperature. This may be achieved either by ensuring an adequate flow of lubricant which acts as a coolant or by surrounding the bearing with chambers in the housing through which cooling water can circulate.

TYPES AND MATERIALS

Journal bearings

A plain journal bearing is essentially a tube which encircles, either totally or partially, a rotating shaft.

The tube is normally held in a fixed housing provides a support for the shaft and allows it to ro with minimum resistance.

There are two principal types of plain jour bearings – solid and split.

A solid bearing is usually called a bushing whe is of the thin-walled type i.e. up to 2mm (5/64") fo ⌀25mm (1") bearing, and a sleeve bearing when of the thick-walled type.

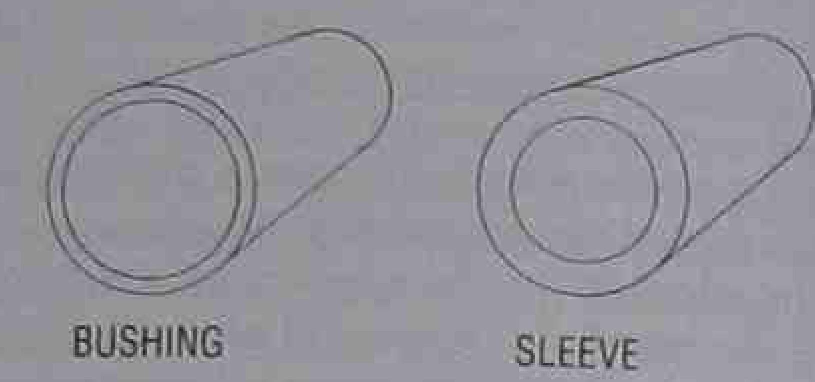


Fig. 6-4 Types of solid journal bearings.

There are various types of split bearing but most common consists of two halves which toget form a full circle. They are used particularly for e of shaft removal which can be accomplished removing the top half of the bearing only.

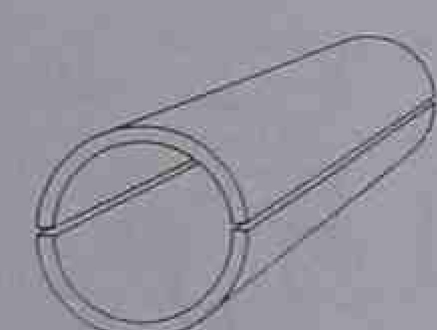


Fig. 6-5 Split journal bearing.

Similar to a split half bearing is a multipart bear which is composed of more than two segments whi fit together around the journal to form a full circ

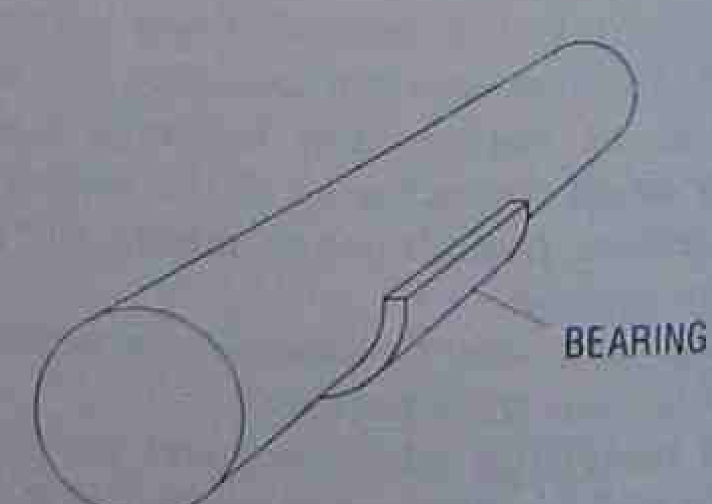


Fig. 6-6 Partial bearing.

A partial bearing is one which supports the shaft from the bottom only and may comprise a half-circle or less. This type of bearing can only be used when loads are relatively heavy and are constant in direction.

Thrust bearings

A plain thrust bearing supports the end thrust of a shaft and restrains axial movement. The bearing surface itself consists of a stationary surface in the form of a pad against which the rotating element bears.

The most common form of plain thrust bearing is the thrust collar arrangement with the bearing face machined with radial grooves as shown in Fig. 6-7.

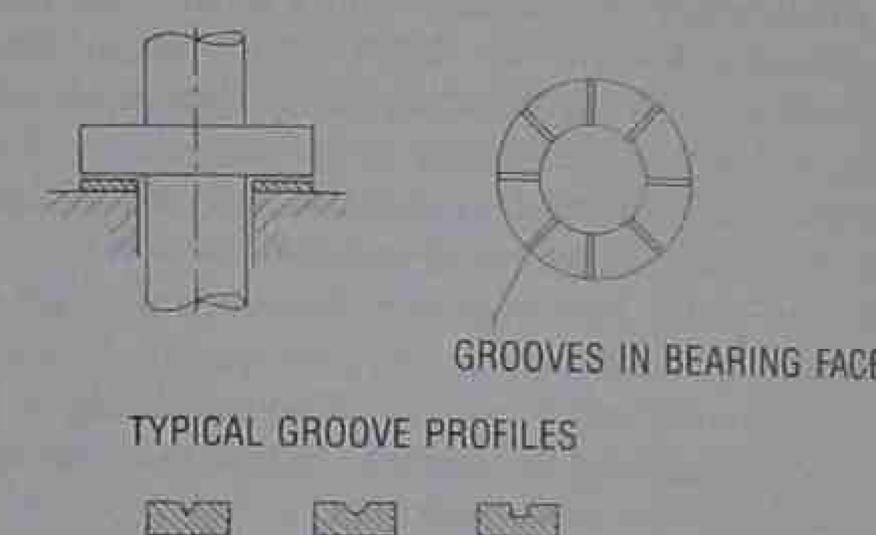


Fig. 6-7 Plain thrust bearing.

Lubricant must be fed into the bearing at the inner diameter so that lubricant flows outwards across the bearing face. Running speeds are limited by lubricant viscosity.

A slight modification to the above is a profiled pad thrust bearing where the pads are shaped to provide a convergent lubricant film and help to form a hydrodynamic wedge. The pad may be profiled to be uni-directional or bi-directional as shown below in Fig. 6-8.

Fig. 6-8 Profiled pad thrust bearing.

Tilting pad thrust bearings are able to accomodate a large range of speeds, loads, and lubricant viscosities. This is achieved by providing pivoted pads which can assume a small angle with respect to the rotating collar surface, and this makes sure that a full hydrodynamic film is maintained between the surfaces.

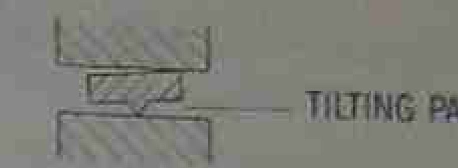


Fig. 6-9 Tilting pad thrust bearing.

Linear bearings

Linear bearings, which are often called ways or guides, support either linear or curvilinear motion and are often associated with reciprocating mechanisms.

There are various forms that linear bearings can take, some of which are shown in Fig. 6-10.

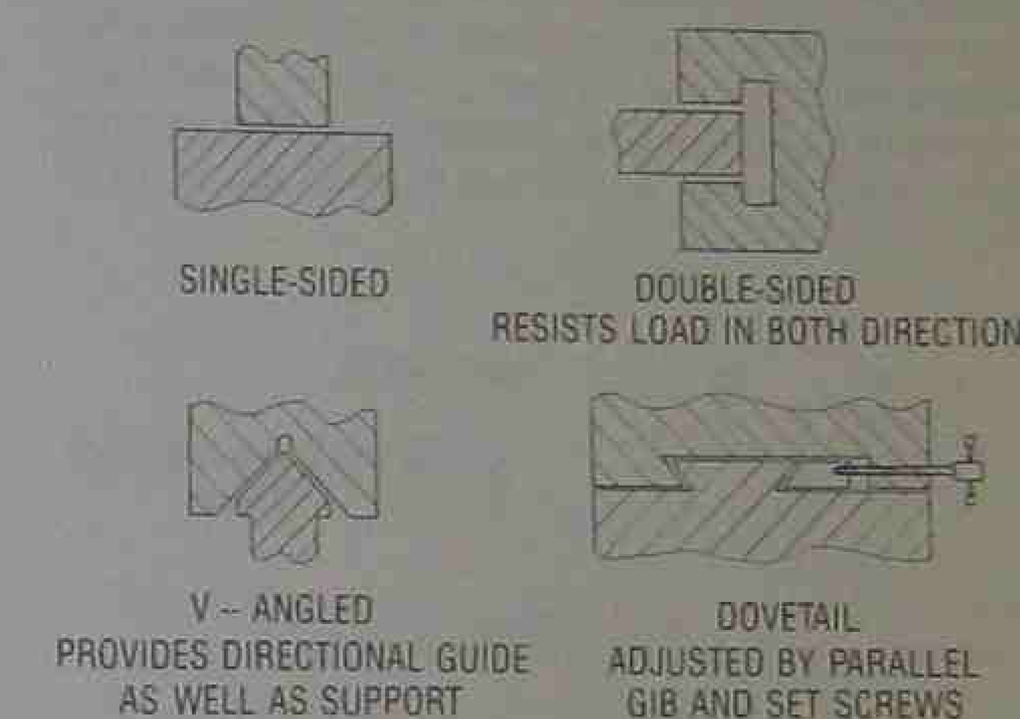


Fig. 6-10 Typical plain linear bearings.

Materials

The requirements of a plain bearing material are complex and, in some senses, conflicting. For example, a bearing material must be hard enough to resist wear and yet soft enough to conform to the contours of the journal during the running-in process. In order to meet these complex requirements, a range of special materials that are used purely as bearing materials has been developed.

A plain bearing material must exhibit the following important characteristics:

Compressive strength To support uni-directional loading without deformation.

Fatigue strength To resist fluctuating dynamic loadings.

Embeddability To embed foreign matter and protect the journal from wear and scoring.

Conformability To tolerate misalignment and deflection under load.

Compatibility To tolerate momentary metal-to-metal contact without seizure.

Corrosion resistance To resist attack due to water or oxidised lubricant products.

The following materials are commonly used:

Whitemetals

The whitemetals or babbitt alloys are the best known of all bearing materials. They are either tin-based or lead-based alloys and contain a significant percentage of antimony. They are normally used as a coating or liner backed by steel, cast iron or bronze. Whitemetals have excellent characteristics in all respects except that compressive and fatigue strength falls rapidly with rising temperature.

Copper-lead alloys

These provide strength and fatigue resistance up to four times that of whitemetals. Conformability and embeddability are lower however, and are sometimes improved by overlaying a thin layer of whitemetal in a tri-metal construction as shown in Fig. 6-11.

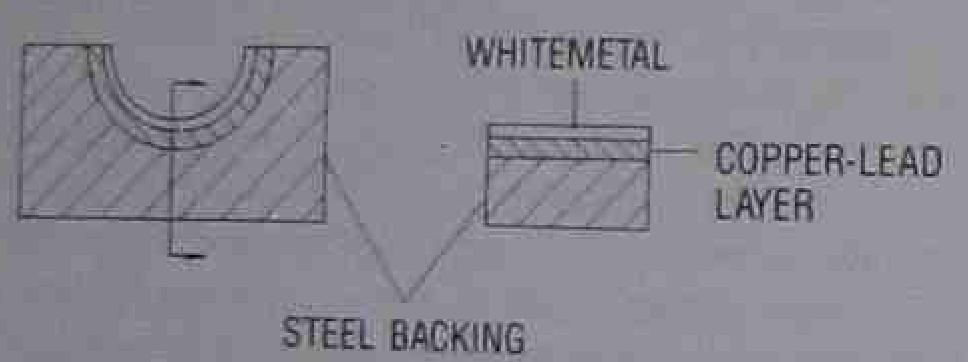


Fig. 6-11 Overlaying a copper-lead alloy with whitemetal.

Bronzes

Bronzes can be produced with a wide range of properties and are probably the most economical of all bearing materials. Four types are used.

Lead bronze Can be used without a steel backing but has low comformability and must be accurately aligned. It is easily cast and machined and is used for moderate speeds and loads.

Tin bronze Has relatively high hardness and is used in applications where loads are high but speeds are low. Tin bronzes require reliable lubrication and a hardened journal.

Aluminium bronze Has good shock and wear resistance and can be used at temperatures of 260°C (500°F) and above. It is best used in heavy duty, low speed applications.

Phospor bronze Also useful in heavy load, high temperature applications.

Aluminium alloys

These may be used as solid aluminium on steel or with an overlay of whitemetal. They have high load capacity, good fatigue strength and excell corrosion resistance, but poor conformability a embeddability. They are best suited to heavy lo at moderate speeds.

For more specific details regarding composition of bearing materials it is necessary consult either the appropriate standards manufacturer's information.

MAINTENANCE PRACTICES

General

There are a number of general considerations t relate to the maintenance of plain bearings.

- Cleanliness is a keynote that must be observ when handling bearings of any kind. Dust and d particles should be kept away from bear surfaces and all bearing components should kept dry and protected at all times.
- Good alignment is essential to plain bearin without self-aligning properties.
- When plain bearings are disassembled all pa should be marked so that they can be reinstall in their original positions. The matching of pa during running-in makes them unsuitable f operation in other positions.
- When fitting plain journal bearings care attention should be paid to the correct tensionin of bolts. Bolting operations are normal conducted by manufacturers with bolts tensione to the same specification recommended in the maintenance instructions. Incorrect tensionin may affect clearances and bearing crush.
- The key to good bearing performance is havi the correct clearances and the correct lubrica
- Bearing surfaces should be given prelimina lubrication before the shaft is turned.
- The journal surface or thrust collar should alwa be checked to ensure that it is perfectly smooth. I the case of a journal, a micrometer should be use to check the size so that replacement bearings w have the correct i.d. (inside diameter).

Assembly

Bushes and sleeves

Bushes and sleeves may be either a press-fit or shrink-fit in the housing.

In the case of a press-fit the following procedu should be adopted.

1 Check the dimensions of the bush and housing a compare with manufacturer's recommendatio to ensure they are correct.
2 Ensure that the mating surfaces are clean, smoo and free of burrs.
3 Press the bearing into the housing using an arb press if possible. Hammering should be avoided as this may ruin the bearing. The bearing should be pressed in in one continuous motion and a lubricant should be used.
4 Check the inside diameter of the bush to see if it has been reduced due to 'close-in' as a result of the press fit.
5 Hone or ream the bush to the recommended size.

In the case of a shrink-fit, the bearing can be assembled either by heating the housing or by cooling the bearing. If the housing is heated, care should be taken to ensure that heating is uniform so that no warping or distortion occurs. Heat up should not be too rapid and normally a temperature of around 100°C should provide sufficient expansion. Avoid heating the housing any more than is absolutely essential. If it is not convenient to heat the housing, the bearing may be packed in dry-ice or chilled in a deep-freeze unit before assembly. As in the case of a press-fit, it is advisable to check the inside diameter of the bearing when temperatures have returned to normal to ensure that it has not been reduced due to 'close-in'.

Split bearings

There are a number of important considerations to be taken into account when assembling split bearings of the precision type.

- It is vital that the surfaces of the bearing and housing are clean. Especially check between the bearing and housing and the joint faces of both the bearing and housing. Also check the oil-ways and oil grooves.
- A split bearing is provided with 'free spread' to ensure that both halves assemble correctly and do not foul the shaft along the joint line when bolted up.

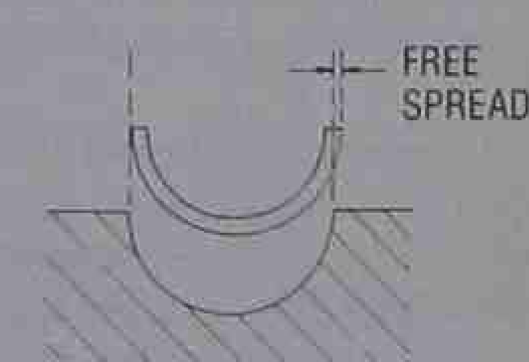


Fig. 6-12 Free spread in a split bearing.

Free spread can be adjusted by tapping the bearing gently with a wooden mallet as shown in Fig. 6-13.

Manufacturer's recommendations should be checked to determine the amount of free spread allowed.

- To ensure that the bearing inserts have good contact with the housing they are made so that the inside diameter at right angles to the parting line is slightly larger than the diameter across the parting line as shown in Fig. 6-14.

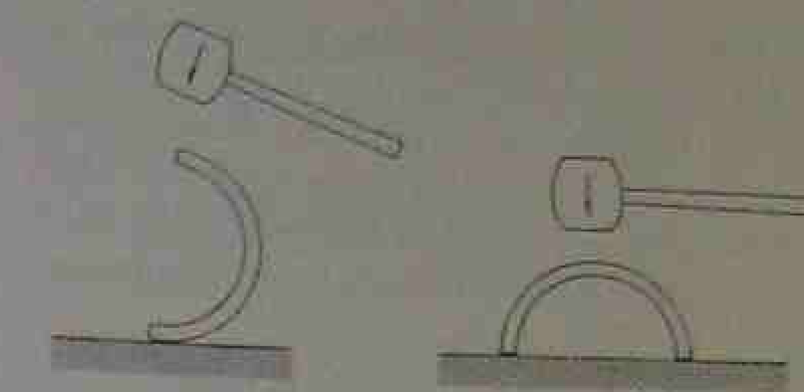


Fig. 6-13 Adjusting free spread.

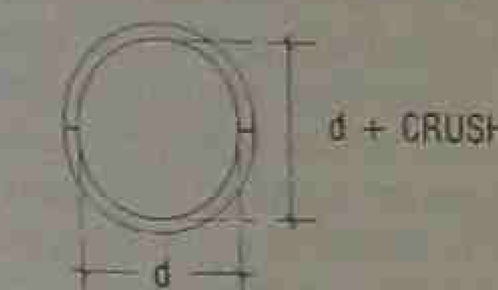


Fig. 6-14 Bearing crush.

This excess is called crush and ensures that when the bearing is bolted up it will be held securely and in full contact with the housing.

The bearing halves should be installed in the housing so that the crush is even on both sides of the housing. Bolts should be torqued up progressively and evenly on both sides.

Clearances

The correct adjustment of clearances is vital to the operation of plain bearings. If clearances are insufficient then metal-to-metal contact will occur causing excessive wear. If clearances are too great then the hydrodynamic wedge will break down and also cause metal-to-metal contact.

In the case of journal bearings there are several ways in which clearances can be checked.

- Measurement of the inside diameter of the bearing and outside diameter of the journal can be made with either a micrometer or gauges. Care should be taken if using a micrometer, to take readings in several positions.
- A soft material that will deform easily, such as plastic thread, can be inserted between the bearing and the journal before the bolts are torqued up. After the bolts have been torqued and released again, the flattened material can be compared with a suitably prepared chart and the clearance read directly.

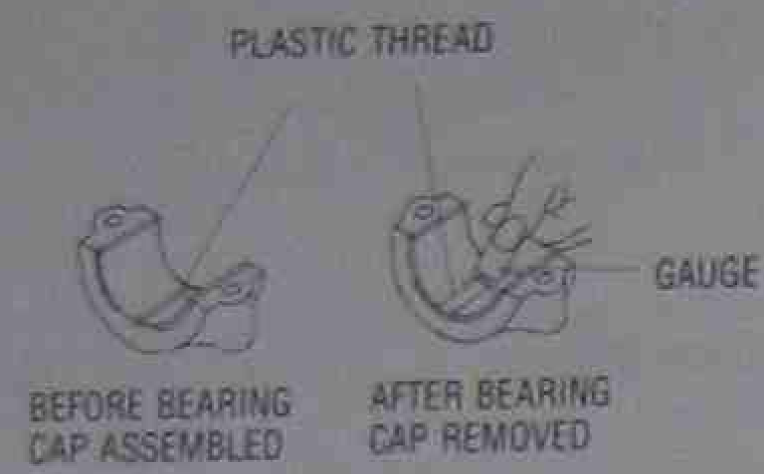


Fig. 6-15 Checking bearing clearance with plastic thread.

- A third alternative is to insert lead or brass shims in between the bearing and the housing. The shims are adjusted to the point where the shaft turns easily but where an extra 0.025mm (0.001") added to the thickness of the shim would cause it to jam. Great care has to be taken not to damage the bearing surfaces if this method is used.

If, whatever method is used, some out-of-roundness is found to exist, then it is the minimum clearance which is critical.

In the case of thrust bearings it is also important to measure the end float to ensure that sufficient clearance exists between the thrust collar and the bearing. This can be done by forcing the shaft hard up in each direction and either measuring the clearance at the thrust faces with gauges or shims or setting up a dial gauge and reading against a suitable shoulder on the shaft.

Relining journal bearings

Although it is less common these days for plain bearing liners to be poured on site it is often still practical for large bearings used in heavy equipment. For smaller bearings, spare liners are usually purchased from the manufacturer and held as stock. However the technique of relining is still used and should be understood.

As a general rule only bearings with a low melting point, such as whitemetal, are relined. Copper and aluminium based alloys have high melting points and the higher pouring temperatures tend to cause distortion of the bearing supports.

The following procedure for relining is recommended:

1 Before starting the operation ensure that proper protective clothing and safety glasses are available. The process is one of casting a molten metal and this may spit or splash during pouring.
2 First clean up the old shell by heating with a torch to remove all traces of the old whitemetal liner. Care should be taken not to overheat the liner and cause distortion.

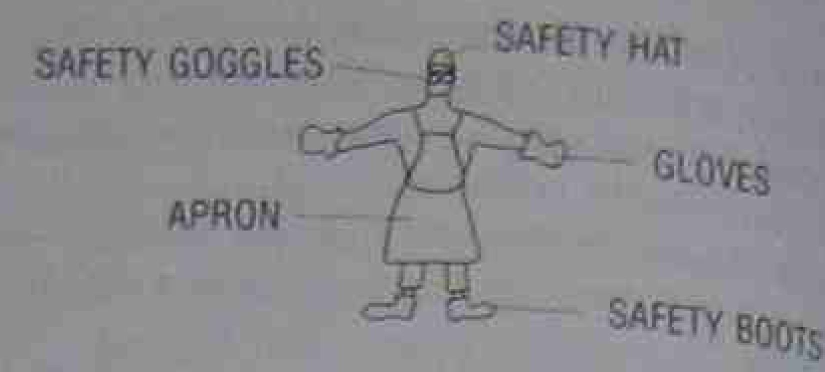


Fig. 6-16 Protective clothing and safety glasses.

3 In order to be properly tinned the steel sh should be chemically clean and slightly etch Clean first in a metal cleaning solution, rinse then dip into an acid solution for a few minu Give a final rinse with clean water.
4 Once cleaned, immerse the shell in hot solder flux at 65°C (150°F).
5 Next tin the shell by dipping it into molten ti 285°C (550°F) for steel and cast iron or 50% 50% lead solder for bronze liners.

Hold the shell submerged just long enough it to reach the bath temperature. Examine tinned surface and if any areas have not tinn completely, wire brush them and reflux a reimmerse them.

N.B. Cast iron is difficult to tin because of t presence of graphite and absorbed oil. This c be overcome by burning off and brushing, or preparation in a molten salt bath.
6 If the bearing is too large or a tin bath is n available then tinning may be accomplished heating with a blowtorch or gas flame and using stick of tin or solder.
7 Set up the bearing shell in a fixture comprising plate and mandrel or core as shown in Fig. 6-1

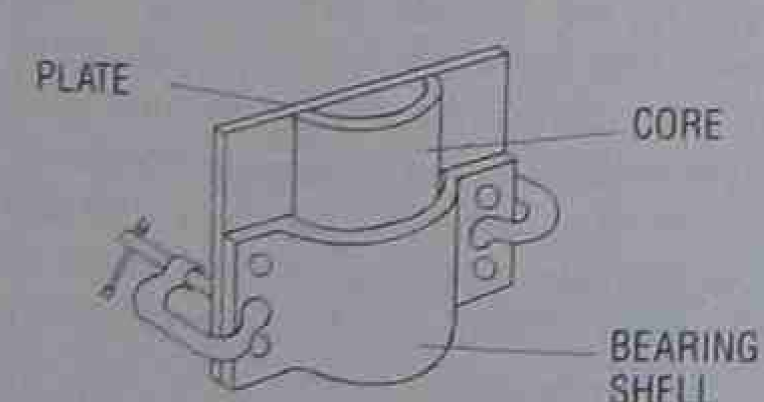


Fig. 6-17 Setting up the mould.

If the bearing is too large to be removed an must be lined on site then register plates must b set up at each end to form the inside radius shown in Fig. 6-18.

In some cases it may be convenient to use th journal as a mandrel in which case it should b coated with graphite to prevent adhesion of th whitemetal.

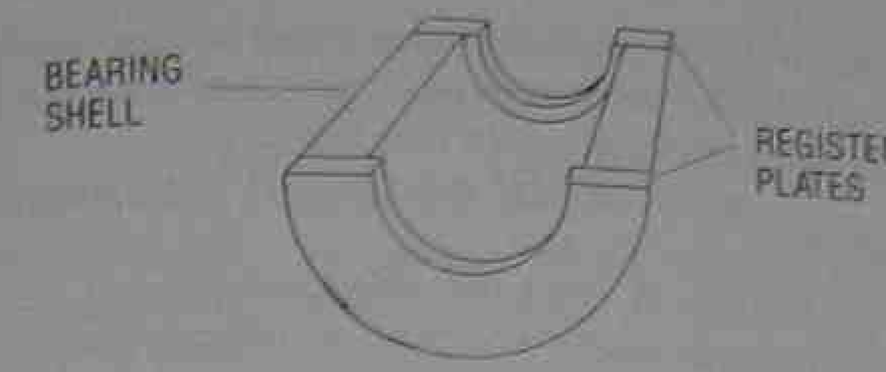


Fig. 6-18 Using register plates to form the inside radius.

The bearing shell should be preheated to 120°C (250°F) immediately before the liner is poured.

8 The whitemetal should be heated to the recommended minimum pouring temperature as given by the relevant standards or by the manufacturers.
9 The liner should be poured in one continuous operation after the whitemetal has been thoroughly mixed and stirred and the surface fluxed and cleared of dross.
10 It is desirable to cool the steel quickly if possible as this minimises shrinkage porosity and improves adhesion. This should be done carefully by water-spraying from the back and bottom of the shell.
11 After the bearing has cooled, dismantle the fixture, check the lining for porosity and remove any excess metal.
12 Check the quality of the bond between liner and shell by tapping the shell with a small hammer. It should give a clear, ringing sound. A cracked sound indicates that the bond is poor.
13 Clamp the two halves of the bearing together and set them up in the lathe ready for machining the inside diameter. Check the journal size and the manufacturer's recommendations and machine the bore of the bearing with the best possible finish.
14 Remount the bearing after applying a coating of Prussian blue (bearing blue) to the journal. Turn the shaft over a couple of times and then inspect the bearing's surfaces. The blue should transfer reasonably evenly to the bearing's surfaces. If it is only present on small areas this means that the bearing is riding on high spots. See Fig. 6-19.

Fig. 6-19 Testing bearing fit with bearing blue.

Scrape the bearing surface using proper scraping tools to remove the high spots. Reassemble and recheck. Repeat the process until the fit is satisfactory.
15 If oil grooves need to be machined, ensure that the edges are chamfered afterwards.

The procedure for relining solid, whitemetal-lined bushings is identical to the above, except that the set-up for pouring is slightly different. See Fig. 6-20.

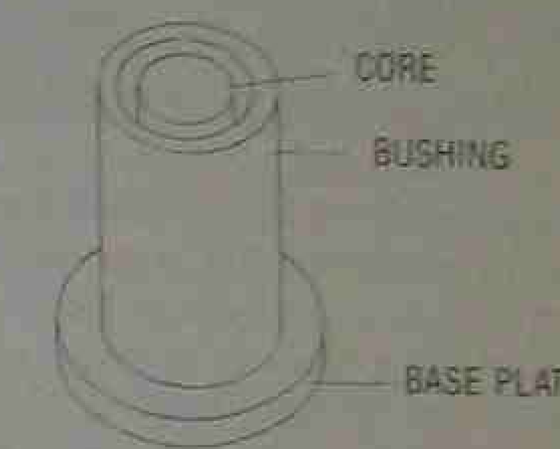


Fig. 6-20 Pouring set-up for relining solid, whitemetal-lined bushings.

Start-up

The following procedures should be adopted during the start-up of new or reconditioned plain bearings.

1 Before the shaft is run, check the alignment with bearings in position. Bluing of the journal will indicate whether the shaft is running true in the bearing. If the wear spots are at one end of one of the bearing halves and at the opposite end of the other as shown in Fig. 6-21 then misalignment is indicated.

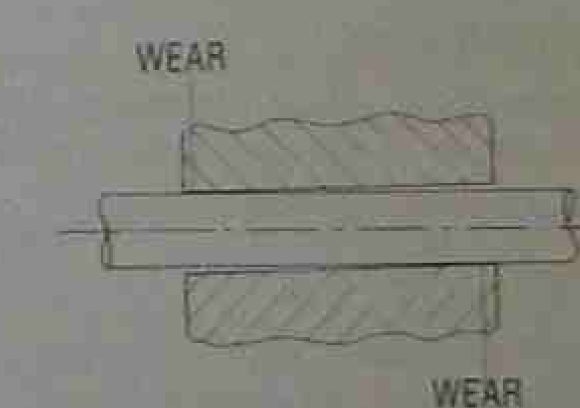


Fig. 6-21 Bluing indicates misalignment.

Alignment should be corrected by shimming the bearing housing.
2 Once the alignment is true, pre-lubricate the bearing surfaces before final assembly to ensure they do not start up dry.
3 Run the shaft and check for noise and vibration.

Check bearing temperatures every half hour for the first two or three hours of operation. If the temperatures are excessive, shut down the machine and inspect the bearing surfaces for wear-in characteristics. Further alignment or scraping may be required before re-starting.

4 Once the operation of the bearing is satisfactory, ensure that it is regularly supplied with the correct lubricant.

FAILURE PATTERNS

Before replacing a damaged bearing it is important to know the precise cause of failure so that the problem can be properly corrected, otherwise similar failures may keep occurring due to the same cause.

Bearing problems and failures can be analysed in terms of effects or symptoms or in terms of causes. From the point of view of fault detection it is useful to look first of all at symptoms and then to trace the possible causes. Every failure is different in some respects so it is extremely difficult to establish definite rules about causes. The same symptom may result from many different causes and systematic troubleshooting techniques are necessary to eliminate all but the real cause. However, it is possible to establish some general rules about the relationship between symptoms and causes that can serve as a guide in the troubleshooting process.

There are two levels at which the symptoms of bearing failure can be detected. The first is what might be called the level of **external** symptoms which are exhibited while the bearing is in operation. The second can be called the level of **internal** symptoms which can only be observed when the bearing is shut down, dismantled and inspected.

Operating symptoms

When the condition of a bearing begins to deteriorate, whatever the cause, then it may exhibit one or all of the following symptoms:

Overheating

The temperature of a plain bearing during operation is a vital indication of its condition. A properly lubricated bearing in good condition should not generate excessive temperatures and should run cool enough to touch. The most efficient way to check bearing temperature is to use a portable sensing device and to measure the normal running temperature of the bearing once it has been run in. Routine checks at regular intervals will then show up any sudden or dramatic increases from the base level.

When using a contact pyrometer or thermocouple to measure temperature it is important to ensure good contact is achieved at a position that reflects the temperature of the bearing.

Vibration

A plain bearing in good condition should run very smoothly, even more so than a rolling element bearing. As clearances increase, some vibration may develop which will tend to accelerate the deterioration of the bearing. As with temperature, hand contact is usually sufficient to establish whether unnecessary vibration is present. It is advisable where possible, and especially for critical equipment items, to establish a normal operating level of vibration using a vibration monitor (see Chapter 10) and to make routine checks to establish when a deviation occurs. As with temperature, a dramatic change in operating characteristics indicates a need for further investigation.

Noise

A plain bearing should not produce high noise levels if it is in good condition. If the noise level increases it may be due either to misalignment or excessive wear. Foreign particles trapped in the bearing may also cause noisy operation. In a reciprocating mechanism the big and little end bearings produce a 'knocking' sound when wear becomes excessive. Noise levels can be measured independently although vibration monitoring is more common. Technicians should develop an 'ear' for the sound of machinery so that changes in both level and characteristics can be detected.

Seizure

Total seizure of a plain bearing is, of course, a symptom which cannot be ignored. If the three characteristics referred to above are carefully monitored then seizure should not occur. Seizure is usually a result of excessive temperature build-up due to lack of lubrication and if temperatures are routinely checked then seizure should be avoided. The danger in allowing undetected deterioration to lead to ultimate seizure of a bearing is that such a sudden catastrophic failure is likely to also cause severe damage to the housing and other machine components. The excessive temperature build-up can result in fire and the sudden shutdown of the machine may cause severe or even disastrous process problems.

Symptoms often occur simultaneously and the presence of one often leads to the others appearing. This means that cross-checks can be made to establish whether a problem exists. For example, if it is suspected that the noise level is increasing, then a check on operating temperature may confirm that the condition of the bearing is deteriorating.

Symptoms found on inspection

The operating symptoms alone rarely, if ever, provide sufficient evidence for the exact cause of failure to be determined. It may be possible, once the external symptoms of failure are detected, to attribute them to the general cause of 'wear' and then merely to proceed to replace the bearing and re-start the equipment. This is a short-sighted approach however, and runs a strong risk of a premature repetition of the same problem. In order to be more precise about the cause of failure, it is necessary to dismantle the bearing and inspect the bearing surfaces to establish what internal symptoms exist. These can generally be identified according to one of the following categories.

Wear

Any two surfaces that are in constant sliding contact, even though properly lubricated, will eventually wear out. This process will, in most cases, be apparent due to a general loss of metal across the bearing surface. The bearing surface may still be reasonably good and the wear only clearly evident by the increased clearances established by a dimension check. Wear occurs due to a combination of the processes of adhesion and abrasion described in Chapter 5.

Mobil

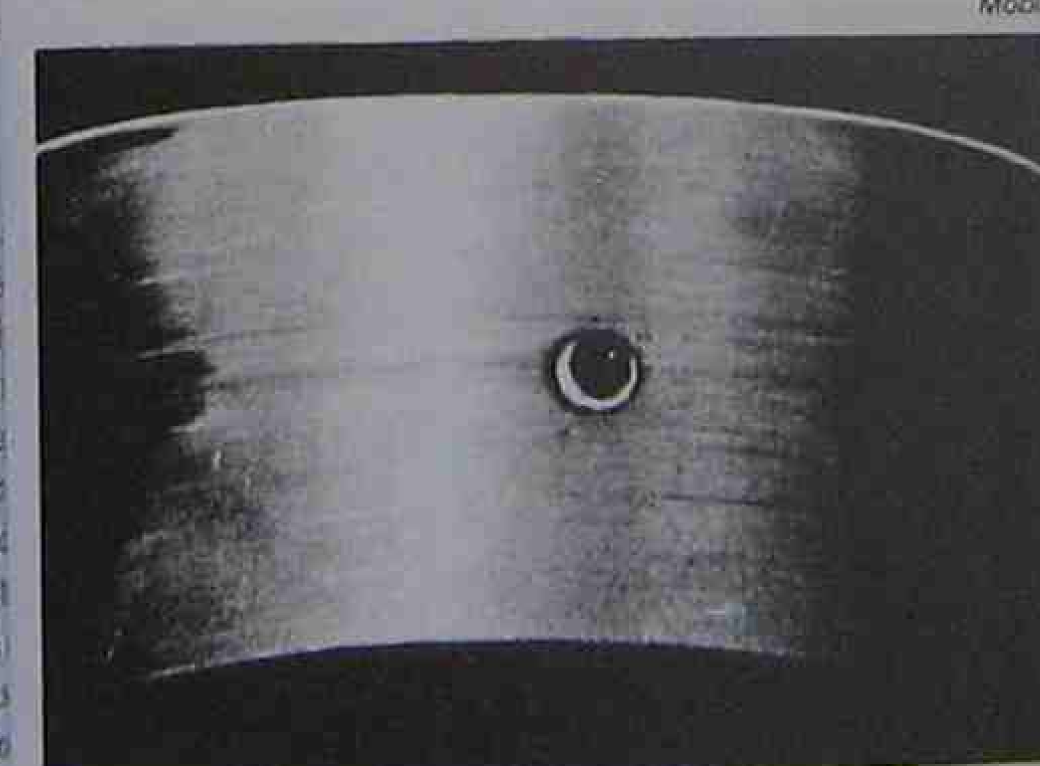


Fig. 6-22 A typical example of general wear.

Fatigue

Fatigue is caused by cyclical stressing due to fluctuating loads and although not as common in plain bearings as in rolling element bearings it can often occur when load or speed becomes excessive.

Mobil

Fig. 6-23 A typical pattern of fatigue failure.

Galling

Sometimes the adhesion due to cold welding of two surfaces in contact becomes so great that grain displacement of the bearing material takes place leaving pits or craters. This is usually associated with high loads which cause excessive pressure between points of contact.

Mobil

Fig. 6-24 Galling of the bearing surface.

Scoring

When excessive amounts of dirt or large dirt particles are present due to contamination of the lubricant or

Mobil

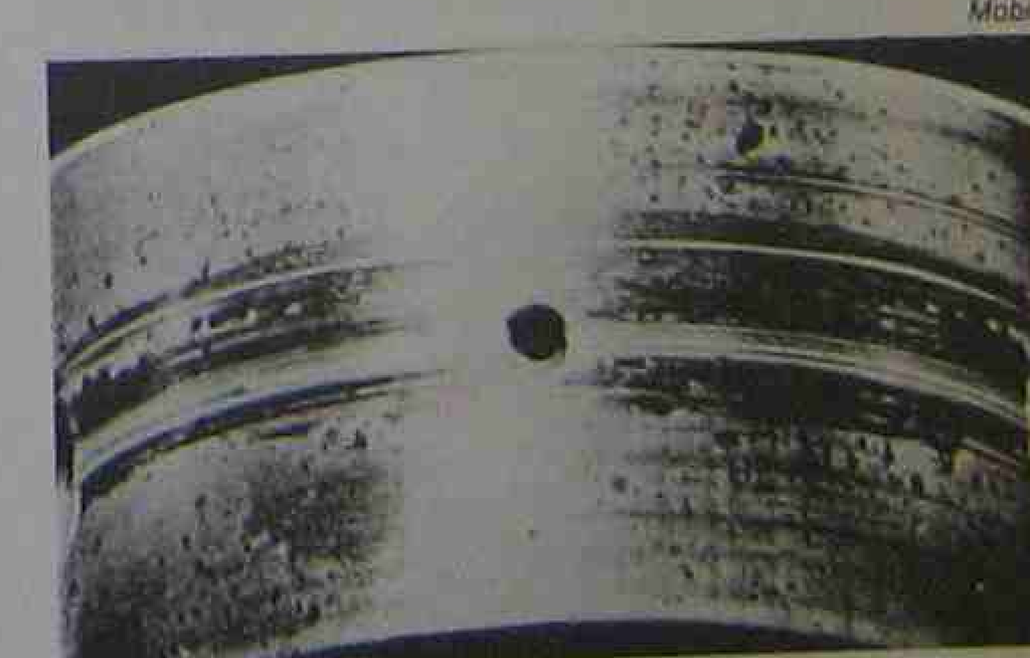


Fig. 6-25 Scoring of the bearing surface.

because of abrasive wear then scoring and erosion of the bearing surface will result.

Wiping

When there is insufficient clearance in the bearing or overheating occurs due to inadequate lubrication then a superficial melting of the bearing surface and a flow of material occurs.

Mobil

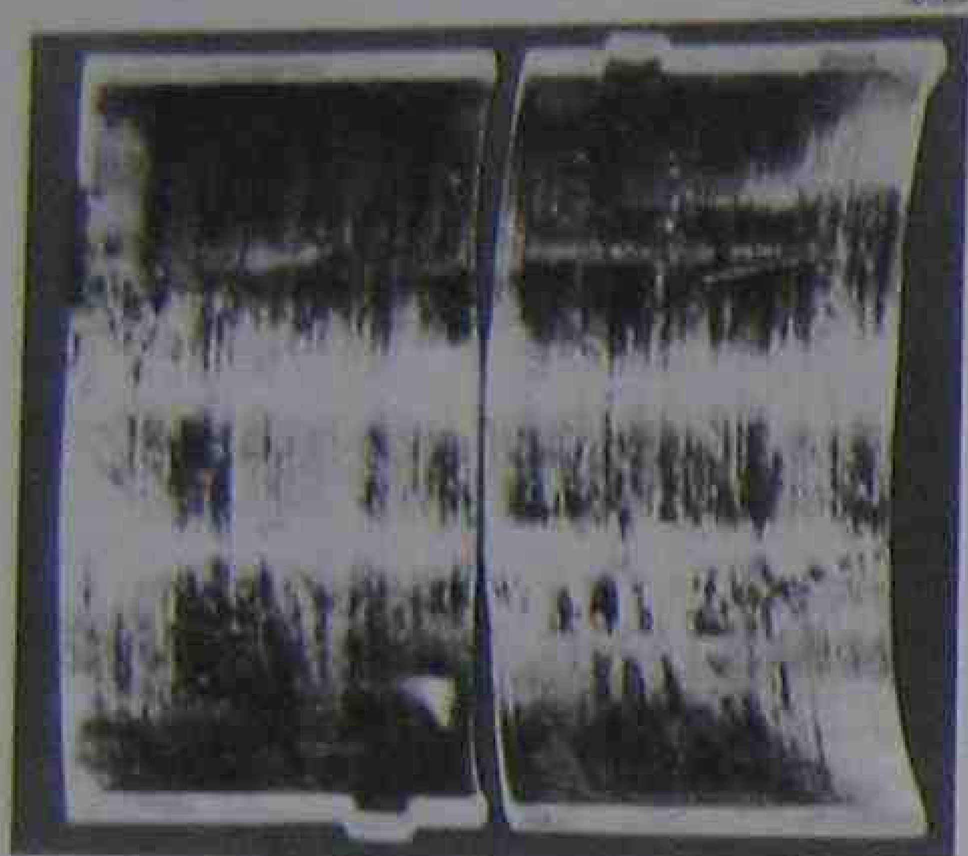


Fig. 6-26 Wiping of the bearing surface.

Fretting

Fretting is most likely to occur in plain bearings between the liner and the housing when the interference fit is inadequate and slight vibratory movement occurs. A characteristic fine brown debris of oxidation material is the usual evidence of fretting or false brinelling as it is sometimes called.

Glacier Metal

Fig. 6-27 Fretting between liner and shell.

Fretting may also occur between the bearing surfaces while the shaft is stationary due to vibrations transmitted from external sources.

Glacier Metal

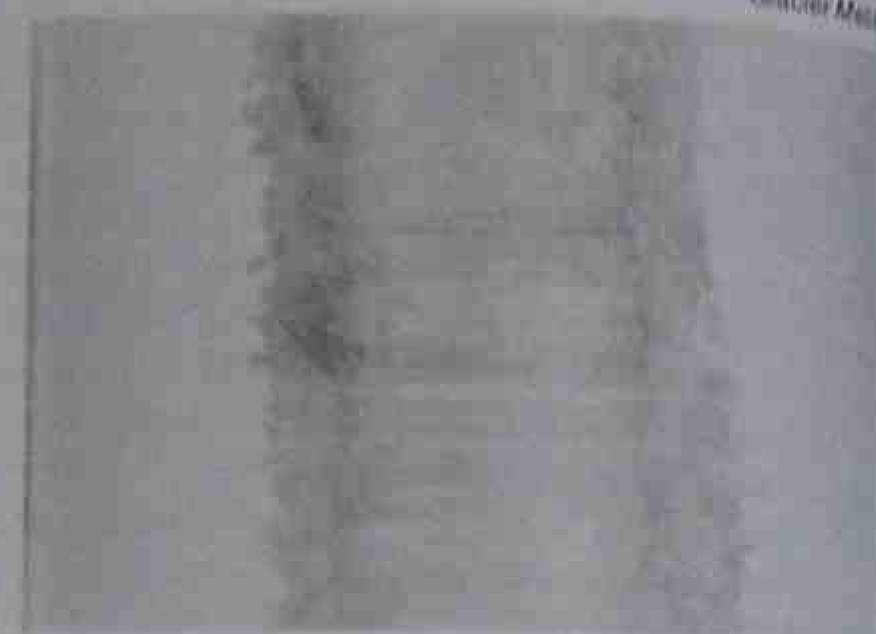


Fig. 6-28 Fretting due to external vibration.

Pitting

Local loss of metal in the form of pits or deep crate can result from a number of causes. This c sometimes be due to stray electric currents that across the bearing and cause pitting of both th surface of the bearing and the journal.

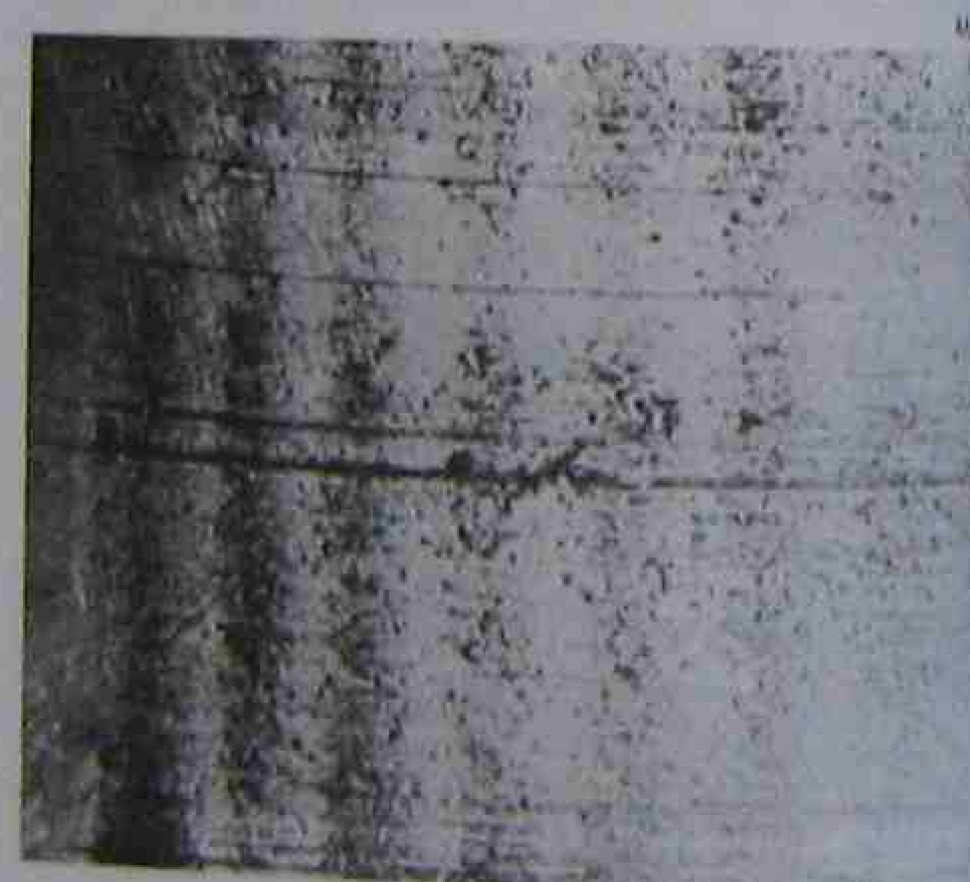


Fig. 6-29 Pitting of bearing surface due to stray electric current

Corrosion

Etching and surface discoloration are all evidence corrosion, and a typical example is shown in Fi 6-30.

Mobil

Fig. 6-30 Typical evidence of corrosion.

Causes of failure

In theory there are an almost infinite number of possible causes of bearing failure but in practice it has been found that most failures can be attributed to one of several quite specific causes. In order of significance, these are as follows:

Dirt

The presence of dirt is the single most destructive cause of plain bearing failure. Large dirt particles which become embedded in the bearing surface can cause excessive wear and reduce the life of the bearing and journal. An excessive quantity of fine dirt particles will cause scoring and rapid wear. Dirt particles can also become trapped between the bearing liner and the housing. When the particles are large this can cause distortion of the liner, reduction of clearances and an impairment of heat transfer. Over-heating and, in some cases, fatigue failure are the likely results.

Dirt particles are most likely to become entrapped during the assembly procedure and the technician must take all possible care to ensure that this does not occur. Contaminated lubricant may also be a source of dirt and recirculating systems should be properly filtered and the filter changed or cleaned at regular intervals. In a crankcase the failure of one bearing may result in contamination of the lubricant by particles of bearing material which may then endanger other bearings. If this condition is suspected then more frequent lubricant changes may be necessary to protect the other bearings.

Inadequate lubrication

Oil starvation is a common cause of bearing failure and may occur for various reasons. The period immediately after overhaul is always critical and it is recommended that the lubrication system be primed to ensure lubricant is present on initial start-up. The lubrication system should be kept under close observation for the period after start-up to ensure that pressures and temperatures are established according to design.

During normal operation, several factors may affect the supply of oil. Leaks, oil pump failure, blocked filters and blocked oil passages are all potential problems. Care must be taken when installing bearing liners to ensure that the oil hole is in line with the oil passage in the housing. It should also be recognised that once a bearing becomes badly worn the hydrodynamic wedge cannot be effectively established and the wear rate will further increase due to inadequate lubrication.

Bearing failure may also occur due to use of an unsuitable lubricant due either to error or poor selection. Lubricant suppliers and equipment manufacturers should be consulted if there is reason for concern about the suitability of the lubricant used. Lubricants should be stored properly and clearly marked so that the likelihood of error is reduced.

Improper assembly

As well as ensuring that cleanliness is observed during the assembly procedure, it is also important to ensure that the bearing is assembled correctly in all other respects. The main considerations are that the bearing should be fitted square and secure in its housing and that it does not get physically damaged during assembly.

Failure to ensure proper assembly may result in uneven or rapid wear and overheating.

Misalignment

If the journal is not properly aligned in the bearing uneven bearing wear and possible damage to the journal will result. The alignment procedures described earlier in this chapter should be followed carefully. The alignment of shaft couplings should also be checked to ensure that misalignment there does not affect the true running of the journal in the bearing. (See Chapter 8.)

Overload

Equipment overloads can occur for many different reasons and are particularly likely to occur when extra output is required. The most common symptoms of overload are overheating and fatigue. When these are observed a review of operating conditions is advisable in order to establish whether any overloading has occurred.

Increased speeds, feed rates, levels, operating temperatures and pressures, etc. can all create overload conditions which lead to machine elements such as bearings having to perform above design limits.

Moisture

The presence of water due to condensation or lack of effective sealing may cause corrosion of the bearing surfaces and the formation of rust.

Lubricant breakdown

Breakdown of the lubricant, because of the formation of organic acids due to oxidation or the presence of specific elements in the oil such as sulphur compounds, may lead to corrosion of the bearing surfaces. An analysis of the lubricant after it has been in service may be necessary to establish whether deterioration has taken place. A review of the oil replacement period may be sufficient to av recurring problems. This should be done consultation with the lubricant supplier.

Summary of the common symptoms and causes of plain bea failure

Symptoms		Causes
Operating	*Inspection*	
Noise	Wear	Dirt
Overheating	Fatigue	Lack of lubricant
Vibration	Galling	Improper assembly
Seizure	Scoring	Misalignment
	Wiping	Overload
	Fretting	Moisture
	Pitting	Lubricant breakdown
	Corrosion	

6.2 ROLLING ELEMENT BEARINGS

PRINCIPLES OF OPERATION

Rolling element bearings, or anti-friction bearings as they are often called, differ from plain bearings in that they incorporate rolling elements, either balls or rollers, which are held between two raceways as shown in Fig. 6-31. A soft metal cage or retainer separates the rolling elements and ensures that they are evenly spaced, but does not carry any load. As a result of the relatively small area of contact between the rolling elements and the races, the frictional resistance to relative motion is comparatively low.

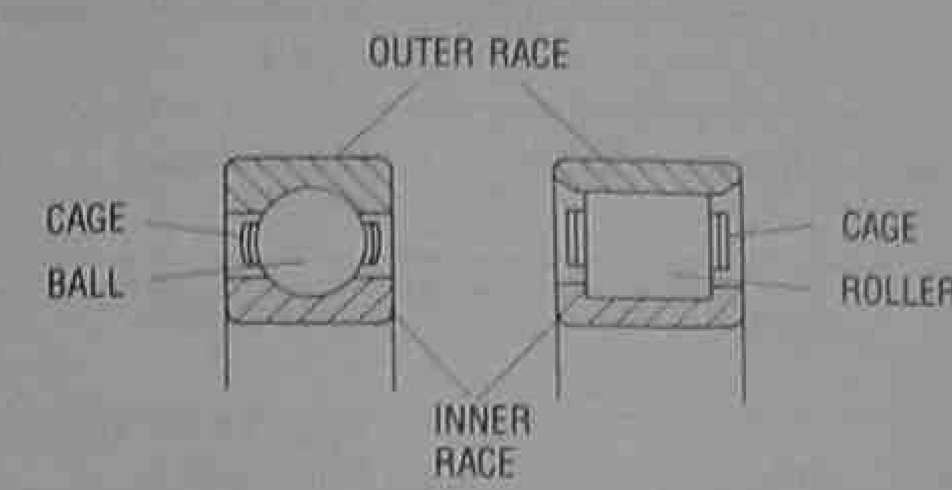


Fig. 6-31 Typical ball and roller bearings.

Because the relative motion between the moving elements is one of rolling rather than sliding, the material requirements are quite different from plain bearings. Instead of requiring bearing materials to be soft in comparison with the journal, both the rolling elements and the raceways of rolling element bearings are made of specially hardened steel. The harder the elements, the smaller the indentation and deformation of the surface and hence the lower the frictional resistance. However, although sliding action between the two moving elements is much reduced, in comparison with plain bearings, it cannot be eliminated entirely, and therefore lubrication is just as critical to rolling element bearings as to plain bearings.

Rolling element bearings are manufactured to very precise standards of accuracy to ensure good performance and long life. Tolerances are held to a thousandth or in some cases a tenth of a thousandth part of a millimetre in order to minimise run-out and to ensure the proper radial and axial clearances required for smooth operation.

The choice of balls or rollers as the rolling elements depends on the operating conditions and the following considerations.

Ball bearings
can operate at higher speeds without overheating,
are less expensive for lighter loads,
have lower frictional resistance at light loads,
are available in a wider range of sizes.

Roller bearings
can carry heavier loads,
are less expensive for heavier loads and larger sizes,
are superior under shock or impact loading,
provide greater rigidity.

The important physical difference between balls and rollers that gives rise to this difference in performance is the variation in the area of contact between the rolling elements and the raceway. The ball has a small area of contact which resembles point contact depending on how much deformation of the ball and raceway occurs. The roller, by comparison, has a greater area of contact which resembles line contact. The difference is shown in Fig. 6-32.

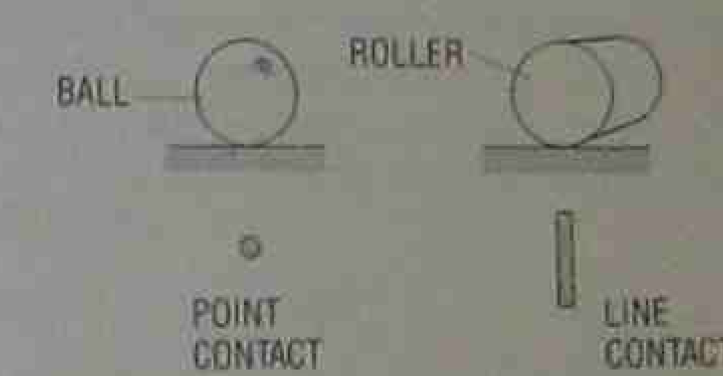


Fig. 6-32 Difference in area of contact between ball and roller bearings.

The greater area of contact of the roller makes it better able to carry heavy loads and withstand impact, but tends to increase frictional resistance at low loads. Ball bearings generate lower resistance at low loads but deform more significantly at high loads which causes frictional resistance to increase. Ball bearings can run at higher speeds due to the smaller area of contact.

TYPES OF BEARINGS

Journal bearings

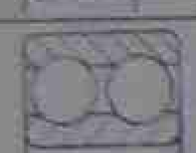
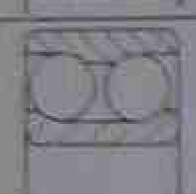
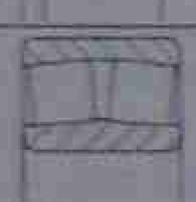
	Type	Radial load	Axial load	Speed	Misalignment tolerance	Axial location
1 	Single row deep groove ball	very good	good	excellent	good	both directions
2 	Single row angular contact ball	very good	very good	good	fair	one direction
3 	Magneto	fair	fair	very good	poor	one direction both directions when used in pairs
4 	Self aligning ball	good	very poor	very good	excellent	both directions
5 	Double row deep groove ball	very good	good	excellent	fair	both directions
6 	Double row angular contact ball	very good	very good	good	fair	both directions
7 	Roller	excellent	very poor	very good	poor	—
8 	Needle roller	excellent	very poor	fair	very poor	—
9 	Tapered roller	excellent	very good	good	poor	one direction
10 	Spherical roller	very good	good	fair	excellent	both directions

Fig. 6-33 Types of rolling element journal bearings.

Thrust bearings

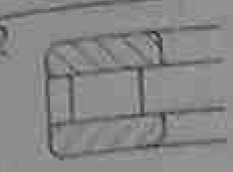
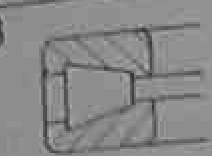
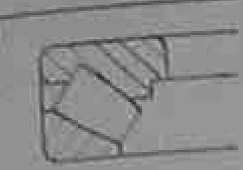
	Type	Radial load	Axial load	Speed	Misalignment tolerance	Axial location
11 	Single row ball	—	very good	fair	poor	one direction
12 	Cylindrical roller	—	excellent	poor	very poor	one direction
13 	Tapered roller	—	excellent	poor	very poor	one direction
14 	Spherical roller	poor	very good	poor	very good	one direction
15 	Double row ball	—	good	poor	poor	both directions

Fig. 6-34 The types of rolling element thrust bearings.

Linear bearings

The construction of slider or linear motion bearings is slightly different from that of journal and thrust bearings.

Ladder bearings

These consist of two hardened steel plates that are separated by a series of balls or rollers held in a cage as shown in Fig. 6-35. They may be used either vertically or horizontally.

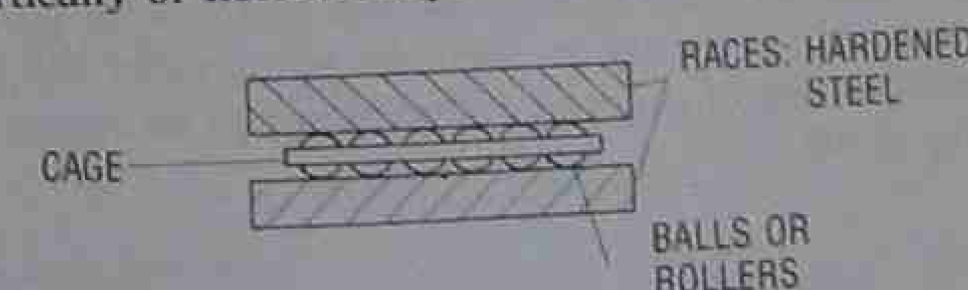


Fig. 6-35 Ladder bearing.

Recirculating ball and roller bearings

In these units the rolling elements are only in contact with the way or slide for a specific distance, then they leave the load area, drop into a return channel and return to the opposite end of the assembly to be fed back into the loaded area. The roller version can carry greater load than the ball but tends to have a shorter life.

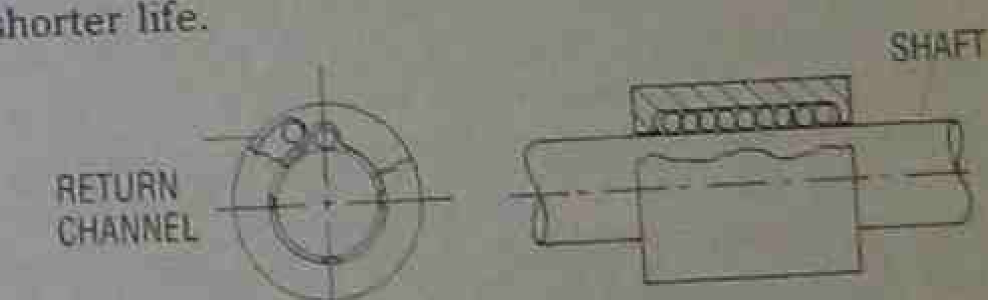


Fig. 6-36 Recirculating ball bearing.

Recirculating roller-chain bearings

An endless chain assembly of rollers is mounted on a solid bearing race attached to a carriage. The rollers are often shaped to fit a ground shaft or way. As the carriage or slide moves, the bearing races roll on the concave rollers which are in contact with the shaft.

Single mounted units have little resistance to side loads and hence two units are often mounted in a V configuration.

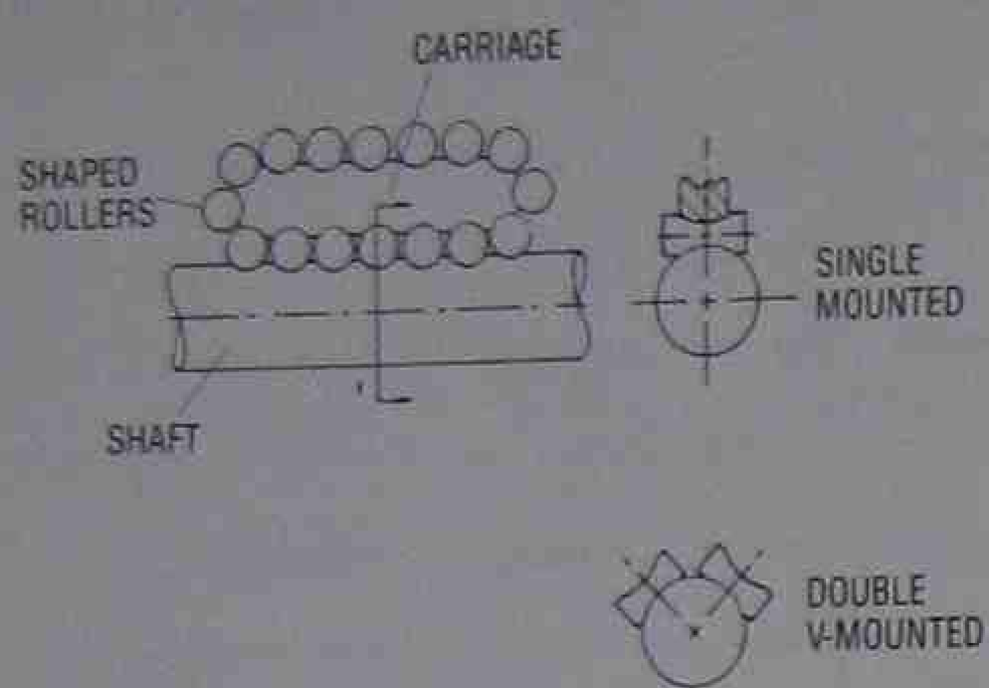


Fig. 6-37 Recirculating roller-chain bearing.

BEARING ASSEMBLY DESIGN

Shaft and housing fits

As a general rule a rolling element bearing should be installed so that its rotating ring is an interference fit and the stationary ring is a slip or push fit. An interference fit is one which requires a press or a driver or, if the fit is especially tight, may require hot mounting, whereas a slip or push fit can be slid into place by hand. Sometimes if loads and speeds are high, both races may be fitted with an interference fit.

The fit required for a particular bearing should be established by consulting the manufacturer or the standards of the ISO (International Standards Organisation).

Thermal expansion

When a shaft is mounted on rolling element bearings some provision may have to be made for thermal expansion of the shaft in the axial direction while at the same time ensuring that all elements of the machine are maintained in the correct relative positions.

In the common case where the shaft rotates and is held by more than one bearing it is normal to have one bearing held in the housing and the other floating and able to accommodate axial movement of the shaft due to expansion or contraction. A typical arrangement is shown in Fig. 6-38.

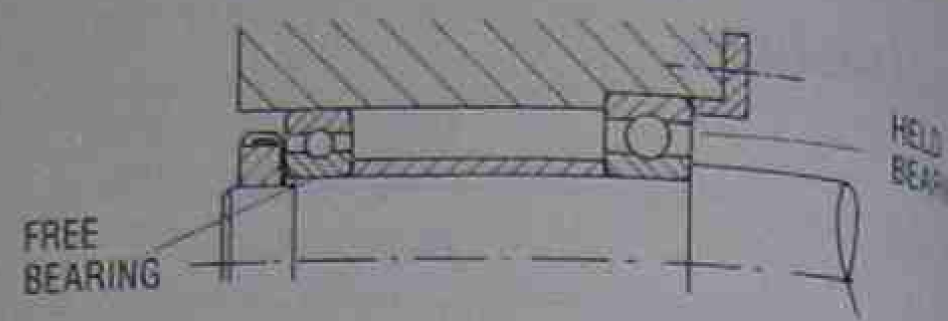


Fig. 6-38 Typical arrangement of free and held bearings.

Types of mountings

There are several standard arrangements that a used to fix a bearing to a shaft so that its axial posit is maintained.

Shaft nut and locking washer

This is the most common arrangement and can used for both single and double row radial bearin It employs a specially designed nut and a tab wash as shown in Fig. 6-39.

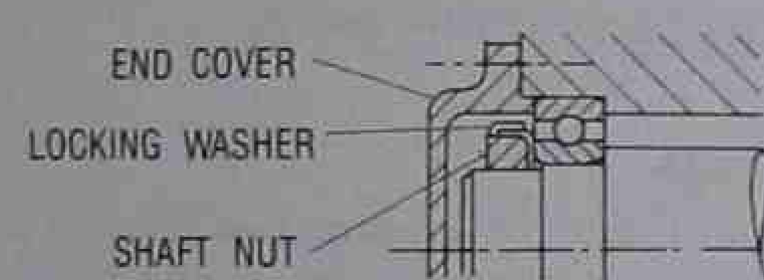


Fig. 6-39 Shaft nut and locking washer arrangement.

Slotted nut and pin

Sometimes a castle nut or specially designed slotte nut may be used in conjunction with a split pin o cotter pin in order to provide adjustment for th inner cones of tapered roller bearings as shown i Fig. 6-40.

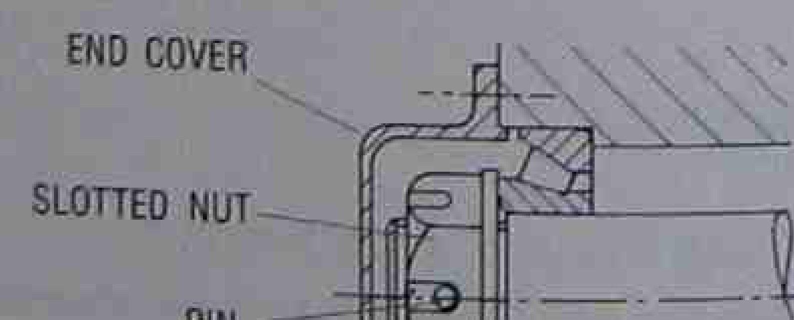


Fig. 6-40 Slotted nut and pin.

End plate and locking bolts

This type of device is often used when the bearing i mounted on the end of the shaft as shown in Fi 6-41.

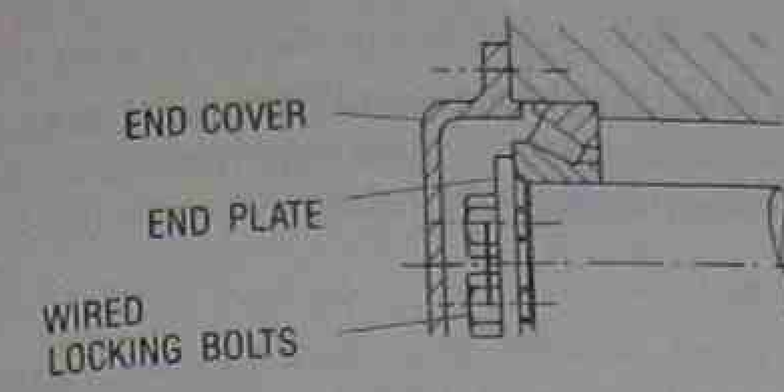


Fig. 6-41 End plate and locking bolts.

End cap and shims

This arrangement is also used for tapered roller bearings or for angular contact bearings where preload has to be set by adjusting shims.

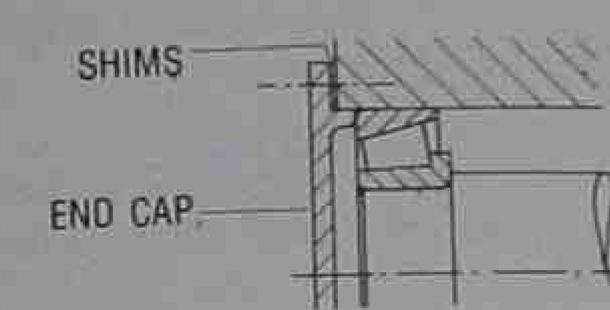


Fig. 6-42 End cap and shims.

Snapring

When it is not suitable for the housing to contain a shoulder then a snapring can be used to locate the bearing in the housing as shown in Fig. 6-43.

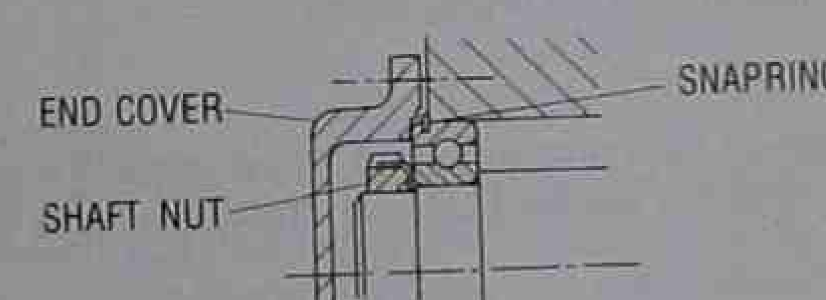


Fig. 6-43 Snapring.

Adapter sleeve

Spherical roller bearings are usually made with a tapered bore and are often mounted as 'floating' bearings on an adapter sleeve as shown in Fig. 6-44.

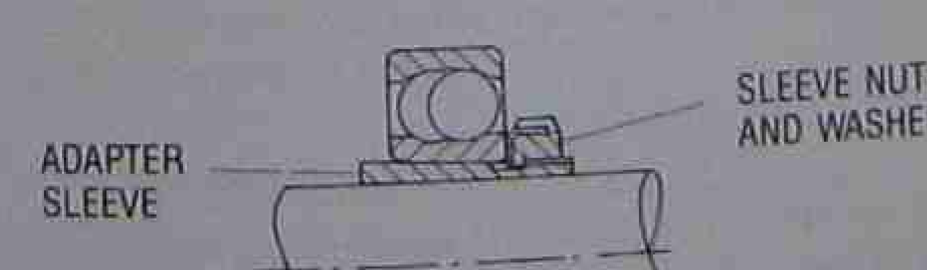


Fig. 6-44 Adapter sleeve.

Withdrawal sleeve

A withdrawal sleeve of the type shown in Fig. 6-45 is also used with self-aligning bearings for ease of dismounting. In this case the bearing must be mounted against a shoulder.

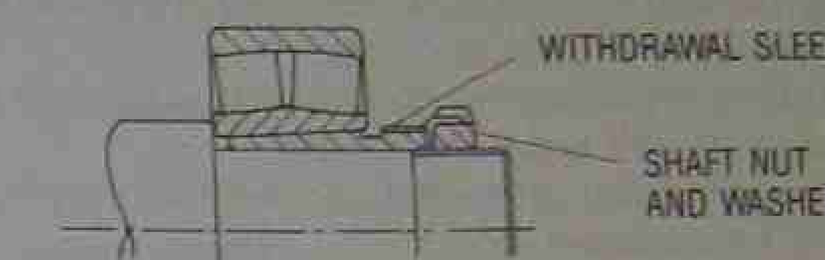


Fig. 6-45 Withdrawal sleeve.

MAINTENANCE PRACTICES

General

Although there are specific maintenance practices associated with particular types of rolling element bearings there are a number of general recommendations that apply to all types. These should be kept in mind whenever bearing maintenance is carried out.

- As with plain bearings, cleanliness is vital to the satisfactory operation of rolling element bearings. Bearings should be kept free of dust and dirt particles and be kept dry and protected at all times.
- Bearings should be handled with clean, dry hands or with clean gloves.
- It is important when mounting and dismounting bearings that the correct tools are used and that they are in good condition. Makeshift arrangements or badly worn tools are likely to lead to damage to the bearing or bearing assembly.
- Bearings should be wrapped in oil-proof paper when not in use.
- Only clean solvents and flushing oils should be used for bearing cleaning.
- Before installing a rolling element bearing both the housing and shaft should be carefully inspected for burrs, nicks and scratches that may interfere with the fitting of the bearing.
- Bearings that are dry and unlubricated or have not been cleaned should not be spun. If compressed air is used to clean a bearing care should be taken to avoid spinning the bearing.
- Cotton waste or dirty rags should not be used to clean bearings, but only clean, lint free rags.
- The slushing compound used to protect a bearing

in storage need not be removed if it is petroleum-based unless it has gone hard or become contaminated.

- The force applied when mounting or dismounting a bearing should always be applied to the ring with the interference fit and should never be applied in such a way that the force is transmitted through the rolling elements.
- Never strike a bearing directly with a hammer or mallet or with a soft metal drift.
- Remember that all sealed and shielded bearings must be mounted cold.
- Bearings should never be heated with a naked flame.

Tools and equipment

The maintenance of rolling element bearings requires specialised tools and equipment and without these many tasks are extremely difficult. The following items should be considered essential:

Mounting dollies and sleeves

When mounting bearings which are a press fit in the housing or on a shaft it is important that pressure is applied evenly around the bearing ring. If the bearing is cocked due to uneven force then surfaces can be damaged and the bearing ring distorted. A set of mounting dollies and sleeves will ensure that bearings are driven home evenly. These can be purchased direct from the bearing manufacturers or can be made up to suit a particular application.

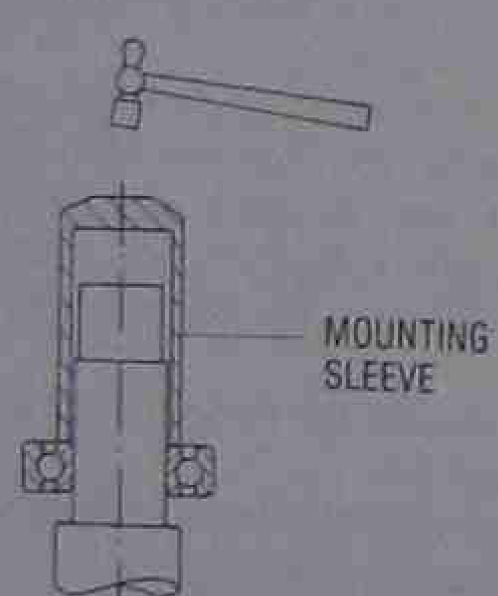


Fig. 6-46 A typical example of a mounting sleeve.

Hammers

Hammers should be made from steel or a soft material and should be free from burrs. Copper or synthetic resin are suitable materials for soft hammers but lead and tin should not be used. Wooden mallets should not be used because of splinters.

Drifts

Only steel drifts should be used and they should be used for bending or straightening tabs on lock washers or for driving shaft nuts.

Arbor press

An arbor press ensures that bearings are driven evenly especially when the fit is tight. It should be used in preference to other methods wh[ere] practicable.

Pullers

Bearing pullers are essential for dismounting a[nd] there are several types available.

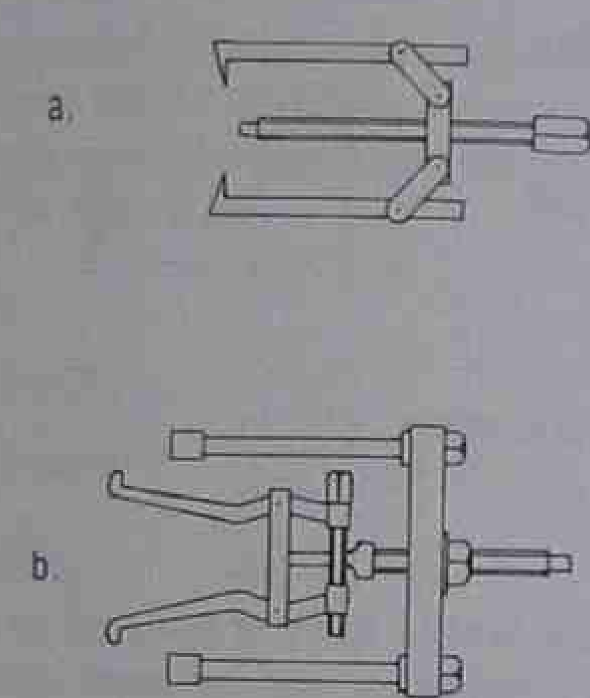


Fig. 6-47 Typical examples of bearing pullers.

Pullers should be maintained in good condition [so] that they operate freely and the claws are free [of] burrs.

Lifting gear

When handling large, heavy bearings it is importa[nt] to have suitable lifting gear available such as tongs [or] slings.

Hook and impact spanners

Hook and impact spanners are needed for use wi[th] adapter sleeve nuts, withdrawal nuts and shaft nuts.

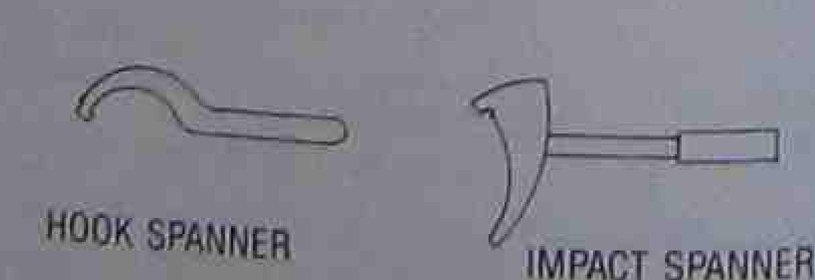


Fig. 6-48 Hook and impact spanners.

Induction heater or oil bath

Modern induction heaters, such as the type shown in Fig. 6-49, are much cleaner and easier to use than an oil bath when bearings have to be heated for a shrink fit.

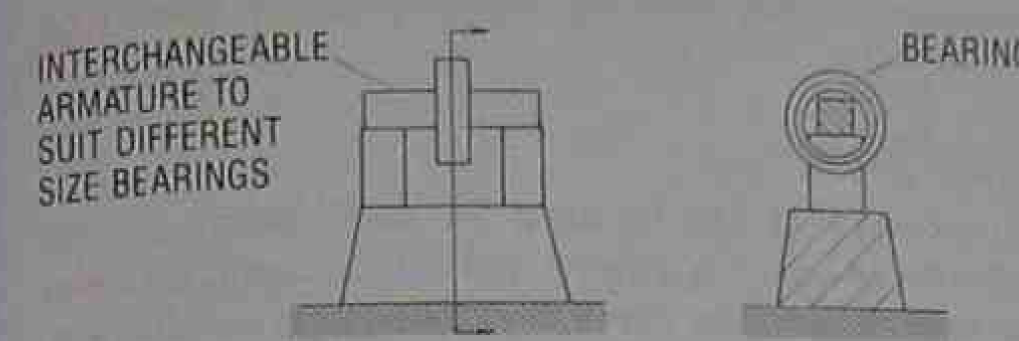


Fig. 6-49 Typical induction heater.

When induction heaters are not available, the traditional oil bath should be used.

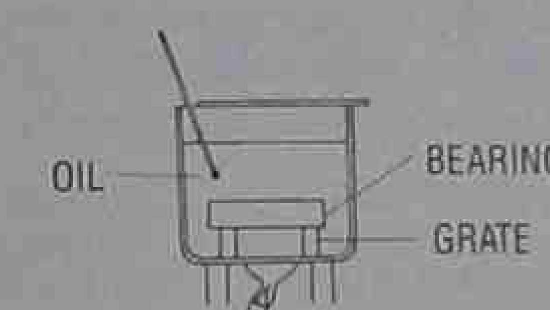


Fig. 6-50 Oil bath.

Gauges

A set of various types of gauges is necesssary for measuring diametral clearances, housing bores and shaft diameters. An ordinary set of feeler gauges is sufficient for measuring clearances but a special set of bore gauges may be required for measuring the bore of housings. Alternatively, standard inside and outside micrometers can be used for measuring diameters.

The above tools represent the minimum required for bearing maintenance. Where bearings are mounted on tapered shafts or on adapter or withdrawal sleeves some hydraulic equipment may also be required.

Oil injection equipment

Where shafts have been provided with oil ducts, hydraulic pressure supplied by an oil injection pump can be used for mounting and dismounting both straight bore and tapered bore bearings. A typical situation is shown in Fig. 6-51.

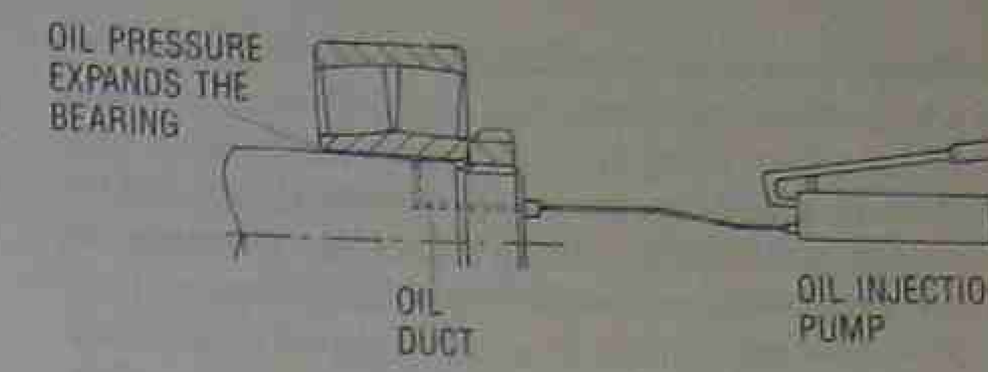


Fig. 6-51 Mounting by oil injection.

Hydraulic nuts

These tools consist of a nut that incorporates a groove in which an annular piston moves when oil pressure is built up behind it. Hydraulic nuts can be used for both mounting and dismounting and employ the same oil injection equipment referred to above.

Fig. 6-52 A typical hydraulic nut.

As well as the above equipment, there are a number of standard items which may be required from time to time and should be readily available. These may include:

straight edge
marking blue
dial gauge
plumb line
solvent
clean rags, etc.

Tools should be kept clean, and in good condition. Bearings are precision items and tools and equipment should therefore be maintained, handled and stored accordingly.

Mounting procedures

Preliminary steps

Before starting the mounting procedure for any bearing, the following points should be considered:

- Check the manufacturer's drawing of the bearing arrangement and ensure that it is clearly understood. Establish which fit is the interference fit.
- Check that the necessary tools and equipment are available after determining the mounting procedure to be used.

- Select a suitable working environment that is clean and adequately lit.
- If an old bearing is to be remounted make sure that it has been properly cleaned and lightly coated with lubricant. Protect the bearing in greaseproof paper until ready for fitting.
- New bearings should be kept wrapped until ready for fitting. Check that the bearing is the correct type and size according to the manufacturer's recommendations.
- Check that the shaft and bearing housing are clean and free from burrs and other surface damage.
- Check the shaft and housing dimensions to ensure that they are correct according to the manufacturer's drawings.
- Check shoulders and abuttments for run-out, especially in the case of thrust bearings. Run-out in the thrust face mounting will cause rapid wear and should be checked with a dial gauge.

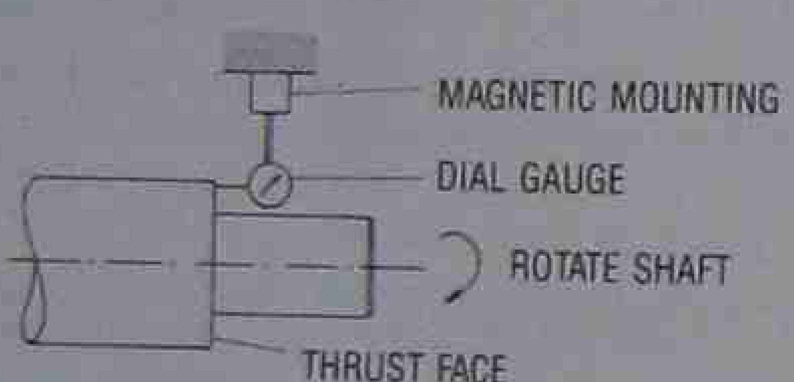


Fig. 6-53 Checking run-out in the thrust face mounting.

Pre-lubrication

Normally, rolling element bearings are not lubricated until after they are mounted, although there may be exceptions, especially when the bearings are inaccessible. For cold mounted bearings, however, a light lubrication of the bearing seat and the shaft journal and housing will assist in the assembly procedure. It is also good practice where shaft nuts and adapter sleeve nuts are used for drive-up, that the threads be lightly lubricated so that they create minimum resistance.

Selection of mounting method

The method to be used for mounting a bearing will depend on the type and size of bearing and on the mounting arrangement. The manufacturer's instructions or service manual should be followed where available.

Bearings with a bore of 100mm (4") or less can usually be cold mounted, whereas larger bearings need to be heated. For bearings with tapered bores, only very large bearings need to be heated.

Cylindrical bore bearings need to be mounted mechanically whereas tapered bore bearings can be mounted using hydraulic mounting tools.

The following methods can be considered standard for mounting rolling element bearings. If doubt exists as to which method should be used the equipment and bearing manufacturers should be consulted.

Arbor press

An arbor press can be used for small bearings with cylindrical bore. A sleeve should be used between the bearing and the press which has flat, parallel and burr-free end faces. The sleeve should bear on the bearing ring with the interference fit.

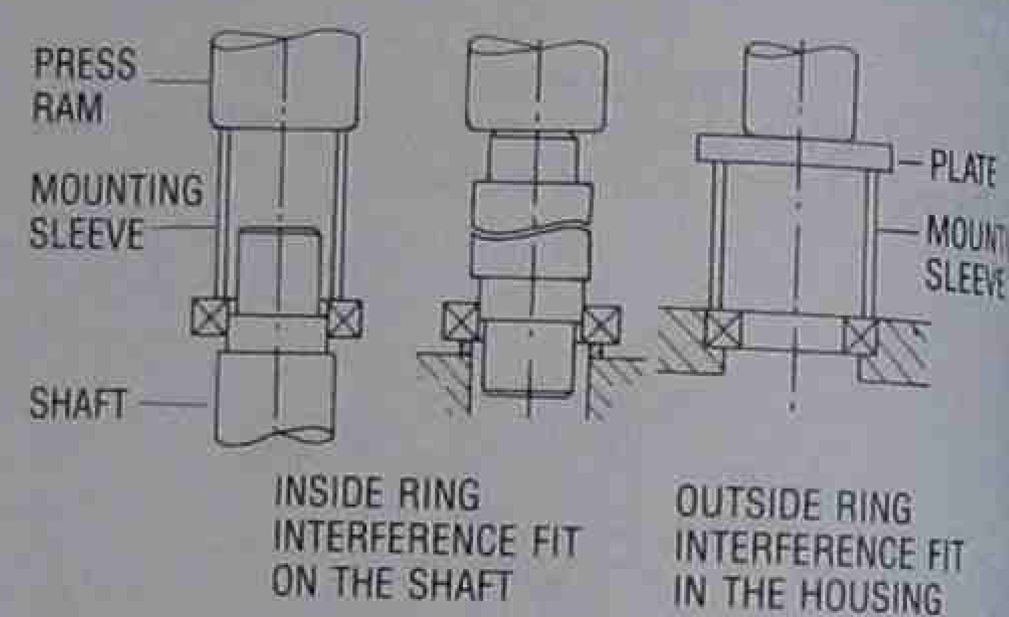


Fig. 6-54 Typical arrangements using an arbor press.

Hammer and dolly

For cold mounting of all cylindrical bore bearings a suitable dolly or sleeve can be used to drive the bearing on to its mounting. The sleeve must bear on the bearing face with the interference fit and care should be taken to ensure that the bearing is driven on smoothly and does not cock over.

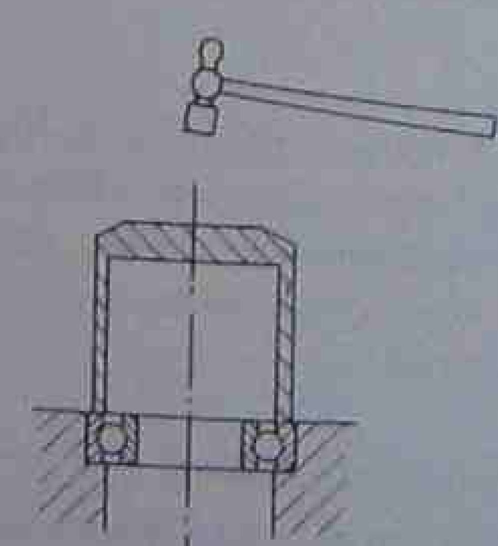


Fig. 6-55 Typical example of the use of a hammer and dolly.

Hook and impact spanners

For small and medium sized bearings (i.e. up to 200 mm (8") bore) with tapered bore, either a hook or impact spanner can be used with a drive nut to drive the bearing on to the tapered shaft or on to an adapter sleeve as shown in Fig. 6-56.

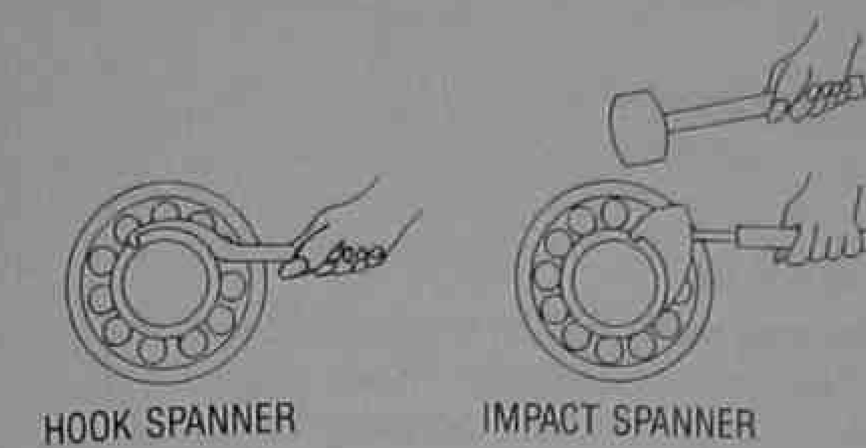


Fig. 6-56 Using hook and impact spanners.

When a drive nut is used in this manner the face of the nut facing the bearing should be coated with a dry lubricant such as molybdenum disulphide and the surface of the shaft or sleeve coated with a light oil.

Hydraulic nut

A hydraulic nut can be used for bearings with tapered bores as shown in Fig. 6-57.

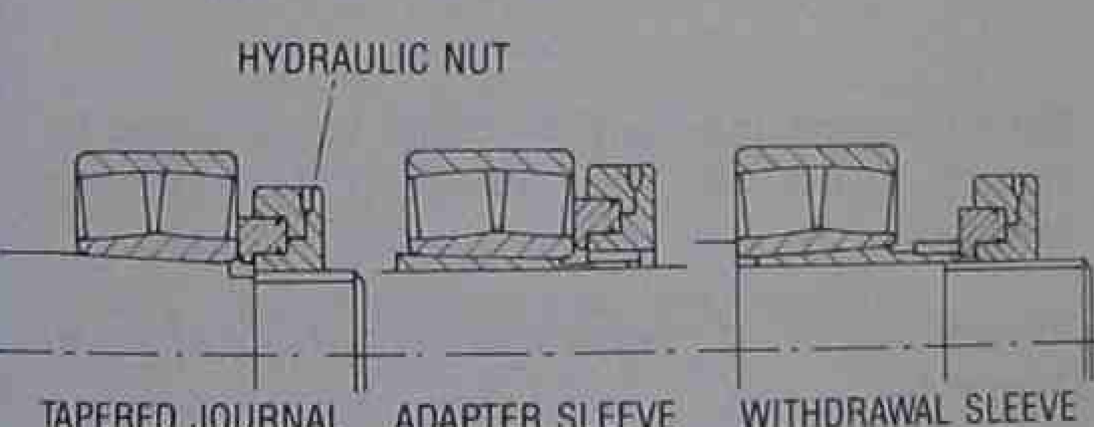


Fig. 6-57 Using a hydraulic nut.

When oil is pumped into the nut the annular piston forces the bearing on to the tapered seat until it reaches a shoulder or the required amount of axial drive-up (p.46).

The hydraulic nut should be used in conjunction with a suitable oil pump and oil of viscosity recommended by the manufacturer.

Oil injection

Oil can be injected between the bearing inner face and the shaft for a tapered bore bearing to expand the bearing and reduce the friction and make it easier to drive the bearing on to the seat as shown in Fig. 6-51.

In order to be able to use this method the shaft must have been specially machined with an oilway and oil grooves to distribute the oil between the bearing and the shaft.

The viscosity of the oil and the pressure developed by the pump should be determined with the supplier of the equipment.

Sometimes withdrawal sleeves are also specially machined with oil ducts and can be used with oil injection as shown in Fig. 6-58.

Fig. 6-58 Withdrawal sleeve machined with oil ducts.

Oil injection plus hydraulic nut

For large bearings on tapered bore shafts or with suitably machined withdrawal sleeves, oil injection can be used in conjunction with a hydraulic nut.

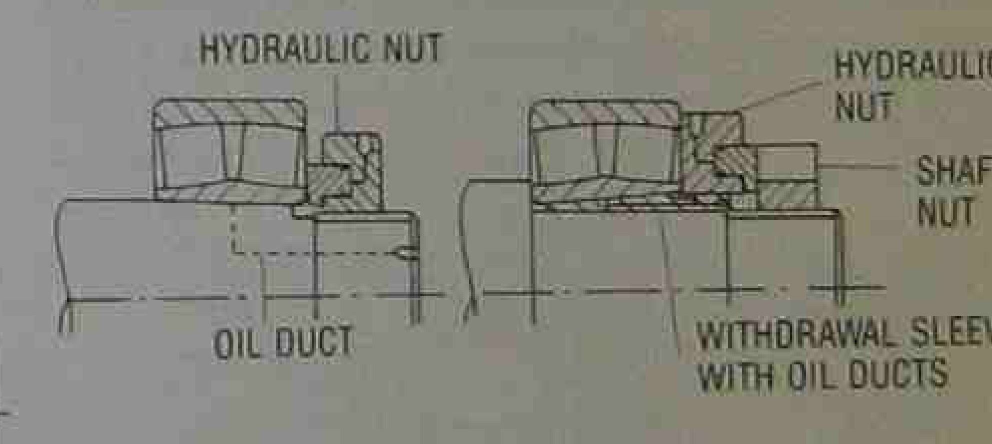


Fig. 6-59 Using oil injection in conjunction with a hydraulic nut.

Hot mounting

For bearings with cylindrical bores and for large bearings with tapered bores it may be necessary to heat the bearing. The bearing should be heated to around 80°C to 90°C above the shaft temperature but never to more than 120°C (250°F). The heating apparatus should be close to the equipment and the hot bearing should be pushed home quickly and smoothly before it cools down or jams. An induction heater of the type shown in Fig. 6-49 should be used where possible.

If an oil bath is used, the bearing should be heated with the oil and not dropped into hot oil and should be kept off the bottom of the bath by a grate to prevent distortion as shown in Fig. 6-50.

If these two methods are not available then a hot plate or an oven may be used.

Whatever method is used a careful check must be kept on the temperature of the bearing by using a surface thermometer.

When a bearing is an interference fit in the housing and it is impossible to heat up the housing it may be necessary to cool the bearing by submerging it in a bath of alcohol cooled by dry ice or a cryogenic liquid.

Special considerations

Adjustment of tapered roller bearings

Tapered roller bearings must be set up either with clearance or with a certain amount of preload, depending on the manufacturer's instructions. The simplest way to do this is to draw up the shaft nut or end plate until there is no play in the bearings and drag becomes noticeable. If end play is required then the shaft nut can be backed off and if preload is required then it can be pulled up harder. The adjustment can be measured by mounting a dial gauge against a shaft shoulder or gear face.

In an arrangement with shims, such as the one shown in Fig. 6-60, once the play has been taken up the gap between the end plate and shaft end can be measured with a feeler gauge and either the end play added or the preload subtracted in order to establish the correct thickness of shim pack.

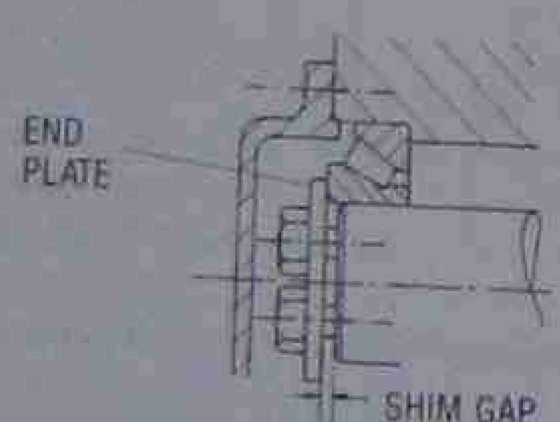


Fig. 6-60 Adjusting the shim pack for tapered roller bearings

Where clearance is required it is always wise to take a final check of end play with a dial gauge against a shaft shoulder as shown in Fig. 6-61.

If a shaft locking nut arrangement is involved then a final check of clearance will be needed after the locking nut has been tightened because this will force the shaft nut towards the bearing by an amount equal to the clearance in the threads.

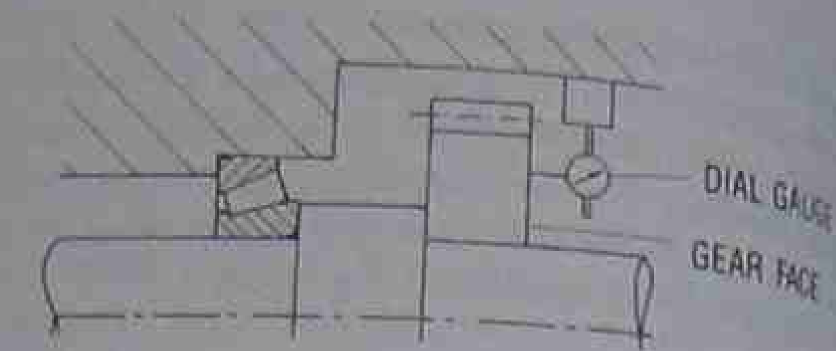


Fig. 6-61 Checking end play.

Axial drive-up

For a bearing mounted on a tapered bore, the deg... of interference between the inner ring and the sh... depends on how far the bearing is driven up t... tapered shaft. This dimension is known as ax... drive-up. In small spherical roller bearings, whe... clearances cannot be measured, the axial drive-u... used as a measure of the interference fit.

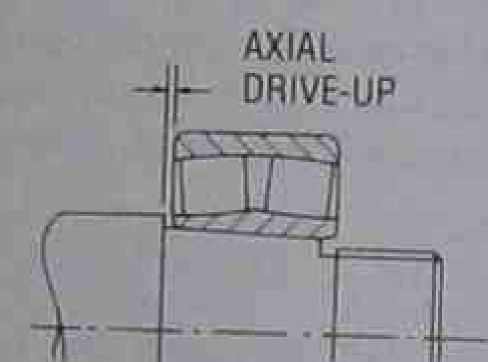


Fig. 6-62 Axial drive-up used as a measure of interference fit.

Manufacturers' information should be consulted f... correct values of axial drive-up for differe... bearings.

Measurement of clearances in spherical rolle... bearings

The internal clearance in a spherical roller bearing ... greater than in a ball bearing and can be measure... with feeler gauges. Because spherical roller bearing... are mounted on tapered seats, as the bearing ... pushed on to the taper the inner ring expands an... reduces the internal clearance in the bearing. Hen... the final clearance is a direct function of th... interference fit on the shaft. Measurement of th... clearance is useful in giving an indication of the sha... fit. This can be accomplished as follows:

- Measure the unmounted radial clearance b... standing the bearing on a clean surface an... rotating the inner ring backwards and forwards t... seat the rollers properly in the outer ring. The... use a feeler gauge to establish the clearanc... between the uppermost roller and the outer ring. Do this by inserting the gauge between two top rollers and then rolling the roller under the blade. Increase the feeler gauge thickness until the roller traps the blade and cannot be withdrawn.

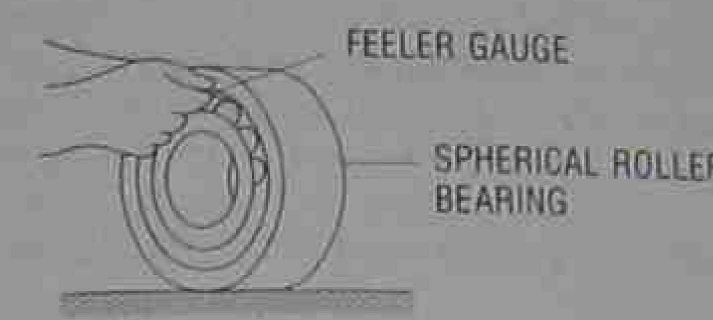


Fig. 6-63 Using a feeler gauge to establish clearance.

- When the bearing is mounted on the tapered shaft or sleeve, the clearance must be measured at the bottom instead of the top. As before, the outer ring should be rotated a few times to ensure that the rollers are properly seated and a feeler gauge inserted between the rollers and the bottom of the outer ring. See Fig. 6-64.

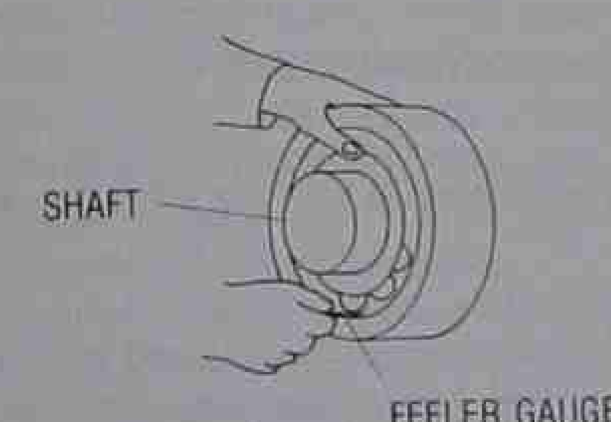


Fig. 6-64 Using a feeler gauge to establish clearance.

- With self-aligning ball bearings the clearances are too small to measure with a feeler gauge. Normal practice is to tighten up the shaft nut and to check the clearance by swivelling and rotating the outer ring. When the clearances are correct the ring should rotate freely but there should be some resistance to swivelling.

Dismounting procedures

The procedure used for dismounting a bearing will depend on the way in which the bearing is mounted and whether the interference fit is on the shaft or in the housing. If the bearing is to be re-used then its relative position should be marked before it is dismounted i.e. which side is 'up' and which way does it face. Whether the bearing is to be re-used or not, care should be taken not to damage the bearing during the dismounting process so that the evidence of failure is not disturbed and can be used to establish cause of failure. The dismounting force, like the mounting force, should always be applied to the ring with the interference fit. Care should also be taken not to damage the surface of the shaft or housing.

The following methods are commonly used.

Interference fit on the shaft

The most common method for dismounting bearings with either a cylindrical bore or a tapered bore is by the use of a puller such as the one shown in Fig. 6-47a.

Bearings mounted on shafts that have been machined with oil ducts can be dismounted using the oil injection method.

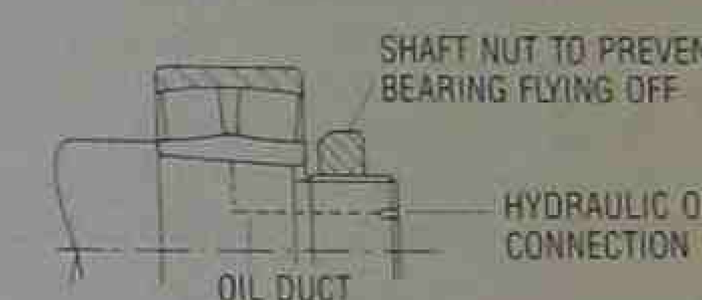


Fig. 6-65 Using the oil injection method.

Interference fit in the housing

When the bearing is an interference fit in the housing then the bearing may have to be hammered off using a dolly or a soft metal drift.

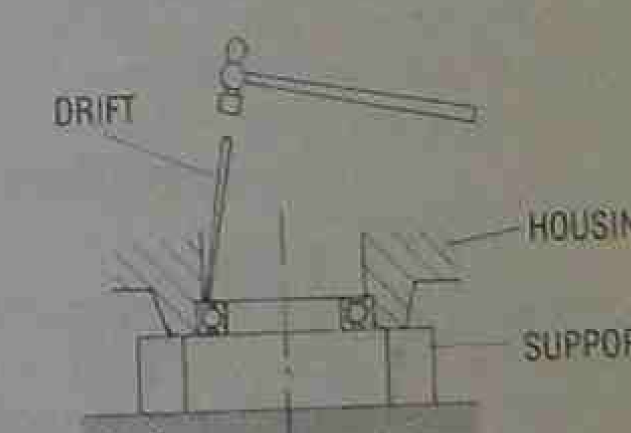


Fig. 6-66 Hammering off the bearing.

Because of the tendency of soft metal drifts to chip they should never be used for mounting. When dismounted in this way bearings should be washed carefully before being re-used.

For bearings where the inner ring can be swivelled an inside puller can be used, such as the one shown in Fig 6-47b.

Bearings on adapter sleeves

A hammer and dolly can be used either to drive the bearing off the sleeve or to drive the sleeve from under the bearing, depending on which way round the sleeve is mounted.

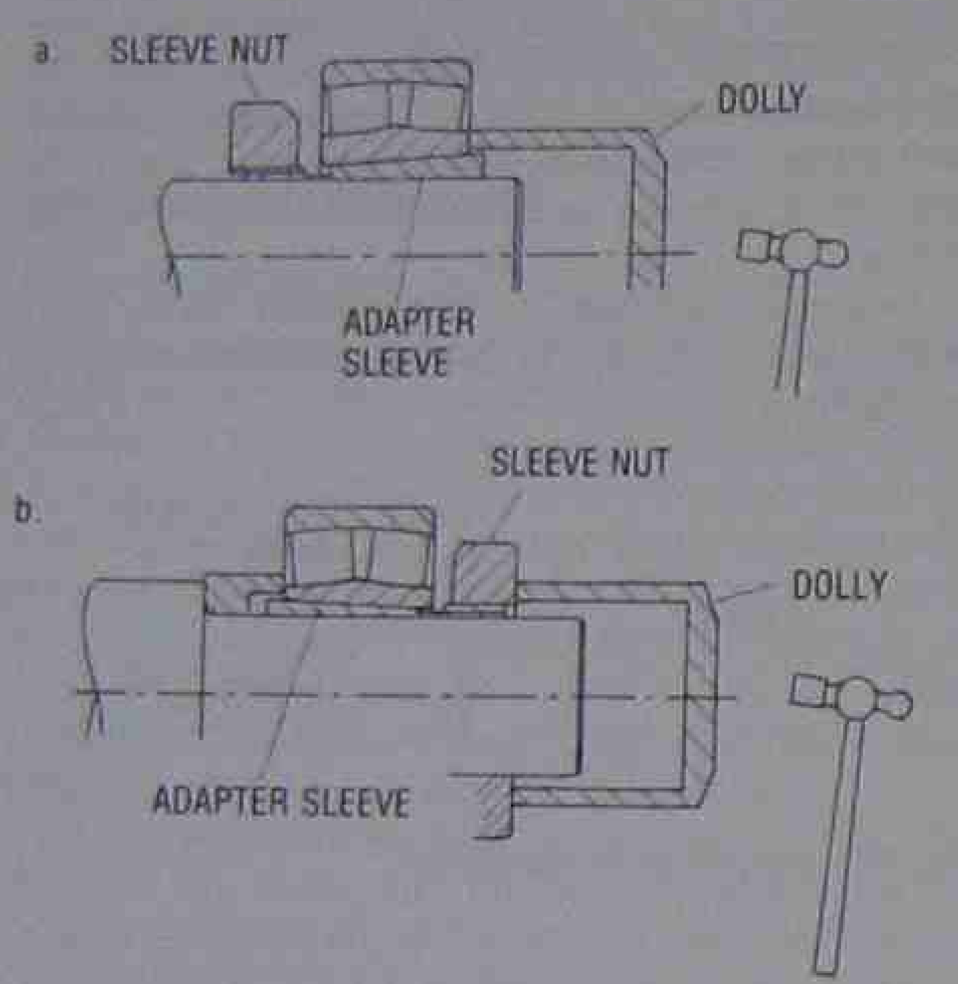


Fig. 6-67 Using a hammer and dolly to dismount a bearing from an adapter sleeve.

The alternative is to use a hydraulic nut as shown in Fig. 6-68.

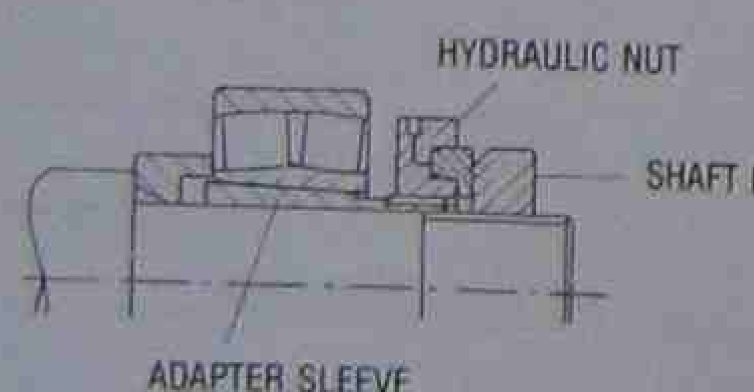


Fig. 6-68 Using a hydraulic nut for dismounting.

Bearings on withdrawal sleeves

For small and medium sized bearings a hook or impact spanner can be used to drive up a withdrawal nut to force out the sleeve as shown in Fig. 6-69.

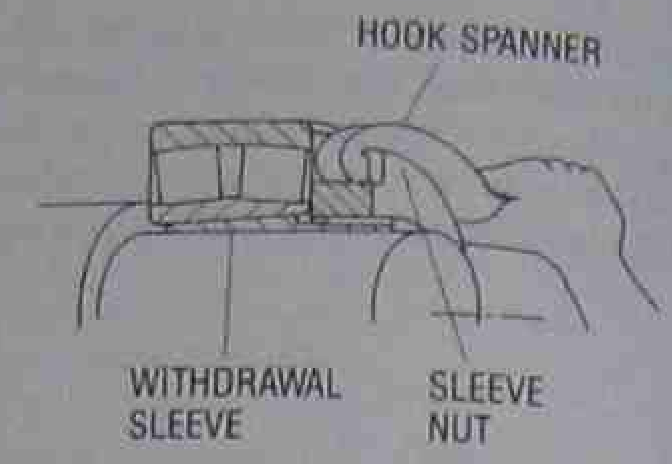


Fig. 6-69 Forcing out the sleeve with a hook spanner.

To make the process easier, the threads and fa[ces] of the withdrawal nut should be lubricated w[ith] molybdenum disulphide.

For large bearings the use of a hydraulic nu[t is] recommended as shown below in Fig. 6-70.

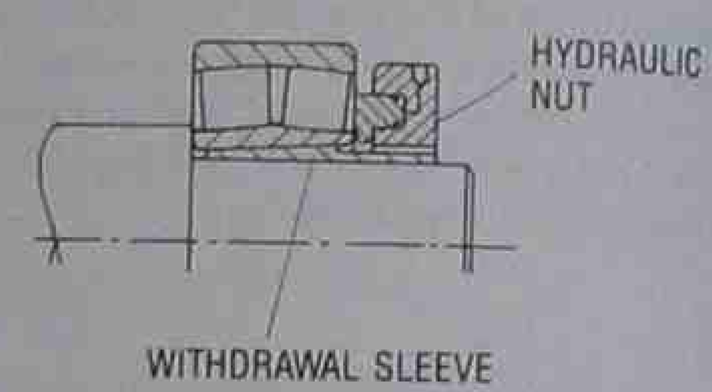


Fig. 6-70 Using a hydraulic nut to remove a withdrawal sleeve.

If the withdrawal sleeve is machined with oil duc[ts] then oil injection can be used in conjunction with th[e] withdrawal nut and an impact spanner, as shown [in] Fig. 6-71.

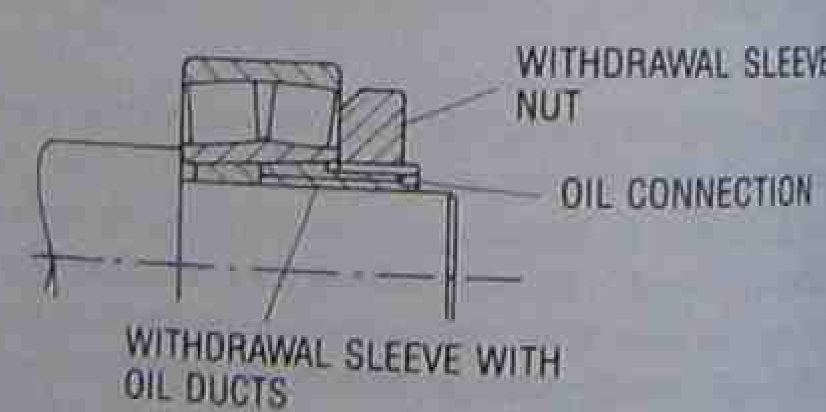


Fig. 6-71 Oil injection used in conjunction with a withdrawal nut.

FAILURE PATTERNS

As in the case of plain bearings, it is important t[o] accurately determine the cause of failure before a

rolling element bearing is replaced or repaired. This requires careful analysis of the symptoms of malfunction and careful inspection of the failed or damaged components. Generally speaking, malfunction and failure of rolling element bearings are caused by much the same problems as affect plain bearings. The symptoms of failure can be classified into those noticed during operation and those found on inspection. The evidence of failure differs to some extent from that shown by plain bearings because of the physical difference in construction.

Operating symptoms

The external symptoms of failure are the same for rolling element bearings as for plain bearings.

Overheating

As the condition of a rolling element bearing deteriorates, the frictional resistance to motion increases and the operating temperature of the bearing begins to rise. Unlike plain bearings, however, where almost any change in condition tends to cause an increase in operating temperatures, only certain causes lead to overheating in rolling element bearings. The two main causes are inadequate lubrication and loss of internal clearance. Because of the sliding contact between the surfaces of a plain bearing almost any change in surface condition will increase the frictional resistance to motion. The rolling contact in a rolling element bearing reacts rather differently.

The temperature of a rolling element bearing can be measured in the same way as a plain bearing and once again it is deviation from the normal running condition that provides a warning of malfunction. A rise of 10°C above normal running temperature is generally considered as cause for alarm with a maximum allowable temperature for a standard bearing being around 125°C (260°F).

Vibration

Unlike plain bearings, some vibration of rolling element bearings is normal due to the action of the rolling elements themselves. With rolling element bearings it is particularly important that a **signature pattern** (see Chapter 10) is established either by vibration analysis or by direct observation. Deviation from that pattern can then be monitored to detect change in condition of the bearing. The acceptable limits for measured vibration vary with the size and type of machine and recommended values are given in Chapter 10. If vibration levels are assessed by direct observation then experience is required to determine the severity of the problem. An understanding of the principles of operation of the bearing and the material characteristics should make it obvious, even to the inexperienced, when vibration levels are becoming unacceptable.

Vibration may develop in a bearing because of defects in the bearing itself, or it may be transmitted to the bearing from other parts of the machine or other adjacent equipment. Vibration produces rapid and repeated blows on bearing surfaces and contributes, sooner or later, to fatigue failure of the bearing material. It can also affect the properties of grease and damage bearing seals and hence lead to loss of lubricant.

When vibration is transmitted to a stationary bearing it can cause fretting corrosion, described below.

Noise

As is the case with vibration, rolling element bearings generate more noise under normal running conditions than do plain bearings. The normal noise pattern may vary due to changes in operating speeds and loads and, as with vibration patterns, a signature should be established from which deviations can be observed.

Although noise can be measured with an acoustic monitoring device (see Chapter 10) it is more usual for direct observations to be made by ear. Hence it is important for maintenance personnel to become familiar with the characteristic noises produced by different types of bearings and the changes produced by different types of defects. For example, dirt tends to produce a crackling sound, damage to the raceways causes a high-pitched whine, and loss of lubricant causes a squeaking sound. The most common aids used in the detection of bearing noise remain the screwdriver and the stethoscope and valuable information about bearing condition can be determined this way.

Seizure

If noise, vibration and overheating are carefully monitored then complete seizure should rarely occur. When it does, it may indicate that overheating has reached the point where the bearing metals have fused and the usual cause of this is lack of lubrication. It is also possible for a very badly worn bearing to seize due to displacement of the rolling elements. The failure of the bearing cage or retainer can also cause the rolling elements to jam.

Symptoms found on inspection

Rolling element bearings also develop symptoms of malfunction that can only be observed when the bearing is dismantled. Careful observation of the bearing surfaces and other evidence is essential if the precise cause of failure is to be established. The only exception here is sealed bearings which are usually discarded and replaced without investigation.

The following symptoms are commonly found as evidence of rolling element bearing malfunction or failure.

Wear

Rolling element bearings are not as susceptible to general wear from adhesion and abrasion as plain bearings, because of the rolling rather than sliding motion of the surfaces, but the presence of foreign material will cause surface damage in the form of scratching and scoring as shown in Fig. 6-72.

SKF

Fig. 6-72 Surface damage caused by the presence of foreign material.

Fatigue

The rolling action of balls and rollers sets up an intermittent transfer of load from one raceway to another and generates cyclical stresses in the loaded area of the raceways. The continuous deflection of the metal causes cracks to develop which lead to flaking of the surfaces.

SKF

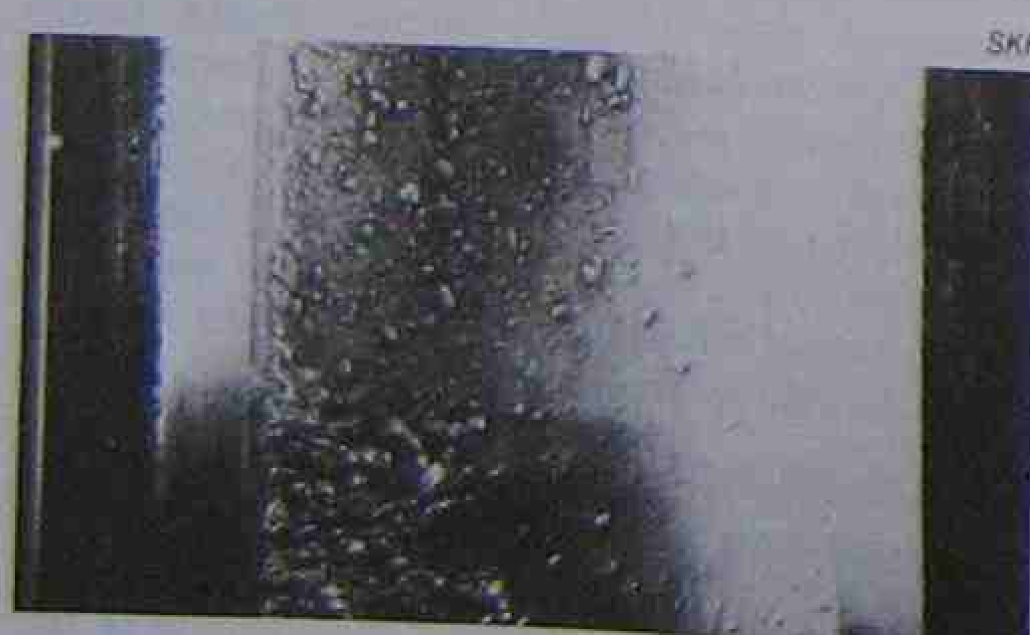


Fig. 6-73 Surface flaking.

As this condition worsens, the metal breaks do further and grain displacement takes place. T condition is known as **spalling**.

Fig. 6-74 Grain displacement due to spalling.

Fretting

Fretting corrosion or false brinelling is most likely occur when a rolling element bearing is dry a stationary. It is due to slight, almost imperceptib motion between the contacting surfaces of the rolli elements and raceways. Such motion may b transmitted to the bearing from an external source may come from the machine itself. Fretting produc a characteristic red-brown dust at the interface.

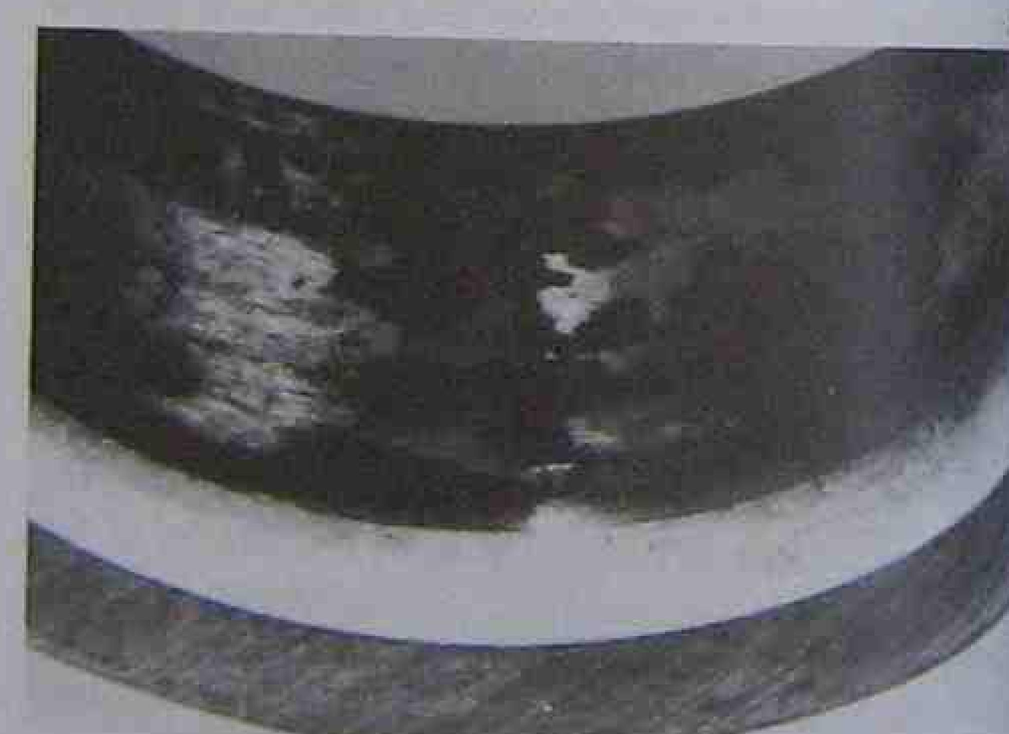


Fig. 6-75 A typical example of fretting.

Fretting corrosion is most likely to occur when machine is in transit or on stand-by.

Brinelling

True brinelling, which false brinelling or fretting said to resemble, consists of indentations in the raceways made by the rolling elements when the bearing is subjected to shock loading.

Mobil

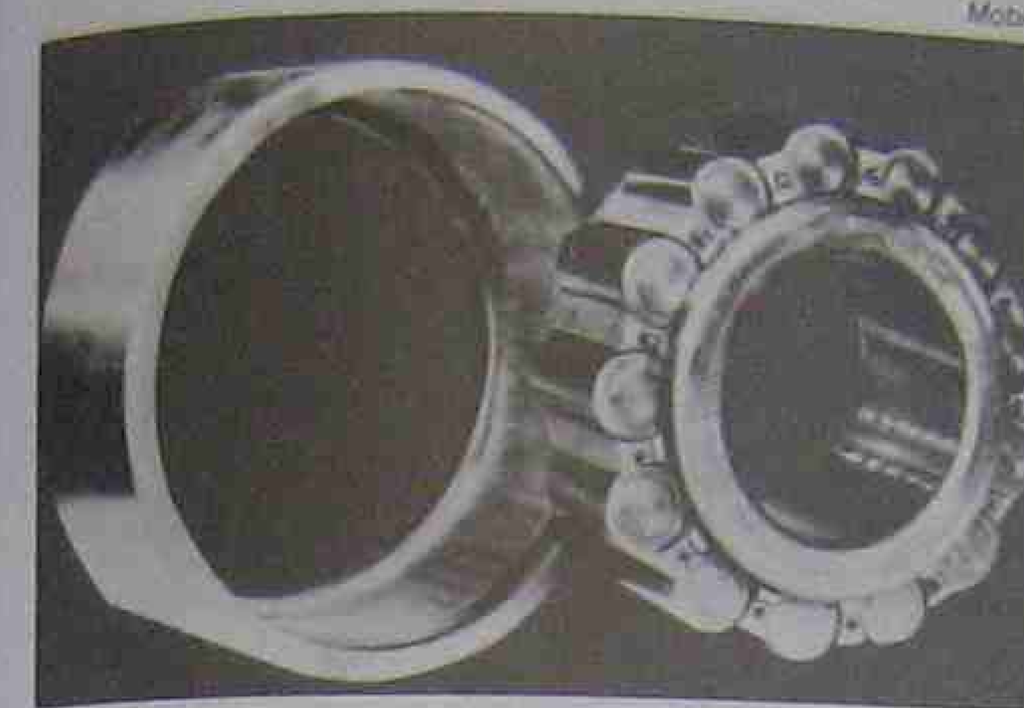
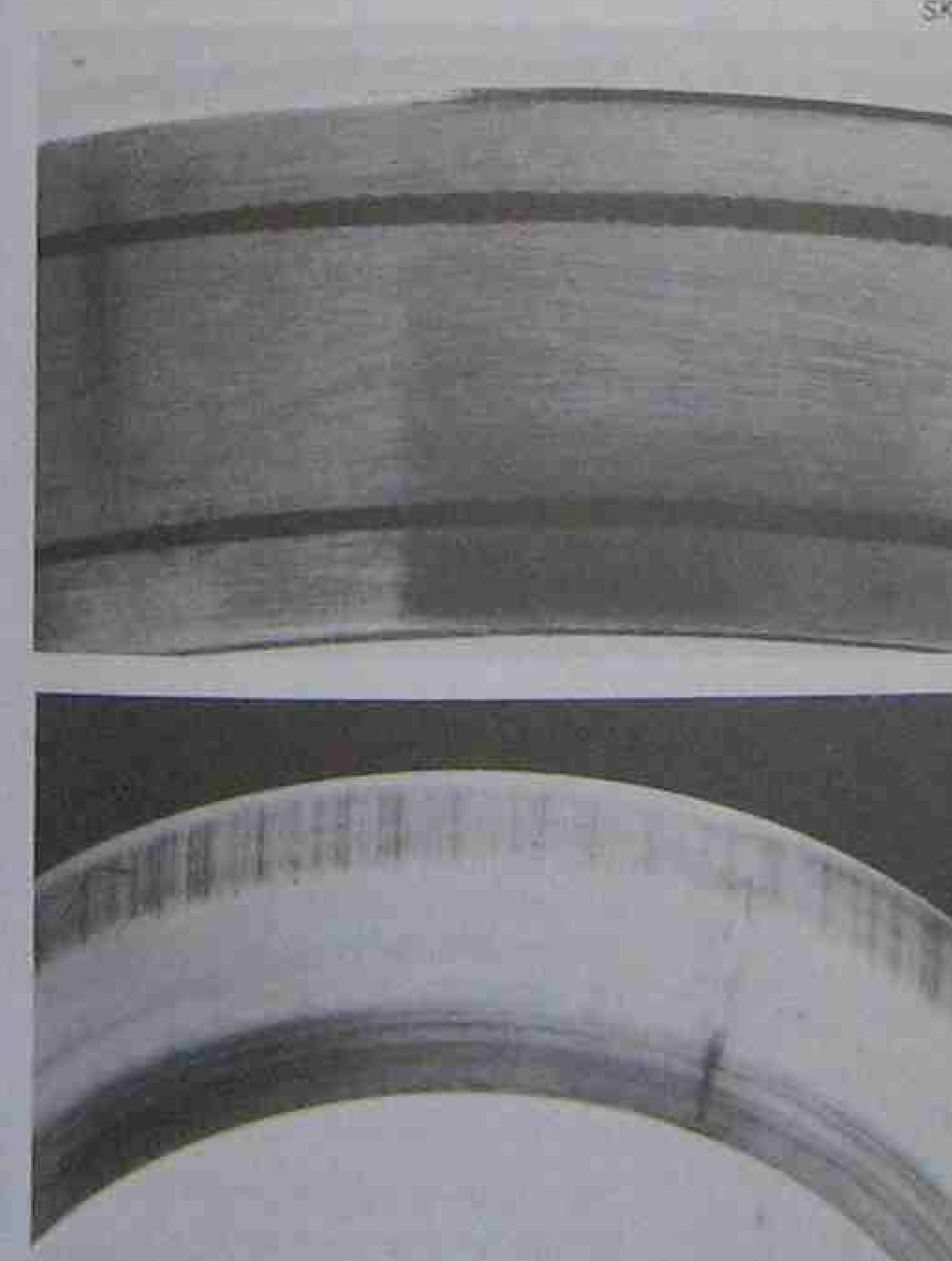


Fig. 6-76 True brinelling.

Pitting and fluting

A pitted or a fluted effect as shown in Fig. 6-77 may appear as a result of stray electric currents arcing across the contact surfaces.

SKF



SKF

Fig. 6-77 Pitting and fluting.

It is usual for the pitting to develop first and if no action is taken to prevent further damage this deteriorates further into a fluted or washboard effect.

Smearing and galling

If the rolling elements fail to rotate properly and rub or slide over the raceways then smearing may occur due to the surface of the raceway 'picking up' on the rolling element as shown in Fig. 6-78.

SKF

Fig. 6-78 Smearing of bearing surface.

In extreme cases this may cause grain displacement and is then referred to as galling which can be identified by the craters produced as shown in Fig. 6-79.

Mobil

Fig. 6-79 Evidence of galling.

Corrosion

Corrosion of bearing surfaces may be evident as pitting or spalling of bearing raceways. Rust and other corrision products may be present.

Mobil

Fig. 6-80 A typical example of corrosion.

Wear patterns

The pattern or load zone produced on the internal surfaces of a ball bearing can be an important clue to the cause of failure. Although other types of bearings also generate specific loading patterns they are much more difficult to analyse than ball bearings.

In order to benefit from the study of load zones it is necessary to differentiate between normal and abnormal patterns. The normal pattern to be expected from a bearing will depend on the wa load is carried and whether it is the inner or the o ring that rotates. Fig. 6-81 shows examples of normal and abnormal load patterns that ca expected to develop under the loading cond stated.

Causes of failure

The same problems that cause malfunction a failure of plain bearings also cause failure of ro element bearings. Rolling element bearings can a be affected by factors that do not usually affect pl bearings due to their different configuration. T the symptoms described above can result from o more of the following:

Dirt

As with plain bearings, the presence of dirt or oth foreign material in the bearing will cause excess wear and premature failure of a rolling eleme bearing. Cleanliness during assembly, consta

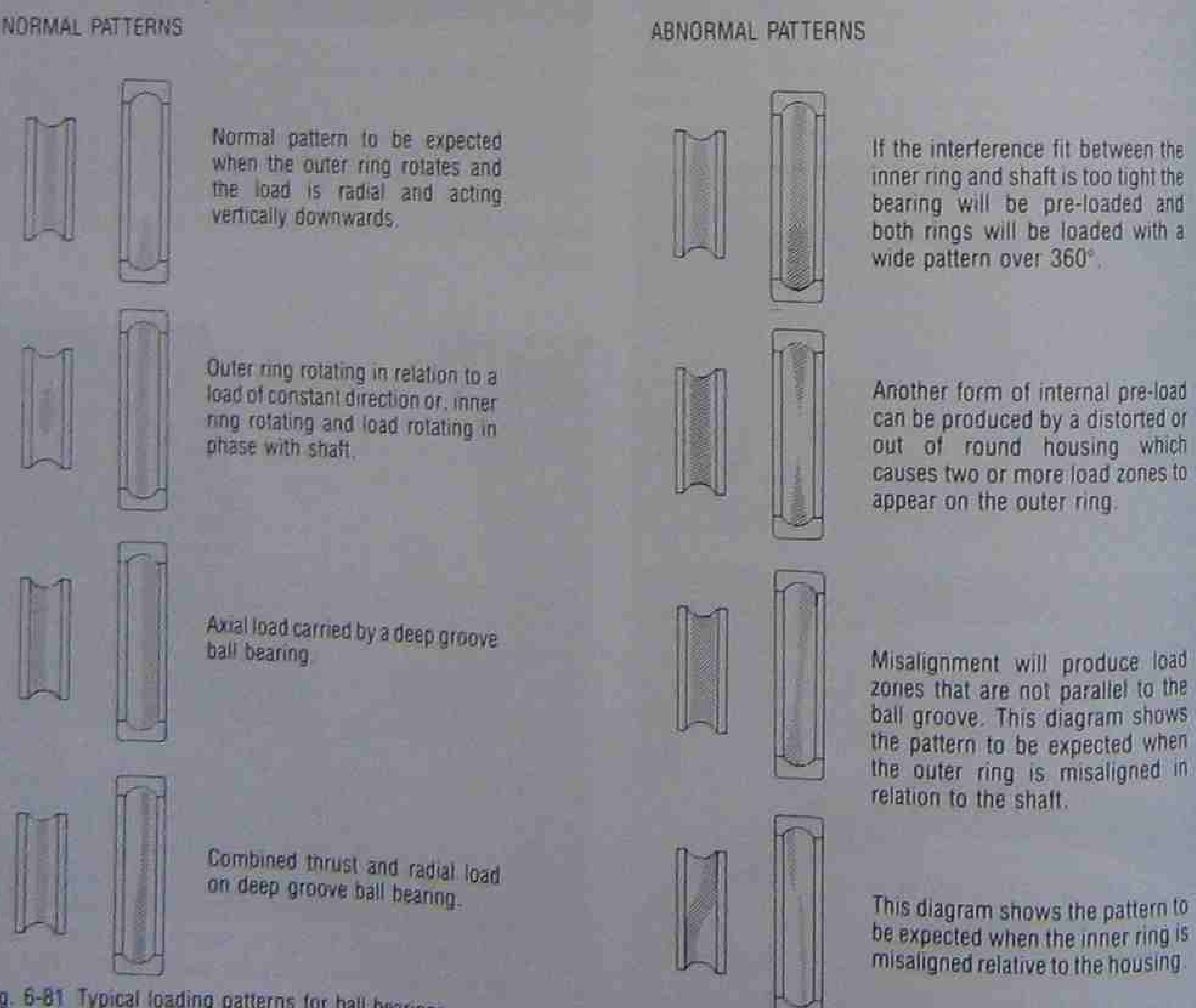


Fig. 6-81 Typical loading patterns for ball bearings.

filtering of lubricants and well maintained seals will help to avoid this problem.

Inadequate lubrication

The oil supply system is as critical for rolling element bearings as for plain bearings and all the recommendations for maintaining an adequate supply of the correct lubricant apply. Failure of the system will lead to rapid deterioration in bearing condition due to smearing and galling, and the likely seizure of the bearing. It is generally recognised that inadequate lubrication is the single most common cause of rolling element bearing failure.

Improper assembly

If a rolling element bearing is not correctly assembled then rapid wear will result from excessive or insufficient pre-loading, misalignment or failure to secure the inner and outer rings correctly. Abnormal wear patterns such as those described in the previous section will result and also the likelihood of fatigue failure is increased.

Misalignment

For rolling element bearings that do not have inherent self-aligning features the alignment of shaft and housing can be critical to the operation of the bearing. Cylindrical roller bearings are the most susceptible to run-out and there are limits to the tolerable run-out for all other types except self-aligning ball bearings and spherical roller bearings. Wear patterns should be studied carefully in order to establish evidence of misalignment and squareness of the housing and shaft alignment should be closely checked if doubt exists.

Overload

If a rolling element bearing is subjected to loads in excess of those for which it was designed, then its life will be shortened and early fatigue failure will occur. The reduction in life span can generally be expected to be proportional to the cube of the load increase. This means if the load were doubled then bearing life would be reduced to approximately one-eighth of normal. Overloading can also occur due to excessive speed which leads to overheating of the bearing. High temperatures may affect the properties of the lubricant and cause centrifugal throw-out of oil and grease. Excessive wear will occur as the effectiveness of the lubricant deteriorates, and at very high speeds fatigue life may be reduced due to the increased loading which results from high centrifugal forces acting on the rolling elements.

Moisture

The presence of water in a bearing, due to leakage into the housing or condensation of moisture due to temperature changes, can cause considerable damage from corrosion. Water may also find its way into the bearing as a result of improper washing and drying techniques used during inspection.

In an oil lubricated bearing, the presence of significant quantities of water will cause emulsification of the lubricant.

Lubricant breakdown

Oxidation of the lubricant may lead to the formation of acidic compounds that may cause corrosion of bearing surfaces. Sometimes lubricant additives may be incompatible with bearing materials and also lead to corrosive attack.

Shock loading

Rolling element bearings are more susceptible to shock loading than are plain bearings because of the point or line contact of the rolling elements. Shock loads cause indentation of the raceways and result in the brinelling effect discussed in the previous section.

Electric currents

Stray electric currents, even at very low voltages, can arc across the lubricant film and lead to pitting and fluting. This is more likely to occur with electrical machinery but can also occur in other machines due to currents that build up statically and short circuit to ground through the bearing. This type of problem may also occur due to the incorrect positioning of earthing leads when welding is being carried out on a machine.

External vibrations

Vibrations transmitted from other parts of a machine, or from another machine altogether, can cause damage to a rolling element bearing whether it is running or stationary. If a bearing is in operation and is subjected to vibrations from an external source then its fatigue life is likely to be reduced depending on the severity of the vibrations. External vibrations transmitted to a stationary bearing may produce fretting corrosion.

Summary of the symptoms and causes of rolling element bearing failure

Symptoms		Causes
Operating	*Inspection*	
Overheating	Wear	Dirt
Vibration	Fatigue (spalling)	Inadequate lubrication
Noise	Fretting	Improper assembly
Seizure	Brinelling	Misalignment
	Pitting and fluting	Overload
	Smearing and galling	Moisture
	Corrosion	Lubricant breakdown
	Abnormal wear zones	Shock loading
		Electric currents
		External vibration

C H A P T E R

POWER TRANSMISSION

As was discussed in Chapter 2, most rotating machines operate in sets which include a driver and driven machine at least and often include a transmission machine such as a gear box as well.

There are various ways in which the output sha[ft of] one machine can be linked to the input sha[ft of] another so that transmission of power can take pla[ce].

7-1 V-belt drives

Although various other types of belt drives are also used, V-belts are the most common. However, many of the maintenance considerations that apply to V-belts are also relevant to other types of belt drive such as flat belts and timing belts.

PRINCIPLES OF OPERATION

V-belts are normally used to transfer power between two shafts whose axes are parallel and some distance apart.

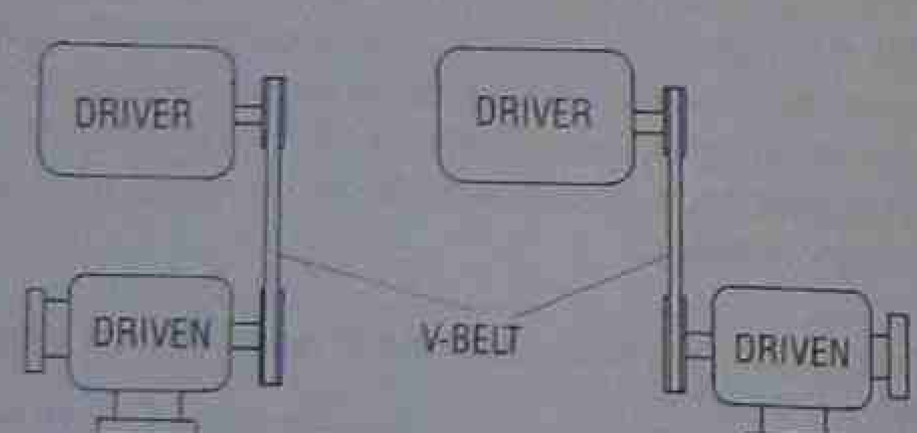


Fig. 7-1 Typical V-belt arrangements.

The belt is mounted on pulleys that are attached to the driving and driven shafts and the drive relies on friction between the belt and the pulleys for its operation. The belt sits in the groove of the pulley and makes contact with the sides of the groove as shown in Fig. 7-2.

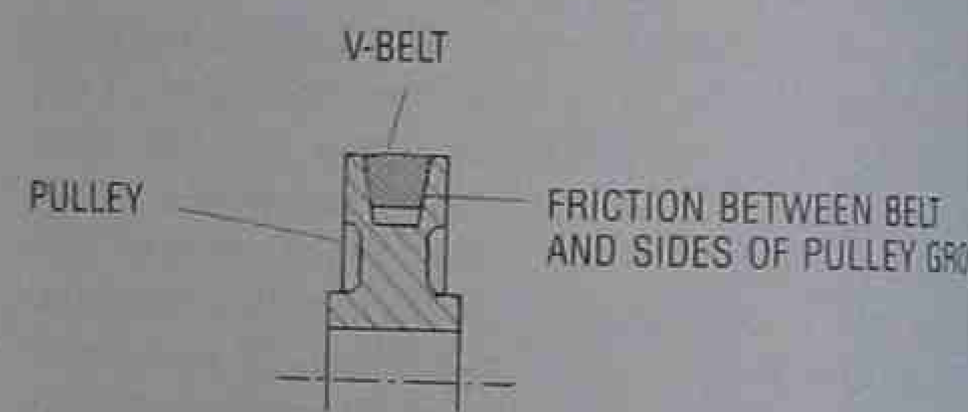


Fig. 7-2 V-belt in pulley.

In order to be able to transmit power, the belt m[ust] be under tension so that it is forced down into t[he] groove. The belt is squeezed and friction devel[ops] between the sides of the belt and the sides of t[he] groove. The depth of the groove is always grea[ter] than the thickness of the belt, however, and the belt should never bottom in the groove. The operation of the belt and its ability to transmit power depend on the size of the friction force and the arc of contact of the belt. The greater the arc of contact the more power the belt can transmit. (Fig. 7-3)

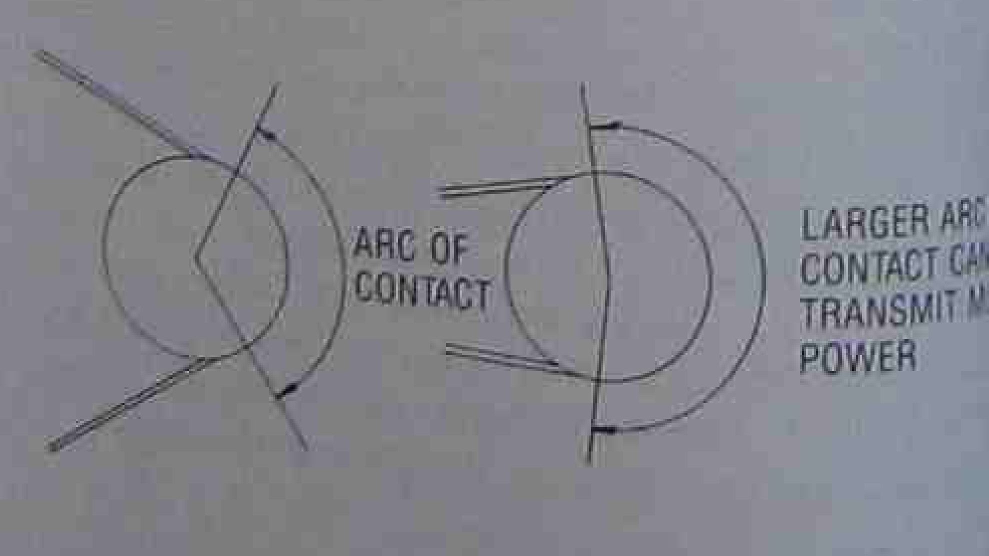


Fig. 7-3 Relationship between power and arc of contact.

As well as performing its primary function of transmitting power, a V-belt can be used to change the speed of the driver output and hence the torque transmitted to the driven unit. There are three basic alternatives as shown in Fig. 7-4.

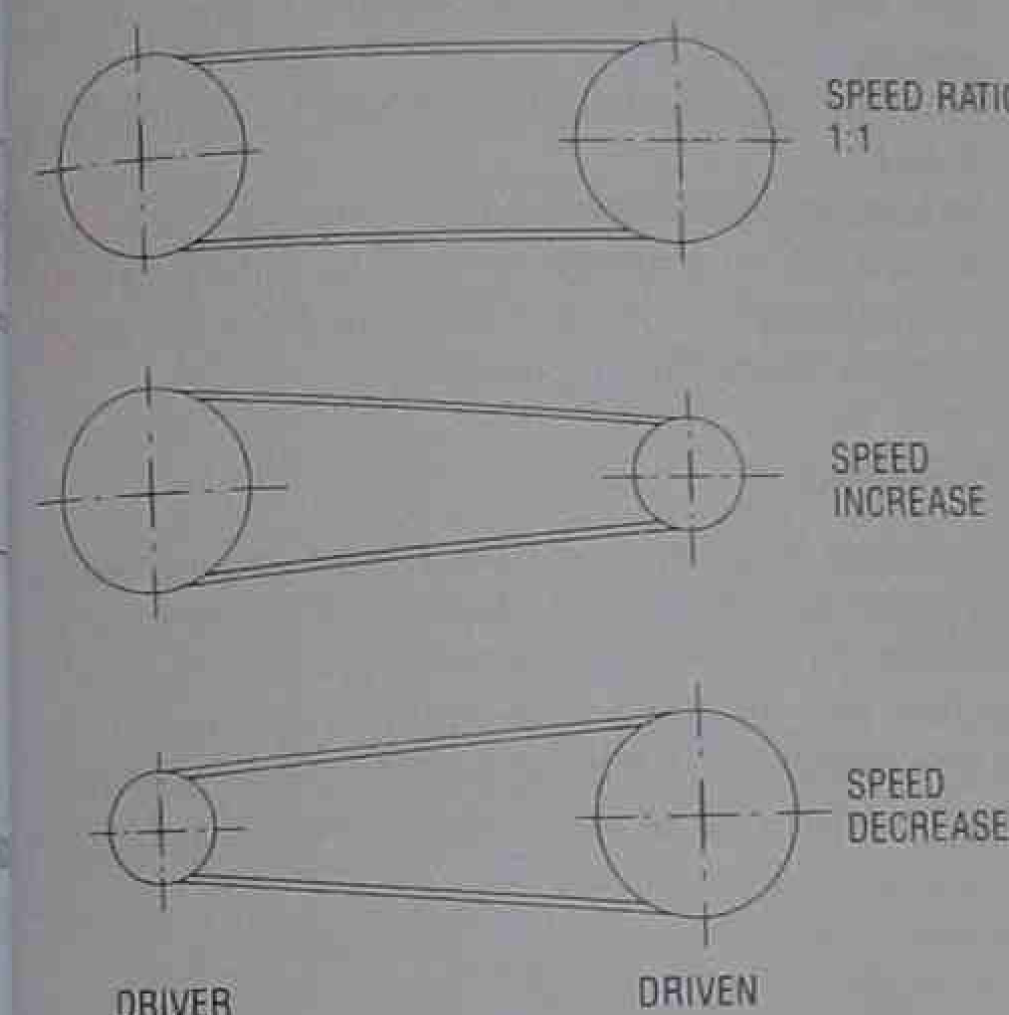


Fig. 7-4 Alternative arrangements for V-belt drives.

The speed ratio between the two pulleys of a belt drive can be calculated from a simple formula.

$$\text{driven speed (RPM)} = \frac{\text{driver pulley diameter (mm)}}{\text{driven pulley diameter (mm)}} \times \text{driver speed (RPM)}$$

It is generally accepted that V-belt drives are limited to belt speeds between 300 and 3000 metres per minute (1000–10 000 feet per minute). If required to operate at higher speeds then dynamic balancing of the pulleys becomes increasingly important.

TYPES AND ARRANGEMENTS

Single V-belt

The most common type is the single belt arrangement whose operation has been described above. In addition to being used to transmit power between parallel shafts, the single belt can also be used for quarter turn drives and angle drives as shown in Fig. 7-5.

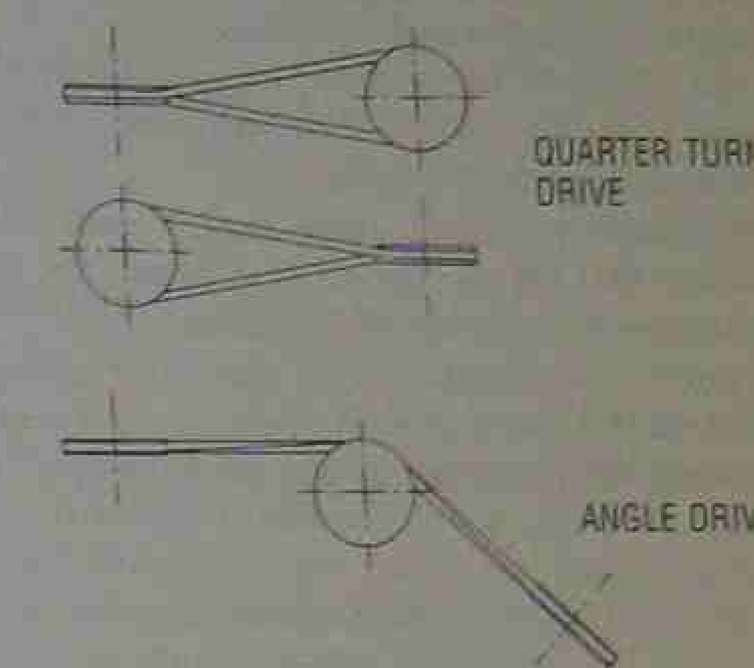


Fig. 7-5 V-belts can be used for quarter turn and angle drives.

Multiple V-belt

In order to increase the capacity of the drive an arrangement which uses several belts mounted on multi-grooved pulleys is often used.

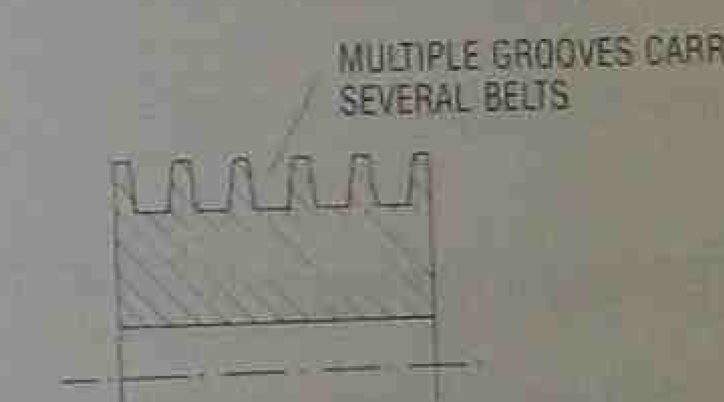


Fig. 7-6 Multi-grooved pulley.

Banded V-belts

In order to overcome the tendency of belts to whip, twist or jump off, a banded V-belt, in which the V-sections are vulcanised to a common band can be used.

Fig. 7-7 A banded V-belt.

Belt tension

V-belts are often tensioned by means of an idler pulley for the following reasons.

- If the relative position of the shafts cannot be adjusted then an idler pulley can be used to assist installation of the belt.
- If the drive is subject to varying loads then a spring loaded idler can provide automatic adjustment of belt tension.
- The inclusion of an idler pulley can help to increase the arc of contact and hence the power transmission capacity of the drive.

It is generally recommended that idler pulleys be mounted on the slack side of a belt, as shown in Fig. 7-8, and positioned close to the drive pulley.

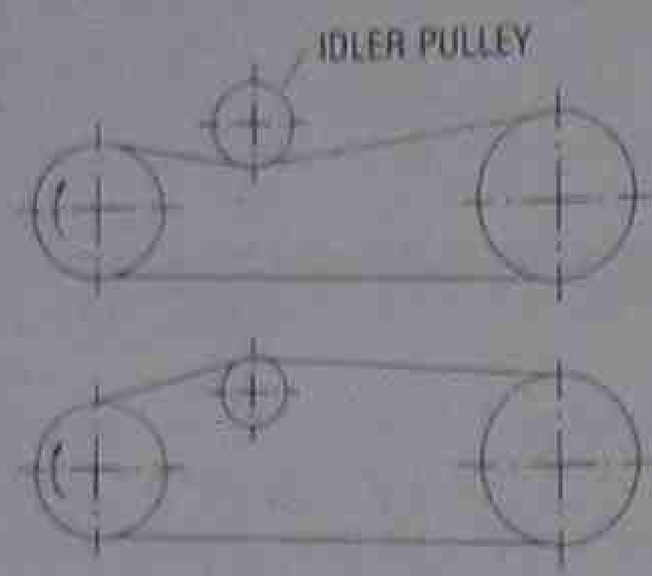


Fig. 7-8 Mounting of idler pulleys.

MAINTENANCE PRACTICES

The following general points should be taken into account in the maintenance of V-belts:

- The operation of V-belts depends largely on the condition and correct positioning of the pulleys.
- V-belt pulleys should be kept clean, free of oil and grease and free from damage and wear.
- Pulleys should be installed parallel and in line with each other.
- Belt tension should be adjusted according to the manufacturer's recommendations.
- V-belts should never be forced or levered on to the pulley as shown in Fig. 7-9.
- Multiple V-belts should be correctly matched to ensure the load is evenly distributed. As well as being the same size it is preferable that all belts are supplied by the same manufacturer.
- Multiple V-belts should be changed as a set. A single new belt will be shorter than the worn, stretched belts and will tend to carry more than its

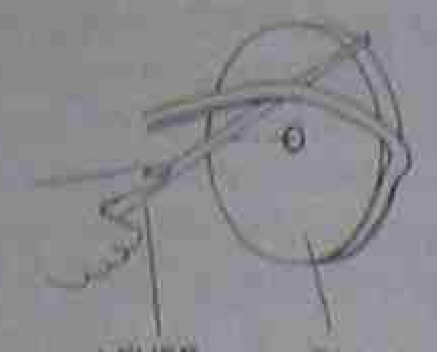


Fig. 7-9 Incorrect method of installing V-belt on pulley.

fair share of load. It is therefore likely to [fail] prematurely.

- No dressings of any kind should be applied to V-belt.
- V-belts should be stored in a clean, dry place and should not be exposed to heat or direct sunlight.
- Do not hang V-belts on nails or small pegs while in storage. Store flat if possible.

Alignment

The correct alignment of the shafts and pulleys is vital to the operation of a V-belt drive. Misalignment of pulleys can occur in several ways.

The first step in aligning the pulleys is to check that the two shafts are level and parallel. This should be done by using a spirit level on the exposed shafts to check for level, and then by using a feeler bar or gauge to check the distance between the shafts on both sides of the pulleys. (Fig. 7-11)

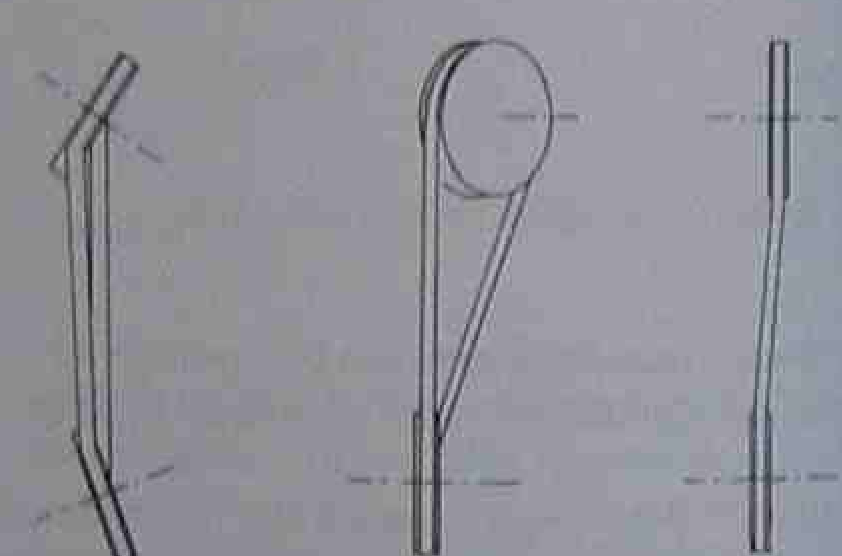


Fig. 7-10 Misalignment of pulleys.

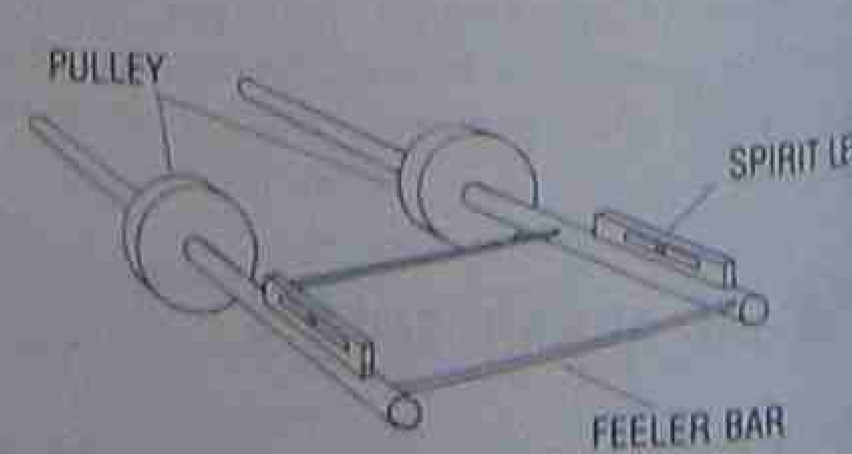


Fig. 7-11 Using a spirit level and feeler bar.

Once the shafts are parallel then the pulleys can be brought into line by using a straight edge across the faces as shown in Fig. 7-12.

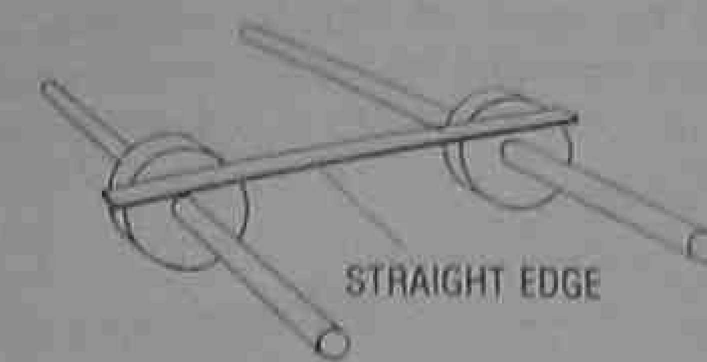


Fig. 7-12 Pulleys brought into line using a straight edge across the faces.

Reposition either of the pulleys until they are properly aligned. If either shaft is subject to end float make sure that it is in its normal running position when the alignment is checked. Rotate the shafts and check the alignment in several positions before it is finally accepted.

Belt tension adjustment

It is important that V-belts run with the amount of tension recommended by the manufacturer. If tension is insufficient then the belt will slip and overheating and wear will result. If the belt is too tight it will also cause overheating as well as damage to bearings.

There are a number of ways in which belt tension can be checked. The most common is to depress the belt and measure the deflection using a ruler and straight edge as shown in Fig. 7-13.

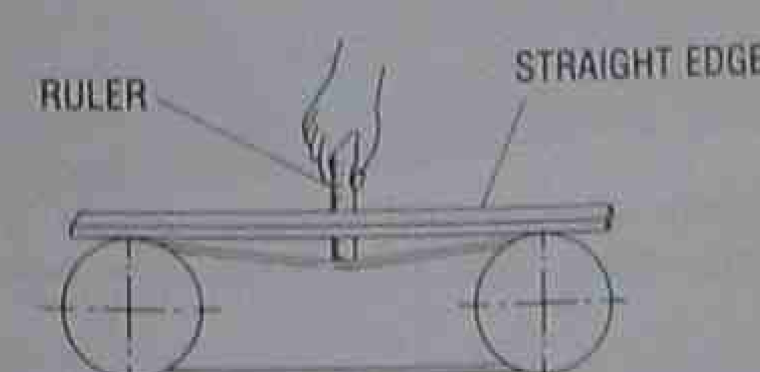


Fig. 7-13 Checking belt tension by depressing the belt and measuring the deflection.

If this method is not considered sufficiently accurate then a spring balance can be used to deflect the belt against a specified pull as shown in Fig. 7-14.

Fig. 7-14 Using a spring balance.

A third alternative is to measure the elongation of the belt under tension. This is done by marking a defined length of the belt and then remeasuring between the same two marks when the belt is under tension as shown in Fig. 7-15.

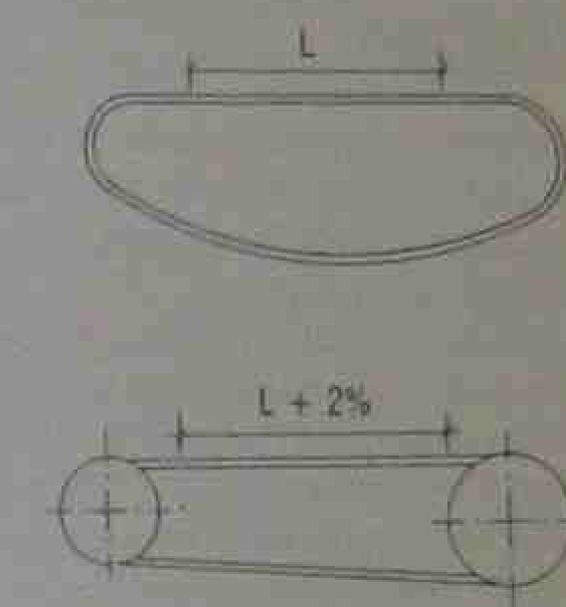


Fig. 7-15 Measuring the elongation of the belt under tension.

For normal drives the elongation of the belt should be around 2%. This may increase for high speed drives and should be checked against manufacturer's recommendations.

Installation of V-belt drives

The following procedure should be adopted when installing V-belts.

1 Check the condition of the pulleys and check for wobble.

Fig. 7-16 Pulley wobble.

2 Align the pulleys according to the procedures outlined above.
3 Reduce the centre distance of the shafts by adjusting the position of the motor. If an idler pulley is used release the tension on the idler so that the belt can be fitted.
4 Fit the belt or belts taking care not to damage them in the process.
5 Adjust the belt tension as recommended above.
6 Run the unit for a short period, say 10 minutes, to allow the belt to seat correctly in the pulley grooves.
7 Readjust the tension.
8 Recheck the tension after 24 hours of operation.
9 Ensure that the drive is protected with a suitable guard at all times during operation. The guard should be provided with ventilation and secure against removal by unauthorised personnel.

FAILURE PATTERNS

Like most machine elements, V-belts do not have an unlimited life and will eventually wear out. Their normal life expectancy will depend on the operating conditions, speed and loading. When belts fail prematurely it is important to determine the precise cause of failure so that corrective action can be taken and performance improved for future operation. As with other machine elements an investigation of failure patterns can be divided into an analysis of symptoms and an analysis of causes.

Symptoms fall into two categories, those apparent during operation and those visible on shutdown of the machine and inspection of the drive.

Operating symptoms

The following conditions can be considered to be evidence of malfunction.

Belt slippage

Any tendency of a V-belt to slip will lead to rapid wear and premature failure. The common causes of belt slippage are insufficient tension, drive overload and the presence of oil or grease on the belt.

Belt squeal

Squealing often accompanies belt slippage and is also caused by overload and insufficient tension. It may also occur when the arc of contact between belt and pulley is insufficient.

Belt ticking or slapping

When the operation of a belt drive is accompanied by a ticking or a slapping sound this is often evidence that some form of mechanical interference is taking place. This may be due to poorly aligned gu... contact with other machine parts.

Belt whipping

If a V-belt starts to whip it is likely to jump out... pulley groove or to roll over and become da... Whipping may be the result of the drive c... being too far apart or due to wobbling p... Sometimes a pulsating load will also cause belt... in which case the suitability of the drive sho... reviewed.

Belts turned over

If the cords in the belt are broken during insta... by levering the belt on to the pulley then the be... stretch excessively and lose strength. Lack of te... may also allow a belt to roll over in the pulley gr... The effect of impulse loads and whipping ma... cause the belt to roll over and this may be over... by installing a spring loaded idler pulley. Once... has turned over it will be damaged and shou... automatically discarded and replaced.

Belt breakage

If a V-belt breaks immediate action is ... required. Breakage may occur due to overloa... the form of shock loads or heavy starting loads... the belt has been properly selected this shoul... occur. A belt that has been weakened by b... levered on to the pulley is very likely to b... prematurely. The presence of foreign objec... material may also damage the belt sufficiently... to break.

Symptoms found on inspection

When the drive is shut down and examined... condition of belts and pulleys will provide evid... from which the cause of failure may be determ... As for all machine elements, every failure is diffe... in some way. However, the following conditio... commonly-found symptoms of V-belt malfunc...

Wear

A properly aligned and tensioned belt will ... along the sides and will eventually need ... replaced. If wear is rapid and leads to premature failure this may be due to misalignment or the presence of dust or other abrasives.

Uneven wear, as shown in Fig. 7-18, may be the result of either misalignment or damage to the pulley grooves.

Gates...

Fig. 7-17 A typical example of normal wear.

Gates Rubber

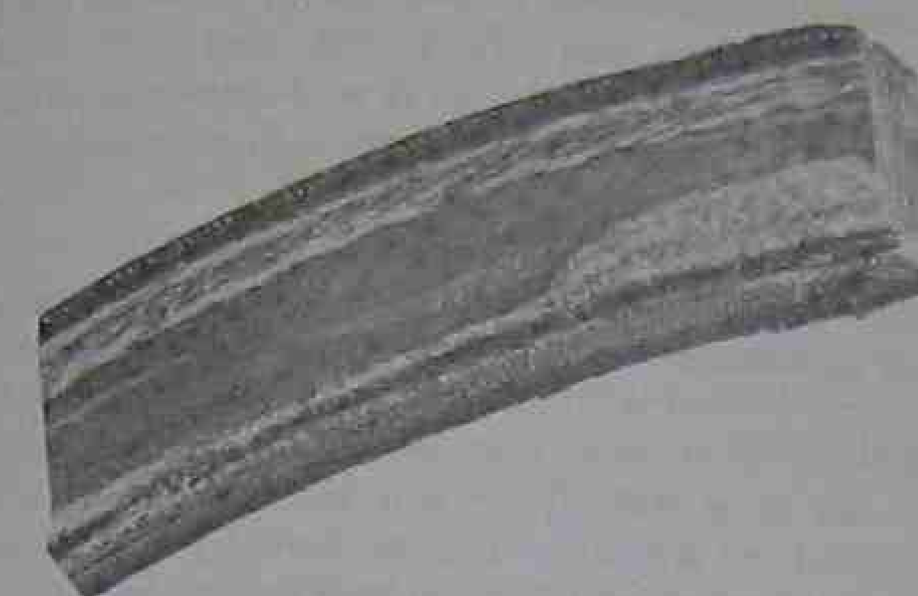


Fig. 7-18 Example of uneven wear.

Cracking

When hardening and cracking appear on the underside of the belt this is usually caused by excessive heat build-up. This may be caused by poor ventilation or by slippage.

Gates Rubber

Fig. 7-19 Hardening and cracking of the belt.

Fraying

Any tendency for the belt to fray along the edges or for the surface to tear and rupture, as shown in Fig. 7-20, is usually evidence that some mechanical interference is taking place or that the pulleys are worn or damaged in some way.

Gates Rubber

Fig. 7-20 Fraying of the belt.

Stretching

If a belt stretches beyond the adjustment range of the tightener then the chances are that the internal cords are broken and the belt should be replaced.

Swelling

If the belt material becomes swollen or spongy it is likely that it has been exposed to oil, grease or other chemicals.

Burns

If a belt shows evidence of burning in one particular area this may indicate either that the belt has slipped during start-up or that the driven unit has jammed or stalled causing the belt to burn when the drive has run on.

Gates Rubber

Fig. 7-21 Evidence of burning in one particular area.

Worn sheaves

In addition to belt damage the pulley grooves may show evidence of wear. This may appear on the sides of the groove (Fig. 7-22), or as a shiny surface on the bottom of the groove (Fig. 7-23) which indicates that the belt has been bottoming. A pulley which shows excessive wear of this type should be replaced and new belts installed.

Fig. 7-22 Dishing of the sides of the groove.

Fig. 7-23 Evidence that the belt has been bottoming.

Causes of failure

The common causes of the types of V-belt failure listed above can be summarised as follows:

Misalignment

As with most mechanisms good alignment is critical. Misalignment of pulleys will cause rapid wear and damage to the belt surface, although alignment tolerances for V-belts are not as stringent as for other drives. The manufacturer's instructions should be consulted to establish the appropriate limits for a particular machine.

Incorrect tension

Insufficient tension will cause the belt to slip and may also lead to belt breakage due to the grab-slip effect. Over-tensioning will increase the wear rate and shorten belt life and may also cause overloading of shaft bearings.

Interference

Any contact between the belt and another part of machine or the belt guard will cause rapid damage in the form of fraying or excessive wear of the belt surface.

Foreign material

Belts and pulley grooves should be kept clean, free of dirt, grit and other contaminants which cause accelerated wear or even belt breakage. Oil, other chemicals will attack the belt and cause deterioration of the material. A well-constructed guard should help to exclude foreign material.

Damaged pulleys

Pulley grooves should be free from nicks, burrs, chips and other damage that may affect belts.

Overloading

The life of a belt drive will be reduced if it is subjected to loads or speeds beyond its design capacity. Excessive wear and belt breakage is likely to occur if loads and speeds are too high.

Overheating

Excessive heat is an enemy of V-belts and can lead to their rapid deterioration. Guards should be made of mesh to allow adequate ventilation, and belt slippage should be corrected as quickly as possible before heat build-up affects the belt material.

Summary of the common symptoms and causes of V-belt failure

Symptoms: Operating	Symptoms: Inspection	Causes
Slipping	Wear	Misalignment
Squealing	Cracking	Incorrect tension
Ticking	Fraying	Interference
Whipping	Stretching	Foreign material
Turn-over	Swelling	Overloading
Breakage	Burns	Overheating
	Worn sheaves	Damaged sheaves

7.2 CHAIN DRIVES

Chains and sprockets provide a positive form of drive which does not slip and they can therefore be used where synchronisation of motion is important. There are various types of chains available, the most common being the roller chain dealt with here. Many of the maintenance practices described also apply to other types of chains such as silent chain, rollerless chain, etc.

PRINCIPLES OF OPERATION

Chains and sprockets fulfil the same basic function as belts and pulleys in transferring power between two parallel shafts. Instead of relying on friction, a chain drive is a positive drive in which the links of the chain engage with specially formed teeth on the sprocket.

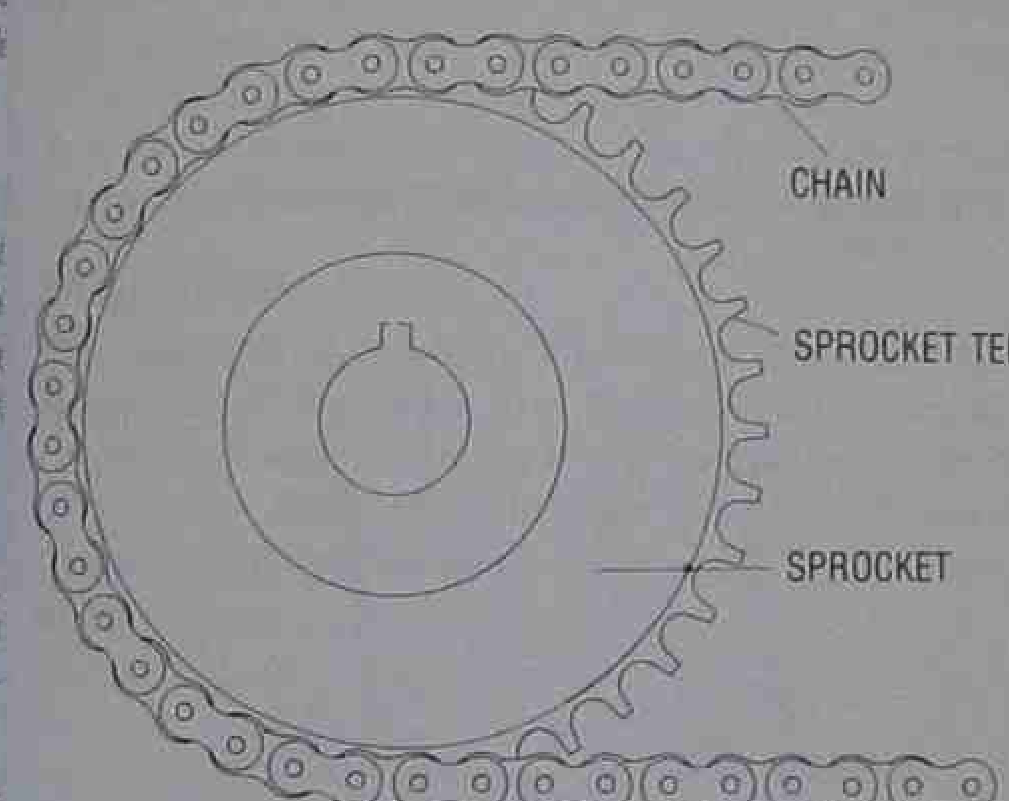


Fig. 7-24 Chain and sprocket.

Standard roller chain is made up of alternate roller links and pin links.

The pitch of the chain is determined by the length of the side plates, and the bushings and pins are press-fitted into the side plates. The pins of a special joining link may be longer and grooved to take spring clips as shown in Fig. 7-26.

The rollers are free to rotate on the bushings, and this reduces the rubbing action between the chain and the sprocket as the chain links roll on to the sprocket and thus avoid excessive wear from sliding friction. Because each roller and bushing functions like a plain journal bearing, lubrication is essential to the operation of chain drives.

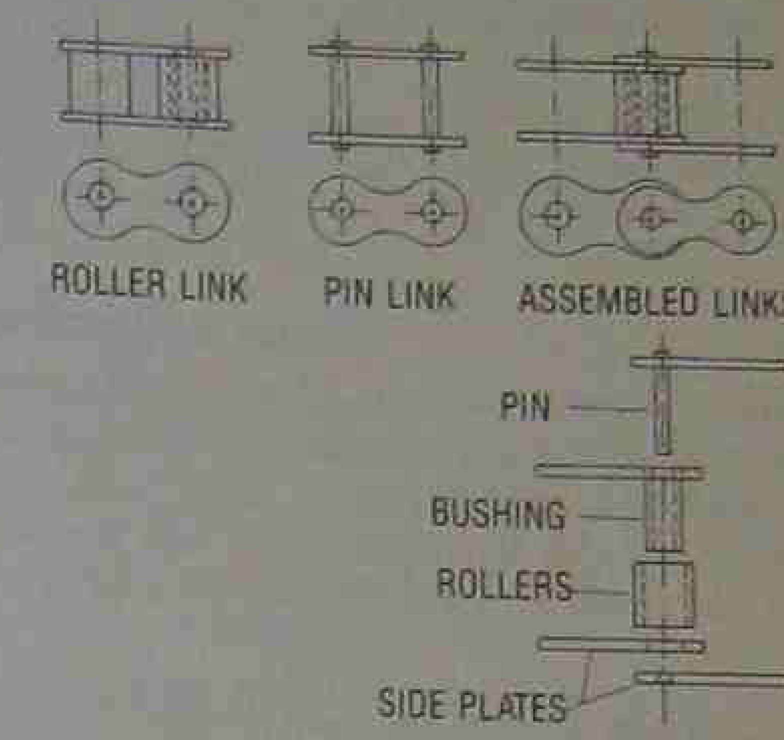


Fig. 7-25 Standard roller chain.

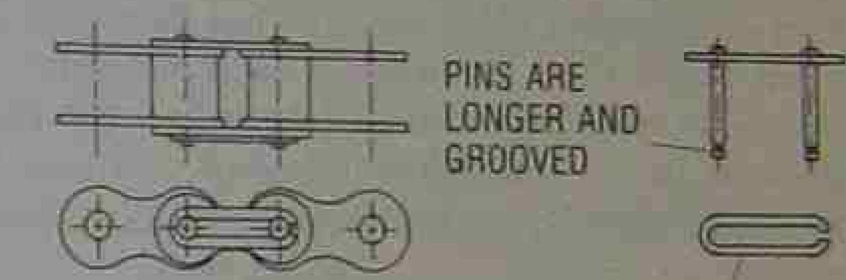


Fig. 7-26 Special joining link.

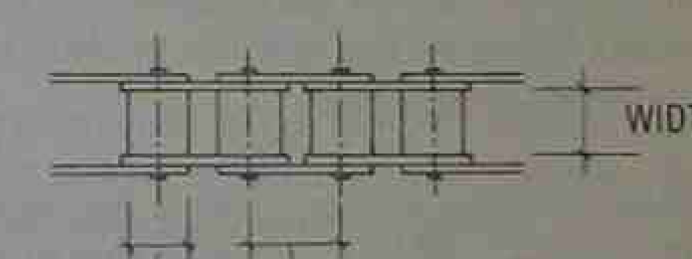


Fig. 7-27 Pitch, width and roller diameter are the critical dimensions of roller chain.

The critical dimensions by which roller chain is identified are the pitch, width and roller diameter.

Chain drive sprockets have teeth cut around the periphery, like a gear, and are specified by the pitch circle diameter, width and number of teeth. They are usually manufactured with an integral hub as shown in Fig. 7-28.

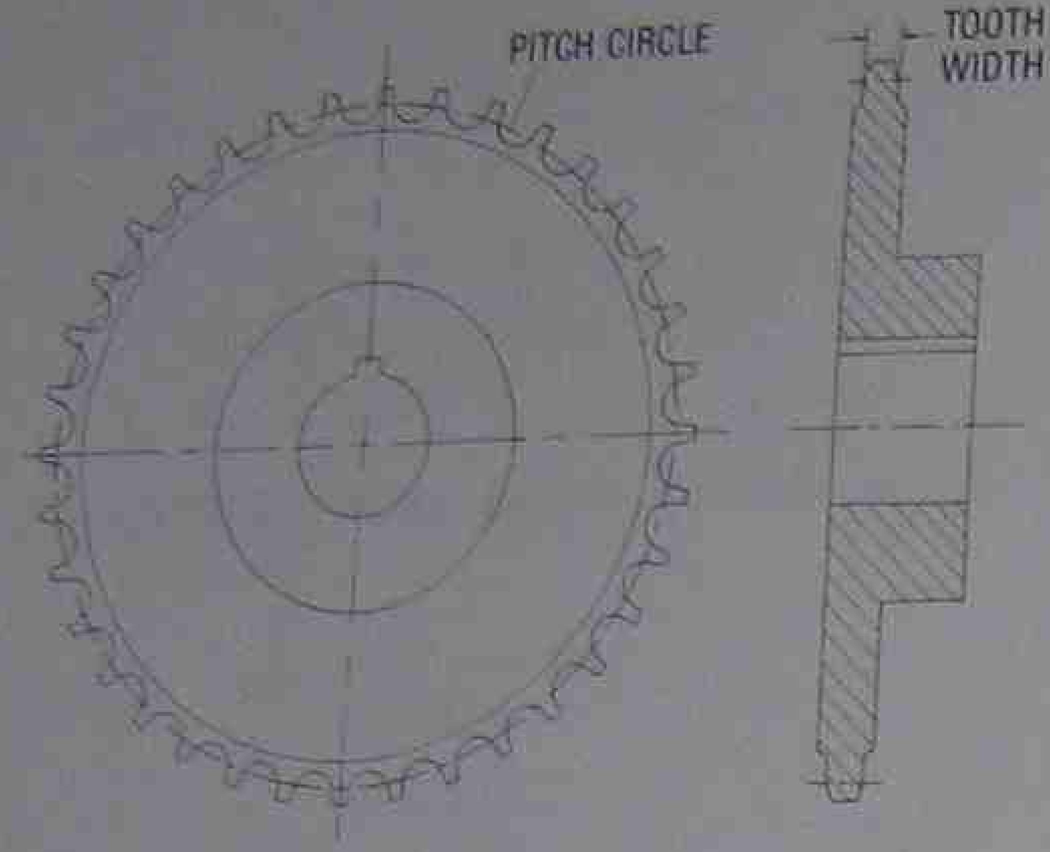


Fig. 7-28 Chain drive sprockets are usually manufactured with an integral hub.

It is normal practice to design chain drives in such a way that the number of chain pitches and the number of sprocket teeth ensure that the same link does not contact the same tooth each revolution. If there are an even number of pitches on the chain there must be an uneven number of teeth on the sprocket and vice versa. This helps to reduce uneven wear.

Chain drives are more sensitive to misalignment than belt drives and must be properly tensioned. They are generally suitable for speeds up to 1350 metres per minute (4500 feet per minute).

The speed of the driven sprocket in relation to the speed of the driver can be determined by using a simple formula based on the number of teeth on the driver and driven sprockets:

$$\text{speed of driven (RPM)} = \text{speed of driver (RPM)} \times \frac{\text{no. of teeth on driver}}{\text{no. of teeth on driven}}$$

TYPES AND ARRANGEMENTS

Standard roller chain is available in single and multi-strand form, and the number of strands required will depend on the power to be transmitted. Double pitch chains are also available. They are cheaper, and are suitable for light loads and low speeds.

Chain drives are used most commonly as horizontal drives and any slack in the chain, resulting from wear, should accumulate on the lower strand as shown in Fig. 7-30.

Vertical drives should be arranged so that accumulated slack falls into the driven sprocket rather than away from it, to prevent misengagement.

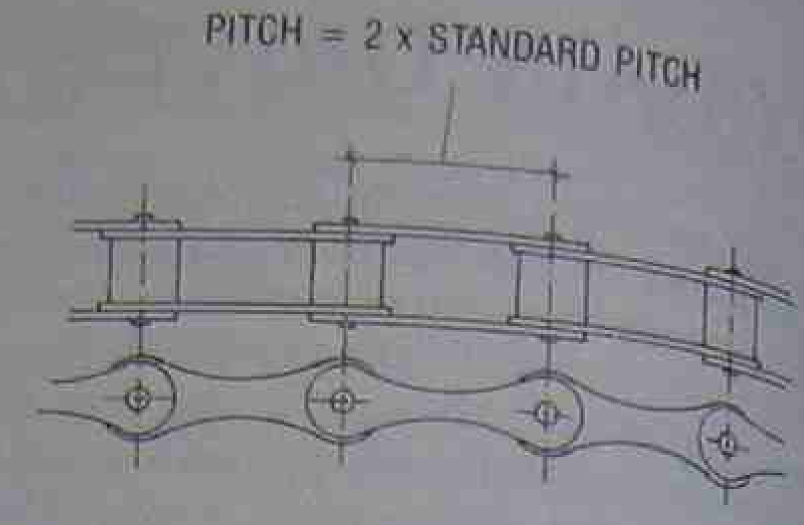


Fig. 7-29 Double pitch chain.

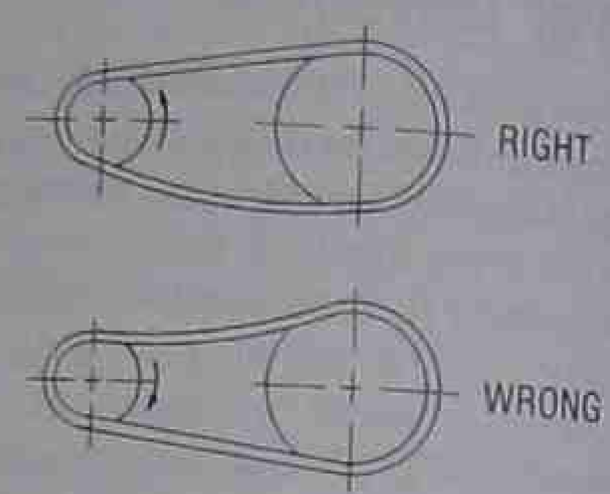


Fig. 7-30 In a horizontal drive, slack should accumulate on the lower strand.

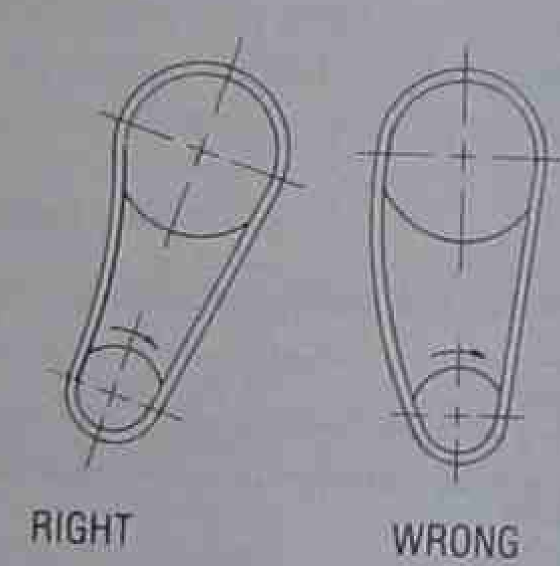


Fig. 7-31 Accumulated slack in a vertical drive should fall into rather than away from, the driven sprocket.

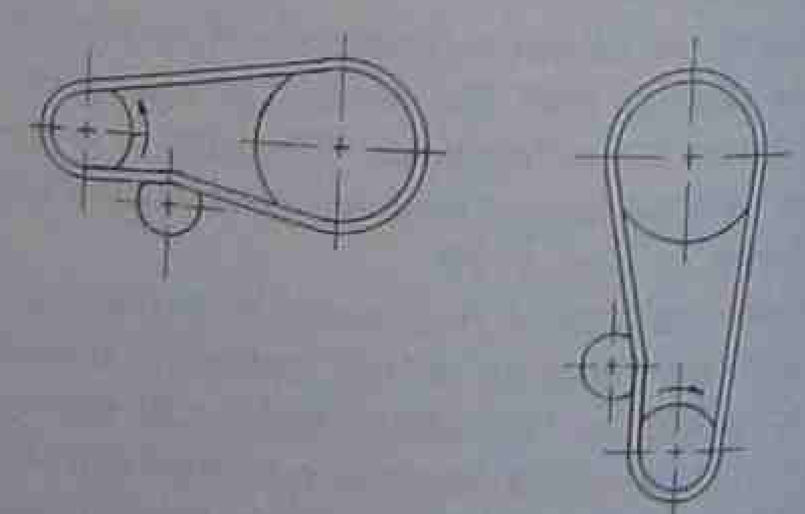


Fig. 7-32 Using a chain tensioner.

Where chain tensioners are used they should be used on the side of the chain where the slack is expected to accummulate. (Fig. 7-32)

MAINTENANCE PRACTICES

The general points to be taken into account in the maintenance of chain drives are:

- As with belt drives, alignment and proper tensioning are critical to the operation of the drive.
- Chain drives should be kept clean and protected from dirt and should be provided with an adequate supply of a suitable lubricant.
- New links should not be installed in chains that have been significantly lengthened by wear.
- New chains should not be installed on badly worn sprockets. Sprockets may be reversed on the shaft to extend their life if necessary.
- New chains should be stored in protective wrappings until ready for use, and protected from excessive heat and moisture.
- Chain drives, like belt drives, should be properly guarded and protected from interference.

Alignment

The alignment of chain sprockets is critical to the operation of the drive and the procedure should be carried out with care. As for V-belt pulleys, the two shafts are first checked for level and parallel alignment using a spirit level and feeler bars or gauges.

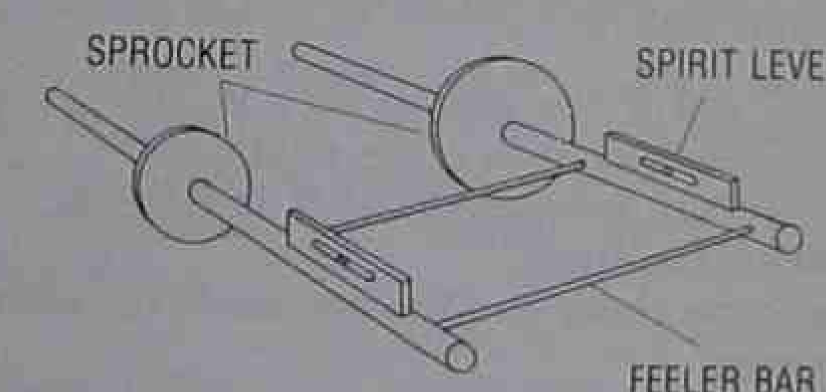


Fig. 7-33 Checking for level and parallel alignment of the shafts.

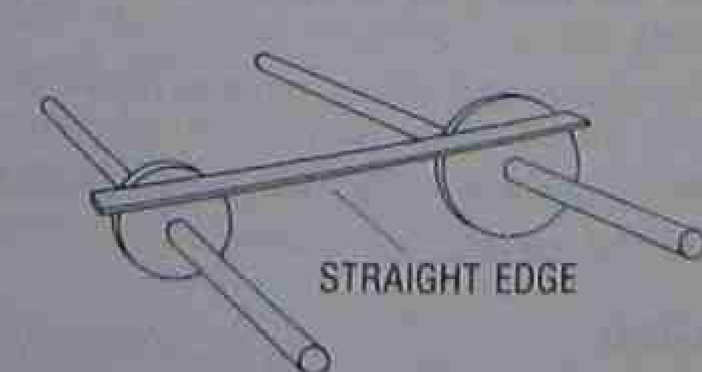


Fig. 7-34 Checking alignment of the sprocket faces.

A straight edge can then be used to check alignment of the sprocket faces as shown in Fig. 7-34.

If either of the shafts is subject to end float then align the sprockets with the shaft in its normal running position.

If the centre distance between the shafts is too great for a straight edge a taut piano wire may be used instead. Rotate the sprockets and check the alignment in several positions.

Slack adjustment

Unlike belt drives, chain drives do not require initial tension but should merely be adjusted to take up the slack. If a chain is too tight it will bind on the sprockets and wear rapidly. If it is too loose it will tend to whip, which will cause vibration and reduced life.

For chains installed on units with adjustable centres, the slack should be adjusted to approximately 2% of the centre distance. In other words, if the centre distance is one metre then the slack should be about 20mm. This can be adjusted by pulling the chain taut on one side and measuring the slack on the other using a ruler and straight edge.

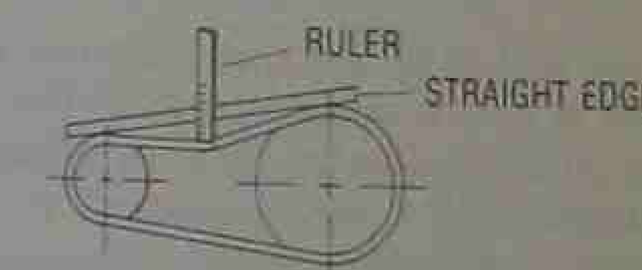


Fig. 7-35 Measuring the slack in a unit with adjustable centres.

For drives on fixed centres, an idler sprocket or spring loaded tightener of some kind will automatically take up the slack.

It is important to recognise that chains stretch during operation and drives with adjustable centres may need to be readjusted from time to time. The so-called stretch is not due to any physical deformation of the chain links but due to wear in the rollers, bushings and pins which must be compensated for.

Installation of chain drives

The following procedure is recommended when installing a chain drive.

1 Check the condition of the sprockets and make sure they are clean and free from damage.
2 Align the sprockets as recommended above.
3 Loosen the tighteners so that the chain will fit over the sprockets.

4 Remove the chain from its wrapping and bring the ends together over one of the sprockets. Use the sprocket teeth to hold the chain and install the final link.

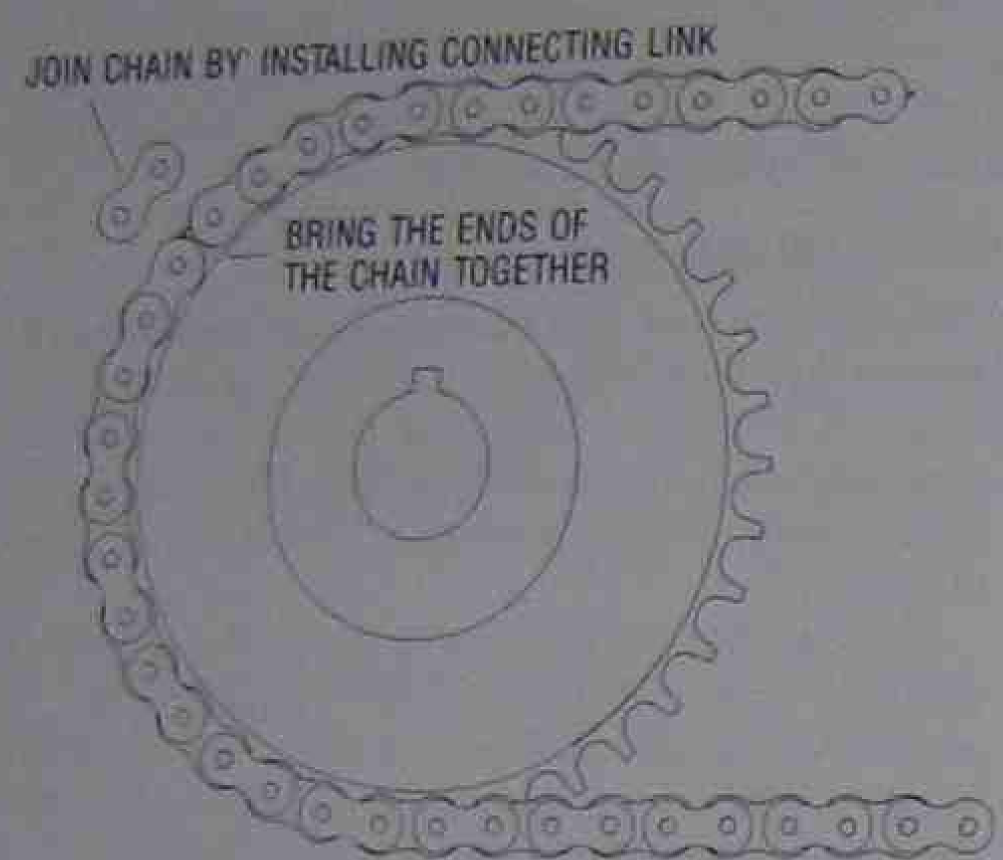


Fig. 7-36 Using the sprocket teeth to hold the chain.

5 Fit the side plate and spring clips or whatever device is used to secure the pins.
6 Take up the slack in the drive as outlined above.
7 Lubricate the chain according to the manufacturer's instructions.
8 Start up the machine and check that the chain runs true, without excessive noise and without binding or whipping.
9 Ensure that the lubrication system is working properly.
10 Install a suitably designed guard to prevent interference with the drive. (Do not attempt to do this while the drive is in operation.)

Chain removal

If the drive adjustment allows the chain to be removed without being broken by the removal of a link then use this method. If the chain then needs to be shortened by removing a link a chain detacher should be used to hold the chain while the link is removed. (Fig. 7-37)

Chain detaching tools are also available to drive the link pins out. (Fig. 7-38)

Cleaning

Whatever the operating conditions it is advisable to remove the chain and clean it from time to time. The debris of normal wear and gummed lubricant will cause wear to the pins and bushings if it is not

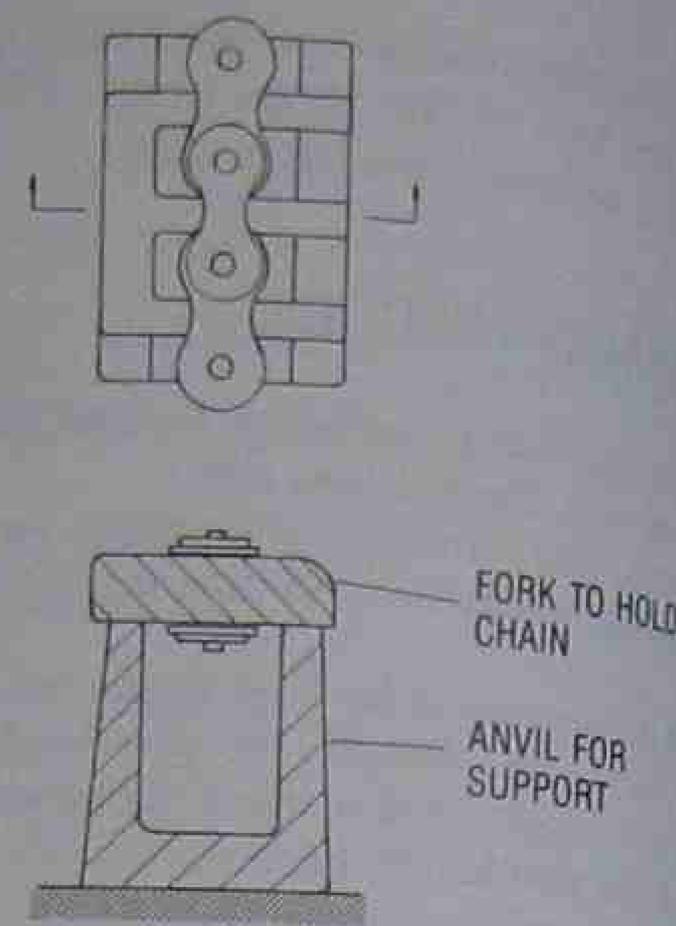


Fig. 7-37 A chain detaching tool.

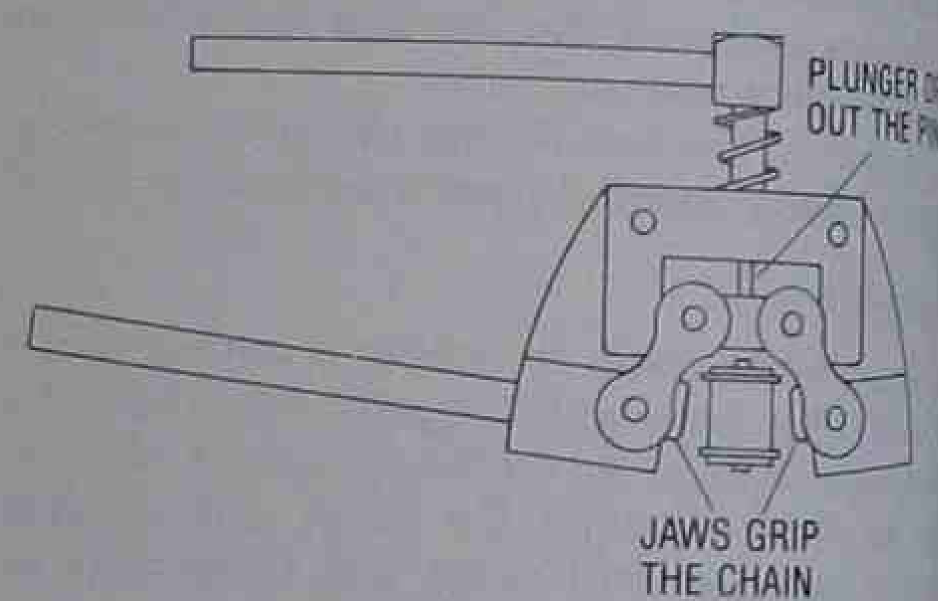


Fig. 7-38 A chain detacher that will drive out the link pin.

removed and if the atmosphere is dusty, rep[illegible] cleaning becomes even more important.

The procedure recommended is as follows:
1 Remove the chain.
2 Check the chain and the sprockets for wear [illegible] corrosion.
3 Wash the chain in kerosene or similar clea[illegible] fluid. Soaking may be necessary for a very d[illegible] chain.
4 Drain off the cleaning fluid and soak the cha[illegible] lubricating oil.
5 Hang the chain and allow excess lubricant [illegible] off.
6 Clean the sprockets and check alignment.
7 Reinstall the chain.

Lubrication

Proper lubrication is critical to the operation of ch[illegible] drives and the achievement of satisfactory ser[illegible] life. The general principles of lubrication apply [illegible] the following points should be remembered.

- Lubrication should be regular and the frequency determined by the operating conditions.
- Whatever lubrication system is employed, the lubricant must penetrate the chain joints.
- Chains should be cleaned regularly so that lubricant can flow into the joints.
- The higher the chain speed the greater the supply of lubricant required.
- Some means of protecting the chain from dirt and other contaminants should be provided.

The most common methods of chain drive lubrication are:

Manual using a brush or oil can suitable for simple drives
Drip feed low horsepower and low speed drives
Automatic using an oil bath or an oil spray suitable for moderate to high speed drives

FAILURE PATTERNS

The common symptoms and causes of chain drive malfunction and failure can be summarised as follows:

Operating symptoms

Noise

Chain drives are generally relatively noisy, certainly more so than belt drives, but if noise becomes excessive it may be an indication of malfunction. As for other indicators of the condition of machine elements, it is useful to establish an operating level when the system is new and properly adjusted so that changes in noise level can be detected. Intermittent ticking or slapping sounds may be associated with interference and should be investigated immediately.

Chain jumping

Chains can have a tendency to climb the sprockets and can reach the point where they jump off. This may be due to wear or to excessive slack, both of which allow the chain to ride high on the sprocket teeth. (Fig. 7-39)

Alternatively, excessive material build-up on the sprocket teeth may interfere with the correct mating of the chain and teeth.

Chain whipping

Too much slack, or pulsating loads, may cause the chain to whip. This may also occur if some of the chain links have become stiff or have seized.

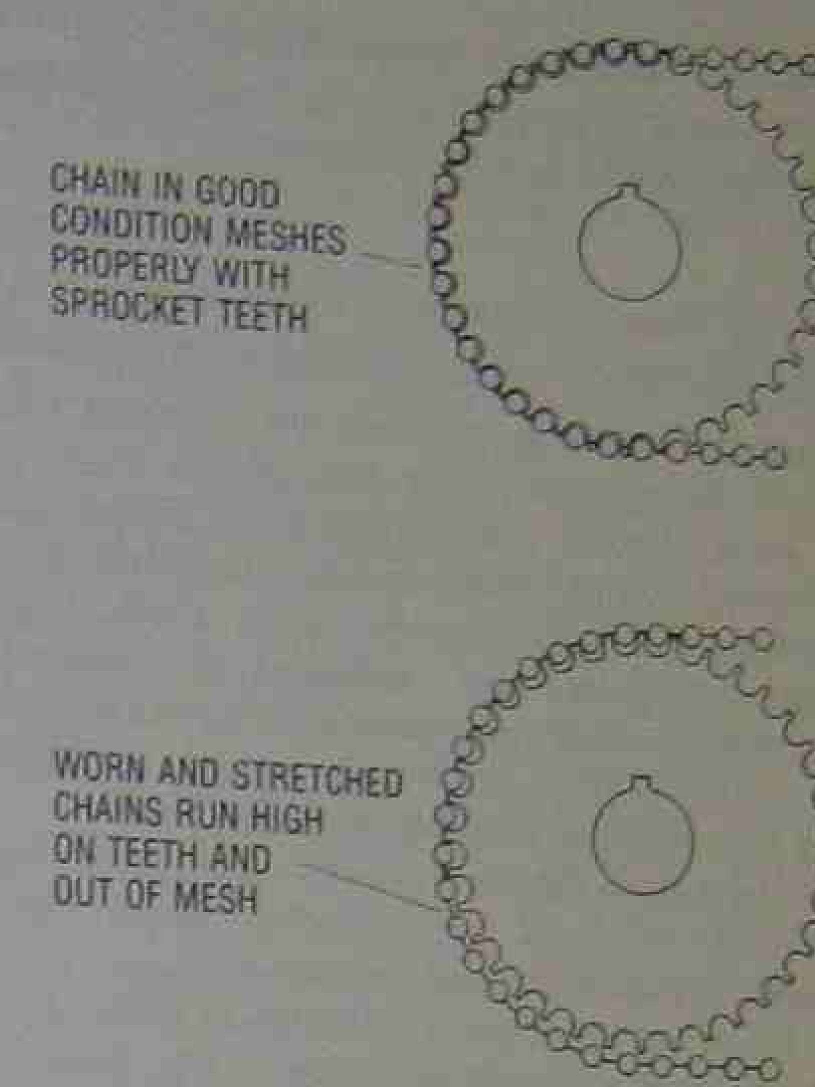


Fig. 7-39 Wear or excessive slack can cause the chain to ride high on the sprocket teeth.

Chain breakage

It is rare for a chain to break completely although this may occur if a condition of malfunction has existed for some time and excessive fatigue has taken place. It can also occur when a chain is badly worn. The lengthened chain may jump a tooth on the sprocket which causes excessive tension and consequent failure of the side plates.

Overheating

Excessive operating temperatures are an indication that the drive is operating at too high a speed, with too great a load, or with inadequate lubrication. It is often difficult to determine the temperature of a chain during operation but it can be checked as soon as the machine is shut down. Once again, a comparison with normal operating conditions must be made in order to detect evidence of malfunction.

Fig. 7-40 Evidence of chain wear.

Symptoms found on inspection

Once the drive is shut down and inspected the following symptoms may be evident.

Wear

Chains may show evidence of wear on the pins, bushings and rollers.

The amount of wear that can be tolerated will depend on how much elongation of the chain it causes. As a general rule, single pitch chains can tolerate up to around 3% elongation whereas double pitch chains can only tolerate up to $1\frac{1}{2}$%. The manufacturer should be consulted for precise tolerances. If wear becomes excessive the chain may jump a tooth on the slack side of the sprocket resulting in excessive loads in the side plates.

If wear occurs on the insides of the side plates and also on the sides of the sprocket teeth then misalignment of the drive is indicated.

Wear to the sprocket teeth will often be evident in the form of 'hooked' teeth as shown in Fig. 7-41.

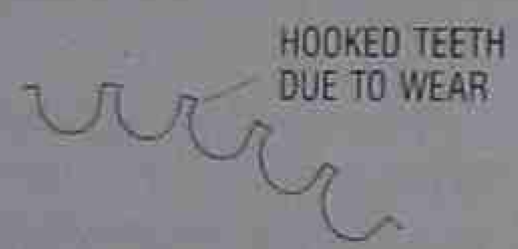


Fig. 7-41 Evidence of wear to the sprocket teeth.

Galling

If lubrication is inadequate at high loads or speeds the bearing surfaces of the pins and bushings may weld and then tear apart. The resultant effect is known as galling.

Fig. 7-42 Galling in a chain drive.

Corrosion

The presence of rust or surface pitting and roughening is evidence that corrosion is taking place.

Fig. 7-43 Evidence of corrosion.

Link seizure

The links of a chain should be free to move wi significant resistance so that the chain can properly with the sprocket teeth. If the chain become tight or seize altogether this will affe operation of the drive and may cause whipp jumping of the chain. Tightness may be due t build-up of foreign material or products of corr which prevent the lubricant from being effec Damage to the side plates from interferenc careless handling may also cause stiffness.

Causes of failure

The common causes of the symptoms listed a are as follows:

Misalignment

Correct alignment is extremely impor Misalignment of sprockets leads to uneven wea chain and sprockets and also results in operation.

Incorrect slack adjustment

If the slack in the chain is either too much or too l the operation will be noisy. The chain may clim on the sprocket teeth and tend to jump when chain is too loose. Excessive wear can als expected.

Lack of lubrication

Inadequate lubrication will affect the operati chain drives just as it will other machine elem Excessive noise and rapid wear will be the obvious symptoms, with chain breakage being ultimate result if no action is taken.

Material build-up

The build-up of foreign material on chain sprockets due to the presence of dust or o contaminants will prevent the lubricant from b

effective, and may interfere with the correct meshing of the chain and sprockets. Proper protection and regular cleaning will help to eliminate this problem.

Interference

Any interference between the chain and guards or other machine elements will cause wear and damage to the chain. It should be recognisable by the increase in noise level.

Vibration

Excessive vibration, transmitted along either shaft to the drive sprockets, will tend to increase wear and may cause failure of the chain fasteners.

Overload

Rapid wear, chain breakage and overheating may all occur if the drive is subjected to loads or speeds above the recommended design limit.

Worn Sprockets

Excessive wear or other damage to sprockets will hasten the deterioration of the condition of the chain. New chains should never be installed on worn sprockets.

Summary of the symptoms and causes of chain drive failure

Symptoms: Operating	Symptoms: Inspection	Causes
Noise	Wear	Misalignment
Jumping	Galling	Slack adjustment
Whipping	Corrosion	Lack of lubrication
Breakage	Link seizure	Material build-up
Overheating		Interference
		Vibration
		Overload
		Worn sprockets

7.3 GEAR DRIVES

Gear drives are used to transmit power from one machine to another where changes of speed, torque, direction of rotation or shaft orientation are required. They may consist of one or more sets of gears depending on the requirements. In most cases the gears are mounted on shafts supported by an enclosed casing which also contains a lubricant.

Most gear drives in use are speed reducers. They reduce the speed of shaft rotation between driver and driven machines and, at the same time, produce a corresponding increase in torque. The recent increased use of high speed machinery, such as centrifugal compressors, has also generated a need for speed increasers.

PRINCIPLES OF OPERATION

A gear is a form of wheel with teeth machined around the outer edge which allow it to engage with another similar wheel or rack.

The most important features of a gear are the tooth profile or cross-sectional shape, and the number of teeth. In modern gears the tooth profile is based on an involute curve which is the shape produced when a line is traced by a point on a cord which is 'unwound' from a cylinder as shown in Fig. 7-44.

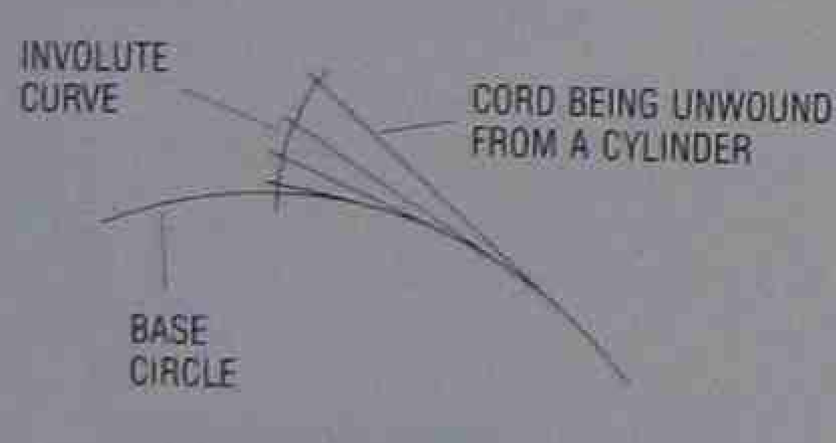


Fig. 7-44 Generating the involute form.

The advantage of the involute form is that when two gear teeth mesh together a constant velocity ratio is maintained with the minimum of sliding and the maximum of rolling action of one tooth against the other. This feature helps to reduce wear and extend the life of the gear.

In order to understand the geometry of gears it is useful to imagine that the simplest form of gears is two plain discs which touch tangentially. If sufficient friction exists between the surfaces in contact then there is no need for special teeth to be cut. However, there is a limit to the torque that can be transmitted by friction and so teeth are cut into the outer edges of the discs to provide a means of positive engagement as shown in Fig. 7-45.

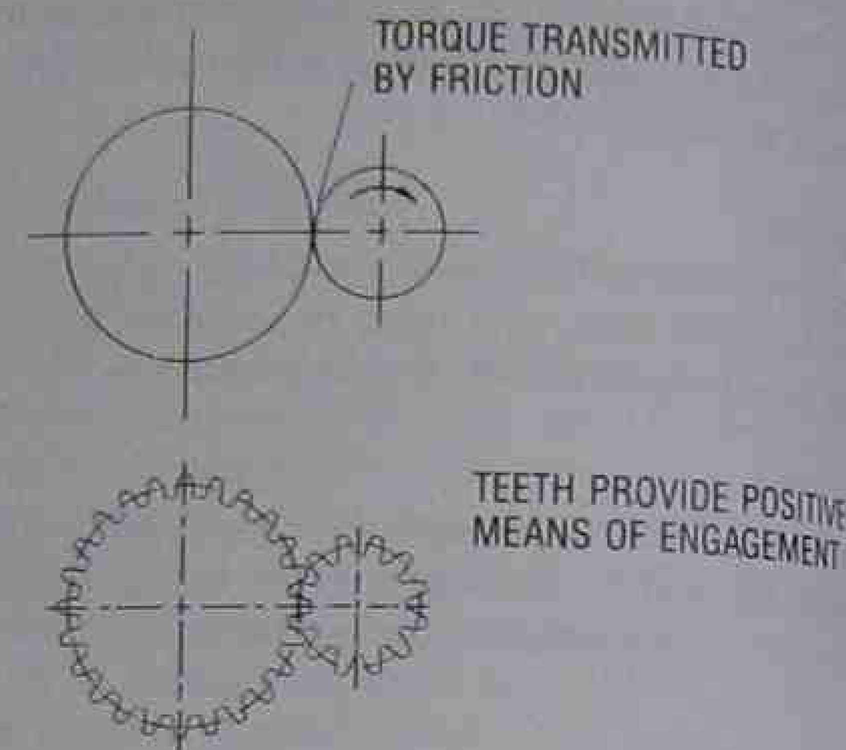


Fig. 7-45 Teeth provide a means of positive engagement.

The imaginary circles on which the gears are c[illegible] are called the pitch circles, and the pitch circle diameter is the major dimension on which gear geometry is based. The other important dimension is the pressure angle. This is the angle between a tangent to the pitch circle and the line of contact of two mating teeth as shown in Fig. 7-46.

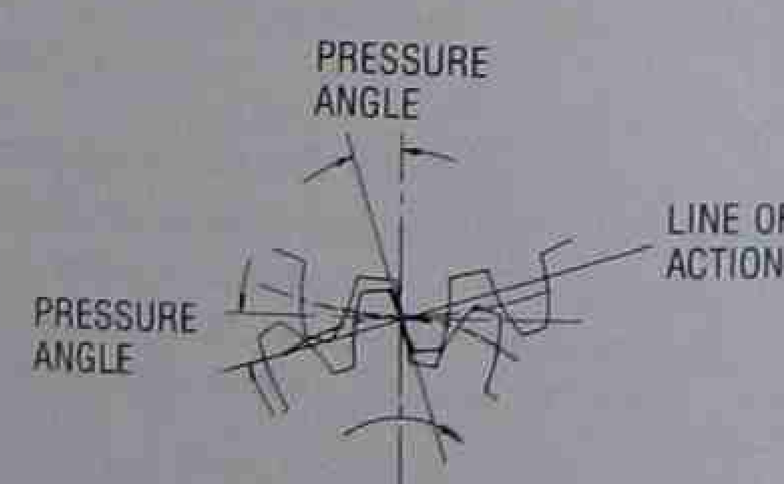


Fig. 7-46 The pressure angle.

A full explanation of the terms used to describe the geometry of a circular gear is given in Fig. 7-47.

If two gears are to mesh properly they must have the same pressure angle. Standard pressure angles of 14.5° and 20° are used with 20° being the most common.

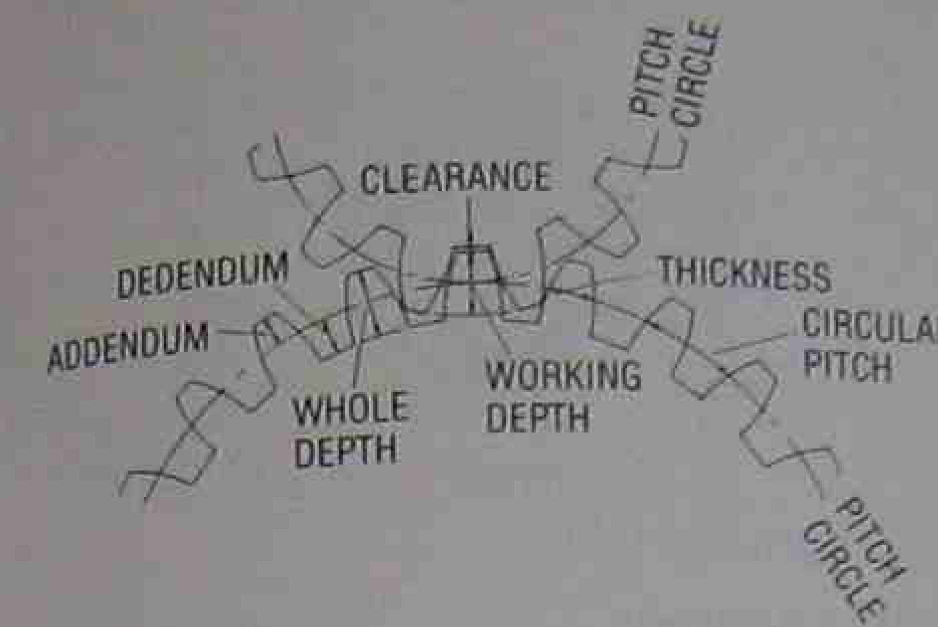


Fig. 7-47 Terms used in circular gear geometry.

In practice, gears are cut to provide running clearance between mating teeth. This is known as backlash.

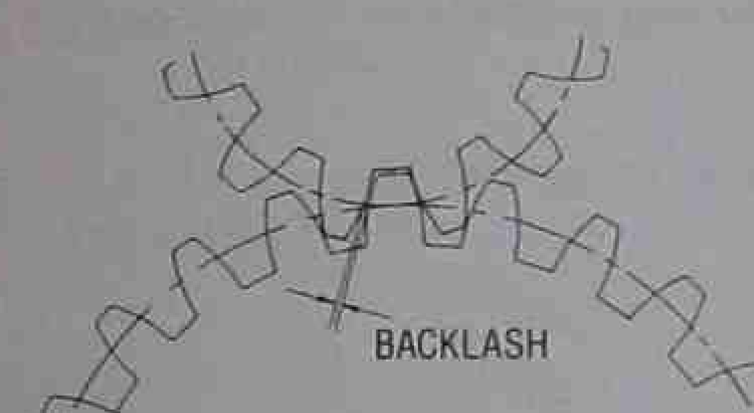


Fig. 7-48 Gears are cut to provide backlash.

The characteristics of mating gears are often described by the term, diametral pitch. This term refers to the ratio of the number of teeth to the pitch circle diameter of the gear and reflects the size and shape of the teeth. Hence two mating gears must also have the same diametral pitch as well as the same pressure angle.

There are several ways in which diametral pitch can be calculated.

- $\text{diametral pitch} = \dfrac{\text{number of teeth}}{\text{pitch circle diameter}}$
- $\text{diametral pitch} = \dfrac{\pi(3.142)}{\text{circular pitch}}$
- $\text{diametral pitch} = \dfrac{\text{number of teeth} + 2}{\text{outside diameter}}$

The speed relationship between two mating gears depends on the number of teeth on each gear and can be determined as follows:

$$\underset{(\text{RPM})}{\text{speed of driven gear}} = \underset{(\text{RPM})}{\text{speed of driver}} \times \frac{\text{no. of teeth on driver}}{\text{no. of teeth on driven}}$$

TYPES AND ARRANGEMENTS

The following types of gears are in common use.

Spur gears

The spur gear is the simplest type of gear and has teeth cut parallel to the axis. Spur gears may be used as external or internal gears or as a rack and pinion.

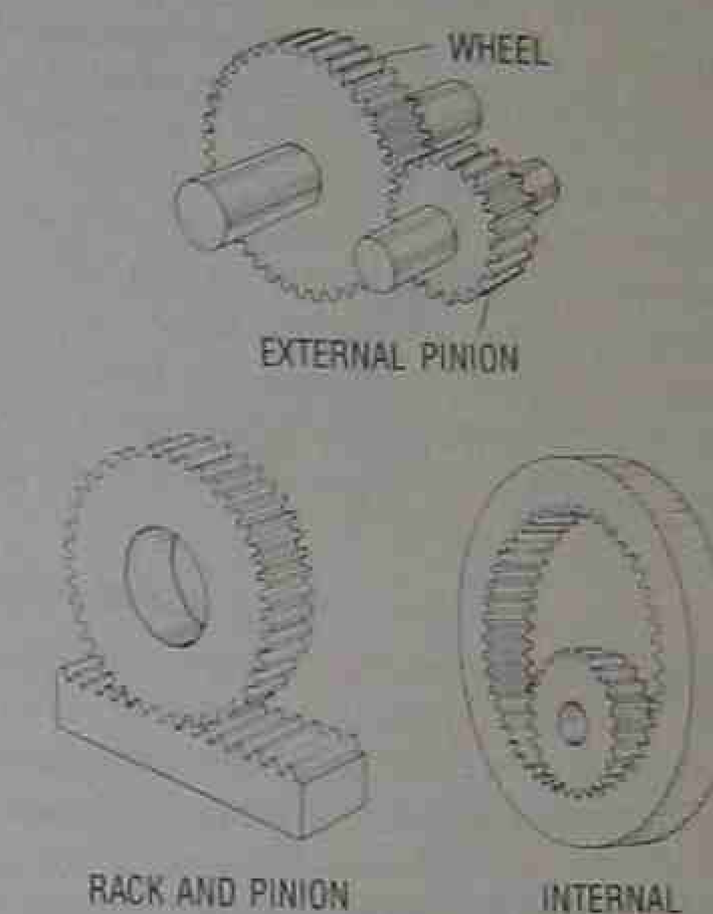


Fig. 7-49 Spur gears.

Spur gears are used to transmit power between parallel shafts operating at moderate speeds. They are simple to manufacture, do not develop end thrust and are the preferred type to be used where practicable.

It is conventional to refer to the large gear as the wheel or bull gear and the smaller as the pinion.

Helical and herringbone gears

Helical gears are also used to transmit power between parallel shafts but have the teeth cut on an angle.

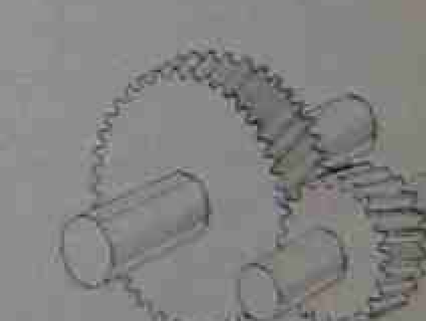


Fig. 7-50 Helical gears have teeth cut on an angle.

The advantage of this design is that several teeth are in mesh at the same time and this results in greater load carrying capacity and smoother operation. Because of the angle of the teeth, helical gears produce end thrust which must be carried by the shaft bearings. This can be overcome by the use of two rows of opposed helical teeth in a 'herringbone' arrangement shown in Fig. 7-51. Herringbone gears are generally not recommended when externally applied end thrust is present or when operating speeds are very high because of the tendency for one helix to carry most of the load.

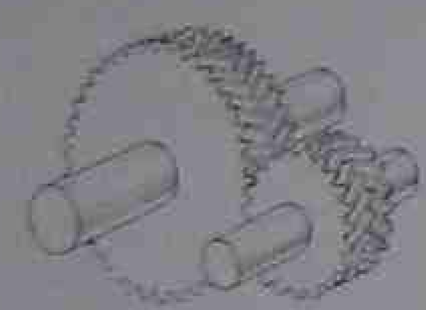


Fig. 7-51 Herringbone gears.

Bevel gears

Bevel gears are used to transmit power between two intersecting shafts, normally at right angles. The teeth on bevel gears may be plain or spiral.

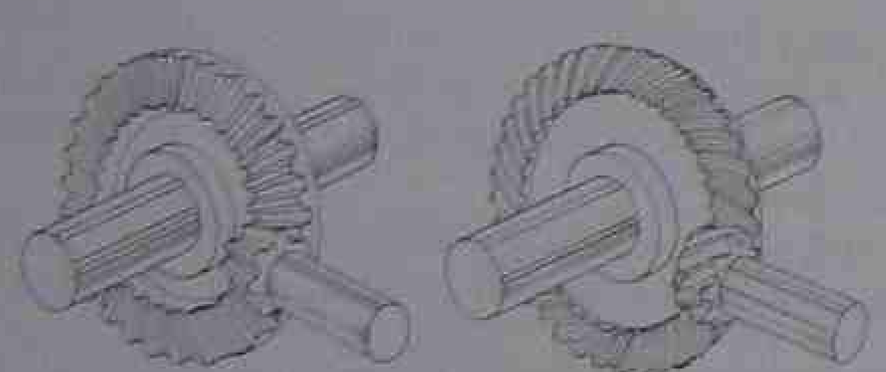


Fig. 7-52 Bevel gear teeth may be plain or spiral.

Spiral bevel gears distribute the load over several teeth, in the same way that helical gears do for parallel shafts, and hence give smoother operation.

Hypoid gears

A variation of the spiral bevel gear is the hypoid gear which is designed to transmit power between two non-intersecting and non-parallel shafts.

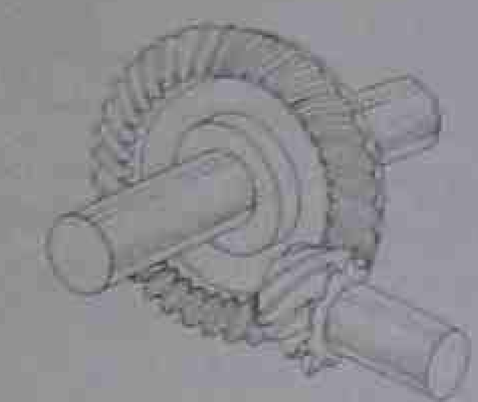


Fig. 7-53 A hypoid gear.

Worm gears

Worm gears are designed to transmit power betwe two non-intersecting shafts at right angles, as sho in Fig. 7-54, and are used when high ratio spe reduction is required. The worm may be cut with o or more threads and must be of the same hand as th wheel.

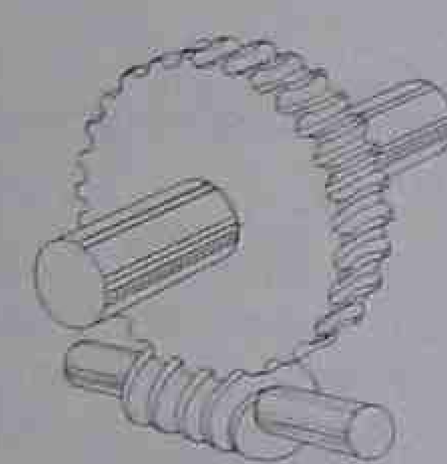


Fig. 7-54 Worm gear.

Whatever type of gear is employed, th arrangement may involve one or more pairs of gea depending on the degree of speed reductio required. (Fig. 7-55)

Most gear drives are mounted in fully enclo casings but large ring gears may be installed as op gears with a suitable guard arrangement.

Gears are generally made from steel or cast ir and are surface hardened in order to increase th wear resistance.

MAINTENANCE PRACTICES

As with other methods of power transmission, th key to satisfactory operation of gear drives is g alignment, proper lubrication and the exclusion dirt and other contaminants.

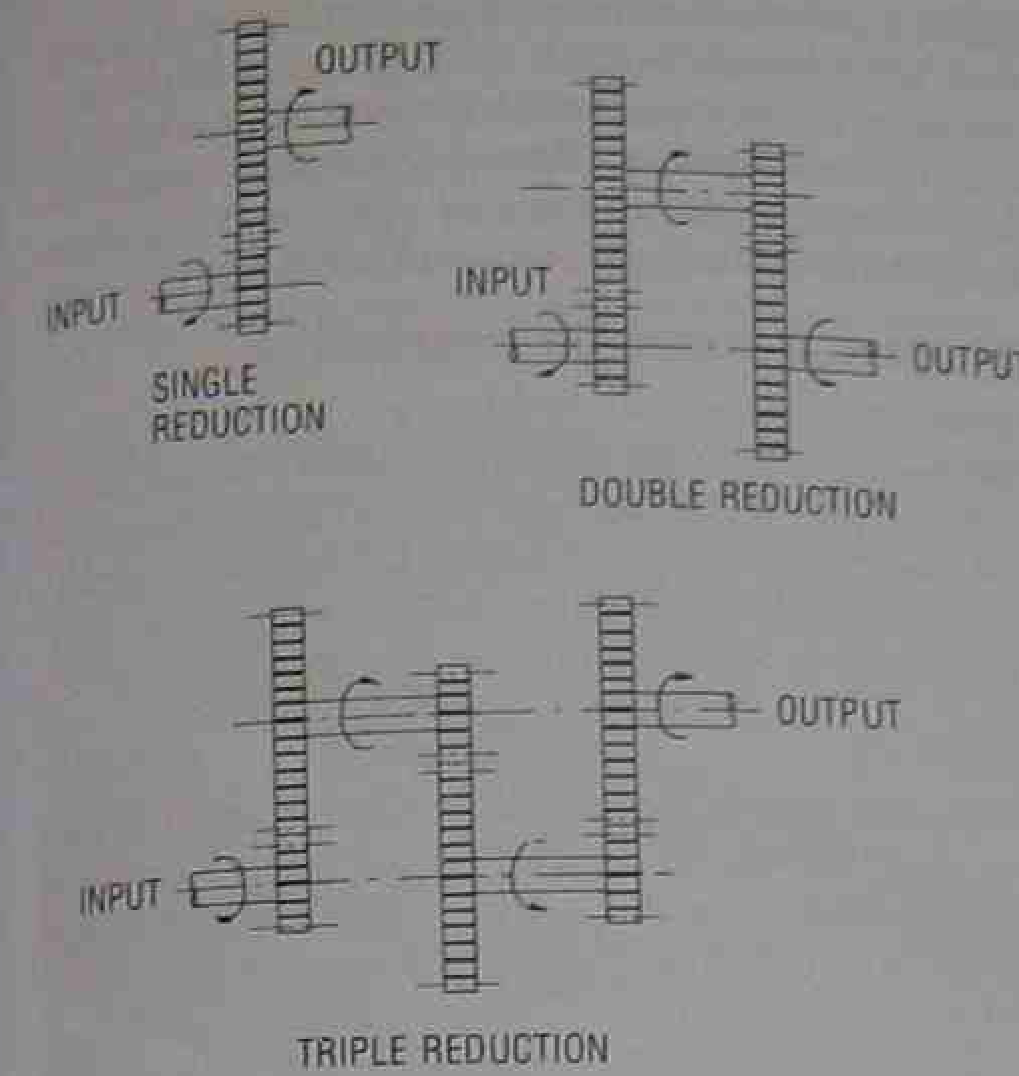


Fig. 7-55 The number of pairs of gears depends on the degree of speed reduction required.

Alignment

In most gear drives the alignment is determined by the machining of the casing and the bearing housings and under normal conditions the gears should be automatically aligned. Care should be taken when mounting a gear box to ensure that no distortion of the casing occurs when the mounting bolts are pulled down. (See Chapter 3.) Excessive wear and run-out in the bearings will result in run-out and wear of gear teeth and should be rectified as soon as possible.

When gears are open, or installed in such a manner that adjustment of the relative positions is possible, either the centre distance between the gears or the backlash between the teeth can be measured. The centre distance between two gears in proper mesh should be equal to half the sum of the pitch circle diameters of the gears. (Fig. 7-56)

If it is more convenient to measure the backlash this can be done using feeler gauges. Care must be taken, whenever the relative position of gears is adjusted, to make sure that the shafts remain parallel so that the gears run true. Parallel alignment of gear tooth faces can be checked by applying marking blue to the pinion teeth and then turning the gear wheel over by hand. The contact pattern on the gear wheel teeth should be even across the face.

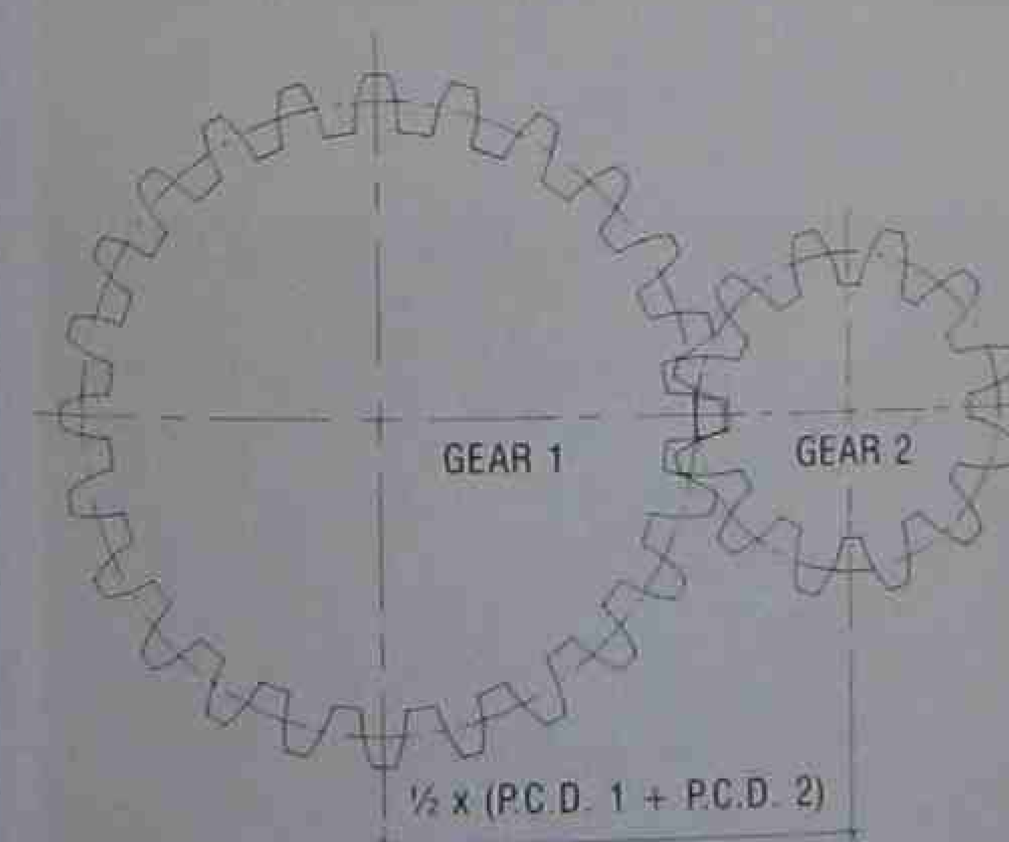


Fig. 7-56 Centre distance determines correct meshing.

Lubrication

Most gear boxes contain a reservoir of lubricating oil in which the lower halves of the gears are submerged. As the gears turn they pick up lubricant which protects the teeth during contact. If maximum gear life is to be achieved the correct lubricant must be used and the correct operating level must be maintained in the sump or reservoir.

A gear box should be checked from time to time for leaks and these should be corrected as soon as possible.

The products of gear tooth wear will collect in the oil reservoir along with any other contaminants which enter the gear box casing. It is therefore necessary to change the lubricant at regular periods as recommended by the manufacturer. This is particularly important during the run-in phase of the machine when the rate of wear-debris production tends to be relatively high.

Gear boxes that are pressure lubricated are normally fitted with a filter and this must be changed or cleaned periodically.

Open gears are usually lubricated by a sump in which the bull gear runs. If the atmosphere is dusty then a large build-up may develop on the gears and it must be cleaned off from time to time.

FAILURE PATTERNS

Like all machine elements which involve relative motion between lubricated components, there are a number of ways in which gear failures occur.

Operating symptoms

The symptoms of gear malfunction found during operation are relatively few in number and easily detectable.

Noise

Even when gears are in good condition they produce a significant amount of noise. This is because of the continuous impact of the gear teeth as they mesh with each other and it will vary with speed and torque transmitted. Every gear combination and gear box has its own distinctive running sound when it is operating satisfactorily and familiarity with that sound will assist the maintenance technician in detecting deterioration in the running condition. Gears in good condition should produce a constant hum with a relatively smooth tone. Once the gears begin to deteriorate, or some malfunction in their operation develops, this noise will change. The sound may become much rougher which could indicate that the gears are not properly in mesh or that they are out of alignment. Misalignment may also cause a rhythmic or pulsating sound to develop. When the surface condition of gear teeth begins to deteriorate, due either to wear or other factors, the sound of the gears tends to increase in pitch and develops into a whine.

Whatever the nature of the sound, whether it be a growl or a whine, the more it increases the more it is an indication of malfunction of the drive assembly. It is not possible to be precise about the nature of the sound that may emanate from a particular problem and the above description can only be considered as a general guide. Increase in noise level, however, can always be treated as positive evidence of a change in the condition of a gear drive.

Vibration

The nature of gears makes it inevitable that their operation is accompanied by a certain amount of vibration. As with noise levels, this will vary according to the type of gears and the speed and load transmitted. An increase in vibration levels will occur when condition deteriorates or a malfunction develops. The most likely causes of an increase in vibration levels are shaft misalignment and teeth running out of mesh. This may occur because of faulty assembly or deterioration in the condition of the shaft support bearings. Wear and deterioration of tooth surface condition, unless they become excessive, are less likely to cause an increase in vibration levels.

Overheating

Generally speaking, overheating of a gear drive is likely to be due to overload or inadequate lubrication and is more likely to occur with enclosed gear boxes than open gears. Serious misalignment or running out of mesh may also cause an eventual increase in running temperature. These conditions should be detected from the change in noise and vibration levels before this becomes significant.

Symptoms found on inspection

Once the external evidence of gear drive malfunction becomes significant then the unit must be shut down and examined so that further evidence can be gathered and cause of failure determined. The following patterns of failure are those most commonly encountered.

Wear

There are various ways in which gear tooth surfaces can wear. In the early stages of gear life, surface irregularities often cause pitting along the pitch line which later disappears when the gears wear in.

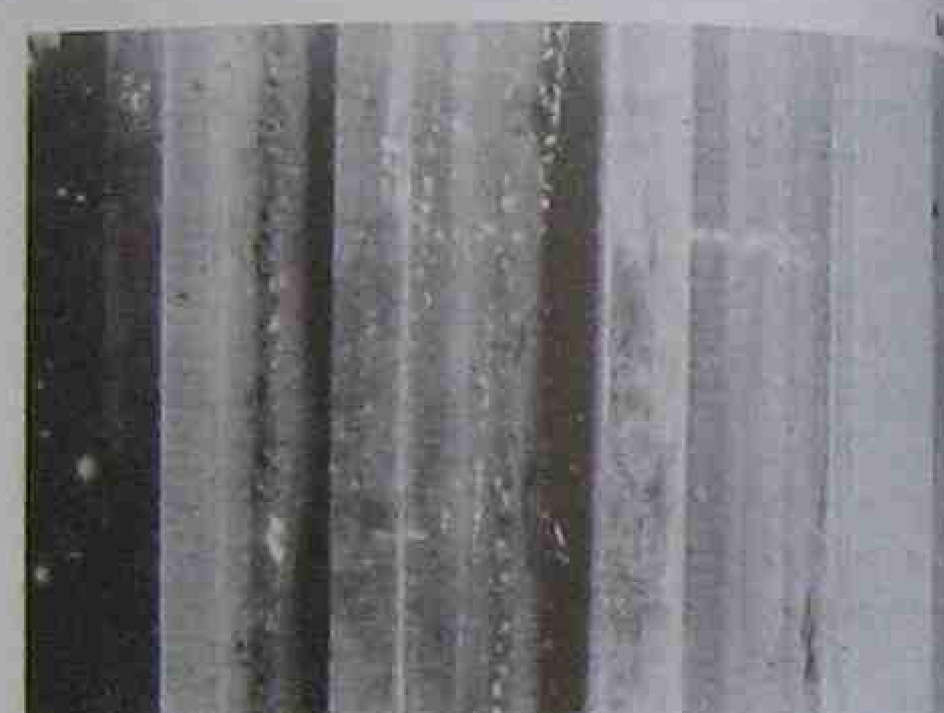


Fig. 7-57 Pitch line pitting.

Fig. 7-58 Normal wear under light to moderate loads.

Normal wear occurs because of metal-to-metal contact between mating teeth and under light to moderate loads will appear as shown in Fig. 7-58.

At higher loads and speeds the adhesion and welding of surfaces that takes place, due to the failure of the lubricant, becomes more extreme and the wear pattern that develops is known as scoring.

Caterpillar

Fig. 7-59 Scoring.

Caterpillar

Fig. 7-60 Abrasive wear patterns.

The most extreme form of this failure mode is known as galling.

When foreign particles such as dirt and grit are present abrasive wear patterns, similar in appearance to scoring, may develop.

The wear patterns that develop due to misalignment and running out of mesh will have quite specific characteristics regardless of the nature of the type of wear. Misalignment will cause an uneven wear pattern to occur across the tooth face as shown in the example in Fig. 7-61.

Running out of mesh will cause either undercutting of tooth faces due to interference (Fig. 7-62) or a shift in the location of the wear pattern towards the tips of the teeth.

Mobil

Fig. 7-61 Wear pattern from misalignment.

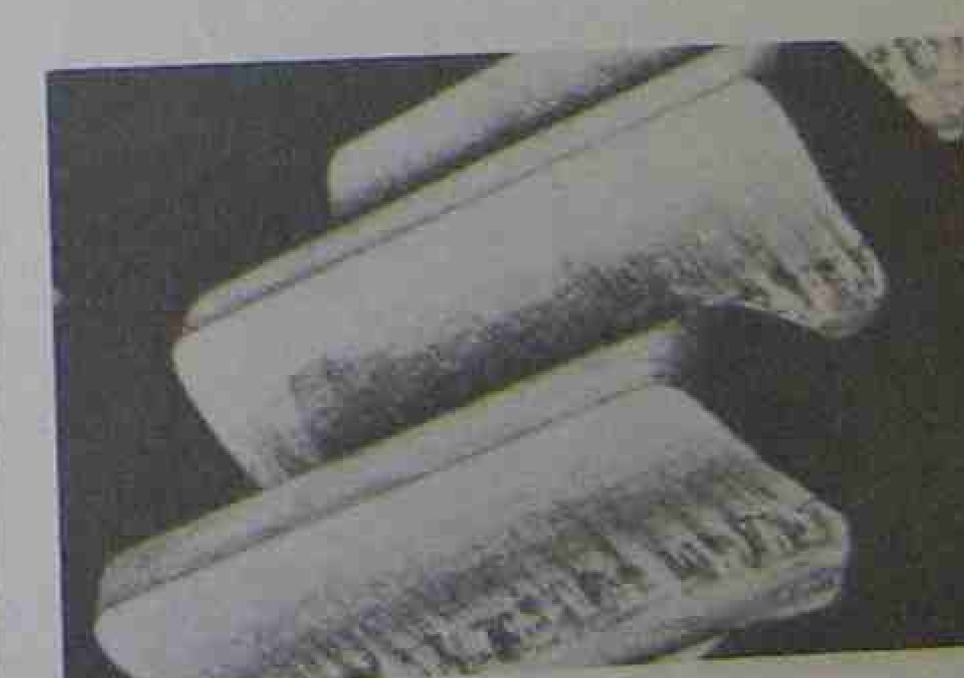


Fig. 7-62 Undercutting of gear teeth.

It is also possible for corrosive wear to occur due to chemical attack from either contaminated lubricant or an additive. This will be evident by etching, not only of gear tooth surfaces, as shown in Fig. 7-63, but also of other gear surfaces.

Caterpillar

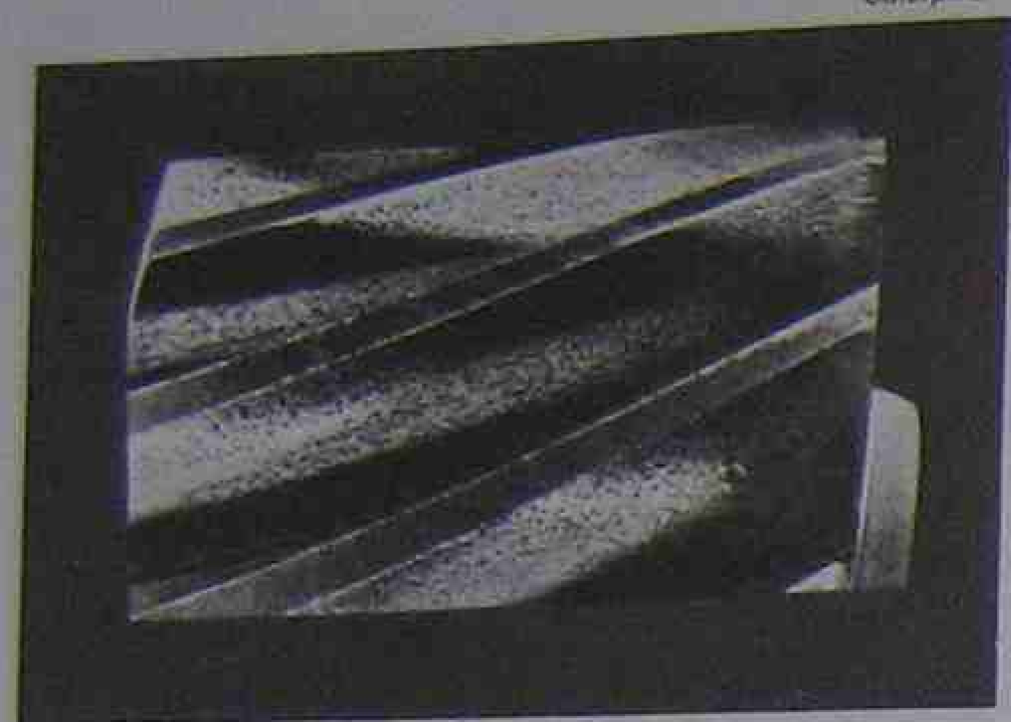


Fig. 7-63 Etching of gear tooth surfaces due to corrosive wear.

Fatigue

The mechanism of fatigue failure in gears follows the same patterns as for other machine elements. Cyclical stressing due to the intermittent contact of gear teeth, especially at high loads, causes sub-surface cracking which then develops into pitting and spalling as shown in Fig. 7-64.

Mobil

Fig. 7-64 Pitting and spalling caused by cyclical stress.

Plastic flow

Deformation of gear tooth surfaces due to the eff of heavy loading is referred to as plastic flow or c flow. This is more likely to occur with soft, duc gears but can also occur with case-hardened ge The surface metal of the gear teeth flows under l and the effect is often described as rippling, ridg rolling or peening.

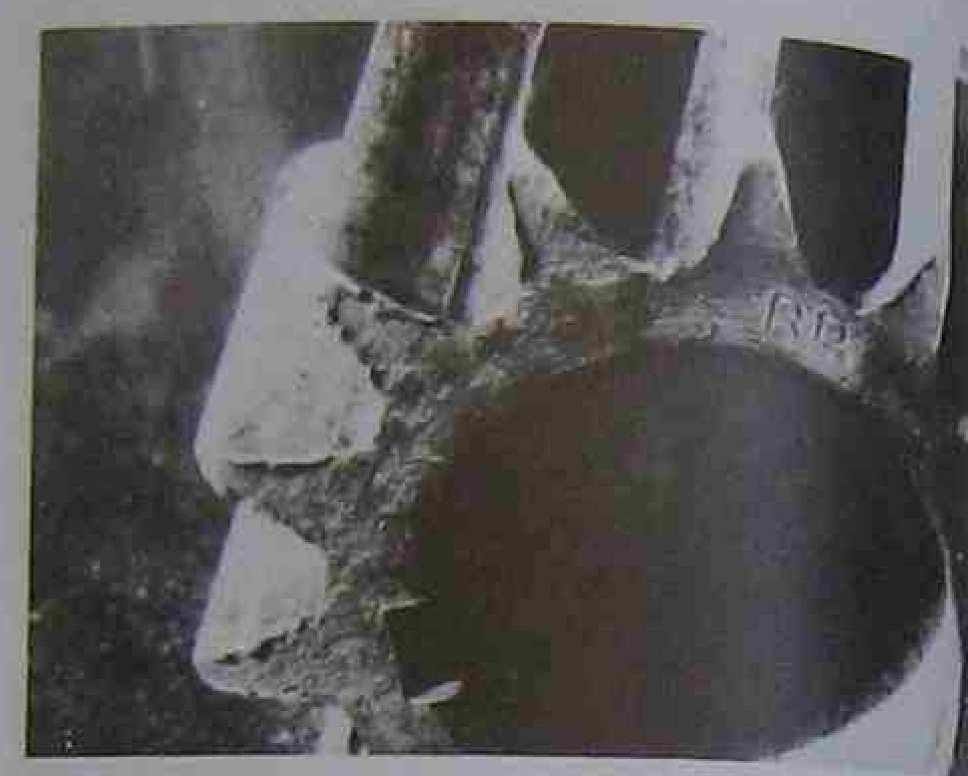


Fig. 7-65 Examples of plastic flow due to peening of tooth edge

Tooth breakage

If regular inspection is carried out then complete partial breakage of gear teeth should rarely occ When it does happen there will usually have be some weakening of the teeth due to a condition th has been present for some time. Once a gear break usually becomes inoperable. The two princip causes are fatigue and heavy impact loads. Fatig failure is often associated with the presence o

Fig. 7-66 Example of gear tooth breakage.

stress raiser such as a quenching crack or some surface defect such as a notch or a tool mark.

Causes

Inadequate lubrication

The operation of gears is vitally dependent on an adequate supply of the correct lubricant and if this is available gears should give almost unlimited life. If the wrong lubricant is used or if lubricant is allowed to deteriorate then the gears will begin to wear. It is also important that lubricant replacement is carried out according to the manufacturer's recommendation. Additives should be carefully selected to suit the particular operating conditions, and lubricant supply systems, including filters, should be properly maintained.

Misalignment

There are two different ways in which gears may be misaligned. They may be **out of parallel** or **out of plane**.

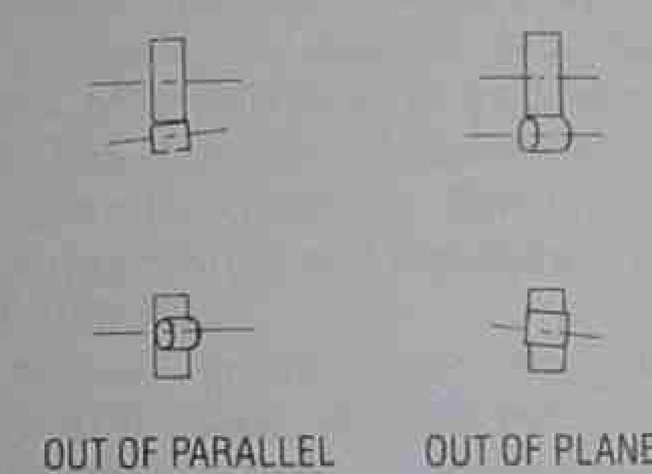


Fig. 7-67 Misalignment of gears.

The uneven wear patterns referred to in the previous section are produced by misalignment and if the condition is allowed to continue then tooth breakage may eventually occur.

Out of mesh

If the gears are set up so that the pitch circles are not touching each other then they can be considered to be out of mesh.

If the centre distance between the gears is too small and there is insufficient backlash then interference occurs and the tip of one tooth tends to dig into the root of the mating tooth and produce excessive wear. This will cause rapid deterioration of gear condition and may result in excessive noise and vibration. If gears are set too far apart the backlash will be excessive and wear will occur close to the tips

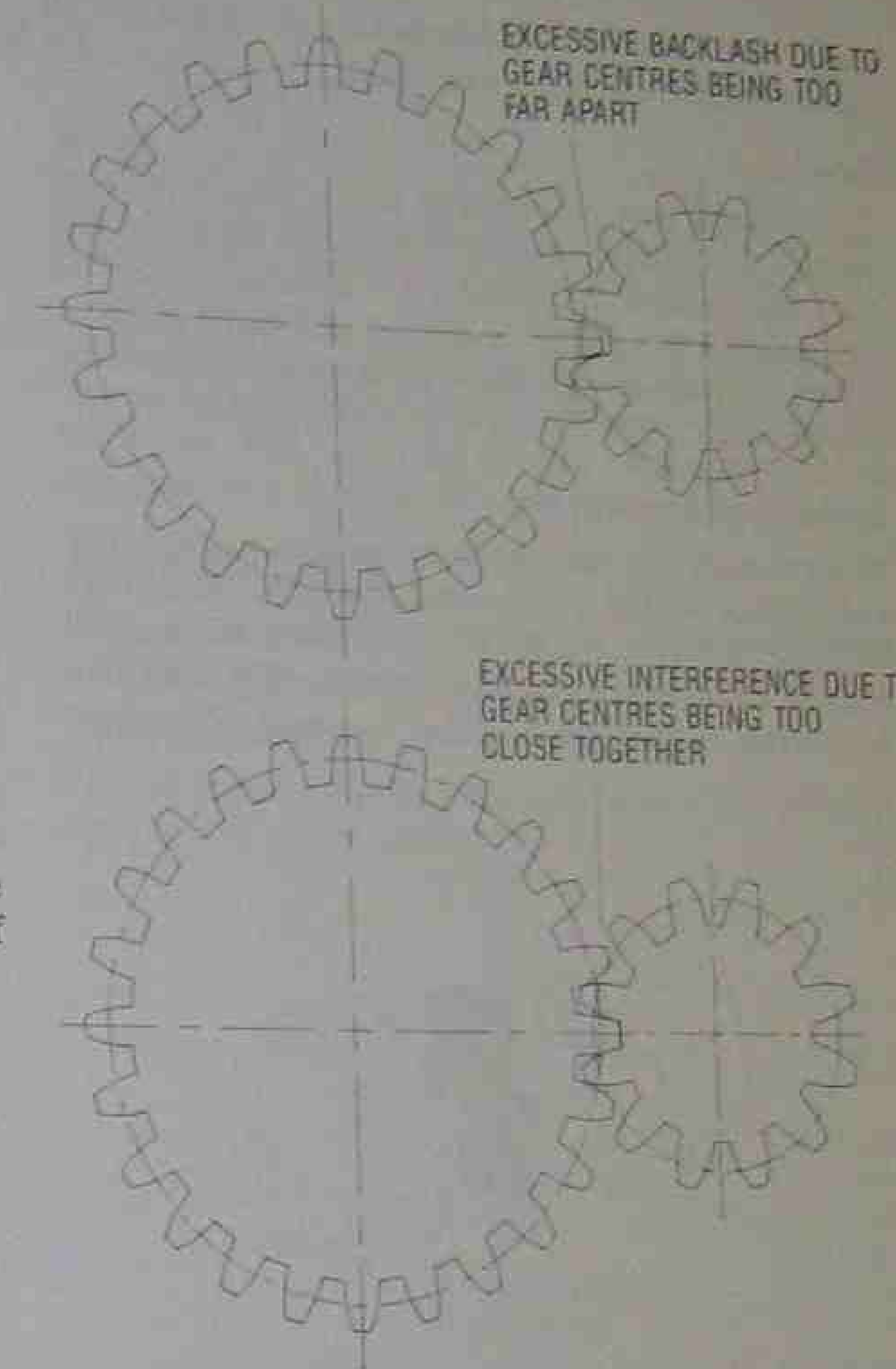


Fig. 7-68 Gears out of mesh.

of the teeth which may even break from the impact. Noise and vibration will also increase due to the large amount of backlash.

Overload

Speeds and loads which exceed design limits and high impact loads will accelerate wear processes and lead to the likelihood of premature failure. Heavy spalling or galling and tooth breakages are the usual consequences of overload conditions. The design limits of the drive should be checked and the unit operated according to the manufacturer's instructions.

Contamination

The contamination of gear lubricant due to the presence of dirt, dust or other abrasives will increase wear rates and cause scratching of tooth surfaces. Particles of wear metal or chips of broken teeth will also cause significant damage. Careful attention should be paid to the condition of seals and filters and the regular replacement of lubricant if contamination is to be avoided.

Moisture

The presence of moisture in a gear box may cause rusting to develop. In order to avoid moisture due to condensation build-up, special breather arrangements may be required whereas ingress of moisture from other sources should be prevented by the oil seals.

Lubricant breakdown

If lubricants are not replaced regularly they may deteriorate to the point where harmful acids may form. If corrosion is detected then lubricant analysis may be required to establish whether any change in properties of the lubricant has occurred. The effect of lubricant additives on particular metals sh also be considered and the lubricant manufac consulted for advice.

Summary of the common symptoms and causes of gear drive fa

Symptoms		Causes
Operating	*Inspection*	
Noise	Wear: adhesive	Inadequate lubricatio
Vibration	scoring	Misalignment
Overheating	abrasive	Out of mesh
	uneven	Overload
	corrosive	Contamination
	Fatigue/spalling	Moisture
	Plastic flow	Lubricant breakdown
	Tooth breakage	

7.4 SHAFT COUPLINGS

Couplings are the devices used to connect two shafts with a common axis of rotation so that one can drive the other. Unlike the other transmission elements discussed in this chapter, couplings do not change the characteristics of the motion they transmit. Speed, torque and direction of rotation all remain the same from driver to driven.

PRINCIPLES OF OPERATION

There are two major categories of shaft coupling. When two shafts are truly aligned the coupling may be **rigid** or **solid**, in which case one shaft merely becomes a direct extension of the other. If some misalignment of the shafts is likely, the coupling must contain a mechanism able to absorb that misalignment and still transmit motion smoothly from one shaft to another. Couplings which accomplish this are known as **flexible** couplings and they operate according to two basic principles.

The simplest types of flexible coupling employ a flexible element to absorb the misalignment. Elastomeric materials such as rubber are normally used for the flexible element although metal elements are also common. This type of coupling requires no lubrication and is generally referred to as **material-flexing**.

The second group may be described as **mechanical-flexing** and involves mechanical components that slide or otherwise move relative to each other to provide the necessary flexibility. This type of coupling requires lubrication to minimise wear.

TYPES

There are various commonly-used couplings which meet the needs of the majority of industrial applications.

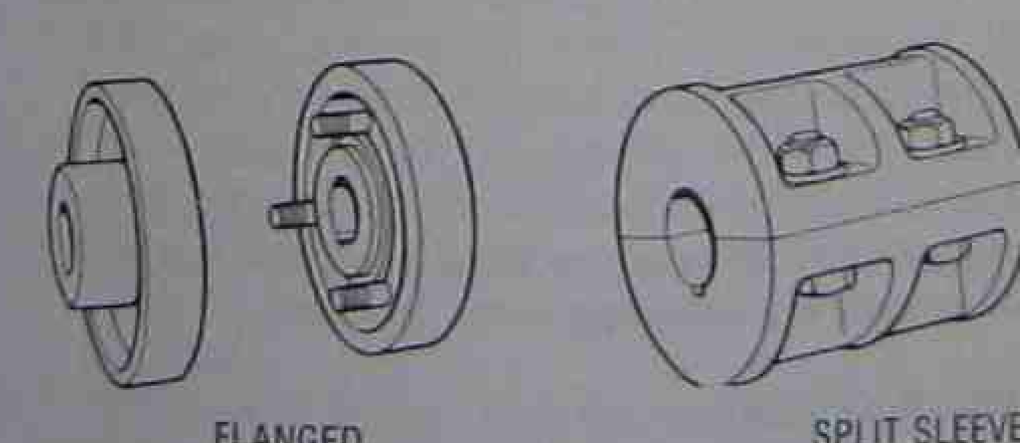


Fig. 7-69 Two common types of rigid coupling.

Rigid

The two most common rigid couplings are the flanged and sleeved types shown in Fig. 7-69.

Rigid couplings cannot normally be used where shafts need to be aligned on installation, as in the case of independent driver and driven units, but are used for line shafting where some means of disconnection is required.

Material-flexing

The most common types of material-flexing couplings are:

Jaw couplings

These utilise a flexible element which fits between two sets of metal jaws.

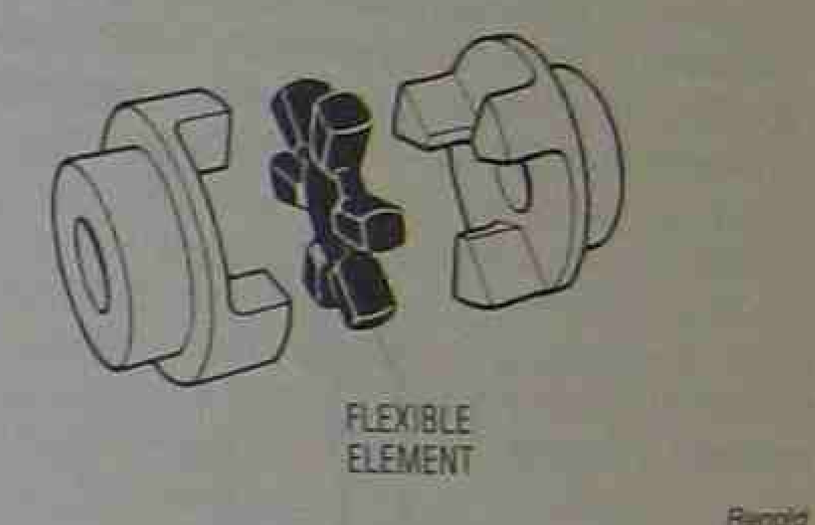


Fig. 7-70 Jaw coupling.

Pin couplings

Two metal flanges are connected by steel studs upon which flexible buffers, usually made of rubber, are mounted. They are sometimes referred to as crown pin or cone ring couplings and a typical example is shown in Fig. 7-71.

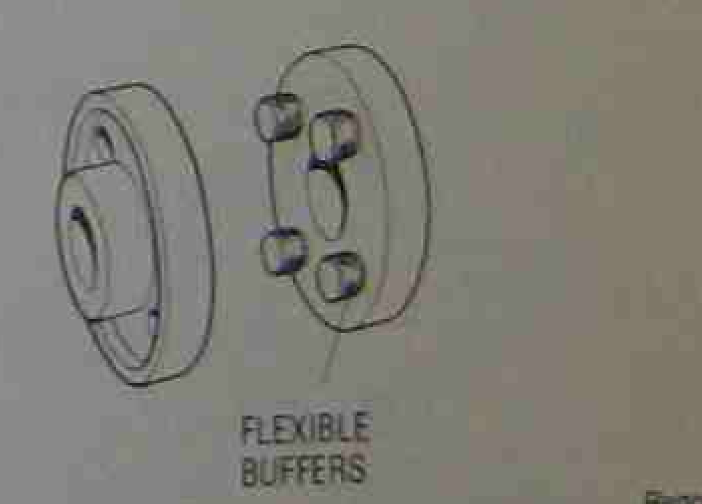
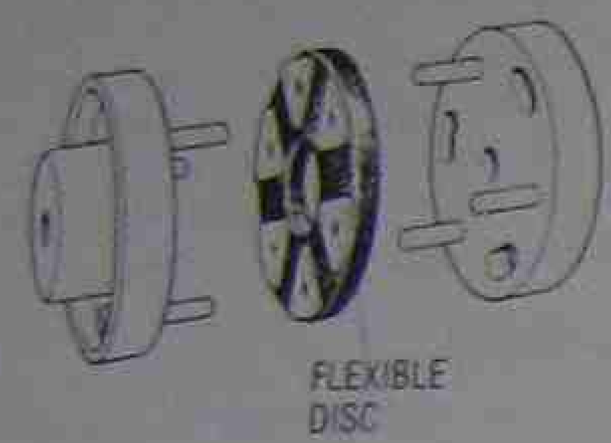


Fig. 7-71 Typical example of a pin coupling.

Disc couplings

In this type of coupling the flanged half bodies are connected with a flexible disc which is usually made of an oil resistant composite material.



Renold

Fig. 7-72 A simple disc coupling.

A more sophisticated type of disc coupling employs a laminated metal disc ring as the flexible element. In order to accommodate both angular and parallel misalignment, two disc elements are normally incorporated within the coupling.

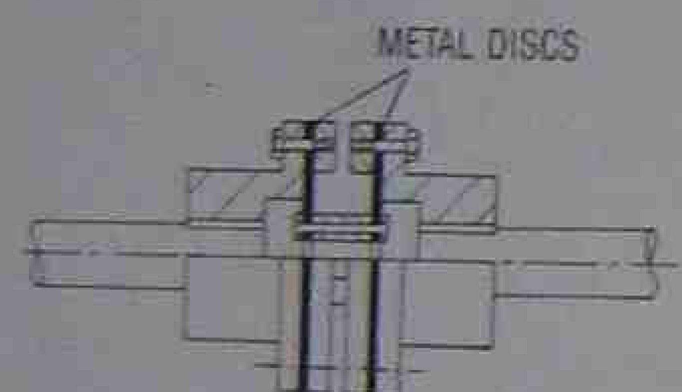


Fig. 7-73 Disc coupling with two disc elements.

Elastomeric couplings

Couplings that rely for flexibility on the elastomeric properties of a specially designed element are available in various designs. The most common is probably the type shown in Fig. 7-74.

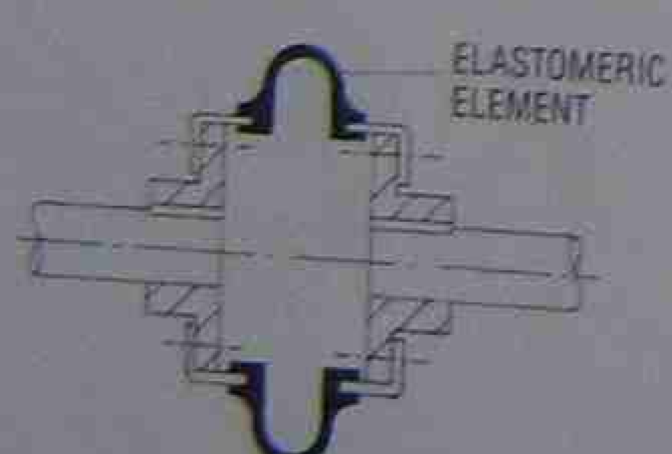


Fig. 7-74 Common type of elastomeric coupling.

Mechanical-flexing

The most common types of mechanical-fl[exing] couplings are as follows:

Gear couplings

The coupling hubs are machined with external [...] teeth which mesh with an internal set of te[eth] machined into the flanged cover as shown in F[ig.] 7-75. Relative movement between the mating se[ts of] gear teeth provides the necessary flexib[ility]. Lubrication is essential and the covers are fitted w[ith] seals to contain the lubricant which is usually gre[ase].

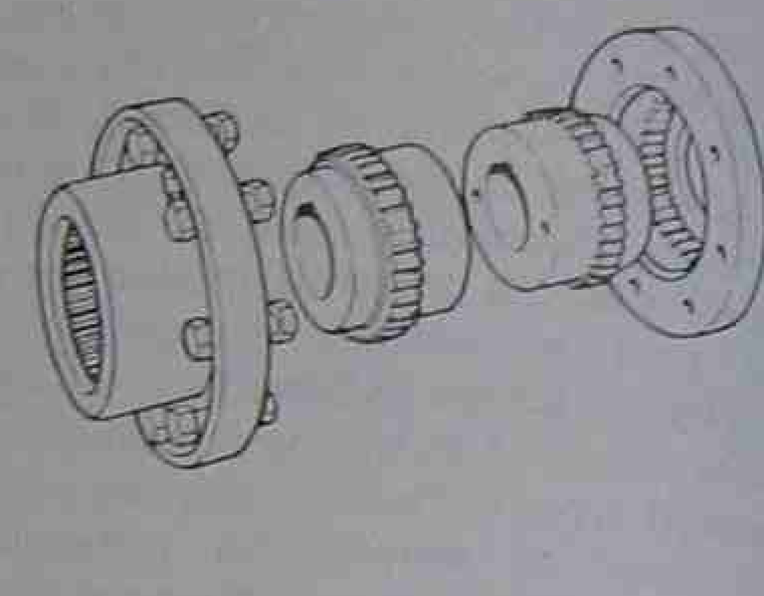


Fig. 7-75 A gear coupling.

Chain couplings

The coupling hubs are machined with sprocket te[eth] and the two halves are connected by a length [of] duplex roller chain.

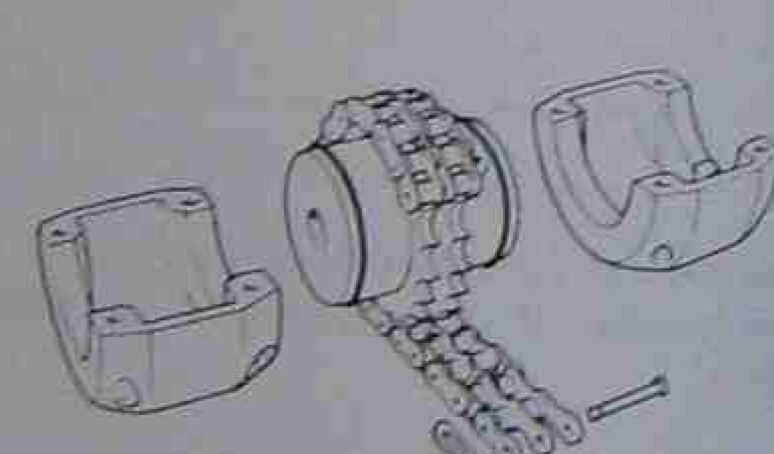
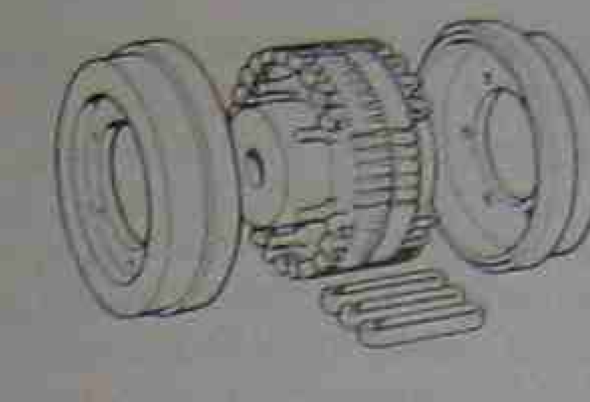


Fig. 7-76 Chain coupling.

An oil-tight cover is provided to contain the gre[ase] and protect the coupling.

Grid couplings

The hubs of grid couplings are machined with ax[ial] grooves or slots into which is fitted a flexible meta[l] grid.



Renold

Fig. 7-77 A grid coupling.

A sealed cover contains grease to lubricate the coupling in a similar way to a chain coupling.

Spacer couplings

These are not a further type of coupling but represent a particular variation of the standard designs that enables length to be added to the coupling as shown in the example in Fig. 7-78.

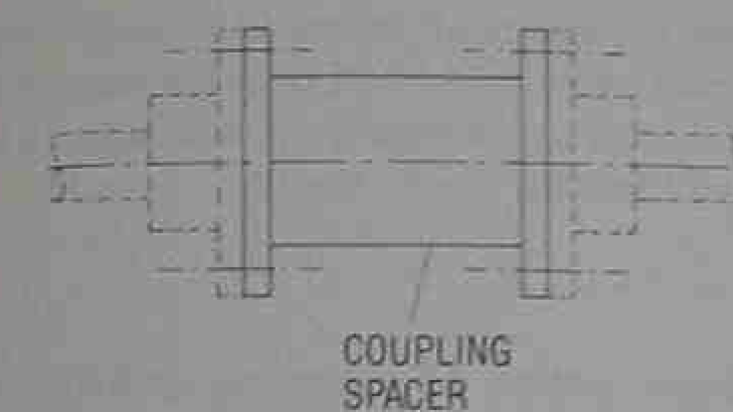


Fig. 7-78 Use of a spacer coupling to add length.

The principal advantage of a spacer coupling is that it allows the coupling hubs to be removed from the shafts without disturbing the position of either machine. Most large machines that cannot be easily moved are fitted with spacer couplings.

MAINTENANCE PRACTICES

The following general considerations are relevant to the maintenance of couplings.

- The most important factor in ensuring that satisfactory coupling performance is achieved is the alignment of the shafts. Although flexible couplings are designed to cope with misalignment there is always a limit to the amount that can be handled without causing rapid wear of the coupling.
- Before a coupling can be installed it must be dissassembled and it is wise to record the order of components so that it can be reassembled in the same order.
- It is vital that the coupling hubs are mounted securely on the shafts so that no relative movement between the hub and shaft can occur. The most common methods of securing the hubs are to use a key and keyway, an interference fit or a tapered bush arrangement.
- Coupling hubs are often supplied with only a pilot hole and must be machined out to fit the shaft. It is vital that the coupling is bored true and concentric otherwise the coupling may be thrown out of balance.
- Before fitting the coupling hubs, the shaft and the bore of the hubs should be inspected to ensure that they are free of nicks, burrs and other damage.
- A rotating coupling can be a safety hazard and should never be operated without a suitable safety guard. The guard should be robust and designed so that personnel cannot come into contact with the coupling during operation.

Lubrication

The lubrication of mechanical-flexing couplings is important if rapid wear is to be avoided. Coupling covers are provided with grease points and the manufacturer's lubrication recommendations should be followed. If machines operate in a dirty or dusty environment it is advisable to strip couplings down from time to time and clean them with a degreasing fluid. New grease should then be applied before restarting the machine.

Assembly

The major consideration when assembling a coupling is to ensure that the two halves are securely fixed to the shafts and that the necessary clearance gap exists between mating faces. Because the procedure varies for different types of couplings it is wise to consult the manufacturer's recommendations. The use of tapered bushes of the type shown in Fig. 7-79 allows

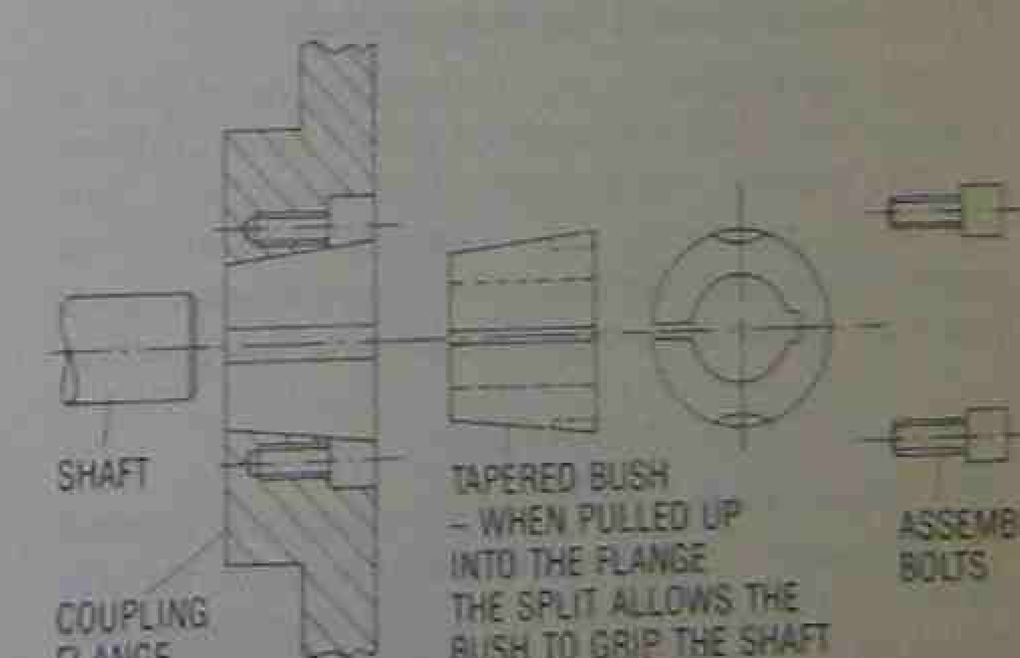


Fig. 7-79 A tapered bush arrangement.

a standard machined coupling to be fitted to a range of shaft sizes by selection of the appropriate bushing.

Alignment

The factor that most significantly affects the performance of shaft couplings is shaft alignment. Because of its importance this topic has been treated separately and is the subject of Chapter 8.

FAILURE PATTERNS

A properly assembled, aligned and lubricated coupling should give virtually unlimited service. Failures occur from time to time, however, and the maintenance technician should be aware of the likely symptoms and common causes.

Operating symptoms

The symptoms of coupling malfunction are relatively few and obvious.

Noise

Under normal operation a coupling should be noise free. If coupling hubs are not tight and wear begins to develop then the coupling may develop a clicking or a rattling sound.

Vibration

Once a coupling begins to vibrate, not only its condition but the condition of other machine elements, such as bearings and seals, will begin to deteriorate. Loose hubs, excessive wear and misalignment are all potential causes of coupling vibration. It is also possible that vibration may be transmitted from other parts of the machine and this too can cause excessive wear or other damage to a coupling.

Symptoms found on inspection

If the performance of a coupling indicates a malfunction then the machine should be shut down at the earliest opportunity and inspected. The following symptoms are common evidence of malfunction. Because there are many different types of couplings the treatment given here is only general.

Wear

The nature of wear shown by a coupling will depend on its construction but excessive wear is usually evidence that the coupling has not been operating satisfactorily. Material-flexing couplings will show evidence of wear to the flexible element and if this is excessive there may even be damage to o[…] components such as pins and flanges. Mecha[…] flexing couplings will show evidence of wear to [...] mechanical elements which accommodate [...] relative movement. Hence the teeth of [...] couplings must be inspected as must the chain [...] sprocket teeth of chain couplings. Wear of [...] couplings is likely to show up in the axial slots [...] hold the grid element. Wear may also be observe[d] [...] the bore of the coupling, and to the shaft, if [...] coupling hub has not been securely fixed.

External damage

The coupling should run well clear of any oth[er] machine parts and external damage may indica[te] interference with the guard or other machin[e] elements. This may appear as extreme wear [...] scoring on the outside of the coupling.

Material damage

The flexible elements of material-flexing couplin[gs] are often made of rubber and may be susceptible [to] attack by oil and other liquids. Swelling and loss [of] elasticity will be the likely result.

Causes

The common causes of coupling failure can b[e] summarised as follows.

Misalignment

As mentioned above if misalignment is too great the[n] the coupling will wear rapidly.

Inadequate lubrication

If mechanical-flexing couplings are not adequately lubricated the life of the coupling will be significantly reduced.

Improper assembly

Failure to mount the coupling hubs securely on the shafts and failure to properly install covers and seals will lead to early malfunction.

Interference

Coupling guards should be designed to giv[e] adequate clearance so that there is no danger [of] interference.

Contamination

Excessive build-up of dirt around the coupling may cause it to bind and prevent if from moving t[o] accommodate whatever misalignment may be present.

Overload

If the operating conditions exceed the design limits of the coupling it is most likely that rapid wear and premature failure will result.

Summary of the failure patterns associated with shaft couplings

Symptoms		Causes
Operating	*Inspection*	
Noise Vibration	Wear External damage Material damage	Misalignment Inadequate lubrication Improper assembly Interference Contamination Overload

7.5 CLUTCHES

Clutches are devices that enable two shafts or rotating elements to be connected or disconnected while at rest or in relative motion. They must be capable of transmitting the maximum torque requirement of the drive system, and of disengaging completely and allowing one shaft to rotate independently of the other. Some clutches allow transmission of motion in one direction only.

The best known application of a clutch is in the transmission system of a motor vehicle. There are many other industrial applications in which clutches are critical machine elements, and they play a particularly important role in the operation of automatically controlled machinery.

PRINCIPLES OF OPERATION

There are two aspects to clutch operation: torque transmission and clutch actuation.

Torque transmission

There are three principle ways in which torque and motion are transmitted from one shaft to another: positive engagement, friction and wedging action.

Positive engagement

The simplest and most basic arrangement is one which relies on positive engagement by means of teeth on coupling halves such as the one shown in Fig. 7-80. Engagement is achieved by allowing one of the clutch halves to slide axially along the shaft. This mechanism has the advantage of being unable to slip but is very limited in its ability to allow engagement on the run, although this can be achieved at low speeds with certain tooth designs.

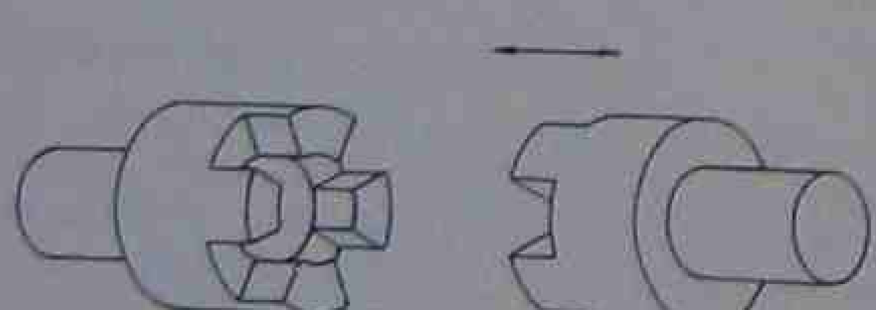


Fig. 7-80 Engagement by teeth.

Friction

Many types of clutch rely on friction to transmit torque. This is the principle used in most motor vehicle clutches. A typical arrangement involves bringing two contacting surfaces together and relying on the friction between them to transmit the torque. Often a special plate covered with material with a high coefficient of friction is interposed between the surfaces to increase the efficiency of the clutch.

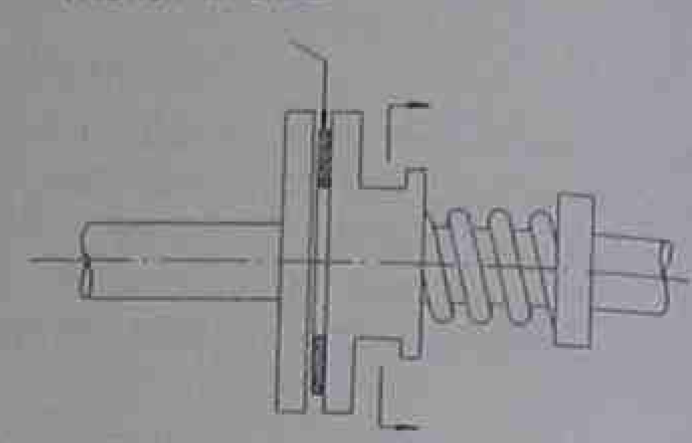


Fig. 7-81 Engagement by friction.

The torque carrying capacity of a friction clutch is directly related to the area of contact between the friction surfaces.

Wedging action

The principle employed in freewheeling and over-running clutches, where motion is required to be transmitted in one direction only, is one that relies on the wedging action of a roller or specially designed element trapped between two races. Rotation of the drive shaft in one direction causes the rollers to wedge and transmit torque to the other shaft, as shown in Fig. 7-82. If the drive shaft rotates in the opposite direction, the driven shaft can either remain stationary or freewheel.

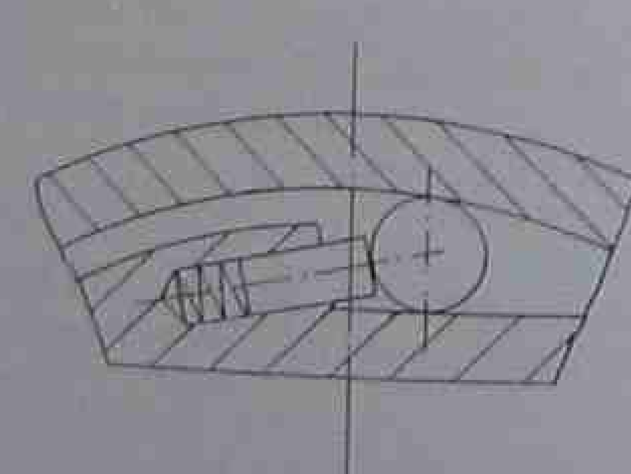


Fig. 7-82 Engagement by rollers.

A variation of this principle involves the use of specially designed wedges called sprags. These are held between concentric races by springs, as shown in Fig. 7-83.

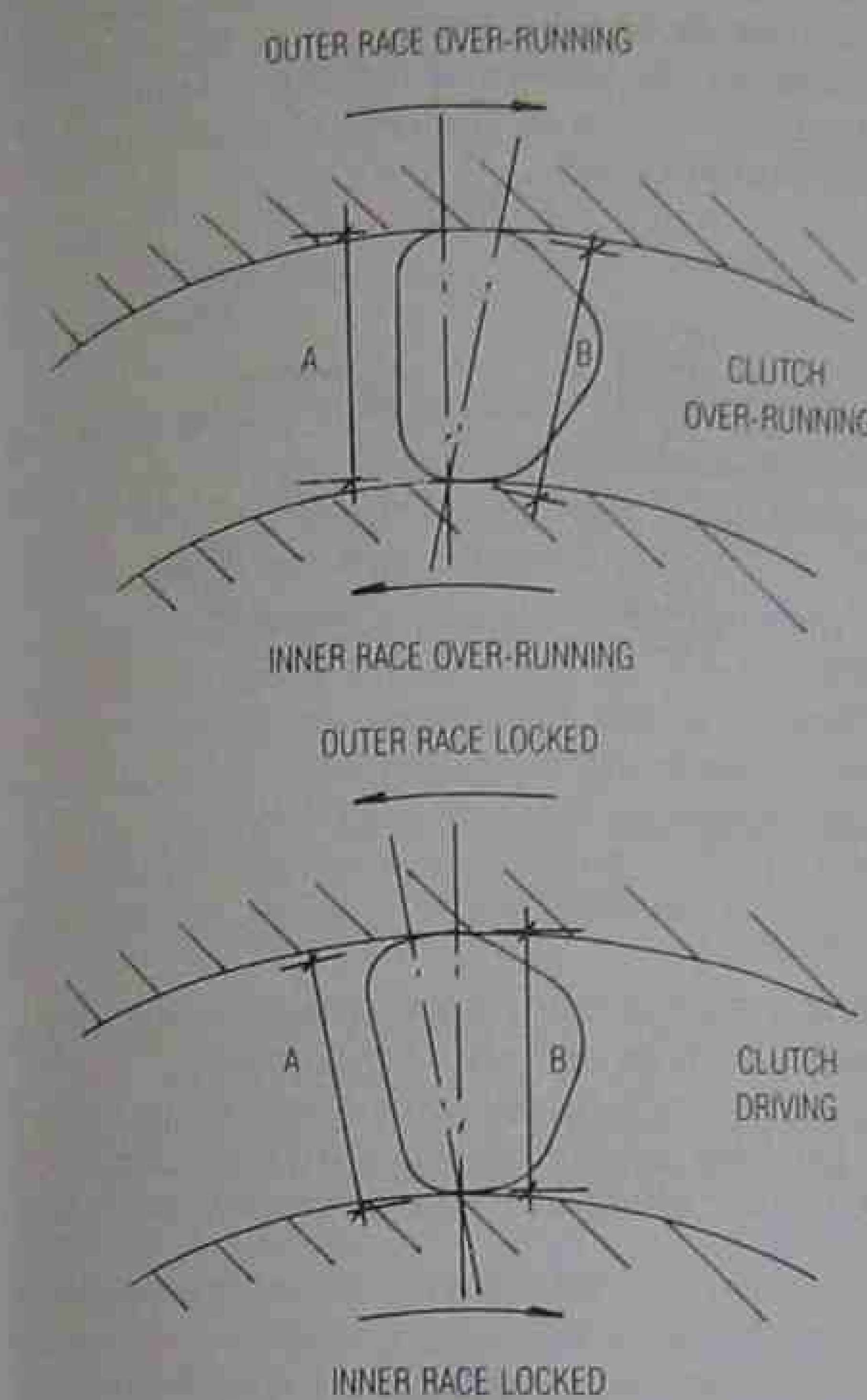


Fig. 7-83 Engagement by sprags.

The sprags are formed with one set of dimensions, 'A', less than the radial gap between the races and the other, 'B', greater than the gap. Rotation of the inner shaft in a clockwise direction will cause the sprag to wedge and drive the outer shaft in the same direction. Similarly, rotation of the outer race in an anticlockwise direction will cause the inner race to rotate. If the inner shaft rotates in an anticlockwise direction, however, the sprag will take up a position such that its shorter dimension will allow it to disengage and the outer race will remain stationary.

Clutches that utilise this principle also allow the driven shaft to 'over-run' the drive shaft by rotating faster and in the same direction.

Clutch actuation

There are four principal ways in which a clutch mechanism can be actuated. They are **mechanical**, **pneumatic**, **electrical** and **hydraulic**. The method used and the design of the system will depend on the application and the conditions under which the system is required to operate.

TYPES AND APPLICATIONS

There are two general industrial applications in which clutches are used: power transmission, and indexing, over-running and backstopping.

Power transmission

The following types of clutches are the ones most commonly used for power transmission.

Dog-tooth

The dog-tooth clutch is a positive displacement type that is very simple in design. One half is operated by a lever and slides on a key or splines. It can be operated only when stationary or moving at very low speeds.

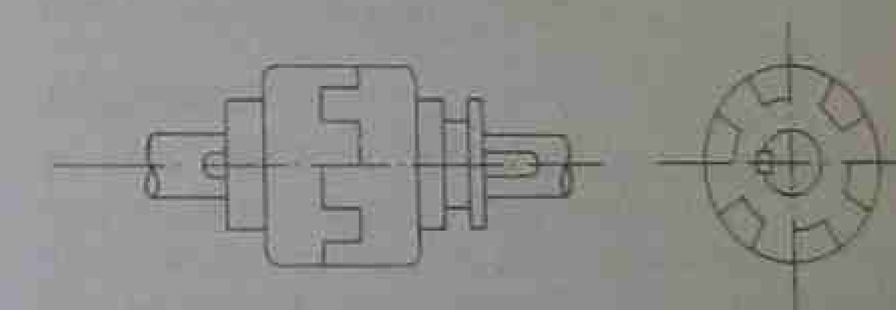


Fig. 7-84 Dog-tooth clutch.

Single-plate friction

This employs a plate with friction material on both sides which is clamped between two steel plates when the clutch is engaged. The friction plate is mounted on splines or a key on the output shaft and can slide axially. The pressure that causes the drive to engage is usually provided by a set of springs, and disengagement is achieved by a lever that acts against the springs and releases the pressure plate.

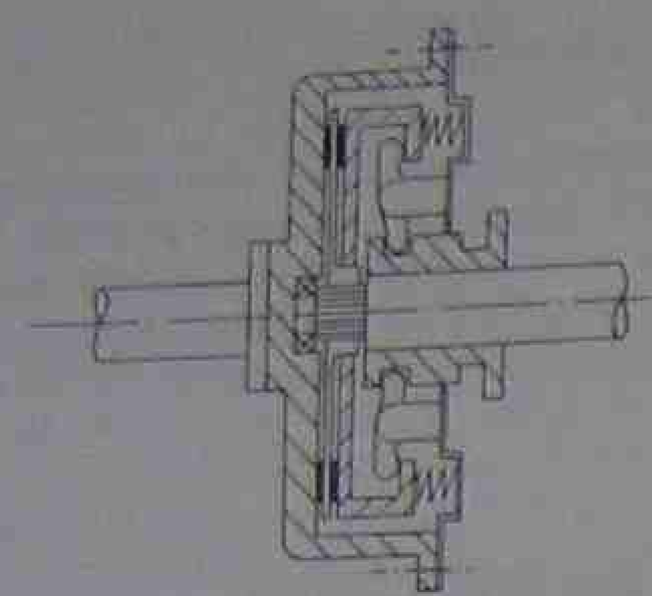


Fig. 7-85 Single-plate friction clutch.

Multi-plate friction

This is a development of the single-plate clutch and has several friction plates (usually five, seven or nine) keyed to one shaft. Interleaved with them are steel pressure plates keyed to the other shaft. As with the single-plate clutch, the pressure is provided by a set of springs or sometimes a single spring.

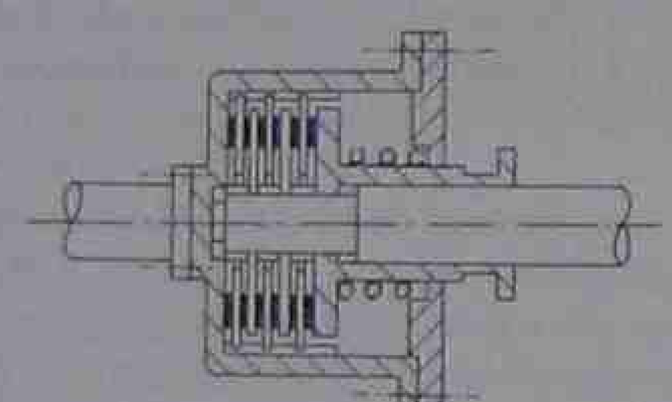


Fig. 7-86 Multi-plate friction clutch.

Cone

A cone clutch has two conical mating surfaces, one of which is usually lined with high-friction material. The torque transmitting capacity of a cone clutch is greater than that of a flat plate clutch of the same diameter because of the increased area of contact.

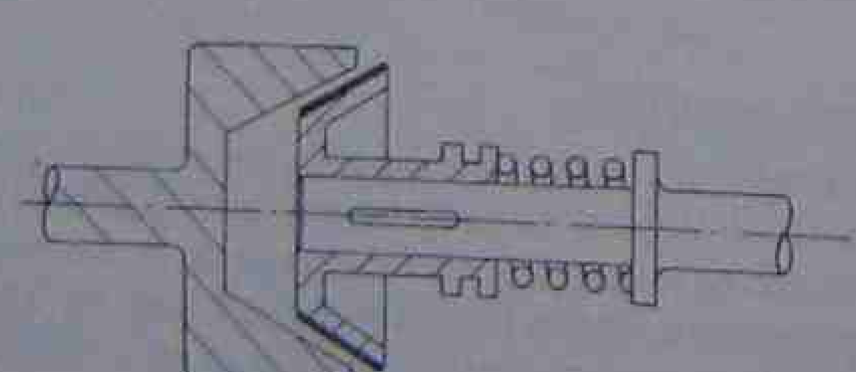


Fig. 7-87 Cone clutch.

Centrifugal

This type is used when engagement of a load has to be achieved at a particular rotational speed. A typical design has spring-loaded weights mounted in radial slots in a member connected to the drive shaft. As the speed of the drive shaft increases, the weights, which are faced with friction material, are thrown out against the surface of a drum mounted on the output shaft. This is the type of clutch commonly used on chain saws.

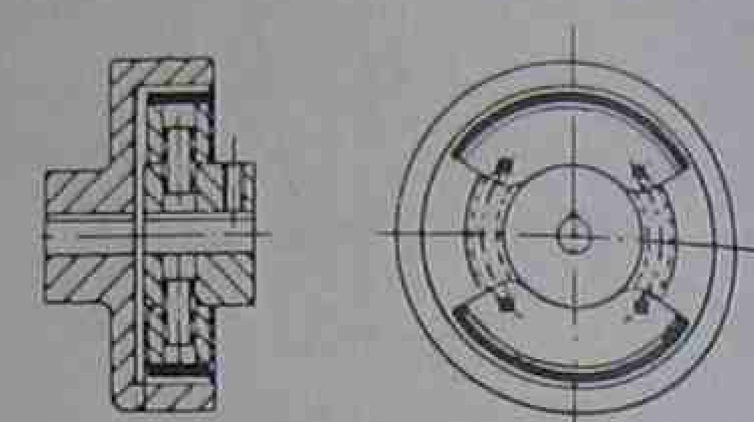


Fig. 7-88 Centrifugal clutch.

Dry fluid

This is a type of centrifugal clutch in which metal particles, such as steel shot, are compacted under the action of the centrifugal force produced by rotation. The particles are contained in a hollow member in which a disc attached to the driven member rotates.

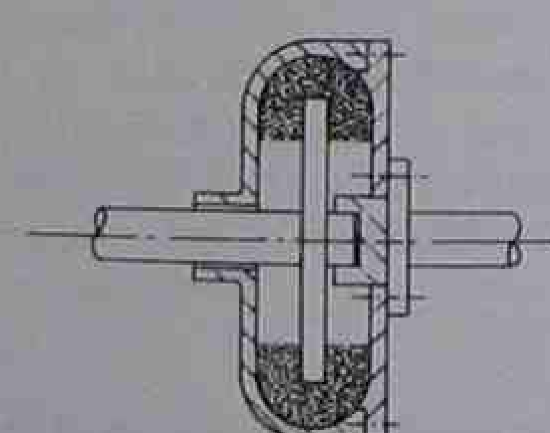


Fig. 7-89 Dry fluid clutch.

Fluid coupling

In a fluid coupling clutch both input and output shafts carry impellers which have radial vanes. The vane spaces are filled with oil which circulates in the vanes when the coupling rotates. The input wheel acts as a pump and the output wheel as a turbine so that power is transmitted. There is always a loss of speed in this type of coupling due to slippage.

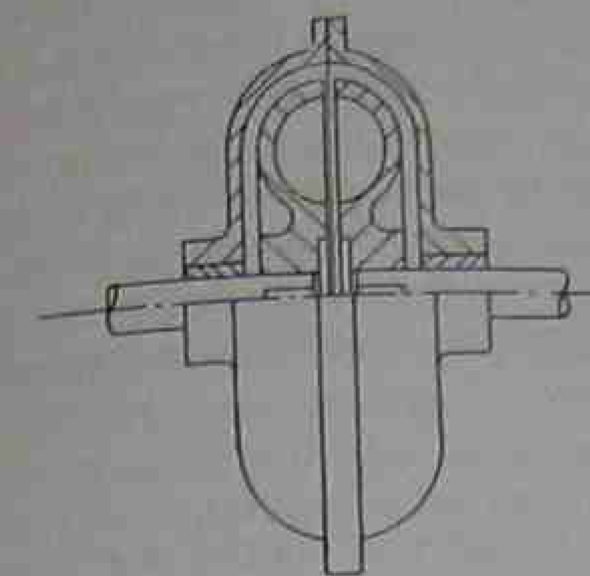


Fig. 7-90 Fluid coupling.

Indexing, over-running and backstopping

There are many industrial applications where a clutch is required to fulfil functions other than, or in addition to, power transmission.

Indexing is the transmission of torque on an intermittent basis either from a continuously rotating shaft or by converting the reciprocating linear motion of a pneumatic or hydraulic cylinder into rotary motion. This function is typically used for indexing conveyors, packaging materials and sheet steel in presses where precise location is required.

Over-running allows the output to rotate faster than the input and is typically used in starter drives for engines or turbines to provide automatic decoupling of starter motor drive when operational speed is reached.

Backstopping is the prevention of undesirable reverse rotation. This is achieved by anchoring one race of the clutch so that the other race is free to rotate in one direction but is immediately locked in reverse. Backstopping is typically used on conveyors, elevators, cranes and pumps.

There are two principal types of clutch that are used in such applications: roller and sprag.

Roller

The freewheeling roller clutch permits a shaft to be driven in one direction only and operates according to the principles previously described.

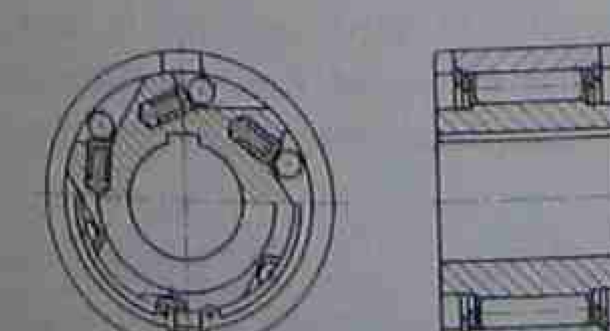


Fig. 7-91 Roller freewheeling clutch.

Sprag

This type, also described above, utilises the wedging action of sprags in an annular space.

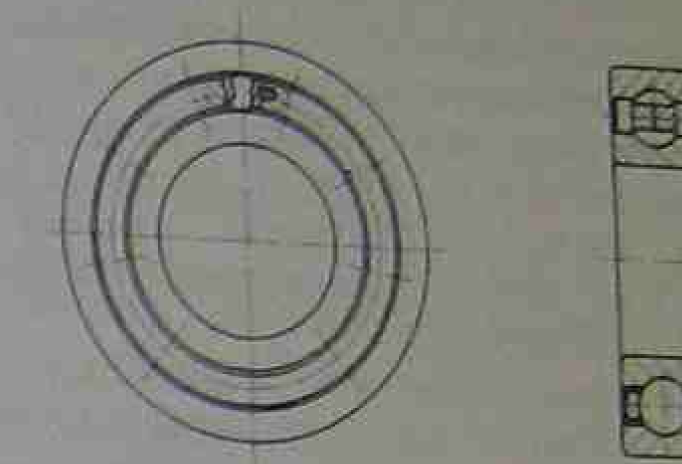


Fig. 7-92 Sprag freewheeling clutch.

Contact free sprag

A variation on the sprag clutch, the contact free sprag is designed such that the sprags disengage totally when the clutch is freewheeling or over-running. The sprags are pivoted so that the action of centrifugal force lifts them clear of the stationary or slower running race. They can be designed to be either internally or externally disengaging.

MAINTENANCE PRACTICES

The maintenance practices associated with clutches are dependent to some extent on the type of clutch concerned. There are a number of general guidelines that should be followed.

Alignment

As with other drive components, the alignment of the two shafts is critical to the operation of a clutch. In the case of coupling type clutches, the requirements are comparable with those for couplings and will normally be specified by the manufacturer. Plate clutches, such as those fitted in vehicles, are also sensitive to misalignment and run out of the flywheel, and bearing housings may need to be checked against manufacturer's recommendations.

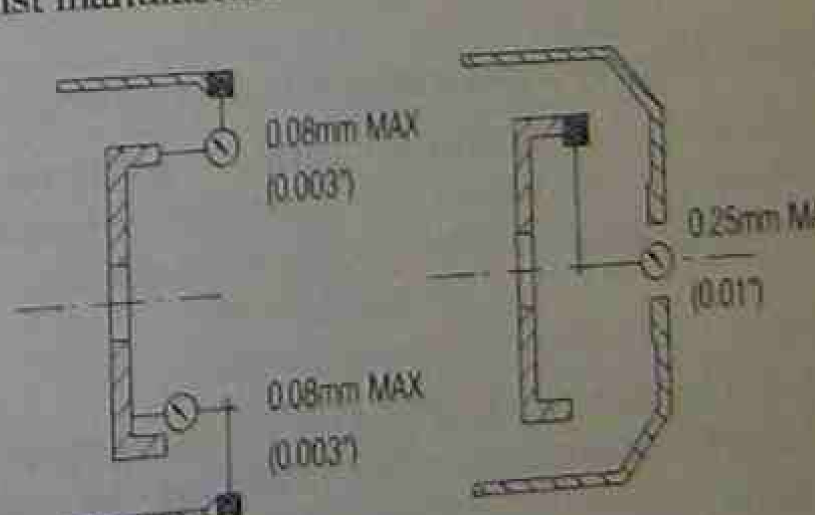


Fig. 7-93 Checking run-out for plate clutches.

Assembly

Assembly should not be attempted unless alignment has already been checked.

The assembly of coupling type clutches requires that the halves be firmly secured to the shafts, usually with parallel keys. Grub screws are often used to prevent sideways movement.

When assembling plate clutches, care should be taken to ensure that no grease or oil comes into contact with the linings. The splines or keyway on which the clutch plate is mounted should be given a light coating of lubricant.

Roller and sprag clutches may be assembled in a number of different ways depending on the application. The major considerations are alignment, secure location, accurate positioning and the avoidance of any undue axial loading on clutch bearings.

Lubrication

Roller and sprag clutches require correct lubrication in order to achieve maximum service life. Generally they will be oil lubricated although grease is also used under special conditions. Under no circumstances should lubricants containing molybdenum disulphide or graphite or similar additives be used as these reduce the coefficient of friction and prevent locking of the clutch at the point of torque transmission.

The engagement of tooth clutches can be assisted with a light coating of lubricant. Other types of clutches do not normally require lubrication except where they contain bearings and these will usually be sealed.

FAILURE PATTERNS

Clutches are required to perform a demanding role and carry fluctuating loads under difficult conditions. They are a potential source of maintenance problems if not properly installed and serviced.

Operating symptoms

Because of the important function it performs, any malfunction of a clutch will usually be very obvious and affect the performance of the drive system.

Noise

When a friction clutch is in good operating condition, it should not generate noise. There may be a slight change in tone between engagement and disengagement due to the loading of a thrust bearing.

Roller and sprag clutches should be silent when driving but may generate noise when freewheeling or over-running. As with other machine elements it is change in noise that indicates a possible malfunction. The noise generated by a clutch under normal operation and in good condition should be established when it is first put into service. Any significant deviation may then be treated as a sign of malfunction.

Shudder

A clutch may shudder or chatter due to grabbing of the friction surfaces. This may cause vibration of the whole transmission. There are a number of conditions that may cause this problem. With friction plate clutches the condition of the friction material and the pressure plate surfaces may deteriorate due to wear or distortion. Oil on the friction material may also glaze and interfere with operation. Loose coupling halves, misalignment and wear of rollers and sprags may also cause a drive to shudder.

Slipping

Most clutches will slip briefly at the point of engagement but should then run without slipping. A slipping clutch may result in either a loss of speed of the driven unit, an over speeding of the drive, or both.

Overheating

A properly engaged clutch should not generate heat. Overheating may be associated with slipping or bearing failure.

Non-engagement

It is possible that a clutch may not engage at all. This would suggest a problem with the actuating system or a jamming of the clutch mechanism.

Non-release

Non-release of a clutch may be a result of problems with the actuating system. It may also be due to binding of the driven plate or half on shaft splines or keyway. An accumulation of dirt or corrosion in roller and sprag clutches may cause a freewheeling mechanism to bind.

Symptoms found on inspection

Once a clutch begins to malfunction it should be taken out of service to avoid further damage. The following are the symptoms most likely to be found on inspection.

Wear

The most common cause of plate clutch malfunction is wear of the friction material that is used to achieve

engagement of the drive. Under normal circumstances, when the clutch has been operating correctly, wear should be uniform across the linings. Uneven wear may suggest misalignment or distortion of the pressure plate. The pressure surfaces will also exhibit signs of wear after extended operation and this will be accelerated if the linings are not renewed at the appropriate time. Wear may also be evident on the face of the thrust bearing.

Wear of keys or splines may cause misalignment and may be a contributing factor to uneven wear of clutch linings.

Careful inspection may be required to reveal evidence of wear of rollers and sprags in freewheeling clutches. The annular races of sprag clutches and the wedging faces of roller clutches may also wear under normal operation.

Oil on linings

Evidence of oil on the friction material linings of a clutch would explain slipping. If the oil has burnt to a glaze then this may explain shudder or grabbing of the clutch.

Mechanical damage

External damage to a clutch may be evidence that it has been struck by some object. The effect of this on alignment should be checked.

Damage to clutch components during assembly or as a result of improper operation may lead to operating problems and further damage.

Friction plate clutches should be carefully inspected for damage to pressure plate surfaces, pressure springs, thrust race and contact surface, splines, keyways and shaft surfaces, and the operating mechanism.

Freewheeling clutches should be inspected for damage to rollers and sprags and the contact races. The mechanism should also be inspected for broken or damaged springs.

When a clutch contains bearings they should also be inspected for damage and for evidence of wear.

Causes

The likely causes of clutch failure are as follows.

Misalignment

As mentioned above, misalignment of shafts will result in uneven wear and premature failure.

Inadequate lubrication

Freewheeling clutches will wear rapidly and fail prematurely if the correct lubrication is not provided.

Improper assembly

When assembling plate clutches it is important that pressure springs are properly located and that housing spigots are properly aligned.

Failure to mount clutch hubs properly on keyed shafts or splines will cause malfunction due to misalignment and vibration.

Interference

Clutches that rotate with the drive should not be allowed to contact a guard or other machine elements.

Contamination

Leakage of oil into a friction plate clutch assembly from shaft bearings will seriously affect operation. Oil seals should be maintained in good condition to protect the clutch mechanism.

Dirt build-up in a clutch mechanism will prevent free movement of components and interfere with operation.

Overload

The application of excessive torque to a clutch is likely to cause rapid wear and lead to premature failure. In the case of some clutches this may be preceded by slipping or other malfunction.

Summary of the failure patterns associated with clutches

Symptoms		*Causes*
Operating	*Inspection*	
Noise Shudder Slippage Overheating Non-engagement Non-release	Wear Oil Mechanical damage	Misalignment Inadequate lubrication Improper assembly Interference Contamination Overload

C H A P T E R

SHAFT ALIGNMENT

Perfect alignment is seldom achieved but good alignment can and should be possible for any machine of which continuous, reliable operation is demanded. Poor alignment can cause vibration and wear and lead to premature failure of bearings, seals, couplings and other machine elements. Despite the fact that flexible couplings can tolerate quite significant degrees of misalignment, it is recommended that shaft alignment should always be as accurate as possible as any misalignment present will set up vibration patterns that will be transmitted through the machine and tend to cause wear. Alignment is a critical aspect of machinery operation and the techniques used should be properly understood.

PRINCIPLES OF ALIGNMENT

Misalignment

Shafts can be misaligned in two ways.

Parallel misalignment

This occurs when the shaft centre-lines remain parallel, but are offset, as shown in Fig. 8-1.

Fig. 8-1 Parallel misalignment

Angular misalignment

This occurs when the shaft centre-lines are out of parallel, although they may intersect at the coupling, as shown in Fig. 8-2.

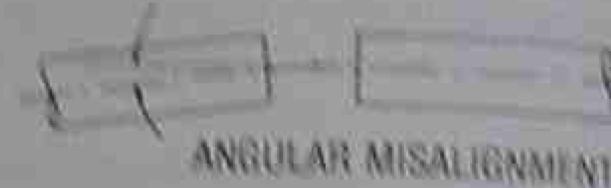


Fig. 8-2 Angular misalignment

It is usual to find that both types of misalignment occur simultaneously and the alignment techniques described are designed to overcome both.

Tolerances

Depending on the size and type of the coupling, coupling manufacturers may indicate that parallel run-out of up to 0.5mm (0.020") and angular run-out up to 15° may be tolerated. It is important to realise that these figures represent the coupling tolerance and not the machine tolerance. Machinery manufacturers normally supply a recommended tolerance for a particular machine and some companies have their own standards which may be even more stringent. The higher the speed and the power transmitted, the more critical alignment becomes.

Adjustment

It is normal practice to adjust the position of the driving machine relative to the driven machine, especially when the driving machine is an electric motor with no interconnecting pipework. The position of driven machines is normally determined by associated process piping. It will have been set up to eliminate pipe strain and should not be disturbed.

Ideally the driving machine should be free to move in any direction with respect to the driven machine. However, if both are mounted directly onto the baseplate without shims, then it is clearly impossible to move the driving machine down with respect to the driven machine. Therefore before the alignment procedure is started a preliminary shimpack of, say, 1.25mm (0.050") should be installed under both machines. These shims should be installed before the interconnecting pipework is aligned and bolted up.

Alignment in the vertical and horizontal directions should be carried out separately to avoid confusion and error. It is normal practice for vertical alignment to be carried out first otherwise the horizontal alignment may be disturbed when vertical adjustment is made.

Dial indicators

The operation of indicators should be clearly understood before they are used in shaft alignment. The important points to remember are that when the plunger is depressed, the gauge reads positive and when it is extended the gauge reads negative.

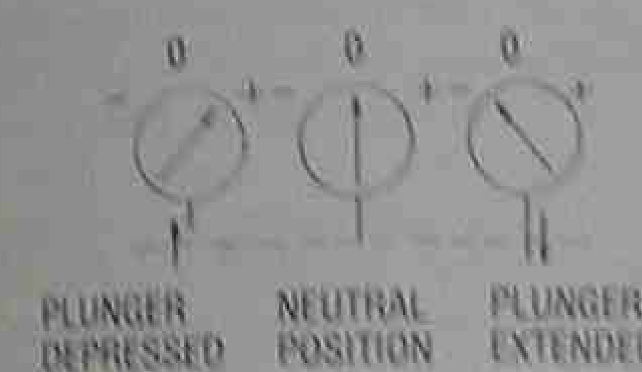


Fig. 8-3 Reading a dial indicator.

It is also important to remember that when a dial indicator is zeroed prior to taking readings, the plunger should be approximately in the mid-range position so that it is free to move in either direction.

Total indicator run-out

When a reading is taken with a dial indicator over 180°, i.e., from top to bottom of a shaft or from side to side, the difference in readings is known as the total indicator run-out (TIR).

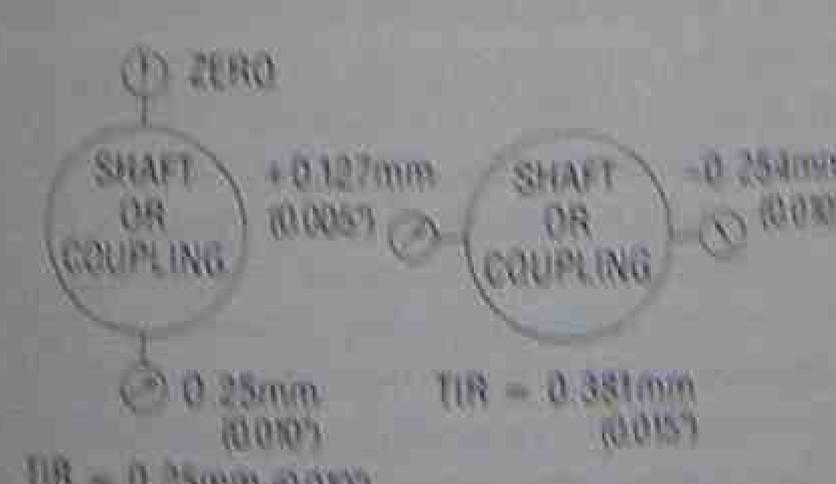


Fig. 8-4 Total indicator run-out.

For these readings to be properly interpreted it must be understood that total indicator run-out represents twice the shaft centre-line offset. As the indicator is turned through 180°, the shaft offset is doubled on the indicator scale, as shown below.

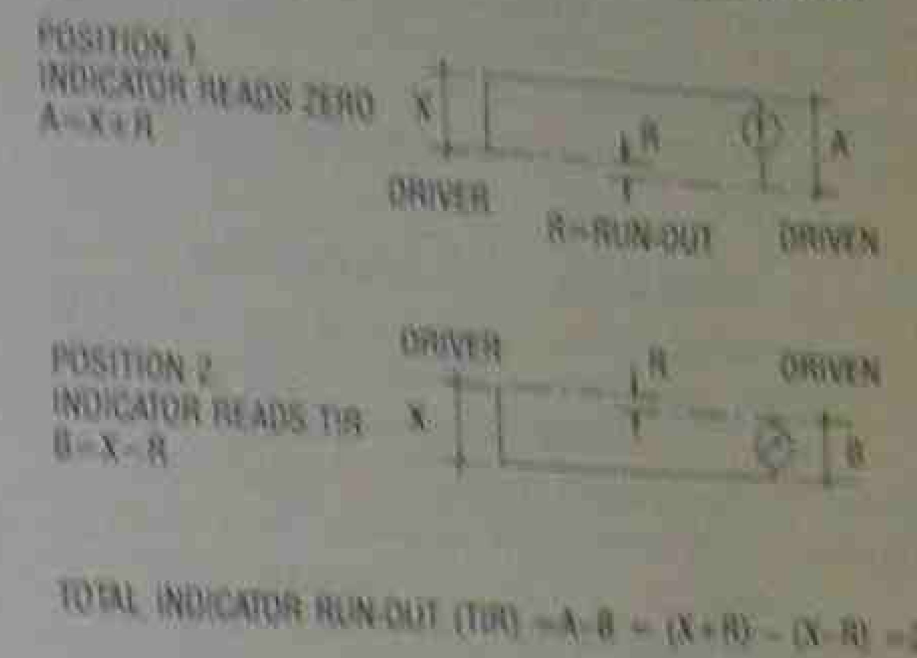


Fig. 8-5 Total indicator run-out= 2 x shaft run-out

Thermal growth

Because of the difference in thermal expansion between the driving and the driven machine during operation, shafts often need to be set up cold with an offset that disappears when the machines are up to operating temperature. This occurs most commonly with steam turbine drivers. Because a turbine undergoes considerable thermal expansion during operation, the shaft needs to be set lower than that of the driven machine in the cold state.

Information about thermal growth is normally supplied by the manufacturer but otherwise can be measured by setting up dial indicators on a fixed reference and recording the actual growth at the operating temperature.

It is generally accepted that a 'hot-check' for machinery alignment is of little value. Not only is it costly and time consuming to bring a machine up to temperature, stop it and determine alignment before it cools down, but the results are highly questionable.

Dowelling

In order to ensure that shaft alignment is maintained when the machine is in operation it is advisable that the position of both machines is secured by dowelling. Two dowels should be used for each unit and they should be located as far apart as possible. The dowel holes should be drilled and reamed in position through the machine mounting into the bedplate. The dowel pins should be parallel ground to give a light drive fit in the holes. The size of the dowels will depend on the size of the machine.

METHODS OF ALIGNMENT

The straight edge and feeler gauge method

The simplest and easiest method, but also the least accurate, is to use a straight edge across the coupling halves to test for parallel run-out and feeler gauges or a taper gauge between the coupling halves to test for angular run-out as shown in Fig. 8-6.

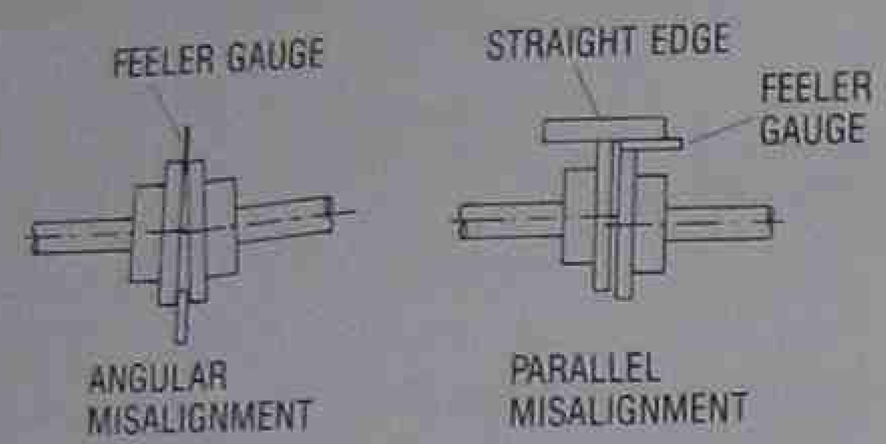


Fig. 8-6 Using a straight edge and feeler gauge.

Adjustments to the shimpacks under the driving machine will need to be made on the basis of trial and error, although repeated alignment work on an individual machine will help the technician to develop a feel for the adjustment required.

It is recommended that, with this method, angular run-out be eliminated first and then the remaining parallel run-out can be eliminated by making equal adjustments at both mountings of the driving machine.

This method is only recommended for use on machines fitted with flexible couplings capable of tolerating up to 15° angular run out, and 0.25mm (0.010") parallel run-out, and for low horsepower, slow speed conditions. The degree of accuracy achieved is unlikely to be consistently better than 0.125mm (0.005").

The face and rim method

This method can only be used for couplings installed with a spacer, and involves the use of two dial indicators supported on a bracket or two brackets set up across the coupling, and mounted on the driven machine as shown in Fig. 8-7. Note that the coupling spacer has to be removed for this method to be used.

As shown in Fig 8-7, parallel run-out is measured by readings taken on the outside diameter of the coupling and angular run-out is measured by readings taken on the face of the coupling half. The support brackets on which the dial gauges are mounted must be rigid and securely attached to the coupling halves. The alignment procedure is carried

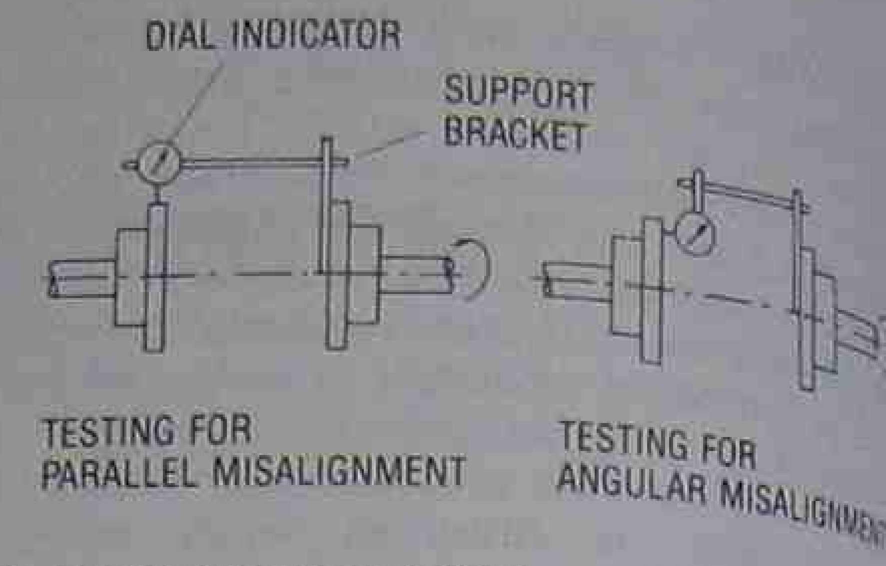


Fig. 8-7 The face and rim method.

out by first zeroing both indicators at top dead cen[tre] and then turning the shaft supporting the ga[uges] through 360° and recording the gauge readings at 9[0°] intervals. A typical set of readings would appear [as] follows.

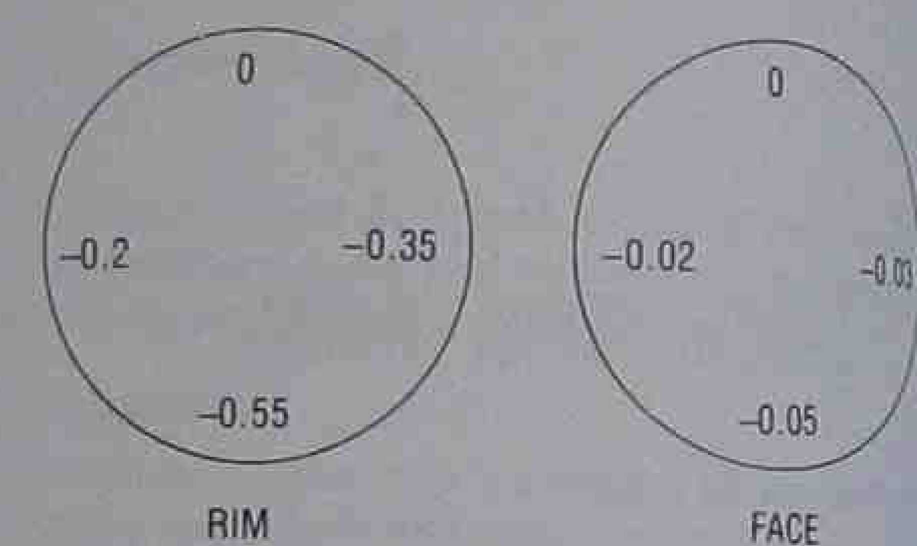


Fig. 8-8 A typical set of readings.

The readings taken on the coupling o.d. or r[im] represent the parallel run-out in both the vertical a[nd] horizontal directions. In this case the driv[ing] machine has been recorded as 0.275mm (0.01[1"]) high at the coupling (remember shaft centre-l[ine] offset is half the total indicator run-out), while t[he] run-out in the horizontal plane is 0.075mm (0.00[3"]). The readings taken on the face of the coupling sh[ow] the angular run-out which, in this case, can be se[en] to be greater in the vertical plane than the horizon[tal].

Although it is theoretically possible to use th[e] readings obtained to calculate the precise adjustme[nt] required at the mountings of the driving machin[e] this is rarely, if ever done in practice. The procedu[re] for adjustment, as with the straight edge and feel[er] gauge method, is essentially one of trial and err[or] backed up by the experience of the technicia[n]. Consequently several sets of readings may b[e] required with progressive adjustments being ma[de] until satisfactory alignment is achieved.

The face and rim method has traditionally been one of the most widely used alignment techniques and is the specific method recommended in many maintenance manuals for rotating machinery. However, for high speed, high horsepower machinery where alignment tolerances become more critical, this method has a number of shortcomings.

Shortcomings of the face and rim method

- The indicator readings obtained by this method reflect not only the misalignment of the shafts but also the out-of-roundness, eccentricity and surface imperfections of the coupling upon which the readings were taken. Even with good quality, high speed couplings, run-out in excess of 0.025 mm (0.001") in the coupling o.d. and face is not uncommon. These errors can be eliminated by determining the geometry of the coupling hub or by turning both shafts together but these precautions are difficult and rarely taken.
- To obtain accurate face measurements it is vital that end float is accounted for. The only way to do this is to ensure that both shafts are hard up against the thrust bearing at all times. Consistent readings are difficult to obtain however, because the axial position of the shaft will depend on how hard it is pushed.
- Face readings are taken across a relatively small diameter and the angular misalignment measured here can be greatly magnified at the outboard end of a long shaft. For example a difference of 0.05 mm (0.002") measured across a 100mm (4") coupling, which might normally be considered as acceptable may represent an error of as much as 0.75mm (0.030") at the rear end of the machine shaft.
- Removal of the coupling spacer is time consuming and increases the risk of coupling damage.

These problems make the face and rim method unsuitable for high speed machinery and have also resulted in it being superseded in general applications by the reverse indicator method which has gained increasing popularity in recent years.

The reverse indicator method

This method involves using dial gauges to take readings on the outside diameter of the coupling hubs (or the shafts) only. No face readings are involved. The gauges are supported on separate brackets mounted on opposite halves of the coupling so that a reading is obtained on each. (Fig. 8-9)

The main advantage of this method is that it allows the relative position of the two shafts to be accurately determined. It is then possible to determine the exact adjustment required at the mountings of the driving

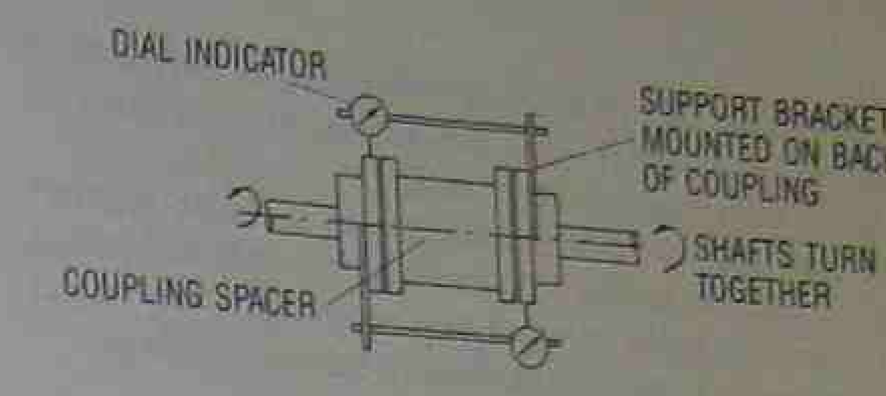


Fig. 8-9 Typical arrangement of the dial gauges.

machine to produce proper alignment. The shortcomings of the face and rim method are overcome in the following manner.

- By turning both shafts simultaneously, errors caused by coupling hub run-out are entirely eliminated.
- Since no face readings are involved, the end float of shafts is of no concern.
- By spanning the entire coupling, angular misalignment is greatly magnified and can be measured with greater accuracy.
- If the support brackets are designed to be mounted on the back of the coupling halves, the coupling spacer does not need to be removed.

One problem that affects the reverse indicator method as well as the face and rim method is the problem of support bracket deflection. This is a matter of concern and should not be ignored if accurate cold alignment is to be achieved. The problem can be minimized by ensuring that the brackets are as rigid as possible, and, if necessary, the deflections can be measured and appropriate corrections made. Deflection of the bracket can be measured by mounting it on a piece of barstock held in a lathe chuck as shown in Fig. 8-10. With the bracket at top dead centre of the barstock the gauge is set to zero. The assembly is then rotated through 180° and the gauge is read at the bottom position.

The difference in indicator readings reflects the deflection in the bracket. The figure can be used to correct readings taken during machine alignment.

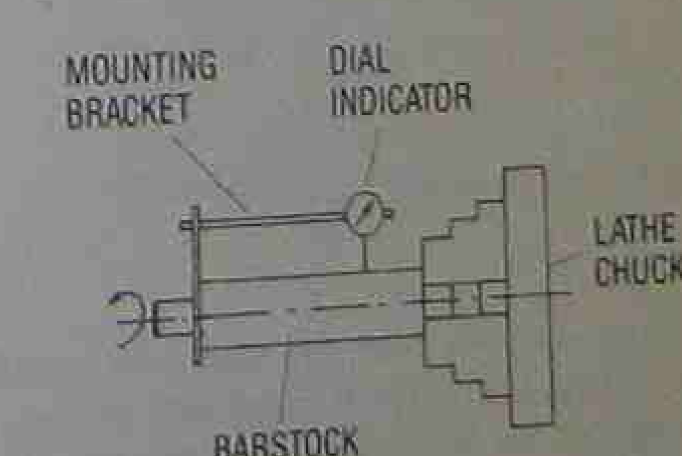


Fig. 8-10 Measuring deflection of the bracket.

Because of the superior features of the reverse indicator method, the procedure is described in detail on a step-by-step basis.

1 Set up the dial indicators on suitable support brackets, preferably so that both indicators read on the same side of the coupling. This is not essential, but saves time in taking readings.

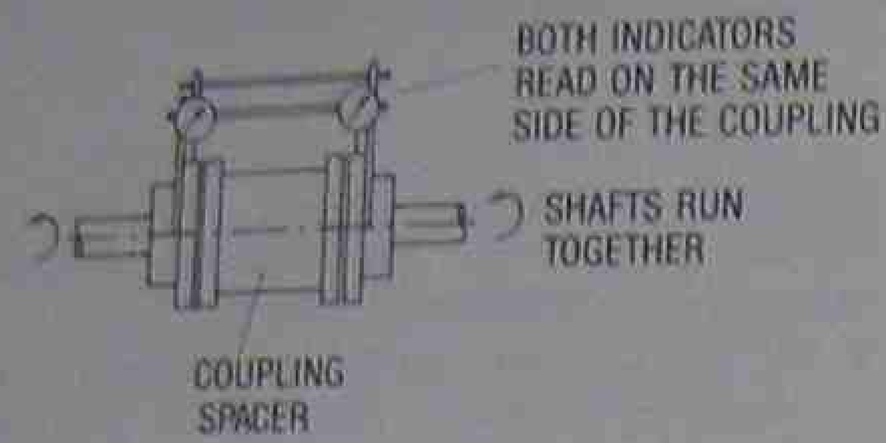


Fig. 8-11 The recommended arrangement for setting up the dial indicators.

2 Adjust both indicators to zero at top dead centre of the coupling hubs.

3 Check that the indicator plungers are in mid-range and readjust if necessary.

4 Check that the indicators can be rotated through 360° without interference.

5 Adjust the driving machine to a roughly central position in the horizontal plane with respect to the driven machine. Indicator readings, if zeroed at top dead centre, should be roughly equal at the 90° and 270° positions, as shown in Fig. 8-12.

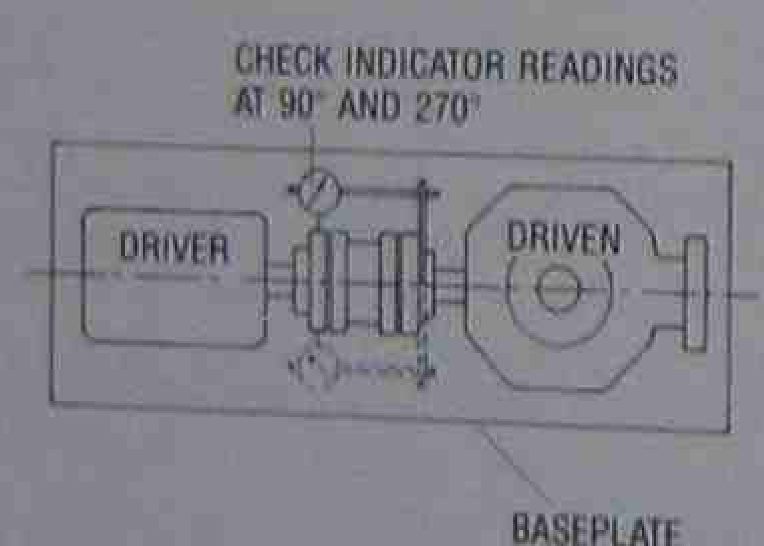


Fig. 8-12 Centralising the driver.

This step ensures that the indicator readings taken during the vertical alignment process are taken across the full diameter of the coupling or shaft. If the two machines are not roughly in line, then the vertical readings taken will give a false value of total indicator run-out as shown in Fig. 8-13.

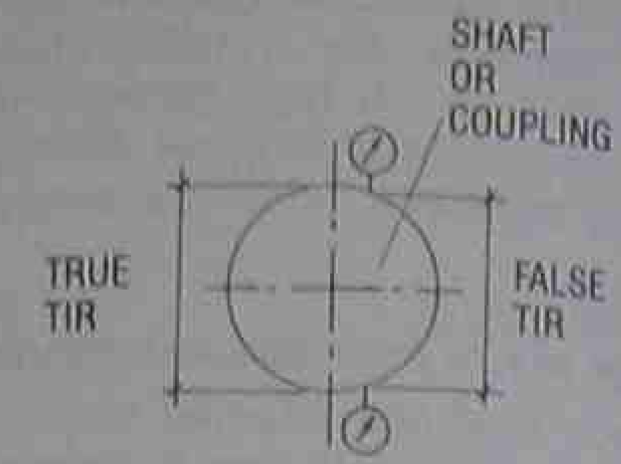


Fig. 8-13 Difference between false TIR and true TIR.

6 Measure the distance between the indicator plungers and the distance between the indicators on the driven machine and the centres of the mounting bolts on the driving machine as shown in Fig. 8-14.

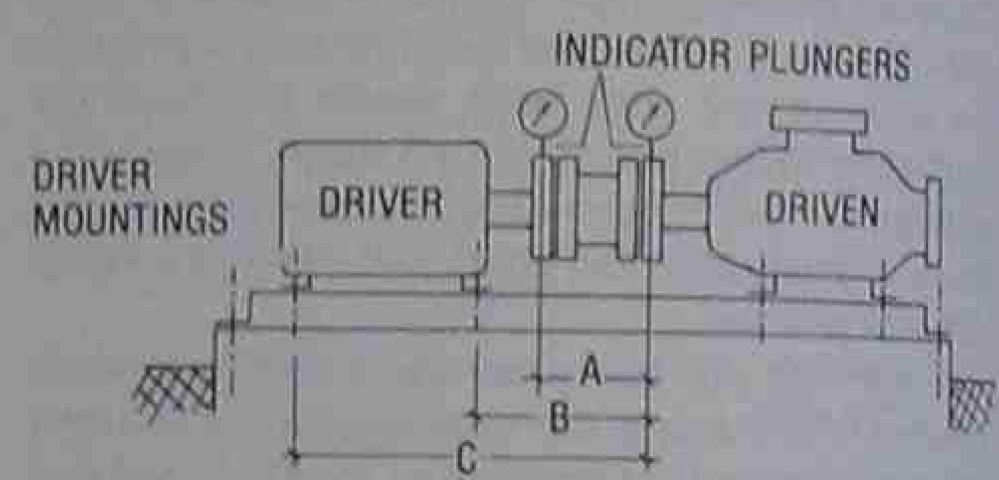


Fig. 8-14 Machine dimensions.

7 Plot the measurements A, B, and C on a sheet of graph paper as shown in Fig. 8-15. A suitable horizontal scale will need to be selected according to the size of the machine.

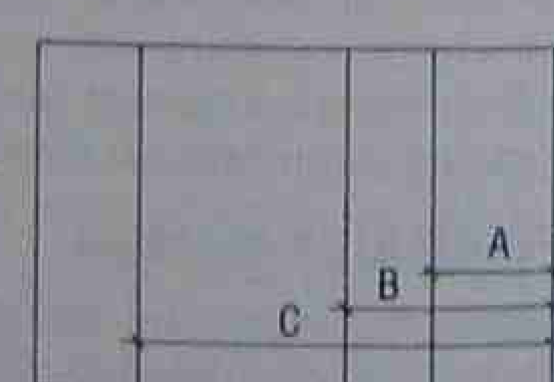


Fig. 8-15 Dimensions marked on graph.

The right hand vertical axis now represents the position of the indicator reading on the coupling hub of the driven machine. The other positions marked on the graph represent the position of the indicator readings on the coupling hub of the driving machine and the front and rear mounting positions of the driving machine.

8 Check the zero reading on the indicators at top dead centre, and then rotate both indicators through 180°. Record the total indicator run-out for each indicator at bottom dead centre.

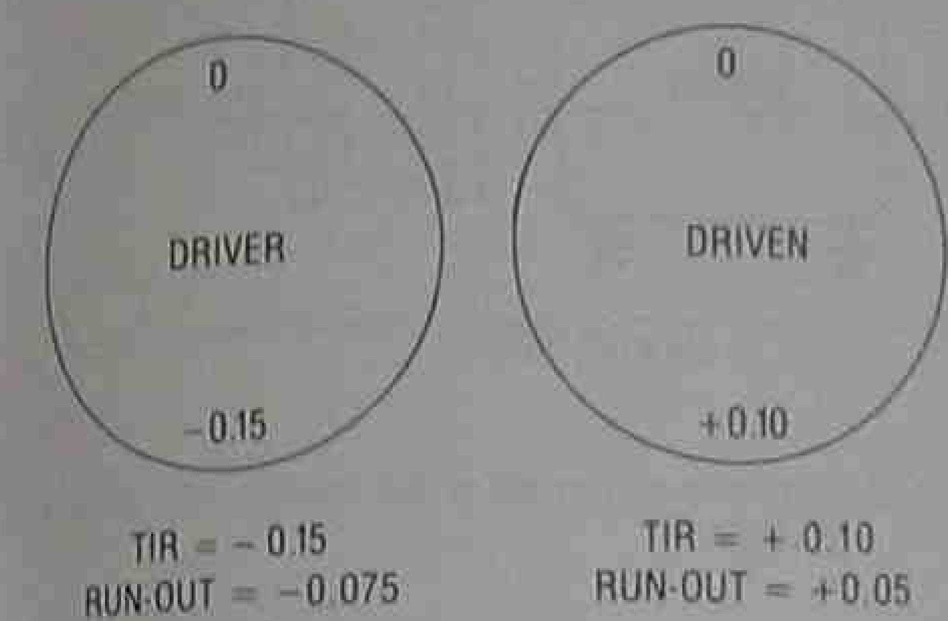


Fig. 8-16 Sample readings.

9 Determine the relative position of the driving machine shaft with respect to the driven machine shaft by halving the total indicator run-out and determining which shaft is high at each point of measurement. See the example given in Fig. 8-17.

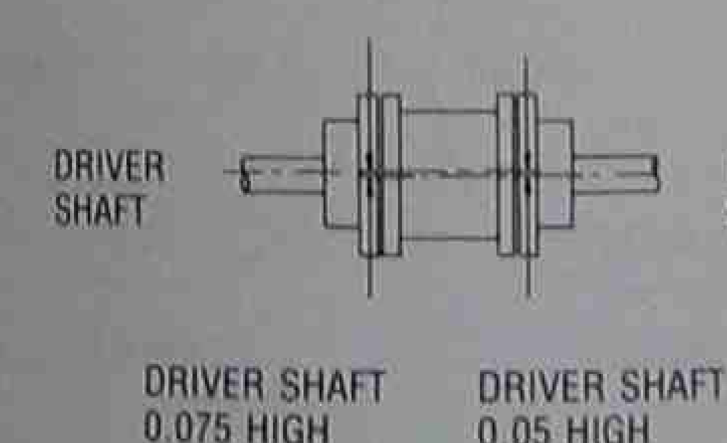


Fig. 8-17 Relative shaft positions at points measured on coupling.

The question of which shaft is high can be resolved by considering whether the indicator reading is positive or negative at bottom dead centre. The following chart shows the various possibilities.

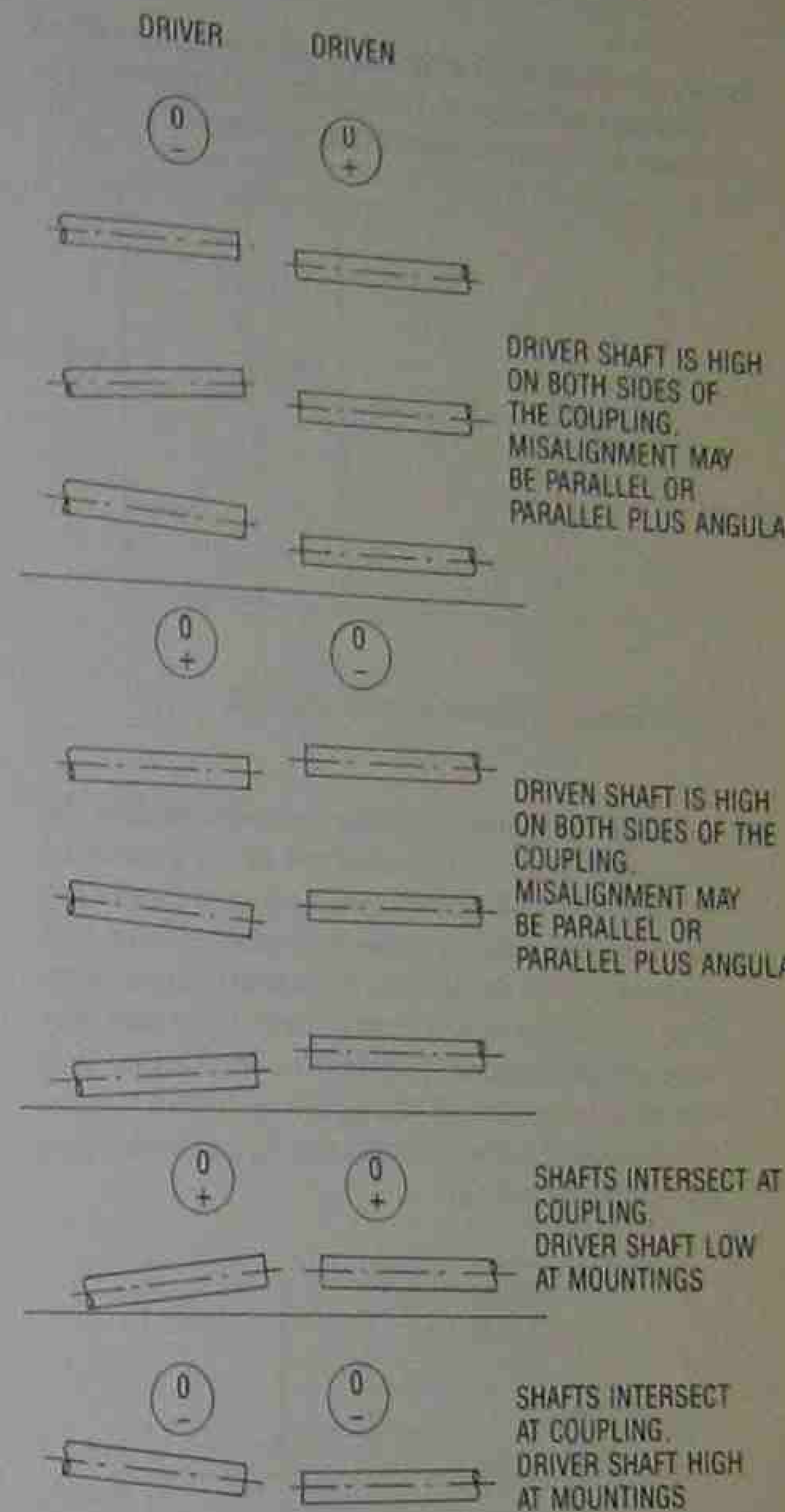


Fig. 8-18 Possible relative shaft positions.

10 Draw a horizontal line on the graph to represent the centre-line of the driven shaft.

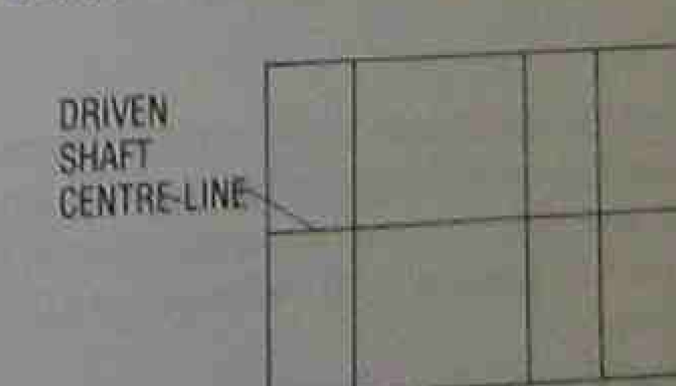


Fig. 8-19 Driven shaft centre-line shown on graph.

11 Plot the positions of the driving machine shaft on the vertical axes of the graph corresponding to the point of measurement as shown in Fig. 8-20.

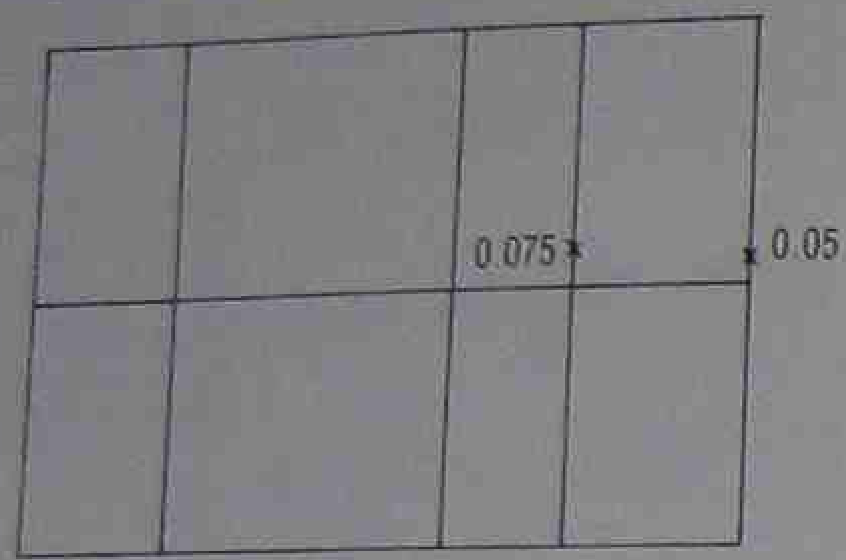


Fig. 8-20 Plotting the position of the driving shaft.

12 Join the two points together, and extrapolate the line back to the positions of the driving machine mountings. The line drawn now represents the position of the driving machine shaft centre-line relative to the driven machine shaft centre-line. The amount of adjustment at both front and rear mountings required to bring the shafts into line can now be scaled off the graph as shown. In the example given, the motor is too high at both front and rear mountings.

Fig. 8-21 Scaling off the required adjustment.

13 Adjust the shimpacks under the driving machine mountings accordingly, retighten the mounting bolts, and then recheck the alignment as before. Repeat the process until the required accuracy is achieved.

14 Repeat the whole procedure to obtain alignment in the horizontal plane. This time the indicators should be set to zero at one side of the coupling and total indicator run-out read at the other.

Jacking screws should be used to adjust the position of the driving machine, and it may be necessary to set up an additional dial indicator, using a magnetic mounting, to measure the movement of the machine, as shown in Fig. 8-22.

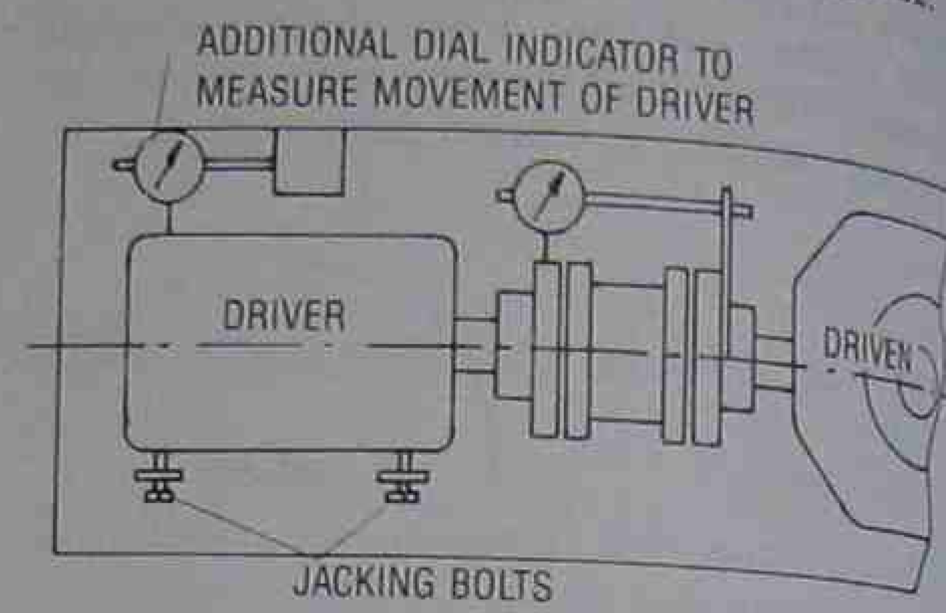


Fig. 8-22 Measuring movement of the machine with an extra dial indicator.

15 When both horizontal and vertical adjustments are complete, a final set of readings at 0°, 90°, 180° and 270° should be taken, with the indicators zeroed at 0°, as a record of the relative shaft positions. Readings should be recorded in the following manner, as if facing the respective coupling halves.

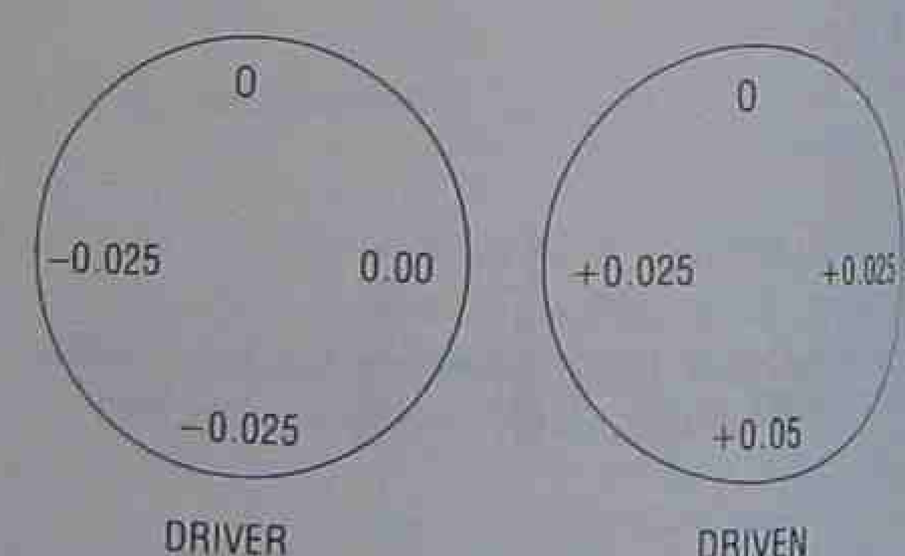


Fig. 8-23 Method of recording readings.

General hints on alignment

- Check the tools. Make sure that dial indicators do not stick.
- Use good quality shimpack material. If the material is damaged, discard and replace with new material.
- Find a good way to turn the shafts. For large machines, make up a clamp-on fixture if necessary.

- If a mechanical end face seal is fitted to the machine, disengage the faces before turning the machine shafts.
- Beware of environmental changes that may affect alignment. Machines exposed to direct sunlight may undergo non-uniform thermal growth.
- Make sure that indicators are positioned to read squarely on the shaft.
- Always check the readings taken. If the readings cannot be repeated, they are not acceptable.
- Stop the shaft at precise 90° increments and turn the shaft in one direction only. If you pass the 90° mark, go round again.

C H A P T E R 9

SEALS

Seals are used to prevent the leakage of liquids, solids and gases from items of rotating machinery and other types of industrial equipment and to stop dirt and other sources of external contamination from entering a machine or piping system.

As industrial equipment has become more sophisticated the conditions under which seals have to operate have become more arduous and sealing technology has developed rapidly in the last twenty years in order to meet these developing needs. The demands of the aerospace industry have had a particularly strong influence on the advancement of seal technology.

Despite the sophisticated developments that have taken place, traditional seal designs are still found in the vast majority of industrial applications.

Before examining particular types of seals it is useful to consider how seals can be classified according to their operating characteristics and application.

The most fundamental distinction to be understood is the difference between static seals and dynamic seals. **Static seals** are those used for sealing surfaces between which there is no relative motion whereas **dynamic seals** are used where relative motion occurs. Static seals are often referred to as **joints** whereas dynamic seals may be referred to as **packing** or **gland packing**.

Dynamic seals can be further grouped into two main categories, **contact seals** where the sealing element rubs against a mating surface under load and **clearance seals** which operate with positive clearance between the sealing surfaces.

The family tree of seals (Fig. 9-1) shows the common groups of seals found in rotating machines.

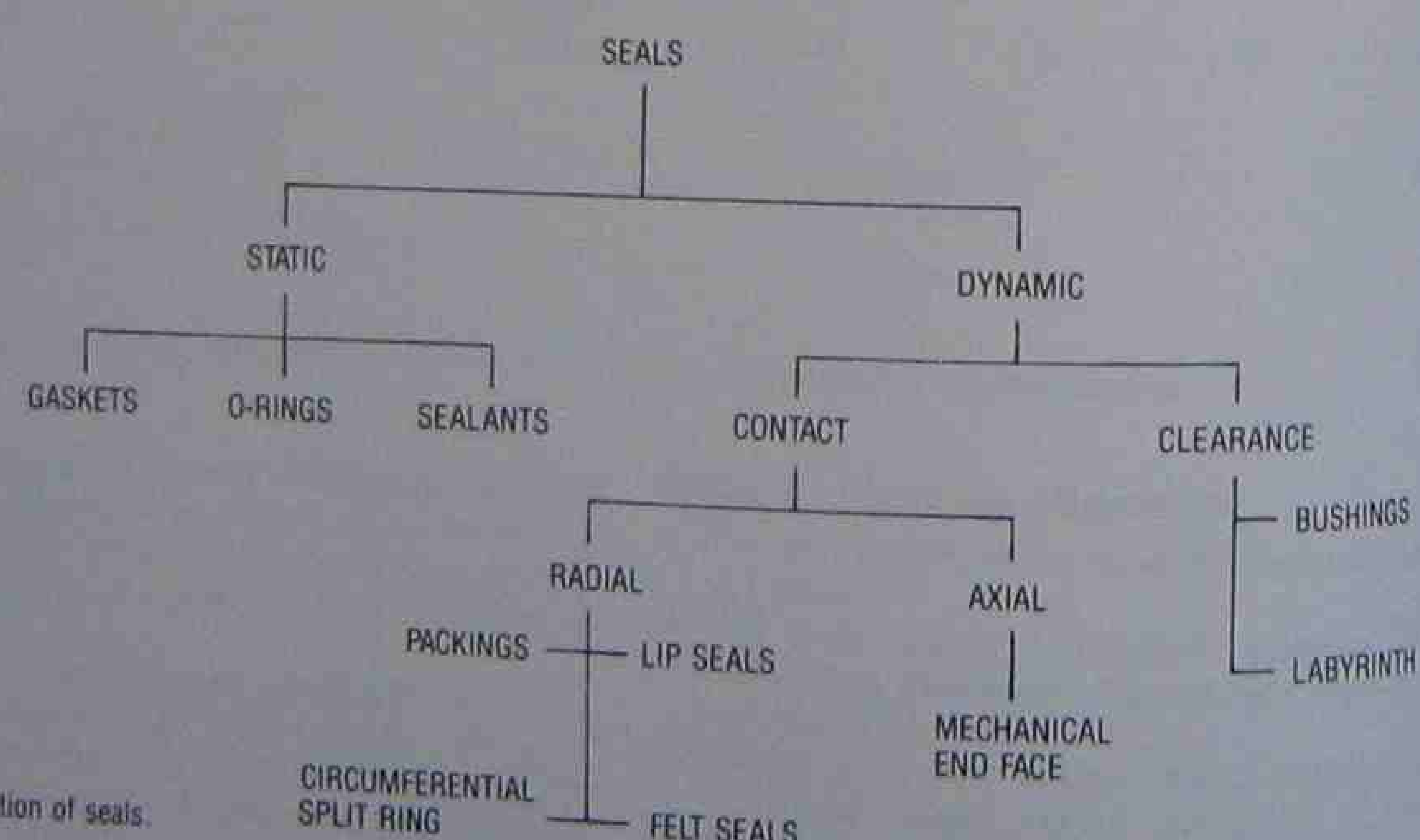


Fig. 9-1 Classification of seals.

9.1 GASKETS

A gasket is a static seal used to contain liquids, solids and gases in all types of machinery, containers and piping systems and may operate under a wide variety of service conditions. Gaskets are normally located between rigid and usually metallic sealing faces.

SEALING PRINCIPLES

A gasket requires an externally applied compressive force to maintain sealing contact. This force is normally provided by a set of flange bolts but various types of clamps or clips may also be used. The compressive force applied to the gasket material between the flange faces must be capable of:

- Accommodating surface variations in the flange faces and in the gasket material itself;
- Overcoming the hydrostatic end-force caused by the internal pressure trying to push the flanges apart;
- Leaving sufficient residual stress to contain the pressure and prevent it from extruding the gasket through the clearance space.

Fig. 9-2 shows the relationship between the forces acting on a gasket.

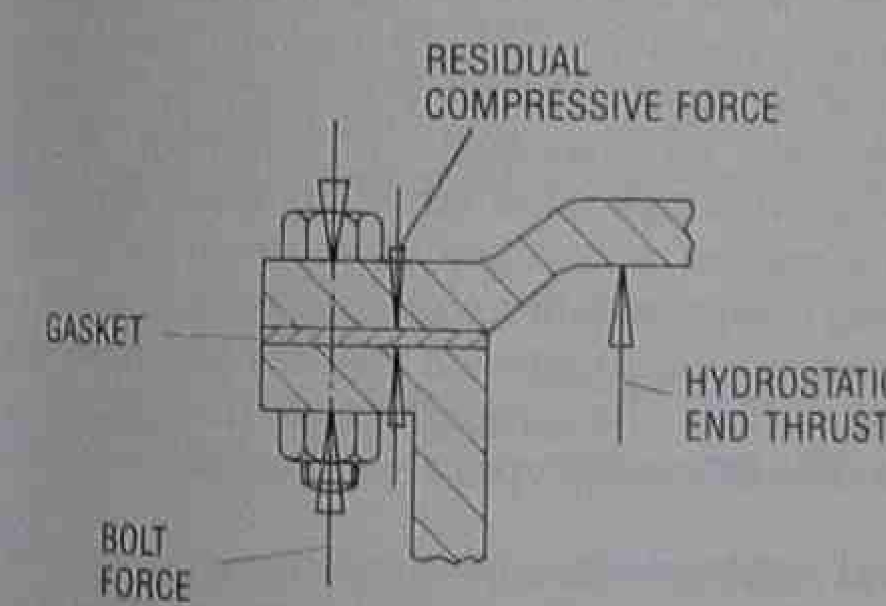


Fig. 9-2 Relationship between the forces acting on a gasket.

To ensure that the gasket continues to seal as the material relaxes in service it is recommended that the residual compressive stress be maintained well above the internal pressure to be contained. A factor of two is usually applied. Hence the compressive load on a gasket can be calculated from the relationship:

residual compressive stress = 2 x internal pressure

This then enables the bolt force or assembly load to be calculated from:

bolt force = hydrostatic end-thrust + residual gasket load

Because a safety factor of two has been allowed in determining gasket stress, no specific allowance for accommodating surface imperfections need be made. However, the ability of a gasket to accommodate surface imperfections will depend on the nature and thickness of the material in relation to the surface finish of the flanges and the internal pressure. A soft gasket material, which may provide a good seal against a comparatively rough surface, may extrude at the working pressure required. It may be necessary to improve the surface finish of the flanges so that a thinner, harder gasket material, capable of containing the working pressure, can be used.

Generally speaking, thinner gasket materials are more suitable for containing higher internal pressures when used between suitable machined flanges because the area exposed to the internal pressure is reduced, and thus the force tending to extrude the gasket is also reduced. See Fig. 9-3.

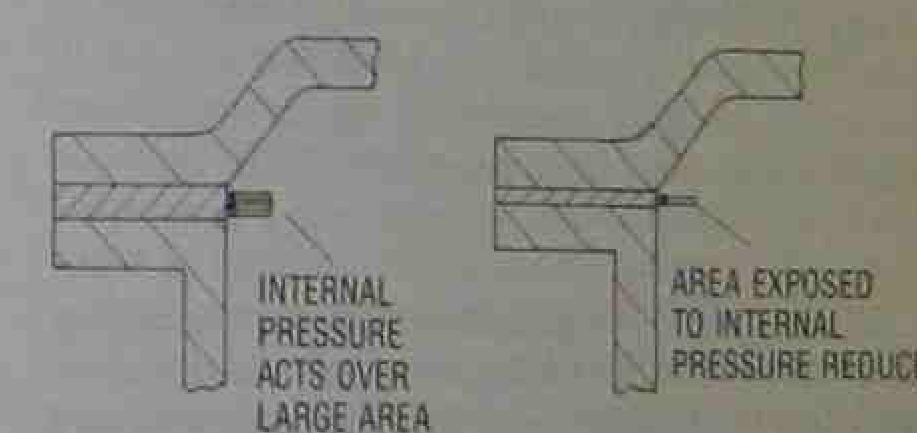


Fig. 9-3 Advantage of thinner gasket materials.

Although a high degree of surface finish would appear to be desirable in all cases, this can be unnecessarily expensive and it should be remembered that some surface roughness helps to provide grip, especially for harder gasket materials.

If good sealing is to be achieved, it is also important that flange faces are parallel and sufficiently rigid to resist distortion.

Although flange design can vary widely, there are a number of standard flange arrangements in common use in most industries and these are shown in Fig. 9-4.

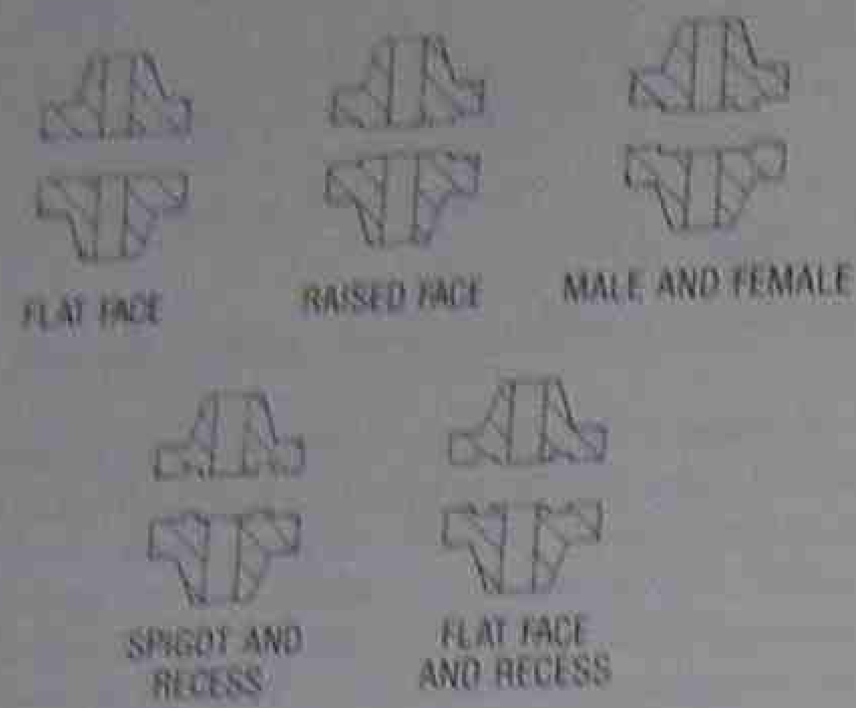


Fig. 9-4 Standard flange arrangements.

Gaskets may either be the ring type, where the outside diameter is located inside the bolt circle, or the full face type, as shown in Fig. 9-5.

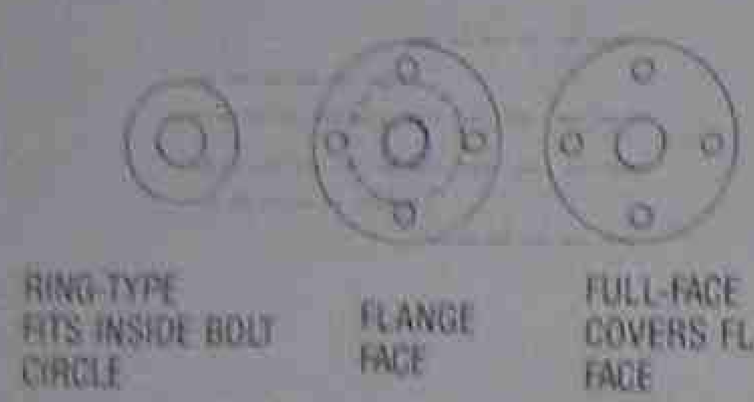


Fig. 9-5 Full-face and ring type gaskets.

The spacing of flange bolts is related to the rigidity of the flange and should be designed to ensure even distribution of gasket load. In most cases, standard flanges are used which take these factors into account.

GASKET MATERIALS

A gasket material generally needs to have the characteristics of softness and deformability, and a high degree of elasticity and flexibility. The material should also be resistant to hardening and fatigue. The softness and deformability enable the gasket to accommodate the surface irregularities, and elasticity is necessary if the material is to be capable of responding to the slight change of shape required as loading fluctuates and flange bolts stretch.

Although the maintenance technician is not normally required to select gasket materials, some understanding of material properties and limitations is important.

Paper

A low cost material whose properties are improved when impregnated with fillers such as wax. Used most commonly in the automotive industry for sealing water, oil and petrol. They can be used up to 120°C (230°F) and 800 kPa (120 psi).

Cork

Useful in low load situations where flange faces are uneven. It has good resistance to oil and solvents but can be adversely affected by water. Can be used up to 50°C (125°F) and 350 kPa (50 psi).

Rubber bonded cork

The properties of cork can be improved by bonding it with an elastomer such as neoprene or nitrile. This produces a material with higher strength and flexibility that resists extrusion. It can be considered as a good general purpose material for low to moderate loads but is not recommended for highly alkaline or acidic conditions. It can be used from −30°C to 150°C (−22°F to 300°F) and up to 350 kPa (50 psi).

Rubber

This is one of the most versatile gasket materials especially when reinforced. The common property of elastomers is their ability to resume their original shape after deformation. For instance, rubber gaskets can be distorted, if required, during assembly. Although there are many types of rubber they all possess a number of common properties including compressibility and sensitivity to extreme temperature. Both high and low temperatures can seriously affect the properties of most types of rubber. The following types are commonly available.

Natural rubber Excellent mechanical properties but limited by low chemical resistance. Will also deteriorate in sunlight and ozone.

Buna-S A suitable substitute for natural rubber with greater resistance to water and heat.

Buna-N Has improved strength and abrasion resistance and also heat resistance. It is particularly suitable for use with mineral oils and some hydrocarbons but is soluble in others and cannot be used with organic acids.

Neoprene Has similar mechanical properties to natural rubber but has improved ozone resistance. Can be used with oils and non-aromatic hydrocarbons and can be made flame resistant.

Butyl Although its mechanical properties are not as good as natural rubber it has high resistance to most chemicals and is widely used for gasketting. It is not recommended for petroleum-based fluids but has good high and low temperature properties.

Viton This is a fluorinated rubber that performs well with most chemicals except esters and ketones. It has good high temperature performance and ozone resistance.

Silicone rubber This is resistant to water and sunlight and can operate at both high and low temperatures. It is not suitable for use with hydrocarbons or at high pressures.

Compressed asbestos fibre (CAF)

This was the most common gasket material, until the health hazards of asbestos became known, because of its high temperature properties and resistance to chemical attack. It is made up of asbestos fibres bonded with an elastomer and has good mechanical properties. It can be used in most application other than with strong mineral acids.

Plastics

Many different synthetic materials have been developed in recent years and particularly since the limitations of asbestos became evident. One which is commonly used as a gasket material is PTFE (polytetrafluoroethylene or Teflon). PTFE is virtually chemically inert and can be used over a wide temperature range from −190°C to 250°C (−310°F to 480°F). The limitation of PTFE is that it has a tendency to cold flow and as a result is most commonly used in the form of an envelope that fits over a gasket cut from another material such as CAF or metal.

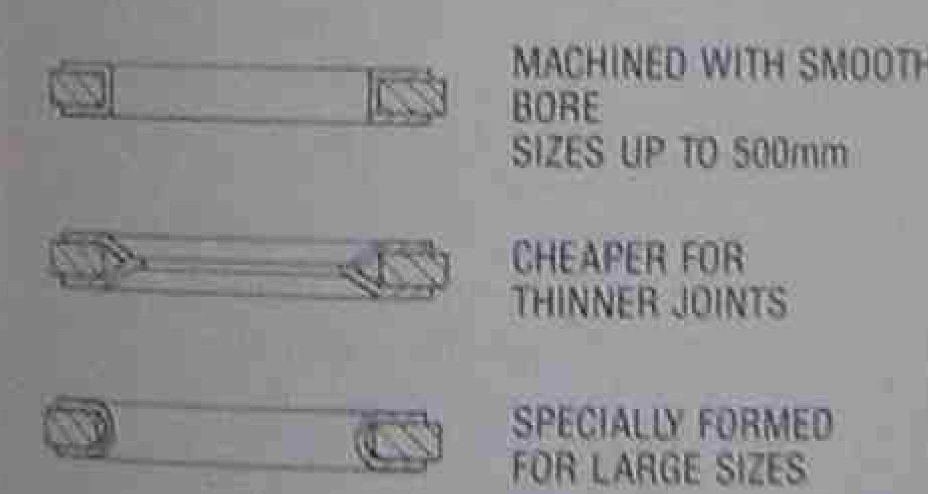


Fig. 9-6 Typical configurations of PTFE envelope gaskets.

Various synthetic fibres, such as aramids, are now used in compounded form with synthetic rubber, such as nitrile, to replace the traditional CAF gasket material.

Metals

When operating conditions become too severe for the kinds of non-metallic gaskets mentioned above, various types of solid metal and semi-metallic gaskets can be used.

Solid metal gaskets made of materials such as lead, aluminium, copper, brass, monel, nickel and alloy steels are used when temperatures and pressures become extreme. They are usually used in a form that concentrates the load over a small area to increase the seating stress rather than as a flat ring.

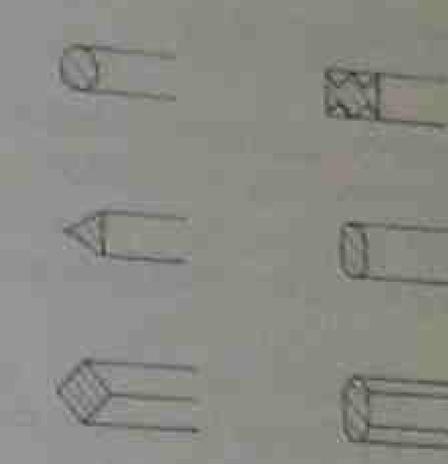


Fig. 9-7 Solid metal gasket sections.

Semi-metallic gaskets come in a number of forms the most common of which are metal clad and spiral wound.

Metal clad gaskets consist of a soft material, such as CAF or millboard, enclosed in a metal covering made from brass, aluminium, copper, monel or stainless steel. They can be made in intricate patterns and are commonly used as cylinder head gaskets and, in the process industry, for heat exchangers and pressure vessels.

Spiral wound gaskets are the most versatile of the semi-metallic type because of their outstanding recovery characteristics. They consist of a V-section metal strip wound into the form of a spiral with a non-metallic filler as shown in Fig. 9-8.

The metal strip is usually made of stainless steel but may be monel, nickel or titanium and the filler may be asbestos, PTFE or graphite. Retaining rings,

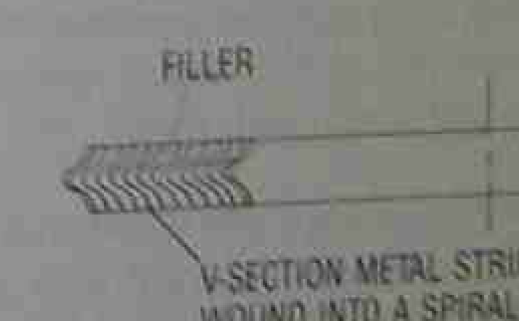


Fig. 9-8 Spiral wound gasket.

either internal or external, are used as shown in Fig. 9-11 to support the spiral wound section and to provide a stop against which the flanges may be tightened to give the gasket the correct amount of compression.

Spiral wound gaskets can operate at temperatures between −250°C and 1000°C (−420°F to 1830°F) and at pressures from vacuum up to 35 000 kPa (5000 psi). They are extremely versatile, and with careful selection of materials are suitable for use with the majority of fluids.

MAINTENANCE PRACTICES

There are a number of general factors that should be considered when removing or installing gaskets.

- It is advisable to have the new gasket cut or fabricated and ready to install before breaking a joint.
- Do not, if possible, make the gasket by hammering on the flange face. This can damage both the material and the flange.
- Use as thin a gasket as possible for the joint conditions.
- For full face gaskets the bolt holes should be the same size as the holes in the flange.
- The gasket inner diameter should be larger than the inside bore of the joint face to prevent the gasket interfering with the fluids contained. The amount will depend on the material. For example, rubber will swell more than CAF and hence greater clearance will be required.
- If a joint has to be broken regularly then a coating of graphite or similar dry lubricant on one or both surfaces may make the gasket easier to remove. If a lubricant is used then a check should be made to make sure it is compatible with the contents of the machine, vessel or pipeline.
- Gaskets on doors and lids that have to be opened frequently can be cemented on one side and smeared with lubricant on the other. The cement chosen must be able to stand up to the operating conditions.

Disassembling a joint

1 Before starting make sure the joint is isolated and that all valves are closed. Drain any residual liquid from the joint and purge any gas if necessary.
2 For piping flanges, loosen and remove all bolts if the gasket is full-faced. For ring type gaskets, loosen all bolts but only remove enough to remove the gasket.
3 In equipment flanges it is recommended that all bolts be removed.
4 Where necessary, spring flanges apart using flange spreaders. If wedges are used care must be taken not to damage the flanges.
5 When the gasket has been removed, clean the joint faces and remove all traces of the old gasket and any jointing compound used.
6 Examine the joint faces for any evidence of scratching, corrosion, erosion or distortion of any kind.

Installing the gasket and assembling a joint

1 Ensure that joint facings are clean and free from burrs.
2 Bolt or stud threads should be clean and lubricated and spot facings on the back of flanges should also be clean.
3 Insert enough bolts in one flange to locate the gasket and make sure it lines up evenly all the way around the inside.
4 With the gasket in place on one flange, bring up the mating flange. Every effort should be made to ensure the flanges remain parallel as they are brought together.
5 Vertical heavy flanges such as those on vessels and heat exchangers should be jockeyed into position using a crane or hoist. These should be positioned on four bolts at an angle of 90° to each other that can be pulled up evenly to allow the flange to find its seat.
6 Insert the remaining bolts and pull them up in the correct sequence as shown in Fig. 9-9.

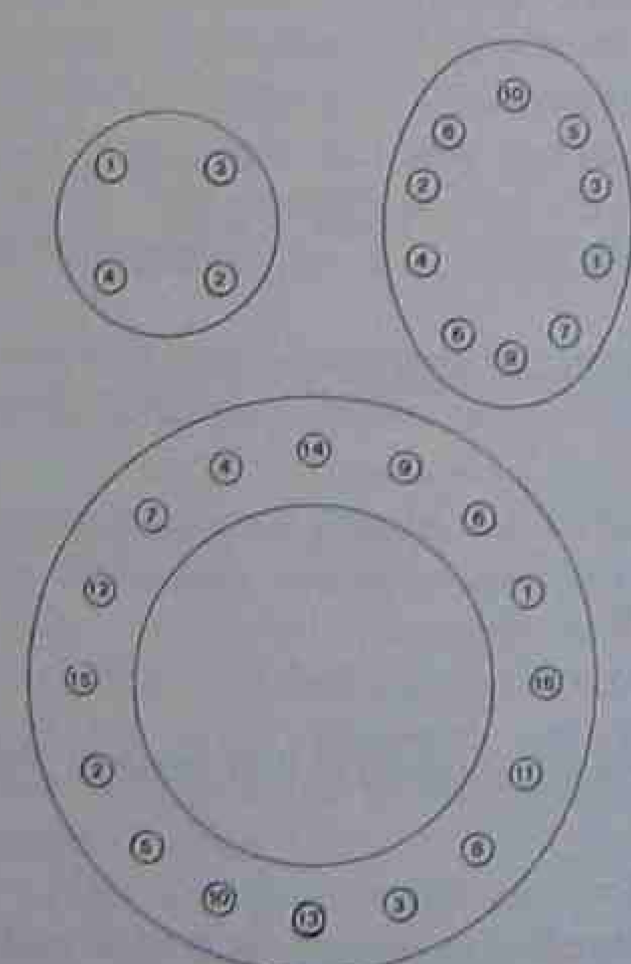


Fig. 9-9 Sequence for tightening bolts.

Do not snug up bolts on the first go round as this can tilt flanges out of parallel. If using an impact wrench, set for about half final torque on the first go round. Ensure that final tightening is uniform.
7 For the best performance in high temperature service make sure that bolts are retightened after 24 hours and then again after one week.

Special considerations for spiral wound gaskets

- Considerable care should be taken in handling these gaskets. Do not remove them from the cardboard backing until they are ready to be installed.
- Rough handling may break the spot welds that hold some of the rings together and the spiral wound section to the support ring.
- A spiral wound gasket should be sized so that the inner and outer wraps are in contact with the flange faces with clearance from the edges as shown below in Fig. 9-10.

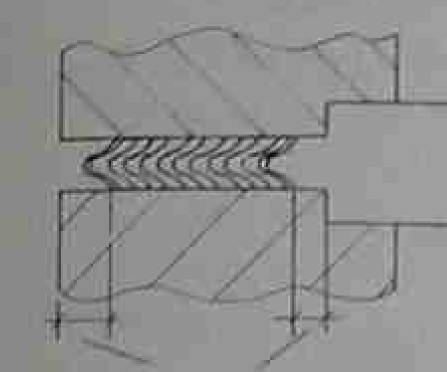


Fig 9-10 Sizing of a spiral wound gasket.

- A spiral wound gasket gives a little as each bolt is tightened and does not have quite the same feel as other gaskets. Hence it is important that the joint is tightened in small steps to ensure the flanges do not tilt and damage the gasket.
- The flanges should be pulled down snug together for a recessed gasket or snug on to the retaining ring to give the correct gasket compression as shown in Fig. 9-11.

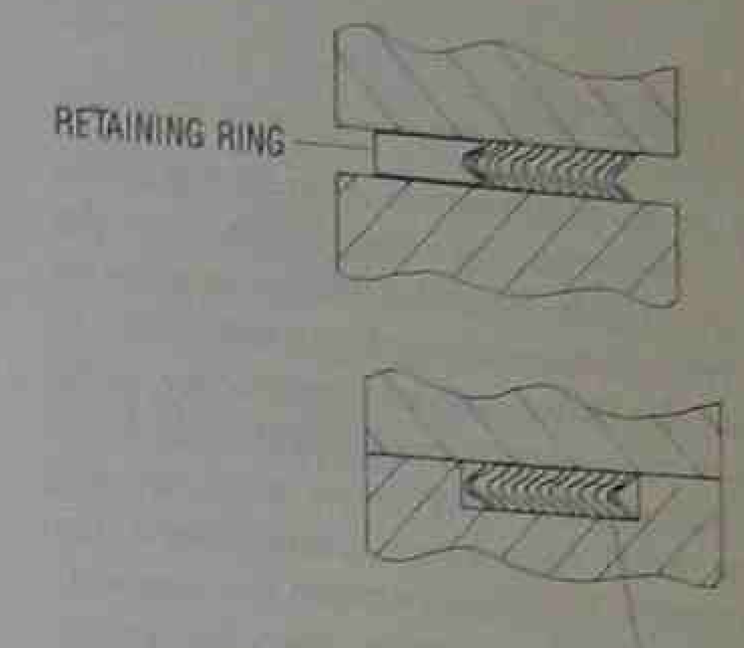


Fig. 9-11 Achieving the correct gasket compression.

9.2 O-RINGS

The elastomeric O-ring is one of the most versatile forms of static sealing arrangement and can also be used as a dynamic seal. The O-ring itself is normally contained within a groove machined into one of the flange faces. The elasticity of the material allows a good seal to be achieved with relatively low contact force.

SEALING PRINCIPLES

A key factor in the performance of an O-ring is the elastomeric properties of the material. An elastomeric material is one which can be repeatedly stretched to twice its normal length and still return to size on release. An O-ring with such properties, once compressed, produces an automatic tightening force and can also adjust to any deformation in the housing as long as the initial compression is maintained. An O-ring will cease to be effective when the material loses its elastomeric properties or when the initial compression of the gasket is lost.

Fig. 9-12 A typical arrangement of a static sealing O-ring.

As can be seen in Fig. 9-12 the O-ring is subjected to an initial compression which is controlled by the size of the ring and the depth of groove. Thus the initial sealing pressure is a result of this compression. When internal pressure is applied through the clearance gap between the flanges this deforms the O-ring further and forces it against the side of the groove, as shown in Fig. 9-13.

As the internal pressure is applied to the O-ring the elastomeric material responds in a manner which is known as self-energizing. This means that the

Fig. 9-13 The O-ring is forced against the side of the groove.

sealing pressure rises in direct response to the increase in internal pressure. However, the total sealing pressure is always higher than the internal pressure by an amount equal to the initial compression of the O-ring:

sealing pressure = internal pressure + initial compression

The maximum internal pressure that can be tolerated by an O-ring will depend on the characteristics of the material and its resistance to extrusion. If the internal pressure is too great then there may be a tendency for the O-ring to extrude into the clearance gap between the flanges as shown in Fig. 9-14.

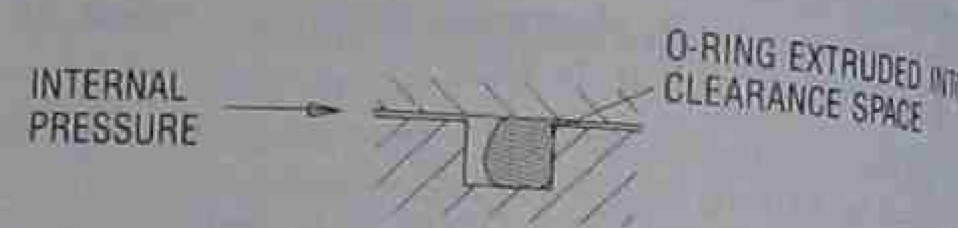


Fig. 9-14 O-ring extrusion.

This is most likely to occur with a static seal if pressure pulsations are sufficient to force the flanges apart. The tendency for an O-ring to extrude can be overcome by the use of a wedge shaped anti-extrusion ring, as shown in Fig. 9-15.

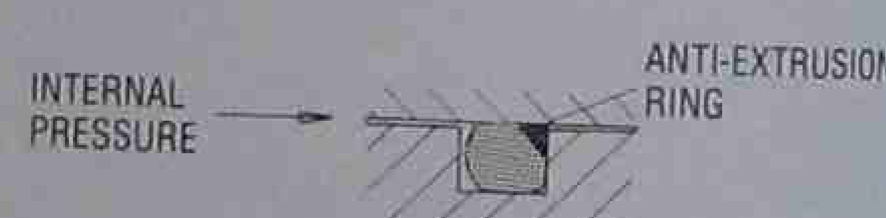


Fig. 9-15 Using an anti-extrusion ring.

MAINTENANCE PRACTICES

The most important concern when handling O-rings is to protect them from damage and to ensure that the correct loading is applied. The following general considerations should be taken into account.

- Ensure that the correct size of seal ring is used in relation to the size of groove.
- Make sure that grooves and recesses are clean and free from sharp edges and burrs.

- Care should be taken to ensure that the ring sits correctly in the groove and cannot get pinched between the flange faces.
- Flange faces should be pulled down evenly and to the correct pre-load recommended by the manufacturer.

FAILURE PATTERNS

The condition of an O-ring after disassembly may provide evidence of the cause of failure.

A ring that has been extruded will show the effects of nibbling along the i.d. as shown in Fig. 9-16.

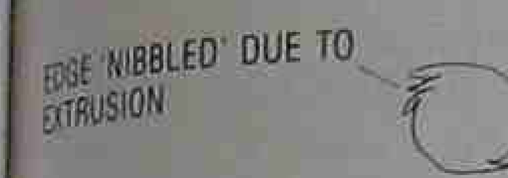


Fig. 9-16 Evidence of extrusion.

Damage caused to O-rings during assembly will usually be evident as nicks or cuts or possibly as twisting.

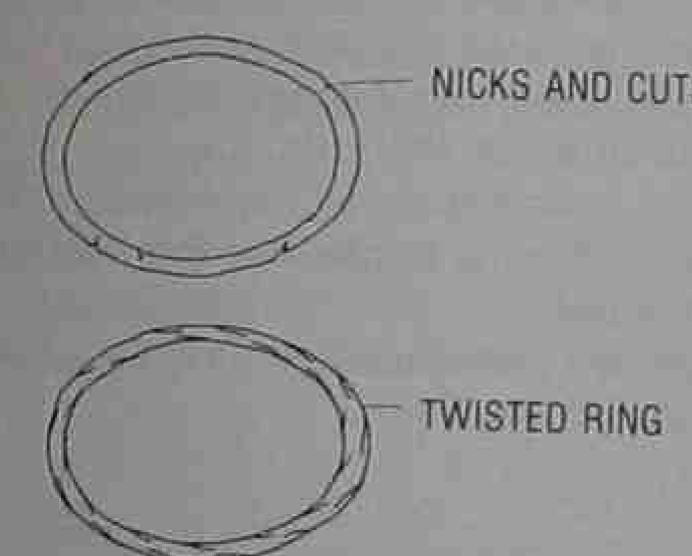


Fig. 9-17 Evidence of damage during assembly.

An overheated O-ring will lose its elasticity and go hard and crack whereas a ring exposed to the wrong chemicals will swell and distort. See Fig. 9-18.

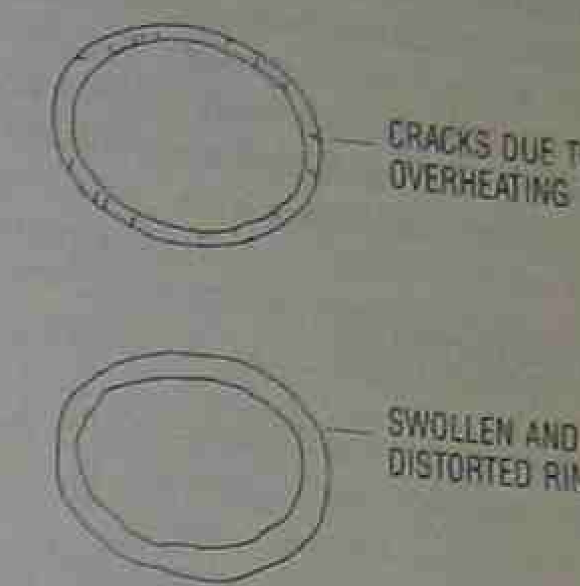


Fig. 9-18 O-ring deterioration.

9.3 SEALANTS

Sealants or liquid gaskets are specially formulated to contain liquids and gases under pressure and can be used as an alternative to gaskets and O-rings in operating conditions that are not too severe. They are relatively inexpensive and are available with a wide range of chemical resistance.

SEALING PRINCIPLES

A sealant performs quite differently from a gasket and allows metal-to-metal contact of the two opposing surfaces. Sealant materials are high viscosity liquids with adhesive properties that fill the voids caused by the surface asperities of the faces as shown in Fig. 9-19.

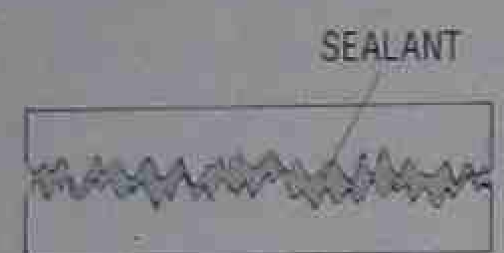


Fig. 9-19 Sealants fill the voids caused by surface asperities.

The fact that a sealant allows metal-to-metal contact to occur means that there is less likelihood of the joint working loose because of the relaxation of conventional gasket material. Once the joint is tight it should remain so and not require readjustment.

The use of a sealant can also reduce or eliminate the machining operations required to produce the degree of surface finish required for gaskets. Sealant materials have sufficient tackiness and viscosity to ensure they fill the voids between the surfaces if they are properly applied.

MATERIALS

The properties of sealant materials can be classified according to their hardening properties. Three basic types are available.

Non-hardening These are mastic type materials with limited adhesive properties. They contain plasticizers to make sure they remain soft.

Hardening-flexible There are various materials available, including neoprene, butyl, acrylic, silicone and polyurethane, that contain curing or setting adhesives but remain flexible when cured.

Hardening-rigid Epoxy, polyester and other resin compounds provide a seal that is rigid when cured.

The majority of sealants used fall into the category of hardening-flexible and common examples include the polyester urethanes used as general jointing compounds and silicone materials used as form-in-place gasket materials.

MAINTENANCE PRACTICES

Because sealants are invariably supplied as a proprietary brand the instructions for application are usually provided by the manufacturer. The important consideration from a maintenance point of view is to ensure that the correct type of sealant is used and that it is correctly applied.

There are several important points to remember in the application of sealants:

- Ensure that surfaces are prepared in the manner prescribed. Dirty or greasy surfaces usually interfere with the performance of the sealant.
- Ensure that the application of the sealant is even and, if recommended, applied to both joint faces. For other than pipe threads, make sure that joint faces are brought together square and that no dirt is allowed to enter the joint.
- Check the curing time of the sealant and make sure that no load is applied to the joint until the sealant has set.
- Do not use old materials that have exceeded their recommended shelf life.

9.4 MECHANICAL END FACE SEALS

Since their development as coolant system seals for the automotive industry, mechanical seals have now become one of the most widely used types of dynamic seal and have undergone considerable technological development. The range of conditions under which a mechanical seal can operate has been significantly increased by the use of new materials and innovative design features. Although relatively simple in construction, they involve complex design features and a high degree of precision in manufacture that necessitates high quality maintenance.

Mechanical seals have been specifically designed to prevent leakage of fluids from between rotating shafts and their housings and as such have replaced the use of soft packings in many situations.

PRINCIPLES OF OPERATION

The primary sealing function of a mechanical seal is achieved by two sealing rings with contacting faces, one of which rotates with the shaft and the other which is fixed in the housing. The contacting faces are lapped to provide an adequate seal and are held together by an axial force created by a mechanical device, such as a spring, and by the hydraulic force of the fluid contained.

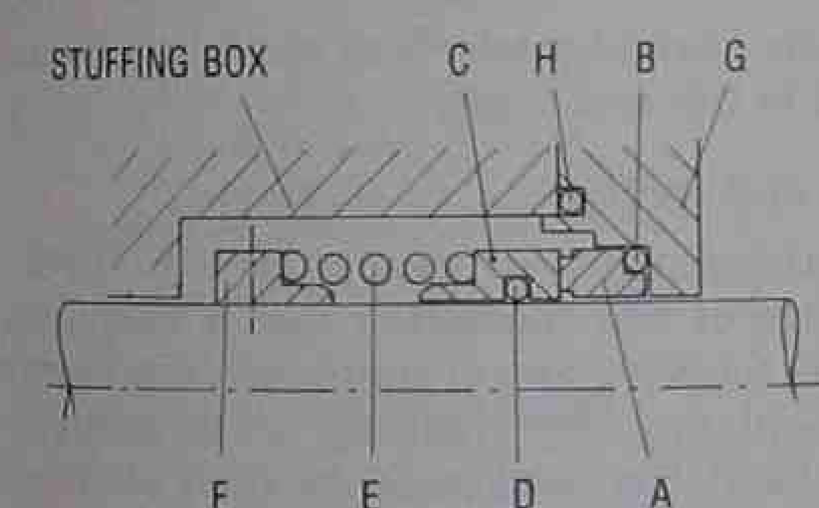


Fig. 9-20 A typical mechanical seal.

It should be noted that there are two secondary seals within a mechanical seal both of which are essentially static in nature and usually involve the use of O-rings. A seal has to be provided between the rotating face and the shaft and another between the stationary face and the seal plate as shown in Fig. 9-20.

The essential components of the mechanical seal are:

A Stationary seal ring This is usually made of carbon and is fitted into a machined recess in the seal plate.

B Stationary seal ring seal This is usually an O-ring which prevents leakage of fluid between the stationary seal ring and the seal plate.

C Rotating seal ring This is locked to the rotating shaft and forced against the stationary seal ring by the combined effort of the spring and the hydraulic pressure of the fluid. Common materials include stainless steel and tungsten carbide.

D Rotating seal ring seal This may also be an O-ring and prevents leakage between the rotating ring and the shaft. It also allows some freedom of movement of the rotating ring to ensure that full face contact of the seal faces is maintained. An alternative to the use of O-rings is the use of a PTFE wedge as shown in Fig. 9-21.

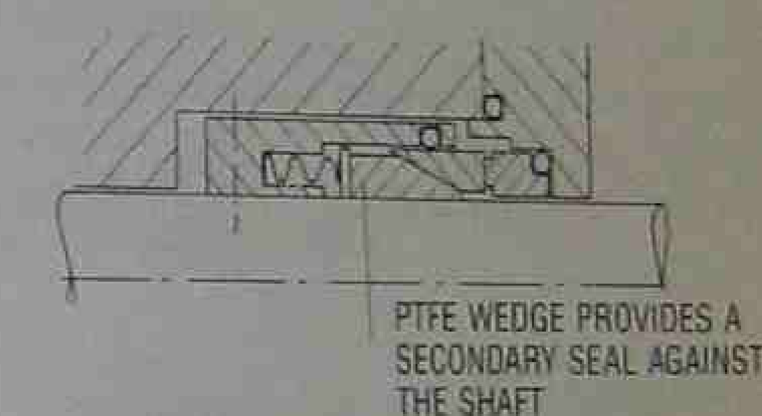


Fig. 9-21 Using a PTFE wedge to prevent leakage between the rotating ring and the shaft.

E Spring A single coil spring is often provided as a means of loading the seal faces.

F Thrust collar This is attached to the shaft by a grub screw and provides a means of driving the spring and a stop against which the thrust is carried.

G Seal plate Comparable to the gland plate in a packed gland. This is the stuffing box cover plate which is also used to carry the stationary seal ring.

H Seal plate seal The seal plate gasket can also be considered as a secondary sealing point.

In addition to the physical characteristics of a seal there are a number of other features that should be understood.

- The contacting seal faces are lapped to precise limits to ensure that they are flat and are supplied in matching pairs.
- The primary sealing faces rely on a supply of lubricating fluid which may be the fluid contained

by the seal or may be a secondary fluid supplied from an external source.

- The combination of mechanical and hydraulic loading on the seal faces can be varied by changing the hydraulic balance of the seal. In the seal arrangement shown in Fig. 9-20 the seal faces are subject to the full effect of the stuffing box pressure. This is shown more clearly in the simplified diagram Fig. 9-22.

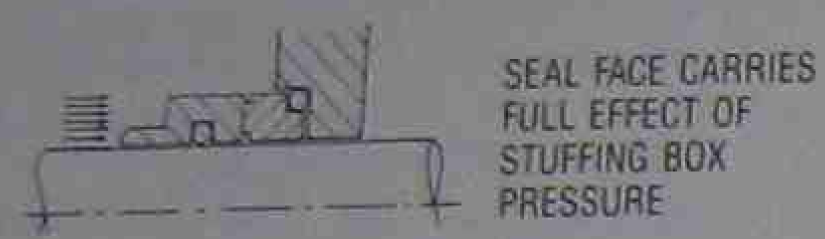


Fig. 9-22 Effect of stuffing box pressure on seal face.

By changing the geometry of the rotating seal ring it is possible to achieve either partial or full balance of the hydraulic pressure as shown in Fig. 9-23.

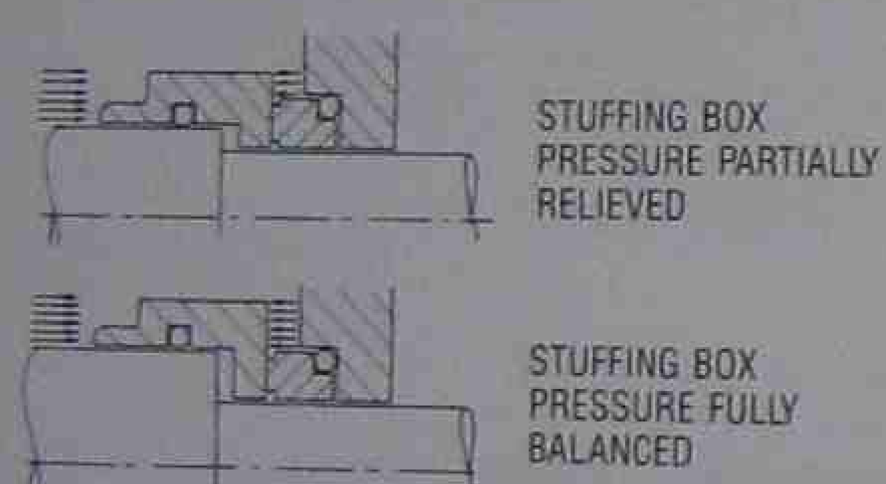


Fig. 9-23 Principles of hydraulic balance.

A seal incorporating this feature is called a **balanced seal** while one without it is referred to as an **unbalanced seal**. Balancing becomes particulaly important when stuffing box pressures are high and are likely to cause excessive loading of seal faces and hence overheating and wear. A balanced seal requires either a step in the shaft or the inclusion of a shaft sleeve in order to provide the necessary offset.

MATERIALS

A range of materials is used for mechanical seals depending on the conditions under which they operate.

Seal faces are available in various combinations including the following:

stainless steel/carbon
lead bronze/carbon
stellite/carbon
chrome oxide/carbon
ceramic/carbon
tungsten carbide/tungsten carbide
tungsten carbide/carbon

The carbon used for seal faces is used in composition form usually with either a metal or a resin filler.

Other seal components such as springs, thrust collar, sleees, etc., are commonly made from stainless steel unless special corrosion resistant properties are required, in which case materials such as Hastelloy or even titanium may be used.

Secondary seal materials are selected on the basis of compatibility with the fluid sealed and the temperature conditions. Synthetic rubbers and PTFE are the most common materials used.

TYPES AND ARRANGEMENTS

There is a range of options available in the design of mechanical seals creating a wide variety of possible arrangements. In addition to the choice between a balanced and an unbalanced seal and the range of materials outlined above, the maintenance technician should be aware of the following alternatives available from most manufacturers.

Mechanical loading

There are several methods of providing mechanical loading to the seal faces.

Single coil spring

This arrangement is shown in Fig. 9-20. The advantage of this method is that it eliminates the necessity for drive pins or similar devices to drive the rotary seal ring. By choosing a coil spring of the correct hand, the rotation of the shaft can cause the spring to tighten against the neck of the rotary seal ring at one end, and the thrust collar at the other, as

Fig. 9-24 Single coil spring tightens with rotation.

shown in Fig. 9-24, providing a positive drive. This helps to overcome the problem of side thrust and tilting moments associated with drive pins.

Multiple coil spring

A single coil spring can be replaced by a number of smaller springs distributed around the shaft.

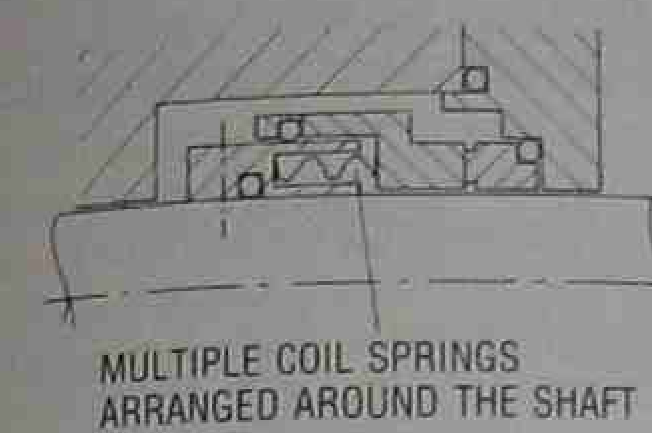


Fig. 9-25 Multiple coil spring.

The advantages of such an arrangement are that a more even distribution of loading on the seal faces can be achieved and the length of the seal assembly can be reduced. They are, however, more susceptible to blockage and interference from solids or sludge.

Bellows

The common alternative to the use of springs is to use a set of metal bellows.

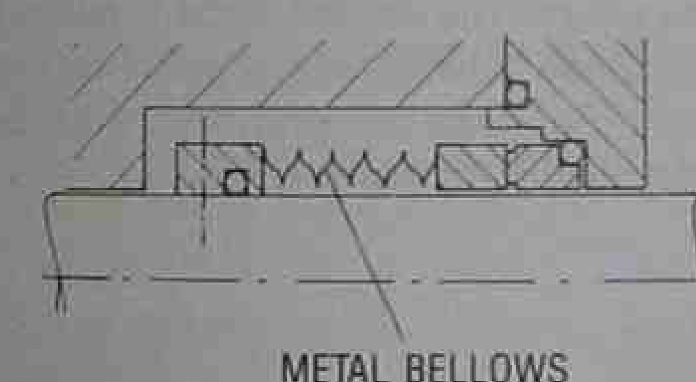


Fig. 9-26 Metal bellows.

One end of the bellows is welded to the thrust collar which in this case is also machined to accommodate a secondary seal. The other end of the bellows is welded to the rotating seal face which is held clear of the shaft by the rigidity of the bellows and is not required to carry a secondary seal.

The advantage of this type of arrangement is that there is no sliding elastomeric seal under the rotating seal ring, and therefore drag or hang-up on the shaft and also wear of the shaft or sleeve are eliminated.

This feature also enables a bellows seal to operate at higher temperatures than a spring loaded seal.

Bellows made from PTFE can also be used to protect the moving parts of a seal, such as the springs, from contact with the sealed fluid thus preventing corrosion. This arrangement is demonstrated in Fig. 9-27.

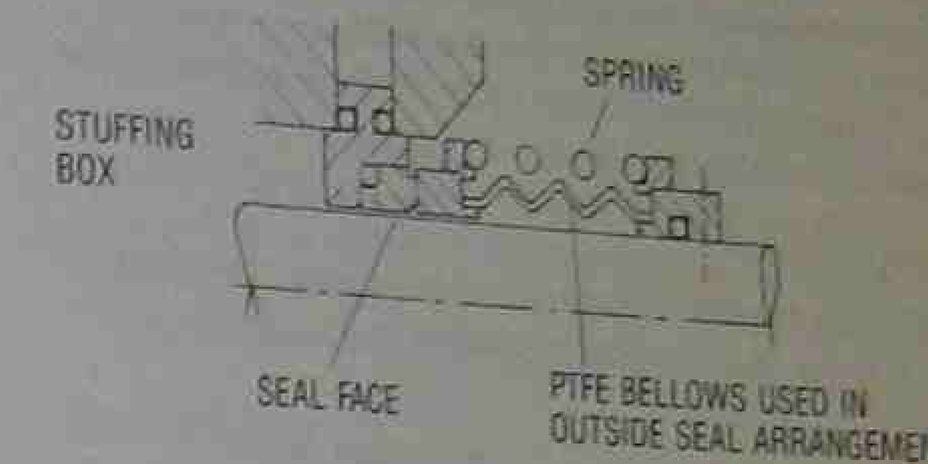


Fig. 9-27 PTFE bellows can protect the moving parts of the seal from contact with the sealed fluid.

Seal location

The most conventional arrangement is to locate the seal inside the stuffing box as shown in Fig. 9-20. It is also possible to locate the seal outside the stuffing box as shown in Fig. 9-28.

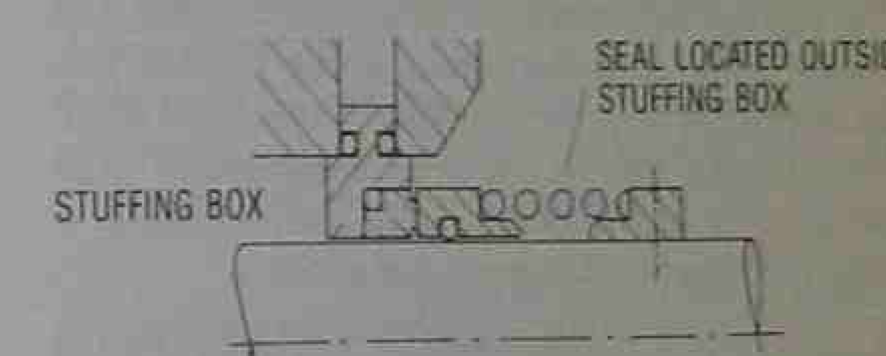


Fig. 9-28 Location of the seal outside the stuffing box.

The advantage of this arrangement is that the sealed fluid is shielded from the moving parts of the seal and corrosion can be eliminated. The major drawback is that the hydraulic pressure now tends to force the seal faces apart. This makes the seal particularly vulnerable to sudden surges of pressure that may open up the seal faces.

Seal combinations

Although the majority of sealing applications can be satisfied by a single seal, in some circumstances a double seal arrangement can be used in order to provide extra protection. This type of arrangement is required when the sealed fluid is not capable of providing the necessary properties to lubricate the seals or when it contains solids or abrasives that could affect the seal faces. The use of a double seal arrangement allows a secondary fluid to be introduced into the space between the two seals. The fluid may be water or a neutral lubricant circulated by an independent system. It is important that the fluid chosen is compatible with the sealed fluid so that contamination can be avoided.

There are a number of ways in which a double seal can be arranged.

Back-to-back

This arrangement consists of two seals mounted in the same stuffing box as shown in Fig. 9-29.

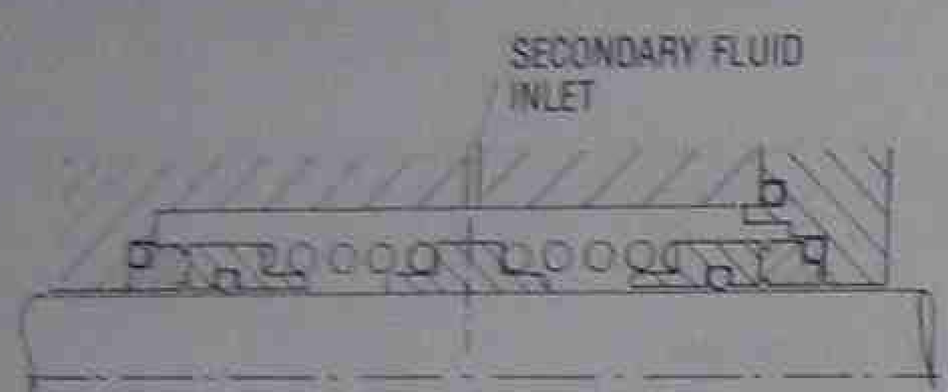


Fig. 9-29 Back-to-back arrangement.

To ensure lubrication of the inboard seal the pressure of the secondary fluid in the stuffing box must be greater than the pressure of the sealed fluid. The example given shows unbalanced seal arrangements but balanced seals can be used if pressure differentials become too great.

Tandem

An alternative to the back-to-back arrangement is the tandem double seal which consists of two seals mounted in the same direction, one inside the stuffing box and the other in a cavity created by a second seal plate as shown in Fig. 9-30.

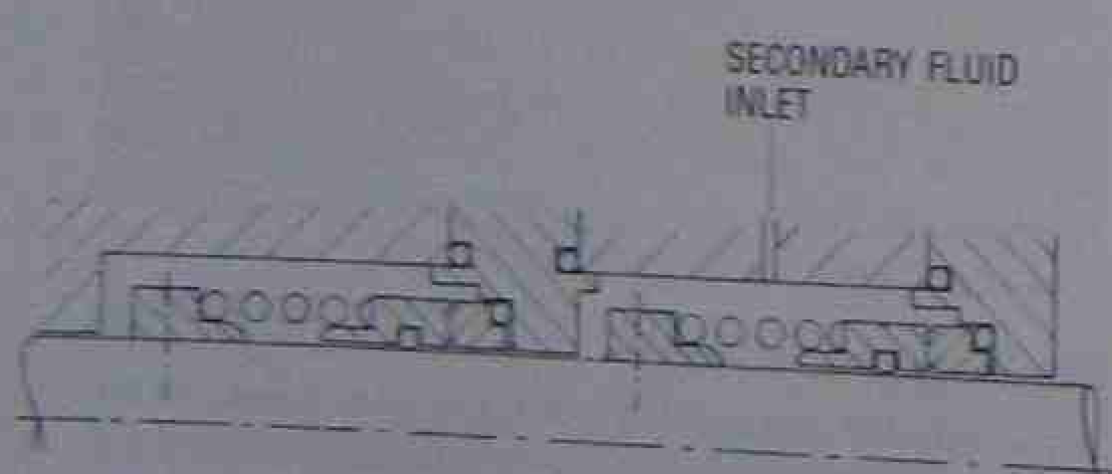


Fig. 9-30 Double seal arranged in tandem.

The inboard seal contains the sealed fluid in the conventional stuffing box but should leakage occur this will be contained by the outer seal which is lubricated by a secondary fluid.

Inside-outside

Where space limitations prevent the use of a tandem seal an arrangement in which a conventional inside seal is backed up by an outside seal can be used. A secondary fluid can be introduced into the cavity between the two sets of seal faces shown in Fig. 9-31.

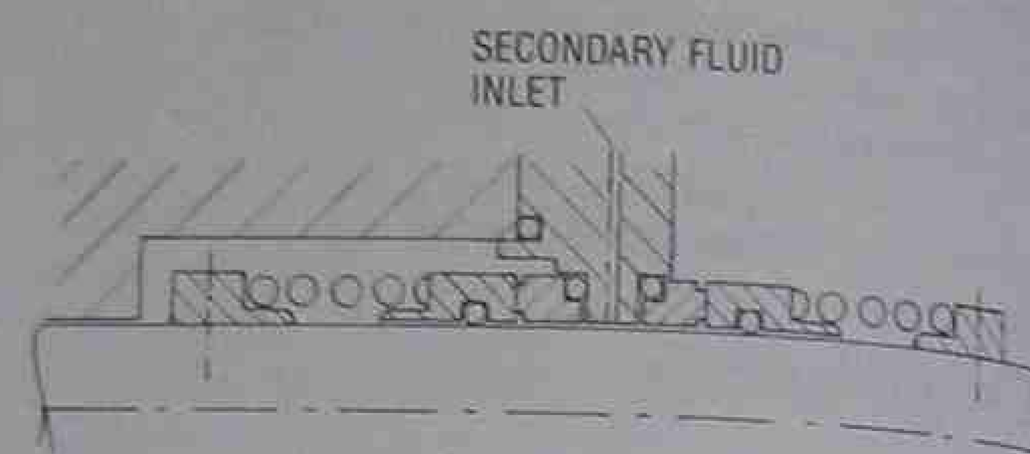


Fig. 9-31 Inside-outside arrangement.

The secondary fluid may be supplied at a higher pressure than the sealed fluid, in which case it lubricates both sets of seal faces. Alternatively, it may be supplied at a lower pressure in which case the outside seal merely acts as a back-up.

Seal and bushing

In situations where the sealed fluid is not hazardous and has adequate lubricating properties but where some control of leakage is required a throttling bush may provide sufficient protection.

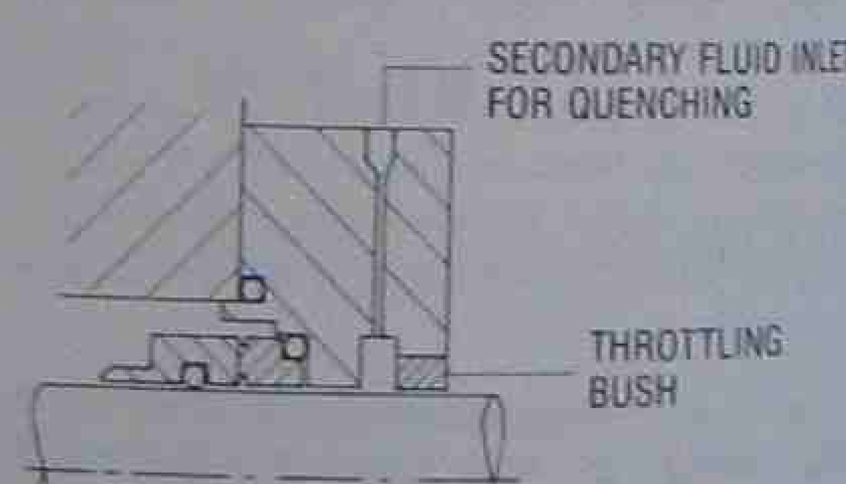


Fig. 9-32 A throttling bush.

This arrangement can also be used to provide quenching to the seal faces by a secondary fluid supplied via the seal plate.

Combination seal arrangements of the types outlined above provide a number of advantages over single seal arrangements:

- They can handle extreme pressure differentials by creating a progressive step-down in pressure across the two seals.
- They provide a buffer zone against the escape of toxic or hazardous fluids.
- They are capable of sealing against a vacuum.
- They are capable of sealing a gas or a liquid which does not possess adequate lubricating properties.
- Fluids containing solids or abrasives can be kept away from the seal faces.
- The hazards associated with the leakage of a high temperature process fluid can be reduced by circulating a lower temperature secondary fluid.

ENVIRONMENTAL CONTROL

An important factor in determining seal performance is the nature of the environmental conditions surrounding the seal face. The factors that affect performance are temperature, corrosion and contamination and these can be controlled by various combinations of flushing and quenching the area around the seal faces.

Flushing

Lubrication and cooling of the seal faces may be accomplished by directing a continuous flow of liquid to the seal interface as shown in Fig. 9-33.

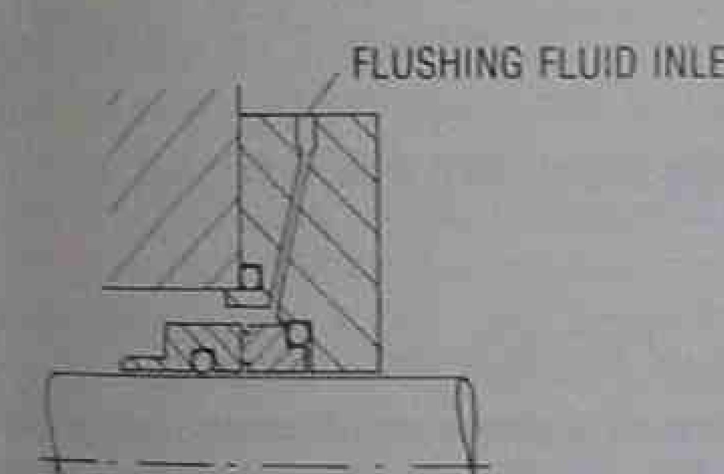


Fig. 9-33 A continuous flow of liquid is directed to the seal face.

The liquid used for flushing may be supplied directly from the pump discharge or may be supplied from an external source and is normally directed into the seal area via a tapping in the seal plate or via the lantern ring connection in the stuffing box. A complete flushing system may need to incorporate a cooling device and some means of solids removal such as a cyclone separator.

The advantages of flushing can be summarised as follows:

- Flushing overcomes the tendency of highly volatile fluids to vaporise around the seal interface.
- Fluids that operate close to their melting point can be prevented from crystallising.
- Seal operating temperatures are reduced.
- Flushing prevents solids from accumulating around the seal interface.
- If an external fluid is used for flushing at a pressure above that of the sealed fluid, then the sealed fluid can be eliminated from the stuffing box. This is useful in the handling of corrosive or toxic fluids as long as the flushing liquid is compatible with the sealed fluid.

Quenching

The creation of a buffer zone behind the seal face by the addition of a backing gland or bushing allows a quenching fluid, which is usually supplied from an external source, to be directed into that zone as shown in Fig. 9-32.

The fluid used for quenching should be clean and should not contaminate the interface area.

The advantages of quenching are:

- Leakage of toxic or hazardous fluids can be prevented from reaching the atmosphere.
- Quenching with a fluid above the temperature of the sealed fluid can prevent crystallisation at the seal interface.
- As well as helping to cool the seal interface, quenching minimises the transfer of heat along the shaft to the pump bearings.

MAINTENANCE PRACTICES

General

A number of general considerations should be taken into account when working on mechanical seals.

- Like bearings, mechanical seals are manufactured to fine tolerances and high surface finish and therefore cleanliness is of the utmost importance.
- The key to good seal performance is to ensure that seal faces are square, correctly loaded and properly lubricated.
- A seal should never be run dry, not even for a few seconds.
- Seal rings are lapped together as pairs and a used ring should never be mated with an unused ring.

Assembly

Before assembly is begun the following points should be considered.

- Check the manufacturer's drawings and instructions to ensure that the assembly procedures are correctly understood.
- Make sure that the necessary tools and equipment are available.
- Select a working environment that is clean and dust free and where adequate lighting exists.
- Seal faces should be left in their protective plastic cover until ready for fitting.
- Check that the shaft is clean and free from burrs and other damage and also check that the stuffing box is clean and that flange faces are free from damage.
- To guarantee the optimum operation of the seal, it may be necessary to check shaft run-out, deflection and stuffing box geometry.

The following checks are recommended:

Shaft run-out

Mount dial indicators as shown in Fig. 9-34 to check shaft run-out and whether or not the shaft is bent.

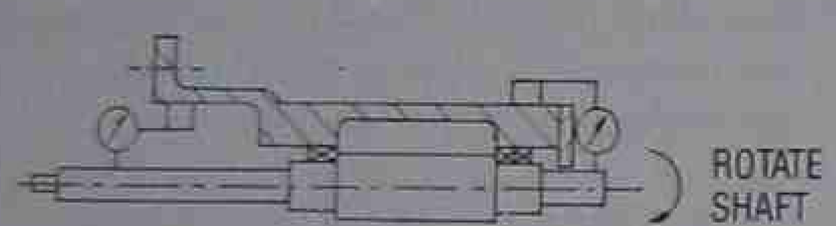


Fig. 9-34 Checking shaft run-out.

Shaft deflection

By lifting the shaft and observing the dial gauge mounted as shown in Fig. 9-35 a check can be made on the amount of shaft deflection that exists due to bearing wear.

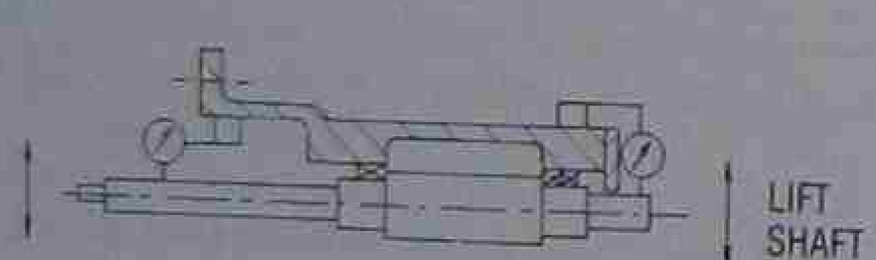


Fig. 9-35 Checking shaft deflection from the dial gauge.

Shaft float

Measure end play by setting up the dial gauge against a shoulder. The measurement recorded should be checked against the manufacturer's recommendations.

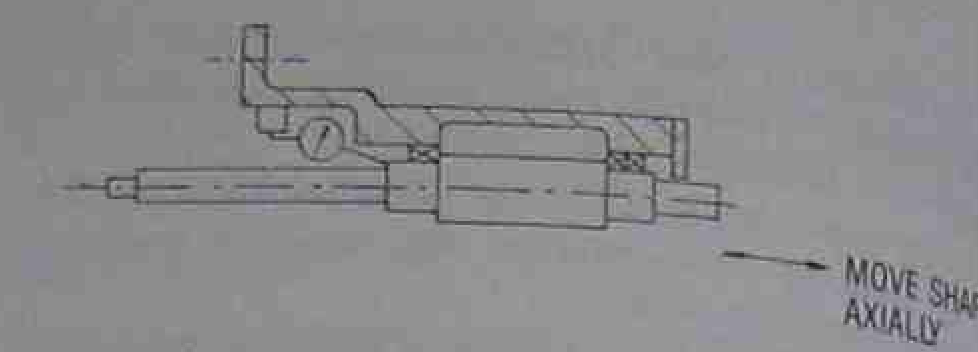


Fig. 9-36 Measuring shaft float.

Sleeve concentricity

If a shaft sleeve is used then this should also be checked for run-out in the same way as the shaft.

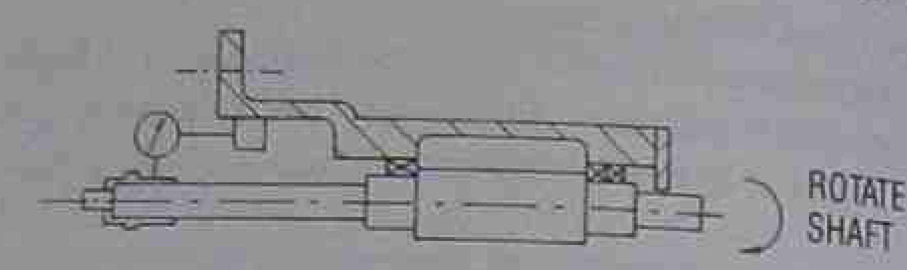


Fig. 9-37 Checking for run-out in the shaft sleeve.

Concentricity of stuffing box bore

A check on the concentricity of the stuffing box bore may be necessary to ensure that the shaft is centralised in the stuffing box. An adapter may need to be made for the indicator and the pump assembled without the seal as shown in Fig. 9-38.

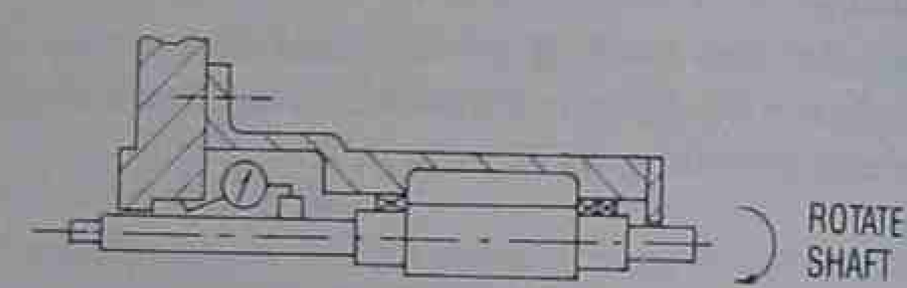


Fig. 9-38 Checking concentricity of the stuffing box bore.

Squareness of stuffing box face

To ensure that seal faces run square a check on the squareness of the stuffing box face can be carried out as shown below. A maximum run-out of 0.075mm (0.003") is recommended. (Fig. 9-39)

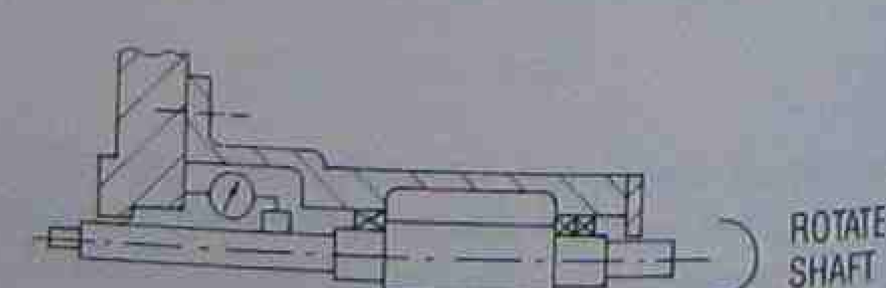


Fig. 9-39 Checking squareness of the stuffing box face.

The assembly procedure itself should be carried out according to the manufacturer's instructions and with consideration given to the following points:

- Shaft shoulders should be chamfered to enable O-rings to be installed without damage.
- If the seal includes a single coil spring make sure that it is the correct hand. The spring should tighten up due to the rotation of the shaft.
- A trace of lubricant may be used to assist the fitting of the rotating seal assembly but no lubricant should be applied to the seal faces.
- To ensure the correct loading of the seal faces the thrust collar must be fastened to the shaft or shaft sleeve in accordance with the dimensions provided by the manufacturer. To provide a reference, a witness mark can be made on the shaft in line with the stuffing box face as shown in Fig. 9-40.

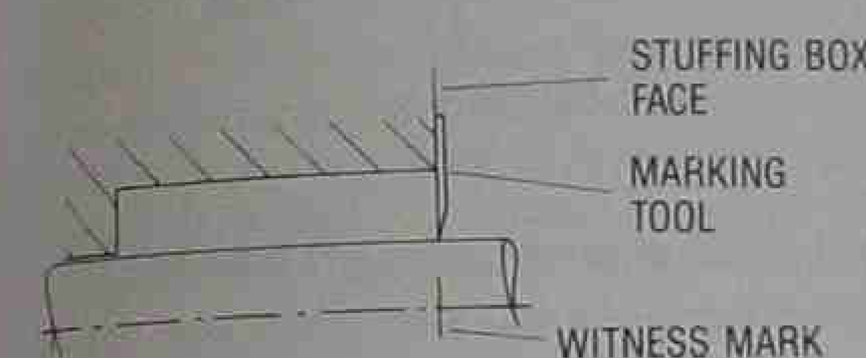


Fig. 9-40 Marking the shaft.

- When fitting the stationary seal ring into the seal plate care should be taken not to twist the ring packing as this may cause damage. A small amount of lubricant on the O-ring will help to push it home square without the need for twisting.
- If flushing connections are provided, and the installation is new or the fluid pumped contains solids, make sure that a strainer is installed in the circulation line to the seal plate.

Start-up

The most important consideration during start-up is that the seal should not start up dry. If the seal is provided with an external flushing connection this must be turned on before the machine is started. Whatever method is used to provide lubrication to the seal faces, steps should be taken to ensure that the stuffing box is flooded before the machine is started. However, only the seal fluid or the secondary flushing fluid should come into contact with the seal faces. Do no apply any oils or other lubricants to the seal faces.

FAILURE PATTERNS

When a seal begins to leak it is often difficult to tell precisely how it is leaking. There are several possible ways in which this may occur and these are shown in Fig. 9-41.

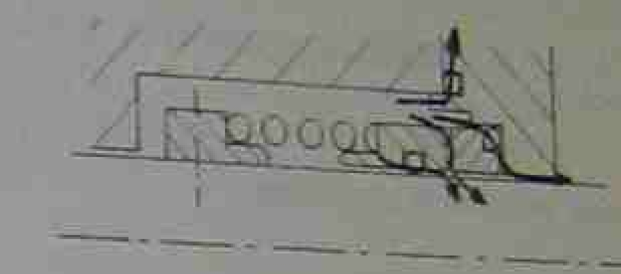


Fig. 9-41 Leakage paths for a mechanical seal.

Leakage of the seal plate gasket will be evident, but it is not so easy to determine whether leakage is occurring across the primary seal faces or the secondary seals.

The majority of seal failures occur due to leakage across the primary faces but careful inspection of all components will often be necessary to make sure that the secondary seals are not to blame.

Seal leakages that occur on start-up are usually due to incorrect assembly such as the fitting of the wrong hand spring, chipped or cracked carbon rings or trapped or damaged O-rings. Inadequate lubrication at start-up may also damage seal faces so that they begin to leak immediately.

There are various symptoms which can be identified by examining the seal components that will assist in determining the cause of failure. The most common causes of seal failure and the effects they produce are as follows.

Vaporisation

When the temperature of the seal interface becomes too great and local boiling occurs, the seal faces pop apart and make a puffing noise.

The stationary carbon face becomes pitted with associated 'comet trailing' and the outside diameter is likely to be chipped. The continual impact of the seal faces as they are blown apart by the vapour causes radial cracks to appear on the rotating face. When the sealed fluid is water, vaporisation may cause the seal faces to blow open and remain open.

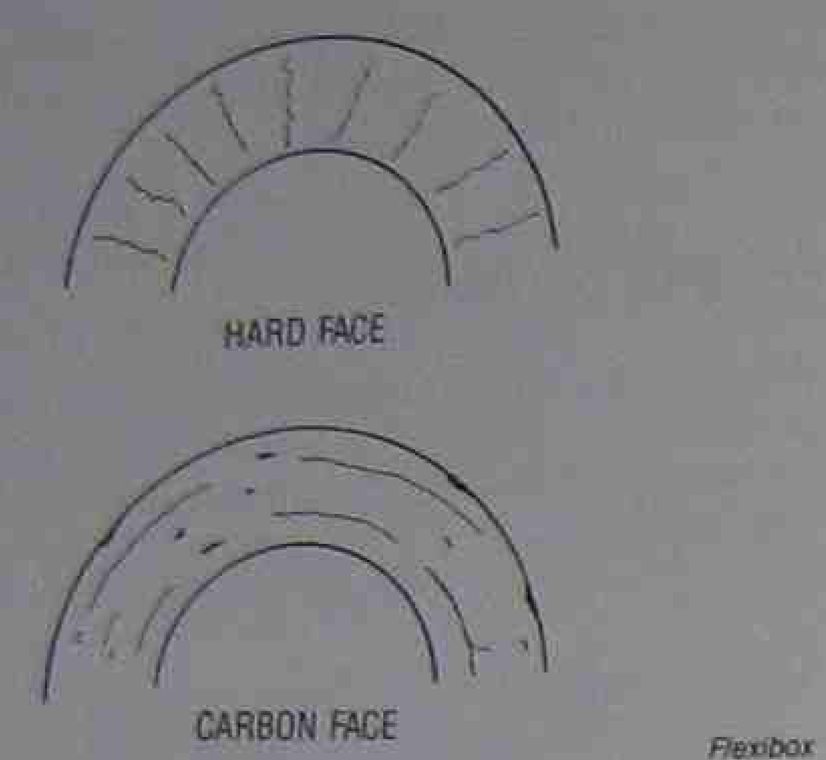


Fig. 9-42 Typical evidence of vaporisation.

Dry running

When the seal faces are inadequately lubricated they are subject to rapid wear in the form of scoring and grooving as shown in Fig. 9-43.

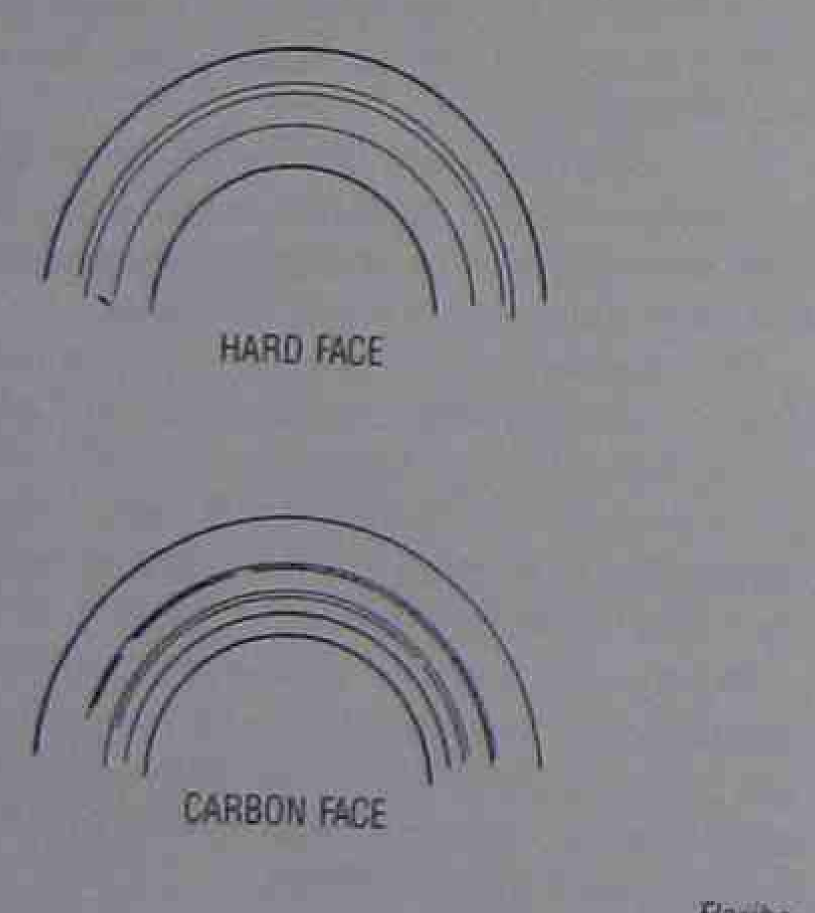


Fig. 9-43 Rapid wear from dry running.

There may also be evidence of overheating in the form of hardening and cracking of the secondary seals if the problem has existed for some time.

Abrasives in sealed fluid

If abrasive matter is present in the sealed fluid this may cause rapid wear of the seal faces and also a solids build-up around the rotating ring secondary seal as shown in Fig. 9-44.

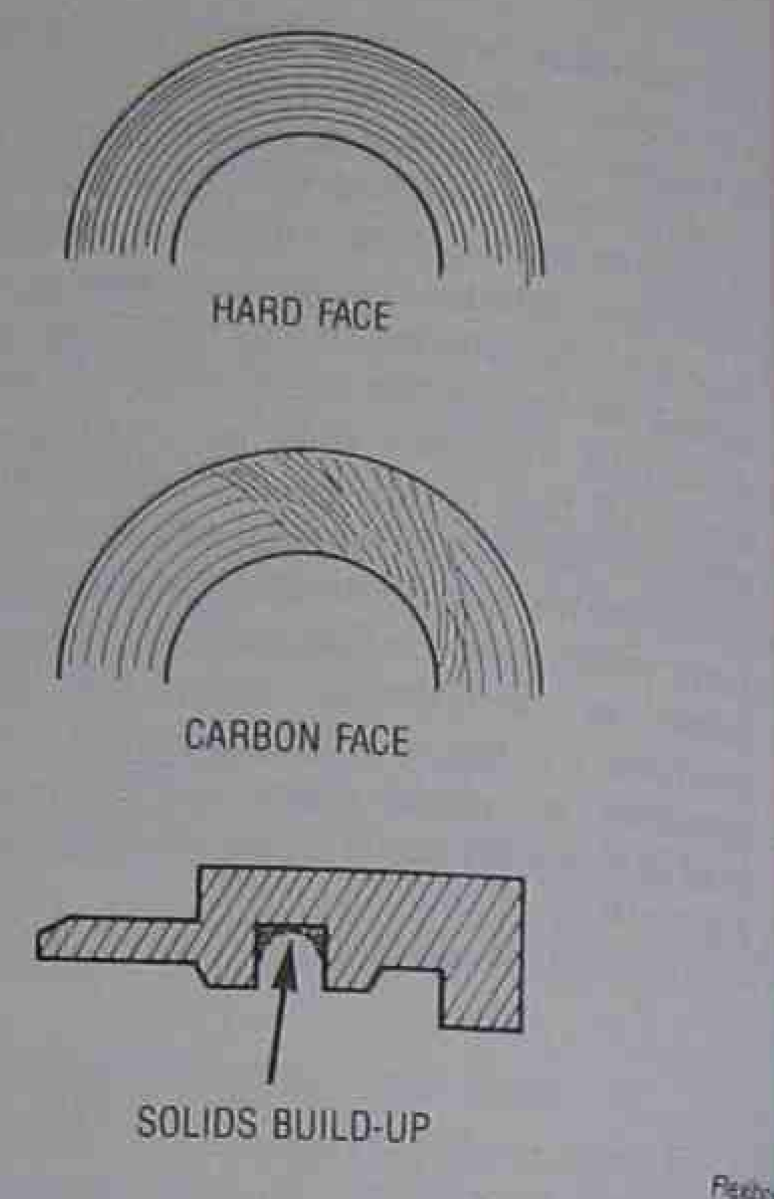


Fig. 9-44 Results of abrasive material in sealed fluid.

The ideal solution to this problem is to exclude the solids from the seal faces by flushing with an external source of clean fluid. Alternatively, wear-resistant seal faces may be required (e.g. tungsten carbide).

Sludging and bonding

These two problems produce very much the same kind of seal face damage. Sludging is associated with the sealing of high viscosity liquids and occurs when the shear stresses between the seal faces exceed the rupture strength of the carbon and tear away the surface of the stationary seal ring. This is especially likely to occur when seal temperatures drop and the viscosity of the interface film increases causing problems on start-up. A similar problem occurs when the fluid actually crystallises while the unit is shut down and forms a temporary bond between the faces. When this is pulled apart on start up damage to the carbon face results.

There are a number of possible solutions to these problems depending on the actual circumstances. Assuming that the liquid viscosity lies within the capabilities of the seal then the important considerations are whether there is adequate circulation of fluid around the seal interface area and whether the temperature of the seal can be prevented from falling below the levels at which problems occur. Fluid circulation will depend to a

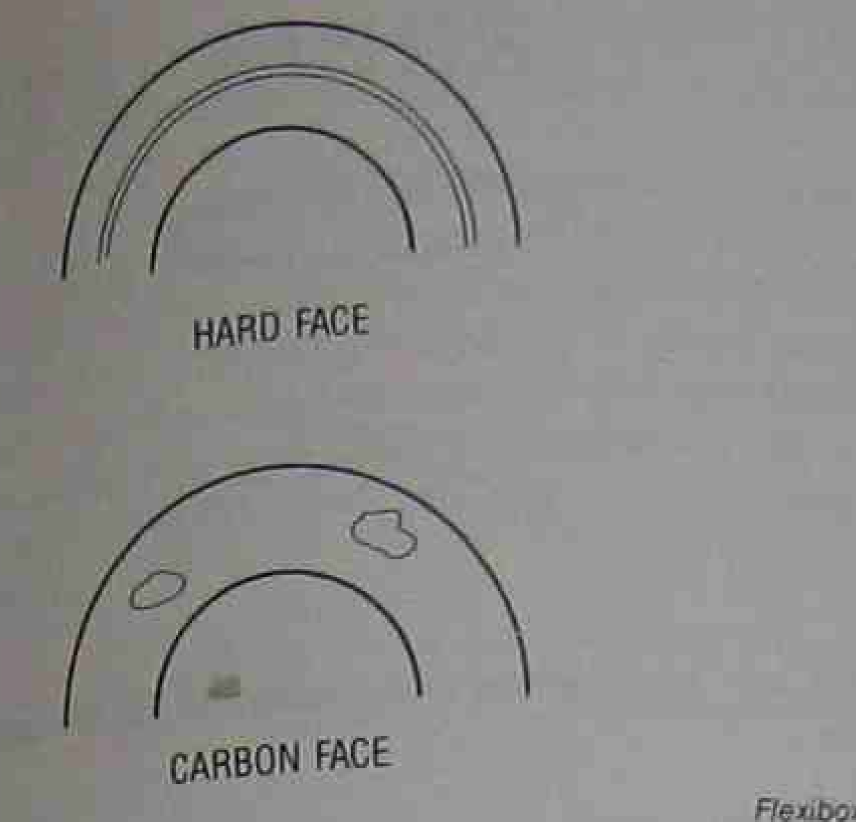


Fig. 9-45 Typical evidence of sludging and bonding.

certain extent on the pressure available in the circulation system and this may need to be boosted if it is not adequate. Some means of preheating the seal before start-up, either by steam tracing, direct injection to the stuffing box or steam circulation through the seal plate may be required to avoid crystallisation.

Distortion

Under some circumstances it is possible for the seal faces to distort. This may be caused by uneven pressure of the drive spring or incorrect assembly. In some cases incorrect storage may also cause the same problem. The results of distortion are uneven running marks on the seal faces as shown in Fig. 9-46.

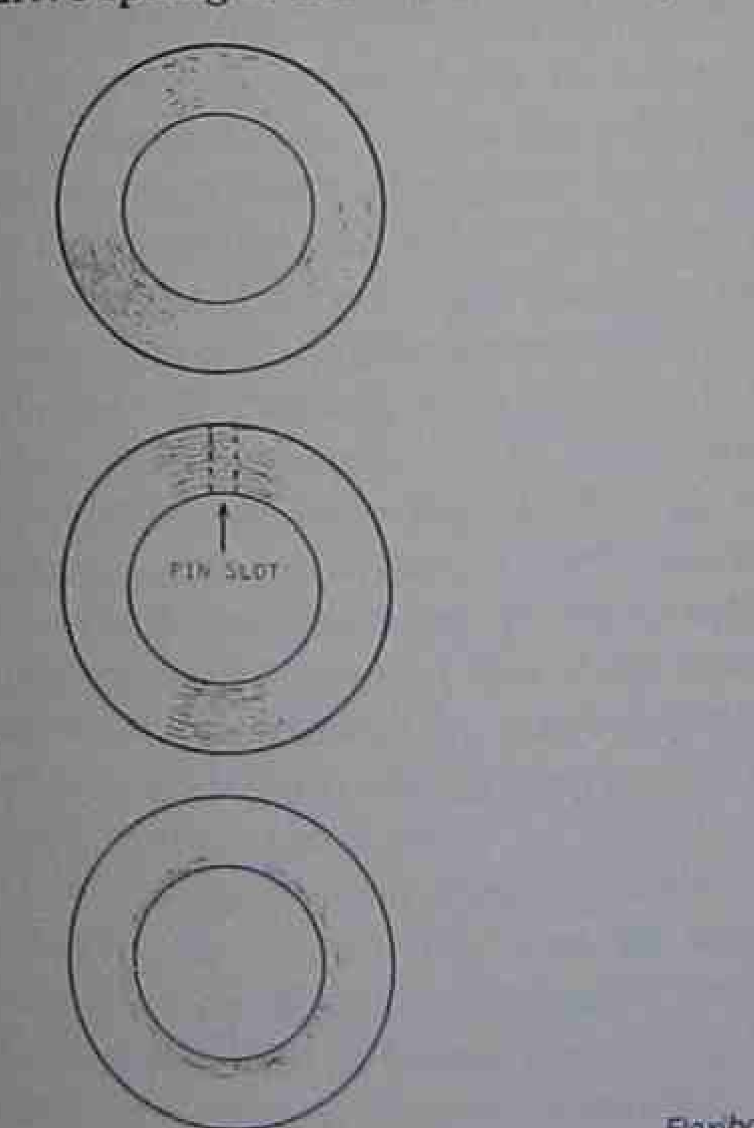


Fig. 9-46 Results of distortion.

The problem of distortion may be overcome by re-lapping the seal faces unless the degree of distortion is too great, in which case new seal rings may be required.

Coking

When the sealed fluid is a hydrocarbon at high temperature, any slight, even minute, leakage past the seal tends to carbonise and cause the secondary seal under the rotating ring to jam up. This prevents it from sliding to take up wear at the primary seal face. This is demonstrated in Fig. 9-47.

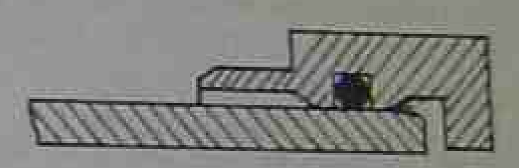


Fig. 9-47 Coking.

The solid build-up of carbonised material on the atmospheric side of the sliding seal eventually jams the rotating seal ring on the shaft or sleeve.

The solution to the coking problem is to provide a steam quench to ensure that the seal temperature stays above that at which coking takes place.

Sleeve damage

It is possible for the sleeve or shaft to become damaged under the rotating ring seal. This, too, prevents the seal ring from sliding along the shaft. The two most likely causes of such damage are vibration and fretting corrosion.

Vibration of the shaft is likely to cause interference between the shaft or sleeve and the underside of the seal ring adjacent to the O-ring groove. This leads to wear and marking of the shaft or sleeve surface as shown in Fig. 9-48.

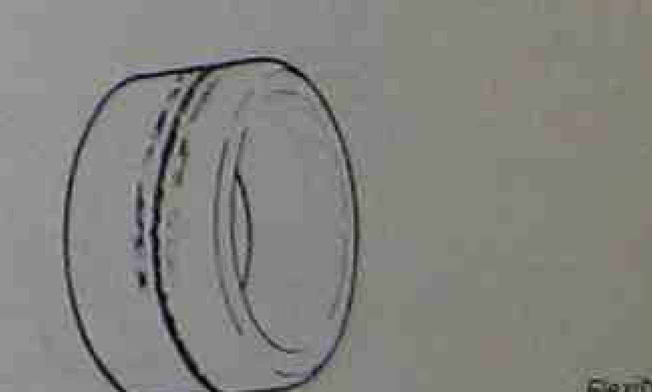


Fig. 9-48 Wear and marking from vibration of the shaft.

When vibration is very slight fretting may occur under the O-ring itself with consequent damage to the shaft or sleeve as shown in Fig. 9-49.

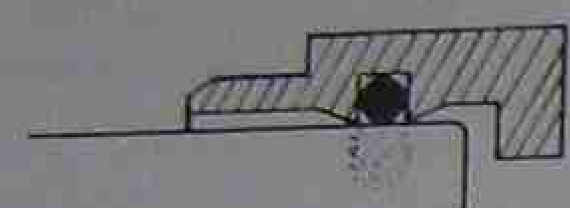


Fig. 9-49 Fretting damage.

Elimination of the source of vibration is the ideal way to solve such problems but if this cannot be achieved totally then hard facing of the sleeve or shaft will help reduce the effects of the problem.

O-ring damage

Damage to the secondary seals can occur in a number of ways. Apart from errors made during assembly it is usually assocated with the conditions of operation. Overheating of O-rings causes hardening and cracking and may indicate either that the O-ring material is not suitable for the duty or that improved cooling of the seal is required.

Extrusion of the O-ring, shown in Fig. 9-50, may be caused by excessive pressure or incorrect clearances.

If the O-ring material is incompatible with the seal fluid then it may appear to be eaten away as shown in Fig. 9-51. The seal manufacturer should be consulted to consider the selection of the seal materials.

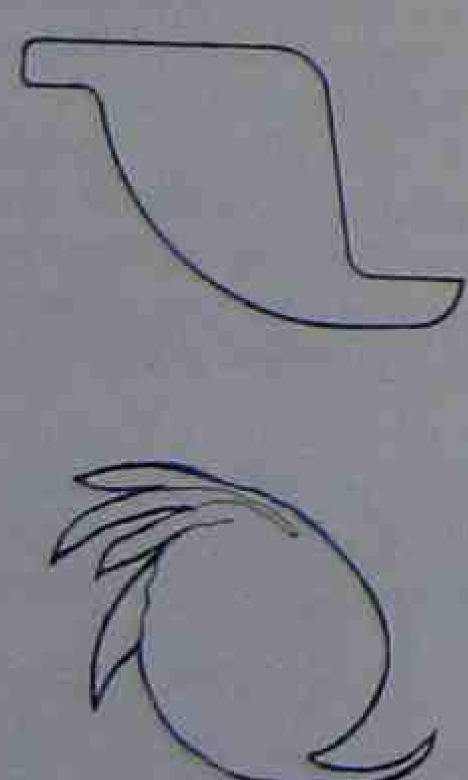


Fig. 9-50 Extrusion of the O-ring.

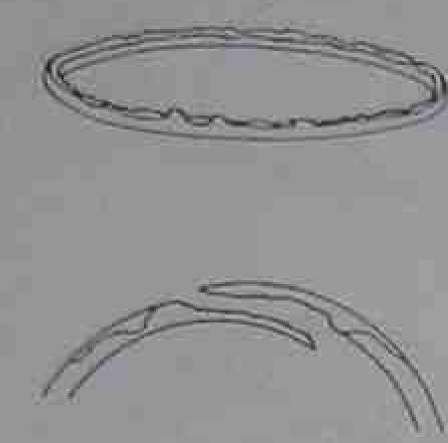


Fig. 9-51 Incompatibility of O-ring material and seal fluid.

Carbon ring erosion

Erosion damage to the carbon ring, shown in Fig. 9.52, may occur due to either the circulation pressure being too high or the presence of abrasive particles in the circulation flow.

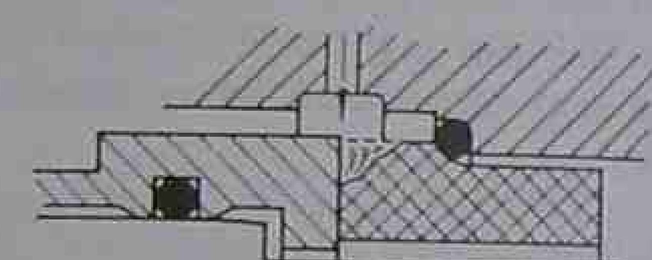


Fig. 9-52 Erosion damage to the carbon ring.

9.5 PACKINGS

The use of packings is one of the oldest approaches to solving dynamic sealing problems. Rings of soft packing material are usually fitted into a stuffing box around the shaft and compressed by a gland ring which forces the packing into contact with the shaft and the stuffing box bore. This method of sealing is also used as a static seal and for reciprocating mechanisms.

SEALING PRINCIPLES

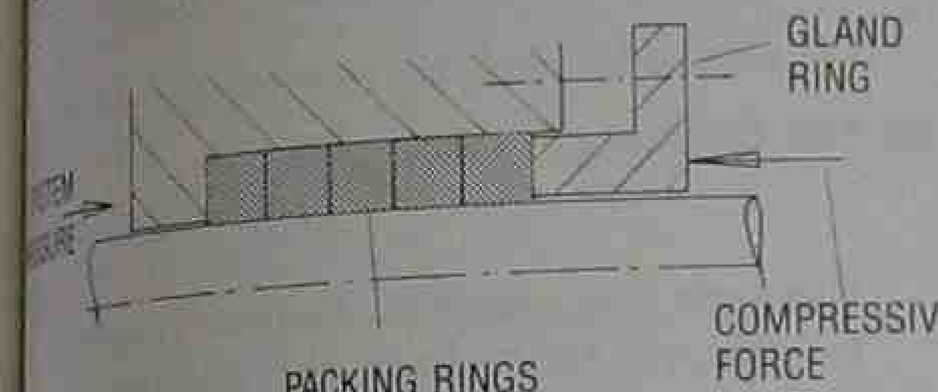


Fig. 9-53 The operating principles of a packed gland.

There are various forms that the packing may take, the most usual being rings of soft woven or braided material that are cut to length and assembled around the shaft inside the stuffing box. A gland ring or follower is pulled up against the stuffing box casing to compress the rings and cause them to expand radially. This radial expansion brings the packing into loaded contact with both the shaft and the stuffing box bore to create the seal. One of the factors affecting the efficiency of the packing is the number of rings. It has been demonstrated by experience that the optimum number of rings is five, due to the manner in which the pressure varies over the packing area. As can be seen from Fig. 9-54, if there are too many rings, the innermost do very little work.

The materials used for packing possess self-lubricating properties which protect the shaft from wear, at least on start-up. However, this lubricant would soon leach out and the packing overheat if it were not supplemented in some way.

In cases where the pump suction pressure is above atmospheric pressure the sealed fluid can be used directly to lubricate the packing by allowing a small amount of controlled leakage through the gland. If the pump suction pressure is below atmospheric pressure, liquid from the pump discharge may be directed into the gland area via a lantern ring which is installed between the rows of packing. If the sealed fluid contains abrasives or is unsuitable for lubrication for some reason a secondary fluid may be directed to the gland area via the lantern ring. These three alternatives are shown in Fig. 9-55.

Whatever arrangement is used it is imperative that the packing is not allowed to run dry. A properly adjusted packed gland should exhibit a slow but steady leakage of liquid from the gland ring.

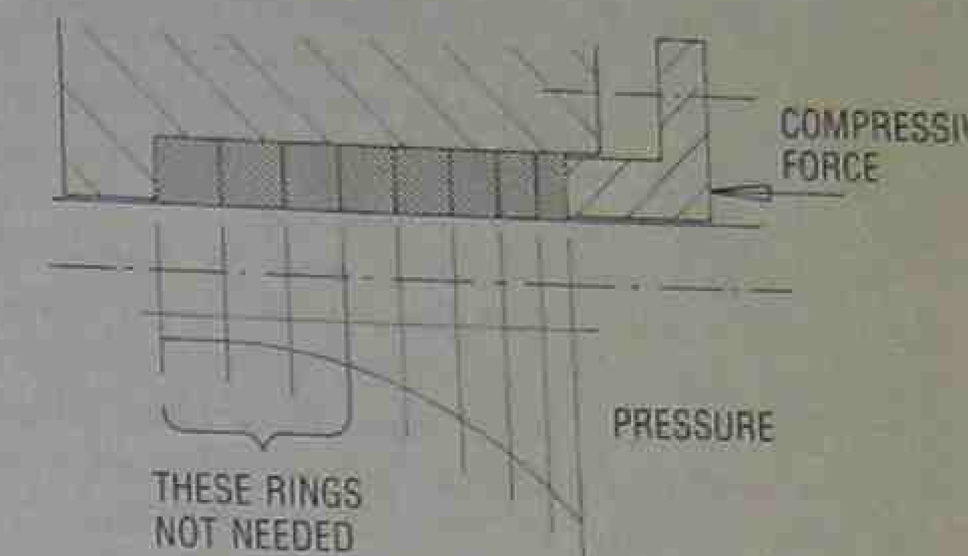


Fig. 9-54 If there are too many rings, the innermost do little work.

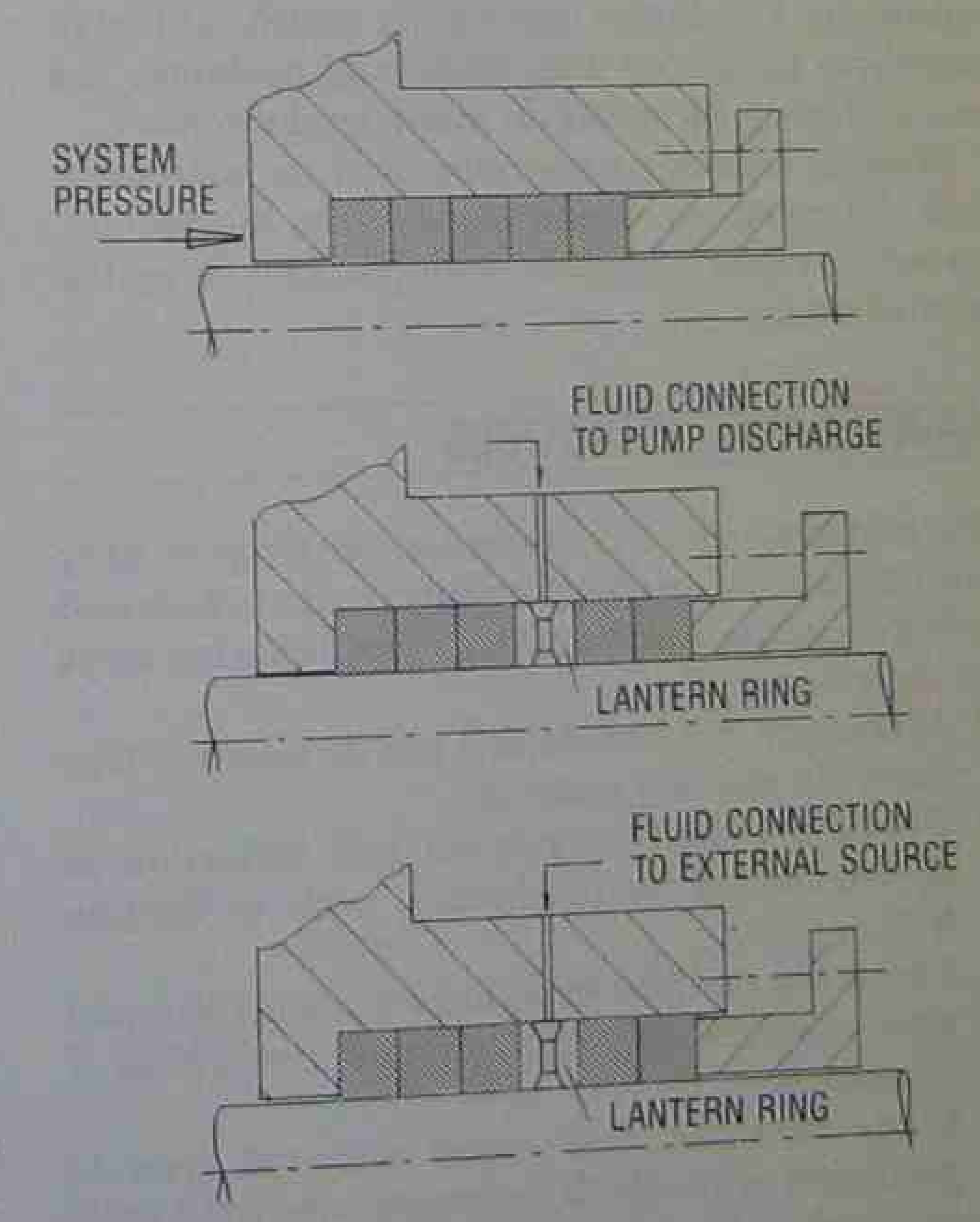
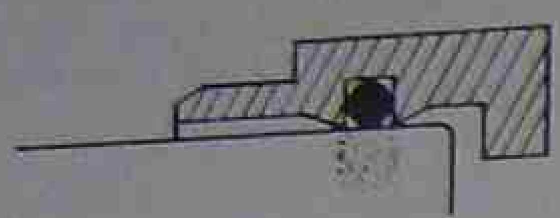


Fig. 9-55 Alternative methods of lubricating a packed gland.

When vibration is very slight fretting may occur under the O-ring itself with consequent damage to the shaft or sleeve as shown in Fig. 9-49.



Flexibox

Fig. 9-49 Fretting damage.

Elimination of the source of vibration is the ideal way to solve such problems but if this cannot be achieved totally then hard facing of the sleeve or shaft will help reduce the effects of the problem.

O-ring damage

Damage to the secondary seals can occur in a number of ways. Apart from errors made during assembly it is usually assocated with the conditions of operation. Overheating of O-rings causes hardening and cracking and may indicate either that the O-ring material is not suitable for the duty or that improved cooling of the seal is required.

Extrusion of the O-ring, shown in Fig. 9-50, may be caused by excessive pressure or incorrect clearances.

If the O-ring material is incompatible with the seal fluid then it may appear to be eaten away as shown in Fig. 9-51. The seal manufacturer should be consulted to consider the selection of the seal materials.

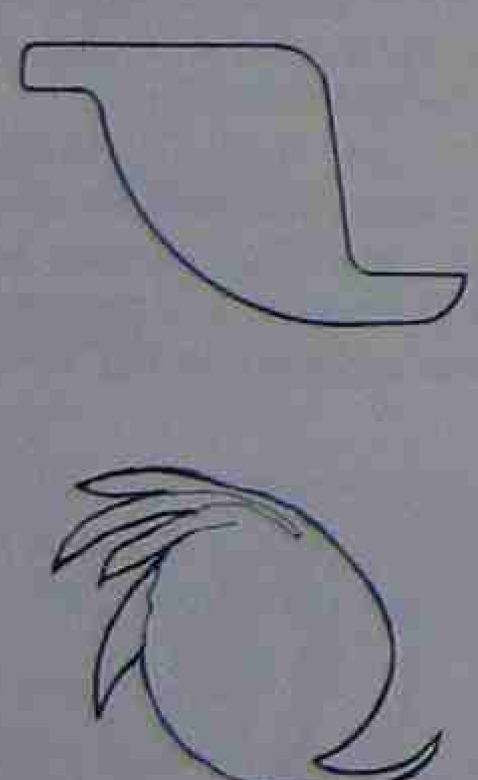
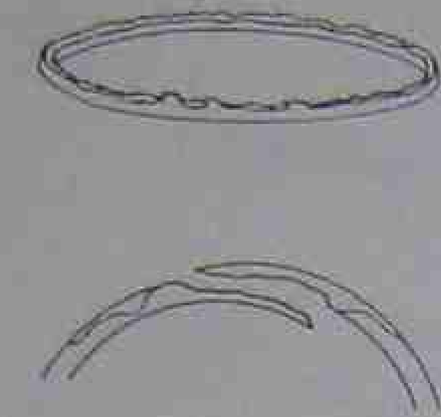


Fig. 9-50 Extrusion of the O-ring.

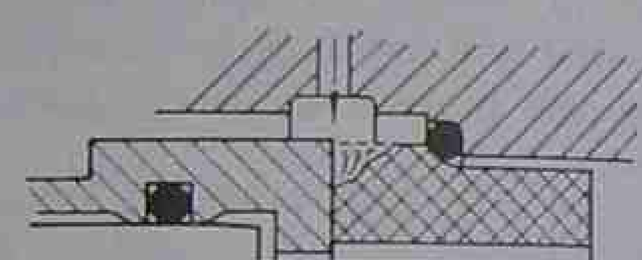


Flexibox

Fig. 9-51 Incompatibility of O-ring material and seal fluid.

Carbon ring erosion

Erosion damage to the carbon ring, shown in Fig. 9.52, may occur due to either the circulation pressure being too high or the presence of abrasive particles in the circulation flow.



Flexibox

Fig. 9-52 Erosion damage to the carbon ring.

.5 PACKINGS

The use of packings is one of the oldest approaches to solving dynamic sealing problems. Rings of soft packing material are usually fitted into a stuffing box around the shaft and compressed by a gland ring which forces the packing into contact with the shaft and the stuffing box bore. This method of sealing is also used as a static seal and for reciprocating mechanisms.

SEALING PRINCIPLES

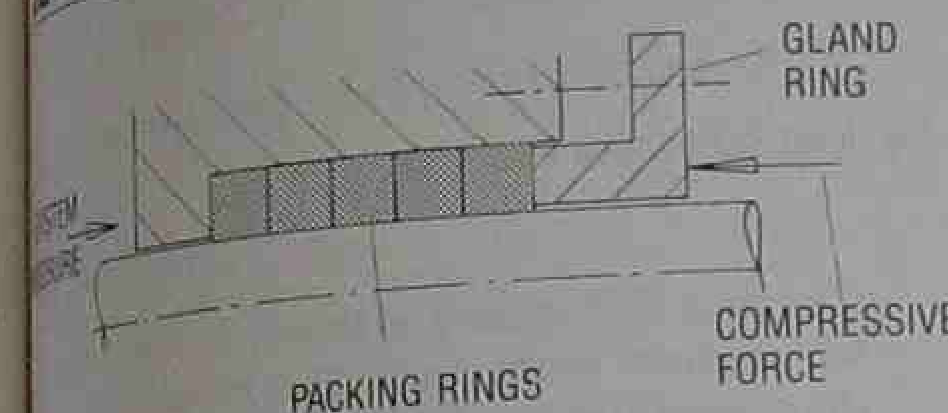


Fig. 9-53 The operating principles of a packed gland.

There are various forms that the packing may take, the most usual being rings of soft woven or braided material that are cut to length and assembled around the shaft inside the stuffing box. A gland ring or follower is pulled up against the stuffing box casing to compress the rings and cause them to expand radially. This radial expansion brings the packing into loaded contact with both the shaft and the stuffing box bore to create the seal. One of the factors affecting the efficiency of the packing is the number of rings. It has been demonstrated by experience that the optimum number of rings is five, due to the manner in which the pressure varies over the packing area. As can be seen from Fig. 9-54, if there are too many rings, the innermost do very little work.

The materials used for packing possess self-lubricating properties which protect the shaft from wear, at least on start-up. However, this lubricant would soon leach out and the packing overheat if it were not supplemented in some way.

In cases where the pump suction pressure is above atmospheric pressure the sealed fluid can be used directly to lubricate the packing by allowing a small amount of controlled leakage through the gland. If the pump suction pressure is below atmospheric pressure, liquid from the pump discharge may be directed into the gland area via a lantern ring which is installed between the rows of packing. If the sealed fluid contains abrasives or is unsuitable for lubrication for some reason a secondary fluid may be directed to the gland area via the lantern ring. These three alternatives are shown in Fig. 9-55.

Whatever arrangement is used it is imperative that the packing is not allowed to run dry. A properly adjusted packed gland should exhibit a slow but steady leakage of liquid from the gland ring.

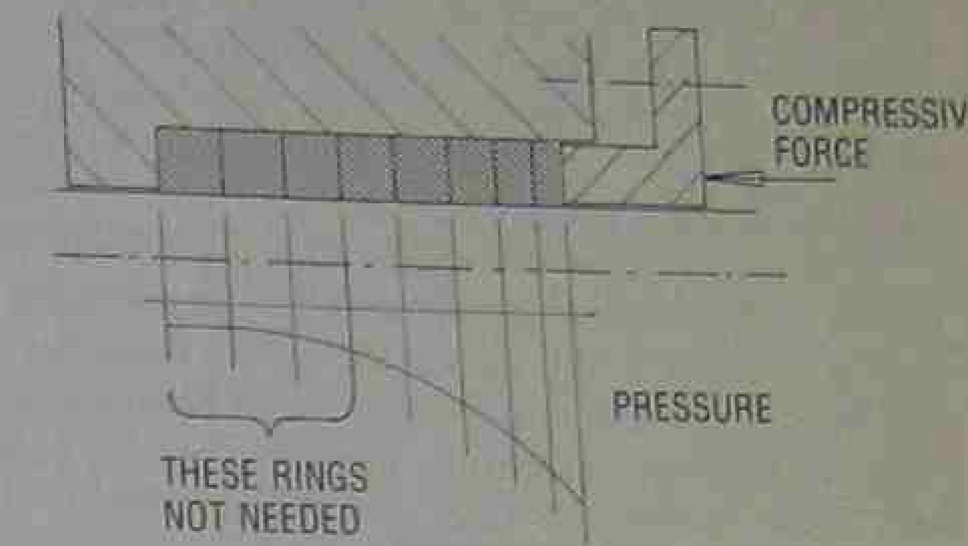


Fig. 9-54 If there are too many rings, the innermost do little work.

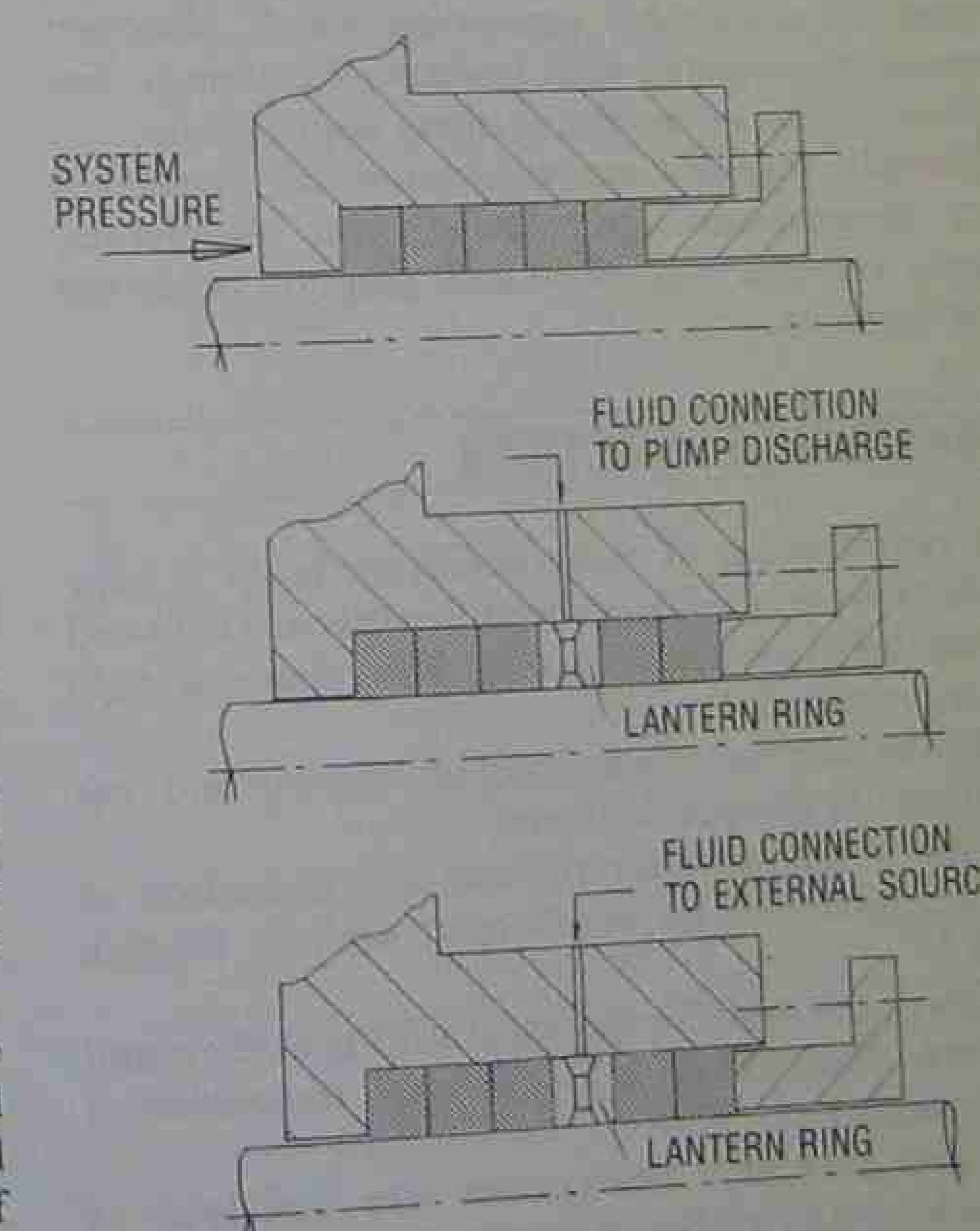


Fig. 9-55 Alternative methods of lubricating a packed gland.

MATERIALS

A wide variety of packing materials is available, the traditional type based on a lubricated fibre yarn remaining the most common. The yarns used may be twisted, plaited or braided into square or round sections using a variety of patterns.

Traditional yarn materials such as hemp, flax, cotton and asbestos (asbestos is no longer used as widely because of the associated health hazards) are still in common use and to these have been added a range of synthetic materials including nylon, rayon and, more recently, PTFE and aramid fibres. Dry lubricants, such as graphite and molybdenum disulphide, are generally more popular than oils and greases while the low friction properties of PTFE have led to its increasing use as a lubricant also. The combination of aramid fibres, which have high tensile strength and heat resistance, and PTFE lubricant has produced a new generation of packing materials which give long life and high chemical resistance.

The other significant area of development in packing materials has been in the use of graphite. Graphite filament yarns have been developed that can operate at extremes of temperature, pressure and speed in the most aggressive of chemical environments. The use of exfoliated graphite foil represents a valuable innovation which, although expensive compared with traditional packings, has proved highly successful in many applications.

Metallic foils are sometimes used for extremely high temperature applications with aluminium, copper and lead lubricated with graphite being the most common.

MAINTENANCE PRACTICES

The performance of compression packings is very much dependent on the procedures that are followed during installation and start-up. The following steps are recommended.

1 Make sure that the stuffing box is clean and free from old packing material.
2 Check the shaft for run-out and deflection as recommended for mechanical seals in Section 9.4.
3 Check the shaft for wear and scoring in the gland area. Replace or build up and remachine if necessary.
4 Make sure that the correct size and type of packing is available. If necessary, check the shaft o.d. and stuffing box i.d.
5 If conventional packing supplied in a continuous length is to be used, then cut it into separate rings. Never wind a coil into the stuffing box. Cut the rings on a mandrel or exposed section of shaft as shown in Fig. 9-56.

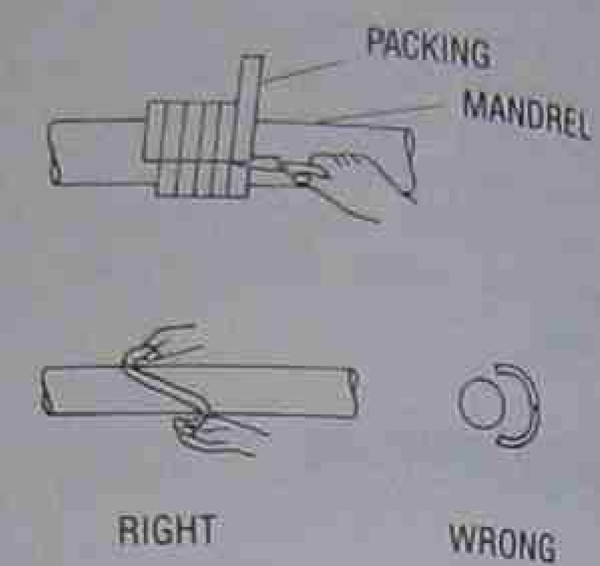


Fig. 9-56 Cut packing into rings.

Cut each ring square as shown and twist the rings to remove them from the mandrel to avoid damaging the inside diamater.
6 Install the rings one at a time and use a tamping tool to make sure that each is properly seated. A light coating of clear oil on the shaft and stuffing box bore may assist installation. Stagger the joints by at least 90° for each successive ring.
7 Install the lantern ring (if included) so that it is in line with the flushing fluid inlet. Remember that the lantern ring will move down the stuffing box when the follower is tightened and so many need to start off slightly behind the fluid inlet.
8 Install the gland ring or follower and pull up finger tight.
9 If a secondary flush is to be used, open the inlet to the gland and ensure that fluid is present.
10 Start up the machine and allow the gland to leak freely for a minute or two to ensure that the packing is well lubricated. Take up the gland ring bolts until leakage is reduced to a tolerable level.
11 After about one hour take up the gland until the leakage rate is around 1-2 drops per second for a Ø25mm (1″) shaft, more for larger sizes.
12 Check the gland periodically to adjust the leakage rate and check the stuffing box temperature. If the gland runs hot increase the leakage rate. Never, for any reason, allow the gland to run without leakage as this will quickly cause over-heating and destroy the packing.

Exfoliated graphite

Unlike most packing materials exfoliated graphite is supplied as ribbon or tape which can be used to form

packing rings *in situ* to suit any sized gland. The principle on which these are prepared is shown in Fig. 9-57.

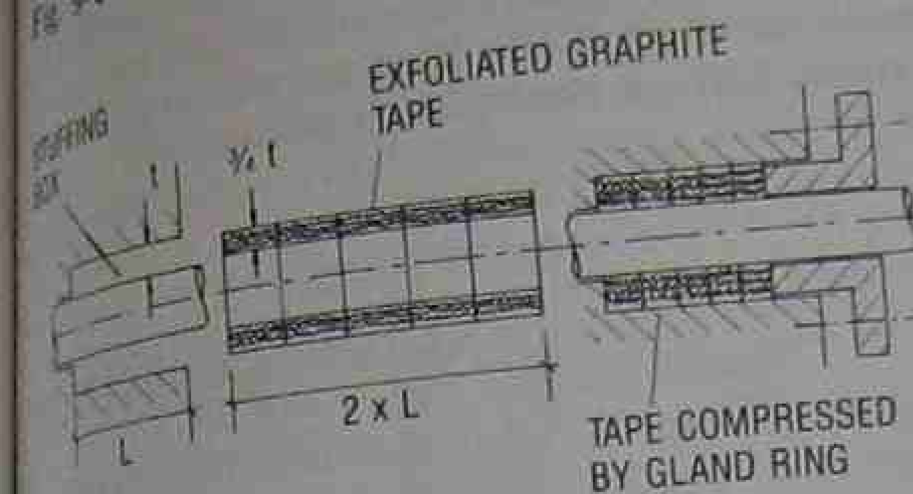


Fig. 9-57 Forming packing rings from exfoliated graphite

The tape or ribbon is wound on to the shaft to form a ring of approximately three-quarters of the thickness of the gland space as shown. As the compression required of the material is normally around 50%, a sufficient number of rings to form a length approximately twice that of the gland will be required. Hence the tape width should be around two-fifths of the final packing depth if five rings are to be used. When all five rings have been built up and slid into the stuffing box the gland ring can be installed and tightened to give the rings compression. As with conventional packings, a slight leakage should be maintained to ensure that the package is lubricated and gives maximum life.

9.6 LIP SEALS

Radial lip seals are used primarily to retain oil and other lubricants in equipment operating with rotating shafts. They are also used to exclude dirt and other contaminants. They are generally only suitable for sealing against low pressures of the order of 35 kPa (5 psi).

SEALING PRINCIPLES

A radial lip seal consists of an elastomeric sealing ring contained in a metal case as shown in Fig. 9-58.

Fig. 9-58 A radial lip seal.

The sealing ring is formed with a knife edge at the contact point which engages with the rotating shaft. A garter spring may be fitted inside the ring to help load the seal. The flexible element may be bonded to the case or the parts may be made separately and then rolled or crimped together.

Sealing occurs by establishing a load at the point of contact between the sealing ring and the shaft. The preload created by the interference fit of the elastomeric ring is supplemented where necessary by the mechanical pressure of the garter spring. A hydrodynamic oil film is formed at the point of contact and should be around 0.025mm (0.001") thick to prevent friction and wear. If the film is any thicker then leakage tends to occur.

The action of the seal relies on the continuous presence of a clean oil film to protect the sealing edge of the elastomeric ring. The shaft surface should be machined to a high order and yet remain rough enough to promote oil retention. A surface finish of the order of 0.025–0.050mm (0.001"–0.002") is recommended.

TYPES AND MATERIALS

A radial lip seal may be of bonded construction in which the flexible sealing element is permanently bonded to the casing or it may be assembled from separate components which are crimped together as shown in Fig. 9-58. The principles of bonded construction are shown below in Fig. 9-59.

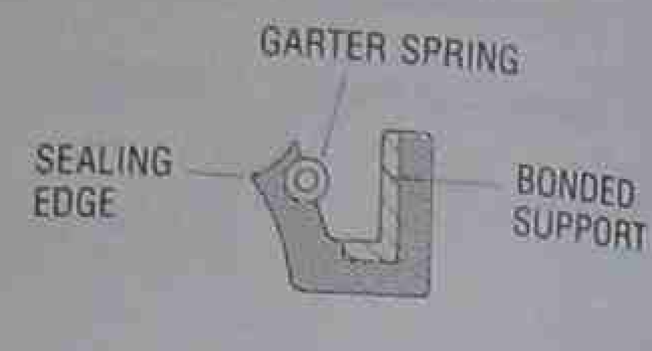


Fig. 9-59 The principles of bonded construction.

The seal may have a single or double lip arrangement with either or both lips being spring loaded. Typical alternatives are shown in Fig. 9-60.

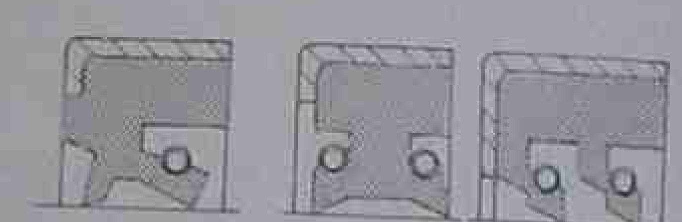


Fig. 9-60 Typical lip seal arrangement.

The second lip of a double lip seal performs the function of a dirt seal or, in a back-to-back arrangement, provides sealing in both axial directions.

In order to increase the sealing efficiency the atmospheric side of the seal lip may have ribs or grooves moulded into it. These create a hydrodynamic pumping action during operation so that any fluid that leaks past the lip is pumped back into the contact area. This feature allows the seal to

Fig. 9-61 Moulded grooves in the atmospheric side of the seal lip help to prevent leakage.

operate at lower lip pressure and hence reduces friction and wear. An example is shown in Fig. 9-61.

Different materials are used for the flexible element of a lip seal depending on the operating conditions. These include nitrile, polyacrylate, fluorocarbon, polyeurethane, PTFE, silicone and leather. The most common of these is nitrile which is compatible with most lubricants and suitable for temperatures up to 130°C (270°F).

MAINTENANCE PRACTICES

One of the most common causes of lip seal failure is damage that occurs during the handling or fitting of the seal. Hence great care should be taken during assembly and the following procedures and precautions are recommended.

1 Before attempting to install a lip seal, make the following checks:
   - Check the seal dimensions against the shaft and housing and make sure they match specifications.
   - Check the seal lip for damage.
   - Check the shaft surface finish and make sure it is free of nicks and burrs.
   - Make sure the seal is clean and that the garter spring is properly located.

2 Make sure that the seal is orientated with the lip facing towards the lubricant.
3 Apply a coating of lubricant to the sealing edge and to the shaft to aid installation.
4 If the seal is metal-cased then a coating of gasket cement around the outside may be required to prevent the seal from leakage around the housing.
5 Use a properly designed tool to push the seal home into the housing.

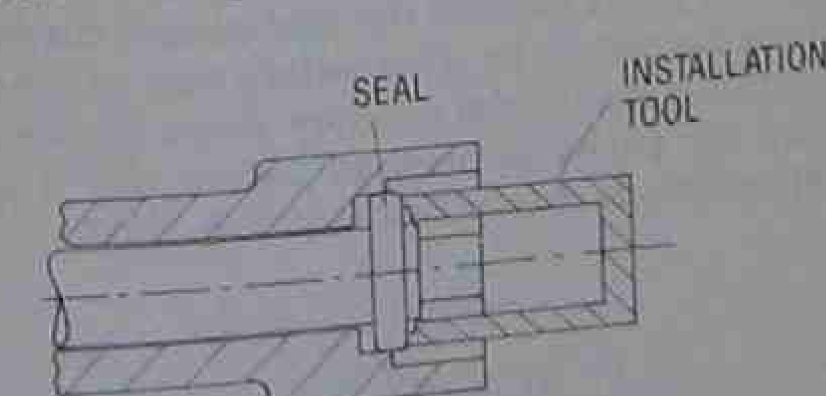


Fig. 9-62 Use the correct tool to push the seal home into the housing.

If the correct tool is not available use a suitable ring that contacts the seal casing around the outside diameter. Never press against the sealing lip.

6 If the seal has to be fitted over a keyway, splines or other sharp corners use a protective cone or sleeve as shown in Fig. 9-63.

Fig. 9-63 Using a protective sleeve.

7 Make sure that the seal is not cocked in the housing and has been pushed home firmly all round.
8 Turn the shaft by hand to make sure it runs freely.
9 The seal may leak slightly on start up until the lip wears in and the hydrodynamic film is established. Once the seal is seated it should run without any measurable leakage.

FAILURE PATTERNS

The most common reason for failure of a radial lip seal is mechanical damage to the sealing edge. The dangers of causing damage during assembly have already been pointed out. They should be avoided if correct procedures are followed. The condition of the shaft is critical and if a seal does not give satisfactory performance then the shaft surface should be carefully inspected for imperfections.

Running conditions can also affect seal performance. The operating speed and temperature must be kept within the design limits of the seal or early failure due to excessive wear can be expected. Inadequate lubrication will also lead to rapid wear and early failure and it should be recognised that operating speeds have an effect on the characteristics of the lubricant. As the speed increases the operating temperature rises and causes the viscosity of the lubricant to decrease and the lubricant film to thin out. This causes increased friction and a further rise in temperature and this whole cycle greatly reduces the life of the seal.

In addition to the above considerations, care should be taken to ensure that the seal is installed the right way round. If hydrodynamic ribs are provided the direction of rotation may be critical and should be checked.

9.7 CIRCUMFERENTIAL SPLIT RING SEALS

Circumferential split ring seals, sometimes referred to as carbon ring seals, are a modified version of a piston ring seal and are designed to seal gases and vapours at high temperatures. They are often found in steam turbines.

SEALING PRINCIPLES

In their simplest form, split ring seals consist of three or more carbon ring segments cut radially to form a set as shown in Fig. 9-64.

Fig. 9-64 Simplest form of split ring seal.

The rings are held together by a garter spring which fits into a machined groove around the outside of the ring segments as shown in Fig. 9-65. A stop or pin prevents the rings from rotating inside a slot in the machine housing.

Fig. 9-65 Typical split ring seal arrangement.

The clearance between the ring and the shaft is sufficiently small to prevent leakage and the internal pressure forces the ring segments against the side of the slot to create an axial seal in a similar way to a mechanical end face seal. Diametral clearances for carbon ring seals usually run around 0.0125mm –0.0625mm (0.0005"–0.0025").

The seal is free to move radially in the housing and can therefore compensate for small irregularities in shaft movement without allowing excessive leakage.

Axial movement of the shaft can also be tolerated by this type of seal.

TYPES AND ARRANGEMENTS

Carbon ring seals are usually installed in sets which are spread axially along the shaft as shown in Fig. 9-66.

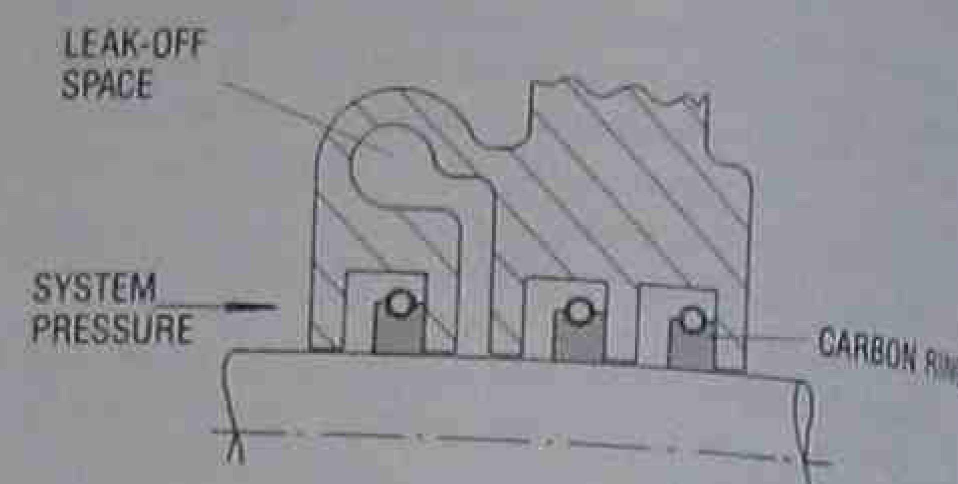


Fig. 9-66 Carbon ring seal set.

A leak-off space is often provided so that any fluid that leaks past the inner ring is allowed to blow off freely and not pass across the outer rings which usually protect a bearing.

More complicated split ring seal arrangements employ up to three sealing ring elements with each consisting of several segments. Some of the rings have joints cut tangentially and are self-compensating for wear. Two primary rings with staggered joints provide the seal against the housing and are backed up by a secondary ring which covers the radial joints in the primary rings. Both sets are loaded by garter springs as before and a typical arrangement is shown below in Fig. 9-67.

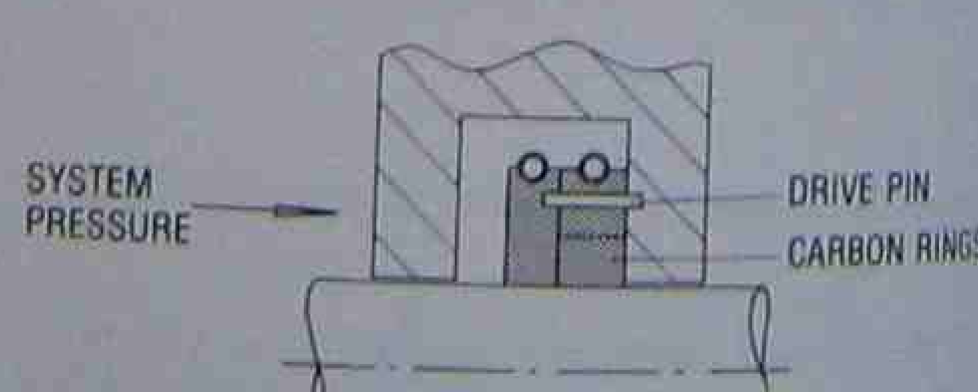


Fig. 9-67 A more complicated split ring seal arrangement.

MAINTENANCE PRACTICES

Carbon ring seals generally require little maintenance if the shaft is running true and the housing is clean and free from damage to the sealing surfaces.

The following points should be considered.

- There is no particular tolerance on carbon ring clearances. Seals should be replaced or repaired when leakage becomes excessive.
- When a machine is shut down, the carbon ring seals should be inspected for wear and sticking. Debris may build up in the seal slot and prevent the sealing ring from moving freely.
- Spring tension should be checked and springs should be replaced if necessary.
- Rings should be dismantled carefully, cleaned, and inspected for damage.
- Rings should always be marked so that they can be reassembled in the same positions.
- Worn rings can be refitted by scraping if necessary. The inner surface should be scraped first to fit the shaft and then the face. The ends of the segments should be scraped last until the complete ring fits the shaft but does not grip it when the garter spring is installed.
- Broken or damaged rings should always be discarded and replaced.
- Before reassembling the seal, the shaft and seal slot face should be carefully examined for damage and build-up of debris.
- Always handle carbon rings with great care because of their fragility.
- When refitting rings make sure that no dust or dirt is trapped against the sealing faces or between the ends of the segments.

9.8 FELT SEALS

Felt seals are primarily used as oil and dust seals and provide a cheaper alternative to radial lip seals under operating conditions that are not too severe.

SEALING PRINCIPLES

Felt is a fabric composed of interlocking wool, or other animal, vegetable or synthetic fibres. When used for sealing purposes it is normally presaturated with a lubricant of slightly higher viscosity than that being sealed. The seal is usually cut from felt sheet and is held in a casing of which there are various designs.

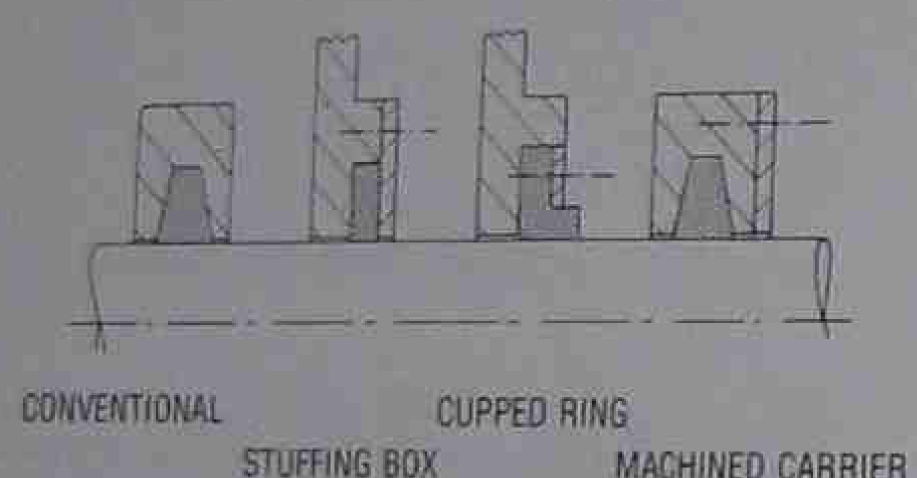


Fig. 9-68 Examples of felt seal designs

The configuration of the felt ring should be such that the difference between the inner and outer diameter is larger than the thickness and it is usual for the thickness to be greater at the i.d. than the o.d. The seal is normally supplied in the form of a cartridge which is designed to fit a machined housing in the bearing assembly.

The interference between the seal and the shaft should be only slight and contact pressure should be low.

Felt seals are not suitable for operation with low viscosity lubricants or for sealing against pressure.

MAINTENANCE PRACTICES

There are a number of considerations that should be kept in mind when handling felt seals.

- Felt is are relatively delicate material and can be easily damaged if it is unduly compressed or stretched.
- Like other types of seals the condition of the shaft at the seal interface will play an important part in the effectiveness of the seal.
- It is normal practice to replace a felt seal, regardless of its condition, whenever a machine is overhauled.

9.9 CLEARANCE SEALS

All the seals discussed so far have been contact seals which rely on direct lubricated contact between a sealing element and the rotating shaft. Clearance seals operate without contact and rely on a small clearance gap to create a pressure drop between the system pressure and the atmosphere.

BUSHINGS

The simplest type of clearance seal is a bushing which operates because of the throttling action provided by a small clearance gap over the length of the bushing.

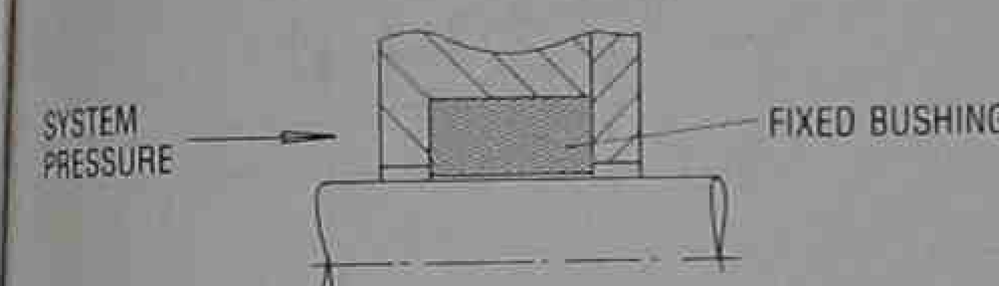


Fig. 9-69 A typical example of a bushing.

The long but relatively small clearance space between the shaft and the bushing creates a restriction to flow which will vary depending on the viscosity of the fluid. Hence bushing seals are not as effective with low viscosity liquids and are no use for sealing gases and vapours.

Types and materials

The simplest type is a fixed bushing of the type shown in Fig. 9-69. The disadvantage of a fixed bushing is that it cannot tolerate any degree of misalignment or eccentricity of the shaft. The only way a fixed bushing can cope with any form of shaft run-out is by increasing the clearance and accepting a higher leakage rate.

To overcome the problem of shaft run-out, a floating bushing may be used. In this arrangement the bushing is allowed to float in the housing and one end is lapped and forms an axial seal against the end of the housing. The bushing is held by a dowel pin so that it does not rotate and may be spring loaded as shown in Fig. 9-70.

For sealing against high pressures, which tend to cause the bushing to stick to the housing, the effect of pressure can be relieved by providing hydraulic

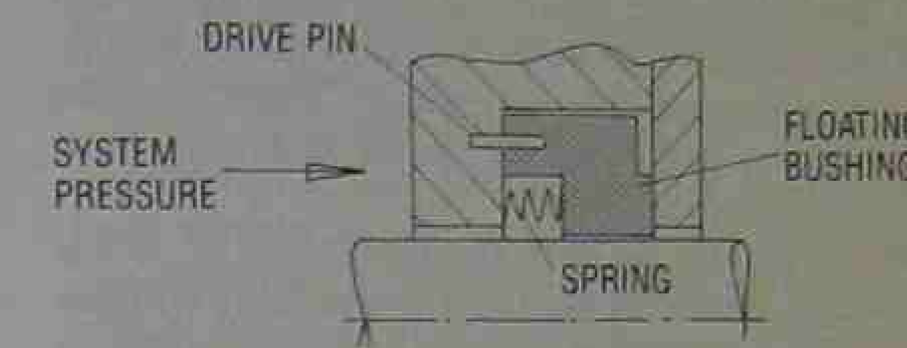


Fig. 9-70 A floating bushing may be spring loaded.

balance in the same way that axial end-face seals are balanced. The end of the bushing is stepped to provide back pressure as shown in Fig. 9-71.

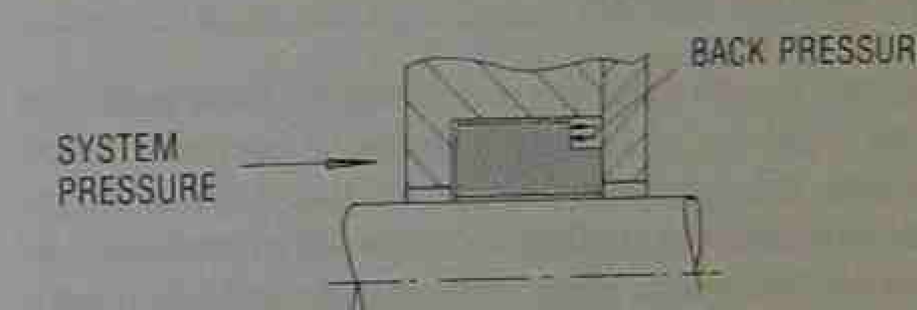


Fig. 9-71 A pressure balanced bushing.

An alternative way to handle high pressure is to replace a single bushing by a series of seal rings, each of which carries only a portion of the total pressure drop. Each ring is free to float in the housing as shown in Fig. 9-72.

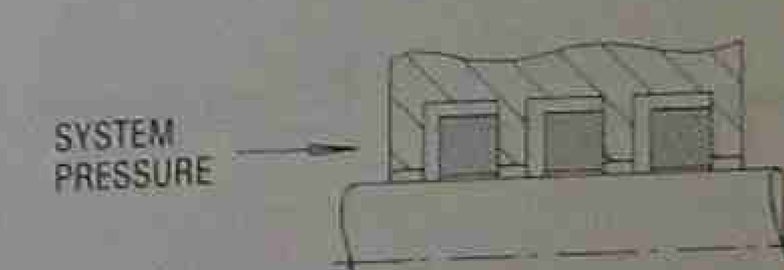


Fig. 9-72 A single bushing can be replaced by a series of seal rings.

Because contact between the shaft and bushing cannot be prevented with certainty it is usual to use a low-friction material for the bushing. Carbon, molybdenum disulphide and PTFE are commonly used, either independently or in composite form, and whitemetals can be used for low temperature service. Higher temperature applications may require the use of aluminium alloys or bronzes.

Maintenance practices

For effective operation, bushings must present the correct clearance gap. Shaft o.d. and bushing i.d. are critical dimensions that should be checked to ensure that the bushing operates effectively. Shaft surfaces should also be inspected for nicks and burrs and excessive machining marks.

Housings should be kept clean and free of debris and bushings should be free to move radially without interference. The ends of floating bushings should be lapped flat and should be free of chips or cracks so that a good axial seal is made with the housing.

Despite the flexibility of a floating bushing, excessive run-out or shaft deflection cannot be accommodated by a bushing seal and should be eliminated.

LABYRINTH SEALS

A labyrinth seal presents a tortuous leakage path to a sealed fluid by providing a series of barriers which dissipate the pressure energy of the fluid in steps. They are used for sealing compressible fluids such as gases and vapours. They can operate at high speeds and temperatures and are often used in high speed machinery such as centrifugal compressors and steam turbines.

Sealing principles

Labyrinth seals can be made in many different forms but the simplest arrangement is the type shown in Fig. 9-73.

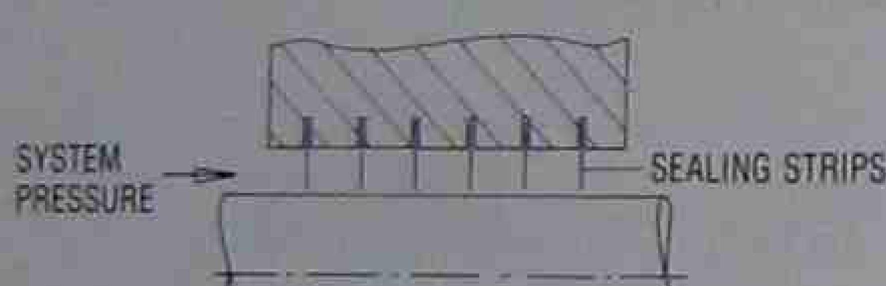


Fig. 9-73 The simplest arrangement of a labyrinth seal.

As the sealed fluid passes through each clearance gap it first accelerates, then decelerates and expands into the following chamber producing turbulence and friction. This results in a loss of pressure energy at each stage with the total pressure differential being spread across the full length of the seal.

The barriers to flow are formed by thin parallel metal strips that are held perpendicular to the shaft and maintain close clearance with the shaft surface. Each chamber has to be wide enough to prevent the fluid from passing directly through from one clearance gap to the next. With a straight through design of the type shown in Fig. 9-73 some carry-over of kinetic energy from one chamber to another is inevitable but this can be eliminated by modifying the seal geometry and introducing a staggered or stepped arrangement as shown in Fig. 9-74.

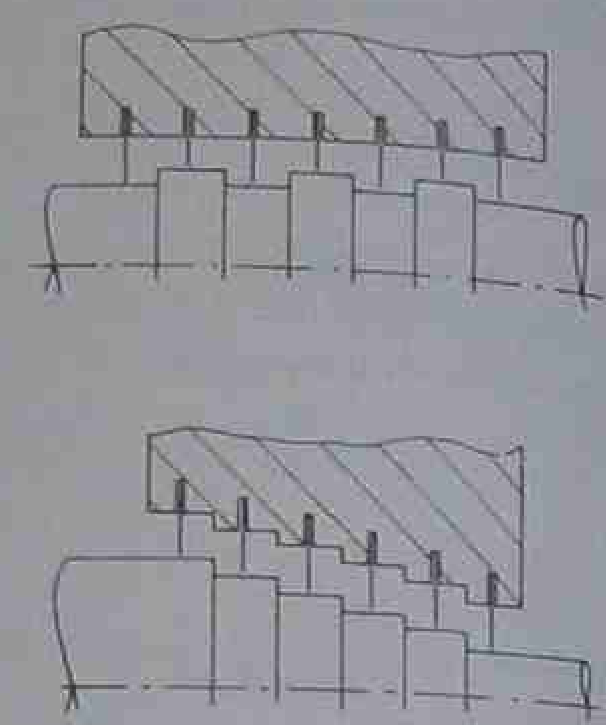


Fig. 9-74 Staggered and stepped arrangement.

Types and materials

Labyrinth seals are usually custom designed to suit each individual machine and materials are selected according to the conditions of operation.

An additional feature that is sometimes desirable involves the provision of a neutral gas purge. The gas is injected into the seal near the high pressure end and then vented to atmosphere either directly or via an eductor system as shown in Fig. 9-75.

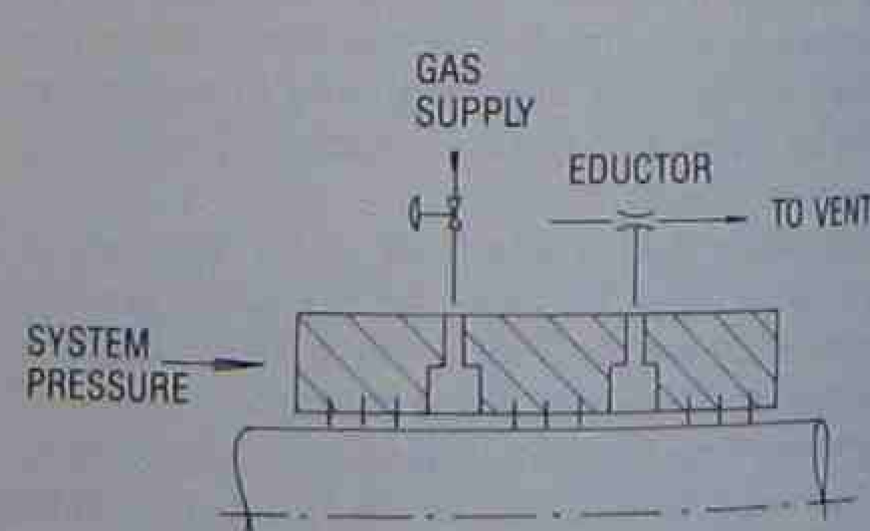


Fig. 9-75 A neutral gas purge.

Maintenance practices

A labyrinth seal should require very little maintenance as long as other machine elements are functioning properly. The clearance gap is vital to the operation of the seal and will depend on the accuracy of machining, concentricity of the housing and the true running of the shaft.

Excessive bearing clearances or any other source of run-out in the shaft will affect seal clearances and may ultimately lead to damage and leakage. Thermal expansion must be allowed for in the design of the seal and changes in the operating temperatures of the machine may affect seal operation. Rigidity of the housing or machine casing is also an important factor in maintaining the correct seal clearances.

When a machine is opened for inspection the sealing tips should be checked for damage and the shaft surface should be checked for interference. If the shaft runs true in the housing and temperature effects are properly allowed for, a labyrinth seal should give unlimited life.

CHAPTER 10

CONDITION MONITORING

The advancement of high technology, especially in the aero-space and nuclear power industries, where safety and reliability are of prime importance, has led to the development of a range of techniques designed to monitor machine operation and generate information that can be used to anticipate breakdown. Monitoring may be carried out on a continuous or an intermittent basis and in some cases can be used to activate a shutdown mechanism.

All condition monitoring techniques rely on monitoring some physical characteristic that reflects the condition of the machine. A normal running level for that characteristic is established when the machine is in good condition and then any significant deviation from that level gives warning that a fault may be developing. This enables a potential fault to be detected before it becomes serious enough to substantially affect machine performance and in time for corrective action to be taken.

These methods are often referred to under the heading of 'Predictive Maintenance' and have been widely adopted in most industries over the last ten years.

Advantages of condition monitoring

Although some of the techniques described in this chapter can be expensive to install, for plant and equipment where there is a high safety and reliability requirement there are distinct benefits to be gained.

- Safety is improved by avoiding the development of dangerous situations that may be hazardous to personnel and other plant and equipment.
- Disruption of production schedules can be reduced by preventing unexpected breakdowns.
- By detecting faults before serious failure occurs, damage to plant and equipment is reduced.
- Condition monitoring eliminates the need to strip down equipment during annual overhaul. As well as indicating when a machine is becoming unserviceable, monitoring also confirms that a machine is operating satisfactorily.
- The equipment can also be used to assist in troubleshooting and to help identify failed components when breakdown does occur.

METHODS

All the methods described below are designed to monitor the condition of on-line machinery and all can be considered as improved, mechanised versions of inspection by personnel. Before the development of these techniques, condition monitoring was carried out by the operator of a machine who observed the running condition using the senses of sight, hearing, touch and smell. This meant that the operator would **look** for obvious signs of defects, **listen** for unusual sounds, **touch** to check overheating and **smell** to detect burning or overheating. This method of inspection can work quite effectively when the operator is experienced and in constant attendance on the machine. In modern industry however, the majority of machines run unattended and although inspection by personnel is still an important element in the monitoring process, mechanised techniques are essential if constant surveillance is to be achieved.

Temperature monitoring

This is one of the simplest methods available and involves the use of thermocouples or resistance thermometers to measure bearing temperatures. The sensing element should be located within 1.25 mm (0.05") of the bearing surface and good thermal contact is vital.

Bearing temperatures tend to fluctuate with load so any preset warning levels must take account of

normal maximum temperature. Warning levels should be based on rise above normal rather than an arbitrary maximum.

This method can give a warning of several hours before breakdown occurs and can be used to raise an alarm and shut down the machine.

Spectrographic oil analysis

Samples of lubricating oil can be analysed using a spectrometer and the proportions of metal elements present can be determined. The oil samples used must be representative of the total contents of the system and should be taken under normal operating conditions. The major source of metals found in lubricating oil is wear debris, and a knowledge of the component materials can indicate the origin of the metals detected.

This method requires regular sampling so that trends can be established. Normal wear will produce a slow but steady increase in metal levels and it is when a sudden increase is recorded that some malfunction is indicated. It should be remembered when interpreting results that metal levels will be high during a running-in period and will also be affected when oil levels are topped up.

Slow deterioration in operating condition can be detected by carefully monitoring trends but failures which occur rapidly may not be detected by this technique because of the time delay between sampling and obtaining the result of the analysis.

This method, because it is based on an intermittent sampling process, is not suitable for automatic alarm and shutdown.

Particle retrieval

As with the last method, this technique analyses the wear debris present in the lubricating oil but instead of measuring metal levels it is more concerned with the physical size of the particles. Wear debris can be collected by installing magnetic plugs in the system downstream of bearings to catch ferrous particles and by backwashing the oil filter element to collect non-ferrous particles. Debris is inspected and classified with the aid of a microscope.

As with the previous technique, this also relies on observing deviations from a trend established by regular sampling and analysis. During run-in large quantities of very small particles can be expected while normal operation would be expected to yield a smaller quantity of wear particles generally below 25×10^{-6} mm (1×10^{-6}") in size. A steady build up of particles larger than 0.25mm (0.01") over three or more samples indicates that wear is increasing at a rapid rate and failure may be imminent.

Again, because this method is based on intermittent sampling, it is not suitable for automatic alarm and shutdown.

Noise monitoring

The noise produced by a machine has traditionally been used as one of the key sources of information about its operating condition. It is still common practice for a technician to hold a screwdriver to a bearing housing in order to detect a malfunction. This simple process can now be systematised using a microphone, amplifier, meter, and continuous recording equipment.

This method is very similar to vibration monitoring which is described below but is not as useful because of the greater difficulty in interpreting results. Hence it is not widely used.

Relative displacement measurement

This method involves the use of devices called **proximity probes** which measure the distance between the probe and a surface such as a rotating shaft. When two probes are set up to view a shaft at right angles, as shown in Fig. 10-1, the outputs can be fed into the *x* and *y* channels of an oscilloscope and a display of the shaft locus (the path of movement of the shaft axis) can be seen.

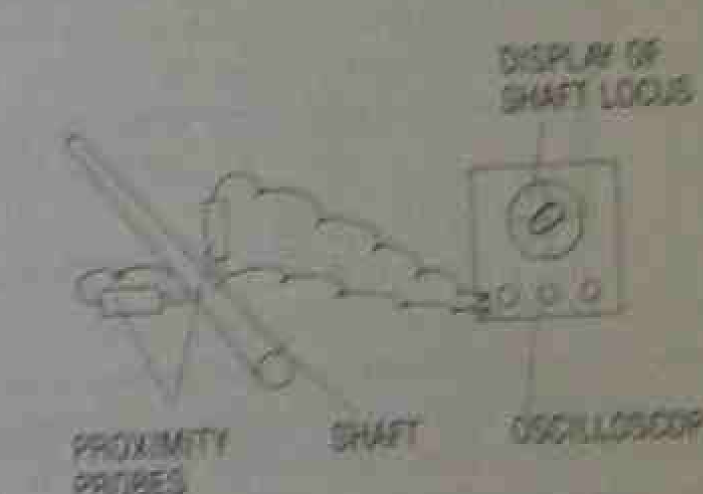


Fig. 10-1 Setting up proximity probes

Excessive shaft movement can be detected by this method and particular problems, such as shaft whirl and oil whirl, can be analysed using the oscilloscope.

Monitoring is continuous and can be used for automatic alarm and shutdown.

Vibration monitoring

The most widely adopted form of condition monitoring over the last ten to fifteen years involves the recording and analysis of machine vibration patterns. Excessive vibration is a reliable indicator of malfunction and by analysing vibration levels at different frequencies accurate diagnosis of the source of vibration is possible.

Vibration is usually measured by a transducer probe which is attached to a suitable point on the machine and is linked by cable to a remote monitoring unit. The probe, which contains a piezo electric accelerometer, may be permanently installed for continuous monitoring or may be hand held and attached to a portable unit for intermittent checking.

The monitoring equipment may be tuned to record vibration levels at all frequencies or, by the use of filters, may be tuned to a selected frequency band. It is usually especially important to monitor vibration levels at motor drive speed and multiples of drive speed.

This method is ideal for continuous monitoring and vibration sensors can be used to initiate automatic alarm and shutdown devices.

As with all other monitoring techniques this method relies on measuring a change from the pattern associated with normal operation. Vibration monitoring allows a signature pattern to be recorded that represents the vibration levels at all frequencies. An initial trace, made with the machine in good operating condition, will represent the base pattern from which deviations can be observed.

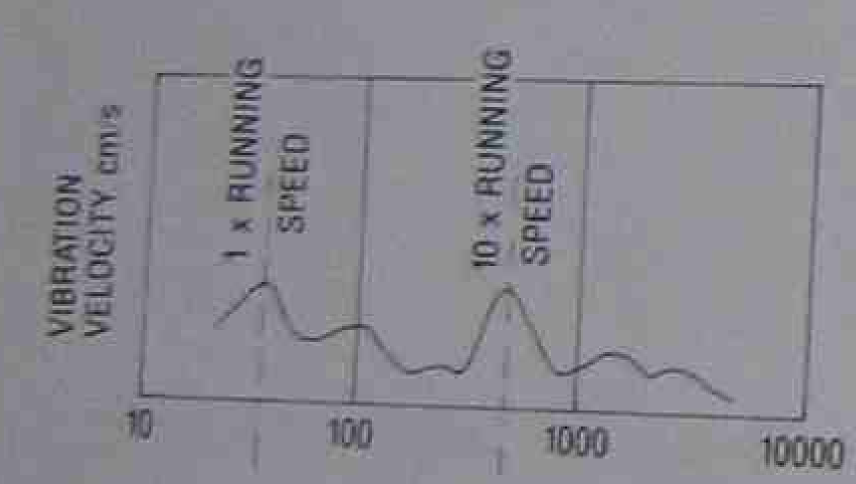


Fig. 10-2 A typical example of a base signature pattern.

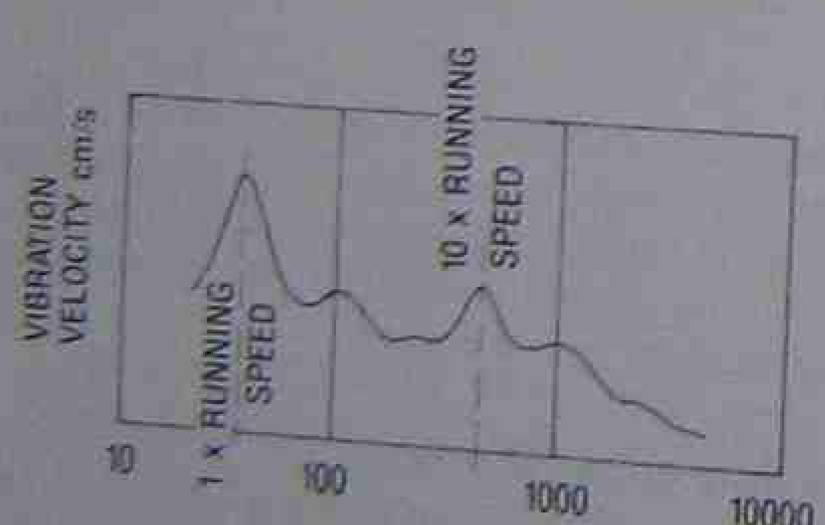


Fig. 10-3 A later trace taken for the machine in Fig. 10-2.

The trace shown represents a pump with ten vanes to the impeller and it should be noted that high vibration levels are indicated at running speed and ten times running speed.

A later trace taken for the machine is shown in Fig. 10-3. It can be clearly seen that there has been a marked increase in the vibration level at running speed which might be caused by a loss of balance.

Different mechanical problems give rise to high vibration levels at different frequencies and hence it is possible to use the signature analysis to predict the potential causes of the problem. Some examples are given below.

Source of vibration	*Dominant Frequency*
Oil whirl	½ x shaft speed
Gear whine	Tooth contact frequency
Bearing defects	Ball or roller speed
Misalignment	2 x shaft speed
Out of balance	Shaft speed
Fan whine	Blade passage frequency

Experience plays a major part in determining the acceptable levels of vibration for particular machines and these will be influenced by the size and type of machine. As a guide to acceptable levels the following chart gives recommended limits for different categories of machine.

Vibration velocity upper limit mm/s rms	Machine class 1 2 3 4 5 6
0.28	
0.45	
0.71	GOOD
1.12	
1.8	
2.8	
4.5	SATISFACTORY
7.1	IMPROVEMENT
11.2	DESIRABLE
18.0	
28.0	
45.0	UNACCEPTABLE
71.0	

1 Small machines such as electric motors up to 15 kW (20 hp).
2 Medium sized machines up to 75 kW (100 hp) without special foundations or rigidly mounted machines up to 300 kW (400 hp) on special foundations.
3 Large machines mounted on heavy foundations that are rigid in the direction of vibration measurement.
4 Large machines on foundations that are soft in the direct of vibration measurement.
5 Machines with unbalanceable parts (e.g. reciprocating mechanisms) on foundations that are rigid in the direction of vibration measurement.
6 Same as 5, but with soft foundations.

CHAPTER 11

TROUBLESHOOTING

Troubleshooting skills are vital to a maintenance technician. Before effective repairs can be carried out the precise cause of breakdown must be determined and although in some cases this may be clearly apparent, in others it will not and a process of investigation will be required in order for the cause to be found. Although every problem is different in some way, the process of troubleshooting, or problem solving, is essentially the same and can be examined in terms of critical elements that apply in all instances. An understanding of these elements will improve the technician's ability to troubleshoot.

Troubleshooting can be described as a logical system of investigation designed to yield the correct cause of breakdown in the shortest possible time and with the least likelihood of error. The term, **breakdown** is used to indicate any machine condition that is considered to be less than satisfactory according to these factors:

performance
downtime
service life
efficiency
safety
environmental impact
cost

which have been described in Chapter 1. Hence breakdown refers not just to dramatic failures that render a machine totally inoperable, but also to failures that lead to an unacceptable reduction in performance. This point is emphasised in order to demonstrate that troubleshooting is not some sort of secret weapon brought out of mothballs on special occasions and used only by experts. Whenever a machine fails to meet the criteria of satisfactory operation then the process of troubleshooting must be employed to determine why. It is therefore just as important for maintenance technicians to understand how the process can be most effectively carried out as for other members of the organisation.

APPROACH TO TROUBLESHOOTING

Troubleshooting, like most skills, can be learned, and it is not necessary to assume that it is something for which only certain talented people are qualified.

One of the biggest obstacles to problem solving for many people is developing the confidence to make judgments and then to back them up. Confidence, however, does not just appear out of thin air but is dependent on other factors. It is by understanding and responding to those factors that confidence can be raised and troubleshooting ability improved.

The ability to make successful judgments and become confident in solving problems is based on the following factors.

Knowledge

In order to be able to assess the condition of rotating machinery and to successfully diagnose the cause of breakdown, it is absolutely vital that technicians have a thorough knowledge and understanding of the physical characteristics of the machine and its construction, the principles upon which the machine operates and the function that it performs. If it is not clear to someone how a machine works then it is unlikely that they will have the ability to diagnose the cause of malfunction when it occurs. The greater the knowledge and understanding of the principles of operation, characteristics and function of an item of rotating machinery, then the greater will be the confidence to make judgments about its condition and the greater the likelihood of those judgments being correct.

This does not mean that it is necessary to possess an encyclopaedic amount of knowledge to be successful at troubleshooting. A certain amount of background knowledge is of course expected, and the purpose of this book is to set out some of that information with respect to key machine elements, but particular problems often require specific research to provide the detailed information necessary to solve the problem. When the occasion arises, the technician should know where to find the relevant information, and how to apply it to the problem. For example, it is always wise, before starting any maintenance task, to consult the description of operation and maintenance instructions supplied by the manufacturer. In most cases this will be sufficient to provide the detailed knowledge required although particularly difficult problems may require deeper research.

Logic

A systematic approach is essential in troubleshooting if blind alleys and false conclusions are to be avoided and causes are to be established in the most efficient manner. Troubleshooting can be thought of as a process of elimination. There may be many possible causes of a particular breakdown and troubleshooting is the process by which all but the actual cause are eliminated. Finding the shortest route from a situation of countless possibilities to the real cause of the problem requires a logical, step-by-step approach.

There is no need to be frightened by the word, logic. It is merely the process of making reasonable deductions from existing information, e.g., hammer-blow + thumb-in-line = pain. The ability to apply logic in this way is really the ability to use **common sense** based on knowledge and experience.

Another way to consider the application of logic is to think of it as a question of **organisation**. When planning any activity there is a sensible (i.e. logical) sequence of events that will give the best result. When wallpapering a room, for example, the room is measured before the wallpaper is bought, not after. Logical troubleshooting is merely an extension of this simple principle. All the steps in the troubleshooting process should bring a solution closer by adding to the understanding of the problem and eliminating irrelevant possibilities.

In most cases the correct solution can usually be selected from a range of obvious alternatives by using informally, the logical principles of an organised approach and common sense. Where obvious solutions fail, however, a more formal and systematic approach is called for. The most formal way in which logic can be applied to a problem is through a systematic diagnostic approach of the kind explained on page 128. This more rigorous type of approach is not usually used unless the problem proves to be especially difficult to solve. This approach requires care and a considerable amount of discipline as cutting corners can easily destroy the benefits of such an approach.

Whether logic is applied informally in the form of common sense or formally through a systematised method it can be thought of as the thread that draws the pieces of the puzzle together to produce the right solution. As with a jigsaw, there is only one way the pieces will fit together to make the picture and similarly with a maintenance problem the correct solution can only be found if the pieces of information and evidence are put together in the right way. Logic is the key to achieving this in the most efficient manner.

Experience

Although much can be accomplished with sound technical knowledge and a rational approach, there is little doubt that experience can be invaluable when it comes to troubleshooting. Unfortunately machinery does not always react according to prediction and although a logical explanation is usually possible once the cause of a problem has been found, logical deduction of the cause from the evidence available may not be so simple. Someone with experience should find it possible to make educated guesses and connect cause and effect, whereas this may be difficult for an inexperienced person.

This does not necessarily mean that only the most experienced should turn their hand to troubleshooting although clearly, all other things being equal, an experienced person may have something extra to offer. Those with less experience should never be afraid to call on the experience of others when necessary. The experience of other people should be considered as one of the resources available to a troubleshooter.

In summary then, there are three factors to keep in mind it comes to troubleshooting. When confronted with a problem, technicians must make sure that they:

- have an adequate knowledge and understanding of the machine concerned,
- use common sense and a step-by-step approach, and
- draw on their own experience and that of others when required.

AIDS TO TROUBLESHOOTING

It is unlikely that problems will be solved quickly and effectively if the relevant information is not available. Therefore the first step in troubleshooting a problem to which there is no obvious solution is to collect all the necessary data to solve the problem. The data required falls into two categories and may be either background data (information regarding the function, design characteristics, maintenance instructions, etc. of the machine), or operational data (information regarding the running conditions at the time of breakdown). In both cases, the maintenance technician who is required to troubleshoot the problem must be aware of all the potential sources of such data.

Background data sources

Manufacturers' information

Most equipment items are provided with an operating and maintenance manual and a set of engineering drawings from which information can be extracted. Additional information can usually be provided by manufacturers on request.

Maintenance history

Most companies keep some sort of record describing the history of equipment items which can yield useful information regarding previous problems. Where no formal record is kept those who have been involved with the equipment may provide information from memory.

Systems drawings

Electrical, hydraulic and pneumatic schematics and control systems drawings are essential tools when troubleshooting systems problems. The ability to read and interpret such drawings is a vital skill that must be developed by those involved in problem solving.

Process drawings

There are various types of systems schematics that can be of great assistance. Process flow diagrams and piping and instrument diagrams (P.&I.D.s) provide useful information regarding the relationships and interconnections between all process equipment items. It is often important to have an understanding of the operation of related items and systems that may have contributed to the problem.

Troubleshooting charts

Manufacturers and technical literature sometimes provide troubleshooting charts designed to help identify the potential causes of common mechanical problems. Although these tend to be very general, they can be a useful starting point in the troubleshooting process.

Operational data sources

Operating records

If an operations log sheet or other form of record is kept, this may provide important information regarding the operating condition of the machine prior to and at the time of failure. Continuous chart recorders are especially useful in helping to determine whether any sudden changes have occurred.

Observers' reports

It is often necessary to interview operating personnel who were in charge of the machine prior to or at the time of failure. Other personnel may also have been in the vicinity of the machine and may be able to provide data. The process of extracting information from others is not always easy and often requires particular communicating skills. These are discussed more fully in 'Communication' on page 126.

Test readings

If the machine is still operable, valuable data can be gained from readings taken by specially installed test equipment. Such test equipment may include multimeters, pressure gauges, temperature gauges, flowmeters, tachometers, dial indicators, etc., which should be installed in positions that yield the most useful information. It is not always necessary to install additional instrumentation to generate this data because use can often be made of in-line instruments.

Condition monitoring equipment

As described in Chapter 10, considerable advances have been made in recent years in the field of fault detection using various types of equipment that can be used to monitor equipment condition either on a continuous or spot-check basis. If this equipment is available it may be the source of invaluable data for the troubleshooter.

Metallurgical analysis

The examination of failed components using metallurgical techniques may generate important information. Microscopic examination of failed surfaces can help to determine the type of failure that has occurred and non-destructive testing methods such as radiographic and ultrasonic inspection, magnetic particle inspection and liquid penetrant inspection can yield information about material structure and condition that may help determine cause of failure.

INVESTIGATION GUIDELINES

Before undertaking the troubleshooting process bear in mind the following guidelines that are recommended to help avoid simple mistakes that can seriously interfere with finding a solution.

- Start the investigation as soon as possible after breakdown has occurred. Time may obscure or obliterate vital evidence.
- Ensure that no evidence is destroyed. Do not disturb or tidy up the scene of failure until a proper examination has been made.
- Collect important pieces of evidence for more detailed examination. Handle and pack them carefully so they are not further damaged accidentally. Accurately identify the parts and components collected.
- Do not be too narrow in the investigation. Check out the surroundings and environmental conditions and approach the point of failure gradually. The cause of failure can often be remote from the point of failure itself.
- Avoid guesswork and drawing easy conclusions unless they are rigorously checked out. Remember that a cause has been established not when it becomes obvious, but when all other possibilities have been eliminated. Be sceptical and always cross-check vital evidence, especially that collected in the form of statements and opinions. People's judgments and perceptions are all fallible and can be subject to subconscious prejudices. The instinct for self-preservation in particular can significantly affect people's ability to be objective.

COMMUNICATION

During the troubleshooting process much of the data will be provided by other people in the organisation, particularly operating personnel, and consequently the technician needs to be skilful at extracting and interpreting information. A technician must first of all know where to go for information and, secondly, how to get it. An awareness of the functions and responsibilities of others in the organisation, particularly those involved with the machine in question, is important as is being on good terms with others and having their confidence.

Very often the key to making sure that the necessary information is forthcoming lies in **asking the right question**. The ability to ask the right questions will depend to a certain extent on a sound knowledge of the equipment and its mode of operation and function. This knowledge will greatly assist the questioner in knowing what to look for.

Information collected should be checked and confirmed wherever possible. Incorrect data can easily be supplied, either deliberately or inadvertently, and it is always wise to cross-check.

The troubleshooter must be prepared to probe thoroughly for what operating personnel may consider irrelevant information. The slight variation in operating procedures that may be adopted by different personnel may well have contributed to the problem unbeknownst to the operator. If the problem is an unusual one without apparent precedent, questions should lead towards establishing what changes took place around the time of breakdown and what was done differently that might have affected the equipment.

It must be remembered that communication is a two-way process. It is one thing to ask the right questions but it is another thing to correctly interpret the answers. Hence **listening** becomes a key element in the process. The first thing to be sure of is that the question has been understood. It is then important to make sure that the answer given actually answers the question asked. This will require careful listening and some cross-checking and confirmatory questions.

The importance of clear communication during the troubleshooting process cannot be over-emphasised. The solution to many a problem has been overlooked because of what somebody 'thought' was said.

SYSTEMATIC DIAGNOSIS

It is generally accepted that the process of troubleshooting can be systematised into a series of fundamental steps that are applicable to fault location for all types of machinery.

These basic steps must be worked through sequentially and can be described as follows:

Problem analysis

This step primarily involves collecting information about the fault so that the problem can be defined as accurately as possible. It is not anticipated that the machine should be dismantled at this stage and, in fact, it may still be in operation. All known information from data sources listed in 'Aids to troubleshooting' should be reviewed so that the nature of the breakdown can be described as accurately and precisely as possible. For instance the term 'pump failure' is a very general description of a problem which does not give much of a lead in the process of troubleshooting. However 'pump shutdown due to excessive seal leakage' or 'pump shutdown due to bearing overheating' defines the problem in much more detail and gives a positive indication of the direction the troubleshooting process should take.

Preliminary inspection

Once the problem has been defined, a more detailed inspection of the equipment can be carried out. In particular, the general area in which the fault is most likely to occur should be investigated. In many cases, where the problem is relatively straightforward, the cause of breakdown may be immediately apparent in which case a repair can be immediately undertaken. If this is not the case, then a number of preliminary questions should be considered before the investigation proceeds further. Such questions include:

- Is there a fault-finding guide for the equipment?
- Have there been any changes or modifications to the machine recently?
- Has a similar fault occurred before?

If the answer to any of these questions is 'Yes', this opens up particular lines of inquiry which should be pursued before the process advances further. If the answer is 'No', then the investigation must move on to the next step.

Fault zone location

If the fault has not been located by this stage then the equipment should be mentally divided into functional zones which can each then be checked for operation. For example, if a petrol engine will not start and the problem is not obvious, it is normal practice to isolate the problem to either the fuel delivery system or the ignition system by checking each separately.

The key to locating the fault zone is to check inputs and outputs rather than to examine the zone itself. For example, if it is necessary to check fuel supply to an engine it is better to disconnect the fuel lead and check fuel flow rather than dismantle the carburettor. The troubleshooter must establish key points in the system where tests can be made to eliminate those zones where operation is unaffected by the fault until, finally, the fault is isolated to one particular part of the machine or system.

Zone investigation

Once the fault has been traced to a particular zone or system then a more thorough investigation can begin. In the case of a single machine element such as a carburettor, the components may now need to be dismantled and examined. If the fault has merely been isolated to a particular circuit or system, then input and output checks may again be required to pinpoint the particular element in the circuit where the fault lies. If it is not possible to make test measurements then it may be possible to eliminate individual elements by bypassing them or substituting components that are known to work.

The more elements that can be eliminated as operating correctly the simpler it becomes to find the faulty element.

Finding the cause

The purpose of troubleshooting, it should be remembered, is not just to locate the fault but also to find the cause. If this is not done then a repair may be made and the machine put back into service and a similar breakdown may recur within a very short time.

As a guide to identifying the cause of failure it is useful to recognise that the cause can be classified according to the manner in which the failure develops:

Wear-out failures Failures attributable to the normal proceses of wear as expected when the component was designed.

Misuse failures Failures attributable to the application of stresses beyond the item's design capabilities.

Inherent-weakness failures Failures attributable to a lack of suitability in the design or construction of the component when subjected to stresses within its stated capabilities.

Recognition of the cause classification will be an important factor in determining what corrective action should be undertaken so that the real cause can be treated.

Replacement or repair

The decision of whether to replace or repair the faulty component may depend on the overall maintenance strategy of the organisation and the downtime involved. If a component is repaired then it should be workshop tested, if possible, before it is reinstalled.

If failure was identified as being due either to misuse or inherent weakness then the repair made must include action to avoid repetition of the failure from the same cause. This may involve changes or modifications to other parts of the machine or system, or may involve the selection of different materials.

Performance checks

Once the repair has been completed it is essential that the performance of the machine is checked to ensure that the fault has been eliminated and that the machine is functioning satisfactorily. Before finally returning the machine to service it is wise to ask the operator to test it to make sure that it is operating correctly.

The systematic approach described above can also be shown in the form of a flow chart. (Fig. 11-1)

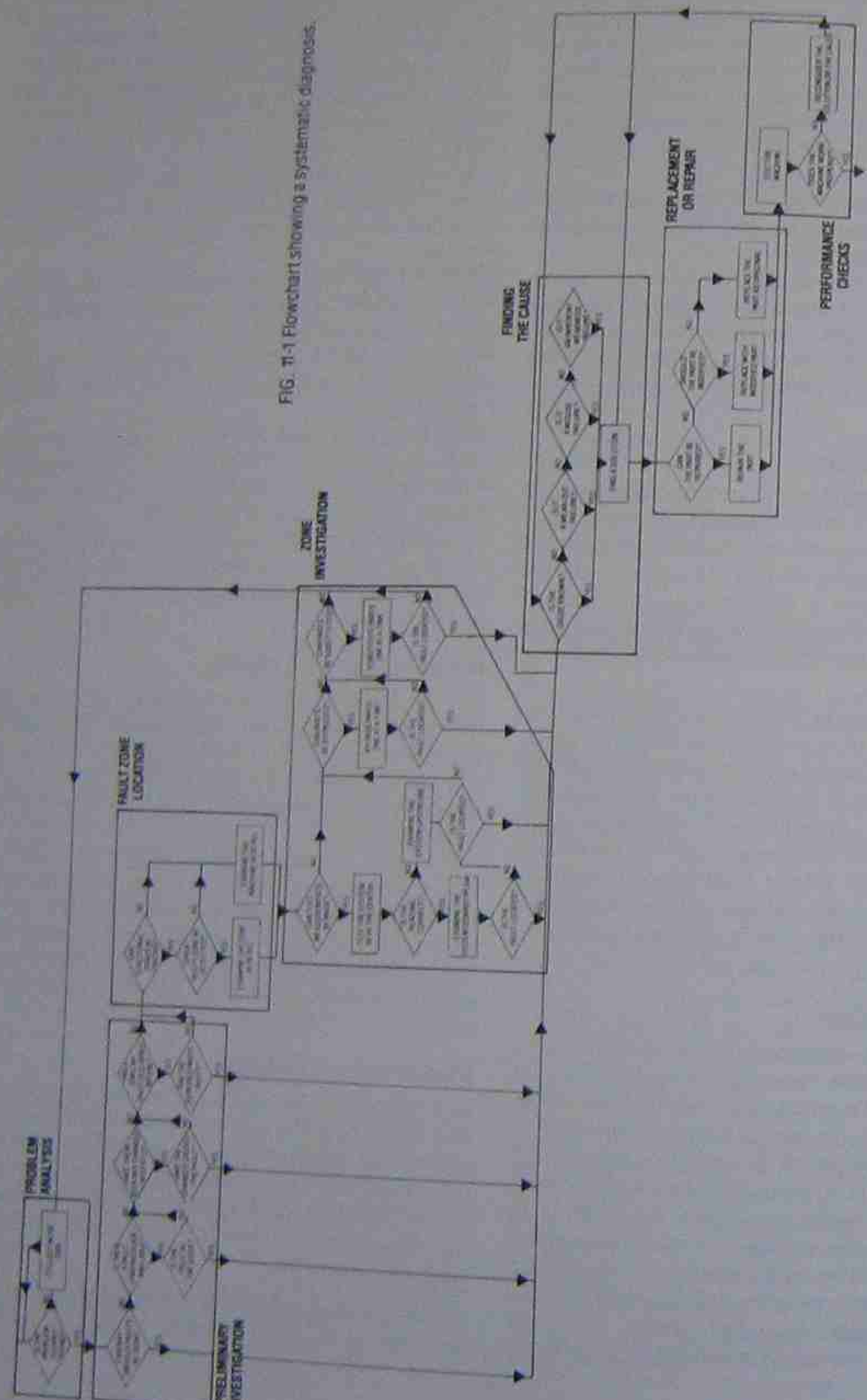


FIG. 11-1 Flowchart showing a systematic diagnosis.

CHAPTER 12

ATTITUDES TOWARDS MAINTENANCE

As Robert M. Pirsig suggests in *Zen and the Art of Motorcycle Maintenance*, the manner in which maintenance work is approached can make all the difference between success and failure. It is often said that someone has the right temperament for a particular job and there is, in fact, a certain kind of temperament that is best suited to maintenance work. The point is that few people are born with the ideal temperament for any particular job but it is possible to learn to control responses and to develop those aspects that are best suited to the job.

Peace of mind

Maintenance work requires thought and attention. The work can often be very exacting, involving fine tolerances, complex assemblies and demanding problems to solve and it requires considerable concentration. Ideally the technician should become involved in the work to such an extent that other thoughts are excluded for the duration of the job.

The prerequisite for good concentration is peace of mind. The ideal state of mind in which concentration can develop is one that is uncluttered with stray thoughts, worries or anxieties, and able to focus solely on the problem at hand. It is rare to achieve sustained peace of mind but it is possible to learn to clear the mind for short periods and to push aside distracting thoughts and emotions. When actors take the stage they leave behind their worries and frustrations, and attempt to become the characters they are portraying. This requires a deliberate mental trick of emptying the conscious mind of the normal stream of thoughts and emotions and replacing them with the details and characteristics of the role to be played. Maintenance technicians should learn to play the same mental trick before undertaking a particular task. They must switch into the role of maintenance technician, empty their minds of all distractions and leave themselves free to concentrate on the task ahead. Peace of mind is a state in which the mind is uncluttered and receptive and free to apply itself to the task ahead.

The other important factor in achieving peace of mind is self-confidence. This was discussed in Chapter 11 with respect to troubleshooting and the same analysis applies to maintenance work as a whole. Self-confidence develops with a knowledge of the subject. Maintenance work has become more demanding as equipment has become more sophisticated and technicians must be prepared to continually upgrade their skills by doing research and asking questions. When a new machine is installed, borrow the maintenance manual and check it over, then when a job comes up you will feel more confident. Before starting any job about which there are doubts, time should be spent asking questions and checking through whatever technical information is available. Often the doubts can be overcome and the job then commenced with self-confidence and peace of mind.

It is almost impossible for anyone to concentrate and apply themselves properly to a task if some major catastrophe has just occurred in their personal lives. In this situation people are likely to make mistakes and hence be a danger to themselves and to others. Under these circumstances people should be treated as if they had a physical injury and put on light duties. In other words, they should be given tasks that are not mentally demanding until they recover. Mental injuries can be far more debilitating than physical ones but, regrettably, this point is often not recognised by employers.

Method

Just as the troubleshooting process demands a methodical approach, so does maintenance work in general. Complex and demanding tasks cannot be approached in a haphazard way if they are to be performed efficiently and without mistakes. There is always a shortest route between the beginning and end of a complex task and the first step that must be taken to find this path, in all cases, is to **stop and think**. This is the cornerstone on which a methodical approach is built and technicians should constantly remind themselves of the need to plan their work and to carefully consider the requirements of each step before proceeding.

For extremely complex tasks technicians can receive help from the Maintenance Planning Department who may go to the extent of producing a critical path network if it is warranted. In most cases, however, they must do their own planning and must try to anticipate the needs of the job in advance and equip themselves to cope with each phase of the job as it arises.

A few minutes spent at the beginning of a job thinking it through may save hours of work later.

Enthusiasm

Enthusiasm is the energy source that drives us along until the task is completed. There should be plenty of enthusiasm available at the beginning of a job but if it is an unpleasant one that does not generally arouse enthusiasm, then some sort of psyching-up process may be necessary. There are plenty of examples of other activities where this process is necessary, especially in sport, and it should not be considered as odd or inappropriate for maintenance technicians to psyche themselves up for a particularly unpleasant or difficult job any more than for an athlete to do so before a world record attempt.

The problem with enthusiasm is that it can easily start to leak away if things go wrong, and once it has dissipated, it is very difficult to replenish. The secret is to find a way of maintaining sufficient enthusiasm to carry through to the end of the job. The way this can best be done is to avoid, wherever possible, the situations that lead to loss of enthusiasm and the principle weapon in this struggle is the one described in the previous section – **method**.

The types of situations that can occur, but that can be avoided with good planning and a methodical approach, are such things as out-of-sequence assembly and unavailability or unsuitability of spare parts. Nothing can be more frustrating or have a more disastrous effect on enthusiasm than to reassemble an item and then find a washer or a spring left over and then have to disassemble and reassemble all over again. The key to avoiding this sort of situation is to adopt a methodical approach to the layout of components and to keep checking the assembly drawings during the process.

The same applies to the problem of spares. It is wise to personally check the availability of spares and correctness of part numbers before a job is started. This way it is known in advance what can and cannot be replaced and more care can then be taken to preserve items for which spares are not available.

The other things that can affect the supply of enthusiasm relate again to the mental state of the individual concerned. Impatience can cause severe problems and will frequently cause mistakes to be made. Thus impatience becomes self-defeating and rather than speeding things up can in fact slow things down to a standstill by undermining the supply of enthusiasm. Anxiety is also a mental state which can seriously interfere with performance and cause mistakes to be made, which in turn saps the supply of enthusiasm. Anxiety usually stems from a lack of confidence which causes people to be tentative and indecisive and thus liable to make mistakes.

The answer to avoiding these problems is to ensure that the necessary peace of mind is established before the job is started and to maintain this with self-control as the job proceeds.

If peace of mind can be maintained and a methodical approach adopted, then enthusiasm should last the distance.

Ego

One of the things that can seriously affect someone's ability to see things clearly is their own ego. It is easy to be convinced of one's own correctness and to ignore the advice of others or to ignore vital evidence that does not fit in with the opinion formed. This is a dangerous condition and can lead to mistakes.

It is vitally important that maintenance technicians remain objective and open-minded at all times. If they have a hunch about something they must be prepared to back it up with evidence and if the evidence does not support the theory, then they must be prepared to drop it rather than press on regardless.

Machines demand respect and do not bend to the will of human beings. Most of the time they are predictable but at other times they can be highly idiosyncratic (unpredictable). It is no good standing before the broken down machine and loudly declaring, 'But, it's just *got* to be the such and such' when the evidence is to the contrary. In those impasse situations the technician must be prepared to recognise that it is he or she who is wrong, not the machine, and then continue to search for the vital evidence that has obviously been missed.

It is recommended that maintenance technicians show a little humility in their approach to their work – arrogance will certainly lead to trouble sooner or later.

The mechanic's feel

Finally, something needs to be said about what can be described as the mechanic's feel.

The properties of materials vary widely and not all can be treated in the same manner. Some materials are elastic and tough and can withstand high loads and stresses, others are brittle or soft and must be treated with great care. Technicians must appreciate different properties and handle different materials accordingly. They must be aware, for instance, when dealing with brass fittings that too much torque can easily strip a thread whereas high tensile steel fittings can withstand much higher stresses.

When dealing with threaded fastenings in general, they should appreciate the difference between **finger tight**, **snug** and **tight** and be able to interpret these three conditions in different situations.

The technician must also be aware of how to handle precision surfaces and those with fine tolerances. Both bearings and seals include components machined to high degrees of accuracy and it is vital that the technician treats such components with the respect needed to protect them.

In order to develop a feel for the materials with which they work, technicians must be aware and think about what they are doing. Like so many of the skills associated with maintenance, the mechanic's feel is not something with which the individual is born but a skill that can be learnt. It can only be learnt, however, if the individual cares enough to want to find out, and takes the trouble to observe, to question and to think about what he or she is doing.

CHAPTER 13

SAFETY

Like maintenance, safe working practice is a preventative activity. In the same way that the objective of maintenance is to prevent breakdown, the objective of safe working practice is to prevent accident, injury and loss. Good maintenance technicians are as familiar with, and as proficient in, safe working practice as they are in good engineering practice. In fact, it would be more appropriate to say that safe working practice is an essential element of good engineering practice.

SAFETY-RELATED INCIDENTS

It is usual to use the word *accident* but there is really no such thing. When things go wrong there is almost always an identifiable cause and incidents are rarely, if ever, the result of some mysterious unseen hand. Safety-related incidents are the result of human error and failure to follow correct procedures and, therefore, can be avoided. They are not just 'accidents' over which no-one has control.

There are a number of ways in which incidents can be categorised, the most useful being according to the consequences.

- **Personal injury** Personal injuries resulting from incidents may range from minor cuts and abrasions that require little or no medical attention to serious disablement or death. Injuries that result in lost time are often recorded as indicators of safety performance.
- **Damage to plant and equipment** Incidents that result in minor damage to plant and equipment are often not recorded unless they cause lost production time. Serious incidents may threaten the local environment as well as causing major damage and loss of production.
- **Environmental damage** Incidents that lead to environmental damage are of increasing concern. Discharge of hazardous liquids and gases and disposal of solid waste are the most likely causes of damage.

Serious incidents may result in extensive damage to the environment, plant and equipment as well as injury to personnel.

Causes of safety-related incidents

Safety-related incidents may occur as the result of a number of factors. It is important that the significance of those factors for the performance of the maintenance function is clearly understood.

Error

Maintenance technicians, like other workers, make mistakes. Mistakes can be costly and result in serious injury and loss. There is always a reason why mistakes are made. Often it is due to carelessness or lack of concentration. It may also be due to lack of familiarity or lack of training. Maintenance technicians must take responsibility for their work, concentrate on the job in hand and, if in doubt, seek direction. The goal of every maintenance technician should be to eliminate error from their work.

Poor work practices

Maintenance technicians must follow correct work practices, some of which are generally applicable while others will be specific to the individual workplace. Part of good engineering practice is the application of sound work practices that reduce or eliminate the possibility of error, protect personnel and equipment, and create conditions in which the highest possible quality of work can be performed. Maintenance technicians must be familiar with, and follow, correct procedures.

Poor communication

Many safety-related incidents occur as a result of poor communication. It is the responsibility of maintenance technicians to ensure that they comply with workplace requirements by communicating verbally with those affected by their work and by following other procedures such as the completion of job sheets and reports, work permits and incident reports. Maintenance technicians are required to work closely with production and other personnel, and good communication is crucial to the effective performance of the maintenance function.

Poor management practices

Maintenance work is conducted within an organisational framework and climate that is established by management. It is management's responsibility to ensure that organisational structures, lines of communication, procedures and systems promote a safe working environment. Management is also responsible for the appointment of qualified staff at all levels and for providing adequate training.

Faulty equipment and materials

It is possible that safety-related incidents may occur as a result of faulty equipment or materials supplied to the enterprise. Such incidents can be guarded against by the institution of adequate testing and inspection procedures, particularly in relation to critical items of equipment and specialised process materials. The responsibility for the development of such procedures lies with management, but the implementation may be a maintenance responsibility.

Poor design

Poor process, system and equipment design can easily lead to unsafe working conditions and the potential for serious incidents to occur. Once again, the responsibility for ensuring that design standards are met lies with management. However, maintenance staff must play an important role by informing the design process and by providing a feedback link to ensure that design problems are identified and corrected in the light of plant experience.

By identifying the possible causes of safety-related incidents in this way it is possible to determine ways in which maintenance staff can contribute to the prevention of such incidents. It must be assumed that safety-related incidents will occur unless active measures are taken to prevent them. Maintenance technicians must, therefore, be proactive rather than reactive in their attitudes to safety, and strive at all times to adopt practices and to take measures that are designed to prevent safety-related incidents from occurring. The remainder of this chapter outlines the principal ways in which this can be achieved.

SAFE WORKING PRACTICE

Workplace behaviour

Strict codes of behaviour must be observed in the workplace in order to protect personnel and minimise the risk of damage to plant and equipment. Maintenance technicians may be required to work in any area of a production plant and, therefore, must exercise caution and pay particular attention to their behaviour. The following guidelines regarding general behaviour should be followed at all times:

- Never fool around in the workplace.
- Never run along corridors, on staircases or anywhere else in the plant.
- Do not play practical jokes.
- Do not play with fire, electricity, compressed air, gas or water systems.
- Never interfere with any machinery or equipment unless authorised.
- Never throw things around the workplace.
- Do not enter restricted areas.
- Never interfere with safety equipment.
- Never distract others from their work.
- Always follow regulations requiring the wearing of protective clothing or use of protective devices.

Maintenance technicians should at all times:

- be thoroughly familiar with the work area;
- concentrate on the job in hand, but also be aware of what is happening around them; and
- observe correct procedures.

Maintenance technicians are the 'ambassadors' of the engineering department and their behaviour in a production plant should be exemplary.

Protective clothing and devices

One of the most fundamental aspects of working safely is wearing the correct protective clothing and using appropriate protective devices. Maintenance technicians must be particularly conscientious in this regard because, by being required to work in differing situations, they are exposed to a wide range of conditions and hazards.

The following guidelines should be observed by maintenance technicians at all times.

- Wear plain, tough, close-fitting overalls and keep them buttoned up. Loose or flapping clothes may get caught in rotating machinery.
- Wear any special protective clothing supplied by the employer.
- Wear safety shoes or boots and keep them in good repair.
- Keep long hair under a tight-fitting cap or net.
- Do not wear jewellery on the job.
- Wear personal protective equipment appropriate to the job in hand.

The following protective equipment should be worn when necessary:

Head protection
- Safety helmet

Eye protection
- Safety glasses or goggles

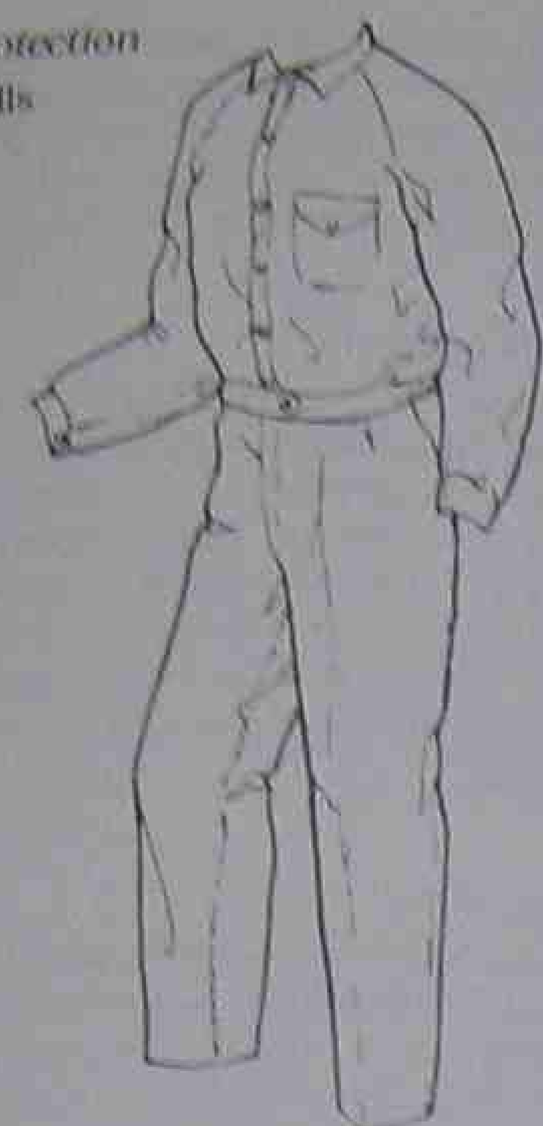
Ear protection
- Ear muffs or plugs

Hand protection
- Strong gloves

Respiratory protection
- Respirator or mask

Foot protection
- Safety shoes or boots

Body protection
- Overalls



All protective clothing and safety devices used should conform to the relevant standards.

Housekeeping

Good housekeeping means keeping the workplace clean, ordered and tidy so that unnecessary hazards are eliminated. It is an important part of working safely and good engineering practice, and involves the following:

- Keeping work areas and benches clean and tidy and free from discarded material.
- Keeping floor areas clean and free from spills of oils or other liquids.
- Returning tools and equipment to their proper places of storage.
- Returning unused materials to the appropriate storage area.
- Disposing of waste and refuse in appropriate receptacles.
- Keeping aisles, accessways and exits clear and free from obstructions.
- Ensuring that all signs and notices can be seen and easily read.
- Ensuring that all necessary safety equipment is kept in good condition and is available for use when required.

Use of hand tools

Good maintenance technicians *never* blame their tools! Tools must be kept in good condition at all times and the right tool for the job must be used. Using an incorrect, improvised or defective tool can lead to safety-related incidents and consequent injury. The following guidelines should be observed:

- Use the correct type and size of tool for the job.
- Check the condition of any tool before it is used.
- Do not use tools that are worn or damaged.
- Keep cutting tools sharp and protect them when not in use.
- Never use a tool for any other than its intended purpose.
- Store and carry tools safely.

Use of portable electric power tools

Electric power tools should be treated with the same respect as hand tools. They must be kept in good condition at all times and used only in the manner intended. Technicians should observe the following guidelines:

- Inspect electric power tools carefully before use. Check the casing, brush caps, switch, lead and plug to ensure that they are not broken or damaged.
- Do not use a tool that overheats and smells of burning.
- Do not carry or suspend a portable tool by its lead.
- When using portable power tools, hold them firmly and securely. Be prepared for the high torque of large capacity tools when they begin cutting or if they jam. Do not put them down until the tool head has stopped rotating.
- When necessary, hold the workpiece firmly in a clamp or vice.
- Do not use tools when they are wet or let tool leads drag through water.
- Protect other workers from sparks, hot metal etc. by using screens or barriers when necessary.
- Protect, cover or signpost tool leads to avoid tripping hazards to others.
- Always use appropriate personal protection.
- Do not leave unattended tools plugged in.

Use of compressed air and compressed air tools

Compressed air can be dangerous if used for anything other than its intended purpose. Maintenance technicians should observe the following guidelines:

- Check that air lines and fittings are in good condition before they are pressurised.
- Hold the end of a hose to stop it whipping when the air is turned on.
- Never use compressed air to clean clothes or any part of the body.
- Never blow down a bench or machine tool with compressed air. Metal filings and chips may be blown 10 metres away.
- Do not allow air hoses to become kinked.
- Follow safe working procedures similar to those for electric power tools for holding, starting, working with, putting down and disconnecting compressed air tools.

Use of explosive powered tools

Explosive powered tools are devices that use an explosive charge (the cartridge) to fire a projectile (the fastener) into hard materials. They can also be very dangerous weapons if used incorrectly. The following guidelines should be observed:

- Observe any warning notice attached to the tool and follow instructions carefully.
- Always use protective shields provided.
- Always use protective eye and ear equipment.
- Protect other workers by displaying a warning notice and using screens and barriers when necessary.
- If a tool fails to fire when triggered, wait for five seconds before removing it from the work surface and unloading it in a safe manner.
- Never attempt to close the breech, or cock an explosive powered tool, by 'crashing' it closed with the hand covering the open end of the barrel.

Machine guards

Maintenance technicians are frequently required to remove, replace and maintain machine guards. Guards are vital for the protection of all workers and maintenance technicians should take a particular interest in their condition and effectiveness. If a technician considers that the condition or design of a machine guard makes it ineffective and constitutes a potential safety hazard, the matter should be attended to or reported to the supervisor.

Danger tags and lock-out

There are several methods by which maintenance technicians are protected against the inadvertent starting of machinery or systems when maintenance work is being carried out.

Danger tags are used to indicate that certain switches or valves must not be operated. The technician working on a machine or system must attach a danger tag to the control switch or valve that supplies power or other input. The name of the technician, date and time should be written on the tag. Only the technician who attached the tag and signed it may remove it. When more than one technician is working on a machine or system they should each attach and remove their own tags.

When a danger tag is considered insufficient protection, equipment should be locked out or isolated by more positive means. Electric circuits can be isolated at the switchboard and the main switch locked in the off position with a padlock. Similarly, valves can be padlocked in the off position. Danger tags are also used in conjunction with locks, and the responsibility for the attachment and removal of both lies with the technician working on the equipment.

To protect maintenance technicians and other workers from hazardous liquids and gases, spade blanks can be installed in pipelines. These should also be used in conjunction with tags to identify the technician concerned.

Every plant will have its own standards and procedures regarding tagging and lock-out of equipment. Maintenance technicians must follow correct procedures at all times.

Work permits

In order to gain approval to carry out work in a plant or plant area a maintenance technician will normally require a permit to work issued by the production department. This permit will ensure that all necessary safety procedures have been complied with and that it is safe for work to proceed. In some cases special permits to enter vessels or carry out specific types of work may be required in addition to a general permit.

The permit system is designed to protect all workers associated with an operation. It ensures that no work can commence until all necessary safety procedures have been complied with and that no plant or equipment can be restarted until all work is satisfactorily completed and there is no danger to maintenance or other workers.

Maintenance technicians must observe the permit system at all times and recognise that it exists for their safety.

Emergency procedures

Maintenance technicians may be working anywhere in a plant when an emergency occurs and so must be familiar with the emergency procedures that apply in each area. Evacuation procedures, assembly points, location of safety equipment, nature of local hazards, and location of accessways and exits should all be clearly understood.

Maintenance technicians should request special training sessions on emergency procedures if they are asked to work in an area that is unfamiliar to them or if they are in doubt about how to protect themselves and others.

HAZARDOUS MATERIALS

Maintenance technicians are likely to encounter many kinds of hazardous materials and it is vital that they are able to identify them and recognise their properties. An international code of symbols is used to indicate the general properties of substances in order that they can be handled appropriately.

Class 1: Explosives

These are substances or articles manufactured or used to produce a practical effect by explosion or a pyrotechnic effect, for example: gunpowder, gelignite, fireworks, fuses and detonators.

Class 2: Gases

Class 2.1: Flammable gases Flammable gases are those that ignite on contact with a source of ignition. Most flammable gases are heavier than air, and as such will flow to low areas, such as drains, pits, valleys, etc. Some gases have a subsidiary risk classification: poison (2.3) or corrosive (8), etc. Examples are acetylene (dissolved) and liquified petroleum gases (LPG).

Class 2.2: Non-flammable compressed gases Non-flammable compressed gases are those that within themselves are not flammable when exposed to a source of ignition. Some of these gases are liquified. Generally most non-flammable compressed gases are heavier than air, in some cases up to 6-7 times heavier. Some non-flammable gases can have a subsidiary risk category of oxidising (5.1) or corrosive (8). Examples are air, refrigerated liquid and oxygen (liquid).

Class 2.3: Poison gases Poisonous gases are gases which are liable to cause death or serious injury to human health if inhaled. Most poison gases have a perceptible irritating odour. Some of these gases can also have subsidiary risks such as being flammable (2.1), oxidising (5.1), corrosive (8) or can in some cases be both oxidising and corrosive. Generally most poison gases are much heavier than air. Examples are chlorine (gas), methyl bromide and nitric oxide.

Class 3: Flammable liquids

Flammable liquids ignite on contact with a source of ignition and have a flash point not higher than 61°C. Substances which have a flash point above 61°C are not considered to be dangerous by virtue of their lower fire hazard. The vapours from all substances of Class 3 have the property of a more or less narcotic effect, and prolonged inhalation may result in unconsciousness or even death. Examples are petrol, kerosene and paint thinners.

Class 4

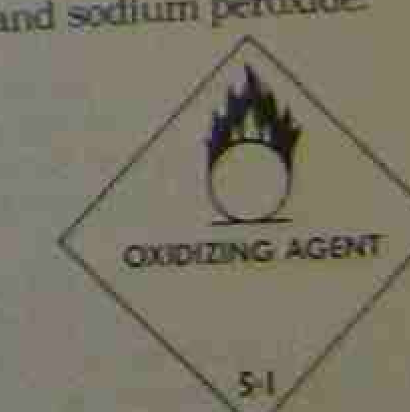
Class 4.1: Flammable solids The substances in this class are solids possessing the properties of being easily ignited by external sources, such as sparks and flames, and of being readily combustible or of being liable to cause or contribute to fire through friction. Examples are sulphur, phosphorus and picric acid.

Class 4.2: Spontaneously combustible substances The substances in this class possess the common property of being liable spontaneously to heat and to ignite. Some of these substances are more liable to spontaneous ignition when wetted by water or in contact with moist air. Some may also give off toxic gases when they are involved in a fire. Examples are carbon, charcoal (non-activated) and carbon black.

Class 4.3: Dangerous when wet The substances in this class are either solids or liquids possessing the common property, when in contact with water, of evolving flammable gases. In some cases these gases are liable to spontaneous ignition due to the heat liberated by the reaction. Some of these substances also evolve toxic gases when in contact with moisture, water or acids. Calcium carbide is an example.

Class 5

Class 5.1: Oxidising substances These are substances which, although in themselves not necessarily combustible, may, either by yielding oxygen or by similar processes, increase the risk and intensity of fire in other materials with which they come into contact. Oxidisers may cause fire when brought into contact with finely divided combustible materials and may burn with almost explosive violence. Examples are calcium hypochlorite (swimming pool 'chlorine') and sodium peroxide.



Class 5.2: Organic peroxides These materials may be either liquids or solids. They support the burning of combustible materials. Under prolonged exposure to fire or heat, containers of these materials may explode. Many organic peroxides may react dangerously with other substances. Violent decomposition may be caused by traces of impurities such as acids. Decomposition of these substances may give rise to evolution of toxic and flammable gases. Examples are benzoyl peroxides and methyl ethyl ketone peroxide (MEKP).

Class 6: Poisons

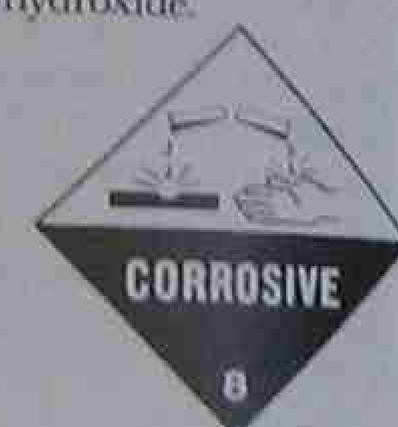
Poisons are substances which are liable to cause death or serious injury to human health if swallowed, inhaled or by skin contact. They are divided into toxic substances (Class 6.1 (a)) and harmful substances (Class 6.1 (b)). These substances can be in solid or liquid form. Nearly all toxic substances evolve toxic gases when involved in a fire or when heated to decomposition. Examples are calcium cyanide and lead arsenate.

Class 7: Radioactive substances

This class includes materials or combinations of materials which spontaneously emit radiation. An example is uranium.

Class 8: Corrosives

These are substances which are solids or liquids possessing, in their original state, the common property of being able more or less severely to damage living tissue. Many substances are sufficiently volatile to evolve vapour irritating to the nose and eyes. A few of these substances may produce toxic gases when decomposed by very high temperatures. Also some substances in this class can be toxic. Poisoning may result if they are swallowed. Examples are hydrochloric acid and sodium hydroxide.



Index

abrasive wear, 17
accidents, 138
adapter sleeves, 41
adhesive wear, 17
adjustable mountings, 10
alignment *see* shaft alignment
anti-friction bearings *see* rolling element bearings
axial drive-up, 45
axial seals, 96

backlash, 69
backstopping clutches, 85
balanced seals, 106
balancing, 6, 12–14
  principles, 12
ball bearings, 37
  load zones, 52
bearing crush, 29
bearings, 6, 25–53
  definition, 25
  loads, 25
  lubrication of inaccessible bearings, 23
  types, 25
bedplates, 8–10
bevel gears, 70
body protection, 140
boundary lubrication, 18
breakdown
  definition, 1, 129
  design and material changes, 2, 3
brinelling
  rolling element bearings, 50
bushes and sleeves, 28–9
bushings, 26, 123–4
  maintenance, 124
  types and materials, 123

carbon ring seals, 120–1
  in mechanical end face seals, 111, 114
centrifugal clutches, 84
chain couplings, 78
chain drives, 61–7
  alignment, 63
  failure, 65–7
  installation, 63–4
  lubrication, 64–5
  maintenance, 63–5
  principles of operation, 61–2
  removal and cleaning, 64
  slack adjustment, 63
  speed formula, 62
  types, 62
circulating oils, 22
circulating system of lubrication, 20
circumferential split ring seals, 120–1
  maintenance, 121
  sealing principles, 120
  types, 120
clearance seals, 96
clutch actuation, 83
clutches, 82–7
  alignment, 85
  assembly, 86
  failure patterns, 86–7
  lubrication, 86
  maintenance, 85
  principles of operation, 82
  types, 83–5
cold flow *see* plastic flow
compressed air tools, 141
condition monitoring, 2, 126–8
cone clutches, 84
contact-free sprag clutches, 85
contact seals, 96
corrosion
  chain drives, 66
  plain bearings, 34
  rolling element bearings, 51
corrosives, 144

danger tags, 141
dangerous when wet substances, 143
dial indicators, 89
  alignment, 90–5
  setting and levelling, 9
diametral clearance
  carbon ring seals, 120
diametral pitch, 69
disc couplings, 78
dog-tooth clutches, 83
drip-feed oiling, 19
driven machines, 5
driving machines, 5
dropping point, 21
dry fluid clutches, 84
dry friction, 18
dynamic balancing, 13–14
dynamic friction, 16
dynamic seals, 96

ear protection, 140
elastohydrodynamic lubrication, 18
elastomeric couplings, 78
emergency procedures, 142
emulsification, 21
end cap and shims, 41
end plate and locking bolts, 40
evaporation
  use of of solid lubricants, 23
exfoliated graphite, 116–17
explosive powered tools, 141
explosives, 142
eye protection, 140

face and rim alignment, 90–1
false brinelling *see* fretting
fatigue, 17
  gear drives, 74
  plain bearings, 33
  rolling element bearings, 50
felt seals, 122
  maintenance, 122
  sealing principles, 122
fire point, 21
flammable gases, 142
flammable liquids, 143
flammable solids, 143
flash point, 21
fluid coupling, 84
fluting
  rolling element bearings, 51
foot protection, 140
fretting, 17
  plain bearings, 34
  rolling element bearings, 50
  use of solid lubricants, 23
friction, 15–17
  causes, 15–16
  clutches, 82
  force formula, 16
  types, 16
full-film lubrication, 18

galling, 17
  chain drives, 66
  plain bearings, 33
  rolling element bearings, 51
  use of solid lubricants, 23
gaskets, 97–101
  assembly load formula, 97
  bolt force formula, 97
  compressive load formula, 97
  installation and joint assembly, 100
  joint disassembly, 100
  maintenance, 100–1

materials, 98-9
sealing principles, 97-8
gaskets, liquid *see* sealants
gear couplings, 78
gear drives, 68-76
alignment, 71
failure, 71-6
lubrication, 71
maintenance, 70-1
principles of operation, 68-9
speed formula, 69
types, 69-70
gear geometry, 69
gear oils, 22
glazing
clutches, 86
graphite, 23
gravity lubrication, 19
greases, 22-3
application, 23
categories, 22-3
measuring of properties, 21
grid couplings, 78
guards (machinery), 6-7, 79

hand protection, 140
hand tools, 140
hardness of greases, 21
hazardous materials, 142
head protection, 140
helical gears, 69-70
herringbone gears, 70
housekeeping, 140
hydraulic balance, 106
hydrodynamic lubrication, 18
hydrostatic lubrication, 18
hypoid gears, 70

indexing clutches, 85
involute curve, 70

jaw couplings, 77
journal bearings, 25
journal bearings, plain, 26-7
clearances, 29-30
relining, 30-1
journal bearings, rolling element, 38

labyrinth seals, 124-5
maintenance, 125
sealing principles, 124
types and materials, 124
ladder bearings, 39
linear bearings, 25
linear bearings, plain, 27
linear bearings, rolling element, 39
lip seals, 118-19
failure, 119
maintenance, 119
sealing principles, 118
types and materials, 118
liquid lubricants, 22
load zones
ball bearings, 52
local welding, 16
lock-out, 141
lubricants, 17-9, 21-4
additives, 24
categories, 22-4
measuring of properties, 21
selection, 18-19
types, 21
lubrication, 6, 15-20
application, 19-20
film conditions, 17-8
principles, 15, 17

machine and engine oils, 22
machine guards, 141
maintenance
attitudes and skills required, 3-4, 135-7
condition monitoring, (human), 2
definition, 1
keeping of records, 2, 131
organisational backup, 4, 131
performance criteria, 1
planned maintenance, 2-3
routine maintenance, 2
satisfactory condition requirements, 1
manual lubrication, 19
manufacturers' recommendations, 2, 131
material-flexing couplings, 77-8
mechanical end face seals, 105-14
assembly, 109-11
environmental control, 109
failure, 111-14
maintenance, 109-11
materials, 106
principles of operation, 105-6
start-up, 111
types, 106-9
mechanical-flexing couplings, 78-9
metal forming dies
lubrication, 24
mixed-film lubrication, 18
molybdenum disulphide, 23, 24
mounting, 6, 8-11
definition, 8
grouting, 10
setting and levelling, 8-10
special mountings, 10-11
multi-plate friction clutches, 83

noise
chain drives, 65
gear drives, 72
plain bearings, 32
rolling element bearings, 49
shaft couplings, 80
noise monitoring, 127
non-flammable compressed gases, 142
nuclear radiation and lubricants, 23

O-rings, 102-3
failure, 103
in mechanical end face seals, 105, 111, 114
maintenance, 102-3
sealing pressure formula, 102
sealing principles, 102
oil injection, 43, 45
oils
categories, 22
measuring properties, 21
operating conditions, 1, 6
operating temperatures
chain drives, 65
gear drives, 72
plain bearings, 26, 32
rolling element bearings, 49
temperature monitoring, 126
use of solid lubricants, 23
organic peroxides, 144
overheating, *see* operating temperatures
overrunning clutches, 85
oxidation resistance of oils, 21
oxidising substances, 143

packings, 115-17
maintenance, 116-17
materials, 116
sealing principles, 115
partial bearings, 26-7
particle retrieval, 127
pin couplings, 77
pitch circle, 61, 62
pitting
plain bearings, 34
rolling element bearings, 51
plain bearings, 25, 26-36
assembly, 28-9
clearances, 29-30
failure, 32-6
lubrication, 26
maintenance, 28-32
materials, 26, 27
principles of operation, 26
relining, 30-1
start-up, 31-2
types, 26-7
plastic flow
gear drives, 74
poison gases, 143
poisons, 144
portable electric power tools, 141
positive engagement, 82
pour point, 21
power transmission, 54-81
predictive maintenance *see* condition monitoring
pressure lubrication, 20
preventative maintenance, 1-3
protective clothing and devices, 139
proximity probes, 127
pumpability, 21

radial seals, 96
radial lip seals, 118-19
radioactive substances, 144
recirculating ball and roller bearings, 39
recirculating roller-chain bearings, 39
refrigeration oils, 22
relative displacement measurement, 127
reverse indicator method of alignment, 91
rigid couplings, 77
roller bearings, 37
roller chains, 61-7
roller clutches, 85
rolling element bearings, 25, 37-53
assembly, 40-1
dismounting procedure, 47-8
failure, 48-53
maintenance, 41-8

mounting
procedure, 43-7
types, 40-1
principles of operation, 37
tools and equipment 42-3
types, 38-40
rolling friction, 16
lubrication, 18
rotating machinery, 5-7
definition, 5
different functions, 5
satisfactory operating conditions, 6-7
run-out see shaft alignment

safe working practice, 139
safety, 138-144
safety related incidents, 138
scoring
gear drives, 73
plain bearings, 33
sealants, 104
maintenance, 104
materials, 104
sealing principles, 104
seals, 6, 96-125
classification, 96
seizure, 17
chain drives, 66
plain bearings, 32
rolling element bearings, 49
use of solid lubricants, 23
shaft alignment, 6, 80, 88-95
methods, 90-5
principles, 88-9
shaft couplings, 77-81
alignment *see* shaft alignment
assembly, 79
failure, 80-1
lubrication, 79
maintenance, 79-81
principles of operation, 77
types, 77-9
shaft nut and locking washer, 40
shutdowns
automatic, 126-8
extended, 2
regular, 2
single plate friction clutches, 83
sleeves, 26, 28-9
slider bearings *see* linear bearings, rolling element
sliding friction, 16
slipping clutches, 86
slotted nut and pin, 40
smearing
rolling element bearings, 51
snaprings, 41
solid journal bearings, 26
solid lubricants, 23-4
spacer couplings, 79
spade blanks, 142
spalling
rolling element bearings, 50
spectrographic oil analysis, 127
spindle oils, 22
spiral wound gaskets, 99-100, 101
splash lubrication, 19
split journal bearings, 26
assembly, 29
spontaneously combustible substances, 143
sprag clutches, 85
spring mountings, 11
spur gears, 69
stability of greases, 21
static balancing, 13
static friction, 16
static seals, 96
statutory regulations, 3
steam cylinder oils, 22
straight edge and feeler gauge method of alignment, 90

temperature monitoring, 126
temperatures *see* operating temperatures
thick-film lubrication, 18
threaded parts
lubrication, 24
thrust bearings, 25
thrust bearings, plain, 27
clearances, 30
thrust bearings, rolling element, 39
total indicator run-out, 89, 92-4
transmission machines, 5, 6, 54-89
troubleshooting, 2, 129-34
aids, 130-1
definition, 129
identification of causes of failure, 17, 133
preliminary guidelines, 131-2
skills required, 3, 129-30, 132
systematic basic steps, 132-4

unbalanced seals, 106

V-belt drives, 54-60
alignment, 56
arc of contact, 54
failure, 58-60
installation, 57-8
maintenance, 56-8
principles of operation, 54-5
speed formula, 55
tension adjustment, 57
types, 55-6
vibration
chain drives, 67
gear drives, 72
plain bearings, 32
rolling element bearings, 49, 53
shaft couplings, 80
vibration isolators, 10-11
vibration monitoring, 127-8
acceptable levels, 122
signature patterns, 122
viscosity, 21
viscosity index, 21

wear (mechanical), 17
chain drives, 65
clutches, 86
gear drives, 72-4
plain bearings, 33
rolling element bearings, 50
shaft couplings, 80
V-belt drives, 58
wear patterns see load zones
wedging action, 82
wiping
plain bearings, 34
withdrawal sleeves, 41
work permits, 142
workplace behaviour, 139
worm gears, 70

Principles of Machine Operation and Maintenance explains how rotating machinery works, and the role of the maintenance engineer in ensuring its proper operation. Stress is laid on the need for the trainee engineer to develop skill in diagnosis and troubleshooting as well as practical expertise in maintenance procedures.

This book is particularly well suited to the needs of students on C&G Schemes 205 and 214. It is also useful for C&G 201 and 228, for BTEC courses at National level and for post-experience training schemes and in-company courses.

Dick Jeffrey teaches mechanical engineering at Footscray College of Technical and Further Education in Australia.

Also available from Butterworth-Heinemann

Basic Engineering Technology (Ed. Timings)
Engineering Science (Bolton)
CAD Systems in Mechanical and Production Engineering (Ingham)
Drafting with AutoCAD (Ingham)

ISBN 0-7506-0293-7

9 780750 602938

BUTTERWORTH HEINEMANN
An imprint of Elsevier Science
www.bh.com

AUSTRALIAN PUMP TECHNICAL HANDBOOK

Produced by the Australian Pump Manufacturers' Association Ltd.

AUSTRALIAN PUMP TECHNICAL HANDBOOK

THIRD EDITION

AUSTRALIAN PUMP MANUFACTURERS' ASSOCIATION LTD.

MTIA House,
214 Northbourne Avenue,
Canberra, ACT 2600
(P.O. Box 817)
Canberra City, ACT 2601)

NATIONAL SECRETARY

Telephone: (062) 47 2233

Contents

Page No.

1.0 DEFINITIONS 2
- 1.1 Units 2
- 1.2 Notation 2
- 1.3 Pumping Terms—
  Pumping, flowrate, total dynamic head, N.P.S.H., pump suction lift, density, specific weight, specific volume, specific gravity, pressure, viscosity.

2.0 PUMP CLASSIFICATION
- 2.1 Rotodynamic— 10
  Centrifugal, peripheral, special.
- 2.2 Positive displacement— 18
  Reciprocating, rotary.

3.0 CHARACTERISTICS OF CENTRIFUGAL PUMPS
- 3.1 Construction— 26
  Impeller, casing, shaft and bearings, axial thrust, radial thrust.
- 3.2 Specific Speed— 34
  Units of specific speed, specific speed and pump type, specific speed and pump efficiency, specific speed and characteristic curves.

4.0 DETERMINATION OF SYSTEM HEAD
- 4.1 Friction losses in pipes and fittings— 38
  Pipe friction, valves and fittings.
- 4.2 Sample calculations of N.P.S.H.A. 43
- 4.3 System Resistance curves— 50
  Effect of displacement of system resistance curve, regulation of flow rate by throttling, regulation by speed, steepness of H-Q curve.

5.0 SELECTION OF CENTRIFUGAL PUMPS
- 5.1 Hydraulic Selection— 56
  Define the pump duty point, calculate NPSHA, select the pump, poor hydraulic selections—their effects.
- 5.2 Mechanical Aspects of Pump Selection. 62

6.0 OPERATION OF CENTRIFUGAL PUMPS
- 6.1 Series Operation 66
- 6.2 Parallel Operation 67
- 6.3 Stable and Unstable Characteristic Curves 68
- 6.4 Affinity Laws 69
- 6.5 Cavitation 72

7.0 PUMP MATERIALS
- 7.1 Preferred Materials 74
- 7.2 Selection of Material 76
- 7.3 Standard Construction Materials— 77
  General purpose water pumps, positive displacement pumps
- 7.4 Plastic Pumps 78
- 7.5 References 79

Page No.

8.0 PIPE SYSTEMS 82
8.1 Suction Pipework 83
8.2 Discharge Pipework 83
8.3 Selecting the Pipe 83
8.4 Valves 83
8.5 Air Vent Valves 84
8.6 Branch Connections 84
8.7 Reduction of Pipe Diameter 84
8.8 Thrust Blocks 85
8.9 Special Considerations 85
8.10 Pressure Surges and Water Hammer 85
8.11 Discharge of Syphon Lines

9.0 INTAKE DESIGN 90
9.1 Intake Design 90
9.2 Option Sump Volumes 90
9.3 Minimum Sump Volume for One Duty Pump System 90
9.4 Minimum Sump Volume for Multi-pump System 93
9.5 Wet Well and Dry Well Pump Stations 93
9.6 Preferred Sump and Intake Designs—
Single sump pumps, multiple pump sumps.

10.0 INSTALLATION AND COMMISSIONING 100
10.1 Foundations 100
10.2 Erection 101
10.3 Coupling Alignment—
Lining up pump and driver, final coupling, major alignment errors, angular alignment, parallel alignment, general, belt driven pumps.
10.4 Pipework Installation 104
10.5 Operation— 104
Before start-up, running, stopping.
10.6 Routine Maintenance 104

11.0 PUMPING DONT'S 108

12.0 PUMP TESTING 110

13.0 CONVERSION TABLES 118

14.0 PUMPING FORMULAE 124

15.0 ENQUIRY INFORMATION 130

Section 1

DEFINITIONS

Section 1—DEFINITIONS

1.1 UNITS

The units used throughout this book are those in common use in the pumping industry and defined in AS1686—Metric Units in Water Supply, Sewerage and Drainage (including pumping) and AS2417—Pumps, The International Acceptance Test Code. In addition, these standards should be read in conjunction with AS1000—The International System of Units (SI) and its Application.

Although it is recommended to work in SI Pump Units, there may be occasions when it is necessary to convert Imperial or Metric Units to SI Units, and tables of conversion factors are given in Section 13.

1.2 NOTATION

The most commonly used pumping terms are:

Term	Notation	Unit customary in practice
Flowrate	Q	L/s
Head	H	m
Pressure	p	kPa
NPSH	NPSH	m
Velocity	v	m/s
Density	ϱ	kg/m^3
Gravitational acceleration	g	m/s^2
Efficiency	η	—
Rotational speed	N	r/min
Area	A	m^2
Absolute viscosity	μ	mPa.s
Kinematic viscosity	ν	mm^2/s
Specific weight	γ	kg/m^2.s^2
Specific volume	$\bar{v}$	m^3/kg
Specific gravity	S.G.	—

1.3 PUMPING TERMS

Pumping:

Pumping can be described as the addition of energy to a fluid to move it along a pipe, into a pressure vessel or to a higher level, i.e. a pump moves fluid from one point to another in a system.

Flowrate:

The flowrate (Q) is the flow (volume of liquid per unit of time) delivered by a pump through the outlet, normally expressed in litres per second (L/s).

Total Dynamic Head:

Total dynamic head (sometimes called differential or generated head) is a measure of the energy imparted to the liquid by the pump, and is equal to the algebraic difference between the total discharge head and the total suction head.

Total dynamic head, where suction lift exists, is the sum of the total discharge head and total suction lift. Where positive suction head exists, total dynamic head is the total discharge head minus the total suction head (Fig. 1.3.1 and 1.3.2).

The total dynamic head against which a pump operates comprises:

(a) Total static head.
(b) Friction losses.
(c) Velocity head.
(d) Entrance and exit losses.

where

(a) Total static head is the difference in elevation between the liquid levels of the suction and discharge. If the pump discharges into a pressure tank, then the total static head is the difference in elevation between liquid levels plus the pressure in the tank expressed in metres of liquid.

(b) Friction head is the equivalent head, expressed in metres of liquid, necessary to overcome friction on the interior surfaces of the pipework system including all valves, bends and fittings.

(c) Velocity head of a liquid moving with a certain velocity is the equivalent static head through which it would have to fall in order to attain that velocity. Velocity head is expressed by the formula:

$$H = \frac{v^2}{2g}$$

where H is the velocity head, m.
v is the average velocity in the pipe, m/s
g is the gravitational acceleration, m/s^2

(d) Entrance and exit losses are usually comparatively small and can be neglected in the majority of industrial applications.

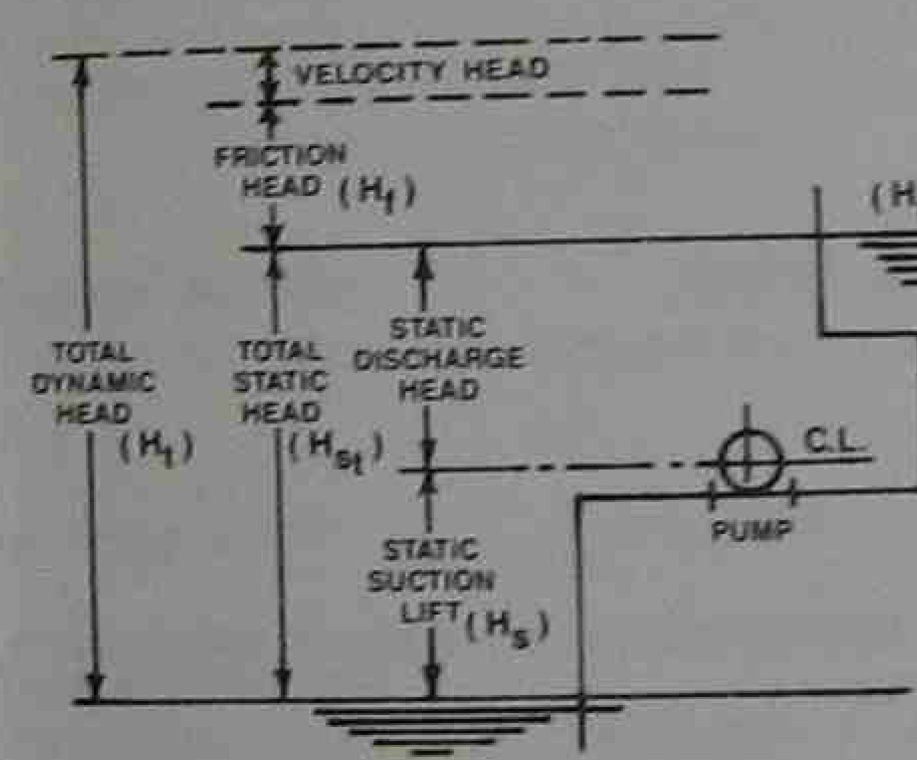


Fig. 1.3.1—Suction lift.

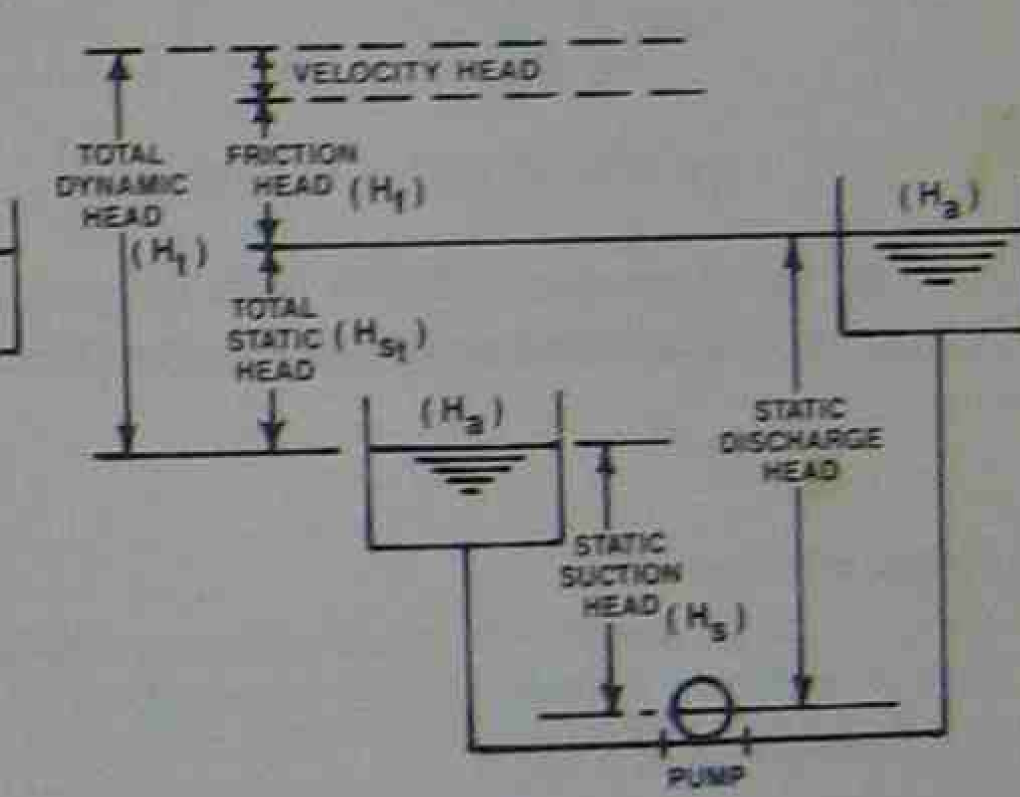


Fig. 1.3.2—Suction head.

Net Positive Suction Head (NPSH)

There are a number of factors which must be considered collectively in order to obtain a complete picture of conditions prevailing at the suction of a pump. The universally accepted practice is to express this calculation in the form of "Net Positive Suction Head" abbreviated as NPSH.

Calculation of NPSH involves consideration of the fundamental fact that every liquid has a vapour pressure which is a function of the liquid and its temperature. Furthermore, if the pressure acting on a liquid is less than its vapour pressure the liquid will boil.

In any pumping system, there is always an absolute pressure available at the suction source. This pressure is reduced in the suction line due to such factors as static elevation, friction and turbulence losses, and energy expended in accelerating the liquid. Finally, there is a pressure drop within the pump itself caused by an increased velocity at the entrance to the impeller and shock losses in the impeller eye.

In order to prevent the liquid boiling, the suction pressure at the pump suction branch must be at least equal to the vapour pressure of the liquid at pumping temperature plus a margin to overcome losses in the pump. This margin of head available above the vapour pressure of the liquid is the nett positive suction head, defined as follows—

NPSH — Nett positive suction head—total head at the pump suction branch over and above the vapour pressure of the liquid being pumped.

NPSHR — NPSH required—is a function of the pump design and is the lowest value of NPSH at which the pump can be guaranteed to operate without significant cavitation. There is no absolute criterion for determining what this minimum allowable NPSH should be, but pump manufacturers normally select an arbitiary drop in total dynamic head (differential head) of 3% as the normal value for determining NPSHR.

NPSHA — NPSH available—is a function of the system in which the pump operates and is equal to the absolute pressure head on the liquid surface (Ha) plus the static liquid level above the pump centreline (negative for a suction lift) (Hs) minus the absolute liquid vapour pressure head at pumping temperature (H_{vap}) minus the suction friction head losses (Hf)

i.e. $NPSHA = H_a + H_s - H_{vap} - H_f$

Alternatively, where the suction pressure head (suction gauge reading) on site is known.

NPSHA — NPSH available at the pump inlet is equal to the absolute pressure head (H_a) plus the suction pressure head referred to pump centreline. (H_{sp}) minus the absolute liquid vapour pressure head at pumping temperature (H_{vap}) plus the suction velocity head $\left(\frac{v^2}{2g}\right)$

i.e. $NPSHA = H_a + H_{sp} - H_{vap} + \frac{v^2}{2g}$

For successful operation NPSHA must be greater than NPSHR.

For sample calculations of NPSH refer Section 4.2

Pump Suction Lift:

The concept of 'suction lift' is normally only applied to atmospheric suction systems, with an open tank or reservoir exposed to one atmosphere of pressure.

For a particular system—

Total Suction Lift (S_L) = Static difference in levels (H_s) + pipe friction loss between revervoir and pump including entry loss (H_f)

= Suction gauge reading corrected to pipe centre line (H_{SP}) and velocity head ($v^2/2g$)

Thus the maximum allowable suction lift is given by:

S_L (Max) = Atmospheric pressure—vapour pressure—NPSHR

Hence the maximum allowable suction lift depends on the local atmospheric pressure, the liquid vapour pressure and the NPSHR of the pump.

The theoretical maximum lift obtainable with an ideal pump with zero NPSHR is equal to atmospheric pressure minus vapour pressure.

Maximum Theoretical Suction Lift with Zero Pump NPSHR

(Basis: Water at 1 standard atmosphere of 101.325 kPa.abs., g = 9.80 m/s)

Extracted from "APMA PIPE FRICTION HANDBOOK" 1982, APMA pp.112, 113.

Temperature °C	Denisty kg/m³	Vapour Pressure kPa	Maximum Suction Lift m
15	999.0	1.704	10.176
20	998.2	2.337	10.119
30	995.6	4.242	9.950
40	992.2	7.375	9.662
50	988.1	12.33	9.190
60	983.3	19.92	8.448
70	977.5	31.16	7.324

For systems with suction vessels at pressures other than atmospheric the NPSH should be calculated—Refer Section 4.2, Examples 5 and 6.

Density:

The density (ϱ) of a liquid is defined as the mass (kg) of that liquid per unit volume (m^3). Thus the units of density are kg/m^3.

For water at 20°C $\varrho = 998.2$ kg/m^3.

Specific Weight:

The specific weight (γ) is defined as the weight per unit volume and is expressed in $kg/m^2.s^2$.

Density and specific weight are related by $\gamma = \varrho.g$

For water at 20°C $\gamma = 998.2 \times 9.8$

$= 9782$ $kg/m^2.s^2$

Specific Volume:

The specific volume $\bar{v}$ of a liquid is the reciprocal of density and is expressed in m^3/kg

$\bar{v} = \frac{1}{\varrho}$

For water at 20°C $\bar{v} = \frac{1}{998.2} = 0.001$ m^3/kg

Specific Gravity:

The specific gravity (or relative density) (S.G.) of a liquid is defined as the ratio of its density at a specified temperature to that of water at some standard temperature. Usually the temperatures are the same and 15.6°C/15.6°C (or rounded off to 15°C/15°C) is commonly used.

$$S.G. = \frac{\varrho \text{ any liquid at specified temperature}}{\varrho \text{ water at } 15.6°C}$$

e.g. for gasoline $\varrho = 749.8$ kg/m^3 @ 15.6°C

for water $\varrho = 999.1$ kg/m^3 @ 15.6°C

$\therefore S.G. = \frac{749.8}{999.1} = 0.7505$

Pressure:

Although it is preferable to express heads in metres of liquid (m) as this is independent of the temperature of the liquid being pumped, head can also be expressed as a pressure in kilopascals (kPa). However, these terms are mutually convertible one to the other as follows:

$$\frac{\text{Pressure in kPa} \times 1000}{\text{Density in kg/m}^3 \times g} = \text{Head in metres}$$

Eg. 1 If the discharge gauge on a pump reads 20 kPa when pumping water at 15°C what is the discharge head in metres?

$$h = \frac{200 \times 1000}{999 \times 9.8} = 20.42 \text{ m}$$

Eg. 2 If in Eg. 1 the pumping temperature was 90°C, the density would be 965.2 kg/m³ hence

$$h = \frac{200 \times 1000}{965.2 \times 9.8} = 21.1 \text{ m}$$

In addition for a given pump, the total dynamic head expressed in metres will remain constant at a given capacity for all fluids (except for viscous liquids). However, the pressure generated (kPa) at a given temperature will be affected by the specific gravity of the fluid pumped.

	SG	Head in m.	kPa
Water	1.0	20	195
Brine	1.2	20	235
Petrol	0.75	20	147

Pressure Terms:

The pressure terms used in discussing pumping applications may be defined as follows:—

a) **Absolute pressure:**
Is the pressure above absolute zero, and is equal to the barametric pressure plus the gauge pressure.

b) **Barometric pressure:**
Is the atmospheric pressure at the altitude where it is measured.

c) **Gauge pressure:**
Is the pressure measured by a gauge and is the pressure above atmospheric pressure at the altitude being considered.

d) **Vacuum:**
Is any pressure below atmospheric, i.e. a negative gauge pressure.

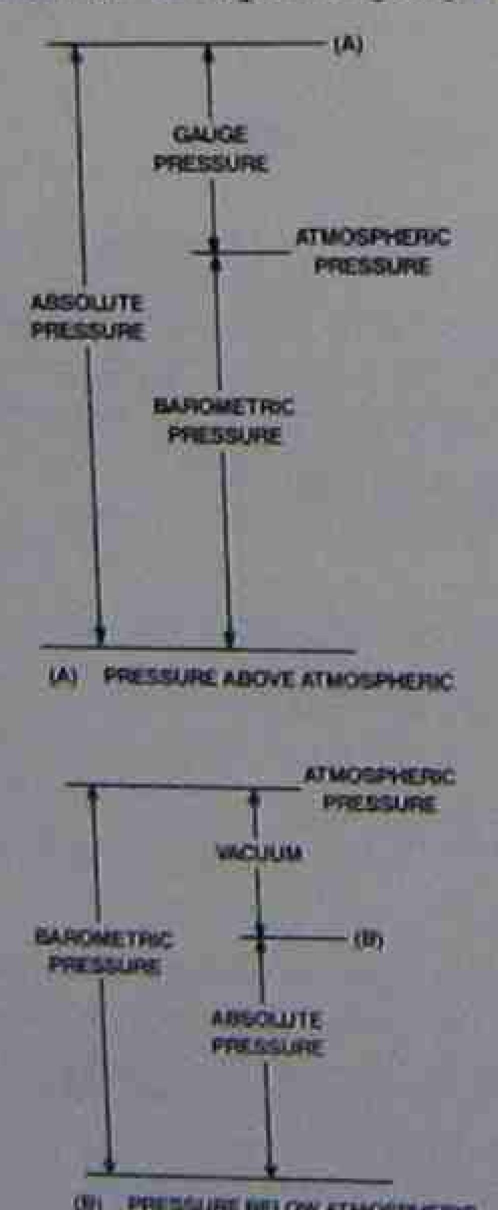


Fig. 1.3.3.

Viscosity:

The viscosity of a fluid (liquid or gas) is that property which offers resistance to flow due to the existence of internal friction within the fluid. This resistance to flow, expressed as a co-efficient of dynamic (or absolute) viscosity is a measure of its tendency to resist internal deformation or shear. Molasses is a highly viscous fluid, water is comparatively much less viscous and the viscosity of gases is quite small compared to that of water.

Dynamic or absolute viscosity "μ" by definition is the ratio of shear stress to velocity gradient in the fluid. The unit of dynamic viscosity is N.s/m² which may be simplified to Pa.s. (Pascal-second). As this unit results in very small values of viscosity for common fluids, it is common practice to express dynamic viscosity in the unit of Centipoise (cP)

1 cP = 10^{-3}Pa.s = 1 m Pa.s (millipascal-second)

Kinematic viscosity "ν" is defined by the ratio of dynamic viscosity to the fluid density (i.e. $\nu = \frac{\mu}{\rho}$) and is commonly used in Reynolds number calculations. The unit of kinematic viscosity is m²/s which also results in very small values for common fluid. It is common practice to express kinematic viscosity in the units of Centistokes (cSt)

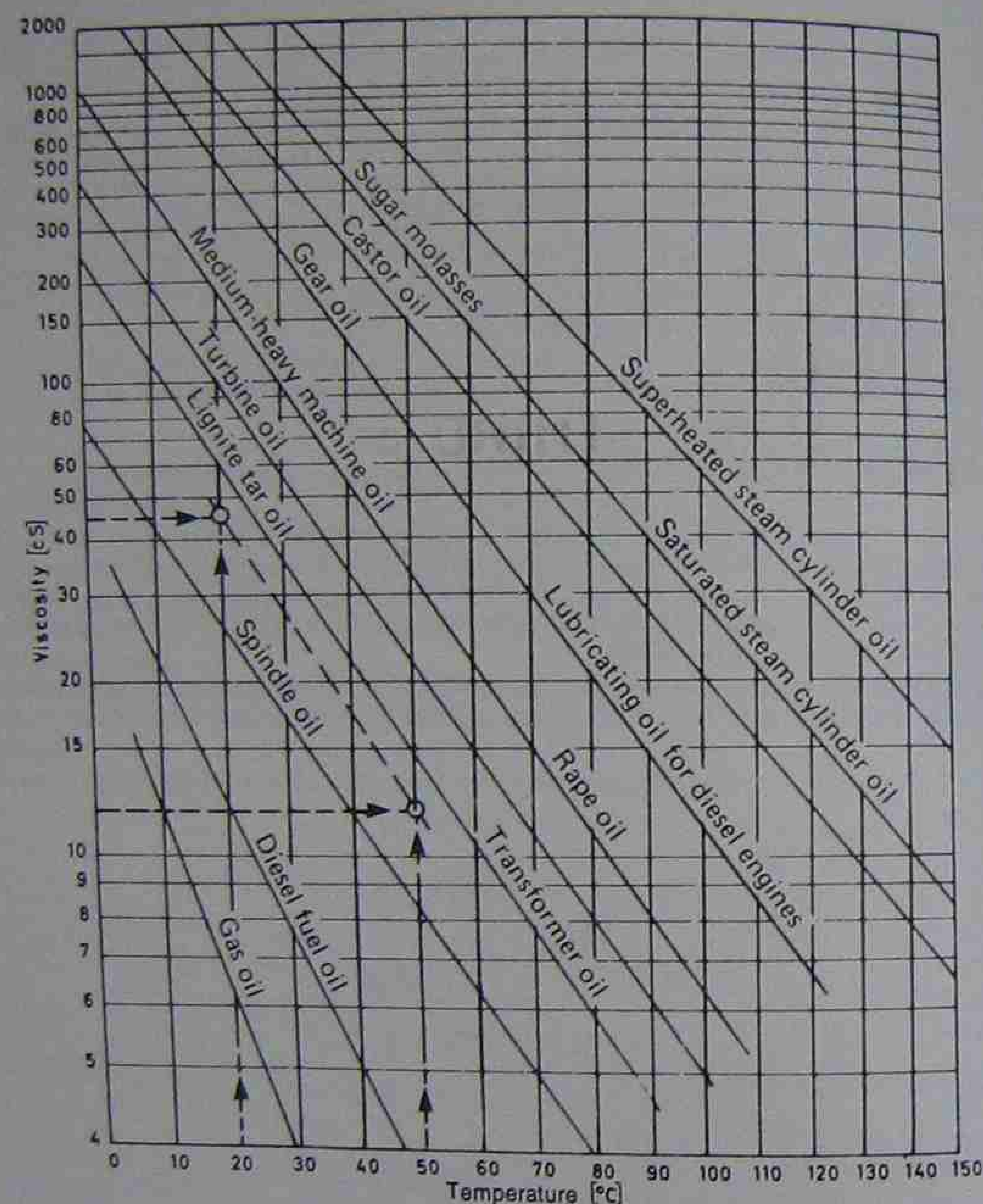
1cSt = 10^{-6} m²/s = 1mm²/s

$$\text{Kinematic viscosity (cSt)} = \frac{\text{dynamic viscosity (cP)}}{\text{density (g/cm}^3\text{)}}$$

The viscosity of the fluid to be pumped should be determined for the range of operating conditions as the performance of a pump is affected by the viscosity which may change with temperature. A reduction in efficiency, increase in power, reduction in head and maximum allowable pump speed, and some reduction in capacity occurs with higher viscosities. Viscosity also affects pipe friction losses, i.e. an increase in pipe friction.

As pump performance curves are normally only available for clean cold water it is recommended that the pump manufacturer be consulted when considering viscous fluid applications.

Effects of Temperature on Viscosity:



The effects of temperature on viscosity. This chart can also be used to estimate the viscosity of other liquids when two sets of corresponding data only are known. These are plotted on the graph and joined by a straight line; the unknown value may then be simply read off. If the viscosity at only one temperature is known, this is plotted on the diagram and a straight line, usually parallel to the adjacent lines, is drawn through this point. All the viscosity-temperature relationships are approximate and do not give accurate values.

Example: Given: Oil having a viscosity of 11.8 cS at (50°C); to find: viscosity at (20°C).
Using the construction given above, the required value is found to be — 45 cS.

NOTE: Viscosity of water at 20°C = 1cS.

Fig. 1.3.4

Section 2

PUMP CLASSIFICATION

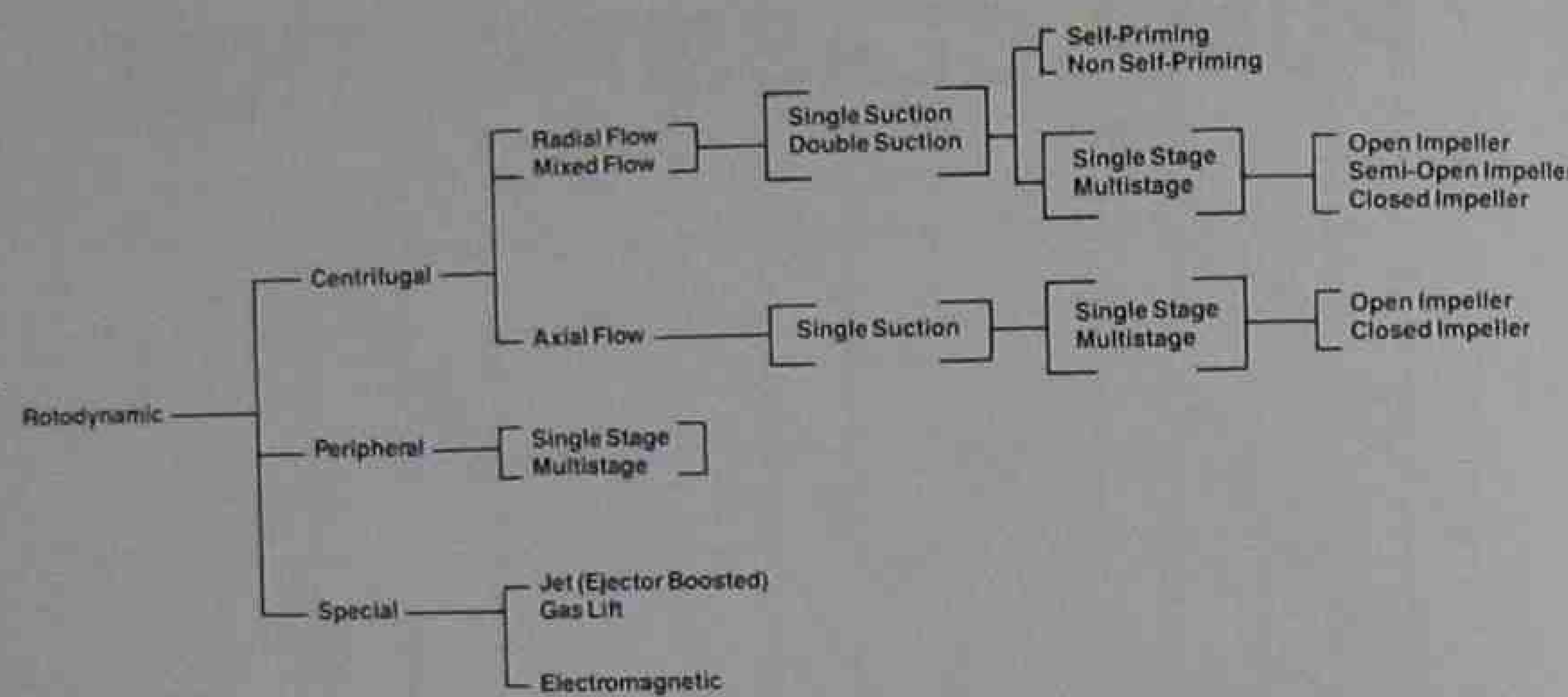
Section 2—PUMP CLASSIFICATION

Although there are a wide range of pumps available for numerous applications they generally fall into two major groups. These are:

1. Rotodynamic Pumps.
2. Positive Displacement Pumps.

2.1 ROTODYNAMIC

Rotodynamic pumps are essentially rotary machines in which energy is continuously imparted to the pumped liquid by a rotating impeller, propeller or rotor.



2.1.1 CENTRIFUGAL

This group of pumps consist of a shaft mounted impeller(s) rotating unidirectionally within a casing. The liquid enters the impeller eye and acquires energy in the form of velocity as it passes through the impeller passages. The velocity head is converted into pressure head by the volute or spiral shaped outer casing of the pump which directs the liquid from the outer perimeter of the impeller to the pump discharge. A less common method of developing pressure head is to surround the impeller with concentric diffusing passages.

Centrifugal pumps can be divided into two main groups depending on the type of impeller.

1. **Radial Flow Impellers.** (Fig. 2.1.1)
   The liquid enters the impeller axially and discharges radially, in effect changing its direction by 90°. In this case the head developed is due to the centrifugal force exerted on the fluid by the impeller.

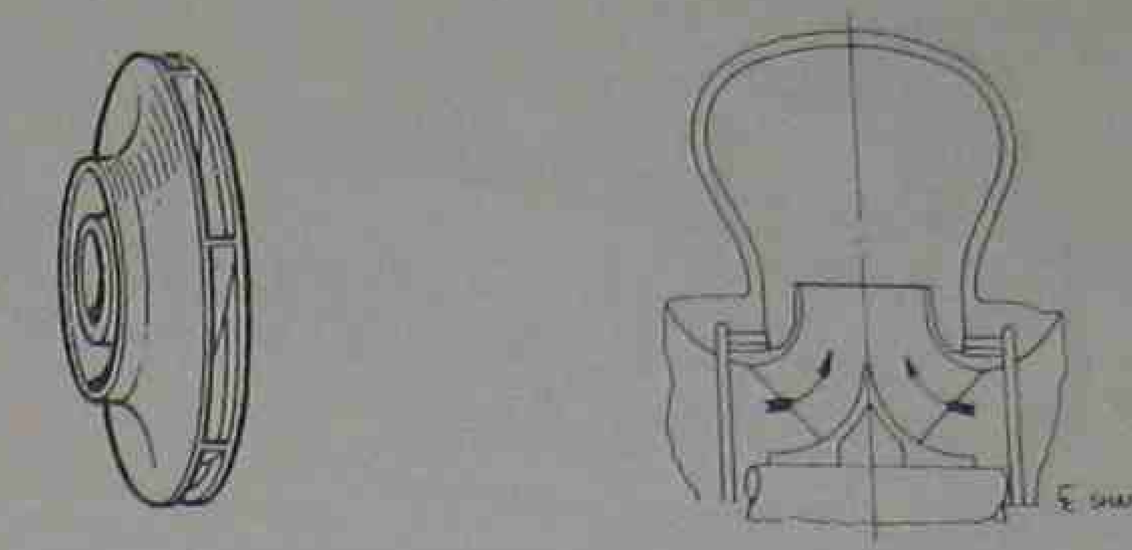


Fig. 2.1.1.

2. **Mixed Flow Impellers.** (Fig. 2.1.2)
   The liquid enters the impeller axially and discharges in both axial and radial directions. In this case the head developed is the result of a combination of the centrifugal force and the lift produced by the vanes on the liquid.

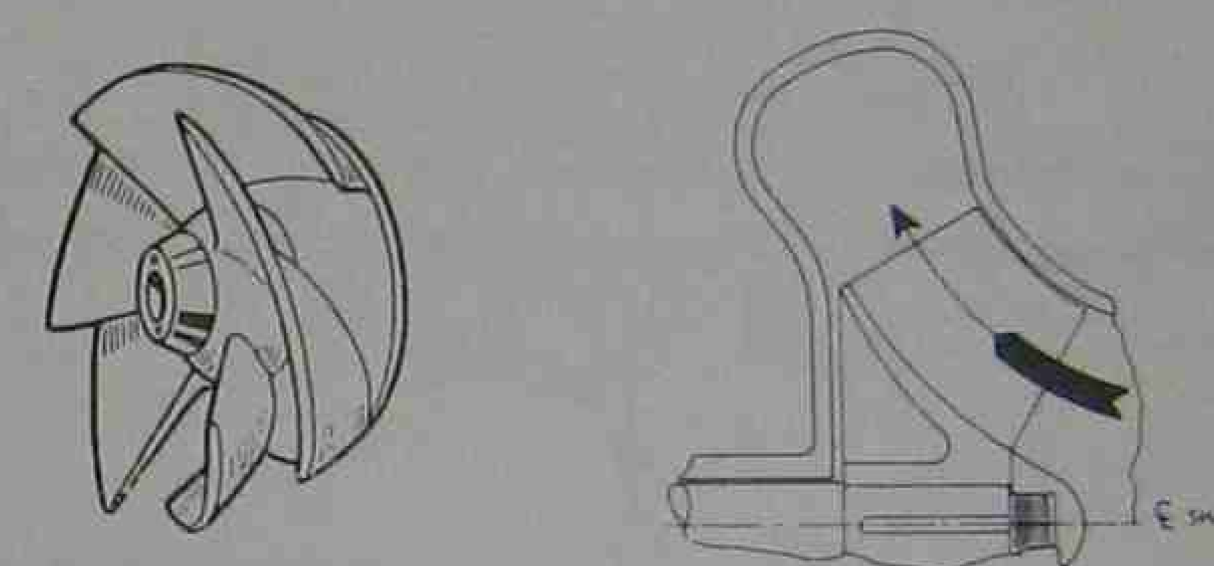


Fig. 2.1.2

3. **Axial Flow Impellers.** (Fig. 2.1.3)
   The liquid enters and leaves the impeller in an axial direction. In this case the head developed is entirely due to the lift produced by the vanes on the liquid.

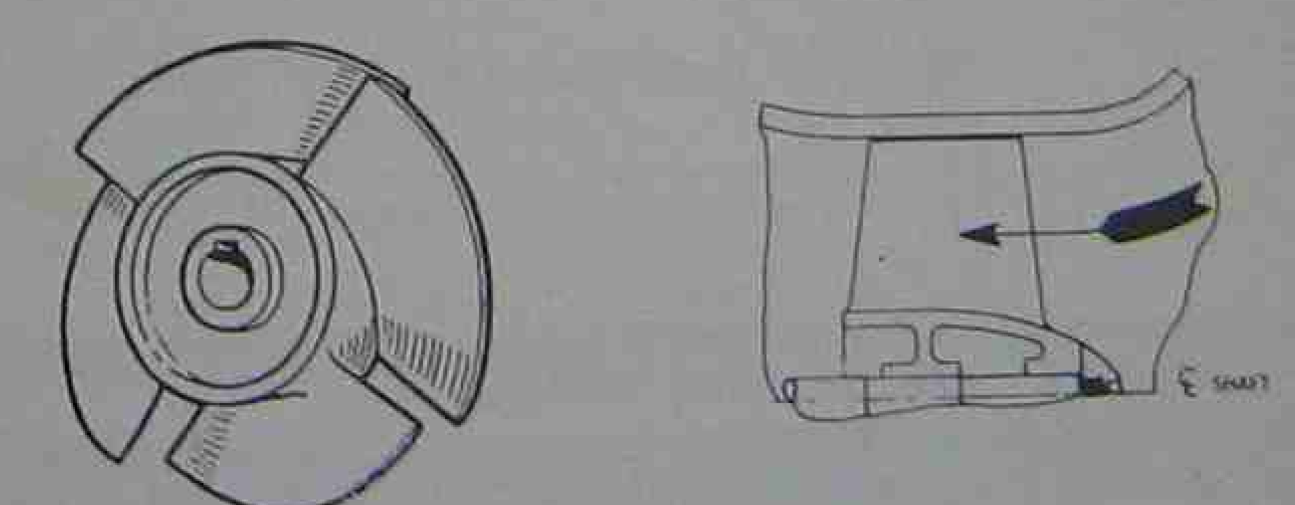


Fig. 2.1.3.

Radial and mixed flow impellers can be subdivided into:

(i) Single suction—the liquid enters the impeller from one side only. (Fig. 2.1.4)
(ii) Double suction—the liquid enters the impeller from both sides. (Fig. 2.1.5)
(iii) Axial flow impellers are of the single suction type. (Fig. 2.1.6)

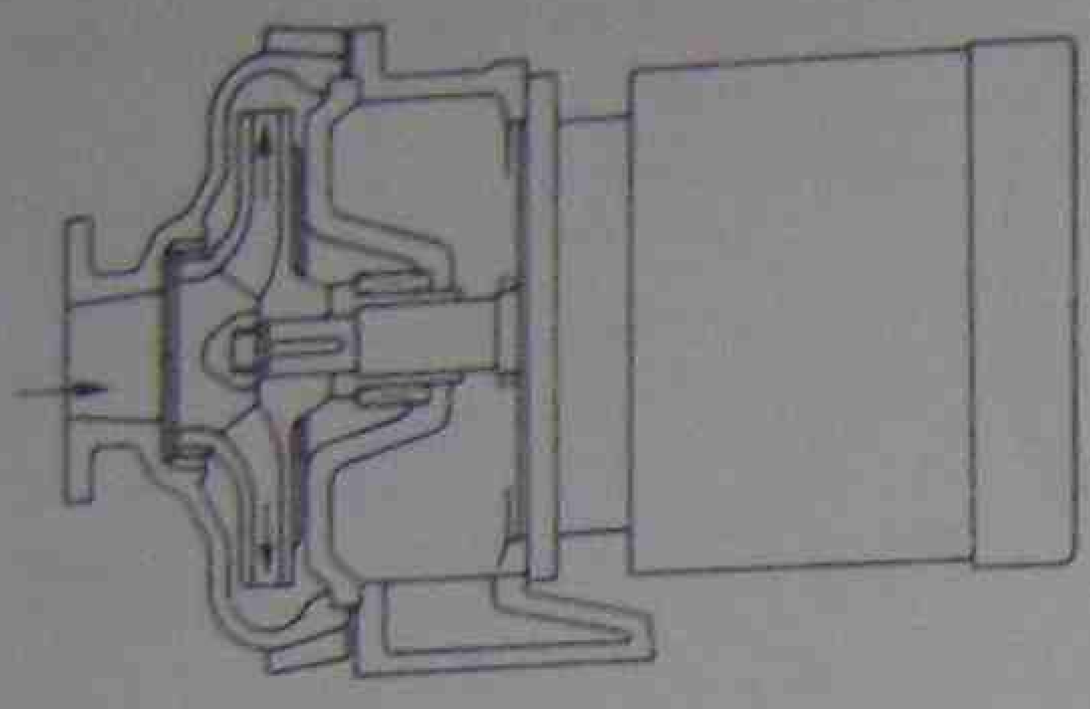


Fig. 2.1.4 Single Suction Close Coupled Pump.

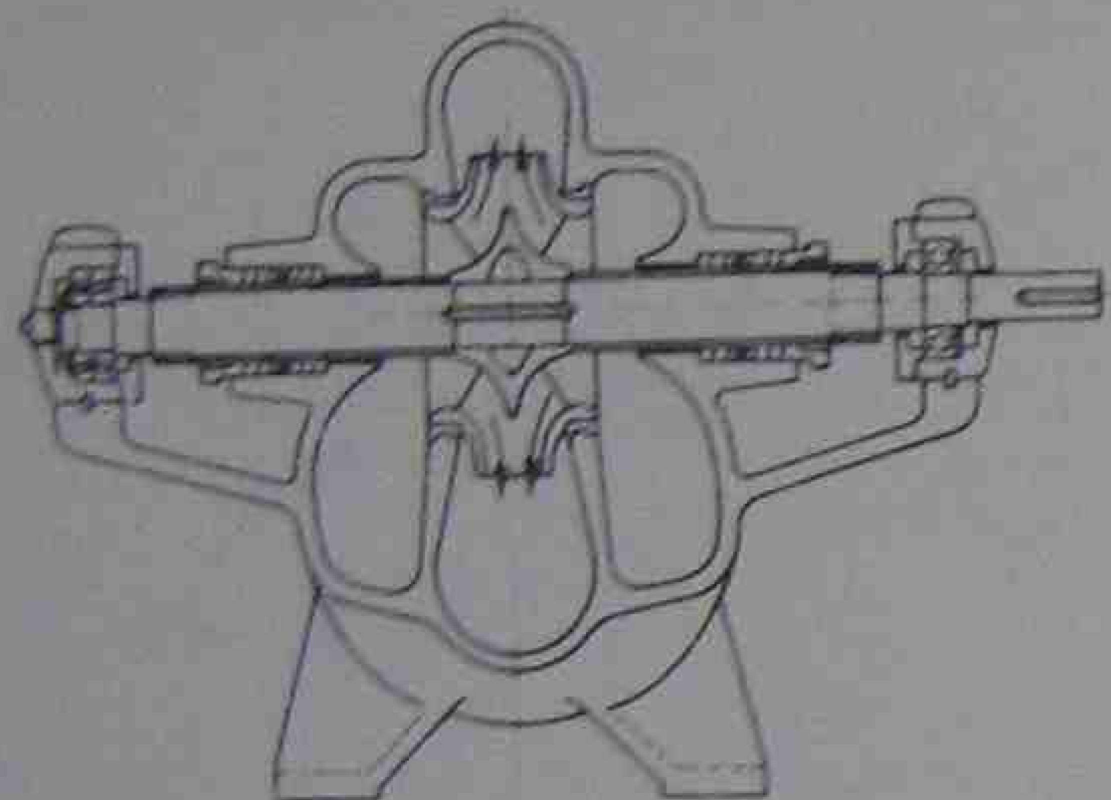


Fig. 2.1.5 Double Suction Axially Split Pump.

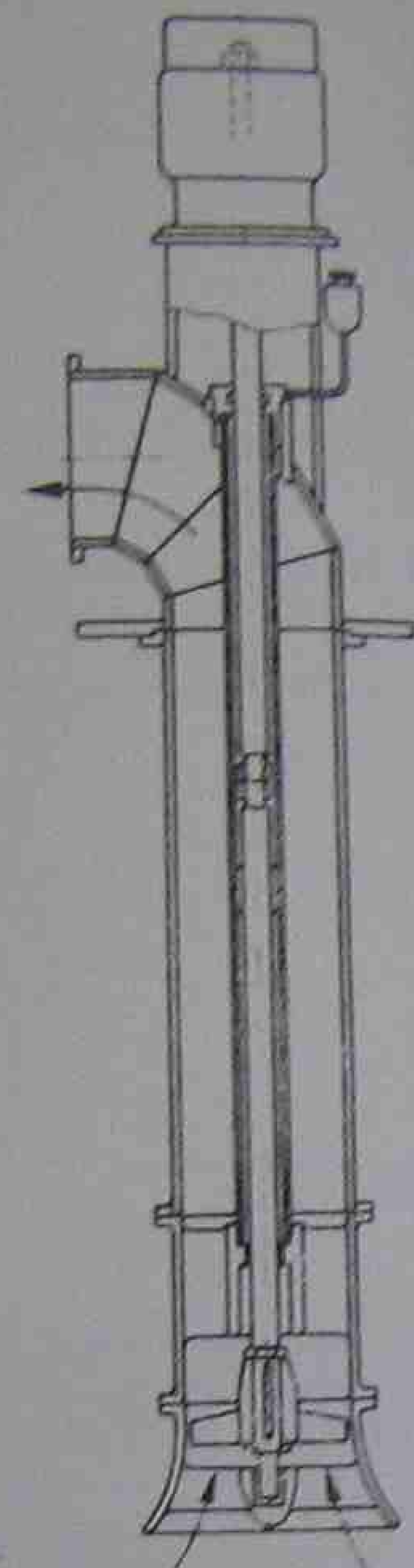
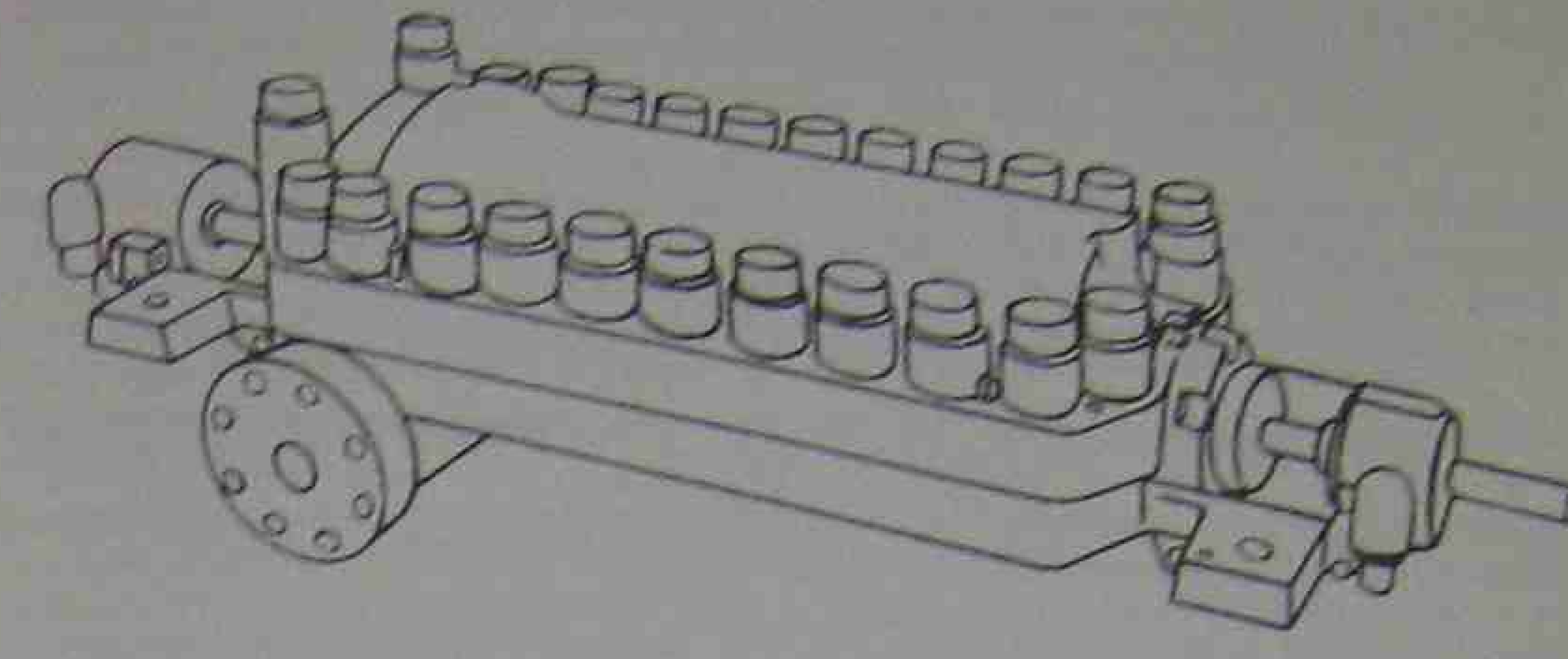


Fig. 2.1.6. Single Suction Axial Flow Pump.

Centrifugal pumps, irrespective of impeller type can be either single stage, whereby the total head is the result of one impeller, or can be multistage.

In multistage units the total head is the end result of a series of impellers all aligned within the one casing. Under these conditions the required head is achieved by the addition of the heads developed by each individual impeller.

Multistage unit casings can be either axially split (Fig. 2.1.7) or radially split. Radially split pumps can consist of a series of ring sections fastened together by external tie bolts (Fig. 2.1.8); alternatively for higher pressures or to cater for thermal shock, the ring sections can be encased in a barrel casing which also contains the suction and discharge branches. (Fig. 2.1.9). Multistage pumps can be either horizontally or vertically disposed.



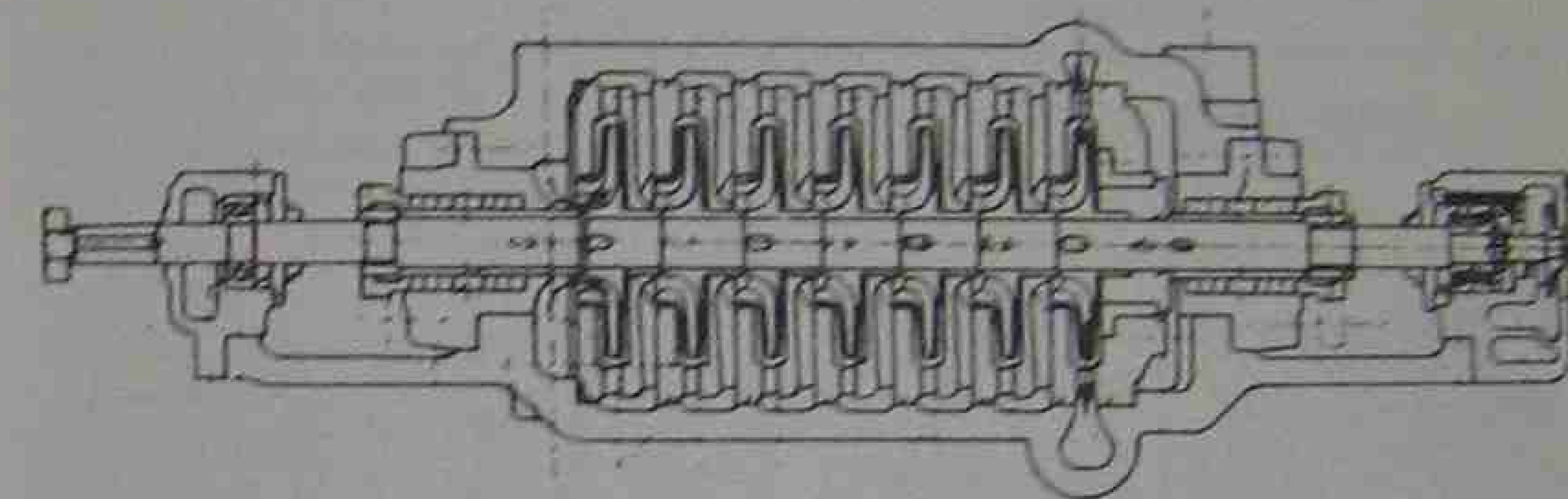


Fig. 2.1.7 Axially Split Multistage Pump.

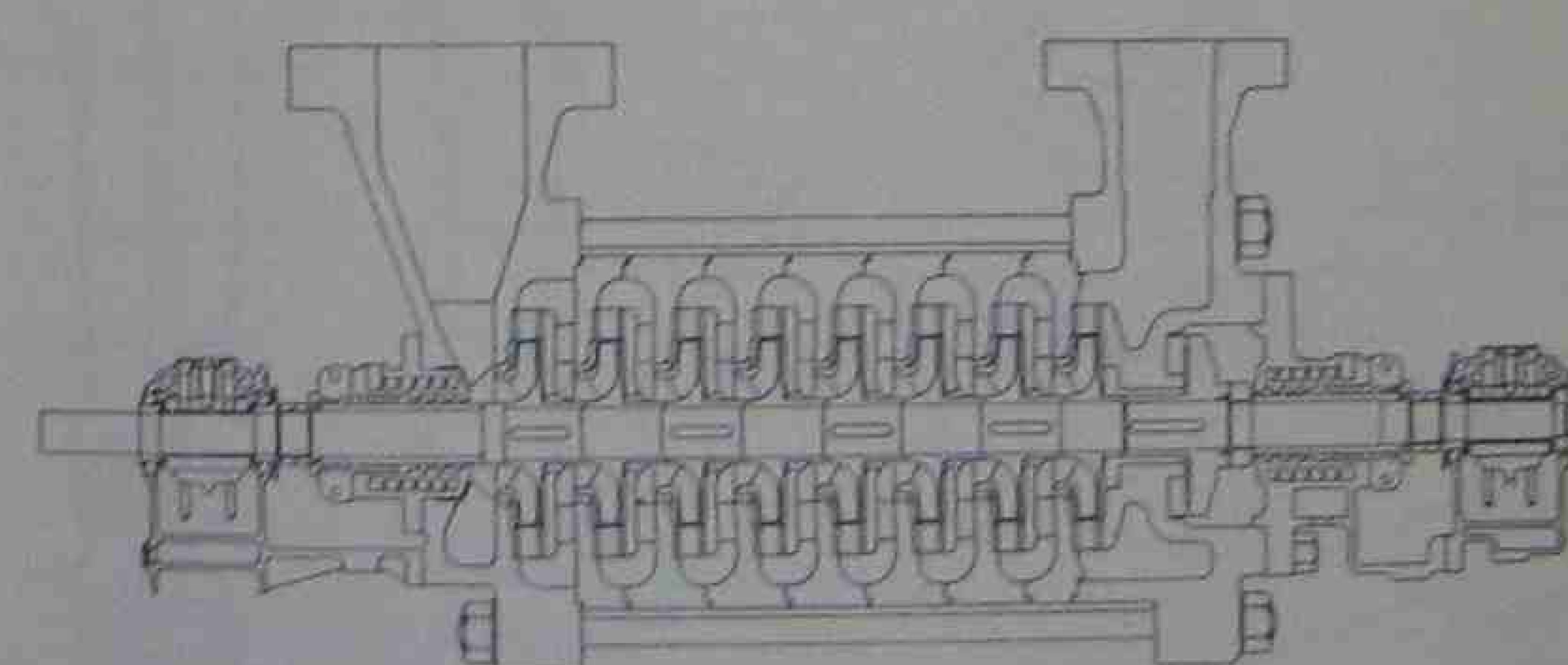


Fig. 2.1.8 Radially Split Ring Section Type Multistage Pump.

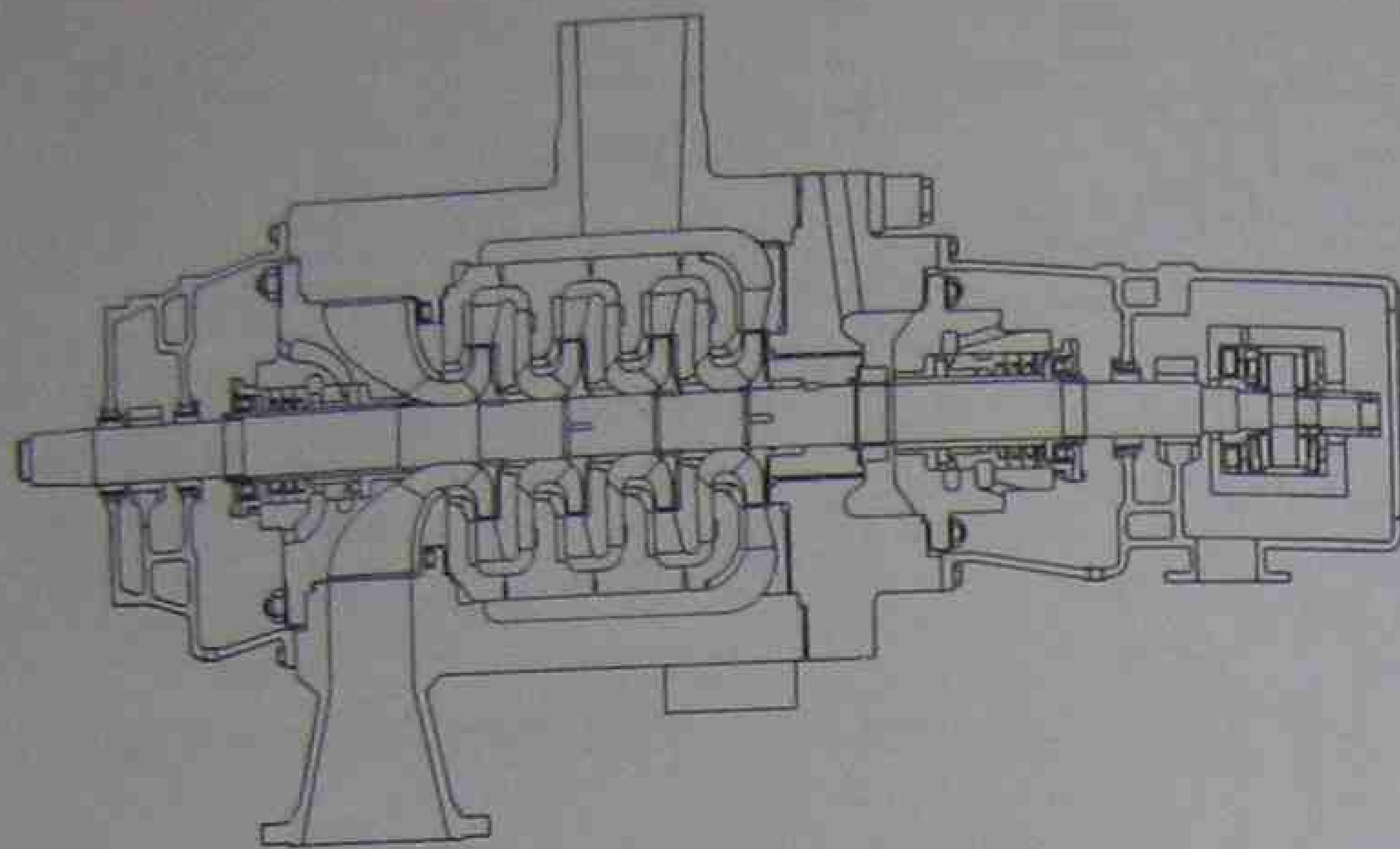


Fig. 2.1.9 Radially Split Barrel Casing Type Multistage Pump.

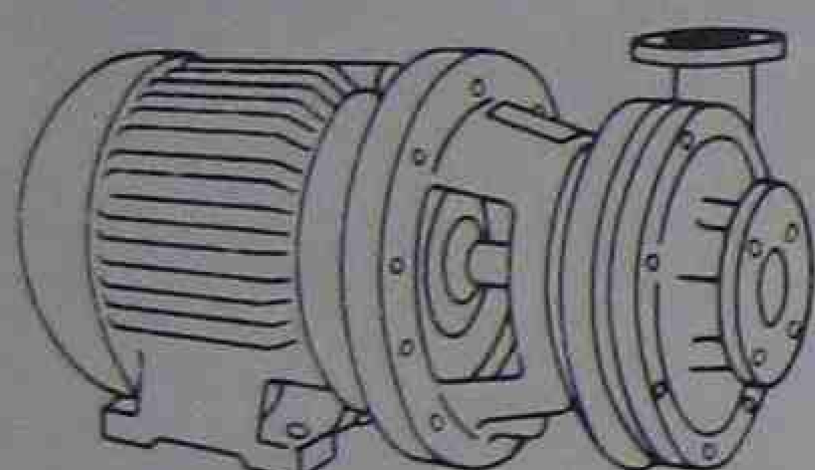


Fig. 2.1.10 Close-Coupled Pump.

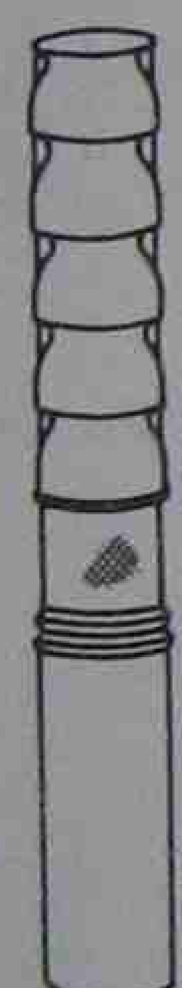


Fig. 2.1.13 Electro Submersible Multistage.

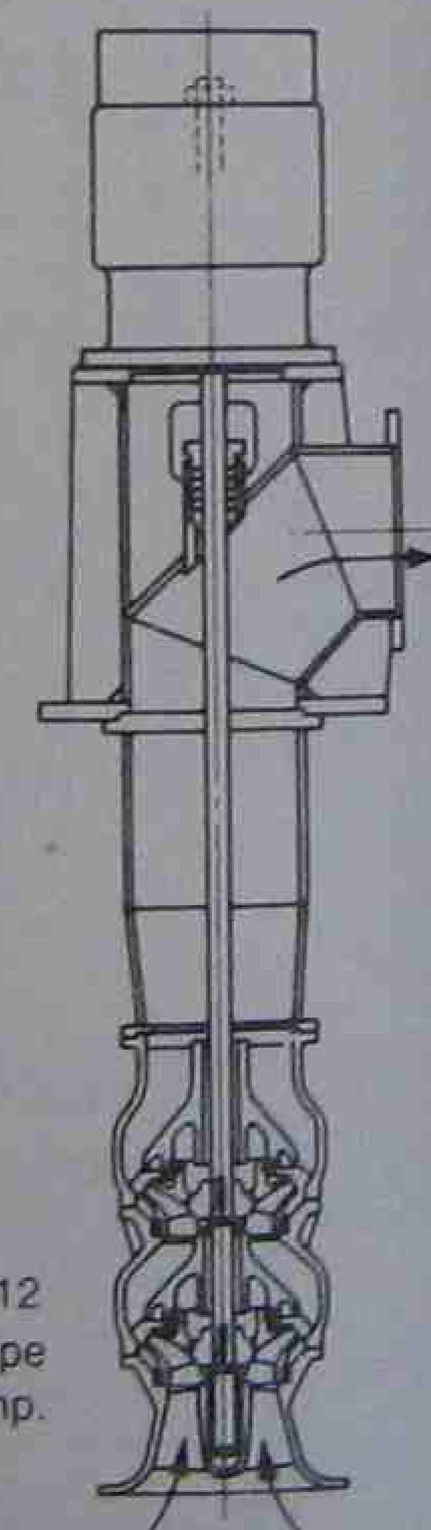


Fig. 2.1.12 Turbine Type Pump.

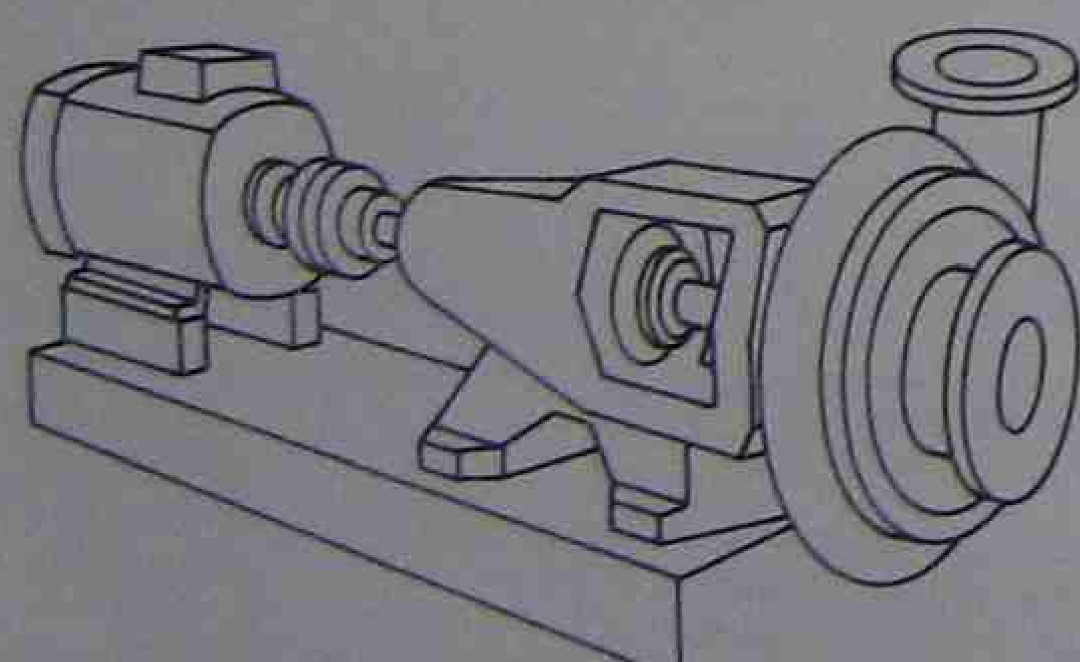


Fig. 2.1.11 Long-Coupled Pump.

Centrifugal pumps can be either self-priming or non self-priming and can have open, semi-open or closed impellers depending on the specific requirements of the particular pump.

Centrifugal pumps can also be identified by their basic mechanical configuration and characteristics:—

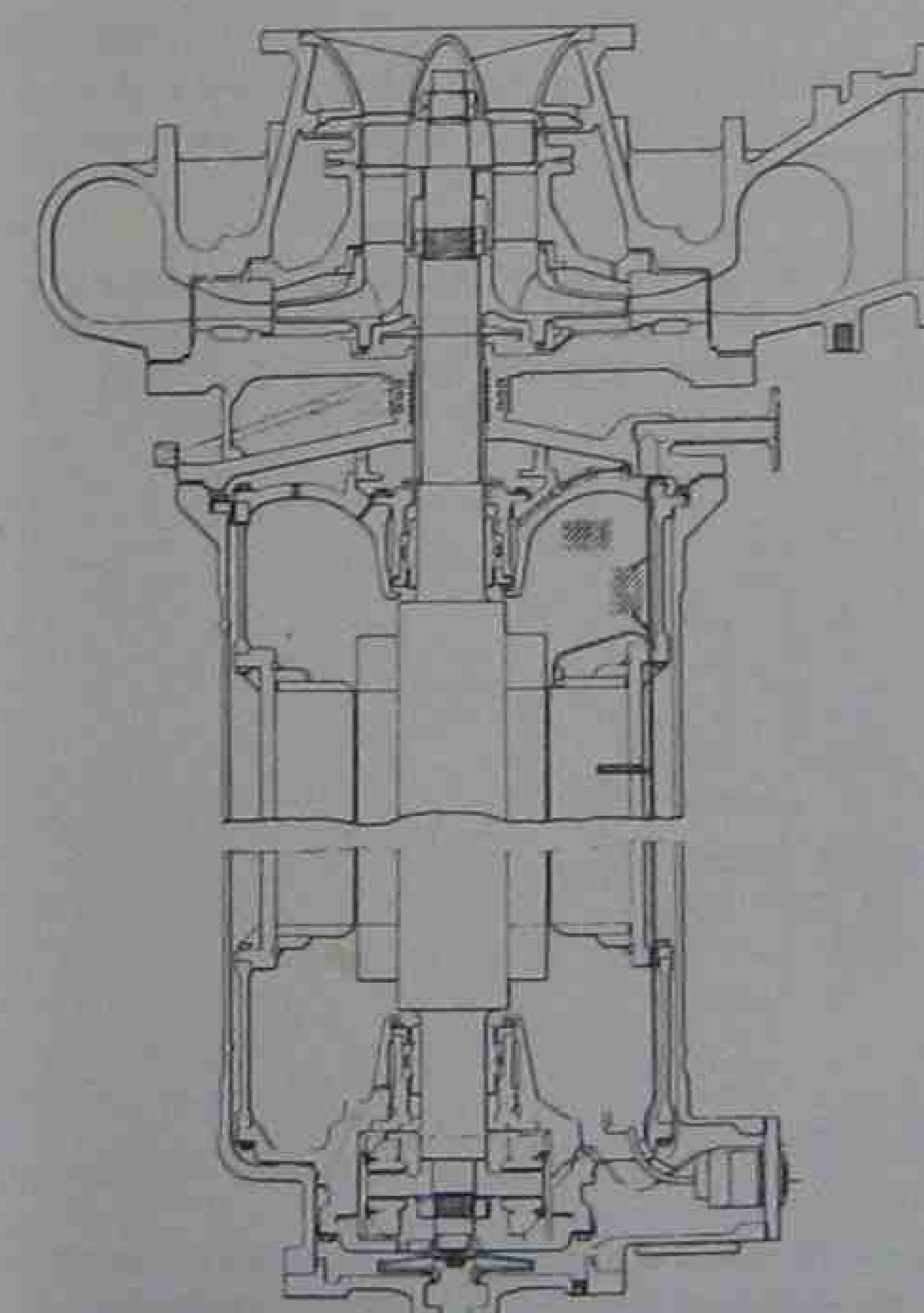
Overhung Impeller Type—the impeller(s) is mounted on the end of the shaft which is cantilevered from its bearing supports. In addition, this type of pump can either be of the close-coupled design, in which the pump casing is fixed directly to the driver frame and the impeller is mounted on the driver shaft (Fig. 2.1.4 & 2.1.10) or long coupled, where the pump is mounted on a base-plate and driven through a coupling. (Fig. 2.1.11)

Impeller Between Bearings Type—the impeller(s) is mounted on a shaft between bearings situated at both ends. (Fig. 2.1.5)

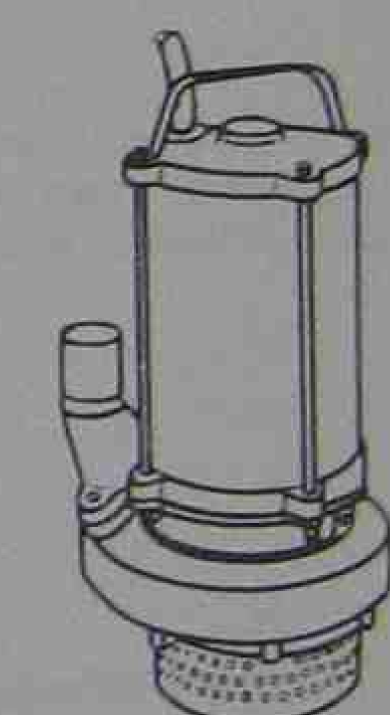
Turbine Type—this rather ambiguous term usually applies to vertical multistage deep well pumps that are constructed with diffuser casings screwed or bolted together. The bearings are lubricated, cooled and flushed by the pumped liquid (Fig. 2.1.12). A special type of turbine pump is the electro-submersible. (Fig. 2.1.13) In these deep well units, the motor is close coupled to the pump and submerged in the well. This type of pump is used for high head applications where long intermediate shafts are undesirable.

In addition to the centrifugal pumps described previously, there are other types which have their own unique characteristics:

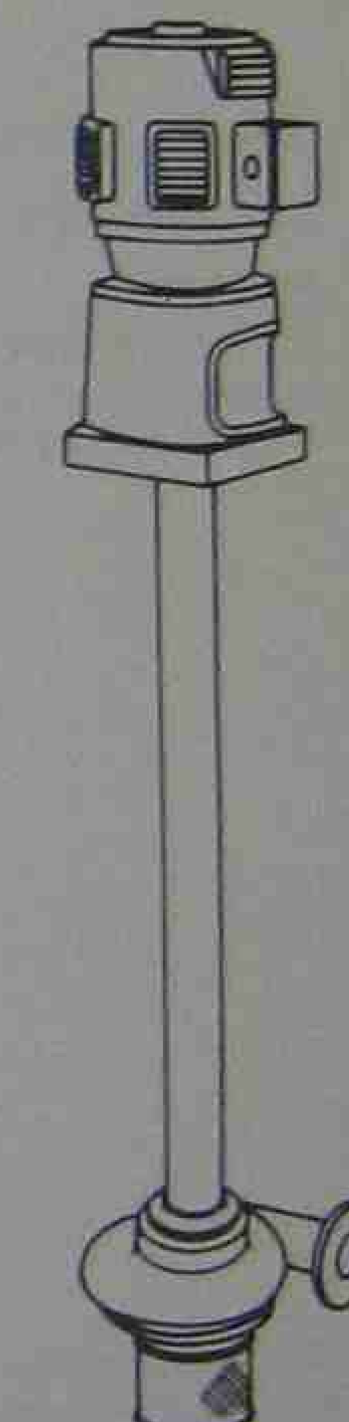
Single stage electrosubmersible pumps, sump pumps, glandless and canned motor pumps, sewage pumps and abrasion resistant pumps. (Fig. 2.1.14)



GLANDLESS CANNED MOTOR PUMP



ELECTRO SUBMERSIBLE



SUMP

Fig. 2.1.14

2.1.2 **PERIPHERAL:**

In a peripheral pump, energy is imparted to the fluid within the cells of a vane wheel impeller. Alternatively, the cells are arranged peripherally on the outer sides of a wheel disc. Peripheral pumps come in various configurations, such as overhung impeller, impeller(s), mounted between bearings, single stage and multistage.

Alternative names are side-channel pump and regenerative turbine pump. (Fig. 2.1.15)

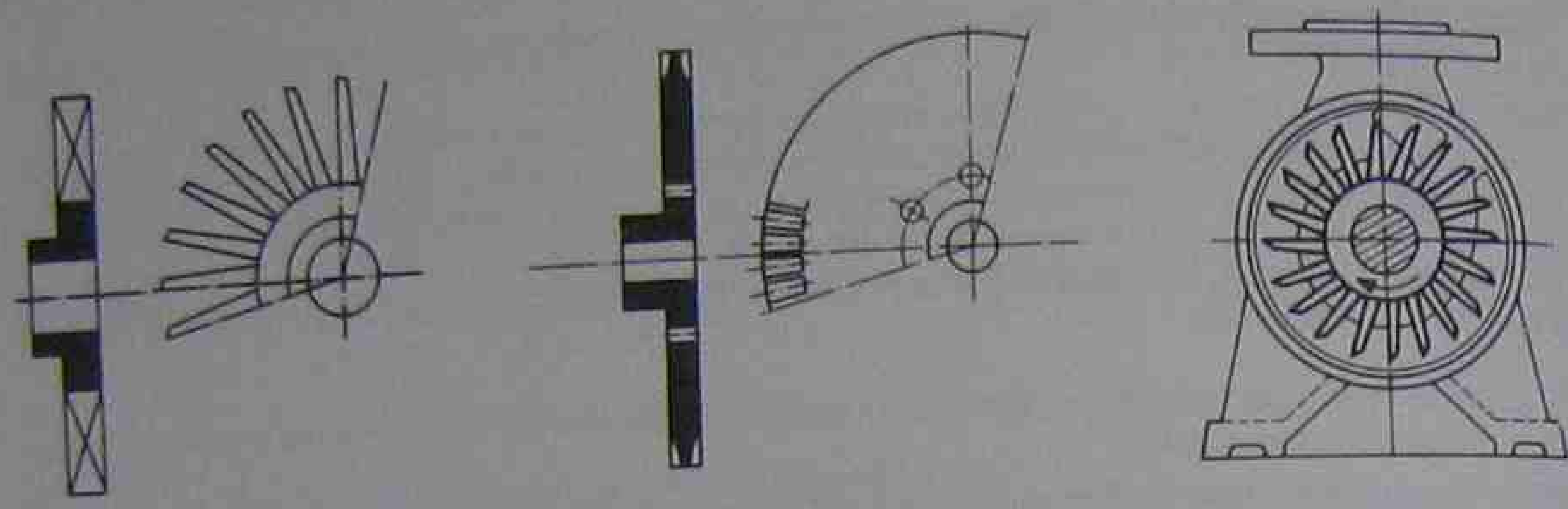


Fig. 2.1.15

2.1.3 **SPECIAL:**

Jet Pump:

The term jet pump describes a pump in which a high velocity jet of fluid is utilised to create a low pressure area in a mixing chamber, causing the suction fluid to flow into this chamber. The venturi is positioned so that gradual velocity conversion occurs with minimum losses. (Fig. 2.1.16)

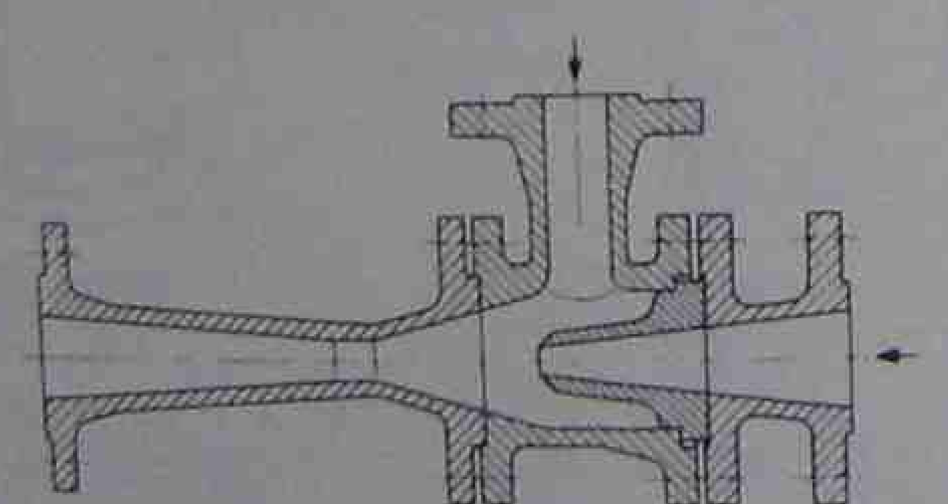


Fig. 2.1.16

When the jet/venturi are attached directly to the pump ahead of the suction impeller and activated by the liquid from the pump it is termed a "shallow well jet pump" (Fig. 2.1.17a). When the jet/venturi are set at the base of an extended suction pipe and activated by the liquid from the pump it is termed a "deep well jet pump". (Fig. 2.1.17b).

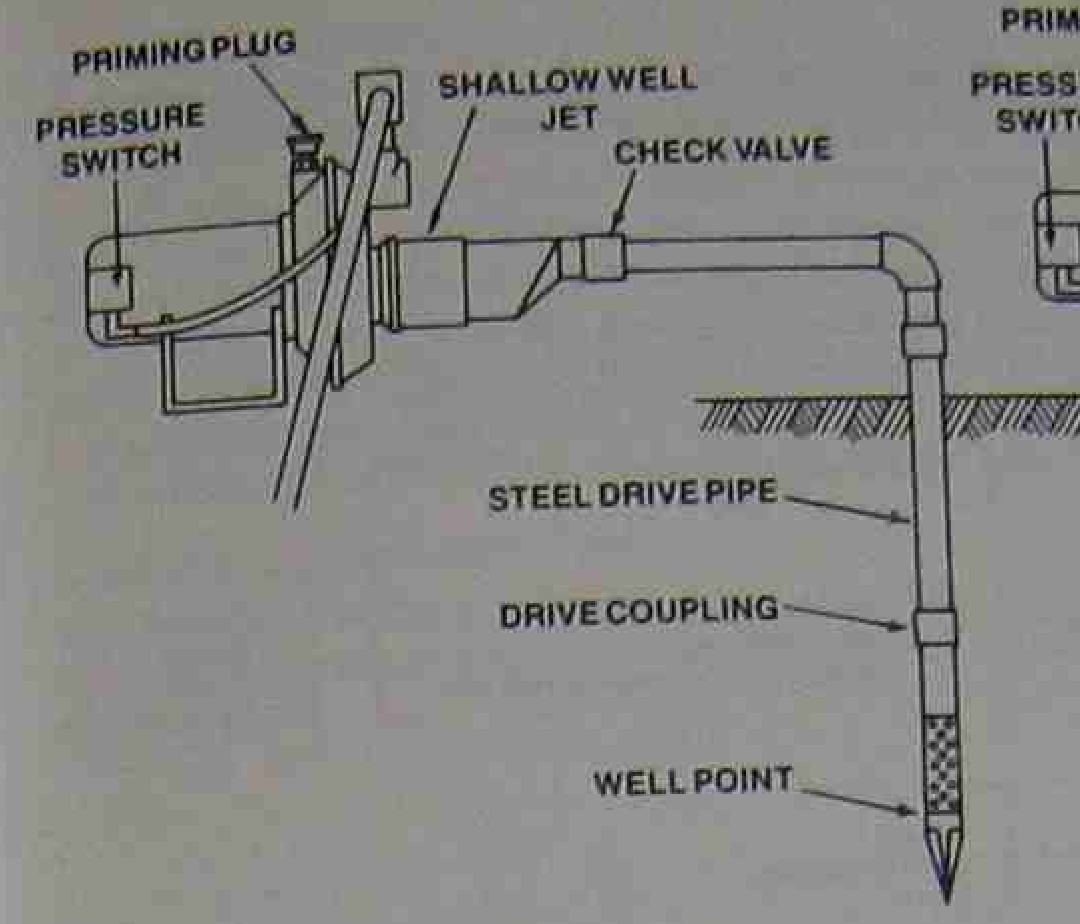


Fig. 2.1.17a Shallow Well.

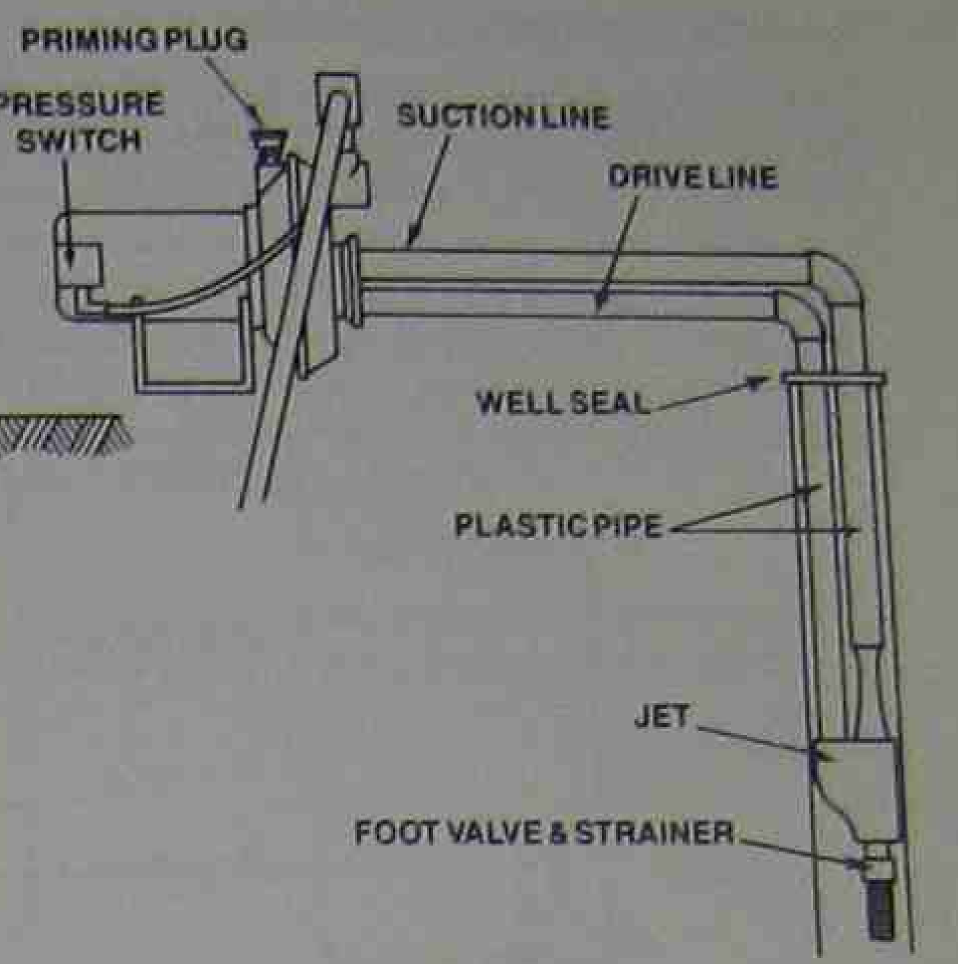


Fig. 2.1.17b Deep Well.

Gas Lift

A gas pressure source is used to lift or 'pump' the liquid handled by mixing it with gas under pressure, usually compressed air. (Fig. 2.1.18)

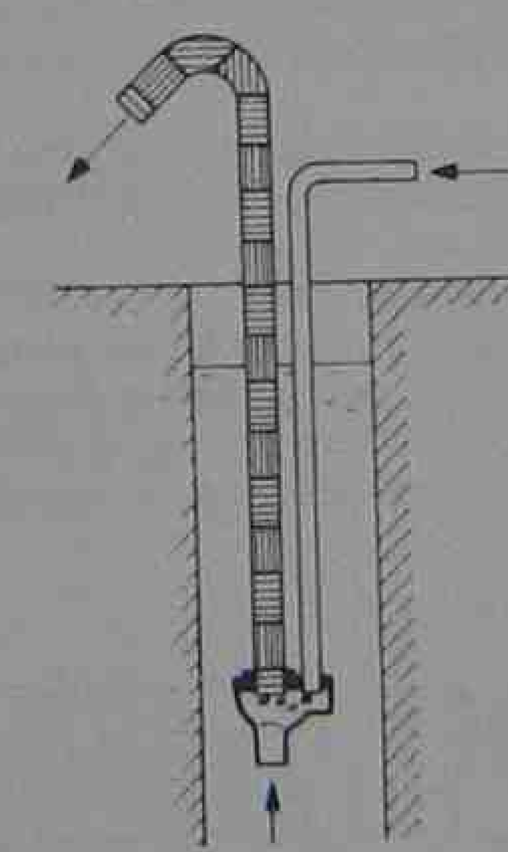


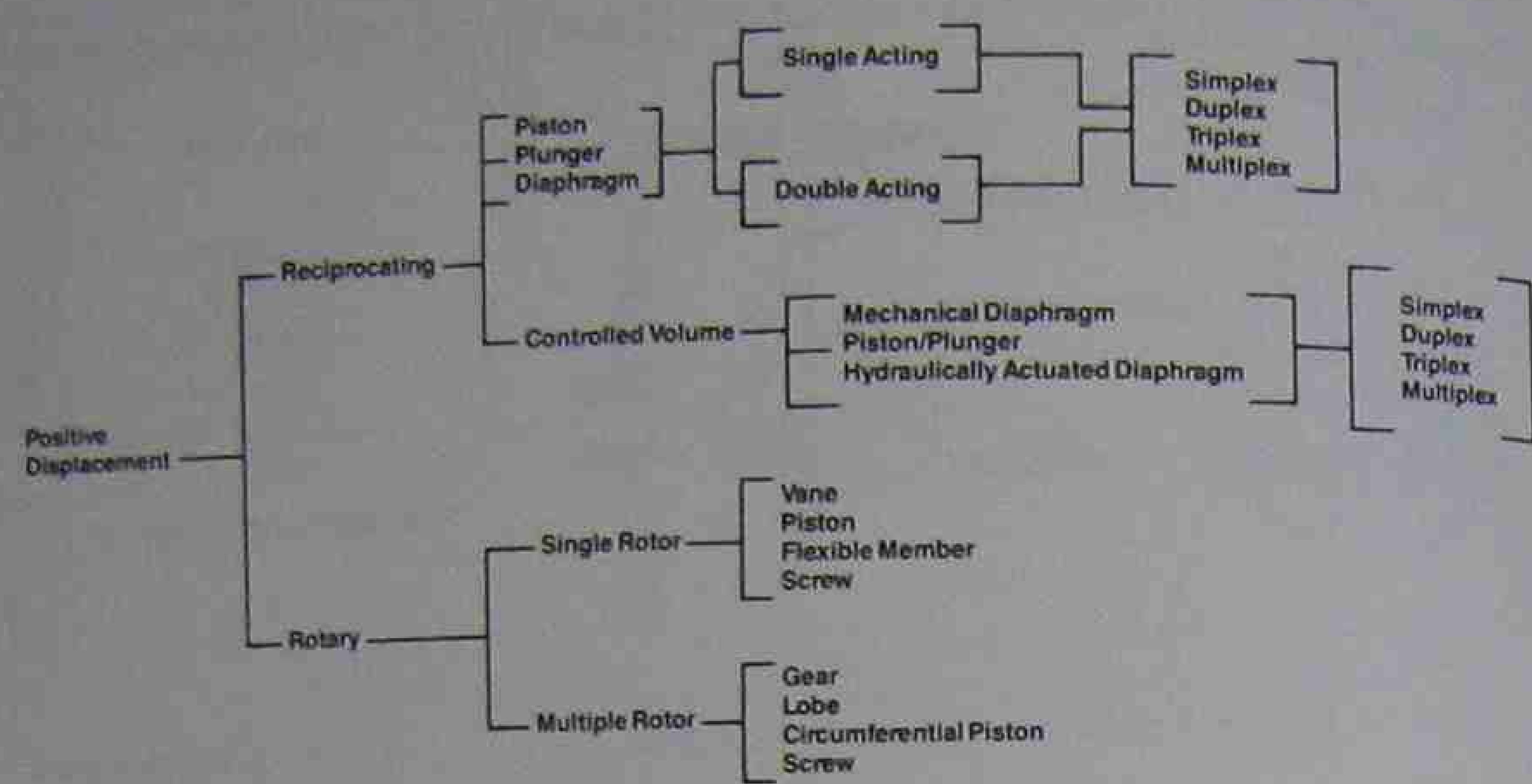
Fig. 2.1.18

Electro-magnetic pump

A glandless pump without rotating parts in which a magnetic field acts upon a susceptible medium (liquid metals in principle).

2.2 POSITIVE DISPLACEMENT

Positive displacement pumps are essentially rotary or reciprocating machines in which energy is periodically added by application of force to movable boundaries of enclosed fluid containing volumes, resulting in a direct increase in pressure.



2.2.1 RECIPROCATING

1. Piston, Plunger Diaphragm

A reciprocating pump set generally consists of a piston/plunger (or diaphragm) inlet and outlet valves and a means by which the piston/plunger can be actuated. This is usually done by a receiprocating engine or an electric motor/steam turbine, crank and connecting rod combination. These pumps can be either.

(a) Single acting in which the liquid is discharged during the forward motion of the piston.
(b) Double acting in which the liquid is discharged during both the forward and backward motions of the piston.

Reciprocating pumps can be further classified as:-

(i) Simplex pumps—contain one single or double acting piston/plunger.
(ii) Duplex pumps—contain two single or double acting pistons/plungers.
(iii) Triplex pumps contain three single or double acting pistons/plungers.
(iv) Multiplex pumps—contain more than three single or double acting pistons/plungers.

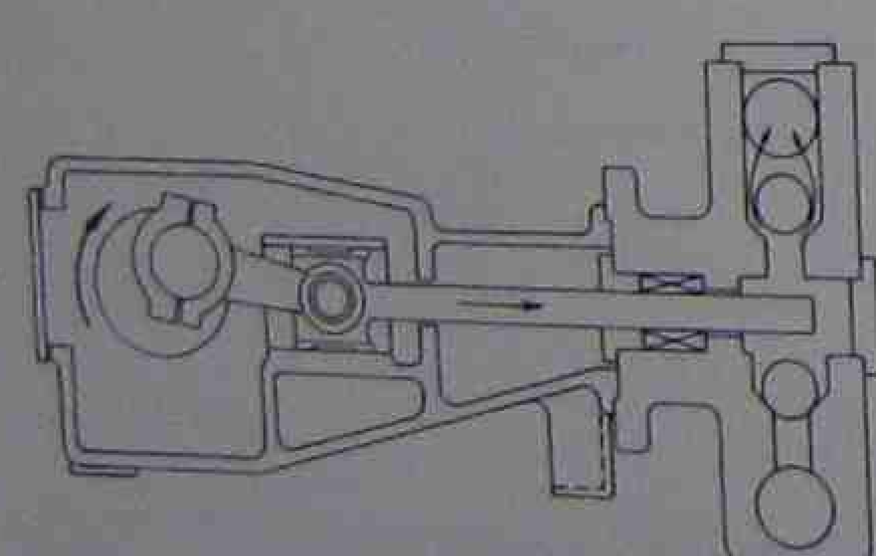


Fig. 2.2.1 Horizontal Single-Acting Plunger Pump.

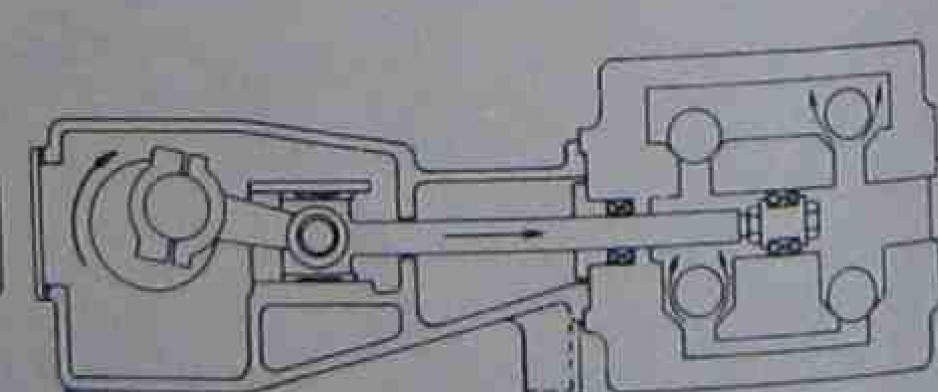


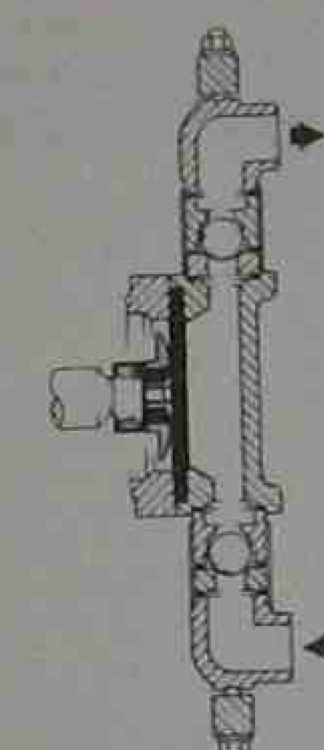
Fig. 2.2.2 Horizontal Double-Acting Piston Pump.

2. Controlled Volume

Another type of reciprocating positive displacement pump is the controlled volume pump which provide precision control of very low flowrates up to 3 L/s. Flow rate accuracy is typically within ± 1%. Other names for these pumps are 'proportioning pumps' and 'metering pumps'. Controlled volume pumps are generally available in three construction styles, piston or plunger; mechanical diaphragm and hydraulically actuated diaphragm type (piston diaphragm). Usually, the driver is an electric motor. Basically, the design criteria that applies to large motor-driven reciprocating pumps also applies to controlled volume pumps. Flow rate variations are normally achieved by manual resetting of the stroke length. Automatic controls are available for stroke length resetting and motor speed. The typical efficiency of this type of pump is around 90%.

Mechanical Diaphragm Type

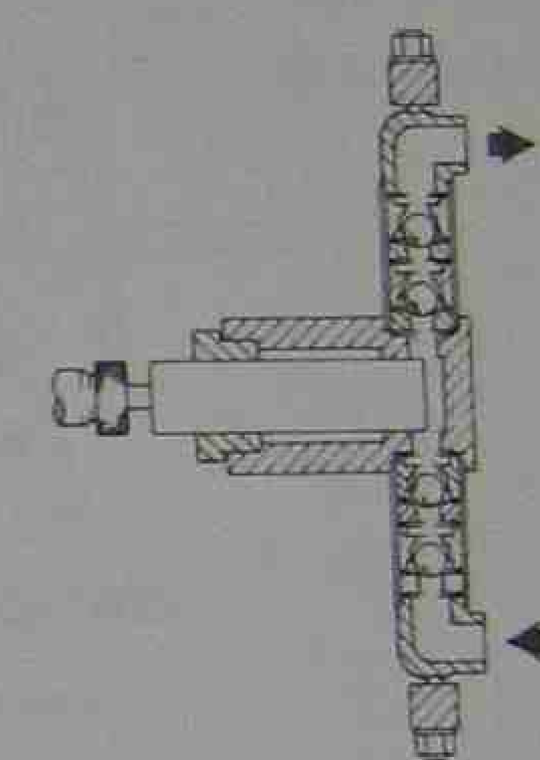
With glandless construction and simple 'wet' end designs, the mechanical diaphragm pump offers the advantage of a positively sealed pumping chamber, in which the risks of corrosion and erosion are negligible.



CONTROLLED VOLUME PUMP

MECHANICAL - DIAPHRAGM LIQUID END

Fig. 2.2.3 Diaphragm.



CONTROLLED VOLUME PUMP

PLUNGER LIQUID END

Fig. 2.2.4 Horizontal Plunger.

Piston/Plunger Type

The piston/plunger pump offers a positive means of metering a wide variety of fluids by utilising a range of gland packing materials and long-life plungers. With this type of pump the flow-rate accuracy and discharge pressure greatly exceed that of the mechanical diaphragm type. The wet end consists of a piston/plunger reciprocating within a cylindrical housing.

Hydraulically Actuated Diaphragm Type

This type of pump combines the glandless construction of the mechanical diaphragm pump and the accuracy, repeatability and high pressure capability of the piston/plunger pump. The diaphragm is hydraulically coupled to the piston or plunger. This type of pump can incorporate diaphragm rupture detection devices and various hydraulic fluids which are compatible with the fluid being pumped.

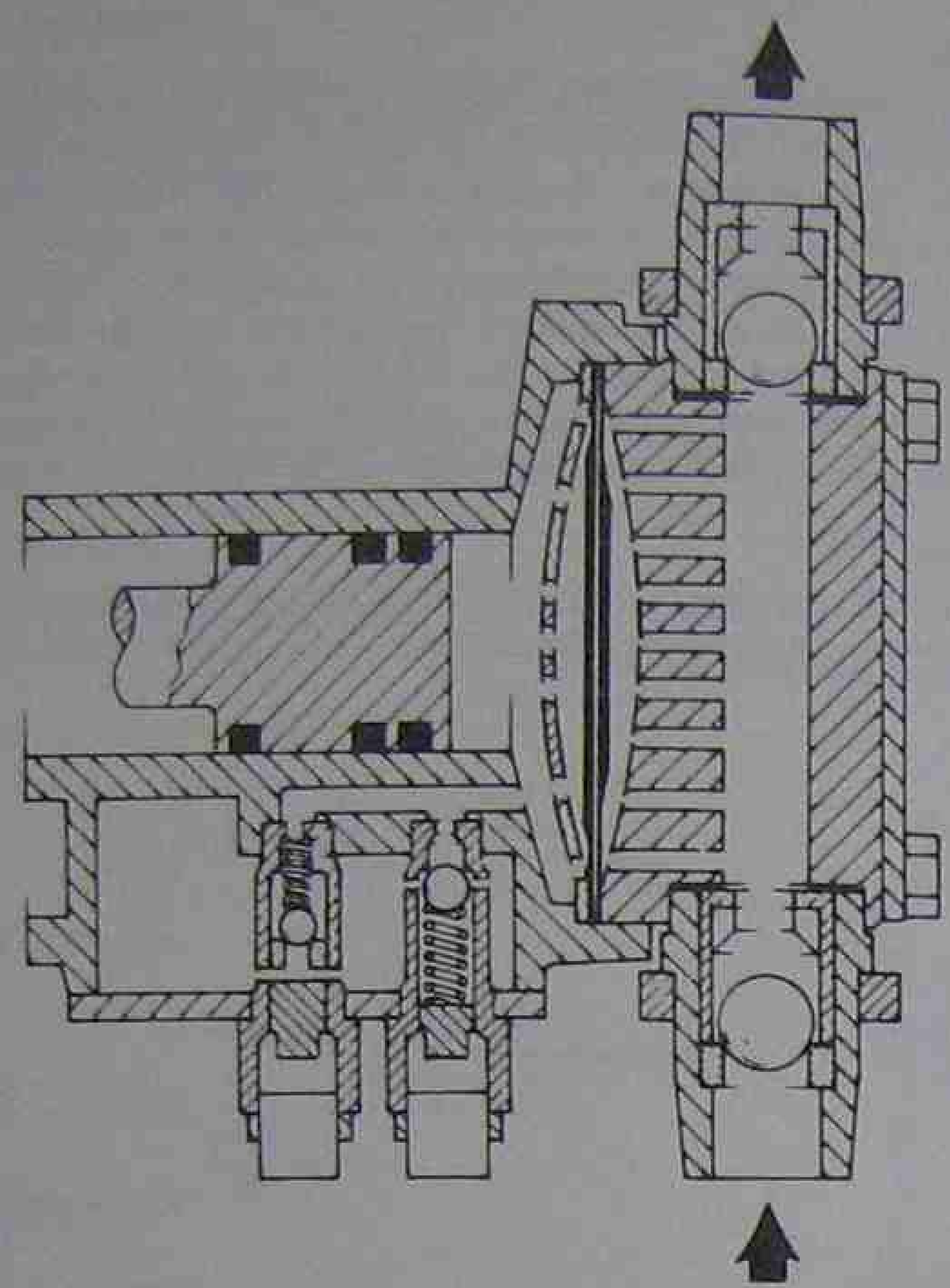


Fig. 2.2.5 Hydraulic Actuated Diaphram.

2.2.2 **ROTARY**

Rotary pumps generally consist of gears, screws, vanes or similar elements enclosed within a casing. They have no separate inlet or outlet valves and the liquid flows through the pump in a uniform stream as a result of the rotation of the elements. These pumps are characterised by their close running tolerances. Rotary pumps can be divided into two general groups:

1. Single Rotor
2. Multiple Rotor

Each group contains four basic types, identified by the type of pumping element.

1. **Single Rotor Types**

(i) **Vane**

The vanes of various forms, including blades, buckets, rollers, or slippers are radially displaced inwards or outwards by a cam which has the effect of drawing liquid into and then out of the casing. There are two types of vane pumps.

(a) Vane in rotor

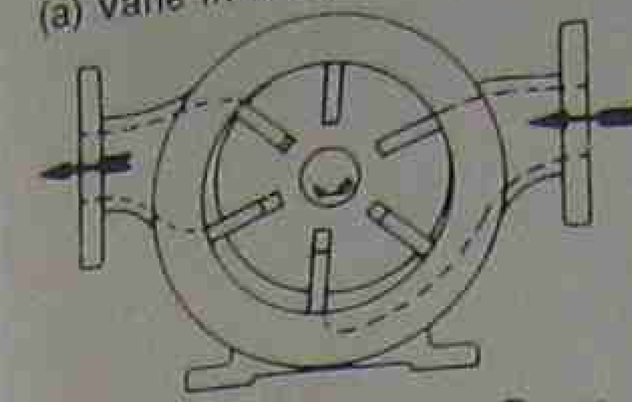


Fig. 2.2.6 Sliding Vane Pump.

(b) Vane in stator

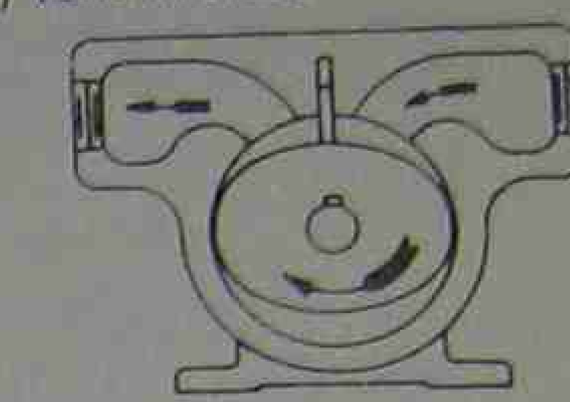


Fig. 2.2.7 External Vane Pump.

(ii) **Piston**

The rotor contains cylinders in which piston like elements reciprocate. As the rotor turns, the movement of the pistons draws the fluid into and out of the casing by the rotation of the cylinder of the piston and cylinder relative to the parts.

There are two variations of this type of pump.

The first in which the pistons reciprocate in an axial direction and the other in which the reciprocating action of the piston is in a radial direction.

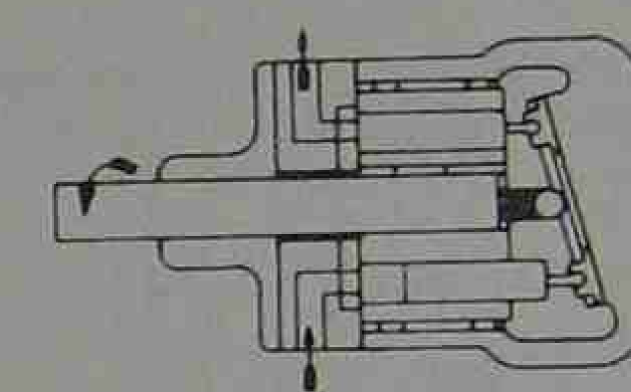


Fig. 2.2.8 Axial Piston Pump.

(iii) **Flexible Member**

The flexible member, either a vane, tube or liner is of sufficient elasticity to accomplish sealing and transfer the fluid within the casing.

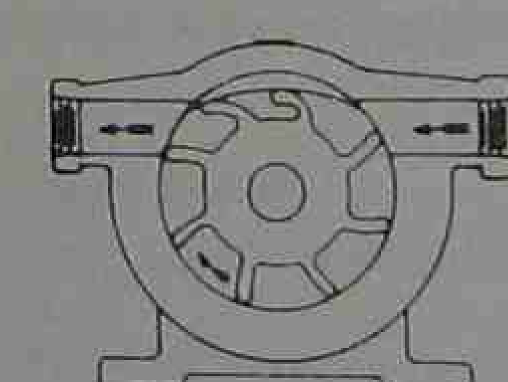


Fig. 2.2.9a Flexible Vane Pump.

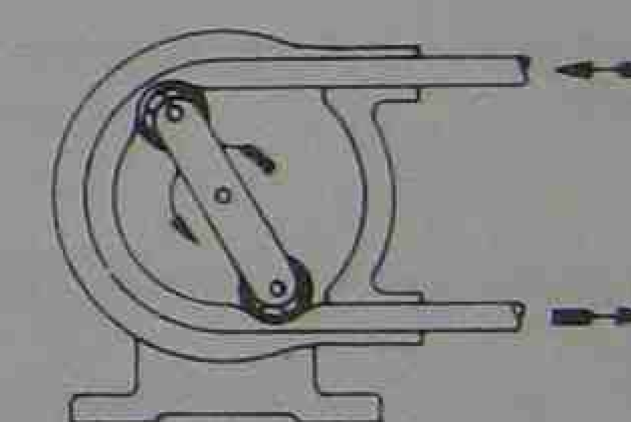


Fig. 2.2.9b Flexible Tube Pump.

(iv) **Screw—Progressive Cavity**

The progressing cavity or helical rotor pump consists of a resilient stator in the form of a double internal helix and a single helical rotor as shown in illustration.

The rotor maintains a constant seal across the stator and this seal travels continuously through the pump giving uniform positive displacement. The single helical rotor rolls in the stator with an eccentric motion.

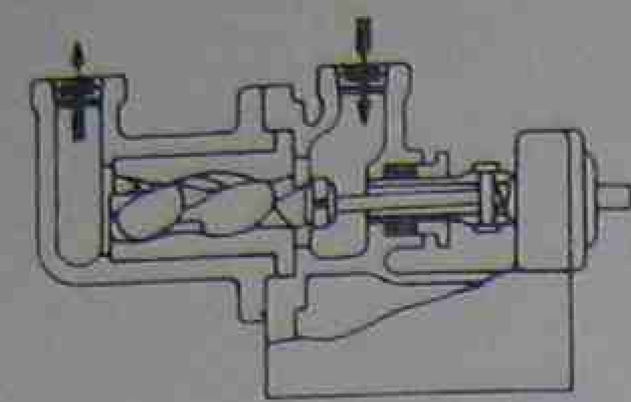


Fig. 2.2.10 Single Screw Pump (progressing cavity).

2. **Multiple Rotor Types**

(i) **Gear**

The meshing of two or more gears provide the pumping action. This meshing of gears also forms part of the moving fluid seal between the inlet and outlet ports.

Gear pumps can be either:—

(a) **External gear—**in which all gear teeth are cut externally and may be of spur, helical or herringbone tooth pattern.

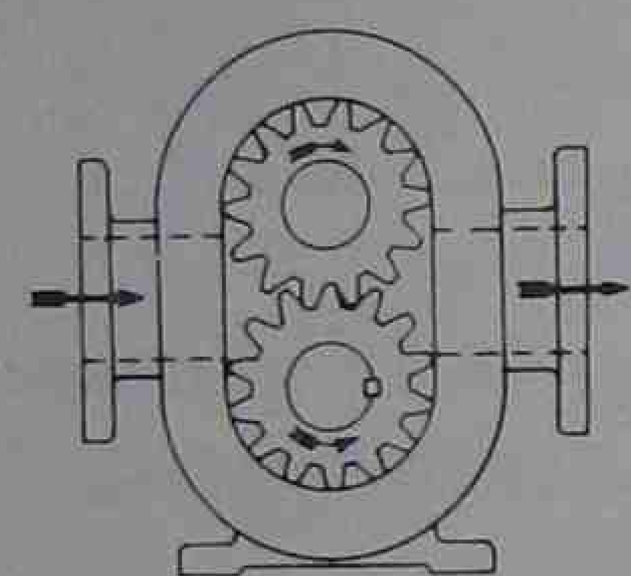


Fig. 2.2.11 External Gear Pump.

(b) **Internal gear—**in which one of the rotors has teeth cut internally and the other has externally cut teeth.

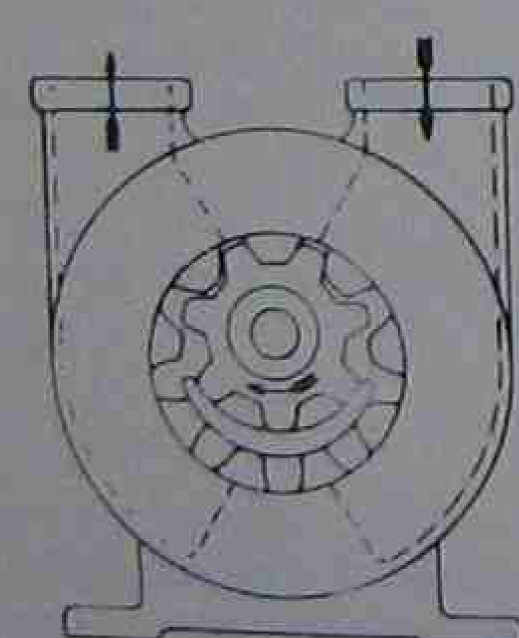


Fig. 2.2.12 Internal Gear Pump (with crescent).

(ii) **Lobe**

The rotors consist of one or more lobes, the interaction of which transports the fluid from the inlet to outlet. As in the gear pumps, the shape of the rotors and their operation provide part of the fluid seal.

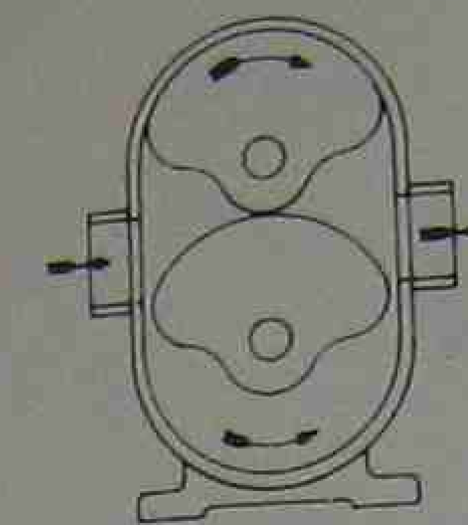


Fig. 2.2.13a Single Lobe Pump.

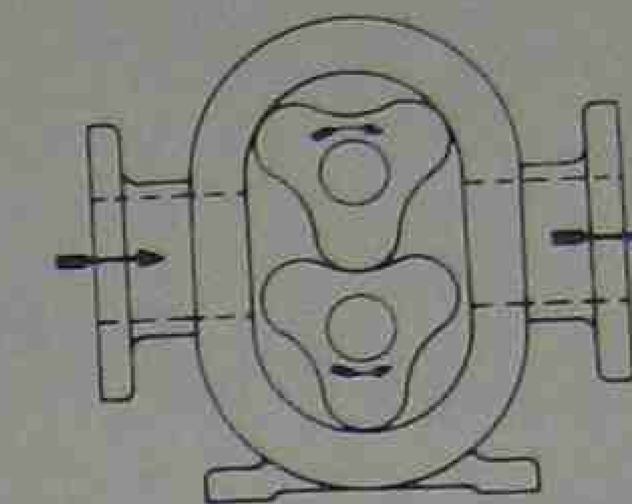


Fig. 2.2.13b Three-Lobe Pump.

(iii) **Circumferential Piston**

The liquid is transported in the spaces between the piston surface to the outlet. The operation is similar to that of a lobe pump, however, a fundamental difference is that the rotors do not form any type of fluid seal.

Circumferential piston pumps can be either internal or external. The internal version must have two or more piston elements and has no need for timing gears, whereas the external type may have one or more pistons and requires timing gears.

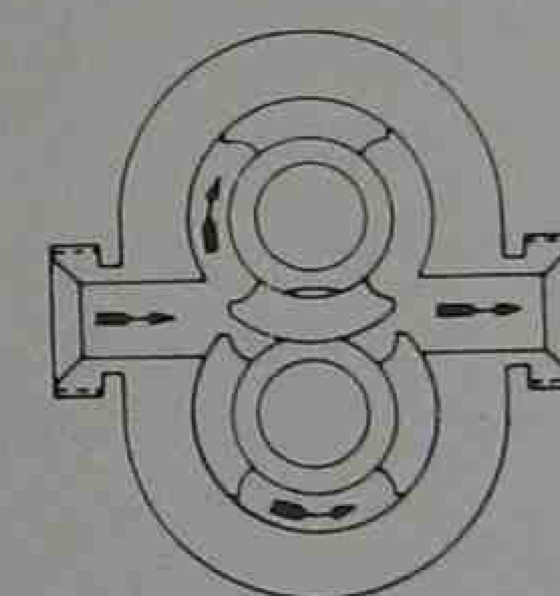


Fig. 2.2.14 Circumferential Piston Pump.

(iv) **Multiple Screw**

Usually the screw type rotors in this type of pump cannot drive each other and timing gears are required.

The principle of operation is similar to that of the single screw pump.

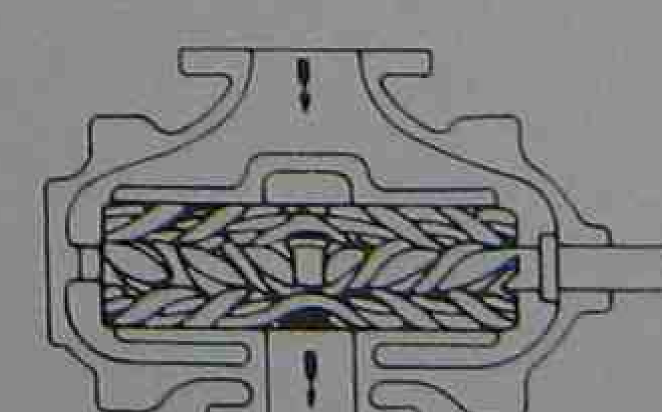


Fig. 2.2.15 Three Screw Pump.

Section 3

CHARACTERISTICS OF CENTRIFUGAL PUMPS

Section 3—CHARACTERISTICS OF CENTRIFUGAL PUMPS

3.1 CONSTRUCTION OF CENTRIFUGAL PUMPS

A centrifugal pump is a machine which moves liquid by accelerating it radially outward in a rotating impeller to a surrounding stationary housing or casing. Thus, a centrifugal pump has two main parts:
1) a rotating element consisting of an impeller mounted on a shaft
2) a stationary element consisting of a casing, stuffing box and bearings.

Impeller

The rotary motion of impeller imparts velocity energy to the liquid some of which is converted into pressure within the impeller passages. Impellers may be classified as radial, axial or mixed flow, depending on design.

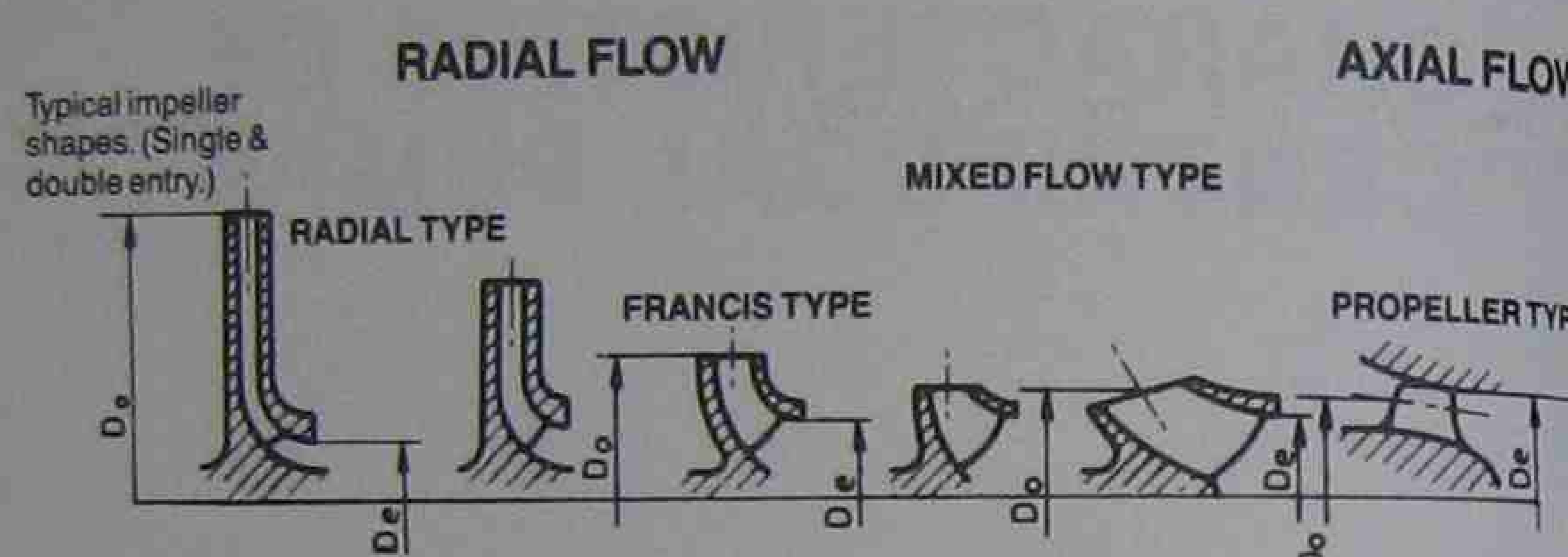


Fig. 3.1

(1) Radial Flow

The radial flow impeller discharges the fluid radially at 90° to the shaft axis.

(2) Axial Flow

The axial flow impeller discharges fluid along the shaft axis. For this reason an axial flow pump is by definition not "centrifugal" in its pumping action.

(3) Mixed Flow

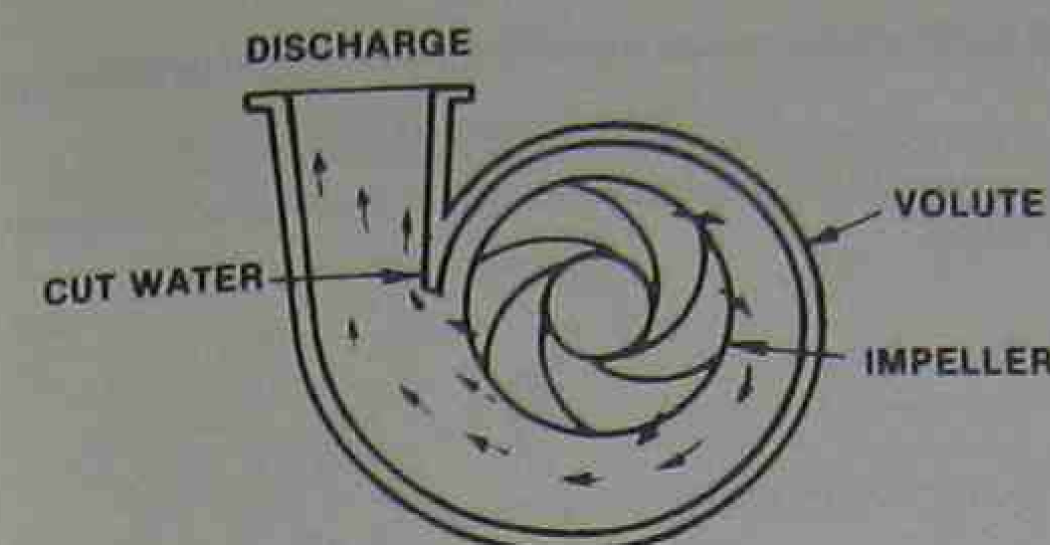
The mixed flow impeller discharges fluid in a conical direction using a combined radial and axial pumping action—as suggested by the title.

(4) Inducers

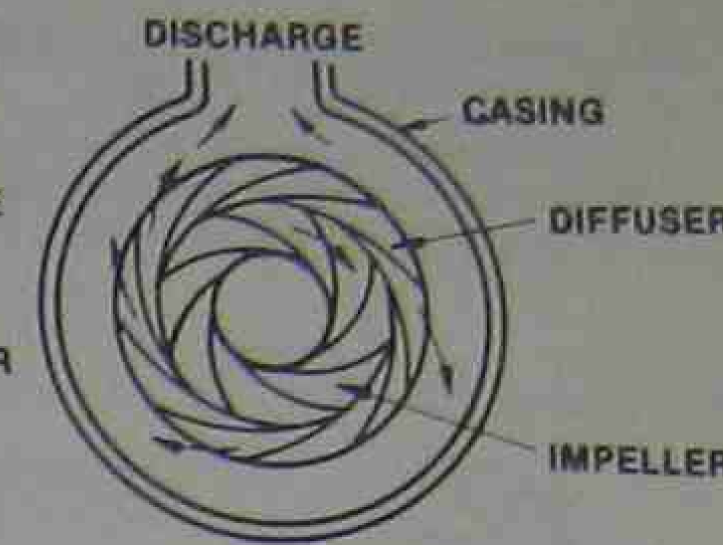
These are fitted on the suction side of first stage impellers. They are basically axial flow impellers with extended vanes. This enables them to operate in a cavitating condition with only a small head drop since the vapour cavity occupies only a relatively small length of passage. It thus generates enough head whilst cavitating to enable the first impeller to operate without cavitation.

Casing

The velocity head of water leaving the impeller is converted into pressure head in the casing either by means of a volute (Fig. 3.1.2a) or by a set of stationary diffuser vanes surrounding the impeller periphery (Fig. 3.1.2b).



VOLUTE TYPE
Fig. 3.1.2a

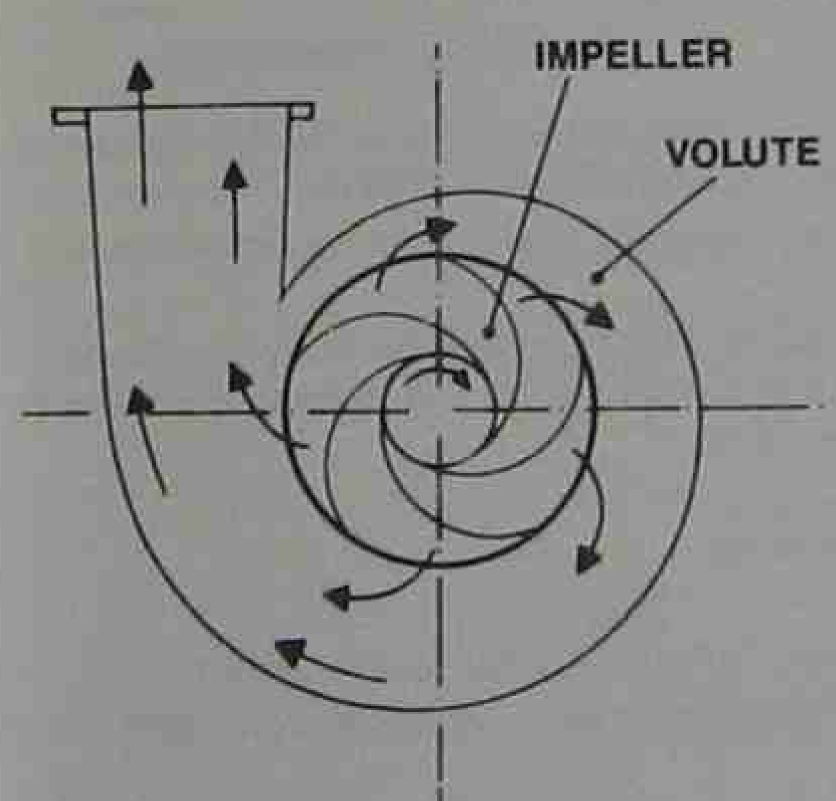


DIFFUSER TYPE
Fig. 3.1.2b

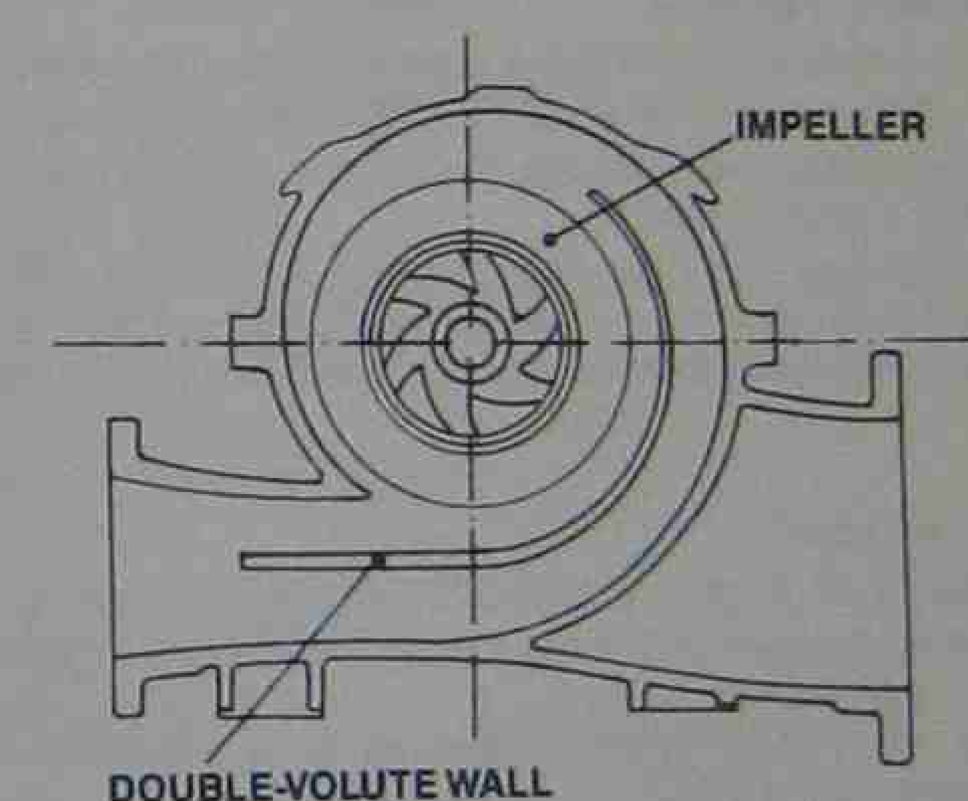
(1) Volute Casing

The volute is the most common form of casing (Fig. 3.1.2a). The volute increases in cross sectional area from the cut-water to the inner end of the discharge cone so as to give a near constant average water velocity.

Most of the conversion from velocity head to pressure head takes place in the discharge cone. However uneven pressure distributions around the impeller may give rise to undesirable radial loading on the shaft, particularly when operating at reduced flows. The double volute design (Fig. 3.1.3b) reduces this radial loading.



SINGLE VOLUTE CASING
Fig. 3.1.3a



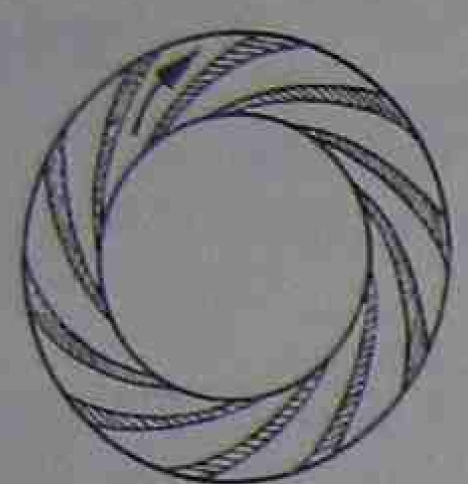
DOUBLE VOLUTE CASING
Fig. 3.1.3b

(2) Diffuser

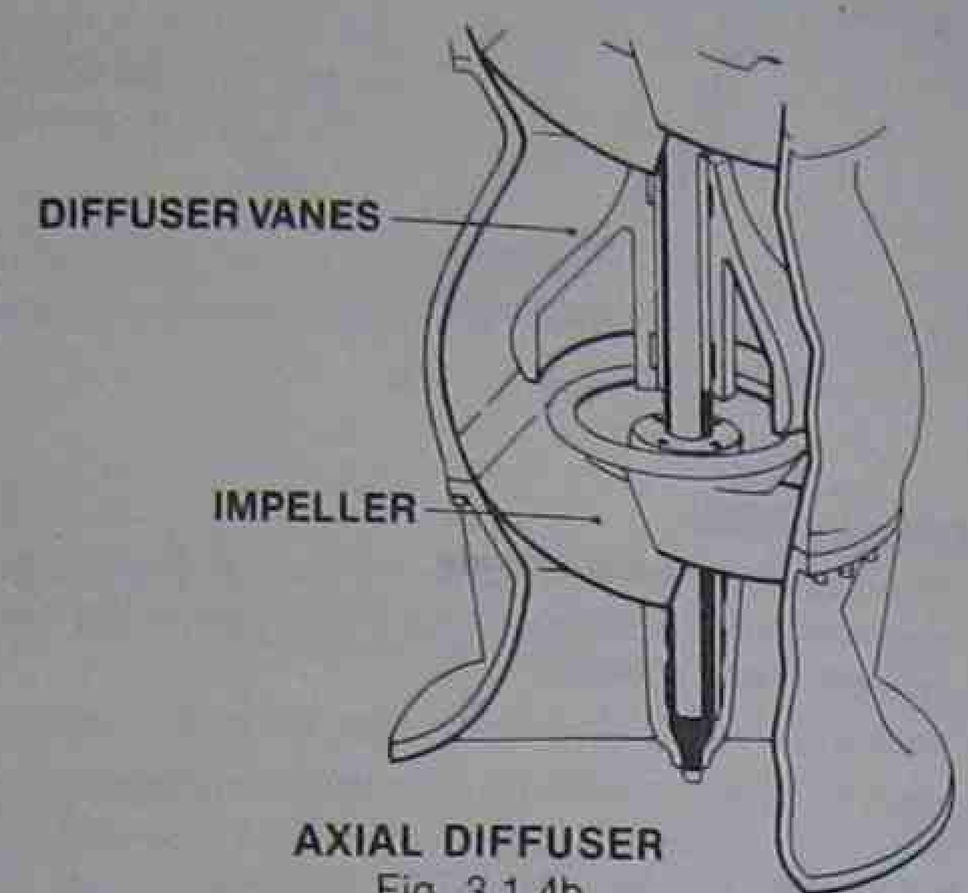
The diffuser fits inside the pump casing and guides the flow smoothly into the discharge pipe (or the next impeller in the case of a multi-stage pump).

The diffuser incorporates a number of vanes which form radially diverging water passages around the periphery of the impeller and recover a significant percentage of the total head. Most diffusers have radial vanes (Fig. 3.1.4a).

Axial diffusers (Fig. 3.1.4b) are used when it is desirable to reduce the outside diameter of the casing or to increase the clearance between the impeller and diffuser vanes in order to reduce vane tip erosion in very high speed pumps.



MULTI-VANE RADIAL DIFFUSER
Fig. 3.1.4a



AXIAL DIFFUSER
Fig. 3.1.4b

Shaft & Bearings

The primary function of the shaft is to transmit the driving torque to the impeller. With the support of the bearings, the shaft must also locate the impeller radially and axially within the casing.

Single stage overhung impeller pumps normally operate in the stiff shaft mode i.e. below the first shaft critical speed.

Mutlistage pumps may often operate in the flexible shaft mode, running above the first critical speed. With these pumps the hydrodynamic support and damping afforded by the internal clearances normally guarantees satisfactory "wet" operation.

Axial Thrust

Axial thrust is generated by the internal pressures acting on the impeller and shaft end.

One component of this axial thrust is that due to, and dependent only on, suction pressure and is shown on Fig. 3.1.5a.

The other major thrust component for a horizontal pump is that created by an "hydraulically unbalanced" impeller as shown in Fig. 3.1.5b. This effect is due to the presence of an unopposed pressure at the impeller back shroud. For small pumps with a suction diameter similar to the shaft diameter, this effect is minimal.

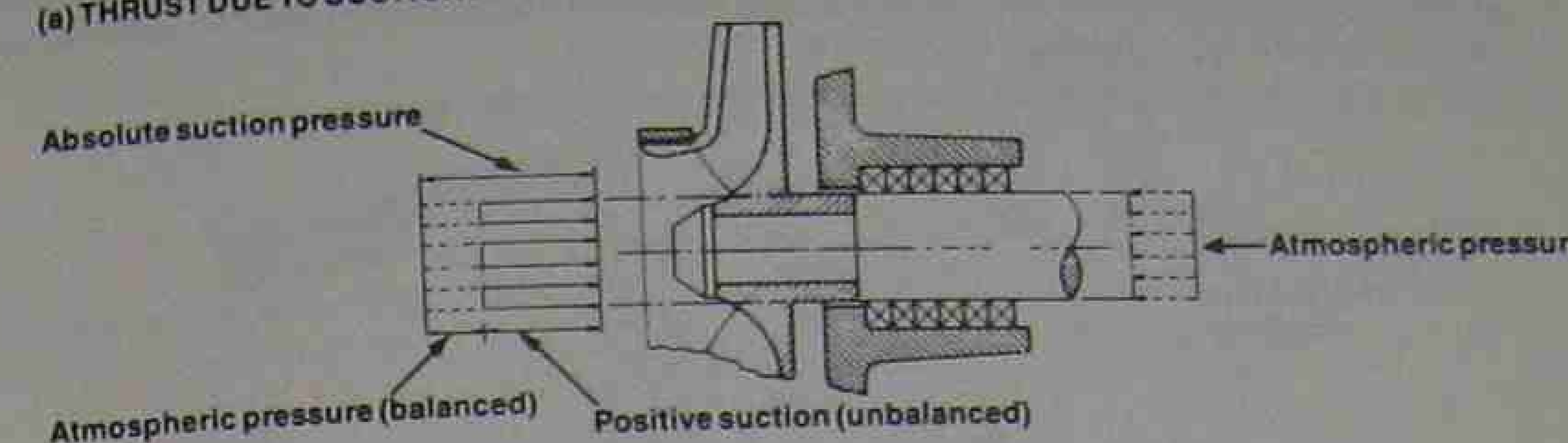
However, for larger pumps and/or higher head pumps the generated thrust can be very large. The most common solution to this problem is to use an "hydraulically balanced" type of impeller fitted with a back ring and balance holes, which reduce the pressure at the rear of the impeller hub by allowing leakage through to the suction side.

A similar reduction in pressure can be achieved through the use of impeller rear (pump out) vanes. Both of these methods are shown in Fig. 3.1.5c.

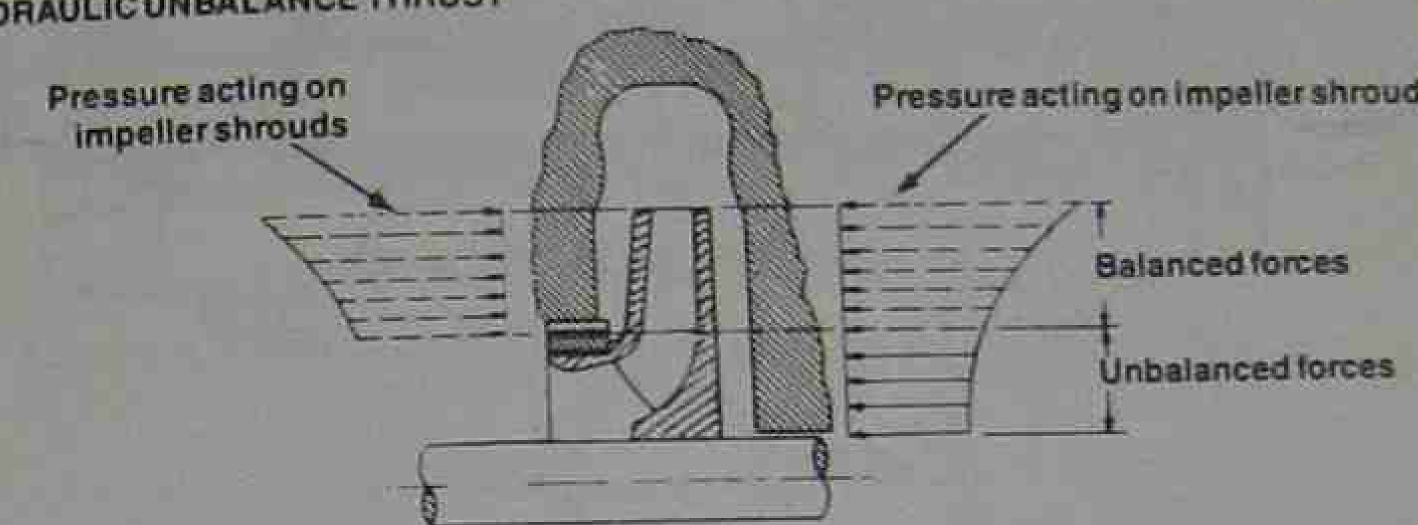
Multistage pumps may have opposed impellers for axial thrust balance, or a balancing device such as a balance disc or balance drum.

AXIAL THRUST

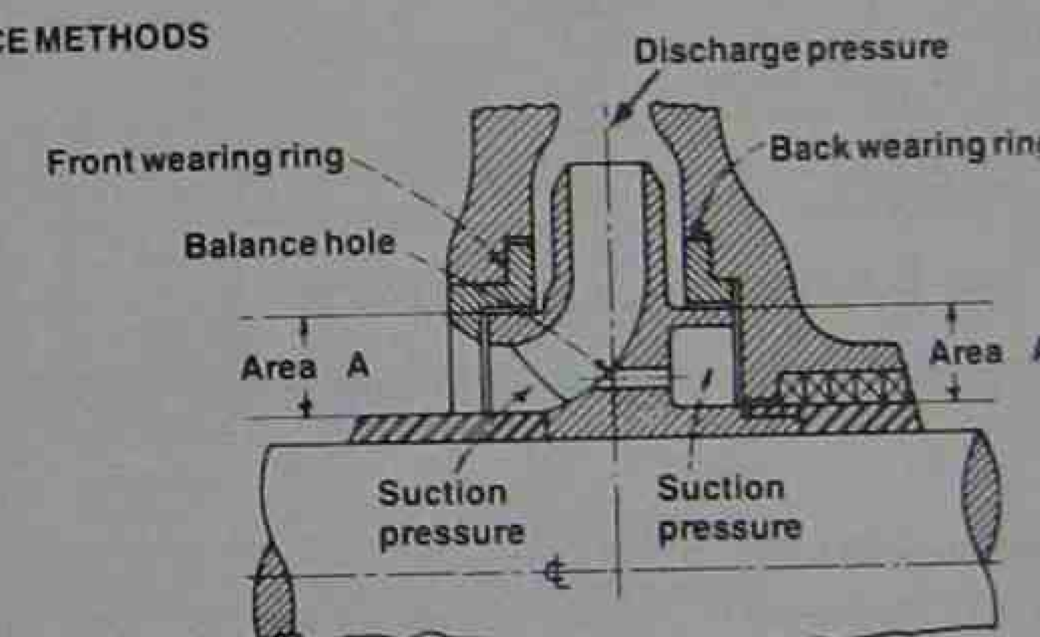
(a) THRUST DUE TO SUCTION PRESSURE



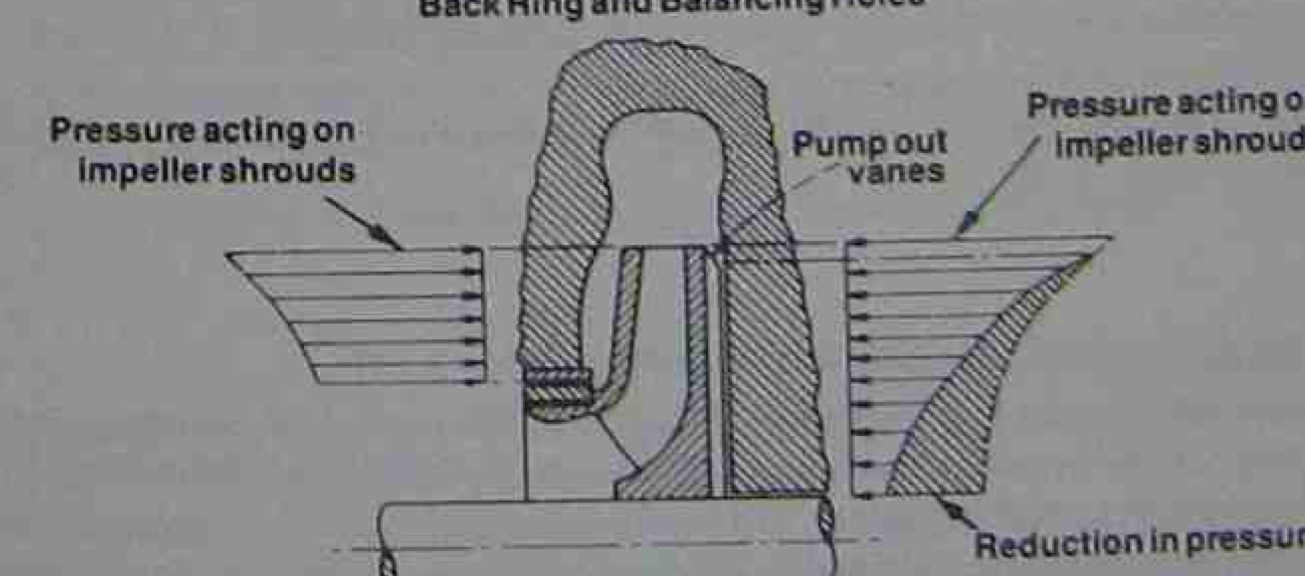
(b) HYDRAULIC UNBALANCE THRUST



(c) BALANCE METHODS



Back Ring and Balancing Holes



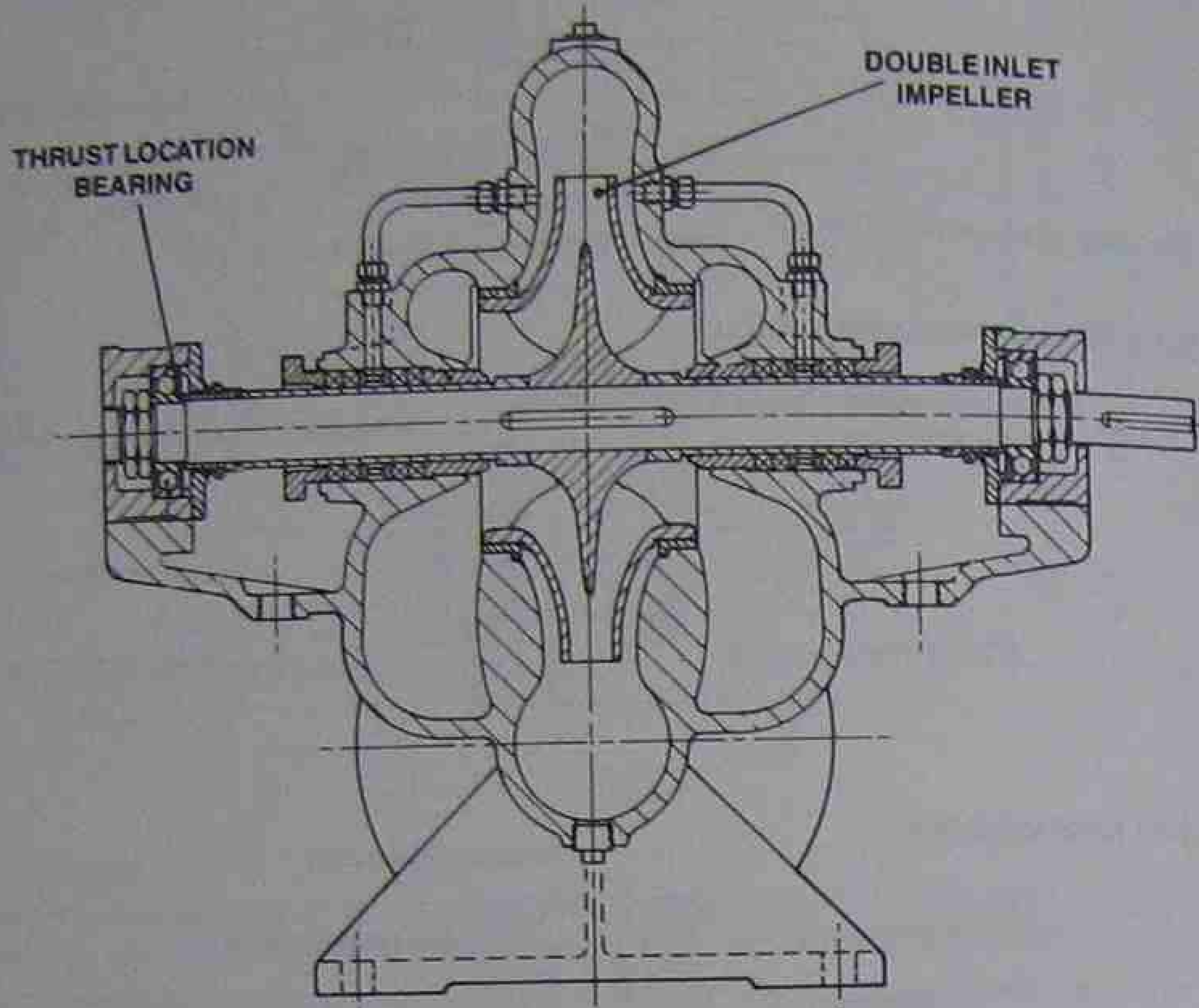
Rear Pump Out Vanes

Fig. 3.1.5

(1) **Single Stage Pump with Double Inlet Impeller** (Fig. 3.1.6)

Theoretically a pump of this type should be in complete axial balance, however, the presence of slight casting differences will cause the flow pattern to each impeller entry (eye) to marginally differ thus creating a residual axial thrust.

This is taken by a ball bearing of the combined radial/thrust type.



HYDRAULIC AXIAL BALANCE

Fig. 3.1.6 Hydraulic Axial Balance With Double Inlet Impeller.

(2) **Single Stage Overhung Impeller** (Fig. 3.1.7)

In this design the wear ring on the back impeller simulates another "eye" at the rear of the impeller. The impeller is almost completely hydraulically axially balanced by the "balance holes" in the back shroud close to the hub which allow leakage through to the suction side.

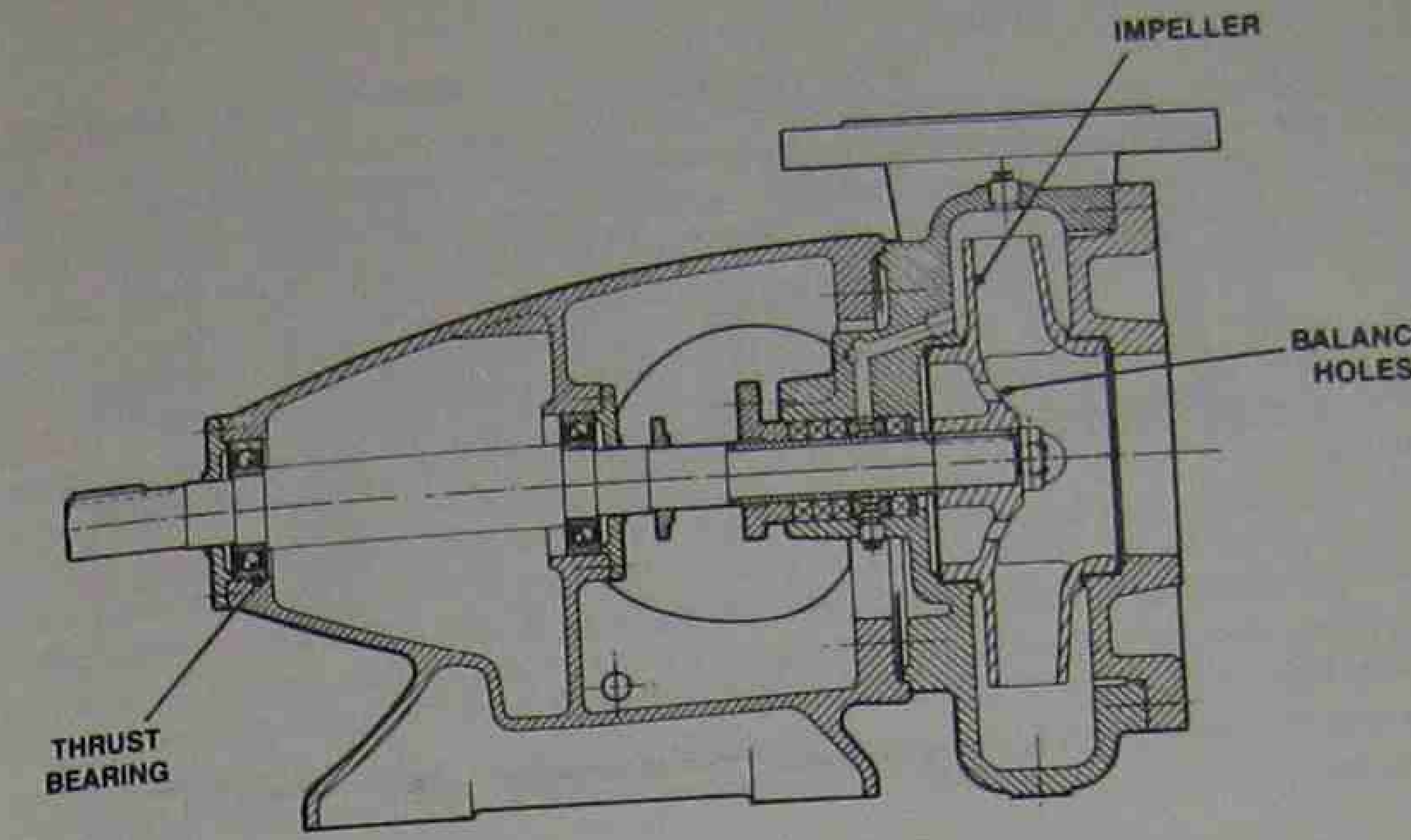
There is an unbalanced load towards the driver due to the suction pressure acting on the shaft area through the stuffing box. A thrust bearing (tilting pad or ball) is fitted to take this normally small remaining axial thrust.

(3) **Multistage Horizontal Pump with Balance Disc** (Fig. 3.1.8)

The unbalanced axial thrust is approximately equal to the pump differential pressure acting on the annular area of the impeller back shroud; this being roughly equivalent to the area of the impeller eye minus the shaft area.

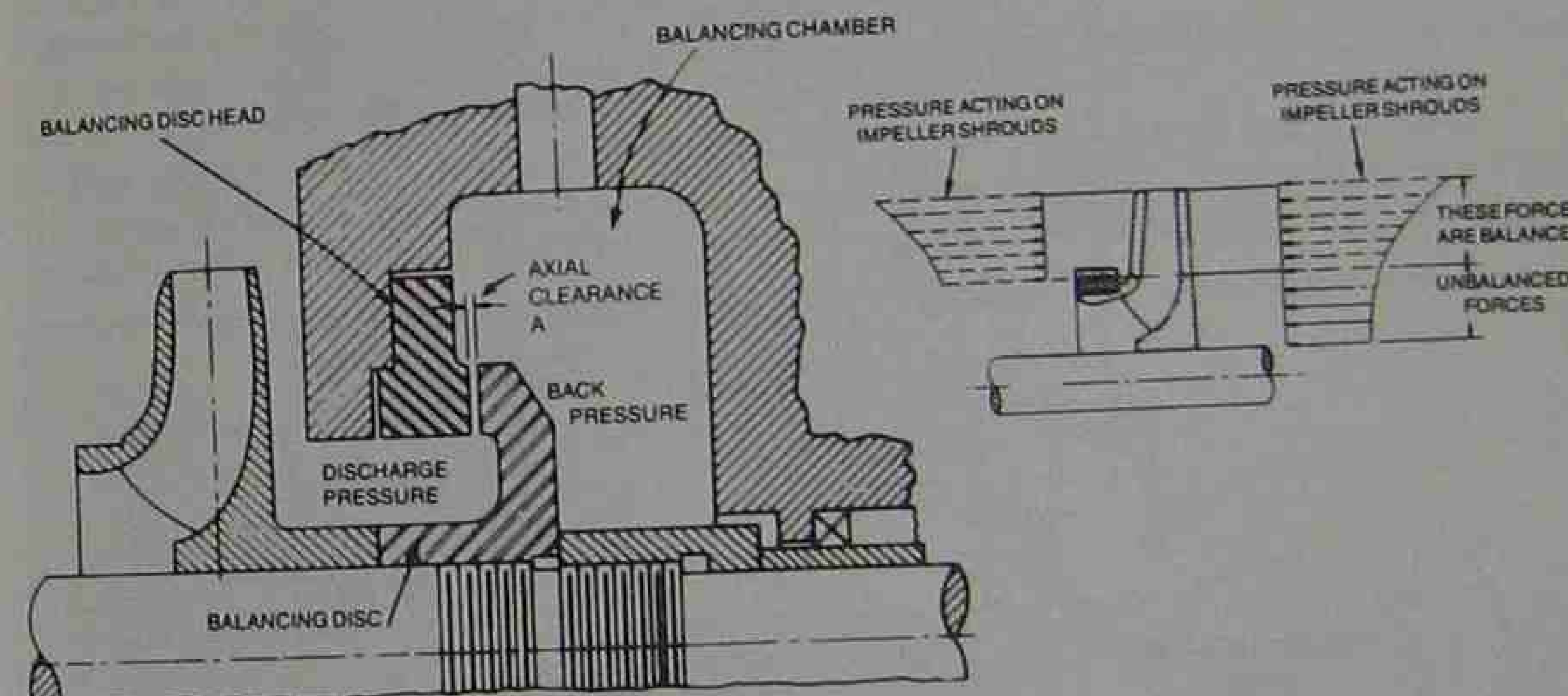
The balance disc rotates with the shaft and automatically adjusts the gap at A so that the pressure in the inner balance chamber opposes the unbalanced axial thrust of the impellers.

This balance device is entirely self compensating and no thrust bearing is required.



HYDRAULIC AXIAL BALANCE

Fig. 3.1.7 Hydraulic Axial Balance Single Stage Overhung Impeller.

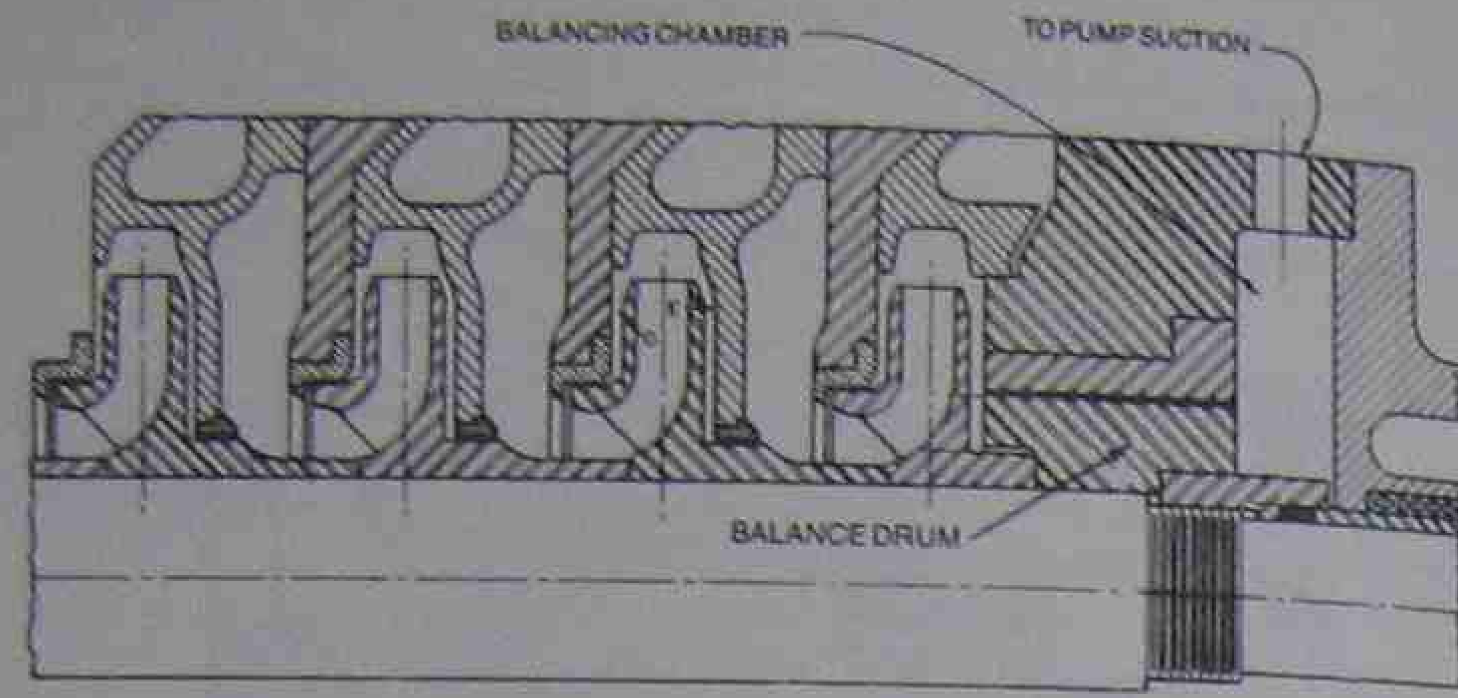


AXIAL BALANCE

Fig. 3.1.8 Axial Balance Typical Balance Disc Arrgt.

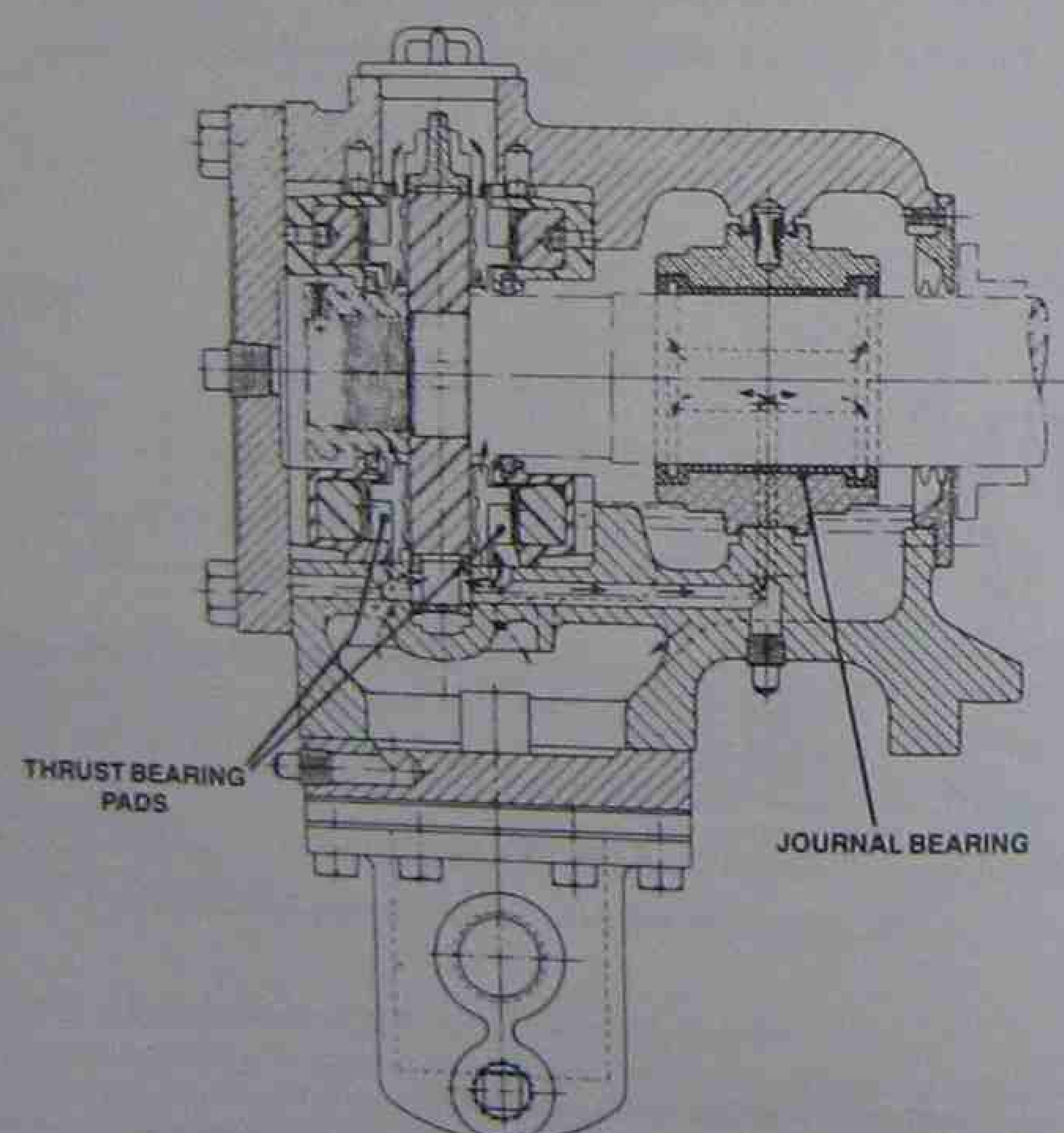
(4) **Multistage Pump with Balance Drum** (Fig. 3.1.9)

The area of this balance drum is approximately the same as the impeller unbalanced area but slightly undersize to ensure a residual unbalanced load in order to keep the shaft in tension. This axial load is taken by a thrust bearing, in this case a tilting pad bearing of the Michell type as shown in Fig. 3.1.10.



AXIAL BALANCE

Fig. 3.1.9 Axial Balance Typical Balance Drum Arrgt.



TILTING PAD THRUST BEARING

Fig. 3.1.10 Axial Balance Devices.

Radial Thrust

When a single volute pump is operated at best efficiency flowrate the velocities and hence the pressures acting on the impeller are uniform around the volute. This is shown in Fig. 3.1.11a.

At flowrates other than best efficiency point, the pressure distribution is no longer uniform.

At reduced flowrates, the pressures increase spirally towards the cutwater (see Fig. 3.1.11b), resulting in a radial reaction F. A similar situation exists at flowrates beyond best efficiency flowrate, with an approximately opposite (in direction) reaction.

Fig. 3.1.11c shows typical variation of radial thrust with flowrate.

Note that maximum radial thrust occurs at zero flow with the minimum at the best efficiency flowrate.

The magnitude of the radial thrust is a function of the total head, impeller diameter and impeller width.

Thus high head pumps with large impeller diameters will experience very high radial thrusts. If pumps of this type are operated at consistently low flowrates, bearing life will be reduced and the bending stresses associated with large shaft deflections may eventually lead to shaft failure.

The manufacturer may design for this problem by supplying an oversize shaft with larger bearings, however, this solution leads to a more expensive unit, especially in the case of large high head pumps.

An alternative is to reduce the value of radial thrust through the use of a double volute as shown in Fig. 3.1.11d. In such a casing the flow is divided into two almost equal streams by two cutwaters 180° apart.

Although the volute pressure inequalities remain, there are now two opposing radial forces which almost cancel.

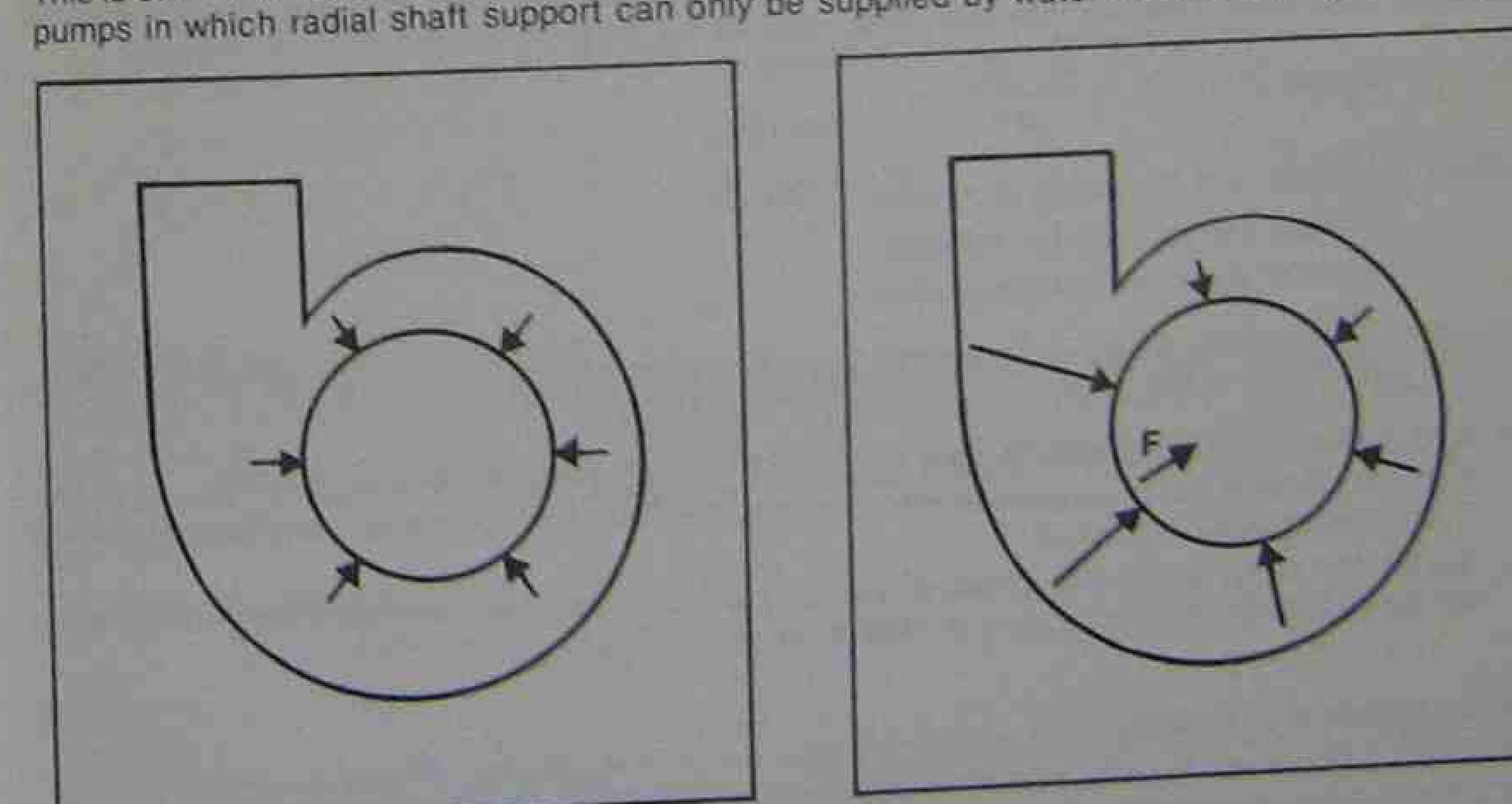
In practice the cancellation is not complete but nevertheless a major reduction in radial thrust at partial capacities is achieved.

Double volute casing pumps are most often found in the petro chemical industry where consistent operation at partial capacities is commonplace.

They are also occassionally found in high head per stage water pumps. Efficiency is slightly lower due to the additional wetted area.

Diffuser casings also virtually eliminate radial thrust in the same way as a double volute.

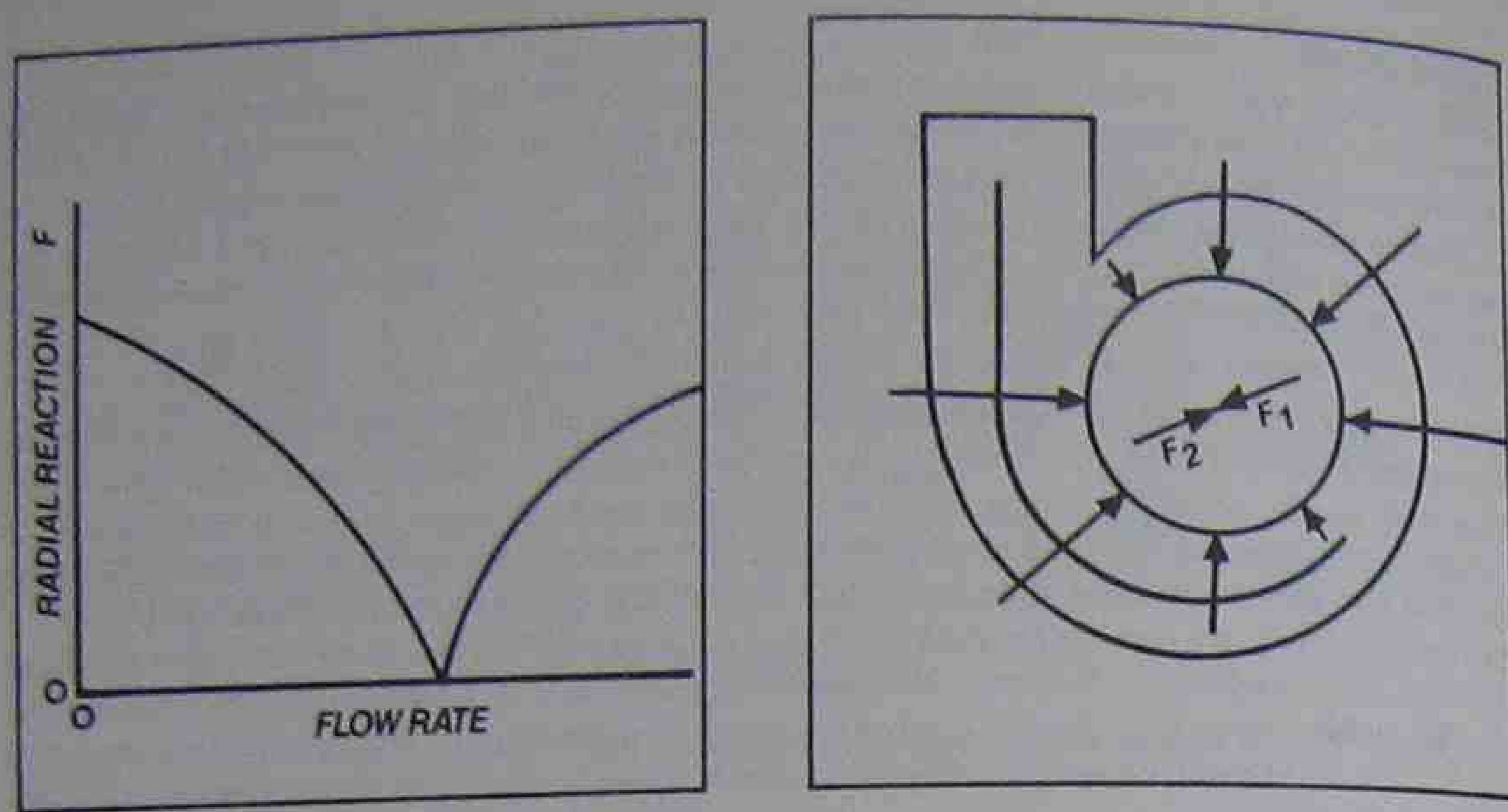
This is one of the reasons behind the reliable operation of diffuser type deep well vertical turbine pumps in which radial shaft support can only be supplied by water lubricated plain bearings.



(a) 100% BEP FLOW

(b) REDUCED FLOW

Fig. 3.1.11



(c) RADIAL THRUST

(d) DOUBLE VOLUTE

Fig. 3.1.11

3.2 SPECIFIC SPEED

The characteristic differences between pumps can be described by a criterion referred to as "specific speed", this being the speed of an ideal pump geometrically similar to the actual pump, which when running at this speed will raise a unit of volume in a unit of time through a unit of head.

The performance of a pump is expressed in terms of pump speed, total head and flowrate. Specific speed is calculated from the formula using data at the best efficiency point as follows:

$$\text{Specific Speed Ns} = \frac{N \sqrt{Q}}{H^{3/4}}$$

where N = speed of the pump in r.p.m.
Q = flowrate in litres per second
H = total dynamic head in metres

Specific speed relates to the geometry of the pump rotor and is independent of the pump size.

When applying the formula above, it should be noted that:

a) With a double suction impeller, half the best efficiency flowrate should be used.
b) With a multistage pump, best efficiency conditions for the first stage impeller only are considered in calculating specific speed.

The normal range of specific speed is from 500 to 12500 and the relationship between specific speed and basic impeller shape is illustrated in Fig. 3.2.1.

Units of Specific Speed

Since specific speed is used only as an index or type number, certain liberties are permissible in selecting the units used.

Consequently the numerical value of Ns will vary according to the units in which H and Q are expressed. The speed of the impeller is always given in r.p.m.

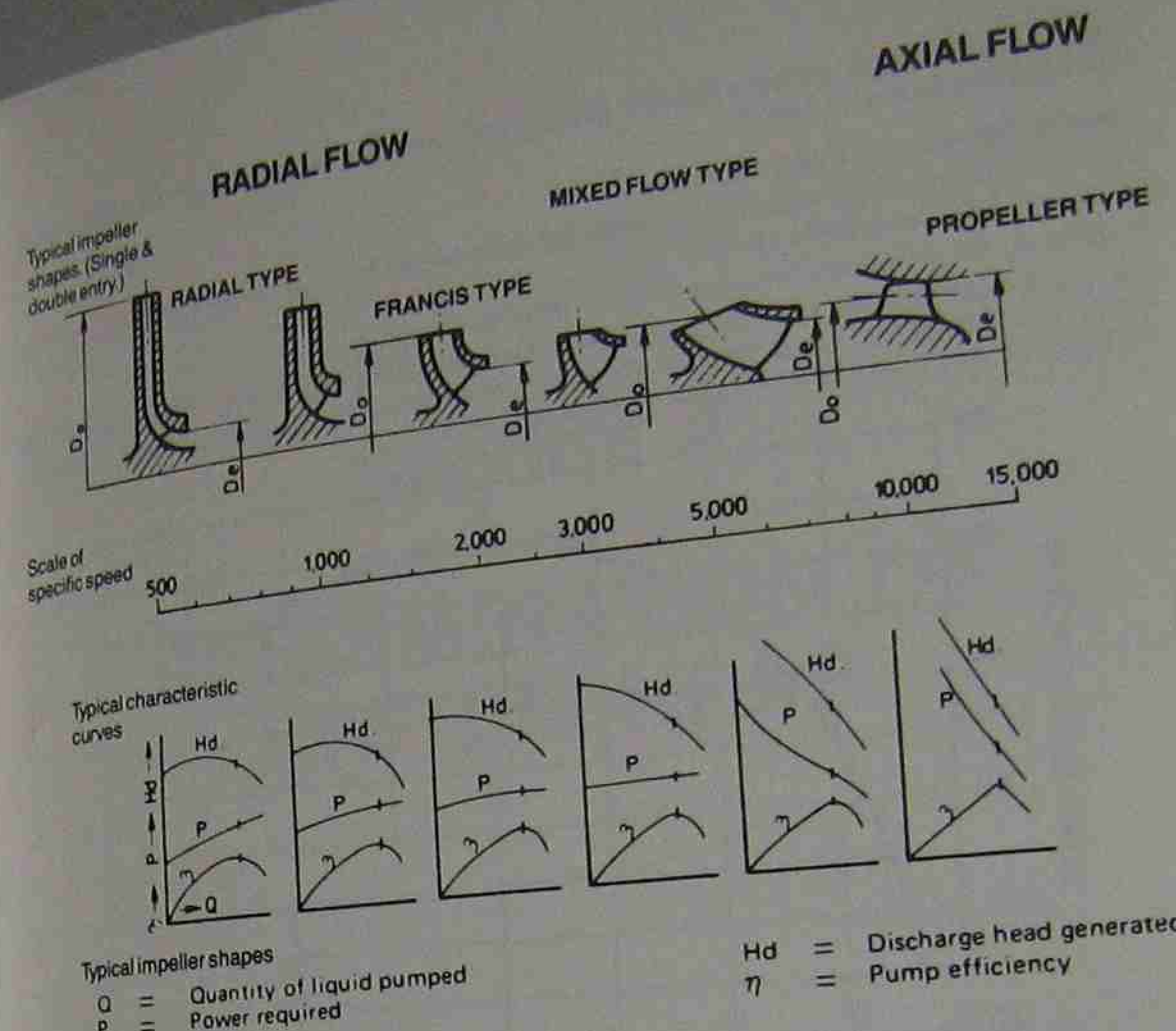


Fig. 3.2.1

The following table shows the relationship of specific speed in various units:

Metric Ns	British Ns	U.S.A. Ns
$\frac{Q\,(L/s)H\,(m)}{1}$	$\frac{Q\,(igpm)H\,(ft)}{1.5}$	$\frac{Q\,(usgpm)H\,(ft)}{1.63}$

Specified Speed and Pump type

By categorising centrifugal pumps in terms of specific speed, the pump user can visualise the type of pump he is likely to need.

e.g. suppose a pump is required to deliver 100 L/s at a head of 89m at 1450 r.p.m. Using the equation:

$$Ns = \frac{1450 \times \sqrt{100}}{89^{3/4}}$$

$$= 500$$

This is a fairly low specific speed radial flow pump. Hence the user can expect a large pump with relatively low efficiency. However, if the speed is doubled at 2900 r.p.m. the specific speed of the pump will be 1000. Consequently the pump can be smaller with a higher efficiency which may allow a less powerful driver, and make the overall pumpset package much less expensive. Civil costs would also be correspondingly reduced.

Hence specific speed can be a valuable guide to the pump user in planning and designing a more cost effective pumping station.

Specific Speed and Pump Efficiency

The maximum attainable efficiency of centrifugal pumps depends to a great extent on pump geometry as categorised by specific speed.

In general, the efficiency increases as the value of Ns rises.

Fig. 3.2.2 illustrates the relationship between pump efficiency and specific speed. The trend of the curve indicates that for economical operation very low values of Ns are to be avoided; a condition that can be overcome by employing multistage pumps.

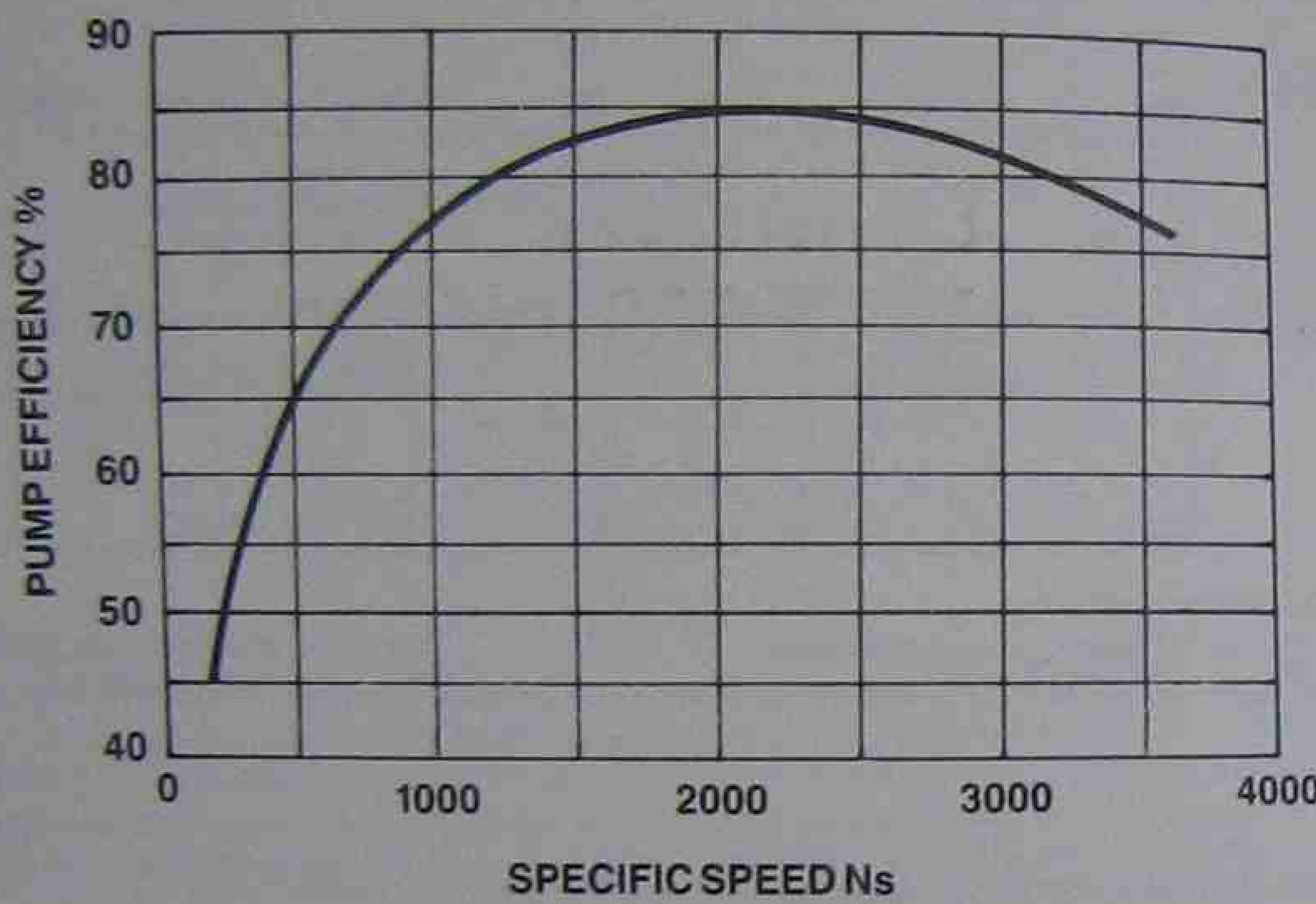


Fig. 3.2.2

Specific Speed and Characteristic Curves:

Specific speed not only relates to impeller geometry, but also the trend of the characteristic curves for total dynamic head, power consumption and efficiency, examples of which are shown in Fig. 3.2.1.

The steepness of the H-Q curve increases as specific speed increases, efficiency curve remains relatively unchanged, except to exhibit a narrow plateau when Ns is high.

At low specific speed, power consumption is a minimum at zero flowrate and rises with flowrate. This is an overloading characteristic and the motor must be sized to meet the extreme conditions that may occur in operation.

At medium specific speed the power curve has a pronounced peak at approximately the design duty. This is a non-overloading characteristic and therefore the pump can work safely over the entire flow range with a motor sized to meet the peak requirements.

High specific speed pumps have a falling power characteristic with maximum power occuring at minimum flow.

Such pumps should not, therefore, be used in zero flow conditions.

Discharge throttle valves or by-pass systems should be avoided with this type of pump otherwise a motor of considerably greater power than necessary for normal duty will be required.

Section 4

DETERMINATION OF SYSTEM HEAD

Section 4—DETERMINATION OF SYSTEM HEAD

4.1 FRICTION LOSSES IN PIPES AND FITTINGS:

The resistance as a liquid flows through a pipe results in a loss of head, commonly called friction loss, and is measured in metres of liquid or head.

The head loss for a given system is a function of the surface roughness of the interior of the pipe wall, the inside diameter of the pipe, the fluid velocity, fluid viscosity and the length of travel. In addition, losses are caused by obstructions in the flow path due to valves, fittings, etc., changes in direction and sudden or gradual changes in the cross-section or shape of the flow path.

PIPE FRICTION:

The friction loss in a pipe may be calculated using the Darcy-Weisbach equation which is:—

$$h_f = f\,\frac{L}{D}\,.\,\frac{v^2}{2g}$$

Where h_f = friction loss between two cross-sections (m of liquid).
- L = Length between two cross-sections (m)
- D = internal diameter of pipe (m)
- v = average fluid velocity (m/s)
- g = acceleration due to gravity (m/s^2)
- f = friction factor - a dimensionless number, which for turbulent flow depends on the roughness of the pipe's interior surface and the Reynolds number.

The Reynolds number R is given by:

$$R = \frac{vD}{\nu}$$

where ν = kinematic viscosity of the fluid (m^2/s)

For laminar flow ($R < 2000$) the friction factor is a function of the Reynolds number only, and the roughness of the pipe's interior has no effect. In this instance the friction factor f becomes:

$$f = \frac{64}{R}$$

For turbulent flow ($R > 4000$) the friction factor is dependant on both the relative roughness and the Reynolds number and can be calculated from the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -2\log_{10}\left[\frac{\epsilon}{3.7D} + \frac{2.51}{R\sqrt{f}}\right]$$

where ϵ = absolute roughness (mm)—see also Fig. 4.1.1.
- = 0.04572 for new commercial steel pipes.
- = 0.12192 asphalt lined cast iron.
- = 0.00152 smooth drawn tubing.

Alternatively, f may be obtained from the diagram developed by Professor L. F. Moody shown in Fig. 4.1.2. This diagram shows the relation between the friction factor f, the Reynolds number R, and the relative roughness ϵ.

A region known as the "critical zone" occurs between R = 2000 and 4000. In this region, the flow may be either laminar or turbulent and the friction factor has the lower limits based on laminar flow and the upper limits based on turbulent flow conditions.

Although the Darcy-Weisbach/Colebrook method offers a mathematical solution to friction loss calculations, other empirical formula have been developed for water flowing under turbulent conditions.

RELATIVE ROUGHNESS FACTORS FOR NEW CLEAN PIPES

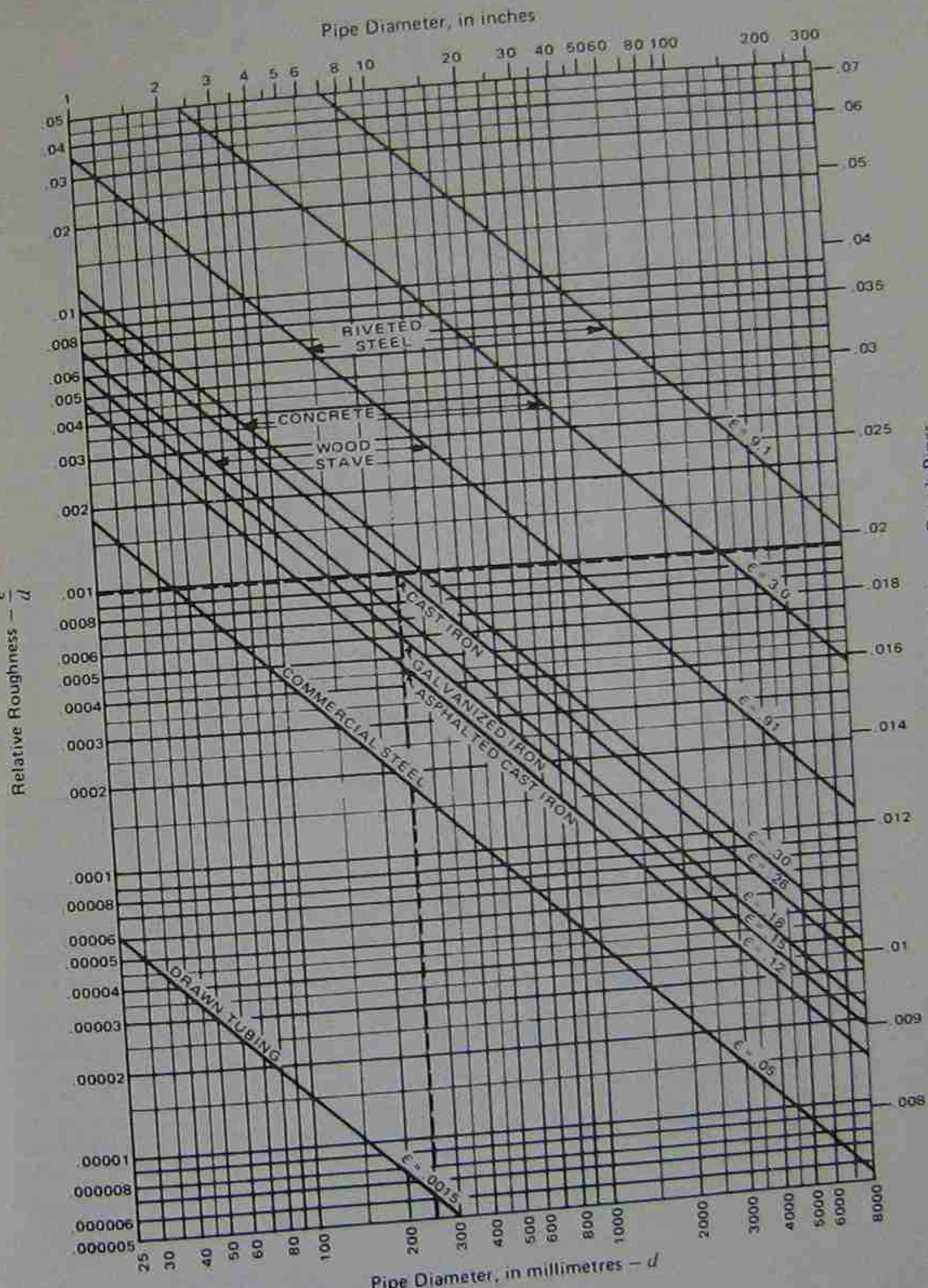


Fig. 4.1.1

MOODY DIAGRAM

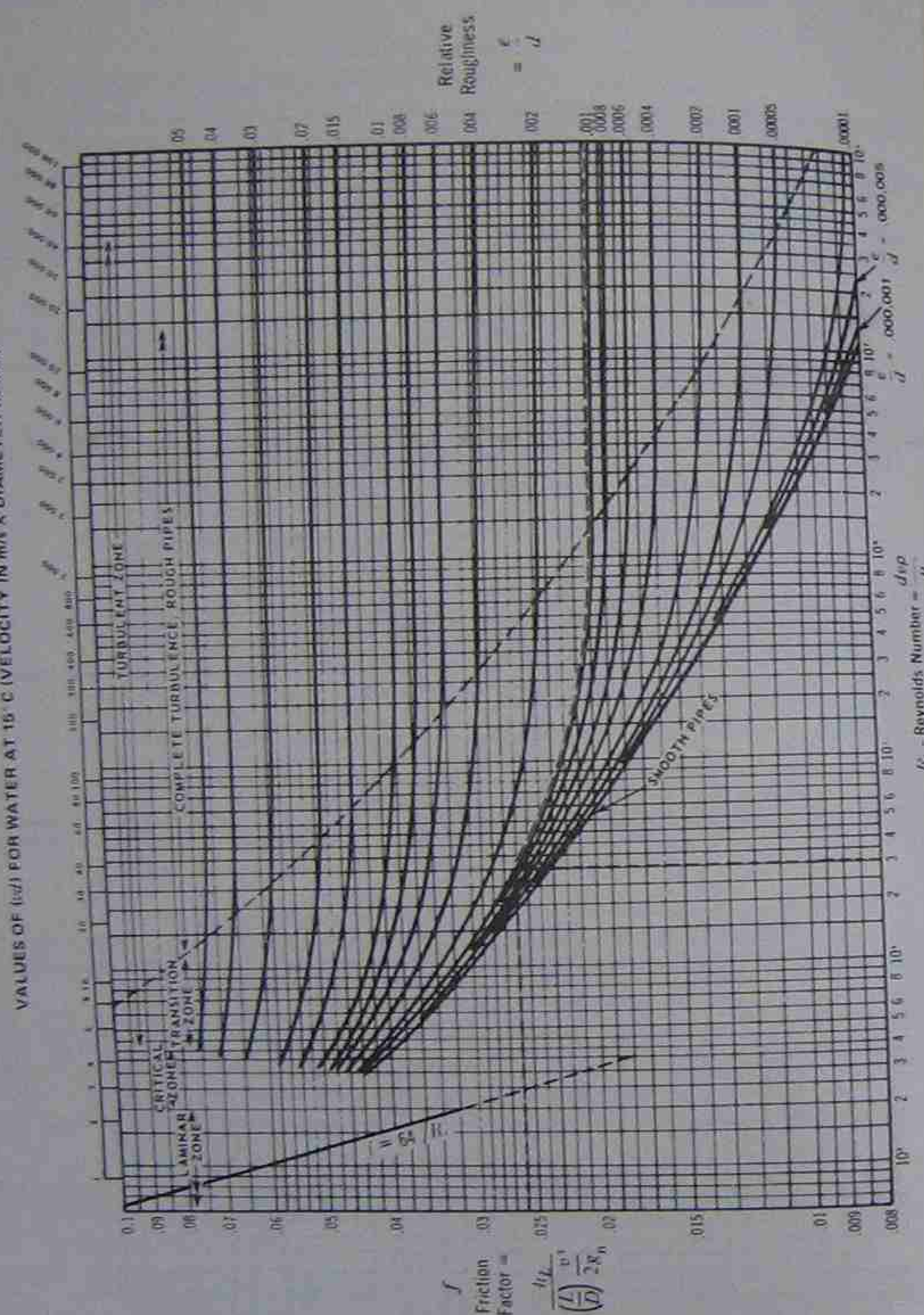


Fig. 4.1.2

One of the most widely used is the Hazen and Williams formula which reads:

$$H_L = \left[\frac{3.35 \times 10^6 Q}{d^{2.63} C} \right]^{1.852}$$

where H_L = head loss (m/100m)
Q = flow rate (L/s)
d = actual internal diameter of pipe (mm)
C = Hazen—Williams constant
= 125-140 Commercial steel pipe
= 135-140 Bitumen lined cast iron pipe
= 140-145 copper tube
= 145-150 P.V.C.

This formula is valid for a fluid having a kinematic viscosity of 1.137 mm^2/s which is the case for water at 15°C. But since the viscosity of water can vary appreciably from 0°C to 100°C, the friction can decrease or increase as much as 40% between the two temperature extremes. However, this formula can be used for any liquid having a viscosity up to 1.137 mm^2/s.

Further information including tables of head loss expressed in metres/100 metres for clean cold water for a range of pipe sizes and materials is given in the Australian Pump Manufacturer's Association publication 'Pipe Friction Handbook'.

VALVES AND FITTINGS:

Flow through a valve, fitting or change in section or configuration causes a head loss which is proportional to velocity head.

$$H_f = K \frac{v^2}{2g}$$

where K = resistance coefficient - defined as the number of velocity heads lost due to a valve, fitting etc..

The resistance coefficient K is usually considered as being independent of friction factor or Reynolds number, and may be considered as a constant for a given obstruction (i.e. valve or fitting) in a piping system under all conditions of flow.

Example

The following example, using the charts and tables contained in the APMA 'Pipe Friction Handbook' illustrates the application of the friction and head loss data in calculating the total system head for a specific system.

Problem:—referring to the accompanying figure, a pump takes water from a sump and delivers it through 380 m of 100 mm diamater schedule 40 steel pipe. The suction pipe is 100 mm vertical, 1.5 m long and includes a hinged disc foot valve and a standard 90 degree flanged elbow. The discharge line includes two standard 90 degree flanged elbows, a flanged swing check valve and an open gate valve. It is required to find the total system head when the rate of flow is 15 L/s and the total static head on the pump is 80 m.

Solution

1. Total length of pipe in system is:
suction : 1.5 m
discharge : 380 m
∴ total : 1.5 + 380 = 381.5 m

For a flow of 15 L/s through 100 mm schedule 40 pipe:

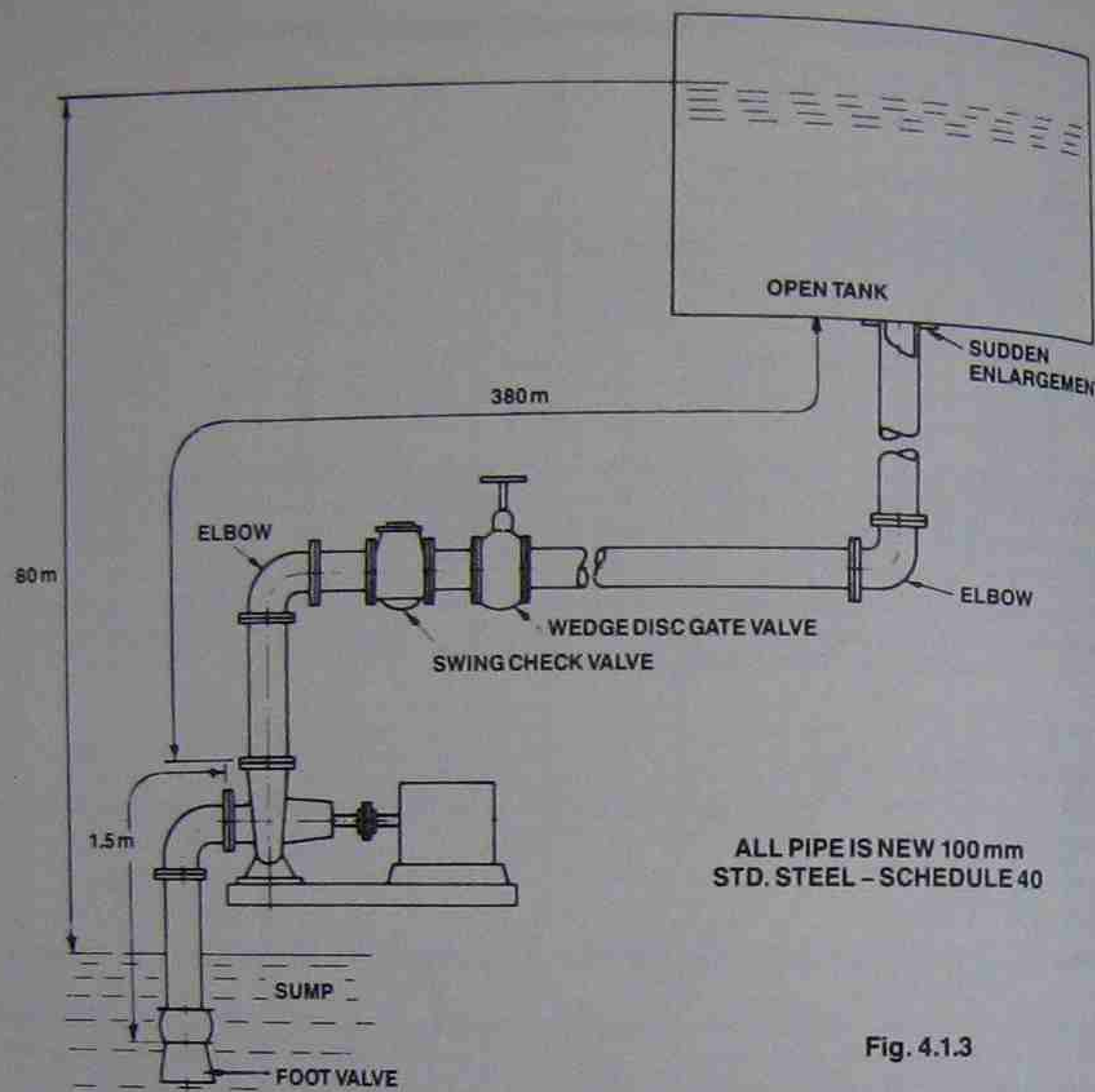


Fig. 4.1.3

Pipe friction loss = 3.15 m/100 m of pipe (Table A)

$\therefore$ The head loss due to pipe friction will be:

$$h_1 = 381.5 \times \frac{3.15}{100} = 12.02 \text{ m}$$

2. The resistance coefficient for the various fittings in the system as obtained from the tables for 100 mm schedule 40 pipe will be:

SUCTION

Standard 90 degree flanged elbow (Section 2.1.2)	K = 0.51
Hinged disc foot valve (Section 2.1.3)	K = 1.28

DISCHARGE

Standard 90 degree flanged elbow (Section 2.1.2)	K = 0.51
Flanged swing check valve (Section 2.1.3)	K = 0.85
Gate valve (Section 2.1.4)	K = 0.14
Sudden enlargement at exit (Section 2.1.5)	K = 1.00

The total resistance coefficient for the fittings on the pump suction and discharge and sudden enlargement at exit will be:

$K = 0.51 \times 3 + 1.28 + 0.85 + 0.14 + 1.0$

$\therefore K = 4.8$

For a flow of 15 L/s through 100 mm schedule 40 pipe.

Velocity head $= \frac{V^2}{2g} = 0.17$ m (Table A)

$\therefore$ The head loss due to fittings in the system will be:

$h_1 = K\frac{V^2}{2g} = 4.8 \times 0.17 = 0.82$ m

3. The total system head is equal to the sum of the total static head and the head losses due to friction in pipes and fittings.
i.e. $H = 80 + 12.02 + 0.82 = 93$ m

4.2 SAMPLE CALCULATIONS OF NPSHA

Basis of Calculations

1 standard atmosphere:	101.325 kpa
Gravitational acceleration:	$g = 9.80$ m/s^2

Physical properties of water are taken from APMA "Pipe Friction Handbook", page 112.

Calculation Method:

Nett Positive Suction Head Available can be simply calculated as follows:

H_{ts} = Total Absolute Suction Head at pump suction nozzle

H_{VAP} = Vapour Head of pumped liquid

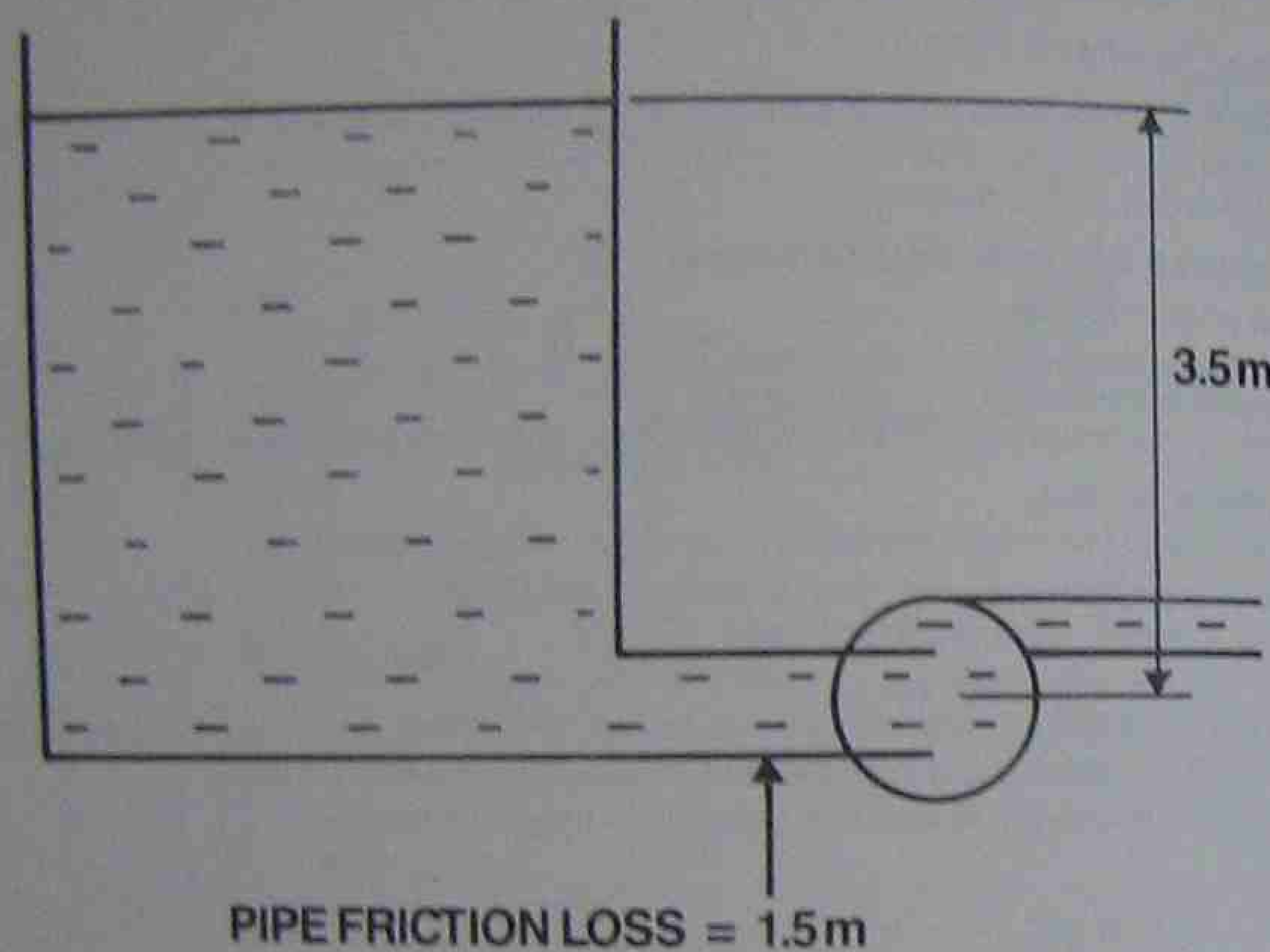
$$\mathbf{NPSHA = H_{ts} - H_{vap}}$$

Note that H_{ts} is "total" and thus must include the velocity head of the liquid at the pump inlet. This should be added to any pressures/heads obtained from gauge readings.

Suction Pipe friction losses would be calculated as shown in section 4.1.

EXAMPLE 1—OPEN TANK

Water at 15°C,
Density: 999 kg/m³
Vapour Pressure: 1.7 kPa(abs)
$g = 9.80\ m/s^2$



$NPSHA = H_{ts} - H_{VAP}$

$H_{ts} = 1\ \text{Atmosphere} + 3.5m - 1.5m$

$= \dfrac{101.325 \times 1000}{9.8 \times 999} + 3.5 - 1.5$

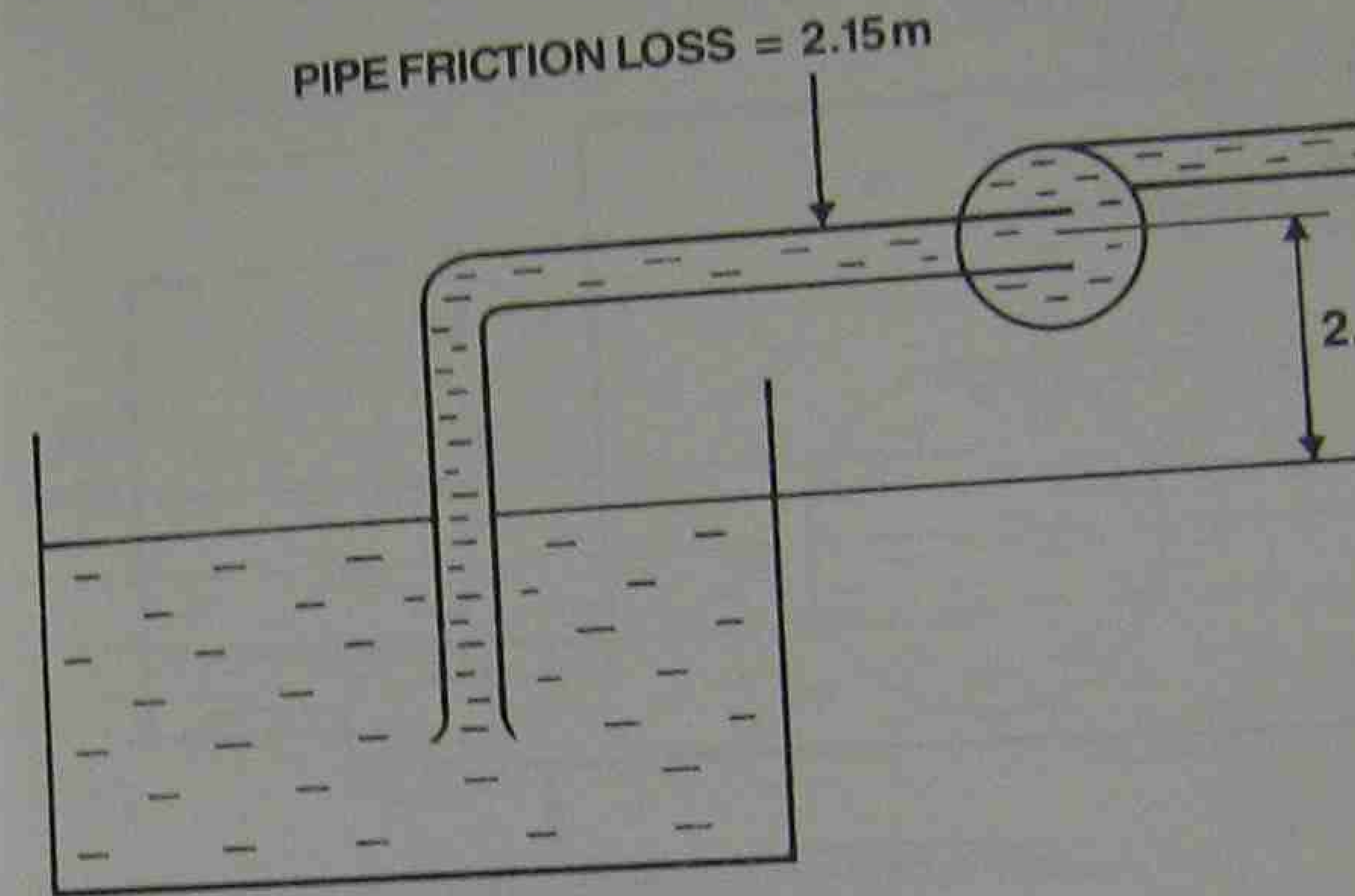
$= 12.35m$

$H_{VAP} = \dfrac{1.7 \times 1000}{9.8 \times 999} = 0.17m$

THUS $NPSHA = 12.35 - 0.17 = 12.18m$

EXAMPLE 2—OPEN TANK

Water at 26°C,
Density: 997 kg/m³,
Vapour Pressure: 3.2 kPa(abs)
$g = 9.80 m/s^2$



$H_{ts} = 1\ \text{Atmosphere} - 2.2m - 2.15m$

$= \dfrac{101.325 \times 1000}{9.8 \times 997} - 2.2 - 2.15$

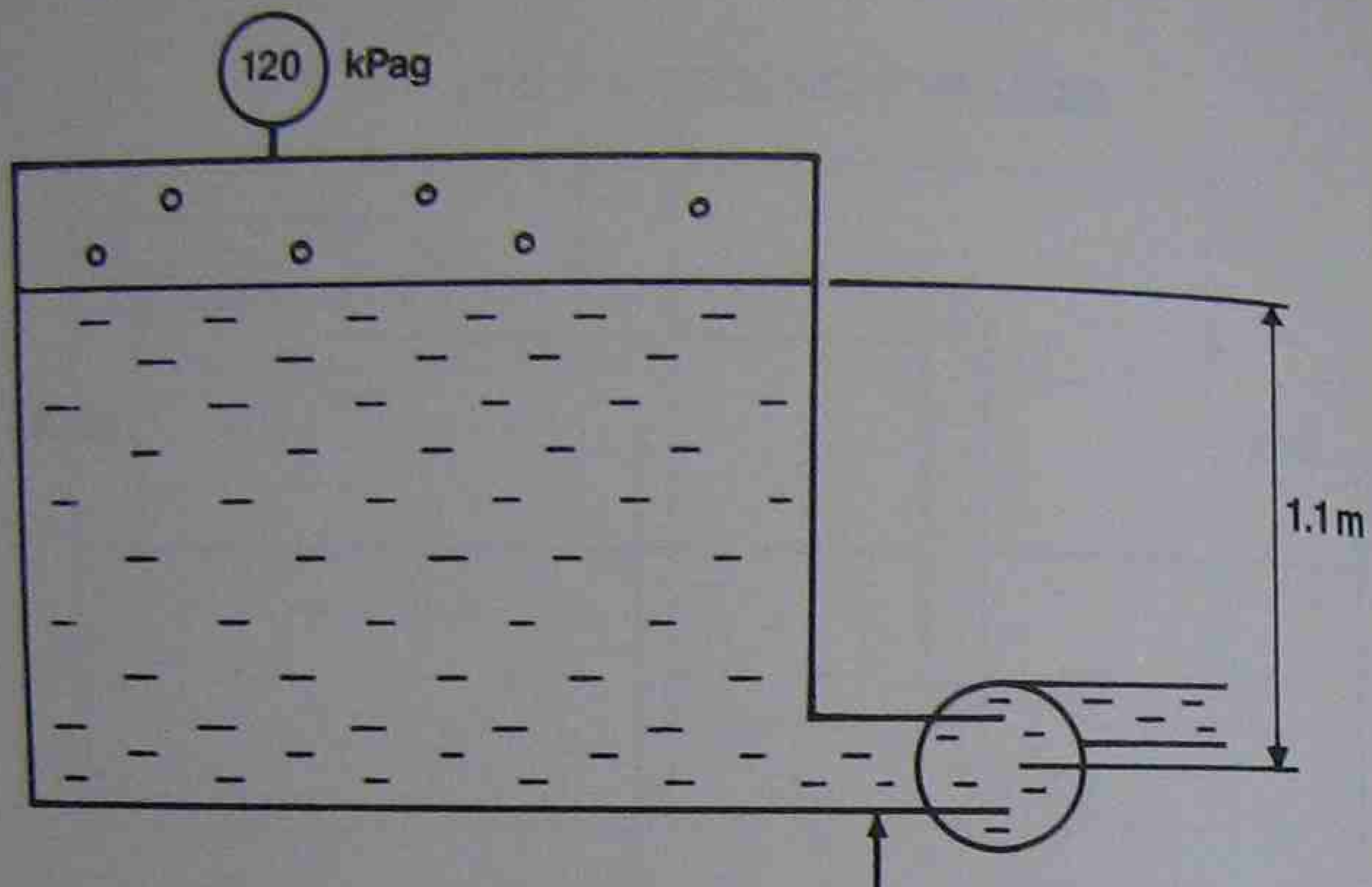
$= 6.02m$

$H_{VAP} = \dfrac{3.2 \times 1000}{9.8 \times 997} = 0.33m$

THUS $NPSHA = 6.02 - 0.33 = 5.69m$

EXAMPLE 3—CLOSED TANK

Water at 120°C,
Density: 943.4 kg/m^3,
Vapour Pressure: 198.5 kPa(abs)
$g = 9.80$ m/s^2



$$H_{ts} = \text{Gauge Reading} + 1 \text{ Atmosphere} + 1.1\text{m} - 1.5\text{m}$$

$$= \frac{(120 + 101.325) \times 1000}{9.8 \times 943.4} + 1.1\text{m} - 1.5\text{m}$$

$$= 23.54\text{m}$$

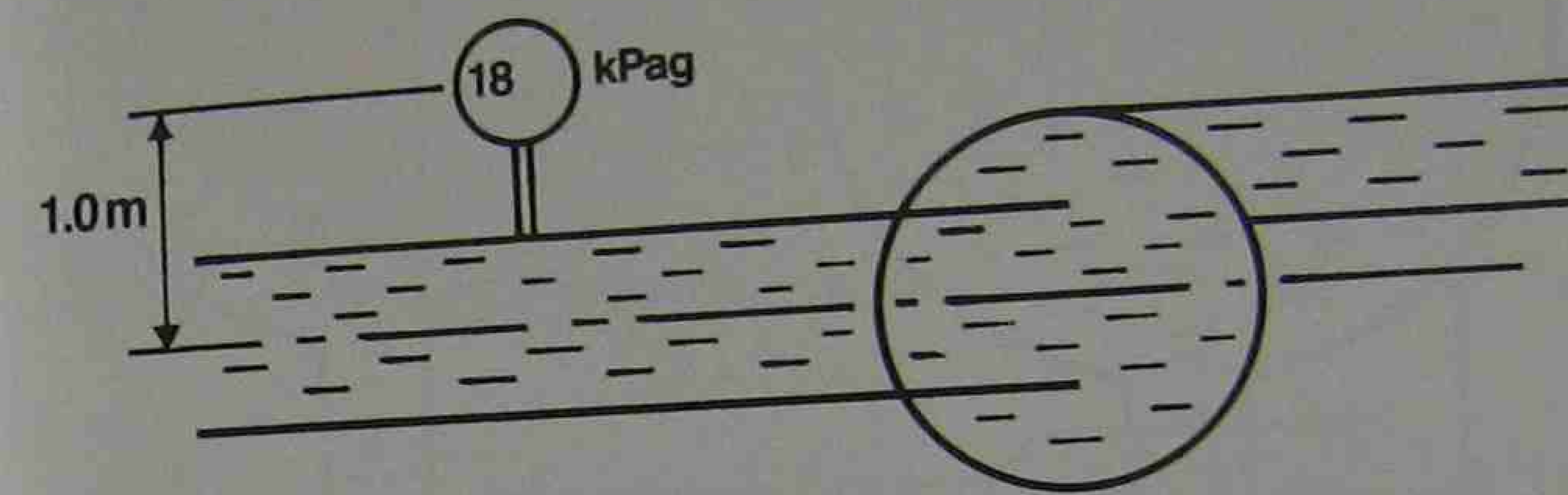
$$H_{VAP} = \frac{198.5 \times 1000}{9.8 \times 943.4} = 21.47\text{m}$$

THUS NPSHA $= 23.54 - 21.47 = 2.07$m

EXAMPLE 4—SUCTION GAUGE READING

Water at 40°C,
Density: 992 kg/m^3,
Vapour Pressure: 7.375 kPa(abs)
$g = 9.80$ m/s^2
Liquid Flow: 14 L/s Suction pipe diameter: 62.7mm
Average Suction Velocity $= \frac{\text{FLOW}}{\text{AREA}} = 4.53$ m/s

Velocity Head $= \frac{v^2}{2g} = 1.048$m



$$H_{ts} = \text{Gauge Reading} + \text{Gauge Height} + 1 \text{ Atmosphere} + \text{Velocity Head}$$

$$= \frac{18 \times 1000}{9.8 \times 992} + 1.0 + \frac{101.325 \times 1000}{9.8 \times 992} + 1.048\text{m}$$

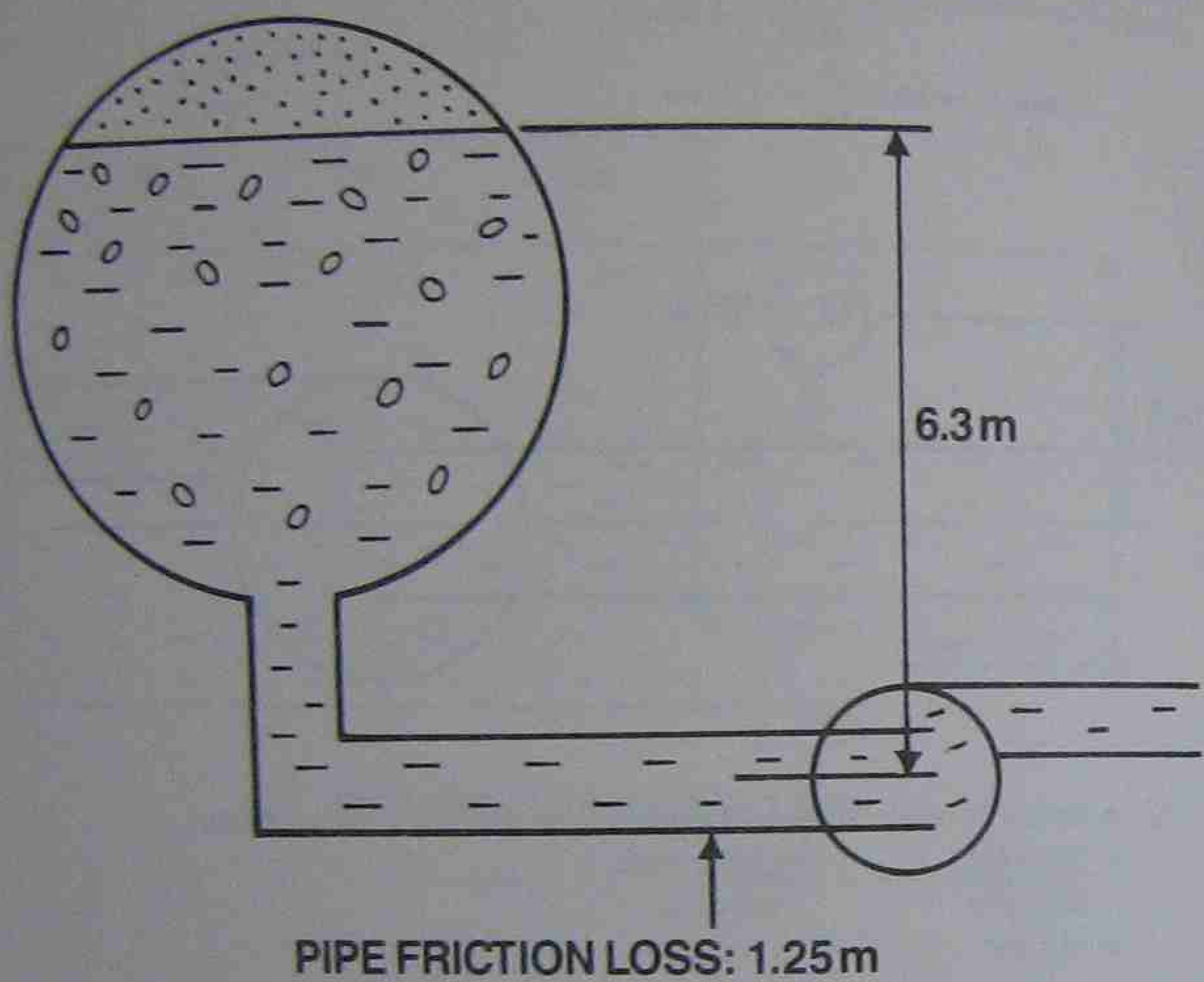
$$= 14.32\text{m}$$

$$H_{VAP} = \frac{7.375 \times 1000}{9.8 \times 992} = 0.76\text{m}$$

THUS NPSHA $= 14.32 - 0.76 = 13.56$m

EXAMPLE 5—SATURATED (BOILING) LIQUID

Water at 140°C,
Density: 925.9 kg/m^3,
Pressure in tank = Vapour Pressure = 361.4 kPa(abs)
g = 9.80 m/s^2



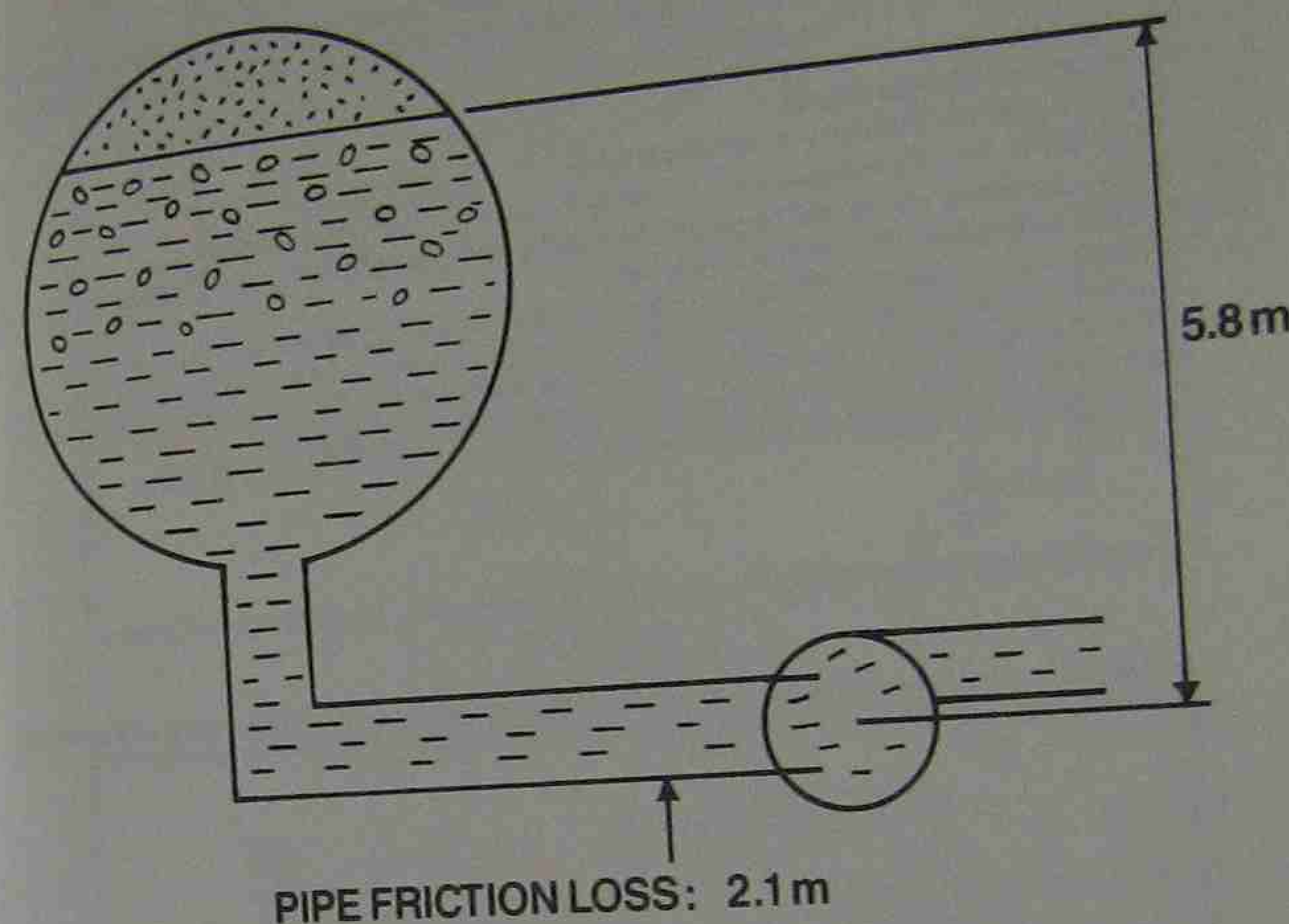
$H_{SI} = H_{VAP}$ + static head – friction loss
Since H_{VAP} will cancel out, the method of calculation can be simplified to:

NPSHA = Static Level – friction loss (for a boiling liquid)

THUS NPSHA = 6.3 – 1.25 = 5.05m

EXAMPLE 6—SATURATED (BOILING) LIQUID

Liquid Propane, Density: 480 kg/m^3



As the liquid is at saturation temperature, the pressure in the vessel will again equal the liquid vapour pressure.

NPSHA = Static Level – Friction Loss

Thus NPSHA = 5.8 – 2.1 = 3.70m

4.3 SYSTEM RESISTANCE CURVE

The quantity Q of liquid delivered by a centrifugal pump, i.e. the operating point, adjusts itself automatically to the system head to be overcome.

The total dynamic head (H_t) on the pump consists of a static component (H_{st}) independent of the flowrate, and friction head loss (H_f) which is proportional to the square of the flowrate.
i.e. $H_t = H_{st} + H_f$

As detailed in Section 4.1, the total friction loss in a piping system is determined by summing all the individual head losses arising from factors which offer resistance to fluid flow. These factors can comprise of:-

1) Inlet and exit losses
2) Suction and discharge piping
3) Sudden or gradual changes in cross-sections
4) Valves, bends and fittings

The system resistance curve can be readily plotted by using the above equation to determine total system head throughout the desired flow range.

e.g., Suppose in a pumping system it is required to deliver 100 L/s at a flow velocity of 5 m/s. The diameter of the pipework throughout the system is uniform and the static head on the pump is 5 m. The head loss in the straight pipework had been evaluated at 1 m and the total resistance coefficient (K) due to all the valves, bends, fittings etc. sums to 4.

The equation for head loss due to valves etc., is

$$H_f = K\,\frac{v^2}{2g}.$$

$$H_f = 4 \times \frac{5^2}{2 \times 9.8} = 5.1\text{ m}$$

$\therefore$ Total $H_f = 5.1 + 1 = 6.1$ m at 100 L/s

As stated previously, head loss due to friction is approximately proportional to flow rate squared.

i.e. $H_f \propto Q^2$

$\therefore$ $H_f = kQ^2$

Hence $k = \frac{H_f}{Q^2} = \frac{6.1}{(100)^2} = 0.00061\ \text{m.s}^2/\text{L}^2$

Now total head $H_t = H_{st} + H_f$

$\therefore$ $H_t = H_{st} + 0.00061Q^2$

which is the equation for the system resistance curve

$\therefore$ for 100 L/s

$H_t = 5 + 0.00061\ (100)^2 = 11.1$ m

for 50 L/s

$H_t = 5 + 0.00061\ (50)^2 = 6.53$ m

Consequently it can be seen that the total system head need only be determined for a single flow rate to allow a reasonable estimate of the system resistance curve.

Effect of Change to System Resistance Curve

If the total loss of head H_f of a pipeline is changed to H_f' by partly closing a bottle valve, or by altered piping condition etc., then the changed head condition will be described by a new system curve R_2 intersecting the H-Q curve at a new operating point A_2 (Fig. 4.3.1).

A shift in the position of the operating point on the pump characteristic curve also occurs when the total static head fluctuates between a maximum and minimum value (Fig. 4.3.2).

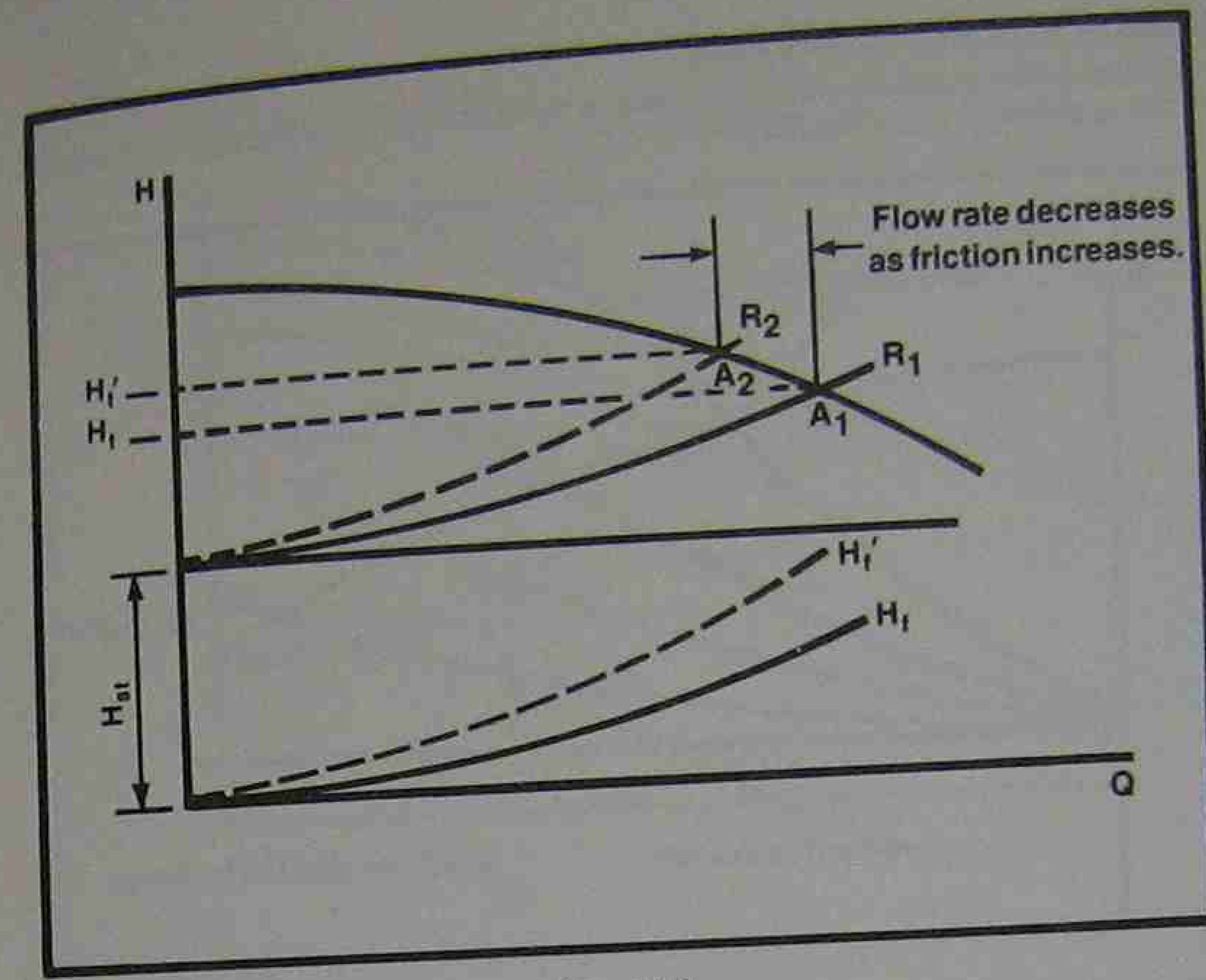


Fig. 4.3.1

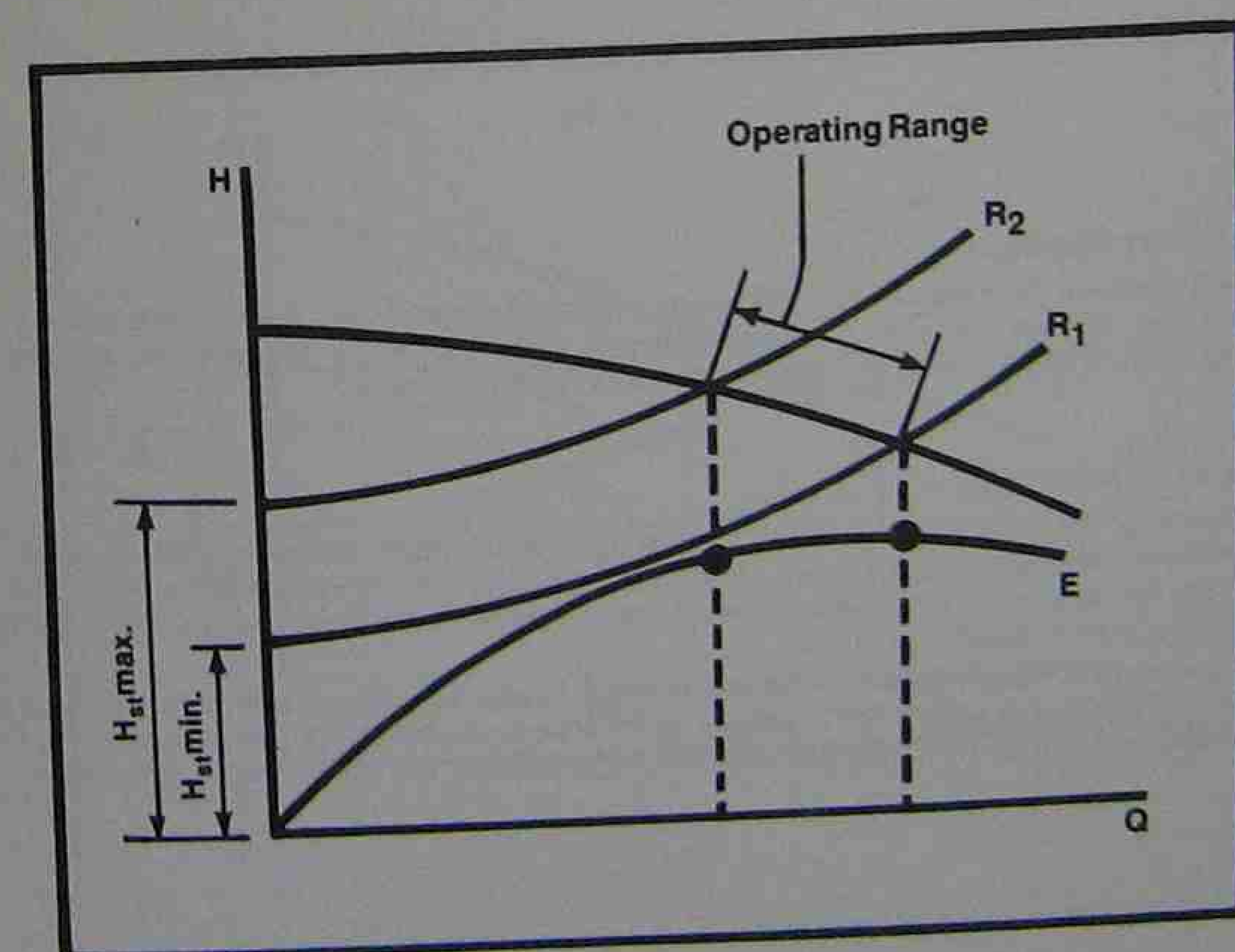


Fig. 4.3.2

Regulation of Flow Rate by Throttling

Each position of the throttling valve produces a different system curve, each of which in turn determines a different operating point (Fig. 4.3.3).

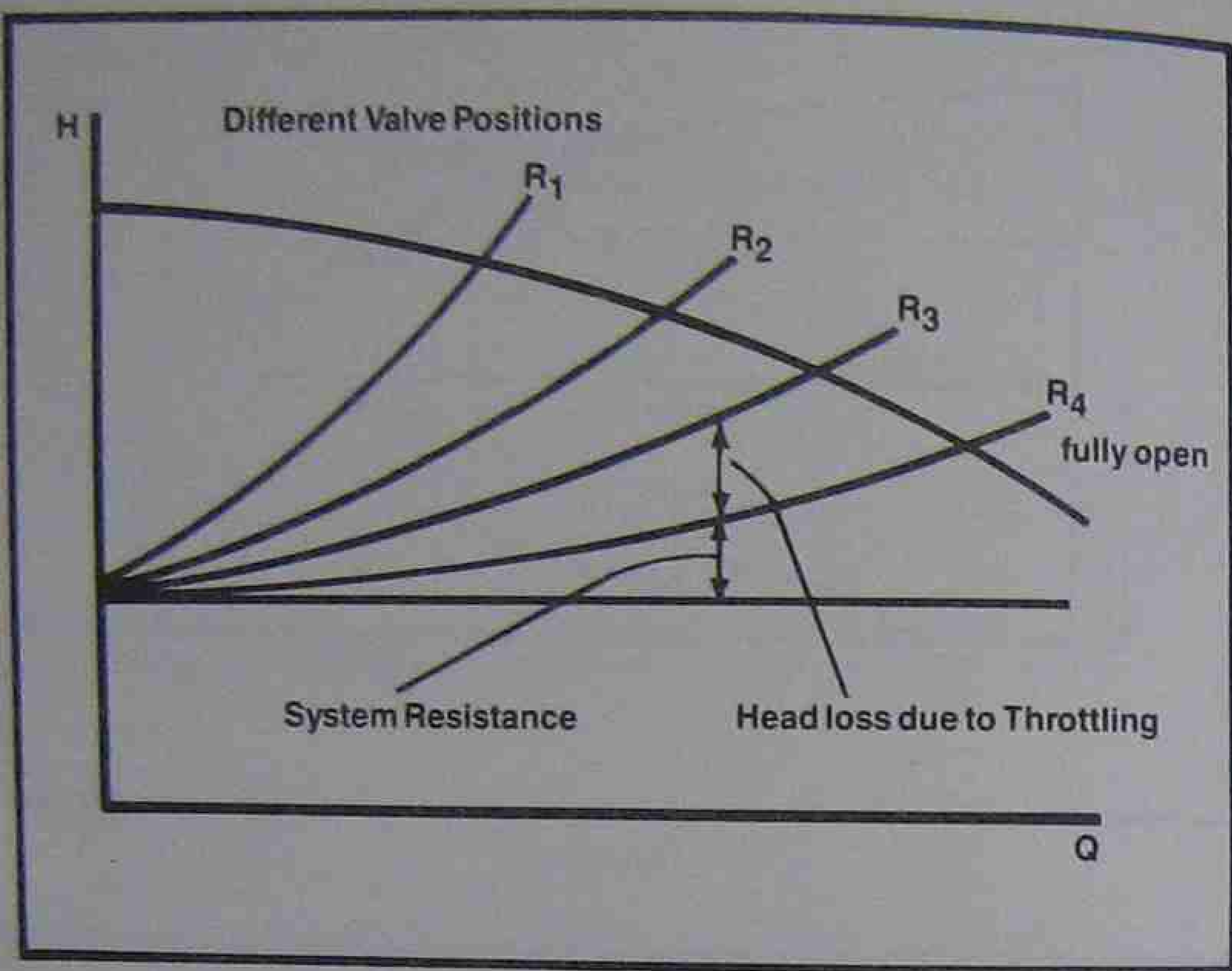


Fig. 4.3.3

Regulation by Speed

Figure 4.3.4 shows H-Q curves for maximum, intermediate and minimum speeds. Each curve intersects the system curve at a single operating point A_1, A_2, A_3 respectively, causing the operating point to move along the system curve between A_1 and A_3.

Steepness of H-Q Curve

This is given by the ratio of shut-off head to the head at optimum efficiency. Flat characteristic curves have a slope of around 10-15% whilst steep curves may have a slope of 50% or more. Flat curves cause comparatively larger variations in flowrate than steep curves when operating against a fluctuating head (Fig. 4.3.5).

Pumps operating consistently with a throttled discharge and against a reasonably constant static head, should preferably have a flat characteristic curve so that under part-load conditions, there is less head loss due to throttling and less throttling work by the control valve. (Fig. 4.3.6)

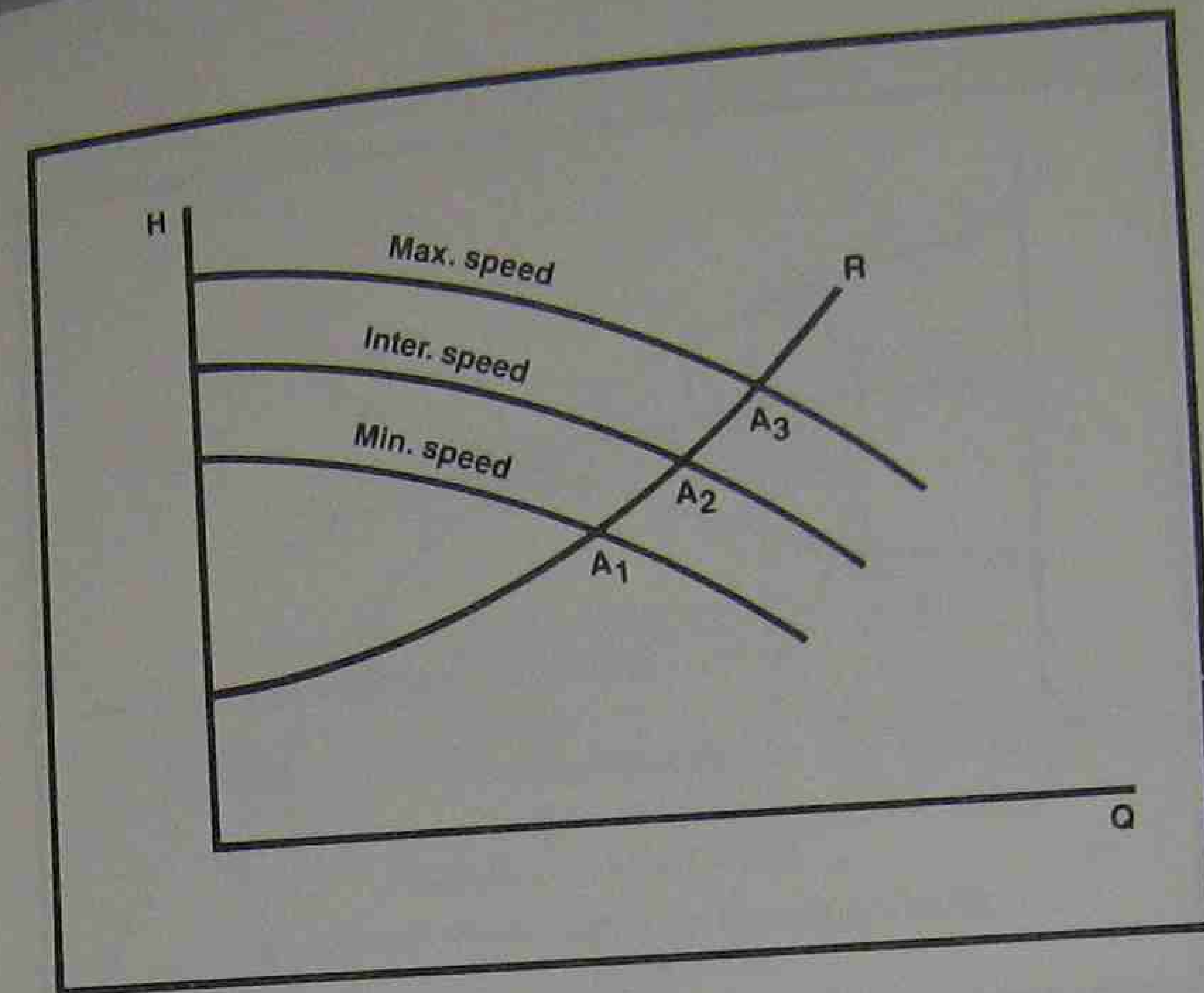


Fig. 4.3.4

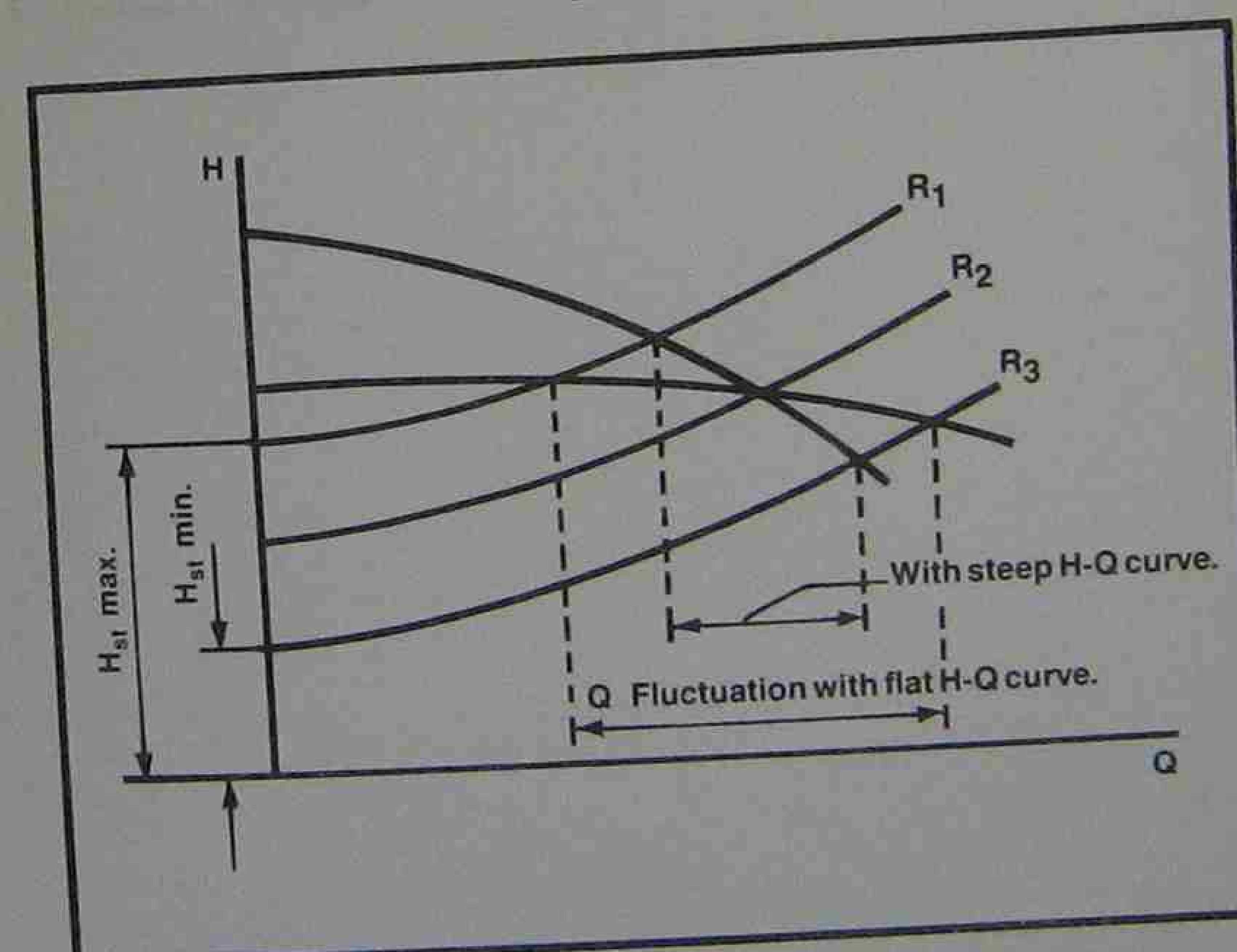


Fig. 4.3.5

Head loss due to throttling is greater with pump A due to steeper curve.

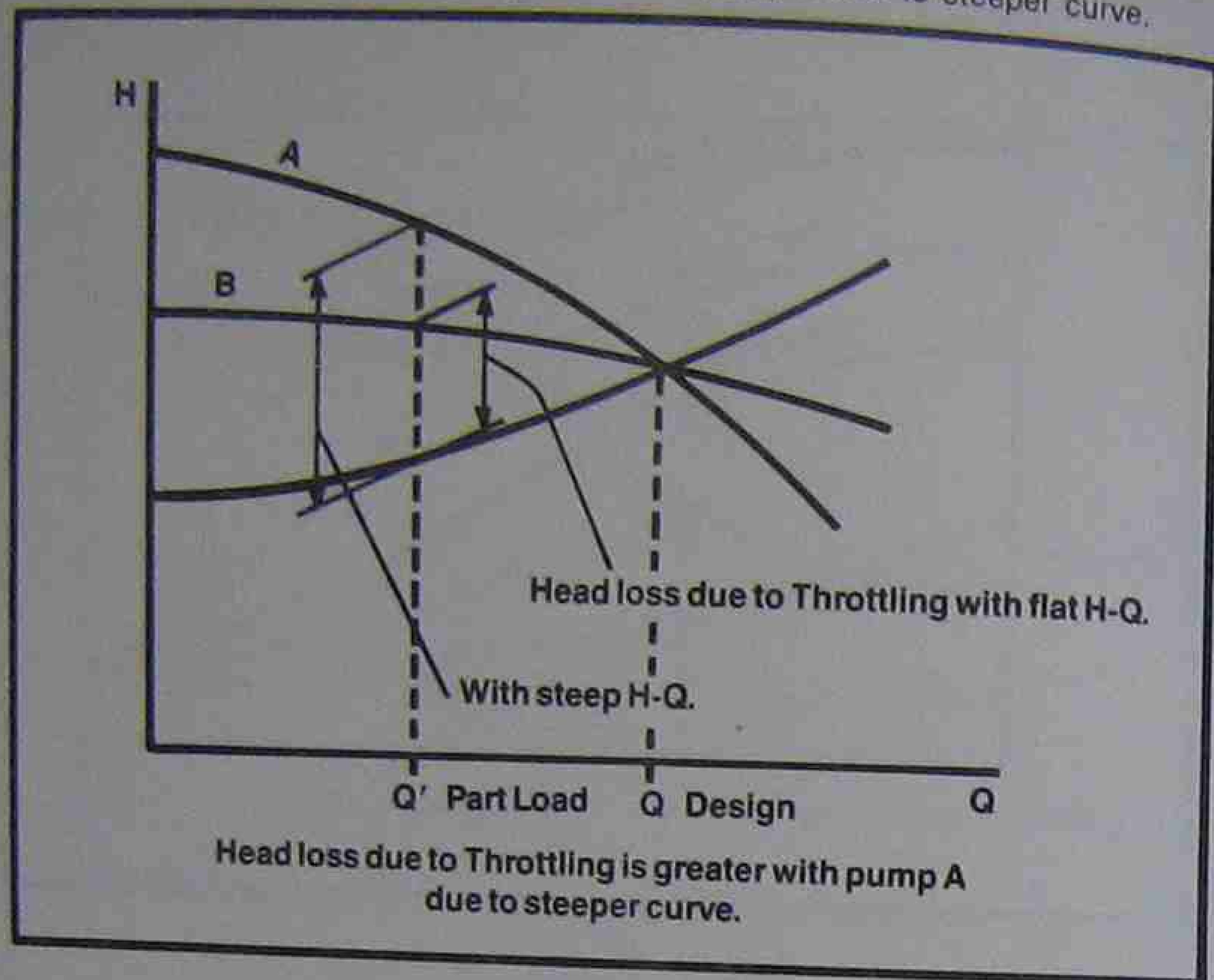


Fig. 4.3.6

Section 5

SELECTION OF CENTRIFUGAL PUMPS

Section 5—SELECTION OF CENTRIFUGAL PUMPS

A centrifugal pump should be matched to the pumping system both hydraulically and mechanically—a mismatched pump will be a continual source of problems.

5.1 HYDRAULIC SELECTION

1) **Define the Pump Duty Point**

Duty Point = rated flow at differential head (total dynamic head)

First Step - system design
quantity—flowrate
system resistance
system curve—duty point

Select the pump to match the system.

Oversizing or undersizing should be avoided as this may result in the pump running outside its allowable operating range.

The flow achieved in a pumping system is the result of the head created by the pump. This is the head necessary to overcome the static head, plus the friction head in the system, and is termed the system resistance head. System resistance can be plotted for different flowrates to produce a system resistance curve as shown in Fig. 51a.

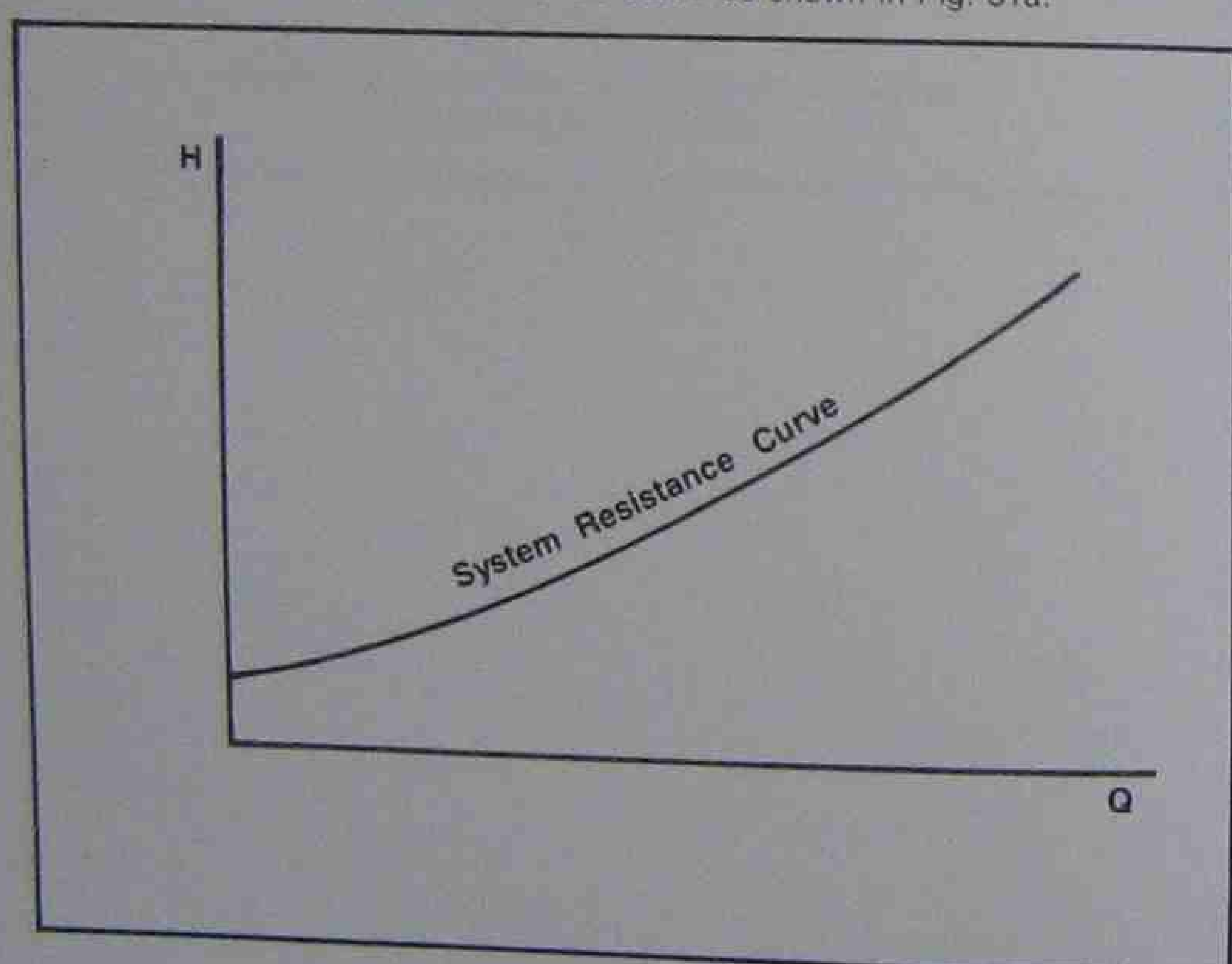


Fig. 5.1a

The operating point of a pump in a system is the flow at which the head developed by the pump is equal to the system resistance head. This is shown graphically in Fig. 5.1b as the intersection between the pump curve and the system resistance curve.

Section 4.0 of this handbook describes how to calculate system head at the duty flowrate. Correct pump selection requires this calculation to be reasonably accurate.

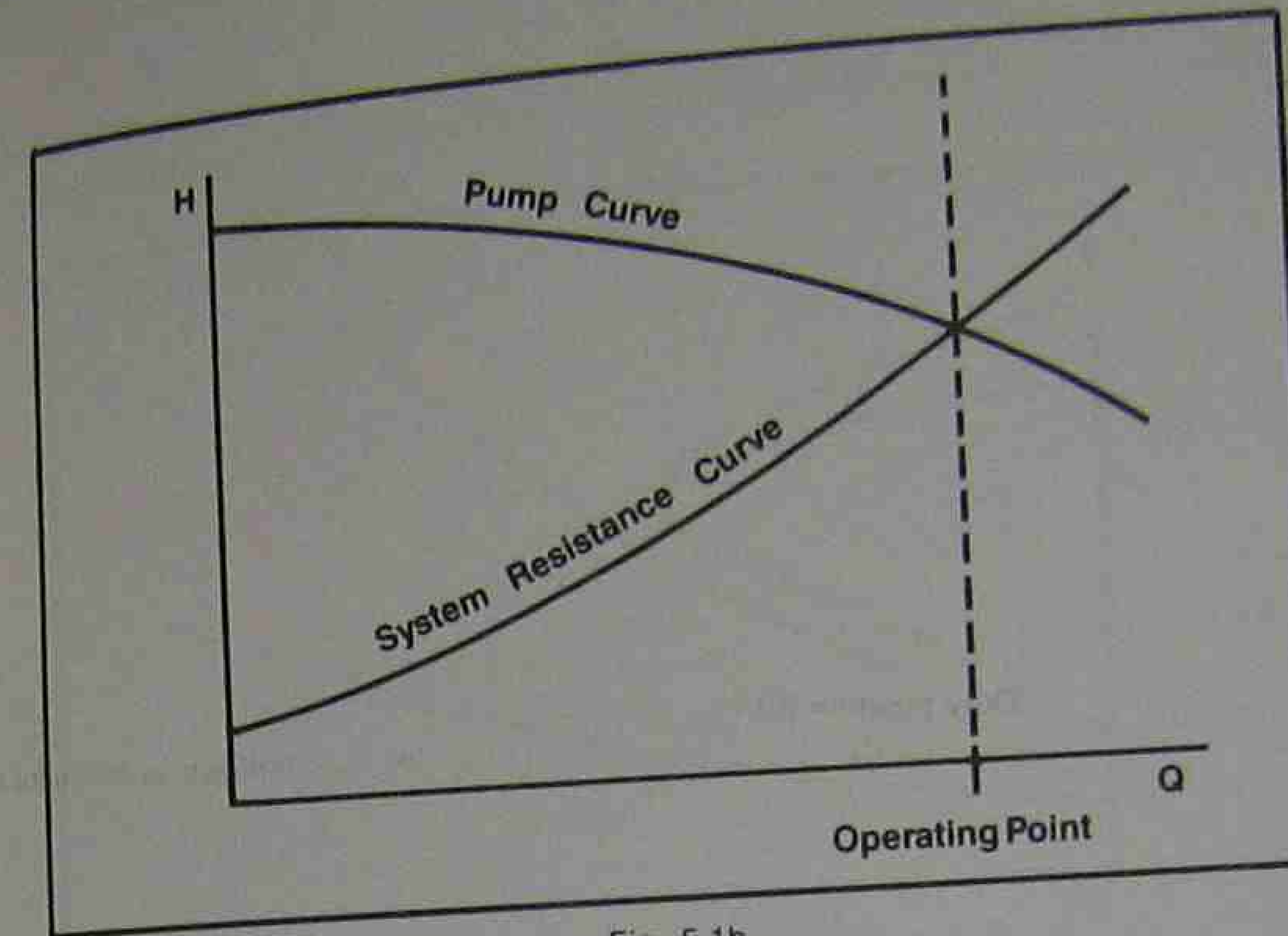


Fig. 5.1b

Safety factors and margins should be avoided since these will result in a pump oversized for the duty. Some of the problems associated with specifying excess head can be predicted from an examination of Fig. 5.2.

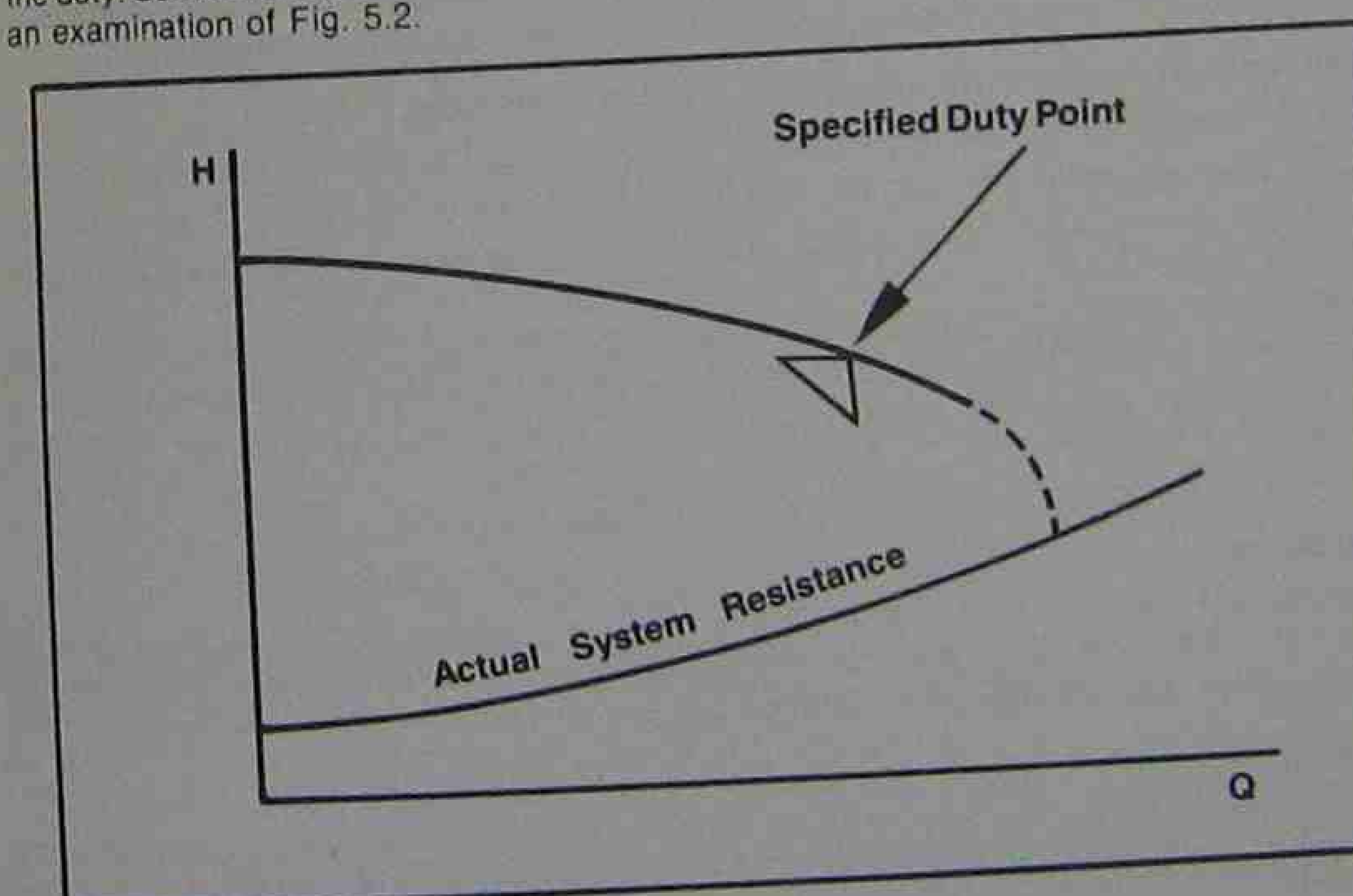


Fig. 5.2

This pump will operate beyond the limit of the published performance curve and would probably cavitate, vibrate and overload its driver.

The key to successful pump selection is to accurately specify the required duty point.

Where doubt exists about the accuracy of system head predictions it is good practice to consider maximum and minimum system characteristics. These can be plotted against pump performance curves to ensure that the selected pump will cover all expected system heads, as shown in Fig. 5.3.

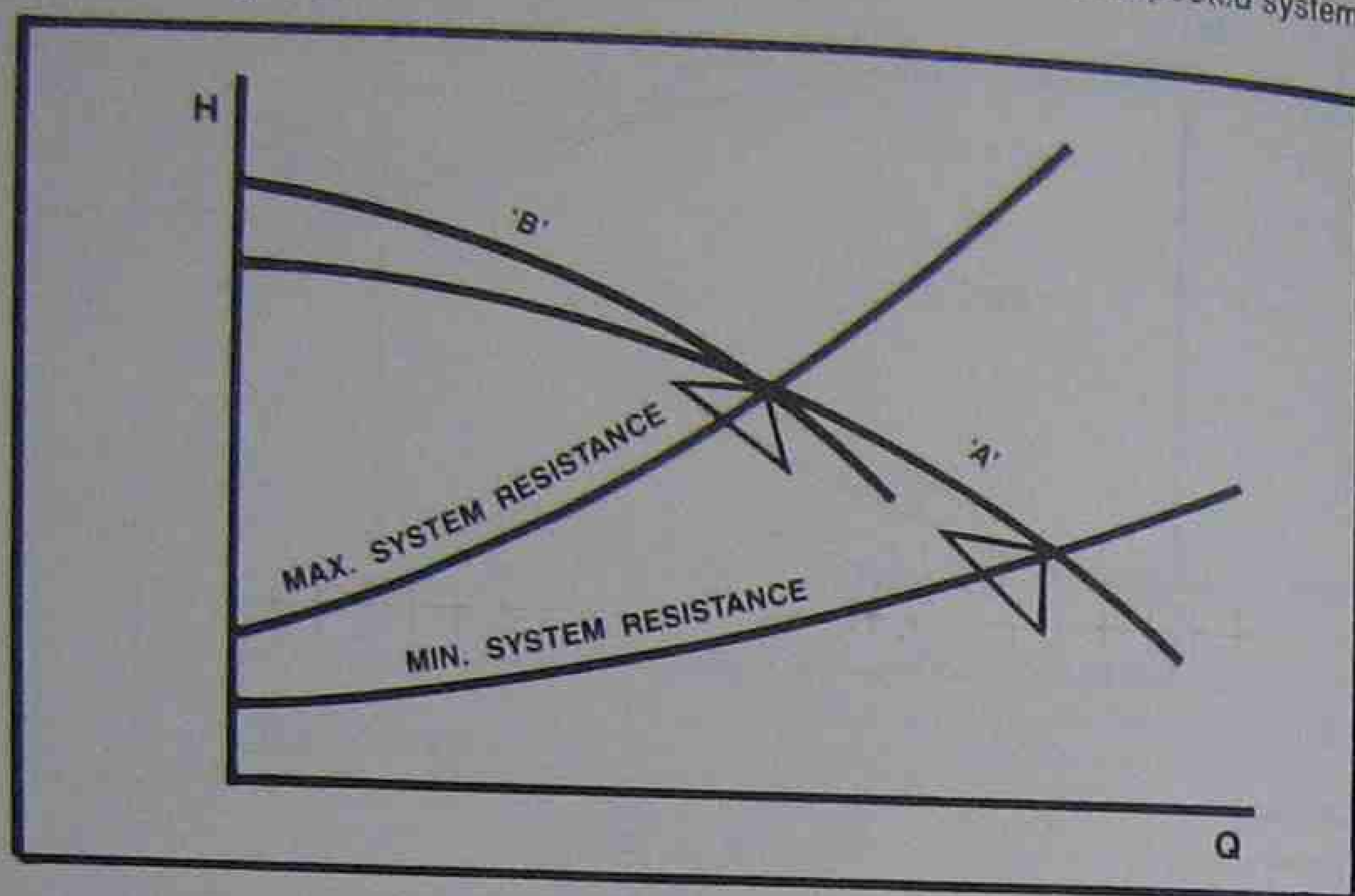


Fig. 5.3

Pump 'B' is acceptable for the maximum head condition only and is not recommended.
Pump 'A' crosses both system resistance curves and is the preferred selection.

2) **Calculate Nett Positive Suction Head Available (NPSHA)**

NPSHA = total absolute head at the pump suction minus the vapour head of pumped liquid.

This is a relatively simple calculation that is described in detail in Section 1 of this handbook.

One of the most significant causes of unsatisfactory pump performance and maintenance problems is cavitation.

To prevent cavitation in a centrifugal pump, the system NPSHA must be greater than the pump Nett Positive Suction Head Required (NPSHR) preferably with a safety margin.

Where a number of possible operating conditions exist, as in Fig. 5.3 above, NPSHA must be calculated for each case.

Sample NPSHA calculations are included in Section 4.2 of this handbook.

Suction Lift

When handling cold water the concept of suction lift may be used to simplify calculations. Pump manufacturers often publish curves showing suction lift performance for clear cold water. As with NPSH a safety margin is recommended.

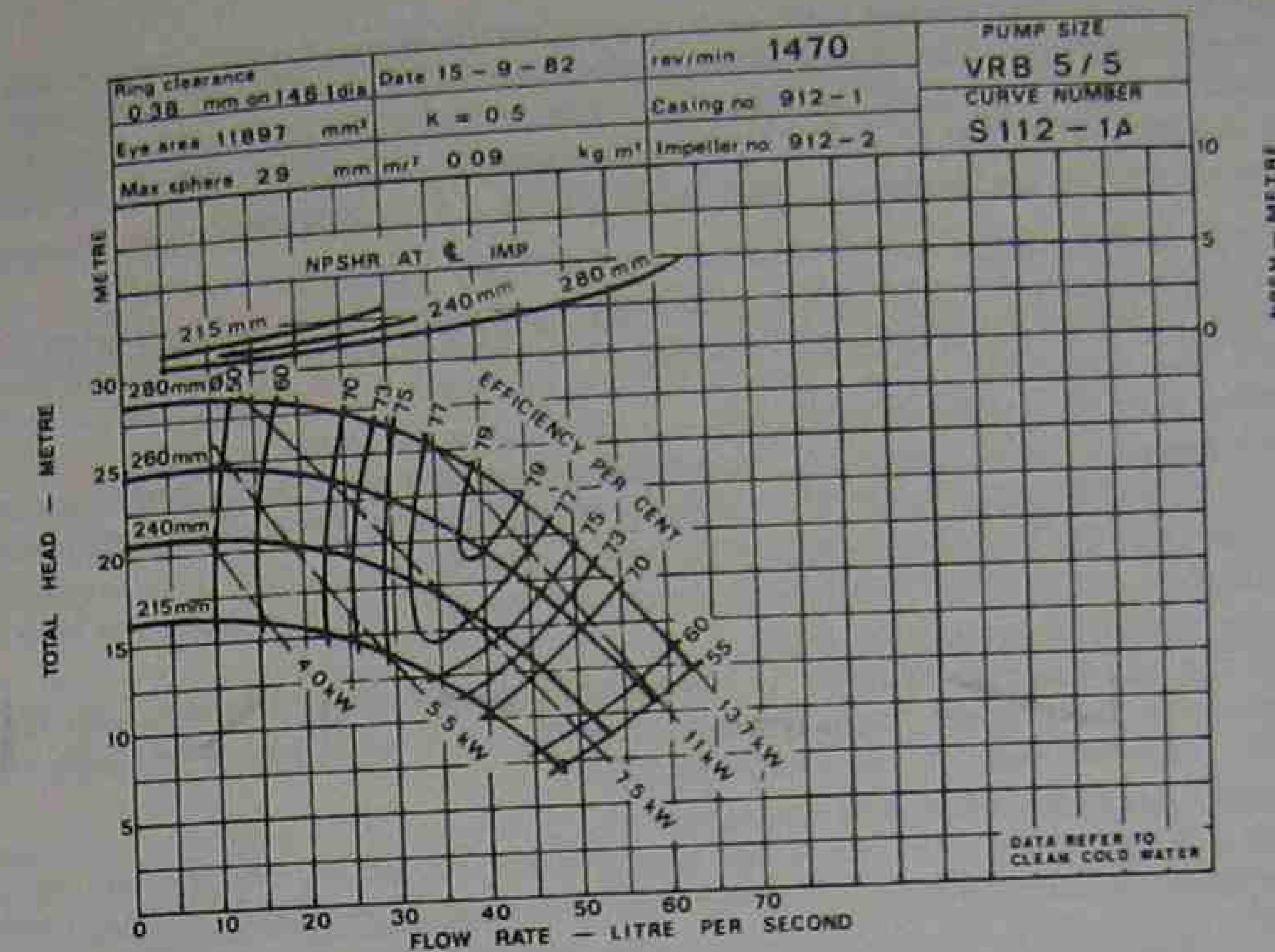
3) **Select the Pump**

Having established the flowrate, the total dynamic head to be generated by the pump and the suction lift or nett positive suction head available, the next step is to select a pump with a performance curve that satisfies all three of these requirements.

Pump manufacturers may present the performance curves of their pumps in several ways.

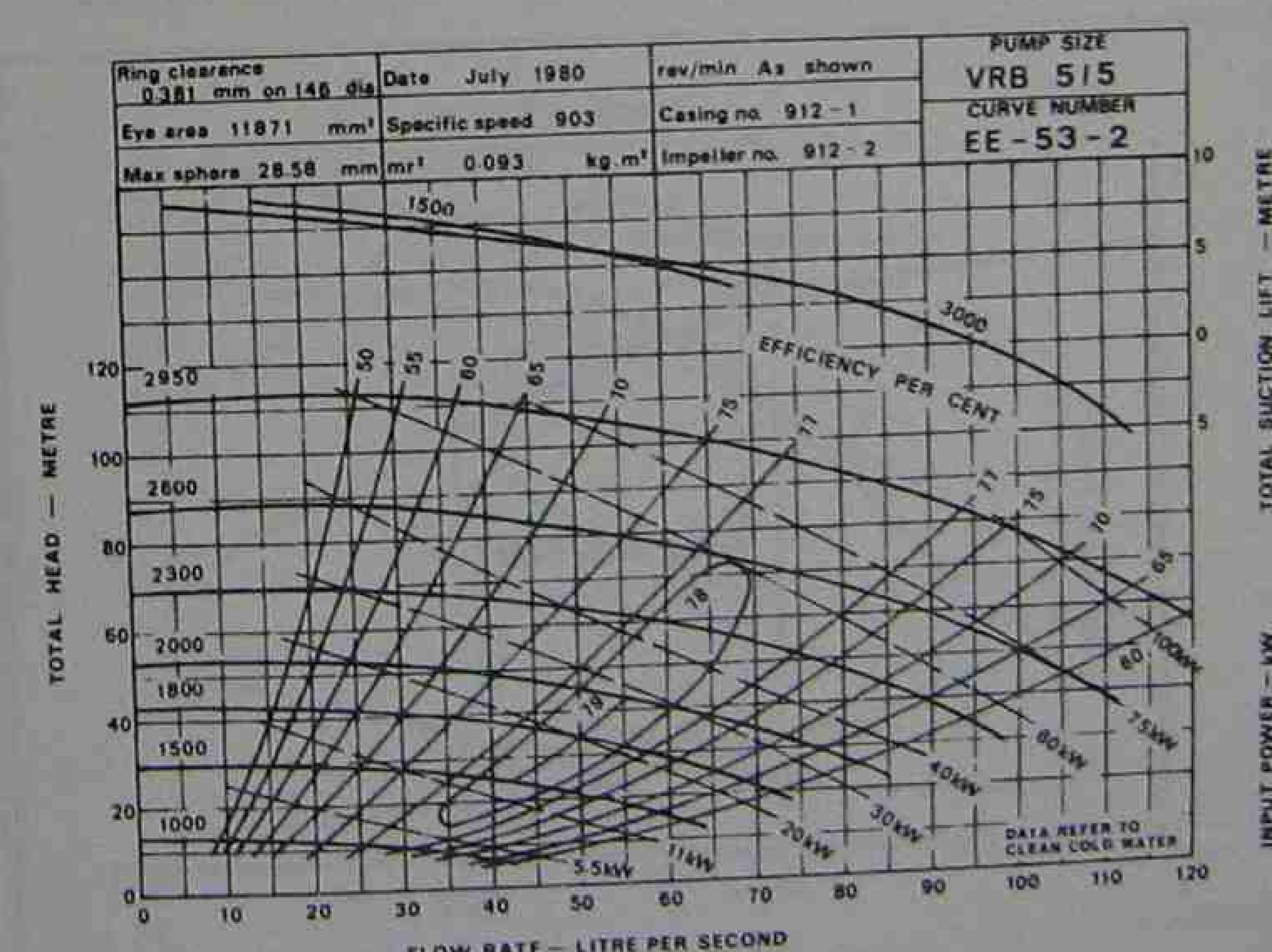
Performance can be presented at constant speed with a family of head—quantity curves corresponding to different impeller diameters or alternatively at varying speeds with a constant impeller diameter. Fig. 5.4a & Fig. 5.4b

In either case, they will also indicate pump efficiency, power absorbed by the pump, and suction lift or NPSH required for the flow range of the unit.



TYPICAL PERFORMANCE CURVES OF A CENTRIFUGAL PUMP AT 1470r.p.m WITH REDUCED IMPELLER DIAMETER.

Fig. 5.4a



TYPICAL PERFORMANCE CURVES OF A CENTRIFUGAL PUMP AT VARYING SPEEDS.

Fig. 5.4b

Therefore, if the required flow, total head, and suction lift are plotted directly on these curves the speed (r/min) and input power (kw) required can be determined. The input power will usually be given for clear cold water, i.e. liquid with a density of 1000 kg/m3. If the liquid intended to be pumped is heavier or lighter than water, more or less power will be required.

Having established the speed and power requirements, a suitable prime mover can be selected.

3(a) Ideal Pump Selection

The ideal selection is a pump running at best efficiency point (B.E.P.) and with NPSHA greater than pump NPSHR by an adequate margin.

This ideal situation is shown in Fig. 5.5.

At B.E.P. flowrate the impeller and volute are operating at pump design conditions and flow through the unit is smooth. Shock, vibration, energy dissipation, and bearing loading are minimised. This pump should have a long trouble-free life.

Unfortunately, compromise is often necessary for the following reasons:

— Finite number of available pumps.
It is not economically feasible to manufacture a specially designed pump for every application. A pump will normally be selected from the range of commercially available units. A perfect match is not always possible.

— Specific speed limitation (see Section 3)
If the specified duty point is one of high head and low flowrate, impeller geometry limitations may affect the pump selection. This may result in operating near pump minimum continuous flow.

— Nett positive suction head.
If the NPSHA is limited, the available pump selections and running speeds may be restricted.

— Variable duty requirements.
A number of disparate operating conditions may have to be met, precluding an ideal selection.

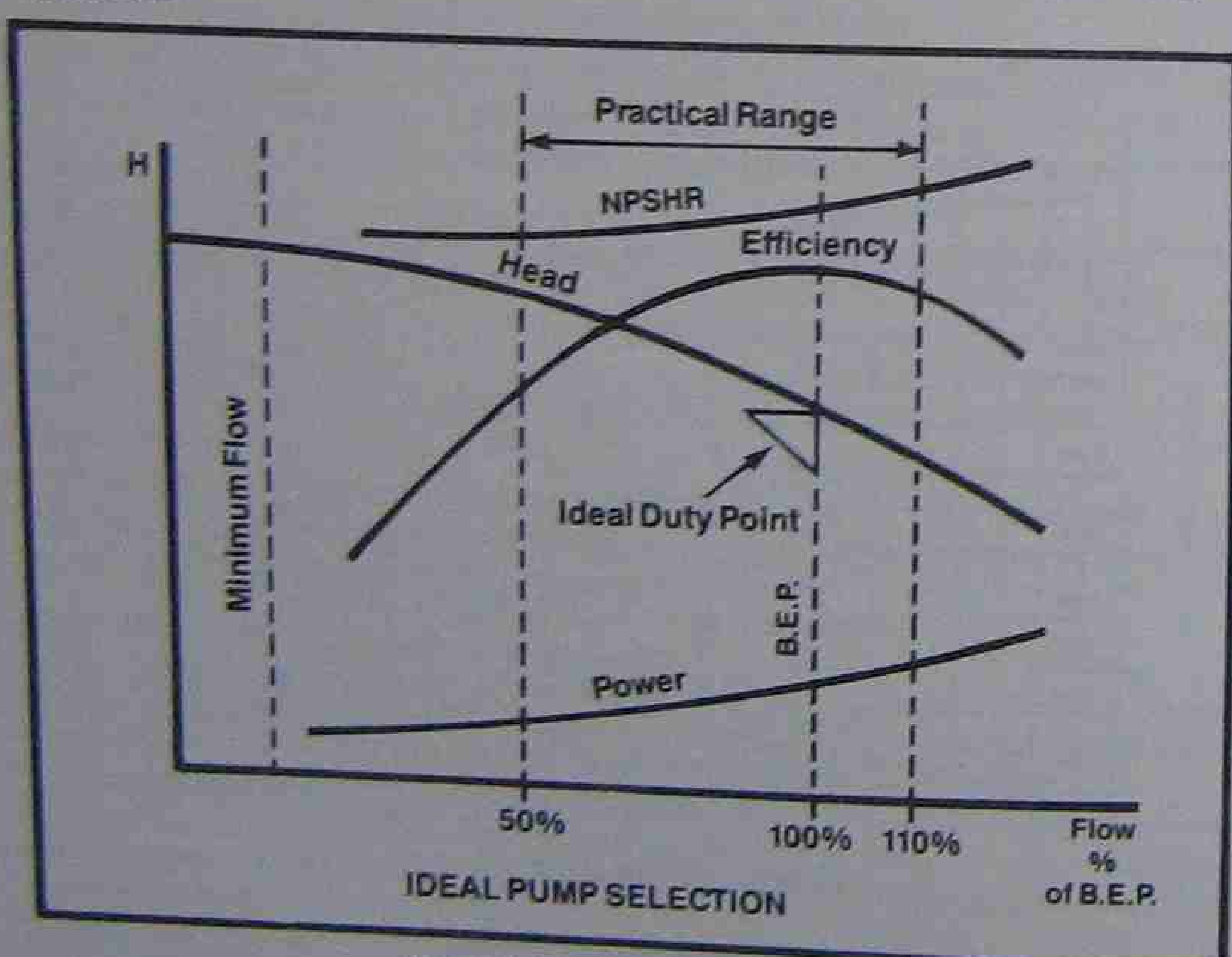


Fig. 5.5 Ideal Pump Selection.

— Mechanical simplicity/cost/reliability.
The "ideal" selection may be expensive or complicated.

Pump selection is often a trade off between efficiency/cost/reliability.

3(b) Practical Pump Selection

A pump selection is normally considered acceptable if the duty point falls within the rate of 50% to 110% of best efficiency flow rate. Also a selection to the left of B.E.P. is normally preferred to one slightly beyond the B.E.P. This allows a greater margin for error in the event that the system designer has overestimated the "actual" system resistance curve. (see Fig. 5.3)

In many cases centrifugal pumps can be successfully operated outside the above recommendations in circumstances where alternative selections are not available or economic. The pump manufacturers advice should be sought in these instances.

Nevertheless, it is always preferable to operate a centrifugal pump as close as possible to its best efficiency point.

4) Poor Hydraulic Selections—Their Effects

(a) Left of B.E.P. (Fig. 5.6)

Operating problems:

— low efficiency
— noise vibration reduced life
— increased radial loads on bearings due to unbalanced volute pressures
— temperature rise due to dissipated energy created by low efficiency

Remedies:

(i) Reselection
— smaller same speed pump if feasible

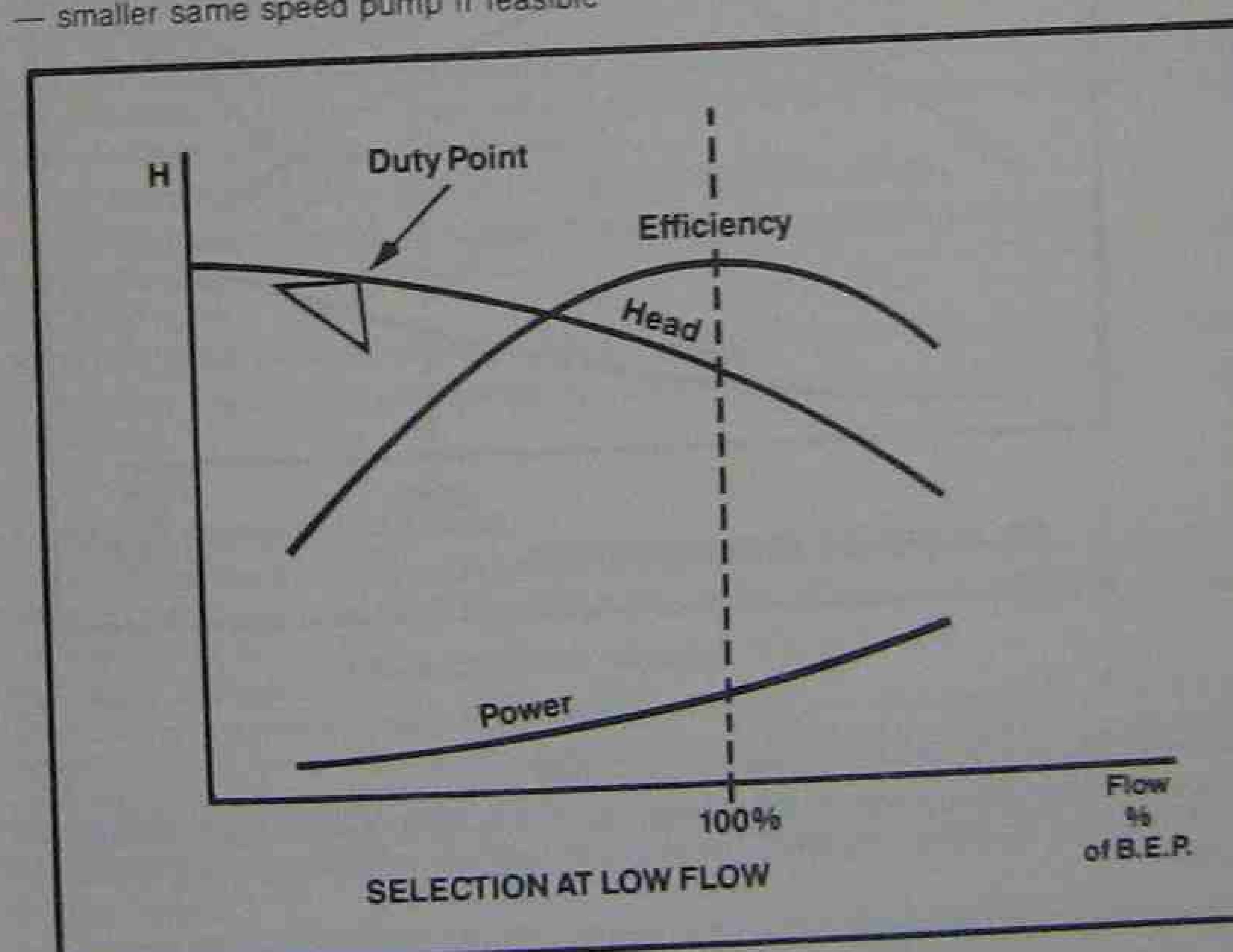


Fig. 5.6 Selection at Low Flow

— smaller high speed pump (efficiency ?)
— smaller multistage pump (expensive)
— positive displacement pump

(ii) Limit adverse effects

— install bypass (increased power)
— use a double volute or diffuser casing to reduce radial loads on bearings
— heavy duty shaft and bearings

(b) Beyong B.E.P. (Fig. 5.7)

Operating problems:

— low efficiency
— increased power
— noise and vibration—reduced life
— poor suction performance (high NPSHR)
— increased radial loads on bearings

Remedies:

— larger pump
— throttle back to lower flowrate

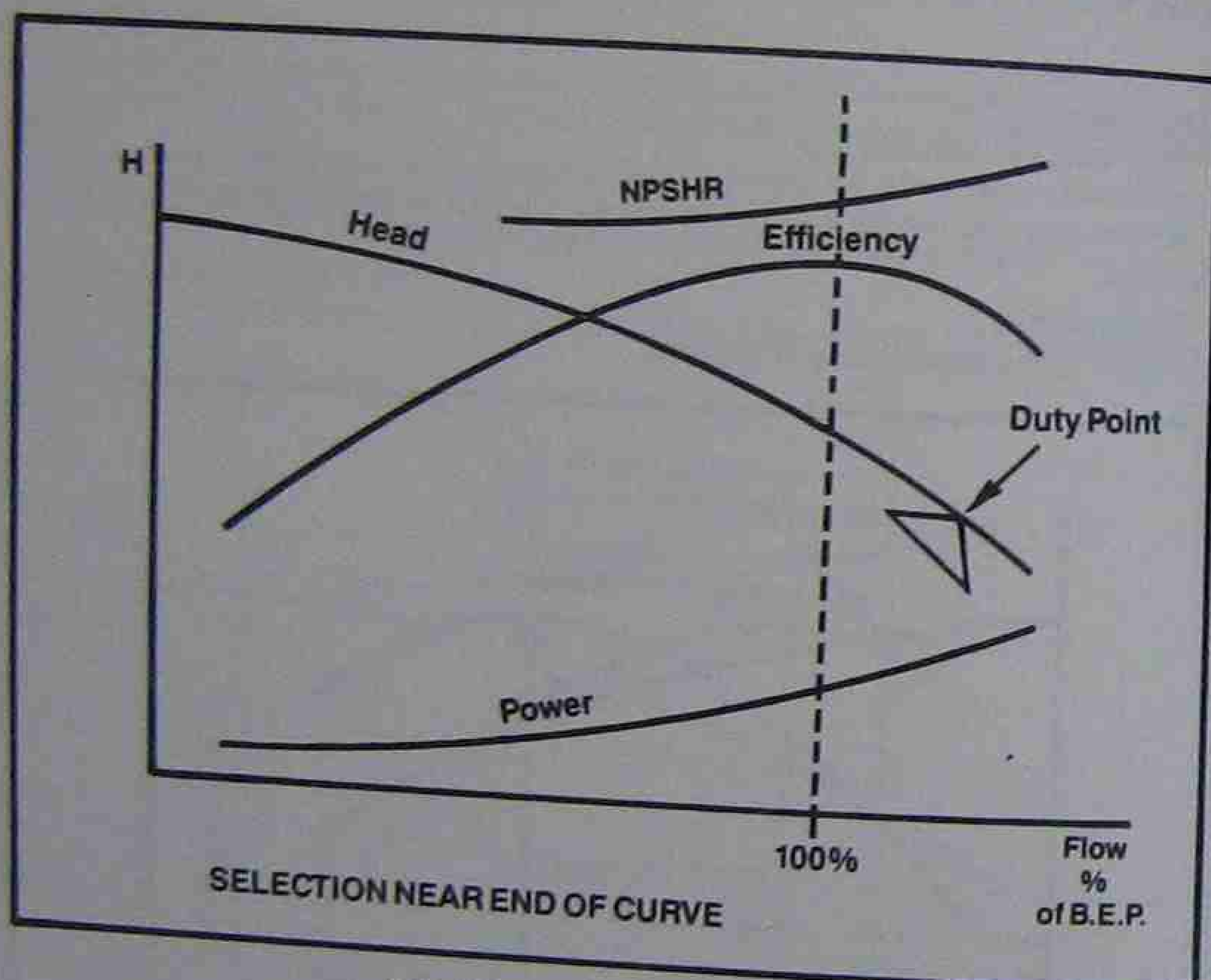


Fig. 5.7 Selection Near End of Curve.

5.2 MECHANICAL ASPECTS OF PUMP SELECTION

Mechanical and constructional features are covered in detail in Sections 3.1 and 3.2 of this handbook, however, it is useful to briefly examine the effect of system conditions on pump mechanical configuration.

5.2.1 Impeller/Casing Geometry

(see also Section 3.2—specific speed and pump type)

Pump geometry to suit a given set of pumping conditions will depend on the chosen speed and number of stages.

e.g. consider the following alternative pump selections for a duty point of 50 L/s at a total dynamic head of 150m.

(a) Single stage, 1450 r.p.m.

Ns = 239, Expected efficiency: 45%
Approximate pump outlet size: 100mm
Approximate impeller diameter: 720 mm

—this is not a practical pump selection due to the large impeller diameter required realative pump size.

(b) Single stage, 2950 r.p.m.

Ns = 486, Expected efficiency: 68%
Approximate pump outlet size: 100mm
Approximate impeller diameter: 360mm

— a reasonable selection with moderate efficiency.

(c) Two (2) stage, 2950 r.p.m.

Ns = 818, Expected efficiency: 74%
Approximate pump outlet size: 100mm
Approximate impeller diameter: 260mm

— better efficiency but more complex and more expensive.

From the trend of the above alternatives it can be seen that pump selection may often be a trade off between practical design, simplicity (low cost), and efficiency.

5.2.2 Maintenance & Reliability

The preceding example illustrates that greater mechanical complexity may result from a desire for improved hydraulic efficiency.

However, a well selected multistage pump may be more reliable than a poorly selected single stage unit. Nevertheless, an increase in the number of stages leads to greater complexity which in turn generally provides greater potential for failure.

Aspects of pump construction that provide greater reliability include:

— reduced complexity/number of stages (assuming good hydraulic selection)
— sensible (larger) running clearances
— short shaft span

It is worth noting that some aspects affecting ease of maintenance may lead to reduced reliability. e.g. easy access to seals and bearings may require a larger shaft span than desirable from reliability considerations.

5.2.3 Effect of Pumping Conditions

Apart from the purely hydraulic considerations of flow rate and total dynamic head, mechanical construction may be influenced by the effects of static pressure, liquid temperature, suction lift, and normal operating point.

(a) High Static Pressures

The obvious requirement associated with high pressure is an adequately designed pressure retaining casing. However, another aspect that must not be overlooked is pressure induced axial thrust on the pump shaft. (see Section 3.1)

The shaft of an overhung pump with a high suction pressure will experience a thrust towards the coupling as shown in Fig. 3.1.5a. This thrust may be extremely high and may lead to rapid bearing failure.

Any pump with a single shaft entry is subject to this effect including pedestal mounted, close coupled, horizontal and vertical pumps.

Measures which may be taken to alleviate this problem include:

—use of an hydraulically unbalanced impeller producing an opposing thrust

—reduced impeller back sealing ring diameter which produces an opposing thrust.

—oversize thrust bearing(s)

—use of double entry between bearings style of pump (Fig. 3.1.6)

(b) High Temperatures

Apart from metallurgical considerations, high temperatures require special consideration to be given to the following:

—centreline support of pump casing
This allows equal thermal expansion above and below the shaft axis, thus retaining shaft/coupling alignment. This is recommended at temperatures above 180°C.

—Increased running clearances to avoid internal contact due to differential expansion.

—Shaft sealing methods, such as high temperature mechanical seals.

—Auxilliary coolant requirements for bearings and seals.

—Extremely high temperatures and pressures may require the use of special gasketing and radially split casings only.

(c) Low Suction Pressure/High Suction Lift

Apart from ensuring the pump NPSHR is suitable for the conditions, operation with suction pressures below atmospheric may result in the ingress of air at the shaft seal(s).
Special precautions to prevent this undesirable occurrence include:

—routing a high pressure flush connection to the stuffing box, thus maintaining pressure above atmospheric at the seal/packing.

—use of a suitable mechanical seal.

(d) Regular Operation at Reduced Flow

When a single volute pump is operated at flowrates other than the best efficiency point, the pressure distribution around the impeller is no longer uniform. (see Fig. 3.1.11b).

This produces a radial load on the impeller and shaft and in extreme cases may result in bearing failure and shaft breakage.

Where regular reduced flow operation cannot be avoided, consideration should be given to the following:

—use of double volute or diffuser style casing

—use of stiffer/stronger shaft assembly

—employ a minimum flow bypass line around the pump

SUMMARY—CENTRIFUGAL PUMP SELECTION

—Accurate system design

—Realistic pump duty point

—Select pump to match the system

—Ensure adequate NPSH margin

—Consider effects of speed and number of stages

—Consider effects of pumping conditions on pump construction

Section 6

OPERATION OF CENTRIFUGAL PUMPS

Section 6—OPERATION OF CENTRIFUGAL PUMPS

A centrifugal pump is self-limiting in the respect that the actual operating point on the H-Q curve is determined strictly by the characteristics of the system in which it operates. The pump will stabilise at a point where pump performance exactly balances the system requirements, the operating point being determined by the intersection of the H-Q curve and system friction curve. To obtain flexibility of operation, a number of pumps can be operated in either series or parallel as described in Section 6.1 and 6.2.

6.1 SERIES OPERATION

Centrifugal pumps are connected in series if the discharge of one unit leads to the suction of the next, i.e. two similar pumps in series operate in the same way as a two stage pump. Each pump imparts energy to the liquid and the resultant total head generated at any quantity is therefore the sum of the individual heads at that quantity (Fig. 6.1).

The main considerations with series operation are:

1) Ensure that the casing of the second or higher stage pump is rated for the higher pressure, i.e. higher strength material, ribbing or extra bolting may be required.
2) The stuffing box of the second stage pump must be designed for the high suction pressure, i.e. a mechanical seal may be necessary.
3) Ensure that all pumps are filled with fluid during start-up and operation and the second pump is started after the first is running.

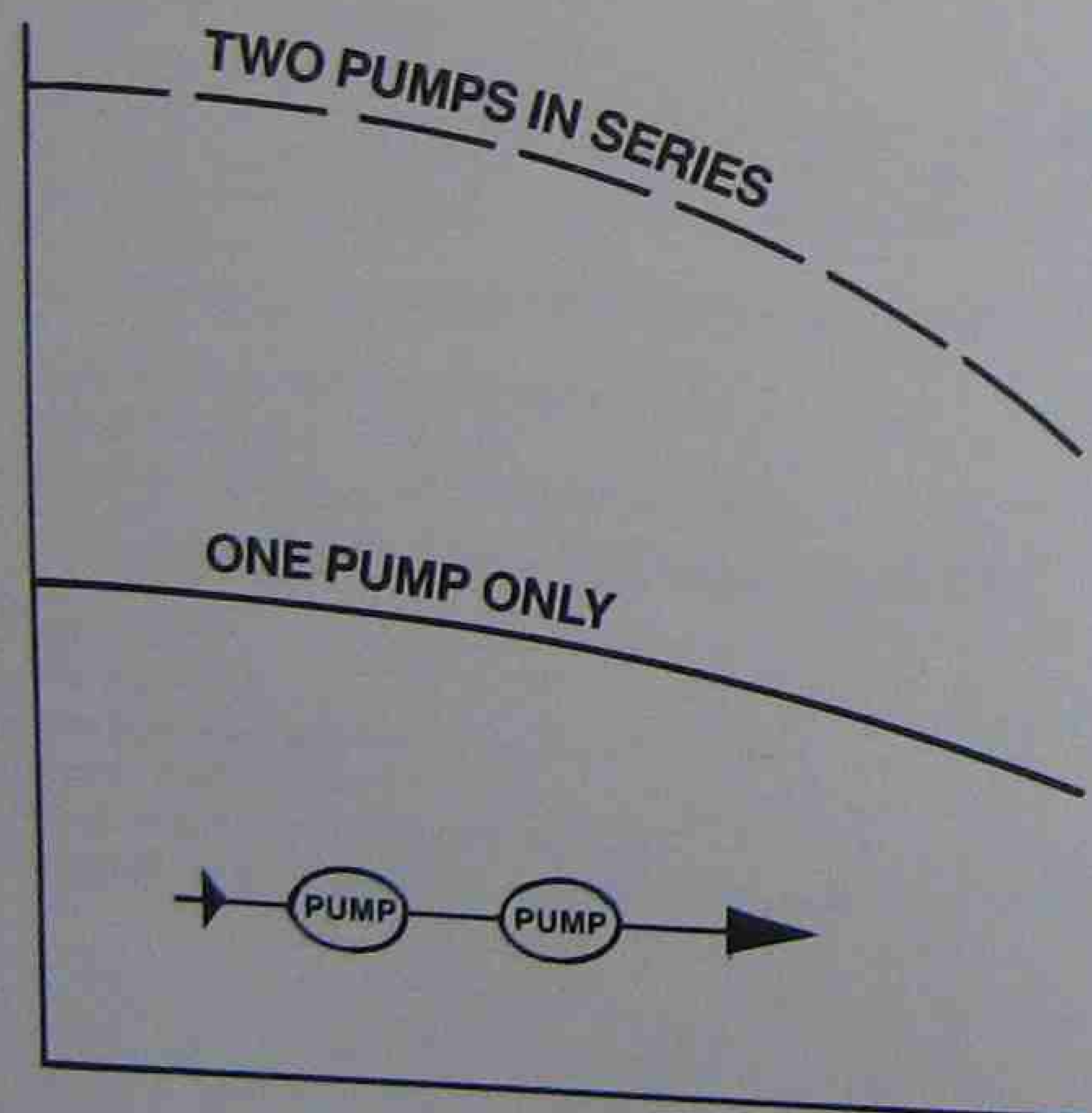
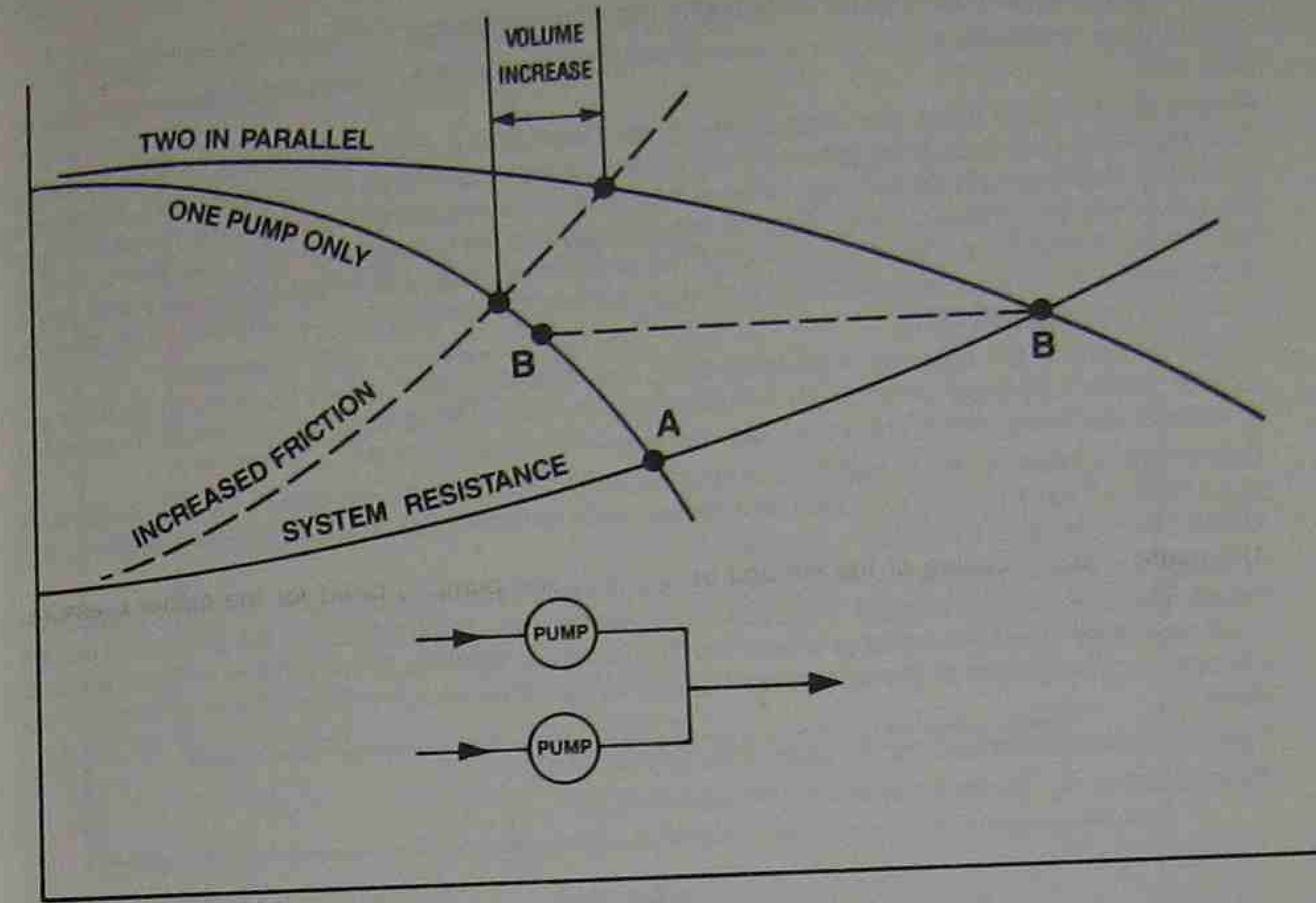


Fig. 6.1 Series Operation.



PARALLEL OPERATION

Fig. 6.2

6.2 PARALLEL OPERATION

When two or more pumps are connected to a common discharge line (and share the same suction conditions) they are installed for parallel operation. The total head is the same across each pump, but the quantity is additive.

Referring to Figure 6.2 one pump by itself will operate at point A. When two pumps are operating in parallel in the system, the operating point on the combined head—quantity curve is B. Therefore each individual pump must be operating at point B on its own curve. We can draw two conclusions from this:

(a) Two pumps operating in parallel will deliver less than twice the flow rate of an individual pump operating by itself in the same system (due to increased friction with two pumps). It will also be noted that the increase in quantity obtained by placing two pumps in parallel is determined by the shape of the system resistance curve. Thus, if there is considerable friction (as indicated by the dotted line), two pumps in parallel may deliver only slightly more than one pump operating by itself.

(b) One pump by itself will operate at a higher flow rate (A) than if it were working in parallel with another pump (B). In other words, it would be operating further out the curve, possibly with greater power requirements. Therefore when a pump is designed for a parallel duty, also ensure that the driver is adequately rated for solo operation.

6.3 STABLE AND UNSTABLE CHARACTERISTIC CURVES

An unstable characteristic curve is one of which the maximum head occurs elsewhere than a shut off.

Referring to Figure 6.3.1, the shut-off head developed by a centrifugal pump with an unstable characteristic curve must be greater than the static head associated with the system resistance curve (as in R1) otherwise the H-Q curve will intersect the resistance curve at two points (R2) and the pump will be unable to start against the completely filled discharge line. The discharge line will therefore require partial emptying until the static head is less than the head developed by the pump at shut-off. Only then will the pump start delivering liquid. However, there are two possible operating points A_1 and A_2.

If several pumps are working in parallel at A_2 some pumps may in fact be working at A_1. If the throttle valve is opened under such conditions, a pump working at A_2 will increase its capacity to A_2' and the pump operating at A_1 will decrease its capacity to A_1'.

Delivery may cease entirely and fluctuations in head and capacity will result. Pumps which have to be throttled back to very low rates of delivery, such as boiler feed pumps, should therefore have stable characteristic curves.

Although not preferred, it is possible to achieve satisfactory parallel operation of centrifugal pumps having unstable characteristics, (Fig. 6.3.2). However, the shut-off head of the combined pumps must be greater than the head at which the first pump is working by an amount at least equal to the opening resistance of the closed non-return valve; otherwise it will be unable to share in the delivery.

System curve R_1: Second pump cannot be switched in.

System curve R_2: Second pump can be switched in.

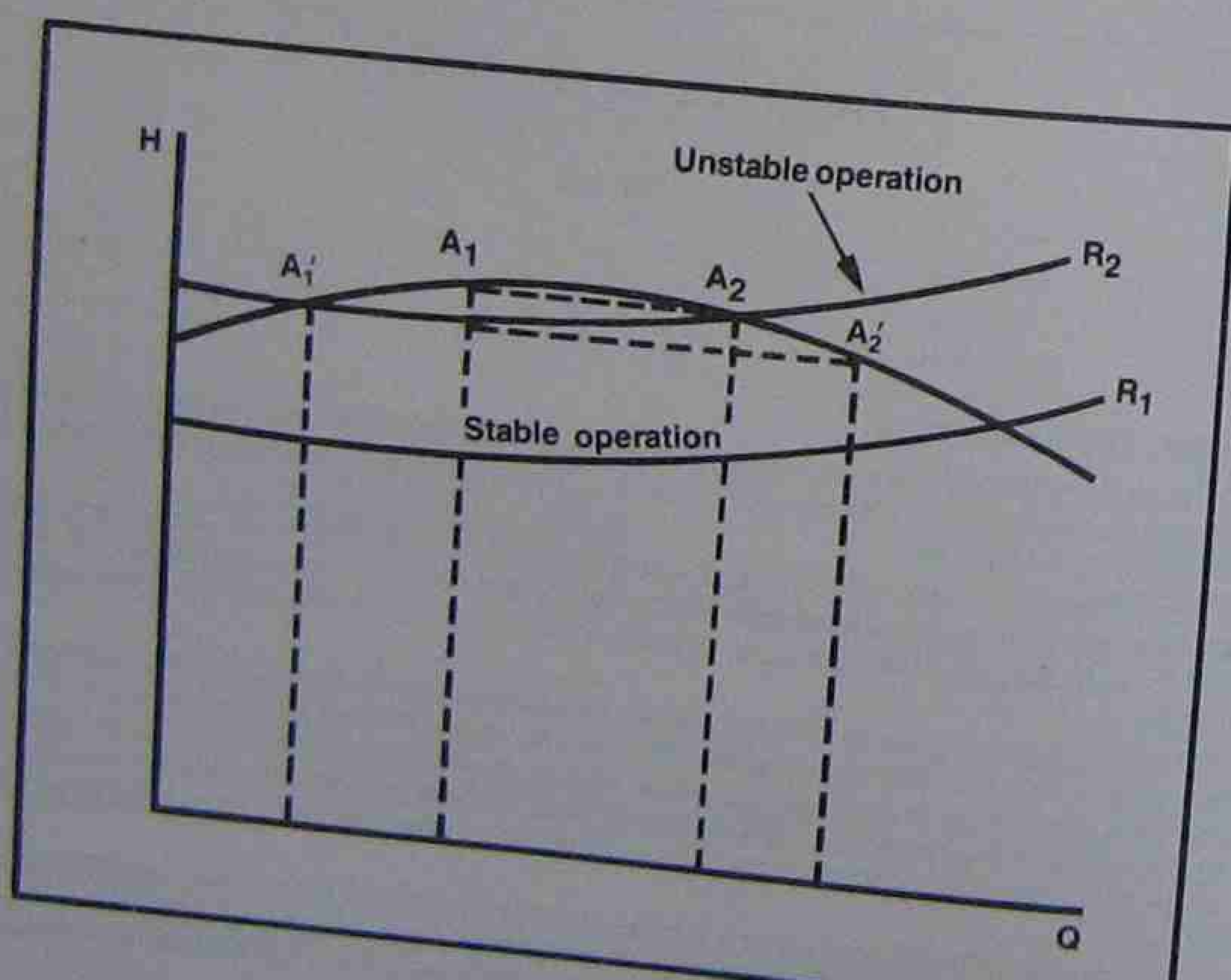


Fig. 6.3.1

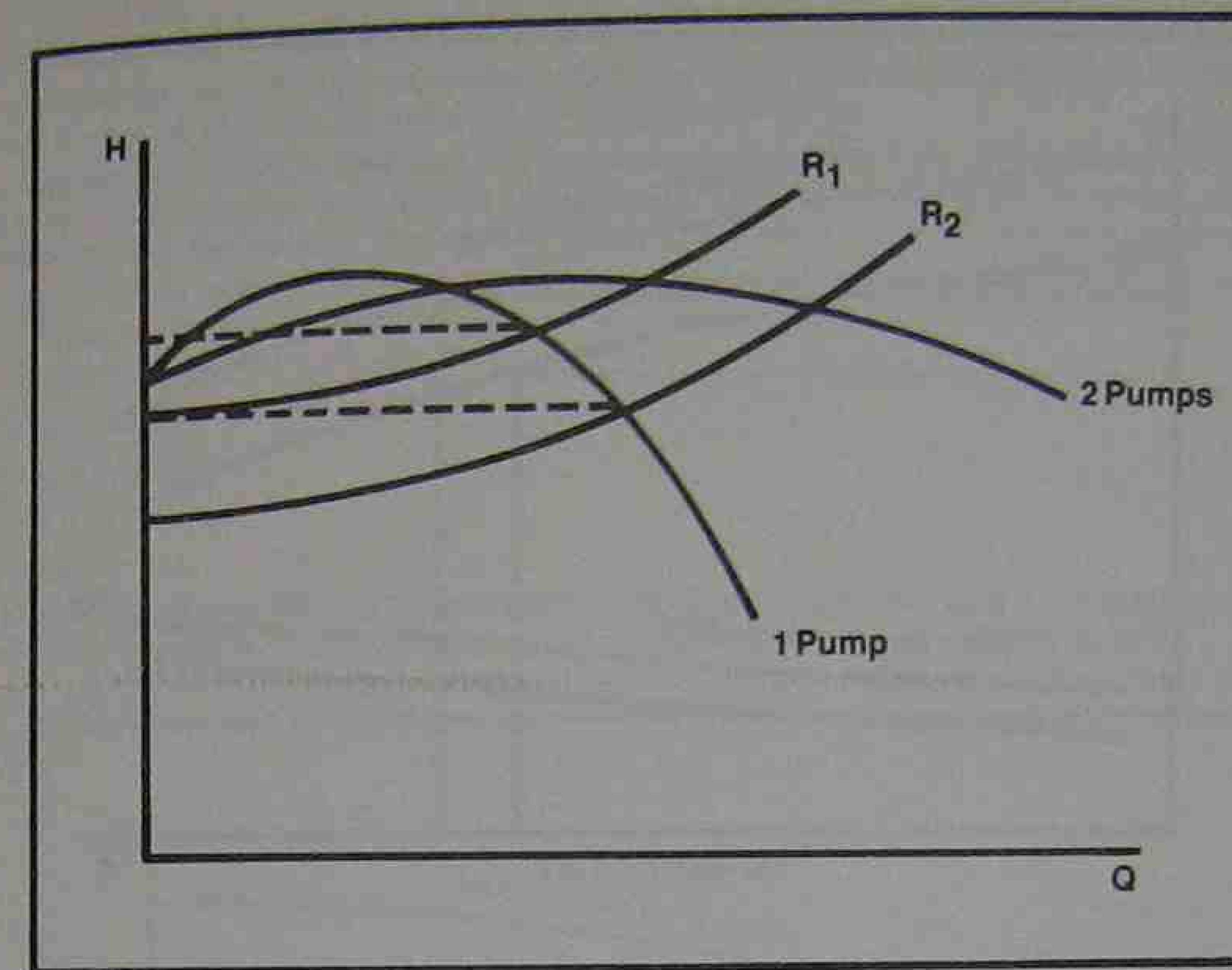


Fig. 6.3.2

6.4 AFFINITY LAWS

There are fundamental laws which can be used to predict changes in pump performance with variations in speed or impeller diameter. It is important in pump applications to be able to develop performance curves corresponding to various speeds or impeller diameters, from standard performance curves. The mathematical relationships between flow rate, head, power and speed which enable this are known as the Affinity Laws.

Speed Variation

For variation in speed with constant impeller diameter, the following rules apply:

a) Pump flow rate (Q) varies directly with the speed
   i.e., $Q_1/Q_2 = N_1/N_2$

b) Pump head (H) varies with the square of the speed
   i.e., $H_1/H_2 = (N_1/N_2)^2$

c) Power absorbed varies with the cube of the speed
   i.e., $P_1/P_2 = (N_1/N_2)^3$

   In using the above formulae, it is assumed that efficiency remains constant.

   In practice, the efficiency is slightly less at lower speeds since friction and drag constitute a larger proportion of hydraulic power.

   Example

   If the pump speed is 1450 RPM and duty is 200 L/s at 30m head; power = 120 kW. What would be the pump duty and duty power if pump speed is increased to 1600 RPM?

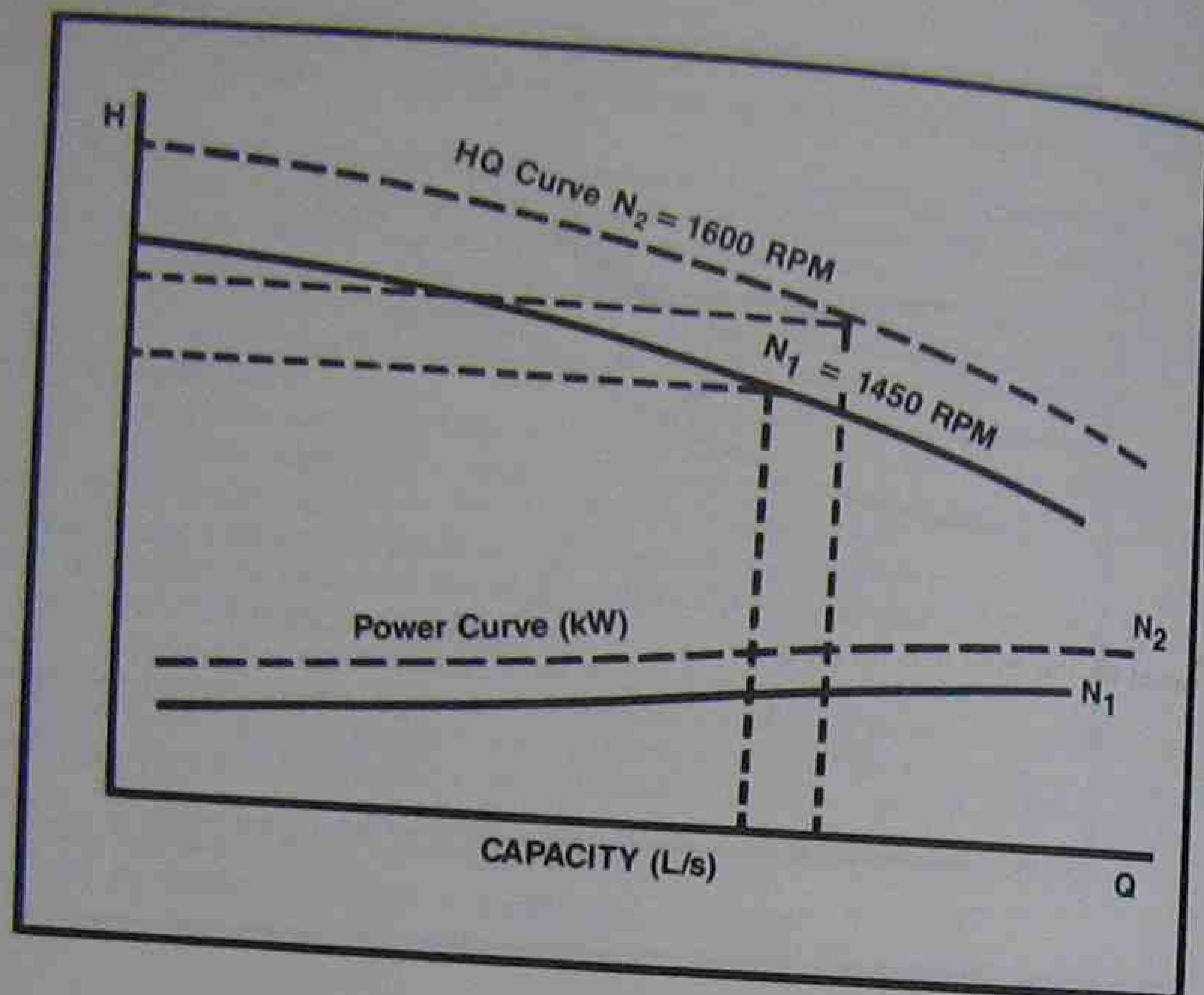


Fig. 6.4.1

$Q_1 = 200$ L/s
$H_1 = 30$m
$N_1 = 1450$ RPM
$kW_1 = 120$ kW
$N_2 = 1600$ RPM

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2}$$

$$\frac{200}{Q_2} = \frac{1450}{1600} \therefore Q_2 = 200 \times \frac{1600}{1450} = 220.69 \text{ L/s}$$

$$\frac{H_1}{H_2} = \left(\frac{N_1}{N_2}\right)^2$$

$$\frac{30}{H_2} = \left(\frac{1450}{1600}\right)^2 \therefore H_2 = 30 \times \left(\frac{1600}{1450}\right)^2 = 36.53 \text{ m}$$

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3$$

$$\frac{120}{P_2} = \left(\frac{1450}{1600}\right)^3 \therefore P_2 = 120 \times \left(\frac{1600}{1450}\right)^3 = 161.23 \text{ kW}$$

Duty at 1600 RPM is 220.69 L/s at 36.53 m head.
Duty power = 161.23 kW

Impeller Diameter Variation

If it is required to permanently reduce the H-Q characteristic of a centrifugal pump, without reducing speed, it may be effected by reducing the impeller diameter.

The resulting reduction in flow rate and head depends on the type of impeller. This is because apart from an alteration in peripheral velocity, there are also changes in the length and overlap of the vanes; in the width of the impeller at exit, and often in the discharge angle as well.

The effect of the reduced impeller diameter on the output can be approximately ascertained by a set of equations which are analogous to the Affinity Laws.

For constant speed:

$Q_1/Q_2 = D_1/D_2$
$H_1/H_2 = (D_1/D_2)^2$
$P_1/P_2 = (D_1/D_2)^3$

Example

If a pump has a duty of 500 L/s at 10m head and duty power is 60 kW using a 0.5m diameter impeller, what would be the duty and duty power if impeller diameter was reduced to 0.4m.

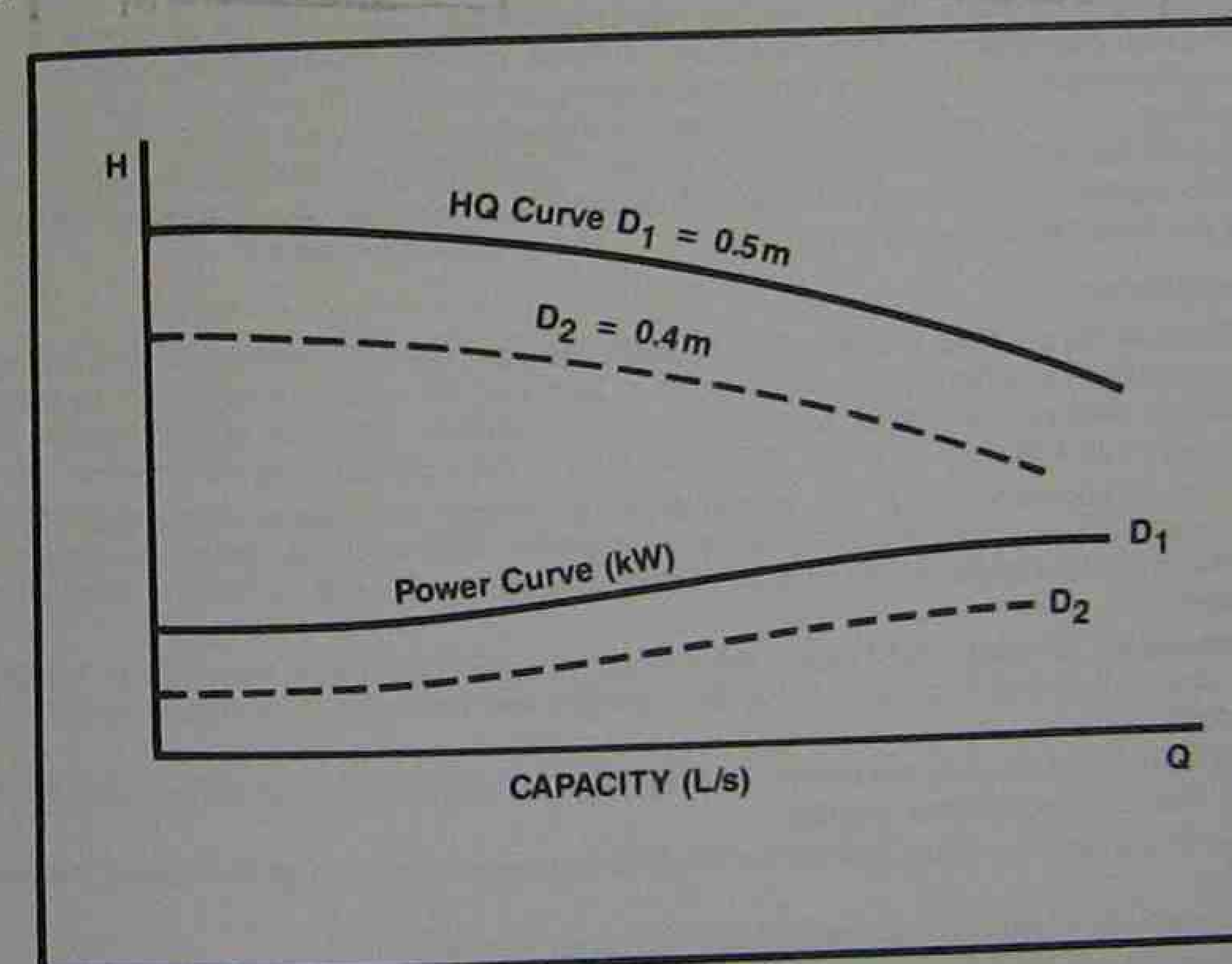


Fig. 6.4.2

$Q_1 = 500$ L/s
$D_1 = 0.5$m
$H_1 = 10$m
$kW_1 = 60$ kW
$D_2 = 0.4$m

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2}$$

$$\frac{500}{Q_2} = \frac{0.5}{0.4} \therefore Q_2 = 500 \times \frac{0.4}{0.5} = 400 \text{ L/s}$$

$$\frac{H_1}{H_2} = \left(\frac{D_1}{D_2}\right)^2$$

$$\frac{10}{H_2} = \left(\frac{0.5}{0.4}\right)^2 \therefore H_2 = 10 \times \left(\frac{0.4}{0.5}\right)^2 = 6.4\text{m}$$

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3$$

$$\frac{60}{P_2} = \left(\frac{0.5}{0.4}\right)^3 \therefore P_2 = 60 \left(\frac{0.4}{0.5}\right)^3 = 30.72 \text{ kW}$$

Duty at 0.4m diameter impeller is 400 L/s at 6.4m head

Duty power = 30.72 kW

It is important to emphasize that the affinity relationships are more accurate for speed variation than for impeller diameter variation. Whenever possible, curves showing tested performance for various impeller diameters should be used in estimating performance for reduced impellers. Should it be necessary to reduce the output of the pump considerably, the impeller should first be machined down to a diameter about 5% larger than calculated. The pumps should be tested with this impeller and the final diameter determined from the H-Q curve plotted from this data. The higher the specific speed of the impeller, the more important it is to observe this precaution.

6.5 CAVITATION

If the **NPSHA** of the pump is less than the **NPSHR**, the pumped liquid will vaporise in the region of the impeller eye i.e. where the local pressure is less than the vapour pressure of the liquid. In this region the fluid will consist of a liquid plus vapour cavities. This can, in the extreme, result in the formation of a vapour lock and the prevention of the fluid entering the impeller.

The more usual occurence is for the fluid to progress through the impeller to a region of higher pressure where the vapour cavities collapse or 'implode'. This implosion takes place with such rapidity that liquid impinges on the impeller vane with sufficient force, to remove material from the surface.

This phenomener is called 'cavitation'. The cavitation cycle comprises therefore of two phase changes; one from liquid to vapour during the initiation and the other from vapour back to liquid during cavity collapse.

Cavitation may be caused by excessive suction lift, insufficient NPSHA or operation at too high a speed. The resulting effects include:

1. Pitting of material surfaces due to the continual hammering action of the collapsing vapour cavities.
2. Significant reduction of performance due to vapour formation.
3. The crackling noise (like gravel going through the pump) caused by vapour cavity collapse.

Severe cavitation usually results in excessive noise, vibration and damage to the pump, whereas mild cavitation may produce nothing more than a small reduction in pump efficiency and moderate wear of pump components.

Cavitation for rotodynamic pumps generally occurs in the eye of the impeller. For reciprocating pumps it usually occurs at the beginning of each stroke within the pump barrel or cylinder and for rotary pumps cavitation will occur if by the end of the opening phase a rotating pocket is not filled with liquid.

In multistage pumps, cavitation normally affects the first stage only.

Section 7

PUMP MATERIALS

Section 7—PUMP MATERIALS

7.1 PREFERRED MATERIALS

The following list of cast and wrought alloys has been compiled by the Australian Pump Manufacturers Association Ltd. in association with the Australian pumping industry.

In the preparation of this list, consideration has been given to the user's requirements, service performance and availability. It is possible to select an alloy from the list to meet the majority of pumping applications and to obtain the necessary material without great difficulty. Wherever possible materials should be selected from the list unless specific applications dictate otherwise.

Any decision to use alloys other than those listed should be carefully considered and discussed with the pump manufacturer.

The near equivalent specifications quoted are not exact equivalents and reference should be made to full detailed specifications for further information.

CAST ALLOYS

Description	Aust. Standard	Near equiv. specifications British	American	German
Grey Iron	1930-T180	1452 GR 180 (1452 GR 12)	ASTM A48-25	1691 GG-20
Grey Iron	1830-T220	1452 GR 220 (1452 GR 14)	ASTM A48-30	1691 GG-20
Grey Iron	1830-T260	1452 GR 260 (1452 GR 17)	ASTM A48-40	1691 GG-25
Austenitic Iron	1833 GR L Ni Cu Cr 15 6 3	3468 GR L Ni Cu Cr 15 6 3 (AUS 101 GR B)	ASTM A436-1B	1694 GGL-Ni Cu Cr 15 6 3
Austenitic Iron	1833 GR L Ni Cr 20 3	3468 GR L Ni Cr 20 3 (AUS 102 GR B)	ASTM A436-2B	1694 GGL-Ni Cr 20 3
Ductile Austenitic S.G. Iron	1833 GR S Ni Cr 20 2	3468 GR S Ni Cr 20 2 (AUS 202 GR A)	ASTM A439-D2	1694 GGG-Ni Cr 20 2
Ductile S.G. Iron	1831/400-250-12	2789 GR420/12 (SNG 27/12)	ASTM A536/60-40-18	1693 GGG-42
Ductile S.G. Iron	1831/600-370-3	2789 GR600/3	ASTM A536/80-55-06	1693 GGG-60
Abrasion-Resistant White Iron	2027-Ni Cr1-550			
Abrasion-Resistant White Iron	2027-Ni Cr2-500			
Carbon Steel	2074-C3	1504-161 GR 430 (592 GR A)	ASTM A216-WCA	1681 G5-45
Carbon Steel	2074-C4-1	1504-161 CR 480	ASTM A216-WCB	
Carbon Molybdenum Steel	2074-L5A	1504-245	ASTM A217-WCI	
Martensitic Corrosion Resisting Steel	2074-H3B	1504-420 C29 (1630 GR B)	ASTM A743-CA15	17445 G-X20 Cr14

CAST ALLOYS—Continued

Description	Aust. Standard	Near equiv. specifications British	American	German
Austenitic Corrosion Resisting Steel	2074-H5C	1504-347C17 (1631 GR B)	ASTM A743-CF8C	17445 G-X7 Cr Ni Nb 18 9
	2074 H6B	1504-316C16 (1632 GR B)	ASTM A743-CF8M	17445 G-X5 Cr Ni Mo 18 10
	2074 H6C	1504-31C17 (1632 GR C)		
Leaded Gunmetal	1565-C83600	1400-LG2	ASTM B584-C83600	1705 G- Cu Sn 5Zn Pb
Leaded Gunmetal	1565-C92410	1400-LG4	CDA924	1705 G-Cu Sn 6Zn Ni
Aluminium Bronze	1565-C95810	1400-AB2	ASTM B148-958	1714 G-Cu A1 10Ni
Manganese-Alum. Bronze	1565-C95710	1400-CMA1	CDA957	—
Leaded Bronze	1565-C93710	1400-LB2	ASTM B584-C93700	1716 G-Cu Pb 10Sn
Leaded Bronze	1565-C93500	1400-LB4	CDA905	1705 G-Cu Sn 10
Copper Tin	1565-C90250	1400-CT1		1705 G-Cu Sn10
Phos. Bronze	1565-C90710	1400-PB1	ASTM B30-C90700	
Aluminium	1874-BP401	1490-LM6	ASTM B85-512A	1725 G A1 Sil2

WROUGHT ALLOYS

Description	Aust. Standard	Near equiv. specifications British	American	German
Free cutting steel	1443/S1214	970-230 MO7 (EN 1A)	AISI 1214	
Carbon Steel	1443/CS1020	970-070 M20 (EN 3B)	AISI 1020	C22
Carbon Steel	1443/CS1030	EN6A	AISI 1030	C35
Carbon Steel	1443/C S1040	970-080 M40 (EN8)	AISI 1040	C45
1% Chrome Molybednum Steel	1444/4140	970-708 M40 (En 19A)	AISI 4140	17200 42 Cr Mo4
Chrome Moly. Steel	1444/X4150	—	—	
Stainless Steel	2837/316	970-316S16 (EN58J)	AISI Type 316	17440 X5 Cr Ni Mo18 12
Stainless Steel	2837/321	970-321S12 (EN58B)	AISI Type 321	17440 X10 Cr Ni Mo18 12
Stainless Steel	2837/416	970-416S21	AISI Type 416	
Stainless Steel	2837/420	970-420S37 (EN56C)	AISI Type 420	17440 X20 Cr 13
Stainless Steel	2837/431	970-431 S29 (EN57)	AISI Type 431	17440 X22 Cr Ni 17
Naval Brass	1567/464	2874-CZ112	ASTM B21 B124-464	17660 Cu Zn39 Sn
Free Cutting Brass	1567-385	2874-CZ121	ASTM B455-385	17660 Cu Zn40 P6
Monel		3071-NA1		

7.2 SELECTION OF MATERIAL

In selecting a suitable material for a pump component, a number of factors have been taken into consideration by the pump manufacturer. The manufacturer's choice will often be a compromise of cost, availability and performance. It frequently happens that the cheapest material is the best material for a particular application. The humble cast iron is a case in point. In contact with certain liquids it may have a better corrosion resistance than more expensive materials and there is also the added advantage of the noise dampering property of cast iron.

Plastics are another example of a low cost material having unique properties, particularly corrosion resistance and strength to weight ratio. Frequently plastics will be selected where cost is not a consideration. The use of plastics in space vehicles is an excellent example of this.

If the pump user is faced with a material option it should not be assumed that cost is a reflection of superiority or inferiority.

Some of the factors affecting material selection are as follows:—

Type of pump
Part of pump
Liquid pumped (temperature, corrosion, erosion, abrasion)
Manufacturing method (casting or machining properties)
Initial cost
Economic life
Material properties (strength, endurance limit, pressure tightness, galling characteristics, etc.)

The final selection of materials may vary between manufacturers for pumps offered for the same operating conditions. One reason for this is that manufacturers tend to specialise in particular methods of manufacturing ie. injection moulding, casting, welding fabrication etc. and the preferred material will be that which is best suited to a particular form of manufacture.

The pattern of material selection for large pumps is, in general, different to that for small pumps. For example, slurry, gravel and coal handling pumps are often selected on the basis of the minimum size of solid that can be handled. They tend to be large and in order to withstand abrasion and impact, are either rubber lined or constructed of hard abarasion resistant cast iron. Additionally to meet the wide variations in required performance standardisation of pump sizes must be limited so that the annual requirement for pumps of a given size is low. It follows that restrictions are placed on the type and scope of materials available for these pumps. A similar set of circumstances apply in the case of pumps for power stations and public utilities such as water supply and sanitation. In these fields of application, ferrous metals are generally used.

In the small domestic pump market such as household pressure units, pool pumps, garden pumps and multistage borehole pumps etc., the material selected for the wetted parts must be resistant to a whole range of natural waters, including chlorinated water and tens of thousands of components may be required per annum. Injection moulded plastics are widely used and a range of materials is available to the manufacturers with excellent strength and corrosion resistant properties.

In between the large industrial class of pump and the domestic pump is a whole range of medium sized pumps designed for a multitude of purposes. This intermediate range of pumps may be categorised into three groups.

(i) Special purpose pumps (eg. marine pumps, food and beverage pumps)
(ii) Chemical pumps
(iii) General purpose water pumps

For group (i) the manufacturer will select materials of construction which, from experience, are known to be resistant to and compatible with the product being pumped.

Group (ii) pumps may be required to handle a specified fluid from a wide variety of chemicals and must therefore be designed for construction in a range of materials and configurations, particularly in relation to shaft sealing.

Group (iii) Water pumps would appear to be the least troublesome where material selection is concerned. However this is not the case. Dissolved salts, gases and pollutants etc. frequently create unexpected and severe corrosion problems. For this reason material options are made available to overcome problems of this nature.

7.3 STANDARD CONSTRUCTION MATERIALS

(a) General Purpose Water Pumps

General purpose water pumps for low and medium pressure usually have cast iron casings, high tensile steel shafts, cast iron or bronze impellers and cast iron or bronze wearing parts.

For higher pressures, the casing is usually manufactured from cast, forged or welded steel with the shaft, impellers and diffusers in stainless steel.

The table below is a general guide to the selection of materials for the range of fluid pH.

pH of fluid	Nature of fluid	Pump material
0-4	Strongly acid	all stainless steel
4-6	acidic	all bronze
6-9	neutral	cast iron or steel (bronze fitted)
9-14	alkaline	all iron

All stainless steel pump—all components in direct contact with the pumped liquid are of corrosion resistant steel of suitable properties for the specific application.

All bronze pump—all components in direct contact with the pumped liquid are of a bronze composition of suitable properties for the specific application.

Bronze fitted pump—the pump casing is cast iron and the impellers are bronze. The impeller rings, bushes and shaft sleeves (if fitted) are also bronze.

All iron pump—all components in direct contact with the pumped liquid are of ferrous metal.

Stainless steel fitted pumps—the pump casing is of a material suitable for the application. The impeller, impeller rings, bushes and shaft sleeves (if fitted) are of a corrosion resistant steel of suitable properties for the specific application.

A term which is often used in conjunction with bronze pumps is 'zinc-free bronze'. The origin of this specification stems from the dezincification corrosion problems experienced with brasses and modified brasses. What is actually required is a material which is not susceptible to dezincification, eg. leaded gunmetal. In fact, the zinc contact can rise to 15% without significant dezincification occurring.

For marine application the following materials are normally recommended:

Casing —leaded gunmetal (or cast iron in some instances)
Impeller —aluminium bronze
Wearing rings—leaded bronze
Shaft —stainless steel or bronze with stainless steel sleeves

(b) Positive Displacement Pumps

With the wide range of materials available most manufacturers have standardised their positive displacement equipment on a range of metallic, plastic and elastomeric materials designed to be employed in the majority of applications. Materials currently in use include:

METALLIC	Various grade of stainless steel
	Monel metal
	Aluminium
	Cast irons (including vitreous lined)
	Special purpose steels (eg. Hastelloy)
PLASTIC	PVC
	Polypropylene
	P.T.F.E.
	Nylon
ELASTOMERIC	(Synthetic rubbers)
	Butyl
	Hypalon
	Nitrile
	Viton

By virtue of its corrosion resistant qualities grade 316 stainless steel is one of the most commonly used materials for general purpose positive displacement pumps. However, stainless steels may be non-resistant to products with high chloride concentrations and to some acids. In these instances, especially in regard to critical components such as pump valves, shaft seals, plungers, pistons, etc. where no corrosion is permissible, other materials such as P20, Hastelloy, ceramic, Monel or titanium may be necessary.

Plastic materials (PVC, polypropylene, PTFE, etc.) are extensively used for their resistance to chlorides and acids in low temperature applications. PTFE, due to its chemical inertness is often used for gaskets and glands, particularly in the food industry since it does not harbour bacteria and hence is not a source of contamination.

Diaghragms, glands, seals, pump stators and rotors etc. are designed in elastomeric materials, specifically selected for their compatability with the product.

High alumina ceramic is ideal for pump plungers, valves and mechanical shaft seals with its extreme hardness and wide resistance to corrosion.

7.4 PLASTIC PUMPS

Plastics, particularly the more recently developed thermoplastic offer considerable attractions to the pump manufacturer; i.e.—lightweight, relatively strong, inexpensively fabricated, excellent corrosion resistance and excellent appearance.

The following concise list of engineering thermoplastics includes the majority of plastic materials used in commercial pump manufacture in Australia.

(a) **Nylons**

Possibly the most extensively used thermoplastics. Nylon, nylon 11 and nylon 23, natural or glass filled, provide high impact strength and toughness. Used for impellers, housings, venturis, etc.

(b) **Acetals**

Acetals provide excellent resistance to wear, can operate at reasonably high temperatures and have good chemical resistance. Acetals are used for poppets, bushes, jets, etc.

(c) **Thermoplastic Polyesters (P.B.T.'s)**

Natural or glass filled P.B.T.'s offer high heat deflection temperatures and are used for components such as diffusers, impellers, pressure tanks, pump housings, etc.

(d) **Polycarbonate**

Polycarbonate is an extensively used material which offers high impact resistance and good dimensional stability. Polycarbonate is commonly used for impellers, casings, etc. Normally used as a glass filled material however natural clarity is sometimes a required property.

(e) **Polysulphone**

Polysulphone is normally used where severe temperature or stress conditions are present. It has a very high heat deflection temperature and excellent chemical resistance. It is normally used glass filled for impellers, housings, etc.

(f) **Modified Polythenylene Oxide (Noryl)**

A commonly used glass filled material provides mouldings with good dimensional stability and a hard scratch resistant surface. Noryl has a tensile strength of sufficient magnitude to allow its use in casings and injector housings, etc.

(g) **Polyphenylene Sulphide (P.P.S.)**

This dimensionally stable material has excellent electrical resistance and high temperature resistance. P.P.S. also has excellent resistance to solvents and is used for pump components which are intended for more severe conditions.

7.5 REFERENCES

It is not intended in this volume to give a list of materials suitable for use with the multitude of liquids which can be pumped. However, if this information is required a comprehensive table indicating the pump materials commonly used for pumping various liquids can be found in the American "Hydraulic Institute Standards for Centrifugal, rotary and reciprocating pumps"—Table 11.

Section 8

PIPE SYSTEMS

Section 8—PIPE SYSTEMS

Many of the problems experienced by pump users are brought about by poor pipeline design or improper application of the pump. Exercise of care in designing the layout of the total system and adherence to good pumping practice will result in a comparative improvement of performance.

Pumping systems, irrespective of the type of pump used, can be split into two fundamentally separate sections, namely; the suction and the discharge.

The suction is that part of the system between the source of the liquid to be pumped and the pump itself. When the pump is above the liquid source, it is said to operate with a suction lift. When the pump is positioned below the liquid level it operates with a suction head.

The remainder of the system constitutes the discharge.

8.1 SUCTION PIPEWORK

(a) It is good practice to keep the suction pipe as short and straight as possible.

(b) Where possible, arrange for the pump to operate with a suction head.

(c) Eliminate all possibility of air pockets being trapped in the suction pipework, e.g. ensure pipework rises continuously towards the pump. See Fig. 8.1a and 8.1b.

(d) Figs. 8.2 and 8.3 indicate where air can be trapped in the suction system, thereby reducing pump performance. When bends are unavoidable in a suction pipe, air release valves must be fitted to the line at point 'A' and a non-return valve or foot valve must be fitted at 'B'; otherwise the pump may not prime.

(e) The suction pipe should be accessible and not for example embedded in concrete.

(f) When choosing footvalves, strainers, bends etc., select those which will provide minimal restriction to flow.

(g) Some care must be taken in the design of the suction inlet. (refer Section 9)

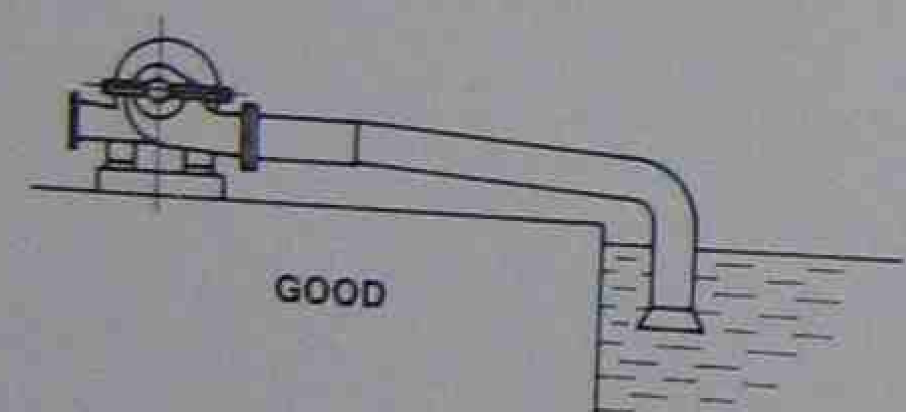


Fig. 8.1a

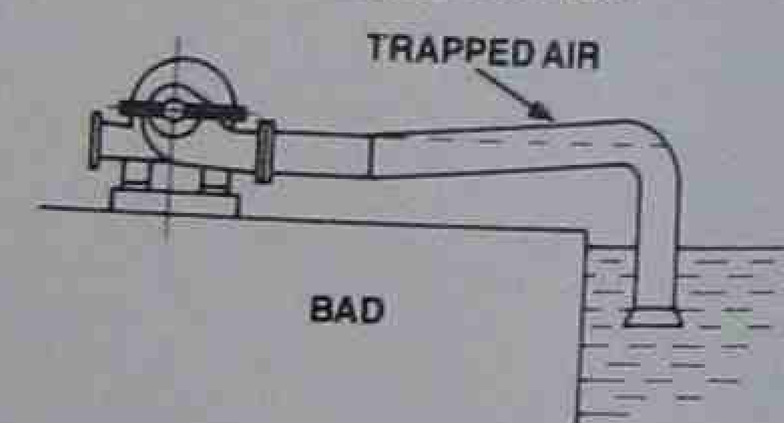


Fig. 8.1b

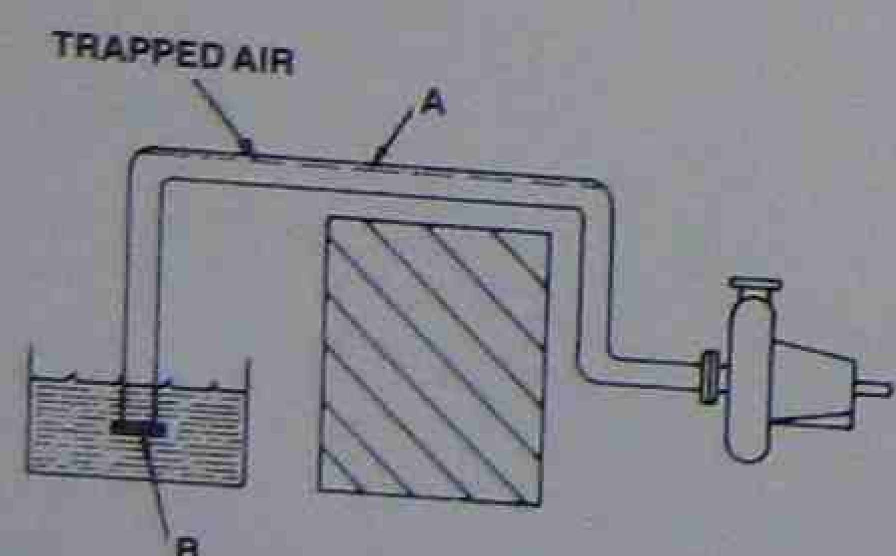


Fig. 8.2

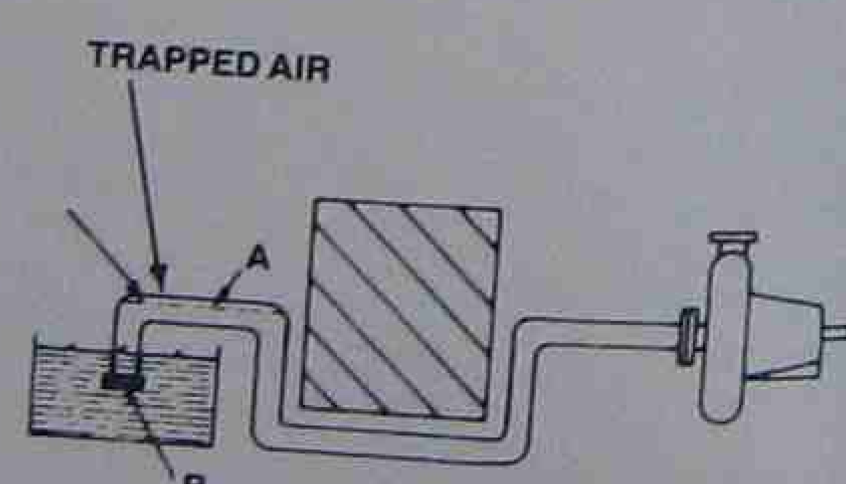
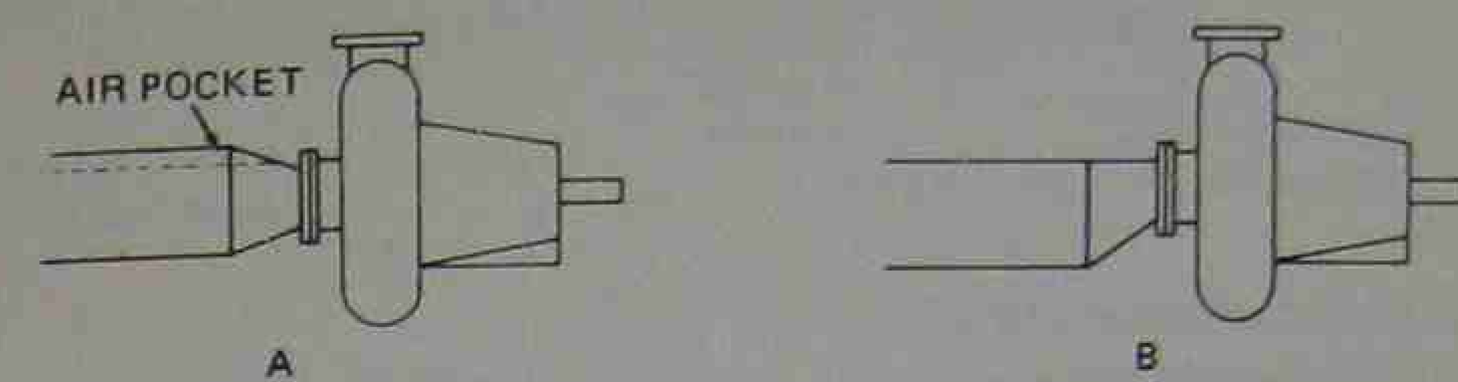


Fig. 8.3

Suction Pipe Size:
The diameter of the suction pipe should be sized according to the flow and allowable head loss; in any case it should be equal in diameter to the pump inlet connection.

If a pipe larger than the pump inlet is to be used, good practice stipulates that an eccentric reducer should be fitted to avoid the possibility of air pockets.



CONCENTRIC – NOT RECOMMENDED   ECCENTRIC – RECOMMENDED

Fig. 8.4

8.2 DISCHARGE PIPEWORK

Discharge pipelines should be designed to minimise losses. All bends, valves, strainers, reducers and other fittings in the line contribute to friction losses.

It should be noted that accurate calculation of friction losses is essential. It follows that the characteristics of the liquid to be pumped must also be considered.

A comprehensive tabulation of head losses that can be expected for given flow rates in commonly used pipes and fittings can be found in the APMA publication "Pipe Friction Handbook".

8.3 SELECTING THE PIPE

Several factors must be considered:

Fluid:		Pipe:	
	Viscosity		Length
	Temperature		Roughness
	Chemical Properties		Strength
	Flow Rate		Cost
	Pressure		

Of these factors, the temperature and chemical properties will probably determine the material from which the pipe is made; or at least exclude the use of certain materials. For example, oil at 300°C could not be pumped through plastic pipes. The pumping of high viscosity fluids substantially increases the friction head, and may necessitate the use of a larger diameter pipeline and long radius bends.

High friction loss can be tolerated in short lengths of pipe, such as may be found in a processing plant. However, on longer pipelines e.g. water supplies, friction becomes a major part of the total head and should be kept to an economic minimum. Average velocities in the pipe of 2m per second are usually found to be acceptable when pumping water over long distances.

8.4 VALVES

It is recommended that a centrifugal pump be fitted with a non-return valve at the discharge. When the pump is shut down the valve prevents reverse rotation of the pumpset and possible damage to the pump and/or driver depending on the mechanical arrangement.

Valves should be installed in such a way that the pump can be isolated for maintenance and inspection purposes.

8.5 AIR VENT VALVES

On pipelines which have crests and troughs, an air vent valve should be fitted at the highest points to allow accumulated air or other gases to escape from the pipe. These valves should be fully automatic in operation and close when liquid enters the valve. Refer Fig. 8.5

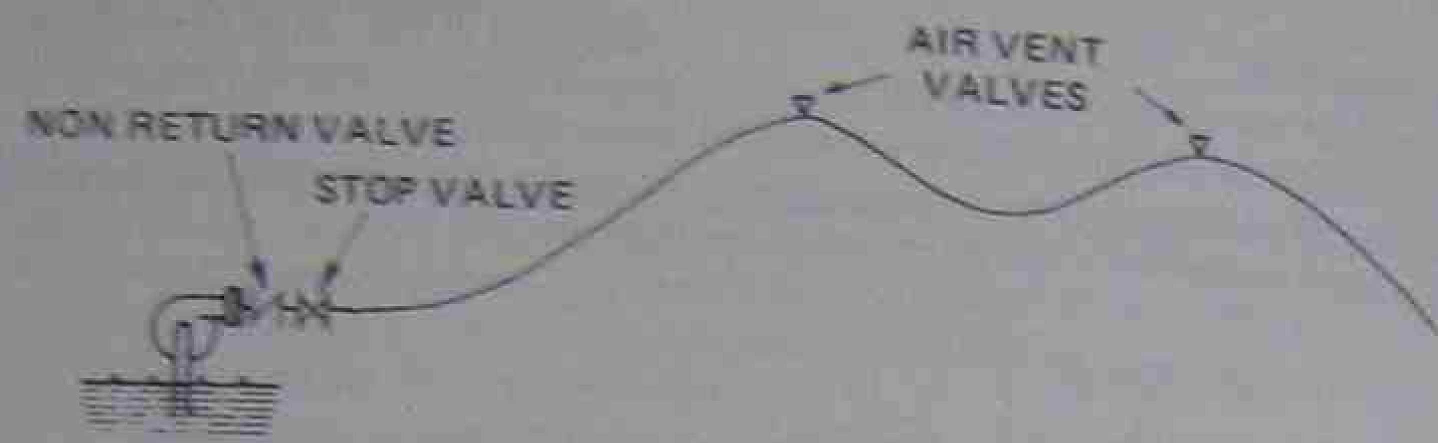


Fig. 8.5

8.6 BRANCH CONNECTIONS

If it is necessary to pipe liquid to more than one discharge point, a "Y" type connection should be used to reduce losses at the branch. Refer Fig. 8.6.

Fig. 8.6

8.7 REDUCTION OF PIPE DIAMETER

If a number of branches are taken off a main pipeline, it may be possible to reduce the diameter of the main pipe, which can be sized to suit the reduced flow.

Sudden changes in diameter are to be avoided as these produce friction losses. The change in diameter should be effected gradually by a taper piece. For best results, the included angle of the taper piece should be 10-13°. Refer Fig. 8.7.

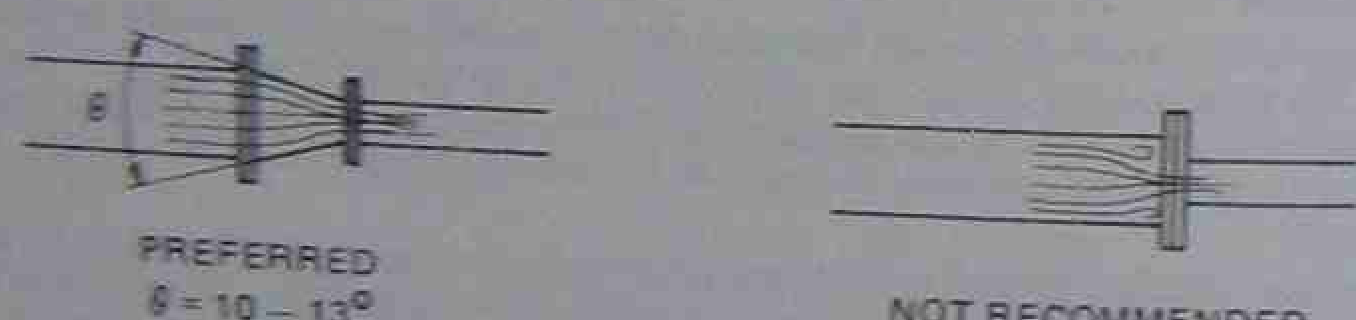


Fig. 8.7

8.8 THRUST BLOCKS

Adequate support and anchorage for pipework must be provided, regardless of whether the pipes are laid above or below ground. Detailed requirements should be sought from the pipe manufacturer to ensure that no mechanical or hydraulic loads are imposed on the pump. Refer Fig. 8.8.

Particular attention should be given to pipework design where antivibration devices are incorporated. e.g. inertia base plates, flexible mounts.

If flexible piping is used particularly on the discharge of a pump, it must be adequately supported and restrained. For example, the coupling alignment may be correct when the pump is shut down, yet become badly misaligned when flexible hoses are allowed to impose forces on the pump during operation.

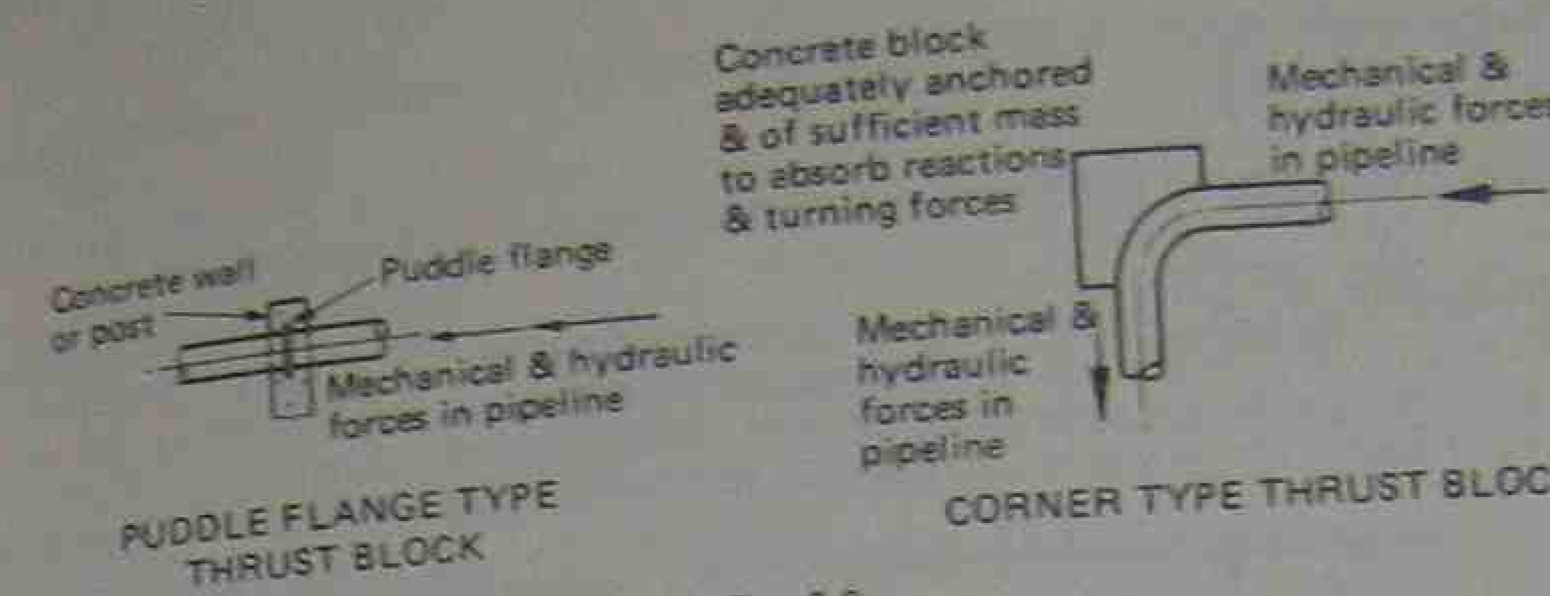


Fig. 8.8

8.9 SPECIAL CONSIDERATIONS

Certain aspects of pump and pipeline design require special consideration and these are usually related to the liquid being pumped, e.g. solids in suspension, chemicals, molten solids, extremes of temperatures. In these instances the pump manufacturer should be consulted.

8.10 PRESSURE SURGES AND WATER HAMMER

Any system in which a fluid is in motion or capable of being set in motion, is susceptible to pressure surge or water hammer phenomena. Surges occur whenever the velocity in a conduit is changed from a steady state condition to another. It is likely that the most severe conditions will correspond to starting or stopping of the pump or 'instantaneous' opening or closure of a valve in the pipeline.

The effects of pressure surges in a system may be compared with the forces on a moving vehicle during periods of acceleration or retardation. When, for example, the vehicle is brought to rest by violent application of the brakes, severe stresses are imposed on parts of the vehicle. The analogy is that a pumping system may be subject to quite dangerous pressures during acceleration and retardation of liquid in the pipes.

Accordingly, if a valve is slammed shut at the end of a delivery line, a transient surge or wave resulting from the sudden gate closure moves towards the pump. If the pipe walls were rigid and inextensible, the speed of the pressure wave would approach the velocity of sound in the liquid—about 1400m/s for water—and since the magnitude of the pressure rise in the pipe due to instantaneous gate closure is proportional to the speed of the wave and the change in water velocity, then extremely high pressures can be propagated. Consequently, if those pressures are not properly controlled, they may cause noise and vibration in the system, piping to burst, or a pump casing to crack.

Water hammer calculations are complex and beyond the scope of this handbook. It is recommended that specialized engineering services be employed in cases where it may be a problem.

It should be emphasised that water hammer effects are always possible and may determine the design loads for the system

The pipeline longitudinal profile is also important and route selection could be influenced by the water hammer implications. Some of these facts are discussed at elementary level in a paper in the Pump Technical Congress, University of Melbourne, 26th November, 1980 and in greater depth by Dr. B. B. Sharp in Water Hammer, Problems and Solutions published by Edward Arnold.

8.11 DISCHARGE OF SYPHON LINES

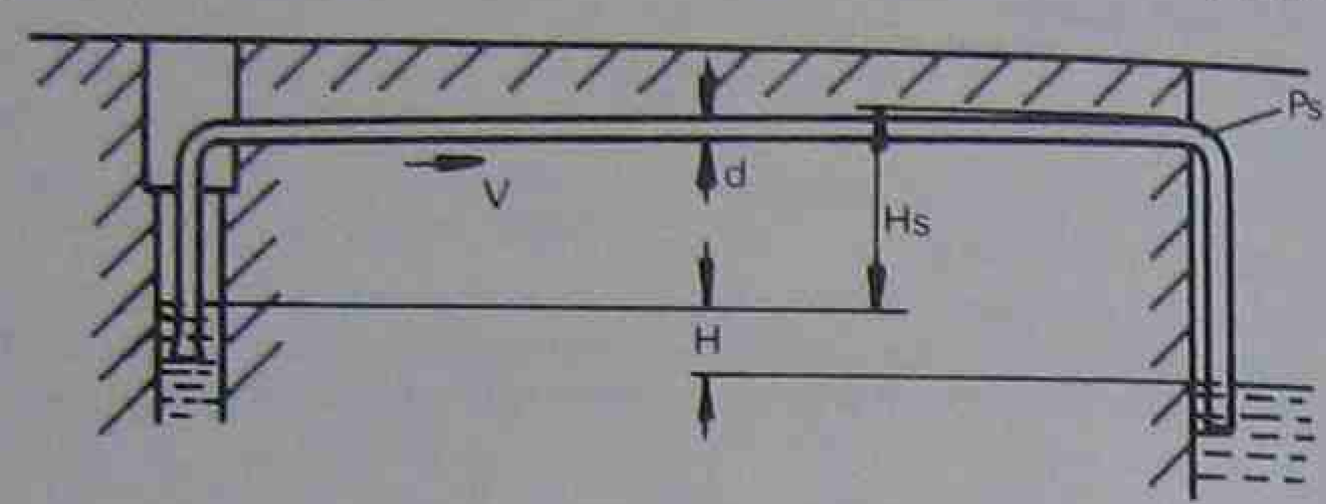
Syphon lines are often used in ground water pumping stations to feed the discharge from industrial wells or boreholes into a collecting well from which the water is then pumped into a reservoir.

The difference in level H (Fig. 8.9) serves as the effective potential head to produce the flow velocity v and to overcome all the friction head losses.

i.e. $H = \left(\frac{v^2}{2g}\right)\left(1+K\right)$

Transforming this equation in terms of syphon discharge and pipe diameter gives:—

$Q = 0.00348\ d^2\sqrt{\frac{H}{(1+K)}}$ where Q = syphon discharge in L/s.
H = differential head in m.
d = pipe bore diameter in mm.
K = sum of the resistance coefficients of pipeline, bends, etc.



SCHEMATIC ARRANGEMENT OF SIPHON LINE

Fig. 8.9

Theoretically, the limit of possible flow up the inlet leg of a syphon is reached when the pressure Ps at the highest point in the system is absolute zero(−10.35m of water). In practice however, this limit is less because when the absolute pressure p_s at the summit becomes equal to the vapour pressure of the liquid, cavitation begins. Consequently, vapour collects at the bend forming so called vapour locks and the flow stops. Hence, for satisfactory operation of a syphon, the maximum static suction lift allowable can be expressed by the following equation:

$$H_s < 0.102\ (p_a - p_v)\frac{v^2}{2g}\ (1 + \Sigma K_s) - S$$

where

H_s = static suction lift in metres
p_a = atmospheric pressure in kPa
p_s = absolute pressure at the highest point in the system
p_v = vapour pressure of liquid in kPa
K_s = the sum of the resistance coefficients for the loss of head from inlet to the highest point of the syphon.
S = a safety margin—a deduction of 1 to 2 m depending on the quality of the pipeline installation.

If the syphon is a long stationary metal pipeline, provision must be made at the summit of the syphon to evacuate the air. This can be most conveniently achieved by means of an automatically controlled vacuum pump. If suitably arranged, this pump can also be used to evacuate the syphon line for the purpose of starting up.

Example 1.

A 150 mm diameter schedule 40 steel syphon line is 200 m long and comprises two right angle elbows and one inlet branch fitting as represented in (Fig. 8.9). The difference in levels is 3.5 m. What approximate discharge can be expected from such an installation?

Using head losses in pipes and fittings obtained from Section 2.1 of the APMA "Pipe Friction Handbook":

Pipe internal diameter = 154.1 mm
K_1 for sharp edged inlet = 0.5
K_2 for right angle elbow = 0.45
Total K_f for fittings = $0.5 + (2 \times 0.45) = 1.4$

K_p for 200 m of the schedule 40 steel pipe is dependent on flow.

Since the flow is unknown the syphon discharge equation must be solved implicitly by assuming a value of flow through the pipe, or since most losses will be in pipe friction, proceed as follows.

i.e. Assume a pipe friction loss of $\frac{1.5\ m}{100\ m}$ for the straight length of steel pipe.

Hence H_f for pipe $= \frac{1.5}{100} \times 200 = 3\ m$

From TABLE A—"Pipe Friction Handbook" for a head loss of $\frac{1.5\ m}{100\ m}$ the flow Q is approximately 30 L/s and the velocity head is 0.132 m.

Now $K_p = \frac{3}{0.132} = 22.7$

$\therefore$ Total $K = K_f + K_p = 1.4 + 22.7$

$\therefore$ $K = 24.1$

Now $Q = 0.00348 d^2 \sqrt{\frac{H}{1+K}}$

$\therefore Q = 0.00348\ (154.1)^2 \sqrt{\frac{3.5}{1+24.1}}$

$\therefore Q = 30.9$ L/s

Since the assumed value of $\frac{1.5\ m}{100\ m}$ pipe friction loss results in a calculated flow of 30.9 L/s which is not equal to 30 L/s as obtained from TABLE A another assumption of friction loss has to be made and the above procedure repeated until the two values are equal.

Example 2

If in the same installation as in Example 1, it were required to obtain 50 L/s from the syphon. What differential head would be needed to produce this flow?

Under these conditions, the equation for discharge of a syphon becomes an explicit equation as all variables are known.

As previously, total K_f for fittings is 1.4 and from TABLE A for a flow of 50 L/s.

Velocity of flow = 2.68 m/s
Head loss = 3.98 m/100 m
Head loss for 200 m of pipe $= \frac{3.98}{100} \times 200 = 7.96$ m

$K_p = \frac{2g(H_f)}{v^2}$

$\therefore K_p = \frac{2 \times 9.8 \times 7.96}{(2.68)^2}$

$\therefore K_p = 21.7$

Hence, total $K = 21.7 + 1.4 = 23.1$

Transposing syphon discharge equation gives:

$$H = (l + k) \left[\frac{Q}{(0.00348\, d^2)} \right]^2$$

$$H = (1 + 23.1) \left[\frac{50}{(0.00348 \times 154.1 \times 154.1} \right]^2$$

$$\therefore H = 8.82 \text{ m}$$

Section 9

INTAKE DESIGN

Section 9—INTAKE DESIGN

9.1 INTAKE DESIGN

An unsatisfactory hydraulic design of an intake or sump can have far-reaching implications on the overall performance of a pumping station.

The fluids under consideration are water and air, the water being pumped and the air being above the free surface or, in some cases, being entrained within the water. However, the results will apply in principle to other fluids used in the process industries as long as allowance is made for effects arising from the difference in the fluid properties. Viscosity and density are the most important properties to be considered.

Poor hydraulic design of a pump sump and intake could arise because of insufficient attention at the design stage or because of site constraints. A summary of the possible consequences is shown in Table 9.1.

9.2 OPTIMUM SUMP VOLUMES

During start-up of an electric motor, higher than normal operating currents are drawn from the power supply. This causes a transient disturbance in the supply, resulting in a voltage dip, and produces significant head energy in the motor. These effects put a constraint on the maximum number of starts in a given time and hence on the minimum sump volume. This must be considered when deciding sump dimensions for optimum flow conditions.

Factors to consider are:

1. required discharge capacity and retention time
2. number of pumps required
3. discharge capacity of each pump
4. operating level of each pump
5. desirability of fixed or variable speed pumps
6. minimum cycle time for each pump

If variable speed pumps are used, the number of starts per hour ceases to be of significance and the minimum sump volume is governed by the control system for varying the pump speed. The water level can normally be held to within 500 mm of a set point for this type of control.

9.3 MINIMUM SUMP VOLUME FOR ONE DUTY SYSTEM (FIXED SPEED PUMP)

It may be shown that the pump starts most frequently when the flowrate into the sump is exactly half the pumping rate (Q litres/sec) and that the cycle time (t secs) is determined as follows:

$$t = \frac{240V}{Q}$$

where V = volume in litres

Hence for a frequency of 10 starts/hours (i.e. $t = 360$ secs), $V = 1\frac{1}{2} \times$ the pumping rate (Q litres/secs) or 6 minutes.

9.4 MINIMUM SUMP VOLUME FOR MULTI-PUMP SYSTEM (FIXED SPEED PUMPS)

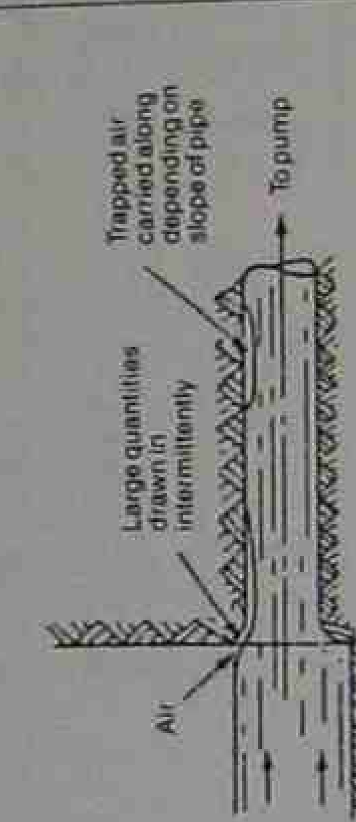
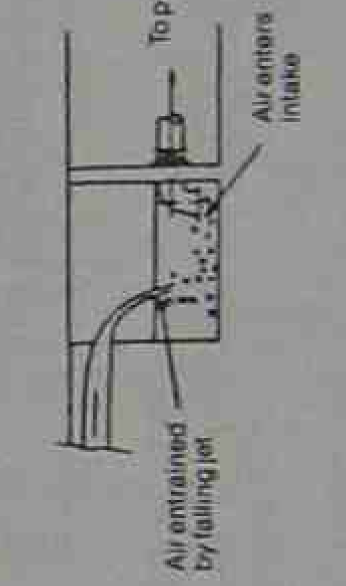
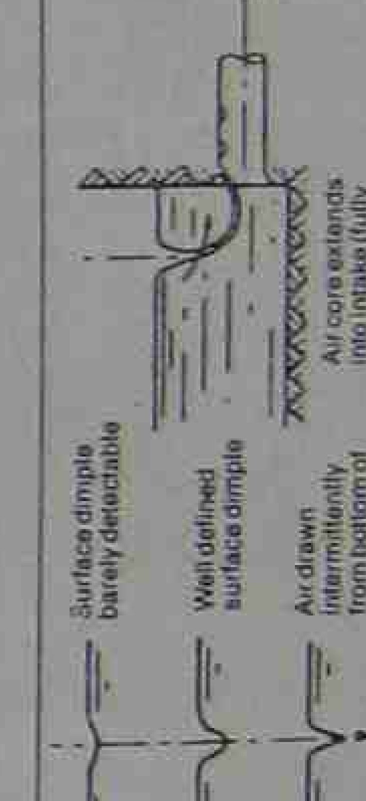
As with the case for a single unit, the volume between start and stop is related to the capacity of the pump started by

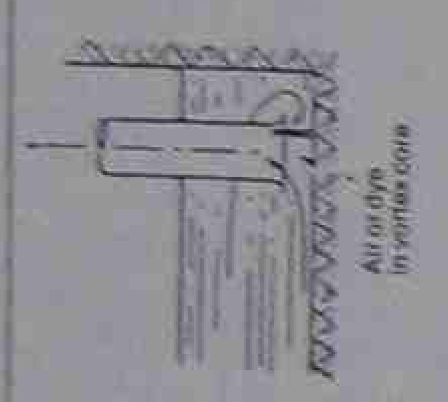
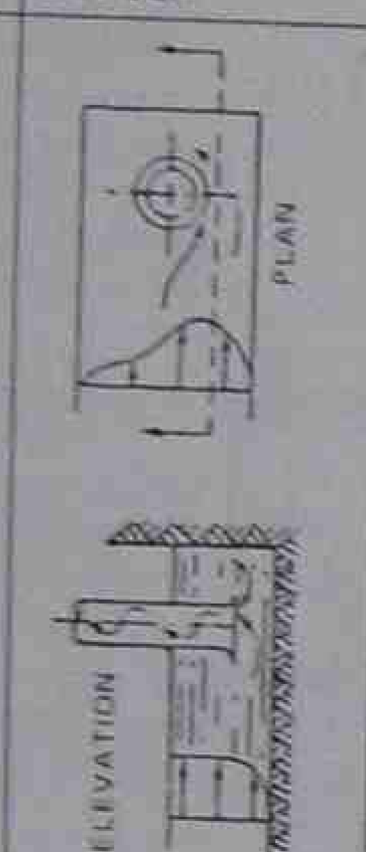
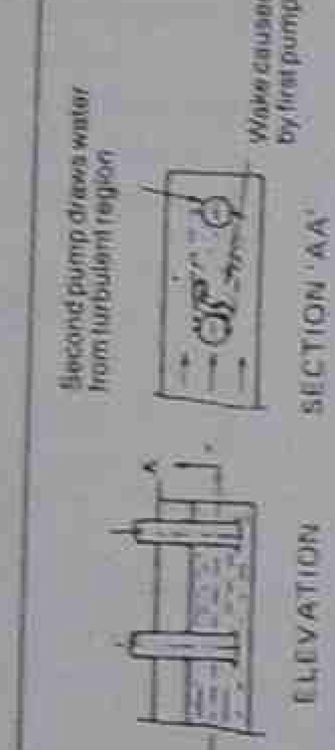
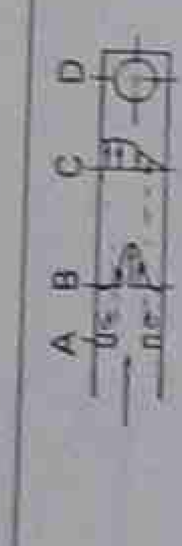
$$t = \frac{240V}{Q}$$

Additional capacity is required to allow a vertical distance of 150 mm between the start or stop levels of consecutive pumps.

For some applications such as sewage, which may become septic, there are restraints on the maximum volume of the pump.

Table 9.1

Flow Condition	Description and Cause	Sketch	Effects on pump and system
A. Air gulping	With the water level at, or only just above, the top of the intake there will be a tendency for air to be drawn into the intake either continuously or intermittently in the form of gulps. There may be distinct local depression in the water surface adjacent to the intake, described as "drawdown", which may accentuate the problem. High intake velocities will also cause increased air flow rates into the intake. The basic reason for this condition is that the sump water level is too low with insufficient cover above the intake.		In the extreme case, the pump may deprime and the delivery fall to zero. If the pump does continue to deliver water it will also discharge large quantities of air which may cause serious problems in the system. For example, the pump may be supplying cooling water to a condenser where air would cause overheating of the tubes. Large quantities of air passing through a pump impeller will cause uneven loading, resulting in vibration, rough running, and probably damage to the bearings in a fairly short time. Although there is not much published data on changes in performance due to air entrainment, it is known that air in quite small quantities leads to a reduction in discharge and loss in efficiency. (In a typical case, a centrifugal pump with 3% free air showed a drop in efficiency of 15%.) Actual figures depend very much on the type of pump, axial pumps being more sensitive to entrained air than centrifugual pumps. Normal design practice should be to exclude all air and this implies that there should be no air-entraining vortices at the intake and that there should be no air entrained in the approach flow to the intake due to other causes.
B. Aerated	Air bubbles may exist in the water of a sump for a variety of reasons, but they are usually due to a falling water jet as the water enters the sump from a weir or culvert that is above the water level in the sump. If there is insufficient time for them to rise to the water surface before they are carried close to the intake section, they will enter the intake and may be carried into the pump. It is best to avoid high level entry into the sump but if this is not possible, then one should provide sufficient separation of the source of aeration from the intake to allow the air bubbles to rise to the surface of the water in the sump. Approximate calculations based on the mean horizontal velocity in the sump, the minimum rise velocity of the air bubbles, and the water depth in sump, can be used to determine the separation distance. A typical minimum figure for the rise velocity is 0.2 m/s, based on air bubbles of between 2 to 5 mm dia. Obviously, larger bubbles will rise faster and so reach the water surface before the smaller bubbles. (However, there is an important class of intake used for sewerage where this procedure cannot always be used and so some air entrainment into the pump may have to be accepted.)		Similar to A above, the severity depending on the quantity of air and the pump type.
C. Air entraining vortices	The extreme case of a steady vortex with an air core between the water surface and the intake feeds air direct into the pump. In less severe cases, the air may enter only intermittently with the vortex being unsteady and less developed. Even if no air is drawn in, the swirling flow associated with the surface can cause problems. Asymmetry or pre-rotation in the flow approaching an intake, coupled with stagnant regions of water above or near the intake, increase the chances of air-entraining vortices forming.		Similar to A above, the severity depending on the quantity of air and the pump type. There are also effects similar to D and E below, the severity depending on the vortex strength and the pump type.

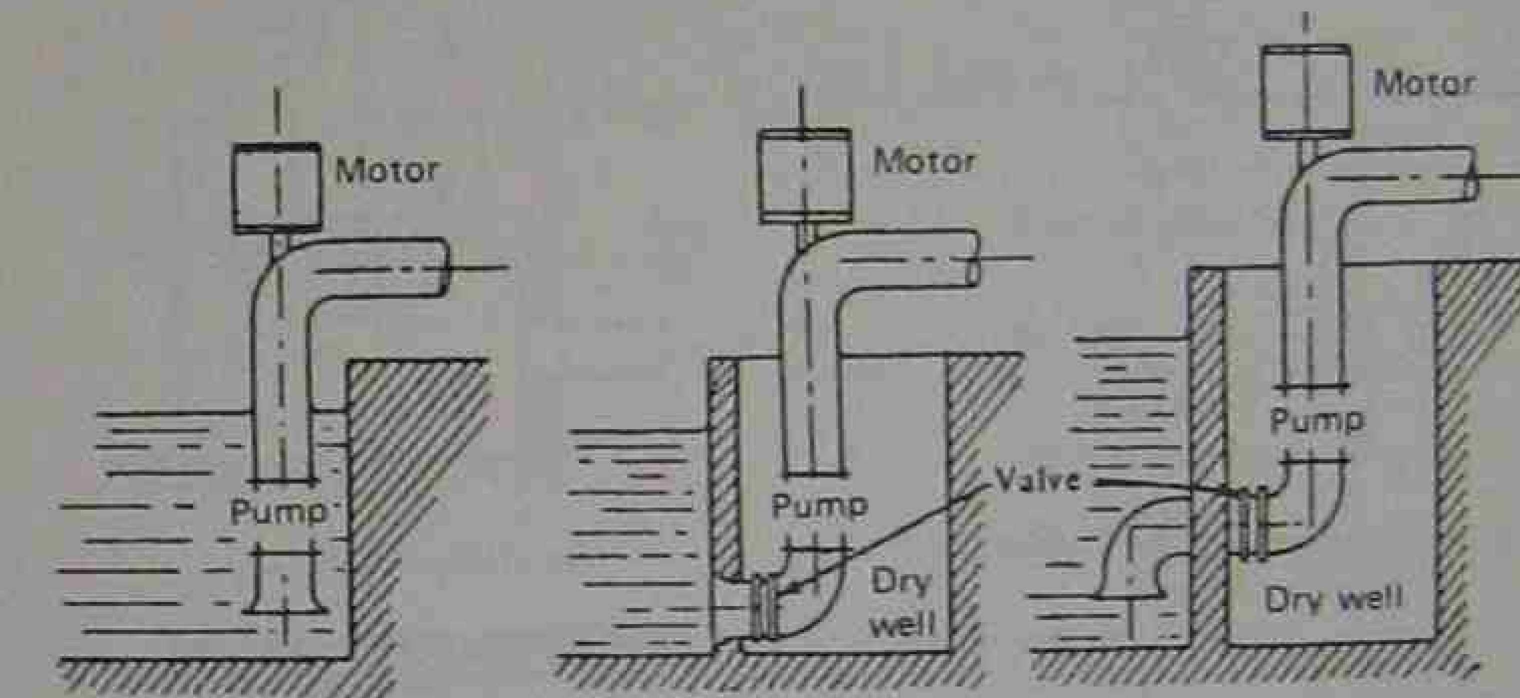
Flow Condition	Description and Cause	Sketch	Effects on pump and system
D. Submerged vortices	These vortices originate from the side walls and floor of the sump rather than the water surface. In models, the vortex core can be seen by injecting dye into the water near where a vortex is anticipated. Sometimes the vortex may be visible as small air bubbles in the water which are centrifuged into the core. There is also the possibility that air in solution can be released by the very low pressure in the rapidly rotating core. There is not a great deal known about the origin of submerged vortices except that they probably stem from swirl in the approach flow and vorticity in the boundary layers on the sump floor and sides. The presence of a submerged vortex core indicates that there is appreciable swirl as well.		Rapid changes in local pressure on the pump impeller as a vortex core is ingested can cause severe vibration and cavitation. Axial pumps are generally more susceptible to damage from vortices than other pump types.
E. Swirling flow	Swirling flow is usually caused by large-scale rotation in the bulk of the fluid in the sump which is then amplified as the flow converges into the intake. In many cases, the swirl may be accompanied by a rapidly rotating inner core even if there is no surface vortex visible.		Severe swirling flow is indentical to D above, but there is an additional effect when the swirl is on a larger and less intense scale but is centred on the pump axis. In this latter case, there can be a gain or reduction in the pump performance depending on the relative directions of the swirl and the pump impeller. This effect is most noticeable in axial and mixed flow pumps.
F. Large-scale turbulance	This is where the eddy size is the same order of magnitude as the intake cross-section area. Unsteady flow patterns arising from obstructions in the sump or poor inlet conditions into the sump are a primary cause. Vortex shedding from pillars or other pumps are a common source of trouble.		Uneven blade loading, changing with time, will cause vibration and noise in the pump. This effect will be worse when the pump is very close to the intake section as in a vertically suspended wet-well installation. If there is a moderate length of conduit between the intake and the pump (say more than 10 diametres) or if there are several bends and changes of cross-section, the initial turbulence may not be so important in defining the flow conditions at the pump inlet flange. In this latter case, the flow conditions will be set by the geometry of the pipework leading to the pump inlet flange.
G. Distorted velocity profile	This can occur for a variety of reasons, but is generally caused by some form of uneven distribution of the flow into the sump. The velocity distribution across the flow in the sump then shows, for example, a higher value on one side than the other, or a higher value at the floor than the surface. This uneven distribution may be steady in time, but it can be a prime reason for swirl and vortex formation at the intake.	 Jet at 'B' caused by gate at 'A' attaches to one side wall only. This produces a distorted velocity profile 'C' at intake 'D'.	The main effect is to promote swirl and vortex formation at the intake section, with the results described in D and E above. However, effects similar to F above could arise with the pump near the intake section, where the non-uniform velocity distribution could lead to asymmetric loads on the pump impeller. (Although this Table discusses flow conditions in the sump, distorted velocity profiles may be caused by bends and changes in section of the pipework between the intake section and the pump, if the pump is not sited very close to the intake.)
H. Stagnations	Large areas of water above or behind an intake are almost stationary. The only movement within these stagnant areas is caused by the shear generated across the 'surface' separating them from the main flow into the intake. Some such stagnant regions are unavoidable, but their extent can be reduced by careful design and filling-in with solid material at the more obvious places.		The boundaries between stagnant regions and the main flow tend to be unstable and fluctuate in position. These regions promote unsteadiness in the main flow and increase the chances of air entraining vortices forming.

9.5 WET WELL AND DRY WELL PUMPING STATIONS

A pump can be installed in a wet or dry well. Fig. 9.5.1 illustrates a vertical suspended pump in a wet well and a similar pump in a dry well. In a dry well, the bellmouth may be direct in the wall or turned down through an angle of up to 90° as shown.

The wet well configuration has the advantage of simplicity and reduced civil costs, and is widely used. However, maintenance can be a problem as it involves de-watering the pump or lifting the pump unit out of the well. The wet well arrangement is most suited for intermittent duty such as pumping storm water in which the well will be dry for most of the time.

The dry well configuration is usually used when reliability is a prime requirement, since the pump is readily accessible at all times. The "turned down" bellmouth of Fig. 9.5.1(c) is a popular arrangement in sewage applications since it allows a lower water cover in the sump compared with a horizontal intake (Fig. 9.5.1(b)), and in general is less prone to vortex action at similar water levels.



(a) Wet well (b) Dry well horizontal flow intake (c) Dry well — turned-down intake

WET AND DRY WELL PUMP INSTALLATIONS

Fig. 9.5.1

9.6 PREFERRED SUMP AND INTAKE DESIGNS

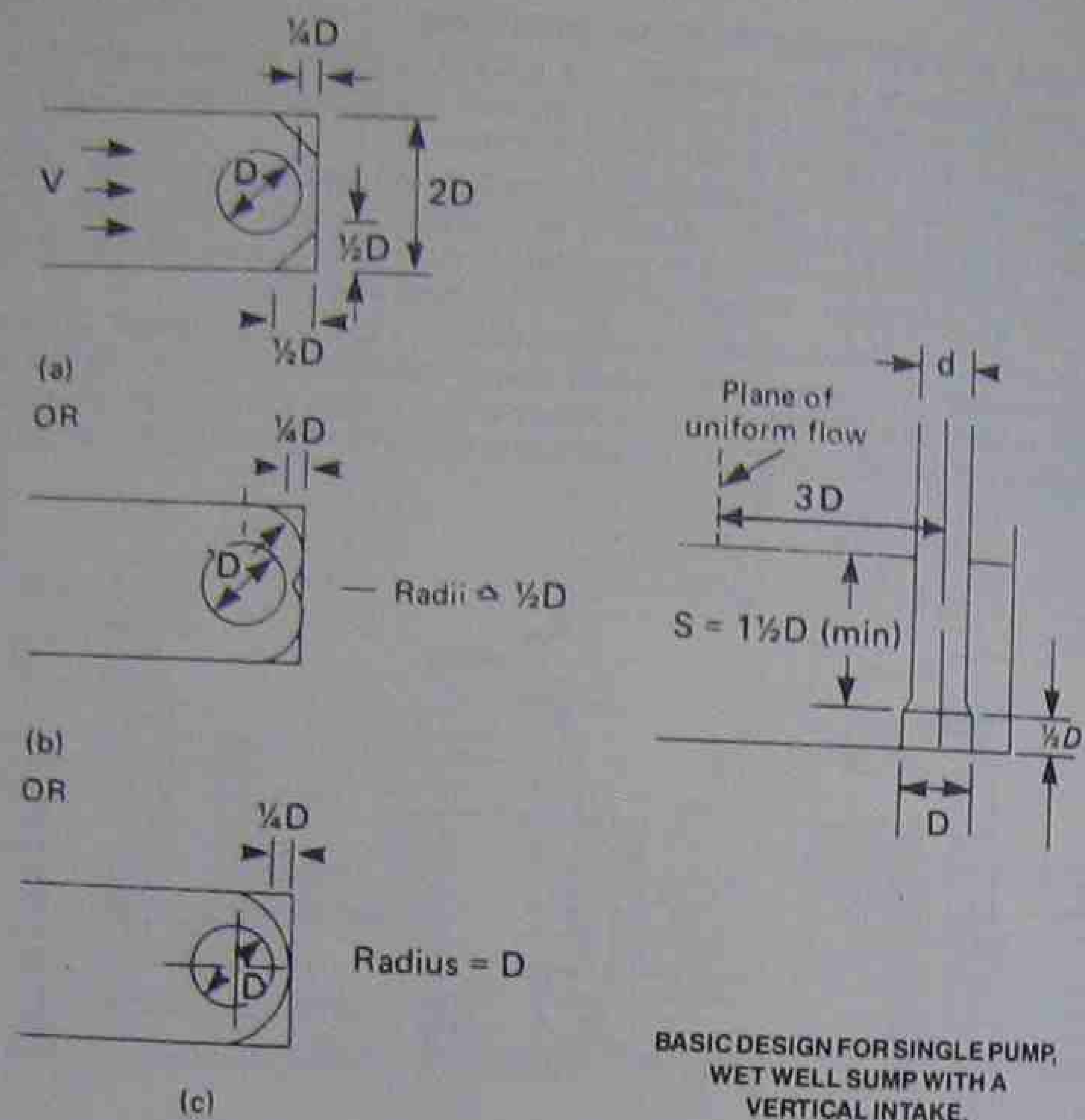
SINGLE PUMP SUMPS

Wet Well Arrangements

A basic design for a simple rectangular sump suitable for vertical spindle axial or bowl type pumps is shown in Fig. 9.6.1.

The dimensions are given in terms of the bellmouth diameter D, since most manufacturers use similar ratios of bellmouth diameter to pump inlet diameter. Typically this ratio D/d is between 1.5 and 1.8. Tolerances on these recommended dimensions cannot be defined, but they are not critical within ±20%. The distance X = ¼D, between the bellmouth and the back wall, may have to be increased for medium size pumps (say 3000 l/s to 7000 l/s) to provide access for maintenance.

It may be convenient to shape the end wall in the form of a circular arc rather than to use corner fillets. As long as the principle of about ¼D minimum space between the bellmouth and the wall is maintained, the design should be satisfactory. Thus, in Fig. 9.6.1, (b) and (c) are alternatives to the standard shape shown in (a). These designs are applicable if there is uniform steady flow



BASIC DESIGN FOR SINGLE PUMP, WET WELL SUMP WITH A VERTICAL INTAKE.

Fig. 9.6.1

through the channel cross-section upstream of the bellmouth. The distance of uniform flow from the bellmouth centre line is about 3D.

Dry Well Arrangement

The basic horizontal arrangement is used if the intake is in the end wall of the sump. For a horizontal intake (Fig. 9.6.2a), the corner fillets are omitted.

The turned down bellmouth shown in Fig. 9.6.2b is the more common arrangement.

The bellmouth may be omitted over part of its circumference, thus allowing the intake to be set with its centre line ½D above the sump floor.

Mean Velocities

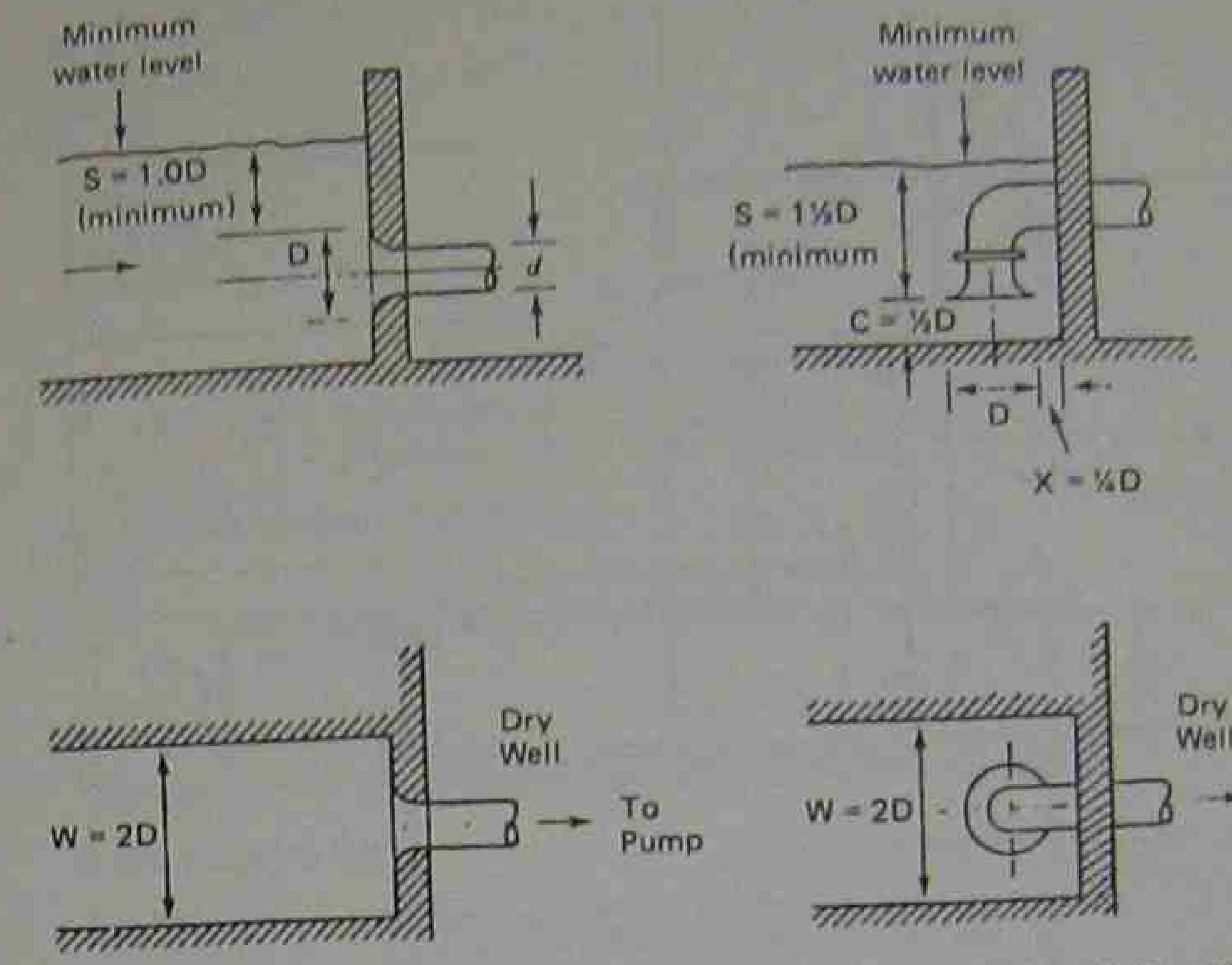
Assuming a typical ratio of bellmouth diameter to pump inlet diameter of 1.75 and a pump inlet mean velocity of 5 m/s (which is typical for a modern axial flow pump):

MULTIPLE PUMP SUMPS

Wet Well Arrangement

The basic designs shown in Fig. 9.6.3 provide two alternative ways of installing three pumps in a sump where uniform steady flow occurs just upstream of the intakes. If the approach flow is less uniform than the ideal case, Fig. 9.6.3b is preferable to Fig. 9.6.3a.

The sump designs given in Fig. 9.6.3 illustrate three pumps. They can be made suitable for other



(a) Horizontal intake (b) Turned-down bellmouth

BASIC ARRANGEMENT FOR SINGLE PUMP, DRY WELL SUMP.

Fig. 9.6.2

configurations by increasing or reducing the width and maintaining similar inter-pump spacing. Great care is required in applying the open sump to a complete system.

Dry Well Arrangement

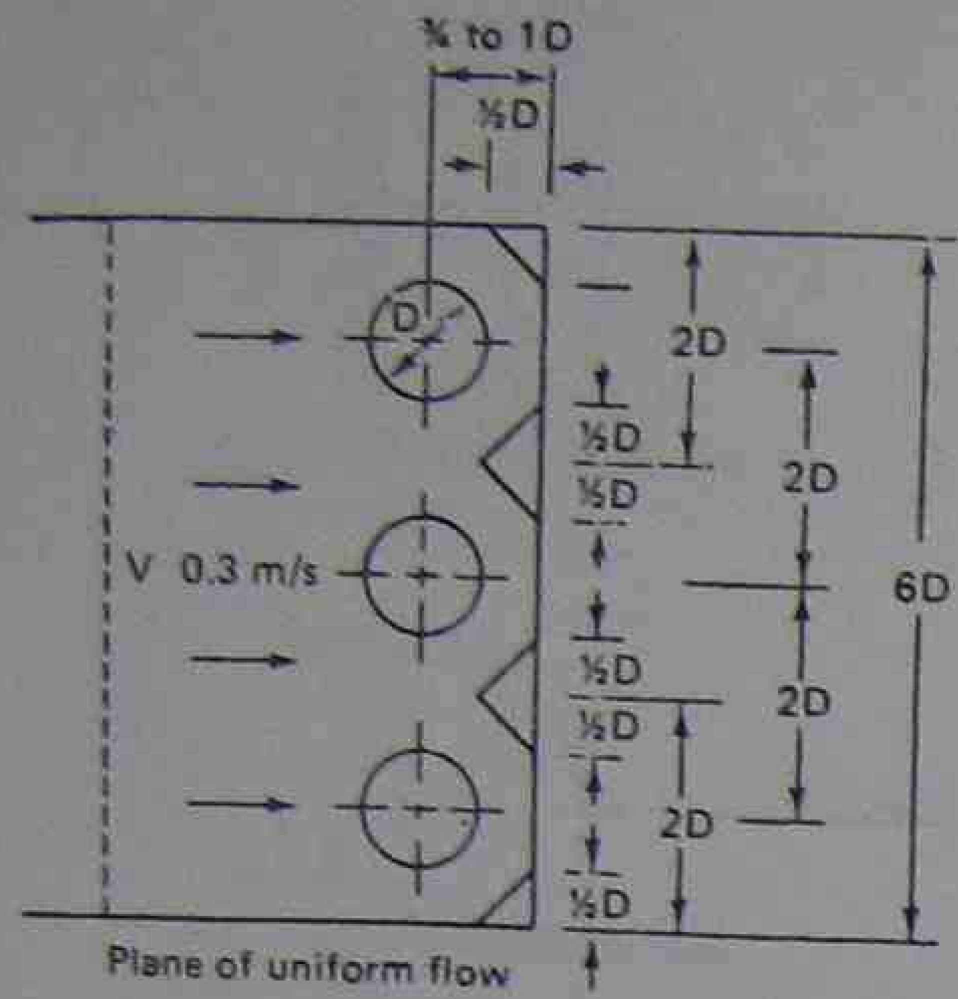
The standard wet well designs shown in Fig. 9.6.3 can be easily adapted for dry well installation of the pumps. The plan dimensions should be the same whether the intake is a turned down bellmouth or a horizontal intake through the end well. The corner fillets or radii need not be used if the intakes are horizontal.

For a given submergence of the intake bellmouth, a horizontal intake in the end wall produces comparatively slightly better flow conditions than a vertical intake. However, a vertical intake has a lower minimum submergence requirement than a horizontal intake, and this explains the common use of a turned-down bellmouth for dry well installations.

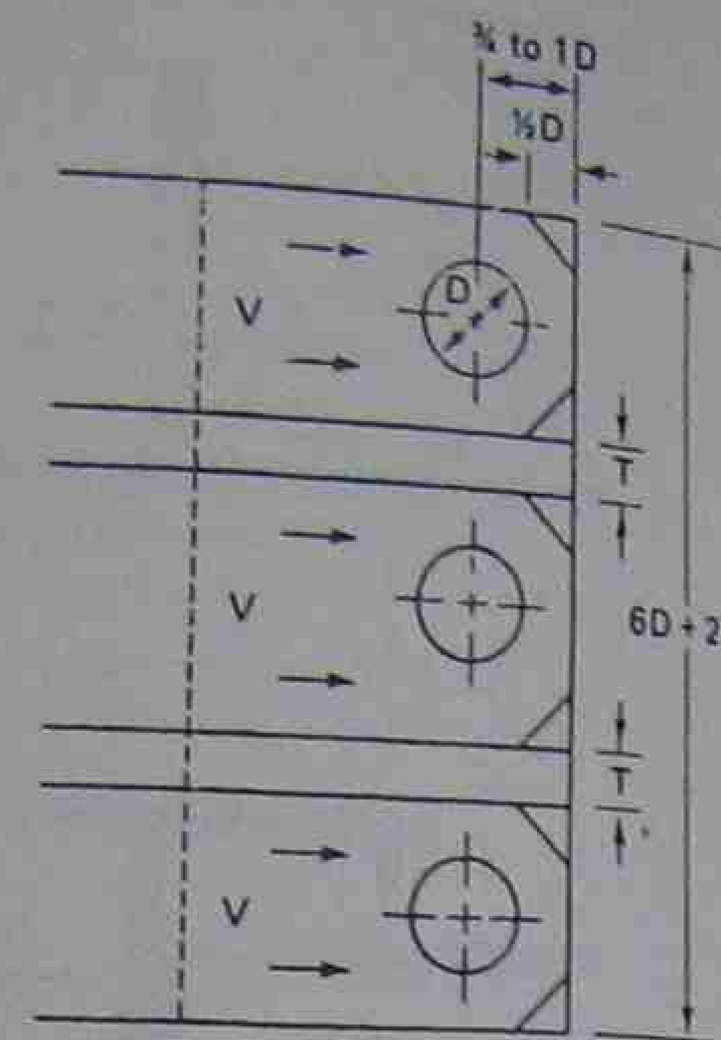
Approaches of the Sump

Examples of multiple sump design which include the approach to the sump, are shown in Fig. 9.6.4. These are not basic designs, but they indicate ways of incorporating the open sump design and the unitised design into an overall system. The problem of providing steady, uniform flow to a multiple sump is compounded because, in the majority of applications, the intakes must function satisfactorily with all possible combinations of pumps in operation.

The flow pattern in an open sump varies appreciably as it is dependent on the number and combination of pumps in operation. In general, a unitised design is preferable, since the approach length (L) of each "unit" can be a conservative value based on the single pump design.



(a) Open sump



(b) Unitised sump

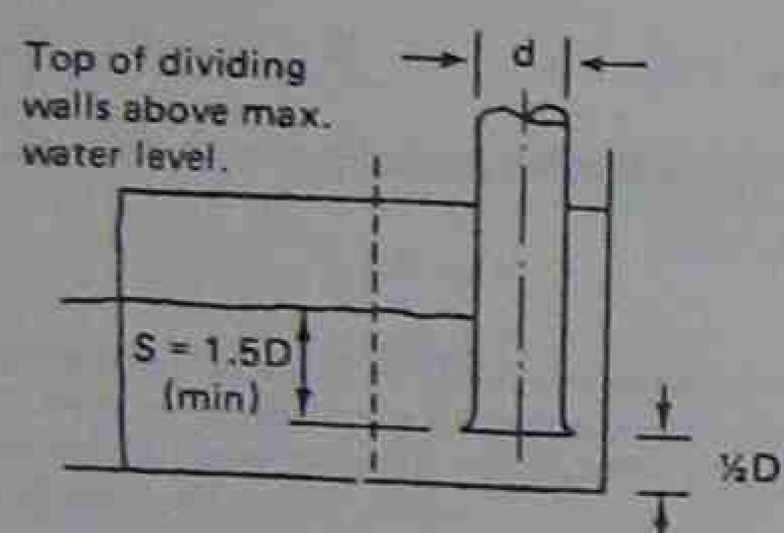
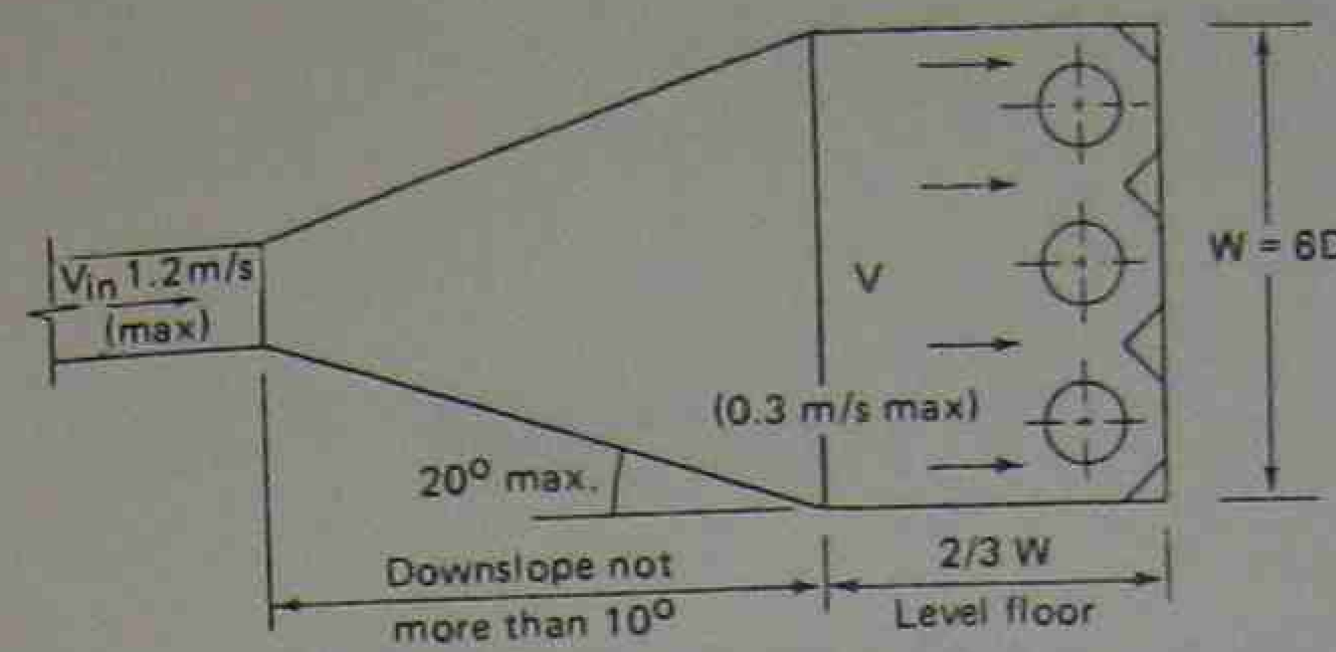
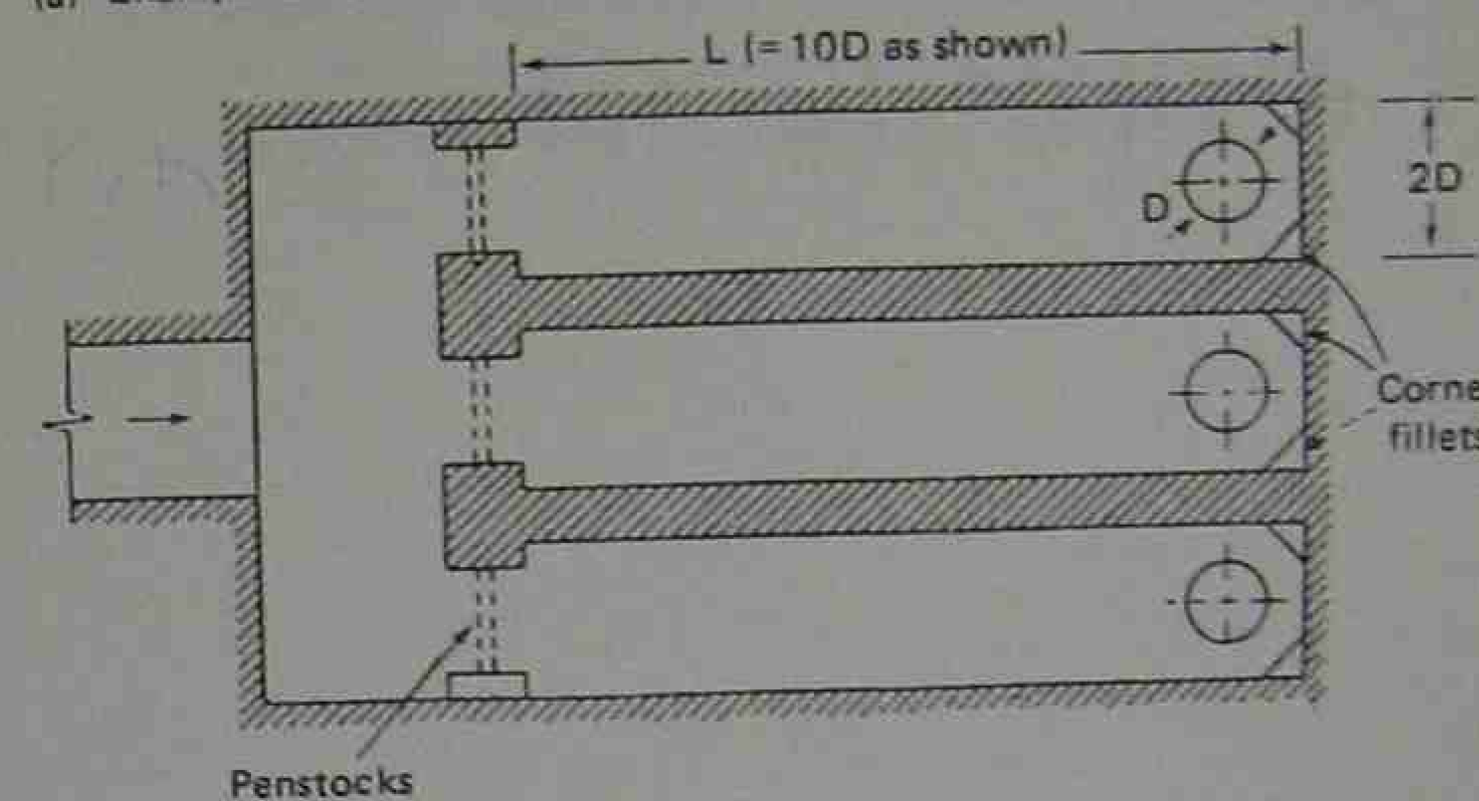


Fig. 9.6.3 Basic sump designs for multiple pumps, wet well arrangement.

The open sump design may well require baffles, splitters or a grid in order to distribute the flow evenly to all the pumps. Since the need for these devices and the siting of them depends on the specific design under consideration, no basic design details can be given with larger pumps. The best way to determine the details is with the aid of a scaled hydraulic model.



(a) Example of open sump design



(b) Example of unitised sump design.

EXAMPLE OF MULTIPLE PUMP SUMPS, INCLUDING THE APPROACH WORKS TO THE SUMP (PLAN VIEWS).

Fig. 9.6.4

Section 10

INSTALLATION AND COMMISSIONING

Section 10—INSTALLATION AND COMMISSIONING

10.1 FOUNDATIONS

1. The pumpset should be supported on a firm foundation preferably a rectangular concrete block of a mass equivalent to 5 times the weight of the complete unit.
2. When pouring the foundation block, holes approximately four times the diameter of the foundation bolts must be left in the block to allow the bolts to be finally positioned before grouting in.
3. Grout allowance—set the top surface of the foundation 20 to 40 mm below the bottom of the baseplate level and provide a rough surface to afford a key effect for the grout. Do not put the baseplate on the foundation until it is of adequate strength. Prior to grouting, the foundation should be clean and wet which will prevent excessive moisture loss from the grout interface.

10.2 ERECTION

1. When pumps and their drivers are mounted on baseplates in the manufacturers works they are correctly aligned, i.e. the couplings are concentric and the faces are parallel in both axes. It is absolutely essential that this alignment be repeated when installing the unit at site. Steel wedges or packers and shims must be inserted between the top of the foundation block and the bottom of the baseplate adjacent to and on each side of the holding-down bolts. Using various thickness packers and shims, the correct coupling alignment will be achieved. It is necessary however to maintain the horizontal centre line or the horizontal flange face (as applicable) at the correct R.L.

   If the baseplate is installed separately without the pump or driver, an engineers level must be used to ensure both driver and pump mounting pads are level in both directions. Again correct R.L.'s must be maintained.

   If the unit has been subjected to maltreatment during transport, it may be necessary to adjust the position of the driver or pump or both on the baseplate to achieve correct coupling alignment.
2. If the unit is to be installed between fixed limits of suction and discharge piping, it is advisable to temporarily connect the piping to the pump before the unit is grouted in. The pumpset can then, if necessary, be moved slightly to suit the layout of the piping.
3. Grout the foundation bolts and fill each pocket with grout to within 25 mm of the top, thereby providing a key for the final grouting of the baseplate. The bedplate must be left in position until the grout has set hard (normally 72 hours).
4. When the foundation bolts are set check the levels of the machined pads on the baseplate with a straight edge and spirit level. A degree of level of 0.1 mm per 1000 mm length of bedplate is acceptable. To obtain this level, it may be necessary to adjust the packing under the bedplate and tighten certain of the foundation bolt nuts to pull the bedplate down at certain points. It is for this reason that foundation bolts are grouted in first. It should be remembered that when an adjustment is made the level in another direction will change.
5. When levelling and alignment is completed and all foundation bolts are pulled down tightly, the preparation for grouting the bedplate can take place.

   Shuttering should be erected around the bedplate to a height of 76 mm or more above the lower flange of the channel to produce a slight head on the grout.

   Grout should comprise one part Portland Cement and two parts sand (no aggregate).

   Mix with as little water as possible to prevent shrinkage. This should be well rammed under the bedplate using a wooden stick.

   Alternatively a commercial non shrink grouting material can be bought, mixed and used strictly to the manufacturer's directions.

   Note: It is important that the grout fills all the cavities between the lower flanges and the foundation and must not shrink after curing.

6. When the grout is set (normally about 48 hours) the shuttering should be removed and a smooth finish given to the grout and foundation surfaces. Uniformly re-tighten all foundation bolts and re-check the unit alignment.
7. The next step is to align the driver with the pump if the driver has been supplied separately. It is important to note that the flexible coupling permits axial movement of both pump and driver shafts and also compensates for thermal expansion. It is to be emphasised that, since pin and bush type couplings are designed to reduce the transmission of shock to the bearings, etc. and do not compensate for misalignment, they must be aligned with the same accuracy as rigid couplings.

   If the unit is installed outdoors, care should be taken to ensure that extremes of ambient temperatures do not adversely affect the alignment.

10.3 COUPLING ALIGNMENT

Lining up pump and driver

1. Mount the pump on the bedplate, ensuring that the pump feet and the bedplate pads are clean and free from burrs. Locate the pump with dowel pins and holding-down bolts. Tighten the holding-down bolts.
2. The faces and periphery of the half couplings should be thoroughly cleaned of all rough or ragged edges as these would produce false alignment readings.
3. Place the driver and/or gear box on the bedplate, having checked that the feet are clean and free from burrs. Insert the holding-down bolts—do not tighten up. Check the height of the pump and driver and/or gear box couplings. Where necessary insert shims (which must be free from oil and burrs) under the driver and/or gear box feet to obtain initial coupling alignment.

Final coupling

1. When running, the half couplings should be separated by a gap dependent upon the type being used and this distance should be confirmed from the general arrangement drawing.

   Note: If the motor is fitted with sleeve type bearings the coupling gap must be set with rotor in its magnetic centre. Most motors have the magnetic centre marked; for further details consult the motor instruction manual.
2. The following equipment is necessary to check coupling alignment.
   (a) Gap gauge and set of feeler gauges
   (b) Dial gauges, or a straight edge (steel rule or similar) and set of feeler gauges.

Major alignment errors

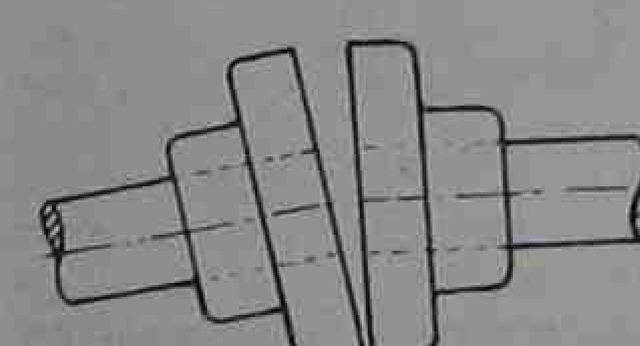


Fig. 10.3.1

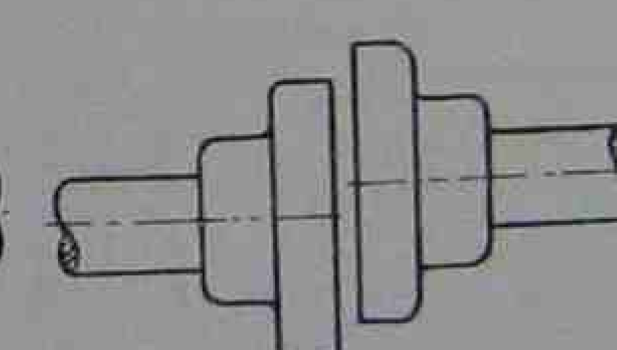


Fig. 10.3.2

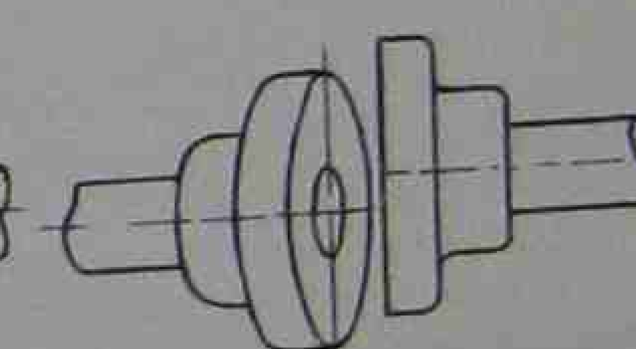


Fig. 10.3.3

1. Errors of Angularity (Fig. 10.3.1): The axes of the shafts are not co-linear and intersect in the plane of the coupling faces.
2. Errors of Eccentricity (Fig. 10.3.2): The shafts are parallel but not concentric.
3. Combinations of 1 and 2 (Fig. 10.3.3): Under these circumstances the axes need not be parallel and they may or may not intersect. By separating the errors of angularity and eccentricity and measuring them independently, the extent of the combination of errors can be determined and corrected. Before lining up, the axial movement of both driving and driven shafts should be

measured. Appropriate allowance can then be made for the clearance between the coupling faces. The lining up procedure involves the separate measurement of the angular and eccentric errors in both vertical and horizontal planes and correction where necessary.

Angular alignment

Errors of angularity should be corrected before attempting to eliminate eccentricity.

1. Remove coupling bolts.
2. Place chalk marks on both coupling halves at points A and B (Fig. 10.3.4), then rotate both coupling halves together and measure the gap at points 1, 2, 3, and 4.

   Alternatively:

   Clamp two dial gauges to the coupling halves as illustrated in Fig. 10.3.5 and zero the gauges.

   Rotate both shafts through 180° and note the gauge readings. If both gauge readings are the same (not necessarily zero), the angular alignment in the vertical plane is correct.

   Rotate both shafts through a further 90° and note the gauge readings. If the readings are the same (not necessarily zero), the angular alignment in the horizontal plane is correct.

   The amount of correction necessary at each point can thus be determined. Where possible, couplings should be aligned to within ± 0.025 mm i.e. 0.050 mm full indicator movement on dial gauge. During this operation, ensure that both shafts are pushed against their axial locations.

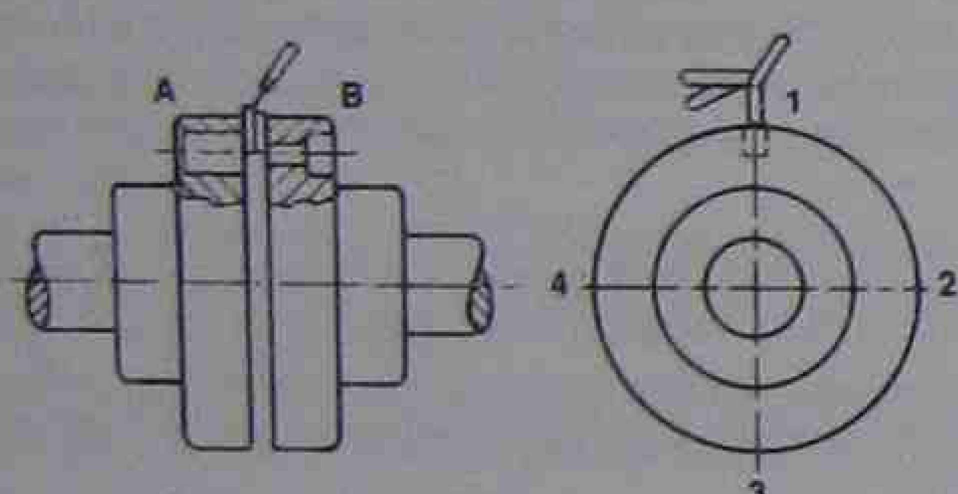


Fig. 10.3.4

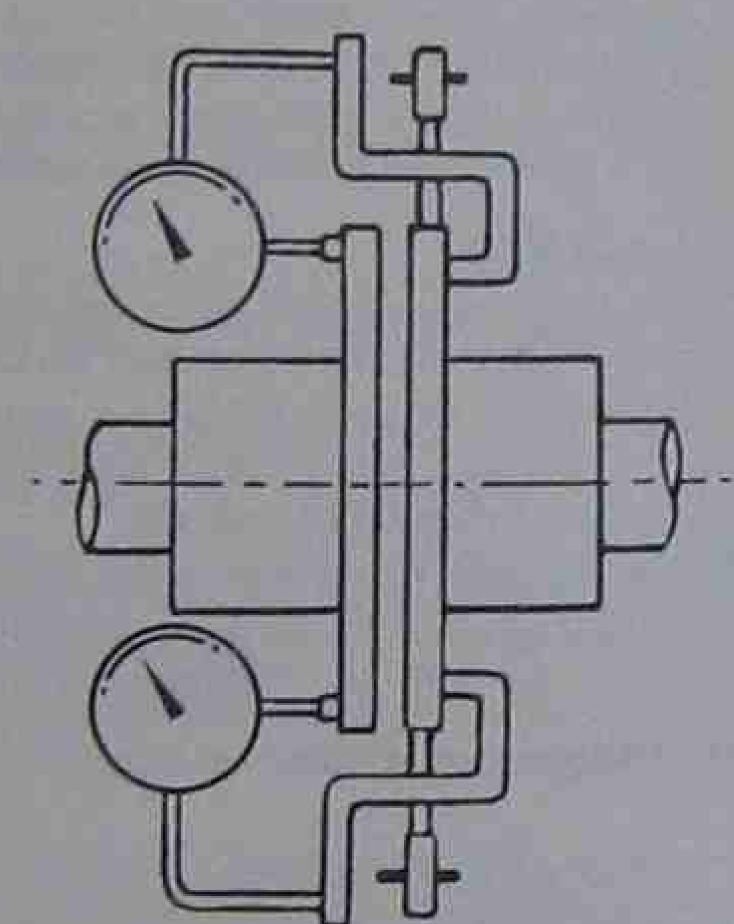


Fig. 10.3.5

Parallel alignment

If this error is in the vertical plane, adjust the shims under the driving unit. If the error is in the horizontal plane it is usually easy to eliminate by moving the driver horizontally until the correct setting is obtained. Concentricity can be checked with a straight edge or dial gauge as shown in Fig. 10.3.6 and 10.3.7. Again, chalk marks should be made on both halves. Rotate the complete coupling and take readings at points 1, 2, 3 and 4. Slight variations in diameter of the two halves can be checked by using a straight edge and a feeler gauge equal to half the difference in concentricity.

Misalignment should not exceed ± 0.025 mm i.e. 0.050 mm full indicator movement on the dial gauge.

If the dial indicator method is used as in Fig. 10.3.7, any sag which may occur in the aligning equipment must be measured and compensated for in the coupling alignment.

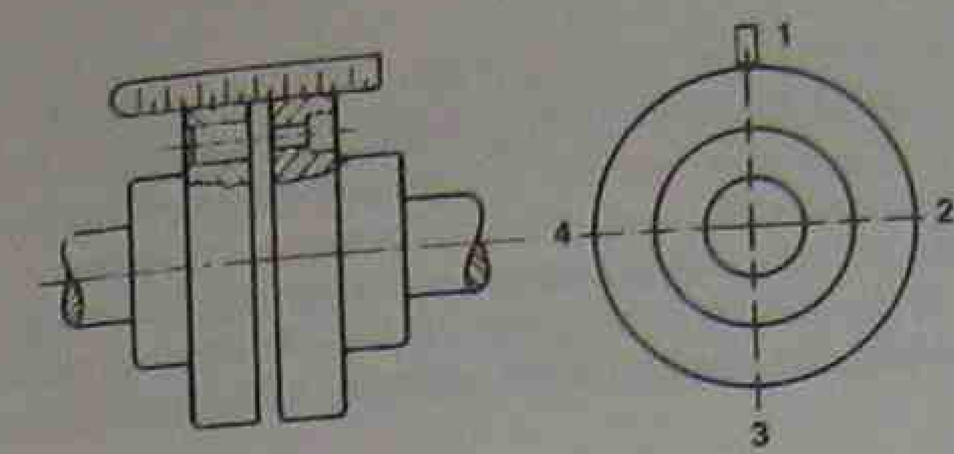


Fig. 10.3.6

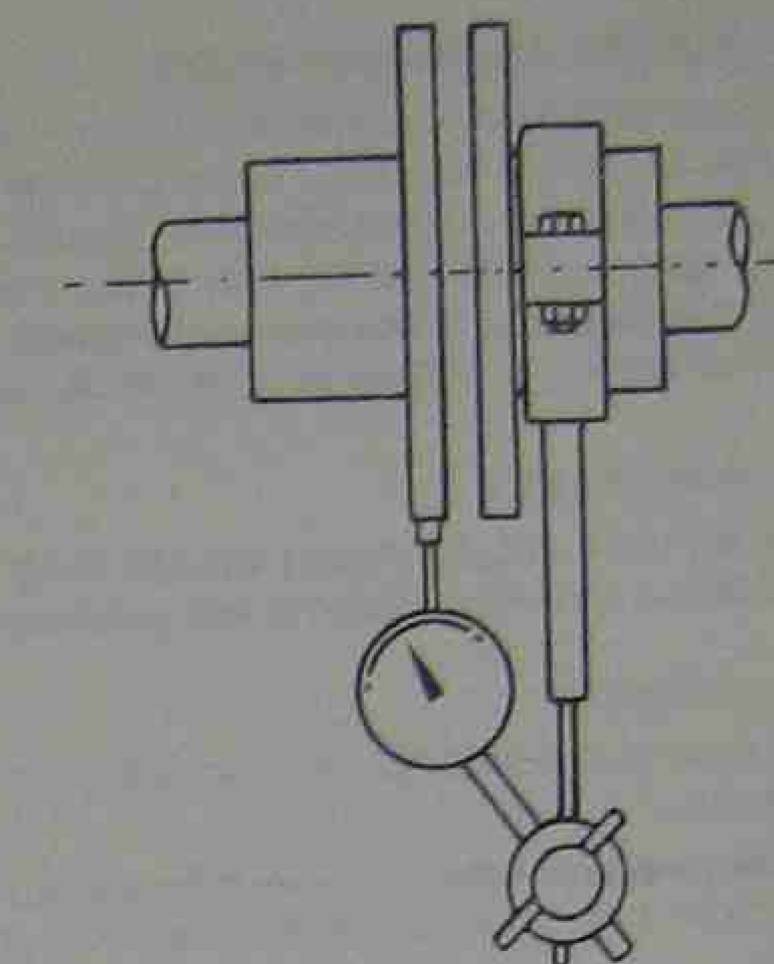


Fig. 10.3.7

General

Once final alignment is complete, the driver and/or gear unit should be tightened down on the bedplate and the alignment rechecked.

Check the coupling alignment after the pumpset has been running for a period of 4 to 6 weeks (or until settlement of foundation is completed).

Some couplings, notably double engagement gear type, laminated metal element spacer type and rubber type couplings, can operate satisfactorily with a relatively greater degree of misalignment. The additional flexibility of these couplings is intended to compensate for moderate misalignment which may develop during operation, and does NOT preclude the necessity for accurate initial alignment. Consult the manufacturer for the alignment procedure of these couplings.

Belt driven pumps

It is important that the driving and driven belt pulleys be in the same plane as, and parallel to, each other. See Fig. 10.3.8.

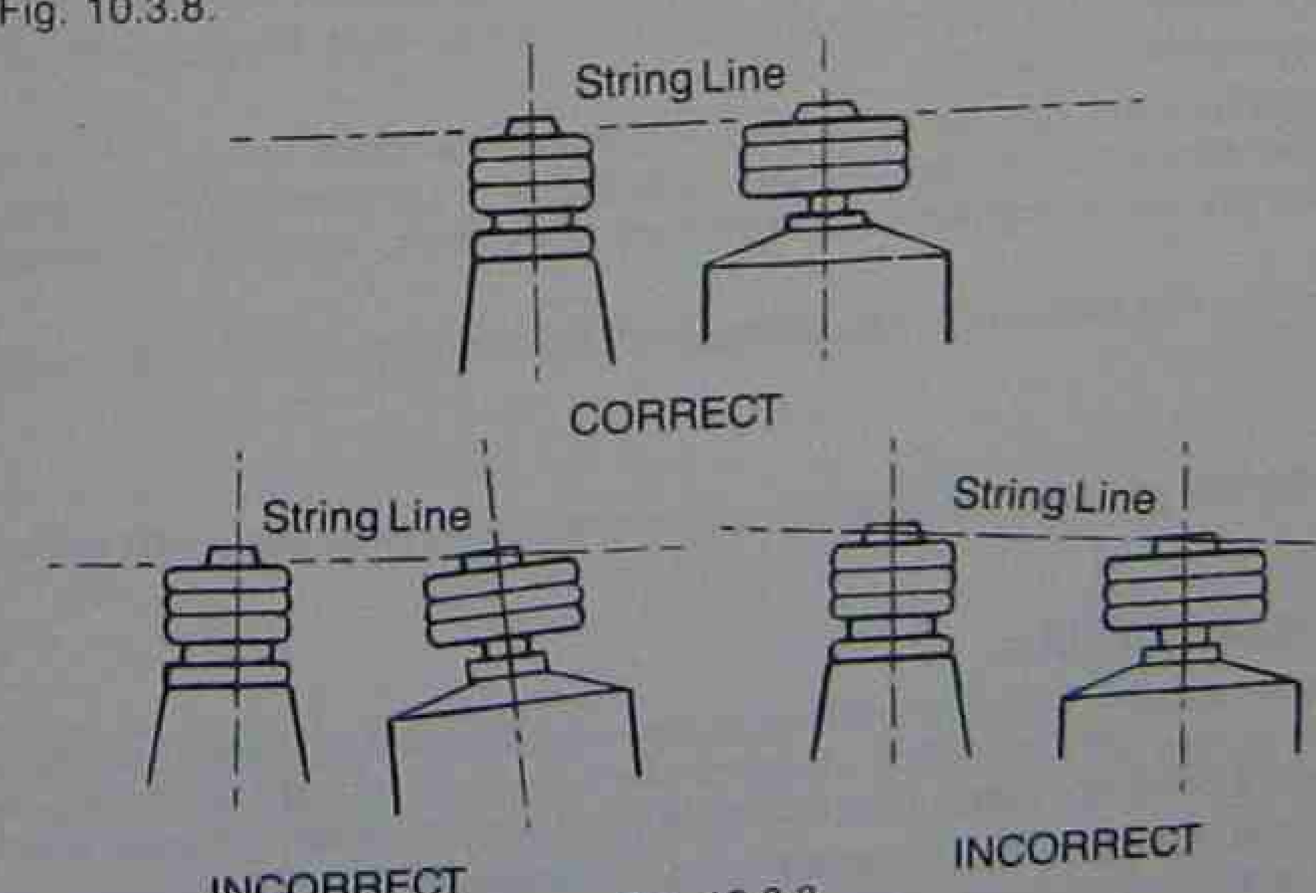


Fig. 10.3.8

10.4 PIPEWORK INSTALLATION

It is essential that no load be imposed on the pump branches as this is liable to disturb shaft and bearing alignment resulting in excessive wear and possibly seizure. The pipework should therefore be independently supported and anchored against pipe thrusts. When erecting the pipework, the pump flanges should be the final closure joints. Check the coupling alignment before and after the pump flange bolts have been tightened. Any alteration in alignment is indicative of excessive pipework loading on the pump branches. If this occurs the pipes must be reworked to remove the load.

10.5 OPERATION

Pumping units should always be operated in accordance with the manufacturers instructions. However, if this information is not available the following general rules should be followed.

Before start-up:

1. Ensure all bearings are charged with the correct lubricant and that the stuffing boxes are suitably packed.
2. Centrifugal pumps must be primed prior to start-up. This involves the removal of air, gas or vapour from the suction pipework and pump casing.
   2.1 If there is a positive suction on the pump, flood the pump by opening the suction valve and allowing the air to be released through the air cock at the highest point on the pump casing.
   2.2 If a suction lift is involved, it is normal practice to fit a foot valve and strainer to the suction pipe inlet. The suction pipe should then be filled from the highest point on the pump casing using an external water supply. This procedure is only necessary when starting up for the first time (or after overhaul or long periods of shut down when leakage of the foot valve may occur).
3. Open external water service valves to bearings and/or stuffing boxes.
4. Ensure the suction valve is open, and for centrifugal pumps that the discharge valve is closed. For axial flow and positive displacement pumps, both valves should normally be open on startup.

Running

When starting the motor for the first time, disconnect the coupling drive and run the motor on no load checking that motor operates satisfactorily and that direction of rotation is correct. Rewire the motor if incorrect. Replace the drive and start the pump against a closed discharge valve. When the unit reaches maximum speed, steadily open the discharge valve (if not already open as in the case of an axial flow pump). If the discharge pressure does not build up, the pump should be stopped and the priming operation repeated.

It should be noted that prolonged operation with the discharge valve closed, or throttled to give flows less than the minimum specified by the manufacturer will damage the pump. Prolonged operation at pressures and quantities vastly different to those for which it was designed is also detrimental.

If the pump output is to be reduced by throtttling, this must be done with the discharge valve and not the suction valve.

Stopping

Slowly close the discharge valve and shut down the driver. Close down external cooling and water services.

10.6 ROUTINE MAINTENANCE

Modern pumps when properly selected installed and given reasonable maintenance, should operate satisfactorily over a long period of time. It cannot be overemphasised that the manufacturer's instructions must be read thoroughly and completely followed to ensure trouble free operation. A high degree of cleanliness of the equipment and surrounding area should be maintained as this will assist in the detection of minor defects.

The main factors in determining if overhaul is required is a falling off in the pump discharge pressure and/or capacity to an unsatisfactory level, or a significant increase in power consumption in relation to quantity delivered.

Depending on operation and environmental conditions, together with a comparison of previous inspections, the frequency of inspection can be altered to maintain satisfactory operation of the plant to suit established operating routines. The checks and inspections carried out during the running-in period will often establish the frequency of future inspections.

Section 11

PUMPING DONT'S

Section 11—PUMPING DONT'S

1) Do not fail to read operating and installation manual before starting pump.
2) Do not fail to check that suction line and/or sump are clean before initial start up.
3) Do not run a pump without any liquid in it.,
4) Do not regulate the flow of a pump with a valve on the pump suction line.
5) Do not run a rotodynamic pump with the discharge valve shut for more than a few moments and a positive displacement pump without a relief valve or continuous by-pass.
6) Do not over tighten a pump packed gland, a slight leakage is desirable.
7) Do not have pump NPSHR greater than system NPSHA.
8) Do not start a pump without checking alignment of pump and motor.
9) Do not put a baseplate or assembled unit on a foundation until it has firmly set.
10) Do not attempt a suction lift greater than the manufacturer's recommendations.
11) Do not make or install suction pipes such that they will accumulate air.
12) Do not install a pump in an application not recommended by the manufacturer.
13) Do not run a pump faster than manufacturer's recommendations.

Section 12

PUMP TESTING

Section 12—PUMP TESTING

Pumps are tested for a variety of reasons, not all of which are solely to meet the customers test requirements.

The final user's interest in the machine is generally confined to the following:—

(a) The total head at the duty flow.

(b) The power consumption at the duty flow which directly indicates the pump efficiency.

(c) The starting capability under simulated site conditions.

(d) The mechanical integrity of the machine.

The user may also be interested in:—

(e) The level of noise emitted by the machine.

(f) The suction performance, defined by NPSH.

(g) A functional check on auxiliary equipment.

The Australian Standard for pump testing is covered by AS2417-1980 parts 1, 2 (class C tests) & 3 (class B tests).

The allowable tolerances and limits of errors are clearly defined in the 3 separate parts of this standard.

Every measurement is inevitably subject to inaccuracies, even if the measuring procedure, the instruments used and the analysis directives fully comply with prevailing acceptance rules. When comparing the test results with the guaranteed characteristics, these inaccuracies should be given adequate consideration.

The rules for deciding whether a pump meets the guaranteed conditions of a contract are expressed in the appropriate test code. It will be observed that these rules vary greatly.

The following comments may be of assistance in appreciating the meaning of the 'ellipse analysis' as specified in some codes. This method is a little confusing compared to those codes in which a definite percentage variation is specified.

Regrettably, many users have assumed that the tolerances listed in say, Part 3 of AS2417, can be applied directly to the guaranteed point. That is to say that at the guaranteed flow the head should be within $\pm 2\%$ of that specified and, at the given head the flow should be within $\pm 4\%$. This being applied in a similar manner to most other codes. This is incorrect.

The tolerances include the maximum permissible limits of overall error and the constructional tolerances.

To make the correct analysis, the equation shown in fig. 12.1 must be evaluated. If the value is greater than or equal to unity then the guarantee is met.

What this really means is that the guaranteed duty point defined by Q_G and H_G on the solid H-Q curve shown, is the centre of an ellipse with the semi axes the limiting errors. Thus if any part of the test curve falls within the ellipse, the guarantee has been met.

The dotted curves show the maximum and minimum limits allowable.

The displayed formula gives a mathematical check on these limits without the need to draw an ellipse.

The guaranteed pump efficiency can be checked for the duty point as defined by the point of intersection if the H-Q curve with a straight line passing through the guaranteed duty point for H-Q and the zero of H & Q axes. The efficiency at this point shall be read on the Q-efficiency curve versus the corresponding abscissa. This is shown diagramitcially in fig. 12.2.

The efficiency at the point of intersection shall be at least 0.972 of that specified (Class 'B' Tests) or 0.95 of that specified (Class 'C' Tests).

To determine the complete hydraulic performance of a pump it is necessary to measure the following:—

(a) **Flow Rate:** This can be done using venturi flow metres, orifice plates, volumetric measuring tanks, weirs, electro magnetic meters or velocity measuring devices used in a known cross

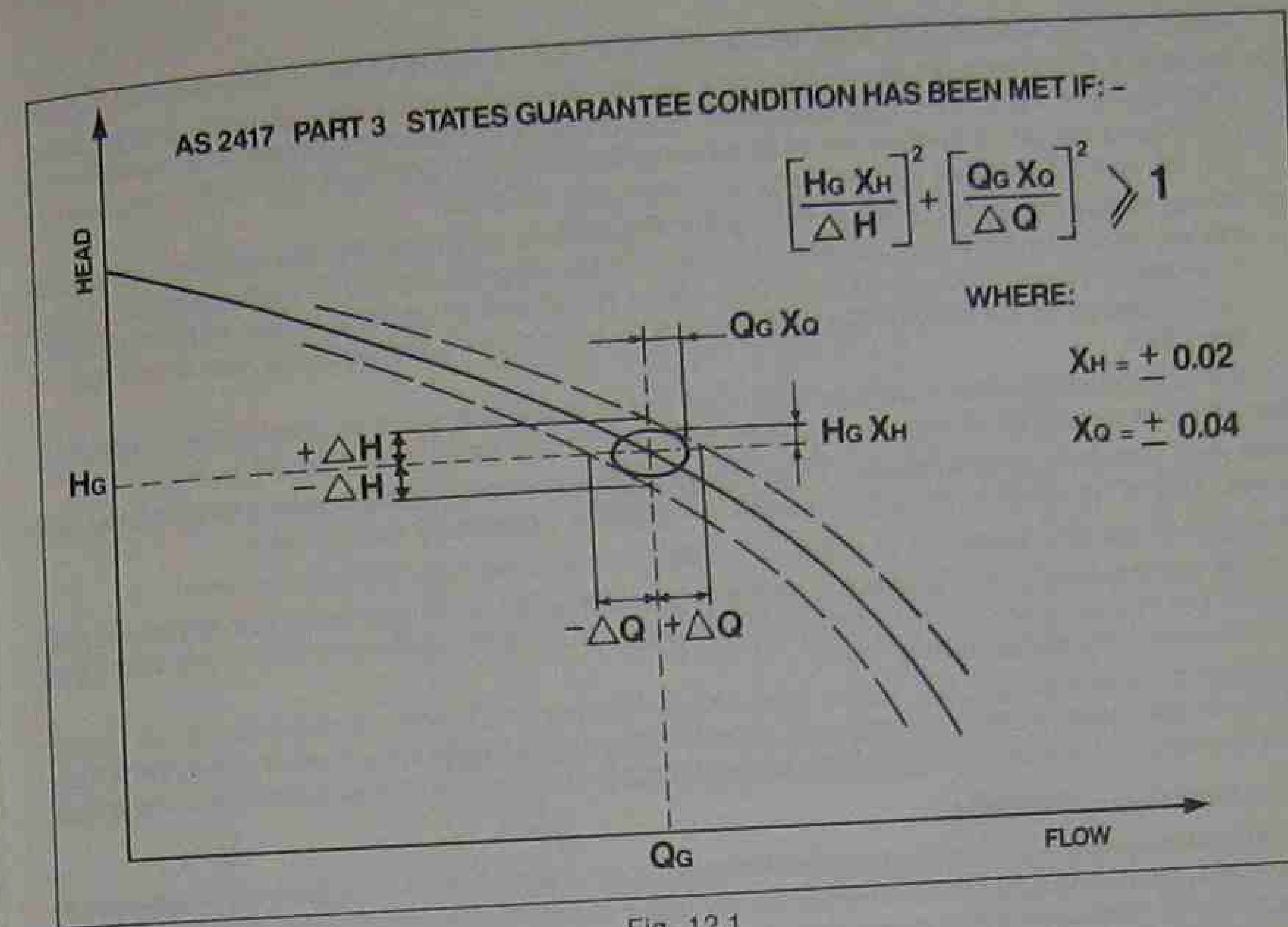


Fig. 12.1

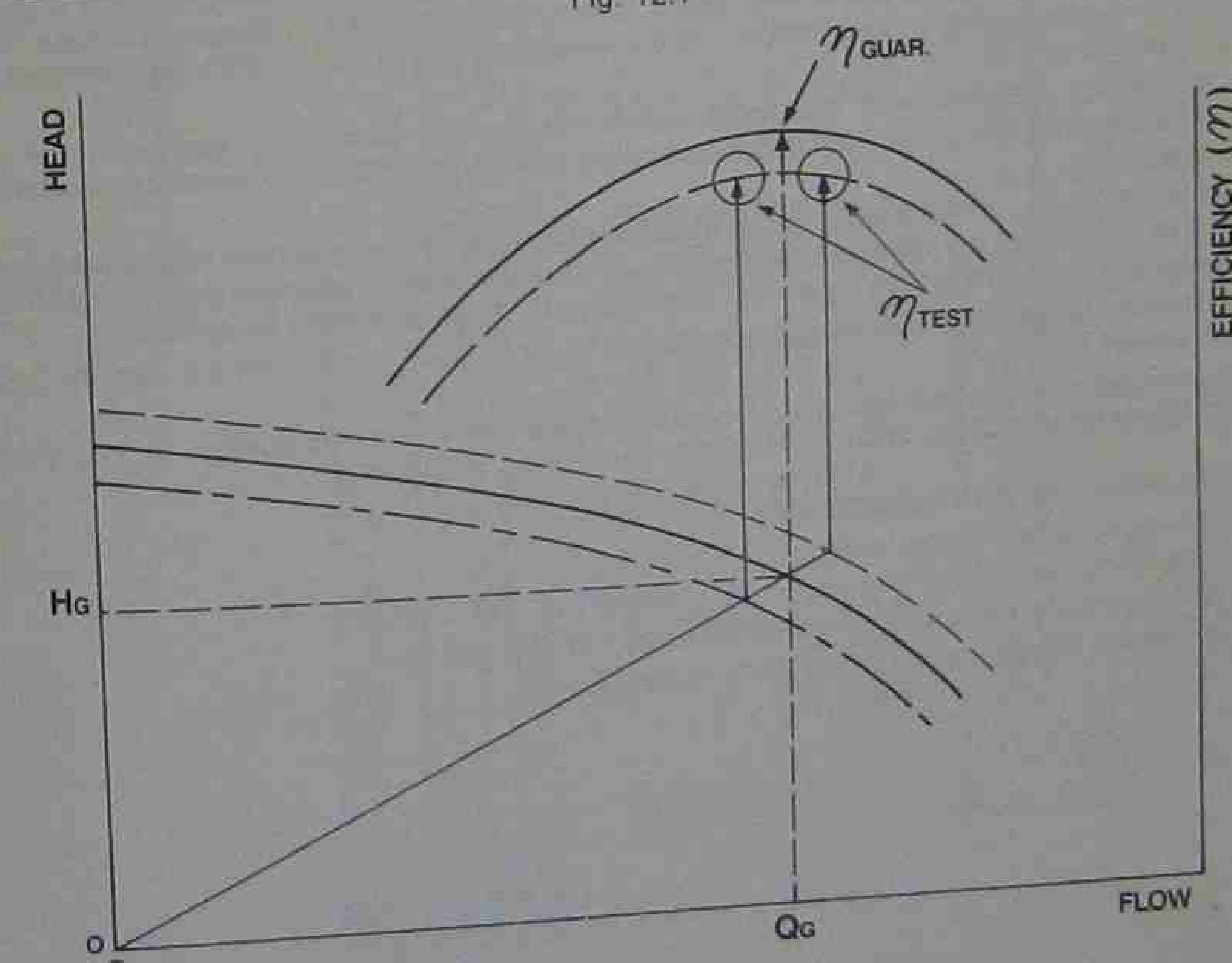


Fig. 12.2

sectional area of pipe. The flow rate is controlled by a throttle valve on the pump discharge.

(b) **Suction & Discharge pressure:** Pressure gauges are the most common devices used for direct pressure measurement but various types of manometers can also be employed. The latest devices in use are transducers which, when in conjunction with computer systems, transform pressure into an electrical signal and are very accurate and reliable.

(c) **Power Measurement:** The majority of pumps tested are driven by electric motors. The normal method used is to measure the input KW using suitable potential and current transformers coupled to single or polyphase wattmeters. The output is calculated using a motor efficiency figure corresponding to the actual load. If the unit is direct coupled, the coupling loss is considered negligible.

If a speed changing device is used between the driver and pump, such as a belt drive or gearbox, then the loss in this device must be known to arrive at the input power to the pump.

Whenever possible, a manufacturer's test on an engineered pump should be at rated speed and power. This is particularly important where low NPSHA conditions exist at site.

The zone of suction instability or onset of cavitation can then be clearly determined.

In most instances the pump is installed on the test bed coupled to its own driver (usually an electric motor), both being mounted on a combination baseplate. Thus the behaviour of the entire unit can be continually monitored during tests.

Very high pressure pumps, e.g., boiler feed units, often present problems to performance testing at the manufacturers works. In many such cases reduced speed tests are conducted and the affinity laws are used to translate the test data to the performance expected at the actual operational speed.

SITE TESTING OF PUMPS

Many clients specify testing of pumps after installation at site to prove the guaranteed performance. If a pump has been works tested and further tests are required after installation, difficulty is often experienced reproducing the works test results. Often instrumentation and layout are such that accuracy of measurement is not possible and the results can differ greatly from that obtained in the test laboratory.

From the pump manufacturers viewpoint, actual site testing should be regarded more as commissioning of the unit. This means ensuring that the unit operates satisfactorily both mechanically and hydraulically.

Manufacturers outlay considerable capital investment to provide and maintain an adequate test facility. The client must realise that provision of such facilities is expensive and that he must bear an appropriate proportion of the cost if he requires sophisticated tests to be carried out.

If improved reliability of operations and a reduction in standby equipment results, then the cost will be justified.

Although modern design techniques are well defined, the works test of a pump is an insurance against future operational problems which may occur after installation.

TYPICAL PUMP PERFORMANCE TEST

Fig. 12.3 shows typical pump performance test sheet.
The pump being tested is shown on the rig illustrated on Fig. 12.4.
A set of calculations have been included for point 1 of the test sheet.
Fig. 12.5 shows the plot of the pump performance curve.

PUMP PERFORMANCE TEST LOG

TEST LOG No. ....

TEST : 03

JOB NO 86SC0020

CLIENT: MYRTLE CREEK WATER & SEWERAGE BOARD.
CHRISTOPHER'S LANE PUMPING STATION.

Barometric Pressure 979 mb; Water Temperature 16.C; Pump Number ZT 5500; Chart Number T45500; Date 31 Jan 86
Pump Size/Type: 200 × 250 – 500 'ZT'; Impeller Dia 449 mm; Inlet Dia 250 mm; Outlet Dia 200 mm
Motor Make: TECO Frame 280MC; Serial No 5C10961; kW 150; V 415; A 256; r/min 1460
Motor Make: TECO Frame 280MC; Serial No 5C10961; kW 150; V 415; A 256; r/min 1460; Drive: Direct Coupled
Motor Efficiency ee@1.00/0.75/0.50 Load = 0.939/0.945/0.945; kW:w 160; Drive: Direct Coupled; Start time 0900
kq 8.671; kp1 – .013595; kp2 .1019; z1 – .3 m; z2 .815 m; kd 3.0526E-5

No	f	p1	p2	W	EkW	ee	kW	L/s	H	ep	eo	k	r/min	NPSH
1	615	198	494	0.928	148.5	0.941	139.8	215.0	55.56	0.838	0.789	0.1918	1483	7.79
2	526	178	534	0.899	143.8	0.942	135.5	198.9	59.16	0.851	0.802	0.2009	1484	7.92
3	408	152	589	0.846	135.4	0.943	127.7	175.1	64.14	0.863	0.814	0.2147	1485	8.08
4	328	135	631	0.806	129.0	0.944	121.8	157.0	68.00	0.860	0.812	0.2281	1486	8.18
5	249	117	663	0.741	118.6	0.945	112.0	136.8	70.84	0.848	0.802	0.2407	1487	8.30
6	191	103	688	0.684	109.4	0.946	103.5	119.8	73.06	0.830	0.785	0.2537	1488	8.40
7	132	91	712	0.612	97.9	0.946	92.6	99.6	75.21	0.793	0.750	0.2730	1489	8.47
8	72	78	731	0.515	82.4	0.945	77.9	73.6	76.83	0.712	0.673	0.3111	1491	8.55
9	21	66	740	0.373	59.7	0.943	56.3	39.7	77.47	0.537	0.506	0.4172	1493	8.63
10	0	60	747	0.311	49.8	0.941	46.8	0.0	78.05	0.000	0.000	0.000	1495	8.68

Finish Time: 1130 Guaranteed Duty—175 L/s, 64 m, 86% Efficiency.

The Pump was tested in accordance with Australian Standard No. 2417 Part 3—1980 [Class B Tests].
The Results obtained comply with the guarantee and the recommendation is that the pump be accepted.

Tested By: Witnessed By:

Fig. 12.3

Notation

f	—Venturi or Orifice Differential Pressure
A	—Rated Full Load Motor Amps
Inlet Dia	—Inlet Diameter in mm (Pipe)
kd	—Velocity Head Conversion Coefficient velocity head $= \frac{v_2^2 - v_1^2}{2g} = kd \times (L/s)^2$
kq	—Venturi or Orifice Flow Coefficient
kp1&kp2	—Factors to convert inlet and outlet readings to m
kW:w	—Instrument Transformer Ratio (Current and Potential Transformers)
Outlet Dia	—Outlet Diameter in mm (Pipe)
p1	—Inlet Pressure (mm Hg)
p2	—Outlet Pressure (kPa)
V	—Rated Motor Voltage (Volts)
w	—Wattmeter reading
z	—Corrections for gauge height $= z2 - z1$
z1	—Distance from pump datum to p1 (m)
z2	—Distance from pump datum to p2 (m)
ee	—Drive Efficiency
EkW	—Electric Kilowatts to drive $= w \times KW{:}w$
eo	—Overall Efficiency (Pump and Drive)
ep	—Pump Efficiency
H	—Total Head. Metres $= (p2 \times kp2) - (p1 \times kp1) + z +$ Vel Hd.1
k	—KW.h./1000 Litres $= \frac{EkW}{3.6 \times L/s}$
kW	—Pump Input Power (Kilowatts) $= EkW \times ee$
L/s	—Litres per second $= kq\sqrt{f}$
NPSH	—Nett Positive Suction Head at Pump Datum (m) = Barom. Pressure $+ (p_1 \times kp1) -$ Vap.Press $+ \frac{v^2}{2g} + z1$
r/min	—Pump Revolutions per Minute
v1	—Fluid Velocity at tapping point p1(m/s.)
v2	—Fluid Velocity at tapping point p2(m/s.)
VP	—Vapour Pressure (PSI) of Water
°C	—Water Temperature (celsius)
g	—Acceleration due to Gravity (9.7982 m/s²)

EXAMPLE CALCULATION

Test Point No 1 Note f, p1, p2, w & r/min are measured

Flow Rate $L/s = kq\sqrt{f} = 8.671\sqrt{615} = 215$ L/s

Total Head $H = (p2 \times kp2) - (p1 \times kp1) + z +$ Velocity Hd

$z = z2 - z1 = 0.815 - (-0.3) = 1.12$ m

Vel.Hd $= \frac{v2^2 - v1^2}{2g} = kd \times (L/s)^2 = 0.000030526 \times 215^2 = 1.41$ m

$\therefore$ Total Head $H = (494 \times 0.1019) - (198 \times -0.013595) + 1.12 + 1.41 = 55.56$m

Pump Input Power $kW = w \times kW{:}w \times ee = 0.928 \times 160 \times 0.941 = 139.8$kW

Pump Efficiency $ep = \frac{L/s \times H}{kW \times 101.972} = \frac{215 \times 55.6}{139.8 \times 101.972} = 0.838$ (83.8%)

Overall Efficiency $eo = ep \times ee = 0.838 \times 0.941 = 0.789$ (78.9%)

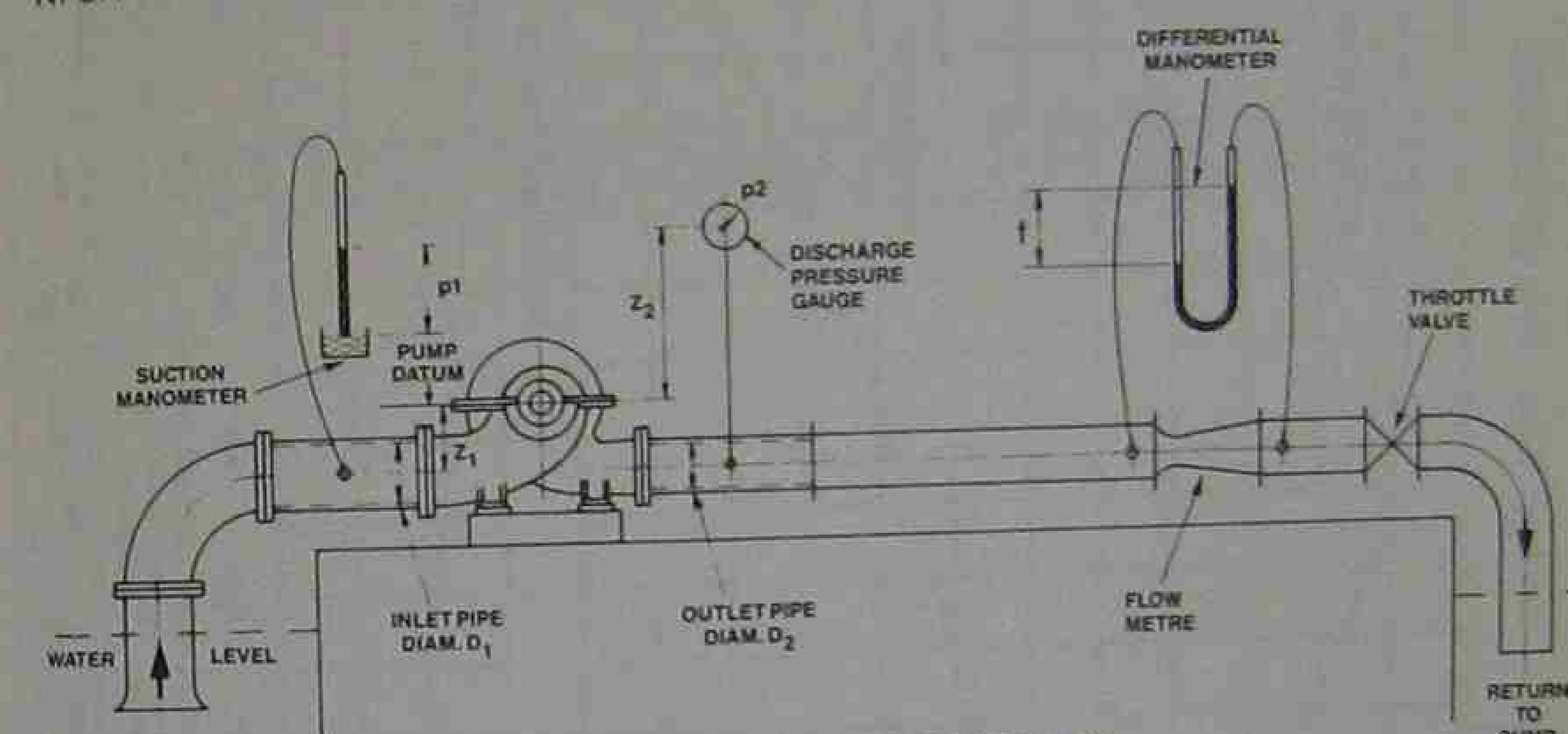
kW.h/1000L $k = \frac{EkW \times 1000}{3600 \times L/s} = \frac{148.5 \times 1000}{3600 \times 215} = 0.1918$

NPSH = Barom Press $+ (p1 \times kp1) -$ Vap. Press $+ \frac{v1^2}{2g} + z1$

Barom Press $= \frac{979}{10 \times g} = 9.99$m Vap. Press at 17°C = 0.19 m

$v1 = \frac{215 \times 1000}{0.7854 \times 250^2} = 4.38$m/s. $v1^2/2g = 0.98$ m

NPSH $= 9.99 + (198 \times -0.013595) - 0.19 + 0.98 - 0.30 = 7.79$m



TYPICAL TEST RIG FOR SPLIT CASE PUMP

TESTED UNDER SUCTION LIFT

Fig. 12.4

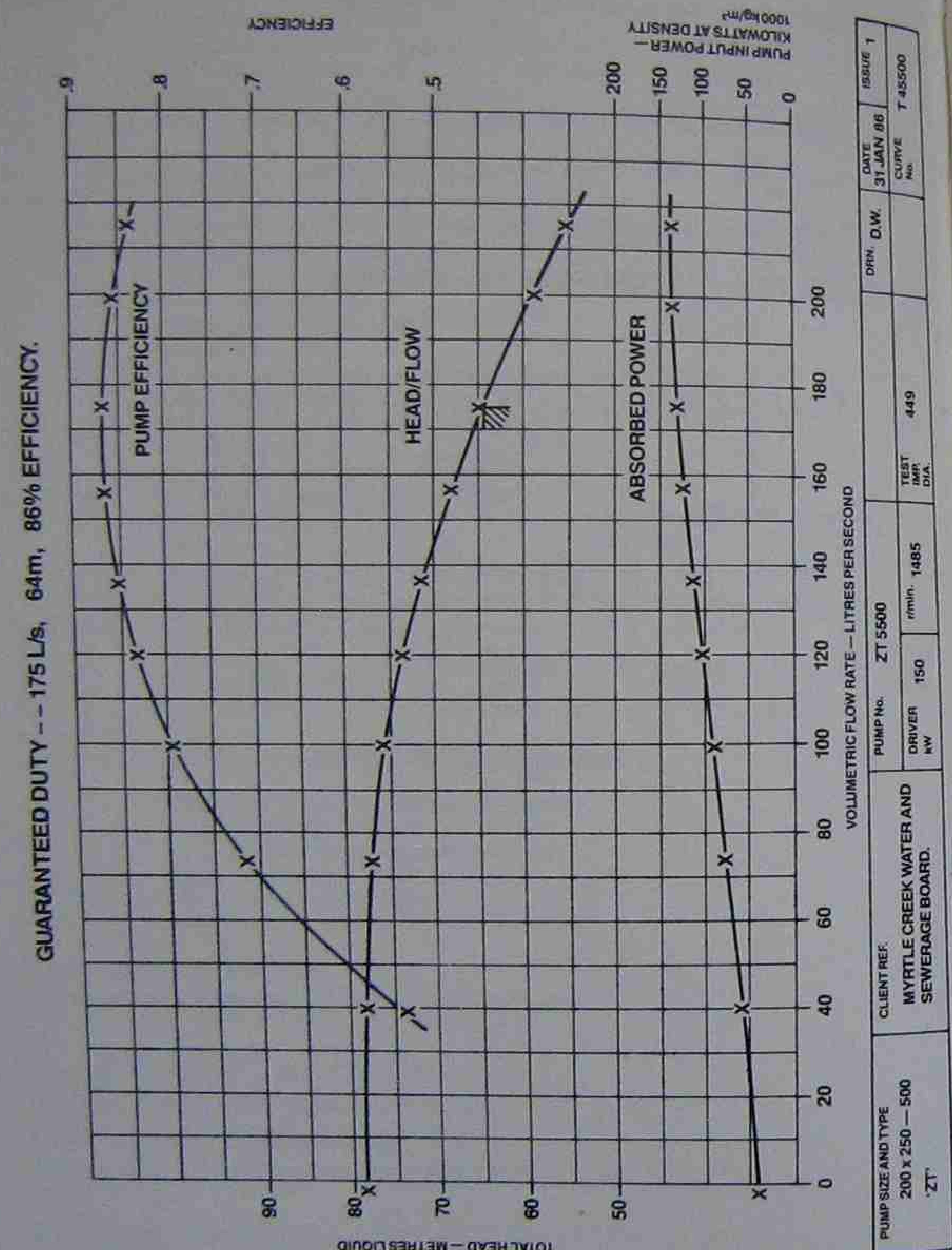


Fig. 12.5

Section 13

CONVERSION TABLES

Section 13—CONVERSION TABLES

The conversion equivalents are generally based on A.S. 1376-1973. In some instances the degree of rounding off has been adjusted for practical use.

Length

millimetre mm	centimetre cm	metre m	inch in	foot ft	yard yd
1	0.1	0.001	0.0394	0.0033	0.0011
10	1	0.01	0.3937	0.0328	0.0109
1000	100	1	39.3701	3.2808	1.0936
25.4	2.54	0.0254	1	0.0833	0.0278
304.8	30.48	0.3048	12	1	0.3333
914.4	91.44	0.9144	36	3	1

1 kilometre = 1000 metres = 0.62137 miles.
1 mile = 1609.34 metres = 1.60934 kilometres.

Area

square millimetre mm^2	square centimetre cm^2	square metre m^2	square inch in^2	square foot ft^2	square yard yd^2
1	0.01	1×10^{-6}	1.55×10^{-3}	1.076×10^{-5}	1.196×10^{-6}
100	1	1×10^{-4}	0.155	1.076×10^{-3}	1.196×10^{-4}
10^6	10 000	1	1550	10.764	1.196
645.16	6.4516	6.452×10^{-4}	1	6.944×10^{-3}	7.716×10^{-4}
92903	929.03	0.093	144	1	0.111
836127	8361.27	0.836	1296	9	1

1 acre = 4046.86 m^2 = 0.404686 hectare.
1 hectare = $1 \times 10^4 m^2$ = 2.471 acre.

Volume

cubic millimetre mm^3	cubic centimetre cm^3	cubic metre m^3	cubic inch in^3	cubic foot ft^3	cubic yard yd^3
1	0.001	1×10^{-9}	6.1×10^{-5}	3.531×10^{-8}	1.308×10^{-9}
1000	1	1×10^{-6}	0.061	3.531×10^{-5}	1.308×10^{-6}
1×10^9	1×10^6	1	61 024	35.31	1.308
16 387	16.39	1.639×10^{-5}	1	5.787×10^{-4}	2.143×10^{-5}
2.832×10^7	2.832×10^4	0.0283	1728	1	0.0370
7.646×10^8	7.646×10^5	0.7646	46 656	27	1

Liquid Measure

cubic metre m^3	litre L	millilitre mL	Imp gallon Imp gal	U.S. gallon U.S. gal	cubic foot ft^3
1	1000	1×10^6	220	264.2	35.3147
0.001	1	1000	0.22	0.2642	0.0353
1×10^{-6}	0.001	1	2.2×10^{-4}	2.642×10^{-4}	3.53×10^{-5}
0.00455	4.546	4546	1	1.201	0.1605
0.00378	3.785	3785	0.8327	1	0.1337
0.0283	28.317	28317	6.2288	7.4805	1

1 U.S. Barrel = 42 U.S. gallons (petroleum measure) = 34.97 Imp. gallons = 0.159 m^3
1 litre = 1×10^6 mm^3 = 1×10^3 cm^3 or 1 cubic decimetre (1 dm^3)
1 litre = 1.76 U.K. pints = 2.133 U.S. pints.

Velocity

metre per second m/s	foot per second ft/s	metre per minute m/min	foot per minute ft/min	kilometre per hour km/h	mile per hour mile/h
1	3.281	60	196.85	3.6	2.2369
0.305	1	18.288	60	1.0973	0.6818
0.017	0.055	1	3.281	0.06	0.0373
0.005	0.017	0.305	1	0.0183	0.01136
0.278	0.911	16.667	54.68	1	0.6214
0.447	1.467	26.822	88	1.6093	1

Mass

kilogram kg	pound lb	hundredweight cwt	tonne t	ton	U.S. ton sh ton
1	2.205	0.0197	0.001	9.84×10^{-4}	0.0011
0.454	1	0.0089	4.54×10^{-4}	4.46×10^{-4}	5.0×10^{-4}
50.802	112	1	0.0508	0.05	0.056
1000	2204.6	19.684	1	0.9842	1.1023
1016	2240	20	1.0161	1	1.102
907.2	2000	17.857	0.9072	0.8929	1

Mass Flow Rate

kilogram per second kg/s	pound per second lb/s	kilogram per hour kg/h	pound per hour lb/h	ton per hour ton/h	tonne per hour t/h
1	2.205	3600	7936.64	3.5431	3.6
0.454	1	1633	3600	1.607	1.633
2.78×10^{-4}	6.12×10^{-4}	1	2.205	9.84×10^{-4}	0.001
1.26×10^{-4}	2.78×10^{-4}	0.454	1	4.46×10^{-4}	4.54×10^{-4}
0.282	0.622	1016	2240	1	1.016
0.278	0.612	1000	2204.6	0.9842	1

litre per second L/s	litre per minute L/min	cubic metre per hour m^3/h	cubic foot per hour ft^3/h	cubic foot per minute ft^3/min	Imp. gallon per minute Imp.gal/min	U.S. gallon per minute U.S. gal/min	U.S. barrel per day U.S.barrel/d (petroleum)
1	60	3.6	127.133	2.1189	13.2	15.85	543.439
0.017	1	0.06	2.1189	0.0353	0.22	0.264	9.057
0.278	16.667	1	35.3147	0.5886	3.666	4.403	150.955
0.008	0.472	0.0283	1	0.0167	0.104	0.125	4.275
0.472	28.317	1.6990	60	1	6.229	7.480	256.475
0.076	4.546	0.2728	9.6326	0.1605	1	1.201	41.175
0.063	3.785	0.2271	8.0209	0.1337	0.833	1	34.286
0.002	0.110	0.0066	0.2339	0.0039	0.024	0.029	1

Force

newton N	kilonewton kN	kilogram force* kgf	pound force lbf
1	0.001	0.102	0.225
1000	1	101.97	224.81
9.807	0.0098	1	2.205
4.448	0.0044	0.454	1

* The kilogram force is sometimes called the kilopond (kp).

PRESSURE AND LIQUID HEAD

newton per square metre N/m^2 (Pa)	kilopascal kPa	bar bar	kilogram force per square centimetre kgf/cm^2	pound force per square inch lbf/in^2	foot of water $ft\,H_2O$	metre of water $m\,H_2O$	millimetre of mercury mm Hg	inch of mercury in Hg
1	0.001	1×10^{-5}	1.02×10^{-5}	1.45×10^{-4}	3.35×10^{-4}	1.02×10^{-4}	0.0075	2.95×10^{-4}
1000	1	0.01	1.02×10^{-2}	0.145	0.335	0.102	7.5	0.295
100000	100	1	1.02	14.5	33.52	10.2	750.1	29.53
98067	98.07	0.981	1	14.22	32.81	10	735.6	28.96
6895	6.895	0.069	0.0703	1	2.31	0.703	51.72	2.036
2984	2.984	0.03	0.0305	0.433	1	0.305	22.42	0.882
9786	9.789	0.098	0.1	1.42	3.28	1	73.42	2.891
133.3	0.133	0.0013	0.0014	0.019	0.045	0.014	1	0.039
3386	3.386	0.0338	0.0345	0.491	1.133	0.345	25.4	1

1 Pascal equals 1 newton per square metre ($1Pa = 1N/m^2$)
1 mm Hg is also known by the name "torr".
The international standard atmosphere (1 atm) = 101 325 pascals or 1.013 25 bar. This is equal to 1.033 23 kgf/cm^2 or 14.6959 lbf/in^2.
1 millibar = 100 pascal (1 mb = 100 Pa)

joule J	kilojoule kJ	megajoule MJ	foot pound force ft lbf	British thermal unit B.t.u	therm	kilowatt hour kWh
1	0.001	1×10^{-6}	0.737	9.48×10^{-4}	9.48×10^{-9}	2.78×10^{-7}
1000	1	0.001	737.56	0.9478	9.48×10^{-6}	2.78×10^{-4}
1×10^6	1000	1	737 562	947.82	9.48×10^{-3}	0.2778
1.356	1.36×10^{-3}	1.36×10^{-6}	1	1.28×10^{-3}	1.28×10^{-8}	3.77×10^{-7}
1055.1	1.0551	1.05×10^{-3}	778.17	1	1×10^{-5}	2.931×10^{-4}
1.0551×10^8	105 510	105.51	7.78×10^7	100 000	1	29.307
3.6×10^6	3600	3.6	2.65×10^6	3412.1	0.03412	1

1 joule = 1 newton metre.

Power

Watt W	kilogram force metre per second kgf m/s	metric horsepower	foot pound force per second ft lbf/s	horsepower hp
1	0.102	0.00136	0.738	0.0013
9.806	1	0.0133	7.233	0.0131
735.5	75	1	542.476	0.9863
1.356	0.138	1.84×10^{-3}	1	1.82×10^{-3}
745.70	76.04	1.0139	550.0	1

1 watt = 1 joule per sec = 1 newton metre per sec.

Density. 1 g/cm^3 = 1000 kg/m^3 = 0.0361 lb/in^3.
1 kg/m^3 = 0.001 g/cm^3 = 0.0624 lb/ft^3.

Specific Volume. 1 cm^3/g = 0.001 m^3/kg = 27.68 in^3/lb.
1 m^3/kg = 1000 cm^3/g = 16.0185 ft^3/lb.

DECIMAL MULTIPLES AND SUB-MULTIPLES OF SI UNITS —PREFIXES:

Factor		
10^{12}	tera	T
10^9	giga	G
10^6	mega	M
10^3	kilo	k
10^2	hecto	h
10	deca	da
10^{-1}	deci	d
10^{-2}	centi	c
10^{-3}	milli	m
10^{-6}	micro	μ
10^{-9}	nano	n
10^{-12}	pico	p
10^{-15}	femto	f
10^{-18}	atto	a

Section 14

PUMPING FORMULAE

Section 14—PUMPING FORMULAE

The following carefully compiled formulae have been included to assist pump users in the planning of pumping plant. See page 125 for definition of terms and units.

Head of liquid from pressure and density.

1) $H = \frac{102p}{\varrho}$ or $H = \frac{p}{9.8\ SG}$

Volumetric flow from mass flow and density

2) $Q = \frac{1000\ Q_m}{\varrho}$ or $Q = \frac{Q_m}{SG}$

Average velocity of flow through a circular section.

3) $v = \frac{1273\ Q}{d^2}$

Average velocity of flow through an area normal to the flow

4) $v = \frac{1000\ Q}{A}$

Velocity head from average velocity

5) $H_v = \frac{v^2}{19.6}$

Velocity head from volumetric flow through a circular section

6) $H_v = \frac{82711\ Q^2}{d^4}$

Force or thrust from pressure and area.

7) $F = \frac{p\ A}{1000}$

Pump input power from pump performance

8) $P = \frac{\varrho H_t Q}{102041\ ep}$ or $P = \frac{SG\ H_t\ Q}{102\ ep}$

Torque from power and speed

9) $T = \frac{9549.3P}{N}$

Pump input power from 3 phase, alternating current electric supply.

10) $P = \frac{\sqrt{3}\ EI \cos\phi\ em}{1000}$

Pump input power from a kilowatt-hour metre.

11) $P = \frac{3600\ K_{mr} r_{em}}{k_{md} t}$

Motor input energy per volume of liquid pumped.

12) $\frac{U}{V} = \frac{\varrho H_t}{367347\ ep.em}$ or $\frac{U}{V} = \frac{SG\ H_t}{367.3\ ep.em}$

Kinematic viscosity from dynamic viscosity and density.

13) $\nu = \frac{1000\mu}{\varrho}$ or $\nu = \frac{\mu}{SG}$

Specific speed

14) $N_s = \frac{N\sqrt{Q}}{H^{3/4}}$

Suction Specific speed

15) $Nss = \frac{N\sqrt{Q}}{(NPSH)^{3/4}}$

DEFINITION OF TERMS AND UNITS

Unless otherwise denoted in the text, the following notation has been used:

A	area	square millimetre
C*	concentration	milligram per litre or kilogram per cubic metre
C_v*	ratio or concentration of solids to slurry by true volume	percent
C_w*	ratio of concentration of solids to slurry by weight	percent
d	diameter normal to flow	millimetre
d_{50}	sieve size that retains 50% of solids by weight	micron
em	motor efficiency	decimal
ep	pump efficiency	decimal
epm*	pump efficiency when pumping slurry	decimal
epw*	pump efficiency when pumping clean cold water	decimal
E	line voltage	volts
ER*	ratio of pump efficiency for slurry to efficiency for water	
F	force	newton
H	head of liquid	metre
H_t	pump total dynamic head	metre
H_{tm}*	pump total dynamic head for slurry	metre
H_{tw}*	pump total dynamic head for clean cold water	metre
H_v	velocity head	metre
HR*	ratio of pump total dynamic head developed for slurry to head developed for clean cold water	
I	line current	ampere
K_{md}	meter disc constant	rev. per kilowatt hour
K_{mr}	meter ratio constant	
M_s	mass of solids pumped	tonnes per hour
N	speed of rotation	rev. per minute
Ns**	specific speed	
Nss**	suction specific speed	
p	pressure	kilopascal
P	pump input power	kilowatt
Q_m	mass flow rate	kilogram per second
Q	volumetric flow rate	litre per second
r	revolutions	
SG	specific gravity of liquid	
SG_m*	specific gravity of slurry	
SG_s*	specific grativy of solids in suspension	
t	time	second
T	torque	newton metre

U	electrical energy	kilowatt hour
μ	dynamic viscosity	millipascal second
v	average velocity	metre per second
V	volume	kilolitre or cubic metre
ν	kinematic viscosity	square millimetre per second
ϱ	mass density of the liquid	kilogram per cubic metre
cosØ	power factor	

In the formulae the acceleration due to gravity for Australia has been taken as 9.80 m/s^2.

* These terms are applicable to slurry applications only

** Specific speed and suction specific speed are type numbers only. Before using these numbers the units must be established.

FORMULAE FOR PUMPING SLURRY

Volumetric flow from mass of solids pumped and concentration.

1) $Q = \dfrac{1000\ M_s}{3.6\ C}$

Specific gravity of slurry from concentration and specific gravity of solids.

2) $SG_m = 1 + \dfrac{C(SG_s - 1)}{1000\ SG_s}$

As stated in the notation.

3) $HR = \dfrac{H_{tm}}{H_{tw}}$

4) $ER = \dfrac{epm}{epw}$

Concentration from the ratio of solids to slurry and the specific gravity of solids.

5) $C = 10\ SG_s\ C_v$

6a) $C = \dfrac{1000\ SG_s\ C_w}{100\ SG_s - C_w(SG_s - 1)}$ 6b) $C_w = \dfrac{100\ C\ SG_s}{1000\ SG_s + C(SG_s - 1)}$

NOTE: For practical purposes it may be assumed that HR = ER.

For concentrations of solids up to 20% by volume, the following expression gives approximate values for HR and ER for centrifugal pumps.

7) $HR = ER = 1 - \dfrac{0.0385\ C(SG_s + 4)\ (SG_s - 1)}{1000\ SG_s + C(SG_s - 1)} \log_e \left(\dfrac{d_{50}}{22.7}\right)$

Pump input power from pump performance when pumping slurry.

8) $P = \dfrac{SG_m\ H_{tm}\ Q}{102\ ER\ epw}$

EXAMPLE

Assume the following data are known.

M_s = 200 tonnes per hour
C = 500 milligram/litre
SG_s = 2.65
d_{50} = 300 microns (0.300 mm)
H_{tm} = 30 metre

1. To find the flow rate of slurry—refer equation (1)

$$Q = \frac{1000\ M_s}{3.6\ C} = \frac{1000 \times 200}{3.6 \times 500} = 111.1 \text{ litres per second}$$

2. To find specific gravity of slurry—refer equation (2)

$$SG_m = 1 + \frac{C(SG_s - 1)}{1000\ SG_s} = 1 + \frac{500(2.65 - 1)}{1000 \times 2.65} = 1.31$$

3. To find HR and ER—refer equation (7)

$$HR = ER = 1 - \frac{0.0385 \times 500(2.65 + 4)\ (2.65 - 1)}{1000 \times 2.65 + 500\ (2.65 - 1)} \log_e \left(\frac{300}{22.7}\right) = 0.84$$

4. To find pump speed and efficiency

$$H_{tw} = \frac{H_{tm}}{HR} = \frac{30}{0.84} = 36 \text{ metres}$$

For the type of solids, flow rate and total head, the type and size of pump may be selected. From the water performance curves for this pump, the speed required for a total head of 36 metres at a flow rate of 111.1 litres per second of water may be determined together with the water efficiency epw for this point. This speed will be the speed necessary to develop the required total head of 30 metres of slurry.

Say epw = 0.75 (from pump performance curve)

5. To find pump input power—refer equation (8)

$$P = \frac{SG_m\ H_{tm}\ Q}{102\ ER\ epw} = \frac{1.31 \times 30 \times 111.1}{102 \times 0.84 \times 0.75} = 67.9 \text{ kilowatts}$$

Section 15

ENQUIRY INFORMATION

APMA ENQUIRY FORM

Company ....

Address ....

.... Postcode ....

Telephone .... Contact ....

DETAILS REQUIRED FOR PUMP SELECTION

Service ....

Operation ....

Installation ....

Control ....

Fluid pumped ....

Pumping Temperature P.T. .... °C

Density at P.T. .... kg/m³

Viscosity at P.T. Kinematic .... mm²/s Dynamic .... mPa.s

Vapour Pressure at P.T. .... kPa

Corrosion/Erosion due to pH value ....

Rate of Flow – normal .... min. .... L/s max. .... L/s

Head Suction – Static (S) .... m

– Friction .... m

NPSH available at pump centreline .... m

Head Discharge – Static (D) .... m

– Friction .... m

Total Head at required Flow Rate m

Drive Type ....

Electricity .... Volts .... Phase ....Hz

Maximum Speed .... r/min.

It is recommended a pump manufacturer be consulted prior to the completion of pump specification and selection. Invariably, the manufacturer will be in a position to advise the solution to a particular application, based on specialist, up to date information and extensive experience in the pumping industry.

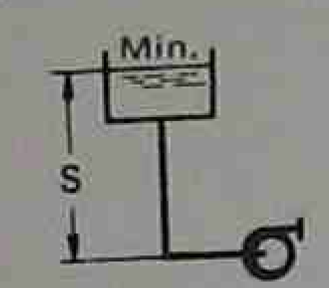
All manufacturers are enthusiastic to supply the most suitable pump for a given installation. It is of great assistance to the manufacturer to receive the fullest information in order that the correct type and size of pump be offered.

Due to the enormous variety of pump types and duties, it is not feasible to provide an enquiry form which covers all possible applications. The following diagrams and standard form are designed to enable a pump buyer to express his basic requirements. Further communication between client and manufacturer will often be necessary before an optimum pump selection can be made.

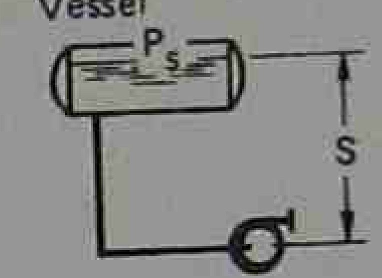
The following diagrams may assist in the completion of the APMA enquiry from. The layouts shown are typical only.

SUCTION

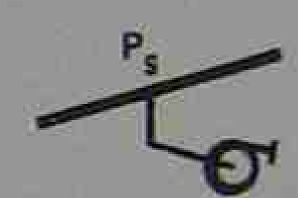
Open Tank



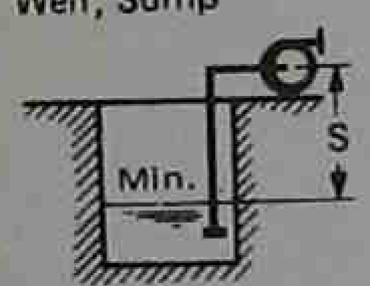
Closed Storage Vessel



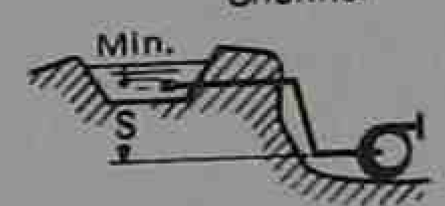
Supply Main



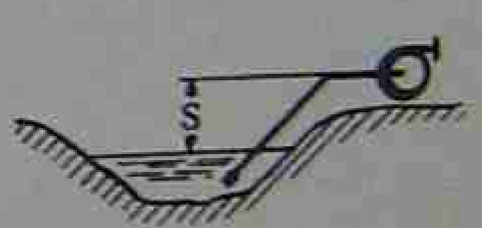
Well, Sump



Reservoir, Irrigation Channel

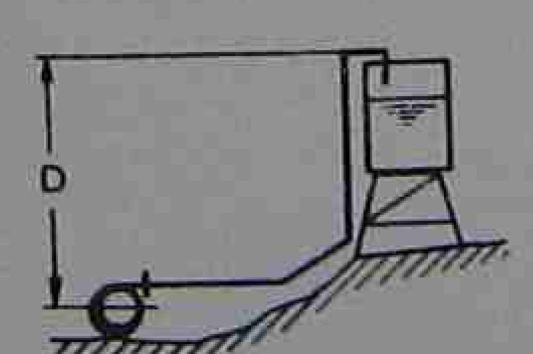


River, Creek, Dam

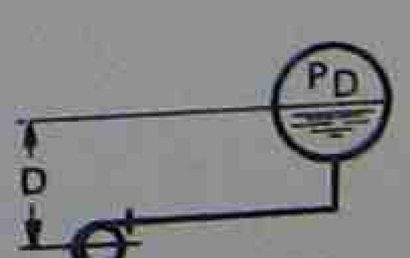


DISCHARGE

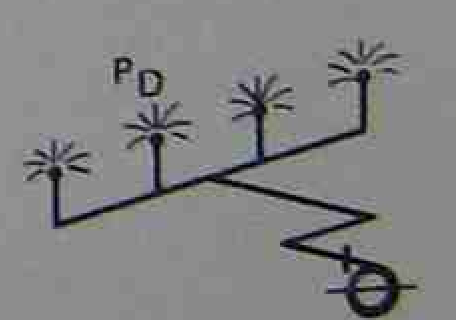
Storage Tank



Closed Storage Vessel



Spray Irrigation



ACKNOWLEDGEMENTS

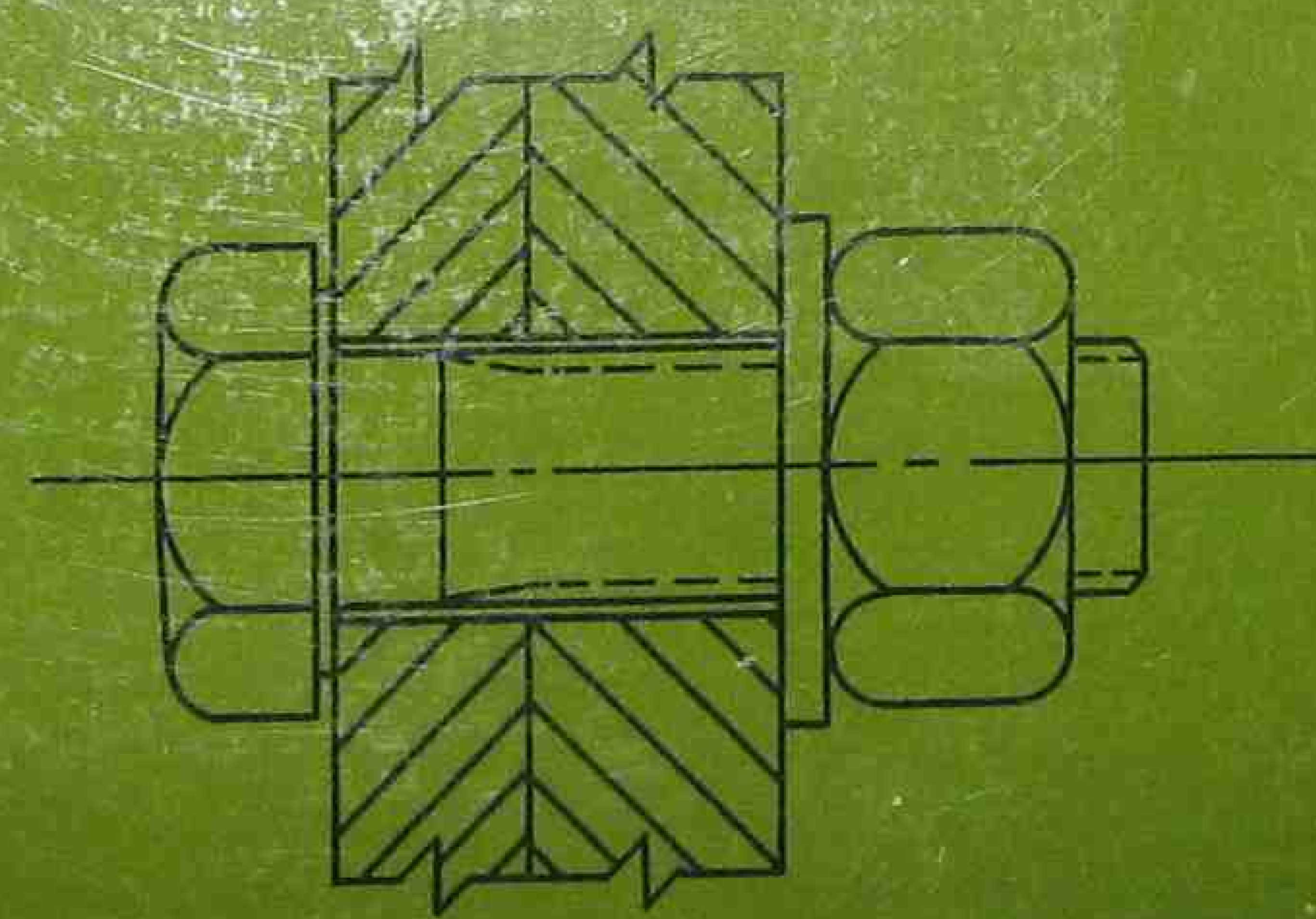
Acknowledgement is hereby given to the following organisations who have permitted the use of material contained in this publication:—

Trade & Technical Press Limited	—	England
Hydraulics Institute	—	U.S.A.
McGraw—Hill Book Company	—	U.S.A.
Ingersoll—Rand	—	U.S.A.
Chapman & Hall Limited	—	England
Institution of Engineers, Aust.	—	Australia
Imported Publications	—	U.S.A.
K.S.B. Pumps	—	West Germany
British Hydro Mechanics Research Association	—	England
Crane Limited	—	England
Sulzer Brothers	—	Switzerland

AISC

Bolting of Steel Structures

STATES
0-1990



A. FIRKINS
T.J. HOGAN

AUSTRALIAN INSTITUTE OF STEEL CONSTRUCTION

NATIONAL LIBRARY OF AUSTRALIA CARD NUMBER AND ISBN 0 909945 59 4

FIRST PUBLISHED 1978

SECOND EDITION 1984 (REPRINTED 1988)

THIRD EDITION — LIMIT STATES DESIGN 1990

THE AUTHORS

Arthur Firkins is Director Technical Services with the Australian Institute of Steel Construction. He is well-known as a lecturer on bolting of steel structures and on the wider topic of economy in fabricated steelwork. He is a member of the committee responsible for the writing of both the SAA Steel Structures Codes, AS 1250-1981 and AS 4100-1990.

Tim Hogan is a Director of the consulting engineering firm of Stigter Clarey and Partners and a former Engineering Manager at AISC. He is Co-Chairman of the SAA Steel Structures committee responsible for the writing of both AS 1250-1981 and AS 4100-1990.

BOLTING OF STEEL STRUCTURES

CONTENTS

		Page No.
1	INTRODUCTION	2
2	SCOPE	2
3	CHARACTERISTICS OF STRUCTURAL BOLTS	
	3.1 Thread form	2
	3.2 Bolt types	3
	3.3 Identification	3
	3.4 Washers	3
4	BOLTING CATEGORIES	4
5	DESIGN OF BOLTS	
	5.1 Force transfer	5
	5.2 Strength of bolts	7
	5.3 Strength limit state for static loads	9
	5.4 Serviceability limit state	11
	5.5 Fatigue	14
6	BOLT LENGTH SELECTION	
	6.1 Plain shank lengths	15
	6.2 Threads included in shear plane	16
	6.3 Threads excluded from shear plane	16
	6.4 Threads included versus threads excluded	18
	6.5 Thread projection	20
	6.6 Material in the grip	20
7	DETAILING	
	7.1 Bolt holes	21
	7.2 Detailing limitations	22
	7.3 Standardized connections	23
	7.4 Washer requirements	24
8	INSTALLATION OF BOLTS	25
	8.1 Introduction	25
	8.2 Snug tightening	26
	8.3 Full tensioning methods	33
	8.4 Clearances	34
	8.5 Inspection	35
9	CORROSION PROTECTION	36
10	ECONOMICAL DESIGN AND DETAILING	37
11	SUMMARY OF BOLT USAGE	39
12	REFERENCES	
APPENDICES		42
A	DETAILS OF COMMONLY USED BOLTS NUTS AND WASHERS	46
B	DETAILS OF SPECIAL BOLTS NUTS AND WASHERS	48
C	DESIGN CAPACITIES OF COMMONLY USED BOLTS— STRENGTH LIMIT STATE	52
D	DESIGN CAPACITIES OF COMMONLY USED BOLTS— SERVICEABILITY LIMIT STATE	53
E	DESIGN CAPACITIES OF COMMONLY USED BOLTS—FATIGUE LIMIT STATE	54
INDEX		

BOLTING OF STEEL STRUCTURES

by

A. FIRKINS

Director Technical Services, AISC

and

T. J. HOGAN B.E. (Hons), M.Eng.Sc., F.I.E.Aust., C.P.Eng.

Director, Stigter Clarey & Partners Pty. Ltd.

— Consulting Engineers

1. Introduction

Bolts are widely used for making connections in structural steelwork, especially field connections. An understanding of all aspects of the use of bolts is consequently vital to designing, detailing, fabricating and erecting steel structures.

The selection of a bolt for use in a structural steelwork connection will need to have regard to a variety of factors including:—

- the nature of the forces to be resisted;
- the design capacity of available bolt types;
- the amount of joint slippage desirable;
- the degree of flexibility/rigidity desired in the joint;
- the cost of the installed fastener.

This manual will discuss a number of these points in some detail.

2. Scope

This publication is intended to provide a state-of-the-art summary of the following items as they relate to the use of bolts in steel structures:

1. the bolt types that are available and commonly used;
2. the characteristics of each bolt type;
3. the method of distinguishing and identifying each bolt type;
4. the design capacities;
5. the method of detailing to arrive at required bolt lengths;
6. the methods of installation;
7. the methods of corrosion protection.

The scope of the publication is limited to:—

- bolt types in common use for steel structures in Australia;
- metric bolts;
- sizes of bolts in common use;
- common methods of application.

3. Characteristics of Structural Bolts

3.1 THREAD FORM

Australian metric bolts incorporate the metric coarse thread series specified in AS 1275 (7)* which is based on ISO (International Standards Organisation) recommendations. Screw threads are to tolerance class 6g of AS 1275 before corrosion protection is applied.

The items of interest in relation to screw threads are the core area (A_c) and tensile stress area (A_s) and these are summarized in Table 1 for bolt diameters in common use. The formula used for calculating A_s (tensile stress area) in AS 1275 is an approximation only, which has been found to correlate reasonably with test results for steels having substantial ductility. Also of relevance are the pitch (p) of the thread and the plain shank area (A_o) which are also quoted in Table 1. The ratios of areas (A_c/A_o and A_s/A_o) are of interest in comparing the capacities of bolts for different design conditions and these are also given in Table 1.

TABLE 1

ISO METRIC SCREW THREADS TO AS 1275 (7)

Nom. dia.	Pitch	Areas			Ratios	
		A_c	A_s	A_o	$\frac{A_c}{A_o}$	$\frac{A_s}{A_o}$
d_f	P	core	tensile stress	shank		
12	1.75	76.2	84.3	113	0.67	0.75
16	2	144	157	201	0.72	0.78
20	2.5	225	245	314	0.72	0.78
24	3	324	353	452	0.72	0.78
30	3.5	519	561	706	0.73	0.79
36	4	759	817	1016	0.75	0.80

*Numbers in brackets indicate References given in Section 12

Explanation to Table 1:

(a) For the area of cross-section at the minor diameter (core area) of external threads—

$$A_c = \frac{\pi(d_1)^2}{4}$$

(b) For the area of the circle having a diameter equal to the mean of the pitch and minor diameters (tensile stress area) of external threads—

$$A_s = \frac{\pi}{4}(D-0.9382P)^2$$

(c) For the nominal shank area—

$$A_o = \frac{\pi}{4}d_f^2$$

where

A_c = minor diameter area (core area)

A_s = tensile stress area

A_o = nominal plain shank area

D = basic major diameter of internal thread (see AS 1275)

P = pitch (see AS 1275)

d_1 = basic minor diameter varying with tolerance class (see AS 1275), taken as (D − 1.22687P)

d_f = nominal bolt diameter

3.2 BOLT TYPES

The two basic types of metric bolt in use in structural engineering in Australia are:—

- **the commercial (Strength Grade 4.6) bolt to AS 1111 (2)**
- **the high strength structural (Strength Grade 8.8) bolt to AS 1252 (6).**

Full details of these bolts are contained in Appendix A.

Commercial bolts are made of low carbon steel with mechanical properties similar to that of Grade 250 material. High strength structural bolts are made by heat-treating, quenching and tempering medium carbon steel. Accordingly, heating or welding to a commercial bolt will cause no significant change in its properties, but either process will cause a significant degradation in the mechanical properties of high strength bolts.

Only a limited range of sizes of these bolts is of interest to structural engineers. The **commercial** bolt is commonly used in the following diameters:—

M12 — purlin and girt applications;

M16 — cleats, brackets (relatively lightly loaded);

M20, M24 — general structural connections, holding down bolts;

M30, M36 —holding down bolts.

Note that the prefix M is used to designate metric bolts with a thread complying with ISO Recommendations and with AS 1275.

The **high strength structural** bolt is most commonly used in diameters:—

M16 — designed connections in small members;

M20, M24 — flexible connections, rigid connections.

(M30, M36) Larger sizes (M30, M36) of the high strength structural bolt should be avoided when full tensioning is required, since on-site tensioning can be difficult and requires special equipment to achieve the minimum bolt tensions specified to be achieved (see 'Installation', Section 8).

Other special types of bolts in limited use in Australia are detailed in Appendix B.

3.3 IDENTIFICATION

The identification of the two different bolt and nut assemblies can be readily made from the bolt head and nut markings indicated in Appendix A. *In addition, a distinguishing feature is the larger bolt head and nut of the high strength structural bolt compared to the commercial bolt.*

The strength of bolts is normally specified in terms of the tensile strength of the threaded fastener. As a consequence, grades of bolts are identified in the following manner:

x • x

Approximately one tenth of the ratio between nominal yield stress and nominal tensile strength expressed as a percentage.

One hundredth of the nominal tensile strength (MPa).

For example a Grade 4.6 to AS 1111 (2) bolt has:

Nominal tensile strength of	4 × 100 = 400 MPa (Actual = 400 MPa)
Nominal yield stress of	0.6 × 400 = 240 MPa (Actual = 240 MPa)

while a Grade 8.8 bolt to AS 1252 (6) has

Nominal tensile strength of	8 × 100 = 800 MPa approx. (Actual = 830 MPa)
Nominal yield stress of (stress at perm. set)	0.8 × 800 = 640 MPa (Actual = 660 MPa)

3.4 WASHERS

Commercial bolts to AS 1111 (2) are not normally supplied with a washer and therefore washers to AS 1237 (4) are ordered separately if required. High strength structural bolts to AS 1252 (6) are normally supplied as bolt/nut/washer assemblies.

4. Bolting Categories

In Australia a standard bolting category identification system has been adopted in AS 4100-1990 "Steel Structures" (5c) for use by designers and detailers. This system is summarized in Table 2.

TABLE 2
BOLT TYPES AND BOLTING CATEGORIES

Bolting Category		Method of Tensioning	Minimum Bolt Tensile Strength (MPa)	Minimum Bolt Yield Strength (MPa)	Bolt Name	Bolt Standard Specification
4.6/S		Snug	400	240	Commercial (Table A.1)	AS 1111 (Ref. 2)
8.8/S						
8.8/T	8.8/TF (friction type joint)	Full Tensioning	830	660	High Strength Structural (Table A.2)	AS 1252 (Ref. 6)
	8.8/TB (bearing type joint)					

Category **4.6/S** refers to commercial bolts of Strength Grade 4.6 conforming to AS 1111 (2), tightened using a standard wrench to a snug-tight condition (see 'Installation', Section 8.2).

Category **8.8/S** refers to any bolt of Strength Grade 8.8, tightened using a standard wrench to a snug-tight condition in the same way as for category 4.6/S. Essentially, these bolts are used as higher grade commercial bolts in order to increase the capacity of certain connection types. In practice they will normally be high strength structural bolts of Grade 8.8 to AS 1252 (6), but any other bolt of Grade 8.8 would be satisfactory such as those to Ref. 1.

Categories **8.8/TF and 8.8/TB** (or **8.8/T** when referring generally to both types) refer specifically to high strength structural bolts of Strength Grade 8.8 conforming to AS 1252 (6), *fully tensioned in a controlled manner to the requirements of AS 4100 (5c)* — see 'Installation', Section 8.3.

The popularity of high strength structural bolts to AS 1252 used in a 'snug-tight' condition leads to the situation where this bolt may need to be fully tensioned on one occasion while on another occasion snug-tightening is sufficient. This could bring misunderstanding and it is necessary for a designer using this bolt type to indicate clearly what level of tensioning (either full or snug) is required in each connection in order to avoid any confusion on site. The bolting category notation given in Table 2 provides a simple way for the designer to specify his requirements.

The system of category designation identifies the bolt being used by using its strength grade designation (4.6 or 8.8) and identifies the installation procedure by a supplementary letter (S — snug; T — full tensioning). For 8.8/T categories, the type of joint is identified by an additional letter (F — friction-type joint; B — bearing-type joint).

It is most important to note that the high strength structural bolt may be specified in three ways:

- snug-tightened — category 8.8/S;
- fully tensioned, friction-type — category 8.8/TF;
- fully tensioned, bearing-type — category 8.8/TB;

— the level of tensioning being, of course, the same for both 8.8/TF and 8.8/TB categories.

In practice, 8.8/S category would mainly be used in flexible joints where the extra capacity of the stronger bolt (compared to 4.6/S category) makes it economic.

It is recommended that 8.8/TF category be used only in rigid joints where a no-slip joint is essential. Note also that 8.8/TF is the only category requiring attention to the faying surfaces — (see 'Surface Coatings on Plies', Section 5.4.5).

A summary of the usage of Grade 4.6 and 8.8 bolts is contained in Section 11.

Design engineers' drawings and shop detail drawings should both contain notes summarizing the category designations.

5. Design of Bolts

5.1 FORCE TRANSFER

5.1.1 Modes

In any bolted structural connection, there are three fundamental modes of force transfer to be considered — two relating to the transfer of shear and one relating to the transfer of tension. These modes are:—

(i) Shear/bearing mode where the forces are perpendicular to the bolt axis and are transferred by shear and bearing on the bolt and bearing on the connected plies;

(ii) Friction mode, which is similar to the shear/bearing mode in that the forces to be transferred are perpendicular to the bolt axis. However the transfer of forces does not rely on shear and bearing; the frictional resistance of the mating surfaces is the prime factor in the force transfer mechanism;

(iii) Axial tension mode, when the forces to be transferred are parallel to the bolt axis.

Often, the modes are combined, since in a large number of connections it is necessary to transfer both parallel and perpendicular force components.

Typical connections subjecting bolts to shear forces only are splices in members, end connections to bracing members and connections of members to gusset plates in trusses. Bolts in tension are found in hanger connections and in bolted moment connections. The latter type of connection may also subject the bolts to combined shear and tension forces.

5.1.2 Bolts in Tension

The behaviour of a bolt in tension is governed by the performance of its threaded length. Thus bolt standards normally specify tensile testing of the threaded bolt as part of their quality control procedures.

The installation of a bolt involves some tightening of the nut on the bolt and this will induce some level of tension in the bolt. Applied tension must overcome this preload and initially applied tension only changes the contact pressure between the connected plies with very little bolt strain and only a minor change in bolt tension.

Further increases in the applied bolt tension eventually reduces the contact pressure to zero and the connected plates are on the verge of separation (Fig. 1). After the plates separate, the bolt force equals the applied tension.

In actual connections, the connected plies are not totally rigid as the above discussion assumes and they actually flex as the tension is applied to the bolted connection. Depending upon the flexural rigidity of the connected plies, additional forces may develop and these forces are referred to as prying forces. The effect of prying forces is to increase the actual tension in the bolt beyond that due to the applied tension itself.

If the connected plies are very stiff then the deformations will be small and little or no prying force will develop, so that the bolt behaves very much like an idealized bolt in tension (Fig. 1a).

When more flexible connected plies are used, the flexural deformation induces prying forces resulting in additional bolt forces (Fig. 1b).

A full discussion of the prying force phenomenon may be found in References 12 and 16.

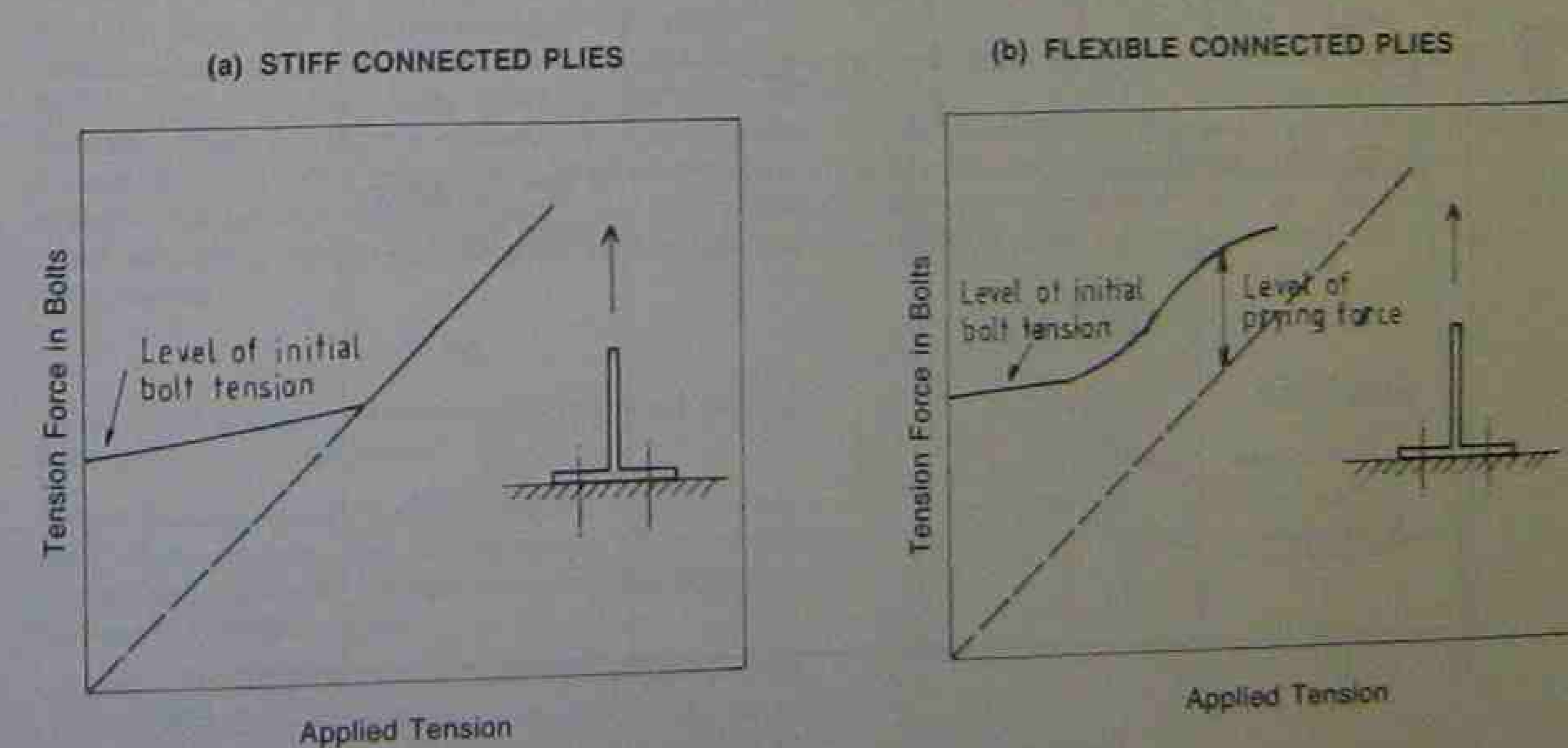


Fig. 1 — Applied Tension and Bolt Tension

5.1.3 Bolts in Shear

The load-extension relationship for the four basic bolting categories in a pure shear connection are illustrated in Fig. 2. Aspects of the behaviour illustrated are reflected in design provisions.

Four characteristic stages are seen to exist for each of the bolting categories in Fig. 2, the change from one stage to another varying with the different bolt categories:

(i) static friction prevents any slip;

(ii) slip occurs as the load exceeds the frictional resistance; (this occurs at a relatively low level for 4.6/S and 8.8/S categories, bolts bear against hole sides);

(iii) fasteners and plates deform elastically;

(iv) yielding of the plates and/or fasteners occurs resulting in either plate fracture or shearing of fasteners.

The load at which slip occurs depends on the level of the tension in the bolt after installation and on the friction coefficient at the interface of the connected plies. Both these factors can be highly variable.

The level of bolt tension is particularly variable with snug tight bolting categories but is more highly controlled with /T bolting categories, where installation procedures are defined with a view to ensuring a predetermined level of bolt tension.

The friction coefficient is very dependent upon the condition of the plies at the interface and generally typical figures are used for design based on test results. A review of typical values of slip coefficient for a variety of surface conditions may be found in Reference 16.

When the frictional resistance is overcome, movement of the joint occurs until the bolts bear against the hole sides. From this stage on, load is transferred by a combination of shear on the bolts and bearing on the ply thickness at the holes. Initially it is the end bolts in the connection which bear first but as the load is increased, the end bolts and holes deform until succeeding bolts come into bearing. Further applied load causes each bolt to deform in proportion to the load it is carrying. A evening out of load occurs if the joint has good ductility.

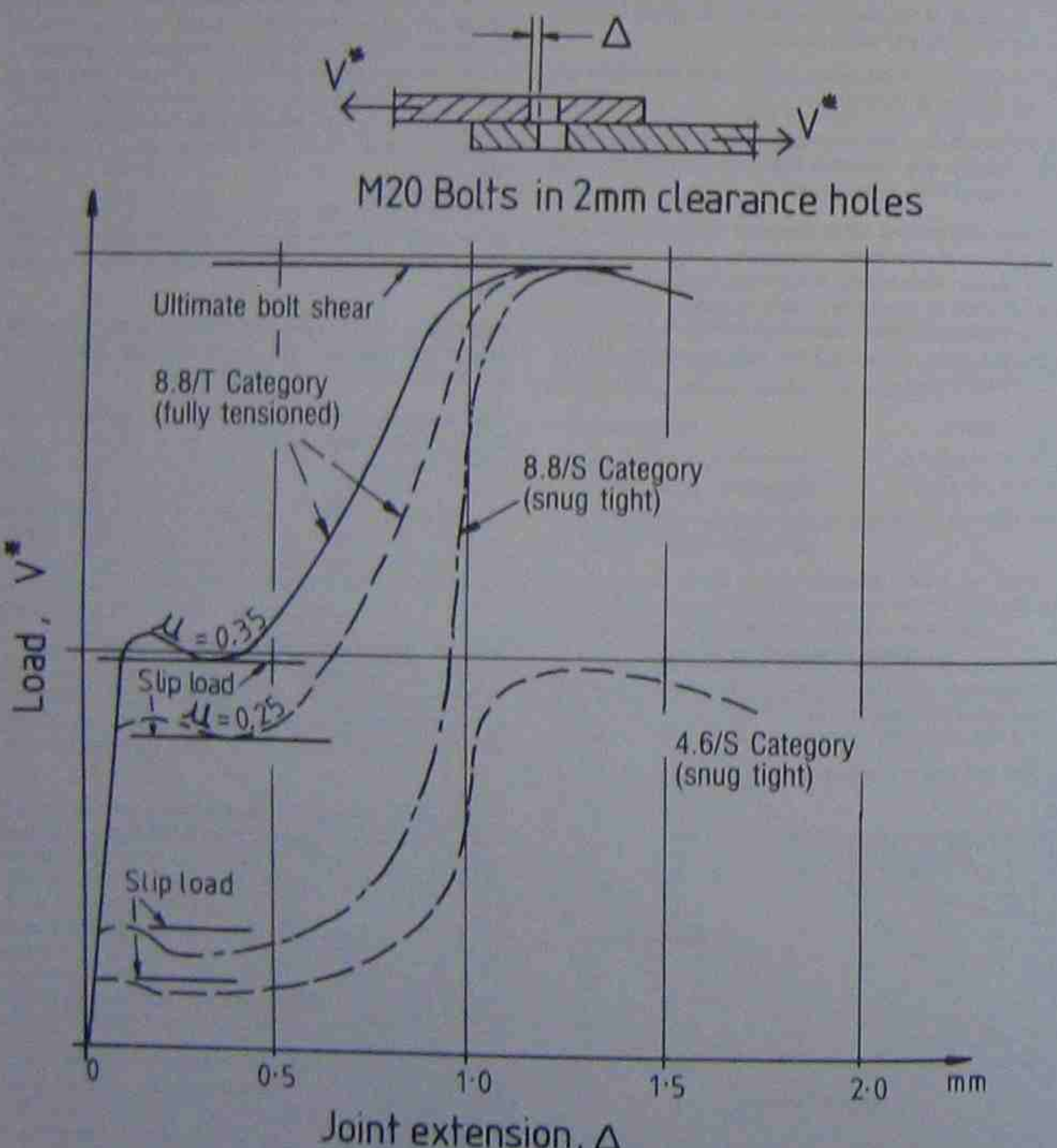


Fig. 2 — Load versus Joint Extension in Shear Connection

5.2 STRENGTH OF BOLTS

5.2.1 Bolts in Tension

The behaviour and strength of a bolt subjected to an axial tension load is governed by the performance of the threaded part of the bolt. The relevant Australian Standards (AS 1111 and AS 1252) specify:—

(i) minimum tensile strength;

(ii) minimum yield stress;

(iii) proof stress.

To determine whether the specified mechanical properties are met, the Standards require direct tension tests on full size bolts. The tensile capacity of a bolt (N_{tf}) is equal to the product of the stress area (A_s) and the tensile strength (f_{uf}), or:—

$$N_{tf} = A_s f_{uf}$$

Values of A_s for the ISO metric thread series may be found in Table 1.

Loading a bolt in direct tension, after preloading by tensioning (8.8/T categories) does not significantly decrease the ultimate tensile strength of the bolt according to Reference 16. Apparently the torsional stresses induced by torquing the bolt have a negligible effect on its tensile strength. Hence, Reference 16 argues that bolts installed in 8.8/T categories can sustain direct tension loads without any reduction in their tensile strength.

5.2.2 Bolts in Shear

Specifications for bolts do not usually require the bolt to be tested in shear, so that the shear strength of bolts does not usually appear in bolt specifications.

Shear strengths have been obtained by a number of investigators by subjecting the bolts to shear induced through plates either in tension or compression. The shear strength has been shown to be influenced by the type of test, with tension-type shear tests giving shear strengths 8%-13% less than compression-type. Reference 16 examines the available data and concludes that for the ASTM A325 and A490 bolts (A325 is equivalent to Australian Grade 8.8; there is no direct equivalent to A490 in Australia, but a bolt of Grade 10.9 would have the same strength), the average shear strength (f_{vf}) was 62% of the tensile strength of the bolt (f_{uf}) with a standard deviation of 3.3%. Hence,

$$f_{vf} = 0.62 f_{uf}$$

This result is based on the use of tension-type tests in order to obtain a lower bound.

Tests on bolted joints have indicated that the level of any initial clamping force has no significant effect on the ultimate shear strength. The factors responsible for this are canvassed in Reference 16.

The shear strength of a bolt is directly proportional to the shear area available, this being the core area (A_c) when considering the threaded part of the bolt or the shank area (A_o) when considering the unthreaded part. Hence, the shear strength is given by:—

V_{fn} = shear strength of threaded part of bolt
$= A_c f_{vf}$
$= 0.62 A_c f_{uf}$

V_{fx} = shear strength of shank of bolt
$= A_o f_{vf}$
$= 0.62 A_o f_{uf}$

Reference 16 reports several interesting facets of shear plane location, these being:—

(i) shear strength is unaffected by the location of the shear plane (either in the shank or the root);

(ii) all available tests indicate that the shear strength is governed by the available shear area;

(iii) the shear and deformation capacity of the bolt are maximized when the shear plane(s) pass through the shank;

(iv) the lowest shear and deformation capacity of the bolt occurs when the shear plane(s) pass through the threaded portion.

For lap splice connections of the type shown in Fig. 3, tests have shown that the length of the bolted connection may have an effect on the strength of the joint.

For short joints, the load on the bolts is reasonably uniform at failure of the joint, but for longer joints the load on the bolts can be quite non-uniform, with the end bolts carrying more load and consequently failing first and unbuttoning then occuring along the joint. A review of this is contained in Ref. 16.

Conventional theories of bolted connection design for joints involving more than one bolt subject to shear loading assume that rigid plate theory applies and that all the bolts are equally loaded. However, both elastic analyses and load measurements (16, 23) indicate that the longer the bolted joint becomes, the less uniform is the load distribution as long as the joint remains elastic.

As a joint is loaded to the extent that either yielding of the plies or bolts or both occur, plastic deformations permit a redistribution of load and a more uniform load distribution will result if the redistribution is allowed to proceed without premature failure of either bolt or plies.

5.2.3 Bolts Subject to Shear and Tension

Tests conducted on bolts subject simultaneously to shear and tension forces indicate that the strength of the bolts is reasonably represented by an elliptical interaction relationship of the type indicated in Fig. 4, so that:—

$$\left(\frac{V_f^*}{V_f}\right)^2 + \left(\frac{N_{tf}^*}{N_{tf}}\right)^2 \leq 1.0$$

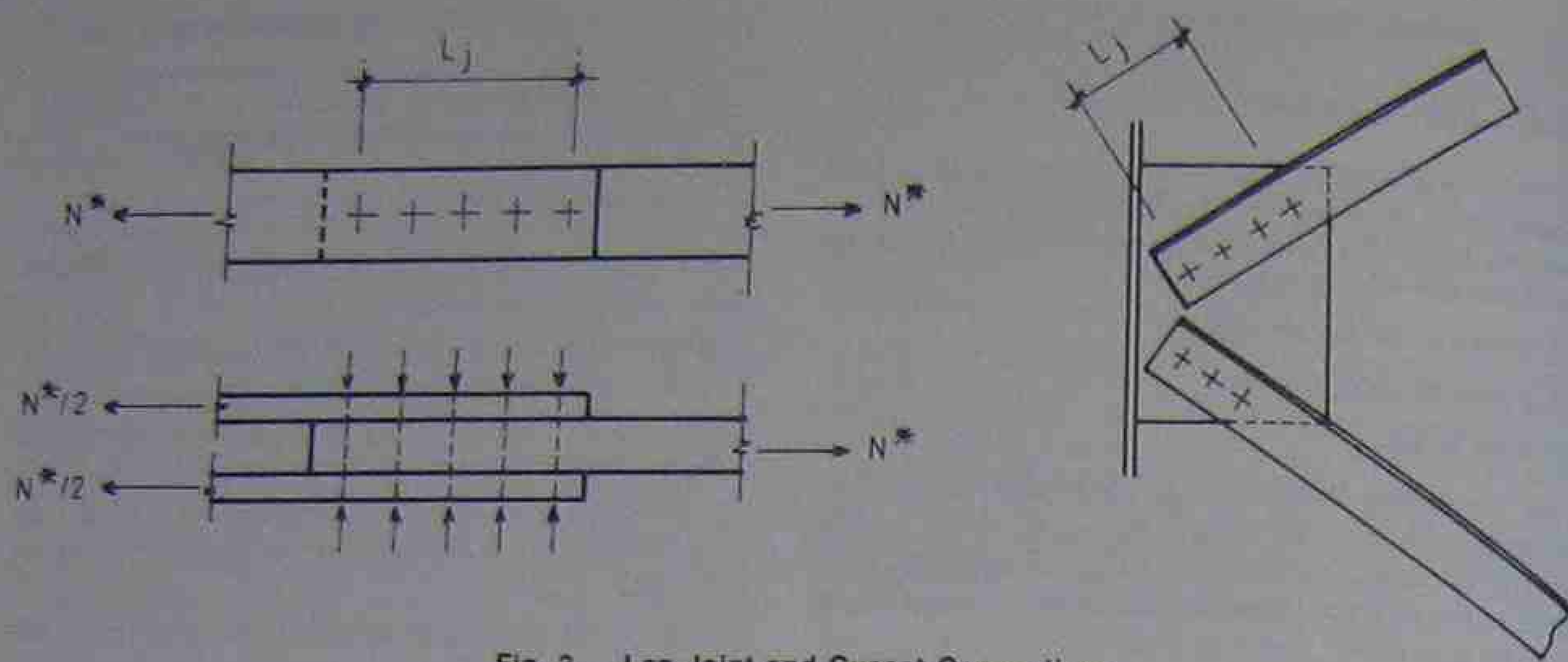


Fig. 3 — Lap Joint and Gusset Connections. Examples of Lap Splice Connections.

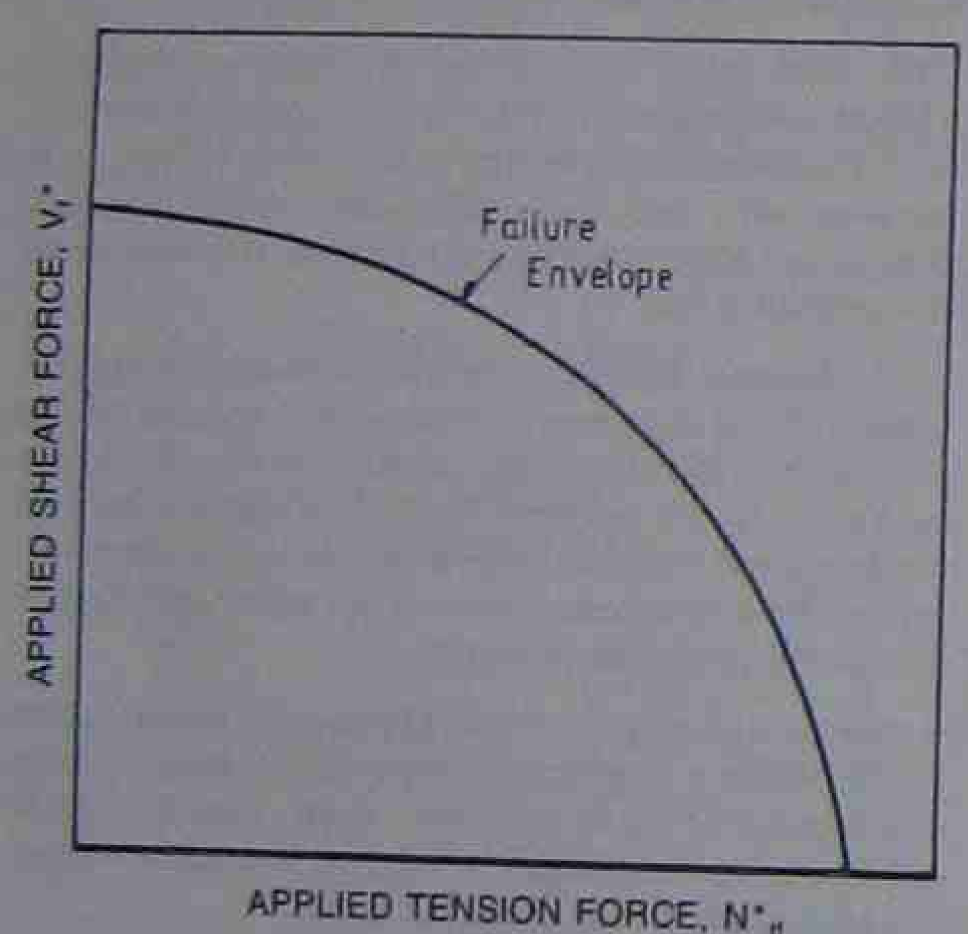


Fig. 4 — Bolts Subject to Shear and Tension.

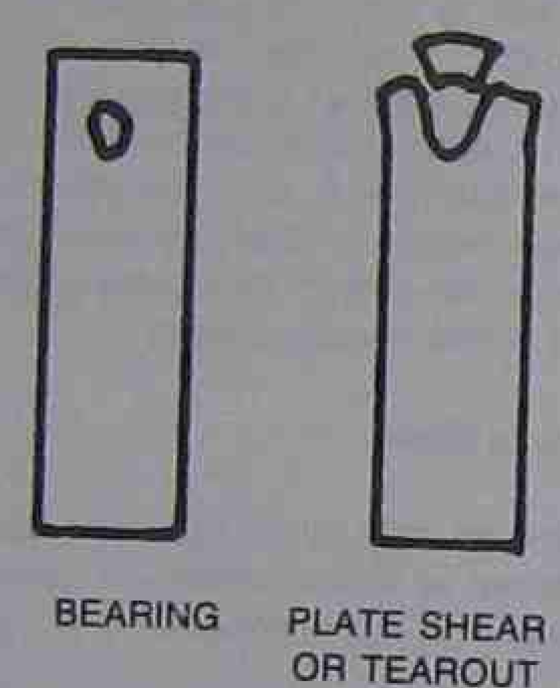


Fig. 5 — Bearing Failure and Plate Tearout Failure

5.2.4 Bearing and Tear Out Failure in Plies

In a bearing joint in which the frictional resistance of the joint is exceeded, the bolt bears against the sides of the bolt holes and load is transferred by shear in the bolts and bearing on the connected plies.

Two failure modes are of interest in this situation in addition to bolt shear or plate tension failures (21):— Fig. 5

(i) local bearing failure of the connected plies;

(ii) shear or tear-out failure of the plate behind the bolt.

In the latter case, the end distance (a_e) is an important geometrical parameter as this distance represents the length of ply which must fail in shear for joint failure to occur. End distance is defined as "the minimum distance from the edge of a hole to the edge of a ply in the direction of the component force plus half the bolt diameter" — see Fig. 6.

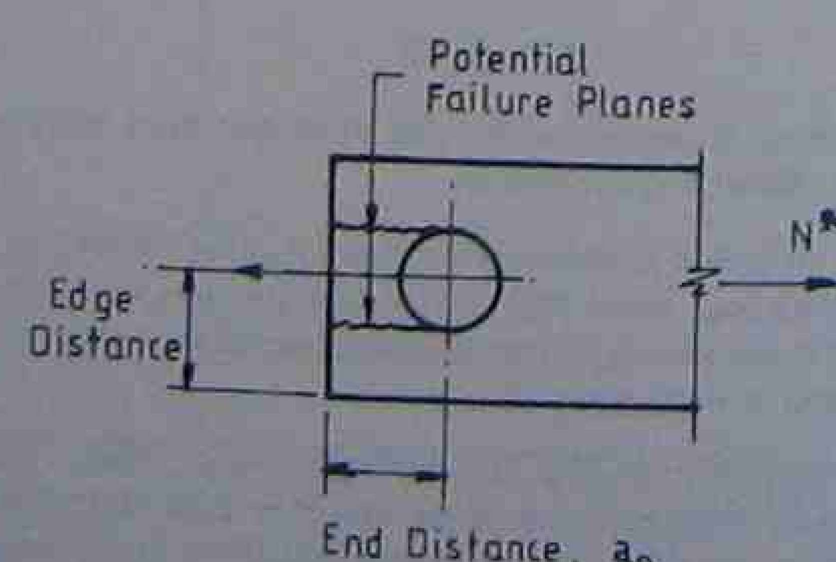


Fig. 6 — End Distance.

The two possible failure modes have been extensively researched and reviews of this research may be found in References 16 and 21.

The research indicates that a local bearing type failure with considerable piling-up of the ply material in front of the bolt occurs at a nominal bearing stress of 4.5 to 4.9 times the yield stress of the ply, which for the normal range of steels is equal to 3.15 to 3.4 times the tensile strength.

Such a failure mode occurs for relatively long end distances generally greater than 3 times the bolt diameter. For shorter end distances, end plate tearout is much more likely.

Various expressions for the failure strength at end plate tearout have been proposed (16, 21) but a simple expression accurate enough for most purposes is:

$$V_b = a_e t_p f_{up}$$

where t_p is the connected ply thickness.

Plate tearout is also theoretically possible between bolt holes although usually the bolt pitch between holes exceeds 3 bolt diameters, so the local crushing occurs in preference to plate tearout (Fig. 7).

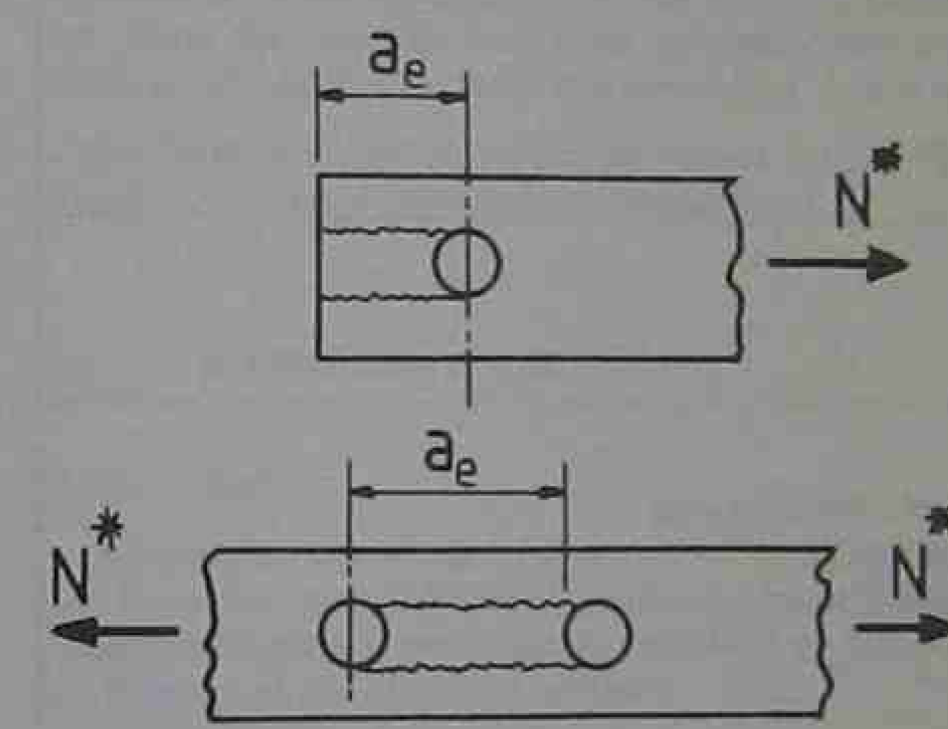


Fig. 7 — End Plate Tearout and Possible Failure Locations

5.3 STRENGTH LIMIT STATE FOR STATIC LOADS

5.3.1. General

All bolted connections irrespective of bolting category are now designed using a single code AS 4100 "Steel Structures" (5c) which is written in limit state format. Accompanying this code is a commentary "AS 4100 Supplement 1. Commentary — Steel Structures" (5d). A discussion of the limit state design method may be found in the Commentary (5d).

Formerly, two codes in permissible stress format were used, namely AS 1250 (5a) for 4.6/S and 8.8/S categories and AS 1511 (5b) for 8.8/TF and 8.8/TB categories.

In AS 4100, the stength limit state design provisions which apply for static load applications are found in Clause 9.3.2. These provisions are summarized in Table 3. The design provisions follow directly from the strength of bolts discussed in Section 5.2.

5.3.2 Lap Connections

As discussed in Section 5.2.2, lap splice connections of the type shown in Fig. 3 require a modification when the applied forces result in shear forces on the bolts. The strength of such a bolted connection is related to the length of the joint.

Conventional theories of bolted connection design for joints involving more than one bolt subject to shear loading assume that rigid plate theory applies and that all the bolts are equally loaded. However, both elastic analyses and load measurements (16, 23) indicate that the longer the bolted joint becomes, the less uniform is the load distribution as long as the joint remains elastic.

As a joint is loaded to the extent that either yielding of the plies or bolts or both occur, plastic deformations permit a redistribution of load and a more uniform load distribution will result if the redistribution is allowed to proceed without premature failure of either bolt or plies.

The relationship used in Clause 9.3.2.1 of AS 4100 is as given in Table 4. The length of the joint is L_j which is indicated in Fig. 3.

TABLE 4

REDUCTION FACTOR FOR LAP CONNECTIONS (k_r)

Length mm	$L_j < 300$	$300 \leq L_j \leq 1300$	$L_j > 1300$
k_r	1.0	$1.075 - L_j/4000$	0.75

TABLE 3

AS 4100 CLAUSE 9.3.2 PROVISIONS
STRENGTH LIMIT STATE — STATIC LOADS

LIMIT STATE	CLAUSE	DESIGN CAPACITY
Shear	9.3.2.1	$V_f^* \leq \phi V_f$ V_f^* = design shear force V_f = $0.62 f_{uf} k_r A_v$ = nominal capacity in shear ϕ = capacity factor = 0.8 f_{uf} = minimum tensile strength of bolt = 400 MPa Grade 4.6 to AS 1111 (2) = 830 MPa Grade 8.8 to AS 1252 (6) k_r = reduction factor for bolted lap connections (see 5.3.2). For all other connections, k_r = 1.0 A_v = available bolt shear area (see 5.3.3) For a single bolt threads included, $A_v = A_c$, core area. For a single bolt threads excluded, $A_v = A_o$, shank area
Tension	9.3.2.2	$N_{tf}^* \leq \phi N_{tf}$ N_{tf}^* = design tension force N_{tf} = $A_s f_{uf}$ = nominal capacity in tension ϕ = capacity factor = 0.8 A_s = tensile stress area
Shear and Tension	9.3.2.3	$\left[\frac{V_f^*}{\phi V_f}\right]^2 + \left[\frac{N_{tf}^*}{\phi N_{tf}}\right]^2 \leq 1.0$
Ply in Bearing	9.3.2.4	$V_b^* \leq \phi V_b$ V_b = nominal capacity of a ply in bearing $V_b \leq 3.2 d_f t_p f_{up}$ (local failure) $\leq a_e t_p f_{up}$ (tearout failure) ϕ = 0.9 d_f = bolt diameter t_p = thickness of the ply a_e = minimum distance from the edge of a hole to the edge of a ply in the direction of the component of force plus half the bolt diameter f_{up} = tensile strength of the ply

5.3.3 Shear Areas

Bolted connections subject to shear may be either installed with the threads of the bolt crossing the shear plane or with the plain shank of the bolt crossing the shear plane. Figures illustrating these conditions may be found in Section 6 (Figs. 9 and 10). In a joint with a number of shear planes, some shear planes may cross the threaded part of the bolt while other shear planes may cross the shank.

Clause 9.3.2.1 (AS 4100) recognizes that the strength of the bolt across any shear plane is dependent upon the available shear area of the bolt at that plane. It allows for all possible combinations by defining the shear area as

$$A_v = n_n A_c + n_x A_o$$

where:— A_c = core area (see Section 3.1)

A_o = plain shank area (see Section 3.1)

n_n = number of shear planes with threads intercepting the shear plane

n_x = number of shear planes with shank intercepting the shear plane

Usually either n_n = 1 and n_x = 0 when there are two plies and threads intercept the shear plane OR n_n = O and n_x = 1 when there are two plies and the shank intercepts the shear plane.

5.3.4 Filler Plates

Where filler plates exceed 6 mm but are less than 20 mm in thickness, the nominal shear capacity V_f specified in Table 3 is required by Clause 9.3.2.5 of AS 4100 to be reduced by 15%. This provision is based on research reported in Reference 16.

5.3.5 Ply in Bearing

Equating the two design capacities for the ply in bearing limit state—one for local bearing failure, the other for plate tearout—results in the conclusion that plate tearout failure occurs in preference to local bearing failure whenever

$a_e \leq 3.2\ d_f$

End distances are normally of the order of 1.75-2.0 d_f so that plate tearout is the criterion of concern for end distance, whereas bolt pitches are typically 3.5-4.0 d_f so that between holes plate tearout is not normally of concern.

5.3.6 Surface Coatings on Plies

At the strength limit state, all bolting categories are assumed to have slipped into bearing against hole sides. The surface coating only affects the load at which slip first takes place so that the surface condition of the plies does not affect the strength limit state. Generally, the ply surfaces only need to be clean and defect free.

For the serviceability limit state, the surface condition of the plies is of major concern—see Section 5.4.5.

5.4 SERVICEABILITY LIMIT STATE

5.4.1 General

Under certain conditions, a bolted connection which does not slip under the serviceability loads may be advisable. This type of connection is known as a friction-type joint and is identified as 8.8/TF bolting category.

AS 4100 (5c) contains design requirements for this type of bolted connection, provisions that are essentially the same as those formerly contained in AS 1511 (5b) where slip was prevented under working loads.

The no slip requirement applies under the serviceability loads—it would be totally unrealistic to have no slip under the strength limit state loads—but a separate check is also required by AS 4100 for the strength limit state.

The design requirements of AS 4100 for 8.8/TF bolting category are summarized in Table 5.

On engineer's drawings the joints should be clearly labelled with the bolting categories 8.8/TF or 8.8/TB. In the absence of any such designation, the fabricator must assume such joints to be friction-type 8.8/TF joints.

5.4.2 Actual Slip

With the bolt hole clearances permitted by AS 4100—see Section 7.1—the maximum amount of slip that can occur with a single bolt in a single hole is 2-3 mm. In actual connections, as the number of bolts in a connection increases, so the potential for slip decreases since the normal inaccuracies in fabrication and erection mean that some bolts in the connection are most likely in bearing mode even before the connection is loaded.

Slip only needs to be restricted where such slip affects the serviceability or behaviour of the structure. Such instances are rare and are mostly restricted to cases of continual reverse loading or fatigue loading.

5.4.3 Initial Bolt Tension

There can be considerable variation in the level of bolt tension possible, unless control is exercised on the bolt installation procedures. The procedures of AS 4100 for bolt installation—see Section 8.3—are intended to ensure that a reliable level of installed bolt tension is achieved so that the design provisions against serviceability slip are themselves reliable.

5.4.4 Hole Types

Different hole types—round, short slotted, long slotted and oversize—are permitted by AS 4100 (see discussion in Section 7.1).

All of the hole types except the standard round hole with 2-3mm clearance may cause a loss of clamping force in the vicinity of the bolt because of loss of area due to the bigger hole. The clamping force is highly localized around the hole and any loss of area has a significant effect on the tension achieved, which in turn affects the slip resistance at the interface.

The factor k_h is intended to compensate for this effect and the values used in AS 4100 are based in recommendations made in Reference 16.

TABLE 5

AS 4100 CLAUSE 9.3.3 PROVISIONS
SERVICEABILITY LIMIT STATE — STATIC LOADS

LIMIT STATE	CLAUSE	DESIGN CAPACITY
Shear	9.3.3.1	$V_{sf}^* \le \phi V_{sf}$ V_{sf}^* = design shear force ϕ = capacity factor = 0.7 V_{sf} = nominal shear capacity — serviceability limit state = $\mu n_{ei} N_{ti} k_h$ μ = slip factor = 0.35 clean as-rolled surfaces or determined by testing in accordance with Appendix J of AS 4100. n_{ei} = number of effective interfaces N_{ti} = minimum bolt tension at installation (see Table 13) k_h = factor for different hole types = 1.0 for standard holes = 0.85 for oversize holes = 0.85 for short slotted holes = 0.70 for long slotted holes
Shear & Tension	9.3.3.3	$\frac{V_{sf}^*}{\phi V_{sf}} + \frac{N_{tf}^*}{\phi N_{tf}} \le 1.0$ N_{tf}^* = design tension force N_{tf} = nominal tension capacity of the bolt = N_{ti}, the installed bolt tension (see Table 13)

5.4.5 Contact Surface Condition

The value of the friction coefficient is highly dependent on the condition of the contact or faying surfaces. The value of the friction coefficient should be determined using a test procedure laid down in Appendix J of AS 4100. The value used in AS 4100 for bare steel surfaces (0.35) is one in long use in Australia (e.g. in AS 1511) and is supported by tests and by Reference 16.

For friction-type joints (8.8/TF category) the condition of the faying (or contact) surfaces is of prime concern, *since the slip factor achieved in practice is directly related to the condition of the faying surfaces*. The slip factor 0.35 given for design purposes in AS 4100 assumes faying surfaces of bare steel to bare steel — i.e. in the "as-rolled" condition.

Often steel members are painted or galvanized and it is important to know what influence this may have on the slip factor. Typical values of the slip factor for various surface preparations are given in Table 6.

TABLE 6

SUMMARY OF SLIP FACTORS

Surface Treatment	Average Slip Factor
Uncoated	
Clean as-rolled	0.35
Flame cleaned	0.48
Abrasive blasted	0.53
Painted	
Red oxide zinc chromate	0.11
Inorganic zinc silicate	0.50
Hot-dip galvanized	
Clean as-galvanized	0.18
Lightly abrasive blasted	0.30-0.40

The values of slip coefficient given in Table 6 should be considered as indicative only and actual values will vary within each generic type of surface treatment according to each manufacturer's own formulation. Major reliance has to be placed on the testing of each formulation using the method specified in Appendix J of AS 4100.

AS 4100 lays down a specific test procedure in Appendix J for establishing the slip factor for a given surface preparation and where the faying surfaces are coated, the slip factor should be determined in accordance with this procedure.

When the Code slip factor of 0.35 has been assumed in design, painted members normally need to be masked at the joints (Fig. 8), unless there is test evidence that the paint system to be used achieves at least this slip factor. Masking adds to the fabrication cost and is to be avoided if possible. One of the attractions of inorganic zinc silicate paint is its high slip factor, which means that no masking of faying surfaces is required.

Reference 16 reports on available test data related to the slip factor associated with faying surface finishes. Among the conclusions reported are:—

(i) Hot-dip galvanizing generally results in a low slip factor (typically 0.09-0.36, average 0.18) due to the soft zinc layer that acts as a lubricant. The slip factor is influenced by the method of pre-treatment prior to galvanizing, with abrasive blasting giving the highest slip factors and pickling the lowest.

(ii) A significant improvement in the slip reistance of galvanized surfaces can be achieved by pre-assembly treatment of the faying surfaces (e.g. wire brushing, light abrasive blasting, chemical treatment). Proper post-galvanizing treatment of hot-dip galvanized surfaces can also achieve a slip factor comparable to that for clean bare steel surfaces. Further testing is required to provide a better estimate of the slip coefficient achievable.

However, it is important to note that any post-galvanizing treatment is labour intensive and therefore costly. A more recent development is the introduction of galvanizing techniques which produce harder final layers of zinc. Extensive testing is now being undertaken to establish higher slip factors for steel processed in modern galvanizing facilities. Designers should check with the galvanizer before assuming a slip factor.

(iii) The slip factors achievable with zinc rich paints with organic vehicles would appear to vary markedly from one commercial mix to another, with some values comparable only to hot-dip galvanized surfaces.

(iv) Inorganic zinc rich paints provide a better slip resistance than zinc paints with organic vehicles. In particular, zinc silicate coatings on blast cleaned surfaces are likely to yield a slip coefficient which is about the same as that provided by blast cleaned base metal (approx. 0.50). Generally, an increase in coating thickness increases the slip resistance.

All the above comments relate to the short-term loading case of the type tested in AS 4100. Under sustained loading, galvanized members have a tendency to continue to slip (or creep) and this is not significantly improved by pre-assembly treatment. Joints treated with organic zinc rich paint show essentially the same behaviour, while inorganic zinc rich paints perform better, generally exhibiting similar slip factors for sustained as for short-term loading.

Also of interest is the fact that the fatigue strength of coated joints is equal to or greater than the fatigue resistance of uncoated joints of similar dimensions (16).

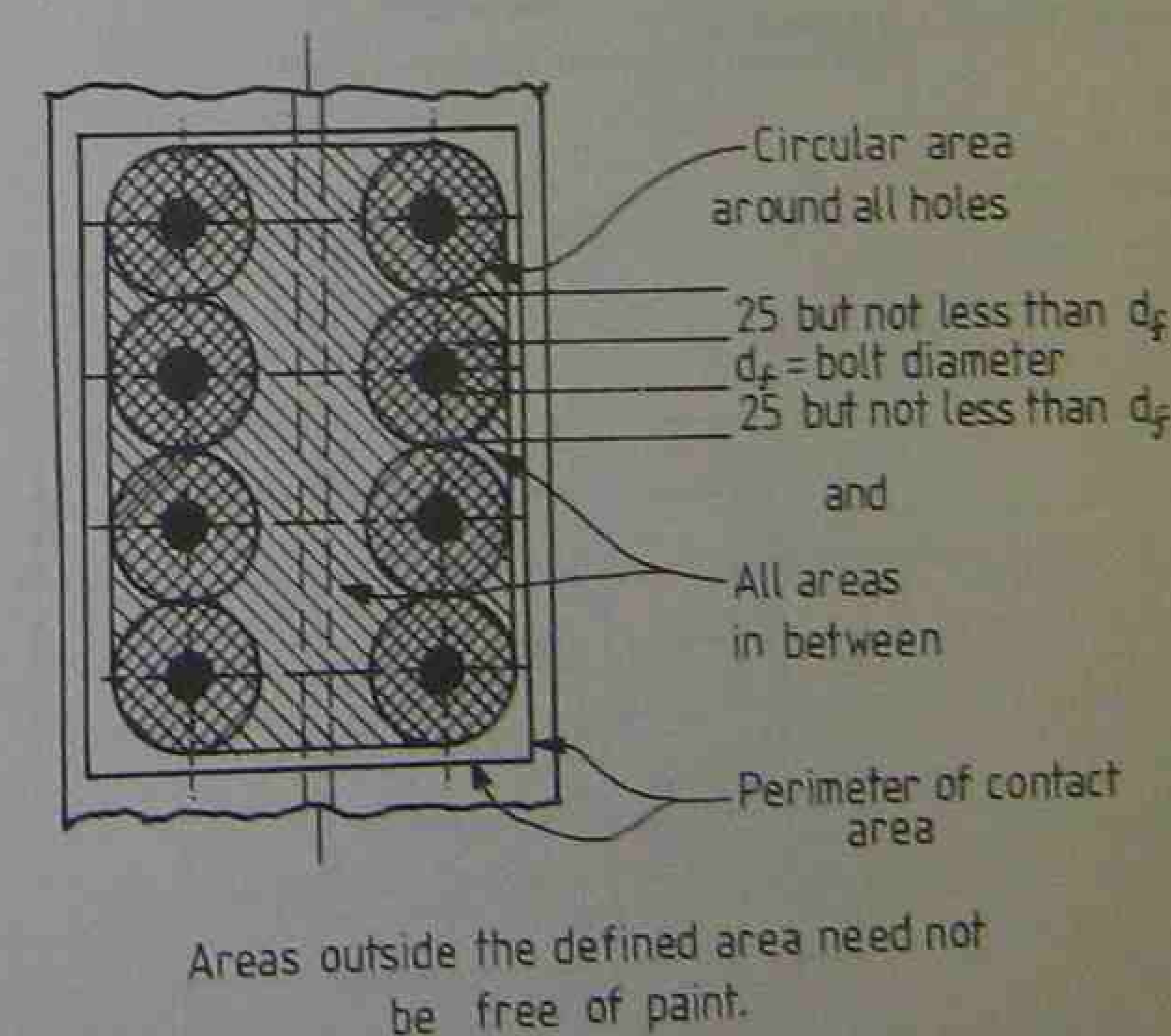


Fig. 8 — Area Requiring Masking (Ref. 19)

5.5 FATIGUE

AS 4100 (5c) also contains provisions for the design of bolted connections subject to fatigue. These provisions are different to those in previous codes such as AS 1250 (5a) and AS 1511 (5b).

It is not possible to review here the fatigue provisions of AS 4100 Section 11 as a commentary is available (5d). Essentially, a detail category is assigned to bolted connections subject to normal stress (tension) and shear stress. This detail category is a number which corresponds to the fatigue strength at 2×10^6 cycles on the appropriate S-N curves, a different S-N curve being used for each detail category.

The detail category takes into consideration the local stress concentration at the detail, the size and shape of maximum acceptable discontinuity for the loading conditions, metallurgical defects and residual stress.

For bolts, AS 4100 provides two detail categories, namely —

detail category 100 — bolts in shear, 8.8/TB bolting category where shear stress must be calculated on the core area A_c.

detail category 36 — bolts in tension, tensile stress being calculated on the tensile stress area, A_s. Additional tension forces due to prying must be taken into account.

For normal stress (tension), the uncorrected fatigue strength (f_f) for detail category 36 subject to n_{sc} cycles of loading or stress is given by

$$f_f^3 = \frac{36^3 \times 2 \times 10^6}{n_{sc}} \qquad \text{when } n_{sc} \le 5 \times 10^6$$

$$\text{or } f_f^5 = \frac{36^5 \times 10^8}{n_{sc}} \qquad \text{when } 5 \times 10^6 < n_{sc} \le 10^8$$

This relationship is shown in Fig. E.1 of Appendix E.

For shear stress, the uncorrected fatigue strength (f_f) for detail category 100 subject to n_{sc} cycles of loading or stress is give by

$$f_f^5 = \frac{10^5 \times 2 \times 10^6}{n_{sc}} \qquad \text{when } n_{sc} \le 10^8$$

This relationship is shown in Fig. E.1 of Appendix E.

For bolts subject to shear force, 4.6/S and 8.8/S bolting categories have no guidance in AS 4100 Section 11 and only bolting categories 8.8/TF and 8.8/TB are recommended. As no slip occurs with category 8.8/TF, no separate design for fatigue of the bolts is required. AS 4100 does contain design provisions for 8.8/TB bolting category.

For bolts subject to tension force, bolting categories 4.6/S and 8.8/S are not recommended and categories 8.8/TF and 8.8/TB are recommended.

The base metal or plies at the bolted connections requires separate assessment, with only the one detail category being relevant:—

Detail Category	Description
140	Bolted lap splices or beam splices

AS 4100 does not contain design provisions for bolts subject to combined shear and tension under fatigue conditions.

Reference 16 contains a review of research on fatigue in bolted connections.

Reference 5d contains a commentary on the above provisions and guidance on where additional information may be found.

6. Bolt Length Selection

6.1 PLAIN SHANK LENGTHS

Plain shank bearing lengths for each bolt type are defined in the relevant Australian Standards (AS 1111 and 1252) as the distance from the bearing surface of the bolt head to the last scratch of the thread.

Using the notation in these Standards (refer also Figs 9, 10):—

ℓ_s = plain shank length
$= \ell - b - a$

where,
ℓ = nominal length of bolt
b = length of thread
a = thread runout

In practice the value of plain shank length ℓ_s depends upon the manufacturing tolerances for b and a, so that:—

$$\ell_{s(max)} = \ell - b_{min} - a_{min}$$
$$\ell_{s(min)} = \ell - b_{max} - a_{max}$$

Values of $\ell_{s(min)}$ are given in Tables 7 and 8 for M20 bolts in commonly used lengths of commercial and high strength types respectively.

Values of b and a may be found in Tables A.1 and A.2 of Appendix A for bolts to AS 1111 and AS 1252 respectively.

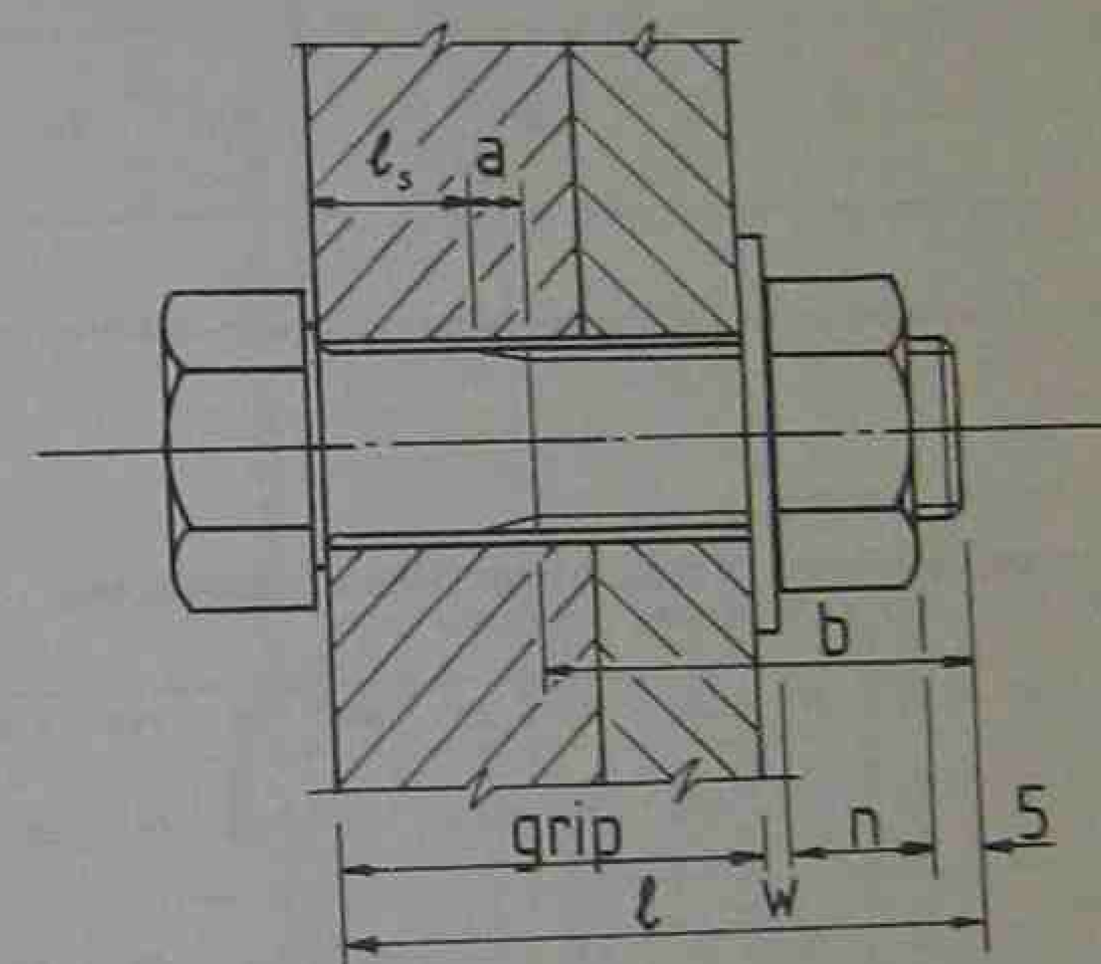


Fig. 9 — Threads included in shear plane.

LEGEND:—
a = thread runout
b = length of thread
ℓ_s = plain shank length
ℓ = nominal bolt length
n = nut height
w = washer thickness

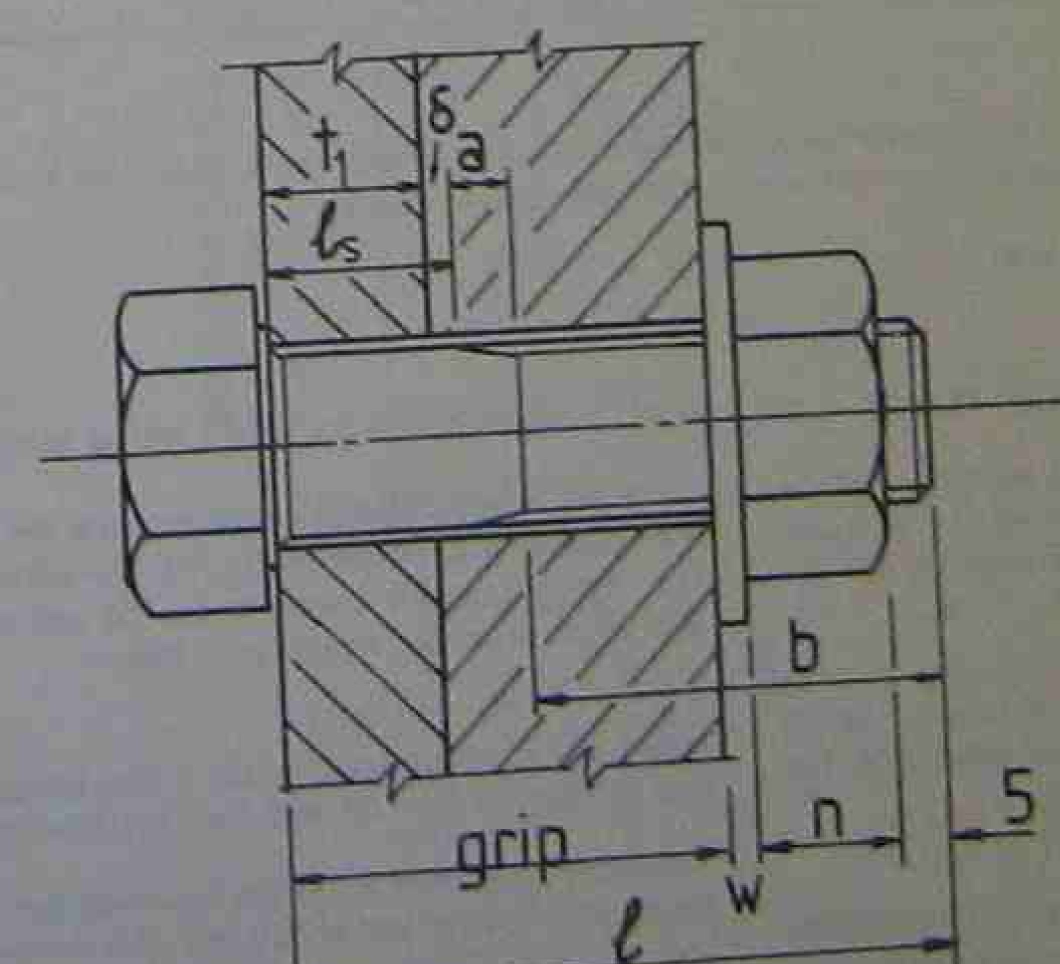


Fig. 10 — Threads excluded from shear plane.

TABLE 7

SHANK LENGTHS AND PERMISSIBLE GRIPS — THREADS INCLUDED IN SHEAR PLANE M20 COMMERCIAL BOLTS (AS 1111)

Bolt length (mm)	40	45	50	55	60	65	70	80	90	100
Classification	Screws					Bolts				
Minimum plain shank lengths ℓ_s (Note 2)	–	–	–	–	–	6.5	11.5	21.5	31.5	41.5
Average minimum grip (Note 3)	10	10	10	10	10	19	24	34	44	54
Average maximum grip (Notes 1 & 3)	14	19	24	29	34	39	44	54	64	74

Notes to Table 7:

(1) The Table assumes a minimum of 2 threads (5 mm) projection through the nut and one washer installed under the nut — see Section 6.2.

(2) From Table 3 of AS 1111.

(3) n_{ave} taken as 18, w_{ave} taken as 3, b_{nom} taken as 46.

TABLE 8

SHANK LENGTHS AND PERMISSIBLE GRIPS — THREADS INCLUDED IN SHEAR PLANE M20 HIGH STRENGTH STRUCTURAL BOLTS (AS 1252)

Bolt length (mm)	45	50	55	60	65	70	75	80	85	90	100	110	120
Minimum plain shank length ℓ_s (Note 3)	10	10	10	10	11.5	16.5	21.5	26.5	31.5	36.5	46.5	56.5	66.5
Average minimum grip (Notes 2 & 4)	10	10	10	14	19	24	29	34	39	44	54	64	74
Average maximum grip (Notes 1 & 4)	15	20	25	30	35	40	45	50	55	60	70	80	90

Notes to Table 8:

(1) The Table assumes a minimum of 2 threads (5 mm) projection through the nut and one washer installed under the nuts — see Section 6.2.

(2) Deduct one washer thickness from grip if additional washer used under head (4.6 max, 3.4 min thickness).

(3) From Table 2.1 of AS 1252.

(4) n_{ave} taken as 21, w_{ave} taken as 3, b_{nom} taken as 46.

6.2 THREADS INCLUDED IN SHEAR PLANE

For the case of **threads included in the shear plane**, (Fig. 9) the average maximum grip (assuming a 5 mm projection of threads through the nut) is given by:—

average maximum grip $= \ell - 5 - n_{ave} - w_{ave}$

The absolute maximum grip would be $(\ell - 5 - n_{min} - w_{min})$ but for detailing purposes it is not possible to presuppose the n and w values applicable. The average minimum grip is assumed to be:—

average minimum grip $= \ell - b_{nom}$

Tables 7 and 8 list the average range of grips for M20 bolts for the case where threads are included in the shear plane for commercial and high strength bolts respectively.

6.3 THREADS EXCLUDED FROM SHEAR PLANE

For the case of **threads excluded from the shear plane**, the situation is as shown in Fig. 10. The critical dimension is t_1, the thickness of the ply under the bolt head, and the bolt length must be chosen such that:—

$t_1 \leq \ell_s - \delta$

where δ is an allowance (usually 3 mm) designed to cover any variation in ply thickness from the nominal thickness.

If a washer is used under the bolt head (for cases where the head is rotated rather than the nut) then:—

$t_1 \leq \ell_s - \delta - w$

where w is the washer thickness.

TABLE 9

BOLT LENGTHS FOR M20 BOLTS TO AS 1252 — THREADS EXCLUDED FROM SHEAR PLANE

Bolts above the line to be used with minimum of 1 washer under nut (* indicates 2 washers required)

GRIP (mm), grips 10–48:

MINIMUM EXTERNAL PLY (mm)	10	12	14	16	18	20	22	24	26	28	30	32	34	36	38	40	42	44	46	48
6	60	60	65	65	65	70	70	75	75	75	80	80	85	85	85	90	90	90	100*	100*
8				65	65	70	70	70	75	75	80	80	80	85	85	90	90	90	100*	100*
10						65	70	70	70	75	75	80	80	80	85	85	90	90	90	100*
12								70	70	70	75	75	80	80	80	85	85	90	90	90
14										70	70	75	75	80	80	80	85	85	90	90
16												70	75	75	80	80	80	85	85	90
18														75	75	80	80	80	85	85
20																75	80	80	80	85
22																		80	80	80
24																				80
26																				
28																				
30																				
32																				
34																				
36																				
38																				
40																				

GRIP (mm), grips 50–90:

MINIMUM EXTERNAL PLY (mm)	50	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	82	84	86	88	90
6	100	100	100	110*	110*	110	110	110	120*	120*	120	120	130	130	130	140*	140*	140	140	140	150*
8	100	100	100	110*	110*	110	110	110	120*	120*	120	120	120	130	130	140*	140*	140	140	140	150*
10	100	100	100	100	110*	110	110	110	110	120*	120	120	120	120	130	130	140*	140	140	140	140
12	100	100	100	100	100	110	110	110	110	110	120	120	120	120	120	130	130	140	140	140	140
14	90	100	100	100	100	100	110	110	110	110	110	120	120	120	120	120	130	130	140	140	140
16	90	90	100	100	100	100	100	110	110	110	110	110	120	120	120	120	120	130	130	140	140
18	90	90	90	100	100	100	100	100	110	110	110	110	110	120	120	120	120	120	130	130	140
20	85	90	90	90	100	100	100	100	100	110	110	110	110	110	120	120	120	120	120	130	130
22	85	85	90	90	90	100	100	100	100	100	110	110	110	110	110	120	120	120	120	120	130
24	80	85	85	90	90	90	100	100	100	100	100	110	110	110	110	110	120	120	120	120	120
26		80	85	85	90	90	90	100	100	100	100	100	110	110	110	110	110	120	120	120	120
28				85	85	90	90	90	100	100	100	100	100	110	110	110	110	110	120	120	120
30						85	90	90	90	100	100	100	100	100	110	110	110	110	110	120	120
32								90	90	90	100	100	100	100	100	110	110	110	110	110	120
34										90	100	100	100	100	100	110	110	110	110	110	120
36												100	100	100	100	110	110	110	110	110	120
38														100	100	110	110	110	110	110	120
40																110	110	110	110	110	120

- When interpolating values, select longer bolt length.
- Table is valid for both single and multiple shear cases.

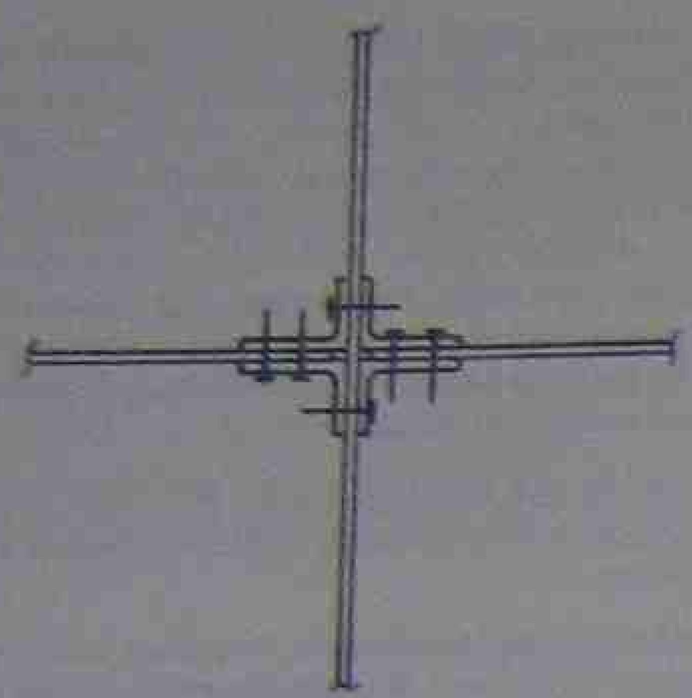


Fig. 12 — Bolt installation to avoid fouling.

(3) In joints with thin plies (e.g. 8 mm angle legs or 8 mm endplates), it is often necessary to use extra washers under the nut where threads are to be excluded from the shear plane in order to ensure that the nut does not run up to the end of the thread.

(4) Because the location of the plain shank relative to the shear plane position is critical for the threads excluded case, such a joint is very sensitive to bolt length selection. This means that bolts have to be selected usually in length increments of 5 mm and results in the stocking of a great number of different bolt lengths and the subsequent difficulty in distinguishing correct bolts for a particular joint on site. Alternatively excessive 'stick-through' must be accepted.

(5) In general structural steelwork, where each joint typically contains only up to 10 bolts, the design economies available using the threads excluded case appear to be outweighed by the subsequent detailing, fabrication and erection difficulties.

Considering that AS 1111 and AS 1252 have adopted the ISO metric bolt dimensions which provide long thread lengths, it is felt in general that designers are better to accept the reduced capacity penalty and design for threads included in the shear plane. *There should never be a reason for considering Grade 4.6 bolts with threads excluded — it is almost certain to be more economic to adopt 8.8/S procedure with threads included.*

However in large structures using Grade 8.8 bolts to AS 1252, and where joints contain upwards of 50 bolts, a good argument can be advanced for the design to be based on threads excluded. In structures such as major power stations where a total of 200,000 bolts is common, the savings can be substantial if consideration is given to alleviating the difficulties mentioned above. These savings result from fewer bolts, fewer holes to be drilled, smaller gusset dimensions and reduced installation time.

Furthermore on large projects there is generally much better control and supervision than on smaller jobs, and it must be stressed that good supervision is required to ensure that the correct bolt length is installed to provide for the threads excluded situation. It is necessary to identify the bolt length required for each joint and for an Inspector to ensure that the correct bolt length is used in that joint. Such checks are not necessary for the threads included case.

6.5 THREAD PROJECTION

AS 4100 requires that the length of a bolt be such that at least one clear thread project through the nut and that at least one thread plus the thread run-out is clear beneath the nut after tightening to either /S or /T bolting category. (Fig. 13)

The methods of calculation presented in Sections 6.1 through 6.3 meet these requirements.

The minimum projection throught the nut of about one thread pitch is intended to ensure that full engagement of the nut thread is achieved. While this is accepted good practice for /S bolting category, it is crucial with /T category in order to achieve the specified minimum bolt tension (see Section 8).

The clearance under the nut is intended to ensure that a nut is never run up to the thread run out on the bolt which constitutes the end of the threaded portion of the bolt. If the clearance is not provided, the nut will not sit firmly against the washer and, in the case of /T category, the necessary turn-of-nut may not have been achieved.

6.6 MATERIAL IN THE GRIP

AS 4100 requires that all material within the grip of the bolt is steel and that no compressible material such as gaskets or insulation be within the grip. This requirement is because predictable performance is only possible with material identical to that for which the design criteria was established.

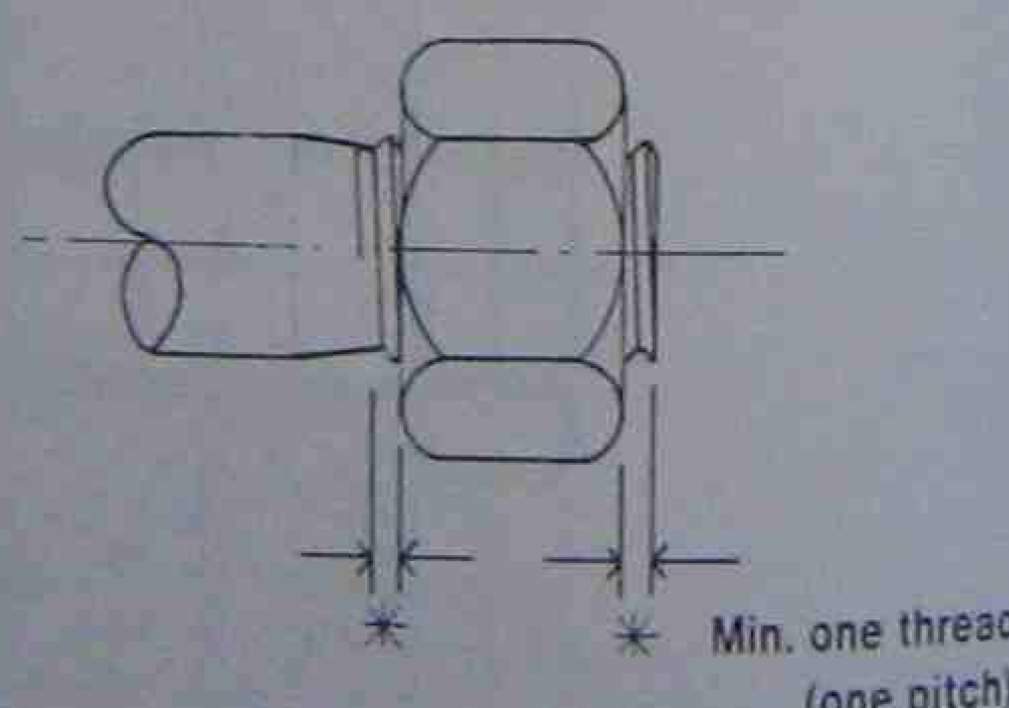


Fig. 13 — AS 4100 requirements for thread projection.

7. Detailing

7.1 BOLT HOLES

The diameter of bolt holes in bolted connections is stipulated in AS 4100 to be larger than the bolt diameter by either:—

2 mm for M24 bolts or smaller
3 mm for bolts larger than M24
6 mm for holes in base plates.

In some applications, the use of slotted or oversize holes may be justified in order to ease erection difficulties. The large oversize holes permitted in base plates is to assist in column erection and is related to the out-of-position of anchor bolts permitted in AS 4100.

AS 4100 also makes provision for the use of short and long slotted holes and oversize holes, and the detailed provisions for such holes are summarized in Table 10. Such provisions generally follow US practice (19) and (24).

The possibility of additional slip with oversize holes must be considered, as should the difficulty in maintaining beam levels in a steel-framed floor. In practice, within normal fabricating tolerances, the total movement in a bolt group using say 4 mm oversize holes will never be 4 mm, 2 mm being more likely. Oversize holes have the advantage that they provide the same clearance in all directions, while slotted holes can be used to make allowance for adjustment in a selected direction.

A review of the investigations of high strength bolted joints with oversize and slotted holes is provided by Kulak, Fisher and Struik (16). Relevant points from this reference are:—

(i) Oversize and slotted holes influence significantly the level of bolt preload, reducing the achieved bolt tension by about 15%. However, the use of hardened washers under the nut and bolt head improves the level of induced bolt tension achieved. The reduction in bolt tension results from the bolt depressing into the plate around the hole, resulting in the nut rotation not producing the degree of bolt elongation desired.

(ii) Immediately after a bolt is tightened, a loss of bolt tension occurs, probably due to creep and plastic deformation in the thread and plastic flow in the plies under the head and nut. Studies on bolts in holes with a standard clearance (2 mm) indicate a loss of from 5-10% of the initial induced bolt tension. The use of oversize or slotted holes has been shown to have no effect on the level of this loss.

(iii) Joints with oversize or parallel slotted holes may undergo substantial displacements once the slip resistance of a joint is exceeded.

(iv) The ultimate strength of a joint is not affected by either oversize or slotted holes.

TABLE 10
AS 4100 PROVISIONS FOR SLOTTED AND OVERSIZE HOLES
(d_f = nominal bolt diameter)

Hole Type	Maximum Size (mm)			Limitations
	General	M20	M24	
Short slotted	Width: $d_f + 2$ Length: $\ngtr$ 1.33 d_f or $d_f + 10$	22 27	26 32	May be used in shear connections. In friction-type joints, slots may be used without regard to direction of loading. In bearing-type, slots must be normal to the direction of the load, bolts must bear uniformly and the joint cannot be eccentrically loaded. May be used in any or all plies of both types provided hardened washers or plate washers are used under bolt head and nut.
Long slotted	Width: $d_f + 2$ Length: $\ngtr$ 2.5d_f	22 50	26 60	May be used in shear connections, but only in alternate plies. In friction-type joints, may be used without regard to direction of loading. In bearing-type, slots must be normal to the direction of the load, bolts must bear uniformly and the joint cannot be eccentrically loaded. Special washer or plate (≥8 mm thick) must cover all exposed long slotted holes.
Oversize	$\ngtr$ 1.25d_f $\ngtr d_f + 8$	25	30	May be used in any or all plies of bearing-type and friction-type connections provided hardened washers or plate washers are installed over the oversize holes.

Bolt holes may be machine flame cut or drilled full size for all grades of steel and all types of bolts, or alternatively, sub-punched 3 mm undersize in diameter and reamed to full size.

Punching of holes has become an economic method of holing structural members and, although some designers are still reluctant to permit punching, research and development in industry has proved the validity of relaxing previous restrictions. AS 4100 reflects this and Table 11 summarizes its requirements in regard to full-size punched holes.

The Commentary to AS 4100 (5d) offers two reasons for restricting the thickness for which full-size punching is permitted.

First, under competent fabrication practices, to avoid an excessively dished area in the immediate vicinity of the hole which may impair the strength of the joint. Second, and particularly in the case of dynamically loaded structures, to avoid metallurgical defects such as severe work-hardening which may impair the strength of the joint.

A slotted hole is either machine flame cut or punched with a die that matches the slot dimensions or formed by drilling two adjacent holes with machine flame cutting used to complete the hole.

Hand flame cutting is not permitted by AS 4100 except as a site rectification measure for holes in column base plate, where it is recognized that some inevitable site correction may be necessary. Hand flame cutting generally produces rough edges of unsatisfactory appearance, although evidence does exist that hand flame cut holes do not adversely affect the performance of the connection (see review of this in (5d)).

The limit on the thickness which may be punched is intended to restrict the amount of local deformation and work-hardening which may occur. The limits in AS 4100 are empirical, based on satisfactory results over 20 years and are the same as in previous codes AS 1250 (5a) and AS 1511 (5b).

TABLE 11
AS 4100 — FULL-SIZE HOLE PUNCHING LIMITATIONS

Bolting Category	4.6/S & 8.8/S	8.8/TF & 8.8/TB	
	Static loading	Static loading	Dynamic loading
Limit on thickness that can be punched	$\leq 5600/f_y$	$\leq 5600/f_y$ ($f_y \leq 360$ MPa)	12 mm
Max. thicknesses:— f_y = 250 MPa = 260 MPa	22.4 mm 21.5 mm	22.4 mm 21.5 mm	12 mm 12 mm

7.2 DETAILING LIMITATIONS

7.2.1 Minimum Edge Distance

Minimum edge distances from the centre of a bolt hole to the edge of a plate or the flange of a rolled section are specified in AS 4100.

These are:—

1.75d_f for sheared or hand flame cut edges

1.50d_f for machine flame cut, sawn or planed edges

1.25d_f for rolled edges of rolled sections

— where d_f is the nominal diameter of the fastener.

These minimum edge distances are based on past successful practice in (5a) and (5b), are related to the expected edge roughness, and are comparable to those in equivalent specifications (such as in (19)).

Table 12 lists these minimum edge distances for commonly-used bolt diameters.

AS 4100 also requires that the edge distance in the direction of a component of force, a_e, should be not less than $V_b^*/\phi t_p f_{up}$ (see Fig. 14 and Section 5.3). It is this requirement which may control the minimum edge distance in many cases.

TABLE 12
AS 4100 — MINIMUM EDGE DISTANCES

Nominal Diameter of Fastener mm	Sheared or Hand Flame Cut Edge mm	Rolled Plate; Machine Flame Cut, Sawn or Planed Edge mm	Rolled Edge of a Rolled Section mm
12	21	18	15
16	28	24	20
20	35	30	25
24	42	36	30
30	53	45	38
36	63	54	45

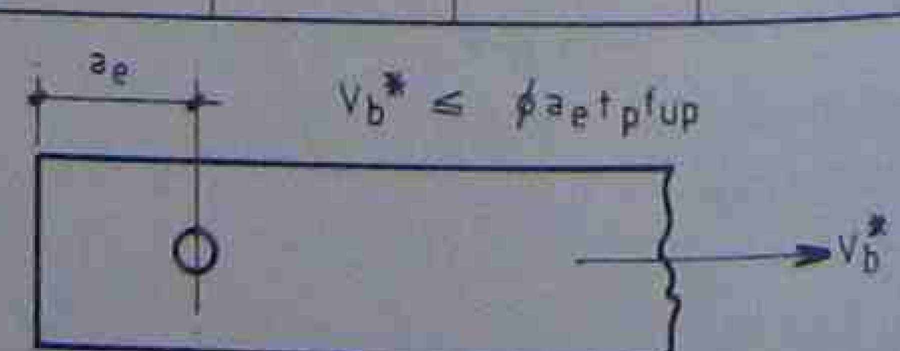


Fig. 14 — Edge distance and applied force

7.2.2 Maximum Edge Distance

AS 4100 specifies the maximum edge distance from the centre of a bolt to the nearest edge. This is defined as 12t_p or 150 mm, whichever is the lesser, where t_p is the thickness of the thinner outer ply.

The values specified for maximum edge distance are purely empirical, based on past successful practice. They are intended to exclude moisture between the connected plies, thus preventing corrosion between the plies, the products of which may force the plies apart. Lesser values should certainly be considered in corrosive applications. The provisions are also believed to prevent curling-up of plate edges (5d).

7.2.3 Minimum Pitch of Bolts

Minimum pitch of bolts is specified in AS 4100 as not less than 2.5 times the nominal diameter of the bolt. However, if it is intended to tension bolts with a special tensioning tool, the minimum distance between the centres of bolt holes shall be appropriate to the type of tool used.

The minimum pitch is actually more related to the tools required to install the fastener and most practical pitches are more like 3.5 times the bolt diameter (such as in (11)).

The minimum pitch may be governed by the AS 4100 bearing stress requirements (see Section 5.3 and Fig. 15). The end distance (a_e) is deemed to include the edge of an adjacent fastener hole. (Fig. 15) in AS 4100.

7.2.4 Maximum Pitch of Bolts

Maximum pitch of bolts is stipulated in AS 4100 as the lesser of 15t_p and 200 mm where t_p may be taken as the thickness of the thinner outside ply. However, in the following cases the maximum distances shall be:

(a) For fasteners which are not required to carry design actions in regions not liable to corrosion . . . the lesser of 32t_p and 300 mm.

(b) For an outside line of fasteners in the direction of the design action . . . the lesser of 4t_p + 100 mm, and 200 mm.

Maximum pitch of bolts must be observed as a safeguard against connected plates getting out of flat, and against the entry of moisture into the joint.

The values specified in AS 4100 for maximum pitch are empirically based on past successful practice, but closer pitches may be preferred if corrosion between the connected plies is considered to be a problem.

7.3 STANDARDIZED CONNECTIONS

In the AISC Standardized Connections (11), the following standard parameters are used for bolted connections of the simple or flexible type designed to transmit vertical shear force only:—

M20 bolts — bolt pitch 70 mm
— edge distance 35 mm
— hole diameter 22 mm
— gauge lines 70, 90, 140 mm

These parameters are used over a range of connections and have proved to be practical over a number of years. They meet all the requirements of AS 4100.

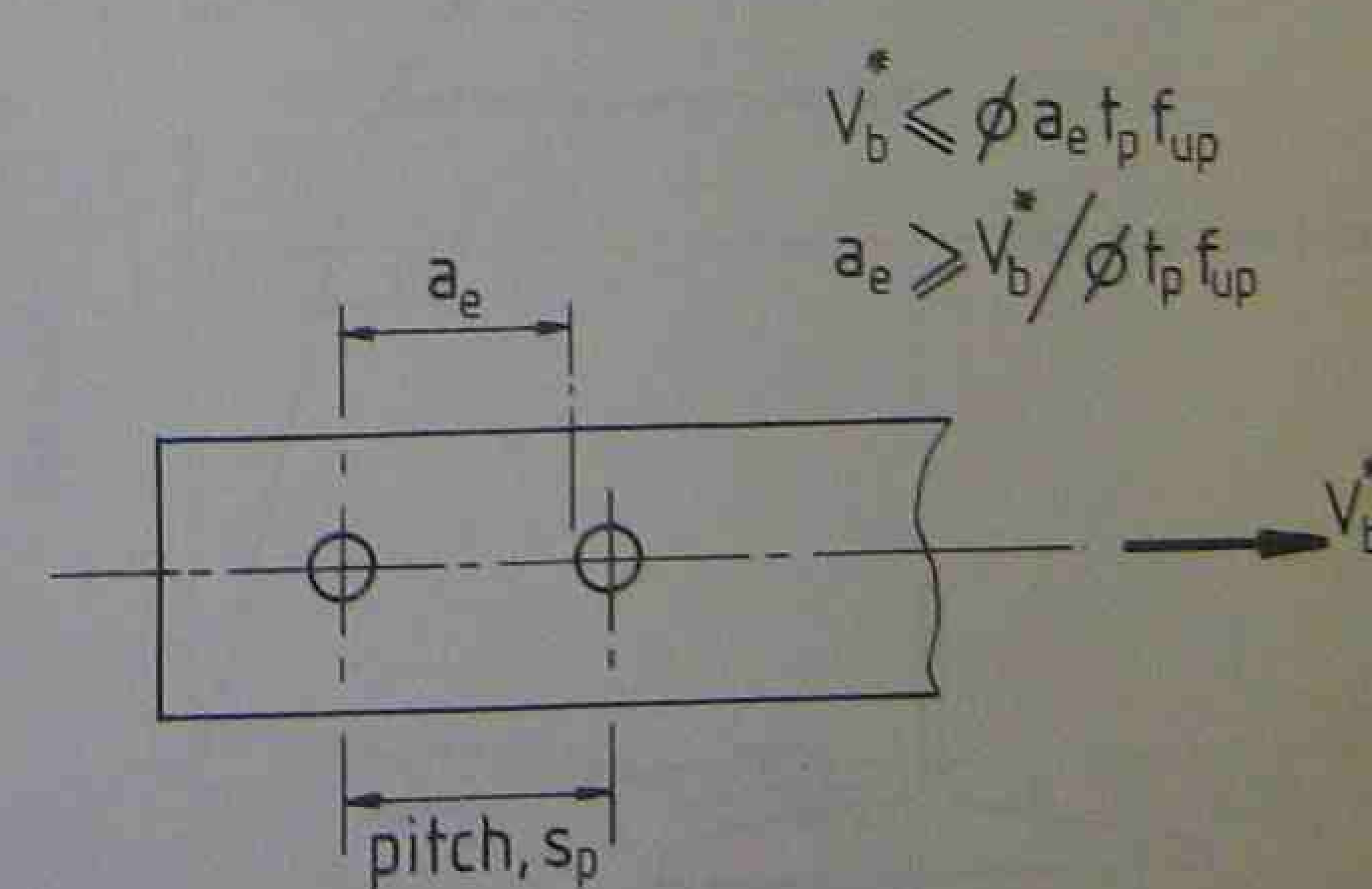


Fig. 15 — Pitch and bearing force design provisions

7.4 WASHER REQUIREMENTS

AS 4100 requires that a washer be used under the rotated part (usually the nut) for all bolting categories. This is common practice in any case but is particularly important for 8.8/T bolting category where the bolt is tensioned.

Where the slope of the surfaces between the parts in contact with the bolt head or nut and the bolt axis exceeds 1:20 (5%), then a tapered washer must be provided against the tapered surface and the non-rotating part must be placed against the tapered washer (Fig.17).

For slotted or oversize holes, hardened washers or plate washers must be used on each side under both bolt head and nut (see Table 10).

Where long slotted holes are used (i.e. up to 2.5 × bolt diameter in length), a special washer or cover plate must be installed to completely cover all of the slots. Such a washer or plate shall be at least 8 mm thick. In long joints it is often economical to provide such a cover plate from a rolled edge flat bar drilled to the nominal bolt centres (as in Fig. 16) and assembled as a single item.

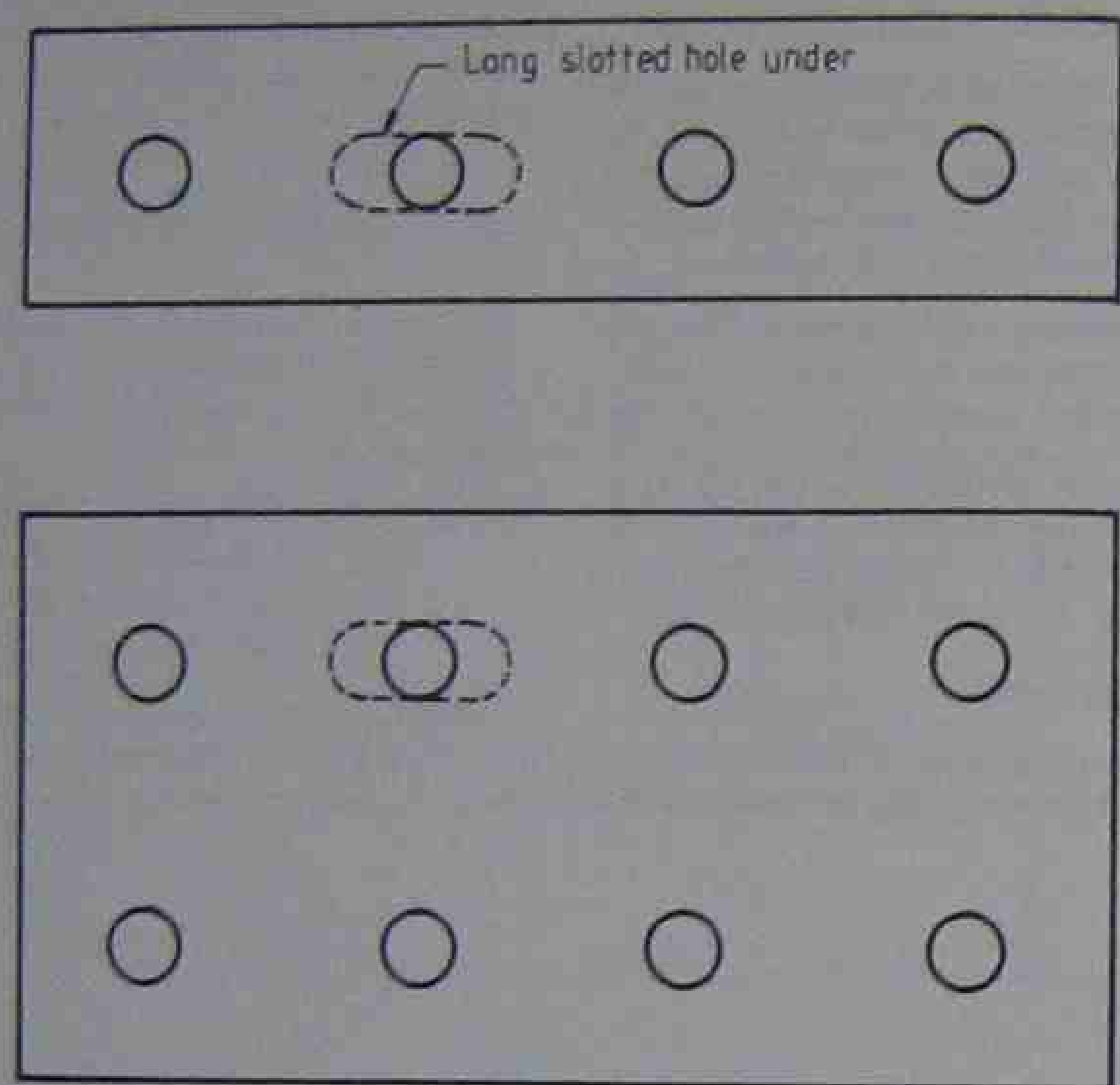


Fig. 16 — Cover plates for long slotted holes.

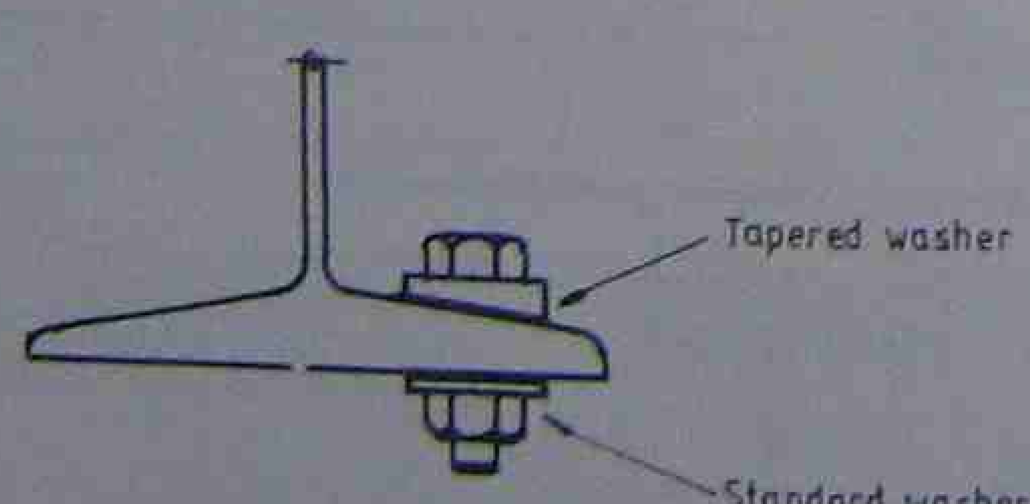


Fig. 17 — Use of a Tapered Washer

8. Installation of Bolts

8.1 INTRODUCTION

In structural bolted connections, the installation of bolts involves the initial lining up of the bolt holes, the insertion of the bolt/nut assembly and the subsequent tightening operation. It has been established that bolt installation can be at least as costly as the initial cost of the bolt/nut assembly, and so it is necessary to consider the installation operation in determining the overall economics of bolted structural connections. In practice, it is the level of bolt tightening which is the governing cost factor in bolt installation.

There are two levels of bolt tension required in structural joints, viz. (i) "snug tightening" — bolting categories 4.6/S and 8.8/S, and (ii) full tensioning to the requirements of AS 4100 — bolting categories 8.8/TF and 8.8/TB.

8.2 SNUG TIGHTENING

This level of bolt tightening is used to describe two situations:—

(a) a final level of bolt tightening in general bolting (such as bolting categories 4.6/S and 8.8/S);

(b) an intermediate level of tightening prior to full tensioning in high strength structural bolting (bolting categories 8.8/TF and 8.8/TB).

The term snug tightening has caused some confusion amongst designers and erectors alike. The term was introduced into the industry with the advent of high strength structural bolting in the 1950s since it is necessary to snug tighten each bolt in a joint to fully compact or bring together the individual plies of the joint before final tensioning is undertaken. Snug tightening is defined in AS 4100 as "the tightness obtained by the full effort of a man using a standard podger spanner". It is in reality the tightness that exists when all plies in a joint are in full contact. Typical tools could be of the order of 450 mm long for M20 bolts and about 600 mm long for M24.* Obviously this is not a very scientific definition, but in fact it describes fairly accurately the level of tightening that has always been attained in general bolting. The important point is that the actual bolt tension achieved in flexible joints using /S categories is not of significance — the behaviour of the bolt under applied loads is well known and accepted. Nor is the actual bolt tension achieved of significance when snug tightening

*Whilst the Code defines snug tightening in terms of hand tools, it can also be carried out using power tools, and this is often a more efficient method. A distinct change in note in the wrench indicates the achievement of snug tight.

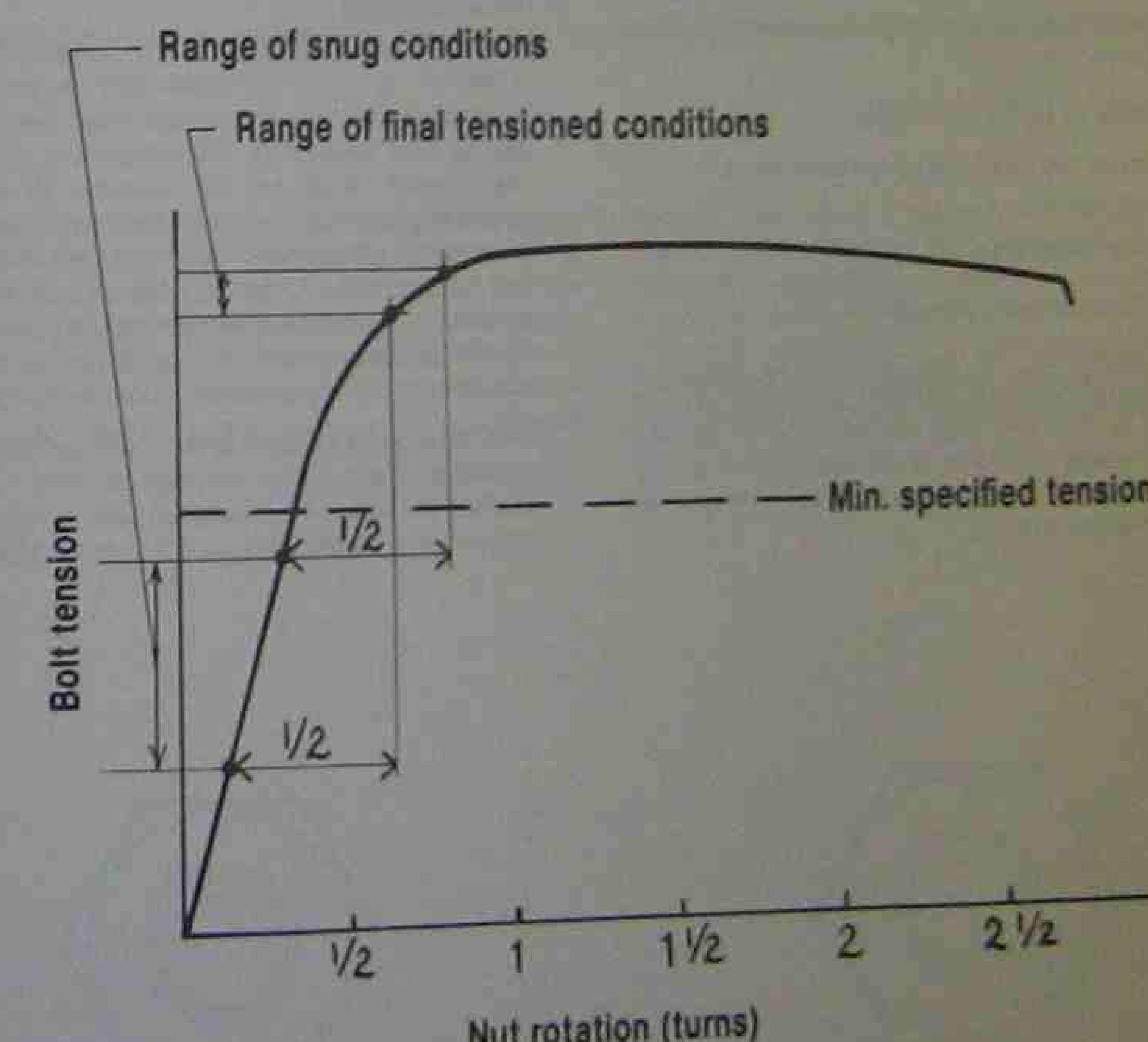


Fig. 18 — Bolt tension and turn of nut.

joints which are to be subsequently fully tensioned, for reasons explained below.

Snug tightening will induce small clamping forces which in practice will be of highly variable levels. This is illustrated in Fig. 18.

Snug tightening is defined as that mode of final bolt tightening always used for 4.6/S and 8.8/S categories. It is also the first step in the full tensioning by 8.8/TF and 8.8/TB categories.

8.3 FULL TENSIONING METHODS

For joints designed in accordance with AS 4100 either as friction- or bearing-type, bolts must be fully tensioned to minimum shank tensions stipulated in Table 13.

TABLE 13
MINIMUM BOLT TENSION IN AS 4100

Nom. Bolt Dia.	Minimum Bolt Tension (kN)
M16	95
M20	145
M24	210

These values are approximately equivalent to the minimum proof loads specified in AS 1252 (see Table A.2).

Tests have indicated that neither the tensile strength nor the shear strength of a bolt is affected by the level of initial tension in the bolt (16).

8.3.1 Methods of Full Tensioning

AS 4100 permits two methods of full tensioning:—

- Part turn of nut
- Direct tension indication.

8.3.2 Part Turn of Nut Method

Controlling tension by the part turn of nut method is primarily a strain control procedure and the desired tension in the bolt can be obtained with accuracy. Since the bolt tension — bolt elongation curve is relatively flat (Fig. 18), variations in the level of snug tightening result in only small variations in final bolt tension.

Tensioning of bolt/nut assemblies by the part turn of nut method should be carried out in accordance with the following procedure (from AS 4100):

(a) Snug tightening

On asssembly, all bolts in the joint must be tightened to the snug condition as described in Section 8.2.

It must be noted that snug tightening, as the first step in full tensioning for 8.8/TF and 8.8/TB categories is to ensure that the steel plies of a joint are fully compacted before final tensioning is undertaken. For this reason in large joints — say of 10 bolts and more — it is necessary to snug tighten from the stiffest part of the joint towards the outer or free edges. If this is not done, uniform compaction may not be achieved, which can lead to variations in final bolt tension and uncertainties in the performance of the joint. If it is not possible at the snug tightened stage to fully compact plies due to out-of-flatness at the interfaces, it is necessary to shim large distortion gaps (in excess of 2 mm) to prevent the energy of final tensioning being used to force the distorted plies together, rather than to induce clamping of the plies by bolt shank tension. With joints with large numbers of bolts two runs over the joint is suggested to check for full snug tight, as the gradual drawing in of the plies may cause those bolts snug tightened first to gradually loosen.

(b) Match marking

After snug tightening, location marks are established on both the nut face and bolt shank — Fig. 19 — which mark the relative positions of bolt and nut, as a control on the level of final nut rotation.

(c) Final tensioning

Nuts are finally rotated by the amount specified in Table 14. In some instances due to problems with accessibility to the nut, the bolt head may be rotated. During final tensioning, the component not turned by the wrench shall not be allowed to rotate. The tensioning operation should be carried out using a pneumatic impact wrench with sufficient torque to apply the full amount of rotation within about 10 seconds. Special impact-type sockets should be used, and these should be given some form of peripheral marking so that the amount of rotation can be observed.

Final tensioning by hand tools is difficult and costly but, if there is no alternative, they can be used. A heavy duty ratchet wrench will be required with a lever length of about 1 m for M20 bolts, and 1.2 m for M24 bolts.

Fig. 19 — Marking for part turn of nut method

TABLE 14
NUT ROTATION FROM THE SNUG TIGHT CONDITION — AS 4100

Bolt length (Underside of head to end of bolt)	Disposition of outer face of bolted parts (see Notes 1, 2, 3 and 4)		
	Both faces normal to bolt axis	One face normal to bolt axis and other sloped	Both faces sloped
Up to and including 4 diameters	1/3 turn	1/2 turn	2/3 turn
Over 4 diameters but not exceeding 8 diameters	1/2 turn	2/3 turn	5/6 turn
Over 8 diameters but not exceeding 12 diameters (see Note 5)	2/3 turn	5/6 turn	1 turn

NOTES:

1. Tolerance on rotation: for 1/2 turn or less, one-twelfth of a turn (30°) over and nil under tolerance; for 2/3 turn or more, one-eight of a turn (45°) over and nil under tolerance.
2. The bolt tension achieved with the amount of nut rotation specified in this Table will be at least equal to the minimum bolt tension specified in Table 13.
3. Nut rotation is rotation relative to the bolt, regardless of the component turned.
4. Nut rotations specified are only applicable to connections in which all material within the grip of the bolt is steel.
5. No research has been performed to establish the turn-of-nut procedure for bolt lengths exceeding 12 diameters. Therefore, the required rotation must be determined by actual test in a suitable tension measuring device which simulates conditions of solidly fitted steel.
6. A review of the research on which the values in this Table are based may be found in Reference 16.

Part-turn tensioning may occasionally induce too high a bolt tension in very short bolts, despite the reduction in the required amount of rotation indicated in Table 14. The occurrence of this condition manifests itself in an abnormal number of bolt breakages during tensioning. If such a condition arises, it may be necessary to establish a reduced degree of nut rotation by carrying out nut rotation-bolt tension tests using a calibration load cell. The reduced rotation values for short bolts given in Table 14 reflect this problem. The reliability of the method depends greatly upon correct snug tightening and upon accurately measuring the degree of nut rotation. It is primarily a method of control which relies on controlling bolt elongation. Some problems may be experienced with galvanized bolts (see Section 8.3.5).

8.3.3 Direct Tension Indication Method

Of the two methods permitted by AS 4100, the direct tension indication method provides the simplest procedure for achieving the specified minimum bolt tension of Table 13 while permitting relatively easy subsequent inspection of the installed bolts.

Several tension indicating devices have been developed to provide a means of checking whether the specified minimum bolt tension of Table 13 has been achieved. In Australia, at the present time, the two most commonly used of these devices are:

- Load indicator washers (used with conventional high-strength structural bolts)
- Swage lock fasteners (proprietary special fasteners).

(a) Load Indicator Washers (see photograph — Fig. 20)

The load indicator washer is a hardened steel washer carrying 4 to 7 protrusions, the actual number depending on the bolt diameter. The washer is normally assembled with these protrusions bearing against the underside of the bolt head leaving an initial gap (Fig. 21a). The bolts are then snug tightened, after which final tensioning is completed, until the gap between the washer face and bolt head is reduced to approximately 0.25 mm by flattening of the washer protrusions (Fig. 21b). At this gap, the bolt tension achieved will be at least the minimum specified in AS 4100 and Table 13. The gap is checked using ordinary feeler gauges, but after a little experience is readily judged by eye.

Where it is necessary, because of joint disposition, to tension bolts by rotating the bolt head, the load indicator washer must be assembled on the nut side of the joint with an additional nut face washer (Fig. 22). Under no circumstances should the load indicator washer be adjacent to the rotating part.

It is important to note that using this method of tensioning, it is still necessary to observe the two-stage tightening procedure, namely initial snug tightening to bring joint plies into effective contact, followed by full tensioning.

Comparing this procedure to the part turn of nut method, it is important to note that a slightly higher nut rotation will be required to produce the minimum specified bolt tension when using load indicator washers than that required without them. This is because in a joint without load indicator washers, all gaps are fully compacted steel-to-steel at the snug tightened point, whereas with the washer extra energy or nut rotation is required to squash the washer protrusions. Thus it is quite common to find cases where the nut rotation required to obtain the correct gap can be three quarters of a turn up to one full turn of the nut.

Fig. 23 indicates a typical relationship between washer gap, induced tension and the turn of the nut.

In reaching a nut rotation of one turn, no danger exists of breaking bolts — in tests breakage of the bolt occurs between 2¼ and 2½ turns typically.

The important thing to remember when using the load indication method of tensioning, is that it is not necessary to measure nut rotation. When the washer gap is reduced to 0.25 mm, then the specified minimum bolt tension has been achieved.

Fig. 20 — Load indicator washer.

(a)

(b)

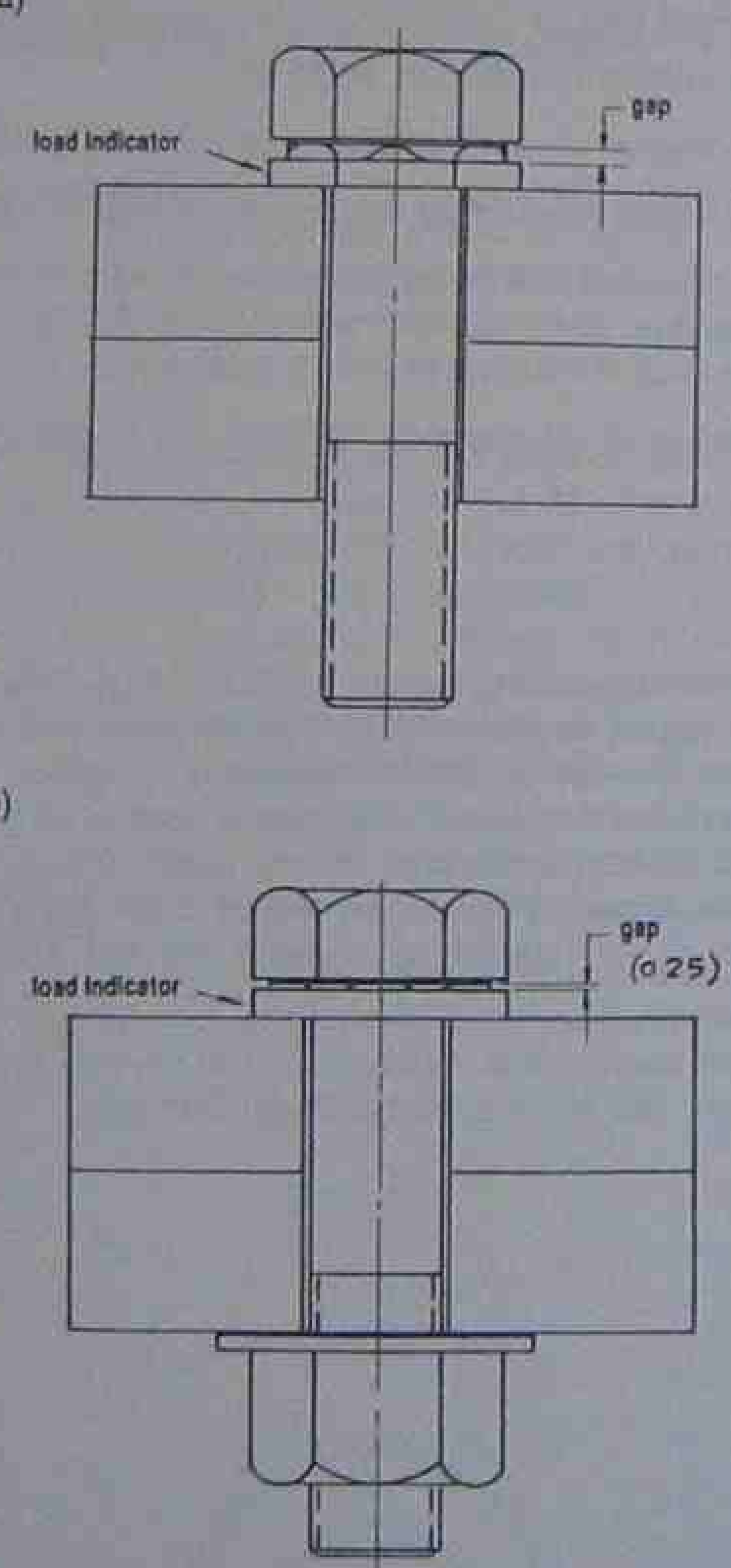


Fig. 21 — Use of load indicator washer.

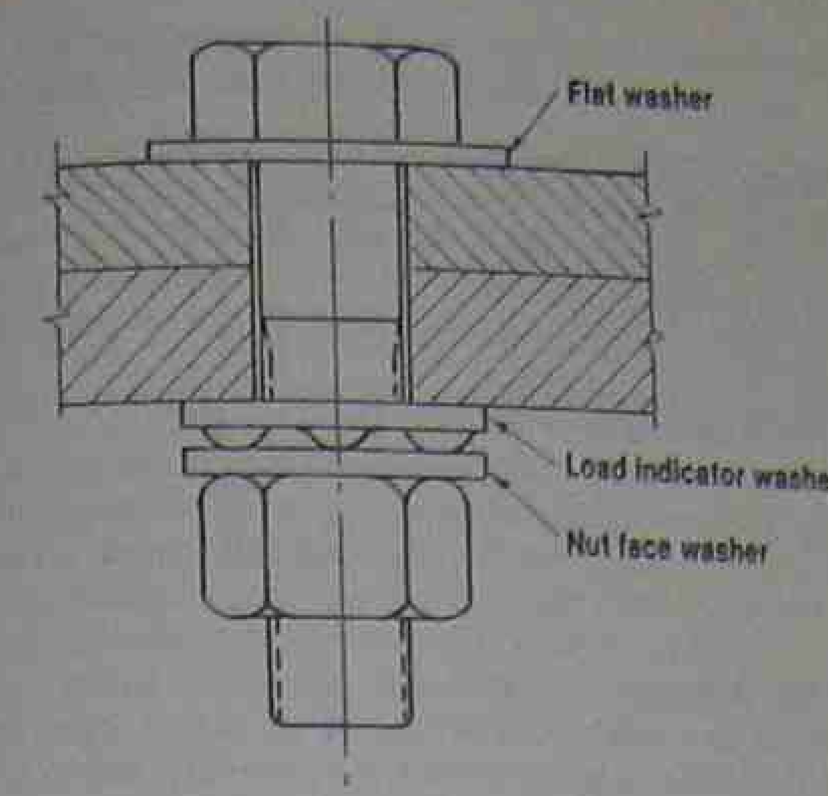


Fig. 22 — Location of load indicator washer when bolt is rotated.

When a structure is exposed to aggressive corrosive conditions it is recommended that the gap be just closed to prevent ingress of corrosive moisture. Again Fig. 23 indicates that a zero gap is achieved at a safe level of bolt tension.

Sometimes problems have been reported in the use of the load indicator washers. These are usually found to be associated with overall bolting procedures and only rarely with the load indicator washers themselves — see Section 8.3.5.

Some instances of cracked or broken washers were found during the introduction of galvanized washers. This was due to embrittlement of the washer material during galvanizing and was quickly solved by the galvanizing industry, which developed procedures to eliminate the problem.

Other cases of broken washers have been traced to the fact that the washer has insufficient clearance on the neck of the bolt and, during tightening, the resulting interference tended to burst the washer. It is most important to confirm that the washer has at least the usual 2 mm clearance on the bolt shank diameter at the junction of shank and bolt head. This is usually very easily checked by examining a random sample of bolts and washers in a batch, and is a necessary first step in establishing correct procedures which will lead to the efficient use of the method.

In spite of the additional cost of the load indicator washer itself, on many projects this method is economical. Experience has shown that the extra cost of washers is offset by large savings in the cost of installation and inspection of joints where bolts are required to be fully tensioned. Currently the cost of a load indicator washer represents about 30% of the total cost of a high strength bolt, nut and flat washer assembly.

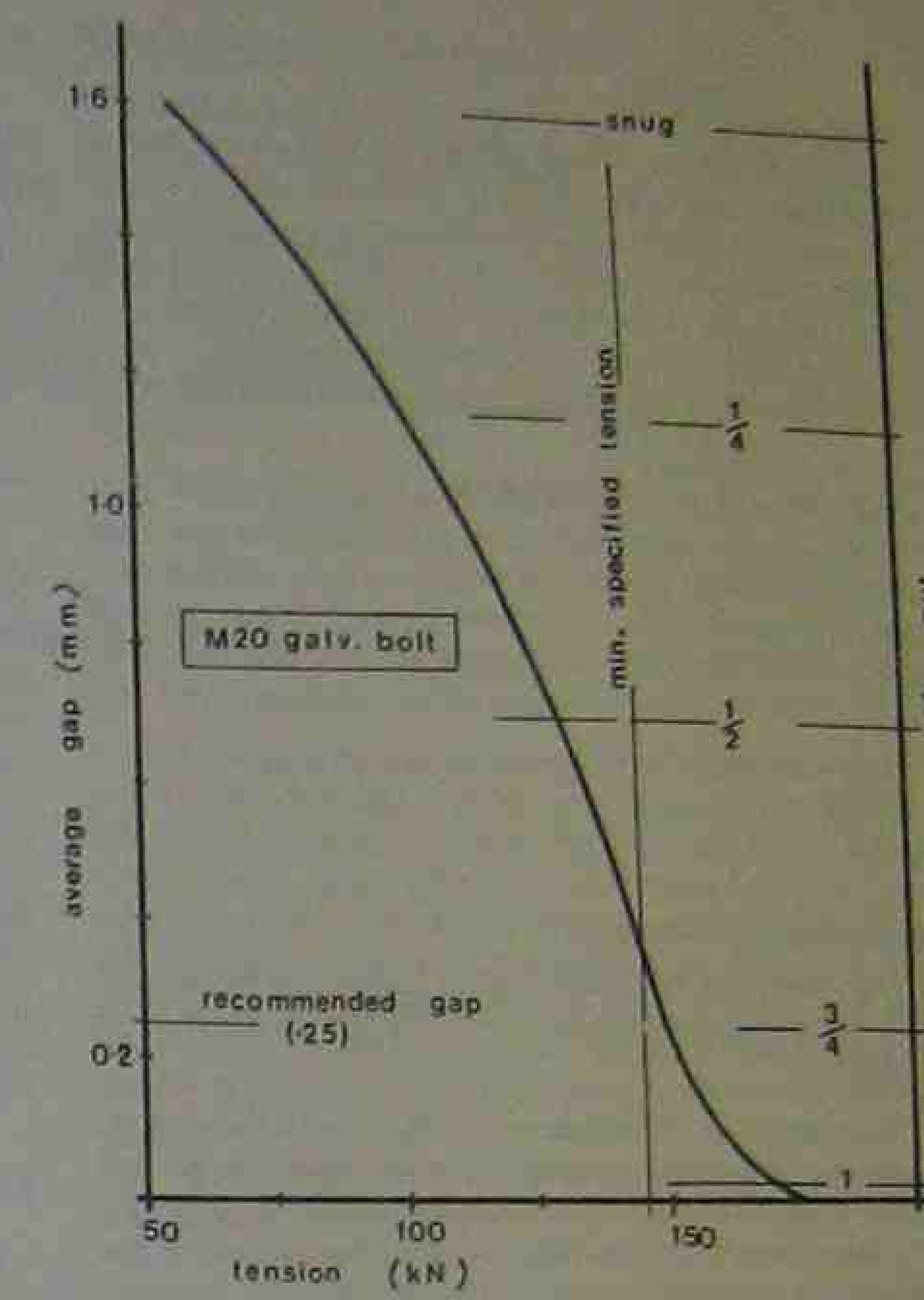


Fig. 23 — Relationship between bolt tension, load indicator gap and turn of nut.

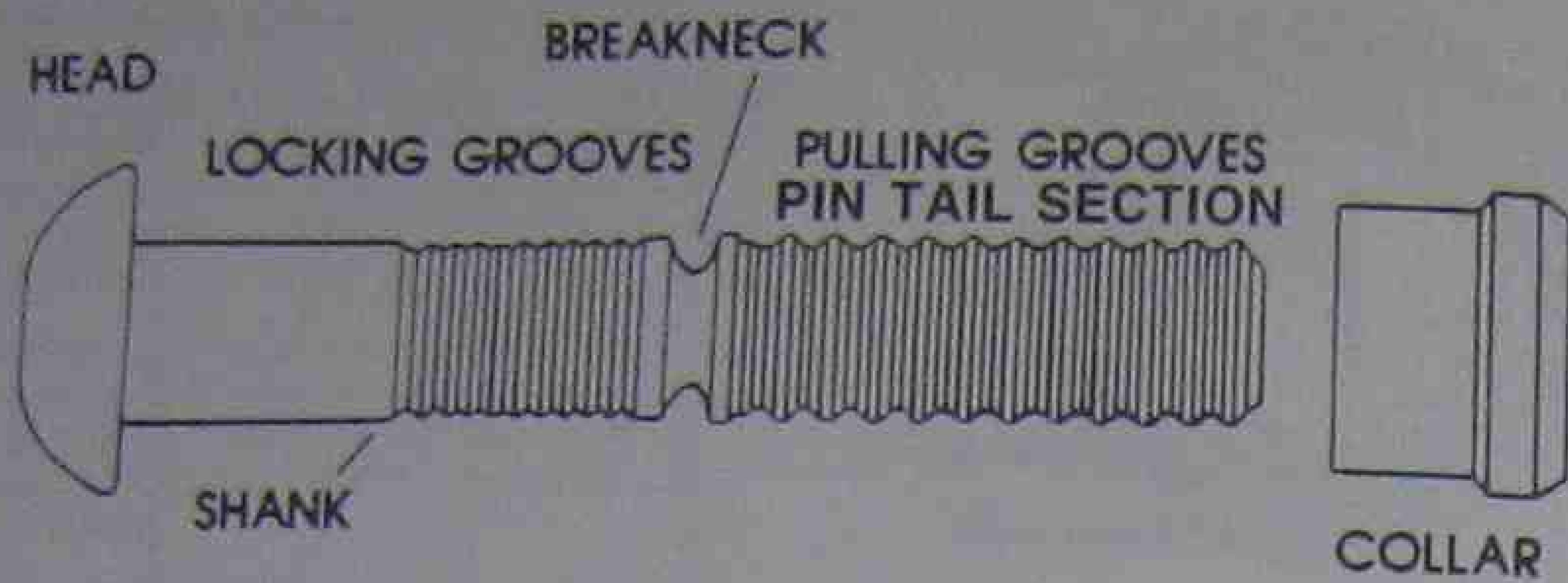


Fig. 24 — Swage lock fastener.

(b) Swage Lock Fasteners

The swage lock fasteners used in Australia at the present time are of the type shown in Fig. 24 and in Appendix B (Table B.2).

The bolt in Fig. 24 is a two-piece high strength fastener consisting of a grooved pin and locking collar. The pin has a forged head, shank, annular (not helical) locking grooves, a breakneck and pin tail. The collar has a plain bore in the pre-assembled state.

The fastener is installed by means of a special installation tool, the collar being swaged onto the locking grooves on the pin while the pin is subject to a tensile force. After the locking collar is swaged, the pin tail section breaks off at the breakneck when correct shank tension is reached. Thus a direct tension indication is provided within the fastener assembly.

This type of high strength structural fastening is of particular value where the prevailing working environment subjects structural connections to severe shock, vibration and dynamic loading.

The system is acceptable within the terms of AS 4100.

8.3.4 Tools

It is of the utmost importance to ensure that only proper tools are used for tensioning. If only hand tools are available, the correct size of podger spanner must be used to snug tighten, while the final tightening must be undertaken by the use of a heavy duty ratchet wrench with a lever length of about 1 m for M20 bolts (about 1.2 m for M24).

More economical bolting will generally result from the use of air impact wrenches. These can be used both to snug tighten and subsequently to fully tension the bolts. Obviously the air wrench must be of a suitable capacity for the bolt sizes under consideration and it is essential to specify the bolt size when ordering the air wrench. Similarly, the wrench will only be effective if a constant air supply at the right pressure is available. Experience shows that for M20 bolts, an air pressure of at least 700 kPa at the wrench is required to tension bolts in the recommended time.

In general, full tensioning of M20 bolts from the snug tight state should be completed in a maximum of 10 seconds with an air wrench. This ensures that the nut is progessively running on the bolt thread over the shortest practical time, and therefore inducing shank tension rather than becoming seized on the bolt thread due to heating over a long impacting time.

8.3.5 Condition of Bolts and Nuts

In the installation of fully tensioned high-strength bolts, the majority of problems are more often than not traced to the condition of the bolt and nut threads at the time of installation.

It is essential that good house-keeping be observed on site. Bolts should be stored in containers in a dry covered area until they are required. If this is not done, some corrosion may take place and while this is not of concern in the overall final behaviour of the bolt, it is critical in the tensioning procedure. Small amounts of corrosion in the thread region of either bolt or nut can create a relatively high coefficient of friction between these threads and this, during the tensioning, will induce high torsional stress in the assembly. The energy of the tensioning will therefore be going into overcoming thread friction rather than serving the real purpose of inducing shank tension.

When using load indicator washers in the assembly, this situation will be revealed by the full effort of the wrench seemingly having fully tightened the bolt but with the washer gap not closed to the recommended limit. This situation is frequently encountered when using plain uncoated bolts if thread corrosion has been allowed to occur.

For galvanized bolts, AS 1252 requires that the galvanized nuts be supplied with a lubricant coating. There is no doubt that this greatly assists in overcoming thread friction and in fact some bolt manufacturers apply a lubricant to the entire bolt after galvanizing. This lubricant of course has a limited life and once lost due to exposure to weather, galvanized fasteners exhibit similar thread friction problems even though no corrosion has taken place. The problem can be overcome by the application of a molybdenum disulphide lubricant at site. This should be applied sparingly to the entire thread length of the bolt and will invariably provide free running of nut on bolt.

The real answer to thread condition variation lies in protecting the bolts and nuts during storage on site.

Too many times bolts can be found in heaps on the site where not only moisture but also sand or cement can be seen on the threads. Keeping the bolts in bags or drums out of the weather for as long as possible will pay great dividends in the tensioning procedure.

8.3.6 Bolts driven into holes

In the assembly of structural joints, the members should be aligned correctly so that bolts can be installed without undue force. This is a particularly important when using fully tensioned bolts. A bolt which has been driven into mis-aligned holes will be difficult to tension correctly because the shank, including the threads, could be severely damaged during the driving. Obviously the damaged thread will cause interference to the nut during tensioning and may cause seizing of the nut on the bolt before the required bolt tension has been achieved.

The use of drift pins is recommended to overcome this problem. The drift pin is made from a bolt blank of the size being used and will align holes in steelwork far more accurately than normal podger spanners (Fig. 25).

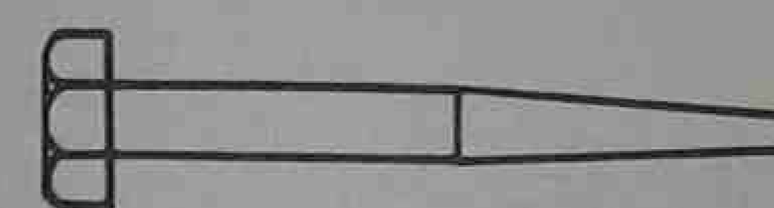


Fig. 25 — A drift pin made from a bolt

8.3.7 Relaxation

Tests have indicated that immediately following completion of full tensioning, there is a 2-11% decrease in bolt tension, the average decrease being 5%. This is believed to result from elastic recovery; creep and yielding of the bolt due to the high stress level might result in a minor relaxation as well (16).

The grip length and number of plies are believed to affect the degree of relaxation, and the extent of relaxation increases over time but most occurs in the first 24 hours. Reference 16 reports that the relaxation with galvanized bolts is twice that for plain bolts, with the amount of relaxation dependent upon the thickness of the galvanized coating.

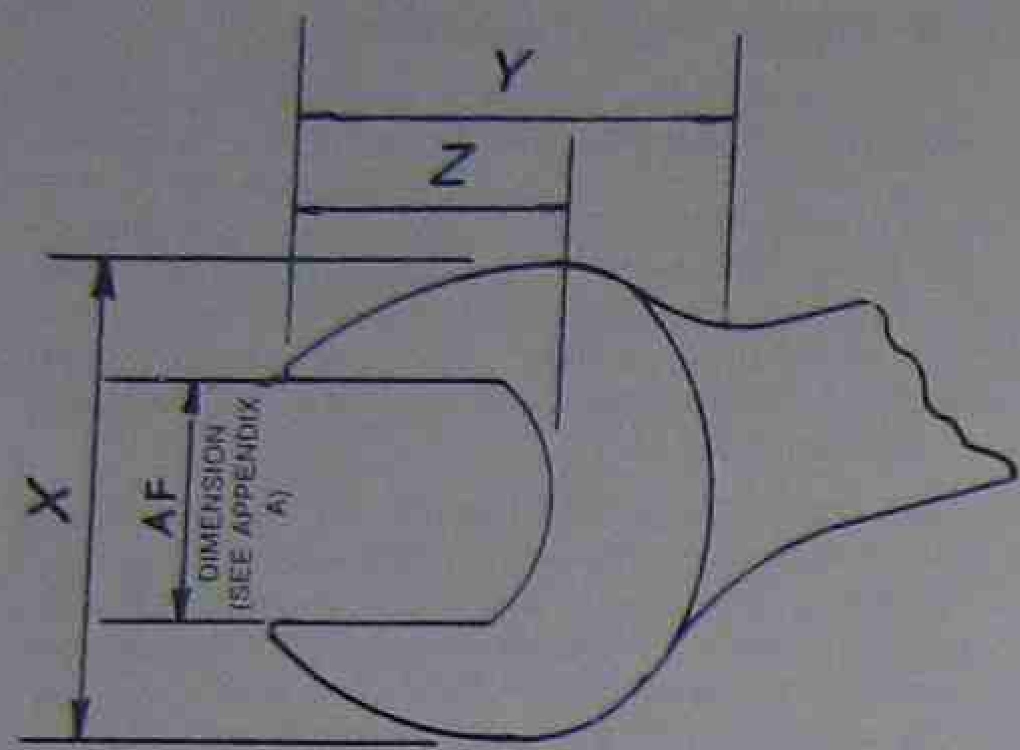
The tensioning procedures of AS 4100 are designed to allow for this relaxation effect.

8.3.8 Re-use of Fully Tensioned Bolts

Since full tensioning induces a bolt tension usually in excess of the elastic limit of the threaded portion of the bolt, re-use and re-tensioning of these bolts will result in a reduced deformation capacity during tensioning, so that the desired minimum bolt tension may not be achieved when the procedures are repeated. Reference 16 advises that plain uncoated bolts may be re-used once or twice provided proper control is established. It does not recommend using coated bolts again.

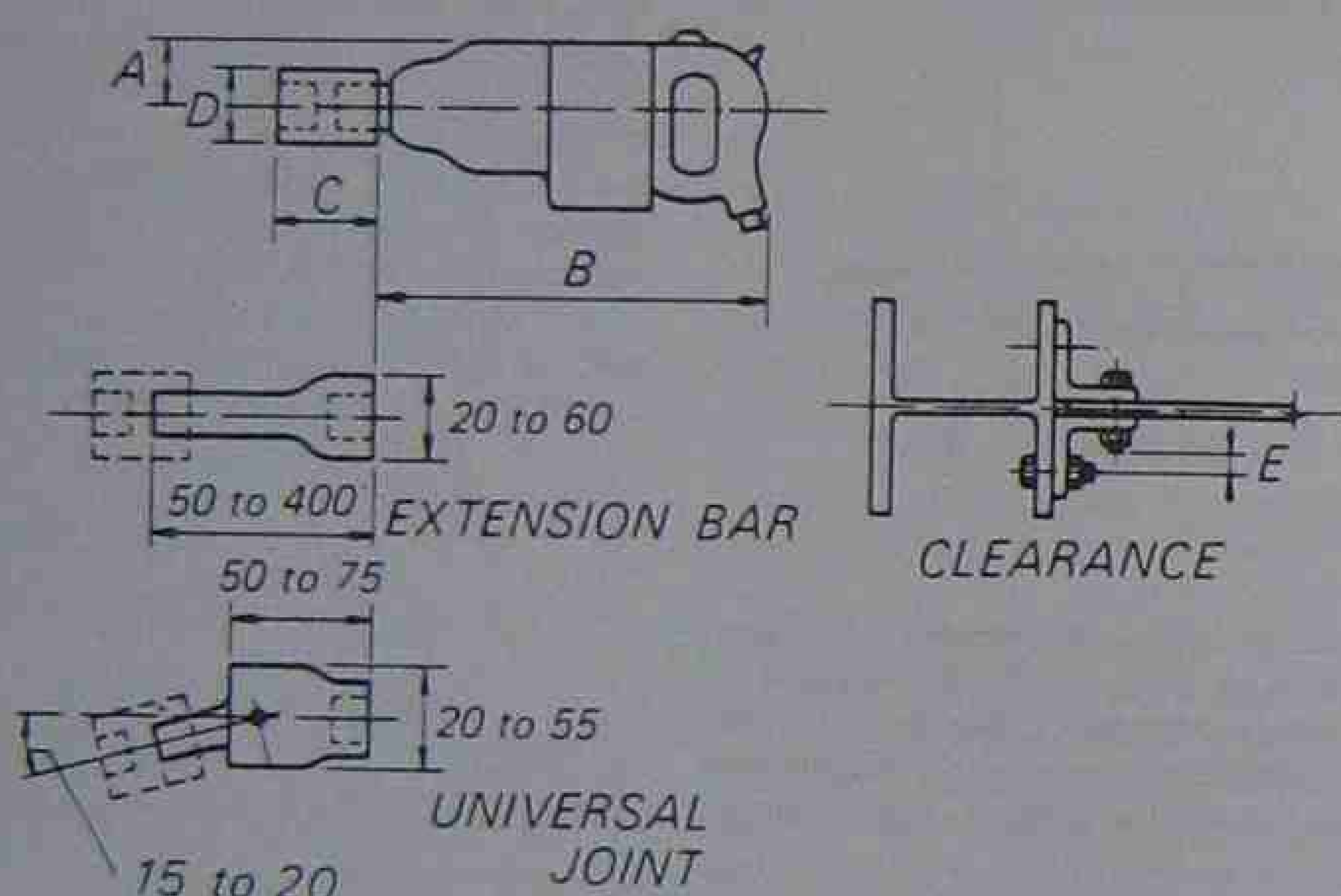
AS 4100 requires that re-tensioning of bolts which have been fully tensioned shall be avoided wherever possible. If re-tensioning must be carried out it is only permitted where the bolt remains in the same hole in which it was originally tensioned and with the same grip. Bolts which have been fully tensioned may not be re-used in another hole.

AS 4100 does not permit retensioning of galvanized bolts due to the relatively soft layer of zinc on the threads. AS 4100 accepts the principle that there will be occasions where bolts may have to be slackened and then retensioned, but prefers that this not occur since the bolts will generally be tensioned beyond their proof load with some plastic deformation likely to occur. It is accepted that a bolt may be retensioned once in the same hole without the danger of bolt breakage on retensioning, but that continued retightening is regarded as undesirable. Tests reported in Ref. 16 indicate that each successive retensioning achieves less induced tension than the previous tensioning, so that the probability of achieving the correct induced tension is reduced. AS 4100 notes that touching up bolts in a joint which may have been loosened during tensioning of other bolts does not constitute retensioning.



Bolt Size	Clearance		
	X	Y	Z
12	40	34	22
16	52	45	28
20	65	56	35
24	78	67	41
30	99	85	53
36	118	102	63

4.6/S and 8.8/S CATEGORIES



8.8/TF and 8.8/TB CATEGORIES

	B	A
Normal Wrenches	to 450	54
Heavy Wrenches	some to 600	65

Normal Sockets*			
Bolt Size	C	D	Clearance E
16	50	38	25
20	60	45	30
24	80	57	35

*Deep length sockets are also available with greater length but same diameter as above.

Fig. 26 — Typical dimensions of hand and power wrenches (for determing erection clearances)

8.4 CLEARANCES

In the detailing of bolted steel structures, attention must be paid to the clearances necessary to install and tighten the bolts.

When using bolts required only to be snug tightened (i.e. 4.6/S and 8.8/S categories) and therefore only requiring the use of standard podger wrenches for tightening, space will be found to be adequate if clearances in accordance with Fig. 26, and standard gauges of members in accordance with Reference 14, are used. Some accessibility problems may be encountered when design is based on the principle of excluding threads from the shear plane in some joints. Because of the relatively long standard thread length on ISO metric bolts the exclusion of threads from shear planes requires the selection of rather long bolts. This was discussed under 'Bolt Length Selection', Section 6.4.

The installation and tightening of high-strength structural bolts for friction-or bearing-type joints, with full tensioning to AS 4100 (i.e. 8.8/TF and 8.8/TB categories), requires more attention to clearance requirements. At the present time the most efficient and economic method of tensioning these bolts is by the use of pneumatic impact wrenches and designers and detailers have to consider the physical dimensions of such tools in relation to clearances required in joints. Fig. 26 shows the dimension of pneumatic impact wrenches and sockets.

Some connection geometries are more sensitive to clearance requirments than others. In particular, the bolted moment end plate connection can have clearance problems if not detailed properly. It will be seen from Fig. 27 that the minimum dimension required from the top of the beam flange to the bolt centreline is determined by the dimension "A" — from the back of the wrench casing to the centreline axis — as tabulated in Fig. 26.

The use of a universal joint does offer some possibility of reducing this dimension, and while this may be seen as an advantage from a design point of view, it should be noted that an impact wrench with a universal joint and socket is generally difficult to handle for an operator some height from ground level and sitting on only the width of the beam flange. In addition, the use of a universal joint reduces the efficiency of the impact wrench and this can make it difficult to achieve the specified minimum bolt tension, especially if located some distance from the source of the compressed air supply.

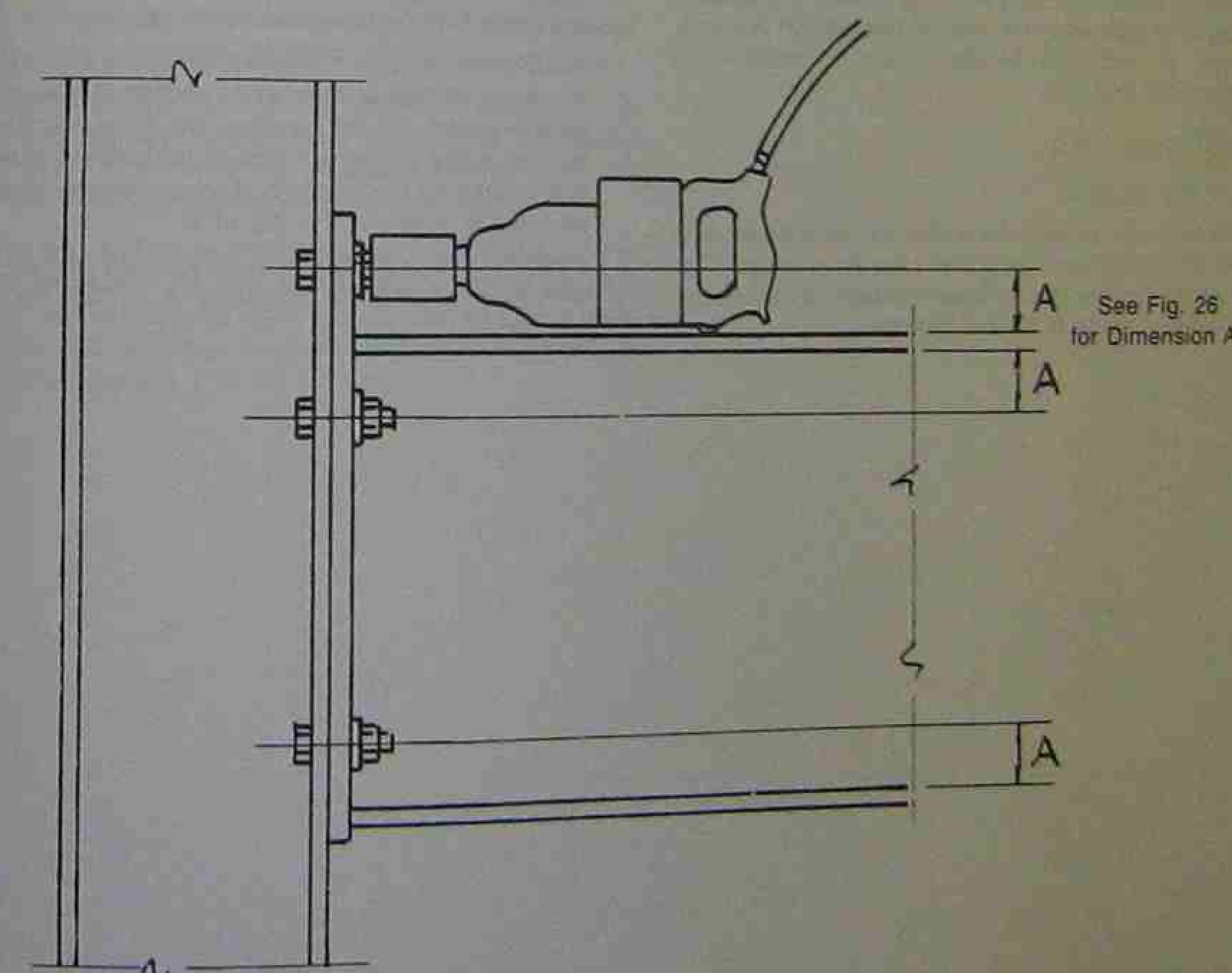


Fig. 27 — Clearance needed for an impact wrench.

8.5 INSPECTION

8.5.1 4.6/S and 8.8/S BOLTING CATEGORIES

In structural joints using 4.6/S or 8.8/S category, the site inspector need only be concerned that the correct bolt type diameter and number have been installed in the joint. Since the level of tightening required is snug and this would have been achieved in the normal course of erection, no further checking is required. A visual check of joints to ensure that the correct numbers of bolts have been installed would reveal any obvious irregularities in the connection assembly. *No inspection to determine the level of tension in the bolt is required by AS 4100, for these bolting categories.*

8.5.2 8.8/TF and 8.8/TB BOLTING CATEGORIES

In joints using 8.8/T category (i.e. 8.8/TF or 8.8/TB) which have to be fully tensioned to AS 4100, inspection must be undertaken in accordance with that Code's requirements. The inspection procedure should be considered in two parts:

(1) Visual Inspection

This should ensure that the correct bolt, nut and washer type have been installed in the joint and snug tightened satisfactorily. Bolts and nuts which show evidence of physical damage should be removed and replaced. This is extremely important when the bolt head shows signs of the bolt having been driven into the holes by force. The resulting damage to the bolt thread can render the tensioning procedure difficult, and make it practically impossible to guarantee the level of bolt tension obtained. The use of drift pins to align holes in members is recommended (Fig. 25).

(2) Bolt Tension Check

(a) Part Turn Method

The inspector should ensure that the correct degree of turn from the snug tight position has been achieved by checking the match marks on the bolt shank and nut face (see Section 8.3.2 and Fig. 19).

Where an impact wrench has been used, either the bolt head or nut will have the appearance of being slightly peened.

(b) Direct Tension Indication Method

The inspector should be satisfied that the manufacturer's specified procedure has been followed and that the development of the specified minimum bolt tension is correctly indicated by the load indicator device.

The great advantage in using load indicator washers in tensioning bolts lies in the simplicity of their use and in the easy means of inspection. The washer having been closed to the correct gap provides a permanent witness that tensioning has been completed, and requires no additional work by the bolting crew to create an inspection facility by match marking.

The feature of the system as far as the inspector is concerned is that he need only check the actual gap on a few bolts by the use of a feeler gauge. After this, a visual comparison will satisfy him that all the bolts are correctly tensioned.

(c) Use of Torque Wrench

When required by the supervising engineer, inspection for gross undertensioning can be carried out by the use of an inspection wrench of either the hand or power operated type. The inspection procedure is outlined in AS 4100, Appendix K. It is important to note that the use of a torque wrench as an inspection device is difficult, tedious and not particularly accurate, and should be used only where there is no alternative — for example where the designer has specified the part turn of nut method but has not required match marking.

Since it is obviously not practicable to check every bolt in a job, a suitable sample would consist of 10% of the bolts in a joint, but not less than two bolts selected at random in any one joint. However the method is really only suitable for the detection of gross undertensioning, as noted in Appendix K of AS 4100.

Extensive site calibration of the inspection wrench to a procedure specified in Appendix K of AS 4100 is required.

9 Corrosion Protection of Bolts

Although bolts can be made in various finishes such as hot-dip galvanized, electro-plated, phosphate-coated, as well as in the plain uncoated finish, it is usual in structural work to deal only with the two extremes of finish, i.e. hot-dip galvanized and plain.

In considering the choice of bolt finish, it is essential to look at the total final protective treatment of the structure. Obviously if a high degree of protection is employed for the steel members because of a severe corrosion environment, the bolts used in connections will have to be equally protected, and the choice of a plain finish will require that the bolts be painted after erection of the structure. This may often be a difficult and costly exercise and the selection of galvanized bolts in such an application would be more economic. However, in mild corrosion environments, where minimum protection is applied to the structure, and where some staining from plain bolts can be tolerated, these bolts will afford the best economic solution. In general, therefore, it is wise to match levels of bolt protection to the applied protection of the whole structure.

It will be seen from Table 15 that if conditions require that bolts in a joint be painted after erection, the alternative use of a galvanized finish is more economic.

Bolts are required to conform to the dimensions specified in Australian Standards before the application of the corrosion protection.

Hot-dip galvanized bolts should comply with AS 1214 (9) while zinc or cadmium electroplated bolts should comply with AS 1897 (10). The coating mass of hot-dip galvanized bolts must be between 300 and 375 grams/square metre.

Assembly tests such as one detailed in Appendix C of AS 1252 (6) are available for testing the anti-seizing properties of an assembly of bolts and nuts which are provided with corrosion-preventative coatings.

For steel bolts to be embrittled after hot-dip galvanizing is rare, especially for the grades of bolt in use in steel structures in Australia, even though fastener manufacture involves cold-working. Guidance on such embrittlement is given in Appendix C of AS 1214 (9).

TABLE 15
INDICATIVE COST-IN-PLACE RATIOS
FOR AN M20 BOLT IN A GROUP

Bolting Category	Plain		Galvanized unpainted
	Unpainted	Painted after	
4.6/S	1.0	2.0	1.4
8.8/S	1.4	2.8	1.7
8.8/T	2.3	4.6	3.4

In using hot-dip galvanized bolts, it is important to remember that the galvanizing process deposits 42-52 μm thickness of zinc to the bolt and the nuts have to be tapped oversize by the manufacturer after galvanizing. (The zinc on the bolt thread galvanically protects the uncoated thread on the nut.) *Consequently only these 'oversize' nuts should be used with a galvanized bolt.* This is particularly important when high strength structural bolts are required to be fully tensioned, since the use of a nut which has not been tapped oversize will, almost certainly, produce a situation where the bolt and nut threads seize before bolt tension is attained. Also important for galvanized high strength structural bolts is the provision of a proper lubricant — applied by the manufacturer — on the threaded parts to prevent seizure of the nut on the bolt during tensioning. Bolts exposed to the weather may lose this lubricant and suitable replacement dry lubricants that can be used in the field are various proprietary compounds incorporating molybdenum disulphide.

10 Economical Design and Detailing — Summary

The essential points to be considered in the economical design of bolted connections are—

(i) Standardize as much as possible for a project (see Reference 11).

(ii) Adopt simple details (see Reference 13).

(iii) Only one bolt diameter and one bolting category should be used in smaller structures; more variety may be justified on a larger structure, but different diameters or categories should be used in accordance with a predetermined philosophy.

(iv) Only one diameter of bolt should be used in any single connection to facilitate the operation of punching or drilling holes, regardless of the size of the structure.

(v) Arrange for a minimum number of field connections by making large sub-assemblies in the shop.

(vi) Bolts in double shear are markedly more efficient and thought should always be given to arranging the connection details accordingly if practicable.

(vii) If possible, avoid bolted connections with a large number of bolts in line parallel to the force, otherwise reduction in bolt efficiency will result. Joints less than 300 mm between first and last bolt are 100% efficient, while longer joints are increasingly less efficient.

(viii) The low design capacity of the 4.6/S bolting category means that it is generally restricted in usage to lightly loaded cleat, bracket or bracing connections.

(ix) The advantage of the 8.8/S bolting category lies in its high design loads and the fact that only snug tightening is required. This bolting category is the most efficient on a cost-in-place per unit capacity. Its use is normally restricted to flexible (simple) connections. It is now the most commonly used bolting category.

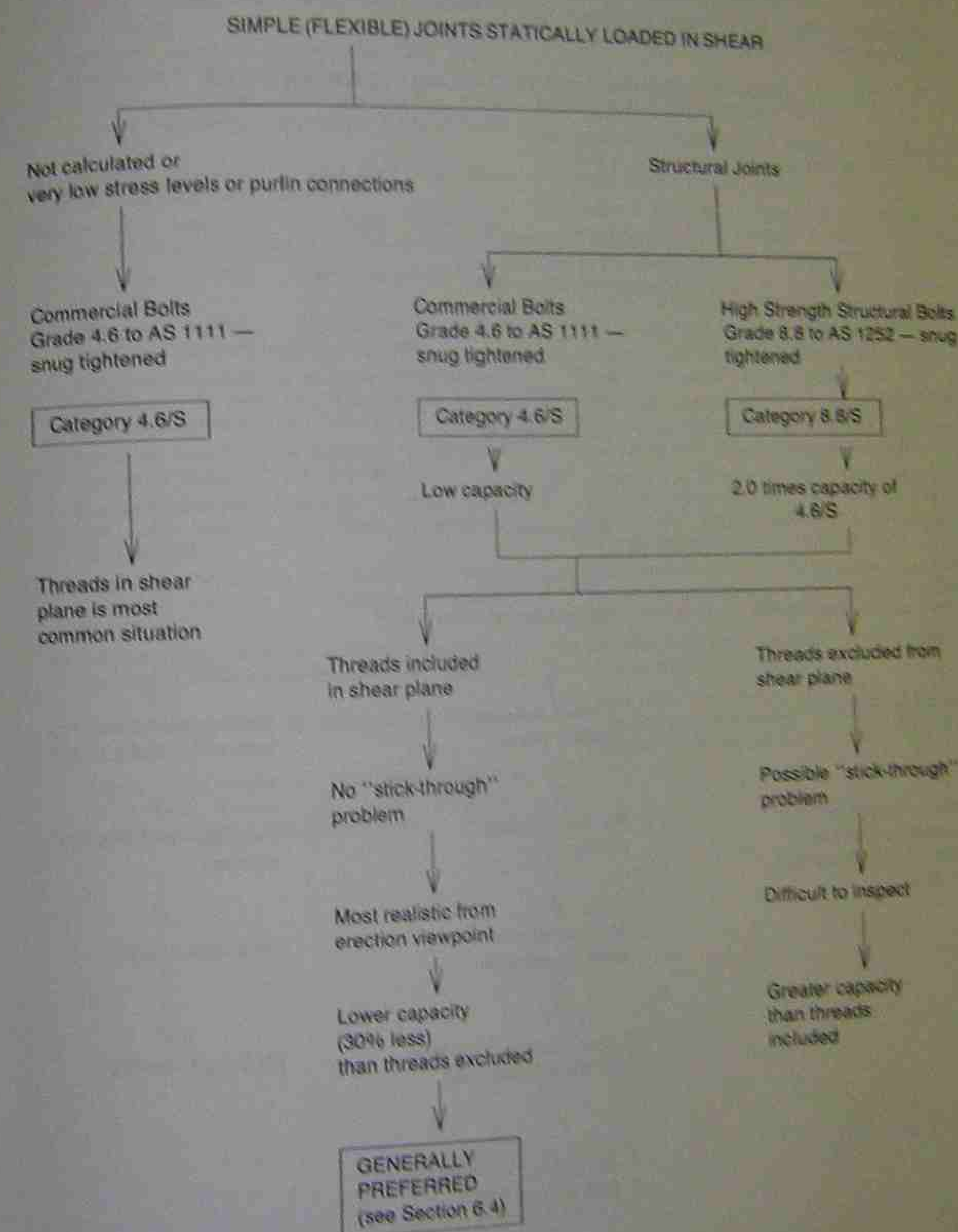
(x) Friction joints 8.8/TF are not very cost efficient and should only be used in joints where slip prevention in the serviceability limit state is a necessity. Bearing joints 8.8/TB are much preferable on a cost basis. 8.8/TF and 8.8/TB categories are recommended for rigid connections.

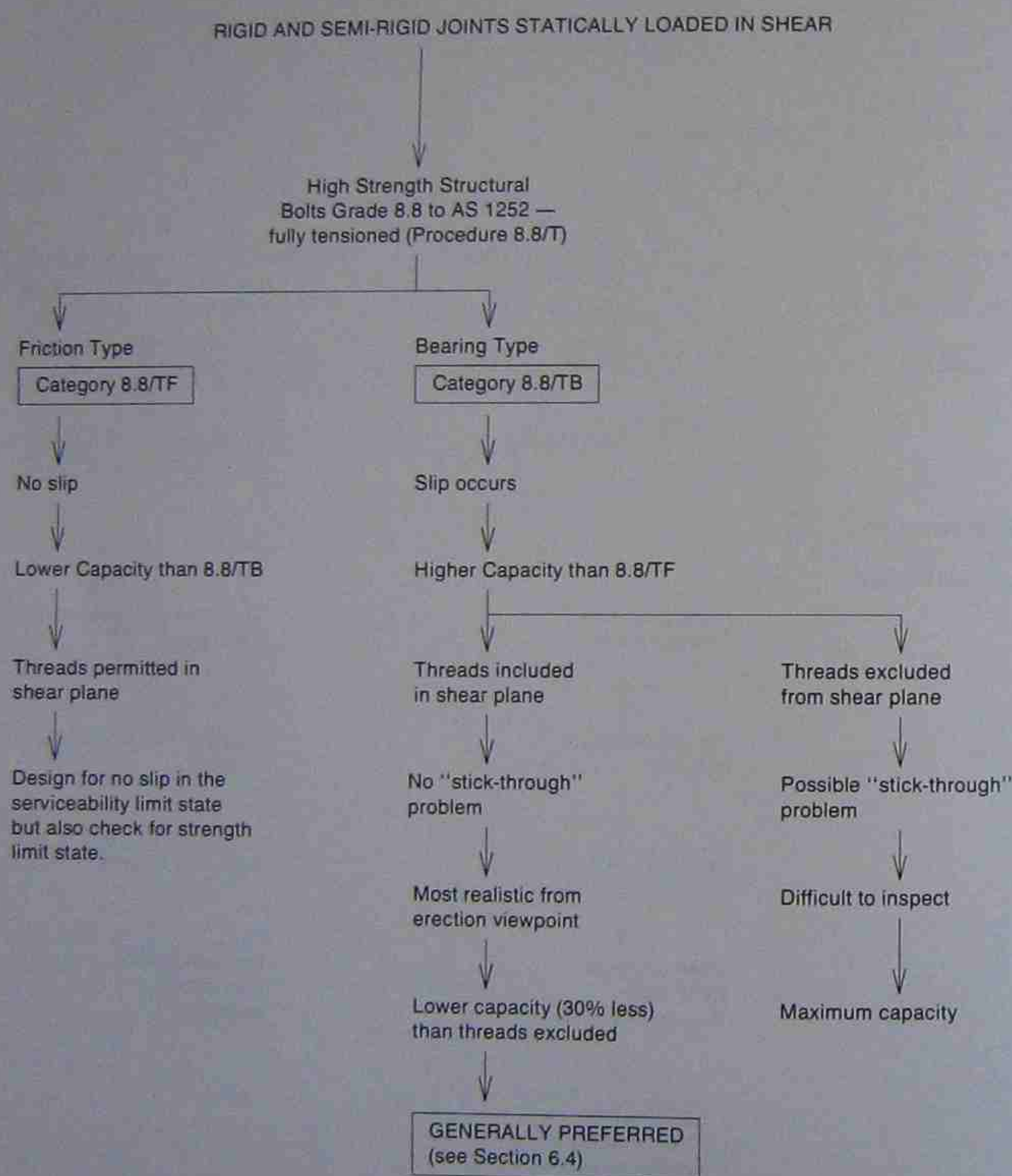
(xi) Try not to mix 8.8/S and 8.8/T bolting categories on the job, but where it is unavoidable arrange the connection details to a predetermined philosophy.

(xii) For economy in bolt numbers it may appear desirable to exclude threads from the shear plane. However, practical reasons outlined in Section 6.4 dictate that usually threads are considered included in the shear plane, unless detailing of the bolts indicates exclusion is certain.

(xiii) Corrosion protection of the bolts should be matched to the end use of the structure.

11 Summary of Bolt Usage





12 References

AUSTRALIAN STANDARDS

1. Standards Association of Australia, "ISO Metric Hexagon Precision Bolts and Screws", AS 1110-1984.
2. Standards Association of Australia, "ISO Metric Hexagon Commercial Bolts and Screws", AS 1111-1980.
3. Standards Association of Australia, "ISO Metric Hexagon Nuts", AS 1112-1980.
4. Standards Association of Australia, "Flat Metal Washers For General Engineering Purposes", AS 1237-1973.

5a. Standards Association of Australia, "SAA Steel Structures Code", AS 1250-1981.

5b. Standards Association of Australia, "SAA High-Strength Structural Bolting Code", AS 1511-1984.

5c. Standards Australia, "Steel Structures", AS 4100-1990.

5d. Standards Australia, "Commentary — Steel Structures", AS 4100 Supplement 1-1990.

6. Standards Association of Australia, "High Strength Steel Bolts with Associated Nuts and Washers for Structural Engineering", AS 1252-1983.
7. Standards Association of Australia, "Metric Screw Threads for Fasteners", AS 1275-1985.
8. Standards Association of Australia, "Fasteners — Bolts Nuts and Washers for Tower Construction", AS 1559-1986.
9. Standards Association of Australia, "Hot-Dip Galvanized Coatings on Threaded Fasteners", AS 1214-1983.
10. Standards Association of Australia, "Electroplated Coatings on Threaded Components", AS 1897-1976.

AUSTRALIAN INSTITUTE OF STEEL CONSTRUCTION PUBLICATIONS

11. Hogan, T.J. & Firkins, A., "Standardized Structural Connections, Details and Design Capacities", Australian Institute of Steel Construction.
12. Hogan, T.J. & Thomas, I.R., "Design of Structural Connections", Australian Institute of Steel Construction.
13. Hogan, T.J. & Firkins, A., "Economical Structural Steelwork", Australian Institute of Steel Construction.
14. Australian Institute of Steel Construction, "Design Capacity Tables for Structural Steel".
15. Firkins, A. & Hogan, T.J., "Bolting of Steel Structures", Second Ed., Australian Institute of Steel Construction 1984.

OTHER PUBLISHERS/PUBLICATIONS

16. Kulak, G.L., Fisher, J.W. & Struik, J.H.A., "Guide to Design Criteria for Bolted and Riveted Joints", John Wiley, Second Edition, 1987.
17. Fisher, J. & Beedle, L., "Criteria for Designing Bearing Type Bolted Joints", Journal of the Structural Division, ASCE, Vol. 91 No. ST5, October 1965, p. 129.
18. European Convention for Constructional Steelwork, "Recommendations for Steel Constructions", Vols. I and II, Document ECCS-EG-76-IE, June 1978.
19. American Institute of Steel Construction, "Manual of Steel Construction", Eighth Edition, 1980.
20. "The Slip Characteristics of High-Strength Friction-Grip Bolted Connections", IMP Technical Report 31, A.J. Carmichael, The University of Newcastle, 1967.
21. Hogan, T.J. & Thomas, I.R., "Bearing Stress and Edge Distance Requirements for Bolted Steelwork Connections", Steel Construction, Australian Institute of Steel Construction, Vol. 13 No. 3, 1979.
22. American Institute of Steel Construction, "Specification for the Design, Fabrication and Erection of Structural Steel for Buildings", 1978 Edition.
23. McGuire, W., "Steel Structures", Prentice Hall, 1968, Chapter 6 "Connections".
24. Allen, R.N. and Fisher, J.W., "Bolted Joints with Oversize or Slotted Holes", Journal of the Structural Division, ASCE, Vol. 94 No. ST9, Sept. 1968.

APPENDICES

A DETAILS OF COMMONLY USED BOLTS NUTS AND WASHERS
TABLE A.1 Metric hexagon commercial bolts
TABLE A.2 High strength steel bolts

B DETAILS OF SPECIAL BOLTS NUTS AND WASHERS
TABLE B.1 Tower bolts
TABLE B.2 Swage lock fasteners

C DESIGN CAPACITIES OF COMMONLY USED BOLTS
— STRENGTH LIMIT STATE
TABLE C.1 4.6/S BOLTING CATEGORY
TABLE C.2 8.8/S BOLTING CATEGORY

D DESIGN CAPACITIES OF COMMONLY USED BOLTS
— SERVICEABILITY LIMIT STATE
TABLE D.1 8.8/TF BOLTING CATEGORY

E DESIGN CAPACITIES OF COMMONLY USED BOLTS
— FATIGUE LIMIT STATE
Fig. E.1 8.8/T BOLTING CATEGORIES

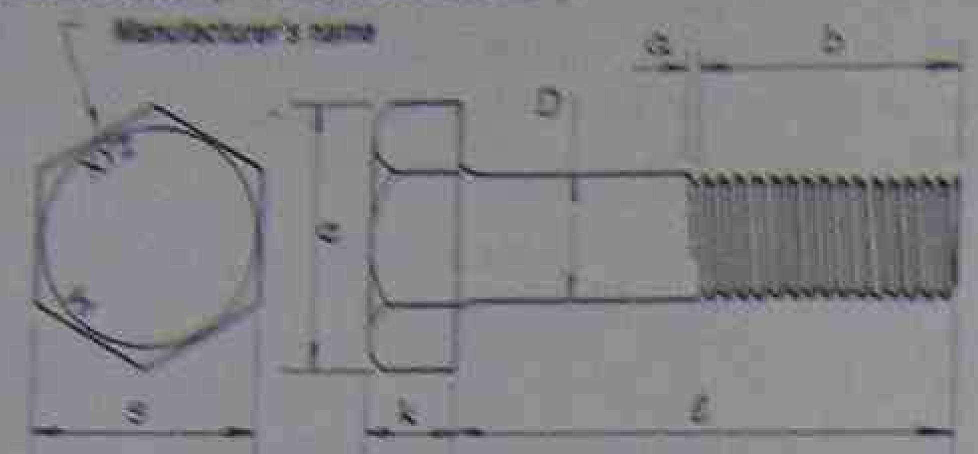
APPENDIX A DETAILS OF COMMONLY USED BOLTS NUTS AND WASHERS

TABLE A.1

METRIC HEXAGON COMMERCIAL BOLTS

GRADE 4.6

(Superseded names: "black bolts", "mild steel bolts").



BOLT GEOMETRY AND HEAD MARKING

STANDARD SPECIFICATION:	AS 1111 (Ref. 2)	
THREAD SPECIFICATION:	AS 1275 (Ref. 7) 8 g tolerance	
GRADE DESIGNATION:	4.6	
NORMAL METHOD OF MANUFACTURE:	Hot or cold forging (generally cold)	
MECHANICAL PROPERTIES:	Tensile strength	400 MPa (nom) (min)
	Yield stress	240 MPa (min)
	Stress under proof load	225 MPa (min)
MOST COMMONLY USED SIZES:	M12, M16, M20, M24, M30, M36	

TENSILE AND PROOF LOADS:

Nom. dia. (mm)	Tensile stress area (mm^2)	Minimum breaking load (kN)	Proof load (kN)
M12	84.3	33.7	19.0
M16	157	62.8	35.3
M20	245	98.0	55.1
M24	353	141	79.4
M30	561	224	126
M36	817	327	184

NOTES: Elongation after fracture = 22% min. Hardness = 114 HB min.

GENERAL DIMENSIONS OF BOLTS (all in millimetres):

Nom. dia.	Pitch P	Shank diameter d_s	Width across flats s	Width across corners e	Height of head k
		nom.	max.	max.	max
M12	1.75	12	18	21	8
M16	2.0	16	24	28	11
M20	2.5	20	30	35	13
M24	3.0	24	36	42	16
M30	3.5	30	46	53	20
M36	4.0	36	55	64	24

THREAD LENGTHS:

Nominal Thread Length, b, (excluding thread runout):

For bolts up to and including 125 mm long $= 2d_s + 6$ mm

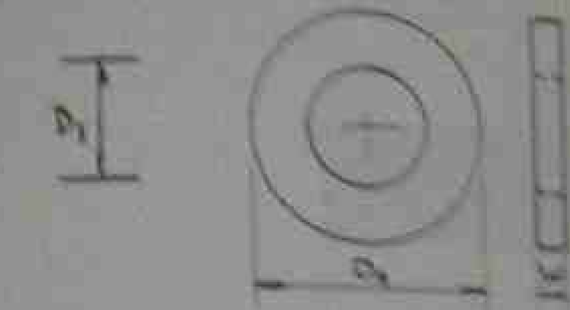
For bolts over 125 mm up to and including 200 mm long $= 2d_s + 12$ mm

For bolts over 200 mm long $= 2d_s + 25$ mm

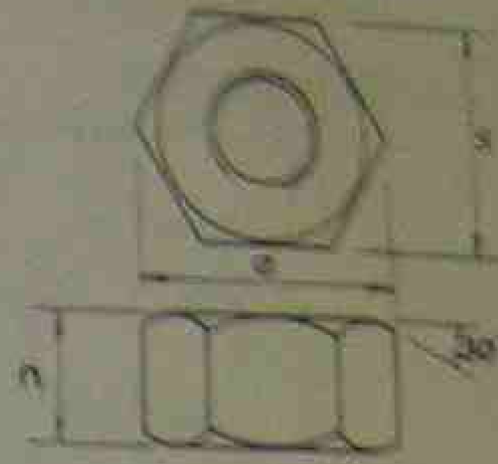
Nom. dia.	Nominal length of thread b nom (mm)	
	$\ell \leq 125$	$125 < \ell \leq 200$
M12	30	36
M16	38	44
M20	46	52
M24	54	60
M30	66	72
M36	78	84

APPENDIX A DETAILS OF COMMONLY USED BOLTS NUTS AND WASHERS

TABLE A.1 (contd)



WASHER



NUT

STANDARD SPECIFICATION: AS 1237 (Ref. 4)

MATERIAL: Mild Steel

GENERAL DIMENSIONS OF WASHERS (mm):

	Nominal size of washer	Thickness (nom)	Inside dia D_1	Outside dia. D_2
			min	max.
Normal Series	12	2.5	14	24
	16	3	18	30
	20	3	22	37
	24	4	26	44
	30	4	33	56
	36	5	39	66

STANDARD SPECIFICATION: AS 1112 (Ref. 3)

THREAD SPECIFICATION: AS 1275 (Ref. 7)

CLASS (Normal): 5

NORMAL METHOD OF MANUFACTURE: Hot or cold forging

MECHANICAL PROPERTIES: Proof load stress 630 MPa (M12, M16 — 610 MPa)

Some manufacturers supply Class 8 nuts with commercial bolts, in lieu of Class 5 nuts. The Class 8 nut, which is stronger, can be identified by a dot at 12 o'clock and a dash at 3 o'clock. Class 5 nuts are unmarked.

PROOF LOADS (class 5 nuts):—

Nom. dia.	Tensile stress area, mm^2	Proof load kN	
		Plain	Galv.
M12	84.3	51.4	43
M16	157	95.8	80
M20	245	154	137
M24	353	222	198
M30	561	561	314
M36	817	817	458

GENERAL DIMENSIONS OF NUTS (mm): Class 5 & 8

Nom. dia.	Pitch of thread	Width across flats s	Width across corners e	Height of normal nuts n
		max.	nom.	ave.
M12	1.75	18	21	11
M16	2.0	24	28	15
M20	2.5	30	35	18
M24	3.0	36	42	22
M30	3.5	48	53	26
M36	4.0	55	64	31

BOLT SIZES GENERALLY AVAILABLE:

Diameter mm	Nominal Lengths ℓ (mm)														
	40	45	50	55	60	65	70	80	90	100	110	120	130	140	150
M12	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
M16	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
M20	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
M24			X	X	X	X	X	X	X	X	X	X	X	X	X
M30					X	X	X	X	X	X	X	X	X	X	X
M36							X	X	X	X	X	X	X	X	X

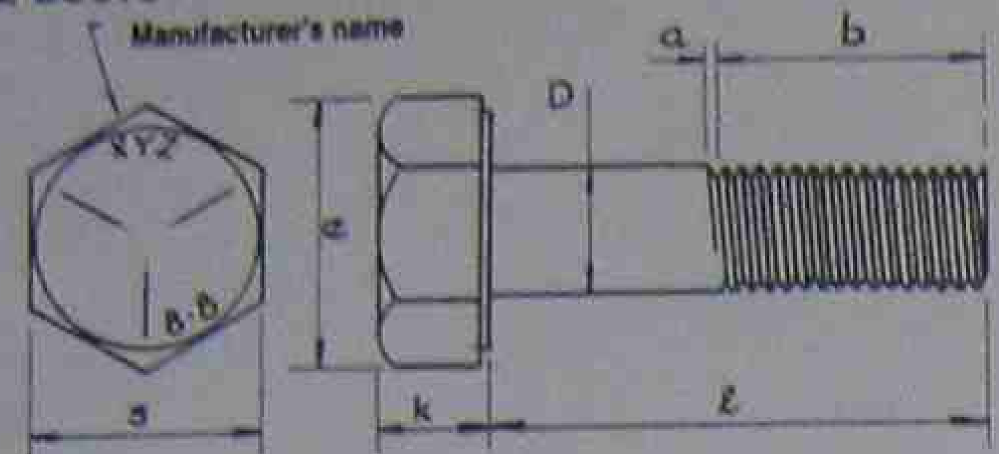
Usually supplied as full thread screws

Commonly available finishes: Plain uncoated; hot dip galvanized [galvanizing specification — AS 1214 (10)]

TABLE A.2

HIGH-STRENGTH STEEL BOLTS

GRADE 8.8



BOLT GEOMETRY AND HEAD MARKING
(Note three radial lines)

STANDARD SPECIFICATION:	AS 1252 (includes nuts & washers) (Ref. 6)
THREAD SPECIFICATION:	AS 1275 (Ref. 7)
GRADE DESIGNATION:	8.8
NORMAL METHOD OF MANUFACTURE:	Hot or cold forging, hardened and tempered
MECHANICAL PROPERTIES:	Tensile strength 800 MPa (nom.), 830 MPa (min.) Stress at perm. set 640 MPa (nom.), 660 MPa (min.) Stress under proof load 600 MPa
MOST COMMONLY USED SIZES:	(M16), M20, M24, (M30), (M36) ( ) available but rarely used

TENSILE AND PROOF LOADS:

Nom. bolt dia.	Tensile stess area A_s (mm^2)	Minimum breaking load (kN)	Proof load (kN)
M16	157	130	94.5
M20	245	203	147
M24	353	293	212
M30	561	466	337
M36	817	678	490

NOTES: Elongation after fracture = 12% min.
Impact strength = 30 J min. Hardness = 242 HB min.

GENERAL DIMENSIONS OF BOLTS
(all in millimetres):

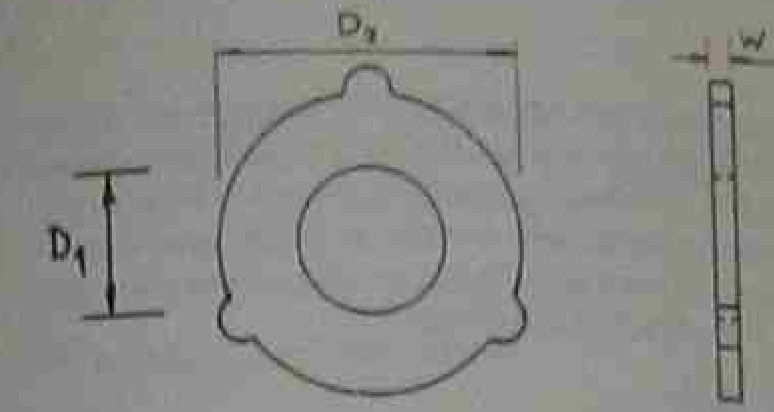
Nom. bolt dia.	Pitch	Thread run out a	Shank dia. d_f	Width across flats s	Width across corners e	Height of head k
	P	max.	nom.	max.	nom.	ave.
M16	2.0	6.0	16	27	31	10.8
M20	2.5	7.5	20	32	37	13.9
M24	3.0	9.0	24	41	47	15.9
M30	3.5	10.5	30	50	58	19.8
M36	4.0	12.0	36	60	69	23.6

THREAD LENGTHS:
Nominal Thread Length, b, (excluding thread runout):
For bolts up to and including 125 mm long $= 2d_f + 6$ mm
For bolts over 125 mm up to and including 200 mm long $= 2d_f + 12$ mm
For bolts over 200 mm long $= 2d_f + 25$ mm

Nom. dia.	Nominal length of thread b nom (mm)	
	$\ell \leq 125$	$125 < \ell \leq 200$
M16	38	44
M20	46	52
M24	54	60
M30	66	72
M36	78	84

TABLE A.2 (contd)

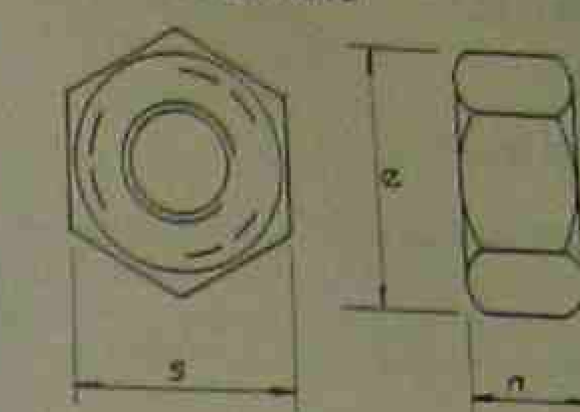
WASHERS



FLAT ROUND WASHER
(Note three nibs)

MATERIAL: Hardened and tempered steel

STRUCTURAL NUTS



NUT MARKING
(Note three broken circumferential lines indented)

THREAD SPECIFICATION:		AS 1275 (Ref. 7)
NORMAL METHOD OF MANUFACTURE:		Hot or cold forging
MECHANICAL PROPERTIES:		Proof load stress
	Uncoated	1075 MPa
	Hot dip galvanized	1165 MPa

GENERAL DIMENSIONS OF WASHERS (mm):

Nominal Diameter of Bolt	Inside D_1 min	Outside Diameter D_2 max.	Thickness w
M16	18	34	3.4 (min)- 4.6 (max)
M20	22	39	
M24	26	50	
M30	33	60	
M36	39	72	

Hardness = 345 HV min
35 HRC min

PROOF LOADS (Property class 8)

Nom. nut dia. (mm)	Tensile stress area (mm^2)	Proof load (kN)	
		Plain	Galv.
M16	157	169	183
M20	245	263	285
M24	353	380	411
M30	561	603	654
M36	817	878	952

GENERAL DIMENSIONS OF NUTS (mm):

Nom. nut dia.	Pitch p	Width across flats s	Width across corners e	Height n
		max.	max.	max.
M16	2.0	27	31	17.1
M20	2.5	32	37	21.3
M24	3.0	41	47	25.3
M30	3.5	50	58	31.3
M36	4.0	60	69	37.6

BOLT SIZES GENERALLY AVAILABLE:

Nom. bolt dia	45	50	55	60	65	70	75	80	85	90	95	100	110	120	130	140	150
M16	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
M20	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
M24			X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
M30						X	X	X	X	X	X	X	X	X	X	X	X
M36									X	X	X	X	X	X	X	X	X

Nominal Lengths ℓ (mm)

bolts with shortened thread lengths
min body length = 0.5 × bolt dia

Commonly available finishes: Plain oiled; hot-dip galvanized [galvanizing specification — AS 1214 (Ref. 9)]

APPENDIX B DETAILS OF SPECIAL BOLTS NUTS AND WASHERS

TABLE B.1

TOWER BOLTS

DESCRIPTION: These are special bolts developed to provide a higher grade bolt with good notch ductility, these being particularly suited to electricity transmission tower members. The bolts have short thread lengths to enable thread to be kept out of the shear plane. Bolts are dimensionally similar to those specified in AS 1111 except that thread lengths have been modified. Step bolts, nuts and washers are also covered in the standard specification.

STANDARD SPECIFICATION: AS 1559 (Ref. 8)

THREAD SPECIFICATION: AS 1275, 8g tolerance (Ref. 7)

GRADE DESIGNATION: 5.8

MECHANICAL PROPERTIES: (carbon steel or weather-resistant steel)
Tensile strength 480 Mpa (min)
Yield strength 340 MPa (min)
Stress under proof load 320 MPa
Shear stress 320 MPa (min)

Charpy V-notch impact at 0°C Average of 3 tests 27 J (min) Individual test min 20 J

MOST COMMONLY USED SIZES: M16, M20 (other diameters M24, M30)

GENERAL DIMENSIONS: consult AS 1559 (dimensionally similar to AS 1111 — see Table A.1)

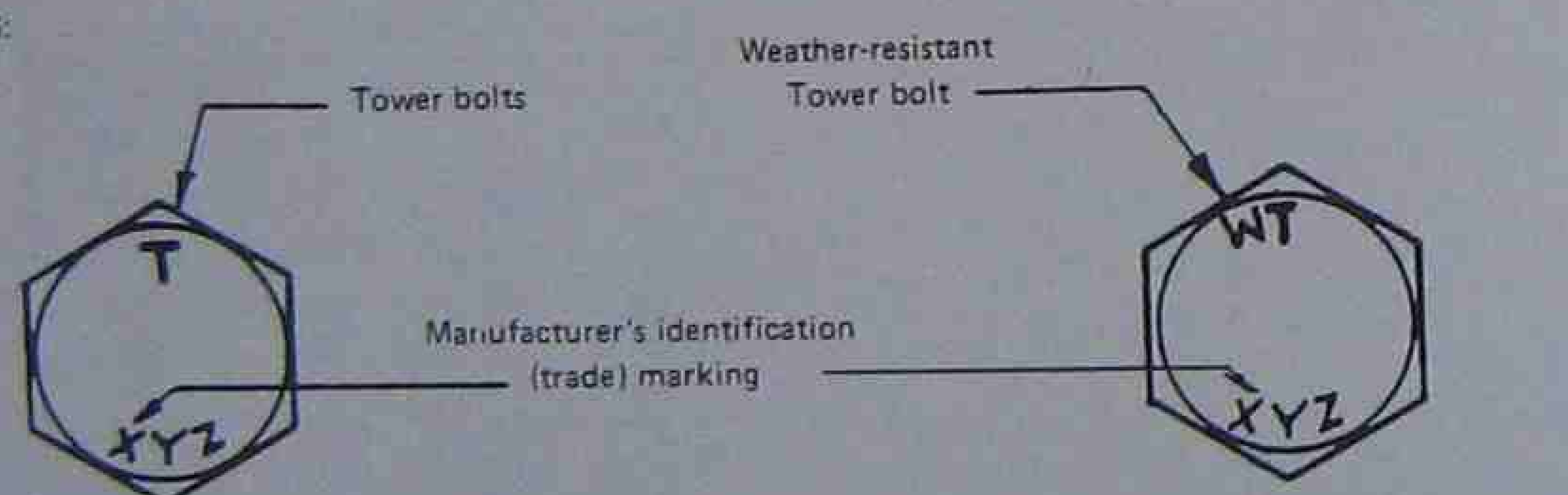
THREAD LENGTHS: consult AS 1559

WASHERS: 5 mm nominal thickness (dimensionally as AS 1237 — see Table A.1)

NUTS: (carbon steel or weather-resistant steel) Class 5 to AS 1112 — (see ISO Metric Hexagon Commercial Bolts, Table A.1). Double nut types — without locking top nut — are permissible.

NORMAL FINISH (bolts, nuts, washers): hot-dip galvanized, unless manufactured from weather-resistant steel (WR 350/2L0).

MARKINGS:



MARKING OF CARBON STEEL BOLT

MARKING OF WEATHER-RESISTANT STEEL BOLT

TENSILE, PROOF AND DOUBLE SHEAR LOADS:

Thread Designation	Test Load (kN) Breaking Load	Proof Load	Double Shear Load
M16	75.4	50.2	129
M20	118	78.0	201
M24	169	113	289
M30	269	180	452

APPENDIX B DETAILS OF SPECIAL BOLTS NUTS & WASHERS

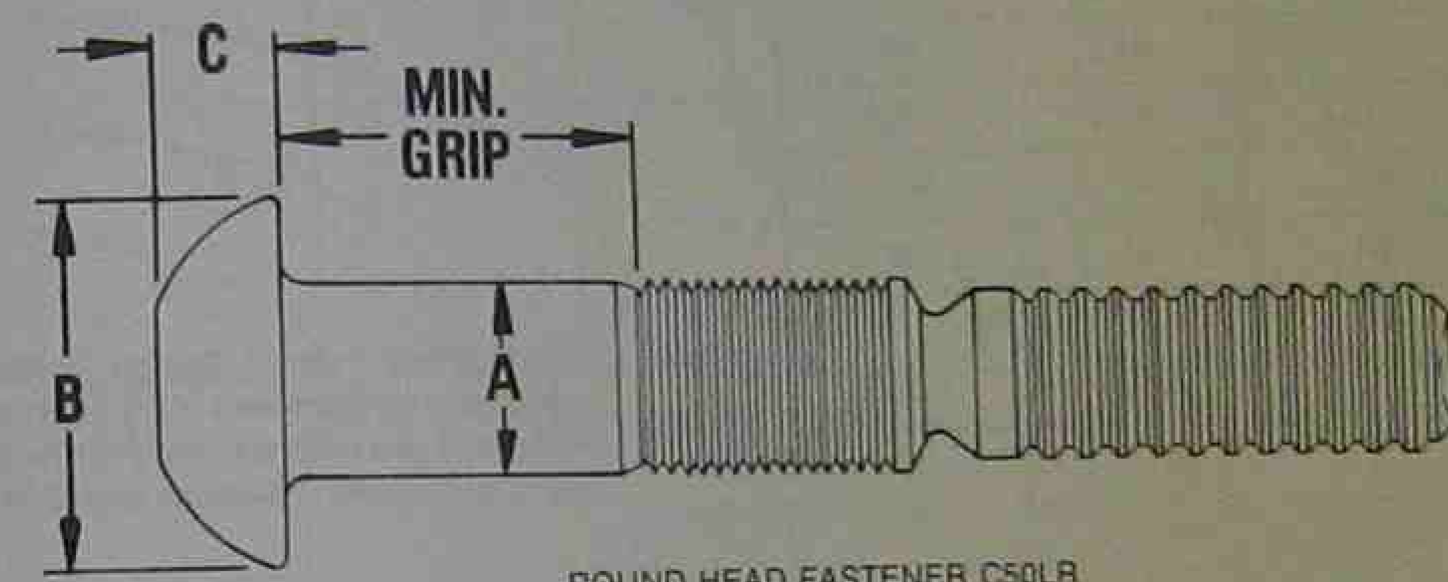
TABLE B.2

SWAGE LOCK FASTENERS C50L SYSTEM

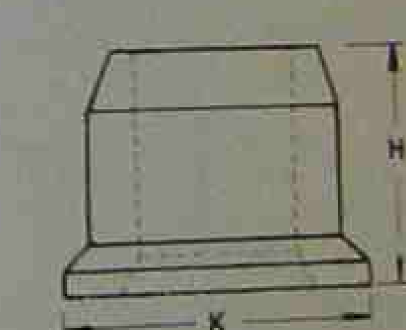
DESCRIPTION: This two piece high strength fastener consists of a pin with two sets of annular grooves and a locking collar with a plain bore. By the use of a special installation tool the collar is swaged into the locking grooves on the pin while the pin is subjected to a tensile force. The pin tail section breaks off at the breakneck when correct shank tension is reached.

STANDARD SPECIFICATION: There is no Australian Standard covering the fasteners themselves, but the system is acceptable as an equivalent High Strength Structural Bolt in terms of AS 4100, Section 2, Clause 2.

TYPICAL DIMENSIONS OF ROUND HEAD FASTENERS AND FLANGED COLLARS (millimetres): C50L SYSTEM



ROUND HEAD FASTENER C50LR



3LC FLANGED COLLAR

Nominal shank dia. A	Maximum hole dia.	Pin: Head dia. B	Pin: Head height C	Pin: Range of grips available	Collar: Greatest dia. K	Collar: Height H
16	18	29	10	6-125	32	24
19	21	35	12	6-125	39	29
22	24	40	14	12-120	41	33
25	27	46	15	12-150	48	38

BOLT MATERIAL: SAE GRADE 5 or ASTM-ASA-325 (Equivalent to Grade 8.8) (imported in imperial sizes to U.S. Standards)

INSTALLATION: Collar installed over bolt and collar tightened using special tool. When collar swaging is complete tool continues to tension until shank breaks flush with top of collar.

APPENDIX C DESIGN CAPACITIES OF COMMONLY USED BOLTS — STRENGTH LIMIT STATE

TABLE C.1

GRADE 4.6

COMMERCIAL BOLTS 4.6/S BOLTING CATEGORY

(f_{uf} = 400 MPa)

Bolt Size	Axial Tension ϕN_{tf}	Shear Values (Single Shear): Threads included in Shear Plane — N ϕV_{fn}	Shear Values (Single Shear): Threads excluded from Shear Plane — X ϕV_{fx}
	kN	kN	kN
M12	27.0	15.1	22.4
M16	50.2	28.6	39.9
M20	78.4	44.6	62.3
M24	113	64.3	89.7
M30	180	103	140
M36	261	151	202
	ϕ = 0.8	ϕ = 0.8	
		4.6N/S	4.6X/S

IMPORTANT NOTE ON 4.6/S BOLTING CATEGORY

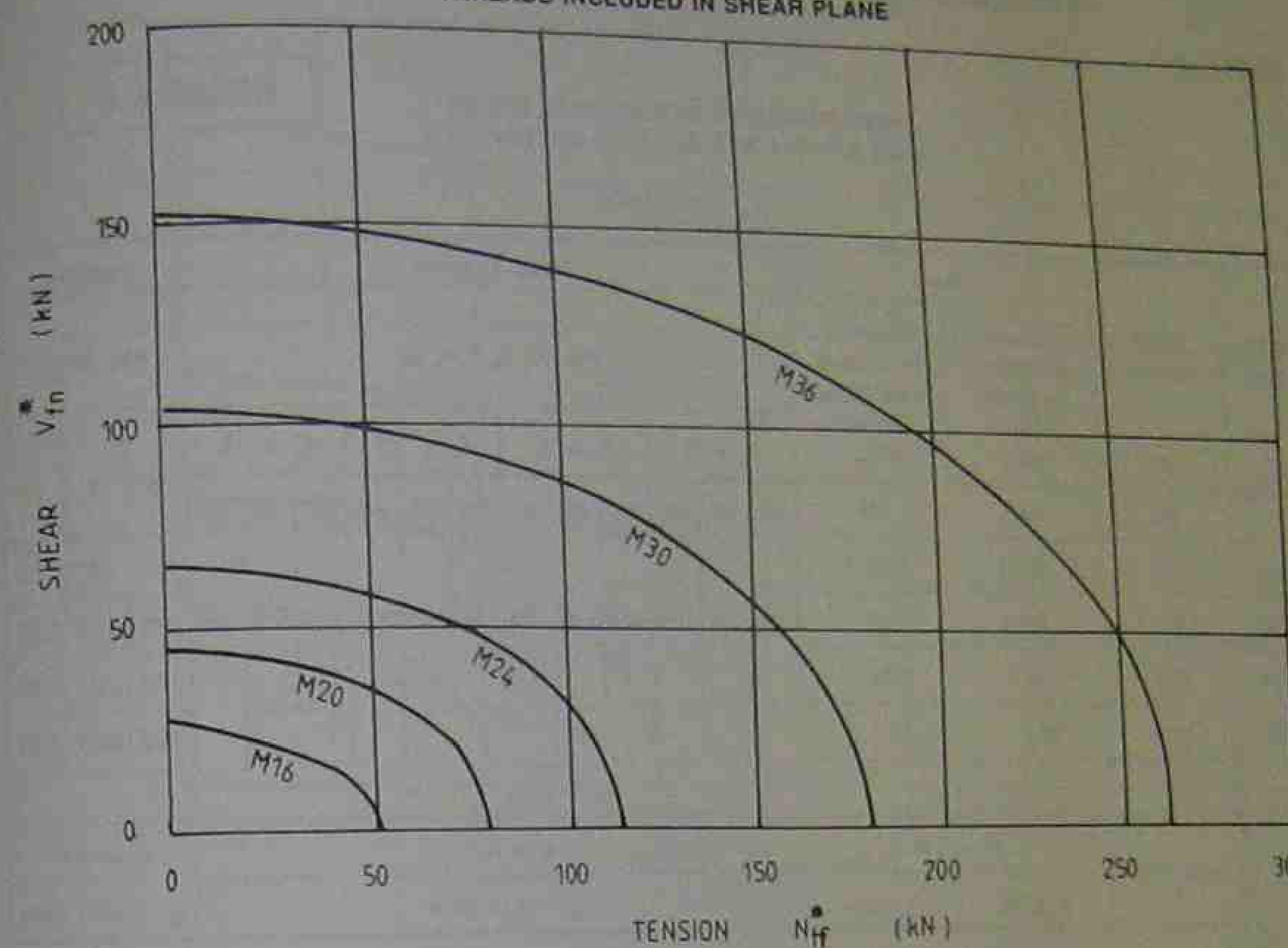
Values for the threads excluded case are provided for the sake of completeness.

However in practical structural bolting situations it is never worthwhile considering 4.6X/S category — 8.8N/S is always more economic. (See also Section 6.4.)

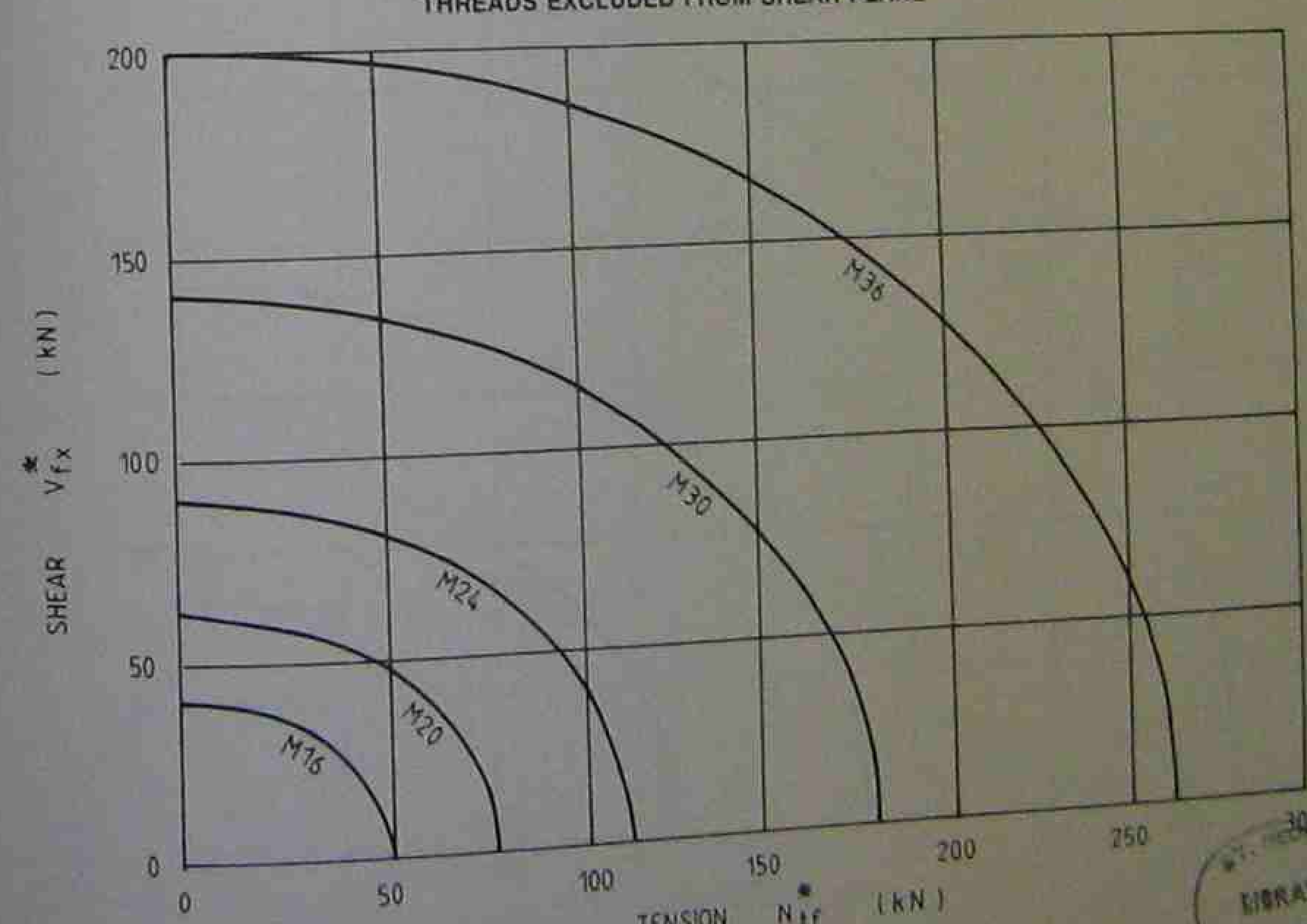
Note: 1. Bearing/Plate Tearout Design Capacity. For all reasonable combinations of ply thickness, bolt diameter and end distance, the design capacity for a ply in bearing (ϕV_b) exceeds both ϕV_{fn} and ϕV_{fx}.

4.6/S BOLTING CATEGORY
SHEAR — TENSION INTERACTION DIAGRAM

THREADS INCLUDED IN SHEAR PLANE



THREADS EXCLUDED FROM SHEAR PLANE



APPENDIX C DESIGN CAPACITIES OF COMMONLY USED BOLTS — STRENGTH LIMIT STATE

TABLE C.2

GRADE 8.8

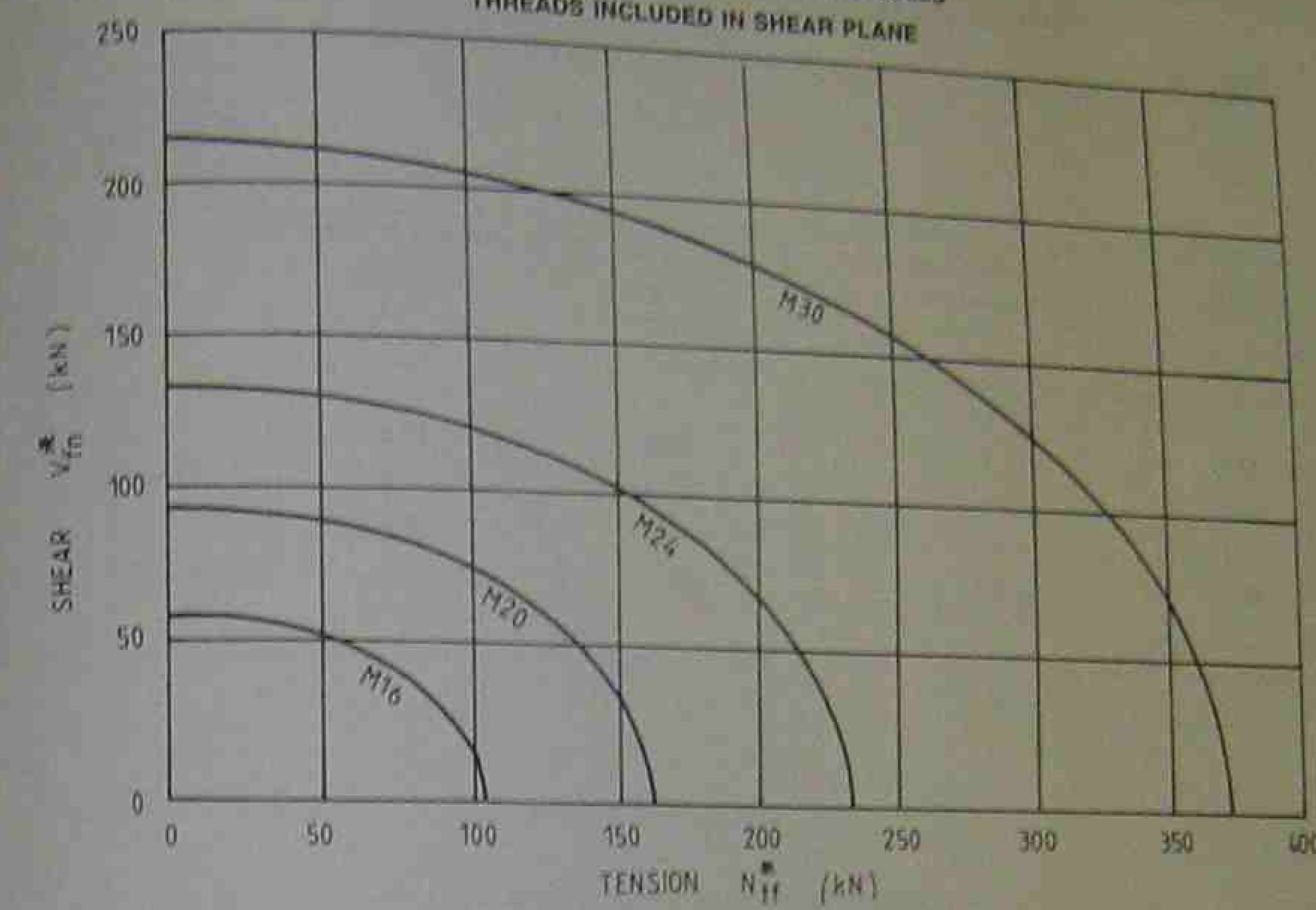
HIGH STRENGTH STRUCTURAL BOLTS
8.8/S 8.8/TB 8.8/TF BOLTING CATEGORIES

(f_{uf} = 830 MPa)

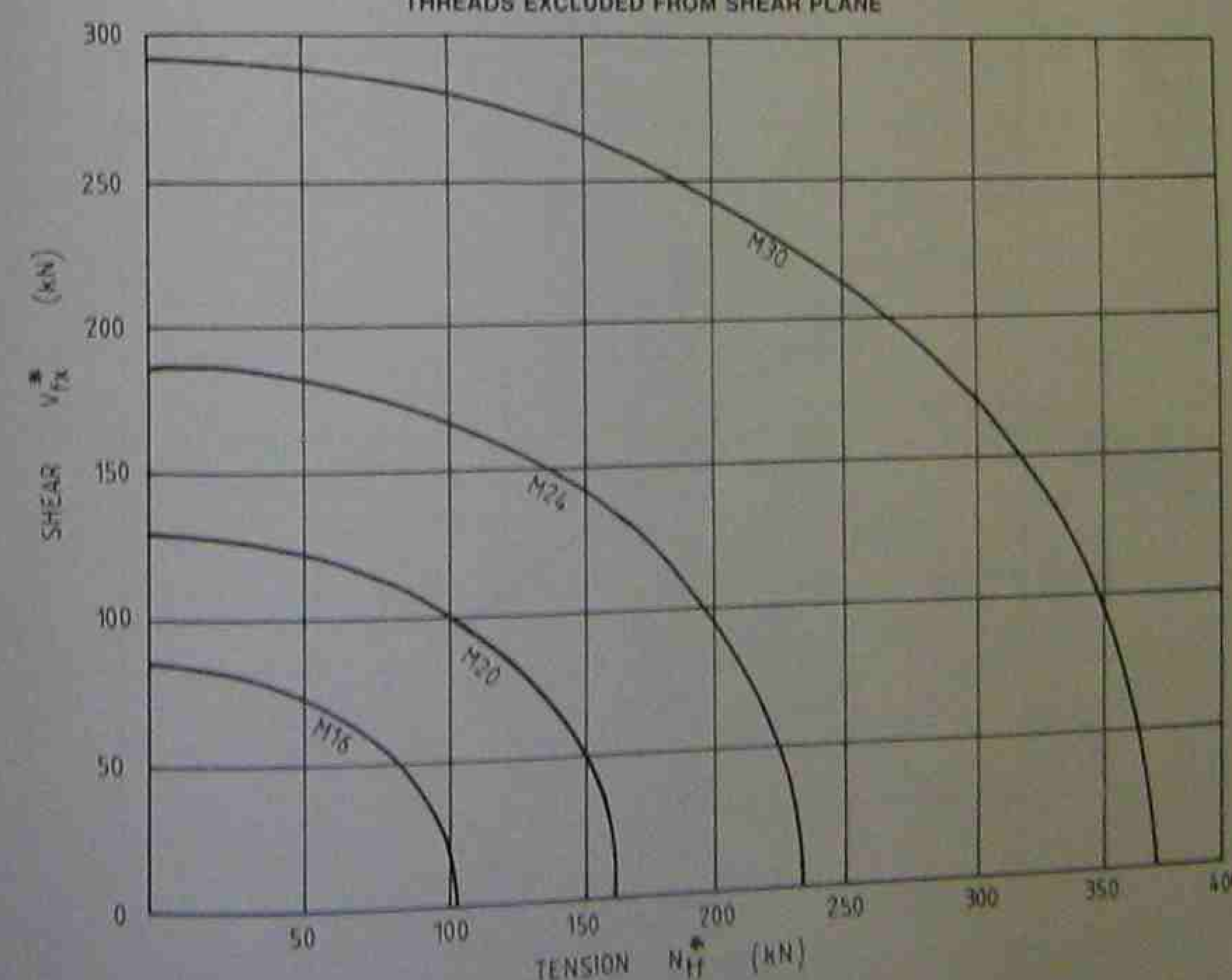
Bolt Size	Axial Tension	Shear		Plate Tearout												Bearing		
		Threads included in Shear Plane	Threads excluded from Shear Plane	ϕV_b for t_p & a_e of:												ϕV_b for t_p		
	ϕN_{tf}	ϕV_{fn}	ϕV_{fx}	t_p = 6			t_p = 8			t_p = 10			t_p = 12			6	8	10
	kN	kN	kN	35	40	45	35	40	45	35	40	45	35	40	45			
M16	104	59.3	82.7													113	151	189
M20	163	92.6	129	78	89	100	103	118	133	129	148	166	155	177	199	142	189	236
M24	234	133	186													170	227	283
M30	373	214	291													213	283	354
				$a_e < a_{emin} = 1.5\, d_f$														
	ϕ = 0.8	ϕ = 0.8		ϕ = 0.9												ϕ = 0.9		
		8.8N/S	8.8X/S	f_{up} = 410 MPa												f_{up} = 410 MPa		

8.8/S 8.8/TB 8.8/TF BOLTING CATEGORY
SHEAR — TENSION INTERACTION DIAGRAMS

THREADS INCLUDED IN SHEAR PLANE



THREADS EXCLUDED FROM SHEAR PLANE



APPENDIX D DESIGN CAPACITIES OF COMMONLY USED BOLTS — SERVICEABILITY LIMIT STATE

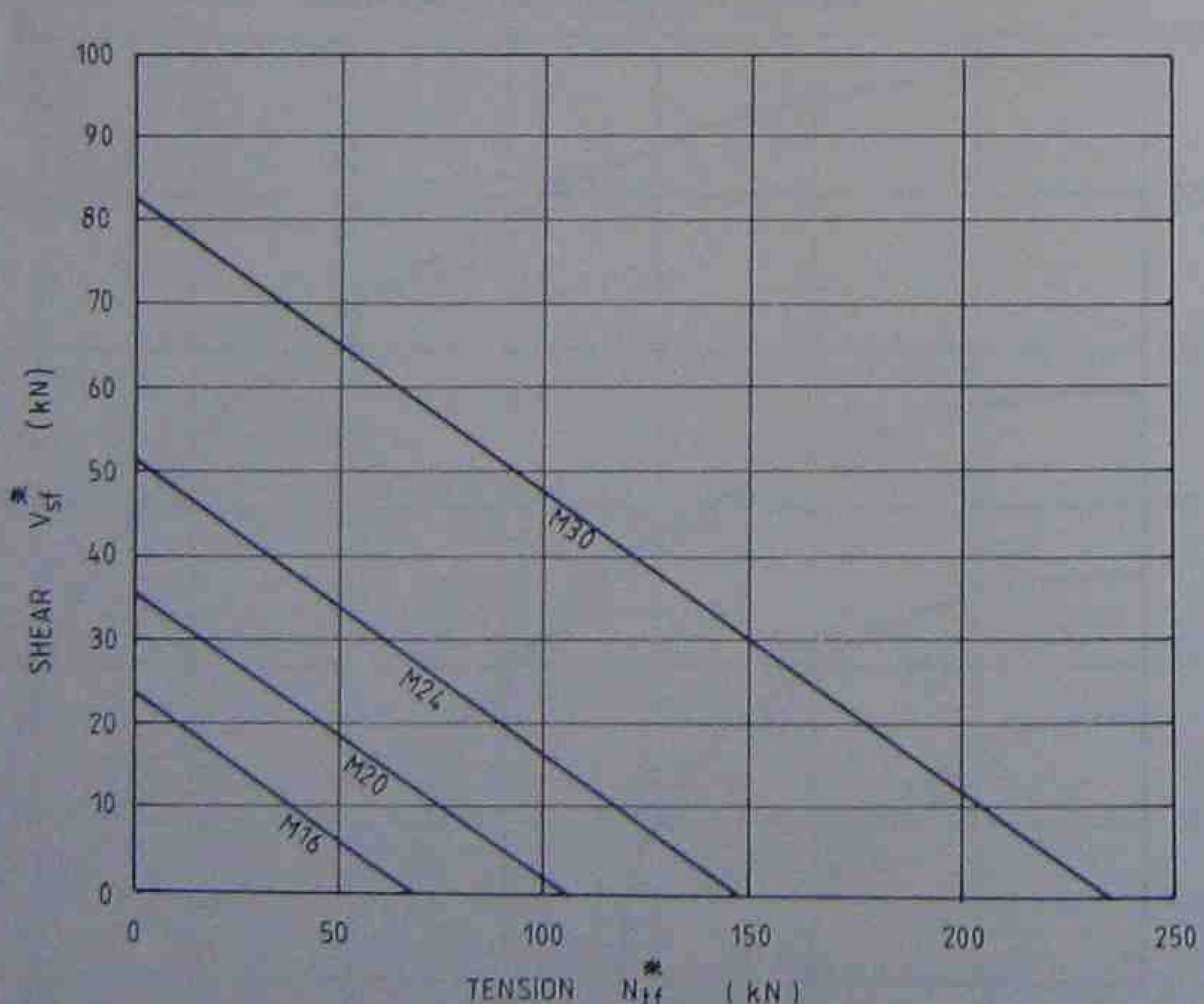
TABLE D.1 HIGH STRENGTH STRUCTURAL BOLTS 8.8/TF BOLTING CATEGORY

(μ = 0.35 n_{ei} = 1 ϕ = 0.7)

GRADE 8.8

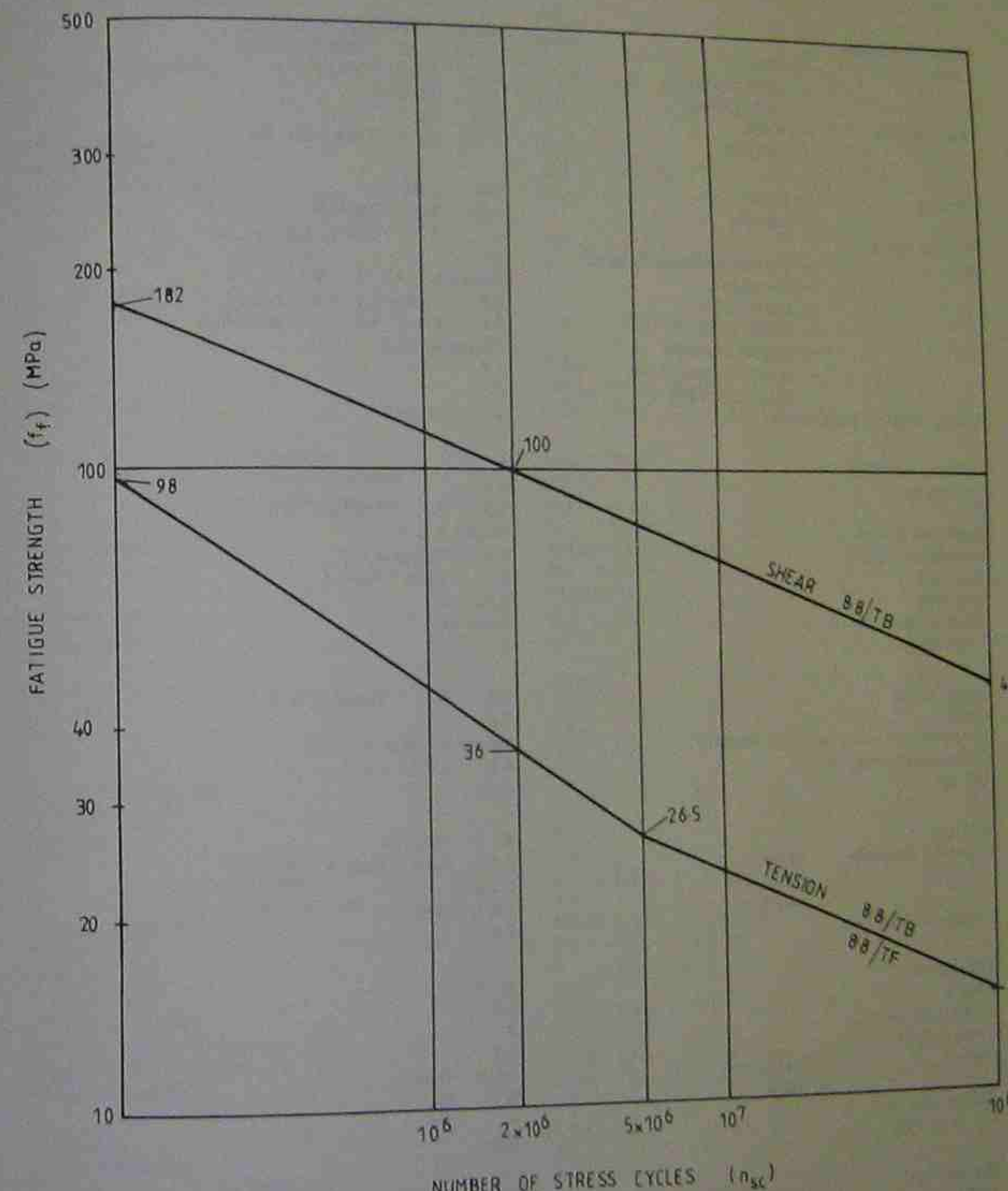
Bolt Size	Bolt Tension at Installation	Design Capacity in Shear (kN) for		
		k_h = 1	k_h = 0.85	k_h = 0.7
	kN			
M16	95	23.3	19.8	16.3
M20	145	35.5	30.2	24.9
M24	210	51.5	43.7	36.0
M30	335	82.1	69.8	57.5

SHEAR — TENSION INTERACTION DIAGRAM
k_h = 1.0



APPENDIX E DESIGN CAPACITIES OF COMMONLY USED BOLTS — FATIGUE LIMIT STATE

FIGURE E.1 HIGH STRENGTH STRUCTURAL BOLTS 8.8/TB and 8.8/TF BOLTING CATEGORIES



BOLTING OF STEEL STRUCTURES

Index

(References are to Section Numbers, or Tables)

Entry	Sub-entry	References
Abrasive blasting		5.4.5, Table 6
Area	— core	3.1, 5.3.3, Table 3
	— plain shank	3.1, 5.3.3, Table 3
	— tensile stress	3.1, Table 3
Axial tension	— mode	5.1, 5.2.1
	— capacity	5.1.2, 5.2.1, Tables C.1, C.2
Bearing	— failure	5.2.4, 5.3.5
	— capacity	5.2.4, 5.3.5, Table 3
Bearing type joint		4, 5.1, 5.2.4, 5.3
Bolt	— diameters commonly used	3.2
	— dimensions	Tables A.1, A.2, B.1, B.2
	— identification	3.3, Tables A.1, A.2, B.1, B.2
	— length selection	6.2, 6.3, 6.4, Tables 7, 8, 9
	— lengths available	Tables A.1, A.2
	— types in common use	3.2
	— usage summary	11
Bolting category designation		4, Table 2
Category designations		4
Clearances		8.4
Codes		5.3, 5.4, Tables 3, 4, 5
Combined actions		Tables 3, 5
Commercial bolts		3.2, Tables A.1, 7
Core area		3.1, 5.3.3, Table 1
Corrosion		8.3.5, 9
Costs in place		Table 24
Cyclic loading		5.5
Design codes		5.3, 5.4, Tables 3, 4, 5
Detailing		7.2
Dimensions of bolts, nuts and washers		Tables A.1, A.2, B.1, B.2
Direct tension indication		8.3.1, 8.3.3, 8.5.2
Drift pin		8.3.6
Economy		9, Table 24
Edge/End distances	— general	5.2.4, 5.3.5, Table 3, 7.2
	— maximum	7.2.2
	— minimum	7.2.1, Table 12
Erection clearances		8.6
Filler plates		5.3.4
Fatigue		5.5
Finishes for bolts		9
Force transfer		5.1
Friction	— coefficients	5.4.5, Table 6
	— mode	5.1, 5.4
Friction type joint		4, 5.4
Full tensioning		8.3
Galvanizing		5.4.5, 8.3.5, 9, Table 6
Grip		6.2, 6.3, 6.4, 6.6, Tables 7, 8, 9
Holes	— generally	5.4.4, 7.1
	— oversize and slotted	7.1, Table 10
	— punched	7.1, Table 11
High strength structural bolts		3.2, Table A.2, 8

Entry	Sub-entry	References
Hot dip galvanizing		5.4.5, 8.3.5, 9, Table 6
Identification		3.3, Tables A.1, A.2
Impact	— wrench	8.4
Inspection		8.5
Installation		8
Interaction		5.2.3, Tables 3, 5
Load/extension relationship		5.1
Load	— indicator washer	8.3.3
	— transfer	5.1
Lap splice connections		5.2.2, 5.3.2, Table 4
Lubricants		8.3.5, 9
Match marking		8.3.2, 8.5.2
Minimum bolt tension		8.3, Table 13
Molybdenum disulphide		8.3.5, 9
Nuts	— dimensions	Tables A.1, A.2
	— oversize	9
Oversize holes		7.1, Table 10
Packing		5.3.4
Painting		5.4.5, 9, Table 6
Part turn of nut		8.3.1, 8.3.2, Table 14, 8.5.2
Pitch	— of threads	3.1, Table 1
	— of holes, maximum	7.2.4
	— of holes, minimum	7.2.3
Plain shank length		6.1, Tables 7, 8
Ply in bearing		5.2.4, 5.3.5, Table 3
Power wrenches		8.2, 8.4
Proof loads		Tables A.1, A.2
Punched holes		7.1, Table 11
Prying forces		5.1.2
References		12
Relaxation		8.3.7
Re-use		8.3.8
Serviceability limit state		5.4
Shear and tension		5.2.3
Shank area		3.1
Shear	— area	5.3.3
	— capacity	5.1, 5.2.2, Tables 3, 5
	— bearing mode	5.1
Slip factors/slip coefficient		5.1.3, 5.4.5, Tables 5, 6
Slotted holes		7.1, 7.5, Table 10
Snug tightening		4, 8.2, 8.5.1
Strength grades		3.3
Strength Limit State		5.3, Table 3
Standardized connections		7.3
Surface coatings on plies		5.3.6, 5.4.5
Swage lock fasteners		8.3.3, Table B.2
Tear out		5.2.4, 5.3.5
Tensile stress area		3.1
Tension	— loads	5.1
	— capacity	5.1, 5.2.1, Tables 3, 5
Tensioning	— full	4, 8.3, 5.4.3
Thread	— form	3.1
	— length	6.1, Tables A.1, A.2
	— pitch	3.1, Table 1
	— projection	6.2, 6.5
	— run out	6.1

AISC

AUSTRALIAN INSTITUTE OF STEEL CONSTRUCTION
4th FLOOR, BP HOUSE, 110 ALFRED STREET
MILSONS POINT, NSW 2061
Telephone 929 6666

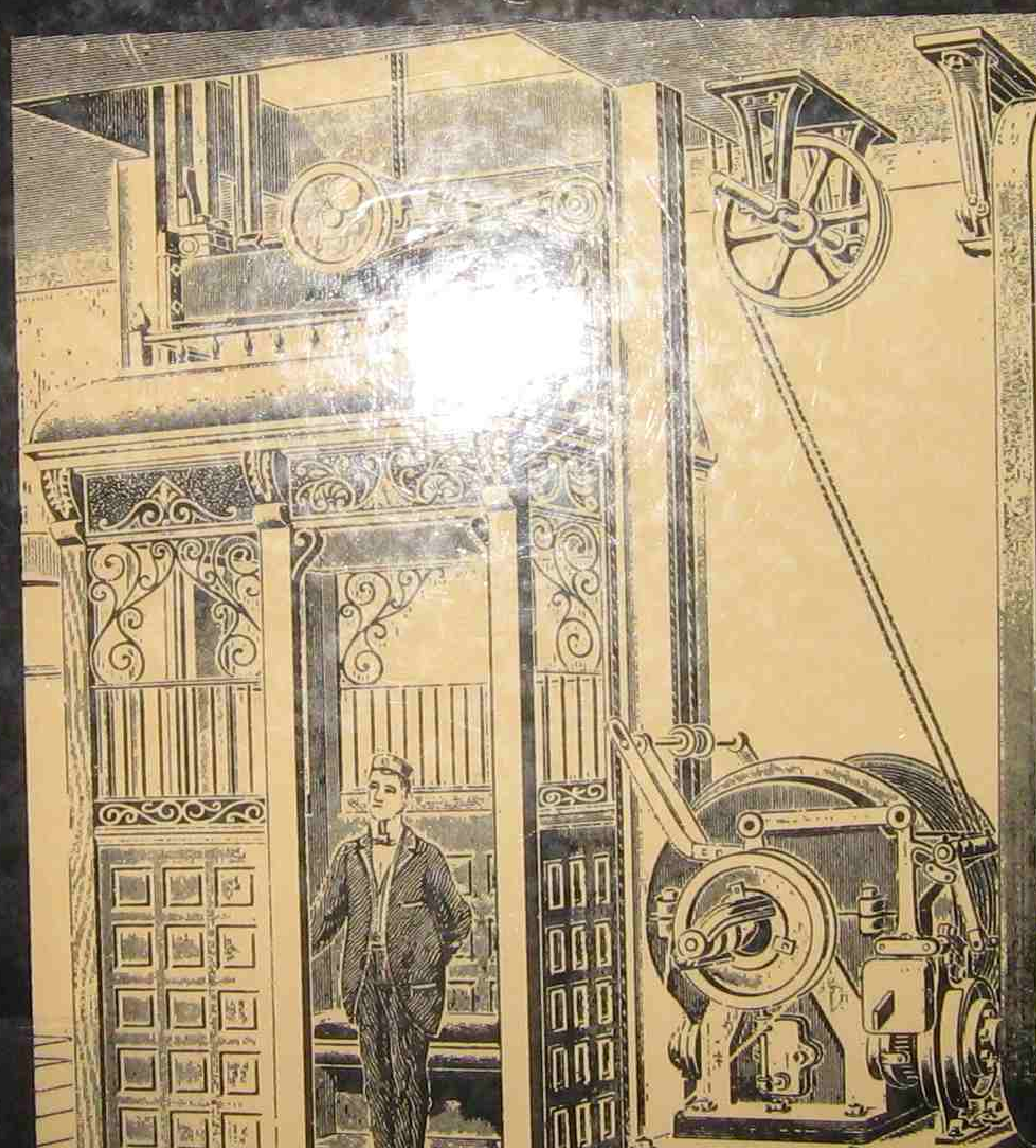
PRINTED BY ROBERT BURTON PRINTERS PTY LIMITED · [illegible] CARLINGFORD STREET [illegible]

Ellis Horwood Series in Transportation

ELEVATOR MECHANICAL DESIGN

principles and concepts

Lubomír Janovský



ELEVATOR MECHANICAL DESIGN:
Principles and Concepts

ELLIS HORWOOD SERIES IN TRANSPORTATION

Series Editor: F. T. BARWELL, Emeritus Professor of Mechanical Engineering, University College, Swansea

The development of transport technology depends on the interaction of scientific advance with social and economic problems. For example, the Maglev transit device, as recently introduced at Birmingham Airport, was rendered possible by the use of semiconductors in advanced electronic controls and the application of modern control theory. The justification for its development was partly environmental, notably the elimination of noise and vibration, and partly economic, i.e. un-manned operation and reduced maintenance cost.

The series treats various transport modes and devices from the functional viewpoint with particular emphasis on the solution of design problems from the economic viewpoint.

Bayesian Decision Methods in Transport Planning and Operations
A. Jessop, Polytechnic of Central London
Underground Transport Systems Technology
B. R. Pursall, University of Sheffield
Freight Transport Calculations
D. S. Wright, Cranfield Institute of Technology

VERTICAL TRANSPORTATION Section

Editor: Dr. G. C. Barney, Director of Networking, University of Manchester Regional Computing Centre and Research Director, Lift Design Partnership.

This important series of books covering the field of lift/elevator and passenger conveyors is published in collaboration with the International Association of Elevator Engineers.

The series will present authoritative texts by internationally renowned experts on all aspects of vertical passenger transportation, which fields include elevators or lifts, escalators and other passenger conveyors. By this means, engineers and others in the industry will have the benefit of the dissemination of knowledge, aiding the training and education of future generations and encouraging and enabling research and development into new advanced systems.

Elevator Technology
G. C. Barney, University of Manchester Regional Computing Centre
Elevator Mechanical Design: Principles and Concepts
L. Janovský, Prague Technical University
Elevator Abstracts including escalators
Editor: G. C. Barney, University of Manchester Regional Computing Centre

ELEVATOR MECHANICAL DESIGN
Principles and Concepts

LUBOMÍR JANOVSKY
Senior Lecturer
Faculty of Mechanical Engineering
Technical University of Prague
Czechoslovakia

Published for
INTERNATIONAL ASSOCIATION OF ELEVATOR ENGINEERS

by
ELLIS HORWOOD LIMITED
Publishers · Chichester

First published in 1987 by
ELLIS HORWOOD LIMITED
Market Cross House, Cooper Street,
Chichester, West Sussex, PO19 1EB, England
The publisher's colophon is reproduced from James Gillison's drawing of the ancient Market Cross, Chichester.

Distributors:
Australia and New Zealand:
JACARANDA WILEY LIMITED
GPO Box 859, Brisbane, Queensland 4001, Australia
Canada:
JOHN WILEY & SONS CANADA LIMITED
22 Worcester Road, Rexdale, Ontario, Canada
Europe and Africa:
JOHN WILEY & SONS LIMITED
Baffins Lane, Chichester, West Sussex, England
North and South America and the rest of the world:
Halsted Press: a division of
JOHN WILEY & SONS
605 Third Avenue, New York, NY 10158, USA

British Library Cataloguing in Publication Data
Janovský, Lubomír
Elevator mechanical design: principles and concepts. —
(Vertical transportation series)
1. Elevators
I. Title II. International Association of Elevator Engineers
III. Series
621.8'77 TJ1370

Library of Congress Card No. 86-27395

ISBN 0-7458-0135-8 (Ellis Horwood Limited)
ISBN 0-470-20804-X (Halsted Press)

Phototypeset in Times by Ellis Horwood Limited
Printed in Great Britain by Butler & Tanner, Frome, Somerset

Table of contents

Foreword . . . 7

List of symbols . . . 9

1 Definition of the mechanical system and component parts . . . 13

2 Suspension of car and counterweight
2.1 Methods of suspension, equalizing gear . . . 17
2.2 Elevator ropes . . . 20
(a) Specification, construction and recommendations for selection . . . 20
(b) Calculation, factor of safety . . . 30
(c) Suspension rope fastenings . . . 33
(d) Life, maintenance and replacement of elevator ropes . 37
(e) Compensating cables . . . 39

3 Types of drive
3.1 Traction drive . . . 42
(a) Roping systems . . . 42
(b) Traction sheave and diverting pulleys . . . 45
(c) Forces on sheave . . . 52
(d) Specific pressure of the rope in the sheave groove . . . 54
(e) Coefficient of friction of the ropes in the grooves . . . 60
(f) Traction under different conditions . . . 64
(g) Rope slip. Wear of sheave grooves . . . 70
3.2 Drum drive . . . 71

4 Gearing
4.1 Design and application . . . 74
4.2 Calculation . . . 80
4.3 Thermal performance calculation . . . 81

5 Brakes
5.1 Types of brakes 86
5.2 Calculation of braking torque. Brake selection 89

6 Counterweight 98

7 Guiding the car and counterweight
7.1 Shape, material and jointing of guide-rails.
Guide-rail brackets 101
7.2 Forces on guide-rails and stress analysis 104
7.3 Types of guide shoes 112

8 Safety gear
8.1 General 116
8.2 Overspeed governor 117
8.3 Instantaneous safety gear 122
8.4 Progressive safety gear 125

9 Car and counterweight buffers
9.1 Specification 134
9.2 Spring buffers (energy accumulation) 135
9.3 Oil buffers (energy dissipation) 142

10 Elevator well and machine room
10.1 General requirements for elevator wells 151
10.2 Well structure, equipment and assembly 152
10.3 Top clearances 153
(a) Traction drive elevators 153
(b) Positive drive elevators 155
10.4 The pit. Bottom clearances 156
10.5 Machine and pulley rooms 156
10.6 Sound insulation 158
(a) Elevator machine 158
(b) Controllers 161
(c) Sound sources located in the well 161

Index 165

Foreword

For many years a handbook dealing with the principles and concepts of electric elevators and the design of mechanical components in particular has been a serious omission. In fact only two books, concerning the field of vertical transportation, are in print at present namely *Vertical Transportation*, second edition, by Strakosch and *Elevator Traffic, Analysis, Design and Control* by Barney and dos Santos. The first book concerns elevatoring, and the process of applying elevator technology to buildings and though it discusses general engineering aspects of vertical transportation systems, it does not deal with the design of elevators themselves. The second book considerably extends the information available on traffic design, systems evaluation, performance measurement and control. This book should fill a gap.

It is the intent of the author to provide a detailed analysis and discuss all aspects of the mechanical design of elevators. Consequently the book contains an appropriate amount of theory with formulae and examples showing both the theoretical method of calculating elevator mechanical components as well as the practical methods for their selection. The information presents the 'state-of-the-art' as currently practised in Europe and the USA. In order that the book is as instructive as possible every section has been amply illustrated and a number of photographs are included. In contrast to many technical books published in the past this book attempts to tell its readers *how* to design an electric elevator from the mechanical viewpoint and also *why* the design is implemented in a particular way.

The book is aimed primarily for engineers, designers, manufacturers, elevator consultants and inspectors, but will also be useful to architects, safety personnel and students, and anyone with an interest or involvement in elevator design, operation, maintenance and safety.

Many of my friends have contributed to this work and their advice, encouragement and help are gratefully acknowledged here. My thanks go especially to Mr William C. Sturgeon, Editor of *Elevator World*, Mr George W. Gibson of Otis Elevator Co., Dr.Ing. Joris Schröder of Schindler Management AG. and Mr Adam Ehrlich of Freissler Otis Ges mbH.

The value of the book has been considerably enhanced by the generous assistance of the following companies and institutions, which I take this opportunity to thank most sincerely:

ASME (American Society of Mechanical Engineers)
British Ropes Ltd.
Dewhurst plc
DRAKO Drahtseilerei Gustav Kocks GmbH
Moline Accessories Company
Siecor Corporation, Republic Wire and Cable
Schindler Management AG.
The Express Lift Company Ltd.
United Technologies, Otis Elevator Company.

Thanks and appreciation are expressed here to the editor, Dr George C. Barney, for his remarkable assistance in the period of the preparation of this book.

Lubomír Janovský
Prague, Czechoslovakia
November, 1986

Authors Note: Those who are of pedagogical nature will note the interchange of *mass* and *weight* within this book. The author is aware of the relationship $W=mg_n$ but has bowed to the common usage in the elevator industry. Otherwise counterweight would be transposed into countermass.

List of symbols

A	heat generated by friction on a brake drum (kJ/h)
C	gear centre distance (mm)
C_1	coefficient taking account of acceleration, deceleration and specific conditions of the installation
C_2	coefficient taking account of the variation in profile of the sheave groove due to wear
D	pitch diameter of the sheave (mm or m)
D_1	worm reference diameter (mm)
D_s	central diameter of the spring (mm)
F	maximum static load imposed on elevator ropes (N) operating force to engage the jaws of a safety gear (N) retarding force of a buffer (N)
F_a	axial thrust (N)
F_o	initial retarding force of an oil buffer (N)
F_r	radial force (N)
F_t	tangential force (N)
F_x, F_y	forces imposed on guide-rails at right angles (N)
G	modulus of elasticity in torsion (shear) (N/mm^2)
H	height of travel (m)
I	moment of inertia of all moving parts of the system (kgm^2)
I_1	moment of inertia of the rotor, brake drum and worm (kgm^2)
I_2	moment of inertia of the worm wheel and sheave (kgm^2)
I_3	moment of inertia of all parts of the system in linear motion (kgm^2)
J	moment of inertia of the cross-sectional area of a guide-rail (mm^4)
J_p	polar moment of inertia (about the perpendicular axis at the centre of gravity of the sectional area) (mm^4)
K	mass of the car (kg)
M	sum of the mass of the empty car and the masses of appropriate portion of the travelling cable and any compensating device, suspended from the car (kg) total load on buffer (kg)

M_1	torque on the worm (Nm)
M_2	torque on the worm wheel (Nm)
M_b	braking torque (Nm)
M_d	dynamic torque (Nm)
M_{st}	static torque (Nm)
M_s	torque on the traction sheave (Nm)
M_t	torsional moment (Nmm)
N	minimum breaking load of one suspension rope (N) normal reaction force (N) number of teeth of the worm wheel
P	output of the driving motor (kW)
P_e	equivalent output (kW)
P_v	power loss (kW)
Q	rated load (kg) heat dissipated from a worm gearbox (kJ/s)
R	resultant reaction force (N)
S	outer surface of a gearbox (m^2) cross-sectional area of a guide-rail (mm^2)
S_p	piston area (m^2)
T	generally tensile force in suspension ropes (N) tangential reaction force (frictional resistance) (N)
T_1	greater static tensile force in suspension ropes on either side of the sheave (N)
T_2	smaller static tensile force in suspension ropes on either side of the sheave (N)
Z	mass of the counterweight (kg)
a	generally acceleration (m/s^2) braking deceleration of the car (m/s^2)
a_o	initial retardation (m/s^2)
b	width of the undercutting of a sheave groove (mm) width of the car (mm)
c	depth of the car (mm) stiffness of the spring (N/mm)
d	nominal rope diameter (mm) diameter of the wire of helical spring (mm)
e	base of natural logarithms
e_y, e_z	eccentricity of the centre of gravity of the rated load in a car (mm)
f	factor of safety of suspension ropes coefficient of friction in a sheave groove
f_r	resonance frequency (1/s)
g_n	standard acceleration of free fall (m/s^2)
h	generally vertical distance (mm or m) total buffer stroke (m)
i	roping factor radius of gyration (mm)
i_G	gear ratio

k	heat transfer coefficient (kJ)/($m^2 \times K \times s$)
l	generally horizontal distance (mm) length of the threaded portion of a drum (mm) length of the spring exposed to torsion (mm)
l_k	maximum distance between guide-rail brackets (mm)
m_L	mass of one fall of suspension ropes (kg)
m_b	mass of supporting beams under the elevator machine (kg)
m_m	mass of elevator machine including the frame (kg)
m_s	mass of the source of vibration (kg)
n	number of suspension ropes number of starts of a worm number of active threads of a spring
n_1	rpm of the worm
n_2	rpm of the motor at the beginning of braking
n_m	rpm of the motor
p	specific pressure (N/mm^2)
q_c	unit weight of travelling cables (kg/m)
q_k	unit weight of compensating cables (kg/m)
q_L	unit weight of suspension ropes (kg/m)
q_o	total area of all escape holes of an oil buffer (m^2)
q_y	total area of escape holes below the piston (m^2)
s	stiffness of resilient mounting of the elevator machine (N/m)
t	lead of thread of a drum (mm)
t_b	braking time (s)
v	rated speed (m/s) velocity of the piston of an oil buffer (m/s)
v_c	rope speed (m/s)
v_p	circumferential velocity on worm reference diameter (m/s)
w	discharge velocity (m/s)
y	variable length of the travelling cable under the car (m)
z	number of operations per hour (1/h) variable distance from the car to its lowest level (m)
α	angle of wrap of the traction sheave angle of the wedge (instantaneous safety gear)
α_n	normal pressure angle (worm gearing)
β	angle of undercutting of a sheave groove
γ	angle of vee groove oil density (kg/m^3)
δ	angle of outer normal lines of the contact area of a round seated groove or undercut groove
ε	angular retardation ($1/s^2$)
ϕ	angle of contact in a radial plane of a round seated or undercut groove angle of friction of worm gearing angle of torsion (rad)
ϕ_1	angle of friction between the jaw and the supporting block of an instantaneous safety gear

ϕ_2	angle of friction between the jaw and the guide-rail
λ	lead angle of the worm thread coefficient of slenderness
μ	actual coefficient of friction between the rope and a sheave groove coefficient of friction of worm gearing discharge coefficient
η_G	tooth efficiency of worm gearing for worm driving
η'_G	tooth efficiency of worm gearing for worm wheel driving
η_L	efficiency of a bearing
η_o	overall efficiency of worm gearing for worm driving
η'_o	overall efficiency of worm gearing for worm wheel driving
η_{RS}	efficiency of the roping system
η_s	efficiency of the sheave
η_2	mechanical efficiency of the system related to the conditions of braking
ψ	coefficient taking account of the percentage of the rated load balanced by the counterweight Wahl's coefficient
σ_k	stress in the guide-rails during the safety gear operation (N/mm^2)
θ	coefficient relating M and (Q + k)
$\Delta\theta$	temperature increment (K)
θ_a	temperature of ambient air (°C)
θ_L	maximum permissible temperature of oil in the gearbox (°C)
ω	angular velocity (1/s) buckling factor

1

Definition of the mechanical system and component parts

Elevator system

The elevator is defined as a permanent lifting equipment serving two or more landing levels, including a car for transportation of passengers and/or other loads, running at least partially between rigid guides either vertical or inclined to the vertical by less than 15°.

Elevators may be classified by several characteristics. The most important characteristic is the drive method, which results in different design principles and different construction of the elevator components.

The classification from this aspect is as follows:

(1) electric elevators,
(2) hydraulic elevators,
(3) pneumatic elevators.

Electric elevators may be:

counter weighted by

(a) *traction drive*, the suspension ropes of which are driven by friction in the grooves of the driving sheave of the machine,
(b) *positive drive*, which are suspended by chains or ropes driven by means other than friction (drum drive).

The British Standard BS 5655 : Part 5 : 1981 distinguishes seven kinds of electric traction elevators and specifies their outline dimensions related to loads, speeds and installation arrangements.

The principal technical parameters of elevators are rated load Q

(kg) and rated speed v (m/s). The equipment is built to these parameters and normal operation is then guaranteed by the manufacturer.

In Europe the values of rated loads have been selected from the R10 series of ISO preferred numbers, while the selection of rated speeds has been accomplished on the basis of the R5 series. Rounded-off values are shown in Table 1.1. Numbers in the upper line represent values of the R5 series, while all numbers are valid for the R10 series.

Table 1.1 — Values of R5 and R10 series

R5 series — multiplying factor $\sqrt[5]{10} \sim 1.6$
R10 series — multiplying factor $\sqrt[10]{10} \sim 1.25$

1.0		1.6		2.5		4.0		6.3		10.0
	1.25		2.0		3.15		5.0		8.0	

The mass of one passenger for the purpose of any calculation is 75 kg.

Further technical parameters are:

(a) height of travel (car rise), the number and location of stops,
(b) dimensions of elevator well, car and machine room,
(c) voltage of the main supply, the number of starts per hour and load factor,
(d) control system,
(e) landing and car doors arrangement and the type of operator,
(f) the number of elevators and their location in the building,
(g) environmental conditions.

The main parts of electric elevators are:

(a) *Suspension means* for car and counterweight, which are represented by either steel wire ropes or chains.

(b) *Driving machine*, which is the power unit, consisting of:

— electric motor,
— mechanical gearing,
— brake,
— sheave, drum or chain sprockets,

2

Suspension of car and counterweight

2.1 METHODS OF SUSPENSION, EQUALIZING GEAR

Cars and counterweights are suspended to steel wire ropes, roller chains or chains with parallel links (Galle type). Since chain elevators are not frequently used nowadays, attention will be given to the rope suspension.

Elevator ropes are attached to the crosshead of the car frame or pass around multiplying pulleys mounted on the crosshead, if a roping system other than 1:1 is employed. Suspension of each rope must be independent.

The minimum number of suspension ropes are three for traction elevators and two for positive drive ones in the USA, while they are two in Great Britain irrespective of the type of drive.

An automatic device for equalizing the tension of suspension ropes should be provided at least at one of their ends. This device is usually represented by an equalizing gear of individual compression spring type. Such a device, consisting of a wedge-type socket equipped with a steel helical spring is shown in Fig. 2.1, with rubber buffers in Fig. 2.2. While the dependence of the axial force upon the compression is linear with helical springs, it is almost linear with rubber buffers in the range of practical forces (up to the maximum allowable compression), but the curve becomes much steeper for excessive forces (Fig. 2.3).

It is necessary to emphasize that the equalizing device of the compression spring type cannot ensure even distribution of tensile forces in the individual suspension ropes. Its application results in favourable load distribution in comparison with rigid suspension but

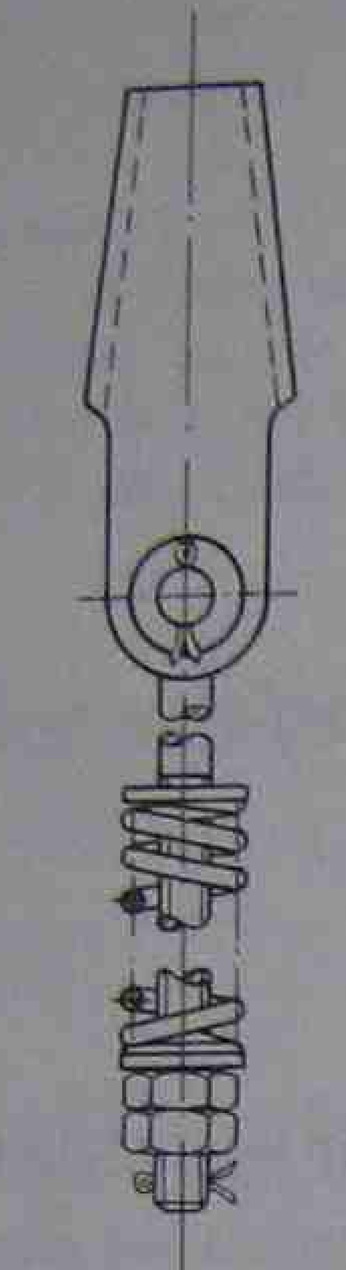


Fig. 2.1 — Suspension device with steel helical spring.

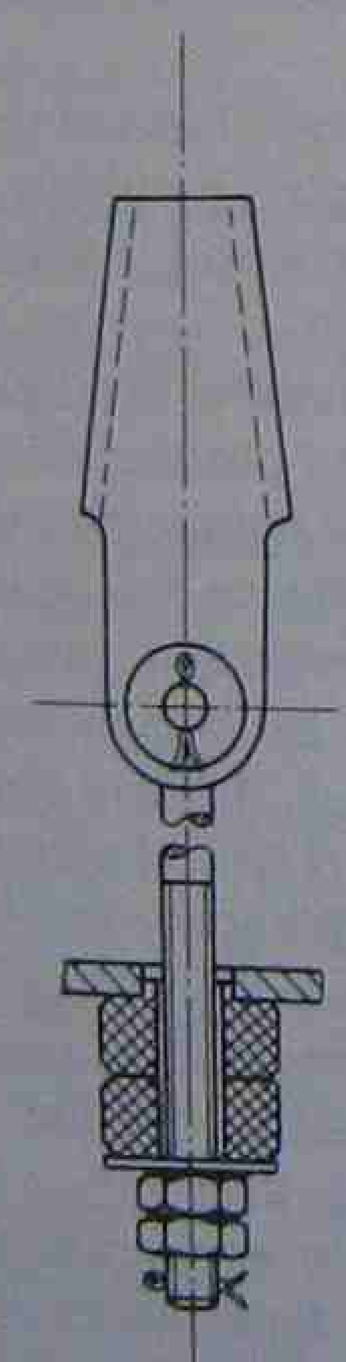


Fig. 2.2 — Suspension device with rubber buffers (Gustav Kocks GmBH).

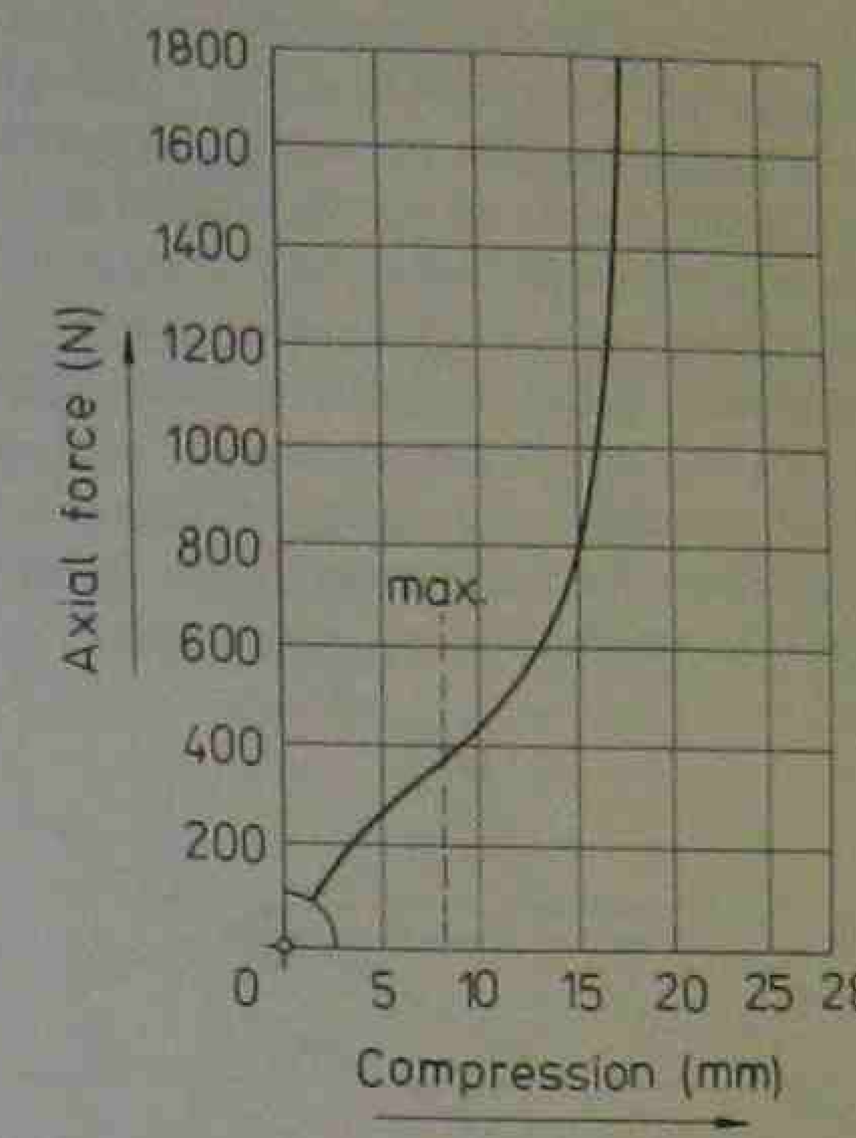


Fig. 2.3 — Axial force to compression diagram for rubber buffers.

in general perfect equalizing cannot be achieved.

The most favourable load distribution on the individual suspension ropes can be accomplished with a balance mechanism, composed of levers and pivots; in Fig. 2.4 schemes for two, three and four ropes are shown. The mechanism is very simple for two ropes; each of them passes over a rope gatherer (not shown in Fig. 2.4) and is attached at one side of the horizontal balance lever, which is free to rotate as one rope stretches more than the other, takes up an inclined position and consequently both ropes are subjected to the same tension. Mechanisms for more suspension ropes are complicated and only exceptionally employed nowadays.

It is of great significance that all suspension ropes be properly installed and anchored at reasonably the same length. It is very difficult to achieve all ropes in equal tension, as many factors affect the load distribution on individual ropes, but improper adjustment at the attachments should be avoided.

There are several methods of checking tension in the ropes. Experienced service men or inspectors should be able to detect differences in tension by feeling the resistance each rope offers to a

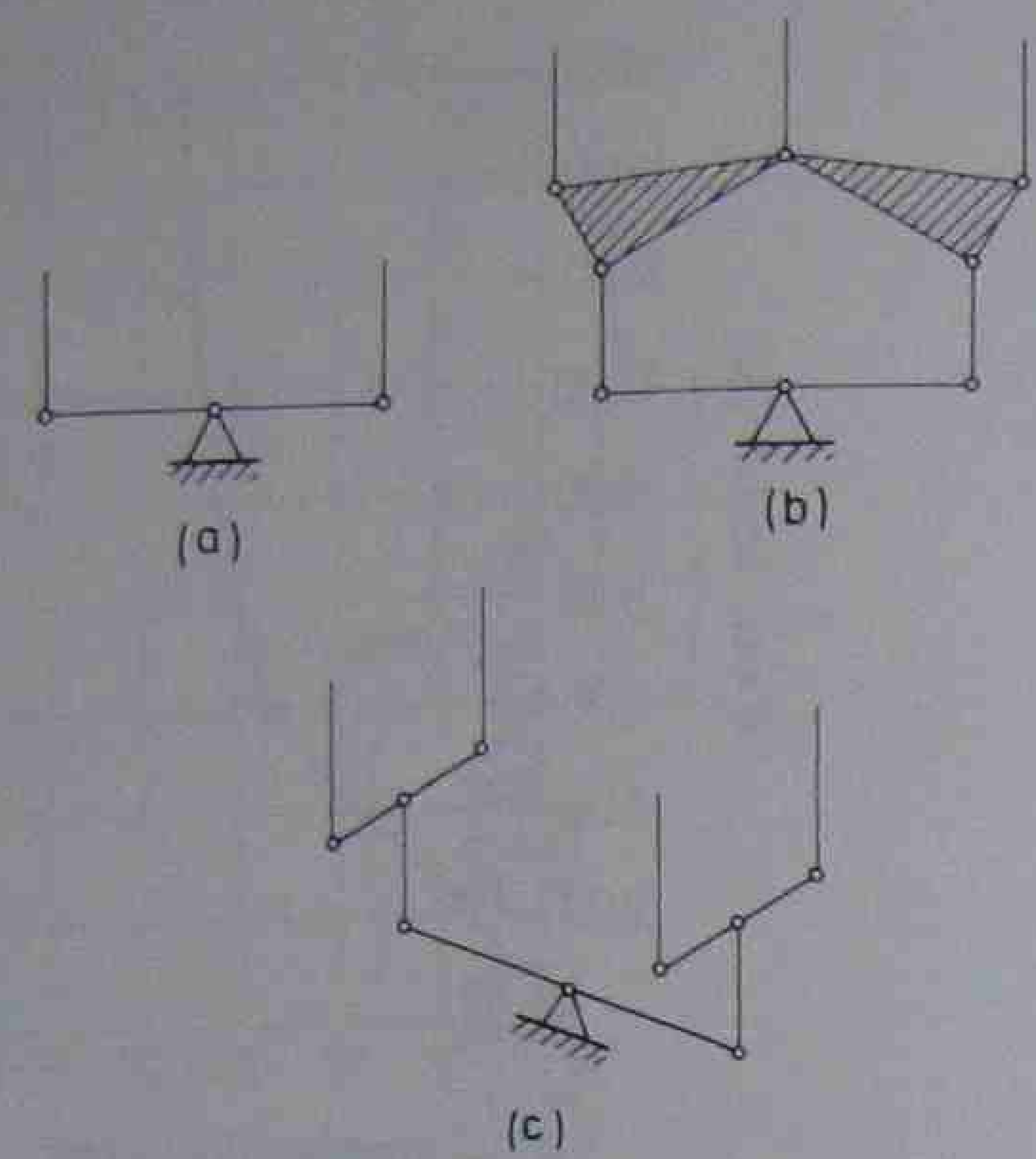


Fig. 2.4 — Balance mechanism for (a) two, (b) three and (c) four suspension ropes.

horizontal pull or by observing the rope vibrations after plucking, however the application of tension meters should be given preference. This device measures rope bending on a fixed length, which is proportional to the rope tension.

2.2 ELEVATOR ROPES*

(a) Specification, construction and recommendations for selection

Elevator ropes are of round stranded construction. The wire material for ropes is manufactured by the open-hearth or electric furnace process or their equivalent.

The tensile grade of wires may be the same throughout the rope — ropes of this kind are known as 'single tensile'. Alternatively, the outer wires may be all of one tensile grade, which is lower than the tensile grade of the inner wires. These ropes are called 'dual tensile'.

The size of the rope is identified by its nominal diameter, which is given by the rope manufacturer. The actual diameter of the rope must be within plus 3% and minus nil of the nominal diameter, when

* IMPORTANT NOTE: At the time of writing both ISO 4344 and BS 302 Standards are emerging. Readers are requested to refer to these standards for specific details, when design projects are being undertaken.

measured under a tension equal approximately to 10% of the minimum breaking load. As a measure of the quality of the rope manufacture the actual diameter measured on a straight portion of the rope under no tension should be within plus 6% and minus nil of the nominal diameter. The actual diameter should be measured carefully at two points located at least one metre apart and at each point two measurements should be taken at right angles. The average value of these four measurements should be within the tolerance mentioned above and the maximum difference between any of the four measurements should not exceed 3% of the nominal diameter. The method of measuring rope diameter is shown in Fig. 2.5.

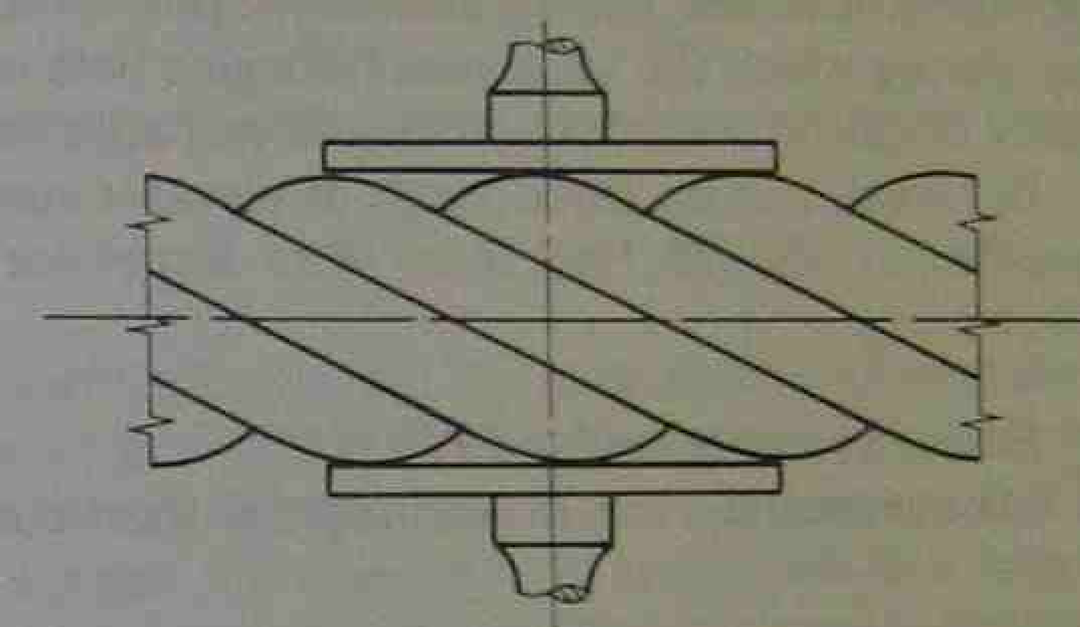


Fig. 2.5 — The method of measuring rope diameter.

A suitable caliper fitted with jaws of sufficient width to cover not less than two adjacent strands must be used.

The nominal diameter of elevator ropes must be at least 8 mm in Great Britain and 3/8 in. (9.5 mm) in the USA. The American Safety Code ANSI/ASME A17.1 specifies the minimum diameter for the outer wires as 0.61 mm.

Elevator ropes are of equal lay construction, in which the wires in the strand are so spun that they all have an equal lay length. The lay length is the distance in a strand or rope, measured parallel to the longitudinal axis, in which the wire in the strand or the strand in the rope makes one complete turn about the axis of the strand or rope respectively. Furthermore, elevator ropes are either of ordinary (regular) lay or Lang's lay. With Lang's lay ropes the direction of lay of the strands is the same as that of the outer wires in the strands, while in the ordinary construction the direction of lay of the outer layer of wires in the strands is opposite to the direction of lay of the strands in the rope. The advantage of Lang's lay ropes is a larger contact surface and consequently a lower unit pressure between the wires of particular strands resulting in a longer life of the rope. On the other hand, ordinary lay ropes can be handled more easily as the tendency to kink and untwist is smaller than with the Lang's lay. Elevator ropes are usually of right-hand lay, which means that the strands are laid up to the right (clockwise direction when looking at the rope end).

For bad environmental conditions, e.g. for outside elevator installations or elevators in chemical plants exposed to extremely corrosive atmosphere, ropes with galvanized wires may be supplied. The wires are provided with zinc protective coating. Two methods of galvanizing may be applied depending on the application for which the rope is intended, namely electrolytic process and hot dip galvanizing, during which the wires pass through a bath of molten zinc. The purity of zinc in protective coating closely approaches 100%.

In Great Britain three kinds of elevator suspension ropes are specified in BS 329:1968 Steel Wire Ropes for Electric Lifts, as follows.

(i) Round strand equal lay 6×19 (9/9/1)

A cross-sectional view of the rope is shown in Fig. 2.6 and all

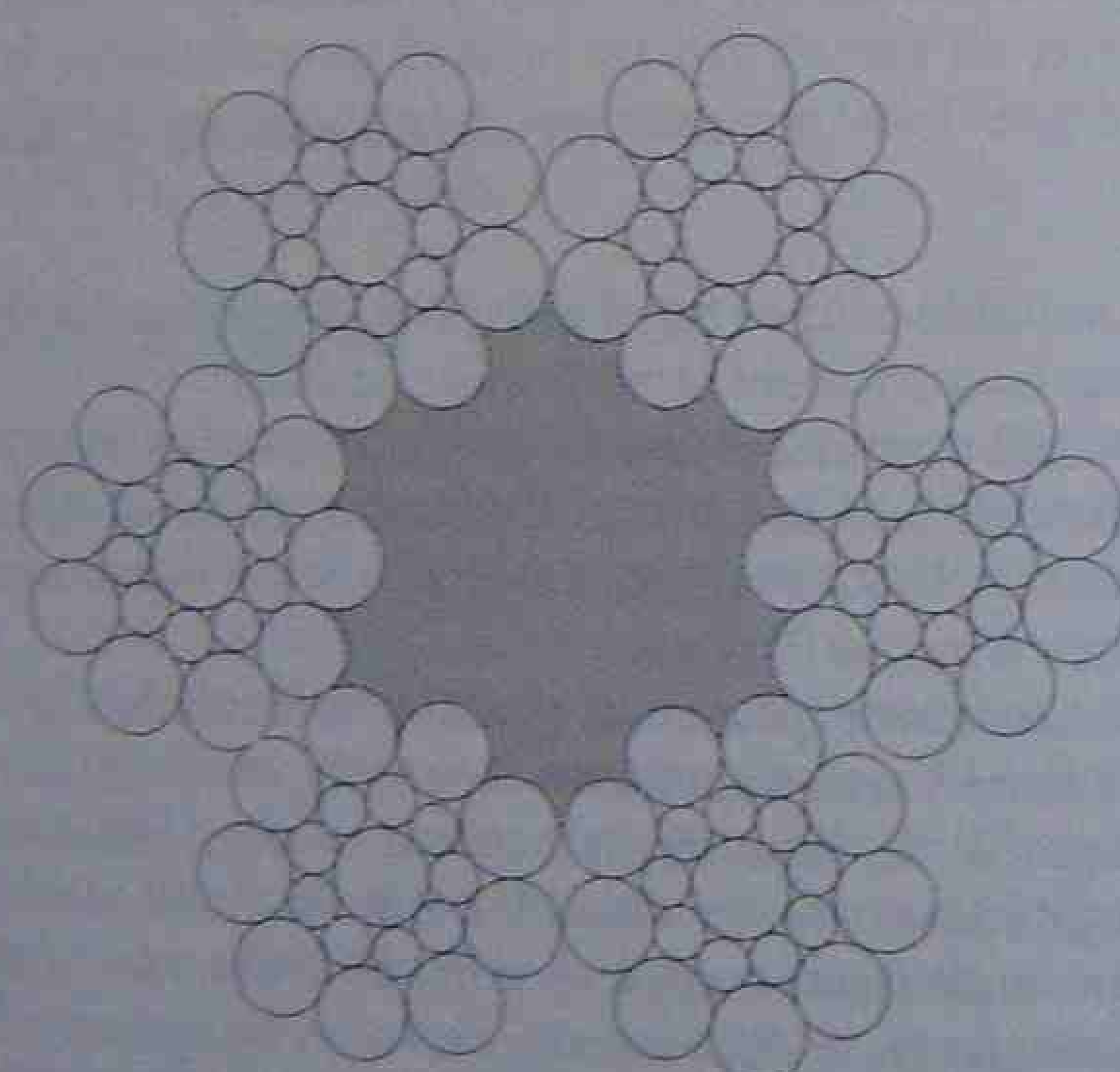


Fig. 2.6 — Round strand equal lay 6×19 (9/9/1) suspension rope.

technical data are quoted in Table 2.1.

The construction 6×19 (9/9/1) means 6 strands, 19 wires in each strand, namely 9 wires in the outer layer, 9 inner wires and 1 central

Table 2.1 — Technical data of 6x19 (9/9/1) elevator rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN) at wire tensile grade (N/mm²) 1370 1170/1770* 1370/1570*	1570
6.5	15.3	19.2	22.0
8	23.2	29.1	33.3
9.5	32.7	41.0	46.9
11	43.8	55.0	62.9
13	61.1	76.8	87.8
16	92.6	117	133

* Dual tensile.

wire (king wire) in the strand. The same number of both the outer and inner wires is typical for the Seale construction. The outer wires are of larger diameter, which results in larger contact area and lower unit pressure between the wires of both layers and as a consequence a considerable improvement in rope life is achieved in comparison with ropes of normal construction used earlier such as 6×19 (12/6/1).

Table 2.2 — Technical data of 6x19 (12/6+6F/1) elevator rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN) at wire tensile grade (N/mm²) 1370 1170/1770* 1370/1570*	1570
8	23.6	29.7	33.9
9.5	33.3	41.9	47.9
11	44.7	56.1	64.1
13	62.4	78.4	89.5
16	94.5	119	135
19	133	168	191
22	179	225	257

* Dual tensile.

(ii) Round strand equal lay 6×19 (12/6+6F/1)

Its cross-section is shown in Fig. 2.7 and technical data are quoted in Table 2.2.

In this rope there are six filler wires of small diameter inserted between the inner and outer wires in each strand filling the interstices between both wire layers. They improve the contact between the layers and facilitate the rope to retain its shape. In calculating the rope strength the filler wires are considered as not bearing any load.

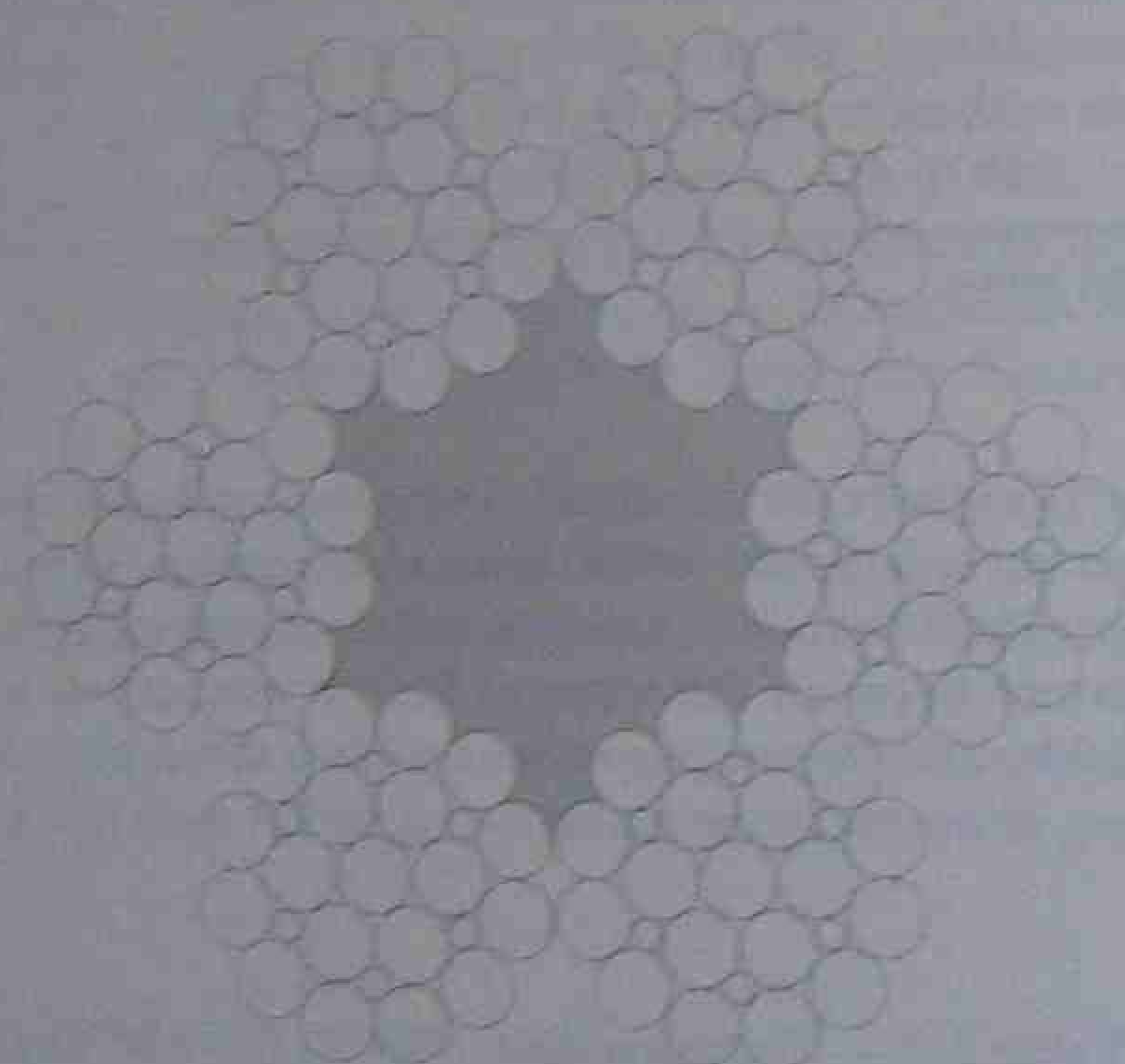


Fig. 2.7 — Round equal lay 6×19 (12/6—6F/1) suspension rope.

(iii) Round strand equal lay 8×19 (9/9/1)

The rope is of the Seale construction, but in contrast to (i) it is composed of eight strands instead of six. It is superior to a six-strand rope in several respects: it is more flexible and fatigue-resistant; it conforms better to the shape of the sheave grooves and is smoother running; the contact area between the wires and the grooves is larger; it sustains a greater number of bendings so that its life may be longer. The rope has less resistance to abrasion, when compared with the equivalent size 6×19 (9/9/1) ropes, because the outer wires are of

a smaller diameter. Also the breaking load is lower than with the 6×19 rope.

In Fig. 2.8 the rope construction is depicted and the technical data

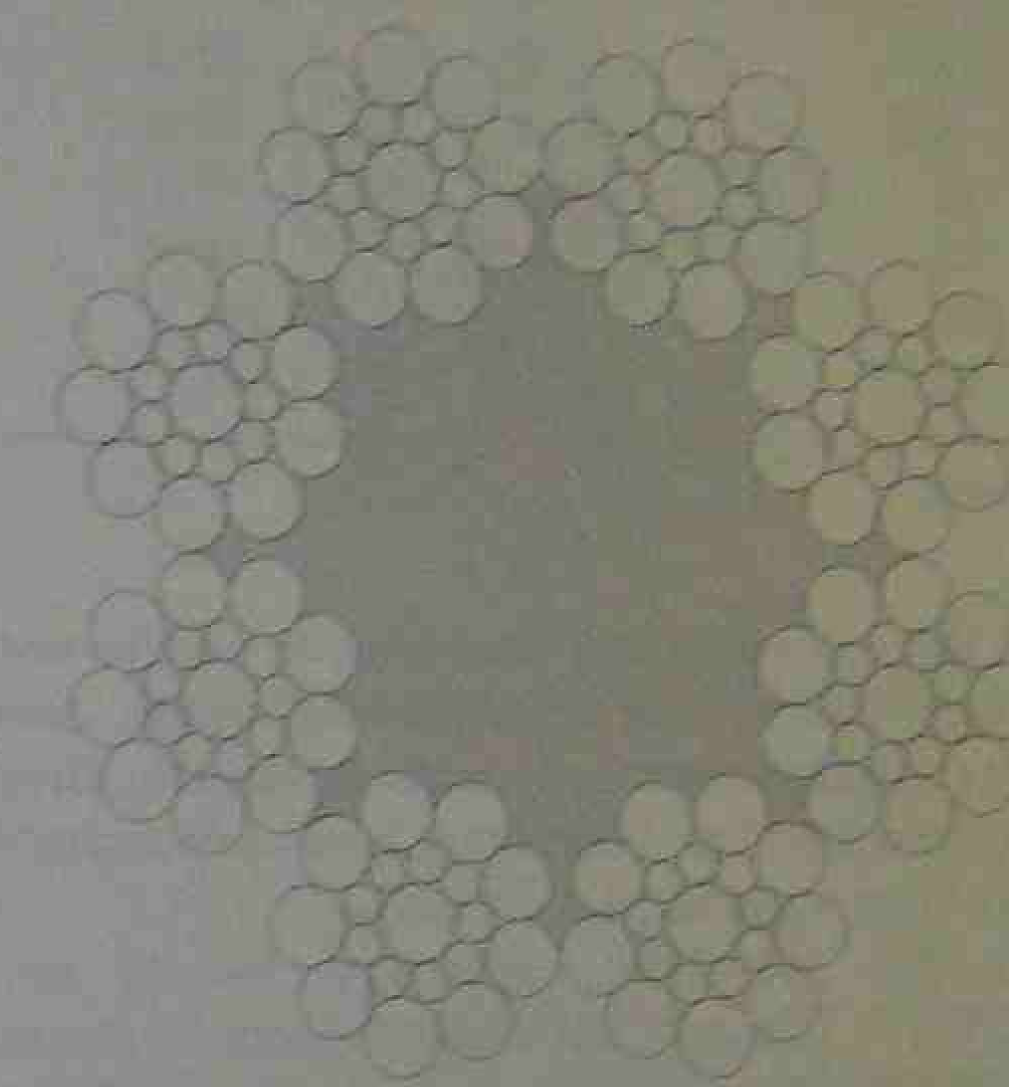


Fig. 2.8 — Round strand equal lay 8×19 (9/9/1) suspension rope.

are reviewed in Table 2.3.

All three kinds of suspension ropes have fibre cores, impregnated with a special lubricant in order to reduce friction between the inner wires and extend their life by preserving them from deterioration from the results of dampness especially when the rope is not in use.

The actual breaking load of the rope must not be less than the minimum value specified in Tables 2.1 to 2.3. It is detected by a tensile test to destruction of a sample of the rope (it should be done for each rope production length) and recorded by the manufacturer. If the actual breaking load is not known the minimum value must be used in calculating the factor of safety.

The weights of ropes given in Tables 2.1 to 2.3 are calculated values only.

Table 2.3 — Technical data of 8x19 (9/9/1) elevator rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN) at wire tensile grade (N/mm^2) 1370 1170/1770* 1370/1570*	1570
8	21.6	25.2	28.8
9.5	30.5	35.6	40.6
11	40.9	47.7	54.5
13	57.2	66.0	76.1
16	86.5	101	116
19	122	142	163

*Dual tensile.

All three kinds of suspension ropes may be also used as ancillary ropes (e.g. governor or compensating ropes). Another two kinds of ancillary ropes are currently manufactured in Great Britain, namely 8×19 (6 and 6/6/1) of Warrington construction and 8×19 (12/6+6F/1) with six filler wires in each strand.

The BS 329:1968 Standard is shortly to be replaced by BS302:1987. The new standard is likely to recommend the use of 1370/1770 N/mm^2 grade dual tensile and 1570 N/mm^2 grade single tensile ropes. The BS 302:1987 and ISO 4344 refer to two main kinds of ropes 6×19 (with 6 strands, 8 to 12 outer wires per strand) and 8×19 (with 8 strands, 8 to 12 outer wires per strand) typical constructions being: 6×19 (9/9/1), 6×19 (12/6+6F/1), 8×19 (9/9/1) and 8×19 (12/6+6F/1).

British Ropes Ltd have introduced the so-called Dyform elevator ropes produced by the drawing of a normal round strand formation through a die, this technique resulting in an increased metallic area in the finished product. The increase of metallic area leads to the reduction of stretch compared with a conventional round strand rope with fibre core and limits the necessity for rope length readjustment. Furthermore, the minimum breaking load is increased while the tensile strength of the wires is held to a level which will give optimum fatigue resistance. Since the surface of the strand is enlarged the specific pressure between the wires and the sheave groove is smaller and consequently the life of the groove is extended. Smooth and quiet operation seem to be further advantages. Dyform ropes of two constructions are manufactured: 6×26 (10/5 and 5/5/1), shown in

Fig. 2.9 and 8×19 (9/9/1), shown in Fig. 2.10, both with fibre cores. The main parameters are quoted in Tables 2.4 and 2.5.

The rope manufacturer Gustav Kocks GmbH in the Federal Republic of Germany has developed an elevator rope especially suitable for high speed and high rise heavy duty installations, designated DRAKO 300T. Fig. 2.11 depicts the cross-section of the rope

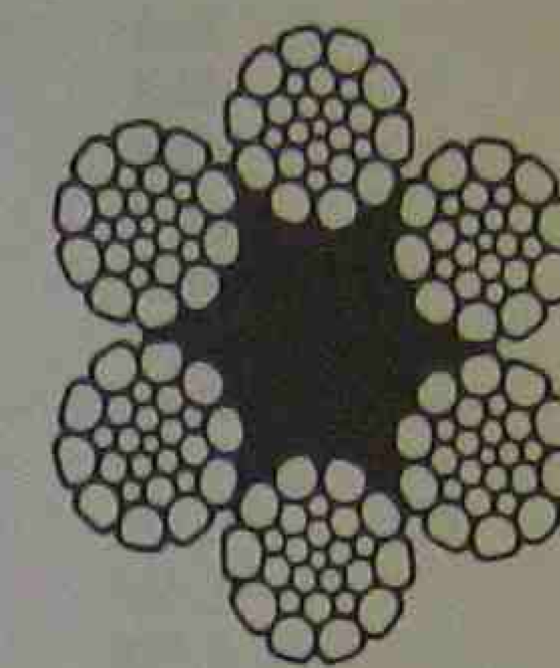


Fig. 2.9 — Dyform rope 6×26 (10/5 and 5/5/1).

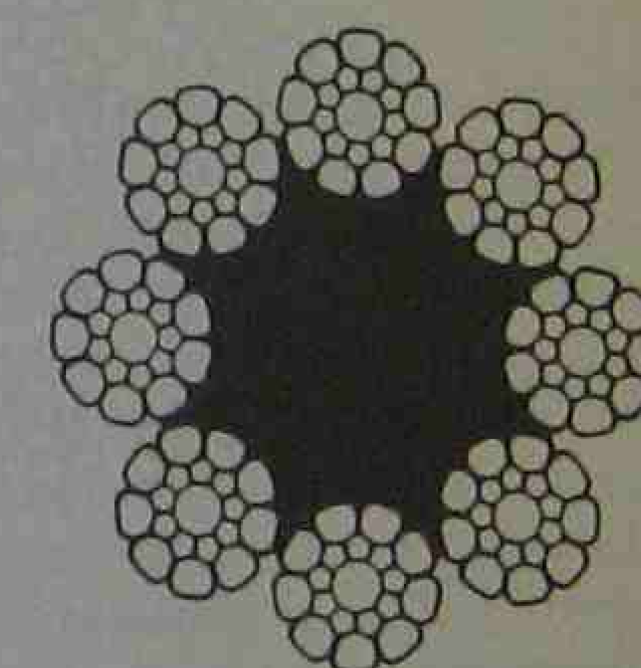


Fig. 2.10 — Dyform rope 8×19 (9/9/1).

Table 2.4 — Technical data of 6x26 (10/5+5/5/1) Dyform rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN)
8	25.6	33.2
9.5	36.1	52.0
11	48.4	70.6
13	67.6	95.1
15.3	81.8	118
16	102	147

Table 2.5 — Technical data of 8x19 (9/9/1) Dyform rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN)
8	23.4	31.4
9	29.6	40.8
10	36.5	53.0
11	44.2	61.7
12	52.6	74.6
13	61.7	91.2
15.5	87.7	128
16	93.4	135

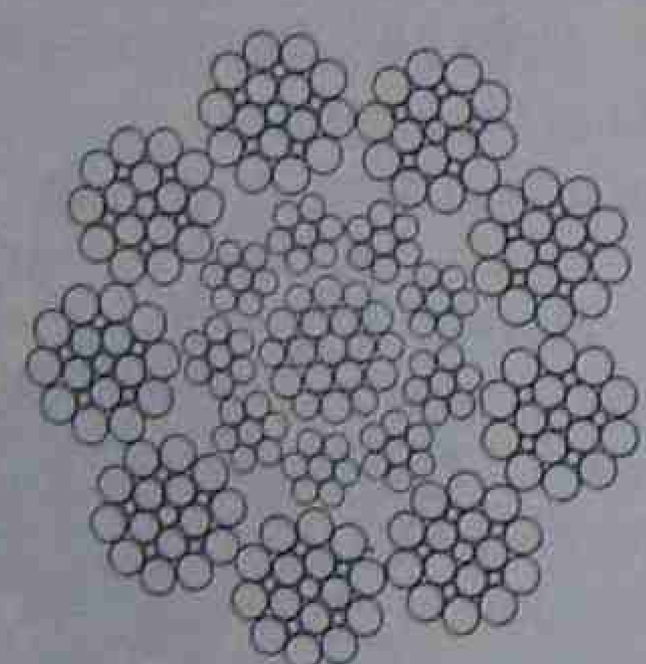


Fig. 2.11 — DRAKO 300T suspension rope.

for the range of nominal diameters 10 to 16 mm, while its technical data are quoted in Table 2.6. The rope is of preformed 9-strand construction of ordinary lay with an integrated wire rope centre. The construction of the strands is 9×16 (10/5+5F/1) with five filler wires in each strand. The integrated rope centre is made of 19 wires and 9 strands of 7 wires each in Lang's lay arrangement. All wires are of the same tensile grade: 1570 N/mm^2. The strands for rope diameters in excess of 16 mm are of the same construction as the British ropes designated in (ii).

The specific load on each wire is reduced with this rope because of

Table 2.6 — Technical data of DRAKO 300T elevator rope

Nominal diameter (mm)	Outer wire diameter (mm)	Metallic area (mm^2)	Weight (kg/100 m)	Breaking load (kN) at wire tensile grade 1570 N/mm^2	
				Calculated	Minimum
10	0.60	46.5	42	73.5	55.1
11	0.65	56.1	50	88	64.6
12	0.70	67.9	59	107	80
13	0.75	79.7	70	125	94
14	0.82	91.1	81	143	107
15	0.90	104.0	93	163	127
16	0.95	120.5	107	189	143
18	0.85	159.6	142	250	188
20	0.95	196.3	175	308	231
22	1.05	237.8	212	373	280

its larger metallic area. The suspension characteristic becomes stiffer because of the special wire core being employed so that both permanent and elastic stretching under different load conditions are reduced to a minimum. The manufacturer states it is less than 50% of the deformation currently occurring with steel ropes with fibre cores. This rope property reduces and in some cases even eliminates the need for automatic relevelling of the elevator car. DRAKO 300T has a smoother and larger surface than comparable ropes of six- or eight-strand construction resulting in an improved contact between the rope and the sheave leading to the decrease of the unit pressure and longer life of both the rope and the sheave grooves.

The difference between the calculated and actual minimum values of breaking load is instructive.

Elevator ropes are preformed as a rule in order to reduce internal stresses created during the current manufacturing process. Preforming is a method of pre-setting the wires in the strands and the strands in the rope, into a permanent final shape which they will take up in the completed rope. Preformed ropes are superior to those made by the usual method of manufacture in several respects:

- the load is evenly balanced on individual strands and wires
- the tendency to kink and untwist is decreased considerably and

the ropes are easier to handle; the rope ends need not be excessively seized to prevent unwinding

- in the event of the breakage of an outer wire the portion which is free does not tend to straighten and stick out of the rope; this prevents the adjacent wires as well as the sheave or pulleys from consequent damage.

For the reduction of rope stretch some manufacturers supply elevator ropes dynamically prestretched, suitable particularly for high rise installations or installations with bottom mounted machines.

(b) Calculation, factor of safety

In calculating elevator rope tension static load is considered only. Consequently the factor of safety must be rather high as it covers the effect of additional stresses such as bending stresses when the rope passes over the sheave and/or pulleys and initial internal stresses in wires due to the method of rope manufacture, the influence of inertial forces in the rope during acceleration and deceleration periods and the effect of uneven distribution of the load because of an imperfect rope equalizer being employed.

In compliance with BS 5655: Part 1: 1979 and EN 81: Part 1, the safety factor of the suspension ropes must be at least

12 in the case of traction drive with three ropes or more
16 in the case of traction drive with two ropes
12 in the case of drum drive

The factor of safety is the ratio between the minimum breaking load of the rope and the maximum static force in this rope and is calculated by the following formula:

$$f = \frac{n \times i \times N}{F} \tag{2.1}$$

where N is the minimum breaking load of one rope (N) (the actual breaking strength, if given by the manufacturer, may be used for calculation); n is the number of suspension ropes; i is the roping factor, i.e. 1 for 1:1 roping, 2 for 2:1 roping, etc. F is the maximum static load imposed on elevator ropes at any position in the well. For the calculation of this force the car is usually supposed to be stationary at the lowest level with its rated load; the effect of the mass of the car, the mass of elevator ropes, the mass of the appropriate

Table 2.7 — Minimum factors of safety for suspension wire ropes (A17.1)

Rope speed (m/s)	Elevators	
	Passenger	Freight
0.25	7.60	6.65
0.38	7.75	6.85
0.50	7.95	7.00
0.63	8.10	7.15
0.76	8.25	7.30
0.88	8.40	7.45
1.00	8.60	7.65
1.125	8.75	7.75
1.25	8.90	7.90
1.50	9.20	8.20
1.75	9.50	8.45
2.00	9.75	8.70
2.25	10.00	8.90
2.50	10.25	9.15
2.75	10.45	9.30
3.00	10.70	9.50
3.25	10.85	9.65
3.50	11.00	9.80
3.75	11.15	9.90
4.00	11.25	10.00
4.25	11.35	10.10
4.50	11.45	10.15
4.75	11.50	10.20
5.00	11.55	10.30
5.25	11.65	10.35
5.50	11.70	10.40
5.75	11.75	10.45
6.00	11.80	10.50
6.25	11.80	10.50
6.50	11.85	10.55
6.75	11.85	10.55
7.00–10.00	11.90	10.55

Note: For conversion of fpm to m/s in this table 1 m/s = 200 fpm was taken.

portion of the travelling cables and any compensation device suspended from the car must be taken into consideration.

In the USA the minimum factor of safety of the suspension ropes is based on the actual rope speed corresponding to the rated speed of the car and is slightly greater for passenger elevators than for freight ones of the same speed. In general the American values are lower in comparison with the British value of 12 as they reach the maximum of 11.9 (passenger) or 10.55 (freight) respectively, for speeds of from 7 up to 10 m/s (Table 2.7).

Example 2.1

Make a correct selection of the suspension ropes for a passenger elevator of the following parameters:

Rated load	$Q = 630$ kg
Mass of the car	$K = 737$ kg
Mass of the counterweight	$Z = 1020$ kg
Height of travel	$H = 33.6$ m
Rated speed	$v = 1.6$ m/s
Roping factor	$i = 1$

No compensating ropes are employed.

The factor of safety f is given by the equation (2.1), from which the minimum breaking load N of all ropes can be expressed.

$$n \times N = \frac{F \times f}{i} = (Q + K) \times g_n \times f = (630 + 737) \times 9.81 \times 12$$
$$= 160\ 923\text{N}$$

The effect of the rope weight may be neglected. The minimum number of ropes will be n=4. For the rope selection Tables 2.1, 2.2 and 2.3 may be used, giving the results shown in Table 2.8.

Table 2.8 — Results for Example 1

Rope type	Nominal diameter (mm)	Wire tensile grade (N/mm^2)	Total breaking load (kN)
6×19 (9/9/1)	11	1370 or dual tensile 1570	220 251.6
6×19 (12/6+6F/1)	11	1370 or dual tensile 1570	224.4 256.4
8×19 (9/9/1)	11	1370 or dual tensile 1570	190.8 218

As seen in the survey above any of the suspension ropes quoted there can be used from the safety aspect, but perhaps the ropes of 1370 N/mm^2 tensile grade would be most convenient. With some elevator installations ropes of the nominal diameter of 13 mm are employed in order to decrease the specific pressure in the sheave grooves and extend life of both the grooves and the ropes (see Example 2).

(c) Suspension rope fastenings

The strength of the rope termination must be at least 80% of that of the rope. The ends of the ropes may be fixed to the car, counterweight or suspension points by means of a metal or resin filled sockets, self-tightening wedge-type sockets, rope grips, hand splicing, ferrule secured eyes or another system of equivalent safety. The fixing of the ropes on a drum can be carried out either on the inside by self-tightening wedges or tapered babbitted sockets or by rope clamps, which are usually applied on the outside of the drum.

The ends of suspension ropes fixed to the car, or the car and counterweight dead ends where multiple roping is used, should be provided with shackle rods of a design facilitating individual adjustment of the rope lengths. The most popular methods only of terminating the suspension ropes will be briefly described.

Self-tightening wedge-type socket

The rope passes round a wedge, the taper of which is usually 1:5 on both sides and corresponds to the taper of the socket.

Tapered babbitted socket

The rope socket may be either cast or forged. If it is made in one piece with the shackle rod it can be of forged steel only. The socket of this design is shown in Fig. 2.12. Dimensions of rope sockets are specified

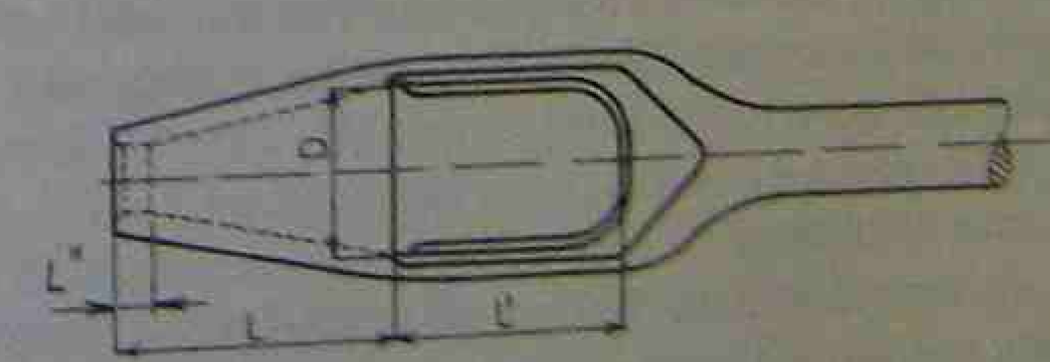


Fig. 2.12 — Tapered babbitted rope socket and shackle rod in one piece.

g) and rated speed v (m/s). The equipment is built to these arameters and normal operation is then guaranteed by the nanufacturer.

In Europe the values of rated loads have been selected from the R10 series of ISO preferred numbers, while the selection of rated speeds has been accomplished on the basis of the R5 series. Rounded-off values are shown in Table 1.1. Numbers in the upper line represent values of the R5 series, while all numbers are valid for the R10 series.

Table 1.1 — Values of R5 and R10 series

R5 series — multiplying factor $\sqrt[5]{10} \sim 1.6$
R10 series — multiplying factor $\sqrt[10]{10} \sim 1.25$

1.0		1.6		2.5		4.0		6.3	10.0
	1.25		2.0		3.15		5.0	8.0	

The mass of one passenger for the purpose of any calculation is 75 kg.

Further technical parameters are:

(a) height of travel (car rise), the number and location of stops,
(b) dimensions of elevator well, car and machine room,
(c) voltage of the main supply, the number of starts per hour and load factor,
(d) control system,
(e) landing and car doors arrangement and the type of operator,
(f) the number of elevators and their location in the building,
(g) environmental conditions.

The main parts of electric elevators are:

(a) *Suspension means* for car and counterweight, which are represented by either steel wire ropes or chains.
(b) *Driving machine*, which is the power unit, consisting of:

— electric motor,
— mechanical gearing,
— brake,
— sheave, drum or chain sprockets

2

Suspension of car and counterweight

2.1 METHODS OF SUSPENSION, EQUALIZING GEAR

Cars and counterweights are suspended to steel wire ropes, roller chains or chains with parallel links (Galle type). Since chain elevators are not frequently used nowadays, attention will be given to the rope suspension.

Elevator ropes are attached to the crosshead of the car frame or pass around multiplying pulleys mounted on the crosshead, if a roping system other than 1:1 is employed. Suspension of each rope must be independent.

The minimum number of suspension ropes are three for traction elevators and two for positive drive ones in the USA, while they are two in Great Britain irrespective of the type of drive.

An automatic device for equalizing the tension of suspension ropes should be provided at least at one of their ends. This device is usually represented by an equalizing gear of individual compression spring type. Such a device, consisting of a wedge-type socket equipped with a steel helical spring is shown in Fig. 2.1, with rubber buffers in Fig. 2.2. While the dependence of the axial force upon the compression is linear with helical springs, it is almost linear with rubber buffers in the range of practical forces (up to the maximum allowable compression), but the curve becomes much steeper for excessive forces (Fig. 2.3).

It is necessary to emphasize that the equalizing device of the compression spring type cannot ensure even distribution of tensile forces in the individual suspension ropes. Its application results in favourable load distribution in comparison with rigid suspension but

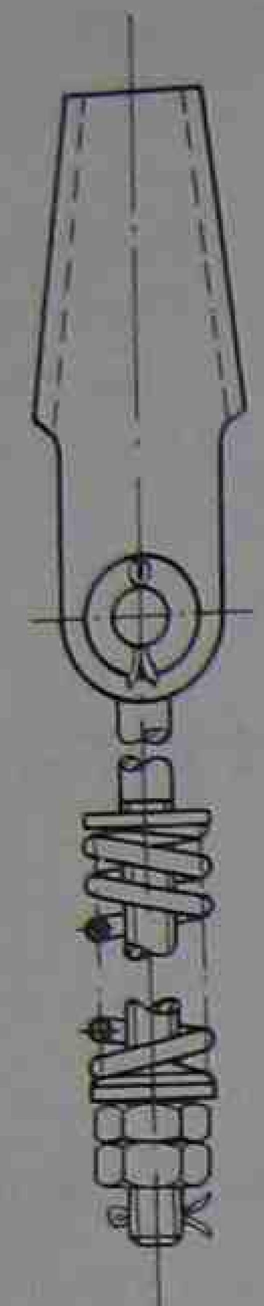


Fig. 2.1 — Suspension device with steel helical spring.

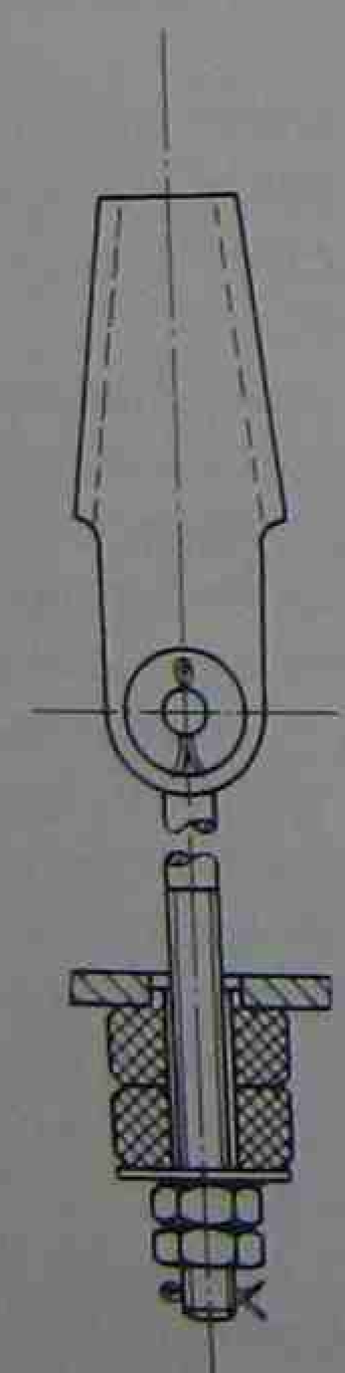


Fig. 2.2 — Suspension device with rubber buffers (Gustav Kocks GmbH).

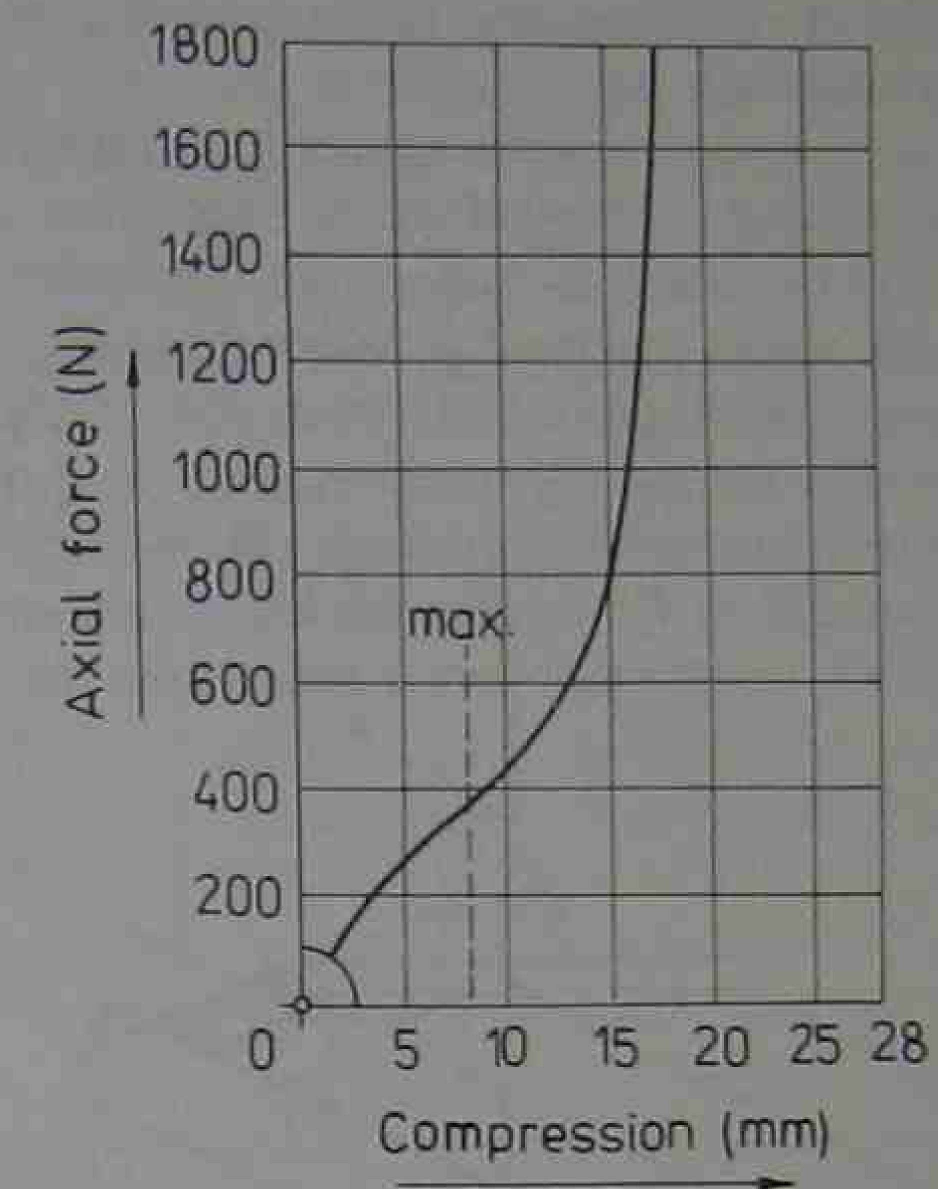


Fig. 2.3 — Axial force to compression diagram for rubber buffers.

in general perfect equalizing cannot be achieved.

The most favourable load distribution on the individual suspension ropes can be accomplished with a balance mechanism, composed of levers and pivots; in Fig. 2.4 schemes for two, three and four ropes are shown. The mechanism is very simple for two ropes; each of them passes over a rope gatherer (not shown in Fig. 2.4) and is attached at one side of the horizontal balance lever, which is free to rotate as one rope stretches more than the other, takes up an inclined position and consequently both ropes are subjected to the same tension. Mechanisms for more suspension ropes are complicated and only exceptionally employed nowadays.

It is of great significance that all suspension ropes be properly installed and anchored at reasonably the same length. It is very difficult to achieve all ropes in equal tension, as many factors affect the load distribution on individual ropes, but improper adjustment at the attachments should be avoided.

There are several methods of checking tension in the ropes. Experienced service men or inspectors should be able to detect differences in tension by feeling the resistance each rope offers to a

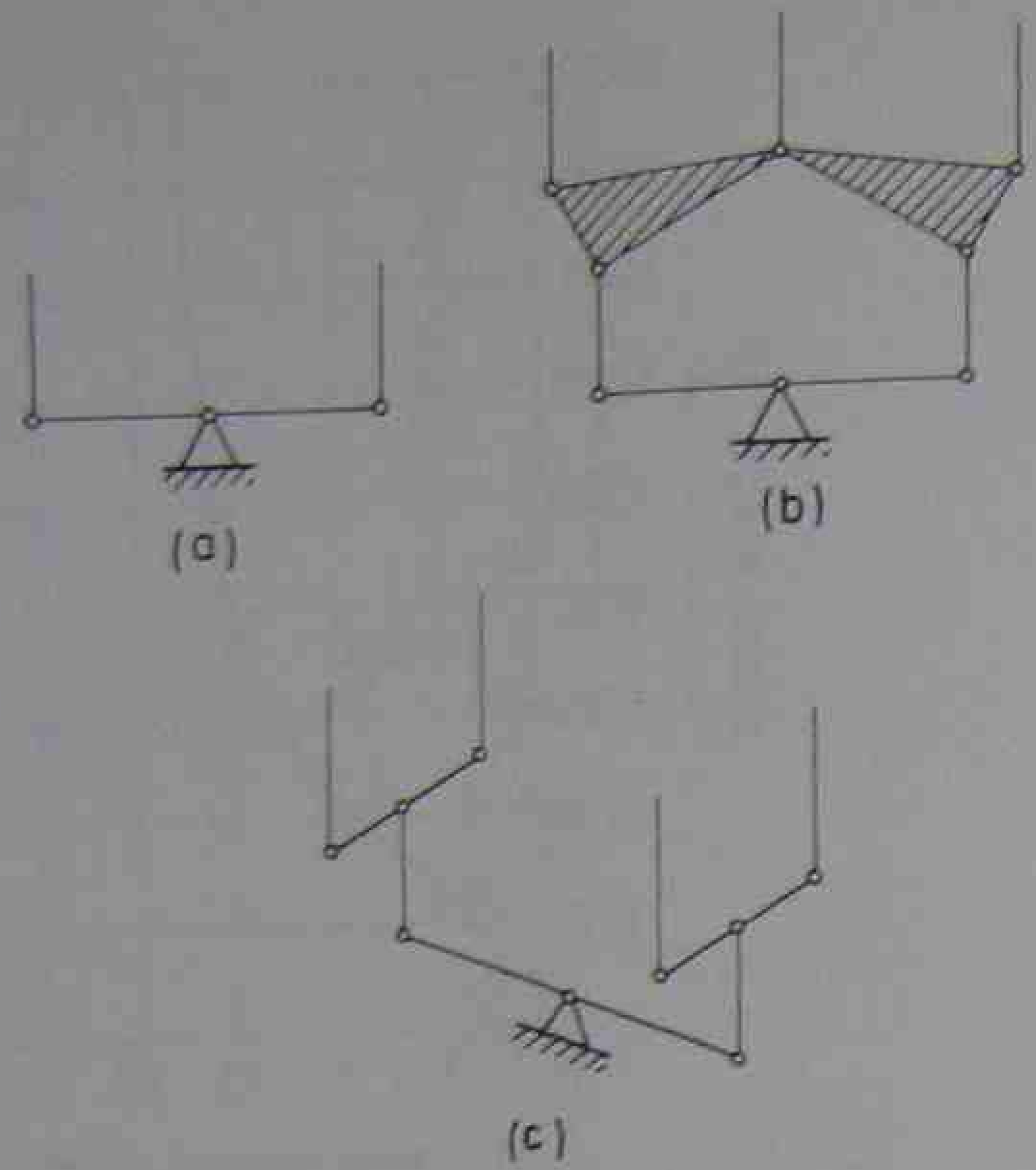


Fig. 2.4 — Balance mechanism for (a) two, (b) three and (c) four suspension ropes.

horizontal pull or by observing the rope vibrations after plucking, however the application of tension meters should be given preference. This device measures rope bending on a fixed length, which is proportional to the rope tension.

2.2 ELEVATOR ROPES*

(a) Specification, construction and recommendations for selection

Elevator ropes are of round stranded construction. The wire material for ropes is manufactured by the open-hearth or electric furnace process or their equivalent.

The tensile grade of wires may be the same throughout the rope — ropes of this kind are known as 'single tensile'. Alternatively, the outer wires may be all of one tensile grade, which is lower than the tensile grade of the inner wires. These ropes are called 'dual tensile'.

The size of the rope is identified by its nominal diameter, which is given by the rope manufacturer. The actual diameter of the rope must be within plus 3% and minus nil of the nominal diameter, when

* IMPORTANT NOTE: At the time of writing both ISO 4344 and BS 302 Standards are emerging. Readers are requested to refer to these standards for specific details, when design projects are being undertaken.

measured under a tension equal approximately to 10% of the minimum breaking load. As a measure of the quality of the rope manufacture the actual diameter measured on a straight portion of the rope under no tension should be within plus 6% and minus nil of the nominal diameter. The actual diameter should be measured carefully at two points located at least one metre apart and at each point two measurements should be taken at right angles. The average value of these four measurements should be within the tolerance mentioned above and the maximum difference between any of the four measurements should not exceed 3% of the nominal diameter. The method of measuring rope diameter is shown in Fig. 2.5.

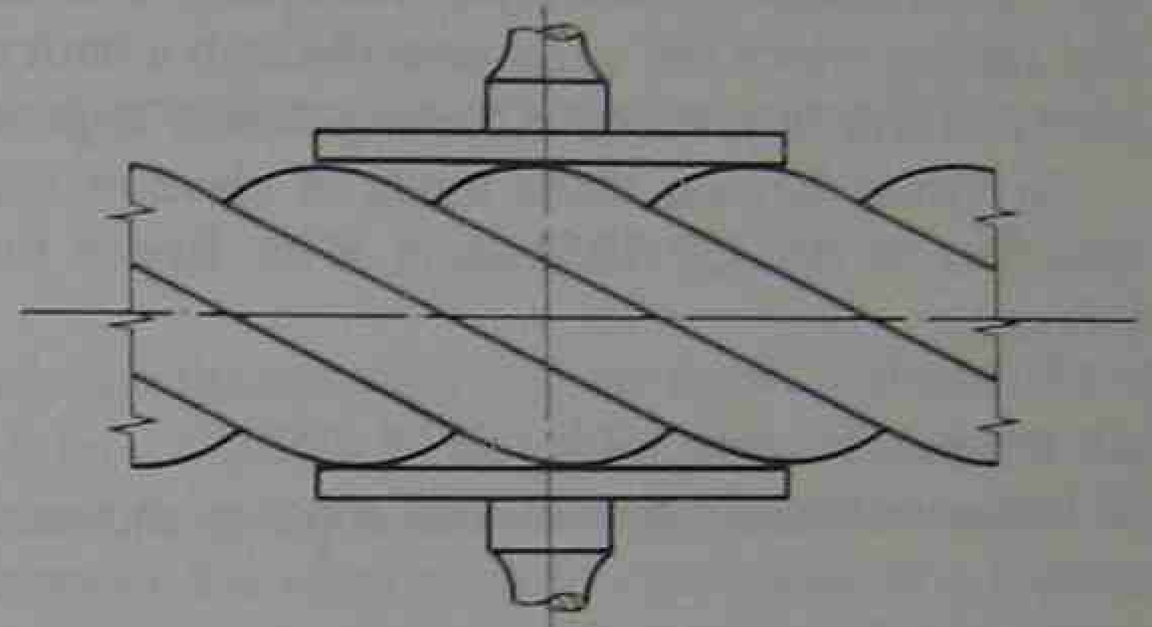


Fig. 2.5 — The method of measuring rope diameter.

A suitable caliper fitted with jaws of sufficient width to cover not less than two adjacent strands must be used.

The nominal diameter of elevator ropes must be at least 8 mm in Great Britain and 3/8 in. (9.5 mm) in the USA. The American Safety Code ANSI/ASME A17.1 specifies the minimum diameter for the outer wires as 0.61 mm.

Elevator ropes are of equal lay construction, in which the wires in the strand are so spun that they all have an equal lay length. The lay length is the distance in a strand or rope, measured parallel to the longitudinal axis, in which the wire in the strand or the strand in the rope makes one complete turn about the axis of the strand or rope respectively. Furthermore, elevator ropes are either of ordinary (regular) lay or Lang's lay. With Lang's lay ropes the direction of lay of the strands is the same as that of the outer wires in the strands, while in the ordinary construction the direction of lay of the outer layer of wires in the strands is opposite to the direction of lay of the strands in the rope. The advantage of Lang's lay ropes is a larger contact surface and consequently a lower unit pressure between the wires of particular strands resulting in a longer life of the rope. On the other hand, ordinary lay ropes can be handled more easily as the tendency to kink and untwist is smaller than with the Lang's lay. Elevator ropes are usually of right-hand lay, which means that the strands are laid up to the right (clockwise direction when looking at the rope end).

For bad environmental conditions, e.g. for outside elevator installations or elevators in chemical plants exposed to extremely corrosive atmosphere, ropes with galvanized wires may be supplied. The wires are provided with zinc protective coating. Two methods of galvanizing may be applied depending on the application for which the rope is intended, namely electrolytic process and hot dip galvanizing, during which the wires pass through a bath of molten zinc. The purity of zinc in protective coating closely approaches 100%.

In Great Britain three kinds of elevator suspension ropes are specified in BS 329:1968 Steel Wire Ropes for Electric Lifts, as follows.

(i) Round strand equal lay 6×19 (9/9/1)

A cross-sectional view of the rope is shown in Fig. 2.6 and all

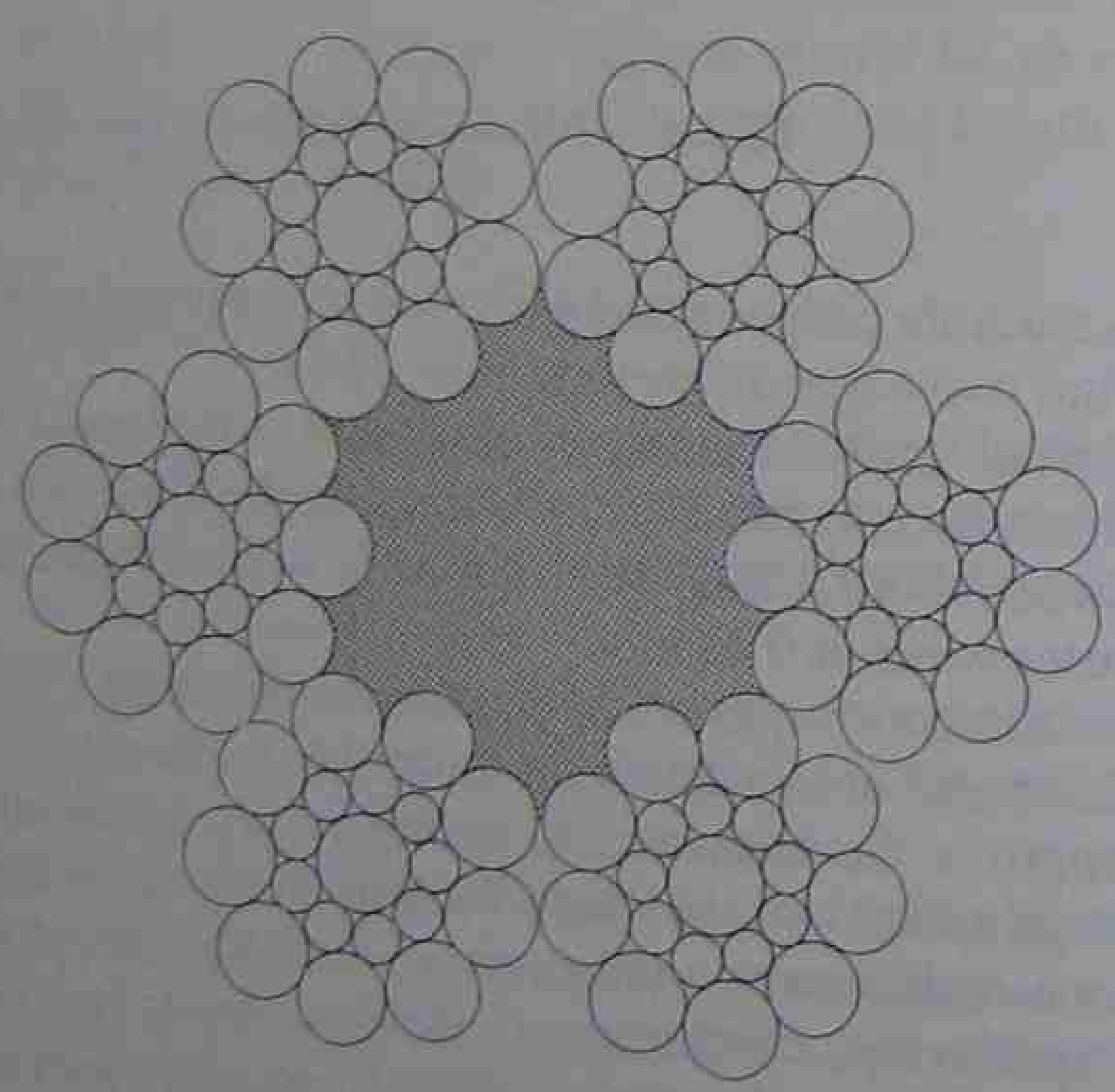


Fig. 2.6 — Round strand equal lay 6×19 (9/9/1) suspension rope.

technical data are quoted in Table 2.1.

The construction 6×19 (9/9/1) means 6 strands, 19 wires in each strand, namely 9 wires in the outer layer, 9 inner wires and 1 central

Table 2.1 — Technical data of 6x19 (9/9/1) elevator rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN) at wire tensile grade (N/mm²) 1370 1170/1770* 1370/1570*	1570
6.5	15.3	19.2	22.0
8	23.2	29.1	33.3
9.5	32.7	41.0	46.9
11	43.8	55.0	62.9
13	61.1	76.8	87.8
16	92.6	117	133

*Dual tensile.

wire (king wire) in the strand. The same number of both the outer and inner wires is typical for the Seale construction. The outer wires are of larger diameter, which results in larger contact area and lower unit pressure between the wires of both layers and as a consequence a considerable improvement in rope life is achieved in comparison with ropes of normal construction used earlier such as 6×19 (12/6/1).

Table 2.2 — Technical data of 6x19 (12/6+6F/1) elevator rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN) at wire tensile grade (N/mm²) 1370 1170/1770* 1370/1570*	1570
8	23.6	29.7	33.9
9.5	33.3	41.9	47.9
11	44.7	56.1	64.1
13	62.4	78.4	89.5
16	94.5	119	135
19	133	168	191
22	179	225	257

*Dual tensile.

(ii) Round strand equal lay 6×19 (12/6+6F/1)
Its cross-section is shown in Fig. 2.7 and technical data are quoted in Table 2.2.

In this rope there are six filler wires of small diameter inserted between the inner and outer wires in each strand filling the interstices between both wire layers. They improve the contact between the layers and facilitate the rope to retain its shape. In calculating the rope strength the filler wires are considered as not bearing any load.

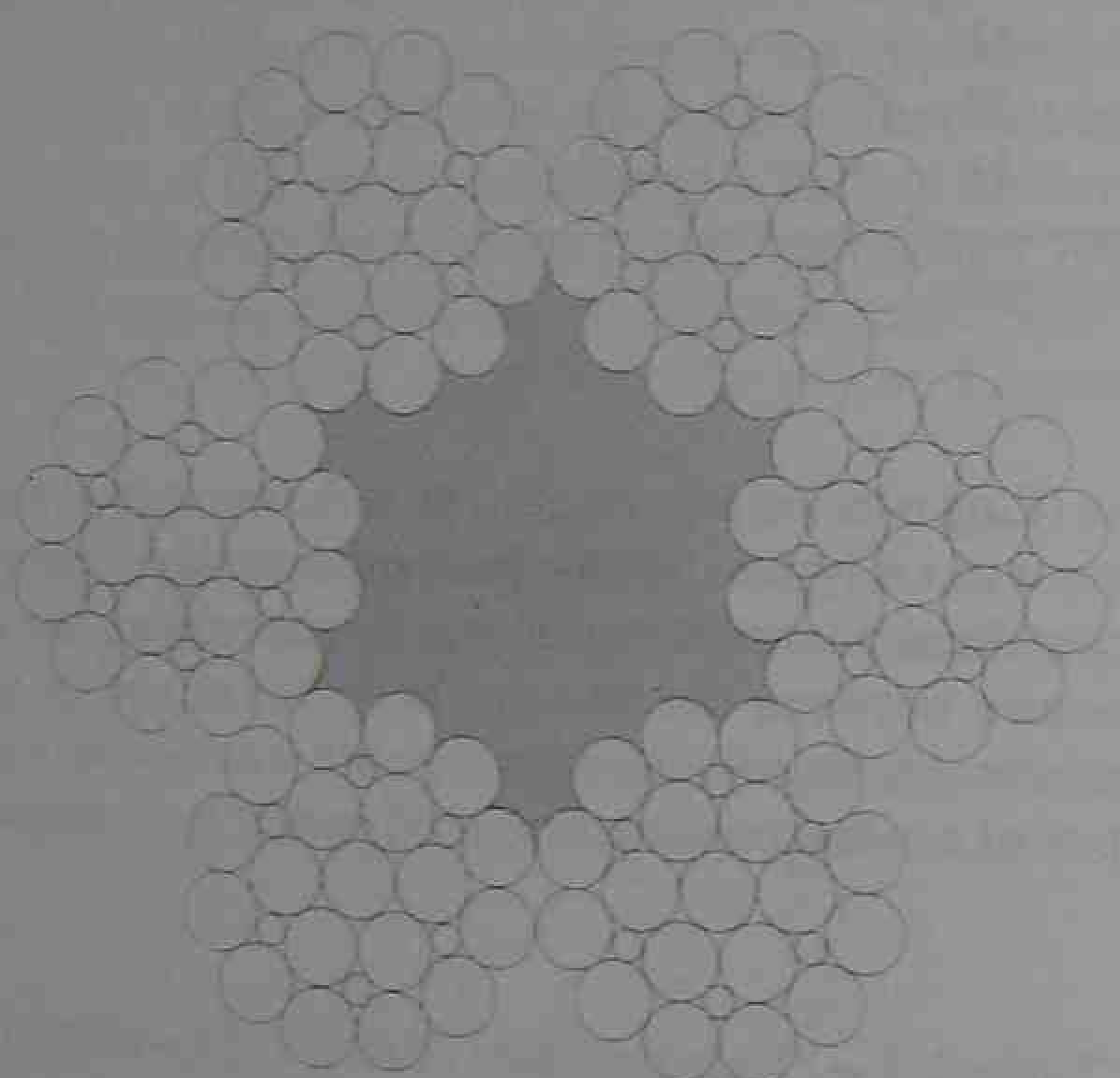


Fig. 2.7 — Round equal lay 6×19 (12/6—6F/1) suspension rope.

(iii) Round strand equal lay 8×19 (9/9/1)
The rope is of the Seale construction, but in contrast to (i) it is composed of eight strands instead of six. It is superior to a six-strand rope in several respects: it is more flexible and fatigue-resistant; it conforms better to the shape of the sheave grooves and is smoother running; the contact area between the wires and the grooves is larger; it sustains a greater number of bendings so that its life may be longer. The rope has less resistance to abrasion, when compared with the equivalent size 6×19 (9/9/1) ropes, because the outer wires are of

a smaller diameter. Also the breaking load is lower than with the 6×19 rope.

In Fig. 2.8 the rope construction is depicted and the technical data

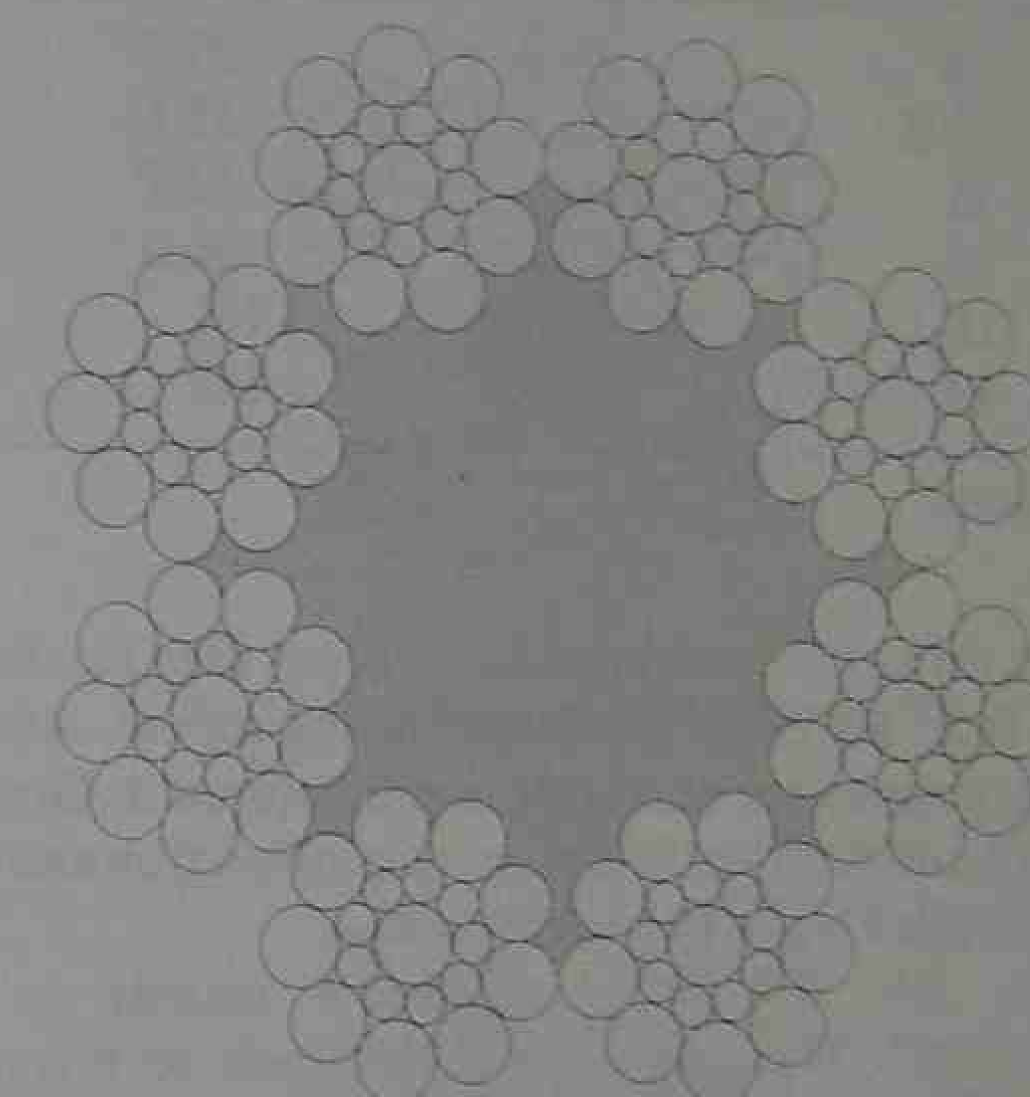


Fig. 2.8 — Round strand equal lay 8×19 (9/9/1) suspension rope.

are reviewed in Table 2.3.

All three kinds of suspension ropes have fibre cores, impregnated with a special lubricant in order to reduce friction between the inner wires and extend their life by preserving them from deterioration from the results of dampness especially when the rope is not in use.

The actual breaking load of the rope must not be less than the minimum value specified in Tables 2.1 to 2.3. It is detected by a tensile test to destruction of a sample of the rope (it should be done for each rope production length) and recorded by the manufacturer. If the actual breaking load is not known the minimum value must be used in calculating the factor of safety.

The weights of ropes given in Tables 2.1 to 2.3 are calculated values only.

Table 2.3 — Technical data of 8x19 (9/9/1) elevator rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN) at wire tensile grade (N/mm^2) 1370 1170/1770* 1370/1570*	1570
8	21.6	25.2	28.8
9.5	30.5	35.6	40.6
11	40.9	47.7	54.5
13	57.2	66.0	76.1
16	86.5	101	116
19	122	142	163

*Dual tensile.

All three kinds of suspension ropes may be also used as ancillary ropes (e.g. governor or compensating ropes). Another two kinds of ancillary ropes are currently manufactured in Great Britain, namely 8×19 (6 and 6/6/1) of Warrington construction and 8×19 (12/6+6F/1) with six filler wires in each strand.

The BS 329:1968 Standard is shortly to be replaced by BS302:1987. The new standard is likely to recommend the use of 1370/1770 N/mm^2 grade dual tensile and 1570 N/mm^2 grade single tensile ropes. The BS 302:1987 and ISO 4344 refer to two main kinds of ropes 6×19 (with 6 strands, 8 to 12 outer wires per strand) and 8×19 (with 8 strands, 8 to 12 outer wires per strand) typical constructions being: 6×19 (9/9/1), 6×19 (12/6+6F/1), 8×19 (9/9/1) and 8×19 (12/6+6F/1).

British Ropes Ltd have introduced the so-called Dyform elevator ropes produced by the drawing of a normal round strand formation through a die, this technique resulting in an increased metallic area in the finished product. The increase of metallic area leads to the reduction of stretch compared with a conventional round strand rope with fibre core and limits the necessity for rope length readjustment. Furthermore, the minimum breaking load is increased while the tensile strength of the wires is held to a level which will give optimum fatigue resistance. Since the surface of the strand is enlarged the specific pressure between the wires and the sheave groove is smaller and consequently the life of the groove is extended. Smooth and quiet operation seem to be further advantages. Dyform ropes of two constructions are manufactured: 6×26 (10/5 and 5/5/1), shown in

Fig. 2.9 and 8×19 (9/9/1), shown in Fig. 2.10, both with fibre cores. The main parameters are quoted in Tables 2.4 and 2.5.

The rope manufacturer Gustav Kocks GmbH in the Federal Republic of Germany has developed an elevator rope especially suitable for high speed and high rise heavy duty installations, designated DRAKO 300T. Fig. 2.11 depicts the cross-section of the rope

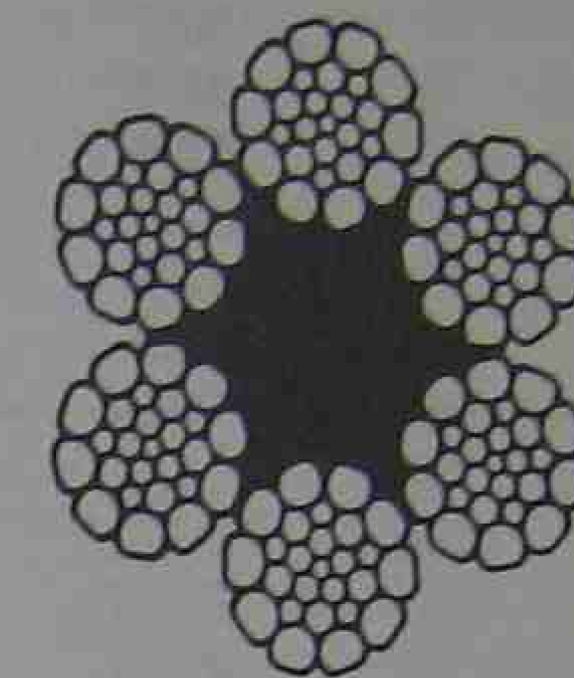


Fig. 2.9 — Dyform rope 6×26 (10/5 and 5/5/1).

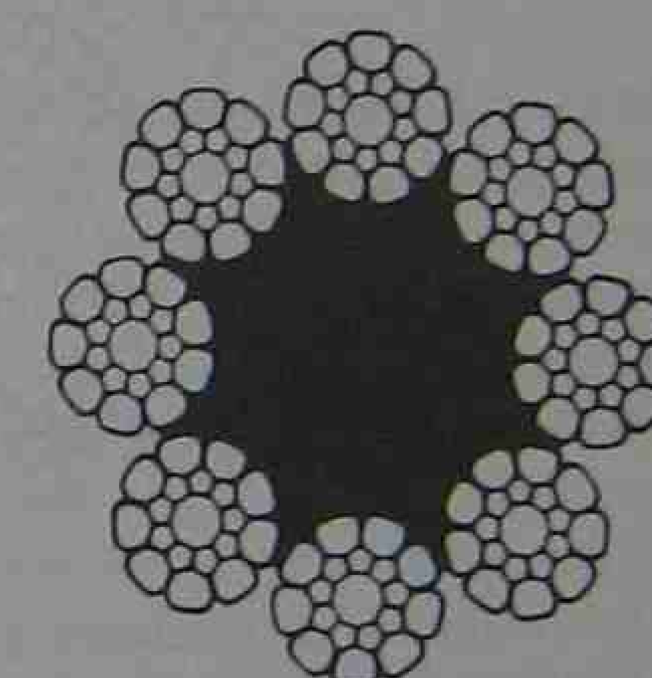


Fig. 2.10 — Dyform rope 8×19 (9/9/1).

Table 2.4 — Technical data of 6x26 (10/5+5/5/1) Dyform rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN)
8	25.6	33.2
9.5	36.1	52.0
11	48.4	70.6
13	67.6	95.1
15.3	81.8	118
16	102	147

Table 2.5 — Technical data of 8x19 (9/9/1) Dyform rope

Nominal diameter (mm)	Weight (kg/100 m)	Minimum breaking load (kN)
8	23.4	31.4
9	29.6	40.8
10	36.5	53.0
11	44.2	61.7
12	52.6	74.6
13	61.7	91.2
15.5	87.7	128
16	93.4	135

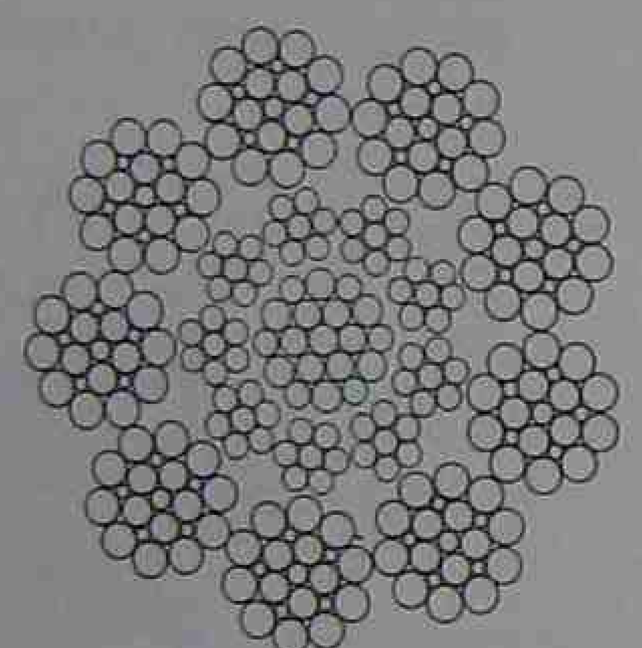


Fig. 2.11 — DRAKO 300T suspension rope.

for the range of nominal diameters 10 to 16 mm, while its technical data are quoted in Table 2.6. The rope is of preformed 9-strand construction of ordinary lay with an integrated wire rope centre. The construction of the strands is 9×16 (10/5+5F/1) with five filler wires in each strand. The integrated rope centre is made of 19 wires and 9 strands of 7 wires each in Lang's lay arrangement. All wires are of the same tensile grade: 1570 N/mm^2. The strands for rope diameters in excess of 16 mm are of the same construction as the British ropes designated in (ii).

The specific load on each wire is reduced with this rope because of

Table 2.6 — Technical data of DRAKO 300T elevator rope

Nominal diameter (mm)	Outer wire diameter (mm)	Metallic area (mm^2)	Weight (kg/100 m)	Breaking load (kN) at wire tensile grade 1570 N/mm^2	
				Calculated	Minimum
10	0.60	46.5	42	73.5	55.1
11	0.65	56.1	50	88	64.6
12	0.70	67.9	59	107	80
13	0.75	79.7	70	125	94
14	0.82	91.1	81	143	107
15	0.90	104.0	93	163	127
16	0.95	120.5	107	189	143
18	0.85	159.6	142	250	188
20	0.95	196.3	175	308	231
22	1.05	237.8	212	373	280

its larger metallic area. The suspension characteristic becomes stiffer because of the special wire core being employed so that both permanent and elastic stretching under different load conditions are reduced to a minimum. The manufacturer states it is less than 50% of the deformation currently occurring with steel ropes with fibre cores. This rope property reduces and in some cases even eliminates the need for automatic relevelling of the elevator car. DRAKO 300T has a smoother and larger surface than comparable ropes of six- or eight-strand construction resulting in an improved contact between the rope and the sheave leading to the decrease of the unit pressure and longer life of both the rope and the sheave grooves.

The difference between the calculated and actual minimum values of breaking load is instructive.

Elevator ropes are preformed as a rule in order to reduce internal stresses created during the current manufacturing process. Preforming is a method of pre-setting the wires in the strands and the strands in the rope, into a permanent final shape which they will take up in the completed rope. Preformed ropes are superior to those made by the usual method of manufacture in several respects:

- the load is evenly balanced on individual strands and wires
- the tendency to kink and untwist is decreased considerably and

the ropes are easier to handle; the rope ends need not be excessively seized to prevent unwinding

- in the event of the breakage of an outer wire the portion which is free does not tend to straighten and stick out of the rope; this prevents the adjacent wires as well as the sheave or pulleys from consequent damage.

For the reduction of rope stretch some manufacturers supply elevator ropes dynamically prestretched, suitable particularly for high rise installations or installations with bottom mounted machines.

(b) Calculation, factor of safety

In calculating elevator rope tension static load is considered only. Consequently the factor of safety must be rather high as it covers the effect of additional stresses such as bending stresses when the rope passes over the sheave and/or pulleys and initial internal stresses in wires due to the method of rope manufacture, the influence of inertial forces in the rope during acceleration and deceleration periods and the effect of uneven distribution of the load because of an imperfect rope equalizer being employed.

In compliance with BS 5655 : Part 1 : 1979 and EN 81 : Part 1, the safety factor of the suspension ropes must be at least

12 in the case of traction drive with three ropes or more
16 in the case of traction drive with two ropes
12 in the case of drum drive

The factor of safety is the ratio between the minimum breaking load of the rope and the maximum static force in this rope and is calculated by the following formula:

$$f = \frac{n \times i \times N}{F} \tag{2.1}$$

where N is the minimum breaking load of one rope (N) (the actual breaking strength, if given by the manufacturer, may be used for calculation); n is the number of suspension ropes; i is the roping factor, i.e. 1 for 1:1 roping, 2 for 2:1 roping, etc. F is the maximum static load imposed on elevator ropes at any position in the well. For the calculation of this force the car is usually supposed to be stationary at the lowest level with its rated load; the effect of the mass of the car, the mass of elevator ropes, the mass of the appropriate

Table 2.7 — Minimum factors of safety for suspension wire ropes (A 17.1)

Rope speed (m/s)	Elevators	
	Passenger	Freight
0.25	7.60	6.65
0.38	7.75	6.85
0.50	7.95	7.00
0.63	8.10	7.15
0.76	8.25	7.30
0.88	8.40	7.45
1.00	8.60	7.65
1.125	8.75	7.75
1.25	8.90	7.90
1.50	9.20	8.20
1.75	9.50	8.45
2.00	9.75	8.70
2.25	10.00	8.90
2.50	10.25	9.15
2.75	10.45	9.30
3.00	10.70	9.50
3.25	10.85	9.65
3.50	11.00	9.80
3.75	11.15	9.90
4.00	11.25	10.00
4.25	11.35	10.10
4.50	11.45	10.15
4.75	11.50	10.20
5.00	11.55	10.30
5.25	11.65	10.35
5.50	11.70	10.40
5.75	11.75	10.45
6.00	11.80	10.50
6.25	11.80	10.50
6.50	11.85	10.55
6.75	11.85	10.55
7.00–10.00	11.90	10.55

Note: For conversion of fpm to m/s in this table 1 m/s = 200 fpm was taken.

portion of the travelling cables and any compensation device suspended from the car must be taken into consideration.

In the USA the minimum factor of safety of the suspension ropes is based on the actual rope speed corresponding to the rated speed of the car and is slightly greater for passenger elevators than for freight ones of the same speed. In general the American values are lower in comparison with the British value of 12 as they reach the maximum of 11.9 (passenger) or 10.55 (freight) respectively, for speeds of from 7 up to 10 m/s (Table 2.7).

Example 2.1

Make a correct selection of the suspension ropes for a passenger elevator of the following parameters:

Rated load	$Q = 630$ kg
Mass of the car	$K = 737$ kg
Mass of the counterweight	$Z = 1020$ kg
Height of travel	$H = 33.6$ m
Rated speed	$v = 1.6$ m/s
Roping factor	$i = 1$

No compensating ropes are employed.

The factor of safety f is given by the equation (2.1), from which the minimum breaking load N of all ropes can be expressed.

$$n \times N = \frac{F \times f}{i} = (Q + K) \times g_n \times f = (630 + 737) \times 9.81 \times 12$$
$$= 160\ 923\text{N}$$

The effect of the rope weight may be neglected. The minimum number of ropes will be n=4. For the rope selection Tables 2.1, 2.2 and 2.3 may be used, giving the results shown in Table 2.8.

Table 2.8 — Results for Example 1

Rope type	Nominal diameter (mm)	Wire tensile grade (N/mm^2)	Total breaking load (kN)
6×19 (9/9/1)	11	1370 or dual tensile 1570	220 251.6
6×19 (12/6+6F/1)	11	1370 or dual tensile 1570	224.4 256.4
8×19 (9/9/1)	11	1370 or dual tensile 1570	190.8 218

As seen in the survey above any of the suspension ropes quoted there can be used from the safety aspect, but perhaps the ropes of 1370 N/mm^2 tensile grade would be most convenient. With some elevator installations ropes of the nominal diameter of 13 mm are employed in order to decrease the specific pressure in the sheave grooves and extend life of both the grooves and the ropes (see Example 2).

(c) Suspension rope fastenings

The strength of the rope termination must be at least 80% of that of the rope. The ends of the ropes may be fixed to the car, counterweight or suspension points by means of a metal or resin filled sockets, self-tightening wedge-type sockets, rope grips, hand splicing, ferrule secured eyes or another system of equivalent safety. The fixing of the ropes on a drum can be carried out either on the inside by self-tightening wedges or tapered babbitted sockets or by rope clamps, which are usually applied on the outside of the drum.

The ends of suspension ropes fixed to the car, or the car and counterweight dead ends where multiple roping is used, should be provided with shackle rods of a design facilitating individual adjustment of the rope lengths. The most popular methods only of terminating the suspension ropes will be briefly described.

Self-tightening wedge-type socket

The rope passes round a wedge, the taper of which is usually 1:5 on both sides and corresponds to the taper of the socket.

Tapered babbitted socket

The rope socket may be either cast or forged. If it is made in one piece with the shackle rod it can be of forged steel only. The socket of this design is shown in Fig. 2.12. Dimensions of rope sockets are specified

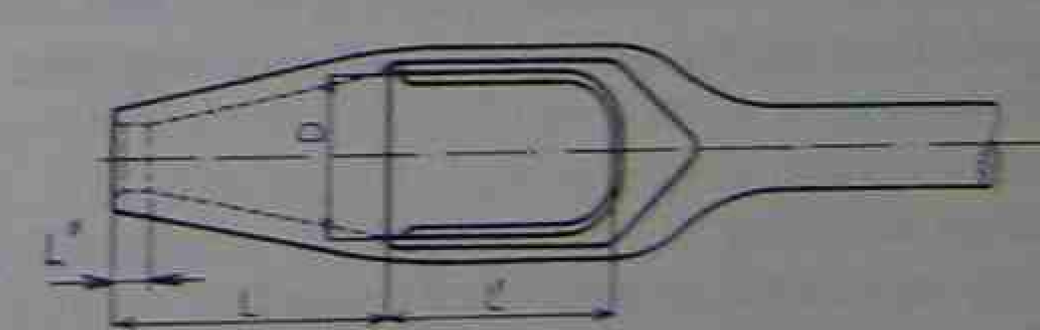


Fig. 2.12 — Tapered babbitted rope socket and shackle rod in one piece.

in the national standard codes. For information principal dimensions given in ANSI/ASME A 17.1 are shown here: L ≧ 4.75 d, L′ ≧ 4 d, 3.2 mm ≦ L″ ≦ 12.7 mm, D = (2.25–3)×d, where d is the nominal rope diameter.

Several methods are currently used for fitting the socket. The principle is as follows: The rope end to be socketed is provided with seizings to prevent the loss of rope lay. The seizings are usually done with annealed iron wire, one of them being close to the cut end of the rope and the second one at a distance equal to the length of the tapered portion of the socket plus the length of the portion of the rope to be turned in. For non-preformed ropes a third seizing should be made at a distance from the rope end equal to the portion of the rope to be turned in. After placing the socket on the rope, the first seizing is removed, the rope end unlaid and the fibre core cut out. The outer surface of all rope strands should be cleaned and made free from grease or dirt with a non-flammable low-toxic solvent. Then the exposed rope strands are bent and turned in, so that the wires form hooks facing inwards. The minimum portion turned in should be of the length equal to 2½ times the nominal rope diameter. When the rope end is positioned definitely in the socket, the bend of the turned-in strands must be slightly overflush with the mouth of the tapered socket at its large end visible after the socket has been babbitted. The thermal procedure may take place afterwards. The socket should be slightly heated to prevent the metal from rapid chilling. The bottom of the socket is sealed on the outside by asbestos yarn or other material of equivalent properties to prevent the babbitt from seeping through and molten white metal is poured in slowly and evenly until the socket is filled to a point level with the top of the opening in the large end. It is necessary that the babbitt is used at the correct temperature and natural cooling takes place after the pouring. The temperature for pouring is between 330 °C and 360 °C and consequently the melting point of the babbitt must be low.

This method of terminating wire rope in elevator installations has proved to be very efficient and reliable and has become popular in spite of its application taking rather a long time.

Resin socketing

This represents an innovated procedure to the preceding method. In this system a thermosetting resin composition, usually based on an unsaturated polyester or an epoxy resin, is used instead of molten white metal as a medium for making a poured termination on elevator rope. The resin composition commonly consists of two or

three components, which being mixed together form a fluid mass that converts to a solid in a few minutes at ambient temperature. Resin socketing does not require special fittings and preparation of the rope end taking place prior to socketing is almost identical with that specified for babbitting.

If correctly carried out solid resin termination becomes functional and is of equivalent level of performance to babbitted rope termination.

Rope grips

Rope grips provide a quick and effective method of fastening rope ends, very popular in some countries. The rope is passed round a thimble forming a return loop and its end is secured by at least three rope grips. The correct method of fixing is shown in Fig. 2.13. The U-

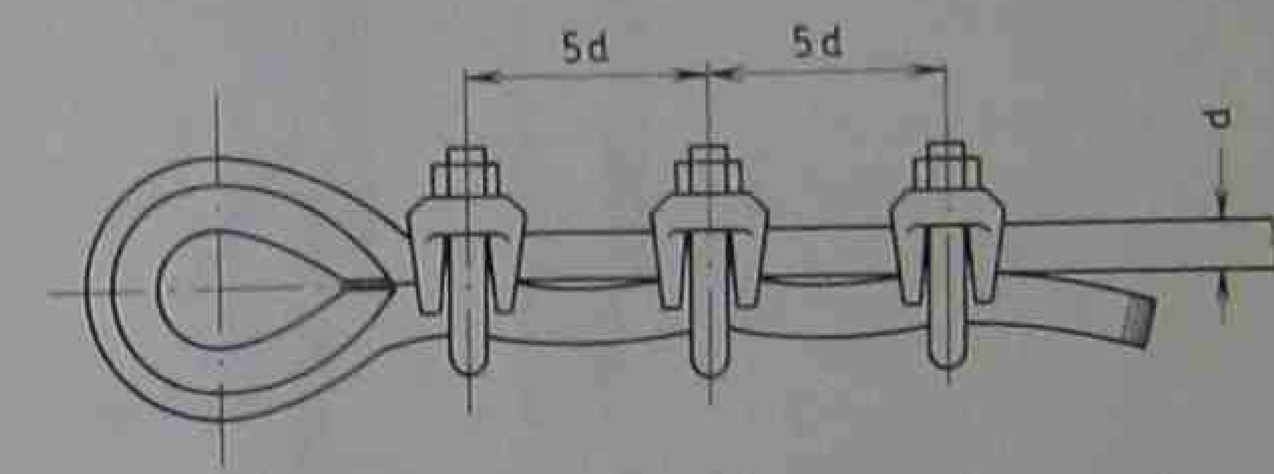


Fig. 2.13 — Method of fixing rope grips.

bolts must always be on the short end of the rope.

Rope splice

The rope forms a return loop round a thimble and its loose end of adequate length is threaded through the loaded rope after removing the fibre core. When the splice has been made it should be bound with stranded wire. This work must be done by means of special tools by an expert to a definite schedule and takes a considerably long time. A five-tuck thimble splice of British Ropes Ltd for a six-stranded rope is shown in Fig. 2.14. The length of the splice is approximately 16 times the nominal diameter of the rope.

Ferrule secured eye

Both the loaded rope and its free end forming a return loop pass through a ferrule made usually of an aluminium alloy. Rope termination is accomplished by pressing the ferrule at predetermined pres-

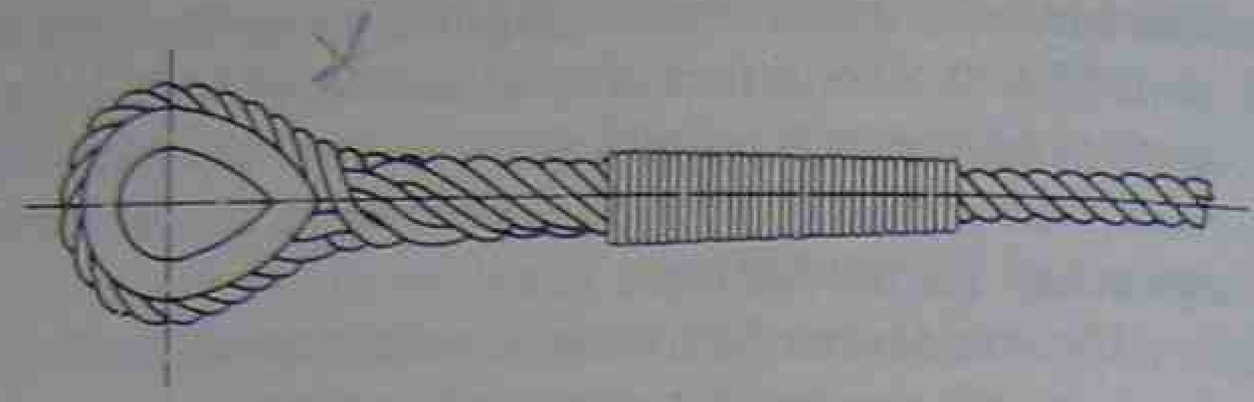


Fig. 2.14 — Rope splice for a six-stranded rope.

sure depending on the composition of the alloy and the size of ferrule (rope). A car suspension gear with this kind of rope termination applied is shown in Fig. 2.15.

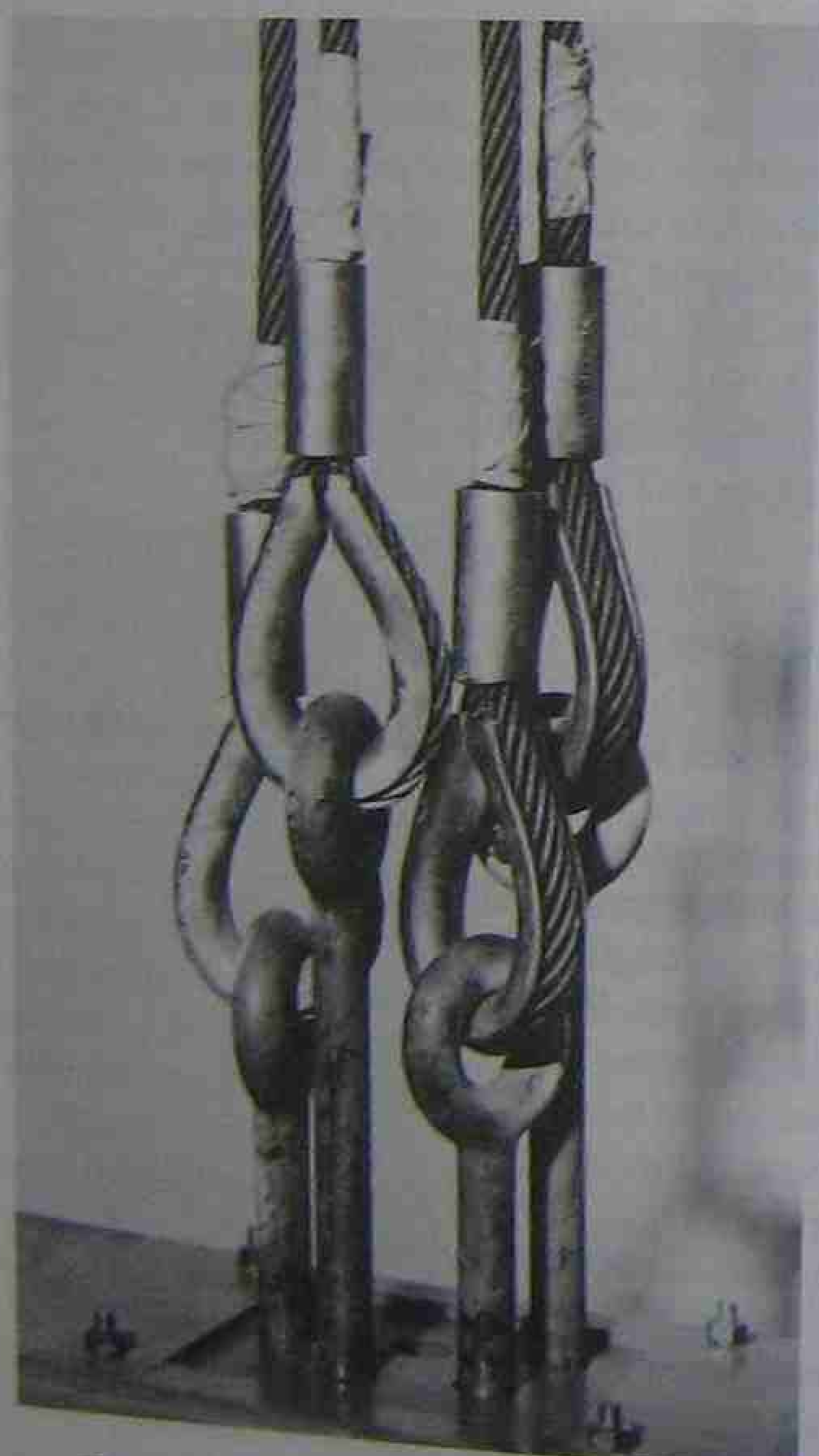


Fig. 2.15 — Rope termination by ferrule secured eye.

(d) Life, maintenance and replacement of elevator ropes

Satisfactory performance of elevator ropes as well as their life depend on a number of factors:

(1) Traction sheave or pulley:
  (a) pitch diameter of the sheave (pulley),
  (b) frequency and direction of bendings,
  (c) shape of the groove,
  (d) material of the sheave and its physical properties.

  With traction sheaves having more than one groove further factors must be considered, namely,
  (e) accuracy with which the grooves are manufactured, particularly the values of their pitch diameters,
  (f) homogeneity of material structure in the area of the sheave grooves.

(2) Elevator rope:
  (a) rope construction,
  (b) material of wires and its physical properties.

(3) Height of travel (car rise).

(4) Rope drag is effected by the angle between the longitudinal axis of the rope and the radial plane of the groove.

(5) Load spectrum:
  (a) absolute magnitudes of tensile forces in elevator ropes,
  (b) the difference of tensile forces on the tight and slack sides of the sheave,
  (c) the difference of tensile forces in particular elevator ropes (at the same point).

(6) Technical parameters of the driving machine and some other elevator components:
  (a) rate of acceleration and deceleration of the car,
  (b) traction capability of the sheave,
  (c) speed of elevator ropes,
  (d) vibration of elevator ropes.

(7) Storage, handling and installation of elevator ropes.

(8) Environment and maintenance.

Some of these factors are mentioned in other sections, so only some of the factors will be commented on here.

A correct determination of the pitch diameter of the traction sheave and pulleys is of great significance and it may be beneficial to increase the diameter above the minimum values specified in safety codes especially with roping systems of higher frequency of bending.

The frequency of bending of any individual portion of the rope passing around the traction sheave is governed by two factors. First, by the frequency of stops of the car at the main floor (indirectly proportional to the round trip time), secondly it is affected by the maximum number of pulleys round which the rope passes while the car is ascending or descending. The effect of the frequency of bending is unfavourable in the matter of rope life; in the event of multiple roping systems and/or double wrap drive being applied it is advisable to increase the ratio sheave diameter to nominal rope diameter above the specified minimum values in order to compensate this effect, particularly when reverse bends occur (roping factors other than 1).

In most lift installations ropes must be lubricated during service to minimize wear on the wires, wear on the sheave or drum groove, oxidation of the wires and atmospheric corrosion. The lubricant also reduces fatigue because it allows correct movement of the wires with respect to each other. The fibre core should never be allowed to reach a state of dryness as an abrasive wearing action between the strands and the core would commence and if allowed to continue for a considerable time it may deteriorate the core to the point where it would no longer support the strands.

For traction elevators rope lubricants are compounded to give antislip properties in addition to lubrication. Elevator ropes should be sufficiently lubricated at all times. The frequency of lubricating depends on many factors including atmospheric and temperature conditions, rope speed, type of elevator service etc. and only inspection can determine this frequency. A suitable method of lubricating should be applied to avoid too much lubricant being placed on the rope exterior. Excessive lubrication may result in insufficient traction and the lubricant might be thrown off the rope when operating at high speed.

The safety of elevator operation requires the ropes to be exchanged, dependent on their deterioration. The main criterion for the indication of the condition of the rope is the number of visible wire breaks on a predetermined rope length. In Table 2.9 the critical number of wire breaks is reviewed for ropes of different construction (by Gustav Kocks).

Table 2.9 — Critical number of wire breaks

Rope construction	Number of visible wire breaks on the length of $6\times d$*	$30\times d$*
6×19	10	19
8×19	13	26
DRAKO 300T	14	29

* d is the nominal diameter of the rope.

The rope must be removed immediately when wire breaks accumulate in one spot, conspicuous contraction (reduction in diameter) or other irregularity of rope shape occurs as a result of the breakage of a strand, excessive rope stretch or disproportionate corrosion is detected during the examination of the rope.

Ropes should be stored in a cool, dry building, where the temperature is kept reasonably constant to prevent condensation. They should not be in contact with the floor nor stored where they are liable to be affected by acid fumes or other corrosive factors.

(e) Compensating cables

With high-rise elevator installations compensating cables of rope or chain type are used to minimize the out-of-balance rope tension on the driving sheave due to the weight of suspension ropes and travelling cables and thus make the load on the sheave and motor constant irrespective of the car position. Unless the effect of the variable weight of suspension ropes is compensated the traction force may become either deficient or excessive and a dangerous situation may be created. Furthermore, the required torque on the elevator motor is reduced with the application of compensating cables. The determination of the size of compensating cables is given in Section 6.

Compensation is usually necessary only for travels over 30 m. For rated speeds in excess of 2.5 m/s only ropes can be used for compensation. They are attached to the bottom of the car, pass down and round tensioning pulleys located in the pit and upwards to be secured to the bottom of the counterweight. (Roping systems are covered in Section 3.1(a).) For rated speeds exceeding 3.5 m/s the tensioning pulley should be equipped with an anti-rebound device. When the anti-rebound device is set in operation an electric safety device must stop the elevator machine.

For speeds under 2.5 m/s chain type compensating cables can be used. They are cheaper and the system is simpler as no tensioning device is required. The first compensating cables of this type were represented by a link chain, which proved to be rather noisy particularly with higher speeds. Therefore it was convenient to employ it for speeds up to 1 m/s only. Furthermore, rubbing one link against the other caused abrasion. Link chains hanging free in the elevator well are liable to converge on a single link forming a point at the bottom of the loop and consequently one half of the chain may rub against another during car movement increasing the noise level and abrasion.

Later on some chain type compensating cables used a sash cord (a rope woven in the links of the chain) or the chain was captured in a plastic coating. These methods resulted in a reduction of both noise and abrasion.

Several years ago Whisperflex compensating cable was introduced by Sincor Corporation, USA. It is mostly of circular cross-section (Fig. 2.16) and composed of three items.

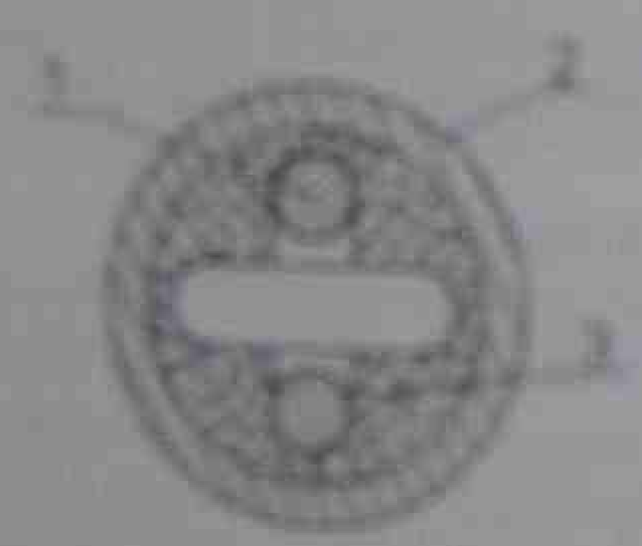


Fig. 2.16 — Cross section of Whisperflex compensating cable (Sincor Corp.). 1. Strength member; 2. Sheath; 3. Core.

- A strength member usually represented by a chain.
- A sheath of black polyvinyl chloride which provides a smooth abrasion resistant surface.
- A mixture of metal particles and plastic, which forms a continuous core.

The metal particles can be ferrous or non-ferrous of any size and shape, preferably between 0.5 and 1.0 mm in diameter. They occupy 30 to 70% of the volume, the rest is occupied by plastic (polyvinyl chloride) elements.

Whisperflex cable may also be manufactured as a flat compensating cable comprising:

(a) Multiple strength members with their respective axes arranged in parallel to one another.
(b) A sheath of flexible polyvinyl chloride.
(c) Metallic particles and plastic mixture.

The application of metal particles results in the increase of the weight per unit linear length so that for a given weight per linear length Whisperflex cable is smaller in diameter in comparison with earlier chain type compensating cables.

The main feature of Whisperflex cable is the minimizing of noise and abrasion. Furthermore, it forms a free hanging loop in the elevator well and has less lateral cable sway, so that it is very unlikely to strike the sidewalls of the well. It has a wave dampening characteristic similar to that of elevator travelling cable.

In Table 2.10 the main technical data of all sizes of Whisperflex

Table 2.10 — Technical data of Whisperflex compensating cable

Cable weight (kg/m)	Chain trade size (mm)	Cable diameter (mm)	Maximum hanging length (m)	Nominal loop (mm)
1.49	3.2	25.1	114	457
2.24	4.8	29.7	152	508
2.98	6.4	32.8	183	533
3.73	7.9	36.1	232	558
4.47	7.9	40.4	194	609
5.22	9.5	41.7	232	609
5.96	9.5	45.5	203	609

compensating cable, manufactured at present, are specified. Whisperflex cable is basically used for speeds up to 1.75 m/s, but future developments should facilitate its application with high-rise, high speed installations as a substitute for compensating ropes and tensioning pulley arrangement, for rated speeds up to 3.5 m/s.

3

Types of Drive

3.1 TRACTION DRIVE

With traction drive the driving sheave is employed as the means for power transmission to elevator ropes. The tractive force is initiated by friction between the ropes and the sheave grooves.

(a) Roping systems

There are a number of different roping systems in existence the application of which depends upon the local conditions, particularly upon the location of the machine, the rated load and rated speed of the car. Great attention should be given to the selection of the roping system so as to achieve long life of elevator ropes, high efficiency of the system and reasonable power consumption. For this purpose the number of pulleys should be decreased to a minimum and if possible, reverse bends of elevator ropes should be avoided.

The machine is usually situated above the well as the overhead position of the machine facilitates the application of the simplest roping schemes and the load exerted upon the building structure is relatively low. With some installations the machine is located in a basement position near the bottom of the well. The initial cost is usually higher in this case and the load imposed on the overhead pulleys at the top of the well and thus on the supporting building structure is also considerably high. For these reasons the basement location of the machine should be avoided whenever possible.

Location of the driving machine in an intermediate position occurs only exceptionally nowadays. Years ago it was used with chain elevators of low rise, but these have been gradually replaced by more

efficient hydraulic elevators.

Schematic diagrams of principal roping systems are shown in Figs. 3.2 to 3.10. In all diagrams the traction sheave is indicated as shown in Fig. 3.1.

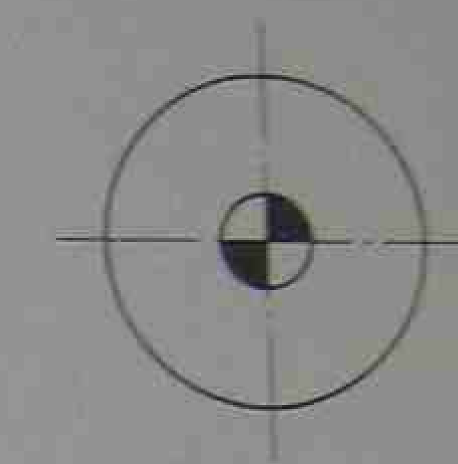


Fig. 3.1 — Designation of traction sheave.

(1) The machine in overhead position

Single wrap drive, roping factor $i=1$	(Fig. 3.2).
Double wrap drive, roping factor $i=1$	(Fig. 3.3).
Single wrap drive, roping factor $i=2$	(Fig. 3.4).
Single wrap drive, roping factor $i=4$	(Fig. 3.5).

(2) The machine in basement position

Single wrap drive, roping factor $i=1$	(Fig. 3.6).
Double wrap drive, roping factor $i=1$	(Fig. 3.7).
Single wrap drive, roping factor $i=2$	(Fig. 3.8).

The arrangement shown in Fig. 3.2 is the simplest possible. A diverting pulley may be provided to deflect the ropes in order to increase the distance from the car hitch to the counterweight, if necessary.

In order to obtain sufficient traction a double wrap drive may be used — see also section 3.1(c). In Fig. 3.3 elevator ropes run from the car over the traction sheave down around the secondary (idler) sheave, then to the driving sheave again, and to the counterweight. Where the diameter of the traction sheave equals the distance between the centrelines of the car and counterweight, the secondary sheave may be located directly under it. Where the distance in question is greater, the secondary sheave serves as a diverting pulley too (Fig. 3.3).

With systems, where the roping factor i is not equal to 1, both ends

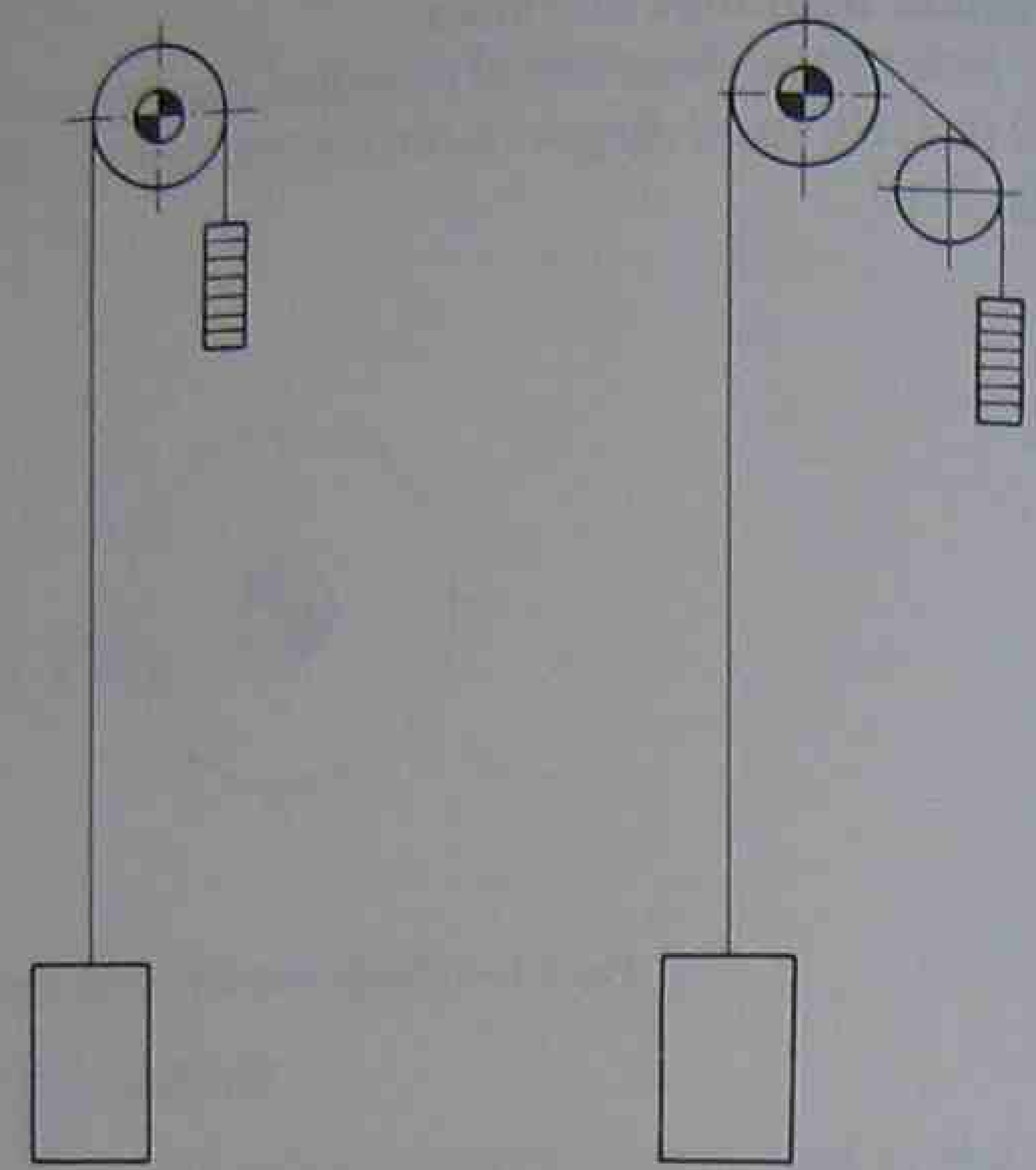


Fig. 3.2 — Roping system for overhead position of the machine, single wrap drive, roping factor 1.

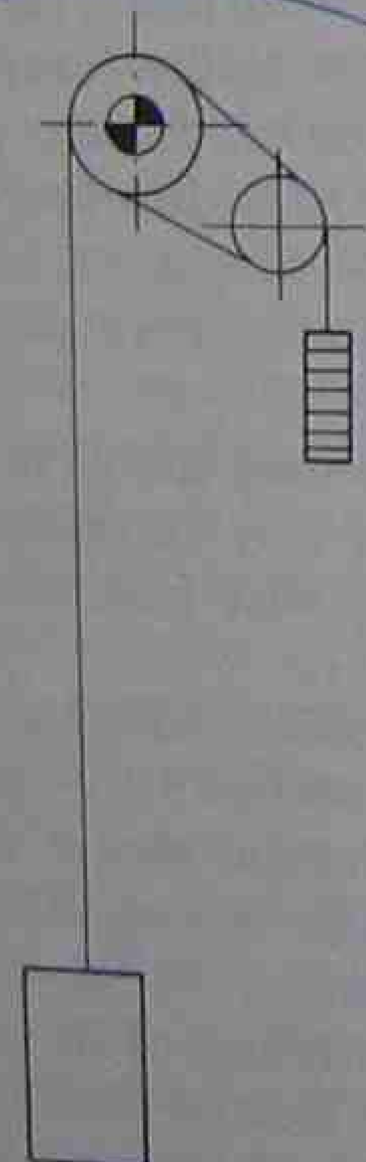


Fig. 3.3 — Roping system for overhead position of the machine, double wrap drive, roping factor 1.

which system used.

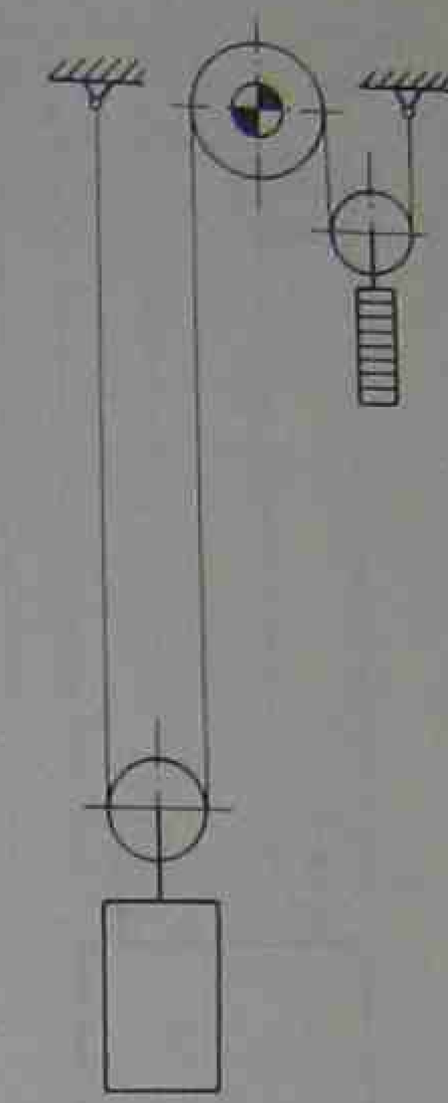


Fig. 3.4 — Roping system for overhead position of the machine, single wrap drive, roping factor 2.

of the elevator ropes are anchored to overhead beams, whilst multiplying pulleys are fixed to the car and counterweight. The theoretical tensile force in elevator ropes is *i* times less than with roping factor 1 and the peripheral velocity of the driving sheave is *i* times greater.

Roping systems with compensating ropes are shown in Fig. 3.9 (roping factor 1) and Fig. 3.10 (roping factor 2). The machine is located in overhead position and single wrap drive is applied.

(b) Traction sheave and diverting pulley

The ratio between the pitch diameter of the sheave or pulley and the nominal diameter of the suspension ropes should be at least 40, regardless of the number of strands, in conformity with both BS 5655 and ANSI/ASME A 17.1. The mimium ratio between the pitch diameter of the tensioning pulleys and the nominal diameter of the compensating ropes is specified equal to 30 in BS 5655, but 32 in A 17.1.

Traction sheaves are provided with finished grooves for suspension ropes. Grooves of two principle shapes are currently employed:

(i) Vee-groove, having an angle of 32 to 40° (Fig. 3.11).

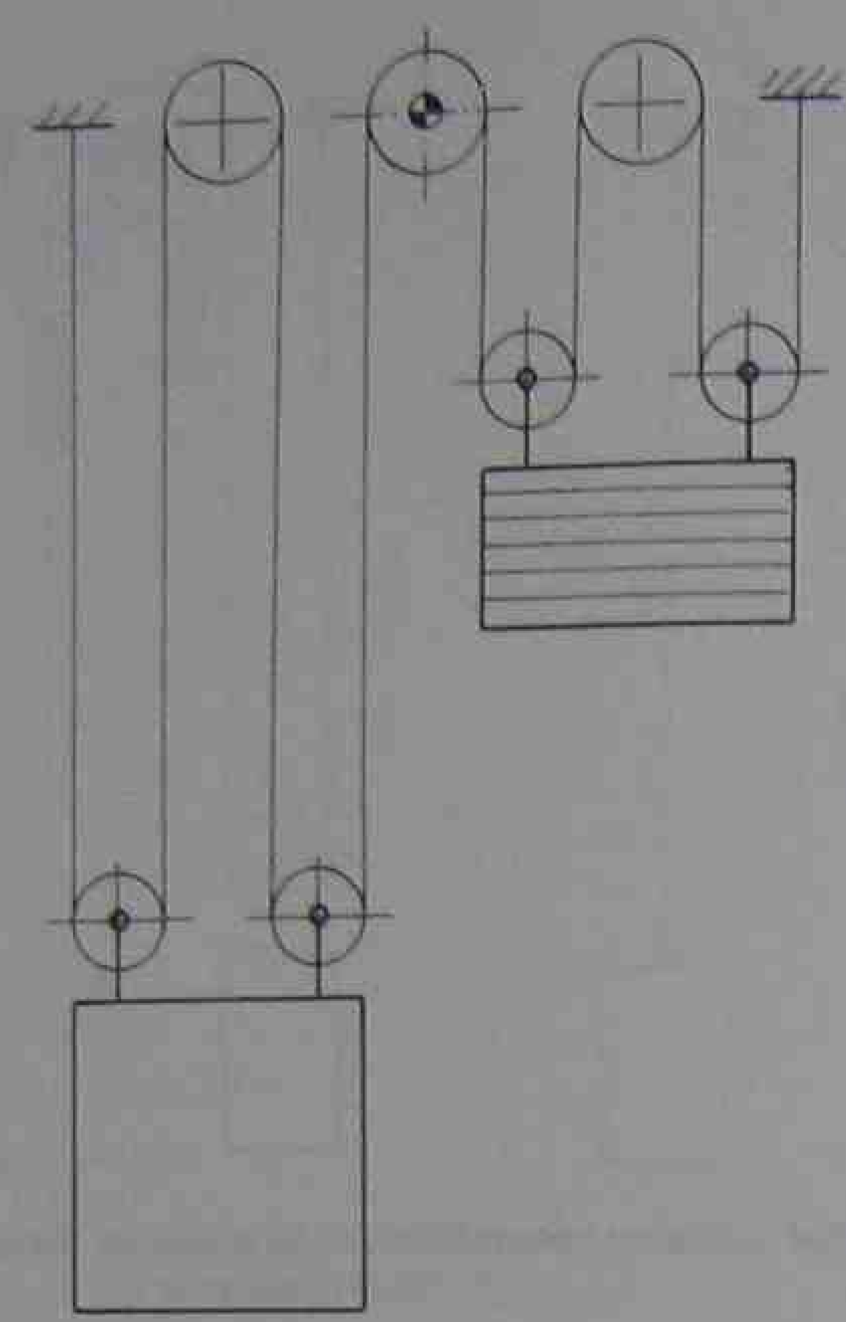


Fig. 3.5 — Roping system for overhead position of the machine, single wrap drive, roping factor 4.

The traction increases with decreasing angle of groove, but so does the specific pressure and resultant wear of both the grooves and suspension ropes.

(ii) U-groove, either round (semicircular) or undercut (Figs. 3.12 and 3.13).

With the round groove the traction is much lower so that the double wrap drive is often used particularly with high-speed elevators. However, this type of groove has great advantages in longer rope life because of lower specific pressure in the contact area between the rope and the groove and of a lower degree of noise, which is especially noticeable at high speeds.

The properties of an undercut groove are intermediate between a vee-groove and a round groove. The width of the cutting is given by the formula

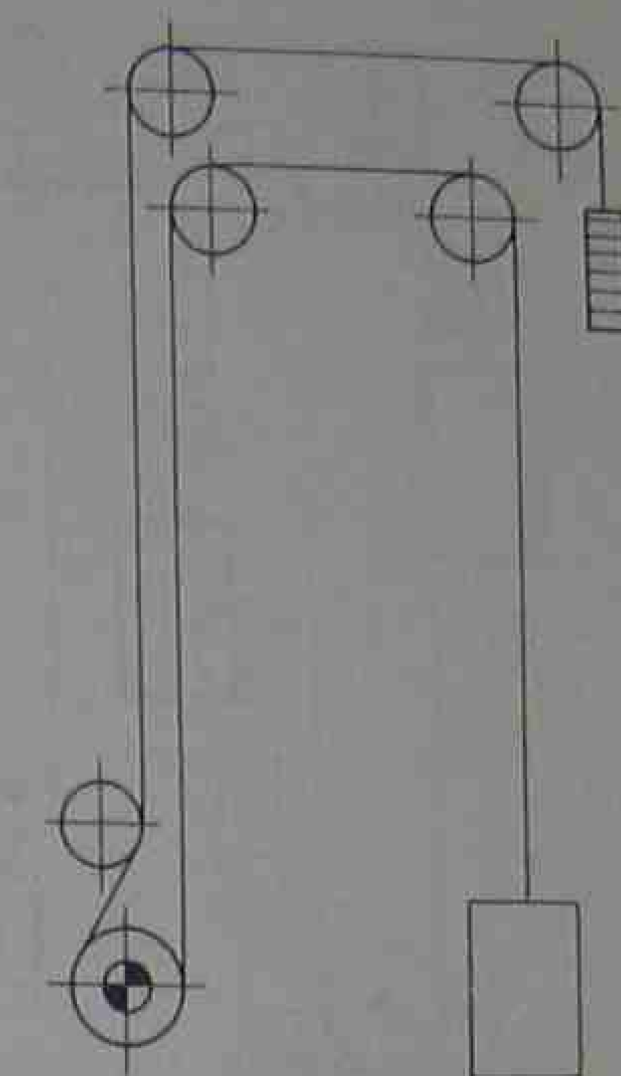


Fig. 3.6 — Roping system for basement position of the machine, single wrap drive, roping factor 1.

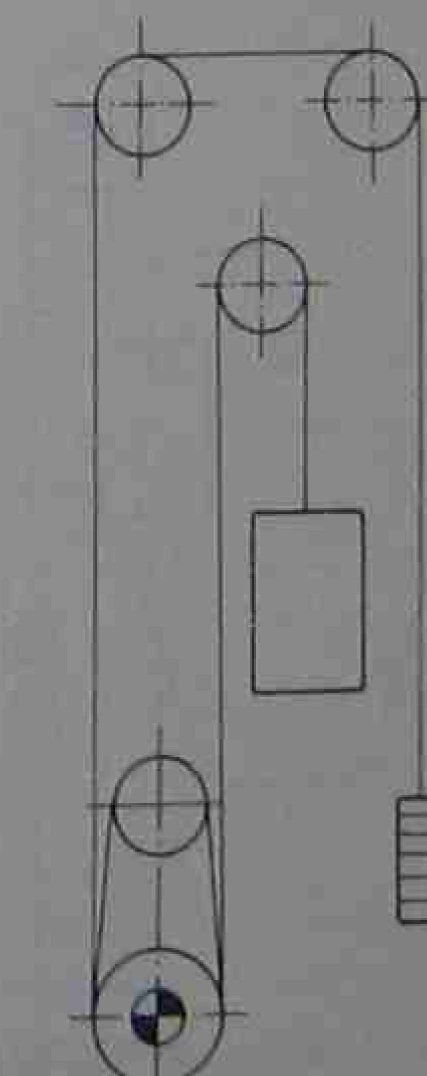


Fig. 3.7 — Roping system for basement position of the machine, double wrap drive, roping factor 1.

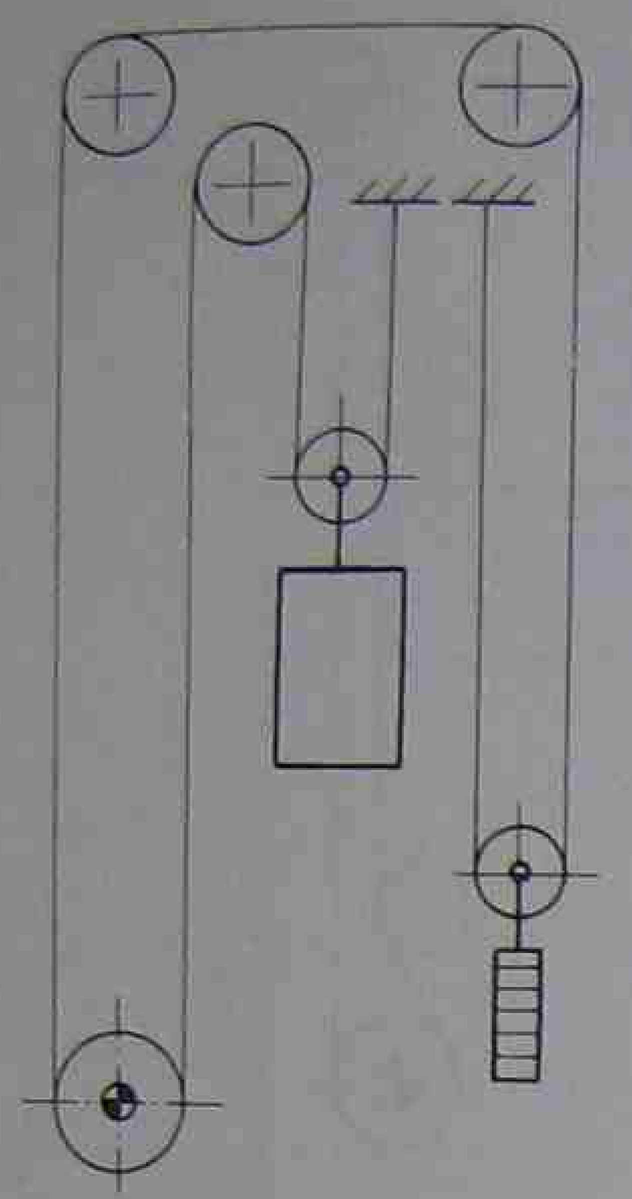


Fig. 3.8 — Roping system for basement position of the machine, single wrap drive, roping factor 2.

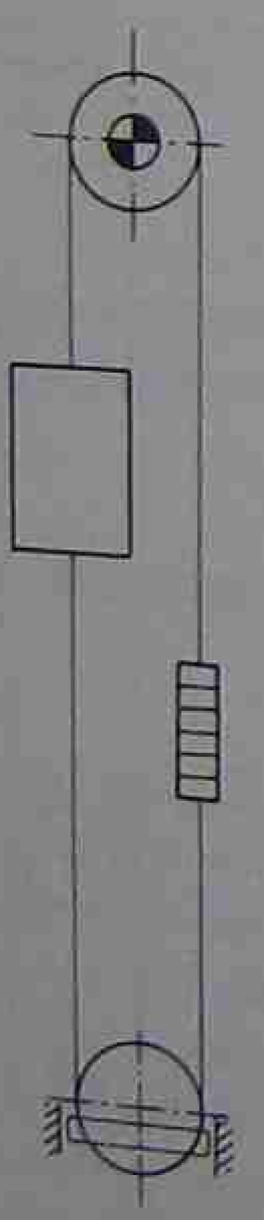


Fig. 3.9 — Roping system with compensating ropes, roping factor 1.

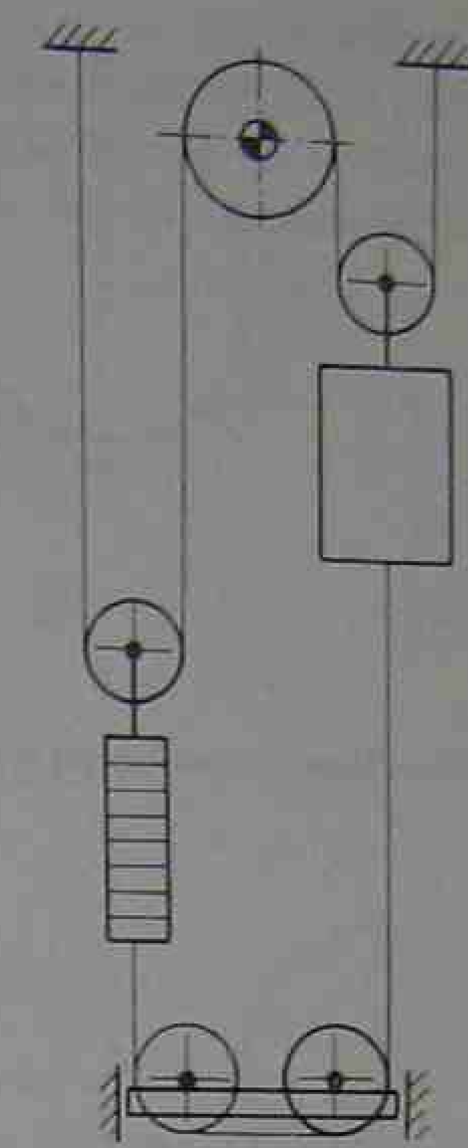


Fig. 3.10 — Roping system with compensating ropes, roping factor 2.

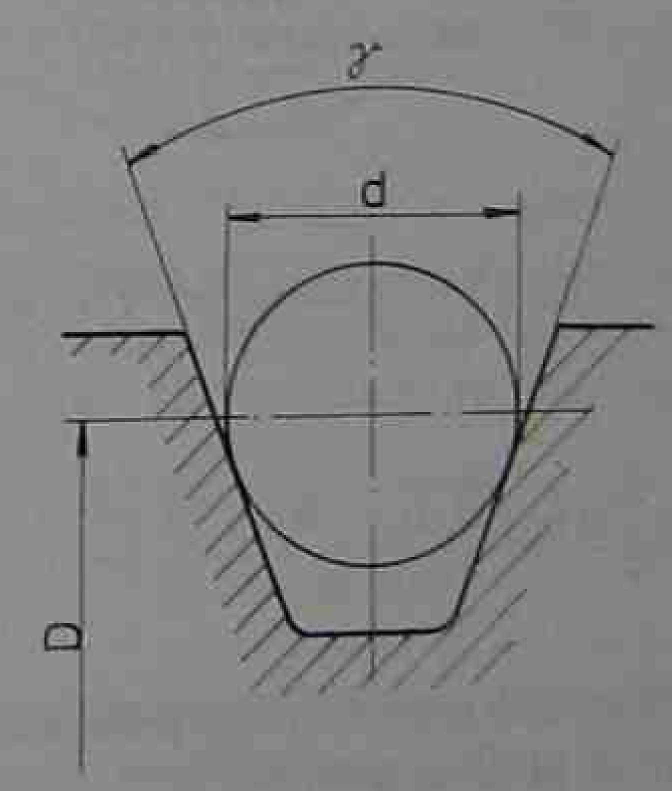


Fig. 3.11 — Vee groove of traction sheave.

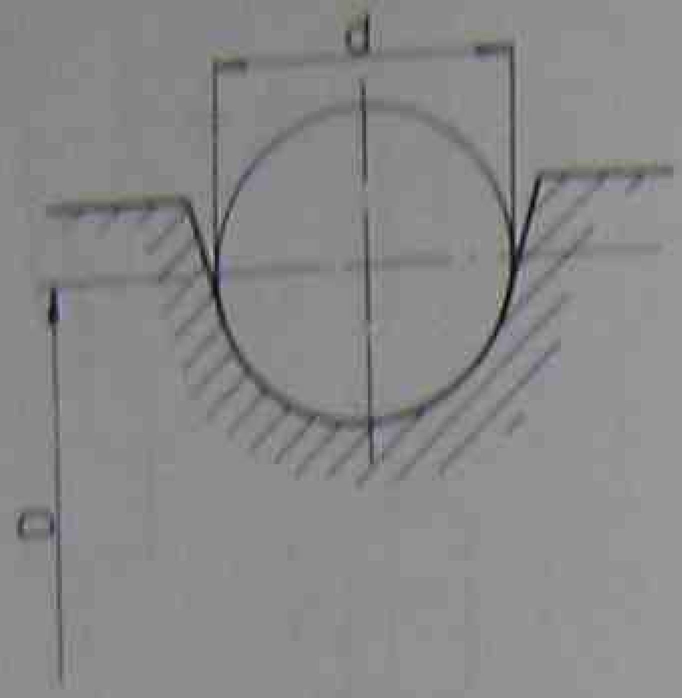


Fig. 3.12 — Round (semicircular) groove of traction sheave.

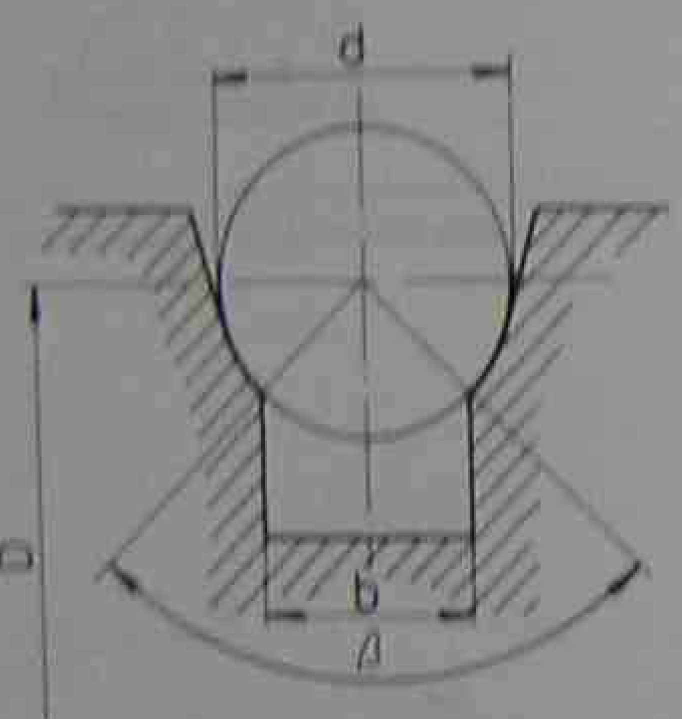


Fig. 3.13 — Undercut groove of traction sheave.

$$\frac{b}{2} = \frac{d}{2} \times \sin \frac{\beta}{2} \qquad (3.1)$$

The angle of the undercutting β should be preferably under 90° and must not be greater than 105°.

Tests and measurements of tensile forces in elevator ropes taken several years ago revealed the load distribution on elevator ropes to be favourable with U-grooved sheaves. From this aspect undercut grooves should replace vee grooves wherever the traction proves to be sufficient.

The diverting pulley is located mostly in the machine room, but if a long span between the centrelines of the car and counterweight is required it is mounted at the top of the well in order to reduce the loss of angle of wrap. In Fig. 3.14 the geometry of the sheave and

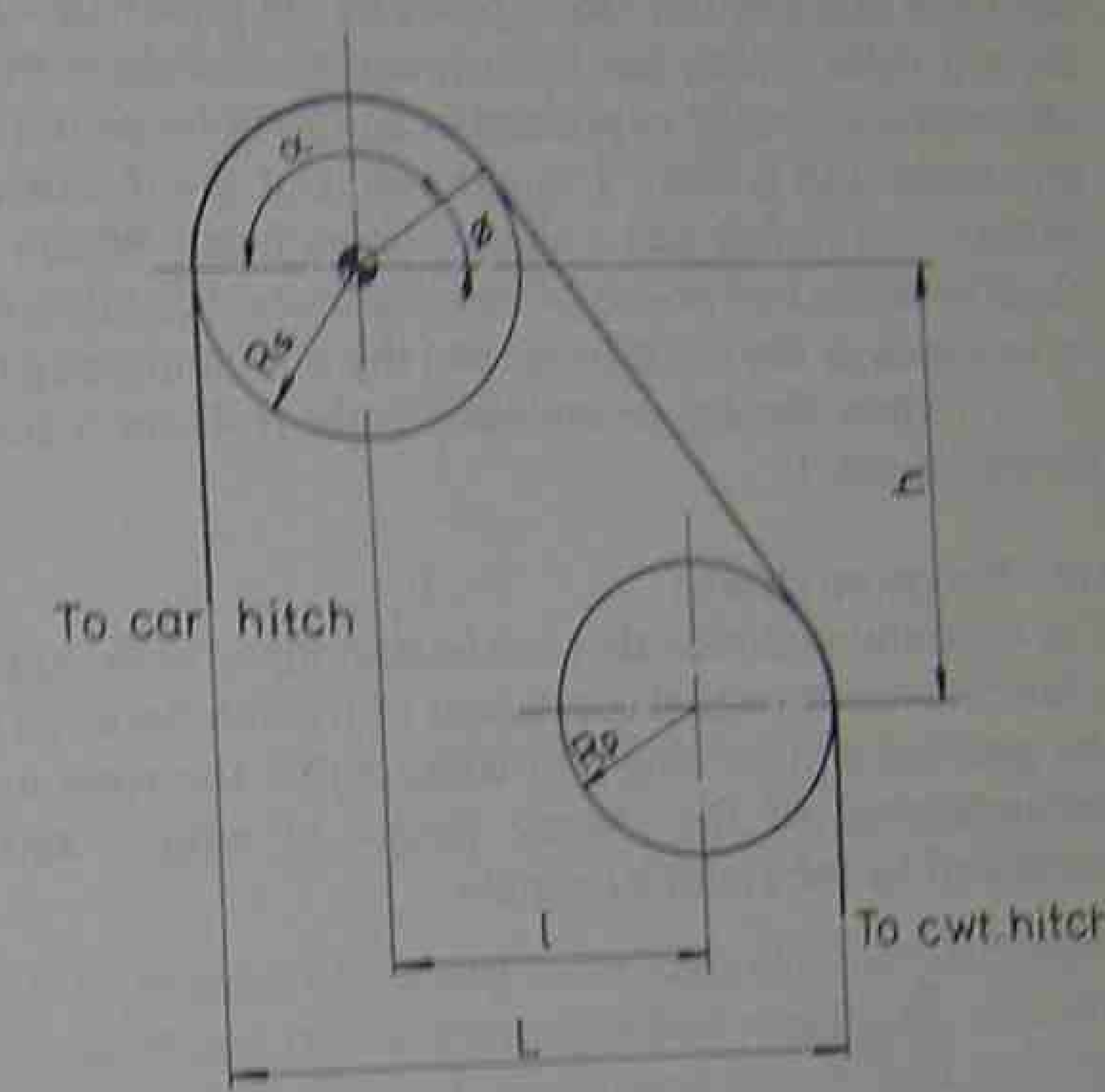


Fig. 3.14 — Geometry of traction sheave and diverting pulley.

diverting pulley is shown. The angle φ between the horizontal axis of the sheave and the point of tangency of the ropes with the sheave represents the loss of the angle of wrap (α) with the application of a diverting pulley, as $\alpha = 180° - \phi$, and is given by the formula

$$\sin \phi = \frac{l\sqrt{l^2 + h^2 - (R_s - R_p)^2} - h(R_s - R_p)}{l^2 + h^2} \qquad (3.2)$$

in relation to the horizontal and vertical distances between the axes of rotation of the sheave and diverting pulley. For $R_s = R_p$

$$\sin \phi = \frac{l}{\sqrt{l^2 + h^2}} \quad \text{or} \quad \operatorname{tg} \phi = \frac{l}{h} \qquad (3.3)$$

Since the distance l is specified at a definite value $l = L - (R_s +$

R_p), the dependence of the angle of wrap upon the vertical distance h becomes evident.

If the axis of the rope leading on to or off a pulley is not coincident with the radial plane of the pulley, drag will occur resulting in rapid wear of rope and pulley groove. To minimize wear the angle between the rope axis and the radial plane of the pulley should not exceed a certain value which has been found to comply with good practice. This value is usually expressed as a ratio in terms of the cotangent of this angle and is 100 : 1 in the event of the distance between two pulleys or a pulley and a sheave being fixed. Where the drag takes place between two points of variable mutual distance so that the drag ratio varies as the car travels then the minimum drag ratio should be 41 : 1 (when the car or counterweight rests on a completely compressed buffer).

(c) Forces on sheave

The maximum traction that can be developed in the sheave grooves is a function of the actual coefficient of friction between the ropes and the grooves and the angle of contact that the rope makes with the circumference of the sheave (angle of wrap), and is generally expressed by the Euler's formula

$$\frac{T_1}{T_2} \leqq e^{f\alpha} \tag{3.4}$$

where T_1/T_2 is ratio between the greater and the smaller static tensile force in the portions of suspension rope situated on either side of the traction sheave, e is the base of natural logarithms, f is the coefficient of friction, dependent upon the shape of the groove, and α is the angle of wrap of the suspension ropes on the sheave (rad).

The static value of the tensile forces T_1 and T_2 depends upon the rated load, the mass of the car or counterweight respectively, the mass of one run of the suspension ropes and the roping factor. If compensating ropes are applied, the influence of their mass as well as of the tensioning force must not be neglected. In relation with Fig. 3.15 forces T_1 and T_2 are calculated for the fully loaded car at the lowest landing and generally expressed roping factor i, though the two-to-one system has been drawn.

$$T_1 = \left(\frac{Q+K}{i} + m_L\right) \times g_n \text{ (N)}; \qquad T_2 = \frac{Z}{i} \times g_n \text{ (N)}, \tag{3.5}$$

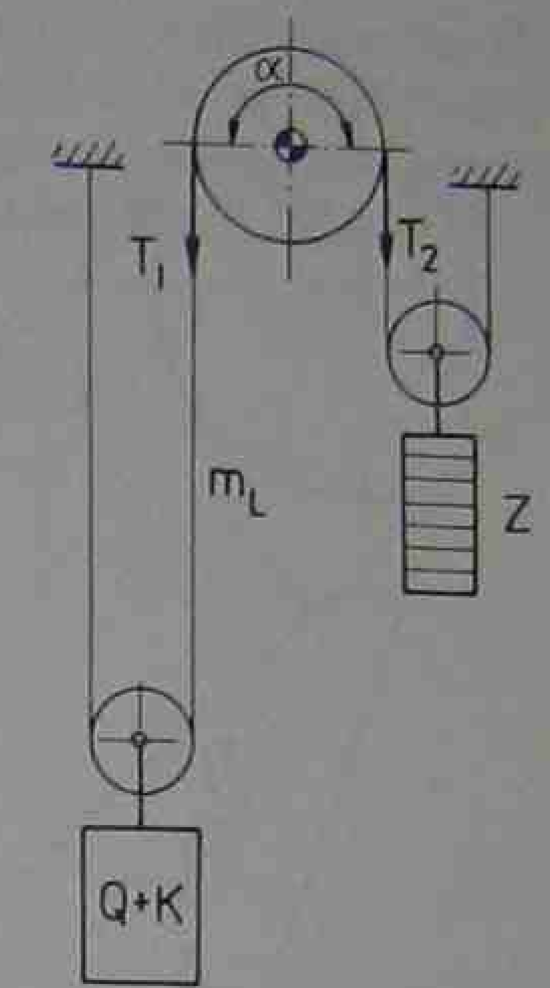


Fig. 3.15 — Diagram of forces on sheave.

where K is the mass of the empty car (kg), Q is the rated load (kg), m_L is the mass of one run of the suspension ropes (kg), Z is the mass of the counterweight (kg), i is the roping factor, and g_n is the acceleration of free fall (m/s^2).

Forces acting upon the sheave in the case of a double wrap drive being employed are depicted in Fig. 3.16; T_3 is the tensile force after the first wrap of the sheave (angle of wrap α_1). Efficiencies of both the driving and secondary sheaves are not taken into consideration.

$$\frac{T_1}{T_3} \leqq e^{f\alpha_1}; \qquad \frac{T_3}{T_2} \leqq e^{f\alpha_2} \tag{3.6}$$

and hence

$$\frac{T_1}{T_2} \leqq e^{f(\alpha_1+\alpha_2)} \tag{3.7}$$

The traction is increased considerably and therefore the application of a double wrap arrangement is very convenient from this point of view. On the other hand, the design of the machine is more complicated, the height is greater and the width of the sheave rim is larger because of the number of grooves being doubled. The number

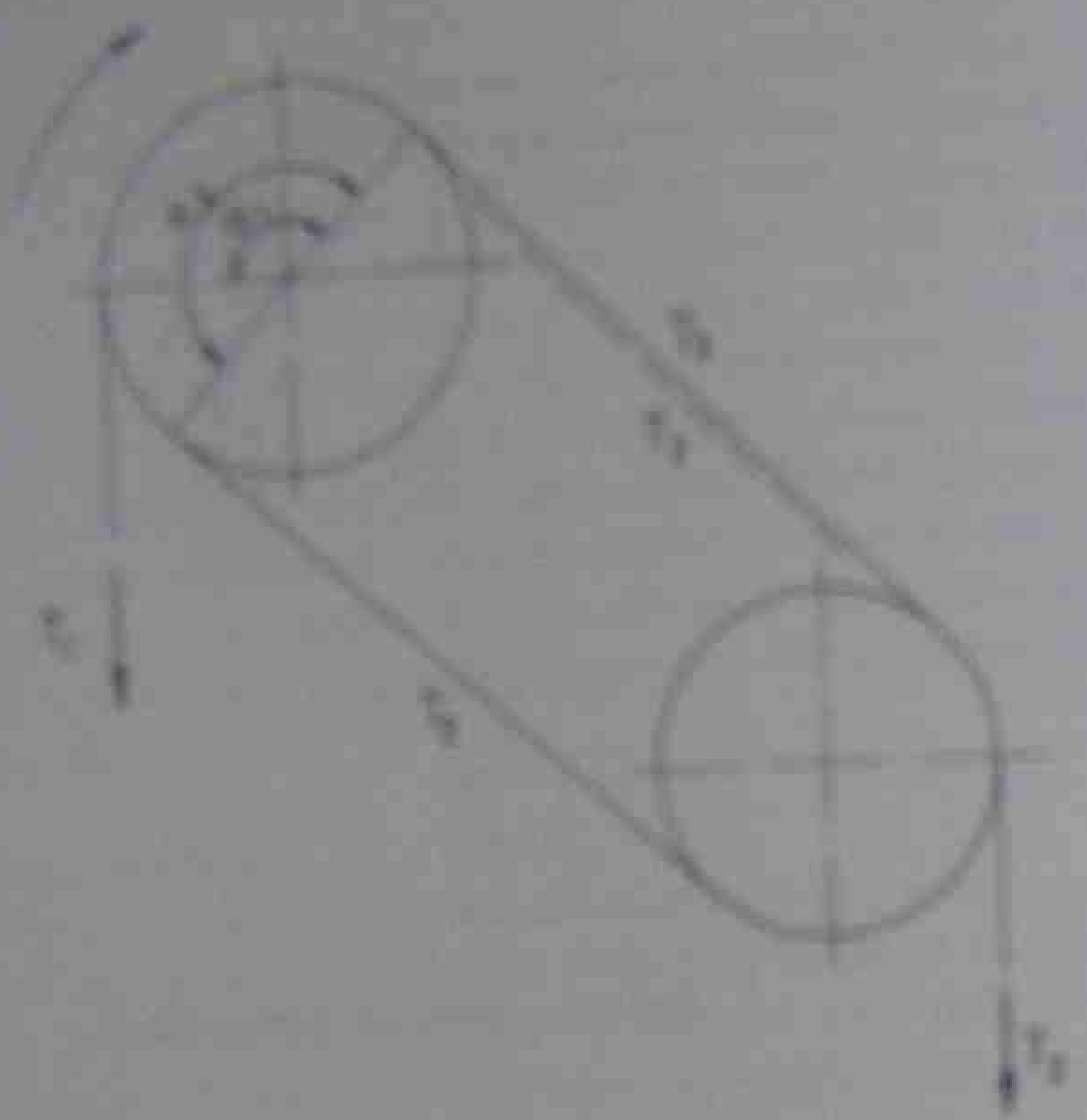


Fig. 3.16 — Forces on sheave with double wrap drive.

of rope bends is greater resulting in additional wear of the suspension ropes. The load imposed upon the sheave is noticeably higher than with a single wrap drive. Frictional resistances are greater and consequently the overall efficiency of the roping system is lower.

(d) Specific pressure of the rope in the sheave groove

The maximum permissible value of specific pressure is specified in BS 5655 as

$$p \leq \frac{12.5 + 4v_c}{1 + v_c} \qquad (3.8)$$

where v_c is the rope speed corresponding to the rated speed of the car (m/s).

In some European countries the maximum pressure is determined in dependence not only on the rope speed, but also on the intensity of traffic.

In calculating specific pressure in the sheave grooves of different profiles the original assumptions of Hymans and Hellborn (1927) will be utilized as they are still valid and used all over the world.

(i) *Vee groove*

The length of deformation of the rope due to the radial effect of the rope tension is assumed to be approximately one-third of the rope diameter and the specific pressure configuration along this distance is assumed to be sinusoidal (Fig. 3.17).

Fig. 3.17 — Distribution of specific pressure in a vee groove.

The maximum value of specific pressure at any point along the wrap of the rope on the traction sheave is then given by the following formula:

$$p = \frac{3\pi \times T}{2D \times d \times \sin\frac{\gamma}{2}} \ (\mathrm{N/mm^2}) \qquad (3.9)$$

where T is the tensile force at the point where specific pressure is calculated (N); D is the pitch diameter of the sheave (mm); d is the nominal rope diameter (mm); and γ is the angle of vee groove (°).

As seen in equation (3.9) the specific pressure is directly proportional to the tensile force, which varies exponentially along the arc of contact (Fig. 3.18), so that the absolute maximum of specific pressure will occur at the point where also the rope tension is maximum.

The abrasion of the groove causes the shape of the vee groove to convert to a round seat profile with a variable angle of contact, the

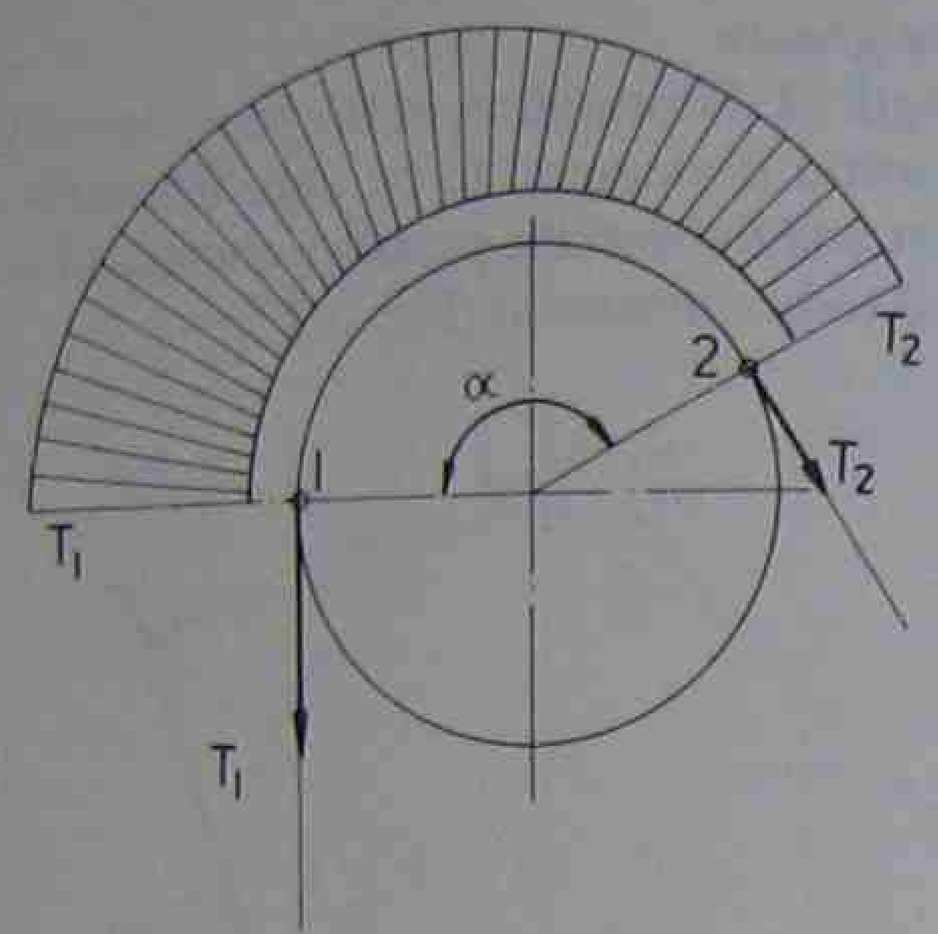


Fig. 3.18 — Variation of tension along the arc of contact of traction sheave.

effect being the same as that of an undercut groove (Fig. 3.19). The specific pressure is decreasing as well as the coefficient of friction and the traction. With the continuation of the abrasion the rope sets deeper in the groove and the arc of contact corresponding to the difference $\delta - \beta$ increases. The angle of the outer normal lines of the contact area δ can reach the maximum value of 180°, whilst the angle β is relatively small. In order to achieve a fixed minimum value of β undercut vee grooves are sometimes employed. The actual value of the specific pressure in a worn vee groove must be assessed for an undercut groove — see (iii).

(*ii*) *Round groove*

As shown in Fig. 3.20 after the abrasion of a round sheave groove has taken place, the rope is setting deeper in the groove and the radial shift of all peripheral points is the same. This means that the wear of the groove in the radial direction is constant along the arc of contact. Wear of the groove is caused by friction dependent upon the specific pressure. If the coefficient of friction is constant then the radial component of the specific pressure must also be constant at any point along the arc of contact. Mathematically expressed, as apparent from Fig. 3.21

$$\frac{p}{\cos\phi} = \text{const} \tag{3.10}$$

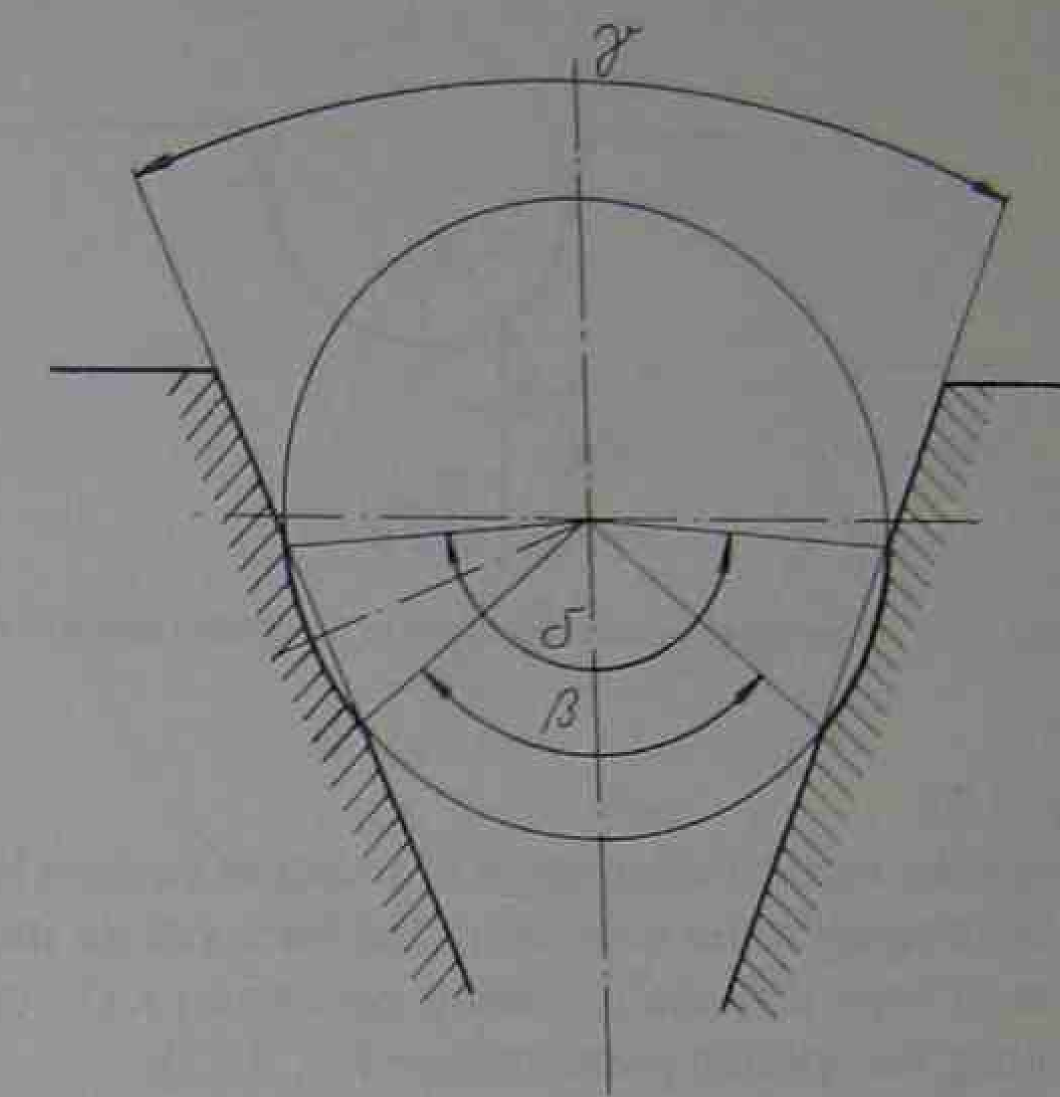


Fig. 3.19 — Vee groove of traction sheave after abrasion.

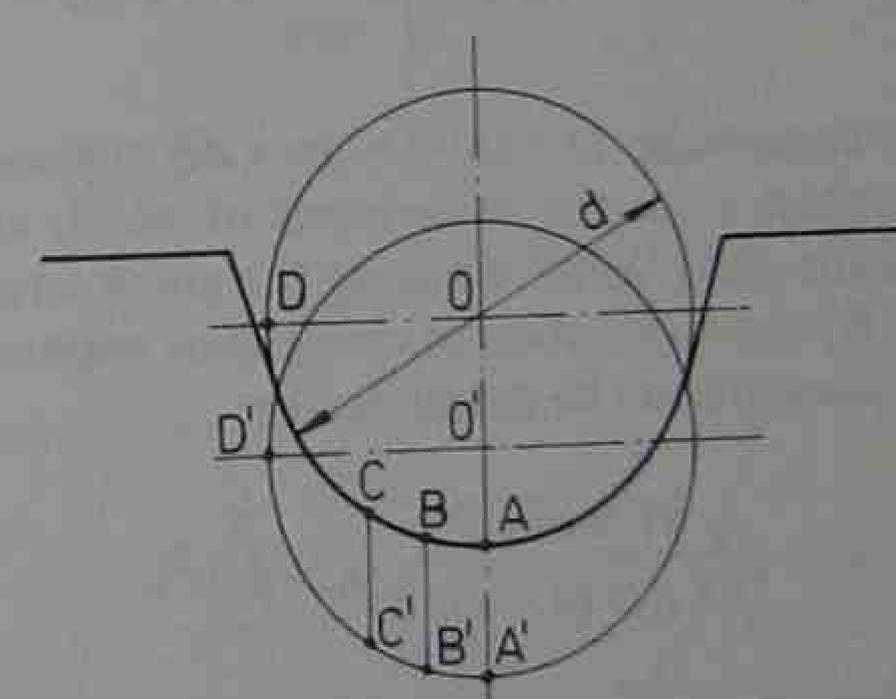


Fig. 3.20 — Radial shift of the rope in a round groove after abrasion.

and hence the specific pressure in a radial plane is generally

$$p = \text{const} \times \cos\phi$$

For $\phi = 0$, i.e. $\cos\phi = 1$, the pressure attains the maximum value so that it occurs at the bottom of the groove. The distribution of the specific pressure along the area of contact is shown graphically

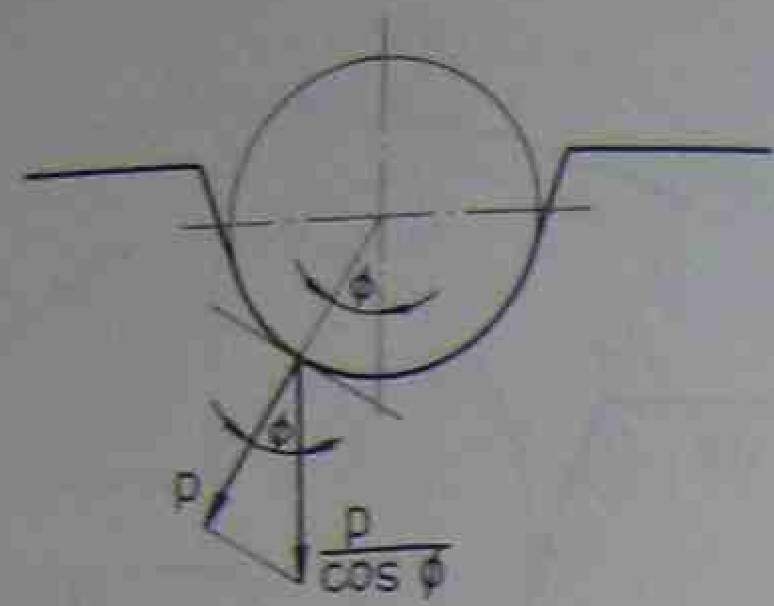


Fig. 3.21 — Specific pressure at any point of the contact area in a round groove.

in Fig. 3.22.

Since the vertical summation of pressure vectors in a radial plane must be in equilibrium with the radial force per an indefinitely small portion of rope dN, the following equation (3.11) can be used for calculating the specific pressure (see Fig. 3.22):

$$dN = \frac{D \times d}{4} \times d\alpha \times \int_{-(\delta/2)}^{+(\delta/2)} p \times \cos\phi \times d\phi \qquad (3.11)$$

The expression $D \times (d/4) \times d\alpha \times d\phi$ represents an indefinitely small contact area, the dimensions of which are $(d/2) \times d\phi$ in the radial plane and $(D/2) \times d\alpha$ along the arc of wrap of the rope on the sheave. By the application of convenient mathematical methods the specific pressure can be calculated as

$$p = \frac{8T \times \cos\phi}{D \times d \times (\delta + \sin\delta)} \text{ (N/mm}^2) \qquad (3.12)$$

which gives the value of general validity.

The maximum specific pressure will occur at the bottom of the groove ($\cos\phi = 1$), at the point where the rope tension T is of maximum value.

(iii) Undercut groove

The distribution of the specific pressure along the area of contact (Fig. 3.23) is analogous to that of a round groove as it is given by the same curve ($p = \text{const} \times \cos\phi$), however there is a principal difference owing to the modified groove profile. The line of contact between the rope and groove is interrupted where the pressure would originally be the greatest, therefore the pressure vectors are spread

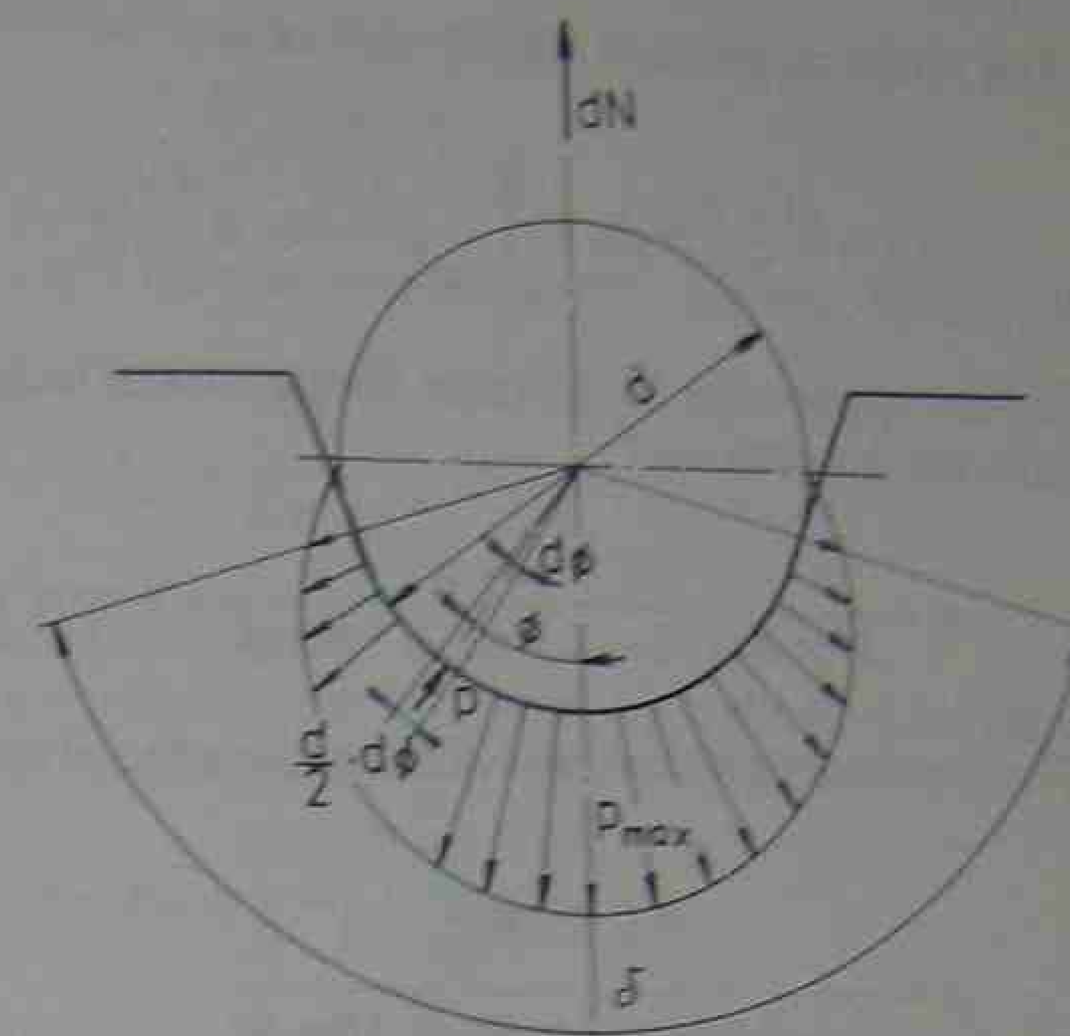


Fig. 3.22 — Distribution of specific pressure in a round groove.

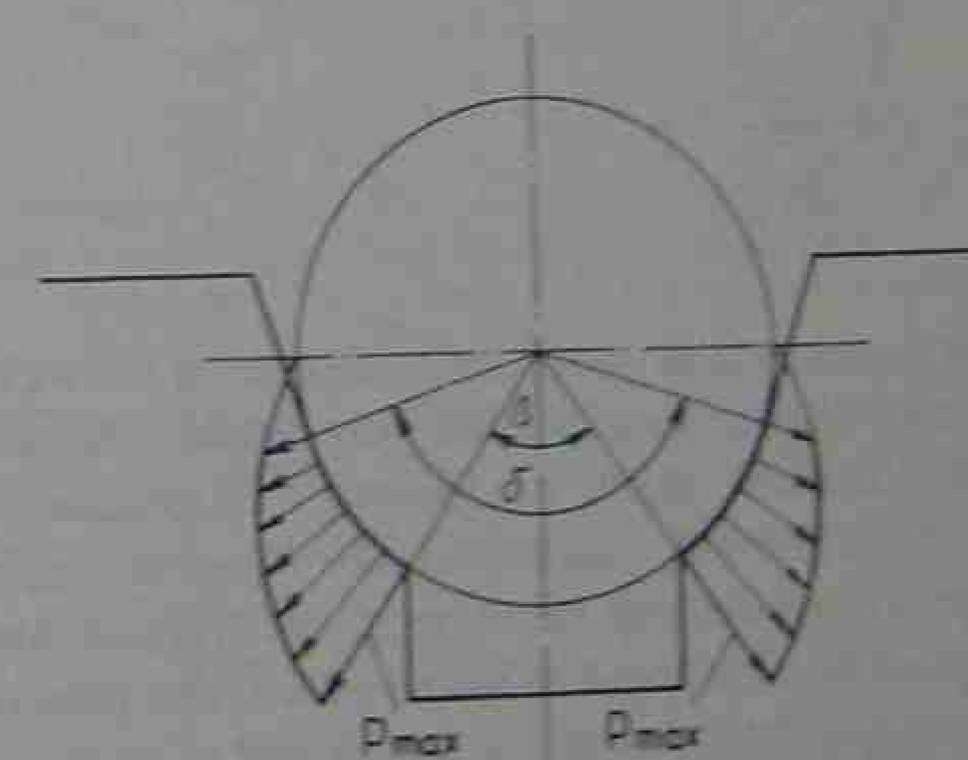


Fig. 3.23 — Distribution of specific pressure in an undercut groove.

over a smaller area of groove contact, thereby inducing groove pressures higher than was found with the round groove.

The initial equation is slightly different

$$dN = \frac{D \times d}{2} \times d\alpha \times \int_{\beta/2}^{\delta/2} p \times \cos\phi \times d\phi \tag{3.13}$$

and by the application of the same mathematical techniques the final formula can be obtained

$$p = \frac{8T \times \cos\phi}{D \times d \times (\delta - \beta + \sin\delta - \sin\beta)} \ (\text{N/mm}^2) \tag{3.14}$$

The maximum pressure will occur at the edge of the undercut, i.e.

$$p = \frac{8T \times \cos\frac{\beta}{2}}{D \times d \times (\delta - \beta + \sin\delta - \sin\beta)} \tag{3.15}$$

(e) Coefficient of friction of the ropes in the grooves

(i) Vee-groove

As generally known the coefficient of friction for a vee-shaped groove is

$$f = \frac{\mu}{\sin\frac{\gamma}{2}} \tag{3.16}$$

where μ is the actual coefficient of friction between the rope and a steel or cast-iron sheave; for the purpose of computation the value of $\mu = 0.09$ is usually taken. γ is the angle of groove (°).

For the currently employed 35° vee-groove $f = 3.325\ \mu$, which results in a considerable increase of the tractive force.

As soon as abrasion has taken place, the groove configuration is converted and the value of the coefficient of friction becomes smaller — see section 3.1(d).

(ii) Round groove

The elementary tangential reaction dF induced by the radial force dN per an indefinitely small portion of rope may be calculated as follows:

$$dF = f \times dN = \frac{D \times d}{4} \times d\alpha \times \mu \times \int_{-(\delta/2)}^{+(\delta/2)} p \times d\phi \tag{3.17}$$

After substitutions

$$p = \frac{8T \times \cos\phi}{D \times d \times (\delta + \sin\delta)}$$

and $dN = T \times d\alpha$ and after integration the final formula becomes:

$$f = 4\ \mu \times \frac{\sin\frac{\delta}{2}}{\delta + \sin\delta} \tag{3.18}$$

(iii) Undercut groove

If the same procedure as in (ii) is applied the final formula is acquired

$$f = 4\ \mu \times \frac{\sin\frac{\delta}{2} - \sin\frac{\beta}{2}}{\delta - \beta + \sin\delta - \sin\beta} \tag{3.19}$$

In general the formula for f may be expressed in the form $f = k \times \mu$ and k may be read off the diagram in Fig. 3.24 for round and undercut grooves.

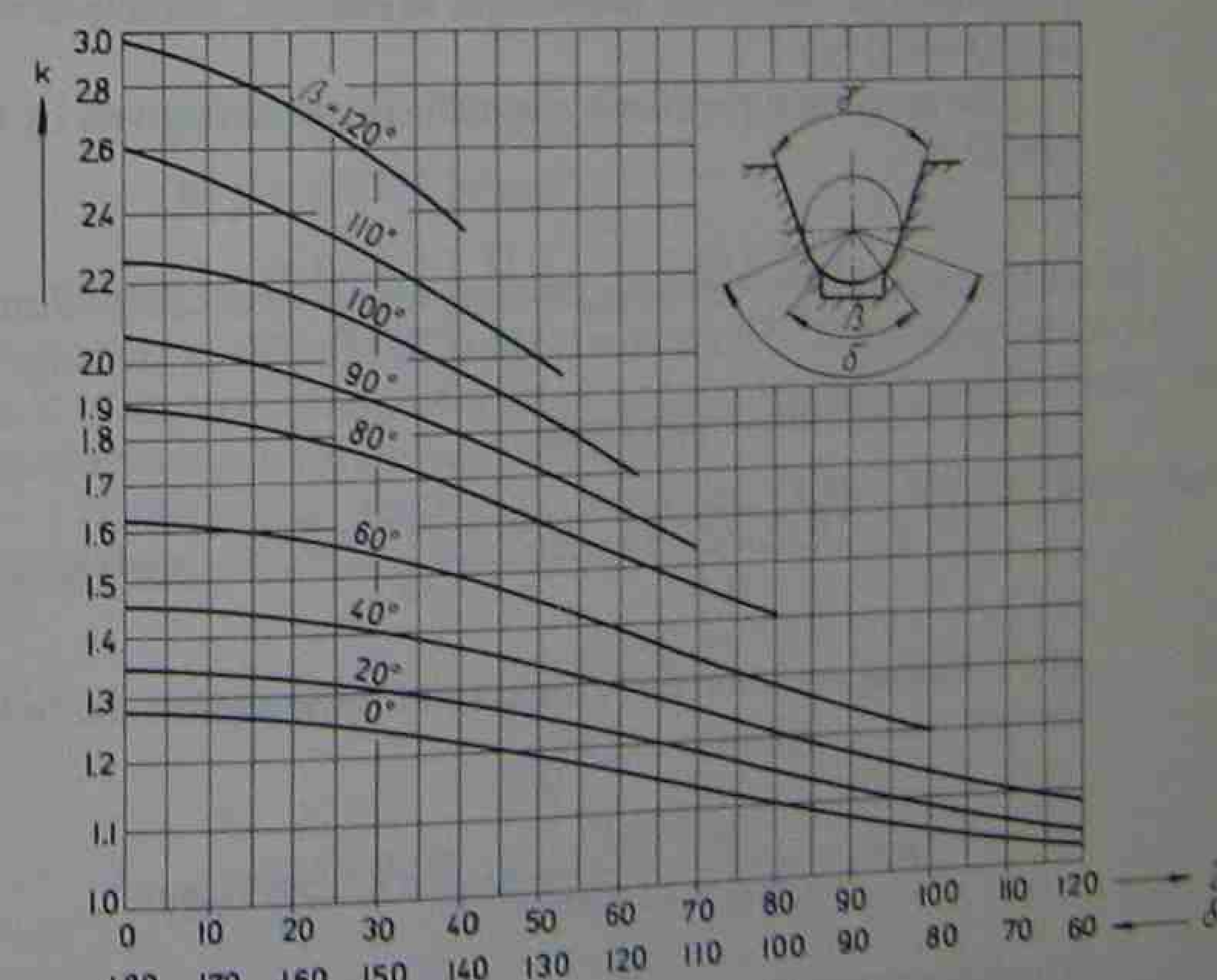


Fig. 3.24 — Diagram for determination of the coefficient of friction in round and undercut grooves.

For the angle δ being equal to 180° the coefficient of friction achieves its maximum value, namely for a round groove

$$f=\frac{4}{\pi}\times\mu=1.273\mu \tag{3.20}$$

for an undercut groove

$$f=4\mu\times\frac{1-\sin\dfrac{\beta}{2}}{\pi-\beta-\sin\beta} \tag{3.21}$$

Example 3.1

For the elevator of Example 2.1 determine the specific pressure and coefficient of friction in the sheave grooves of the following profiles:

(1) vee groove, $\gamma=35°$
(2) round groove, (a) $\delta=167°$; (b) $\delta=180°$
(3) undercut groove, (a) $\beta=90°$, $\delta=180°$; (b) $\beta=105°$, $\delta=180°$.

Calculate the values for two sheave diameters, namely $D=560$ mm and $D=610$ mm.

The maximum permissible specific pressure is given by the equation (3.8)

$$p\leqq\frac{12.5+4v_c}{1+v_c};\ p_{\max}=\frac{12.5+4\times1.6}{1+1.6}=7.269\ \text{N/mm}^2$$

The maximum tensile force

$$T=\frac{(K+Q)\times g_n}{n}=\frac{(630+737)\times9.81}{4}=3352.5\ N$$

(1) Utilizing equation (3.9) the result for the sheave diameter of 560 mm is:

$$p=\frac{3\pi\times3352.5}{2\times560\times11\times\sin17.5°}=8.5288\ \text{N/mm}^2$$

which is excessive:
for the sheave diameter of 610 mm

$$p=7.8298\ \text{N/mm}^2\ \text{(also excessive)}$$

If a vee groove is to be employed, it is necessary to either increase

the number of the suspension ropes to five or use ropes of a larger nominal diameter (13 mm).

The coefficient of friction

$$f=\frac{\mu}{\sin\dfrac{\gamma}{2}}=\frac{0.09}{\sin17.5°}=0.299$$

(2) The specific pressure is calculated from equation (3.12) for the sheave diameter of 560 mm

$$\text{(a) } p=\frac{8\times3352.5}{560\times11\times\left(\dfrac{167}{180}\times\pi+\sin167°\right)}=1.386\ \text{N/mm}^2$$

$$\text{(b) } p=\frac{8\times3352.5}{560\times11\times\pi}=1.3859\ \text{N/mm}^2$$

for the sheave diameter of 610 mm

$$\text{(a) } p=1.273\ \text{N/mm}^2$$

$$\text{(b) } p=1.2723\ \text{N/mm}^2$$

The decrease of the specific pressure with the alteration of the angle δ from 167° to 180° is of no importance. The pressure is very low as it is the coefficient of friction and the traction capability may be insufficient as a result (see Example 3).

Coefficient of friction — equation (3.18) will be used for calculation

$$\text{(a) } f=4\times0.09\times\frac{\sin83.5°}{\dfrac{167}{180}\times\pi+\sin167°}=0.1139$$

$$\text{(b) } f=\frac{4\times0.09}{\pi}=0.1146$$

(3) For calculation of the specific pressure equation (3.14) is used. For the sheave diameter of 560 mm

$$\text{(a) } p=\frac{8\times3352.5\times\cos45°}{560\times11\times(\frac{\pi}{2}-1)}=5.3936\ \text{N/mm}^2$$

$$\text{(b) } p = \frac{8 \times 3352.5 \times \cos 52.5°}{560 \times 11 \times (\pi - \frac{105}{180} \times \pi - \sin 105°)} =$$

$$= 7.7257 \text{ N/mm}^2 \ldots \text{ excessive}$$

For the sheave diameter of 610 mm

(a) $p = 4.9515$ N/mm^2

(b) $p = 7.092$ N/mm^2

Coefficient of friction — equation (3.19)

$$\text{(a) } f = 4 \times 0.09 \times \frac{1 - \sin 45°}{\frac{\pi}{2} - 1} = 0.1847$$

$$\text{(b) } f = 4 \times 0.09 \times \frac{1 - \sin 52.5°}{\pi - \frac{105}{180} \times \pi - \sin 105°} = 0.2168$$

In all cases apart from those indicated as excessive, the suspension ropes satisfy the condition concerning the specific pressure in the sheave groove. Before making the final decision about the technical parameters of the traction drive further aspects must be taken into consideration (e.g. life of ropes and sheave grooves) and calculation of the traction capability must be carried out (see Example 3).

(f) Traction under different conditions

The tractive force must be sufficient enough at any time regardless of the car load, car position and direction of travel, i.e. the validity must be maintained of equation (3.4)

$$\frac{T_1}{T_2} \leqq e^{f\alpha}$$

To avoid the necessity of exact computation of the actual traction, which might be rather complicated, especially with multiple roping systems, an extended formula of general validity has been adopted in BS 5655 and EN 81, as given in equation (3.22)

$$\frac{T_1}{T_2} \times C_1 \times C_2 \leqq e^{f\alpha}, \tag{3.22}$$

where T_1/T_2 is the ratio between the greater and the smaller static forces in the portions of rope situated on either side of the traction sheave; two cases are taken into consideration, (i) car stationary at the lowest landing level with a load equivalent to 125% of the rated load, and (ii) unloaded car stationary at the highest landing level. C_1 is the coefficient taking account of acceleration, deceleration and specific conditions of the installation

$$C_1 = \frac{g_n + a}{g_n - a},$$

where g_n is the standard acceleration of free fall (m/s^2) and a is the braking deceleration of the car (m/s^2). The minimum permissible values of C_1 are given in Table 3.1.

Table 3.1 — Values for C_1

C_1	Rated speeds (m/s)
1.1	≤ 0.63
1.15	0.63 to ≤ 1.0
1.2	1.0 to ≤ 1.6
1.25	1.6 to ≤ 2.5

For rated speeds in excess of 2.5 m/s C_1 must be calculated for each particular case, but the minimum value must not be less than 1.25. C_2 is the coefficient taking account of the variation in profile of the groove due to wear

$C_2 = 1.0$ for round and undercut grooves
$C_2 = 1.2$ for vee grooves

Great attention must be given to the determination of the tractive force particularly with high rise elevator installations, where the weight of the ropes on the counterweight side may be sufficient to provide traction while the counterweight is resting on the buffers and the machine is rotated in the 'up' direction. Precautions must be taken to prevent the car from being raised under these conditions.

Maximum available traction is limited by the angle of wrap (α) and the coefficient of friction of the ropes in the grooves (f) in relation with equation (3.1) and consequently the maximum traction is developed when the quantity $f \times \alpha$ is maximized. This may be achieved by increasing (i) f, (ii) α, or both.

(i) Better traction capability through a higher coefficient of friction may be achieved by using non-metallic groove liners. They have been used for many years with traction-type machines in mines (Koeppe); their driving sheaves of large diameter have been provided with liners installed in short segments individually mounted.

The principal requirements for groove liners are summarized:

- high coefficient of friction under all circumstances (both static and dynamic, on application of rope lubricants)
- resistance to specific pressure in the contact area between the rope and the liner
- resistance to wear (abrasion)
- resistance to rope lubricants
- stability of physical properties for different environmental conditions (temperature, humidity)
- resistance to ageing
- inflammability or low level of smoke and/or toxic gas originating when a thermal decomposition takes place

The first groove liners for traction elevators, that appeared in the mid-1960s, were of similar design. They were composed of short polyurethane segments arranged with resilient treads. The segments could be easily mounted into place, the last one being cut to fit the space left. The rope spacing of the groove was increased with the application of groove liners (Fig. 3.25) and the machining of the sheave grooves was more costly than with conventional sheaves. However, the traction was increased as well as the applicability of

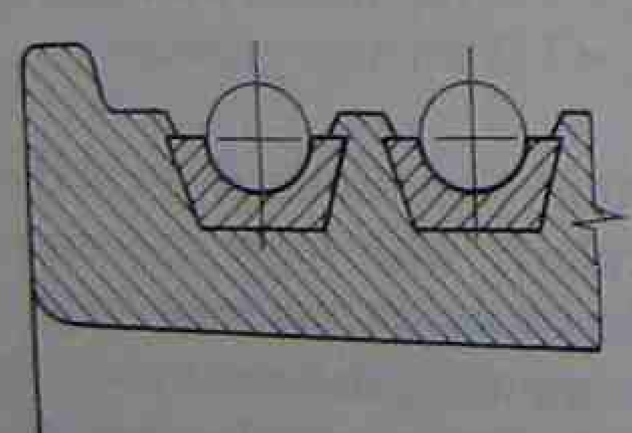


Fig. 3.25 — Original arrangement of sheave inserts.

single wrap drive and life of elevator ropes was extended because of lower specific pressure in the round groove of the inserted liner. A reduction of noise level associated with the contact of steel ropes in metal grooves was achieved.

The current design of groove liners manufactured by the Otis Elevator Co. and designated as Cable-Save™ is a treaded flexible ring of polyurethane placed in a machined vee groove of the driving sheave.

The principal requirement from the safety aspect concerning the application of non-metallic groove liners is the maintenance of sufficient traction in the event of the liners having failed for any reason (e.g. being destroyed by fire). In compliance with ANSI/ASME A 17.1 a car with a load equivalent to 125% of the rated load must be stopped safely and held in its position in such a case. This requirement is satisfied by seating the liners in 35° vee grooves of the machined portion of the sheave rim (Fig. 3.26). If a failure of

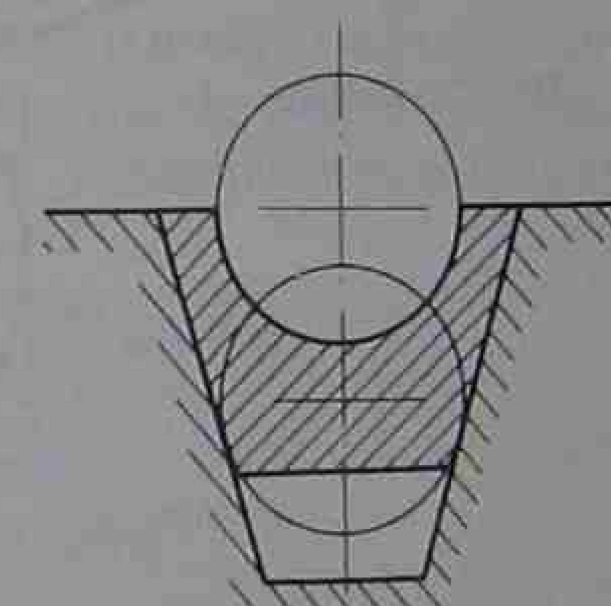


Fig. 3.26 — Present arrangement of sheave inserts.

polyurethane liners occurs the rope will make a line contact with the groove along the arc of wrap and exceedingly high specific pressures will be induced, but they can be considered permissible because of the relatively short time of the car motion.

Further advantages of non-metallic groove liners are the elimination of field regrooving of the sheave as wear occurs on liners only and the reduction in machine vibrations transmitted to the car. Polyurethane liners have been used on both geared and gearless machines with car speeds up to 5 m/s.

(ii) The angle of wrap (α) may be increased considerably by the application of double wrap drive — see section 3.1(c). Another solution of the problem appeared in the early 1980s, when the Otis Elevator Co. introduced the so-called 'long wrap drive'. This roping arrangement embodies the arc of contact around the driving sheave of approximately 270°C as shown in Fig. 3.27. The ropes leading to the car are interwoven with the ropes leading to the counterweight via a diverting pulley. In order to avoid any undesirable interference of

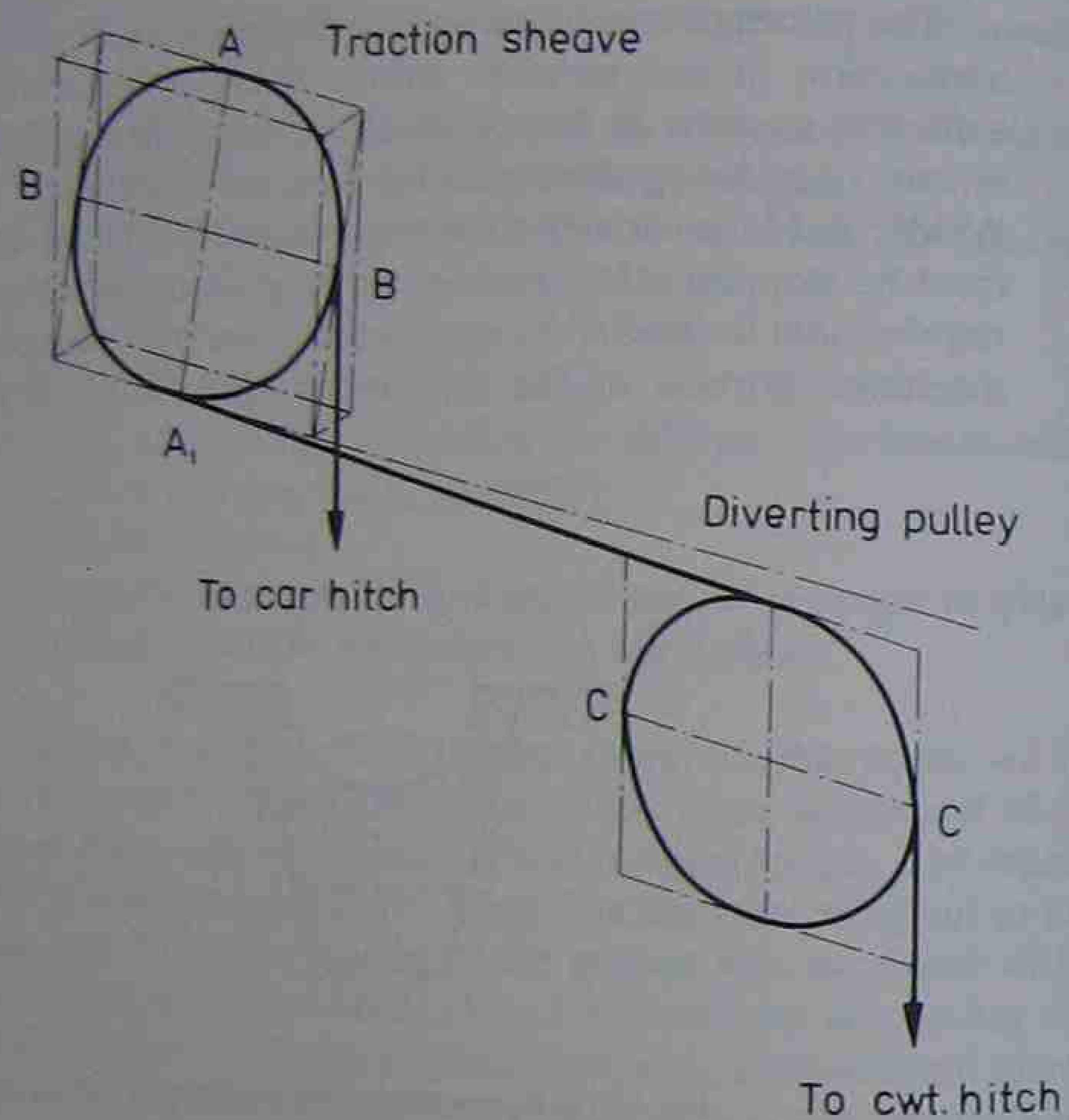


Fig. 3.27 — Diagram of the 'long wrap' arrangement of the machine.

the rope leads, sufficient clearance must be ensured. For this purpose the plane of the driving sheave is slightly tilted from the vertical plane, while the diverting pulley is shifted perpendicularly and rotated round its vertical axis so that its horizontal axis C–C is set parallel to the offset rope lead. Straight lead from the tangent point A_1 and parallel to axis B–B does not give sufficient clearance at cross-over.

Both the driving sheave and diverting pulley are provided with non-metallic groove liners, the purpose of which is not only to

increase traction, but also to minimize wear of ropes on account of the rope drag occurring on the sheave as a result of its plane being tilted. Several thousands of geared machines with car speeds up to 1.75 m/s have been arranged in this way.

Example 3.2

Calculate the traction capability of the driving sheave of the elevator mentioned in Examples 2.1 and 3.1.

Assume that a diverting pulley is used with the machine and the angle of wrap is consequently $\alpha = 165°$. For calculation the extended equation (3.22) will be used. Calculation of the ratio T_1/T_2 will be carried out in the case of the car stationary at the lowest landing level with a load equivalent to 125% of the rated load.

$$\frac{T_1}{T_2} = \frac{1.25\,Q + K}{Z} = \frac{1.25 \times 630 + 737}{1020} = 1.4946$$

Another case which is usually taken into consideration, namely an unloaded car stationary at the highest landing level is not so dangerous as the ratio T_1/T_2 is smaller than in the preceding case:

$$\frac{T_1}{T_2} = \frac{Z}{K} = \frac{1020}{737} = 1.384$$

$$\frac{T_1}{T_2} \times C_1 \times C_2 \leqq e^{f\alpha}$$

Assuming that the rate of deceleration will not exceed $a = 1.0\ \mathrm{m/s^2}$ the coefficient C_1 will be of the following magnitude

$$C_1 = \frac{g_n + 1}{g_n - 1} = 1.227$$

$C_2 = 1$ for round and undercut grooves

$$\frac{T_1}{T_2} \times C_1 = 1.4946 \times 1.227 = 1.8339$$

$$1.8339 \leqq e^{f\alpha}$$

The minimum value of the coefficient of friction $f_{\min}$ can be easily calculated from the equation

$$1.8339 = e^{f_{min} \times \alpha} \tag{3.23}$$

$$f_{min} = 0.2106$$

The examination of the results obtained in Example 3.1 proves the tractive force to be sufficient enough only with the undercut sheave groove of the angle $\beta = 105°$.

(g) Rope slip. Wear of sheave grooves

Rope slip

The relative motion of the rope in a sheave groove has two components and is given by the superposition of both.

(i) The rope stretch (elongation) is proportional to the tension and varies accordingly, while there is practically no tangential deformation of a steel or cast-iron sheave. Since there is a considerable difference between the tensile forces on the tight and slack sides of the sheave, a change in tension occurs in each individual portion of the rope whilst passing around the sheave, resulting in a slippage.

(ii) Rope slip caused by other ropes the speed of which is different because of different pitch diameters of the sheave grooves. This factor is of considerably greater influence.

Wear of sheave grooves

Because of a rope to sheave interaction taking place during the elevator operation most of the factors effecting life of elevator ropes, see section 2.2(d), will also affect life of the sheave grooves.

An analysis was carried out and tests were undertaken in Czechoslovakia several years ago. It was shown that the life of the grooves was remarkably affected by factors causing an uneven distribution of tensile forces in the elevator ropes — first of all by different pitch diameters of the grooves. This uneven distribution resulted in different magnitudes of specific pressure between the ropes and sheave grooves.

With different pitch diameters peripheral velocities of individual ropes will be different and consequently some ropes will slip in their grooves being either accelerated or decelerated by the others, their tension varying in dependence upon the diameter of the groove and the direction of car travel.

Wear is caused by both the tangential and radial effect of the rope. The tangential effect is represented by the rope slipping in the groove and is notably dependent upon the specific pressure. A comparison of differences between the theoretical travel of each rope and a fictitious travel calculated for a sheave groove of an average pitch diameter

revealed that wear of particular grooves did not correspond to the relative distances and consequently the tangential effect of the rope could not be of primary importance.

The radial effect is induced by the tension in the rope that initiates the specific pressure between the rope and the groove. The magnitude of the pressure determines the rate of rope deformation and thus the relative radial motion of the rope in the groove. Average radial forces acting upon a unit contact area of 1 mm in length were calculated for each rope and proportional values were obtained as well as proportional values of wear of all grooves after a considerable number of car trips. Comparison of the unit radial forces and wear for each groove revealed the primary significance of the radial effect of the rope as the proportional values of the unit radial forces and wear were almost identical.

Since the magnitudes of specific pressure in individual grooves are directly proportional to the rope tension, see section 3.1(d), all forces should be of the same value if even wear is to be achieved. This is conditioned by the sheave geometry — the difference between the pitch diameters of individual grooves should be minimized.

3.2 DRUM DRIVE

Apart from hydraulic elevators all earlier types of elevator machines were of winding-drum type; there were two sets of suspension ropes employed, one end of the ropes of each set being securely fastened on the drum, the other ends to the car or counterweight, respectively. One set of ropes was wrapped clockwise round the drum and the latter one anti-clockwise, so that while one set was unwrapping the other one was being wrapped on the drum. This layout and machine design cannot be taken into consideration as no counterweight can be used with drum elevators in conformity with both EN 81 and ANSI/ASME A 17.1. Furthermore, A 17.1 specifies another three restrictions for drum drive application, namely,

(1) Winding-drum machines cannot be used for passenger elevators.
(2) The rated speed of the elevator must not exceed 0.25 m/s.
(3) The height of travel must not exceed 12.2 m.

The drum is threaded and only one layer of rope is wound on it to extend the life of both the rope and the grooves. The grooves are of round shape with the radius of $(0.53–0.56) \times d$, where d is the nominal rope diameter. For n ropes in a set, the grooves must be multiple thread (n-threaded). When the car rests on its fully compressed buffers, one and a half turns of rope should remain in the grooves of the drum. Dimensions of the threaded portion of a drum are denoted in Fig. 3.28.

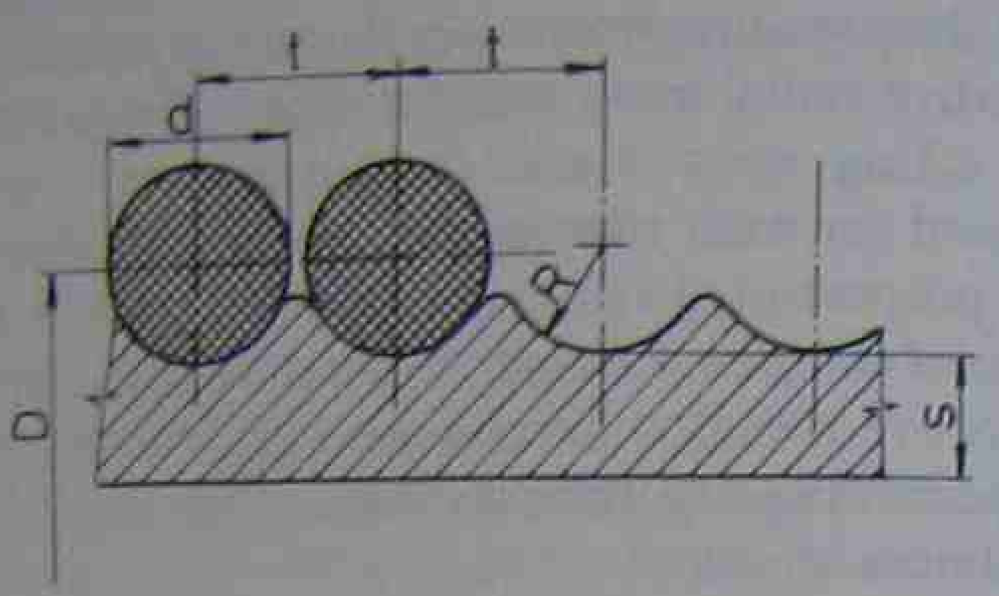


Fig. 3.28 — Profile and dimensions of the threaded portion of a drum.

The ratio between the pitch diameter of the drum and the nominal diameter of the suspension ropes must be at least 40.

The number of turns for one rope is given by the formula

$$z = \frac{i \times H}{\pi \times D} + 1.5\,, \tag{3.24}$$

where i is the roping factor and H is the car travel (m).

The length of the threaded portion of the drum for one rope is

$$l = z \times t \text{ (mm)}\,, \tag{3.25}$$

where t is the lead of thread (mm) — see Fig. 3.28.

If the number of suspension ropes be n, the length will be increased accordingly, i.e.

$$l = z \times t \times n \text{ (mm)}.$$

The fleet angle of the rope (the angle between the longitudinal axis of the rope and the axis of the groove) should not exceed 4° either side of the groove axis.

The rope anchorage on the drum is usually internal wedge type socket or at least two rope clamps.

Dimensions of the drum are particularly dependent upon the rated load and the car travel, these two parameters being limiting

factors for the drum drive application. The drum drive can hardly be used for travels in excess of 30 m and the practical maximum number of suspension ropes is three. The upper position of the machine is inconvenient as in the case of the suspension ropes leading directly onto the drum lateral forces on the guide rails are induced.

Because of the many advantages of the traction drive the drum drive is seldom employed nowadays, except with service elevators (dumbwaiters) for low rises and lower position of the machine, or unconventional freight elevator installations where elevators currently manufactured cannot be used.

4

Gearing

4.1 DESIGN AND APPLICATION

Gearless machines are usually used for rated speeds over 2 m/s, while for lower speeds geared machines must be applied. Spur gearboxes were used from time to time in the past, but with the advancement of design and production techniques the worm gearing has become the accepted standard for geared elevator machines. Helical geared machines have been introduced by the Otis Elevator Co. recently, employing a double reduction of high efficiency.

The worm gearing is sometimes used in combination with a belt drive (e.g. trulevel machines for accurate levelling equipped with two motors) or an additional pair of spur gears (heavy duty freight elevators). Indirect-drive machines, utilizing V-belt drives or toothed drive belts must include at least three belts operating in parallel as a set. The minimum factor of safety related to the breaking strength of the belt is 10.

Since other types of gearing are used only exceptionally the worm gearing will be given attention in this chapter. The application of a worm gear speed reducer brings several advantages.

(a) It is very compact and of the smallest dimensions of all types of gearing for the given ratio and transmitted power.
(b) It has a minimum number of moving parts thus minimizing maintenance and replacement.
(c) The sliding action of the worm gearing results in a quiet operation.
(d) It has an inherently high shock load resistance.

The worm is usually cut from forgings of alloy steel which provide

a tough core of high strength and are suitable for case hardening to get a hard working surface. The material is mostly nickel or nickel chromium steel, but some companies prefer 0.4 per cent or 0.55 per cent carbon steel for light duty gearing, normalized. The hardened worms are ground and polished to provide a perfect tooth profile and maximum smoothness of the surface in order to minimize friction and wear. The rims of worm wheels are made of centrifugally cast bronze, machined to mate with the worm. The bronze alloy may be of phosphor, copper–tin or copper–tin-nickel composition of low coefficient of friction. Centrifugal casting results in a fine and perfect homogeneity of material structure with a great fracture resistance and good sliding properties.

The worm shaft is always supported by two radial bearings and one axial bearing for sustaining the axial thrust. It can be placed either in the upper position (over-driven worm-gear) or in the lower position (under-driven worm-gear) below the worm wheel. The upper position seems to be used less frequently nowadays, mostly with light and medium duty machines. Its advantages are easy sealing of the gearbox, easy control of the gearing and the worm-wheel shaft being located low above the machine framework. On the other hand, the lubrication of the worm is generally worse than with under-driven worm-gears, particularly during starting periods when metal-to-metal contact may occur under high loads. Also in braking periods the worm-wheel velocity may not be high enough to provide a sufficient supply of oil to the teeth contact area.

A typical under-driven worm-gear is shown in Fig. 4.1.

Worm thread surfaces are involute helicoids with a normal pressure angle of 15 or 20 degrees. It should be noted that as the pressure angle is increased above 20 degrees, the tooth is subjected to increased compressive force and it becomes necessary to use lubricants suitable for higher unit pressures.

The number of starts of the worm n is directly related to the gear ratio i_G.

$$i_G = \frac{N}{n}, \tag{4.1}$$

where N is the number of teeth of the worm wheel; as a rule

$N \geqq 36$ for pressure angle 15°
$N \geqq 24$ for pressure angle 20°.

If 85 is accepted as the maximum number of teeth of the worm

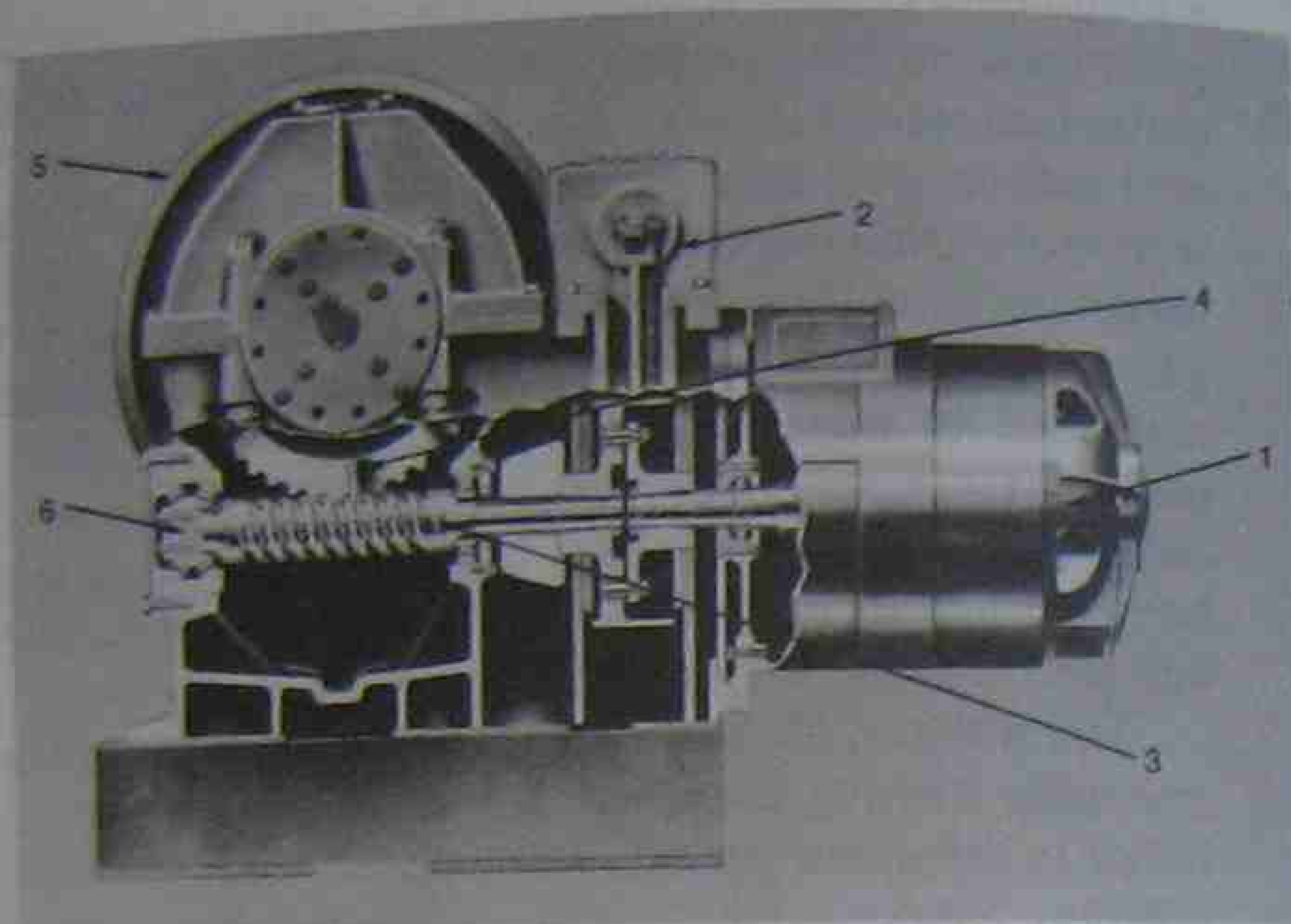


Fig. 4.1 — A typical under-driven worm-gear (Otis Elevator Co.).

1. a.c. driving motor
2. d.c. brake
3. Worm and worm shaft machined in one piece
4. Worm wheel
5. Traction sheave
6. Tapered roller bearings

wheel (in conformity with the experience of most elevator manufacturers to avoid excessive dimensions of the gearing) the maximum gear ratio dependent on the number of starts will be:

Number of starts	Maximum gear ratio
1	85
2	42
3	28

The layout of the low-speed shaft may be accomplished in several different ways. The shaft may be supported by:

(a) Two bearings in the gearbox with the traction sheave overhung (Fig. 4.2) for relatively small or medium loads.
(b) Two bearings one of which is located in the gearbox, the housing of the other being an integral part of the gearbox casting on the other side of the sheave; the worm wheel is overhung (Fig. 4.3).
(c) Three bearings — used sometimes for large capacity heavy duty freight elevators.

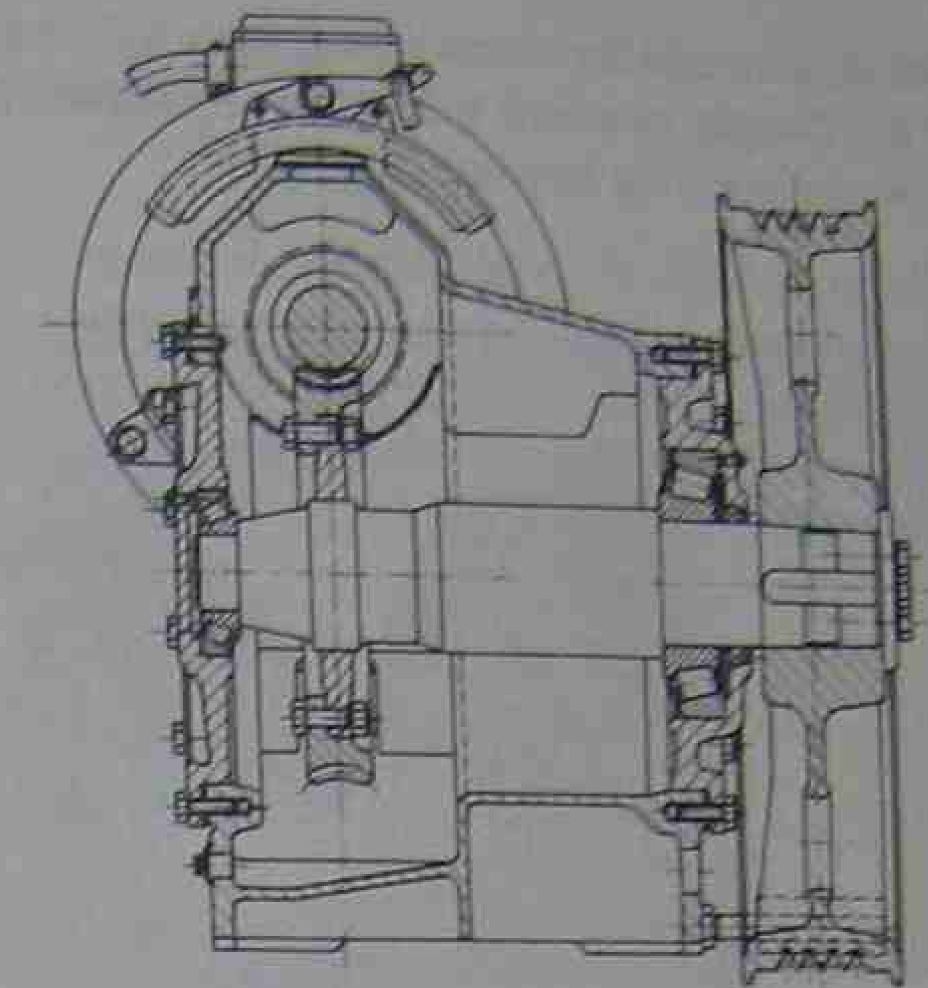


Fig. 4.2 — Cross-sectional view of the low-speed shaft with the sheave overhung (Otis Elevator Co.).

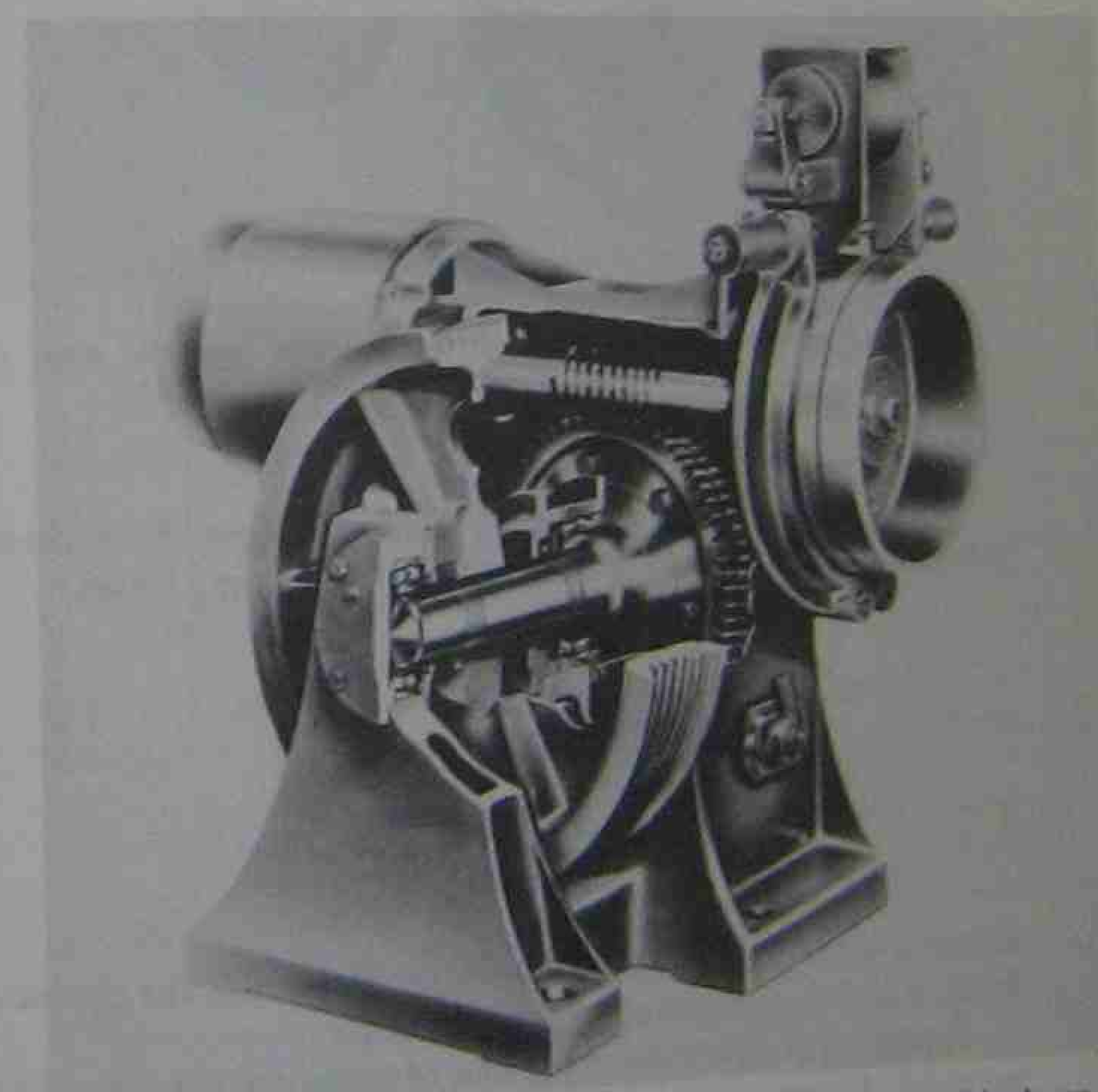


Fig. 4.3 — A low-speed shaft supported by two bearings on either side of the sheave (Otis Elevator Co.).

(d) The sheave and the worm wheel have a common centre (spider), to the flanges of which their rims are bolted; they are supported by a non-rotating journal (Fig. 4.4.).

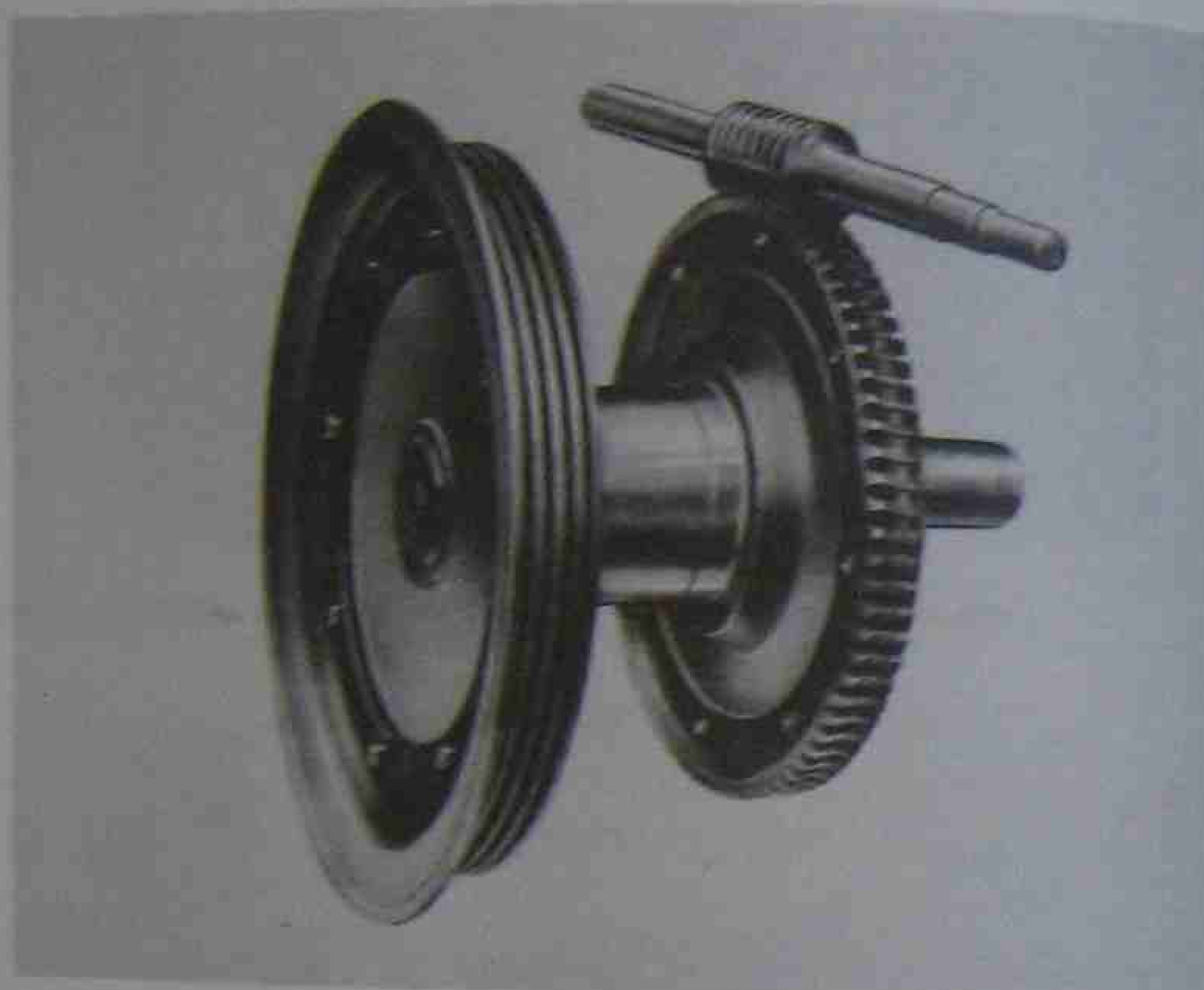


Fig. 4.4 — Worm wheel and traction sheave with a connecting spider (Wm Wadsworth & Sons, Ltd).

The tooth efficiency of worm gearing η_G may be expressed by equation (4.2) (excluding bearing and oil-churning losses)

$$\eta_G = \frac{\mathrm{tg}\lambda}{\mathrm{tg}(\lambda + \phi)}, \tag{4.2}$$

$$\mathrm{tg}\phi = \frac{\mu}{\cos\alpha_n},$$

where λ is the lead angle of the worm thread, μ is the coefficient of friction and α_n is the normal pressure angle.

Equation (4.2) is valid in the case of worm driving only. When the reversal of power transmission takes place (worm wheel driving), the tooth efficiency will be given by the equation:

$$\eta'_G = \frac{\mathrm{tg}(\lambda - \phi)}{\mathrm{tg}\lambda} \tag{4.3}$$

The coefficient of friction depends upon a number of factors, namely the material, surface finish, speed, the kind of the lubricant, tooth load, accuracy and rigidity of mounting. Experimentally determined values of μ are currently used, including the bearing losses on the worm and worm-wheel shafts, which are relatively small and difficult to separate.

In BS 721 $\mathrm{tg}\phi$ is depicted directly as dependent on the rubbing speed of the corresponding tooth surfaces (pressure angle 20 degrees), see Fig. 4.5. The values given on the chart are based on the use of phosphor bronze wheels and case hardened ground and polished steel worms, lubricated by a mineral oil having a viscosity of between (60–130) × 10^{-6} m²/s at 60°C. These materials working in

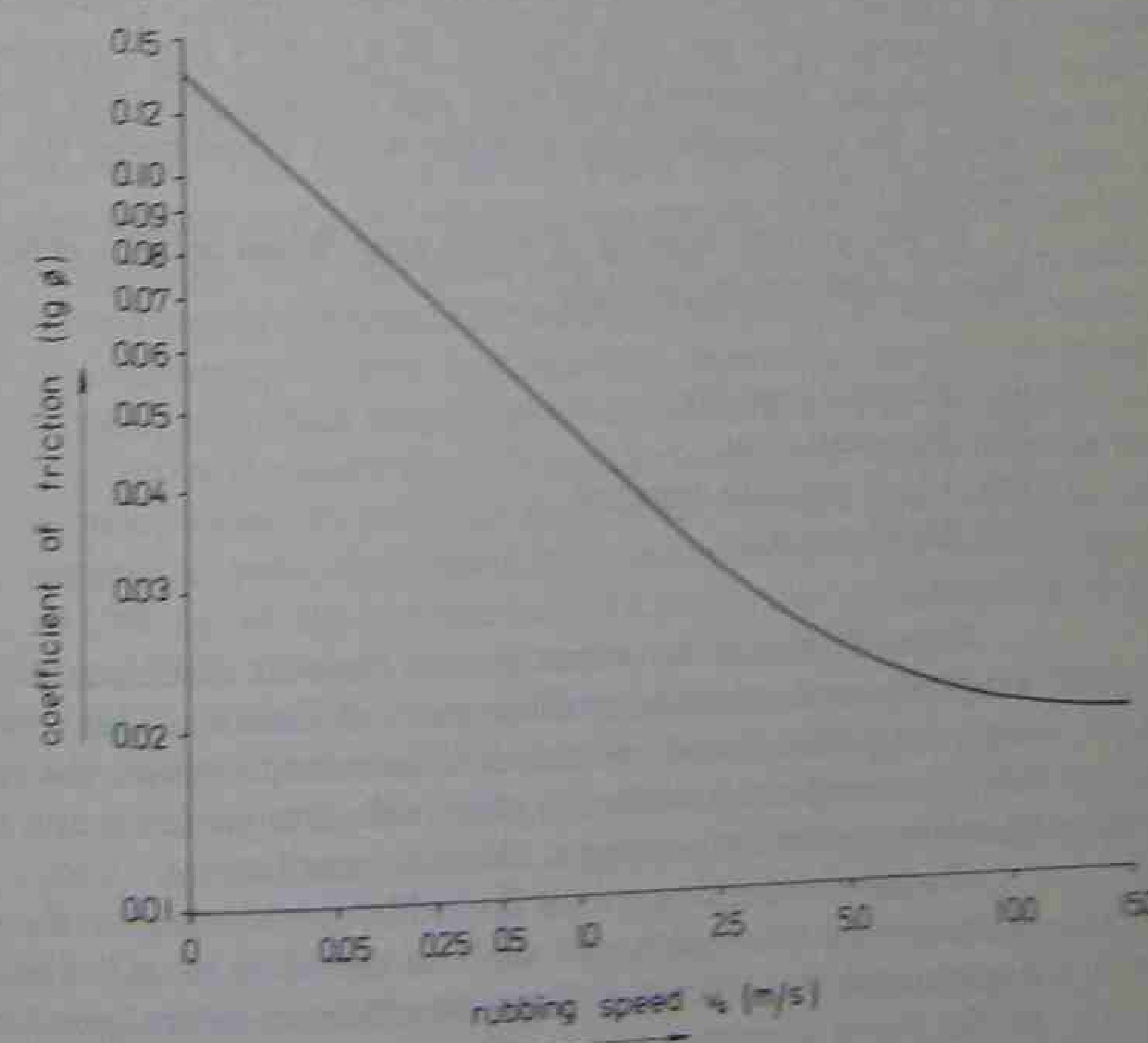


Fig. 4.5 — Diagram for coefficient of friction in relation to the rubbing speed of worm gearing.

conjunction have been found to be the best from the aspect of a low coefficient of friction; this combination also gives the greatest resistance to wear.

The rubbing speed is given by the formula

$$v_s = \frac{v_p}{\cos\lambda} \quad \text{(m/s)} \tag{4.4}$$

where v_p is circumferential velocity on worm reference diameter (m/s).

4.2 CALCULATION

There is no uniform method for rating the worm gearing specified by the US manufacturers, while BS 721: Worm Gearing is valid in Great Britain.

In compliance with BS 721 the permissible torque is limited either by consideration of surface stress or bending stress (wear or strength) in both the worm threads and worm-wheel teeth. Consequently the determination of the load capacity of a pair of gears involves four calculations, concerned with wear and strength of worm and worm wheel, the permissible torque on the worm wheel being the least of the four values.

In general the capacity of worm gear drives may be determined from several aspects, namely

(a) Thermal capacity.
(b) Wear capacity.
(c) Tooth strength capacity.
(d) Shock capacity.

Capacity ratings for worm geared elevator machines are based primarily on the thermal performance calculations. The wear capacity ratings should also be carried out in order to ensure the required life of worm gearing, while the gear-tooth strength is never a limiting factor for the worm gearing in elevator installations.

Choice of worm diameter is limited by considerations of strength, deflection of the high-speed shaft in the view of a correct teeth engagement and efficiency. Since the efficiency of the gear increases as the worm diameter is reduced the worm diameter should be as small as possible consistent with adequate strength at the root section of the worm and permissible deflection.

4.3 THERMAL PERFORMANCE CALCULATION

The temperature of the lubricant in the area of the engaging teeth is the limiting factor for the thermal capacity of gearing. If the temperature is too high, the oil film may fail at moderate tooth pressures, with the result that the structure of the metal may be affected by the progressive increase in temperature, causing disintegration of the bronze worm wheel. Since the efficiency of worm gearing is generally lower than with other types of gearing and the heat generated is directly proportional to the power loss the heat to be dissipated from a worm drive may be considerable.

The factors that influence the rate of heat dissipation from a worm gearbox are:

(1) The surface area of the gearbox
(2) The movement of the lubricant inside the box
(3) The motion of the air outside the box which may be forced by means of a fan fitted on the worm shaft.

The natural heat dissipation under static conditions is to some extent dependent on the design of the gear-case, but for gearboxes of similar design it is approximately proportional to the surface area of the case. The motion of the worm and worm wheel initiates a disturbance of the oil bath and consequently increases the rate of heat dissipation. The heat generated by the power loss has to be transferred mainly through the medium of the oil to the walls of the case, the efficiency of this heat transfer depending on the oil speed. Cooling efficiency may be increased considerably by the application of a fan on the worm shaft. It is important to bear in mind that the fan is more effective on large gears than on small ones owing to the greater air velocity attained at a given speed. For instance at a speed of 1000 r.p.m. the increase in heat capacity due to the fan varies from about 25% on small boxes to about 60% on large boxes.

In the USA *The Thermal Rating of Worm Gearboxes* by H. Walker, and AGMA Standard Practice 440.01 and 440.02 are utilized, while the calculation by Niemann (Niemann and Winter, 1983) is usually used as a part of the criteria in Europe. The principles of the latter calculation will be presented in this chapter.

The heat dissipated from a gearbox at the maximum permissible temperature rise Q must be greater than or at least equal to the power loss P_v, i.e.

$$Q \geqq P_v \tag{4.5}$$

The power loss P_v may be calculated from the equations

$$\eta_o = \frac{P_1 - P_v}{P_1} = \frac{P_2}{P_2 + P_v} \quad \text{for worm driving} \tag{4.6}$$

or

$$\eta_o' = \frac{P_1}{P_1 + P_v} = \frac{P_2 - P_v}{P_2} \quad \text{for worm wheel driving} \tag{4.7}$$

where η_o is overall efficiency of worm gearing including bearing and oil-churning losses for worm driving, and η_o' is overall efficiency of worm gearing for worm wheel driving. Index 1 refers to the high-speed shaft, index 2 to the low-speed shaft.

The heat dissipated from the gearbox at a stationary rate of heat transfer

$$Q = \Delta\theta \times S \times k \quad \text{(kJ/s)} \tag{4.8}$$

where $\Delta\theta$ is temperature increment (difference between the maximum permissible temperature of the outer surface of the gearbox and the temperature of the ambient air) (K), S is the outer surface of the gearbox (m^2) and k is the heat transfer coefficient (kJ)/($m^2 \times K \times s$).

The formula for the temperature increment $\Delta\theta$ is as follows:

$$\Delta\theta = \frac{\theta_L - \theta_a}{1.03 + 0.01\sqrt{0.1 n_1}} - 1.5 \quad (K) \tag{4.9}$$

where θ_L is maximum permissible temperature of oil in the gearbox (°C) (quoted by the oil companies); θ_a is the temperature of ambient air (°C) and n_1 is the r.p.m. of the worm.

The outer surface S may be calculated

$$S \approx 9 \times 10^{-5} \times C^{1.85} \quad (m^2) \tag{4.10}$$

for well designed gearboxes from the aspect of cooling (well arranged cooling ribs) or

$$S \approx 9 \times 10^{-5} \times C^{1.8} \quad (m^2) \tag{4.11}$$

for gearboxes of current design for equations 4.10 and 4.11. Where C is gear centre distance (mm).

The heat transfer coefficient:

$$k \approx 6.6 \times 10^{-3} \times \left[1 + 0.4\left(\frac{n_1}{60}\right)^{0.75}\right] \tag{4.12}$$

for under-driven worm-gears and a fan mounted on the worm shaft

$$k \approx 6.6 \times 10^{-3} \times \left[1 + 0.23\left(\frac{n_1}{60}\right)^{0.75}\right] \tag{4.13}$$

for under-driven worm-gears in the case of no fan being employed. The dimensions for equations 4.12 and 4.13 are given by: $kJ \times m^{-2} \times K^{-1} \times s^{-1}$.

With over-driven worm-gears the coefficient k is about 20% less than the values presented above. The worm wheel should be immersed in oil by some 30% of its diameter in this case.

In the event of the load and/or the speed being variable the equivalent output P_e is decisive for the worm gearing rating, given by the formula

$$P_e \approx \frac{P_1 \times t_1 + P_2 \times t_2 + \ldots}{t_1 + t_2 \ldots} \quad (kW)\ , \tag{4.14}$$

where P_1 is the output of the motor during the time period t_1, P_2 is the output of the motor during the time period t_2 etc.

Forces due to power interaction worm/worm wheel and tension in elevator ropes in the case of worm driving are depicted in Fig. 4.6. as well as the torques on both shafts. Individual forces are given by the following formulas:

$$\text{Tangential force} \quad F_t = \frac{2M_1}{D_1} \tag{4.15}$$

$$\text{Radial force} \quad F_r = \frac{F_t \times \text{tg}\,\alpha_n \times \cos\phi}{\sin(\lambda + \phi)} \tag{4.16}$$

$$\text{Axial thrust} \quad F_a = \frac{F_t}{\text{tg}(\lambda + \phi)} \tag{4.17}$$

where M_1 is torque on the worm (Nm), D_1 is worm reference diameter (m), α_n is the normal pressure angle, λ is the lead angle of the worm thread, and ϕ is the angle of friction.

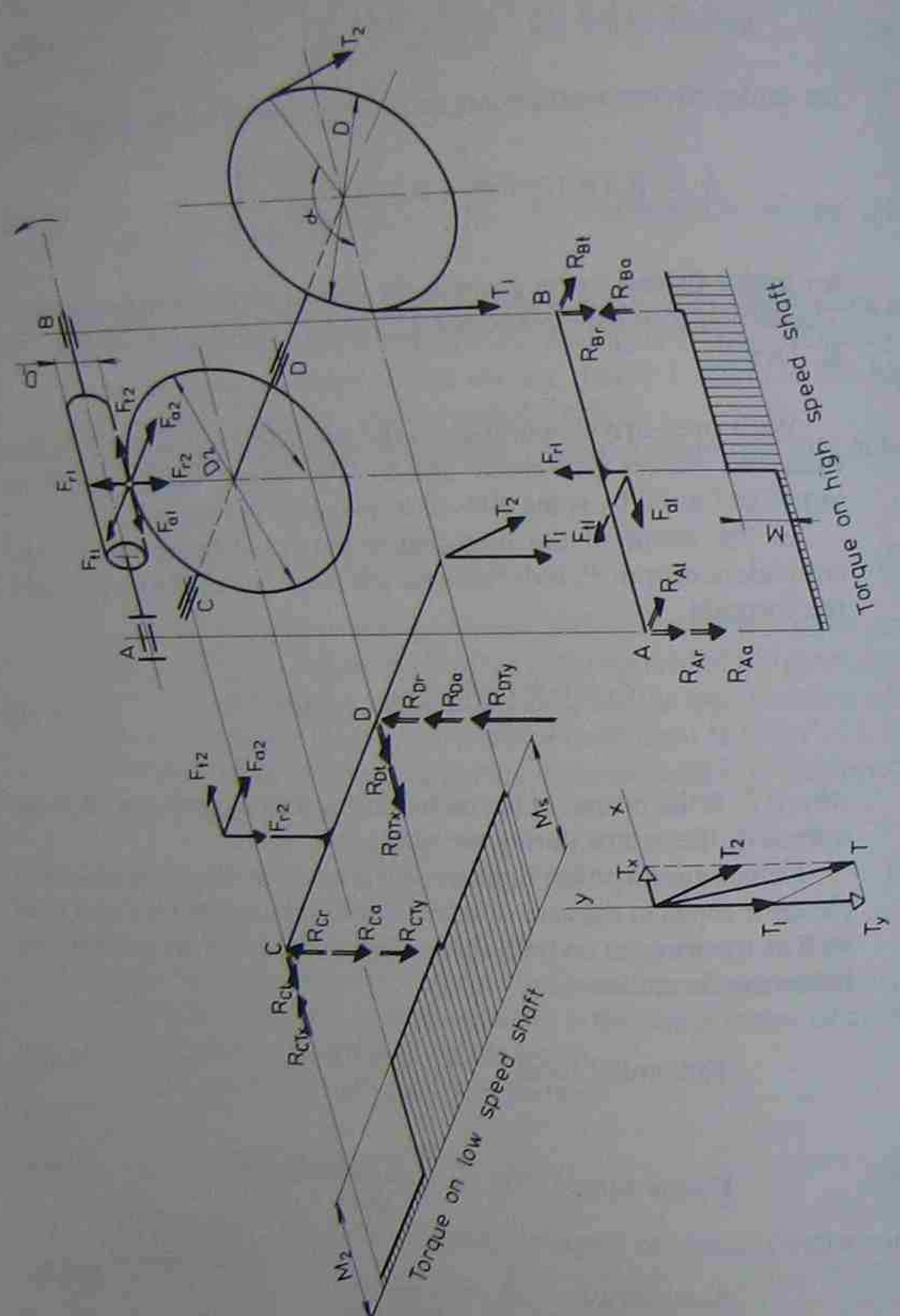


Fig. 4.6 — Forces acting upon worm gearing system with the worm driving.

$$M_1 = M_m \times \eta_L^2 = 9550\frac{P}{n_m} \times \eta_L^2 \quad \text{(Nm)} . \tag{4.18}$$

where P is the output of the driving motor (kW), n_m is the r.p.m. of the motor (1/min), η_L is the efficiency of one bearing.

The meaning of further symbols used in Fig. 4.6 is as follows:

M_2 is torque on the worm wheel (Nm)

$$M_2 = M_1 \times i_G \times \eta_G \quad \text{(Nm)} , \tag{4.19}$$

where i_G is gear ratio, η_G is the tooth efficiency of worm gearing and M_s is torque on the traction sheave

$$M_s = M_2 \times \eta_L^2 \times \eta_s = \frac{T_1 - T_2}{2} \times D \quad \text{(Nm)} , \tag{4.20}$$

where T_1, T_2 are tensions on either sides of the sheave (N), D is the pitch diameter of the sheave (m), and η_s is the efficiency of the sheave.

If the direction of rotation is reversed, a change in direction of the tangential forces and axial thrusts will take place causing a change in the configuration of reaction forces on the supports (bearings). In the event of the worm wheel driving, not only the tooth efficiency will be changed, but also formulae for F_a and F_r will be altered, their denominators being converted to $tg(\lambda - \phi)$ and $\sin(\lambda - \phi)$, respectively. A complete analysis should be done to find the most dangerous load for each component of the system.

5

Brakes

5.1 TYPES OF BRAKES

The elevator braking system, which must be set in operation automatically in the event of loss of power supply and/or loss of supply to the control circuits, must be provided with an electromechanical friction brake. This brake must be capable of stopping the machine when the car with the 125% of rated load is travelling at its rated speed and hold the system at rest afterwards. The retardation must not be in excess of that resulting from the operation of the safety gear or by stopping the car on its buffers.

The brake is usually mounted on the high-speed shaft (motor shaft), because of the braking torque being relatively small here, provided that the shaft is coupled to the sheave (drum, sprockets) by direct mechanical means. With indirect-drive machines, utilizing V-belts, toothed drive belts or drive chains, the brake must be located on the traction sheave (drum) assembly side of the machine so as to be fully effective in the event of the belt set or chain set failure.

The brake must be applied by compression springs or by gravity. It can be released either electromagnetically or electrohydraulically. The interruption of the current must be controlled by at least two independent electric devices. Braking should occur as the electric circuit operating the brake is interrupted.

When the machine is fitted with a manual emergency operating device, the brake must be so designed to enable releasing by hand and constant effort must be exerted to keep the brake open in such a case.

The most common form of elevator brake is an electromagnetic brake, consisting of a spring assembly, brake shoes with linings and a magnet assembly. The release of the brake is effected by energizing

the solenoid; when de-energized the brake shoes grip the brake drum under the influence of the compression spring and induce the braking torque.

Elevator brakes are mostly of external contracting type, equipped with two shoes, but occasionally brakes of internal expanding type occur with gearless machines of large dimensions. Band brakes are not allowed to be used with elevators and the application of disc brakes is rare.

The brake shoes may be either fixed on the operating arms or pivoted with self-aligning effect. Brakes of the first design are simple, but precise adjustment and frequent inspection is necessary, to ensure the specific pressure is evenly distributed in the contact area between the brake shoes and the drum to avoid uneven wear of the brake linings. With brakes of the latter design this is easier to achieve, but they are of more complicated construction. Friction spring devices must be fitted to the brake shoes to prevent 'trailing' on the brake drum.

The magnet may be mounted directly on one operating arm and exert a horizontal force for the brake release, or located in vertical position and act upon the arms through a system of linkages.

The brake shoes are provided with linings of high coefficient of friction, which should be well bedded in; they are usually fastened by means of countersunk-head rivets, made of brass or copper. The lining is mostly made of woven fibre for light-duty application and of zinc bonded asbestos for medium and heavy-duty brakes. The dimensions should be sufficient enough to decrease the specific pressure and minimize wear. Although it is desirable that the coefficient of friction of the lining is high it should not be too high as the final stopping might be jerky. This may be avoided by using the zinc bonded asbestos lining in which the zinc acts as a permanent dry lubricant and reduces brake drag to a minimum.

The fulcrum pins are made of bronze for smaller brakes and greased for long-life lubrication, while they are of steel in self-lubricated bronze bearings for larger models.

In Fig. 5.1 an elevator brake, designed for operation from d.c. or rectified a.c. supplies, is shown. This type is currently provided with operating coils rated for half hour duty, but may be used for one hour duty or continuous service with reduced braking torque. Base and operating arms are of high grade cast-iron. The brake linings are mounted directly on operating arms; the spring is located below the centres of rotation of the arms. Adjustment of the braking torque is easily effected by means of a single screw situated at the bottom end

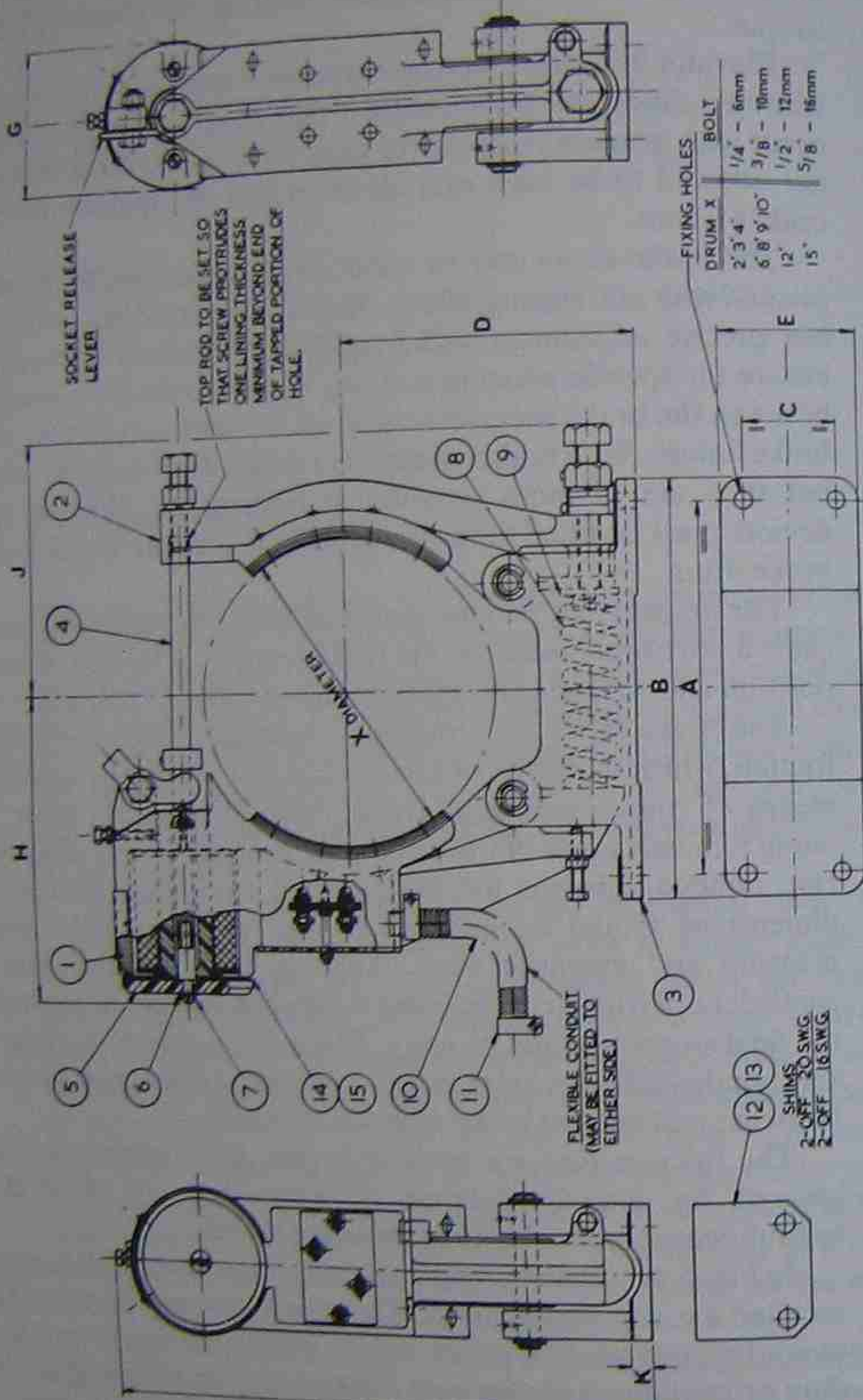


Fig. 5.1 — Electromagnetic d.c. brake (Dewhurst plc).

of the operating arm, by which the compressive force of the spring can be preset. The d.c. magnets are extremely quiet in operation and are therefore used where silence is essential, for example for passenger elevators.

Another d.c. brake of rather different construction is shown in Fig. 5.2. The braking force is induced by two compression springs and

Fig. 5.2 — Electromagnetic d.c. brake (Moline Accessories Co.).

the magnet is located between the extended upper ends of the operating arms. The linings are mounted directly on operating arms.

Both brakes are of simple construction with minimum parts which makes them easy to maintain. Original clearances between the brake linings and the drum are close so that rapid setting is easily accomplished.

5.2 CALCULATION OF BRAKING TORQUE. BRAKE SELECTION

In compliance with EN 81: Part 1 the braking torque must be sufficient to safely stop a car with a load equivalent to 125% of the rated load and hold it at rest afterwards. The torque is composed of two parts: the static component necessary to hold the system at rest and the dynamic component for absorbing the kinetic energy of all moving parts of the system. The braking torque will be calculated in the event of the car stopping at the lowest landing as this condition is decisive for braking. A diagram of the system is shown in Fig. 5.3;

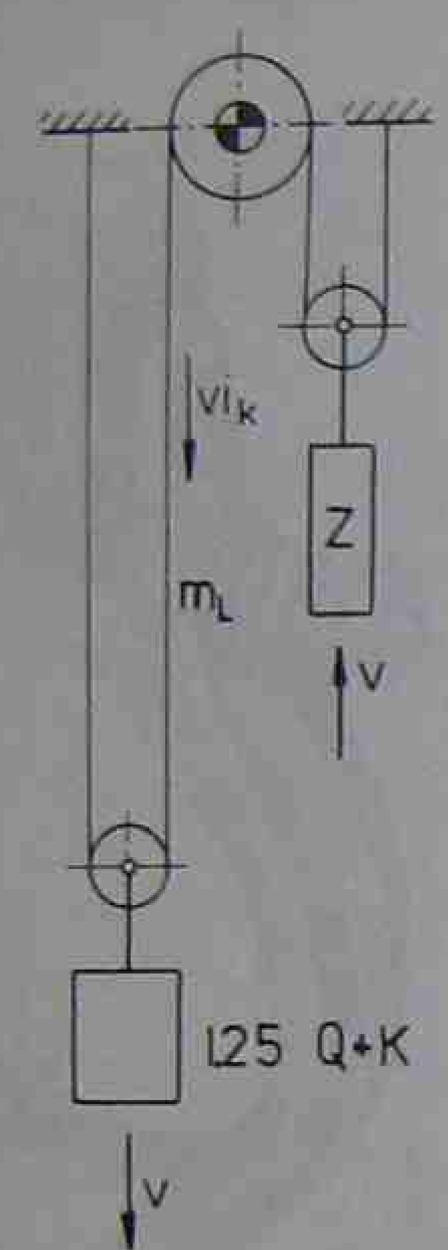


Fig. 5.3 — Diagram for calculation of the braking torque.

although it is drawn for the roping factor $i = 2$, a general calculation will be carried out.

Calculation of the static torque M_{st}

M_{st} is given by the formula:

$$M_{st} = \left(\frac{1.25Q + K - Z}{i} + m_L\right) \times g_n \times \frac{D}{2i_G} \times \eta_2 \quad \text{(Nm)} \; , \tag{5.1}$$

where Q is rated load (kg), K is mass of the car (kg), Z is mass of the counterweight (kg), i is the roping factor, m_L is the mass of one fall of suspension ropes (kg), g_n is the acceleration of free fall (m/s^2), D is

the sheave diameter (m), i_G is the gearing ratio, and η_2 is the mechanical efficiency of the system related to the conditions of braking. It is possible to calculate η_2 as a product.

$$\eta_2 = \eta_{RS} \times \eta_s \times \eta'_G \; , \tag{5.2}$$

where η_{RS} is efficiency of the roping system, η_s is efficiency of the sheave, and η'_G is efficiency of mechanical gearing between the motor and the sheave for reversed power transmission. It is necessary to consider the direction of power transmission especially with worm gearing application as its efficiency is affected by this factor to a large extent (see section 4.2).

Calculation of the dynamic torque M_i

$$M_i = I \times \varepsilon \quad \text{(Nm)} \; , \tag{5.3}$$

where I is the moment of inertia of all moving parts of the system referred to the high-speed shaft (brake drum shaft) (kgm^2), and ε is the angular retardation of the high-speed shaft (1/s^2).

The total moment of inertia I may be calculated as

$$I = I_1 + I_2 + I_3 \; , \tag{5.4}$$

where I_1 is the moment of inertia of the rotor, brake drum and worm (kgm^2), I_2 is the moment of inertia of the worm wheel and sheave (kgm^2), and I_3 is the moment of inertia of all parts of the system in linear motion (kgm^2).

If the moment of inertia of the worm wheel and sheave (${}^1I_2$), referred to their axis of rotation, is known, the transmission to the high-speed shaft can be easily carried out on the basis of the conservation of kinetic energy

$$I_2 = {}^1I_2 \times \frac{\eta'_G}{{i_G}^2} \tag{5.5}$$

The same principle in calculating I_3 can be applied. The moment of inertia (${}^1I_3$) referred to the low-speed shaft (traction sheave shaft) will be calculated by utilizing the following equation, expressing the equality of the translational and rotational energies.

$$\frac{1}{2}\,{}^1I_3 \times \omega^2 = \frac{1}{2}[(1.25Q + K + Z) \times v^2 + m_L \times (i \times v)^2] \times \times \eta_{RS} \times \eta_s \; , \tag{5.6}$$

where ω is the angular velocity of the low speed shaft (1/s)

$$\omega = \frac{2i \times v}{D}$$

v is the levelling speed of the car and counterweight (m/s). Hence

$${}^1I_3 = (1.25Q + K + Z + m_L \times i^2) \times \frac{D^2}{4i^2} \times \eta_{RS} \times \eta_s \qquad (5.7)$$

and

$$I_3 = {}^1I_3 \times \frac{\eta'_G}{i_G{}^2} \qquad (5.8)$$

Having substituted for ${}^1I_3$ from the preceding equation the final formula for I_3 is obtained:

$$I_3 = (1.25Q + K + Z + m_L \times i^2) \times \frac{D^2}{4i^2 \times i_G{}^2} \times \eta_2 \qquad (5.9)$$

Since the braking force is constant with a friction brake the motion during the braking period is uniformly retarded and the angular retardation may be calculated as follows:

$$\varepsilon = \frac{\pi \times n_2}{30 t_b} \quad (1/s^2) \ , \qquad (5.10)$$

where n_2 are revolutions per minute of the motor at the initial instant of the braking (1/min) and t_b is the braking time (s).

If the braking time be specified the braking torque required for a safe operation may be determined from the equation

$$M_b = M_{st} + M_i \quad (\text{Nm}) \ . \qquad (5.11)$$

The specification of the braking time requires a certain experience. Kinematic equations for the uniformly retarded motion may be of use in this respect and facilitate the estimation.

If l be the distance of the car from the destination when the brake is set in operation and v the initial car velocity, then:

$$t_b = \frac{2l}{v} \quad (s) \ . \qquad (5.12)$$

Furthermore, if a is the rate of retardation (m/s^2), then

$$t_b = \frac{v}{a} \quad (s) \ . \qquad (5.13)$$

It may be easier to assess the rate of retardation and then calculate the braking time. The rate of retardation should not be specified at too large a value, bearing in mind the braking conditions; it will be much higher when a car with the same load stops at the highest landing as in this case the static torque will contribute to the braking according to the equation

$$M_b = -M_{st} + M_i \ . \qquad (5.14)$$

After the selection of the brake has been made, it is advisable to calculate the rate of retardation in the case of a car stopping at the highest landing to make sure that the braking torque is not excessive and the passengers will not be subjected to physical discomfort.

Selection of the brake

The selection of a brake for a given application is related to the braking torque to be applied to the drum and the thermal capacity of both the brake and the drum.

The kinetic energy of all elevator moving parts, converted into heat on the brake drum in unit time, must be dissipated to prevent the temperature of the lining from becoming excessive.

The heat A generated by friction between the brake shoes and the drum per hour is given by the formula

$$A = M_b \times \omega_a \times t_b \times z \times 10^{-3} \quad (\text{kJ/h}) \ , \qquad (5.15)$$

where ω_a is the average angular velocity during the braking time (1/s),

$$\omega_a = \frac{\pi \times n}{60}$$

where n are revolutions per minute of the motor at the beginning of the brake action (1/min), t_b is average braking time (s), and z is the number of operations per hour (1/h).

The actual number of operations per hour corresponds to the number of car stops and is therefore dependent upon the elevator operating conditions. An analysis should be done of the peak hours to determine the traffic pattern and consequently the maximum number of stops per hour, the average load and the average braking time. The correct selection of the brake is conditioned by the knowledge of these parameters. In manufacturers' catalogues the number of operations per hour is specified in addition to the braking torque. With most elevator installations the thermal calculation of the brake is not necessary. Nowadays electrical braking to almost zero speed is applied practically with all intensive traffic passenger elevators. With light duty passenger elevators and freight elevators the number of stops per hour is relatively low and consequently the kinetic energy absorbed within one hour is also likely to be low. However, where the levelling speed is relatively high or large moving parts are involved the thermal performance should be analysed and a complete calculation carried out.

Example 5.1

Determine the braking torque of an electromagnetic brake for a passenger elevator of the following parameters:

Rated load	$Q = 630\,\text{kg}$
Mass of the car	$K = 737\,\text{kg}$
Mass of counterweight	$Z = 1020\,\text{kg}$
Rated speed	$v = 1.6\,\text{m/s}$
Height of travel	$H = 33.6\,\text{m}$
R.p.m. of the motor	$n = 1500\,\text{1/min}$
Moment of inertia of the motor	$I_m = 0.205\,\text{kg m}^2$
Traction sheave diameter	$D = 610\,\text{mm}$

Roping system 1:1, no compensating ropes or chains employed.

Provided that electric braking is applied to almost zero speed, the mechanical braking torque under normal operating conditions will be approximately equal to the static torque — see equation (5.1). However, in the event of the power supply being interrupted at the instant the car approaches the lowest landing level with the load equivalent to 125% of the rated value, the brake must be capable of stopping the car safely and to prevent it from overrunning the destination landing. Both the static and dynamic torques must be taken into consideration.

The static torque

$$M_{st} = (1.25Q + K - Z) \times g_n \times \frac{D}{2i_G} \times \eta_2 \quad (\text{Nm}) , \qquad (5.1a)$$

where gear ratio i_G,

$$i_G = \frac{n}{n_s} = \frac{1500}{50.09} = 29.943$$

r.p.m. of the sheave n_s,

$$n_s = \frac{60v}{\pi \times D} = 50.09\ \text{1/min}$$

and the mechanical efficiency of the system — see equation (5.2)

$$\eta_2 = \eta_s \times \eta'_G = 0.96 \times 0.85 = 0.816.$$

For the purpose of this calculation let us estimate the partial efficiencies $\eta_s = 0.96$, $\eta'_G = 0.85$.

$$M_{st} = (1.25 \times 630 + 737 - 1020) \times 9.81 \times \frac{0.61}{2 \times 29.943} \times 0.816 = 41.13\ \text{Nm}.$$

The dynamic torque — see equation (5.3)

$$M_i = I \times \varepsilon \quad (\text{Nm}) ,$$

where the total moment of inertia of the system — see equation (5.4)

$$I = I_1 + I_2 + I_3 .$$

Let us assess the moment of inertia of the brake and the worm $I_b = 0.2\,\text{kgm}^2$. Then

$$I_1 = I_m + I_b = 0.205 + 0.2 = 0.405\,\text{kgm}^2 .$$

Because of large gear ratio I_2 is relatively small (usually up to $0.3I_1$); assume $I_2 = 0.2I_1$.

By equation (5.9)

$$I_3 = (1.25Q + K + Z) \times \frac{D^2}{4i_G^2} \times \eta_2$$

$$= (1.25 \times 630 + 737 + 1020) \times \frac{0.61^2}{4 \times 29.943^2} \times 0.816 =$$
$$= 0.2154 \text{ kgm}^2$$
$$I = 0.405 \times 1.2 + 0.2154 = 0.7014 \text{ kgm}^2$$

Angular retardation

$$\varepsilon = \frac{\pi \times n}{30 t_b} \quad (1/s^2) \ . \tag{5.10}$$

The braking time t_b can be calculated on the basis of the rate of retardation a. The rate of retardation should be specified in relation to the braking distance under normal operating conditions, otherwise the car might overrun the destination level in the case of a failure of the power supply taking place when the car is at the point where electric braking should commence. In the speed to time diagram (Fig. 5.4) curve 1 depicts the speed to time relation when electric braking is

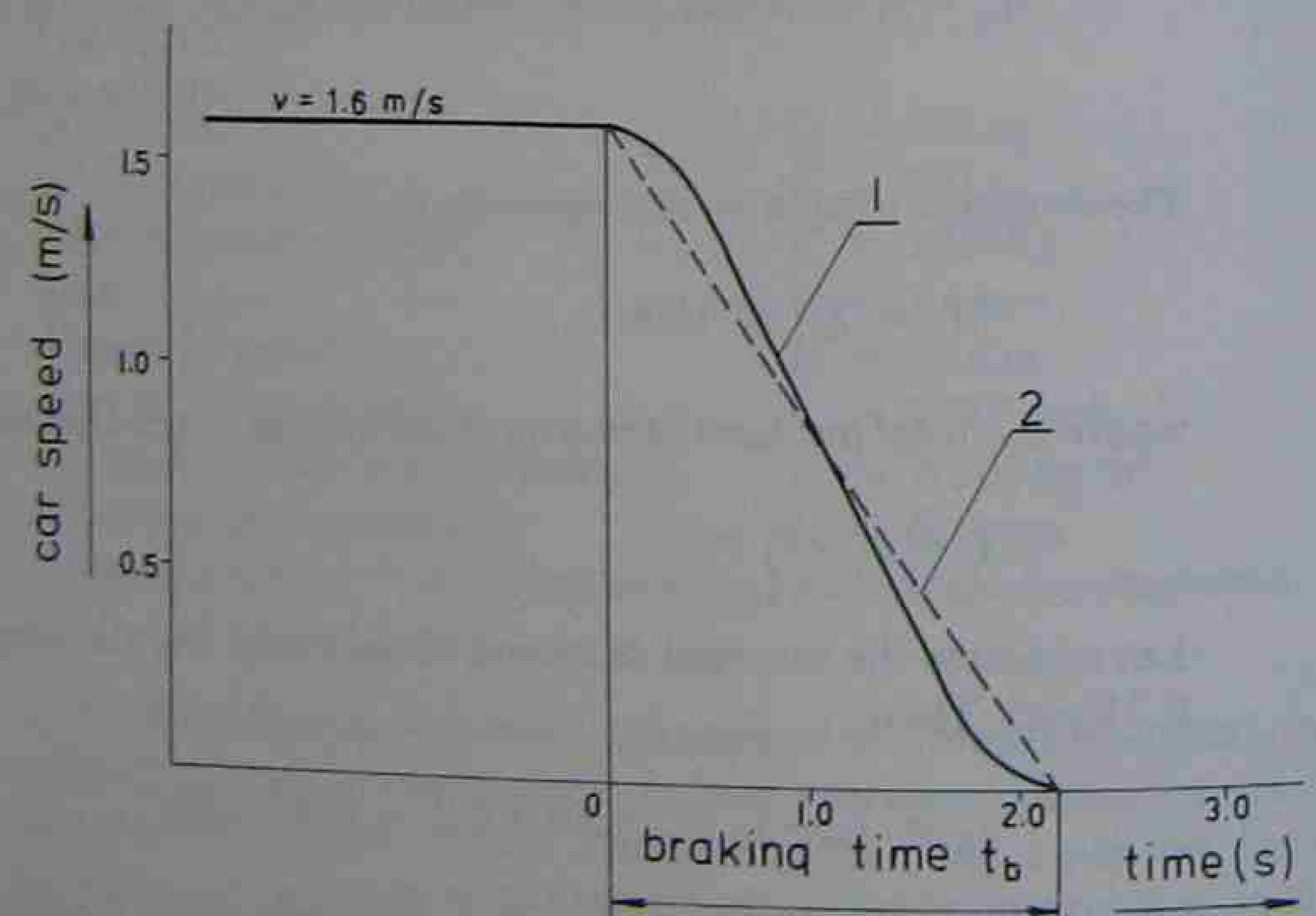


Fig. 5.4 — Diagram for determination of the rate of retardation.

applied, whilst the dashed line 2 corresponds to the mechanical brake operation of equal duration. As seen in Fig. 5.4, a represents the

average rate of retardation due to the electric braking.

Assume, that $a = 0.75 \text{ m/s}^2$.

$$t_b = \frac{v}{a} = \frac{1.6}{0.75} = 2.133 \text{ s} \ .$$

Then

$$\varepsilon = \frac{\pi \times 1500}{30 \times 2.133} = 73.642 \ 1/s^2$$

and

$$M_i = 0.7014 \times 73.642 = 51.65 \text{ Nm} \ .$$

The total braking moment required is

$$M_b = M_{st} + M_i = 41.13 + 51.65 = 92.78 \text{ Nm} \ .$$

A certain inaccuracy has occurred in the calculation as 1500 r.p.m. was considered to be the initial value for the braking period. Since the motor was working as a generator prior to the brake application its speed would be slightly in excess of the synchronous value, but this difference is not of prime significance and could not affect the brake selection.

The actual braking torque should be as close to the calculated value as possible as the retardation of the car when it is moving in an upward motion prior to the braking action, might be excessive. Assuming that the actual braking torque is 95 Nm then it is possible to calculate the rate of retardation in the event of power failure taking place when the car is stopping at the highest landing level.

$$M_b = M_i - M_{st}$$

$$95 = 0.7014 \times \frac{\pi \times 1500}{30 \times t_b} - 41.13$$

$$t_b = 0.8093 \text{ s}$$

$$a = \frac{v}{t_b} = \frac{1.6}{0.8093} = 1.977 \text{ m/s}^2$$

The rate of retardation may be considered rather high, but it may be taken for granted as the brake will stop the car running at the rated speed in the case of power failure only.

Again, the r.p.m. of the high-speed shaft is different from 1500 being slightly lower in this case.

6

Counterweight

The counterweight is used with traction and chain drive elevators for balancing the mass of the car and a portion of the rated load, this portion being usually 45 to 50%. With high rise installations the mass of the counterweight for ideal balancing is also affected by the mass of the travelling cable.

In relation with Fig. 6.1 expressing the ideal balancing of both sides of the traction sheave the equation of equilibrium can be formulated:

$$\begin{aligned}(K+\psi\times Q)\times g_n+(H-z)\times q_L\times g_n+z\times q_k\times g_n+y\times{}_{e}\times g_n\\ = Z\times g_n+z\times q_L\times g_n+(H-z)\times q_k\times g_n\ ,\end{aligned} \qquad (6.1)$$

where Z is the mass of the counterweight (kg), ψ is the coefficient taking account of the percentual portion of the rated load balanced by the counterweight, H is the car travel (m), q_L is the unit weight of suspension ropes (kg/m), q_k is the unit weight of compensating cables (kg/m), q_e is the unit weight of travelling cable (kg/m), y is the variable length of the travelling cable under the car (m), z is the variable distance from the car to its lowest level (m), g_n is the standard acceleration of free fall (m/s^2).

The relation between y and z may be easily expressed, neglecting the fact that the length of the travelling cable is greater by a few metres than $H/2$, which makes little difference with high rise installations:

$$y = \frac{z}{2}\ .$$

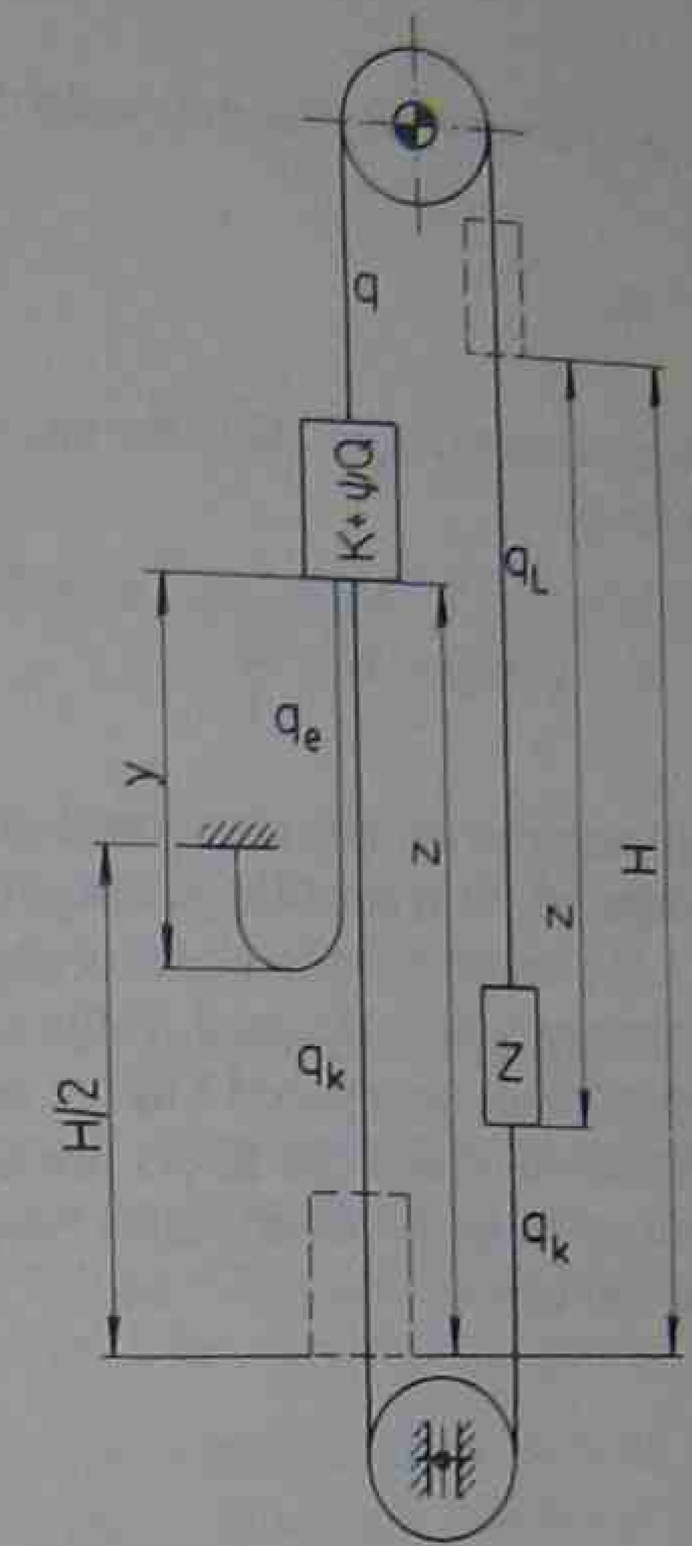


Fig. 6.1 — Diagram for calculation of the counterweight.

After the substitution in (6.1) has been carried out:

$$\begin{aligned}K+\psi\times Q+H\times q_L+z\times\left(q_k-q_L+\frac{q_e}{2}\right)\\ = Z+H\times q_k+z\times(q_L-q_k)\end{aligned} \qquad (6.2)$$

Equation (6.2) must be satisfied for any car position, i.e. for any z. Therefore it is valid to equate the independent members as well as to equate those members containing the independent variable z i.e.:

$$K+\psi\times Q+H\times q_L = Z+H\times q_k \qquad (6.3)$$

$$q_k-q_L+\frac{q_e}{2} = q_L-q_k \qquad (6.4)$$

Equation (6.4) determines the unit weight of compensating cables

$$q_k = q_L - \frac{q_c}{4} \tag{6.5}$$

Having substituted for q_k in (6.3) the correct mass of the counterweight is obtained:

$$Z = K + \psi \times Q + H \times \frac{q_c}{4} \tag{6.6}$$

The counterweight is usually composed of a steel frame (in which the fillers are secured) by eye bolts passing through the top member of the frame for the suspension to elevator ropes and guide shoes for guiding the counterweight in the well. If the fillers are made of metal and the rated speed does not exceed 1 m/s, a minimum of two tie-rods may be employed, on which the fillers are secured. The fillers are mostly cast-iron sections or steel plates, sometimes prefabricated concrete blocks are used.

7

Guiding the Car and Counterweight

7.1 SHAPE, MATERIAL AND JOINTING OF GUIDE-RAILS. GUIDE-RAIL BRACKETS

The functions of the guide-rails are as follows:

(1) To guide the car and counterweight in their vertical travel and to minimize their horizontal movement.
(2) To prevent tilting of the car due to eccentric load.
(3) To stop and hold the car on the application of the safety gear.

The car and counterweight must be both guided by at least two rigid steel guide-rails, which are usually cold drawn T-sections of either 370 N/mm^2 grade or 520 N/mm^2 grade steel. In the USA a suitable non-metallic material may be used for guide-rails where steel may present an accident hazard, as in chemical or explosive plants, provided the rated speed of the car does not exceed 0.76 m/s. In the past few years round guide-rails have been successfully used for hydraulic elevators and counterweights without safety gears.

In the USA T-section guide-rails must conform to the nominal weights and dimensions shown in Fig. 7.1 and Table 7.1. Both

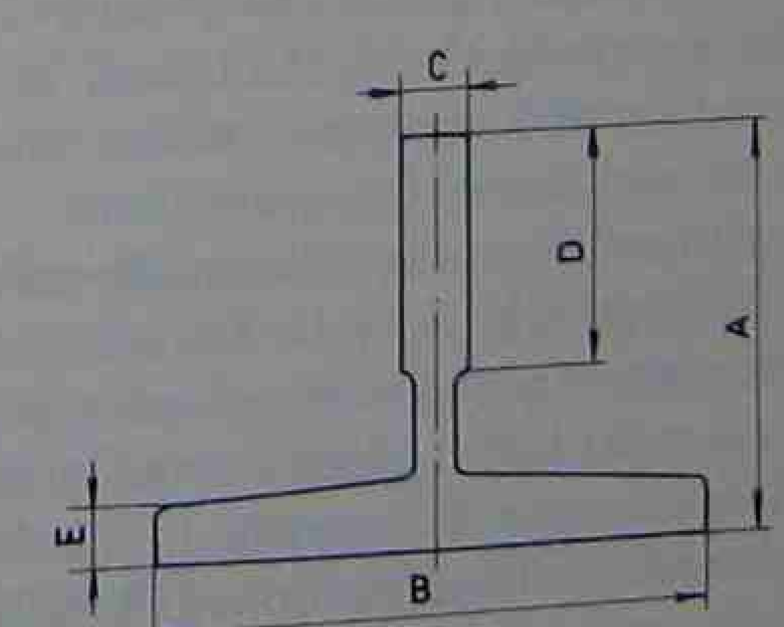


Fig. 7.1 — T-section guide-rail.

nominal weights and dimensions are quoted in pounds per foot and inches respectively in conformity with the American practice in guide-rail designation and can be easily converted to SI units (see note below Table 7.1).

Table 7.1 — T-section guide-rail dimensions

Nominal weight (lb/ft)	Nominal dimensions (inches)				
	A	B	C	D	E
8	$2\frac{7}{16}$	$3\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{4}$	$\frac{5}{16}$
11	$3\frac{1}{2}$	$4\frac{1}{2}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{5}{16}$
12	$3\frac{1}{2}$	5	$\frac{5}{8}$	$1\frac{3}{4}$	$\frac{5}{16}$
15	$3\frac{1}{2}$	5	$\frac{5}{8}$	$1\frac{31}{32}$	$\frac{1}{2}$
18.5	$4\frac{1}{4}$	$5\frac{1}{2}$	$\frac{3}{4}$	$1\frac{31}{32}$	$\frac{1}{2}$
22.5	4	$5\frac{1}{2}$	$1\frac{1}{8}$	2	$\frac{9}{16}$
30	5	$5\frac{1}{2}$	$1\frac{1}{4}$	$2\frac{1}{4}$	$\frac{11}{16}$

Note: 1 lb/ft=1.49 kg/m; 1 in.=25.4 mm.

Guide-rail surfaces used for guiding the car and counterweight should be sufficiently smooth to facilitate a proper operation of the guiding members. If the guides are not made of drawn steel, the rubbing surfaces must be machined for rated speeds exceeding 0.4 m/s.

It is of prime importance that the guide-rails are plumb and that the distance between them is maintained equal along their full length as uneven guide-rails, improper installation and/or rough surfaces may cause vibrations of the car and/or counterweight and may create dynamic forces variable during the travel. These forces would result in the fluctuation of frictional resistance on guide-rails and consequently in the variation of the motor load.

The design and construction of guide-rails joints are specified in detail in ANSI/ASME A 17.1. The ends of the guide rails are precisely machined with a tongue and matching groove centrally located in the web. Also the backs of the guide flanges are machined to form a flat surface for the fishplate, to which the ends of each guide-rail are fastened by at least four bolts. The width of the fishplate must not be less than the width of the guide-rail. An example of guide-rail jointing is shown in Fig. 7.2.

Fig. 7.2 — Guide-rail jointing.

The layout of a guide-rail in the well, as usually accomplished, is shown in Fig. 7.3. The guide rail is supported at the bottom end in the pit and the guide-rail brackets are positioned at regular distances from each other along the rail. Hinged guide-rails are very rare at present; they were sometimes used with small capacity elevators.

The brackets, their fastenings and supports, such as building beams and walls, must be capable of resisting the horizonal forces imposed due to uneven loading of the car with a total deflection limited to a value that will not affect the normal operation of the elevator. In the USA the total deflection at the point of support must not be in excess of 3.2 mm. Furthermore, they must safely withstand the application of the car or counterweight safety gear.

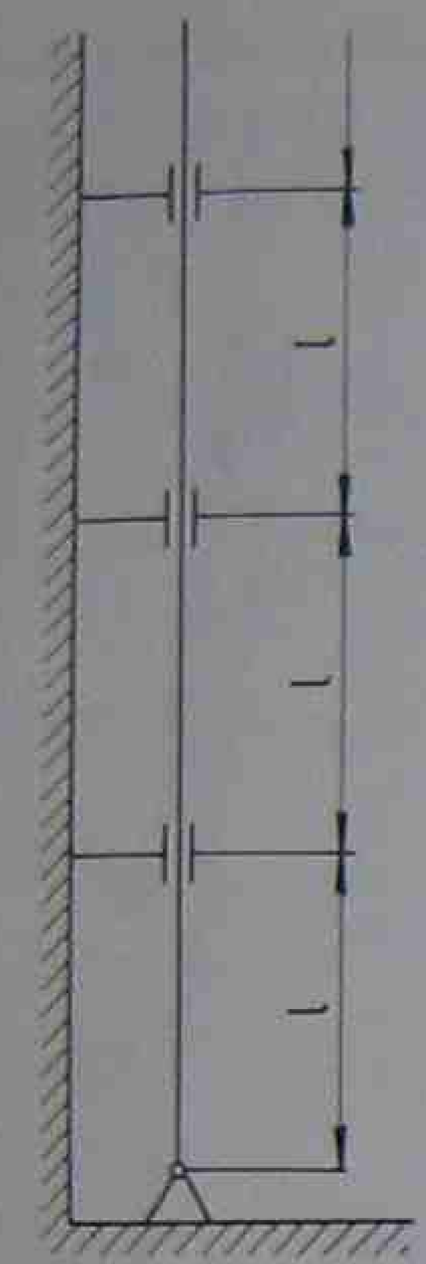


Fig. 7.3 — Arrangement of a guide-rail in the well.

7.2 FORCES ON GUIDE-RAILS AND STRESS ANALYSIS

In calculating the guide-rails three operating conditions should be taken into consideration:

(1) Running conditions with the load unevenly distributed on the car floor.
(2) Safety gear operation.
(3) Loading or unloading respectively.

In compliance with EN 81.1 and BS 5655 the evaluation of the stress in the guide rails is carried out for (2), while the calculation of deflection concerns quite different operating conditions, namely (1).

The stress in the guide rails during the safety gear operation is calculated in case of buckling and is given by a simple formula

$$\sigma_k = \frac{F_b \times \omega}{S} \ (\text{N/mm}^2) \qquad (7.1)$$

where F_b is the braking force due to the safety gear operation (N), S is the cross-sectional area of the guide-rail (mm^2) and ω is the buckling factor, see Tables 7.2 and 7.3 for steels of 370 N/mm^2 grade or 520 N/mm^2 grade respectively, as a function of the coefficient of slenderness λ.

Table 7.2 — Buckling factor ω for steel of 370 N/mm^2 grade

λ	0	1	2	3	4	5	6	7	8	9	λ
20	1.04	1.04	1.04	1.05	1.05	1.06	1.06	1.07	1.07	1.08	20
30	1.08	1.09	1.09	1.10	1.10	1.11	1.11	1.12	1.13	1.13	30
40	1.14	1.14	1.15	1.16	1.16	1.17	1.18	1.19	1.19	1.20	40
50	1.21	1.22	1.23	1.23	1.24	1.25	1.26	1.27	1.28	1.29	50
60	1.30	1.31	1.32	1.33	1.34	1.35	1.36	1.37	1.39	1.40	60
70	1.41	1.42	1.44	1.45	1.46	1.48	1.49	1.50	1.52	1.53	70
80	1.55	1.56	1.58	1.59	1.61	1.62	1.64	1.66	1.68	1.69	80
90	1.71	1.73	1.74	1.76	1.78	1.80	1.82	1.84	1.86	1.88	90
100	1.90	1.92	1.94	1.96	1.98	2.00	2.02	2.05	2.07	2.09	100
110	2.11	2.14	2.16	2.18	2.21	2.23	2.27	2.31	2.35	2.39	110
120	2.43	2.47	2.51	2.55	2.60	2.64	2.68	2.72	2.77	2.81	120
130	2.85	2.90	2.94	2.99	3.03	3.08	3.12	3.17	3.22	3.26	130
140	3.31	3.36	3.41	3.45	3.50	3.55	3.60	3.65	3.70	3.75	140
150	3.80	3.85	3.90	3.95	4.00	4.06	4.11	4.16	4.22	4.27	150
160	4.32	4.38	4.43	4.49	4.54	4.60	4.65	4.71	4.77	4.82	160
170	4.88	4.94	5.00	5.05	5.11	5.17	5.23	5.29	5.35	5.41	170
180	5.47	5.53	5.59	5.66	5.72	5.78	5.84	5.91	4.97	6.03	180
190	6.10	6.16	6.23	6.29	6.36	6.42	6.49	6.55	6.62	6.69	190
200	6.75	6.82	6.89	6.96	7.03	7.10	7.17	7.24	7.31	7.38	200
210	7.45	7.52	7.59	7.66	7.73	7.81	7.88	7.95	8.03	8.10	220
220	8.17	8.25	8.32	8.40	8.47	8.55	8.63	8.70	8.78	8.86	220
230	8.93	9.01	9.09	9.17	9.25	9.33	9.41	9.49	9.57	9.65	230
240	9.73	9.81	9.89	9.97	10.05	10.14	10.22	10.30	10.39	10.47	240
250	10.55										

For steel qualities with intermediate strengths, linear interpolation may be applied to determine the value of ω.

Table 7.3— Buckling factor ω for steel of 520 N/mm² grade

λ	0	1	2	3	4	5	6	7	8	9	λ
20	1.06	1.06	1.07	1.07	1.08	1.08	1.09	1.09	1.10	1.11	20
30	1.11	1.12	1.12	1.13	1.14	1.15	1.15	1.16	1.17	1.18	30
40	1.19	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27	40
50	1.28	1.30	1.31	1.32	1.33	1.35	1.36	1.37	1.39	1.40	50
60	1.41	1.43	1.44	1.46	1.48	1.49	1.51	1.53	1.54	1.56	60
70	1.58	1.60	1.62	1.64	1.66	1.68	1.70	1.72	1.74	1.77	70
80	1.79	1.81	1.83	1.86	1.88	1.91	1.93	1.95	1.98	2.01	80
90	2.05	2.10	2.14	2.19	2.24	2.29	2.33	2.38	2.43	2.48	90
100	2.53	2.58	2.64	2.69	2.74	2.79	2.85	2.90	2.95	3.01	100
110	3.06	3.12	3.18	3.23	3.29	3.35	3.41	3.47	3.53	3.59	110
120	3.65	3.71	3.77	3.83	3.89	3.96	4.02	4.09	4.15	4.22	120
130	4.28	4.35	4.41	4.48	4.55	4.62	4.69	4.75	4.82	4.89	130
140	4.96	5.04	5.11	5.18	5.25	5.33	5.40	5.47	5.55	5.62	140
150	5.70	5.78	5.85	5.93	6.01	6.09	6.16	6.24	6.32	6.40	150
160	6.48	6.57	6.65	6.73	6.81	6.90	6.98	7.06	7.15	7.23	160
170	7.32	7.41	7.49	7.58	7.67	7.76	7.85	7.94	8.03	8.12	170
180	8.21	8.30	8.39	8.48	8.58	8.67	8.76	8.86	8.95	9.05	180
190	9.14	9.24	9.34	9.44	9.53	9.63	9.73	9.83	9.93	10.03	190
200	10.13	10.23	10.34	10.44	10.54	10.65	10.75	10.85	10.96	11.06	200
210	11.17	11.28	11.38	11.49	11.60	11.71	11.82	11.93	12.04	12.15	220
220	12.26	12.37	12.48	12.60	12.71	12.82	12.94	13.05	13.17	13.28	220
230	13.40	13.52	13.63	13.75	13.87	13.99	14.11	14.23	14.35	14.47	230
240	14.59	14.71	14.83	14.96	15.08	15.20	15.33	15.45	15.58	15.71	240
250	15.83										

For steel qualities with intermediate strengths, linear interpolation may be applied to determine the value of ω.

The breaking force F_b may be evaluated by means of the following formulae:

for instantaneous safety gear

$$F_b = 25\ (Q + M)\ \text{(N)} \tag{7.2a}$$

for captive roller-type safety gear

$$F_b = 15\ (Q + M)\ \text{(N)} \tag{7.2b}$$

for progressive safety gear

$$F_b = 10\ (Q + M)\ \text{(N)}, \tag{7.2c}$$

where Q is rated load (kg) and M is the sum of the mass of the empty car and the masses of the appropriate portion of the travelling cable and any compensating device, suspended from the car (kg).

The value of σ_k must not exceed:

140 N/mm² for steel of 370 N/mm² grade
210 N/mm² for steel of 520 N/mm² grade

Finally

$$\lambda = \frac{l_k}{i}, \tag{7.3}$$

where l_k is maximum distance between guide-rail brackets (mm) and i is the radius of gyration (mm)

$$i = \sqrt{\frac{J}{S}}\ \text{(mm)}, \tag{7.4}$$

where J is moment of inertia of the cross-sectional area (mm⁴); for determination of J assume that buckling would occur in the plane of the smallest rigidity in bending of the guide-rail.

As seen in Fig. 7.4 the guide-rail is supposed to behave like a plain beam of two hinged supports, subjected to the braking force in its longitudinal axis. The stress analysis carried out by Janovský (1981) for different elevator operation conditions revealed the fact that calculation of buckling in the plane of the smallest rigidity of the guide-rail should be taken into consideration with small capacity elevators, but is not decisive with large capacity passenger and

Fig. 7.4 — Diagram for calculation of guide-rails.

freight elevators. Janovský considered the guide-rail to be a continuous beam of a variable number of supports. In fact, the braking force on the application of safety gear is acting in parallel to the longitudinal axis of the guide, its eccentric position creating a bending moment in addition to the axial force. Numerical calculations proved the resultant stress in combined pressure and bending in the plane of the bending moment to be decisive for most elevators.

Under normal operation (running) conditions the load may be unevenly distributed in two perpendicular directions. In Fig. 7.5 a pictorial drawing of guide-rails and all forces exerted upon them due to uneven car loading (off-centring of the load) are shown. Forces F_y are exerted in the plane of the guide-rails (y–y), while F_{z1} and F_{z2} are acting in z–z planes at right angles to y–y plane. Each guide-rail is subjected to bending due to F_y and combined bending and torsion due to F_z. The forces may be calculated as follows:

$$F_y = \frac{Q \times g_n \times e_y}{h} \tag{7.5}$$

$$F_{z1} = \frac{Q \times g_n \times e_z \times (b + 2e_y)}{2h \times b} \tag{7.6}$$

$$F_{z2} = \frac{Q \times g_n \times e_z \times (b - 2e_y)}{2h \times b} \tag{7.7}$$

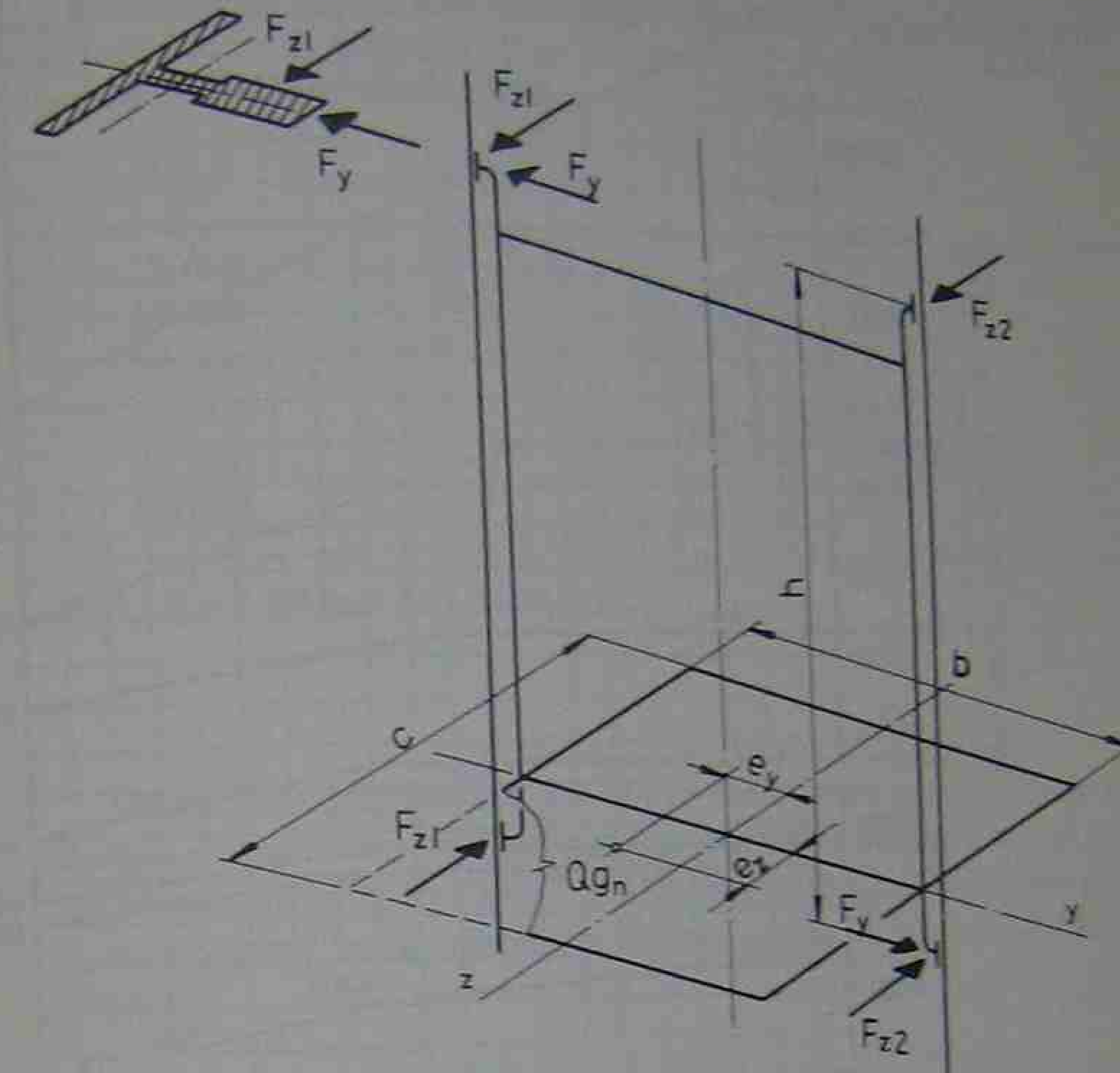


Fig. 7.5 — Forces on guide-rails due to uneven load distribution.

The total deflection of a guide-rail must be determined by the superposition of partial deflections induced by individual forces.

The American Safety Code ANSI/ASME A 17.1 specifies the maximum stress in a guide-rail or its reinforcement, due to the horizontal forces imposed on the rail during loading, unloading or running, calculated without impact, as 103 N/mm^2 based on the class of loading and the maximum permissible deflection as 6.3 mm.

In the case of a single car or counterweight safety gear being used, the maximum suspended weight of the car and its rated load, or the maximum suspended weight of the counterweight, including the weight of any compensating ropes or chains and travelling cables, per pair of guide-rails, can be read off the diagram in Fig. 7.6 for the size of the guide-rail and the bracket spacing used. All quantities are quoted in American gauge which can be easily converted to European gauge (see Fig. 7.6).

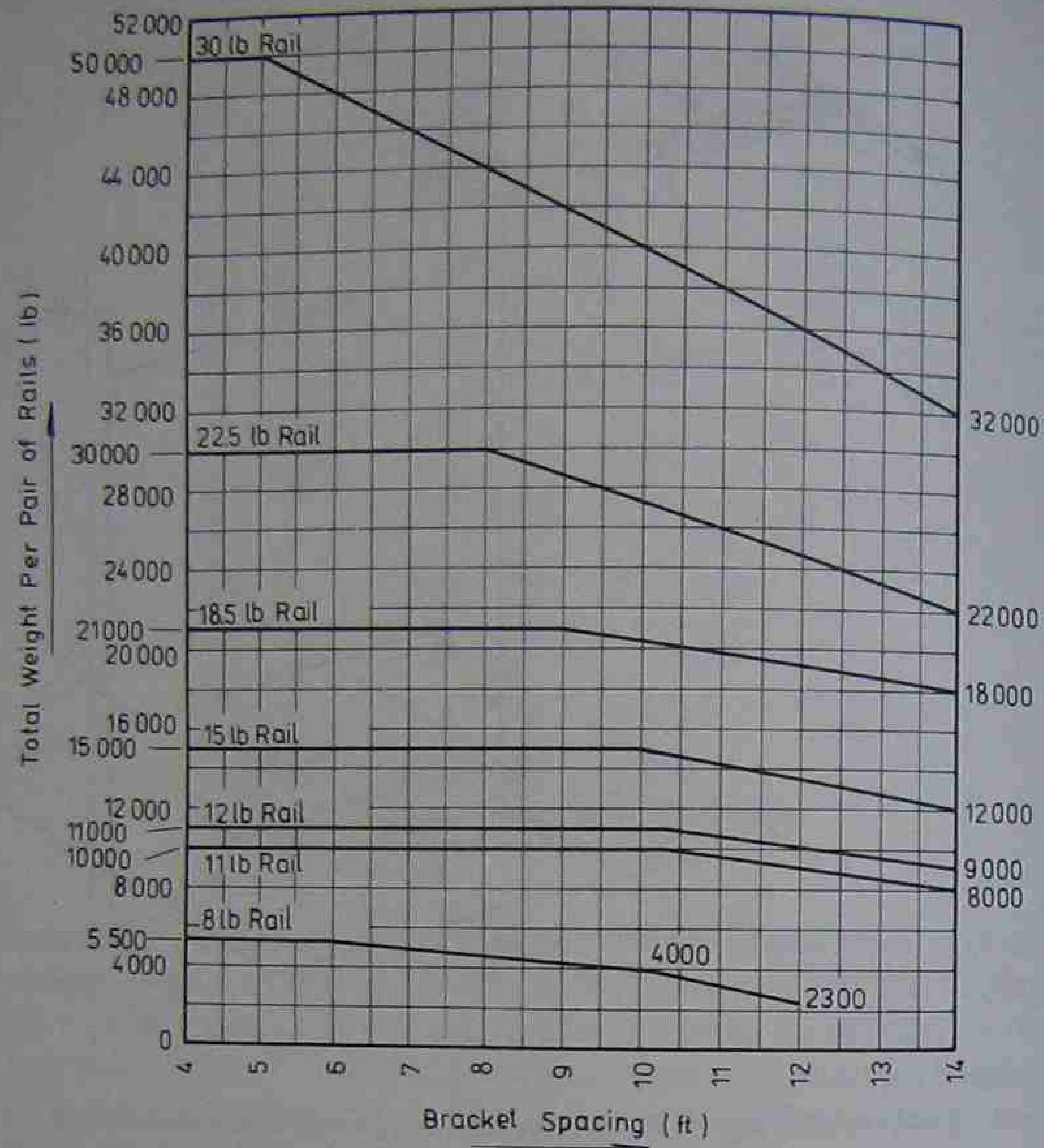


Fig. 7.6 — Diagram for guide-rail calculation.
Note: 1lb = 0.454 kg; 1ft = 0.305 m

With two (Duplex) car or counterweight safety gears employed the loads specified in Fig. 7.6 may be increased by the multiplying factors shown in Table 7.4.

Table 7.4 — Load multipliers for different vertical distances between safety gears for duplex safety gears

Vertical distance (ft)	Multiplier
18 or more	2.0
15	1.83
12	1.67
9	1.50

Note: 1 ft = 0.305 m.

Table 7.5 — Guide-rails for counterweights without safety gears

Weight of counterweight (lb)	Nominal weight of guide-rail (lb/ft)
15 000	8
27 000	11
29 000	12
40 000	15
56 000	18 ½
80 000	22 ½

Note: 1 lb = 0.454 kg; 1 lb/ft = 1.49 kg/m.

Where counterweights are not provided with a safety gear the weight of the counterweight per pair of guide-rails must not exceed the values specified in Table 7.5. The bracket spacing in the case of no reinforcement being employed should not be in excess of 4.88 m (16 ft).

Example 7.1

Evaluate the stress in the guide rails during safety gear operation in compliance with BS 5655. Technical parameters of the elevator:

Rated load $Q = 630$ kg
Mass of the car $K = 737$ kg

No compensating ropes or chains are used; the mass of the travelling cable can be neglected for the purpose of this calculation. Progressive safety gear is employed. The car is guided by two guide-rails of the following size and parameters:

T 68/82/9 — made of steel of 370 N/mm^2 grade
Cross-sectional area $S = 1070$ mm^2
Minimum moment of inertia $J = 307\,000$ mm^4.
Maximum distance between guide rail brackets $l_k = 2500$ mm.

The stress in guide-rails (Equation 7.1)

$$\sigma_k = \frac{10(K+Q)\times\omega}{S}$$

The radius of gyration (Equation 7.4)

$$i = \sqrt{\frac{J}{S}} = \sqrt{\frac{307\,000}{1070}} = 16.938 \text{ mm}$$

The coefficient of slenderness (Equation 7.3)

$$\lambda = \frac{l_k}{i} = \frac{2500}{16.938} = 147.59$$

The buckling factor (from Table 7.2).

$$\omega = 3.68$$

$$\sigma_k = \frac{10\times(630+737)\times 3.68}{1070} = 47 \text{ N/mm}^2$$

The guide rails will easily withstand the safety gear application as the maximum permissible stress is 140 N/mm^2.

7.3 TYPES OF GUIDE SHOES

Both the car and counterweight must be guided on each guide-rail by upper and lower guiding members attached to the frame.

There are two principal types of guide shoes in existence, namely

(1) Glide shoes (slipper guides).
(2) Roller guides.

Glide shoes are currently used for low and medium speeds to approximately 2 m/s. Sliding friction may represent a considerable resistance to car motion especially in the case of the shoes being spring-loaded, so that a constant pressure can be exerted on the guide-rails. The shoes are mostly of the swivel type, the journal cushions being made of neoprene or similar material. The shoes are made of cast-iron or steel and lined with special gibs. It has become a standard practice today to use plastic materials for gibs, such as

nylon, of low coefficient of friction, good sliding properties and wear resistance. Furthermore, the jibs facilitate the absorption of shock loads and rail misalignment. Molybdenum disulphide is sometimes added to produce an abrasion resistant material providing an even longer service.

With the application of glide shoes the guide-rails are lubricated to diminish the frictional resistance and wear and improve riding conditions. Automatic lubricators are currently employed at present, using heavy oil or grease. In Fig. 7.7 a simple automatic device is shown; it is mounted directly on the car and counterweight upper guide shoes. It assures an even application of protective film of heavy oil over the entire bearing surface of the guide-rail. The rate of flow can be adjusted by a readily accessible screw. No brushes or wipers are employed.

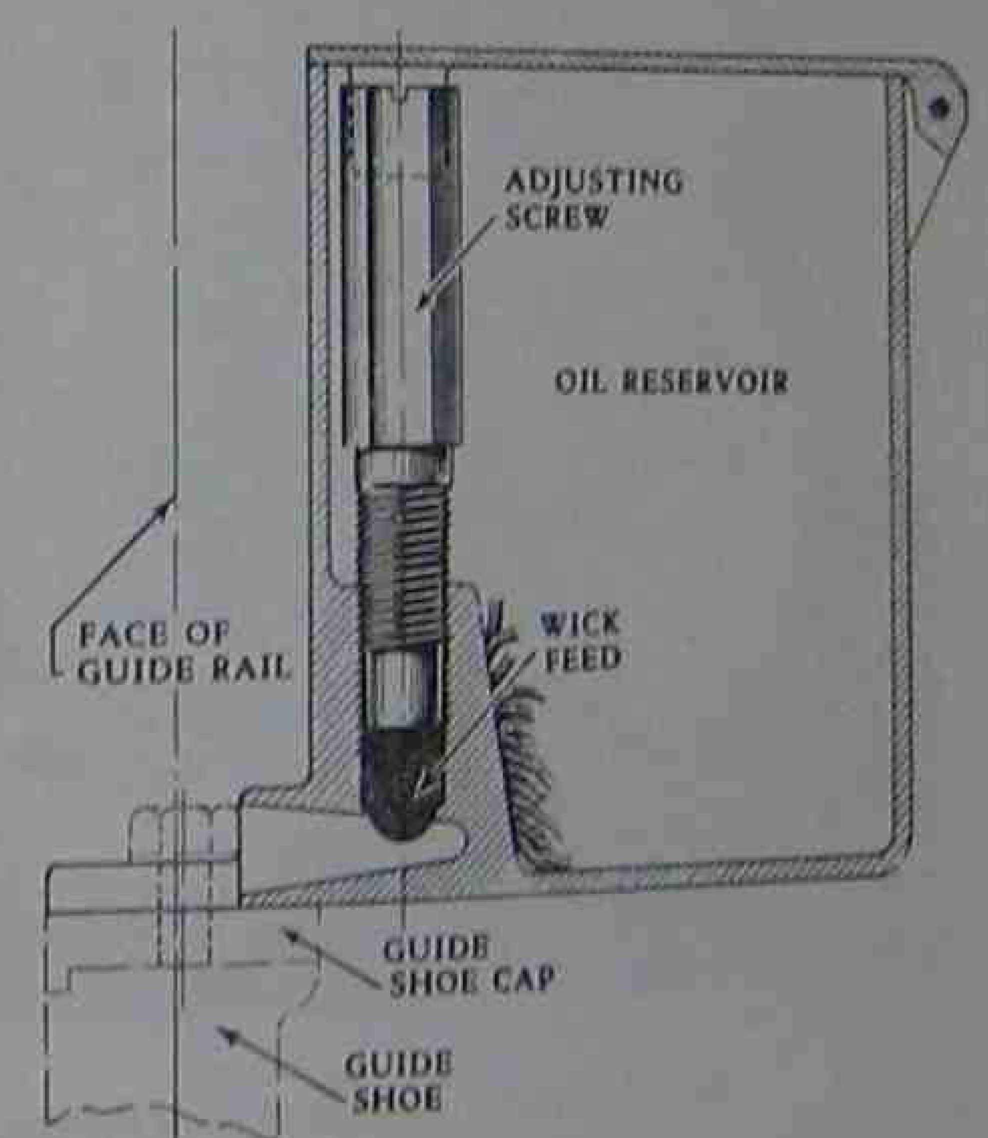


Fig. 7.7 — Guide-rail lubricator (Otis Elevator Co.).

To achieve constant properties of the fluid film after the application of a lubricant to the guide-rail is not an easy matter as the film is exposed to environmental conditions, such as dirt and dust settling, variation of ambient temperature and humidity, extensive exposure to the air oxygen etc., which strongly affect both its physical and chemical properties.

Roller guides are essential on high-speed elevators, but the smooth ride and the saving in power resulting from the reduction in friction justifies their use on medium speed elevators too.

Roller guides are composed of three spring-loaded rollers, which are in permanent contact with the guide-rail as a result. Since the rollers are provided with rubber or polyurethane tyres noise and vibrations are minimized and riding quality is improved considerably as there is no metal-to-metal contact. The rollers are of large diameter nowadays, which results in quiet operation and smaller frictional resistance. Furthermore, they are mounted on ball bearings. Each roller is supported by a pivoted rocker-arm which automatically adjusts itself to the guide-rail. For heavy duty conditions six rollers are sometimes used in pairs on spring-loaded rocker levers.

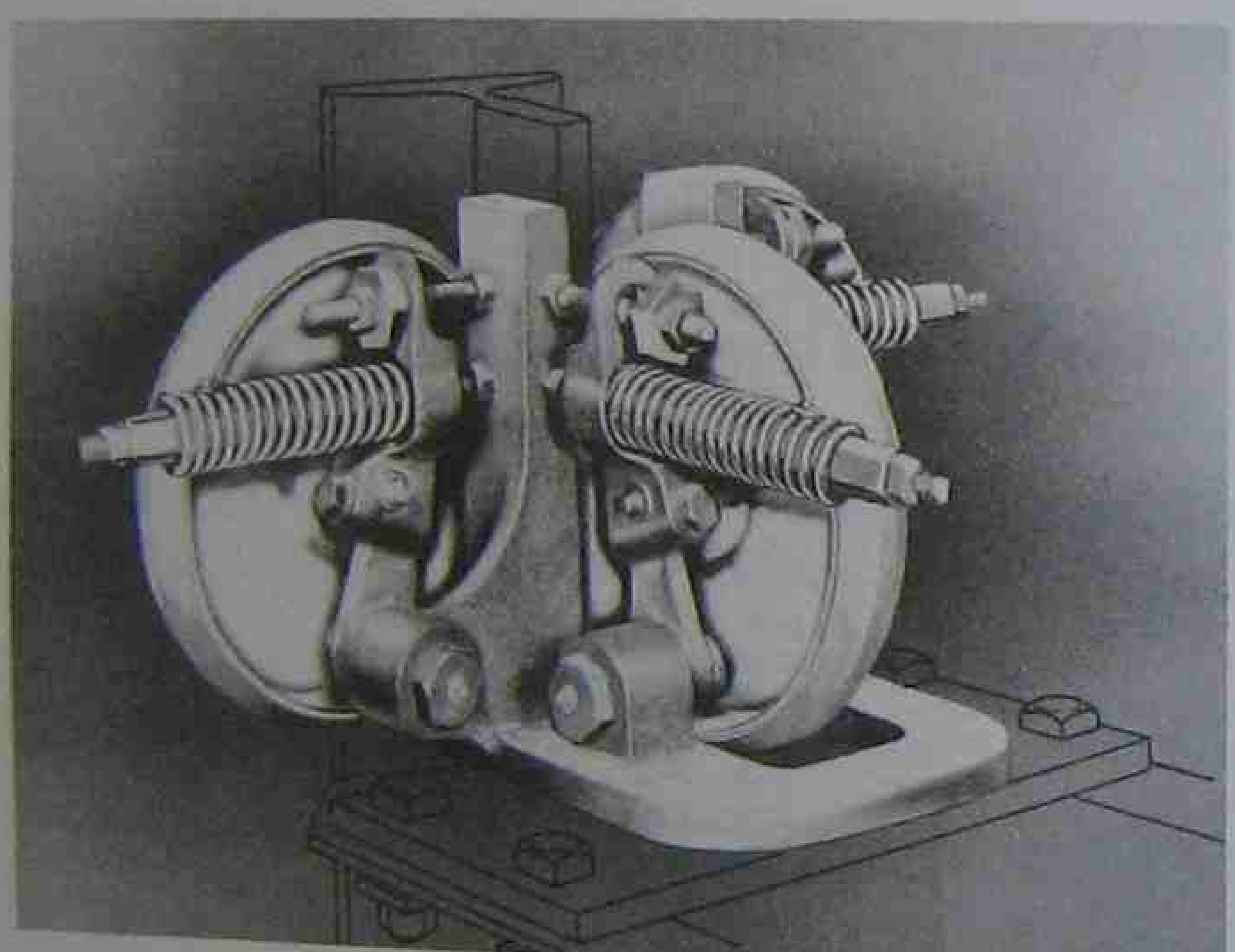


Fig. 7.8 — Roller guide shoes (Otis Elevator Co.).

Roller guides operate on dry, unlubricated rails, so that the danger of accumulation of oil or grease in the pit is avoided and a fire hazard eliminated.

In Fig. 7.8 a typical roller guide is shown, while in Fig. 7.9 the helical steel springs were replaced by rubber cushions.

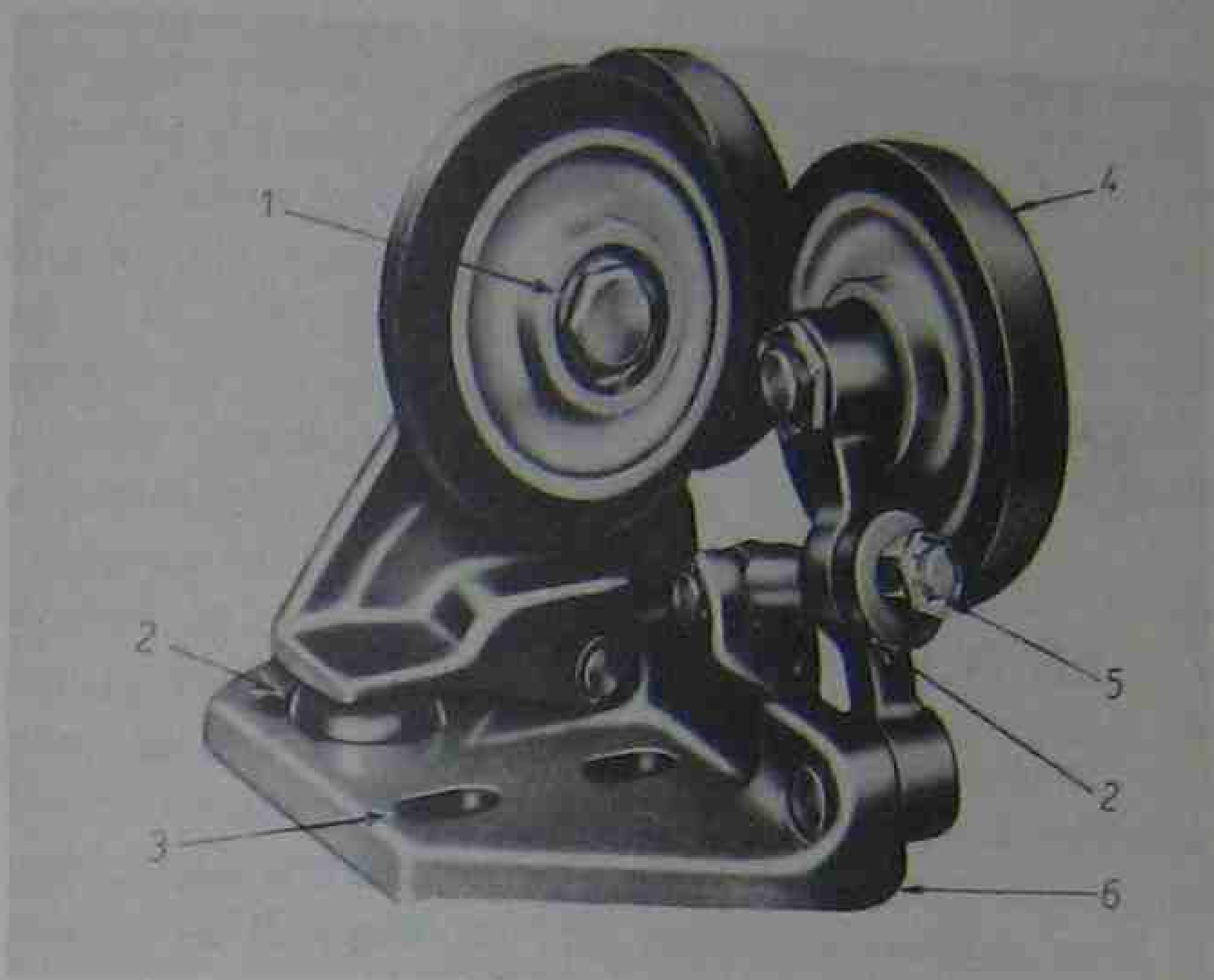


Fig. 7.9 — Roller guide shoes with rubber cushions (Moline Accessories Co.).

1. Two large precision ball bearings per wheel
2. Rubber cushions
3. Mounting slots for interchangeability with other shoes
4. Rubber tyred rollers
5. Bolts for the adjustment of alignment and pressure
6. Cast body with machined base.

8

Safety gear

8.1 GENERAL

The car of every elevator which is suspended by wire ropes or chains, and which may be entered by persons for the purpose of transportation or mechanical handling of goods at landings, must be provided with a safety gear. The counterweight must be equipped with a safety gear when occupied spaces are located under the well.

The safety gear is a mechanical device for stopping the car (or counterweight) by gripping the guide-rails in the event of the car speed attaining a predetermined value in a downward direction of travel irrespective of what the reason for the increase in speed may be.

The safety gear is preferably located below the lower members of the car frame and operates on one pair of guide-rails. The operation should be simultaneous on both guide-rails; the floor of the car with the load uniformly distributed must not incline more than 5% from its normal position (3.1% in the USA).

The predetermined value of the car (counterweight) speed at which it must be stopped is the tripping speed of the overspeed governor. The counterweight safety gear may be either tripped by the failure of the suspension gear, or by a safety rope if the rated speed does not exceed 1 m/s. A device must be mounted on the governor that would initiate the stopping of the motor before or at the moment of safety gear operation.

Car safety gears are classified on the basis of their performance characteristics. They are as follows:

(1) *Instantaneous type*, which exerts a rapidly increasing pressure on the guide-rails during the stopping period. The stopping time and distance are very short; no flexible medium is introduced to limit the retarding force and the stopping distance. This type can be employed for rated speeds not exceeding 0.63 m/s in Europe, but up to 0.76 m/s in the USA. The behaviour of a car or counterweight at the application of this kind of safety gear cannot be exactly predicted nor calculated and must be examined experimentally.

(2) *Instantaneous type with buffered effect* incorporates an elastic system of either the energy accumulation type with buffered return movement or the energy dissipation type. It is usually represented by one or more oil buffers interposed between the lower members of the car frame and a safety plank applied on the guide-rails and develops retarding forces during the compression stroke of the buffers. The stopping distance is equal to the effective stroke of the buffers. They may be used for rated speeds up to 1 m/s in Europe and up to 2.5 m/s in the USA.

(3) *Progressive type* applies limited pressure on the guide-rails during the stopping interval. After the safety gear is fully applied retarding forces are reasonably uniform. The stopping time and distance are related to the mass of the movable system being stopped and the speed at which the application of the safety gear is initiated. This type must be used for rated speeds in excess of 1 m/s (in Europe).

If duplex safety gears are employed on the car, they must be all of the progressive type.

The release of the safety gear must be carried out only by the motion of the car in an upward direction; after its release the safety gear must be ready for another operation.

8.2 OVERSPEED GOVERNOR

A schematic arrangement of the overspeed governor system is shown in Fig. 8.1. The governor is usually located in the machine room; it is provided with the governor rope (1), passing round the governor pulley (2) down to a tensioning pulley (3) in the pit and back again to the governor pulley. The system is driven by the car to which the governor rope is attached at the point (4). When the tripping speed of the governor rope is achieved the governor stops the rope. Since the car continues in a downward motion the tension in the governor rope

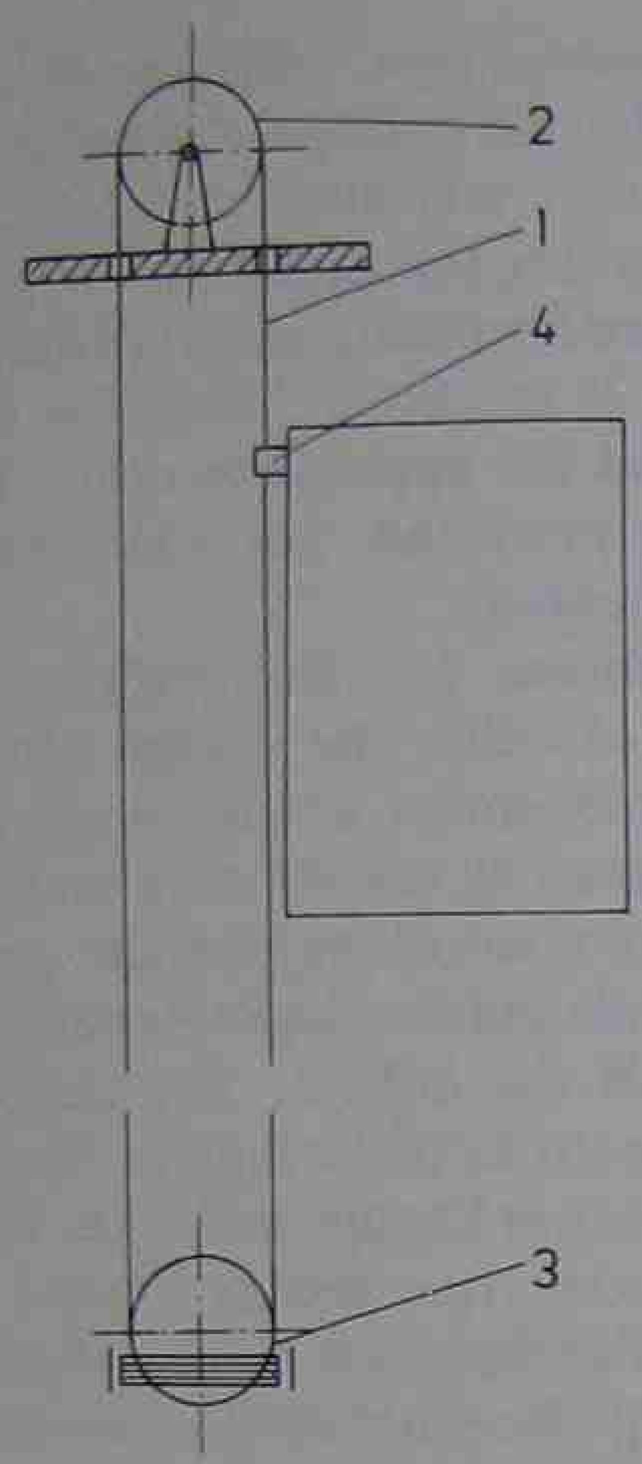


Fig. 8.1 — Arrangement of overspeed governor system: 1. Governor rope; 2. Governor pulley; 3. Tensioning pulley; 4. Attachment point.

is increased exceeding the value necessary to engage the safety gear and the safety gear is consequently set in operation.

In compliance with EN 81 the tripping speed should be at least 115% of the rated speed and less than:

(a) 0.8 m/s for instantaneous safety gears except for the captive roller type.
(b) 1 m/s for safety gears of the captive roller type.
(c) 1.5 m/s for instantaneous safety gears with buffered effect or progressive safety gear used for rated speeds ≦ 1.0 m/s.
(d) $1.25v+(0.25/v)$, where v is rated speed (m/s) for other types of safety gears.

For rated speeds in excess of 1 m/s it is recommended that the tripping speed be close to the upper speed limit. The tripping speed of counterweight overspeed governors should be higher than that for the car, but not by more than 10%.

In the USA the lower limit of the car governor tripping speed is the same as specified in EN 81. The upper limit is given in Table 8.1

Table 8.1 — Maximum car governor tripping speeds

Rated speed (m/s)	Maximum car governor tripping speed (m/s)	Maximum car speed at which governor overspeed switch operates (m/s)
0–0.635	0.889	0.889[a]
0.762	1.067	1.067[a]
0.889	1.270	1.143
1.017	1.423	1.281
1.143	1.565	1.408
1.270	1.713	1.540
1.525	2.008	1.804
1.779	2.297	2.069
2.033	2.592	2.333
2.287	2.887	2.602
2.541	3.177	2.862
3.050	3.762	3.573
3.558	4.346	4.128
4.067	4.931	4.682
4.575	5.516	5.241
5.083	6.100	5.795
5.592	6.710	6.375
6.100	7.320	6.954
6.609	7.930	7.534
7.117	8.540	8.113
7.625	9.151	8.693
8.134	9.761	9.273
8.642	10.371	9.852
9.151	10.981	10.432
9.659	11.591	11.011
10.167	12.201	11.591

a Governor overspeed switch not required on car governors.
For conversion fpm to m/s: 1 m/s ~196.7 fpm was taken.

dependent upon the rated speed as well as the car speeds at which the governor overspeed switch operates. For rated speeds in excess of 7.62 m/s (1500 fpm) the maximum tripping speeds must not exceed 120% of the rated speed. The counterweight governor tripping speed is the same in the USA as in Europe.

The force induced in the rope by the governor when tripping occurs should be at least either 300 N or twice the necessary force to engage the safety gear, whatever is greater.

The governor rope diameter should be at least 6 mm (9.5 mm in the USA) and for the calculation of the minimum breaking load a safety factor of at least 8 (5 in the USA) must be taken. The minimum ratio between the pitch diameter of the governor pulley and the nominal diameter of the rope is specified as 30 in Europe, while it is related to the rated speed and the number of strands in the USA (Table 8.2).

Table 8.2 — Multipliers for determining governor pulley pitch diameter (USA)

Rated speed (m/s)	No. of strands	Multiplier
≦1.07	6	42
≦1.07	8	30
>1.07	6	46
>1.07	8	32

The response time of the overspeed governor should be as short as possible to facilitate a rapid operation of the safety gear and prevent the car speed from becoming excessive. For progressive safety gears the maximum permissible movement of the overspeed governor rope in order to operate the safety mechanism is specified in ANSI/ASME A17.1. For car safety gears it depends on the rated speed as shown in Table 8.3. For counterweight safety gears it is specified as 1.07 m regardless of the rated speed.

From the design aspect overspeed governors may be of pendulum

Table 8.3 — Maximum permissible movement of car governor rope

Rated speed (m/s)	Maximum distance (m)
up to 1.02	1.07
1.02 to 1.91	0.914
over 1.91	0.762

type or centrifugal type. Centrifugally-operated governors are either of geared vertical shaft type or of the horizontal shaft pattern. The latter type is most frequently employed nowadays. Simple construction resulting in a short response time and reliability in operation and minimal space requirements for installation are its principal features. In Fig. 8.2 a typical overspeed governor of this type is shown. It is

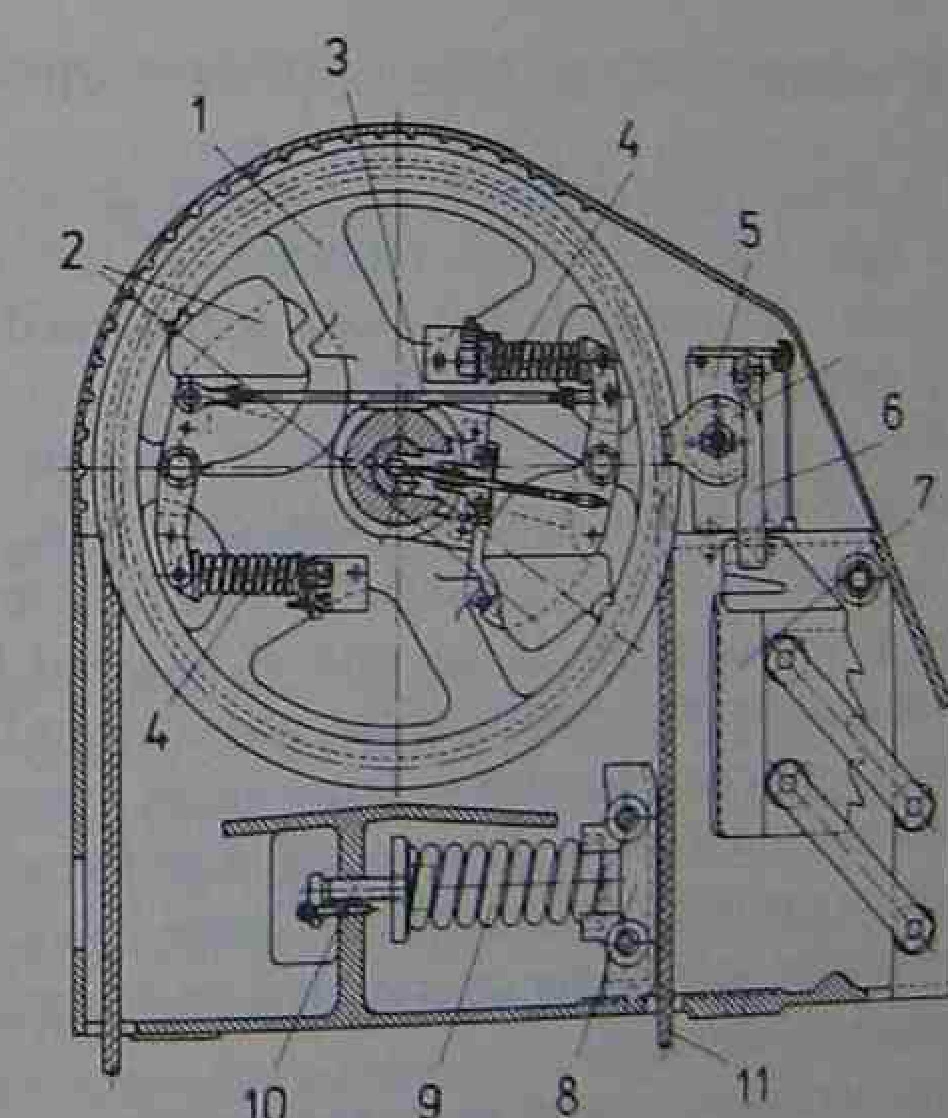


Fig. 8.2 — Centrifugal overspeed governor (Otis Elevator Co.). 1. Governor pulley; 2. Two pivoted flyweights; 3. Rod; 4. Helical springs; 5. Switch; 6. Latching device; 7. Swinging jaw; 8. Fixed jaw; 9. Helical spring; 10. Adjusting bolt; 11. Governor rope.

provided with two pivoted flyweights (2), linked together by a rod (3) to ensure simultaneous movement and secured in position by helical springs (4). The governor pulley (1) rotates in a vertical plane; if the speed of the elevator car exceeds the rated speed by a predetermined value the flyweights are driven outwards due to the centrifugal force and trip an overspeed switch (5), which cuts off power to the elevator and sets the brake. If the car speed continues to increase, the further outward motion of the flyweights causes them to trip a latching device (6), that normally holds a swinging jaw (7) of the governor clear of the governor rope. When the swinging jaw is released it clamps the governor rope (11) against the fixed jaw (8). This jaw is spring-loaded (helical spring (9)) and pre-set (adjusting bolt (10)) to give the tension required in the governor rope to operate the safety gear as the governor rope slides through the jaws during the safety gear operation. Governor jaws must be of such shape and minimum length that no appreciable damage to or deformation of the governor rope results from the stopping action of the jaws.

8.3 INSTANTANEOUS SAFETY GEAR

Instantaneous safety gear may be of three types:

(1) *Wedge-type instantaneous safety gears* were very popular years ago, but have been replaced by the eccentric cam type. However, the principle has been maintained with flexible guide clamp safety gear (see Fig. 8.7).

Wedge-shaped jaws are inserted in tapered cast-iron blocks attached to the lower members of the car frame. Two jaws operate on each guide-rail. They are connected by a system of rods and linkages to the governor rope. If the rope is stopped by the governor the relative motion of the car and the rope causes the operating rods of the safety gear mechanism to move in an upward direction and the jaws connected to the rods to engage with the guide-rails. As soon as the jaws come into contact with the guide-rail a wedging action takes place and the safety gear operation is no longer dependent upon the operating mechanism.

The necessary condition for the wedging action to take place is

$$\alpha \leqq \phi_2 - \phi_1, \tag{8.1}$$

where α is the angle of the wedge, ϕ_2 is the angle of friction between the jaw and the guide-rail, and ϕ_1 is the angle of friction between the jaw and the supporting block.

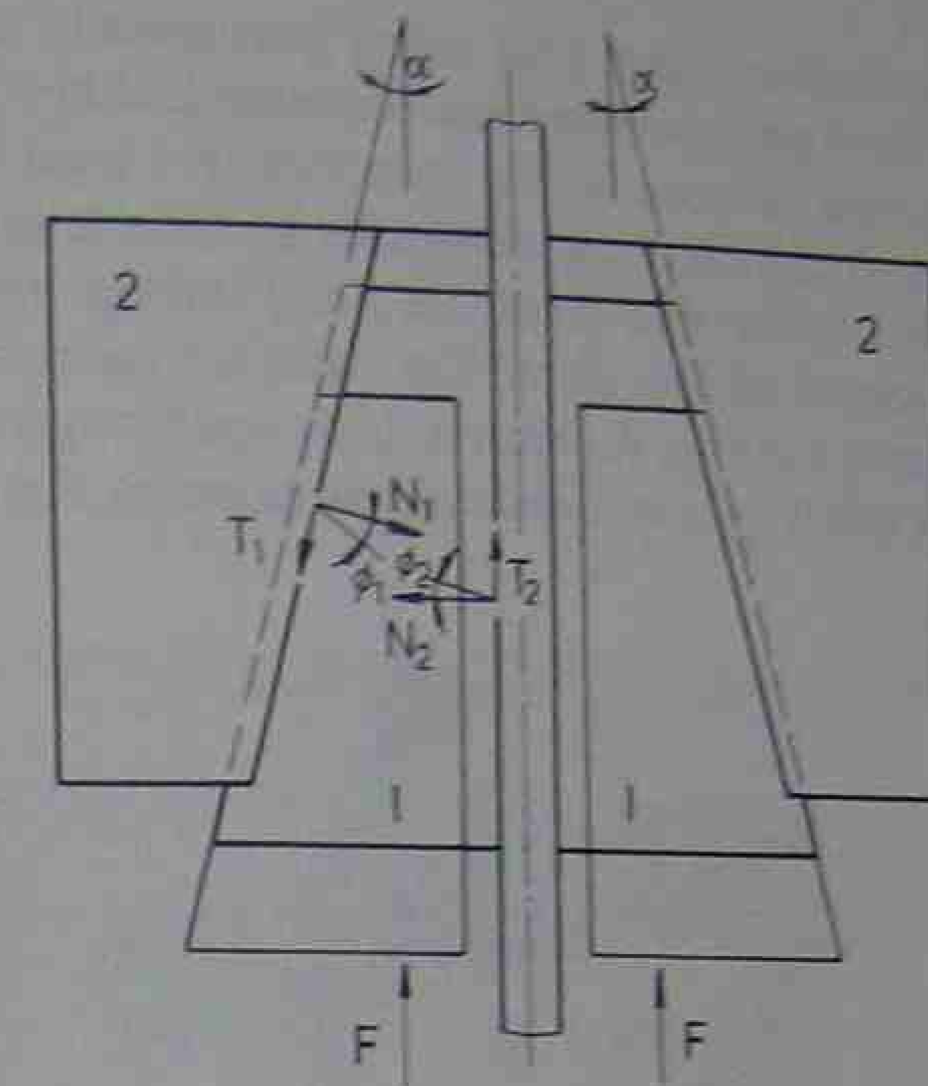


Fig. 8.3 — Diagram of safety gear block with wedge-shaped jaws. 1. Jaws; 2. Safety gear block.

In Fig. 8.3 a diagram of the jaws (1) of one safety gear block (2) are shown in both retracted and operating positions. All forces acting upon one wedge-shaped jaw are depicted in Fig. 8.3; the designations are as follows:

N — normal reaction force.
T — tangential reaction force (frictional resistance).
F — operating force to engage the jaws.

The rail-gripping side of the jaw is often notched in order to increase the coefficient of friction between the jaw and the guide-rail to satisfy equation (8.1) and to minimize the accumulation of oil, grease and dirt on the face of the jaw.

The stopping time and distance are very short and the rate of retardation is high with all instantaneous safety gears. The higher the coefficient of friction between the jaws and the guide-rail the greater the impact which results in the discomfort of the pasengers and heavy strains on both the car frame components and the guide-rails. For this reason it is advisable to provide a smooth surface on the rail-gripping side of the jaw and arrange for rolling friction on the opposite side. This is currently done with flexible guide clamp safety gears, where the wedge-shaped gibs move on chromium plated and case hardened rollers. In practice the angle of wedge of the jaw is $\alpha = 6$ to $7°$.

(2) *Eccentric cam type safety gear* usually consists of two hardened steel cams, serrated and eccentric in shape, for each guide-rail. In this type two connecting camshafts are used with cams keyed at either end; they are linked together and rotate in opposite directions when the safety gear is actuated; the linkage ensures simultaneous movement of all four cams. The cams are restrained in the retracted position by a spring. As a rule a single operating rod is attached to one cam. A safety gear of this design is shown in Fig. 8.4.

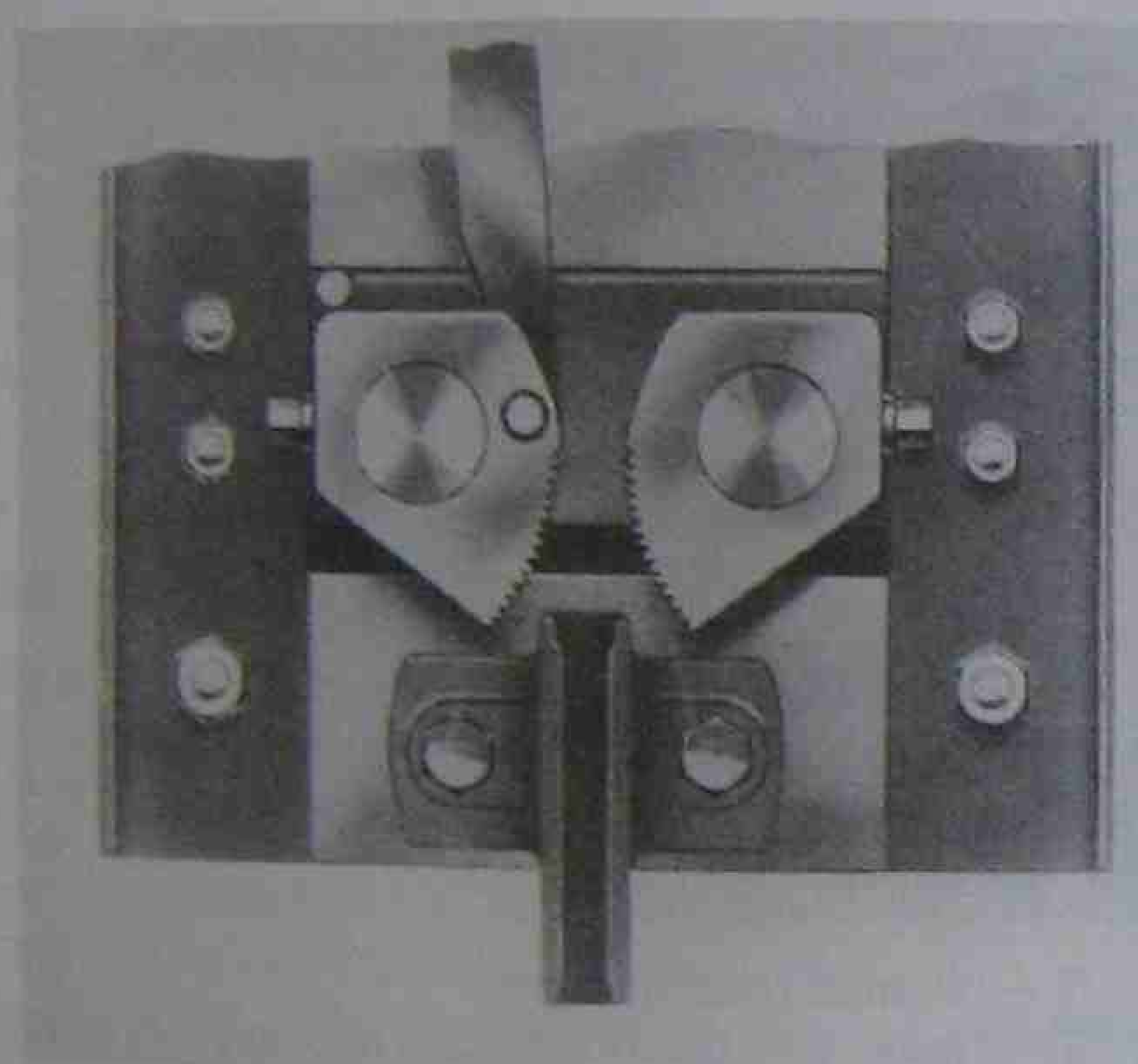


Fig. 8.4 — Instantaneous safety gear of eccentric cam type (Express Lift Co.).

With the counterweight only one cam is sometimes used for one guide-rail and a buffer plate is provided on the opposite side. Since the rail-gripping components of the safety gear are not symmetrical there is a tendency for the guide-rail to be forced out of the vertical plane unless the safety gear block can move in horizontal direction in order to take up the clearance between the fixed buffer plate and the guide rails. This action may extend the time from stopping the governor rope to the actual operation of the safety gear.

As seen in Fig. 8.5 the resultant force between the cam and guide-rail R should be so located as to give a turning moment $a \times R$ of the designated direction, otherwise further operation would be conditioned by a continuous pull on the operating rod and the pressure

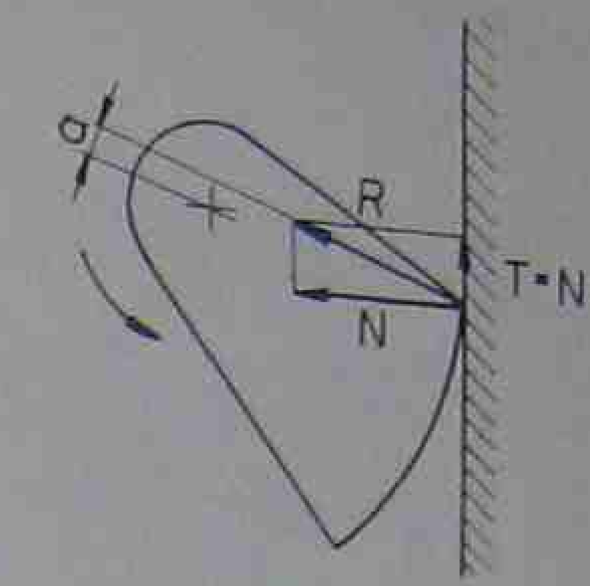


Fig. 8.5 — Diagram of forces on eccentric cam.

exerted on the guide-rail would be proportional to the tension in the governor rope. In Fig. 8.5 N is normal reaction force and T is friction between the cam and guide-rail.

The contact pressure is relatively high at the application of this type of safety gear as a small area of the cam engages with the guide-rail. Consequently a heavy strain may occur in the peripheral portion of the cam resulting in the breakage of a tooth and also the surface of the guide-rail may deteriorate.

(3) *Roller type safety gears* (Fig. 8.6) are usually employed on low speed heavy freight elevators.

A hardened knurled steel roller (1) is guided in a tapered steel jaw (6), which forms a self-aligning fixed jaw on the opposite side of the guide-rail. The roller is mounted on an actuating lever (2), to which the operating rod (4) is attached. Simultaneous operation on both guide-rails is ensured by mounting both actuating levers on a common shaft (3). When the roller engages with the guide-rail the jaw (6) moves in horizontal direction in the safety gear block (5) so that the clearance on the opposite side may easily be taken up.

For further operation (i.e. after the engagement has been achieved), independent of the function of the safety gear mechanism, the following condition must be satisfied:

$$\alpha \leqq \phi_1 + \phi_2, \tag{8.2}$$

where α is the angle of tapered jaw, ϕ_1 is the coefficient of friction between the roller and the tapered jaw and ϕ_2 is the coefficient of friction between the roller and the guide-rail.

8.4 PROGRESIVE SAFETY GEAR

In the case of the free fall of a car with rated load the average retardation after the progressive safety gear has been applied should

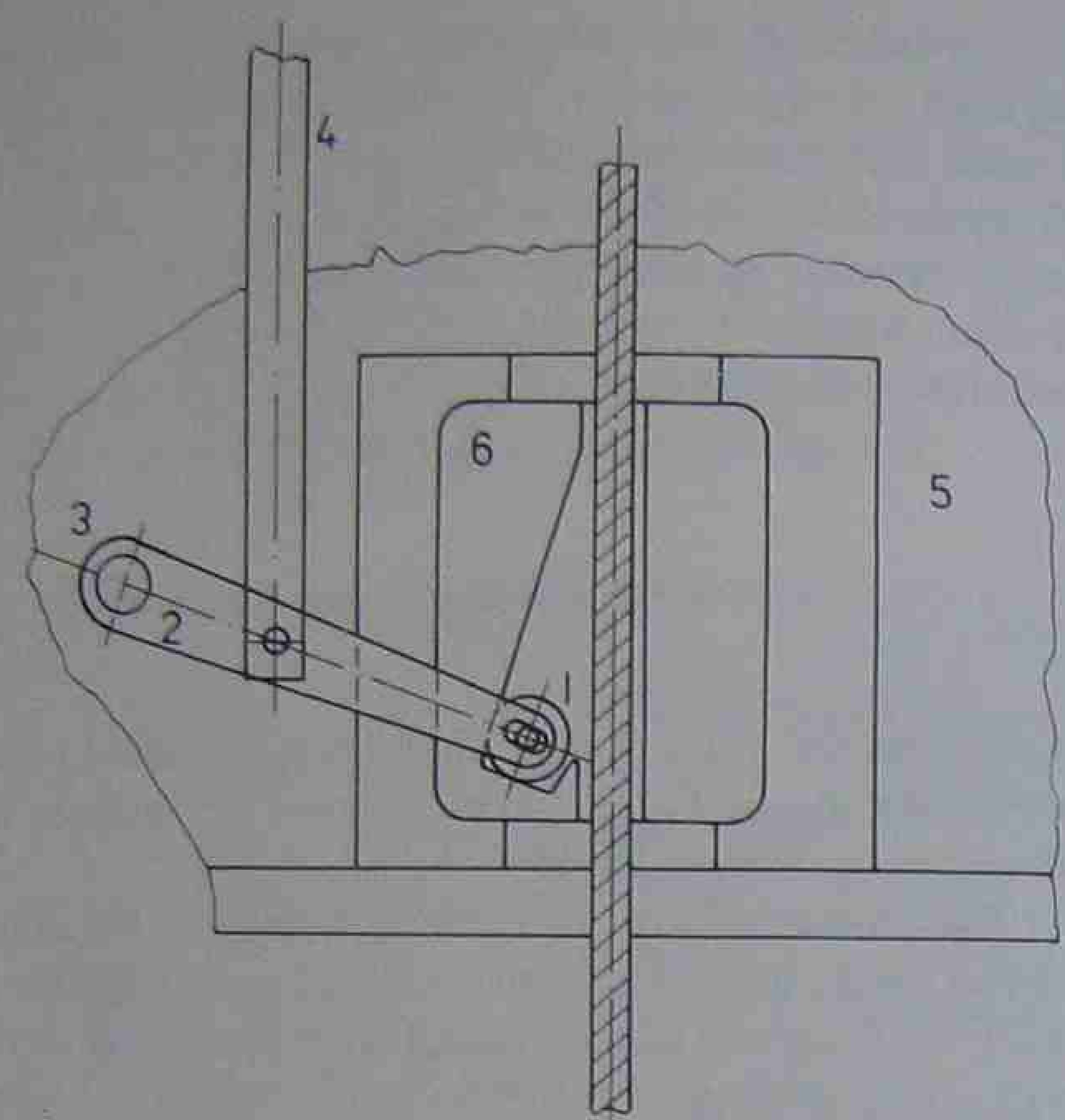


Fig. 8.6 — Diagram of roller type safety gear. 1. Roller; 2. Actuating lever; 3. Common shaft; 4. Operating rod; 5. Safety gear block; 6. Jaw.

lie between 0.2 g_n and 1.0 g_n, where g_n is standard acceleration of free fall. Maximum and minimum stopping distances are specified in A17.1 on the basis of governor tripping speeds.

A great number of progressive safety gears of various designs were introduced by elevator manufacturers in the past, however only a few of them have proved to have outstanding properties in all respects and have been employed until the present time.

Flexible guide clamp safety gears have been used most frequently. They consist of two safety clamps (one per guide-rail) securely bolted to the bottom members of the car frame and connected by a system of rods and linkages to ensure simultaneous action. A clamp for heavy duty installations is illustrated in Fig. 8.7, while a simple design for light and medium duty elevators is shown in Fig. 8.8. The function of the clamps of both designs is identical; the difference lies in the manner in which the pressure between the gibs and the guide-rail is controlled. The clamp assembly (Fig. 8.9 corresponds to Fig. 8.7) comprises two wedge-shaped gibs (1), moving on hardened steel chromium-plated rollers (2), mounted in a non-ferrous cage (3) and

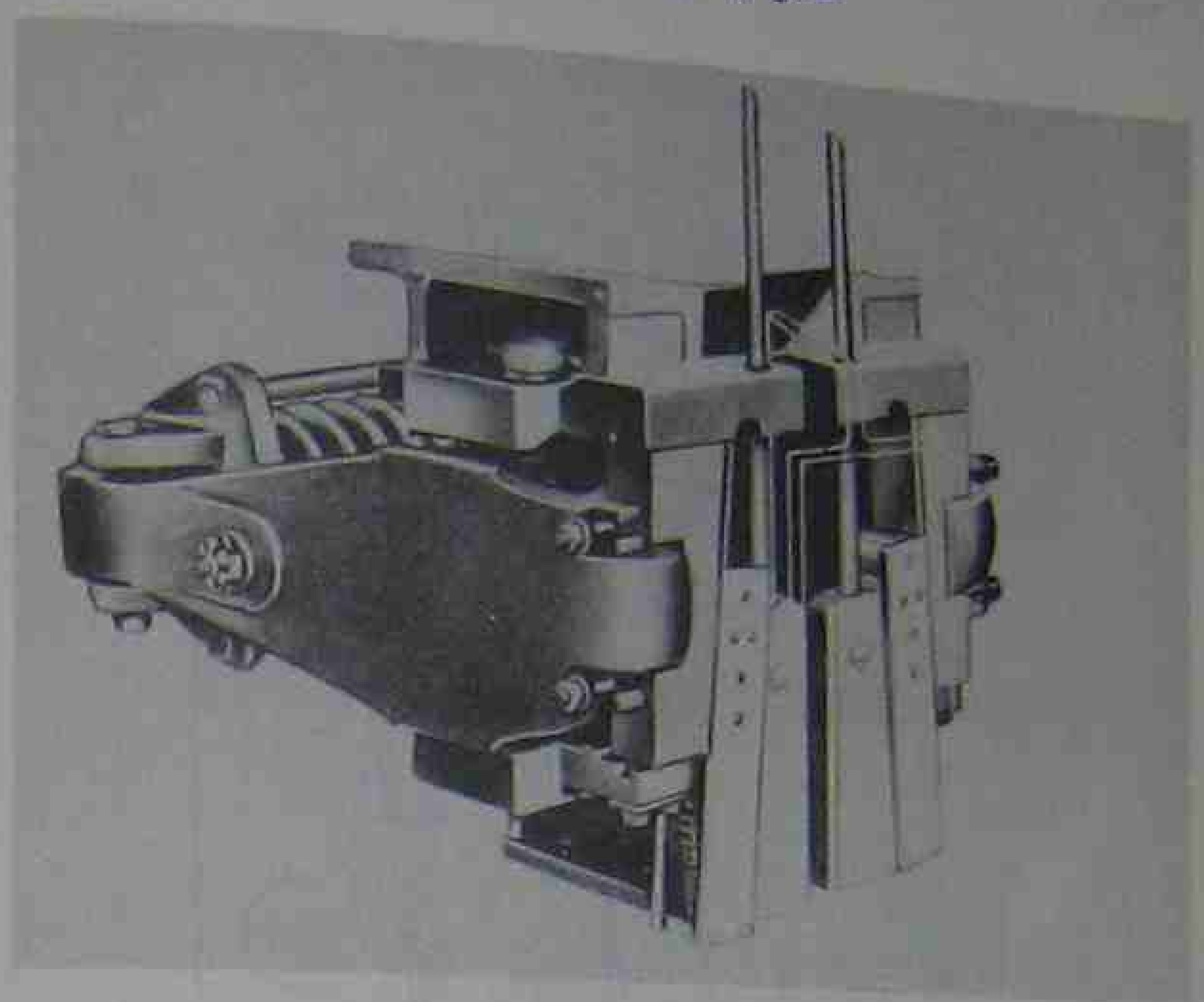


Fig. 8.7 — Flexible guide clamp safety gear for heavy duty elevators (Otis Elevator Co.)

Fig. 8.8 — Flexible guide clamp safety gear for light and medium duty elevators (Otis Elevator Co.).

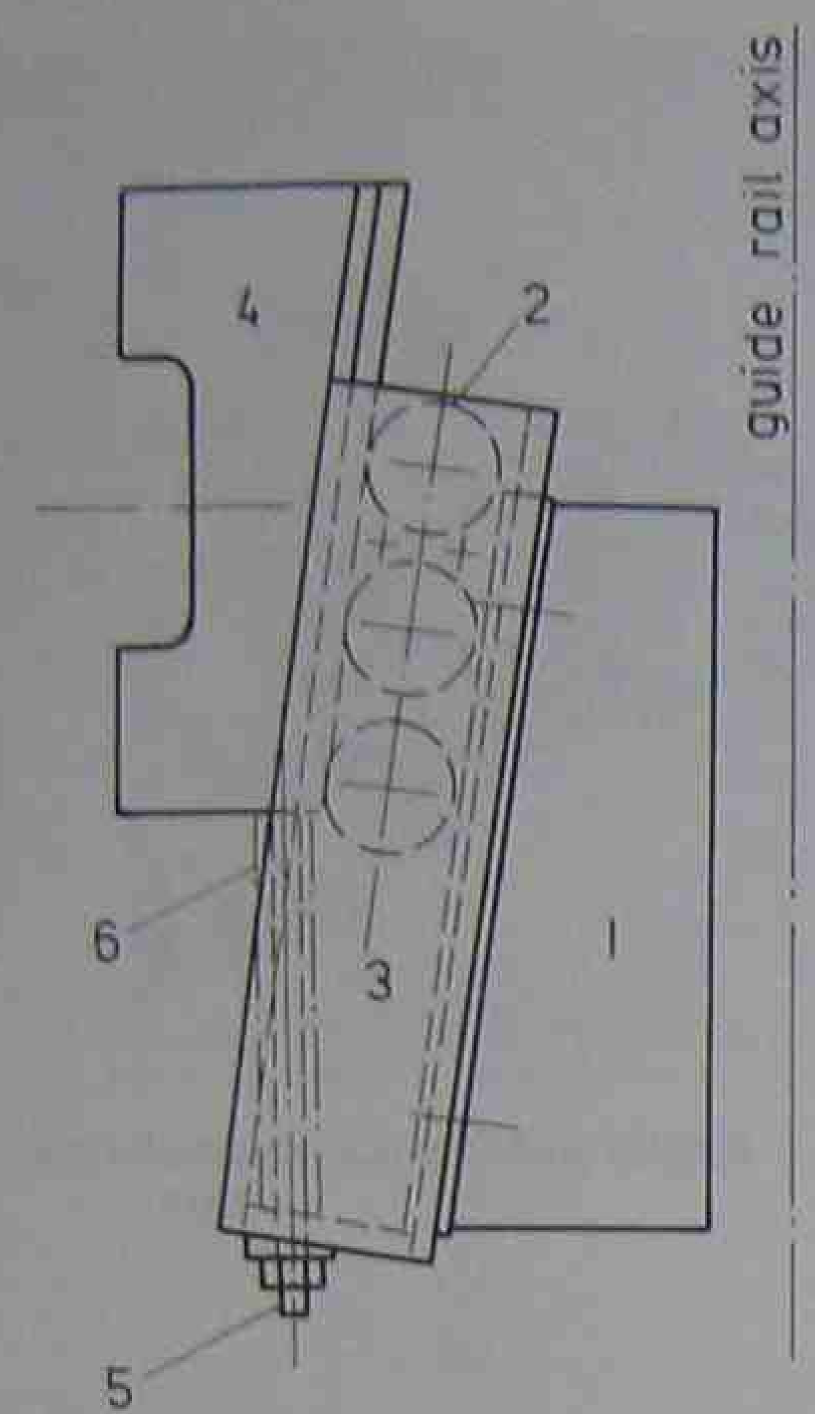


Fig. 8.9 — Clamp assembly of flexible guide clamp safety gear. 1. Gibs; 2. Rollers; 3. Cage; 4. Jaw; 5. Screw assembly; 6. Helical spring.

running in the hardened steel tracks of the jaw (4). The relative position of the cage (3) to the jaw (4) can be adjusted by means of a screw assembly (5), the cage being secured in its position by a helical spring (6). A two-sided recess in the jaw (4) facilitates the relative motion and accurate alignment of the gib against the guide-rail.

When the safety gear is actuated by the overspeed governor, the gibs are raised by operating rods and brought into contact with the guide-rail. A wedging action follows, independent of any member of the safety gear operating mechanism. As seen in Fig. 8.10 the jaws (1) are mounted on rocker arms (2), forming tongs and rotating around vertical fulcrum pins. The pressure between the gibs (4) and the guide-rail is controlled by an equalizing steel spring (3) so that the pressure cannot exceed a predetermined maximum value. After the maximum value has been achieved a constant retarding force is exerted causing the car to come to a smooth sliding stop at a relatively low rate of retardation.

A U-spring is used with the safety clamp illustrated in Fig. 8.8, substituting the helical spring and both rocker arms.

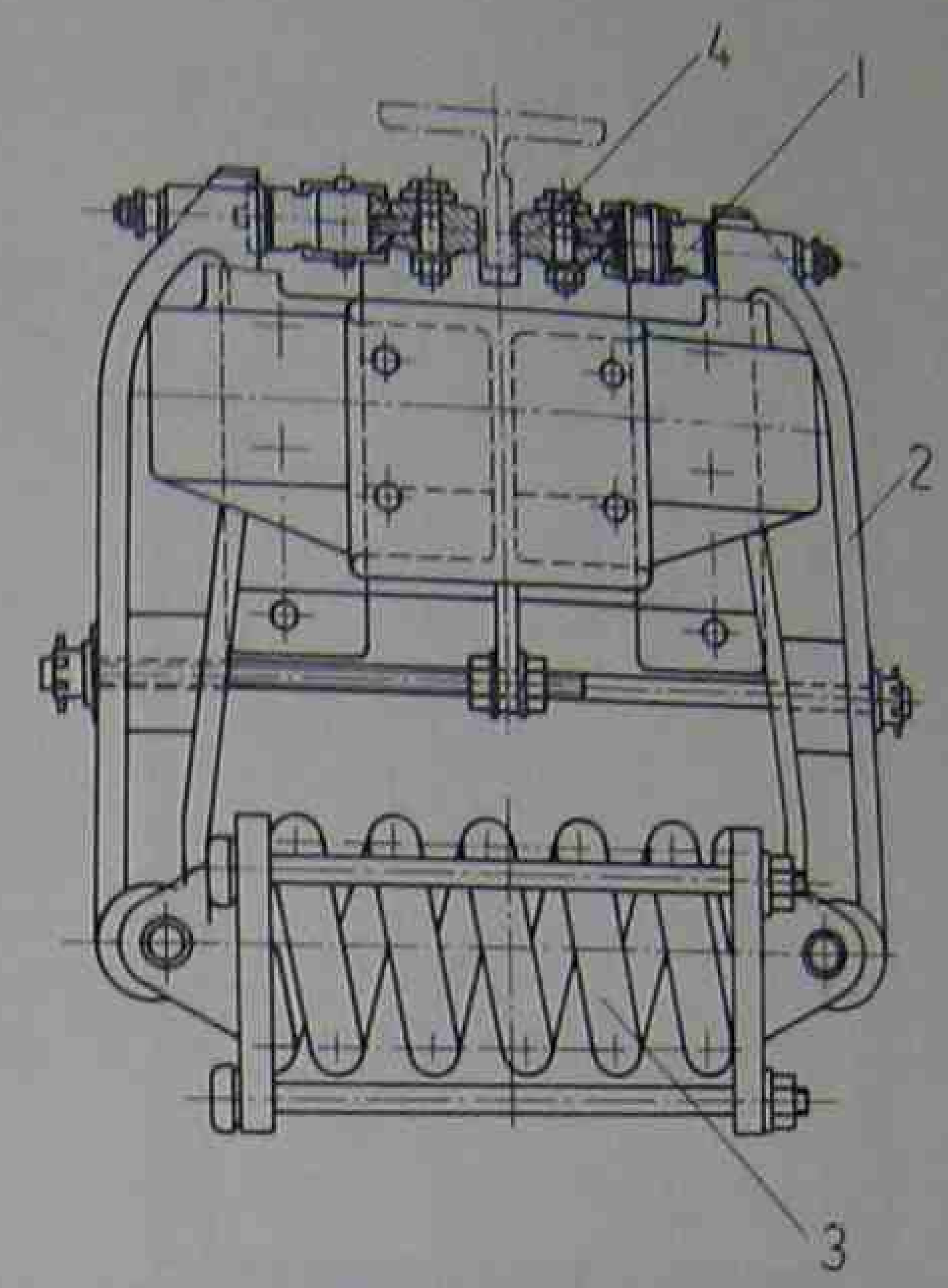


Fig. 8.10 — Rocker arms and equalizing spring of flexible guide clamp safety gear. 1. Jaws; 2. Rocker arms; 3. Equalizing spring; 4. Gibs.

The operating mechanism is depicted in Fig. 8.11 and Fig. 8.12. Both ends of the governor rope (1) are attached to the double-arm lever (2), pivoted on a short horizontal shaft (3), providing linkage between components operating on each side of one guide-rail. Operating rods (4) and (6) are connected directly to the wedge-shaped gibs (7) and (8) of the safety clamp (9). The other side of the safety gear system is of the same design; the linkage of both sides is accomplished by a connecting rod (10), provided with a right-hand thread at one end and a left-hand thread at the other so that alignment of both sides of the system can be easily attained by revolving the rod. This arrangement is currently used in high rise installations; an adjustable spring is incorporated in the linkage system to prevent the safety gear from being actuated due to minor impulses on the governor rope while the car is travelling at normal speed. In lower rise installations the upper end of the governor rope is usually attached to the operating mechanism and the other end to the car frame. With this arrangement the system is kept at rest under normal conditions by weights attached to the operating rods.

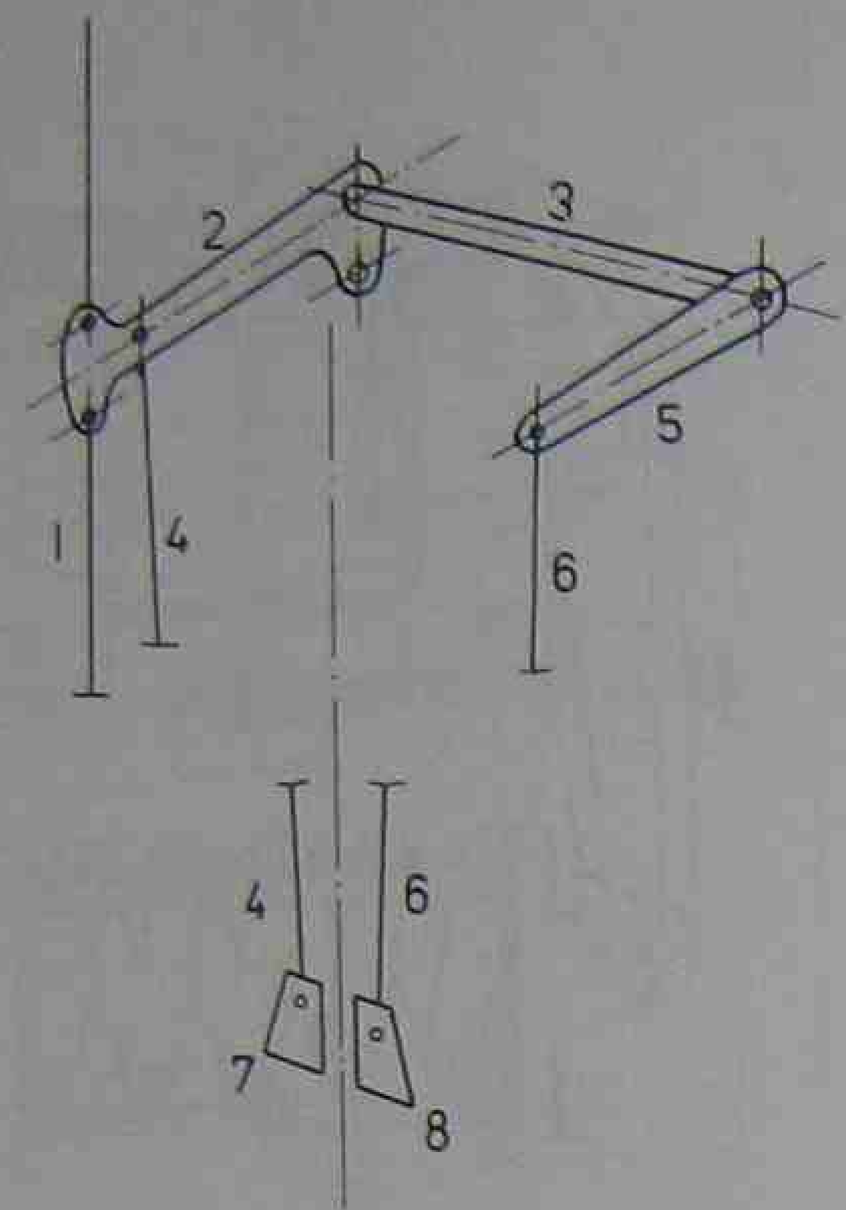


Fig. 8.11 — Operating mechanism of flexible guide clamp safety gear. 1. Governor rope; 2 & 5. Lever; 3. Shaft; 4 & 6. Operating rods; 7 & 8. Gibs; 9. Clamp; 10. Connecting rod.

After the car comes to a stop the safety gear gibs continue to grip the rails and hold the car stationary. The safety gear may be released by lifting the car in the upward direction. This reverses the wedging action and the gibs slide back into their original position; no readjusting is necessary to make the safety gear ready for another operation.

This type of safety gear offers the following features:

(a) Smooth sliding movement of the car after actuating the safety gear.
(b) Because of the wedging action of the gibs the final stopping of the car is independent of the governor rope and operating mechanism.
(c) Short response time due to a simple and effective operating mechanism.
(d) Because of a large contact area no damage is caused to the gibs and guide-rails.
(e) Easy release; no resetting is necessary.
(f) Easy alignment of the gibs and adjustment of the operating mechanism.

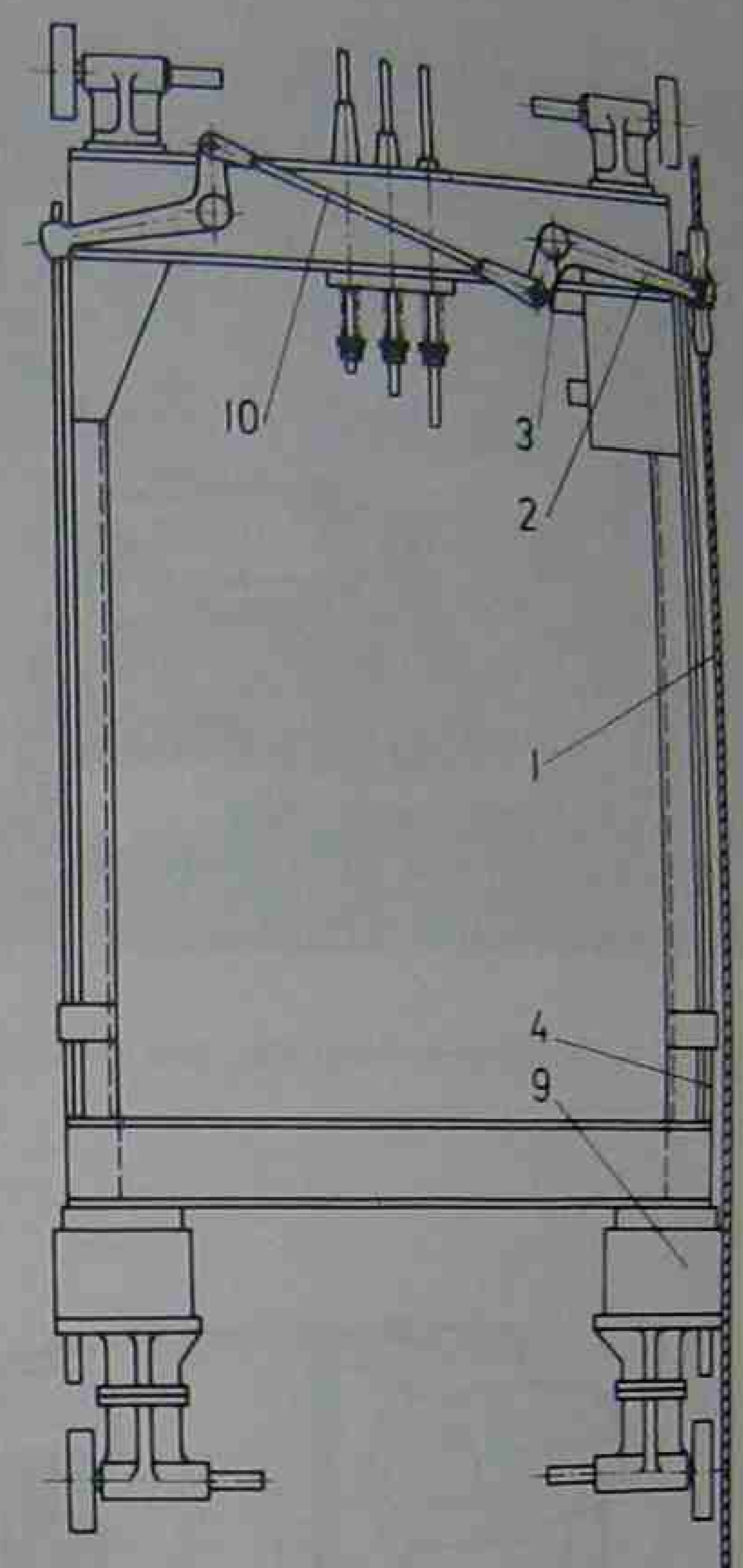


Fig. 8.12 — Operating mechanism of flexible guide clamp safety gear. 1. Governor rope; 2 & 5. Lever; 3. Shaft; 4 & 6. Operating rods; 7 & 8. Gibs; 9. Clamp; 10. Connecting rod.

Flexible guide clamp safety gear incorporating a set of Belleville springs instead of a conventional helical spring is shown in Fig. 8.13.

In recent years a progressive safety gear of roller type has become popular in Europe, particularly on a.c. passenger elevators. The diagram in Fig. 8.14 shows the principal components of one safety clamp. The tapered portion of the clamp has been replaced by two flat springs, forming the track for a hardened knurled steel roller actuated directly by an operating rod. On being lifted the roller engages with the guide-rail and wedges itself between the guide-rail and the springs. The pressure exerted upon the guide-rail is

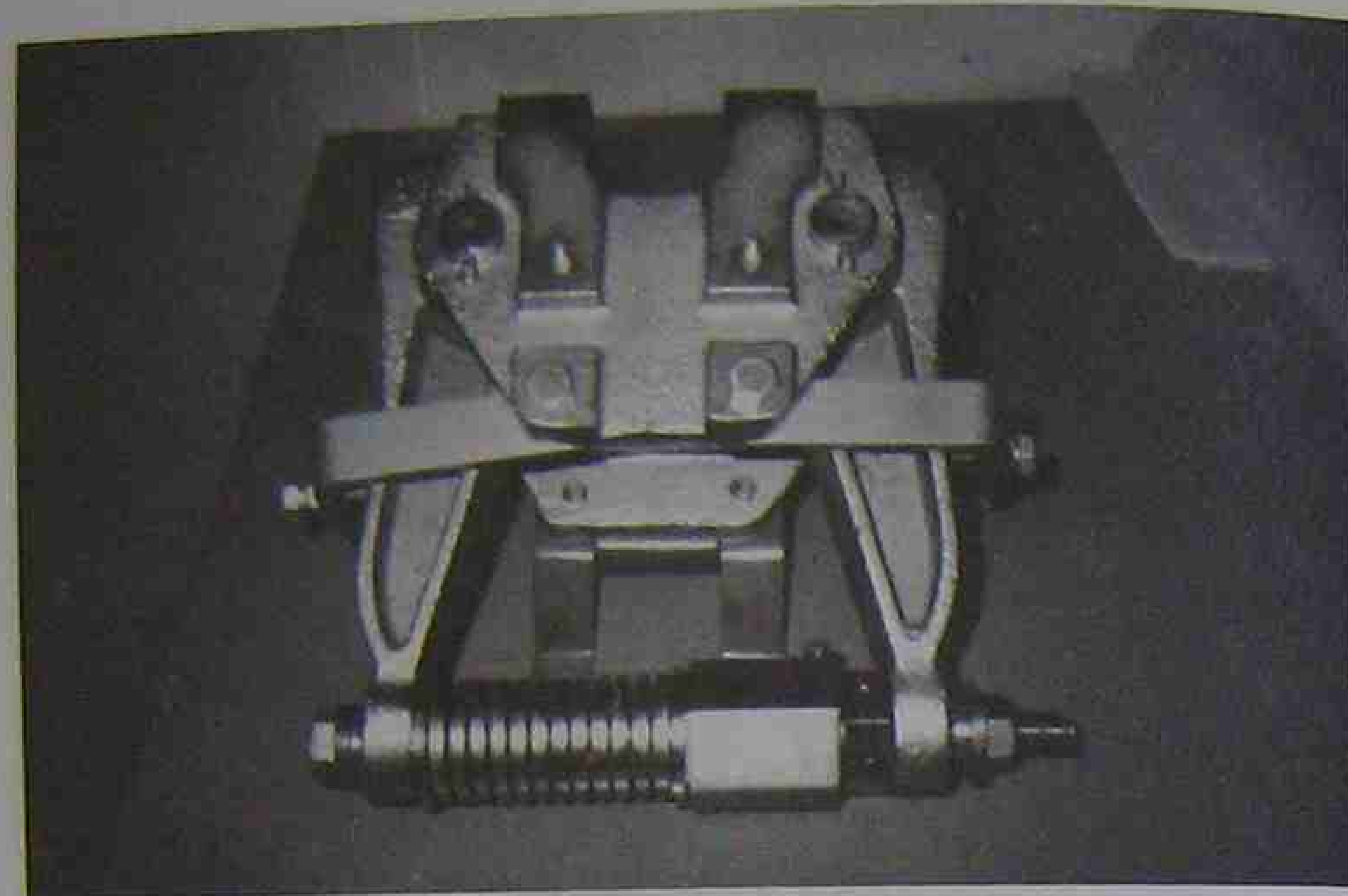


Fig. 8.13 — Flexible guide clamp safety gear (Schindler Management AG).

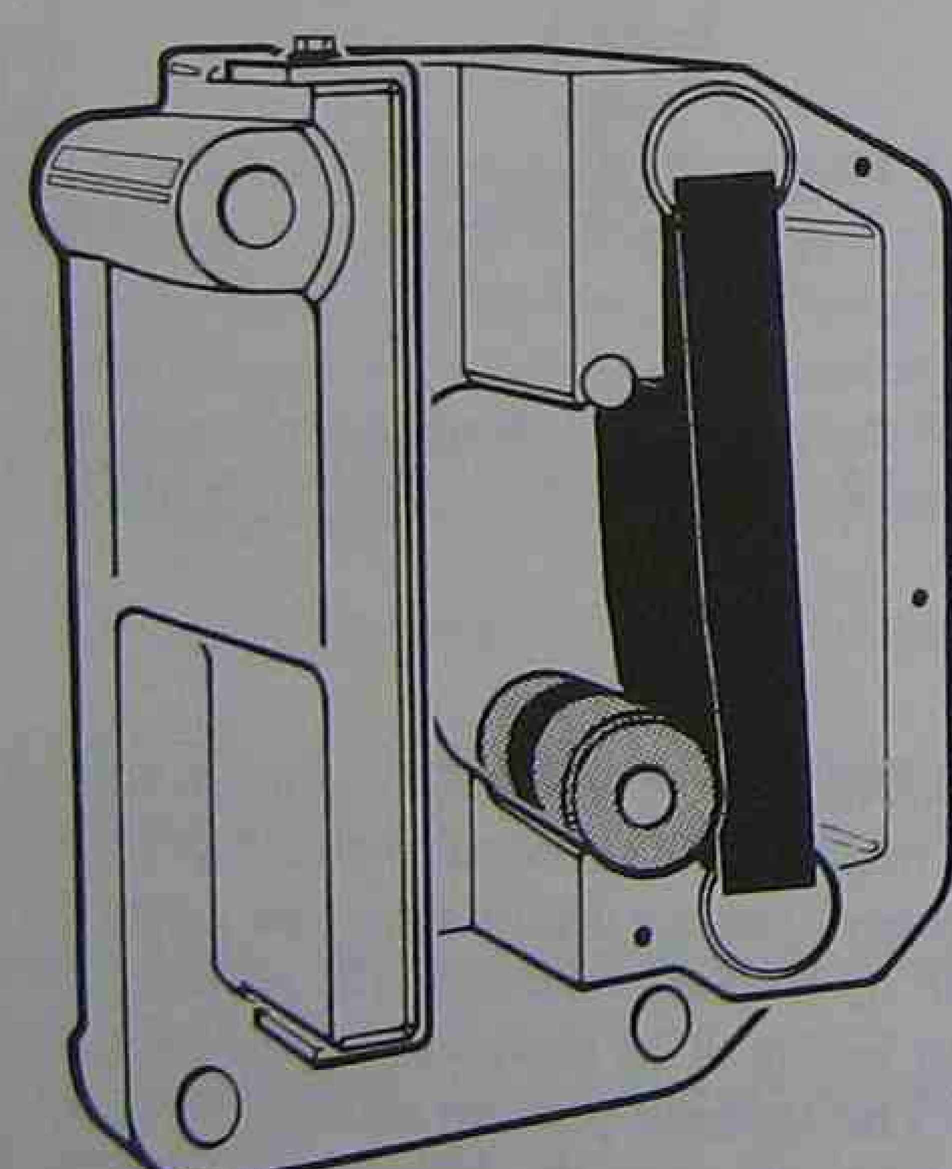


Fig. 8.14 — Progressive safety gear of roller type (Otis Elevator Co.).

controlled by the springs; however, it is desirable that the stiffness of the springs is not too high as the contact stress might be excessive because of the small contact area and might cause a damage to the guide-rail.

9

Buffers

9.1 SPECIFICATION

Elevators must be equipped with buffers located in the pit at the bottom limit of travel for both cars and counterweights, to constitute the final emergency device. If the buffers are attached to the car or counterweight a pedestal at least 0.5 m high must be provided at the end of the travel. Pedestals are not required for the counterweight buffers if it is impossible to gain involuntary access under the counterweight.

Positive drive elevators must also be provided with buffers on the car top to function at the upper limit of travel. If a counterweight is provided, the car buffers must not function until the counterweight buffers are fully compressed.

There are two principal types of buffers in existence:

(1) Energy accumulation type
(2) Energy dissipation type

(1) The Energy accumulation buffer type with or without buffered return movement can be used for rated speeds up to 1.0 m/s or 1.6 m/s respectively.

The total possible stroke must not be less than twice the gravity stopping distance corresponding to 115% of the rated speed v, i.e.

$$2 \times \frac{(1.15v)^2}{2g_n} \approx 0.135v^2 \tag{9.1}$$

and in no case less than 65 mm.

The stroke must be covered under a static load of between 2.5

and 4 times the mass of the car plus its rated load (or the mass of the counterweight) in Europe, whilst the multiples are 2 and 3 in the USA.

(2) The Energy dissipation buffer type may be used irrespective of the rated speed of the elevator. The total possible stroke must be at least equal to the gravity stopping distance corresponding to 115% of the rated speed, i.e. $0.0674v^2$.

Reduced stroke buffers may be used when a device is provided checking that the slowdown of the machine is effective before arrival at terminal landings and if necessary causing the car speed to be reduced in such a way that the speed at which the car comes into contact with the buffers is not in excess of that for which the buffers were designed. This speed may be used instead of the rated speed when calculating the buffer travel; however, the stroke must be at least:

(a) 50% of $0.0674v^2$, if the rated speed v does not exceed 4 m/s.
(b) 33% of $0.0674v^2$, if the rated speed v exceeds 4 m/s.

In any event the stroke must not be less than 420 mm.

With the rated load in the car, in the case of free fall, and with the speed of impact being equal to that for which the buffer is calculated, the average retardation must not exceed g_n. Retardation of more than $2.5g_n$ must not last longer than 0.04 s.

Strokes required for particular types of buffers in relation to the rated speeds are illustrated in Fig. 9.1.

9.2 SPRING BUFFERS (ENERGY ACCUMULATION)

The principal component of an energy accumulation buffer is usually a helical spring made of either round or square wire. Volute springs are rare nowadays.

A typical buffer is illustrated in Fig. 9.2.

A buffer composed of three helical springs in parallel is shown in Fig. 9.3. It is convenient to use two or three helical springs in parallel where the retarding force is to be increased, as the height of the buffer is less than with a single spring of the same characteristic.

The buffer in Fig. 9.4 consists of two helical springs of square section, each welded to a steel baseplate and mounted on a fixing plate. Registration pieces for the guide-rails may be seen at each end of the fixing plate.

The determination of the dimensions of a spring buffer is accomplished on the basis of the strength analysis. As a rule helical springs,

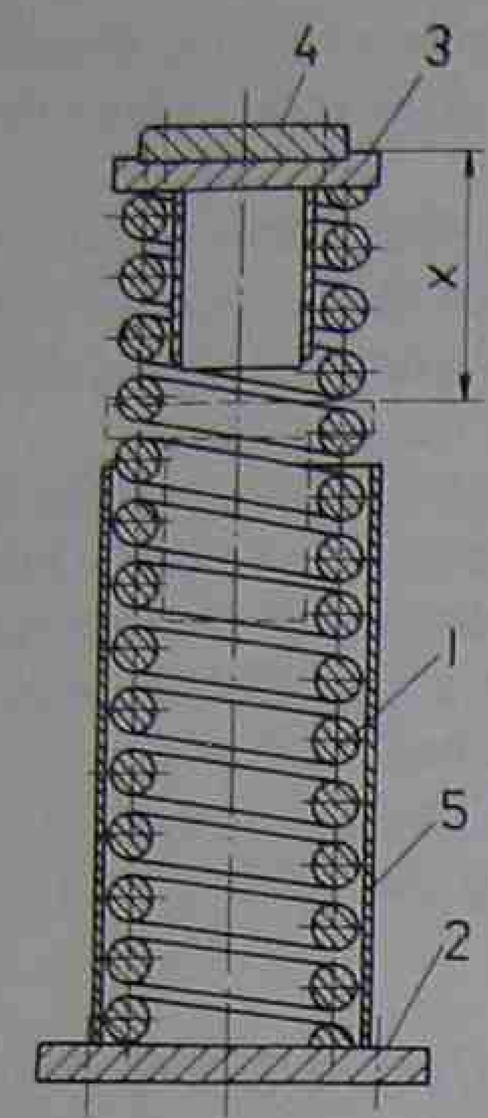


Fig. 9.2 — Helical spring buffer of round section.

1. Helical spring of round wire
2. Lower buffer plate
3. Upper buffer plate
4. Rubber contact block for absorbing first impact
5. Cylinder for guiding the spring

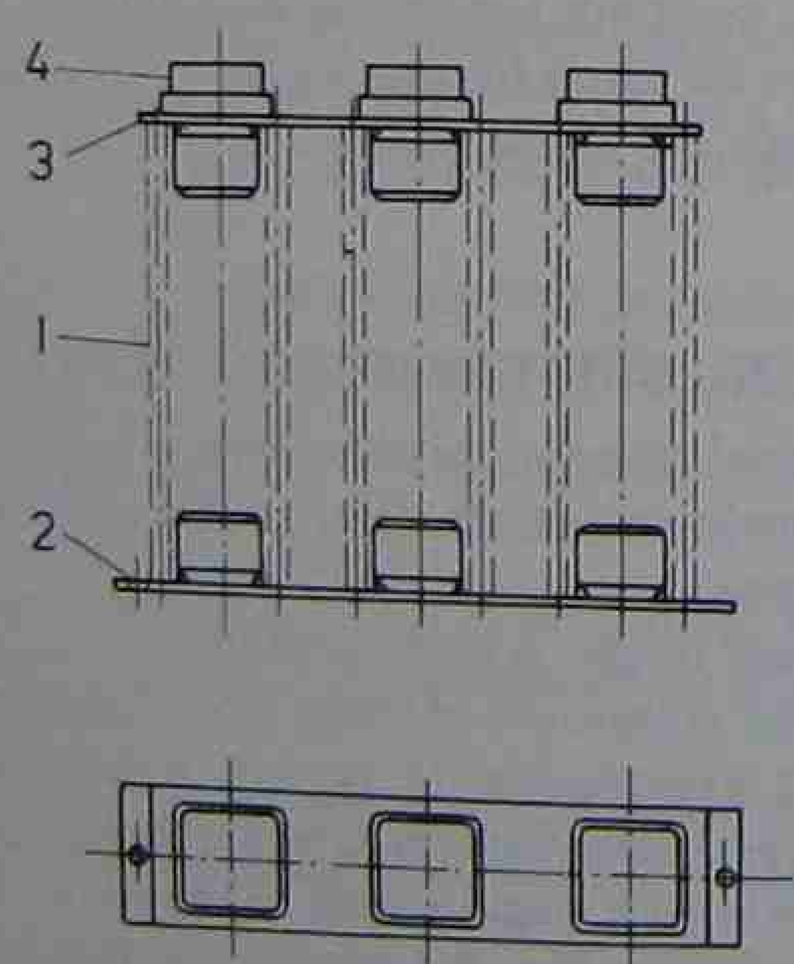


Fig. 9.3 — Triple spring buffer.

1. Springs
2. Lower buffer plate
3. Upper buffer plate
4. Rubber bumpers

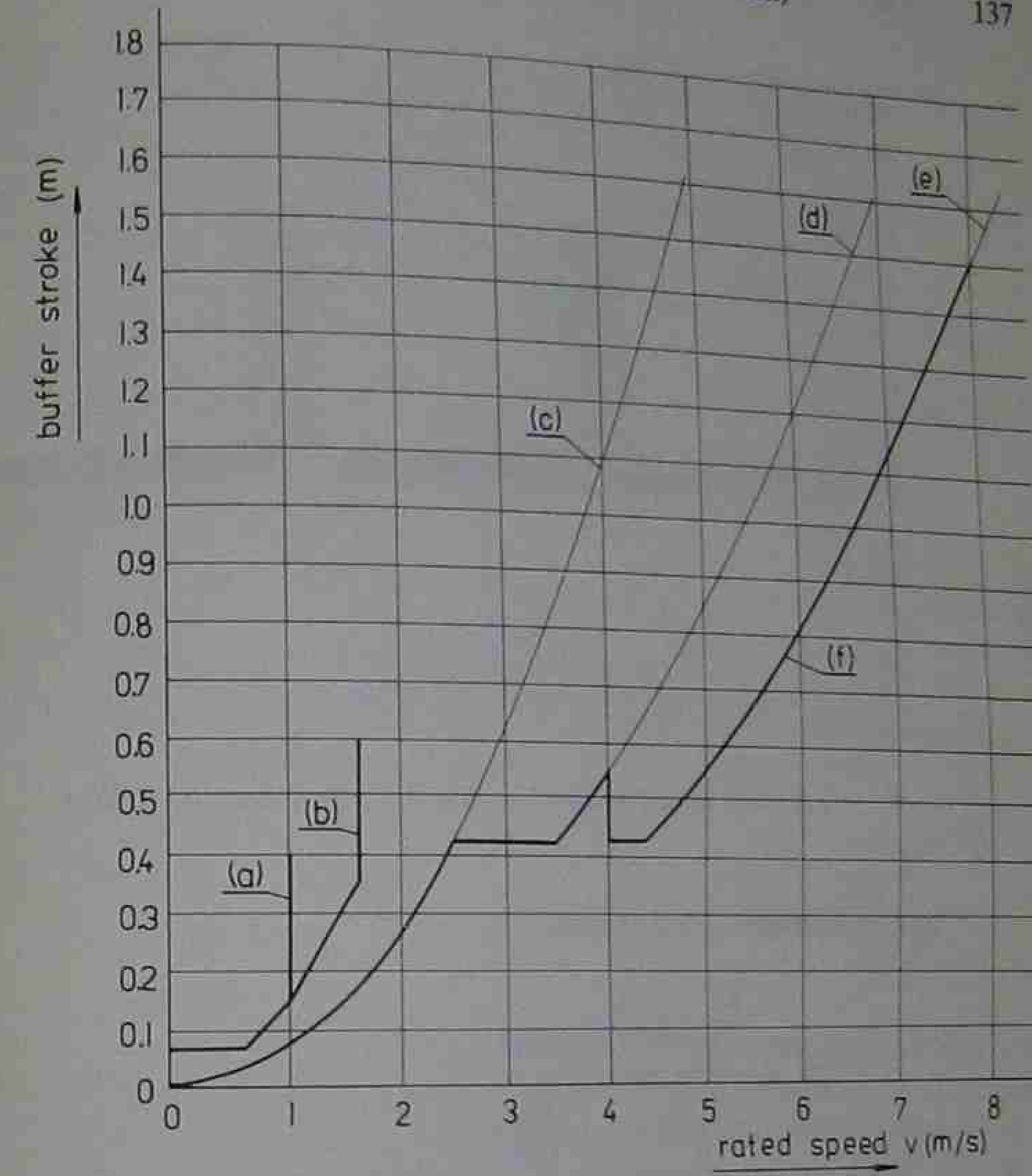


Fig. 9.1 — Graph illustrating the strokes required for buffers. (a) Energy accumulation; (b) energy accumulation with buffered return movement; (c) energy dissipation without reduction in stroke; (d) energy dissipation with reduction of 50%; (e) energy dissipation with reduction of 33%; (f) minimum stroke possible when advantage is taken of all the possibilities.

made of round steel wire, with a constant angle of lead of all active threads are used. Only the stress in torsion is taken into consideration as the effect of other kinds of stress is negligible.

A method of calculation is applied assuming an even distribution of stress over the cross-sectional area of the wire. Since in fact the stress increases with the decreasing distance from the axis of the spring, and it achieves its maximum value at the point nearest to the axis, a correction must be made using the so-called Wahl's coefficient ψ. The value of ψ depends upon the ratio of D_s/d and is given

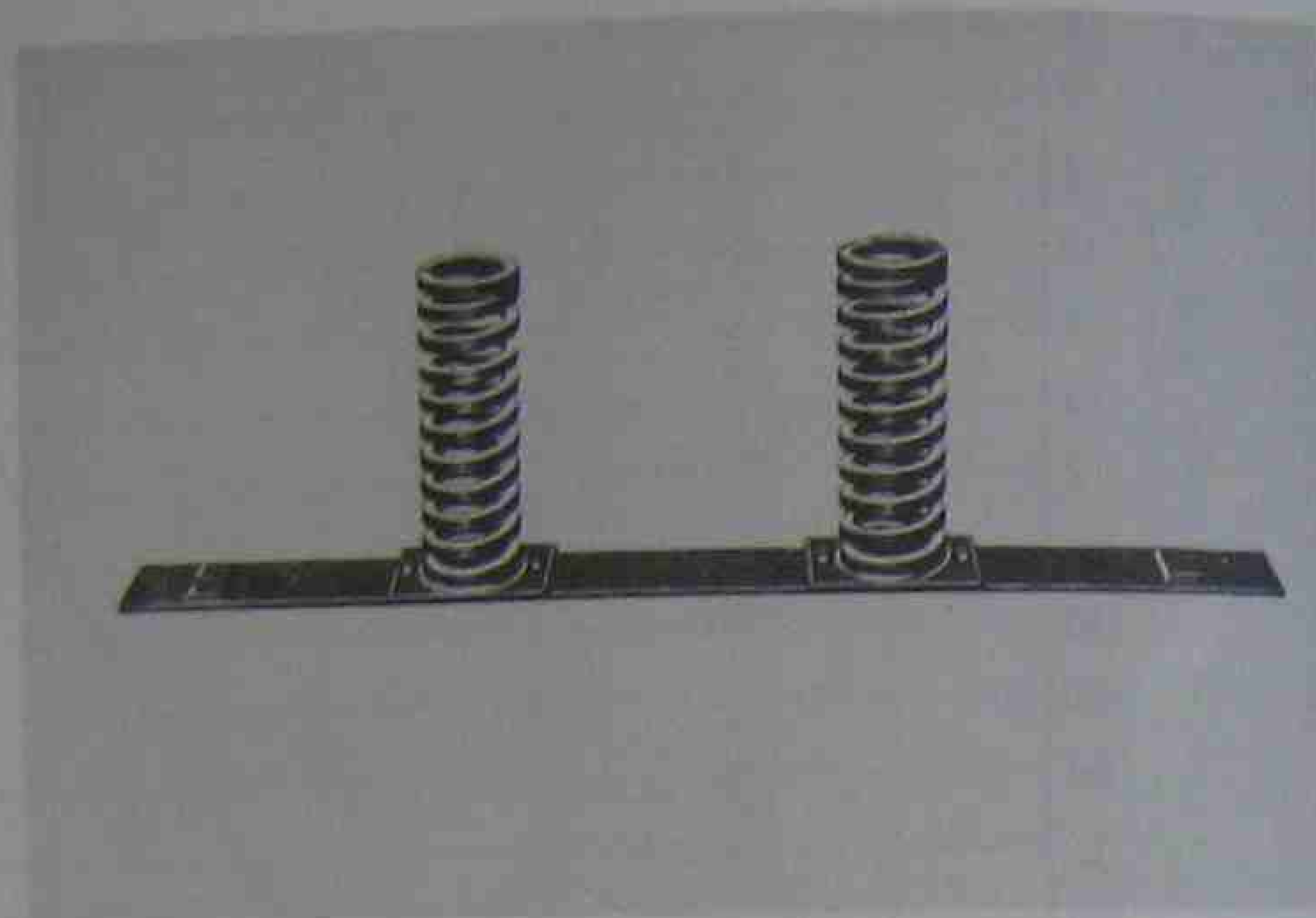


Fig. 9.4 — Helical spring buffers of square section (Express Lift Co.).

by the formula

$$\psi = \frac{\frac{D_s}{d} - 0.25}{\frac{D_s}{d} - 1} + \frac{0.615}{\frac{D_s}{d}} , \tag{9.2}$$

where D_s is the central diameter of the spring (mm) and d is the diameter of the wire (mm).

The relation between ψ and D_s/d is illustrated by a graph, Fig. 9.5, so that the value of ψ can be easily obtained.

The calculation procedure is as follows:

(1) The value of D_s/d must be estimated and coefficient ψ can be obtained from Fig. 9.5.
(2) Determination of the maximum permissible stress in torsion τ_p. Usually,

$$\tau_p \approx 0.28\sigma_{ts} \quad (\text{N/mm}^2) , \tag{9.3}$$

where σ_{ts} is the tensile strength of the wire (N/mm²).

Fig. 9.5 — Diagram for Wahl's coefficient.

The value of the permissible stress in each individual case may be determined greater or less than quoted above in dependence upon the mode of loading, quality and heat treatment of the material, environmental conditions and required life of the spring, on the basis of either experimental tests or the experience of the designer.

(3) From the equation of strength in torsion

$$\frac{8F_{max} \times D_s \times \psi}{\pi \times d^3} \leqq \tau_p , \tag{9.4}$$

where F_{max} is the force in case of the spring being fully compressed (N), the diameter of the wire can be expressed:

$$d \geqq \sqrt[3]{\frac{8F_{max} \times \psi}{\pi \times \tau_p} \times \frac{D_s}{d}} \quad (\text{mm}) . \tag{9.5}$$

In compliance with the specification for energy accumulation buffers (see section 9.1) the maximum force of the spring buffer must not exceed the value of:

$$F_{max} = 4(Q+K) \times g_n \quad \text{in Europe} \tag{9.6a}$$

or

$$F_{max} = 3(Q+K) \times g_n \quad \text{in the USA} \tag{9.6b}$$

The ratio of D_s/d was estimated in item (1.)

(4) From the ratio of D_s/d the central diameter of the spring D_s can be calculated.

(5) Determination of the number of active threads of the spring.

The compression of the spring x can be expressed both from the equation

$$F = c \times x \quad \text{(N)} , \tag{9.7}$$

where c is the stiffness of the spring (N/mm), and the equation for the angle of torsion ϕ

$$x = \phi \times \frac{D_s}{2} . \tag{9.8}$$

The angle of torsion

$$\phi = \frac{M_t \times l}{G \times J_p} \quad \text{(rad)} , \tag{9.9}$$

where M_t is torsional moment (Nmm), l is the length of the spring exposed to torsion (mm), G is the modulus of elasticity in torsion (shear) (N/mm^2), and J_p is the polar moment of inertia of the wire cross-section (about the perpendicular axis at the centre of gravity of the sectional area) (mm^4).

As is generally known

$$M_t = \frac{F \times D_s}{2} \quad \text{(Nmm)} , \tag{9.10}$$

$$l = \pi \times D_s \times n \quad \text{(mm)} , \tag{9.11}$$

where n is the number of active threads of the spring,

$$J_p = \frac{\pi \times d^4}{32} \quad \text{(mm}^4\text{)} \tag{9.12}$$

After the substitution in equation (9.8):

$$x = \frac{8F \times D_s^3 \times n}{G \times d^4} \tag{9.13}$$

By the combination of equations (9.7) and (9.13):

$$\frac{F}{c} = \frac{8F \times D_s^3 \times n}{G \times d^4} \tag{9.14}$$

and hence the number of active threads

$$n = \frac{G \times d^4}{8D_s^3 \times c} . \tag{9.15}$$

(6) Determination of the stiffness of a spring.

The stiffness of the spring c, which is necessary to be known before the calculation of n can be carried out, can be determined on the basis of the specification quoted in section 9.1. The minimum stroke is specified as $h_{min} = 0.135v^2$ and the maximum retarding force of the buffer should be between $2.5(Q+K) \times g_n$ and $4(Q+K) \times g_n$. The characteristic of helical springs (diagram force to compression) is linear. In Fig. 9.6 lines 1 and 2

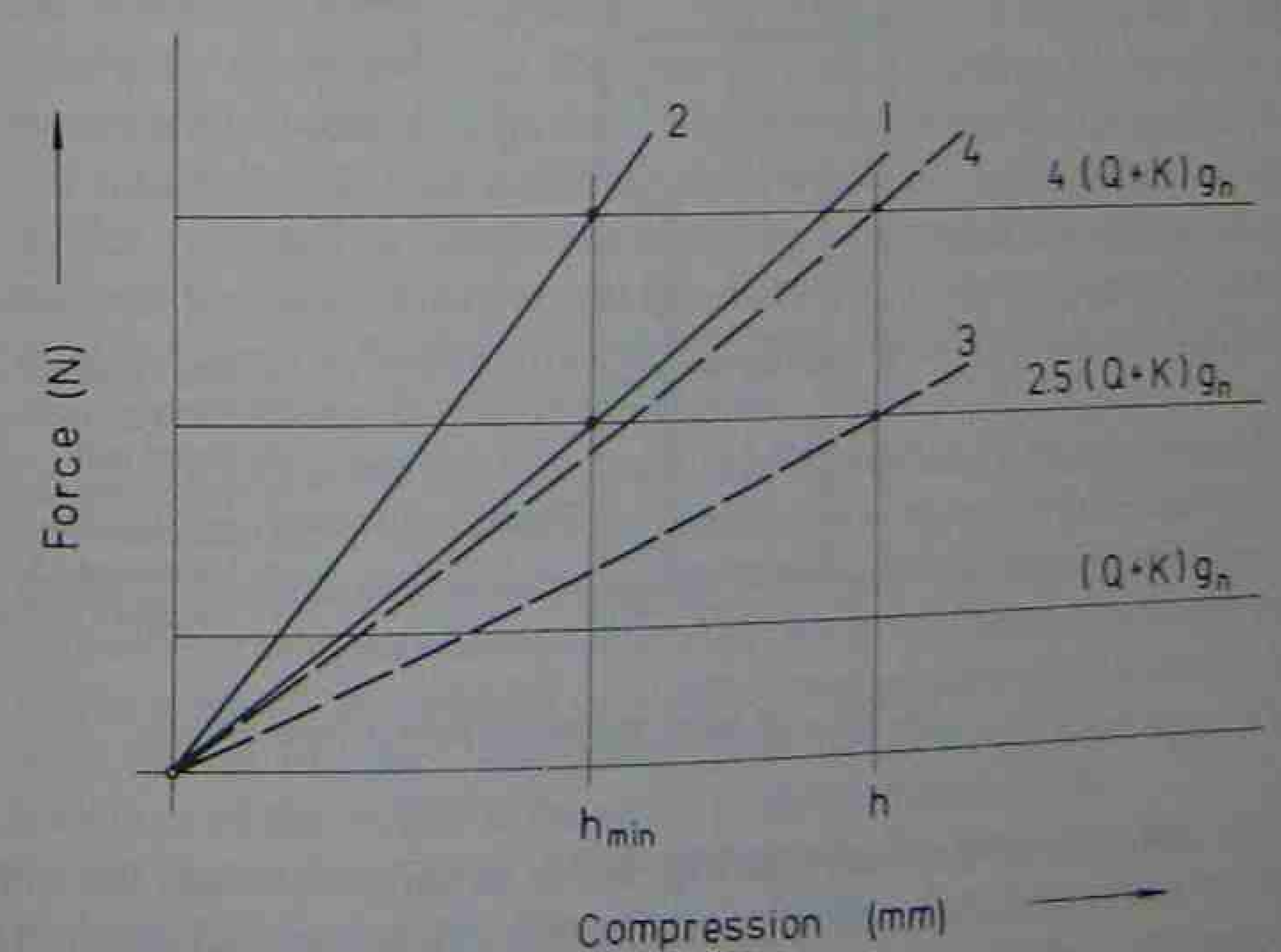


Fig. 9.6 — Diagram force to compression for spring buffers.

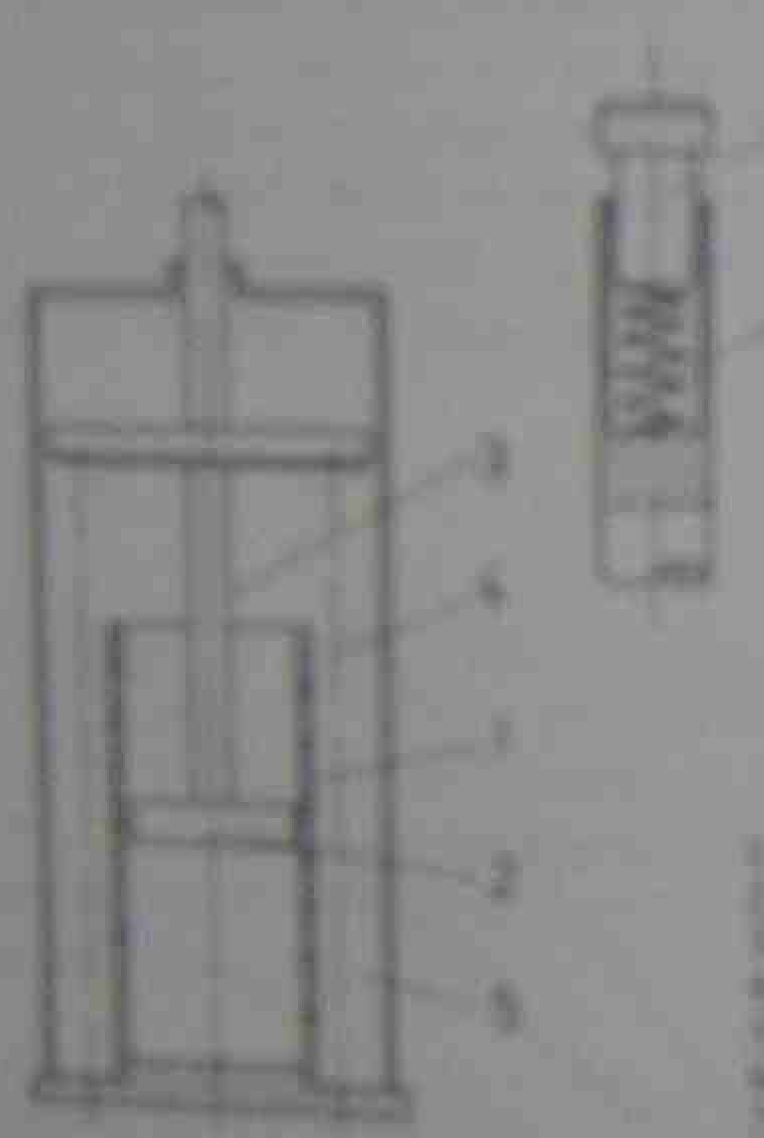
correspond to the limit forces at the minimum stroke, while they convert to lines 3 and 4 when a longer stroke is taken into consideration; the stiffness of the spring is then decreased.

9.3 OIL BUFFERS (ENERGY DISSIPATION)

In contrast to the progressive retarding force of spring buffers, oil buffers can be designed to induce a constant force during the stopping action, resulting in constant retardation.

Although the construction of oil buffers may be different in detail, the general principle is retained. The buffer must be capable of converting the kinetic energy of the car (counterweight) at the instant of impact into heat as well as the potential energy due to the loss of level, equal to the stroke of the buffer.

Schematic diagrams of oil buffers of different design are shown in Figs. 9.7, 9.8 and 9.9. All of them are of spring return type. The



1. Inner cylinder
2. Piston
3. Piston rod
4. Compression spring
5. Oil reservoir
6. Contact block
7. Ancillary spring

Fig. 9.7 — Oil buffer with escape holes in inner cylinder.

function of the buffer in Fig. 9.7 is as follows: when the car overtravels the lower landing, it strikes the rubber contact block (6) at the top of the piston rod (3), which together with the ancillary spring (7) absorbs the first impact. Further descent of the car drives the piston

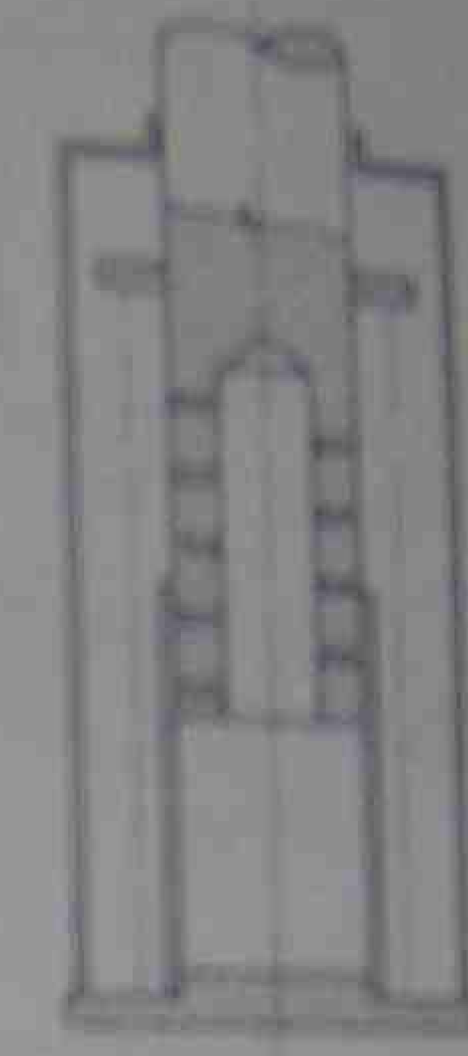


Fig. 9.8 — Oil buffer with escape holes in plunger.

(2) into the oil-filled inner cylinder (1), provided with a series of escape holes. The oil is forced through the escape holes into the outer oil reservoir (5), the number and total size of the holes being gradually decreased and consequently the restriction to the oil flow increases producing sufficient oil pressure to retard the motion of the car and to bring it to a smooth stop. When the car is lifted and the buffer released, a compression spring (4) returns the piston to its normal position. This permits the oil to flow from the reservoir back into the inner cylinder through the escape holes so that the buffer is automatically ready to function again.

In Fig. 9.8 a similar construction is illustrated. A plunger provided with a chamber at the lower end and a series of escape holes in its wall is the principal component of the buffer. The hydraulic cylinder is smooth without any holes. When the plunger is forced downward the upper edge of the cylinder gradually overlaps the escape holes in the wall of the plunger, decreasing their number and total size. The function of the buffer is the same as of the previous one.

In Fig. 9.9 a rather different construction is shown. When the plunger (2) is forced downward into the oil cylinder (1) by the descending car or counterweight, the oil escapes through an annular orifice into the chamber inside the hollow plunger, the rate of flow

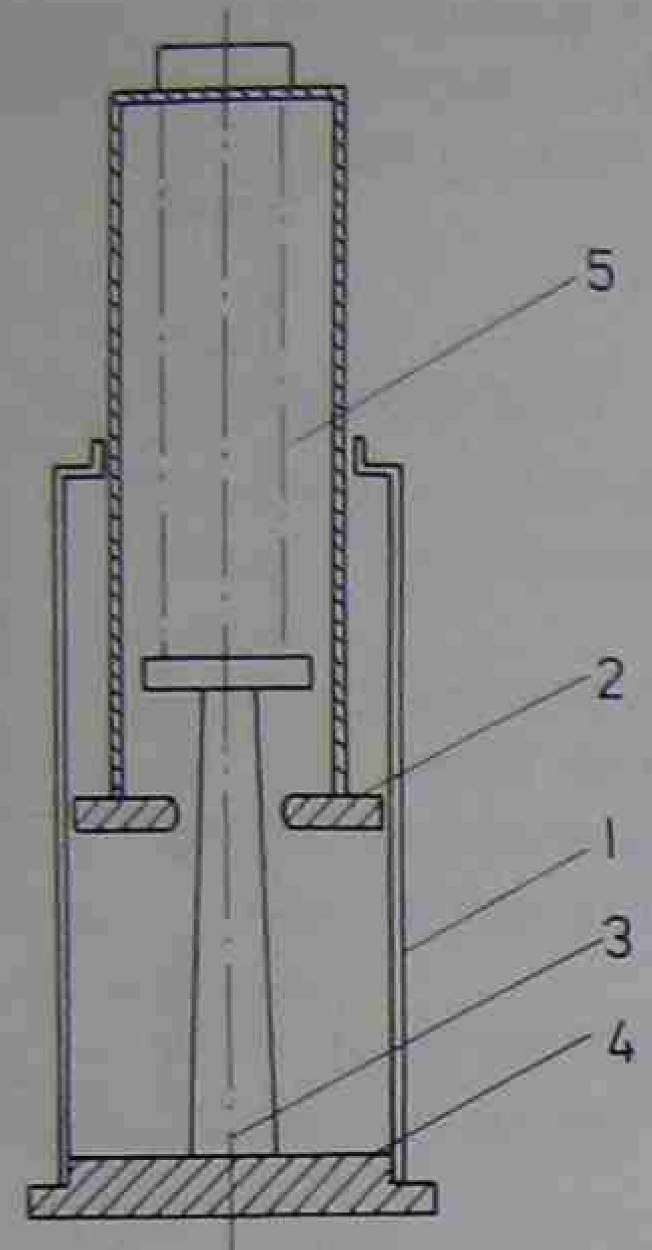


Fig. 9.9 — Oil buffer with tapered steel pin and annular orifice.
1. Hydraulic cylinder
2. Plunger
3. Tapered steel pin
4. Lower buffer plate
5. Return spring

being controlled by the tapered pin (3). The orifice is gradually reduced in size as the plunger descends resulting in a constant retarding force. A compression spring (5) inside the plunger returns the plunger to its operating position after the car or counterweight have been lifted.

An oil buffer for shallow pits is shown in Fig. 9.10. The car (counterweight) does not operate the piston of the buffer directly, but by means of a double lever, which results in a reduced stroke of the buffer (less than a half of the car trajectory).

The buffer should be so designed that the retardation a of the car (counterweight) after striking the buffer is approximately constant and equal to the standard acceleration of free fall g_n.

The equation of continuity representing the equality of volumes of oil displaced by the piston and forced through the escape holes in the cylinder is

$$v \times S_p = w \times q_y \times \mu \ , \qquad (9.16)$$

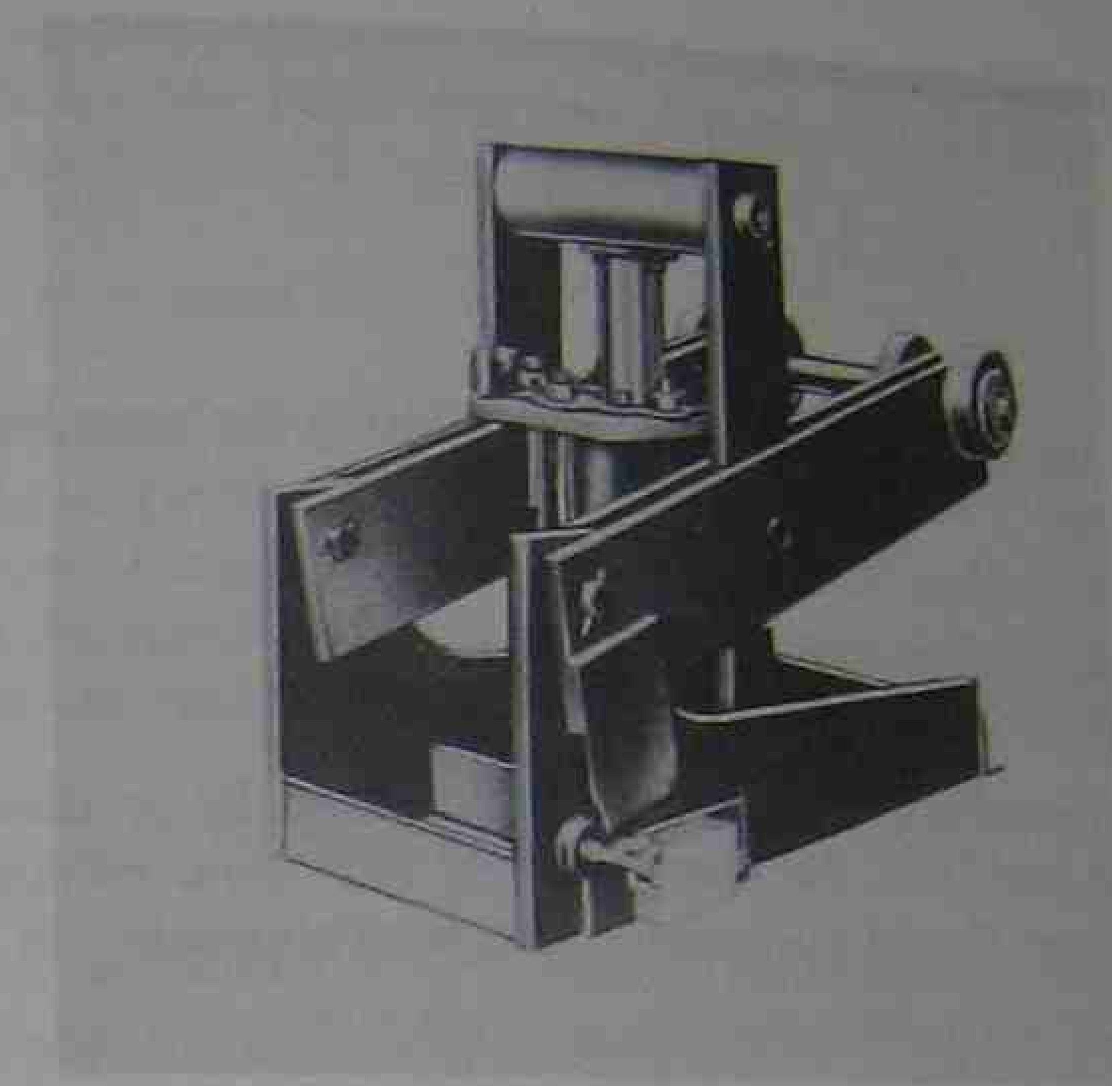


Fig. 9.10 — Oil buffer of lever type for shallow pits (Moline Accessories Co.).

where v is the velocity of the piston (m/s), S_p is the piston area (m^2), w is the discharge velocity (m/s), q_y is the total area of escape holes below the piston (m^2) and μ is the discharge coefficient.

The discharge velocity is:

$$w = \sqrt{\frac{2p}{\gamma}} \ \text{(m/s)} \ , \qquad (9.17)$$

where p is specific oil pressure in the cylinder (N/m^2) and γ the oil density (kg/m^3).

A formula for the specific pressure is:

$$p = \frac{\gamma}{2\mu^2} \times S_p^2 \times \left(\frac{v}{q_y}\right)^2 \qquad (9.18)$$

Hence the retarding force:

$$F = p \times S_p = \frac{\gamma}{2\mu^2} \times S_p^3 \times \left(\frac{v}{q_y}\right)^2 \qquad (9.19)$$

Substitution in equation (9.19) for $v = \sqrt{2g_n \times y}$, corresponding to free fall, produces:

$$F = \frac{\gamma \times g_n}{\mu^2} \times S_p^3 \times \frac{y}{q_y^2} \tag{9.20}$$

In the event of F being constant during the stopping action of the buffer

$$y = \text{const.} \times q_y^2 \tag{9.21}$$

Equation (9.21) formulates the parabolic relationship between both variables; the distribution of individual escape holes in the cylinder should correspond to this relationship. In Fig. 9.11 the

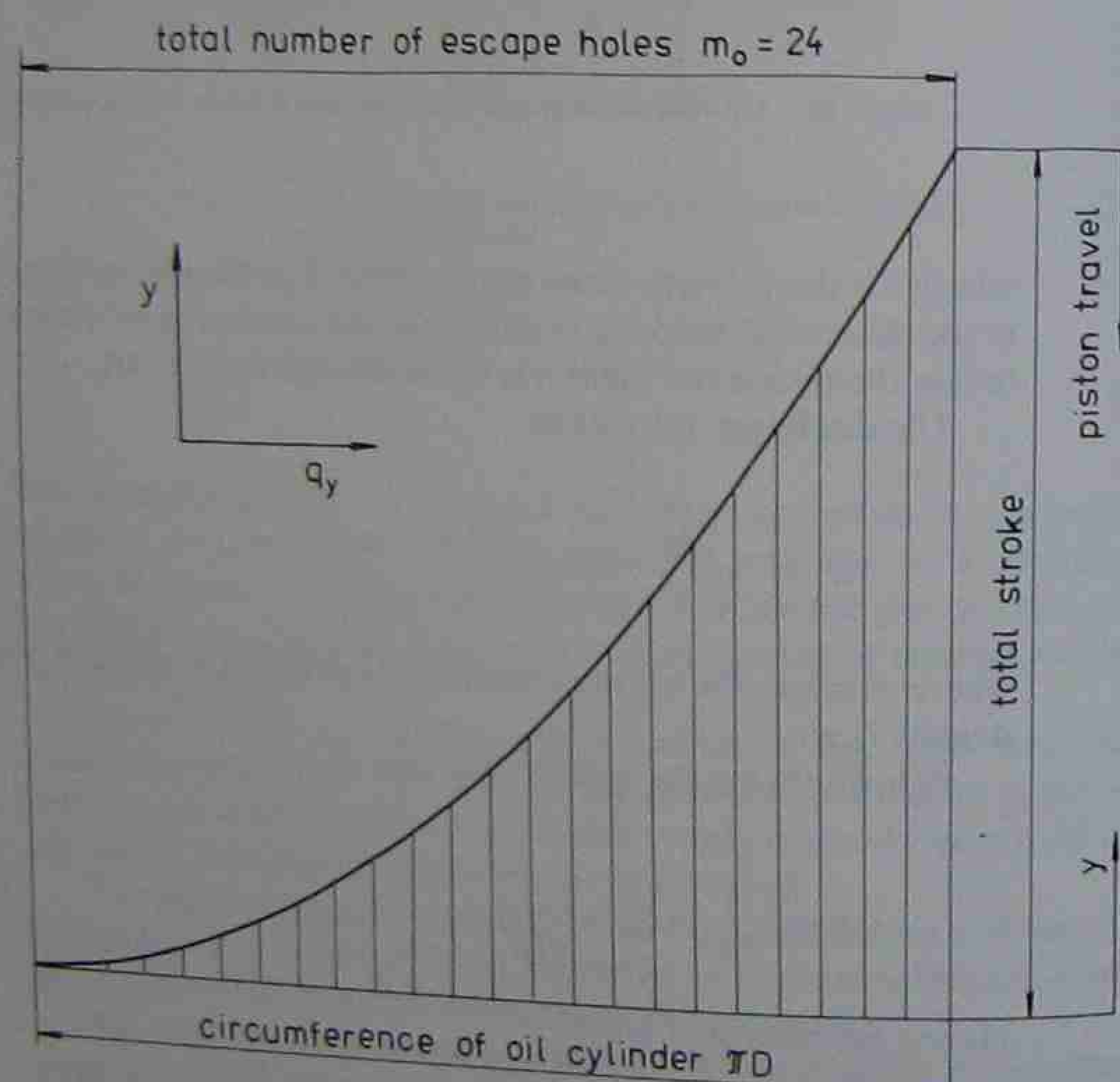


Fig. 9.11 — Distribution of escape holes in the oil cylinder.

cylinder surface has been developed in a plane and the location of 24 escape holes has been indicated. Equation (9.21) is of general validity, so that it is also possible to write

$$h = \text{const} \times q_o^2 \tag{9.22}$$

where h is total stroke of the buffer (m) and q_o the total area of all escape holes (m^2).

As seen in equation (9.19) the initial retarding force F_o depends only on the speed of impact v_o (initial speed) and does not depend upon the car load. Its value may be easily calculated for $a = g_n$

$$F_o = (Q + K) \times (g_n + a) = 2(Q + K) \times g_n \tag{9.23}$$

If the car rated load Q be altered, the retarding force F will vary accordingly. In Fig. 9.12 the original constant line (i) for the total load

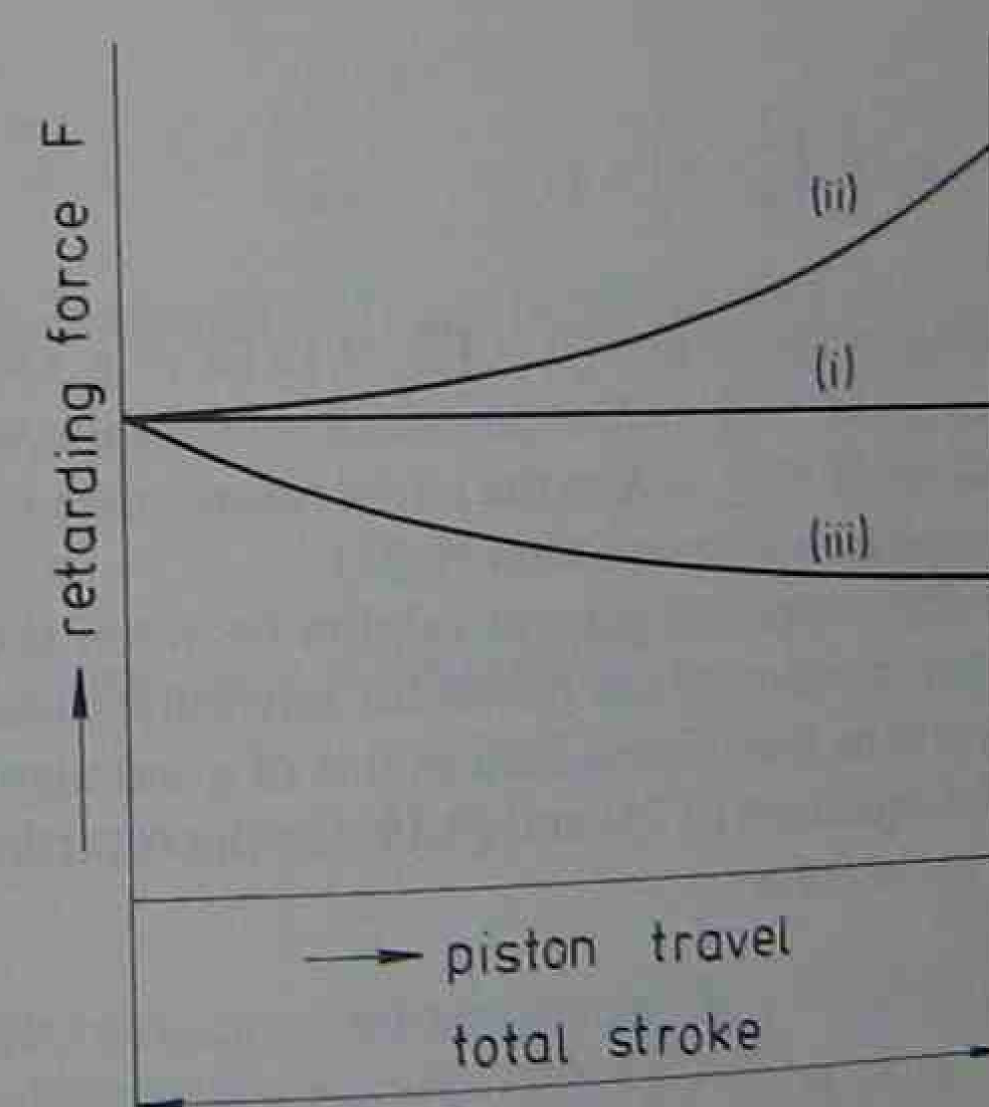


Fig. 9.12 — Diagram retarding force to piston travel.

$M = Q + K$ will convert to (ii) for $M > Q + K$ or (iii) for $M < Q + K$, as the mechanical work of the retarding force must be equal to the sum of the kinetic energy of the car (counterweight) at the moment of impact and the potential energy corresponding to the total stroke of the buffer.

The variation of the rate of retardation is depicted in Fig. 9.13. If the total load is indicated:

$$M = \theta \times (Q + K) \qquad (9.24)$$

where θ is a coefficient; then the retarding force

$$F = M \times (g_n + a) = \theta \times (Q + K) \times (g_n + a) \qquad (9.25)$$

and the retardation

$$a = \frac{F}{\theta \times (Q + K)} - g_n \qquad (9.26)$$

Having substituted in equation (9.25) for $F = F_o$ from equation (9.23) a formula for the initial retardation is obtained

$$a_o = \left(\frac{2}{\theta} - 1\right) \times g_n \qquad (9.27)$$

Consequently the line (i) in Fig. 9.13 ($a = g_n$) will convert to the curve (ii) for $M > Q + K$ as the initial retardation will be less than g_n or to (iii) for $M < Q + K$ as the initial retardation will be greater that g_n in compliance with equation (9.27).

The dependence of general validity between the retarding force and instant position of the piston for any buffer load and any initial velocity will be formulated here as it is of great significance.

Initial equations (9.25) and (9.19) for the retarding force F can be written in the form

$$F = M\left(g_n + v \times \frac{dv}{dy}\right) \qquad (9.28)$$

and

$$F = C \times \left(\frac{v}{q_y}\right)^2 \quad \text{respectively.} \qquad (9.29)$$

Similarly a formula for F_o can be obtained, namely:

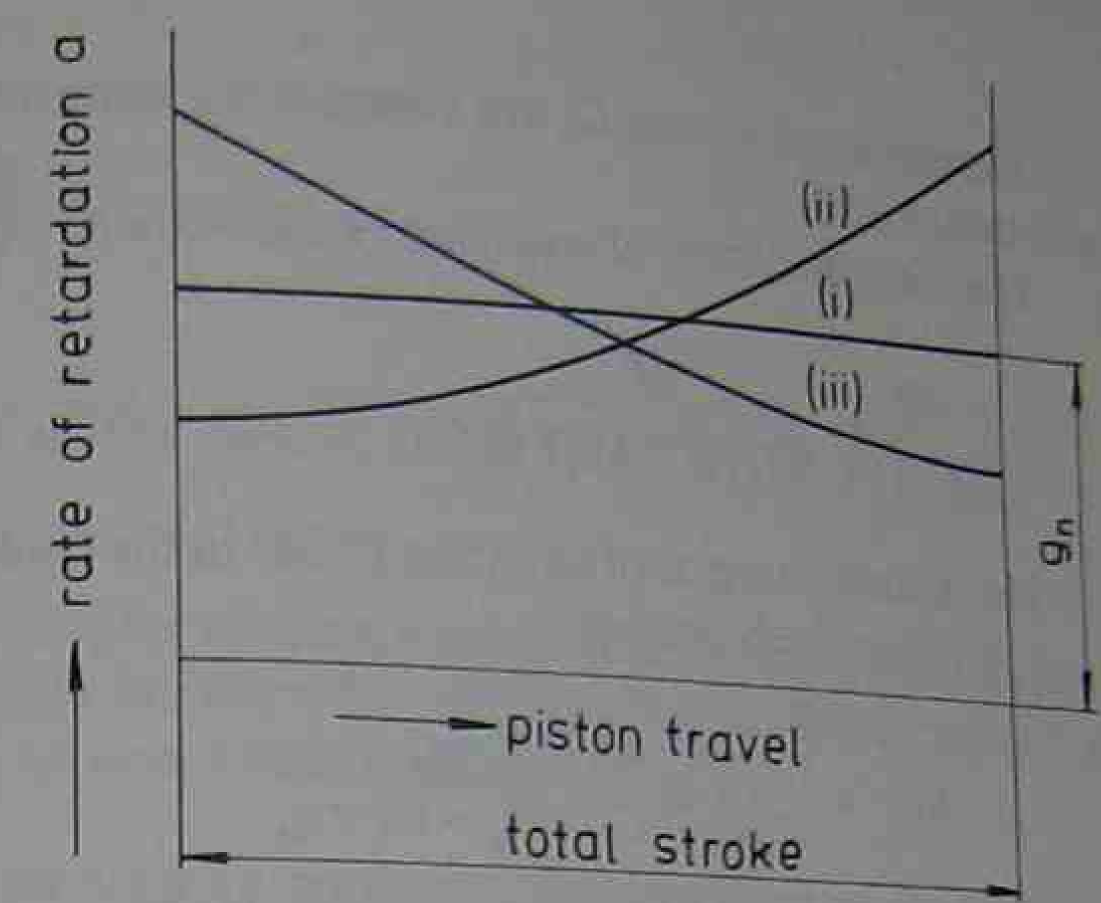


Fig. 9.13 — Diagram rate of retardation to piston travel.

$$F_o = C \times \left(\frac{v_o}{q_o}\right)^2$$

Utilizing equations (9.21) and (9.22) the ratio F/F_o can be obtained:

$$\frac{F}{F_o} = \frac{v^2}{v_o^2} \times \frac{h}{y} \qquad (9.30)$$

and after rearranging this equation

$$y \times F = F_o \times \frac{h}{v_o^2} \times v^2 \qquad (9.31)$$

By differentiation and substitution for $v \times (dv/dy)$ from equation (9.28), a differential equation is obtained:

$$\frac{y}{dy} = \frac{1}{dF}\left[\left(\frac{F_o}{M} \times \frac{2h}{v_o^2} - 1\right) \times F - \frac{2g_n \times h \times F_o}{v_o^2}\right] \qquad (9.32)$$

For the purpose of simplification take

$$C_1 = \frac{2h \times F_o}{v_o^2} \quad \text{and} \quad C_2 = \frac{C_1}{M} - 1\,. \qquad (9.33)$$

The values of C_1 and C_2 are constant for definite initial velocity and buffer load.

The simplified form of equation (9.32) is then:

$$\frac{dy}{y} = \frac{dF}{C_2 \times F - C_1 \times g_n} \tag{9.34}$$

The definite integral of equation (9.34) in the limits of $h - y$ and $F_o - F$ respectively leads to:

$$\ln\frac{y}{h} = \frac{1}{C_2} \times \ln\frac{C_2 \times F - C_1 \times g_n}{C_2 \times F_o - C_1 \times g_n} \tag{9.35}$$

After removing the logarithms and raising the equation to the power C_2 the final formula for the retarding force F becomes:

$$F = \frac{(C_2 \times F_o - C_1 \times g_n) \times \left(\frac{y}{h}\right)^{C_2} + C_1 \times g_n}{C_2} \tag{9.36}$$

As seen in equation (9.36) the initial force F_o is affected only by the initial velocity v_o, while at the end of the stroke the retarding force F depends especially upon the buffer load M.

A similar relationship between the retardation a and the instant position of the piston can be formulated, utilizing equations (9.27) and (9.36)

$$a = \frac{(C_2 \times F_o - C_1 \times g_n) \times \left(\frac{y}{h}\right)^{C_2} + C_1 \times g_n - C_2 \times M \times g_n}{C_2 \times M} \tag{9.37}$$

10

Elevator well and machine room

In this chapter account is taken of the European Standard EN 81 and the British Standard BS 5655: Part 5 only. Specifications according to A 17.1 have not been included.

10.1 GENERAL REQUIREMENTS FOR ELEVATOR WELLS

An elevator well is a space in which both the car and the counterweight, if there is one, travel. It is usually located inside the building though a number of prefabricated wells were installed on the outside of buildings in some European countries in the past. In the Transamerica Building in San Francisco, the shape of which is of a truncated tetrahedral pyramid of 260 m in height, the elevator wells are located inside the building up to the 29th floor, come out at this level and continue upwards on the outside (Fig. 10.1).

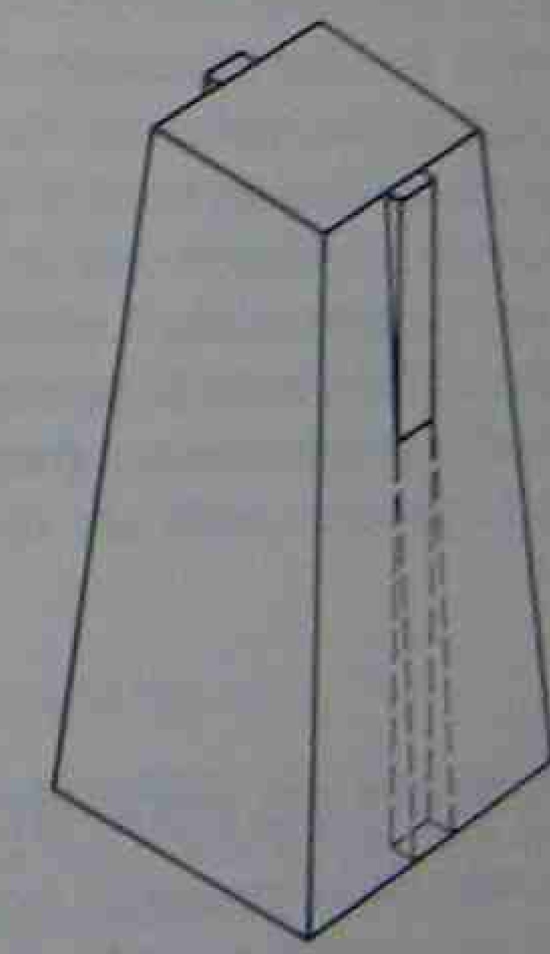


Fig. 10.1 — The Transamerica Building in San Francisco.

If the well is required to contribute to the protection of the building against the spread of fire, it must be totally enclosed by solid walls, floor and ceiling. If not, the well must be provided with walls of a minimum height of 2.5 m above any points accessible to persons; furthermore at the entrance face a mesh or perforated panels must be used from the height of 2.5 m above the level in case the car door is not locked mechanically.

The only permissible openings in the well enclosure are landing doors, inspection and emergency doors to the well, ventilation openings, vent openings for escape of gas and smoke in the event of fire and permanent openings between the well and the machine or pulley rooms. This specification does not relate to elevator wells for observation elevators, one side of which remains open along the total height of travel.

If the elevator car is not provided with doors, the wall facing the car entrance must be so designed that a force of 300 N applied at a right angle to the wall at any point, evenly over an area of 500 mm^2, in a round or square section, will not cause an elastic deformation of the wall exceeding 10 mm.

Elevator wells should not be located above a space accessible to persons. If such a space exists, the base of the pit must be designed for an imposed load of at least 5000 N/m^2. Furthermore, either the counterweight must be equipped with a safety gear or below the counterweight buffer a solid pillar extending down to firm foundations must be installed.

Any equipment that does not belong to the elevator cannot be installed in the well, with the exceptions of fire detectors and heating equipment excluding hot water or steam heating; however, all control and adjustment devices must be located outside the well.

The well must be ventilated, but must not be used to provide ventilation of any portion of the building not determined for the service of elevators. It also must be provided with permanent electric lighting for repairs and maintenance purposes.

Dimensions of elevator wells are specified in BS 5655: Part 5: 1981.

10.2 WELL STRUCTURE, EQUIPMENT AND ASSEMBLY

Elevator wells may be of concrete, brick-built or steel structure. Concrete wells are usually assembled of plane panels or spatial blocks.

The walls, floor and ceiling of the well must be made of durable incombustible materials which do not contribute to the creation of dust and have sufficient mechanical strength. The structure of the

well must sustain the loads which may be applied by the machine, by the guide-rails during the safety gear operation, loading and/or unloading the car or in case of off-centring of the load in the car (see section 7.2), by the action of the buffers or by the anti-rebound device of compensating ropes.

In the well, car and counterweight guide-rails are fixed by means of guide brackets. With concrete and brick-built wells guide-rails should be so supported on the brackets that a relative vertical shift at the points of fixing is possible to eliminate the effect of different thermal expansion.

Landing doors represent well equipment of great significance, however, they will not be dealt with here.

There have been two different methods in existence for the installation of individual mechanical components and electric appliances in the well.

The original method, often called a 'wet process', was carried out in the same way for many decades. Brackets, holders and/or anchor bolts for fixing the devices were embedded additionally in wall pockets, which had been left free in the brickwork or concrete structure during the erection.

A new method was introduced in Europe after World War II, resulting in a considerable reduction in assembling operations on the site. A vast production of prefabricated panels and/or spatial blocks, often completely assembled, took place; they were of a very high technical standard due to the ideal conditions of manufacture, assembly and control in the factory. For fixing car and counterweight guide-rails and electric devices in the well metal sections of C-profile are currently employed, embedded in the walls, to which attachment elements are fastened by means of T-bolts. In Fig. 10.2 simple attachment elements are illustrated, facilitating a rapid and effective assembly of guide-rails and electric switches in the well.

10.3 TOP CLEARANCES

(a) Traction drive elevators

When the counterweight rests on its fully compressed buffer, three conditions must be satisfied:

(1) The guided travel of the car, still possible in the upward direction, must be equal to or greater than the minimum travel, given by the formula

$$0.1 + 0.035\, v^2 \text{ (m)}, \tag{10.1}$$

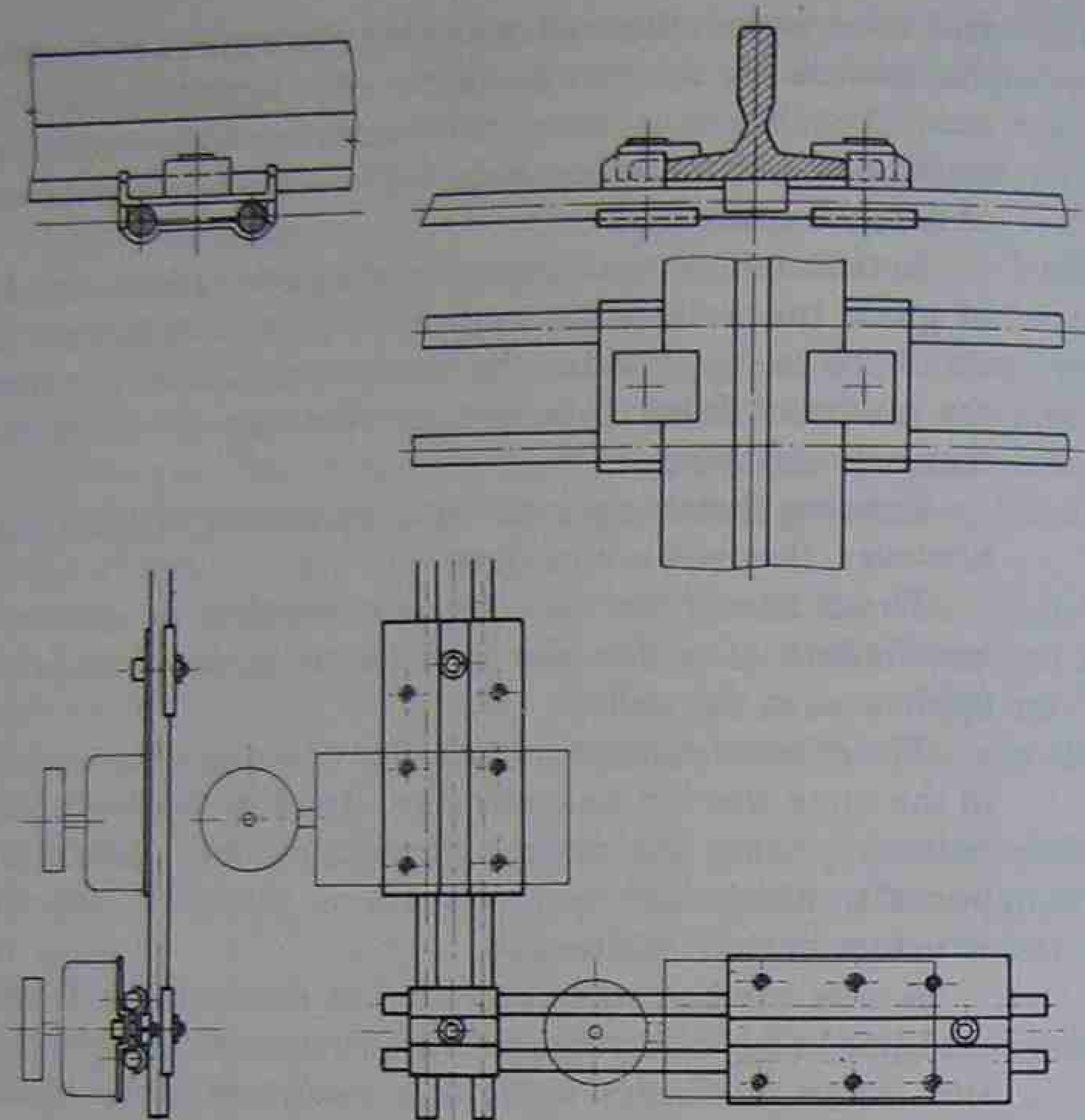


Fig. 10.2 — Attachment elements for guide-rails and electric switches.

where v is rated speed (m/s).

(2) The free height above the roof of the car enclosure must be at least $1 + 0.035\ v^2$

(3) The free distance between the lowest part of the ceiling of the well and the highest part of the guide shoes, of rope attachments and any part of vertical sliding doors, if any, must be at least $0.1 + 0.035\ v^2$.

Also, between the lowest part of the ceiling and the highest part of any other equipment fixed on the car roof, must be at least $0.3 + 0.035\ v^2$.

Similarly when the car rests on its fully compressed buffers, the guided travel of the counterweight, which is still possible in the upward direction, must be at least $0.1 + 0.035\ v^2$.

When the retardation of the elevator is positively monitored, the value of $0.035\ v^2$ in all formulae may be reduced to (i) $0.5 \times 0.035\ v^2$

for elevators of rated speed $\leqq 4$ m/s, (ii) $0.333 \times 0.035\ v^2$ for elevators of rated speed > 4 m/s. In no case may this value be less than 0.25 m.

For elevators provided with compensating ropes having a tensioning pulley equipped with an anti-rebound device the values of $0.035\ v^2$ may be replaced by a figure related to the possible travel of the pulley in question plus 1/500 of the car travel, with a minimum of 0.2 m.

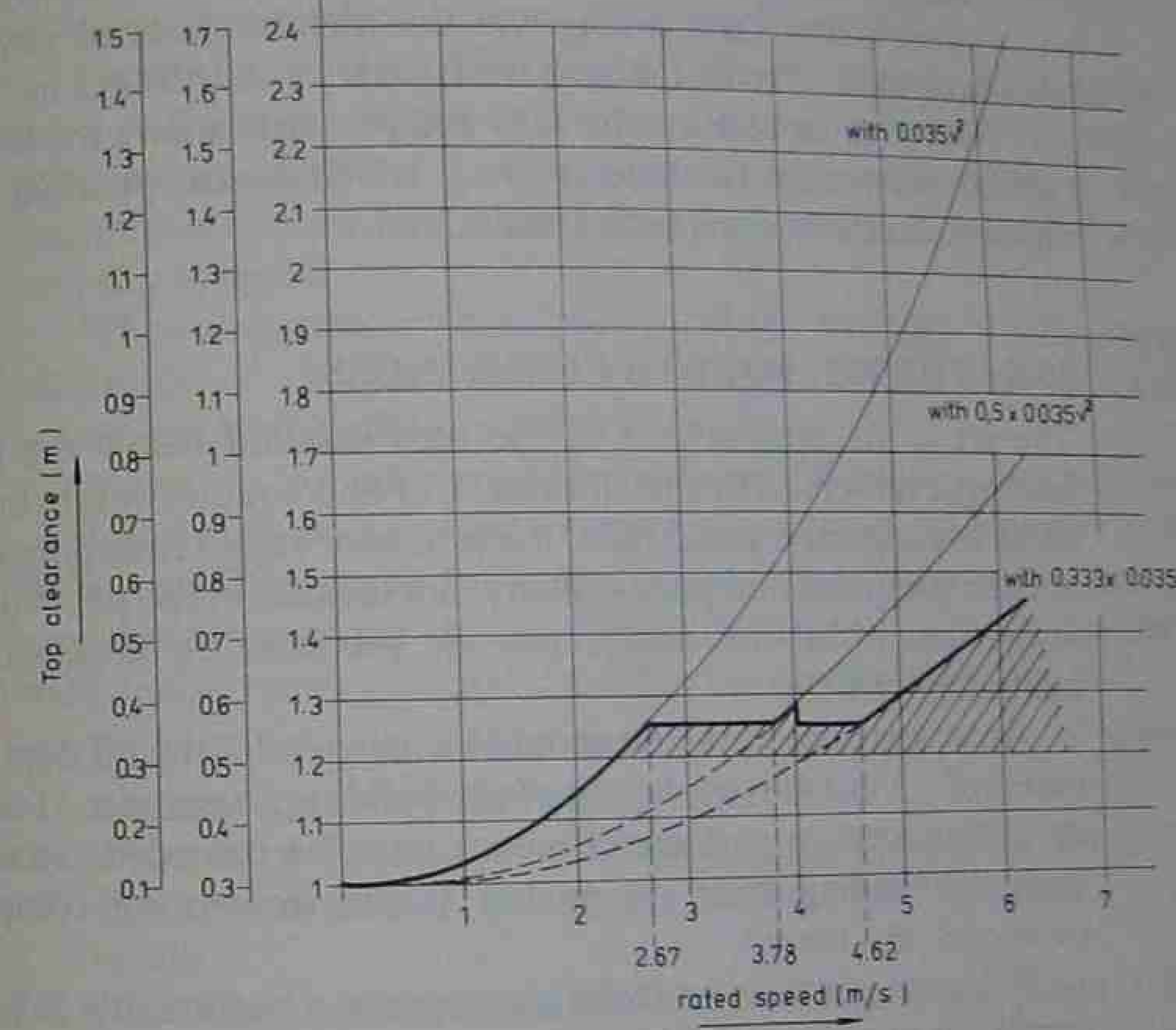


Fig. 10.3 — Top clearances for traction drive elevators.

In Fig. 10.3 a graph illustrating the top clearances for traction drive elevators is shown. The minimum clearances possible when maximum advantage of the possibilities specified in this section are taken is indicated by the heavy lines. The cross-hatched area of values is valid for elevators with compensation pulleys equipped with an anti-rebound device.

(b) Positive drive elevators

When the car is at the top floor the guided travel of the car, still possible in the upward direction before the upper buffers come into action, must be at least 0.5 m.

When the upper car buffers are fully compressed, the following conditions must be satisfied:

(1) The free height above the roof of the car enclosure must be at least equal to 1 m.

(2) The free distance between the lowest part of the ceiling of the well and the highest part of the guide shoes or any part of vertical sliding doors, if any, must be at least 0.1 m. Also between the lowest part of the ceiling and the highest parts of any other equipment, fixed on the car roof, must be at least 0.3 m.

When the car rests on the fully compressed buffers the travel of the counterweight (if there is one), which is still possible in the upward direction, must be at least 0.3 m.

10.4 THE PIT. BOTTOM CLEARANCES

The pit is the bottom part of the well situated below the lowest landing level served by the elevator. Car and counterweight buffers, bases of supported guide-rails, the governor rope tensioning device and the compensation pulley, if any, are located in the pit. After the installation of all this equipment the pit must be impervious to infiltration of water.

An access door to the pit should be provided if the pit depth is in excess of 2.5 m and if the layout of the building so permits. If there is no other access, a permanent means must be provided, accessible from the landing door, to enable a safe descent of competent personnel into the pit.

When the car rests on its fully compressed buffers, the following conditions must be satisfied:

(1) There must be sufficient space in the pit to accommodate a prism block of the dimensions 0.5 m × 0.6 m × 0.8 m, resting on one of its faces.

(2) The clear distance between the bottom of the pit and (a) the lowest parts of the guide shoes, of safety gear blocks, toe guards or any part of vertical sliding doors, if any, must be at least 0.1 m. (b) the lowest part of the car, except for items detailed in (a), must be at least 0.5 m.

10.5 MACHINE AND PULLEY ROOMS

In the machine room the elevator machine or machines and associated equipments are located. In a pulley room pulleys for double

wrap drive, diverter or deflection pulleys are placed and also overspeed governor or governors and electrical equipment may be located. Machine rooms are preferably located above the well, either inside the building or in a penthouse on the roof.

Diverter and deflection pulleys may be installed in the well, provided that examinations, inspections and maintenance operations can be carried out from the car roof or from outside the well. Installation of traction sheaves and overspeed governors in the well is very rare, but possible on conditions specified in EN 81.

The structure of machine and/or pulley rooms must withstand the loads and forces exerted during the elevator operation. The walls, floor and ceiling must be of incombustible materials, which do not assist the creation of dust. The floor must be made of or covered with non-slip material.

When required by the function of the building, the structure should be so designed to absorb substantially the vibrations and sounds created by the operation of elevators.

Access to the interior of the machine and pulley rooms must be easy and safe at any time. Access routes must be provided with sufficient illumination and must be of at least 1.8 m in height. Access routes for personnel should preferably incorporate stairs. Ladders may be used provided that the following conditions are satisfied:

— They will be exclusively used for the purpose of access to the machine room or pulley room respectively.
— When in position, they will form an angle of between 70 and 76° with the horizontal, unless they are fixed and less than 1.5 m in height.
— One or more holds within easy reach will be placed adjacent to the top end of the ladder.
— Fixed attachment points must be provided for ladders that are not fastened.

Means of access must be provided for the transportation of heavy equipment during erection and, if desirable, its replacement. For this purpose beams for electric or hand-operated hoists should be provided under the ceiling of the machine room, conveniently positioned to permit the handling of heavy equipment.

Access doors must have a minimum width of 0.6 m and a minimum height of 1.8 m. They must not open towards the inside of the room. Access trap-doors for personnel must give a clear passage of at least 0.8 m × 0.8 m. When closed they must be able to sustain a vertical force of 2000 N at any point.

Machine rooms must be ventilated either naturally or mechanically to ensure normal operation of the elevator and prevent the machine from getting overheated. The ventilation system should be so designed to avoid air swirl leading to excessive dust penetration and infiltration of fumes and humidity into the equipment. The ambient temperature in machine rooms should be maintained between +5°C and +40°C (as well as in pulley rooms in case they contain electrical equipment).

Machine rooms must contain the main switches for each elevator. By means of the main switch the power supply to the machine can be interrupted in normal operational conditions of the elevator. The switch must be positioned close to the point of access to the machine room. If more elevator machines are located in the same machine room, the control mechanism of the main switches must allow the elevator concerned to be easily identified. The switch must not cut the circuits feeding: car lighting and ventilation, lighting of machine and pulley rooms, socket outlets on the car roof and in the machine and pulley rooms, lighting of the well and alarm device.

A stop switch must be installed in the pulley room, easily accessible from the entrance to the room, to stop the elevator and keep it stationary in order to avoid a hazard during the examination, inspection and maintenance operations.

Adequate electric lighting of the machine room must be provided on the basis of at least 200 lux (108 in the USA) at the floor level. The supply for the lighting must be independent of the power supply to the machine.

Dimensions of machine and pulley rooms are specified in BS 5655:Part 5:1981.

10.6 SOUND INSULATION

The structure of most buildings, particularly residential and commercial ones, hotels, hospitals etc., must guarantee vibration absorption and sound insulation of sources, located inside the building. Vibrations of elevator equipment initiated as a consequence of building sway are not considered in this section.

There are several sources of vibration and noise associated with the operation of elevators.

(a) Elevator machine

The machine, mounted on a steel frame, is usually supported by heavy concrete beams with extensive properties as vibration dampers. Between the machine frame and the supporting beams rubber pads or silentblocks are inserted having the characteristic of a

rubber spring. In Fig. 10.4 complete elevator machines are mounted on concrete beams and rubber pads are inserted between these beams and a heavy concrete block, functioning as a vibration absorber.

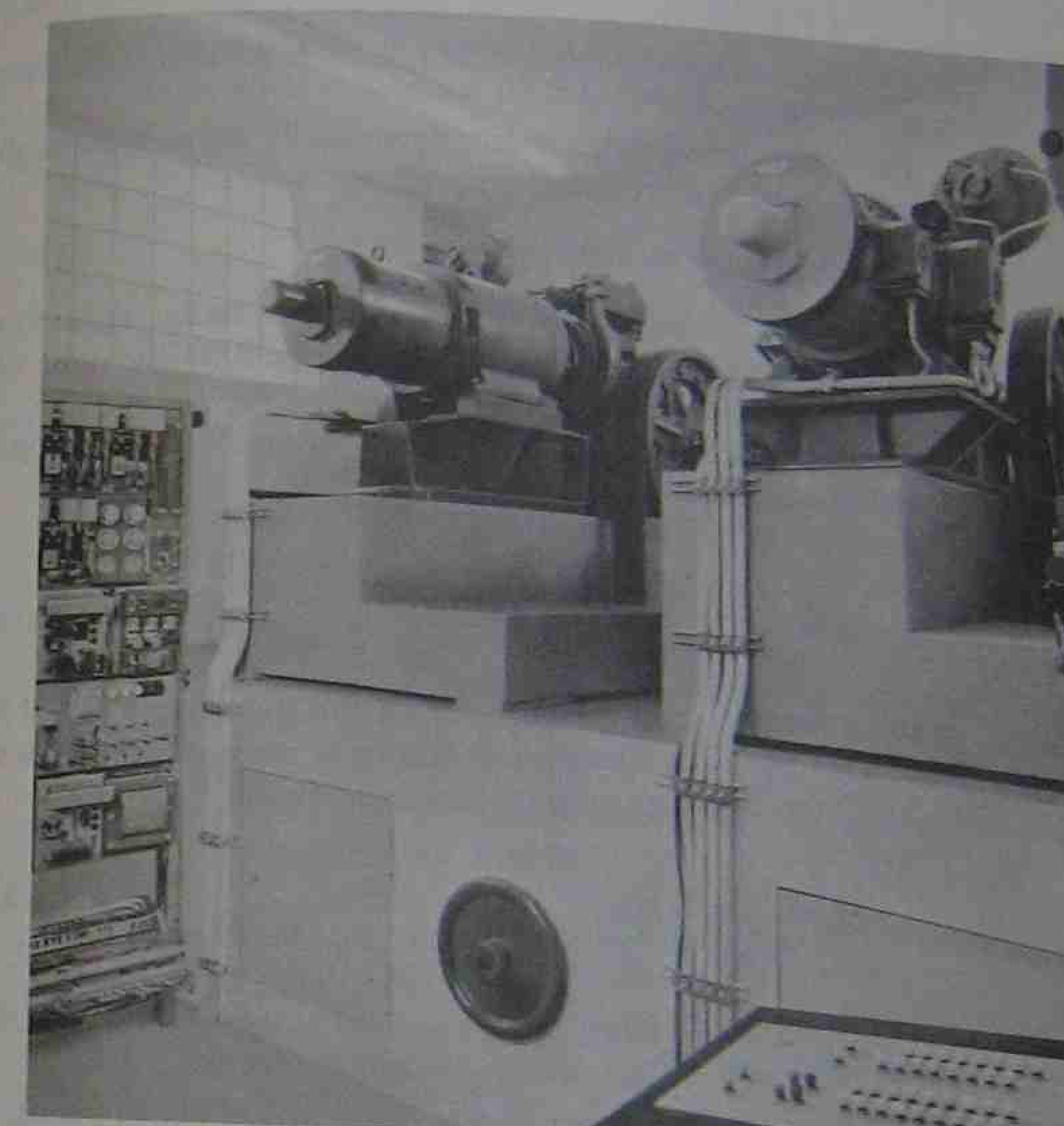


Fig. 10.4 — Elevator machines supported by a heavy concrete block.

After a certain simplification the machine (including the frame) supported by heavy concrete structure may from the mechanical viewpoint be considered as a single-mass vibration source, spring cushioned (resilient mounting). The properties of such a system are determined by the resonance frequency, given by the following parameters:

(1) Stiffness of the resilient mounting (N/m).

(2) Mass of the source with dynamic effect (kg).

In the event of the machine resting on resilient inserts (pads), the stiffness of the mounting is given by the formula

$$s = n \times s_1, \tag{10.2}$$

where n is the number of inserts and s_1 is the stiffness of one insert (N/m).

The mass of the vibration source dynamically affecting its neighbourhood m_s may be expressed generally as

$$m_s = m_m + m_b \text{ (kg)}, \tag{10.3}$$

where m_m is the mass of the machine including the frame (kg) and m_b is the mass of the supporting beams, forming one solid body with the machine (see Fig. 10.4) (kg).

The resonance frequency f_r of a system continuously oscillating in vertical direction (undamped vibration)

$$f_r = \frac{1}{2\pi} \times \sqrt{\frac{s}{m_s}} \text{ (1/s)} \tag{10.4}$$

The value of f_r must not lie between the frequencies f_1 and f_2, given by the following formulae, as otherwise resonance would take place with resultant high magnitudes of amplitude:

$$f_1 = \frac{1}{\sqrt{2}} \times \frac{n}{60} \text{ (1/s)} \tag{10.4a}$$

$$f_2 = \sqrt{2} \times \frac{n}{60} \text{ (1/s)}, \tag{10.4b}$$

where n is the revolutions per minute of the drive motor (1/min).

Furthermore, the resonance frequency must be less than the frequency of the line current, e.g. in Europe

$$f_r < 50 \text{ 1/s}. \tag{10.5}$$

Bearing in mind all conditions mentioned above, there are two possibilities for the design of elevator machine supporting structure.

(1) The machine is supported in a manner resulting in a low resonance frequency of the system, satisfying the following conditions:

$$f_r < \frac{1}{\sqrt{2}} \times \frac{n}{60}$$
$$f_r < 50 \text{ 1/s} \tag{10.6}$$

(2) The machine is supported in a manner resulting in a high resonance frequency of the system, satisfying the following conditions:

$$\sqrt{2} \times \frac{n}{60} < f_r < 50 \text{ 1/s}. \tag{10.7}$$

When projecting and calculating the frame and supporting structure of the elevator machine the maximum permissible load for the rubber pads or silentblocks must be taken into consideration and these inserts should be positioned under the machine or supporting beams in such a way that an equal distribution of the load on each insert is achieved.

(b) Controllers

The simplest solution seems to be to mount the controller on an insulated baseplate. If this solution cannot be accepted, rubber springs of honeycomb shape, vulcanized to upper and bottom protective steel plates, may be inserted between the controller and the wall or the floor respectively (Fig. 10.5).

(c) Sound sources located in the well

(1) Landing doors.
(2) Guiding the car and counterweight.
(3) Suspension ropes, compensating cables and governor rope; tensioning devices.

Although efficient provisions can be accepted to reduce substantially the sounds associated with the motion of the car, counterweight and other components, the most reliable method for preventing the noise diffusion is the installation of sound insulation to the elevator well. In such a case the well is best assembled of prefabricated spatial blocks, completely detached from the building structure by a dilatation joint. The enclosure of the well forms an independent self-balancing structure without rigid joints with the building structure.

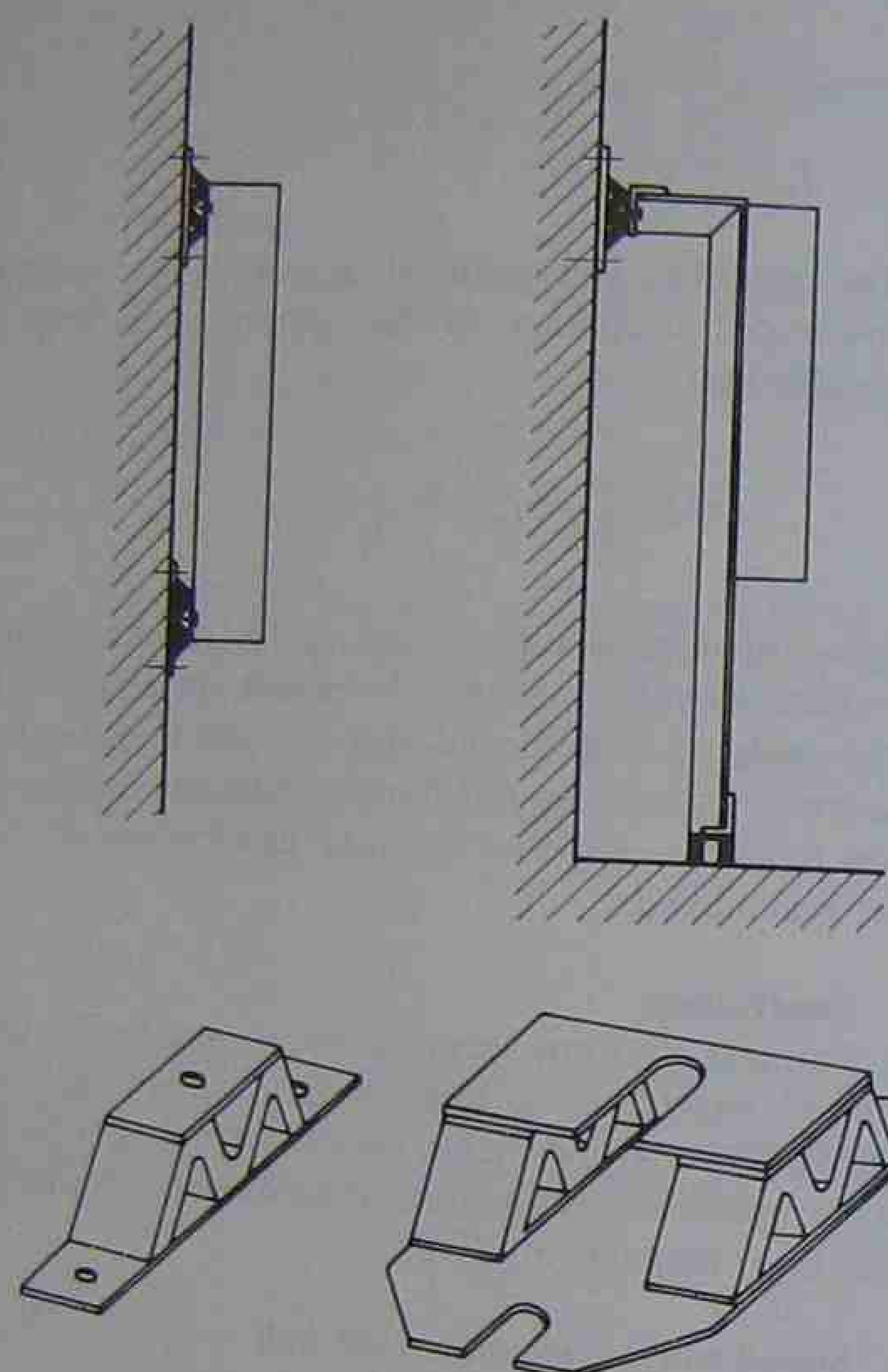


Fig. 10.5 — Elevator controller on resilient mounting.

The bottom part of the well (pit) is insulated from the building by inserts of a sound absorbing material, such as ferrocork plates. To increase the stiffness of the well structure and avoid its lateral movement the interstice between the well and building structures may be filled with sound absorbing material (ferrocork inserts, polystyrene) at floor levels.

Bibliography

Cholewa, W. and Hansel, J., *Problem of Plastic Lining Application in Friction Drives*. International Convention for Elevator Engineering ELEVCON 86, Nice, 1986.

de Crouy-Chanel, F., *International Standardization in the Lift Industry*. International Convention for Elevator Engineering ELEVCON 86, Nice, 1986.

Franzen, C. F. and Englert, Th., *Der Aufzugbau*. F. Vieweg & Sohn, Braunschweig, 1972.

Gibson, G. W., *New Concepts in Traction Drives*. International Lift Symposium Amsterdam, 1984.

Hymans, F. and Hellborn, A. V., *Der neuzeitliche Aufzug mit Triebscheibenantrieb*. Julius Springer, Berlin, 1927.

Janovský, L., *The Distribution of Tensile Forces in Elevator Ropes with Traction Drives and the Resultant Effect upon Wear of Sheave Grooves*. International Lift Symposium, Amsterdam, 1984.

Janovský, L., *Stress analysis for guide rails of electric elevators*. Technical University of Prague, dissertation, 1981.

Janovský, L. and Doležal, J., *Výtahy a eskalátory*. SNTL, Prague, 1980.

Laney, R. and McCallum, W., *Whisperflex Compensating Cable*. International Convention for Elevator Engineering ELEVCON 86, Nice, 1986.

Niemann, G. and Winter, H., *Maschinenelemente III*. Springer-Verlag, Berlin, 1983.

Phillips, R. S., *Electrical Lifts*. Pitman Publishing, London, 1973.

Strakosch, G., *Vertical Transportation: Elevators and Escalators*,

2nd edition. John Wiley & Sons, New York, 1983.
Terminology for lifts, escalators and passenger conveyors. Fédération Européenne de la Manutention, Section VII, 1981.
The Selection and Inspection of Elevator Ropes. Elevator World Educational Package, Mobile.
Walker, H., *The Thermal Rating of Worm Gearboxes.*

CODES AND STANDARDS

ANSI/ASME A 17.1—1984 Safety Code for Elevators and Escalators.
BS 329:1968 Steel Wire Ropes for Electric Lifts.
BS 721:1963 Worm Gearing.
BS 5655:Part 1:1979/EN 81:Part 1 Lifts and Service Lifts—Safety Rules for the Construction and Installation of Electric Lifts.
BS 5655:Part 5:1981 Lifts and Service Lifts—Specification for Dimensions of Standard Electric Lift Arrangements.
BS 302:1987 Stranded Steel Wire Ropes — Part 4 Specification for Ropes for Lifts.
ISO 4344: Steel Wire Ropes for Lifts.

Index

ancillary ropes, 26
anti-rebound device, 39

bottom clearances, 156
brakes, 86–89
  calculation, 89–93
  selection, 93–94
braking torque, static, 90
  dynamic, 91
breaking load, see elevator ropes
buckling factor, 104–106
buffers, 15, 134
  required strokes, 134–135
  energy accumulation type (spring buffer), 135–139
  energy dissipation type (oil buffer), 134, 142–150
  of lever type, 145

Cable-Save™, 67
car, 15
compensating cables, 39
counterweight, 15, 98–100
critical number of wire breaks, 39

diverting pulley, 45, 51
double wrap drive, 43–46
DRAKO 300T rope, 26, 29
driving machine, 14
drum drive, 71–73
Dyform elevator ropes, 27–28

elevator, definition, 13
  technical parameters, 13–14
  traction drive, 13, 42, 153
  positive drive, 13, 155
elevator brake, see brakes
elevator pit, see pit
elevator ropes, 20–31, 36
  minimum number, 17
  nominal diameter, 21
  preforming, 29
  calculation, 30
  factor of safety, 30
  breaking load, minimum, 23, 26–29
  life, 37–39
  load spectrum, 37
  lubrication, 38
  replacement, 39
elevator well, 15
  general requirements, 151
  structure, 152
equalizing gear, 17
escape holes in oil cylinder, layout, 146
Euler's formula, 52

ferrule secured eye, 35

gearing, 74
gearless machine, 74
gear ratio, 76
glide shoes, see guide shoes
governor, 15, 115
  governor pulley, 118
  governor rope, 121
  centrifugally operated, 121–122
groove, see sheave groove
groove liner, 66–68
guide rail, 101
  shape, 101–102
  jointing, 102–103
  layout, 103
  calculation, stress analysis, 104, 108
  lubrication, 113
guide shoes, 112
  glide shoes, 112
  roller guides, 114

Lang's lay ropes, 21
long wrap drive, 68
low speed shaft, layout, 76

machine room, 156–158

oil buffer, see buffers
ordinary lay ropes, 21
over-driven worm-gear, 75
overspeed governor, see governor

pit, 156
preforming, see elevator ropes
pulley room, 156

resin socketing, 34
resonance frequency, 160–161
roller guides, see guide shoes
rope, drag, 37, 52
  fastenings, 33–36
  grips, 35
  slip, 70–71
  socket, 33–34
  splice, 35
  strand, 21–22
roping factor, 30
roping systems, 42
round groove, 46, 56–58
  specific pressure, 54–60
  coefficient of friction, 60–62
rubbing speed, 79

safety gear, definition, 15
  general, 116
  instantaneous, 117, 122
    with buffered effect, 117
    wedge type, 122–123
    eccentric cam type, 124–125
    roller type, 125
  progressive, 117, 125–133
  flexible guide clamp, 126

Seale rope construction, 23
self-tightening wedge-type socket, 33
sheave, see traction sheave
sheave groove, 45
  wear, 70–71
sheave insert, see groove liner
single wrap drive, 43–48
sound insulation, 158
  elevator machine, 158–161
  controllers, 161
  sources in the well, 161–162
spring buffers, see buffers
suspension, of car and counterweight, 17
  methods, 17
suspension ropes, see elevator ropes

tapered babbitted socket, 33
technical parameters, see elevator
top clearances, 153–156
traction capability, 64–65
traction sheave, 38, 45, 49–50, 54
tractive force, 64
travelling cable, 98
tripping speed, 118–120

undercut groove, 46, 50
  specific pressure, 61–62
  coefficient of friction, 60
under-driven worm-gear, 75–76

vee-groove, 45, 49
  specific pressure, 54–55
  coefficient of friction, 60

Wahl's coefficient, 137–139
Whisperflex cable, 40–41
worm gearing, 74–79
  calculation, 80
  tooth efficiency, 78–79
  thermal performance calculation, 81–85

zinc protective coating, of wire ropes, 22

ELEVATOR TECHNOLOGY
Editor: G. C. BARNEY, Director of Networking, University of Manchester Regional Computing Centre

Undoubtedly, the arrival of the computer era and the enhanced performance of control and electro-mechanical engineering has transformed the performance of elevator systems. Reflecting technological advance during the last decade, this timely book offers the first wide-ranging, modern survey of present information on a full range of elevator topics.

ELEVATOR ABSTRACTS
Editor: G. C. BARNEY, Director of Networking, University of Manchester Regional Computing Centre

This significant classification of work carried out in the elevator industry will be useful to all practitioners. Over 1000 entries by some 800 authors are included. All entries are catalogued against some 80 key words for easy reference.

BAYESIAN DECISION METHODS IN TRANSPORT PLANNING AND OPERATIONS
A. JESSOP, Transport Studies Group, Polytechnic of Central London

Shows how the body of techniques known as Bayesian decision theory can be applied to problems of transport planning and management. The publishers know of no such comparable work, based on the experience of teaching of the subject to post-experience, postgraduate and undergraduate students for more than 15 years. It assumes little mathematical knowledge, using the 'black box' approach to give students key results; and enables them to grasp the techniques presented, carefully explaining the basic points.

ELEMENTARY ELECTRIC POWER AND MACHINES
P. G. McLAREN, Lecturer in Engineering, University of Cambridge, and Fellow of Churchill College

"keen students have good pointers for further reading: they will certainly find more on power electronics essential if they want to get the best out of modern motors" — Austin Hughes, Department of Electronic Engineering, University of Leeds, in *The Times Higher Education Supplement*

COMPUTER-AIDED ENGINEERING: Heat Transfer and Fluid Flow
A. D. GOSMAN, Department of Mechanical Engineering, Imperial College of Science and Technology, University of London; B. LAUNDER, Department of Mechanical Engineering, University of Manchester Institute of Science and Technology; and G. REECE, Department of Engineering Mathematics, University of Bristol

This book, together with its set of computer programs, is the first of a unique new sequence of computer-aided engineering texts relating different kinds of physical phenomena to engineering systems. It contains four topics, comprising a user's guide and instructional manual covering the physical and mathematical model upon which the topic is based.

VIBRATION ANALYSIS AND CONTROL SYSTEMS DYNAMICS
C. F. BEARDS, Department of Mechanical Engineering, Imperial College of Science and Technology, University of London

"a splendid textbook . . . the arguments are well presented . . . a really useful text" — N. W. Grimes in *British Book News*

"of use to the practising vibration and control engineer . . . its real usefulness is at undergraduate level, for which its clear text, illustrations and many worked examples, make it admirably suited" — R. Stirling in *The Aeronautical Journal*

published by
ELLIS HORWOOD LIMITED
Publishers · Chichester

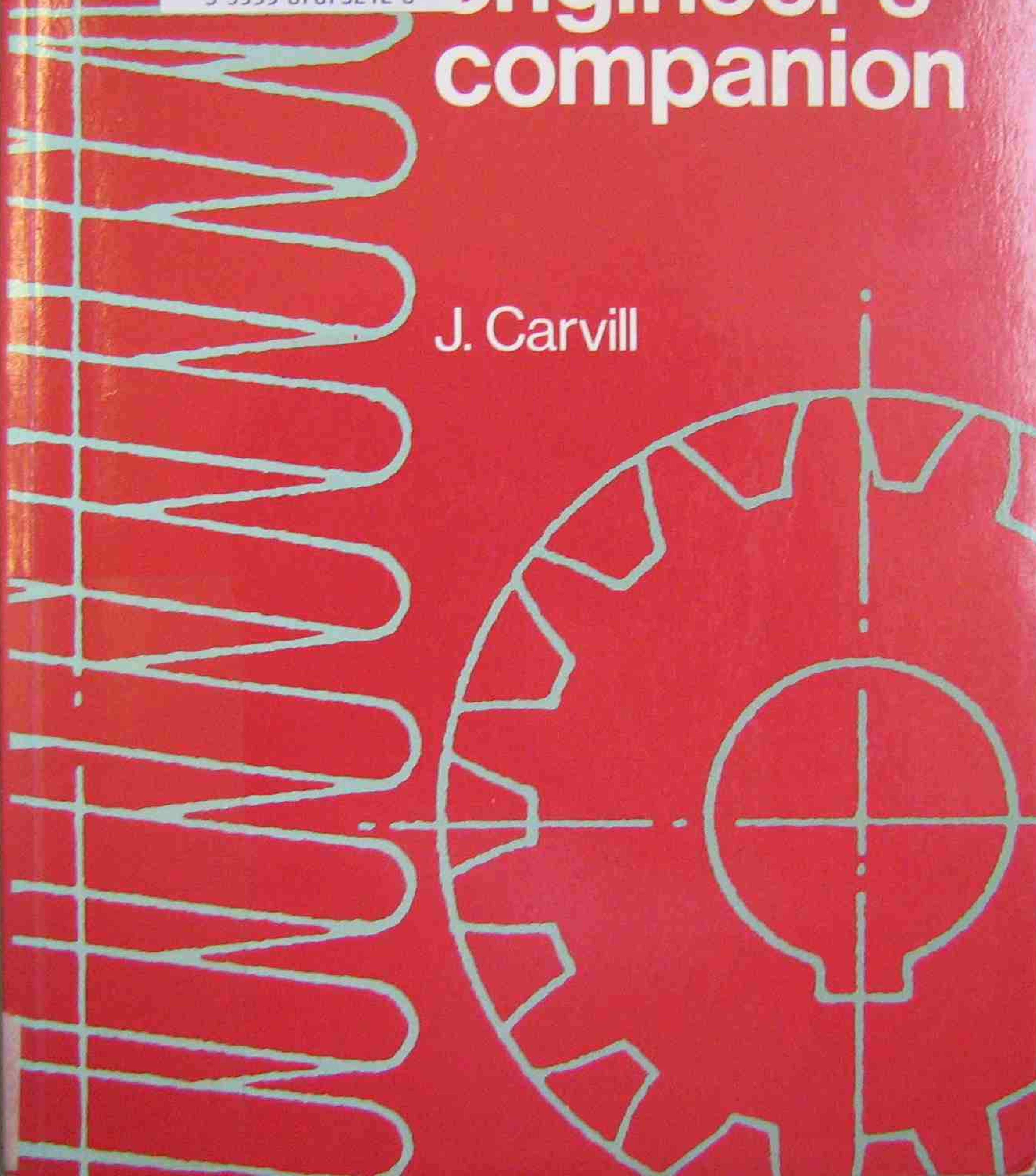
distributed by
HALSTED PRESS a division of
JOHN WILEY & SONS
New York · Chichester · Brisbane · Toronto

Ellis Horwood Library Edition ISBN
Halsted Press Library Edition ISBN 0

Butterworths

The student engineer's companion

J. Carvill



The Student Engineer's Companion

J. Carvill

Butterworths

London Boston Singapore
Sydney Toronto Wellington

Contents

1. BASIC ENGINEERING COMPONENTS

1.1 Fasteners 1
1.2 Screw threads 13
1.3 Springs 15
1.4 Gears 19
1.5 Fluid seals, joints and gaskets 23
1.6 Pipes, pipe fittings, ducts and valves 29
1.7 Shafts and rods 38
1.8 Bearings 42
1.9 Engineering design features 47
1.10 Stock materials 57

2. POWER TRANSMISSION ELEMENTS

2.1 Brakes and clutches 60
2.2 Shaft couplings 66
2.3 Belt, rope and chain drives 70
2.4 Mechanisms 74

3. WORKSHOP EQUIPMENT

3.1 Hand tools 82
3.2 Machine tools 95
3.3 Soft-soldering, brazing and welding equipment 107
3.4 Metal-forming equipment 112

4. ENGINEERING MATERIALS

4.1 Metals 118
4.2 Plastics and other non-metallic materials 120

1. Basic Engineering Components

1.1 FASTENERS

BOLTS

The bolt is widely used in engineering to fasten machine parts together, often in conjunction with a nut, to form a non-permanent connection between the parts. It has a head (usually hexagonal but which may also be square or round) and a shank of circular cross-section which is screwed with a V thread for part of its length. When the shank is screwed for its whole length it is often called a *screw* or *machine screw*.

Bolts are available in a wide range of shank diameters and lengths with various thread pitches.

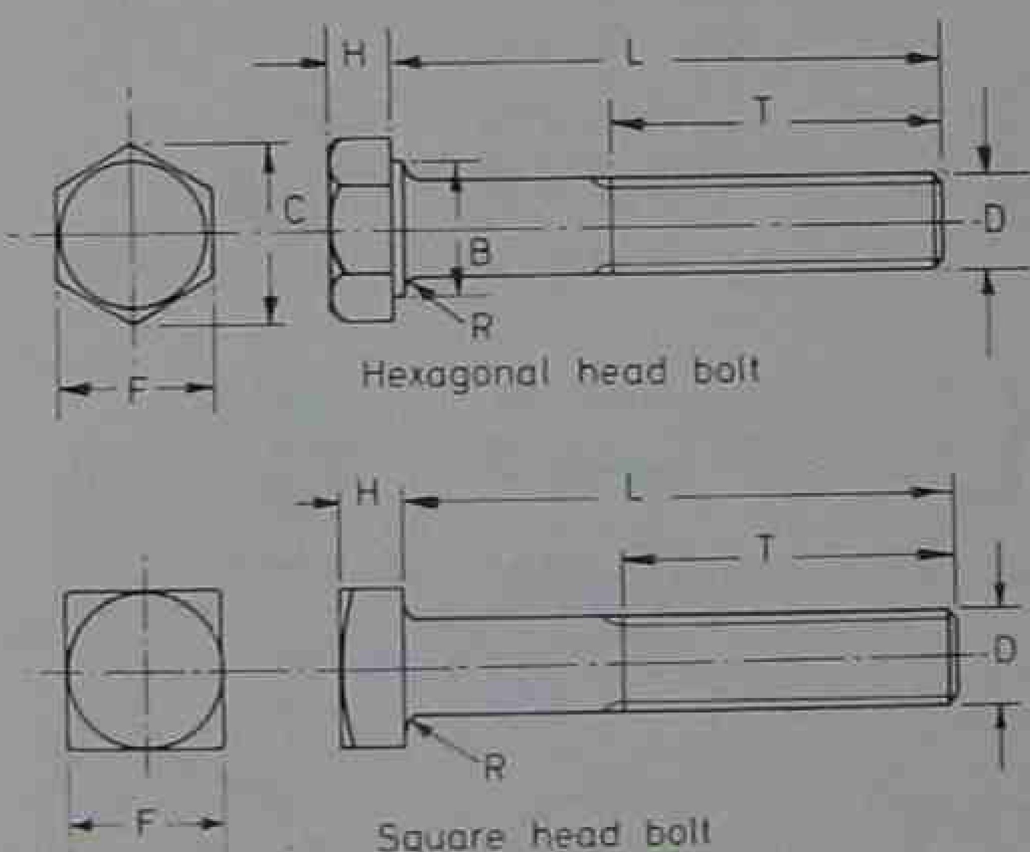


Figure 1.1 *Types of bolt*

Materials Most bolts are made of low or medium carbon steel by forging or machining, and the threads may be cut or rolled. Forged bolts are termed 'black' and machined bolts 'bright'. They are also made in high tensile steel (HT bolts), alloy steel, stainless steel, brass and other non-ferrous metals and alloys. In some cases they are protected from corrosion by galvanising or plating.

Types of thread In Britain metric bolts (ISOM) have largely replaced Whitworth (BSW) and British Standard Fine (BSF). For small sizes British Association (BA) threads are used. In the U.S.A. the most common threads are 'unified fine' (UNF) and 'unified coarse' (UNC).

Dimensions and proportions British and U.S. bolts have fractional inch sizes, e.g. ¼ in., ½ in., 1 in., with standard lengths, e.g. 1 in., 2½ in., etc., and metric bolts are made with diameters of integral numbers of mm, as shown in Table 1.1.

Table 1.1
EXTRACT FROM TABLE OF METRIC BOLT SIZES (mm)

Nominal Size	*D*	*H*	*F*	*Thread pitch Coarse*	*Fine*
M10	10	7	17	1.5	1.25
M12	12	8	19	1.75	1.25
M16	16	10	24	2.0	1.5
M20	20	13	30	2.5	1.5

D = Outside or major diameter of thread
L = Length of shank
T = Length of thread
H = Height of head
F = Distance across flats
C = Distance across corners
R = Radius of fillet under head
B = Bearing diameter

The main dimensions of bolts are: outside or major diameter of thread, length of shank, length of thread, height of head, hexagon size across flats and across corners. In addition, the pitch of the threads is given and sometimes the diameter at the bottom of the threads. The expression 'length of bolt' refers to the shank length.

Loading The total load on a bolt is the sum of the initial tightening load and the load imposed by the machine parts fastened by the bolt. The tightening load is often controlled by the stipulation of a limiting tightening torque and special torque spanners are available for this purpose.

Bolted joint A bolt may be used with a nut and washer to fasten parts together. The washer prevents damage to the surface of the part adjacent to the nut when the nut is tightened. In this case the bolt is referred to as a *through bolt*.

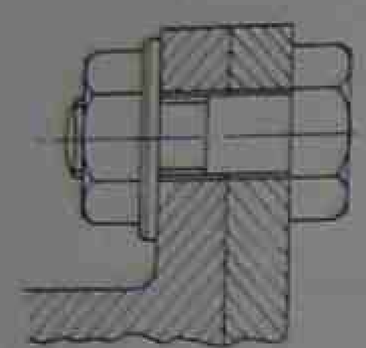


Figure 1.2 *Bolted joint (through bolt) application*

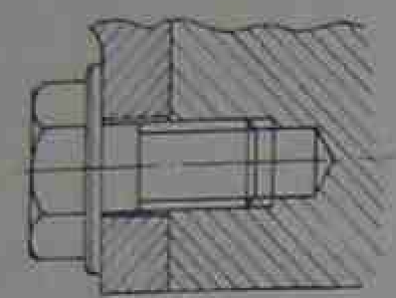


Figure 1.3 *Tap bolt application*

Tap bolt In circumstances where a nut cannot be accommodated it may be replaced by a threaded hole in one of the machine parts connected by the bolt. Passing through a clearance hole in the first part, the bolt is screwed into the threaded hole in the second part. Closer spacing of the bolts is achieved by the use of socket-head screws which are described later.

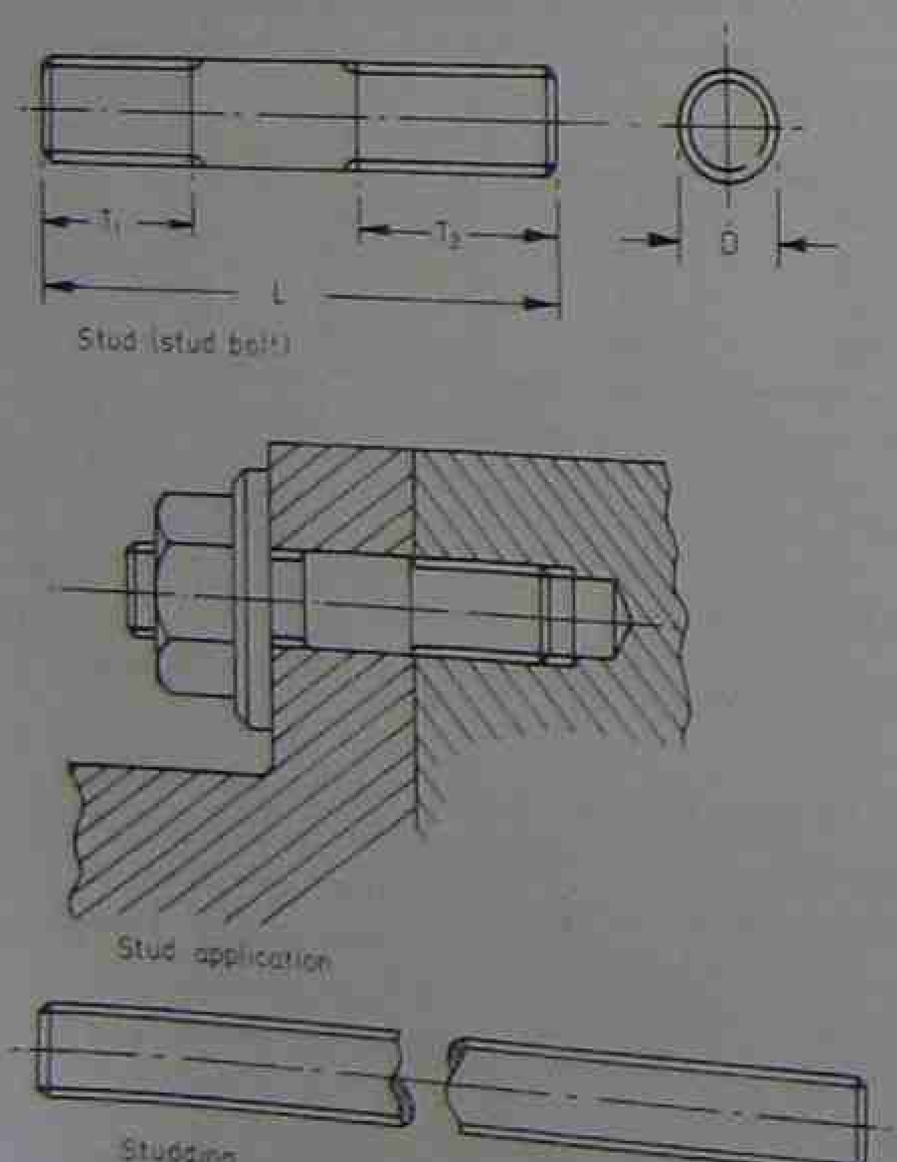


Figure 1.4 *Stud and application*

Stud (stud bolt) In cases where a tap bolt may have to be removed and replaced at frequent intervals, damage to the thread in the hole may occur. In such situations it is advisable to use a stud or stud bolt.

A stud consists of a piece of round bar threaded at each end with a plain middle section. The threads may have different pitches or be of opposite hands, i.e. one right hand and the other left hand. One end of the stud is screwed into the threaded part using two nuts or a special *stud box*, and the other part fastened by means of a nut and washer. The stud is left in place when the parts are dismantled.

Lengths of screwed rod known as *studding* are available for use as studs.

Uniform strength bolt Bolts under high impulsive load have a tendency to break at the bottom of the threads where the cross sectional area is smallest, and the V shape tends to produce cracks.

In a uniform strength bolt part of the shank is reduced in diameter to that at the bottom of the thread. Under high shock loads this part stretches and relieves the stress in the threads.

Alternatively, the shank may be drilled to reduce the area of cross section.

Uniform strength bolts are used for big-ends of connecting-rods in petrol and diesel engines.

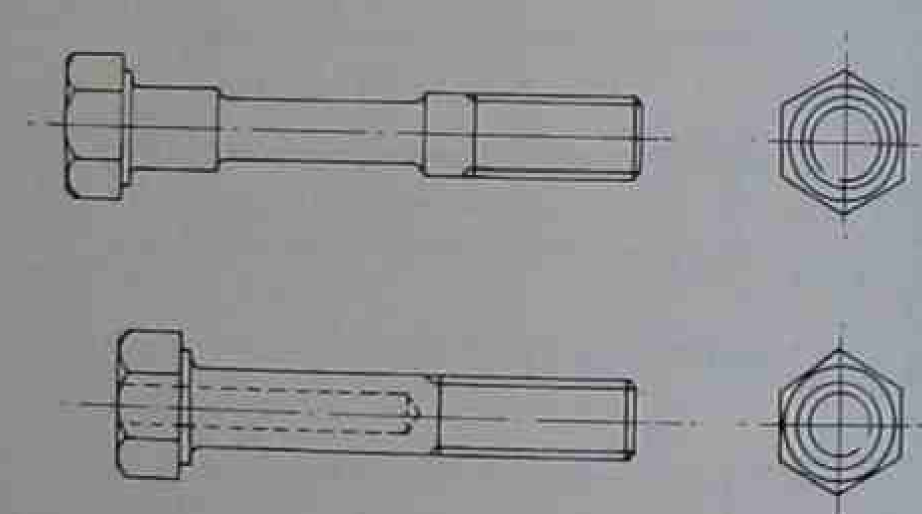


Figure 1.5 *Uniform strength bolts*

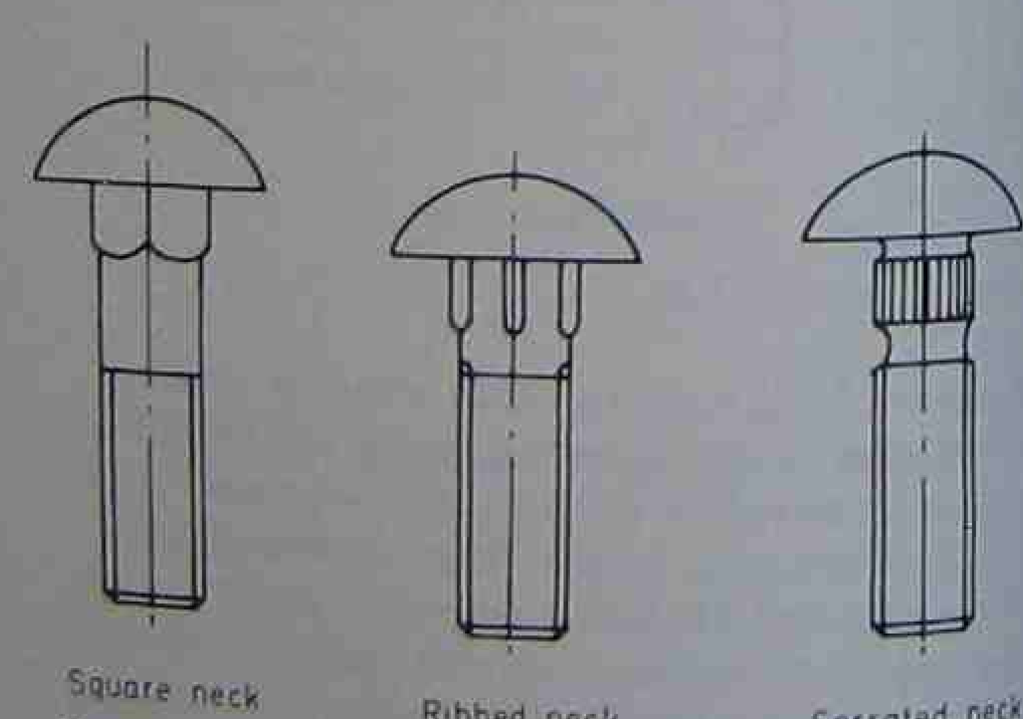
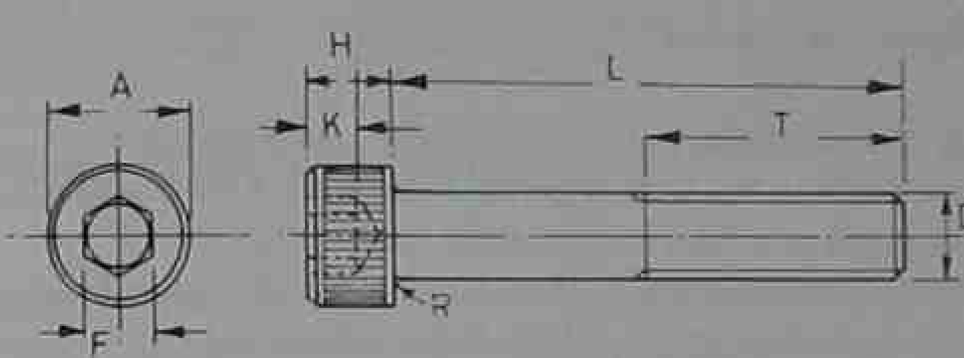


Figure 1.6 *Coach bolts (carriage bolts)*

Coach bolt (carriage bolt) Coach bolts, or carriage bolts, usually have round heads and are black bolts made of low carbon steel with coarse threads. They are used to fasten metal parts to wood. Ribs, fins or a square on the neck under the head act as locking devices. Square nuts are used with coach bolts.

Hexagon socket head screw (or bolt) A wide variety of screws (or bolts) are available which have a hexagonal recess or socket in a circular head requiring a special key or wrench for tightening. The head has many forms: cap, countersunk and button. These screws are invariably made of high tensile steel and have a coating of black oxide due to heat-treatment.

Socket screws are mostly used as tap bolts and the heads are often located in a recess for a neat appearance.



Typical metric sizes (mm)

D = 10.0	R = 0.6
A = 16.0	F = 8.0
H = 10.0	K = 5.5

L/T according to application

Figure 1.7 *Hexagon socket head screw*

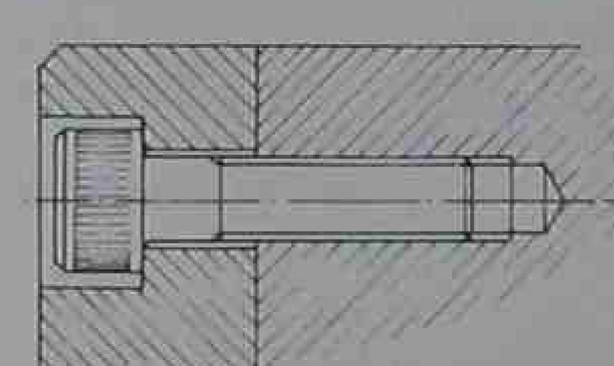


Figure 1.8 *Hexagon socket head screw application*

Figure 1.9 *Hexagon socket wrench (Allen key)*

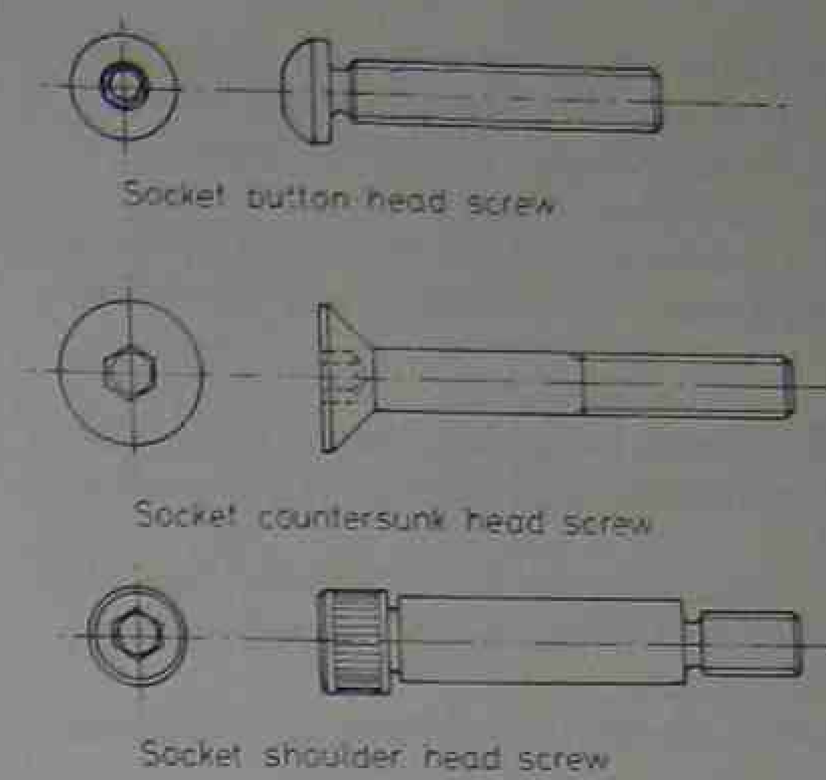


Figure 1.10 *Types of socket head screws*

T bolt The T bolt has a specially-shaped head which suits the T slots in bedplates and machine-tool tables and allows the bolt to slide in the slot without rotating.

It is used extensively for holding down work to be machined and is available in mild steel and high tensile steel.

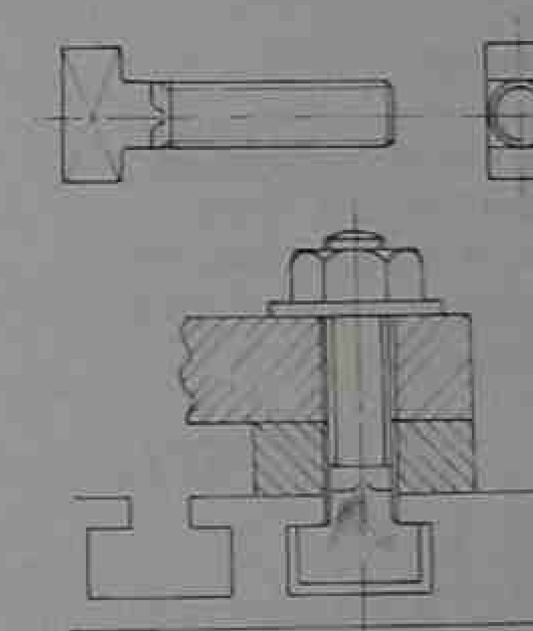


Figure 1.11 *T bolt and application*

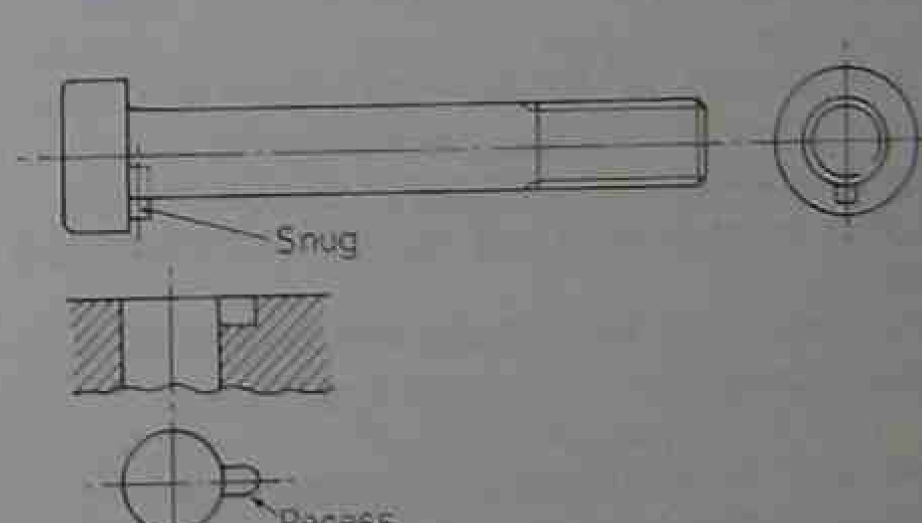


Figure 1.12 *Cheese head bolt*

Cheese head bolt Large bolts are often made with a circular head known as a *cheese head*. This shape eliminates the necessity for a hexagon. To prevent rotation of the bolt when being tightened, a pin or *snug* driven or screwed into the shank, just below the head, engages with a recess in the face of the adjacent part.

Rag bolt (foundation bolt) Rag bolts, or foundation bolts, are used for attaching machinery etc. to concrete or masonry.

Made of steel or iron, the rag bolt has a flat, tapered and roughened head which ensures a good bond when grouted into concrete.

The *indented foundation bolt* has a round body with indentations.

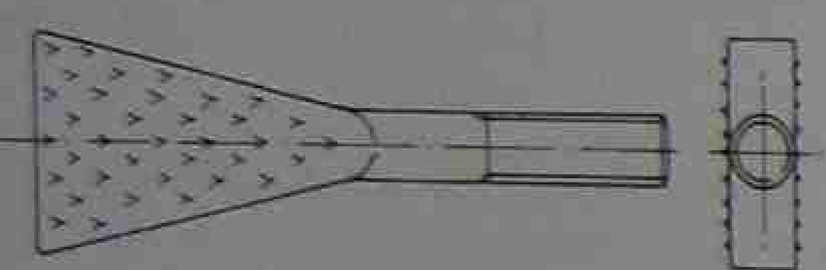


Figure 1.13 *Rag bolt*

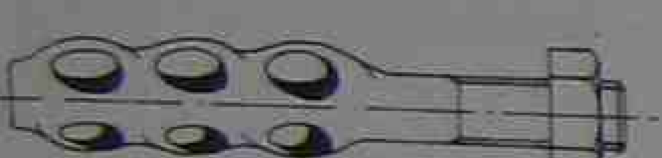


Figure 1.14 *Indented foundation bolt*

Rawlbolt (anchor bolt) A proprietary bolt for anchoring machinery to a floor or wall. A hole is drilled and the bolt inserted. When tightened, the segmented shell is expanded by a cone on the screw to give the bolt a tight fit in the hole.

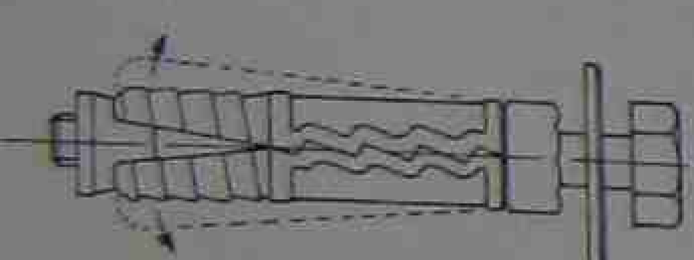


Figure 1.15 *Rawlbolt*

Eye bolt An eye bolt consists of a steel ring to which a screwed shank is attached, and it is usually permanently fitted to heavy machinery to provide an anchorage point for a rope, chain or hook used for lifting purposes.

The proportions and material used are controlled by strict standards.

Fitted bolt In most cases through bolts are fitted into holes slightly larger than the bolt diameter which are known as 'clearance' holes. Sometimes, however,

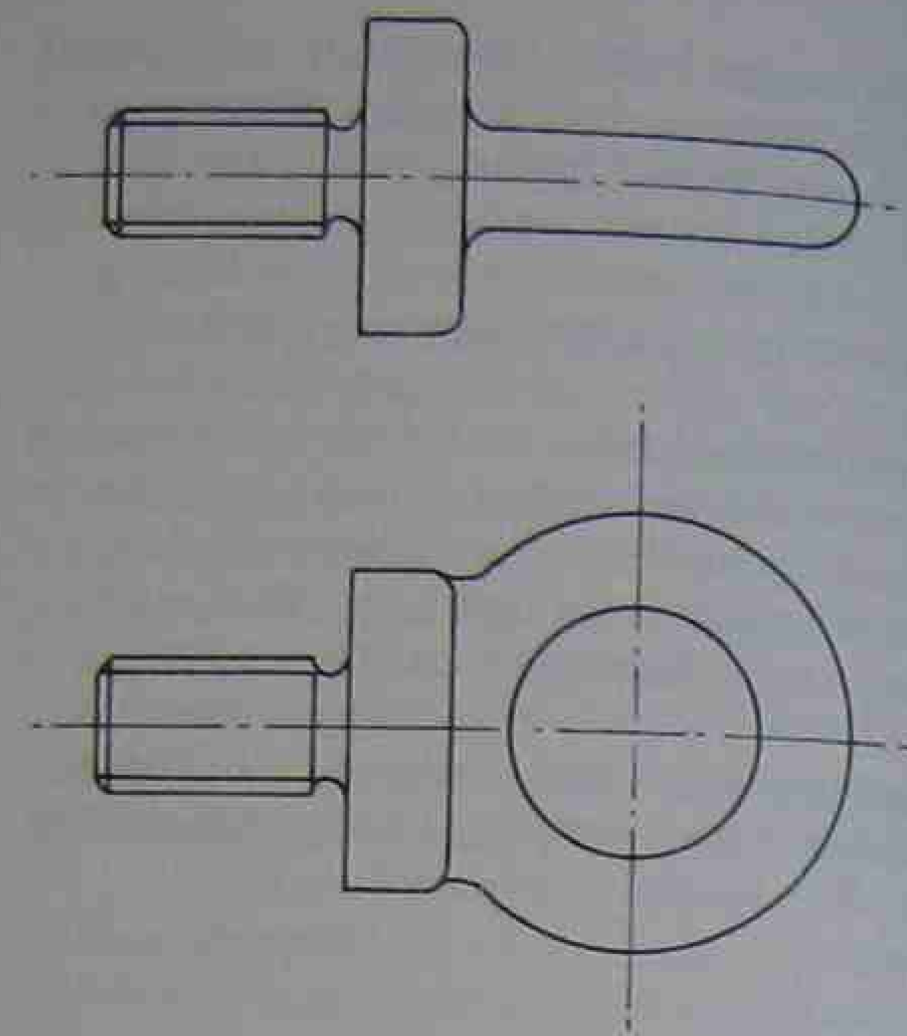


Figure 1.16 *Eye bolt*

the shank of the bolt is accurately machined and fitted into a reamed or bored hole with a very small clearance and this results in an exact location of the parts bolted together. Fitted bolts are often used in solid bolted and flanged shaft couplings, e.g. for a ship's propeller shaft. An alternative method of location is to use dowels in conjunction with bolts.

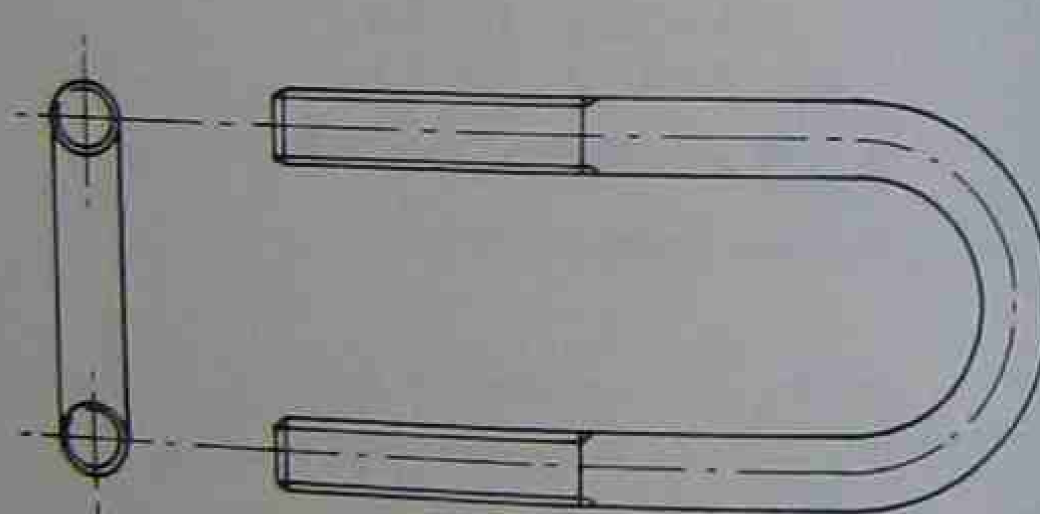


Figure 1.17 *U bolt*

U bolt This consists of a piece of circular bar bent in the form of a U and with the ends threaded. It is used for fastening round, or half round, objects such as pipes and shafts to flat surfaces.

SCREWS

This name is given to a wide variety of threaded fasteners with various types of head used with metal, wood, plastics, etc.

The name is sometimes used instead of bolt, as in the case of socket screws, but usually refers to small screwed fasteners used for light assemblies.

Most screws employed in engineering are made of steel or brass, sometimes plated, using British, metric or American threads. For small screws ranging in size from about 1.5mm to 6mm British Association and metric threads are used.

Special threads are employed for wood screws and self-tapping screws.

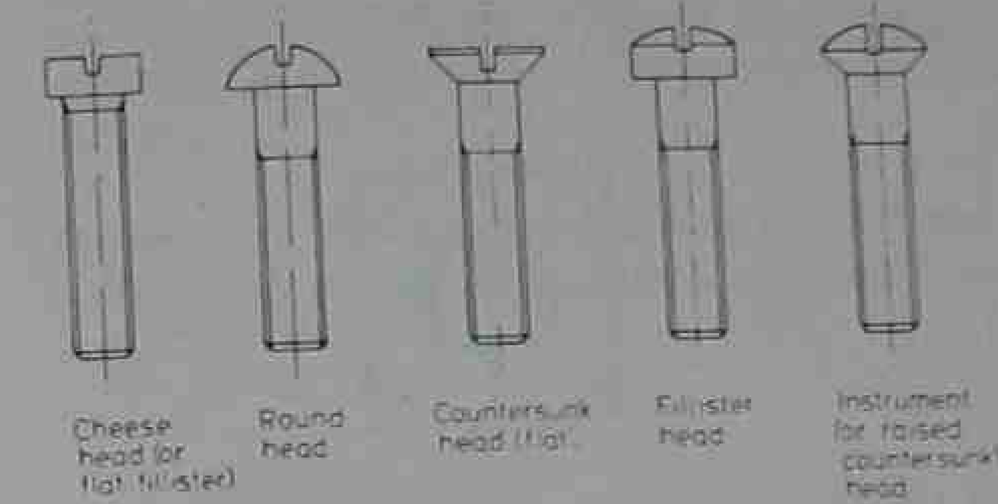


Figure 1.18 *Slotted head machine screws*

Slotted head machine screw This has a rectangular-section slot cut in the head to suit a screwdriver. There are many types of head, including round, cheese (flat fillister), fillister and countersunk or flat.

They are available in various threads and in both steel and brass which may be either cadmium or chromium plated.

Set screw Set screws are used to prevent relative motion between machine parts and often they take the place of keys on shafts where the transmitted torque is small. Most set screws do not have heads but have either a slot or a hexagon socket, and these types are known as *grub screws*.

Hardened steel is used in most cases and a variety of points is available.

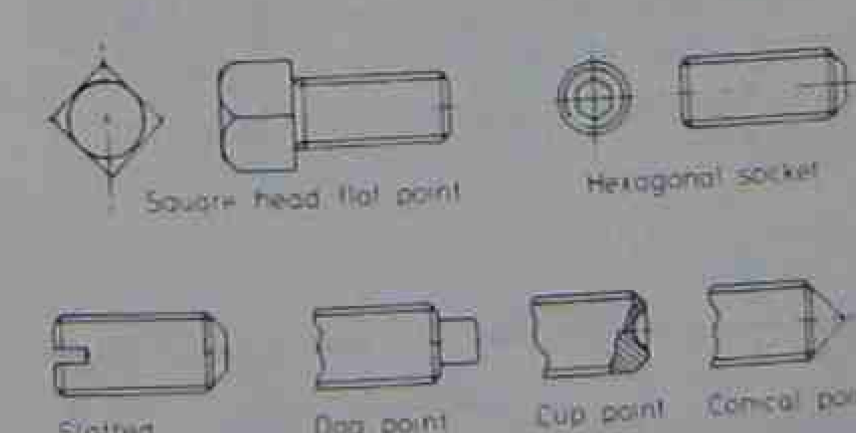


Figure 1.19 *Set screws*

Self-tapping screw (thread-forming and cutting screw) Self-tapping screws have a coarse screw thread on a tapered shank and are made of hardened steel.

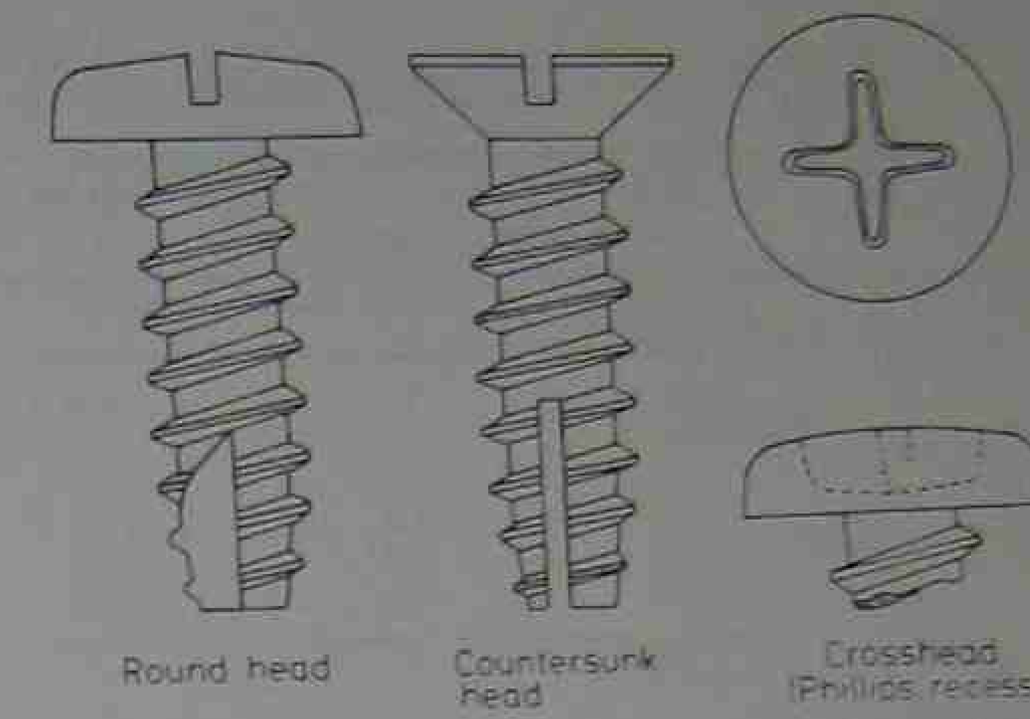


Figure 1.20 *Self-tapping screws*

They eliminate the necessity for a threaded hole or nut by cutting a thread in the material into which they are driven.

The shank may have either a blunt or a pointed end and it sometimes has longitudinal grooves which help to cut the thread in the manner of a screw tap. The heads are round, button or countersunk and have either slots or cross-shaped recesses, the latter requiring the use of a cross-head screwdriver.

These screws are used extensively for the assembly of sheet metal parts, soft castings and plastics.

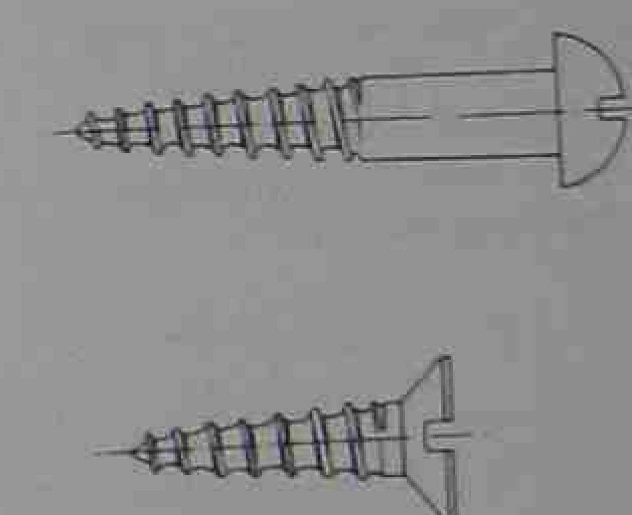


Figure 1.21 *Wood screws*

Wood screw This type is often used in engineering to attach sheet metal to wood. Wood screws are made of steel, brass, gunmetal and copper, and may be painted or electroplated. The heads are round or countersunk with either slots or star recesses.

Wood screws are available in a wide range of diameters from 2–10mm and in many lengths.

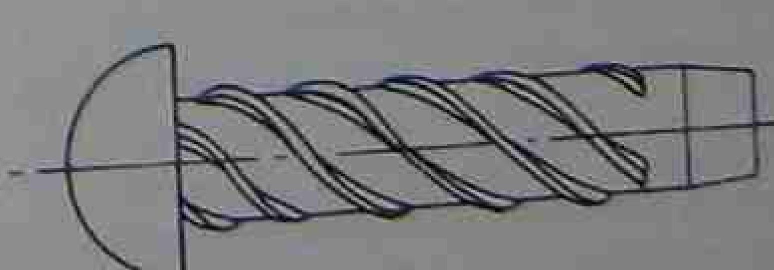


Figure 1.22 *Drive screw*

Table 1.2
ISO METRIC PRECISION HEXAGON NUTS AND THIN NUTS (mm)

Nominal size and diameter	*Thread pitch (coarse)*	*Width across flats*	*Width across corners*	*Thickness of nut Normal*	*Thin*
d	p	s	c	m	t
M5	0.8	8.0	9.20	4.0	–
M6	1.0	10.0	11.50	5.0	–
M8	1.25	13.0	15.00	6.5	5.0
M10	1.5	17.0	19.60	8.0	6.0
M12	1.75	19.0	21.90	10.0	7.0
M16	2.0	24.0	27.70	13.0	8.0

Drive screw Drive screws are hardened steel pins with very coarse pitch multistart screw threads. They are hammered or pressed into unthreaded holes in which they rotate to form a mating thread.

They are used for the rapid attachment of parts such as nameplates to castings, clips, etc. (Figure 1.22).

NUTS

A nut is a collar, usually made of metal, with a threaded hole into which is fitted a bolt, stud or screwed bar. Together with a bolt it provides the most widely used means of fastening parts together.

Nuts may be hexagonal, square or round in shape. Steel nuts are available in either black or bright condition and may be forged or machined. Black nuts may be machined on one or both faces and bright nuts have one or both faces chamfered.

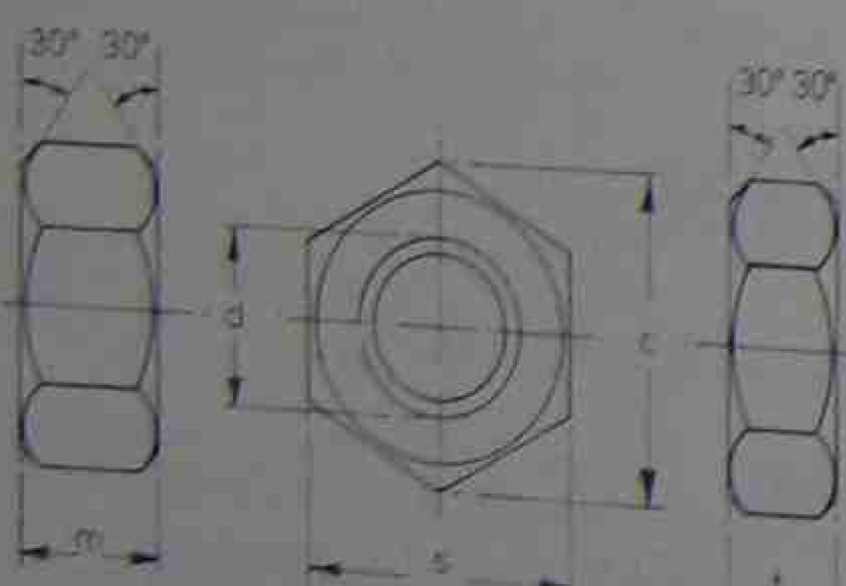


Figure 1.23 *ISO metric precision hexagon nut and thin nut*

Square nut Square nuts are usually obtained in the black, or unmachined, state and provide a cheap alternative to hexagon nuts.

Round nut (ring nut) These are often used for attaching parts to shafts and have slots or holes so that they may be tightened by using a special key.

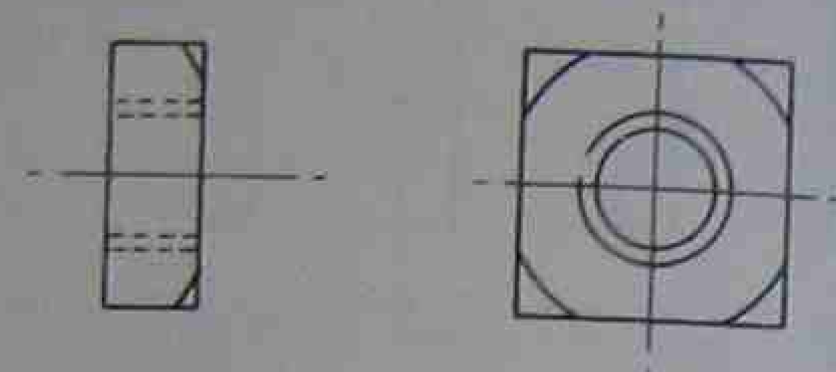


Figure 1.24 *Square nut*

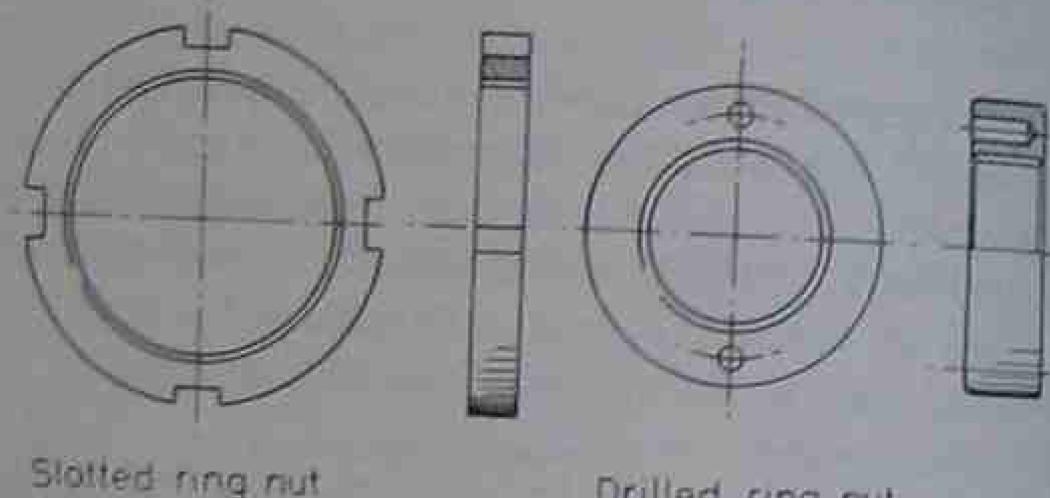


Figure 1.25 *Ring nuts*

Cap nut (crown nut, dome nut) In these one end of the threaded hole is closed and rounded to protect the end of the bolt or stud and give a neat appearance. They are made in steel or brass and are usually chromium plated.

Wing nut These nuts have wing-like projections for hand tightening. They are made in steel and brass and are used where frequent removal and replacement of parts is required.

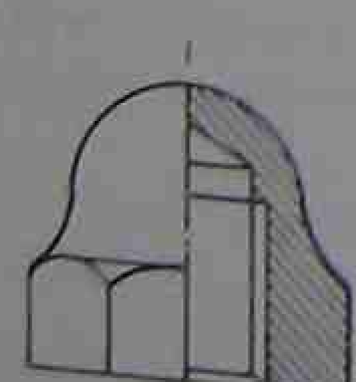


Figure 1.26 *Cap nut (crown nut, dome nut)*

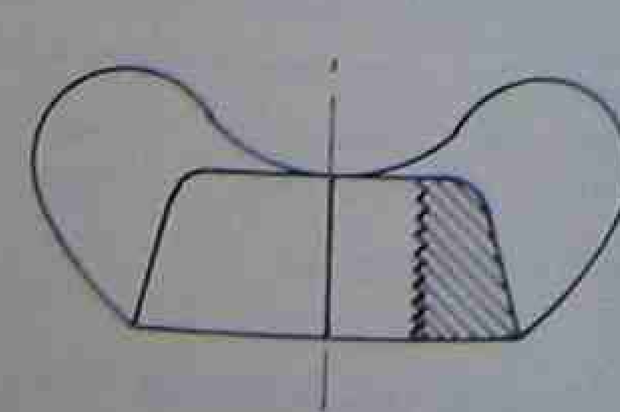


Figure 1.27 *Wing nut*

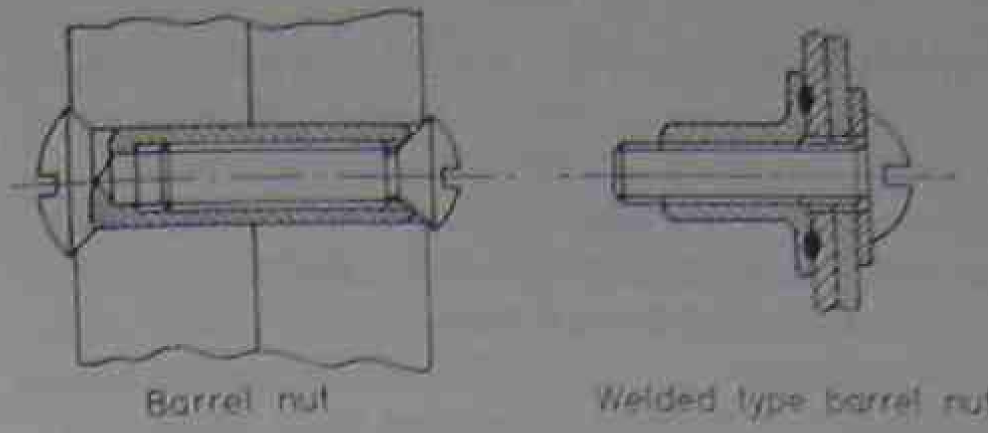


Figure 1.28 *Barrel nuts*

Barrel nut This nut has a tubular form. One type has a slotted head similar to that of the mating screw, while another has a flange which is welded to sheet metal.

Captive nut A nut which is loosely fastened to a machine part so that it is held in position until a bolt or screw is fitted.

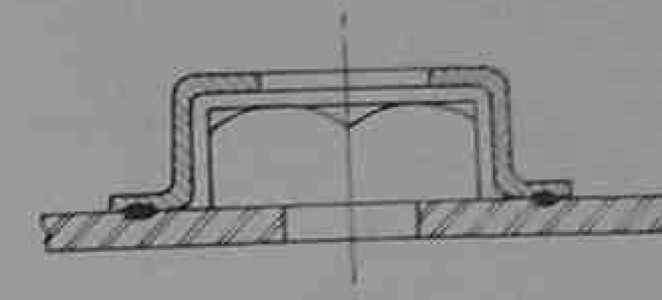


Figure 1.29 *Captive nut*

Locking nuts Nuts subject to shock loads and vibration have a tendency to work loose and cause damage or failure in machines. A wide range of locking devices is available including special nuts, lock washers and adhesives.

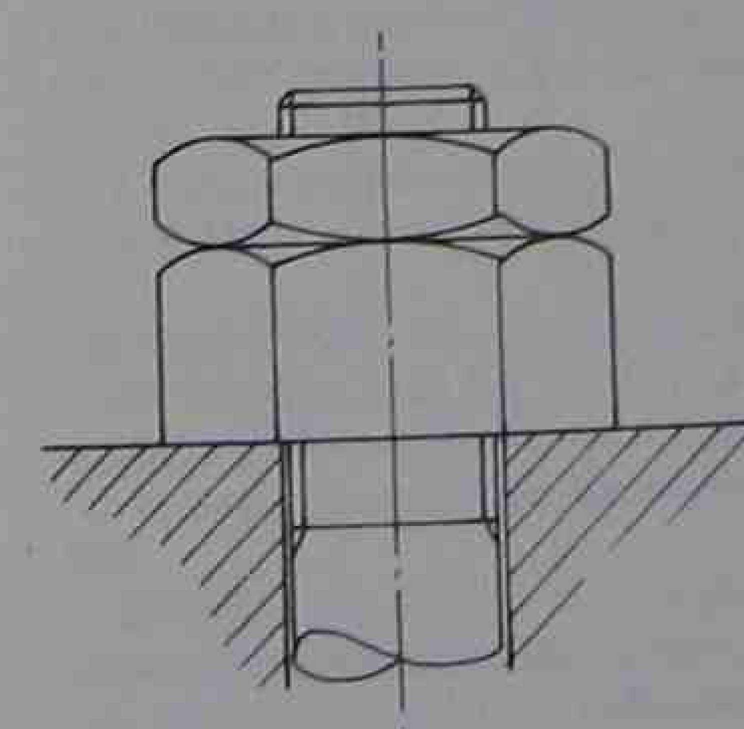


Figure 1.30 *Locked nuts (jam nuts)*

Locked nuts (jam nuts) A nut can be locked by tightening another nut against it and for this purpose a thin nut is used for the sake of economy. Ideally this should be situated below the normal-sized nut although this necessitates the use of a thinner spanner. Two spanners are required when locking the nuts.

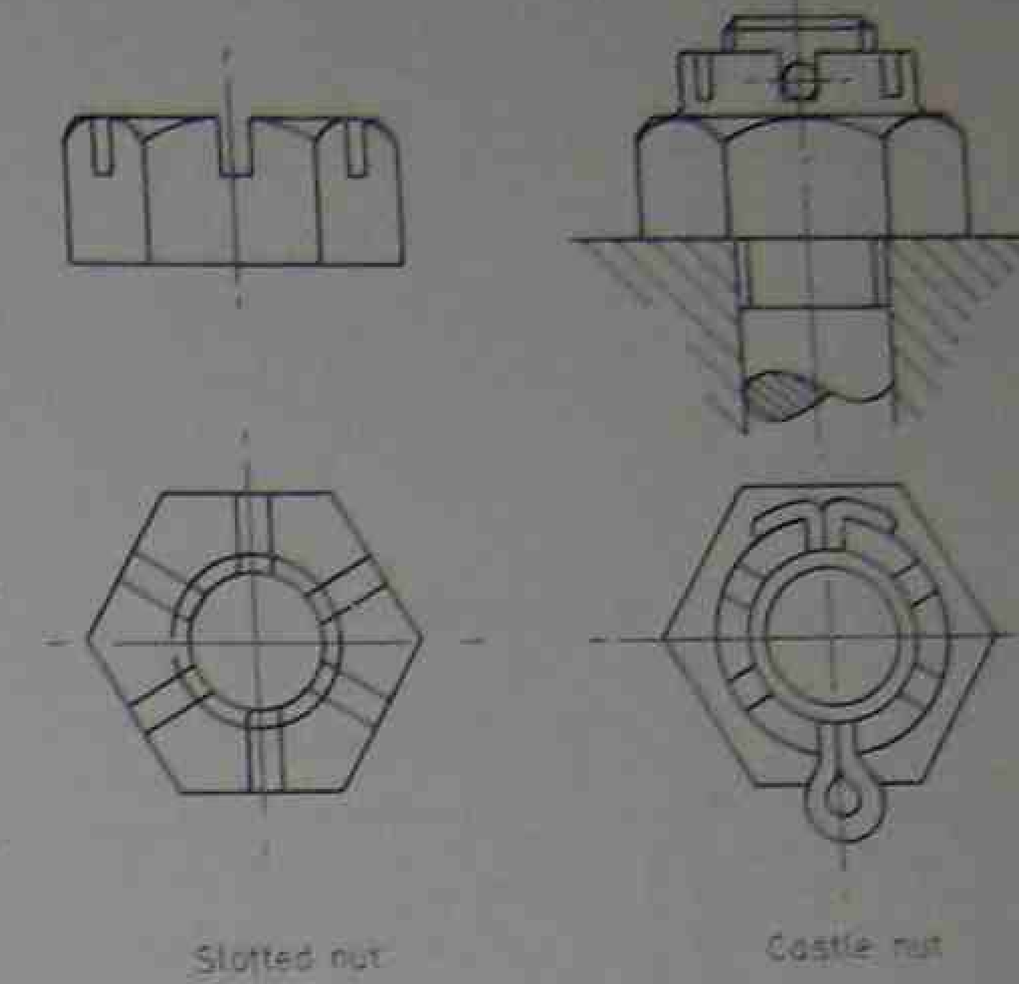


Figure 1.31 *Slotted and castle nuts*

Slotted nut and castle nut A slotted nut is a hexagonal nut with six radial slots cut in the top face two of which line up with a hole in the bolt so that a split pin may be passed through to lock the nut. Alternatively, wire can be used to lock a group of nuts.

In a castle nut the slots are cut in a circular section of the nut above the hexagon.

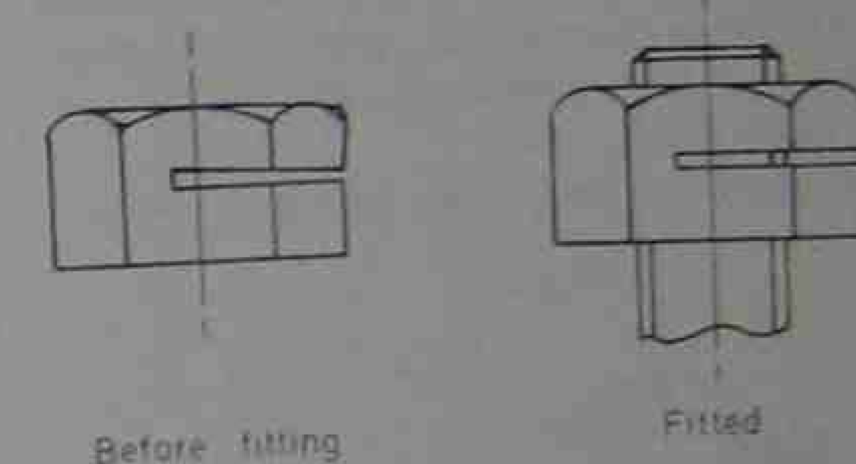


Figure 1.32 *Split nut*

Split nut A slot cut in the side of a hexagon nut is closed before fitting. The bolt forces the slot open with a resulting high frictional force which reduces the tendency for the nut to loosen.

Spring lock nut (compression stop nut) This is a hexagonal nut similar in appearance to a castle nut but the slots in the top of the nut form tongues which are initially pressed down to apply a frictional force on the bolt when fitted (Figure 1.33).

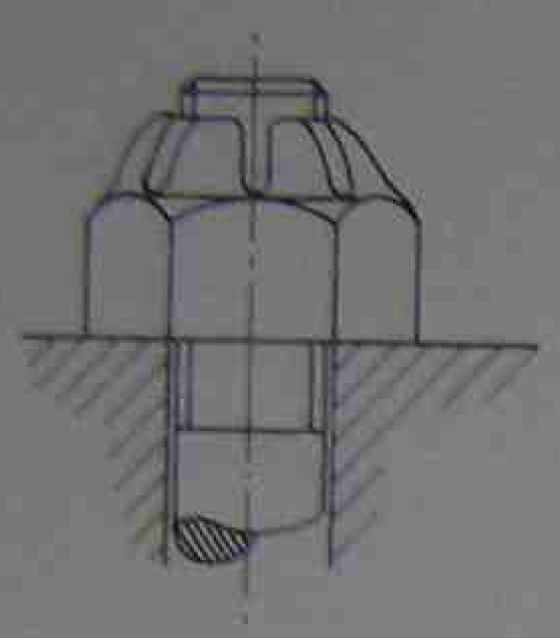


Figure 1.33 *Spring lock nut (compression stop nut)*

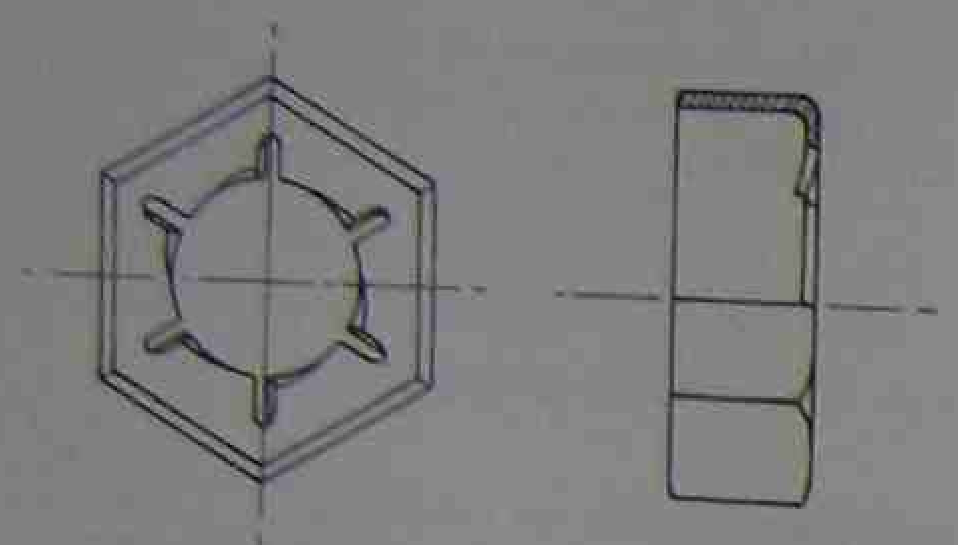


Figure 1.34 *Stamped spring nut*

Stamped spring nut This is stamped out of spring steel sheet in a variety of patterns with a hexagonal form and projections which engage with the bolt thread to give a high degree of friction.

Elastic stop nut (Nyloc nut) A ring of material such as fibre or nylon is inserted into a groove in the bore of a hexagon nut to provide a high frictional force when the nut is fitted.

The Nyloc nut is a proprietary type with a nylon insert.

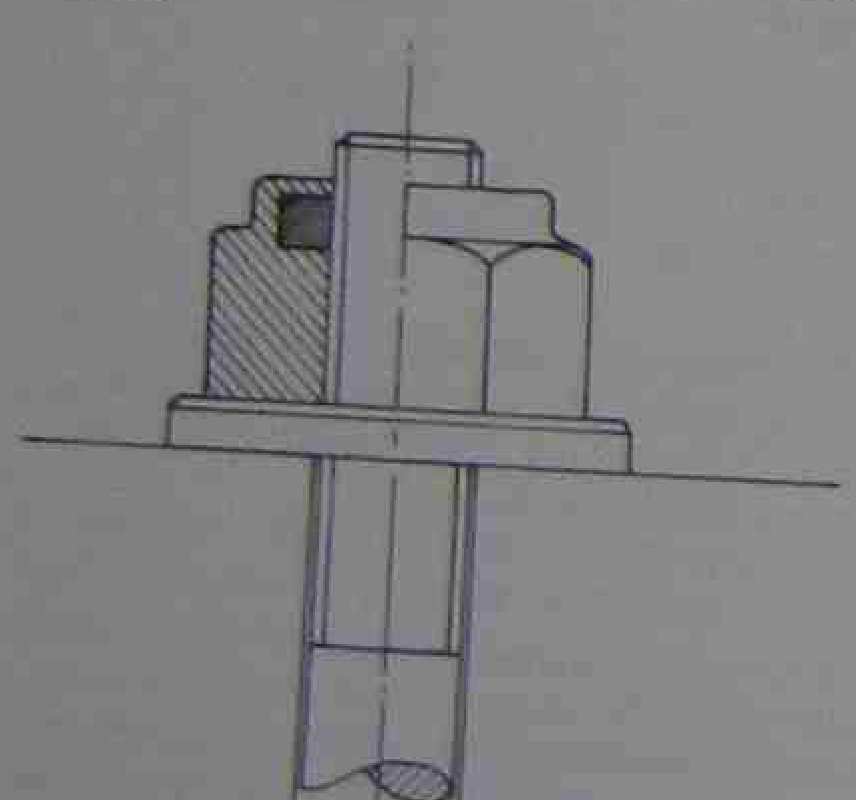


Figure 1.35 *Elastic stop nut (Nyloc nut)*

WASHERS

A washer is an annular disk of metal, plastic, rubber, etc., usually flat, which is placed either under a nut or between the surfaces of a joint to distribute the load when the nut or joint is tightened.

Most washers are made of steel but brass is used in conjunction with brass screws and nuts. Washers of copper, aluminium, fibre and leather are used extensively for sealing fluids.

Plain washer (flat washer) This is a flat washer, usually made of steel, and used under a nut to prevent damage to the face and to distribute the load. Cheap washers are punched out of black plate but more expensive ones are machined and have a bevelled edge for improved appearance. In addition to washers of 'normal' proportions, 'narrow' and 'wide' varieties are available.

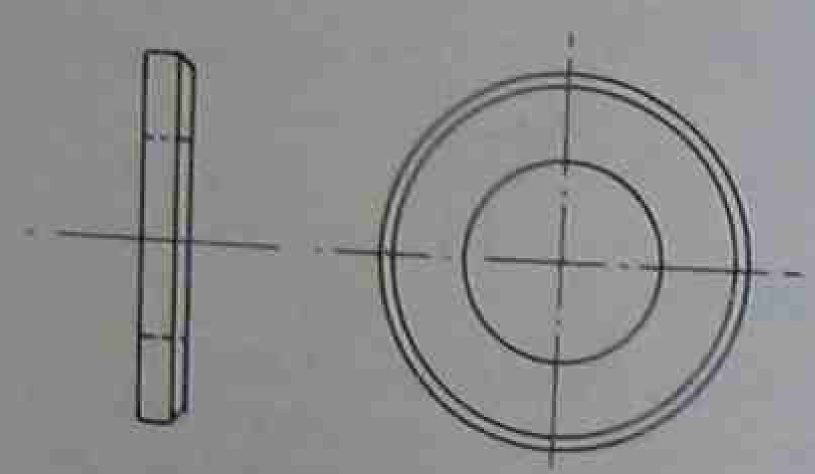


Figure 1.36 *Plain washer (flat washer)*

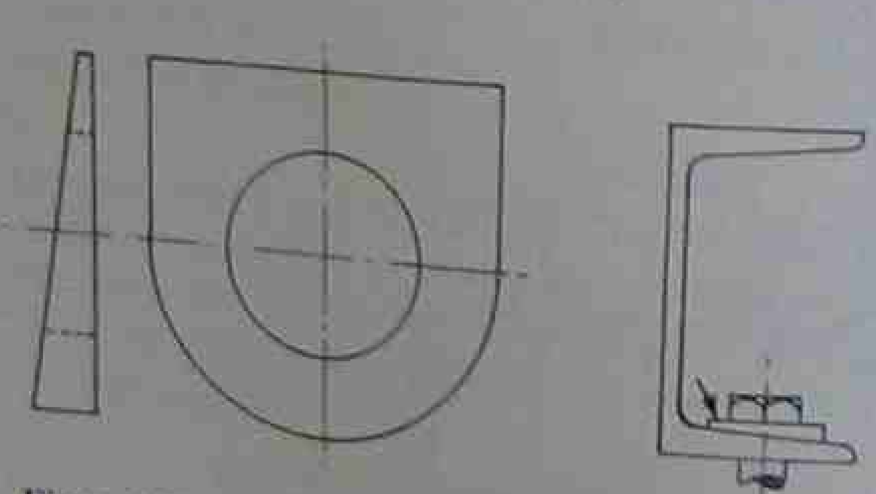


Figure 1.37 *Taper washer and application*

Taper washer A washer where the thickness varies from one side to the other to allow for the taper on the flanges of rolled steel sections such as channel and I beams.

Lock washer To prevent the loosening of nuts due to shock and vibration, lock washers are used extensively as an alternative to locknuts. There are two main types, those which rely on increased friction between the nut and the face, and those which use the faces of the nut to give a positive fixing.

Helical spring lock washer This consists of one or more turns of a helical spring made of rectangular section spring steel wire. When the nut is tightened the washer is compressed to cause a large friction force between the nut and the face. This is aided by sharp ends on the washer which cut into the faces to give positive locking.

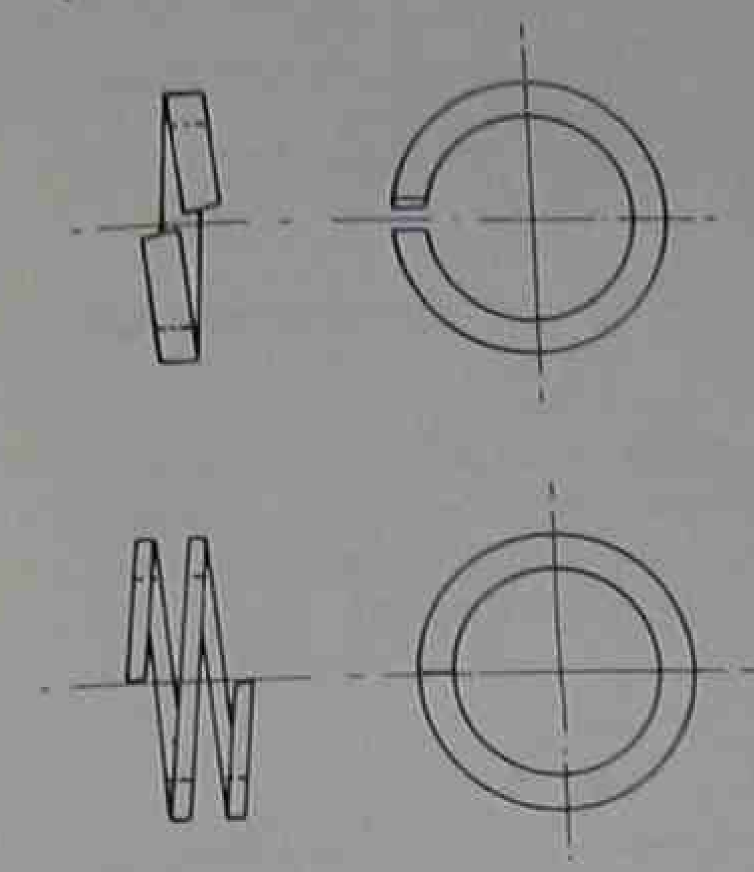


Figure 1.38 *Helical spring lock washer and two-coil spring lock washer*

Serrated lock washer (tooth lock washer) These are made of spring steel and consist of annular disks with serrations on either inner or outer diameter. The resulting projections are twisted and have sharp edges. When the nut is tightened the projections are flattened and cut into the faces of the nut and the part in contact.

A conical washer with external serrations is available for use with countersunk head screws.

Tab washer The tab washer is made from sheet metal and has a hole for the bolt or stud with tabs on the periphery which are bent at right angles against the faces of the nut and against a face on the adjacent part.

Alternatively, the tab may be punched into a hole previously drilled in the part.

RIVETS

A rivet is a metal pin with a circular shank and a head. It is used to make a permanent joint between two or more pieces of plate. The shank is passed through mating holes in the plates and 'closed' by forming a head on the projecting shank by hammering or pressing.

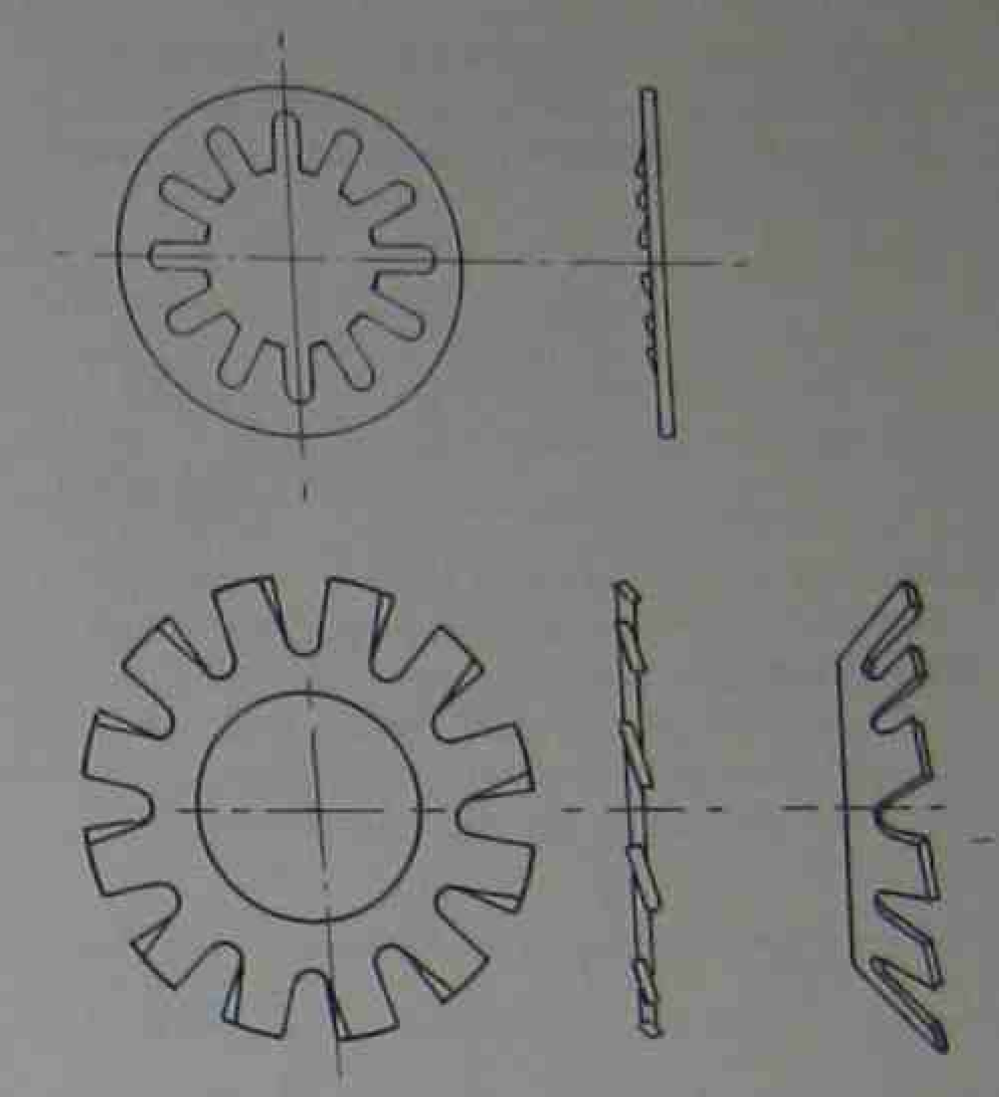


Figure 1.39 *Internally serrated lock washer (tooth lock washer) and externally serrated lock washer, flat and for countersunk hole*

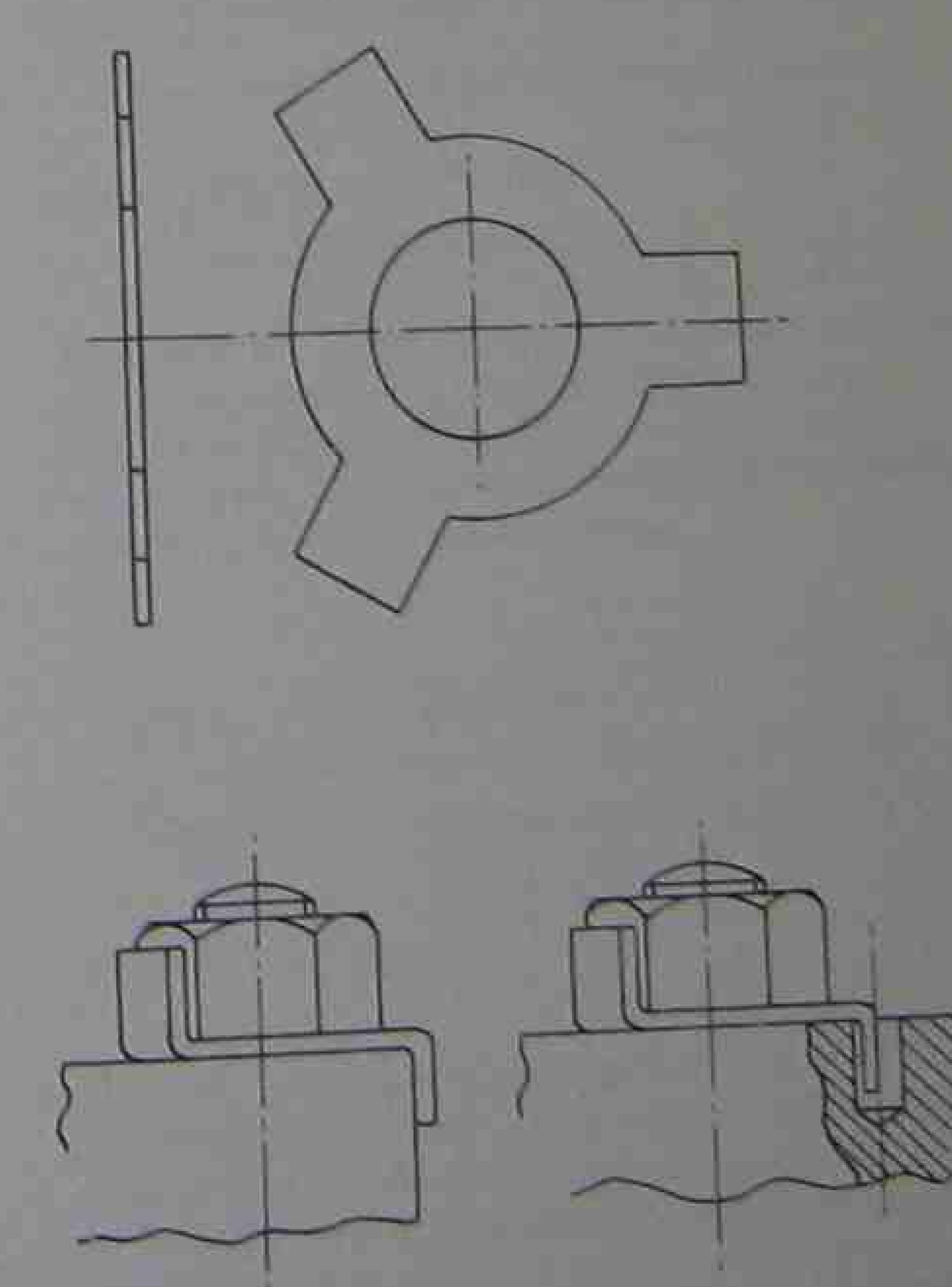


Figure 1.40 *Tab washer and application*

Steel rivets are often closed when red hot but rivets of softer metals such as copper and aluminium are closed cold. The heads may be round, countersunk, pan-shaped, etc.

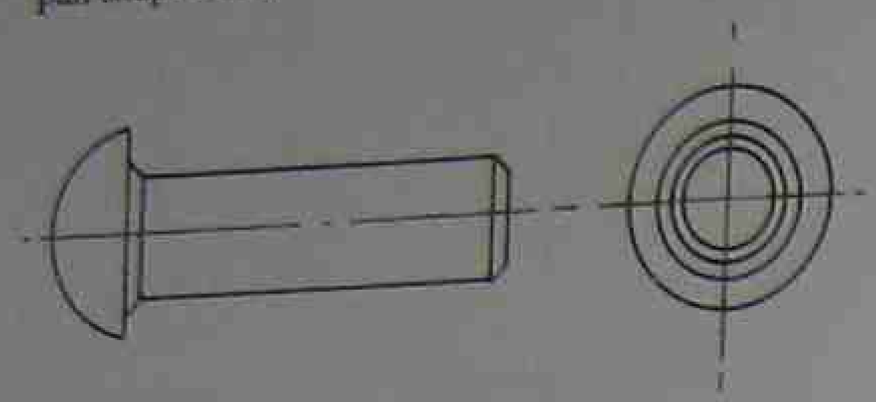


Figure 1.41 *Rivet*

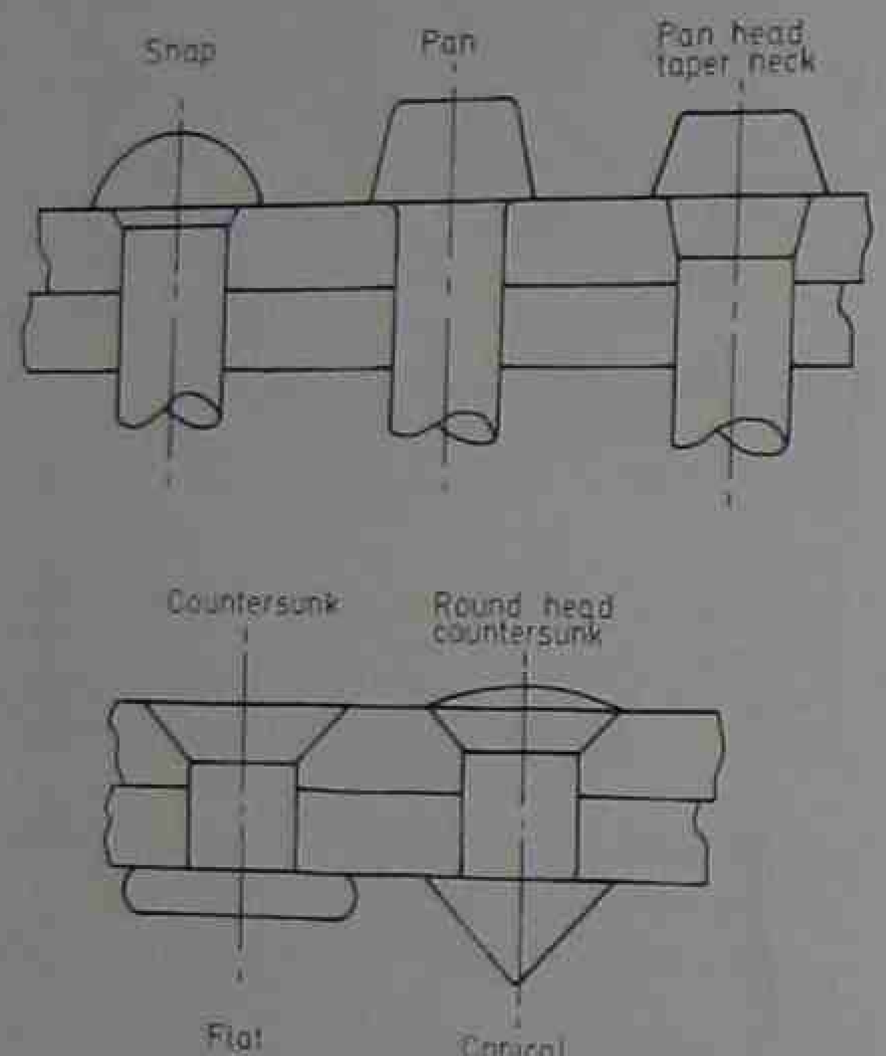


Figure 1.42 *Types of rivet*

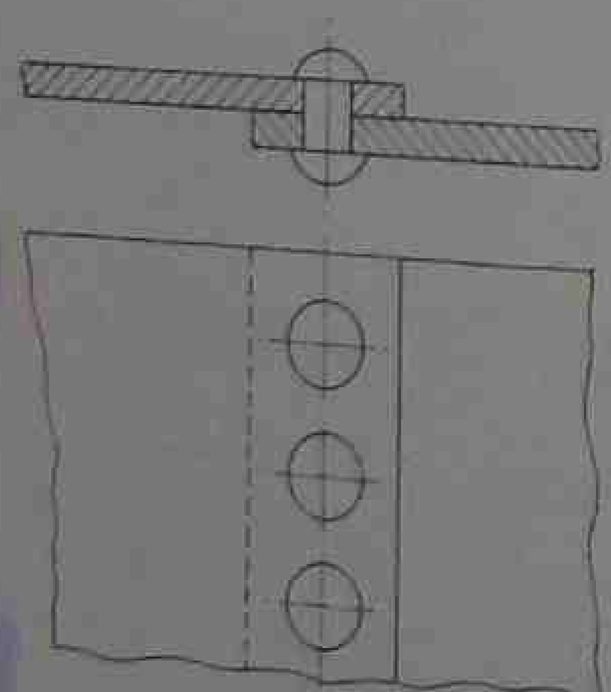


Figure 1.43 *Riveted lap joint*

Riveted joint Metal plates may be joined together by overlapping the edges and riveting using one or several rows of rivets. Alternatively, the plates may be placed edge-to-edge and a butt joint made with butt straps on one or both sides of the joint. One or more rows of rivets are passed through the plates and straps on each side of the joint.

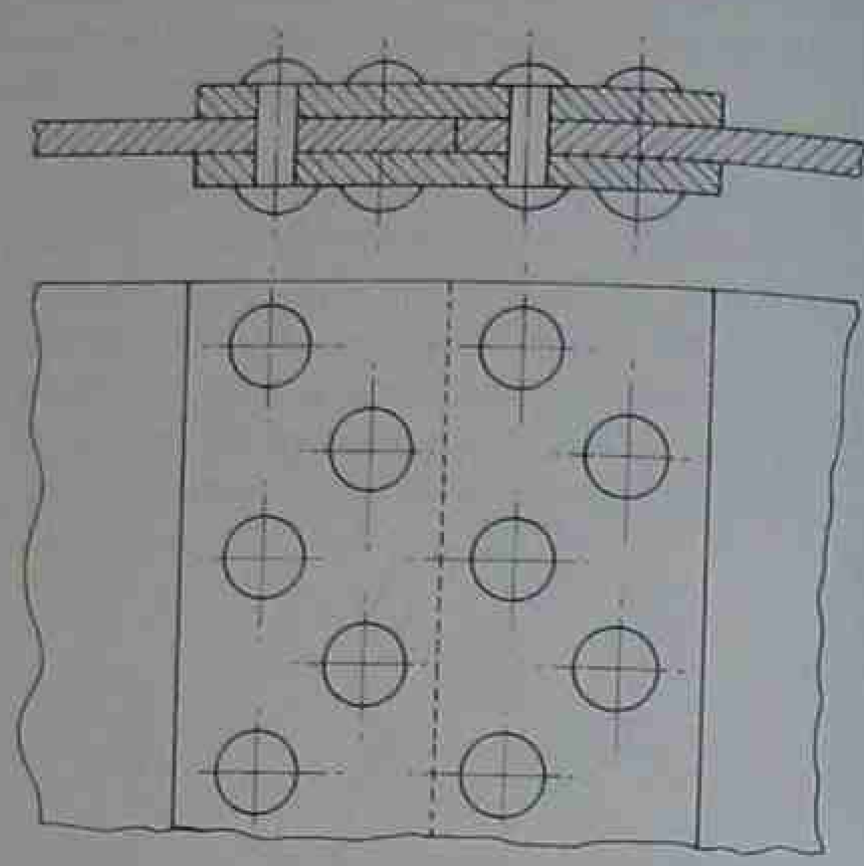


Figure 1.44 *Double riveted butt joint with two straps*

Flush rivet (aircraft type) The head of this rivet is flat and countersunk so that it is flush with the face of one of the plates. This is advantageous in aircraft construction where a smooth surface is required for aerodynamic reasons. The rivets are mostly made of aluminium.

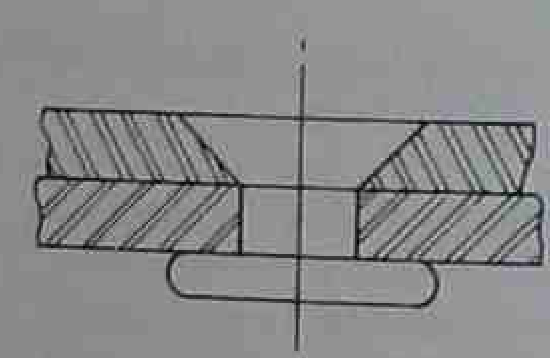


Figure 1.45 *Flush rivet*

Tubular rivet This consists of a piece of soft metal tubing the ends of which are deformed by a special tool. They are used for joining thin metal sheets.

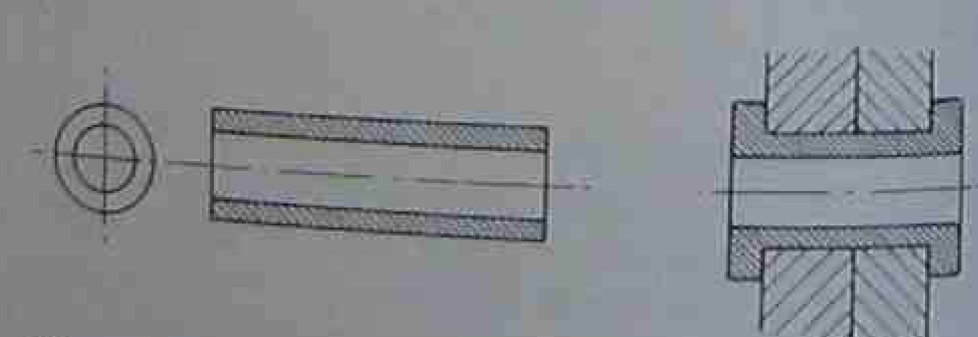


Figure 1.46 *Tubular rivet*

Pop rivet This is a type of tubular rivet which initially has a hard steel pin passing through it. When the rivet is fitted by means of a special tool the head of the pin closes the rivet and snaps off. Riveting is done from one side of the plate.

Figure 1.47 *Pop rivet*

Explosive rivet The end of the shank of this rivet is hollowed out to take a small explosive charge. When this is exploded the protruding shank expands to form a joint. This type is used extensively in the aircraft industry.

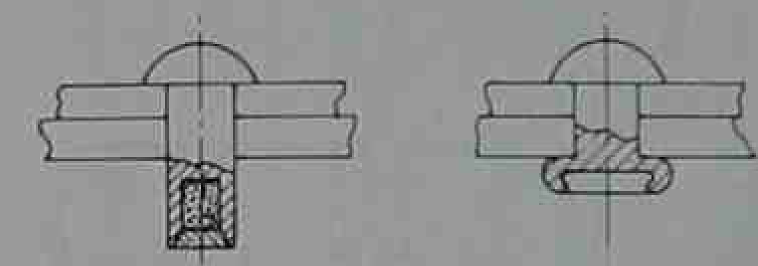


Figure 1.48 *Explosive rivet*

PINS

This term refers to a large range of components consisting basically of a piece of rod or bar, usually of circular section and either solid or hollow. They are used for fixing, locating and load carrying.

Plain pin This is simply a piece of bar which, in most cases, is machined to a good finish and accuracy. It is used for locating parts.

Dowel pin A dowel pin is a straight circular pin, sometimes with a head, which is accurately fitted into holes to locate two or more parts together. This is often used in conjunction with bolts and studs.

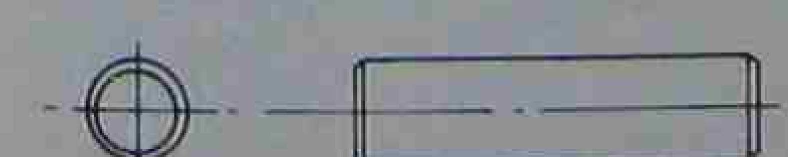


Figure 1.49 *Plain pin*

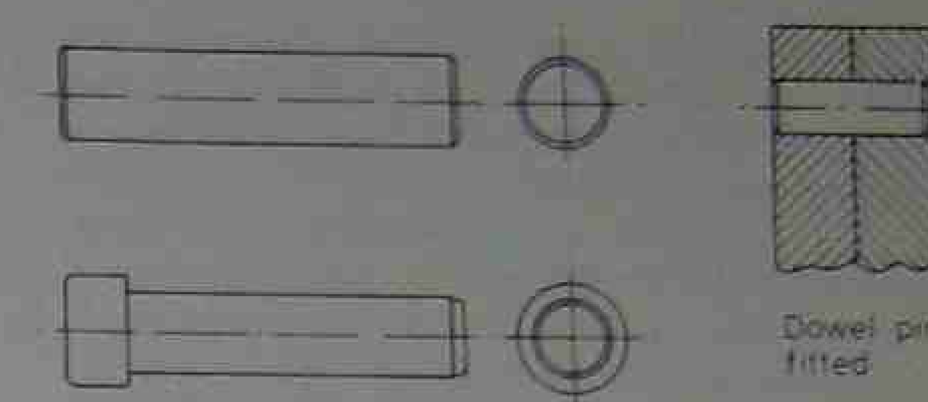


Figure 1.50 *Dowel pins*

Taper pin A type of dowel which has a fine taper so that a tight fit is obtained when it is lightly tapped into a hole which has been drilled and then finished with a taper reamer. Taper pins are often used in conjunction with a sleeve for connecting shafts transmitting low power (*see* Section 2.2 on Shaft couplings).

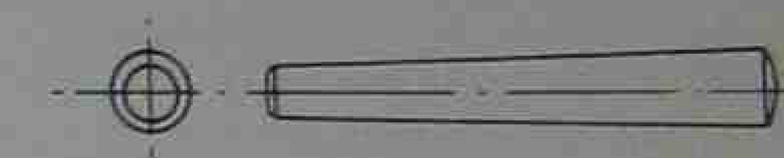


Figure 1.51 *Taper pin*

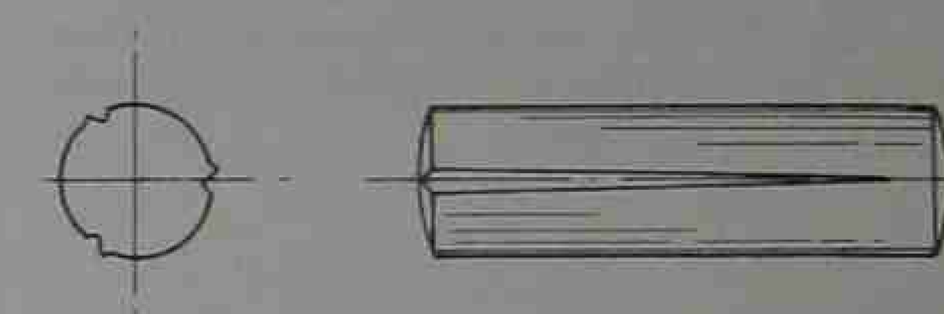


Figure 1.52 *Grooved pin*

Grooved pin This is a straight, circular and solid pin which has longitudinal grooves with raised edges formed by rolling. A tight fit in the hole is achieved when the pin is hammered in. Grooved pins are useful as keys for light power transmission.

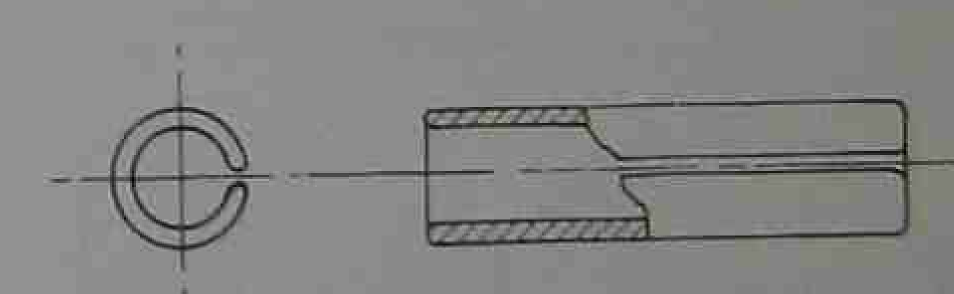


Figure 1.53 *Roll pin*

Roll pin A roll pin is a spring steel tube, with a longitudinal slit, which is driven into a slightly smaller hole so that the slit closes to give a tight fit.

It is easier to fit than solid pins and taper pins, and an example of its use is for attaching hand wheels to valve spindles.

Figure 1.54 *Split pin (cotter pin)*

Split pin (cotter pin) A pin formed from half-round wire folded to give a shank and a head. The pin is passed through mating holes in parts and the protruding ends bent outwards to secure it. It is used mainly for locking slotted nuts.

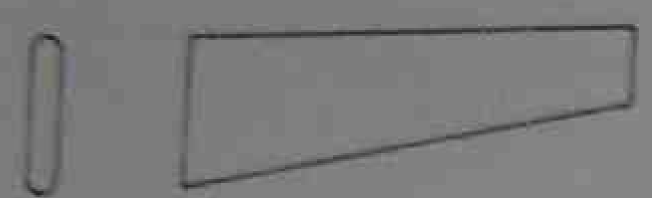


Figure 1.55 *Cotter*

Cotter A tapered pin of rectangular cross-section. It is used to provide a rigid joint between rods under an axial force. The cotter fits into slots in the ends of the rods and may easily be removed if desired (*see* Section 1.7 Cottered joint).

CLIPS

Clips are used for attaching pipes, tubes and cables to other parts to prevent rattling and provide location relative to those parts.

Band hose clip In this type of clip a steel band formed into a circle is tightened onto a hose by means of a screw and nut.

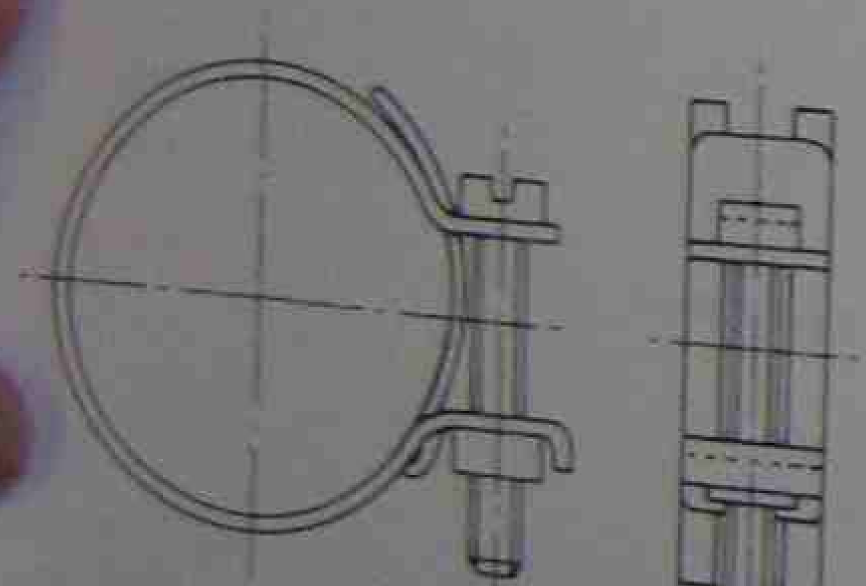


Figure 1.56 *Band hose clip*

Worm-drive hose clip (Jubilee clip) In another type, known as a worm-drive *Jubilee clip*, the screw engages with serrations in the steel band instead of a nut. These clips are used for clamping rubber and plastic hoses to metal pipes.

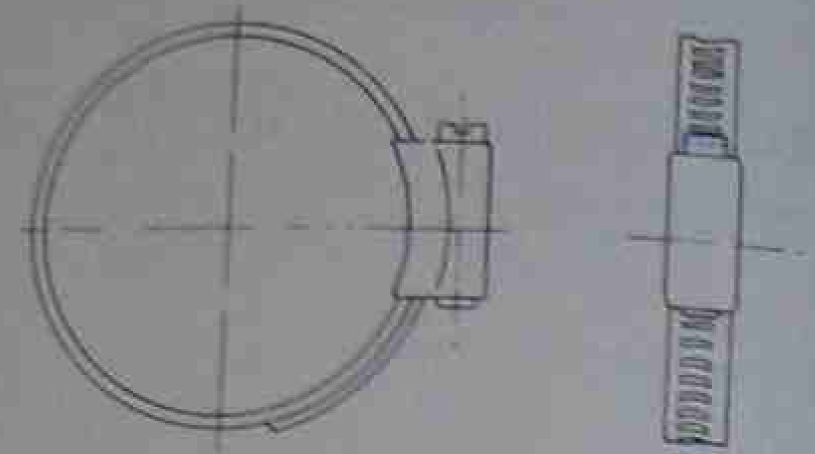


Figure 1.57 *Worm-drive hose clip (Jubilee clip)*

Spring wire hose clip A loop of spring wire with projecting ends is used to clamp a hose onto a pipe. The clip is fitted or removed by opening the loop with pliers.

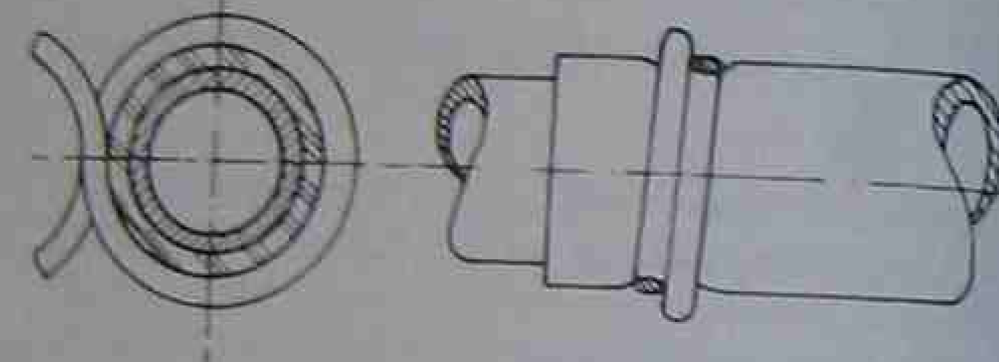


Figure 1.58 *Spring wire hose clip*

Pipe and cable clips These are used for fastening pipes, tubes and cables to machines to provide location and prevent vibration. They are usually made of metal strip, sometimes ribbed for strength, and are fastened by screws or rivets.

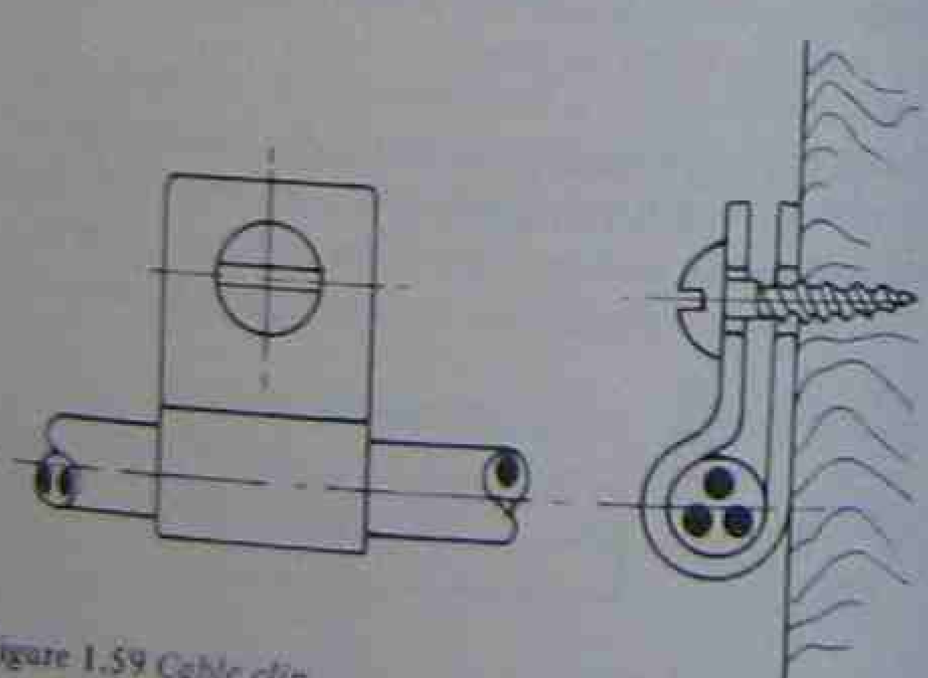


Figure 1.59 *Cable clip*

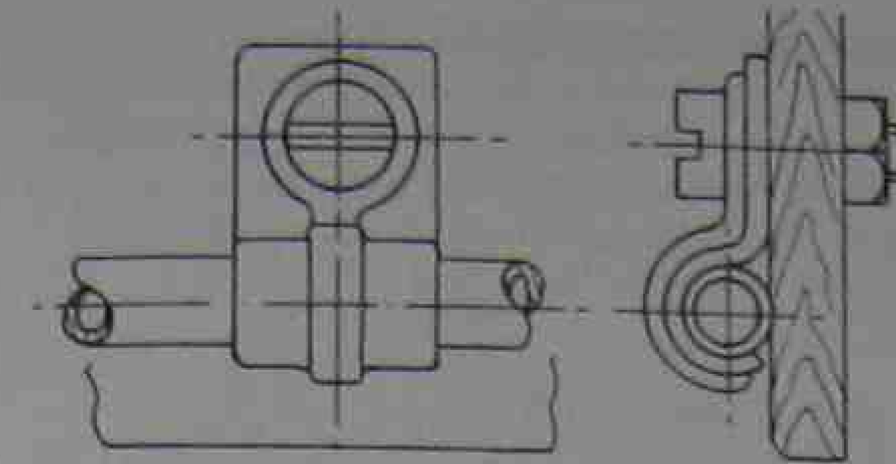


Figure 1.60 *Pipe clip*

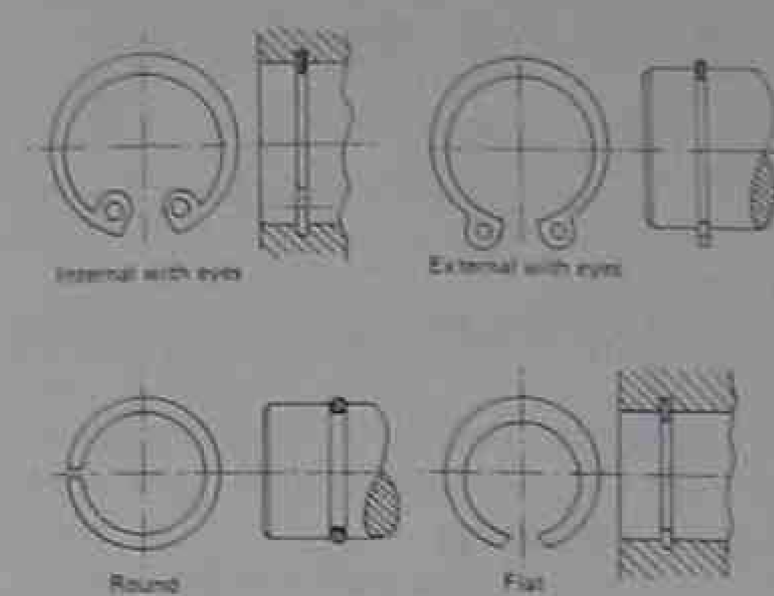


Figure 1.61 *Circlips (retaining rings)*

Circlip (retaining ring) A spring steel clip in the form of an incomplete ring which fits tightly into a circumferential groove on a shaft or in a bore and locates parts axially. Circlips may be of circular or rectangular cross-section. Rectangular section circlips may have internal or external eyes for easier fitting using a special tool.

1.2 SCREW THREADS

A screw thread is formed by cutting or forming a helical groove, or thread, on the surface of a circular bar or in a circular hole. The thread may be right-handed or left-handed, and of various cross-sections such as V, square, trapezoidal, etc. External threads of large diameter are usually produced by machining with a single point tool. Smaller bar is screwed in a lathe using a die box, or by hand using stocks and dies. Threads on bolts, screws and studs are often produced by rolling. The bar is formed by means of a pair of flat or circular dies having the thread form. The method is cheaper for large quantities and gives a better finish as well as a higher strength.

Internal threads of large size are machined with a single point tool. Smaller holes are machine-tapped in quantity or hand-tapped for small numbers off.

V threads are used mainly for fasteners while square and trapezoidal threads are used for power transmission.

V THREAD

A V thread is in the form of an isosceles triangle with the 'crest' and 'root' either flattened or rounded. The main dimensions are: pitch (the distance between adjacent threads); major and minor diameters; V angle; area of cross-section at the bottom of the threads.

V threads are used almost exclusively for fasteners such as nuts and bolts.

The main types of thread form are: Whitworth (BSW) 55°; British Standard Fine (BSF) 55°; British Association (BA) 47.5°; Metric (ISO) 60°; USA Standard 60°; Unified Coarse and Fine (UNC-UNF) 60°.

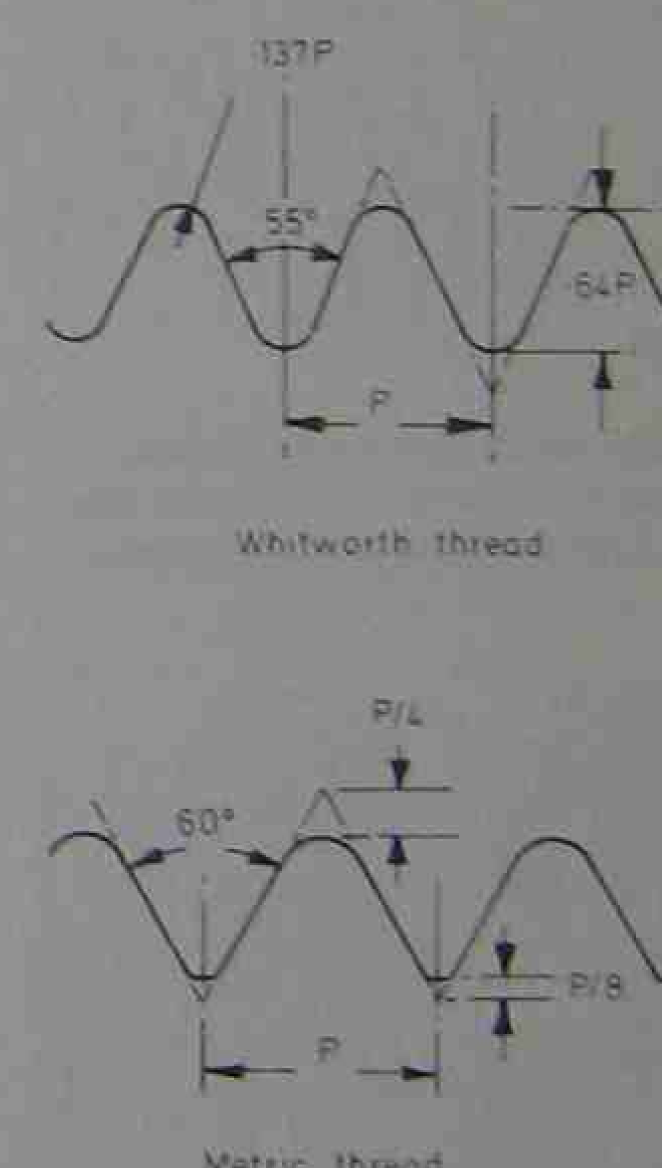


Figure 1.62 *Thread details*

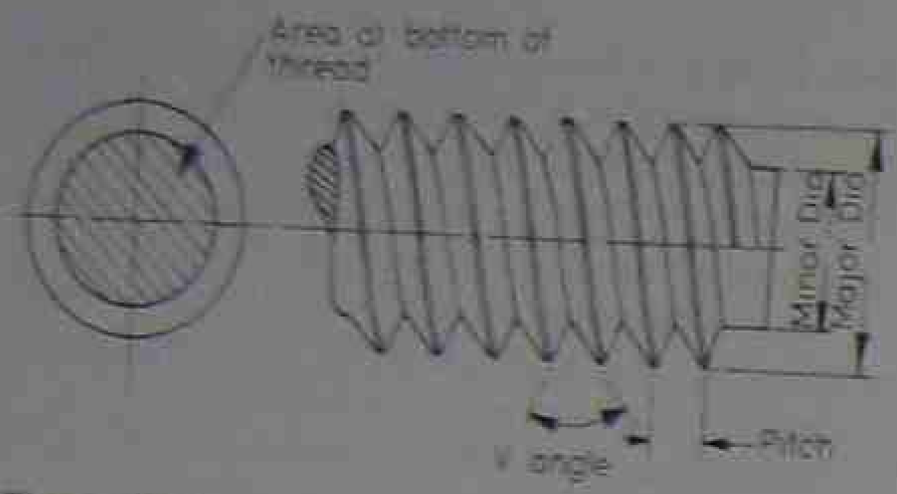


Figure 1.63 *V screw thread*

SQUARE THREAD

The square thread is mechanically strong and is used mainly for power transmission. There is no radial force on the nut and friction is low.

Square threads, and Acme and buttress threads, are formed by machining on a lathe, whereas V threads are often cut by hand with taps and dies.

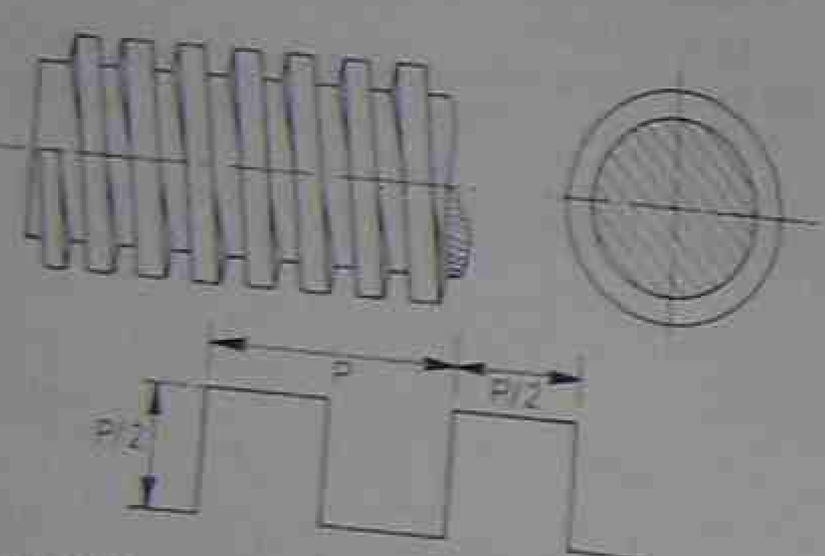


Figure 1.64 *Square thread*

ACME THREAD

Used for power transmission, this is a trapezoidal thread of greater root strength and easier to cut than the square thread. It is sometimes used for the lead screw in lathes.

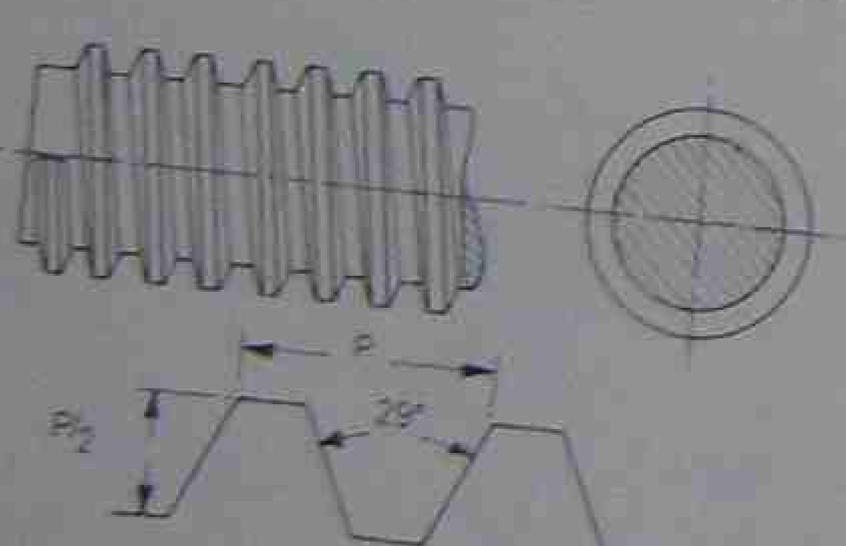


Figure 1.65 *Acme thread*

BUTTRESS THREAD

This is a thread used for power transmission which combines the advantages of both square and Acme threads. The load must be applied in one direction only, and that is on the vertical face.

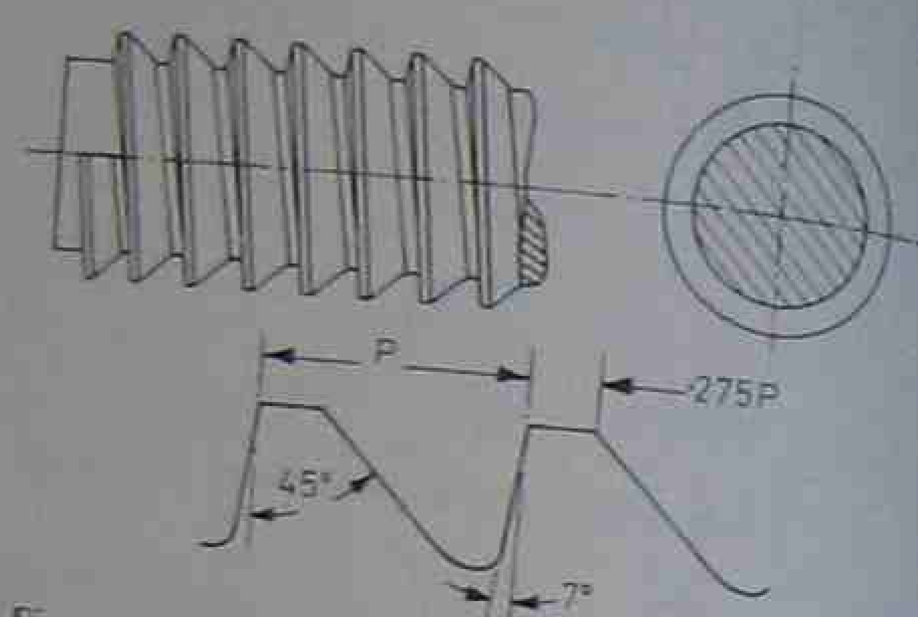


Figure 1.66 *Buttress thread*

MULTI-START THREAD

To obtain a larger pitch without increasing the depth of thread and reducing the strength, two or more threads may be cut on the same screw side by side. The nut advances N times the pitch, where N is the number of thread 'starts' in each revolution. The advance is known as the 'lead'.

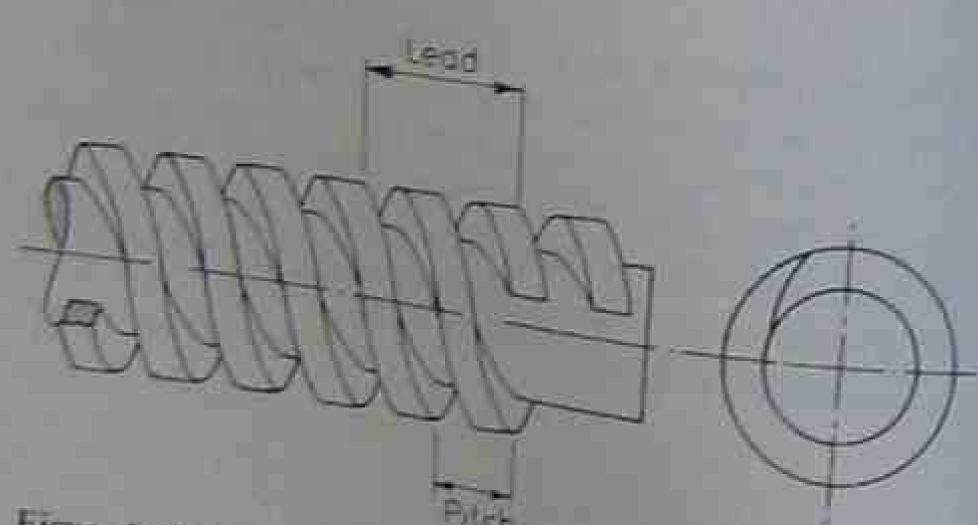


Figure 1.67 *Two-start square thread*

BALL-BEARING POWER SCREW

This is an extremely low friction power screw in which the screw and nut have opposing helical grooves of part-circular cross-section to suit ball bearings situated between them.

Power is transmitted between the screw and nut via the ball bearings which circulate continuously through a tube attached to the outside of the nut.

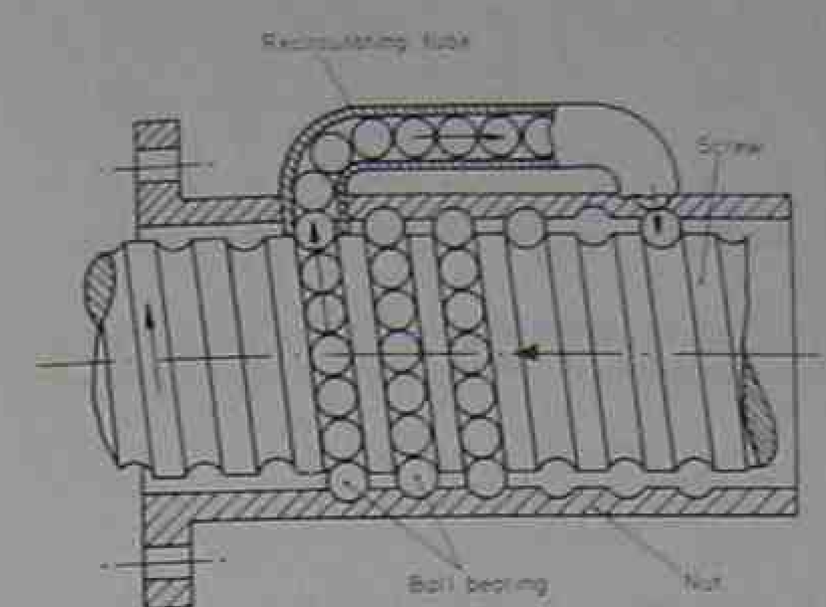


Figure 1.68 *Ball-bearing power screw*

EFFICIENCY OF SCREW THREADS

The efficiency of screw threads is defined as the ratio of the work done by the nut to the work put into the screw.

This may be as low as 20% for a poorly-lubricated V thread but up to 80% for a well-lubricated square or Acme thread. The ball bearing screw has an efficiency approaching 100%.

1.3 SPRINGS

Springs are used extensively in engineering to control movement and to apply forces in machines. They limit impact forces, reduce vibrations by storing energy, and are often used to measure forces.

HELICAL COIL SPRING

This consists of a wire of circular, square, rectangular or other section wrapped around a cylinder to form a helix.

HELICAL COMPRESSION SPRING

In a helical compression spring the coils are sufficiently open to allow shortening of the spring under a compressive load. The ends of the spring are usually flattened and ground to provide a seating and the wire is generally round, but rectangular section wire is also used.

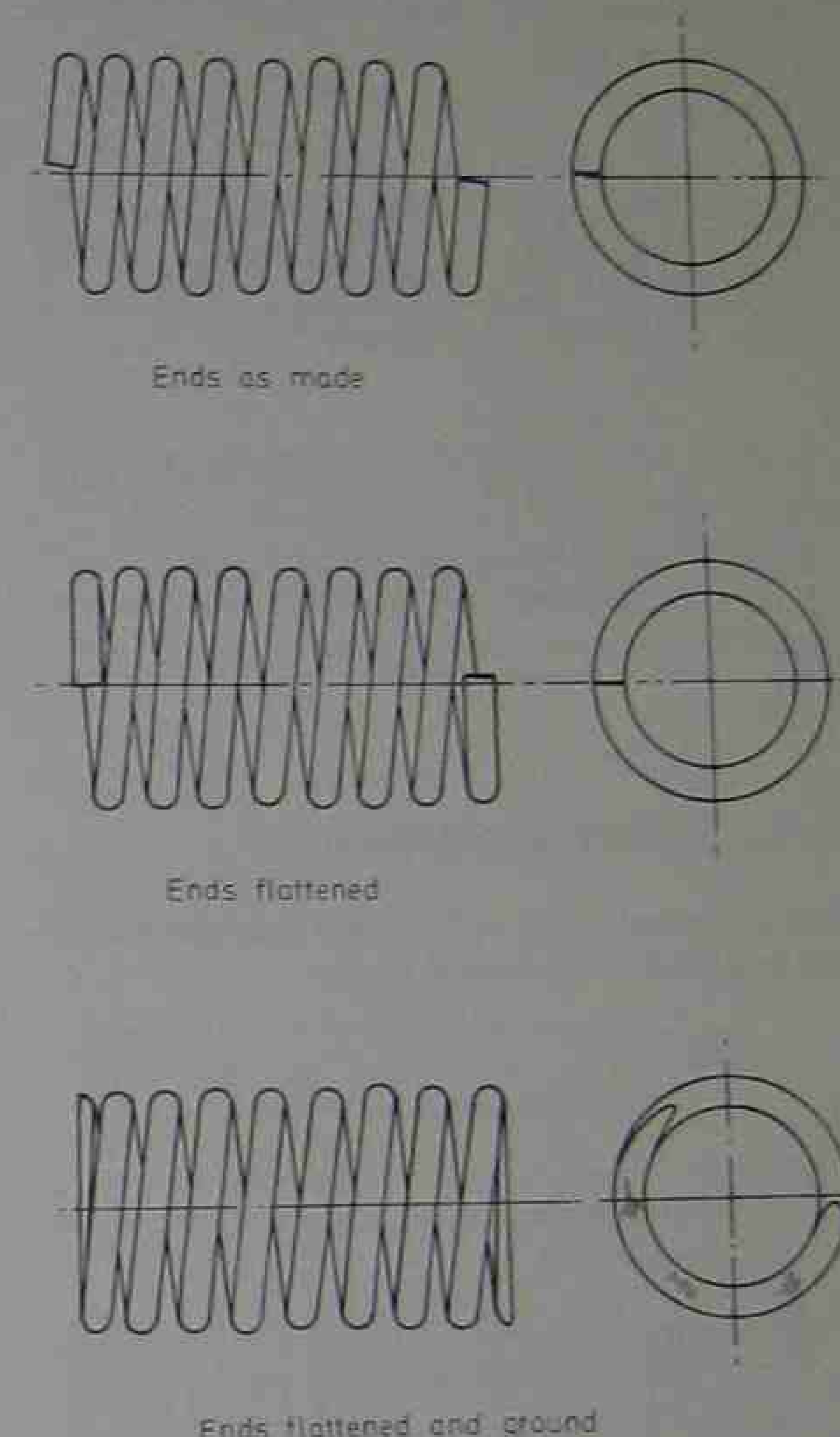


Figure 1.69 *Helical compression springs*

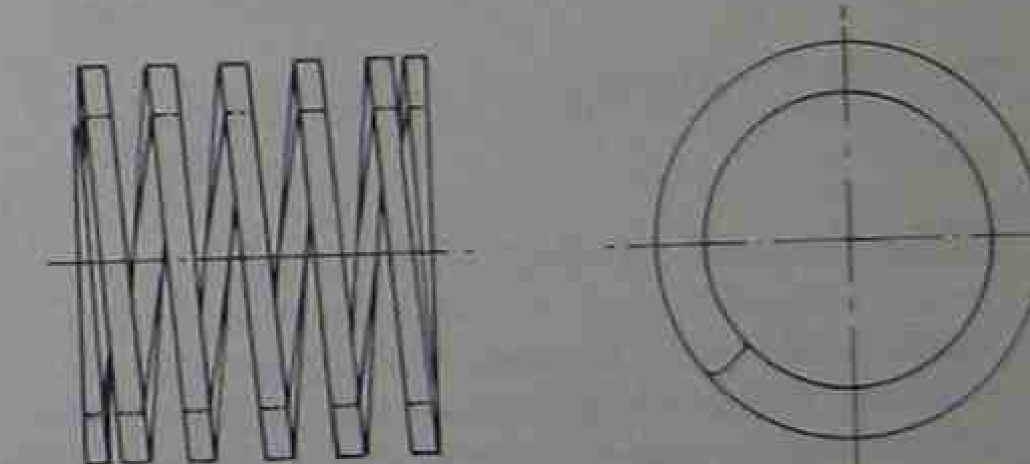


Figure 1.70 *Rectangular-section spring*

HELICAL TENSION SPRING

The ends of helical tension springs are formed into hooks of various types and the springs are usually pre-tensioned so that a small initial load is required to open the coils.

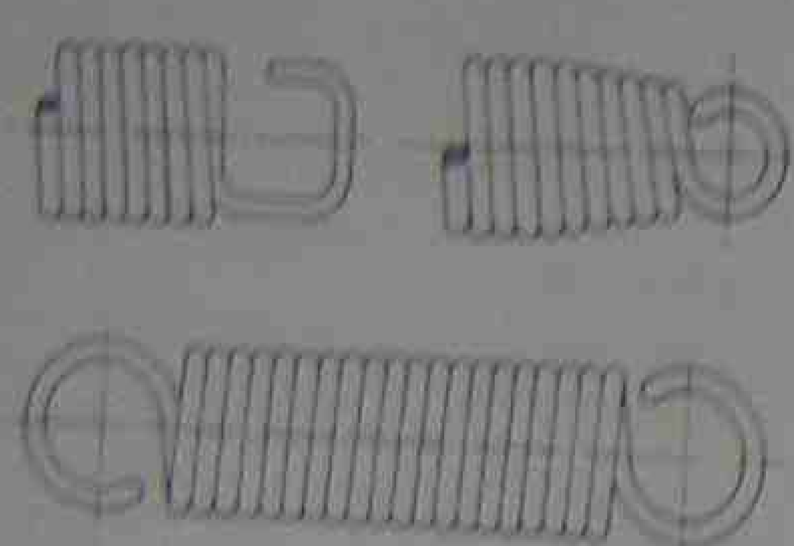


Figure 1.71 Helical tension springs

HELICAL TORSION SPRING

This is a helical spring with suitable ends which is subject to a twisting moment or torque.

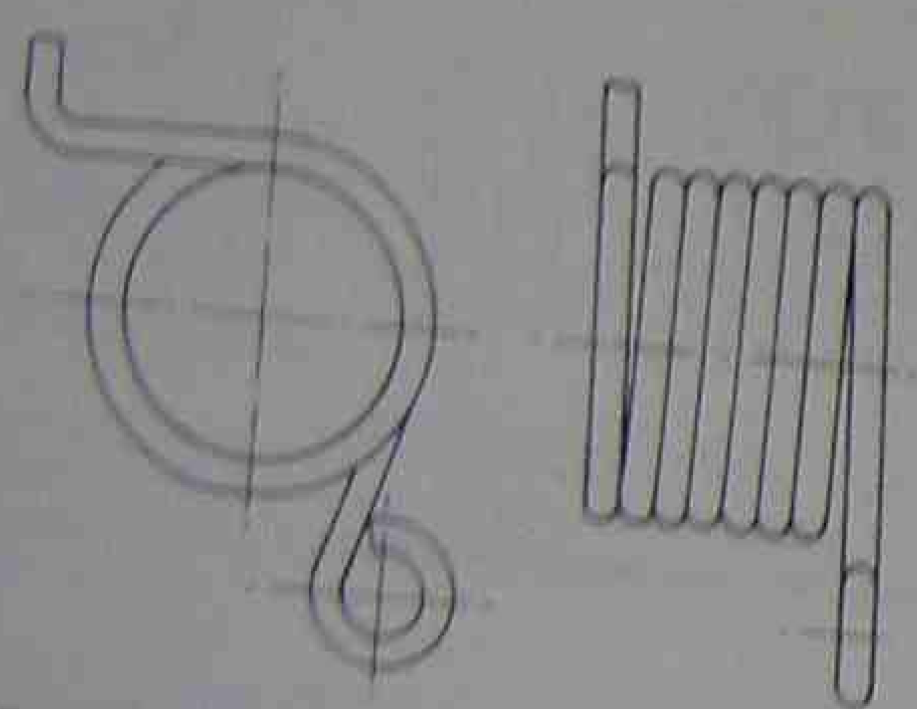


Figure 1.72 Helical torsion spring

HELICAL SPRING DATA

The important dimensions are: mean diameter of coils, wire size, free length, loaded length, clearance between coils, number of coils.

It is also necessary to know: design load, material, spring rate or stiffness (the load per unit deflection).

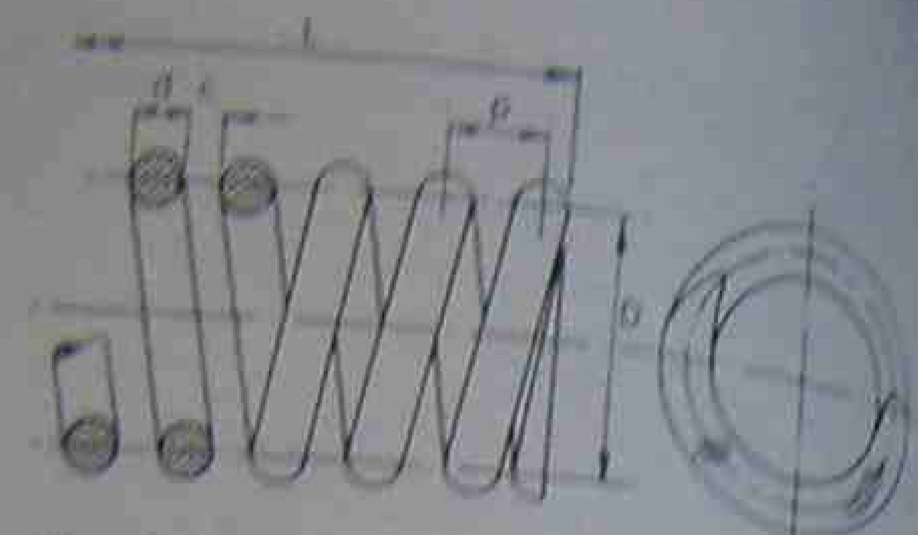


Figure 1.73 Helical spring dimensions: D = mean diameter of coils; d = wire diameter; c = coil clearance; L = spring length; p = pitch

HELICAL SPRING MATERIALS

Most springs are made of hard, drawn, spring steel but many other metals are used including chrome vanadium steel, phosphor bronze, beryllium copper and stainless steel. Drawn steel wire is often termed *piano* or *music wire*.

SPIRAL TORSION SPRING

In this spring, the wire of round or rectangular cross-section is wound in a flat spiral and the spring is deformed under the application of a twisting moment or torque. Small round wire spiral springs are used in locks and catches and as return springs for control rods. The clock spring is a case in which rectangular wire is used.

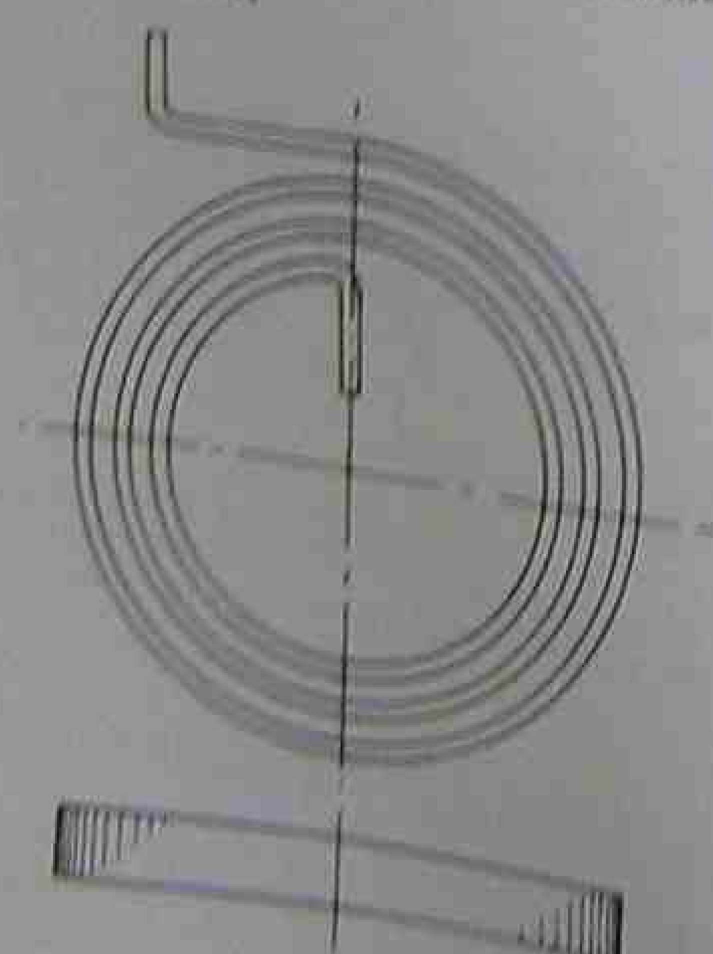


Figure 1.74 Spiral torsion spring (rectangular section)

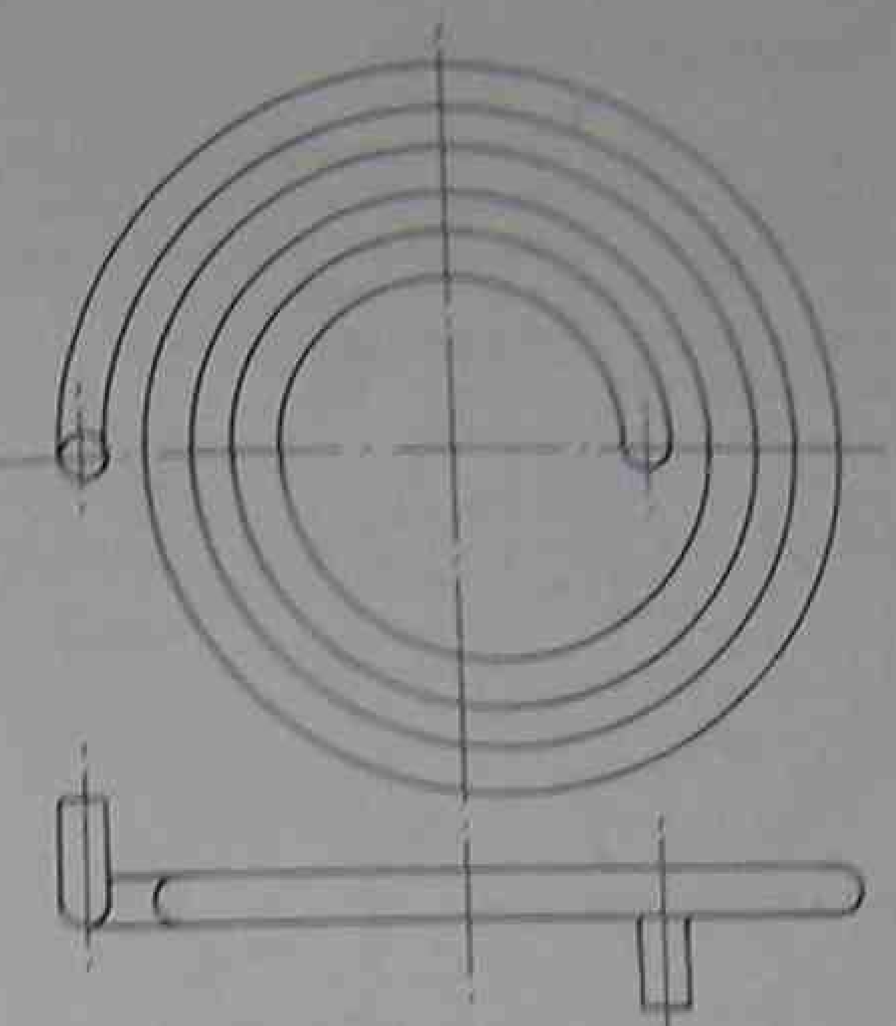


Figure 1.75 Spiral torsion spring (round wire)

CONICAL SPRING (UPHOLSTERY SPRING)

This is a helical compression spring in which the coil diameter changes from one end to the other so that a conical form is obtained. The larger diameter coils, which are the least stiff, close first so that the spring becomes stiffer as the load is increased.

An advantage is that the length of the spring when fully compressed is only the diameter of the wire. These springs are used extensively for upholstery.

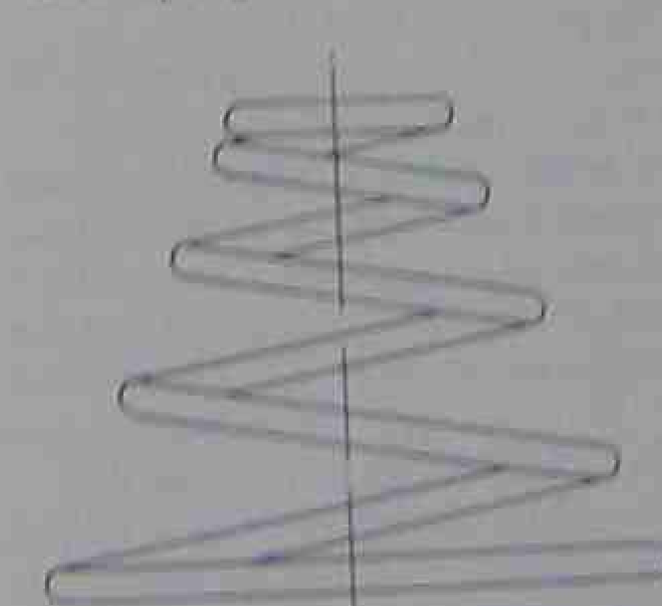


Figure 1.76 Conical spring (upholstery spring)

VOLUTE SPRING

A volute spring is made of thin rectangular strip wound into a conical spiral with each successive coil overlapping the previous one, and the cone flattens under sufficient load.

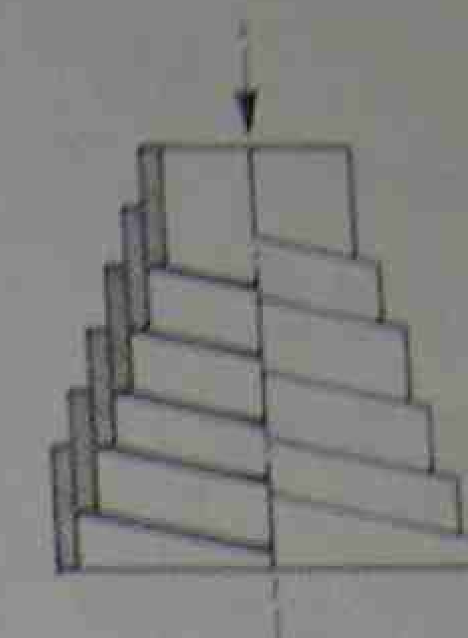


Figure 1.77 Volute spring

LEAF SPRINGS

These springs consist of one or several flat or slightly curved bars of steel held at one end and loaded at the other (cantilever) or held at both ends and loaded at the centre (beam).

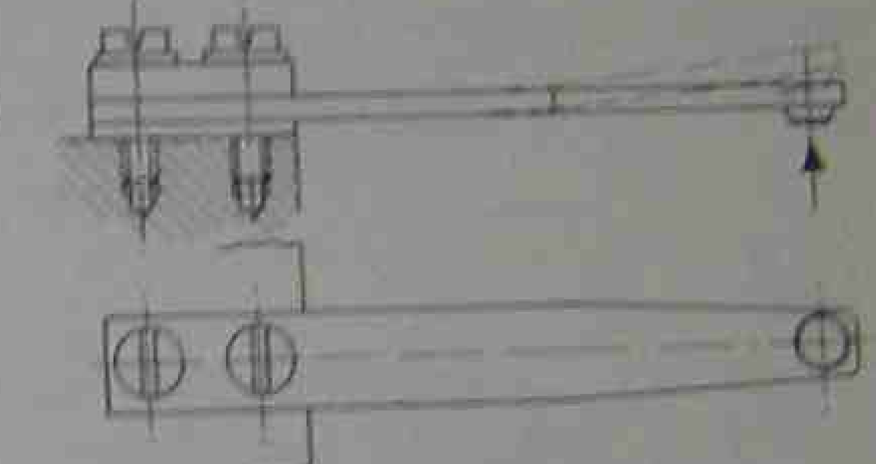


Figure 1.78 Cantilever leaf spring (electrical contact)

Figure 1.79 Beam leaf spring

LAMINATED LEAF SPRING (CARRIAGE SPRING)

A spring used extensively for vehicle suspensions. It consists of several flat strips clamped together and these vary in length to give a higher strength. The ends of the longest strip (usually at the top) are formed into eyes which take the suspension bolts, and the load is applied at the centre. Half of one of these springs may be used as a cantilever spring (Figure 1.80).

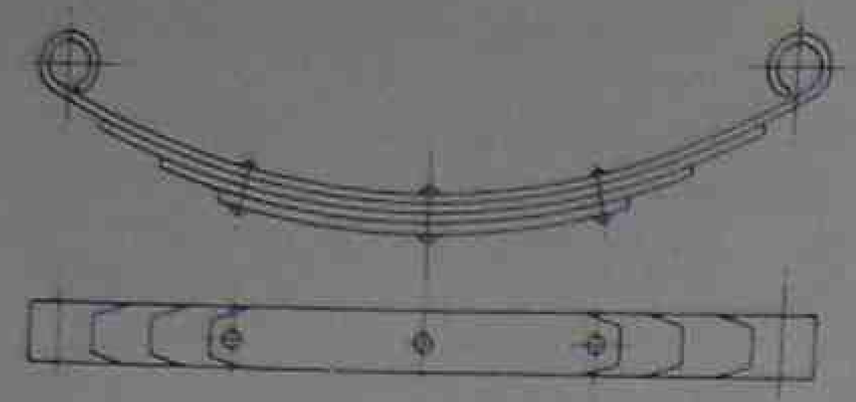


Figure 1.80 *Laminated leaf spring (carriage spring)*

RUBBER SPRING

Rubber, or synthetic rubber, bonded to metal plates is used extensively for springs under all types of loading. The spring may take the form of a block of rubber between two flat plates or a sleeve between two cylinders. Rubber springs have high internal damping and are therefore useful for damping vibrations.

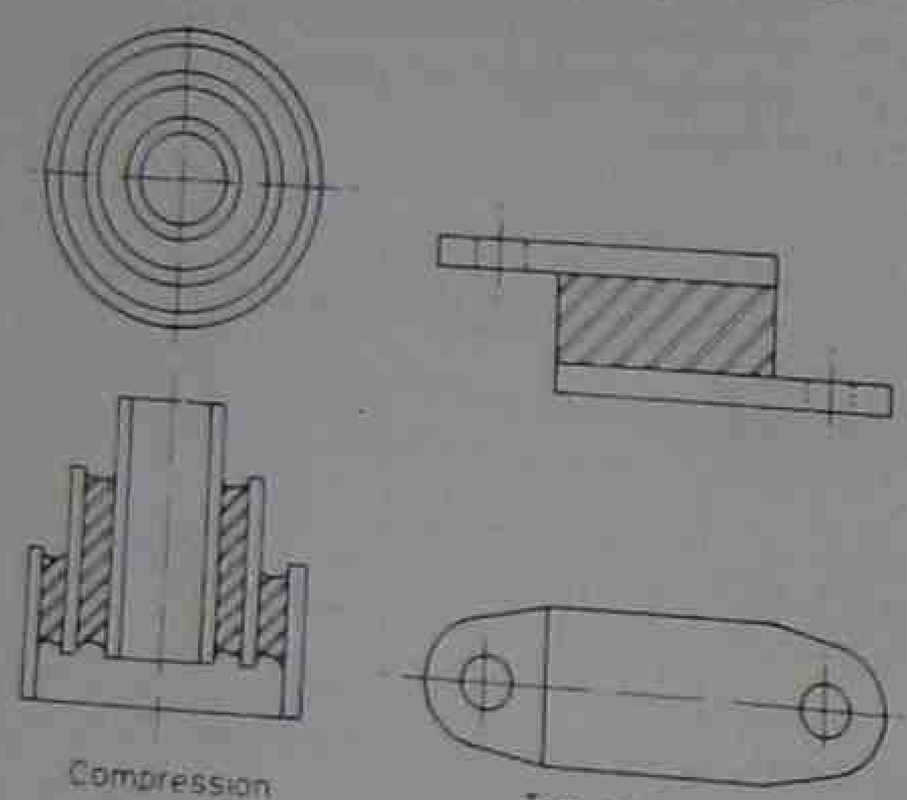


Figure 1.81 *Rubber springs*

AIR SPRING (PNEUMATIC SPRING)

In an air spring the elasticity of the air contained in a cylinder or rubber bellows is used to provide cushioning. The bellows type spring is sometimes used in vehicle suspensions.

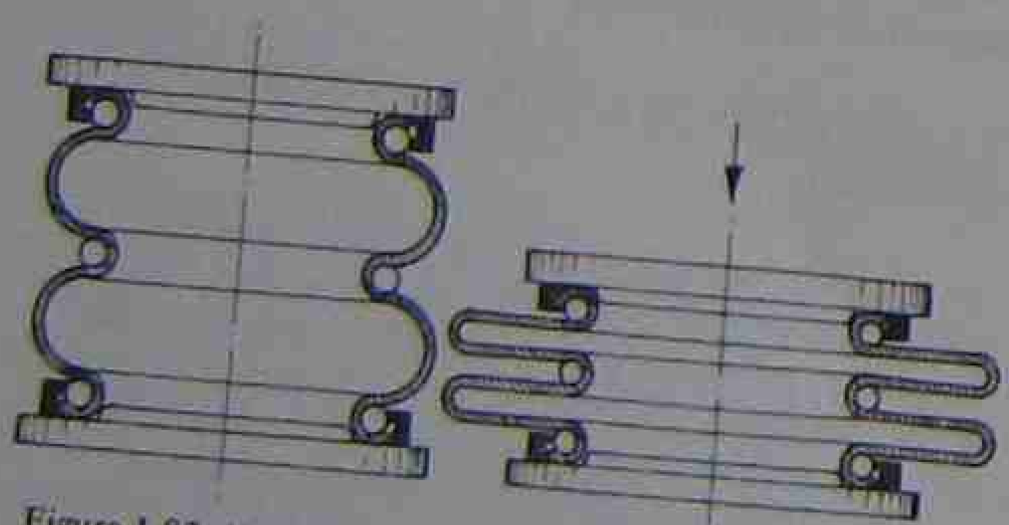


Figure 1.82 *Air bellows spring*

TORSION BAR SPRING

A straight, solid or hollow bar usually of circular but sometimes rectangular cross-section which may be used as a torsion spring. One end is clamped and the other provided with a lever to which the load is applied. They are used a great deal in vehicle suspensions.

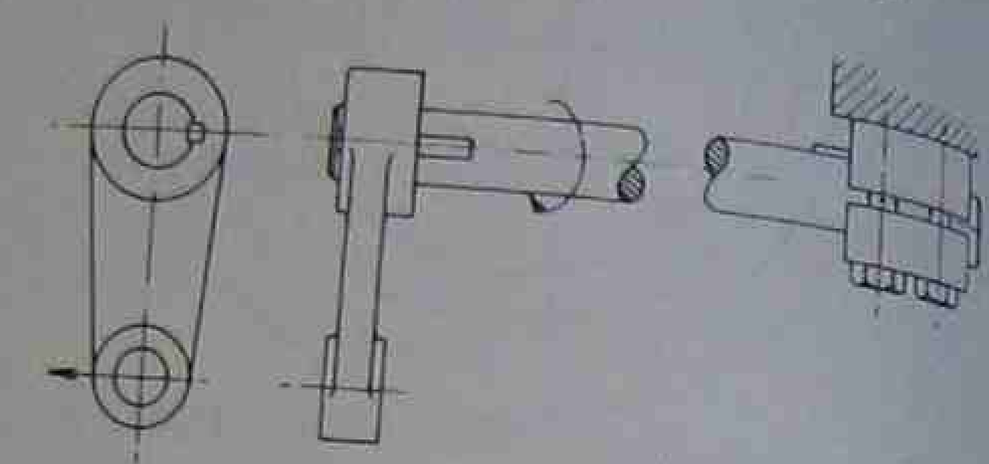


Figure 1.83 *Torsion bar spring*

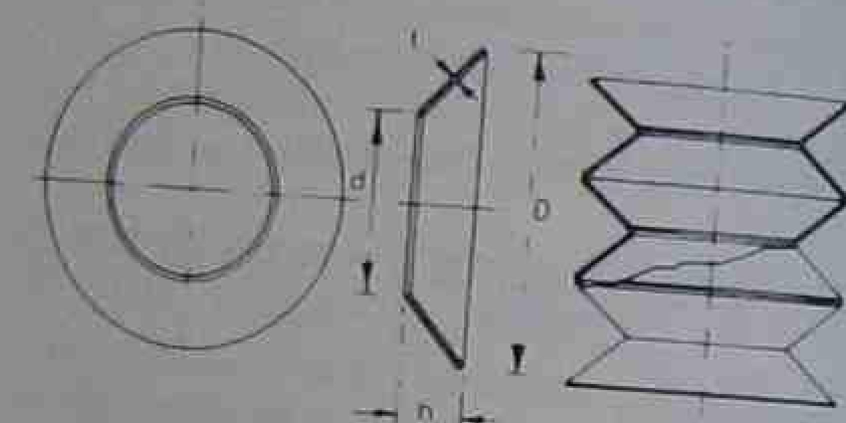


Figure 1.84 *Belleville washer (diaphragm spring)*

BELLEVILLE WASHER (DIAPHRAGM SPRING)

The Belleville washer is an annular steel ring which is slightly dished (or coned) and is loaded axially. To decrease the stiffness, a number of washers may be used in series. This type of spring saves considerable space and provides a wide variation in the shape of the load deflection curve as the ratio h/t is altered.

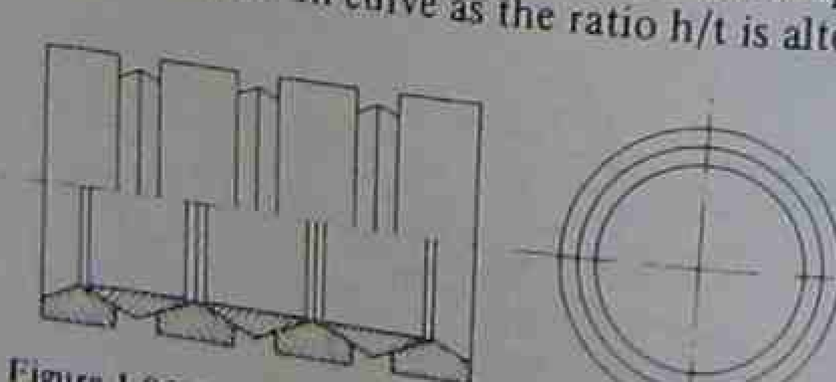


Figure 1.85 *Ring spring*

RING SPRING

This has alternate internally and externally bevelled rings stacked in a cylinder which is axially loaded. Under axial compression the internal rings contract and the external rings expand. These springs have a very high stiffness and high frictional clamping.

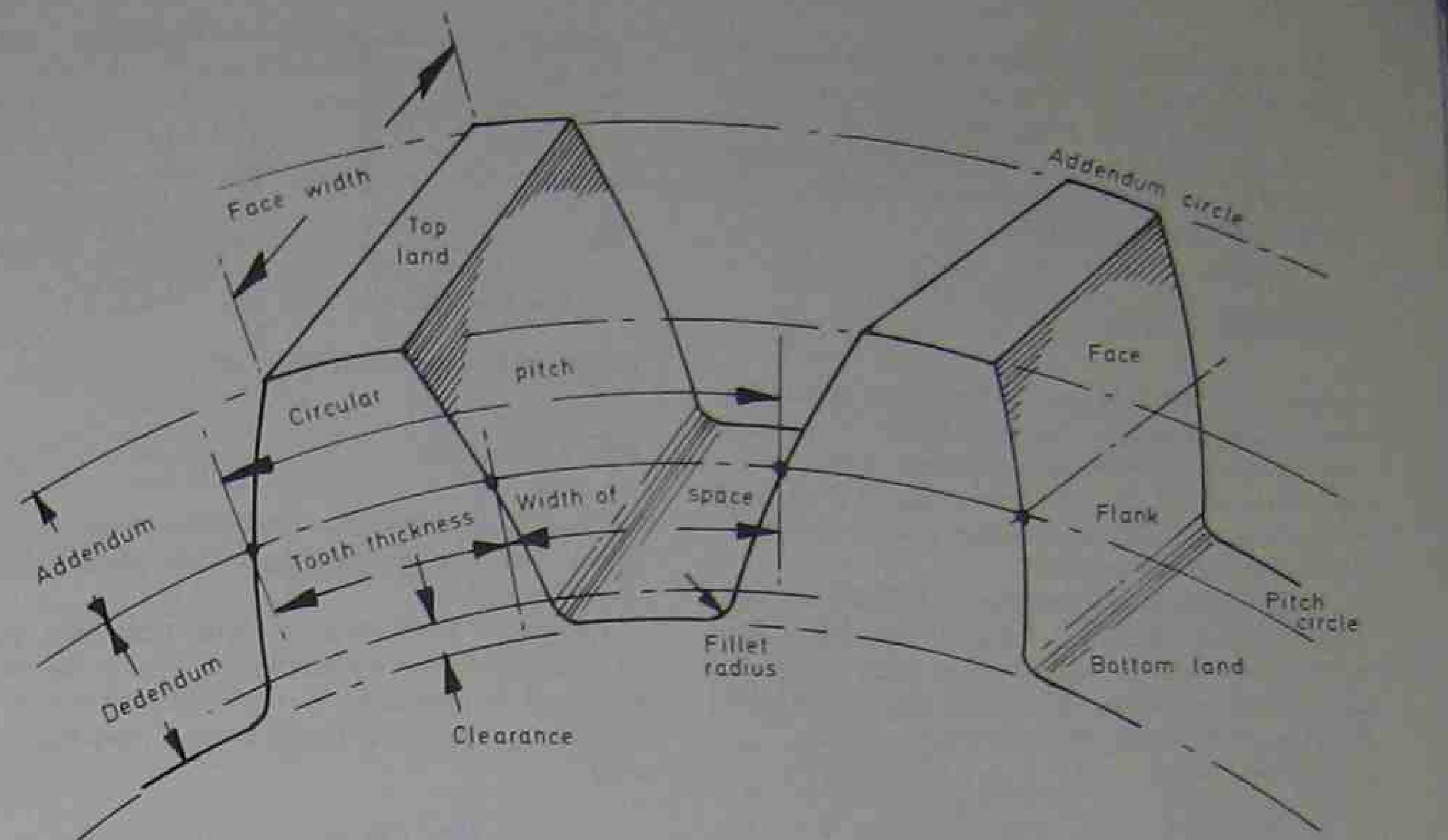


Figure 1.86 *Nomenclature of gear teeth*

1.4 GEARS

Gears are rotating machine elements which transmit motion and power without slip by means of a series of engaging projections known as *teeth.*

SPUR GEARS

Spur gears are disks with straight teeth on their peripheries which run parallel to the axis of rotation. The shafts carrying mating gears are parallel to one another and the ratio of the numbers of teeth on the two gears determines the ratio of the speeds and is inversely proportional to it.

The diameter of the equivalent friction disk is known as the 'pitch circle' and the distance between corresponding points on two adjacent teeth on the pitch circle is called the 'circular pitch'.

The number of teeth per inch of pitch circle diameter (British sizes) is the 'diametral pitch', and the pitch circle diameter in mm divided by the number of teeth (metric sizes) is the 'module'.

The height of that part of a tooth above the pitch circle is known as the 'addendum' and that part below it the 'dedendum'. The corresponding circles are the 'addendum circle' and the 'dedendum circle'.

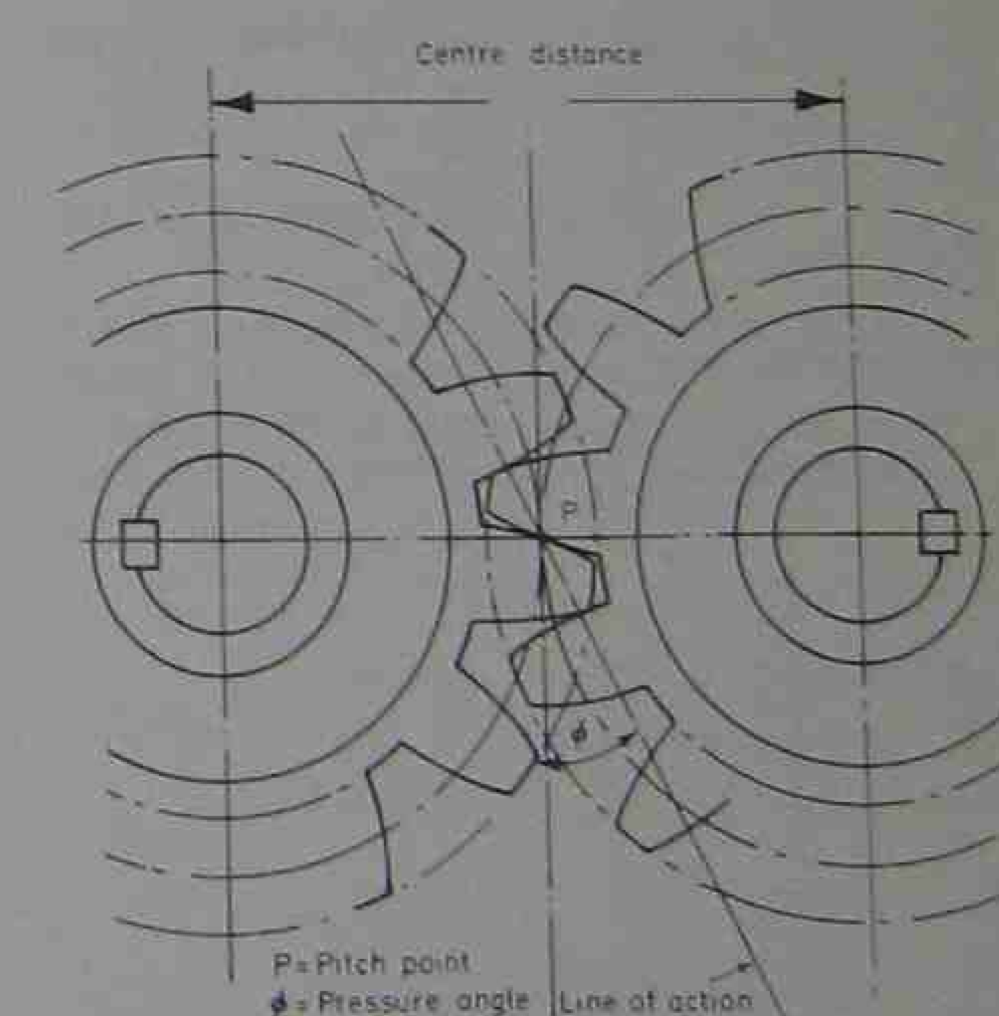


Figure 1.87 *Spur gears in mesh*

INVOLUTE GEAR

Most gears are made with the mating faces in the form of an 'involute' curve. An involute is the locus of (or curve described by) a point on a string as it is unwrapped from around a cylinder. This shape gives the correct rolling action when the teeth mesh. The path along which the teeth make contact is straight and is called the 'line of action'. The angle of this line to the tangent is the 'pressure angle'. The distance between the centres of a pair of meshing gears is called the 'centre distance'.

HELICAL GEAR

The teeth of a helical gear are formed on a helix on the cylindrical surface, that is, in the form of a coarse screw thread. The gradual engagement of the teeth and the resultant smooth transfer of load from tooth to tooth permits the use of higher powers with less noise than for straight teeth.

To eliminate axial thrust *double helical gears* are employed. Two gears with helices of opposite hand are mounted on each shaft. The two gears are often made in one piece, sometimes with a gap between the two sets of teeth. These are often known as *herringbone gears*.

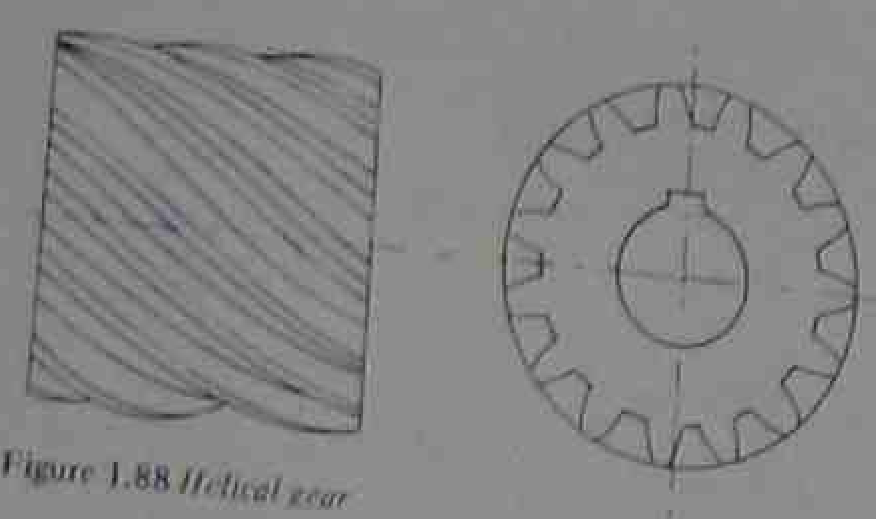


Figure 1.88 *Helical gear*

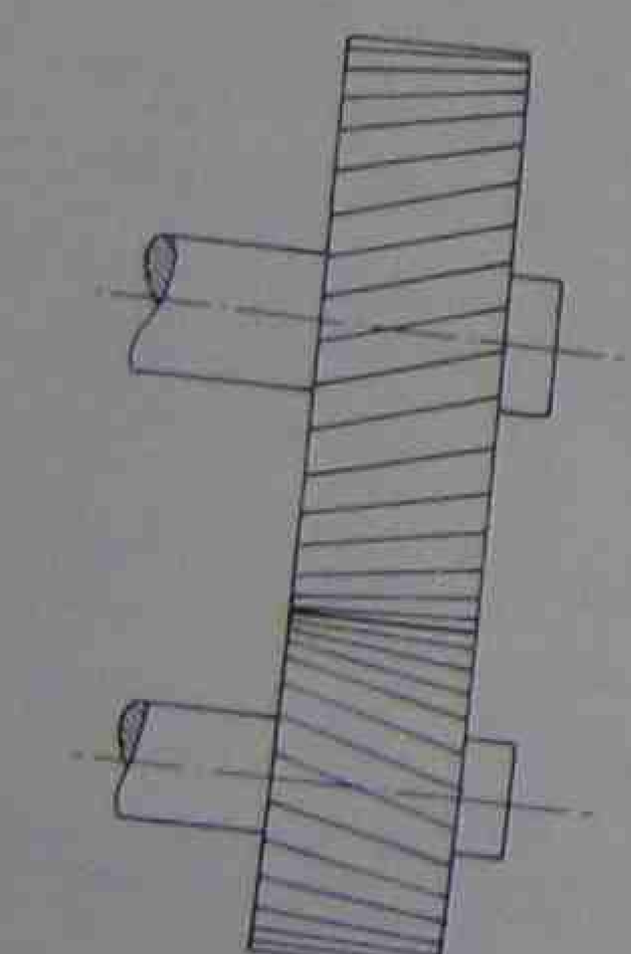


Figure 1.89 *Helical gears in mesh*

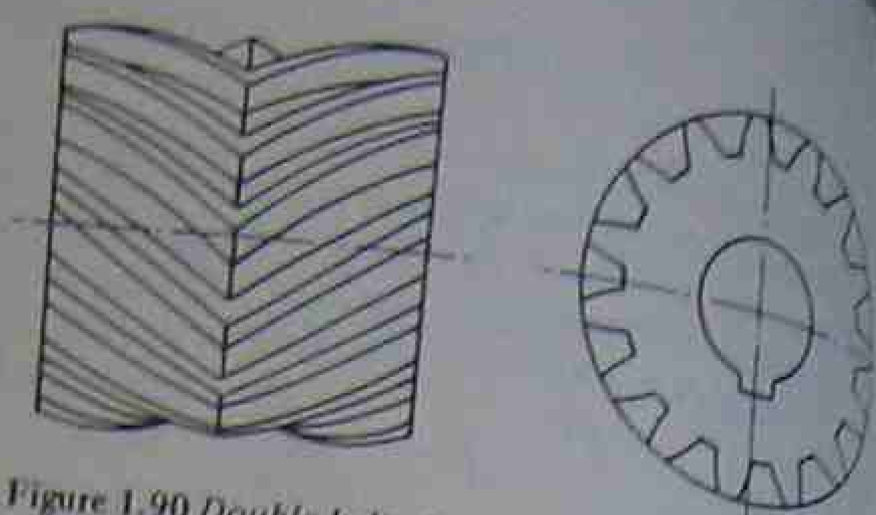


Figure 1.90 *Double helical gear*

BEVEL GEAR

On bevel gears the teeth are cut on a conical surface so that gears with intersecting but non-parallel shafts may be mated.

On *straight bevel gears* the teeth are straight, and on *spiral bevel gears* the teeth are curved in the form of a spiral. In both cases the shafts may be at any angle to one another although in most cases the angle is 90°.

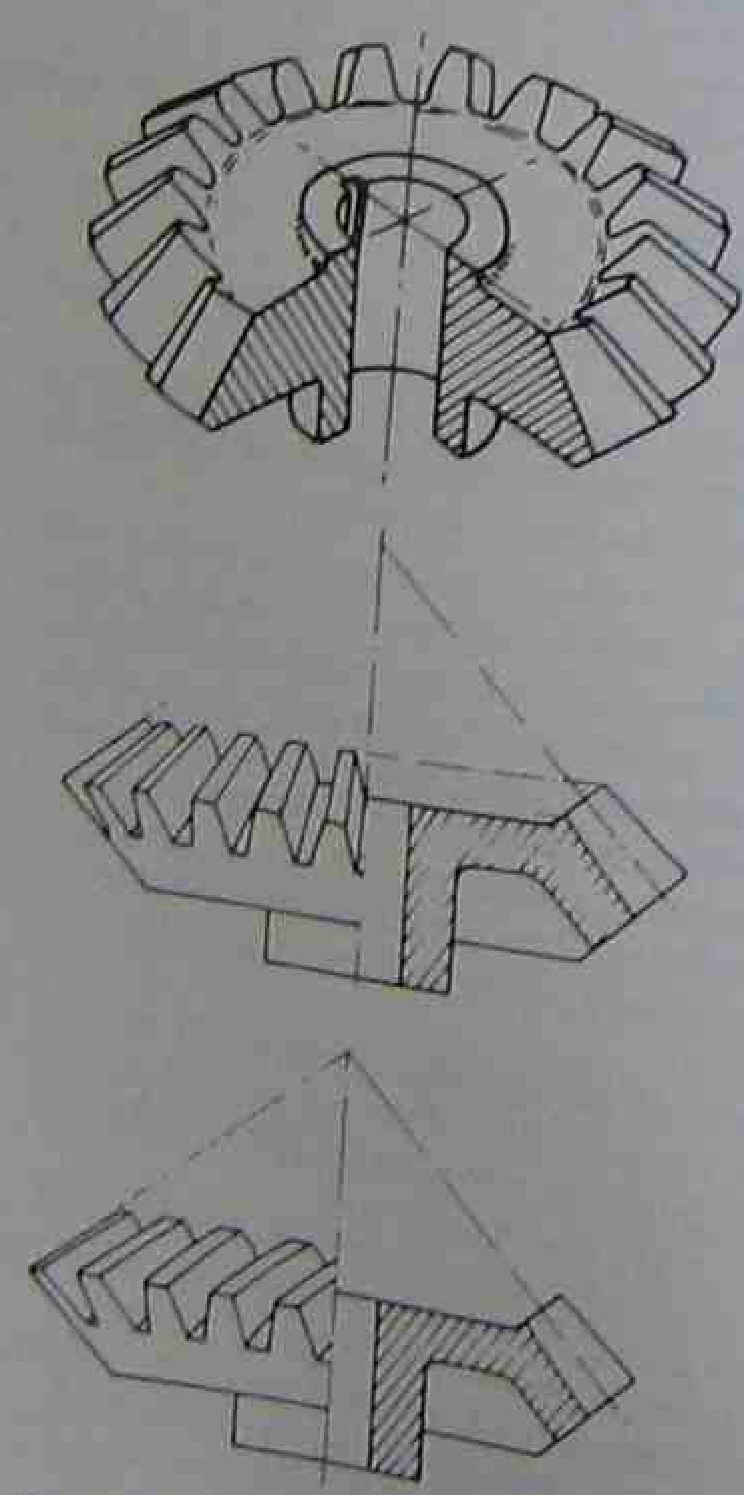


Figure 1.91 *Straight bevel gears (top and centre), and spiral bever gear*

Skew bevel gears have straight teeth but the axes are non-intersecting, and *hypoid gears* have curved teeth the axes of which are non-parallel and non-intersecting.

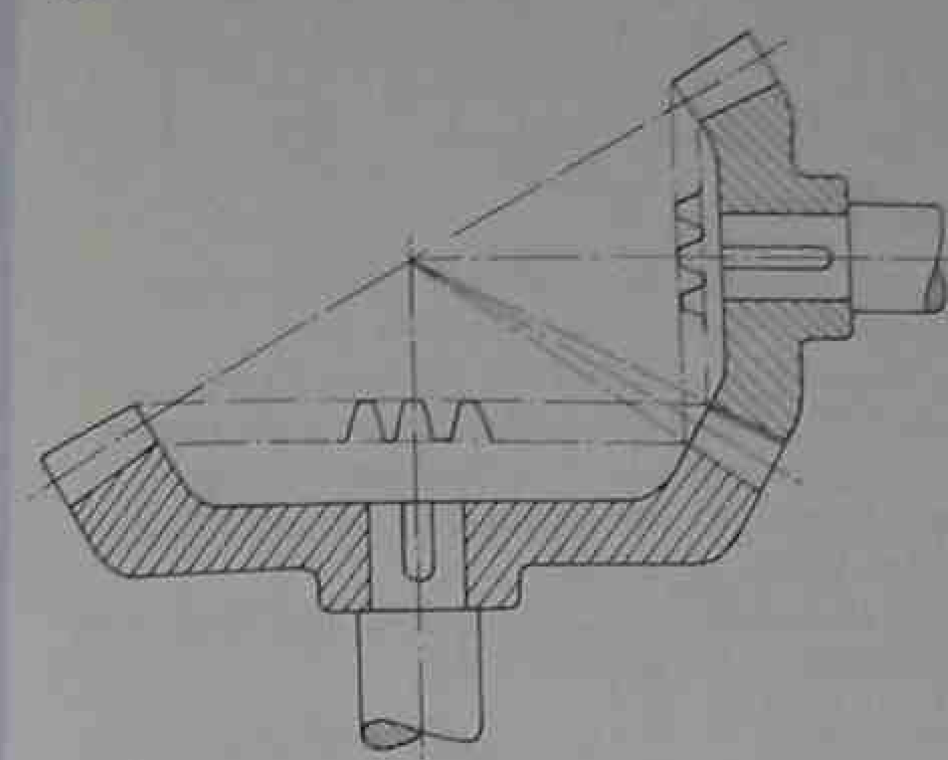


Figure 1.92 *Pair of bevel gears*

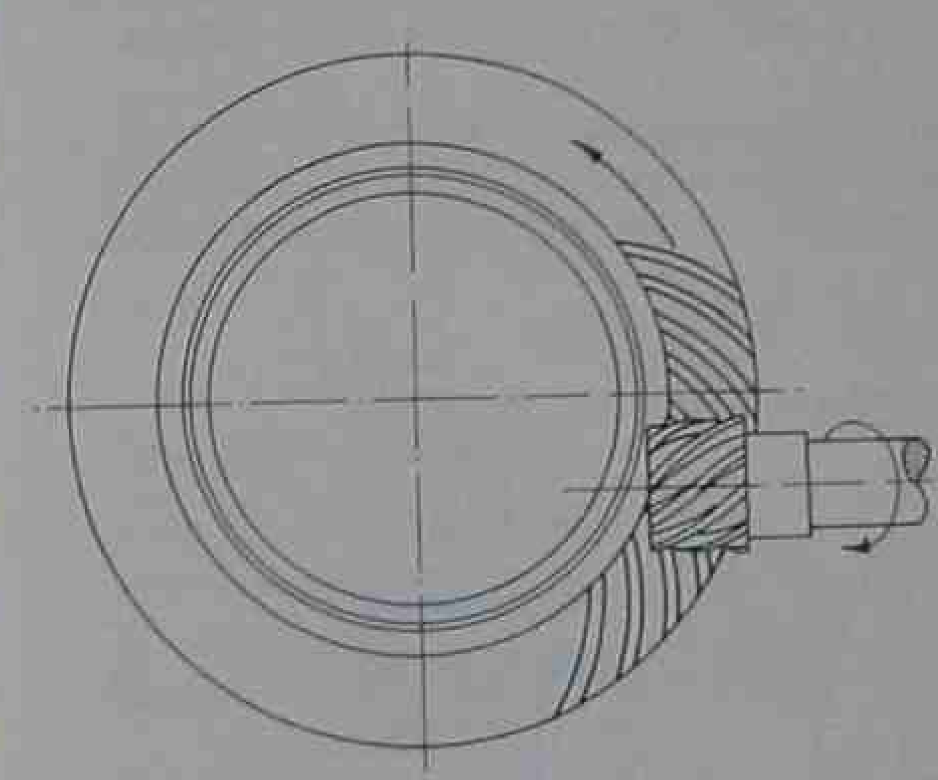


Figure 1.93 *Hypoid gears*

WORM GEAR

On worm gears the shafts are at right angles to each other but do not intersect. In effect the worm is a single or multi-start, trapezoidal-form screw thread which meshes with a much larger diameter worm wheel with teeth curved to suit the worm diameter.

Worm gears are used for speed reduction and have ratios of up to about 300:1.

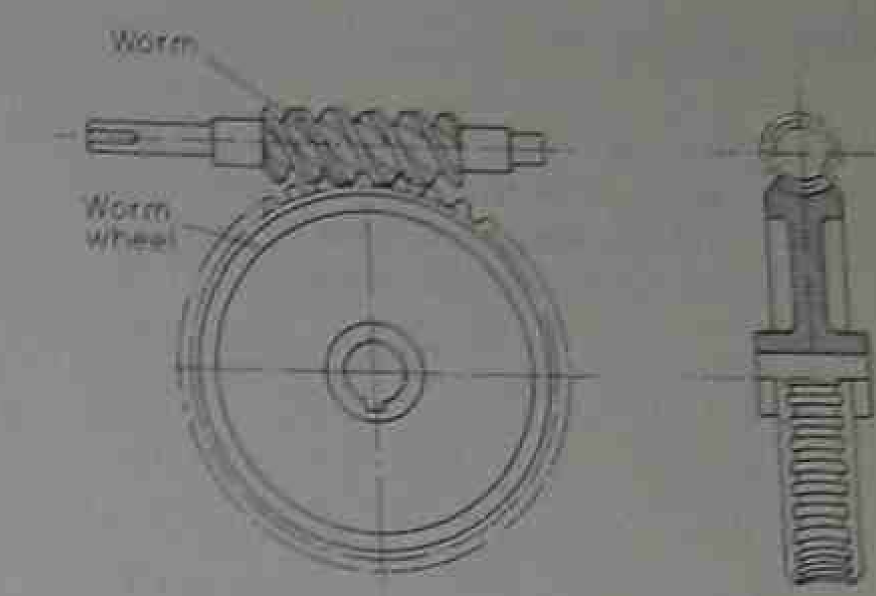


Figure 1.94 *Worm gear*

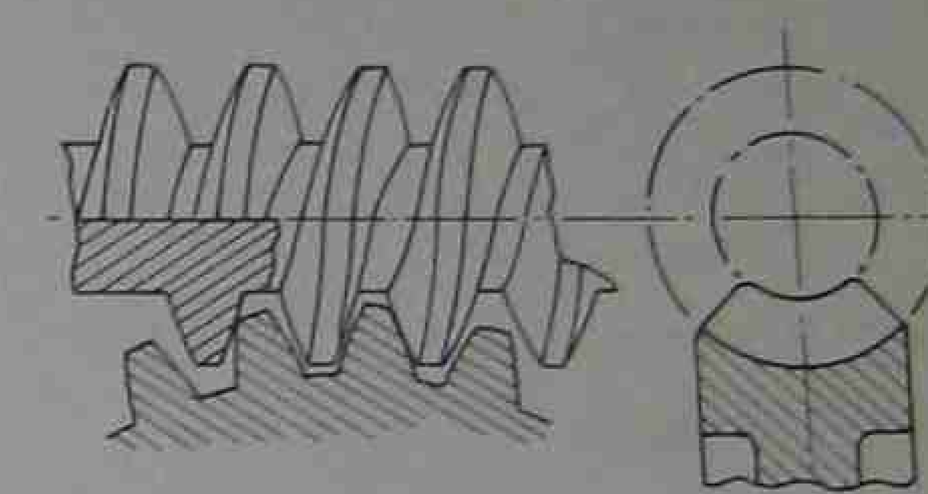


Figure 1.95 *Detail of worm gear*

RACK-AND-PINION GEAR

A rack is a straight bar with teeth which mesh with mating teeth on a gear, or *pinion*. The rack moves in a straight line as the pinion rotates. If the pinion teeth are of involute form the rack teeth will be straight-sided.

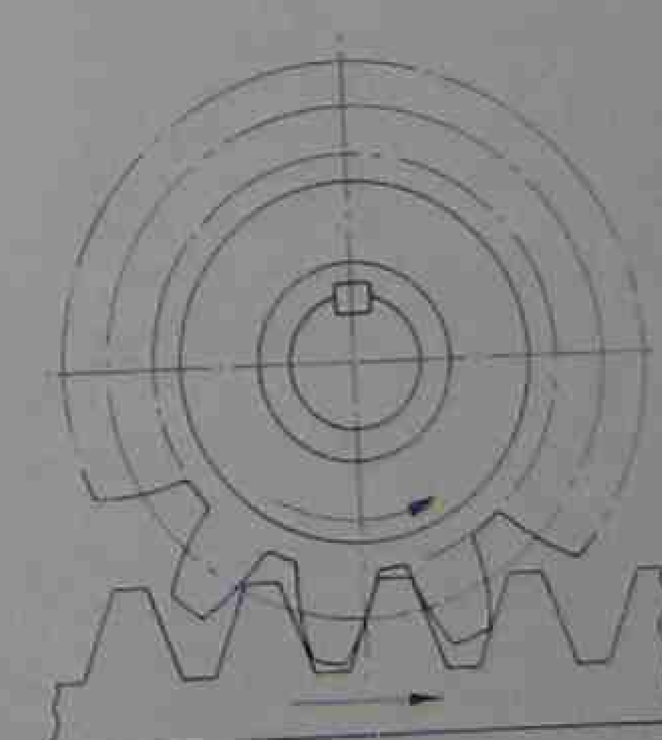


Figure 1.96 *Rack-and-pinion gear*

INTERNAL GEAR

An internal gear has teeth cut on the inside of an annular ring which mesh with teeth on a gear with external teeth which is situated inside the annular ring. Both gears rotate in the same direction.

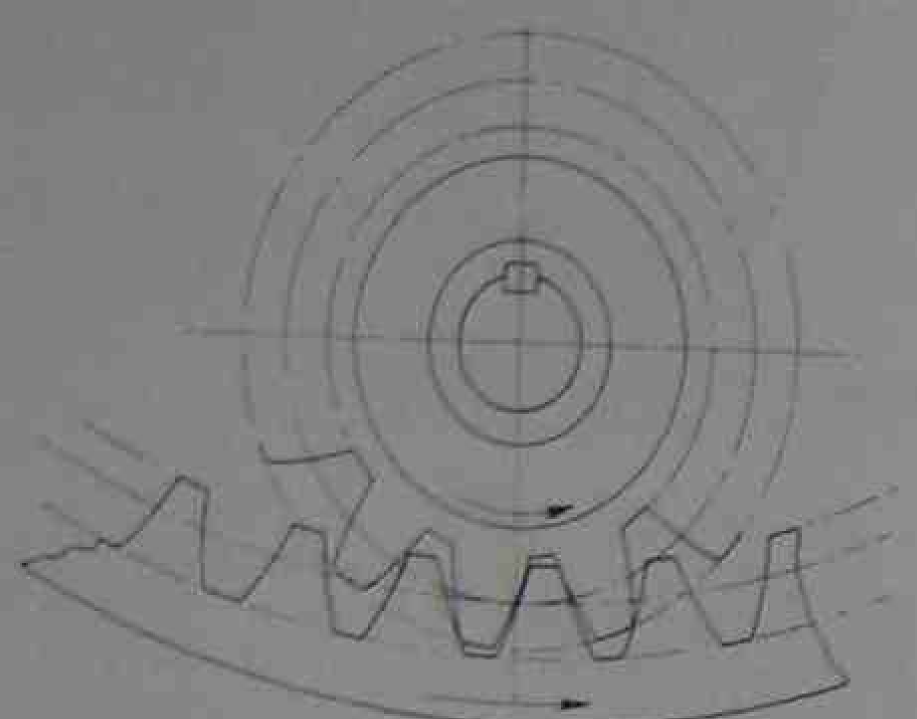


Figure 1.97 *Internal gear*

EPICYCLIC GEAR

An epicyclic gear comprises a *sun gear* keyed to a central shaft, and several *planet gears* which are meshed with and revolve around the sun gear. The planet gears are mounted on spindles held in position by a carrier attached to a sleeve running on the sun gear shaft.

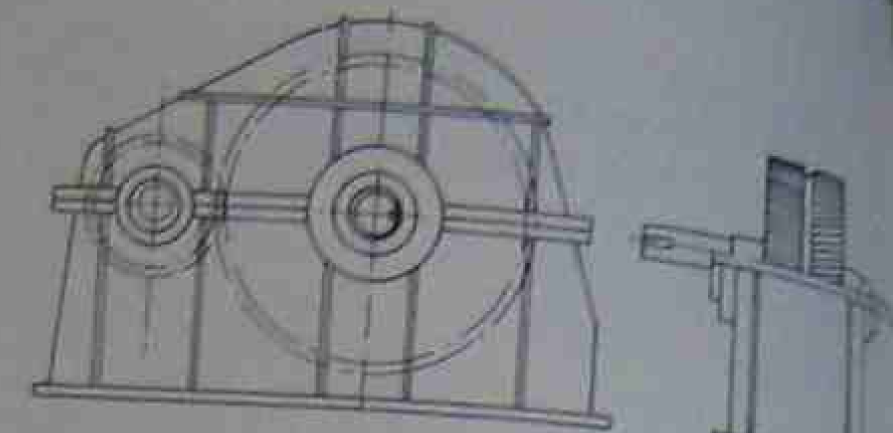


Figure 1.99 *Gearbox*

A second co-axial shaft carries a *ring gear* whose internal teeth mesh with the planet gears.

Various gear ratios can be obtained depending upon which member is held stationary, the inner gear, the outer gear or the planet gear ring. In an epicyclic gearbox the various elements are held by friction brakes as desired.

An advantage of epicyclic gears is that their input and output shafts are concentric.

GEAR TRAIN

When more than two gears are employed the system is known as a *gear train*. When these gears are mounted in a casing with a lubrication system the assembly is called a *gearbox*. An intermediate shaft is called a *layshaft*.

In an automobile gearbox some of the gears are on splined shafts and can be moved by using a hand operated gear selector to give different speed ratios.

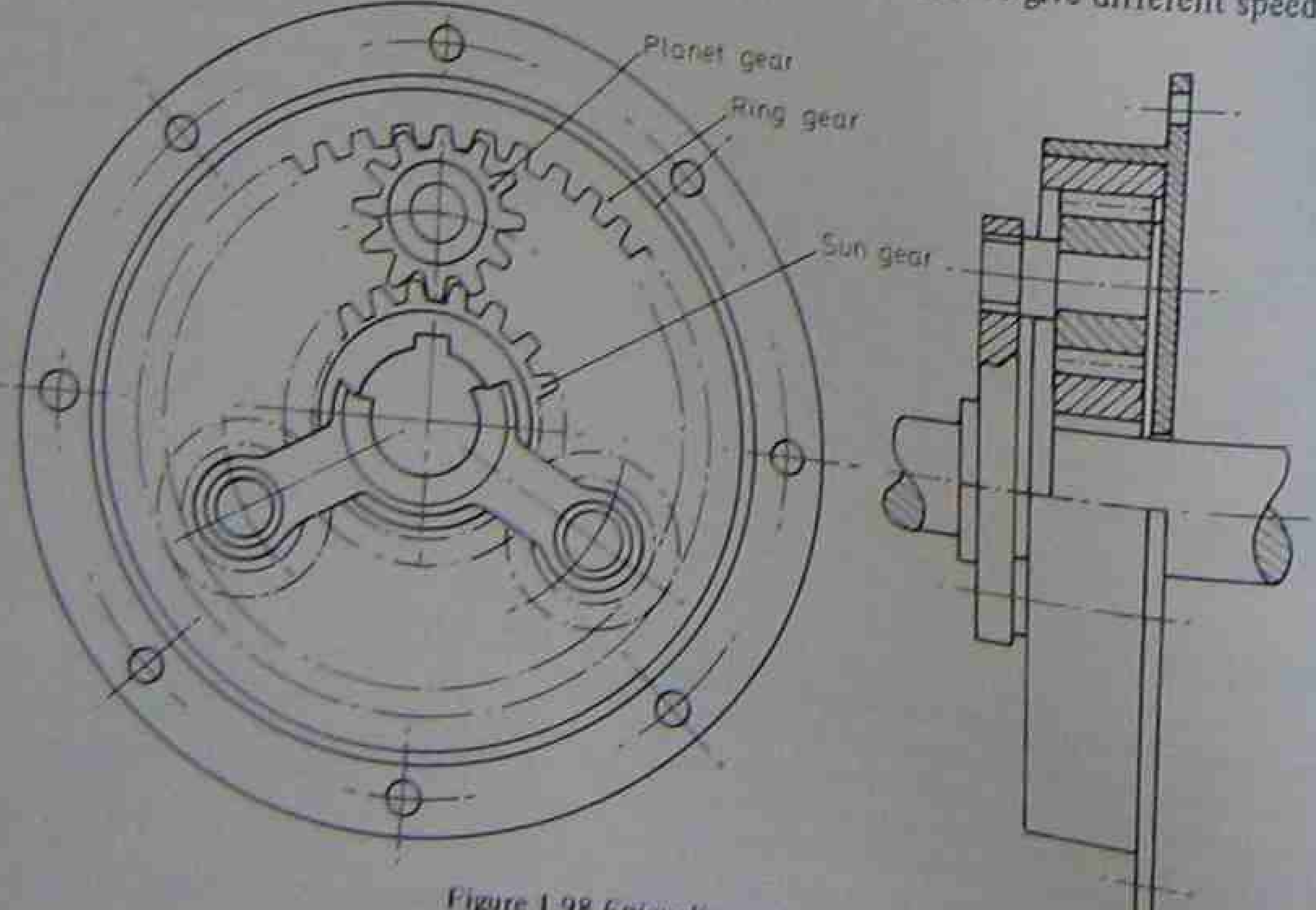


Figure 1.98 *Epicyclic gear*

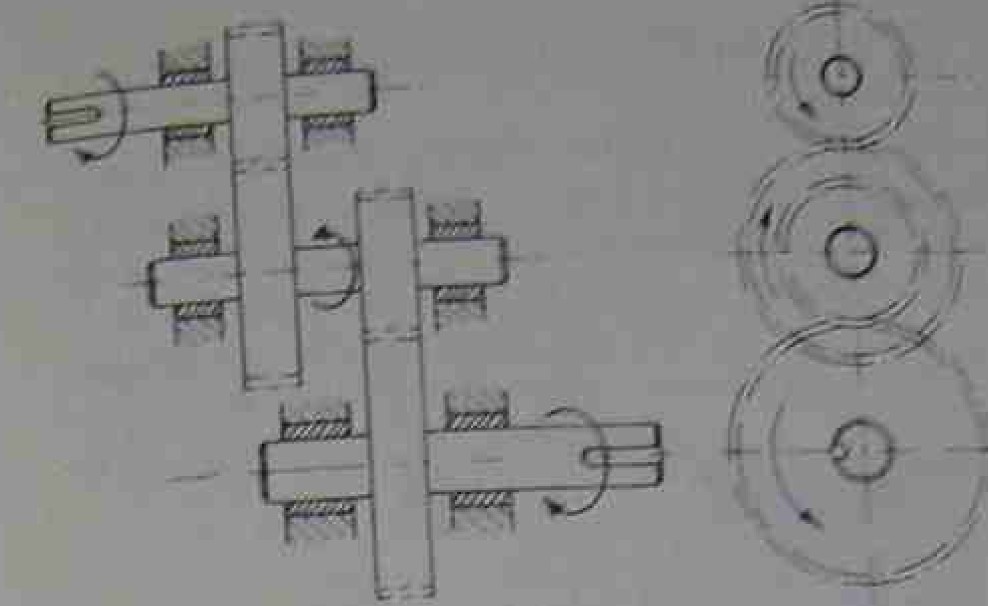


Figure 1.100 *Double reduction gear train*

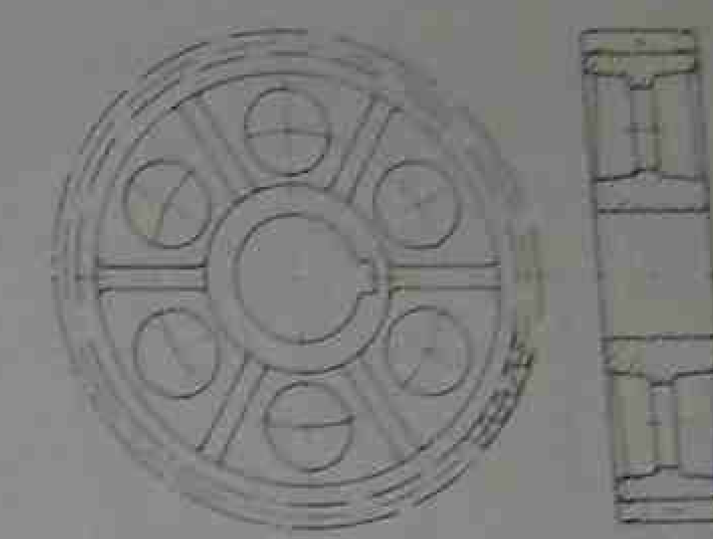


Figure 1.102 *Cast steel gear*

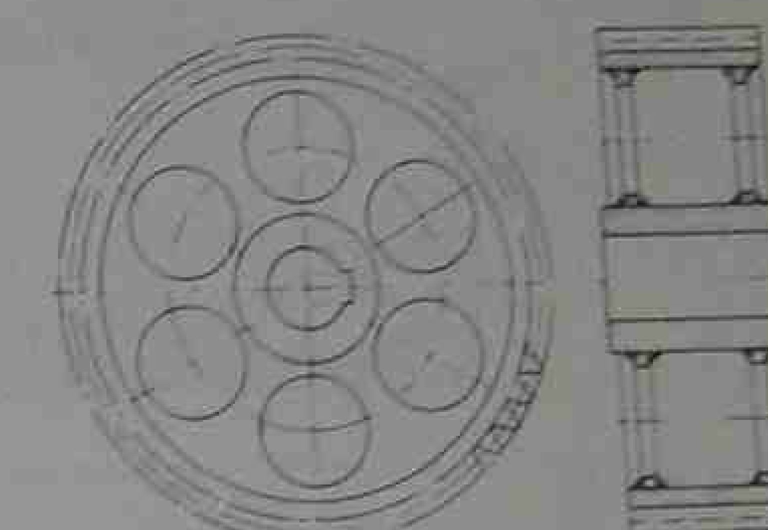


Figure 1.103 *Welded steel gear*

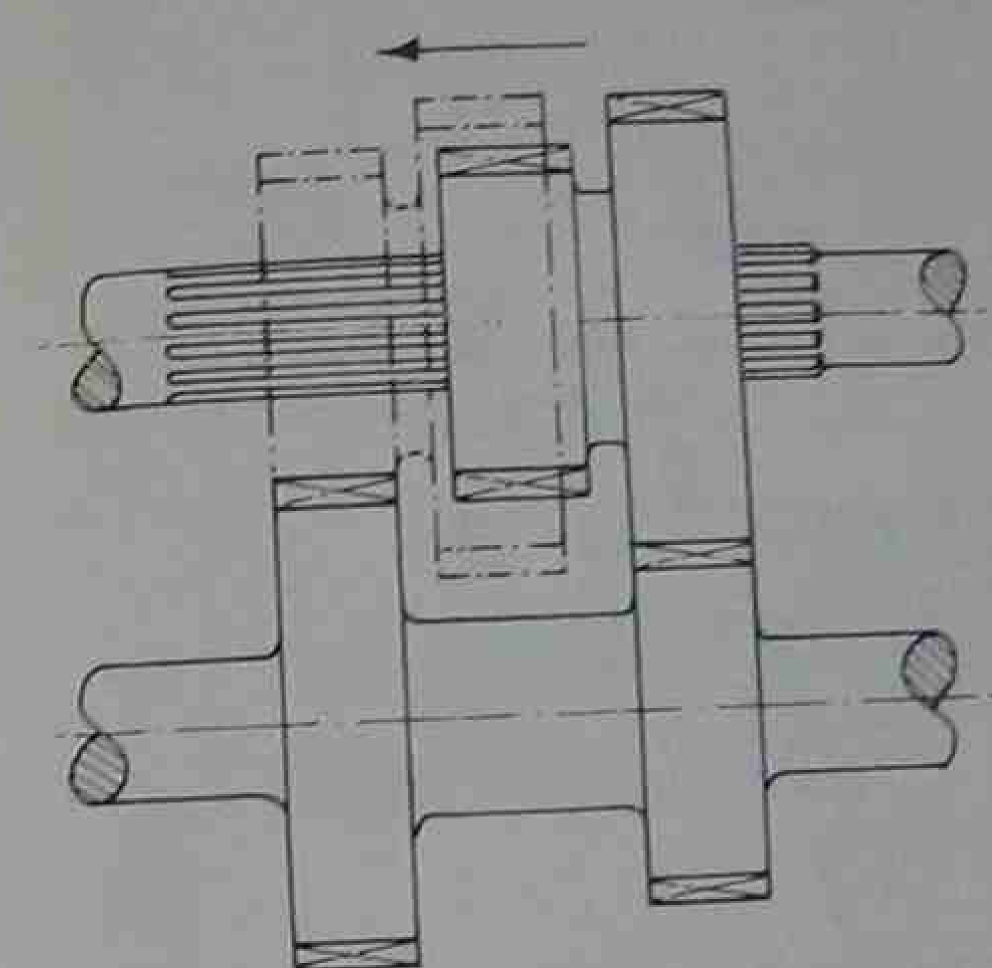


Figure 1.101 *Gear changing*

Figure 1.101a *Splined gear shaft*

GEAR MATERIALS

Gears are made from cast iron for light duty, and cast and alloy steels for heavy duty. Bronze is used for worm gears. Plastics, particularly laminates such as Tufnol, are used in special cases.

Small gears are usually cast or forged, and large gears, e.g. marine gears, are welded fabrications. Steel gears often have the teeth case-hardened and ground to a high surface finish.

1.5 FLUID SEALS, JOINTS AND GASKETS

FLUID SEALS

Fluid seals are devices for preventing unwanted leakage of liquids and gases in machines. The seal may be between two fixed parts or two parts with relative motion. Seals between fixed parts are generally known as *joints, gaskets* and *sealing washers*. Seals preventing the leakage of lubricants on rotating and sliding shafts and other moving parts are called *oil seals*. The housing for a shaft seal is sometimes referred to as a *gland*.

Various types of jointing cement are used with or without gaskets, e.g. Locktite.

STUFFING BOX (GLAND)

A stuffing box, or gland, is a recess in a casing (surrounding a shaft) containing sealing material, or *gland packing*, which is compressed by means of an adjustable ring to prevent leakage along the shaft. This ring is often termed a *gland-ring* or simply *a gland* (Figure 1.104).

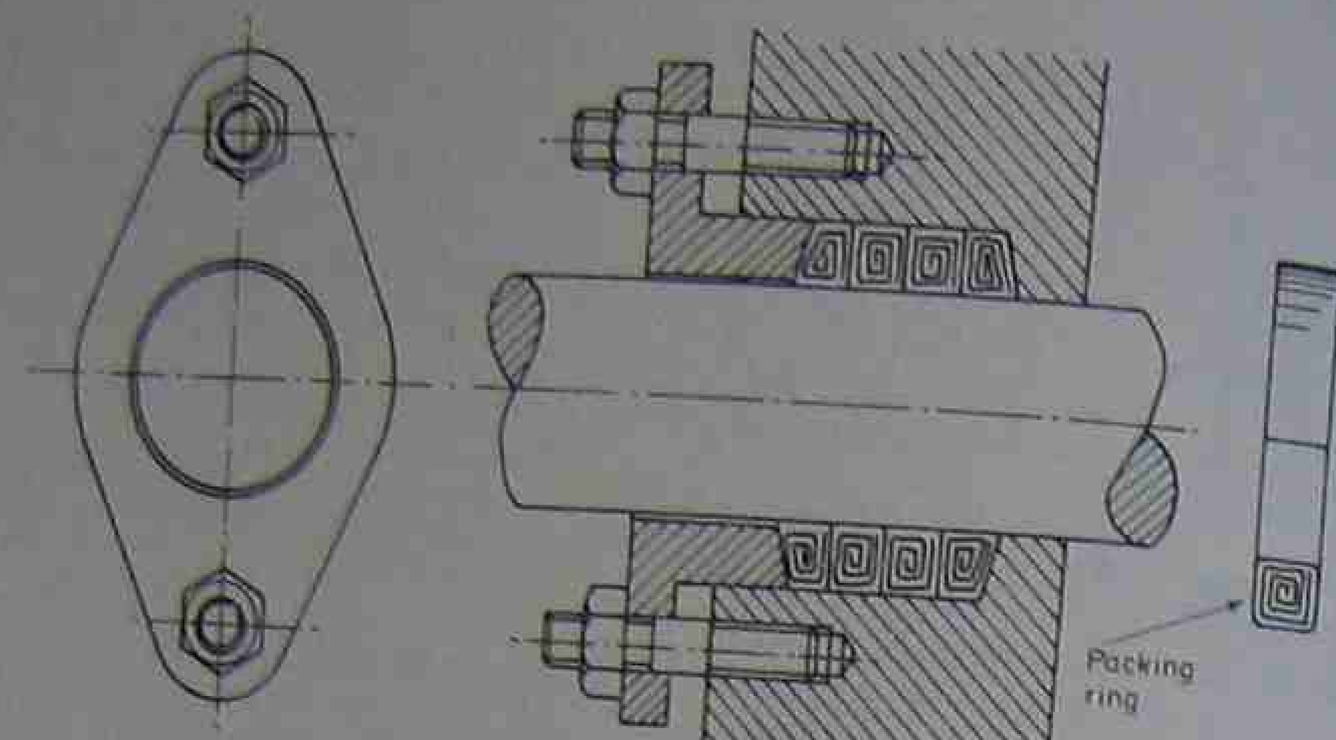


Figure 1.104 *Stuffing box (gland)*

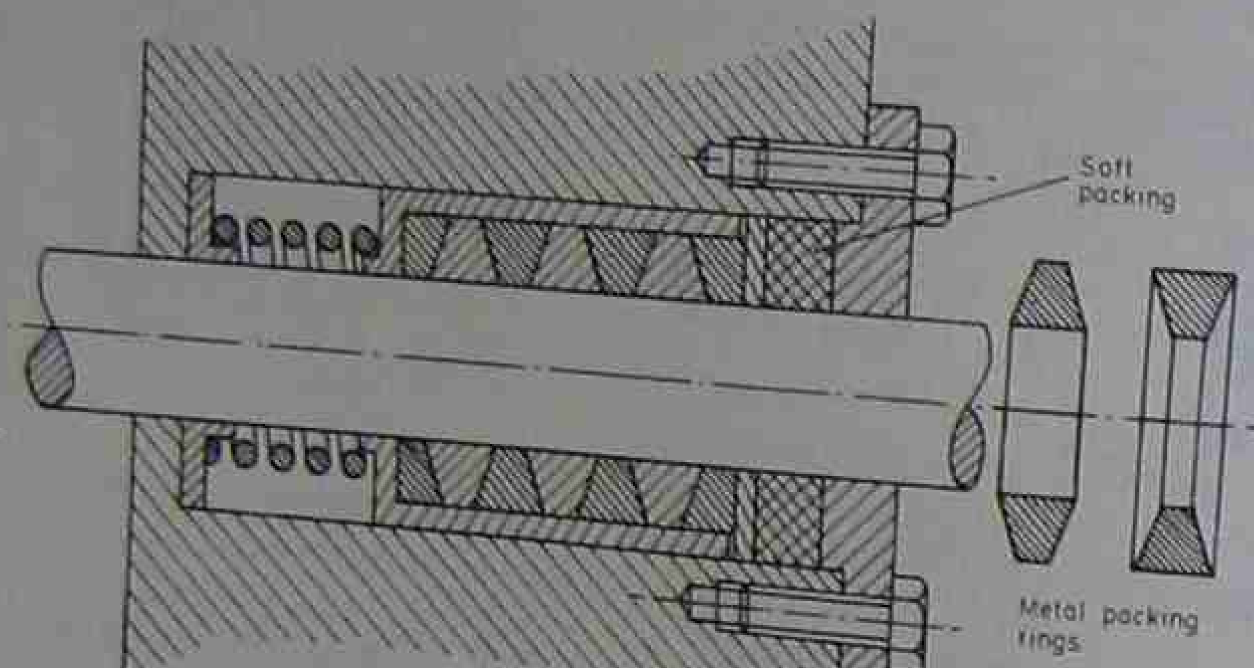


Figure 1.105 *Metallic packing*

GLAND PACKING

The packing in a stuffing box consists of a spiral, or several rings, of woven asbestos, cotton or hemp impregnated with grease and graphite to reduce friction.

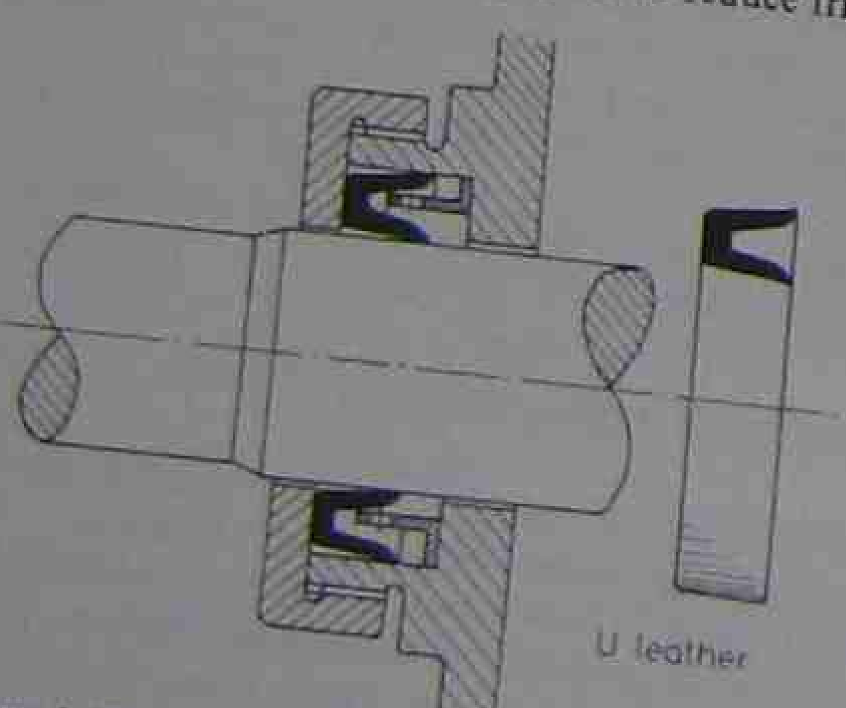


Figure 1.106 *Shaft seal with U leather*

METALLIC PACKING

This consists of alternative triangular-section rings of gunmetal and whitemetal, or sometimes plastic and metal. The rings are compressed, together with a ring of soft packing, either by tightening with bolts or by means of a spring.

U SEAL

A moulded U-section seal of leather or synthetic rubber is held in position by a screwed or bolted ring. The fluid pressure forces the lip of the seal on to the shaft or cylinder, thus improving the sealing action.

This type of seal is used where there is a large pressure difference across it.

LIP SEAL WITH GARTER SPRING

This is a moulded synthetic rubber seal, usually with a metal insert, in which the lip of the seal is held onto the shaft by a circumferential *garter spring*.

It is suitable for low pressures and is used in IC engines, automobile gearboxes, etc.

A double, or duplex, seal is used to prevent the entry of dirt from the atmosphere.

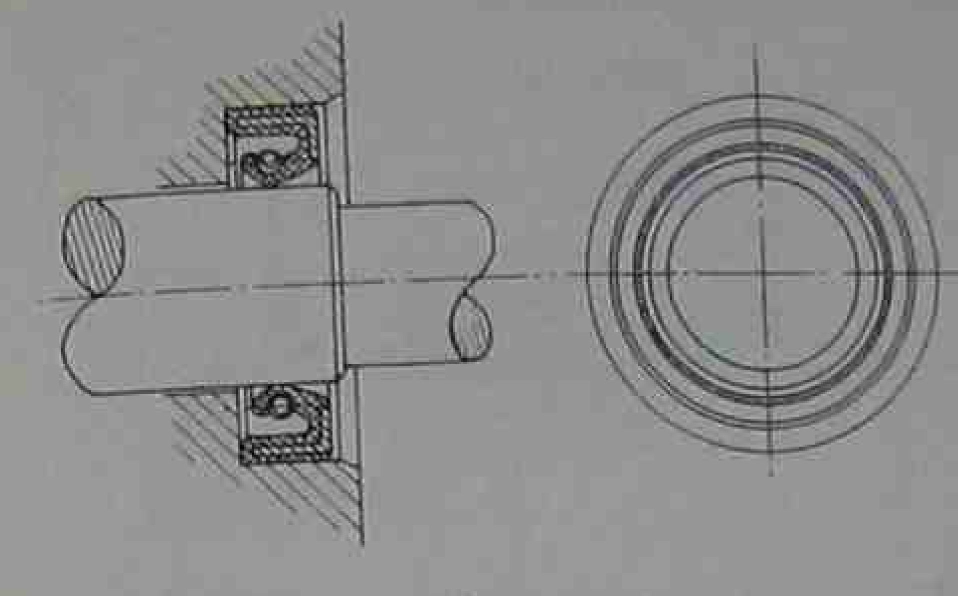


Figure 1.107 *Lip seal with garter spring*

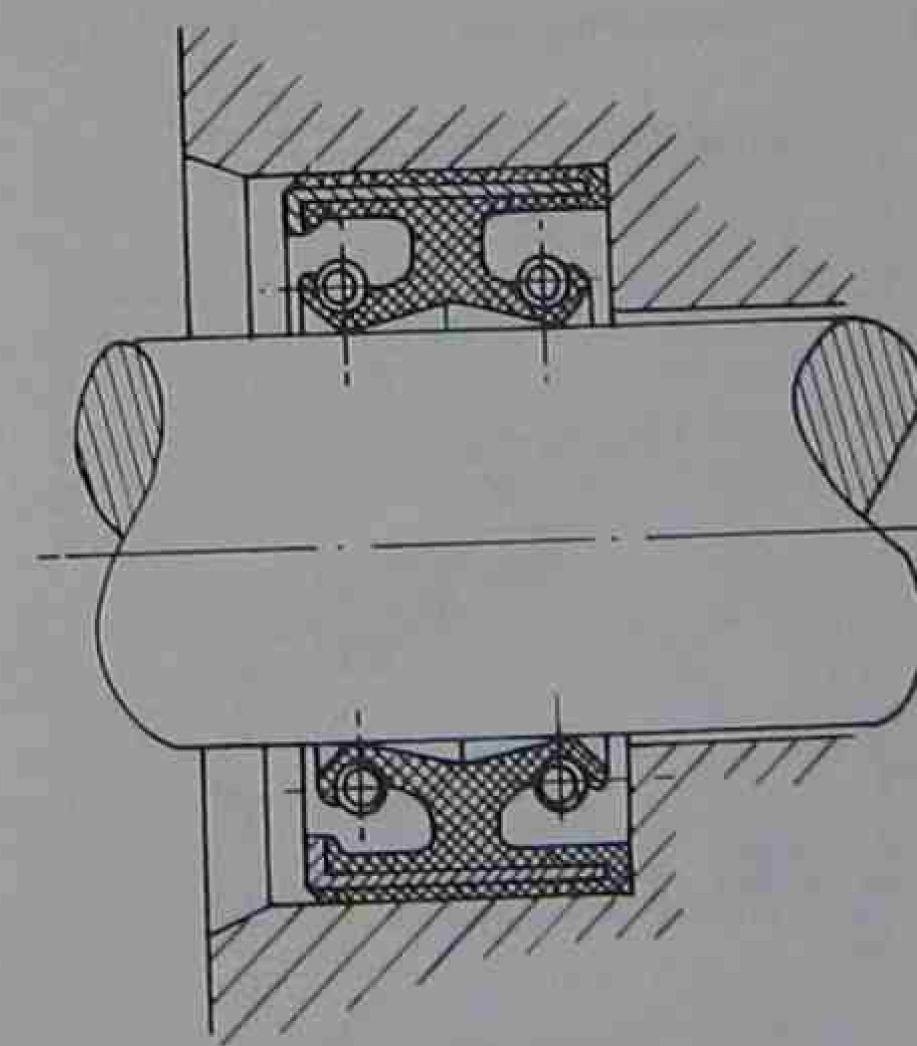


Figure 1.108 *Duplex lip seal*

CUP SEAL

A synthetic rubber or leather seal with a single lip, used for sealing hydraulic and pneumatic pistons. Sealing is aided by compression from the holding flange.

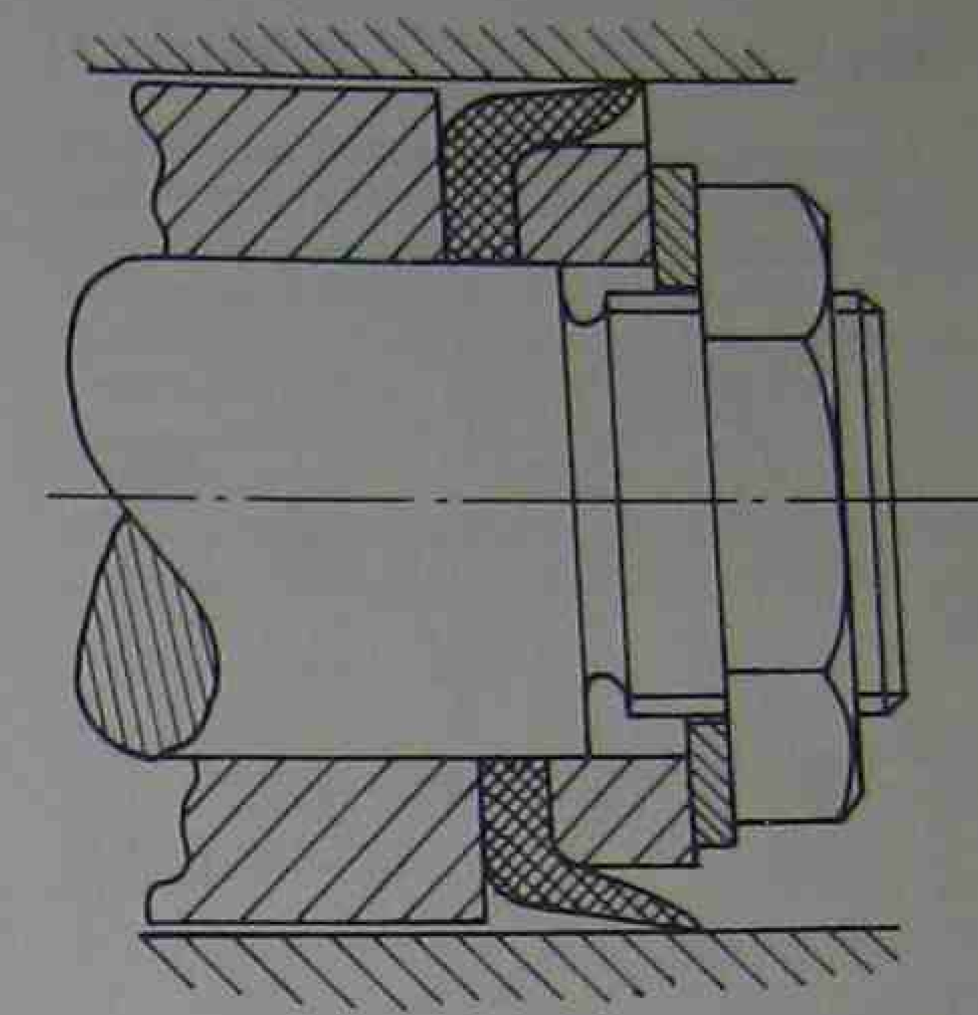


Figure 1.109 *Cup seal*

HAT PACKING

A synthetic rubber packing in the form of a flat washer with a raised sealing lip on the inside edge, used for sealing rods. It relies on controlled compression by a nut to give effective sealing.

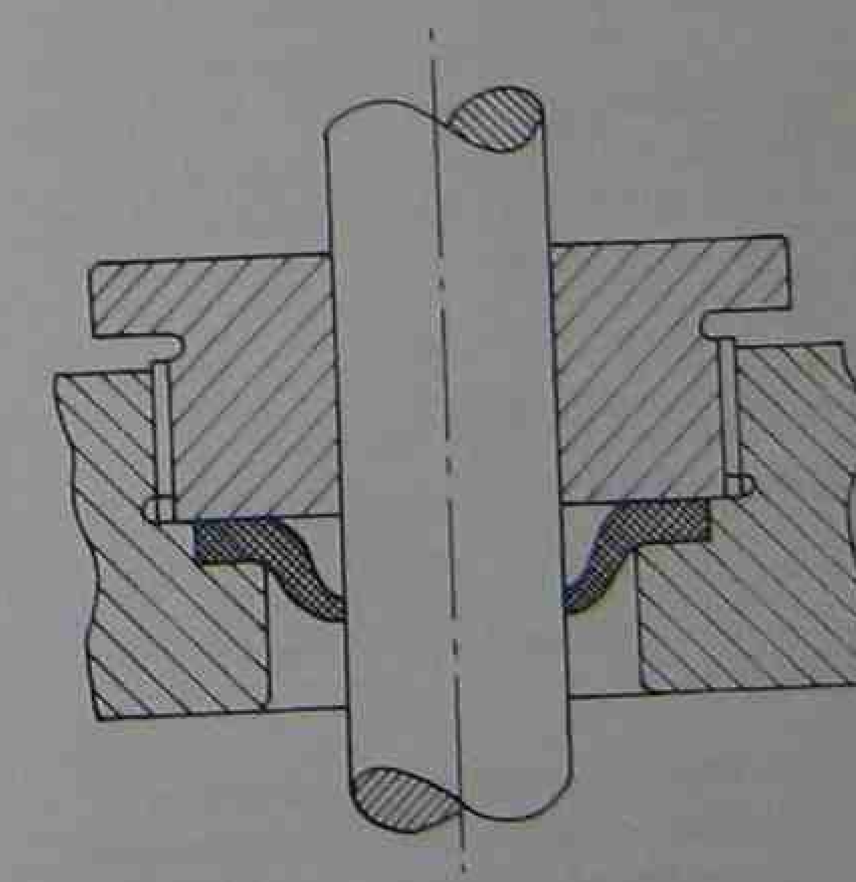


Figure 1.110 *Hat packing*

FELT RING

A felt ring provides a simple and cheap form of seal to retain lubricant and exclude dirt. It is suitable for grease-lubricated, low-speed rolling bearings.

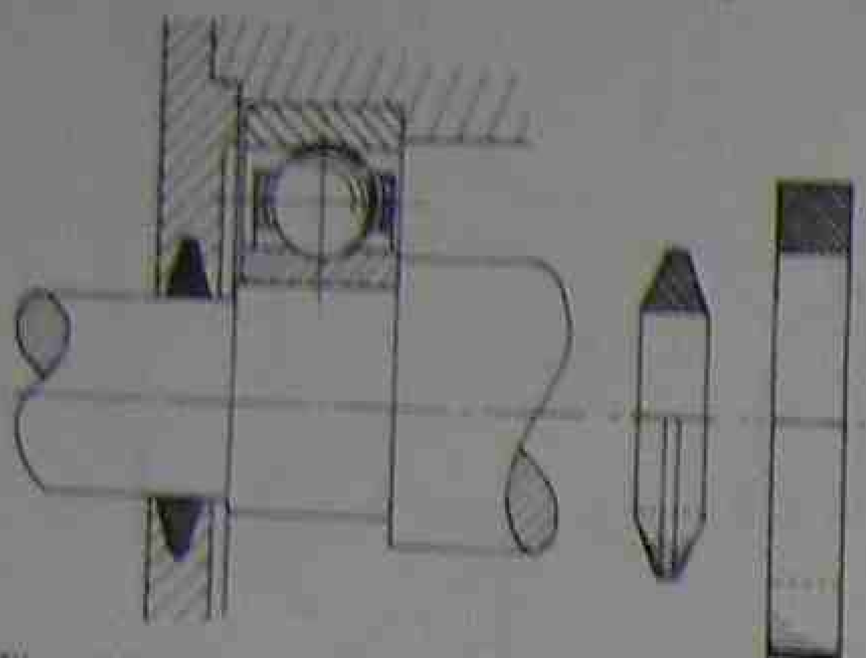


Figure 1.111 *Felt sealing rings*

O RING

An O ring is the simplest form of hydraulic packing. It is a synthetic rubber ring of solid circular cross-section made in a wide variety of ring and cross-section diameters.

O rings are suitable for static sealing, e.g. in pipe joints, cylinder end covers and valve spindles, and also for low speed dynamic sealing of pistons and piston rods in hydraulic and pneumatic cylinders.

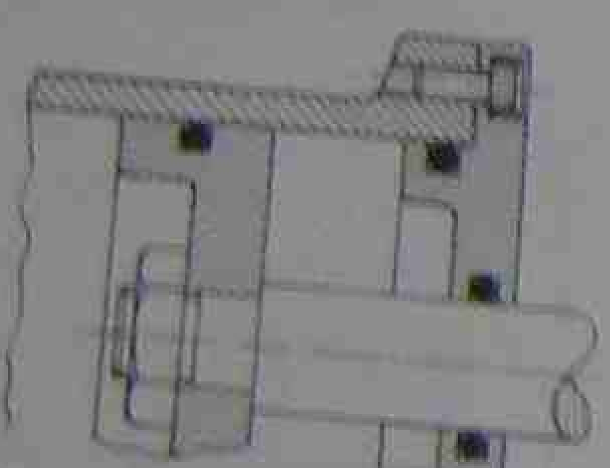


Figure 1.112 *Hydraulic cylinder with O ring seals*

CARBON SEALS

In these seals a block or blocks of carbon held in a housing run with zero clearance on the moving surface.

In one type, segmental carbon rings are held on to a rotating shaft by garter springs, and these are used in small steam turbines.

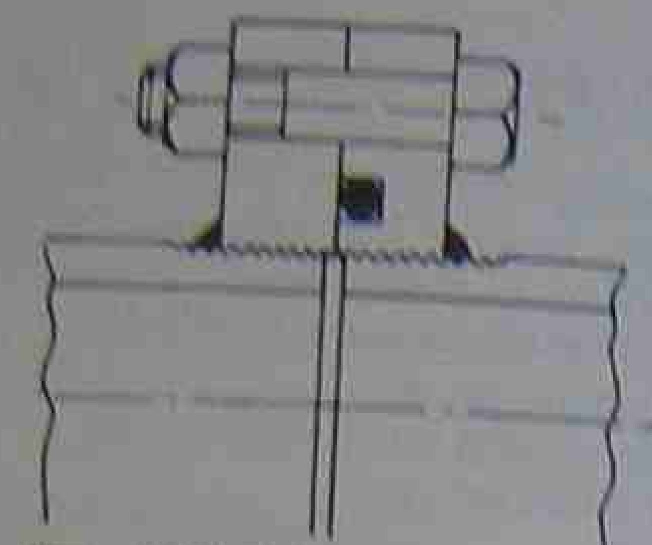


Figure 1.113 *O ring in pipe joint*

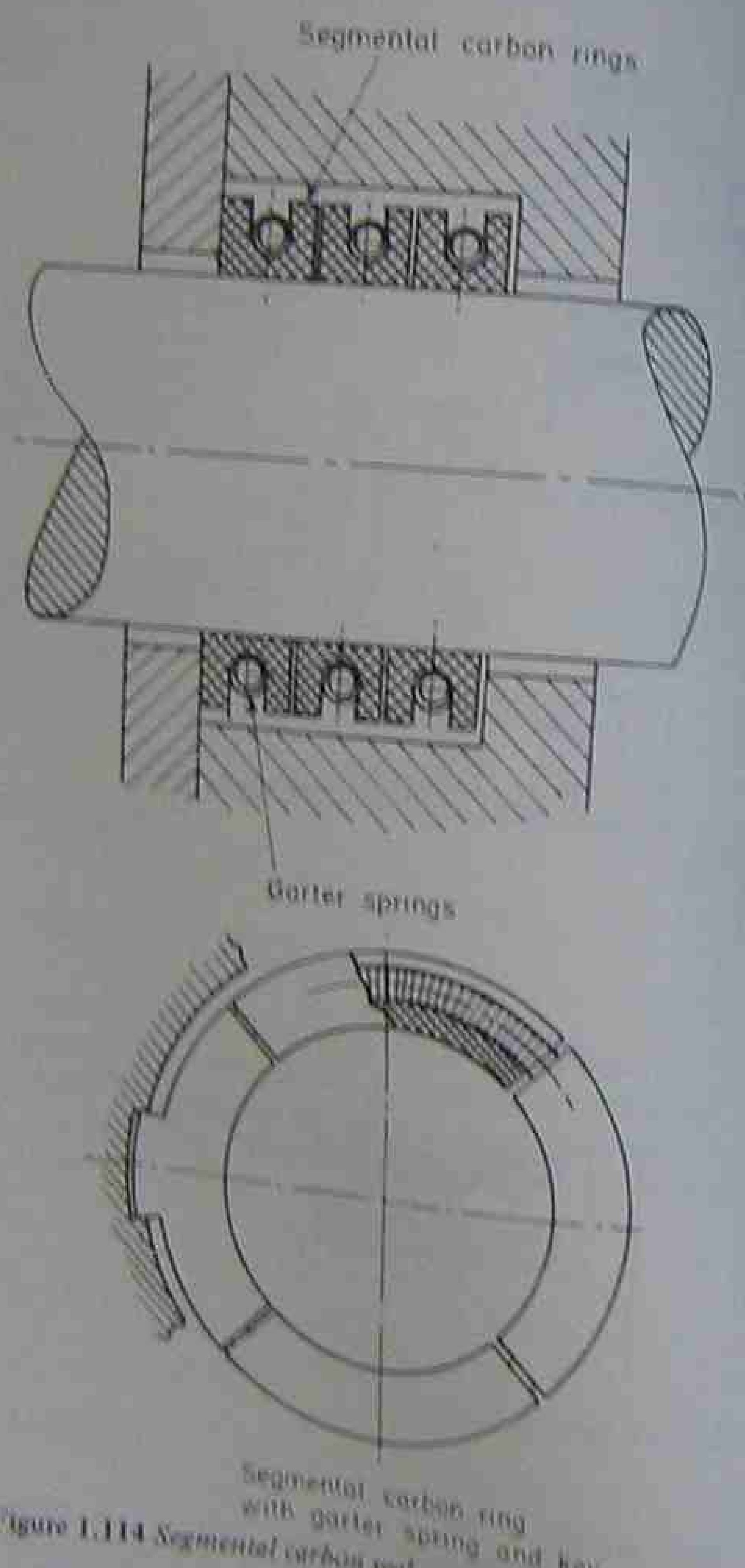


Figure 1.114 *Segmental carbon seal*

Another type has an annular ring of carbon held on to a thrust collar on the shaft by a spring, and this type is commonly used in automobile engine water pumps.

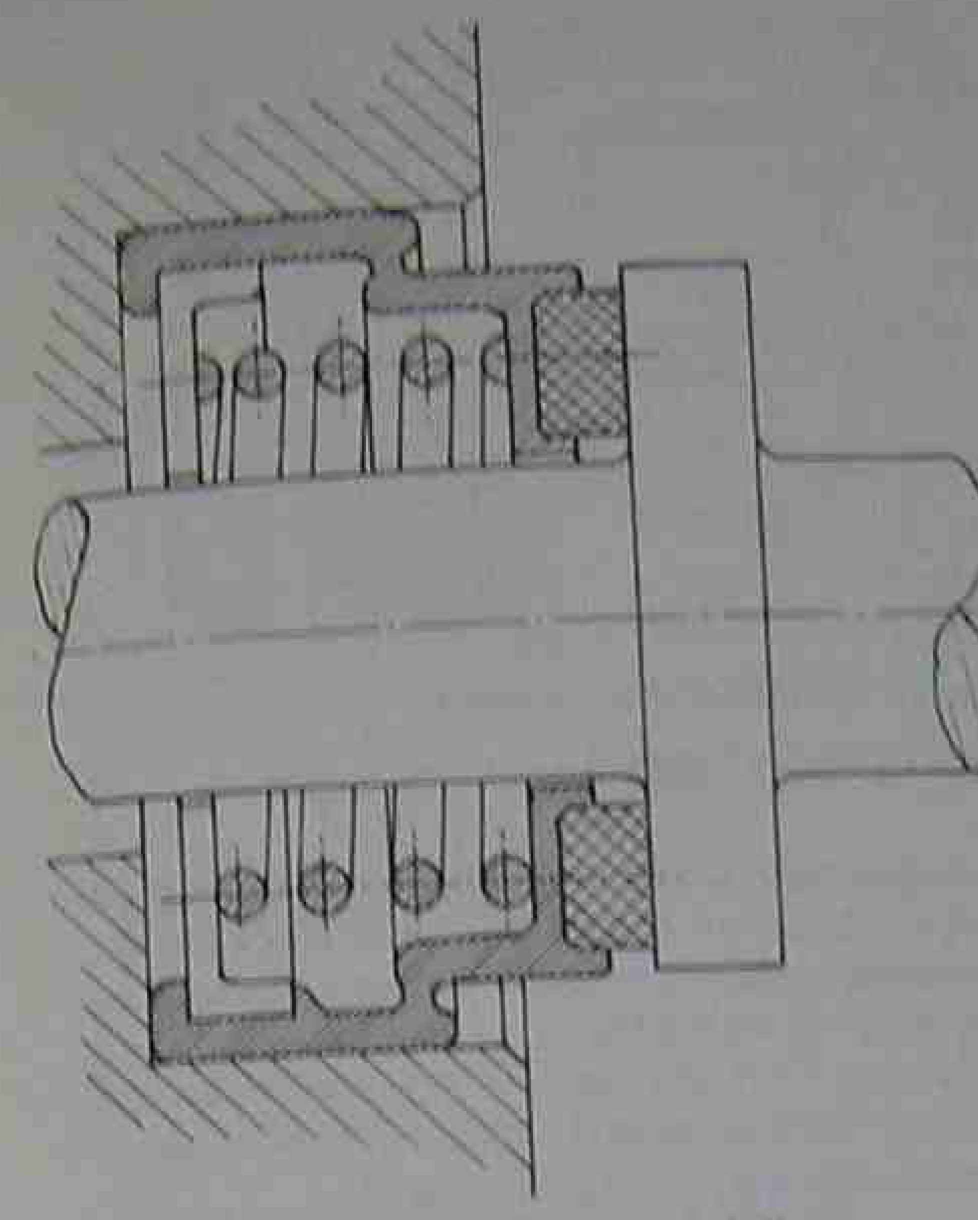


Figure 1.115 *Carbon face seal (spring-loaded)*

BELLOWS SEAL

This seal consists of an annular ring of low friction material running on the face of a thrust collar on a shaft and mounted on a bellows which flexes, thus allowing for any misalignment. The bellows also acts as a spring to give the required pressure between the surfaces.

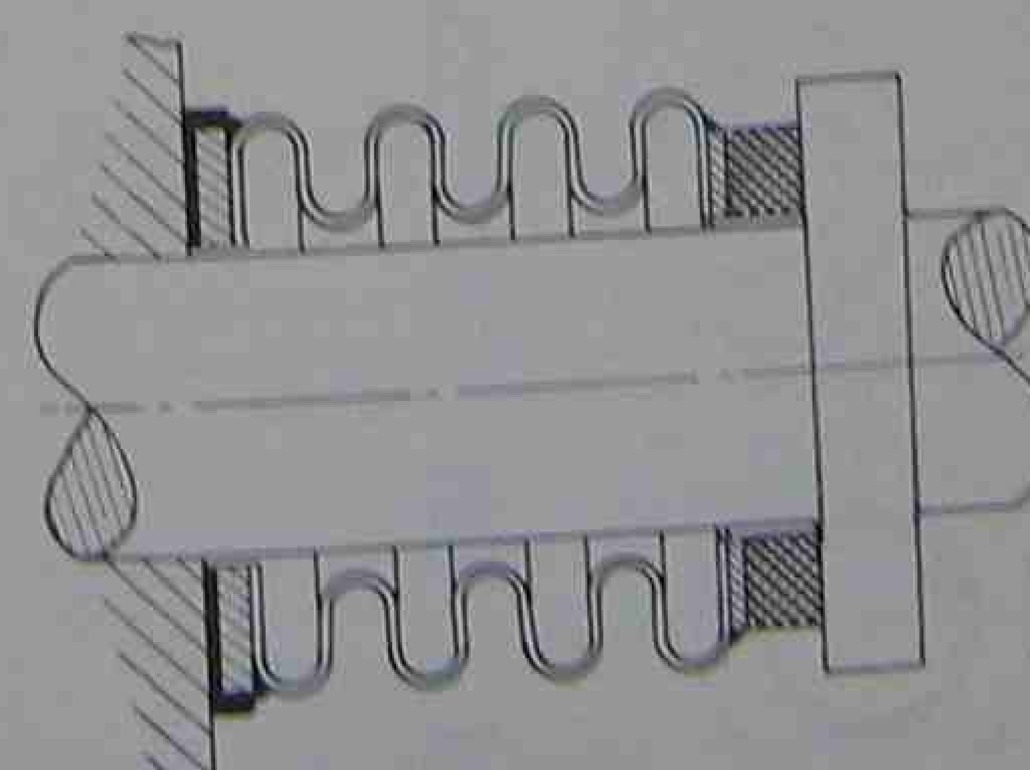


Figure 1.116 *Carbon face seal (with bellows)*

PISTON RING

The pistons of IC engines, reciprocating pumps and compressors are sealed against gas or air pressure by rings on the piston. The rings, which are made of cast iron or steel, are of rectangular cross-section and fit into grooves of the same shape in the piston. The ring is cut across so that it springs out against the cylinder wall to ensure a good seal.

A typical petrol engine piston has two compression rings for gas sealing, and an oil scraper ring which prevents oil from entering the combustion chamber.

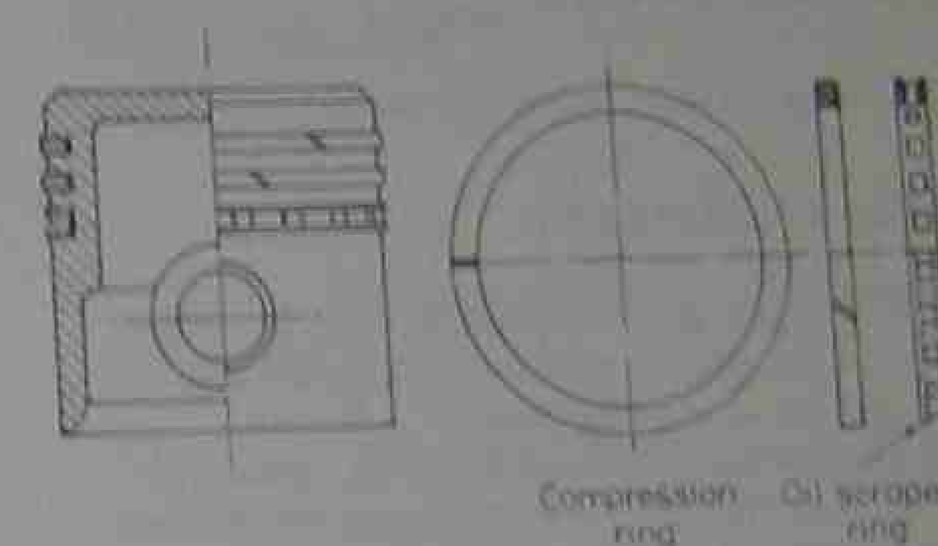


Figure 1.117 *IC engine piston rings*

LABYRINTH GLAND

A labyrinth gland is a type of seal in which there is no contact between seal and moving part. This is important where there may be side movement of a shaft relative to the fixed part. It is inevitable, however, that some leakage must occur.

Labyrinth glands in steam and gas turbines consist of a series of restrictions formed by projections on the shaft and/or casing. The pressure of the steam or gas is broken down by expansion at each restriction.

A type used for rolling bearings has a series of grooves in a fixed sleeve surrounding the shaft.

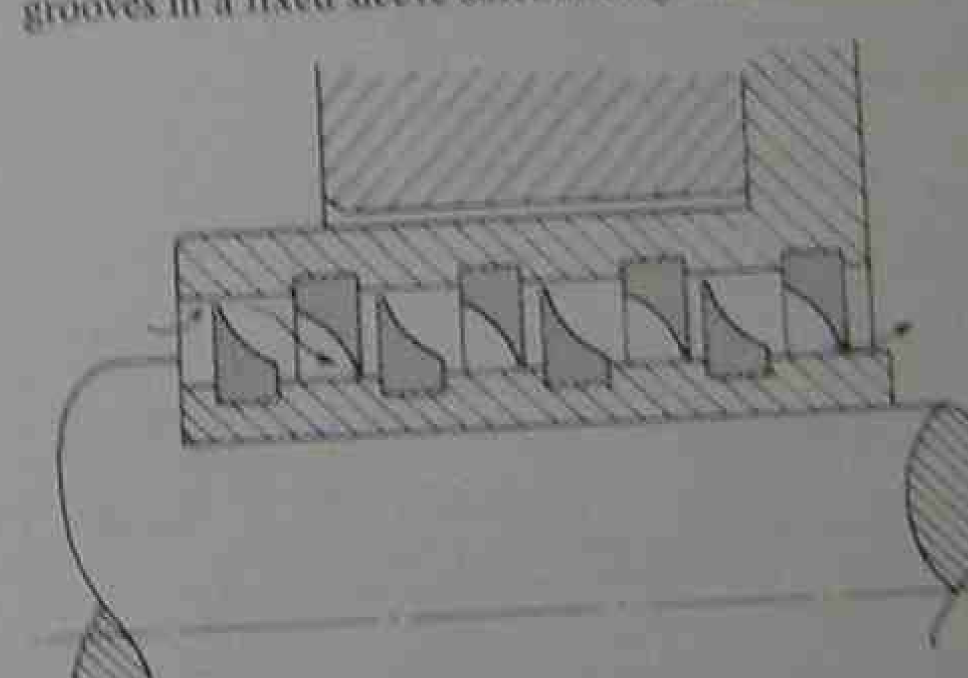


Figure 1.118 *Labyrinth gland for steam turbine*

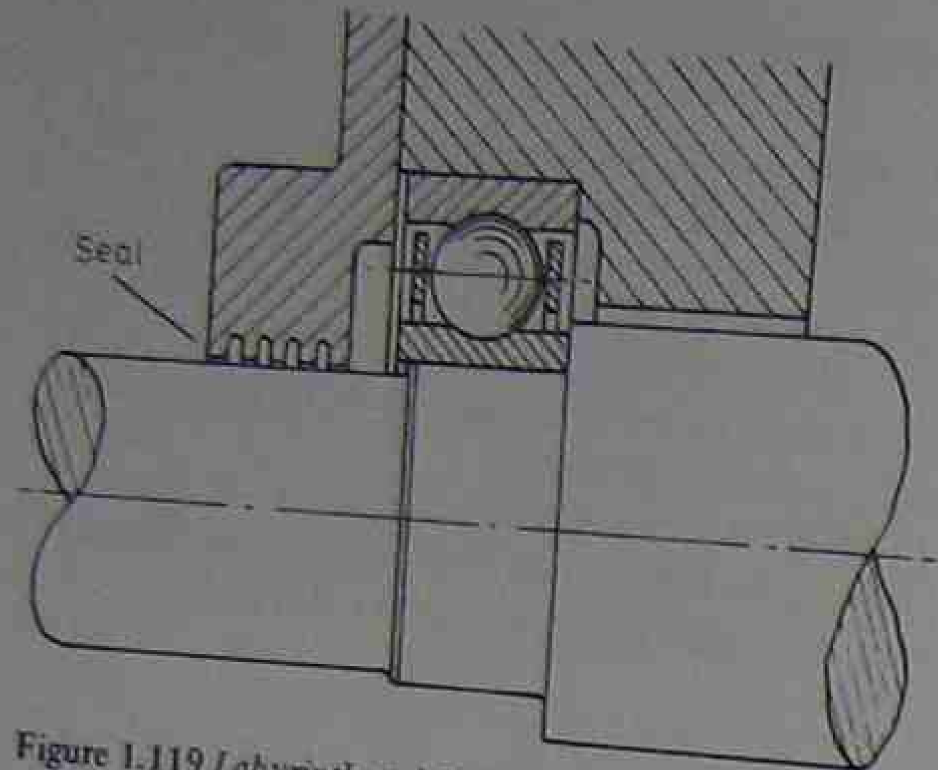


Figure 1.119 *Labyrinth seal for ball bearing*

JOINT AND GASKET

These are static seals consisting of specially-shaped pieces of flat jointing material compressed between mating faces on machine parts.

Soft materials such as cork, rubber, paper and asbestos are used for low pressure, low temperature applications and will accommodate a high degree of roughness of the faces.

For high pressures and temperatures metallic gaskets are used. These are made of soft metals, such as copper or aluminium, and are often used in the form of a sandwich with a soft non-metallic material.

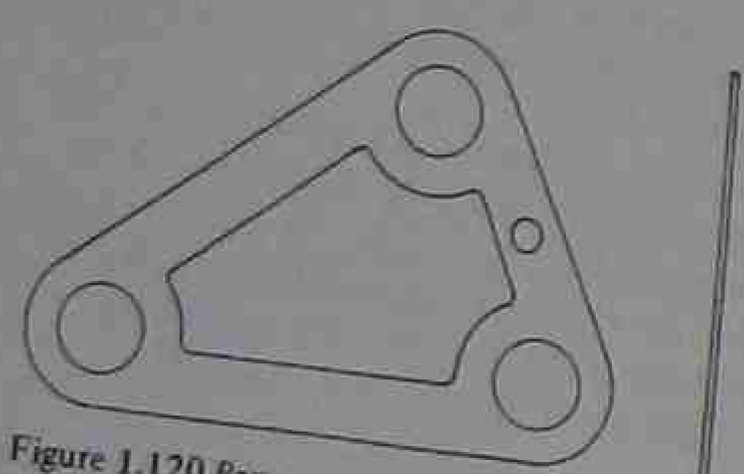


Figure 1.120 *Paper gasket*

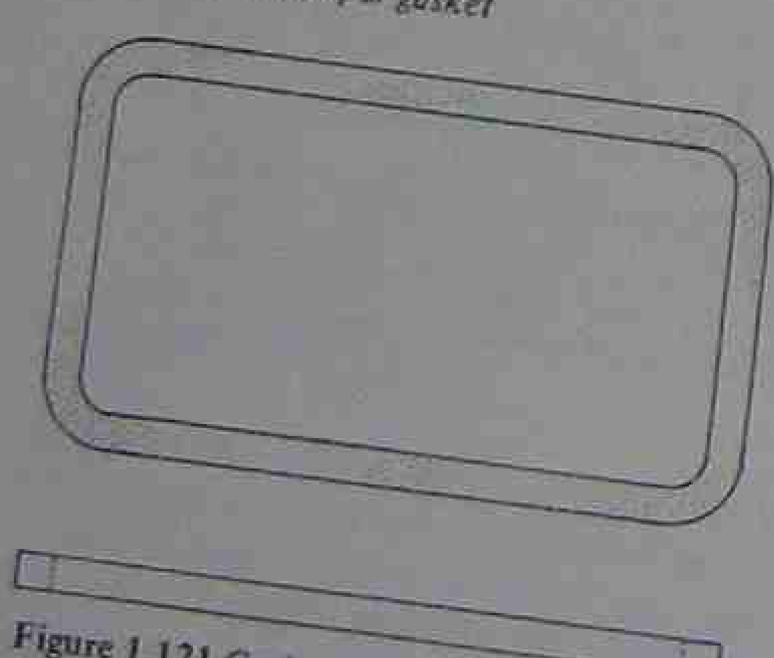


Figure 1.121 *Cork gasket*

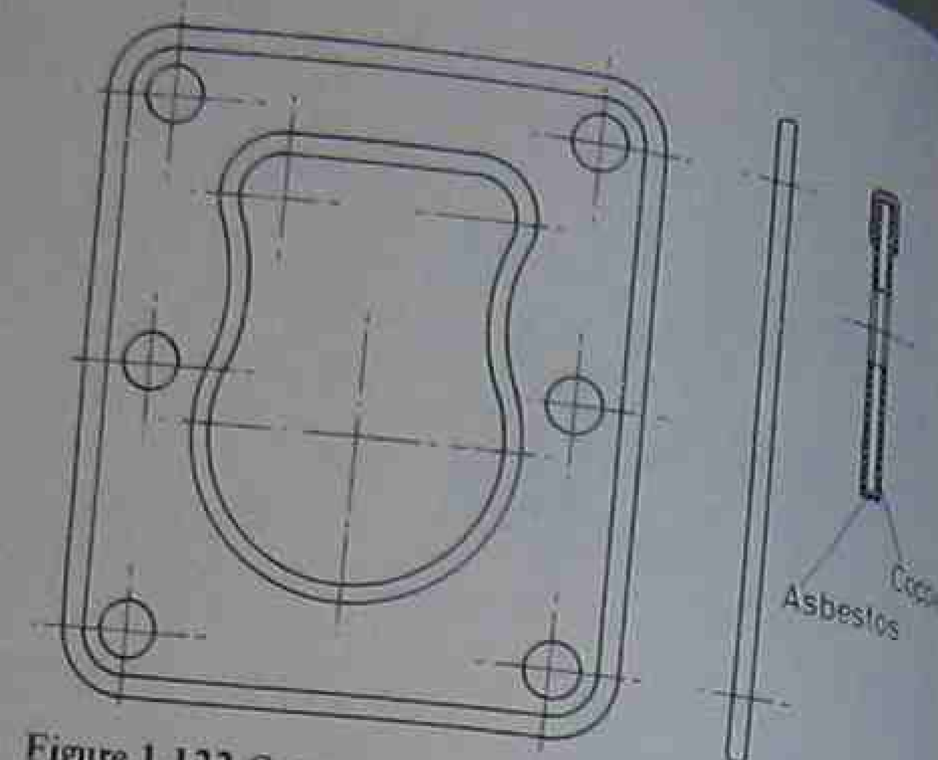


Figure 1.122 *Copper-asbestos gasket*

SEALING WASHER

Sealing washers are used to prevent leakage of fluids past the threads of bolts and screwed plugs. Metal washers are made of annealed copper or aluminium and compress on tightening.

Leather, fibre and plastic sealing washers are also used and there are several designs of metal sealing washers with synthetic rubber inserts.

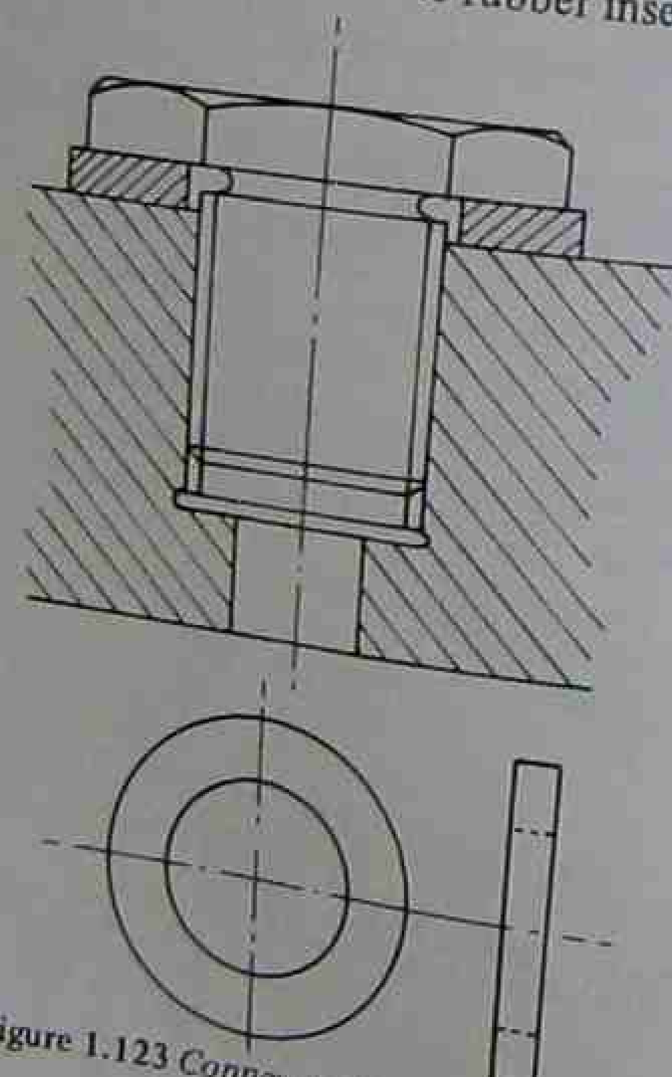


Figure 1.123 *Copper or aluminium sealing washer*

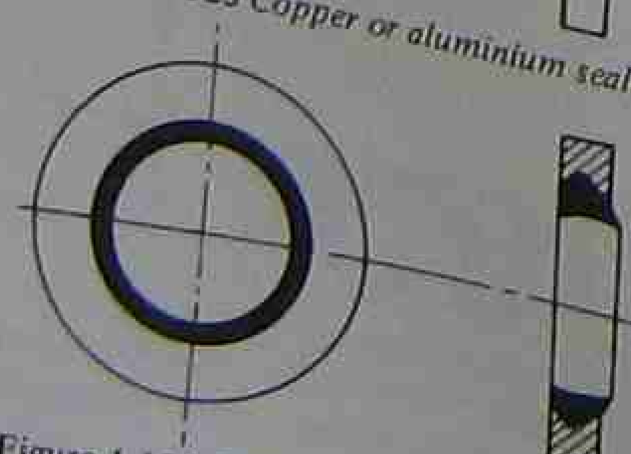


Figure 1.124 *Steel sealing washer with rubber insert*

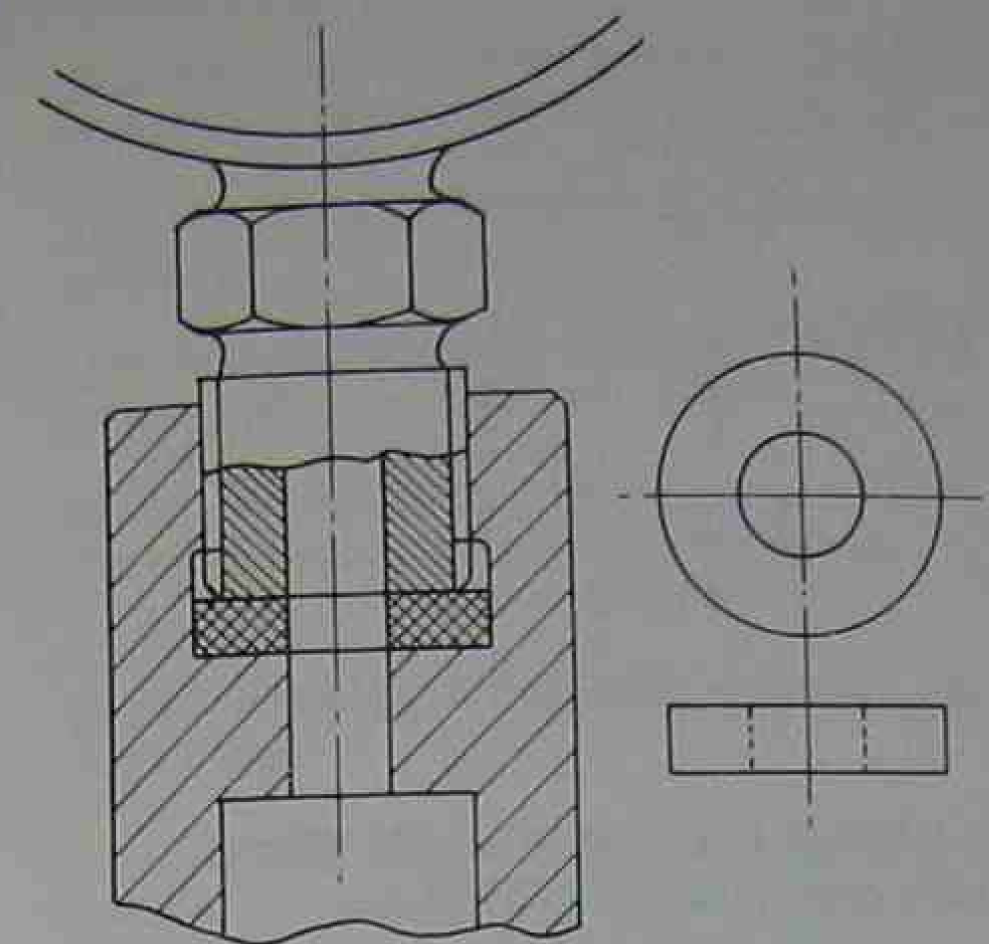


Figure 1.125 *Leather sealing washer for pressure gauge*

1.6 PIPES, PIPE FITTINGS, DUCTS AND VALVES

Pipes are tubes of metal, plastics, rubber, glass, etc., usually circular in section, used for conveying fluids or mixtures of solids and fluids.

The term *duct* is generally applied to pipes of large size made of sheet metal which are used for gases such as air.

A wide range of standard pipe and duct fittings, such as bends, couplings and valves is available.

SCREWED IRON PIPE JOINTS AND FITTINGS

These are made of wrought iron or malleable cast iron and they have ends to suit screwed fittings. They are used for water, air and gas mains.

The pipe ends are threaded and their fittings are provided with internal matching threads. To ensure perfect sealing the threads are treated with various jointing materials and cements before assembly.

The main types of fitting are:

Socket A threaded sleeve for joining straight pipes.

Elbow For joining pipes at right angles.

Slow Bend For right-angled bends but with a larger radius of curvature than the elbow.

T joint For joining three pipes, two of them in line and the other at right angles to them.

Reducer A straight fitting for joining pipes of different diameter.

Cap For closing the end of a pipe. A cap has an internal thread and it is fitted round the end of a pipe.

Plug This has an external thread and is used for closing the end of a fitting.

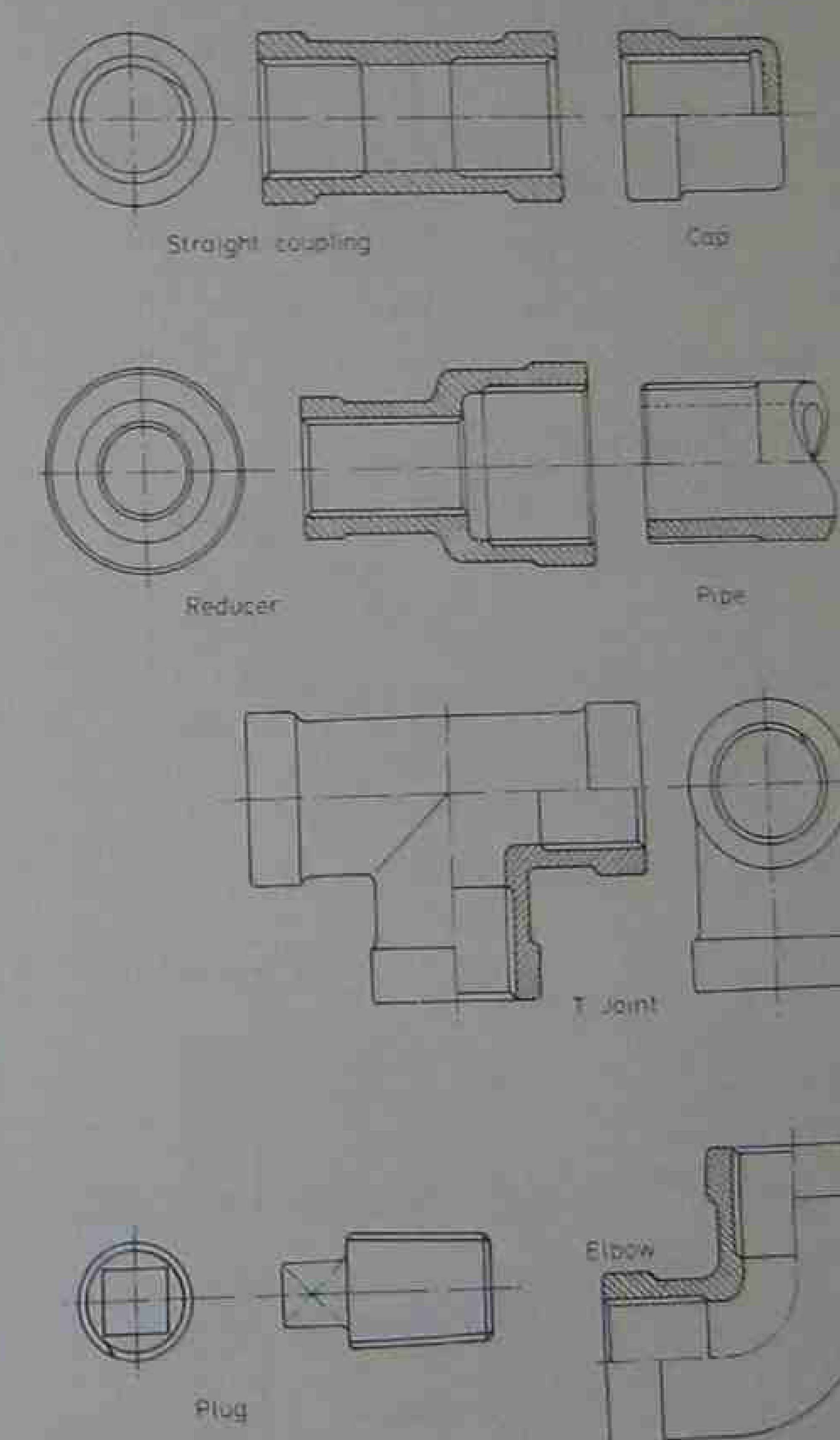


Figure 1.126 *Screwed iron pipe fittings*

FLANGED PIPE JOINTS

Medium and large bore pipes are often joined by bolted flanges either screwed or welded, or both screwed and welded, to the ends of the pipes. The joint may be 'face-to-face' or sealed with a washer or gasket.

Heavy-duty pipe joints have screwed and welded flanges with flanges and bolts which are larger than those used in low or medium pressure joints. A common type of *hydraulic pipe joint* has two-bolt flanges of heavy construction and a soft copper sealing ring which fits into a recess in one of the flanges. *Flanged pipe bends* are available with various ratios of bend radius to pipe bore.

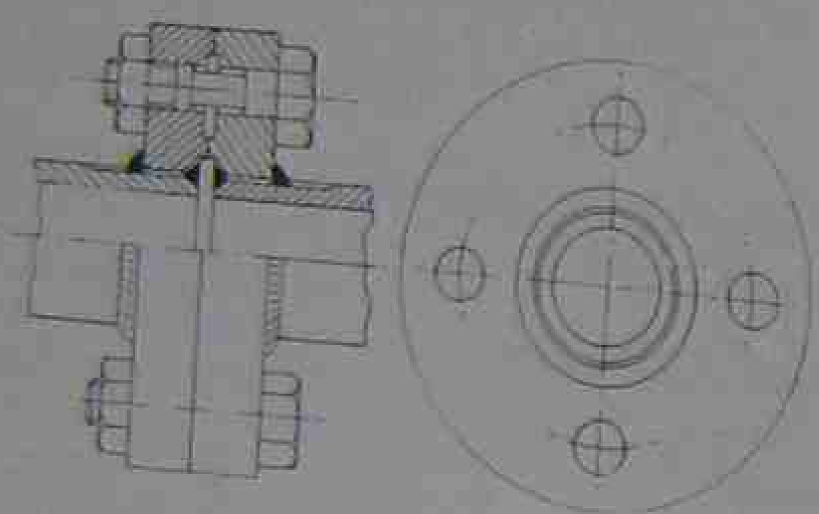


Figure 1.127 *Flanged pipe joint*

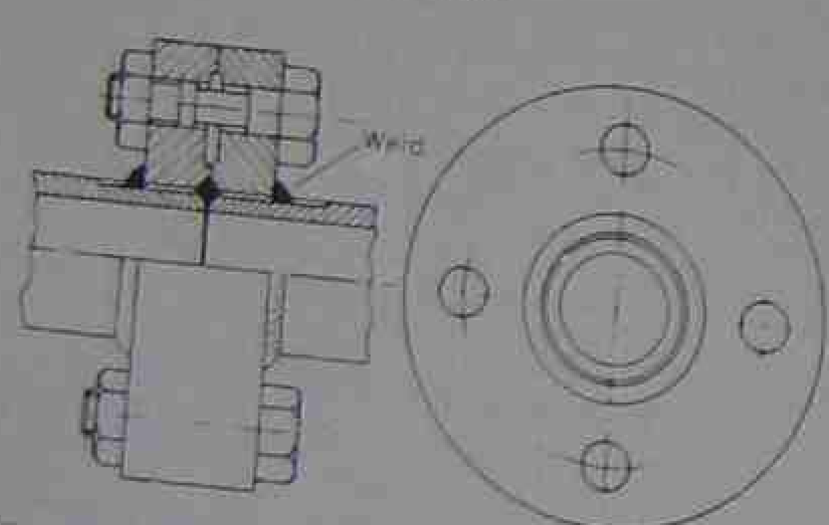


Figure 1.128 *Heavy-duty pipe joint*

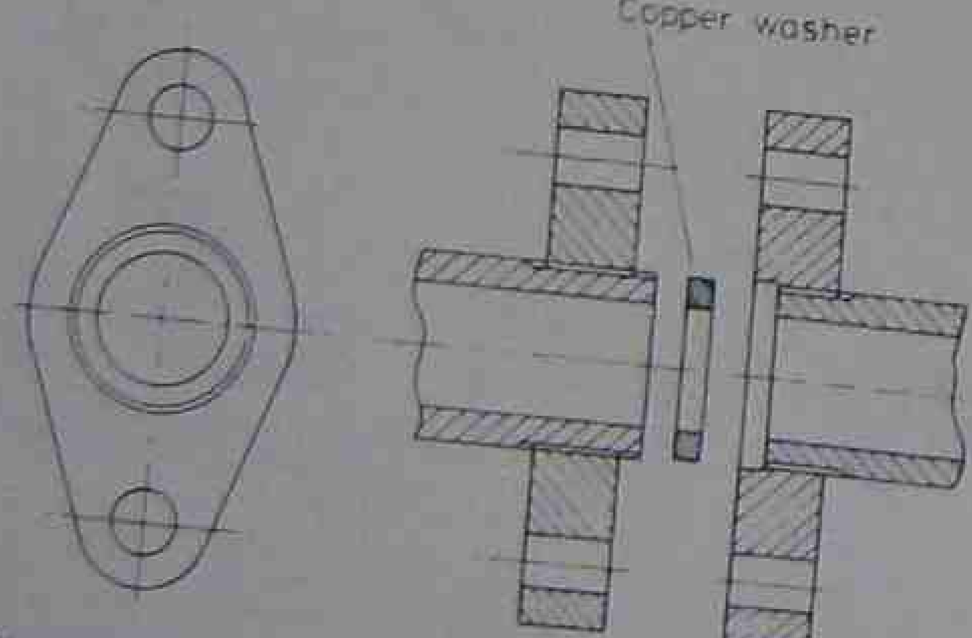


Figure 1.129 *Hydraulic pipe joint*

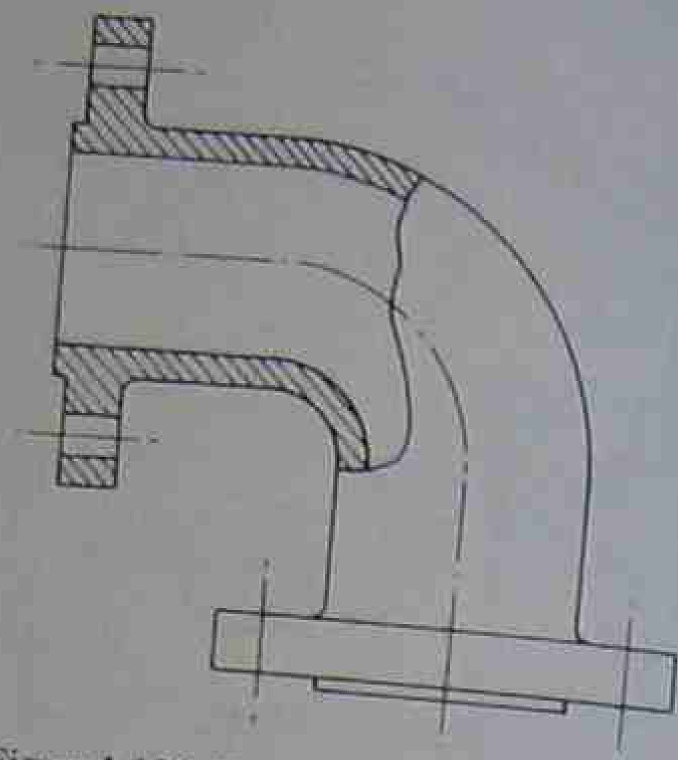


Figure 1.130 *Flanged pipe bend*

SOLDERED COPPER PIPE FITTINGS (CAPILLARY PIPE FITTINGS)

These fittings, designed for use with copper pipes, contain solder and only require to be heated by a blowlamp to make the joint.

The pipe, which must first be cleaned, is inserted into the fitting containing an internal ring of soft solder which, when melted, flows by capillary action and bonds the pipe to the fitting.

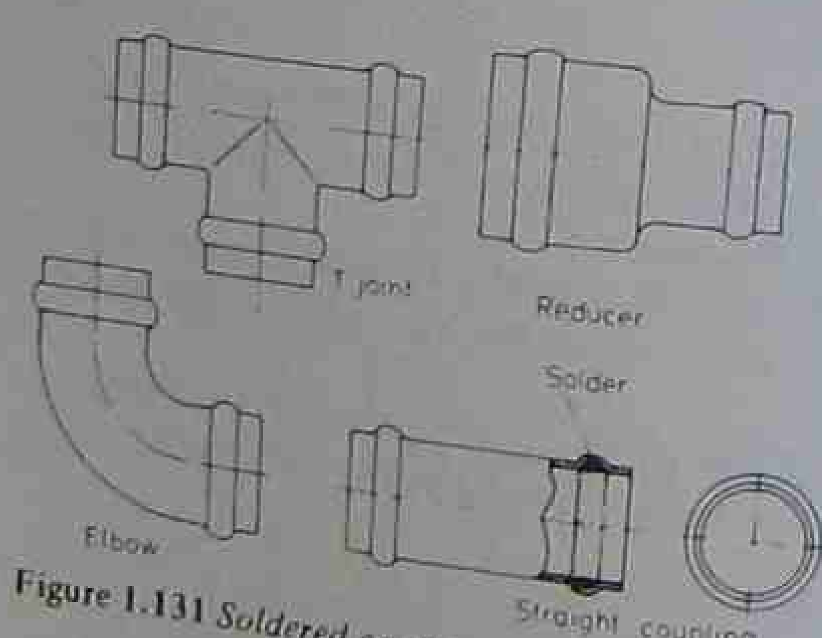


Figure 1.131 *Soldered copper pipe fittings*

PIPE AND DUCT MATERIALS

Pipes for all kinds of fluids are made from metals, plastics, rubber and canvas, etc.

Copper pipe This is made in bore sizes from a few mm up to 15cm and is used for water, oil, gas, and low pressure steam.

Heavy gauge steel Used for hydraulic pipes.

Lap-welded wrought iron Pipes of large and medium bore for conveying water are made from this material.

Solid drawn steel Used in the making of large and medium bore pipes for steam.

Brass For small size pipes as used in lubricating systems.

Galvanised steel/aluminium alloy sheet Metals used in the manufacture of ducting.

Plastics Polythene, PVC, nylon, etc. are examples of plastics utilised for the making of flexible pipes.

SCREWED BRASS PIPE COUPLINGS AND FITTINGS

A number of types of screwed pipe couplings are available and these may be used with copper pipes of up to about 40mm bore. They usually have a ring, known variously as a nipple, ferrule, olive or sleeve, fitted over the end of the pipe and secured to it by soldering, brazing or compression. The joint is made by means of a nut which holds the ring tightly onto the part to which the pipe is to be attached.

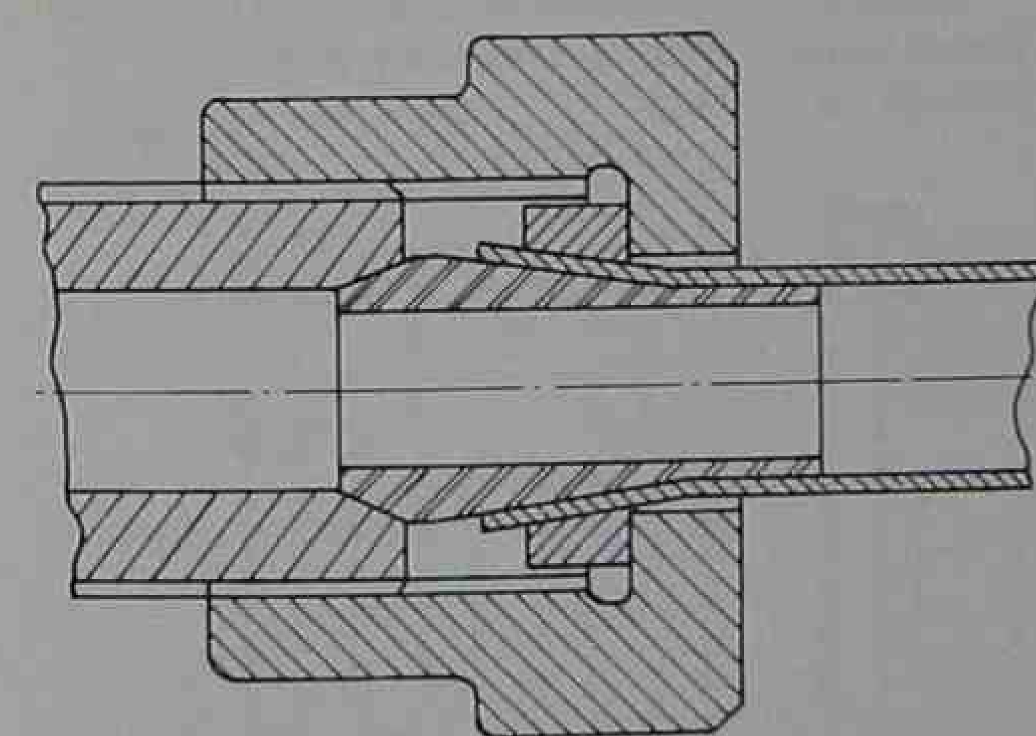


Figure 1.132 *Screwed brass pipe coupling (nut-and-sleeve)*

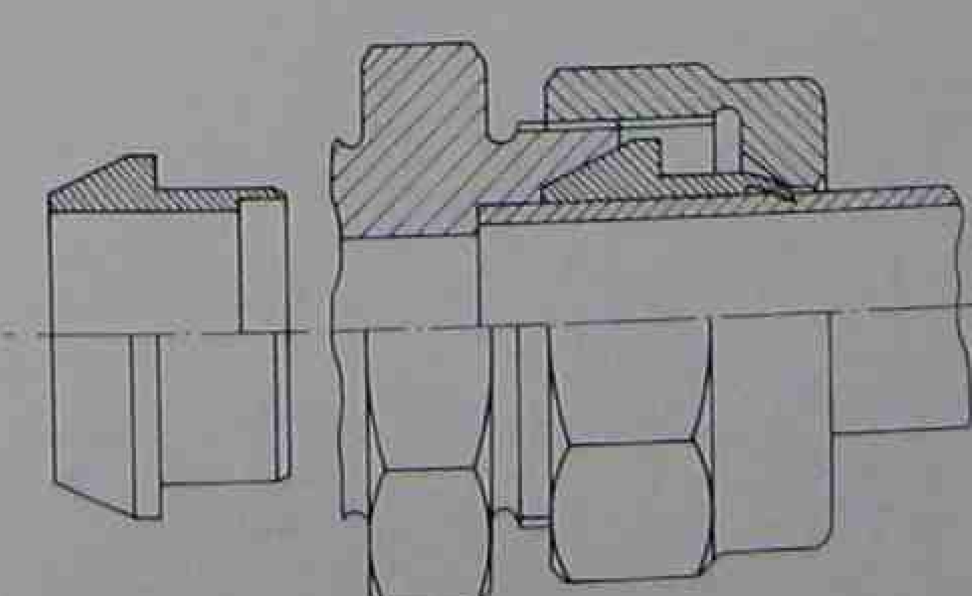


Figure 1.133 *Ermeto pipe coupling (steel pipe)*

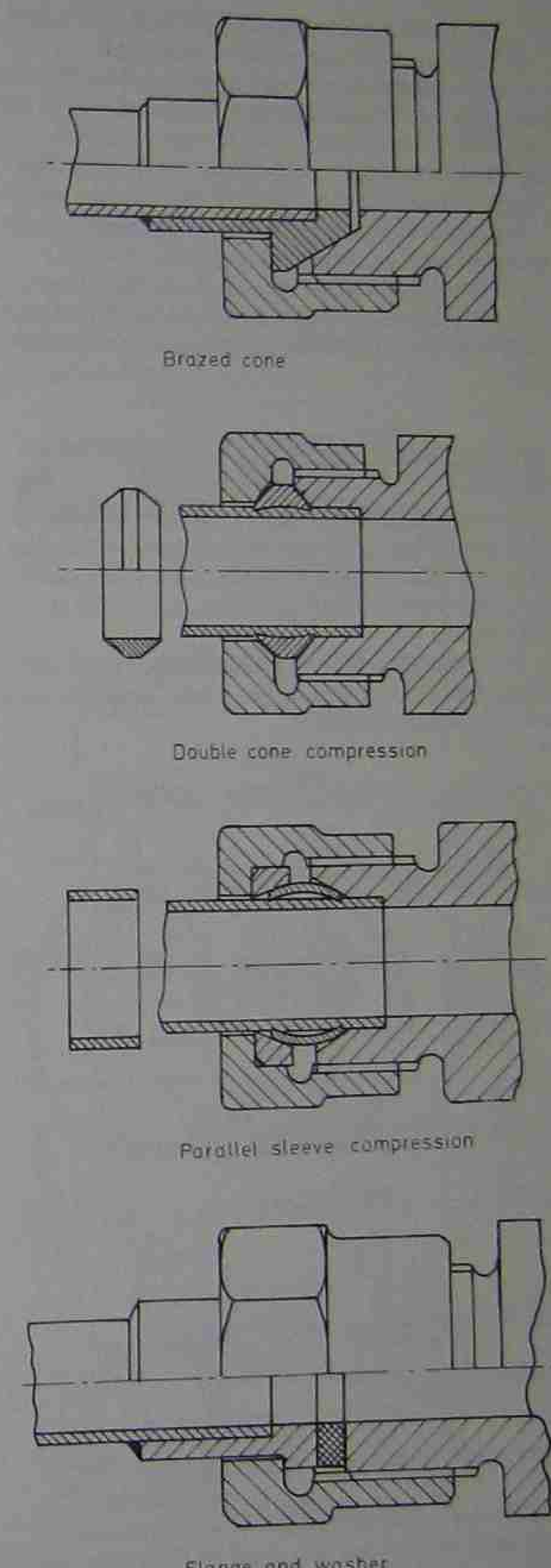


Figure 1.134 *Screwed brass pipe couplings*

The *nut-and-sleeve coupling* has a tapered sleeve which fits into and opens out the bore of the pipe, and a coned ring on the outside of the pipe. When the nut is tightened the pipe is nipped between the sleeve and the ring.

Sleeves attached by soldering or brazing may be single coned or double coned. Straight sleeves are compressed into the pipe by tightening the nut so that they deform and become curved.

One type of compression coupling, the *Ermeto coupling*, has a coned steel sleeve used with steel pipe, and this system is suitable for high pressure hydraulic pipes.

SCREWED BRASS PIPE FITTINGS

The *straight coupling* is used to connect two pipes in line: the *stud coupling* connects a pipe to some other parts; the *elbow* connects pipes at right angles; the *T coupling* connects two pipes in line and a third at right angles.

For each of these fittings the ends may be of different sizes (reducer), and one or more ends may have a female (internal) thread.

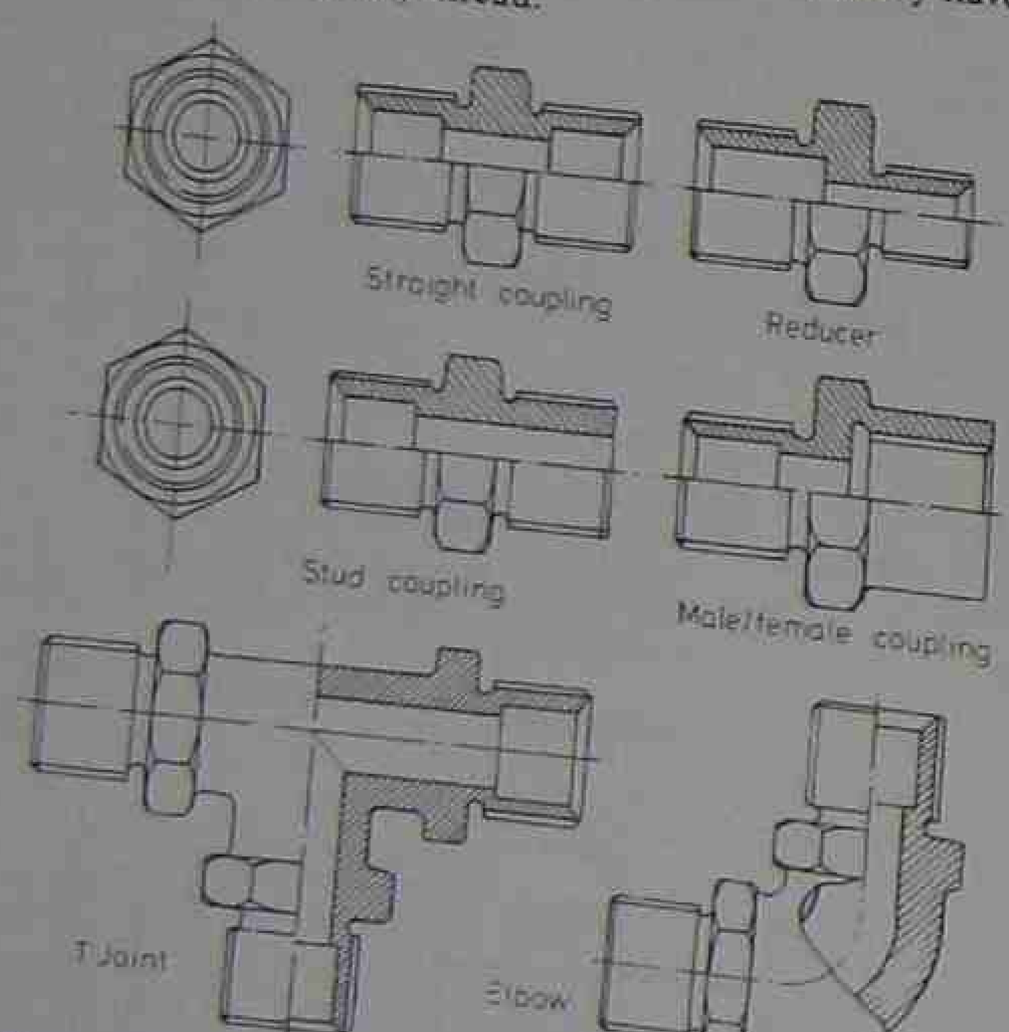


Figure 1.135 *Screwed brass pipe fittings*

EXPANSION JOINTS AND BENDS

When there may be relative movement of the ends of a pipeline it is necessary to introduce flexibility, and this can be done by using flexible pipes or flexible joints.

The *sliding expansion joint* consists of two tubes, one sliding inside the other with a gland to prevent leakage, and rods limiting the movement.

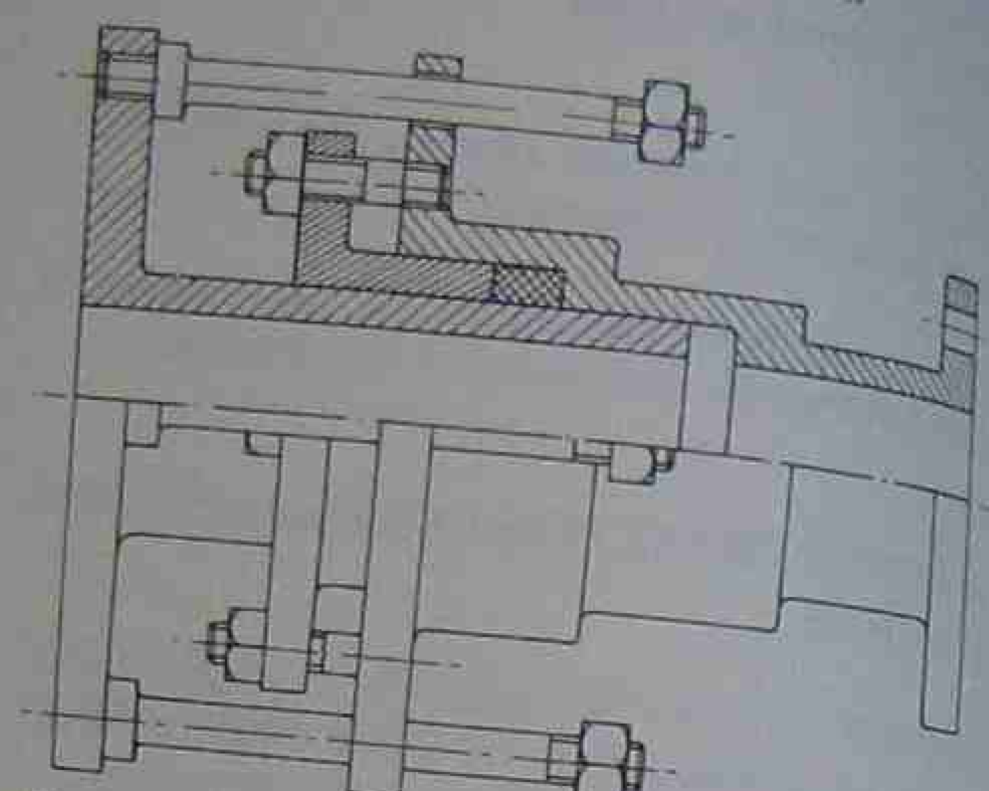


Figure 1.136 *Sliding expansion joint*

An alternative, the *bellows joint*, has a section of pipe replaced by a metal bellows which expands or contracts with relative movement of the pipes. If only an angular movement is desired a hinge is fitted.

Flexibility can be obtained from either U or lyre-shaped bends in which the elasticity of the pipe material itself is used.

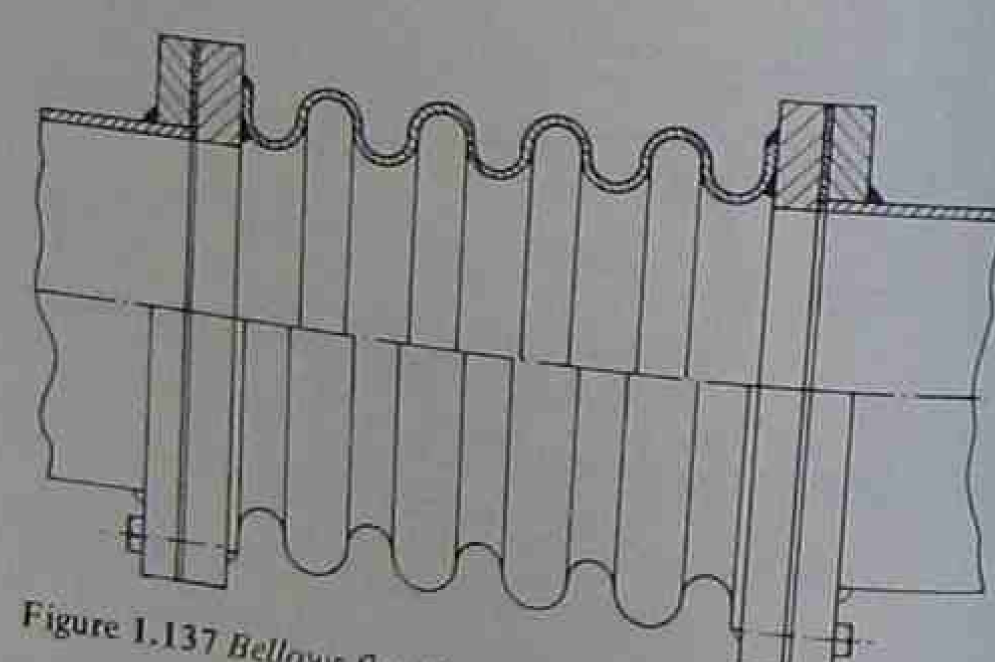


Figure 1.137 *Bellows flexible pipe joint*

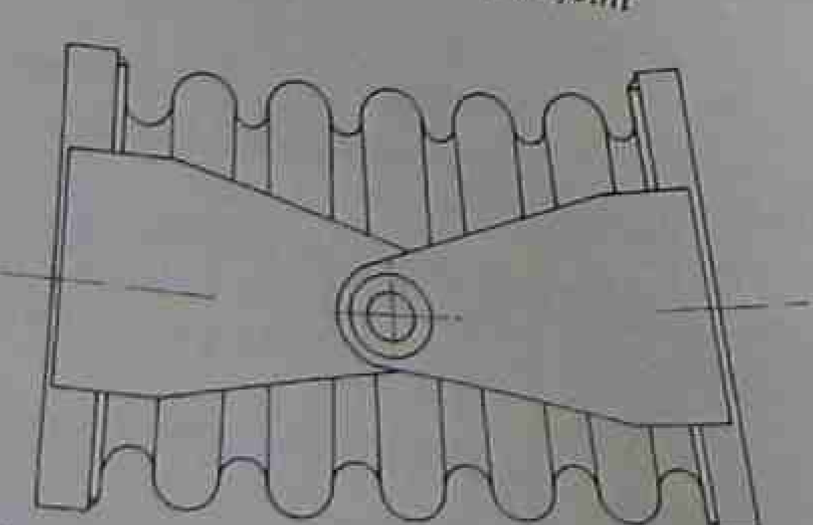


Figure 1.138 *Hinged bellows joint*

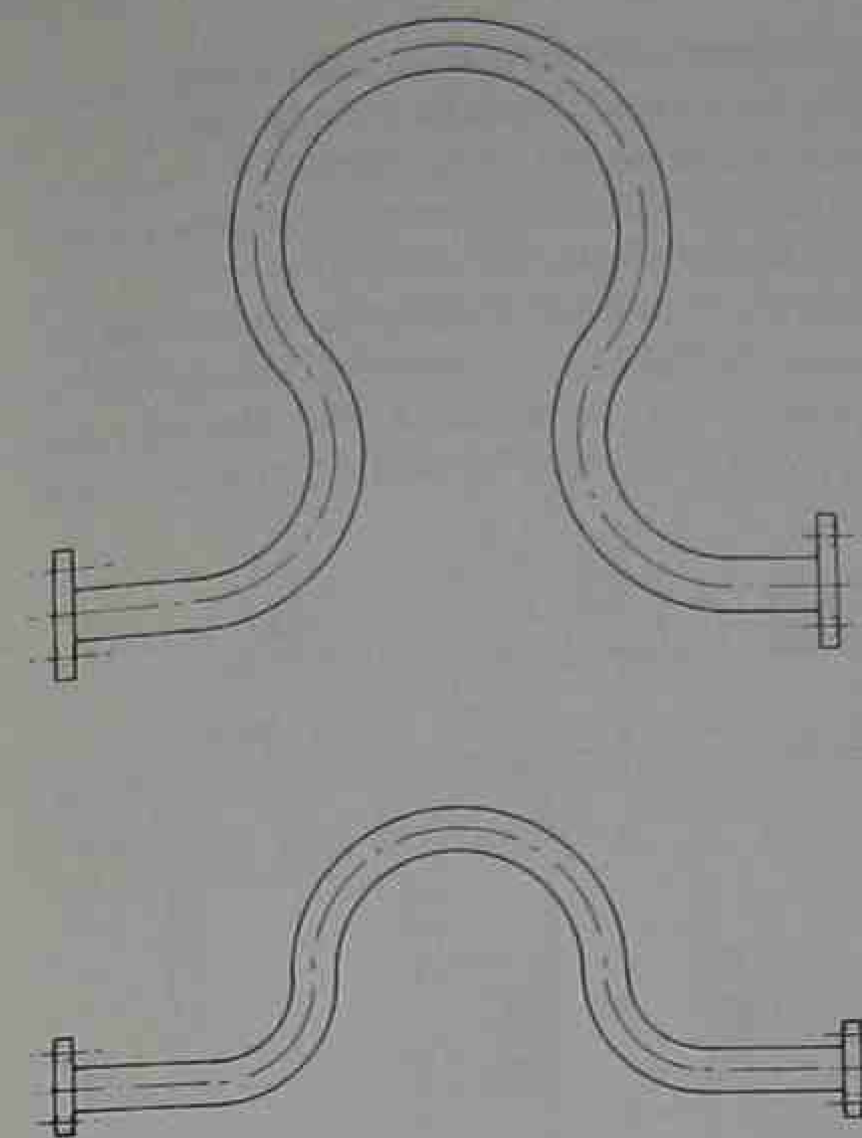


Figure 1.139 *Flexible expansion pipe bends*

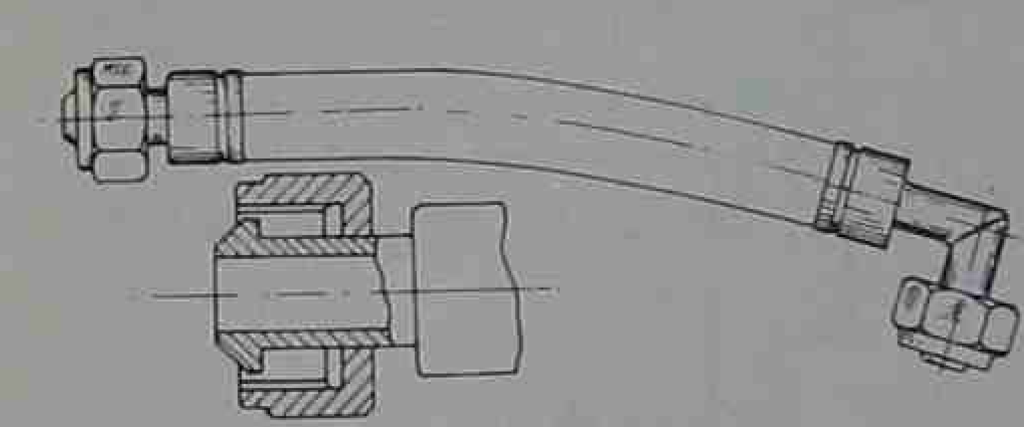


Figure 1.140 *Hydraulic hose and fittings*

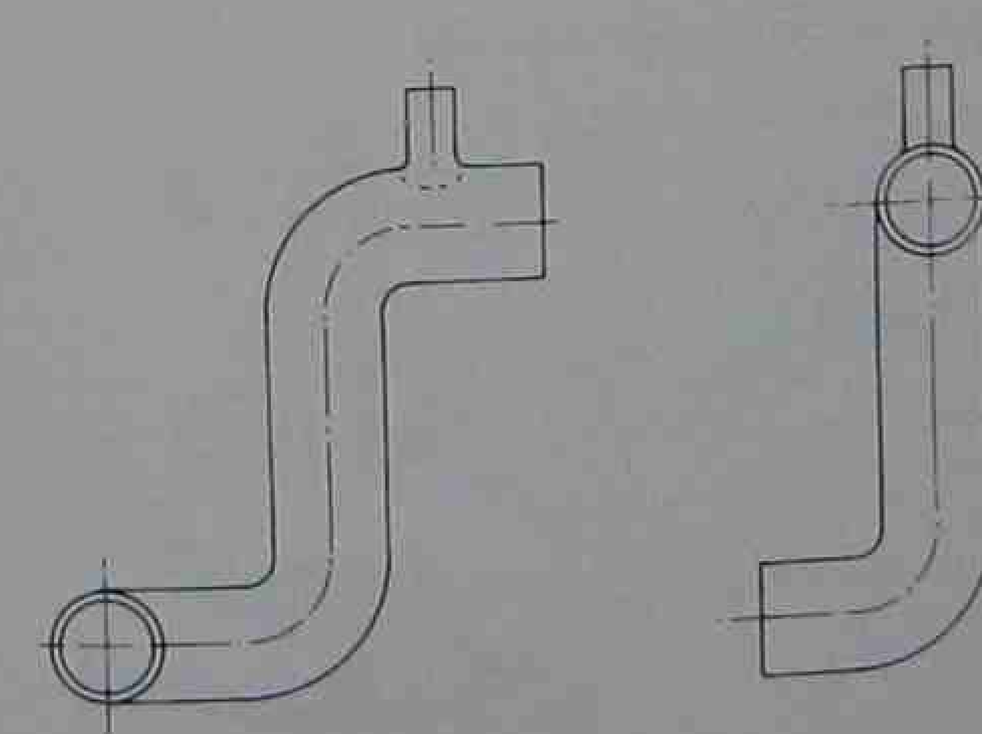


Figure 1.141 *Rubber hose for automobiles*

FLEXIBLE PIPES

Flexible pipes are used extensively in engineering.

Hydraulic hose, made of synthetic rubber reinforced with steel braiding and canvas, is able to withstand high pressures. Screwed couplings are attached to the ends.

Braided plastic petrol pipe is used for IC engines.

Canvas-reinforced rubber pipes (or hoses) are used for water cooling systems in automobiles.

Flexible metal pipes include the types shown in *Figure 1.143* and also the bellows type similar to that shown in *Figure 1.147*.

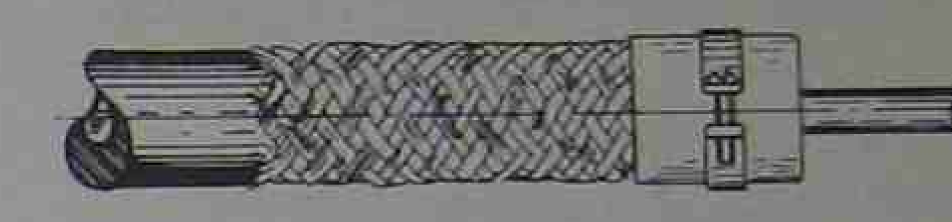


Figure 1.142 *Flexible petrol pipe*

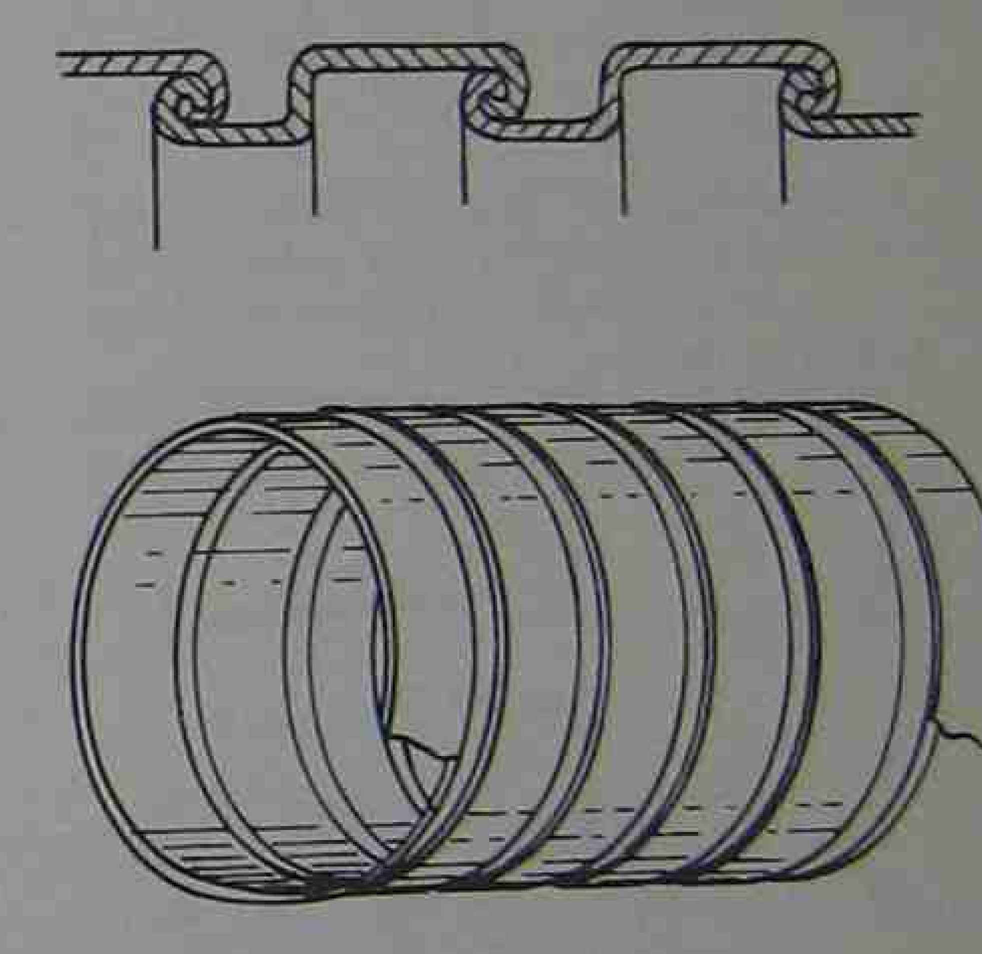


Figure 1.143 *Flexible metal pipe*

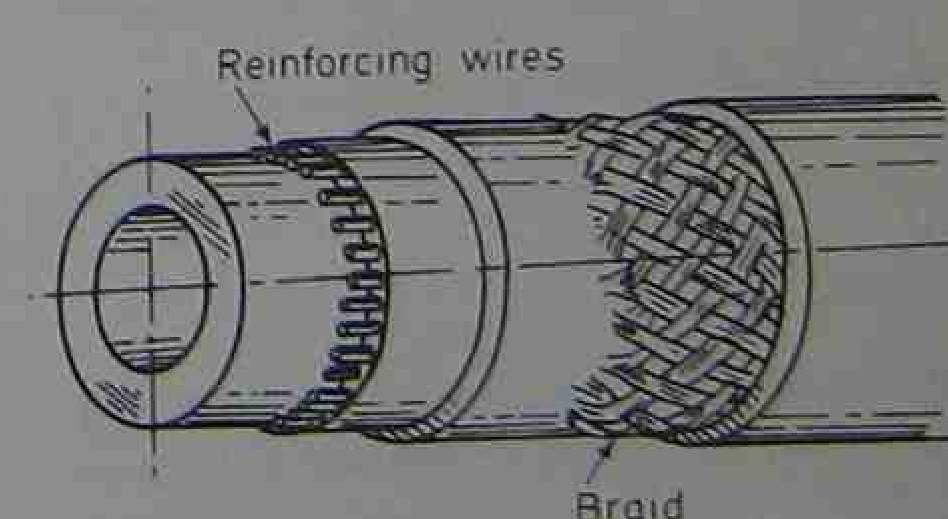


Figure 1.144 *Construction of high pressure hydraulic hose*

FITTINGS FOR DUCTING

Ducts made of sheet metal are usually joined by bolted flanged joints with gaskets of rubber, cork and proprietary jointing.

Fittings used include bends, transition pieces for joining ducts of different cross-section, and branched pipes.

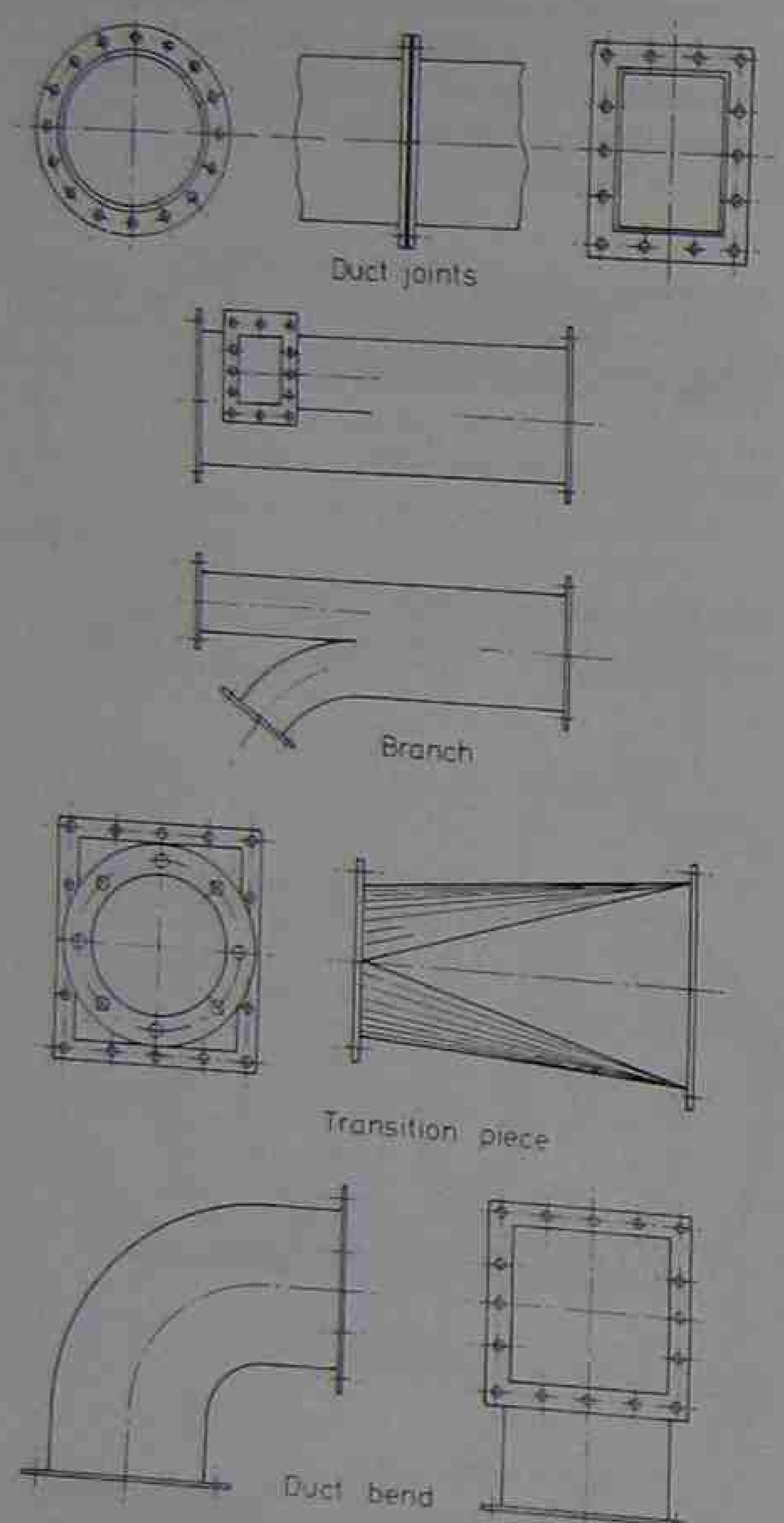


Figure 1.145 *Ducting and fittings*

VALVES

Valves are devices for controlling the flow of a fluid and they may be manually or automatically operated.

Stop valve Stop valves are used for shutting off the flow of steam, gas or liquids. One type is known as a *screw shut-down valve* or, because of its shape, a *globe valve*. This has a vertical spindle passing through a nut and carrying a chamfered valve disc which rests on a conical valve seat when the valve is closed. The spindle is provided with a gland.

For small bore systems the *needle screw-down valve* is used. In this type the spindle is tapered to a conical point which closes onto a seat. The inlet and outlet pipes are connected by coned screwed unions.

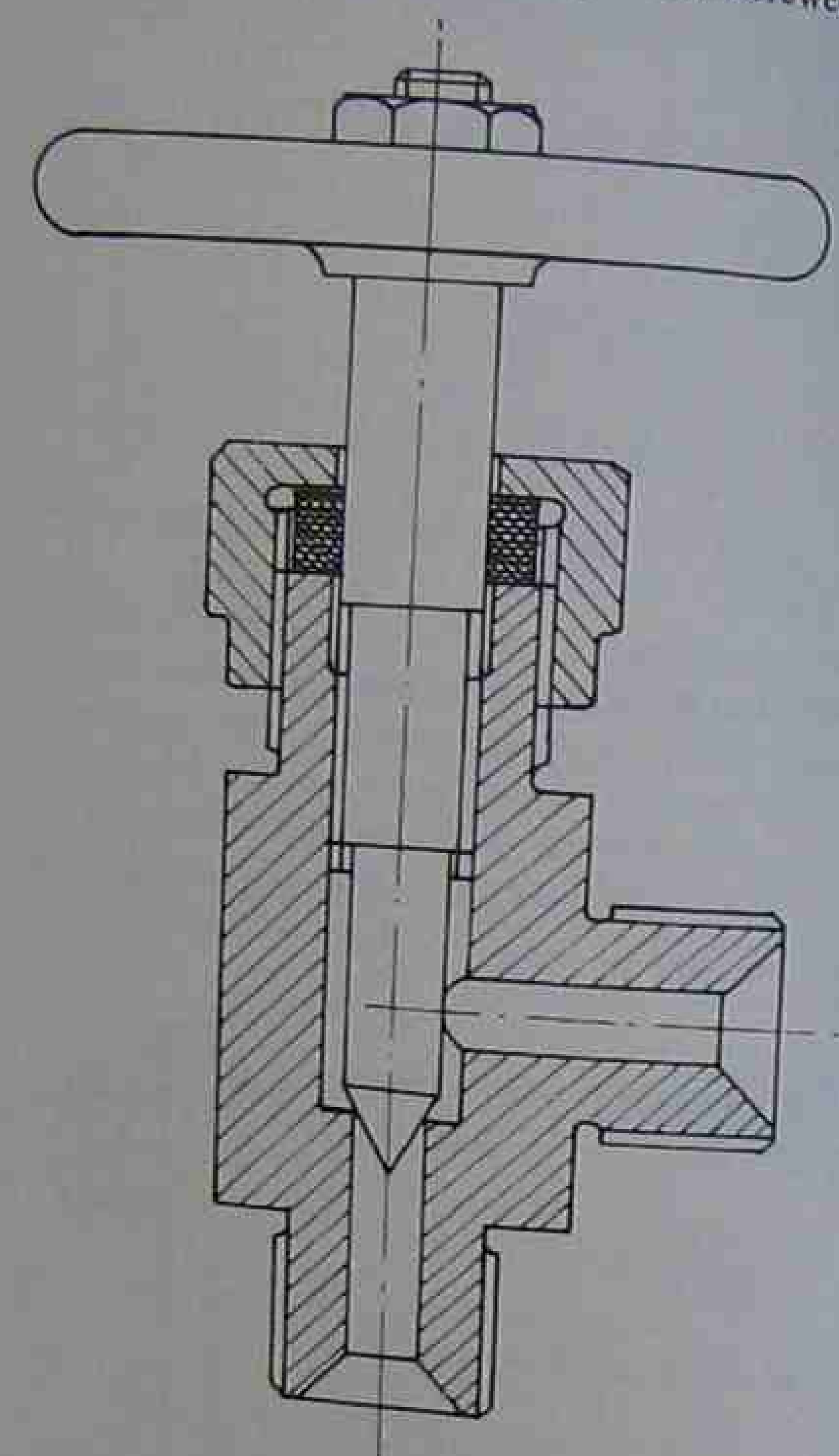


Figure 1.146 *Needle screw-down valve*

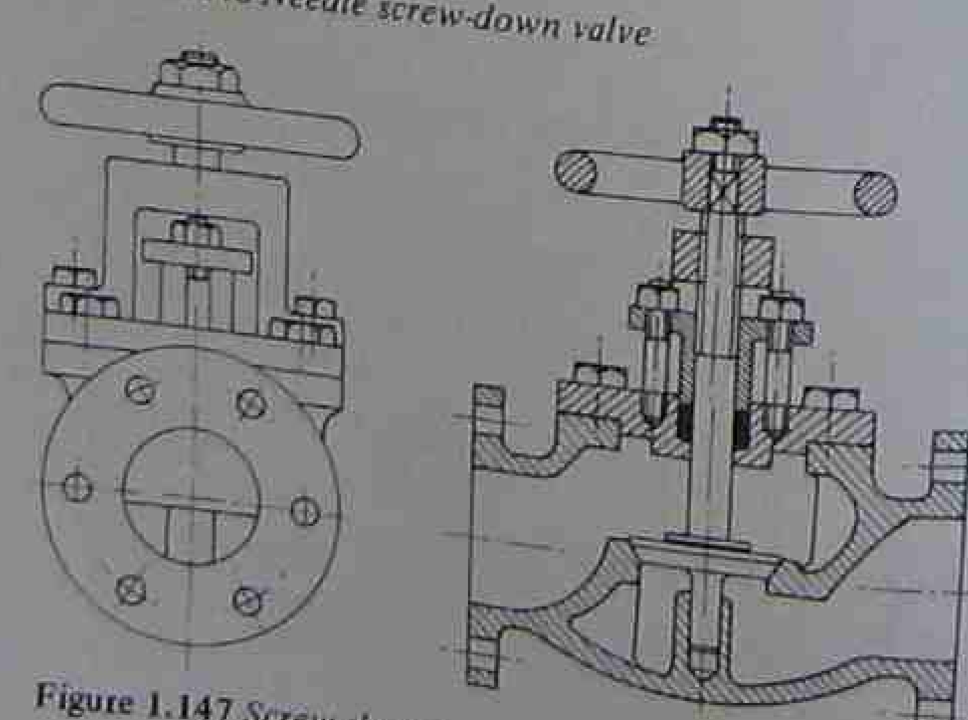


Figure 1.147 *Screw shut-down valve (globe valve)*

Steam safety valve The valve is normally kept closed by a dead weight, a weighted lever, or a spring which is initially compressed. At the required pressure the valve opens and allows steam to flow until the pressure has fallen sufficiently. The operating pressure is adjusted by moving the weight on the lever or altering the spring compression.

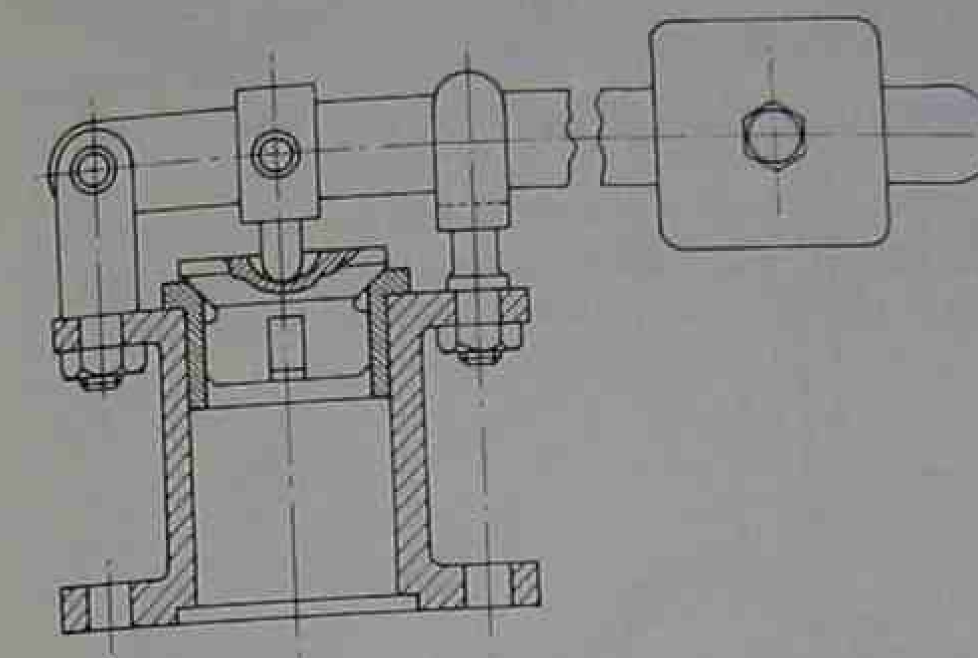


Figure 1.148 *Dead-weight safety valve*

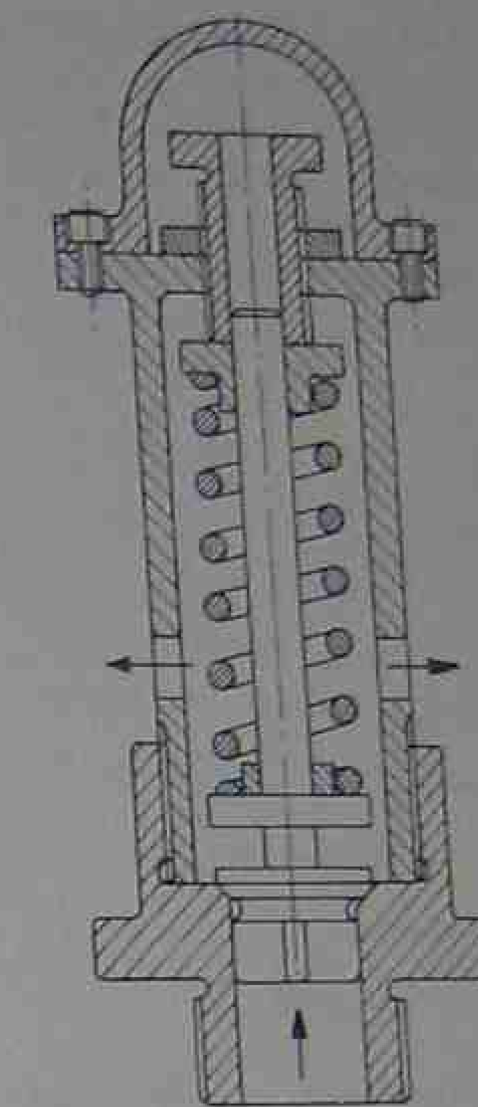


Figure 1.149 *Spring-loaded safety valve*

Hydraulic relief valve A valve used in hydraulic systems to prevent excessive pressure and subsequent damage.

A typical design has a spring-loaded pilot valve which opens at the desired pressure and causes a flow through a small hole in a large piston which normally holds the main valve shut. The flow results in a pressure difference across the piston and the main valve opens to allow the oil to flow to a tank.

Non-return valve, check valve These are valves which permit flow in one direction only.

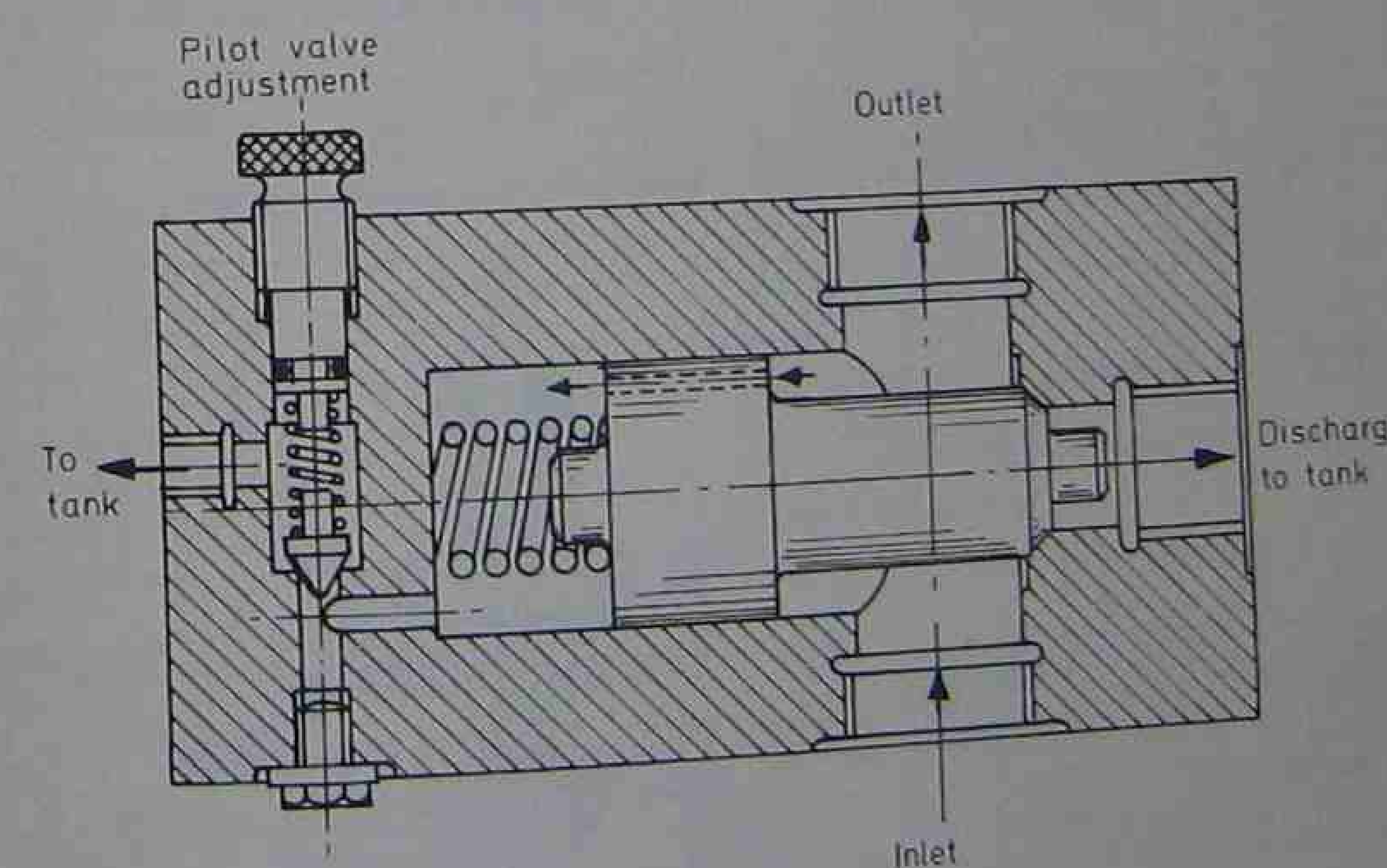


Figure 1.150 *Hydraulic relief valve*

In one type a ball, needle or disk, etc. is held onto a seat by means of a loaded spring and it opens with the fluid flow. In another valve a spring-loaded piston covers a port, and a third has a hinged flap held closed by gravity and opened by the flow of fluid. This last type is used to prevent blow-back when an explosion takes place in gas pipes.

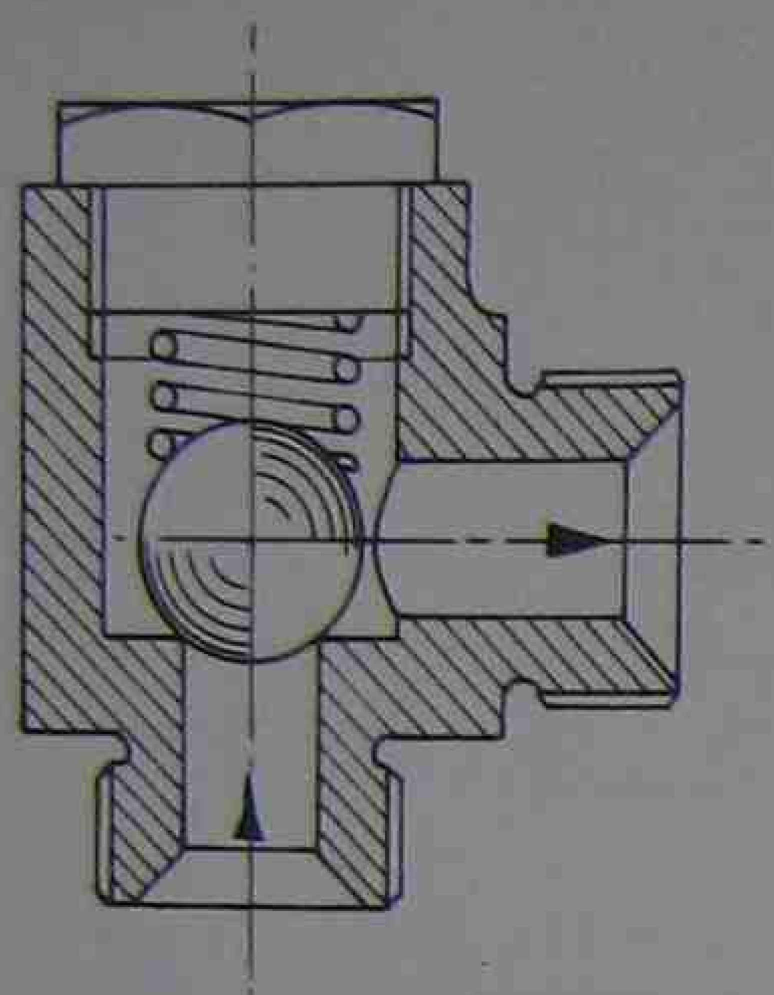


Figure 1.151 *Ball-type non-return valve*

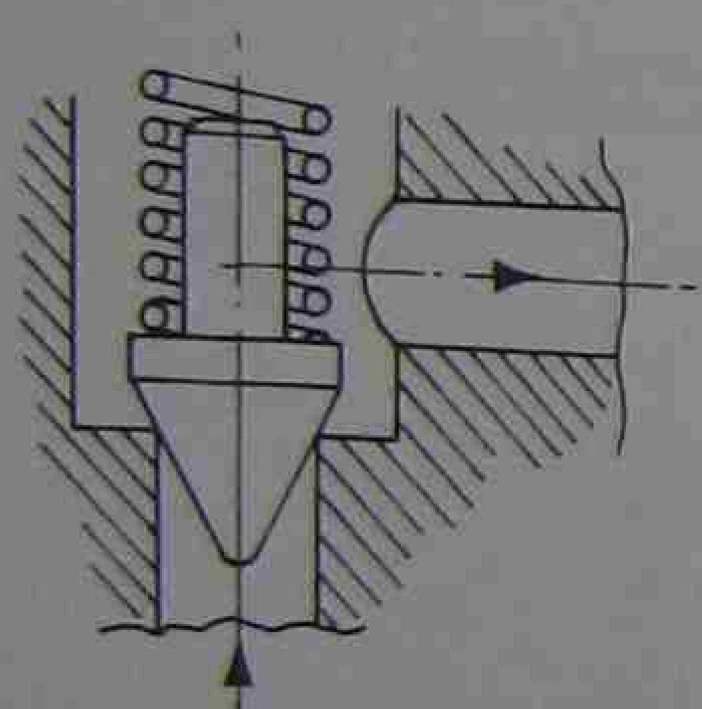


Figure 1.152 *Needle-type non-return valve*

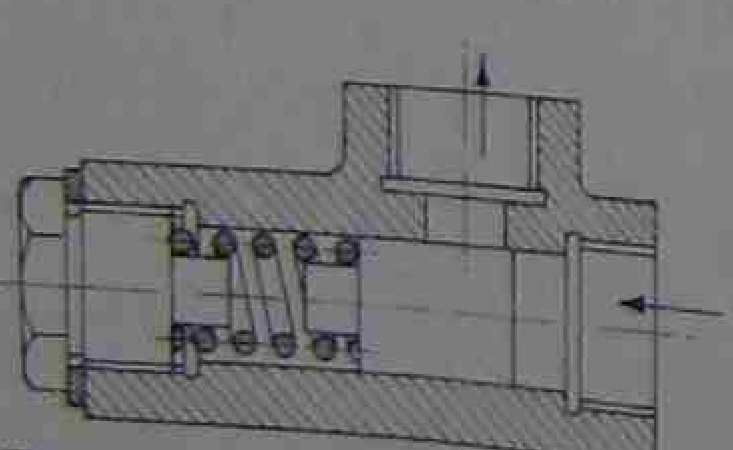


Figure 1.153 *Piston-type non-return valve*

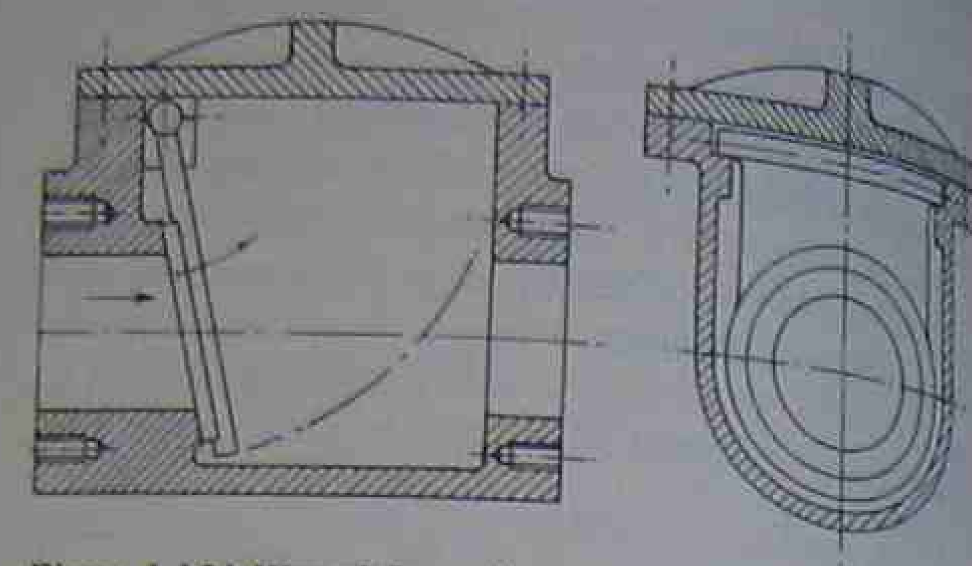


Figure 1.154 *Hinged flap valve*

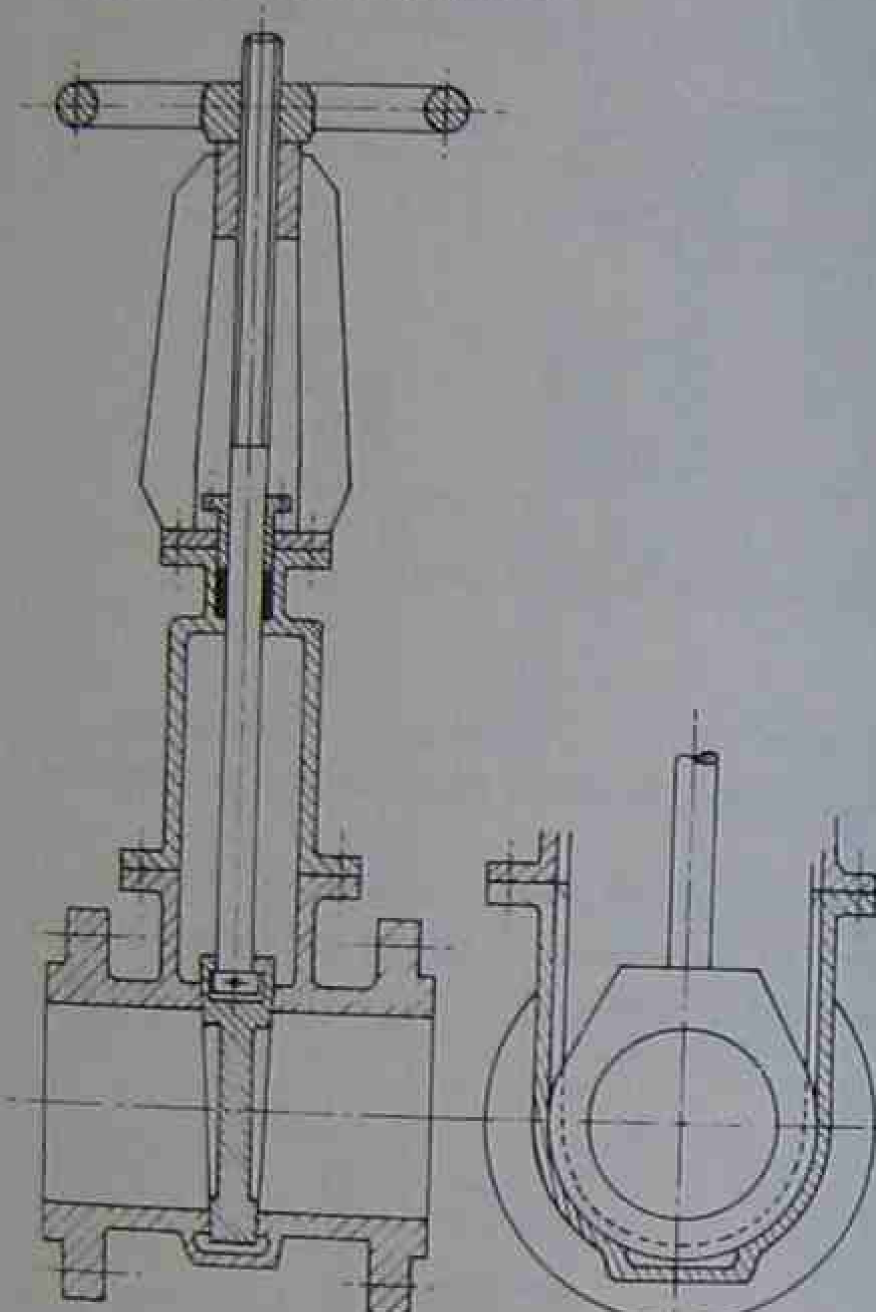


Figure 1.155 *Gate valve (sluice valve)*

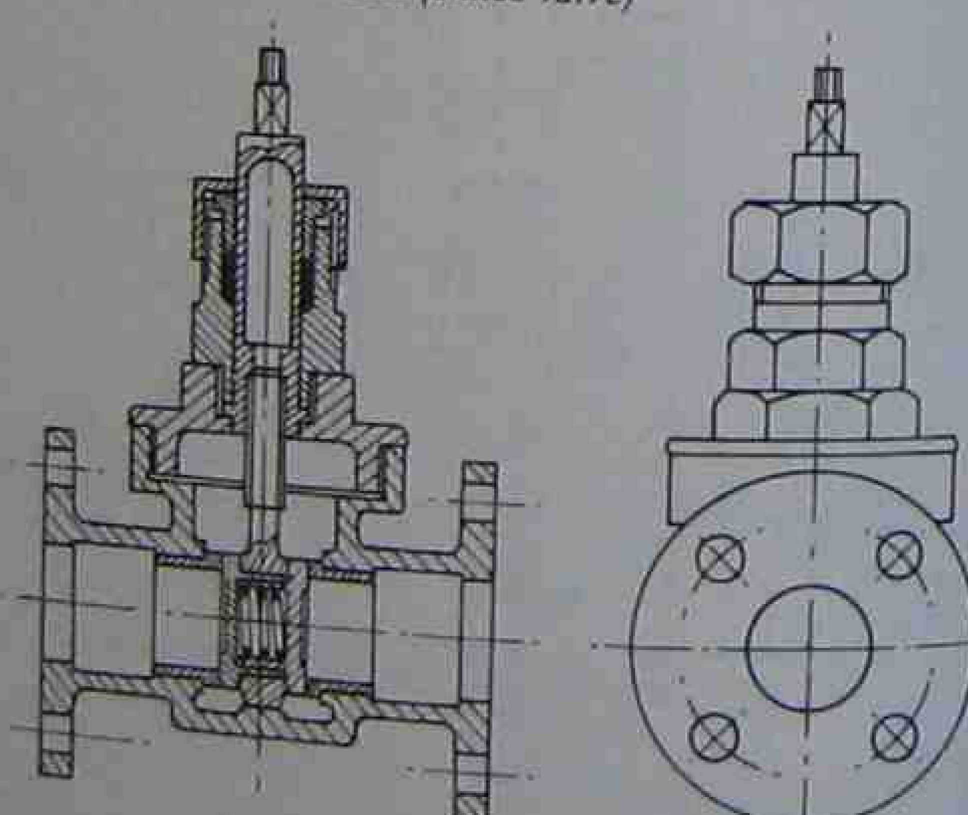


Figure 1.156 *Parallel face gate valve*

Gate valve The gate valve is intended as a shut-off valve but may be used to control the flow.

A *gate*, or shutter, which may be wedge-shaped or have spring-loaded parallel faces, is moved by a screw on the handle across the bore thus shutting off the flow. The gate slides in machined faces.

The wedge type of gate valve is also known as a *sluice valve*.

Butterfly valve The butterfly valve has a disk, pivoted about its diameter, which fits exactly into the bore of a pipe. The disk is rotated on the pivot to close or partially close the bore. This valve is used extensively in automobile engine carburettors for the throttle valve but is also made as a separate unit in large sizes.

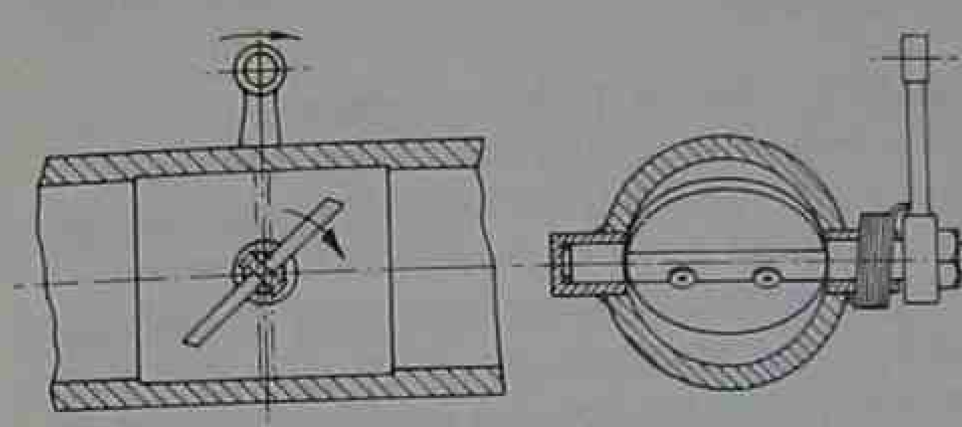


Figure 1.157 *Butterfly valve*

Saunders valve This is a proprietary shut-off and control valve in which a rubber diaphragm is moved down by a screw onto a metal bridge. The resistance to flow when fully open is small. The inside of the valve may be lined with materials resistant to heat, corrosion, etc.

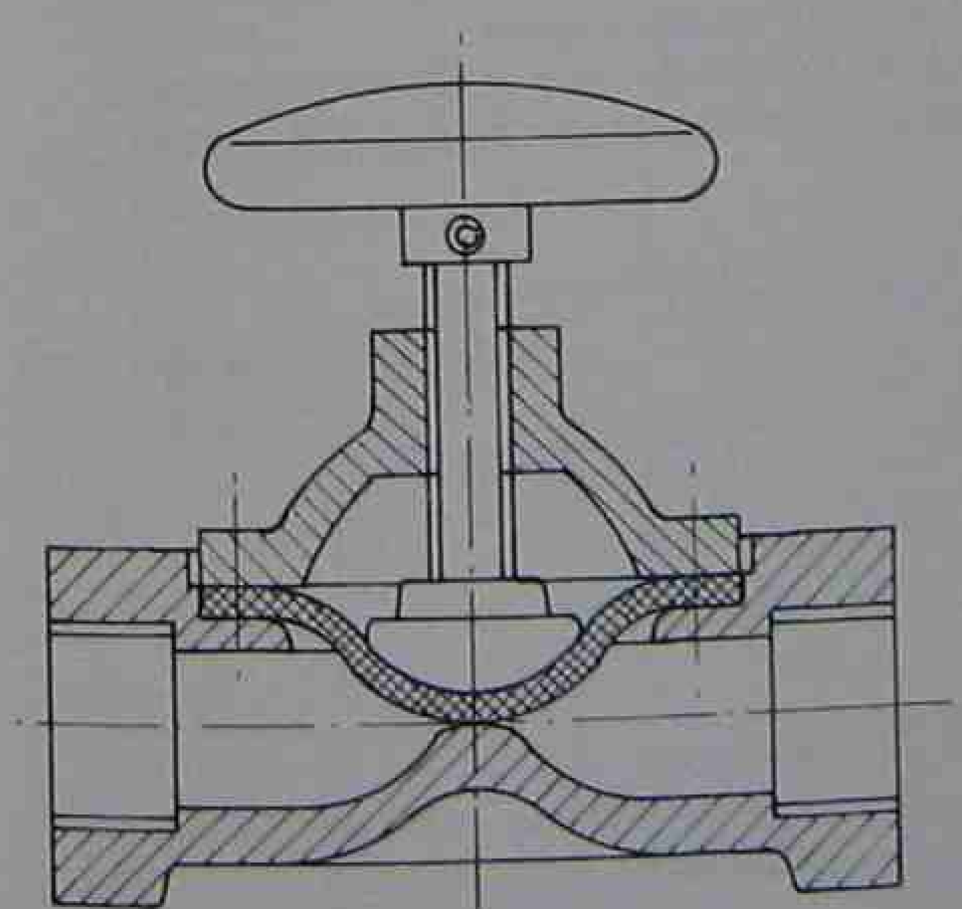


Figure 1.158 *Saunders valve*

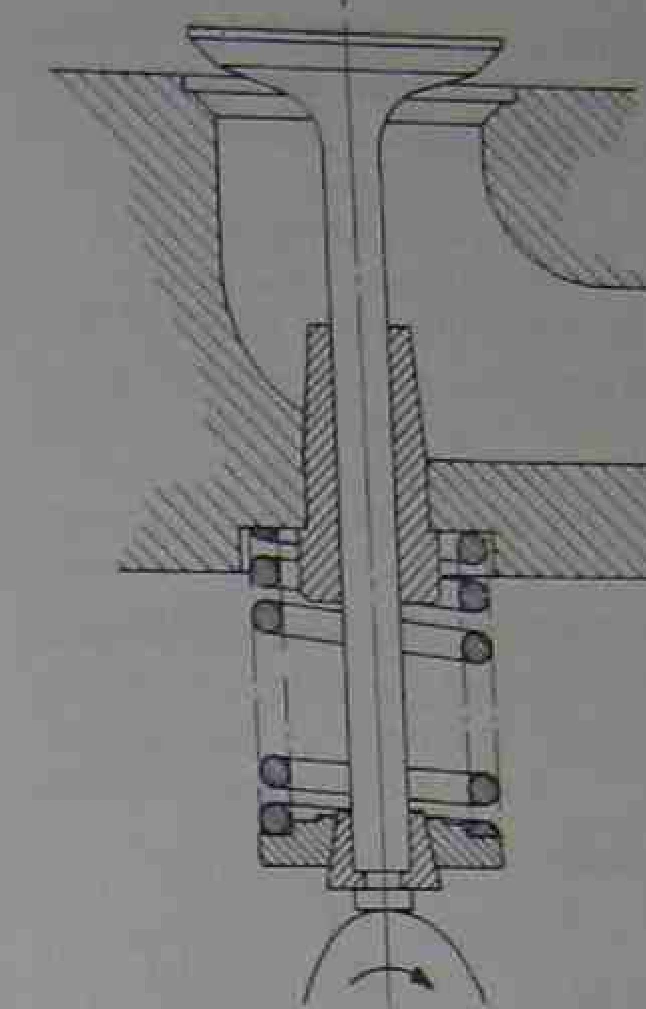


Figure 1.159 *Automobile-type poppet (mushroom) valve*

Poppet valve (mushroom valve) Most internal combustion engines use this type as inlet and outlet valves. The head rests on a conical seat and the valve is lifted by a cam against a compressed spring.

Sleeve valve A sleeve valve consists of a sleeve, pierced with *ports*, or openings, sliding or rotating in a cylinder also having ports which are made to open and shut. This valve is used in some IC engines.

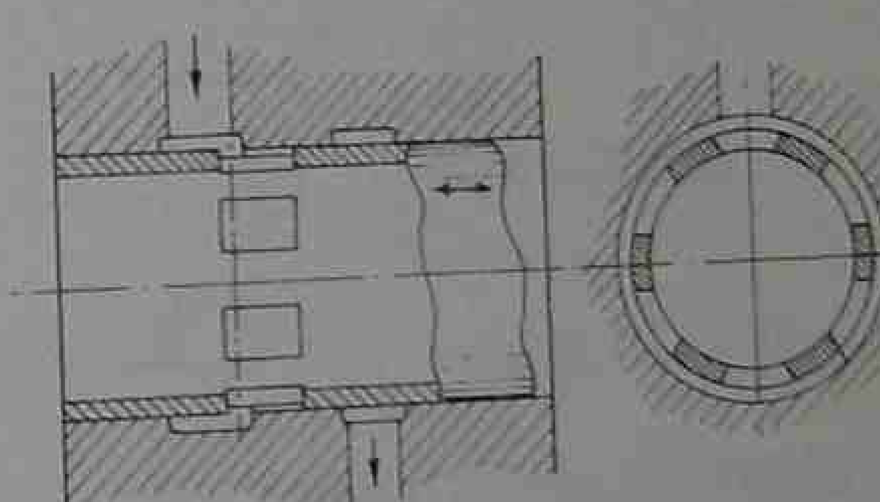


Figure 1.160 *Sleeve valve*

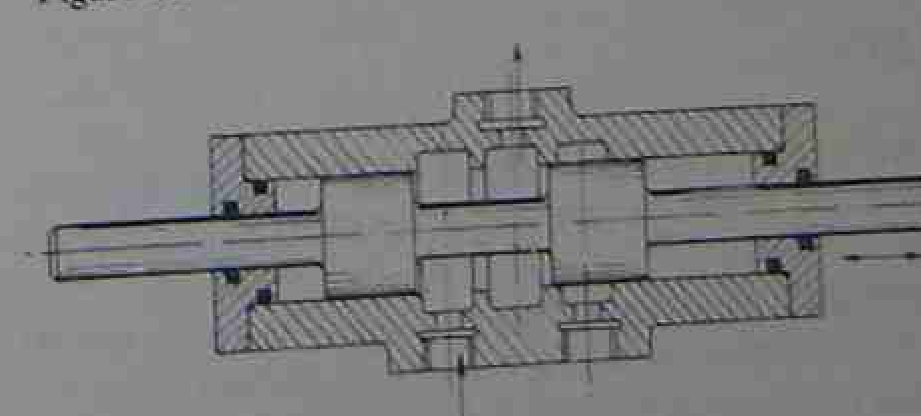


Figure 1.161 *Spool valve*

Spool valve Spool valves are used in hydraulic power circuits to control the flow of fluids. Pistons on a sliding rod open and close ports in a cylinder.

Plug cock The plug cock is used mainly for controlling gases. A taper plug fits into a tapered hole (in the body) at right angles to the flow direction. A hole drilled through the plug coincides with the bore of the body, and to close the cock the plug is turned through a right angle. For this purpose the plug may be fitted with a handle or have a square head to suit a key or spanner.

A similar valve of larger size has a spherical plug.

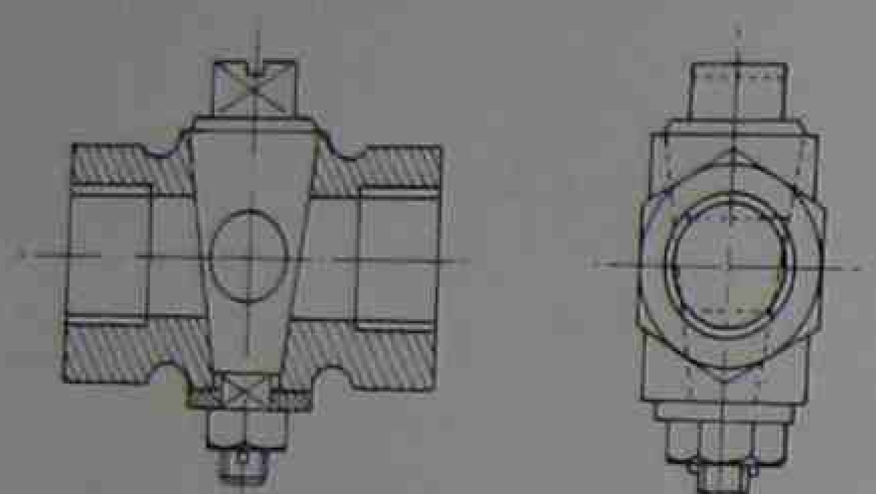


Figure 1.162 *Plug cock*

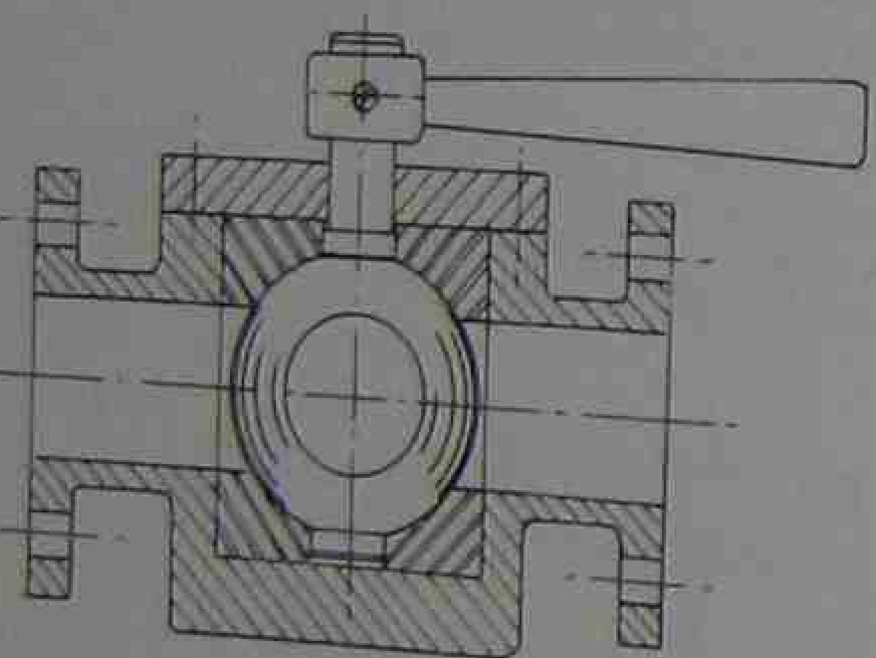


Figure 1.163 *Spherical plug cock*

1.7 SHAFTS AND RODS

SHAFTS

A shaft is a machine element used for transmitting motion and power by means of rotation. It is invariably of circular cross-section, may be solid or hollow, rotates in bearings, and carries gears, rotors, wheels, etc. It is subject to torsion and to transverse and longitudinal loads which are often fluctuating. Small diameter shafts are also called *spindles*.

Materials Most shafts are made of low or medium carbon steel, the later often heat treated. The surface may be heat treated to harden it to limit wear, as fo bearings and seals. Shafts are also made of high-strength steel and aluminium alloys and are sometimes protected by plating or by nonferrous sleeves, e.g. in pumps.

Applications Shafts are used extensively in machines of all kinds, e.g. machine tools, gearboxes, automobile and marine engines, aero-engines, domestic appliances etc.

Surface finish The surface finish is generally that produced by turning but some sections are ground to a high finish for bearing journals and where there are contact oil seals.

Solid shaft Most shafts have a solid circular cross-section with steps or shoulders for the location of parts mounted on them. The ends of the shaft and the ends of sections are often chamfered to prevent burring, and internal corners are often given a radiused fillet to prevent stress-concentration. The shaft may have non-circular sections, e.g. square, hexagonal, etc.

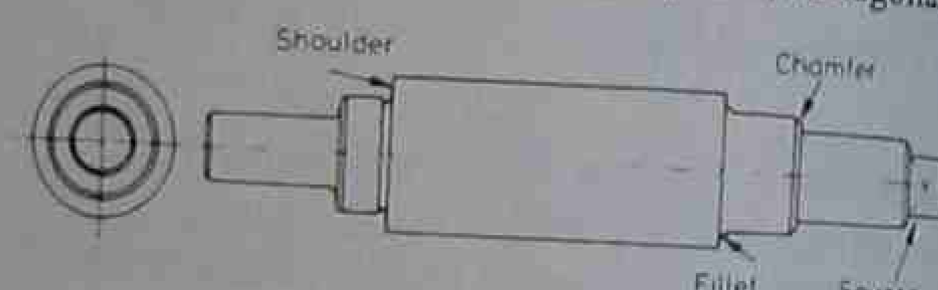


Figure 1.164 *Solid shaft*

Hollow shaft A hollow shaft is much stronger both in torsion and in bending than a solid shaft of the same weight. Often used where lightness is essential, as for example in aero-engines.

Another application is that of concentric shafts where a solid shaft may run inside a hollow one.

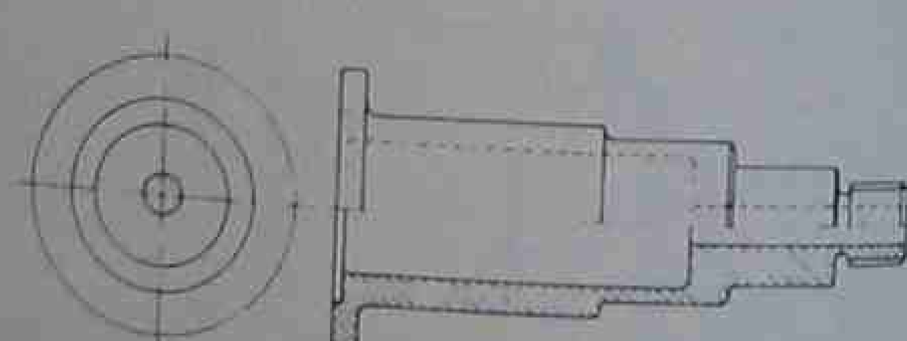


Figure 1.165 *Hollow shaft*

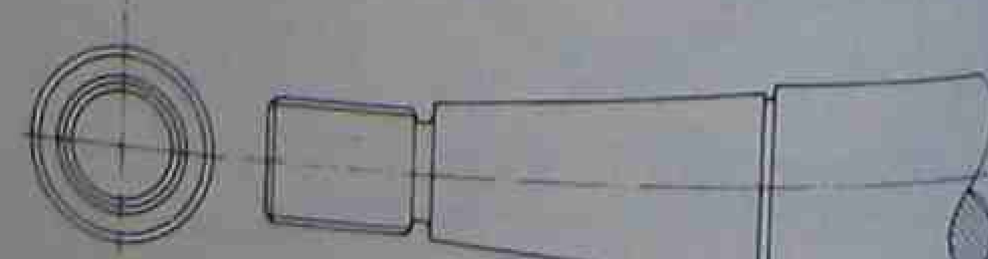


Figure 1.166 *Taper shaft*

Taper shaft Parts of shafts are sometimes tapered to suit components with a matching taper bore. A nut is used to give an extremely tight fit and a key provided to prevent relative rotation.

Screwed shaft Parts of a shaft may be screwed to take nuts which are used to attach components rigidly to the shaft. Hollow sections of shaft may also have internal screw threads.

The screw may be used for power transmission, as for example on the lead screw of a lathe, with a thread of square or buttress section.

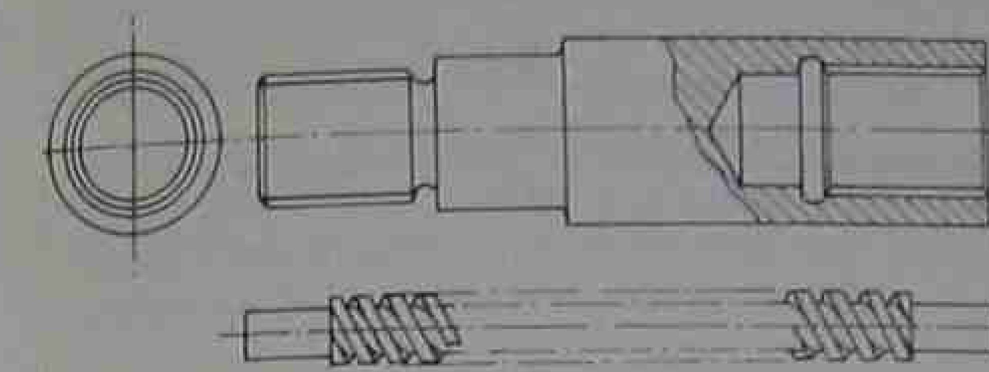


Figure 1.167 *Screwed shafts*

Crankshaft A crankshaft is a shaft having one or more levers, called *cranks* keyed to it or integral with it. The cranks have pins, known as *crankpins*, which are parallel to the shaft. The crank is sometimes fitted with a balance weight to offset the weight of the crank, crankpin and any part mounted on the crankpin.

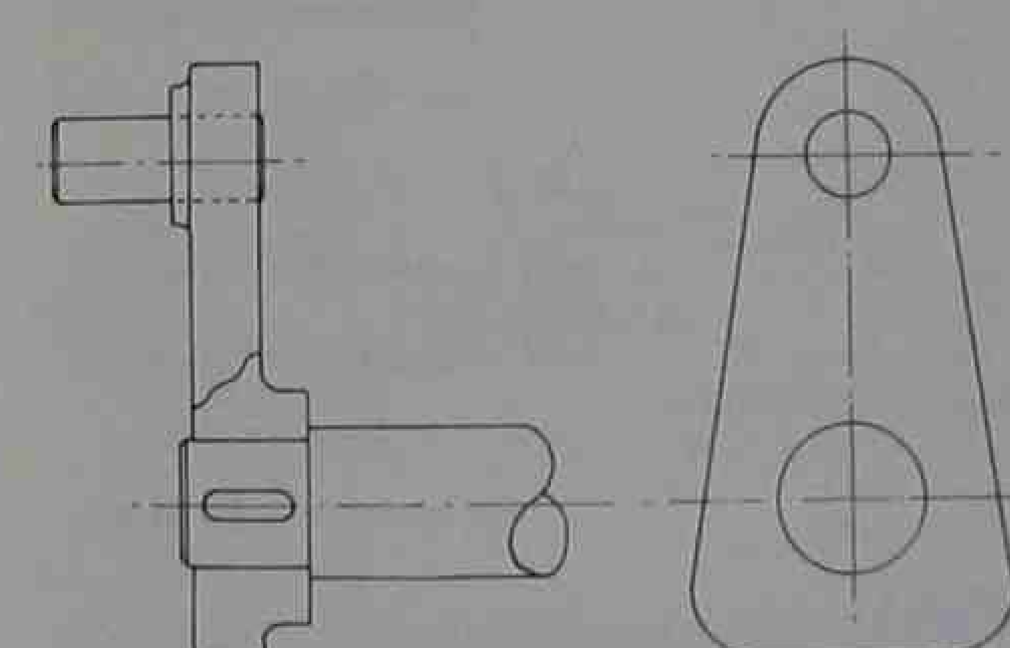


Figure 1.168 *Overhung crank*

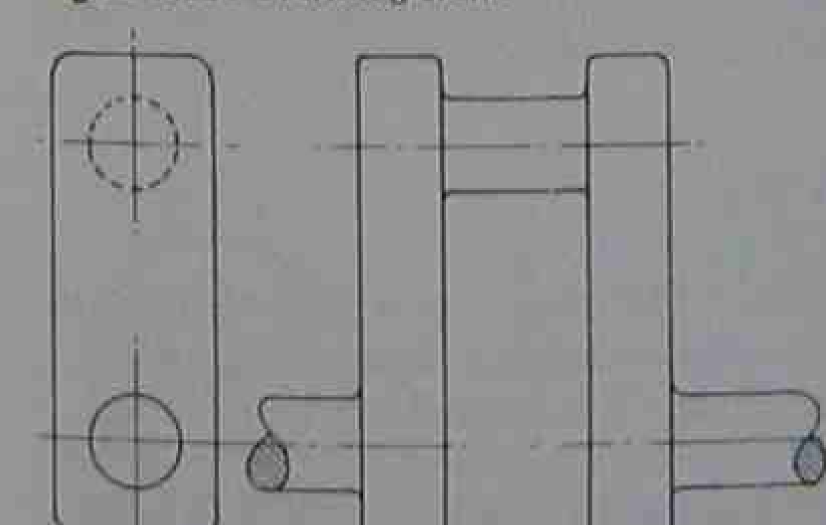


Figure 1.169 *Crank*

Camshaft A shaft on which a cam (or cams) is fitted. A typical example is the camshaft of an automobile engine.

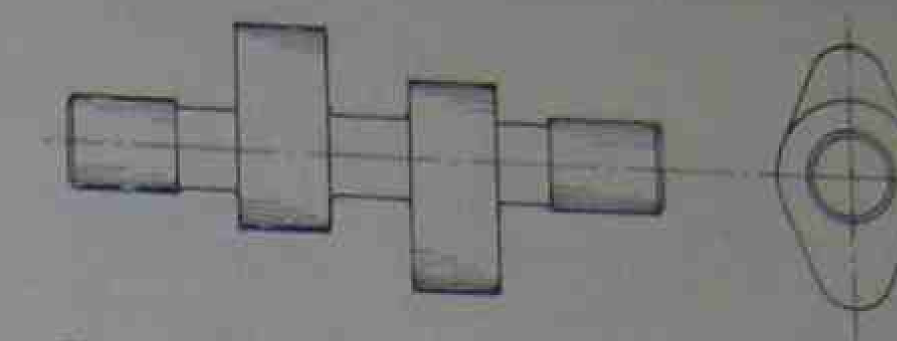


Figure 1.170 *Camshaft*

Grooves in shafts Grooves are often machined circumferentially in shafts for the location of *circlips* and for supplying oil to bearings. They may be of semi-circular or rectangular section.

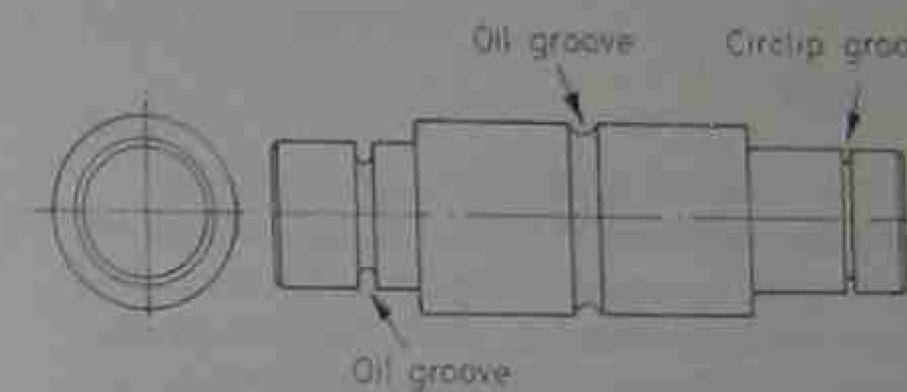


Figure 1.171 *Grooves in shaft*

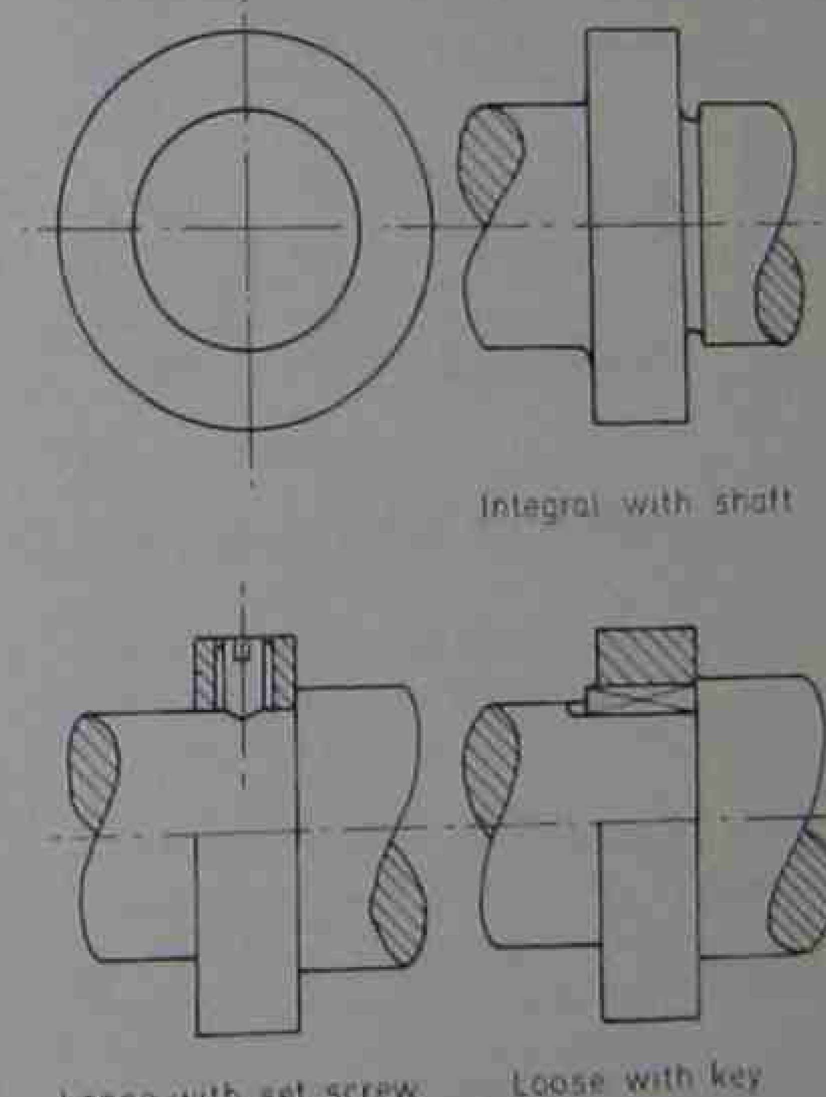


Figure 1.172 *Shaft collars*

Shaft collars A collar is a short section of increased diameter (on a shaft) used to locate parts mounted on the shaft, or to take an axial load from a thrust bearing.

Oilways Holes are drilled in a shaft to supply oil to bearings supporting it. A typical example is an automobile crankshaft in which oil is fed through a system of oilways to the main and connecting-rod bearings.

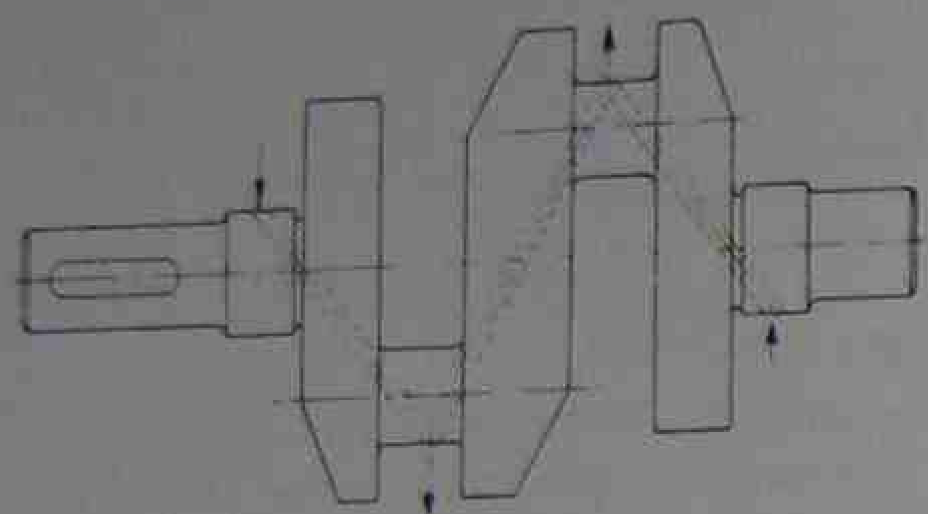


Figure 1.173 *Oilways in crankshaft*

SPLINES

Splines consist of a number of longitudinal ribs equally spaced around the circumference of a shaft which engage in corresponding grooves in the hub of a part to be mounted on the shaft. The form of the splines may be rectangular or triangular (usually referred to as serritions), or involute as in the case of gear teeth. The hub is often made a sliding fit on the shaft where axial movement is required.

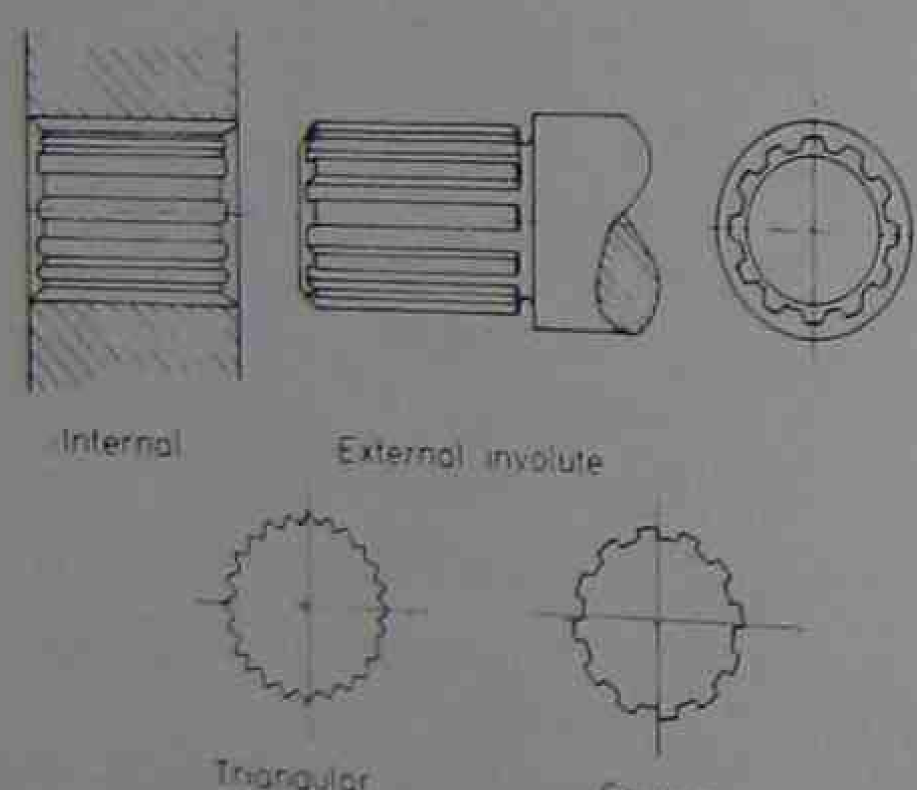


Figure 1.174 *Splines*

KEYS AND KEYWAYS IN SHAFTS

A key is used to prevent a machine part from moving relative to another part in a given direction. On shafts they prevent the rotation of a part relative to the shaft but may allow sliding along it. The key must be strong enough to transmit the shaft torque.

Materials Because they are subject to severe stresses, keys are made of high tensile steel or steel alloys.

Applications Keys are used for couplings, pulleys, gears, clutchplates, flywheels, etc. There are three classes of fit: 'free', where the hub slides easily; 'close', where an accurate fit is required; 'normal', where a minimum of fitting is necessary, as in cases of mass production.

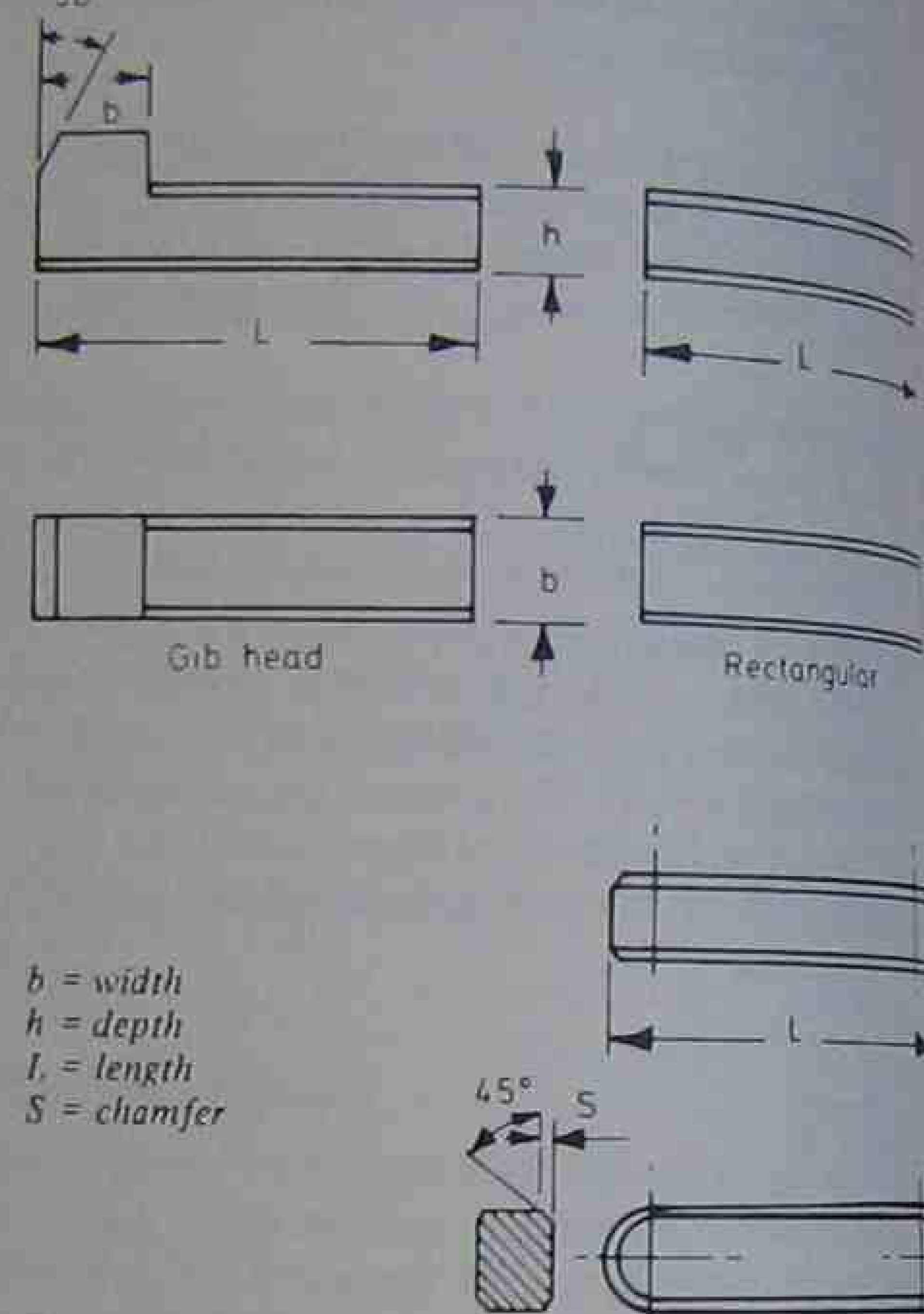


Figure 1.175 *Shaft keys*

KEYWAY

A keyway is a slot in a shaft into which a key is fitted, and it is usually produced by milling.

TYPES OF KEY

Rectangular key This consists of a piece of rectangular-section bar with chamfered edges having a slight taper to assist fitting. It is located in a longitudinal rectangular slot, or keyway, which has been milled in the shaft to a depth of half that of the key. A corresponding keyway is machined in the bore of the part to be keyed to the shaft.

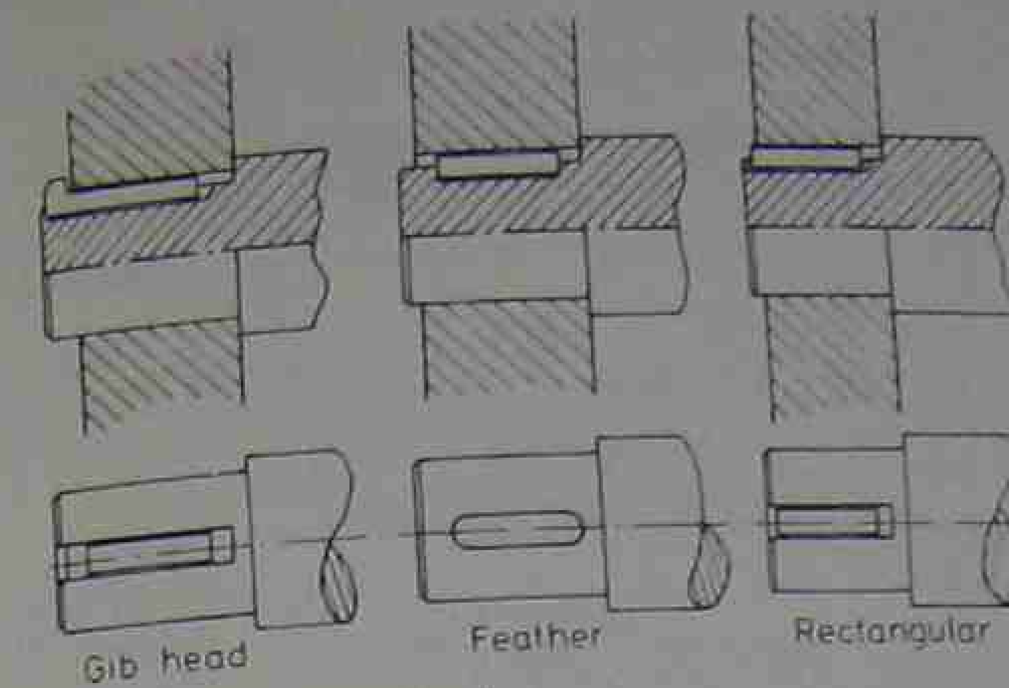


Figure 1.176 *Key applications*

Feather key A rectangular key with radiused ends fitting into a keyway of the same shape in the shaft which is end milled and closed at each end. The keyway in the hub is open at one or both ends.

Gib head key The gib head key is used at the end of a shaft and is tapered to give a rigid fixing when hammered into place. A head is provided to assist removal of the key.

Woodruff key This is in the form of a circular segment which fits into a slot of the same shape. It is often used on taper shafts and can be tilted for easy assembly.

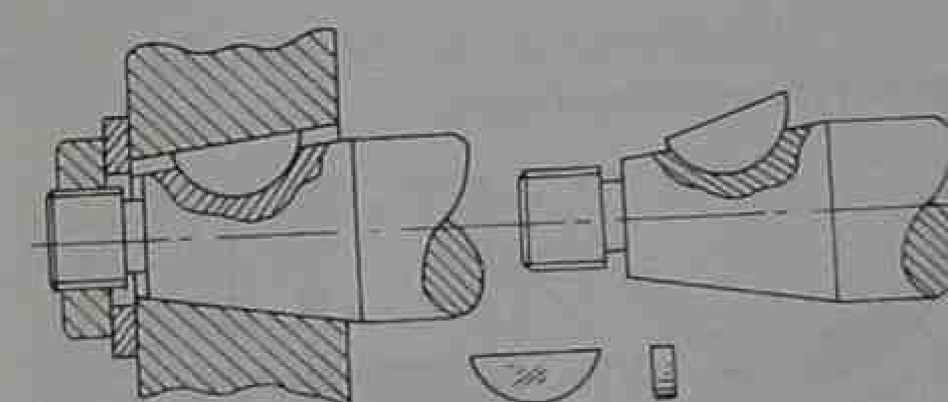


Figure 1.177 *Woodruff key*

Saddle key For a saddle key a rectangular slot is required in the hub but only a 'flat' is necessary on the shaft. It is intended for light duty only.

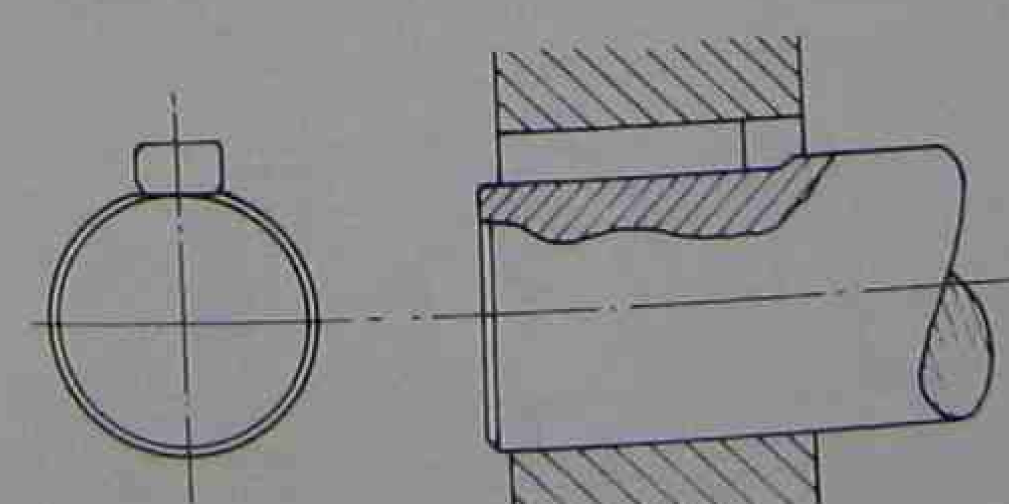


Figure 1.178 *Saddle key*

Round key This type provides a cheap method of keying requiring only a hole to be drilled (parallel to the axis of the shaft) which is half in the hub and half in the shaft. The key is a piece of stock bright bar or a taper pin.

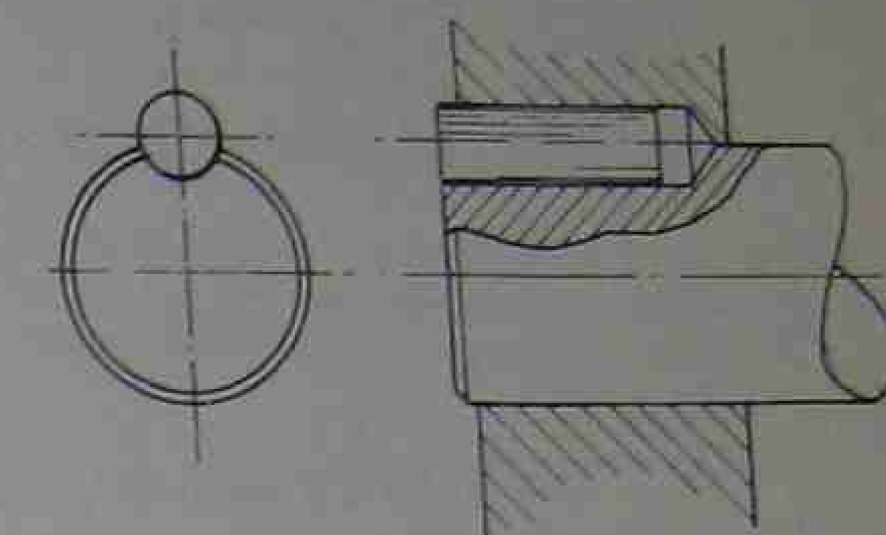


Figure 1.179 *Round key*

RODS

Rods are bars, usually circular in cross-section, which transmit tensile or compressive forces, or limited rotary movement. Rods in tension are known as *tie bars* or *tie rods*.

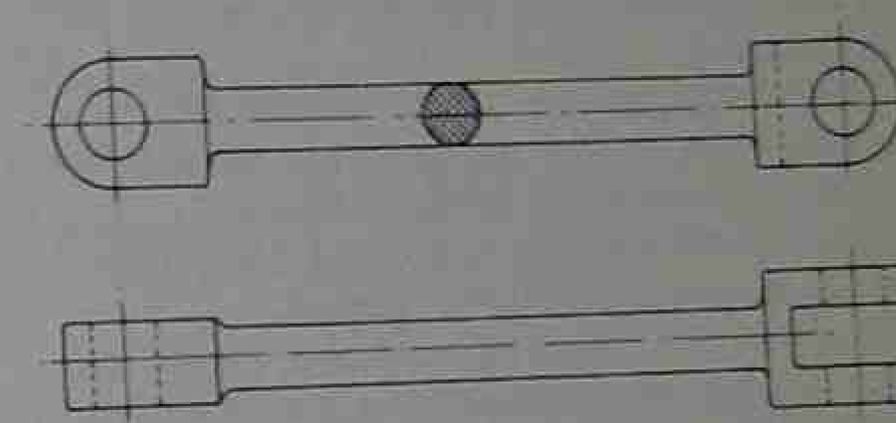


Figure 1.180 *Tie rod*

PIN JOINT, KNUCKLE JOINT

This is a hinged joint, connecting two or more rods, used as a tie for roof trusses.

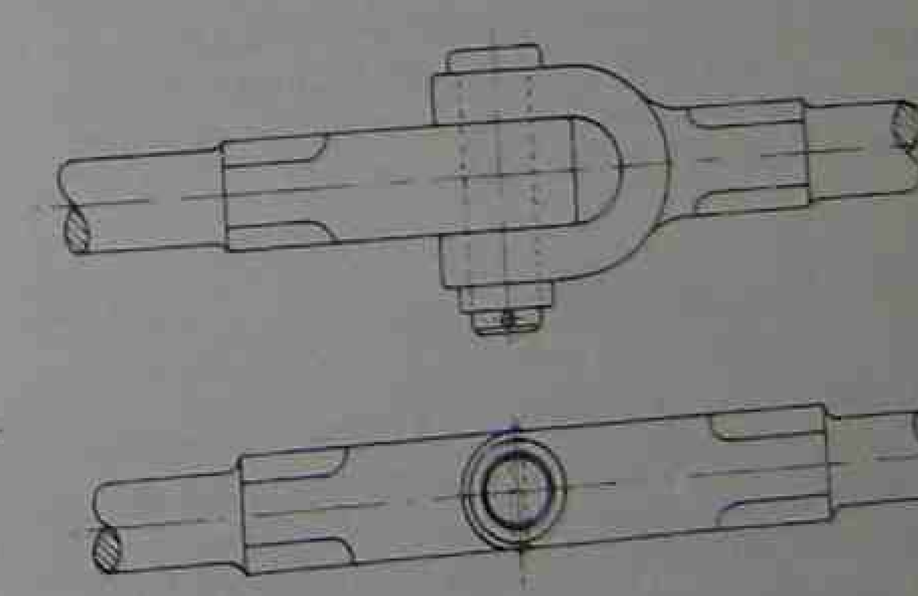


Figure 1.181 *Pin joint for rods*

COTTERED JOINT

A rigid joint connecting two rods in which a cotter pin is used.

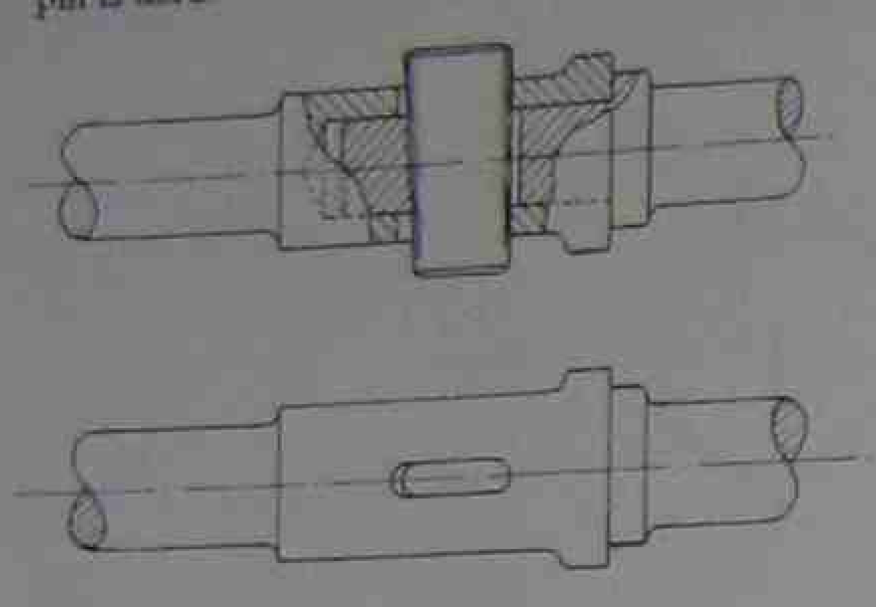


Figure 1.182 *Cottered joint for rods*

1.8 BEARINGS

Bearings are used in engineering to restrain or guide the movement of one machine part relative to another with the minimum of friction and wear. In most cases they are used between rotating parts, such as shafts, and machine frames, but they may also be used for linear motion.

Shaft bearings may be *journal bearings* (the term 'journal' refers to the shaft) which take radial forces, or *thrust bearings* which take axial forces.

A lubricant is required between the sliding surfaces, and this may be solid, liquid or gaseous.

There are two main classifications of journal and thrust bearings: *sliding contact*, e.g. a shaft running in a plain sleeve; *rolling contact*, e.g. ball and roller bearings.

PLAIN JOURNAL BEARING, SLEEVE BEARING

This comprises a *bush* or sleeve, made of brass, bronze, etc., held in a stationary member in which a shaft rotates. It is supplied with a suitable lubricant through holes and/or grooves. The gap between the shaft and the bush is known as the 'clearance'.

PEDESTAL BEARING

The bearing may be mounted in a support bolted to a flat surface. This is known as a pedestal and it may be a casting, forging or weldment. The pedestal is sometimes split at the centreline of the shaft and the bush made in two pieces called *brasses* or *shells*.

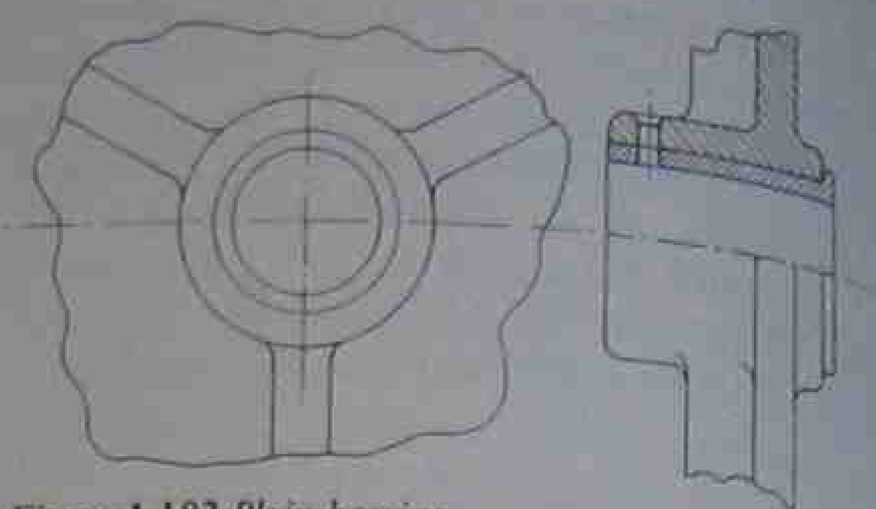


Figure 1.183 *Plain bearing*

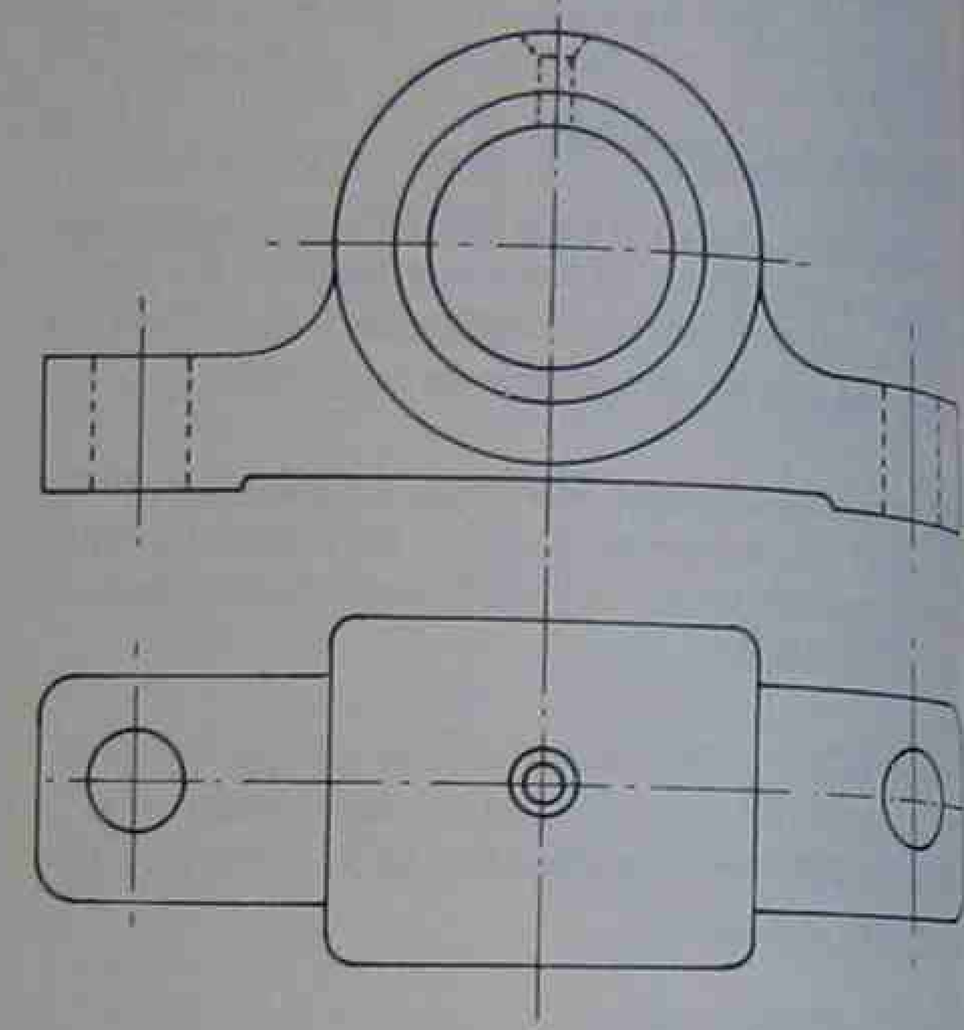


Figure 1.184 *Simple pedestal bearing*

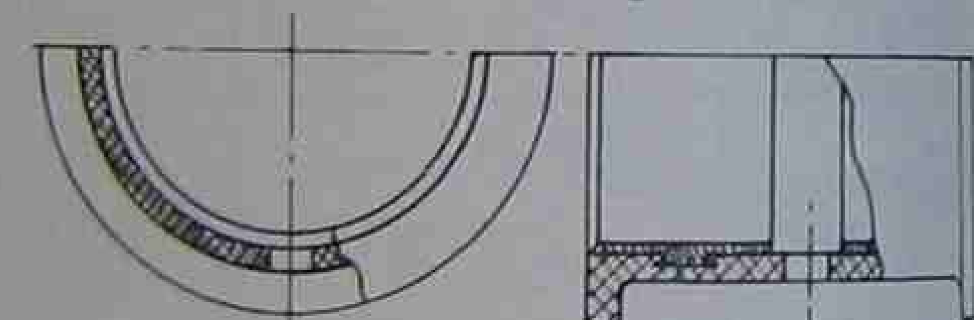


Figure 1.185 *Journal bearing shell*

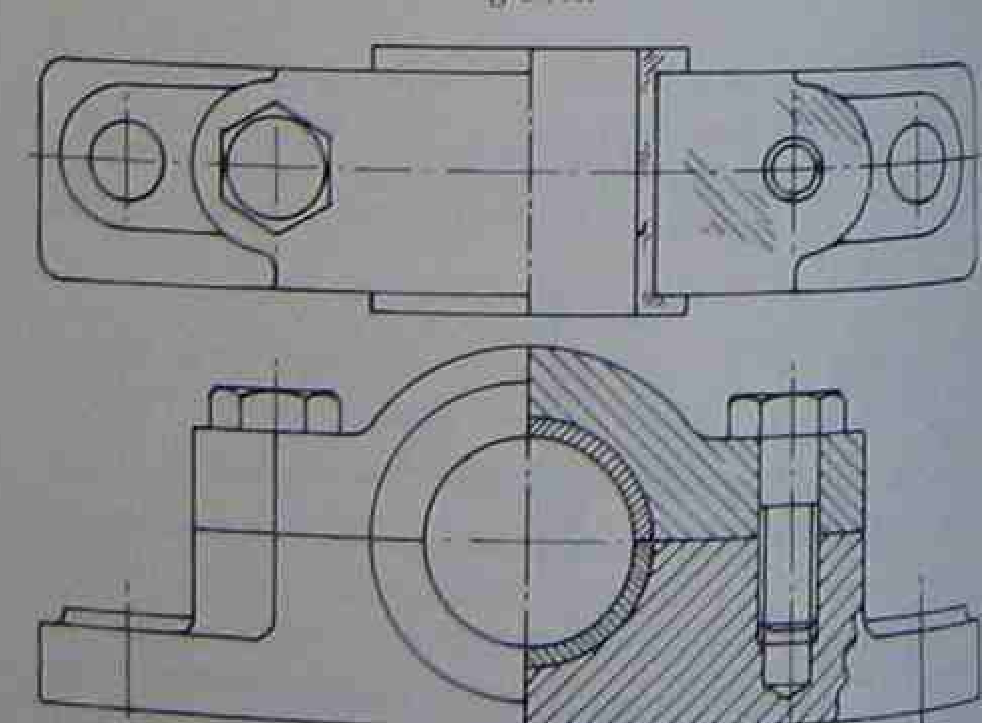


Figure 1.186 *Split or halved pedestal bearing*

RING OILER BEARING

A loose ring rolls on the shaft by friction and picks up oil from a bath. The oil flows from the shaft into the bearing.

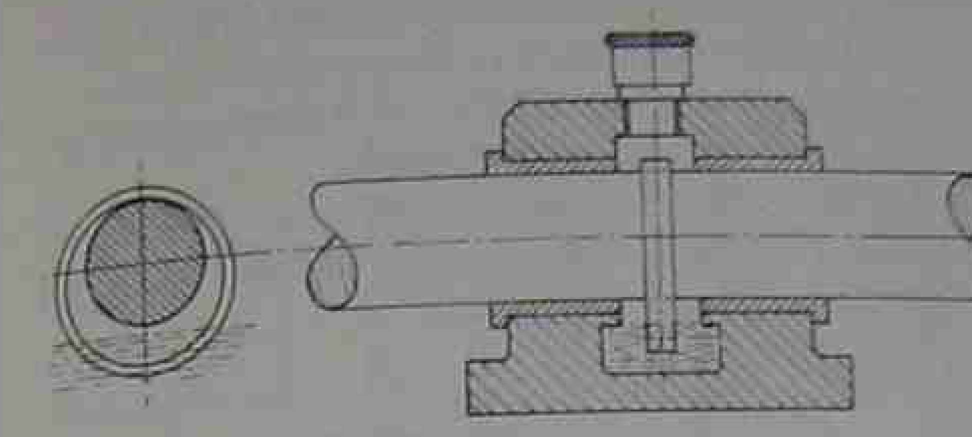


Figure 1.187 *Ring oiler bearing*

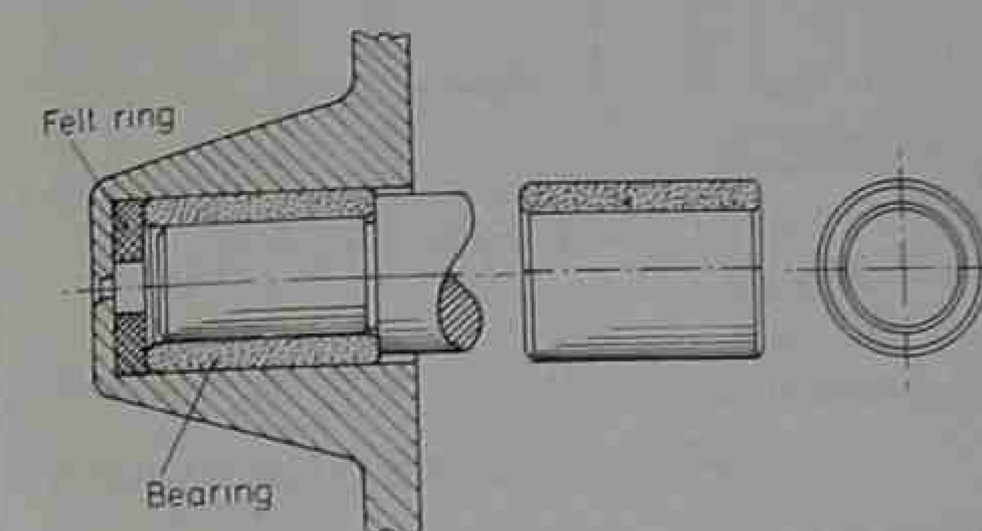


Figure 1.188 *Automobile dynamo porous bearing*

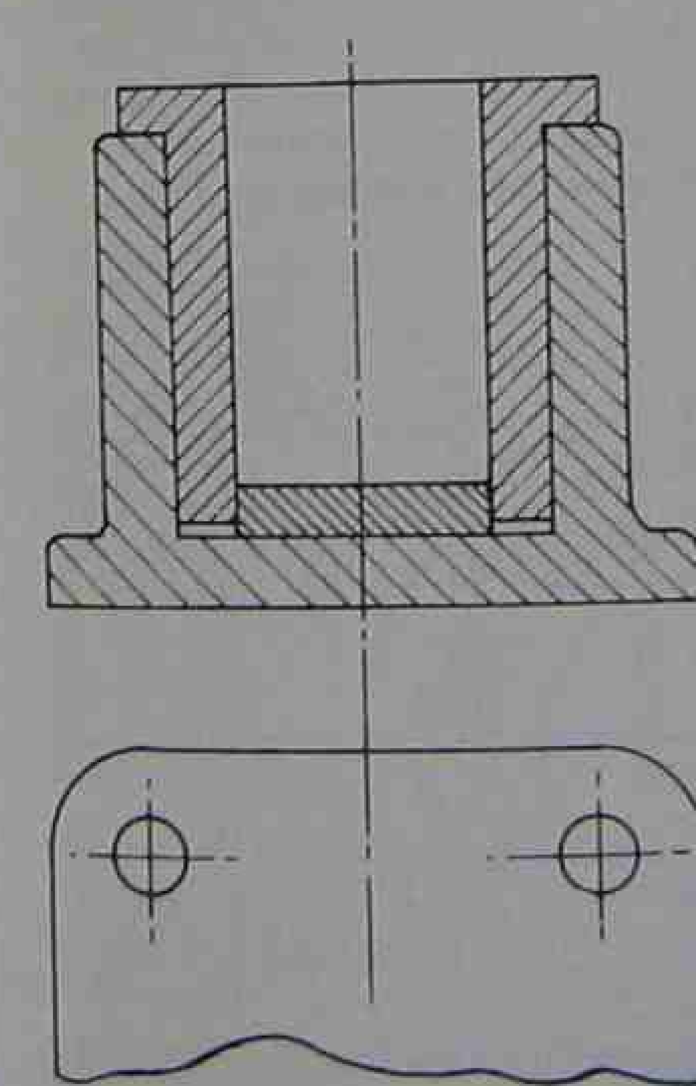


Figure 1.189 *Footstep thrust bearing*

POROUS BEARING, SELF-OILING BEARING

This is a bush made of porous metal containing oil or grease, often requiring no further lubrication during its life.

FOOTSTEP THRUST BEARING

A bearing used for the lower end of a vertical shaft. A bush takes the horizontal load and a hardened steel pad takes the weight of the shaft or any thrust.

PLAIN THRUST BEARING

A flat ring of brass, bronze, etc. taking the thrust on a shaft via a collar.

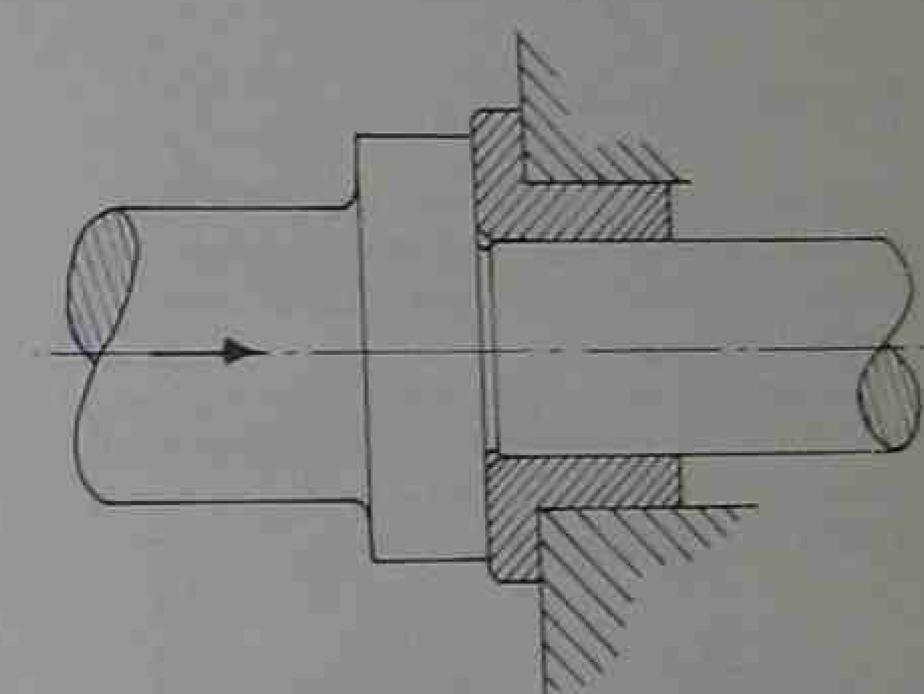


Figure 1.190 *Plain thrust bearing*

MICHELL THRUST BEARING (PIVOTED SEGMENT BEARING)

A thrust bearing in which a thrust collar on a shaft bears on segmental 'pads' faced with bearing metal. These tilt to provide a high-pressure oil 'wedge'. This bearing is used for marine propeller shafts and other large, high-duty applications (Figure 1.192).

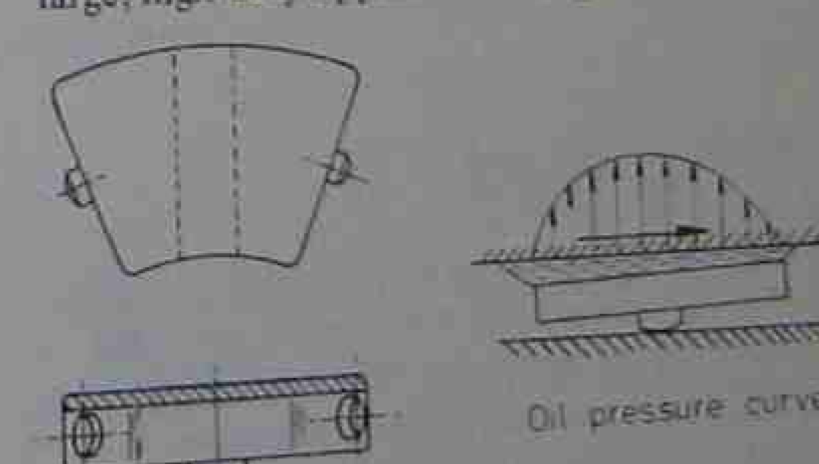


Figure 1.191 *Thrust pad*

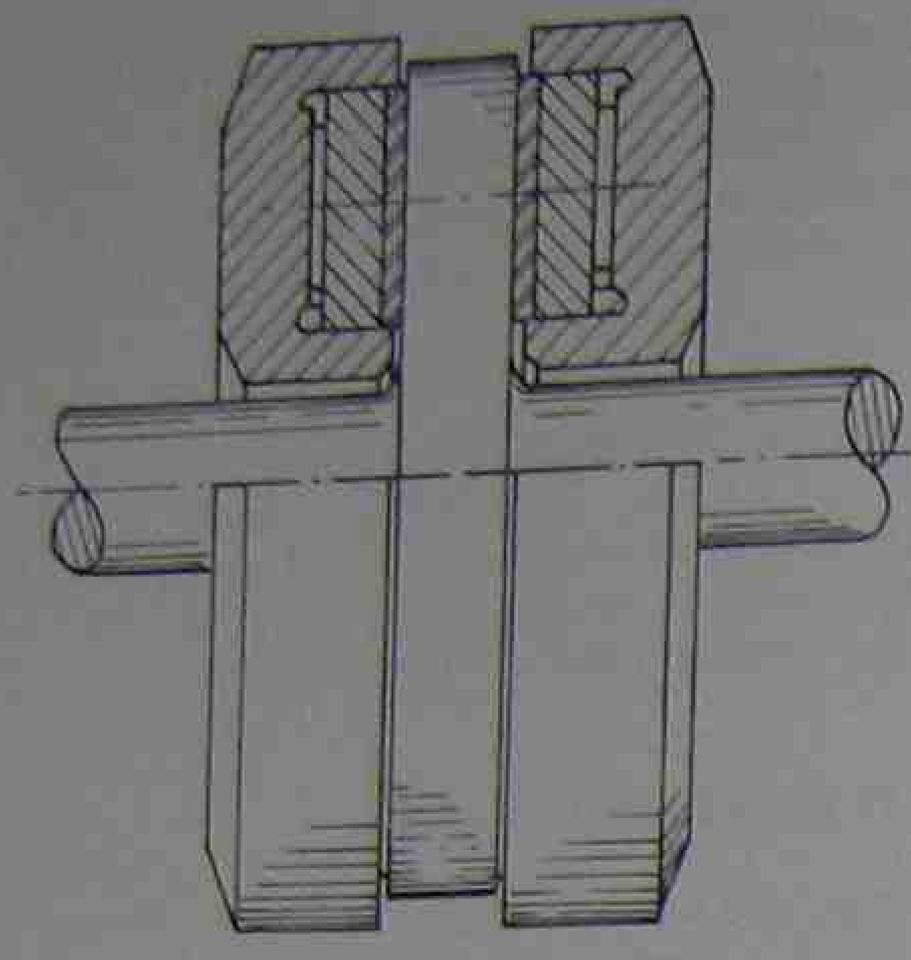


Figure 1.192 *Double Michell thrust bearing*

GAS BEARINGS

Gases, such as air, nitrogen, etc., may be used as lubricants in both journal and thrust bearings. The gas is introduced through radial holes in a sleeve surrounding the shaft.

Gas bearings are used for low friction and high speed applications and where contamination by oils is undesirable. A typical example is the air bearings in a dentist's drill driven by an air turbine.

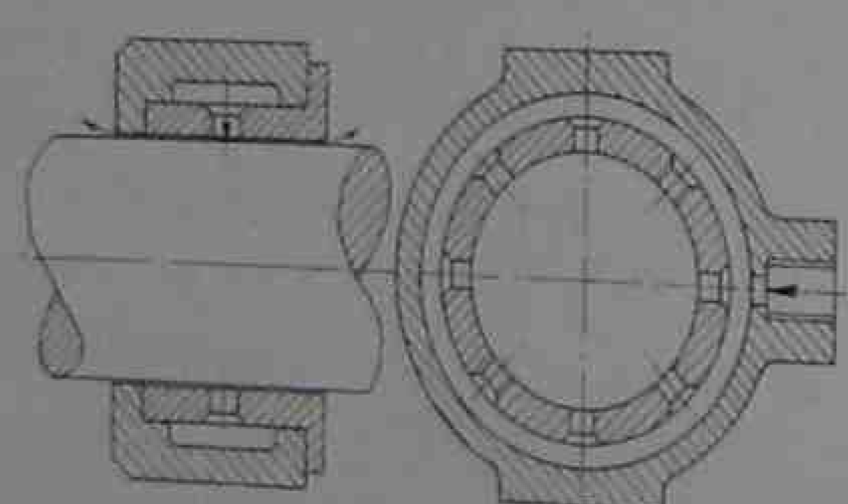


Figure 1.193 *Air bearing*

BEARING MATERIALS

Plain bearings use brass, bronze and gunmental bushes. Linings for shell bearings and thrust pads use white metal (an alloy of tin, antimony and copper) and lead alloys which also include tin, antimony, copper, etc. Also used are various plastic materials such as nylon, Tufnol and PTFE.

ROLLING CONTACT BEARINGS

This is a general name given to a wide range of bearings in which hardened steel balls or rollers run between housings (or *'races'*) on the shaft and the fixed member.

They are often termed *anti-friction bearings* because of their extremely low frictional resistance, especially at low speeds. The small clearances involved permit accurate alignment of shafts.

Ball and roller bearings, although more complicated than plain bearings, may compare favourably in price when made in large quantities.

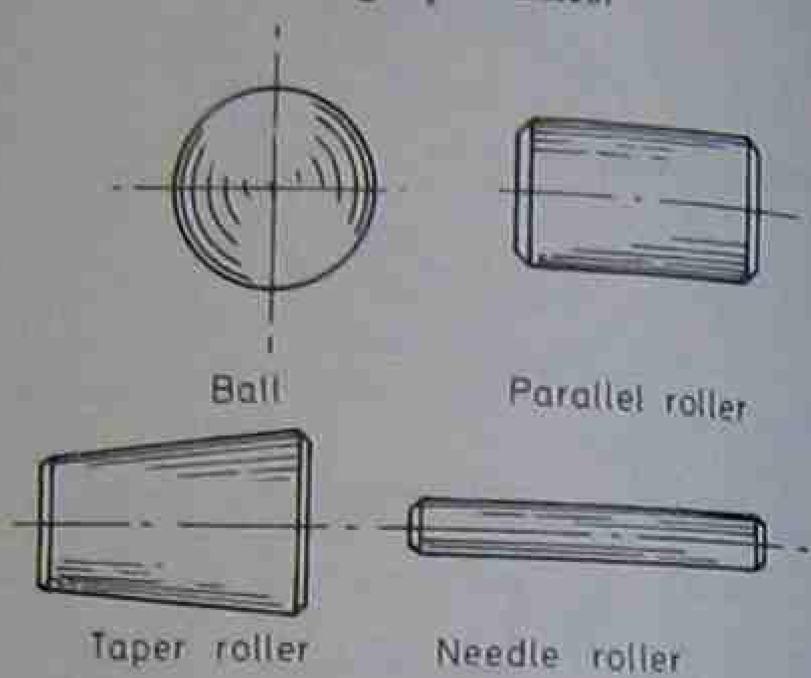


Figure 1.194 *Types of bearings*

SINGLE ROW BALL BEARINGS

The bearing consists of inner and outer grooved rings of hardened steel between which run hardened steel balls held and spaced in a retainer or cage. This bearing will take a certain amount of axial thrust but for higher thrusts *angular contact bearings* are more suitable.

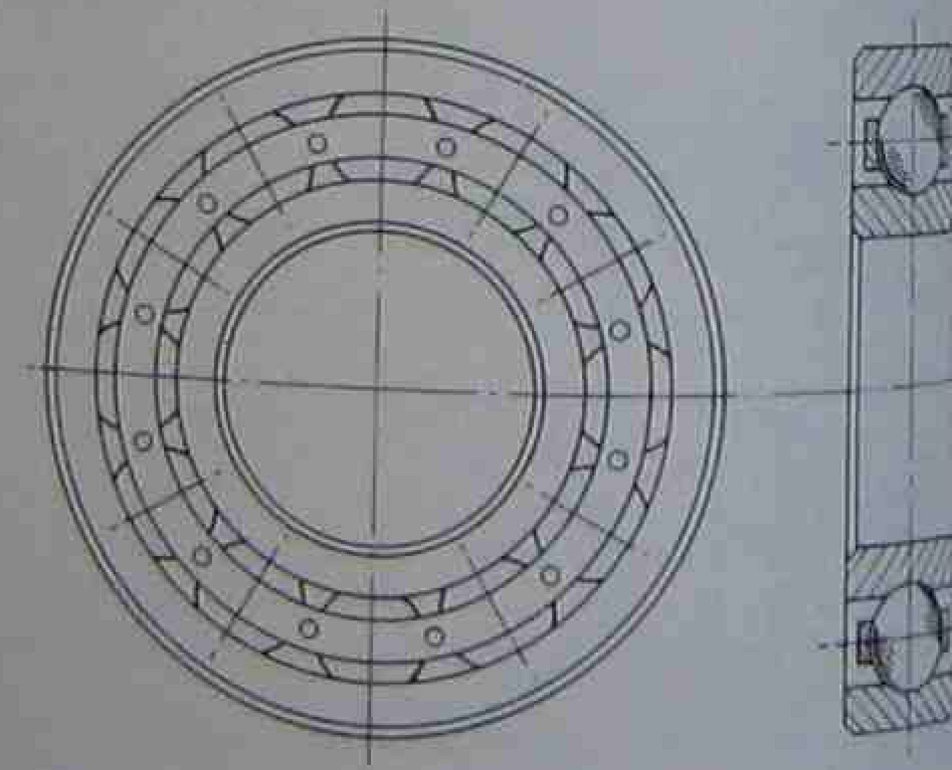


Figure 1.195 *Standard ball bearing*

Self-aligning bearings are used when there is doubt about the alignment of the housing.

Sealed bearings packed with grease require no further lubrication during the life of the bearing.

Deep groove bearings are designed for higher thrusts.

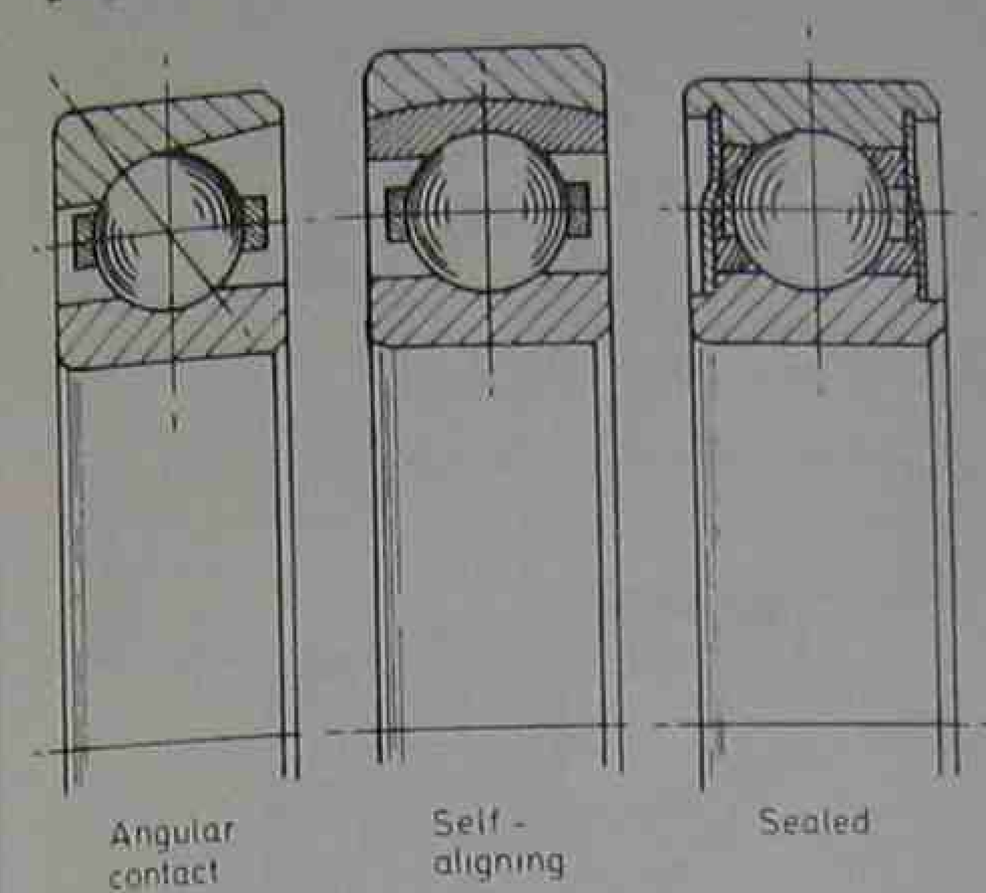


Figure 1.196 *Ball bearings*

DOUBLE ROW BALL BEARINGS

Double row ball bearings designed to take radial and/or thrust loads will withstand higher loads without increasing the outer diameter.

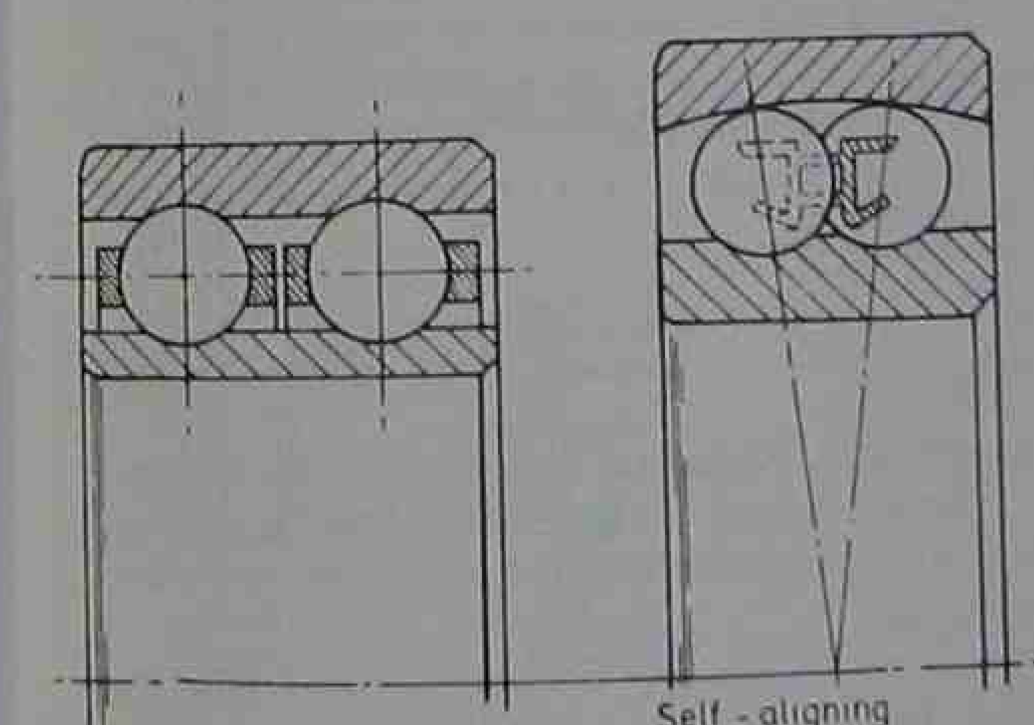


Figure 1.197 *Double row ball bearings*

ROLLER BEARINGS

Roller bearings will withstand much greater radial loads than ball bearings but parallel rollers will not take thrust loads. These bearings may be designed so that either the inner or outer ring may be detached.

If the rollers are tapered and arranged in the form of a cone, a *taper roller bearing* capable of carrying very high radial and thrust loads is obtained. These are often used in opposing pairs.

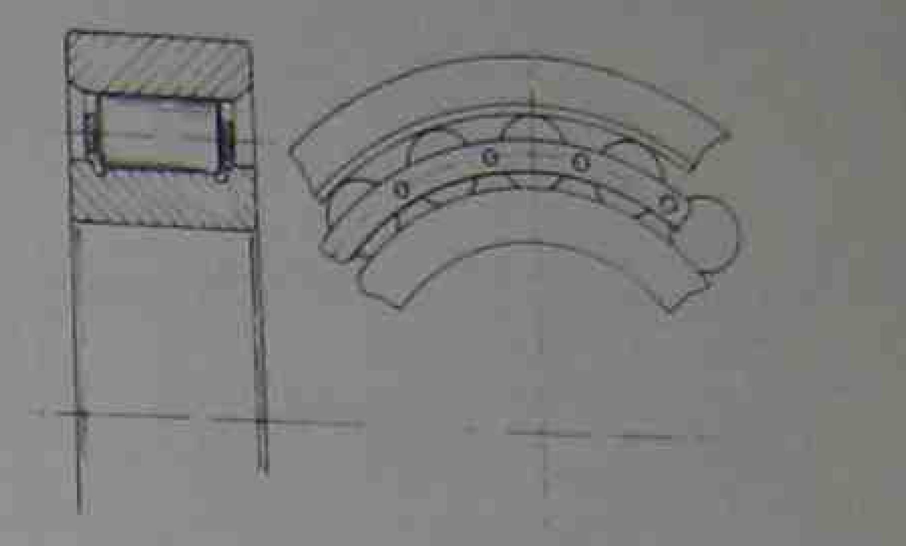


Figure 1.198 *Roller bearing*

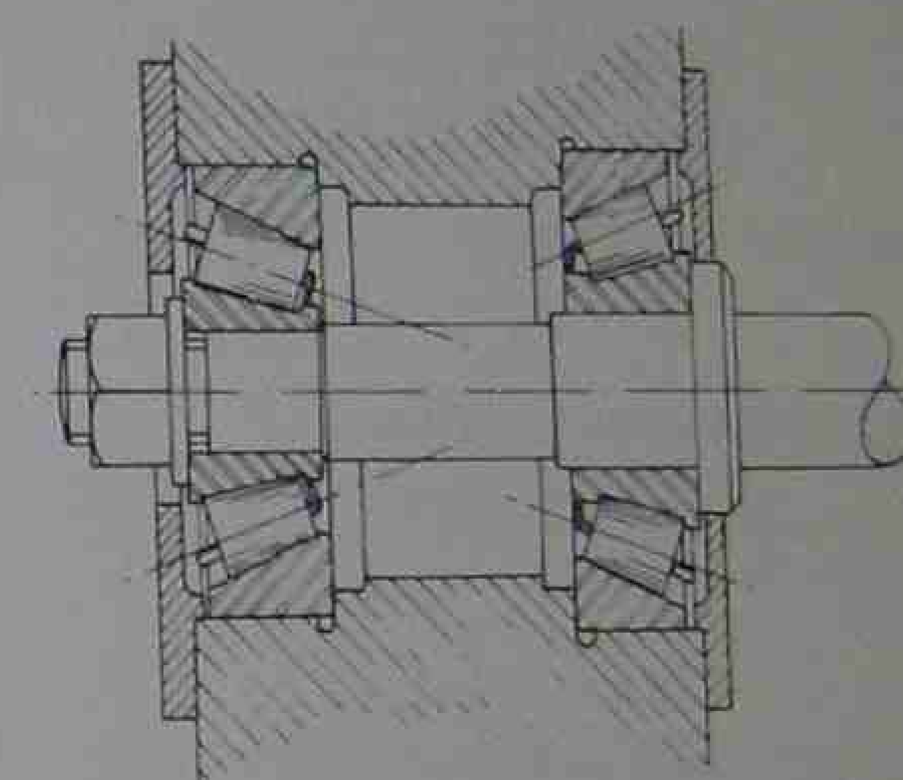


Figure 1.199 *Opposed taper roller bearing*

NEEDLE ROLLER BEARINGS

Where space is restricted very small diameter needle rollers are used. The rollers run on the surface of the shaft which must be hardened and ground. Needle rollers are used in automobile gearboxes and backaxles.

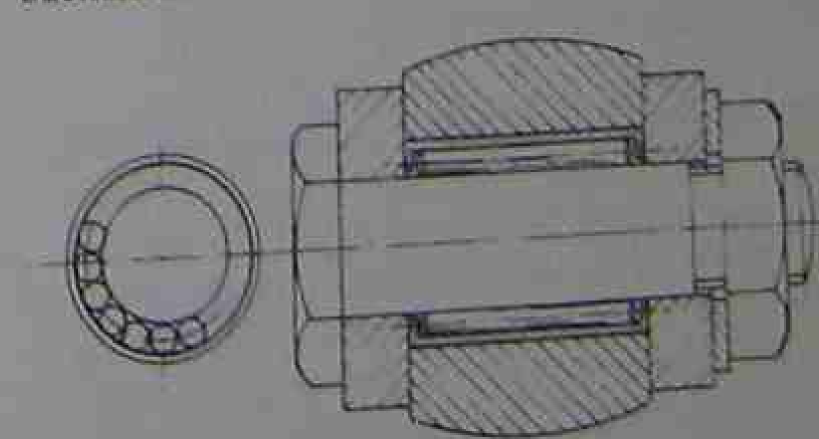


Figure 1.200 *Needle roller bearing*

BALL THRUST BEARINGS

The balls are mounted in a cage and run between the faces of two annular rings, and these may be used to take the weight of a vertical shaft in a footstep bearing.

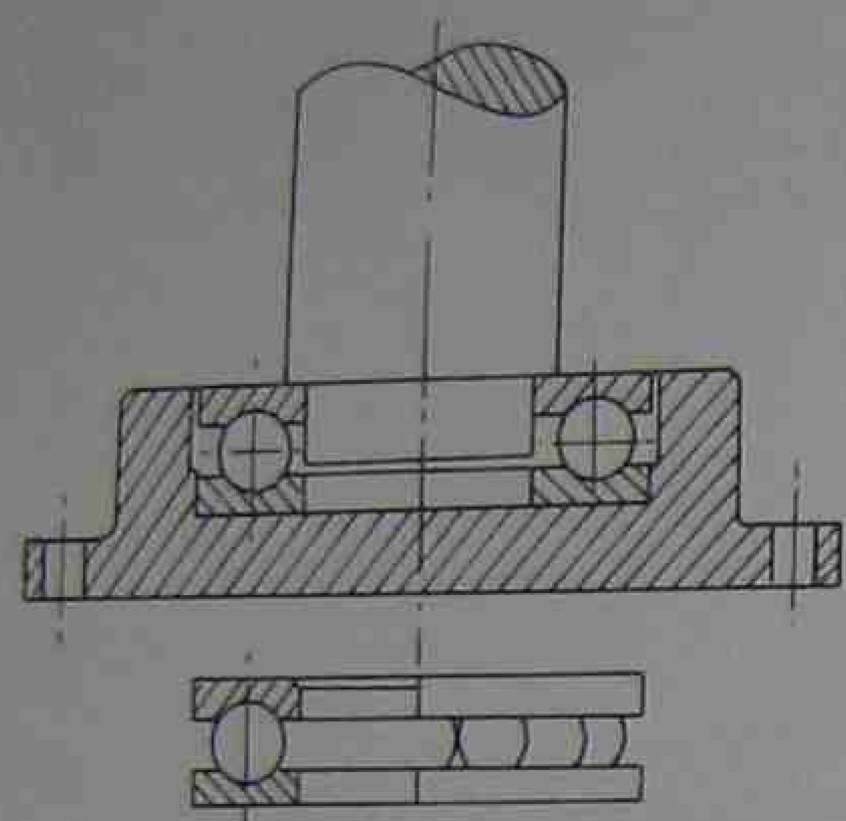


Figure 1.201 *Ball thrust bearing*

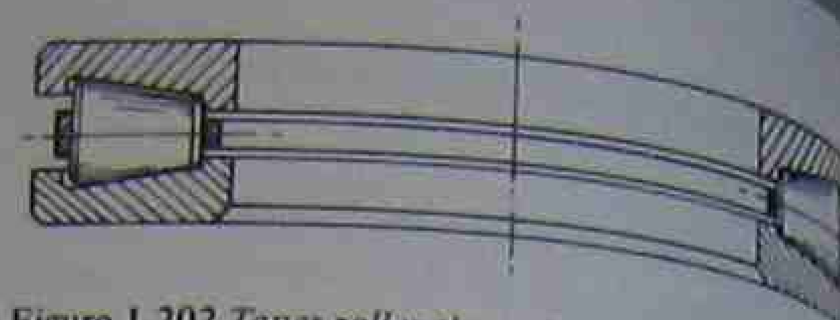


Figure 1.202 *Taper roller thrust bearing*

ROLLER THRUST BEARINGS

These are similar to ball thrust bearings except that tapered rollers are used.

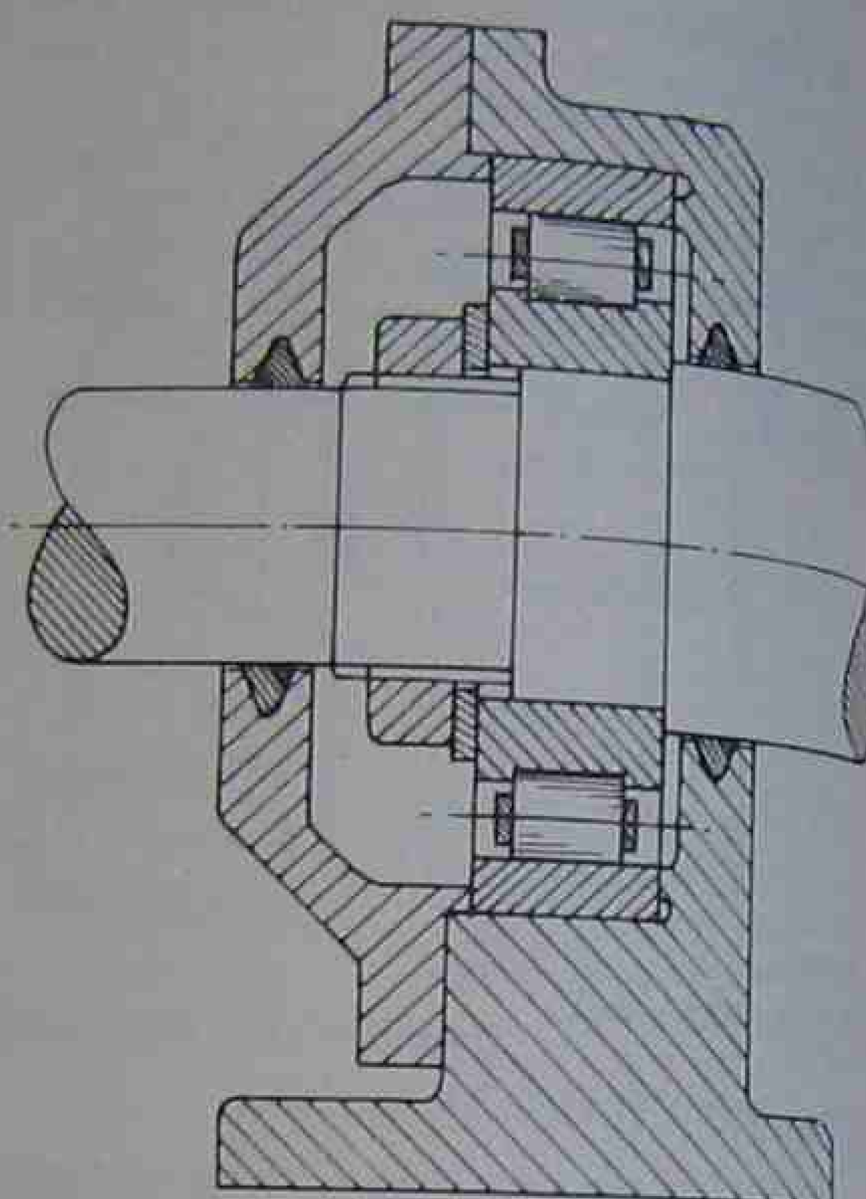


Figure 1.203 *Roller bearing in sealed housing*

BEARING HOUSINGS

The fixed part into which a bearing fits is known as a 'housing', and the outer ring of the bearing must be a good fit in this housing to ensure correct operation. As some form of closure is necessary to prevent loss of lubricant and keep dirt out, oil seals are often included.

BEARING LUBRICATORS

Plain bearings are usually lubricated by oil, and this is pressure fed for high duty cases. Ball and roller bearings may be lubricated by grease or oil. Grease may be fed to the bearing by means of a grease nipple and gun or by a screw-down greaser, or the bearing may be packed with grease and sealed for life. Oil may be fed from a sight feed drop oiler or by splash feed, oil bath, oil jet or oil mist.

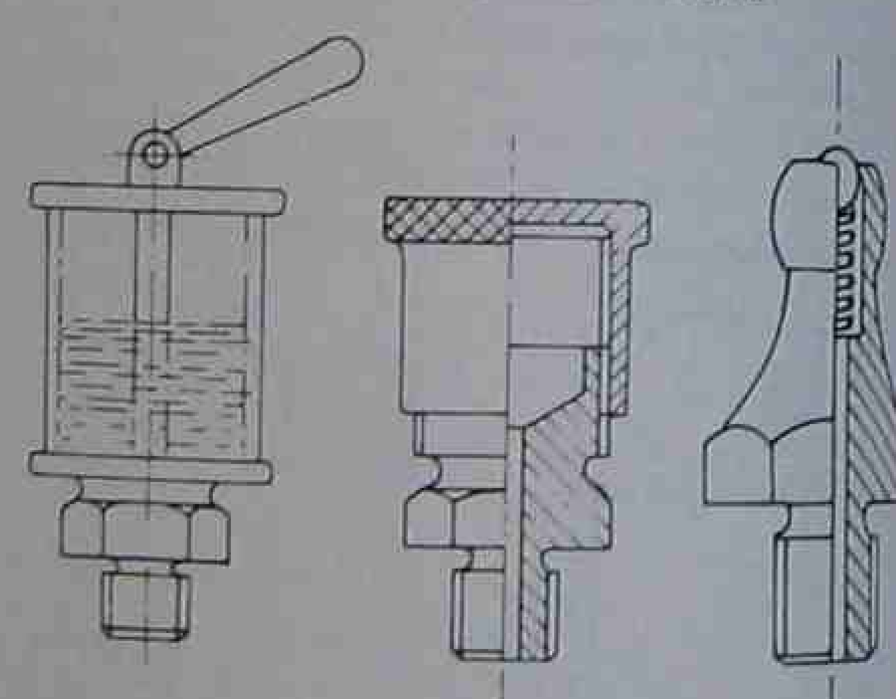


Figure 1.204 *Lubricators*

LUBRICANTS

Most bearings are lubricated with petroleum oils and grease made from those oils, but vegetable oils are used in special cases. Air and other gases may be used

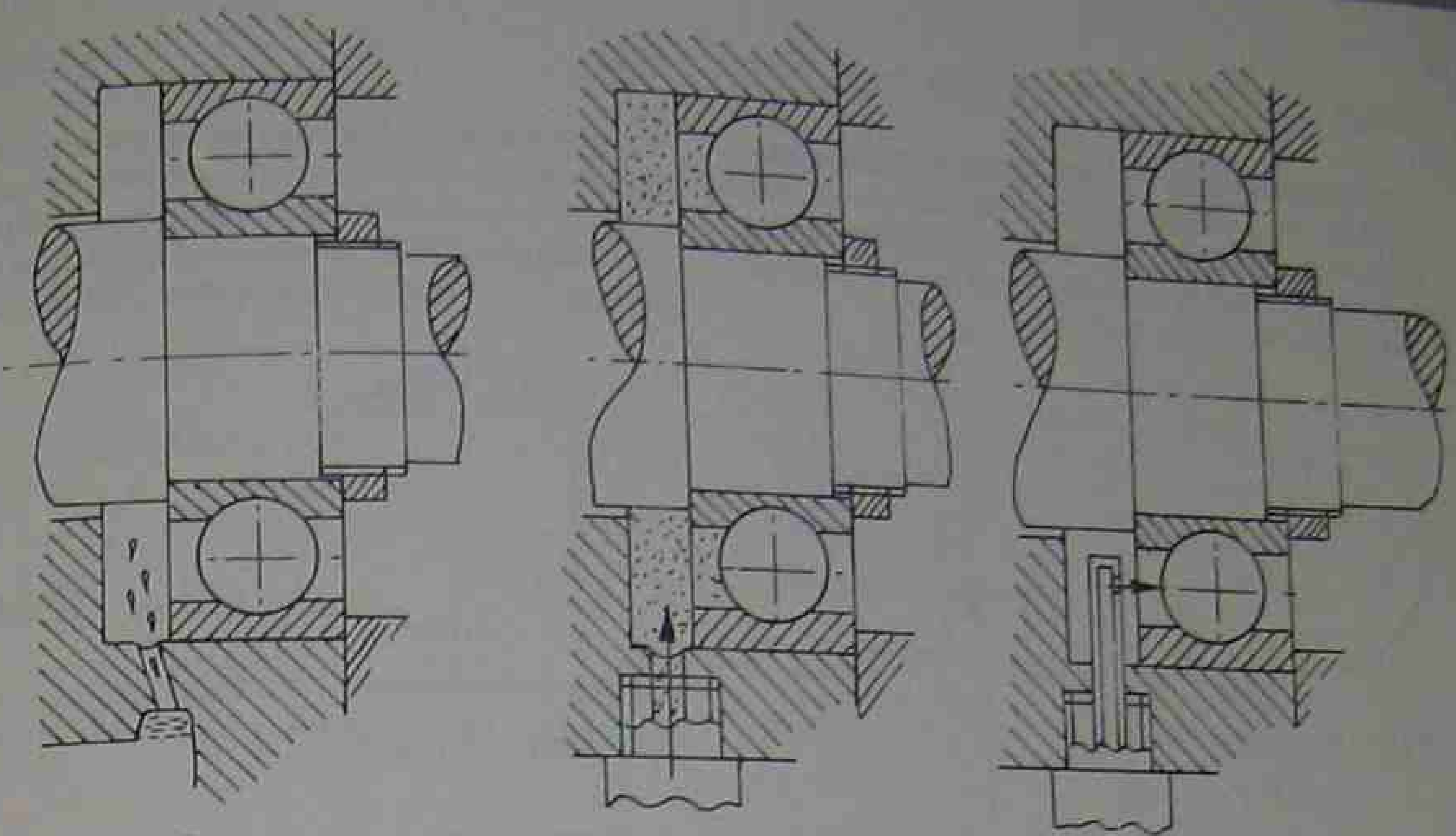


Figure 1.205 *Bearing lubrication*

1.9 ENGINEERING DESIGN FEATURES

In addition to the many standard components encountered in engineering there is a large number of features which are integral with them. These often have unusual and sometimes rather curious names with the same name being given to several items of quite different appearance. The following describe in alphabetical order the more common features to be found.

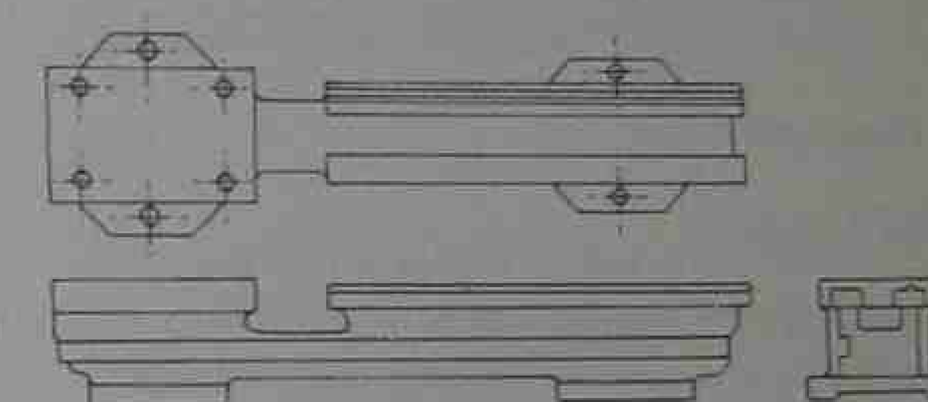


Figure 1.206 *Bedplate (for lathe)*

BEDPLATE, BED

A bedplate is a welded or bolted fabrication, or a casting, usually attached to the floor, on which machines, engines and machine tools etc. are mounted.

BODY

This is a term used for the main part of an assembly to which other smaller parts are attached. For example, the main part of a valve in which the spindle, seating, etc. are located.

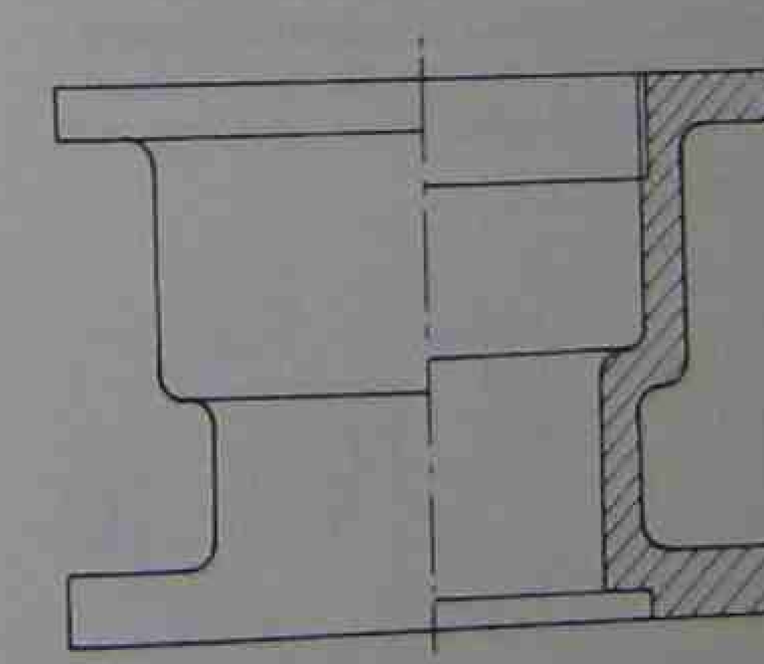


Figure 1.207 *Body (for safety valve)*

BOX (*see* Casing)

BOSS

A boss is a short, usually cylindrical, projection (on a machine part) which often has a central hole in which a pin, shaft or bolt, etc. is located. It is used on castings to give a flat seating for a bolt head or nut, or on levers to give a longer bearing.

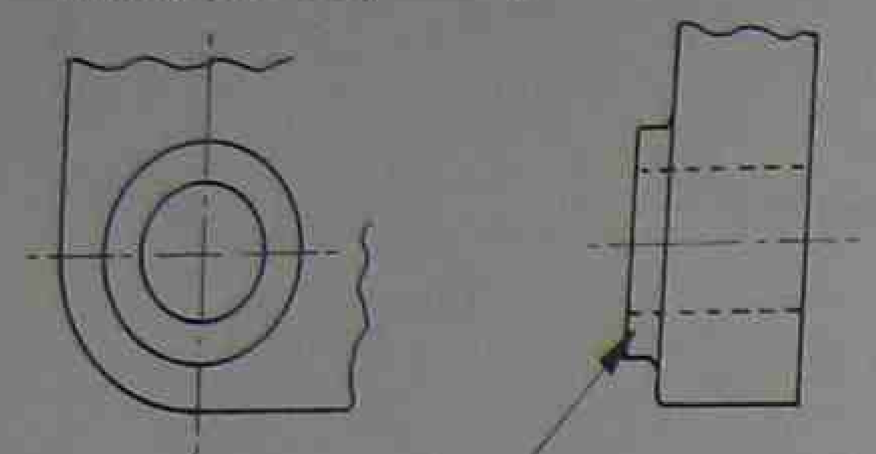


Figure 1.208 *Boss*

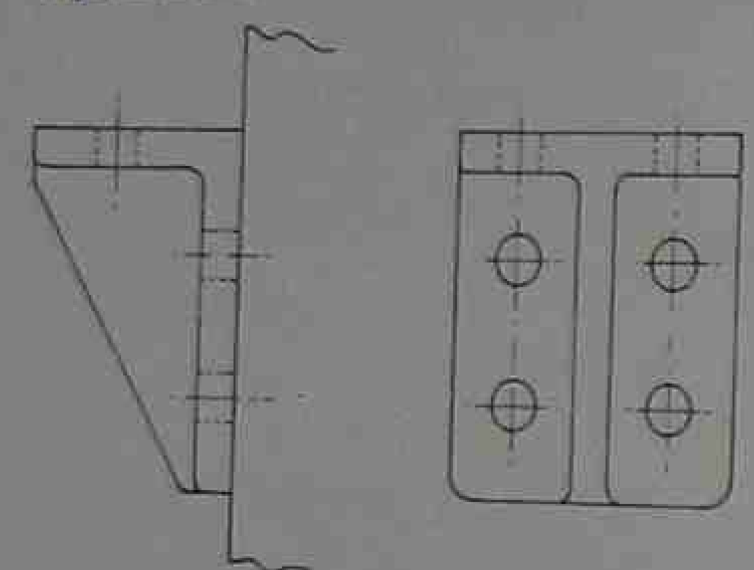


Figure 1.209 *Bracket*

BRACKET

A bracket is a support for machine parts often attached to a wall or vertical surface by bolts, rivets or welding.

BUSH

A bush is a cylindrical sleeve which fits into a hole and acts as a bearing for rotating or reciprocating shafts or rods. The bush is often made of material different from the part into which it is fitted, e.g. a brass bush in a steel part (*see* Sleeve).

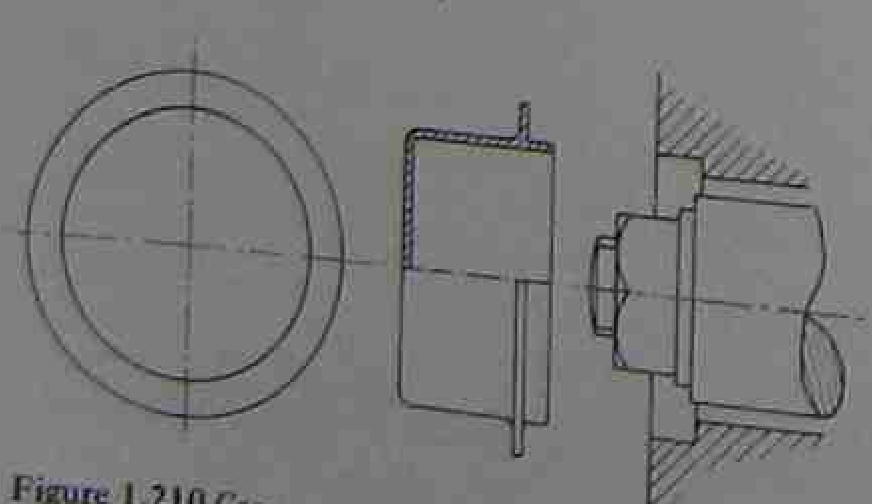


Figure 1.210 *Cap*

CAP

A cap is a metal or plastic cover bolted or pressed into position and used to retain oil or grease, or to protect, and sometimes to improve, the appearance of a machine part. The removable half of a journal bearing housing is also referred to as a cap.

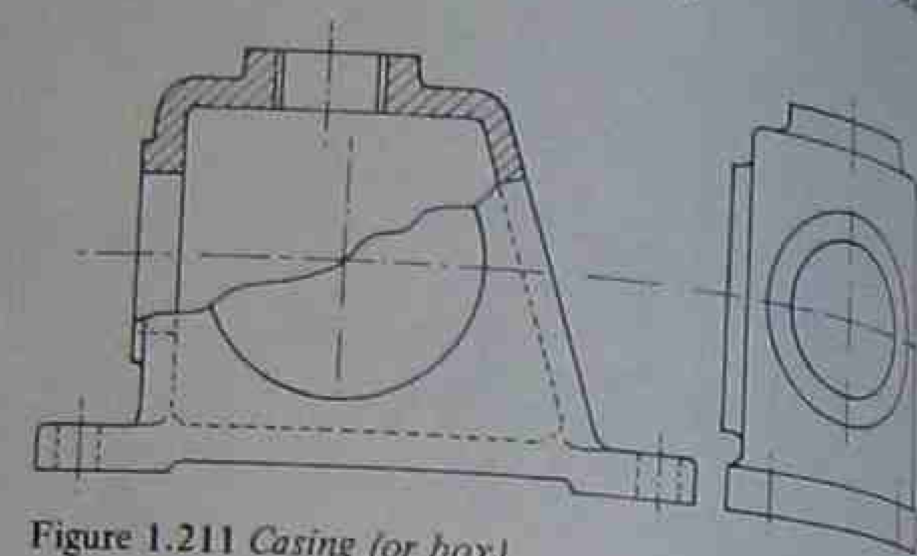


Figure 1.211 *Casing (or box)*

CASING (BOX)

A casting, forging or weldment in the form of a box or container which holds machine parts, such as those of a pump or turbine, is called a 'casing'.

CASTING

A casting is a metal or plastic article formed by pouring liquefied material into a suitably-shaped mould and then allowing it to solidify before being removed from the mould.

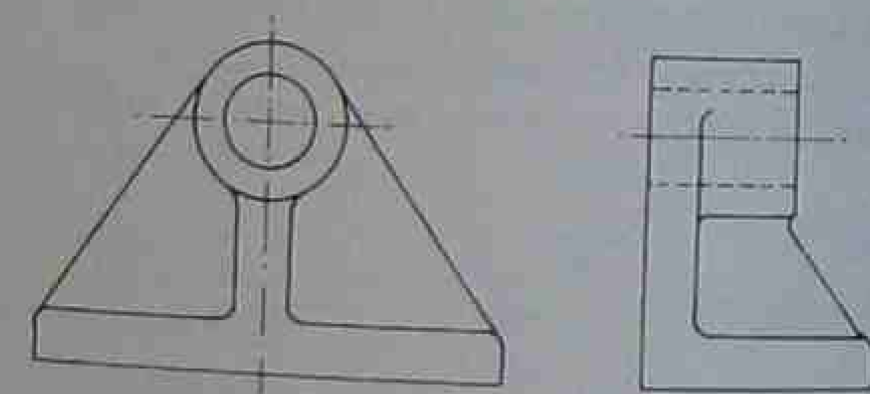


Figure 1.212 *Pedestal casting*

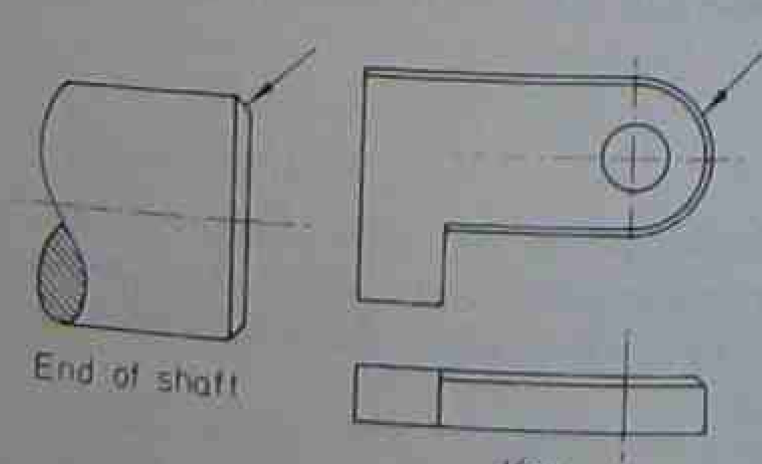


Figure 1.213 *Chamfer*

CHAMFER

The profile obtained by bevelling an edge or corner. It is usual to chamfer the corners produced by sudden changes in diameter on a shaft. This reduces the possibility of damage and eases the fitting of parts such as bearings, seals, collars, etc.

COLUMN (*see* Pillar)

COVER (COVERPLATE)

A cover is a plate, usually flat, which is bolted, riveted, welded or pressed into position to cover a cavity in a casing, etc.

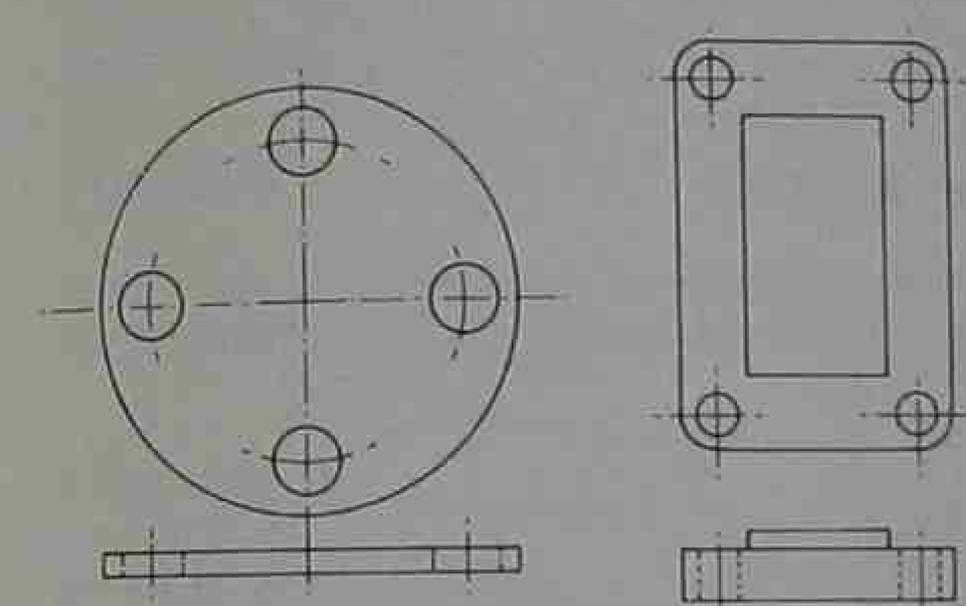


Figure 1.214 *Covers (coverplates)*

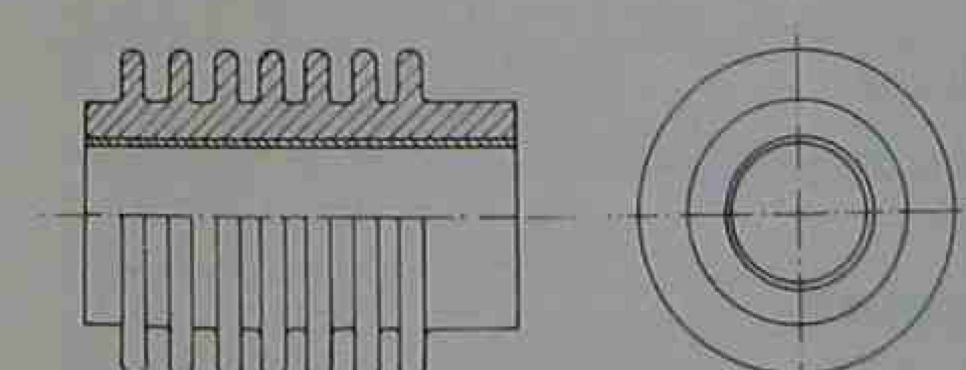


Figure 1.215 *Cylinder (with cooling fins and liner)*

CYLINDER

A cylinder is a circular tube in which a piston or plunger slides, e.g. as in pumps, engines or hydraulic rams. It is also a tubular container for liquids and gases.

DISTANCE PIECE

Any piece of material which is placed between two parts to maintain them at a fixed distance apart is termed a *distance piece*. It may be a plate or, as in the case of a shaft, a cylindrical sleeve (*see* Spacer).

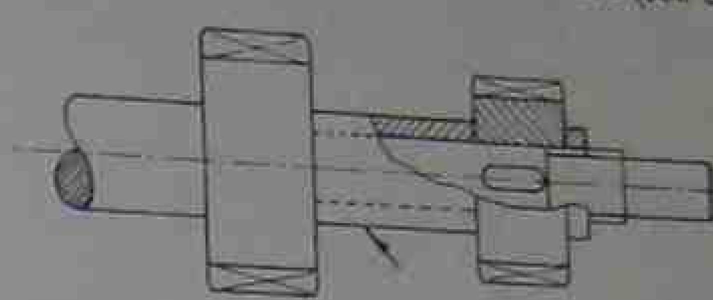


Figure 1.216 *Distance piece*

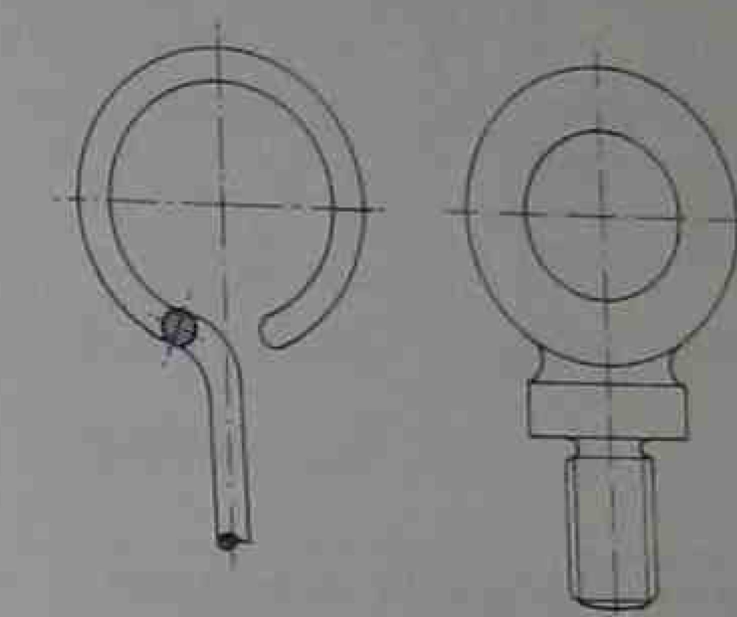


Figure 1.217 *Eye (wire and eye bolt)*

EYE

An eye is a loop formed at the end of a metal rod or wire by bending, or a ring which has been machined, cast, or forged on a component.

FACE

A face is a flat surface produced on a component by hand-filing or machining to which other parts may be attached, or on which they may slide, as opposed to the rough surface of a casting, forging etc.

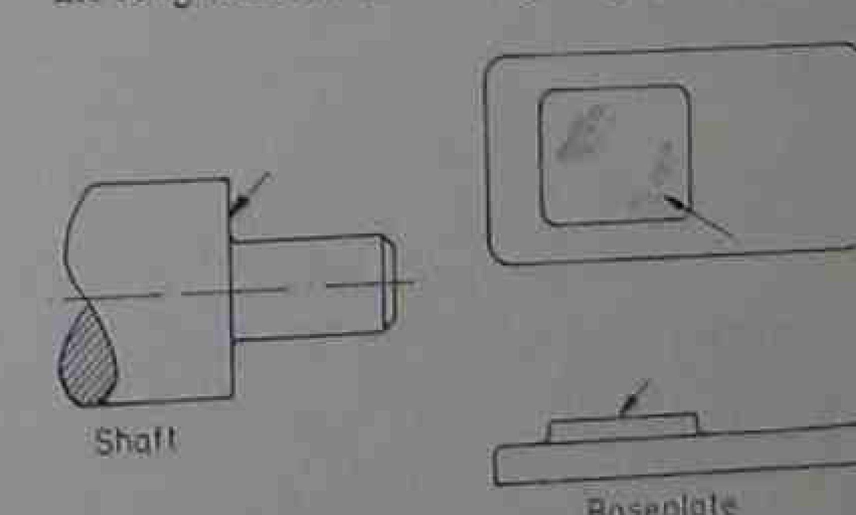


Figure 1.218 *Face*

FAUCET (RECESS)

A faucet is a circular recess into which a matching *spigot* fits, as in the case of a pipe joint.

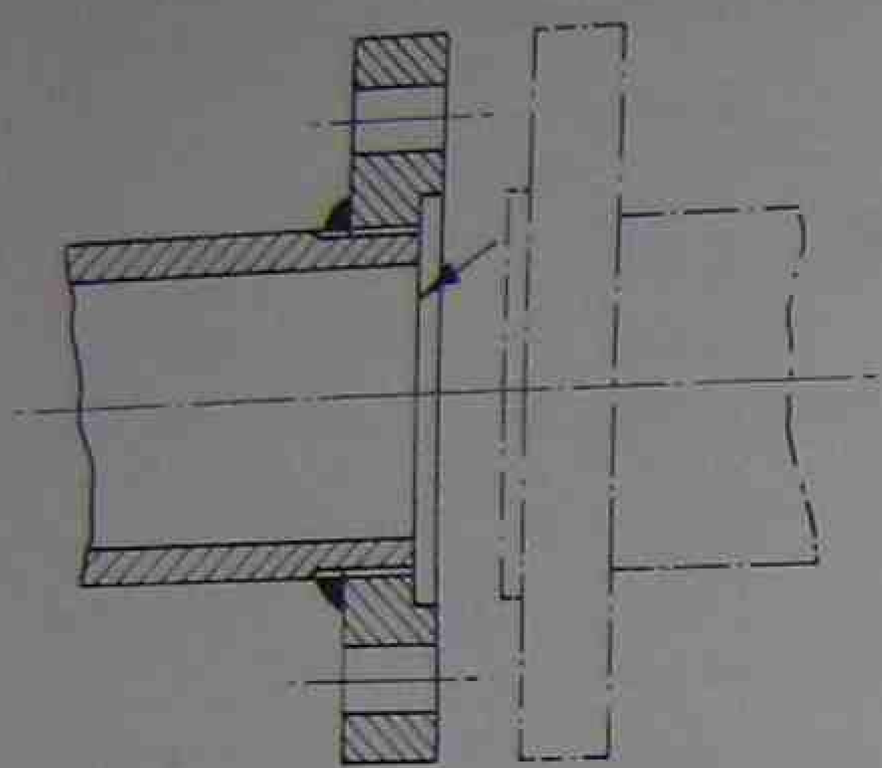


Figure 1.219 *Faucet (or recess)*

FILLET

This is a radius in a corner formed by two flat surfaces, and it is used to improve the quality of castings and the strength of machined parts (*see* Radius).

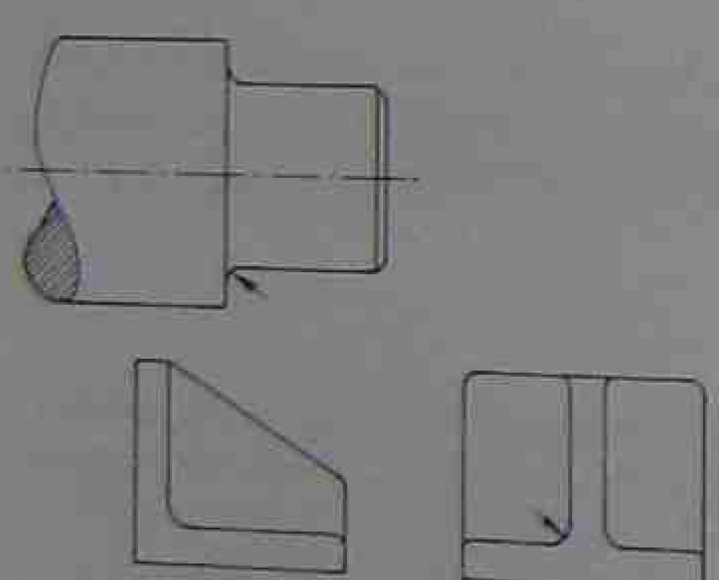


Figure 1.220 *Fillet*

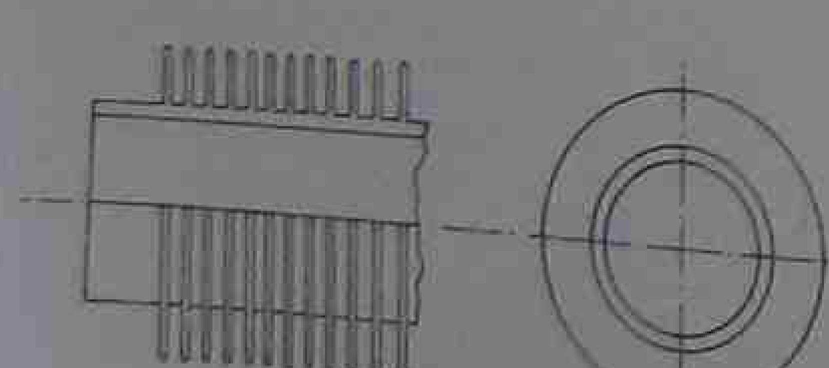


Figure 1.221 *Fins (on pipe)*

FIN

A fin is one of a number of thin ribs on the outer surface of engine cylinders, gear units, coolers, etc., used to dissipate heat to the surrounding air.

FLANGE

A flange is a projecting rim or lip, such as the rim of a wheel which runs on rails, the top and bottom parts of an I-section beam or channel section, the disk-like projection at the end of a pipe, or a bolted shaft coupling etc.

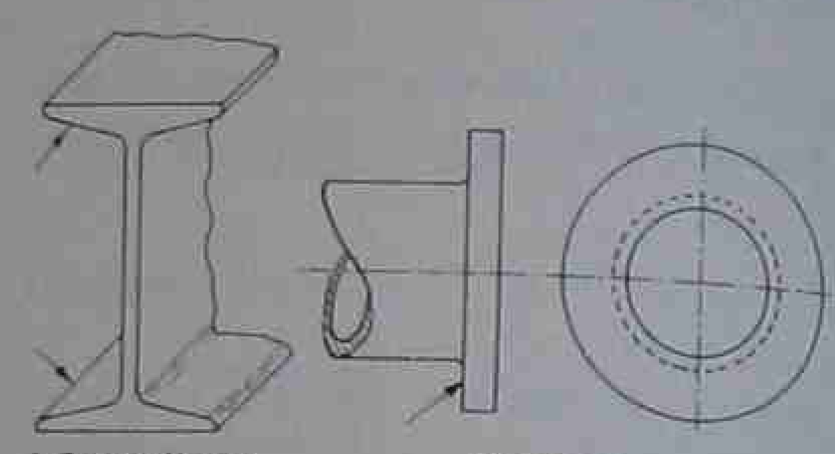


Figure 1.222 *Flange*

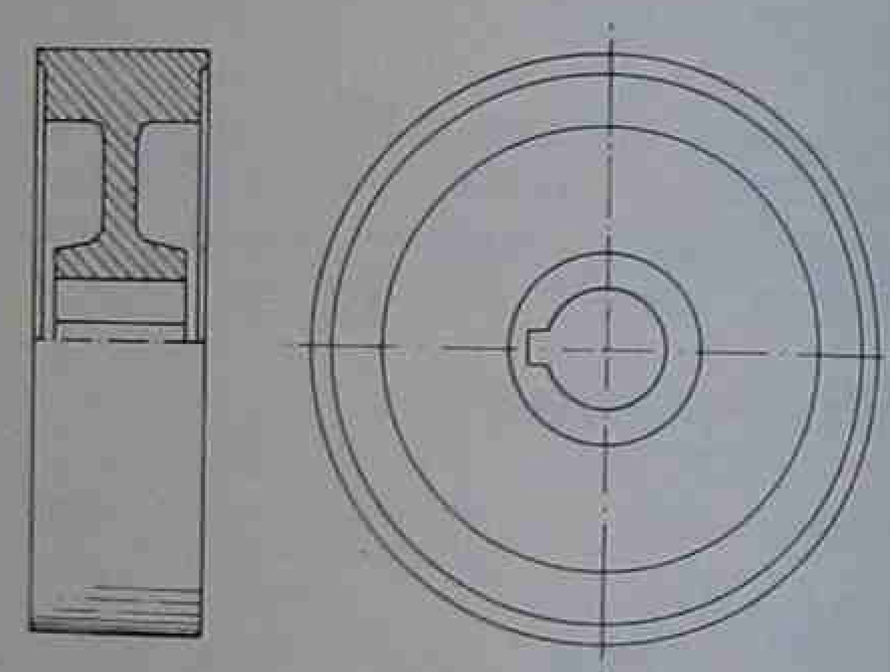


Figure 1.223 *Flywheel*

FLYWHEEL

A flywheel is a rotating disk with most of its mass concentrated at the rim. It is used for storing rotational energy and hence smoothing out vibrations.

FOOT

A foot is a projection on a casting, forging or weldment by which any of these is supported on a usually horizontal surface.

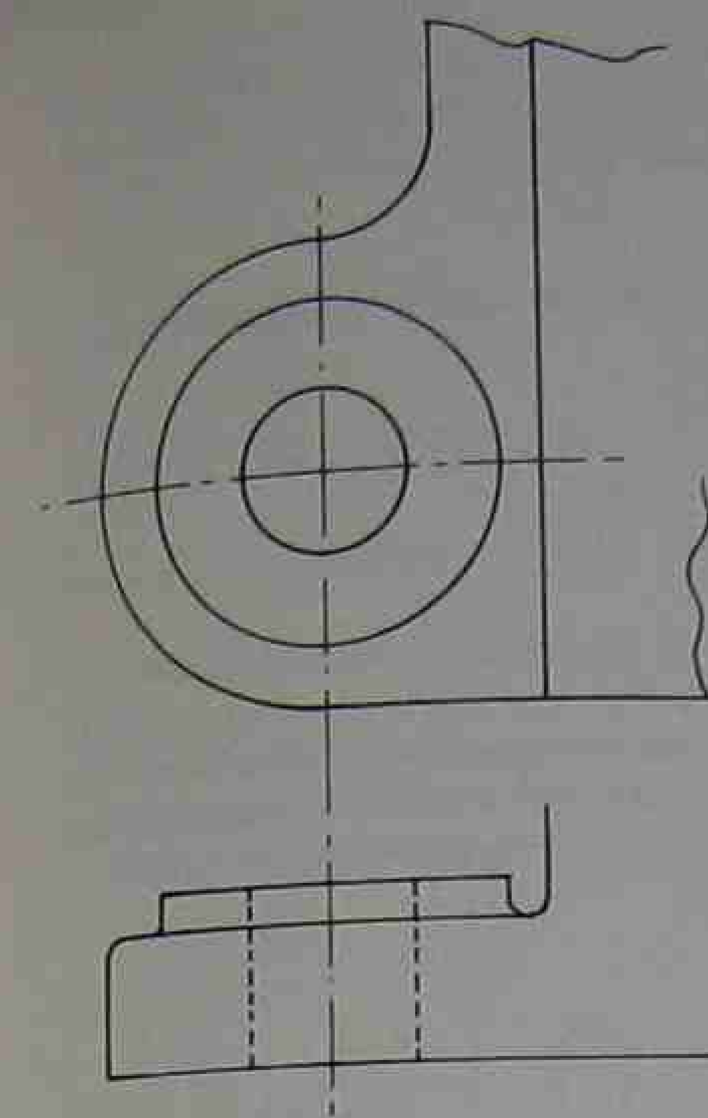


Figure 1.224 *Foot*

FORGING

A part made by hammering or pressing hot malleable metals either by hand or machine is known as a forging.

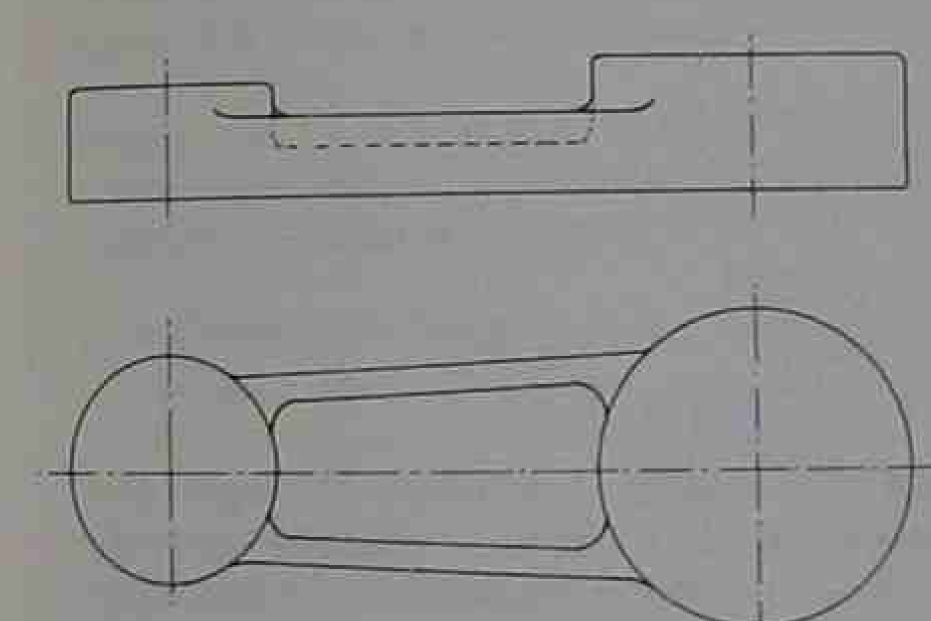


Figure 1.225 *Forging (for lever)*

FORK, FORK END

This is a feature at the end of a rod or lever consisting of two prongs which may have holes in them to take a pin.

HANDLE

A handle is a bar or lever shaped to give a good hand grip while applying straight or rotational motion.

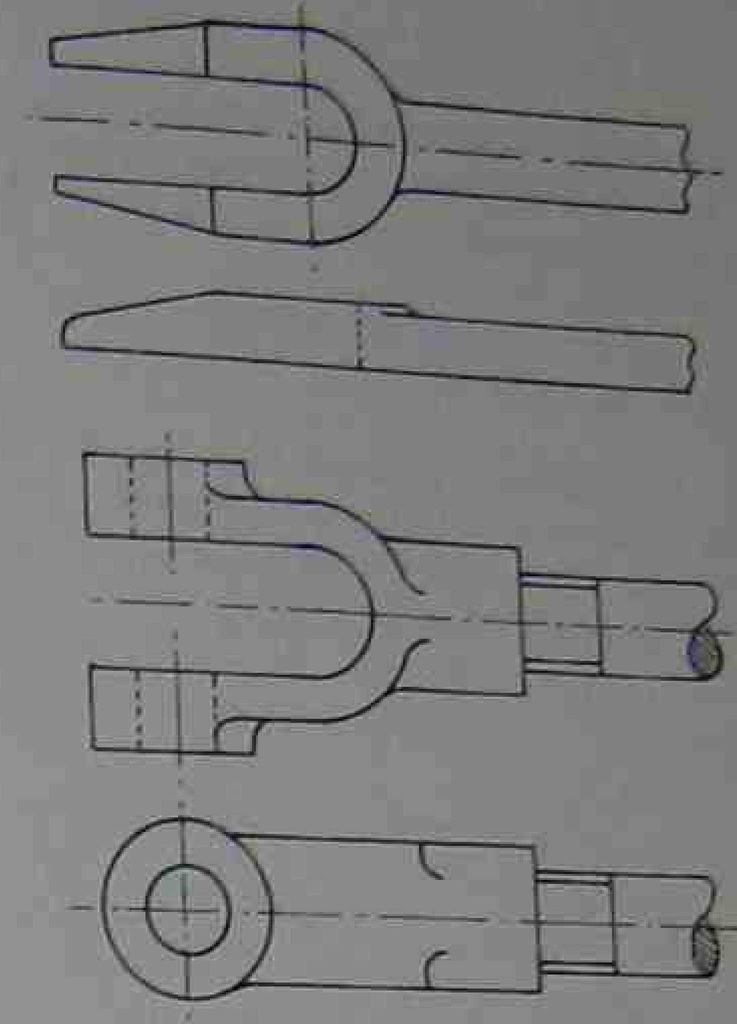


Figure 1.226 *Forks*

HANDWHEEL

This is a wheel with a heavy rim designed to provide a good hand grip when operating valve spindles, machine tool controls, steering gear, etc. Sometimes a handle is fitted to the rim to improve control.

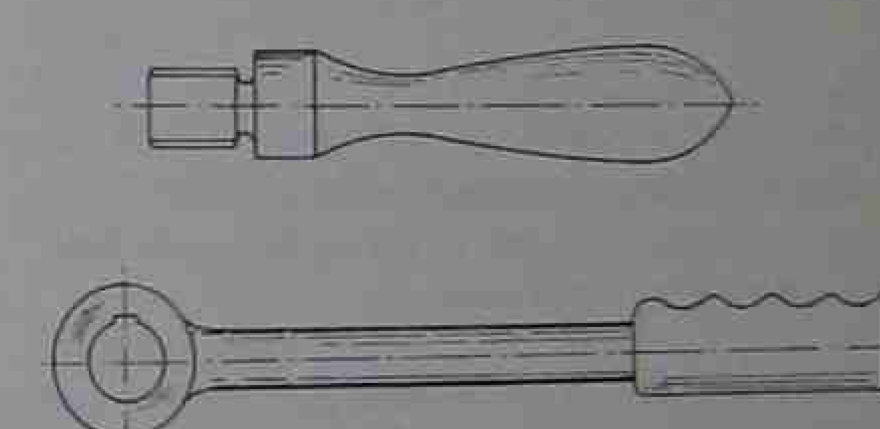


Figure 1.227 *Handles*

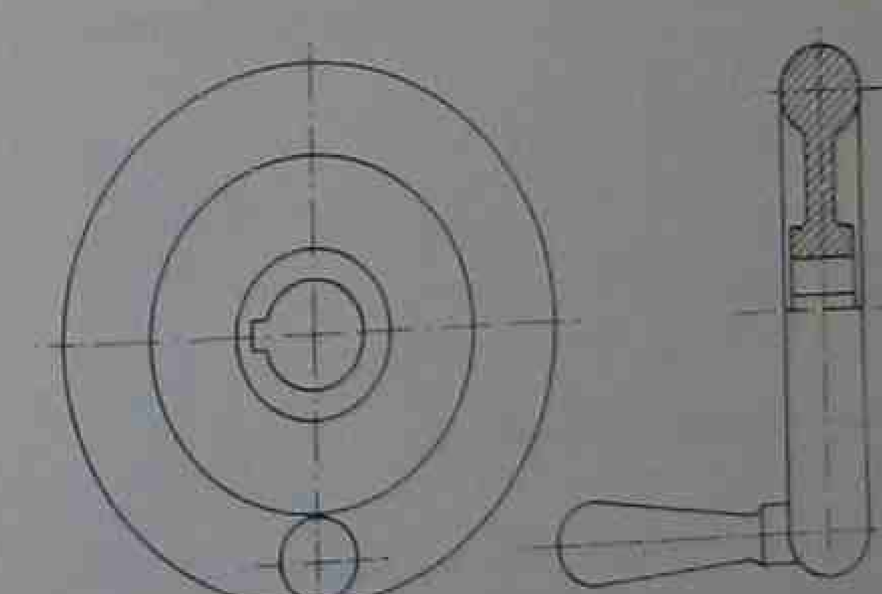


Figure 1.228 *Handwheel*

HOLES

In engineering components holes may be produced by numerous methods including casting, forging, punching, drilling and boring, and they may be reamed, tapped, countersunk, etc.

The following describes some of the types of hole most frequently encountered.

Bored hole A bored hole is produced by means of a single point boring tool on a lathe or boring mill. Greater accuracy is obtained than with a drill and larger holes can be produced.

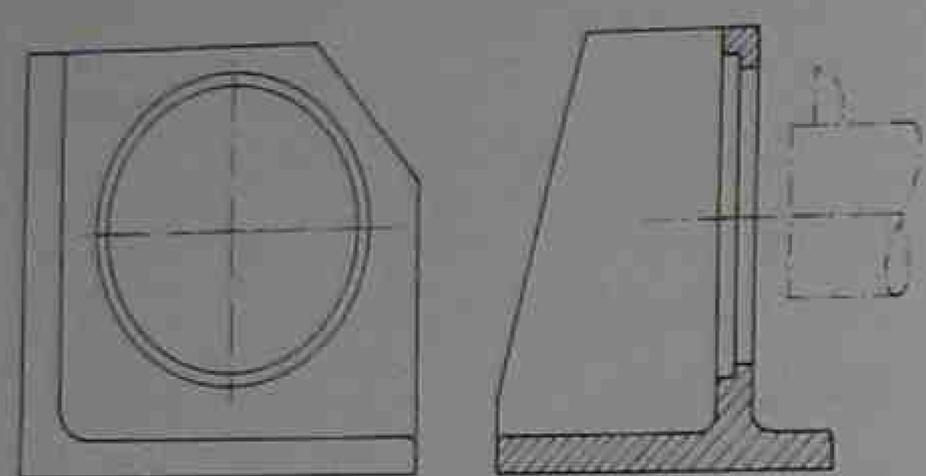


Figure 1.229 *Bored hole*

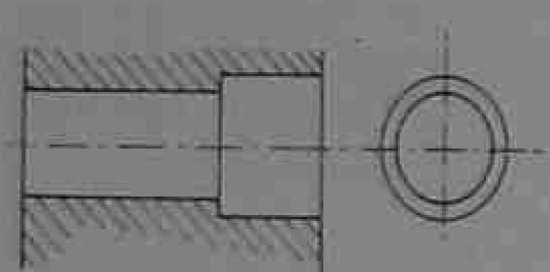


Figure 1.230 *Counterbored hole*

Counterbored hole A counterbored hole has its diameter increased for part of the depth, e.g. to provide a recess for a screw head.

Countersunk hole A countersunk hole has a conical section at the entrance to take the head of a countersunk screw, or a bolt or rivet.

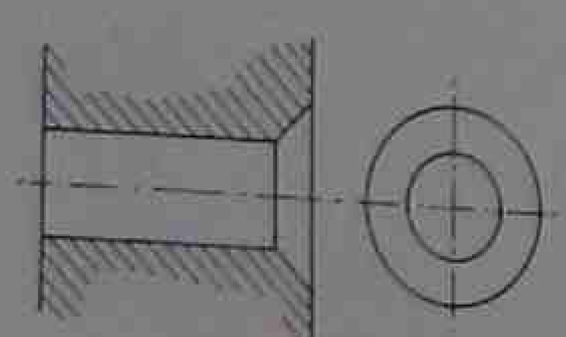


Figure 1.231 *Countersunk hole*

Drilled hole A drilled hole is produced by a rotating twist drill and it may be either a 'through' hole or a 'blind' hole.

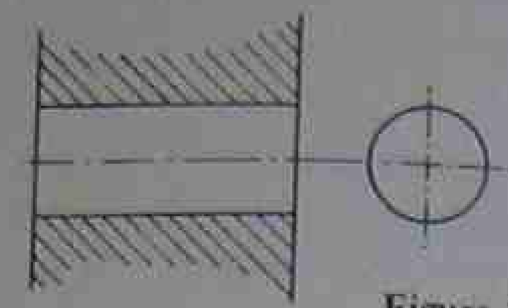


Figure 1.232 *Drilled through hole*

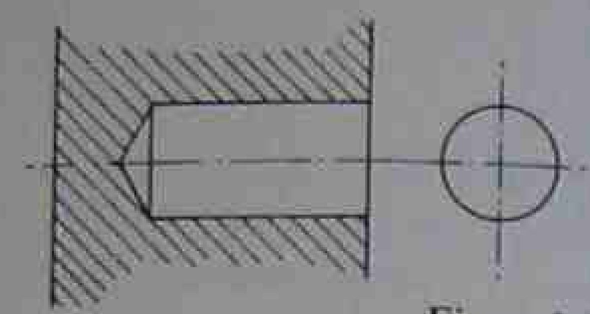


Figure 1.233 *Drilled blind hole*

Elongated hole An elongated hole is produced from a hole initially circular which has been extended into a slot by filing, or by joining two adjacent circular holes.

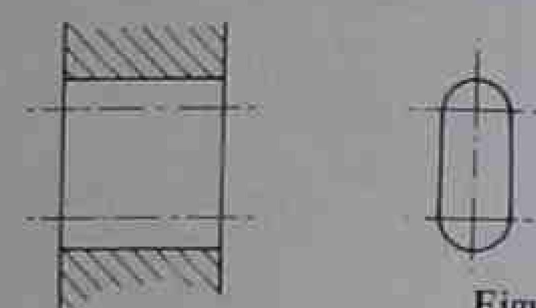


Figure 1.234 *Elongated hole*

Flat bottomed hole A drilled blind hole generally has a conical bottom but this can be made flat with a special tool to produce a 'flat-bottomed' hole.

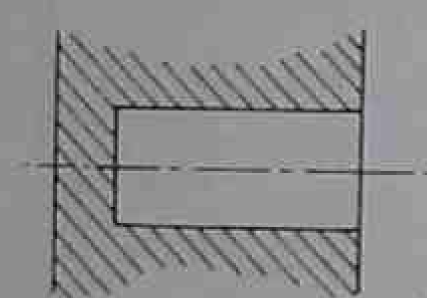


Figure 1.235 *Flat-bottomed hole*

Punched hole This is a hole of any shape produced in a sheet of metal by a hand- or machine-operated punch. A piece of metal the shape of the punch is forced out by shearing action.

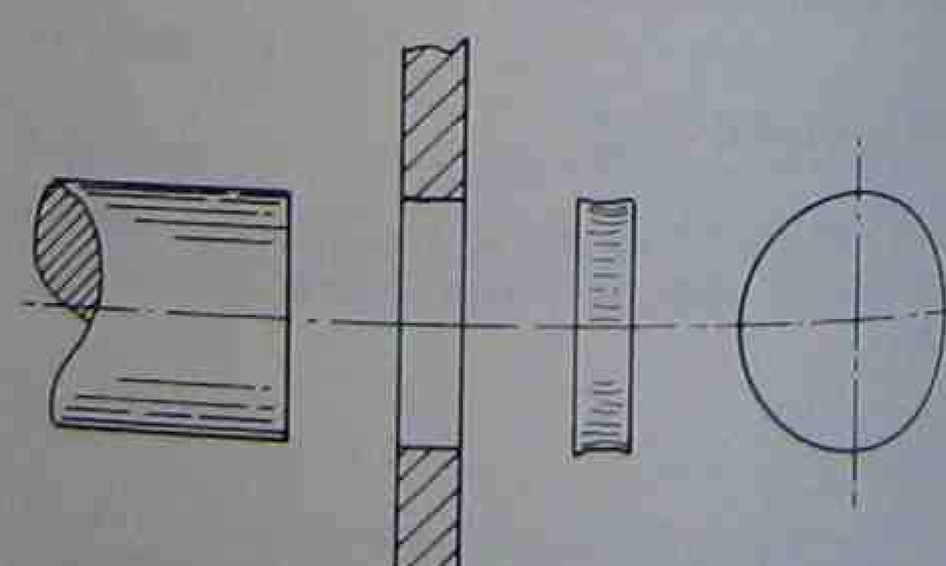


Figure 1.236 *Punched hole*

Reamed hole A reamed hole is an accurately-sized hole made by slightly enlarging a drilled hole by means of a fluted cutter called a *reamer*.

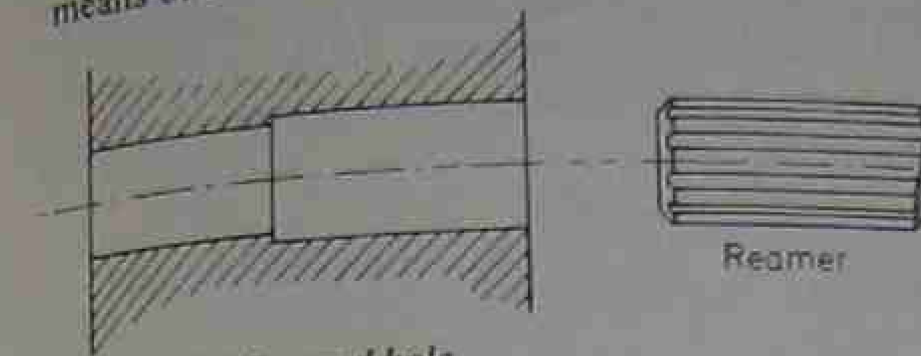


Figure 1.237 *Reamed hole*

Spotfaced hole (knifed hole) A spotfaced hole has a flat circular face cut at the entrance to take the head of a bolt etc. This is necessary when the surface is initially rough.

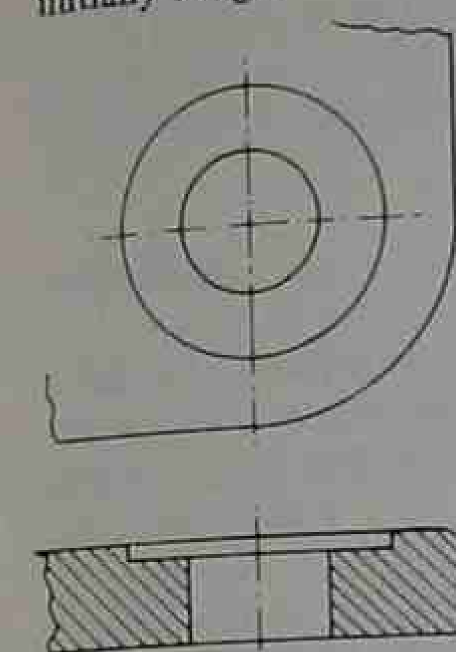


Figure 1.238 *Spotfaced hole (knifed hole)*

Square hole A square hole, or a hole of any non-circular shape, may be produced by hand filing or with a special tool called a *broach*.

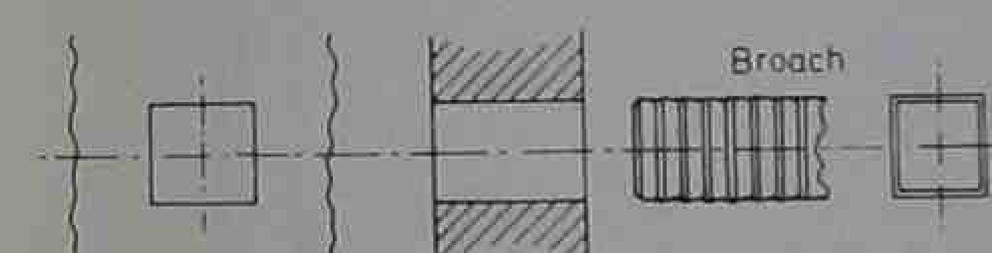


Figure 1.239 *Square hole (using a broach)*

Tapped hole This is a circular hole in which a screw thread has been cut for all or part of its length.

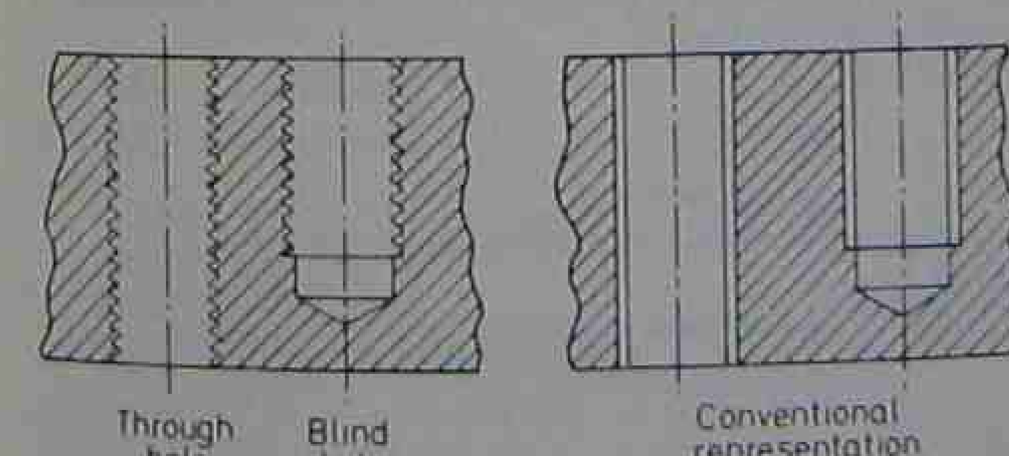


Figure 1.240 *Tapped holes*

HOUSING

A component which contains or 'houses' other components is known as a *housing*.

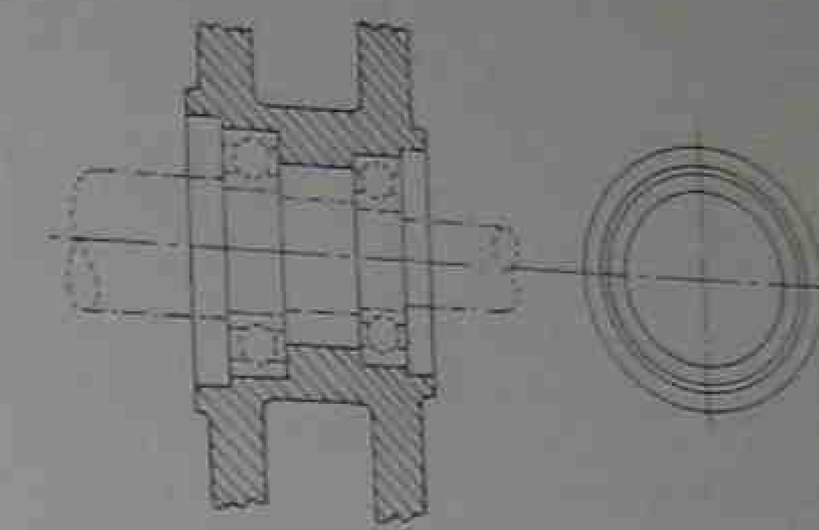


Figure 1.241 *Housing (for bearings)*

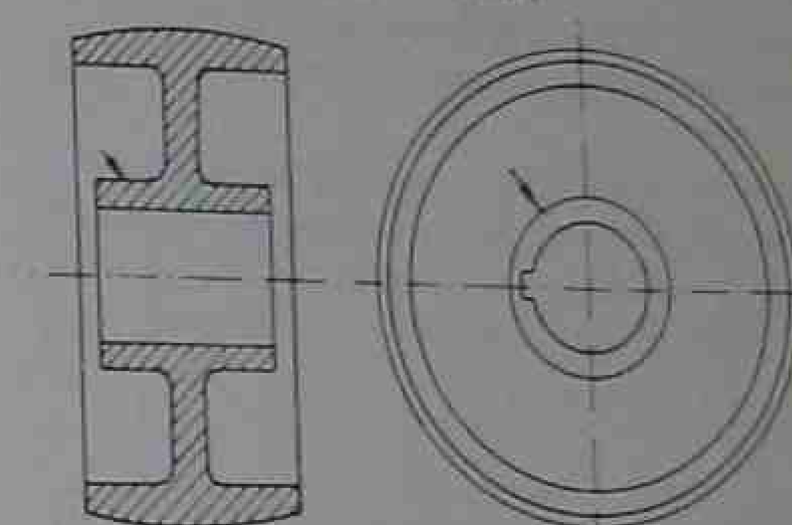


Figure 1.242 *Hub (of pulley)*

HUB

A hub is the heavy central section of a wheel, disk or pulley which usually has a hole for mounting on a shaft.

KNURLING

Knurling consists of serrations formed on the cylindrical surfaces of levers, handles, circular nuts, etc. to ensure a good hand grip.

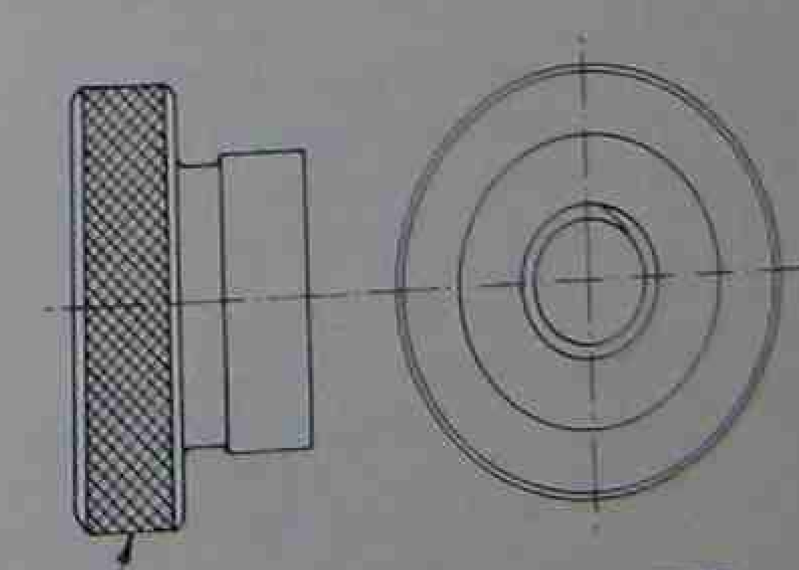


Figure 1.243 *Knurling*

LUG

A lug is an ear-shaped projection on a casting or forging, etc., often pierced by a hole.

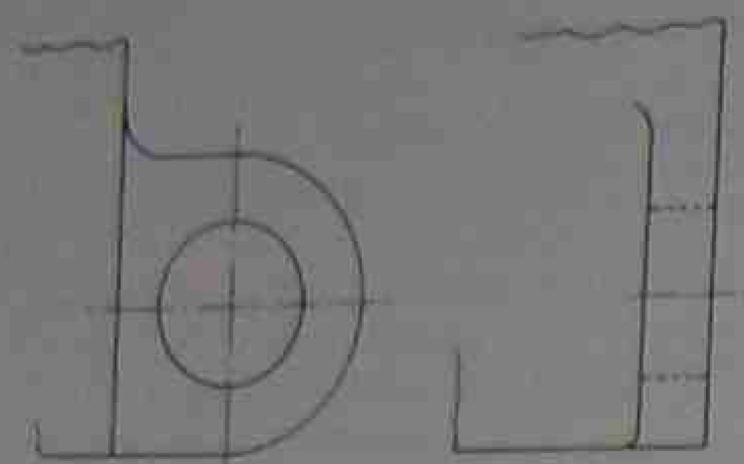


Figure 1.244 *Lug*

PEDESTAL

A support for other components, such as a shaft, mounted on a horizontal or vertical surface.

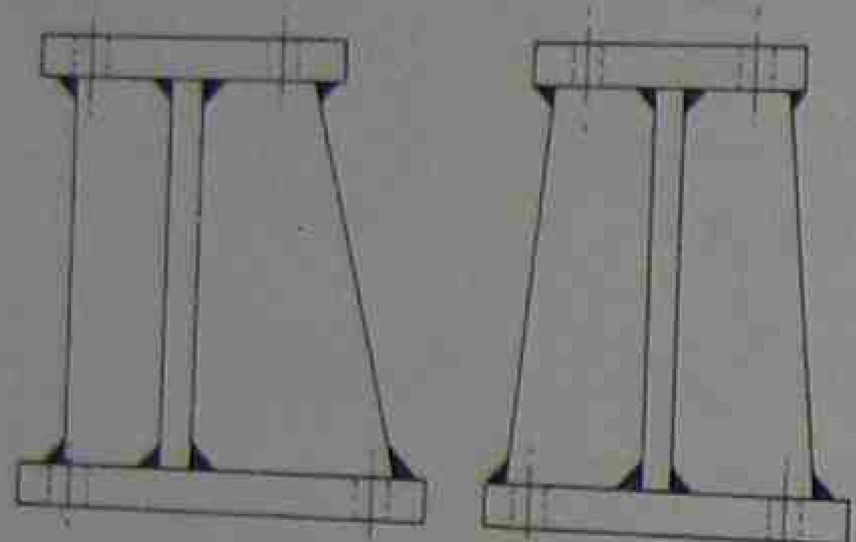


Figure 1.245 *Pedestal*

PILLAR

This is a bar or rod usually circular and vertical which holds two components apart. It is also called a *column*.

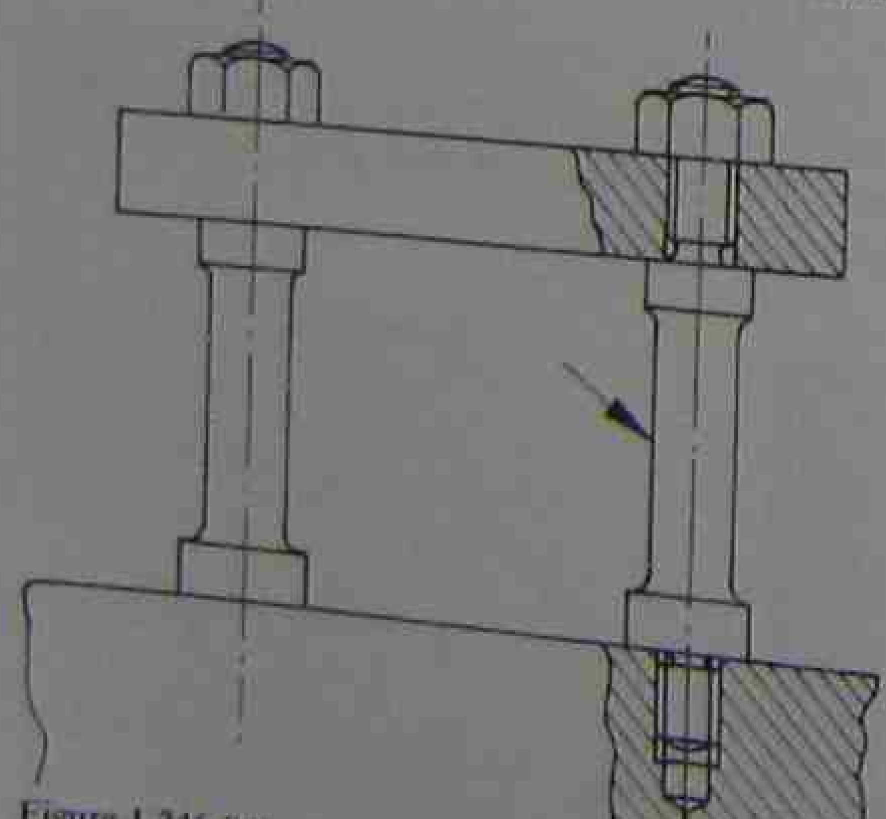


Figure 1.246 *Pillar (or column)*

PISTON

A piston is a cylindrical object which slides in a cylinder filled with a gas or liquid, as in an engine or pump. The piston is connected to a piston rod or connecting rod and is fitted with sealing rings (*see* Piston rings).

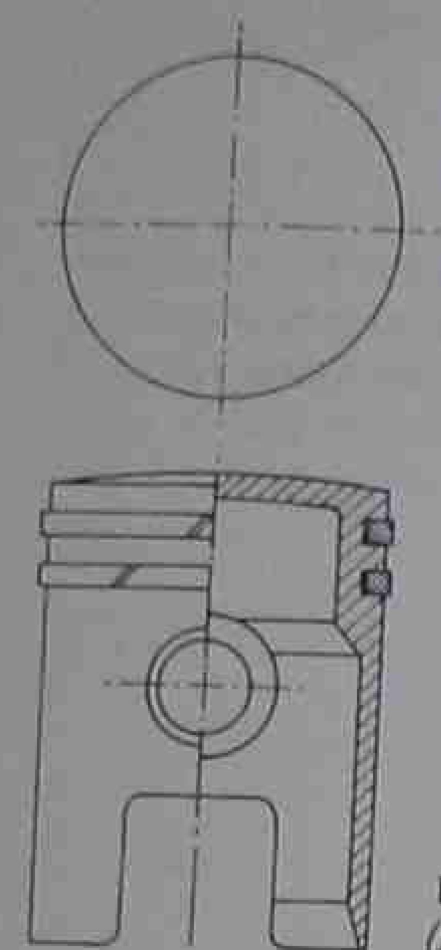


Figure 1.247 *Piston (for auto engine)*

PLATE

A plate is a piece of thin flat material, or any part which is generally flat in shape.

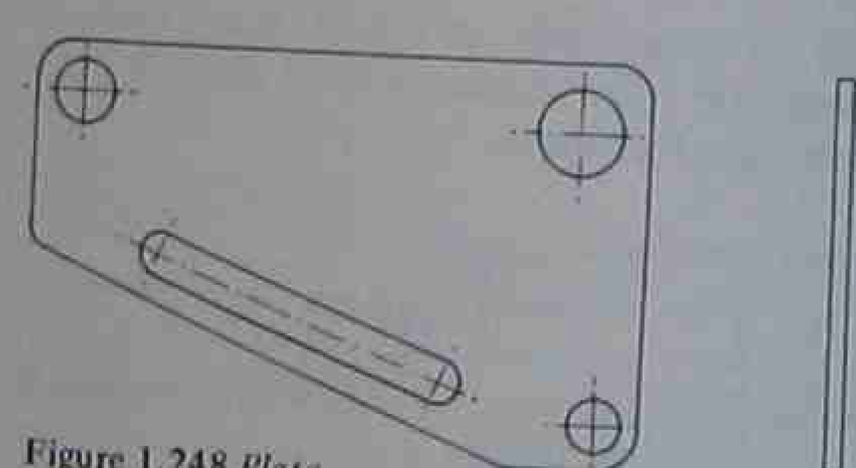


Figure 1.248 *Plate*

PLUNGER

Another name for a piston which is usually applied to small pump pistons.

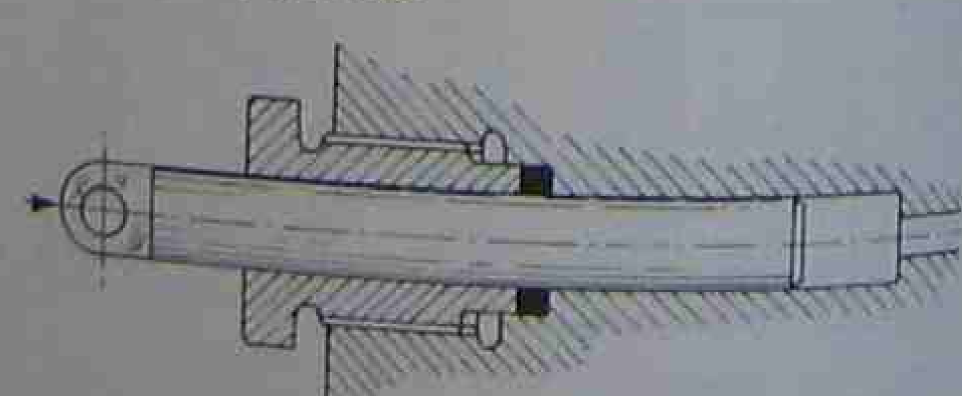


Figure 1.249 *Plunger (for pump)*

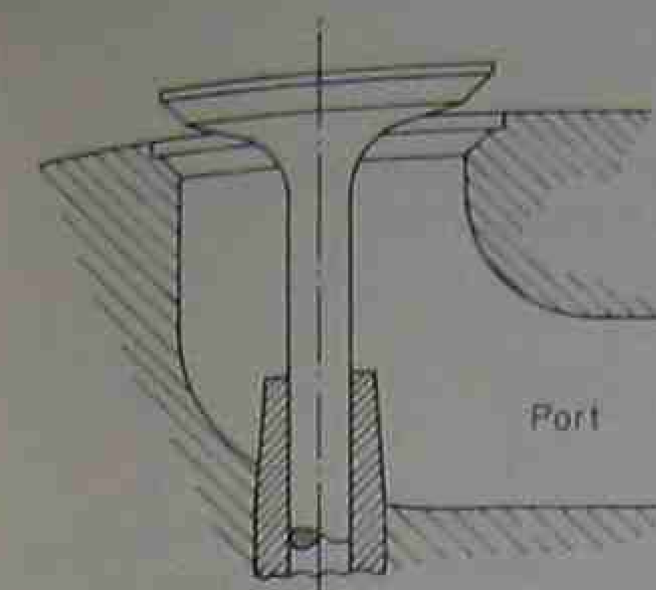


Figure 1.250 *Port (for IC engine valve)*

PORT

A port is a hole of any shape through which fluids or gases flow in engines, pumps, etc. They are often variable in size and are used to control the flow.

RADIUS

A radius is a modification to a sharp internal or external corner on a component to round it off with part of a circular arc for reasons of appearance, strength and safety (*see* Fillet).

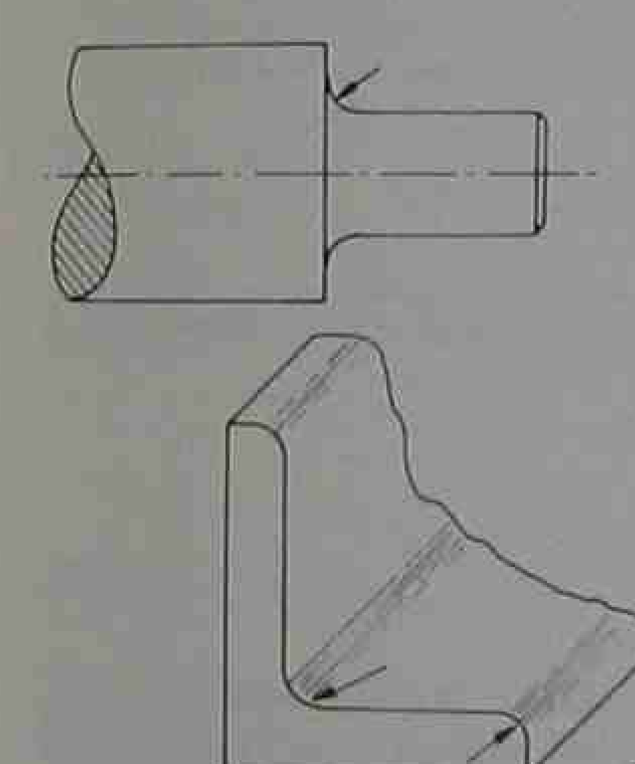


Figure 1.251 *Radius*

RECESS (FAUCET)

A recess is a depression or hollow in the surface of a part into which another part may be fitted.

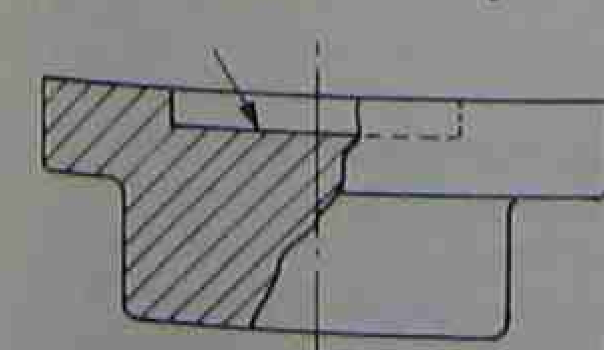


Figure 1.252 *Recess*

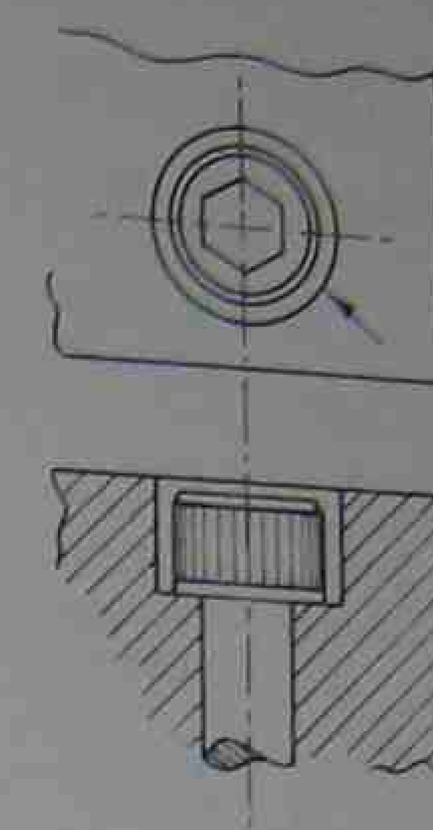


Figure 1.253 *Recessed socket head screw*

RIB

A rib is a raised thin section on the surface of a casting, forging, weldment, etc., used for strengthening and stiffening. It is also one of a series of members built into aircraft wings to which the skin is attached.

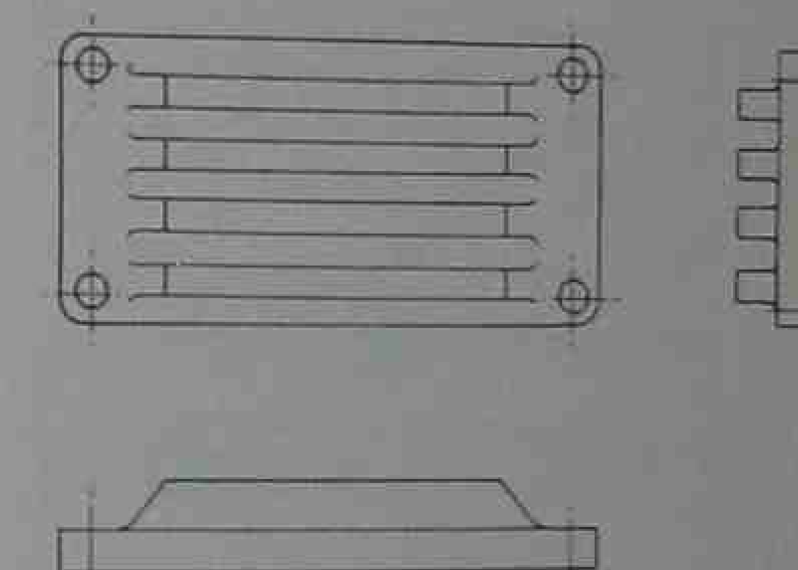


Figure 1.254 *Ribbed plate*

ROTOR

A rotor is a disk or drum mounted on a rotating shaft, e.g. turbine rotor, electric motor rotor. The term is also applied to a system of aerofoils such as those supporting a helicopter.

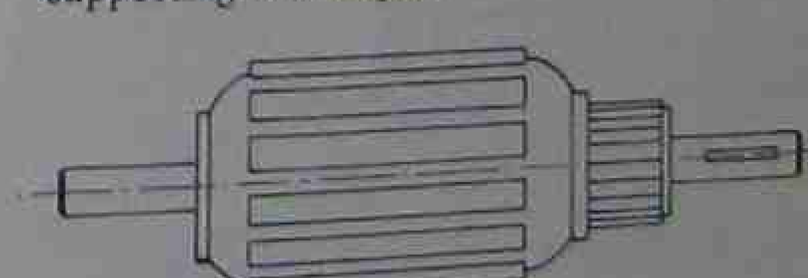


Figure 1.255 *Rotor (for electric motor)*

SLEEVE

This is a thin circular tube machined on the inside and outside.

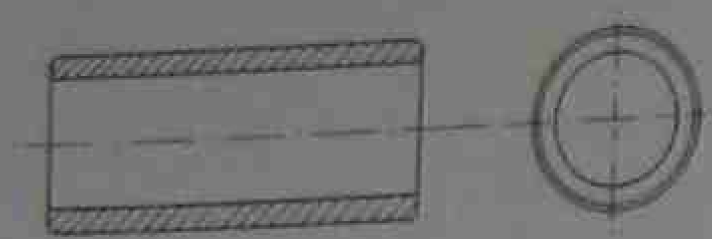


Figure 1.256 *Sleeve*

SLOT

A slot is a long groove or aperture, straight or curved, in a machine part in which another part slides or is mounted.

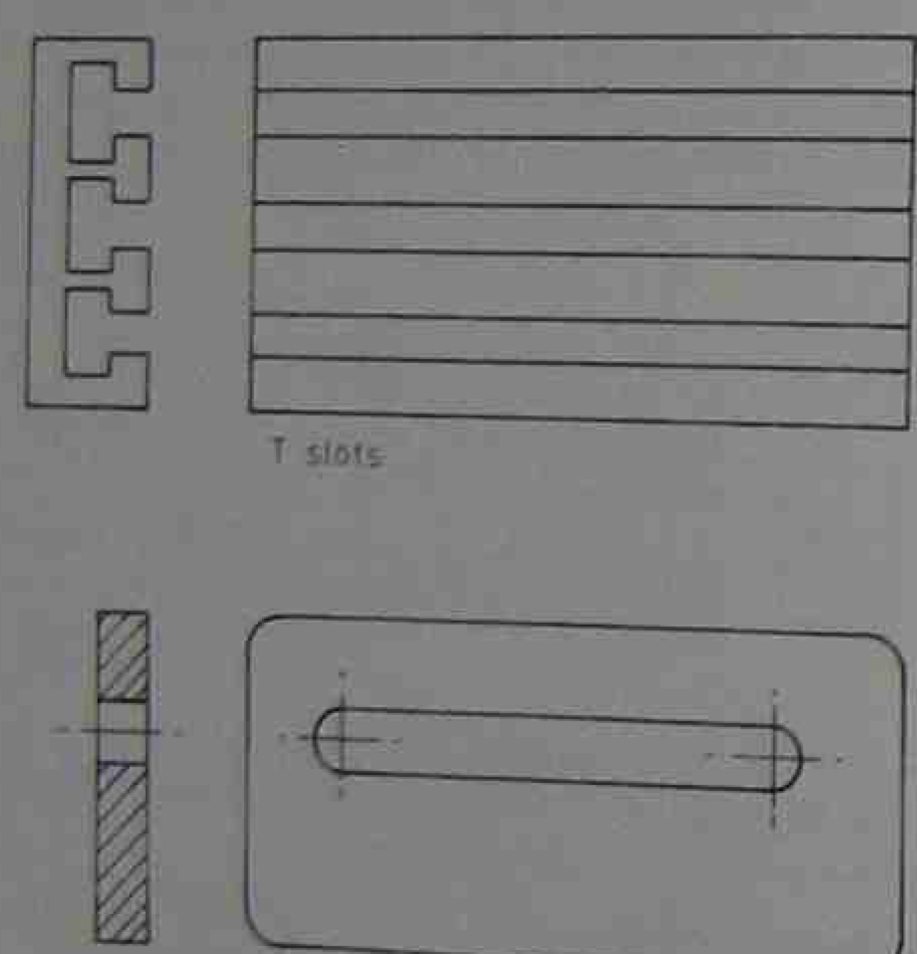


Figure 1.257 *Slots*

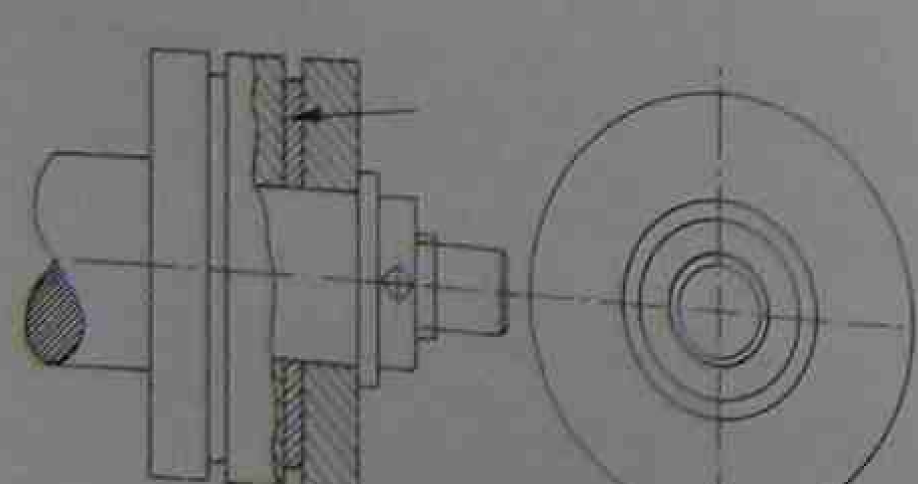


Figure 1.258 *Spacer*

SPACER

A spacer is another name for a *distance piece* but it usually refers to a thin plate.

SPIGOT

This is a raised part on the face of a circular flange on a pipe or shaft coupling which fits accurately into a corresponding recess (*faucet*) in the mating flange.

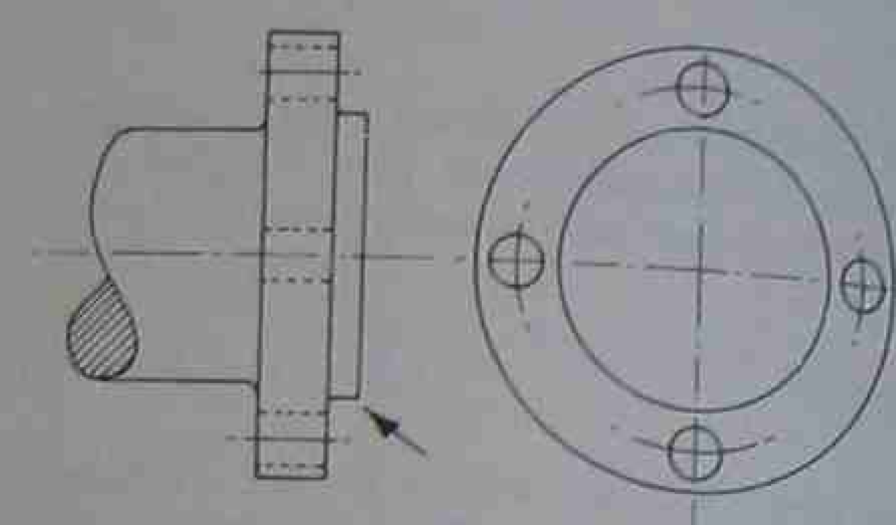


Figure 1.259 *Spigot*

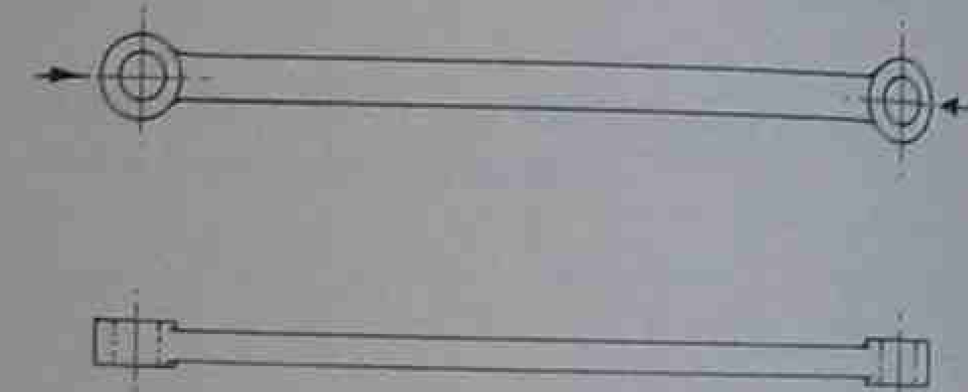


Figure 1.260 *Strut*

STRUT

A light structural member which takes a compressive load.

WEB

A web is a relatively thin part of a casting, forging or weldment joining two thicker parts, e.g. the central section of a channel or I beam. The thin disk joining the hub and rim of a wheel or pulley is also known as a web.

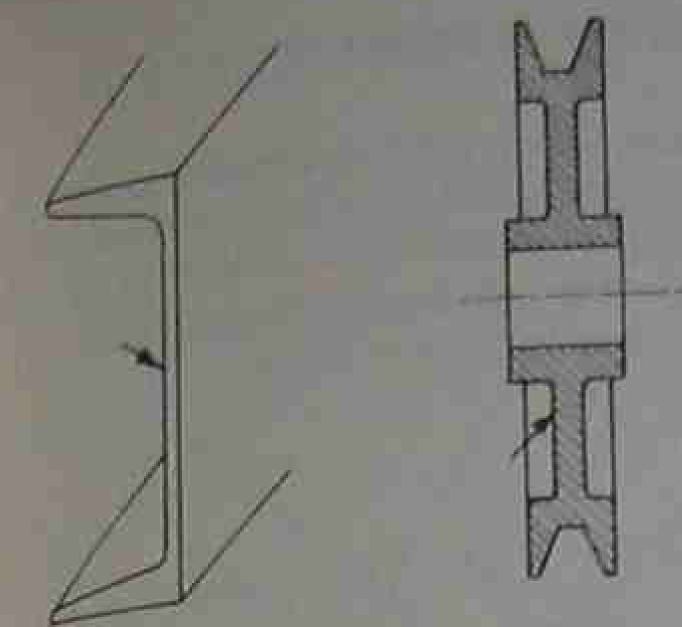


Figure 1.261 *Web (of channel section and pulley)*

WELDMENT

This refers to any welded assembly, usually fabricated from steel plate, bar, tube, etc.

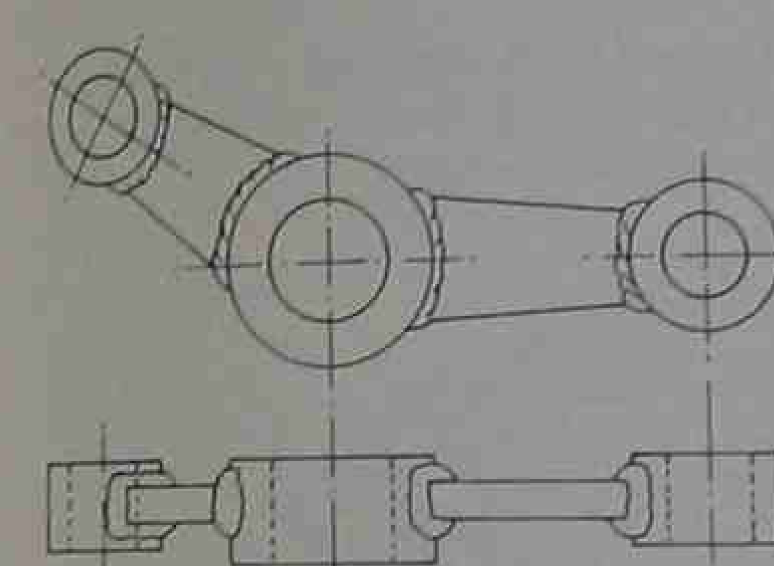


Figure 1.262 *Weldment (lever)*

WHEEL

This is the name given to a wide variety of rotating disk-shaped objects, such as flywheels, gearwheels and handwheels.

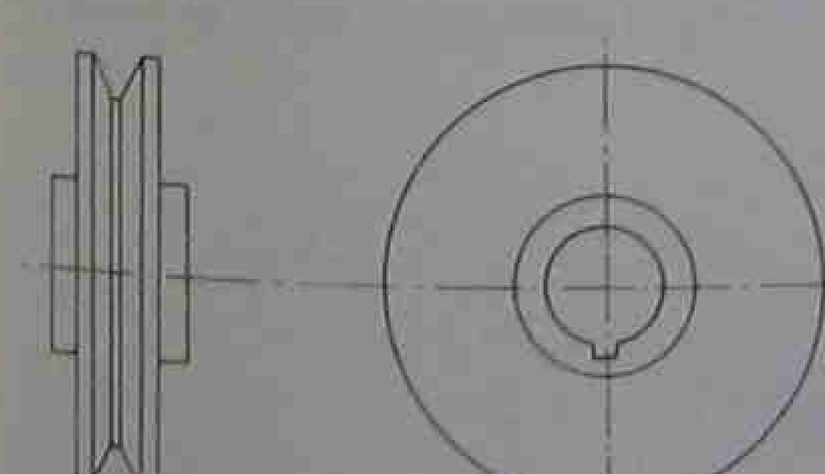


Figure 1.263 *V pulley wheel*

1.10 STOCK MATERIALS

STEEL SHEET

The term 'sheet' refers to thicknesses of less than 5mm ($^3/_{16}$ in.). When made it is rolled into coils and then cut into standard-sized pieces for sale.

Sheet is usually made in low carbon steel cold rolled to give a good finish and it may be given a protective coating of tin, zinc (galvanised), or plastics such as PVC. Various kinds of corrugated sheet are produced.

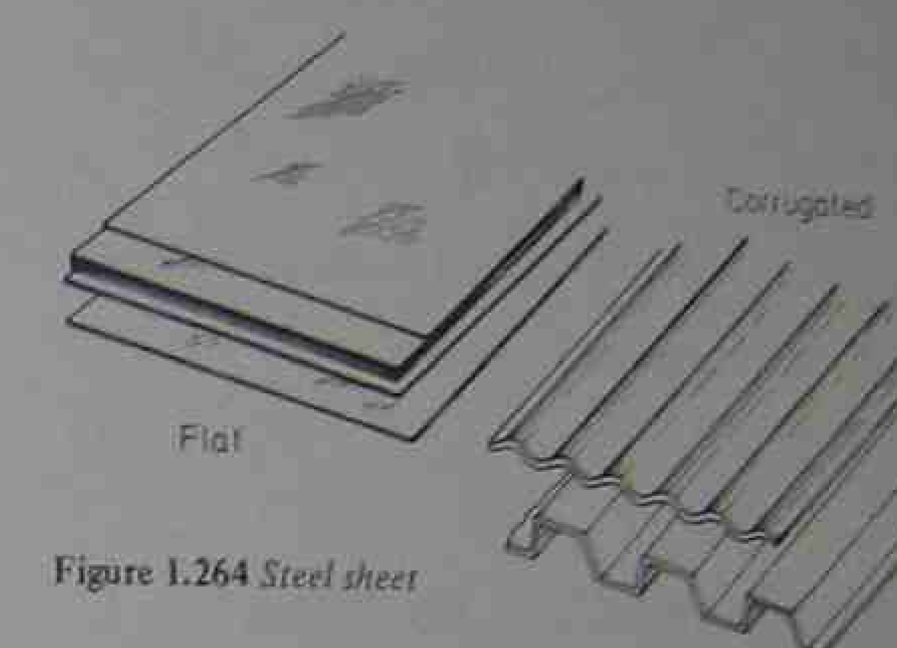


Figure 1.264 *Steel sheet*

Figure 1.265 *Steel plate*

STEEL PLATE

This is steel from 6mm (¼ in.) thick up to 150mm (6 in.) thick. Steel plate is produced in various grades and in sizes up to 2.5m (8 ft.) in width.

STEEL STRIP

This is narrow plate or sheet made in carbon, alloy and stainless steel.

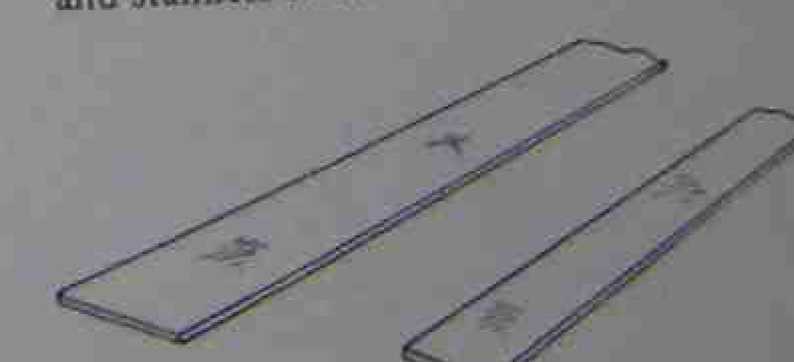


Figure 1.266 *Steel strip*

HOT-ROLLED STEEL SECTIONS

The following sections are produced by hot rolling using specially-shaped rollers: I beam, angle, channel and T for structural work; round, rectangular and hexagonal bar, and sheet for stamped, drawn and punched parts.

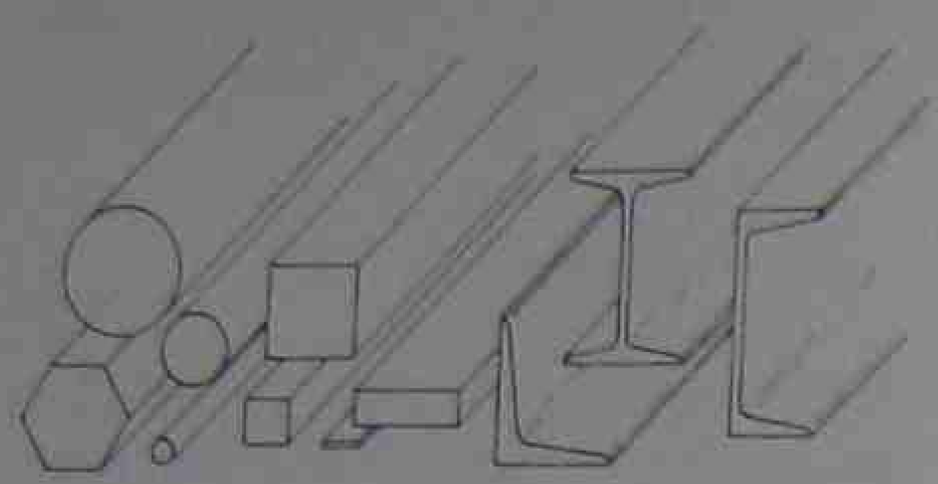


Figure 1.267 *Hot rolled steel sections*

COLD-ROLLED STEEL SECTIONS

Cold rolling of the above sections produces a smooth surface which can be plated or coated.

BRIGHT STEEL BAR

Bar produced by hot rolling has a rough oxidised surface and is referred to as 'black' bar. 'Bright' bar, of higher accuracy, is produced by the drawing process in sizes up to 40mm (1½ in.) diameter.

Square, rectangular and hexagonal bright bars are used extensively for the production of small machined parts.

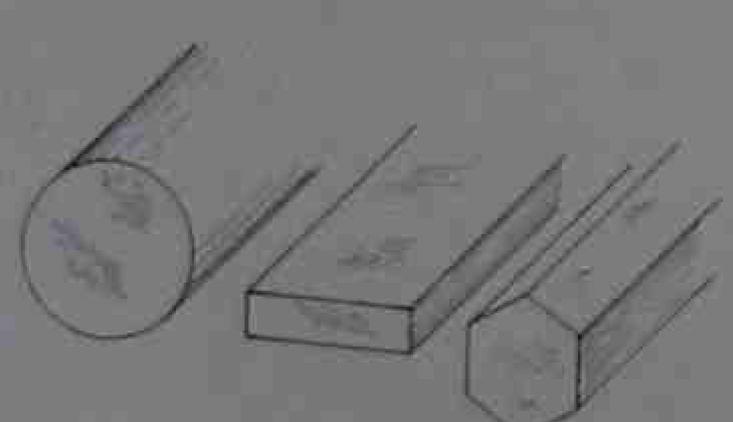


Figure 1.268 *Bright steel bar*

HOLLOW STEEL SECTIONS

Hollow round sections in small diameters are used for small machined parts. Large hollow round and hollow rectangular sections are of great value in all kinds of structural work. Rectangular sections range in size from a few millimetres to 400mm (16 in.).

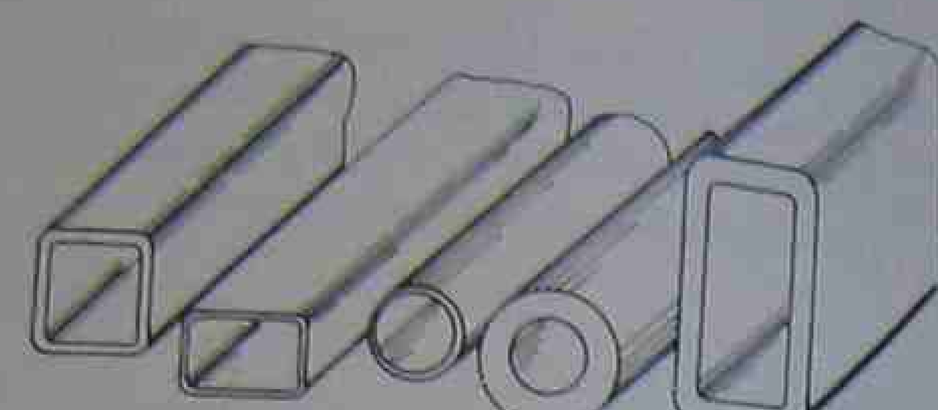


Figure 1.269 *Hollow steel sections*

STEEL CHEQUER PLATE

Chequer plate is used for flooring and it has a raised pattern to give a good foothold. It is used in ships' engine rooms, power stations, etc.

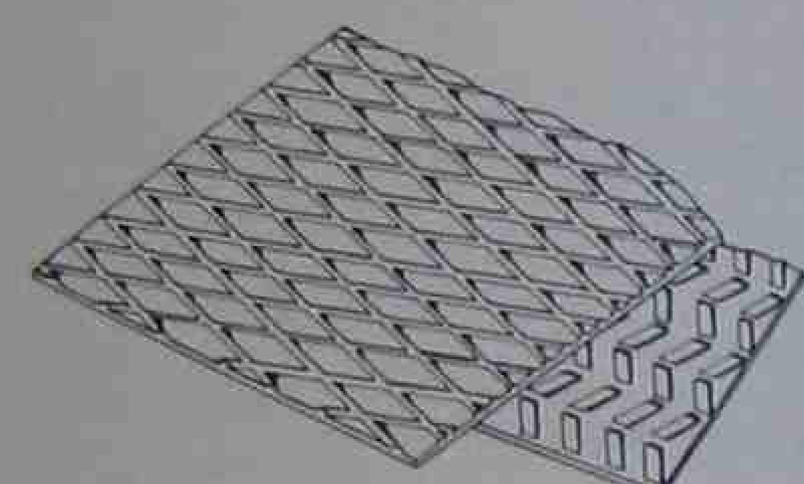


Figure 1.270 *Chequer plate*

WIRE

High strength carbon and alloy wire is used for wire ropes and small springs, while wire of lower-grade steel is used for wire mesh. Copper wire is made in a large range of diameters and is often insulated with varnish or a plastic coating. Tinned copper wire is also made.

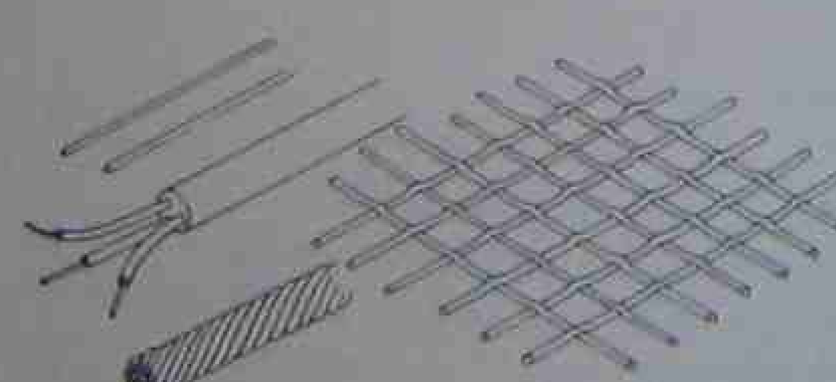


Figure 1.271 *Wire*

NON-FERROUS METALS

Brass, bronze, copper, aluminium and aluminium alloys are used to make solid and hollow round bar, rectangular bar and hexagonal bar. These metals are also used for a wide range of gauges of sheet and plate.

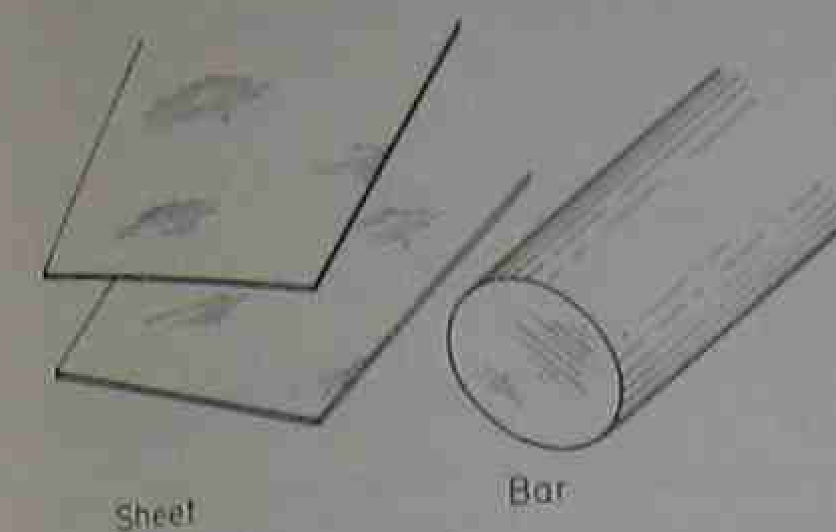


Figure 1.272 *Non-ferrous metals*

TUBE

Light gauge metal hollow sections are referred to as *tubes* and these are made in mild steel, high-strength alloy steel, stainless steel and aluminium alloys.

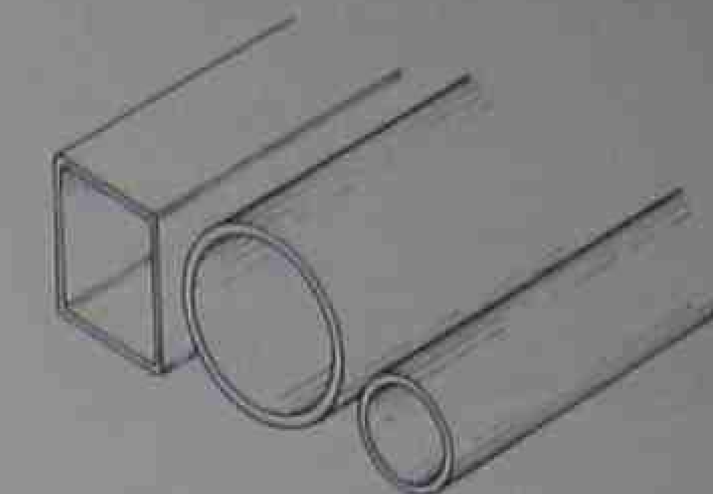


Figure 1.273 *Light gauge tube*

2. Power Transmission Elements

2.1 BRAKES AND CLUTCHES

BRAKES

A brake is a device which applies a resistive force to a moving body in order to retard or stop it. It may also be used to absorb and/or measure power. In most cases the motion is rotational.

Disk brake A brake in which a segmental block, or an annular flat ring of friction material, is forced against the face of a rotating disk. Automobile disk brakes employ two segmental blocks (or pads) on opposite sides of the disk operated by hydraulic pistons located in a *calliper*.

Disk brakes are less prone than drum brakes to 'fading', i.e. a fall in efficiency due to prolonged application.

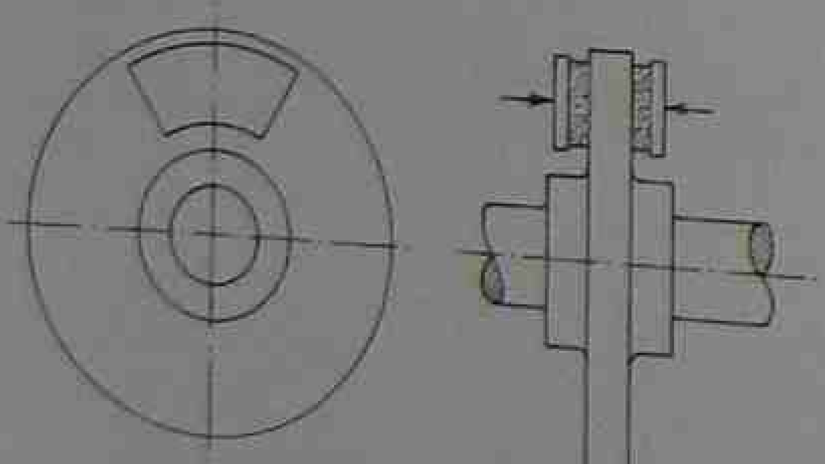


Figure 2.1 *Disk brake*

Single block brake This is a brake in which a block made from, or lined with, a material of high coefficient of friction is held against the rim of a rotating wheel in order to cause retardation of the wheel. The force may be applied manually, by a spring or by other means.

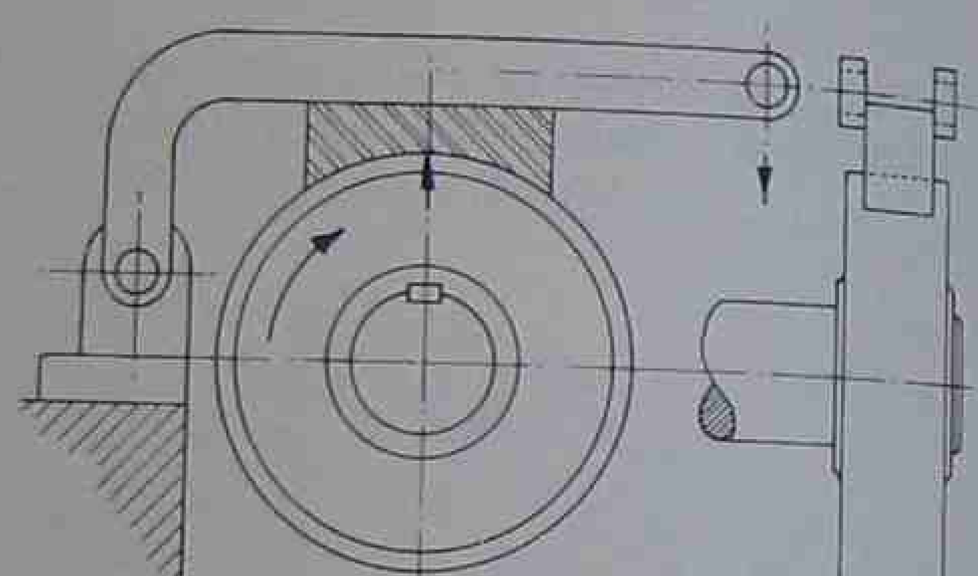


Figure 2.2 *Single block brake*

Double block brake In this brake two blocks situated on opposite sides of a wheel or drum are actuated simultaneously by a bell crank lever.

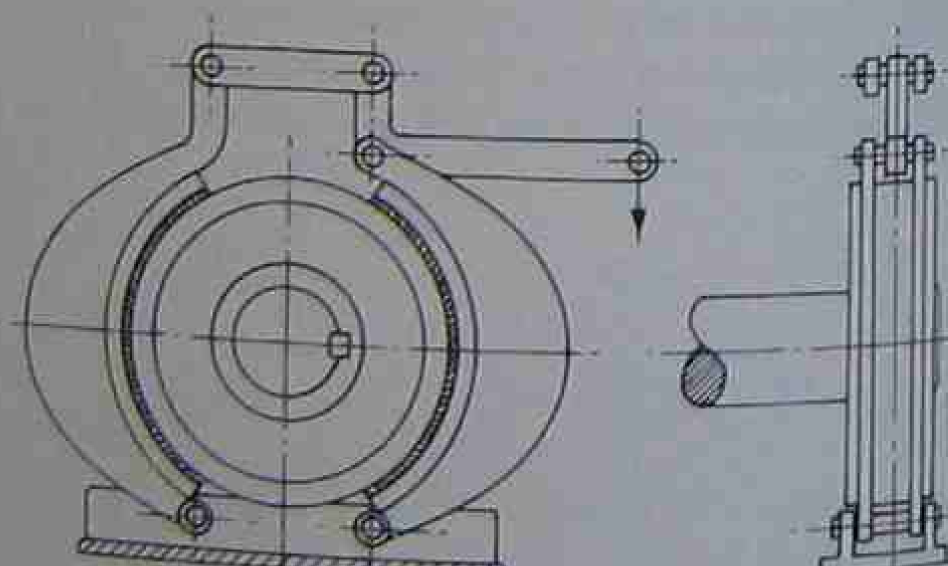


Figure 2.3 *Double block brake*

Spring-set brake This is a double block brake in which the blocks, or shoes, are normally held in contact with the drum, thus preventing its rotation. A compressed spring applies the load and the operating lever compresses the spring still further to release the drum. The lever may be operated by hand, electric solenoid, or by a hydraulic or pneumatic cylinder. This type of brake is used in lifts since it is 'fail-safe'.

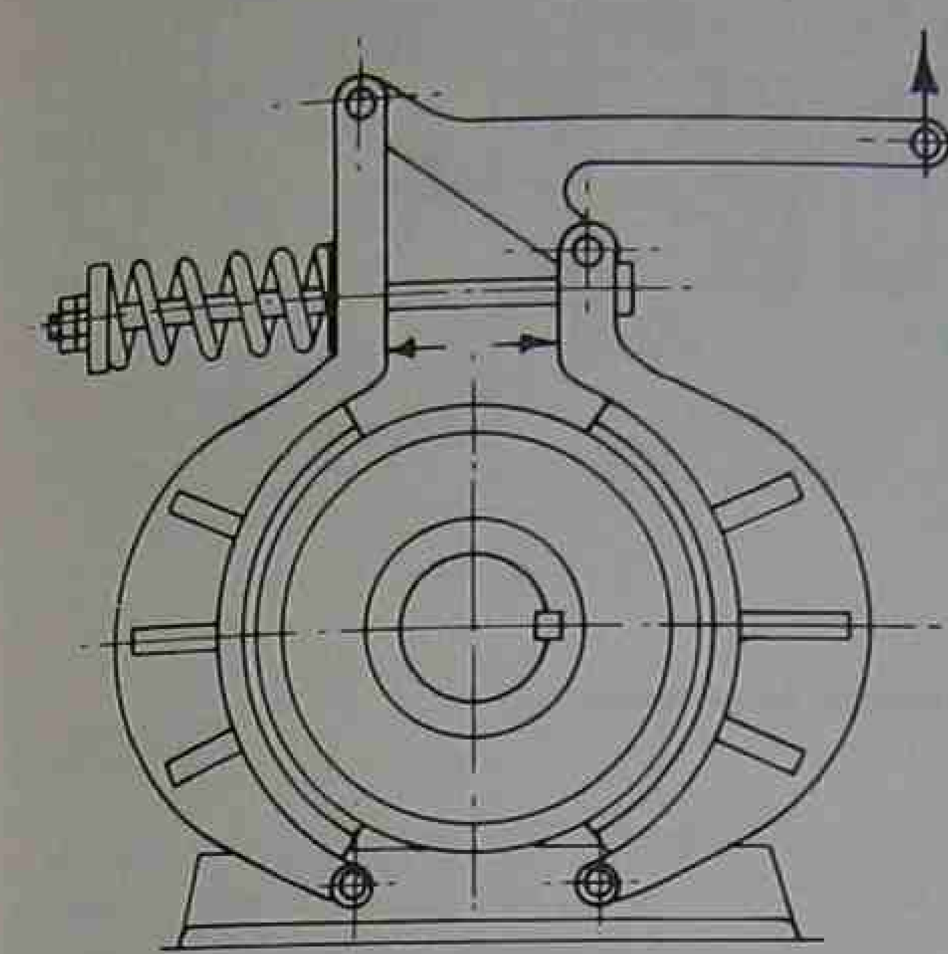


Figure 2.4 *Spring set brake*

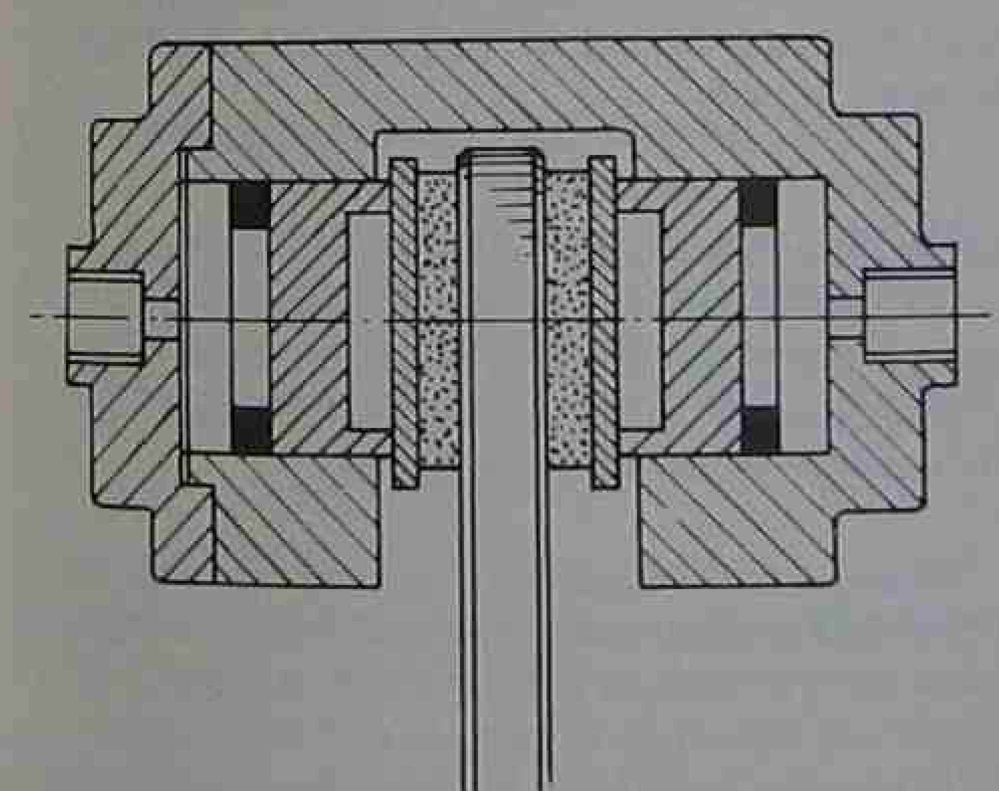


Figure 2.5 *Automobile disk brake calliper*

Multi-plate (disk) brake This type comprises several annular friction disks interleaved with steel disks. The steel disks run on a splined shaft, and the friction disks are prevented from rotating by pins. The brake is actuated by compressing the disks.

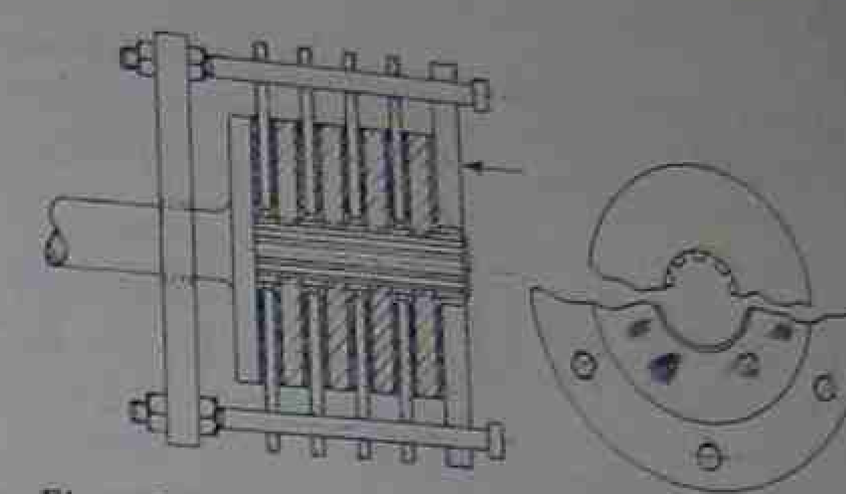


Figure 2.6 *Multi-plate (disk) brake*

Centrifugal brake In a centrifugal brake, shoes are thrown outwards by centrifugal force against the action of springs. The speed at which the brake operates is determined by the spring force.

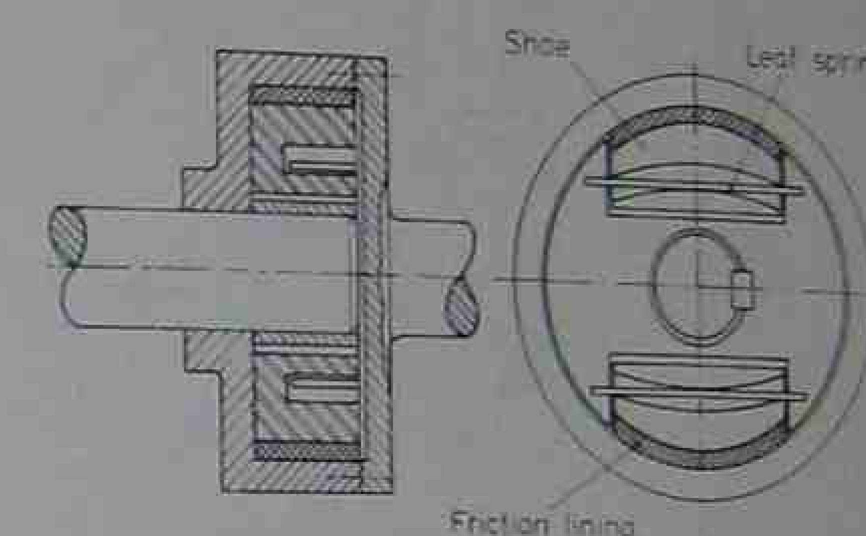


Figure 2.7 *Centrifugal brake*

Internal expanding shoe brake (drum brake) A brake in which two curved shoes are forced outwards against the inside of the rim of a rotating drum. The shoes are lined with friction material, known as brake linings, riveted or bonded to their outer surface.

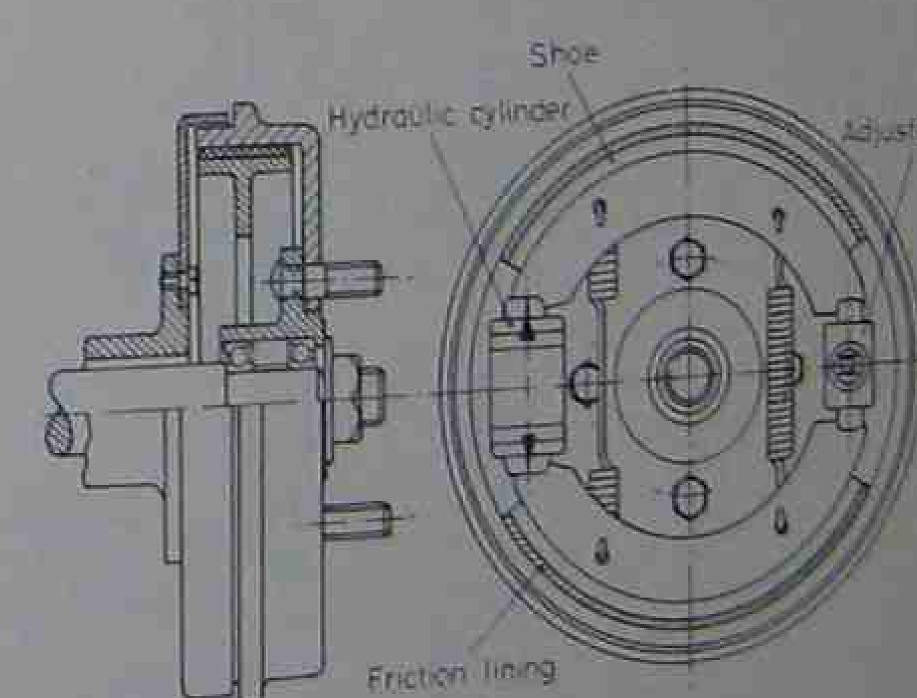


Figure 2.8 *Internal expanding shoe brake (automobile-type drum brake)*

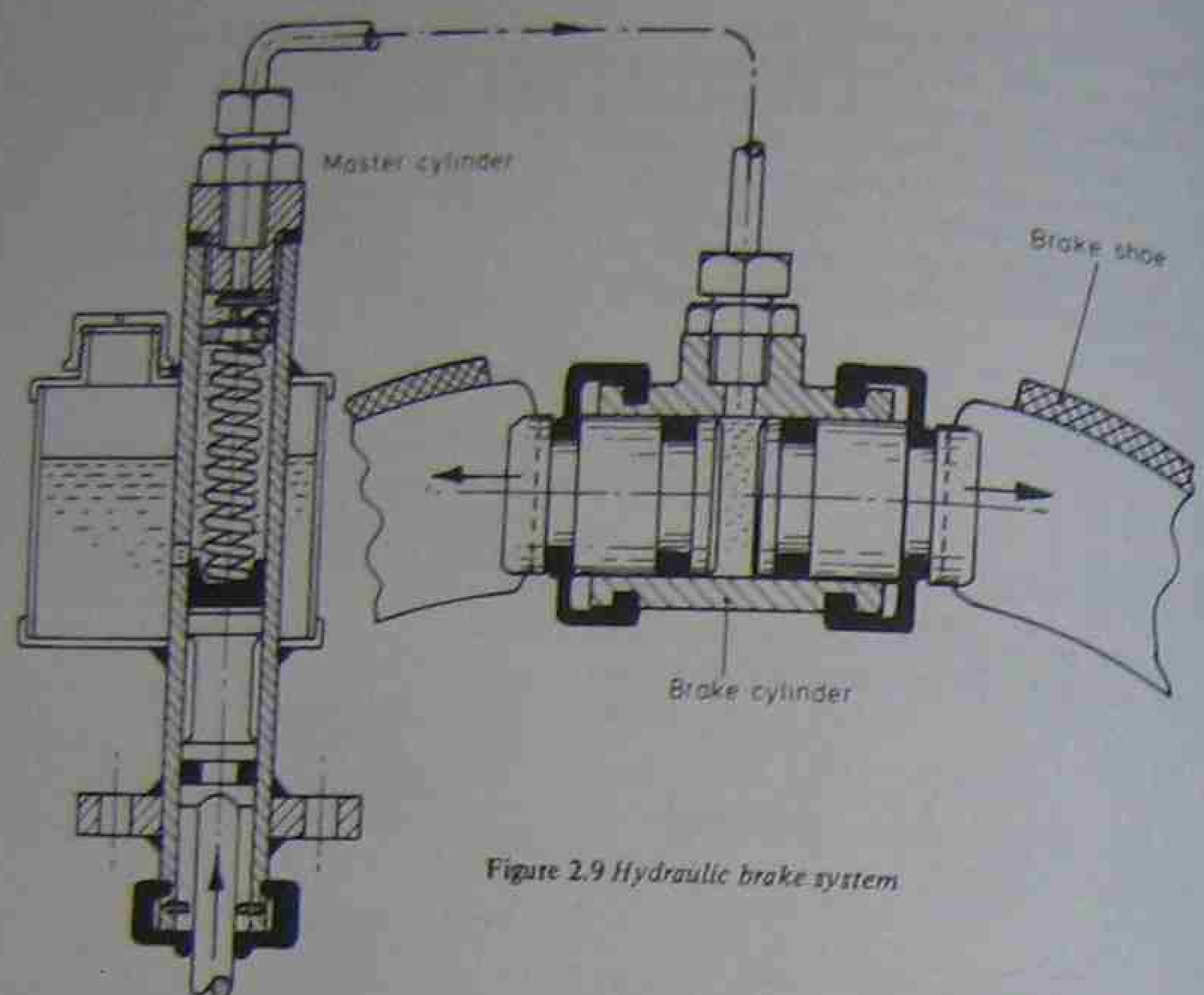


Figure 2.9 *Hydraulic brake system*

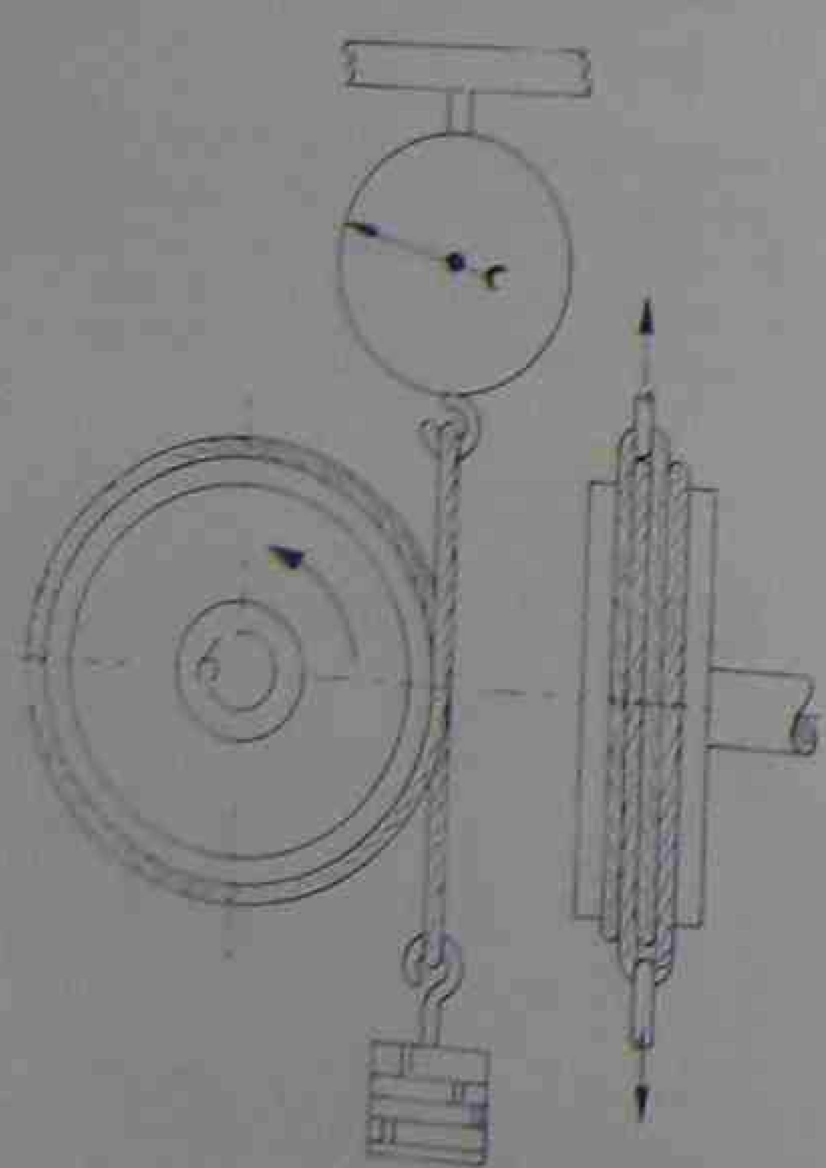


Figure 2.10 *Rope brake*

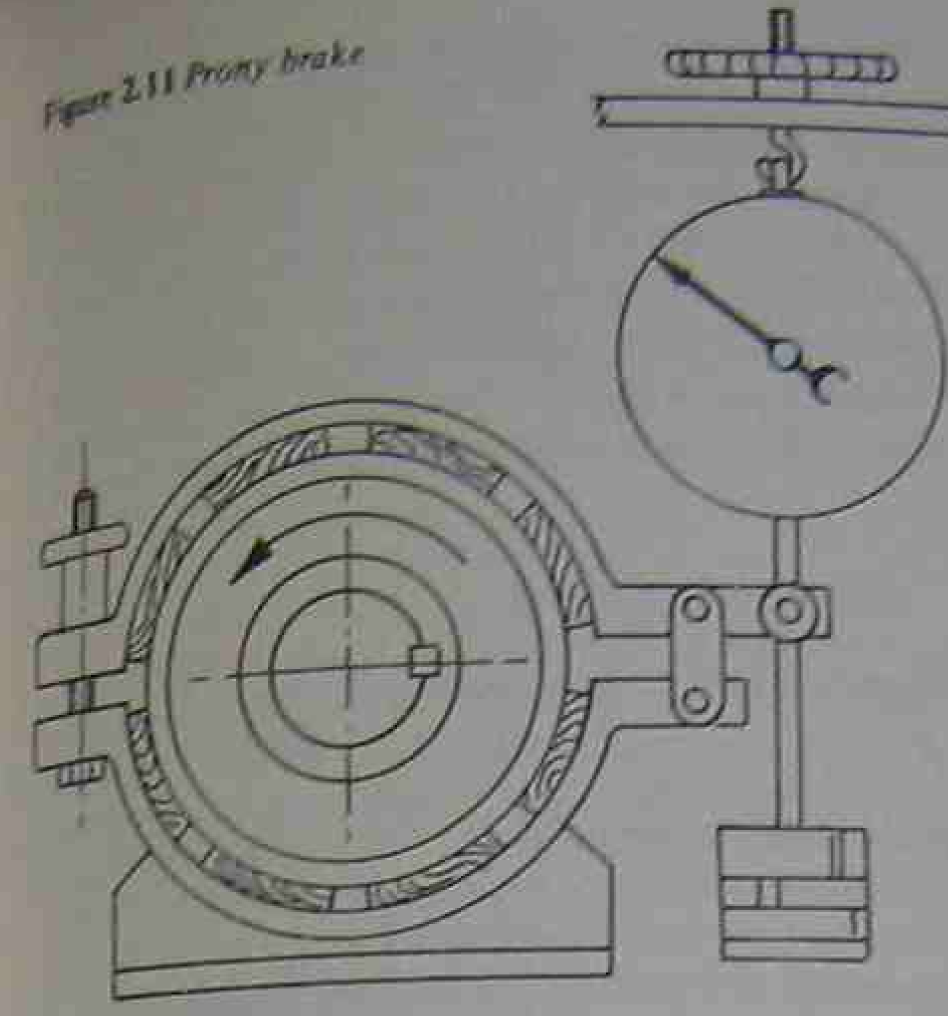
Power brake Brakes may be energised by electric solenoids, or by pistons using hydraulic or pneumatic pressure or a vacuum. In automobile brakes foot pressure applied to a master cylinder pressurises the brake fluid and this actuates pistons in contact with the shoes or pads.

Power absorption brake or dynamometer Brakes are used in the testing of engines to absorb and usually measure the power of a rotating shaft. These brakes, or *dynamometers* as they are called, may be of the friction type, hydraulic or electrical.

Rope brake or dynamometer (band brake) Rope of one or more strands, or a flexible belt, is looped around a drum and tightened to provide a friction force on the drum. The torque can be found by measuring the tangential force with a spring balance attached to the rope or belt.

Prony brake The Prony brake is used for the measurement of power in which friction blocks run on the outside of a drum. The torque on the block carrier is balanced by weights and a spring balance, or alternatively, by a load cell.

Figure 2.11 *Prony brake*



CLUTCHES

A clutch is a device which enables two shafts or rotating elements to be connected or disconnected while at rest or in relative motion.

Friction clutch In a friction clutch power is transmitted from one rotating element to another by means of the frictional force between them.

Single-plate friction clutch (disk clutch) This is a friction clutch in which the contacting surfaces consist of flat annular rings held together with sufficient force to prevent slipping when power is transmitted through the clutch. The force is usually supplied by one or more springs and the clutch is disengaged by a lever.

Clutch plate (clutch disk) A clutch plate is one of the plates in a friction clutch which is lined with a material having a high coefficient of friction when in contact with a metal surface. The friction material is either riveted or bonded to a metal backing plate.

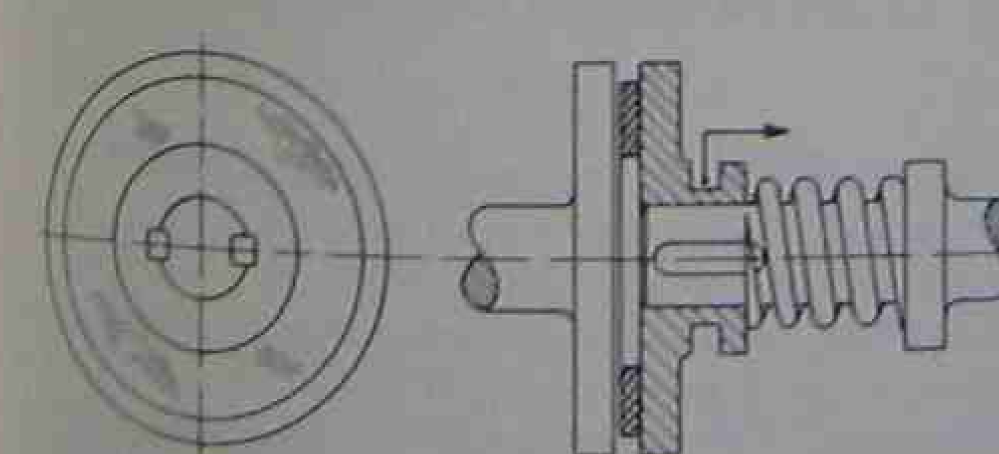


Figure 2.12 *Single-plate (disk) friction clutch*

Automobile friction clutch This clutch employs a plate with friction material on both sides which is clamped between two steel plates when the clutch is operated. The friction plate is carried on splines or keys on the output shaft so that it can slide axially, and one of the steel plates is attached to the input shaft. The other steel plate rotates with the input shaft and is spring loaded so that it normally holds the clutch in the engaged position. The clutch is disengaged by a lever moving a release sleeve on the shaft.

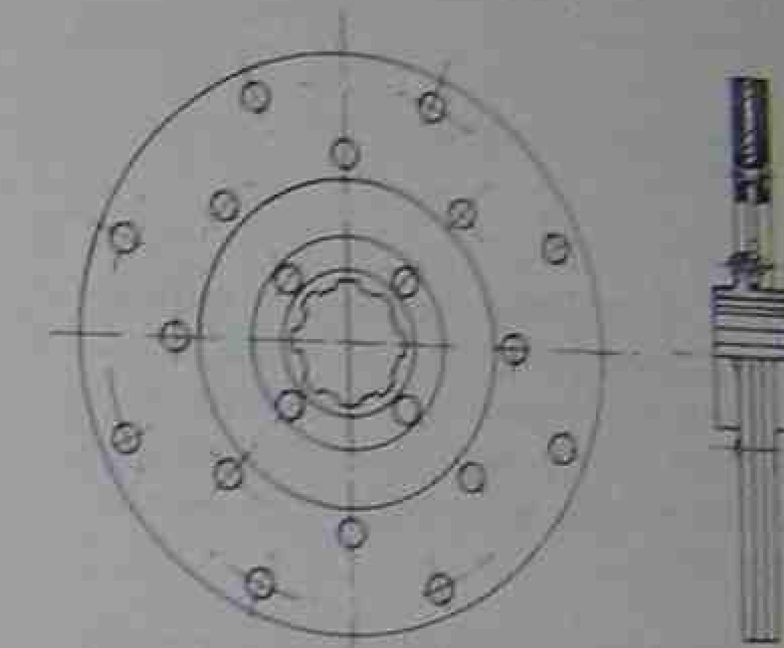


Figure 2.13 *Automobile-type clutch plate (disk)*

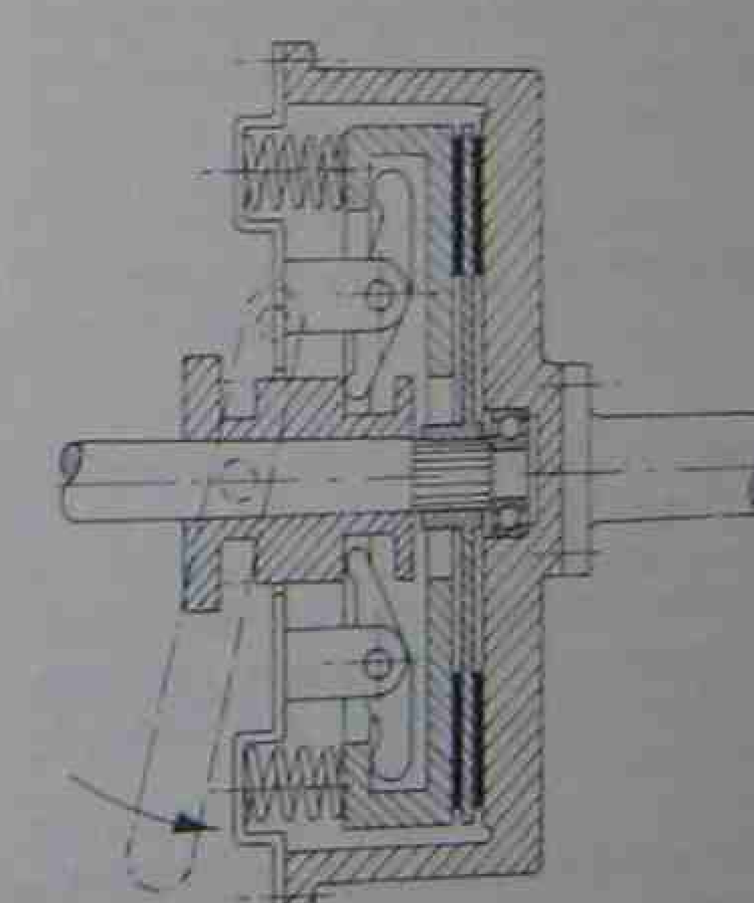


Figure 2.14 *Automobile friction clutch*

Multi-plate (disk) friction clutch The multi-plate automobile clutch has evolved from the single-plate clutch. Here several friction plates are keyed to one shaft and rotate with it. Interleaved with them are steel pressure plates keyed to the other shaft. All the plates are compressed together by a spring and released by a lever-actuated sleeve.

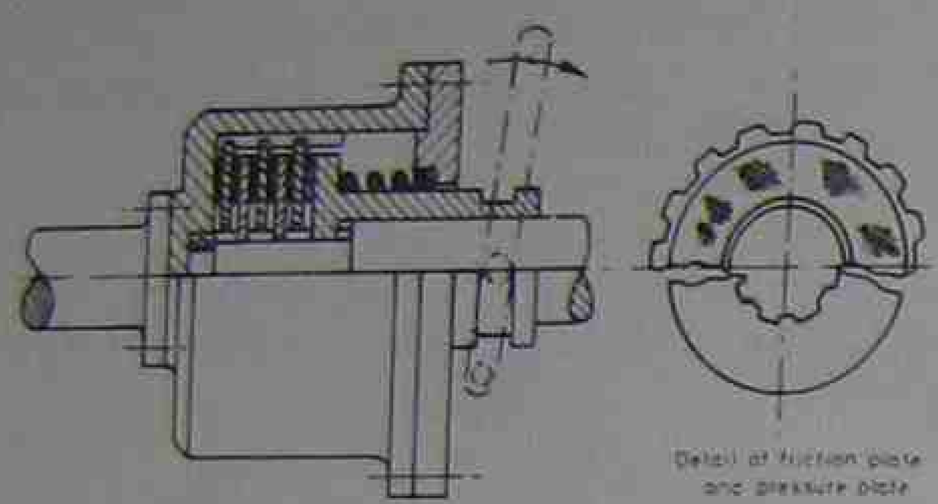


Figure 2.15 *Multi-plate (disk) friction clutch*

Cone clutch This has two conical mating surfaces one of which may be lined with a high-friction material. The torque capacity of a cone clutch is higher than that of a flat plate clutch of the same diameter. This increases as the angle of the cone to the axis decreases.

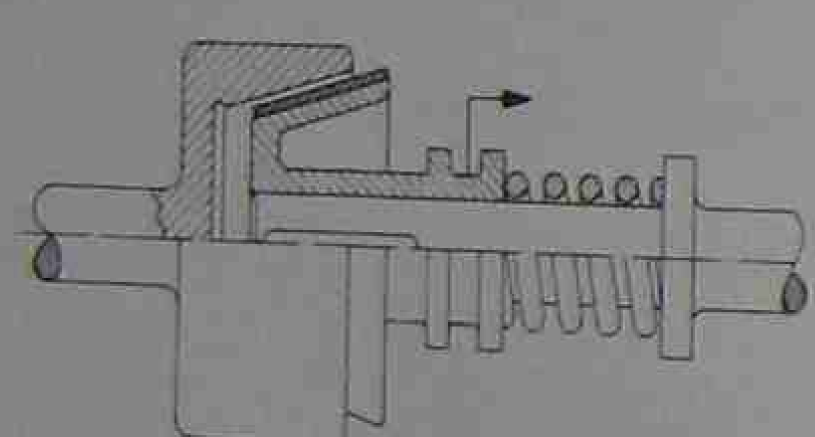


Figure 2.16 *Cone clutch*

Dog clutch (claw clutch) One half of this clutch is fastened permanently to one shaft while the other half slides on a feather key, or on splines, and is moved by a lever. Mating teeth on each half engage with one another and transmit the drive. The clutch can be operated only when stationary or moving at very low speeds.

Freewheeling or over-running clutch A freewheeling, or over-running, clutch permits a shaft to be driven in one direction of rotation only. It slips when the speed of the driven shaft exceeds that of the driving shaft.

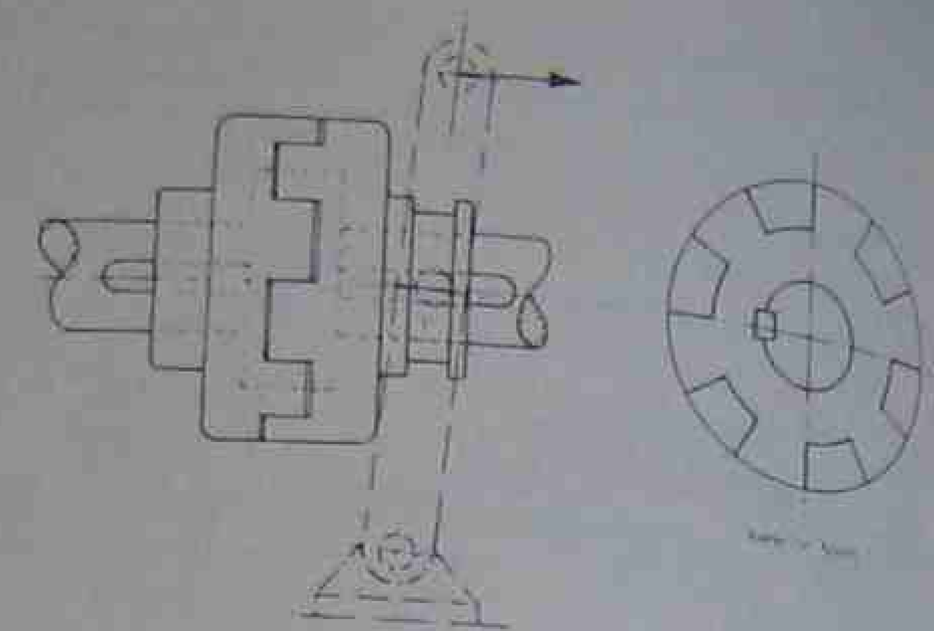


Figure 2.17 *Dog, or claw, clutch*

Roller freewheeling clutch Each roller is located in a wedge-shaped space between the inner and outer races. In one direction of rotation the rollers run up the wedge and lock to give a drive. In the other direction the rollers lie at the bottom of the ramp and the clutch slips.

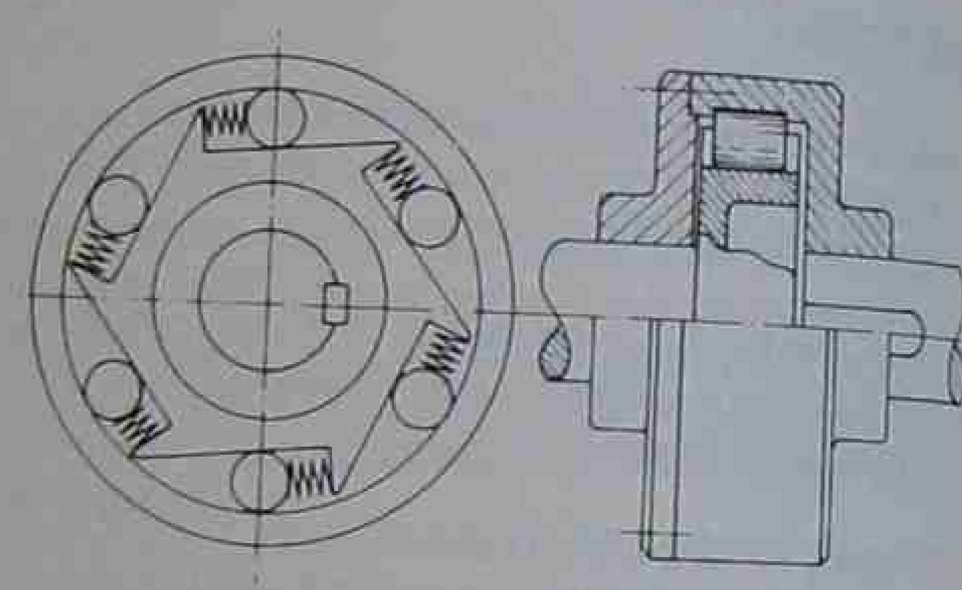


Figure 2.18 *Roller freewheeling clutch*

Sprag clutch Inner and outer races are connected to the input and output shafts. Wedge-shaped objects called *sprags* are situated in the annular space between the races. The sprags are shaped so that in one direction of rotation they wedge between the two races and the clutch is engaged. In the other direction the sprags tilt so that they are freed and the clutch slips.

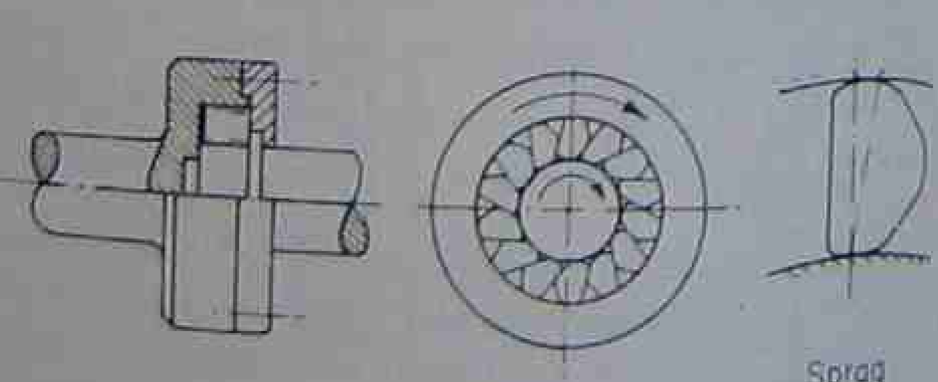


Figure 2.19 *Sprag clutch*

Centrifugal clutch A centrifugal clutch is used when the engagement of a prime mover with a load is to be achieved at a predetermined rotational speed. A typical design has spring-loaded weights mounted in radial slots in a member connected to the driving shaft.

The outer faces of the weights, which are faced with friction material, engage with the inner surface of a drum attached to the driven shaft at a speed which is determined by the tension on the springs (*see* Figure 2.8).

Dry powder clutch This is a type of centrifugal clutch in which metal particles, such as steel shot, are compacted under the action of the centrifugal force produced by rotation. The particles are contained in a hollow driving member in which a disk attached to the driven member rotates.

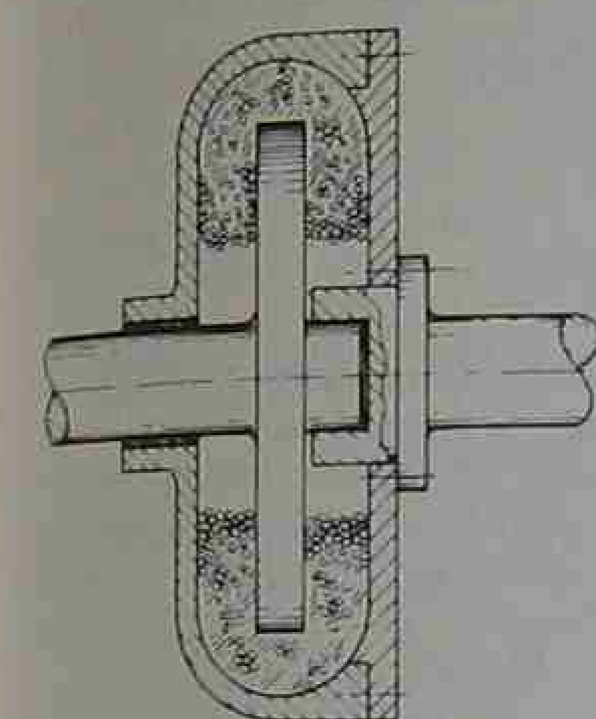


Figure 2.20 *Dry powder clutch*

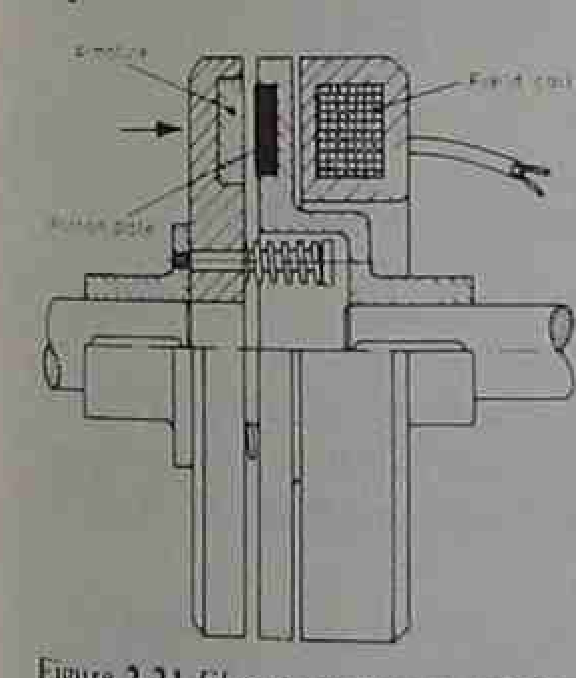


Figure 2.21 *Electromagnetic friction clutch*

Electromagnetic friction clutch In this clutch the output shaft carries a rotor with a friction facing. An armature in the form of a disk is driven by the input shaft and can move axially against springs. A field coil, which is either fixed or free to rotate with the output shaft, is energised to engage the clutch by producing a magnetic field which draws the rotor and armature together.

Fluid coupling (fluid flywheel) In a fluid coupling both input and output shafts carry impellers which have radial vanes. The vane spaces are filled with oil which circulates in the vanes when the coupling rotates. The input wheel acts as a pump and the output wheel as a turbine so that power is transmitted. There is always a loss of speed due to slip.

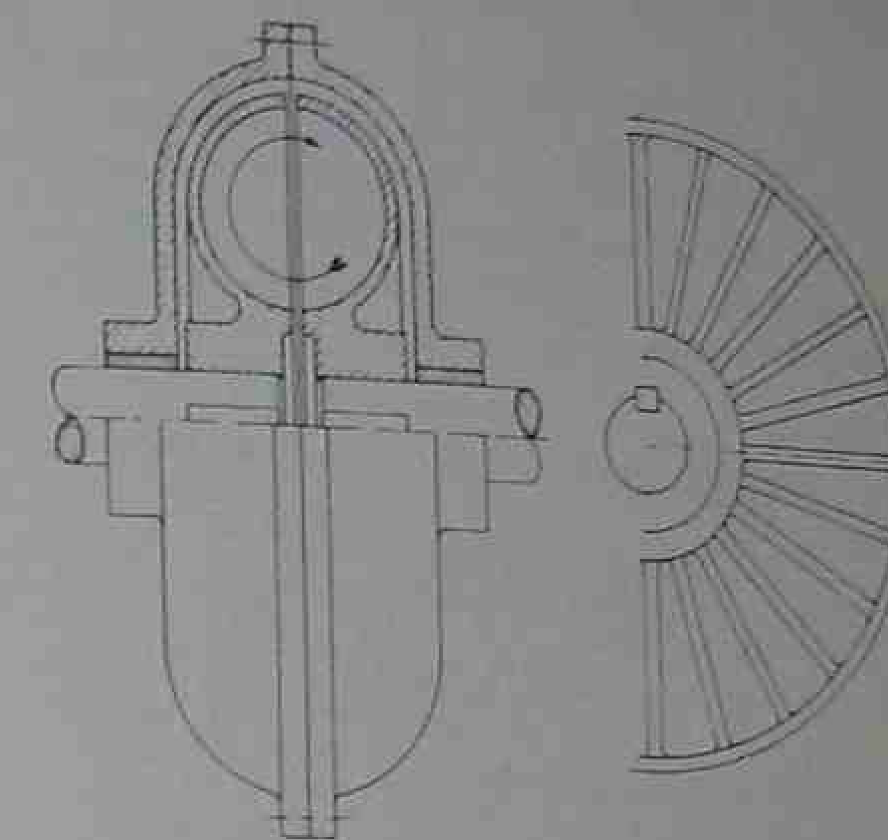


Figure 2.22 *Fluid coupling*

FRICTION MATERIALS FOR BRAKES AND CLUTCHES

Most brake and clutch linings are made of asbestos in a resin binder or rubber compound which will take a moderately high pressure. Powder metal on a steel backing is suitable for higher pressures and cork is used for light duty.

The coefficient of friction depends upon whether the lining is wet (with oil) or dry. Typical values are listed in Table 2.1.

Table 2.1

Material on Steel	*Coefficient of Friction* Dry	Wet
Cork	0.3	0.1
Asbestos in rubber compound	0.3–0.4	0.1
Asbestos in resin binder	0.3–0.4	0.1
Powdered metal	0.2–0.4	0.05–0.08

2.2 SHAFT COUPLINGS

A shaft coupling is a device which is used to connect two shafts together either rigidly, or flexibly when misalignment may be present.

SOLID BOLTED FLANGED COUPLING

Flanges on the shafts are rigidly connected by bolts. The flanges may be keyed to the shafts or integral with them. The shafts must be accurately lined up. Location is achieved by means of a spigot on one flange which fits into a recess in the other, or by using fitted bolts.

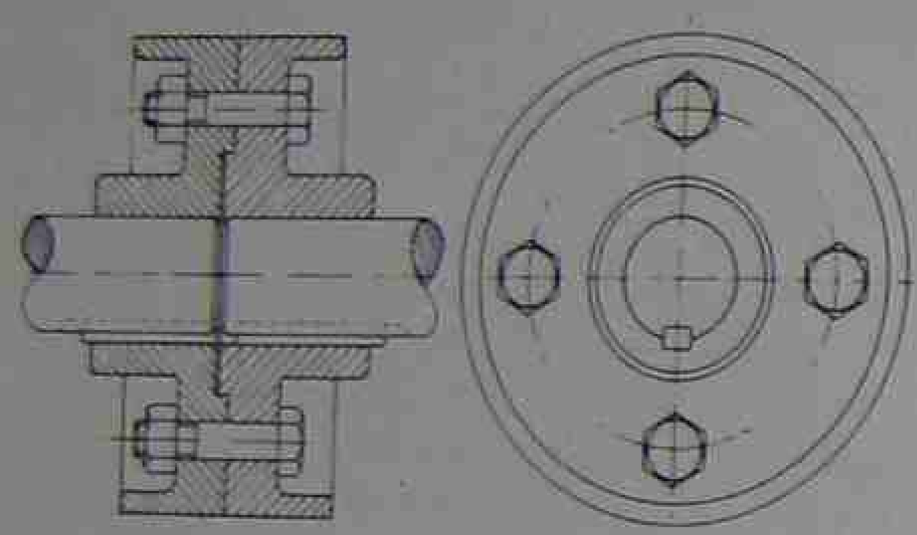


Figure 2.23 *Solid bolted flanged coupling*

MUFF COUPLING

Shafts may be rigidly connected by a muff coupling consisting of a sleeve halved longitudinally with one half keyed to both shafts. The halves of the sleeve are bolted together.

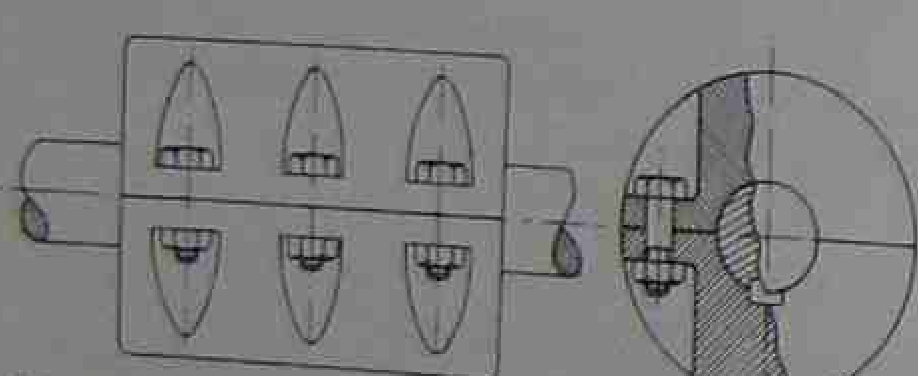


Figure 2.24 *Muff coupling*

COMPRESSION COUPLING

A split, double conical sleeve fits tightly over the shafts (which must have the same diameter) and is compressed onto the shafts by bolted flanges. The connection is made by friction between the sleeve and the shafts, and keys are unnecessary. This type of coupling can easily be dismantled without disturbing the shafts.

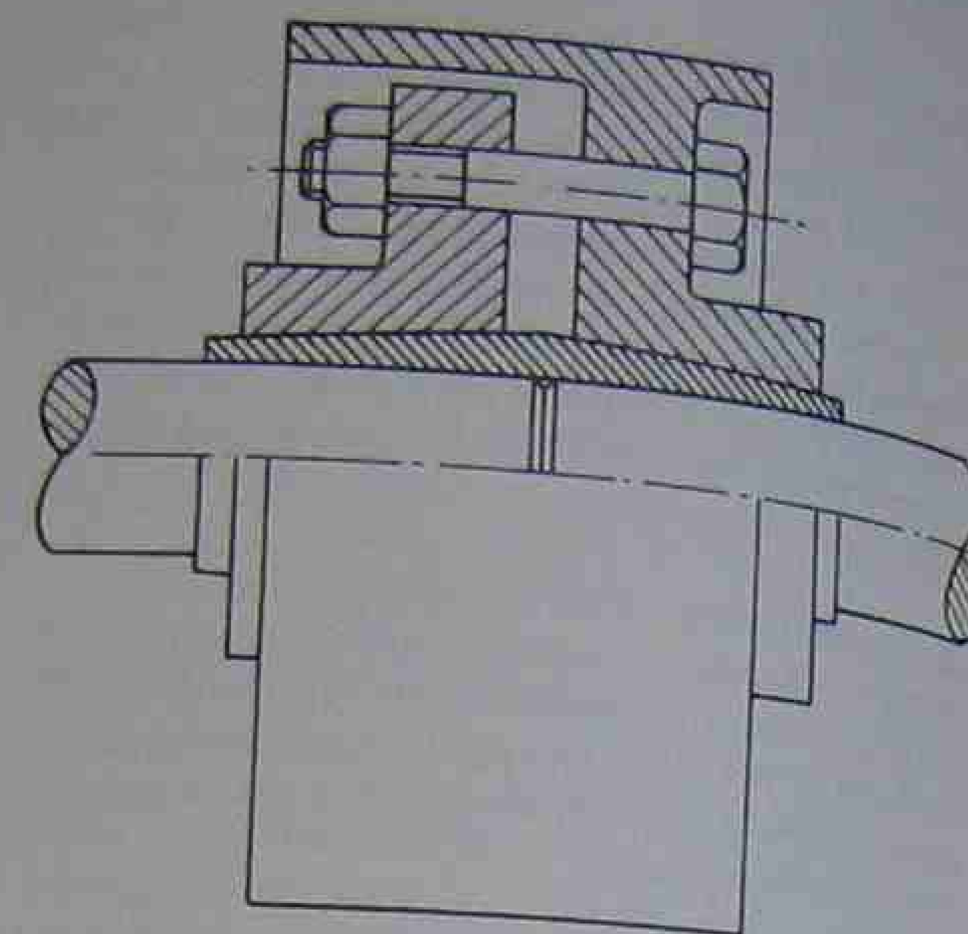


Figure 2.25 *Compression coupling*

CLAW COUPLING

Flanges keyed to the shafts have mating teeth cut on their faces which engage to give a drive.

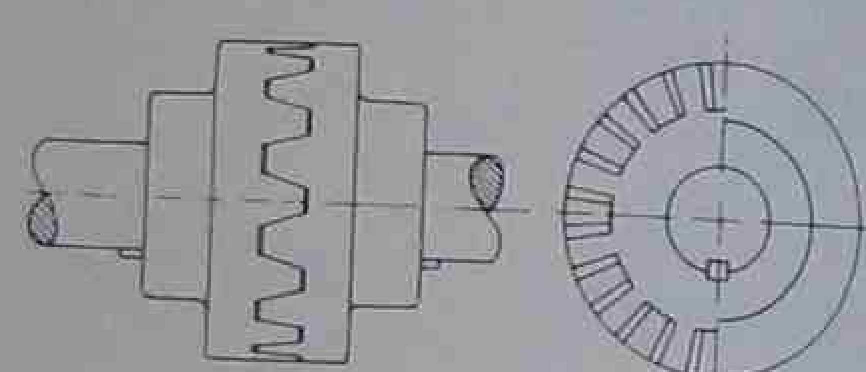


Figure 2.26 *Claw coupling*

SLEEVE COUPLING

This is a simple type of coupling consisting of a sleeve fitted over the two shafts and fastened to them by set screws or pins. It is used for light drives.

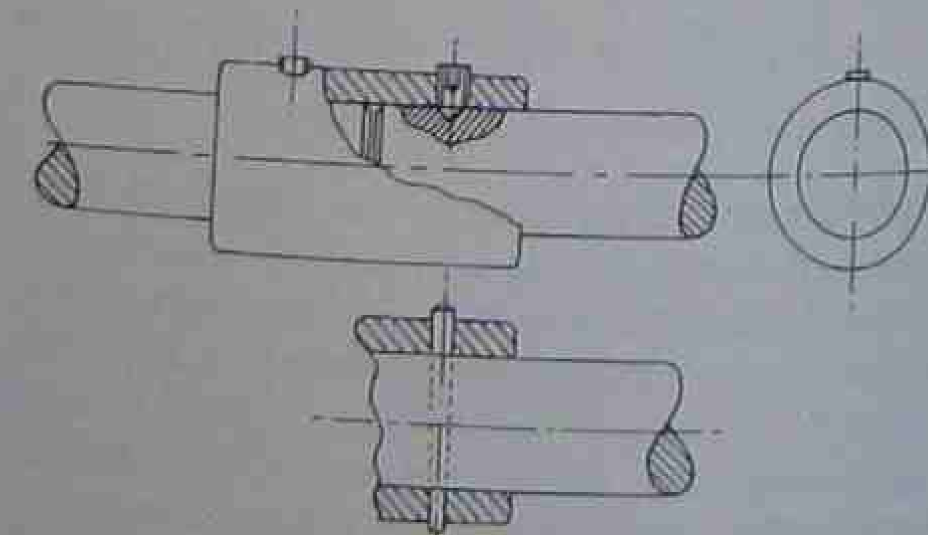


Figure 2.27 *Sleeve coupling*

FLEXIBLE SHAFT COUPLINGS

These couplings are designed to accommodate misalignment of shafts and also to help reduce torsional vibrations and absorb sudden torque variations.

Coupling design depends upon which of the following types of misalignment is present:–

Angular When the axes of the shafts are at an angle to one another

Parallel When the shafts are parallel but the axes are not coincident

Axial When the axes are coincident but there is relative axial movement.

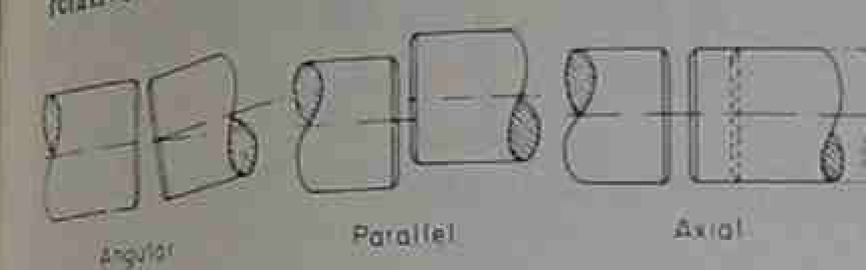


Figure 2.28 *Misalignments of shafts*

BONDED RUBBER COUPLING

A simple type of coupling for light loads in which steel sleeves, keyed to the shaft and locked with screws, are bonded to a rubber or synthetic rubber section.

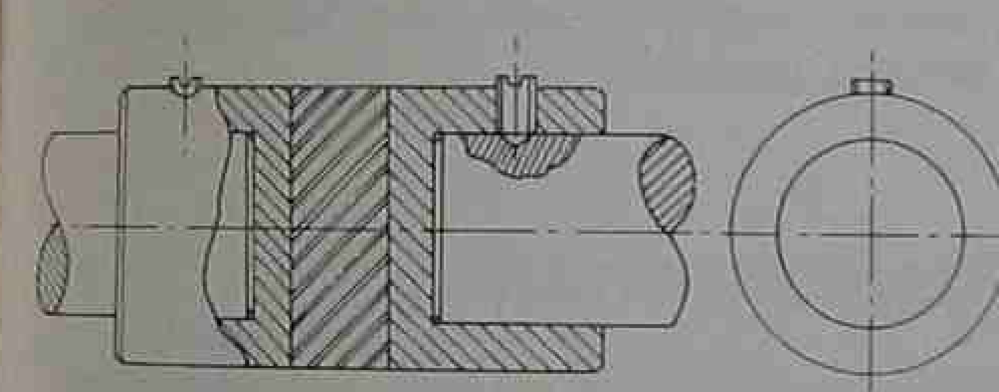


Figure 2.29 *Bonded rubber coupling*

RUBBER-BUSHED PIN-TYPE COUPLING

This is a bolted flange coupling where the bolts are rigidly attached to one flange but pass through rubber bushes set in the holes in the other flange. It is used mainly for angular misalignment.

DISK-TYPE FLEXIBLE COUPLING

A flanged coupling which has a rubber disk, bonded between two steel disks, bolted to the flanges through alternate holes.

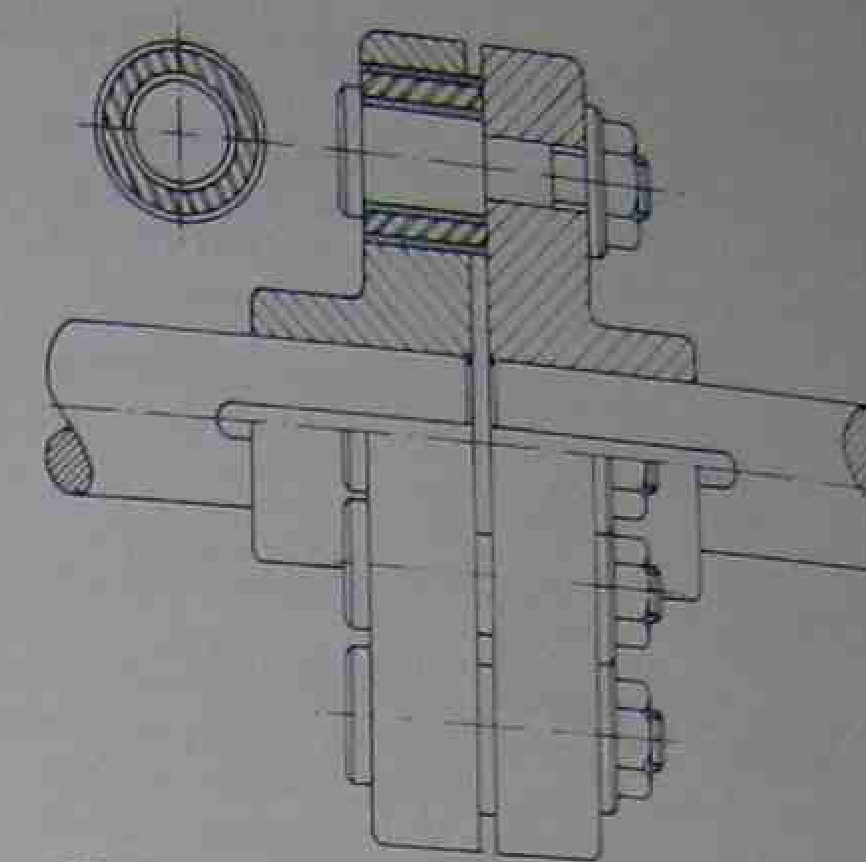


Figure 2.30 *Rubber-bushed pin-type flexible coupling*

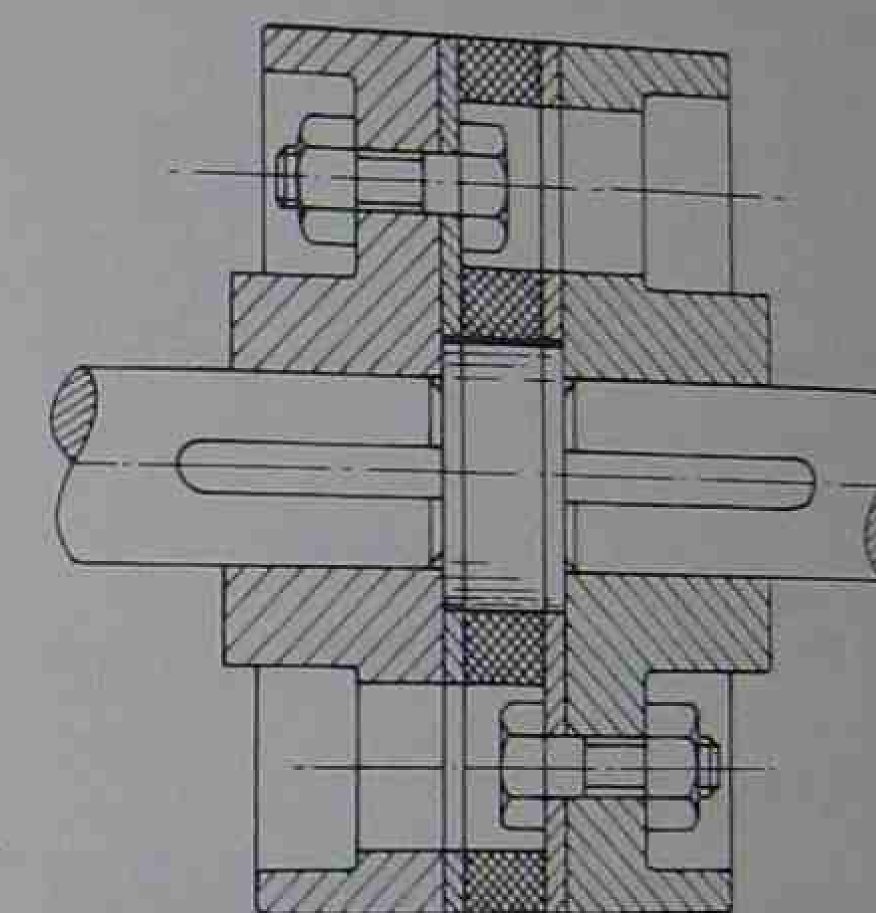


Figure 2.31 *Disk-type flexible coupling*

MOULDED RUBBER INSERT COUPLING

Flanges keyed to the shafts have projections which fit into moulded recesses in a rubber insert through which the drive is transmitted.

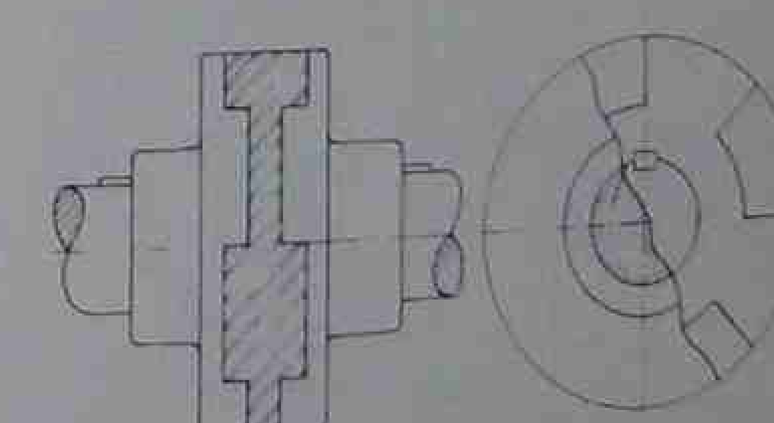


Figure 2.32 *Moulded rubber insert coupling*

RUBBER-TYRE-TYPE FLEXIBLE COUPLING

In this coupling the flanges on the shafts are connected by a rubber tyre clamped to each flange. The coupling, which is made under the names Fenaflex and Periflex, gives high flexibility and shock-absorbing capacity.

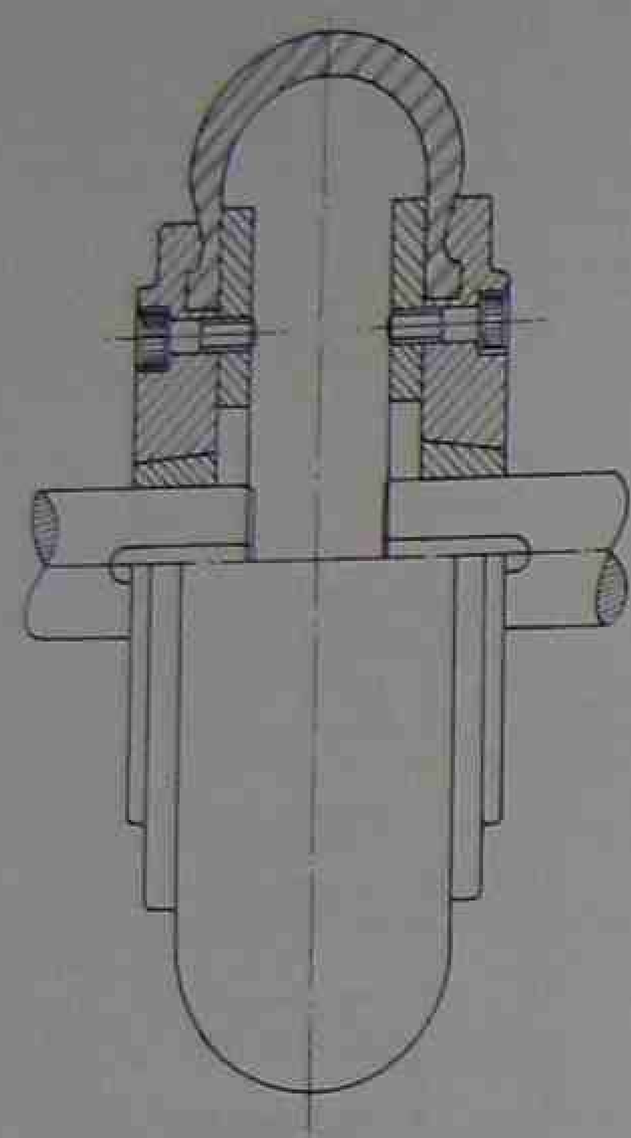


Figure 2.33 *Rubber-tyre-type flexible coupling*

GEAR COUPLING

Flanges keyed to the shafts have involute teeth cut on their periphery which engage with mating internal teeth at each end of a sleeve packed with lubricant.

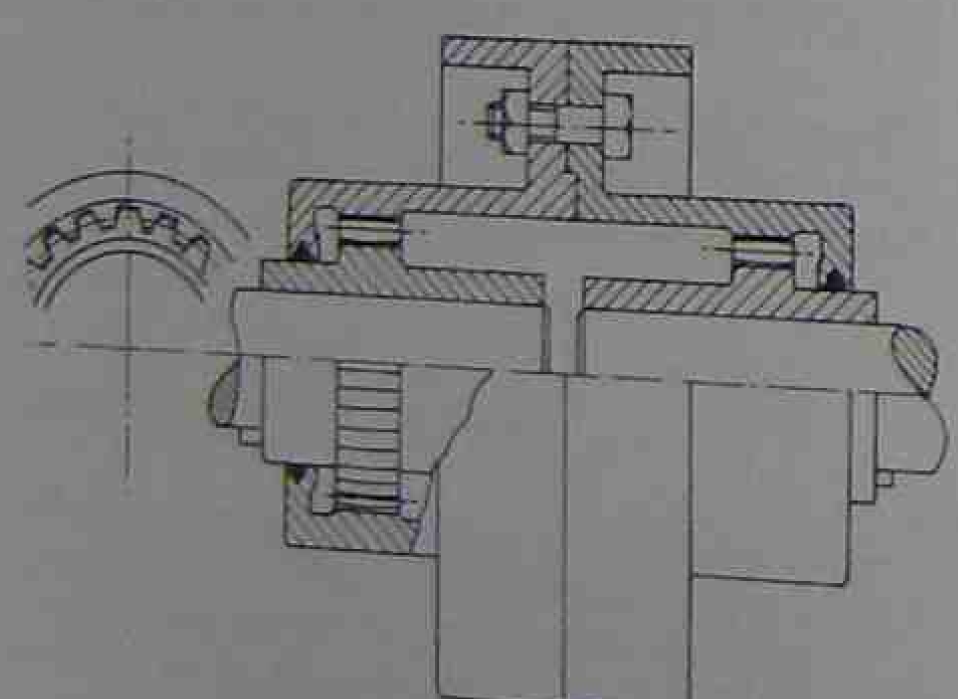


Figure 2.34 *Gear coupling*

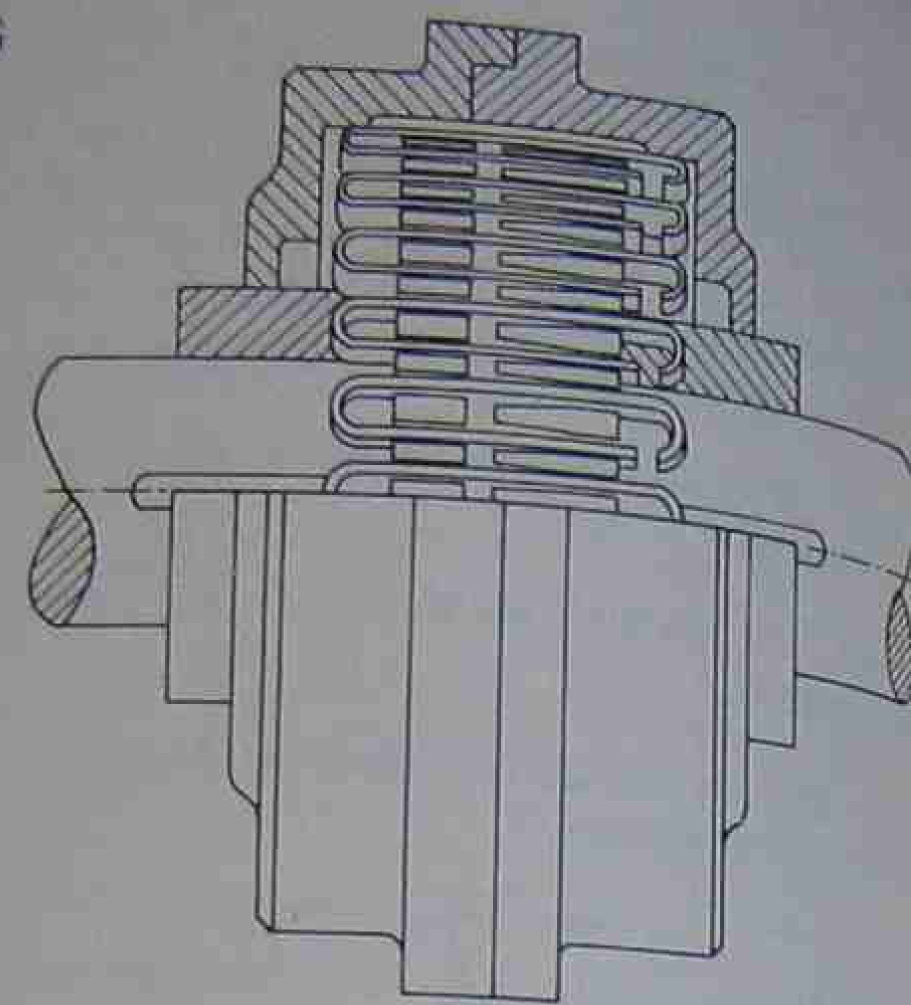


Figure 2.35 *Metal spring coupling*

Parallel and angular misalignment are accommodated but there is no torsional flexibility.

METAL SPRING COUPLING

Loops of spring steel, set in axial slots in the shaft flanges, transmit the drive and provide angular and torsional flexibility.

OLDHAM COUPLING

Steel flanges keyed to the shafts have diametral keys on their faces which mate with two corresponding slots set at right angles to one another on a disk between them. The disk is usually made of brass. This coupling is suitable for parallel misalignment only.

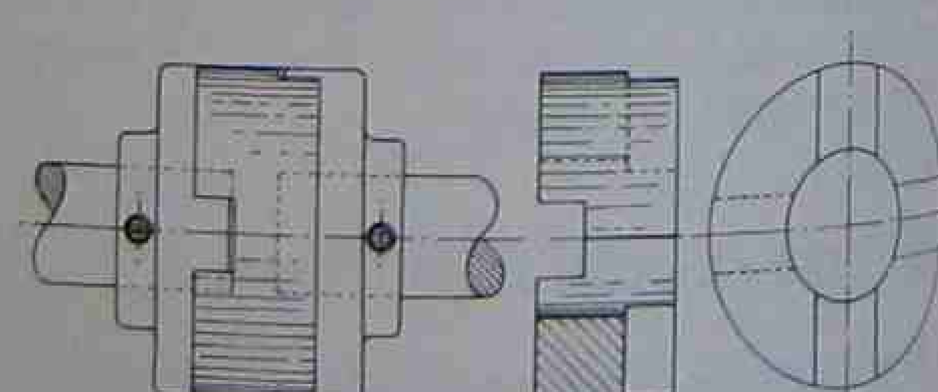


Figure 2.36 *Oldham coupling*

METAFLEX COUPLING

This is a flexible coupling in which the shaft flanges are connected through sets of thin steel laminations which provide longitudinal and angular flexibility but not flexibility in torsion. Parallel misalignment can be accommodated if two sets of laminations are used.

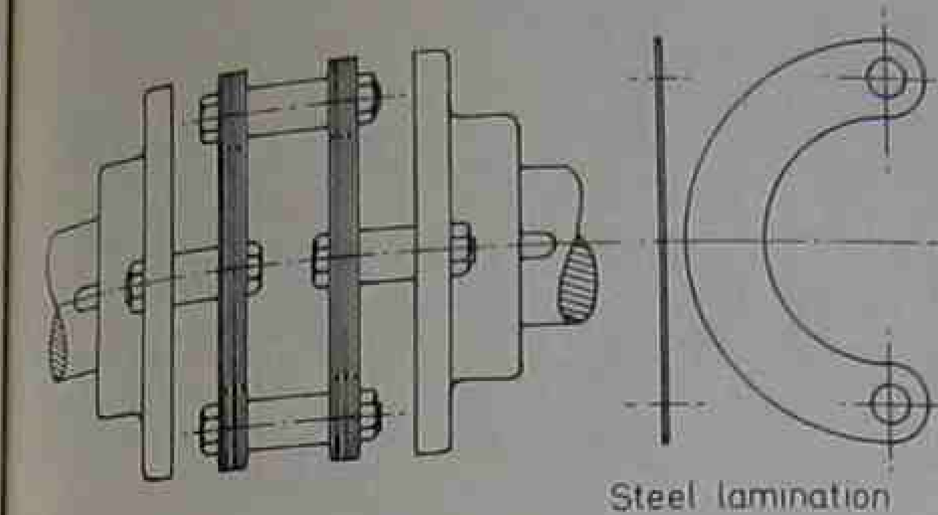


Figure 2.37 *Metaflex coupling*

HOOKE'S-TYPE UNIVERSAL JOINT

This joint provides a coupling which will accept up to about 20° of angular misalignment. It consists of a fork on each shaft joined by a cross-piece with bearings. Two joints, one at each end of a shaft, will accommodate a large parallel misalignment.

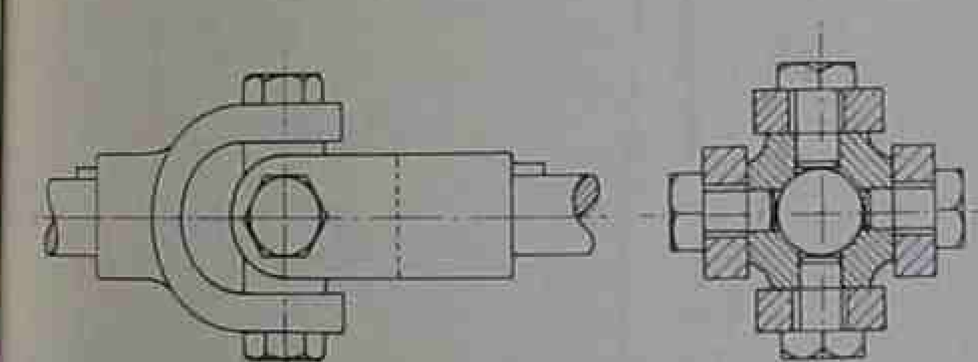


Figure 2.38 *Hooke's-type universal joint*

CONSTANT-VELOCITY UNIVERSAL JOINT

The Hooke's joint suffers from the disadvantage of speed fluctuations which increase with angular misalignment. Constant-velocity joints, of which there are several types, overcome this problem. In the type shown the inner and outer races have circular arc grooves which are curved and in line with the shaft axis. A ball in each groove transmits the drive and allows angular misalignment of the shafts.

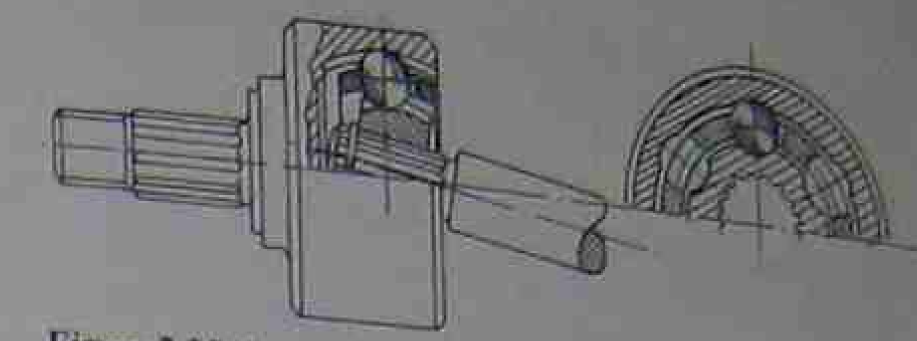


Figure 2.39 *Constant-velocity universal joint*

AUTOMOBILE-TYPE UNIVERSAL JOINT

This is a development of the Hooke's joint which is used in pairs to transmit power to the road wheels of vehicles. The bearings have needle rollers and the forks are mounted on splined shafts to allow for changes in length between joints.

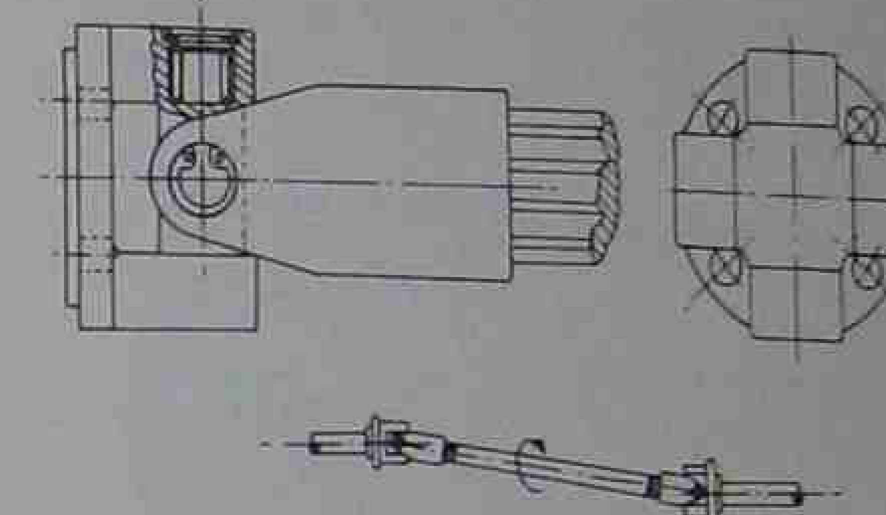


Figure 2.40 *Automobile-type universal joint*

CHAIN COUPLING

The flanges are provided with sprockets which take a ring of duplex (or double) roller chain which connects the flanges. A rubber cover is usually fitted.

This coupling has only a slight amount of flexibility.

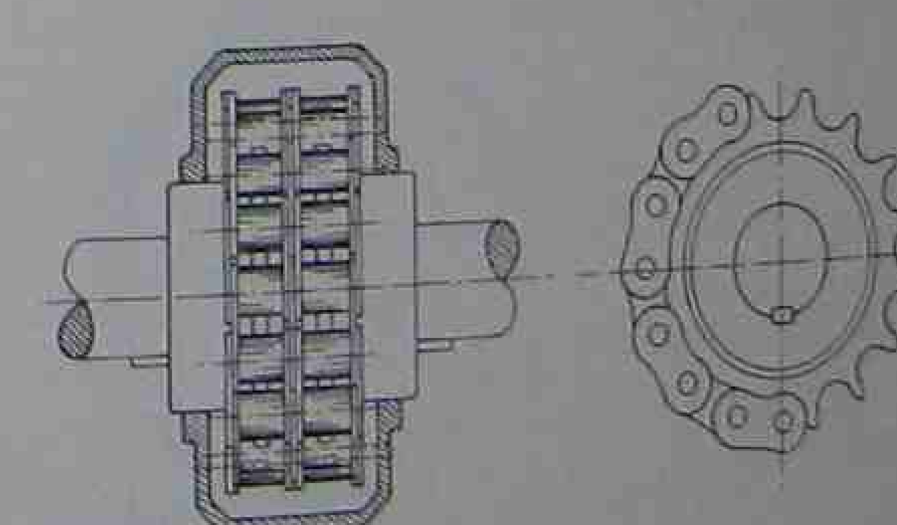


Figure 2.41 *Chain coupling*

2.3 BELT, ROPE AND CHAIN DRIVES

BELT DRIVE

A belt drive is used to transmit rotation from one shaft to another by means of a flexible belt running on pulleys on the shafts. The belt is usually flat or of V section, and it is tensioned to prevent slipping. In general, the pulleys have different diameters so that either an increase or a decrease in speed is obtainable.

FLAT BELT DRIVE

Flat belts are made of leather, reinforced rubber, impregnated canvas or woven fabric and belt speeds of 30 m/sec are possible. To prevent the belt from slipping sideways off the pulley, the latter may be cambered or have flanges. A flat belt may be twisted to give a drive through a right angle or a reversal of the direction or rotation.

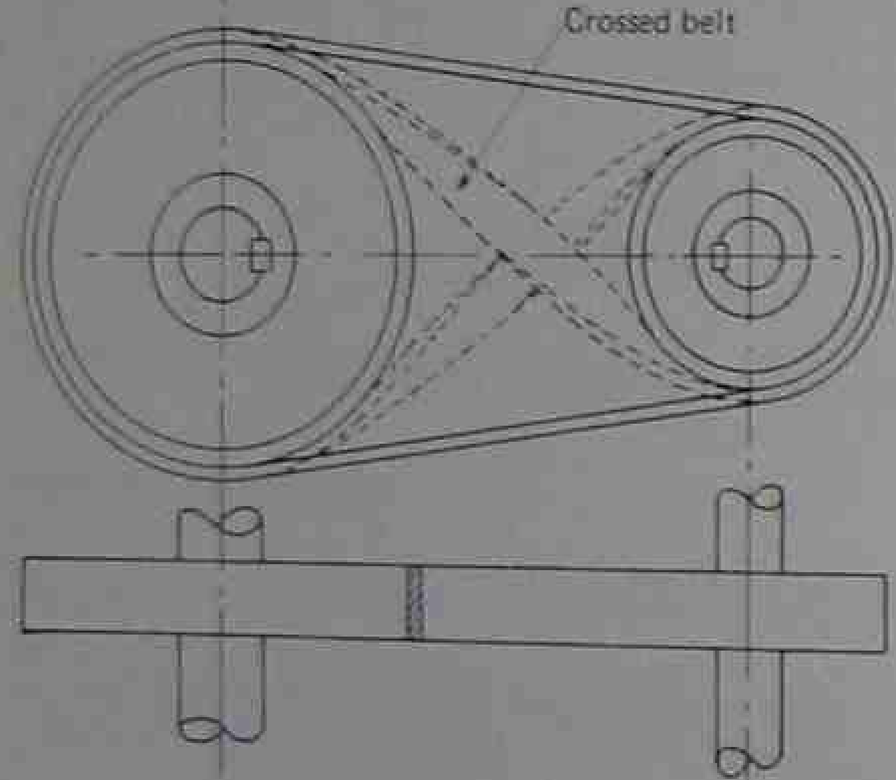


Figure 2.42 *Flat belt drive*

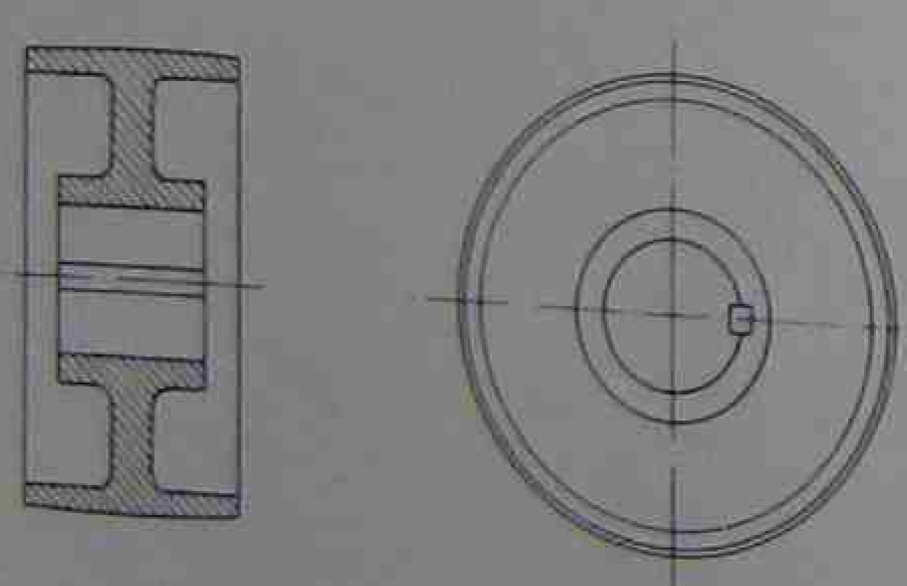


Figure 2.43 *Cast pulley for flat belts*

STEPPED PULLEY DRIVE

Stepped pulleys are used in pairs to provide a selection of speed ratios. In effect, each of the two pulleys consists of several pulleys of different diameters joined together, with the two pulleys being mounted in opposite ways on the shafts. The speed ratio can be changed while running.

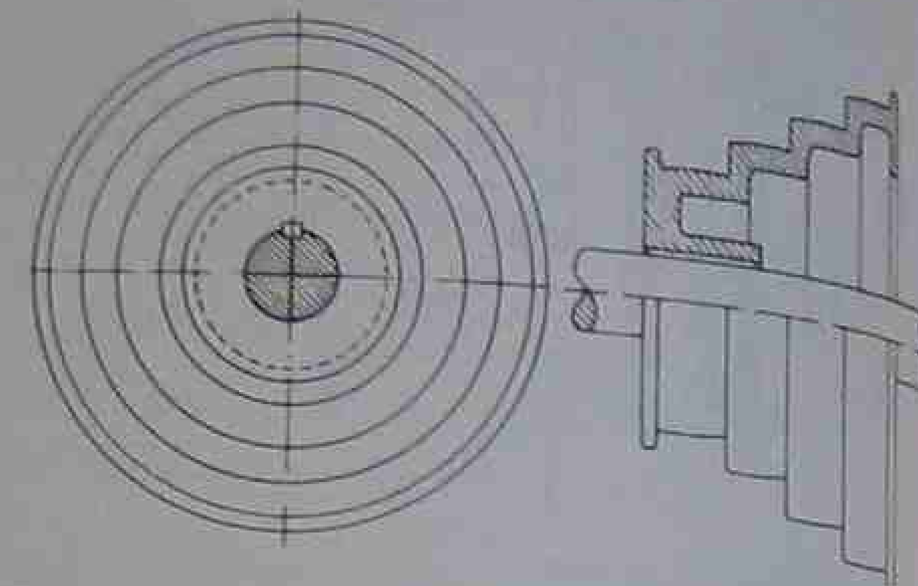


Figure 2.44 *Stepped pulley (flat belt) drive*

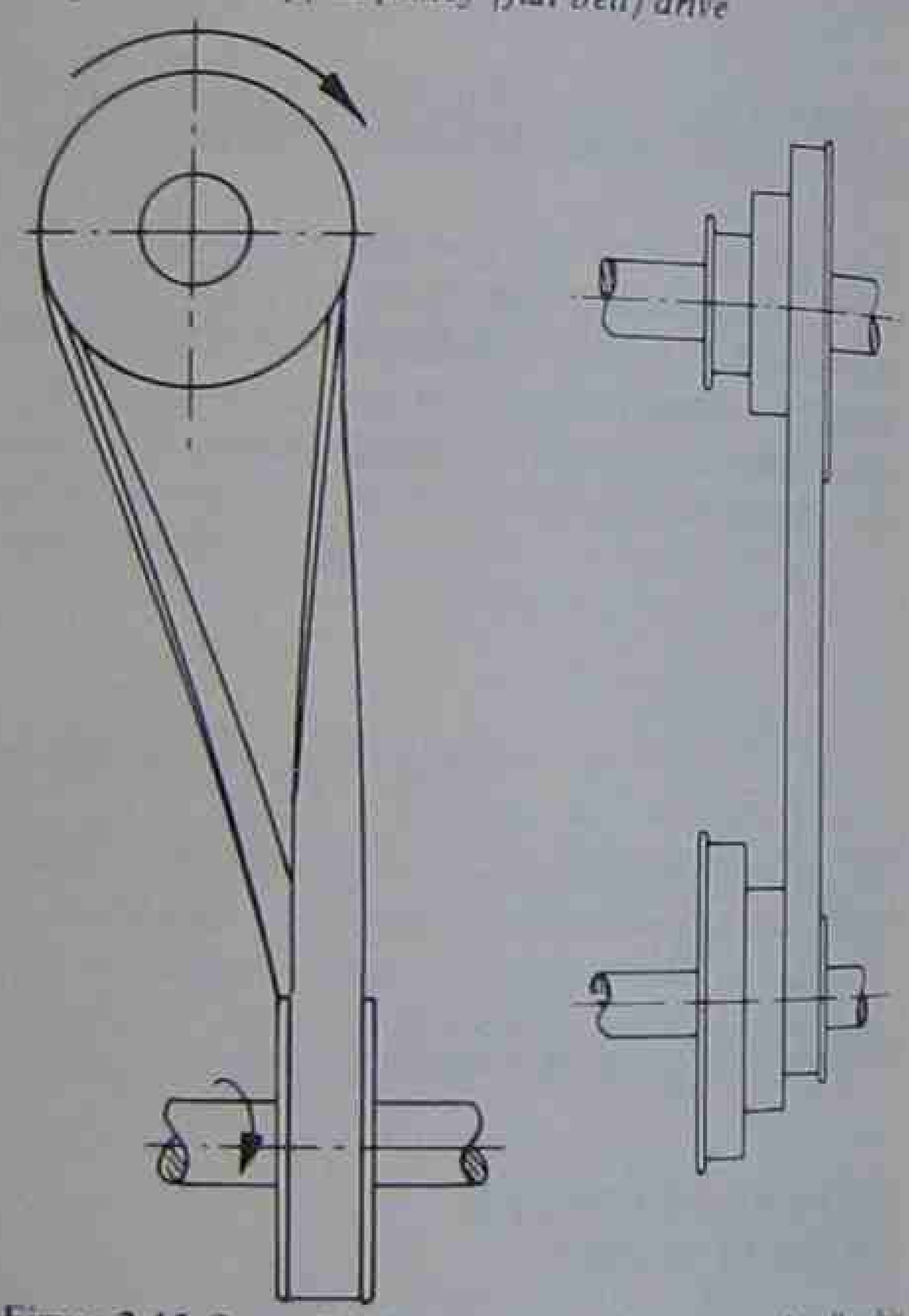


Figure 2.45 *Crossed belt drive (left), and stepped pulley drive*

V BELT DRIVE

A V belt has certain advantages over the flat belt. It is more compact than the flat belt and because of its wedge action it can transmit about three times the power for the same belt strength. It has a tapered cross-section with an angle of about 40° and this fits into a groove of the same shape in the rim of the pulley. V belts are made of rubber in which are embedded nylon cords encased in rubber-impregnated woven cotton. They are used for electric motor drives, automobile fan belts, etc.

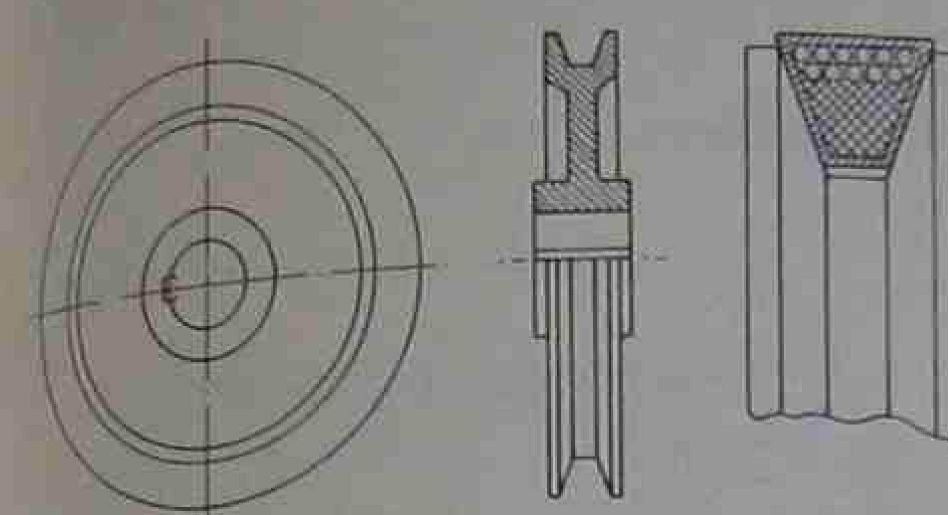


Figure 2.46 *V pulley and section through V belt*

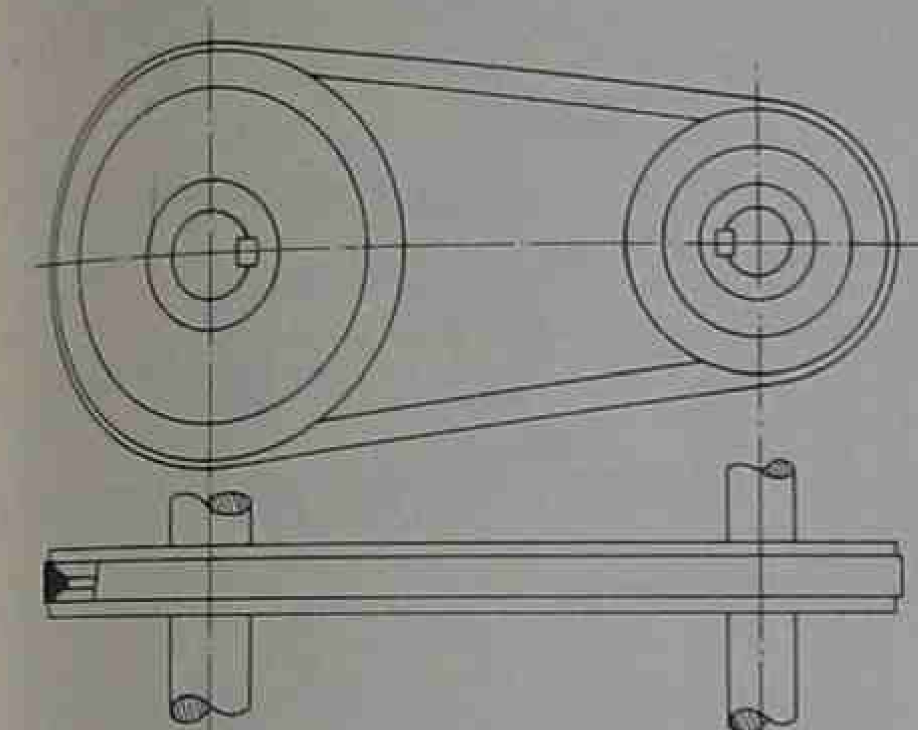


Figure 2.47 *V belt drive*

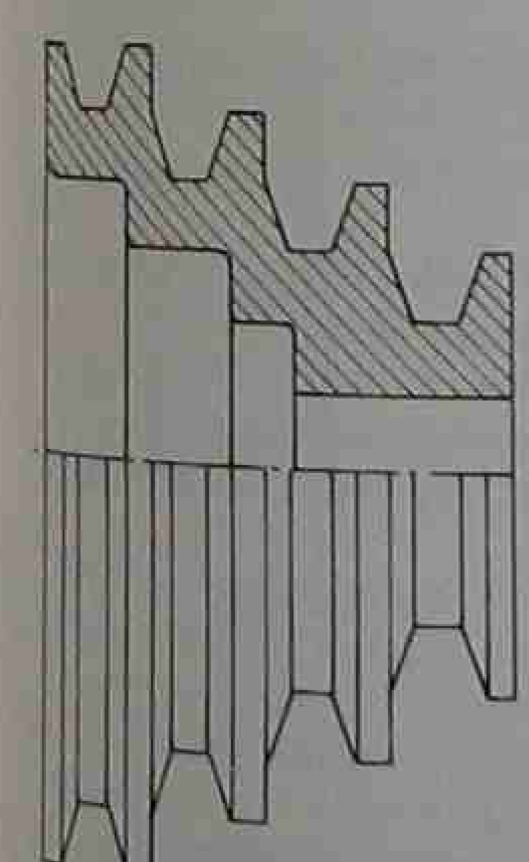


Figure 2.48 *Stepped V pulley*

Stepped V belt drive Stepped V pulleys are available to permit speed selection, but to allow alteration of the belt position the drive must be stopped.

Multiple V belt drive To permit higher powers to be transmitted the multiple V belt drive is used. In this type each pulley has several grooves of the same diameter with a belt in each groove, and this variation is safer in the event of belt failure.

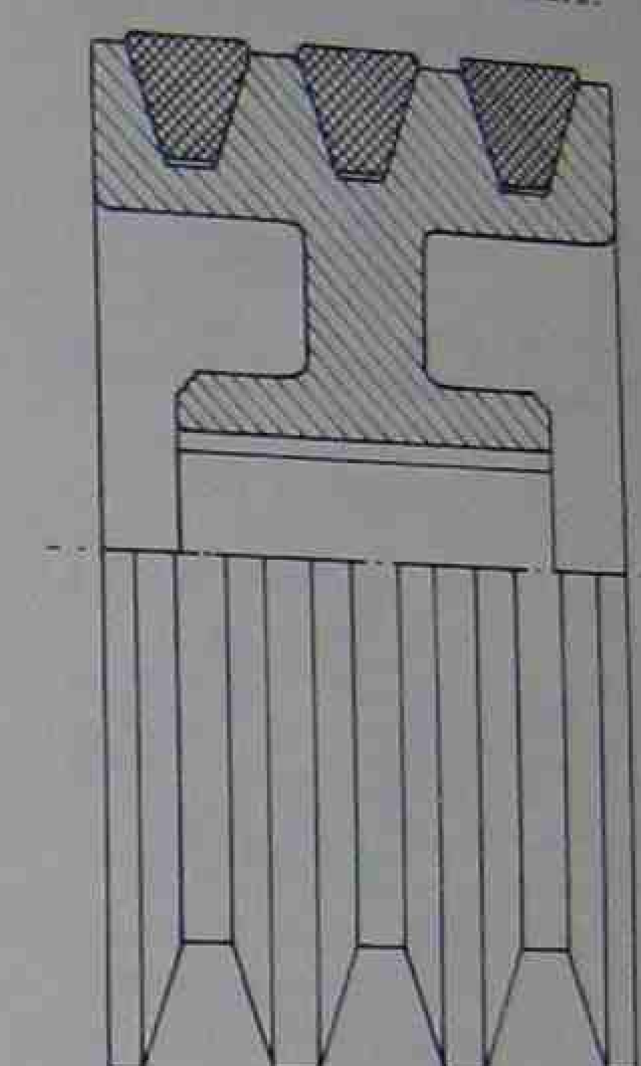


Figure 2.49 *Multiple V pulley*

Ribbed belt drive This is an alternative to the multiple V belt in which a single belt with longitudinal triangular cross-section ribs is employed.

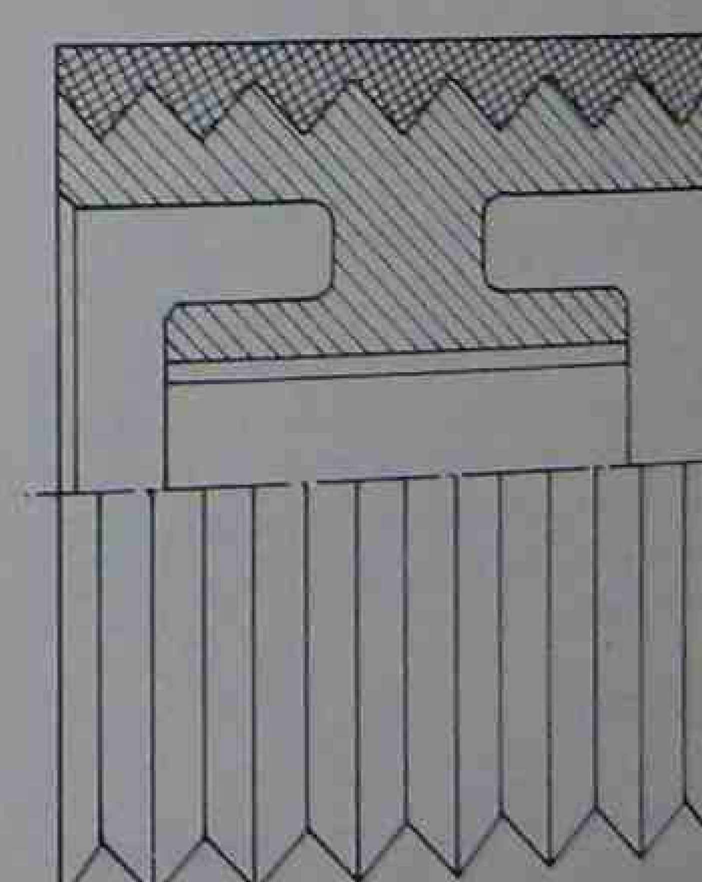


Figure 2.50 *Ribbed belt drive*

VARIABLE SPEED BELT DRIVE

This is a parallel belt drive in which the speed ratio can be altered while running by changing the effective diameters of the pulleys. Each shaft carries two coned pulleys with the narrow ends of the cones facing one another. One cone of each pulley slides on keys on the shaft and a wide V belt connects the pulleys. The speed is varied by moving one pair of cones closer together and the other pair further apart, or vice versa. The pulleys may be grooved to prevent possible slip.

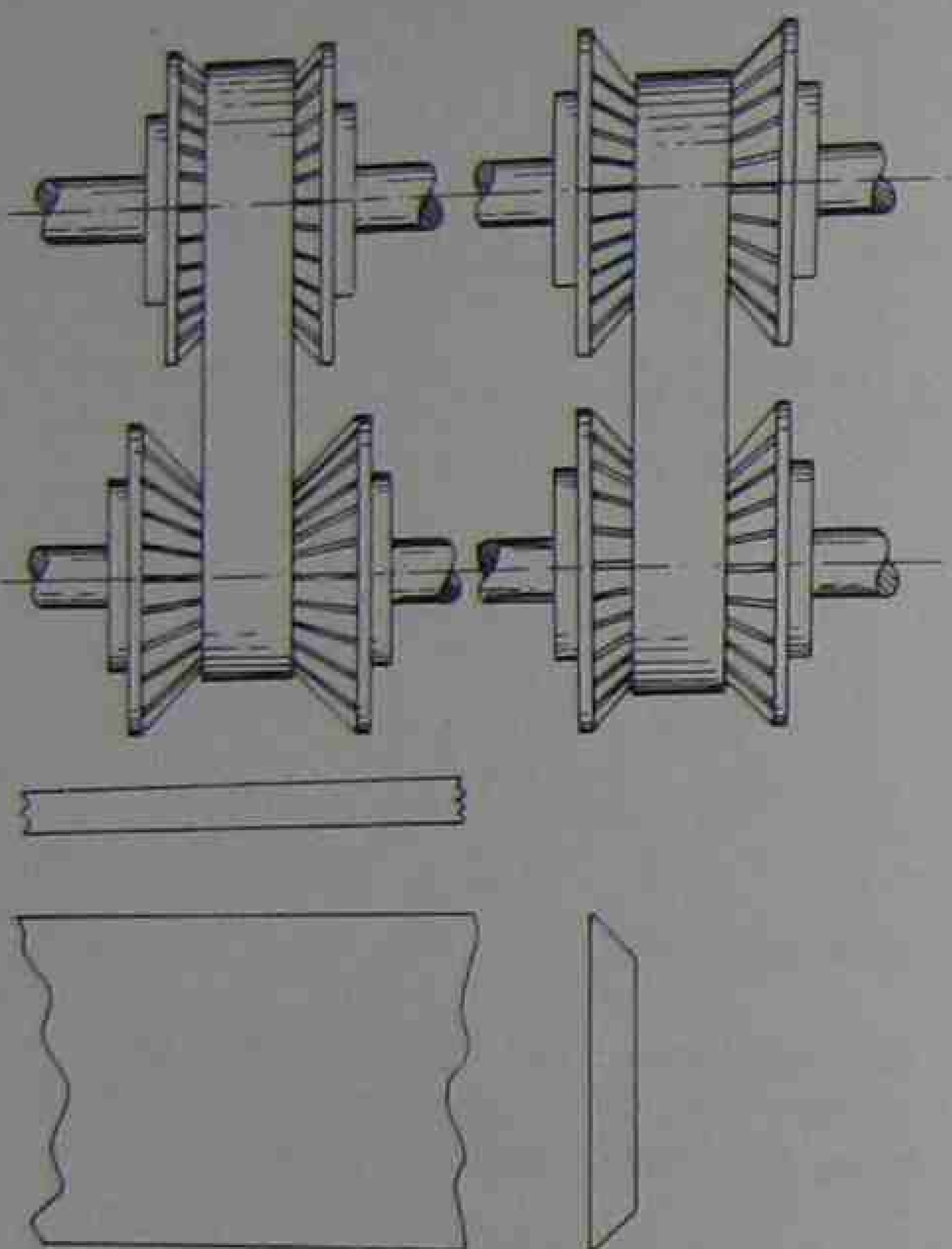


Figure 2.51 *Variable-speed belt drive and belt*

TIMING BELT DRIVE

A timing belt has transverse teeth on the inside surface which mate with grooves on the pulleys. The belt has steel wire reinforcement which enables it to transmit about three times the power at three times the speed of a conventional belt. There is no slip and exact speed ratios are maintained.

BELT CONVEYOR

This is a device for transporting loose material and machine parts using a wide flat belt running on rollers.

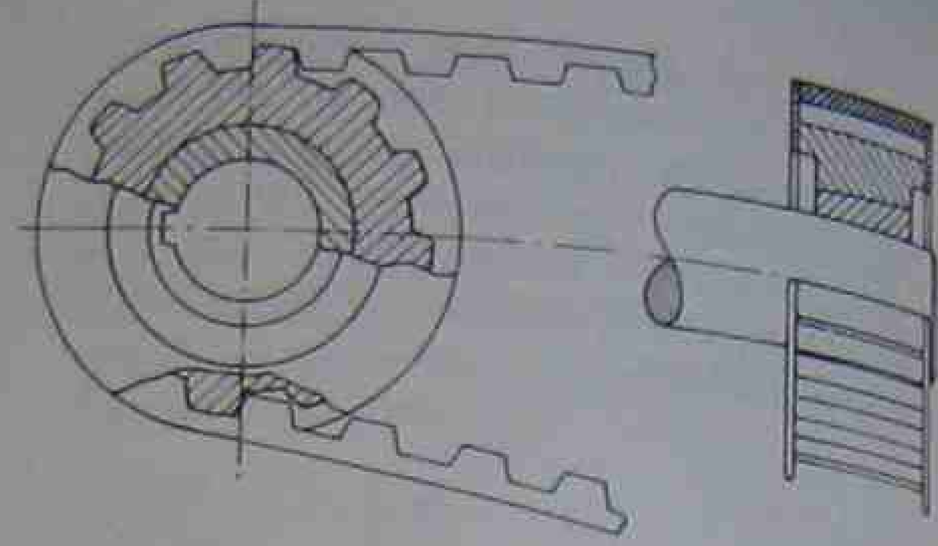


Figure 2.52 *Timing belt drive*

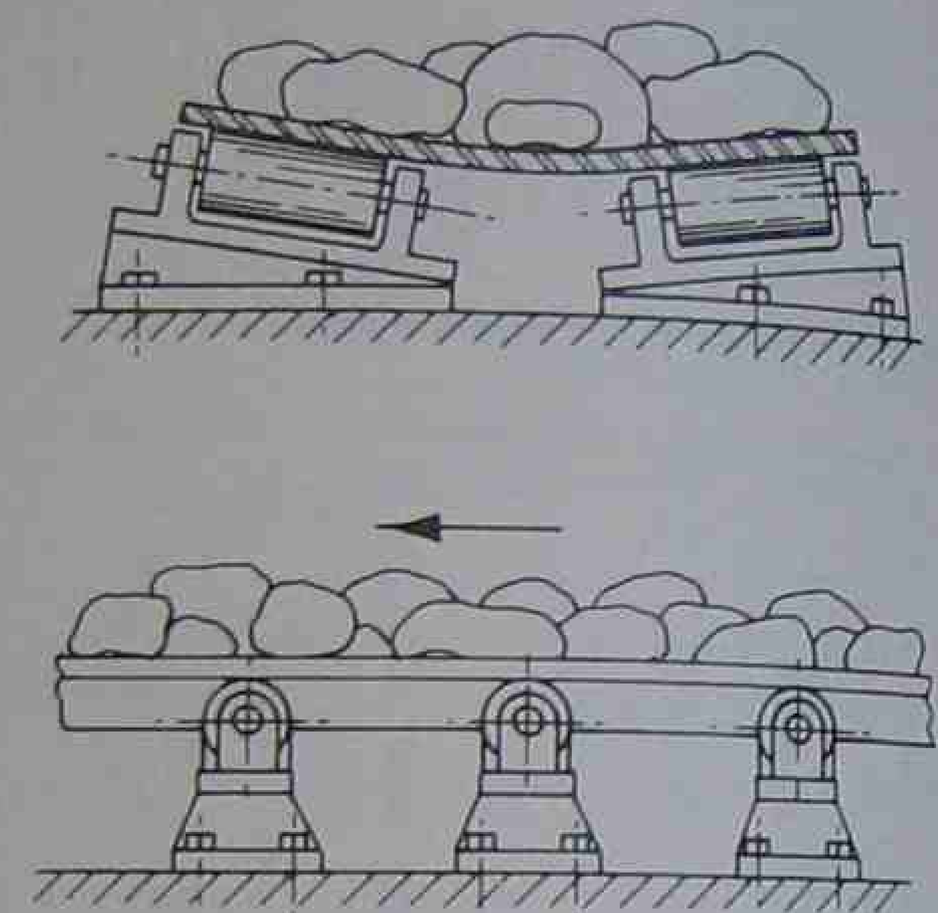


Figure 2.53 *Belt conveyor*

ROPE DRIVE

Rope drives are used extensively for lifting and transporting gear. Manilla rope is satisfactory for relatively light loads but steel wire rope is used for heavy loads.

Pulleys have V grooves with a large radius at the bottom of the groove, and the large diameter of the pulleys limits bending stress in the wire. Wire rope is made of several strands, each consisting of a number of smaller wires twisted about a core of hemp saturated with lubricant.

Ropes are used in lifts, hoists and earth-moving machines, etc. One application is in a *block-and-tackle* hoist in which the rope is passed around the pulleys in two *sheaves* one of which is attached to a fixed point and the other carrying a hook for the load. Each sheave has two or more pulleys running freely on the same shaft, and a load several times the pull on the rope can be lifted with this equipment.

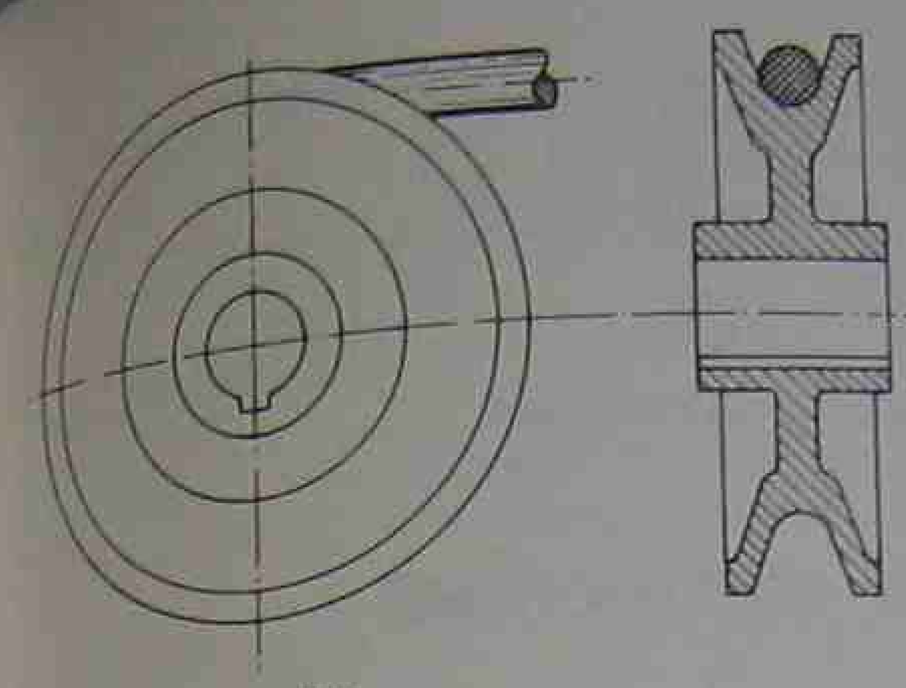


Figure 2.54 *Rope pulley*

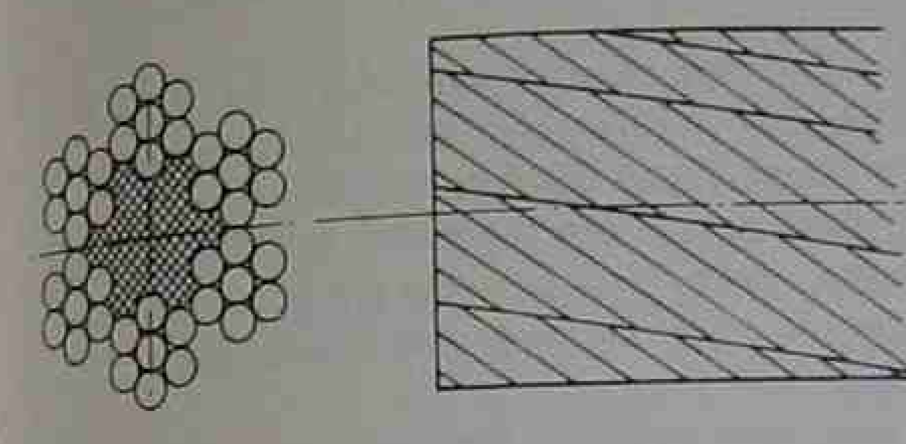


Figure 2.55 *6 x 7 wire rope*

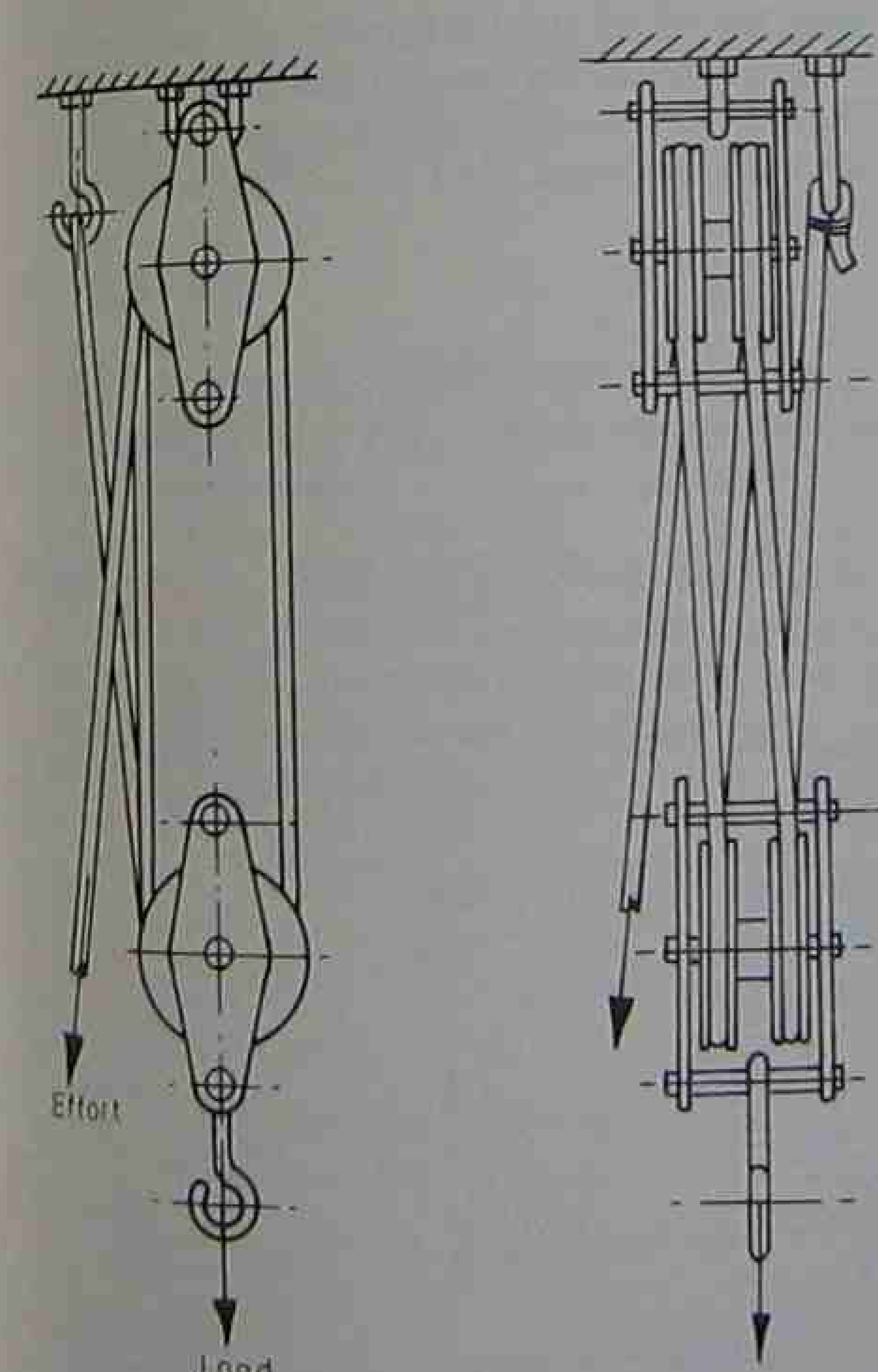


Figure 2.56 *Block-and-tackle*

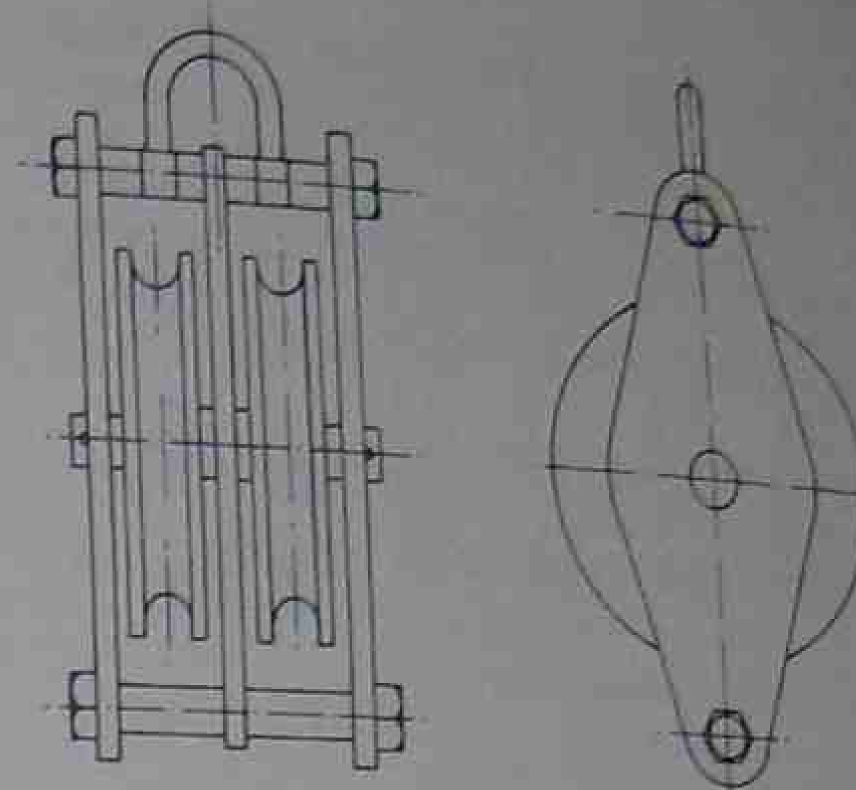


Figure 2.57 *Two-pulley sheave*

CHAIN DRIVE

A chain drive consists of a continuous series of links which engage with teeth on pulleys attached to parallel shafts. This type of drive has no slip and it will transmit much more power than belt drives.

The most common type is the *roller chain drive* as used on bicycles. This has hardened steel rollers, joined by links, which engage with toothed wheels called *sprockets*. Higher powers can be transmitted by using double or treble chains known as 'duplex' and 'triplex' chains.

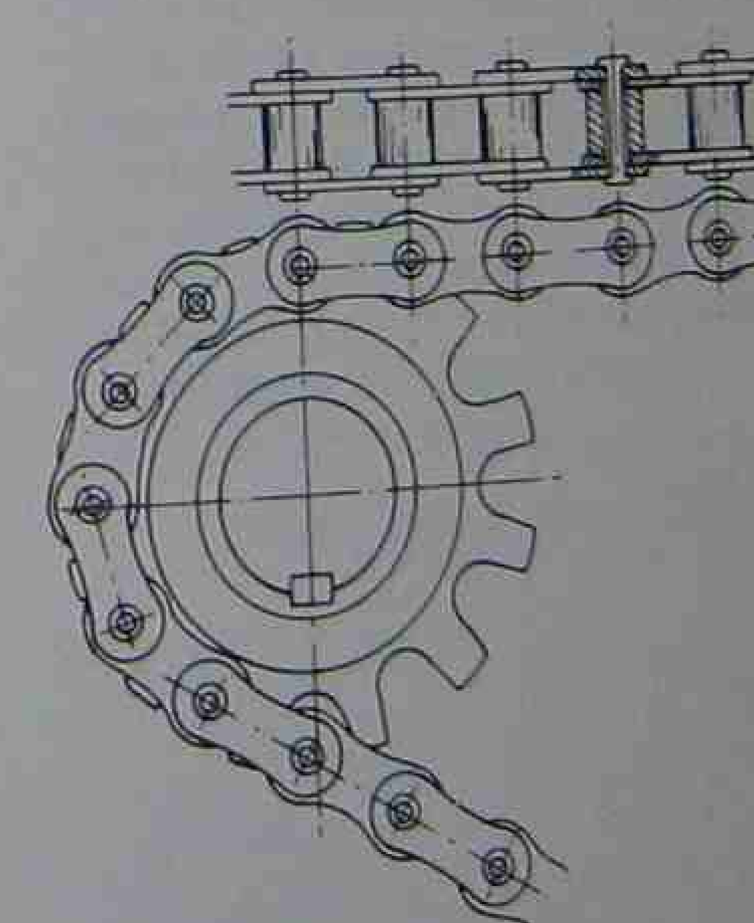


Figure 2.58 *Roller chain and sprocket drive*

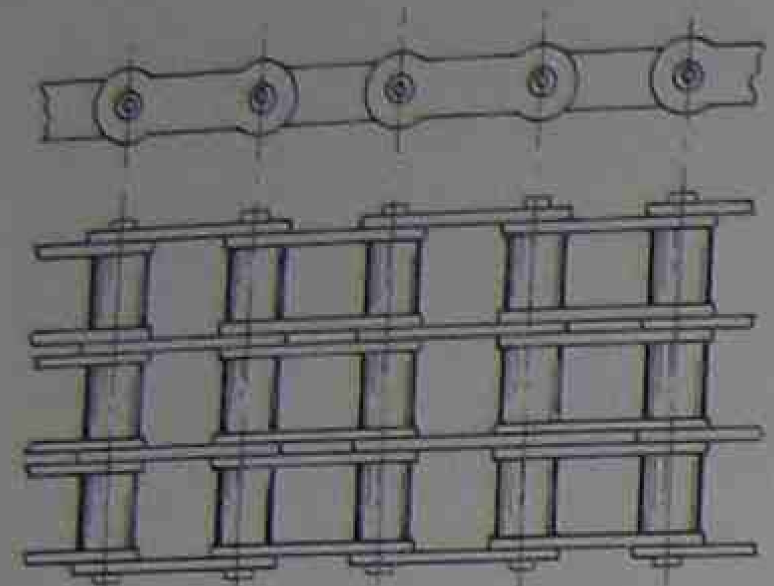


Figure 2.59 *Triplex roller chain*

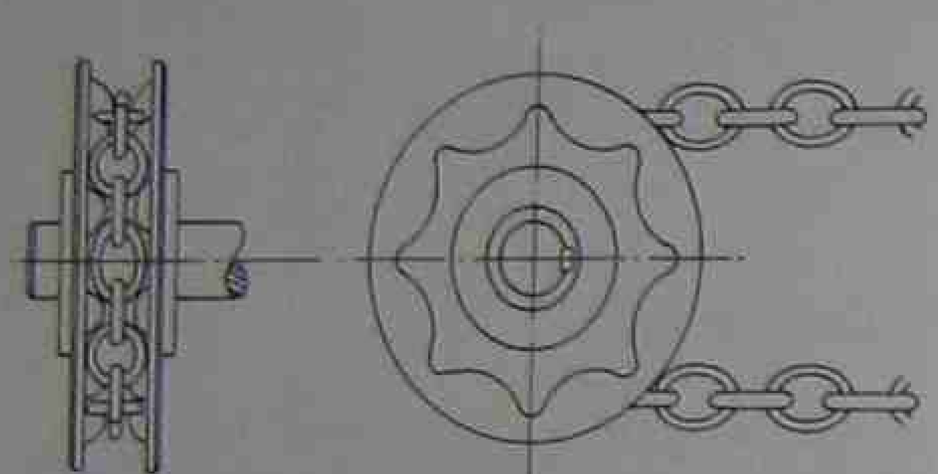


Figure 2.60 *Chain drive*

For large centre distances it is often necessary to use a tensioner to take up the slack in the chain. This tensioner consists of a free-running sprocket on a spring-loaded lever.

Another type of drive uses malleable iron links of rectangular shape which can easily be connected or disconnected when required. Known as *link belt chains* they are suitable only for low speed drives such as, for example, on agricultural machinery.

For low power drives a *bead chain* may be used. This chain has beads spaced on a wire which engage with recesses in the pulleys and it can be used for non-parallel shafts.

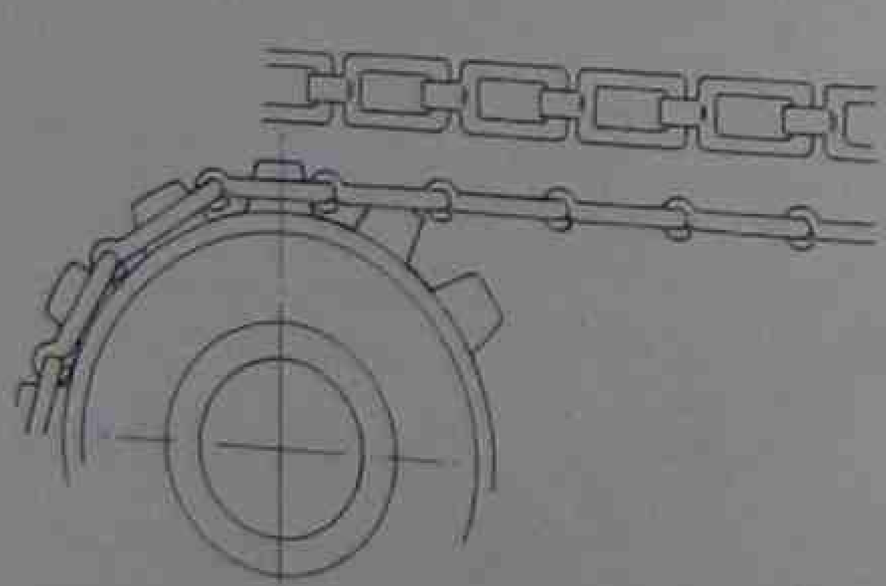


Figure 2.61 *Link belt chain drive (malleable iron)*

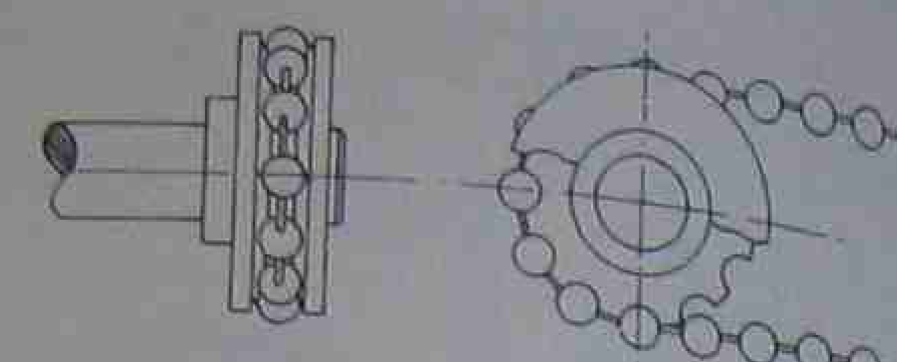


Figure 2.62 *Bead chain drive*

2.4 MECHANISMS

LEVERS

A lever is a rigid beam pivoted at a point known as the 'fulcrum', with a 'load' at another point and an 'effort' applied at a third point which balances the load. The fulcrum and the points at which the load and effort are applied have pin joints. Levers used as machine components can be of rectangular, elliptical, I-section, etc.

Levers may be of the first, second or third order: A *first order lever* has the fulcrum between the load and the effort; the *second order lever* has the load between the effort and the fulcrum; the *third order lever* has the effort between the load and the fulcrum.

The *bell crank lever* has the fulcrum at the junction of two arms which are at an angle to one another.

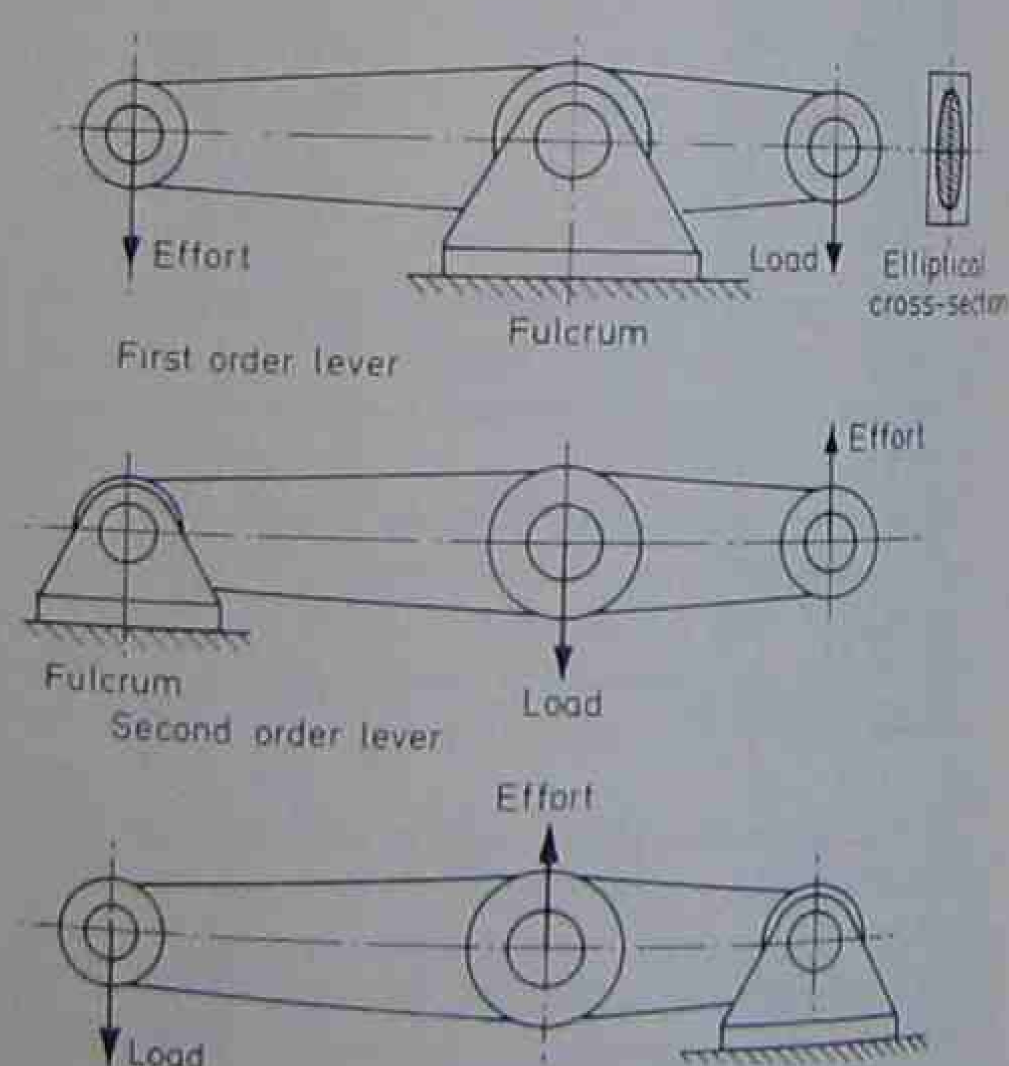


Figure 2.63 *Orders of levers*

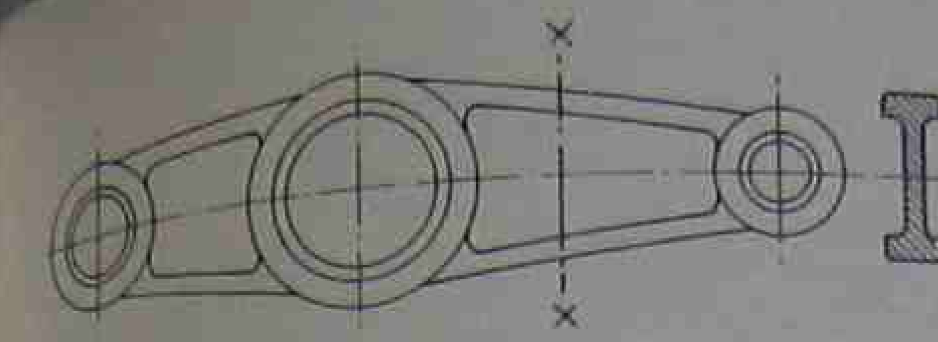


Figure 2.64 *I-section lever*

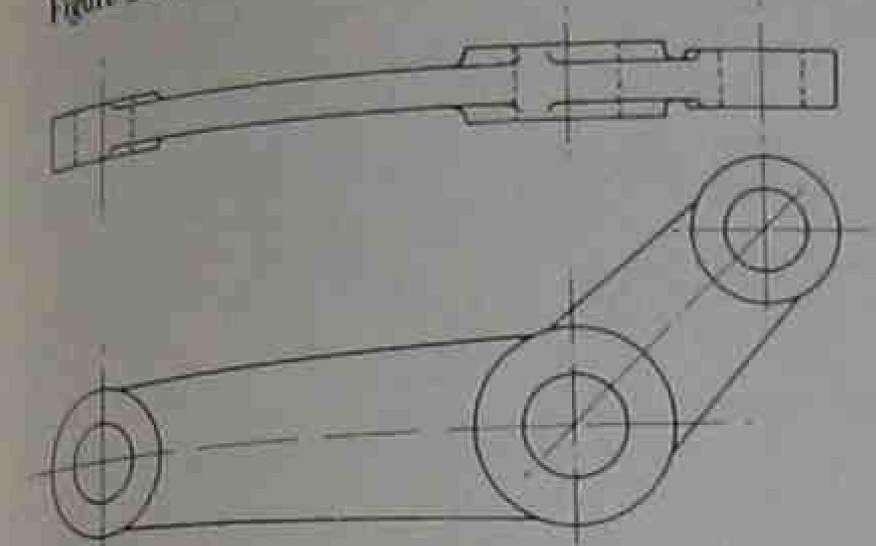


Figure 2.65 *Bell crank lever*

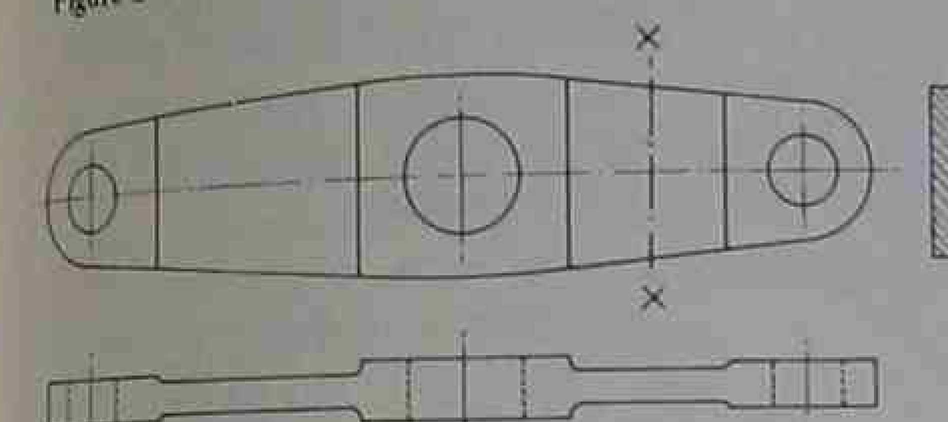


Figure 2.66 *Rectangular-section lever*

LINKAGES

Linkages are combinations of levers, rods and cranks which convert one type of motion into another, e.g. rotary to reciprocating.

Four-bar chain This consists of four links connected at their ends by pins in the form of a trapezium. One of the links may be made to rotate continuously and another link is usually part of a machine frame. The rotating link is called a *crank*.

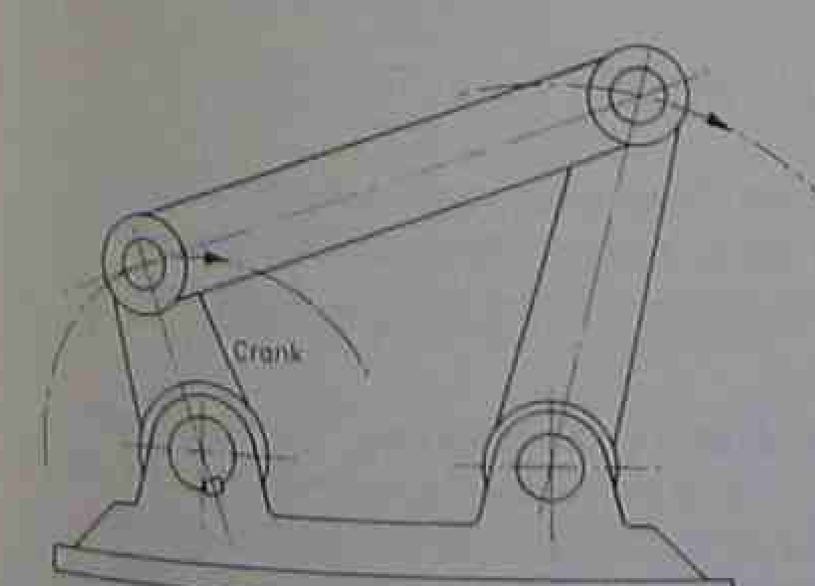


Figure 2.67 *Four-bar chain*

Crank-slider mechanism In this mechanism a link known as a *connecting rod* is connected at one end to a rotating crank and at the other end to a sliding block, or *slider*. As the crank rotates the slider moves with reciprocating motion. This mechanism is used in the IC engine.

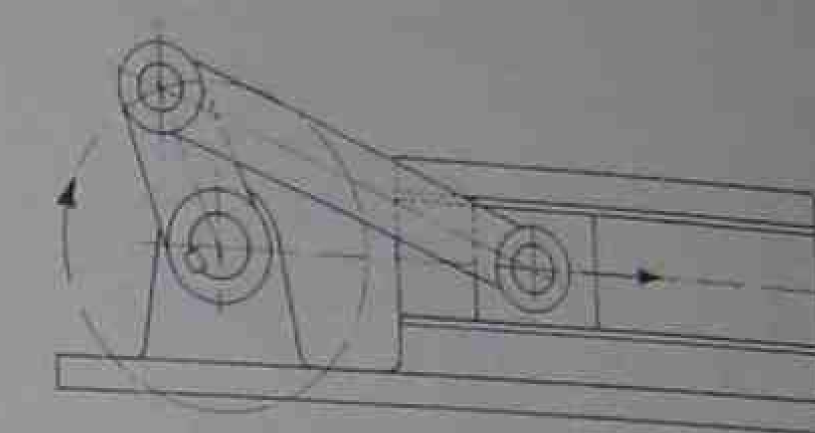


Figure 2.68 *Crank-slider mechanism*

Eccentric-crank and slider mechanism An eccentric is a crank-slider mechanism in which the crank and crank pin are replaced by an eccentric circular disk mounted on the shaft.

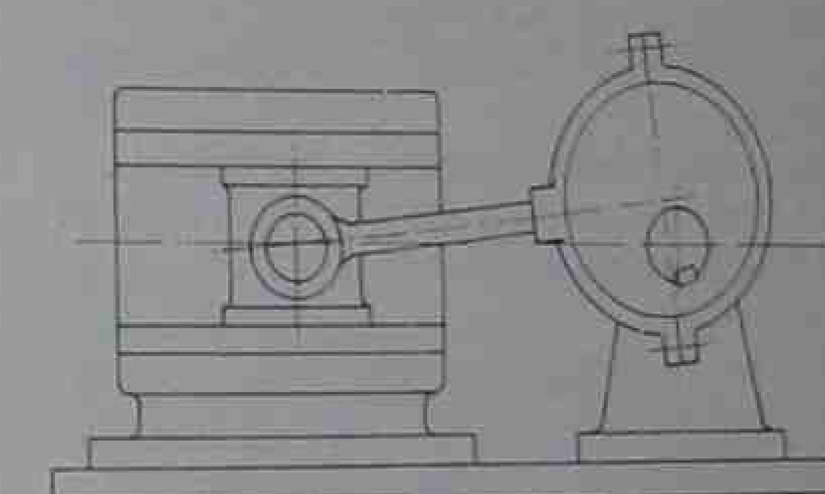


Figure 2.69 *Eccentric-crank and slider mechanism*

Parallel motion mechanism This is a four-bar chain in which opposite links are of equal length, and two opposite links rotate in the same direction. The pantograph uses a parallel linkage for the purpose of reproducing a motion to a different scale as, for example, in the scaling-up of drawings.

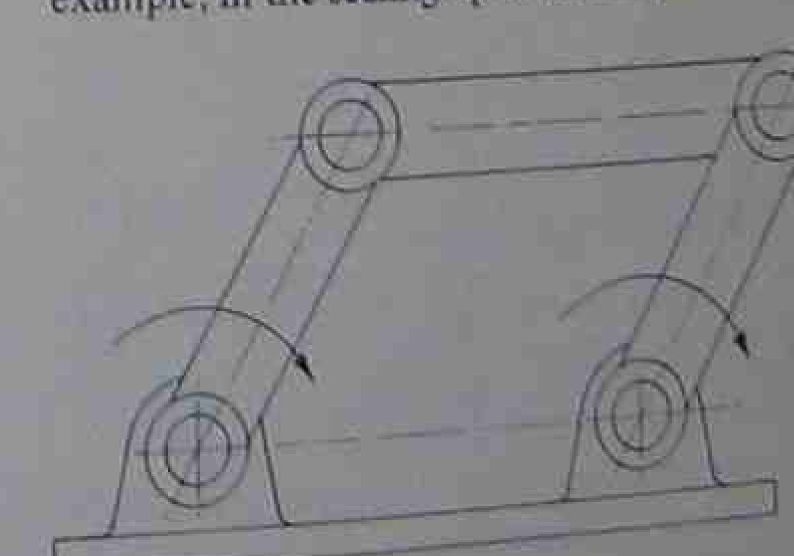


Figure 2.70 *Parallel motion mechanism*

Ratchet mechanism A ratchet mechanism produces an intermittent motion from a reciprocating or oscillating motion. It is useful for indexing in machine tools. The ratchet may be driven by a crank and connecting rod, as shown.

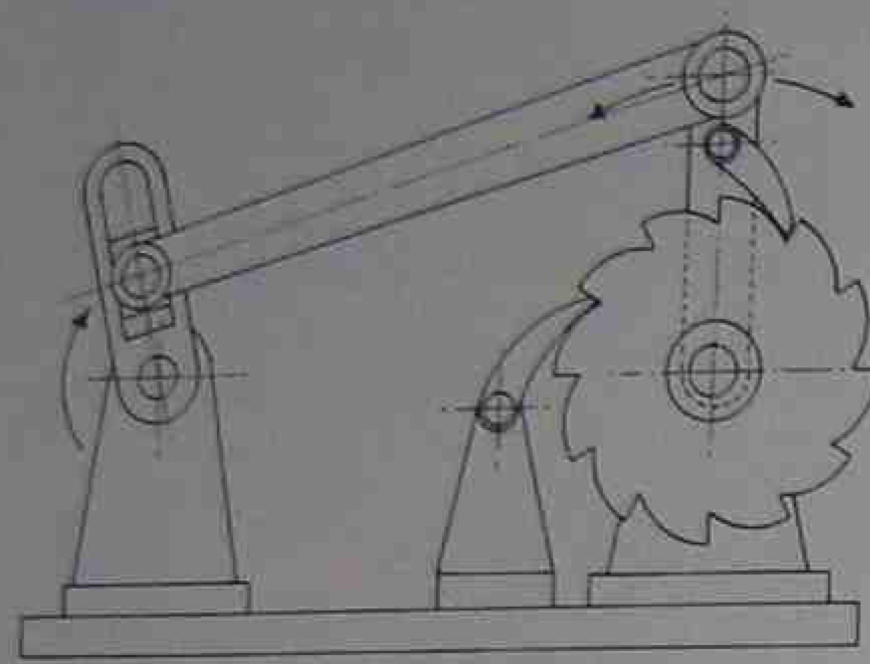


Figure 2.71 *Ratchet mechanism*

Quick-return mechanism (oscillating-crank mechanism) In this mechanism the end of a rotating crank is attached to a block which slides in an oscillating link which in turn moves a reciprocating bar. The motion of the bar is more rapid in one direction than in the other.

This motion is used in machine tools, such as a shaper, where the cutting stroke of a tool is required to be slower than the return stroke.

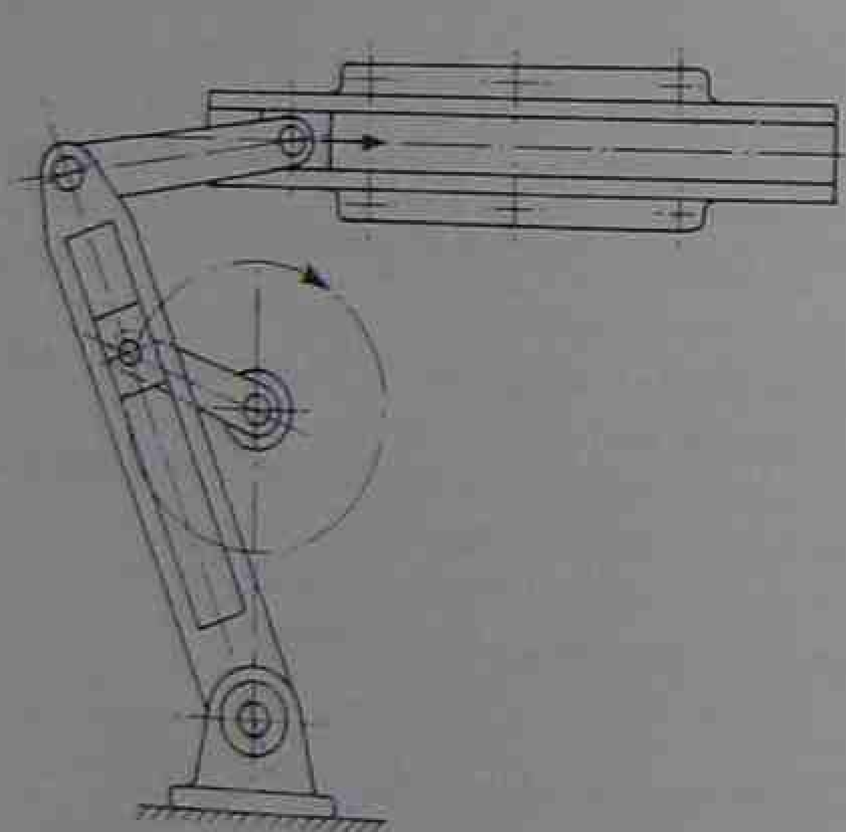


Figure 2.72 *Quick-return mechanism*

Scotch yoke mechanism A scotch yoke consists of a crank pin which slides in a slot in a crosshead attached to a reciprocating rod. The slot is at right angles to the rod. Uniform rotation of the crank produces simple harmonic motion in the rod.

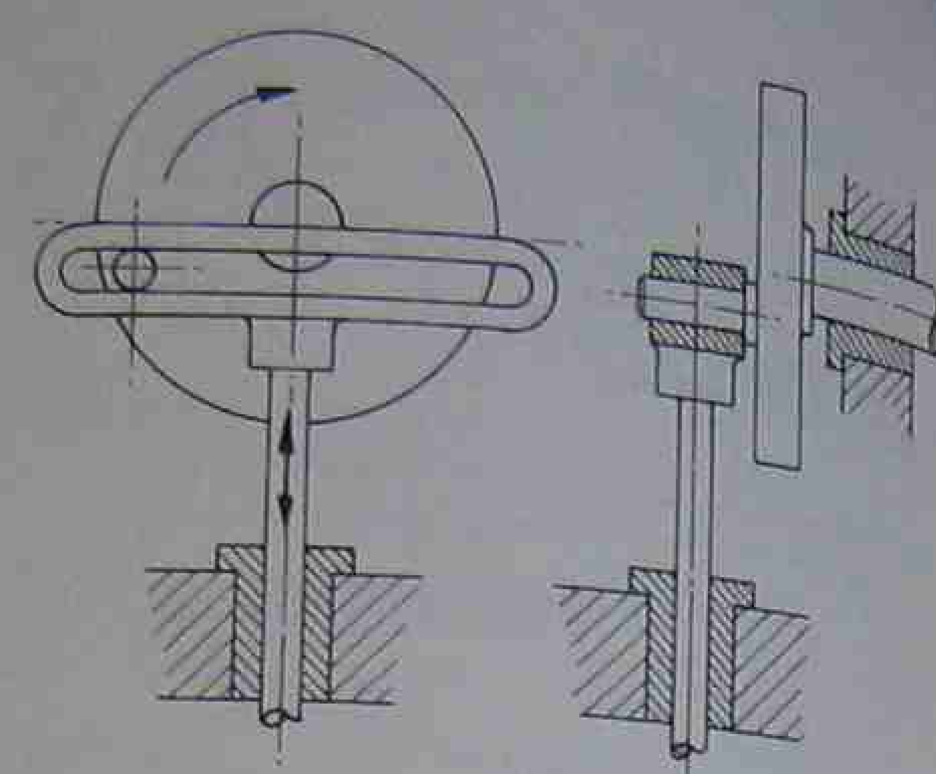


Figure 2.73 *Scotch yoke*

Geneva wheel mechanism In the Geneva wheel mechanism a rotating crank pin engages with radial slots in a disk attached to a parallel shaft to give an intermittent and opposite rotation in the second shaft. The mechanism is used in machine tools where an intermittent action is required.

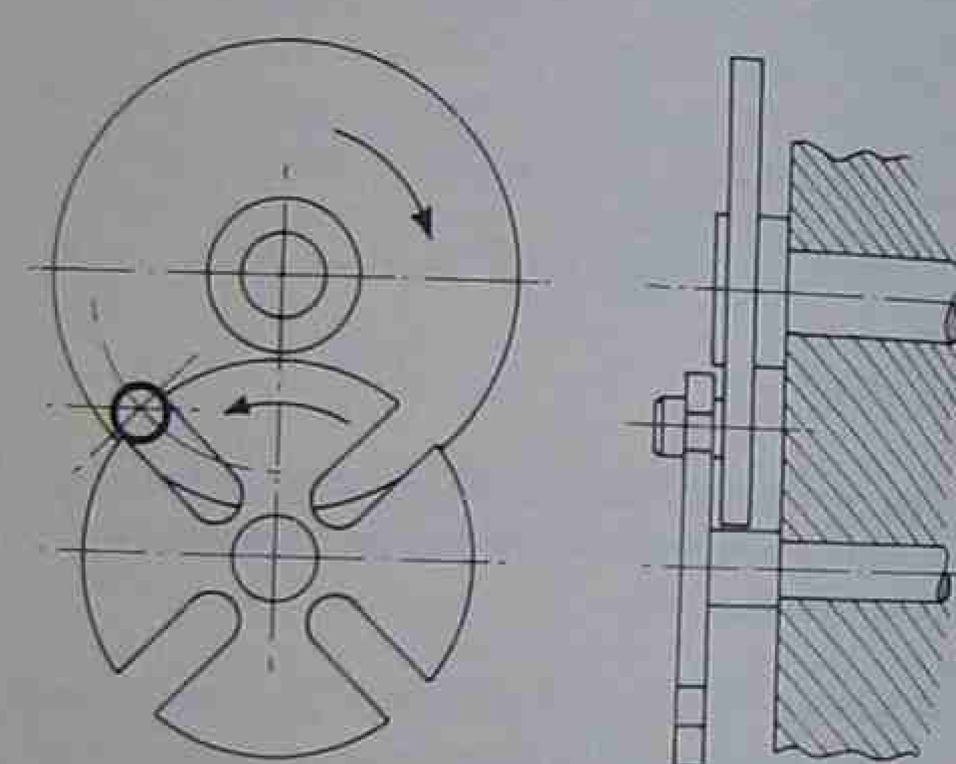


Figure 2.74 *Geneva wheel mechanism*

GOVERNORS

Governors are devices for controlling the flow of steam, gas, air or water to engines, turbines, etc., by maintaining constant speed under varying load.

Centrifugal dead-weight governor In this type of governor, weights on pivoted arm rotate and fly outwards under centrifugal force, the restoring force being the gravitational force on the weights. The movement of the weights operates the valve control.

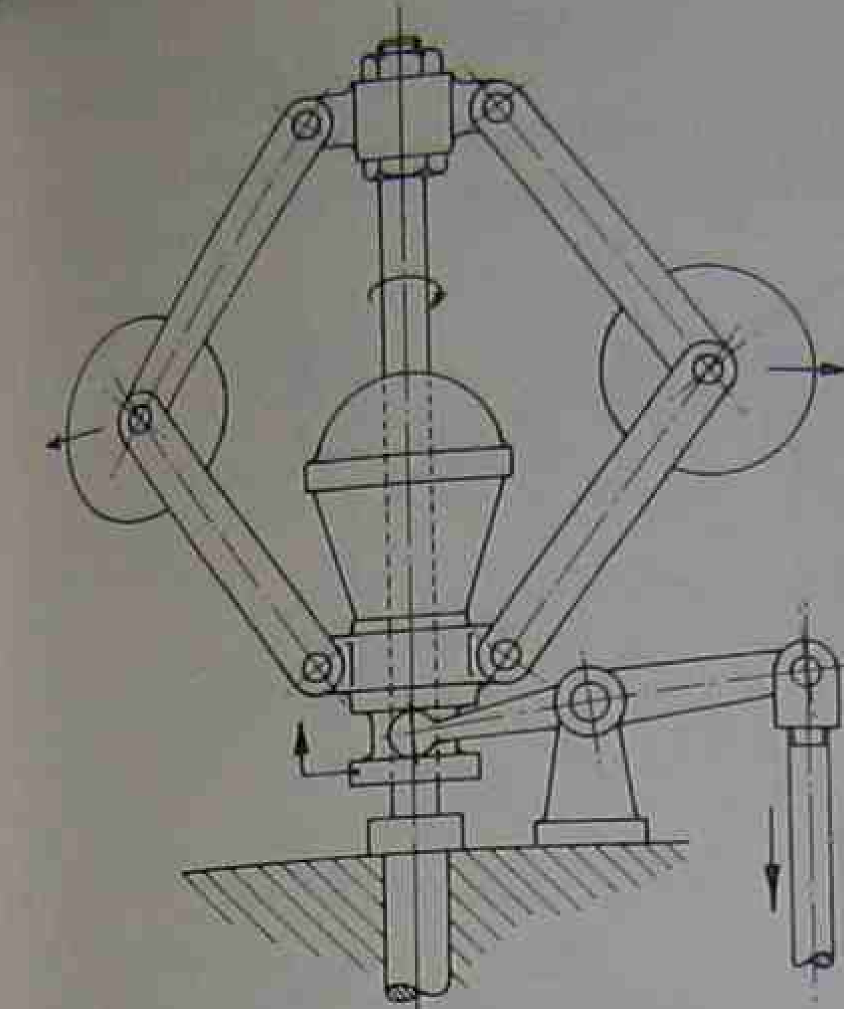


Figure 2.75 *Centrifugal dead-weight governor*

Hartnell governor (spring-loaded) In the type known as the Hartnell governor, the weights are attached to a spring which provides the restoring force. The spring force may be adjusted to control the speed at which the governor operates.

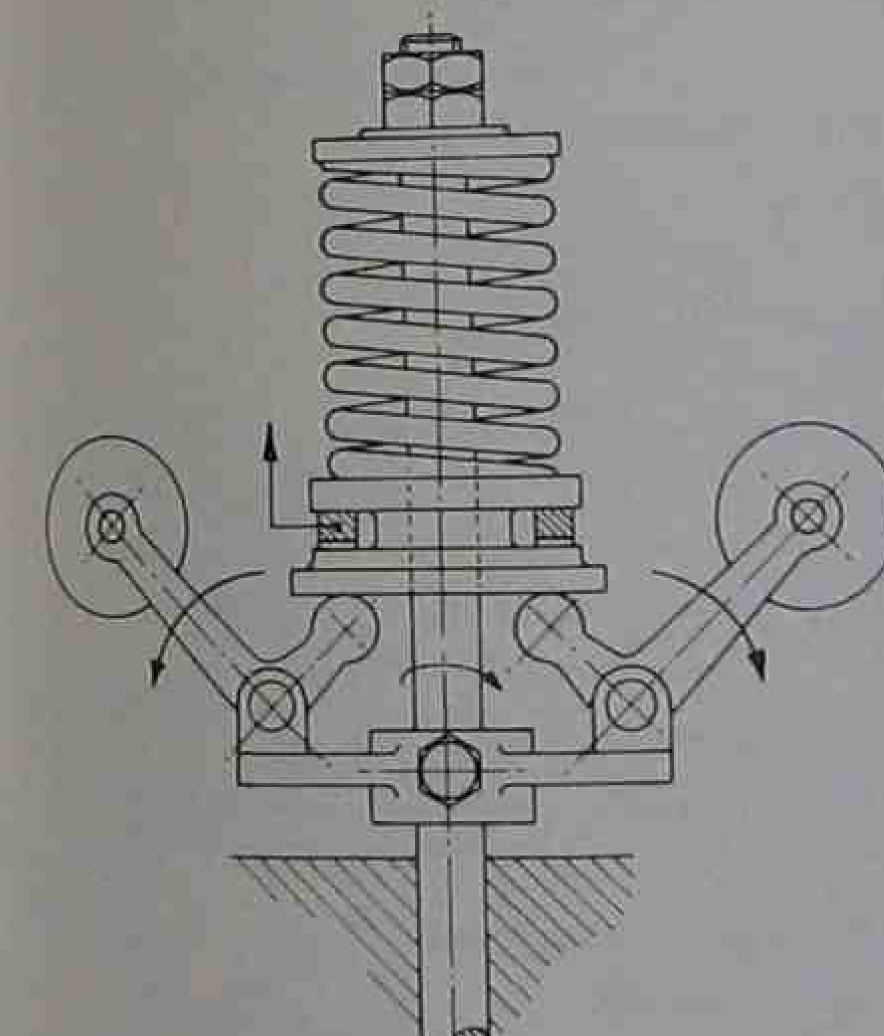


Figure 2.76 *Hartnell governor (spring-loaded)*

JACKS

Jacks are devices using levers, gears or hydraulic intensifiers to provide a large lifting force from a small manual effort.

Screw jack This has a nut rotated by bevel gears when these are turned by a handle. The nut moves a vertical screw supporting the load.

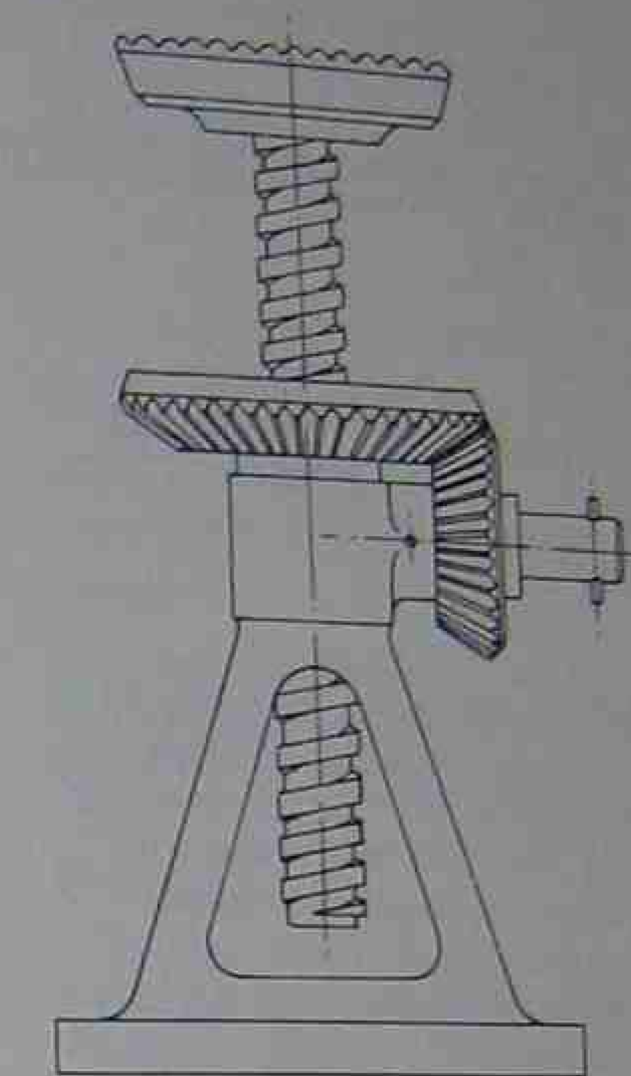


Figure 2.77 *Screw jack*

Hydraulic jack In a hydraulic jack a hand-operated lever connected to a pump with a small diameter plunger supplies oil to a large diameter ram which lifts the load. The load is proportional to the ratio of the piston areas and the handle leverage.

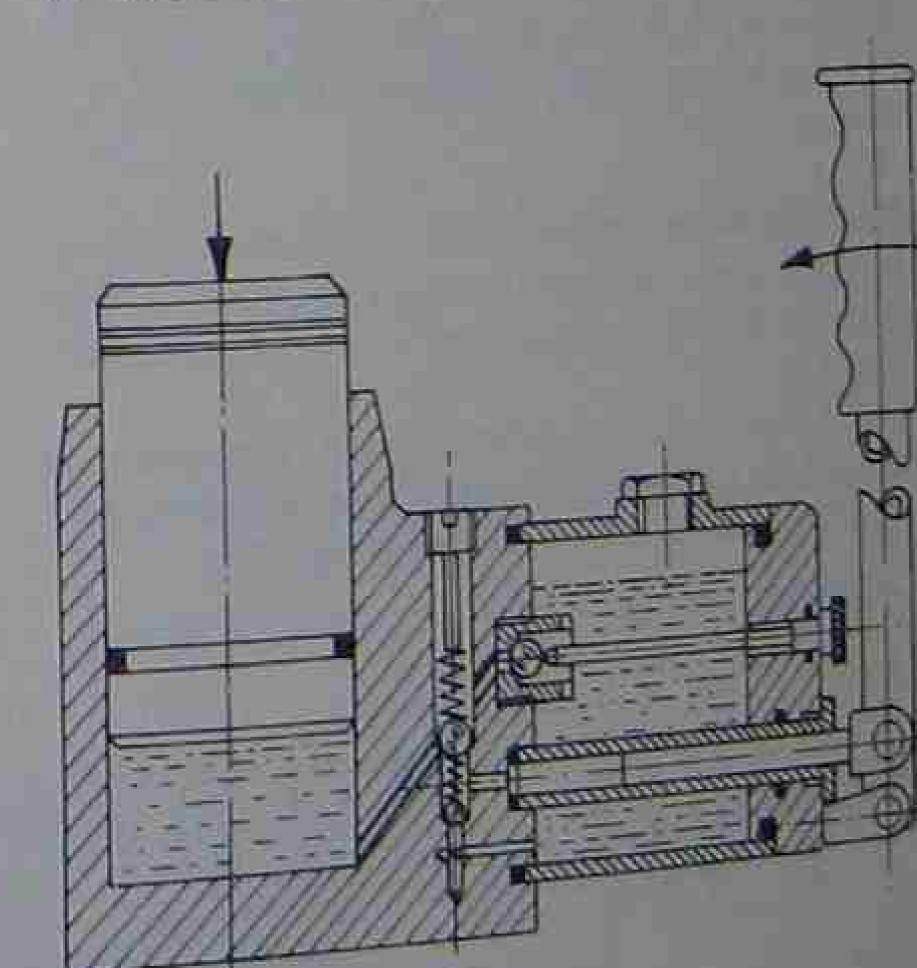


Figure 2.78 *Hydraulic jack*

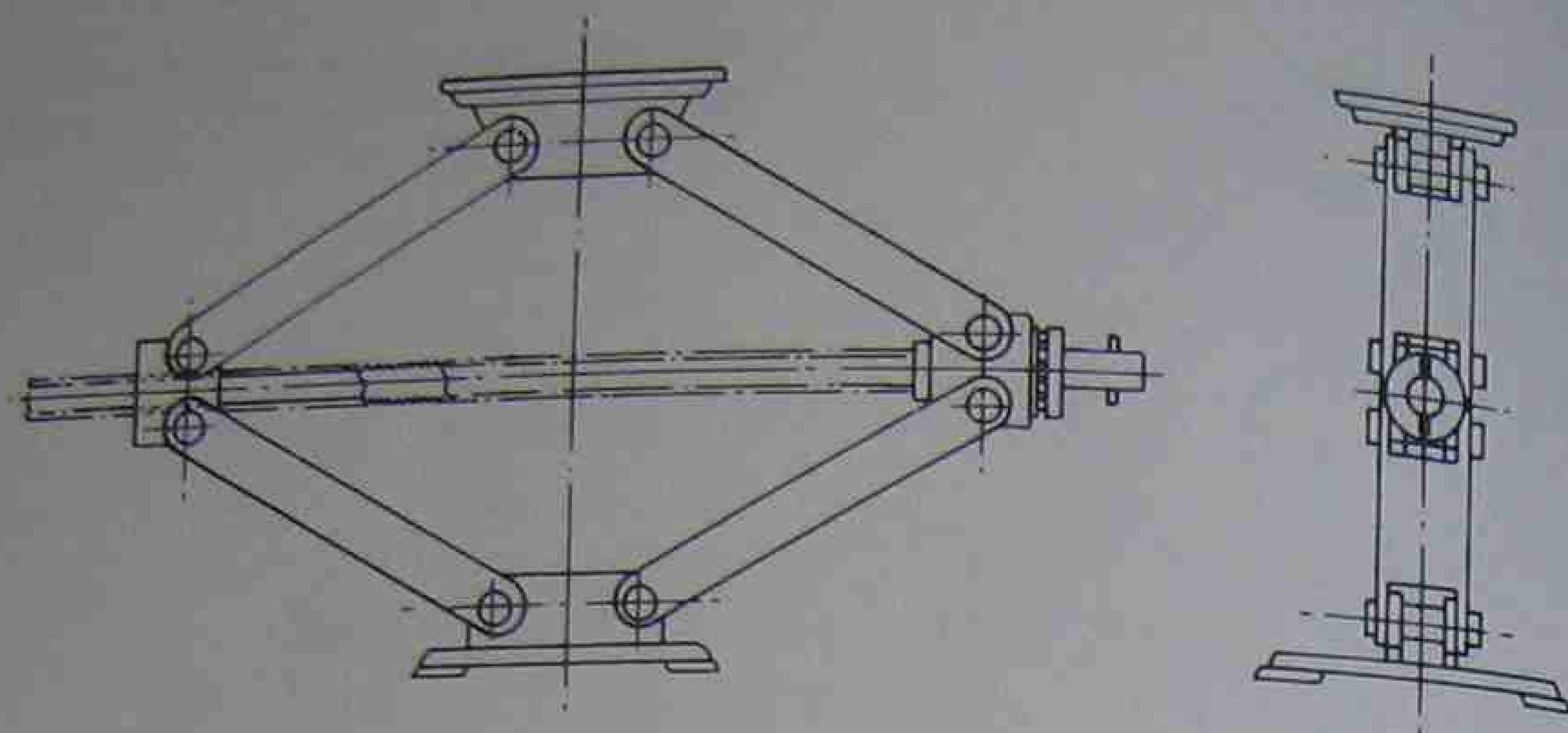


Figure 2.79 *Scissors jack*

Scissors jack A scissors jack uses a screw and nut in combination with links. Four links are arranged in the form of a parallelogram with a screw connecting two opposite corners. The other two corners are attached to the base and load-application points respectively. Turning the screw brings the first two corners together and moves apart the base and load points, thus lifting the load.

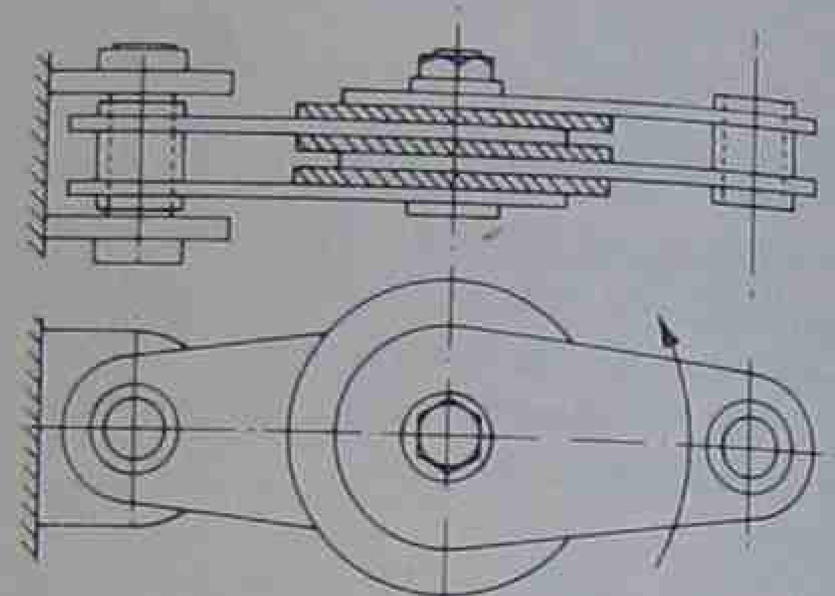


Figure 2.80 *Mechanical friction damper*

SHOCK ABSORBERS

These are devices installed between a machine and its mountings to damp out any undesirable vibrations or shock loads. They utilise mechanical or fluid friction to absorb the power.

Mechanical friction damper This has friction disks clamped between two elements which have relative motion.

Hydraulic shock absorber This has a piston inside a cylinder which forces oil through a small hole or valve orifice to provide resistance to the motion and hence retard it.

Two types are used on automobiles: the older type which has a piston connected to a lever; the telescopic type in which one tube slides inside another.

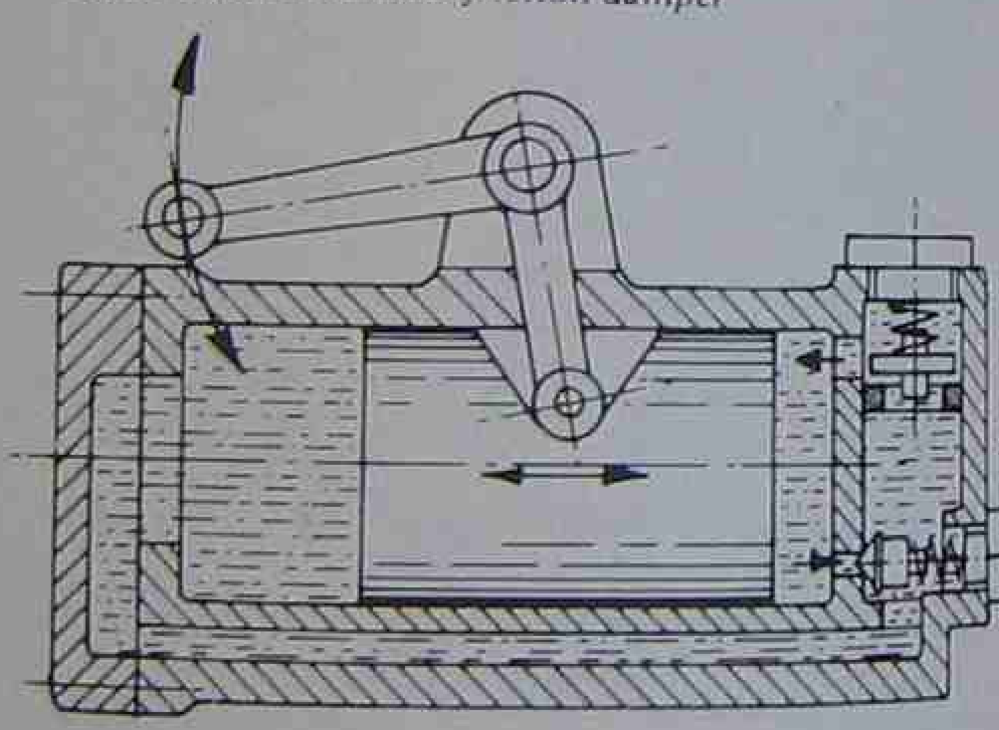


Figure 2.81 *Hydraulic shock absorber*

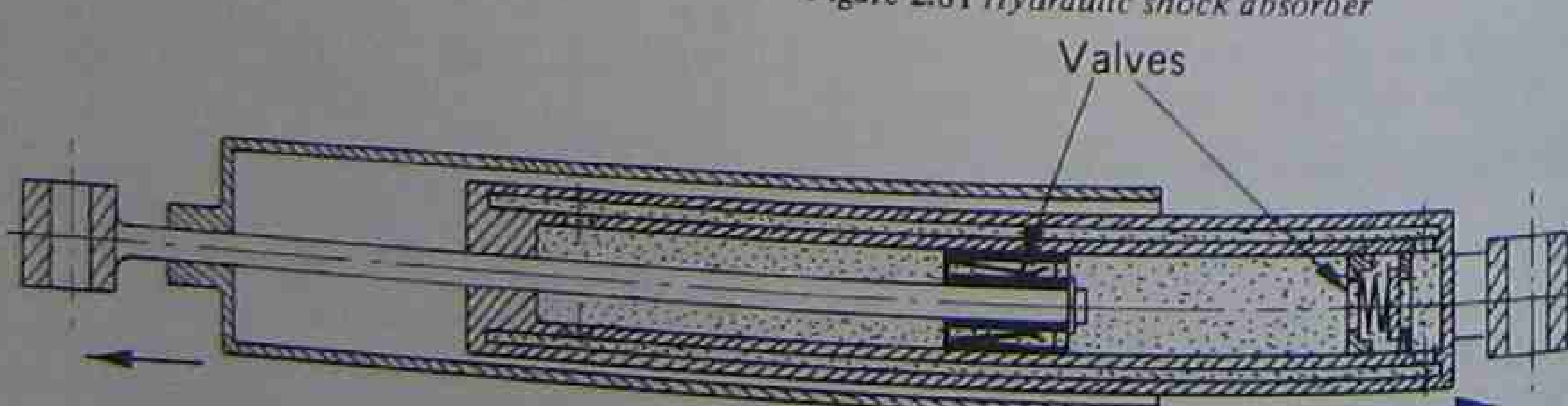


Figure 2.82 *Telescopic hydraulic shock absorber*

STEERING MECHANISMS

These are mechanisms between the steering wheel of a vehicle and the steering linkage connected to the road wheels.

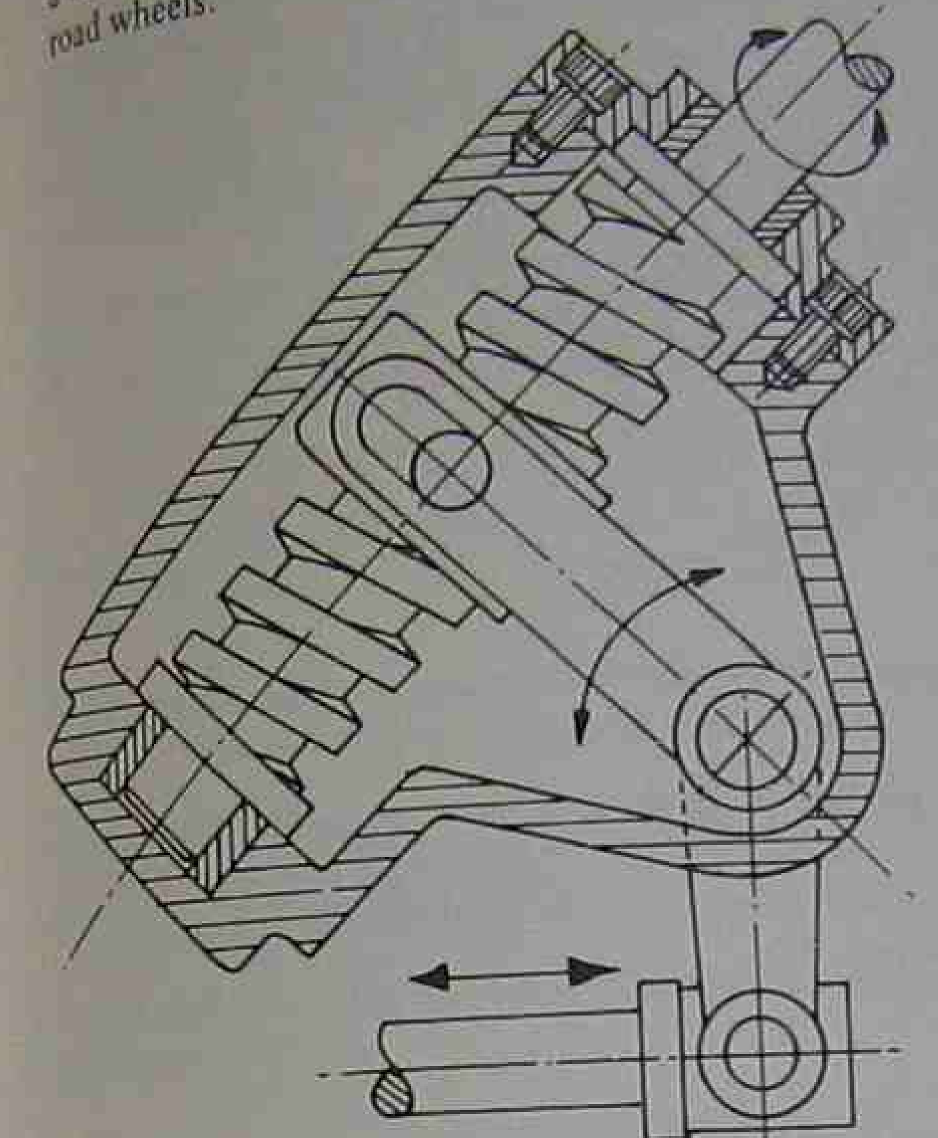


Figure 2.83 *Screw-and-nut steering*

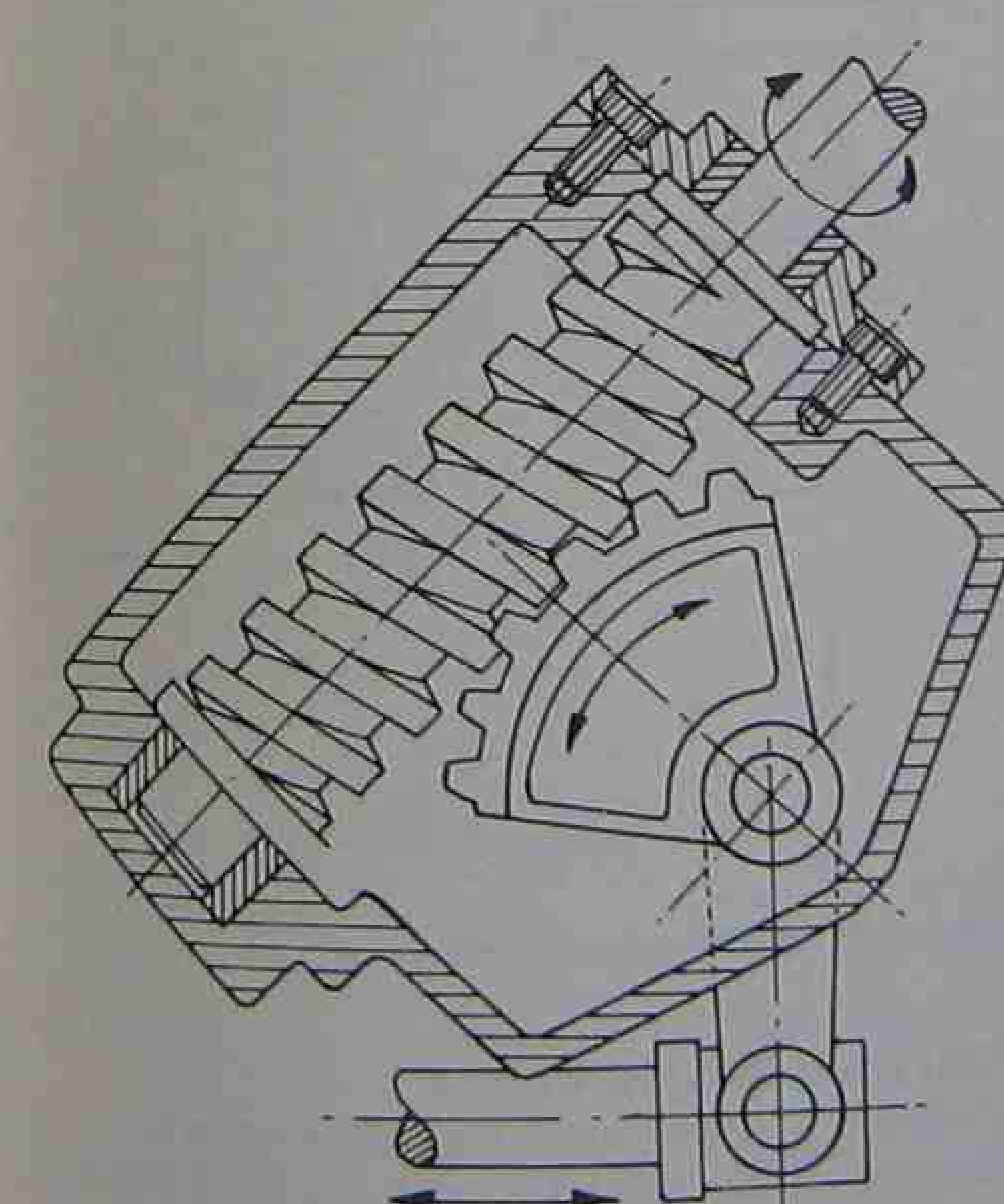


Figure 2.84 *Worm-and-wheel steering*

Screw-and-nut steering A square-threaded nut connected to the steering linkage is driven by a mating screw on the steering column.

Worm-and-wheel steering This has a worm on the steering column which engages with a toothed quadrant which has a lever connected to the steering rod.

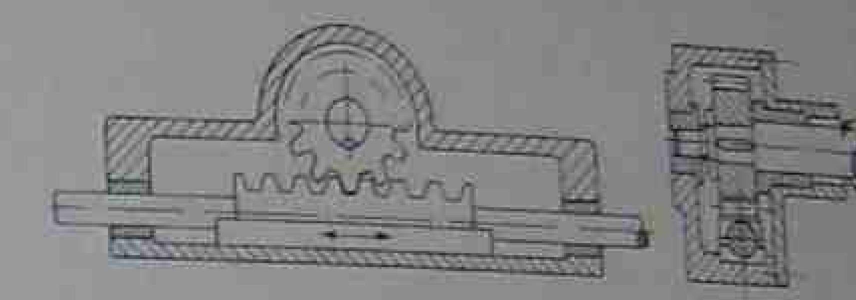


Figure 2.85 *Rack-and-pinion steering*

Rack-and-pinion steering A small gear or pinion on the steering column drives a toothed quadrant carrying a lever pinned to the steering rod.

CAM

A cam is a mechanism which involves sliding and which converts one type of motion into another, e.g. rotary to reciprocating. The cam itself may have any type of motion, rotary, reciprocating or oscillating.

RADIAL CAM (PLATE CAM)

This cam has a radial 'profile' and rotates continuously to give motion to a sliding or rocking 'follower'. The total movement of the follower is called the 'lift' and the highest point on the cam is known as the 'nose'. The profile is constructed on the 'base circle', and the period when the follower is stationary is known as the 'dwell'.

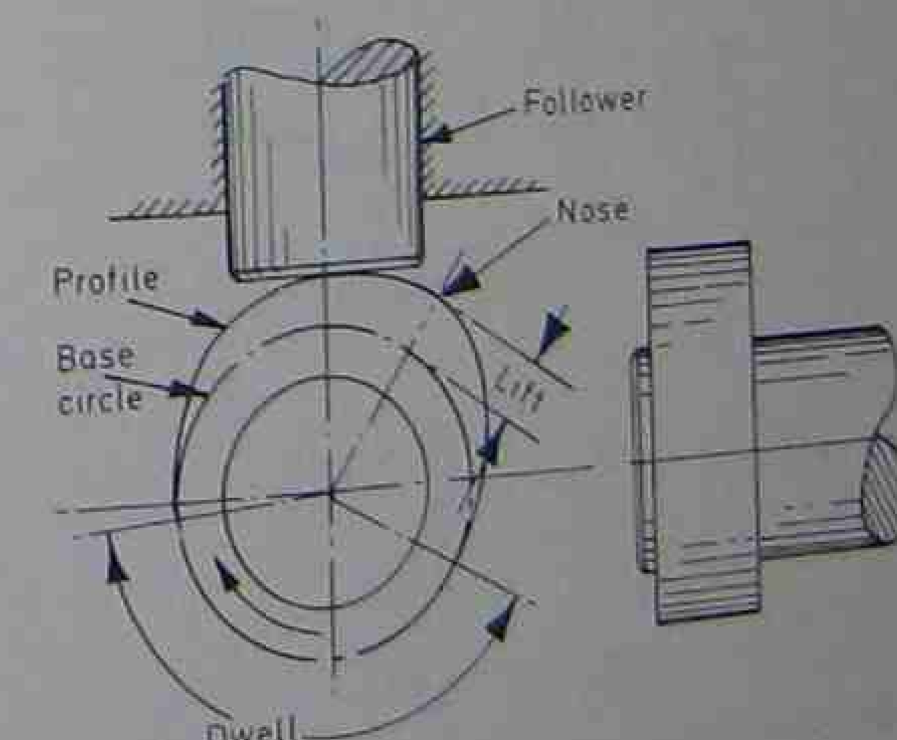


Figure 2.86 *Cam details*

CAM FOLLOWERS

That part of a cam mechanism which is driven by the cam is known as the cam 'follower'. This may have reciprocating or rocking motion and have different types of end in contact with the cam, e.g. knife-edge, flat, domed or with a roller.

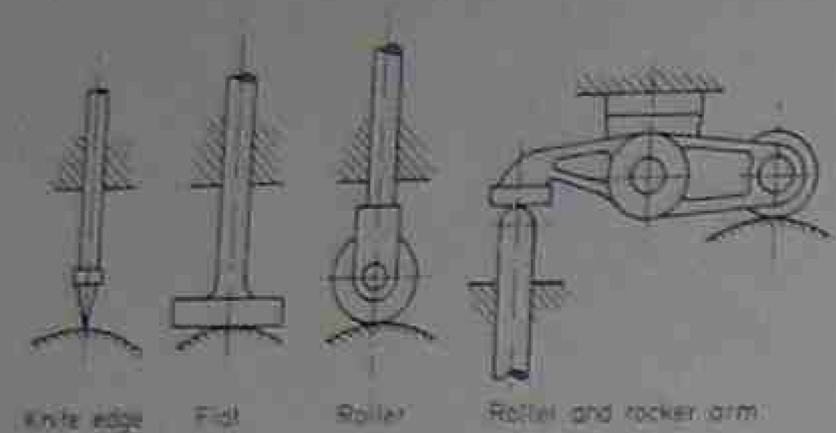


Figure 2.87 *Cam followers*

TYPES OF RADIAL CAM

The profile of the cam is designed to give the required type of motion to the follower.

Constant velocity cam A cam which produces a constant speed of the follower.

Constant acceleration/deceleration cam This is used to limit high acceleration and deceleration of the follower. For the first half of its travel the follower accelerates uniformly and for the second half it decelerates uniformly.

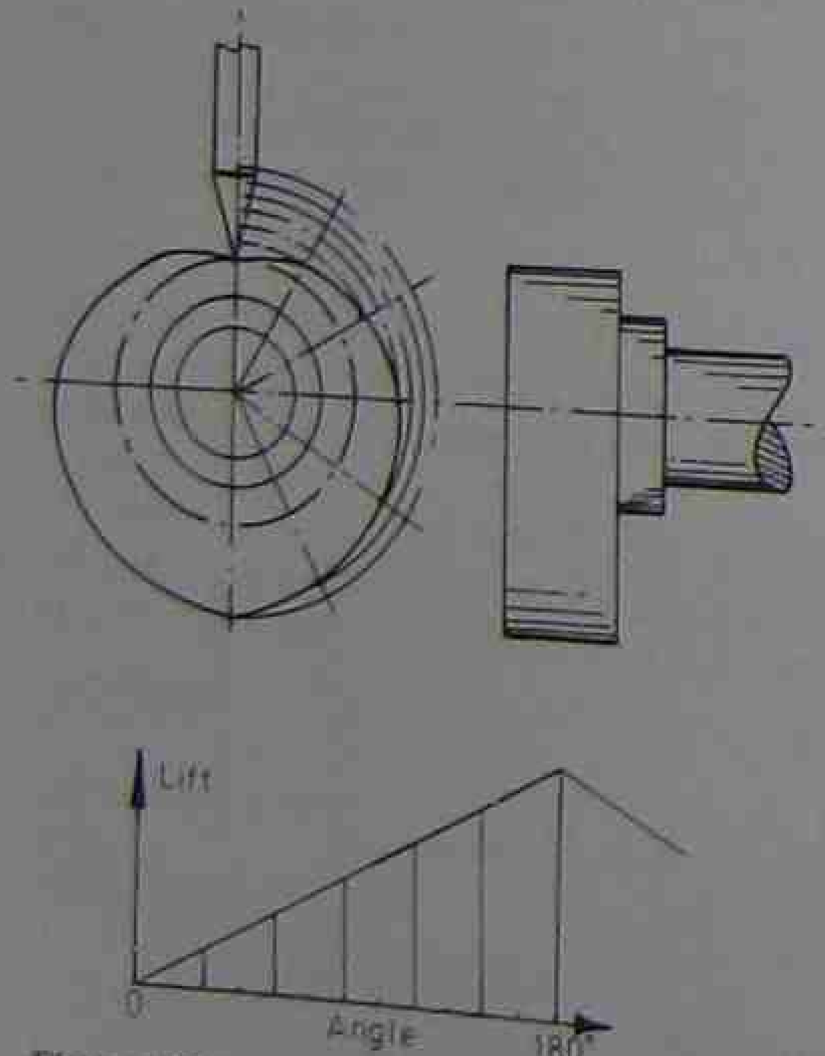


Figure 2.88 *Constant velocity cam*

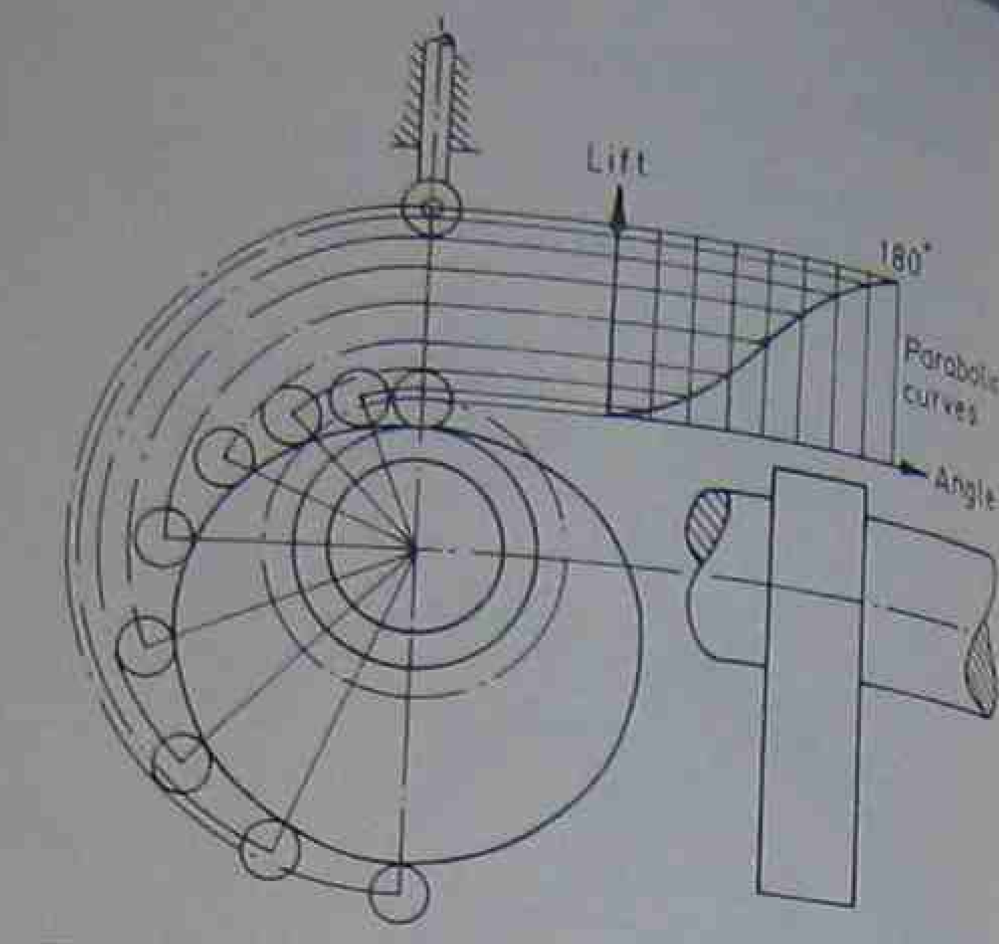


Figure 2.89 *Constant acceleration/deceleration cam*

Simple harmonic motion cam A cam which produces an oscillating motion of that type. An eccentric circle cam with a flat follower produces simple harmonic motion.

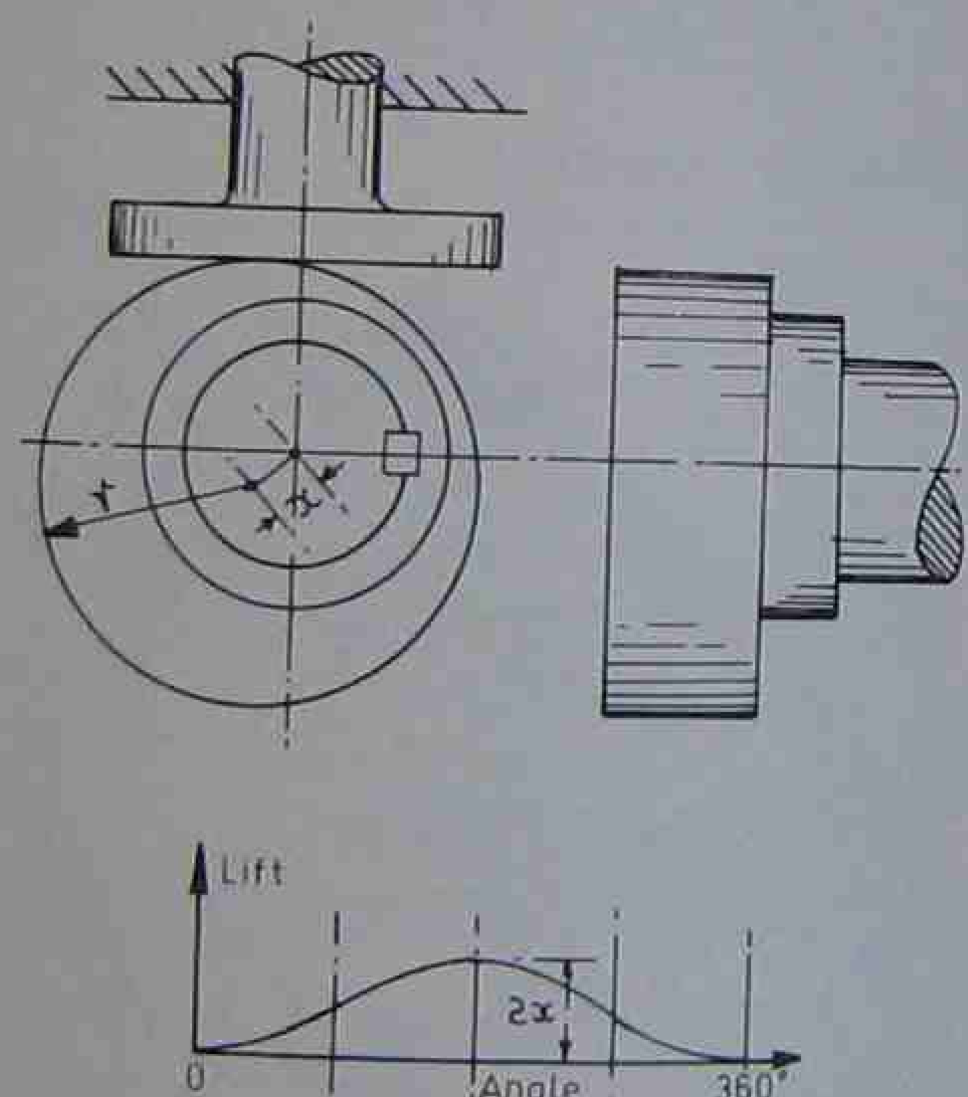


Figure 2.90 *Simple harmonic motion cam*

POSITIVE RETURN CAM

A radial type of cam where the follower ends in a side-mounted peg which fits into a slot in the face of the cam. With this cam a return spring is not required on the follower.

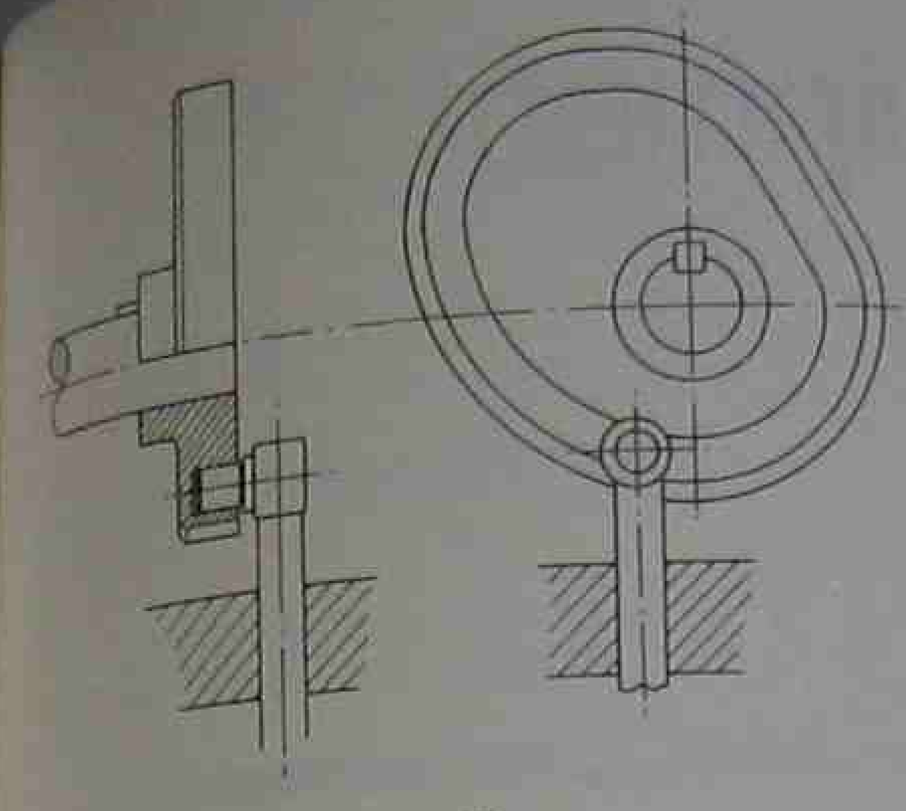


Figure 2.91 *Positive return cam*

Some cams are named according to their shape.

Tangent cam This has a profile comprised of two circular arcs joined by tangents.

Circular arc cam The profile of this type is composed of four circular arcs.

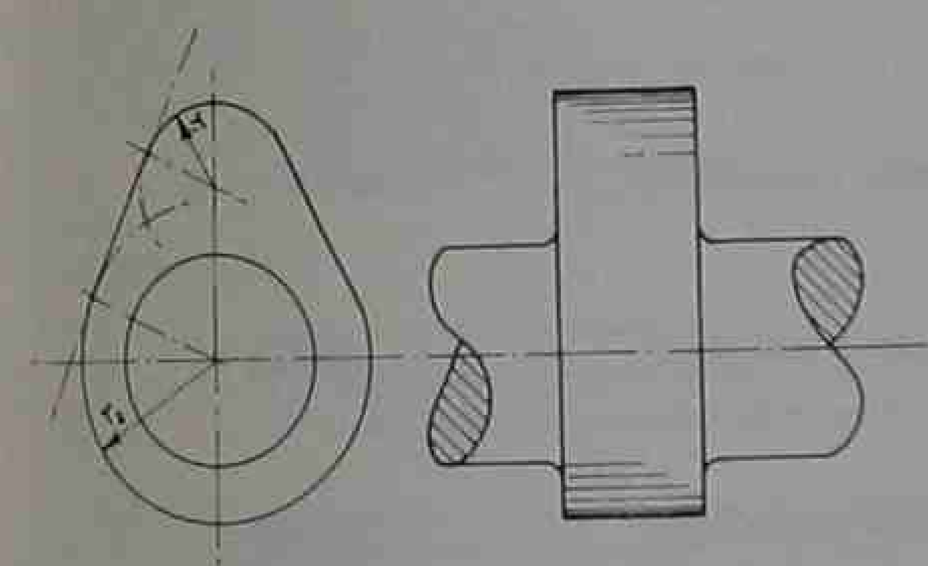


Figure 2.92 *Tangent cam*

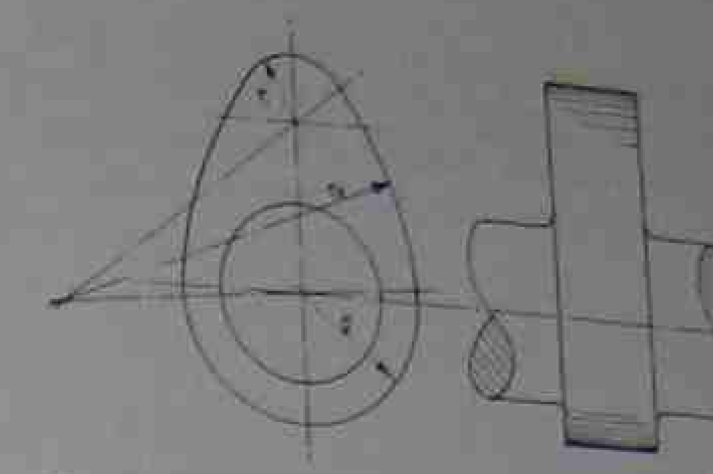


Figure 2.93 *Circular arc cam*

Face cam (axial cam) The cam profile of this type is on the end of a rotating cylinder and it imparts to the follower a sliding motion parallel to the shaft axis.

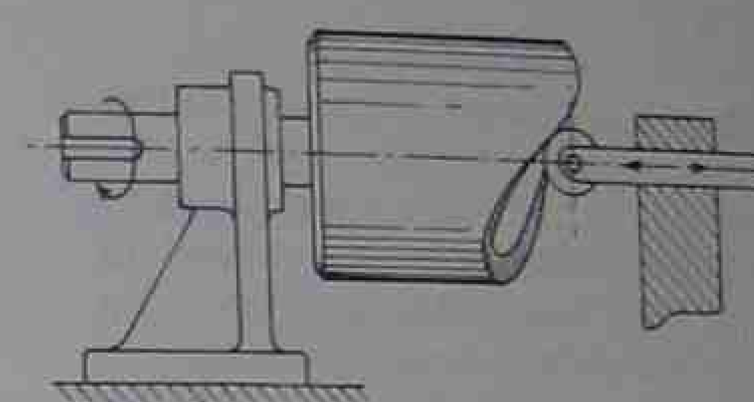


Figure 2.94 *Face cam*

Globoid cam This cam is a rotating cylinder with a concave circular arc profile (globoid). The follower, which is of the rocker type, has a peg which follows a groove cut in the cylinder.

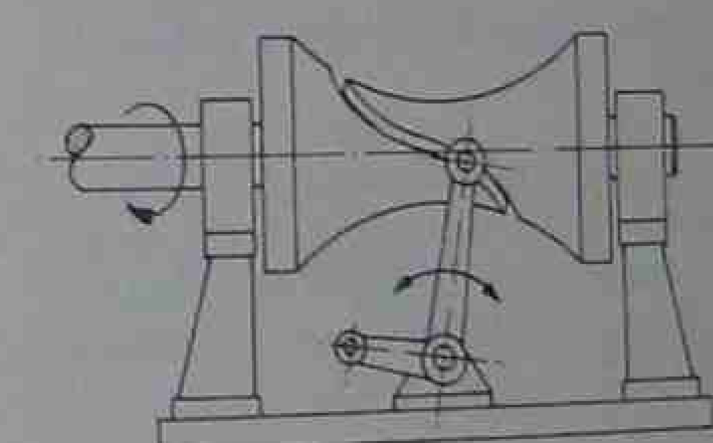


Figure 2.95 *Globoid cam*

3. Workshop Equipment

3.1 HAND TOOLS

HAND FILE

A hand file is a metal cutting tool consisting of a bar of hardened and tempered high carbon steel on which cutting teeth have been formed. It has a tapered 'tang' which fits into a handle usually made of wood. The cross-section of the bar may be rectangular, square, round, half-round, triangular, etc., and the pitch of the cutting teeth varies from fine to coarse. Files are often tapered along their length but may be parallel.

Nowadays, work is machined to such fine limits that use of the hand file is restricted. In many cases bench- and band-filing machines are used.

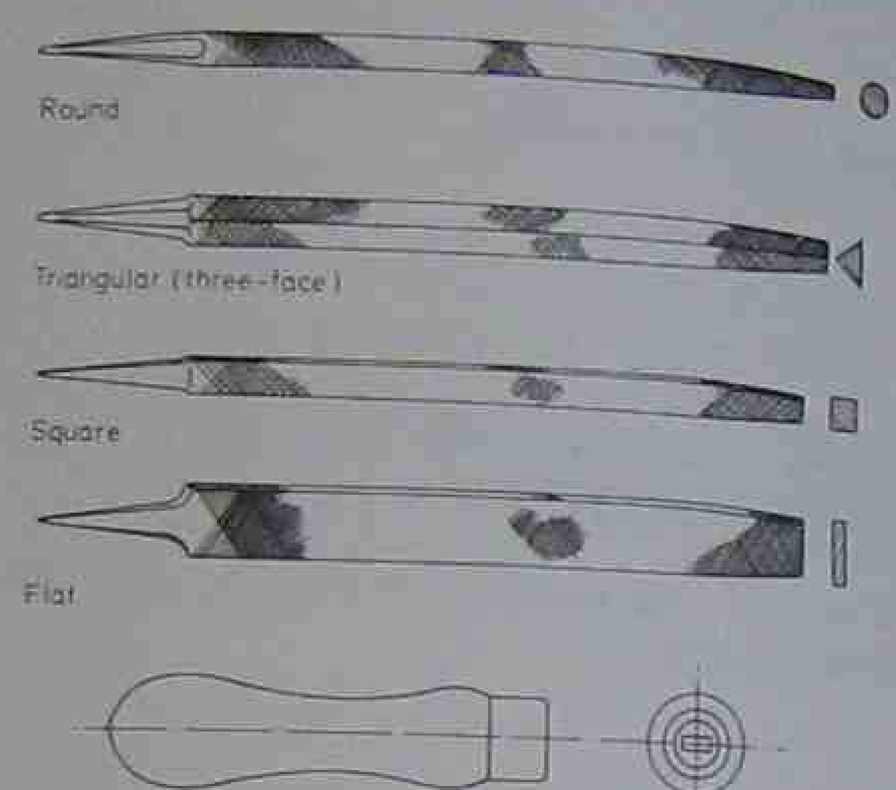


Figure 3.1 *Hand files and handle*

Needle files (pattern files) These are small, fine-tooth files with various cross-sections and sometimes with integral handles, used for diemaking and general toolroom work.

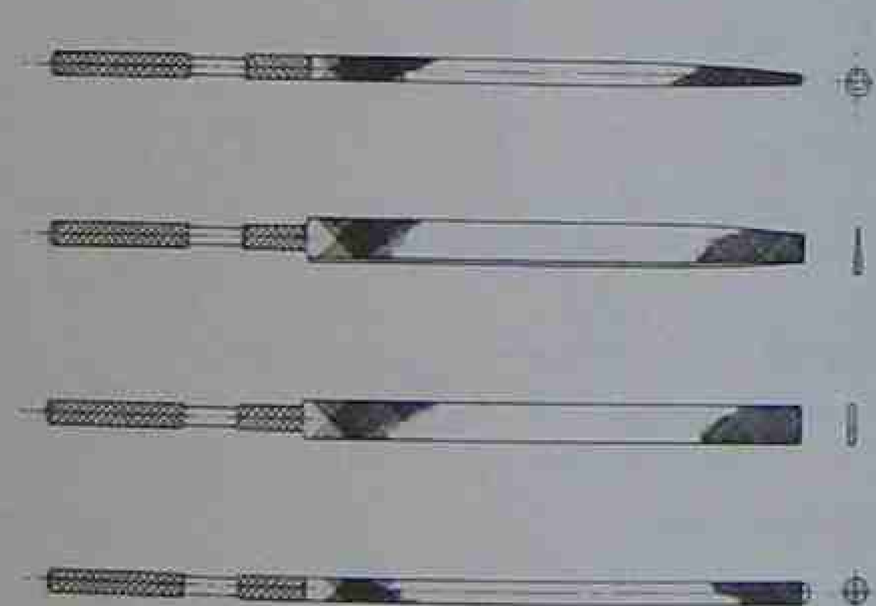


Figure 3.2 *Needle files (pattern files)*

Rotary files These are short sections of hardened steel with various profiles, such as cylindrical, conical and spherical, which fit into the chuck of a hand-held power tool. They are useful for filing awkward recesses in dies, etc.

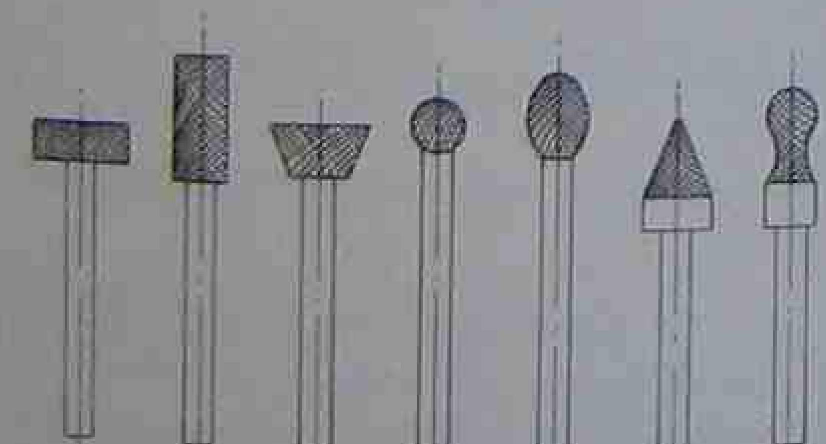


Figure 3.3 *Rotary files*

SCRAPERS

Scrapers are hand tools consisting of bar of various cross-sections, such as rectangular, half-round and triangular, which have sharp edges for removing metal from a flat or curved surface by scraping. The blades are made of hardened high-carbon or tool steel and fit into handles.

RASP

The rasp is a very coarse file used on soft metals, such

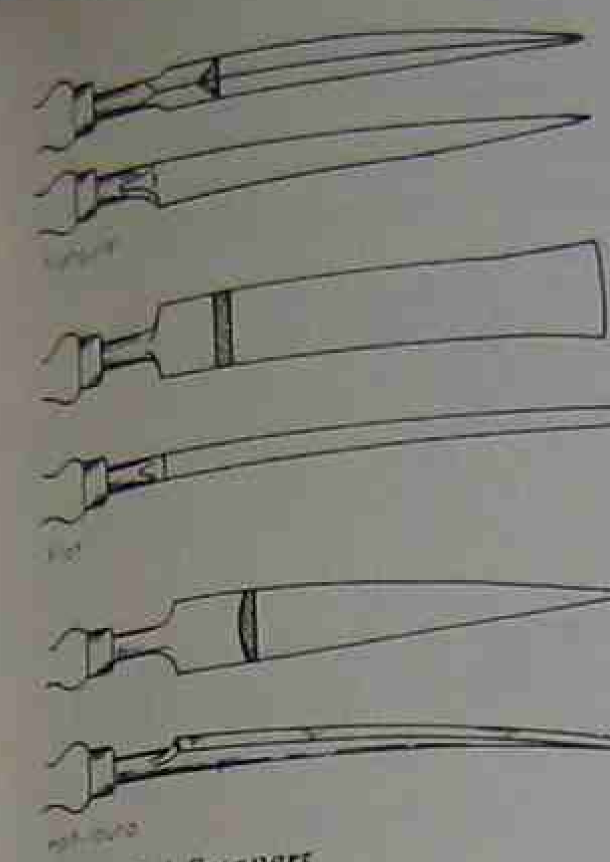


Figure 3.4 *Scrapers*

as lead-tin alloy, and non-metallic material, such as wood, hardboard and plastics.

The teeth of rasps are cut with a pointed punch which produces a series of sharp projections. The usual cross-section is half-round.

A proprietary rasp known as Surform has a thin, replaceable blade.

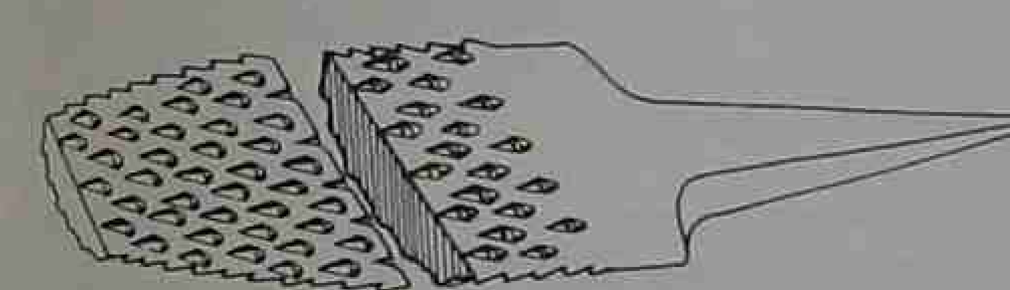


Figure 3.5 *Rasp*

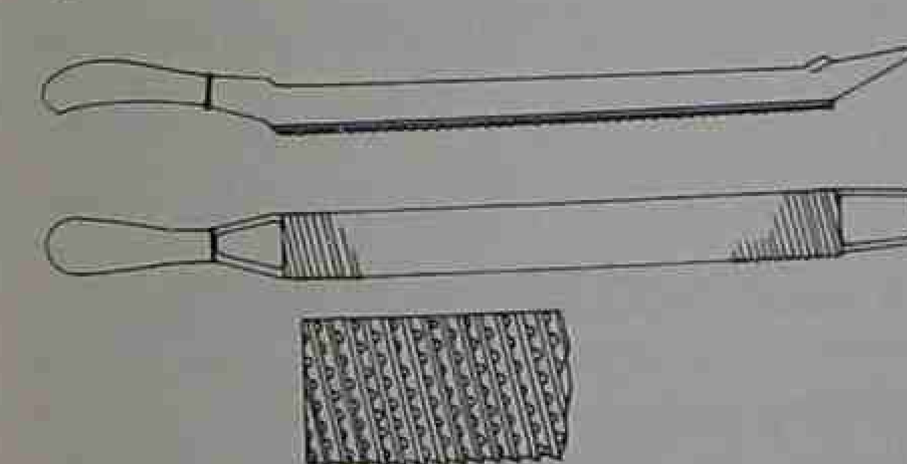


Figure 3.6 *Surform rasp*

HACKSAW

A hacksaw is a metal-cutting hand tool with a thin, narrow blade held in an adjustable frame. The blades are replaceable and the frame can accommodate blades of different lengths.

Hacksaw blades are made of carbon or high-speed steel with either the teeth or the whole blade hardened. The pitch of the teeth varies from 0.8mm ($^{1}/_{32}$ in.) to 1.8mm ($^{1}/_{14}$ in.). For general use with mild steel 16-18 teeth per inch are suitable, but fine pitch is used for thin sheet and tube and a coarse pitch for brass, copper and cast-iron.

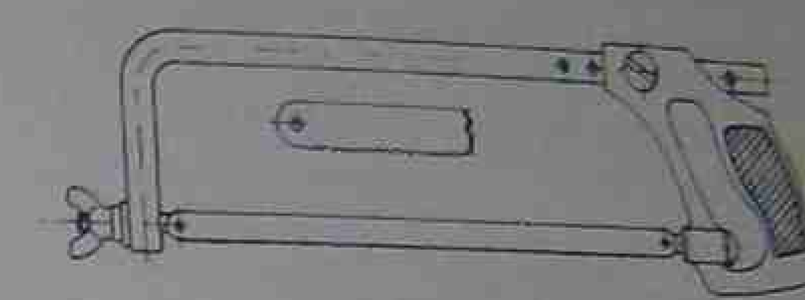


Figure 3.7 *Hacksaw*

CHISEL (SET)

Two main types of chisel, or *set*, are used by blacksmiths, the *hot chisel* for cutting hot metal and the *cold chisel* for cutting cold metal. Both are normally fitted with a handle and the chisel is held on the work and struck with a hammer. In engineering a fitter uses a shorter, hand-held cold chisel.

The most important types are the flat, the crosscut, the half-round and the diamond-pointed chisel.

Chisels are made from high carbon steel of octagonal cross-section varying in size from 10–20mm ($^{3}/_{8}$–$^{3}/_{4}$ in.), and alloy steel is also used.

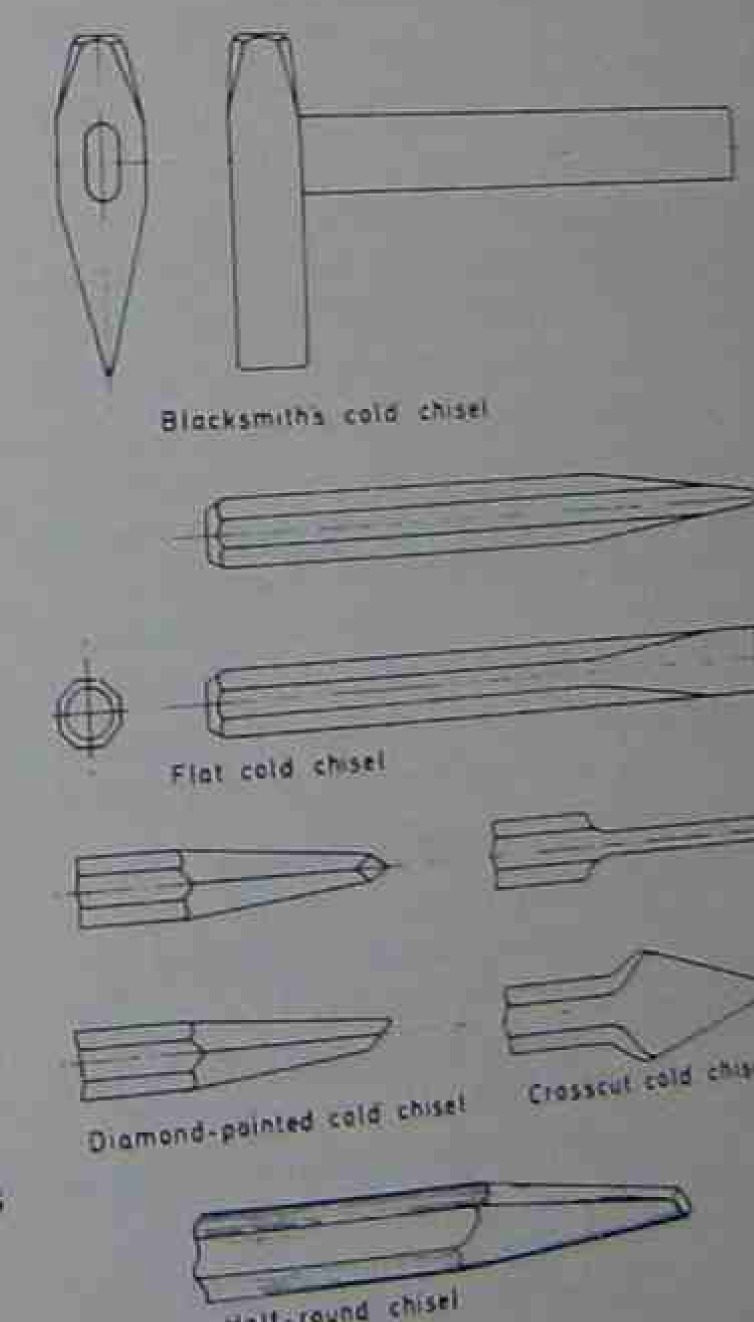


Figure 3.8 *Chisels*

CENTRE PUNCH

The centre punch is used to give small, round indentations when struck with a hammer. The marks are used in conjunction with scribed lines when marking-off for the centres of drilled holes and to provide a hole for one leg of a pair of dividers when marking off circles. Punches are made of hardened and tempered cast steel.

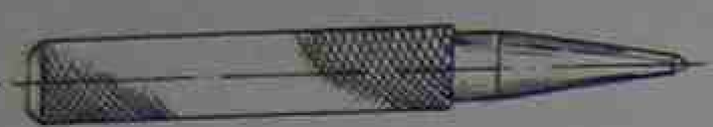


Figure 3.9 *Centre punch*

DRIFTS

Drifts are punches made of hardened and tempered tool steel with specially shaped ends used for finishing off non-circular holes, e.g. square, rectangular and hexagonal.

One type of square drift has teeth on its faces to improve the cutting action.

Pin drift A drift with a small-diameter end used for driving pins out of shafts, etc. They are made in several sizes.

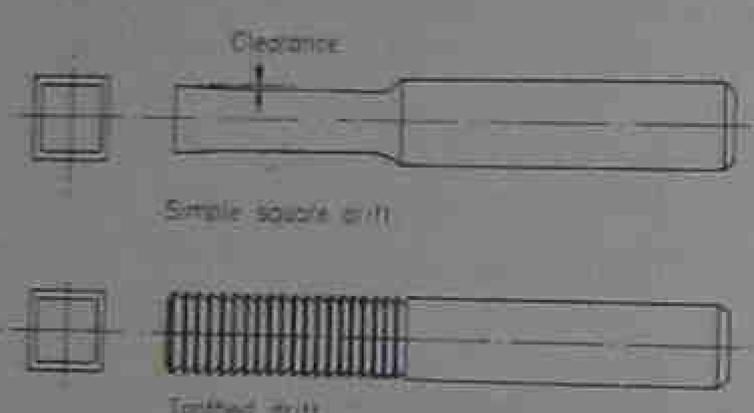


Figure 3.10 *Drifts*

Figure 3.11 *Pin drift*

TWIST DRILL

A twist drill is a manually- or machine-rotated tool with cutting edges for producing circular holes in metal, plastics, wood, etc. It consists of pieces of hardened steel bar with usually two spiral grooves, or flutes, ending in two angled cutting edges. The flutes assist in removing the cuttings.

Twist drills range in size from a fraction of a mm to over 10 cm. The shank, that is the end opposite to the cutting end, is either straight to suit a drill 'chuck' or has a taper, known as a 'Morse taper', to suit the spindle of a drilling machine.

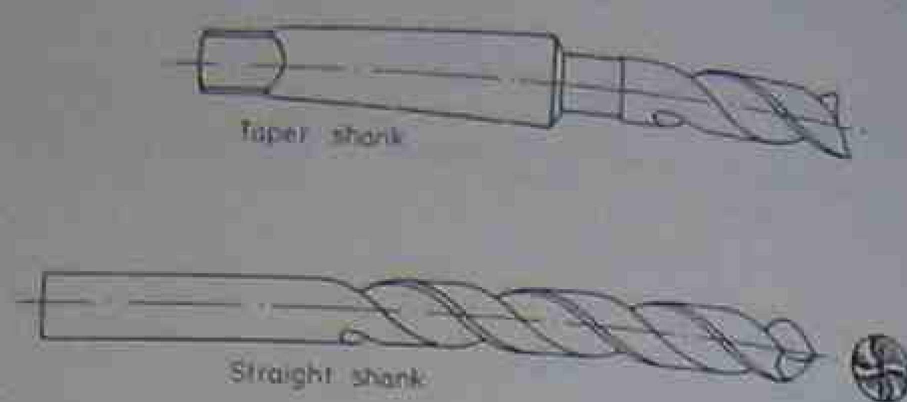


Figure 3.12 *Twist drills*

HAND DRILL

A hand drill is a tool with a chuck which takes twist drills. The drill is rotated by turning a geared handle. The name is also applied to hand-held electrically powered drills.

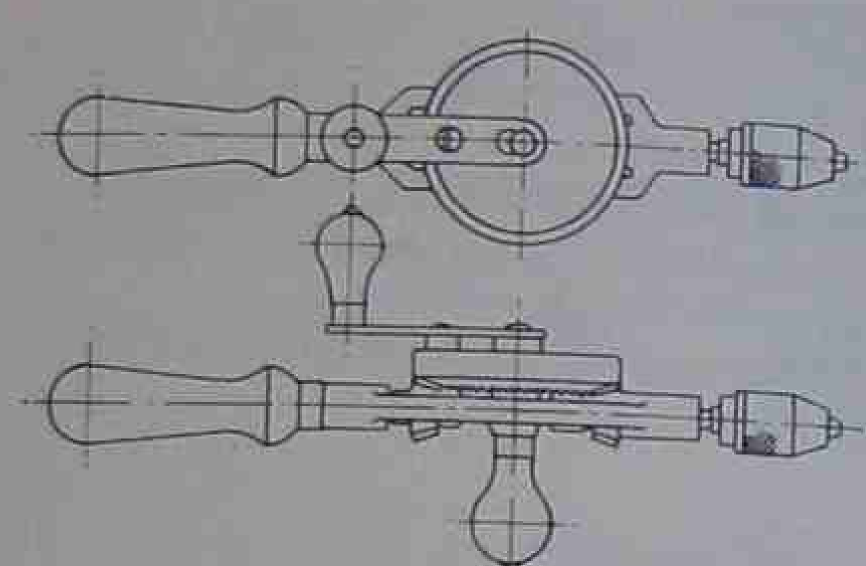


Figure 3.13 *Hand drill*

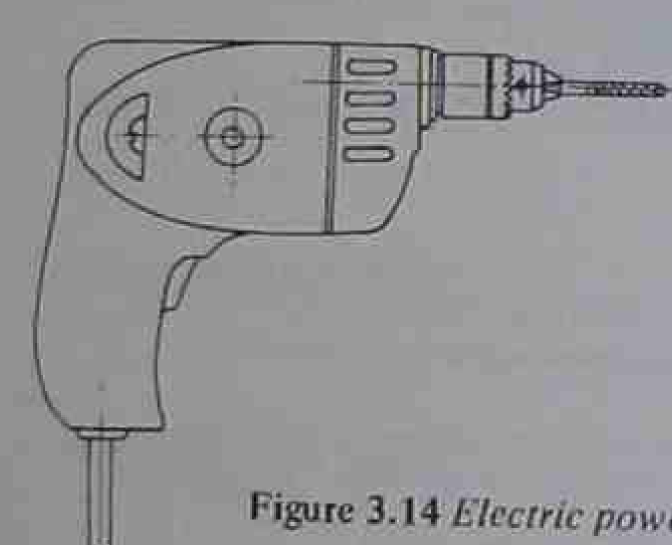


Figure 3.14 *Electric power hand drill*

MASONRY DRILL

A twist drill with hardened inserts on the cutting edges suitable for drilling bricks, concrete, etc.

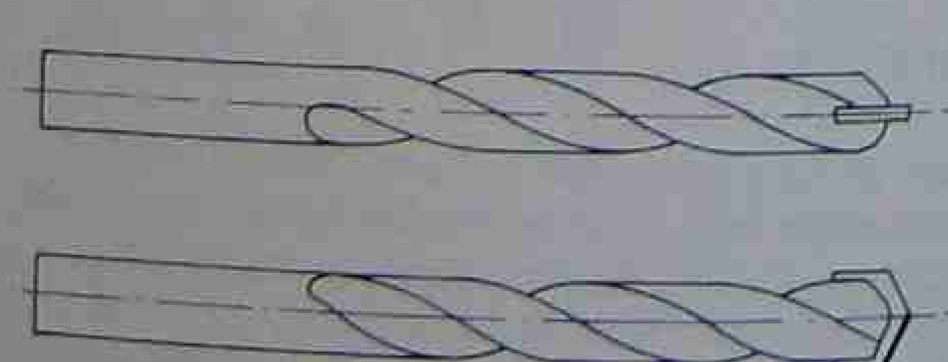


Figure 3.15 *Masonry drill*

COUNTERSINKING TOOL

A rotating cutting tool with a conical end having cutting teeth used for countersinking drilled holes.

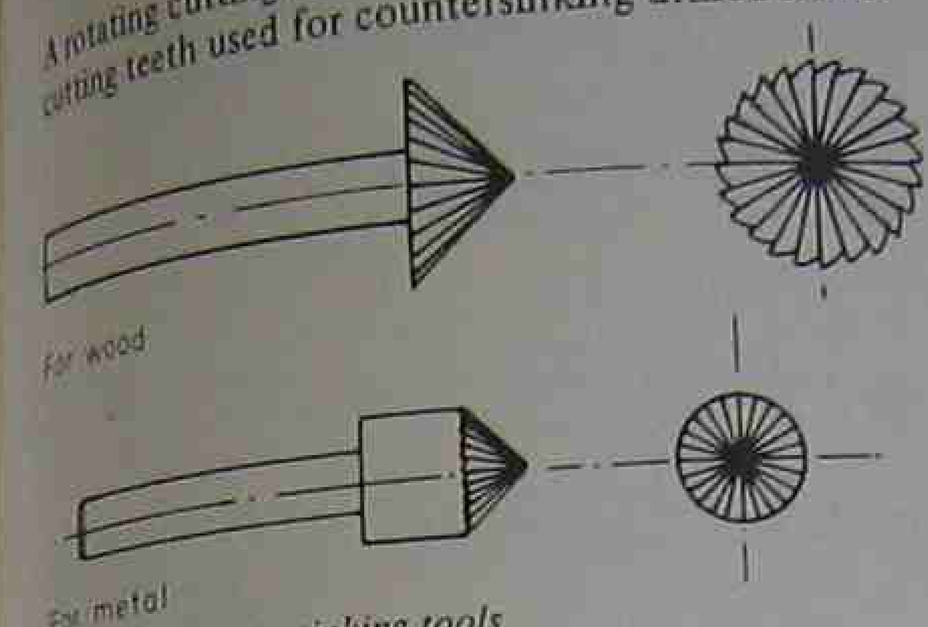


Figure 3.16 *Countersinking tools*

REAMER

A cylindrical cutting tool similar in appearance to a drill but with a flat end and either straight or helical cutting edges on the cylindrical surface. Hand reamers have a long lead (tapered section) to align the reamer with the hole.

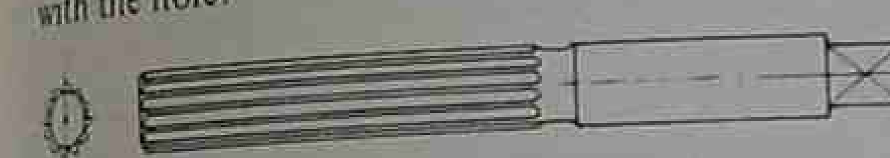


Figure 3.17 *Reamer*

TAPER REAMER

This is a small reamer with the cutting length tapered to produce holes for taper pins.

Figure 3.18 *Taper reamer*

TAP

A tap is a tool used to produce an internal screw thread in a drilled or bored hole. It is essentially a screw with three or four longitudinal grooves providing cutting edges which, when hardened, will cut a mating thread inside a hole. The shank has a square end which will fit either a machine spindle or a hand-tapping wrench.

A set of taps consists of three taps with the same thread but with different tapers at the end. The first has a long taper for starting the thread, the second a shorter taper for finishing through holes, and the third tap is a plug tap for taking the thread to the bottom of a blind hole. The hole must be initially drilled to the tapping size, that is to a diameter slightly smaller than the diameter at the bottom of the threads.

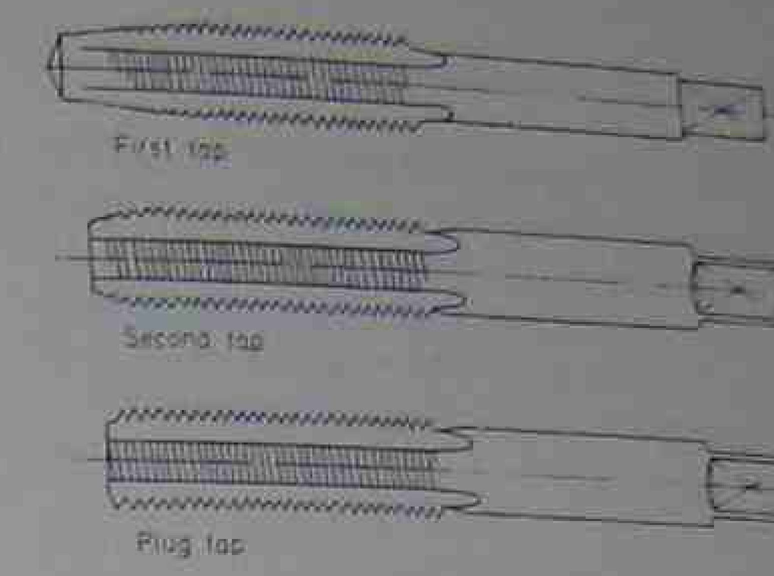


Figure 3.19 *Set of screw taps*

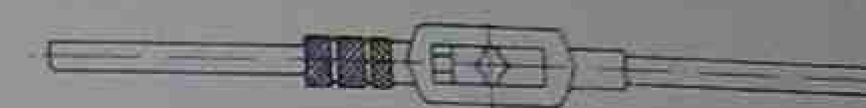


Figure 3.20 *Tap wrench*

TAP WRENCH

A tap wrench which is used for manual tapping, comprises a bar with a central adjustable hole (to suit the square on a tap shank) and handles.

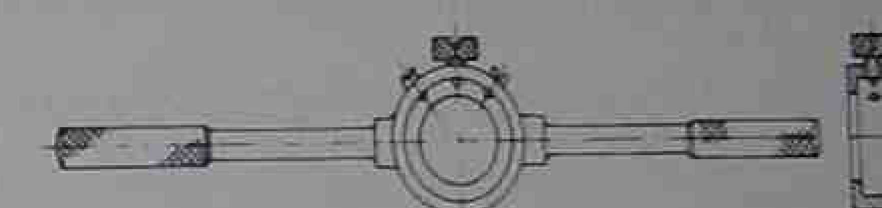


Figure 3.21 *Die stock*

DIE

A die is used to cut an external screw thread on a cylindrical surface. It consists of a hardened, circular steel nut with an internal thread which is relieved by three or four longitudinal grooves to produce cutting edges. The die is split so that the diameter of the hole can be reduced by means of two screws and increased by means of a third pointed screw. The die is held in a circular recess in a 'stock' fitted with handles for manual operation.

The first cut is made with the die opened slightly, and second or third cuts are made with it progressively closed until the correct depth of thread is obtained.

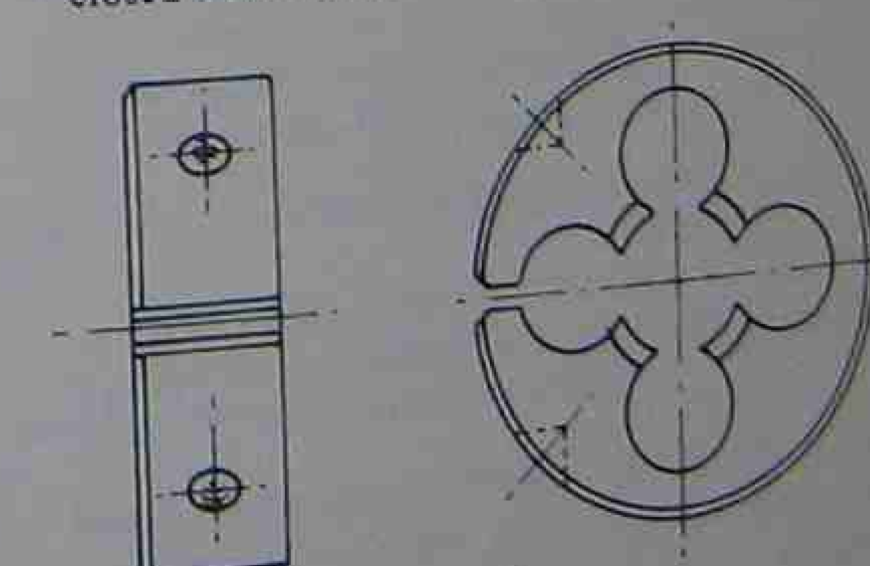


Figure 3.22 *Thread-cutting die*

HAMMERS

A hammer is a hand tool with a heavy steel head on the end of a hand-held shank usually made of wood but sometimes made of steel tube with a handgrip.

Hammers are available in many sizes and weights from 100 gm to 10 Kg, and with many shapes of head, and they are used to give blows of varying force.

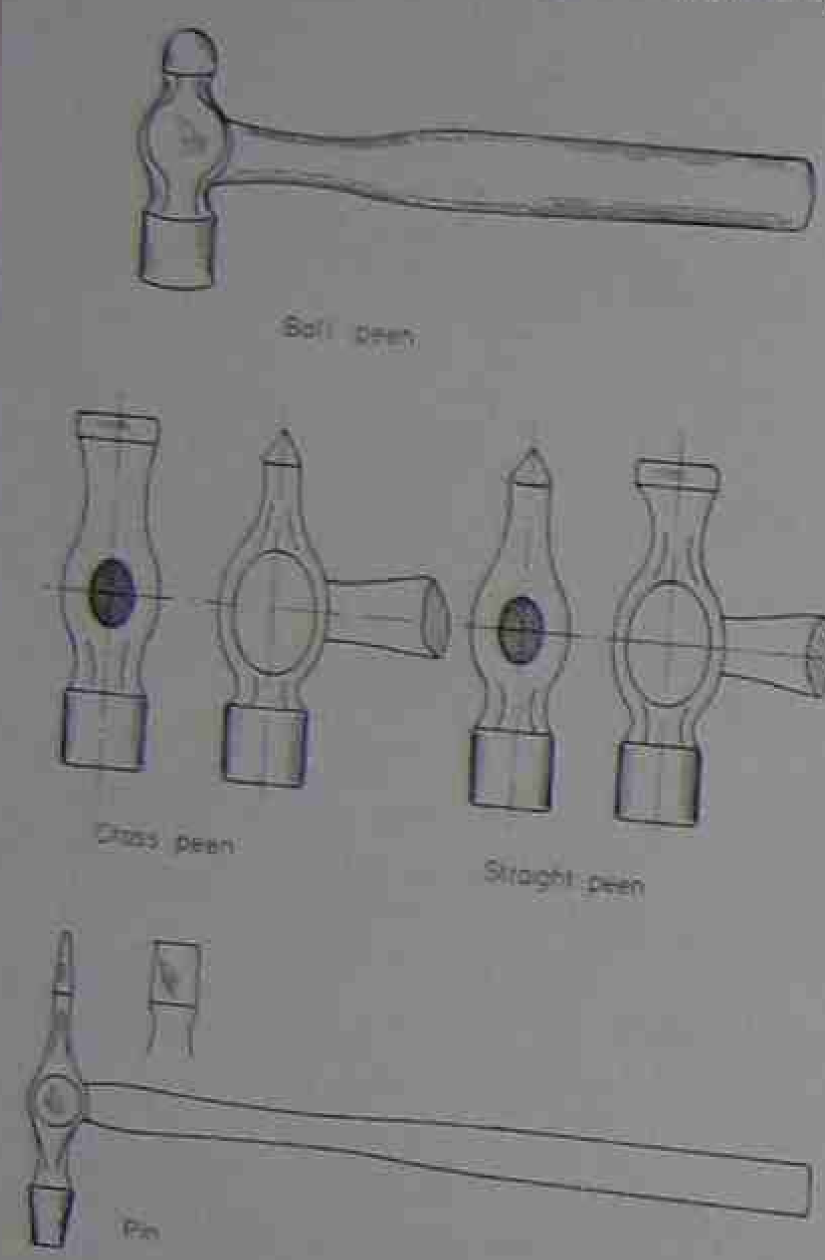


Figure 3.23 *Hammers*

Figure 3.24 *Sledge hammer (maul)*

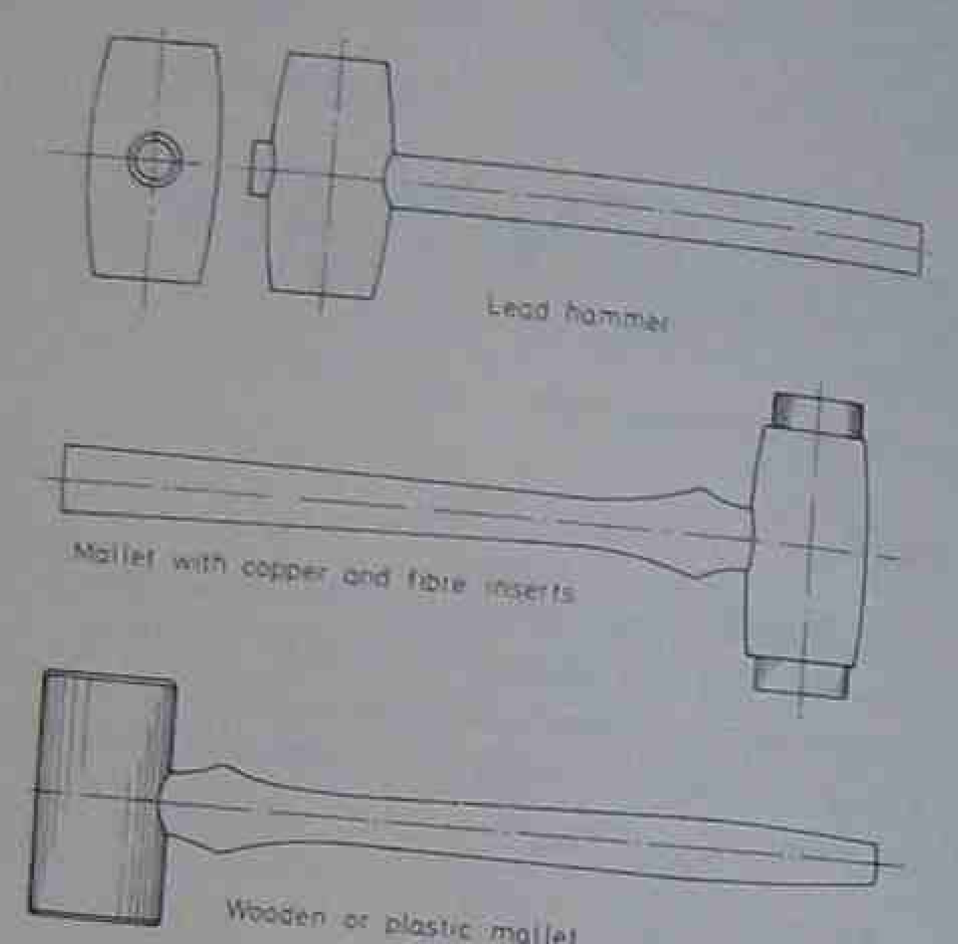


Figure 3.25 *Soft hammers (mallets)*

Ball peen hammer This is the most common type of hammer which has a ball-shaped end on the head opposite to the striking face. It is used for all kinds of engineering work, the ball end being useful for sheet metal forming, and it is made in weights from 100 gm to 1.5 Kg.

Cross peen and straight peen hammers These have blunt, chisel-shaped ends on the head opposite the face.

Pin hammer A name often applied to the smallest hammers, and these may be ball peen or cross peen.

Sledge hammer (maul) This is a heavy, double-faced hammer weighing up to 15 Kg which is swung using both hands.

Soft hammers (mallets) To avoid damage to machined surfaces soft hammers, or mallets, are used, the main types being: lead hammer, with iron handle; mallet, with copper and fibre inserts in an iron head; mallets with heads of boxwood, plastic or hide.

RULES

Engineers' rules are made from hardened and tempered steel marked off with high accuracy. They are made from strips 10–30 mm wide and in lengths from 10 cm (4 ins). Folding pocket rules are available with an extended length of 60cm (24 ins.).

Figure 3.26 *Engineers' rule – 12 in. (300mm)*

Extending spring rules are used where great accuracy is unnecessary.

In engineering, rules are used for marking-off, setting callipers and dividers, etc.

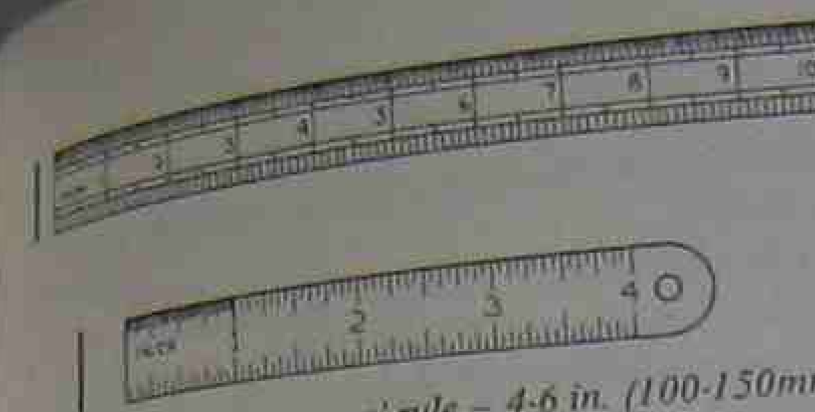


Figure 3.27 *Small engineers' rule – 4-6 in. (100-150mm)*

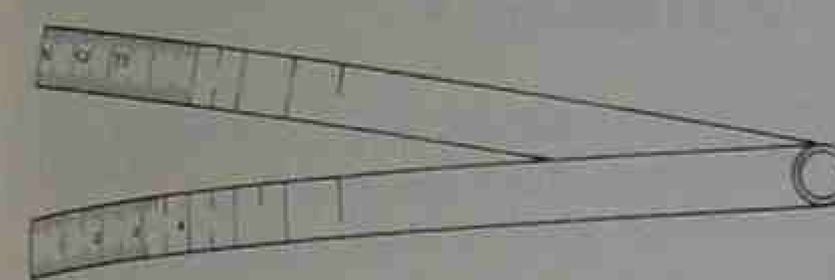


Figure 3.28 *Folding rule*

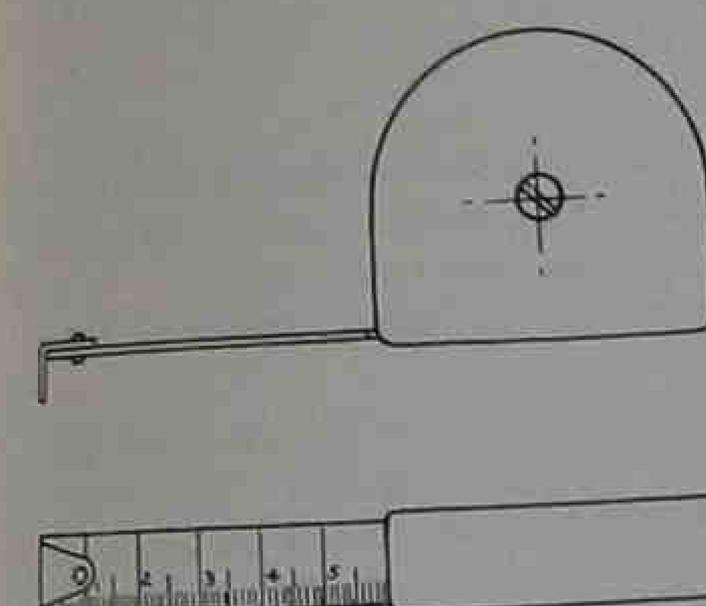


Figure 3.29 *Spring tape rule*

THICKNESS GAUGE (FEELER GAUGE)

Thickness, or feeler, gauges are thin blades of spring steel of exact thickness used for measuring small gaps (or clearances) between parts. They are usually made in sets of various thicknesses pinned together at one end.

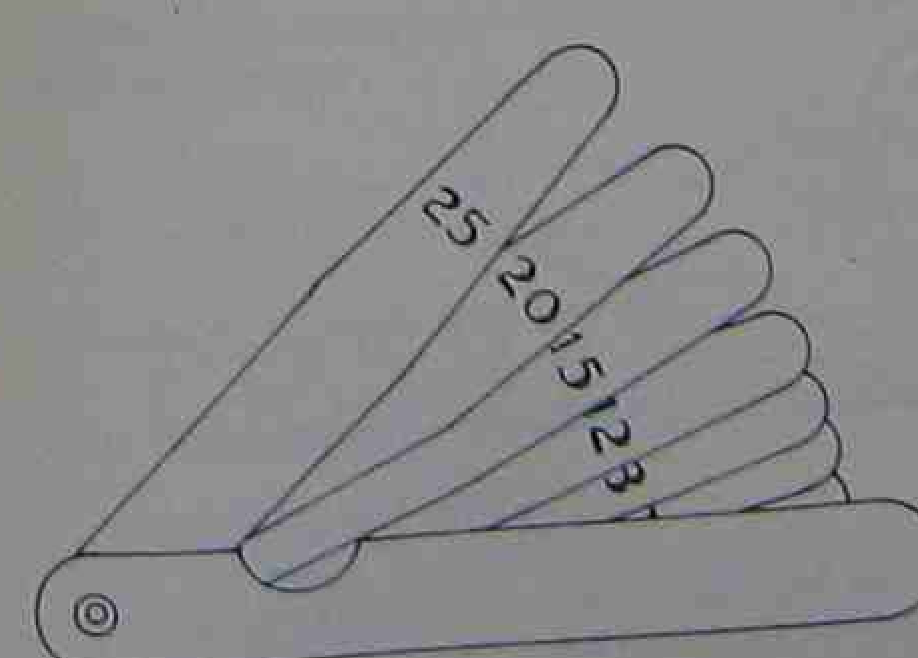


Figure 3.30 *Thickness gauge (feeler gauge)*

THREAD GAUGE (SCREW PITCH GAUGE)

The thread gauge has several blades, each with a number of teeth of different standard pitch and form, and mounted at either end of a holder. These blades are used when machining threads so that the thread form and pitch may be checked.

Another type of gauge is used for setting thread-cutting tools.

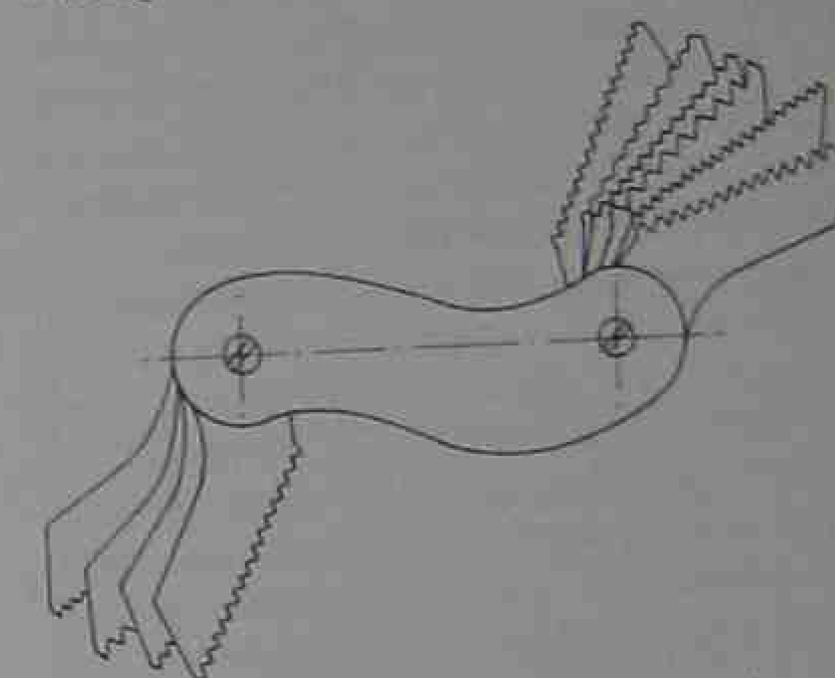


Figure 3.31 *Thread gauge (screw pitch gauge)*

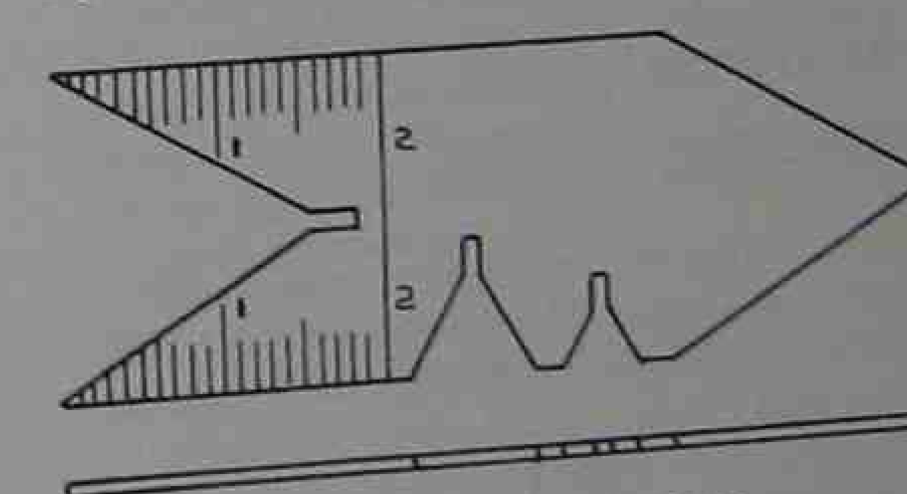


Figure 3.32 *Screw-cutting tool setting gauge*

SCRIBER

A scriber is a thin steel rod with a pointed end or ends, one of which may be bent at right angles to allow access to holes. Some scribers may have a knurled handle in the middle.

The scriber is used for marking-off, and it produces a fine scratched (or scribed) line on a machined face.

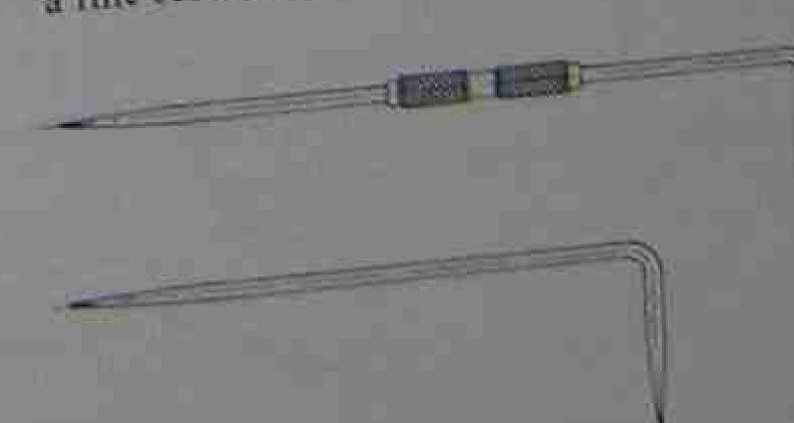


Figure 3.33 *Scribers*

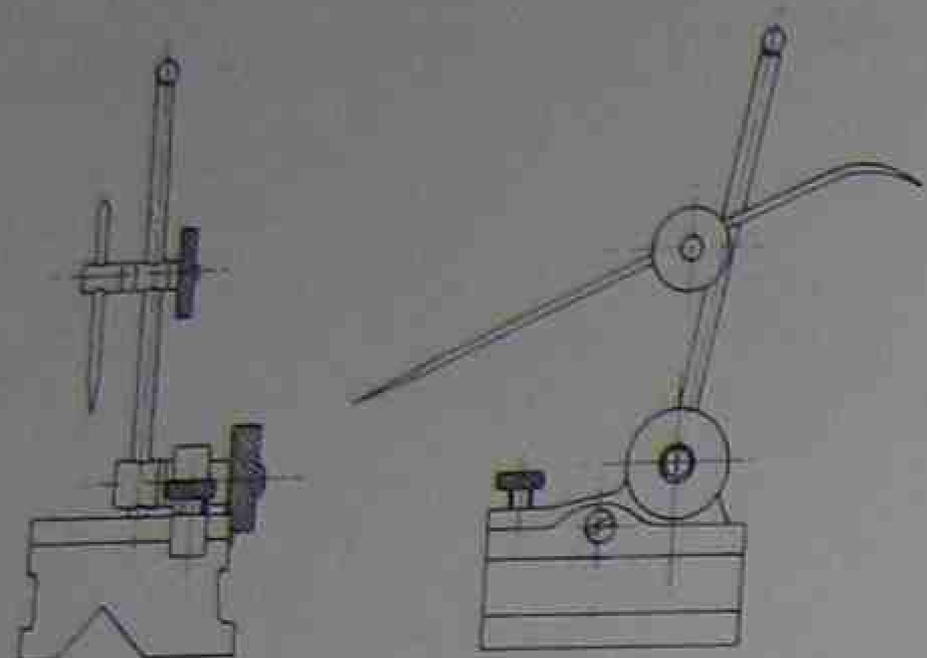


Figure 3.34 *Surface gauge with scriber*

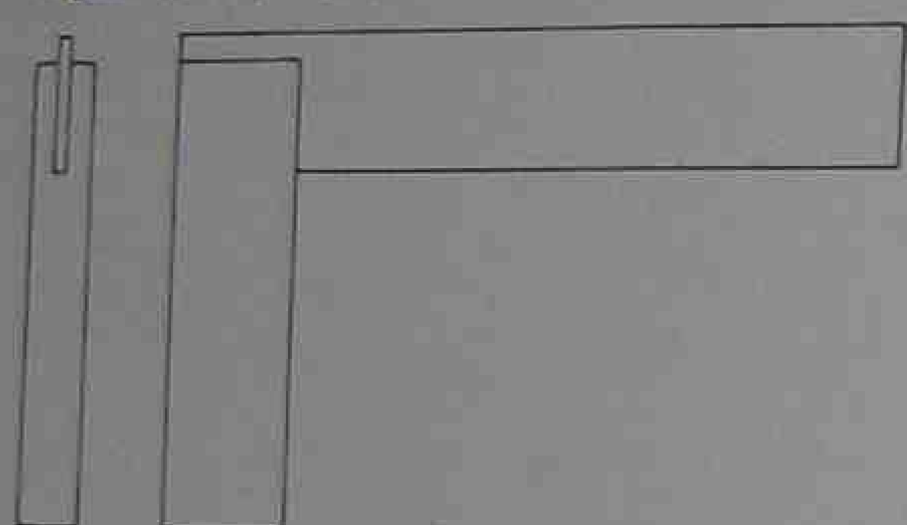


Figure 3.35 *Square (try square)*

SQUARE (TRY SQUARE)

This is made from two rectangular pieces of steel joined at right angles, and is used to check internal and external right angle corners, especially when hand-fitting.

SURFACE GAUGE

A surface gauge has three main parts, a base, a post and a scriber.

The base is ground flat and has a machined V groove for use on round work. The post can be tilted by means of a screw and carries either a scriber for marking-off, or a dial gauge.

COMBINATION SET

This consists of a steel rule with a slot (or keyway) along its length, to which one of three heads may be attached, namely, a square, a centre finder and a protractor.

DIVIDERS

Dividers consist of a pair of adjustable hinged points used for measuring and transferring sizes and also for scribing circles and radii when marking-off.

The hinge may be of the 'firm joint' type or with a spring, as shown.

OUTSIDE CALLIPERS (CALLIPERS)

An adjustable pair of curved and hinged legs used for measuring the thickness of parts and diameters of shafts, etc. The callipers may have a 'firm joint', as shown, or be of the 'spring joint' type.

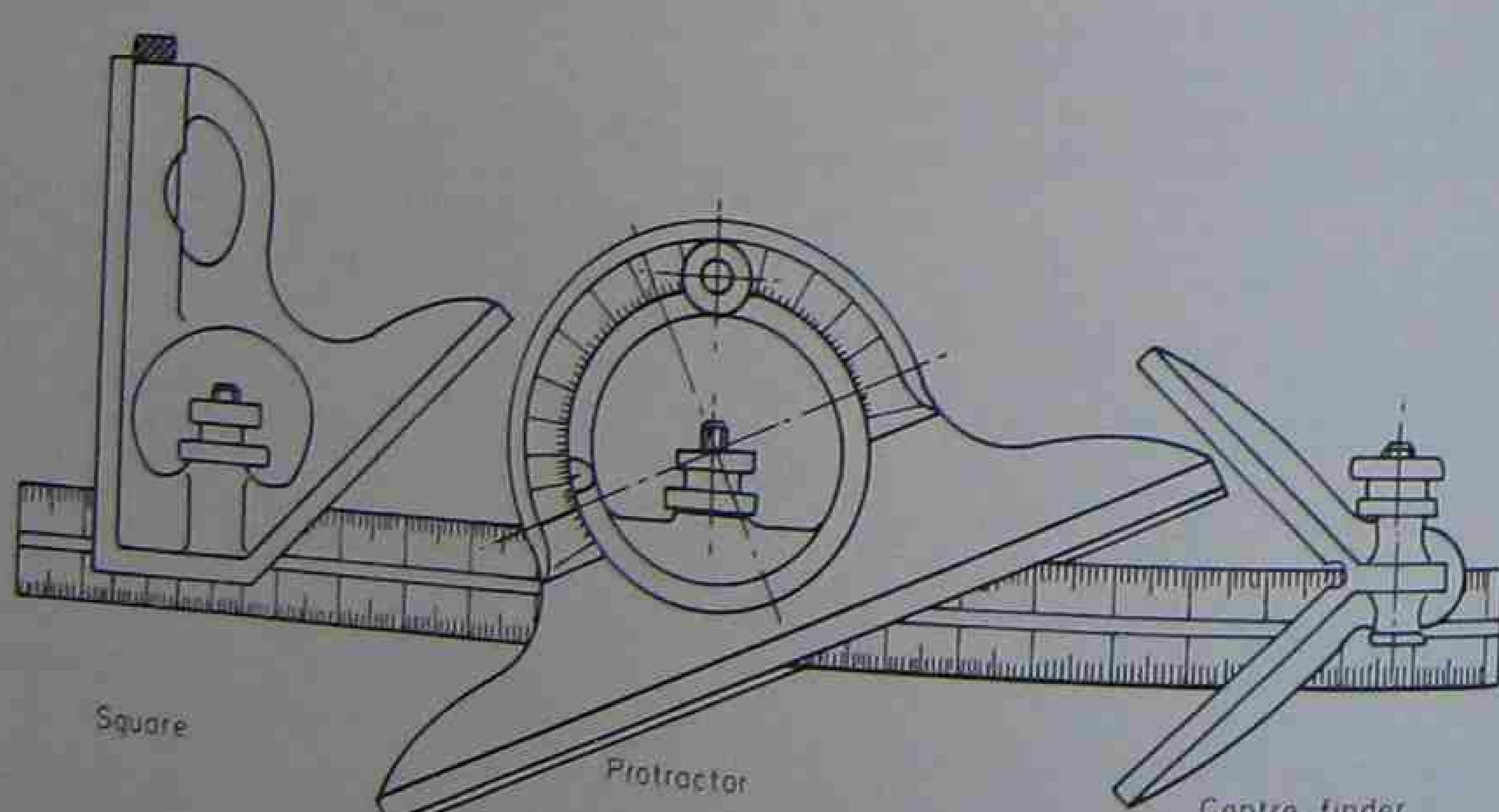


Figure 3.36 *Combination set*

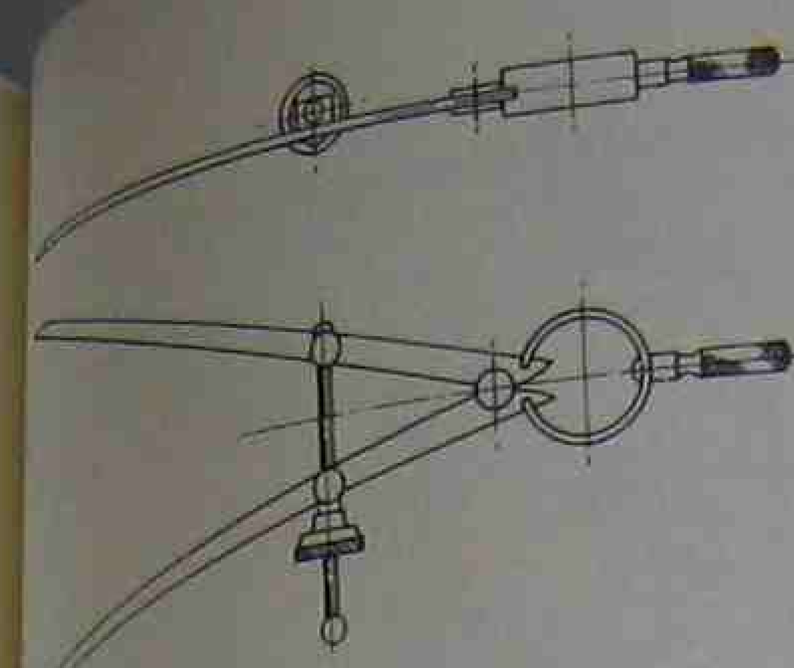


Figure 3.37 *Spring dividers*

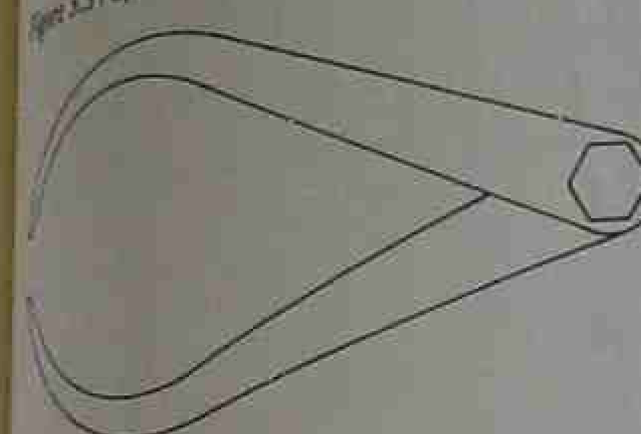


Figure 3.38 *Outside callipers*

INSIDE CALLIPERS

The hinged legs are bent outwards so that inside measurements such as the bore of a pipe may be measured.

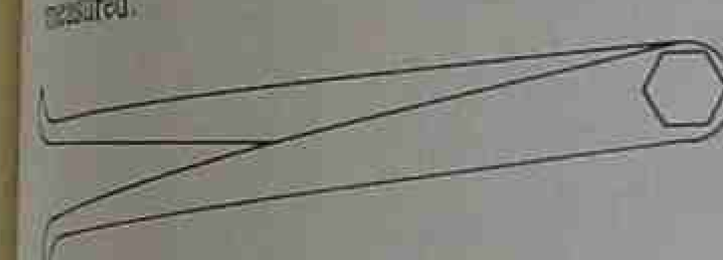


Figure 3.39 *Inside callipers*

JENNY CALLIPERS

Also known as *hermaphrodite callipers* or *odd legs*, these have one leg curved inwards and one straight leg which sometimes has a replaceable point. They are used for measuring the distance from a point to an outside surface.

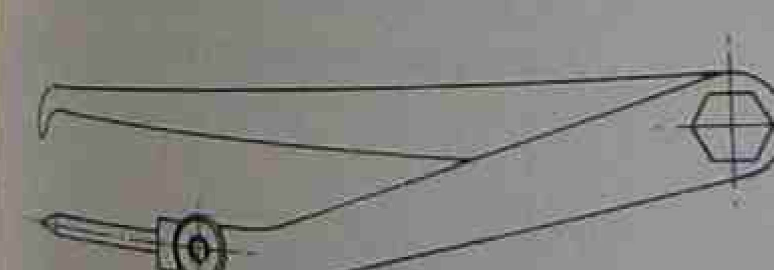


Figure 3.40 *Jenny (hermaphrodite, odd leg) callipers*

GEAR-TOOTH CALLIPERS

A gear-tooth calliper has an 'inside' leg and an 'outside' leg. It can be used for measuring the distance from an inside edge to an outside edge.

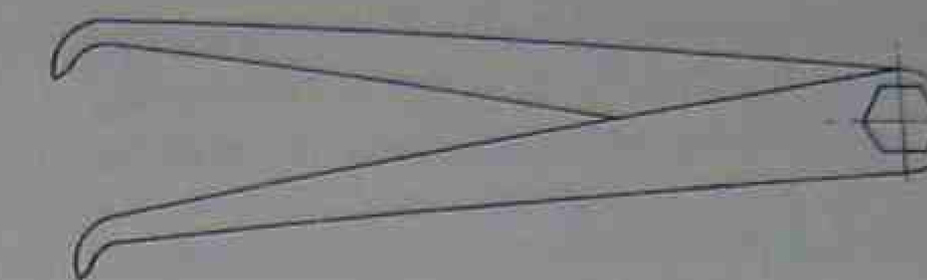


Figure 3.41 *Gear-tooth callipers*

MICROMETERS

Micrometers are instruments used for the accurate measurement of internal and external dimensions of objects, particularly those of cylindrical shape. The measurement is based upon the advance of an internally-threaded thimble rotated on a precision screw.

Micrometers are made in a very large range of types and sizes, the most common being the *outside micrometer* measuring up to 25mm in 0.01mm intervals (1 in. in 0.001 in. intervals). It has a fixed graduated barrel screwed to take the graduated thimble which is knurled to give a good finger grip and which has a movable anvil attached to it. The barrel is attached to a semi-circular frame at the opposite end of which is mounted a fixed anvil.

The object to be measured is placed between the anvils and the thimble is rotated until the object is nipped by them. The size is then read from the graduations on thimble and barrel.

Large outside micrometers are supplied with extensions for the fixed anvil.

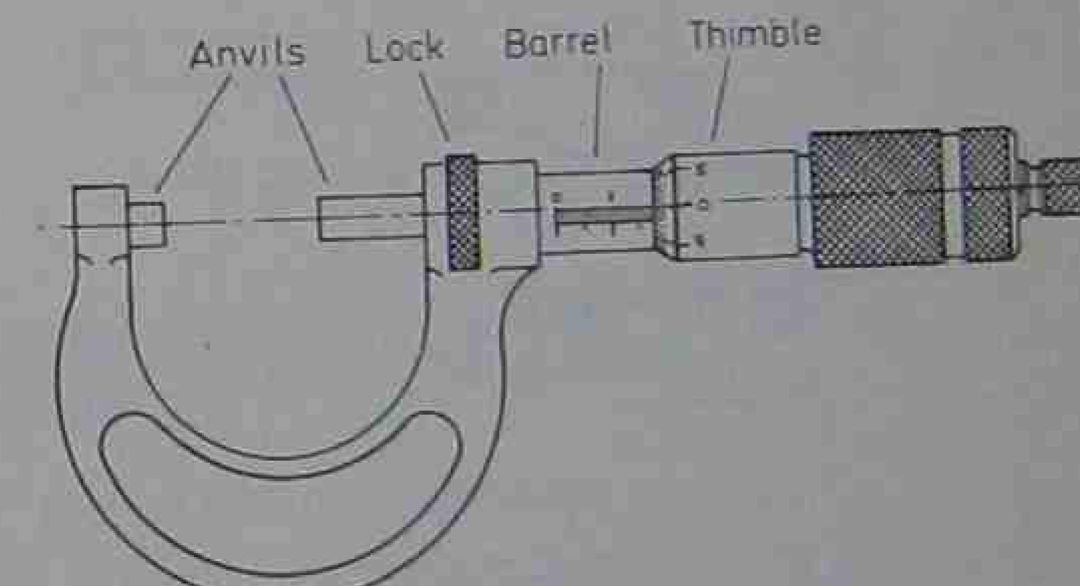


Figure 3.42 *Outside micrometer (1 in. – 25mm)*

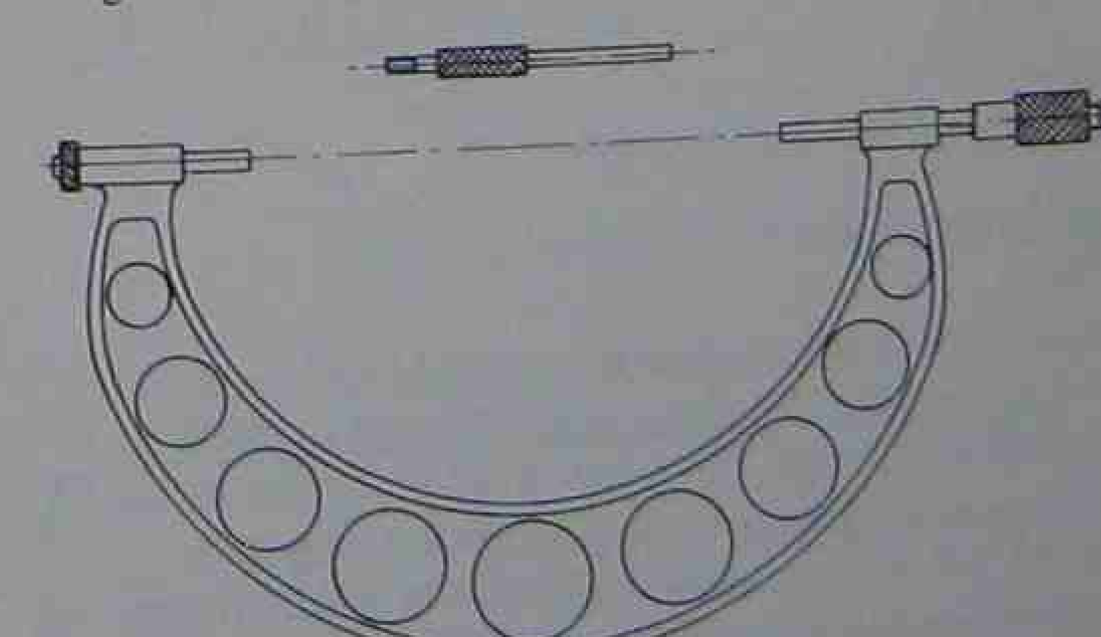


Figure 3.43 *Large outside micrometer with extension rod*

An *inside micrometer* has the fixed anvil projecting from the end of the thimble opposite to the moving anvil, and extension rods may be used to extend the range.

The barrel and thimble assembly, called a *micrometer head*, can be obtained separately and can be attached to any precision measuring instrument.

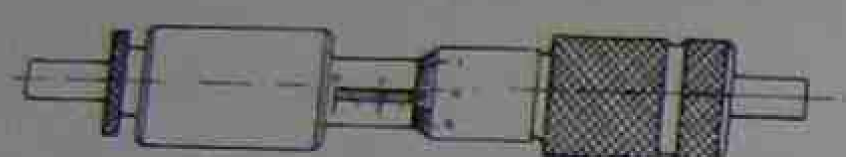


Figure 3.44 *Inside micrometer*

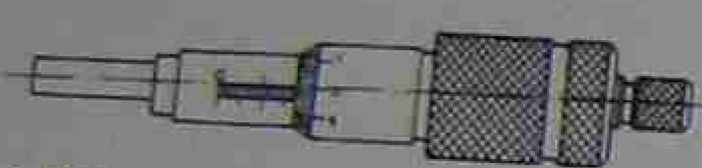
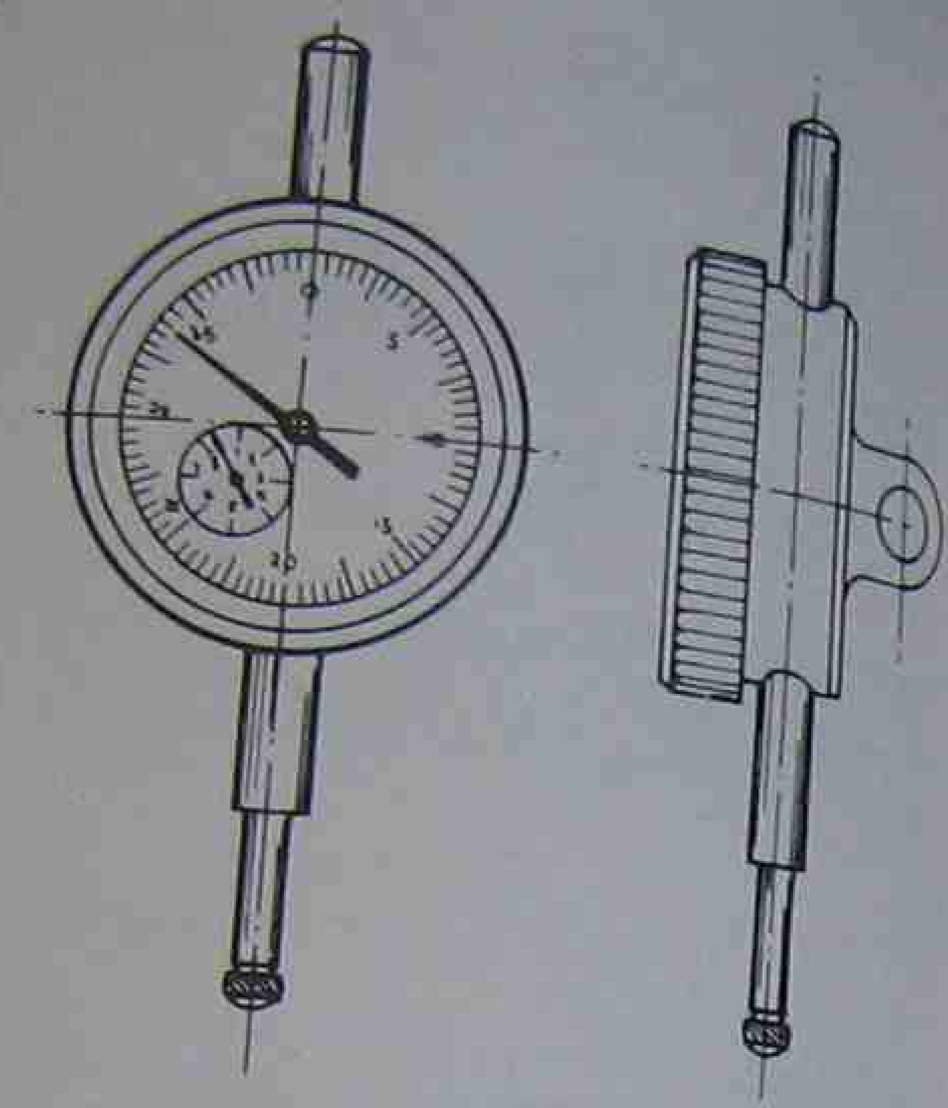


Figure 3.45 *Micrometer head*

VERNIER CALLIPER GAUGE

This instrument, used for internal and external measurement, has a long flat scale to which a fixed jaw is attached at one end, and a sliding jaw with a cursor running along the scale. A scale on the cursor is read in conjunction with the fixed scale.

Outside measurements are made between the jaws and inside measurements over projections on the jaws.



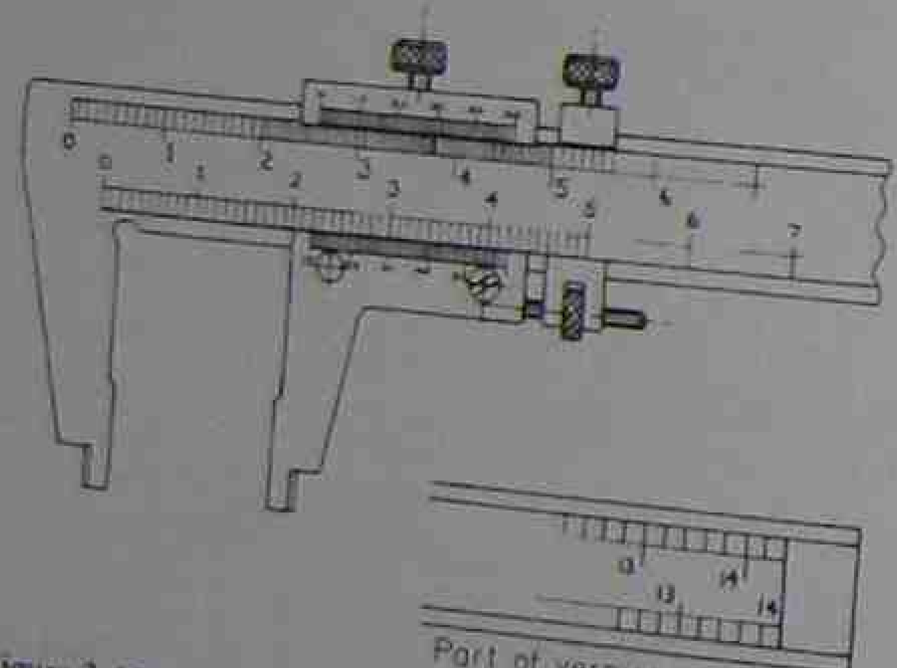


Figure 3.46 *Vernier calliper gauge*

DIAL GAUGE

The movement of a plunger is magnified and displayed on a dial on which intervals of 0.002 mm (0.0001 in.) are possible. The dial has a fixing lug by which it can be attached to a surface gauge post or a magnetic base test set. When a bell-crank lever is attached, the combination is referred to as a *Dial Test Indicator-lever operated*.

Figure 3.47 *Dial gauge*

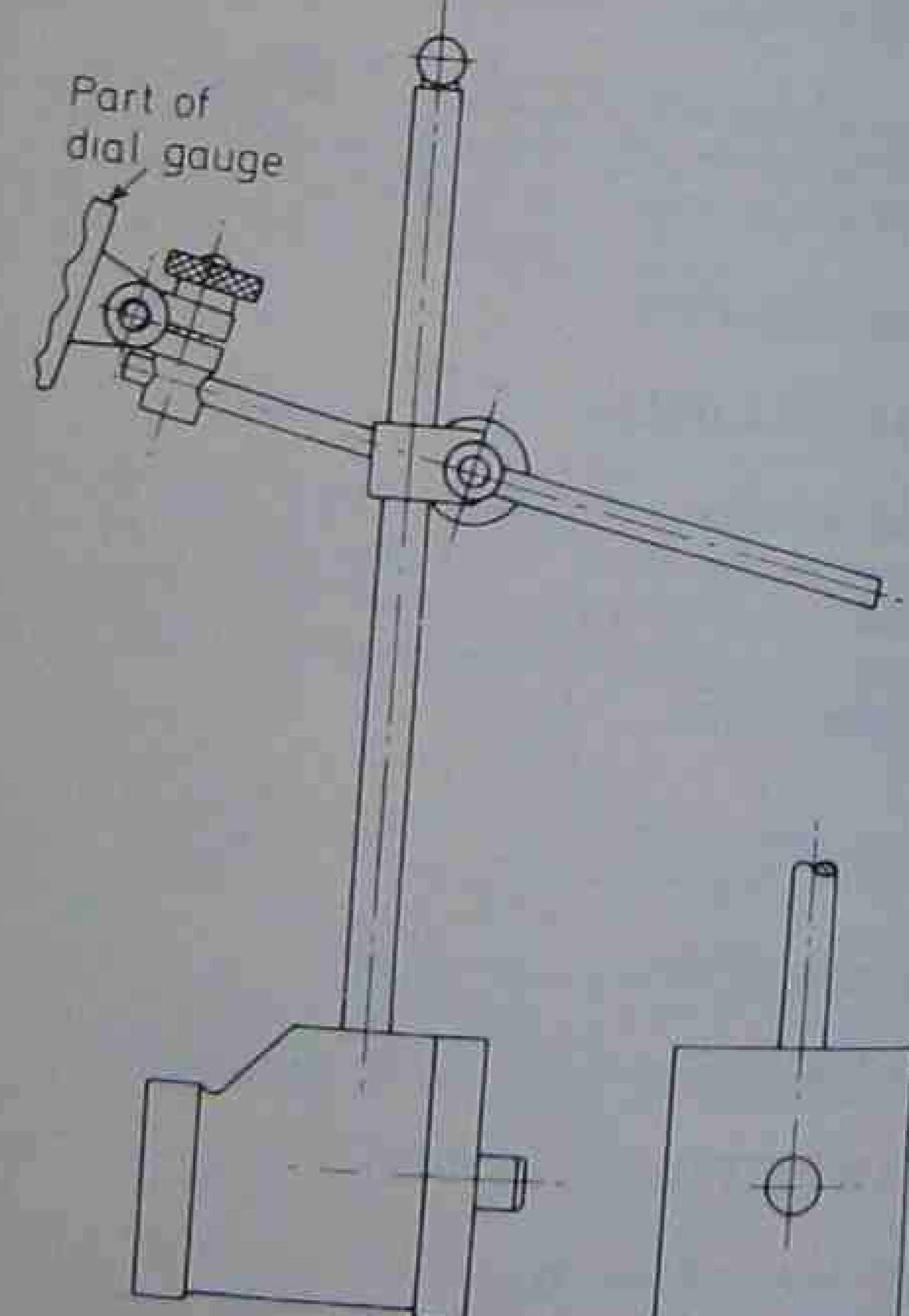


Figure 3.48 *Magnetic base test set*

V BLOCK

V blocks are used for holding cylindrical objects while they are being machined or marked off. They have a ground base and sides with a ground V in the top. Sometimes the sides have grooves to take a clamp for holding the object rigidly onto the block.

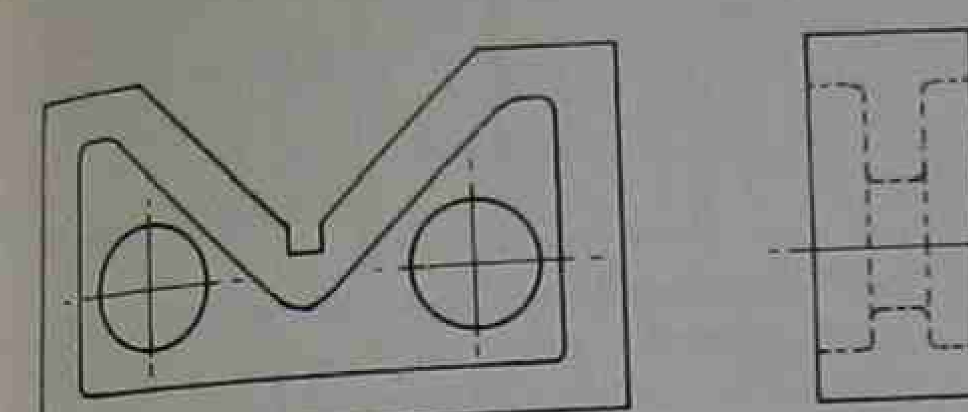


Figure 3.49 *V block*

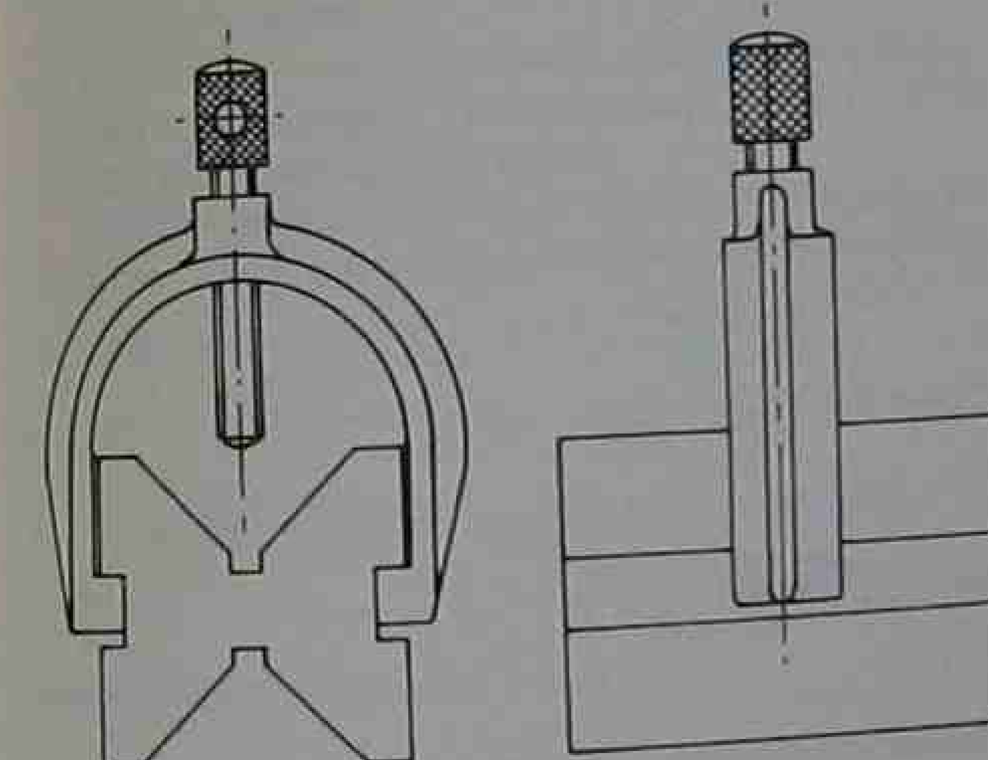


Figure 3.50 *V block with clamp*

SURFACE PLATE

A heavy plate of cast iron or steel with a flat surface of high accuracy used for marking off and gauging. A larger version is called a *surface table*.

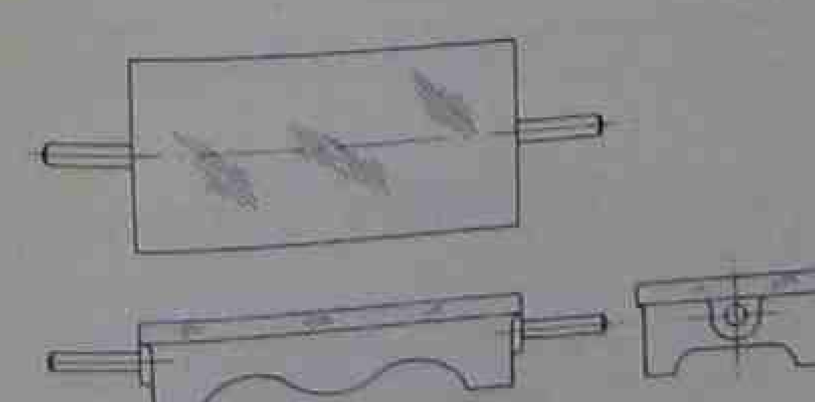


Figure 3.51 *Surface plate*

CLAMPS

Various types of clamp are used for temporarily holding down during machining, bolting, riveting etc. The range includes the *C clamp*, the *toolmakers' clamp* and the *strap clamp*.

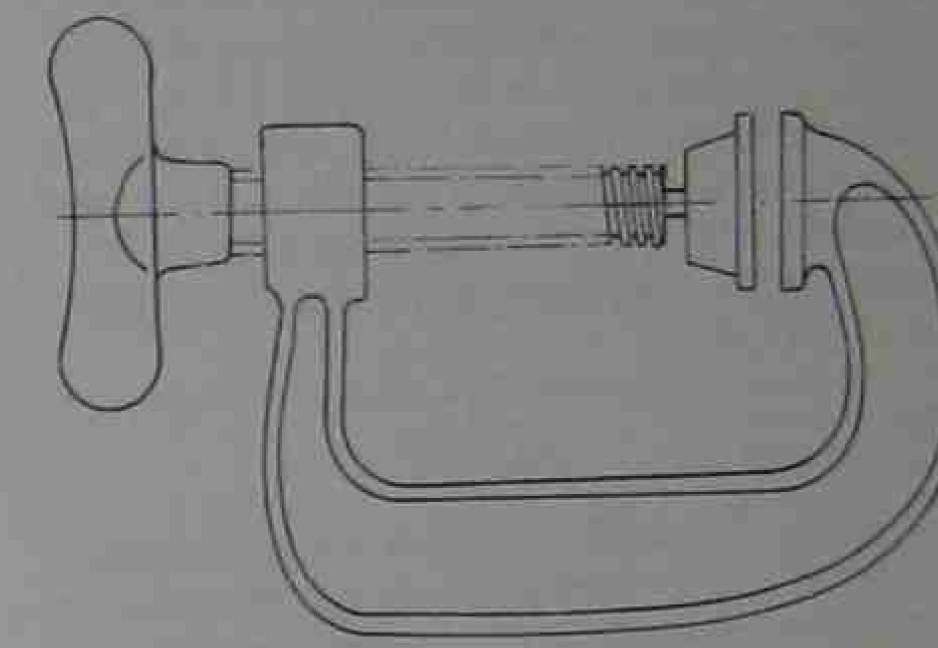


Figure 3.52 *C clamp*

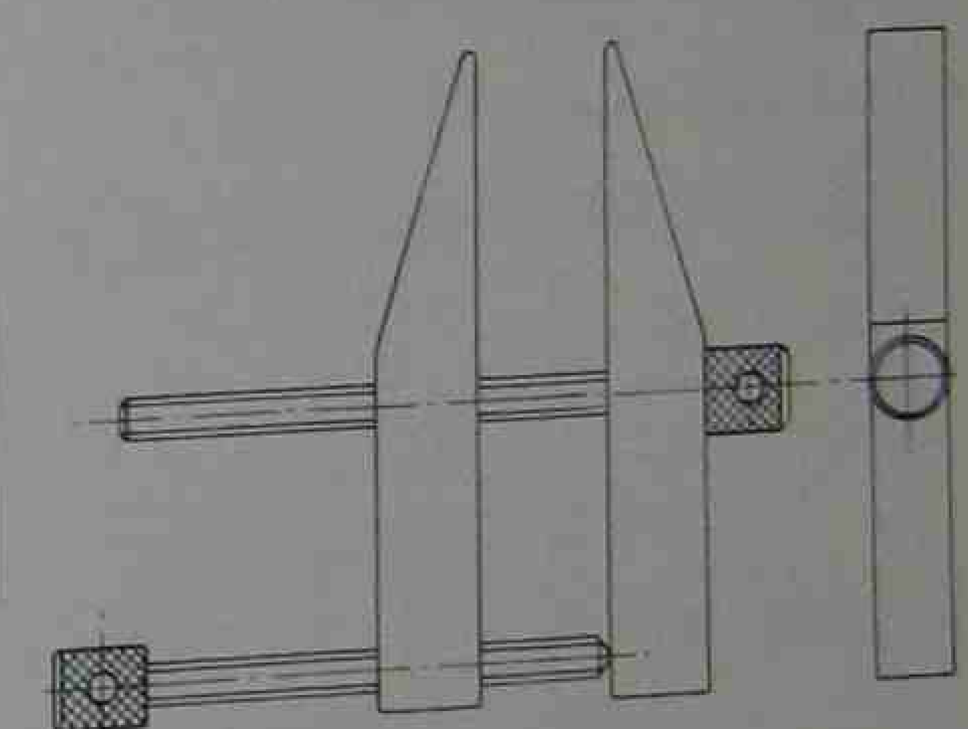


Figure 3.53 *Toolmakers' clamp*

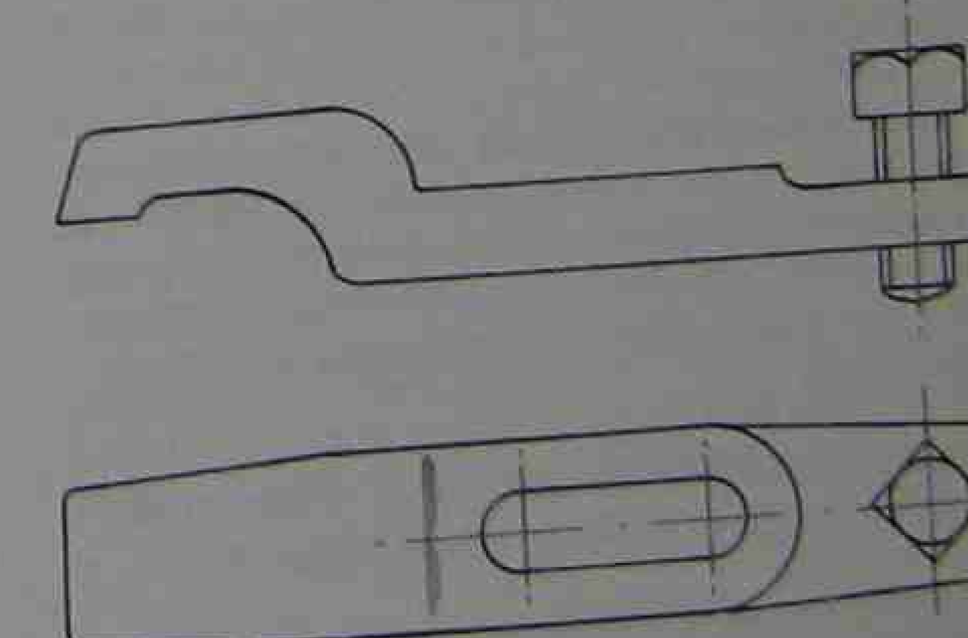


Figure 3.54 *Strap clamp*

VICE (VISE)

A vice is a bench or machine-mounted clamping device for holding work while manual or machine operations are being carried out. It comprises two jaws which are brought together by a hand-operated screw to clamp the work. The jaws are made of hardened steel and are serrated to improve their grip on the work. To avoid accidental marking of soft metals, 'grips' of lead, copper, aluminium or fibre may be fitted. The *bench vice* is used for general fitting, and the *machine vice*, which is often able to tilt and swivel, is mounted on the table of a machine, e.g. a drilling machine. *Pipe vices* are used for holding pipes while they are being cut or screwed.

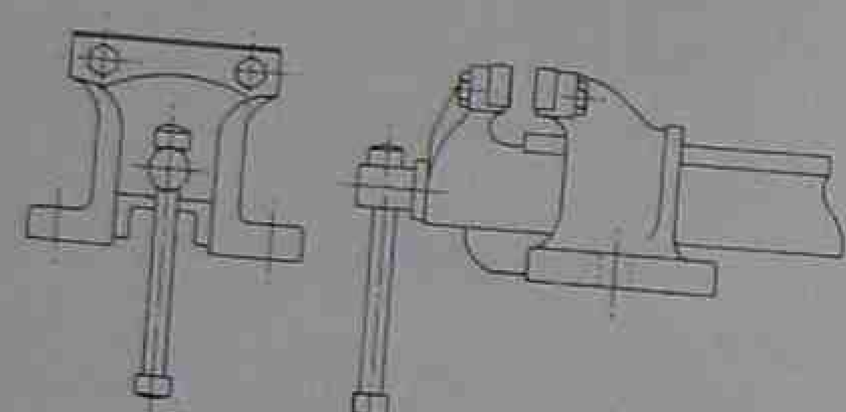


Figure 3.55 *Engineers' vice*

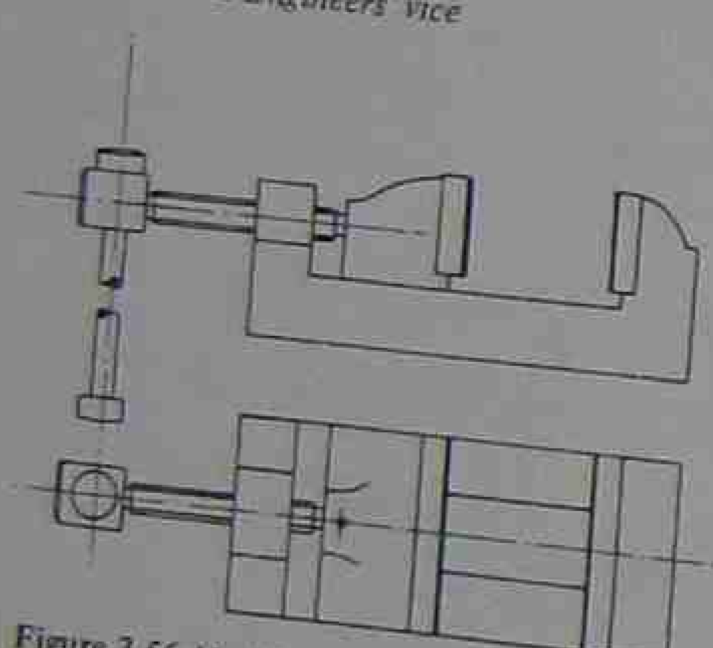


Figure 3.56 *Machine vice*

PLIERS

Pliers are hand tools used for gripping small components, bending, wire cutting, etc. Two handles with a common pivot have jaws with serrations (pipe grips), and usually have blades for side cutting and notches for wire cutting. For electrical uses the handles are insulated.

Long-nose pliers (needlenose pliers) These are useful for holding small work and bending loops in wire.

Round-nose pliers These are used for bending loops in wire and narrow strip and have no serrations.

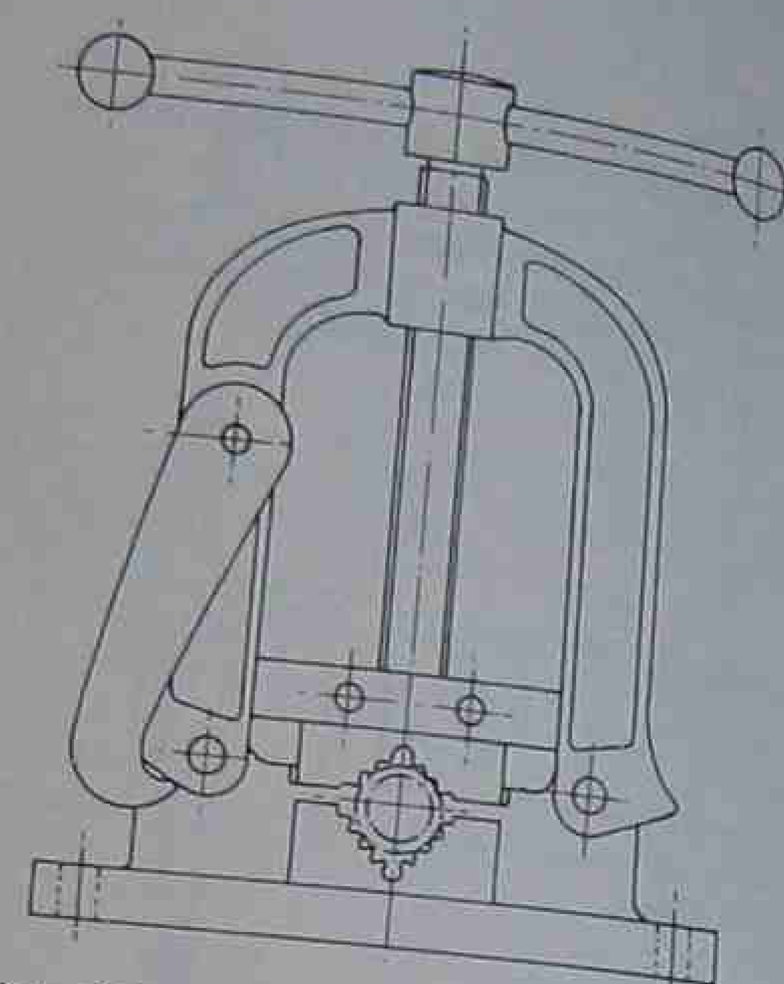


Figure 3.57 *Pipe vice*

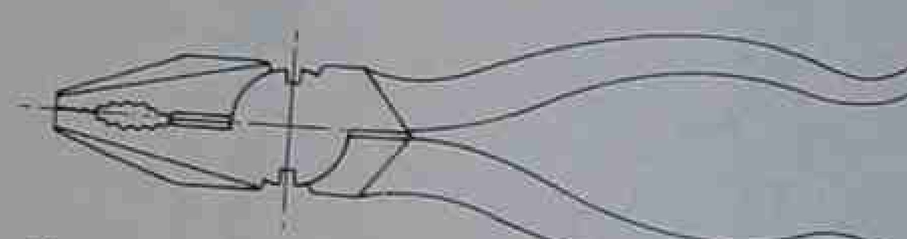


Figure 3.58 *Pliers*

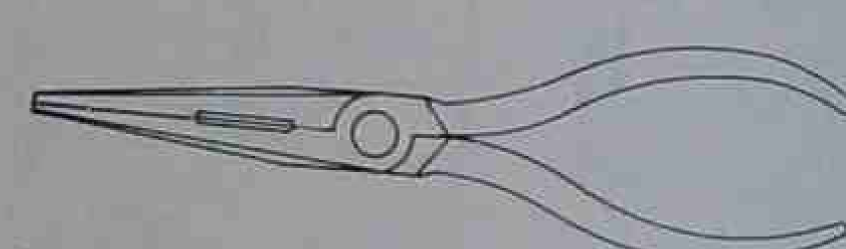


Figure 3.59 *Long-nose (needlenose) pliers*

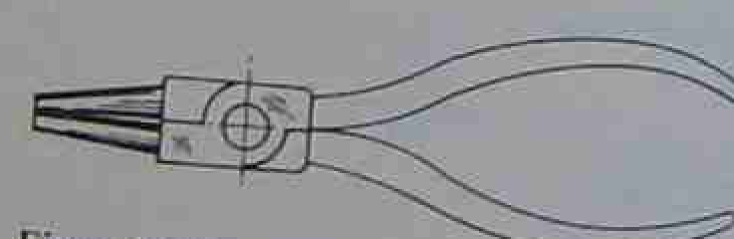


Figure 3.60 *Round-nose pliers*

WIRE CUTTERS

Wire cutters have blades suitable for cutting copper and small gauge iron wire.

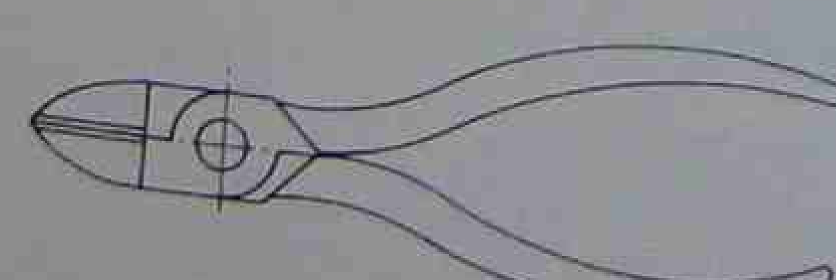


Figure 3.61 *Wire cutters*

SHEARS (SNIPS, TINSNIPS)

Hand tools of the scissor type for cutting thin sheet metal, jointing, etc.

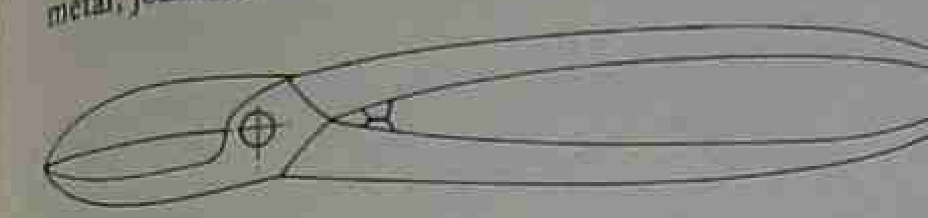


Figure 3.62 *Shears (snips)*

SCREWDRIVERS

These are tools used for hand-driving screws. They consist of a usually round bar of carbon tool steel heat-treated to give hardness and toughness with a handle at one end and the other end formed to suit the recess in the screw. This may be a slot, hexagonal hole or cross recess.

Screwdrivers are made in sizes ranging from extremely small jewellers' and instrument-makers' types to large ones over 40cm in length.

Ratchet screwdrivers can be adjusted to be rigid in both directions of rotation or to slip in either direction.

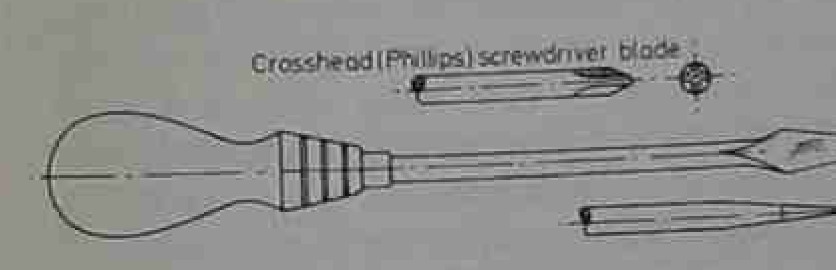


Figure 3.63 *Wooden-handled screwdriver*

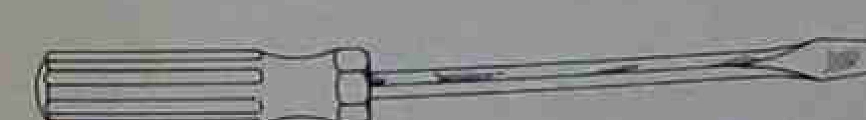


Figure 3.64 *Heavy screwdriver with plastic handle*

Figure 3.65 *Instrument makers' screwdriver*

Figure 3.66 *Ratchet screwdriver*

SPANNERS (WRENCHES)

A spanner, or *wrench*, is a device employing leverage and it is used for tightening or releasing nuts, bolts, and other screwed fasteners. Most spanners are hand-operated but power-driven spanners are available.

Open-ended spanner One or both ends of this spanner has open parallel jaws which, in the double-ended type, are usually of different sizes. These jaws fit the flats on nuts and bolt heads. They are obtainable singly or in sets of different sizes.

Figure 3.67 *Open-ended (open-jaws) spanner (wrench)*

Ring spanner (box wrench) The ends of this spanner are in the form of rings with twelve internal corners to suit hexagon nuts and bolts. The hexagon size is different at each end and sets of up to twelve spanners are available. The spanners may be flat or have the ends offset.

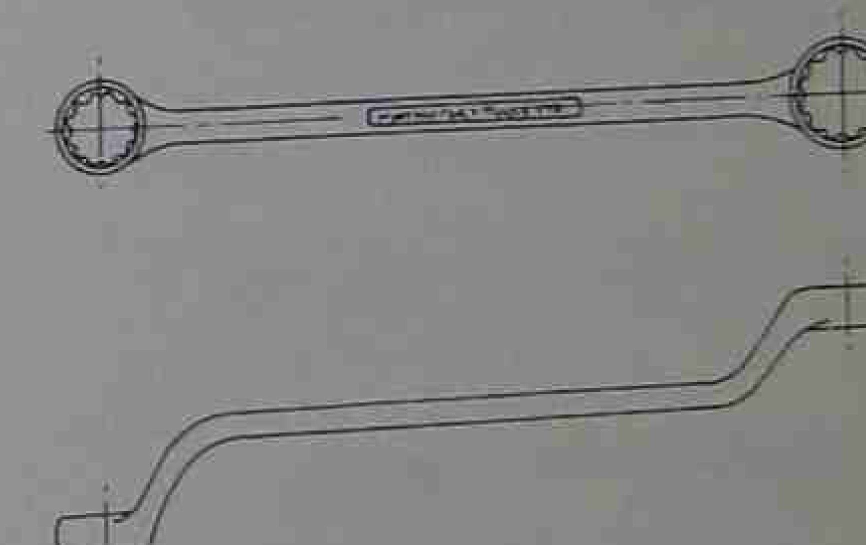


Figure 3.68 *Ring spanner (box wrench)*

Tubular box spanner Tubular spanners are made of tough alloy steel circular tube formed at the ends into different sizes either of hexagon or square. Holes are drilled through the tube to take a *tommy bar* which provides the required leverage.

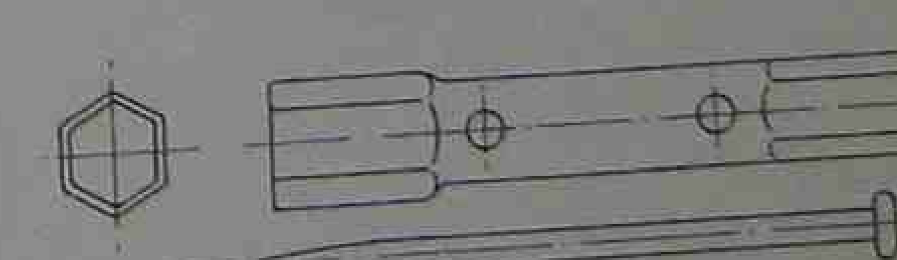


Figure 3.69 *Tubular box spanner (socket wrench) and tommy bar*

Socket spanners (socket wrenches) These are short circular tubes of alloy steel with twelve internal corners at one end to suit hexagonal nuts, etc. and a square socket at the other end to suit various attachments. Sets of sockets contain up to 24 different sizes.

The following attachments are available: bar handle, socket handle, brace, extensions, universal joint.

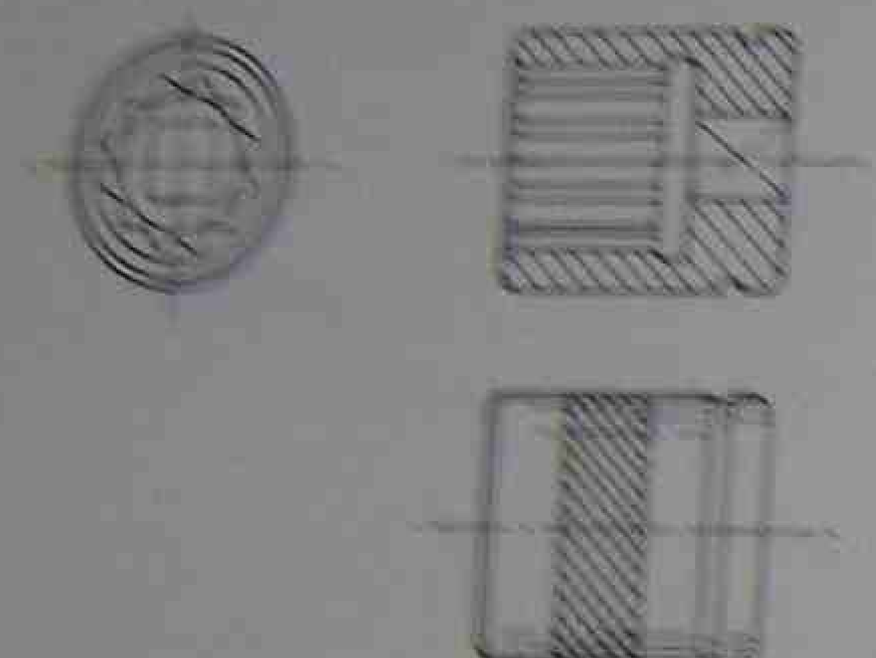


Figure 3.70 *Socket for socket spanner (wrench)*

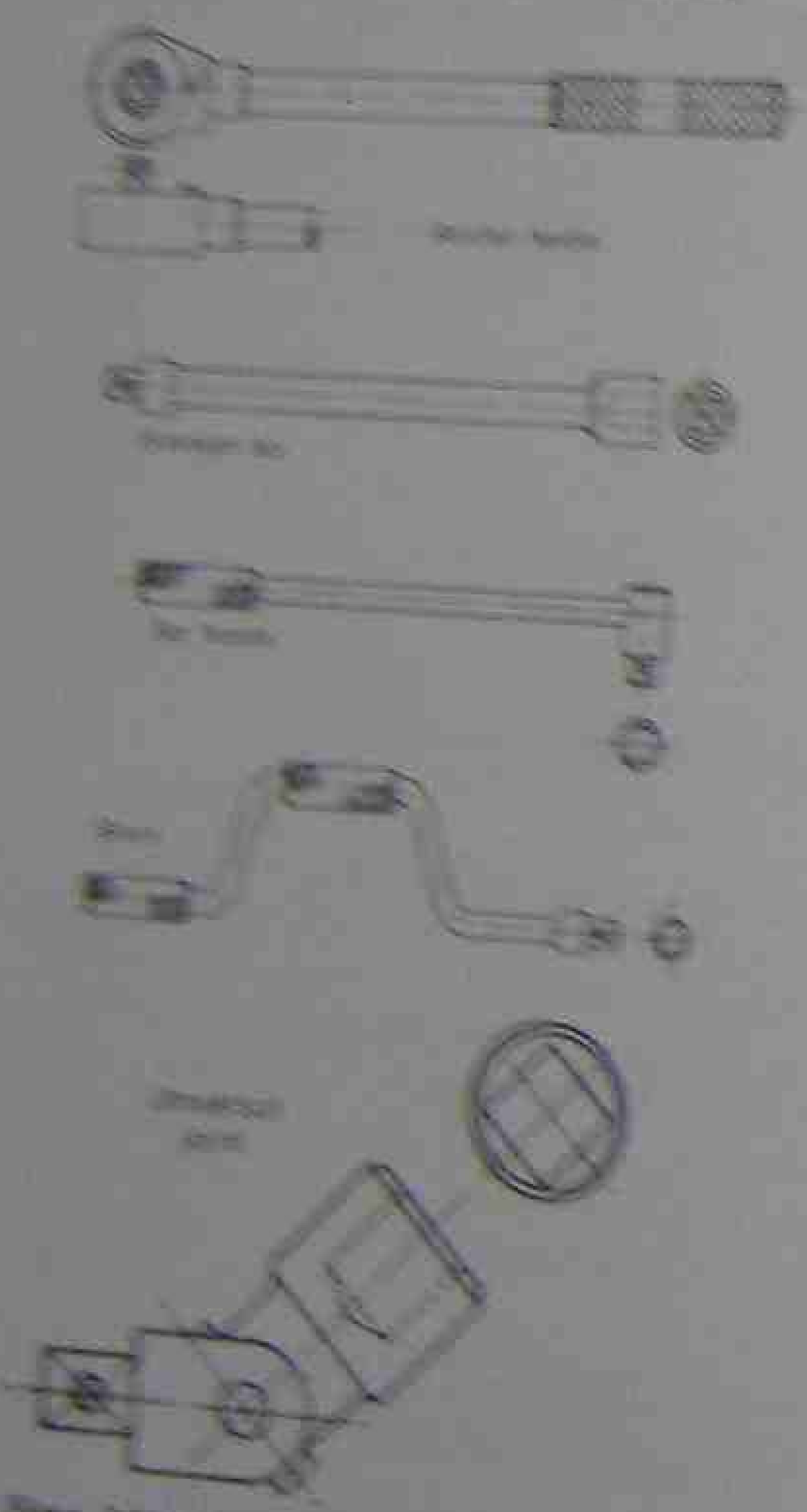


Figure 3.71 *Attachments for socket spanners (wrenches)*

Adjustable spanner (wrench) An open-jawed spanner with one jaw adjustable so that the gap in the jaws can be altered to suit different sizes of nuts and bolts.

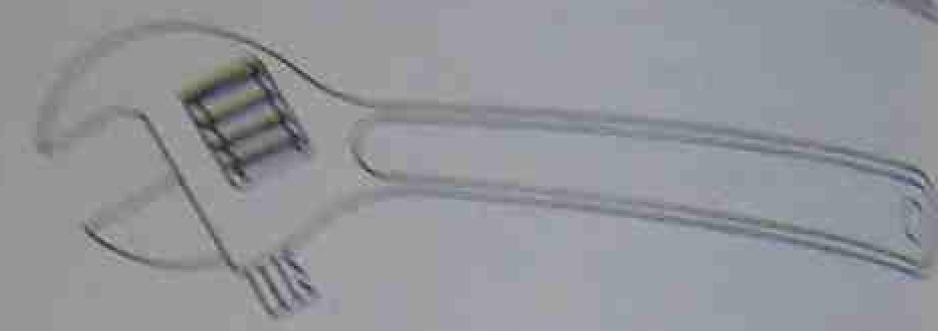


Figure 3.72 *Adjustable spanner (wrench)*

Face spanner (wrench) At the end of the handle a flat C-shaped section has a circular pin at each end of the C which engage with holes in a ring nut.

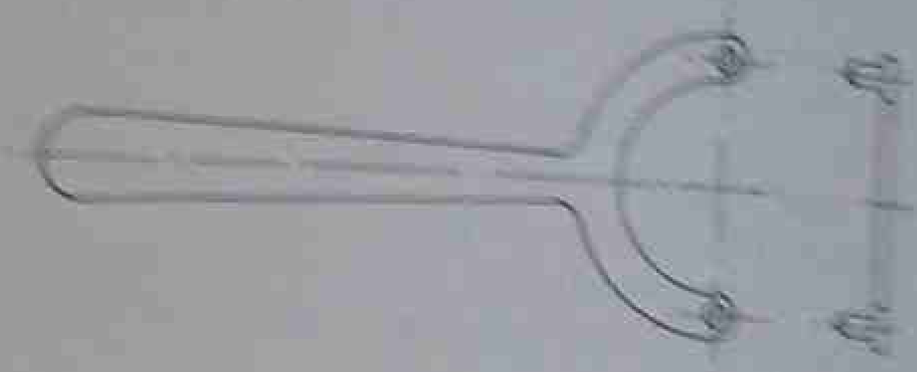


Figure 3.73 *Face spanner (wrench)*

C spanner (wrench) The end of this spanner is curved into the form of a C with a square peg at the end which engages with a notch in the circumference of a ring nut.

Figure 3.74 *C spanner (wrench)*

Hexagon socket wrench (Allen key) A piece of hexagon bar with the end bent at right angles, used in conjunction with hexagon socket screws.

Figure 3.75 *Hexagon socket wrench (Allen key)*

Torque spanner A torque spanner is used to tighten nuts to a predetermined torque to avoid overstraining the threads. There are two types, one has a dial which indicates the torque and another which can be set to slip at the required torque.

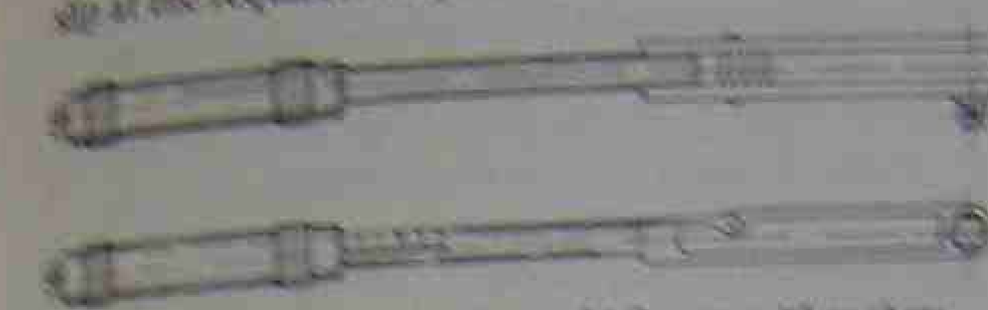


Figure 3.76 *Torque spanner (wrench) for use with sockets*

Pipe wrench (Stillson wrench) A type of adjustable spanner designed to grip pipes and other circular objects.

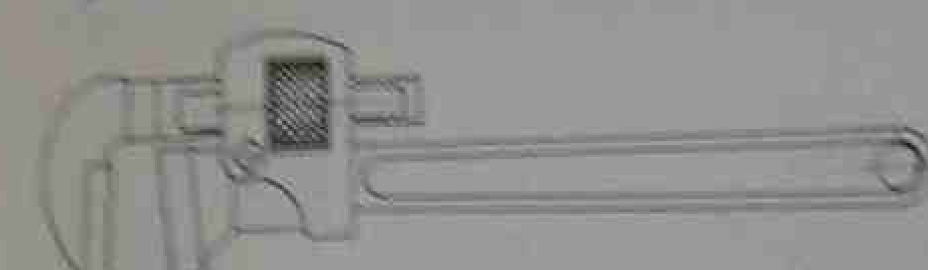


Figure 3.77 *Pipe wrench (Stillson wrench)*

3.2 MACHINE TOOLS

The term 'machine tool' applies to any power-driven, non-portable machine designed to shape usually metal parts. It includes milling machines, shapers, lathes, drilling machines, borers, gear cutters, etc.

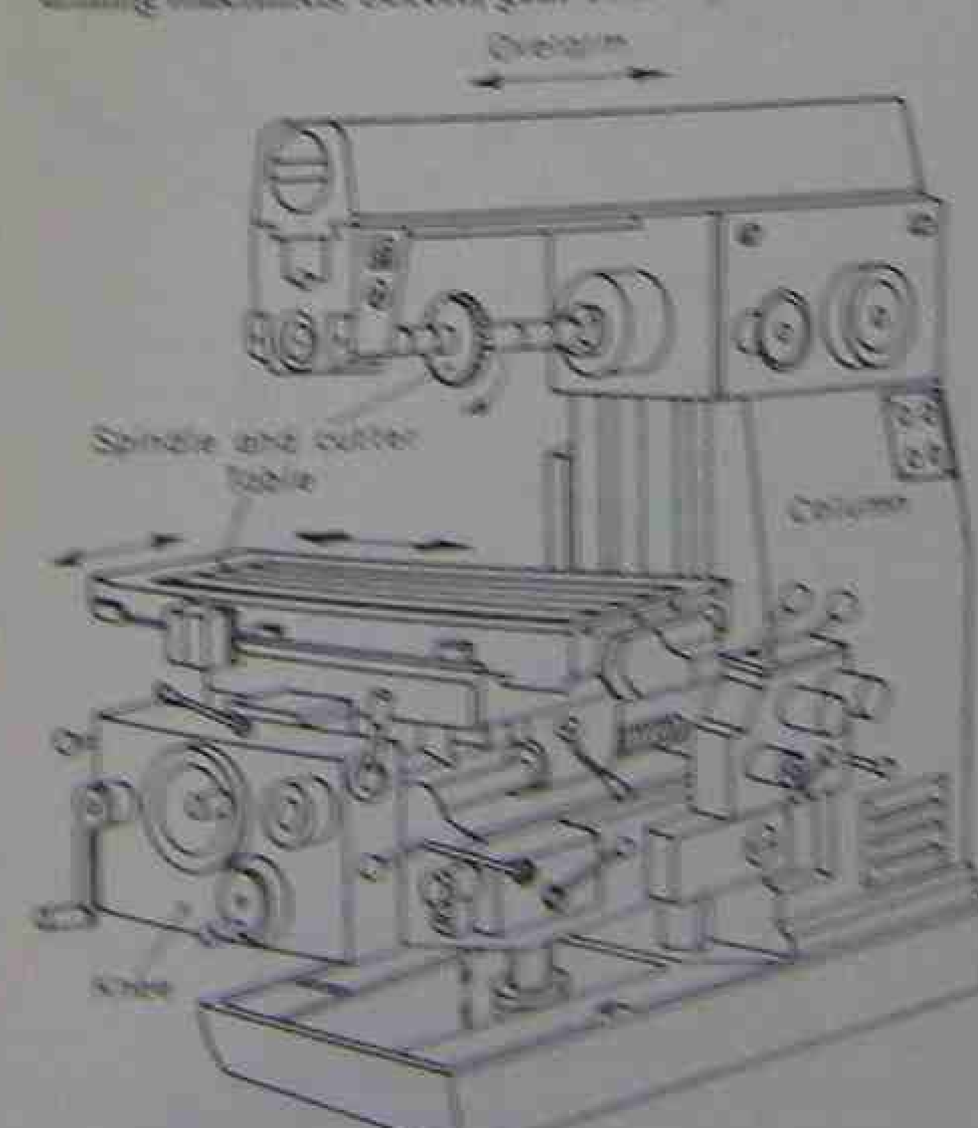


Figure 3.78 *Horizontal milling machine*

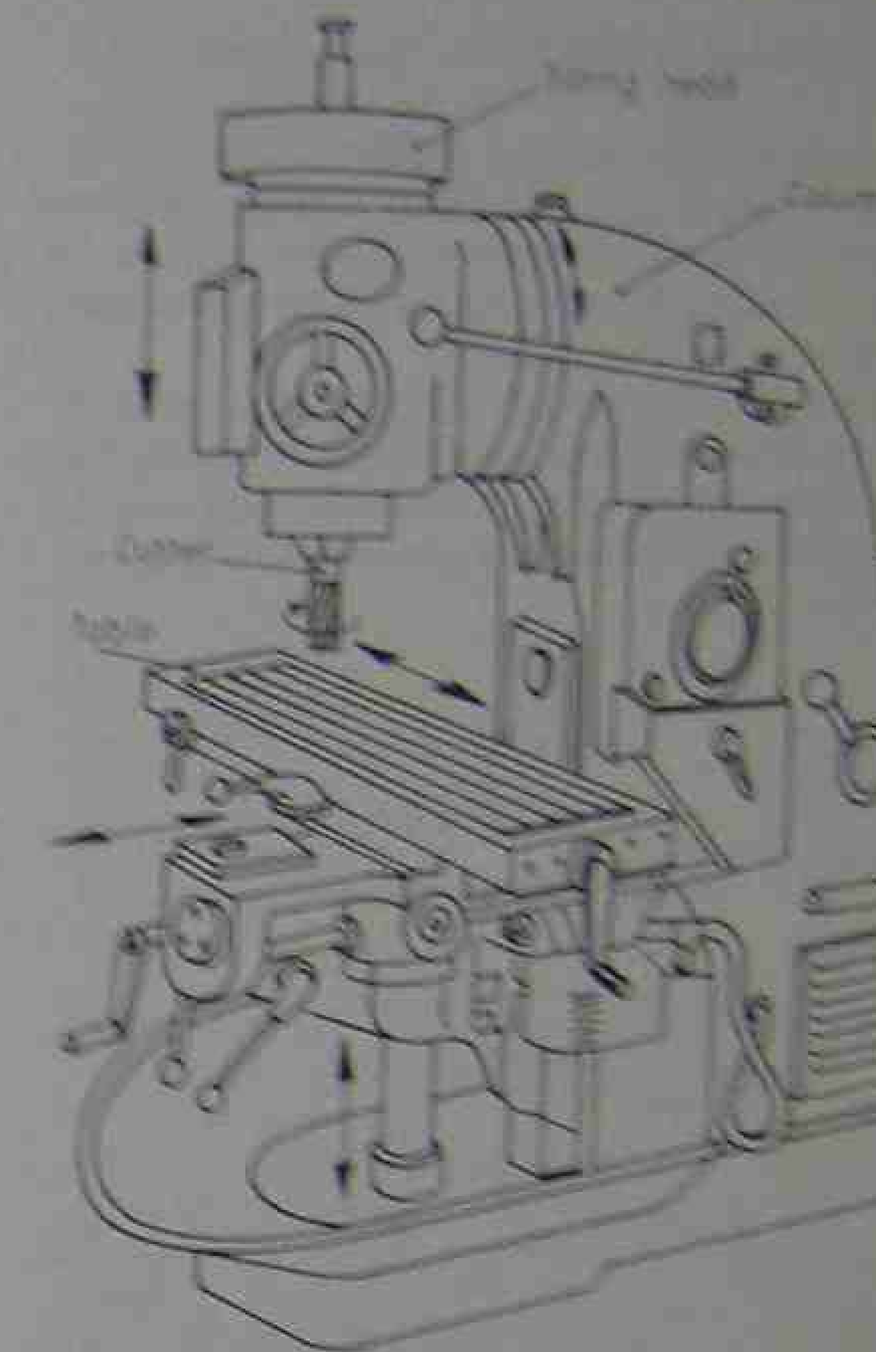


Figure 3.79 *Vertical milling machine*

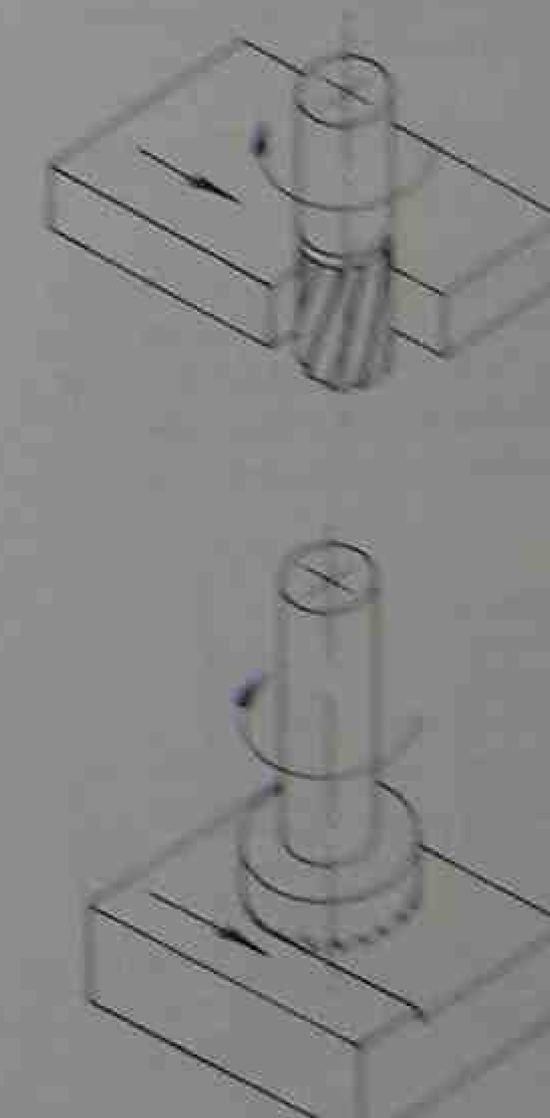


Figure 3.80 *Vertical milling operations*

MILLING MACHINE

A milling machine produces mainly flat surfaces by means of a rotating cutter. The work is mounted on a table which can be driven in three directions, and the milling cutter is driven at various speeds by an electric motor mounted on a column.

The two most common types of milling machine are the horizontal machine which has a spindle with a horizontal axis, and the vertical spindle machine which may have a tilting head to incline the spindle axis to the vertical.

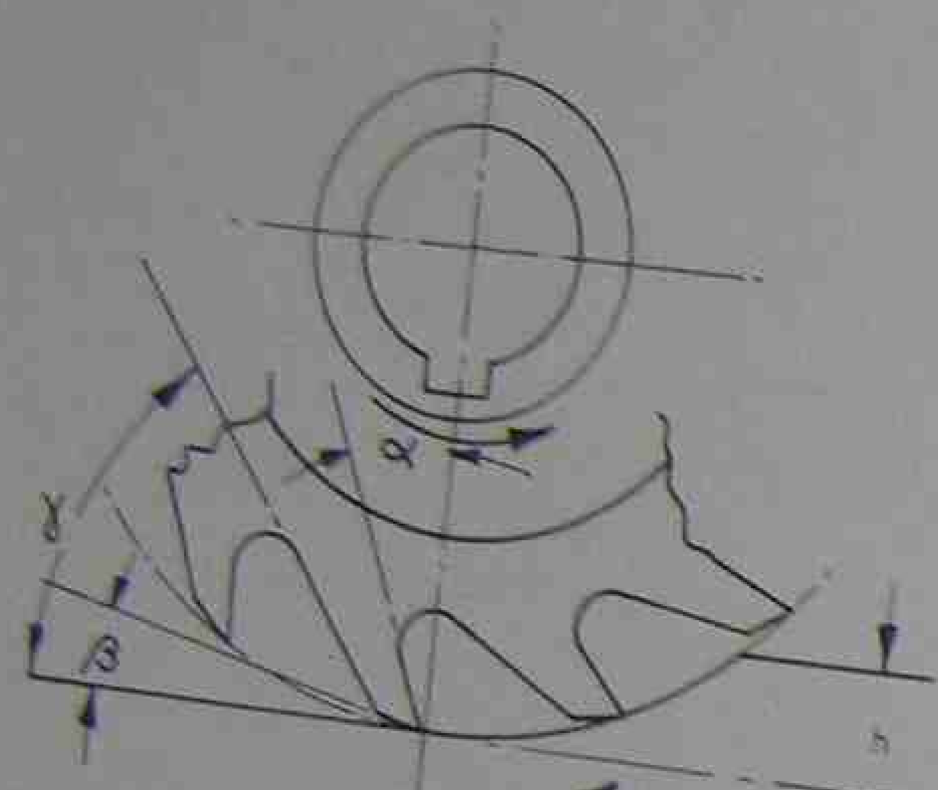


Figure 3.81 *Milling cutter nomenclature: α-rake; β-primary clearance; γ-secondary clearance; h-depth of cut*

MILLING CUTTERS

Milling cutters have cutting teeth around the circumference and/or on the end of a disk or cylinder. The teeth may be parallel to the axis of rotation, or helical, and may be contoured. Several cutters may be 'ganged' together to produce complicated profiles.

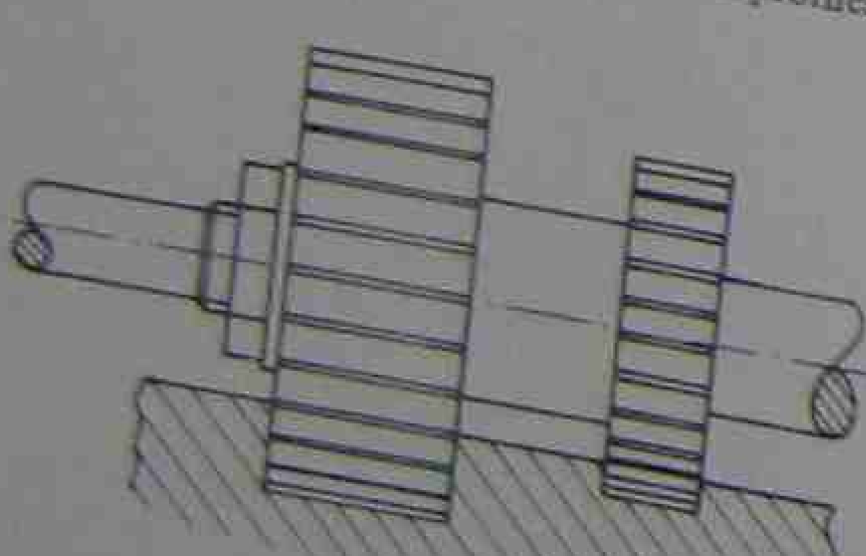


Figure 3.82 *Ganged milling cutters*

Slot milling cutter A disk with teeth on the rim which cut on the periphery only. It is used for machining slots or as a slitting saw, and is mounted on a shaft or *arbor* in a horizontal milling machine.

Figure 3.83 shows the conventional 'up-cutting' action, but if the feed is reversed a 'down-cutting' action is obtained, as shown in Figure 3.83a. This tends to drive the work in the feed direction, and a backlash eliminator in the table drive is required.

Side and face milling cutter This has cutting edges on the sides of the disk so that it can be used for side cutting as well as edge cutting.

Contour milling cutter These have special profiles such as concave or convex circular arcs, and they are used for producing internal and external radii, etc.

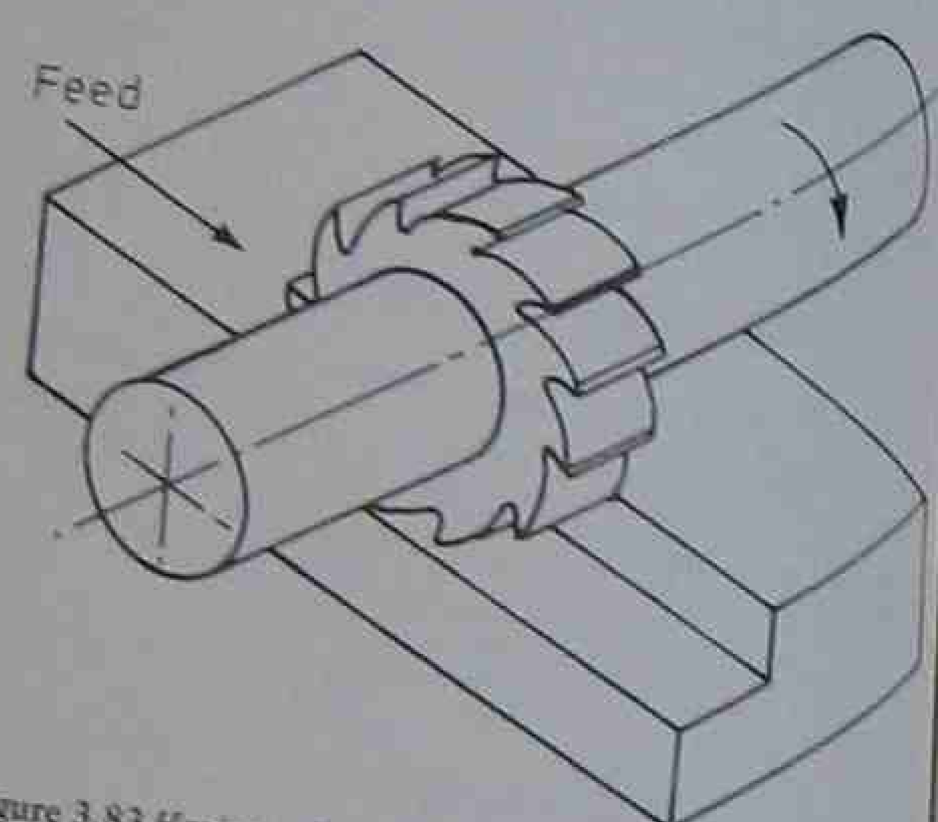


Figure 3.83 *Horizontal milling – up cut*

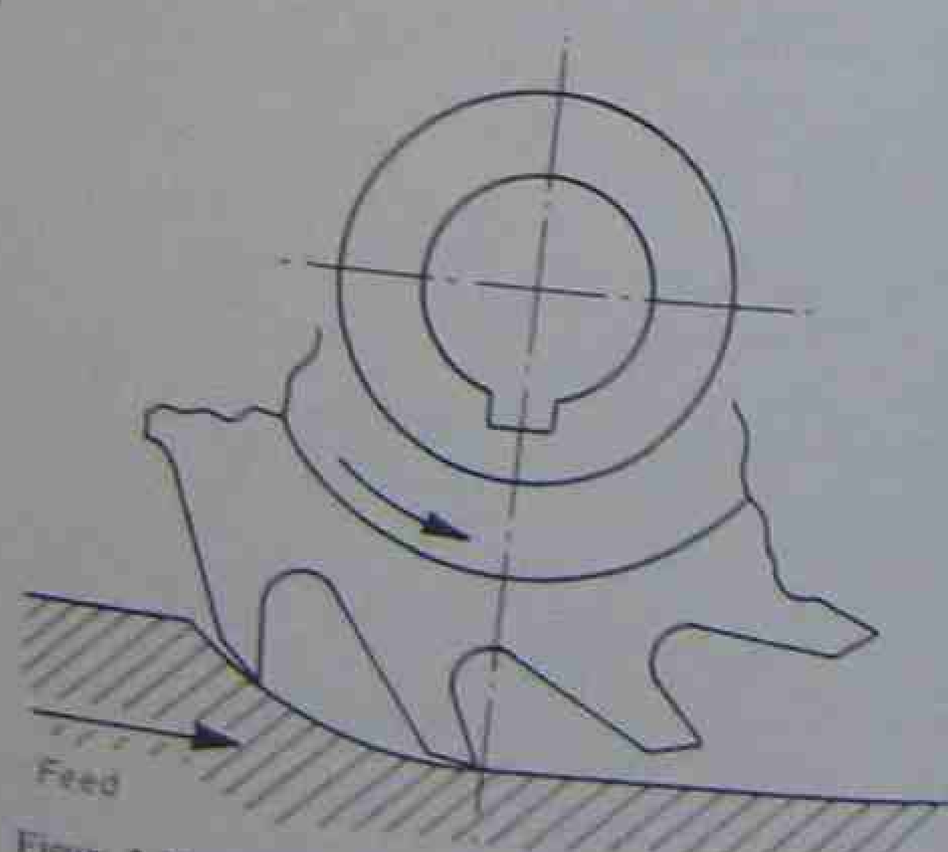


Figure 3.83a *Horizontal milling – down cut*

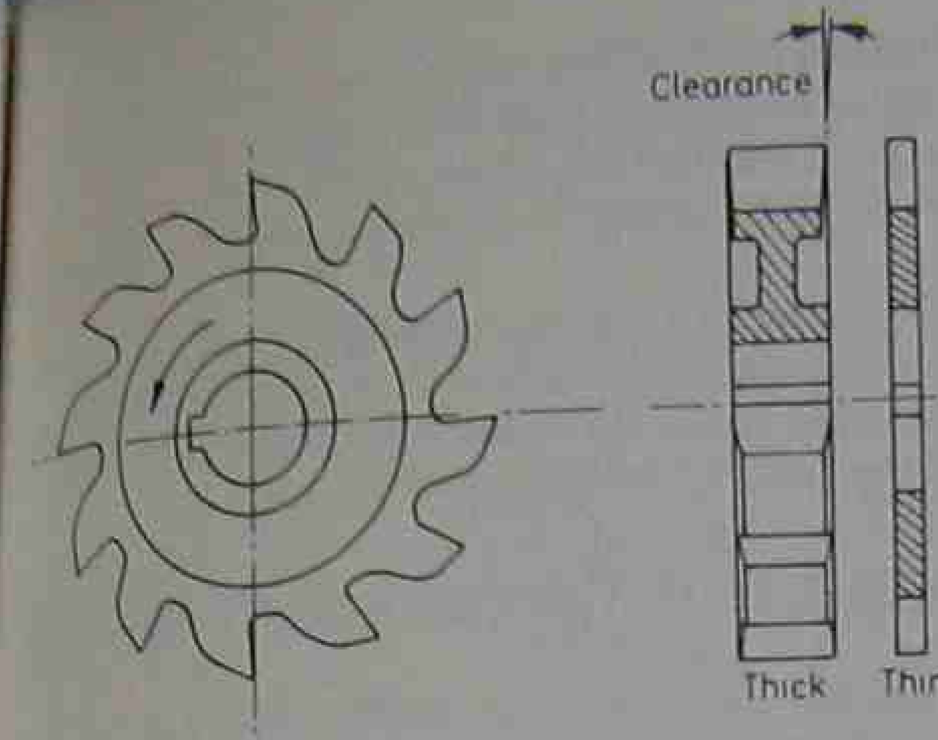


Figure 3.84 *Slot milling cutter*

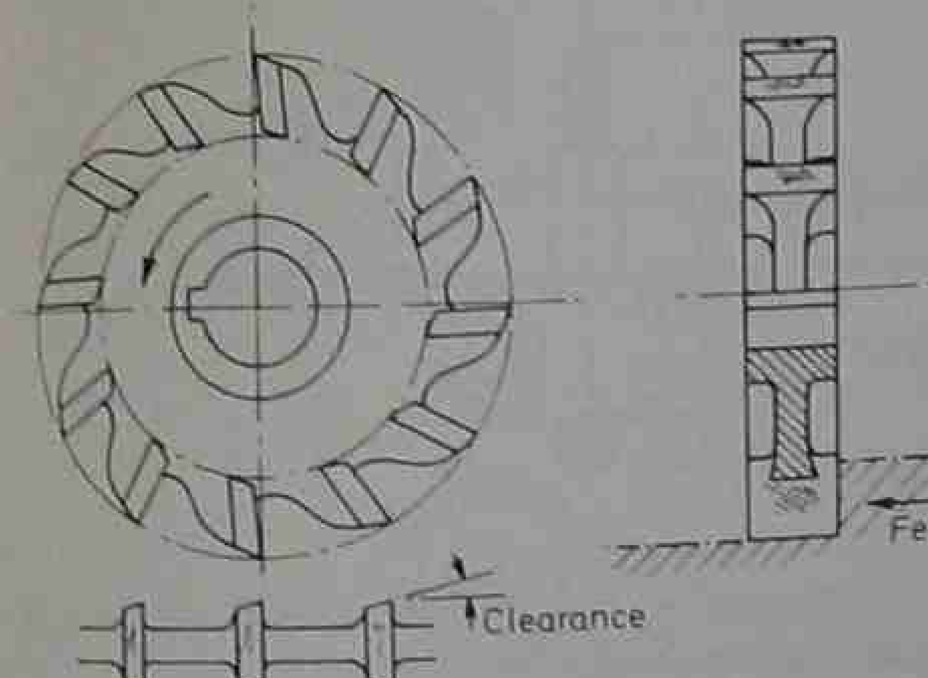


Figure 3.85 *Side and face milling cutter*

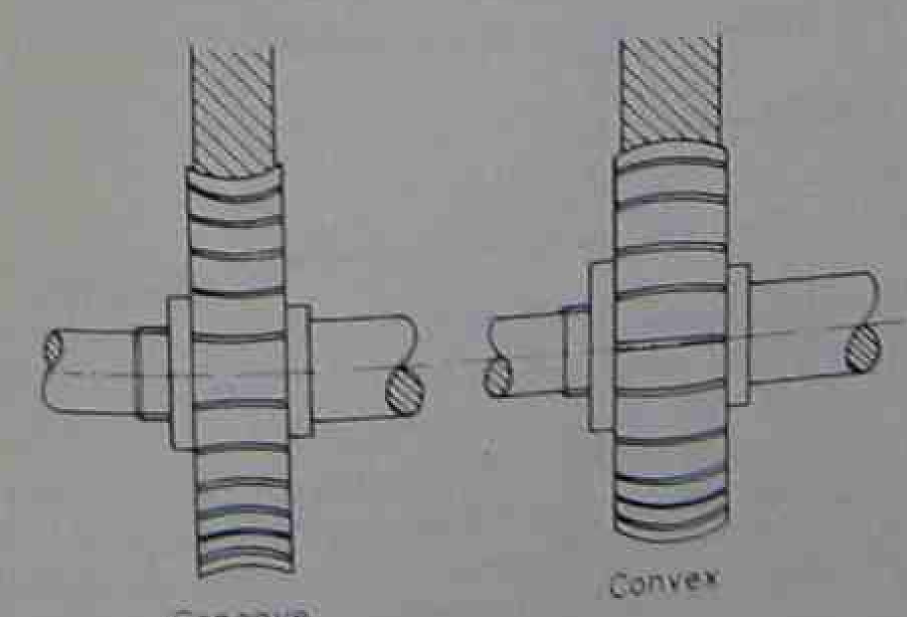


Figure 3.86 *Contour milling cutter*

Slab or rolling milling cutter A large cylindrical cutter used for machining horizontal surfaces. Usually it is mounted on a horizontal arbor, and to prevent vibration helical teeth are preferred.

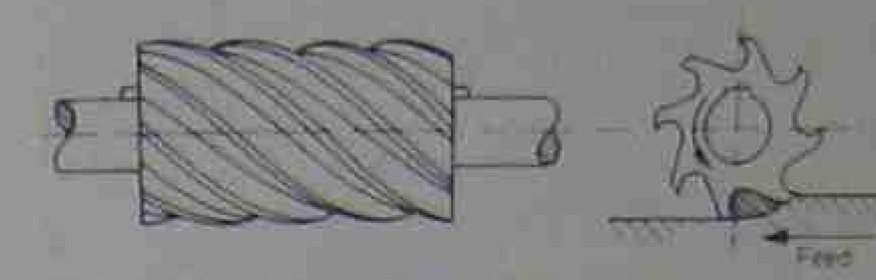


Figure 3.87 *Slab or rolling milling cutter*

End milling cutter Small end mills are made from solid bar with straight shanks to suit a collet. Larger cutters with separate shanks tapered to suit the machine spindle nose are known as *shell end mills*.

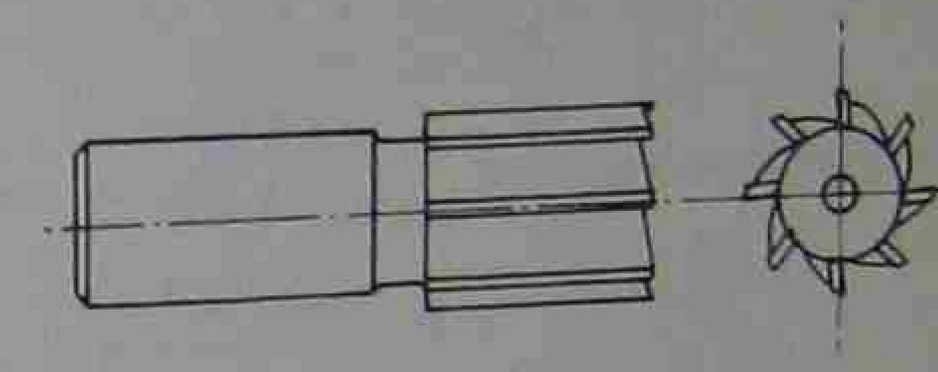


Figure 3.88 *End milling cutter*

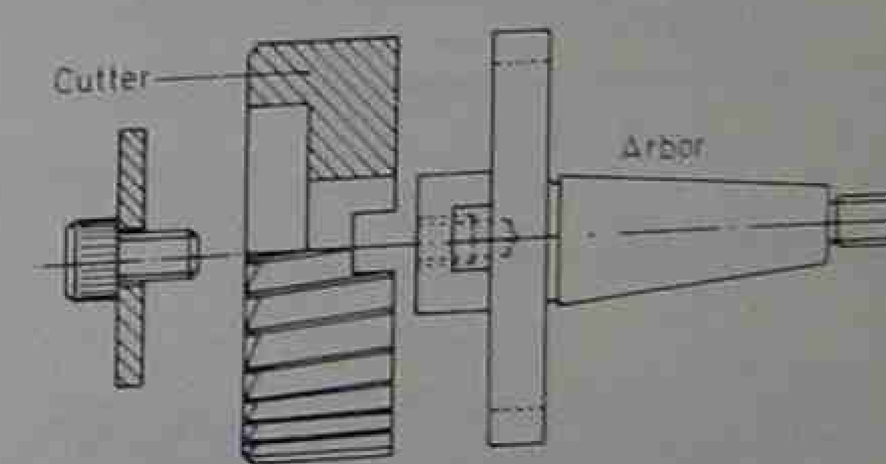


Figure 3.89 *Shell endmilling cutter*

Angular tooth milling cutters The teeth are on a conical surface so that angled surfaces may be machined. Typical angles are 45° and 60° to the horizontal. One type is known as a *dovetail cutter*.

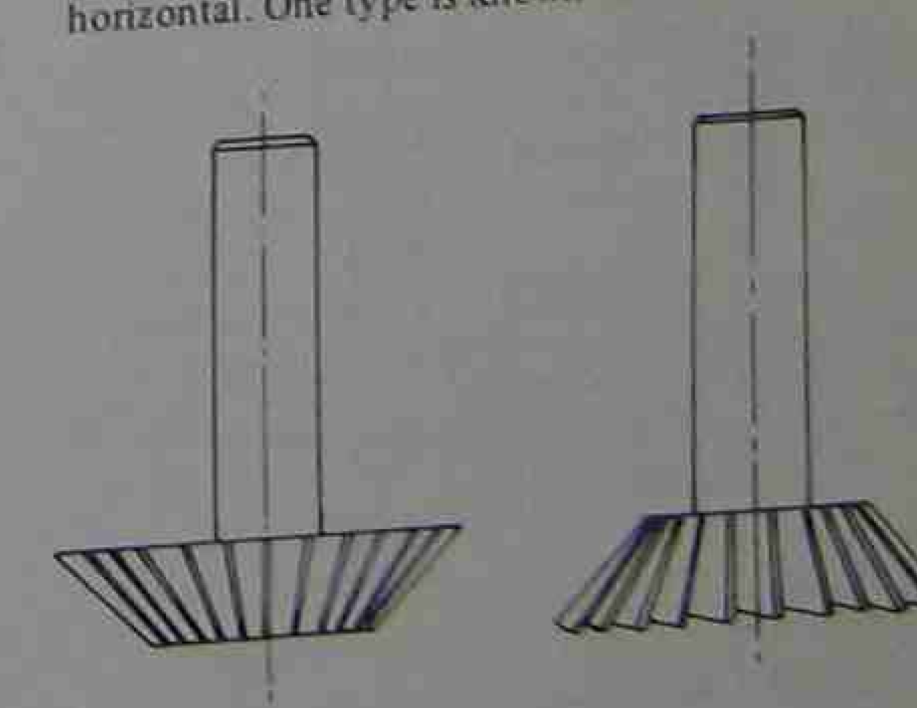


Figure 3.90 *Angular tooth milling cutter*

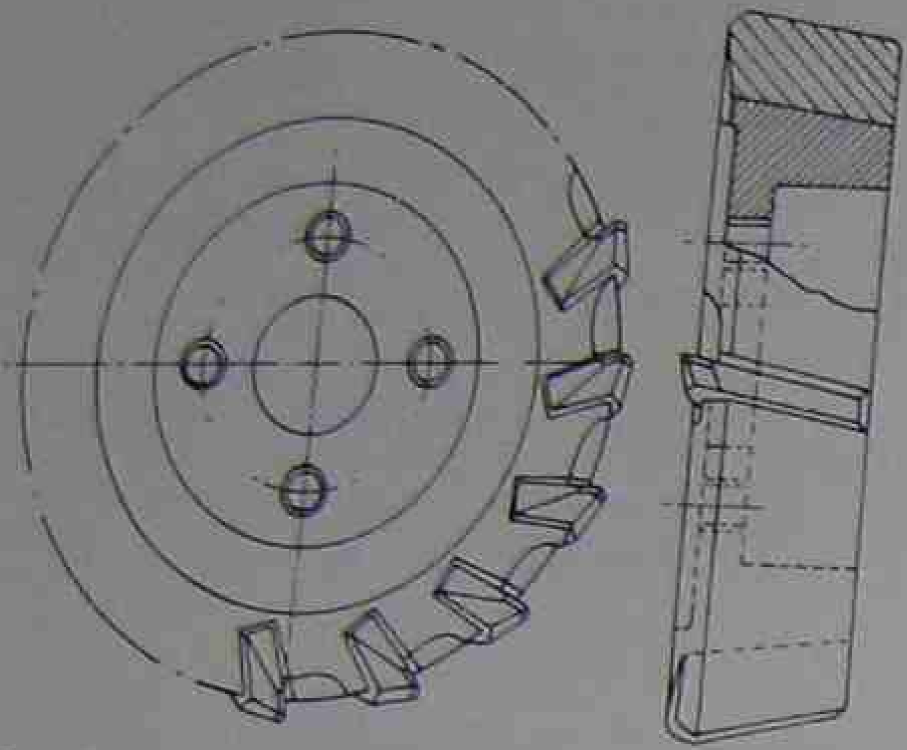


Figure 3.91 *Inserted tooth cutter*

Inserted tooth milling cutter Large, end or peripheral milling cutters are made which have separate teeth inserted into the disk which can be replaced when worn. Alternatively, the teeth can be re-sharpened.

Fly cutter To machine large surfaces a single-point cutter mounted on a disk or bar may be used. The bar-mounted cutter can be set at any radius and several cutters may be used to reduce the load.

DRILLING MACHINES (DRILL PRESSES)

These are machines which provide rotation at various speeds for drills to produce holes in a wide variety of materials. They vary in size from small bench-mounted machines to very large radial-arm floor-mounted machines.

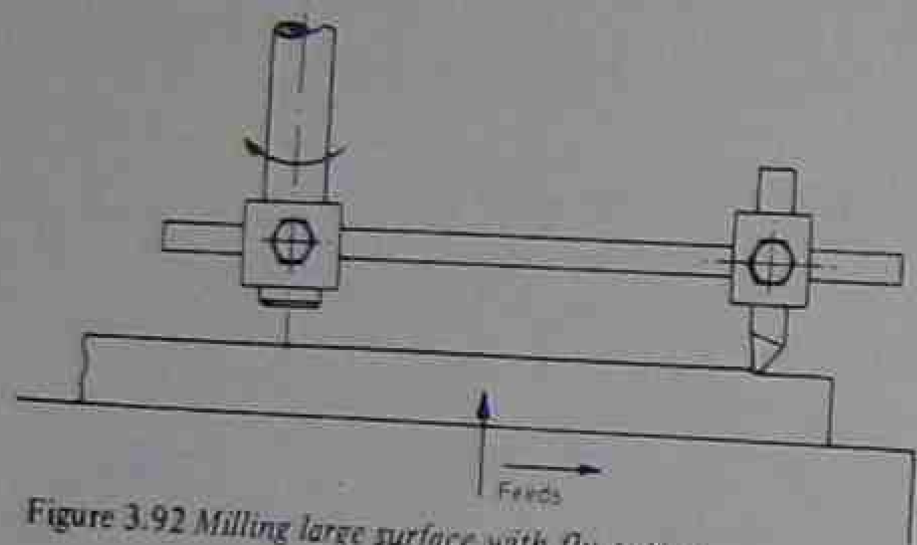


Figure 3.92 *Milling large surface with fly cutter*

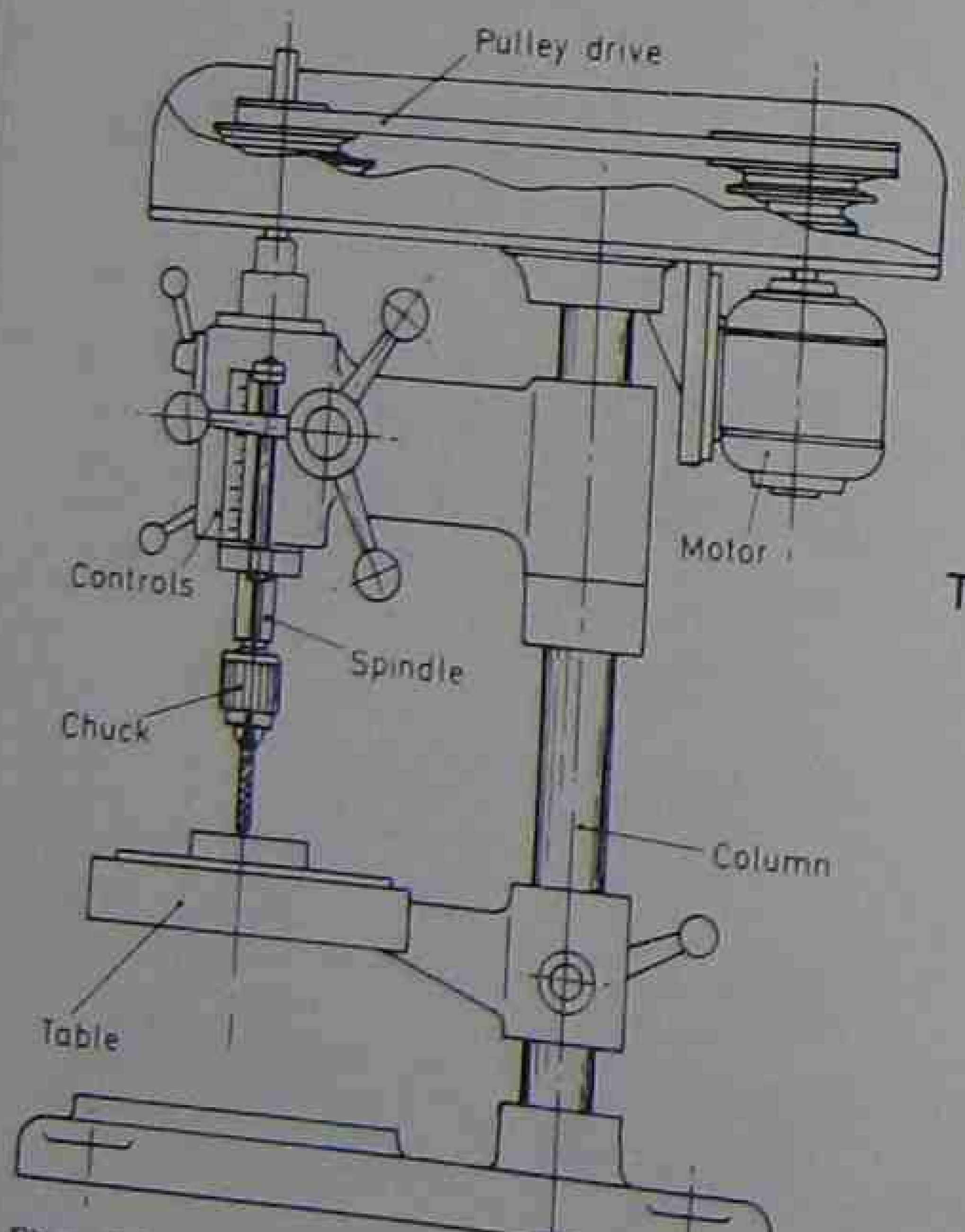


Figure 3.93 *Bench drilling machine*

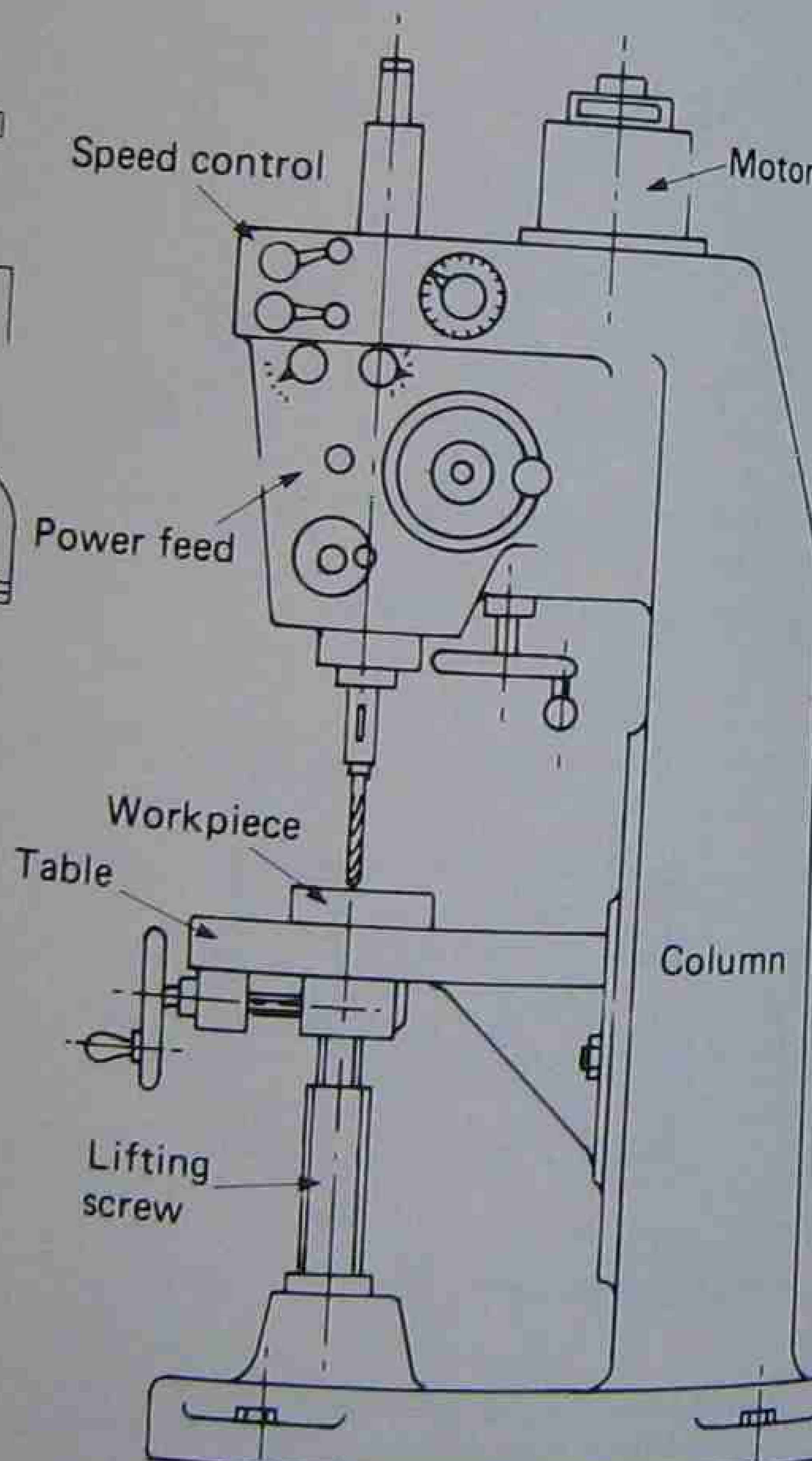


Figure 3.94 *Vertical box column drilling machine*

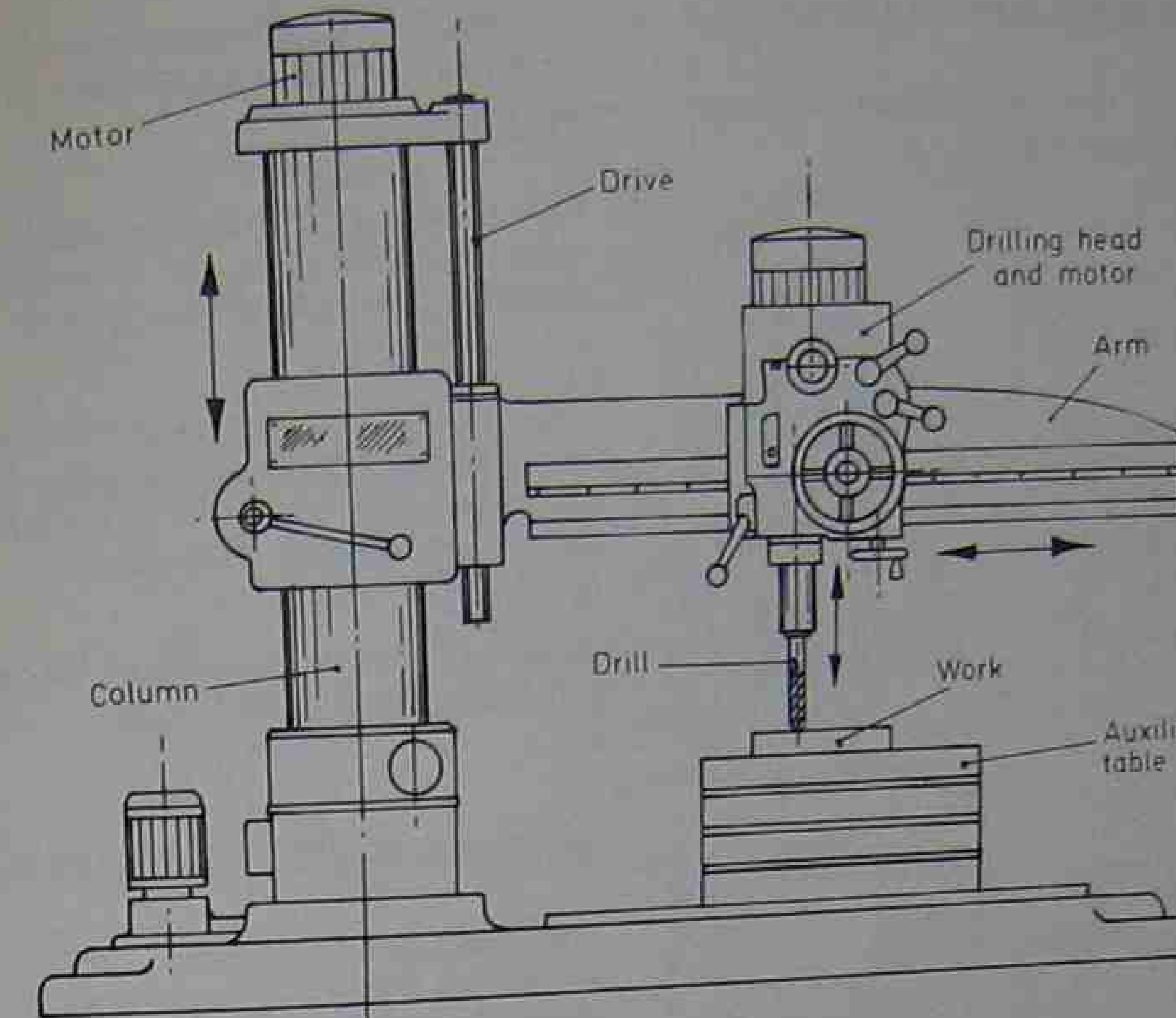


Figure 3.95 *Radial-arm drilling machine*

Radial-arm drilling machine The drilling head with the motor, speed change gearbox and drill spindle is mounted on a horizontal slide on the arm. The arm can be raised or lowered by an auxiliary motor and swung horizontally on a vertical post. Work is attached to the bedplate or to an auxiliary table. This machine is suitable for drilling heavy casting, forgings and weldments.

Bench drilling machine In this machine the vertical spindle is driven by an electric motor through a set of stepped V pulleys to give several speeds. The drill is either held in a chuck or has a taper shank which fits directly into the spindle. There is a table which can be adjusted vertically or swung horizontally on a post, and an adjustable stop controls the depth of hole. The work is clamped to the table or held in a machine vice bolted to the table.

Bench drilling machines for light work with drill sizes of up to about 10mm diameter and speeds of up to 9 000 rev/min are often referred to as *sensitive drilling machines*. Heavier duty machines take drills of up to 25mm diameter.

Vertical box column drilling machines This is a floor-mounted machine with a heavy cast column carrying a head with a vertical spindle. A multi-speed gearbox fitted to the drive gives a wide range of speeds and there is a power feed. Work is mounted on a table which can be raised or lowered by a screw.

DRILL CHUCK

Straight shank drills are held in a chuck with jaws operated by a key, but taper shank drills fit directly into the machine spindles via taper adaptors. Drills of up to 20 mm (¾ in.) may be accommodated.

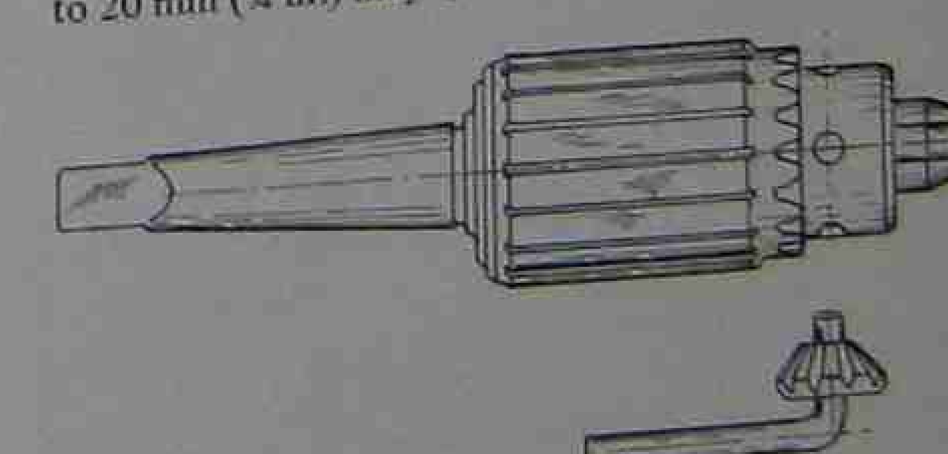


Figure 3.96 *Chuck and key*

DRILLING MACHINE OPERATIONS

In addition to drilling through and blind holes other operations are possible as follows:

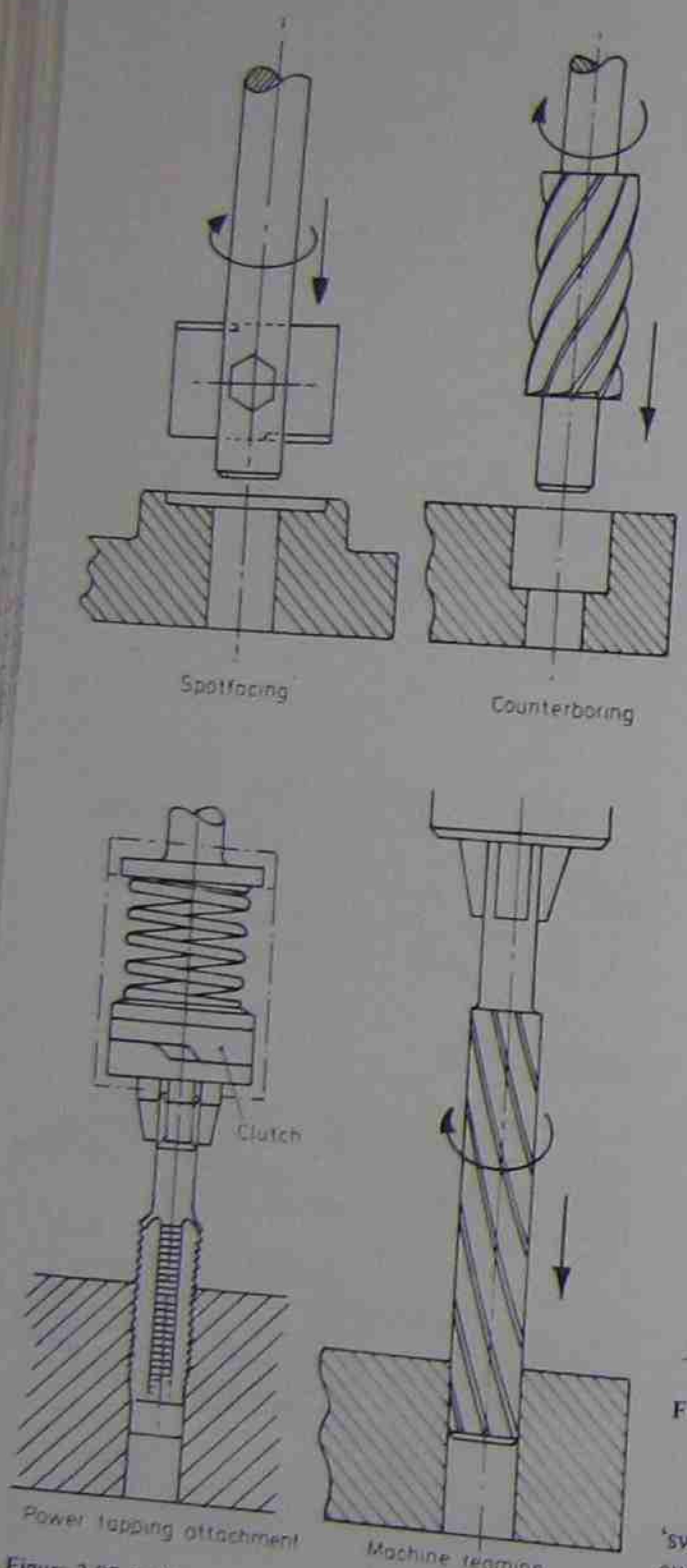


Figure 3.97 *Drilling machine operations*

Spotfacing A blade-type cutter attached to a bar is used to cut a face at the entrance to a hole in a rough casting etc.

Counterboring Enlarges the drilled hole to take the head of a socket screw, etc. A special tool, similar to an end mill but with a pilot, is used.

Tapping May be done on the machine using a tap holder with a spring-loaded clutch which slips when the tap reaches the bottom of the hole until the direction of rotation is reversed.

Reaming May be done using a low speed and plenty of cutting fluid.

SHAPING MACHINE (SHAPER)

A shaping machine, or shaper, is used to produce flat surfaces by means of a single-point reciprocating tool. The surfaces may be vertical, horizontal or angled.

The machine consists of a reciprocating arm driven by a variable-speed, quick-return mechanism, with a tool head which has a vertical feed and a table with feed in two horizontal directions. The 'head' is mounted in a vertical slide and carries the 'clapper box' which in its turn carries the 'tool post'. The clapper box is designed to tilt on the return stroke to avoid rubbing of the tool.

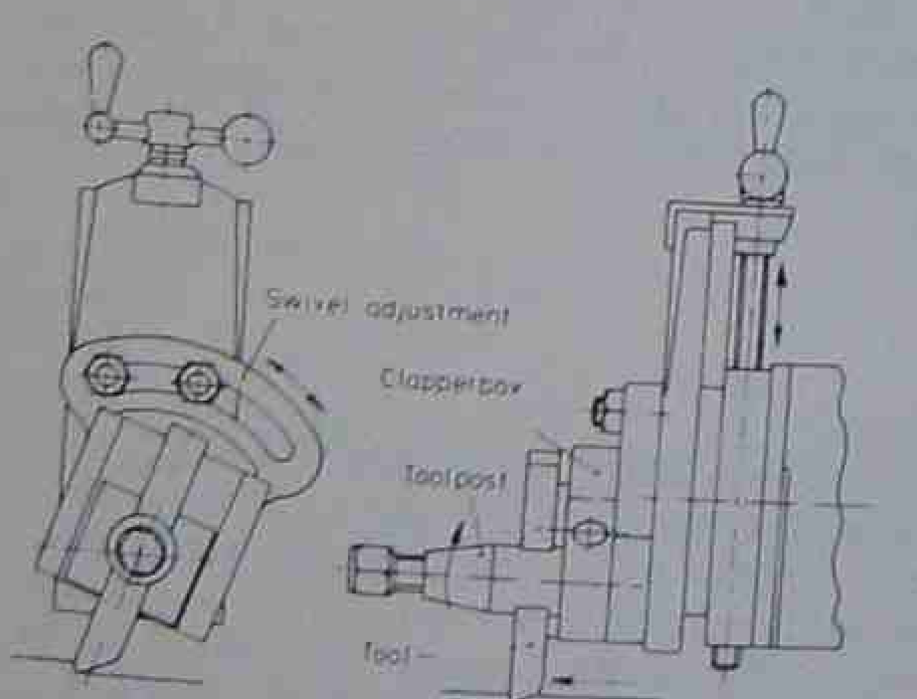


Figure 3.98 *Shaping machine tool head*

Two typical tools are the 'straight' tool and the 'swan-necked' tool. The swan-necked tool has the cutting edge behind the clapper box pivot. Bending the tool tends to lift the cutting edge and prevent 'digging-in'.

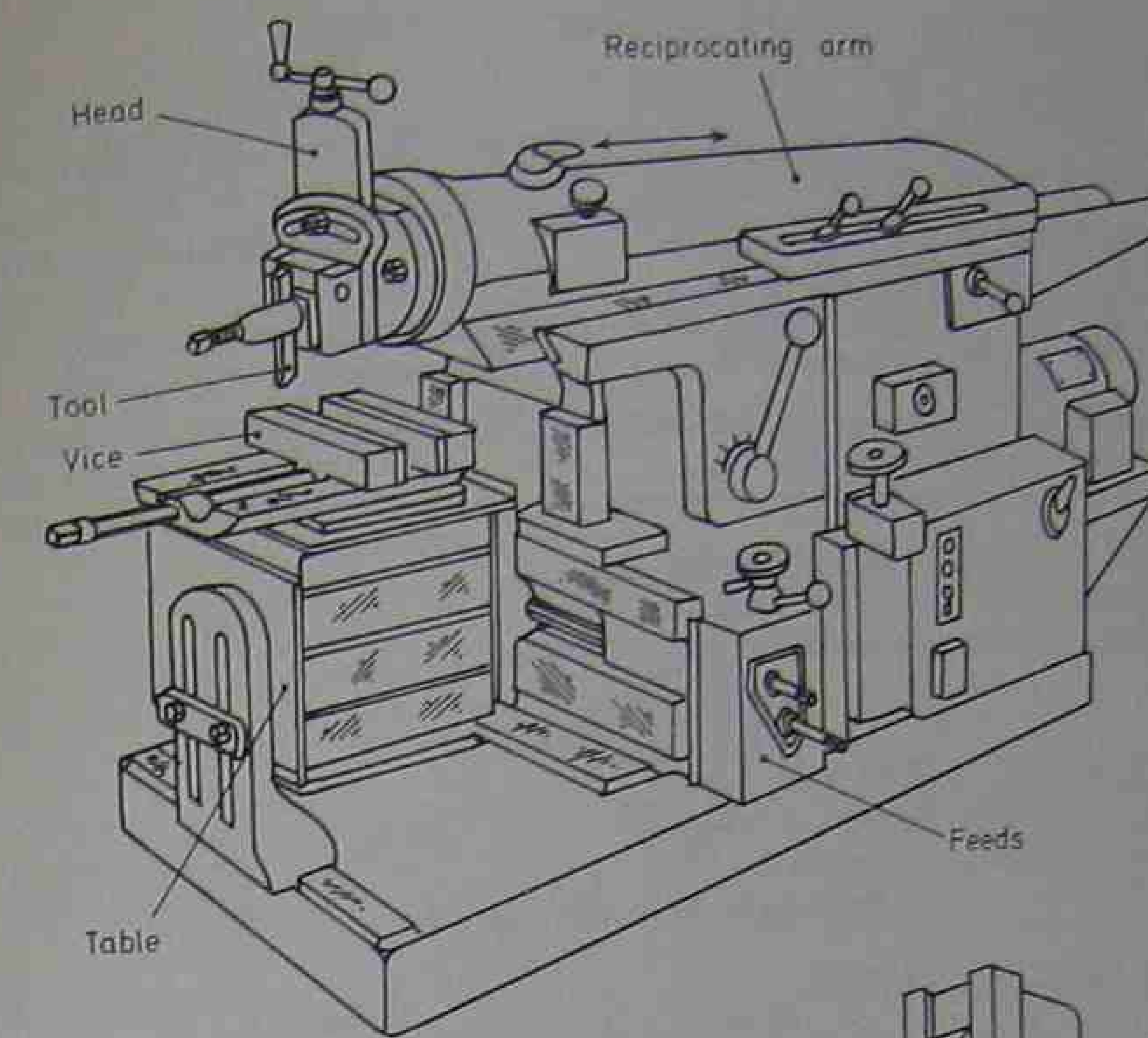


Figure 3.99 *Shaping machine (shaper)*

VERTICAL SHAPER (SLOTTER)

Vertical shapers, or slotters, are used for internal cutting and planing at angles. The table has a rotary feed to give curved machined surfaces.

These machines are used for cutting keyways in gears, pulleys, etc.

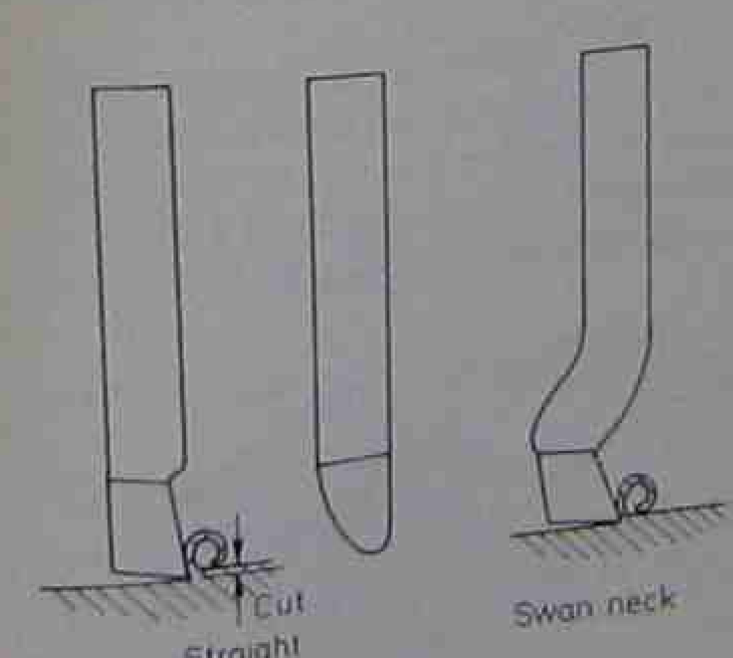


Figure 3.100 *Shaper tools*

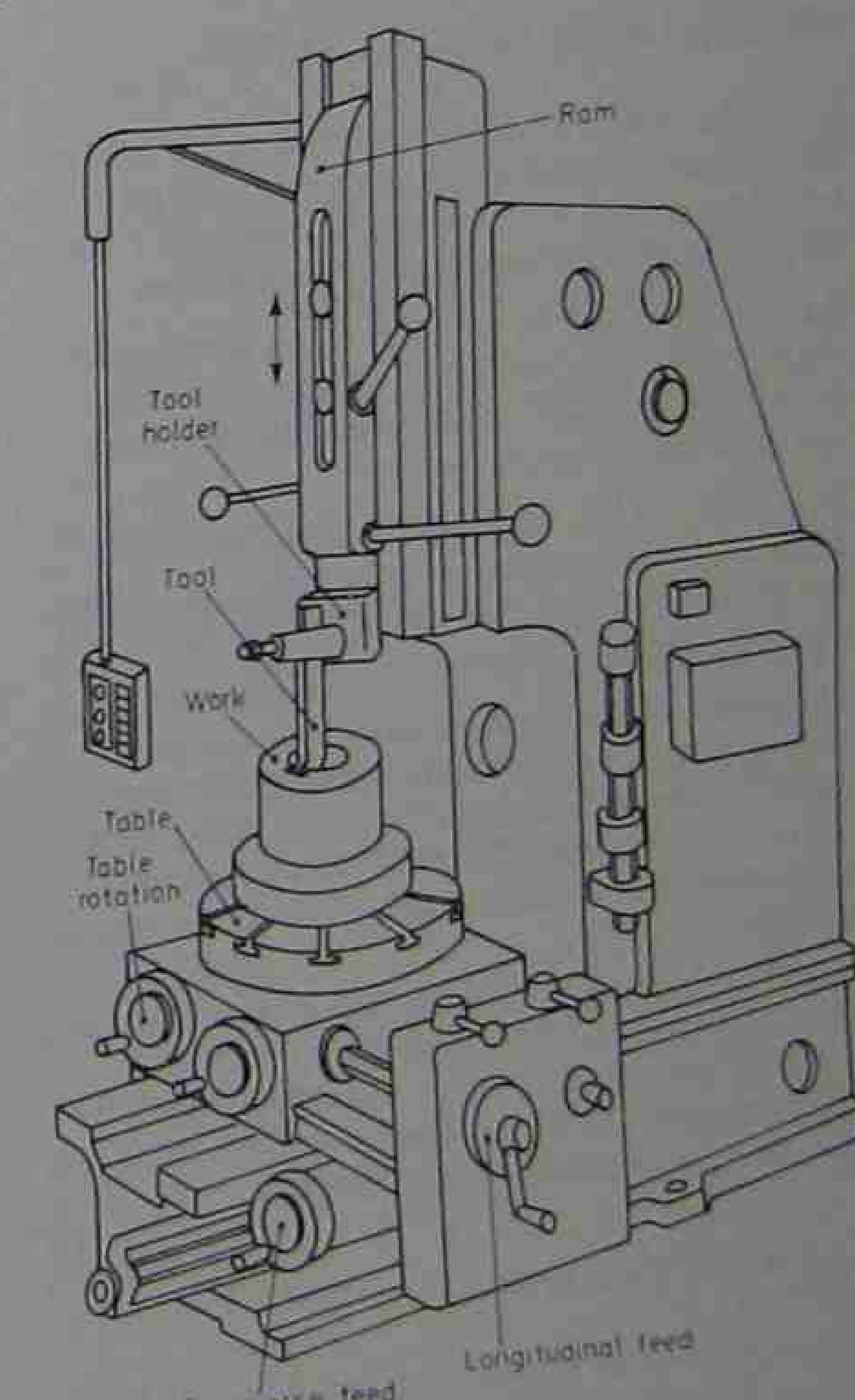


Figure 3.101 *Vertical shaper (slotter)*

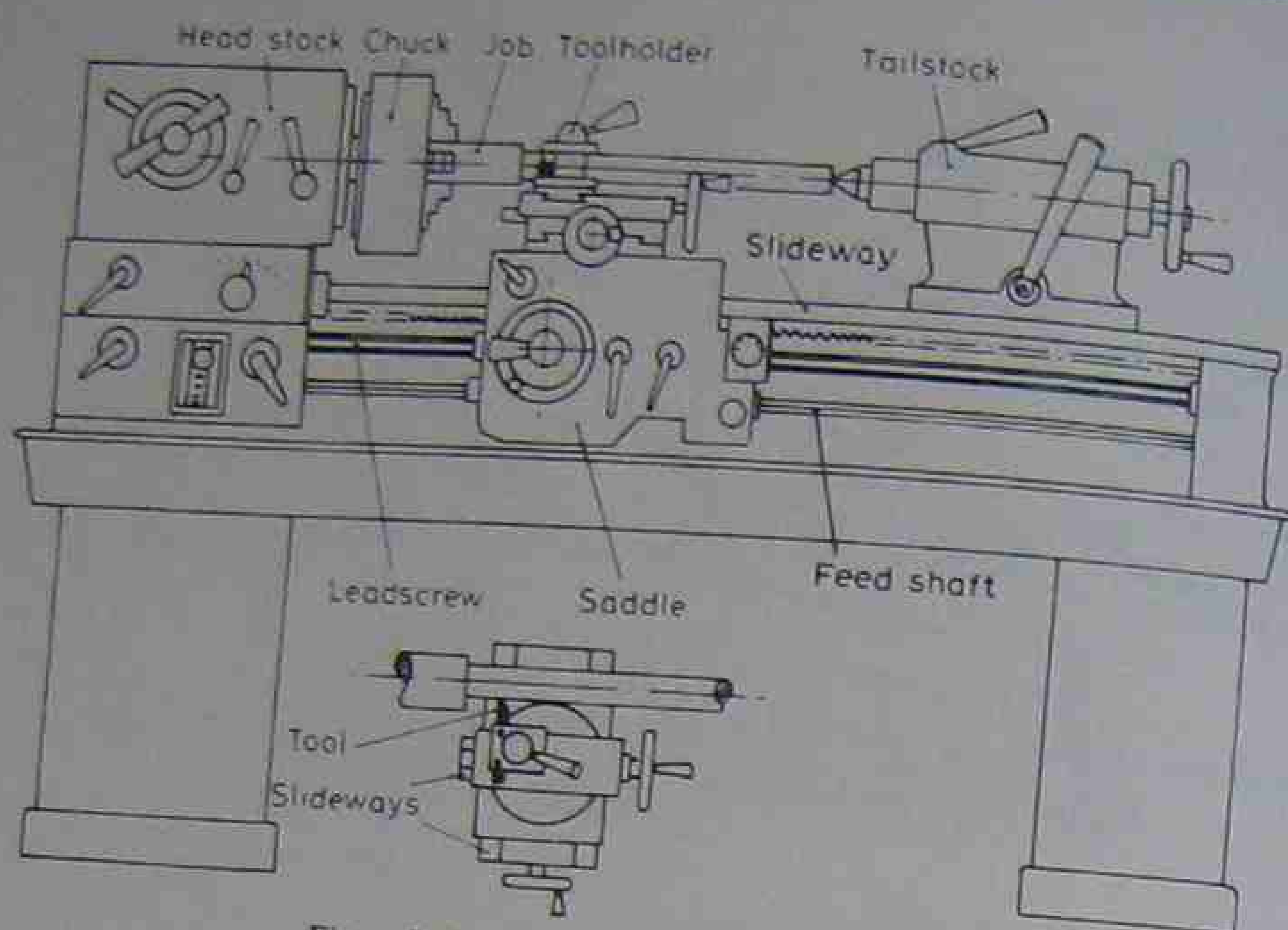


Figure 3.102 *Centre lathe (engine lathe)*

CENTRE LATHE (ENGINE LATHE)

The centre lathe is a machine tool used to produce flat, cylindrical and conical surfaces with a single point tool and the work rotating. It consists of a bed, headstock, tailstock and saddle with a leadscrew.

Bed A casting resting on the floor and carrying the other parts of the lathe. Running along the length of the bed is a slideway on which the tailstock is fixed and on which the saddle moves.

Headstock A box mounted at one end of the bed and containing a spindle driven by speed change gears and an electric motor. The end of the spindle takes a chuck or faceplate to hold the workpiece.

Tailstock This is mounted at the opposite end of the bed to the headstock. It holds a conical 'centre' which can support the end of the workpiece, or a drill chuck when required. The tailstock can be moved along the bed on a slideway and locked in the desired position.

Saddle This carries the toolholder and the slides and turntable which control the tool position. It is traversed along the bed either manually or by a *feed shaft* (lay shaft). When carrying out screw-cutting it is driven by a *lead screw* geared to the motor.

LATHE OPERATIONS

The most common operation performed on the lathe is that of 'turning' cylindrical surfaces. Others are parting-off, boring, facing, drilling and screwcutting.

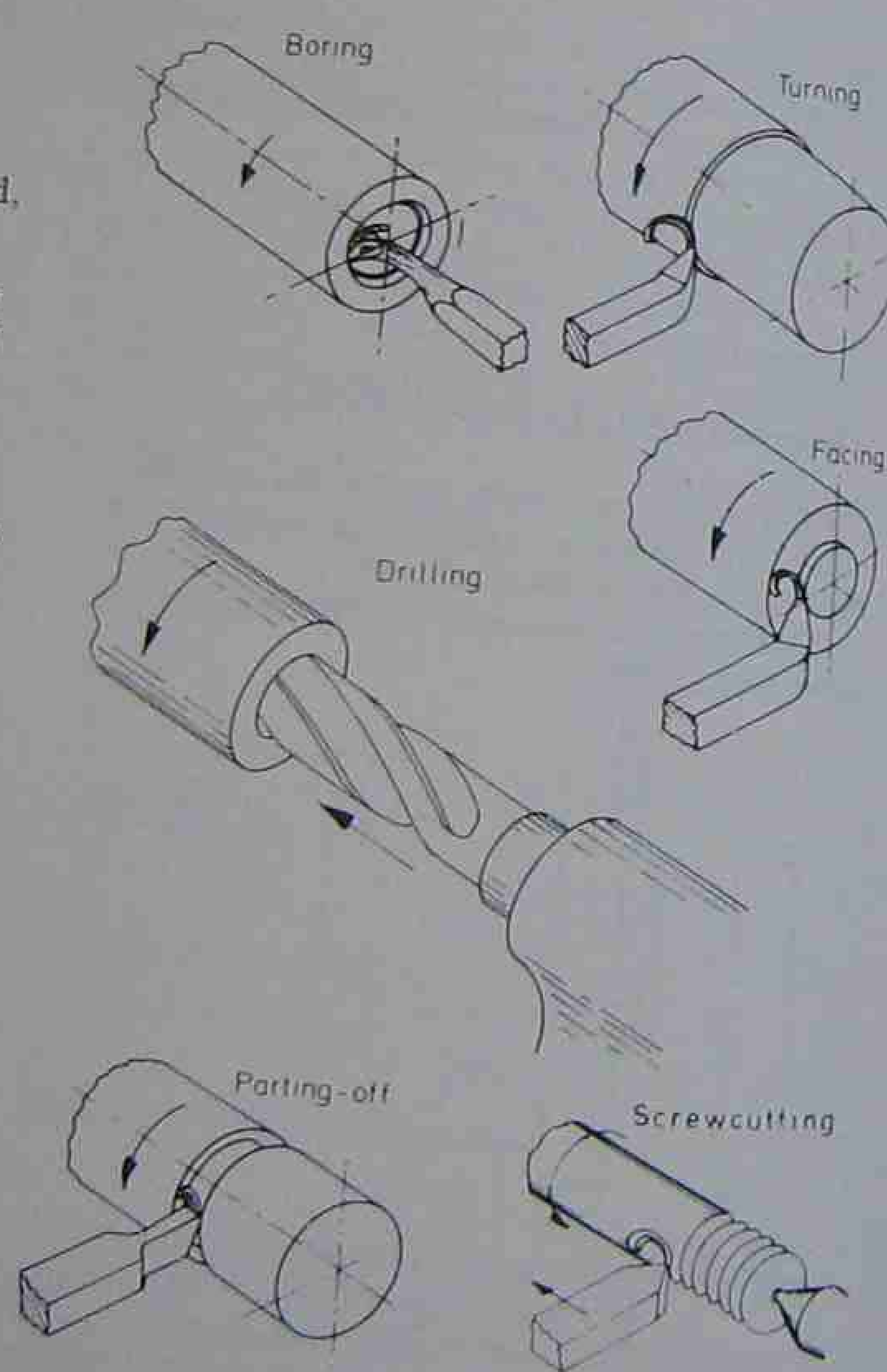


Figure 3.103 *Lathe operations*

LATHE TOOLS

There are numerous types of lathe tool designed to suit specific operations. The principle ones are: bar-turning, turning and facing, parting-off, facing, boring and screw-cutting.

Some tools are made from one piece of tool steel, others have high-speed steel tips welded to carbon steel shanks and some have tungsten carbide tips brazed on to a steel shank. A tool-holder with interchangeable tips can also be used.

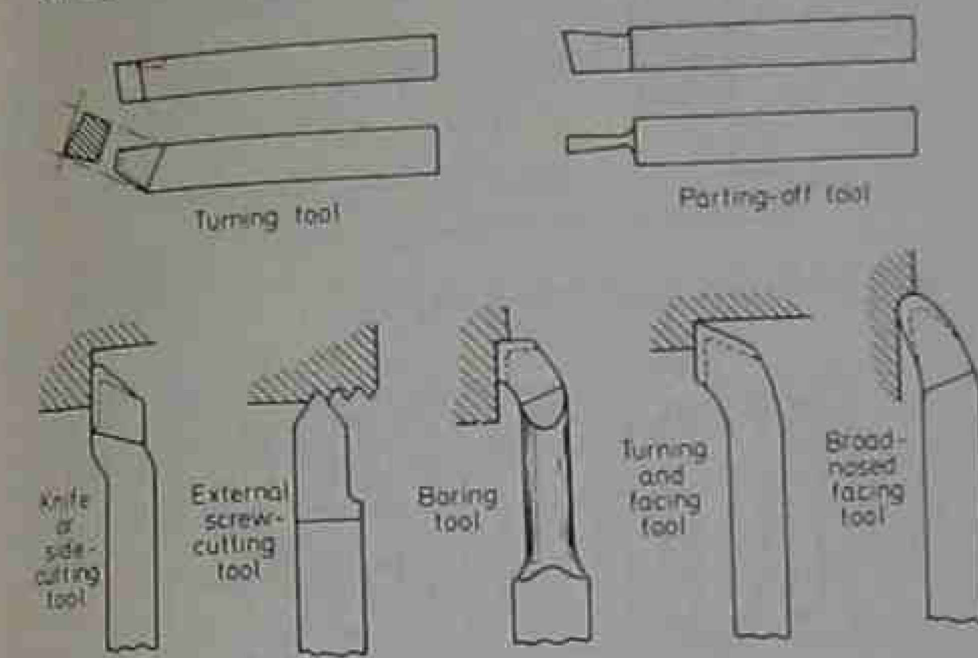


Figure 3.104 *Lathe tools*

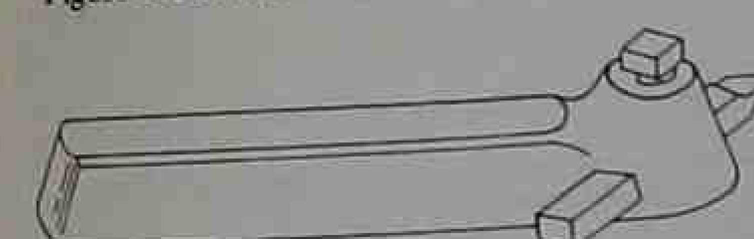


Figure 3.105 *Lathe tool holder*

LATHE CHUCKS

The chuck is a device for holding the workpiece in a lathe, and the three main types are: *three-jaw self-centring, four-jaw independent, collet.*

Self-centring three-jaw chuck This has a cylindrical body with jaws in three radial slots driven by a scroll plate in the body. As the scroll plate is rotated by means of a bevel gear operated by a hand key, the jaws move in the slots simultaneously.

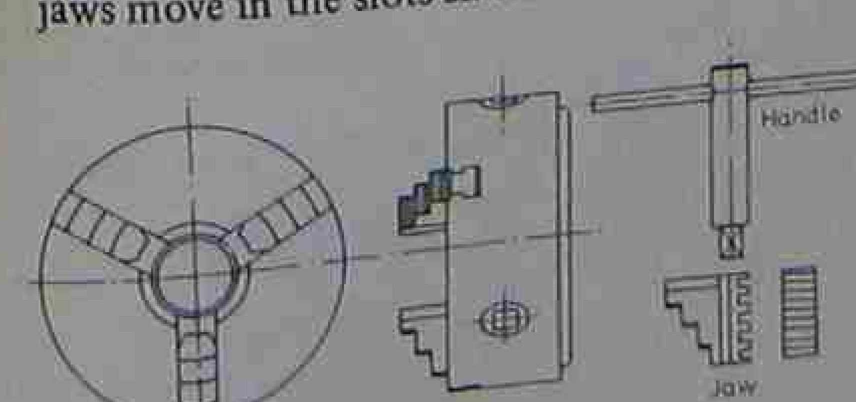


Figure 3.106 *Self-centring three-jaw chuck*

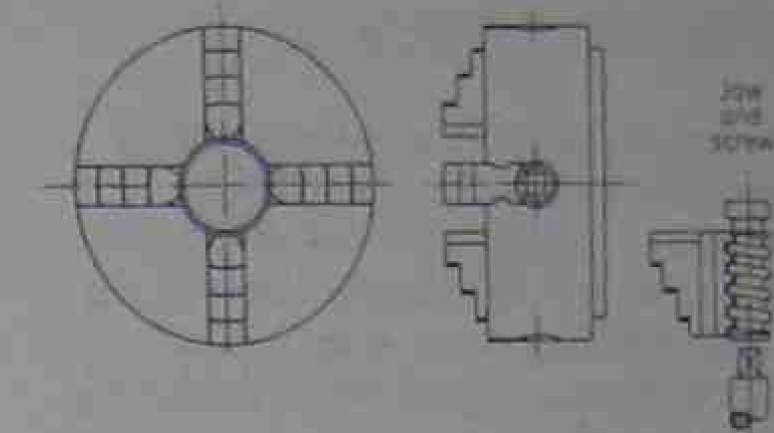


Figure 3.107 *Independent four-jaw chuck*

Independent four-jaw chuck Each of the four jaws is moved radially by a square thread on a key-operated screw which engages with a mating thread on the back of the jaw.

Collet chuck (collet) Collets are used to hold standard sizes of round, square and hexagonal stock bar. The bar is held in a tapered sleeve tightened by means of a screwed collar.

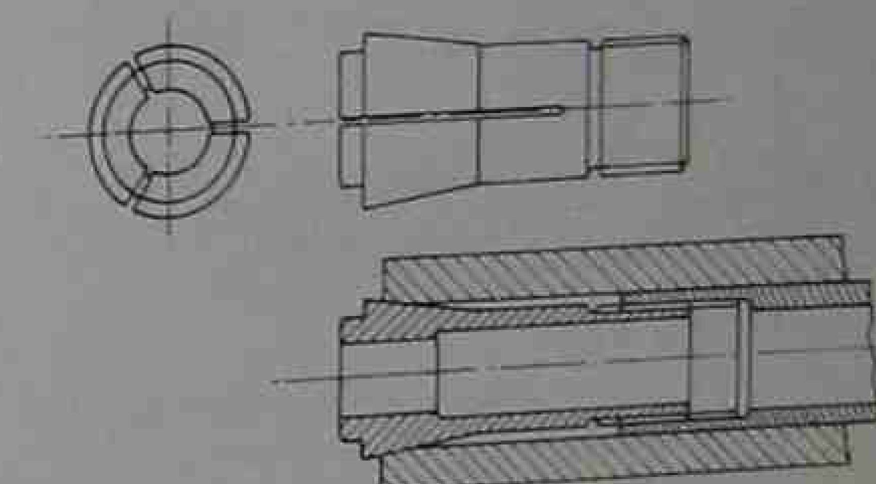


Figure 3.108 *Spring collet*

FACE PLATE

A face plate is used instead of a chuck for objects of irregular shape such as castings. It is a flat circular plate screwed to the spindle and pierced with slots to which the workpiece is clamped. It is often necessary to use a balance weight to prevent vibration.

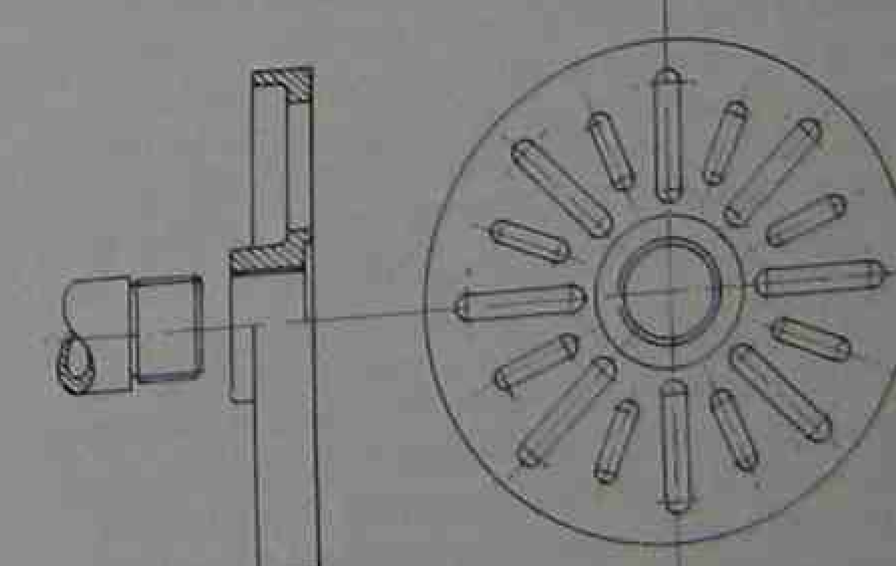


Figure 3.109 *Face plate*

TURNING BETWEEN CENTRES

Objects such as shafts are often mounted between centres instead of in a chuck. 'Centre holes' are drilled in the ends of the shaft by a centre drill, and conical centres in the spindle nose and tailstock are fitted into them. A driving pin on a 'catch plate' locked to the spindle nose engages with a driving dog clamped to the workpiece.

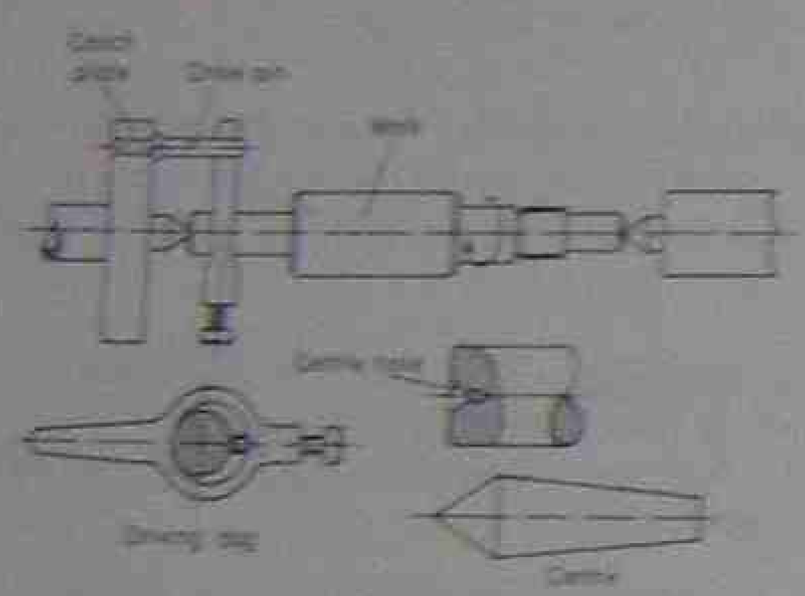


Figure 3.110 *Turning between centres*

Figure 3.111 *Centre drill*

GRINDING MACHINES

Grinding machines are used to produce flat, cylindrical and other surfaces by means of high-speed rotating abrasive wheels. Grinding is often a means of giving a more accurate finish to a part already machined but it is also a machining process in its own right.

Surface grinding machine This is used to produce a highly-finished flat surface. The work is held on a horizontal reciprocating table by clamps or by a magnetic chuck. A grinding wheel rotates on a horizontal axis at right angles to the table motion. This table is movable along the wheel axis and the wheel can be moved vertically to give the required cut.

The magnetic chuck is a flat steel plate with permanent magnets embedded in it, and it is particularly useful for holding down thin plate.

Cylindrical grinding machine A machine similar in layout to a centre lathe but with the tool replaced by a seperately-driven grinding wheel. Cylindrical surfaces are produced by traversing the wheel along the axis, and complex profiles may be produced by specially-shaped grinding wheels.

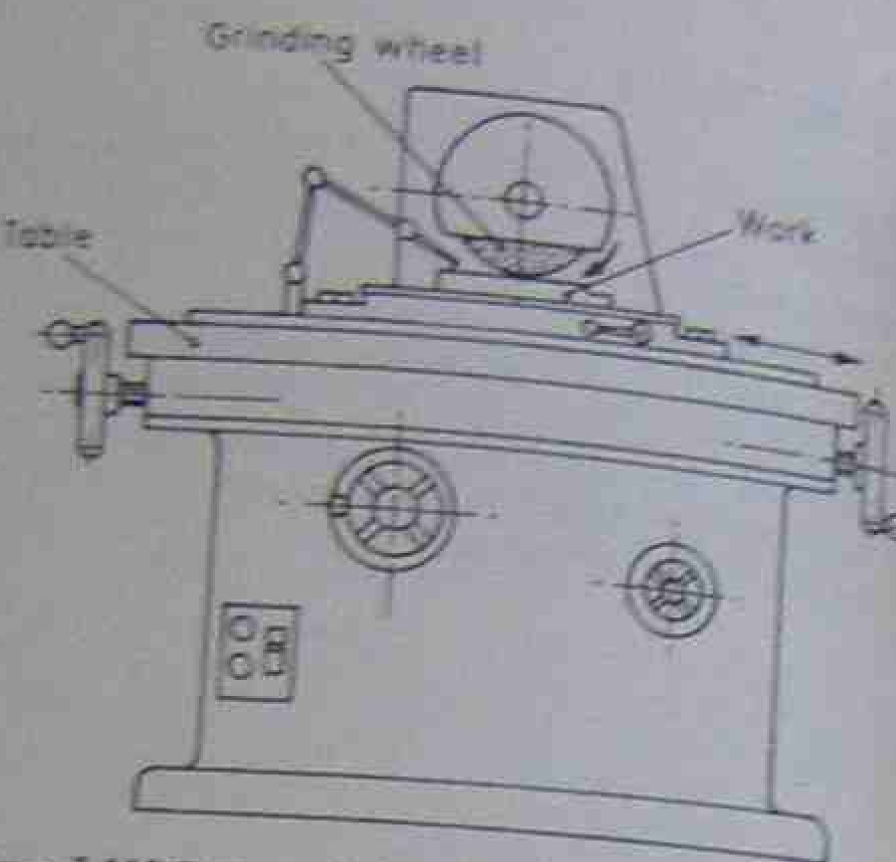


Figure 3.112 *Surface grinding machine*

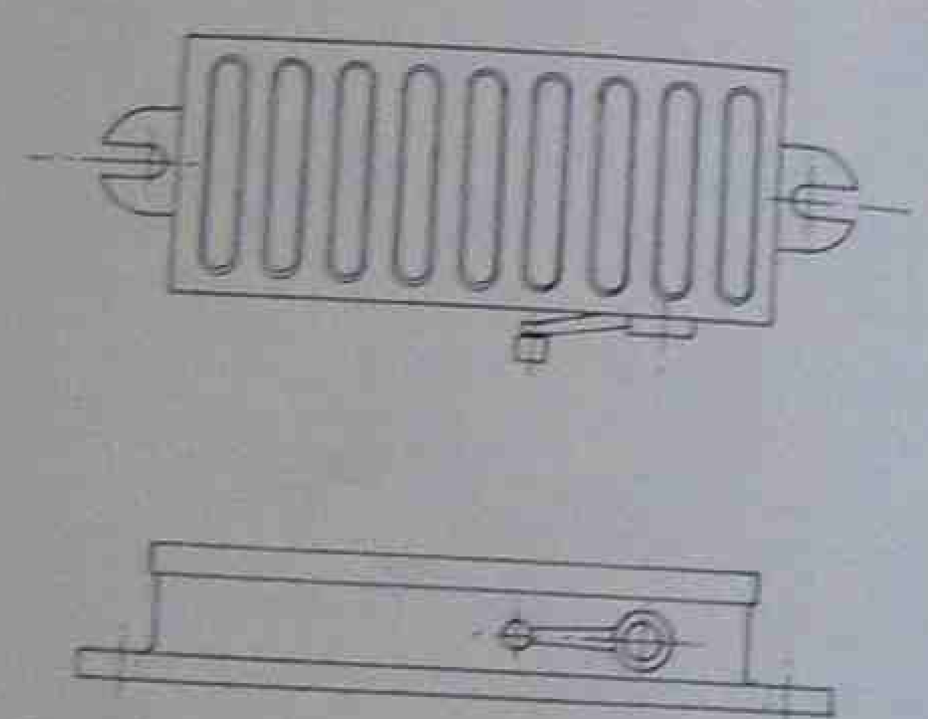


Figure 3.113 *Magnetic chuck for surface grinding machine*

GRINDING WHEELS

Grinding wheels are made in a vast number of materials, shapes and sizes. Typical materials are bonded abrasive powders such as aluminium oxide (Al_2O_3), silicon carbide (Si C), or diamond. Some typical shapes are shown.

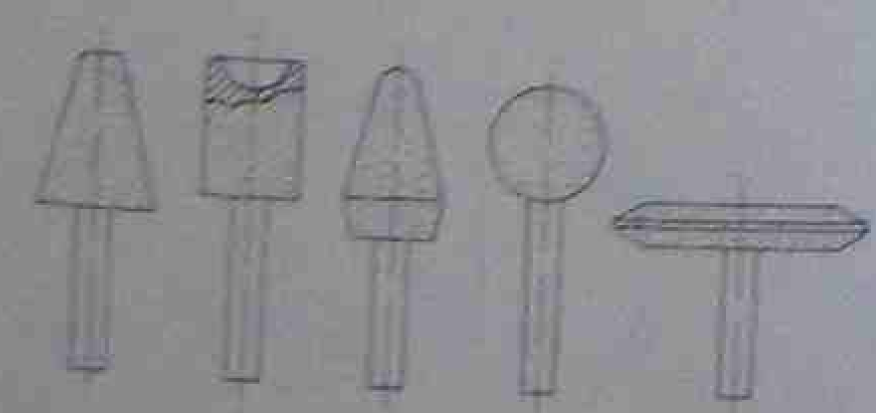


Figure 3.114 *Contour grinding wheels*

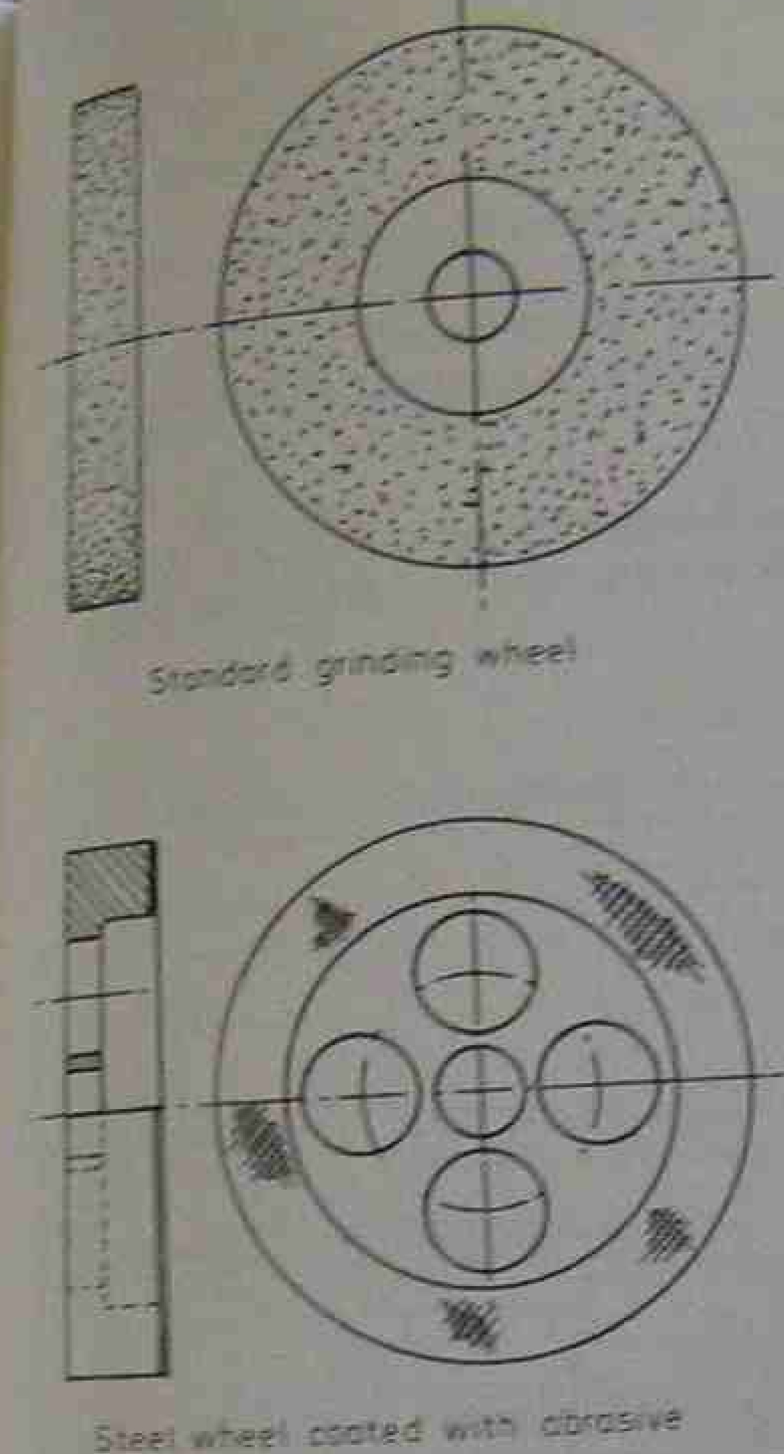


Figure 3.115 *Grinding wheels*

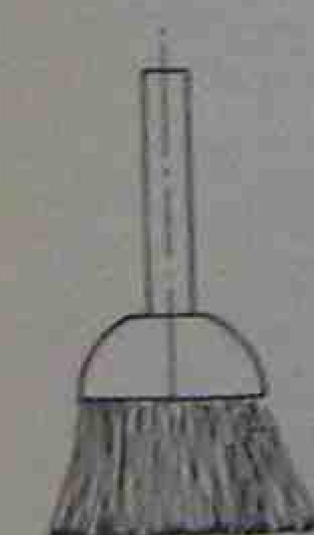


Figure 3.116 *Rotary wire brush*

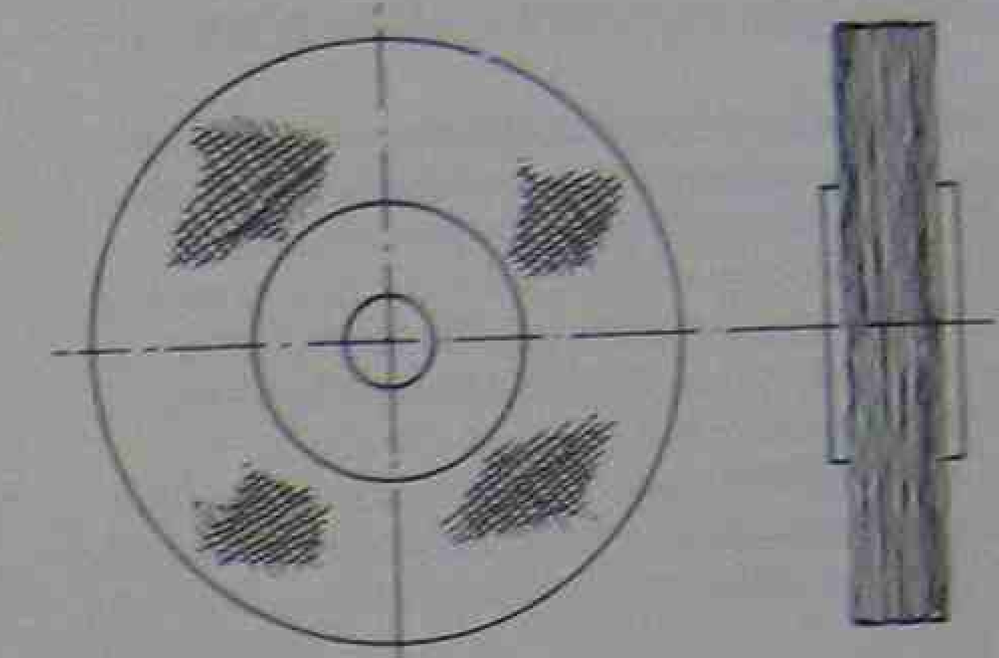


Figure 3.117 *Polishing (buffing) mop*

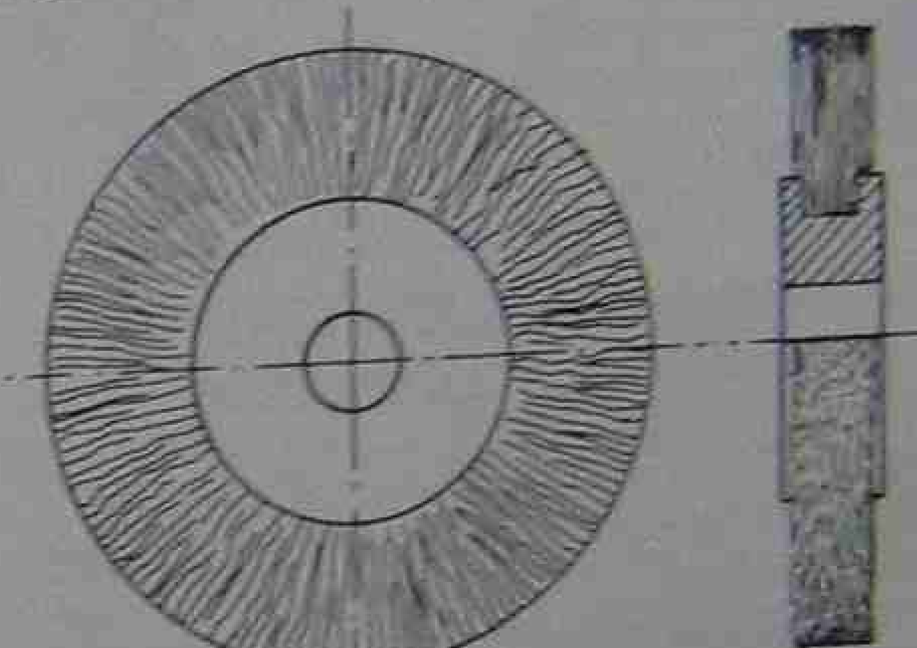


Figure 3.118 *Rotary wire brush*

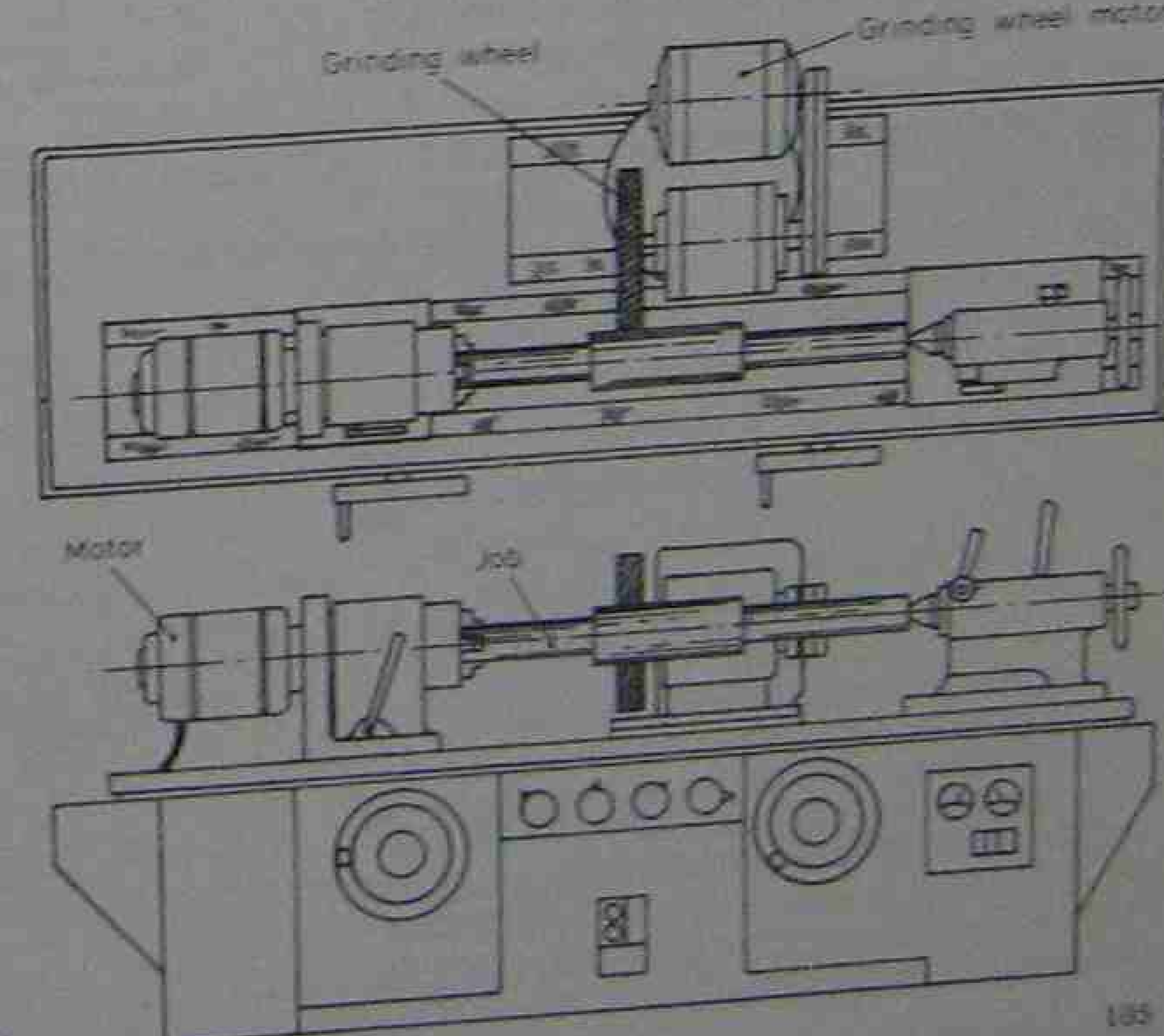


Figure 3.11 ... nding machine

BENCH AND PEDESTAL GRINDERS

These machines are used for manual grinding operations such as sharpening tools for lathes, shapers, etc., and for sharpening chisels, screwdrivers and other hand tools.

The wheels are mounted on the shafts at both ends of an electric motor. Wire brush wheels and buffing mops may be fitted.

These machines and the hand-held grinder are generally referred to as 'off-hand' grinders as opposed to 'precision' grinders.

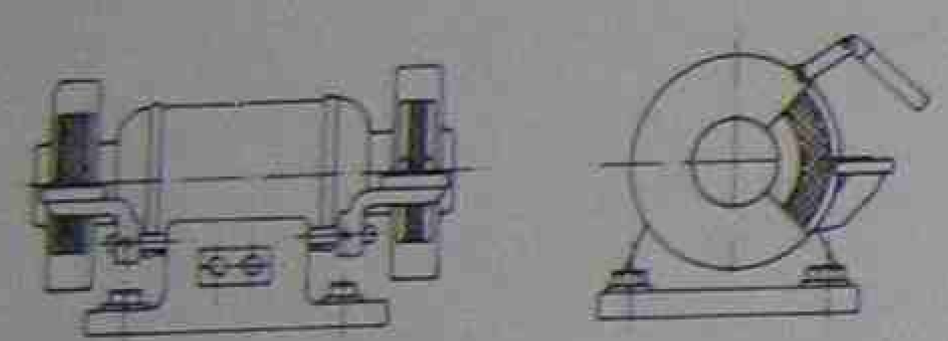


Figure 3.120 *Bench grinder*

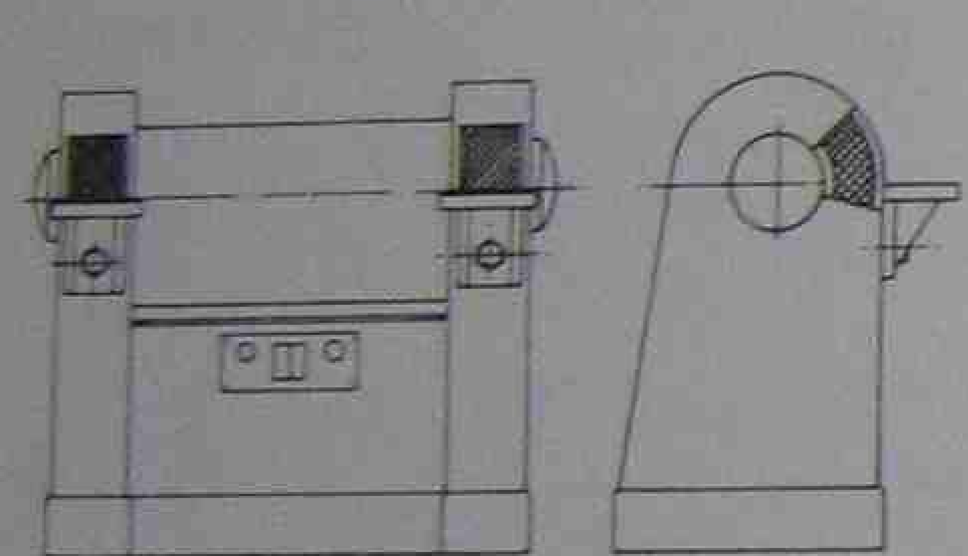


Figure 3.121 *Heavy duty pedestal grinder*

BORING MACHINES

To make large diameter holes boring is used as an alternative to drilling.

In a boring machine the work is mounted on a stationary table and the boring bar rotates on a vertical spindle. In a *boring mill* the work is usually very large and it rotates while the boring bar remains fixed.

Horizontal boring machines These are used for very large work and they also have drilling and milling facilities.

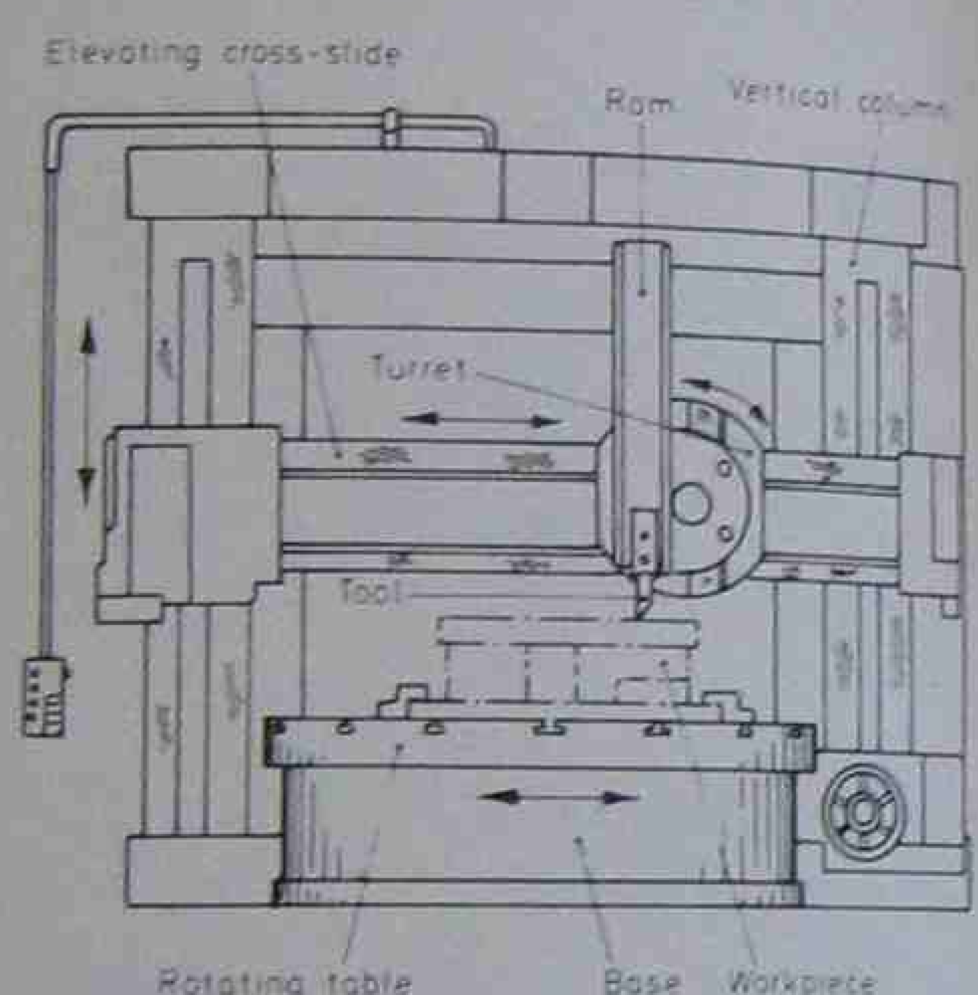


Figure 3.122 *Vertical boring mill*

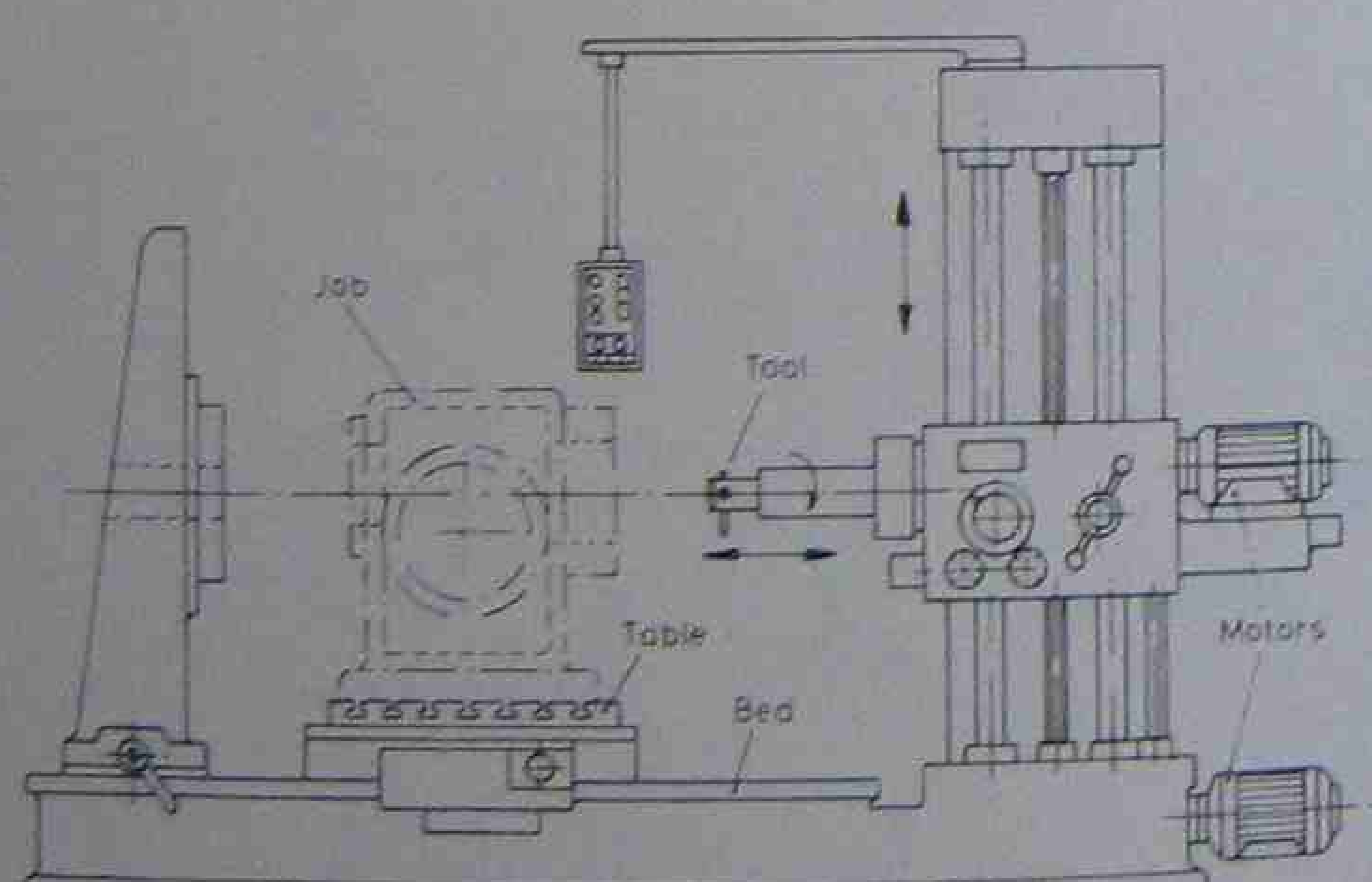


Figure 3.123 *Horizontal boring machine*

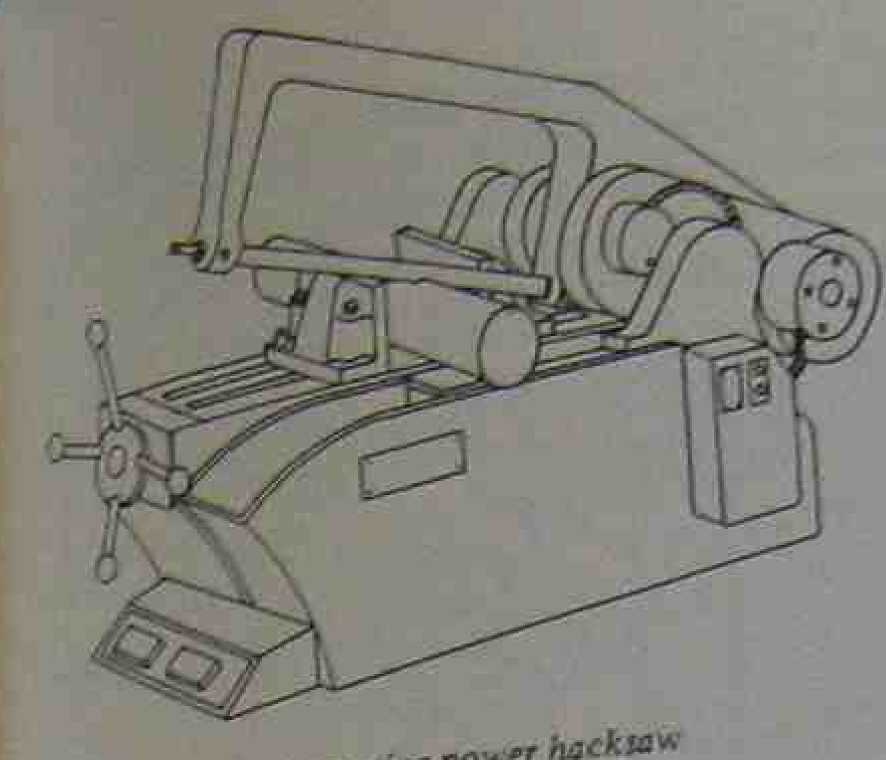


Figure 3.124 *Reciprocating power hacksaw*

POWER SAWS

There are two main types of power say, the *reciprocating power hacksaw* and the *bandsaw*.

The reciprocating saw has a blade and action similar to that of a hand hacksaw. The work, usually heavy bar, is clamped in a vice with the feed being controlled by a *dashpot* which dampens vibrations of the saw.

The light duty bandsaw has a continuous flexible blade driven vertically past the work which is moved by hand on a horizontal table. This saw is suitable for cutting wood, plastics and thin sheet, usually of soft metals.

The reciprocating power saw is being replaced by heavy duty bandsaws capable of cutting very thick, tough metal. The work is clamped on a table which may be tilted and is fitted with a feed mechanism.

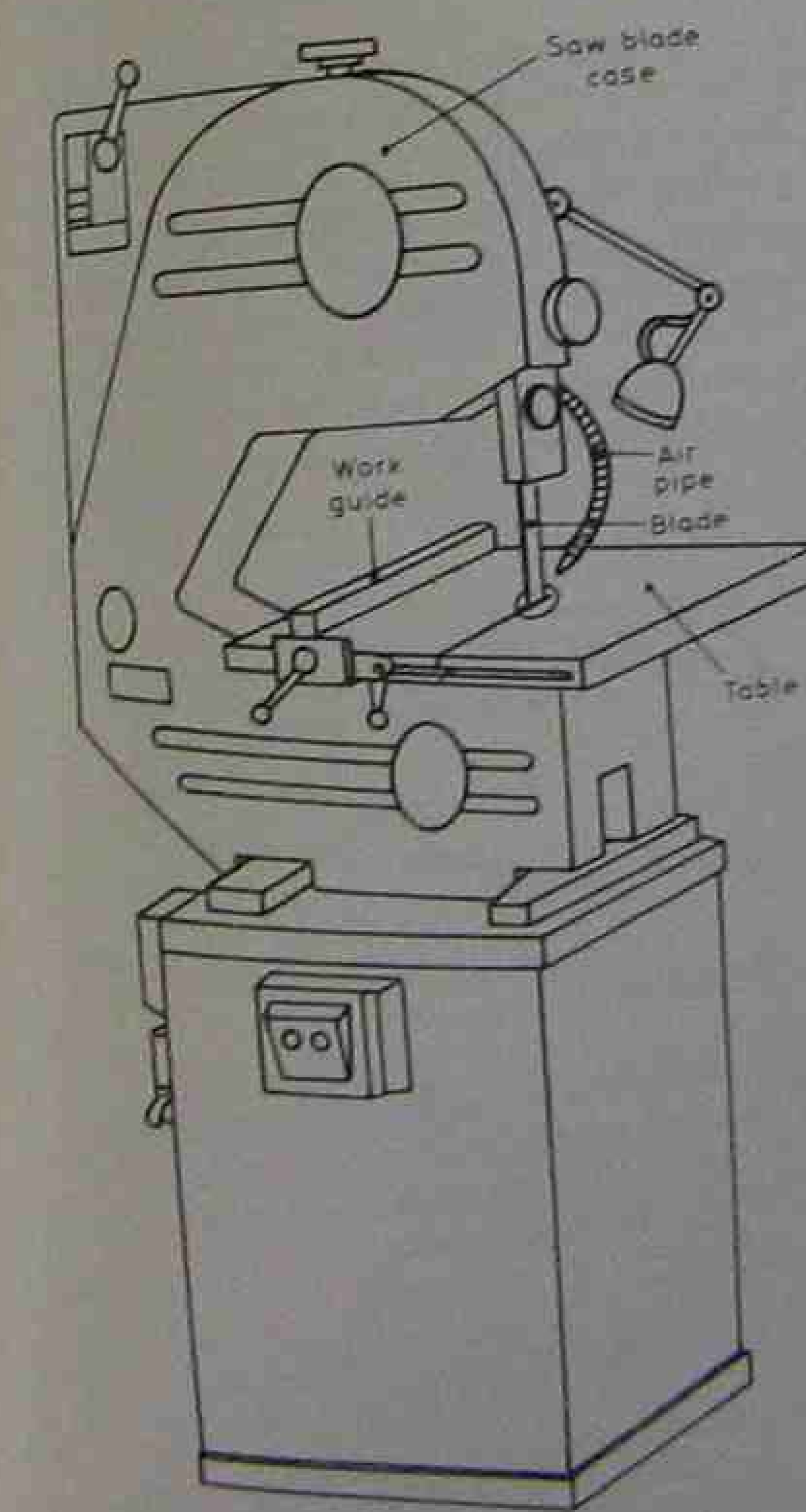


Figure 3.125 *Light duty bandsaw*

3.3 SOFT-SOLDERING, BRAZING AND WELDING EQUIPMENT

SOFT-SOLDERING

Soldering is the hot bonding of metal parts by adhesion using a thin film of low melting point alloy known as solder. Small joints are made by applying heat with a soldering iron while large joints are made by 'sweating' the parts together in a gas flame.

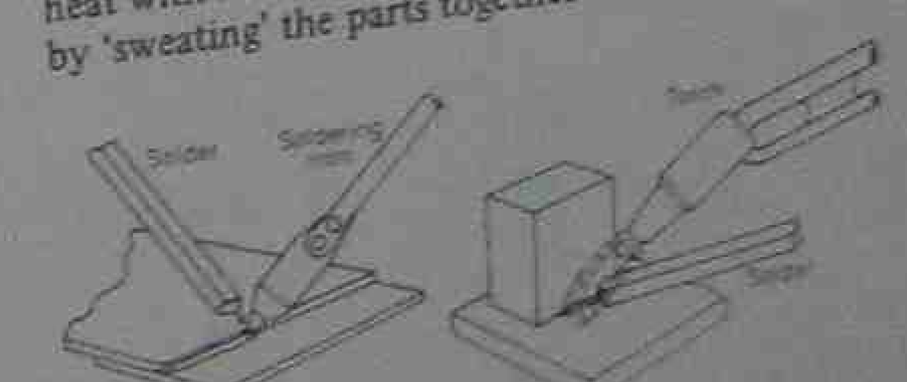


Figure 3.126 *Soft soldering*

SOLDERING IRONS

For heavy engineering work soldering irons have heavy copper 'bits' mounted on an iron shank with a wooden handle. Bits may be of either the 'straight' or the 'hatchet' type, and both may be heated by a gas flame or have an internal electric heating element. Electric irons are made in small sizes for electronic work. The bit is heated to the required temperature and before use the working edge is coated with solder, or 'tinned'.

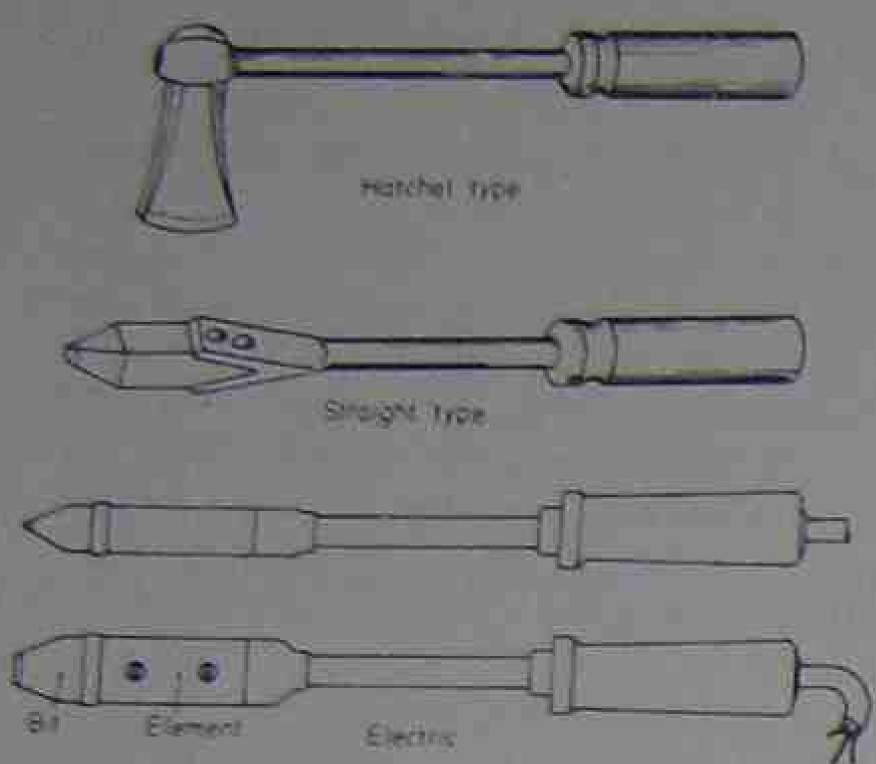


Figure 3.127 *Soldering irons*

SOFT SOLDER

Soft solder is a mixture of tin and lead which melts at the relatively low temperature of 183°C. The proportion is approximately two parts of tin to one part of lead. It is available in bar form for general engineering work and also in the form of wire with a resin flux core.

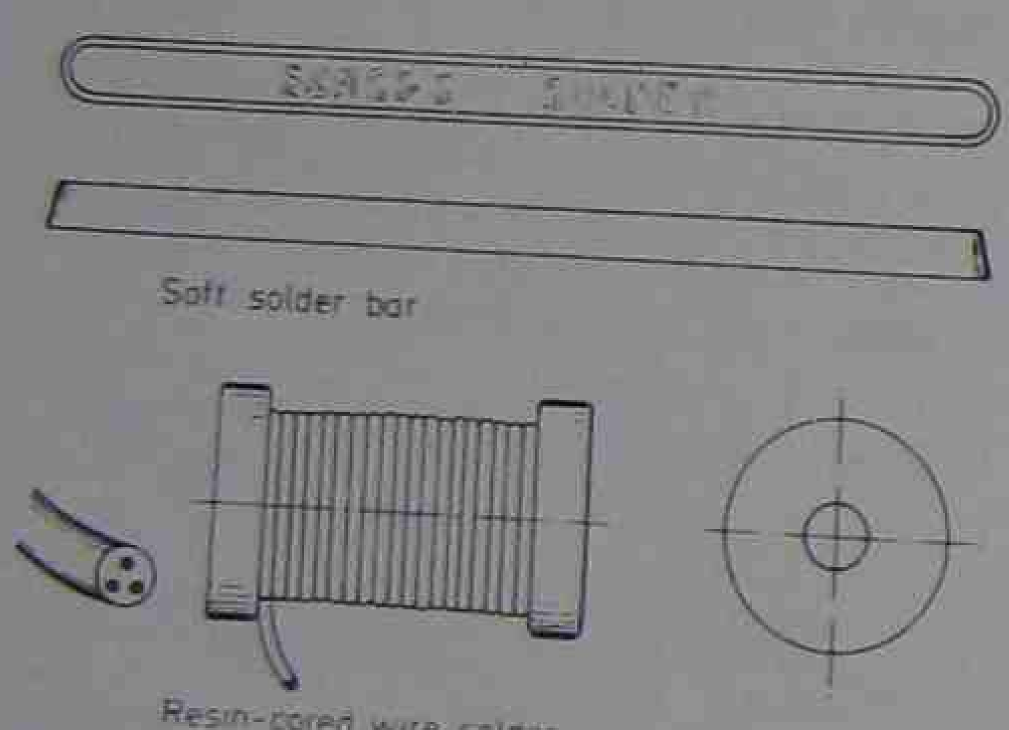


Figure 3.128 *Types of solder*


Figure 3.129 *Silver solder*

SILVER SOLDERING

Silver solder is an alloy of silver, copper and zinc with a melting point of about 700°C, that is, lower than that required for brazing. It may be used for joining brass and copper. The solder is supplied in a strip form.

FLUX

Flux is a substance which, when heated, provides a liquid or gaseous shield which excludes oxygen and prevents oxidisation of the surface of the metal being soldered. It may be an active flux such as zinc-chloride (killed spirit) or a passive flux such as resin.

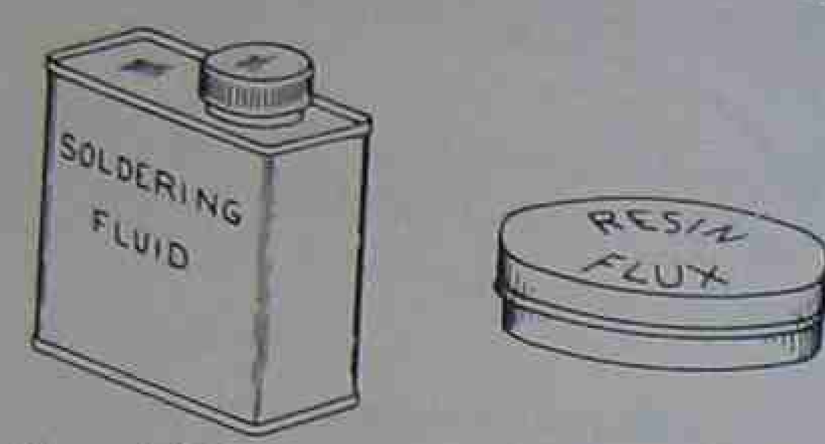


Figure 3.130 *Soft solder fluxes*

BRAZING (HARD SOLDERING)

In brazing, the joining metal is brass consisting of varying proportions of copper and zinc known as 'spelter'. The brazing temperature of about 900°C is much higher than that for soft soldering. Spelter is supplied in the form of round rod of small diameter or in granular form mixed with flux.

Borax is generally used as brazing flux in the form of a powder which is mixed to a paste with water. The heat of brazing melts the borax to form a protective slag.

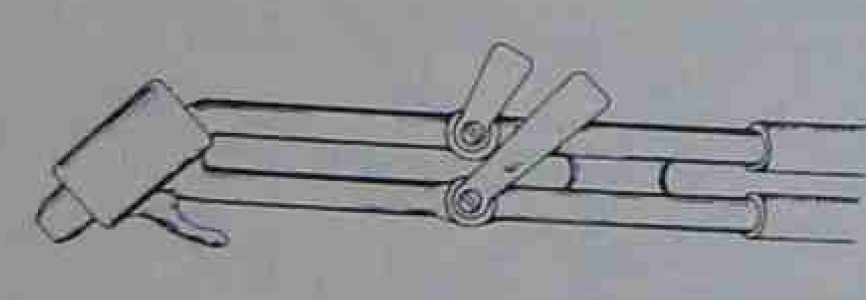


Figure 3.131 *Gas-air brazing torch*

BRAZING TORCH

A special torch is required for brazing. The usual type is supplied with mains gas and air from a blower, and it is fitted with adjustable gas and air taps for controlling the flow and mixture of gases.

BRAZING HEARTH

Brazing is best carried out in a hearth designed for the purpose. The hearth is made from steel lined with fire brick and has an electrically-driven air blower mounted on it.

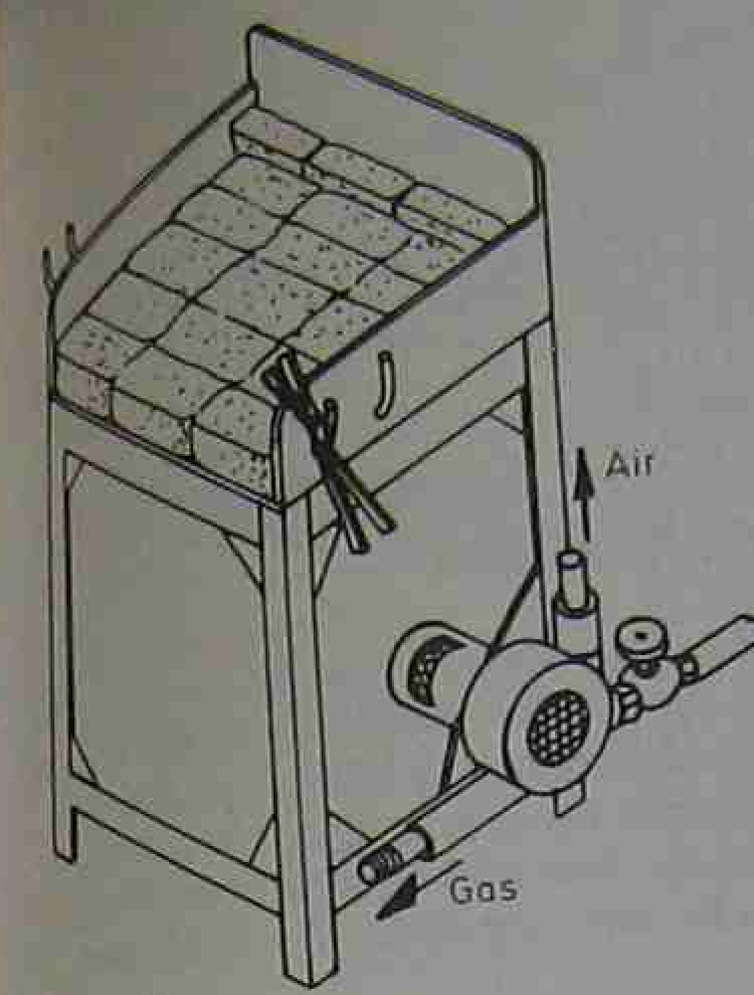


Figure 3.132 *Brazing hearth*

WELDING

Welding is a process for joining metals, plastics, etc. by heating them locally until they can be joined by fusion or forging. The main methods of fusion welding are: electric arc, gas torch, friction and resistance welding.

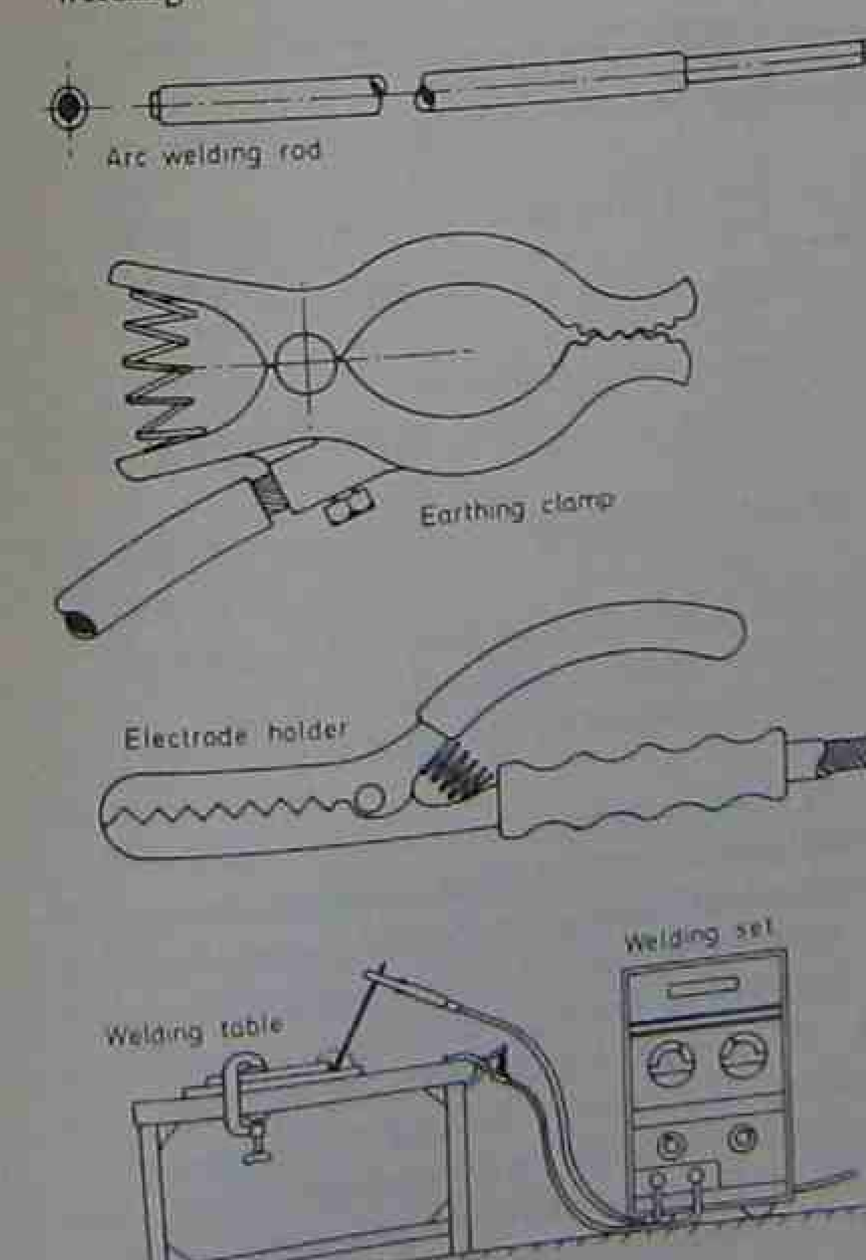


Figure 3.133 *Arc welding equipment*

ARC WELDING

In this process the heat of fusion is generated by an electric arc struck between two electrodes, one of which is the workpiece and the other a welding rod. The welding rod is made of a material similar to that of the parts being welded and is coated with a solid flux which melts when heated. It is used to 'fill' the welded joint.

Power is obtained from an a.c. or d.c. *welding set* providing a regulated low voltage, high current supply to an *electrode holder* and an *earthing clamp*. The work is done on a special *welding table* made of steel on which the work is clamped and to which the earthing clamp is attached to complete the electric circuit.

TYPES OF WELD

Most welds in machine elements are fillet welds although butt welds are used a great deal for pressure vessels. These and other types of weld commonly used are described in the following:

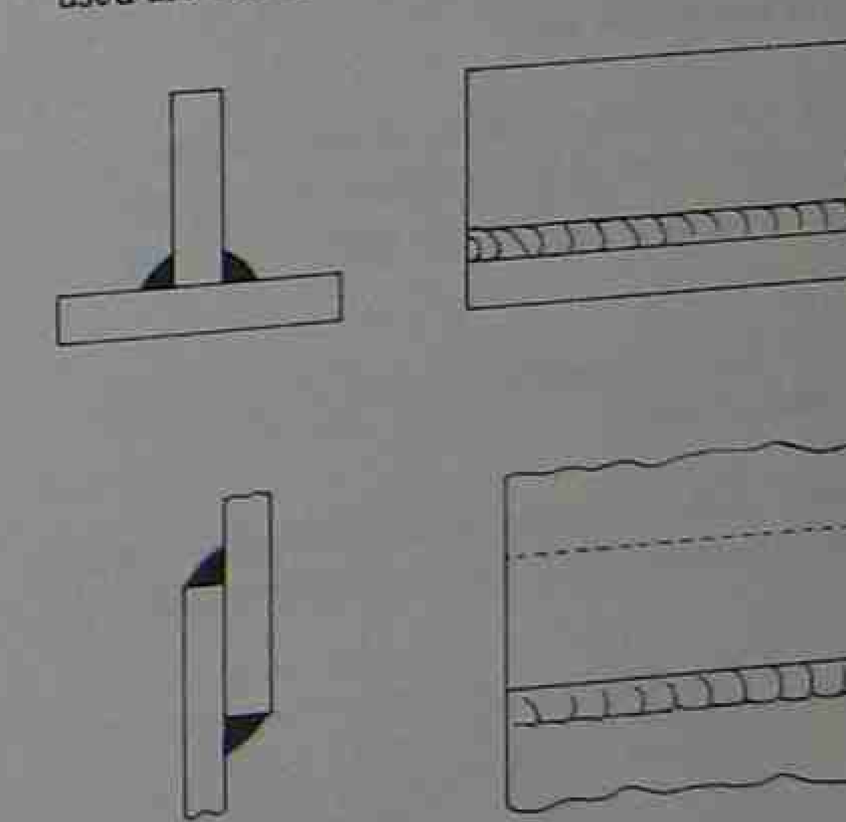


Figure 3.134 *Fillet welds*

Fillet weld A weld joining two plates at right angles, or overlapping plates. The weld is formed in the corner and is of approximately triangular shape.

Intermittent weld In the interests of economy and to reduce distortion, fillet welds are often intermittent.

Butt weld A weld joining two plates end to end to form a continuous plate. The plates are usually prepared by chamfering and the weld may be on one side or both sides of the plates (*see* Flash weld).

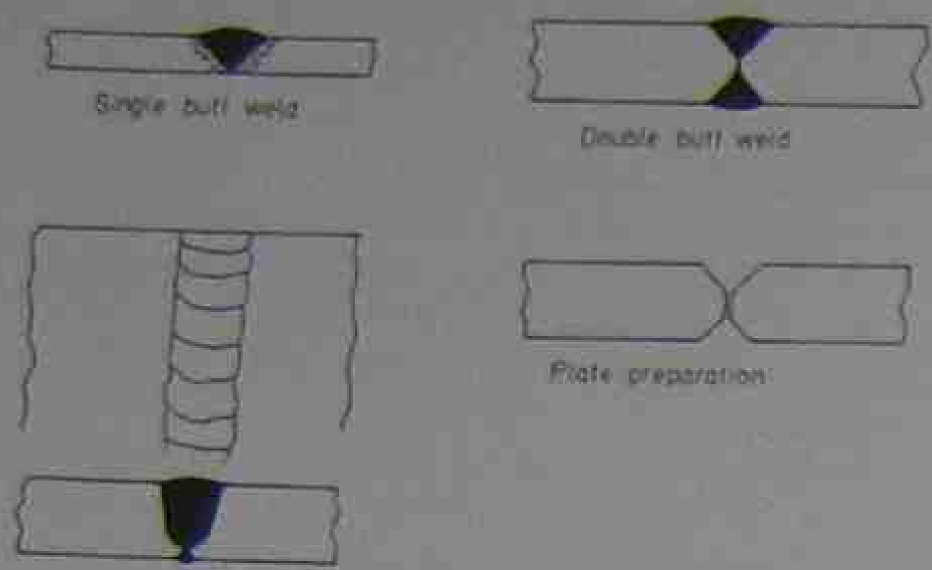


Figure 3.135 *Butt welds*

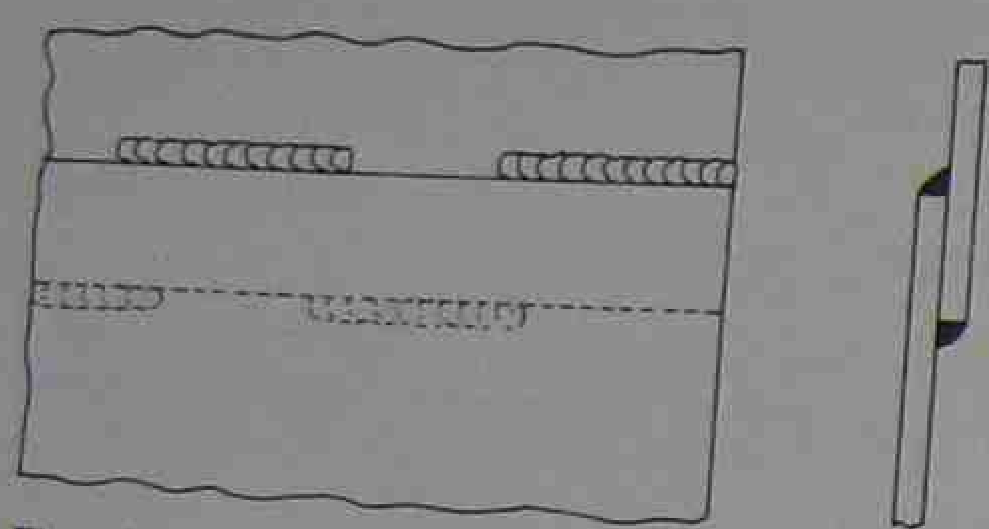


Figure 3.136 *Intermittent weld*

Plug weld A method of joining plates in which a series of holes in one plate are filled with weld to join it to the other plate to give a result similar to riveting.

Slot weld A similar method to plug welding using slots instead of holes.

Tack weld A short intermittent weld used to hold parts together temporarily before further welding.

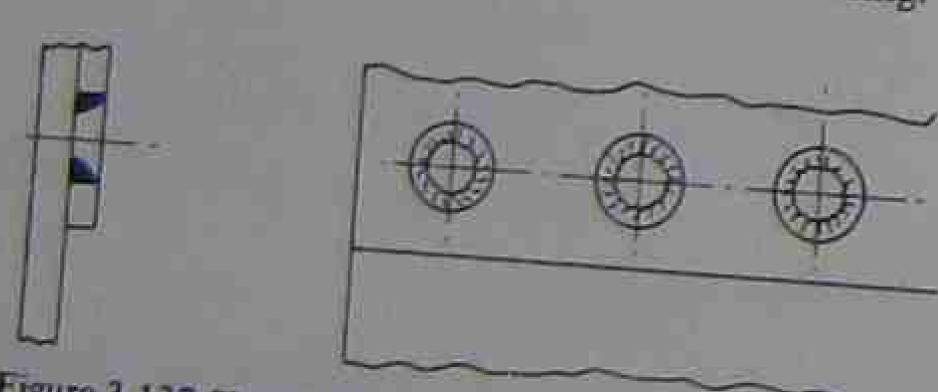


Figure 3.137 *Plug weld*

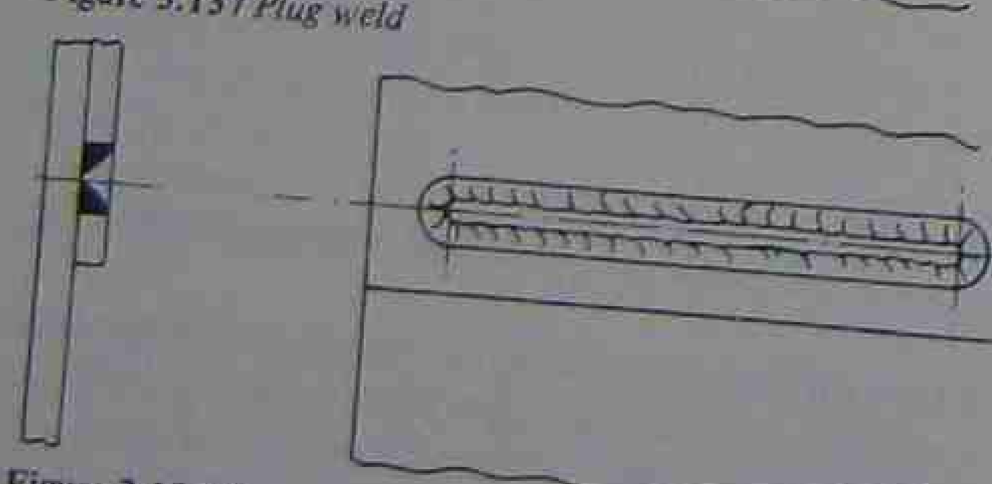


Figure 3.138 *Slot weld*

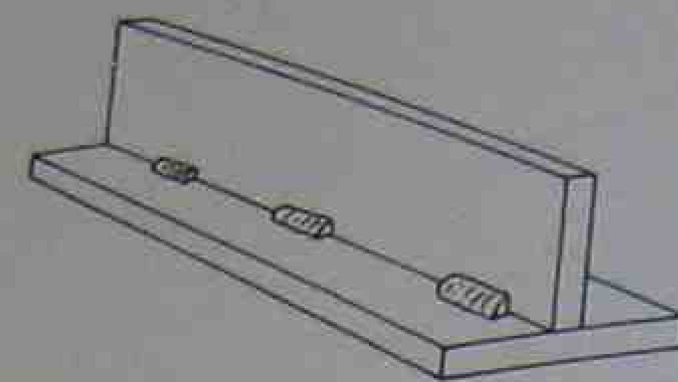


Figure 3.139 *Tack weld*

RESISTANCE WELDING

Welding is produced by fusion due to the heavy current flowing between two metal objects when they are held in close contact by electrodes. The current depends upon the nature of the surface and the contact pressure. This method is used for joining sheet metal and for attaching studs, rods and bosses etc. to plates.

Spot weld A type of resistance weld where thin plates are joined by local spots of fusion. It is used extensively for lightly-loaded light metal parts.

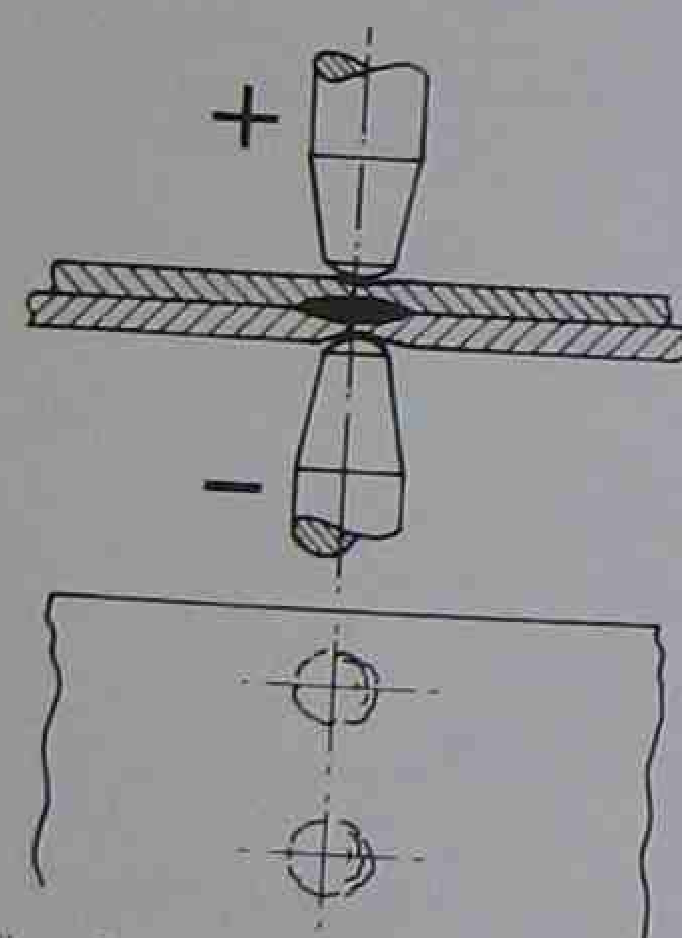


Figure 3.140 *Spot welding*

Resistance seam weld A continuous process in which two overlapping plates are joined by fusion due to the current flowing between two rotating wheels.

Flash weld A method of butt welding bars by resistance welding. The ends of the bars are connected to an electrical supply and brought together under pressure to complete the circuit. The arc produced causes fusion and the power is cut off to allow the joint to solidify.

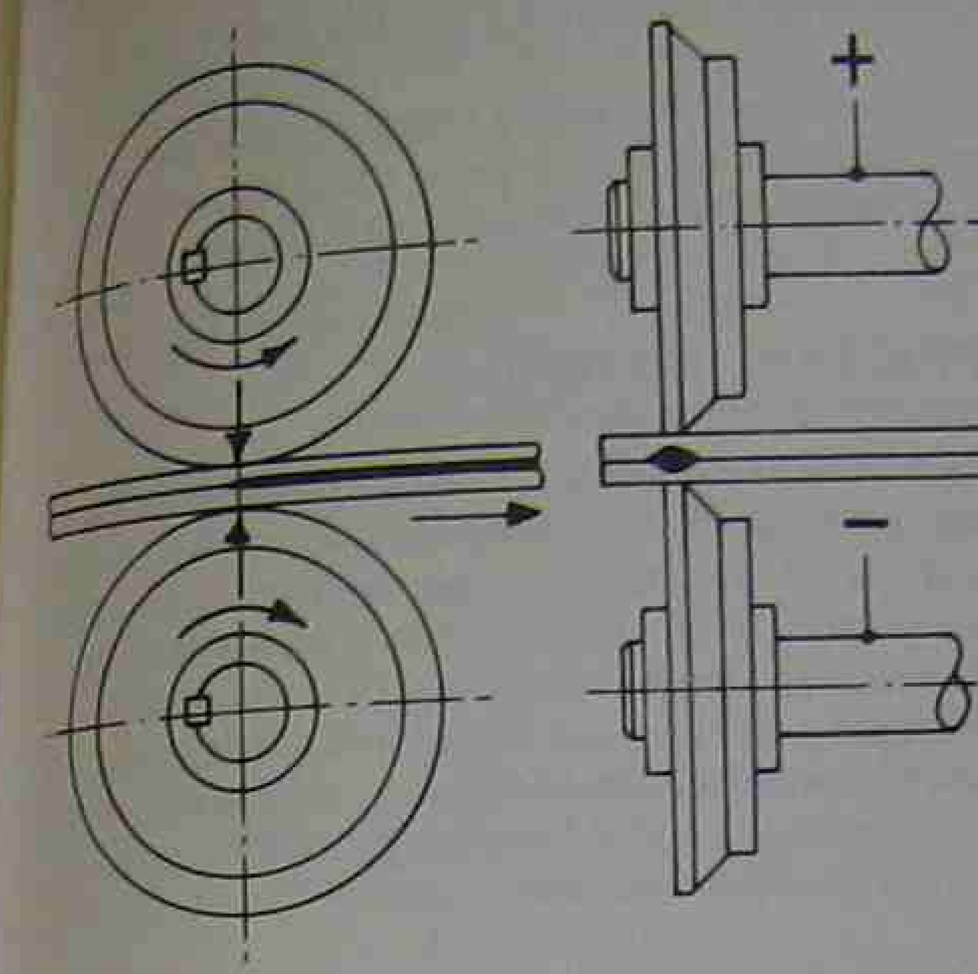


Figure 3.141 *Resistance seam welding*

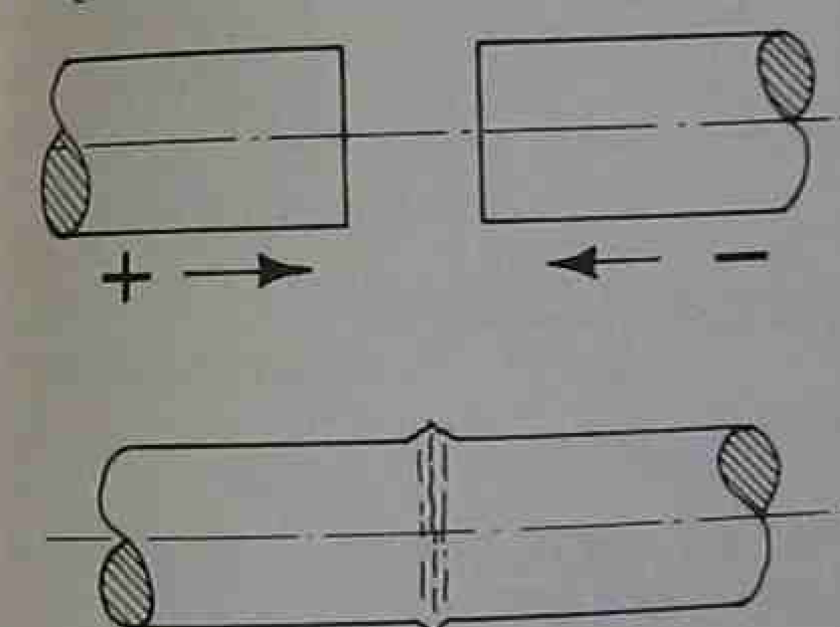


Figure 3.142 *Flash welding*

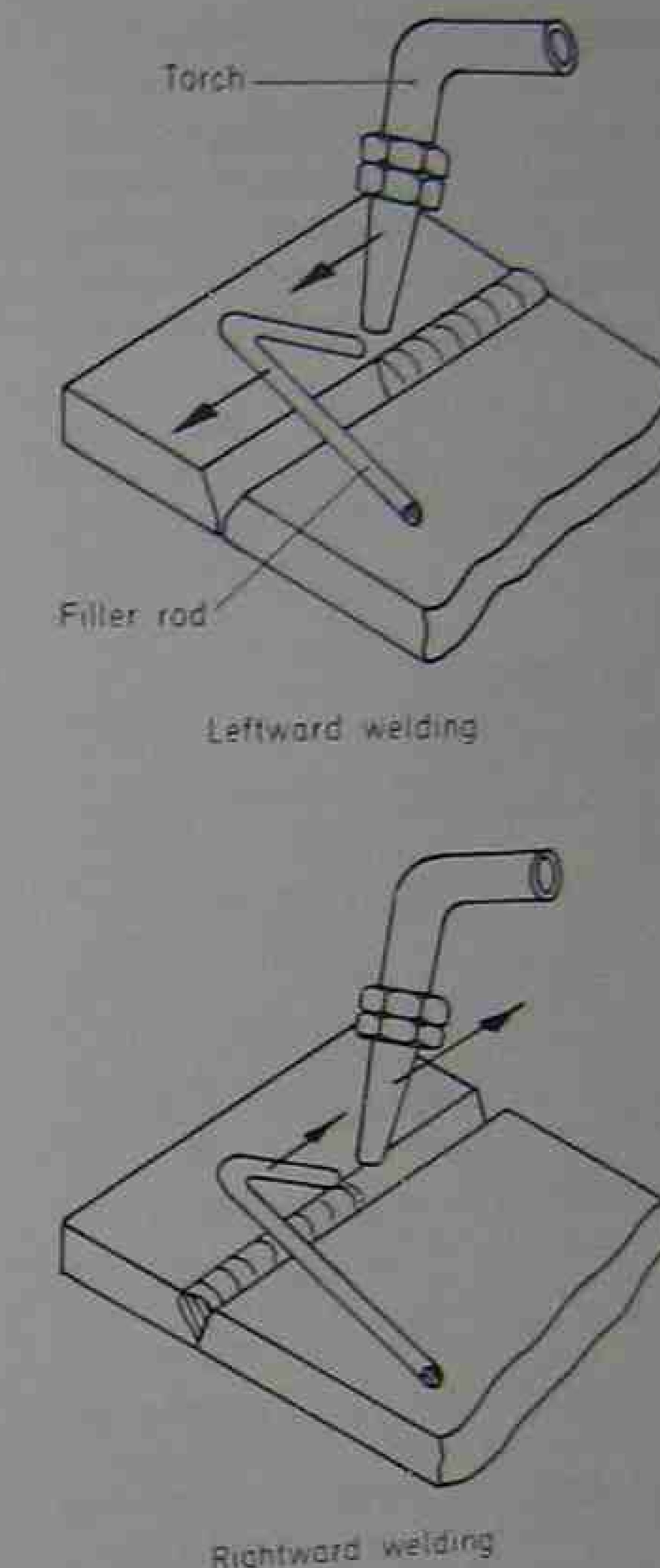


Figure 3.143 *Gas welding*

GAS WELDING

Gas welding is suitable for most sheet metals up to a thickness of a few mm.

The heat of fusion is obtained by burning pure oxygen with gases such as acetylene, butane, propane or hydrogen in a specially designed torch. A filler rod is fed by hand.

Welding may be carried out to the left or right, these methods being known respectively as 'leftward welding' and 'rightward welding'. The former is used for metal thickness of up to 3mm and the latter for over 3mm, but gas welding is not often used for plate over 3mm.

Gas welding equipment The welding torch, fitted with gas and air control valves and interchangeable nozzles, is connected to gas cylinders by flexible hoses leading from regulators on the bottles.

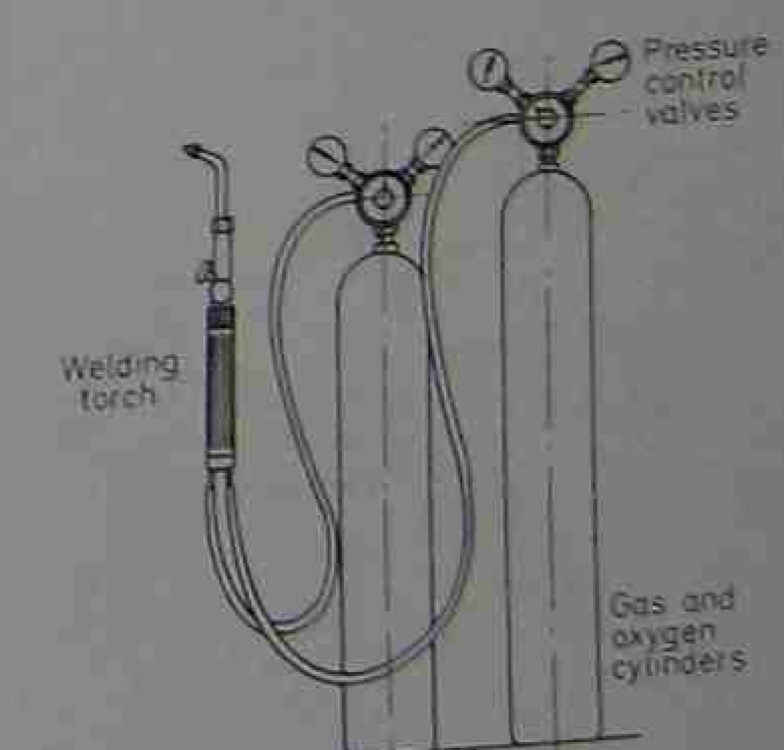


Figure 3.144 *Gas welding equipment*

FLAME CUTTING

Flame cutting is a method of producing shaped flat parts cut from steel plate, especially of large thickness. The torch may be hand-held or mounted in a welding machine in which a template of the required shape is followed by the cutting flame.

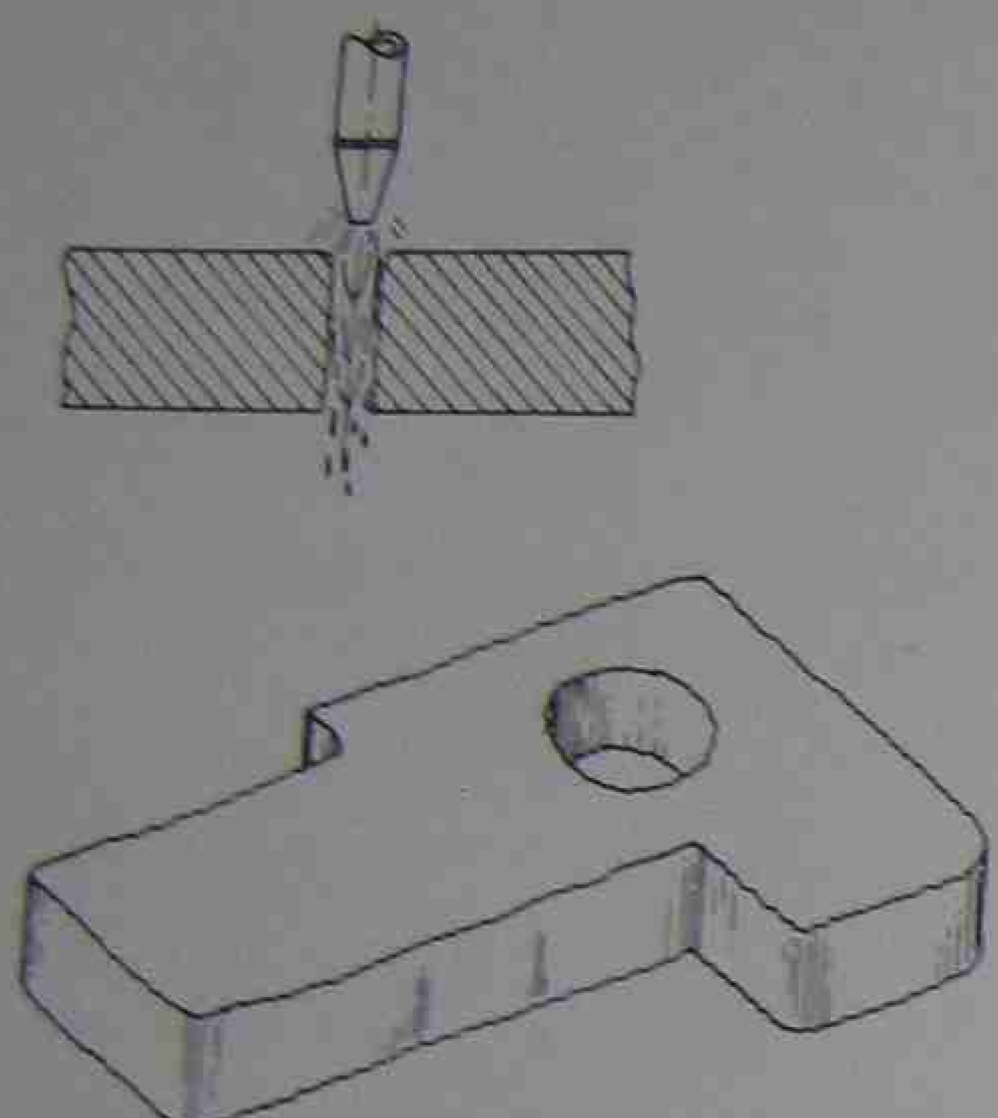


Figure 3.145 *Flame cutting*

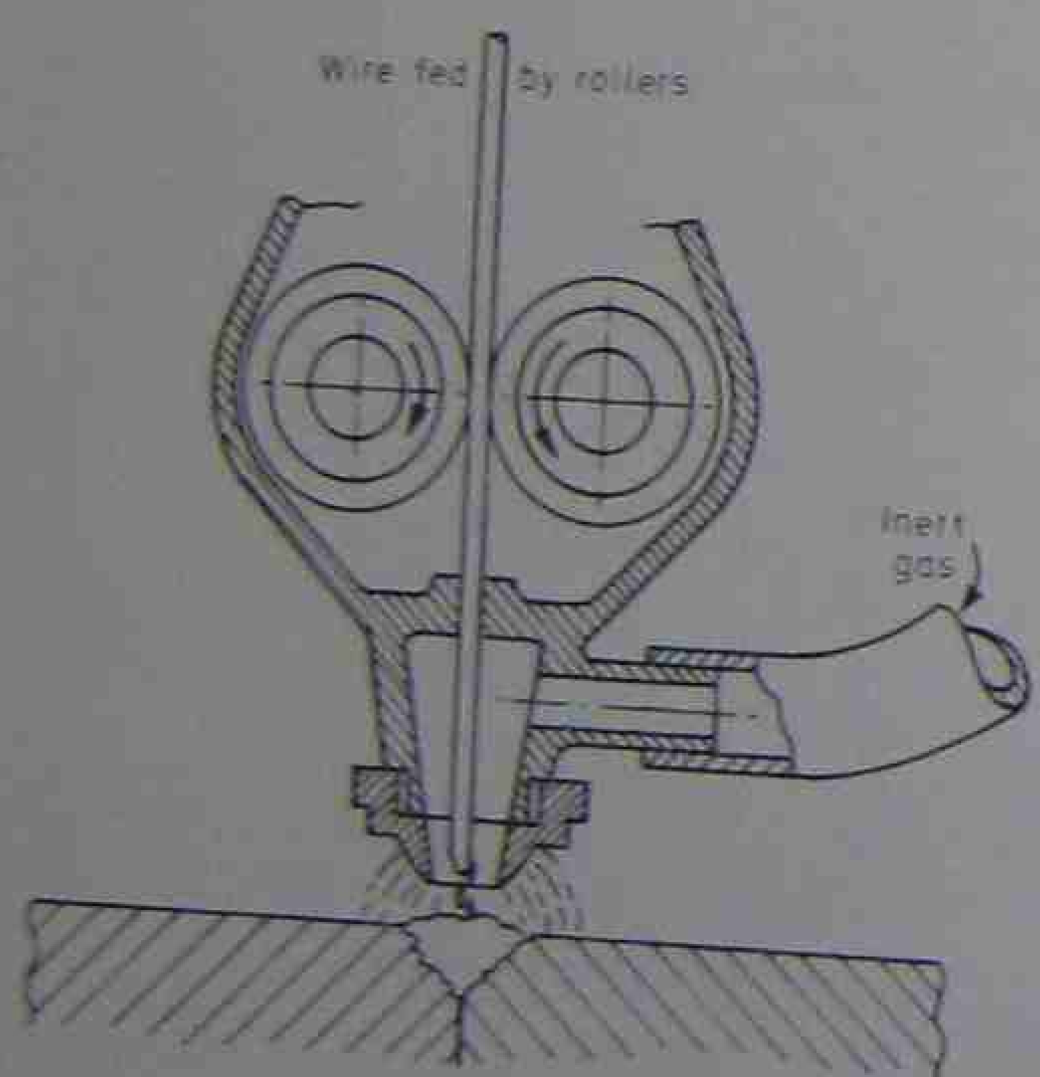


Figure 3.146 *Gas metal arc welding*

GAS SHIELDED METAL ARC WELDING

In this process, also called Metal Inert Gas (MIG), an inert gas such as argon is used as a flux for arc welding. The electrode is a continuously-fed consumeable wire.

FRICTION WELDING

Parts may be welded by the heat generated by mechanical friction. One of the parts is rotated in a chuck and the other part, which does not rotate, is brought into contact with it under pressure. When the resulting friction has produced sufficient heat to fuse the contacting surfaces, the non-rotating part is freed and allowed to rotate. On cooling, the parts are joined. This method is useful for joining dissimilar metals which are otherwise difficult to weld.

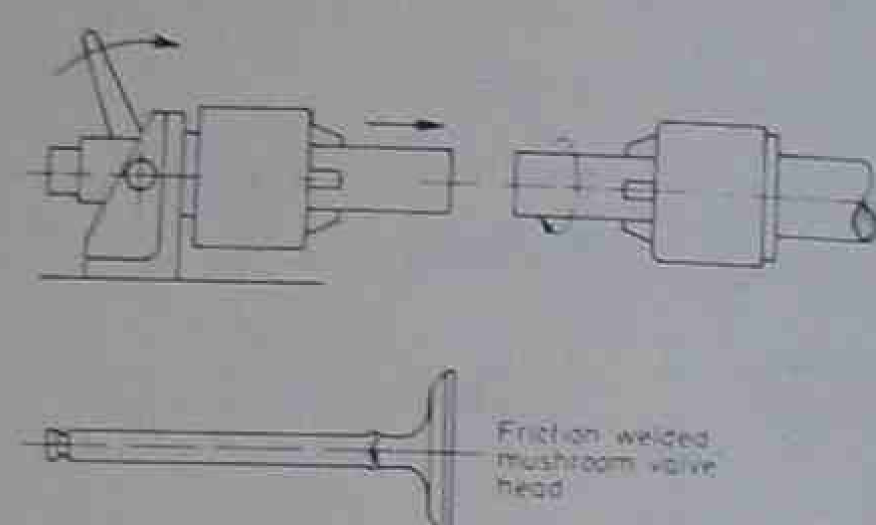


Figure 3.147 *Friction welding*

3.4 METAL-FORMING EQUIPMENT

CASTINGS

A casting is produced by pouring molten metal, or any other material, into a suitably shaped mould and allowing it to solidify. The term 'casting' also refers to the actual process.

Most castings are *sand castings* which are made in a mould using a special moulding sand. The mould is made in a *moulding box* divided horizontally into halves which are located with dowels and bolted together during casting. A wooden replica, or *pattern*, of the part to be cast is embedded in the sand to produce a cavity. Any holes required in the casting are produced by inserting *cores* into the mould. These cores are previously made in a *core box* out of baked sand. The molten metal is poured into the mould through runners until it fills the cavities and appears in risers. When the metal has set the sand is removed, the sticks of metal in the runners and risers are removed, and the casting is cleaned by chipping, grinding and sandblasting.

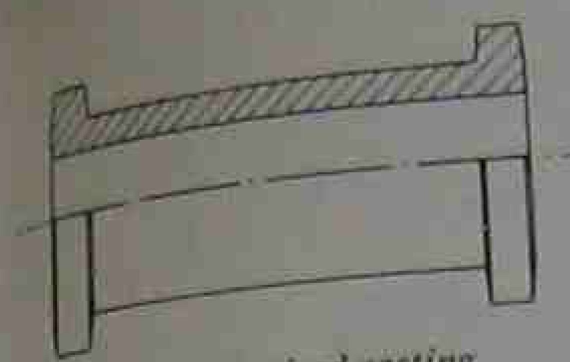


Figure 3.148 *Required casting*

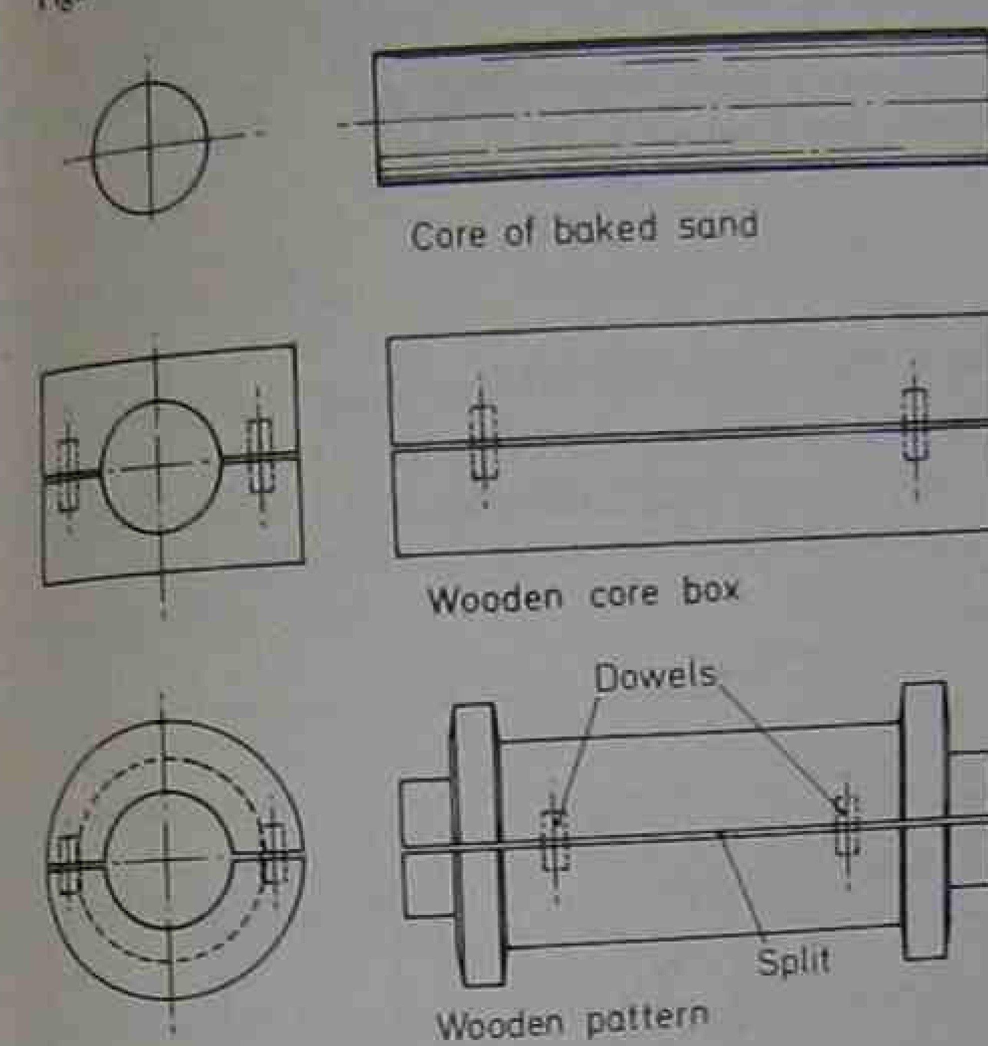


Figure 3.149 *Casting*

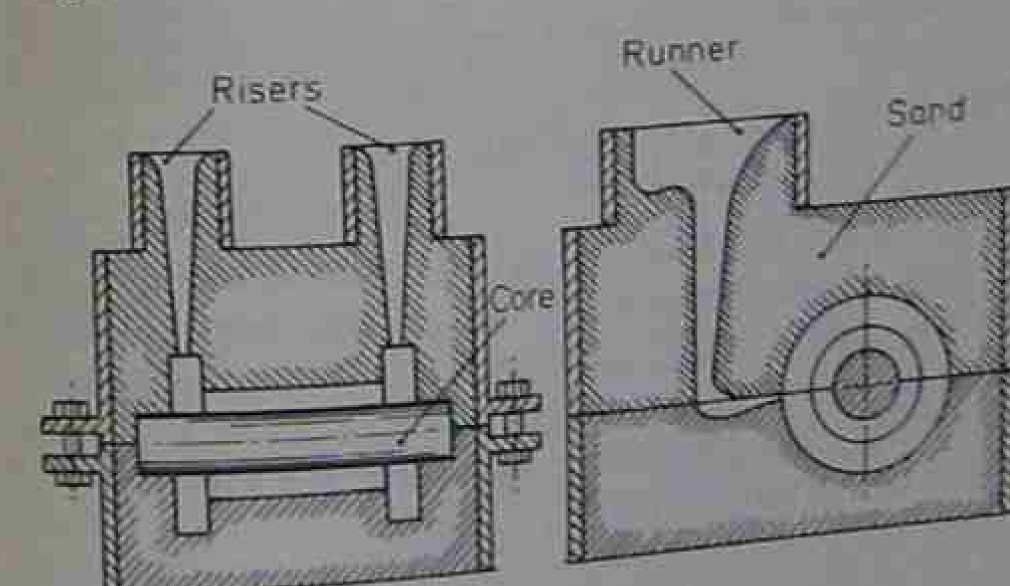


Figure 3.150 *Moulding box*

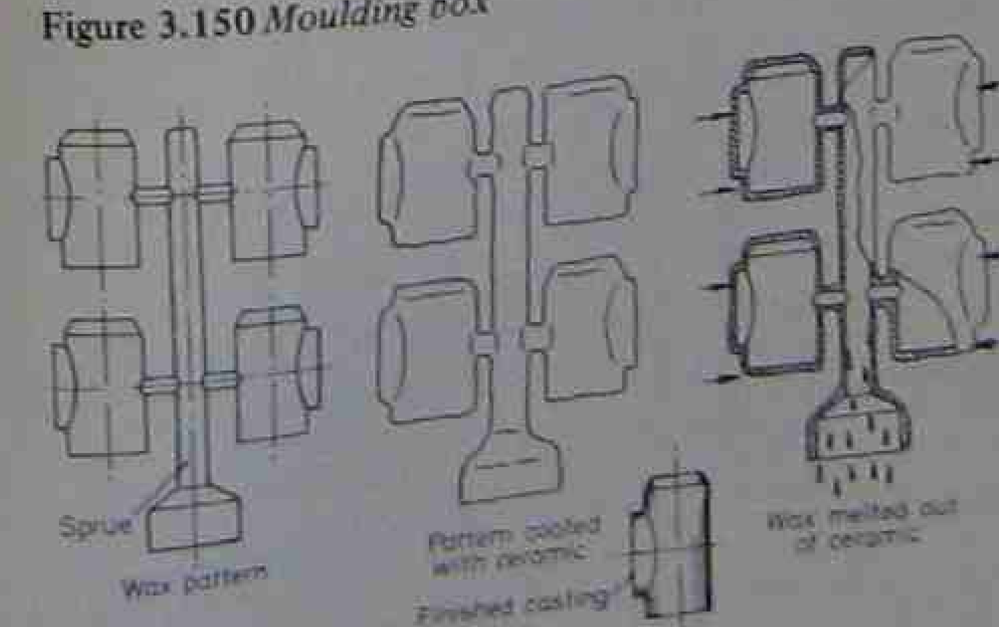


Figure 3.151 *Investment casting*

Investment casting (lost wax casting) A permanent metal mould is made and from it wax patterns are produced. These are coated with several layers of ceramic slurry, each layer being allowed to harden. The coated patterns are baked and the wax melts and runs away leaving a cavity which is filled to give a precision casting.

Die casting The mould is made of steel in several parts dowelled together. Molten metal is fed into the mould under gravity or pressure and when solid it is ejected by pins.

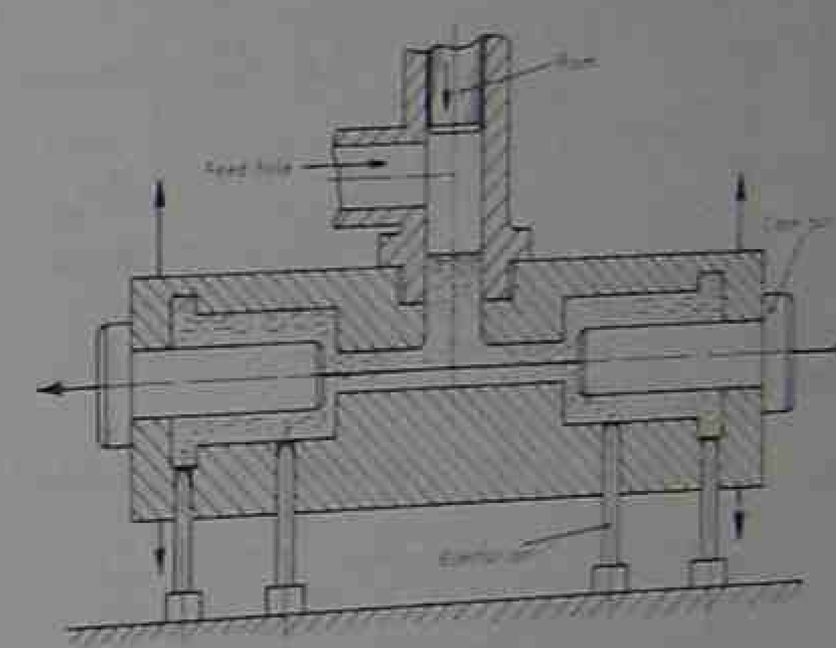


Figure 3.152 *Die casting*

Centrifugal casting Cylindrical and circular components, such as piston rings, cylinder liner pipes, gear wheels and locomotive wheels, may be cast in a rotating mould in which centrifugal pressure in the molten metal produces a fine-grain high-quality casting. Molten metal is poured into the mould while it is rotating and allowed to solidify before rotation ceases.

Wheel-shaped objects are cast with the axis vertical whereas pipes are cast with a horizontal axis.

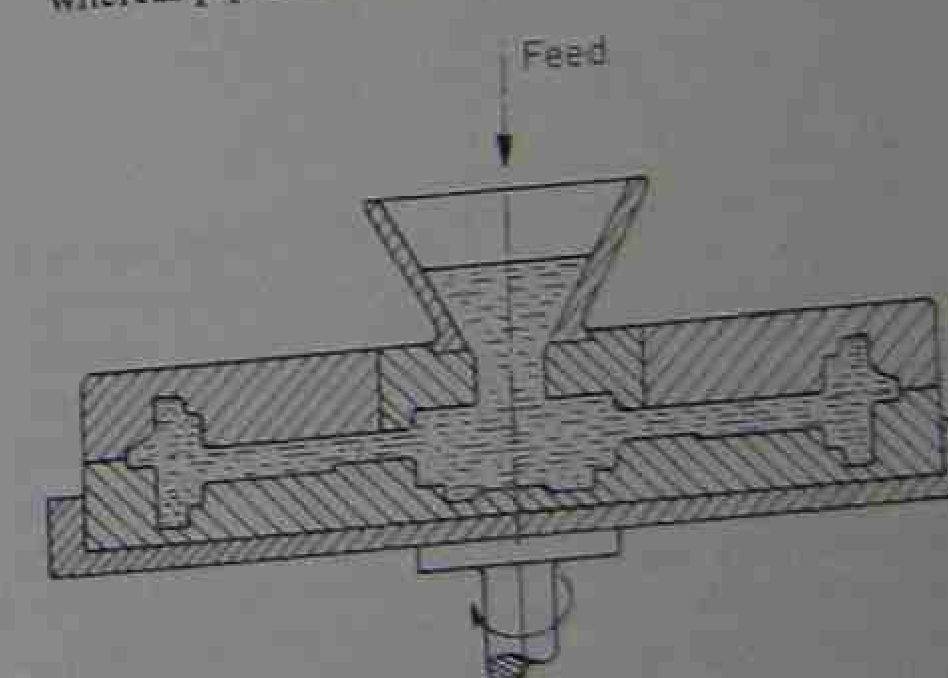


Figure 3.153 *Centrifugal casting – vertical axis*

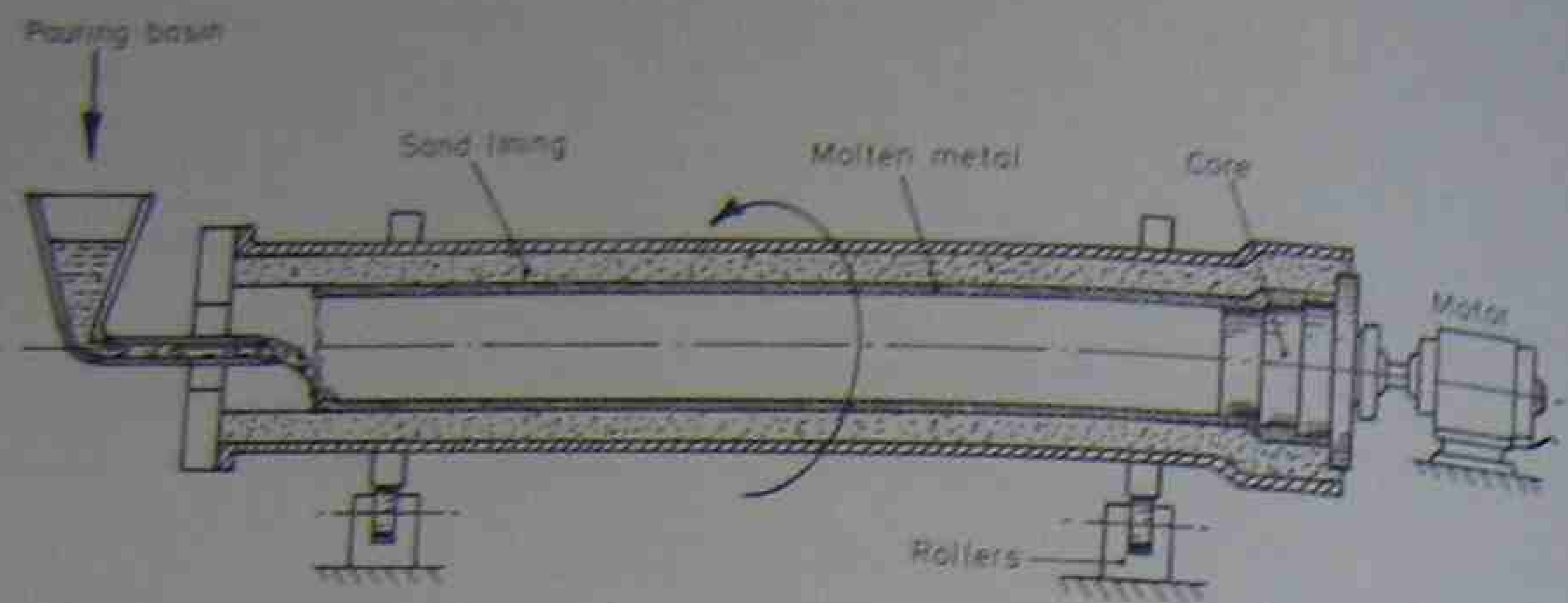


Figure 3.154 *Centrifugal casting of pipe – horizontal axis*

FORGING

Forging is the forming of metal parts by hammering, pressing or bending to the required shape, usually while the metal is at red heat.

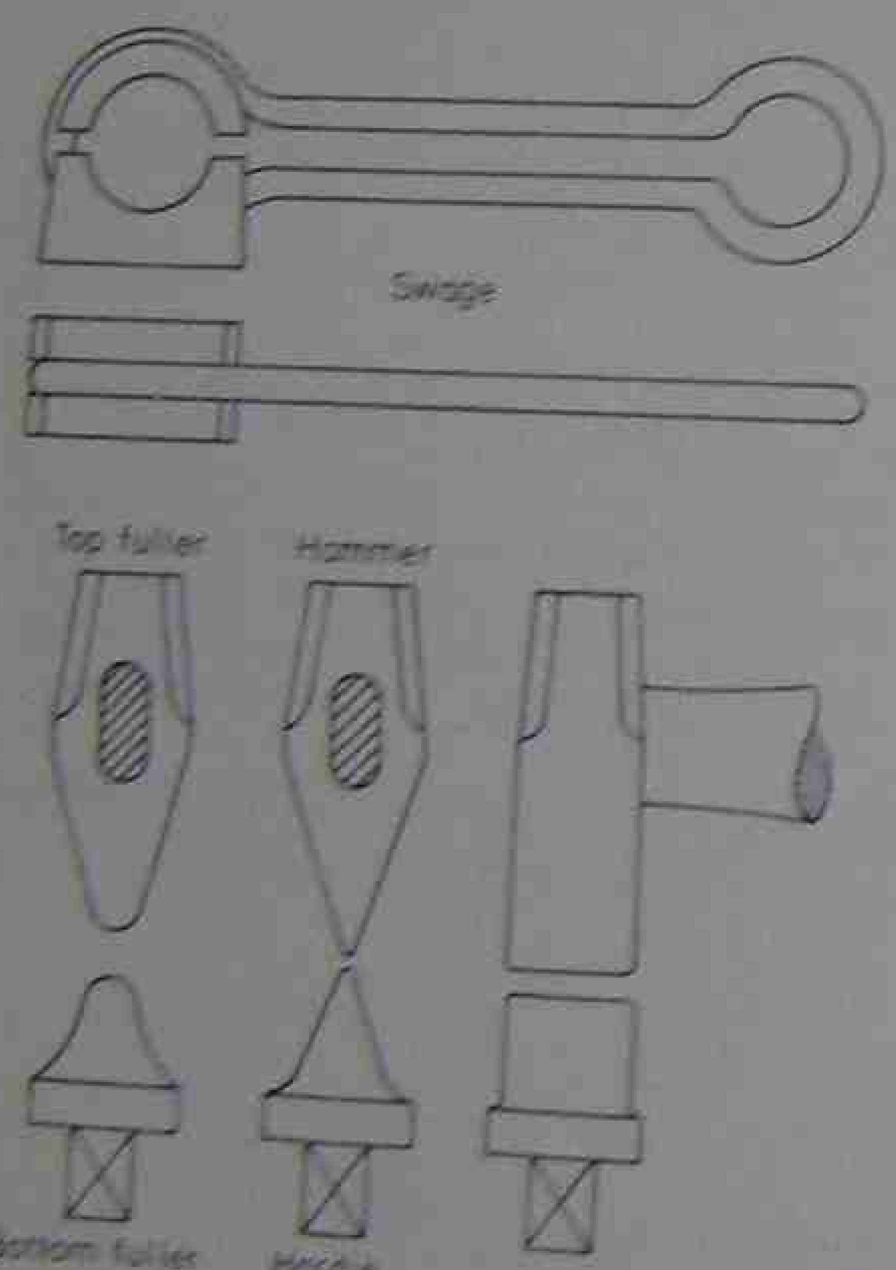


Figure 3.155 *Hand forging tools*

Hand forging This is the process used by a blacksmith where parts are formed on an *anvil* using special hammers, chisels and swages. The anvil has a hardened top face used for cutting on and a soft 'beak' for bending bars. In the top face is a square hole called a 'Hardie hole' used for holding various tools.

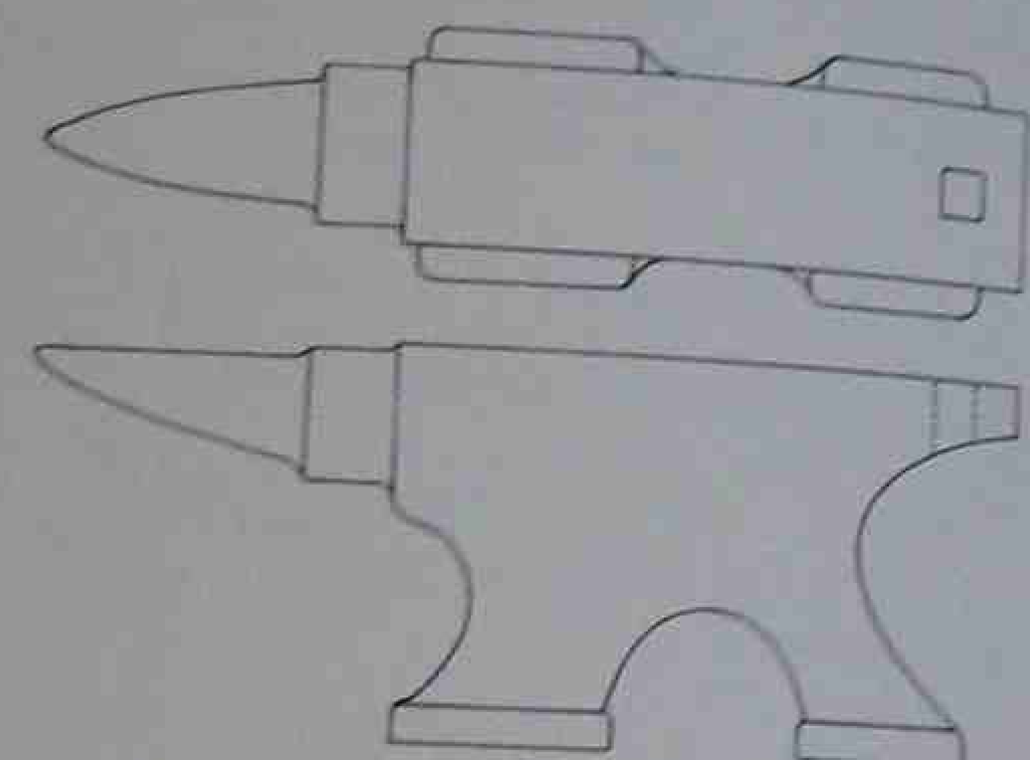


Figure 3.156 *Anvil*

Drop forging machine This is a machine used for forging in which pneumatic or hydraulic pressure is used to provide the force. Hot metal blanks are compressed between hard steel dies one of which is attached to the powered hammer and the other to the machine table.

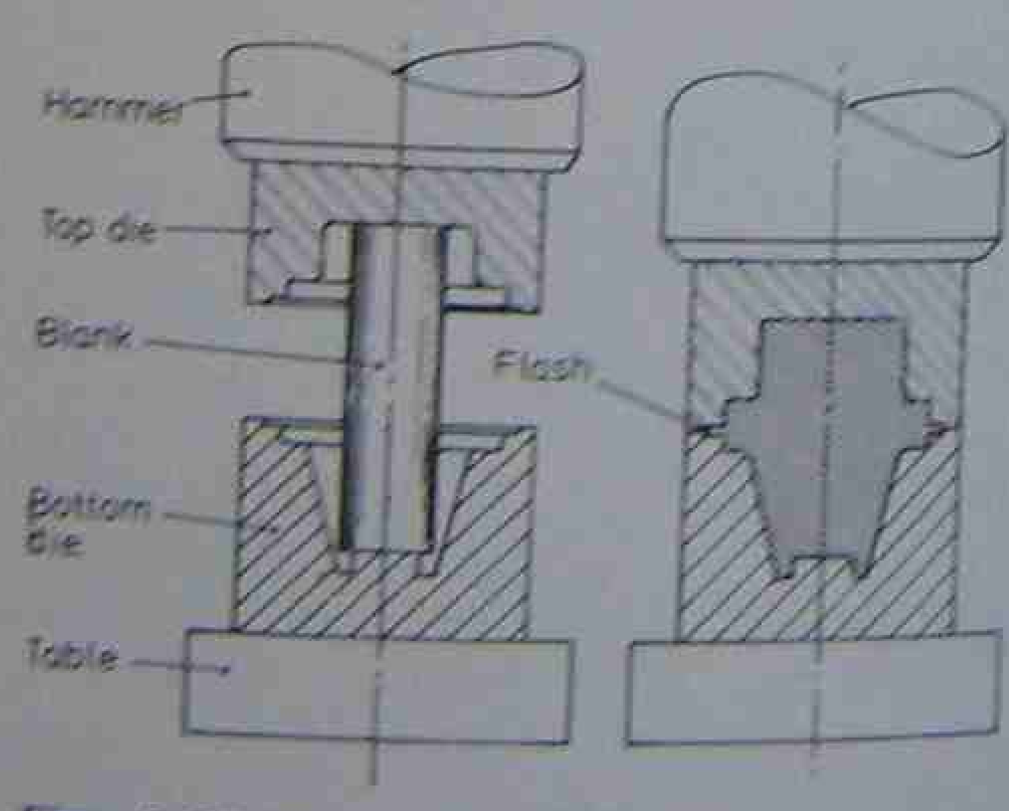


Figure 3.157 *Drop forging machine*

Figure 3.158 *Forging with flash removed*

ROLLING MILL

Red-hot ingots of steel, or other metals, are passed through successive pairs of rollers specially shaped to produce flat sheet, I, T, channel, angle or any other section bar. Final cold rolling is sometimes carried out to give a better finish.

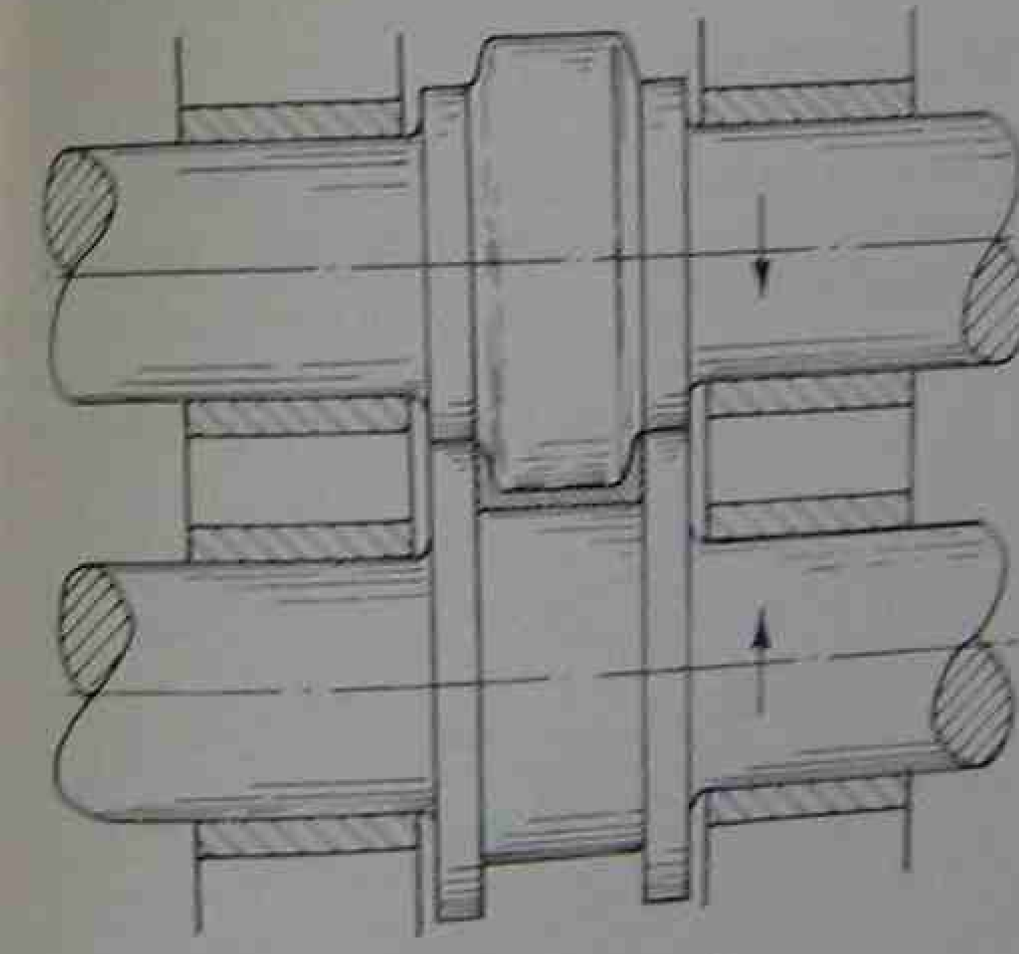


Figure 3.159 *Rolling mill (rolling channel)*

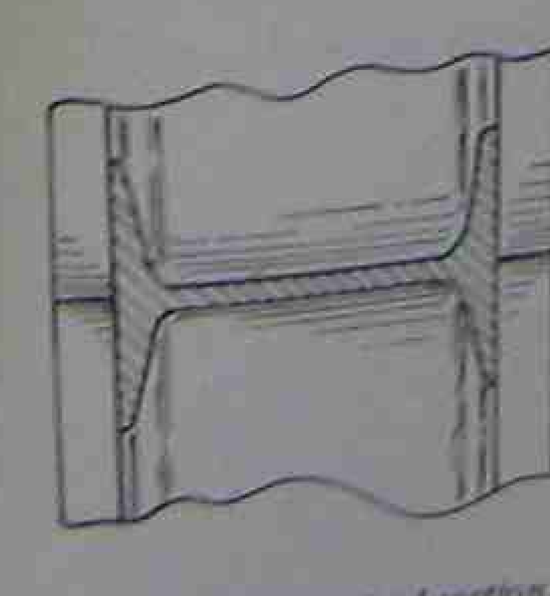


Figure 3.160 *Rolls for I section*

EXTRUSION

When metal is sufficiently soft, either at normal temperatures or when heated, it can be forced under pressure to flow through a die like toothpaste from a tube to form bars of any desired cross-section.

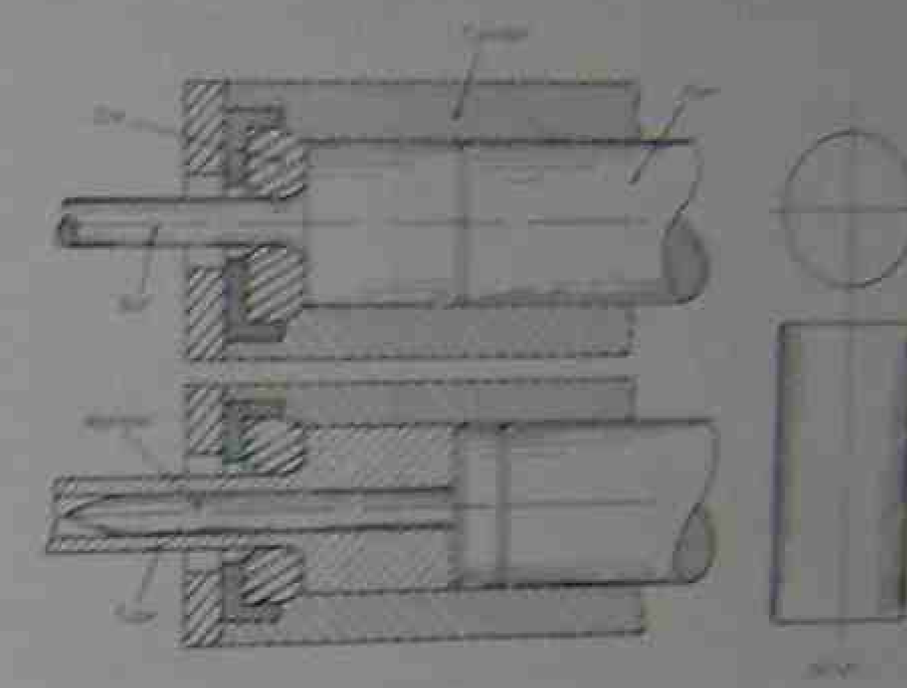


Figure 3.161 *Hot extrusion*

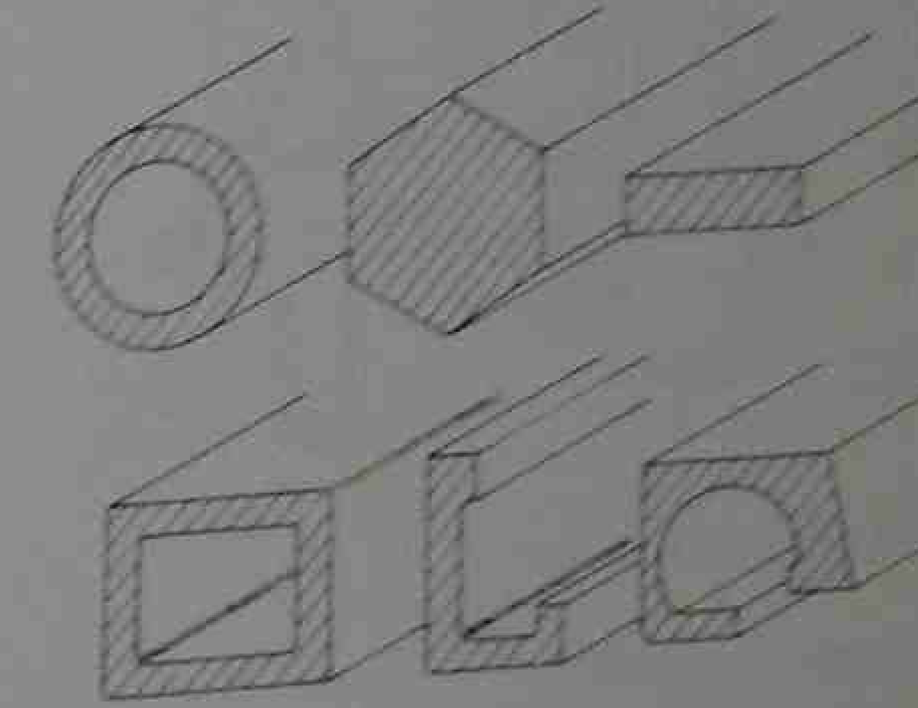


Figure 3.162 *Hot extruded sections*

Hot extrusion A piece of metal bar, or *billet*, is heated to the desired temperature and fed into a cylinder. It is then forced by a ram through a *die* of the correct shape to produce long lengths of bar which can be cut to the required length. Hollow sections can be made using a mandrel placed in the orifice.

Cold extrusion Soft metals such as copper, zinc and aluminium can be extruded when cold, and sometimes it is necessary to carry out the process in several stages.

Impact extrusion A metal which is plastic when cold may be formed by the impact of a high velocity punch to form a tube. A 'slug' of metal is given a single blow which 'splashes' it up the sides of the punch. This process is used for the manufacture of toothpaste tubes, battery cases, ignition coil cans, etc.

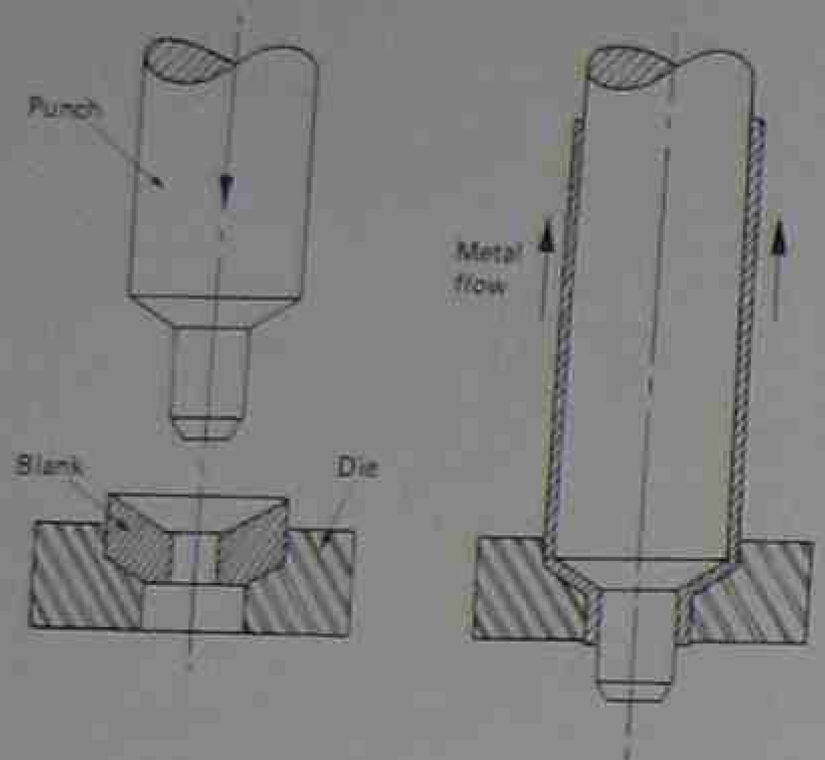


Figure 3.163 *Impact extrusion*

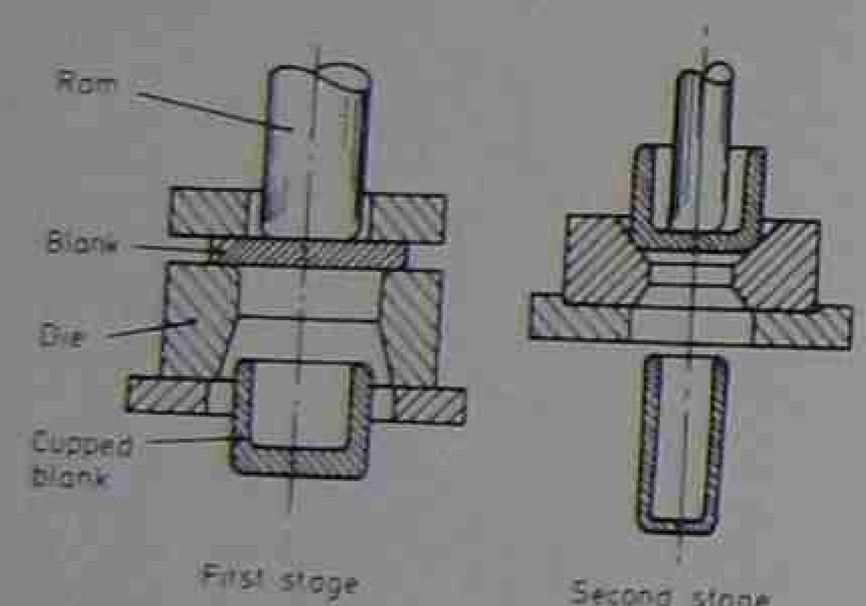


Figure 3.164 *Deep drawing*

DEEP DRAWING

Deep drawing is the forming of sheet or plate into box- and cup-shaped articles by pressing it with a shaped punch into a die. It involves considerable plastic deformation of the material. The process is used for cartridge cases, washing machine tubs and many electrical fittings.

PRESS

A press is used for a wide range of processes such as punching, piercing, blanking, notching, bending, drawing, folding, flanging, etc. It may be operated by means of a crank connected to a heavy flywheel or by hydraulic power.

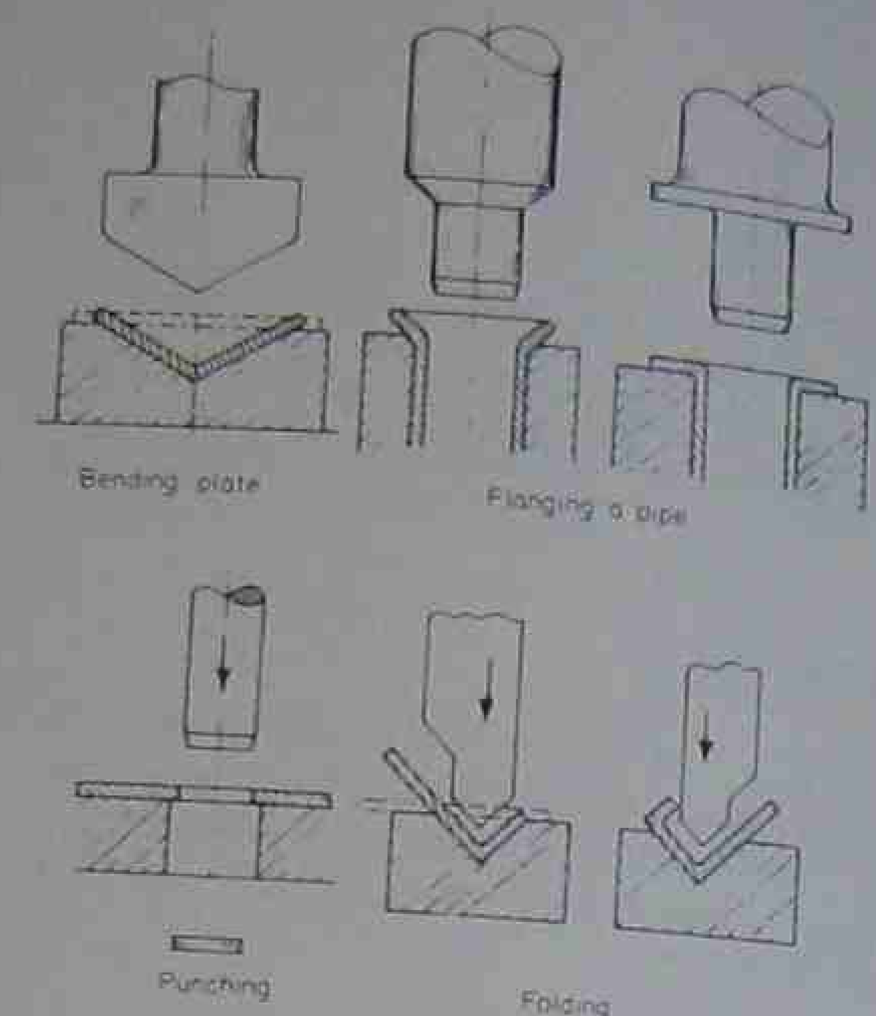


Figure 3.165 *Press work*

SHEARING MACHINE

A shearing machine consists of a fixed blade and a manually- or power-operated moving blade used for cutting sheet metal.

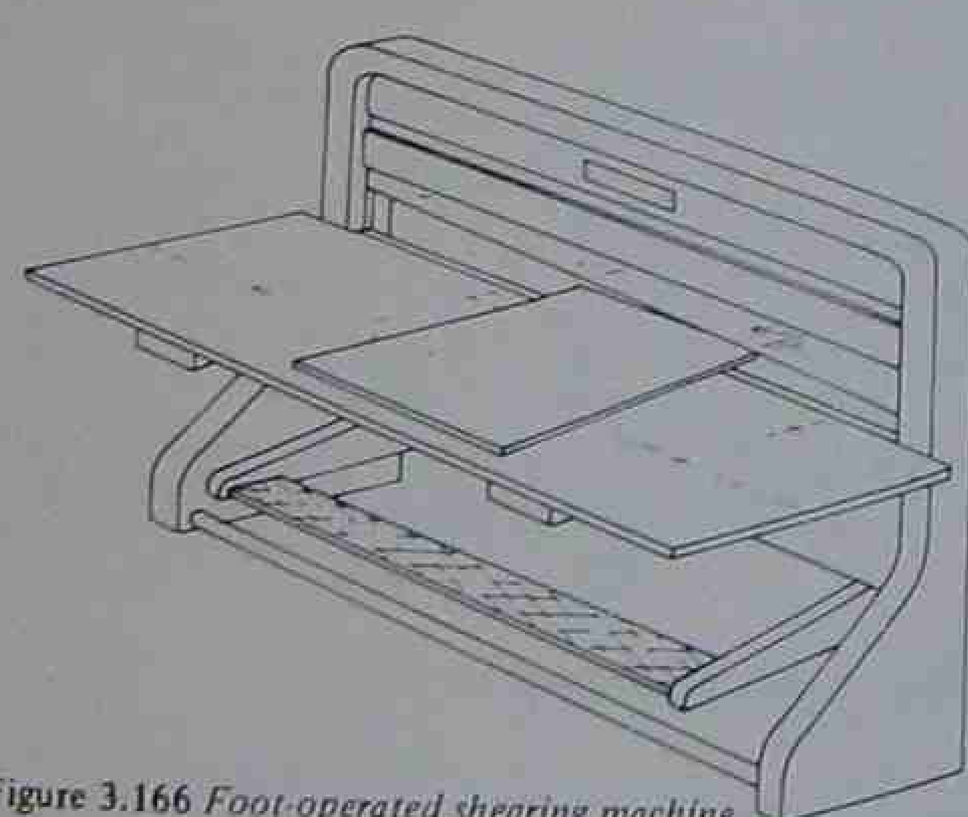


Figure 3.166 *Foot-operated shearing machine*

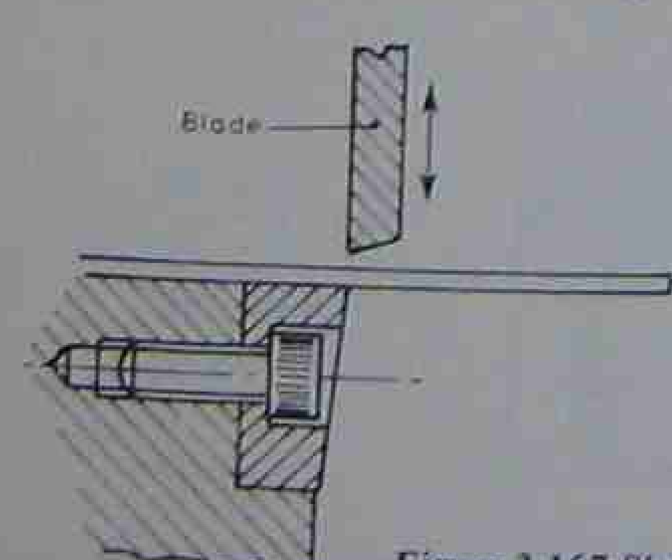


Figure 3.167 *Shearing operation*

WIRE DRAWING

A method for producing small diameter wire by drawing it while cold through successively smaller dies. Because of work hardening, the yield strength of the material of the wire increases at each drawing.

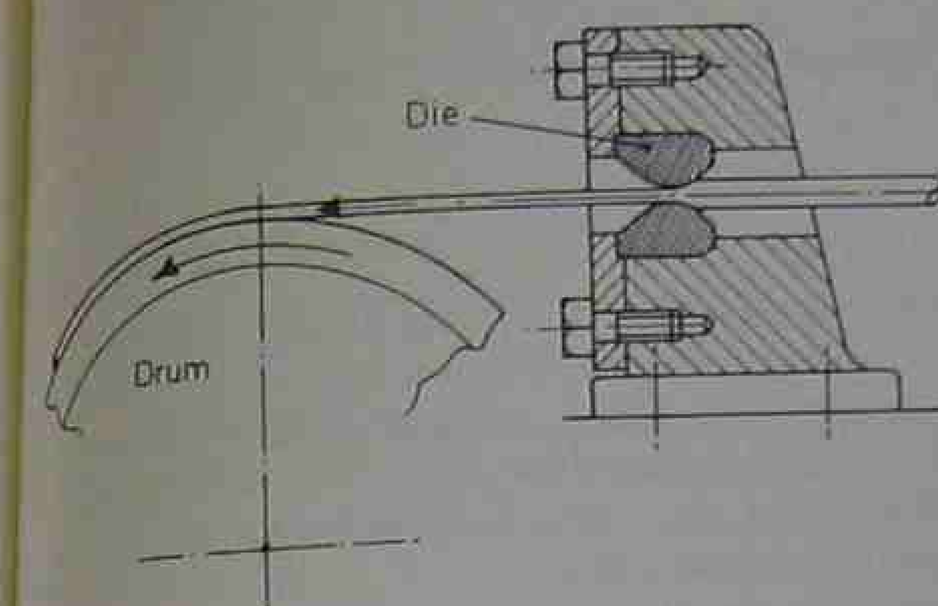


Figure 3.168 *Wire drawing*

SPINNING

Spinning is the forming of items out of sheet metal by rotating a thin metal disk at high speed, usually in a lathe, and pressing it into a cup or cone shape over a former using either a forming tool or a roller.

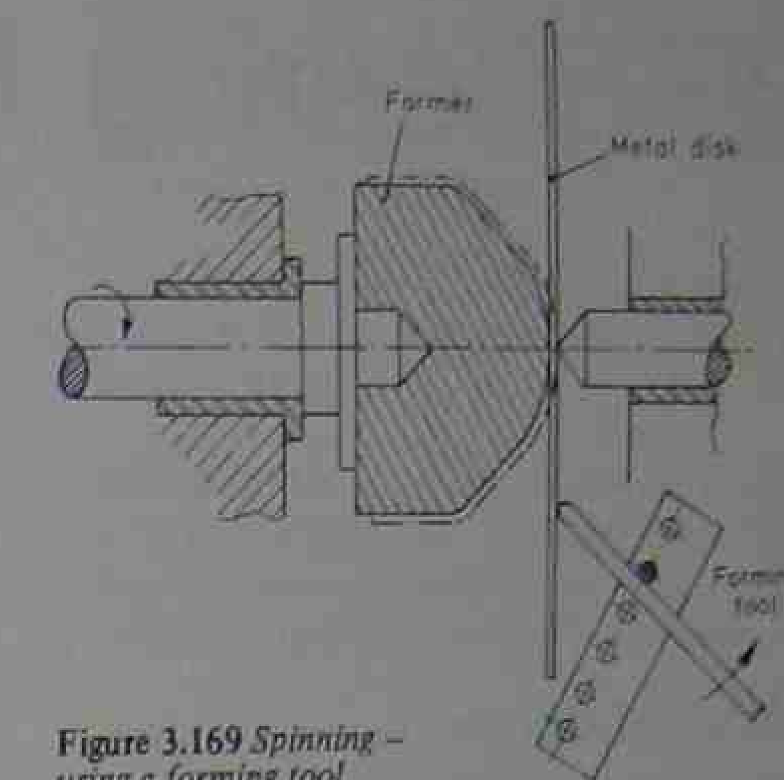


Figure 3.169 *Spinning – using a forming tool*

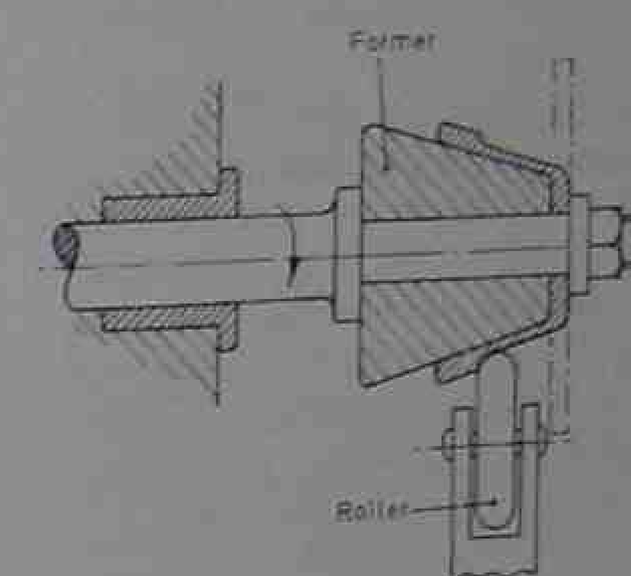


Figure 3.170 *Spinning – using a roller to form*

4. Engineering Materials

4.1 METALS

CAST IRON

Cast iron contains a considerable amount of carbon (2.4–4.5%) and is too brittle for 'working'. It is, however, very fluid when molten and is suitable for intricate castings.

Though quite strong in compression it is weak in tension and shear, but the strength can be increased considerably by the addition of alloying materials such as nickel, chromium and copper. These alloy cast irons can be used for gears, camshafts and crankshafts and are cheaper than steel.

SPHEROIDAL GRAPHITE (SG) IRON

In this iron (also called Nodular Iron) the graphite is in the form of small spheres resulting in increased ductility. The mechanical properties approach those of steel combined with good castability.

MALLEABLE CAST IRON

This is tougher and more shock-resistant than cast iron and is suitable for automobile parts, such as levers and pipe fittings.

There are three main types, *white heat* with superior castability, *black heat* with superior machineability, and *pearlitic* superior to the other two but difficult to produce.

STEEL

Steel is essentially an alloy of iron and iron carbide with small amounts of other elements in controlled quantities. A wide range of steels is available with properties varying considerably depending upon the alloying elements and subsequent heat treatment.

Carbon steel The amount of carbon determines the strength of steel. The main types are: dead mild (or low carbon), mild, medium carbon and high carbon.

Dead mild (low carbon) The carbon content is 0.07–0.15%. It is used for pipes, wire, nails, boiler plates, chains, etc., and it is worked when hot but does not machine easily.

Mild The most widely used of engineering materials whose carbon content of 0.15–0.25% allows it to be easily worked, machined and welded. Typical applications are ships' plates, forgings, nuts and bolts, gears and shafts.

Medium carbon A type which can be heat-treated to give greater strength, medium carbon steel is also easily machined. Its uses are machine parts, forgings, castings, springs and drop hammer dies.

High carbon This type contains 0.5–1.5% carbon which gives it great strength when heat-treated. At the lower carbon content it is used for screwdrivers, hammers, spanners, chisels, forging and pressing dies, while the highest carbon content is used for drills, lathe tools, hacksaws, ball bearings, taps and dies.

Alloy steels The addition of alloying elements to steel makes it more responsive to heat treatment, and this results in increased hardness, strength and toughness and greater resistance to corrosion. The main alloying elements are nickel, chromium, manganese, molybdenum, vanadium, tungsten, silicon and cobalt.

Nickel-chrome Nickel-chromium steels are among the most important alloy steels used in engineering. Heat treatment gives a wide range of properties with good resistance to shock and good ductility. They are used for high-tensile bolts, hardened gears, machine tools, etc.

Stainless A corrosion-resistant steel with about 12% chromium and other elements, used for turbine blades, and in sheet form for chemical and food containers.

Silver A bright-drawn high-carbon steel in the form of circular rod, containing manganese and chromium.

Tool High carbon steel to which manganese, tungsten and chromium have been added. When hardened it is used for dies, and to a limited extent for machine tools having been largely replaced by tungsten carbide inserts or brazed tips.

TUNGSTEN CARBIDE

An extremely hard material used for machine tool tips.

COPPER

A reddish, ductile metal of low strength but which is a very good conductor of heat and electricity. It is used for making pipes, electricity cables, gaskets and washers for fluid sealing, and is extensively alloyed with zinc, tin, aluminium, etc.

Copper nickel alloys There is a wide range of these alloys with amounts of nickel varying from 2–70%. Alloys with 40–45% nickel are used for resistance wire, e.g. Constantan, and they have a high resistivity and small resistance/temperature coefficient. Alloys containing about 70% nickel are known as *monel metal*. These are as strong as steel and have excellent resistance to corrosion by sea water, salt solutions and organic acids. They are suitable for steam turbine blades and condenser tubes.

Beryllium copper If 2% beryllium is added to copper the result is a very hard alloy as strong as tool steel. It is used for springs, bellows, bourdon tubes in pressure gauges and for non-sparking tools.

BRASS

This is the most widely used alloy of copper. It consists of 30–50% zinc and the remainder copper, to which a small amount of lead may be added to improve machinability. Brasses with a low zinc content are used for cold working in the production of condenser tubes, cartridge cases, gas and electric light fittings. Brasses with 39–46% zinc are easily hard worked and are used for extruded sections, pump parts, hydraulic fittings, nuts, bolts and screws.

BRONZE

'Bronze' refers to a copper-tin alloy which is used for electrical parts, hydraulic components, bearings and gears.

Gunmetal This is a bronze with 2% zinc added to improve the fluidity when casting and increase hardness. It is used extensively for castings, particularly those of a complicated form.

Lead bronze The addition of lead improves machinability in bronzes and improves the wear resistance of bronze bearings.

Phosphor bronze Phosphor bronze contains from 0.1–1.0% phosphorus. It is used for heavy-duty bearings, gears and non-ferrous springs, and is hard with good resistance to wear and corrosion.

Manganese bronze This is a bronze to which ferro-manganese has been added. It is used for very high strength castings.

ALUMINIUM

A light, ductile metal which is a good conductor of heat and electricity. It is used for electrical conductors and parts of switchgear, kitchen utensils, wrapping foil, window frames and as a base for many alloys.

Aluminium is extremely soft and ductile but is difficult to cast. Its properties can be greatly improved by adding small amounts of copper, silicon, manganese, magnesium, iron, zinc, nickel, bismuth and titanium. Aluminium alloys are classed as *wrought* and *casting* and may be *heat-treatable* or *non-heat-treatable*.

Silicon lowers the melting point and improves castability. Copper, magnesium, manganese, zinc, titanium and nickel increase the strength. Lead and bismuth increase machinability.

Duralumin This most important of aluminium alloys contains copper, manganese, magnesium and silicon. It is as strong as mild steel with a third of the weight and is used for forgings, stampings, bar, sheet and rivets.

Aluminium-copper-zinc alloys These have a low melting point and are easily cast. They are cheap and are used for gearboxes and crankcases.

Y alloy This is an aluminium alloy containing 4% copper, 2% nickel and 1.5% magnesium. It is used extensively for castings such as cylinder heads and pistons for IC engines.

MAGNESIUM ALLOYS

Magnesium is only two thirds the weight of aluminium, and its alloys are useful for producing lightweight forgings and castings.

NIMONIC ALLOYS

An important range of alloys composed mainly of nickel and chromium with small amounts of titanium and carbon. They are very strong up to temperatures of 900°C and are therefore suitable for gas-turbine and jet-engine applications.

ZINC

Zinc is used as a protective coating for sheet metal, wire, nails, etc. It is also used in alloys with aluminium, copper and magnesium. Zinc alloys are used for die castings.

CHROMIUM

Chromium is used in electroplating to provide a surface with a high polish and high corrosion-resistance. It is alloyed with steel to give extremely high strengths.

CADMIUM

A fairly expensive metal used for plating and in certain types of battery.

TITANIUM

An expensive metal used in the production of aircraft parts alloyed with aluminium, vanadium, manganese, tin, etc. It has high strength, low weight, good heat- and corrosion-resistance.

LEAD

Lead is a heavy, soft, ductile metal with little mechanical strength. Because of its good resistance to corrosion it is used for roofing, cable sheathing and lining chemical apparatus. It is used extensively in nuclear work for radiation shielding and is alloyed with other metals for bearing metals and solders.

WHITE METAL (BABBITT METAL)

The name given to a range of tin-based alloys containing copper and antimony and sometimes lead. The antimony is important in ensuring low friction in bearings. These alloys are expensive and used only in high performance applications.

FUSIBLE ALLOYS (LOW MELTING POINT ALLOYS)

Alloys of bismuth, lead and tin have very low melting points and are used for solders and fusible plugs.

4.2 PLASTICS AND OTHER NON-METALLIC MATERIALS

PLASTICS

The term 'plastics' covers a wide range of man-made materials which can be moulded to the required shape by the application of heat and pressure.

There are two main types of plastic, *thermoplastic* and *thermosetting*. Thermoplastic materials become soft and pliable when heated and can be moulded into the required shape. They can be reheated and remoulded repeatedly. Thermosetting plastics suffer a chemical change when subjected to heat and pressure and they cannot be softened by reheating.

PVC (POLYVINYL CHLORIDE)

Rigid PVC is a thermoplastic material used for pipes and ducts, plasticised PVC is used for cable covering, mouldings, fabric and flexible sheet. It is flame- and water-resistant.

PERSPEX (ACRYLIC RESIN)

Perspex, a thermoplastic material, is one of the most common acrylic resins. It can be made very clear or coloured and has good optical properties. It is available in sheet, bars, tubes, etc.

POLYSTYRENE

A thermoplastic which can be moulded into complicated shapes with fine detail. It is an excellent electrical insulator but is not weather resistant.

POLYTHENE (POLYETHYLENE)

A thermoplastic polymer with good electrical properties, especially at high radio frequencies, which is suitable for use as weatherproof sheeting.

POLYPROPYLENE

This is similar to polythene but has greater heat resistance.

CELLULOSE PLASTICS

These include celluloid used for drawing office equipment, Cellophane used in film form for packaging, and rayon used for clothing, cements, petrol pipes and photographic film.

NYLON

A plastic used in fibre form for fabrics and also moulded into parts requiring light weight, low friction and flexibility. It has good self-lubricating properties when used for bearings.

EXPANDED PLASTICS

Polythene, polystyrene, polyurethane and PVC can be foamed by the introduction of gas bubbles during manufacture. These foam plastics are extremely light and are used for heat insulation, packaging and model making.

BAKELITE

A hard phenolic thermosetting plastic which is very cheap and is used for electrical insulated parts, tool handles, bonding material for grinding wheels, etc.

TUFNOL

A laminated plastic, i.e., one containing fabric, paper, etc. Available in sheet, rod and tube, and used widely in engineering for bearings, gears, pulleys and insulated parts.

PTFE (POLYTETRAFLUORETHYLENE)

Marketed under the names Fluon and Teflon this is a plastic with high chemical inertness and heat resistance used for extremely low-friction bearings and non-stick surfaces.

EPOXY RESINS

Polymers such as Araldite are widely used for structural plastics, adhesives and for encapsulation.

GLASS FIBRE

Fine threads of glass are used as a reinforcement in synthetic resin mouldings for boat hulls, pump impellers, roof lights, etc., and also as a structural material.

CARBON

Amorphous carbon is used for fluid seals in pumps and steam turbines. *Carbon fibres* are extremely strong and can be used to increase greatly the strength of epoxy resin mouldings and metal components.

DIAMOND

A form of carbon used for the tips of oil-well drills etc., and as a dust on dressing tools for grinding wheels.

RUBBER

Natural latex combined with sulphur, carbon black and other materials is widely used in engineering where flexibility is required. Examples of its use are tyres, flexible couplings and anti-vibration mountings.

SYNTHETIC RUBBER

A wide range of synthetic rubbers, such as Neoprene, Buna and butyl, are used for flexible mountings and couplings and oil seals, etc. In their properties they are often superior to natural rubber.

CERAMICS

This is a general name for all non-metallic, non-organic materials but usually it refers to materials which will withstand very high temperatures. They are extremely hard and also wear- and corrosion-resistant and can be moulded into a variety of shapes before firing. They are used for burner nozzles, gas-turbine parts and electrical insulation.

ASBESTOS

A natural mineral material used for fireproofing. It can be woven into cloth or made into board. Sindanyo is the name of a form of asbestos board used for low voltage insulation.

PORTLAND CEMENT

A material made by burning a mixture of clay and lime. Mixing it with sand, aggregate and water produces concrete, and this is usually reinforced with steel bars to make structural columns and beams.

ABRASIVES

Materials such as carborundum and emery are used for making grinding wheels and sharpening stones.

WOOD

Wood is used in engineering in its natural form, and also in plywood, chipboard and blockboard.

Index

Abrasives, 104, 121
Absorption brake (dynamometer), 62
Acetylene gas (welding), 111
Acme power transmission thread, 14
Acrylic resin (Perspex), 120
Active flux, 108
Addendum/circle (gear), 19
Adjustable spanner (wrench), 94
Adjuster (brake), 61
Air bearing lubrication, 46
Air bellows spring, 18
Air (gas) bearings, 44
Air (pneumatic) spring, 18
Allen key (hexagon socket wrench), 3, 94
Alloy cast iron, 118
Alloy elements (steel), 118
Alloy metals, 118–120
Aluminium (aluminum), 119
Aluminium
  alloys, 119
  bar/tube, 59
  -copper-zinc alloys, 119
  gasket, 28
  oxide (abrasive), 104
  rivet, 10
  sealing washer, 28
Anchor bolt (Rawlbolt), 4
Angle of V (screw thread), 13–14
Angle milling, 97
Angle section steel bar, 58, 115
Angular contact bearing, 44
Angular tooth milling cutter, 97
Anti-friction bearings, 44
Anvils, 89, 114
Arbor, 96–97
Area (bottom of screw thread), 1, 13
Arc welding/plate/rod, 109
Argon gas (welding), 111
Asbestos gland packing, 24
Asbestos (Sindanyo), 121
Attachments, socket spanner, 94
Auto. brake, 61
Auto. disk brake calliper, 61
Auto. dynamo porous bearing, 43
Auto. friction clutch, 63
Auto-type clutch plate (disk), 63
Auto-type drum brake, 61
Auto-type poppet (mushroom) valve, 37
Auto-type universal joint, 69
Axial (face) cam, 81

BA/BSF/BSW bolt/screw threads, 1, 13
Babbitt (white) metal, 120
Bakelite, 121
Ball-bearing components, 44–45
Ball-bearing power screw, 15
Ball peen hammer, 86
Ball thrust bearing, 46
Ball-type non-return valve, 36
Ball-types of bearing, 44–46
Band/rope brake (dynamometer), 62–63
Bandsaws (power), 107
Band-type hose clip, 12
Bar, boring, 106
Bar handle, 94
Barrel (micrometer), 89
Barrel nuts, 7
Bar solder, 108
Bars/sections, 57–59, 115
Base circle (cam), 79
Bead chain/drive, 74
Beak (anvil), 114
Bearing lubrication, 46–47
Bearing materials, 44
Bearings/components, 42–47
Bed/bedplate (design feature), 47
Bed (horizontal borer), 106
Bed (lathe), 47, 102
Bell crank lever, 74
Bell crank lever (double block brake), 60
Belleville washer (diaphragm spring), 18
Bellows flexible pipe joint, 32
Bellows joint, hinged, 32
Bellows seal, carbon faced, 27
Belt conveyor, 72
Belt drives/pulleys, 70–72
Belt materials, 70–71
Bench drilling machine, 98
Bench grinder, 106
Bench/pedestal grinders, 106
Bench vice (vise), 92
Bending (presswork), 116
Bends, duct/pipe, 29–30, 32–34
Beryllium copper, 119
Bevel gears, 20–21
Billet (extrusion), 115
Bit, soldering iron, 107–108
Blacksmiths' anvil, 114
Blacksmiths' cold chisel, 83
Black steel bar, 58
Black steel bolts, 1

Blanking (presswork), 116
Blanks, 114, 116
Blind drilled hole, 52
Block-and-tackle, 72–73
Body (design feature), 47
Bolt dimensions, 1, 3
Bolted/flanged pipe joint, 30, 34
Bolts/bolt heads, 1–4
Bolt hole clearance, 2, 4
Bolt load/strength, 1–2
Bolt threads, 1, 13–15
Bolt tightening torque, 2
Bonded rubber flexible shaft coupling, 67
Bonded rubber spring, 18
Borax (brazing flux), 108
Bored hole, 52
Boring, 102, 106
Boring bar, 106
Boring lathe tool, 103
Boring machines, 106
Boring mill, 106
Boring tool, 103
Boss (design feature), 48
Bottom fuller (forging), 114
Box, core (casting), 113
Box/casing (design feature), 47–48
Box spanner (wrench), 93
Boxwood-head hammer, 86
Brace, 94
Bracket (design feature), 48
Brake components, 60–62, 65
Brakes, 60–63
Branch (ducting), 34
Brass, 119
Brasses (bearings), 42
Brass pipe joints/fittings, 31–32
Brazed cone pipe coupling, 31
Brazing (hard soldering), 108–109
Brazing torch, 108
Bright bar, 58
Bright bolt, 1
Bright steel bar, 58
Broach, 53
Broad-nosed facing tool, 103
Bronzes, 119
Brush, rotary wire, 105
Buffing (polishing) mop, 105
Buna, synthetic rubber, 121
Bush (design feature), 48
Bush (sleeve), bearing, 42, 48
Butane gas (welding), 111
Butterfly valve, 37
Butt joints, 10, 109–110
Button head socket bolt/screw, 3
Buttress power transmission screw thread, 14
Butyl, synthetic rubber, 121

C clamp, 91
C spanner (wrench), 94
Cable clip, 12
Cadmium/plating, 120
Cage, ball bearing, 44
Callipers, 61, 88–90
Cam details, 79
Cam followers, 80
Cams, 79–81
Camshaft, 39
Cantilever leaf spring, 17
Canvas belt, 70
Cap (crown/dome) nut, 6
Cap (design feature), 48
Cap head socket bolt/screw, 3
Capillary pipe fitting, 30
Cap, pipe fitting, 29
Captive nut, 7
Carbon, 121
Carbon face seals, 27
Carbon fibres/for sealing, 121
Carbon seals, 26–27
Carbon segmental seal, 26
Carbon steels, 118
Carborundum (abrasive), 121
Carriage (coach) bolt, 2–3
Carriage (laminated leaf) spring, 17–18
Casing/box (design feature), 47–48
Casting, 48, 112–114
Casting (design feature), 48
Casting dies, 113
Casting, lost wax (investment), 113
Casting methods, 112–114
Cast iron, 118
Castle nut, 7
Cast pulley, flat belt, 70
Cast steel gear, 23
Catch plate (lathe), 104
Cellophane, celluloid, cellulose, 120
Cement, 121
Centre distance (gear), 19
Centre/drill (lathe), 104
Centre (engine) lathe, 102–104
Centre finder (combination set), 88
Centre holes (shaft), 104
Centre punch, 84
Centres, turning between (lathe), 104
Centrifugal brake, 61
Centrifugal casting, 113–114
Centrifugal clutch, 65
Centrifugal dead-weight governor, 76–77
Ceramics, 121
Chain flexible shaft coupling, 69
Chain pulley, 74
Chains/chain drives, 73–74
Chain sprocket, 73–74
Chamfer (design feature), 48–49
Chamfer (shaft), 38
Channel bar/section, 58, 115
Check (non-return) valves, 35–36
Cheese-head bolt, 3–4
Cheese-head screw, 5
Chequer plate, 58
Chisels (sets), 83
Chromium (alloy element), 120
Chromium plating, 120
Chucks, 84, 102–104
Chuck with key, 99
Circlips (retaining rings), 13
Circlip groove (shaft), 39
Circular arc cam, 81
Circular pitch (gear), 19
Clamps, 91, 109
Clapper box (shaper), 100
Claw (dog) clutch, 64
Claw shaft coupling, 66

Clearance (bearings), 42
Clearance (bolt hole), 2, 4
Clearance (gears), 19
Clips, 12–13
Clutch components, 63–65
Clutches, 63–65
Coach (carriage) bolts, 2–3
Cobalt (alloy element), 118
Cold chisel, 83
Cold extrusion, 115
Cold-rolled steel sections, 58
Cold rolling, 115
Collars (shaft), 39, 43
Collet chuck (collet), 103
Column (drilling machines), 98–99
Column (milling machine), 95
Column (pillar) (design feature), 49, 54
Combination set, 88
Compression coupling (shaft), 66
Compression pipe couplings, 31
Compression piston ring, 27
Compression springs, 15, 17–18
Compression stop (spring lock) nut, 7
Concrete, 121
Cone clutch, 64
Coned pulley, 72
Conical head rivet, 10
Conical (upholstery) spring, 17
Connecting rod (crank-slider), 75
Constant acceleration/deceleration cam, 80
Constantan, 119
Constant-velocity cam, 80
Constant-velocity universal joint, 69
Contour grinding wheels, 104
Contour milling/cutter, 97
Conveyor belt, 72
Copper, 118–119
Copper-asbestos gasket, 28
Copper (capillary) pipe fittings, 30
Copper/fibre inserts (mallet), 86
Copper nickel alloys, 119
Copper rivet, 10
Copper sealing washer, 28
Core, core box/pins (die casting), 113
Cork gasket, 28
Corrugated sheet steel, 57
Cotter, 12, 42
Cottered joint (rods), 42
Cotter (split) pin, 12
Cotton belt, 70
Cotton gland packing, 24
Counterbored hole, 52
Counterboring, 100
Countersinking, 52, 85
Countersinking tool, 85
Countersunk head (flat) screw, 5
Countersunk hole, 52
Countersunk rivet, 10
Couplings (pipe fittings), 29–32
Cover/coverplate (design feature), 49
Crankpin, 39
Cranks, 39, 75
Crankshaft, 39–40
Crank-slider mechanism, 75
Crest (screw thread), 13
Crosscut cold chisel, 83
Crossed belt drive, 70
Crosshead, 76
Cross-head (Phillips) screw, 5
Cross-head (Phillips) screwdriver, 93
Cross peen hammer, 86
Crown (cap/dome) nut, 6
Cup seal, 25
Cutters, milling machine, 95–98
Cutters, wire, 92
Cylinder (design feature), 49
Cylinder (extrusion), 115
Cylindrical grinder, 105

Dashpot, 107
Dead mild (low) carbon steel, 118
Dead-weight safety valve, 35
Dedendum/circle (gear), 19
Deep drawing/dies, 116
Deep groove bearings, 45
Dial gauge (indicator), 90
Diametral pitch (gear), 19
Diamond (abrasive), 104
Diamond-pointed cold chisel, 83
Diamond tool tips, 121
Diaphragm spring, 18
Die casting, 113
Die casting core/ejector pins, 113
Dies, 85, 113–117
Die stock, 85
Disk brake, 60
Disk clutch, 63–64
Disk-type flexible shaft coupling, 67
Distance piece (spacer) (design feature), 49
Dividers (spring), 88–89
Dog (claw) clutch, 64
Dome (cap/crown) nut, 6
Double block brake, 60
Double butt weld, 110
Double cone pipe joint, 31
Double helical gear, 20
Double Michell thrust bearing, 44
Double reduction gear train, 23
Double riveted butt joint, 10
Double row ball bearings, 45
Dovetail milling/cutter, 97
Dowel/pin, 11
Drawing (presswork), 116
Drawing, wire, 117
Drifts, 84
Drill chuck/key, 99
Drilled holes, 52
Drilling head, 99
Drilling (lathe), 102
Drilling machine operations, 100
Drilling machines (drill presses), 84, 98–99
Drills, 84, 104
Drive chains, 73–74
Drive screw, 5–6
Driving dog/pin (lathe), 104
Drop forging machine/dies/blank, 114
Drum brake, 61
Drum, wire drawing, 117
Dry fluid clutch, 65
Duct/pipe materials/fittings, 29–34
Duplex lip seal, 25
Duralumin sheet, 119
Dwell (cam), 79
Dynamometer (absorption brake), 62

Earthing clamp (welding), 109
Eccentric, 75
Eccentric circle cam, 80
Eccentric-crank and slider mechanism, 75
Efficiency of screw threads, 15
Effort on levers, 74
Ejector pin (die casting), 113
Elastic stop (Nyloc) nut, 8
Elbow pipe fittings, 29–32
Electrical contact (cantilever leaf) spring, 18
Electric power hand drill, 84
Electric soldering iron, 108
Electrode/holder (welding), 109
Electromagnetic friction clutch, 65
Elliptical cross-section lever, 74
Elongated hole, 52
Emery (abrasive), 121
Endmilling/cutter, 97
Engine (centre) lathe, 102–104
Engineers' rules, 86–87
Engineers' vice (vise), 92
Epicyclic gear, 22
Epoxy resins, 121
Ermeto pipe joint, 31
Expanded plastics, 121
Expanded polystyrene, 121
Expansion joints/bends, 32
Explosive rivet, 11
Extension bar (socket spanner), 94
External circlip, 13
External involute shaft splines, 40
Externally serrated (tooth) lock washer, 9
External screw-cutting lathe tool, 103
Extruded sections, 115
Extrusion billet/cylinder/dies/mandrel, 115
Extrusion, cold/hot, 115
e bolt, 4, 49
e (design feature), 49
e, wire, 49

bric belt, 70
e (axial) cam, 81
e (design feature), 49
e milling/cutter, 97
e plate (lathe), 103
e spanner (wrench), 94
ing (lathe), 102
cet (recess) (design feature), 50
ther key (shaft), 40–41
er (thickness) gauge, 87
sealing rings, 26
e sealing washer, 28
handle, 82
, hand, 82
r rod (welding), 111
(design feature), 50
(shaft), 38
weld, 109
er screw head, 5
design feature), 49–50
nder lever, 74
l bolt, 4
cutting, 112
e (coupling), 66–69
e (design feature), 50
ed pipe bend, 30

Flanged pipe joints, 30–34
Flanged shaft coupling, 66
Flange/washer screwed brass pipe coupling, 31
Flanging (presswork), 116
Flap non-return valve, 36
Flash welding, 110–111
Flat belt drive, 70
Flat-bottomed hole, 52
Flat cam follower, 80
Flat circlip, 13
Flat cold chisel, 83
Flat file, 82
Flat fillister screw head, 5
Flat head rivet, 10
Flat (plain) washer, 8
Flat scraper, 83
Flat sheet steel, 57
Flexible bellows pipe joint, 32
Flexible expansion pipe bends, 33
Flexible hoses/pipes/fittings, 33
Flexible shaft couplings, 67–69
(*see* Shaft couplings)
Fluid flywheel (fluid coupling), 65
Fluid seals, 23
Flush rivet, 10
Fluxes, 108
Fly cutting/cutter, 98
Flywheel (design feature), 50
Foamed plastics, 121
Folding (presswork), 116
Folding rule, 87
Followers (cam), 79–80
Foot (design feature), 50–51
Foot-operated shearing machine, 116
Footstep thrust bearing, 43
Forging (design feature), 51
Forging dies/hammer/methods, 114
Fork/fork end (design feature), 51
Former/forming tool (spinning), 117
Foundation bolts, 4
Four-bar chain, 75
Four-jaw independent chuck, 103
Freewheeling (over-running) clutches, 64
Friction clutches, 63–65
Friction coefficient, 60, 65
Friction disk/lining, 61, 63–65
Friction materials (brakes/clutches), 65
Friction pad, 60
Friction plate (clutch), 64
Friction-welded poppet (mushroom) valve, 112
Friction welding, 112
Fulcrum (lever), 74
Fullers (forging), 114
Fusible alloys, 120

Ganged milling cutters, 96
Garter spring, 25–26
Gas (air) bearing, 44
Gas bearing lubricant, 46
Gas bottle, 111
Gaskets (seals), materials, 28
Gas pressure control valve (welding), 111
Gas shielded metal arc welding, 112
Gas welding/torch, 111
Gate (sluice) valve, 36
Gauges, 87–88, 90
Gearbox, 22
Gear changing, 23
Gear flexible shaft coupling, 68
Gear layshaft, 22
Gear materials, 23
Gears, 19–23
Gear shaft, splined, 23
Gears in mesh, 20
Gear teeth data, 19
Gear-tooth callipers, 89
Gear train, 22–23
Geneva wheel mechanism, 76
Gib head key (shaft), 40–41
Gland packings, 23–24
Gland-ring, 23
Gland (seal), 23
Glass fibre, 121
Glass pipe, 29
Globe valve, 34
Globoid cam, 81
Governors, 76–77
Grease bearing lubrication, 46
Grease nipple, 46
Greaser, screw-down, bearing lubrication, 46
Grinding machines (grinders), 104–106
Grinding wheels/materials/motor, 104–106
Grooved pin, 11
Grooves in shafts, 39
Grub screw, 5
Gunmetal, 119

HT (high tensile) bolt, 1
Hacksaws, 83
Half nut, 6, 7
Half-round chisel, 83
Half-round file, 82
Half-round scraper, 83
Halved (split) pedestal bearing, 42
Hammers, 86, 114
Hand/electric power drills, 84
Hand files, 82
Hand forging tools, 114
Handle, bar, 94
Handle (design feature), 51
Handle, file, 82
Handle, ratchet, 94
Hand reaming/tapping, 85
Handwheel (design feature), 51
Hardie/Hardie hole (hand forging), 114
Hard solder/soldering (brazing), 108
Hartnell spring-loaded governor, 77
Hatchet-type soldering iron, 108
Hat packing (seal), 25
Headstock (lathe), 102
Hearth, brazing, 108–109
Heavy-duty band saw, 107
Heavy-duty pedestal grinder, 106
Heavy-duty pipe joint, 30
Heavy screwdriver, 93
Helical coil spring, 15
Helical compression springs, 15, 17
Helical gears, 20
Helical reamer, 85
Helical rectangular-section spring, 15
Helical spring data/materials, 16
Helical spring lock washer, 9
Helical tension/torsion springs, 16–17
Hemp (gland packing), 24
Hermaphrodite callipers, 89
Herringbone gear, 20
Hexagonal bar, 58–59
Hexagonal head bolt, 1–2
Hexagon nut, 6–8
Hexagon socket head screw (bolt), 3
Hexagon socket wrench (Allen key), 3, 94
Hide-head hammer, 86
High carbon steel, 118
High-pressure hydraulic hose, 33
High tensile (HT) bolt, 1
Hinged bellows expansion pipe joint, 32
Hinged flap non-return valve, 36
Holes, 52–53, 104, 114
Holes (design feature), 52–53
Hollow rectangular/round sections, 58/59
Hollow shaft, 38
Hollow steel sections, 58
Hooke's-type universal joint, 69
Horizontal axis centrifugal casting, 113–114
Horizontal boring machine, 106
Horizontal milling machine, 95
Horizontal milling, up/down cuts, 96
Hose clips, 12
Hose pipe fittings, 33
Hot chisel, 83
Hot extrusion/extruded sections, 115
Hot-rolled bar, 115
Hot-rolled steel sections, 58
Hot rolling, 115
Housing (bearing), 46, 53
Housing (design feature), 53
Hub (design feature), 53
Hub, pulley, 53
Hydraulic brake system, 62
Hydraulic cylinder, 26, 61–62
Hydraulic hose/fittings, 33
Hydraulic jack, 77
Hydraulic pipe joint, 30
Hydraulic relief (safety) valve, 35
Hydraulic shock absorbers, 78
Hydrogen gas (welding), 111
Hypoid gears, 21

IC engine piston rings, 27
I-section hot-rolled bar, 115
I-section lever, 75
I-section steel bar/beam, 58, 115
ISO metric precision hexagon nuts, 6
Impact extrusion tools, 116
Indented foundation bolt, 4
Independent four-jaw chuck, 103
Indicator (dial) gauge, 90
Inserted tooth milling cutter, 98
Inside callipers, 89
Inside micrometer, 90
Instrument makers' screwdriver, 93
Instrument screw thread, 5
Integral collar (shaft), 39
Intermittent weld, 109–110
Internal circlip, 13
Internal expanding shoe brake, 61
Internal gear, 22
Internally serrated (tooth) lock washer, 9

Internal shaft splines, 40
Investment (lost wax) casting, 113
Involute gears, 19–23
Iron, 118
Iron pipe, 29

Jacks, 77–78
Jam (locked) nuts, 7
Jenny callipers, 89
Jet bearing lubrication, 46–47
Joints/gaskets, 28
Journal/bearing/shell, 42
Jubilee hose clip, 12

Keys/keyways (shafts), 40–41
Killed spirit (zinc chloride) (flux), 108
Knifed (spotfaced) hole, 53
Knife-edge cam follower, 80
Knuckle (pin) joint (tie rods), 41
Knurling (design feature), 53

Labyrinth gland (steam turbine), 27
Labyrinth seal (ball bearing), 27–28
Laminated leaf (carriage) spring, 17–18
Laminated plastics, 121
Laminations (*see* Metaflex flexible shaft coupling)
Lap joint, riveted, 10
Lathe, centre/engine, 102–104
Lathe components/operations/tools, 102–104
Layshaft (gear), 22
Lead/lead alloys, 120
Lead bronze, 119
Lead-head hammer, 86
Leadscrew (lathe), 102
Lead, screw thread, 14
Leaf spring, 17–18
Leather belt, 70
Leather sealing washer, 28–29
Left-hand screw thread, 13
Leftward welding, 111
Levers, 74–75
Lift (cam), 79
Light-duty bandsaw, 107
Light gauge tube, 59
Line of action (gear), 19
Linkages (mechanisms), 75–76
Link belt chain/drive, 74
Lip seal with garter spring, 25
Loads on components
- Belleville washer, 18
- block-and-tackle, 72–73
- bolt/bolt tightening, 1–2
- buttress thread, 14
- cell (brake), 62
- conical spring, 17
- helical gear, 20
- jacks, 77–78
- leaf spring, 17–18
- lever, 74
- ring spring, 18
- roller bearing, 45–46
- shock absorber, 78
- spring design, 16
- strut, 56

Locked (jam) nut, 7
Lock washers, 8–9
Long-nose (needlenose) pliers, 92
Loose collar (shaft), 39
Lost wax (investment) casting, 113
Low carbon steel, 118
Low melting point alloys, 120
Lubricants/lubricators (bearings), 46
Lubrication (bearings), 47
Lug (design feature), 54
Lyre flexible expansion pipe bend, 33

Machine reaming/tapping, 100
Machine screw, 5
Machine tools, 95–107
Machine-tool tables
- auxiliary drilling, 99
- bandsaw, 107
- borers, 106
- drilling machine, 98–99
- milling machines, 95
- shaper, 100–101
- surface grinder, 104
- vertical shaper, 106

Machine vice (vise), 92
Magnesium alloys, 119
Magnetic base test set, 90
Magnetic chuck (surface grinder), 104
Major diameter (thread), 1, 14
Male/female coupling (pipe fitting), 32
Malleable cast iron, 118
Malleable iron screwed pipe fittings, 29
Mallets (soft hammers), 86
Mandrel (extrusion), 115
Manganese (alloy element), 118
Manganese bronze, 119
Manilla rope, 72
Marking-off, 87
Masonry twist drill, 84
Master cylinder (brake), 62
Maul (sledge hammer), 86
Mechanical friction damper, 78
Mechanisms (linkages), 73–81
Medium carbon steel, 118
Mesh (gears), 19–20
Mesh, wire, 58
Metaflex flexible shaft coupling, 69
Metal alloys, 118–120
Metal inert gas (MIG) welding, 112
Metallic gasket, 28
Metallic packing seal, 24
Metal spring flexible shaft coupling, 68
Metric bolt/screw threads, 1, 13
Michell thrust bearing/pad, 43
Micrometer head, 90
Micrometer components, 89–90
Mild carbon steel, 118
Milling cutter nomenclature, 96
Milling cutters, 96–98
Milling machine components/operations, 95–98
Milling machines, 95–96
Minor diameter (thread), 1, 14
Misalignment (shafts), 67–68
Mist bearing lubrication, 47
Module (gear), 19
Molybdenum (alloy element), 118

Monel metal, 119
Mop, buffing/polishing, 105
Morse taper (twist drill), 84
Motor, grinding wheel, 105
Mould (casting), 113
Moulded rubber insert shaft coupling, 67
Moulding box components (casting), 113
Moulding sand (casting), 113
Muff shaft coupling, 66
Multi-plate (disk) brake, 61
Multi-plate (disk) friction clutch, 64
Multiple V belt drive, 71
Multiple V pulley, 71
Multi-start power screw thread, 14
Multi-start worm gear, 21
Mushroom (poppet) valve, 37, 112

Natural rubber, 121
Needlenose (long-nose) pliers, 92
Needle (pattern) files, 82
Needle roller/bearing, 44–45
Needle screw-down valve, 34
Needle-type non-return valve, 36
Neoprene, 121
Nickel (alloy element), 118–119
Nickel-chrome steel, 118
Nimonic alloys, 119
Non-ferrous metals, 59, 118–120
Non-metallic materials, 120–121
Non-return (check) valves, 35–36
Nose (cam), 79
Notching (presswork), 116
Nut-and-sleeve pipe joint, 31
Nuts, 6–8
Nyloc (elastic stop) nut, 8
Nylon, 120
Nylon cords (belt), 71

O-ring seal, 26
Odd leg callipers, 89
Off-hand grinder, 106
Oil lubrication, 47
Oil grooves (shaft), 39
Oil jet/mist/splash bearing lubrication, 47
Oil scraper piston ring, 27
Oil seals (*see* Seals)
Oilways (shafts), 40
Oldham flexible shaft coupling, 68
Open-ended (open-jaws) spanner (wrench), 93
Opposed taper roller bearings, 45
Orders of levers, 74
Oscillating crank mechanism, 76
Outside callipers, 88–89
Overhung crank, 39
Over-running (freewheeling) clutches, 64
Oxygen (welding), 111

PTFE (plastic), 121
PVC (plastic), 120
Packing (seals), 23–28
Pan head taper neck rivet, 10
Paper gasket, 28
Parallel-face gate valve, 36
Parallel motion mechanism, 75

Parallel roller (bearing), 44
Parallel sleeve compression pipe joint, 31
Parting-off (lathe), 102
Parting-off lathe tool, 103
Passive flux, 108
Pattern (casting), 113
Pattern (needle) files, 82
Pedestal bearing, 42
Pedestal/bench grinders, 106
Pedestal casting, 48
Pedestal (design feature), 54
Perspex (acrylic resin), 120
Phillips recess (cross-head) screw, 5
Phillips cross-head screwdriver, 93
Phosphor bronze, 119
Piercing (presswork), 116
Pillar (column) (design feature), 49, 54
Pin drift, 84
Pin hammer, 86
Pinion (gear), 21, 79
Pin (knuckle) joint (tie rods), 41
Pins
- crank-, 39
- core (die casting), 113
- dowel, 11
- driving (lathe), 104
- ejector (die casting), 113
- grooved, 11
- plain, 11
- roll, 11
- split (cotter), 12
- taper, 11

Pipe bends, 30, 33
Pipe casting, centrifugal, 113–114
Pipe clip, 13
Pipe couplings/fittings, 29–33
Pipe/duct joints (couplings), 29–34
Pipe/duct materials/fittings, 29–34
Pipe expansion joints, 32
Pipes, 29–34
Pipe (Stillson) wrench, 95
Pipe vice (vise), 92
Piston (design feature), 54
Piston (hydraulic) shock absorber, 78
Piston rings, 26–27
Piston rod, 26
Piston-type non-return valve, 36
Pitch circle/point (gear), 19
Pitch (screwthread), 1, 13–14
Pivoted segment (Michell) bearing, 43
Plain (flat) washer, 8
Plain/plain journal bearings, 42
Plain pin, 11
Plain thrust bearing, 43
Planet (epicyclic) gear, 22
Plastic-head mallet, 86
Plastic pipes, 31
Plastics, 120–121
Plate (design feature), 54
Plate (radial) cam, 79
Plates
- arc welded, 109
- bed, 47
- brake friction, 60–61
- catch (lathe), 104
- clutch friction, 63–64
- clutch pressure, 64

Plates *continued*
cover-, 49
deep drawn, 116
flame cut, 112
gas welded, 111
non-ferrous metal, 59
pressure (clutch), 64
presswork on, 116
ribbed, 55
steel, 57
steel chequer, 58
surface, 91
welded, 109–112
Plating, cadmium/chromium, 120
Pliers, 92
Plug cock valve, 38
Plug (pipe fitting), 29
Plug weld, 110
Plunger (design feature), 54
Pneumatic (air bellows) spring, 18
Polishing (buffing) mop, 105
Polyethylene, 120
Polypropylene, 120
Polystyrene, 120
Polytetrafluorethylene (PTFE), 121
Polythene, 120
Polyurethane, 121
Polyvinylchloride (PVC), 120
Poppet (mushroom) valve, 37, 112
Pop rivet, 11
Porous bearing, 43
Port (design feature), 55
Ports, 36–37, 55
Positive return cam, 81
Power absorption brakes (dynamometers), 62
Power saws, 107
Power tapping/attachment, 100
Power transmission screw threads, 14–15
Precision grinders, 104–105
Press/presswork, 116
Pressure angle (gear), 19
Pressure plate (clutch), 64
Pressure regulator valve (gas welding), 111
Profile (cam), 79
Prony brake, 62–63
Propane gas (welding), 111
Protractor (combination set), 88
Pulley block, 72–73
Pulley drive (drilling machine), 98
Pulley hub, 53
Pulleys, 70–74
Punch, centre, 84
Punched hole, 52
Punch (impact extrusion), 116
Punching (presswork), 116
Pure copper, 118

Quick-return mechanism, 76

Races (bearing), 44
Rack-and-pinion gear, 21
Rack-and-pinion steering, 79
Radial (plate) cam, 79
Radial-arm drilling machine, 99
Radius (design feature), 55
Rag (foundation) bolt, 4
Ram (die casting), 113
Ram (slotter), 101
Rasp, 83
Rasp (Surform-type), 83
Ratchet handle, 94
Ratchet mechanism, 76
Ratchet screwdriver, 93
Rawlbolt (anchor bolt), 4
Rayon, 120
Reamed hole, 53
Reamers/reaming, 85, 100
Recessed socket-head screw, 3, 55
Recess (faucet) (design feature), 50
Reciprocating power hacksaw, 107
Reciprocating surface-grinder table, 104
Rectangular bar, 58–59
Rectangular key (shaft), 40
Rectangular-section lever, 75
Rectangular-section spring, 15–16
Reducers (pipe fittings), 29, 32
Resin, acrylic (Perspex), 120
Resin-cored wire solder, 108
Resin flux, 108
Resistance seam weld, 110–111
Resistance welding, 110–111
Resistance wire, 119
Retainer (bearing), 44
Retaining rings (circlips), 13
Ribbed belt/drive, 71
Ribbed neck coach bolt, 2–3
Ribbed plate, 55
Rib (design feature), 55
Right-hand screw thread, 13
Rightward welding, 111
Ring (epicyclic) gear, 22
Ring oiler bearing, 43
Ring (round) nut, 6
Ring spanner (box wrench), 93
Ring spring, 18
Riser (casting), 113
Riveted joints, 10
Rivet materials, 10
Rivets, 9–11
Rods
arc welding, 109
brazing, 108
connecting, 75
cottered joint for, 42
filler, 111
pin (knuckle) joint for, 41
piston, 26
screwed, 2
sealing of, 26
tie, 41
Rolled steel sections, 115
Roller and rocker arm cam follower, 80
Roller bearings, 45–46
Roller cam follower, 80
Roller chain/sprocket drive, 73–74
Roller forming tool (spinning), 117
Roller freewheeling clutch, 64
Rolling-contact bearing, 44
Rolling/rolling mill, 115
Rolling (slab) milling/cutter, 97
Roll pin, 11
Rolls for I section, 115

Root (screw thread), 13
Rope (band) brake (dynamometer), 62–63
Rope drive/pulley, 72–73
Ropes, 72–73
Rotary files, 82
Rotary wire brush, 105
Rotating machine-tool table, 106
Rotor (design feature), 55
Round bar, 58–59
Round file, 82
Round head countersunk rivet, 10
Round head screw, 5
Round key (shaft), 41
Round-nose pliers, 92
Round (ring) nut, 6
Round-section circlip, 13
Rubber belt, 70
Rubber-bushed pin-type flexible shaft coupling, 67
Rubber gasket, 28
Rubber hose (auto), 33
Rubber insert sealing washer, 28
Rubber, natural/synthetic, 121
Rubber, pipe, 29, 33
Rubber springs, 18
Rubber-tyre-type flexible shaft coupling, 68
Rules, 86–87
Runner (casting), 113

Saddle key (shaft), 41
Saddle (lathe), 102
Safety valves, 35
Sand casting/equipment, 112–113
Saunders valve, 37
Sawing machines, 107
Scissors jack, 78
Scotch yoke mechanism, 76
Scrapers, 82–83
Screw-and-nut steering, 79
Screw-cutting /die (hand), 85
Screw-cutting (lathe), 102
Screw-cutting tool setting gauge, 87
Screw-down bearing greaser, 46
Screwdrivers, 93
Screwed brass pipe couplings/fittings, 31–32
Screwed iron pipe fittings, 29
Screwed rod, 2
Screwed shaft, 39
Screw heads, 5
Screw jack, 77
Screw pitch (thread) gauge, 87
Screws, 4–6
Screw shut-down (globe) valve, 34
Screw taps, 85
Screw threads/details/dimensions, 1, 13–15
Scribers, 87
Scriber with surface gauge, 88
Sealed ball bearing, 45
Sealing (rods), 26
Sealing washers, 28–29
Seals/joints/gaskets, 23–29
Seam weld, 110
Second order lever, 74
Segmental carbon ring seal, 26
Self-aligning ball bearing, 45
Self-centring three-jaw chuck, 103
Self-oiling bearing, 43
Self-tapping screws, 5
Sensitive drilling machine, 99
Serrated lock washer, 9
Serrated neck coach bolt, 2–3
Set screws, 5
Set (*see* Chisels)
Shaft collars, 39, 43
Shaft couplings, 66 (*see* Flexible shaft couplings)
Shafts (types/details), 22–23, 38–41, 43
Shaping machine/components/tools (shaper), 100–101
Shearing/machine, 116
Shears (snips/tinsnips), 93
Shear/tension rubber spring, 18
Sheaves (pulleys), 72–73
Sheet metals
duralumin, 119
non-ferrous metal, 59
steel, flat/corrugated, 57
Shell endmilling/cutter, 97
Shells (journal bearing), 42
Shock absorbers, 78
Shoe (brake), 61–62
Shoulder (shaft), 38
Shoulder socket bolt/screw, 3
Side and face milling/cutter, 97
Side-cutting (knife) tool, 103
Sight feed drop oiler (bearings), 46
Silicon (alloy element), 118
Silicon carbide (abrasive), 104
Silver solder (soldering), 108
Silver steel, 118
Simple harmonic motion cam, 80
Simple square drift, 84
Sindanyo (asbestos), 121
Single block brake, 60
Single helical gear, 20
Single plate (disk) friction clutch, 63
Single row ball bearing, 44
Single-start worm gear, 21
Skew bevel gear, 21
Slab (rolling) milling/cutter, 97
Sledge hammer (maul), 86
Sleeve (bush) (bearing), 42
Sleeve (design feature), 56
Sleeve shaft coupling, 66
Sleeve valve, 37
Slider, 75
Slideway (lathe), 102
Sliding contact bearing, 42
Sliding expansion pipe joint, 32
Slot (design feature), 56
Slot milling/cutter, 96–97
Slotted head screw, 5
Slotted nut, 7
Slotter (vertical shaper), 101
Slotting machine, 101
Slot weld, 110
Slow bend pipe fitting, 29
Sluide (gate) valve, 36
Snap head rivet, 10
Snips (tinsnips/shears), 93
Snug (pin), 3–4
Socket head screws/bolts, 3, 5
Socket pipe fitting, 29
Socket screw wrench (Allen key), 3, 94
Socket spanner (wrench)/attachment, 94
Soft hammers, 86

Soft solder/soldering methods, 107–108
Soldered copper (capillary) pipe fittings, 30
Soldering iron bit, 107
Soldering irons, 107–108
Soldering torch, 107
Solders, 107–108
Solid bolted flanged coupling, 66
Solid shaft, 38
Spacer (distance piece) (design feature), 49
Spanners (wrenches), 2–3, 93–95
Spelter (brazing), 108
Spherical plug cock valve, 38
Spigot (design feature), 56
Spindle, machine tool, 95–99
Spindle nose (lathe), 104
Spindle (small shaft), 38
Spinning/equipment, 117
Spiral bevel gear, 20
Spiral torsion spring, 16–17
Splash bearing lubrication, 47
Splined gear shaft, 23
Splines (shaft), 40
Split (cotter) pin, 12
Split (halved) pedestal bearing, 42
Split nut, 7
Spool valve, 37
Spotfaced (knifed) hole, 53
Spotfacing, 100
Spot weld, 110
Sprag/over-running clutch, 64
Spring collet (chuck), 103
Spring data, 16
Spring dividers, 89
Spring-joint callipers, 88
Spring-loaded carbon face seal, 27
Spring-loaded safety valve, 35
Spring (compression stop) lock nut, 7
Springs, 15–18
Spring set brake, 61
Spring tape rule, 87
Spring washer, 9
Spring wire hose clip, 12
Sprocket (chain drive), 73–74
Sprue (lost wax casting), 113
Spur gear, 19–23
Square bar, 58–59
Square (combination set), 88
Square head bolt, 1
Square hole, 53
Square neck coach bolt, 2–3
Square nut, 6
Square power transmission screw threads, 13–14
Square shaft splines, 40
Square (try square), 88
Stainless steel, 118
Stamped spring nut, 8
Standard ball bearing, 44
Standard grinding wheel, 105
Standard lip seal, 25
Standard pliers, 92
Steam safety valve, 35
Steel angle section, 58, 115
Steel bar, 57–58
Steel chequer plate, 58
Steel grinding wheel (abrasive coated), 105
Steel I section, 58, 115
Steel pipe, 31

Steel rivet, 10
Steels, 118
Steel sealing washer (rubber insert), 28
Steel sections, 58
Steel sheet, 57
Steel strip, 57
Steering mechanisms, 79
Stepped pulley drives, 70–71
Stillson (pipe) wrench, 95
Straight bevel gear, 20
Straight peen hammer, 86
Straight pipe coupling, 29, 32
Straight reamer, 85
Straight shank twist drill, 84
Straight shaper tool, 101
Straight soldering iron, 108
Stock, die, 85
Stop valve, 34
Strap clamp, 91
Strip (steel), 57
Strut (design feature), 56
Stud brass pipe coupling, 32
Stud/stud bolt/box, studding, 2
Stuffing box (gland), 23–24
Sun (epicyclic) gear, 22
Surface finish, shaft, 38
Surface gauge with scriber, 88
Surface grinder, 104
Surface plate, 91
Surface table, 91
Surform rasp, 83
Swage (forging), 114
Swan neck shaper tool, 101
Sweating (joints), 107
Swivel adjustment (shaper), 100
Synthetic rubbers, 121

T bolt, 3
T-section bar, 115
T slot, 3, 56
T-joint pipe fitting, 29–30, 32
Tab washer, 9
Tables, 91, 95, 98, 104, 106, 109
Tack weld, 110
Tailstock (lathe), 102
Tangent cam, 81
Tap bolt, 2
Taper neck pan head rivet, 10
Taper pin, 11
Taper reamer, 85
Taper roller (bearings), 44–46
Taper (shaft), 38–39
Taper shank twist drill, 84
Taper washer, 8
Tapped/through blind holes, 53
Tapping, 85, 100
Tapping attachment, 100
Taps, 85
Tap wrench, 85
Telescopic hydraulic shock absorber, 78
Tensioner, chain drive, 73
Tension/shear rubber spring, 18
Thermoplastic/thermosetting, 120
Thickness (feeler) gauge, 87
Thimble (micrometer), 89
Thin nuts, 6
Third order lever, 74

Thread-cutting die, 85
Thread-cutting screw, 5
Thread form, 13
Thread-forming screw, 5
Thread (screw pitch) gauge, 87
Threads (see Screw threads)
Three-face (triangular) file, 82
Three-jaw self-centring chuck, 103
Through bolt, 2
Through drilled hole, 52
Thrust bearings, 43–46
Thrust collar (shaft), 39, 43
Thrust pad (Michell bearing), 43
Tie rod (bar), 41
Timing belt/drive, 72
Timing pulley, 72
Tin, 119–120
Tinning, 107
Tinsnips (snips/shears), 93
Tips for lathe tools, 103
Titanium, 119
Tommy bar, 93
Tool holder (lathe), 102–103
Toolmakers' clamp, 91
Tool post (shaper), 100
Tools, lathe, 102–103
Tool steel, 118
Tooth clearance (gears), 19
Toothed belt, 72
Toothed drift, 84
Toothed pulley, 72
Tooth (internally/externally serrated) lock washer, 9
Tooth thickness (gears), 19
Top face (anvil), 114
Top fuller, 114
Torches
 brazing, 108
 gas welding, 111
 soldering, 107
Torque spanner (wrench), 2, 95
Torsion bar spring, 18
Torsion spring, 16–18
Transition piece (ducting), 34
Triangular scraper, 83
Triangular shaft splines, 40
Triangular (three-face) file, 82
Triplex roller chain, 74
Try square (square), 88
Tube, 59
Tubular box spanner (socket wrench), 93
Tubular rivet, 10
Tufnol, 121
Tungsten (alloy element), 118
Tungsten carbide, 118
Tungsten carbide tool tip, 103
Turning between centres (lathe), 104
Turning/facing lathe tool, 103
Turning (lathe), 102
Twist drills, 84
Two-coil spring lock washer, 9
Two-pulley sheave, 73
Two-start power transmission thread, 14

UNC/UNF bolt/screw threads, 1, 13
U bolt, 4
U flexible expansion pipe bend, 33
U leather seal, 24

US bolt/screw threads, 1, 13
Unified coarse thread (UNC), 1, 13
Unified fine thread (UNF), 1, 13
Uniform strength bolt, 2
Universal joints, 69
Universal joint spanner attachment, 94
Upholstery (conical) spring, 17

V belt/drive, 70–71
V block/clamp, 91
V pulleys, 70–71
V threads, 13–14
Valves, 34–38
Vanadium (alloy element), 118
Variable speed belt/drive, 72
Vernier calliper gauge, 90
Vertical axis centrifugal casting, 113
Vertical boring machine, 106
Vertical box column drilling machine, 98
Vertical milling machine/operations, 95
Vertical shaper (slotter), 101
Vices (vises), 92
Volute spring, 17

Washers, 8–9, 28–29
Wax pattern (investment) casting, 113
Web (design feature), 56
Wedge gate valve, 36
Welded barrel nut, 7
Welded butt joint, 109
Welded plate, 109–112
Welded steel gear, 23
Welding equipment/gases, 109, 111
Welding joints/rods/methods, 109–112
Welding set, 109
Welding table, 109
Weldment (design feature), 57
Wheel (design feature), 57
Wheel-shape casting, 113
White (Babbitt) metal, 120
Whitworth screw thread, 1, 13
Wing nut, 6
Wire, 58, 119
Wire brush, rotary, 105
Wire cutters, 92
Wire drawing, 117
Wire drawing die/drum, 117
Wire eye, 49
Wire mesh, 58
Wire rope, 72–73
Wood, 121
Wooden-head mallet, 86
Woodruff key (shaft), 41
Woodscrews, 5
Work between centres (lathe), 104
Work/workpiece (lathe), 102–104, 106
Worm, 21
Worm-and-wheel steering, 79
Worm gears/wheel, 21
Woven fabric belt, 70
Wrenches (see Spanners)
Wrought iron, 118
Wrought iron pipe/fittings, 29

Y alloy, 119

Zinc, 119
Zinc chloride (killed spirit) (flux), 108

The Student Engineer's Companion is an easily-followed guide to over 800 workshop tools, components, processes and machines for students and others who need an authoritative reference work. The text is supplemented by several hundred illustrations, mostly in the form of engineering drawings and often with third angle projection, and alternative American terms are given where appropriate. As an aid to quick reference the book is divided into four sections covering basic engineering components, power transmission elements, workshop equipment and engineering materials, and there is a comprehensive index for easy cross-reference.

By the same author

Famous Names in Engineering

Illustrates and records the lives of those engineers and scientists whose names are well-known, not because of themselves, but because their names have been given to a theory, unit or an effect.

1981 96 pages illustrated soft cover 0 408 00540 8

Other related [illegible]

Dictionary [illegible]nical Engineering—Third edition
J.L. Nayler a[illegible] Nayler

1985 400 [illegible] illustrated hard cover 0 408 01505 5

ISBN 0-408-00438-X

9 780408 004381

Butterworths
Borough Green,
Sevenoaks,
Kent TN15 8PH