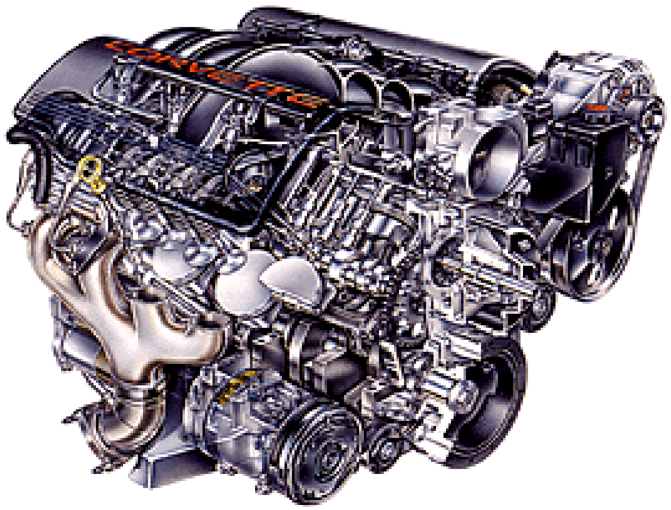
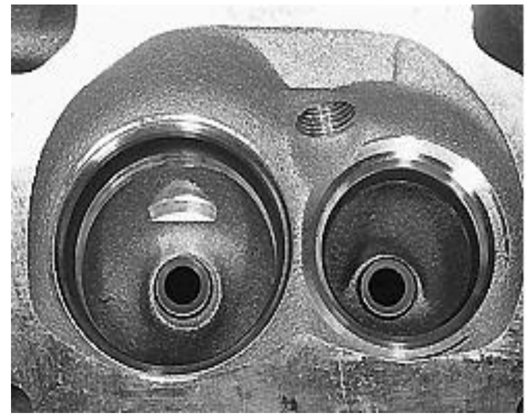
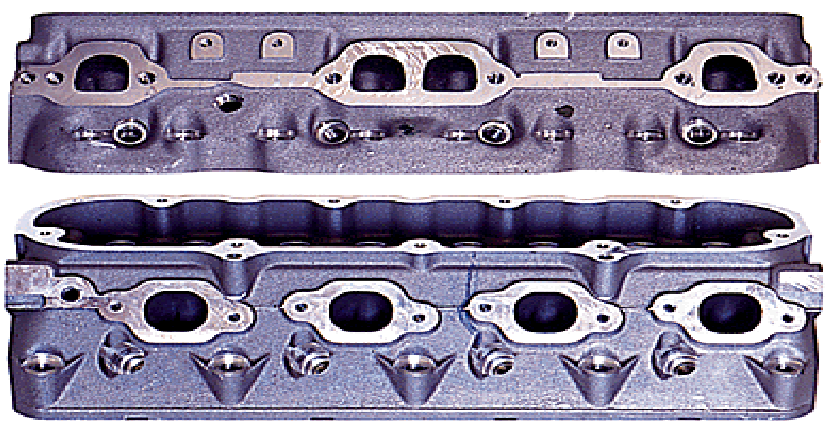
The internal combustion engine is a rich source of examples of almost every conceivable type of heat transfer. There are a wide range of temperatures and heat fluxes in the various components of the internal combustion engine. Internal combustion engines come in many sizes, from small model airplane engines with a 0.25 " (6 mm) bore and stroke to large stationary engines with a 12" (300 mm)

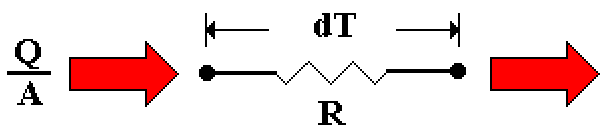
About 25 % of the air/fuel mixture energy is converted to work, and the remaining 75% must be transferred from the engine to the environment. The heat transfer paths are many, and include many different modes of heat transfer.
In this module, we will discuss the heat transfer processes in the engine components, then consider the engine parameters and variables which affect the heat transfer processes. We present various models of the heat transfer processes, and example calculations of the heat transfer rates and coefficients.
Once you have explored the core sections, you can also go to the advanced sections, where current research topics and problems are presented.


CORVETTE





The conduction and convection heat transfer in engines are processes that occur in series and parallel with each other. A series path is convection through the cylinder gas boundary layer, conduction across the cylinder wall, and convection through the coolant liquid boundary layer; and a parallel path is conduction through the cylinder wall and through the piston crown. In heat transfer resistance modeling, we look for regions which have relatively large temperature differences, and compute the heat transfer resistance across those regions



The thermal resistance is defined as the ratio of the temperature difference, dT, to the heat transfer Q. This is analogous to Ohm's law, in which the electrical resistance is defined as the ratio of the voltage drop across a resistor to the current flow across the resistor.

$DV = I\ R$ or $R = DV/\ I$ (Ohm's Law)

$DT = (Q/A)\ R$ or $R = DT\ /\ (Q/A)$ (on a per unit area basis)

Conduction resistance

$$R_{cond} = \frac{\Delta T}{Q/A} = \frac{L}{k}$$

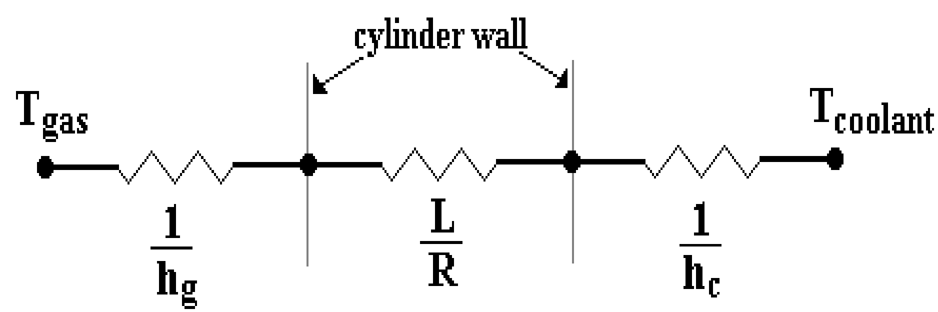
Convection resistance

$$R_{conv} = \frac{\Delta T}{Q/A} = \frac{1}{h}$$

The resistance model is very useful in determining the heat transfer in a complex steady state heat transfer situation. It is assumes that the heat transfer is primarily one dimensional across the resistance element, so as the problem becomes more multidimensional, the accuracy decreases.

Heat transfer to coolant

For the heat transfer from the engine cylinder to the coolant, a series path can be assumed. For example:



For example, assume that the cylinder gas temperature is 1200 K, and the coolant temperature is 300 K. The cylinder thermal conductivity is 80 W/mK, and its thickness is ½" (0.012 m). Also assume that the convection coefficient is 200 on the gas side, and 1000 on the coolant side.

Then

The thermal resistance of the gas layer, R_{gas}, is $1/h = 1/200 = 50 \times 10^{-4}$

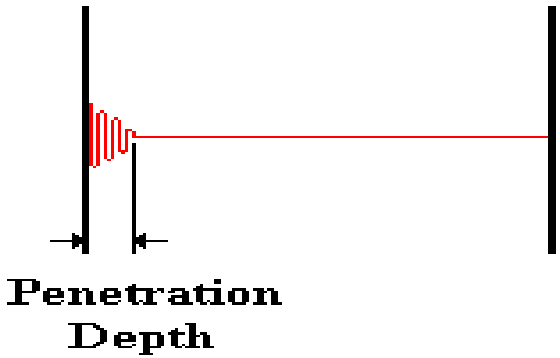
The thermal resistance of the cylinder wall, R_{wall} is $L/k = 0.012/80 = 1.5 \times 10^{-4}$

The thermal resistance of the coolant, $R_{coolant}$ is $1/h = 1/1000 = 10 \times 10^{-4}$

The largest resistance is the gas side resistance, R_{gas}.

This means that the heat transfer in this case is relatively insensitive to the type of material used in the wall. If the cylinder was made of aluminum instead of steel, the overall heat transfer would not change significantly.

For the above resistances, the overall heat transfer is about 150,000 W/m2.



The heat transfer from the hot combustion gases is includes forced convection through the hot gas boundary layer, conduction through the cylinder wall, and forced convection (including boiling) into the fluid coolant in the head, engine block, and piston.

There is a small (about 5 %) radiative component of heat transfer from the gas to the cylinder walls.

The heat transfer process is periodic due to the piston motion. However, the engine speed is usually high enough so that the temperature fluctuations only penetrate about a millimeter into the cylinder wall.

We use the unsteady heat conduction equation to examine the effect of the periodic combustion on the temperature profile in the cylinder wall.

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2}$$

where T is temperature C, *t* is time (sec), *x* is distance (m), and a is the thermal diffusivity (m^2/s). Scale analysis of the above equation gives an approximate relation for the penetration depth *x*:

$$x \cong (\alpha t)^{1/2}$$

An engine speed, w, of 1000 rpm, has a time scale *t* of

$$t = 1/\omega = \frac{1}{1000}\left(\frac{\min}{rev} * \frac{60\sec}{\min} * \frac{rev}{2\pi\, rad}\right) = 0.01\sec$$

The thermal diffusivity of steel is about a = 20 x 10 $^{-6}$ m^2/s, so the penetration depth is approximately

$$x \cong (\alpha t)^{1/2} = \left[\left(20x10^{-6}\right)\left(10^{-2}\right)\right]^{1/2} = 0.5mm$$

Since the penetration depth is a very small fraction of the cylinder wall thickness, the cylinder wall can be assumed to have a temperature profile that is not changing in time

Cylinder Heat Flux and Temperatures

The surface temperature of the cylinder can be measured with thermocouples, and the resulting cylinder heat flux deduced from Fourier's conduction equation and the unsteady heat conduction equation. The figure below is a representative graph of the cylinder heat flux as a function of crank angle.

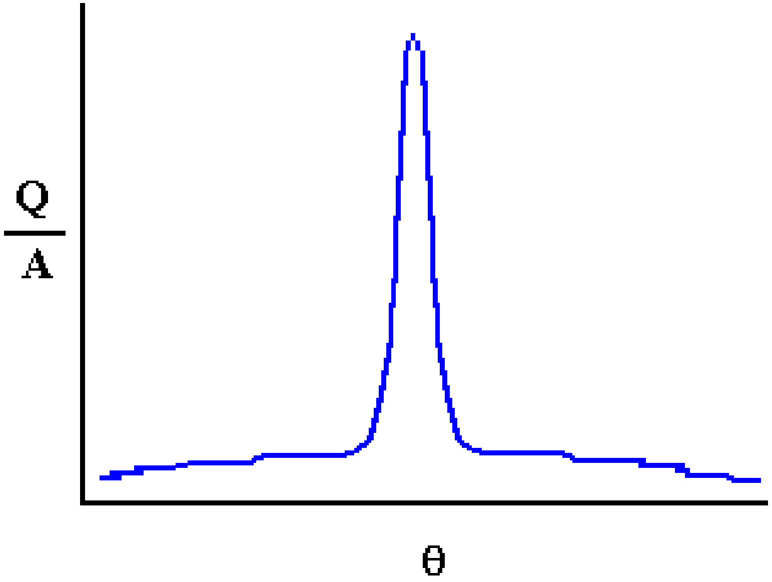


Figure 15. Heat Flux versus Crank Angle

The heat flux begins rising when the combustion flame impacts the cylinder wall, has a maximum at peak cylinder pressure when gas temperatures peak, typically 20 degrees after Top Dead Center (TDC).

The peak heat fluxes are on the order of 1 to 3 MW/m2. The heat flux increases with increasing engine load and speed.

As the cylinder wall temperature increases, the piston and ring friction will be reduced, decreasing the fuel consumption, the heat flux to the wall from the combustion gases will decrease, and the formulation of pollutants such as nitrous oxides also increases.

A correlation used to compute the space averaged instantaneous heat transfer coefficient (Woschni, 1967 , "A universally applicable equation for the instantaneous heat transfer coefficient in the internal combustion engine",

SAE Paper 670931): is:

Nusselt # = 0.035 (Reynolds #)$^{0.80}$

$$\frac{h(\underline{x},t)b}{k} = a\left(\frac{Ub}{\nu}\right)^m$$

Where:
b = cylinder characteristic length (m), usually chosen to be the cylinder bore.

k = gas thermal conductivity, (W/mK), typical value 0.06

n = gas kinematic viscosity, (m2/s), typical value 100 x 10-6

U = average gas velocity (m/s)

During intake, compression, and exhaust, the average gas velocity U is proportional to the mean piston speed, $U_{piston.}$

For intake and exhaust, $U = 6.18\ U_{piston}$

For compression, $U = 2.28\ U_{piston}$

The instantaneous heat transfer coefficient during combustion depends on the gas speed and cylinder pressure, which change significantly during the combustion process. During combustion and expansion, the gas speed U (m/s) induced by combustion is given by the following equation (Woschni, 1967) :

$$U = 2.28 U_p + 0.00324 T_o \frac{V}{V_o} \frac{\Delta P_c}{P_o}$$

Where To, Vo, and Po are reference intake cylinder temperature (K), volume (m3), and pressure (Pa), V is the instantaneous cylinder volume (m3), and DP_c is the instantaneous pressure rise due to combustion relative to the unburned gas pressure at the same crankshaft angle.

The unburned gas cylinder pressure is determined from thermodynamic analysis.

An example pressure computation is given in the Cylinder Pressure page. For engines with significant swirl, the gas velocities are higher.

The gas properties in the correlation equation are evaluated at the instantaneous average cylinder temperature determined from the ideal gas law:

$$T_g = \frac{P\,M}{\rho R}$$

Where:

P = instantaneous cylinder pressure (kPa)

r = instantaneous mixture density, mass/volume (kg/m3)

M = averaged molecular weight (kg/kmol)

R = universal gas constant, 8.314 kJ/kmol K

If P= 500 kPa, r = 5 kg/m3, M = 40 kg/kmol

$$T_g = \frac{P\,M}{\rho R} = \frac{500 \cdot 40}{5 \cdot 8.314} = 480\,C = 753\,K$$

The thermal conductivity k varies as T^3/4 , and the viscosity varies as T^0.62 , so that the instantaneous heat transfer coefficient can be written as

$$h(\underline{x},t) = 3.26 b^{-0.2} P^{0.8} T^{-0.55} U^{0.8}$$

If b=0.1m, P = 500 kPa, T = 1000 K, and U = 10 m/s,

$$= 3.26 \cdot 01^{-0.2} \cdot 500^{0.8} \cdot 1000^{-0.55} \cdot 10^{0.8}$$

$$= 105 \mathrm{~W/m^2} K$$

For the overall average heat transfer from the gas to the cylinder coolant, convection type heat transfer equations are used.

$$Q / A = h\,(T_{gas} - T_{cool})$$

Where:
Q = overall heat transfer (W/m2)
A = reference cylinder area (m2)
Tgas = effective gas temperature, typically 800 C
Tcool = coolant temperature, typically 80 C
h = heat transfer coefficient (W/m2 K)

The heat transfer coefficient depends on the engine geometry, such as the exposed cylinder area and bore, and the piston speed. Due to the complex gas flow in the cylinder, it varies with location in the cylinder and in time with changing piston position. The value of the heat transfer coefficient is found from a Nusselt number - Reynolds number type correlation.

Nusselt # = a (Reynolds #)^{m}

$$\frac{hb}{k} = a\left(\frac{Ub}{\nu}\right)^m$$

Where:
b = cylinder characteristic length (m), usually chosen to be the cylinder bore.
k = gas thermal conductivity, (W/mK), typical value 0.06
n = gas kinematic viscosity, (m2/s), typical value 100 x 10-6
U = characteristic gas velocity (m/s)
The characteristic gas velocity U is typically the mean piston speed:

$$U = U_{piston} = 2 * \frac{RPM}{60} * Stroke$$

However, some correlations use the average intake velocity:

$$U = U_{intake} = \frac{\dot{m}}{\rho A_{piston}}$$

and other correlations require a characteristic combustion velocity. The coefficient a and exponent m are found from engine experiments. There are three types of heat transfer coefficients used in the computation of engines heat transfer.

Table III. Types of Heat Transfer Coefficients

	Averaging	Used for
h(x,t)	averaged over time & space	overall steady state energy balance calculations
h(x,t)	instantaneous, space average	heat transfer versus crank angle
h(x,t)	instantaneous, local	local calculations of thermal stress

The peak values of the instantaneous and local coefficients can be many times higher than the averaged values.

A frequently used correlation for a space and time averaged heat transfer coefficient is that of C. F. Taylor ("The Internal Combustion Engine in Theory and Practice", MIT Press, 1985), which uses the mean piston speed for the characteristic velocity.

$$\frac{h(\underline{x},\underline{t})b}{k} = 104\left(\frac{U\,b}{\nu}\right)^{3/4}$$

For a 1000 rpm engine with a 0.1m stroke, combustion gas temperature of 1000C, and coolant temperature of 80 C

$$U = U_{piston} = 2*\frac{RPM}{60}*Stroke = 2*\frac{1000}{60}*0.1 = 3\,m/s$$

Where:
b = 0.1 m
k = 0.06 W/mK
n = 100 x 10 -6 m2/s
the average heat transfer coefficient is

$$\frac{h(\underline{x},\underline{t})\,0.1}{0.06} = 10.4\left(\frac{3\cdot 0.1}{100x10^{-6}}\right)^{3/4} = 4215$$

$$h(\underline{x},\underline{t}) = 2529\ W/m2\,K$$

The average heat transfer per unit area from the cylinder to the coolant is

$$Q/A = h\,(T_{gas} - T_{cool}) = (2529)(1000-80) = 2.33\ MW/m2$$

For a reference cylinder area of p b^2/4, the average heat transfer is

$$Q = \frac{Q}{A} * \pi b^2 = (2.33x10^6)(\pi\, 0.1^2/4) = 18273\,W$$

There are two types of coolants used to remove the heat from the engine block and head: air, and water. With air as a coolant, the heat is removed through the use of fins attached to the cylinder wall. With water as a coolant, the heat is removed through the use of fluid filled internal cooling passages.

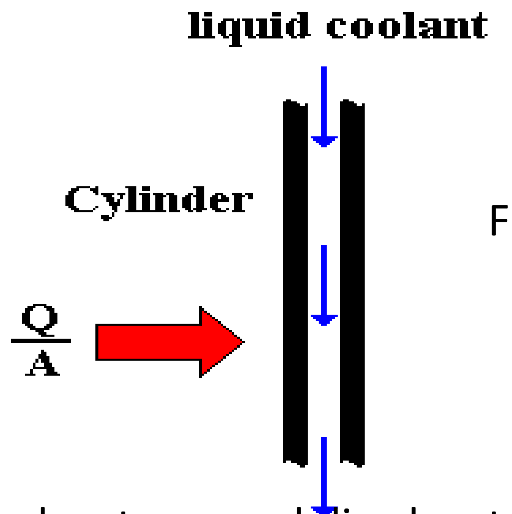


Figure 17. Liquid Coolant

Both have various advantages and disadvantages. Air systems are much noisier, since there are no water jackets to absorb the combustion sound. Air systems use fins to increase the heat transfer rate. An analysis of the heat transfer from a fin is performed with the Fin Performance Applet.

Water systems can freeze unless additives are used. The water cooling system is usually a single loop where a water pump sends coolant to the engine block, and then to the head. The coolant will then flow to a radiator or heat exchanger and back to the pump.

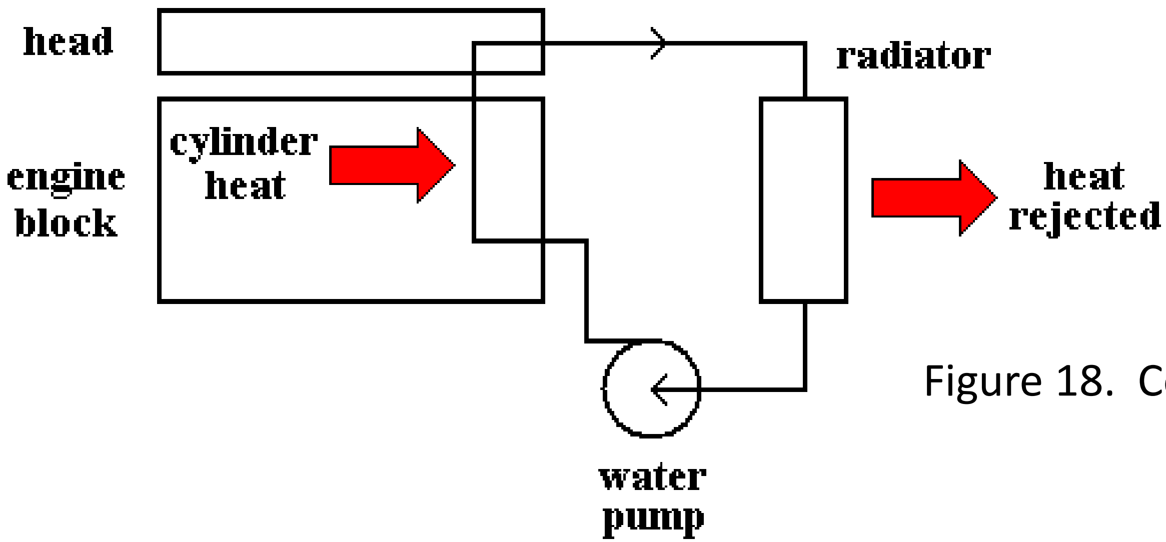


Figure 18. Cooling System Loop

During engine warmup, a thermostatically controlled valve will recycle the coolant flow through the engine block, bypassing the heat exchanger. As the engine heats up, the valve will open up, and allow the coolant to flow to the radiator.

Nucleate Boiling of Coolant

The heat fluxes and surface temperatures near the exhaust manifold and port are high enough so that nucleate boiling can occur in the coolant at those locations. The boiling heat transfer coefficients are much larger than single phase forced convection, so that the metal temperatures are lower.

For heat fluxes of the order of 1.5 MW/m2 and above, the resulting surface temperature of the cooling jacket will be about 20 to 30 C above the saturation temperature , typically, 130 C (400 K).

The convection boiling process is very complex, as bubbles formed on the jacket surface are swept downstream and condense in cooler fluid.

The surface temperature of the jacket throughout the engine block will be fairly uniform. The saturation or boiling temperature can be raised by increasing the pressure or by adding an additive with a high boiling point, such as ethylene glycol.

The design of the cooling passages in the engine block and head is done empirically. The primary design consideration is to provide for sufficient coolant flow at the high heat flux regimes, such as the exhaust valves.

The air and fuel mixture inducted into the cylinder is convectively heated by the intake manifold and valve.

The heat transfer from the valve is modeled as flow over a cylinder, and the heat transfer in the intake manifold is modeled as entry length turbulent flow in a curved pipe.

The hot exhaust gases convect heat to the exhaust manifold and the exhaust valve. The exhaust manifold conducts a portion of this heat to the head and radiates to the engine block.

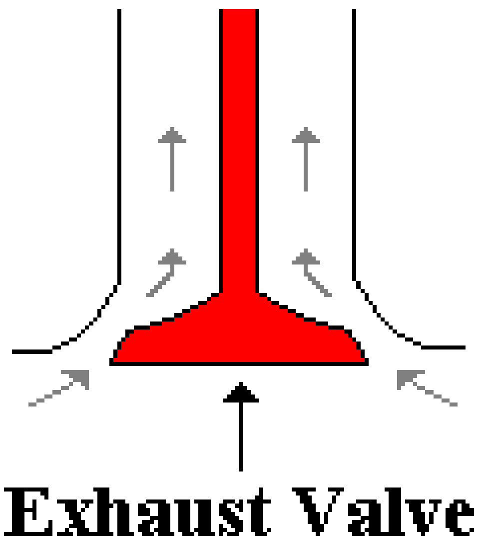
The exhaust heat transfer from the gas to the valve port, due to the pulsing flow, is about 50% greater than given by steady flow correlations.

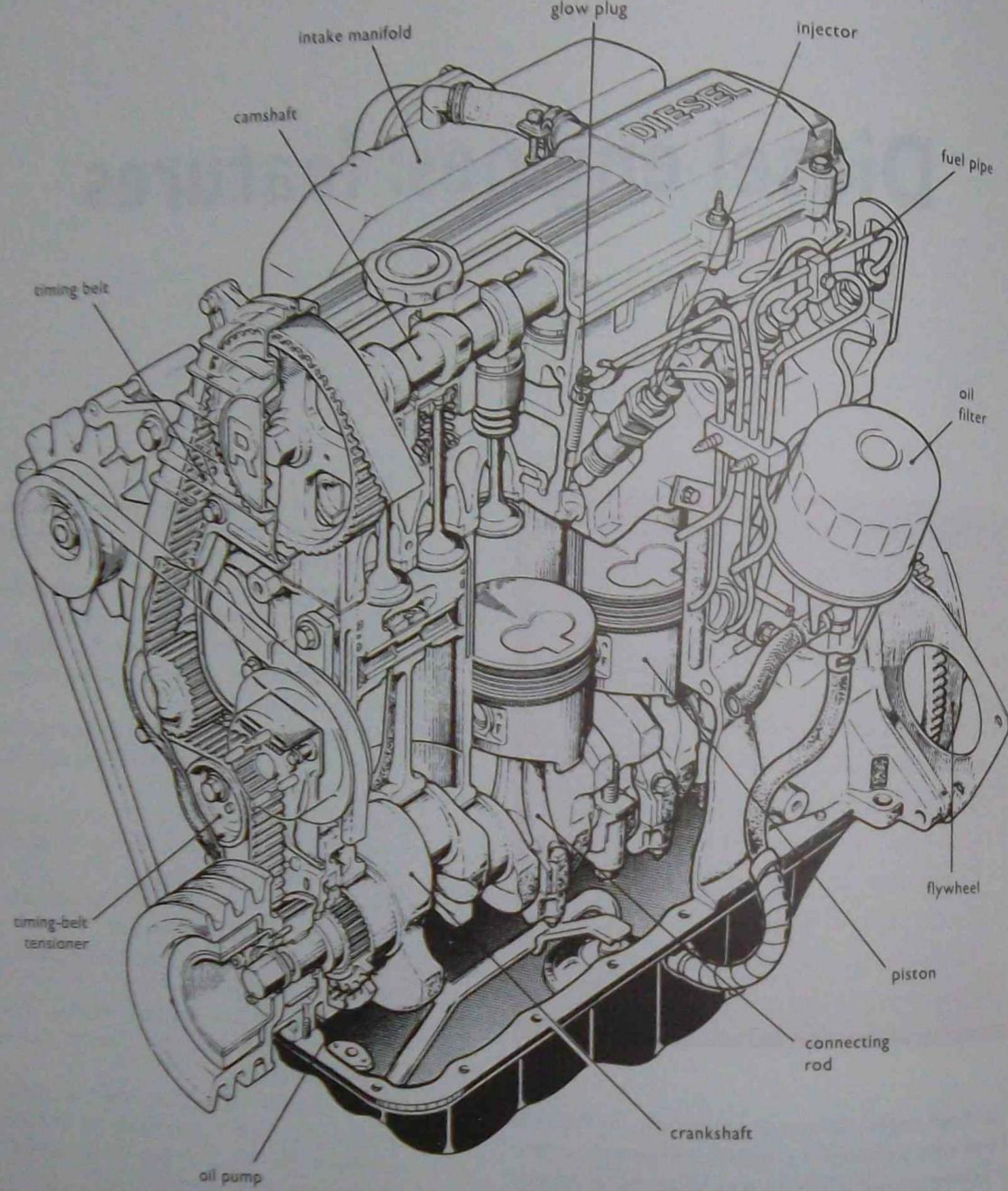
A correlation from Malchow (Malchow et al., 1979, "Heat transfer in the straight section of an exhaust port of a spark ignition engine", SAE Paper 790309) for the exhaust port heat transfer is:

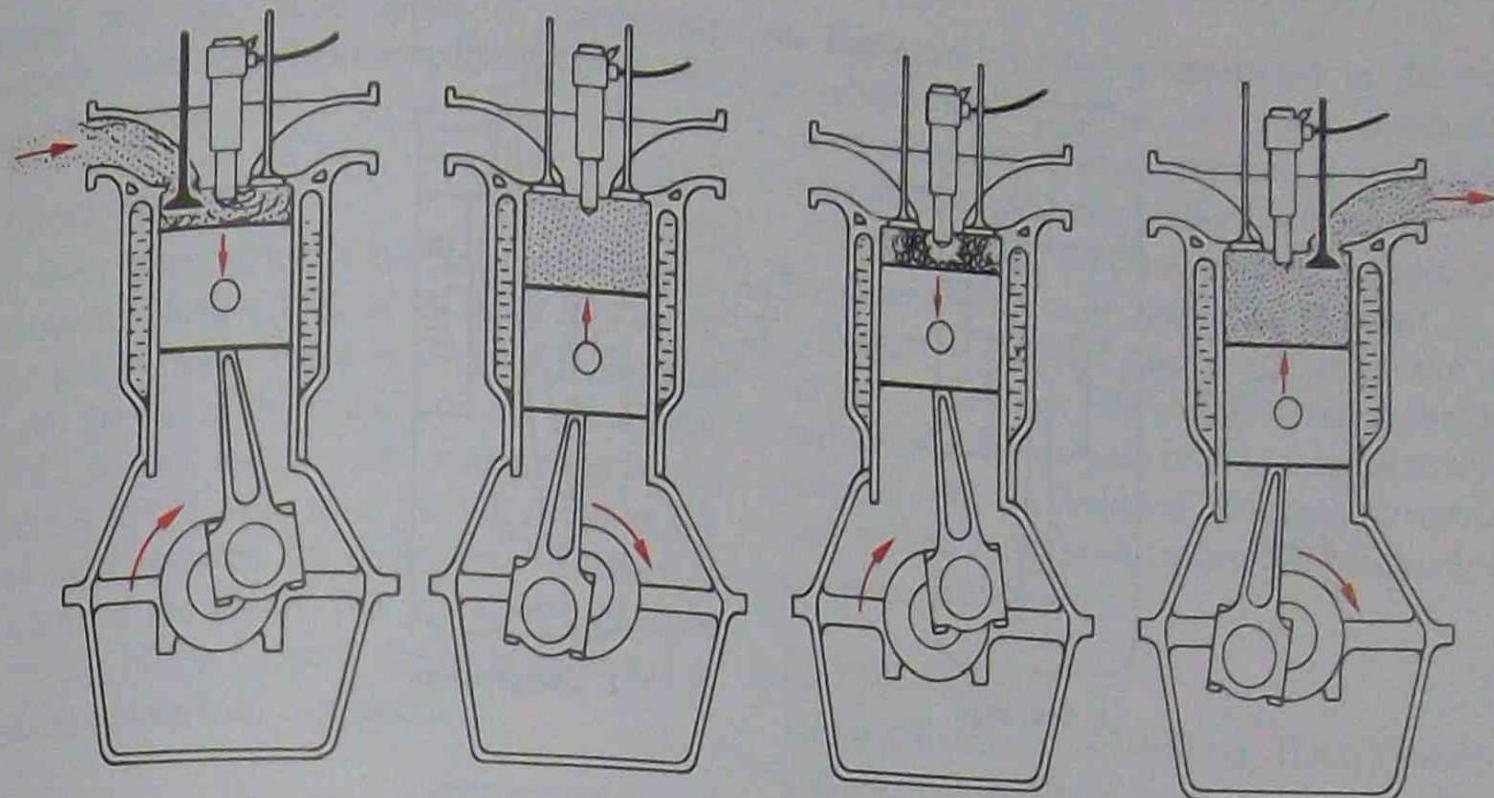
$$Nu = 0.048\, Re^{0.78}$$

where the characteristic length for the Nusselt and Reynolds numbers is the exhaust port diameter.

Figure 20. Exhaust Valve




glow plug
intake manifold
injector
camshaft
fuel pipe
timing belt
oil
filter
flywheel
timing-belt
tensioner
piston
connecting
rod
crankshaft
oil pump



1. Intake 2. Compression 3. Power 4. Exhaust

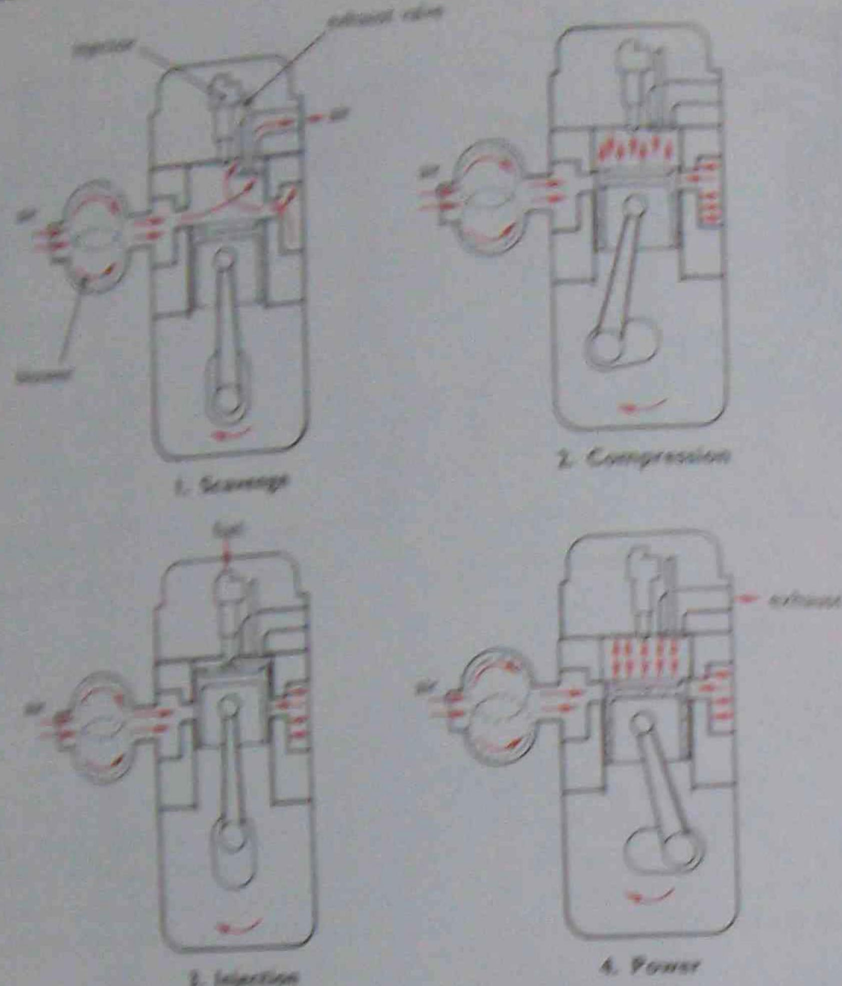
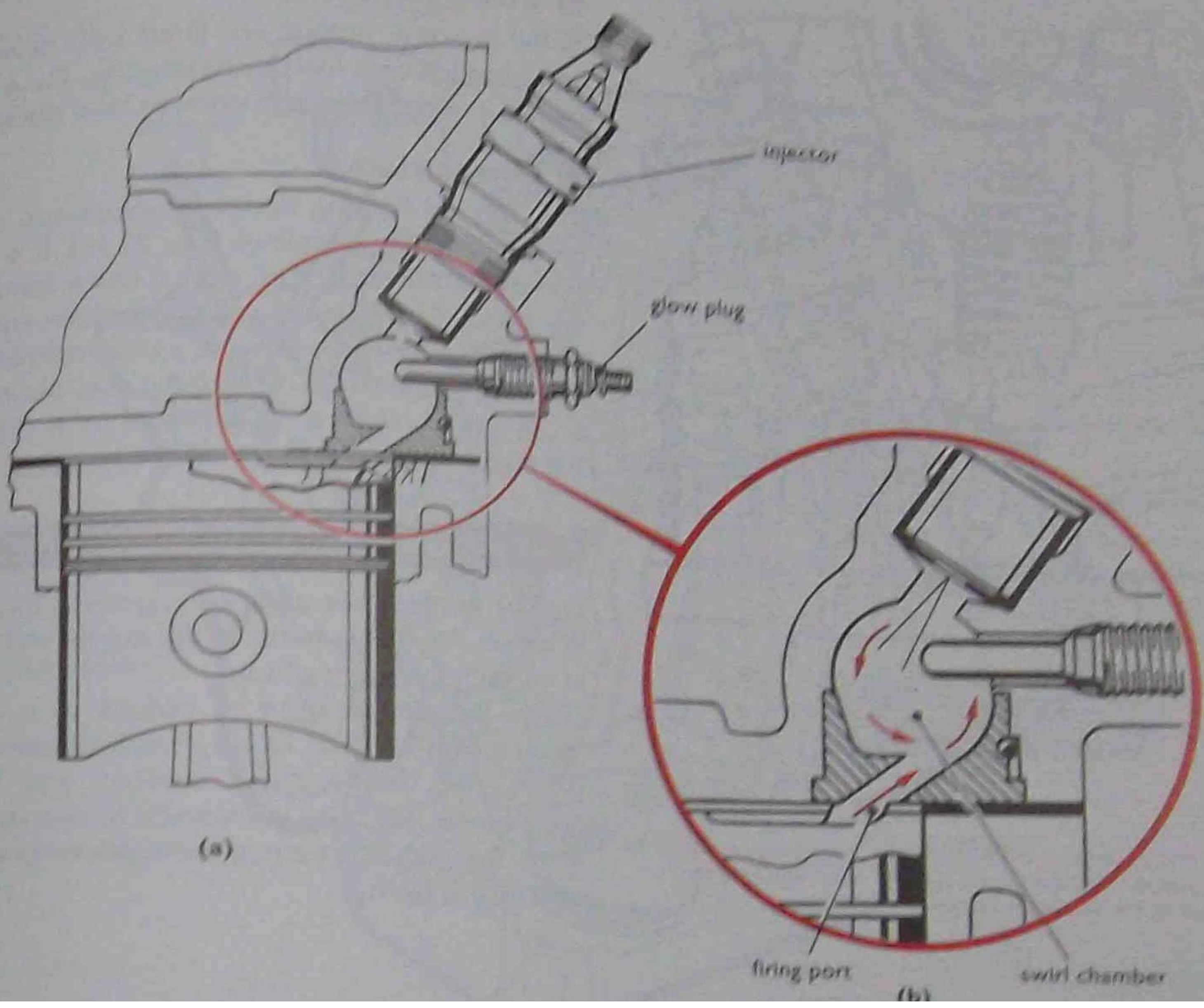


Fig. 22.3 Two-stroke diesel cycle
1. **Scavenge**—air is blown into the cylinder and exhaust gases leave the cylinder
2. **Compression**—air is blocked off at the inlet port and the exhaust valve is closed
3. **Injection**—fuel is sprayed into the cylinder
4. **Power**—the piston is forced down the cylinder and the exhaust valve opens towards the bottom of the stroke


injector
glow plug
(a)
firing port
swirl chamber
(b)

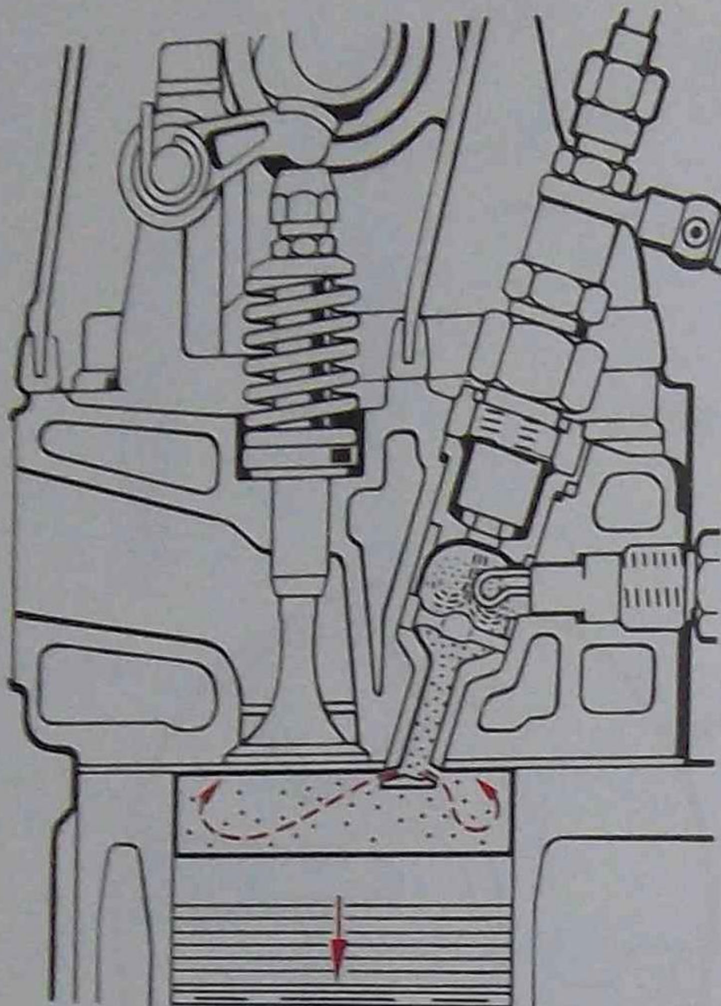


Fig. 22.10 Precombustion chamber; the injector sprays fuel into a small combustion chamber in the cylinder head which is fitted with a glow plug for starting purposes MERCEDES

Excess-fuel device

Some injection pumps are fitted with devices that provide additional fuel during starting. The extra fuel makes it easier for combustion to take place.

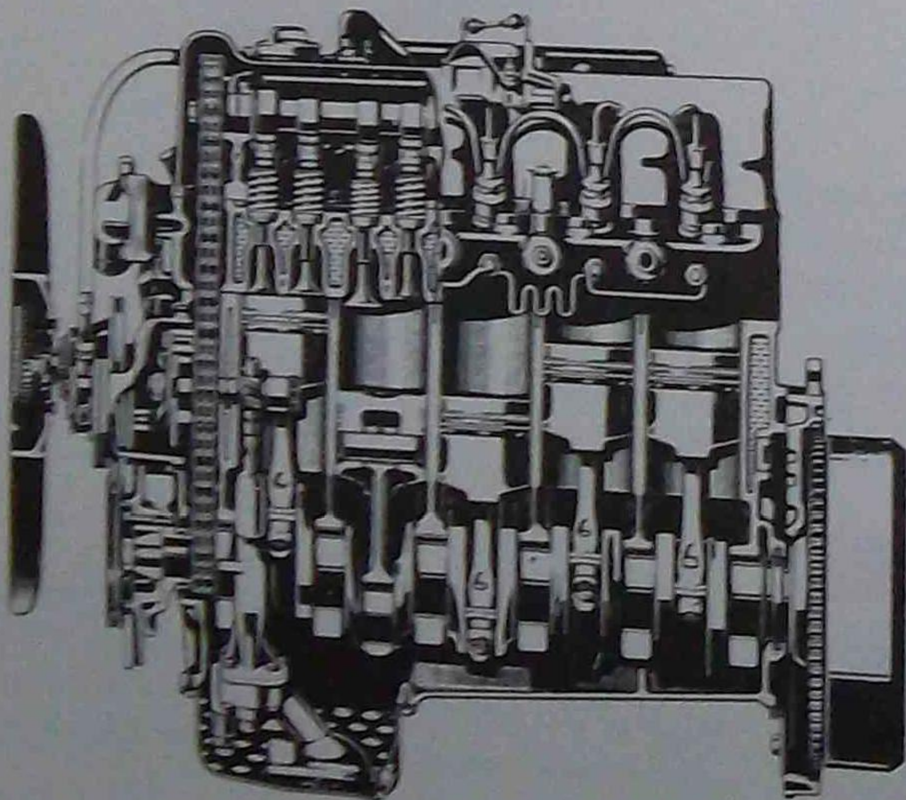
Thermostart

This type of heating device is sometimes provided on direct-injection engines (Fig. 22.15). It is fitted in the intake manifold. The device has a central body surrounded by a heater coil and ignition coil. When operated, a ball valve delivers a small amount of fuel, and this is ignited. The burning fuel heats up the air in the manifold so that heated air is drawn into the engine. This makes starting easier.

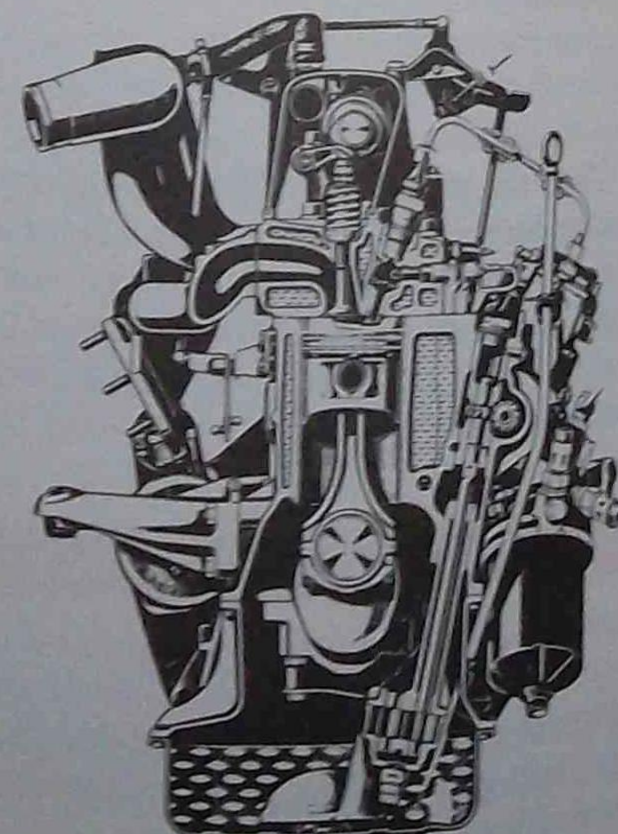
Combustion of fuel in the engine

One of the features of diesel engines is the manner in which the fuel is burnt in the engine and the way in which power is produced.

The fuel comes from an injector as a very fine spray which consists of very small droplets of fuel. When the droplets are injected into the hot compressed air, they commence to evaporate, and when the vapour reaches the right temperature, it will start to burn.



Longitudinal section



Cross-section

Fig. 22.11 A five-cylinder diesel engine; features to be seen are the overhead camshaft, chain drive, slipper-type pistons, crankshaft and bearings, valves and operating gear, injector, glow plug, water-jackets, oil pump and the injection pump

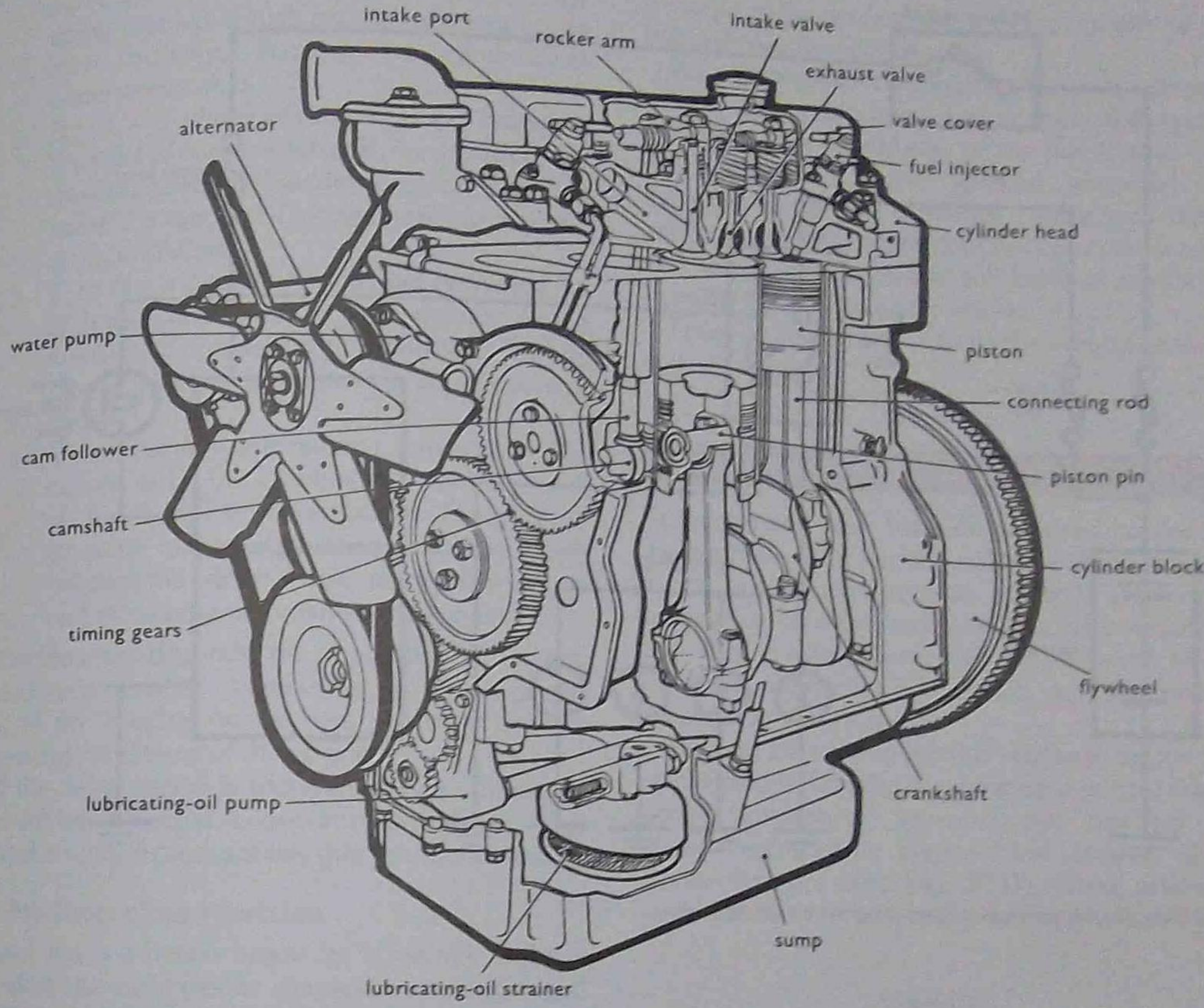


Fig. 22.12 *Basic diesel engine with direct injection; this is a larger engine* PERKINS

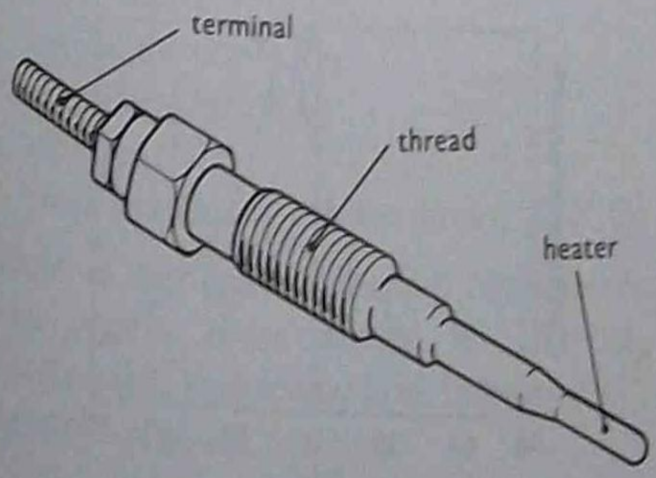


Fig. 22.13 A glow plug, which provides heat during starting
TOYOTA

The centre of the droplet is comparatively cool, so that it does not burn immediately. While this all happens very quickly, there is progressive burning of each droplet of fuel as it turns to vapour.

Once ignition is started, there is a rapid rise in both pressure and temperature, so that fuel still being sprayed from the injector commences to burn immediately. Burning then continues, even after injection has ceased, until all the fuel has been burnt.

Four phases of combustion

The combustion process in a high-speed diesel engine, as described above, can be considered as consisting of four phases or stages. Figure 22.16 illustrates these in a graph, which shows the increase in pressure in the cylinder compared with the rotation of the crankshaft from 90° before TDC to 90° after TDC. There are two lines on the graph:

1. The dotted line shows how the pressure in the cylinder would change during and after the compression stroke if no fuel were injected. The rise and fall of pressure would be uniform throughout the strokes.

surrounding the droplets from the injector spray reaches a high-enough temperature to start burning. Burning (or combustion) commences at B.

2. *B to C—a phase of rapid burning.* Following the start of combustion at B, the droplets burn rapidly, causing a sudden pressure rise until point C is reached. During this phase, burning is uncontrolled.
3. *C to D—a phase of controlled burning.* Fuel is burnt uniformly as it continues to be injected. Combustion is 'controlled' by the rate of injection until point D, when injection ends.
4. *D onwards—afterburning.* Injection has ceased at D, but combustion continues until all the fuel has been burnt. There is no increase in pressure shown on the graph because the piston is now moving downwards and increasing the volume of the cylinder.

Combustion depends on a number of factors, including combustion chamber design, the design and size of the injector nozzle, the fuel used and the operating conditions of the engine.

If the delay period is too long, a large amount of fuel will be subjected to uncontrolled burning. This causes a sudden pressure rise that produces a knock.

Products of combustion

Diesel fuel is a hydrocarbon (as is petrol), and this provides the combustible elements of hydrogen and carbon.

The products of combustion in a diesel engine that is operating efficiently and in good mechanical condition are:

1. water vapour (H_2O)
2. carbon dioxide (CO_2)
3. unused oxygen (O_2)
4. nitrogen (N)
5. small amounts of carbon monoxide (CO)

Because of the high combustion temperatures in a diesel engine, there should be little or no hydrocarbon (HC) remaining from unburnt fuel if the fuel system is in good condition.

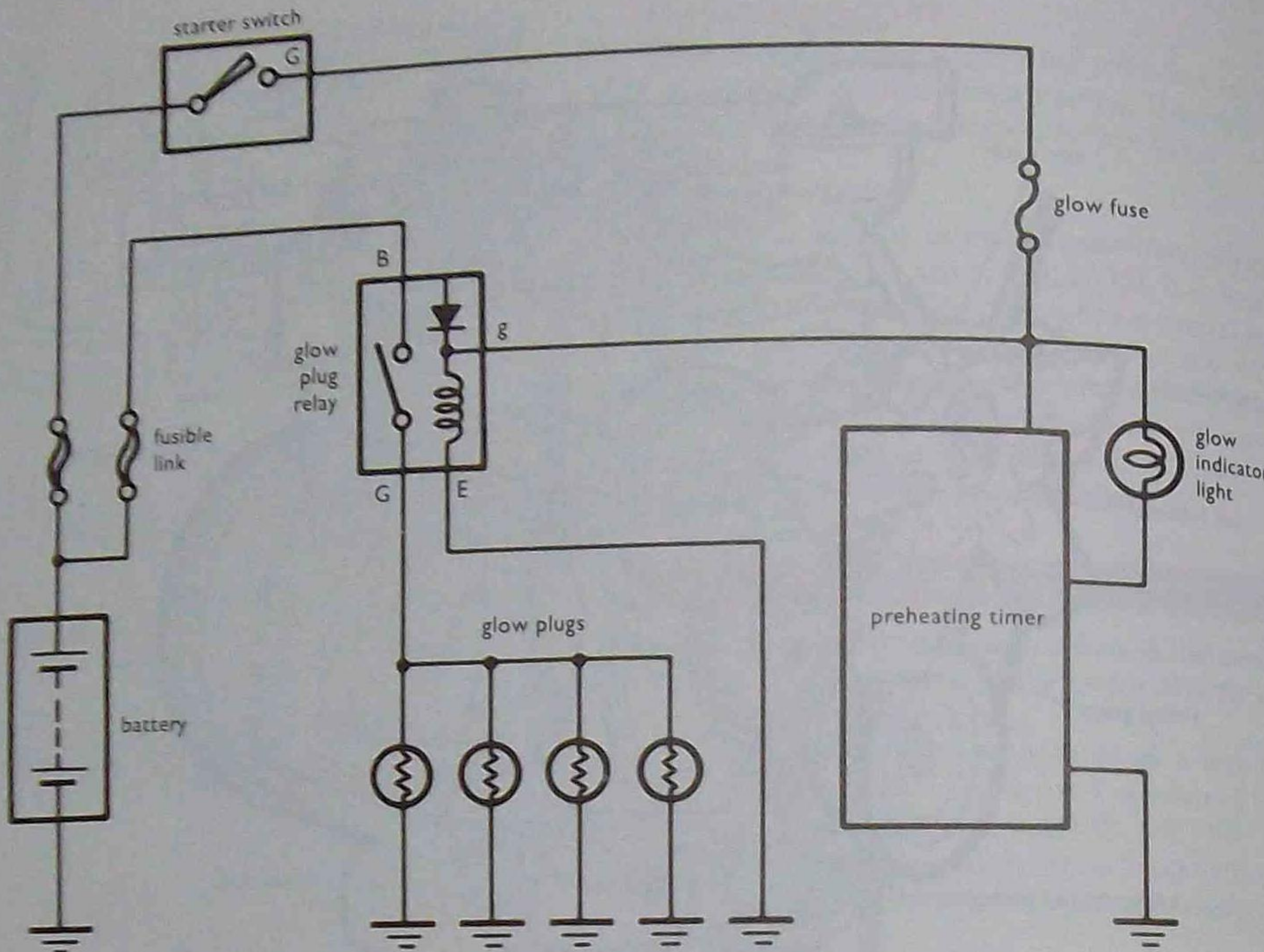


Fig. 22.14 *Electrical circuit for glow plugs with a timing device* TOYOTA

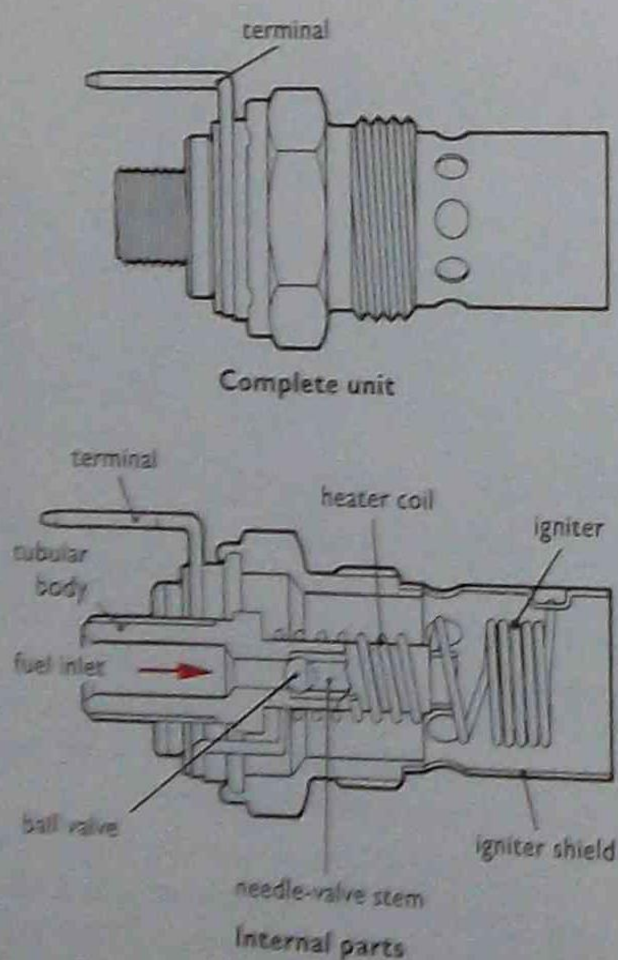


Fig. 22.15 Thermostart starting aid LUCAS/CAV

2. The solid line shows the change in pressure during the compression and power strokes, with the pressure increasing rapidly as a result of combustion.

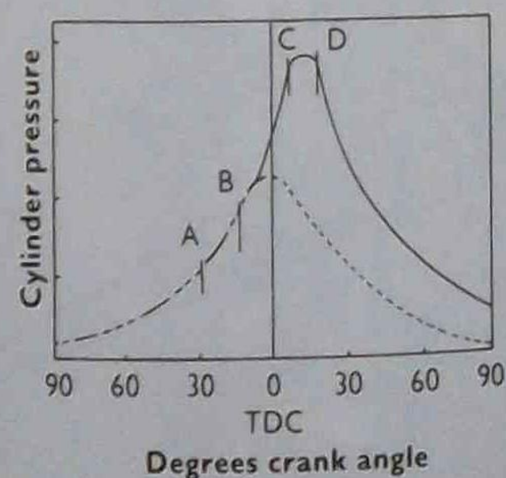


Fig. 22.16 *Phases of combustion in a diesel engine*

The four phases of combustion indicated on the graph are as follows:

1. *A to B—the delay period.* Injection commences at A, but combustion does not occur immediately. This is delayed until the vapour

which would otherwise go to waste, is used to increase the mass of the air charge to the cylinders. Figure 22.18 shows a basic turbocharger arrangement.

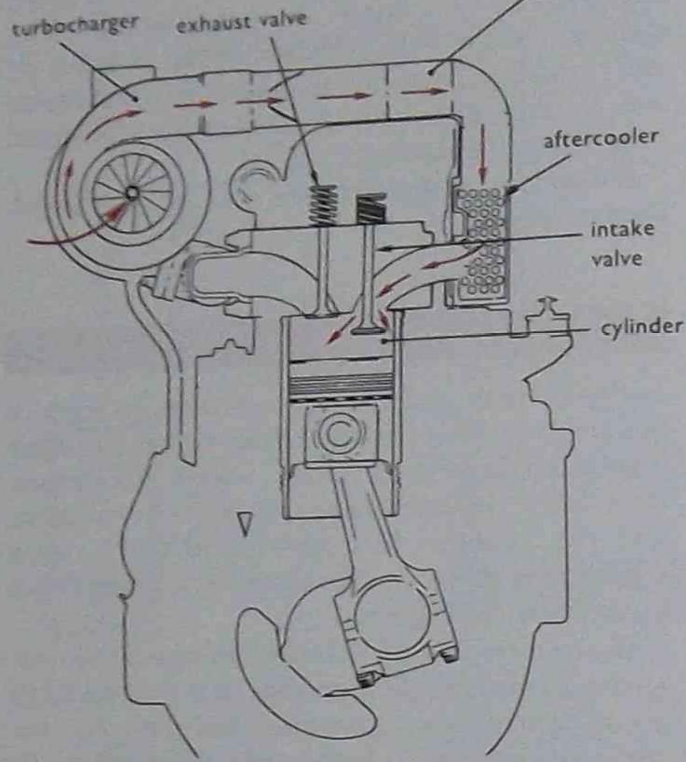


Fig. 22.18 *Turbocharger with aftercooler showing intake airflow* CUMMINS

The turbocharger has a rotor, which consists of a shaft with a turbine wheel on one end and a compressor wheel or impeller on the other. The rotor is mounted in a housing to form the exhaust-driven turbine at one end and the compressor at the other. Exhaust gases directed through the turbine cause the rotor to turn at high speeds and operate the compressor.

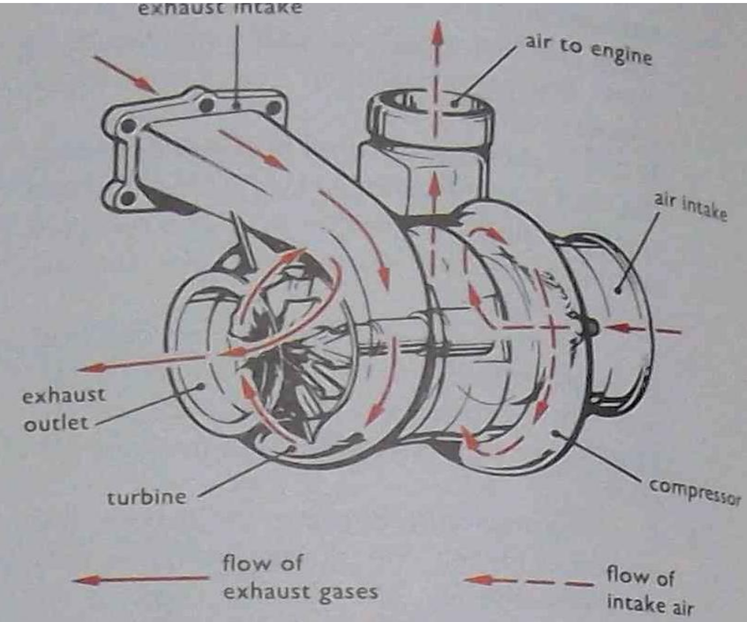
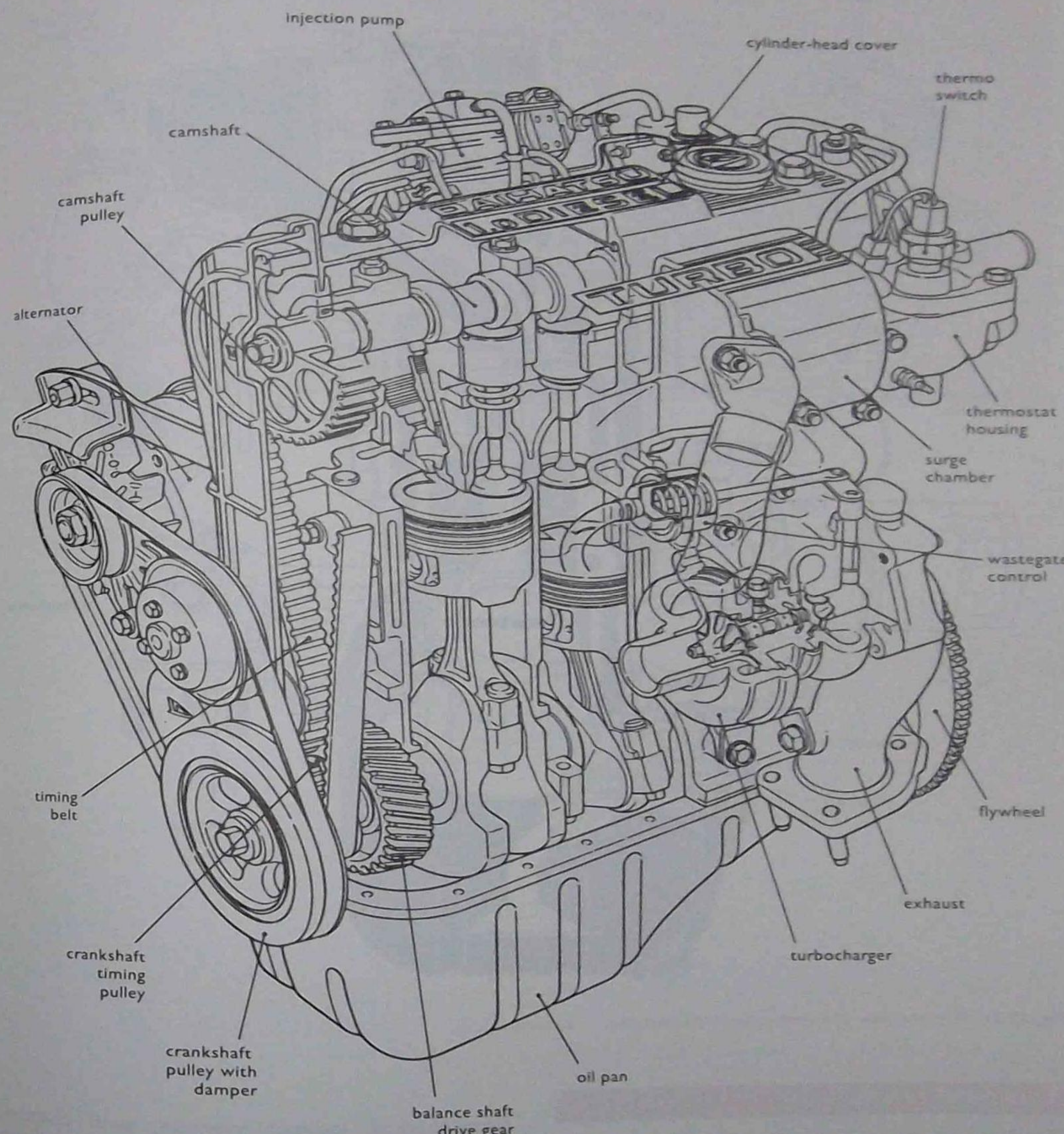


Fig. 22.19 *Turbocharger with parts named* CUMMINS

engine flows around the turbine housing and is then directed inwards by the nozzle blades, so that it has a high velocity before entering the turbine wheel.

Turbochargers not only use the waste energy in the exhaust gas, but they also respond to engine demands. If more fuel is injected into the engine, the energy in the exhaust gases will increase. Turbocharger speed will increase and there will be a greater mass in the air charge entering the engine. However, turbocharger speed must be limited to prevent excessive speed and pressure, and this is done with a wastegate control. (Refer to Chapter 15 for more information on turbochargers and wastegate control.)


injection pump
cylinder-head cover
thermo switch
camshaft
camshaft pulley
alternator
TURBO
thermostat housing
surge chamber
wastegate control
timing belt
flywheel
exhaust
turbocharger
crankshaft timing pulley
crankshaft pulley with damper
oil pan
balance shaft drive gear

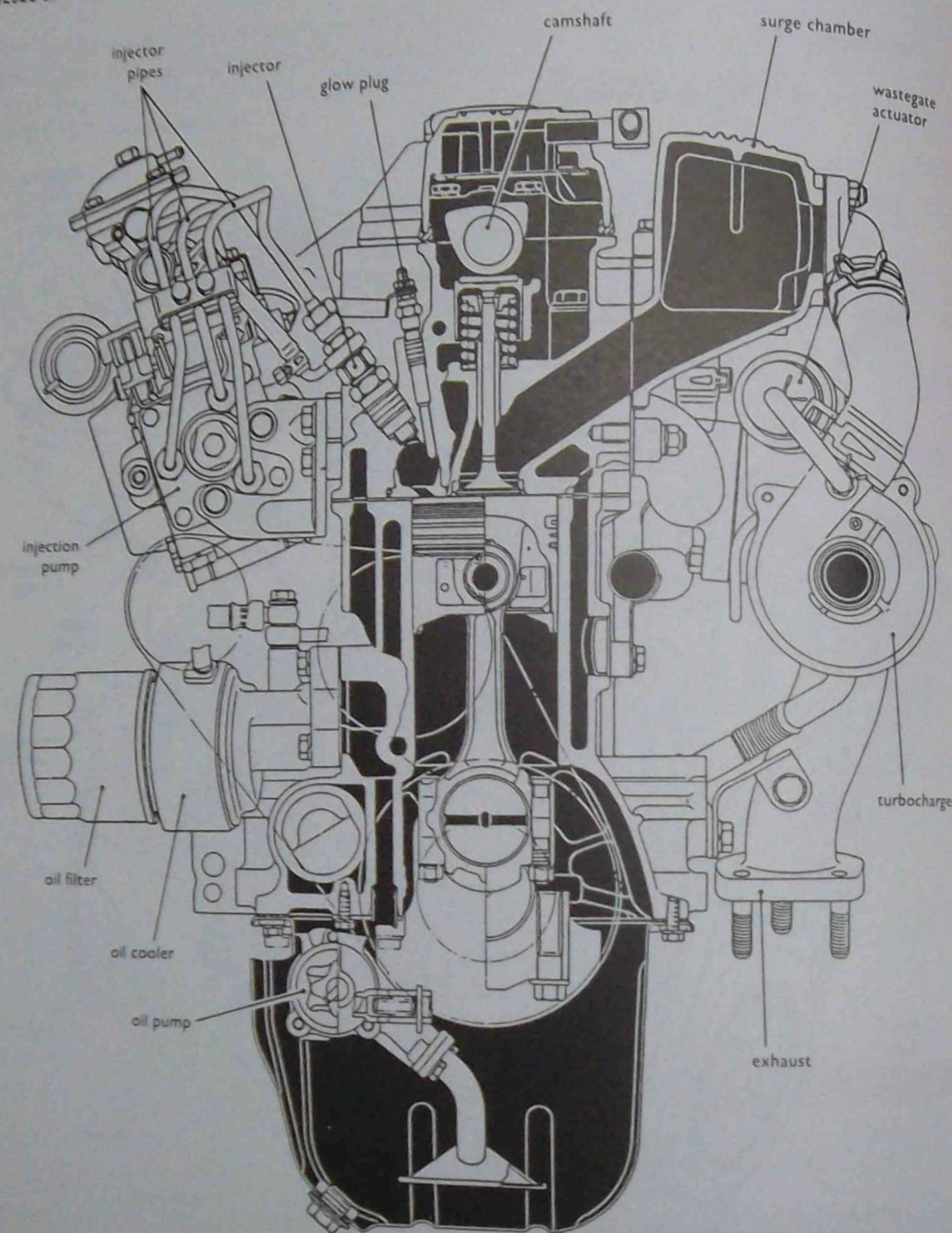


Fig. 22.21 Sectional view of a three-cylinder diesel engine DAIHATSU

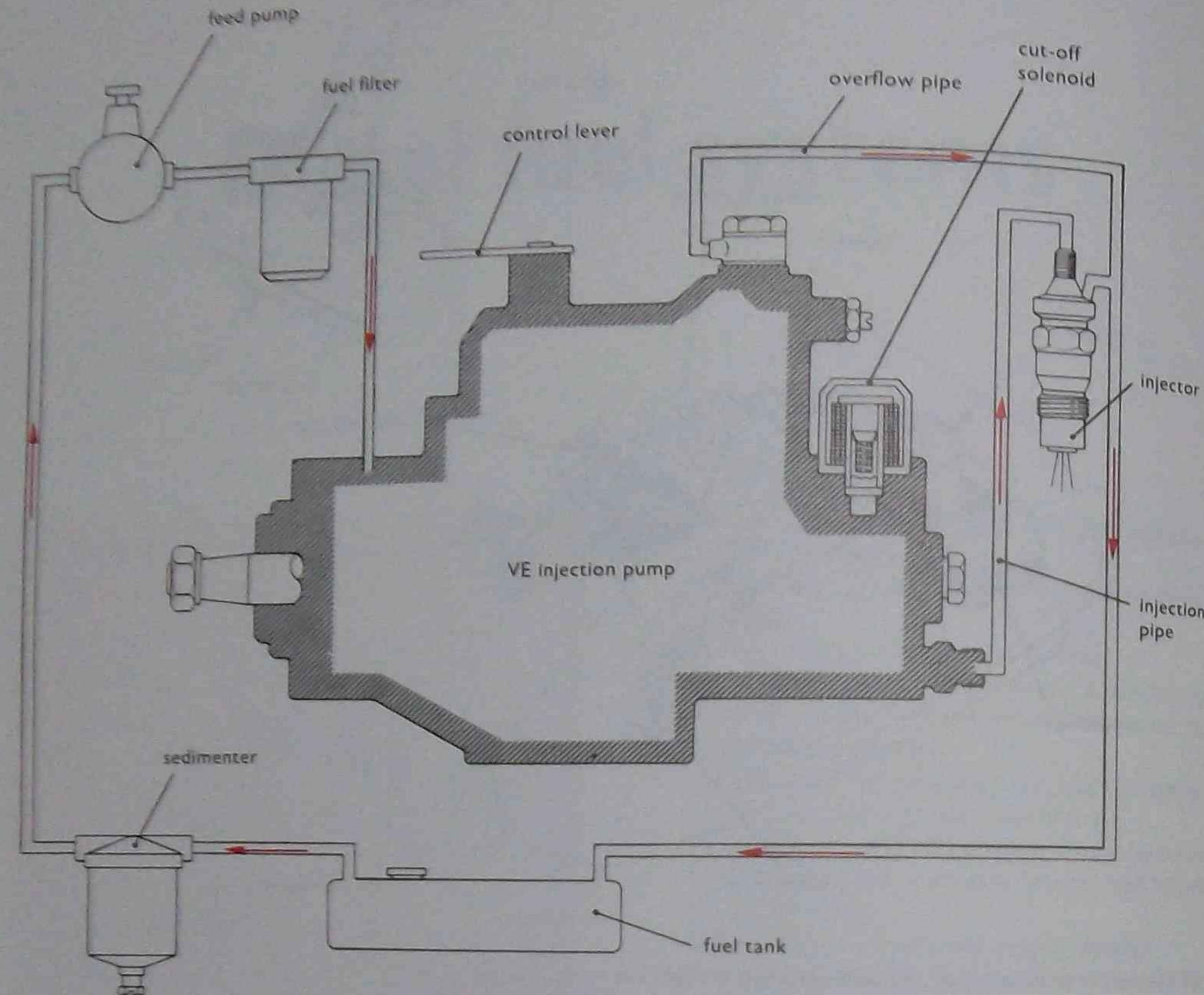


Fig. 23.2 Schematic arrangement of a fuel system with a distributor-type pump ZEXEL

distributor-type pump ZEXEL

from the top of the filter to the tank, and a leak-off pipe from the injector. The fuel flow in the system is marked on the diagram.

Figure 23.4 shows an engine with the components installed. This type of engine would be used in a commercial vehicle.

Fuel supply pumps

All diesel fuel systems have some form of supply pump which takes fuel from the tank and delivers it to the injection pump. They can be referred to as *supply pumps*, *lift pumps* or *transfer pumps*. Vane pumps, diaphragm pumps, plunger pumps and gear pumps are all used, but this depends on the type of system.

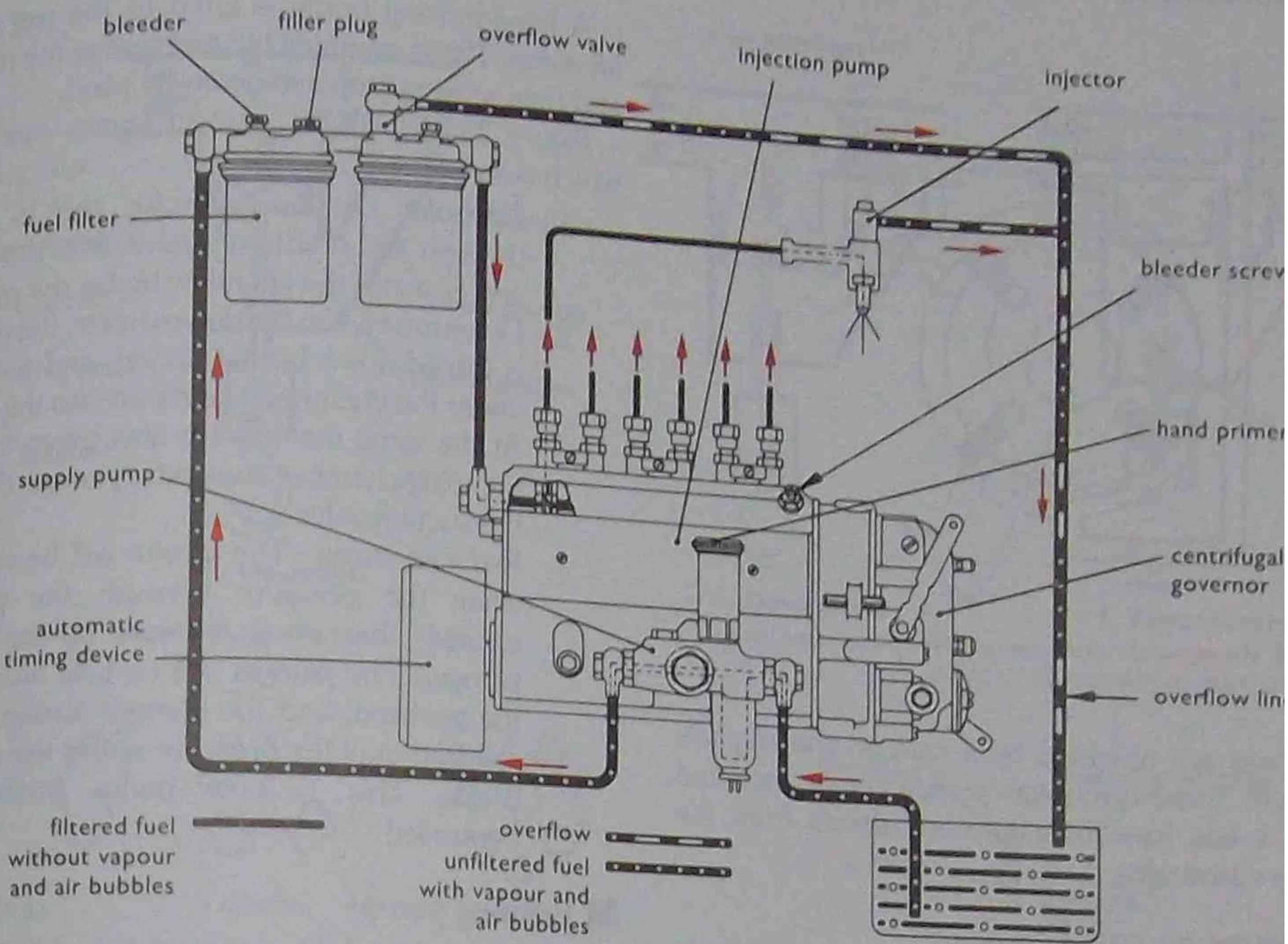
Vane pumps

Vane pumps are used with distributor-type injection pumps. The vane pump is located inside the injection pump housing. It is used to take fuel from the fuel tank and supply it to the pumping element.

The vane pump (Fig. 23.5) is driven by the injection pump shaft. It has a rotor which is mounted off-centre in the pump housing. Slots in the rotor carry the vanes, which slide backwards and forwards as the rotor turns. Fuel taken into the pump inlet is carried around between the vanes and discharged from the outlet.

Diaphragm pumps

These are similar to the diaphragm pumps used on petrol engines with carburettors, but they have a hand priming lever. They are mounted to the cylinder



ig. 23.3 *Diagram of a fuel system with an in-line injection pump* BOSCH

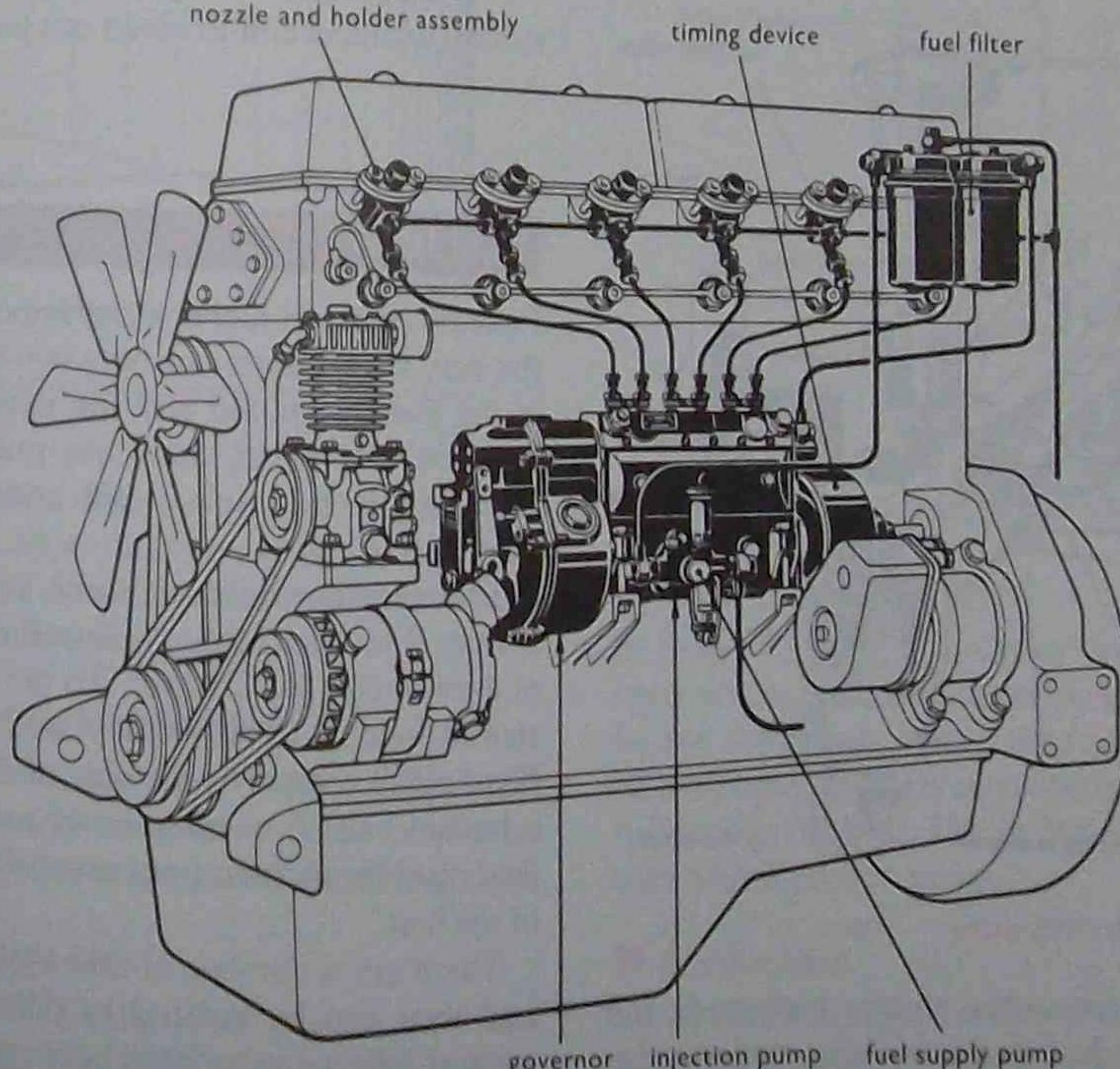


Fig. 23.4 *An in-line injection pump and other fuel system components installed on a commercial vehicle engine* BOSCH

on the injection pump's camshaft (Fig. 23.6).

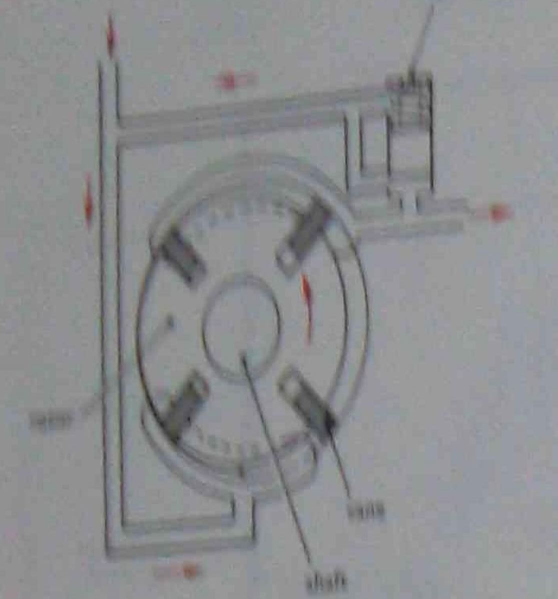


Fig. 23.5 Arrangement of a vane-type feed pump used with a distributor-type injection pump

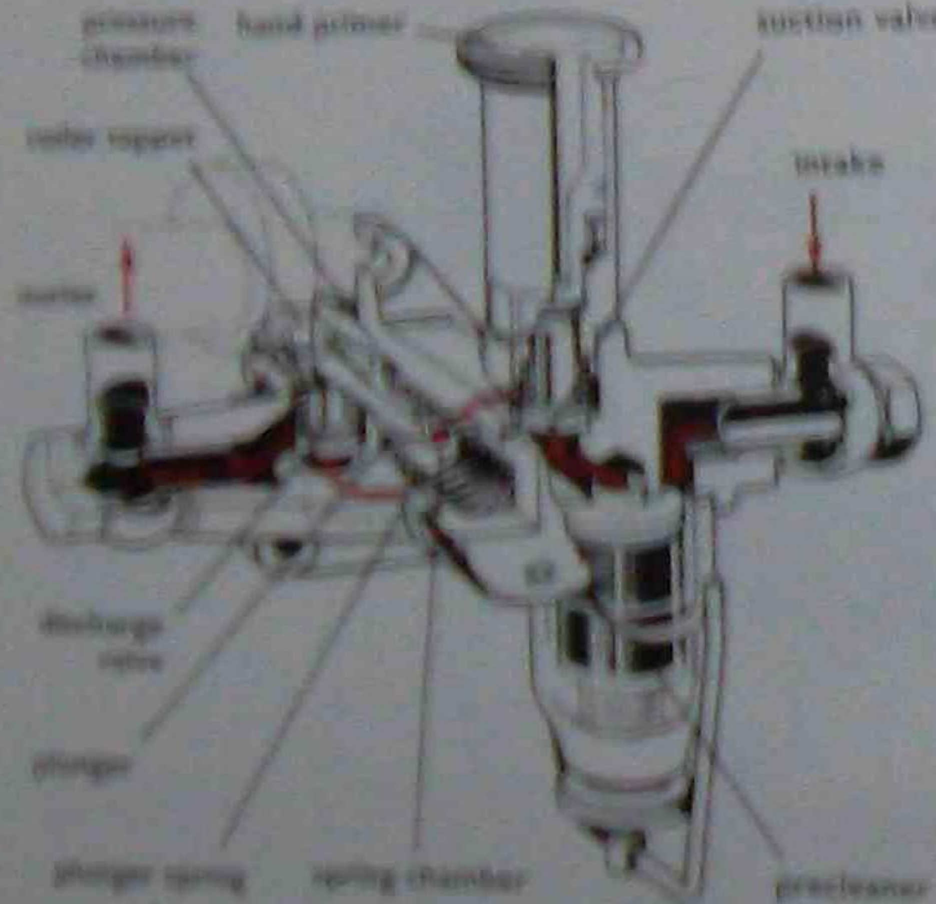


Fig. 23.6 Plunger fuel supply pump

The eccentric moves the plunger backwards and forwards to take fuel in through the suction valve and pump it out through the discharge valve, so maintaining a flow of fuel.

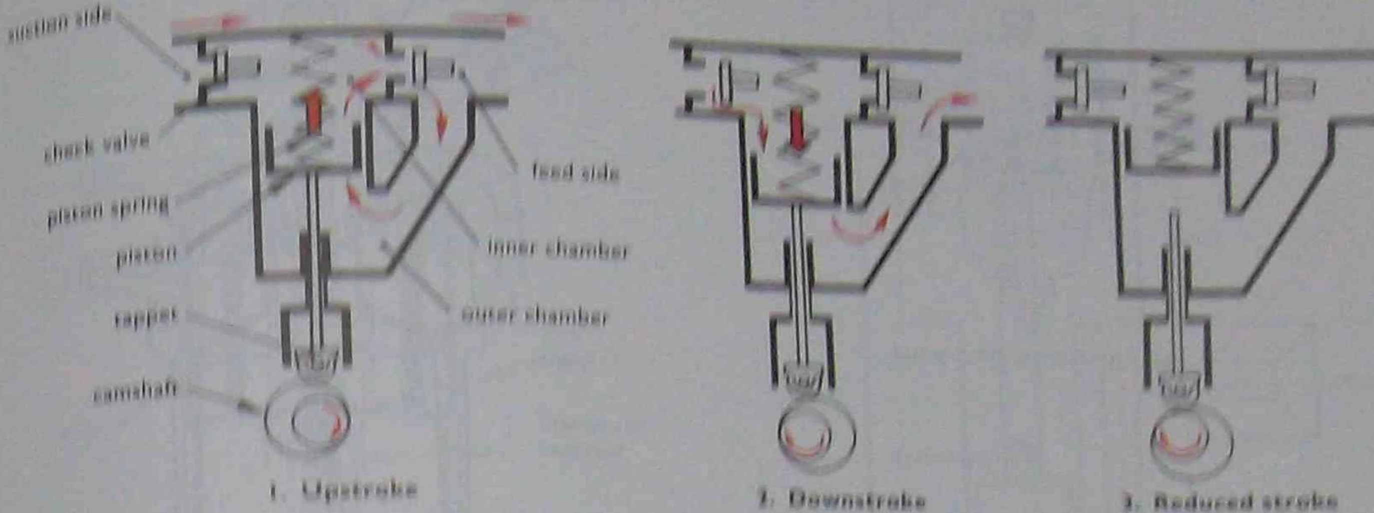


Fig. 13.7 *Plunger supply pump operation* ZEXEL

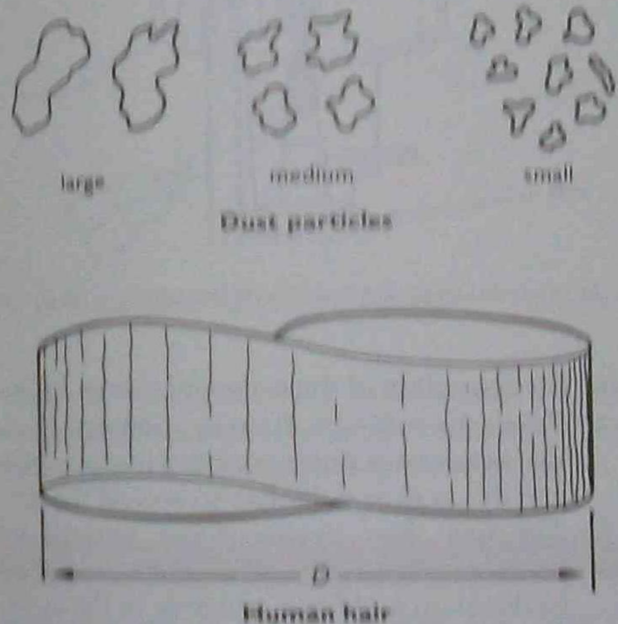


Fig. 13.8 *Size of dust particles compared with human hair: D diameter of hair*

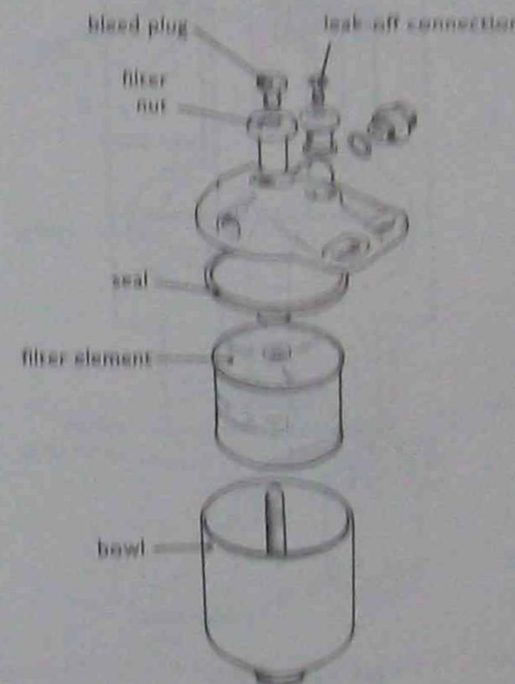


Fig. 13.9 *Fuel filter with replaceable element* LUCAS/CAV

Faulty sealing of a filter on the suction side of a

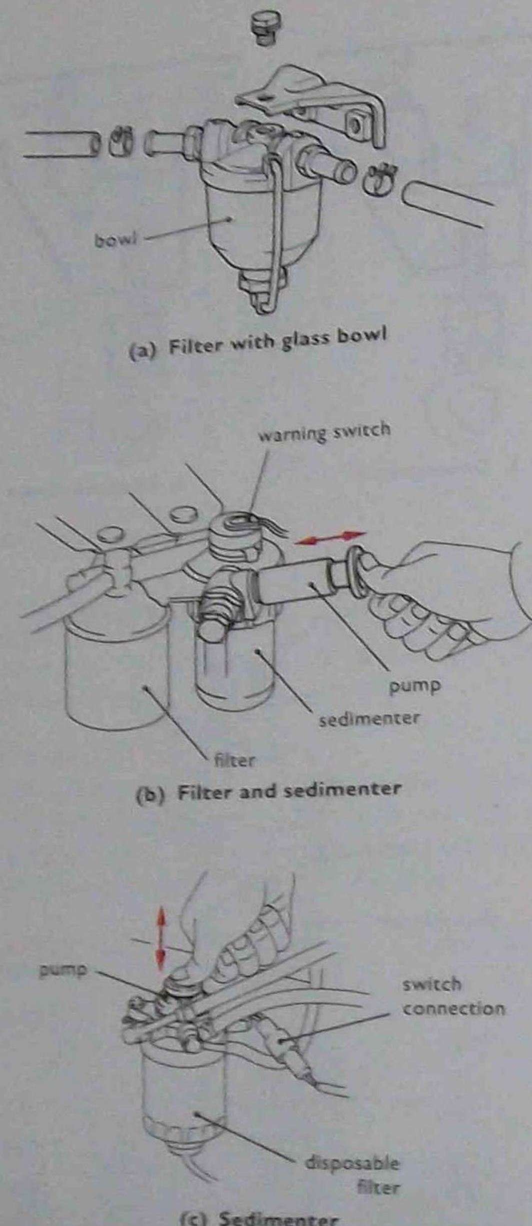


Fig. 23.10 Types of diesel fuel filters TOYOTA

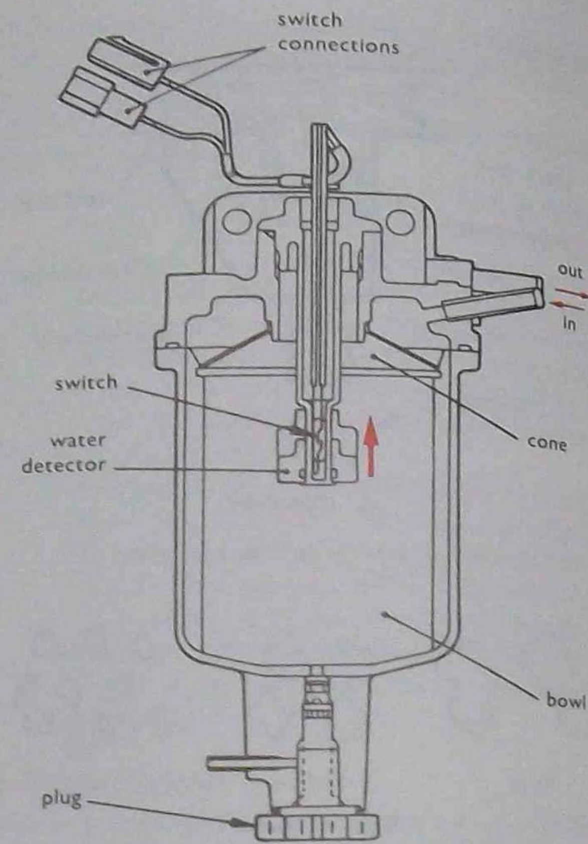


Fig. 23.11 *Sedimenter with water-level detector* FORD

The filtering action of the sedimenter is as follows:

1. When the engine is running, fuel that enters the sedimenter flows over the top of a cone, and this acts as a diffuser to spread the fuel.
2. The fuel then passes down towards the bottom of the sedimenter where it changes its direction and flows upwards to the outlet.
3. As the fluid changes direction, any heavy particles or water in the fuel fall to the bottom to remain as sediment, and this can be drained off.

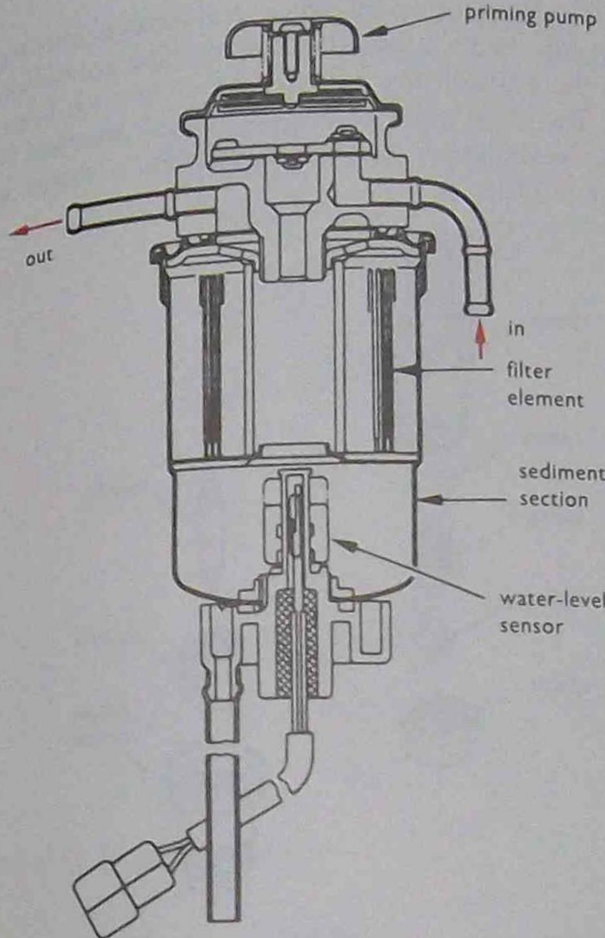
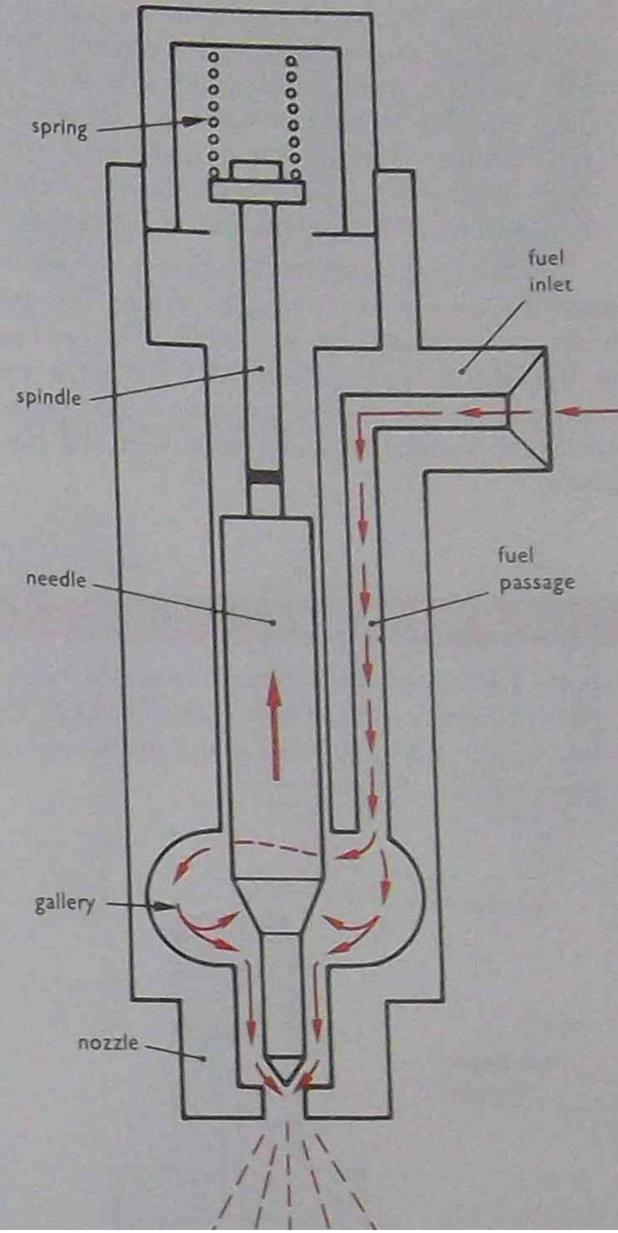


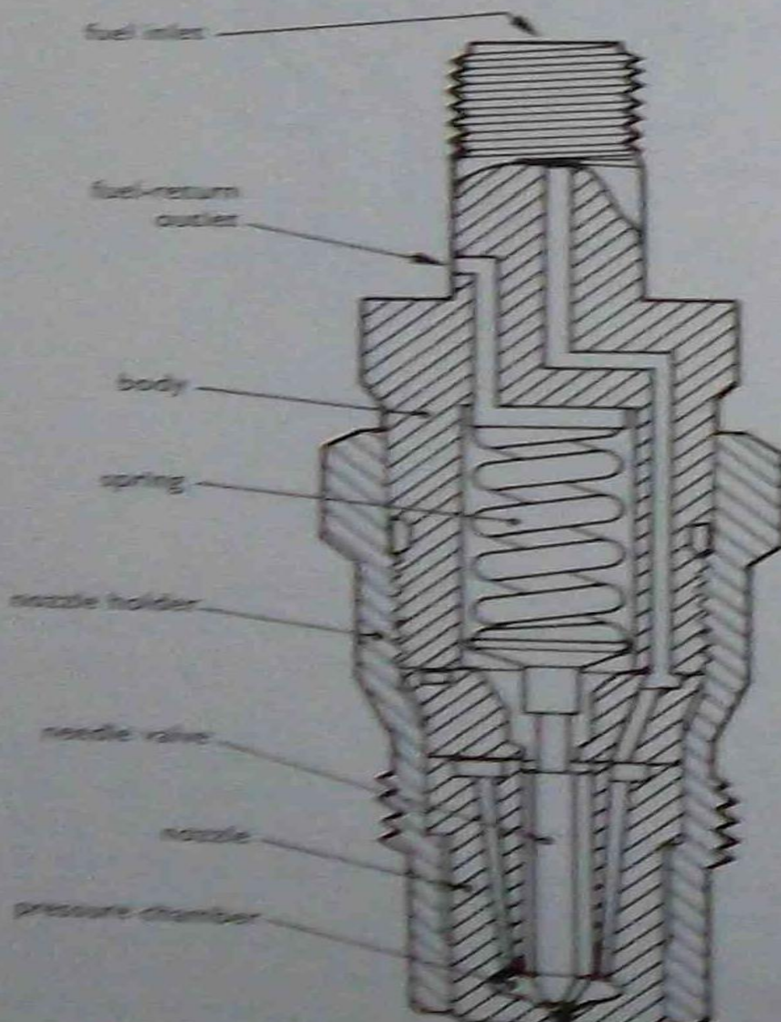
Fig. 23.12 *Combined fuel filter and sedimenter* DAIHATSU

Edge filters consist of a small square bar which is a close fit in a bore. Fuel is filtered when it passes through the narrow gap between the two parts.



Types of injectors

Figure 23.14 is a sectional view of the type of injector used in passenger cars and light commercial vehicles. It operates in the same way as the simple injector just described.



This injector is threaded, and screws into the cylinder head. It has the injector pipe connection at the top with the leak-off pipe connection below it. The other parts of the injector are identified in the illustration. A dismantled injector is shown in Figure 23.15.

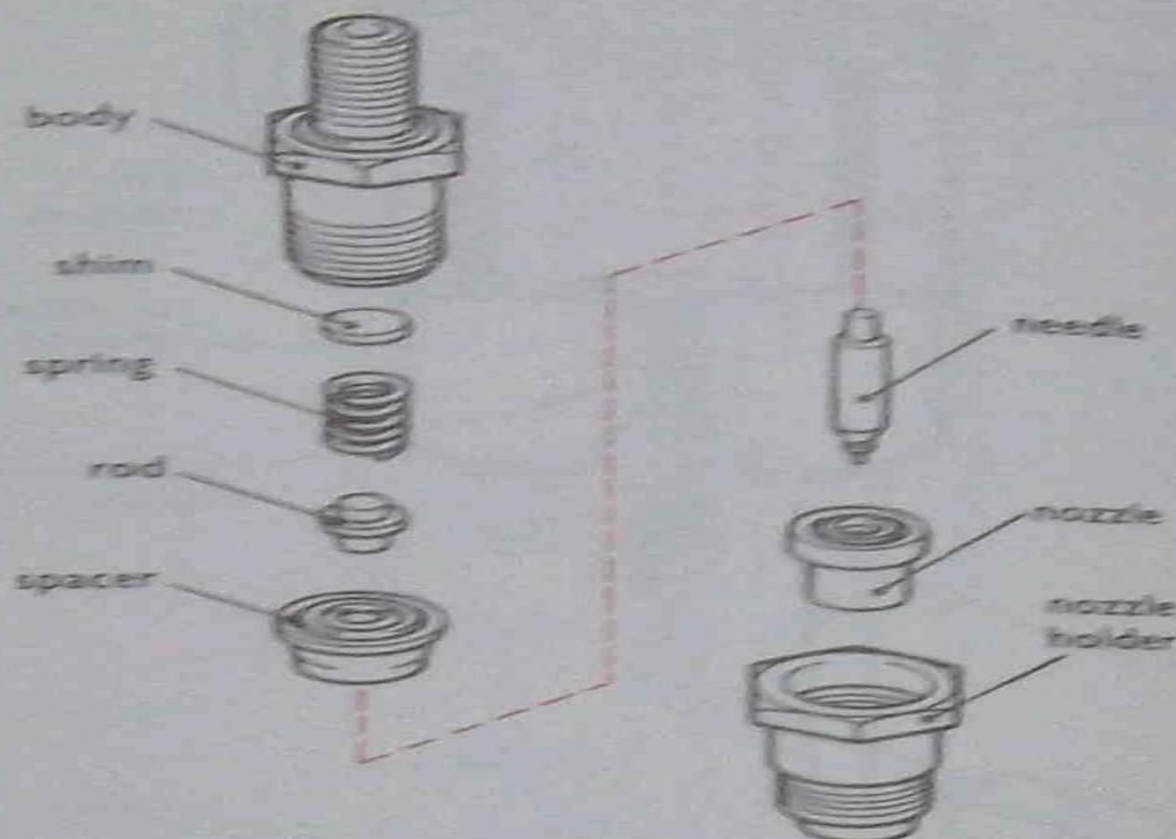


Fig. 23.15 *Dismantled parts of an injector* TOYOTA

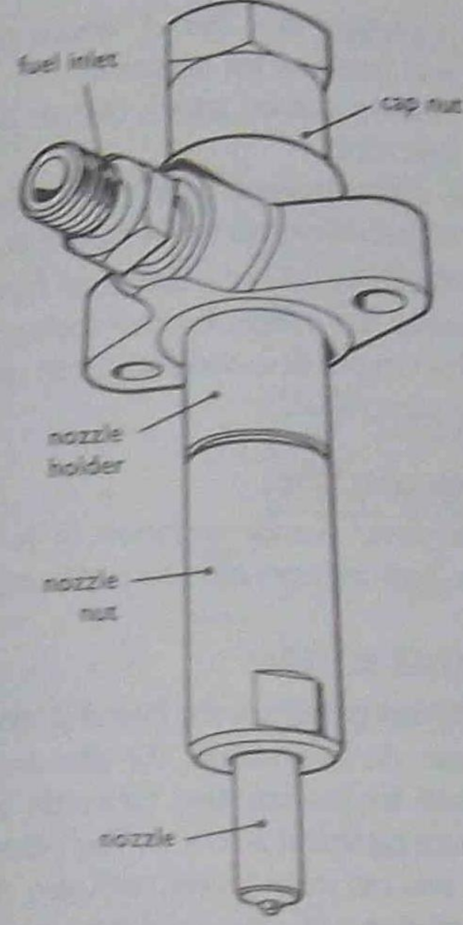


Fig. 23.16 *Flanged-type injector*

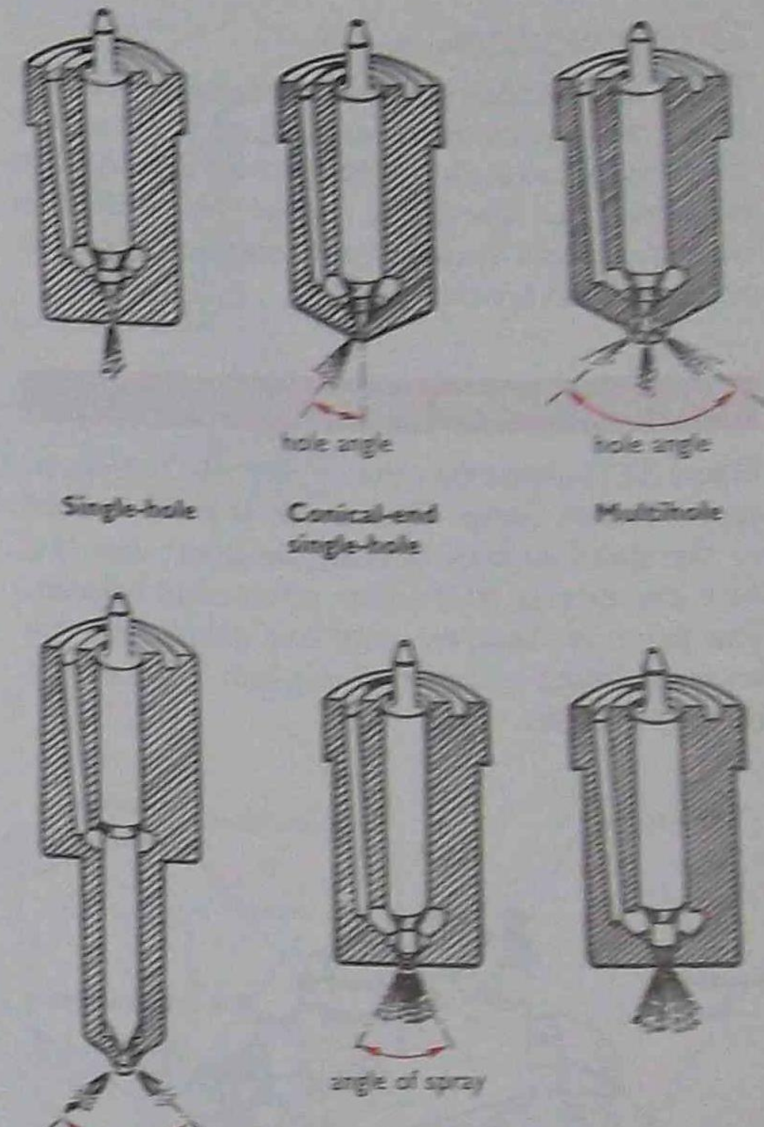


Fig. 23.16 Flanged-type injector

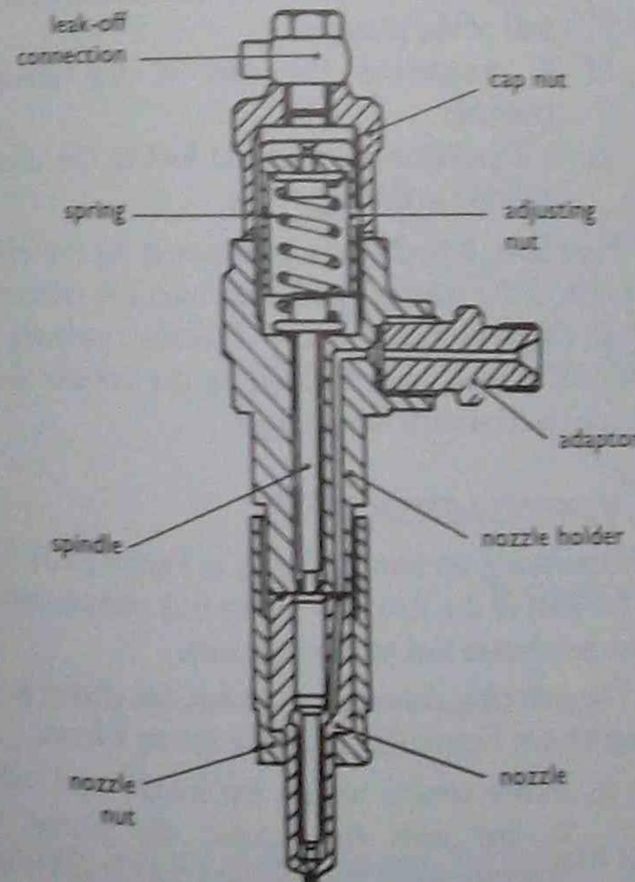


Fig. 23.17 *Internal construction of a flanged-type injector* LUCAS/CAV

☐ MULTIHOLE NOZZLES

These nozzles have two or more holes drilled in the end of the nozzle. The number of holes and their size and position depend on the requirements of the engine.

Fig. 23.18 *Types of injector nozzles* LUCAS/CAV

☐ LONG-STEM NOZZLES

This type of nozzle has a long stem which is an extension of the underside of the nozzle. The end of the stem carries the normal holes and valve seat. The long stem enables the part of the nozzle that has fine clearances (between the needle and the nozzle) to be kept away from the combustion chamber so that this part can operate in a comparatively cooler area of the cylinder head.

☐ PINTLE NOZZLES

This type of nozzle has a much larger hole than other types, and the end of the needle valve is formed into a pin or 'pintle' which protrudes through the hole. By modifying the shape and size of the pintle, injectors can produce different spray patterns. The spray can be varied from a small hollow cone to a hollow cone with an angle of 60°.

Pintle nozzles are designed for use in engines with indirect injection, that is, for those that have combustion chambers of the air cell, swirl chamber or precombustion type.

Distributor-type injection pump

Figure 23.19 shows the external parts of a distributor-type injection pump. This design of pump is fitted to the diesel engines of many passenger cars and light commercial to medium commercial vehicles. The pump is flange-mounted and driven from the engine's timing chain or timing belt. It rotates at camshaft speed.

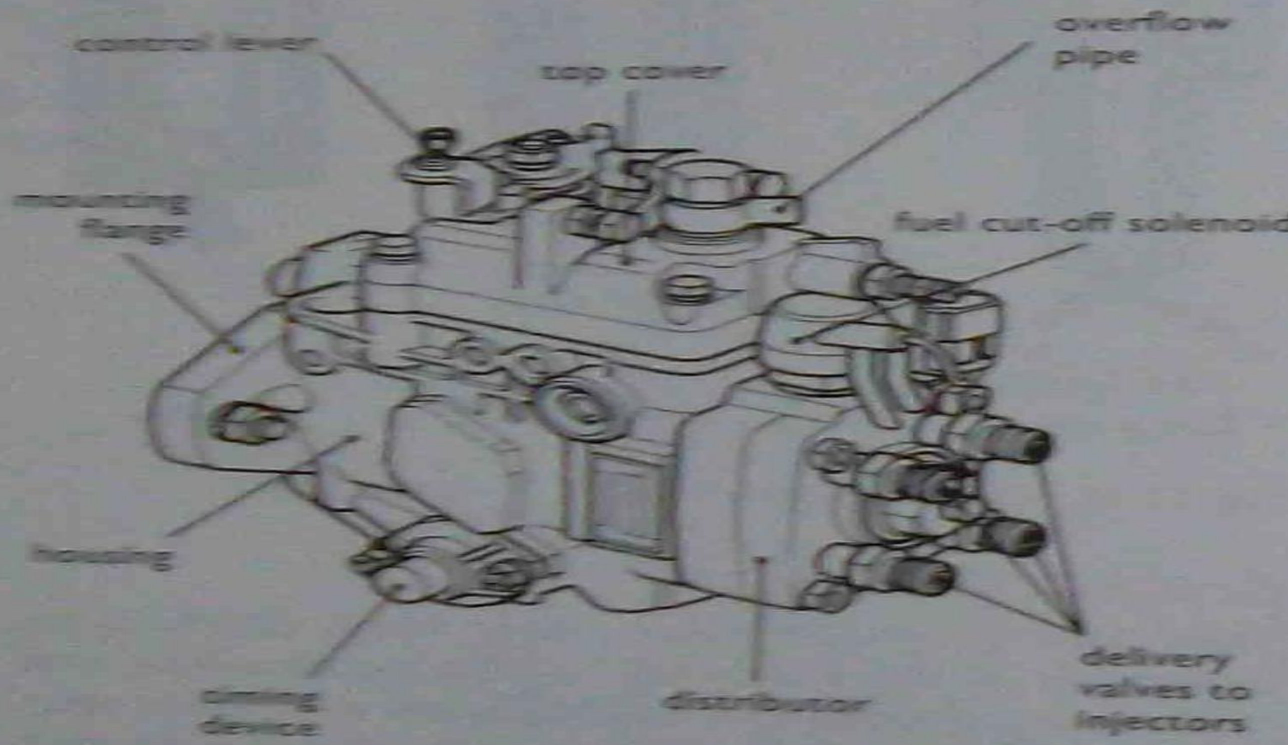


Fig. 23.19 *External parts of a distributor-type injection pump* TOYOTA

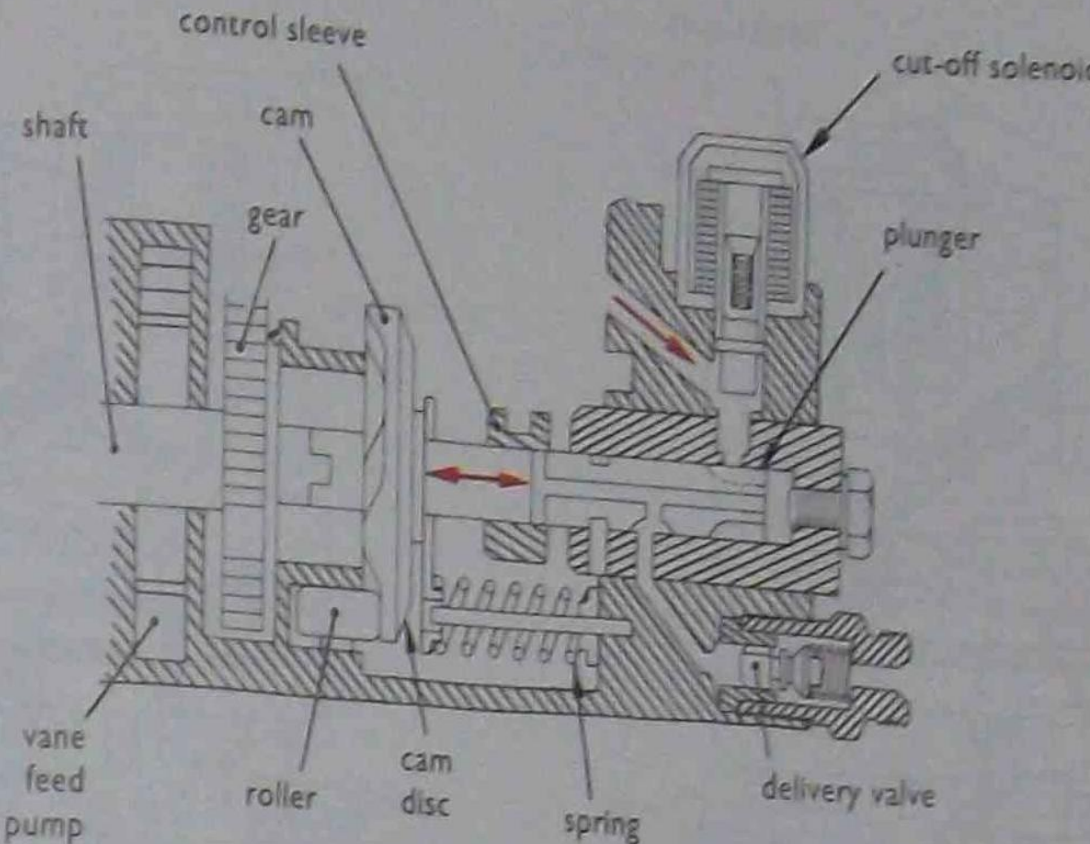


Fig. 23.20 *Schematic arrangement of a distributor-type injection pump* ZEXEL

Fig. 23.20 … type injection pump ZEXEL

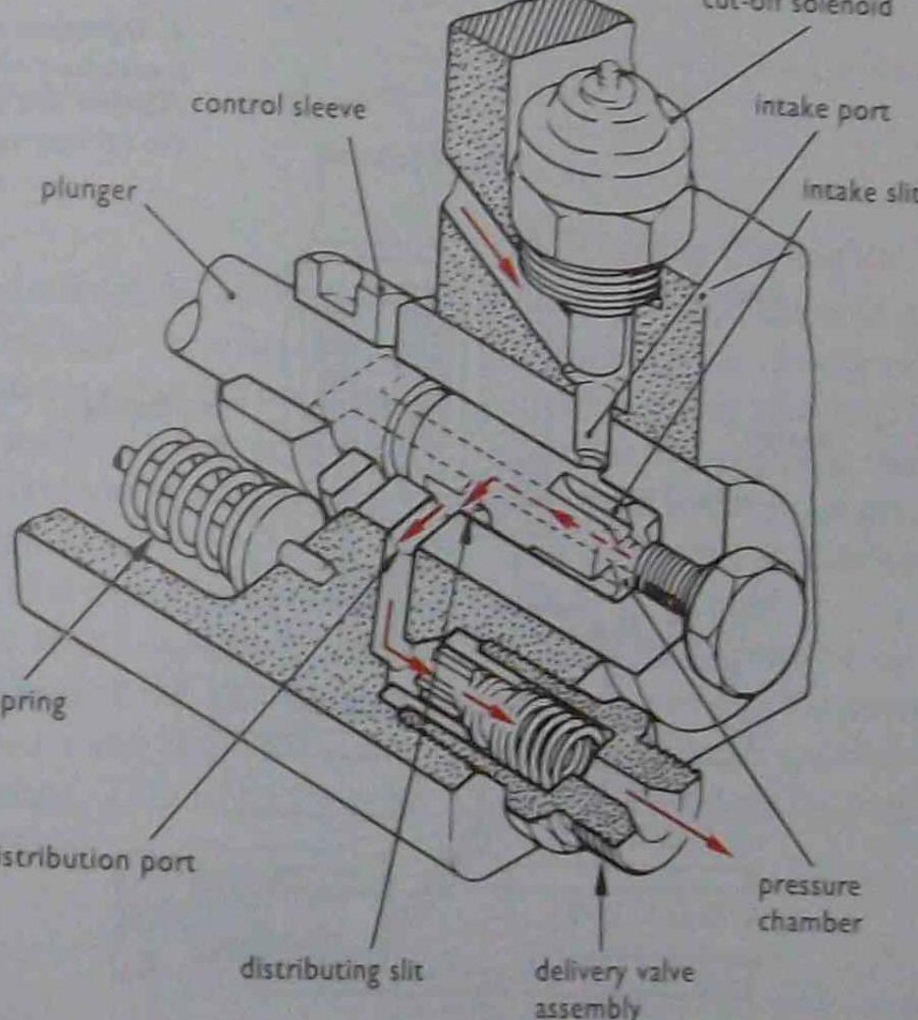
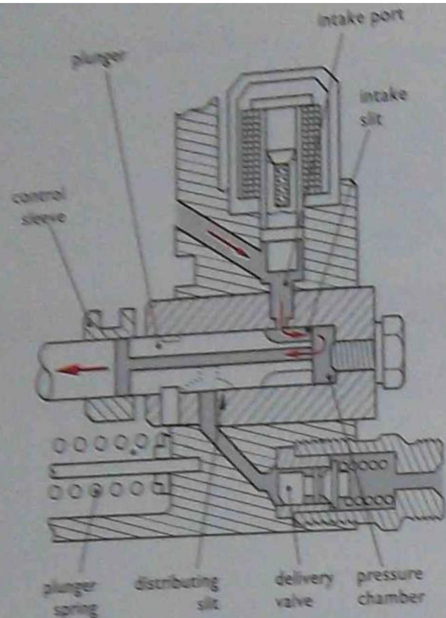
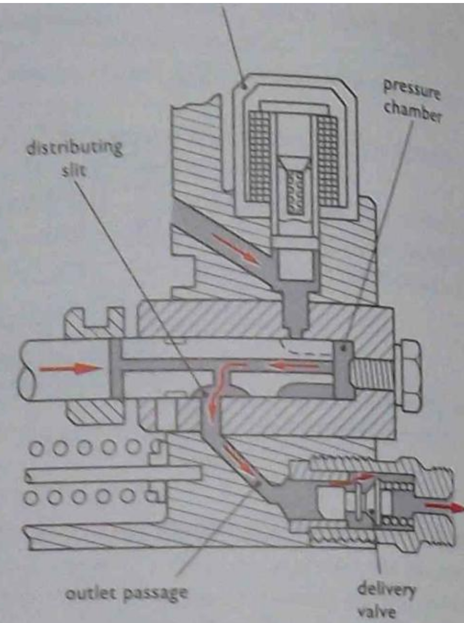


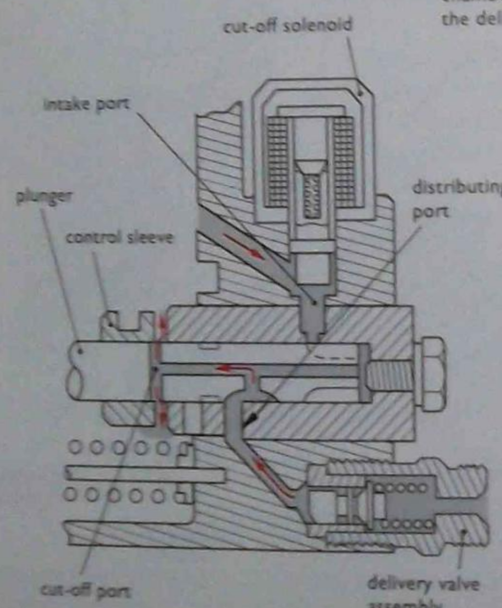
Fig. 23.21 Pumping section of a distributor pump ZEXEL



1. Intake stroke. Fuel fills the pressure chamber and the centre of the plunger.



2. Injection stroke. Fuel is pressurised in the pressure chamber and delivered from the delivery valve.



3. End of delivery. The cut-off port in the plunger releases pressure.

Fig. 23.22 The pumping action of the plunger in a distributor-type injection pump ZEXEL

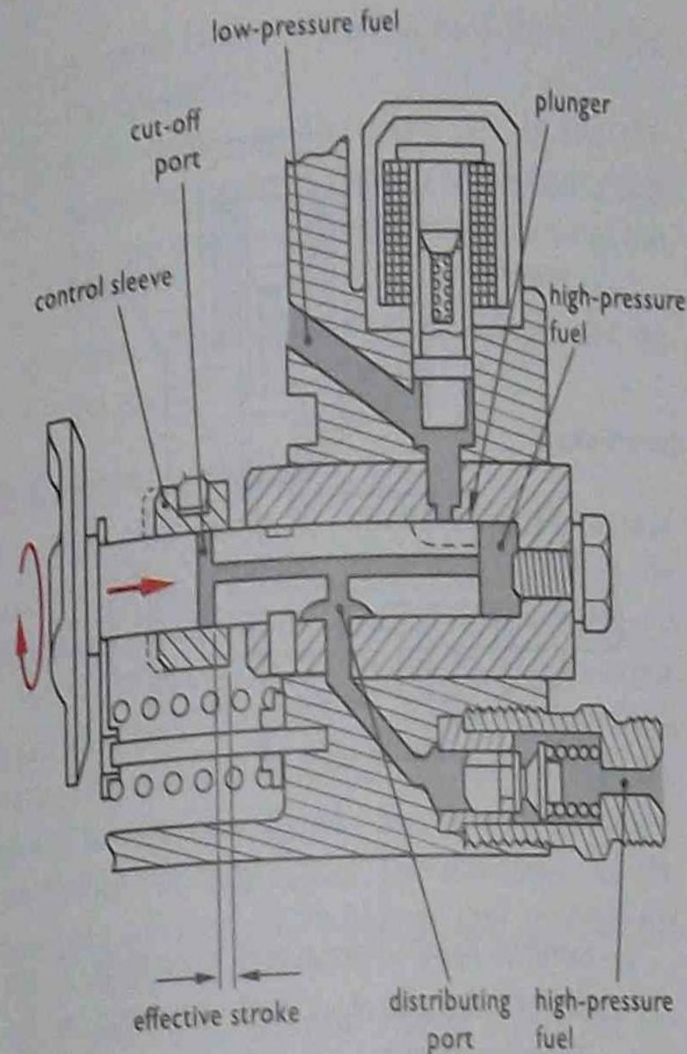


Fig. 23.23 *The control sleeve is used to adjust the effective stroke of the pump* ZEXEL

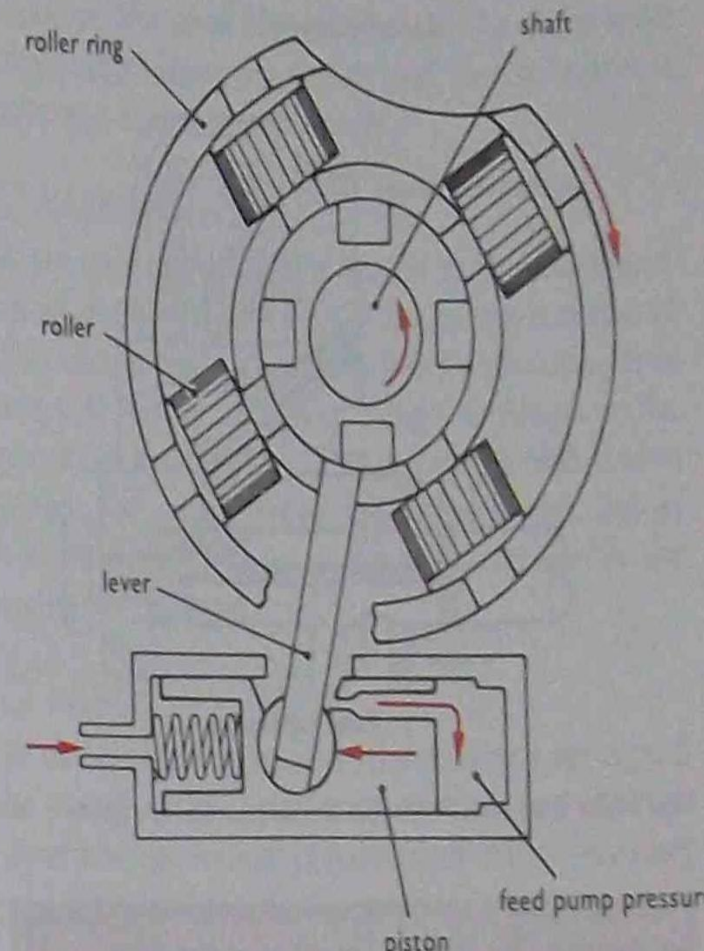


Fig. 23.25 *Timing device used on a distributor injection pump* ZEXEL

Timing advance

The arrangement for automatically advancing the injection is shown in Figure 23.25. The roller assembly which operates the cam disc is rotated a few degrees in a direction opposite to pump rotation. This causes the plunger action to commence earlier, and so injection timing is advanced.

The movement of the roller assembly is controlled by a piston in a cylinder which is subject to feed pump pressure. As the speed of the injection pump increases, feed pump pressure also increases to move the piston against the spring. This turns the roller assembly to advance injection.

Stopping the engine

The fuel cut-off solenoid is used to stop the engine. It does this by closing off the fuel intake passage to the pumping plunger (Fig. 23.26).

When the engine is switched on, the solenoid is energised and its plunger is raised. This opens the fuel intake port so that fuel from the feed pump can reach the plunger.

When the engine is switched off, the solenoid is de-energised and the plunger is pushed down by its spring to close off the intake port and block the supply of fuel. This stops the engine.

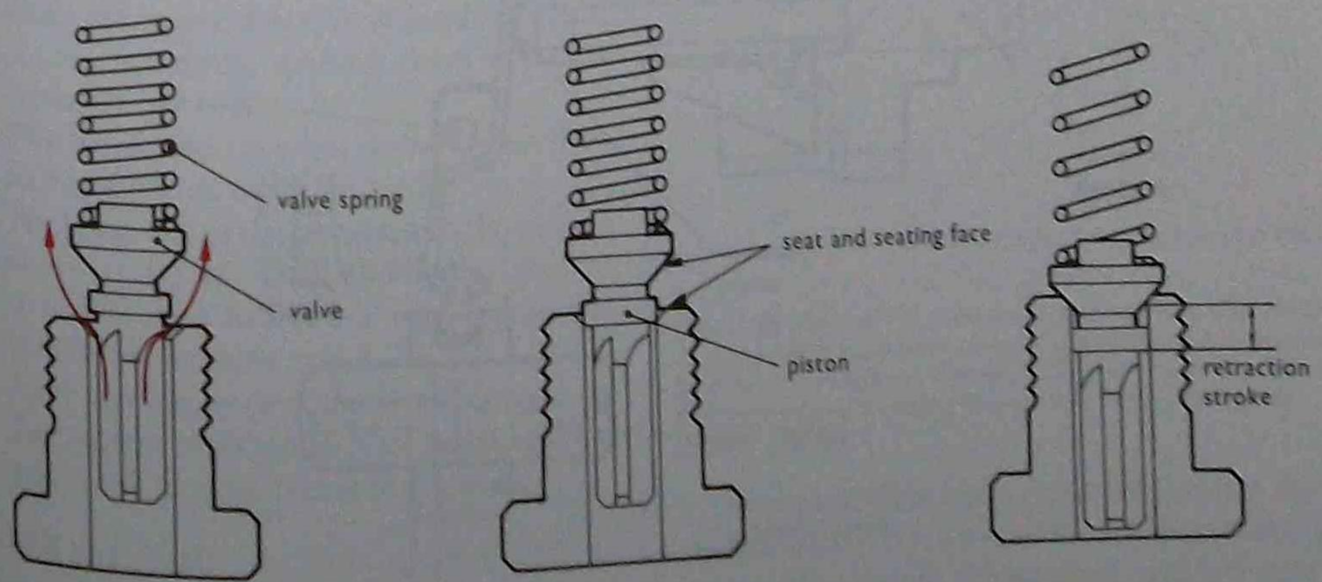
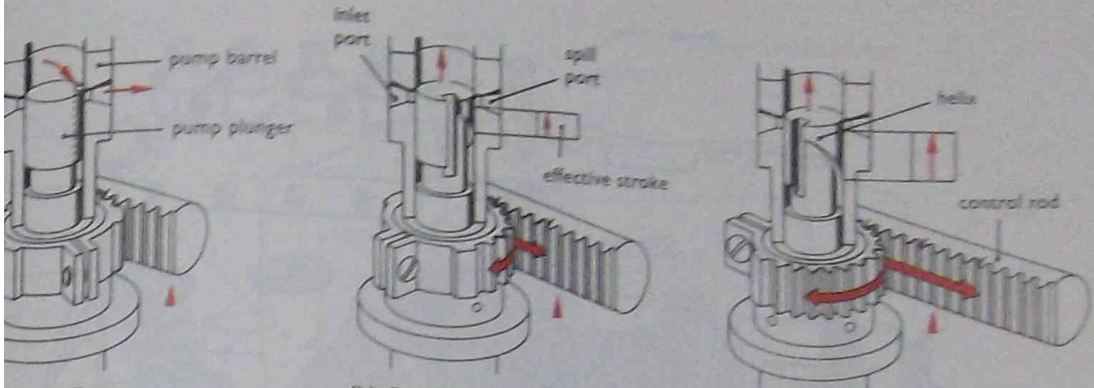


Fig. 23.24 Action of a delivery valve; the retraction stroke causes a pressure drop in the injector pipe ZEXEL



No delivery (b) Partial delivery (c) Maximum delivery

Governor

A governor for an in-line injection pump is shown in Figure 23.36. There are a number of different designs. This is a mechanical governor with flyweights which are connected through linkage to the control rod of the injection pump. The control lever is connected to the driver's accelerator.

The governor has a number of springs and linkages which allow the flyweights to control the engine. In principle, its basic operation is similar to that of the simple governor described below.

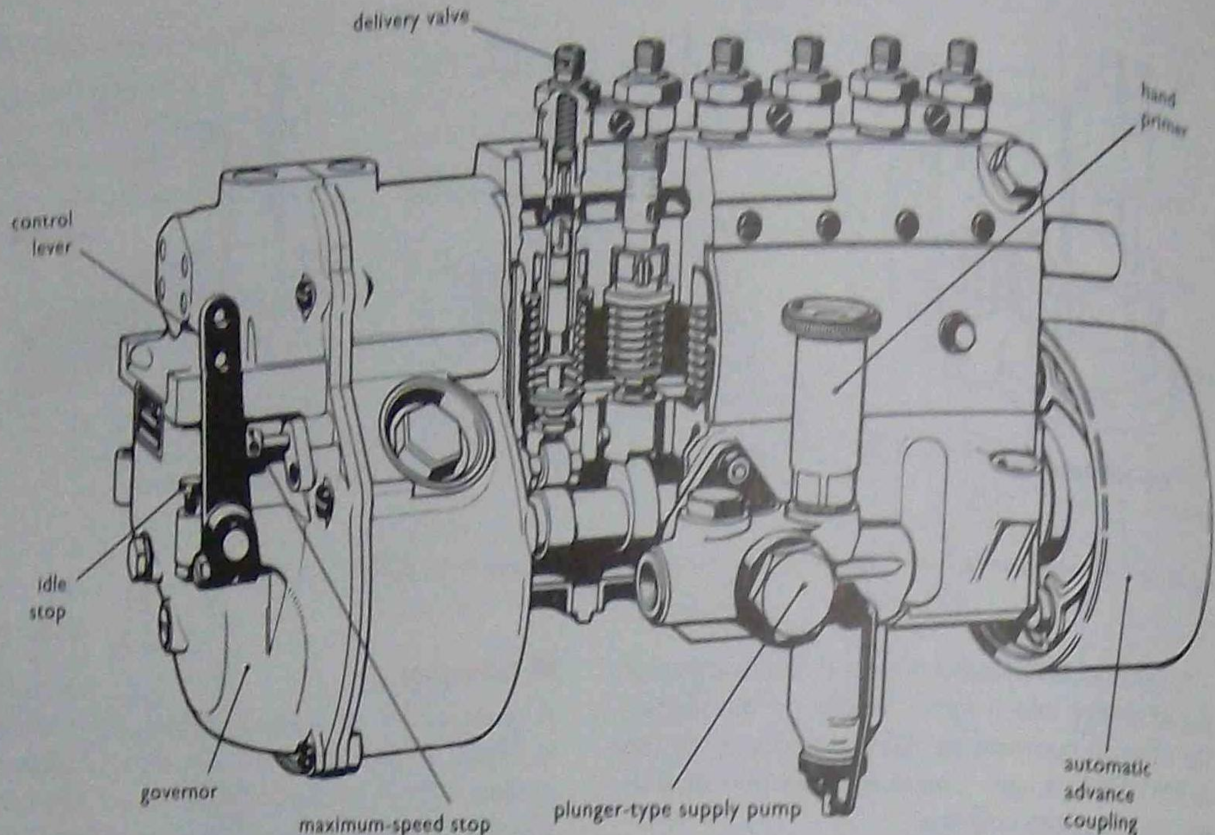


Fig. 23.35 *In-line injection pump with a mechanical governor* BOSCH

Fig. 23.36 Mechanical governor for an in-line injection pump BOSCH

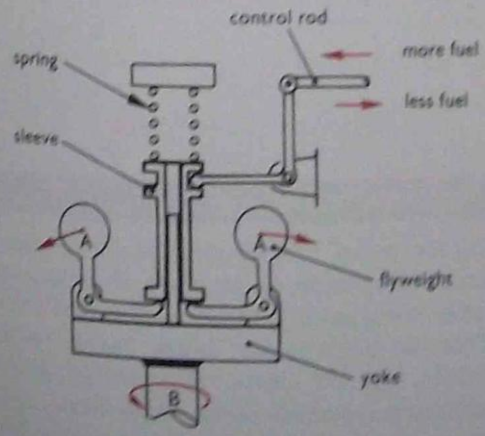


Fig. 23.37 Principle of a mechanical governor. Centrifugal force on the flyweights moves them outwards. Movement is transferred to the control rod; the spring restrains movement

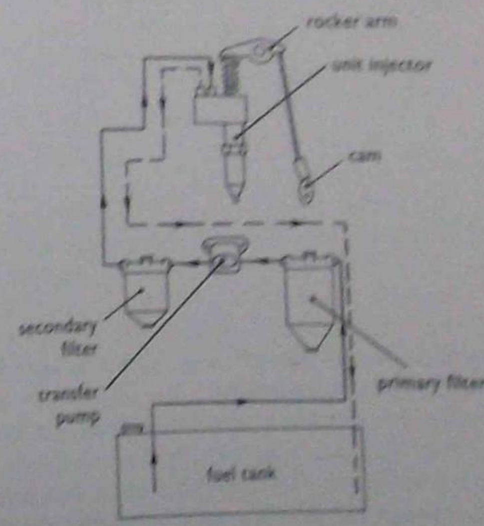


Fig. 23.38 Arrangement of a diesel fuel system with a unit injector

described.

Unit injector system

In this system, the functions of the injection pump element and the injector are combined within the injector itself. The injector is operated by a rocker arm and pushrod from the camshaft.

Referring to the diagram in Figure 23.38, fuel is taken from the tank by a gear-type transfer pump. It passes first through the primary filter, then through the pump to the secondary filter, and on to the injector.

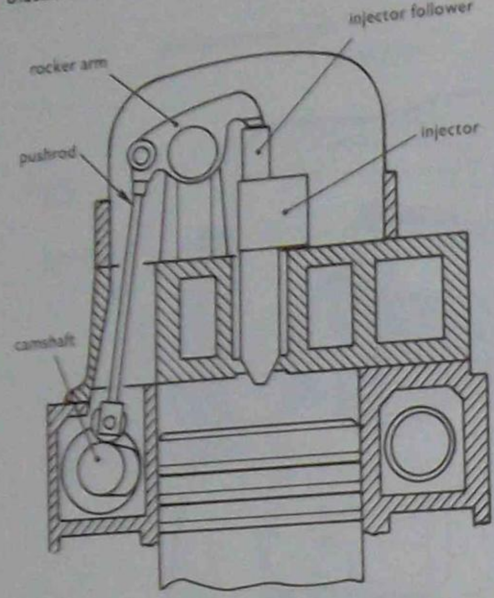


Fig. 23.39 *Arrangement of a unit injector in an engine*

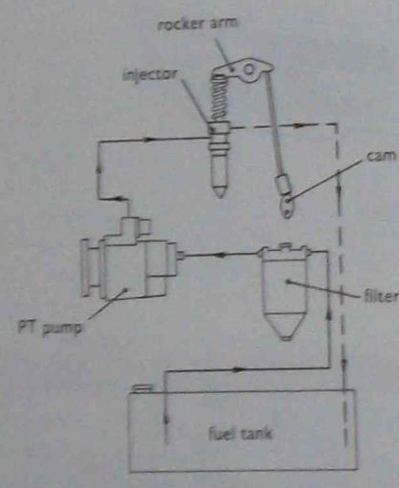


Fig. 23.40 *Basic PT fuel system for a diesel engine*

☐ INJECTOR OPERATION

The injector injects fuel, and also meters it. Metering is based on the pressure at the injector and the time that the metering orifice in the injector remains open. The pressure is varied by the fuel pump for different engine operating conditions, while the time interval is established by the engine's rotational speed, which determines the rate of motion of the injector plunger.

A simplified injector and its operating mechanism is illustrated in Figure 23.41.

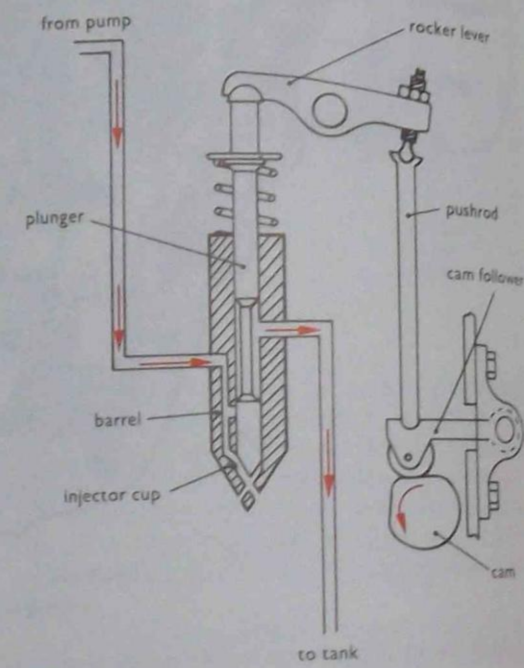


Fig. 23.41 *Simplified PT injector and operating mechanism* CUMMINS

Technical terms

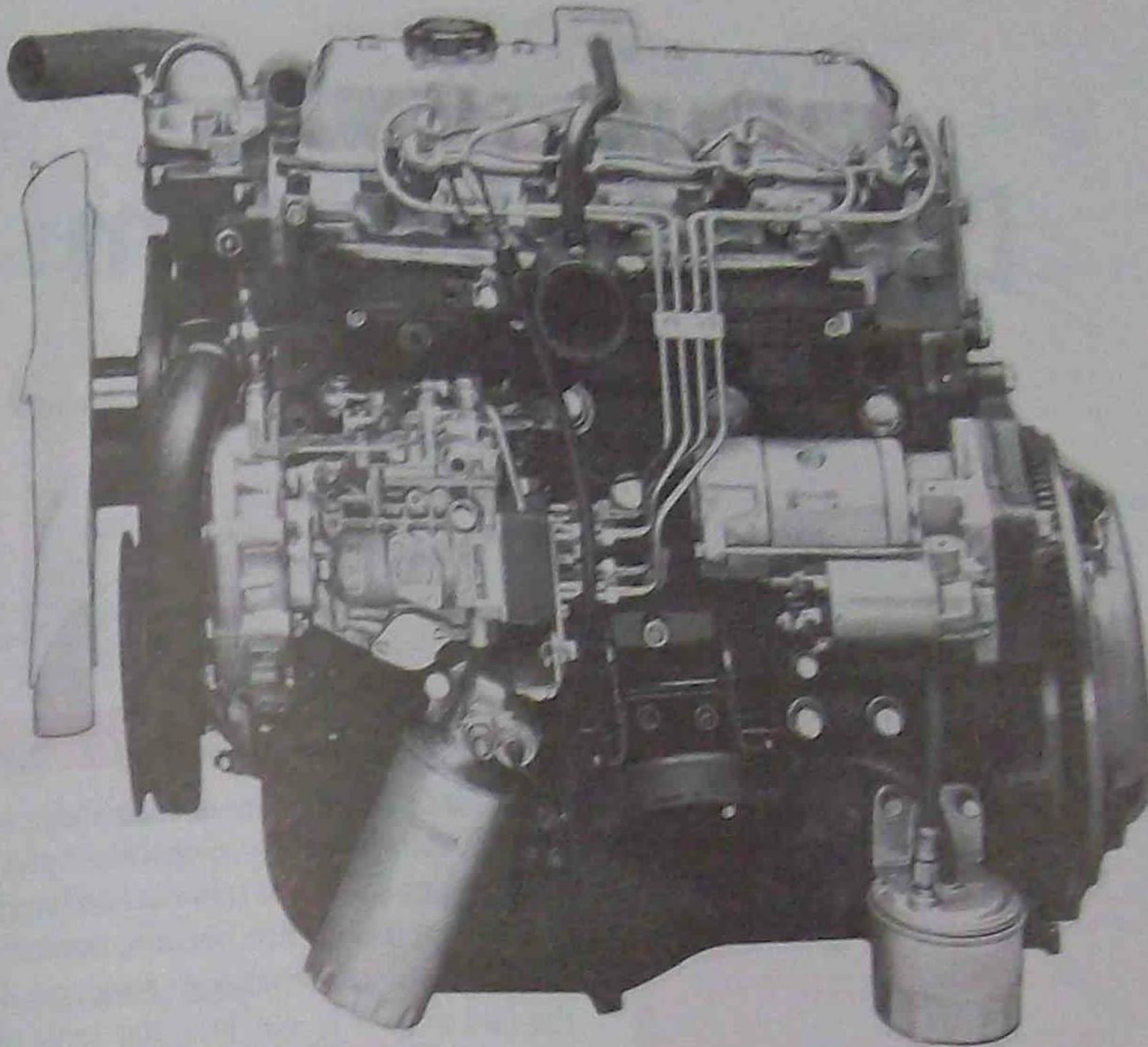


Fig. 23.42 *A four-cylinder diesel engine with a VE distributor pump flange-mounted to the rear of the timing case; injector pipes connect the distributing section of the pump to the injectors. The engine is used in light commercial vehicles* MAZDA