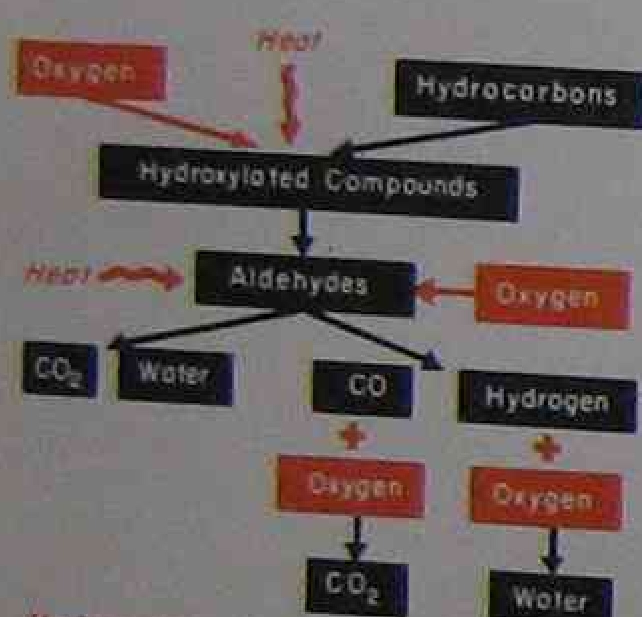
of flue gas from gas- and oil-fired furnaces often reveals presence of chemical compounds known as *aldehydes*. Formaldehyde is a fairly familiar member of this group. Nothing in our simple carbon-to-carbon-dioxide and hydrogen-to-water reaction explains the presence of these compounds. How did these strangers get into the parade?

Part of the answer lies in our comparison of two types of gas burners. Under one set of conditions, as we have seen, oxygen associates with the hydrocarbon molecules, producing *hydroxylated* compounds that are unstable. These form aldehydes which, in turn, break down or oxidize until formaldehyde is produced. What happens next depends on how much oxygen is present. Formaldehyde may break down with heat to form carbon monoxide and hydrogen, or it may burn to water and carbon dioxide or carbon monoxide. Any carbon monoxide and hydrogen formed burns in the normal way.

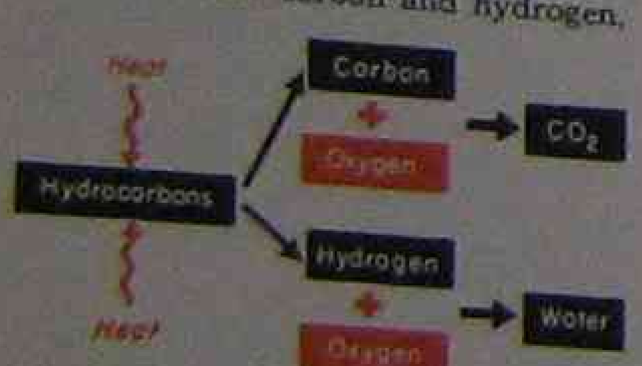


Hydroxylation burning of hydrocarbons

So it is clear that quite a few changes occur between the beginning and end of the burning operation and that many intermediate products form and disappear. If combustion is incomplete, some of these compounds remain and show up in the flue gas. The faint acrid odor sometimes observed indicates the presence of one group of intermediate compounds, aldehydes.

Cracking Process.

From our gas-burner example, we also saw that under different conditions, hydrocarbons are cracked instead of hydroxylated. Such cracking breaks them down into carbon and hydrogen,



Cracking process in hydrocarbon burning

which burn to carbon dioxide and water. We have seen that it is the incandescent carbon which gives such a flame its characteristic yellow color, and it is the carbon which gives it soot-and-smoke-forming propensities.

We have talked about these two ways hydrocarbons burn as if they were entirely separate. In virtually all practical burning of hydrocarbons, however, both processes go on simultaneously and the general character of the flame depends on which predominates.

What fixes nature of hydrocarbon flame?

This, in turn, depends on surrounding conditions. Early mixing and preheating of hydrocarbons and air, plus time for the oxygen molecules to enter the hydrocarbon molecules, favor hydroxylation. Rapid heating, and lack of mixing time, favor thermal decomposition.

We've belabored this business of hydrocarbons for two reasons: (1) Biggest part of practical combustion work involves them; natural gas and oil are hydrocarbon fuels, so is the volatile matter of coal. (2) It shows clearly that while our familiar chemical equations for combustion are useful in calculating air requirements, etc, we must know more about what happens between the starting point and the finish if we are to design and operate successful firing equipment and furnaces. *The actual combustion process may consist of several reactions and their nature depends on the form in which the basic elements carbon and hydrogen appear in the fuel, and the way burning takes place.*

Elementary Fuels.

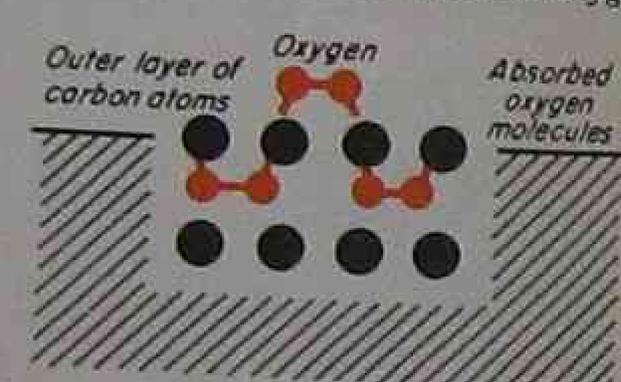
Thus, to understand practical combustion problems we must take a look at each of the forms in which carbon and hydrogen appear in commercial fuels and the steps through which they pass in burning. This isn't as tough as it sounds, since for our purposes we can consider all complex fuels as combinations of three simple or elementary fuels: (1) gaseous hydrocarbons (2) solid carbon and (3) mixture of carbon monoxide and hydrogen. Each of these general groups contains a variety of fuel substances which differ in minor respects (charcoal and coke in the solid-carbon class, for example), but each group displays a generally similar pattern of combustion.

Some commercial fuels are simple mixtures of these elementary fuels. For example, coal gas is essentially a mixture of gaseous hydrocarbons and carbon monoxide plus hydrogen. With other familiar fuels, our elementary fuels come into being during the burning process; fuel oil volatilizes into gaseous hydrocarbons before actual combustion. Some cracking may also occur, yielding solid carbon and hydrogen.

Coal is another fuel that decomposes into elementary fuels before actual combustion starts. When first exposed to the fire's heat gaseous hydrocarbons, carbon monoxide and hydrogen are distilled off, leaving solid carbon behind. Hence to know what actually happens when coal burns we must know the individual burning processes of all three elementary fuels.

How Carbon Burns.

We've got the toughest of these under our belts, having already seen how gaseous hydrocarbons burn. The way carbon burns has puzzled the experts for many years but careful experiments seem to back up the current theory. This assumes that oxygen penetrates the carbon surface to break away atoms which hook up with the oxygen



Oxygen combines with outer carbon atoms

in a loose sort of carbon-oxygen compound that is unstable. It is not a true chemical compound because proportions of carbon and oxygen atoms are not fixed, nor is it entirely a mixture, since the oxygen cannot be driven off. Depending on temperature and other conditions, this *physiochemical* compound breaks up into carbon dioxide and carbon monoxide. If there is an excess of oxygen, carbon monoxide is oxidized to carbon dioxide. If carbon is in excess, dioxide is reduced to monoxide.

Burning H_2 and CO.

Hydrogen and carbon monoxide are grouped together as an elementary fuel because they commonly occur together and their burning processes are alike. In ordinary combustion work, conditions are such that both burn according to the familiar simple reactions:

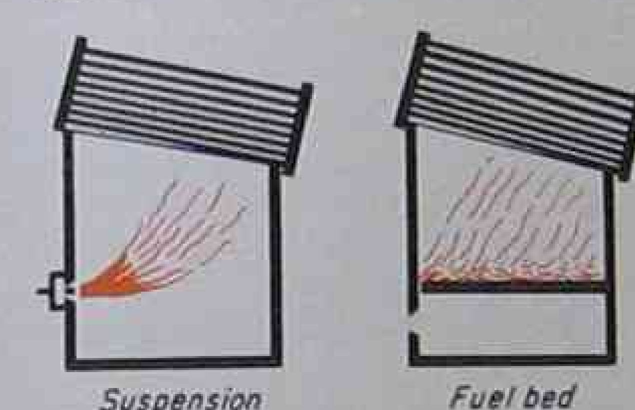
$$2H_2+O_2=2H_2O$$
$$2CO+O_2=2CO_2$$

Hydrogen burns more than 2.5 times as fast as carbon monoxide when they burn together with excess oxygen.

Actual Firing Job.

Now, with these fundamentals out of the way, we're primed to tackle the job of burning commercial fuels in actual furnaces. Just what is this job? In broadest terms, it includes: (1) preparing the fuel and air (2) converting the complex fuel into elementary fuels (3) bringing these fuels and air together in the right proportions and at the proper temperature for ignition and combustion, and (4) transferring heat from the products of combustion to the boiler or other surfaces while retaining enough heat in the combustion zone to maintain volatilization and ignition. All these actions are going on at the same time in any furnace, and each particle of fuel traces the entire sequence, in order, in its brief passage through the furnace.

Broadly speaking, there are two ways to handle this job: (1) in *suspension* or (2) on a *fuel bed*. Let's look at both



Practical fuel burning takes two forms

to see how they differ and, more important, in what ways they are alike.

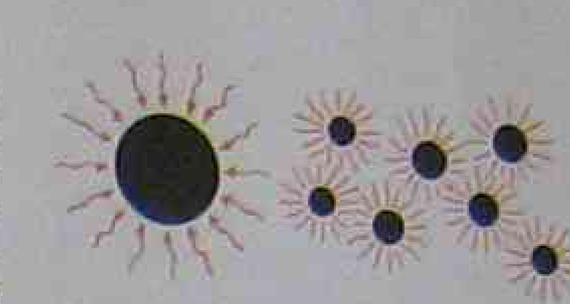
Suspension Firing.

When we think of suspension, or *burner firing*, gas comes immediately to mind as the simplest example. Here we have a fuel already in the gaseous state, ripe for quick mixing with air. Gas burners thus need only proportion the volume of air and gas to insure thorough mixing. As we shall see later when examining actual burner designs, they differ mainly in how and when this mixing occurs.

Burners for oil and coal, in addition to proportioning fuel and air, and mixing them, must *prepare* the fuel. For oil, this means making it ready for conversion from liquid to gaseous hydrocarbons. For coal, it involves distilling off the volatile matter (gaseous hydrocarbons again, plus hydrogen and carbon monoxide). This must be done in the instant after the fuel enters the furnace.

Preparing Fuel.

It is the furnace heat that does the conversion; job of the firing equipment is to put fuel into shape that makes best use of this heat. For suspension firing of both oil and coal the answer is the same—it must be broken up into many small particles to expose as


Small particles show more area to heat

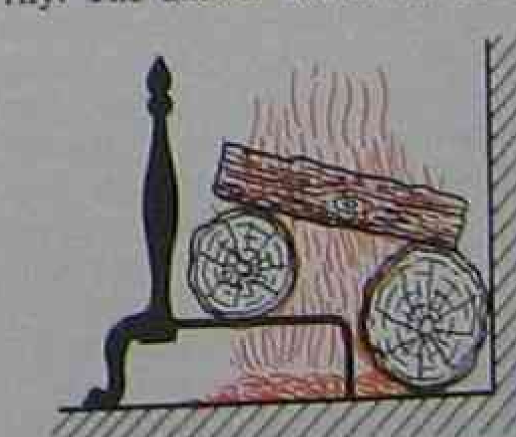
much surface as possible. With oil, *atomization* is obtained in a variety of ways by the burner; with coal *pulverization* is handled in a separate unit, the burner merely mixing the finely ground particles with air and injecting them into the furnace.

In addition to small particles, we need fast motion between particles and air to strip away the microscopic "coats" of gas that form on the particles and thus expose fresh surfaces to heat and air.

Furnace's Job.

The furnace takes over where the burner leaves off, providing the conditions for continuous complete combustion. Within the combustion zone, fuel must be vaporized or distilled, mixed with air, ignited, and the resulting reactions between fuel and oxygen carried to a finish. Chief responsibility for bringing fuel and air together rests with the burner; the furnace can do little to correct failure of the burner in this respect. On the other hand, the heating and igniting job is predominantly the furnace's.

The difficulty met in trying to burn a single log in a fireplace illustrates the part played by good furnace design. As anyone who has tried will testify, a single log is one of the hardest things in the world to keep burning, while a number of logs will blaze merrily. The answer lies in the fact that


Logs reflect heat to each other

when there are several logs, each reflects heat to the others, for distillation of volatile matter and ignition. In the same way a furnace must maintain a heat supply to prepare and ignite incoming fuel.

Soot and Smoke.

With either coal or oil, some carbon formerly associated in hydrocarbon compounds breaks away as free carbon and, of course, a substantial part of most coal is solid carbon. Thus we always have the problem of burning carbon particles, which, as we have seen, proves extremely difficult even with ample time. Here is the germ of smoke and soot. Naturally, if the furnace is small and cold areas (boiler tubes and the like) are badly located, carbon cannot possibly burn to completion.

All these steps—bringing air and fuel together, raising mixture to ignition temperature, sweeping away successive layers of gas from fuel particles, burning carbon as far as possible—occur while fuel and air travel from burner to furnace outlet, an extremely short

In combustion, time equals distance

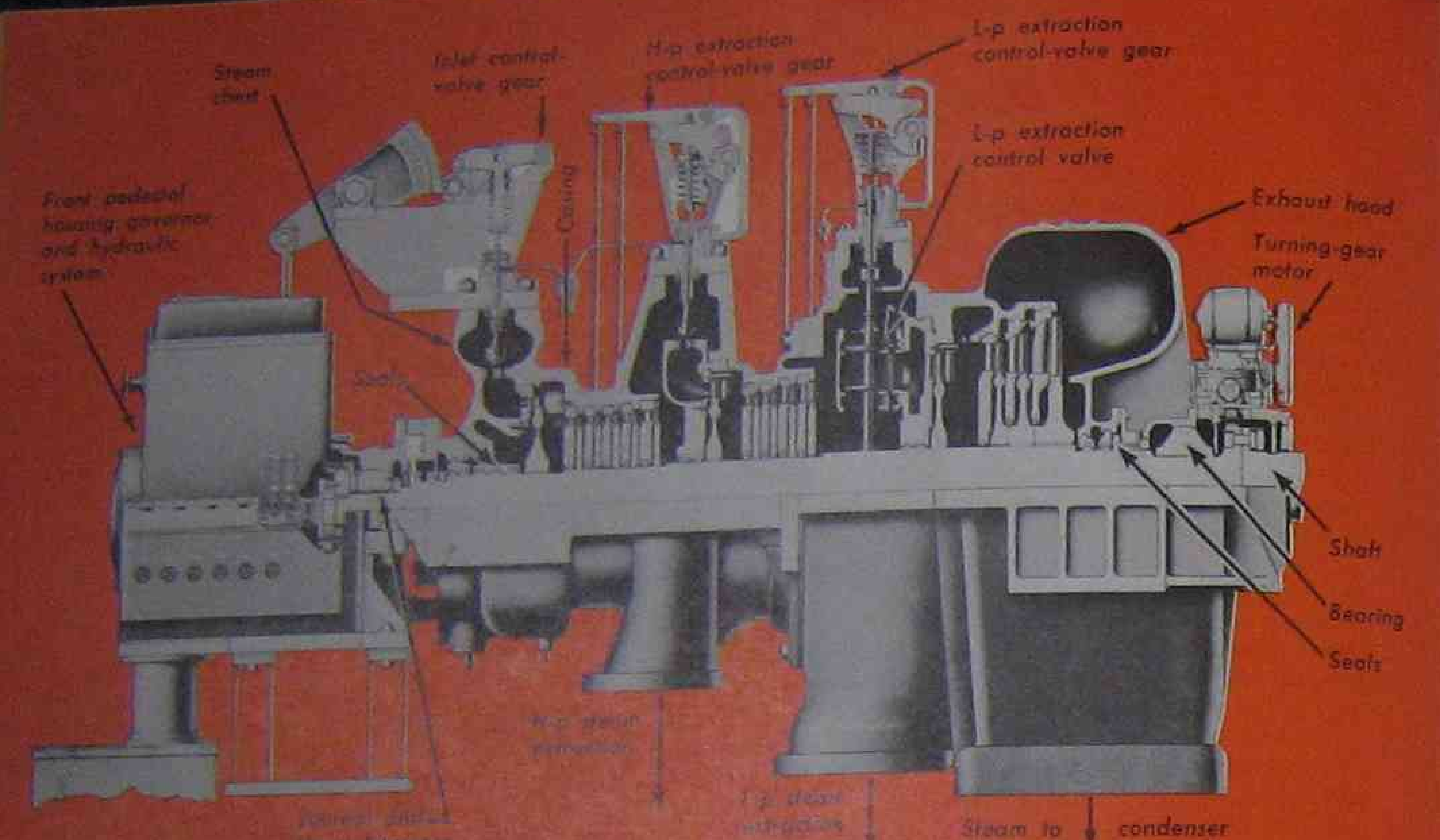
time. Just how much depends on the distance traveled and the speed, and whether or not flow is *turbulent*.

Turbulent Mixing.

By turbulence, we mean a condition in which fuel and air whirl and eddy in irregular paths instead of flowing in streamlines from burner to furnace outlet. This has several beneficial effects: (1) There is an increase in the time available for combustion. (2) Better mixing of fuel and air results. (3) Fuel and air move past each other at higher velocity, thus helping to sweep away combustion products and expose fresh surfaces. It has been well said that successful combustion depends on three "T's"—Temperature, Time and Turbulence.

Ash and Slag.

So far, we have ducked the question of mineral impurities (ash) found in many fuels and particularly in coal. These, of course, do not burn. In suspension firing, if nothing else happened, some particles would be carried out of the furnace with the gas stream (*flyash*) while others would drop out of suspension in furnace or boiler passes. This simple picture is complicated by the fact that, depending on the nature of the ash, and to some extent on surrounding conditions, ash becomes plastic and sticky. If it comes in contact with relatively cold surfaces while in this plastic state it forms *slag* coatings, reducing rate of heat trans-



DOUBLE-AUTO-EXTRACTION [illegible] Extraction-control valves hold [illegible] by regulating its flow to following stages

this by converting leaving kinetic energy of the last stage into internal energy.

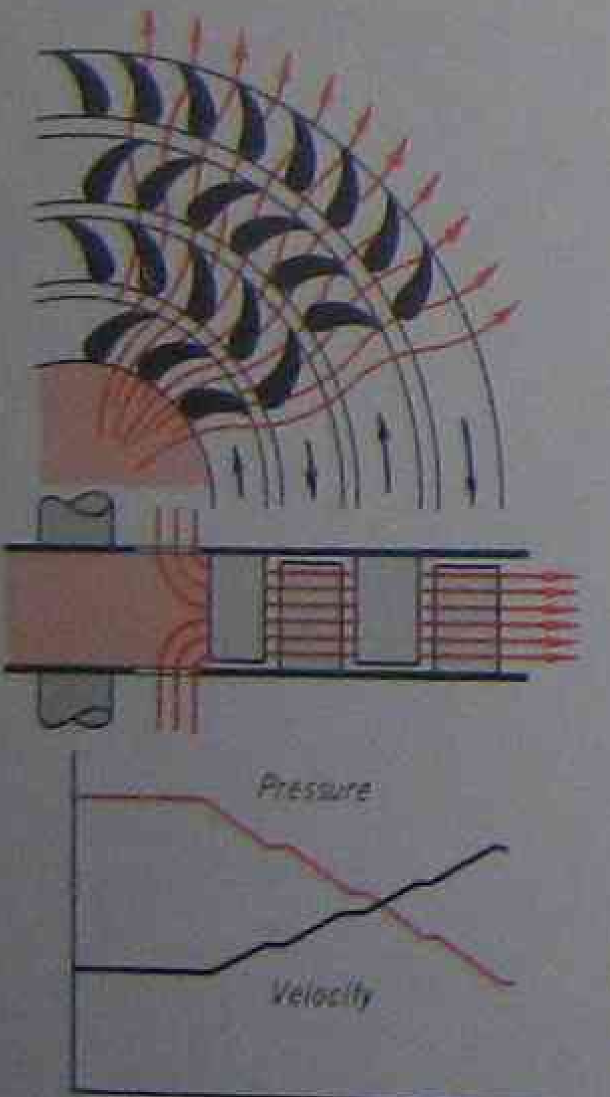
Rupture diaphragms in the exhaust hood prevent excessive pressure buildup, should the condenser lose its vacuum. A *governor* geared to the shaft in the front bearing pedestal operates the governor valves through a hydraulic servomotor system. Main governor can be mechanical or hydraulic; it opens and closes the valves in response to fall and rise of shaft rpm.

An *overspeed governor* is usually located in the shaft at the front bearing pedestal. Its function: shut a main stop valve ahead of the governing valves when shaft rpm exceeds rating by about 10% (15% on small turbines).

Disk-and-diaphragm type of turbine shown above is used where impulse staging predominates with higher pressure drops per stage, and percentage of reaction is relatively low. This unit handles mechanical or generator drive. It features two points of *automatic extraction* where part of the steam may be withdrawn from the turbine at constant pressure. Steam chest is integral with the casing. Control and extraction valves respond to both shaft speed and steam pressure at the extraction points.

In this turbine the moving buckets ride on *wheels* or *disks* that are shrunk on and keyed to the shaft. The nozzles are carried in diaphragms that have centered holes through which the shaft passes. Seals at these openings set a limit to steam leakage bypassing the nozzles; construction of the diaphragm limits the cross-sectional area available for leakage through the seal.

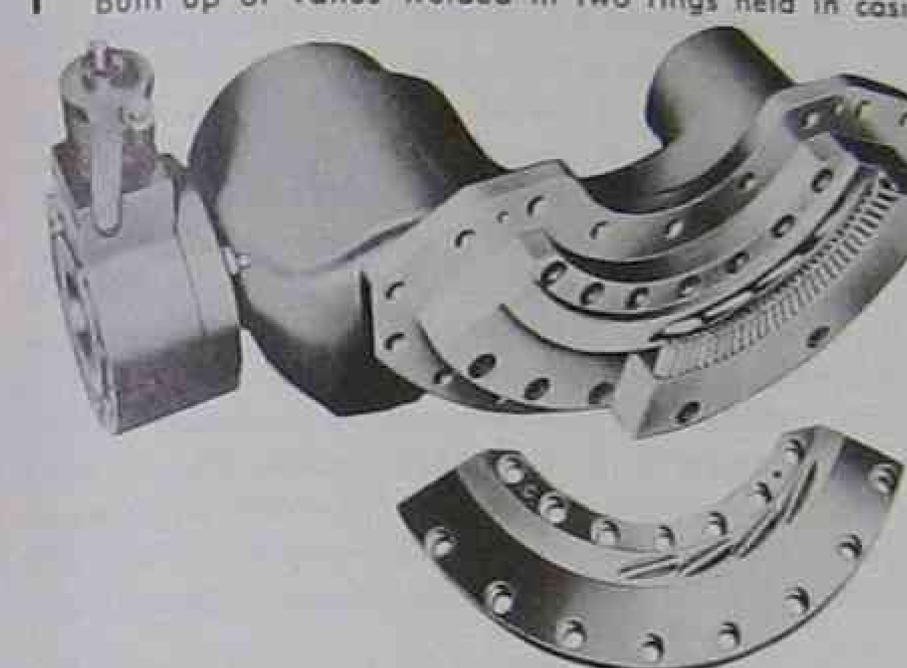
All turbines need a *lubricating system* for their bearings. Usually it acts as the hydraulic system for actuating valves and servomotors, as well. The oil reservoir is often remote, at a lower level to facilitate drainage.



RADIAL-FLOW double-rotation reaction unit has only moving blades. Intermeshing "cylinders" of blades mount alternately on oppositely rotating plates. This turbine has two shafts, usually drives alternators. It can handle high-speed steam jets because there are no stationary blades; steam can be extracted

KEY ELEMENTS in turbine design

1 Nozzle block has three groups of converging nozzles built up of vanes welded in two rings held in casing



2 Steam chest holds nozzle block. Block has converging-diverging nozzles, plus a row of reversing blades

Every stage of a turbine has two basic elements: (1) stationary nozzle and (2) moving bucket or blade. Design of these parts depends on factors such as entering steam conditions, exhaust steam pressure, shaft speed, rated capacity, steam flow, where part is located in the unit.

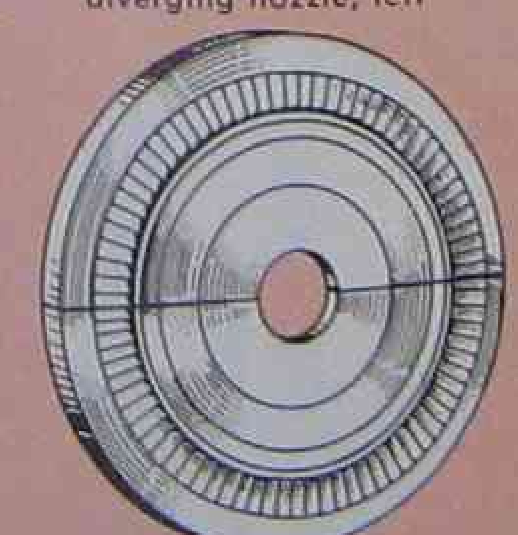
Nozzles may be either converging or converging-diverging, as we saw on p 2. Selection depends on pressure ratio of steam across the nozzle. Fig. 1 shows upper half of nozzle block for the first stage of an intermediate-size turbine. Nozzles are grouped so they can be controlled by individual valves. These are typical converging nozzles: in cross section, the steam passage is rectangular.

Some shapes of vanes making up this type nozzle are shown on p 2. In the high-pressure end of the turbine, nozzle vanes are usually welded into the nozzle diaphragm. In the low-pressure end, nozzle vanes may be cast as an integral part of a diaphragm.

Fig. 2 is a steam chest for a small-capacity turbine. Converging-diverging nozzles are drilled into the block and a set of stationary reversing buckets attached for the 2-row control or velocity-compounded stage. Steam passages through the nozzles have a circular cross section.

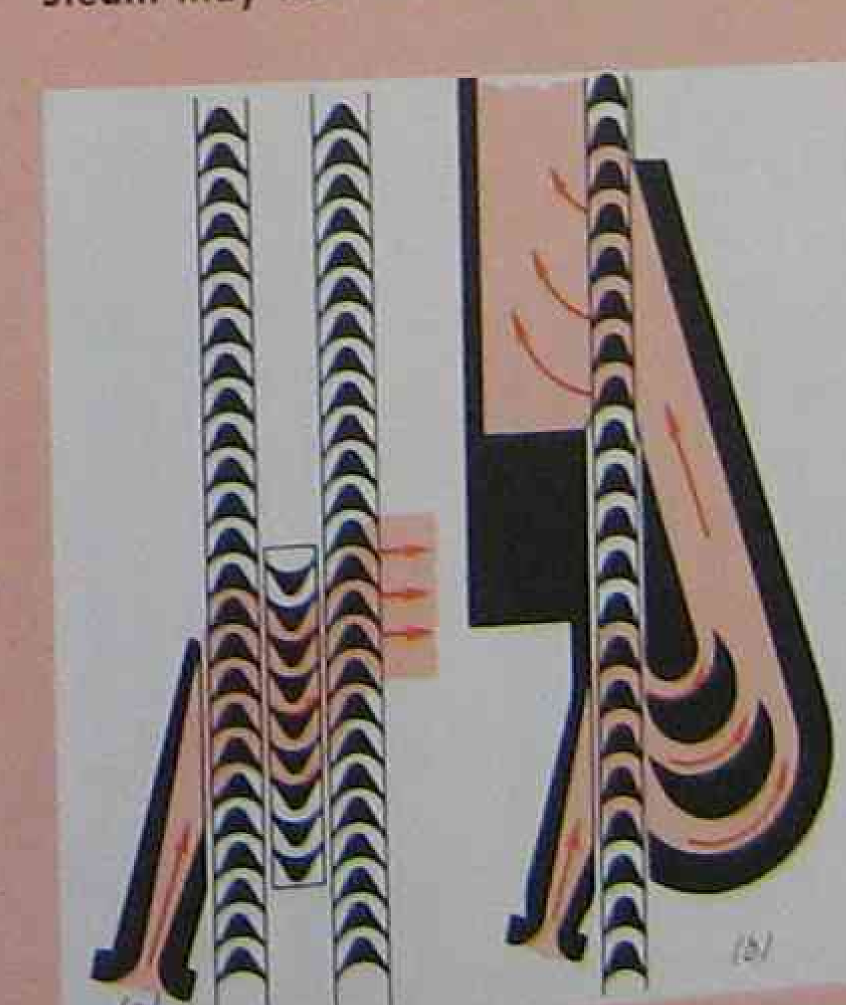
Single converging-diverging nozzle, Fig. 3, has attached reversing blading. This nozzle fits a small turbine with a 2-row single wheel. Typical nozzle diaphragm for a lower stage of a disk-and-diaphragm turbine is shown in top drawing, Fig. 4. Its two halves dovetail to make it tight against leakage. Lower drawing shows a nozzle row for a drum type unit; no inner diaphragm is needed because the drum occupies this space. Fig. 5*a* shows nozzle, wheel, stationary-blading layout for the 2-row single-wheel unit.

3 Reversing blading, right, attaches to a converging-diverging nozzle, left

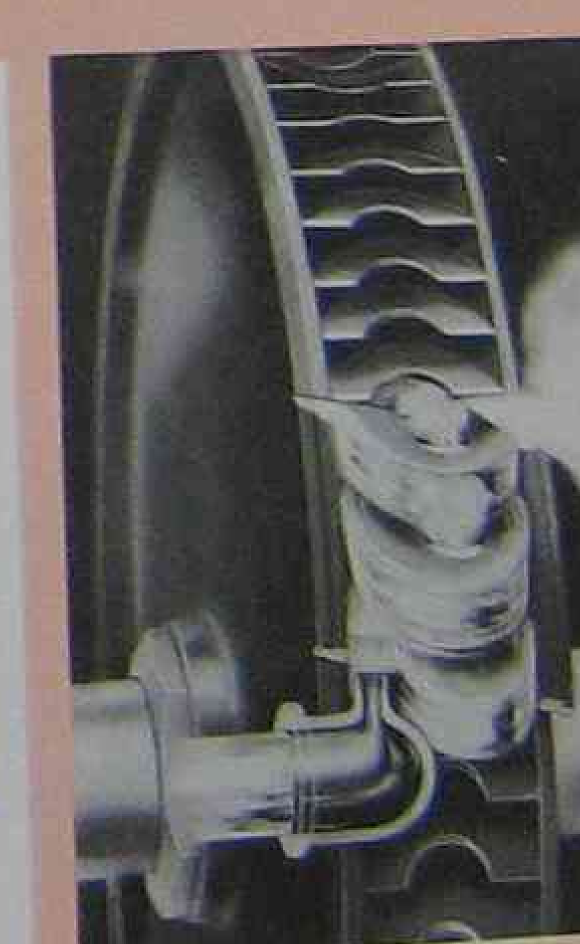


4 Typical nozzle setups are for disk-diaphragm unit, above; drum type, below

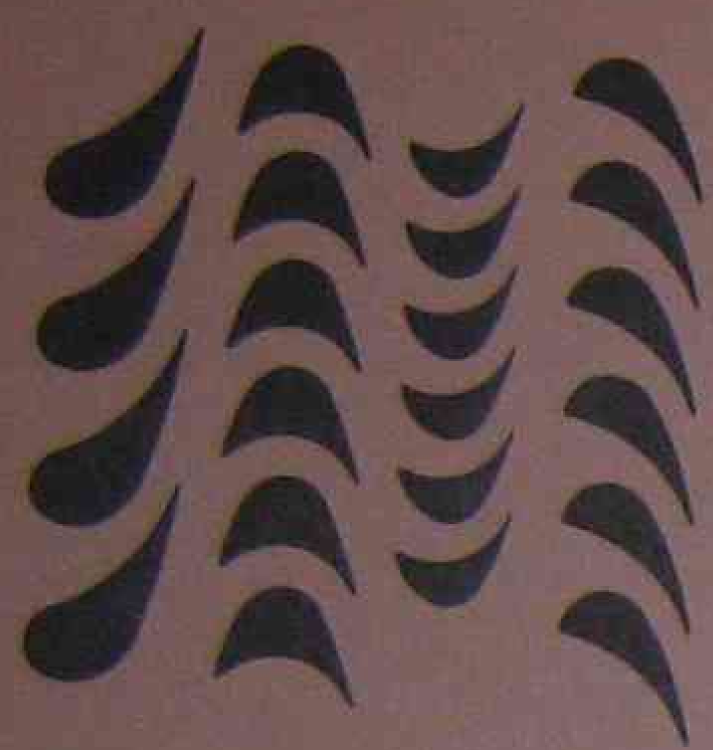
Steam may flow more than once through blade row



5 (a) Velocity-compounded stage passes steam once through moving blades (b) steam flows twice through blades



6 Velocity-compounded impulse wheel; three passes



7 Control stage for large unit has low percentage reaction, steam-pressure drop

Velocity compounding can be done with a single row of moving buckets. Steam may pass just once through the blades, as in Fig. 5*a*. In 5*b*, stationary reversing nozzle at right guides high-speed steam into a second pass through the moving buckets. In another method of velocity compounding, Fig. 6, steam flowing from a converging-diverging nozzle makes three passes through buckets milled into the edge of a solid wheel. Stationary reversing chambers return the high-speed steam to the wheel buckets along a helical path. Control stage for large turbine, Fig. 7, differs slightly from those on p 2 since vanes have a small percentage of reaction built in.

Blades or buckets take many forms. Fig. 8 shows buckets of two succeeding stages in a disk-and-diaphragm turbine, along with the intervening nozzle. Buckets receive the working force of the steam, transmit it as a moving torque into the wheels or disks that carry them.

Height of buckets and diameter of stages increase in succeeding stages to accommodate volume of the steam, which expands as its pressure drops. Shroud bands cover the bucket tips to keep steam from spilling out radially.

In high-efficiency turbines every effort is made to confine steam flow to the working passages through nozzles and buckets. Sealing strips between bucket shrouds and casing diaphragms, Fig. 8, minimize steam leaking past the bucket tip into the following nozzle.

Some turbines use pure impulse buckets without any pressure drop across them. In other units, Fig. 7, stage may have some low degree of reaction—10% or less—to produce a small pressure drop across the bucket. This small drop can be very useful—it keeps all buckets running full of steam, shaft thrust positive in one direction.

In Fig. 9, blades of five stages mount on their disks or wheels. Side- or axial-entry roots attach them to the wheel rims. Shrouds, usually riveted to the blade tips, are used to fasten groups of blades together on a wheel. They prevent steam from spilling, minimize bucket vibration.

Reaction blading for drum type turbines, Fig. 10, has stationary blades fastened to the upper stationary-blade ring, which is supported by the turbine casing. Moving blades dovetail into mating slots on the drum rotor.

Since steam pressure drops across both moving and stationary blading, radial seal strips opposite the shrouds minimize steam leakage past the blades. Turbine of Fig. 10*a* uses two seal strips per blade row; 10*b* uses three.

In many turbines that use velocity-compounded control stages, steam flows over only a part of the total periphery of the stage. In the "idle" section, buckets churn the stagnant steam. This windage loss reduces turbine efficiency. To minimize it, shields fit closely over the idle part of the bucket travel. They limit amount of steam whipped about by the nonworking buckets, Fig. 11.

When blade height becomes a significant part of the total stage diameter, ratio of steam and bucket speed changes over the length of the bucket. Then warped buckets, Fig. 13, must be used. Fig. 12, an exploded view, shows a nozzle diaphragm, steam vortex flow, a bladed or bucketed wheel and the leaving exhaust steam. Ideally, steam enters the nozzles in an axial direction and leaves in a circumferential direction, forming a vortex flow or eddy that is contained by the turbine casing before steam enters the moving buckets. To avoid cross currents in the vortex flow, product of linear velocity of steam and radius of circle it travels in must be constant: $v_1 r_1 = v_2 r_2$. Steam pressure must be higher at outer than inner radius.

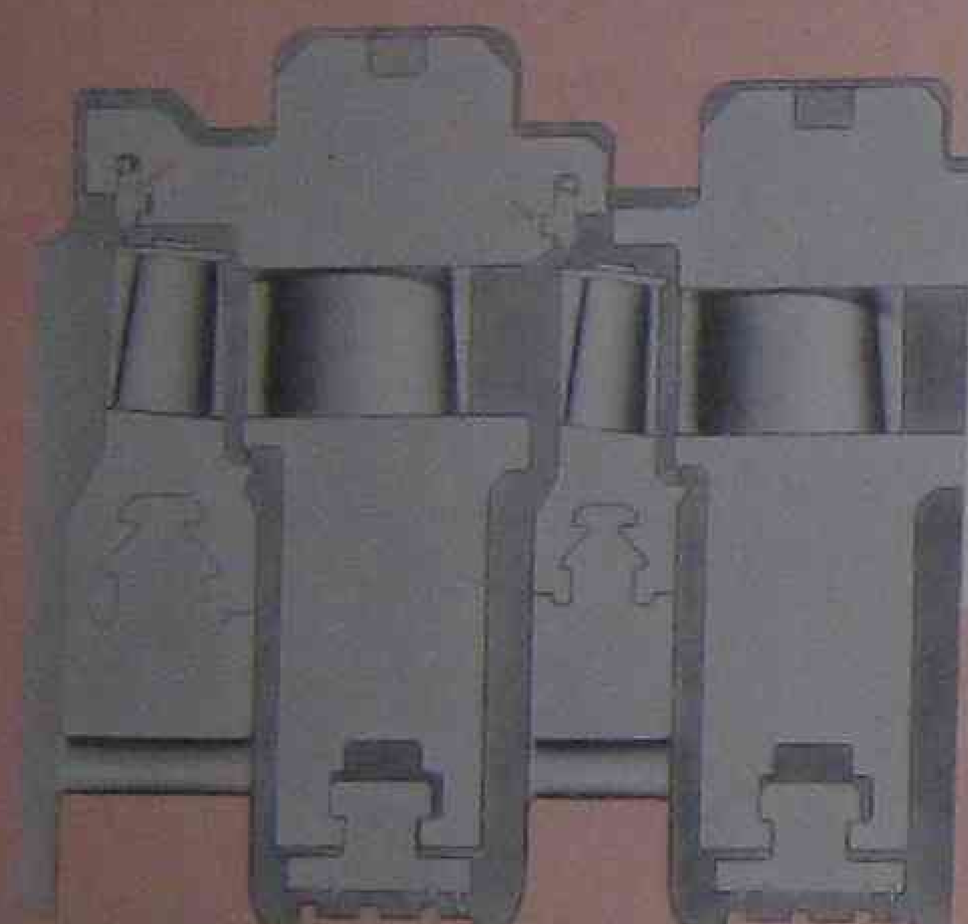
Steam leaves nozzles at the inner radius with higher linear speed than at outer radius. But buckets' linear speed increases with radius. So we get a steadily growing ratio of blade to steam speed as we move from root to tip.

Fig. 13 shows velocity diagrams at root and tip of a blade that receives a steam jet moving in vortex flow. Root of the blade has been designed for impulse flow, which is equivalent to 0% reaction, no pressure drop.

Entrance angle of the blade is fixed by angle of approach of the steam's relative speed, so steam slides smoothly over the blade. In the ideal situation, absolute steam speed should just about double the blade speed. At blade's exit edge, vector difference of relative steam and blade speed shows that steam has a residual absolute speed in an axial direction.

In our example, blade speed at its tip is about double the absolute steam speed. Relatively, then, steam approaches the blade from a direction almost opposite to its motion. Blade section must be twisted to receive the steam smoothly all the way up the blade. But since entering steam pressure is higher at the tip than at the root, there will be a pressure drop through the blade. This means we must use a reaction-blade section, with relative steam speed higher at the blade exit. A pure reaction force acts at the blade tip, a pure impulse force at the root.

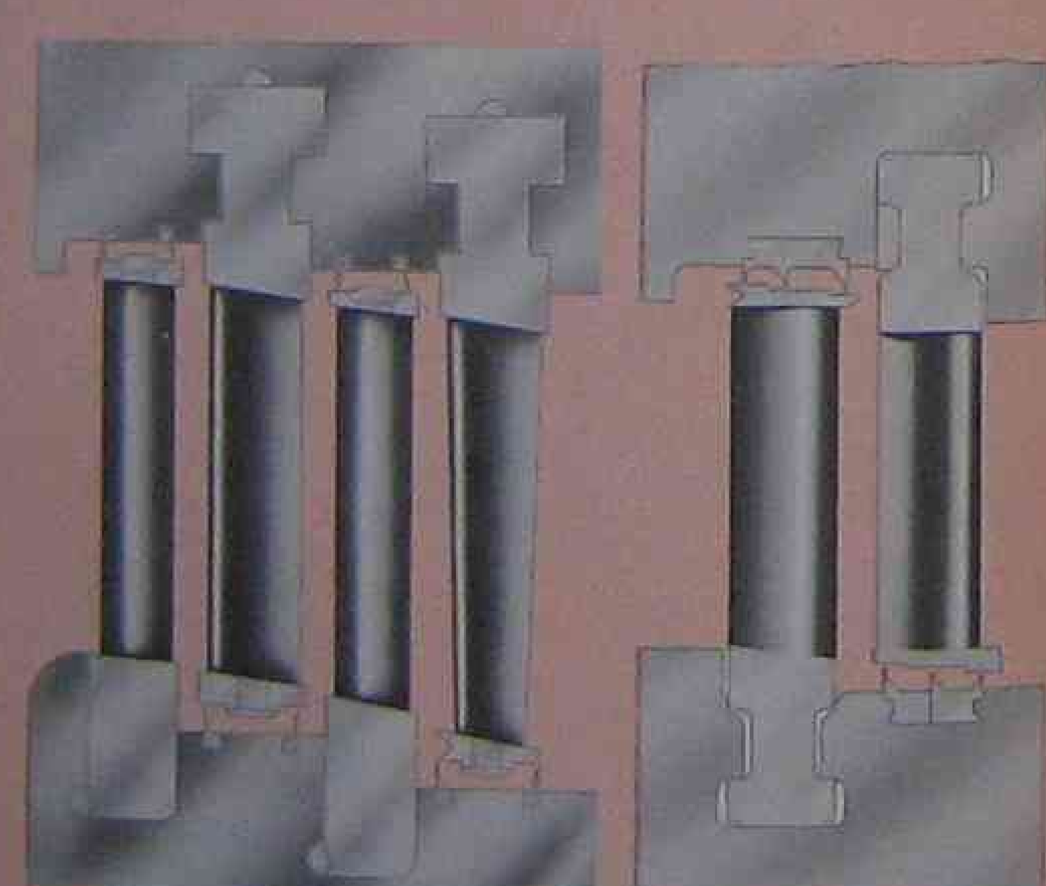
At the tip exit, vector difference of relative steam speed and blade speed indicates that steam leaves with low velocity in an axial direction, same as at the root.



8 Two rows of moving buckets with nozzle between have sealing strips over shroud bands to limit steam leakage

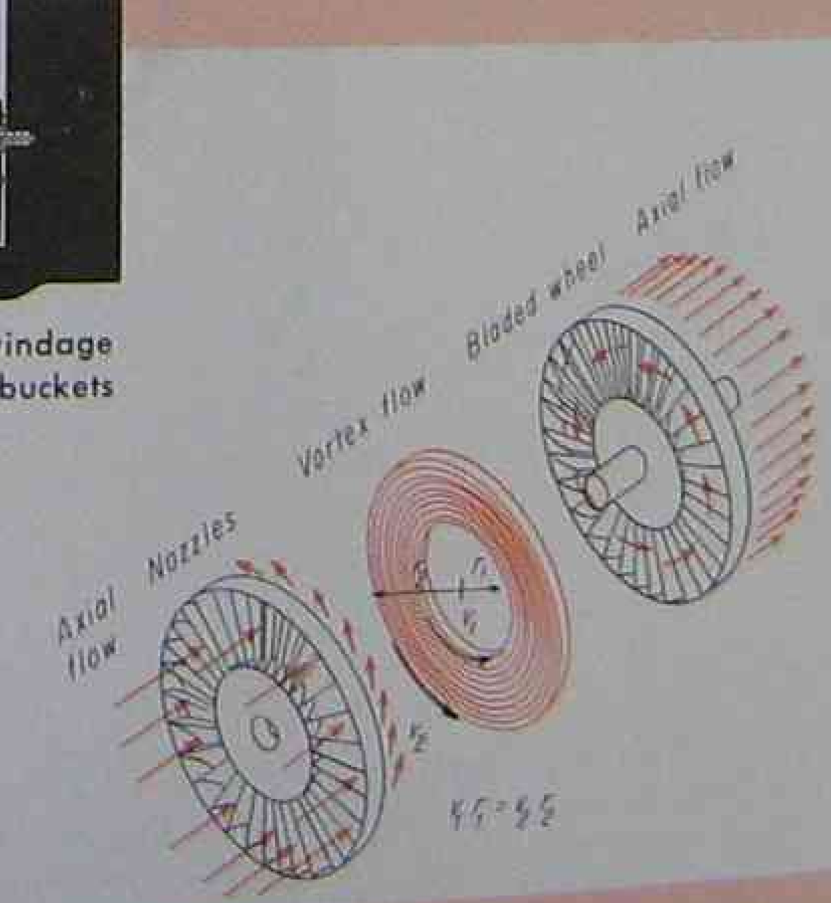
Designs of blades, roots, shrouds, seals and shields influence unit efficiency

9 Side-entry blades dovetail into disks to transmit force to turbine shaft. Shrouds tie groups of blades together

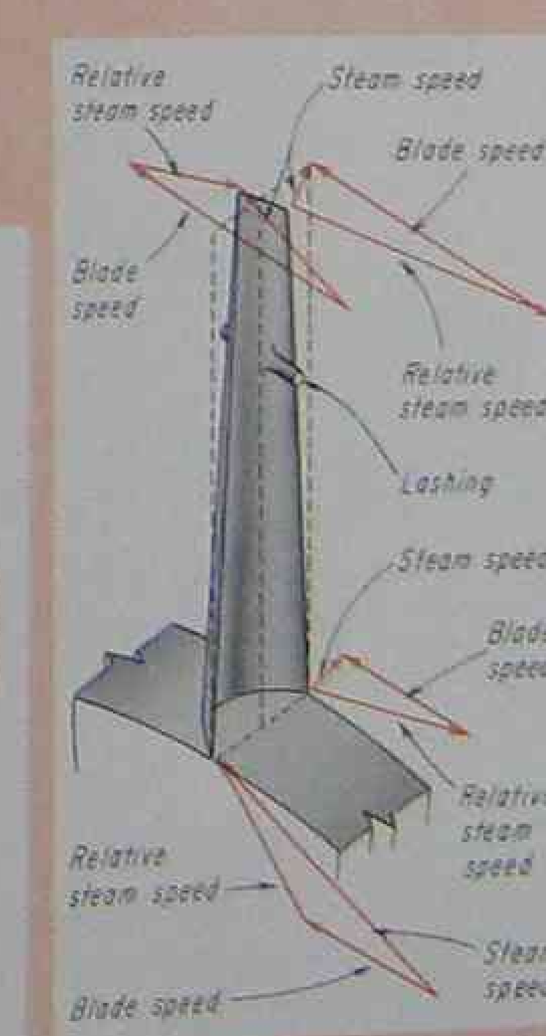


10 Reaction stages of drum type unit need sealing strips, which limit steam flow past stationary, moving blading

11 Shields cut windage loss on idle buckets

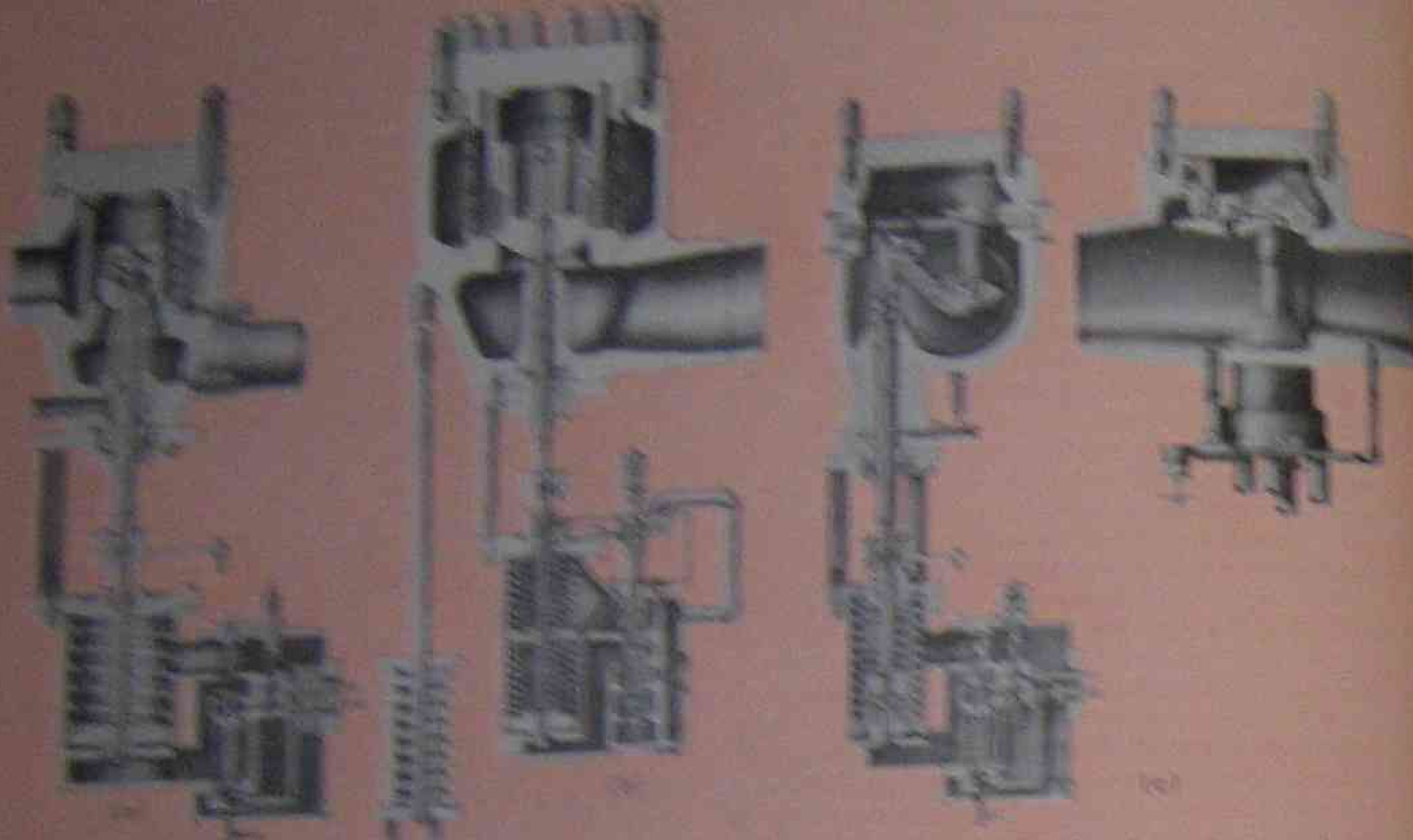


12 Steam flows axially in entering-stage nozzles, leaves with vortex flow. Twisted blades return it to axial

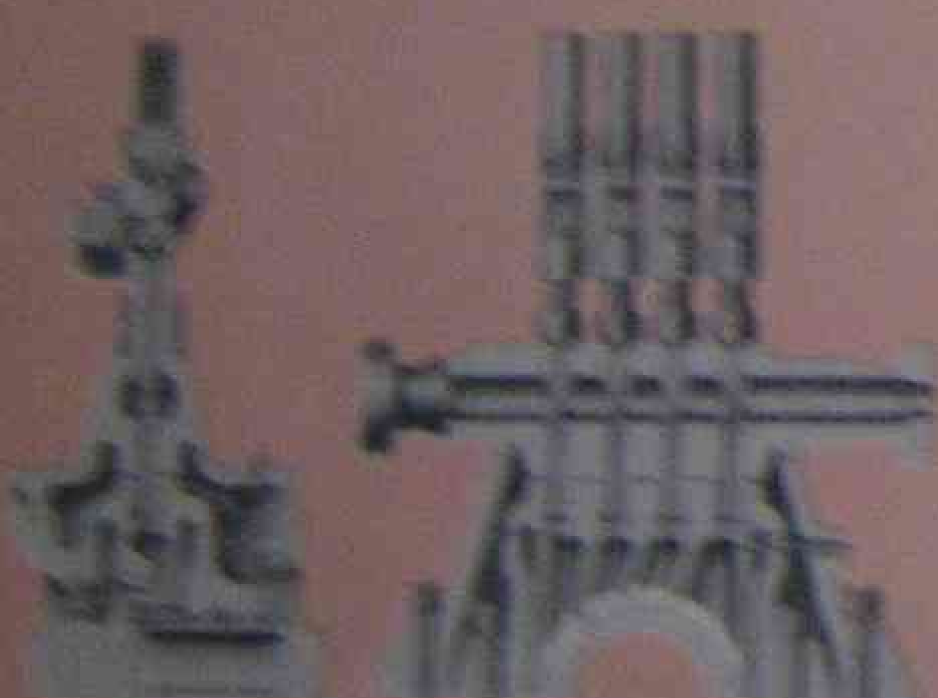


13 Twisted blade works with impulse flow at root, reaction at tip

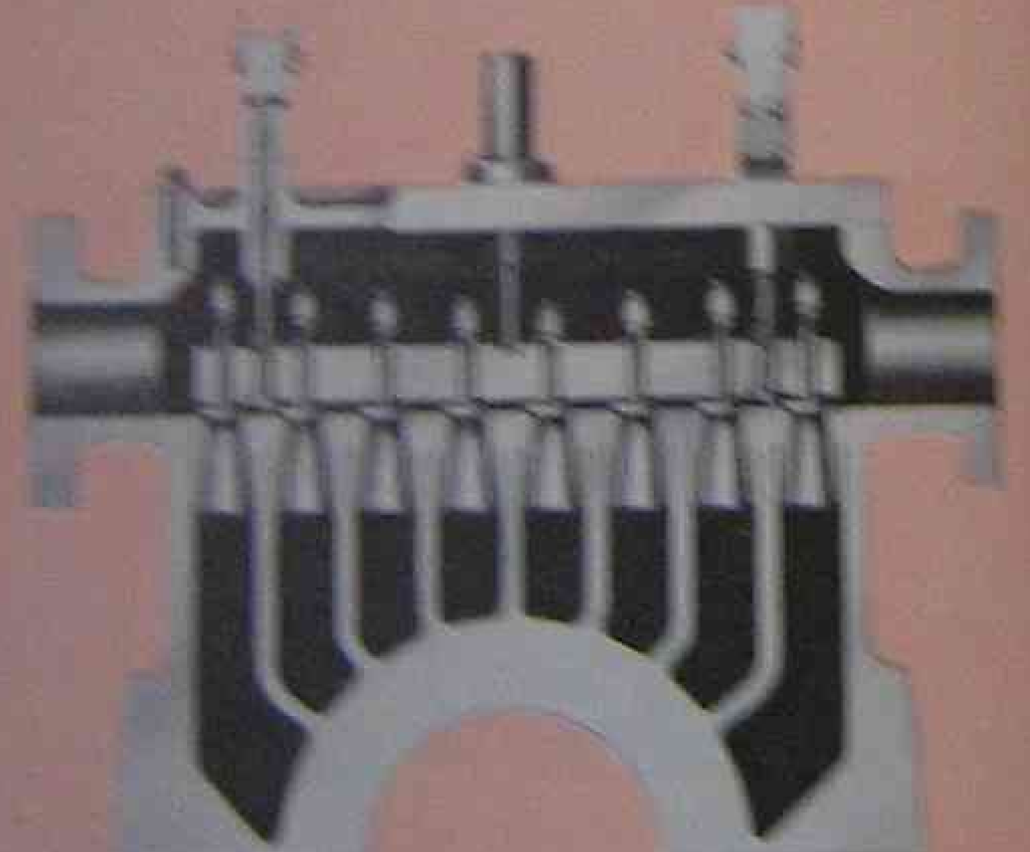
Automatic throttle, stop, check and steam-chest governing valves control flow and pressure of steam through the turbine



14 (a) A throttle-stop valve controls steam to all nozzles; in tripping, shuts off flow in emergencies. (b) Reheat-unit intercept valve cuts flow from reheater on large load changes. (c) It's backed up by stop valve in series.



15 Free-expanding nozzle chest has nozzle groups controlled by cam-operated poppet valves in steam chest



16 Crossbar in steam chest, lifted by two rods, opens poppet valves in sequence to control flow to the nozzles

17 Combined throttle-stop valve and steam chest for 4-valve may be placed in front of the turbine or to one side. Flexible pipes tie it to nozzle groups

18 Piston-operated grid valve controls flow in automatic-extraction unit

19 Balanced throttle valve controls steam flow to smaller turbines

Rotors for disk-and-diaphragm turbines may be machined from a single forging for smaller-capacity units, or built up of separate disks shrunk and keyed on a forged and machined shaft. Edges of the disk are milled to hold the buckets. Shaft usually has integral journals for the bearings and also carries a thrust collar. Drum type turbine rotors increase their diameter from front to rear end of the unit. They're usually made as one forging, then milled to receive the blade roots.

Casings hold the blade rings, bearings and other stationary parts. A small unit's casing may split vertically, but all large ones split horizontally at the shaft center line. For high-pressure, high-temperature service, front elements have been built with two casings—steam flows between the inner and outer one. This design allows thinner shells, so expansion and contraction aren't so much of a problem. Casings may be cast or built up of welded plate.

Valves control the flow of steam through a turbine. Combined stop-throttle valve, Fig. 14a, regulates steam flow during warmup by opening the inner valve first. When the unit is ready to take load, main valve is opened wide and nozzle-chest valves pick up the control function. In an emergency, the stop valve slams shut when oil pressure is released from the operating cylinder below. This oil pressure can be regulated manually during startups. Fig. 17 is a throttle-stop valve on a steam chest.

Reheat turbines need both intercept and stop valves, Fig. 14b and c, between boiler reheater and turbine-reheat inlet. Steam in the reheater and its connecting lines can seriously overspeed the unit during a large load drop or loss. The intercept valve closes part way during rapid load changes; both valves shut on overspeed.

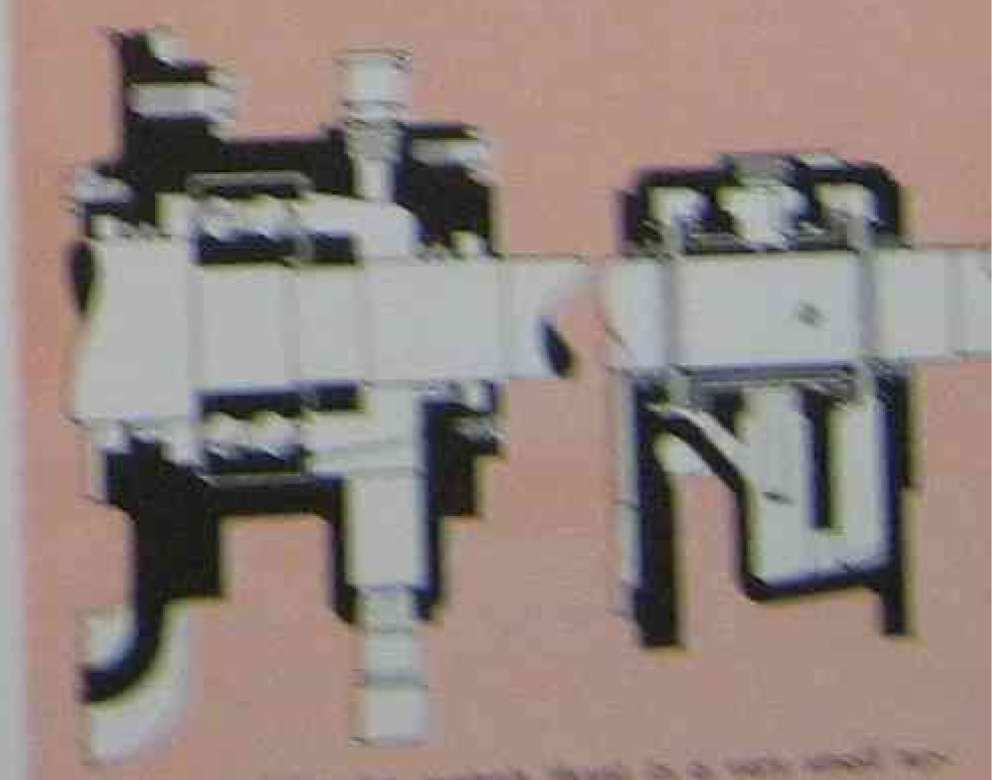
Changes in steam flow to meet varying loads are controlled by valves in a steam chest, Fig. 15, 16, 17. These admit steam to groups of nozzle chests, an arrangement giving high part-load efficiency. Valves in Fig. 15 are opened by cams on a governor-controlled shaft. In Fig. 16 the bar in the steam chest is lifted by two rods moved under governor control. Stems have varying lengths to open the valves in sequence. Fig. 17 shows an entirely separate steam chest; here the valves are lifted by an external hinged lever.

Balanced governor-controlled single-admission valve, Fig. 19, handles small-steam-turbine flow. It may feed several paralleled nozzle chests on the unit; intervening hand valves can cut them in and out as wanted.

In automatic-extraction turbines, three different types of valves may control steam flow through the downstream sections. On p. 8, cam-operated poppet valves control the first opening, with cam-operated spool valves for the second. Piston-operated grid valve, Fig. 18, may be used for

Bearings range from journals to simple ball types.

20 Journal bearing has spherical seat for self-alignment. Groove in top builds oil wedge, stops shaft whipping

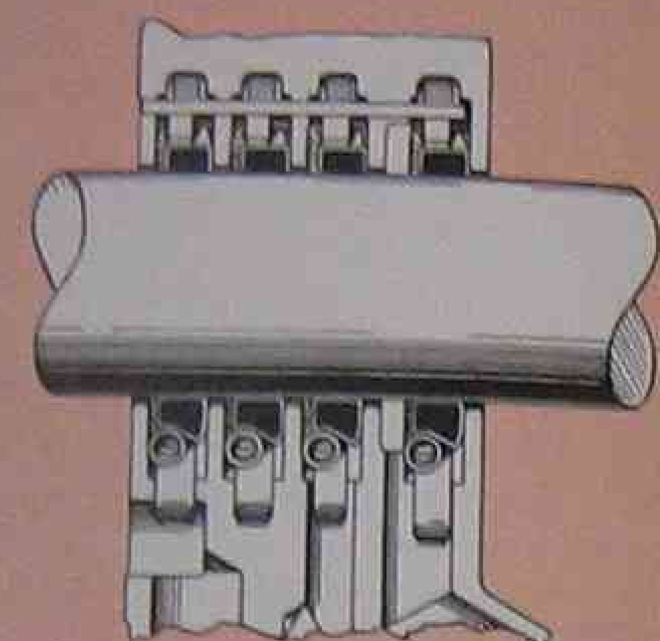


21 (a) Ball bearing controls thrust in a very small turbine. (b) Ring-oiled journal is used on small units

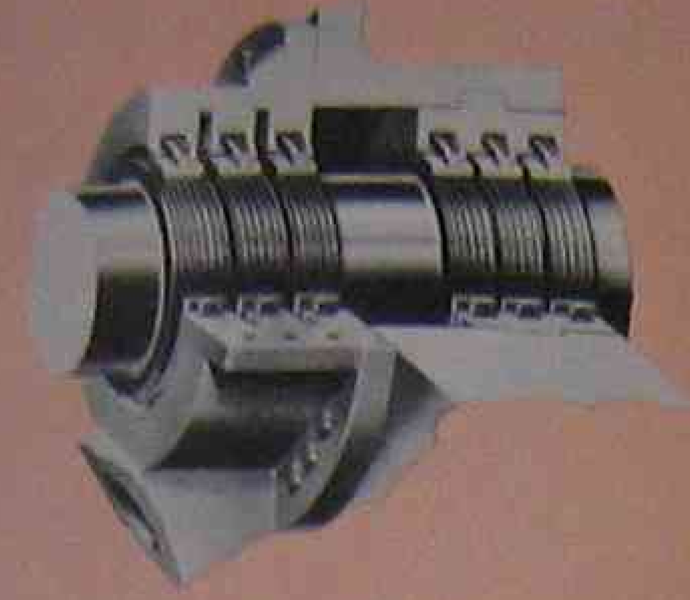
Thrust bearings hold rotor in position; shaft seals limit steam leakage

22 In Kingsbury type of thrust bearing, tilting shoes form oil wedges

23 Ball-seat bearing has tapered lands for shaft thrust bearing



24 Carbon packing rings form simple seal against shaft steam leakage



25 Stepped labyrinth-gland seal limits both shaft steam and air leaks

low pressure drop across the valve—about 50 psi or less.

Bearings range from pressure-lubricated journal types for large units, Fig. 20, through ball bearings for very small turbines, Fig. 21*a*, and ring-oiled journal bearings for small units, Fig. 21*b*. Larger bearings are almost universally designed with oil grooves in their top halves to build an oil wedge that presses down on the journal.

Steam-pressure differential across most turbine stages creates a net thrust along the shaft. This must be counterbalanced to keep the rotor in proper position. Fig. 22 shows a Kingsbury thrust bearing. Individual movable thrust shoes bear on leveling plates, left. Thrust collar, fixed to the shaft, pushes on the shoes from the right, and holds clearance between moving and fixed components.

Fig. 23 shows bearing surfaces, on the side of a journal-bearing housing, for a tapered-land thrust bearing. A thrust collar fixed on the shaft pushes against the surfaces to hold the shaft in place. Oil wedges build up between collar and tapered lands.

Steam leakage can't be avoided entirely. Some steam will leak out of turbine casings where the shaft must pass through. In condensing turbines, air tries to leak along the shaft into the low-pressure condensing space.

We've seen sealing arrangements used to limit leakage around blades. Same idea comes in for shafts. In a slant-top carbon packing-ring seal, Fig. 24, carbon packing rides directly on the shaft. Springs anchored at the top hold packing segments in place and against the sealing surfaces at right. Steam that does manage to leak past the seals may be led to a lower-pressure stage of the turbine, to a heater, or vented to atmosphere. Condensed steam usually accumulates in the last section of the seal, then drains to waste.

Step-type labyrinth gland, Fig. 25, also controls shaft leakage. Intermediate leakoffs direct the steam to lower turbine stages or heaters. The large intermediate chamber may connect to the suction of a blower which holds a vacuum lower than the turbine's last stage. This would draw in l-p steam from the right and air from the left. Blower discharges the mixture to a condenser where the steam is recovered. L-p steam might be fed into the outer left chamber to block air flow into the seal.

Seal strips have tapered edges so any accidental rubbing will wear them down quickly without overheating the shaft. Steps milled in the shaft match longer seal strips, forming a long, tortuous path with high flow resistance. Enough axial distance is needed between strip and step to avoid contact when shaft and casing expand or contract at different rates. Where movement is relatively large, the shaft may be smooth with the seal strips all of equal depth. Some turbines use labyrinth seals in series, then add a set of carbon-ring seals at the shaft ends.

MATERIALS include steels, newer alloys

The many parts of a steam turbine work under varying service conditions. Effective design selects, from appropriate materials for each part, combination promising long life at low cost. Critical parts of a modern turbine are those handling the entering steam. It's at high pressure and temperature, because higher entering conditions mean more energy can be converted to shaft work. But using too costly materials can wipe out fuel savings.

Casings . . .

as well as steam and nozzle chests are usually cast of steel. A sample composition for 825-F service may be 0.3 C, 1.0 Mn, 0.6 Si, 0.06 S, 0.05 P. At the other extreme, 1050-1100 F may use austenitic steel with 0.08 C, 16.00 Cr, 13.00 Ni, 2.00 Mo and 0.8 Cb. Wrought parts: 0.08 C, 18.00 Cr, 9.00 Ni, 0.8 Cb.

Rotors . . .

are made up of shafts, disks or drums and blades or buckets; each element offers a choice of materials. Shafts for lower temperatures vary from hot-rolled heat-treated carbon-steel bar stock to alloy-steel forgings. Up to 1050 F the shaft may be a forging: 0.37 C, 1.00 Mn, 0.35 Si, 0.035 S, 0.035 P, 1.25 Cr, 1.50 Mo, 0.50 Ni and 0.30 V.

At 1050 F, drum would be a ferritic-alloy forging: 0.30 C, 0.50 Ni, 1.00 Cr, 1.25 Mo and 0.25 V.

Wheels for 650-F service may be 0.45 C, 0.90 Mn, 0.15 Si, 0.035 S, 0.035 P. H-t wheels are usually of the same composition as h-t drums.

Buckets . . .

made of cold-rolled drawn stock may be stainless steel: 0.06 C, 0.25 Mn, 0.50 Si, 0.03 S, 0.03 P, 11.5 Cr, 0.40 Mo, 0.5 Ni. Milled blades for h-t service would be about the same as the drawn blades, but made from hot-rolled bars. Shroud bands would be annealed stainless strips, perhaps with the Mo left out. Stellite shields may be brazed on leading edges of l-p buckets running in the wet-steam region.

Nozzle rings and diaphragms

for 500-F service are often cast iron or Meehanite; as temperatures go higher, materials range through steel plate, cast steel, steel and stainless-steel forgings. Some diaphragm nozzles for 950 F are rolled, some cast aluminum-chromium steel.

Seal and gland packing . . .

around the shaft varies from carbon to stainless steel, leaded bronze, leaded nickel-brass, non-hardened stainless iron and corrosion-resistant chrome-molybdenum materials. Springs to hold packing in place vary: Inconel, monel, stainless steel.

Bearings . . .

use high-tin babbitt as the inner lining, cast on bronze, steel or cast-iron backs. Their shells are often cold-rolled steel. Journals and thrust collars are usually an integral part of the shaft, and of the same material, though some journals are built up of sprayed metal to make a long-wearing hard surface. Oil rings, like smaller bearings, are usually brass—a softer material than the journal.

Bolting . . .

of high-pressure casings running at high temperature raises special problems because of creep. Gradual elongation or growth of materials under stress relaxes bolt's hold on the casing joints. One builder uses a 13-chrome tungsten-moly-vanadium alloy steel for 925 F and above. This material resists temper embrittlement and oxidation, has higher notched-bar rupture strength.

As bolts creep they must be tightened periodically; when they're strained to their rated level they must be replaced. If bolt material has a lower coefficient of thermal expansion than the flange, less initial tightening stress is required.

Piping . . .

materials range from carbon steel for temperatures below 900 F and medium pressures to Type 316 stainless steel for service up to 1150 F. All materials lose rupture strength as temperature rises. For example, at 800 F, Type 316 has a 100,000-hr rupture strength around 47,000 psi; at 1150 F it drops to 14,000 psi. H-p piping must have heavy walls.

Inlet pipe seals . . .

for turbines in the 1000-1050-F range may be stellite. These sealing rings let pipe connections between separate steam chest and nozzle chests move axially and transversely during startups and shutdowns. Piping oxidation at joints must be prevented. It freezes the seal rings; resulting rigid connection transmits piping expansion and contracting forces to the turbine casing—perhaps causing serious misalignment. In one design, stellite rings bear against a terminal ferritic-stainless-steel section welded on the pipe leads.

Governing valves . . .

at the front end of a turbine are usually made, like the steam chest, of a carbon-chrome alloy, one of the alloy steels. Stems must resist oxidation to prevent freezing in the packing. One manufacturer plates them with 13-chrome tungsten-moly-vanadium to be sure of good service above 950 F.

TURBINE DESIGNS span broad system needs

In industry, steam turbines handle an almost endless variety of jobs. Top steam conditions are 4500 psig and 1200 F, lows are subatmospheric pressure and 100 F.

Capacities begin at less than 1 hp for mechanical drives, run up to 1,000,000 kw for generator drives—with larger sizes in sight. In the U.S.A., turbine-generators usually run at 1800 and 3600 rpm; other countries often use 1500 and 3000 rpm. Industrial turbines run up to 20,000 rpm for lower capabilities with smaller diameter rotors.

Basic types of turbines, opposite page, divide into two main classes: condensing units exhausting steam at less than atmospheric pressure; *noncondensing* units exhausting at higher than atmospheric.

These classes may be further sorted according to steam flow in the turbine: (1) *straight-flow* (2) *reheat* (3) *automatic-extraction* (4) *nonautomatic-extraction.*

Straight-flow uses full-throttle steam from nozzles to exhaust. In reheat, main steam flow exhausts from the unit at an intermediate stage. It's resuperheated, usually in the boiler furnace, then returns at the next lower stage for further expansion to exhaust. Some units have two stages of reheating.

Automatic-extraction unit bleeds off part of the main steam flow at one, two or three points. Valved partitions between selected turbine stages control extracted-steam pressure at the desired level. Proportion of main steam flow extracted for process at various points may be controlled in any manner within design limitations. When extracted steam flow through the unit does not produce enough shaft power to meet the demand, more steam flows through to exhaust. These turbines are put between steam-supply and process-steam headers, diagram below. Automatic governing systems correlate steam flows, pressures, shaft speed and shaft output for any one unit.

Nonautomatic-extraction turbines bleed steam at as many as nine different stages. Pressure of extracted steam at each stage varies with the turbine shaft load. But pressure variations like this can seldom be tolerated for process work. So these units usually work as generator drives; extracted steam is used for feedwater heating.

Generator-drive turbines often combine condensing, reheating and nonautomatic-extraction. Mechanical-drive turbines are designed to meet varied needs of industry for prime movers to drive fans, pumps, compressors and other machines. This service generally calls for small noncondensing units with exhaust steam used for heating.

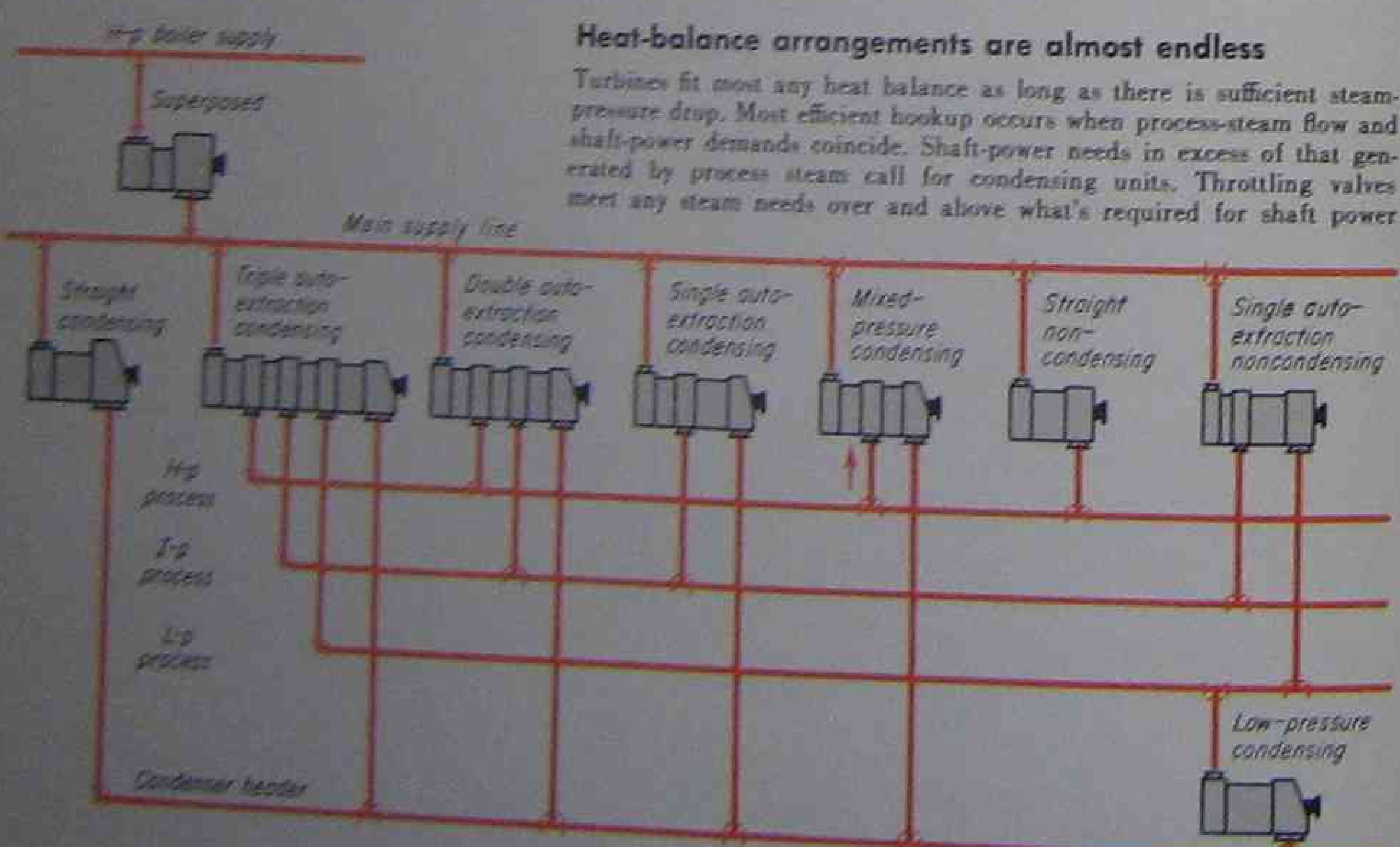
Special types of turbines include the double-flow-exhaust, sometimes used for mechanical-drive units. This design permits high shaft speeds and low backpressures where steam volume grows enormously and calls for very large passages between last-stage buckets.

Mixed-pressure turbines take steam from h-p or l-p sources or both to generate shaft power. Automatic-extraction units may also act as mixed-pressure turbines.

Straight-noncondensing turbines work over a wide range of throttle and exhaust conditions. When extra shaft power and steam are needed, an additional h-p turbine can be added. Supplied with steam by a new h-p boiler, superposed unit in diagram below exhausts at high backpressure into existing main steam-supply lines. Typical turbines of each class and construction are illustrated on the following pages.

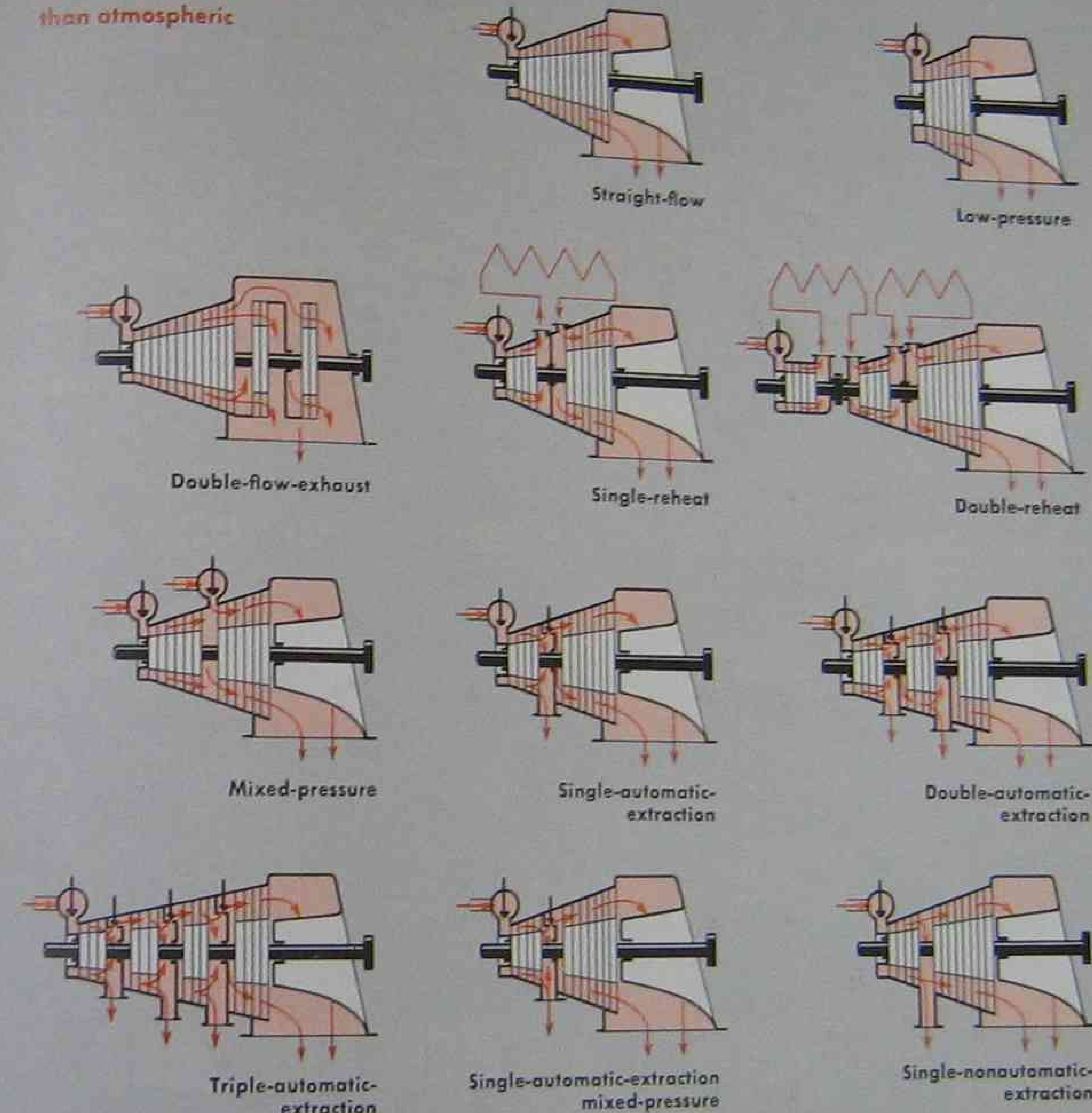
Heat-balance arrangements are almost endless

Turbines fit most any heat balance as long as there is sufficient steam-pressure drop. Most efficient hookup occurs when process-steam flow and shaft-power demands coincide. Shaft-power needs in excess of that generated by process steam call for condensing units. Throttling valves meet any steam needs over and above what's required for shaft power

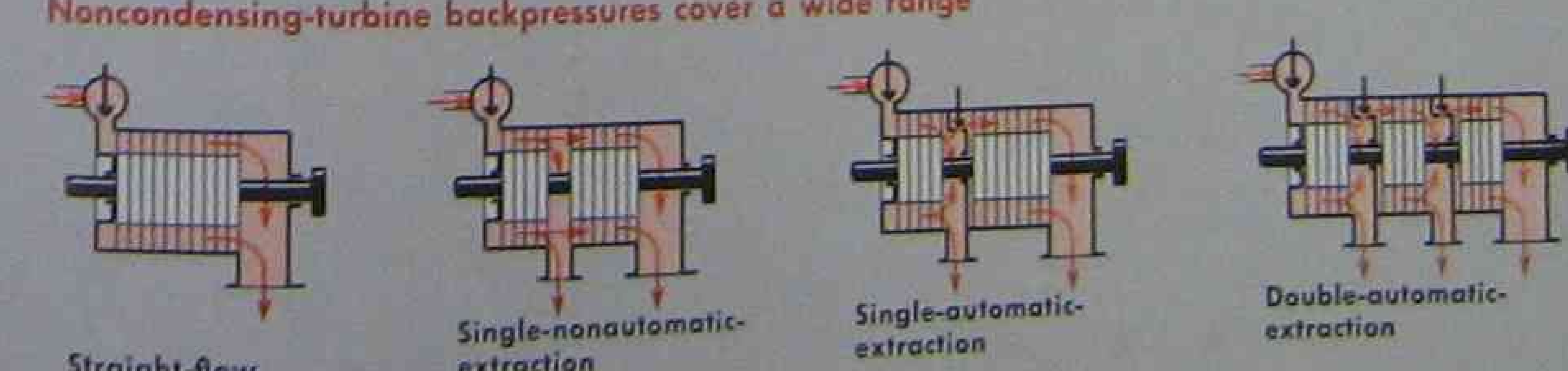


Fifteen basic turbine types: condensing and noncondensing

Condensing turbines exhaust at backpressures less than atmospheric

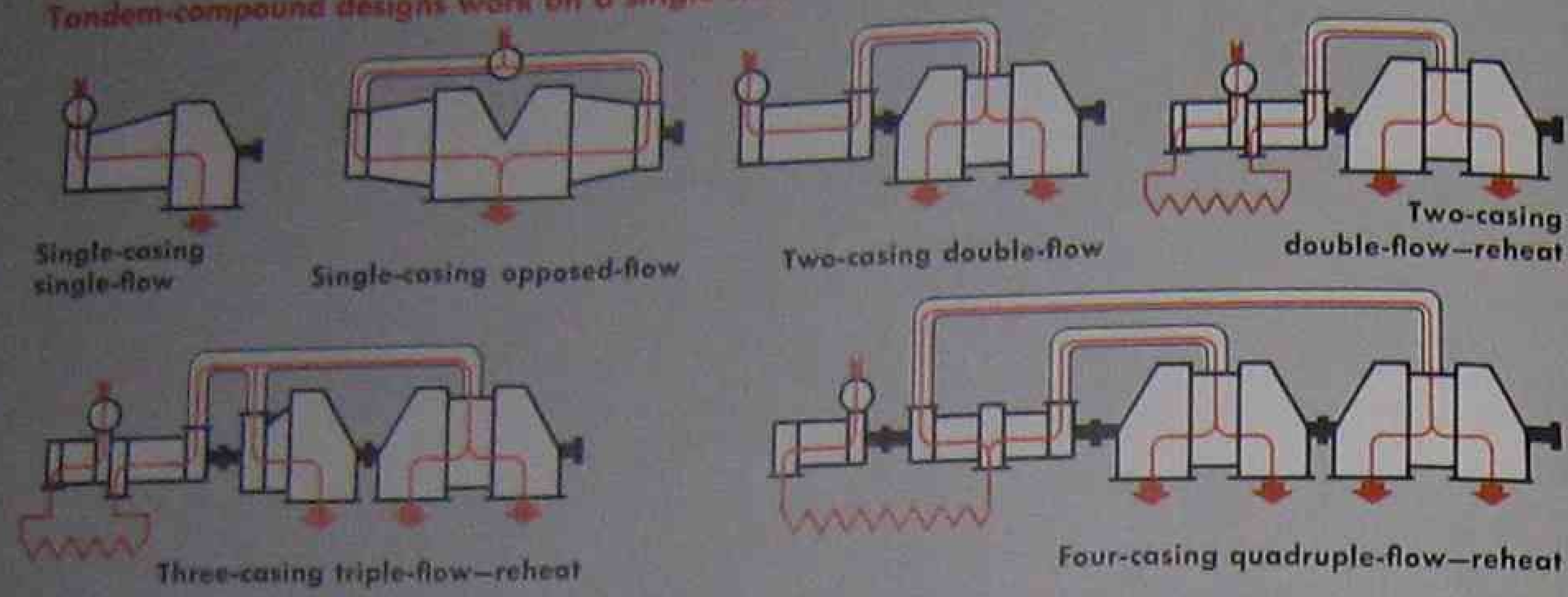


Noncondensing-turbine backpressures cover a wide range

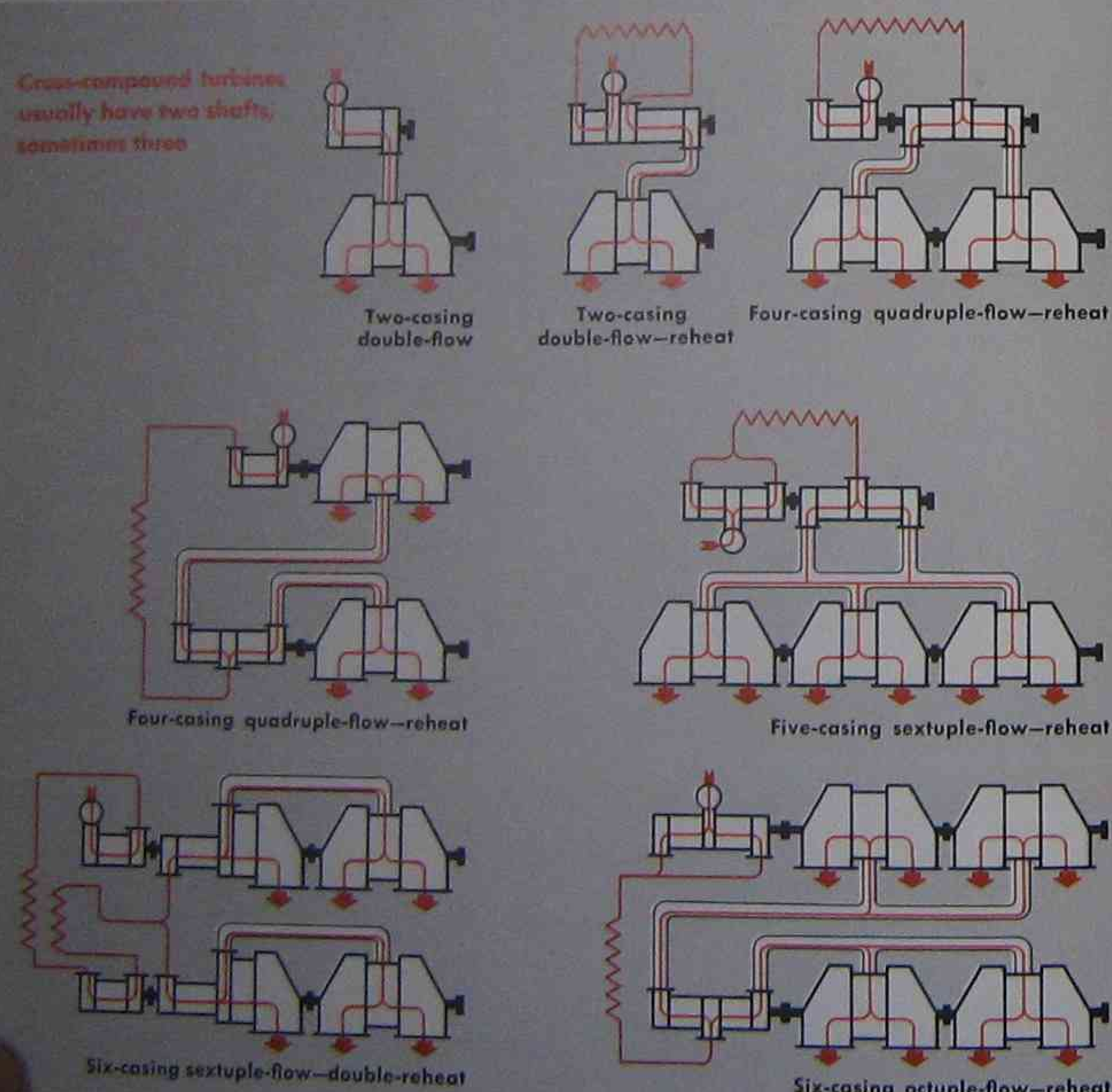


Casing and shaft arrangements depend on capacity, steam conditions

Tandem-compound designs work on a single shaft



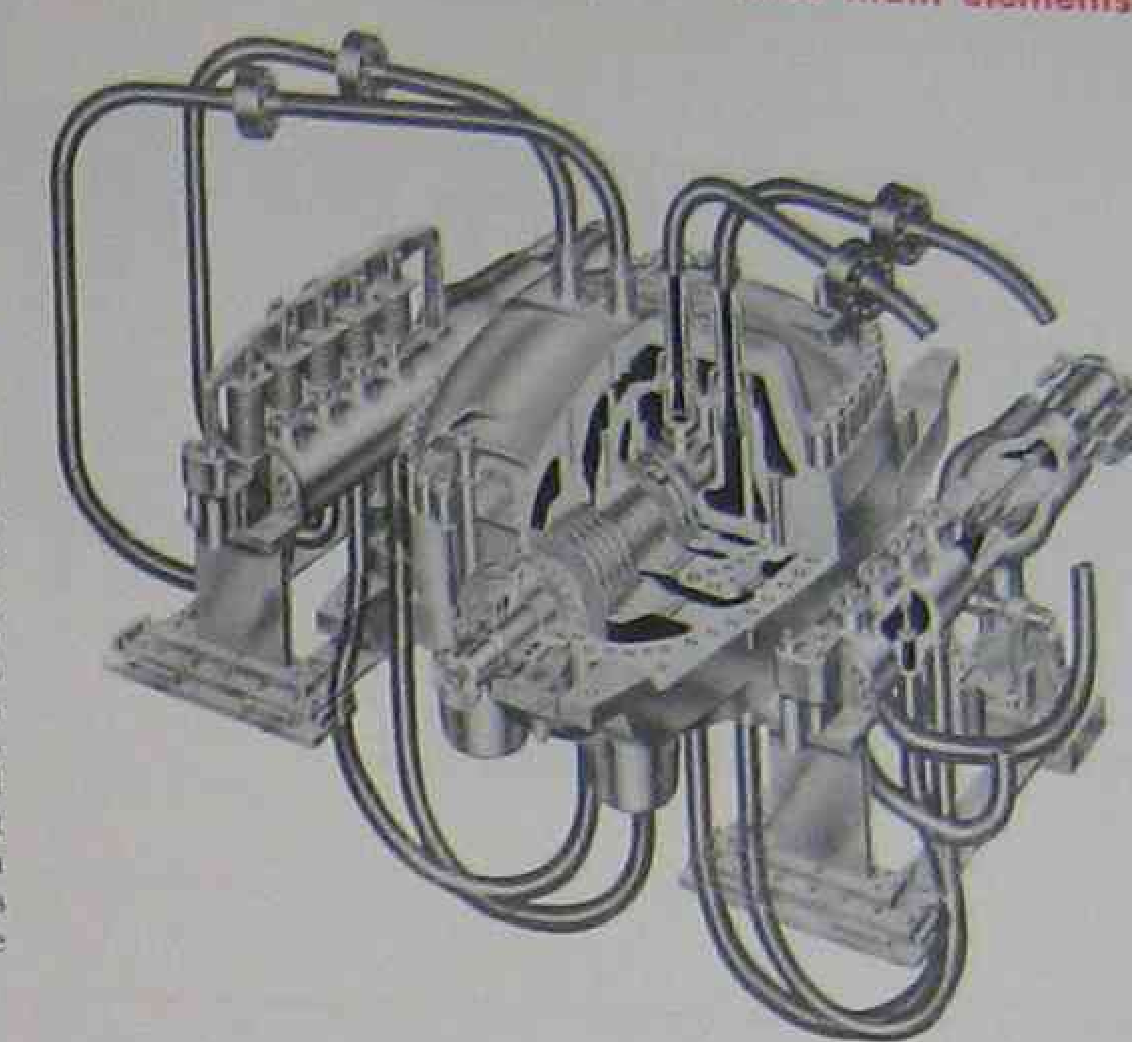
Cross-compound turbines usually have two shafts, sometimes three



Large compound turbine units may have three main elements

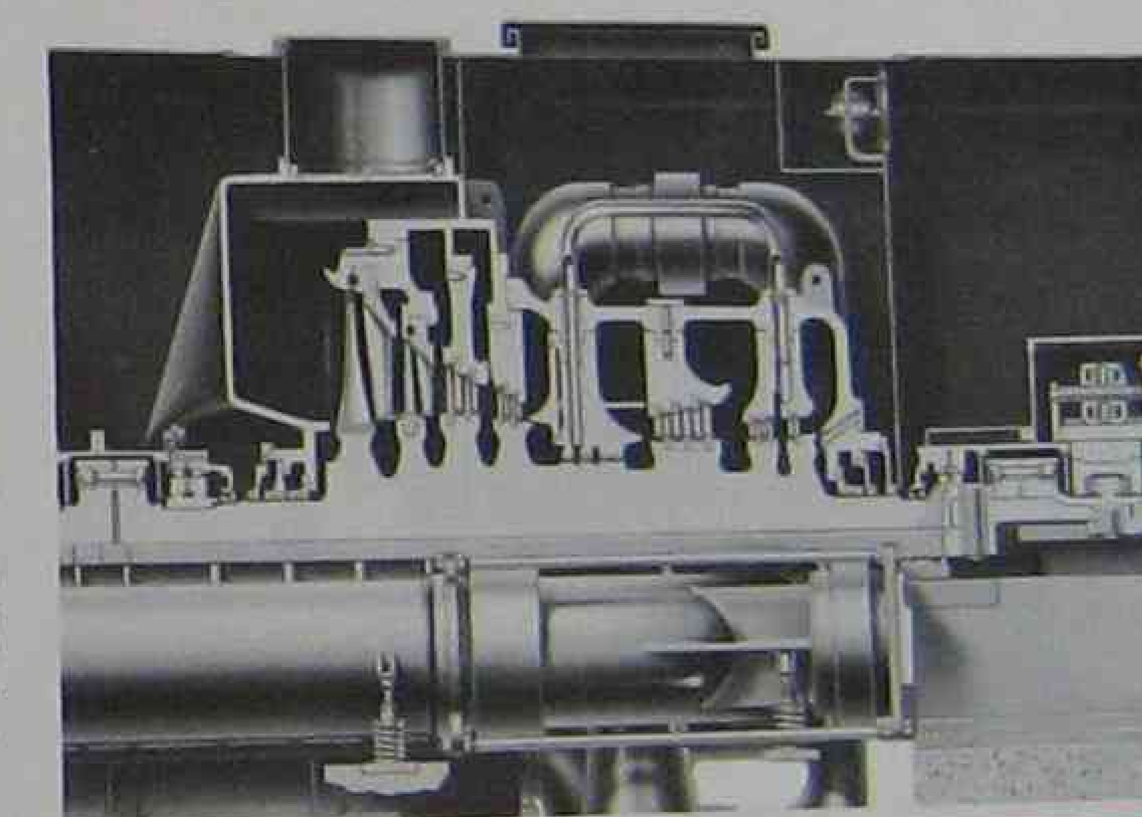
First element receives h-p h-t steam from boiler

High-pressure high-temperature casing for a tandem- or cross-compound turbine features separate steam chests with integral stop-throttle valves. Flexible inlet piping connects chests to separate nozzle chambers in turbine casing. Double-casing construction allows steam cooling for high-temperature parts; it also means the inner and outer casings can have thinner shell sections, reducing thermal stresses in shells and flanges. Exhaust from 3600-rpm element of this type usually goes to reheater in boiler furnace. Shaft gland leakage is normally piped to a lower stage of the turbine where it does useful work.



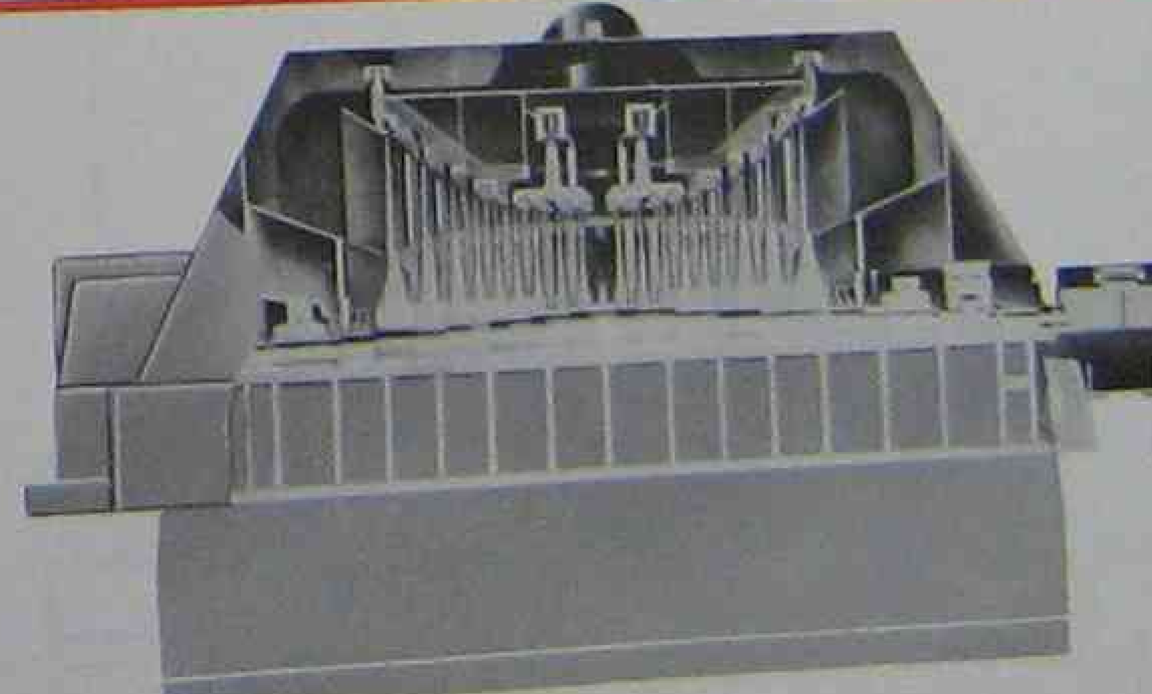
Intermediate-pressure casing takes steam from reheater

Intermediate-pressure casing is shown here along with one-third of the exhaust-flow turbine of a triple-flow unit. Steam from the boiler reheater flows through intercept and stop valves before entering initial stage of this casing (to right of center). After passing through three reaction stages, one-third of total flow goes through three more stages at right and then through crossover to enter last six stages before exhausting to condenser below. Two-thirds of flow leaves intermediate stages through cross-under, then enters double-flow low-pressure element, not shown.

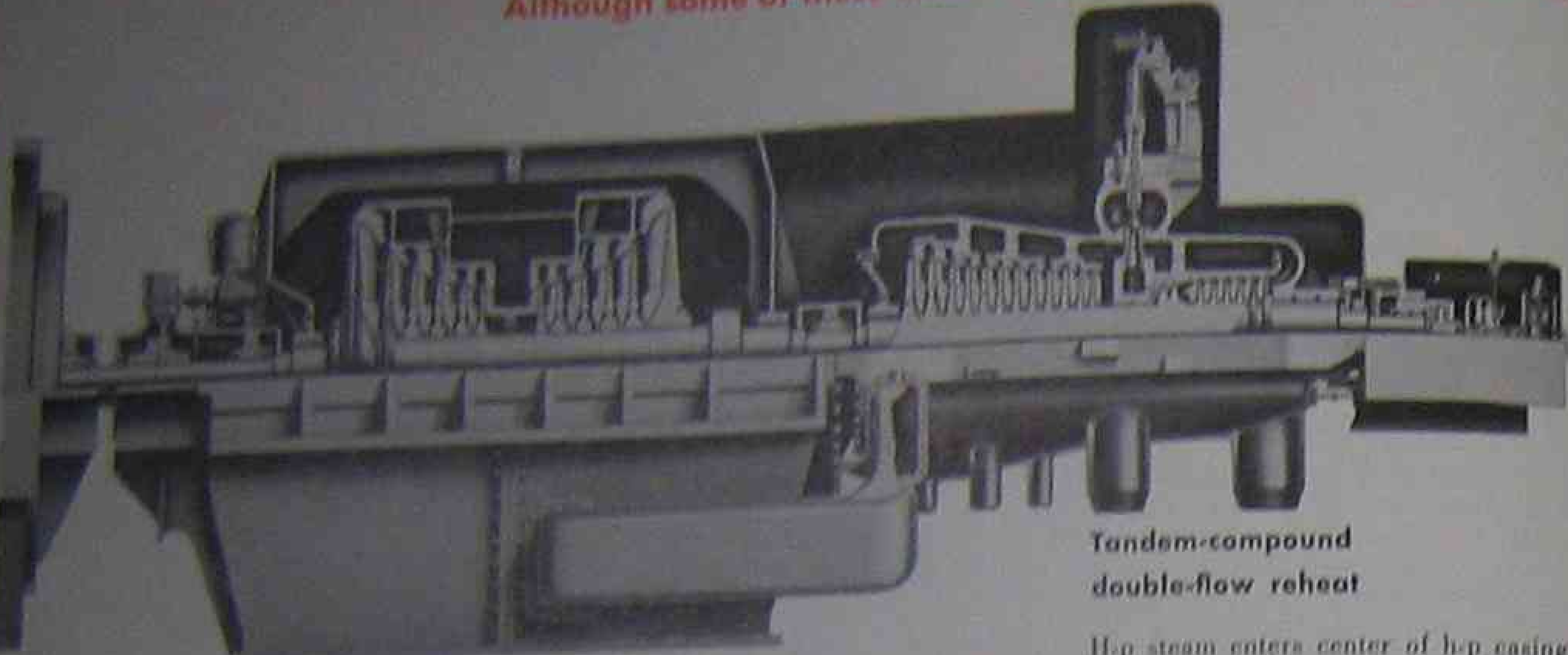


Low-pressure casing extracts remaining energy

Double-flow low-pressure casing receives steam at center through crossover from an i-p casing. Steam divides, flows through paralleled l-p stages in opposite directions. Last stages are followed by diffusers, which recover some energy from steam velocity before the steam flows downward in exhaust hoods to condenser below. Buckets are warped in the latter stages because they make up a large part of total stage diameter.

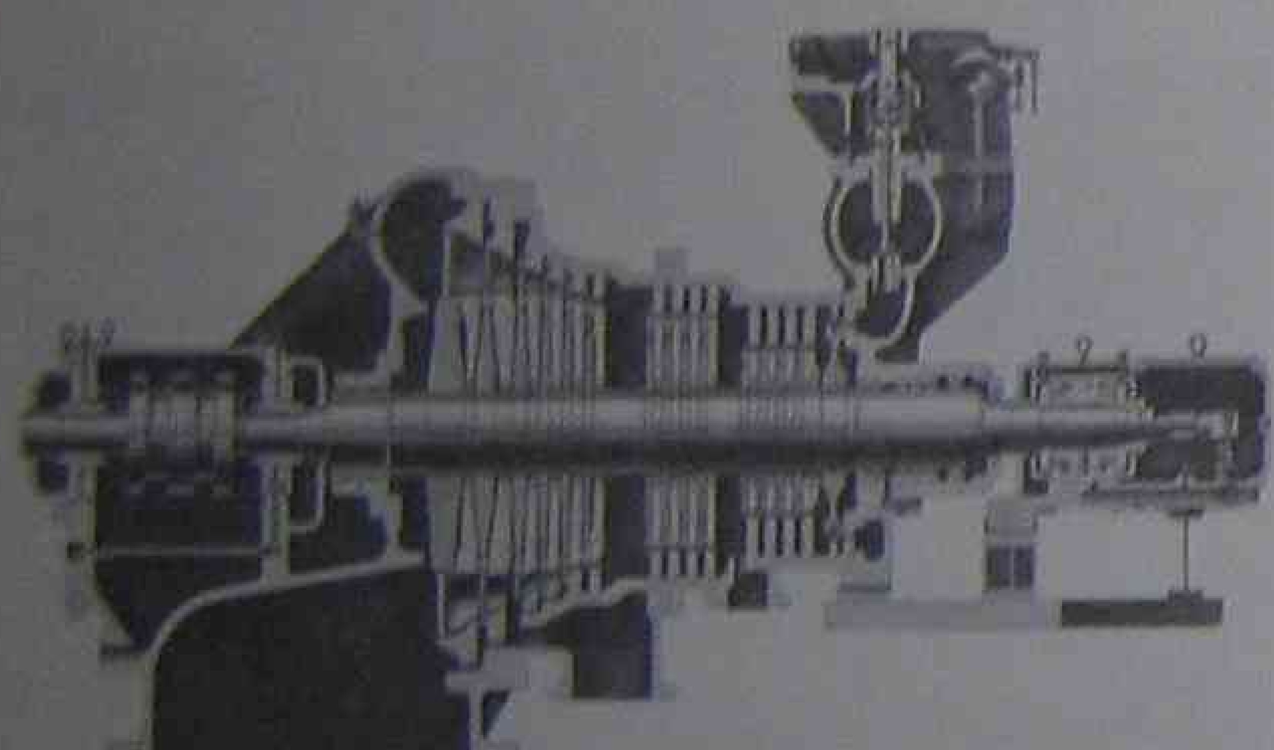


Although some of these steam turbines drive electric generators,



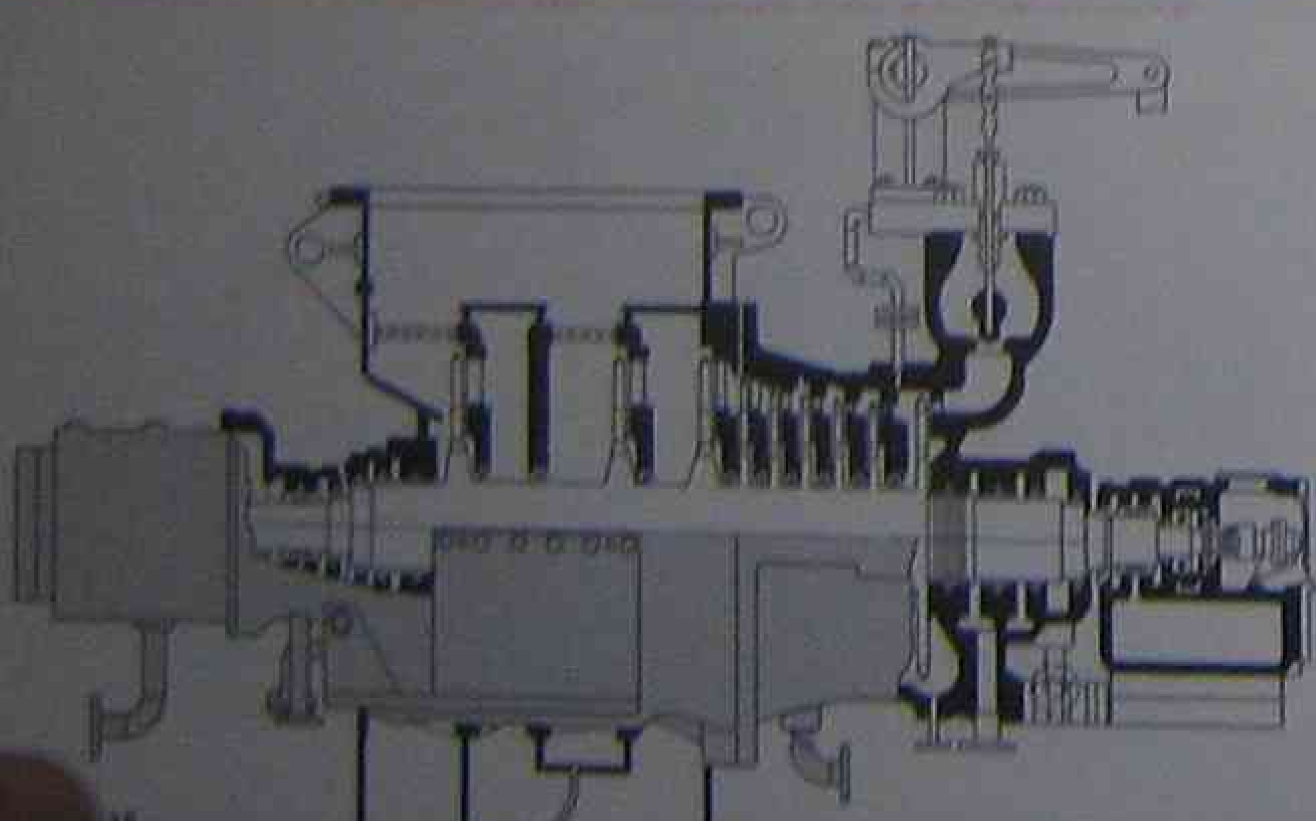
Tandem-compound double-flow reheat

H-p steam enters center of h-p casing, leaves at right end. Reheated steam enters at center and flows to left end, then through crossunder to double-flow l-p turbine exhausting to condenser.



Single-casing multistage condensing

Integral steam chest has cam-operated governing valves; velocity-compounded first stage has full arc admission. Two nonautomatic extraction points in this disk-and-diaphragm design withdraw steam for feedwater heating. Last five stages use twisted blading, varying reaction along the blade length; other stages have impulse blading. Ball-seated front journal bearing carries thrust-bearing surface on each of its sides for two thrust collars shrunk on main shaft. Diaphragms fit directly in single casing.



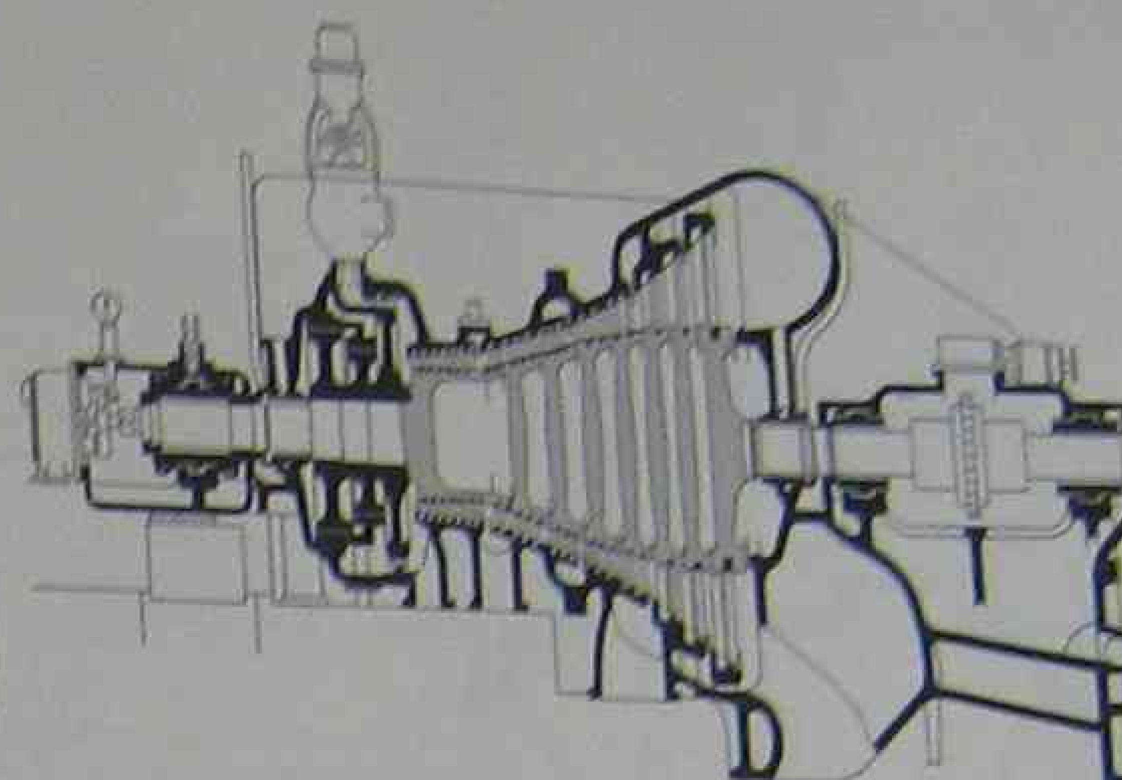
Single-casing double-flow-exhaust condensing

Steam expands from 600 psig, 750 F to 3.5-in.-Hg abs. Double-flow arrangement of last stage permits shorter blades so unit can run at higher speed—up to 6500 rpm—at good engine efficiency for mechanical-drive service. This unit has its exhaust opening at top. It can develop 13,100 shaft hp at rated speed. Disk-and-diaphragm design uses impulse blading in all its stages, has one point of nonautomatic extraction for boiler-feedwater heating.

other general-purpose units offer a wider shaft-speed range . . .

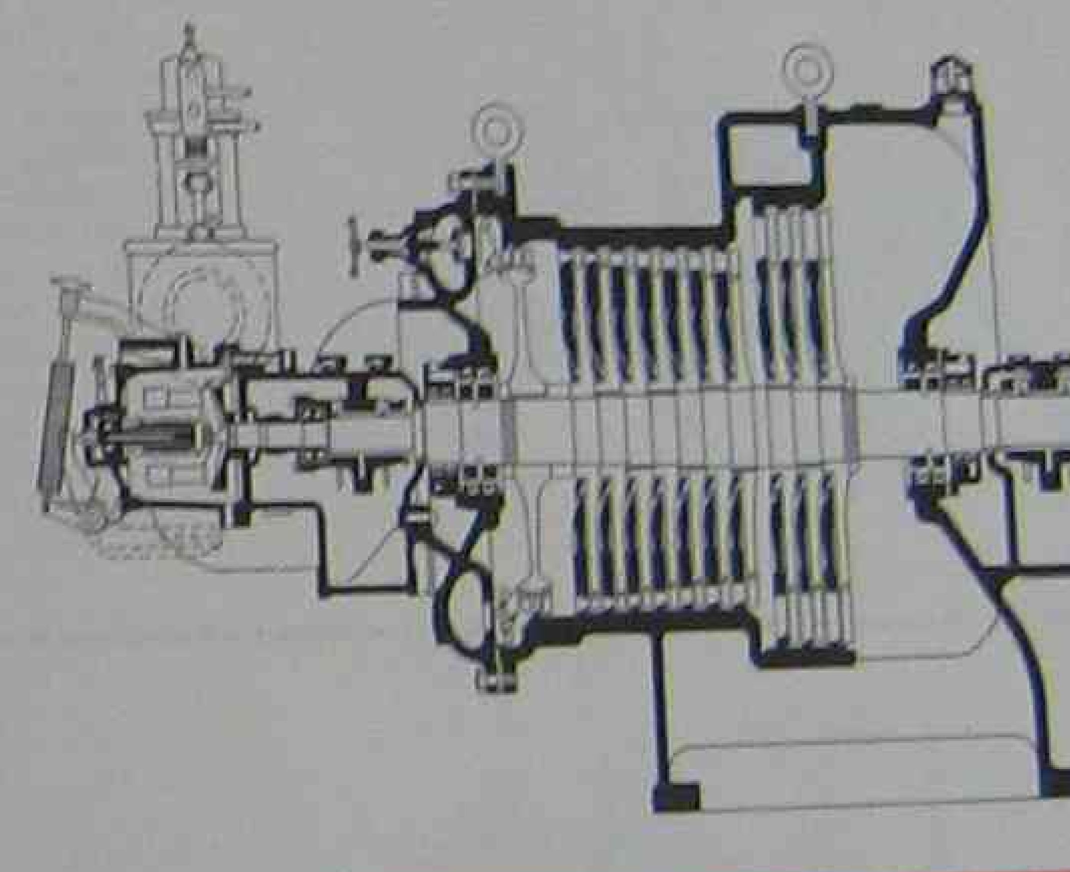
Single-casing condensing

Velocity-compounded first stage precedes all reaction stages in this drum type design. Dummy piston at left of first stage helps the thrust bearing counterbalance unbalanced force of reaction stages. This European design features a drum made up of forged sections welded together. After heat treating, drum is slotted to receive reaction blading. Steam is removed for feedwater heating at four nonautomatic-extraction points. Turning gear mounts on shaft between two journal bearings, at right.



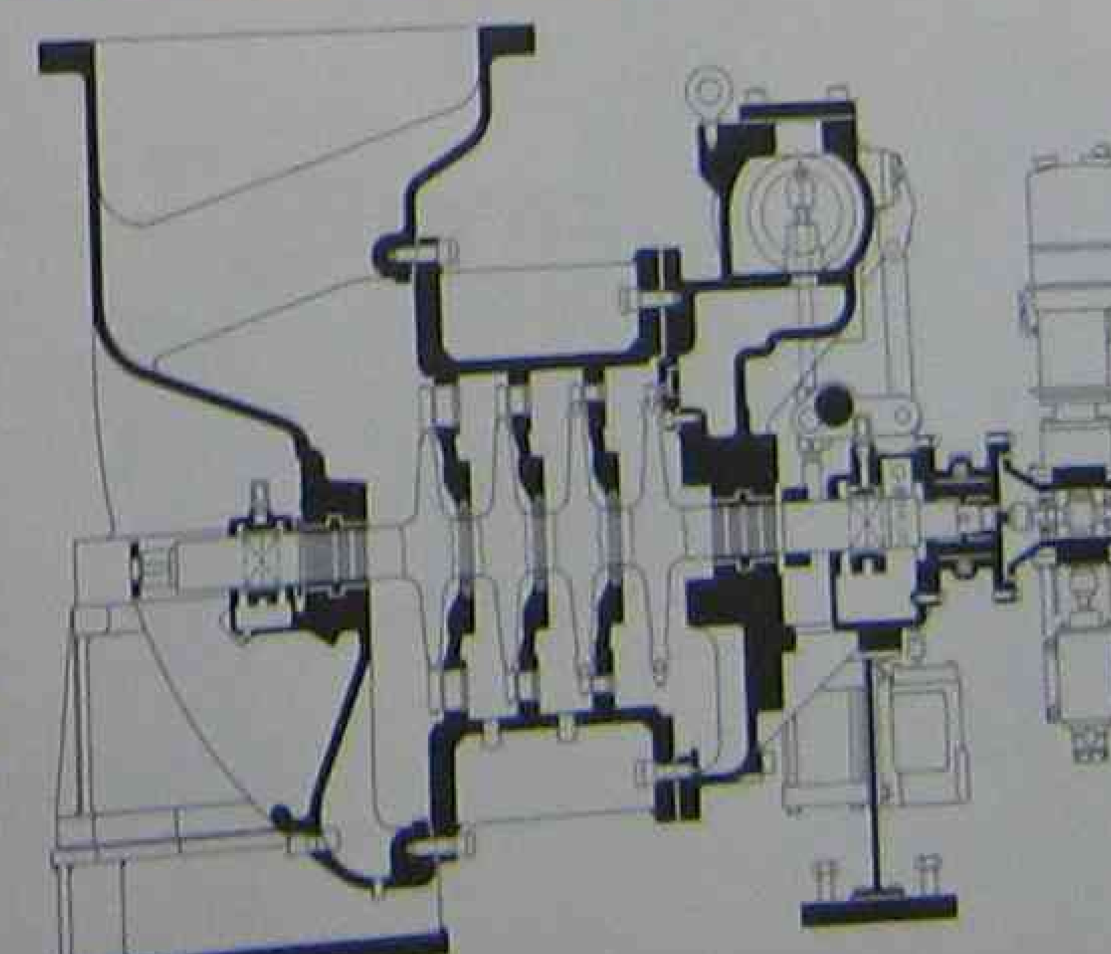
Single-casing condensing

Velocity-compounded first stage is followed by ten impulse stages. Ball-thrust bearing keeps shaft aligned axially. On left end of shaft, a centrifugal governor controls steam-flow valve through linkages. Hand valves, as on upper-left casing, control nozzle groups for more efficient part-load operation. This unit can be fitted with nonautomatic-extraction openings, to withdraw steam for feedwater heating. Carbon-ring seals are used at diaphragms and casing glands; brass oil rings, dipping into reservoirs to pick up the oil, lubricate this steam turbine's two journal bearings.

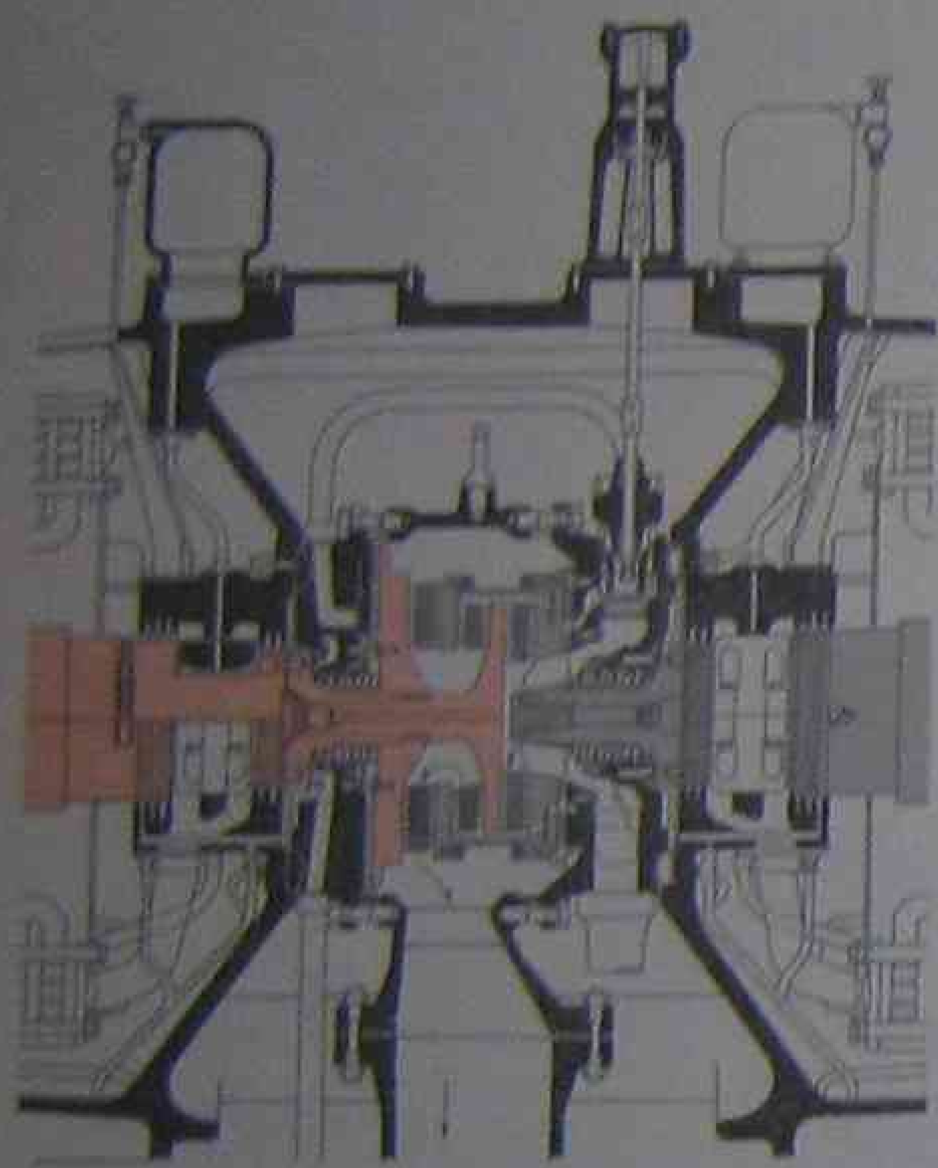


Four-stage condensing

Steam chest is equipped with bar-lift control valves; lift rods move bar from below. This disk-and-diaphragm design uses impulse staging only and exhausts through an upward-directed hood. All shaft seals are of the stepped-diaphragm type. Since inlet-steam conditions are moderate the unit does not need a velocity-compounded first stage. Governor and speed changer mount on right end of shaft. Thin web beam under front end of turbine allows axial expansion during startups with minimum strain on casing and keeps the turbine's shaft properly aligned.



. . meeting the needs of both electric-generator and mechanical drives



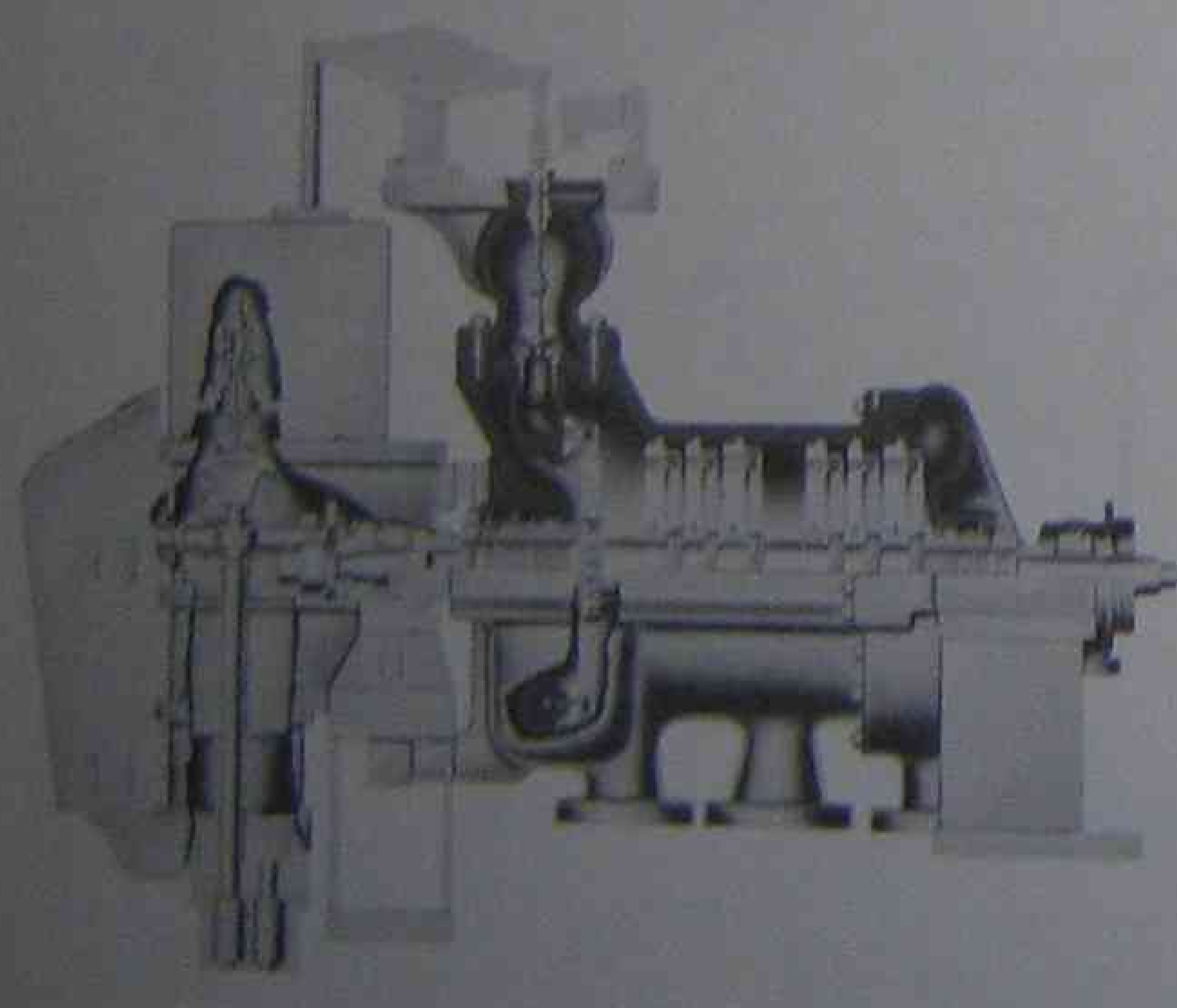
Radial-flow double-rotation

This unit drives two alternating-current generators, one on each shaft. Generators are tied together electrically, to keep the oppositely rotating shafts in synchronism for best blade-steam-speed ratio of the reaction stages. Multidisk turbine is arranged so h-p steam from supply line and throttle valve enters from below. It flows first into the annular steam chest, then through holes in the overhung blade disk to the center area at the shaft.

Oppositely rotating shaft and blade assemblies are shown in light gray and light color. Oppositely moving blades have seals to minimize steam leakage past their tips.

Steam flows outward radially through first concentric set of blades. Then it turns 180 deg to flow inward radially through a second set of concentric blades. It again makes a 180-deg turn to flow radially outward through the third set of blades. From here steam flows into annular space leading to exhaust pipe at bottom of turbine. Bypass valve to right of disk in annular steam chest lets h-p steam skip first set of blades to enter the second set, providing overload operation at reduced efficiency.

Backpressure turbine develops 7500 kw. It can be designed for automatic or nonautomatic extraction of partly expanded steam. Strip type labyrinth seals on the moving blade rings minimize steam leakage past the blades, while concentric labyrinth seals between overhung disks and inner casings cut down leakage short-circuiting the blading. Labyrinth-gland seals at the two shafts control steam flow through these clearances. This unit handles generator drives only.



Noncondensing nonautomatic-extraction

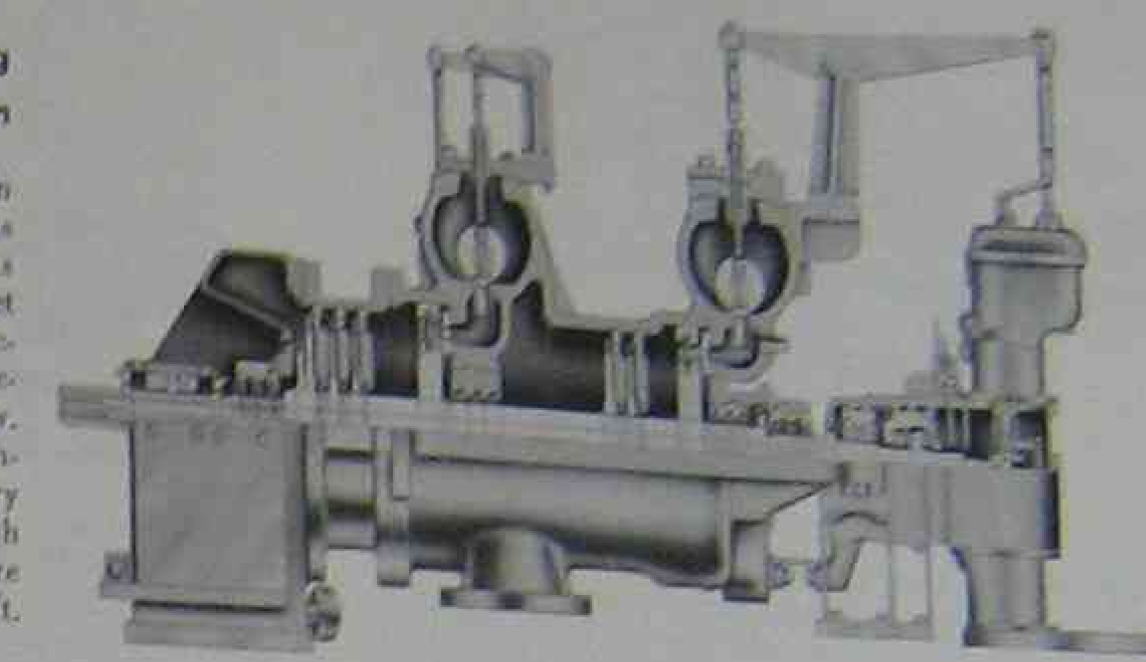
In addition to generator service, industry uses turbines as prime movers to drive pumps, compressors and other types of machinery. Their prime advantage is heat-balance flexibility: throttle steam can be used at boiler or process pressure; exhaust steam can be used for heating and process, or condensed. Extraction and high-backpressure or topping service are possibilities, too. If extraction or exhaust steam can be used, a steam drive has definite economic advantage.

The central-station boiler-feed-pump-drive turbine at left takes steam at high pressure from main line or at lower pressure from reheat line. Sketch shows alternate-nozzle-box design for h-p, below, and l-p, above. Control valves are double-lift poppet type. Unit has seven impulse stages with two nonautomatic extraction points for feedwater heating. It exhausts noncondensing to main cycle.

Noncondensing single-automatic-extraction

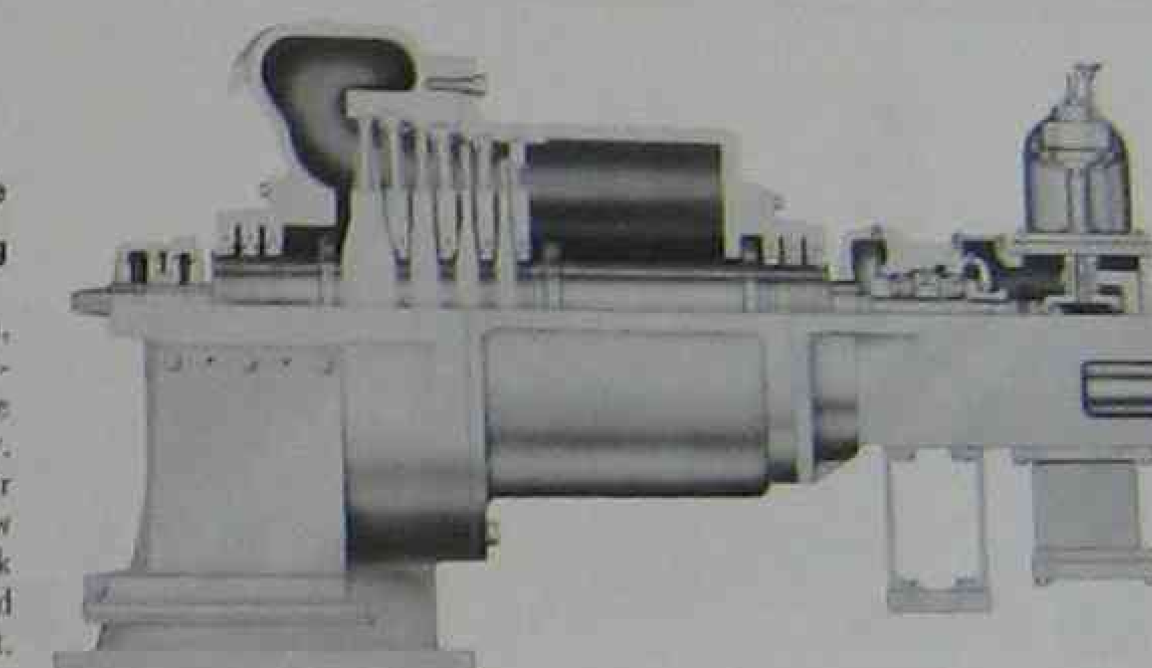
First stages are velocity-compounded in h-p and l-p sections. H-p section has two impulse stages and l-p section has three. Secondary-steam-chest poppet valves control steam flow into the second section, maintain constant extraction-steam pressure for process flow.

Unit is fitted with step-labyrinth interstage and shaft seals, Kingsbury thrust bearing, multiple valves in each steam chest. Disks of this unit are shrunk onto the main turbine shaft.



Low-pressure condensing

This turbine takes low-pressure steam, at 100 psig or less, available from process exhaust lines. Its four stages use twisted blading for high stage efficiency. For high shaft speed, unit has rotor milled out of a solid forging; for low speeds, steel disks would be shrunk on a solid shaft. Journal bearings and Kingsbury thrust bearing position shaft.

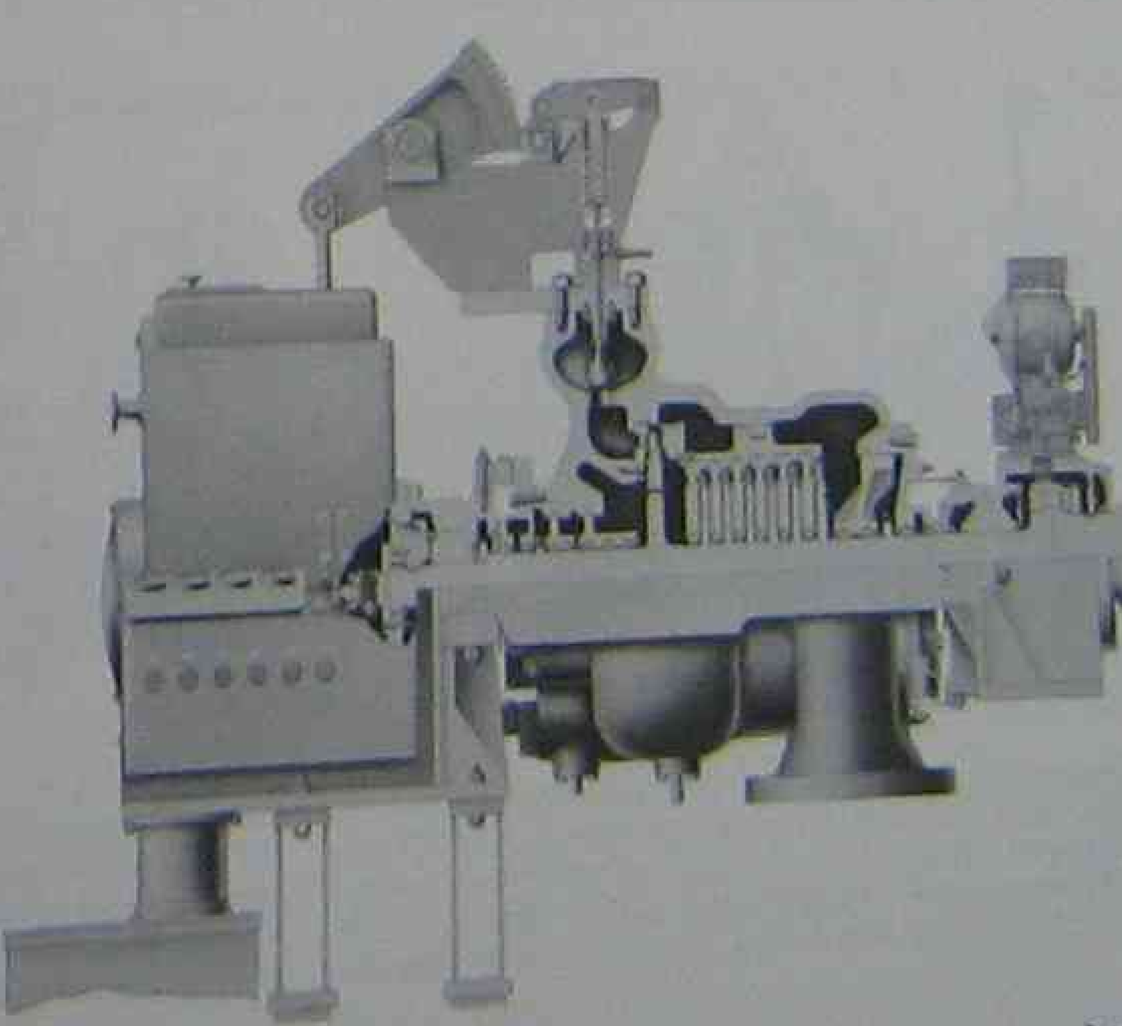


Noncondensing or backpressure

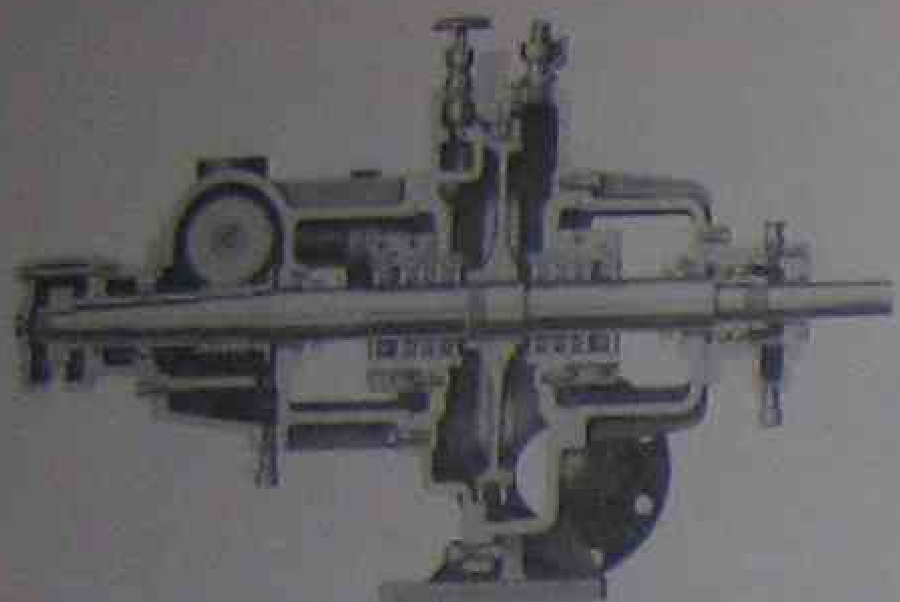
Seven impulse stages are controlled by a multivalved steam chest. Since backpressure is high, this unit needs only a small exhaust hood. Turbine is used where lower-pressure process steam and electric generation are needed simultaneously by an industrial plant.

The exhaust-pressure governor holds whatever exhaust-steam pressure the process requires. But as a result, steam flow through the turbine and its shaft output vary. The turbine must run in parallel with other generators that can meet variations in electric load; it cannot follow them when under exhaust-pressure governing.

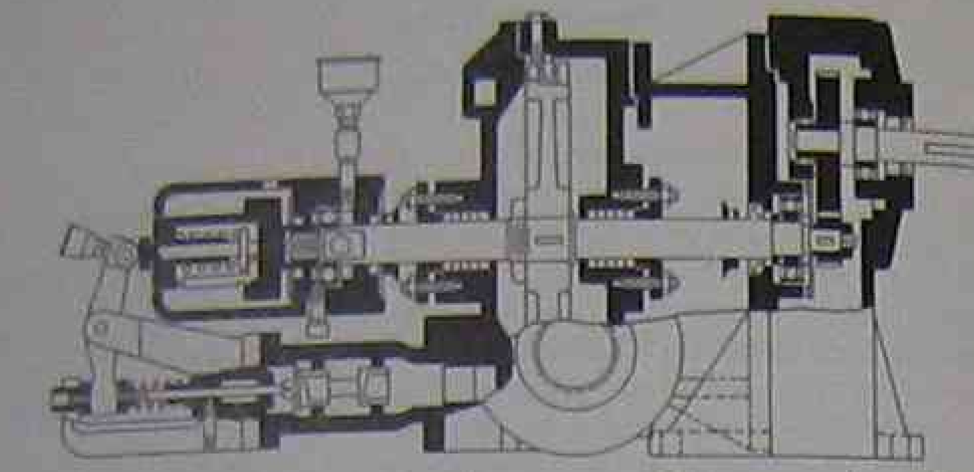
This turbine may also "top" lower-pressure turbine-generators, taking steam from a new h-p boiler and exhausting at lower pressure into the existing steam main of existing turbines.



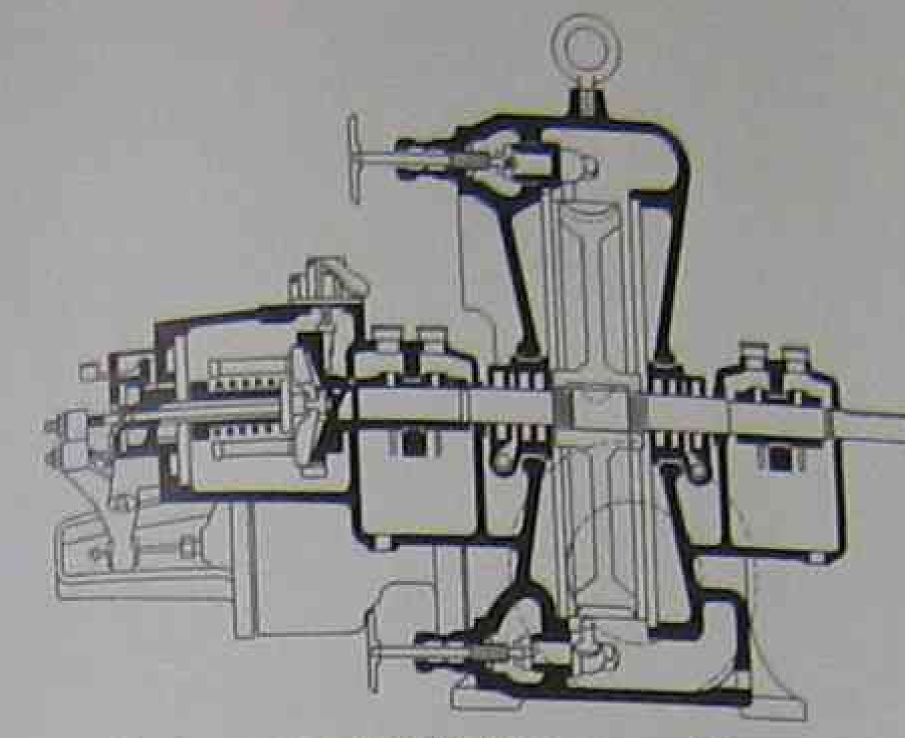
Many mechanical-drive units are single stage, velocity compounded



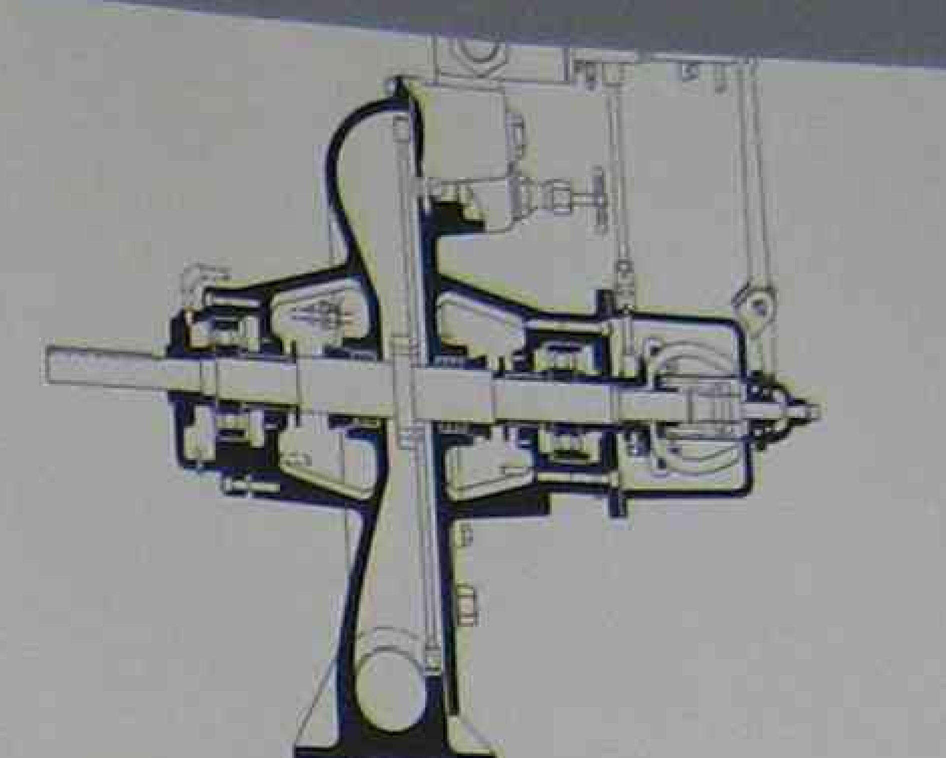
Single-wheel velocity-compounded turbine may be designed for a range of capacities—from less than 1 hp up to 250 hp and more. Design uses one of two standardized blading types for all conditions: 450-psig 750-F and down at inlet; 0- to 80-psig exhaust. Unit may exhaust to process.



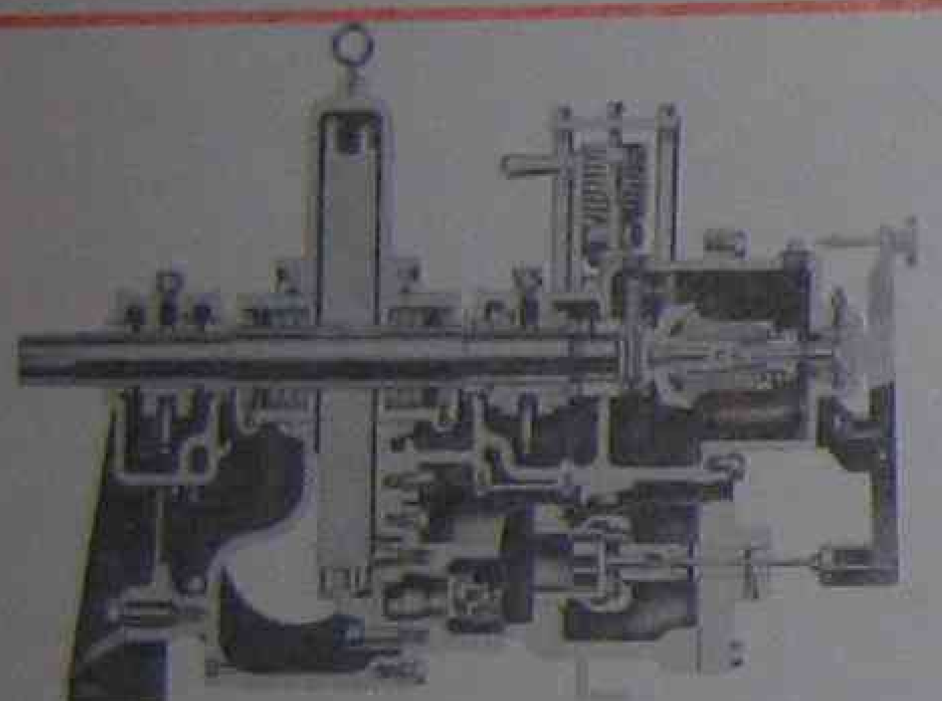
Mechanical-drive design for up to 100-hp capacity takes steam at 600 psig, 750 F, exhausts at 75 psig. Shaft speed of 3500 rpm drives load at up to 300 rpm through integral reduction gear. Flyball governor on shaft end controls balanced valves through linkage. Carbon rings seal shaft.



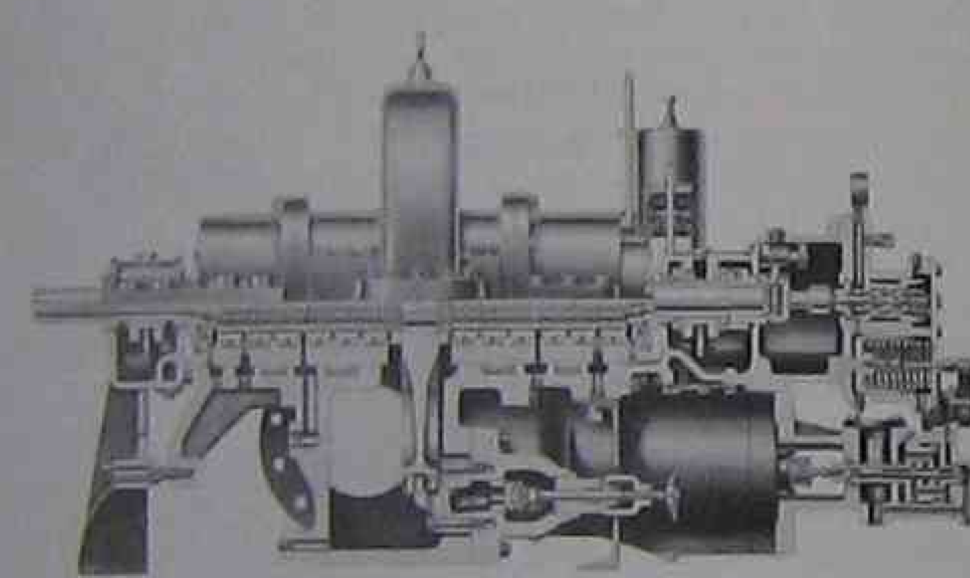
Single-stage re-entry turbine uses solid rotor with circular blades milled out of periphery. Wheel is pressed on and keyed to shaft. This unit develops up to 650 hp with 650-psig saturated steam exhausting at atmospheric. Hand-valve nozzle control helps realize better part-load efficiency.



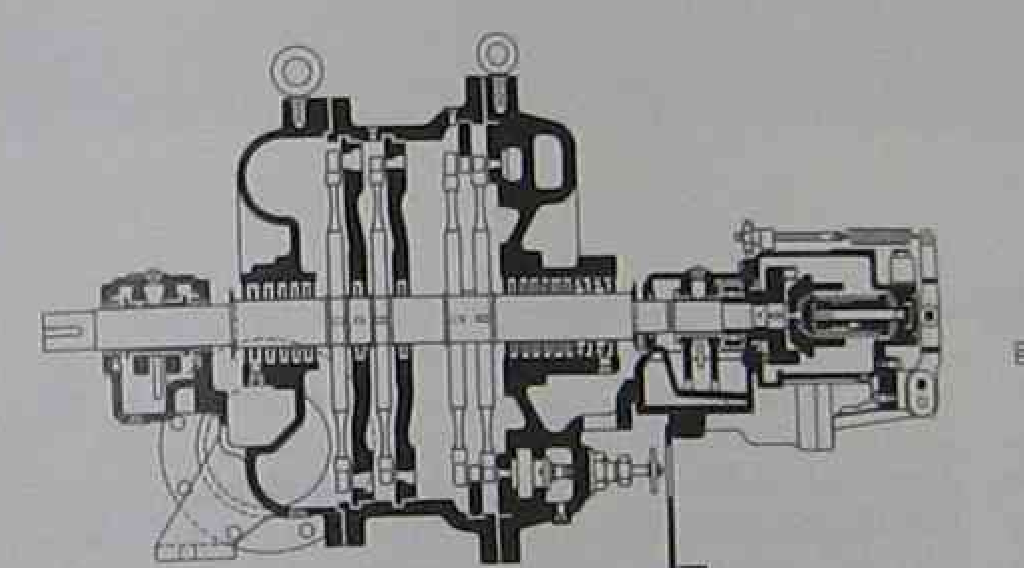
Single-stage impulse turbine has 100-hp maximum rating with 275-psig 525-F steam and 50-psig backpressure, runs at shaft speeds up to 4000 rpm. One of its two groups of nozzles is fitted with hand-controlled valve for part-load efficiency. Stuffing boxes use semimetallic packing at glands.



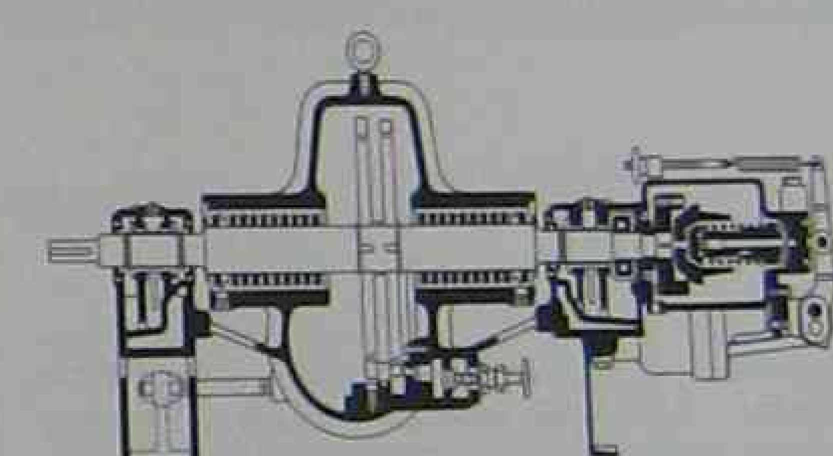
Velocity-compounded turbine has close-fitting casing at top, reduces windage loss on wheel and blading. Converging-diverging nozzles are used because steam's pressure ratio runs less than 53%. Overspeed-emergency-governor trip is in shaft at left of constant-speed-governor flyweight assembly.



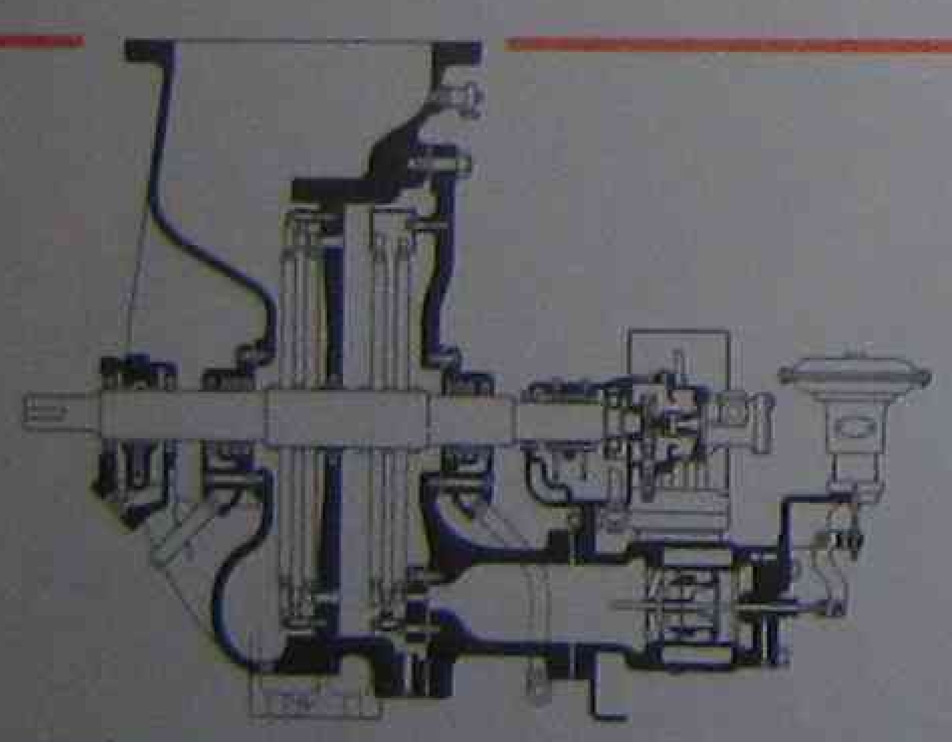
Velocity-compounded turbine with single wheel and high backpressure features extensive spring-loaded shaft sealing. Steam sealing and gland leakoffs must be used to control steam leakage along shaft. A shaft-driven oil pump provides governing. Hand valves up part-load efficiency.



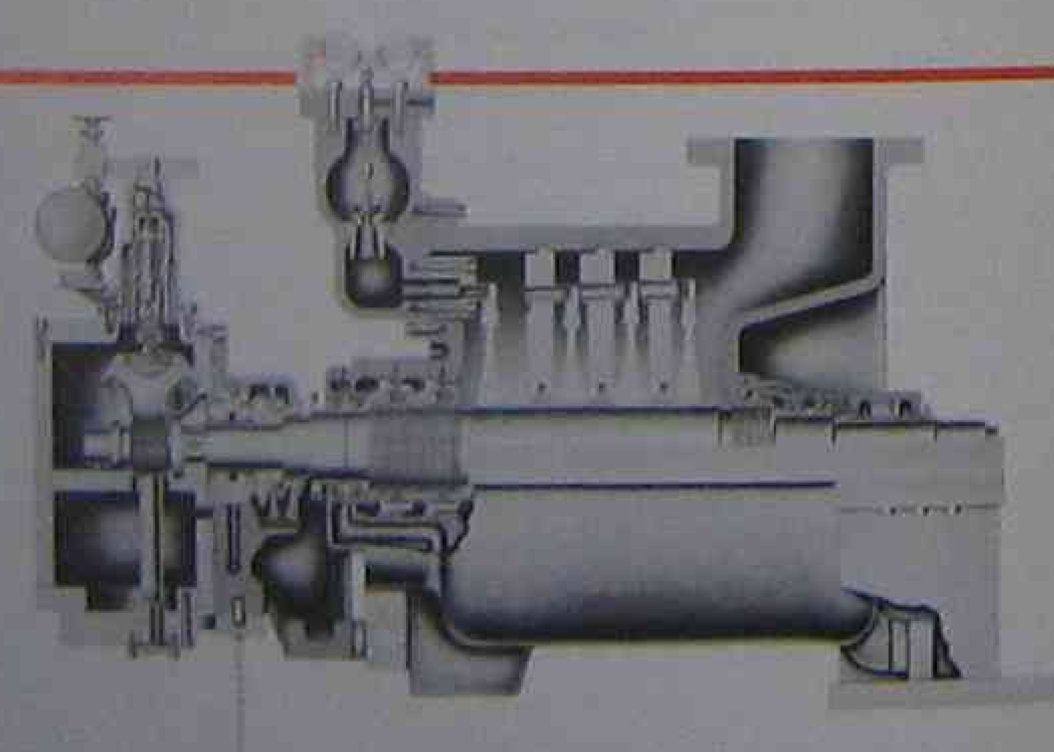
Multistage turbine's velocity-compounded first stage uses two separate wheels. This type may be condensing or non-condensing, runs at shaft speeds up to 10,000 rpm. It has carbon-ring seals, ring-oiled journal bearings and a double-thrust ball bearing to control position of the shaft.



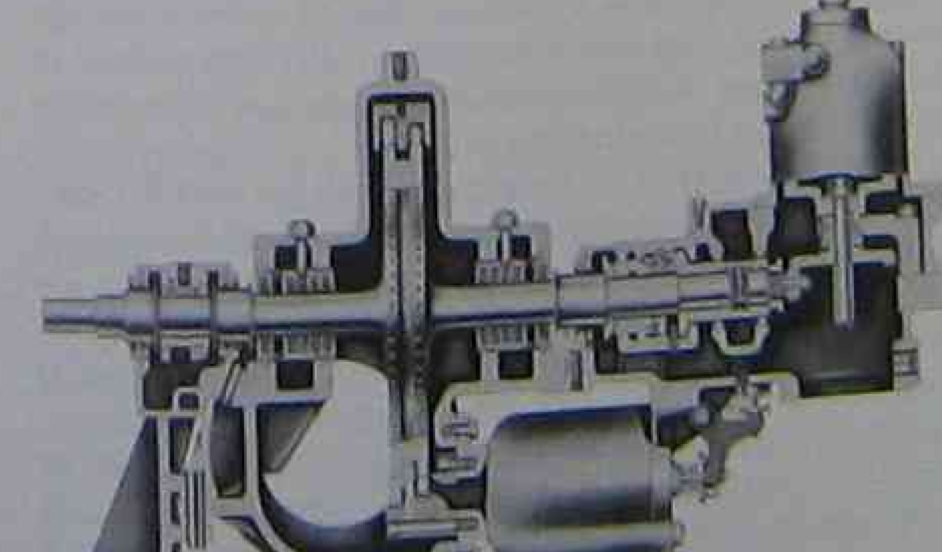
Velocity-compounded 2-wheel turbine is built for 165-psig backpressure. Each shaft gland has ten carbon-ring seals. Gland leakoffs dispose of a little condensed-steam leakage. Hand speed changer attached to governor linkage varies speed range by 20%; maximum: 4500 rpm.



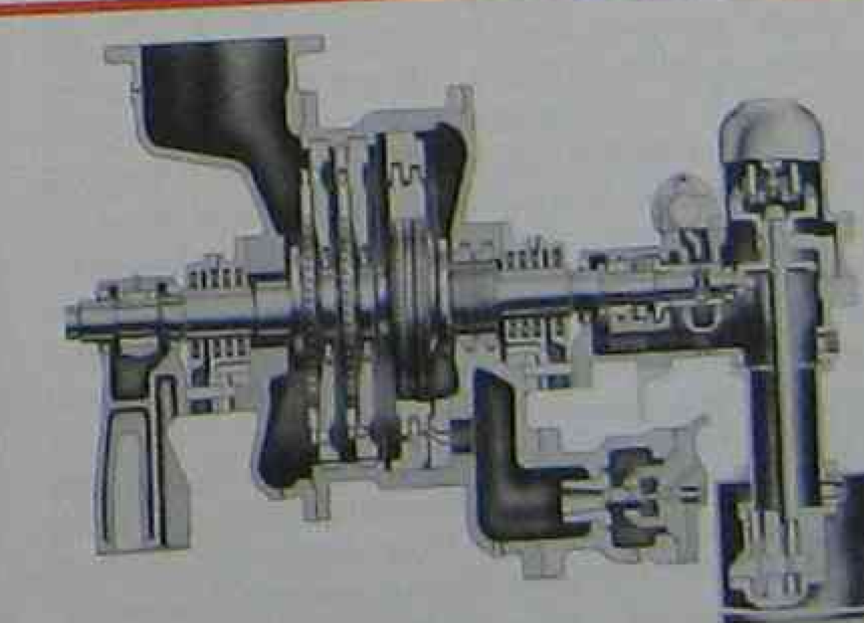
Variable-speed turbine handles compressor drive, with speeds ranging from 3500 to 6000 rpm. Turbine uses 250-psig 500-F steam exhausting at 2- to 6-in.-Hg abs. There are two velocity-compounded stages; upper half of first-stage wheel is shielded, which helps avoid excessive windage losses.



Noncondensing high-backpressure turbine is built to run up to 10,000 rpm and above. For high speeds, rotor would be a machined solid forging. Unit has four impulse stages. Flyball governor, left, positions pilot valve of hydraulic-governor system controlling bar-lifted valves in steam chest.

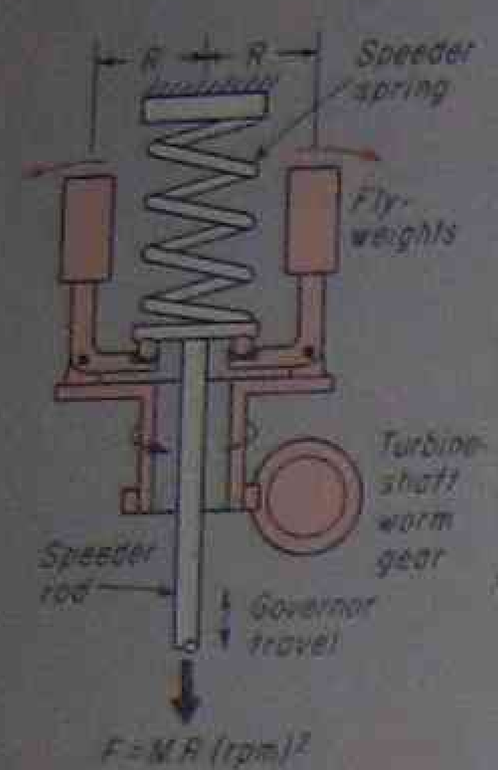


High-speed turbine is built for inlet steam of 1450 psig and 1000 F to exhaust at up to 75 psig. Upper half of velocity-compounded wheel is closely shielded against windage losses. The 12,500-rpm shaft is designed to run well above first-critical speed but well below the second-critical speed.

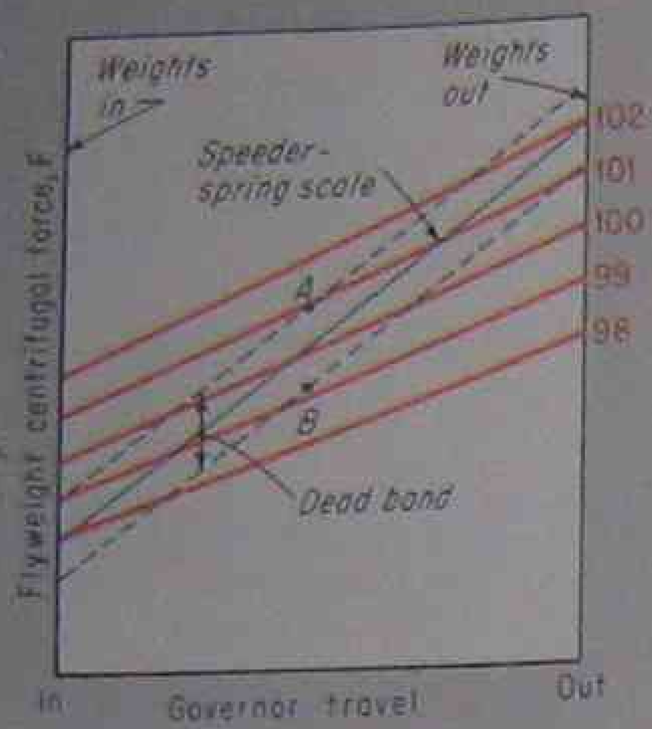


Condensing turbine has velocity-compounded first stage followed by two impulse stages. Combined labyrinth and carbon seals are used at h-p end; l-p end has carbon rings only. Shaft drives gear type oil pump and flyball governor at right end; thrust bearing is included to position the shaft.

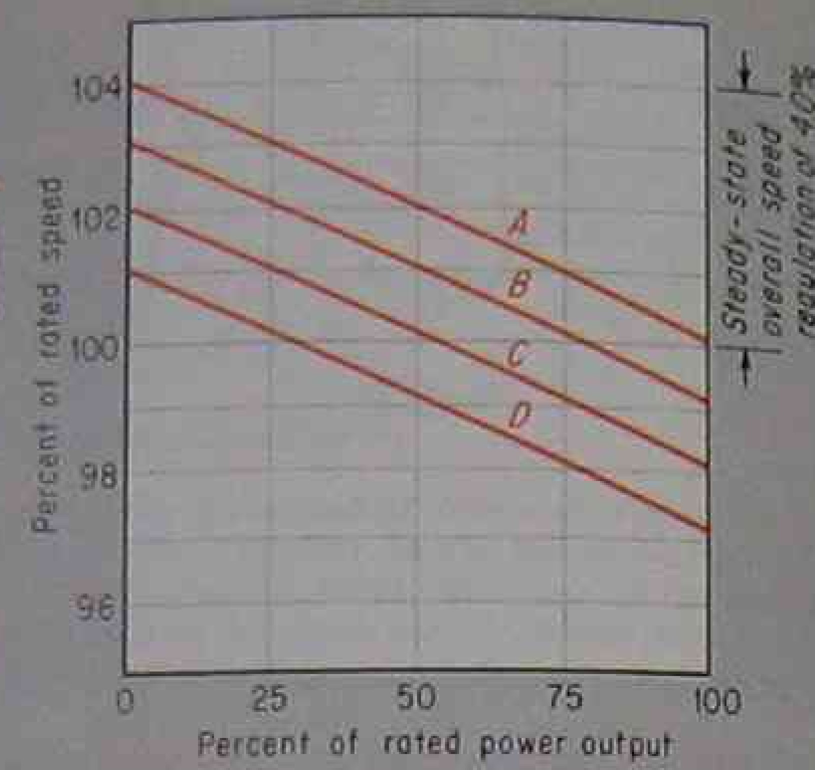
Flyweight governor opposes centrifugal force of pivoting weight with mechanical force of spring



1 Flyweight governor balances the force of spring and weights, moves speeder rod

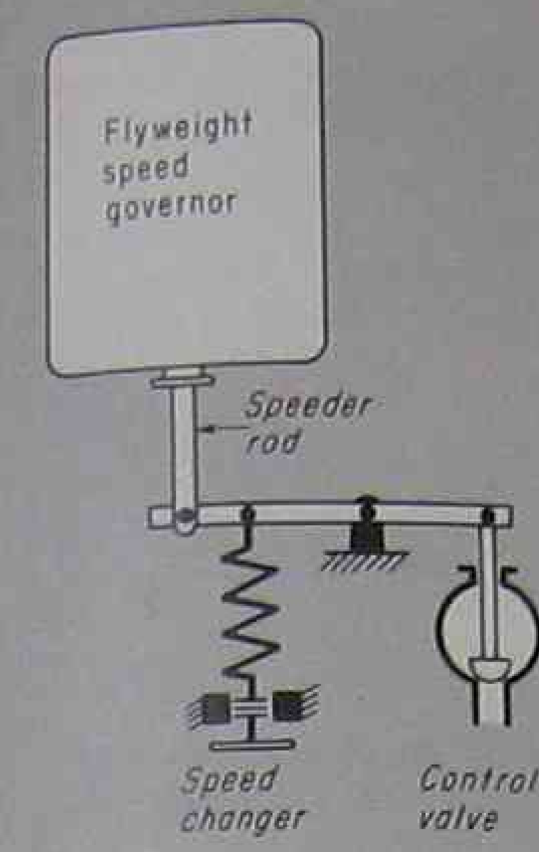


2 Speeder-spring force must grow faster than the weight force as speed rises

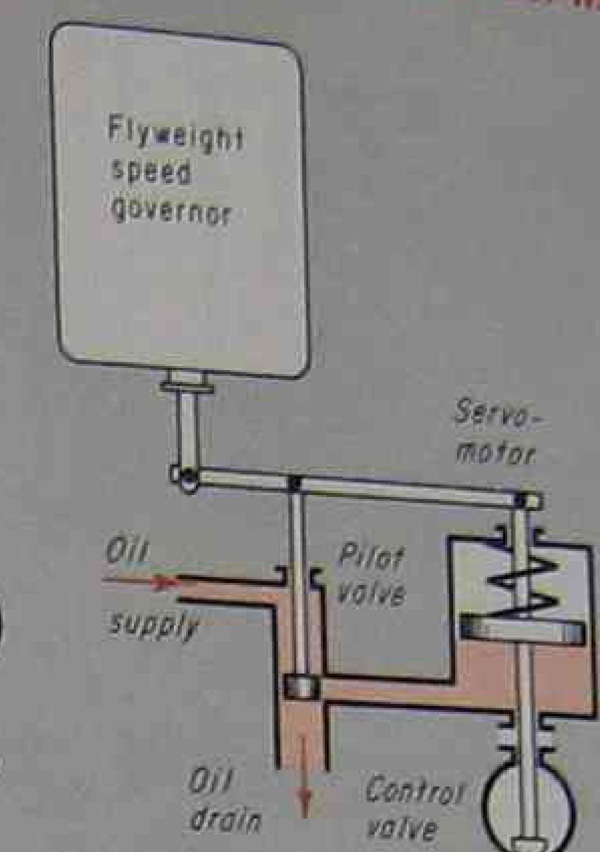


3 Speed-power curves show how a speed changer shifts droop characteristic

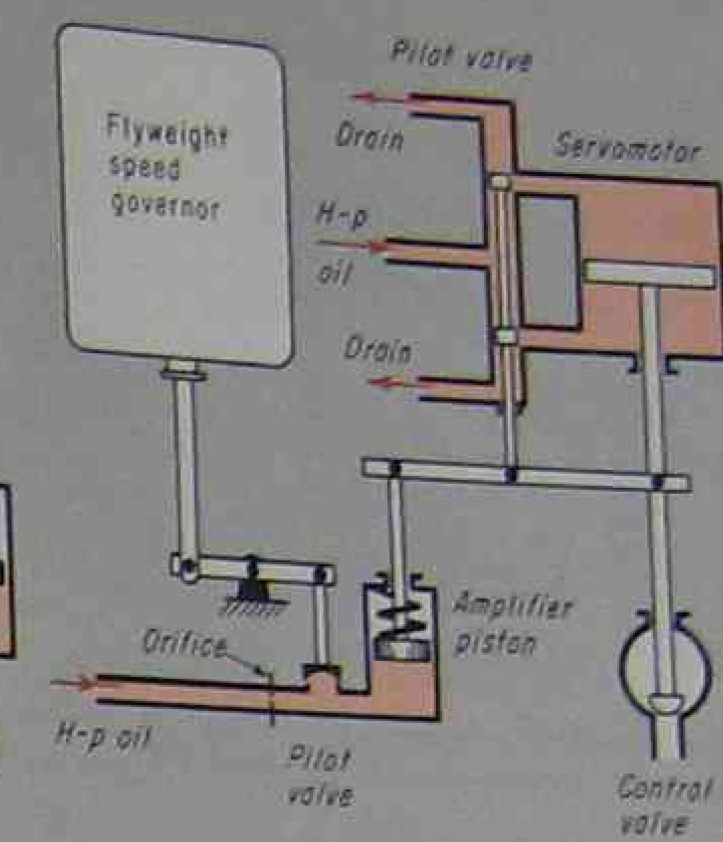
Basic turbine governor adds refinements to boost its force



4 Speed-changer spring alters force acting on the steam-control-valve lever



5 Pilot-valve-controlled servomotor lets weak governor force move steam valve



6 Double-relaying system helps governor handle the very large steam valves

GOVERNING controls shaft speed

A turbine's governing system functions to control steam flow through the unit. It usually does this to keep some factor constant. Often we want to hold steady at all loads: (1) turbine shaft speed (2) exhaust-steam pressure (3) automatic-extraction pressure (4) inlet-steam pressure (5) driven-equipment discharge pressure (6) driven-equipment differential pressure or (7) any combination of these factors. Since shaft speed is most frequently controlled, let's study it to see how turbines are governed.

Turbine speed will stay constant even without governing, just as long as shaft load and throttle-valve opening are unchanged and steam conditions remain steady. But when shaft load decreases with constant throttle-valve opening, the turbine speeds up because it's getting too much steam. To bring shaft speed to normal the valve must throttle steam flow to match the new load. When shaft load rises and valve opening is constant, the turbine slows down; to return shaft speed to normal, the valve must open wider, admit more steam. A shaft-speed governor can make these changes automatically.

A flyweight assembly measures shaft speed. Fig. 1 shows one form of it: two flyweights mount on a plate which turns about a vertical axis driven by the turbine shaft through a worm and gear. Weights are pivoted so as centrifugal force tilts them outward, they compress the stationary *speeder spring* and lift up the *speeder rod*. Speeder-rod movement can be hooked up to open and close the turbine throttle valve, Fig. 4.

How does a governor behave? As a first step to understanding, we can remove the speeder spring in Fig. 1 and find out how flyweights act by themselves. First let's hold the weights in their innermost position—weights in—with the speeder rod, and turn the assembly at its rated 100% speed. We'll exert a downward resisting force F on rod while weights turn through circle with radius R.

If weights move through a circle with larger radius, we find that force on speeder rod is larger, though speed is still 100%. Force $F = MR\ (rpm)^2$ where M is the flyweights' mass, R is radius of their motion and *rpm* the number of turns per minute.

Plotting force F against governor travel or speeder-rod position for various speeds from 98 to 102%, we get the family of straight-line curves in Fig. 2, which ideally converge at a common point off to the lower left of the chart. Slope of each speed curve has a scale reading in lb force per inch of governor travel.

A speeder spring with a scale to match the 100% curve develops the same resisting force F for a given lb per inch of compression or governor travel. Spring scales set at the middle of governor travel at 100% speed would be fine as long as all factors stayed constant. But as soon as load on the unit decreased, the shaft would speed up. Then weights would develop more force than the spring over the range of governor travel. Weights would fly to their weights-out limit, shutting the steam valve.

As steam flow stopped, unit would slow down. Slightly below 100% speed the springs' mechanical force would overbalance the weights' centrifugal force and slam them to weights-in, completely opening the steam valve. So if weight and spring scales are equal they're continually fighting, producing wild hunting from fully shut to wide-open throttle valve, without hope of reaching a balance. Obviously, this is no way to govern a turbine.

Speed droop. To get out of this dilemma, spring force must grow faster than weight force as speed rises. Spring scale should be steeper (larger) than the flyweight scale, solid black line in Fig. 2.

Suppose the turbine is running at 100% speed, where the speeder-spring curve crosses the red 100%-speed curve. When shaft load drops, shaft speed rises. Then increased force of the flyweights momentarily overbalances the spring force, raising the speeder rod to decrease throttle-valve opening. Decreased steam flow limits shaft speed rise, so spring force again balances the higher fly-weight force at the new higher speed. Governor can't travel to its weights-out limit because the spring exerts greater force than the weights beyond the new point of balance.

If the turbine is running at 100% speed and load rises, the shaft slows down. Then flyweights' lesser force lets the overbalancing spring force lower the speeder rod to open the throttle valve wider. Increased steam limits the speed drop so diminishing spring force matches weight force at the lower speed. Governor won't travel to its weights-in limit because the weights exert more force than the spring below the new balance point.

At each point in governor travel the turbine runs at a definite speed, slower at full load and faster at no load. For the spring scale in Fig. 2 speed varies by 4% from full to no load. This is the governor's *regulation* or *speed droop*. Fig. 4 shows speeder-rod and control-valve hookup.

Frictional effects, ignored in the ideal model, play a big part in design of a working governor. Suppose the unit is running at 98% speed in Fig. 2 at the weights-in position, and then speeds up. Weights' force rises, but because of friction in the linkage and glands the governor doesn't move immediately. By the time the turbine reaches 99% speed, force is large enough to overcome friction and the governor travels along the upper dotted curve.

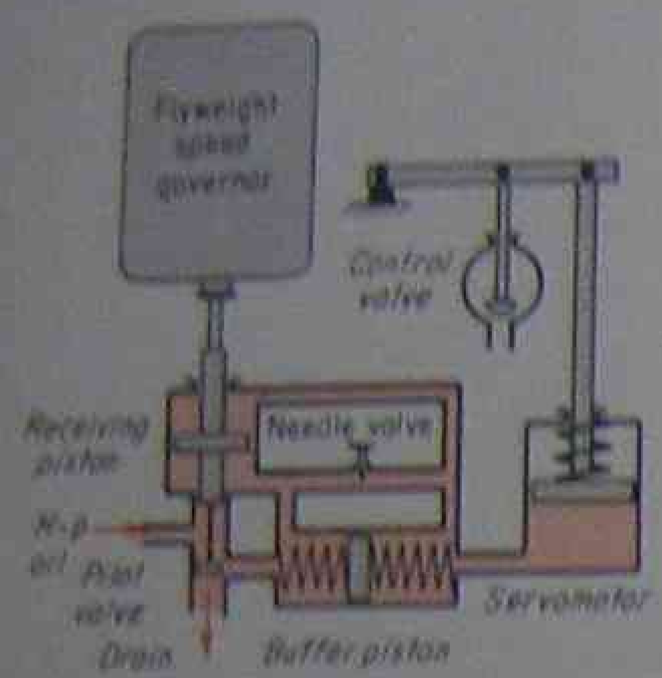
Now let's bring speed up to 101% at A by reducing turbine load. Next, we'll start to load it again; the turbine slows down to 99.4% speed at B before the governor responds by moving toward its weights-in limit along the lower dotted line. Vertical distance between upper and lower dotted lines measures the governor's *dead band*, which in turn defines *sensitivity*: speed change needed to produce a corrective movement in governor travel.

Speed-output curve, Fig. 3, shows how a governor varies shaft speed with load. A given spring scale A produces 100% speed at full load. As turbine is unloaded the shaft speed rises until it reaches 104% speed at no load. But usually we want to keep shaft speed constant at all loads. So a *speed changer*, Fig. 4, is added. It puts additional spring force on the lever controlling the steam valve, is regulated by hand or remote-controlled motor.

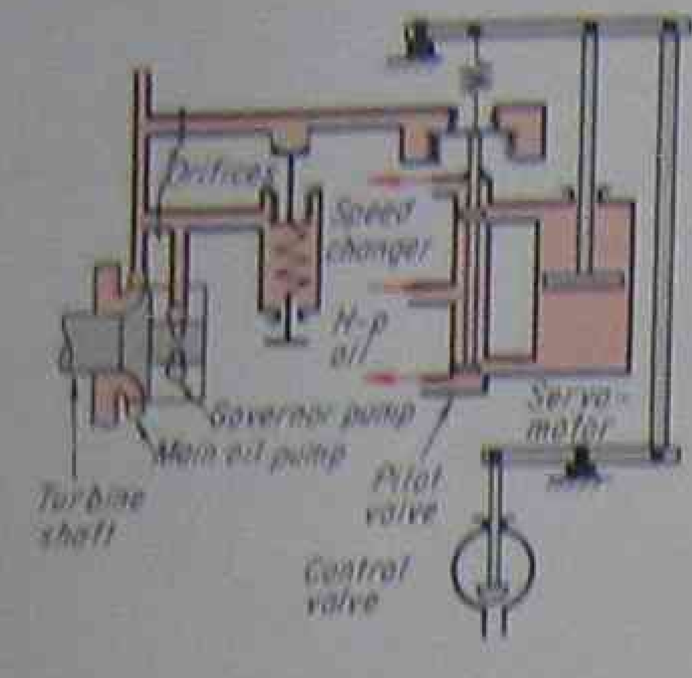
Suppose we have a 100% load at 100% speed and bring load down to 75% in Fig. 3. Speed rises to 101%. We would reduce this to 100% by loosening the speed-changer spring, allowing the steam valve to close a little more. Fig. 3 shows characteristics for speeder spring adjusted to hold 100% speed at 25, 50, 75 and 100% loads.

Relayed governor, Fig. 5, is used where speeder rod doesn't develop enough force to move the steam valve. A *floating lever* links speeder rod to a *pilot valve* and the *servomotor* that actually moves the control valve. Initial speeder-rod movement pivots the lever on the servomotor stem and moves the pilot-valve stem. This admits h-p oil to the spring-loaded servomotor cylinder or drains oil

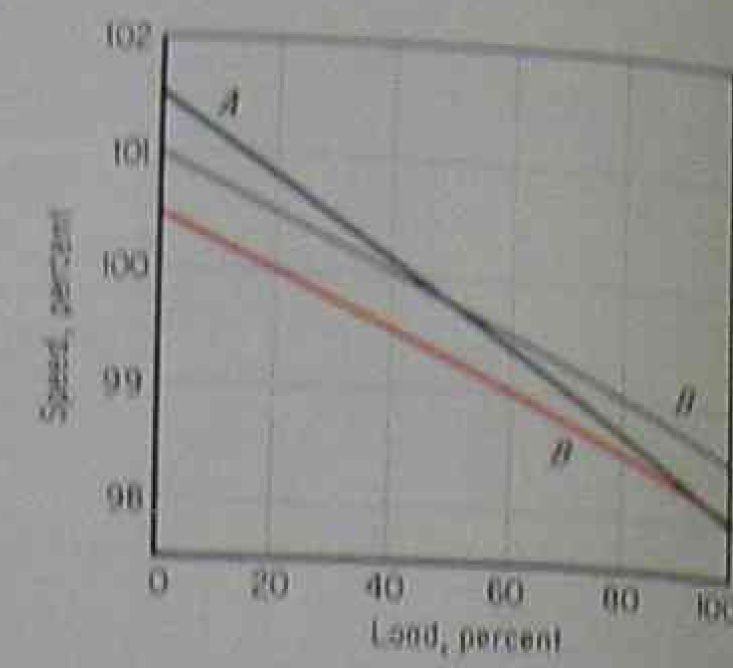
Generator load division depends on governor droop



7 Isochronous governor uses transient speed changes to move to new setting



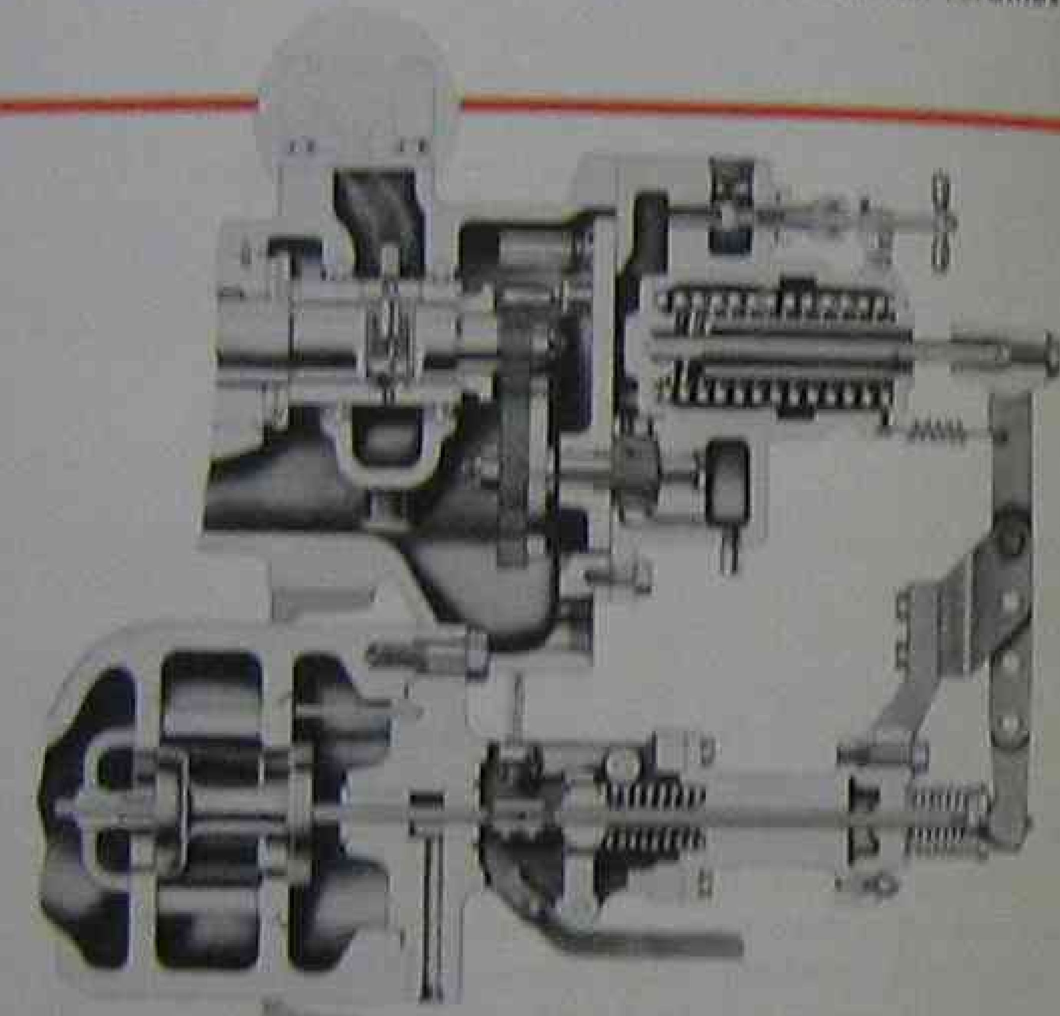
8 Hydraulic governor has an oil pump as its speed-sensitive measuring element



9 Governor-droop characteristics determine load division between turbines

Designs vary with size of unit, shaft speed, needed accuracy

13 Direct-acting mechanical governor has weights mounted on leaf springs, tied to steam-valve stem. It holds steady speed



14 Hydraulic-orifice governor's shaft-driven gear pump supplies oil for governing and lubrication; its speed ratio is 3:1

from it, moving the control valve to its new position. Motion also moves the pilot valve to neutral so it won't overshoot the new position of the control valve.

Turbines with large steam valves need a relayed governor plus amplification, Fig. 6. A cup type pilot valve bleeds h-p oil from the line to control the first-step amplifier piston. This piston controls the pilot valve of the main servomotor. Servomotor is not spring-loaded; to move it, h-p oil is admitted on one side, drained from the other.

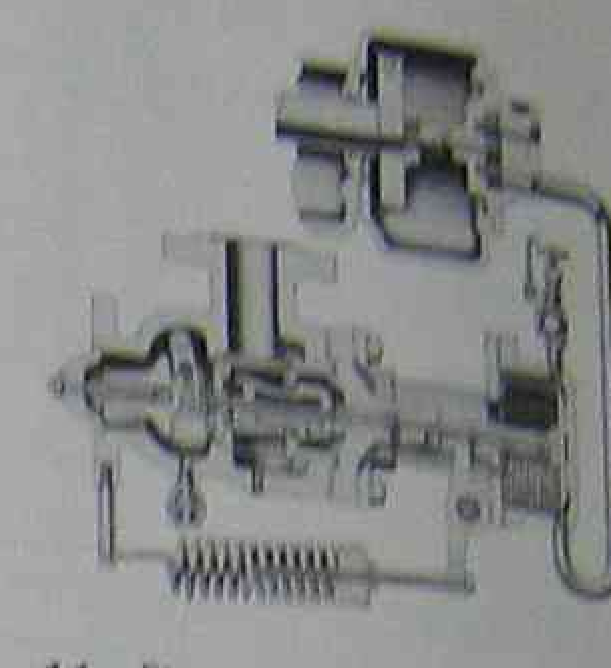
Isochronous governor, Fig. 7, holds the same *steady-state* speed on a turbine at all loads without a speed changer. When load first varies it produces a *transient change* in shaft speed, but that's gradually eliminated as load stays constant at its new level. On a *change* in load the speeder rod moves the receiving piston, and pilot valve in turn unbalances the spring-balanced buffer piston while moving the servomotor to its new position. After the initial correction, buffer piston moves back into balance. It equalizes pressure across the receiving piston, repositioning governor flyweights at rated 100% speed with the new shaft load.

Hydraulic governor, Fig. 8, has a special centrifugal pump instead of flyweights as its speed-sensing element. Oil pressure varies as the square of the shaft speed. Governor pump controls throttled pressure developed by the main oil pump; this acts on the bellows in the speed changer. Bellows controls the cup type valve which is part of the pilot valve, regulating pressure in another bellows which positions the servomotor pilot valve.

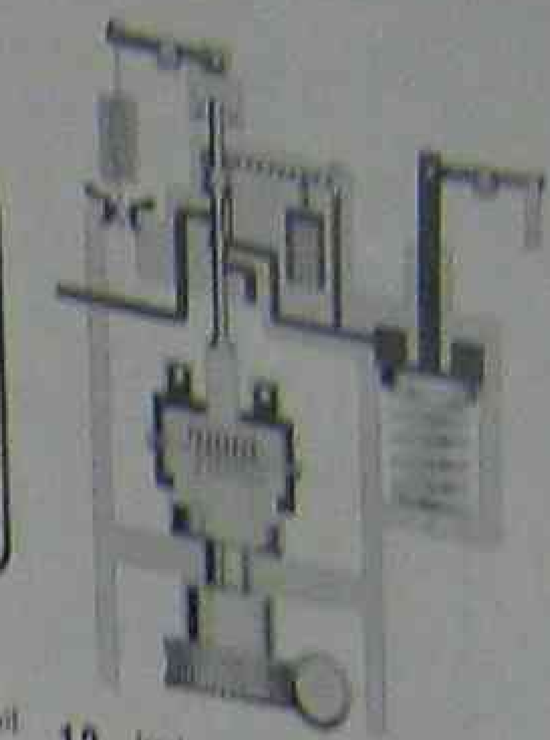
Generator load division, Fig. 9, depends directly on the governor droops of the turbines that drive the generators. Paralleled synchronous generators run at the same speed, just as if they were connected mechanically.

When load rises, speed falls until total output of all

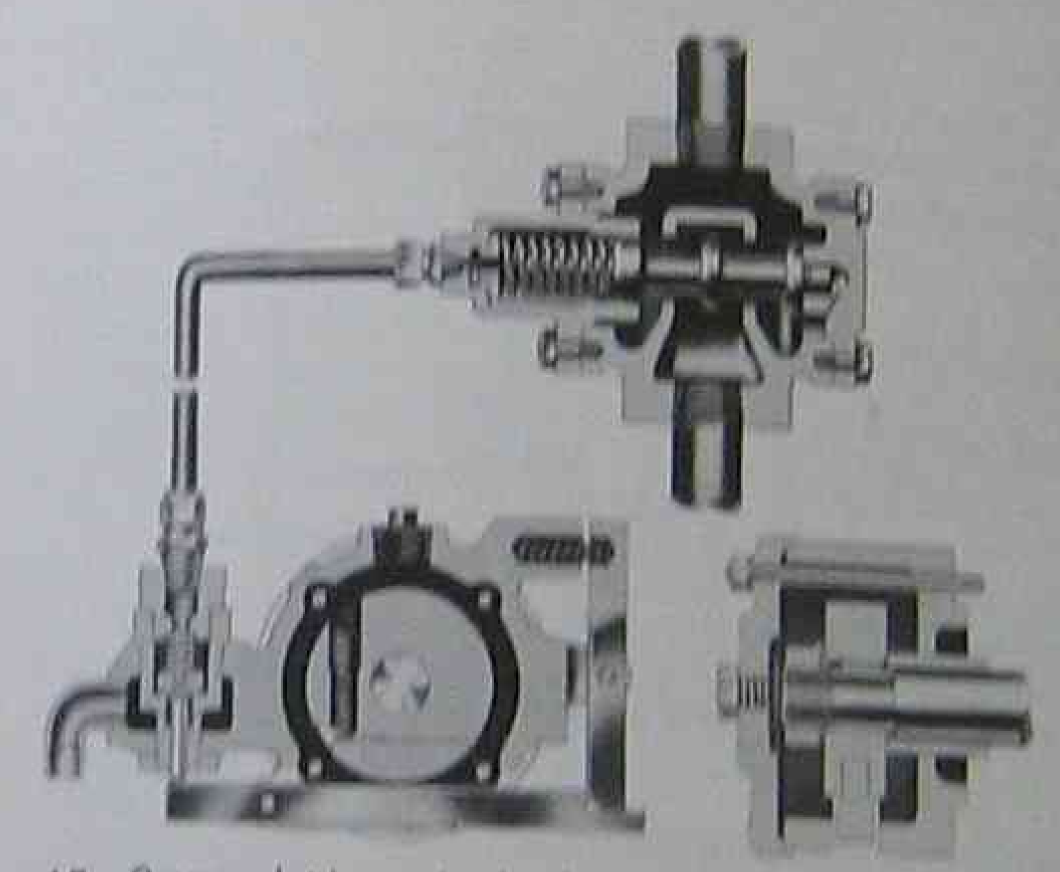
10 Direct-acting flyweight governor for small-size turbine: weight pivots roll



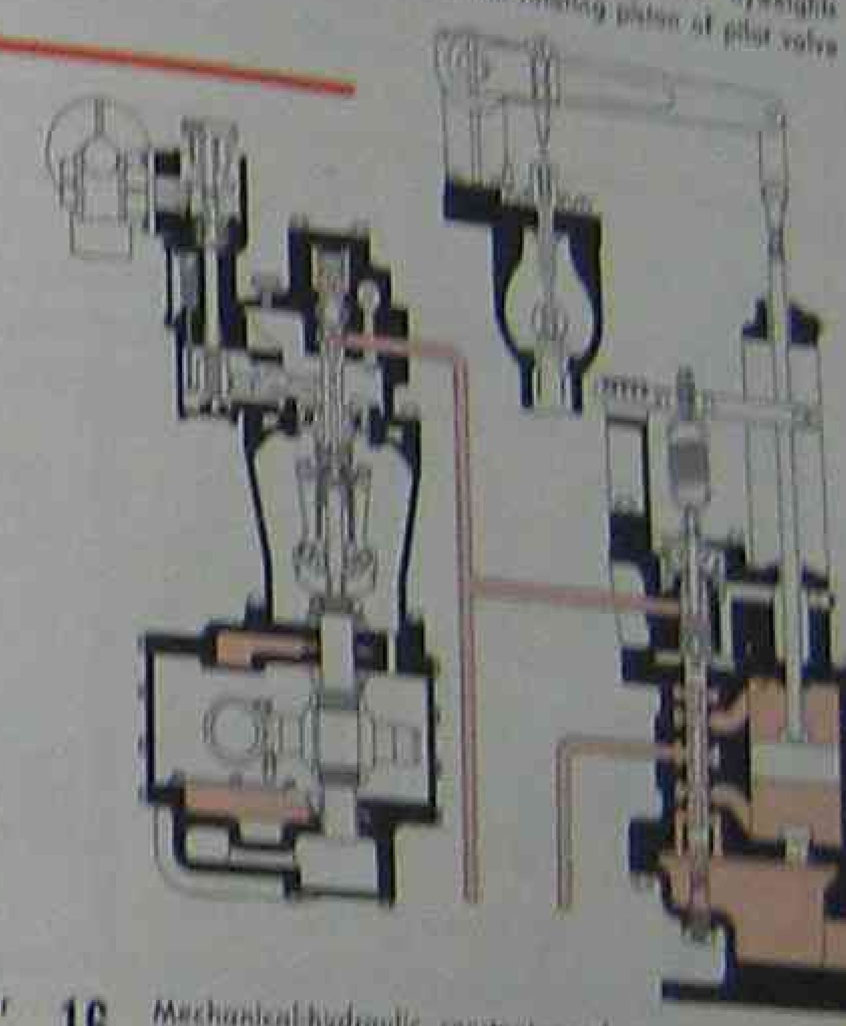
11 Direct-acting hydraulic governor: oil pump moves valve through bellows



12 Isochronous governor's flyweights control rotating piston of pilot valve



15 Overspeed trip: spring-loaded weight on shaft-trip lower lever opens pilot valve, left, which then closes main valve



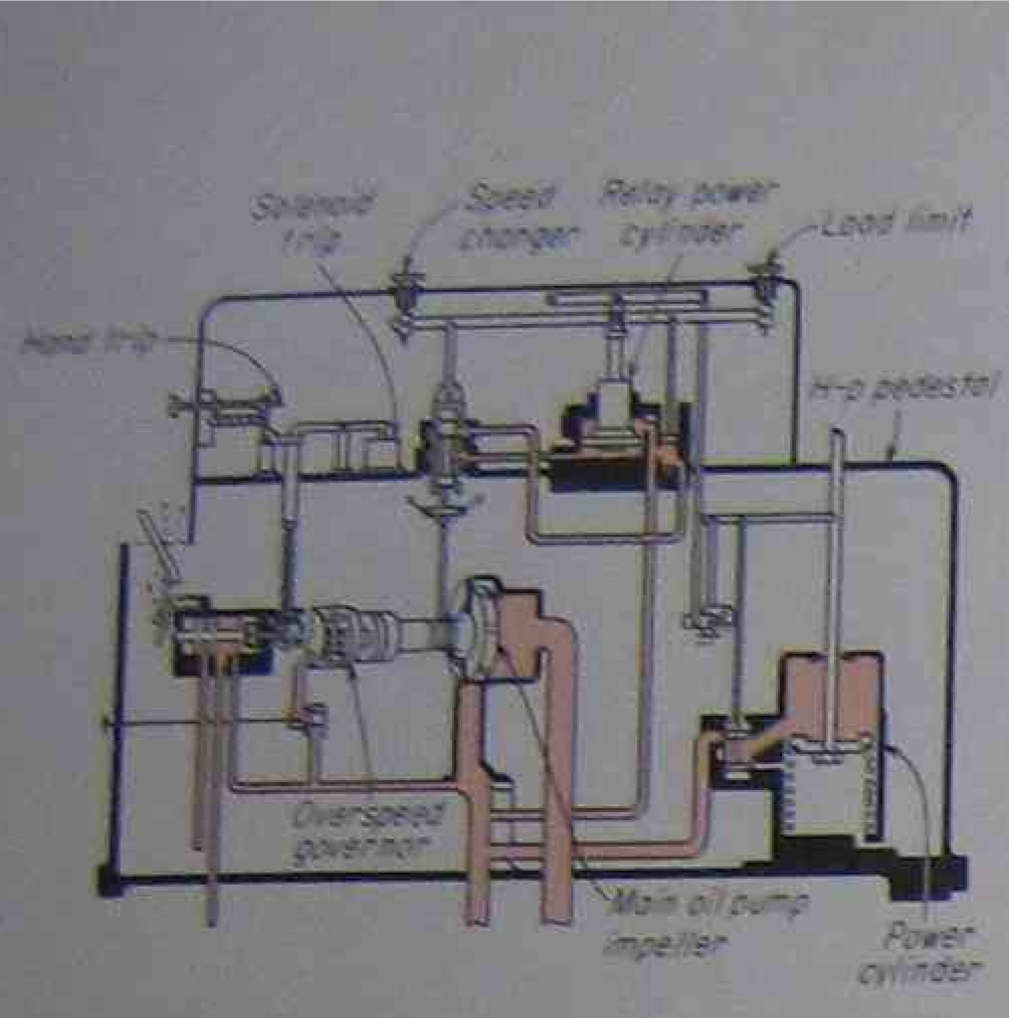
16 Mechanical-hydraulic constant-speed governor head positions rotating pilot. Pilot controls double-relay oil system

units meets the new demand. In Fig. 9, two machines *A* and *B* have the droops shown by the black curves. At 100% speed *A* carries 43% load and *B* 40% load. If load rises until speed falls to 99%, *A* carries 72% load and *B* 80% load. To restore speed to 100% with this same division, speed changers on both units are adjusted to lift both curves the same amount.

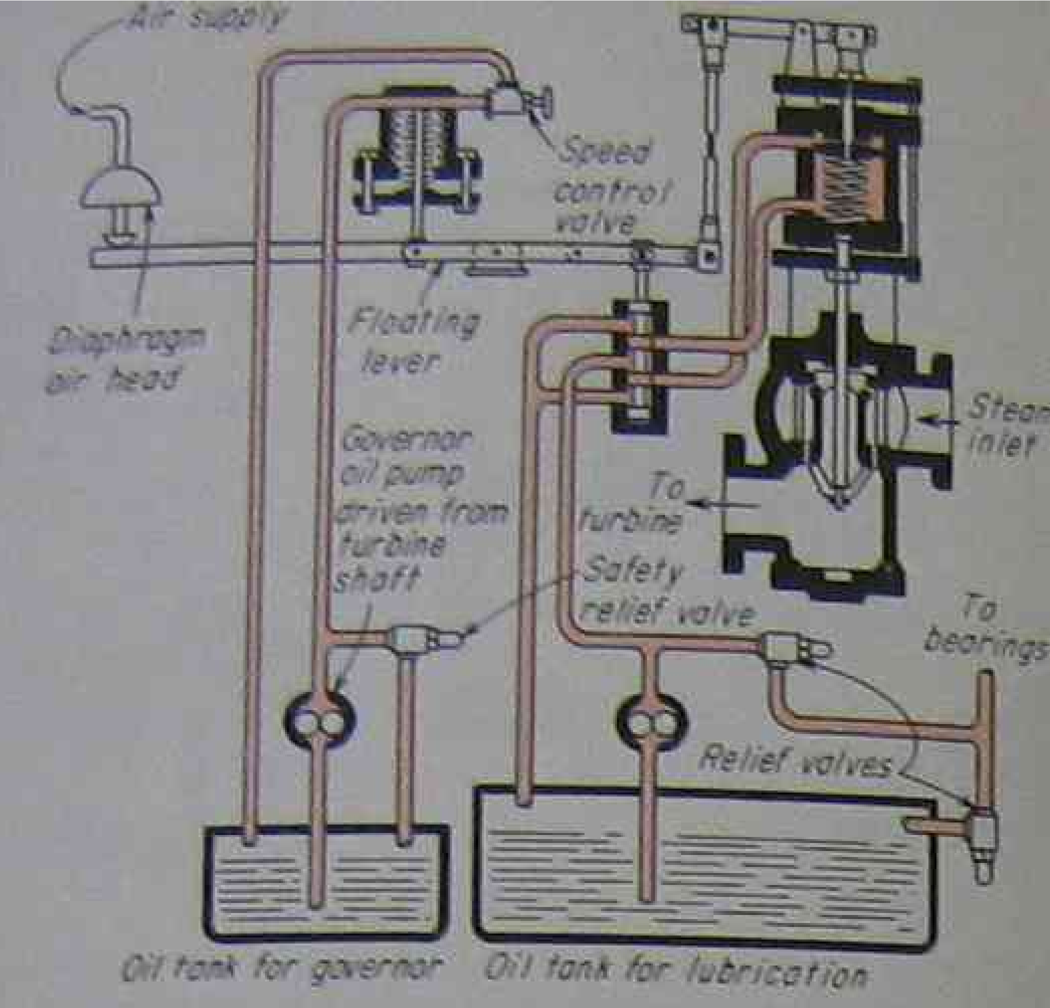
Returning to 100% speed in Fig. 9, suppose we want to drop load from the total 40 + 43% to 63%. If *B* is to take all the reduction, its droop must be lowered to the red curve. In general, unit with least droop will always take largest share of the total load.

Governor-system designs vary with size of turbine, speed of shaft, degree of accuracy—and with the manufacturer. Direct-acting mechanical governor in Fig. 10 links directly to the steam-control valve. To minimize needed force, steam valve is a balanced double-seated design. Both governor weights pivot on rolling surfaces with circular sections. To the left of the governor head a spring-loaded unbalanced pin acts as the overspeed trip. When shaft speed exceeds the safe limit, centrifugal force snaps the pin to an outer position so it trips a lever that closes the turbine stop valve. A speed-changer knob on upper end of governor lever tightens spring (not shown) to shift the unit's droop curve.

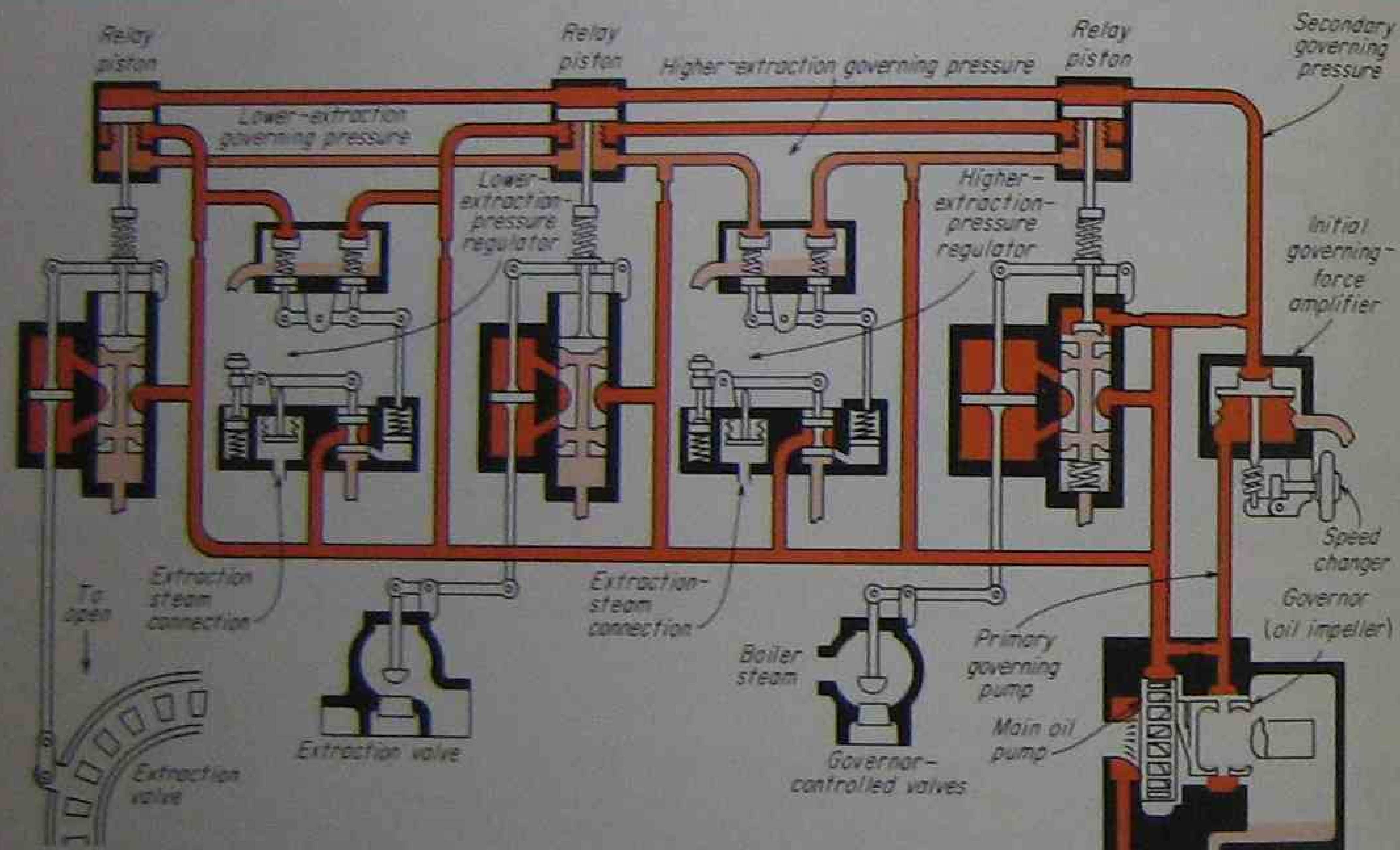
Hydraulic governor in Fig. 11 has a 3:1 speed-adjustment ratio. Its gear type positive-displacement oil pump develops a pressure that varies with shaft speed. Pressure acts on the bellows, and this directly positions the balanced double-ported steam-control valve. Needle valve above the bellows bypasses oil back to tank to vary the bellows pressure and hold any desired speed. Turbine stop valve is at left of main governing valve. Fig. 14 shows a hydraulic governing system for a m-d turbine.



17 Turbine front pedestal houses the mechanical governor, oil pump, relay valves and servomotors for regulating unit



18 Variable-speed oil-relay governor's air motor also acts on floating lever to control the main steam flow into turbine



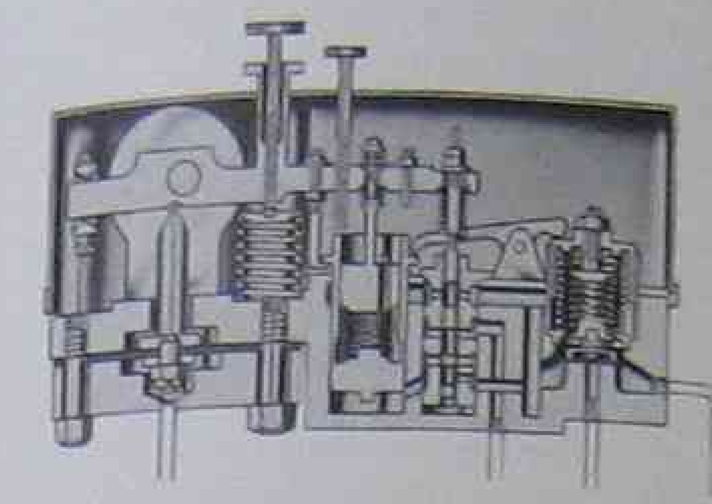
21 Double-automatic-extraction turbine uses combination of three signals on three relay pistons to hold constant pressure at each of its extraction openings and maintain a constant shaft speed

Isochronous governor, shown schematically in Fig. 12, mounts flyweights on a rotating head at upper ends in torsion bushings. Weights work through strap springs to rotate the pilot valve above. Pilot controls oil pressure to servomotor. Speed changer, upper left, acts on governor through the pilot-valve stem.

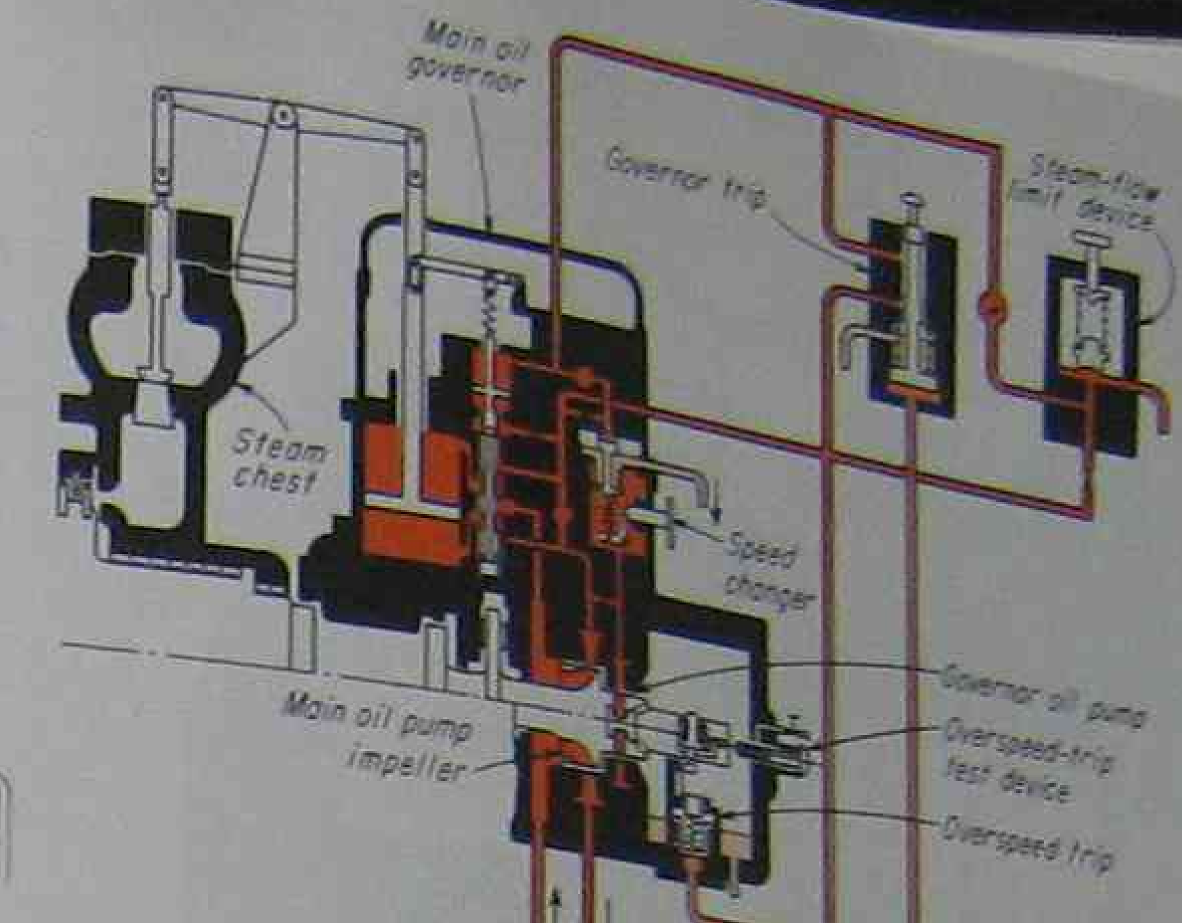
In one direct-acting mechanical governor for small turbines, Fig. 13, flyweights mount on flexible springs and directly move the nonrotating steam-control-valve stem which projects through the rotating governor shaft. Fig. 15 shows the same unit's overspeed-trip governor. Spring-loaded weight on the shaft swings out on overspeed to trip the lever and open the pilot valve, lower left. This relieves steam pressure in back of the steam stop valve above, slamming it shut.

Mechanical-hydraulic governing system for a multi-valved turbine, schematic, Fig. 16, uses rolling-contact flyweights to position the pilot valve above. Speed-changer

As turbines grow complex, governing systems expand to handle more functions



19 Exhaust-pressure regulator controls oil pressure to turbine's main servomotor



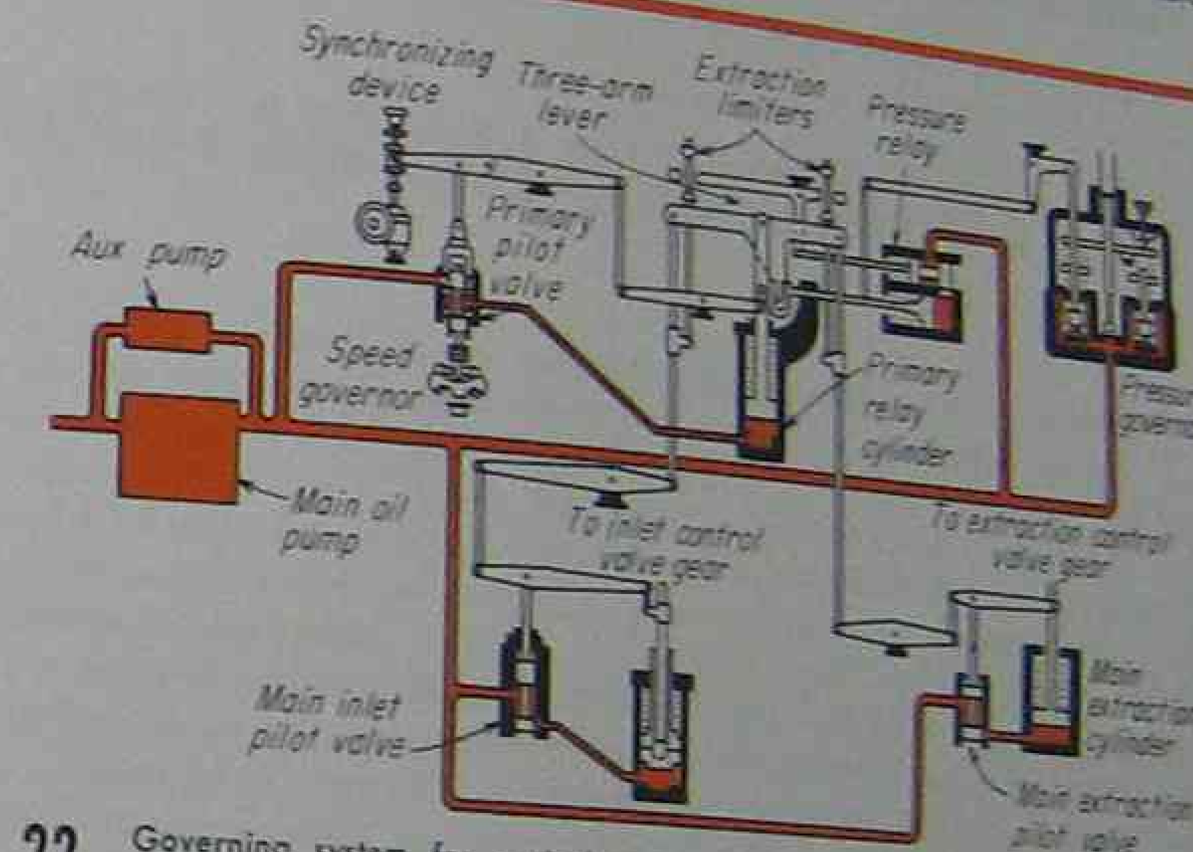
20 Completely hydraulic governing system for large unit needs only mechanical links between servomotor piston, relay and main steam valves; oil serves elsewhere

assembly stands at upper left. Pilot valve controls pressure to the primary servomotor; it controls the secondary pilot valve, lower right. This positions the secondary servomotor or power piston, which opens and closes main steam-control valves.

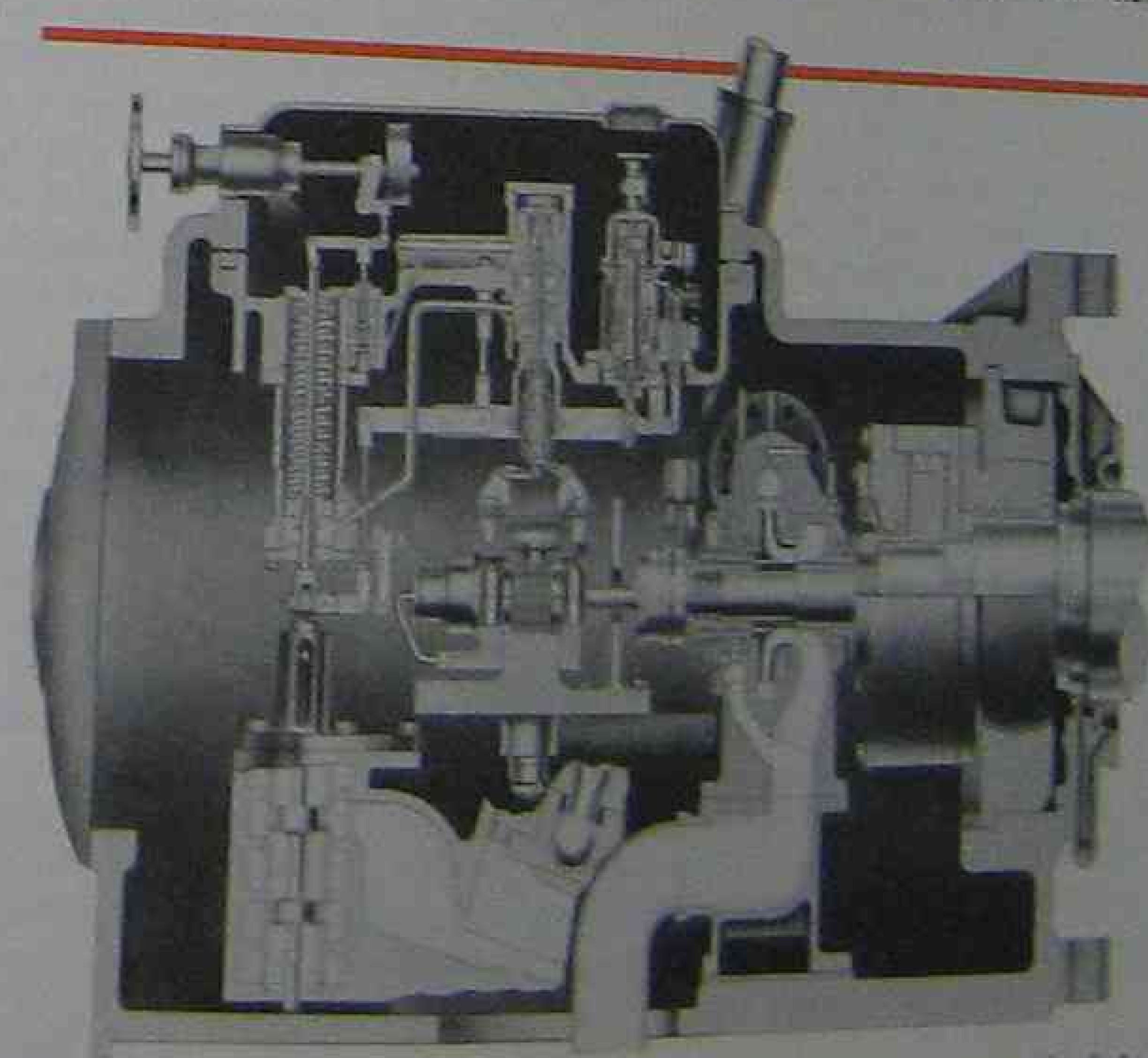
Fig. 17 diagrams a governing pedestal for a medium-size turbine. Fig. 18 is a schematic for a variable-speed oil-relay governor with an air-motor control on the floating lever. Fig. 19 shows the pressure regulator that controls steam valves on a backpressure unit. Fig. 20 has some details of a hydraulic type governor.

A double-auto-extraction unit's governing system, Fig. 21, combines three signals on three relay pistons to hold constant pressure at each extraction opening and constant shaft speed. Speed changer after governor impeller bleeds relaying-oil pressure, to shift droop characteristics as needed. Pressure regulators modify relaying-oil pressure to control extraction pressures.

Governing system for a single-automatic-extraction turbine needs fewer elements, Fig. 22; it controls main inlet and extraction valves. Fig. 23 takes a section through front standard of a reheat unit showing the flyweight governor, oil pump, relay valves, servomotors and other elements needed to automatically control this type of a steam turbine.

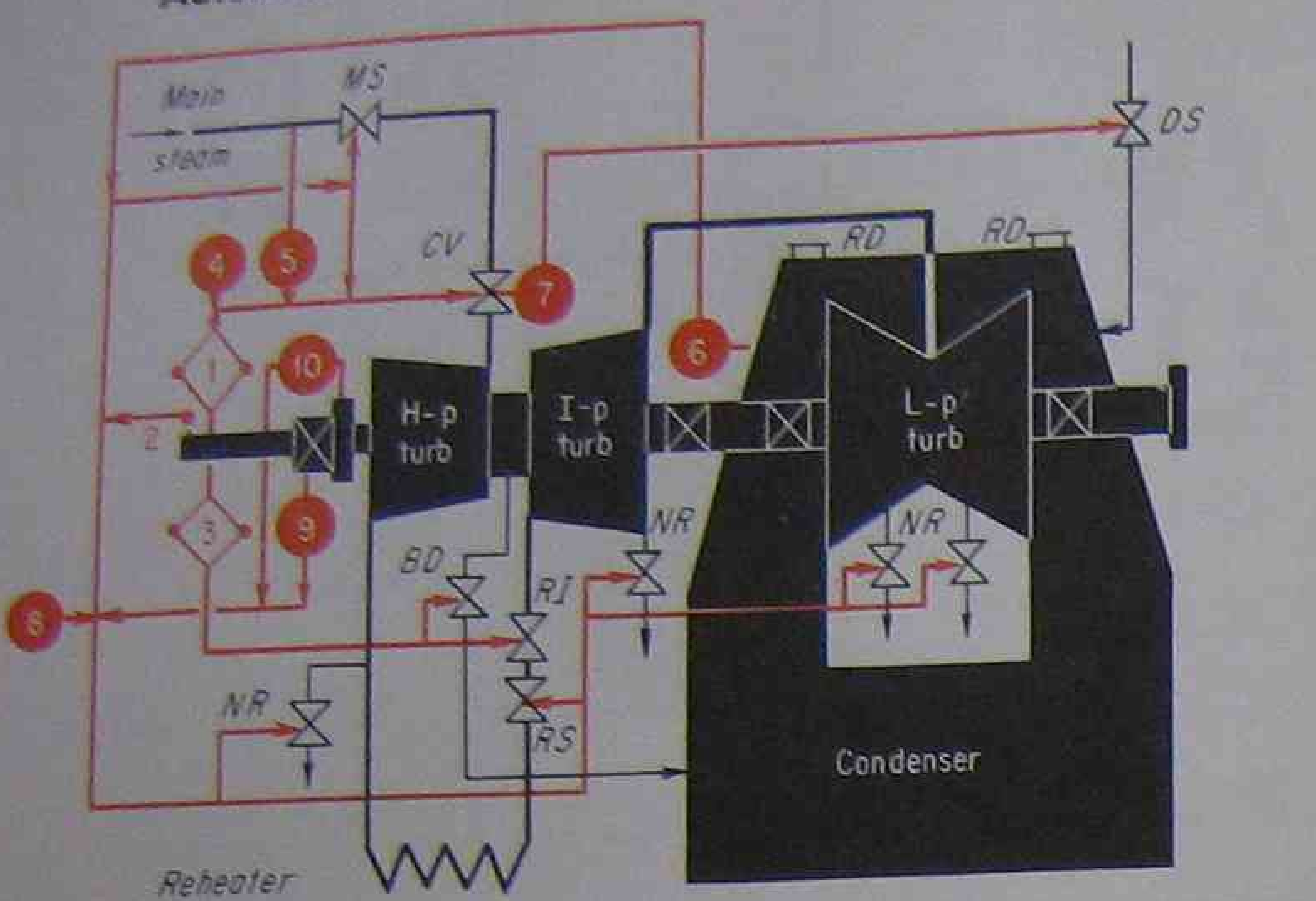


22 Governing system for a single-automatic-extraction turbine must control main inlet and extraction valves. Links move pilot valves to control servomotors



23 Governing system for reheat turbine also controls intercept valves, partly closing

Automatic load control uses these basic

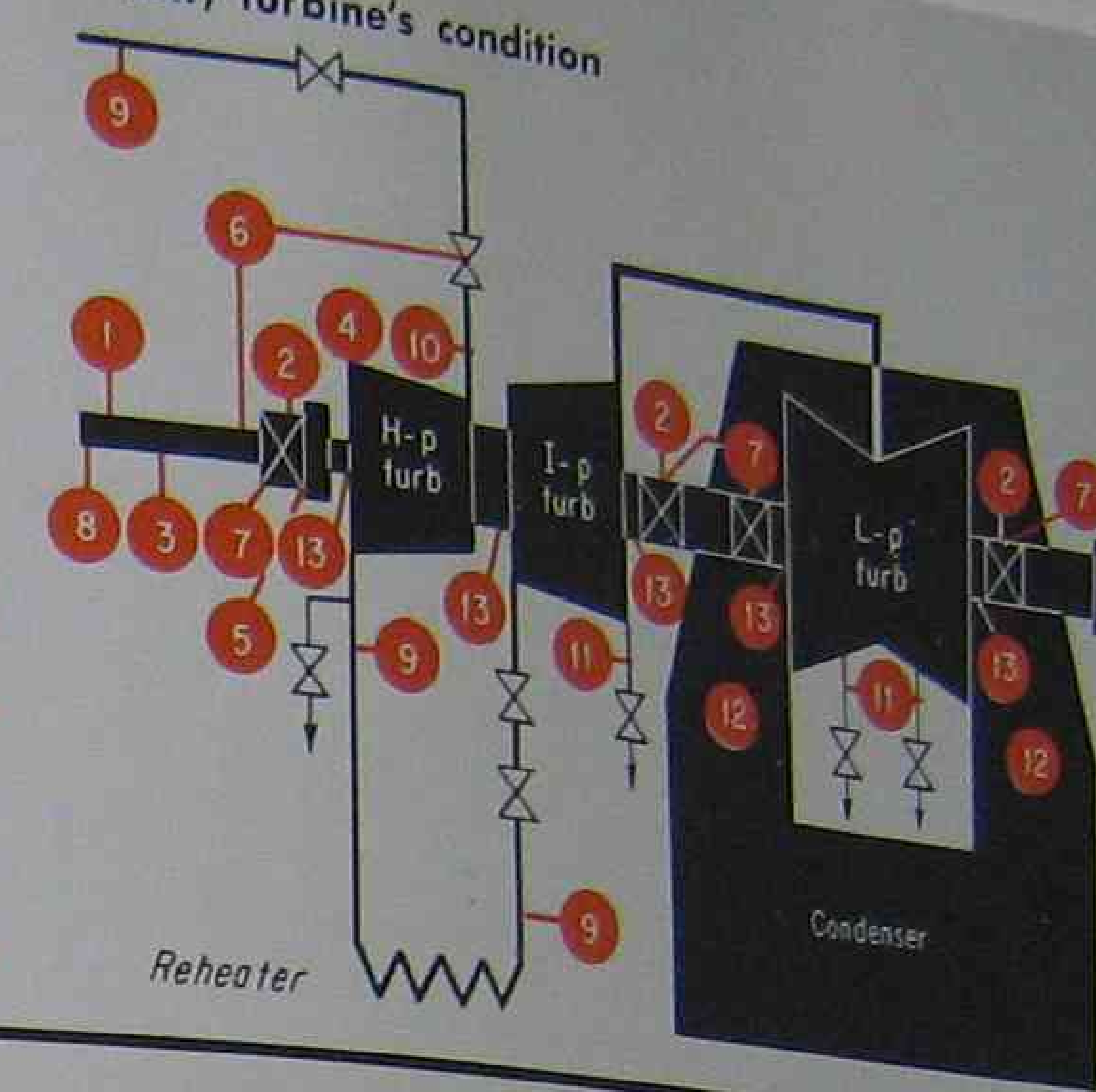


BD — Gland blowdown valve
CV — Turbine control valves
DS — Desuperheating spray valve
MS — Main steam stop valve
NR — Extraction nonreturn valve
RD — Relief diaphragm
RI — Reheat intercept valves
RS — Reheat stop valves

1 — Speed governor
2 — Overspeed governor
3 — Pre-emergency governor
4 — Load-limiting meter
5 — Initial-pressure regulator
6 — Low-vacuum trip
7 — Control-valve position meter
8 — Solenoid trip; hand or relay
9 — Low bearing-oil-pressure trip
10 — Thrust-position trip

Instruments give data on steam flow, turbine's condition

1 — Spindle-eccentricity meter
2 — Shaft-vibration meter
3 — Spindle-position meter
4 — Casing-expansion meter
5 — Differential-expansion meter
6 — Speed and governor-valve-position recorder
7 — Bearing-oil pressure, temperature, sight-flow glass, temperature alarm
8 — Tachometer
9 — Steam pressure, temperature, flow
10 — Nozzle-group pressures
11 — Extraction-steam pressure, temperature
12 — Exhaust-hood pressure, temperature
13 — Gland-steam pressure, temperature



CONTROLS AND INSTRUMENTATION provide for safe, efficient operation

Shaft speed is one of the most important variables in running a turbine. We have seen how some of the many available governing systems control speed in turbines large and small. For safety, the operator must always have on hand certain minimum information about his unit. As turbines grow larger, automatic aids or controllers are required.

Small-turbine needs in controls and instruments represent a minimum: speed governor, overspeed trip, throttle-pressure gage, throttle thermometer, exhaust-pressure gage, tachometer. Where the turbine controls or affects exhaust-header pressure, its speed governor includes an exhaust-pressure regulator to adjust the governor control.

Gland-steam pressures should be known to hold adequate shaft sealing. Turbines that use circulating-oil lubricating systems have pressure gages at pumps, bearing inlets and filters; thermometers are needed at oil-cooler inlets and outlets and at bearing outlets.

List of requisite equipment grows with turbine capacity and rising steam conditions. As an example, let's look at the gear needed by some reheat turbines for automatic control, above left, and instrumentation, right.

To control load automatically we depend on:

Speed governors to measure shaft speed and adjust governor-valve openings to pass the needed steam flow. Valves, in turn, keep speed within governor's regulation range. To hold exact speed, speed-changer spring or bellows tension must be adjusted manually or by an automatic frequency-measuring device. When generators work in parallel, the speed changers are used to divide total load between the turbines on the line.

Load-limiting meter overrides the speed governor to hold maximum load at any point the operator wants. The limiter controls steam flow during startup by setting governor valves as needed.

Pre-emergency speed governor controls governor valves or reheat-intercept valve or both during sudden drops in load. When turbine speed exceeds about 101% the governor starts closing the intercept valve; at 105% it shuts the valve completely. At about 104% speed, steam trapped in the reheater is bled down until spent. Speed governor takes control when speed drops below about 101%.

Initial pressure regulator keeps boiler and turbine from running with fast-dropping steam pressure and temperature. When pressure drops to about 90% of rating, regulator closes the control valves to hold the pressure. At about 80% pressure the control valves are in no-load position. As pressure rebuilds, speed control is returned to the governor.

Overspeed trip backs up the normal speed-governing system, including main and pre-emergency speed governors. At about 110% speed the trip goes into action, closing the main stop valve ahead of the control valves to shut off all main steam supply. Simultaneously it: (1) closes the reheat stop valve ahead of the reheat intercept valve to bottle up steam in the reheater (2) closes all check valves in extraction-steam lines, so bleed condensate can't evaporate and flow into the turbine (3) opens the gland-blowdown valve so steam flowing through the gland from h-p to i-p casing can blow down directly to the condenser instead of flowing through i-p and l-p turbines to cause overspeeding.

Low-vacuum trip shuts down every steam valve in the turbine the moment exhaust vacuum falls below about 20-in. Hg. This prevents overheating the exhaust hood and overstressing the last turbine stage. An associated alarm at the hood warns when its temperature exceeds safe limits. This may happen when all steam is shut off and the generator motors: last-stage blading churning the l-p steam trapped at exhaust may overheat both exhaust hood and shaft.

Exhaust-pressure relief must be provided when steam pressure in the exhaust hood rises above atmospheric. Rupture diaphragms in the hood burst at about 5 psig. They pass full steam flow so exhaust pressure doesn't exceed about 8 psig.

Desuperheating sprays are needed by some reheat units when their loads drop to less than 5%. A meter on the main-steam control valves or exhaust-hood thermostat operates the desuperheating-spray valve automatically to cool l-p end of the turbine during low-load operation.

Solenoid trip shuts down the turbine by closing all valves. It may be operated by hand or hooked up to control circuits of its boiler and generator.

Low bearing-oil-pressure trip guards against wiping main bearings when lubricating-oil supply fails. This trip activates the emergency shutdown systems, promptly closing all the steam valves.

Thrust trip measures position of the rotor relative to the thrust-bearing cage. If its position deviates from safe limits, trip operates the emergency shutdowns, closing all steam valves.

Other types of controls tie in with various special applications of the turbine. Their use depends on probabilities of emergencies arising, and on an economic balance: cost of guarding against the condition vs cost of possible damage.

Instrumentation on a turbine and its piping tells operator condition of the steam flowing through all parts of the unit and condition of the turbine structure. Available instruments range widely. Some are indicators, others recorders; some may be vital parts of the automatic-control system we just talked about.

Spindle-eccentricity meter checks on rotor's straightness. After an extended shutdown, shaft will bow if it's heated unevenly. Then it would vibrate excessively when brought up to operating speed. This meter tells operator amount of shaft bowing that may be present, before he runs up his rotor speed.

Shaft-vibration meter measures vibration amplitude at several of the turbine bearings. Out-of-balance rotor is quickly detected, as well as any buildup of boiler-steam deposits on the blading. A record of vibration readings taken at regular intervals shows up internal changes.

Spindle-position meter indicates axial position of the turbine spindle relative to stationary parts of the thrust bearing. It's needed because high thrust loads or wear of the bearing shoes may cause excessive movement.

Cylinder-expansion meter measures total elongation of the turbine casings. Instrument usually stands at the governor pedestal, since unit is anchored at its exhaust end.

Speed and governor-valve-position recorder helps operator run the turbine during startups and emergencies. It gives him better basis for judgment when pressure and temperature are off normal.

Bearing-oil condition throughout the lube system must be known to assure continuous smooth operation.

Steam conditions—pressure, temperature, flow—at all major openings in the turbine are informative. They tell the operator a lot about his machine, help him analyze cause of misoperation in an emergency.

Gland-steam conditions must be properly supervised by the operator at all times, if he's to run the turbine at maximum efficiency. Leakage steam that bypasses the turbine stages does no useful work.

LUBRICATION AND HYDRAULIC SYSTEMS

Lubrication is needed to minimize turning friction in main bearings, thrust bearing and reduction gears, cool the journal and other bearing surfaces. Friction heats journals and bearings, and the shaft conducts some heat from hot parts of the turbine to the bearings. Lube oil carries the heat away, keeps bearing temperature at a safe level, prolonging life of bearing and oil.

Smaller steam turbines usually get along with the simplest type of oil system to lubricate their bearings. From this minimum, systems range upward to the complex hydraulic set ups of the larger central-station steam turbines. In these units, oil serves as a power-transmitting fluid for speed- and load-control, high-pressure, overspeed and protective systems as well as a main-bearing lube.

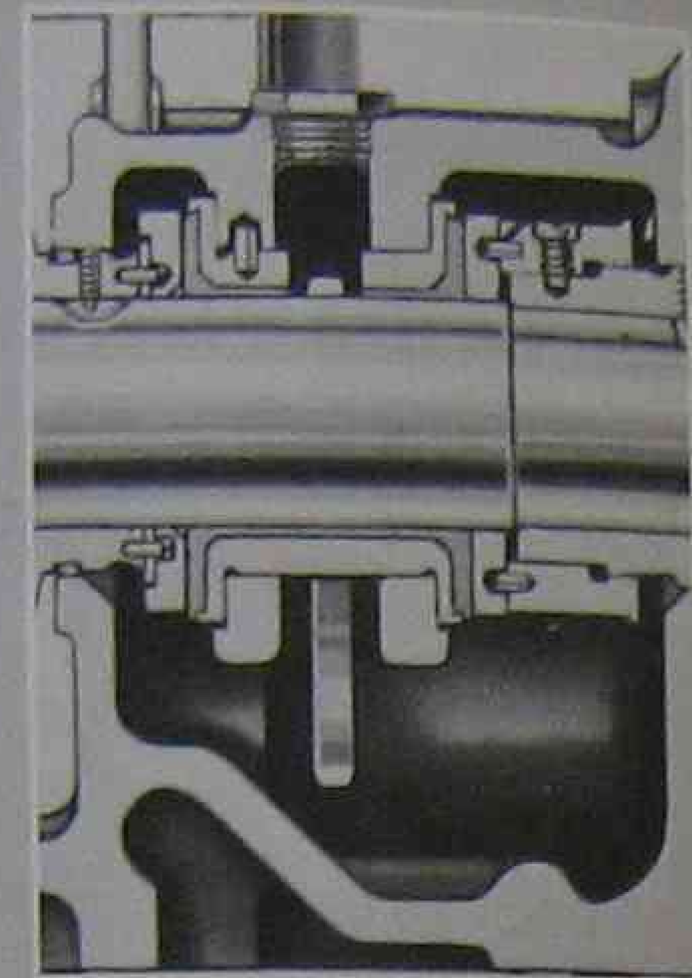
Bearings carry full weight of turbine rotors, let them spin free with minimum friction. Metal-to-metal contact would ruin bearing lining and journal in short order, so an oil film is forced between the two surfaces to separate them. Frictional effects take place between surface and oil, but mainly in the film.

Lube oil must wet the metal surfaces it separates, yet stay cohesive enough to resist being squeezed from area of maximum pressure. Low unit bearing pressures and high shaft speeds help form the needed oil wedge between surfaces. A good lubricating system is designed to prolong life of the oil.

Small bearings need a relatively small amount of oil. In the ring-oiled bearing, Fig. 1, a ring rides loosely on the journal. Lower arc of the ring dips into the oil reservoir below. As the shaft turns the ring revolves; oil from the reservoir clings to the ring and is carried with it to the top of the journal. Here the oil transfers to the journal. Grooves in the bearing surface let the oil flow in an axial direction to fill the space between journal and bearing. The continuously flowing oil drains back to the reservoir after passing through the bearing.

For smaller turbines, the reservoir has enough outer surface area to radiate heat picked up by oil to the room air. Larger turbines using ring-oiled bearings may need cooling coils in their reservoirs.

Larger bearings would not get enough lubrication or good cooling from a simple ring-oiling arrangement. They need forced-circulation oil systems. Since forced circulation requires pumps to produce pressure, economics usually combines the lube system with the fluid-power hydraulic system that actuates turbine controls.



1 Ring-oiled journal bearing: brass ring runs through oil reservoir below

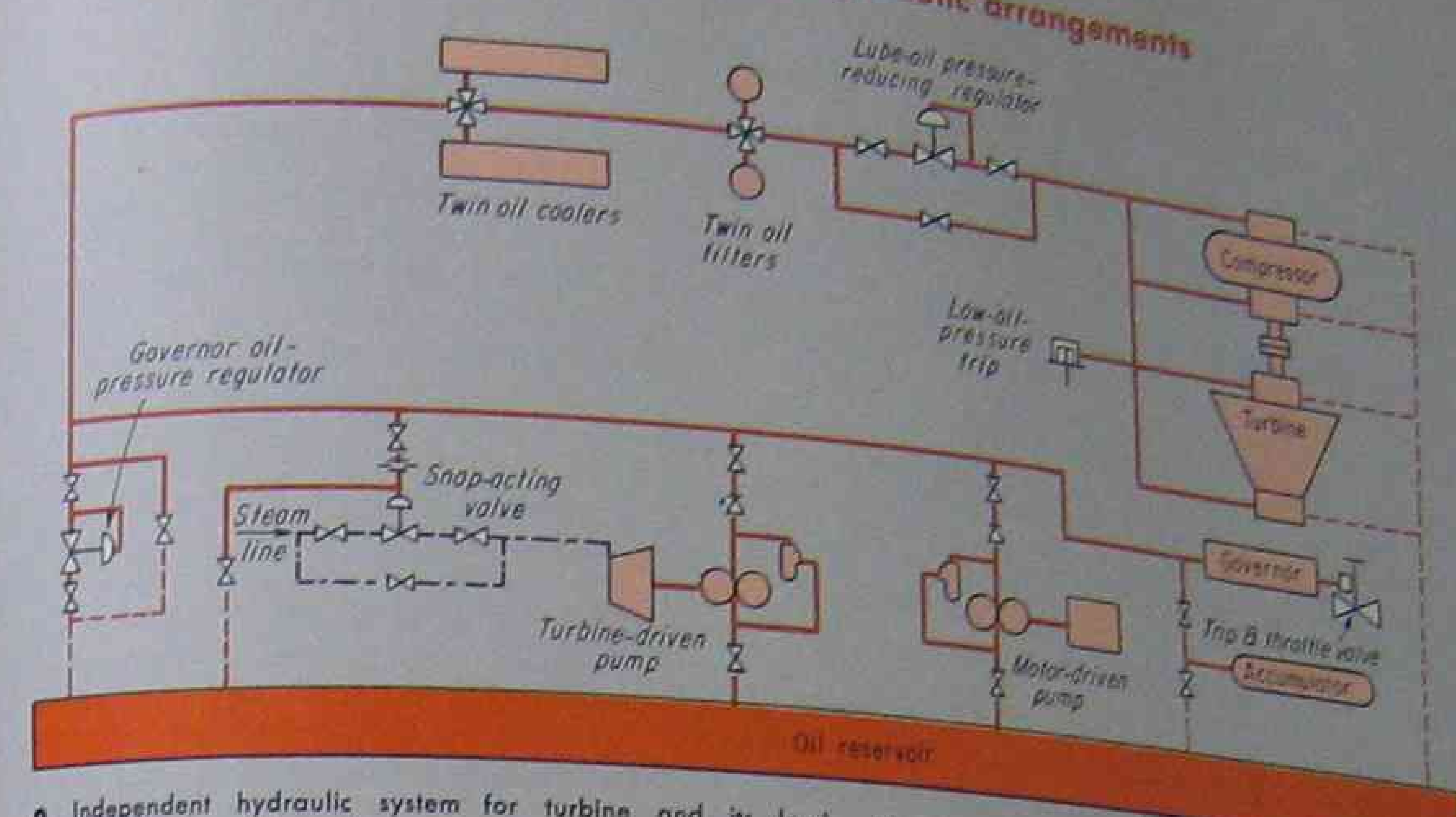
Basic independent oil system, Fig. 2, combines reservoir, pumps, coolers, pressure controls, accumulator, filters and piping. Two pumps are used for service continuity: if one fails in service the other starts fast so oil-pressure won't decay to the point where it trips any of the shutdown controls. Pump capacity, boosted by the accumulator, must supply the high transient oil flow the control system needs during large changes in shaft load.

Positive-displacement rotary pumps in Fig. 2 have a flooded suction protected by strainers. The motor-driven pump normally runs with the oil-turbine-driven unit as standby. Bypass relief valves protect pumps and drivers against accidental shutoff of discharge valves. A snap-acting valve automatically brings the turbine-driven standby pump into operation on drop in oil pressure.

Backpressure regulator and reducing valves hold oil-system pressure at operating level. Twin oil coolers and filters in front of the lube-pressure regulator remove heat and clean the oil before it flows to the bearings. At coolers and filters, 4-way valves let one of the elements come out of service so it can be cleaned without shutting down the main turbine.

Oil filters usually are sized to remove dirt particles larger than 3 to 5 mils diameter with a pressure drop of 3 to 5 psi at design temperature and flow. In addition, there's usually auxiliary cleaning and conditioning equipment—either continuous or batch operation.

Various purifying methods may be used. Centrifuging, filtering or settling mechanically separate the contaminants from the oil, but do not cause chemical changes. Adsorbent purifiers using fuller's earth or similar materials and chemical treatments tend to restore neutralization number, but care is needed when using them on oils



2 Independent hydraulic system for turbine and its load supplies lube oil and governing-system fluid power. Two oil pumps with different drivers assure continuous oil supply in emergencies. Oil circulates continuously through bearings

containing inhibitors. Heating oil to about 200 F assists in breaking any emulsion. When emulsion persists, oil should be discarded and replaced with fresh lubricant.

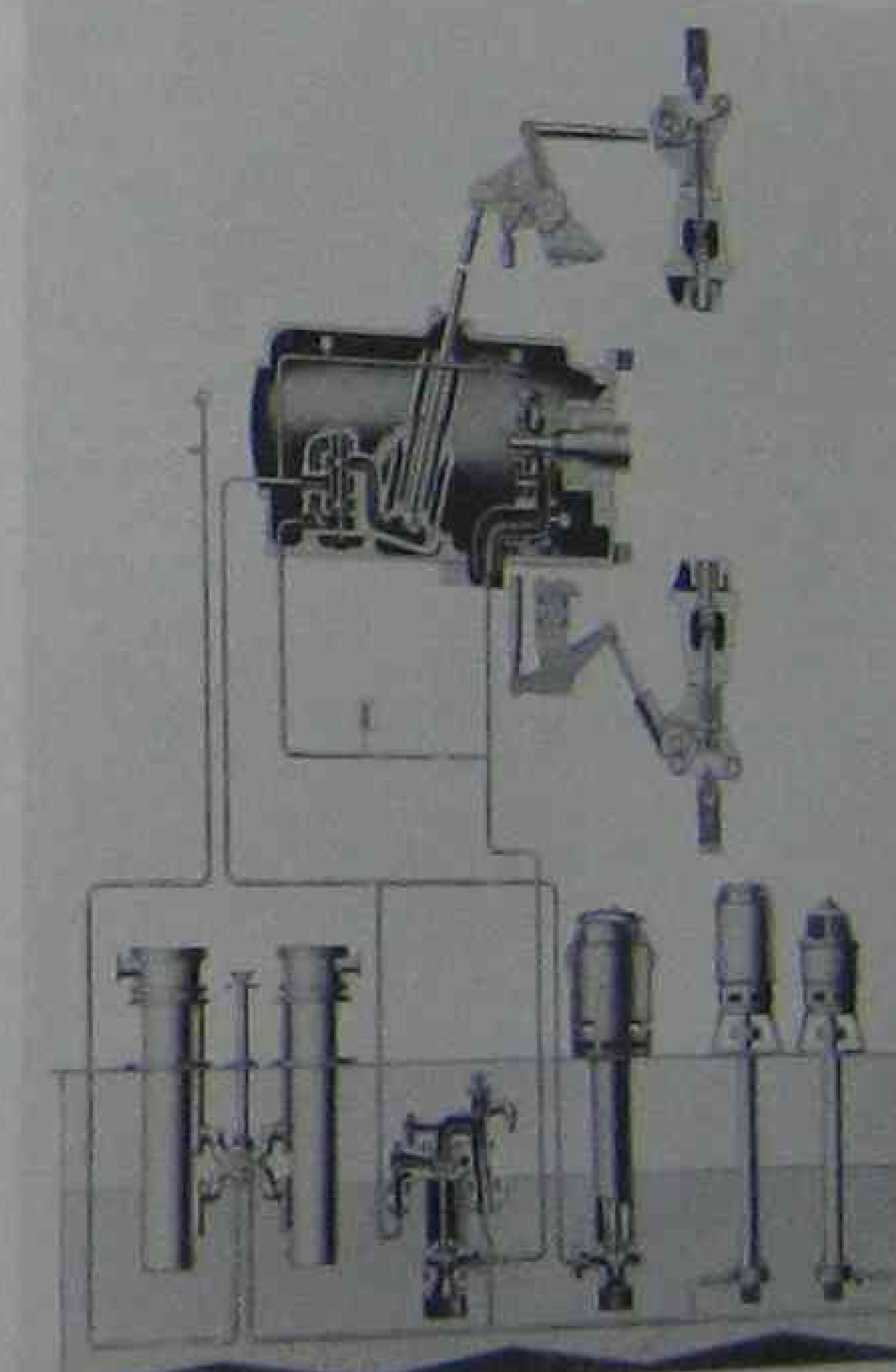
Piping that passes near hot lines or machine parts may be totally enclosed in an outer sheath. If any leak develops in the oil lines, sheath piping drains the leakage back to the oil reservoir. This for safety's sake—most lube oils ignite when sprayed on surfaces at temperatures of 675 F and above.

An alternate solution involves fire-resistant lubricants in the main bearings of large central-station units. Much study has centered on F-R lubes. High cost is one factor, but main deterrent to their adoption by industry has been effect on generator insulation.

Large-turbine hydraulic systems, Fig. 3, have undergone an evolution in the past decade. Their oil reservoirs, usually located in the plant basement, house pumps, regulators, oil coolers and booster pumps or ejectors. Leakage from any of these drains right into the reservoir.

A centrifugal oil pump, directly mounted on the main turbine shaft, is primed by an oil-driven booster pump. Booster mounts below oil level in the reservoir and feeds the shaft-pump suction with oil at 10 to 20 psig. Oil from the shaft pump goes to the small oil-turbine wheel at 200 psig, and that drives the direct-connected booster pump. After discharging from the oil turbine at 30 to 40 psig, oil flows through the coolers on its way to the bearings through a pressure-reducing valve.

This oiling system takes advantage of centrifugal-pump characteristics. Normally it pumps only the bearing's lube needs. Servo mechanism, which needs momentarily large oil flows, goes next to the shaft-mounted pump to reduce size of oil piping between reservoir and turbine.



3 Hydraulic system for large turbine: shaft-driven centrifugal is backed up by motor-driven and booster pumps

These turbine-oil recommendations are typical

Oil properties	Forced circulation: Direct connected	Forced circulation: Gear drive	Ring oiled: Water cooled	Ring oiled: No cooling
Viscosity, ssu, at 100 F	140-170	250-350	250-350	375-525
130 F	76-92	120-165	120-155	180-230
210 F	40-45	47-60	40-45	50-70
Flash point, F	330	350	330	350
Neutralization No.	0.20 max	0.20 max	0.20 max	0.20 max
Oxidation test, hr	1000	1000	1000	1000
Operating conditions				
Min oil temperature before start	50	70	50	—
Bearing inlet temperature, F	110-120	110-120	140-160	140-180
Bearing outlet temperature, F	140-160	140-160	140-160	—

A motor- or steam-driven auxiliary backup pump starts automatically when oil pressure falls below operating level. System in Fig. 3 includes further backup pressure: small ac and dc motor-driven pumps. Ac pump also works as the turning-gear oil pump when the unit coasts down.

Meter orifices in the oil-feed pipe to each bearing divide flow so each bearing gets its adequate share. A basket strainer ahead of each orifice guards against plugging.

As oil drains from bearings and the thrust bearing, some of it flows through an illuminated sight box and over a thermometer. Abnormal operation on any individual bearing can be caught immediately.

Oil should be high quality refined petroleum. It should hold no water, acid, alkali, soap, grit, resin or anything else that might damage it or the turbine parts it contacts. But in case a little water does entrain, the oil should protect ferrous parts of the turbine from rusting. A good turbine oil separates freely from water, has no tendency to emulsify permanently.

Turbine manufacturers usually recommend that the oil vendor undertake responsibility for its conformance to specs, table above. Each manufacturer specifies limits that will satisfy his unit's needs.

Using oils near the upper viscosity limits raises friction losses in the bearings, hence bearing outlet temperatures. Extremely high temperatures speed oil's oxidation rate, abbreviate its useful life. Most satisfactory operation uses the lower-viscosity oils, running with moderate temperatures for longer oil life.

Maintenance of turbine oils means keeping them free of foreign materials that wander into the system. An oil-purification system can be added as accessory to the lube system—it should be checked out with the turbine-oil vendor and turbine manufacturer.

Turbine oil needs a regular analysis to guard against serious lube failure. Analysis reveals progressive changes in oil's properties and possibly system malfunctions.

Water is the frequent oil contaminant; it comes from: (1) steam leakage from shaft glands (2) condensation of moisture in entrained air (3) leaky oil-cooler tubes. Whenever a turbine is idle, its operators should take the opportunity to drain water accumulated in the bottom of the reservoir and other low points of the lube system. Excessive water accumulation should be checked and remedied when found.

Oil foams when it's whipped and mixed with air. Foaming may cause: (1) loss of pressure, which interferes with the control system (2) poor lubrication (3) increased oil oxidation (4) oil overflow and loss. Sometimes just a slight reduction in circulating pressure, or running the oil at a slightly higher temperature, eliminates foaming. Or there may be too much or too little oil in the system.

Additives improve lube performance by inhibiting deteriorating factors. Benefits: longer oil life, cleaner systems, rust prevention and lower wear rates. On the other hand, additives must not affect the oil's base properties.

Grease may be the main bearing lube on a small unit. It's used on many turbine auxiliaries: needle and roller bearings on valve gear, racks and sleeve bearings on control-valve gear, the synchronizing-device gearbox, governor linkage pins and bearings, front pedestal expansion slides and exhaust-hood side feet.

Greases must be high-temperature lubricants, applied by hand, with a pressure gun or by hand-compression cups. They must not separate on standing or when heated below their dropping temperature, must remain solid at operating temperatures, hold together under centrifugal force without separation of soap and oil components. For good operation, grease must resist oxidation, be free of dirt, fillers, abrasives, excess moisture, free acid or lime.

Schedule of oil changes or greasings depends on turbine service and the working environment. Constant vigilance over condition and supply of lube is the price of reliable, continuous steam-turbine operation.

PERFORMANCE varies with turbine cycle

As we've said, turbines convert internal energy of steam into mechanical shaft work, Fig. 1. Amount of energy that steam has available for conversion depends on its initial temperature and pressure and on the way it expands to the lower pressure in the turbine.

A *Mollier chart*, Fig. 2, tells how much energy steam has at each state in terms of enthalpy and how much energy is available for a given expansion. First, the point where the curves for initial pressure and temperature cross tells us enthalpy of the steam h_1 in Btu per lb. Dropping straight down from this point to the curve for the exhaust pressure, we find ideal enthalpy at the exhaust h_{2s}. Then:

$$\text{Available energy} = h_1 - h_{2s}$$

An ideal turbine would convert all this energy to work at 100% *engine efficiency*. Actual turbines can't achieve this high performance. Fig. 2 charts *condition lines* for 100, 75, 50, 25 and 0% engine efficiencies. Energy initially in the steam stays in the steam if it isn't converted to work. We find this by:

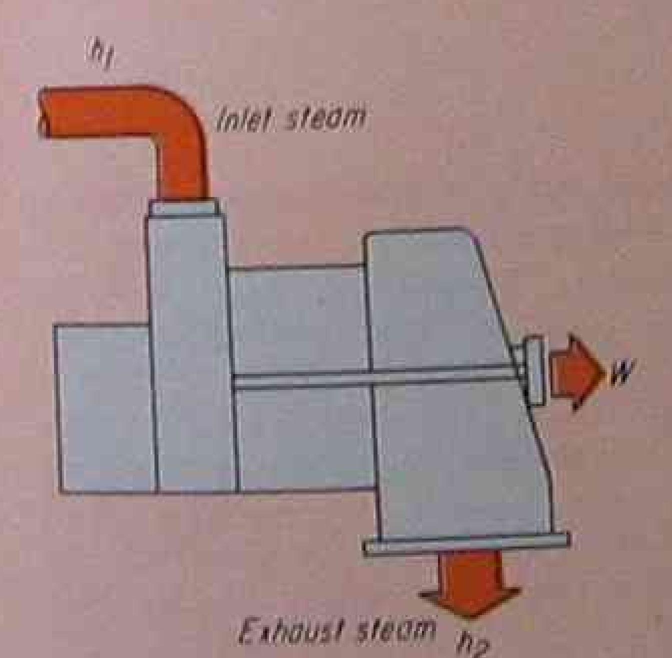
$$h_2 = h_1 - (\text{avail energy}) \times (\text{engine efficiency})/100$$

Turbine design generally aims for highest engine efficiency compatible with the cost of energy. Engine efficiency also varies for a given turbine with the load it's carrying. Fig. 3 shows the condition line for a single-valve turbine at different loads. As the control valve throttles steam flow with dropping load, steam pressure in the nozzle chest drops. This reduces the available energy and the exhaust enthalpy rises with dropping load.

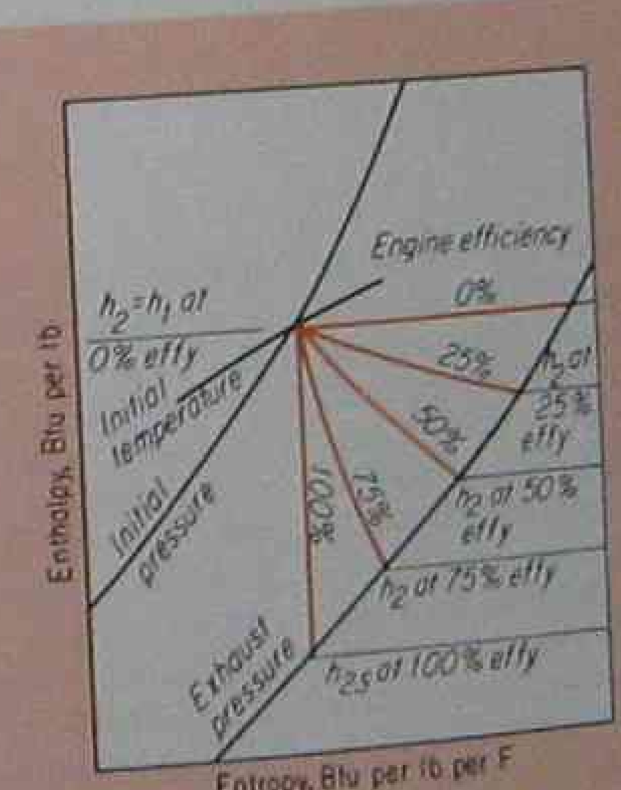
Noncondensing turbines. Fig. 4 shows the variation in available energy in steam used by these units. To find actual work developed, multiply available energy from this graph by engine efficiency. Exhaust moisture will be less than the ideal of Fig. 4.

Fig. 5 shows typical engine efficiencies at rated capacity for single-stage turbines ranging from 30 to 500 shaft hp. Efficiencies rise with both capacity and shaft speed. To find engine efficiencies at half capacity, multiply by the half-load factors.

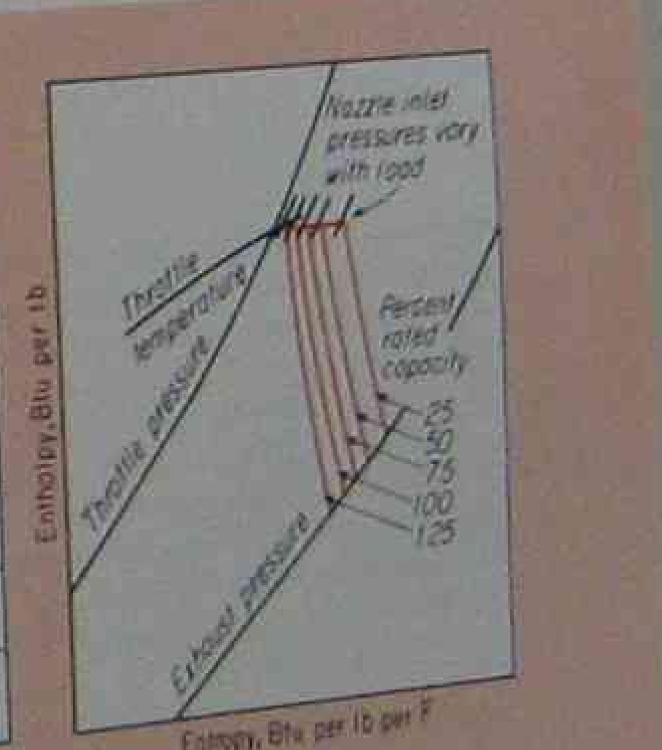
Condensing turbines. Fig. 6 shows the available en-



1 Turbine converts internal energy of steam into useful shaft-work output



2 Engine efficiency measures available energy changed to shaft-work output

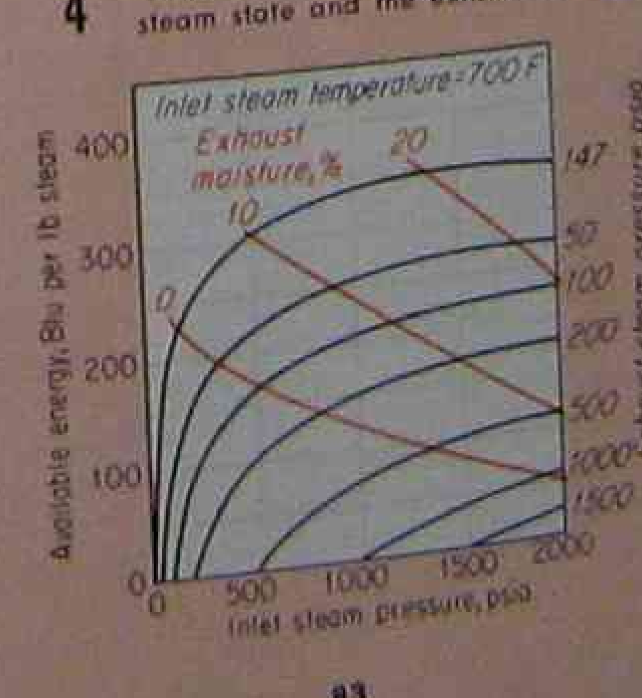


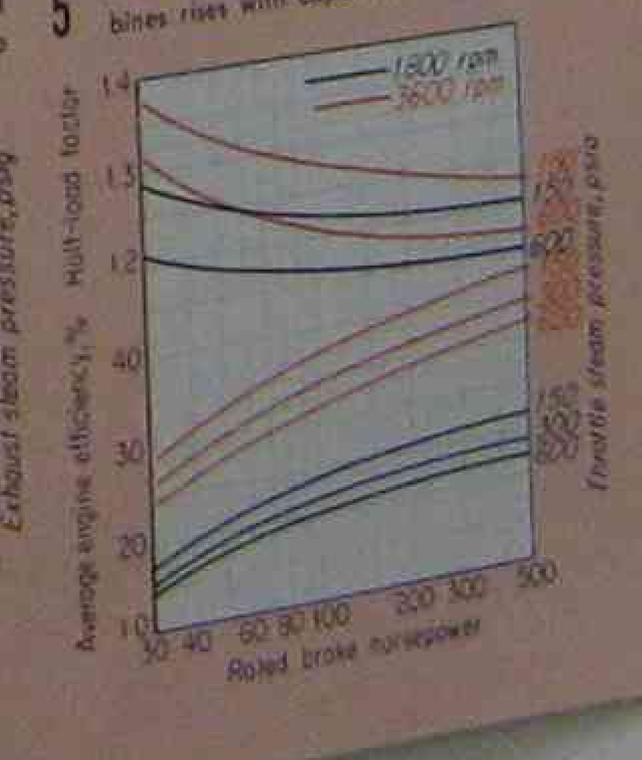
3 Turbine condition line shifts as the governing valve throttles steam flow

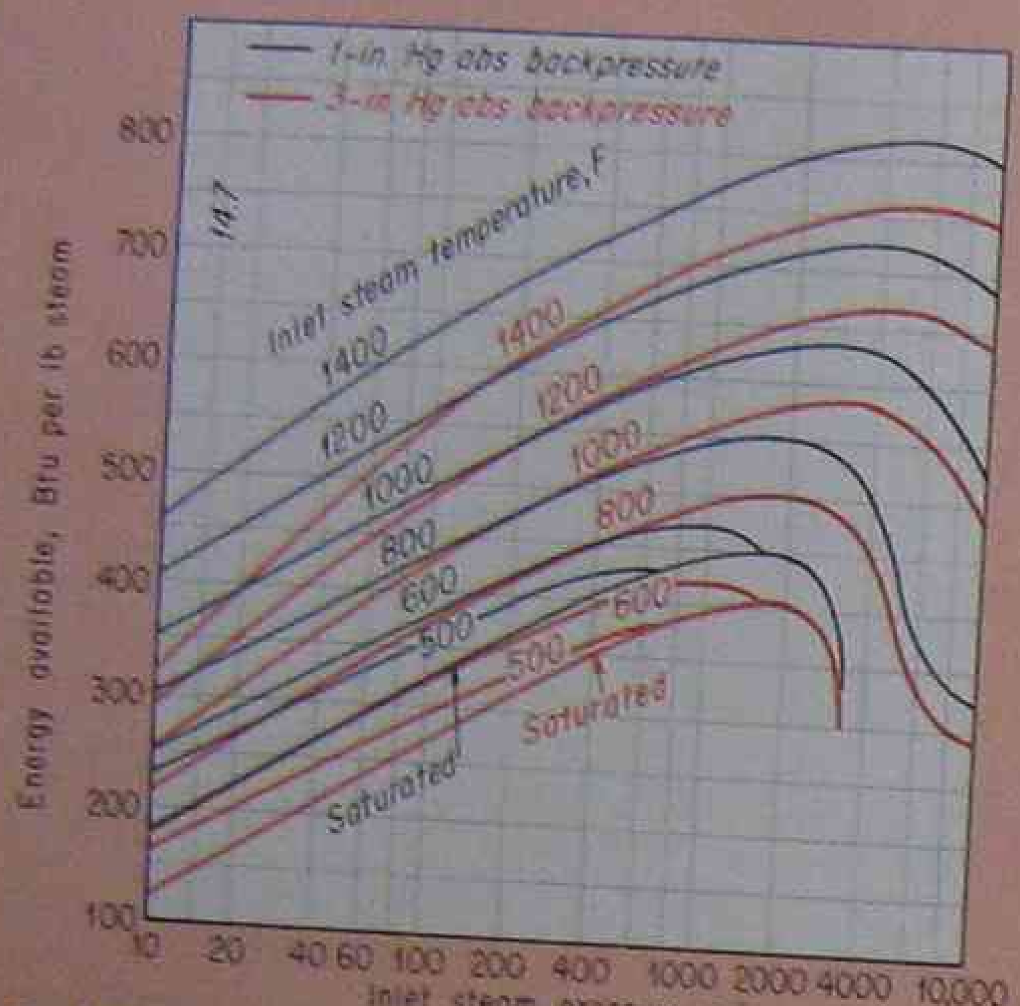
4 Available energy depends on initial steam state and the exhaust pressure

5 Engine efficiency of single-stage turbines rises with capacity, shaft speed

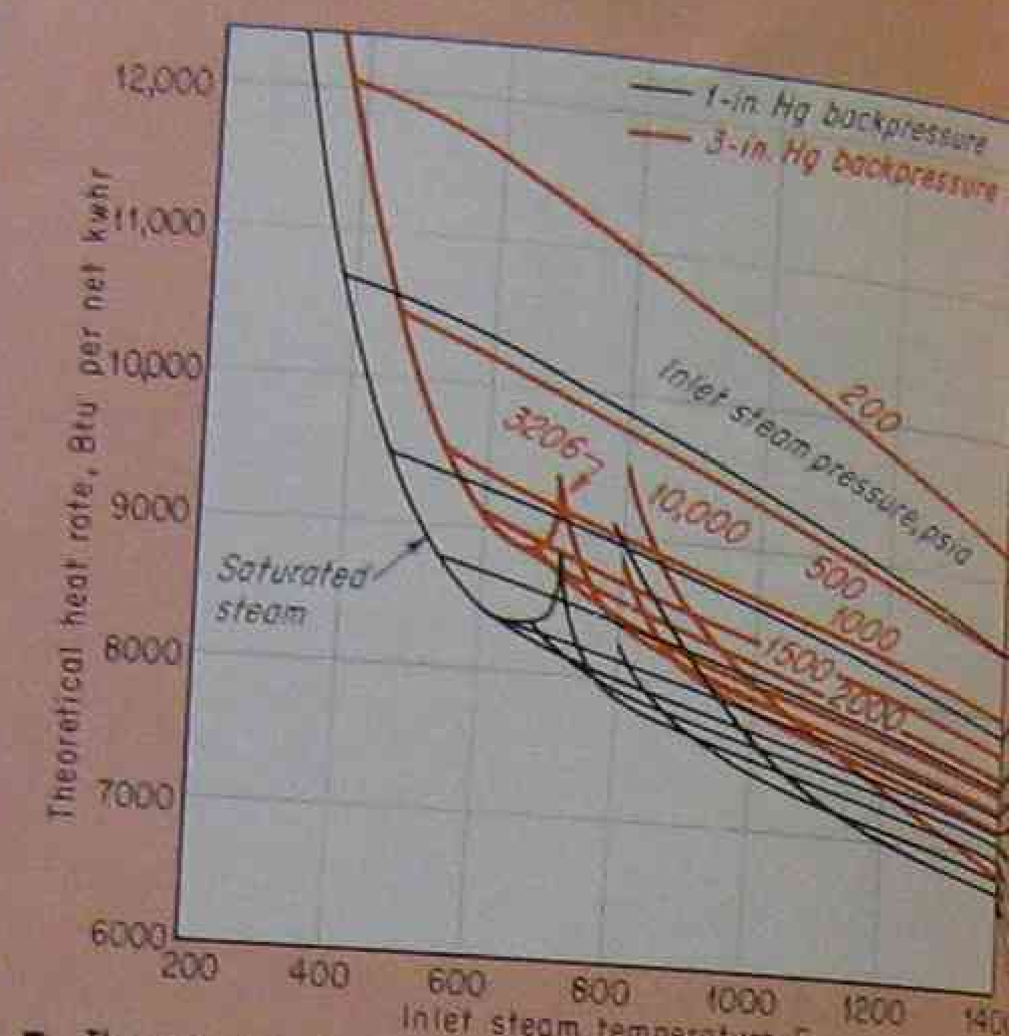
Inlet and exhaust steam conditions fix shaft power generated by a turbine







6 Available energy for condensing turbines depends on inlet steam pressure and temperature as well as backpressure



7 Theoretical heat rate of condensing turbines generally improves with rising throttle-steam pressure and temperature

Estimating turbine performance

Given: 5000-kw turbine, inlet steam at 800 F, 1000 psia; exhaust at 1-in.-Hg abs; engine efficiency of 74%.

Solution: From Fig. 7:

Theoretical heat rate = 8400 Btu per kwhr

Actual heat rate = 8400/0.74 = 11,350 Btu per kwhr

Thermal efficiency = 3413/11,350 = 0.301 or 30.1%

From Fig. 6:

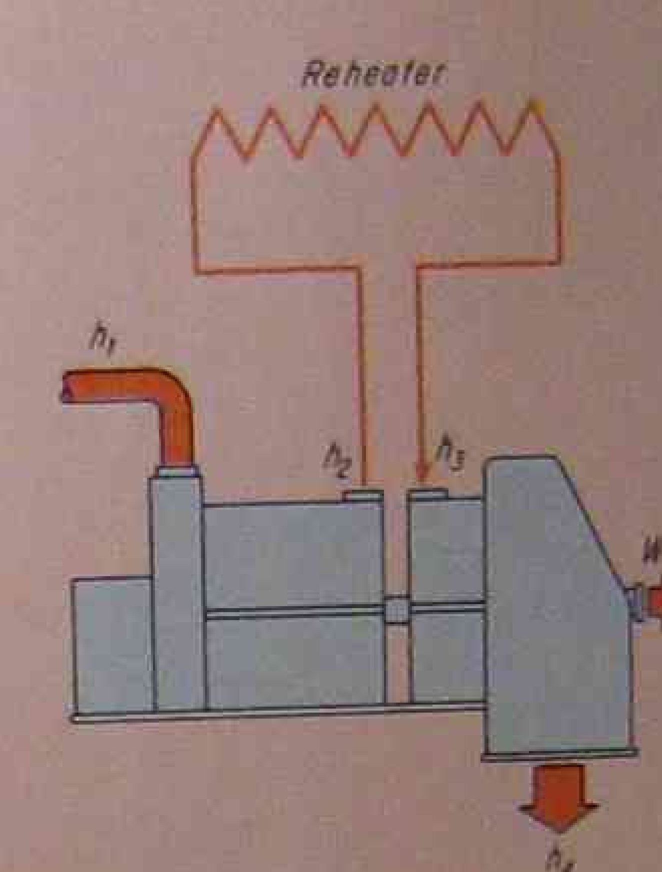
Available energy = 545 Btu per lb

Energy released = 545 × 0.74 = 403 Btu per lb

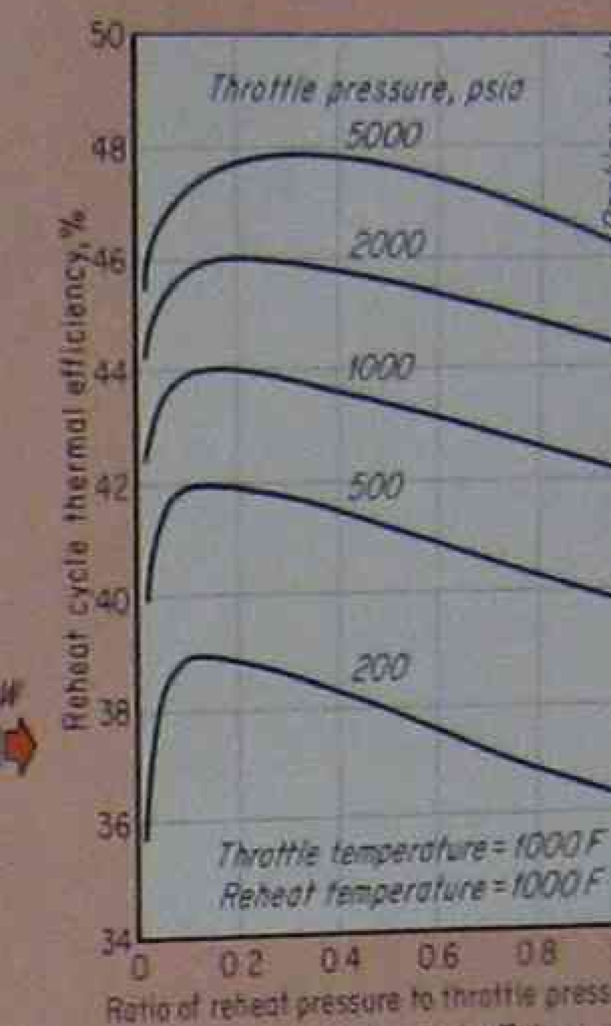
Full-load output = 5000 × 3413 = 17,060,000 Btu per hr

Steam flow = 17,060,000/403 = 42,300 lb per hr

Full-load steam rate of turbine = 42,300/5000 = 8.46 lb per kwhr



8 Reheat turbine has all its steam reheated at an intermediate pressure



9 Maximum gain in thermal efficiency is made with reheats at low pressures

ergy in steam used by these units with 1-in.- and 3-in.-Hg-abs backpressure. The energy generally rises with initial steam temperature and pressure, with some exceptions beyond 5000-psia pressure and around the critical point at 3206 psia and 705 F.

Set of red curves shows that higher backpressure reduces available energy at all inlet-steam levels. In Fig. 6 we see the reason for intensive development of high-pressure high-temperature turbines in the past decade: higher available energy makes better use of expensive materials.

Theoretical heat rates of condensing turbines are charted in Fig. 7, which corresponds to Fig. 6. Theoretical heat rate drops with increasing inlet-steam temperature and pressure. Again, exceptions to this general trend come around the critical-point area. The red curves show that rising backpressure raises the heat rates.

Heat rate relates directly to the overall thermal efficiency of the turbine:

Thermal efficiency = 3413/(*heat rate*)

where heat rate is in Btu per kwhr.

Actual heat rates of condensing turbines can be estimated roughly from Fig. 7: divide theoretical heat rates by the unit's engine efficiency, example on facing page.

Reheat turbines, Fig. 8, have seen wide acceptance in large central stations during the last decade. Here the steam expands in the front end of the turbine. At an intermediate stage, the entire flow passes through a resuperheater in the boiler furnace. Reheated steam then goes back to the turbine at the lower pressure and expands through its remaining stages to the condenser.

Fig. 9 shows effect on ideal-cycle thermal efficiency of reheating at different initial throttle pressures. Throttle and reheated temperatures remain constant at 1000 F at the different pressures. Straight-condensing theoretical efficiency is given at the 1.0 ratio of reheat to throttle pressure (marked *Rankine cycle*).

As reheat pressure drops the thermal efficiency rises until pressure ratio is about 0.3 at 5000 psia, 0.1 at 200 psia. Any further reheat-pressure drop rapidly depresses thermal efficiency. In general, reheating gains about 2% in thermal efficiency. When actual engine efficiencies are considered, relative gain can be larger, about 4 to 5%.

Regenerative cycles, Fig. 10, extract steam from several stages in the turbine to heat feedwater being returned to the boiler. Raising feedwater temperature to highest possible level at h_r means less fuel is needed to evaporate each lb to the throttle-steam state. Amount of heating that the extracted steam can do depends on throttle pressure of the cycle, Fig. 11.

Boiler has to add $(h_1 - h_r)$ Btu to each lb of steam. In the straight-condensing cycle, h_r is saturated-liquid enthalpy at the condenser pressure. In the regenerative cycle h_r ideally can be raised to the saturation enthalpy of the throttle pressure (if it's less than the critical pressure of steam, 3206.2 psia).

Fig. 12 shows reduction in straight-condensing-cycle heat rates for a range of throttle pressures and temperatures when feedwater is heated regeneratively to the maximum possible. But these curves assume that heating is done reversibly in an infinite number of heaters. Actual regenerative cycles must be limited to relatively few heaters, so we can't achieve ideal performance. Fig. 13 shows

effect of the number of heaters and percentage of possible feedwater enthalpy rise on the actual regenerative-cycle heat rate.

For example, let's assume the 5-mw turbine we were studying, above, has three feedwater heaters. Let's estimate actual heat rate with 1-in.-Hg-abs backpressure in the condenser, feedwater heated to 70% of maximum possible rise.

Straight-condensing heat rate was 11,350 Btu per kwhr. From Fig. 11, maximum possible rise = 495 Btu per lb. Actual enthalpy rise = 495 × 0.7 = 346 Btu per lb. From Fig. 12, theoretical reduction in straight-condensing heat rate = 14.8%. From Fig. 13, reduction for number of heaters and actual enthalpy rise = 0.71. Correction factor for actual regenerative cycle = 1 − 0.148 × 0.71 = 0.895. Then full-load regenerative-cycle heat rate = 11,350 × 0.895 = 10,160 Btu per kwhr.

For any given number of heaters there's a certain enthalpy rise at which economy gain becomes a maximum, as Fig. 13 shows. For higher or lower enthalpy rises the gain will be less. For example, if one heater takes steam from the throttle to produce 100% feedwater enthalpy rise, using main steam gives no economy gain because the heating steam has done no work in the turbine. At the other extreme, a heater tied to the exhaust stage can't do any heating because steam and condensate have the same temperature.

Let's extend our study a little more by finding the heat rate of our 5-mw turbine when it uses ten heaters for feedwater heating to 90% of maximum possible rise. From Fig. 13, heat-rate reduction for number of heaters and actual enthalpy rise = 0.89. Then correction factor for the actual regenerative cycle becomes 1 − 0.148 × 0.89 = 0.868. Full-load 10-heater regenerative-cycle heat rate becomes 11,350 × 0.868 = 9850 Btu per kwhr. Adding (10 − 3) = 7 heaters reduces the heat rate by (10,160 − 9850) = 310 Btu per kwhr.

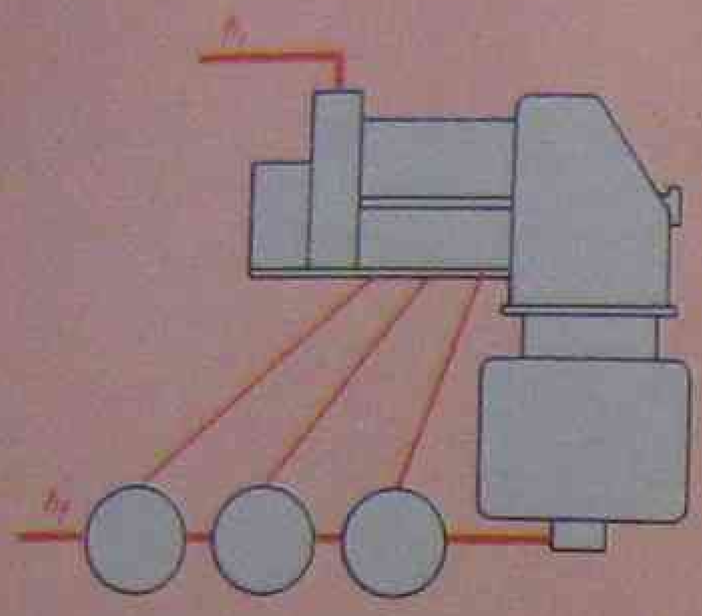
To decide on number of heaters, we would have to compare carrying charges on the extra heaters and piping to the reduction in annual fuel costs. If saving is greater than carrying charges, the larger number of heaters would probably be justified.

Reheat-regenerative cycles combine steam-reheating and regenerative-feedwater-heating principles to achieve high overall thermal efficiencies. Coupling high efficiencies with high throttle pressures and temperatures, these modern turbine cycles appear to be trending to the ultimate in overall thermal performance.

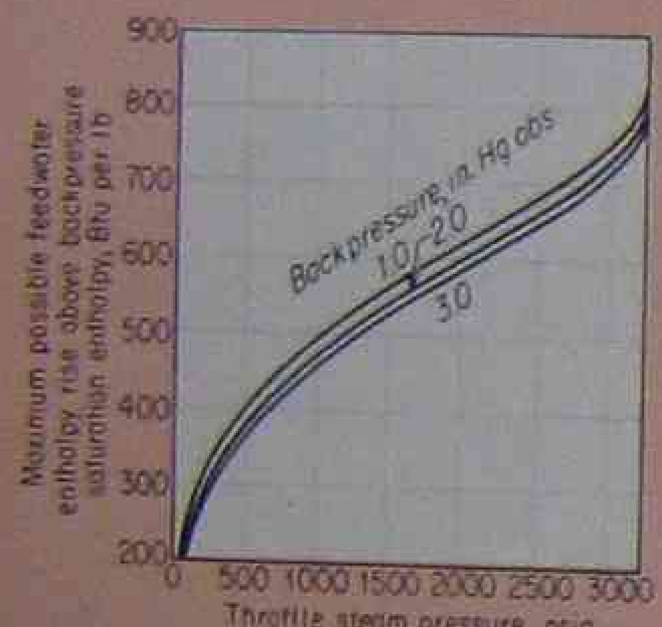
Fig. 14 shows full-load heat rates that can be achieved by turbines ranging in capacity from 100 to 500 mw, with throttle pressures from 1000 to 6000 psig. Initial and reheat temperatures are kept at 1000 F for all units and backpressure is constant at 1.5-in.-Hg abs. All units use six stages of regenerative feedwater heating. Their gross heat rates are figured on generator-terminal outputs. Deducting feedwater-pump energy input from generator output gives net heat rates.

Difference in these curve families shows large work input the feedwater pump needs at high throttle pressures. Taking this into account, we find that optimum net heat rate for the 100-mw unit comes at about 3400-psig throttle pressure. Best pressure for the 500-mw unit: about

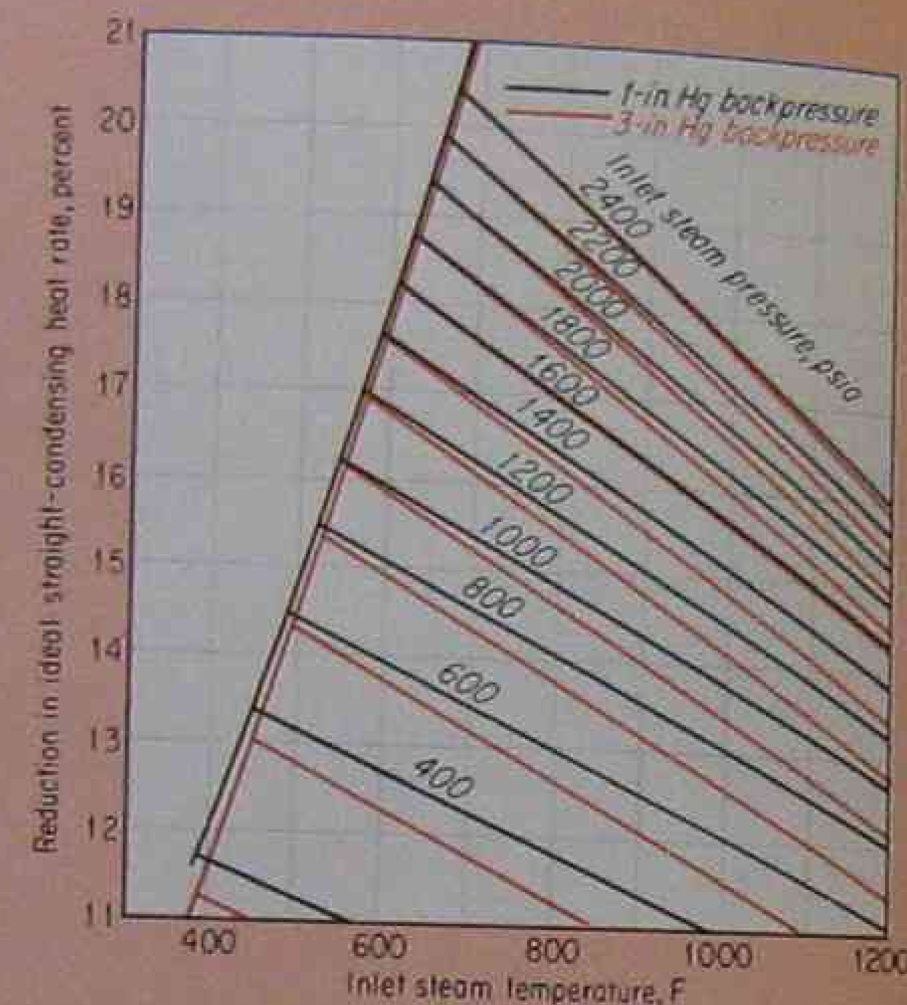
Regenerative cycles conserve energy



10 Regenerative heating cycle extracts steam to heat returning feedwater



11 Feedwater can be heated through higher ranges as steam pressure rises



12 Rising steam pressure improves gain in regenerative-cycle heat rate, but that's offset by rising inlet-steam temperature

4400 psig. Spread of the curves shows performance gain that can be made by going to larger-capacity units at higher throttle-steam pressures. Capacities, steam conditions, cycles, investment and fuel costs—all must be carefully coordinated to arrive at best choice for a central-station generating unit.

Typical large condensing turbines vary in their heat-rate performance with changing load and different back-pressures, Fig. 15 to 18. As units become larger and steam conditions higher, heat rates over the range of outputs become lower and lower. This is partly because of larger size, partly because higher steam conditions can be justified for the larger units, Fig. 14. This gives them an inherent thermodynamic advantage.

In Fig. 15 to 18, heat rates are usually minimum near maximum capacity. This follows because throttling isn't needed to control steam flow at full load, Fig. 3. As drawn, these curves are the *mean of valve loops*. The true curve shows a series of concave downward loops that reach a minimum where valves of a nozzle chest are fully open. In contrast, at loads where the valve is just starting to crack open there's a large throttling loss; curve takes a sudden jump upward. Our curves show the overall trend, but true heat rates may be about 2% different from the trend at the valve-opening and mid-valve points.

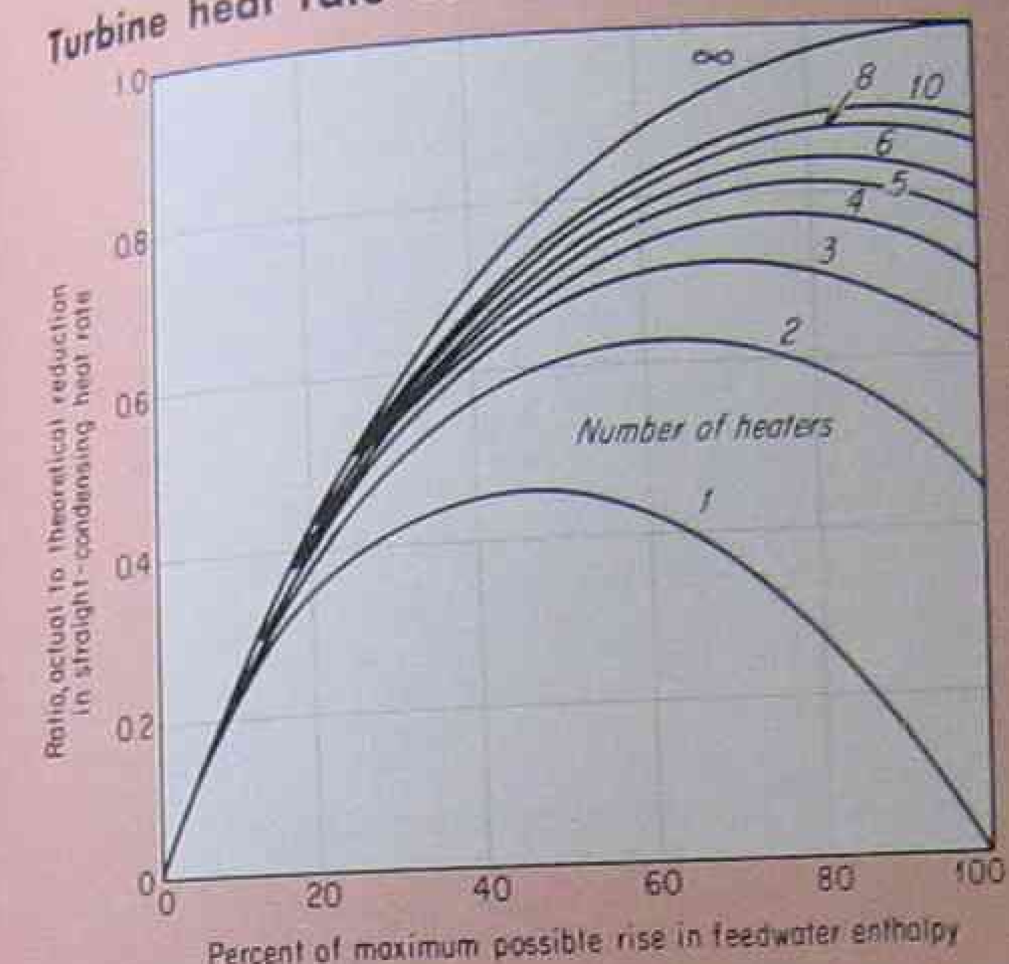
Mechanical-drive turbines often run at variable speed, in contrast to the central-station units running at constant speed. The m-d turbine's shaft output largely depends on shaft speed. Fig. 5 shows engine efficiencies usually achieved by single-stage m-d turbines.

Fig. 19 charts throttle-steam flows for a multistage and a single-stage m-d turbine. Upper scale shows corresponding shaft speed for the shaft output plotted below. Plot of steam flow vs output is called a *Willans line*. For a constant-speed single-valve turbine this line is almost straight. But as we see here, the Willans line becomes curved for variable-speed turbines.

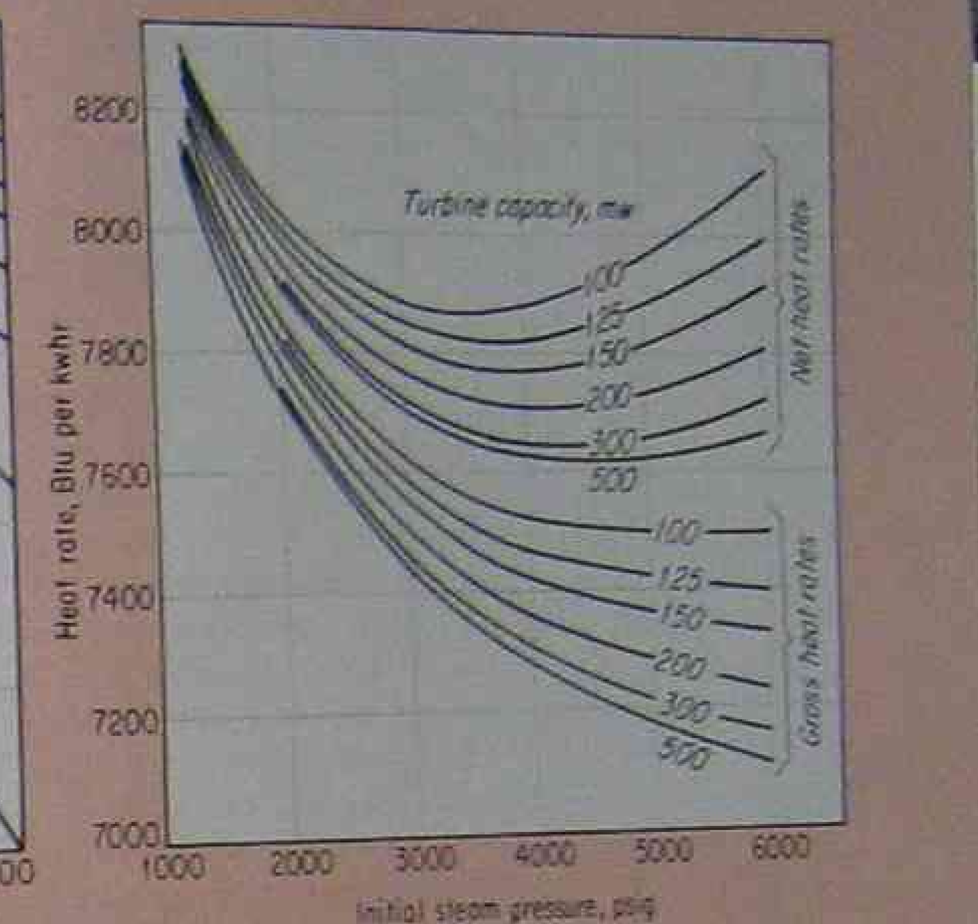
This comparison shows that for the given steam conditions with both turbines exhausting at 150 psig, the single-stage turbine is more efficient at low loads and less efficient at high loads than the multistage unit. Choice between turbines for a given application can only be made after a close study of operating conditions.

Automatic-extraction turbines can be used when both electric power and process steam are needed at the same time. If process steam is needed at two different pressures and electric-energy needs are equal to, or less than, energy available from process-steam flow through a turbine, a noncondensing single-automatic-extraction turbine can be used. However, if electric-energy needs are higher than process-steam flows can generate, a condensing double-automatic-extraction unit should be used.

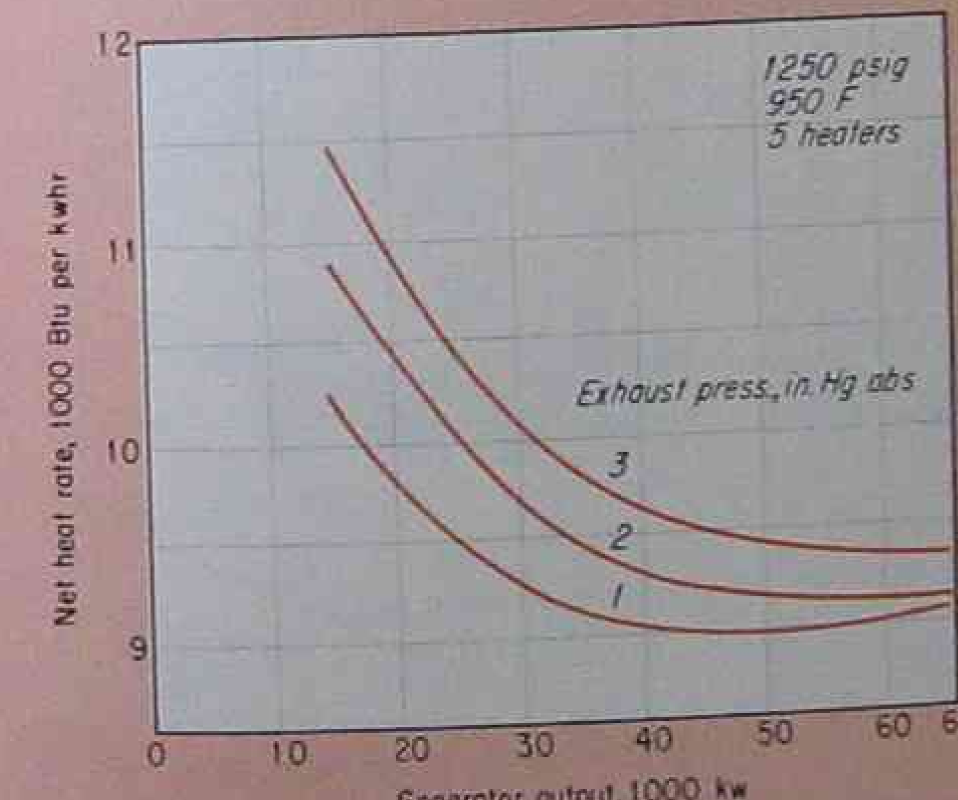
Turbine heat rate varies with load and steam cycle



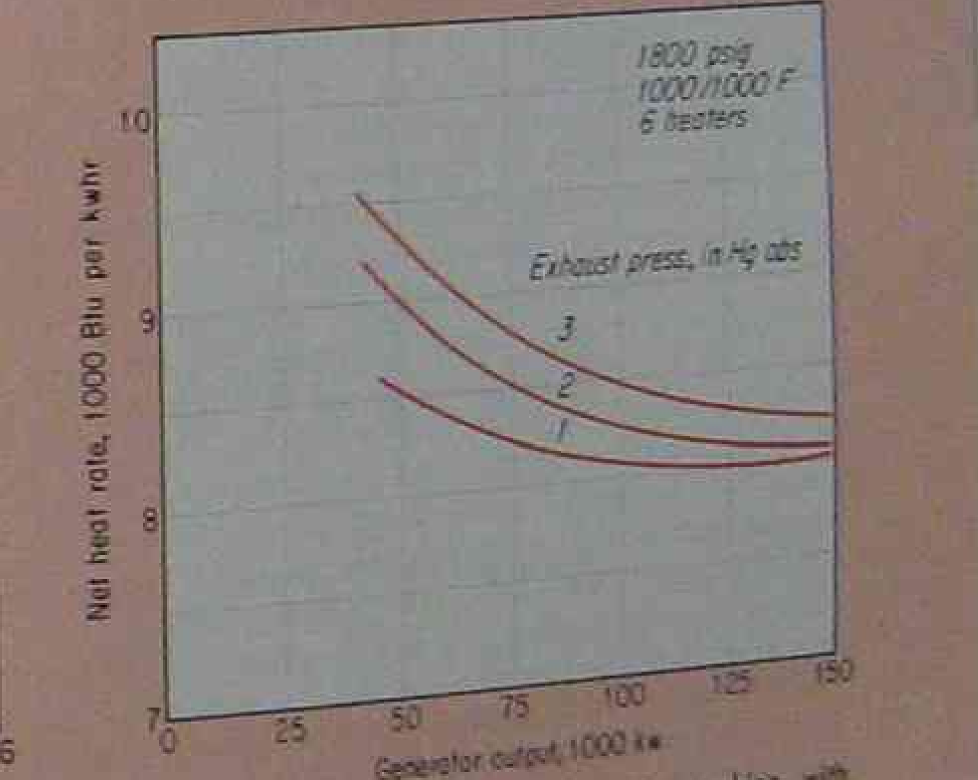
13 More extraction points, properly placed in an actual turbine regenerative cycle, improve the heat rate



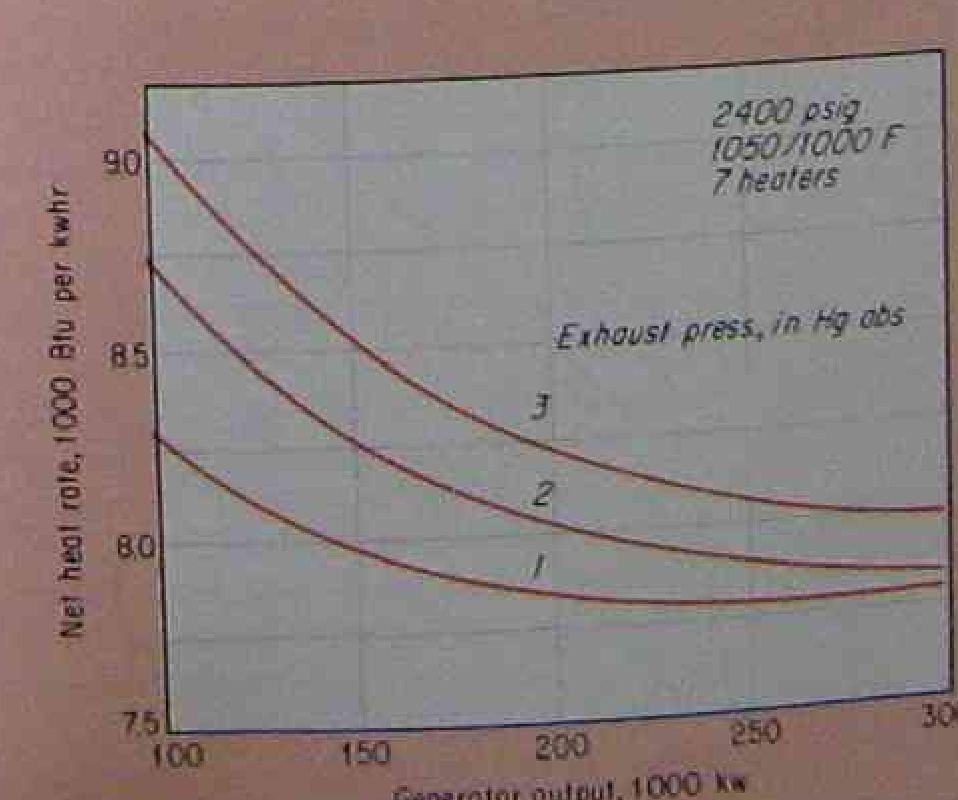
14 Full-load heat rates predicted for turbines with six heaters, 1000/1000-F steam, 1.5-in.-Hg-abs exhaust pressure



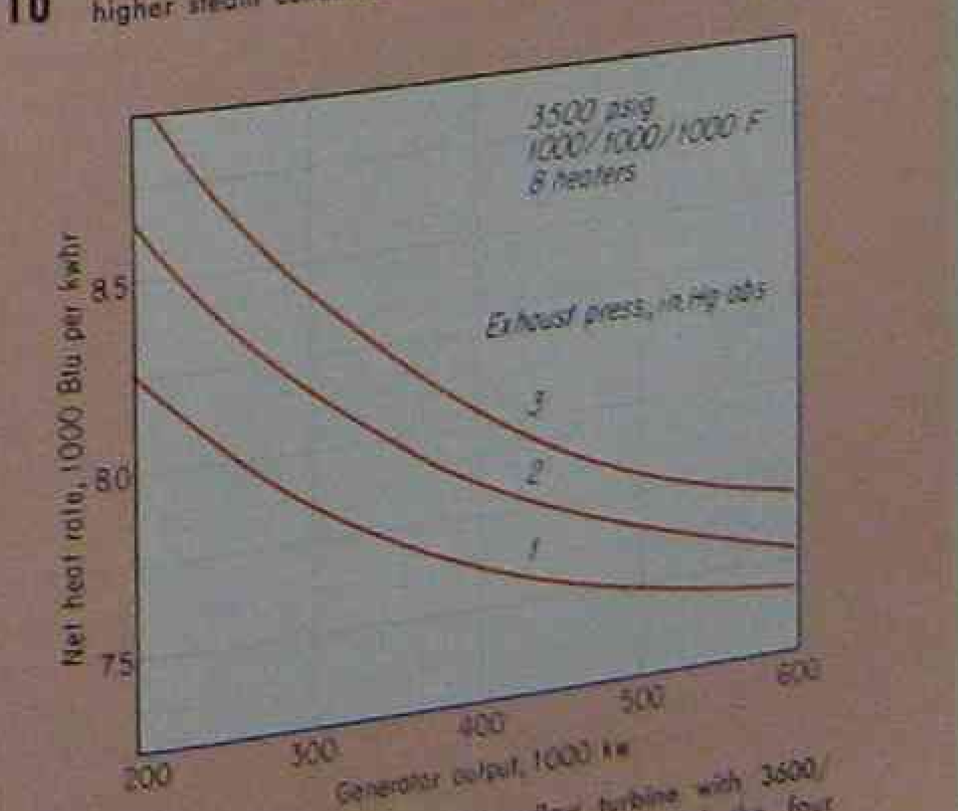
15 Typical heat rates of 66-mw tandem-compound double-flow 3600-rpm unit vary with load and backpressure



16 A 150-mw tandem-compound double-flow turbine with higher steam conditions works at lower overall heat rates



17 A 300-mw tandem-compound four-flow turbine with still higher inlet steam shows more heat-rate improvement



18 A 600-mw cross-compound four-flow turbine with 3600/1800-rpm shafts shows best performance of the four

Performance of automatic-extraction turbines is given in terms of steam flow, both throttle and extraction, versus shaft output in hp or kw. Heat rate of the energy produced by noncondensing automatic-extraction turbines does not vary greatly. Energy chargeable to the turbine is only the relatively small amount "skimmed off" the steam before it flows out through extraction and exhaust openings to the process lines.

Condensed-steam energy of condensing automatic-extraction units is charged to the turbine. So most efficient generating condition comes at maximum-extraction flows (the same as minimum exhaust-section flow). For a given kw output, throttle flow is maximum when extraction is maximum and heat rate minimum; conversely, throttle flow is minimum and heat rate maximum at zero extraction.

In sample performance curve of a condensing single-automatic-extraction turbine, Fig. 20, the family of parallel curves defines required throttle-steam flow at given kw output and extraction flow. Each parallel line is the turbine's Willans line at a constant value of extraction flow. Lower ranges of kw output limit amount of steam that may be extracted.

When output is all produced by extracted steam, exhaust section of the turbine is idle. Then the blades churn steam trapped in these stages and rapidly raise its temperature to a point where the blades may fail. To prevent this, a small "cooling" steam flow is sent through the exhaust section to keep blading temperature at a safe level. Cooling steam carries off windage loss of the idle blades.

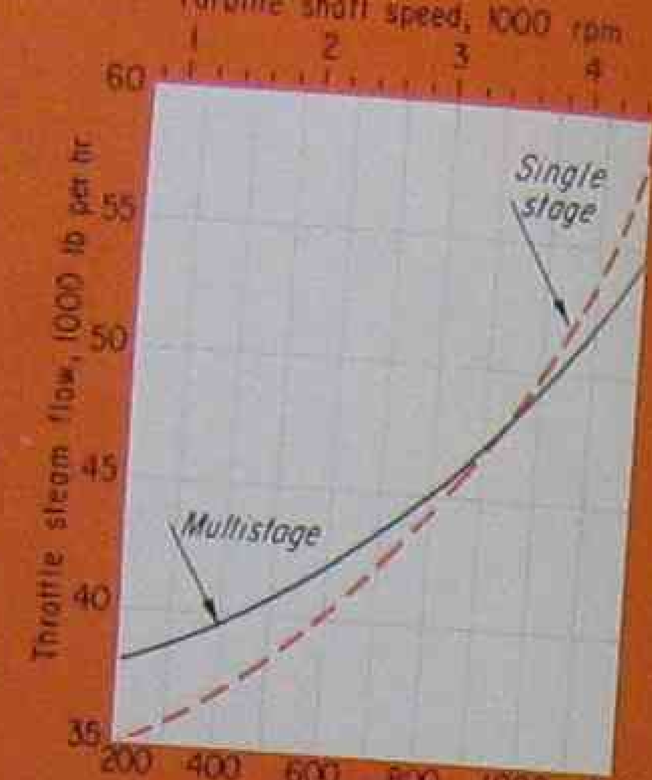
In Fig. 20, the *minimum flow to exhaust* curve shows relation between kw output produced on extracted steam alone and corresponding throttle-steam flow. This curve intersects each of the constant-extraction-flow curves at the throttle flow that equals sum of the cooling steam and extraction steam. In expanding through the first section of the turbine from throttle to extraction stage, the cooling steam does its share of work. But in passing through the exhaust section from extraction stage to exhaust flange it doesn't do any work—only cools last-stage blading.

Exhaust sections often pass just as much exhaust flow as needed to produce rated output at zero extraction. Additional output in the overload range can be generated by admitting more steam at throttle and extracting it. This is the *maximum flow to exhaust* curve. Other limits are *maximum throttle flow* and *maximum generator output*.

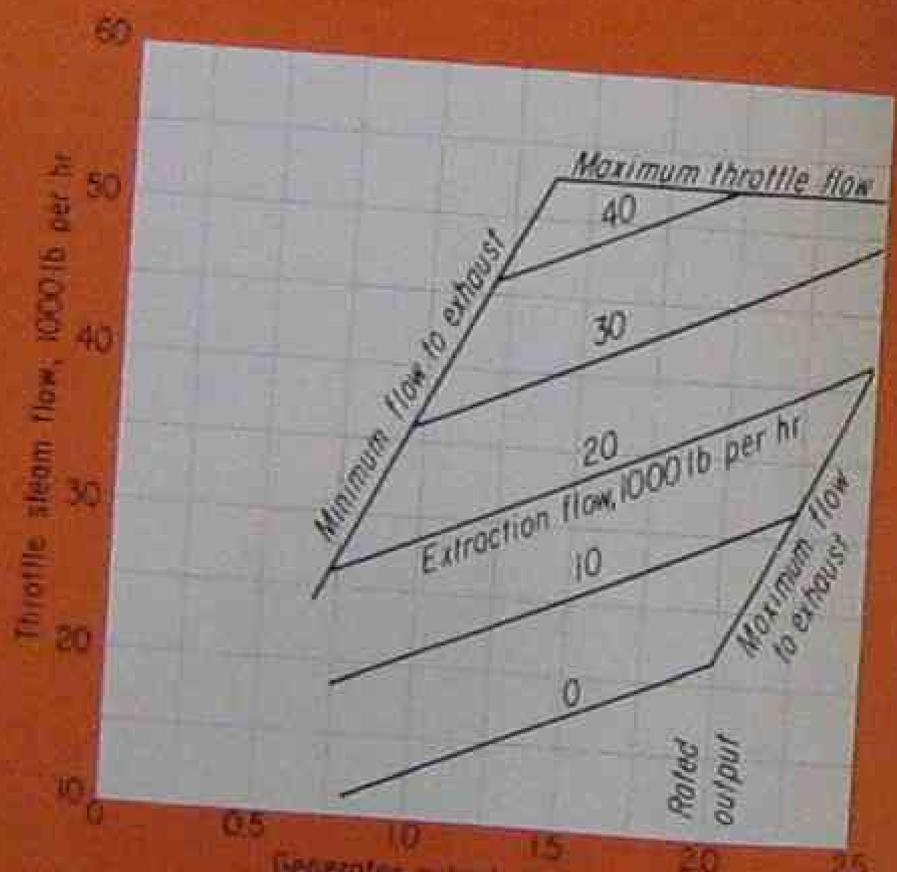
Fig. 21 shows performance curves for a condensing double-automatic-extraction turbine. The second extraction opening needs an additional family of curves on the chart. Two more limits must be considered: *minimum* and *maximum flow to intermediate section*. These depend respectively on cooling-steam-flow need and maximum steam-flow capacity of the intermediate section ahead of the turbine exhaust section.

POWER gratefully acknowledges the willing assistance of engineers of the many steam-turbine manufacturers, who patiently assembled basic data for this report.

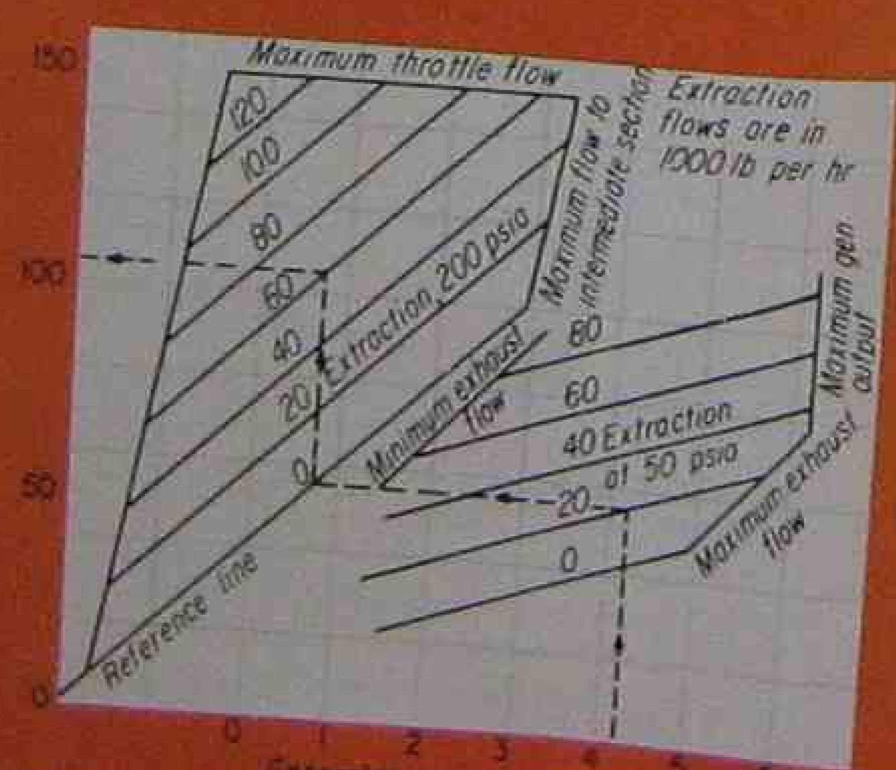
M-d and auto-extraction units show these typical performance curves



19 Performance of single- and multi-stage unit: 400 psig, 650 F, 150 psig



20 Single-automatic-extraction condensing turbine generates rated 2 mw at 600 psia, 800 F, 200 psia, 2-in. Hg abs



21 Double-automatic-extraction cond[illegible]e generates rated 5 mw at 850 psia, 800 F, 2[illegible], 1-in. Hg

Fuels and Firing

By L N ROWLEY, *Managing Editor*, J C McCABE and B G A SKROTZKI, *Associate Editors*

From those dawn times when our hairy ancestors squatted about a flame before a cave, fire has been an intimate associate. It gave cozy warmth and kept back the creeping dark. It smelted ores, made metal malleable for forging. In later days it generated steam and pushed mightily on engine pistons.

No wonder the ancients viewed fire as one of the "elements," along with the other cornerstones of their world—earth, air and water. No wonder fire has been and still is worshipped by savage peoples. No wonder traces of that age-old feeling appear in our sentimental appreciation of a glowing hearth.

Down through the centuries, as fire came ever to play a greater part in man's life and work, the nature of "burning" remained secret. Not until the classical experiments of Lavoisier, Priestley and other 18th-century scientists swept aside the mystical ideas of the Middle Ages did the process stand revealed as a reaction with oxygen from the air.

From that time on we have gained steadily in our knowledge of burning. Learned that it is more complex chemically than we first thought. Learned that good combustion is mainly a mechanical problem. Learned many practical things about fuels and efficient firing. Learned above all how little we know.

Today the reaction by which fuel and oxygen combine and give off heat is still the most important single process in our universe. Heat is the power engineer's stock in trade and combustion a process he lives with daily. As fuels become ever more expensive to obtain, and as energy demands skyrocket, nothing is more necessary than an understanding of the manner in which fuels burn and how to put them to work with peak effectiveness.

To aid that understanding is the aim of this special section, POWER's 70th. In 48 practical pages, it tells how fuels burn, gives facts about commercial fuels, describes firing equipment, shows how to select fuels wisely for new plants and old, and outlines short-cut methods for checking efficiency.

P W SWAIN, EDITOR

DECEMBER 1948 POWER

How Fuels Burn

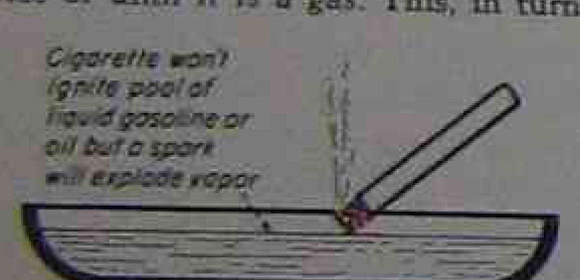
All around us, at all times, oxygen combines with other elements. Iron and oxygen combine to form the *oxide* we know as rust. Silver tarnishes, copper takes on a soft green oxide coat. This general process is *oxidation*. Burning, or *combustion*, is a special form of oxidation. In it, oxygen combines rapidly with materials we call *fuels* and substantial amounts of heat are liberated.

Under some conditions, combustion may be self-starting. For example, coal piled outdoors combines slowly with oxygen in the air, giving off heat. If the heat doesn't get away, temperature rises, the reaction speeds up and eventually becomes rapid enough to be called burning.

How Burning Starts.

Such *spontaneous combustion* is relatively uncommon. We are more familiar with combustion that begins with heat applied from some outside source. A fire laid in the grate will not light itself. Nor, for that matter, will a match. We start the burning process by striking it to generate friction heat enough to set it aflame, then use the flame's heat to light kindling, and the kindling's heat to start the logs. This process of using an easily ignited object to provide heat for ignition of another is common in engineering.

We are all aware that some things burn more readily than others. In general, this depends on how easy it is to turn the particular substance into a gas, *because nothing truly burns unless or until it is a gas*. This, in turn,



Nothing burns unless or until it is a gas

depends on the nature of the substance and on its amount compared with the amount of heat available to start the process. It's easier to start wood burning than coal, and easier to ignite a twig than a log.

A candle illustrates the point well. To make wax burn, we turn it into a gas. With a wick, heat from a match flame does the trick easily; without a wick it can't be done. Why? Without a wick, the match's heat is so small in comparison with the mass of wax that it can do no more than melt a little. None is vaporized, because there just isn't heat enough. Adding a wick changes the picture.

The wick draws up, by capillary action, an amount of melted wax so small in relation to the match flame that enough heat is available to raise the wax to the temperature needed for vaporization. Once wax is vaporized,

Wick draws up melted wax by capillarity

or made like a gas, burning is relatively easy to start. Burning the wax gives off enough heat to continue the process of melting, vaporizing and igniting.

Although the wick is eventually consumed, it contributes nothing to the burning. It is a purely mechanical device to create conditions needed to start and maintain combustion. It finds its counterpart in burners, stokers and furnaces, for while combustion is essentially a chemical reaction, most practical problems of fuel burning are mechanical.

Combustion of Gases.

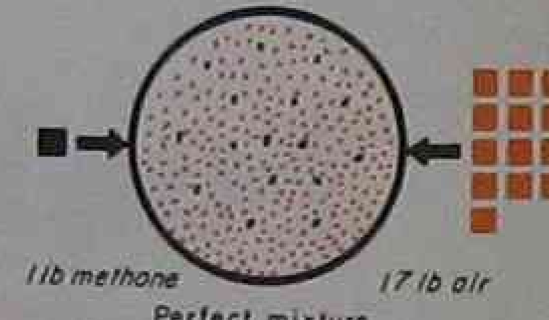
Since our fuels burn as gases, or as if they were gases, we need some acquaintance with the rules governing combustion of gases. Assume we have a gas, either a fuel that is normally gaseous, or a gas given off from a solid or a liquid fuel. In burning this gas, or any gas, it is first necessary to have a gas-air mixture that will ignite and then to raise this mixture to the ignition temperature and hold it there. Let's look first at this business of combustible mixtures.

To start with, we are working with two basic fuel elements—hydrogen and carbon. Sulphur and some other elements burn and give off heat, but common practice considers the reactions negligible and the elements impurities. Hydrogen is normally a gas and can only be liquified at an extremely low temperature—more than 400 F below zero. On the other hand, carbon is a solid that cannot be completely vaporized at temperatures much under 6300 F. Heating values are high: 62,000 Btu per lb for hydrogen and 14,500 Btu per lb for carbon.

Carbon and hydrogen exist in almost innumerable combinations called *hydrocarbons*. Many of these compounds, like methane (CH_4), are normally gaseous and form a major part of most important fuel gases.

Air Required.

Although the form in which these two elements appear in any actual fuel is vastly important to its burning, as we will see later, it makes no difference from the chemical balance-sheet angle. The important thing here is that each element or compound needs a certain amount of oxygen for complete combustion. It thus becomes easy (see p 114) to figure exactly how much oxygen we must have to burn a given amount of gas and, further, just how much air we need. If we mix exactly this much air with the gas we have a *perfect* or *theoretical* mixture. The relation be-



Perfect-mixture amounts for air, methane

tween gas and air is called the *fuel-air ratio*. This is usually expressed by weight. For any particular fuel, this ratio can be figured from the chemical analysis.

Ignition Temperature.

Now we have a combustible gas and the exact amount of air to burn it completely. We need still another element—heat. If such a perfect mixture is heated gradually, rate of chemical combination increases until a point is reached where the reaction no longer depends on heat from an outside source and practically instantaneous combustion occurs. Lowest temperature at which this happens is the *ignition temperature*. It may be defined as the temperature at which heat is generated by the reaction faster than it is lost to the surroundings, and combustion thus becomes self-propelling. Below this point the gas-air mixture will not burn freely and continuously unless heat is supplied to it.

Actual ignition temperature depends to some extent on surrounding conditions, and the term usually refers to combustion in ordinary air. Among other factors, pressure affects ignition temperature, which is lower at high pressures.

These statements apply equally to mixtures that are not perfect, that is, to mixtures a little leaner or a little richer than perfect. But for each gas there is a point where the mixture becomes too lean or too rich for ignition and continued burning. Almost every automobile owner has cussed a flooded carburetor that produced too rich a mixture and made starting difficult. Most of us have also seen the same engine miss fire when the carburetor is adjusted for too lean a mixture. Thus, for each gas there is a range of mixtures that will ignite and continue to burn. These are called *inflammable mixtures*. Limits of this range are known as limits of inflammability.

Practical Mixing.

Thus far we have looked at this business of mixtures and burning as if the right amount of gas was mixed with the right amount of air, and the entire mixture heated and burned in one operation. In actual practice this happens only rarely. Nearest approach is in the cylinder of a gas engine. In

Gas engine has near-perfect mixture

boiler firing, and other furnace work, we meet not with a single mass of ready-made gas-air mixture but rather with air and gas streams, which must be brought together and heated somehow to start and maintain ignition. The way this mixing is accomplished, and its effectiveness, makes all the difference in the nature of combustion and its completeness.

The familiar Bunsen burner of the laboratory and the fishtail burner once used in gas-lighting systems show up these differences clearly. The Bunsen burner displays a relatively small, blue, *nonluminous* flame, while the fishtail burner shows a larger, yellow, *luminous* flame. With the same gas, why the difference?

Yellow-Flame Burning.

The answer lies in the way air and gas mix, and shows clearly how the time and place of this mixing affect the process by which the fuel elements, particularly the hydrocarbons, burn. In the fishtail burner, gas flows through a slotted opening in a thin flat stream. The dark area near the burner tip is

Fishtail gas burner shows yellow flame

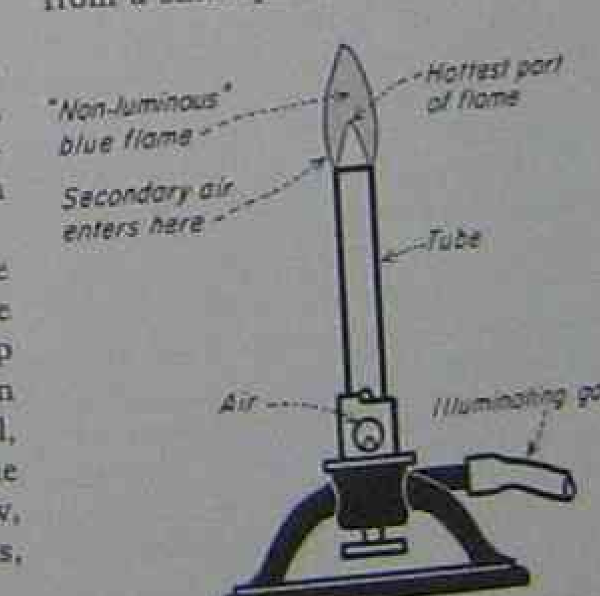
cold gas in the process of picking up and mixing slightly with air. Suddenly the mixture reaches correct proportion and starts to burn rapidly only a few hundredths of an inch from relatively cold, too-rich mixture.

Here we run into an action commonly known as "cracking" or, more fully, *thermal decomposition*. When hydrocarbons are subjected to high temperature they crack or break down chemically into other carbon and hydrogen combinations and eventually into the basic ingredients, carbon and hydrogen. This is what happens to some of the gas near the edge of the flame where heat is intense. The cracking action sets free hydrogen, which burns in the flame, and carbon particles, which glow to incandescence and make the flame yellow and highly visible.

Normally carbon burns out pretty well by the time it reaches the flame edge. But if we put any cold object in the flame, the carbon particles are chilled and the burning stops. Resulting deposit of lamp-black or *soot* is identical with similar deposits in boilers. If these microscopic particles of carbon "that didn't finish burning" stay suspended in the products of combustion, we have *smoke*.

Blue-Flame Burning.

In the Bunsen burner, gas issues from a small jet at the base of a mix-



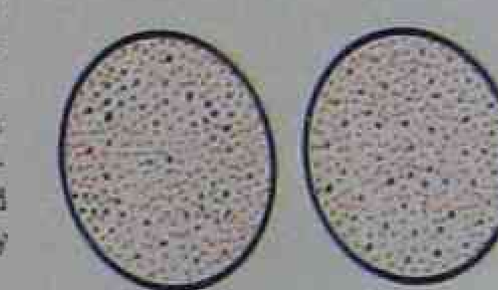
Bunsen burner operates with blue flame

ing tube. Air is drawn in through a side opening. Gas and air mix before reaching the flame area and the result is a nearly colorless "blue" flame.

The secret lies in the fact that gas and air mix before reaching the combustion zone and are heated more slowly than in the fishtail burner. This gives oxygen a chance to penetrate the hydrocarbon molecules and form oxygenated compounds. The process by which oxygen associates with hydrocarbons is known as *hydroxylation*. As will be explained more fully later, the new compounds thus formed burn with an entirely different flame. There is no cracking and no carbon is formed. Thus a change in the point of mixing, and rate of applying heat, produces an entirely different burning process.

Good Mixing Vital.

This simple example yields several ideas of great importance in this combustion business. First is the key part played by mixing. In actual flames and combustion spaces we have widely different mixtures side by side. In one area the mixture may be too rich to burn, as it was close to the tip in the fishtail burner. In other areas we may



Poor mixing versus good mixing

have almost all air and hardly any fuel. This unsatisfactory situation may prevail even though the average relation of fuel to air over the entire space may be correct for good burning. By evening out these inequalities, thorough mixing makes the actual furnace condition more nearly like the "statistical" average.

The example also shows what happens when carbon breaks away from its compounds. Microscopic particles formed are carried along in the stream with little or no motion relative to the air. This makes them extremely difficult to burn even if the process is not stopped, as by chilling.

Finally, and perhaps most important, we see that combustion is not quite so simple as it seems. It is true that we start with only two fuel elements, and that by simple equations we can show that they combine with oxygen to form water, carbon dioxide and, under some conditions, carbon monoxide. This is much like saying that two men traveling from New York to Chicago have identical journeys. Their experiences would depend on how they went, and by what route, and only by knowing these could we tell something about their trips.

Burning Hydrocarbons.

Burning of gaseous hydrocarbons is a good case in point. Complete analysis