CORRESPONDENCE TUITION SCHEME

COURSE FOR POWER PLANT OPERATORS

LESSON 6 - THE DEVELOPMENT OF THE MODERN STEAM TURBINE

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1 SELECTION OF SIZE AND TYPE OF TURBINE

1.1 Advantage of Large Output

The main advantage to be gained from increasing the output of steam turbines is a saving in capital cost. It is invariably true that a 60 MW turbine costs less than two 30 MW turbines and this rule seems to apply to all sizes without limit. This is mainly because doubling the output does not double the quantity of material, or the labour involved in manufacturing and erecting, and many ancillary items have to be included whatever the output of the set.

At the same time, for a given set of steam conditions, doubling the output means doubling the steam flow and hence the blade height and the centrifugal stress in the last row of blades becomes a limiting feature. This can be overcome to some extent by introducing multi-flow low pressure cylinders, but even so the stress in the last blade row before the L.P. cylinder imposes some limit. The problem is partially alleviated by the steam which is bled from the turbine, although this is not the primary object of bled steam.

When the pounds per second of steam entering the turbine can no longer be increased, the heat per pound of steam must be increased in order to obtain a greater output, that is, by raising the pressure and temperature of the steam. This also increases the efficiency of the whole boiler-turbine cycle, enabling a greater proportion of the heat input to the boiler to result in useful work.

Higher steam pressures, however, mean thicker cylinder casings, and higher temperatures mean more expensive material, so that the cost of the turbine is raised by these increases. Nevertheless in terms of increased output and decreased running costs such measures are well worthwhile. Higher steam conditions lessen the initial volume per pound of the steam and the possibility arises of the high pressure blades being too short to be efficient. For this reason high steam conditions are best applied to large machines.

1.2 Speed Limitations

The speed of a turbo-generator is determined by the system frequency and the number of poles on the rotor. In a 2-pole machine there will be one complete cycle per revolution, so that, for a frequency of 50 cycles, the speed must be $50 \times 60 = 3{,}000$ r.p.m.; similarly a 4-pole machine would give two complete cycles per revolution, so that, for the same frequency, the speed must be $\frac{50 \times 60}{2} = 1{,}500$ r.p.m. Since a generator cannot have less than two poles the maximum speed is 3,000 r.p.m. Further, since a gearbox for any but the smallest turbo-generators would be prohibitive in cost, the turbine is connected directly to the generator rotor.

From the point of view of the turbine designer, the greater the speed the cheaper and more efficient the machine becomes and therefore 3,000 r.p.m. is the standard speed adopted. There are some 1,500 r.p.m. machines, such as the large sets of pre-war design and the low pressure turbine and generator of the very few cross-compound machines.

The designer always endeavours to obtain the maximum exhaust area without excessive centrifugal stress, so as to reduce the unused kinetic energy in the exhaust steam.

In the case of the radial flow turbine the two shafts rotate in opposite directions at either 1,500 r.p.m, or 3,000 r.p.m. giving a relative speed between the blades of 3,000 r.p.m. or 6,000 r.p.m.

1.3 Turbines using Reheated Steam

The extent to which the initial pressure and temperature of the steam can be raised is limited. The temperature can be raised to 1,050°F. without necessitating the use of expensive austenitic steels and at this temperature a pressure of 1,800 p.s.i.g. would give an exhaust wetness of 12%, which is about the maximum permissible if serious blade erosion is to be avoided.

Since considerable gains in output can be achieved by reheating the steam after it has done work in the H.P. turbine, all new units for conventional stations use the reheat cycle. Steam from the H.P. cylinder exhaust is returned to a reheating section of the boiler and returns at a similar temperature to that at the H.P. inlet.

Reheating the steam provides three advantages:-

(a) The efficiency of the plant is increased.

(b) The wetness at the exhaust end of the turbine is considerably reduced.

(c) A greater output can be obtained from a given size of machine.

The main disadvantage is that the steam in the pipes to and from the reheater provides a store of energy which can cause the turbine to overspeed in the event of sudden loss of load. To prevent this, interceptor valves which work in conjunction with the main stop valve, are fitted at the reheat steam inlet to the I.P. cylinder.

Other disadvantages are more complex pipework, increased capital cost and operational complications.

1.4 Turbines for Nuclear Stations

Reactor technology is still in its infancy and so far the highest steam conditions that can be produced in a nuclear station are very much lower than in a conventional station. In particular the temperature relative to the pressure is low by normal standards, causing the wetness at the L.P. exhaust to be unduly high. Consequently blading erosion is expected to be more severe.

The problem can be overcome to a limited extent by using two supplies of steam at different pressures and temperatures, thus making better use of the heat from the heat exchanger. The lower pressure steam is admitted at a suitable point in the turbine, where it mixes with the steam which has passed the high pressure stages.

Additional variable speed turbines may be required, either to drive the gas circulators directly, or to provide a variable frequency supply to synchronous motors driving the reactor gas coolant circulators.

2 BASIC TURBINE DESIGN CONSIDERATIONS

2.1 Emergency and Governor Valves

Steam entering a turbine has to pass at least two valves. The first, the emergency stop valve, is opened either by hand or hydraulically and closes automatically if an emergency condition should arise. It is also used for shutting down the turbine. The second, the governor valve, is completely under the control of the governor and is used for regulating the speed or load of the set.

On all but the smallest sets there are two or more such valves in parallel so that they can control a large quantity of steam without becoming too large. It is essential that these valves operate extremely rapidly, as any delay may cause the rotor to overspeed to a dangerous extent once the load is lost. An emergency stop valve may take ½ of a second to close fully from the time the tripping signal is received.

On reheat sets interceptor valves have to be provided where the reheated steam returns to the I.P. turbine, in order to prevent steam trapped in the reheater from accelerating the rotor to a dangerous extent. These valves deal with a lower pressure and greater volume of the steam are considerably larger than the H.P. emergency stop valves.

Similarly the dual pressure sets used in nuclear power plants have two sets of emergency and governor valves.

Control valves used on turbines are generally of three types: double beat, single beat or flap. Fig. 1 shows some designs of valves used for modern turbines:-

(a) is a "double-beat" valve having two seatings, the object being to balance the forces due to steam pressure. It is suitable for most pressures but not for high temperatures, as differential expansion between the valve and cage would cause one or other seating to "weep".

(b) is another double-beat valve of the hollow type, in which the steam from one seating is led through the centre of the valve. The thinner walls promote even heating and lessen differential expansion:

(c) is a modern spherical valve, used for controlling high temperature steam. Being a "single-beat" valve, with one seating, the pressure forces are not balanced, and a large operating force is required:

(d) is a similar valve fitted with an internal pilot valve, which by opening first equalises the pressures, and provides initial fine control:

(e) is a cylindrical valve, in which steam pressure is prevented from acting on the back of the valve by a fine annular clearance:

(f) is a governing valve of the "mushroom" type

(g) is a flap valve, used for reheat emergency valves, where the steam pressures are moderate, and the specific volumes (and, hence, the valve diameters) are large.

A double beat valve has the advantage of being balanced as far as steam thrust is concerned, so that only a relatively small force is required for its operation. On the other hand it has two seats, which due to expansion or wear may not both seat properly.

A single beat valve has only one seat, but requires powerful relays and return springs for its operation if the steam pressure is high. This valve is often spherical in shape, with a loosely mounted foot to ensure tight closure. Sometimes there is a small pilot valve which opens first and equalises the steam pressure before the main valve opens.

Interceptor valves are sometimes of the flap type, in this way they may have a large area without a correspondingly large spindle leakage.

When valves are only slightly open, the steam leaks through with a high velocity and wears grooves in the seating. This is known as wire-drawing and causes the valve to weep. It is therefore bad practice to open valves only slightly.

Governing valves are contained in a steam chest which may be an integral part of the cylinder casting or a separate entity at the side of the turbine. The latter is the more common design, due to the difficulty of making a sound high temperature steel casting incorporating an integral steam chest. Separate steam chests are connected to the turbine by long loop pipes to take up expansion movements.

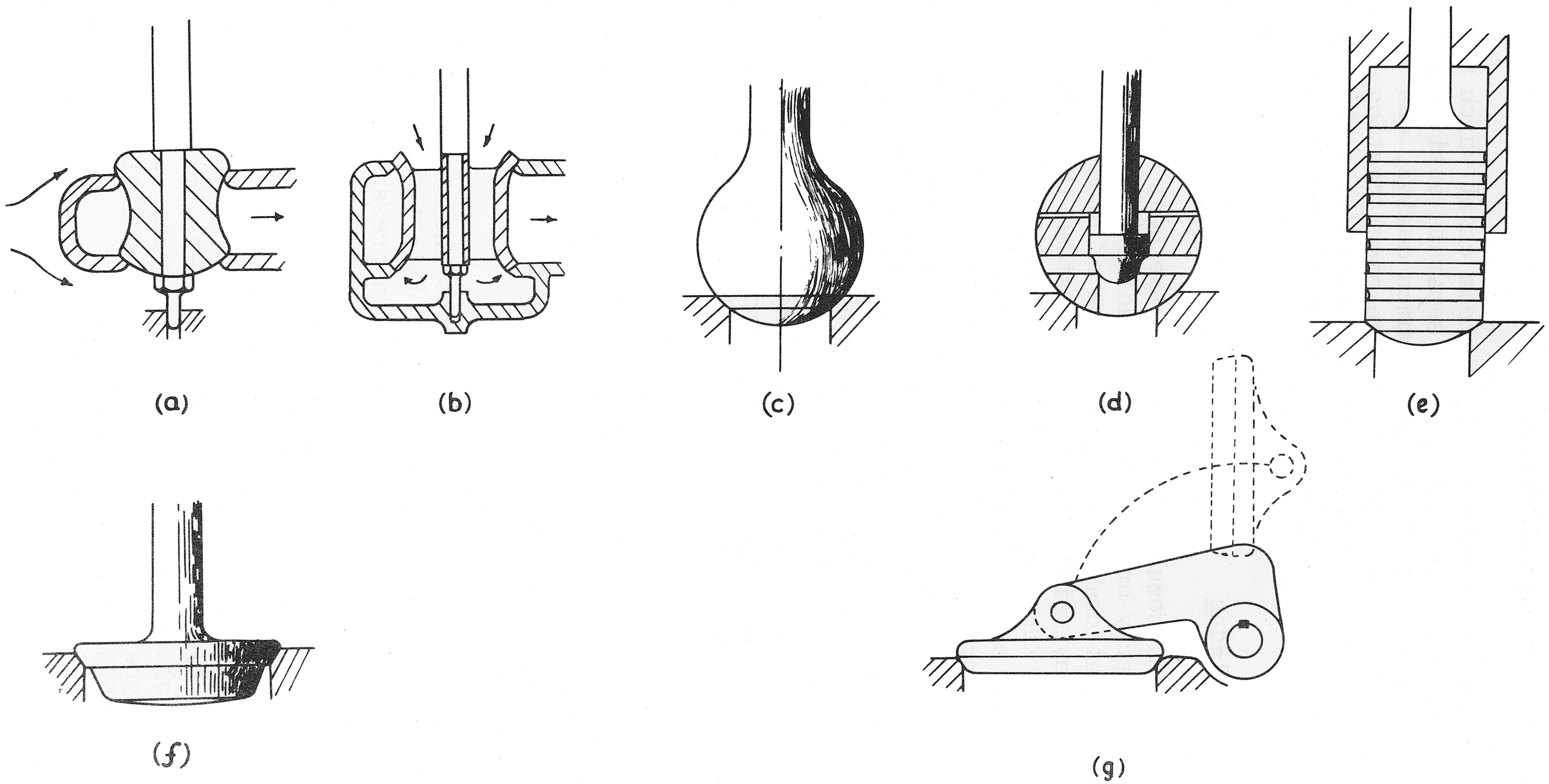

Figure 1: TYPES OF STEAM VALVE

2.2 Nozzles and Blades

The function of a nozzle is to convert the heat energy of the steam into kinetic energy or motion. In blading of the impulse type the nozzle remains stationary and directs jets of steam on to the moving blades. In reversing the path of the steam the moving blades or buckets absorb the kinetic energy of the steam, and the rotor turns. As the steam passes through the nozzle it loses pressure and total heat, but in passing through the moving blades the pressure remains constant and the steam merely gives up its kinetic energy.

In blading of the reaction type, both stationary and moving blades act as nozzles and buckets, so that there is a pressure drop across each. Propulsion is therefore effected partly by impulse and partly by reaction as in jet propulsion. Where these effects are equal the process is known as 50% reaction and the moving and fixed blades are then identical in profile. This lends itself to easy manufacture.

Although this blading is usually loosely termed reaction type blading, pure reaction (100%) implies that all the pressure drop takes place in the moving blades. While this is perfectly feasible it is not usually adopted in steam turbines owing to the difficulty of preventing leakage past the blade tips.

In long blades it is necessary to prevent the steam being flung to the outer periphery by centrifugal force, this is done by creating, in the space between fixed and moving blades, a pressure which is lower at the root than at the tip, that is to say, by increasing the degree of reaction from root to tip. Such blading appears to have a twisted shape and is known as vortex blading.

Most modern blading is of the vortex design whether impulse or reaction type, H.P. blading is too short for there to be much variation along the blade and may be of either type. L.P. blading tends to be of one type only, with near-impulse conditions at the root and as much as 65% reaction at the tip.

It is sometimes desirable to employ a large heat drop in the first stage of the H.P. turbine, in order to limit the maximum pressure and temperature which the cylinder has to withstand. In such cases the nozzles form part of a separate nozzle chamber and the kinetic energy is absorbed by a double row of impulse blades. Such a stage is known as a velocity-compounded or Curtis stage.

The primary aim at the L.P. end of the turbine is to provide as large a leaving area as possible in order to reduce the leaving loss. Normally centrifugal stress limits this area but the limitation may be somewhat overcome by using a Baumann exhaust (See Fig. 2). This involves adding a further "tier" to the penultimate stage, where extra blade height can be accommodated within the allowable limits of stress. This outer tier exhausts directly to the condenser, thus providing a larger effective area and smaller leaving loss.

2.3 Turbine Casings

A turbine casing has to withstand the pressure of the internal steam and must also form a stiff beam resting on the pedestals, so that the stationary blading is accurately aligned. When the steam pressure is very high a double shell construction is used, giving adequate strength without resorting to very thick metal sections which would take a long time to warm up.

Cylinders may be arranged for single flow, double flow or reversed flow, the latter being used on some H.P. turbines to centralise the hottest parts and simplify expansion. (See Fig. 3)

It is essential that H.P. and I.P. cylinders be efficiently and evenly lagged, otherwise besides heat being lost, serious distortion may occur.

A turbine with impulse type blading is divided into a number of stages at different pressures by diaphragms located in grooves. Diaphragms are not necessary for reaction blading since the pressure drop per stage is small and half of this pressure drop occurs across the moving blades.

Cylinder flanges impede quick warming up by reason of their large mass and can cause transverse distortion. This can be avoided by warming the flanges with steam led through special passages during the warming-up period.

L.P. casings are extremely bulky in order to provide a sufficiently large exhaust area for the low density exhaust steam. They may be made of cast iron, or fabricated from steel plate.

H.P. and I.P. casings are cast in steel, the composition depending on the maximum temperature. Alloy steel containing chromium, molybdenum and vanadium is used for temperatures up to l,050°F., and carbon steel is used for more moderate temperatures.

2.4 Shafts Couplings and Bearings

A shaft of a turbine may be constructed in one of the following ways (See Fig. 4):-

(a) The hollow drum rotor, used for high temperature reaction bladed H.P. turbines or I.P. turbines after reheating.

(b) The solid drum rotor, used instead of the hollow drum where quick warming to a high temperature is not necessary.

(c) The built-up rotor, consisting of discs shrunk on to a shaft, used mainly for L.P. rotors, and for moderate temperature H.P. or I.P. rotors. If used with higher temperatures there is a danger of discs expanding and becoming loose, but at low temperatures it enables discs of very high strength material to be used.

(d) The welded disc rotor, used in some cases for reaction type L.P. rotors. It eliminates the need for large forgings, but requires very careful welding and subsequent stress relieving.

(e) The integral disc rotor, used for high temperature H.Po rotors with impulse blading, and in many cases for I.P. and L.P. rotors as well. A large forging is required which is difficult to make and to inspect. Much labour and waste material is incurred in machining the discs from the solid. If sound, however, such a rotor contains relatively low stresses and is very reliable.

A turbine shaft is located axially by a thrust bearing and expansion takes place away from this point. If the rotor and casing at each section are at the same temperature and have the same coefficient of expansion, then whether in a hot or cold condition the axial blading clearances will be the same. If however, during start-up, load-changing or shut down, temperature differentials alter due to different rates of heat absorption, the axial blading clearances will change. It is at such times that there is a danger of blade rubbing, and this is a major factor in limiting the rate at which turbines can be started and loaded.

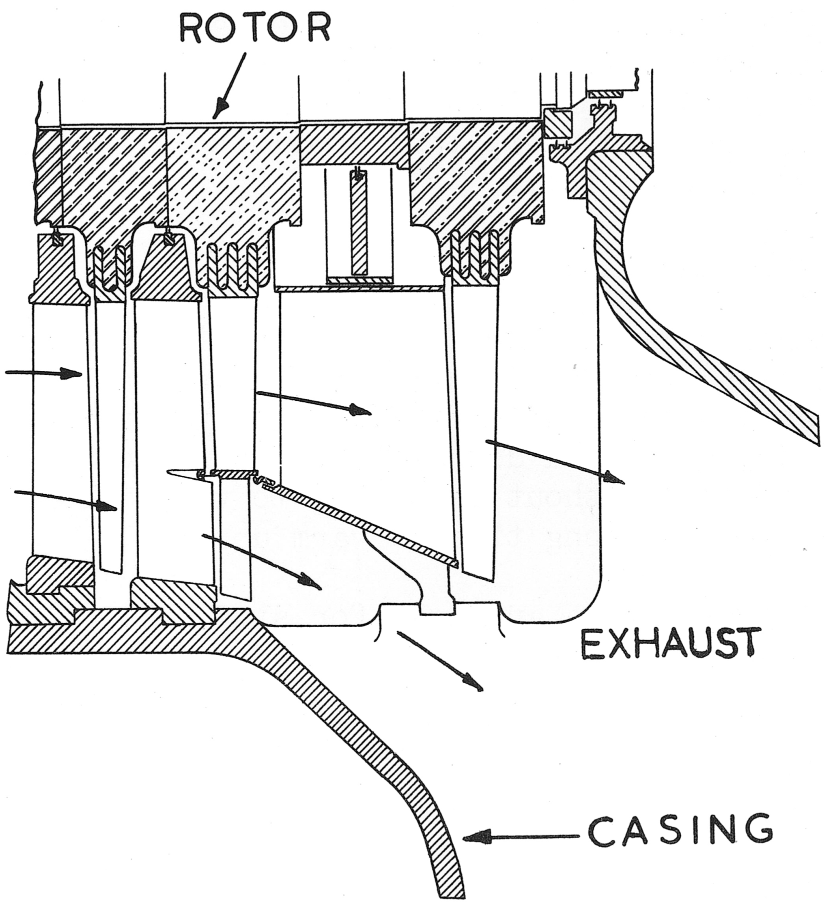

Figure 2: MULTI - EXHAUST

Formerly turbine rotors were connected by means of flexible couplings, with the idea of ensuring that each would expand independently from its own thrust bearing. Such couplings, however, would only permit sliding when lightly loaded and therefore unless loading was delayed while expansion took place, coupling lock and "snatching" was liable to occur. They were effective, however, in accommodating minor angular misalignment.
The higher steam temperatures and outputs of modern turbines have led to all shafts being solidly coupled. They are located from a single thrust bearing placed as near as possible to the hottest part of the shaft in order to minimise the possibility of blade rubbing. On reheat machines the thrust bearing is usually placed between the H.P. and L.P. turbines, the H.P. turbine being arranged with its exhaust at the governor end.

Solidly coupled shafts combine in effect to form a single rotor and alignment must be very accurate. It is essential that the foundations should be free from differential settlement and that the supporting structure under each bearing should be extremely rigid.

Bearings are invariably of the plain journal type, with cast-in white metal linings. (See Fig. 5). Usually each rotor is supported by a bearing at each end, but in some cases there is only one bearing between two cylinders. The centre lines of the bearings are aligned to the natural curve which the coupled shaft assumes under gravity. In accordance with the general principles of bearing lubrication, the shaft when running rests on a wedge of high pressure oil and no wear should take place. At very low shaft speeds this wedge is dispersed and wear can then occur. Before putting the turbine on the turning gear, the heavy L.P. shafts are lifted by high pressure oil supplied from the jacking pump.

Thrust bearings on the steam turbines are always of the Michel type, incorporating tilting pads which set up an oil wedge in front of the thrust collar. A second set of surge pads is provided on the other side of the collar in case abnormal steam conditions cause the direction of thrust to be reversed. Normally the shaft is arranged so as to be in tension.

3 DESIGN OF ASSOCIATED EQUIPMENT

3.1 Governor System

The function of the governor is to control the: speed of the turbine when the generator becomes disconnected from the grid and the electrical load when it is connected to the grid, in which case the speed is tied to the frequency of the system. It is the combined effect of all the governors of the sets on the system which helps to maintain the frequency at any instant until loading correction is made effective by the system control engineer.

Governors may be mechanical, hydraulic or electric. normally those on the C.E.G.B. system are mechanical. When a turbine is run up to speed, the governor takes over control when the speed is about 6% below normal and thereafter automatically keeps the speed at the desired value. This value can be adjusted from the control room by means of speeder gear.

The governor operates on the principle of a spring balance in which the centrifugal force of two revolving weights acts on a spring, the extension of which controls the governing valve power relays. (See Fig. 6). If the speed rises, the centrifugal force on the weights increases and they move slightly outwards against the spring, this motion is transmitted to the power relays and causes the governing valves to close. In this way the rate of entry of steam to the turbine is reduced and the speed falls back to its proper value.

The governor should be very sensitive. The weights are designed to be as small as possible and they operate the smallest possible number of moving parts. Relays are used to amplify the power provided by the weights. A primary relay adjacent to the governor provides power for moving the control linkage and power relays coupled to each valve provide power to open the valves against the combined spring and steam pressure.

Fig. 7 shows a typical relay. The pilot valve is normally in the "shut-off" position but is displaced when a change of position of the control rod occurs. Oil is thereby admitted to the cylinder, causing the piston to move against the spring. In doing so it resets the pilot valve to the "shut-off" position and moves a distance proportionate to that moved by the control rod.

A number of turbines are fitted with low vacuum unloading gear and low steam Pressure unloading gear. Low vacuum may cause the low pressure blading to overheat and low steam pressure may be accompanied by dangerous slugs of water entering the turbine. Therefore when a certain value of vacuum or steam pressure is reached the turbine will automatically unload itself gradually as the conditions deteriorate further down to a minimum load of about 10%.

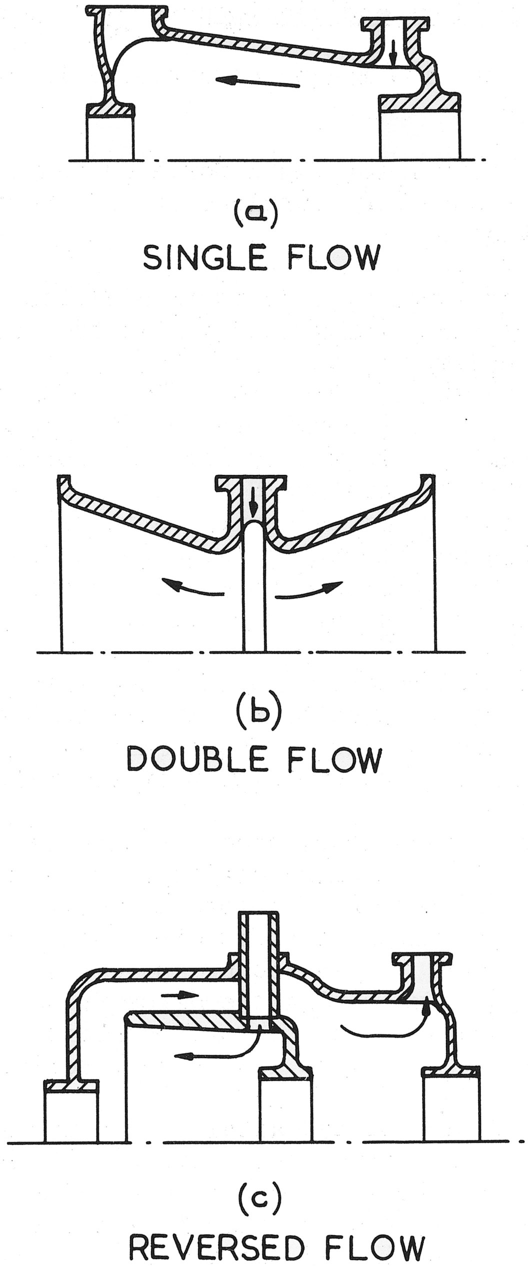

Figure 3: CYLINDER FLOW ARRANGEMENTS

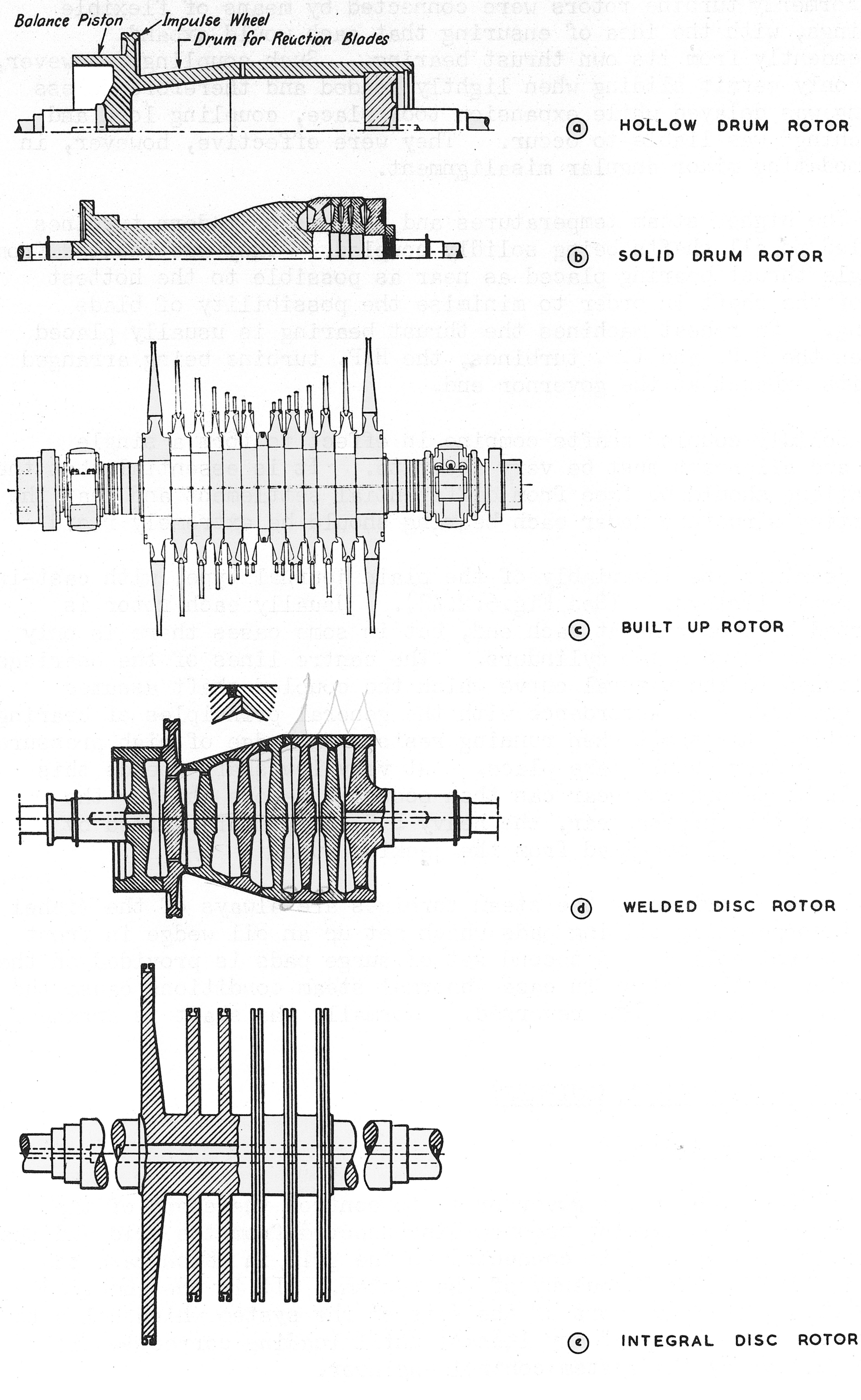

Figure 4: TYPES OF TURBINE ROTOR.

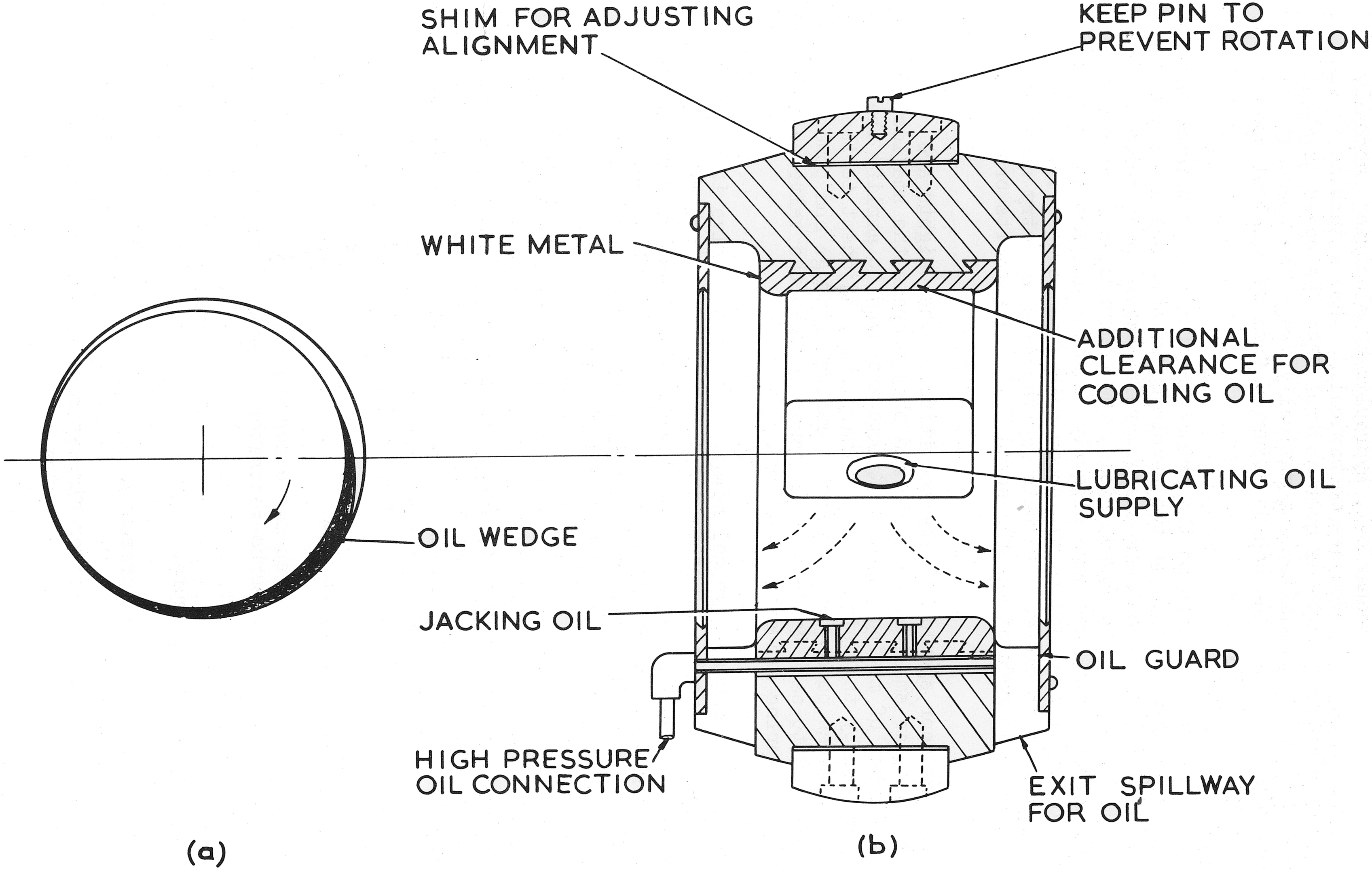

Figure 5: JOURNAL BEARING

3.2 Emergency Tripping System

A number of contingencies may occur which necessitate the immediate shut-down of the turbo-generator. Among such contingencies are

- Low relay oil pressure.
- Low lubricating oil pressure.
- Opening of main generator circuit breaker by its electrical protection equipment following a fault.
- Overspeed in excess of 10%.
- Low vacuum beyond the range of the unloading gear.
- Low steam pressure beyond the range of the unloading gear.
- Thrust bearing failure.

An automatic tripping system ensures that if any of these contingencies occur, emergency stop valves shut off the steam supply and the generator circuit breaker is opened. Other possible contingencies are catered for by a push button in the control room and a lever on the turbine governor pedestal, either of which can actuate the tripping gear.

The various detecting devices each cause the relay oil pressure to collapse if the contingency arises. The emergency stop valves are held open by relay oil pressure against a strong spring and failure of this pressure causes immediate closure. The generator circuit breaker relay and the control room push button collapse the oil pressure by means of a solenoid operated drain valve. Counterwise an oil pressure switch energises the generator electrical tripping circuits, whenever the relay oil pressure ceases to be maintained.

Emergency stop valves are generally opened manually and are used for starting up the set. On reheat machines additional interceptor valves are necessary to prevent steam remaining in the reheater pipes from accelerating the rotor to a dangerous degree.

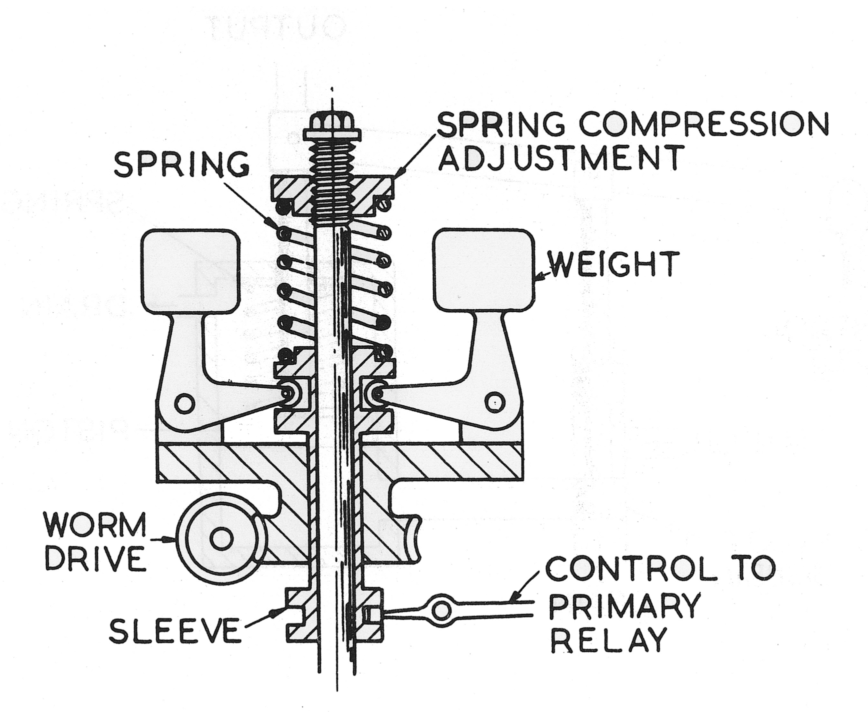

Figure 6: CENTRIFUGAL GOVERNOR

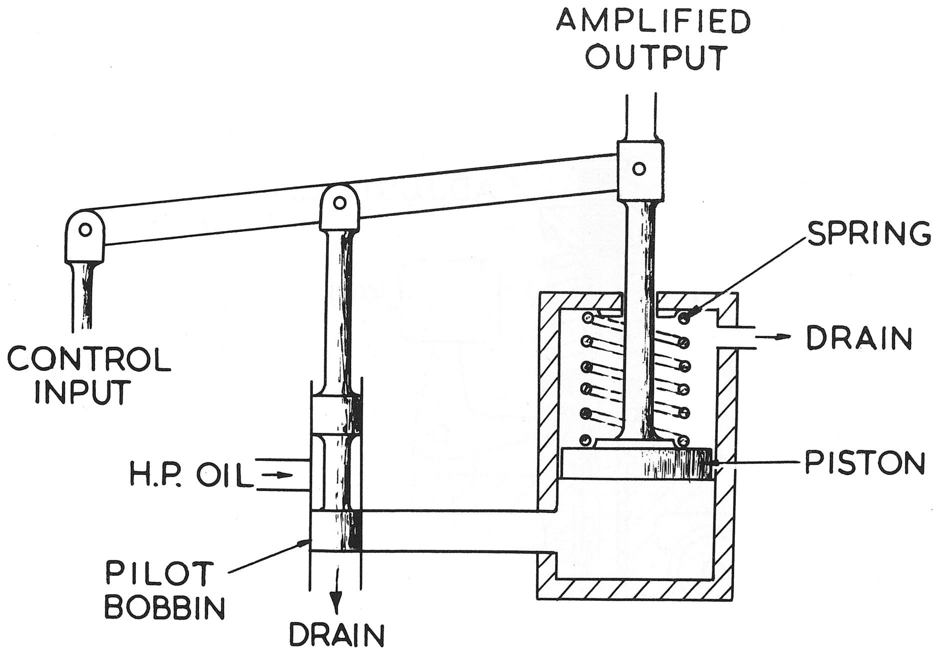

Figure 7: SINGLE-ACTING RELAY

3.3 Lubrication System (See Fig. 8)

To provide continuous lubrication for a single rotating shaft seems a simple enough problem, yet when every eventuality has been provided for, the system becomes complicated.

The main oil circuit consists of:-

1. The main oil tank, large enough to enable the oil to rest for several minutes to allow time for defoaming and settlement of solid particles.
2. The main oil pump, driven directly from the turbine shaft and generally providing the full quantity of oil at high pressure. It may be of the gear type, which requires reduction gearing from the main shaft but no priming, or of the centrifugal type, which is mounted directly on the shaft and needs a priming device since it does not create a suction lift.
3. The valve chamber(s), containing pressure reducing valves which provide high pressure oil for the governing and tripping circuits, and low pressure oil for bearing lubrication. It may also contain bypass relief valves and non-return valves.
4. The oil coolers, which are water-cooled heat exchangers and reduce the temperature of the oil to that required at the bearings.
5. The thrust and journal bearings themselves.

Oil is returned from the bearings by gravity through large diameter pipes to accommodate the foam. A plentiful supply of oil is required to remove heat from the shaft.

A separately driven auxiliary oil pump is required for use when the turbine is starting and stopping, Or in emergency if the main pump fails. It may be steam driven, but nowadays it is usually A.C. electrically driven, as the mains supply has a greater availability than the steam supply. As a further safeguard there is generally a D.C. electrically driven flushing pump, supplied from the station batteries and providing lubricating oil only.

If a centrifugal main oil pump is used, it must be provided with means of priming. Normally an ejector is fitted, which makes use of a small quantity of high pressure oil to lift a larger quantity of oil from the tank to the pump. When starting, the high pressure oil comes from the auxiliary pump but thereafter it is bled from the delivery pipe of the main pump.

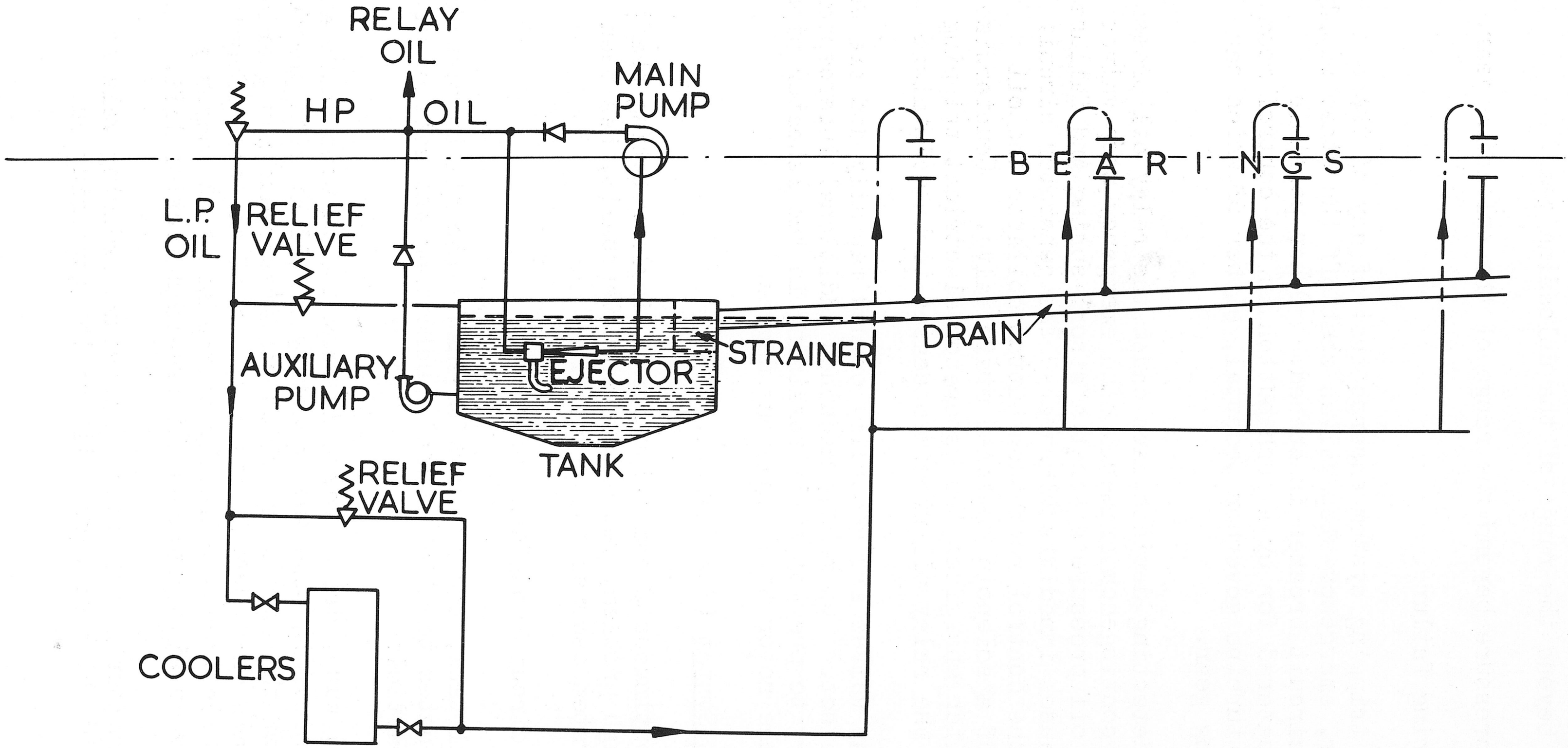

Figure 8: TYPICAL LUBRICATION SYSTEM

3.4 Gland System

It is impossible to completely prevent the leakage of steam through the clearance space where the rotor passes through the end of each cylinder. However by carefully designed glands the loss may be reduced to a minimum.

Glands invariably have packings of the labyrinth type, Which provide a number of fine clearances in series. As the steam leaks through these clearances it is throttled, that is to say, its pressure falls and its velocity increases. This velocity is largely lost due to the eddying motion of the steam in the spaces between the fine clearances. Thus after passing through each clearance the pressure is reduced and less leakage occurs. The number of clearances or packing rings required depends upon the steam pressure.

To reduce the clearance area, glands are made with a diameter as small as possible and clearances as fine as possible. The diameter is limited by considerations of shaft strength and the radial clearance, by the clearance within the bearings and by possible shaft distortion.

Glands must allow for axial expansion of the shaft and casing to take place without causing a rub. Further, if a rub should take place due to shaft vibration, it is desirable that the heat generated shall be minimised in order to prevent serious frictional heating of the shaft and possible distortion. A typical modern gland comprises stationary fins on spring loaded sectors, the shaft merely being smooth or castellated. If a rub should occur the sectors relieve the generation of heat and can be replaced readily if damaged. (See Fig. 9(a)).

Other designs of glands consist of fine fins which allow large axial displacement and staggered fins, designed to bend if a rub occurs so as to relieve the interference automatically. (See Figs. 9(b) and (c))

At suitable points along the H.P. and I.P. shaft glands the leakage steam may be led away to a point further down the turbine, where it does a certain amount of useful work. The final leakage steam which is at a pressure just above atmospheric, is led to the L.P. glands for sealing purposes, the residual vapour being vented either to atmosphere or to a vapour condenser vented to atmosphere.

Sealing steam is required whenever the cylinder pressure is below atmospheric such as at the L.P. glands and for other glands when at low loads. As stated above, the L.P. glands are usually sealed with leakage steam during normal running, but require a special supply of desuperheated steam for sealing the glands at low loads.

The use of sealing steam may be avoided by providing water paddle seals which seal the clearance space by means of an annular belt of water held in place by centrifugal force. Such seals, however, absorb a considerable amount of power from the shaft. (See Fig. 10)

3.5 Turning Gear

When a turbine comes to rest the residual heat tends to convect towards the upper half of the cylinder, causing the top of the rotor to cool more slowly than the bottom. This would cause the rotor to "hog" unless steps are taken to rotate the shaft until it is thoroughly cool.

To do this electrical turning gear is used, similar in principle to the starter motor on a car.

A motor drives the rotor through reduction gearing, which is engaged by means of a sliding pinion. The turning gear must also be used during the warming up period. When the turbine starts to run under its own power, the pinion is automatically thrown out of mesh to disconnect the turning gear drive mechanism.

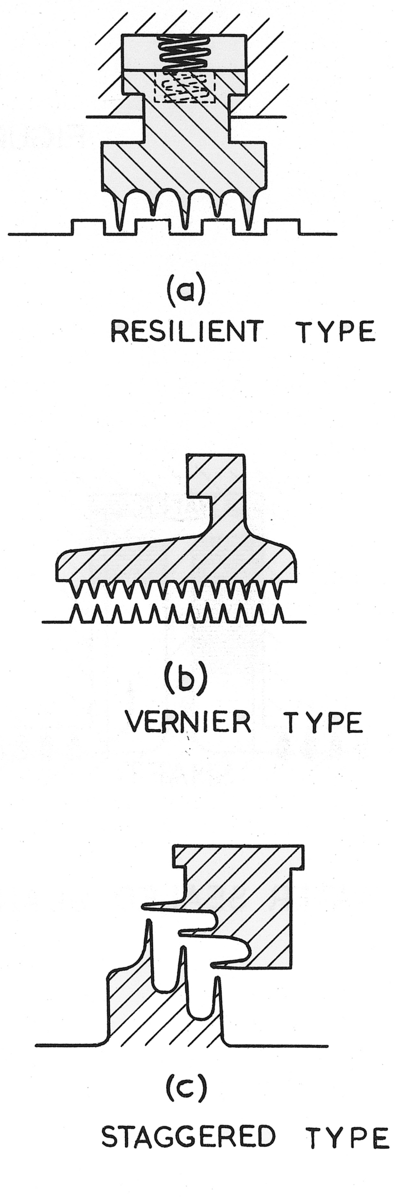

Figure 9: LABYRINTH GLAND DESIGNS

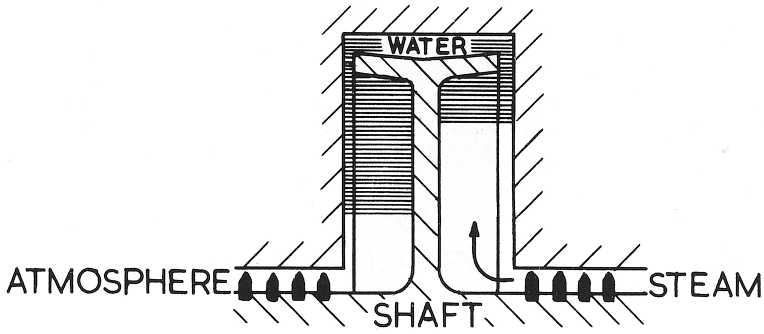

Figure 10: WATER SEALED GLAND

When the rotor is started, very high pressure jacking oil is used to slightly lift the journals of the heavy L.P. rotors by means of small orifices at the bottom of the bearings. This saves wear of the white metal during the time before a proper oil wedge has been formed and also enables a smaller turning motor to be used. The pressurised jacking oil is provided by an electrically driven reciprocating pump.

During the turning period, lubricating oil is supplied from an A.C. electrically driven flushing pump. The motor of this pump is electrically interlocked with the turning gear motor.

Turning gear is often referred to as "barring gear", since on old sets the rotor was turned by hand using a long bar and worm gear.

A very slow speed of turning is sufficient to prevent the rotor bending, but by increasing, the speed to about 30 r.p.m, sufficient turbulence is set up to prevent casing distortion as well.

4 FUTURE TRENDS

All turbines now being installed by the Central Electricity Generating Board for coal-fired stations Use the reheat cycle. Outputs range from 100 to 550 MW and efficiencies from 41 to 46% are expected.

The capital cost per KW installed has, fallen in spite of considerable increases in the cost of labour and materials. The conventional cycle, however, is approaching its full exploitation, and therefore further savings with this cycle will depend mainly on size increase. Further possible refinements are supercritical steam pressures, higher steam temperatures and two stages of reheat. These possibilities are being pursued, but their benefits may not outweigh their disadvantages.

Steam turbines for nuclear stations also continue to increase in size and it is anticipated that their steam conditions will rise as reactor techniques improve.

A possible future step may be the combined gas-steam cycle, in which the advantages of the gas turbine and steam turbine cycles are joined in one unit to give a considerable improvement in overall efficiency.

Questions on Lesson 6 - The Development of the Modern Steam Turbine

Please answer any four of the following questions

1. Why do modern turbines of large output employ reheat? What are the advantages and the difficulties?
2. Explain the fundamental difference between impulse type and reaction type blading. How are these two principles incorporated in the blading arrangements of modern turbines?
3. Describe three different types of turbine rotor construction, indicating the use of each.
4. Why is it necessary to employ a number of pumps to supply oil to a turbine. For what purpose is each used?
5. What is meant by differential expansion. What precautions does it involve?
6. Discuss the reasons for the current trend to install larger turbines.