NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment and System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-1 - Turbine Construction

0.0 INTRODUCTION

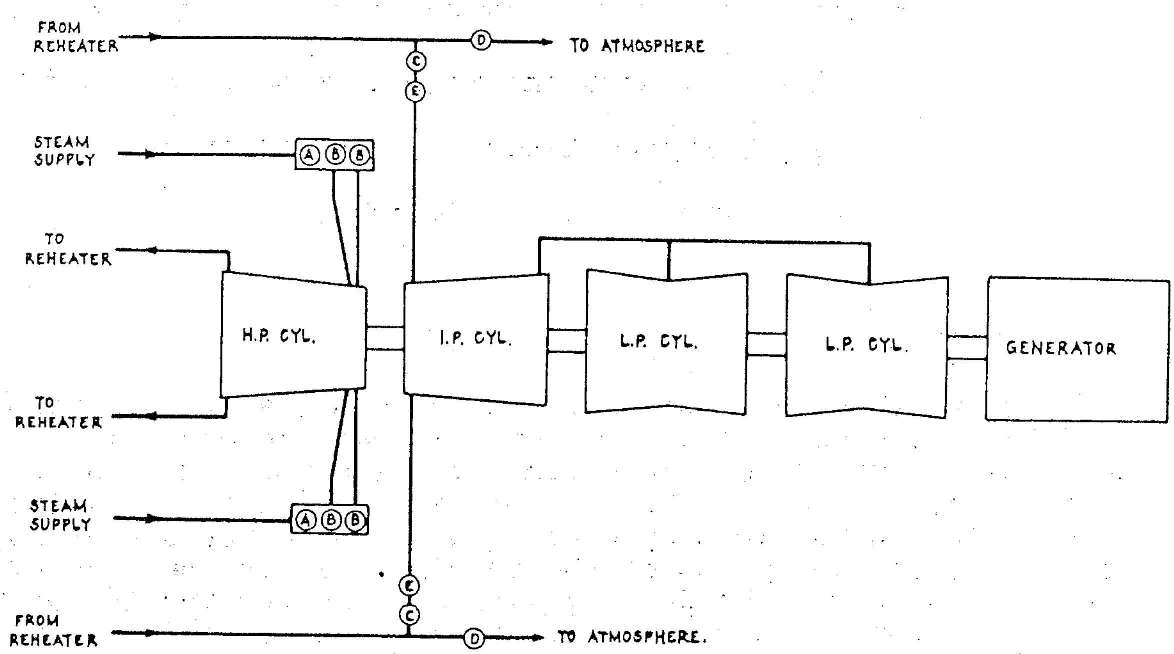
Up to this point we have discussed how a steam turbine works and the different shaft arrangements that are possible. We have also discussed one or two of the detailed construction aspects such as gland seals. This lesson will continue to develop the discussion on turbines by dealing with the construction aspects of the stationary parts of the turbine. Or another way of saying it is that we will be dealing with how the turbine is put together as far as the stationary parts are concerned. The turbine spindles will be covered in a later lesson.

1.0 INFORMATION

Reheat Piping and Valves

We have mentioned reheat tandem-compound and cross-compound units but have not said how the flow of steam is controlled in case the unit has to be shut down in a hurry. Figure 1 shows a typical arrangement of control and overspeed protective valves for a reheat turbine. For large steam turbines such as 200 MW or over, steam is supplied to the turbine through two main steam pipes which connect directly from the boiler to two cast-alloy-steel chests disposed symmetrically about the H.P. end of the turbine. Each of the main steam chests contain a combined stop and emergency valve (A), which can be operated locally or from the control room and can be tripped by the action of various automatic electrical and hand-operated protective devices.

From each chest, the steam flows to the high-pressure cylinder through two governing valves (B) arranged in parallel in each chest, then through four bore loop pipes which enter the casing radially at top and bottom. The use of separate chests and U-loops between the chests and casing ensures that the lateral stress due to pipe expansion and casing expansion will be equalized from both sides of the casing. The U shape of the pipe loops ensures flexibility of piping. Thus the thrust loads on the turbine casing due to metal expansion is minimized.



A = emergency stop valves
B = governing valves
C = reheat emergency stop valves
D = relief valves
E = intercept governing valves

Figure 1. Arrangement of control and overspeed protective valves for reheat turbine.

There are two steam lines leading to the reheater. These are called cold reheat lines. There are also two steam lines leading from the reheater to I.P. cylinder. These are called hot reheat lines.

One can readily imagine that the reheater and hot and cold reheat lines have a capacity for a large quantity of steam. If there should be a sudden rejection of load and the emergency stop valves closed, the large quantity of steam still in the reheater and hot and cold reheat lines would continue to flow through the I.P. & L.P. cylinders if there were nothing to stop it. This would produce enough work output to cause the turbine shaft to overspeed dangerously. In order to avoid this, and be able to stop the flow to the I.P. cylinder, reheat emergency stop valves (C) and intercept governing valves (E) are located in each hot reheat line near the I.P. cylinder.

The intercept governing valves are not required to control the flow of steam to the unit when the load is being increased or decreased at a steady controlled rate. They are normally fully open above 25% of full load and only begin to close after around 3% rise in speed above normal has occurred. Once closure begins it proceeds rapidly and the overall time of closure of the intercept valves is less than that of the main governing valves. As soon as the intercept valves are closed the relief valves (D) open and allow the steam to escape to atmosphere. Just as emergency stop valves are fitted at the H.P. inlet to the turbine to act as a second line of defence should the governing valves stick or for some other reason be slow in closing, it is considered advisable to reinforce the protection provided by the intercept valves by including reheat emergency stop valves just ahead of the intercept valves. These stop valves are under the control of the overspeed governor and other emergency devices available to the control room operators.

H.P. Inlet Piping and Valve Arrangement

Figure 2 shows a high-pressure, high temperature casing for a tandem -or cross- compound turbine with flexible loop pipes connecting separate steam chests to separate nozzle chambers in the turbine casing. Figure 3 a) depicts a cross-section of the turbine casing showing the way the loop pipes connect into the casing to the nozzle chambers.

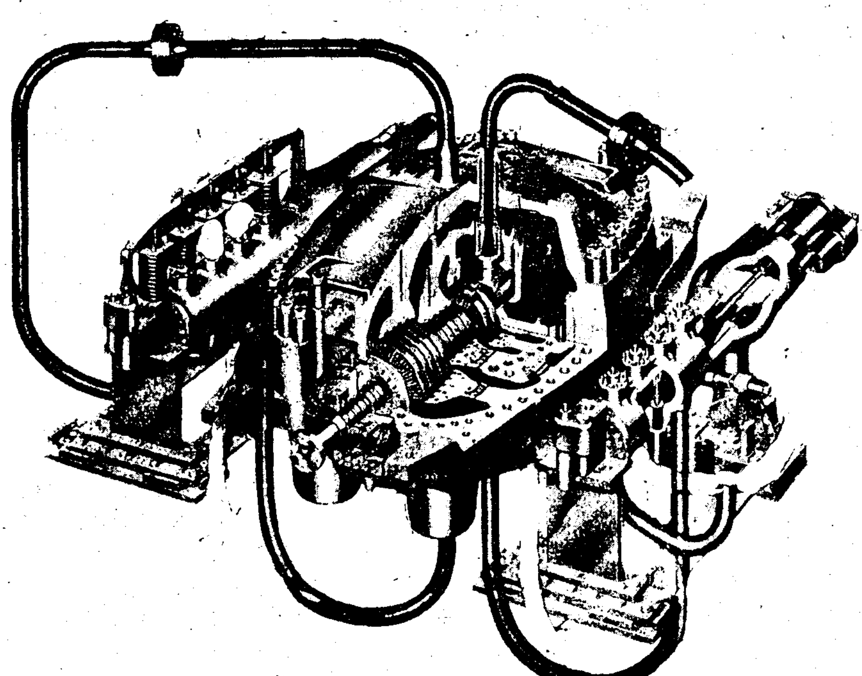


Figure 2 - High pressure high temperature casing for a tandem- or cross - compound turbine features separate steam chests with emergency stop valves. Flexible U-shaped or loop pipes connect chests to separate nozzle chambers in turbine casing.

For large units there are normally 4 governor valves and 4 inlet loops.

In steam power station practice there are two different methods of controlling steam flow into a turbine, and thus controlling work output of the turbine.

Method I Throttle Control. When a unit is expected to be operating at full load practically all the time, then all four governing valves are opened in small steps at the same rate until at full load they are all fully open at the same time.

When a valve is only partially open it throttles steam - I.E. its outlet pressure is lower than its inlet pressure but enthalpy remains the same. Therefore this method is referred to as throttle control of steam. In a throttle-governed turbine the steam flow is controlled at all partial loads by varying the pressure in front of the nozzles and all nozzle chambers are used simultaneously. This type of control simplifies the turbine governor and the inlet nozzle design and reduces stresses in the casing due to temperature differences at the inlet nozzles. However, throttling produces losses in steam energy and therefore at part load the turbine efficiency with this method is poor.

Method II Nozzle Governing. When a unit is expected to be operated at part load for considerable periods of time a sequence of governor valve opening is used which is referred to as nozzle governing. In a nozzle-governed turbine the output is governed by the number of nozzle chambers used and hence by the number of nozzles in use at partial loads. With this method only two governing valves are opened simultaneously at low loads. This two valves are called primary valves and the inlet loops are arranged so that steam will enter only two nozzle chambers 180° apart as can be seen in figure 3 a). That way the casing will be heated uniformly all around minimizing the possibility for distortion due to uneven heating. Since there are four separate nozzle chambers, and they are not interconnected, there will be no steam flow through two of them. When both valves are fully open the unit will be operating at 50% full load. At this point the nozzles in the two chambers are fully utilized.

Above 50% load the next valve, which is referred to as the secondary valve is opened step by step. The inlet loop for this valve is also labelled in figure 3 a). When this valve is fully open the unit will be at 75% full load and the nozzles in three nozzle chambers will be fully utilized.

The last or tertiary valve is required only when the unit is operating between 75% and 100% of full load. At 100% load the nozzles in all four nozzle chambers will be fully utilized. With this method only one or at the most two valves are throttling steam at one time and the nozzles are more efficiently used. Therefore this is a more efficient method for part load operation. At 100% load all valves would be fully open and no throttling would be taking place in either method. In North American practice for conventional stations, method II is generally used.

At this point two definitions are in order:

Maximum Continuous Rating (M.C.R.) - refers to the maximum output that a turbine can produce continuously. This is also the same as 100% full load.

Economic Continuous Rating (E.C.R.) - Refers to that load for which the turbine is designed to operate most efficiently under continuous conditions. If it is expected that the turbine will be operating at full load most of the time then of course it would be designed to operate most efficiently at 100% full load, in which case E.C.R. would be the same as M.C.R. However, if it is expected that the unit will be operating at part load most of the time then the unit may be designed to operate most efficiently at 75% full load. (The E.C.R. is then 75% of full load). The unit can still operate at M.C.R. or 100% full load when it is required, however, in this case the tertiary loop would have its inlet to the turbine at such a point that it would bypass the first two or three stages. This is so because the first two or three stages would not have a large enough flow area to be able to handle the steam flow required for 100% full load. Bypassing the first two or three stages means a loss in efficiency and therefore the turbine would be operating at a lower efficiency at M.C.R. than at E.C.R.

Double Casing

A turbine cylinder is basically a pressure vessel. Its weight is normally supported at each end. It is therefore designed to withstand hoop stresses in the transverse plane, and to be very stiff in a longitudinal direction to maintain accurate clearances between the diaphragms or fixed blades and the rotor.

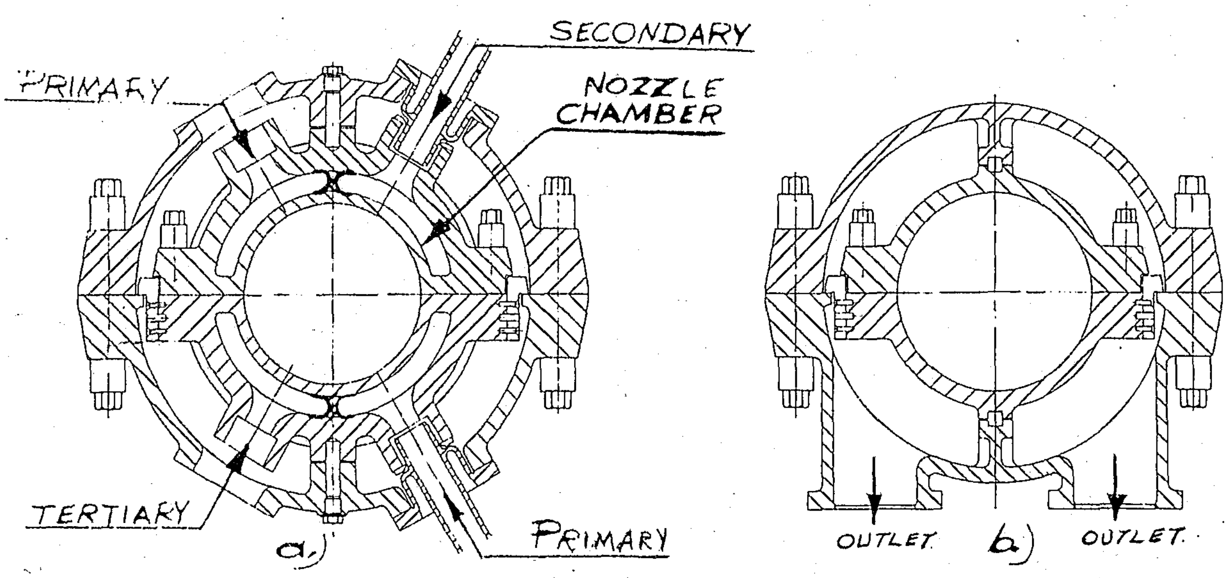


Figure 3. Cross section through double-shell high-pressure turbine.

Stress complexities are introduced by the gland housings, the horizontal flange and steam entry and exit passages. Furthermore, the external mass of the flanges may be such that (over 1 ft.thick for large units), when starting, they warm up more slowly than the remainder of the shell, and the different rates of expansion set up temperature stresses and distortion. This distortion can be minimized by incorporating passages in the flanges, through which steam is circulated during start-up so that they will warm up at the same rate as the rest of the casing.

Where steam conditions are above 1000 psi. the high pressure cylinder is generally of double shell design as shown in figure 3. In this case steam at exhaust pressure fills the space between shells, enabling each shell to be designed for a relatively small pressure differential as shown in figure 4. This, allows a reduction in shell thickness and results in an increase of metal area in contact with the steam. It permits quicker warming of the turbine when starting without undue temperature stress. The castings are simpler to manufacture and are as a result sounder.

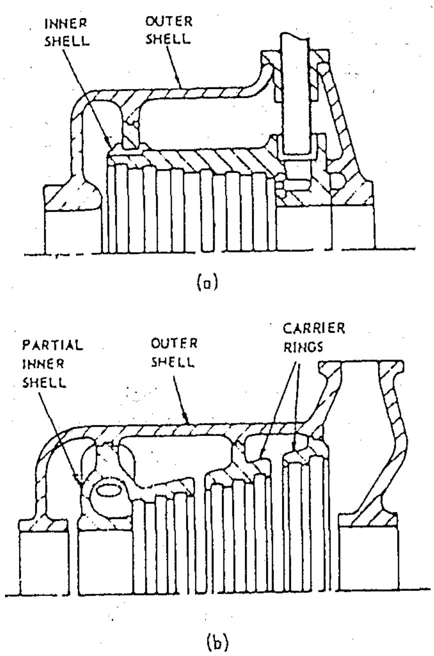


Figure 4. Double shells and carrier rings.

On turbines using reheated steam, the steam enters the intermediate pressure cylinder at high temperature but medium pressure and, therefore, the design of the casing is an easier problem than for the high pressure cylinder, the wall thickness being smaller. Nevertheless, a partial double shell is sometimes, adopted as illustrated in figure 4b).

The carrier rings shown are used to support the stationary blades or diaphragms when steam conditions are moderate. This permits the cylinder casting to be of simple design and the same pattern can be used for different stage arrangements.

We have been discussing individual cylinders so far. To see how they all fit together refer to figure 5. The turbine foundation supports H.P. and L.P. casings, bearing pedestals and steam chests. The H.P. inner cover with its diaphragms or blade rings fits into the outer cover. In this particular case the I.P. diaphragms and their carrier rings are supported by the outer part of the H.P. cylinder. The drum type spindles or rotors fit into the lower halves of the assembled casings. Fit is tight, with very small clearances between moving and stationary parts. When it is in place, the H.P. spindle is covered by the top halves of the diaphragms, and the inner cylinders. Upper half of diaphragms, and their carrier rings correspondingly cover the L.P. spindle.

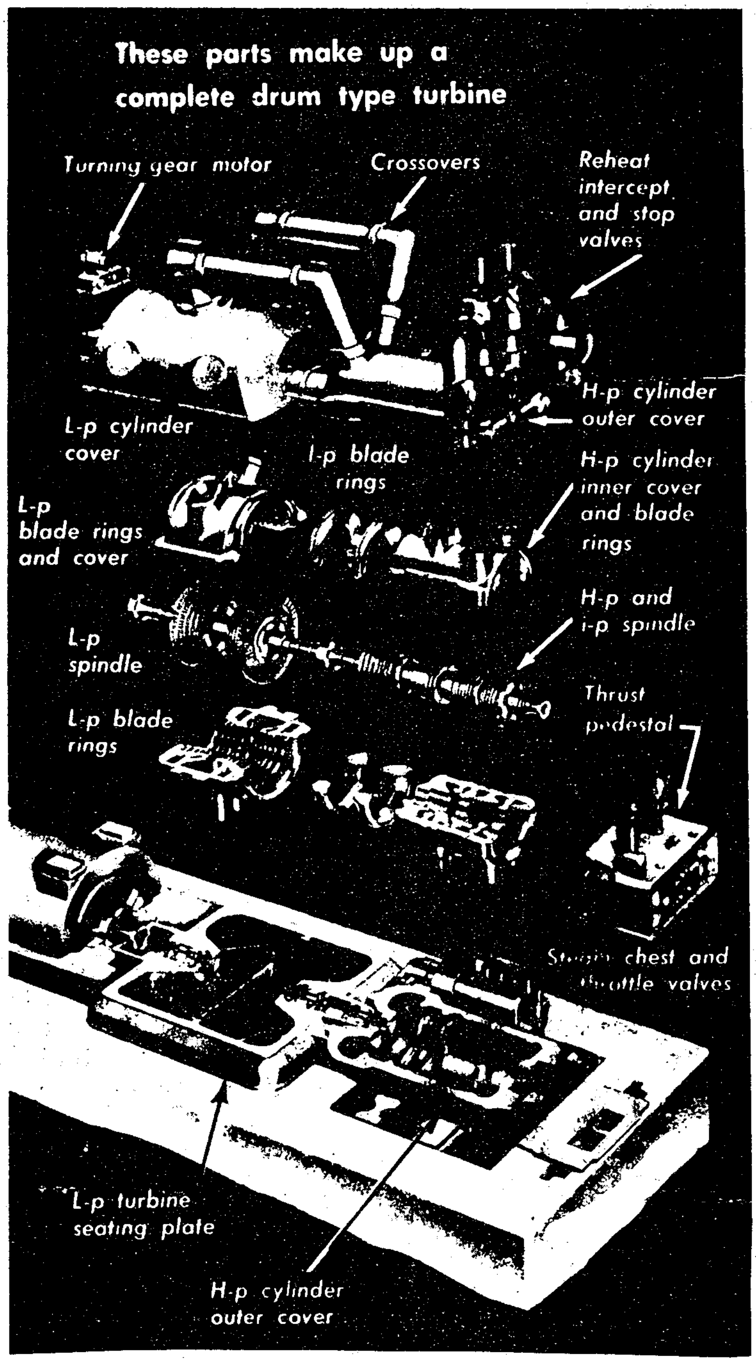


Figure 5. Diagram showing the parts for a large tandem compound turbine and how they are put together, including double casings and carrier rings.

H.P. outer cover and L.P. cylinder covers with crossover pipes and main steam connections, plus a turning-gear motor, complete the unit.

Dummy piston

In a type of turbine called a reaction turbine, which we will be discussing later on, there is a pressure drop across each row of blades and a considerable force is set up which acts on the rotor in the direction of steam flow. In order to counteract this force and reduce the load on the thrust bearings, dummy pistons are machined out of the rotor forging at the steam inlet end. It is usual to have a dummy piston for each cylinder except where the steam flow can be arranged so that the forces cancel each other out as in a dual flow L.P. or I.P. cylinder.

The dummy piston diameter is so calculated that the steam pressure acting upon it in the opposite direction to the steam flow balances out the force on the rotor blades in the direction of steam flow. It is preferable that the dimensions be so arranged as to keep a small but definite thrust towards the exhaust end of the turbine. To help maintain this condition at all loads a balance pipe is usually connected from the casing, on the outer side of the balance piston, to some tap-off point down the cylinder, as shown in figure 6. The turbine inlet nozzles would be located in the casing to the right of the dummy piston which means that the right hand side of the dummy piston would be at relatively high pressure. At the point where the tap-off for the balance pipe is located in the casing, steam pressure is relatively low which means that the left-hand side of the dummy piston would be at a relatively low pressure. Consequently there would be an overall thrust to the left on the dummy piston as the arrows indicate. This balances the thrust to the right due to the force exerted by the steam on the blades.

Extraction Belts and Drains

Figure 7 illustrates the type of arrangement used for extraction belts, (or bleeding point). This type of annulus completely encircles the casing at this point and a pipe connected to this annulus would lead to a feedheater.

As steam flows through a turbine and gives up its heat energy, some condensation will take place to form fine droplets of water. This is especially true in the case of turbines in nuclear power stations where the steam at inlet is already at saturation conditions. The percent moisture at the exhaust can be as much as 14%.

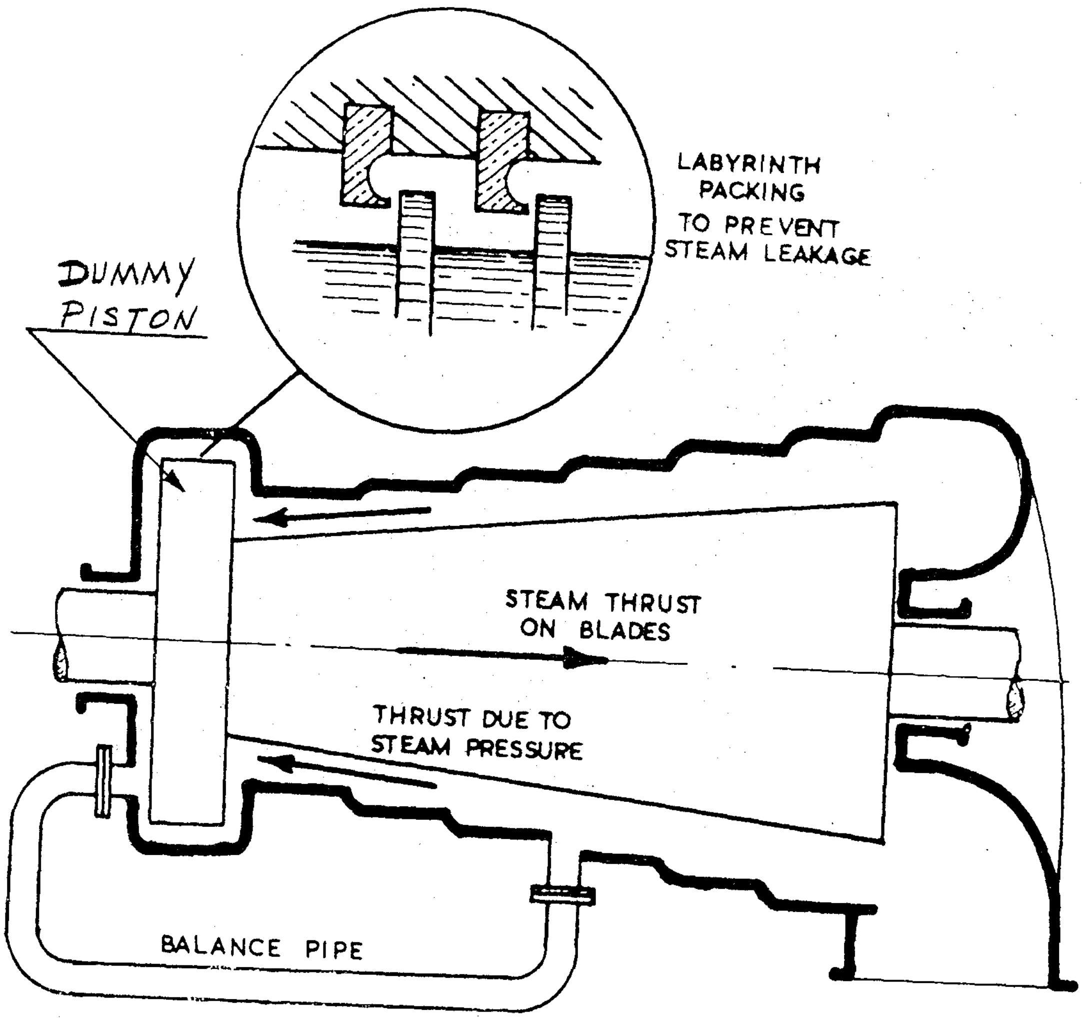


Figure 6. Dummy piston and balance pipe

The L.P. blades have to operate at high speed in wet steam and the droplets of water can cause severe erosion of the blades. Therefore, aside from the steam separator used in nuclear stations to remove moisture from the steam, another method can be used. The water droplets are denser than the steam and as they strike the moving blades they will tend to be flung radially outwards due to centrifugal force and collect on the inner surface of the casing. For this reason grooves are machined in the casing where condensate is expected to collect as is shown in figure 7. Small pipes connected to these grooves drain the condensate to the condenser.

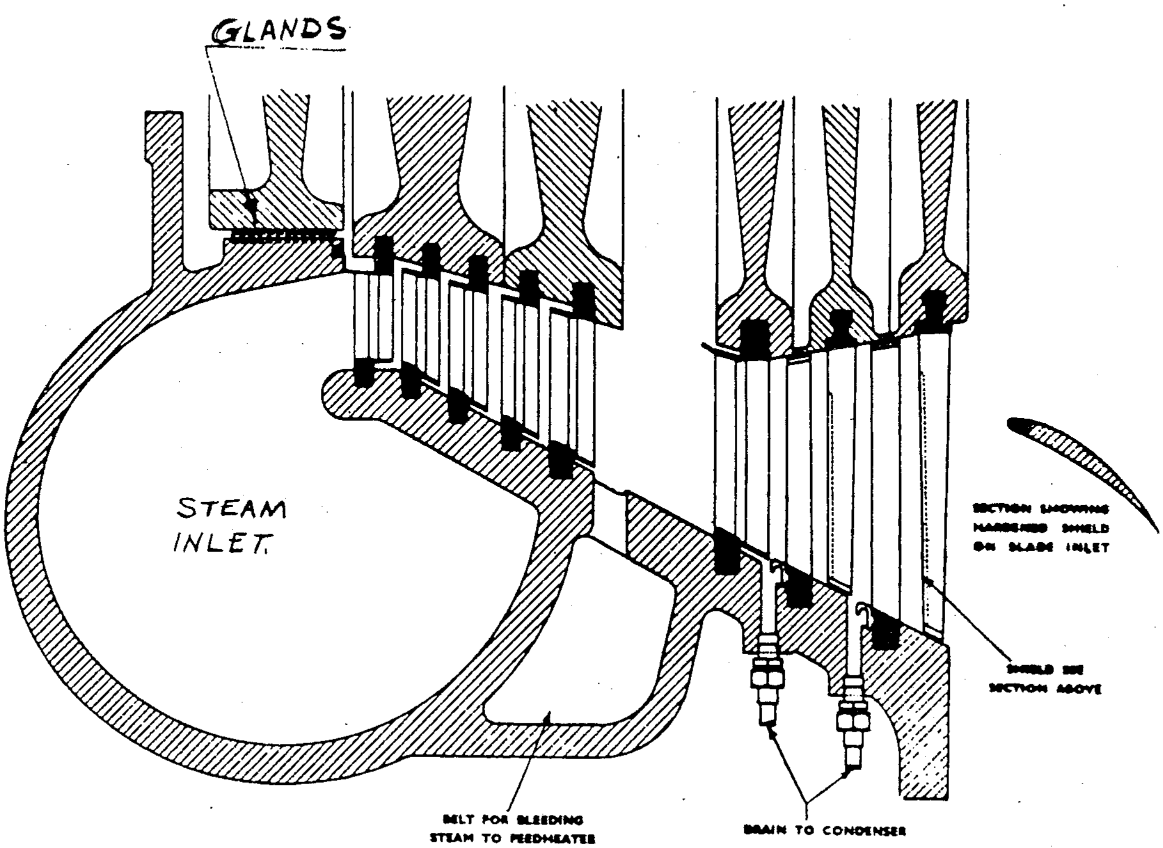


Figure 7. Quarter section of a turbine showing a steam extraction belt and drains to remove condensate from inside the turbine.

Exhaust Annulus

We said previously that there was a limit as to the length to which the last row of moving blades can be built because of the stress exerted on the blade due to centrifugal force. The maximum blade lengths in use at present are approximately 30 inches for 3600 rpm. shafts and approximately 46 inches for 1800 rpm. shafts. The length of the last stage blades will determine, of course, the flow area for the steam at the exhaust end of the turbine. We know that volume of flow is equal to area times velocity. For best efficiency the velocity leaving the last stage has to be as low as possible. Since the volume of steam flow is enormous, the exhaust area therefore has to be large. For large units the total area of the spaces between the last stage of moving blades (or exhaust area)may have to be around 150 to 200 sq. ft. Of course this area could never be made available by using only one exhaust and for this reason multiple exhausts (2 or 3 double flow L.P. cylinders) are used to obtain optimum exhaust annulus areas. The exhaust area and turbine efficiency are interrelated - i.e. within certain limits the greater the exhaust area the higher the efficiency.

The last-stage annulus area required for turbines of a given rating may be roughly estimated by the following relationship:

$$\text{Optimum annulus area} = \frac{\text{generator output kw.}}{\text{1000 x average exhaust pressure, in Hg.abs.}}$$

D. Dueck.

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4 - Turbine, Generator & Auxiliaries

-1 - Turbine Construction

A - Assignment

1. Draw a sketch showing the arrangement of control and overspeed protective valves for a reheat turbine. Label all the components.
2. What is the purpose of the intercept governing valves?
3. What do we mean by nozzle governing? When is this method of governing employed?
4. Define economic continuous rating (E.C.R.).
5. What is the advantage in using double casings for turbines?
6. What is the purpose of the dummy piston in a turbine?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-2 - Turbine Rotors

0.0 INTRODUCTION

This lesson will describe how turbine rotors are constructed, how they behave during rotation, how they are coupled together and aligned and how their expansion is taken care of.

A turbine rotor is often also referred to as a turbine spindle.

1.0 INFORMATION

Turbine rotors may be of four main types as follows:

Forged steel drum rotor
Disc rotor
Solid forged rotor
Welded rotor

Drum Rotors

In the drum type rotor the H.P. steam inlet end of the rotor is a single steel forging and the exhaust shaft and disc another separate forging. After machining, the drum is shrunk on to the exhaust end disc forging and secured by bolts and driving dowels. Grooves are machined in the body of the drum to take the necessary blades. Figure 1 shows a drum type rotor.

Because it is designed with the same thickness of material as the casing, it promotes even temperature distribution. The drum type rotor is limited in its application because of the excessive stresses which would occur if it were made in large sizes. It is therefore, suitable for small machines or high pressure cylinder machines with more than one cylinder.

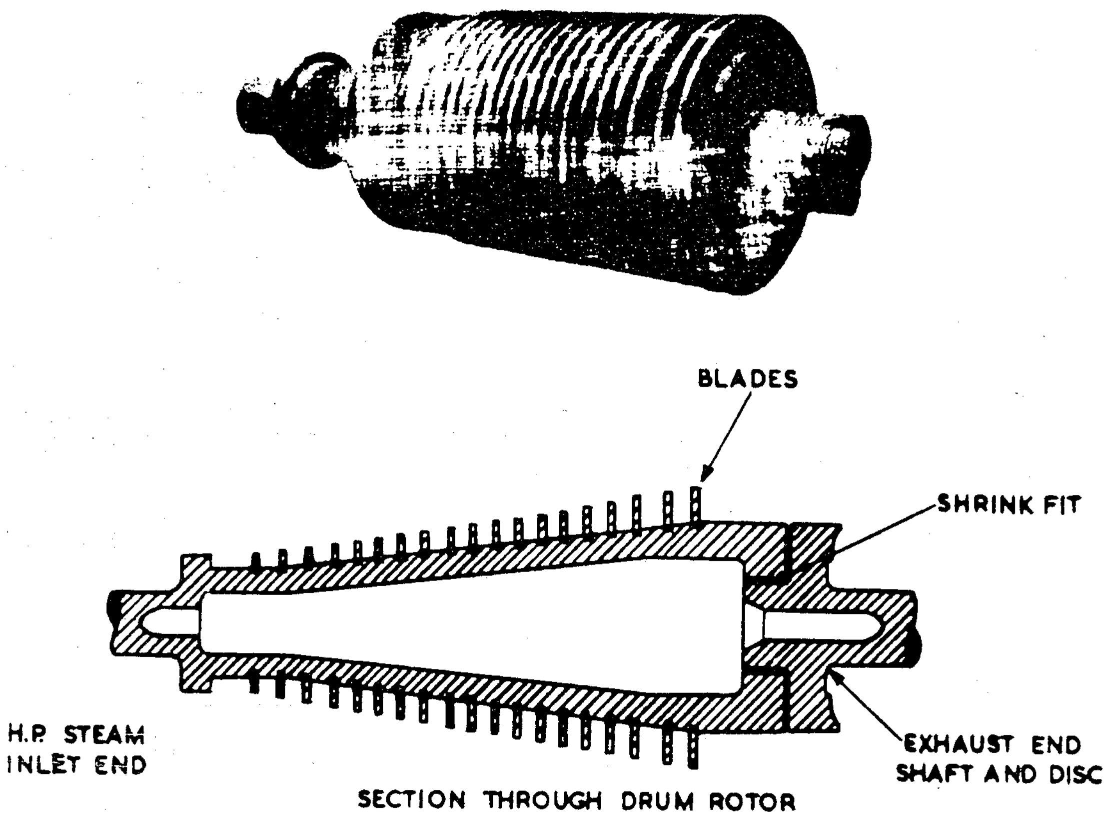


FIGURE 1. Drum type rotor

Disc Rotors

The disc rotor is made up of a number of separately forged discs or wheels and the hubs of these wheels are shrunk and keyed on to the central shaft. The outer rims of the wheels have suitable grooves machined to allow for fixing the blades. The shaft is sometimes stepped so that the wheel hubs can be threaded along to their correct positions. Suitable clearances are left between the hubs to allow for expansion axially along the line of the shaft. Figure 2 shows a disc type rotor.

This type of shaft is relatively inexpensive since the discs and shaft are relatively easy to forge and inspect for flaws. The disadvantage of these rotors is that there is danger of the wheels coming loose particularly at the high temperature and where at times the wheels may become hot and the shaft cool. Another source of trouble under conditions of high temperature and stress is the phenomenon of creep, which again could cause the shrink fit to disappear after prolonged running.

Disc type rotors are used mainly for low pressure machines.

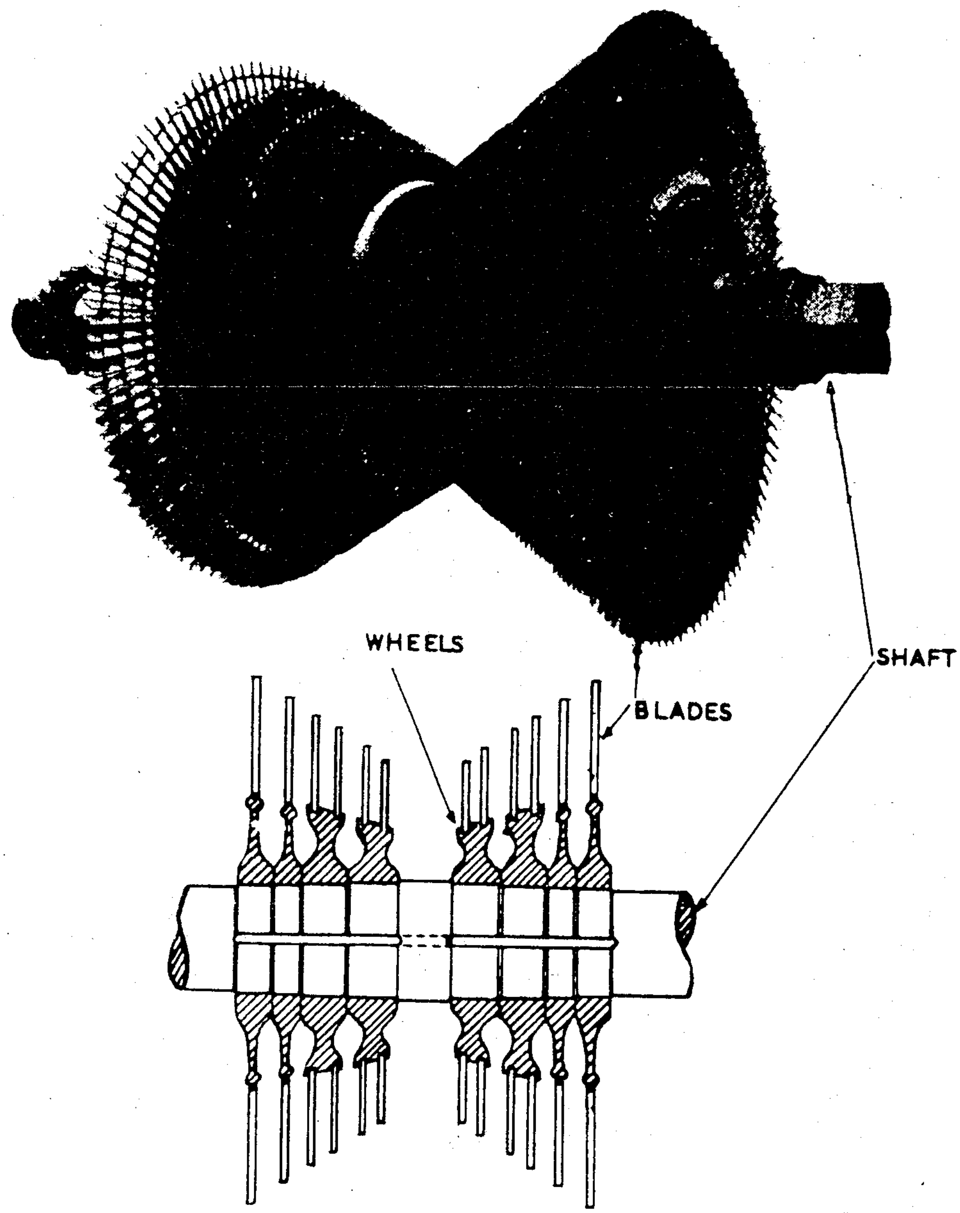


FIGURE 2. Disc type rotor

The main problem with low pressure rotors is one of centrifugal stress, the last stage disc being the most heavily stressed part of the turbine. (The safety factor, based on yield point may be about 2 at 15% overspeed.) The centrifugal load of the large rotating blades sets up a tensile stress in the rim of the wheel, which increases with decreasing radius, its maximum value being at the bore of the hub. Therefore the larger the bore of the hub, the greater is the maximum stress. If the bore is very small then the hoop stresses are lessened. If there is no hole, the hoop stresses throughout the disc are theoretically halved. This fact is made use of to some extent by solid forged rotors which have only a small hole used for inspecting the forging and in certain welded low pressure rotors. These two types of rotors are described below.

Solid Forged Rotors

Rotors of this type have wheels and shaft machined from one solid forging, the whole rotor being one complete piece of metal.

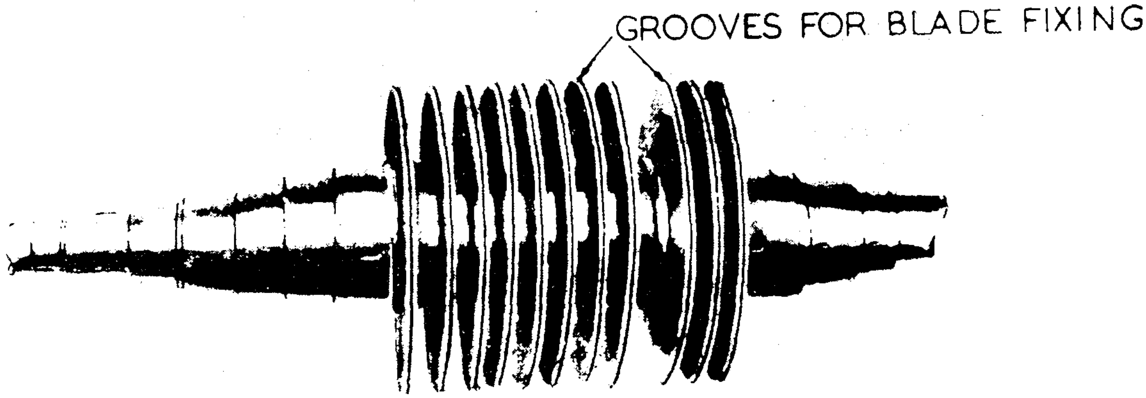
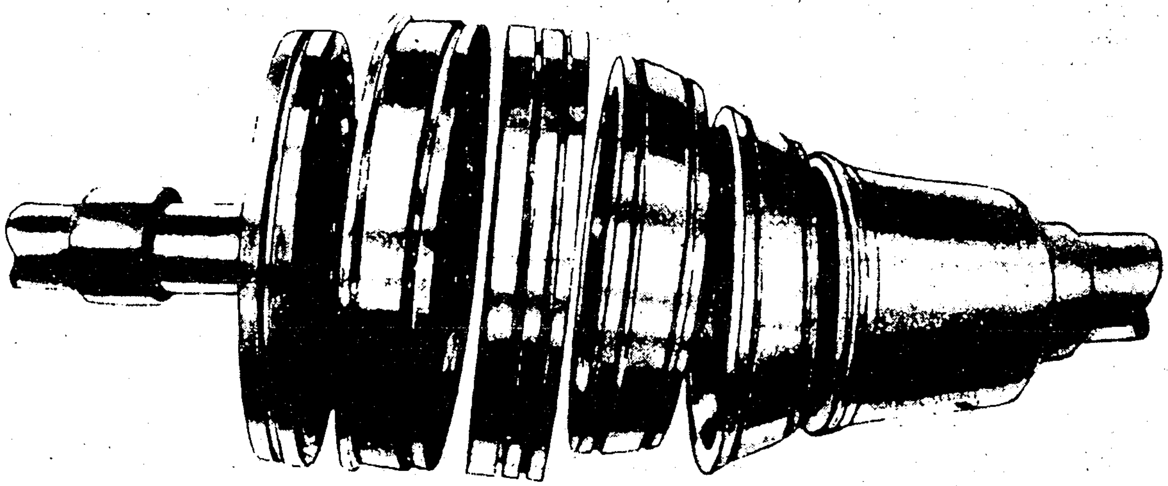


FIGURE 3. Solid forged rotor already machined.

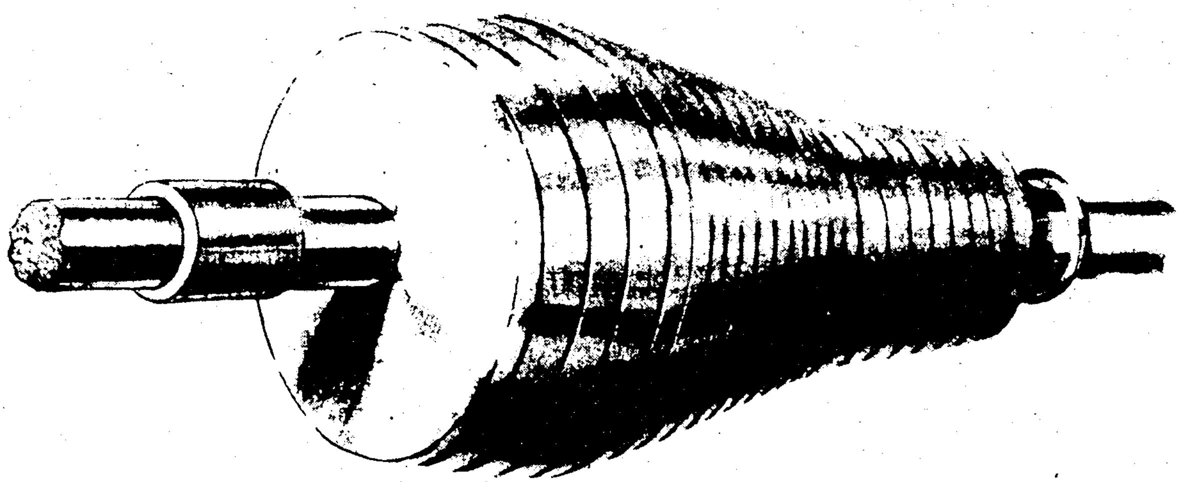
This results in a rigid construction and troubles due to loose wheels of the shrunk type are eliminated. Grooves are machined in the wheel rims to take the necessary blading. Figure 3 shows a solid forged rotor. This type can be used for the H.P., I.P. and even for the largest L.P. rotors. However, they are expensive to forge, and a fair number of forgings have to be rejected because they do not pass inspection tests. Nevertheless, they have other good features which override their disadvantages.

Welded Rotors

Welded rotors are built up from a number of discs and two shaft ends. These are joined together by welding at the circumferences and because there are no central holes in the discs the whole structure has considerable strength. Small holes are drilled in the discs to allow steam to enter inside the rotor body to give it uniform heating when coming on load. Grooves are machined in the discs to carry the blades. Figure 4 shows the welded type rotor.



ROTOR SHOWING DISCS BEFORE WELDING



ROTOR AFTER WELDING DISCS TOGETHER

FIGURE 4. Welded rotor

Balancing

When fully assembled the rotor is balanced statically and dynamically. Static balance means that the weight is evenly distributed around the axis of the shaft, and may be checked by rolling the rotor on horizontal knife-edge supports.

Dynamic balance means that the moments of the out-of-balance weights along the axis, about either bearing, add up to zero. This is checked by spinning the rotor on resilient bearings, detecting the vibration, and adding weights until it is negligible. A modern balancing machine which eliminates to a large extent the trial and error processes used in the past enables balancing to be carried out with a high degree of accuracy. Nevertheless a small out-of-balance force always remains.

Critical Speed

A stationary shaft supported between bearings has a natural frequency of vibration, depending on the size of its diameter in relation to the distance between the bearings. If its speed of vibration corresponds to its natural frequency the residual out-of-balance force causes resonance--i.e. causes an increase in intensity by sympathetic vibration, which can build up to a dangerous extent. This speed is known as the critical speed of the shaft. It is sometimes above the normal running speed and sometimes below. If above, the shaft is said to be "stiff". If below, the shaft is said to be "flexible". The critical speed should be passed as quickly as possible when running up the turbine. Standards require that the critical speed shall not be within 20% of the running speed. The prediction of critical speeds is very complex and depends on such factors as the flexibility of the supporting structure, among others.

During the manufacture of turbine shafts great efforts are made to ensure that the forging is stable; that is, that the physical properties of the forging do not change in service. An unstable shaft is likely to develop a deflection when in service, producing out-of-balance forces and consequent vibration. There are three types of instability:

1. Permanent, caused by non-symmetrical coefficients of expansion across a diameter. This can be avoided by close metallurgical control while the rotor is still in the ingot stage.

2. Temporary, caused by "locked-up" stresses in the rotor. This is relieved by rotating the shaft in a special furnace, at high temperature, both before and after machining.

3. Transient, caused by differences in conductivity and emissivity. This is normally overcome by the use of turning gear.

Transient Bending of Turbine Rotors

Transient bends have presented a most important operational problem, both as to control and consequences. They have occurred not only at initial commissioning but also after years of service, and with every type and construction of rotor. They are judged to be of several kinds, including bends due to transient thermal gradients in rotor body, to effects from gland sleeves and less demonstrably, to reactions from couplings and oil-film whirl in bearings.

Transient thermal gradient in a rotor body is the phenomenon whereby a rotor lying stationary in a hot casing acquires, over a period of time, a systematically increasing bend and thereafter, over a further period of time systematically straightens. This, as we explained previously, is because of a higher temperature at the top of the rotor than at the bottom as it cools. The bend is normally upwards.

As a quantitative measure of the problem, it may be noted that the mass of a typical 3600 rpm. rotor for example, a body diameter of 15", and 9 ft. between journal centres, and weighing 5 or 6 tons, would be displaced 0.001" upwards by the bend produced by a 'top' to 'bottom' temperature difference of less than 3°F. It may further be noted that a displacement of 0.004" would produce an unbalanced centrifugal force, when the shaft is rotating, equal to the weight of the rotor.

The rotor straightens if the temperature gradients are removed and this is done by using the turning gear on a turbine shaft. The rotation of the shaft while cooling ensures that there is even cooling and that no temperature gradients exist.

Use of Modal Shapes for Critical Speeds

With modern turbine-generators there is not only one rotor to consider as far as critical speed is concerned, but there may be four or five rotors coupled together to make up one shaft. The objective of the designer is to ensure that the critical speed is not within 20% of normal operating speed. In doing this the designer has to consider three different sets of coupling conditions and two different sets of temperature conditions.

Let us say we are considering a 3-cylinder turbine connected to a generator(alternator) as shown in figure 5. The 3-cylinder turbine is tested by itself for overspeed in the manufacturing plant; the generator rotor is tested independently for overspeed and finally the rotors all connected together in the final installation must be able to pass the overspeed test. These are the three sets of coupling conditions to be considered when evaluating whether the shaft's critical speeds are acceptable. In addition the turbine may be running under cold conditions during test and under hot conditions during normal operation. The critical speeds will be different for both conditions.

Consequently, the designer calculates the critical speed for each individual rotor and these speeds are then related to the corresponding critical speed of the complete unit. One way of doing this is by the use of modal shapes as is shown in figure 5. These modal shapes are then used to decide where design changes should be made in order to alter undesirable critical speeds (if any) of the complete coupled set. Critical speed changes can be made by changing the location of bearings. As is apparent from figure 5, moving the bearings for one rotor further apart would tend to result in a lower critical speed, while moving them closer together would tend to result in a higher critical speed. It does not matter whether the critical speed is above or below the normal operating speed just as long as it is not within 20% of the operating speed.

Figure 5 illustrates the type of results one would get for a turbine generator operating at 3000 rpm. The results would not be exactly the same for a 3600 rpm machine but there is enough similarity in the shape of the modes so that this can be used as an example. In this particular machine the I.P. and L.P. rotors are connected by what is called a long 'layshaft' between overhung couplings without additional bearings.

The first mode indicates a critical speed for the generator rotor at 1429 rpm. The second mode indicates a critical speed for the L.P. rotor at 1976 rpm. and so on. Each of the rotors has at least one critical speed as shown. However, in the fifth mode, notice that the shape of the curve for the generator rotor resembles a sine wave. This is actually an exaggerated portrayal of the shape the shaft takes when it is operating at its second critical speed, which is quite different from the generator's first critical speed. Vibrations at second critical speeds are more severe and more serious than first critical speeds. Rotors sometimes also have third critical speeds but there comes a point beyond which they would disintegrate. Don't confuse the term second critical speed. The overall shaft would run rough at 1429 rpm, 1976 rpm, 2176 rpm etc., but the only rotor actually experiencing a second critical speed in figure 5 is the one for the generator.

Since running at critical speed for any length of time is very undesirable, a machine with characteristics as illustrated in figure 5 could be run up to operating speed in the following sequence: After initial warm-up it could be brought up to 1000 rpm., and thereafter could be accelerated through successive steps to 1450 rpm., 1900 rpm., 2500 rpm and finally to 3000 rpm. This could be done without fear of holding the speedconstant at a critical value.

Couplings

The shaft is made of shorter sections or rotors, as we have described previously, and coupled together because of the difficulty of manufacturing and handling long shafts. Couplings are primarily devices for transmitting torque and they may either be flexible, semi-flexible or rigid. Flexible couplings are capable of absorbing small amounts of angular misalignment, as well as axial movement.

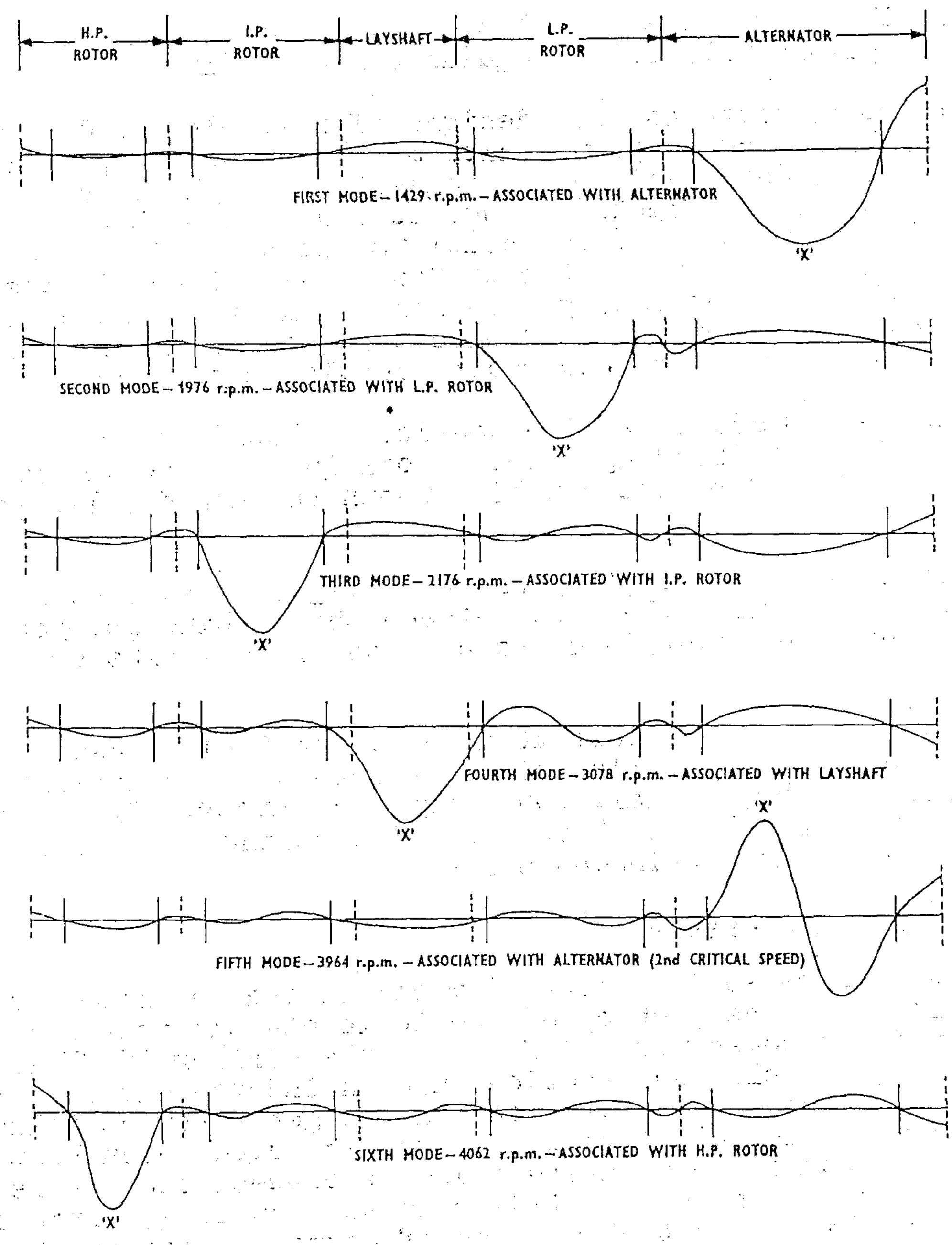


FIGURE 5. First six modes of vibration of a solidly coupled 3000 rpm. turbine-generator consisting of a 3-cylinder turbine connected to a generator, in which the I.P. and L.P. rotors of the turbines are connected through a long layshaft. Unit maximum values are marked by "x". Full and broken vertical lines denote bearing and coupling locations respectively.

Couplings (Cont'd)

Double flexible couplings can also accommodate eccentricity. Semi-flexible couplings will allow angular bending only.

Figure 6 illustrates some designs of flexible couplings in common use. The claw coupling which may be either single or double, is robust and slides easily when transmitting light loads. With heavy loads, however, friction tends to cause it to become rigid. The 'Bibby' type is adopted for couplings up to medium sizes. In addition to other features, it provides torsional resilience, the torsional stiffness increasing with load. The gear-tooth coupling transmits torque by internal and external gear teeth of involute form, which are curved to accommodate angular misalignment.

All these couplings require continuous lubrication. This is normally achieved by feeding a jet of oil into an annular groove, from which it is led to the coupling teeth through drilled passage-ways.

The semi-flexible coupling shown in figure 7 a) requires no lubrication and is normally used between the turbine and generator. It consists of a bellows-piece and has one or more ribs or convolu-tions.

The rigid coupling illustrated in figure 7 b) is used for high torque transmission where flexible couplings are impracticable. A spigot locates the two half-couplings and numbered fitted bolts join the flanges as shown. This type of coupling makes the several rotors behave as one continuous shaft.

History of Coupling Rotors

With the increase in rating of machines and the member of cylinders employed the problem of how to locate the rotors in the axial direction becomes more difficult. The earlier 60 MW sets, following long-established practice, had an independent thrust bearing in each turbine and also one in the generator as illustrated in Figure 8 a). Flexible type couplings were used throughout with provision to accommodate relative movement between the rotors due to thermal expansion by axial sliding in the coupling. Frictional resistance to sliding in the coupling of course reacts on the thrust bearing and because of this the generator thrust bearing was elim-inated and rigid coupling introduced between the low-pressure tur-bine and the generator as shown in figure 8 b). The next step was to eliminate the thrust bearing in the low pressure turbine and rigidly couple the low-pressure and intermediate-pressure turbine shafts together, leaving a flexible coupling only between the inter-mediate-pressure and high-pressure turbines. This is the present standard arrangement for three-cylinder 60 MW turbines and has proved highly satisfactory in service. The same arrangement was retained for the first 100 MW design shown in figure 8 (c), but for the 120 MW design the final (Cont'd on page 13.)

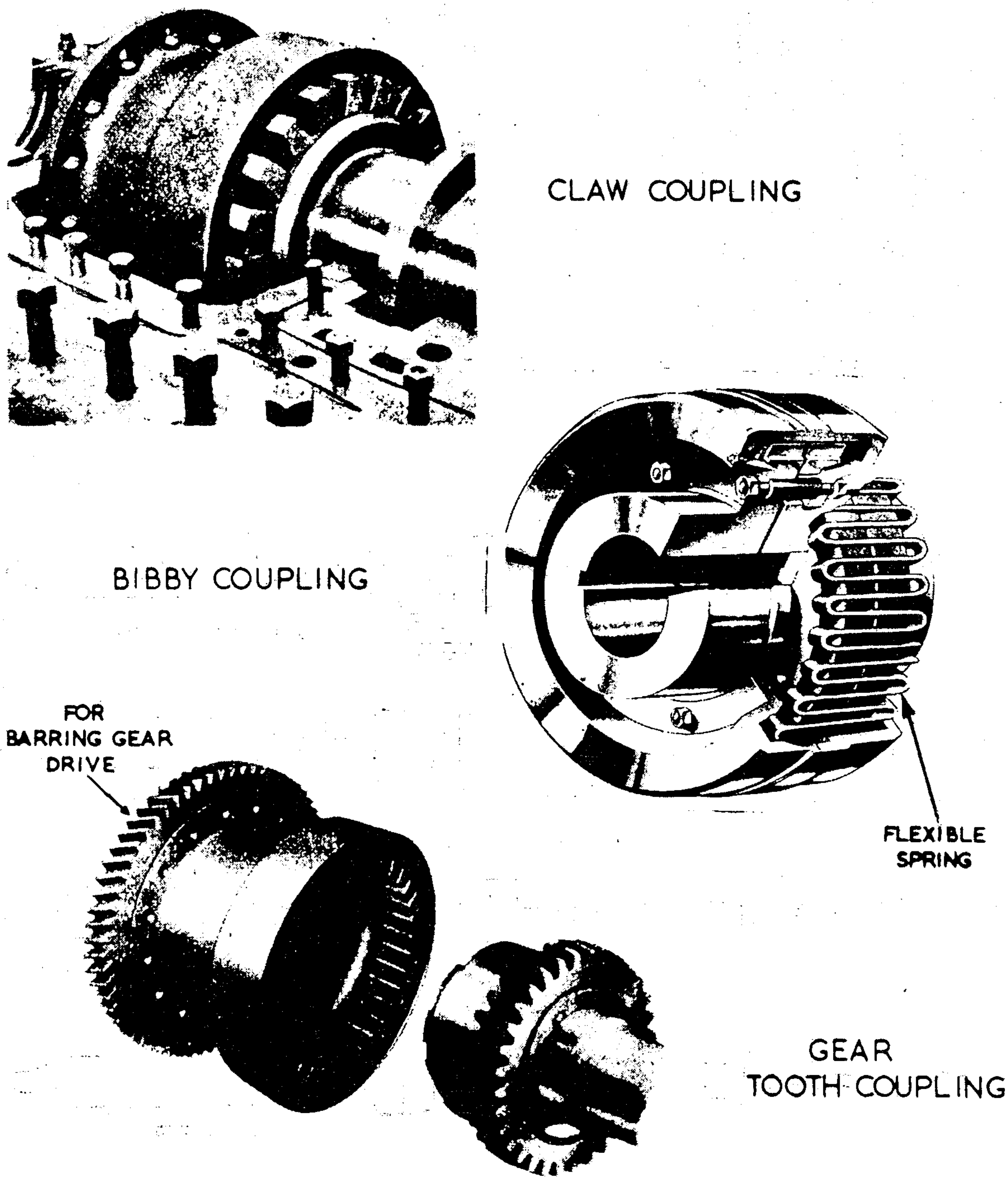


FIGURE 6. Flexible couplings

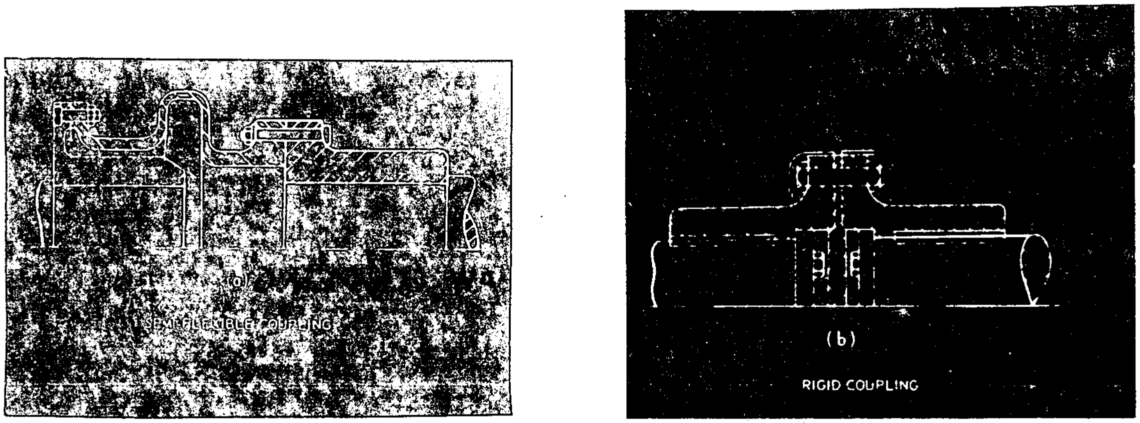
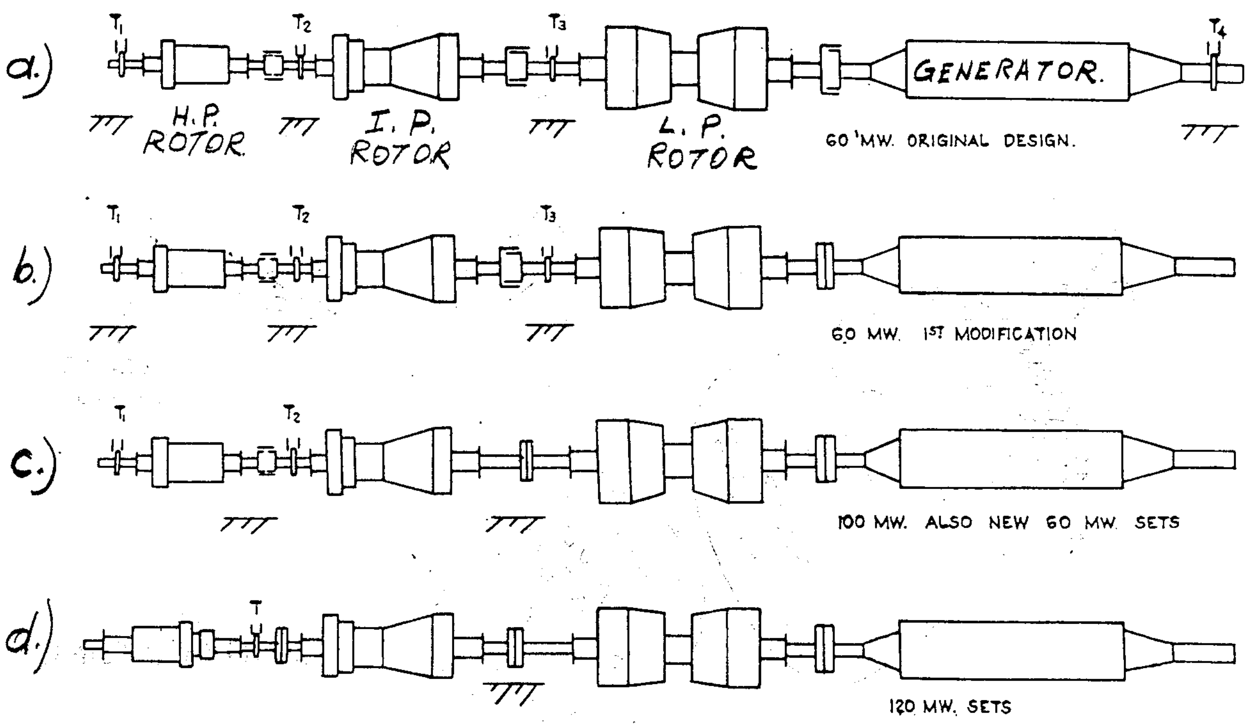


FIGURE 7. Semi-flexible and rigid coupling



T *denotes thrust bearing.* //// *denotes anchorage.*

FIGURE 8. Evolution in arrangement of couplings, thrusts and cylinder anchorages.

History of Coupling Rotors (Cont'd)

step was taken of adopting rigid couplings throughout. This involves turning the high pressure turbine around so that its inlet end is adjacent to the intermediate-pressure turbine, in an attempt to cancel out the thrust on the I.P. and H.P. rotors. The entire rotor assembly is located by a thrust bearing at the inlet end of the intermediate-pressure turbine.

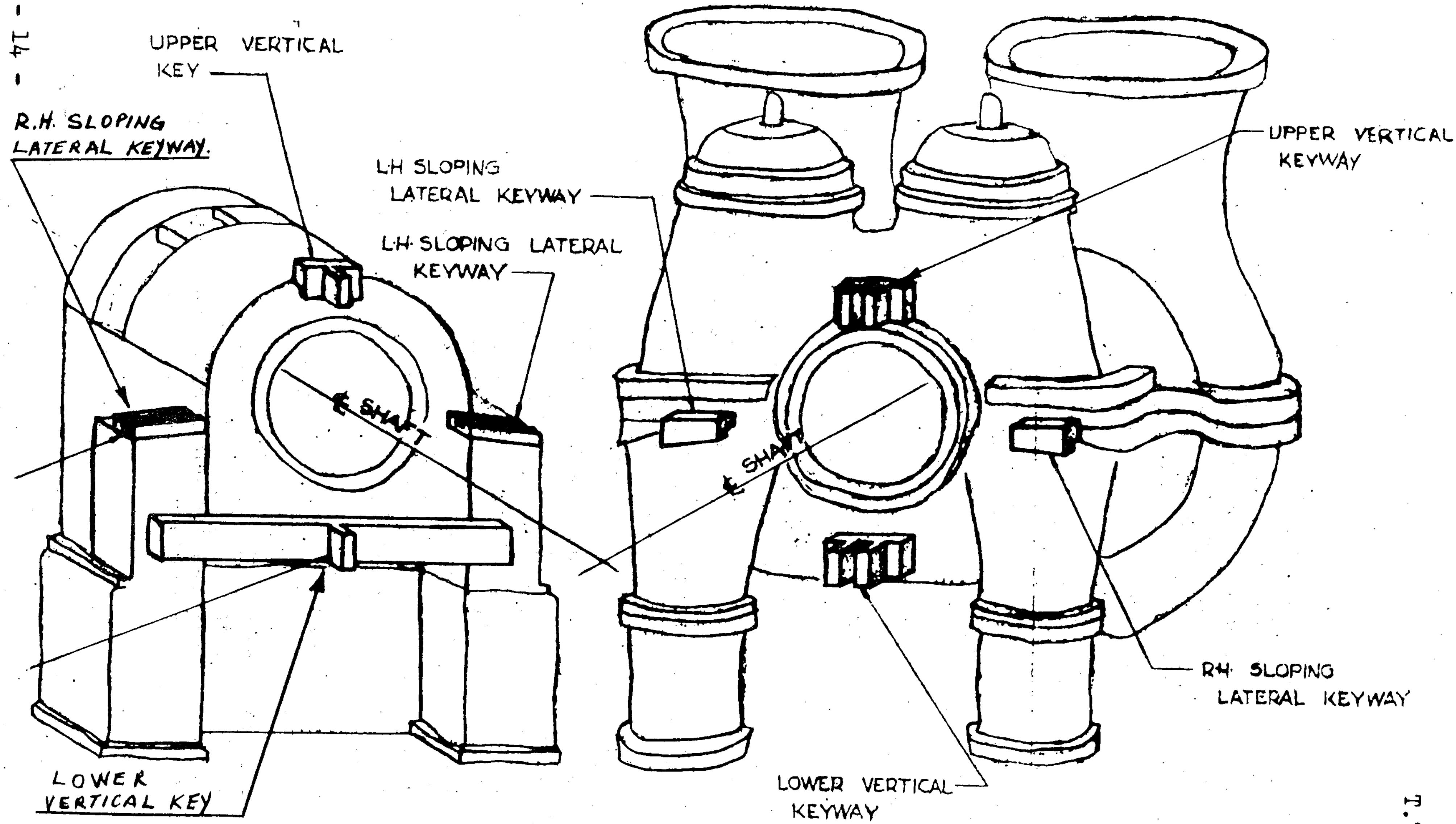
The employment of rigid couplings throughout is highly advantageous in that it eliminates a source of heavy and incalculable thrust loading. It does, however, mean that differential axial expansions between rotor and casing are cumulative from cylinder to cylinder and to the generator. Special care has to be taken to allow for this in the blade and gland clearances and at the hydrogen seals on the generator.

Anchoring of Cylinders

The turbine cylinders are supported by soleplates which are firmly bolted and embedded in the concrete foundations. The pedestals located in between the cylinders which support the bearings are also supported in a similar way. The cylinders and pedestals can slide on these soleplates in an axial direction. In fact graphite is often put between the surfaces to reduce sliding friction. Figure 8 d) shows that the 120 MW turbine is anchored to the foundation between the low pressure and intermediate pressure cylinders. Hence any movement of cylinders due to expansion must take place to the right and to the left from the anchorage point.

The pedestals and cylinder casings are all keyed together so that any movement in one produces movement in all of them. That is, as the I.P. cylinder expands it will move to the left, pushing the pedestal that contains the thrust and journal bearing, as well as the H.P. cylinder, along. This expansion may well be an inch or more. How, one might well ask: If the cylinders move that much, and considering the small clearances between rotating and stationary blades and glands, how come the blades and glands don't foul each other? The answer is that the thrust bearing moves along with the pedestal to which it is fixed, and drags the shaft along with it at the same rate as the cylinders expand. The shaft will expand too, of course, but in the opposite direction to the cylinders. The net result is that the difference in expansion between the rotor and casings is small enough to avoid any fouling.

In the case of turbines which have two double-flow L.P. cylinders the anchor point is generally between the two L.P. cylinders. The L.P. cylinder temperatures are not much above ambient temperatures and therefore they experience very little expansion.



INLET END PEDESTAL HIGH PRESSURE CYLINDER END

FIGURE 9. KEYING ARRANGEMENT OF FRONT END OF A SMALL UNIT TO INLET END PEDESTAL.

In order to maintain transverse alignment as the cylinders move axially, central sliding keys between the pedestals and foundations are used as shown in figure 9. Sloping compensating keys and keyways which are slightly below centre line on either side of the casing and pedestal maintain horizontal alignment during radial expansion of the cylinder.

Figure 10 shows the modern method of keying the high pressure cylinders and pedestals together. The paw supports are generally horizontal with the palms level with the horizontal joint. In this way concentricity is unaffected by expansion of the cylinder. The palms incorporate transverse keys which transmit axial expansion of the cylinder as shown.

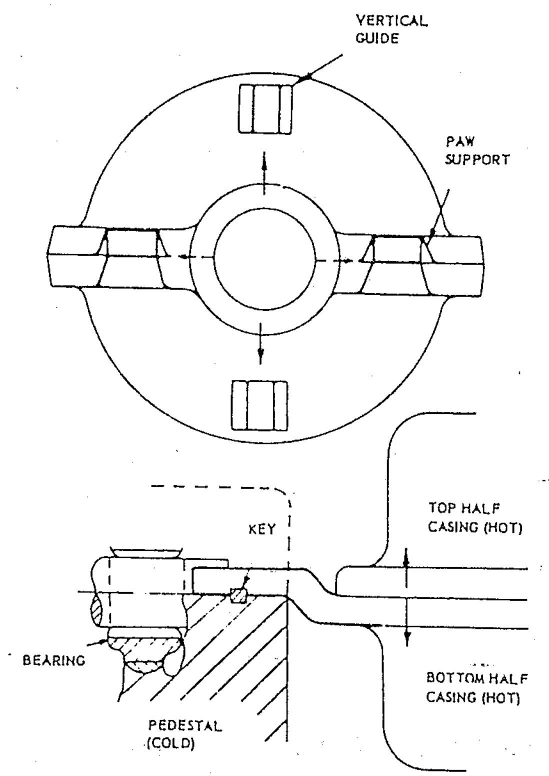
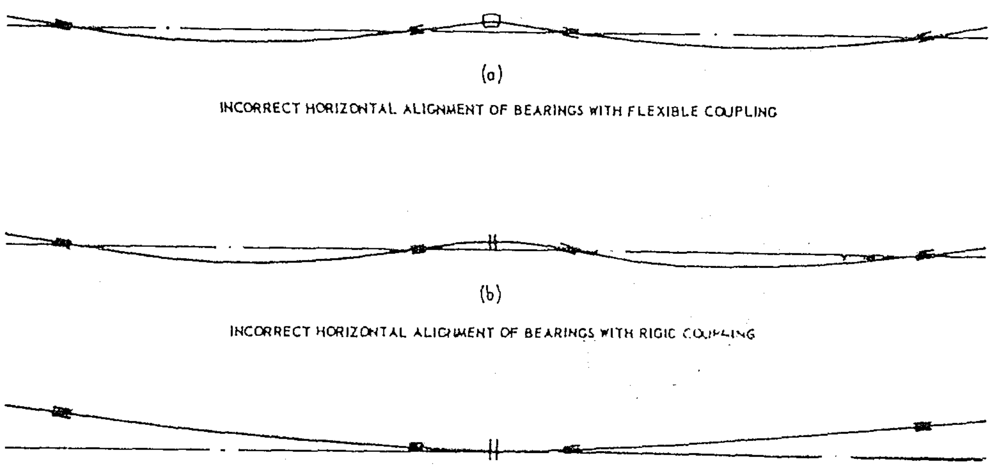


FIGURE 10. Cylinder support paws.

Shaft Alignment

The smooth running of a long line of four or five heavy rotors, all rigidly coupled together, must be expected to depend very greatly on the maintenance of good alignment of the bearings. From the very beginning, the most careful attention must be given to the foundations to ensure that they shall not settle or distort or vary in any way. Similarly, the structure of the low-pressure turbine which carries the more heavily loaded bearings must be extremely rigid and unvarying. The bearing shells are directly supported by transverse vertical plates of great depth which run uninterruptedly the full width of the exhaust space plus the supporting feet and are welded to the side plates, which enclose the exhaust, and to the bearing cradles. The thickness of the transverse plates ranges from 2" to 3" and the complete structure is one of great rigidity.

A long shaft naturally bends under its own weight but it nevertheless revolves about its curved centre line. Figure 11 shows how this fact influences the alignment of the bearings. Diagrams a) and b) are both incorrect and in practice such shafts would be lined out as shown in c). The two bearings near the centre are aligned horizontally. After the shafts are in position, the outer bearings are



(a)

INCORRECT HORIZONTAL ALIGNMENT OF BEARINGS WITH FLEXIBLE COUPLING

(b)

INCORRECT HORIZONTAL ALIGNMENT OF BEARINGS WITH RIGID COUPLING

(c)

CORRECT ALIGNMENT OF BEARINGS TO NATURAL DEFLECTION OF SHAFTS

FIGURE 11. Shaft alignment

raised to bring the coupling into alignment. On a large turbine the outer bearings may be about $\frac{3}{4}$ inch above the level of the central bearings.

The position of the bearings may be altered by putting shims under the pedestal or by placing shims directly underneath or on the sides of the bearing seats.

Figure 12 shows an alignment curve for a 3-cylinder turbine, generator and exciter. The general principle of alignment is that, assuming the coupling faces to be true with the shafts, the shafts are aligned in such a way that a continuous curve is formed with their natural deflections from governor to exciter. The shafts will retain their natural deflection at any speed other than the critical speeds.

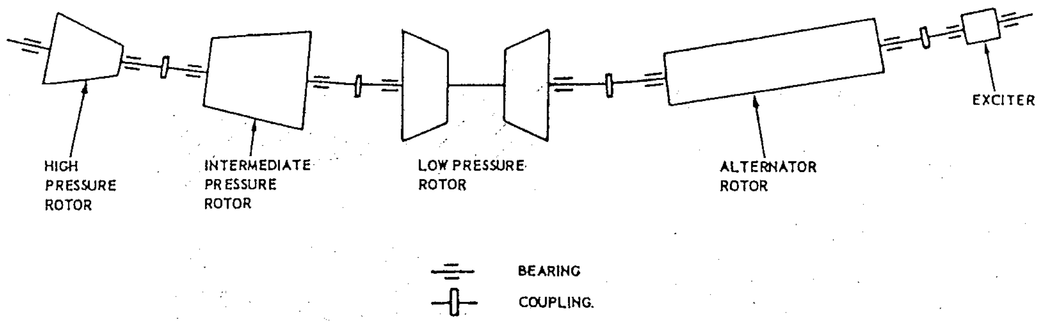


FIGURE 12. Alignment curve for a turbine-generator.

Position of Shaft in Bearings

When a normal journal sleeve bearing is at rest the shaft lies at the bottom of the bearing in the manner shown in figure 13 a). The weight of the shaft tends to squeeze the oil out of the bearing until only that adhering to the surfaces remains.

As rotation commences the first action of the shaft will be to climb up the bearing, as shown in figure 13 b) to a point where it begins to slip. At this stage there is a sliding of metal on metal and boundary lubrication takes place by oil adhering to the metal.

As the speed of the shaft increases the lubricant is dragged over top of the shaft and down, by virtue of its viscosity until it forms a thin pad under the shaft and film lubrication is established. The shaft now assumes a central position, but due to the oil film is raised slightly in the bearing as shown in figure 13 c).

A further increase in speed increases the pumping action on the oil and further establishes the oil film. When this is accomplished the shaft assumes a position as in figure 13 d). It will be seen that the bearing has created the ideal wedge-shaped oil film.

When a jacking oil pump is used figure 13 a), and b) are eliminated and there is no metal to metal rubbing.

The minimum thickness of the oil film depends upon details of design but an average value is approximately ½ thousandths of an inch per inch radius of shaft. For example the minimum film thickness for an 8 inch diameter shaft would be 2 thousandths of an inch.

The clearance between shaft and bearing shell is again a matter of design but is generally about 1 thousandth of an inch per inch of shaft diameter.

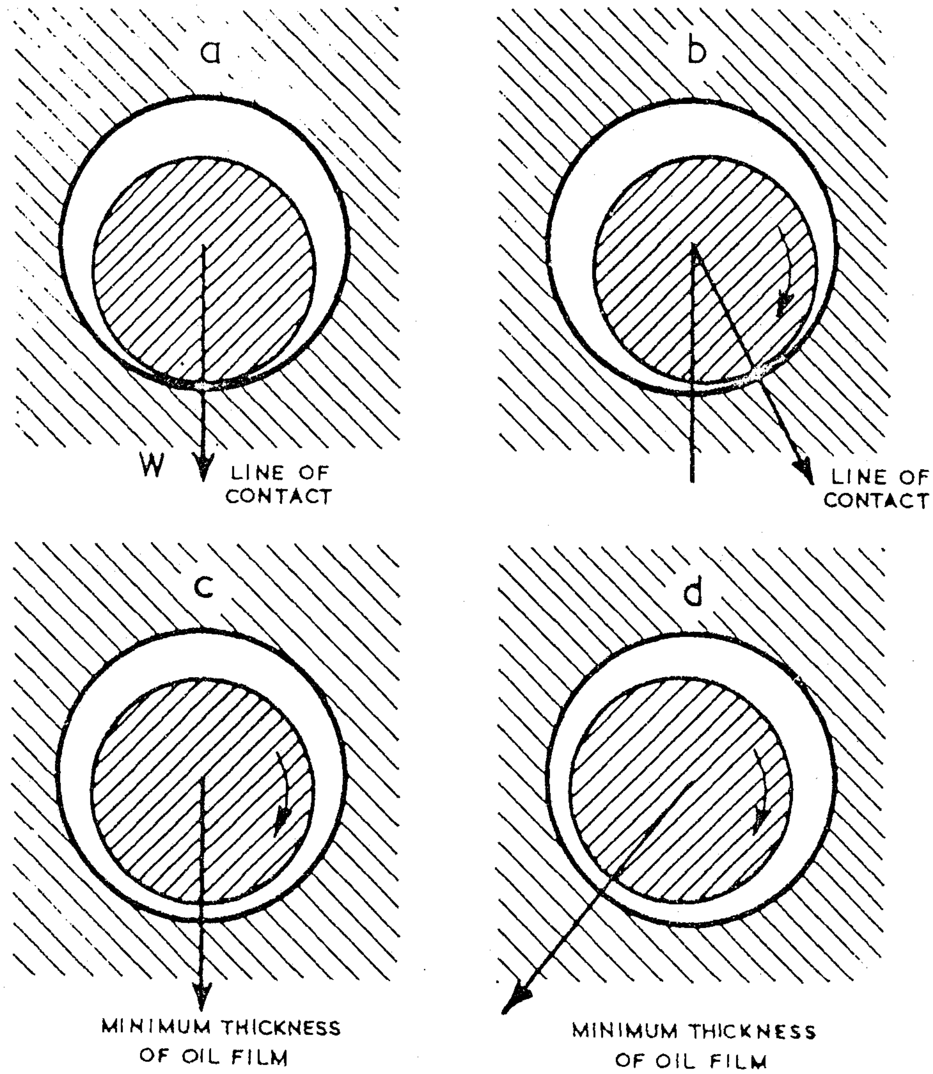


FIGURE 13. Position of shaft in bearing.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine Generator & Auxiliaries

-2 - Turbine Rotors

A - Assignment

1. What four main types of turbine rotors are there?
2. What do we mean by dynamic balance?
3. What is the critical speed of a shaft? Why is it undesirable to operate a shaft near its critical speed?
4. Name and describe 3 types of instability that can occur in a shaft.
5. What is the current practice as far as coupling rotors together is concerned?
6. Describe why there is no fouling of blades, glands and generator hydrogen seals due to expansion in a modern turbine generator.
7. Draw a correct alignment curve for a turbine generator.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-3 - Foundations

0.0 INTRODUCTION

The foundation which supports the turbine generator, exciter and condensers is referred to as the turbine-generator "block". Normally in an operational steam station little attention is paid to the turbine-generator block because great care is taken in its design. However, a poorly designed and constructed block could result in considerable difficulty as far as safe operation of the unit is concerned.

This lesson will discuss a few important points concerning the turbine-generator block.

1.0 INFORMATION

The turbine-generator has to be designed so that it effectively fulfills the following purposes:

1. It has to safely support the load due to turbine, generator, exciter and condenser.

2. It has to absorb vibration due to operation of the unit and prevent spread of this vibration to the rest of the building.

3. It has to remain level throughout the life of the station. A sag in one place could mean serious shaft misalignment, resulting in a rough running unit.

4. It has to provide sufficient passageways for main steam lines, extraction steam lines, C.W. piping electric cables, etc. leading to and from the unit.

Figure 1 illustrates a typical turbine generator block for a 120 MW unit. In order to design and construct the foundation so that it does not sag at any point it is very important to know what type of material the foundation rests on. For this reason soil tests at the site are carried out before design begins. Pouring the foundation on top of solid rock is desirable but it is very seldom possible to find such a site. Very often it happens that the load bearing material is quite poor. The only solution to this problem is then to drive piles into the ground up to the point where they strike solid rock or some other suitable load bearing material.

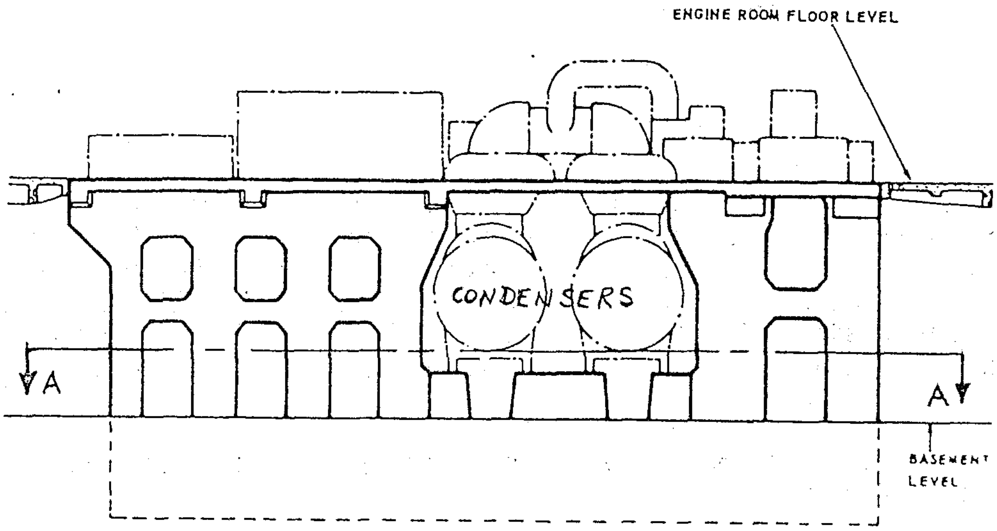


FIGURE 1. Elevation view of a turbine-generator block.

These piles may be made of huge timbers or steel I-beams. For some of Ontario Hydro's steam power stations piles as long 90 ft. to 130 ft. have been driven into the ground.

Over top of the piles a foundation to support the block is then poured, which is located below the basement floor level shown in dotted lines, figure 1. This part may consist of 8 ft. of solid reinforced concrete.

For operational reasons, it is convenient to mount the turbine-generator at the same level as the control room, and it has to be high enough so that the condensers can fit underneath the L.P. exhausts. For this reason the turbine floor is often 28 ft. or more above the basement floor level. The turbine-generators contain heavy rotating machinery, and a considerable amount of vibration is transmitted to the supporting structure. Therefore, although it would be possible to support a turbine-generator on a steel structure carried on concrete foundations, greater rigidity and vibration absorption is obtained by supporting it on a reinforced concrete block. The wider transmission of vibration is prevented by making the block completely isolated from the rest of the building and other foundations. This is achieved by a 1 inch thick layer of bituminous cork filler placed between the block and concrete floor at each floor level.

In figure 1 can be seen the base supports for the condensers; this type of base support for any piece of equipment is commonly referred to as a "plinth". For the condenser the plinth is around 5 ft. thick.

Figure 2 illustrates a sectional plan of the turbine generator block. This section has been taken at a level just above the condenser plinths as shown by the lines A--A in figure 1. It shows that the

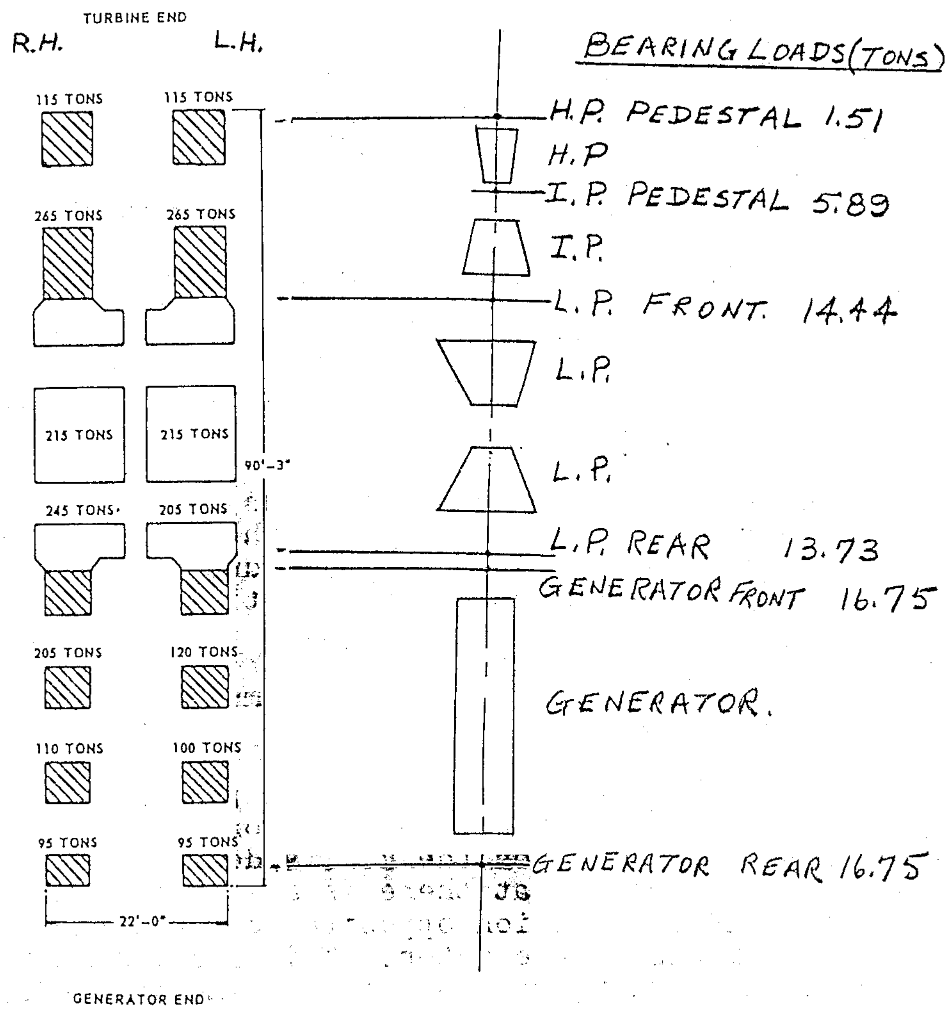


FIGURE 2. Sectional plan of turbine-generator block.

thick concrete floor carrying the turbine-generator is supported by four pairs of concrete columns at the generator end shown cross-hatched and by two pairs of concrete columns at the turbine end.

The intervening space (not cross-hatched), which houses the condensers is bridged by a concrete floor, at the turbine floor level, to carry the low pressure cylinder. This floor receives immediate support from a pair of steel columns located between the condensers (not shown in figure 1). Opposite the sectional plan of the columns are shown the relative positions that the turbine cylinders and generator would occupy, together with the loads borne by the bearings.

Figure 2 also shows the total loads for a typical 120 MW unit carried by the concrete columns and plinths including self-weight of the concrete columns.

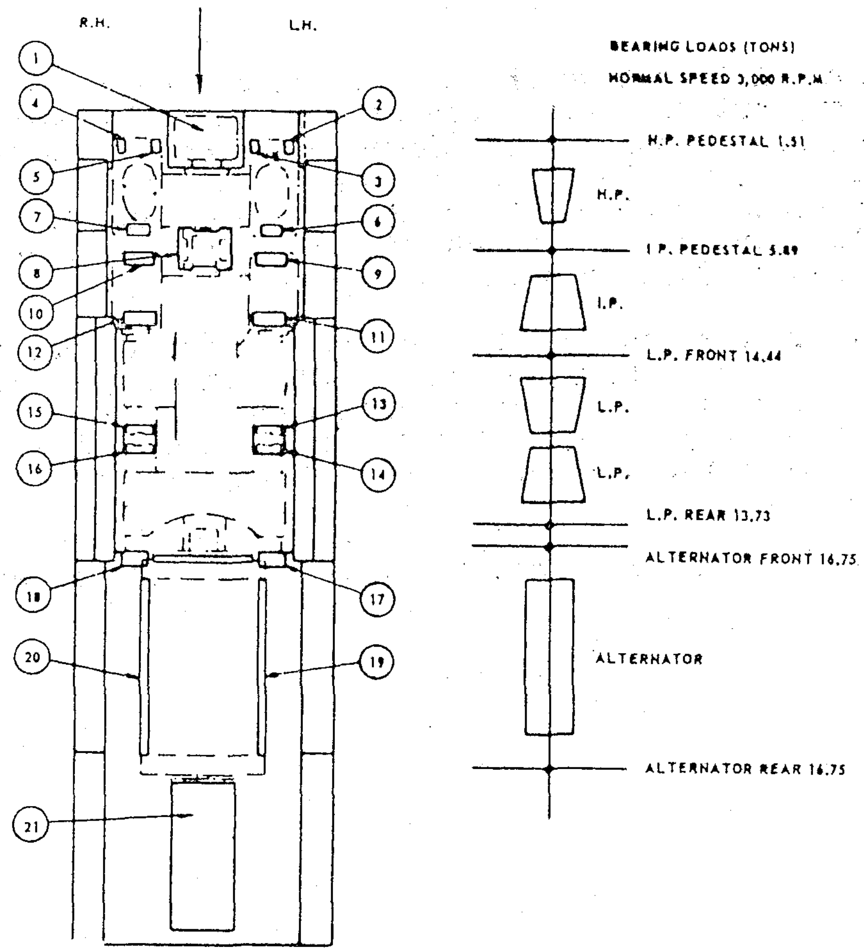
Block Loads at Turbine Floor

The loading of the block at turbine floor level due to the turbine-generator is shown in figure 3 together with an associated table of loads. The values listed in the table are ones that the designer has to know before he starts designing the turbine generator block.

The encircled numbers point to the various pedestals, stools and soleplates on which the loads are concentrated. Opposite these numbers in the table are given the loads under various conditions. R.H. stands for right hand, and L.H. stands for left hand. This follows the conventional designation looking from the governor end of the turbine. All the loads are given in tons.

Static load refers to the weight on various soleplates when the equipment is stationary. Friction load lists the longitudinal or axial forces exerted on the foundation columns due to sliding friction as the turbine expands and contracts. The friction forces will be acting towards the governing end as the turbine expands and towards the generator end as the turbine contracts. For this reason they are shown as positive and negative. The block columns have to be able to withstand these forces.

The column listing torque reactions are of particular interest. You will notice that for the turbine the reactions on the L.H. side are downward while the reactions on the R.H. side are upward. In a lesson later on you will learn that there is a reaction on the stationary blades which acts in a direction opposite to the direction of forces on the moving blades of the rotor. This reaction force is trying to rotate the stationary part of the turbine in a direction opposite to the rotor rotation. Since in this example there is a downward force on the L.H. side and an upward force on the R.H. side the reaction forces on the stationary part of the turbine are acting in a counter-clockwise direction. (From this one can deduce that the shaft is rotating in a clockwise direction.) The torque values listed are for a 3000 rpm machine. For a 3600 rpm machine these values would be slightly larger.



	Type of Load	Static	Friction	Torque Reactions	Distribution of Turbine Rotor Weights	Distribution of Alternator Rotor Weights
1	H.P. Pedestal Sole Plate	21.5	±[illegible].75	L.H.[illegible] Down R.H.[illegible] Up	1.51	-
2	L.H. H.P. Steam Chest Front Outer Stool	4.4	±3.5	-	-	-
3	L.H. H.P. Steam Chest Front Inner Stool	4.4		-	-	-
4	R.H. H.P. Steam Chest Front Outer Stool	4.4	±3.5	-	-	-
5	R.H. H.P. Steam Chest Front Inner Stool	4.4		-	-	-
6	L.H. H.P. Steam Chest Rear Stool	8.8	±4.25	-	-	-
7	R.H. H.P. Steam Chest Rear Stool	8.8	±4.25	-	-	-
8	Centre Pedestal Sole Plate	46.0	±23.0	L.H.7.72 Down R.H.7.72 Up	5.89	-
9	L.H. I.P. Steam Chest Front Stool	6.0	Nil Spring Supports	-	-	-
10	R.H. I.P. Steam Chest Front Stool	6.0	Nil Spring Supports	-	-	-
11	L.P. Front Cylinder L.H. Front Sole Plate	40.0	±20.0	2.64 Down	4.99	-
12	L.P. Front Cylinder R.H. Front Sole Plate	40.0	±20.0	2.64 Up	4.99	-
13	L.P. Front Cylinder L.H. Rear Sole Plate	30.5	±30.0	1.42 Down	2.23	-
14	L.P. Rear Cylinder L.H. Front Sole Plate	29.5	(Centre Column)	(Centre Column)	2.98	-
15	L.P. Front Cylinder R.H. Rear Sole Plate	30.5	±30.0	1.42 Up	2.23	-
16	L.P. Rear Cylinder R.H. Front Sole Plate	29.5	(Centre Column)	(Centre Column)	2.98	-
17	L.P. Rear Cylinder L.H. Rear Sole Plate	22.05	Anchor Point ±67.37	0.69 Down	3.[illegible]	-
18	L.P. Rear Cylinder R.H. Rear Sole Plate	22.05	Anchor Point ±67.37	0.69 Up	3.[illegible]	-
19	Alternator L.H. Sole Plate	103.0	-	9.3 Up	-	16.75
20	Alternator R.H. Sole Plate	103.0	-	9.3 Down	-	16.75
21	Exciter Bedplate	18.0	-	-	-	-

Exciter Bedplate area to carry extra 33.5 tons static load during alternator rotor withdrawal.
Anchor Point (friction load) midway between Rear L.P. Sole Plates on centre line of machine.
Alternator Torque reactions opposite to turbine.
Turbine Rotor and Alternator Rotor Weights have been included in the static loads.
For Turbo-Block design purposes take Short Circuit Torque as 9 x Normal Torque on R.H. Sole Plate.
This figure should be doubled for shock load.
All loads given in tons.

FIGURE 3

As an example let us take the L.P. front cylinder sole plates. Each soleplate supports a static load of 40 tons. The one on the L.H. side has in addition to this a 2.64 downward force due to torque reaction during operation. So the total load on the L.P. front cylinder L.H. front sole plate is 40 + 2.64 = 42.64 tons during operation. Similarly the load on the L.P. front cylinder R.H. front soleplate would be 40 - 2.64 = 37.36 tons during operation.

Notice that in the case of the alternator (generator) the reaction torque is upward on the L.H. side and downward on the R.H. side. We said the rotor was rotating clockwise and due to magnetic effect there is a force acting to try and drag the stator around with it. For this reason the rotational force is opposite on the generator stator to that on the turbine casings.

The remainder of the information in figure 3 is self-explanatory.

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NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-3 - Foundations

A - Assignment

1. Why should the turbine block be free from sag in any one place?
2. What are the main reasons for using a reinforced concrete turbine block rather than a steel structure?
3. Why is the turbine block isolated from the rest of the steam station building?
4. What is the reason for uneven loading between the L.H. and R.H. sides of the turbine generator block?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-4 - Nozzles

0.0 INTRODUCTION

Nozzles play a very important role in turbines.

In the next lesson we want to describe two different types of turbines which use nozzles--namely reaction turbines and impulse turbines. However, before we can do so we need to discuss the function of nozzles. A nozzle is essentially a device that converts heat or internal energy to kinetic energy.

We have said in the lesson on "How a Steam Turbine Works", T.T.3 level, that for turbine use, a nozzle must form a smooth jet of steam travelling at high velocity. This jet should keep its shape so that it impinges effectively on a blade. We also said that the velocity at which the jet travels depends upon the pressure difference across the nozzle, that is between nozzle entrance and exit, and the initial temperature of the steam.

We further said that pounds-per-second flow also depends on the cross-sectional area of the nozzle. The larger the area, the greater the weight of flow. These two factors are directly related; doubling area doubles flow, tripling area triples flow, etc.

Different fluids such as water, gas or steam behave in different ways as they pass through a nozzle. This lesson will describe only how steam behaves as it passes through a nozzle and the different shapes of nozzles required for different pressures.

It is assumed that the reader has studied the course on "Heat & Thermodynamics" T.T.2 level.

1.0 INFORMATION

The following example will give an introduction as to the way steam behaves as it flows through a nozzle. Figure 1 illustrates steam flow with discharge pressure variations. The inlet pressure remains constant. By keeping inlet and discharge pressures equal across a nozzle obviously there will be no flow. That is, with discharge pressure equaling 100% inlet pressure, relative flow is zero, as shown in the lower graph. Considering the curve labelled 3A, we see that as the discharge pressure drops to 90% of inlet pressure,

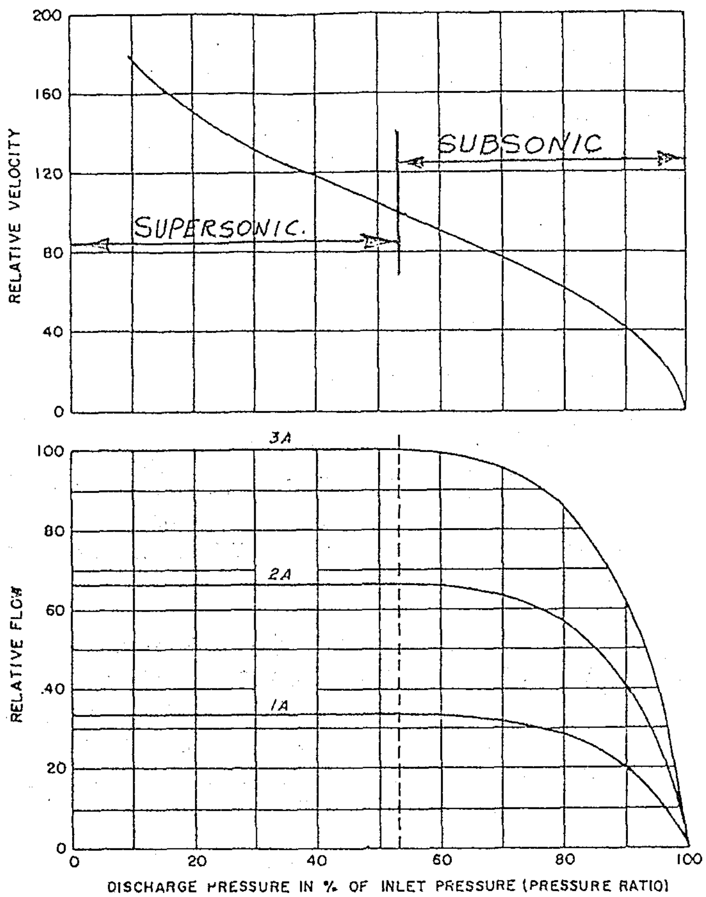


Figure 1

Figure 1. When initial steam pressure is kept constant and the discharge pressure lowered behind a nozzle, the leaving-steam velocity increases (top graph). The rate of flow, however, rapidly increases until discharge pressure drops to about half initial pressure (bottom graph). Further drop in exhaust pressure makes no change in maximum flow occuring at about 50%.

flow begins, and rises rapidly to a relative value of over 60. As discharge pressure drops to 80% of inlet pressure, flow rises to over 80. Flow rises less rapidly as pressure at discharge continues to drop the inlet pressure staying constant. At the discharge pressure which is about 54.7% of inlet pressure (dotted line), the flow reaches its maximum value of 100. Then as we continue lowering the discharge pressure, we find that the flow rate stays constant. It will stay constant, even if the discharge pressure becomes a perfect vacuum.

The three curves in figure 1 show how the flow would vary through three nozzles with cross-sectional areas of say 1, 2, and 3 sq. in. You'll find that at all pressure ratios the flow values for 1A, 2A, and 3A are in exact ratio to the areas.

Critical Pressure Ratio

From the foregoing discussion on 'Steam Flow' let us define the following symbols:

Let P_1 = the initial steam pressure before the nozzle.

and P_2 = the final steam pressure after the nozzle.

Then we say that $\frac{P_2}{P_1}$ = pressure ratio of the nozzle.

In figure 1, we have said that as the discharge pressure decreased, the nozzle steam flow increased until the flow reached its maximum at the dotted line. We said that this maximum flow occurred when the discharge pressure P_2 was 54.7% of inlet pressure, P_1.

This is the same as saying:

$$\frac{P_2}{P_1} \times 100 = 54.7\% \text{ or } \frac{P_2}{P_1} = 0.547$$

$$\text{or } P_2 = 0.547\ P_1$$

The point at which the flow in the nozzle reaches its maximum is known as its critical point. The pressure ratio $P_2/P_1 = 0.547$ at this point is known as the <u>critical pressure ratio</u>. The pressure P_2 corresponding to the critical pressure ratio is called the <u>critical pressure</u> p_c.

By convention, when we say that the pressure ratio is less than critical we mean that the discharge pressure is less than 54.7% of inlet pressure. When we say that the pressure ratio is above critical we mean that the discharge pressure is greater than 54.7% of the inlet pressure.

(<u>Note</u>: the reader should not confuse the above mentioned critical pressure regarding nozzles, with the critical point of 3207 psi. describing a state condition of steam in the steam tables. These are two different subjects.)

Experiments have shown that the flow will remain constant at the maximum value of critical flow when the exhaust pressure is reduced below that corresponding to the critical pressure ratio. The critical pressure ratio is not the same for all gases and will vary for the same gas for changes in pressure and temperature. For steam the critical pressure ratio is different for the superheated and moist states. The value varies from 0.547 to 0.549 for superheated steam when the initial pressure P_1 changes from 0 psia. to 1500 psia., respectively, and from 0.580 to 0.594 for saturated or moist steam over the same pressure change.

Converging and Converging-Diverging Nozzles

Two different types of nozzles are generally used depending on the steam pressure.

(a) If the discharge pressure is greater than the critical pressure ($P_2 \geqq P_c$) the weight of flow is dependent on the discharge pressure and the nozzle has to be convergent as shown in figure 2(a). (P_c stands for critical pressure.)

The converging nozzle is also referred to as a non-expanding nozzle.

(b) If the discharge pressure is less than the critical pressure ($P_2 < P_c$) then the weight of flow is independent of the discharge pressure and the nozzle has to be convergent-divergent as shown in figure 2(b). The reason for this will be explained a bit later on. The converging-diverging nozzle is also referred to as an expanding nozzle.

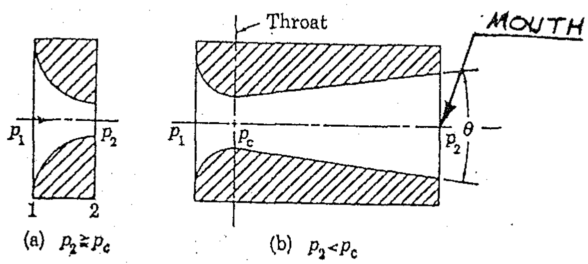


Figure 2

Figure 2. Nozzles. The convergent nozzle (a) needs only a well-rounded entrance with the area at section 2 proportioned to give the correct amount of discharge with a particular exit pressure P_2. The divergent nozzle (b) has a throat area designed to pass the required amount of steam, an angle θ of some 6 - 10°, and an area at the exit section 2 that results in neither overexpansion nor underexpansion.

By convention the narrowest part of any nozzle is referred to as its throat as labelled in figure 2(b). The outlet of any nozzle is referred to as its mouth. For a convergent nozzle then, the throat and mouth would be one and the same thing.

Velocity

We have said previously that an object that has weight and speed has kinetic energy. The steam jet leaving a nozzle has considerable kinetic energy. It can be expressed by the following equation:

$$\text{Kinetic energy, ft.lbs.} = \frac{\text{weight, lb. x velocity}^2\text{, ft./sec.}}{64.34}$$

Thus when steam is at rest in a boiler drum or pipeline, its kinetic energy is zero. But when it charges out through a nozzle to a lower-pressure, it acquires kinetic energy. The high pressure steam approaches a nozzle inlet with relatively low speed, so that its initial kinetic energy is relatively negligible. The question is: Where does the steam get its kinetic energy? If we measure the temperature of the steam before entering the nozzle, it will have a given value. Now if we could measure the temperature of the steam jet by means of a thermometer travelling at the same speed as the jet, we would find the temperature had dropped considerably.

So there's the clue. Temperature is an index of energy in the steam. The steam loses enthalpy in passing through a nozzle and acquires kinetic energy. In other words, enthalpy converts to kinetic energy. Here is the basic example of changing enthalpy to "available" mechanical energy.

For the steam nozzle, the general energy equation as described in the course on "Heat and Thermodynamics" T.T.2 level, reduces to:

$$\frac{V_1^2}{2gJ} + h_1 = \frac{V_2^2}{2gJ} + h_2$$

This assumes the process is a diabetic.

By transposing we get:

$$V_2 = \sqrt{2gJ(h_1-h_2) + V_1^2}$$

If the initial velocity is small, V_1 may be neglected and therefore the ideal velocity of the steam after an isentropic expansion is

$$(\sqrt{2gJ} = \sqrt{2 \times 32.17 \times 778} = 223.7):$$

$$\text{Ideal } V_2 = 223.7\sqrt{h_1-h_2}$$

However, due to friction of flow, the nozzle is not 100% efficient. It is more like 90% to 95% efficient. So we find that actual nozzle exit velocity for steam or gases is:

$$\text{Actual } V_2 = 223.7\sqrt{e_n(h_1-h_2)}$$

where: V_1 = steam velocity at nozzle inlet, ft./sec.

V_2 = steam velocity at nozzle outlet, ft./sec.

h_1 = enthalpy of steam at inlet conditions, Btu/lb.

h_2 = enthalpy of steam at outlet conditions, Btu/lb.

g = acceleration due to force of gravity, ft./sec/sec.

J = joule's constant, 778 ft. lbs. = 1 Btu.

e_n = nozzle efficiency, obtained by testing.

Example

Steam at 200 psia., 500°F expands through a nozzle with an efficiency of 95% to a pressure of 100 psia. Find the jet velocity V_2 and kinetic energy.

From steam tables h_1 = 1269 Btu/lb. and by using proper equations it can be shown that $h_2 \doteq$ 1204 Btu/lb.

$$\therefore V_2 = 223.7 \sqrt{e_n(h_1-h_2)} = 223.7\sqrt{.95(1269 - 1204)}$$

$$V_2 = 1{,}760 \text{ ft./sec.}$$

Kinetic energy for one pound steam is:

$$K.E. = \frac{w.V_2^2}{2g} = \frac{1 \times (1760)^2}{2 \times 32.17} = 48{,}200 \text{ ft. lb.}$$

This means that each pound of steam can do work of 48,200 ft.lb. by being brought to rest from a speed of 1760 ft./sec.

The power developed depends on the rate of flow in lbs. per minute:

$$\text{Power} = \frac{(\text{flow, lb./min. x work/lb. steam, ft.lb.})}{33{,}000}$$

If the steam flow through the nozzle is 1000 lb./min., the jet power would be:

$$\text{Power} = \frac{1{,}000 \times 48{,}200}{33{,}000} = 1{,}460 \text{ H.P.}$$

or since 1 H.P. = 0.746 kw.

Power = 1460 x 0.746 = 1,088 kw.

Sonic Velocity

At the critical pressure ratio where steam flow has reached a maximum for a certain inlet pressure the velocity of fluid at the nozzle throat is the same as the velocity of sound in the fluid. A drop in pressure below the critical pressure will therefore not be transmitted upstream, since pressure variations travel at sonic velocity, and hence no additional flow would be induced. For this reason the velocity attainable at the throat of a nozzle, no matter what the conditions, can never be higher than sonic velocity.

Figure 3(a) shows a convergent nozzle, through which is flowing wet steam. V_2 represents the nozzle exit velocity and V_s represents sonic velocity. When $P_2 = 0.7\ P_1$ the throat velocity is subsonic.

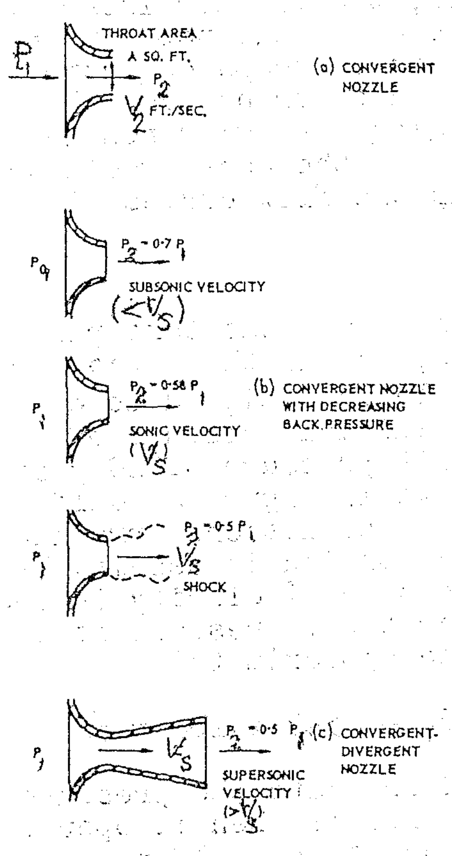


Figure 3. Flow through nozzles.

At $P_2 = 0.58\ P_1$ the throat velocity is shown as V_s or sonic velocity. (For wet steam 0.58 is the critical pressure ratio, as we had mentioned previously.)

Reason for Divergent Nozzle

We said previously that a converging nozzle should only be used if the exit pressure is equal to or greater than the critical pressure and a converging-diverging nozzle should be used when exit pressure is less than critical pressure.

There are times when a converging nozzle has been used when the exit pressure was less than critical.

Under this condition the steam will expand up to the throat as though the outlet pressure were at critical pressure. However, as soon as the steam has left the throat its pressure will become that of the surroundings, or less than critical pressure, with a corresponding increase in specific volume and velocity. The expansion from the inlet to the throat will be gradual, but the expansion after leaving the nozzle will be in the form of an explosion or shock as shown in figure 3 (b) where $P_2 = 0.5\ P_1$. Obviously the first part of the expansion, that from the inlet to the throat, would be far more efficient than the latter part where the explosion effect would cause a dissipation of energy.

Therefore, when it is necessary to design a nozzle for pressure ratios less than critical, it is customary to add a diverging section after the throat of the nozzle as shown in figure 3 (c). The extended nozzle passage beyond the throat allows the expansion to take place in an orderly fashion without the loss of energy encountered in the explosion. This then is the reason for the converging-diverging nozzle for pressure ratios less than critical.

You will notice that in figure 3 (c), even though the velocity at the throat can still not exceed sonic velocity, the velocity at the mouth is greater than sonic, or we say it is supersonic. This occurs because the steam is still expanding in the diverging part of the nozzle from the critical pressure at the throat to something less than critical pressure at the mouth. Hence, the further increase in velocity above sonic velocity. Referring back to the top part of figure 1 you will notice that the velocity curve keeps on increasing even for ratios less than critical. This can occur because that velocity was measured at the mouth of a converging-diverging nozzle. Had the velocity been measured at the throat, then the curve would have levelled off at the critical ratio, just as the flow curves did.

Nozzle Pressure for Flow Variations

In order to understand the several phenomena that take place in a nozzle under different conditions of flow, assume that a converging-diverging nozzle is designed to expand steam from an initial pressure of 100 psia. to an exit pressure of 20 psia. The nozzle shown in figure 4 would correspond to these conditions, provided the converging portion is assumed lengthened for the sake of clarity in the sketch. The critical pressure at the throat would be approximately:

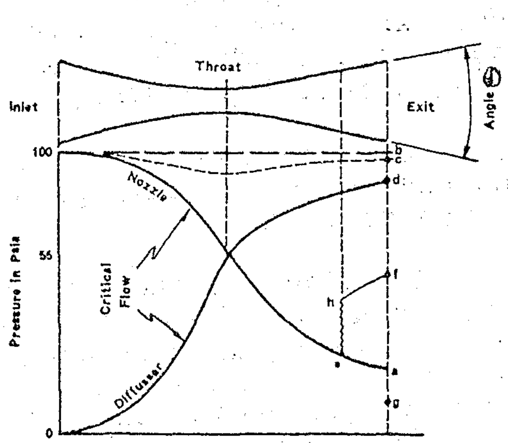


Figure 4. Pressure in a nozzle for various flow conditions.

$$P_2 = 0.547 \times P_1 = .547 \times 100 = 54.7 \text{ psia.}$$

Line 'a' would indicate the pressures throughout the nozzle when it operates under design conditions. If the outlet pressure were the same as the inlet pressure, 100 psia., there would be no flow and the same pressure would exist at all points in the nozzle, curve b.

If the outlet pressure were c, the throat pressure would not decrease to the critical pressure, and, therefore, the flow would not reach the maximum. This condition exists in a metering Venturi where the pressure drop from the inlet to the throat need be only sufficient to give an accurate measurement and should be kept low to reduce the loss as much as possible.

Before discussing the other possibilities for the exit pressure, let us assume that this same nozzle is to be used as a diffusor. Then the steam inlet velocity would be very high and the inlet pressure low. Instead of expanding, the steam will be compressed in the diffusor, and the kinetic-energy will be converted into pressure energy. This is the action that takes place in the diffusor of air ejectors used in connection with condensers. The outlet pressure would be at d, and the adjacent line would give the pressure history of the steam during its passage through the diffusor.

Returning again to the nozzle action, if the inlet pressure were 100 psia. as previously assumed and the outlet pressure d, the pressure gradient throughout the nozzle would follow the critical flow line starting at 100 psia. and decreasing to the critical pressure at the throat. From there on the pressure gradient would follow the diffusor curve to the outlet pressure of d.

So far all the cases considered for exit pressures have involved smooth orderly flow conditions. Such would not be the case, however, if the outlet pressure were at f. If the nozzle should have to operate with that back-pressure, the expansion would proceed along the path a until, after passing through the critical pressure point, it reached e which is a pressure less than the exhaust pressure. At that point a new phenomenon called compression shock would occur and the pressure gradient would rise suddenly and then proceed to f. The compression from h to f will take place as in a diffusor and will be gradual. An expansion of this type is called <u>overexpansion</u> and is caused by the nozzle mouth area being too large.

One more possible operating condition would be with the outlet pressure as at g. The pressure drop from a to g would occur at the mouth of the nozzle with very rapid expansion of the steam similar to that which would occur if the nozzle did not have the diverging section. This condition is called <u>underexpansion</u>.

Note that it is impossible for a converging nozzle to have compression shock.

Conditions such as a, f, and g are common in steam nozzles used in turbines, while conditions such as b, c, and d are very rare.

The ratio of the nozzle mouth area to the nozzle throat area is the nozzle expansion ratio, or, briefly, the <u>nozzle ratio</u>. For any set of conditions, it can be calculated by applying the continuity equation to the mouth and throat areas. Theoretically, this would give:

$$\text{Nozzle Ratio} = \frac{\text{mouth area}}{\text{throat area}} = \frac{V_t \times v_m}{V_m \times v_t}$$

where: V_t = velocity at the throat, ft./sec.

V_m = velocity at the mouth, ft./sec.

v_t = specific volume at the throat, cu. ft./lb.

v_m = specific volume at the mouth, cu. ft./lb.

Nozzle Flow

The weight of flow through a nozzle when exit pressure is above critical is calculated from the continuity equation as follows:

$$w = \frac{AV_2}{v_2} \text{ --- (1.)}$$

where: w = steam flow, lbs./sec.
A = throat area, sq. ft.
V_2 = nozzle exit velocity, ft./sec.(converging nozzle.)

v_2 = specific volume at nozzle exit, cu. ft./lb.

When the pressure ratio is critical or less, the maximum flow that can pass through a nozzle is given by:

$$w = 0.309\ A\ \frac{\sqrt{P_1}}{v_1}\ \text{lb./sec.} \text{ ------------------------------ (2.)}$$

where: w = steam flow, lbs./sec.
A = throat area of nozzle, sq. ins.
P_1 = inlet pressure, psia.

v_1 = specific volume at nozzle inlet, cu.ft./lb.

The above equation is derived by using higher forms of mathematics, which is beyond the scope of this course, and therefore is given without proof.

Notice that the flow in equation (2.) is independent of the exit pressure P_2. Therefore when the pressure ratio is critical or less, and if the throat area remains the same, then all we have to do to to increase flow through a nozzle is to raise the inlet pressure P_1.

How does this agree with what we said regarding figure 1 where we indicated that the flow remained constant with a ratio equal to or less than critical pressure ratio? We must remember that the conditions were different there. In that particular case we said that inlet pressure would remain constant and exit pressure would be reduced. Now, if we were prepared to raise the inlet pressure for the case in figure 1, we could increase the flow to a very large extent even though critical pressure conditions or less existed at the nozzle exit.

From the above discussion and equation (2.) we can deduce the following characteristic phenomena for nozzles:

If $P_2 > P_c$, the weight of flow is dependent on P_2.

If $P_2 \leqq P_c$, the weight of flow is independent of P_2.

where: P_2 = nozzle exit pressure, psia.

P_c = critical pressure, psia.

Equation (2.) tells us an important fact about the turbine inlet nozzles. The stop valve pressure is generally fixed at a certain value by design. Thus, if the turbine exhaust can handle the flow, then it can be seen that for the steam conditions given by P_1 and v_1 the maximum flow through the turbine, and hence the maximum power output is limited by the throat area of the first row of nozzles.

Supersaturation

For condensation to occur, it is necessary that there be nuclei around which the drops of water form. These nuclei may be dust particles, electrons, or other drops of water. If the steam is slightly superheated at the beginning of expansion and if it is nearly free of solid particles, there will be few nuclei to form the basis of condensation. If might be expected then that, if the change of state takes place very rapidly (the time of flow in a nozzle is of the order of 0.001 sec. or less) and if the nuclei are scarce, actual condensation might lag behind theoretical condensation. This is what apparently happens. Even though equilibrium conditions may suggest a moisture content of several percent at the throat, actual condensation may not be evident until some considerable further expansion.

Where condensation has not occurred according to theory, the steam is in an unstable state and is called supersaturated steam. Supersaturated steam is steam that has been cooled below the saturation temperature corresponding to its pressure. As soon as there is a very small amount of condensation, water particles are available as nuclei for other condensation, and the steam reaches a state of equilibrium almost instantly thereafter.

The degree of supersaturation is defined as:

$$\frac{\text{actual pressure of the vapor}}{\text{saturation pressure corresponding to actual temperature}}$$

Supersaturated steam is denser than steam in equilibrium at the same pressure. The velocity of supersaturated steam is less than the velocity of steam in equilibrium both having expanded from the same initial condition.

Nozzle Designs

About the simplest and most inexpensive non-expanding nozzle to manufacture is sketched in Figure 5. The cross section is circular, but since the nozzle is at an angle with the blades, the mouth is elliptical. The hole is drilled and reamed. The angle α is the nozzle angle. To obtain the most energy from the steam jet, the blade must not be shorter than the height of the nozzle; usually it is made slightly taller than the height of the nozzle. It will be shown later that the angle α should be as small as possible; generally it is about 15 degrees and in exceptional cases as large as 20 degrees. The more that this angle is decreased, the longer will be the length of the nozzle and the greater the friction loss; but the nozzle mouth will also become wider, causing a very bad flow condition. This bad effect is sometimes lessened by reducing the distance between adjacent nozzles until the mouths overlap.

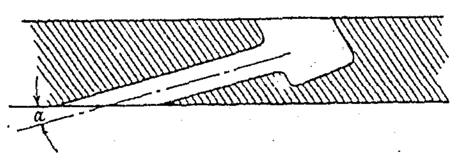


Figure 5. Sketch of a reamed non-expanding nozzle.

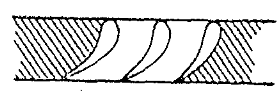


Figure 6. Sketch of a curved foil nozzle.

From aerodynamics it can be determined that the foil shape (teardrop) would be the most efficient nozzle design. The inlet is well rounded and the outlet sharp, Figure 6. The angle α is again about 14.5 or 15 degrees. The flow characteristics of this type of nozzle are very favorable, and this design is also used for reaction blades which will be explained in the next lesson.

Expanding nozzles could be designed so that the diverging portion would provide uniform decrease in pressure with respect to length, or uniform drop in enthalpy with respect to length, or uniform increases in specific volume with respect to length. In the first two cases, the mouths would flare rapidly, which would direct the jet of steam over a wide area and would make it ineffective. In all cases the cost of manufacturing such nozzles would be prohibitive with no particular advantages.

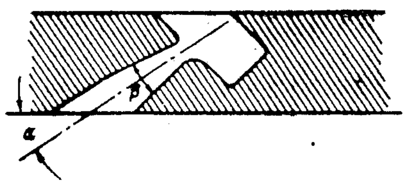


Figure 7. Sketch of an expanding nozzle.

The common design of expanding nozzles, shown in Figure 7, is inexpensive. The passages are reamed after drilling. Angle α is the same as for the reamed non-expanding nozzle, and angle β, the divergence angle, depends on the nozzle ratio. This angle should not be over 15 degrees as the steam will not follow the sidewalls for greater angles. If the steam were to lose contact with the sidewalls, the nozzle would become a non-expanding nozzle with the pressure ratio less than critical and would have the loss of explosion as in the non-expanding nozzle. The cylindrical extension shown in figure 7 is to guide the steam. Trapeziform and rectangular nozzles have been used in the past, but they were more inefficient than other designs.

Milled nozzles for efficiency are shown in figure 8.

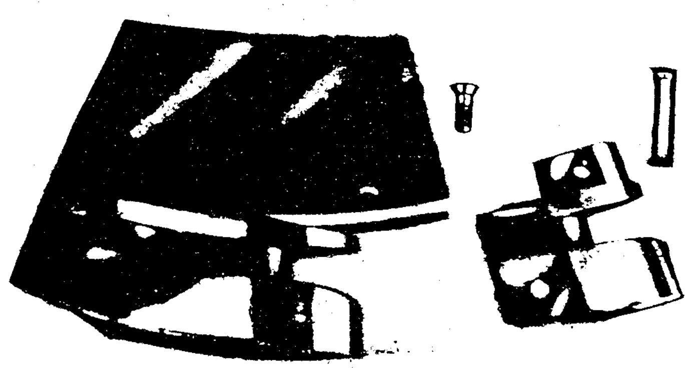


Figure 8. Details of built-up nozzle construction.

Summary

In this lesson on nozzles we have covered a fairly wide field and it would be advantageous to summarize the most important points. This is done below:

a.) <u>Converging and Converging-Diverging Nozzles</u>

Both Nozzles have the following points in common:

1. Inlet pressure held constant--decreasing exit pressure will increase velocity and flow through the nozzle until critical pressure is attained at which time flow and velocity at the the throat will no longer increase even though the exit pressure is decreased below critical pressure.

2. Increasing inlet pressure with the situation in #1--i.e. with exit pressure less than critical, will definitely increase the weight of flow.

3. Velocity of flow at mouth is:

$$V_2 = 223.7 \sqrt{e_n(h_1-h_2)}$$

4. Weight of flow when pressure ratio is greater than critical:

$$w = \frac{AV_2}{v_2} \quad \text{lb./sec.}$$

5. Weight of flow when pressure ratio is critical or less is:

$$w = 0.309\, A\sqrt{\frac{P_1}{v_1}} \quad \text{lb./sec.}$$

6. If exit pressure P_2 is greater than critical pressure P_c, then weight of flow is dependent on P_2.

   If exit pressure P_2 is equal to or less than critical pressure P_c, then weight of flow is independent of P_2.

7. Increasing throat area will increase weight of flow. Doubling area doubles flow, tripling area triples flow, etc.

b.) <u>Converging Nozzle</u>

The following phenomena are characteristic of a converging nozzle only:

1. The velocity at the throat can never exceed sonic velocity.

2. Shock waves are established at exit in the case of converging nozzles if the exit pressure is less than critical pressure.

3. A converging nozzle is a non-expanding nozzle.

c.) Converging-Diverging Nozzle

The following phenomena are characteristic of a converging-diverging nozzle only:

1. The velocity at the throat can never exceed sonic velocity. Velocity at the mouth, however, can attain supersonic values when operating below the critical pressure ratio.

2. When the nozzle mouth area is too large, compression shock will occur and this is called overexpansion. When nozzle mouth area is too small underexpansion occurs.

3. A converging-diverging nozzle is an expanding nozzle.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-4 - Nozzles

A - Assignment

1. Define critical pressure ratio and critical pressure for a nozzle.
2. Draw a converging nozzle and a converging-diverging nozzle.
3. To what part of a nozzle do the terms "throat" and "mouth" refer?
4. Why are nozzles required in steam turbines?
5. Steam with a heat content of 1285.3 Btu/lb. expands through a nozzle with an efficiency of 95% to a pressure and temperature at which it has a heat content of 1200 Btu/lb. Calculate the nozzle discharge velocity. Assume inlet velocity is negligible.
6. Under what conditions is a convergent-divergent nozzle used? Why?
7. Define nozzle ratio.
8. When is the weight of flow dependent on discharge pressure? When is it not?
9. What is meant by supersaturation?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-5 - Turbine Blading

0.0 INTRODUCTION

At the T.T.3 level we described in a general way how a turbine worked. This lesson will now consider more closely how the turbine transforms the heat energy in the steam to mechanical energy at the turbine coupling and will describe different types of Blades.

1.0 INFORMATION

It has already been mentioned that the high velocity steam gives up its kinetic energy to the turbine wheels and thus the spindle revolves. To illustrate the enormous speeds at which the steam travels after expansion, take a simple numerical example. Starting with steam at 600 lbs. p.s.i.a. and a temperature of 800°F it will be seen from the steam tables that such steam has a total heat content of 1,407 B.T.U's per lb. If it is now expanded down to 300 p.s.i.a. at 620°F it would have a total heat of 1.326 B.T.U's per lb. The difference between these two total heats which is 81 B.T.U. per lb. of steam has been spent in increasing the velocity of the steam. In the expansion considered the ideal velocity would increase from zero to:

$$V = 223.7\sqrt{h_1 - h_2}$$

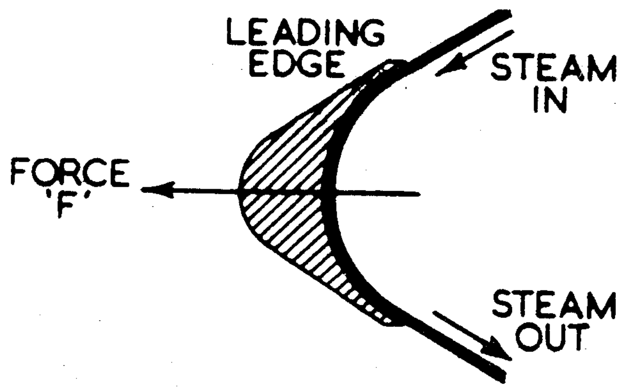
$$= 223.7 \times \sqrt{81} = 2000 \text{ ft/sec.}$$

or 1,365 m.p.h. This high velocity steam is applied to the leading edge of the turbine blade as shown in Figure 1; it glides along the blade surface and leaves the trailing edge in the direction shown. The shape and angle of the blade and the initial direction of the steam are so designed that this takes place without shock to the blade.

Thus each pound of steam with a velocity of 2,000 ft. per sec. meets the blade and due to blade profile is turned back in approximately the opposite direction at almost the same speed as shown in Figure 1. Hence, a force F in the direction shown is exerted on the blade, transmitted through the wheel to the spindle and thus to the turbine coupling to drive the generator.

This force F is exactly similar to that exerted on the hand if a weight is whirled around at the end of a piece of string. (See Figure 1.) This force is known as centrifugal force which means a force exerted away from the centre.

We have explained previously that it is possible to reduce the steam from a high pressure to a low pressure all in one stage and this would result in a single stage turbine. However, the velocity of steam would be very high. It can be proved that maximum efficiency of an impulse turbine is obtained when the blade



SECTION OF TURBINE BLADE

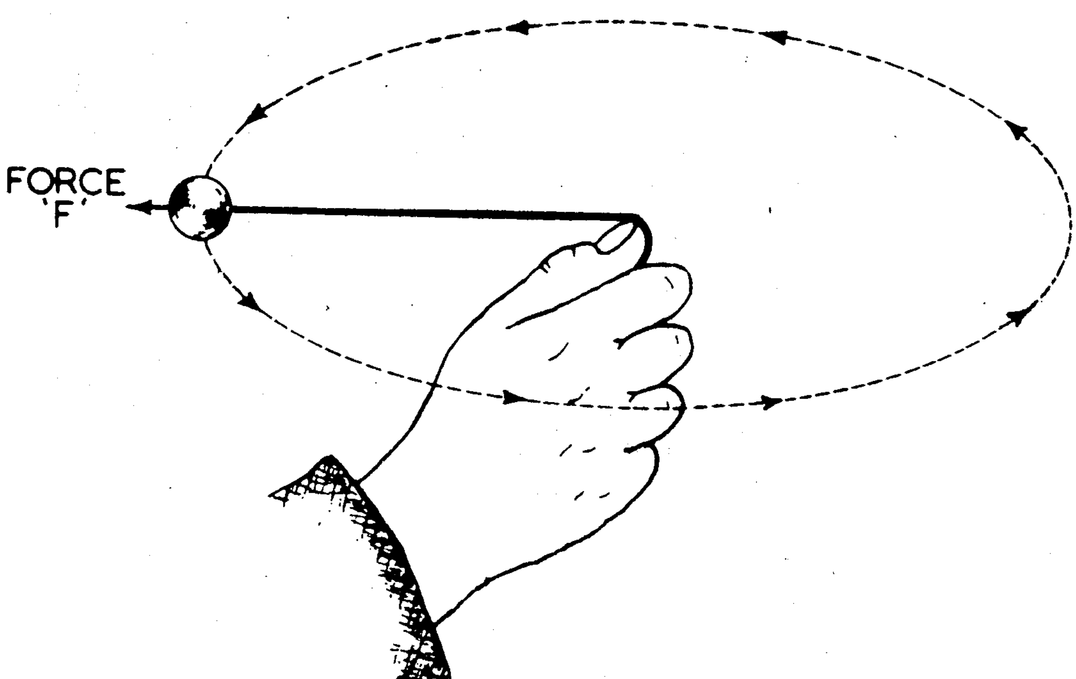


Figure 1 Centrifigual Force

speed is approximately half the speed of the steam. The blade speed depends on two factors: 1.) the speed of the shaft, and 2.) the mean blade diameter. If the steam velocity is very high it may be that for maximum efficiency the blade would have to travel at such a speed that it would tear apart due to centrifugal force. So for this reason we said that the steam is expanded in a number of stages. The machine rpm. is fixed at 3600 or 1800 by the system frequency of 60 cycles per second. Greater efficiency occurs at the higher speed.

In general there are two main types of steam turbine blading design, which are known as:

<u>Impulse Type</u>

<u>Reaction Type</u>

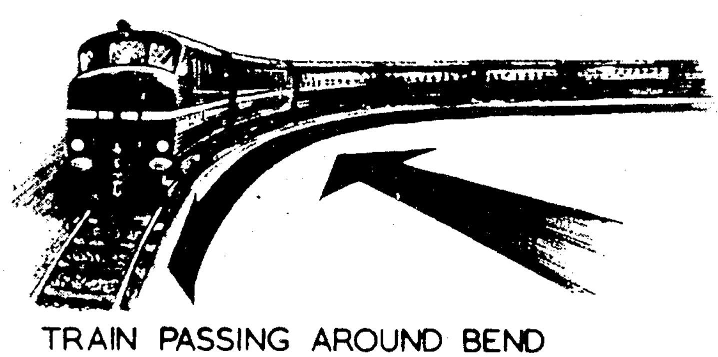
The turbine can have either a curtis stage or a rateau stage as the first stage in addition to normal stages. These terms will be explained later on.

Having decided that it is necessary to obtain high velocity steam to drive the turbine, then the way this high velocity steam is obtained determines whether the machine is impulse or reaction, or a combination of both.

<u>Impulse Type Machines</u>

As previously shown the blade obtains no motive force from either the static pressure of the steam or from any impact of the jet. The blade is so designed that the steam will glide on and off without any tendency to strike it, thus giving an <u>IMPULSE</u> to the blade which causes the spindle to rotate.

The principle is exactly similar to that of a train gliding around a railway curve. The train exerts an outward thrust on the railway line due to centrifugal force but at no point does the train actually strike the rail. (<u>See Figure 2.</u>)



<u>Fig. 2</u> Impulse Effect

In an impulse machine the steam is expanded to the required pressure in FIXED NOZZLES and the high velocity steam does work on the MOVING BLADES as previously described. The essential feature of an impulse turbine is that ALL the pressure drop occurs in the nozzles there is NO pressure drop over the moving blades themselves.

Since there is no difference in pressure between the two sides of any moving blade there is little or no tendency for steam to leak past the blade tips and so the problem of sealing between the two sides of the wheel is considerably simplified. Since the leakage is negligible all the steam expanded will do useful work.

Reaction Type Machines

In the reaction type of machine the principle is slightly different. Work is still being done by the impulse due to the reversal of direction of the high velocity steam in the moving blades, but the FIXED AND MOVING BLADES are so designed that the steam expands as it passes through BOTH, thus giving, in addition, a REACTION EFFECT due to the expansion of the steam through the moving blades.

A simple example of the use of the reaction effect is shown by the ordinary rotating lawn sprinkler. Here the water jets cause a reaction effect on the nozzles of the sprinkle and thus cause them to rotate around the pivot. (See Figure 3)

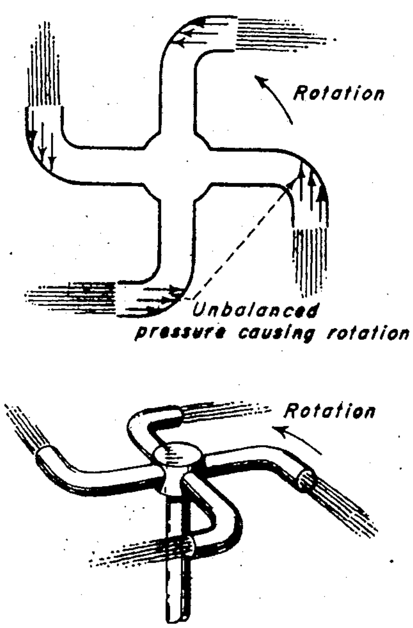


Fig. 3 Unblanced-pressure forces acting in a direction opposite to jet flow in a lawn sprinkler make it rotate. These are the so-called "reaction" forces.

Another example of this reaction effect is felt by the fireman holding a high pressure water hose and the same effect is used to drive a jet aeroplane or a rocket.

As we said in the nozzles of an impulse turbine the pressure drop is only in the stationary blades. However, these stationary blades also experience a reaction force. Hence, there is a force acting in the direction opposite to steam flow at the curved wall of the nozzle. This force would tend to cause the stationary blades or diaphragms to rotate in a direction opposite to the direction of the shaft. For this reason the diaphragms have to be securely anchored to the to the casing which, in turn, is securely anchored to the foundation. In a reaction turbine part of this force is put to work in the moving blades, however, the rotating force on the stationary blades exists also.

Since in a reaction type machine a pressure also occurs across the MOVING blades, it is necessary to provide effective sealing at the blade tips. This must be done to prevent leakage of steam past the shrouding of the wheel and consequent loss in efficiency, particularly at the high pressure end of the machine. In spite of this apparent disadvantage, the reaction type machine has certain advantages. The steam to blade speed ratio for maximum efficiency is not the same as for the impulse machine and, under certain circumstances, a reaction type is more efficient in practice than the corresponding impulse type would be. The majority of modern machines use both impulse and reaction principles. The general tendency is to use impulse at the high pressure end, where the question of sealing is important and reaction at the low pressure end, where the sealing is not so important because of the large volumes of steam involved.

Considerable work has been done by manufacturers on blade shapes or profiles and now blades in which the expansion is partly impulse and partly reaction are becoming much more common. In fact some blade manufacturers now make the low pressure blading with a ratio of impulse varying from the tip to the root of the blade thus further reducing tip leakage at the low pressure end.

In a machine containing both impulse and reaction stages it is not usually possible to tell by inspection which are the impulse and which are the reaction stages.

Mehtods of Reducing Rotor Speed

As already stated, it is necessary to use many rows of blades in the turbine in order to efficiently absorb the kinetic energy of the steam. <u>This process of absorbing the energy in several steps is known</u> as <u>Compounding</u>, and the following are the four main methods of thus reducing the most efficient rotor speed down to 3,600 r.p.m. or lower. A turbine designer can use two techniques for compounding:

1. <u>compounding steam pressure</u> through several rows of stationary nozzles or

2. <u>compounding steam velocity</u> through 2 to 4 rows of moving blades.

Before we proceed any further let us reword the above two techniques in another way and at the same time introduce the names by which these two types of stages are known.

Rateau Stage named after Professor Rateau is one turbine stage and consists of one nozzle or a row of stationary nozzles admitting steam to one row of moving curved blades. A series of rateau stages are used when the technique of pressure-compounding is employed in a turbine.

Curtis Stage named after its inventor C.G. Curtis also consists of one row of stationary nozzles but the high velocity steam jet thus formed acts on two or three rotating rows of blades, (with stationary guide blades in between moving blades) before leaving the turbine or before entering another set of nozzles. The curtis stage is used when the technique of velocity-compounding is employed.

The moving blades of a curtis stage are attached to a wheel which has a bigger diameter than the neighboring rotor wheels and is known as a velocity wheel. Fig. 4 illustrates a velocity wheel, in the foreground.

Both the rateau stage and curtis stage are impulse type of stages. That is the pressure drop all takes place in the stationary nozzles and there is no pressure drop across the moving blades.

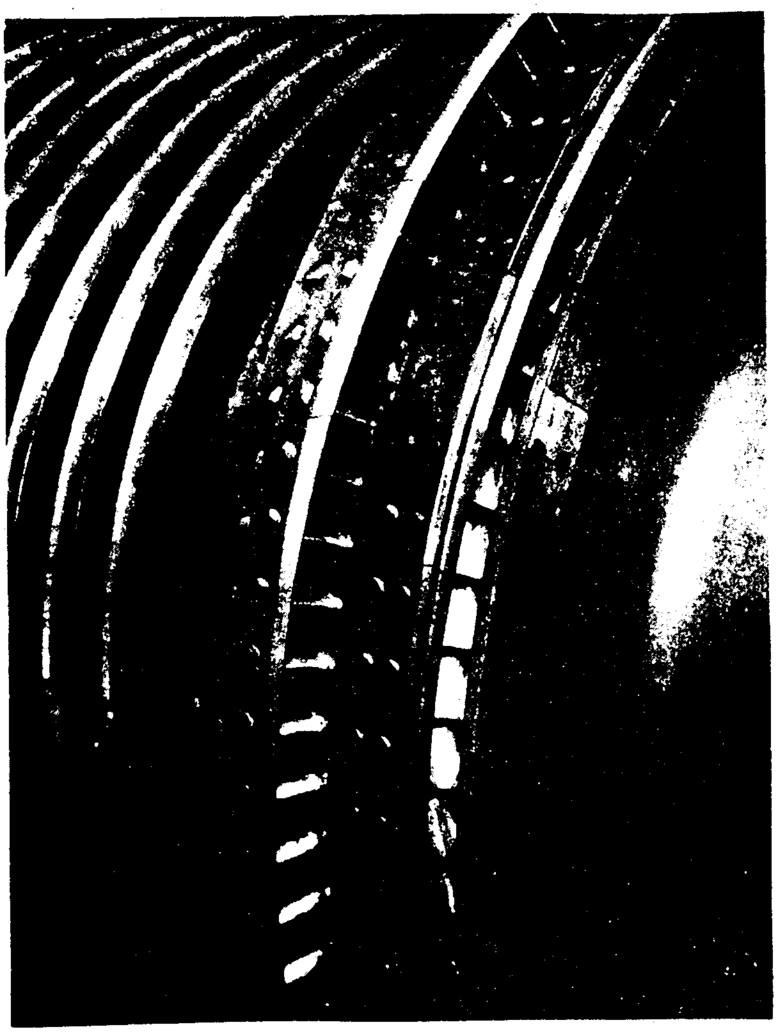


Figure 4

Figure 4. Part of a turbine rotor. The first two rows of blades are attached to the velocity wheel forming part of the curtis stage.

Compounding for Pressure using (Rateau Stages)

In this type of compounding, the rows of moving blades are keyed to the shaft in series, but in between each there is a row of fixed nozzles and the total pressure drop of the steam is divided equally between all of the fixed nozzles. This is illustrated in figure 5. The steam from the boiler is passed through

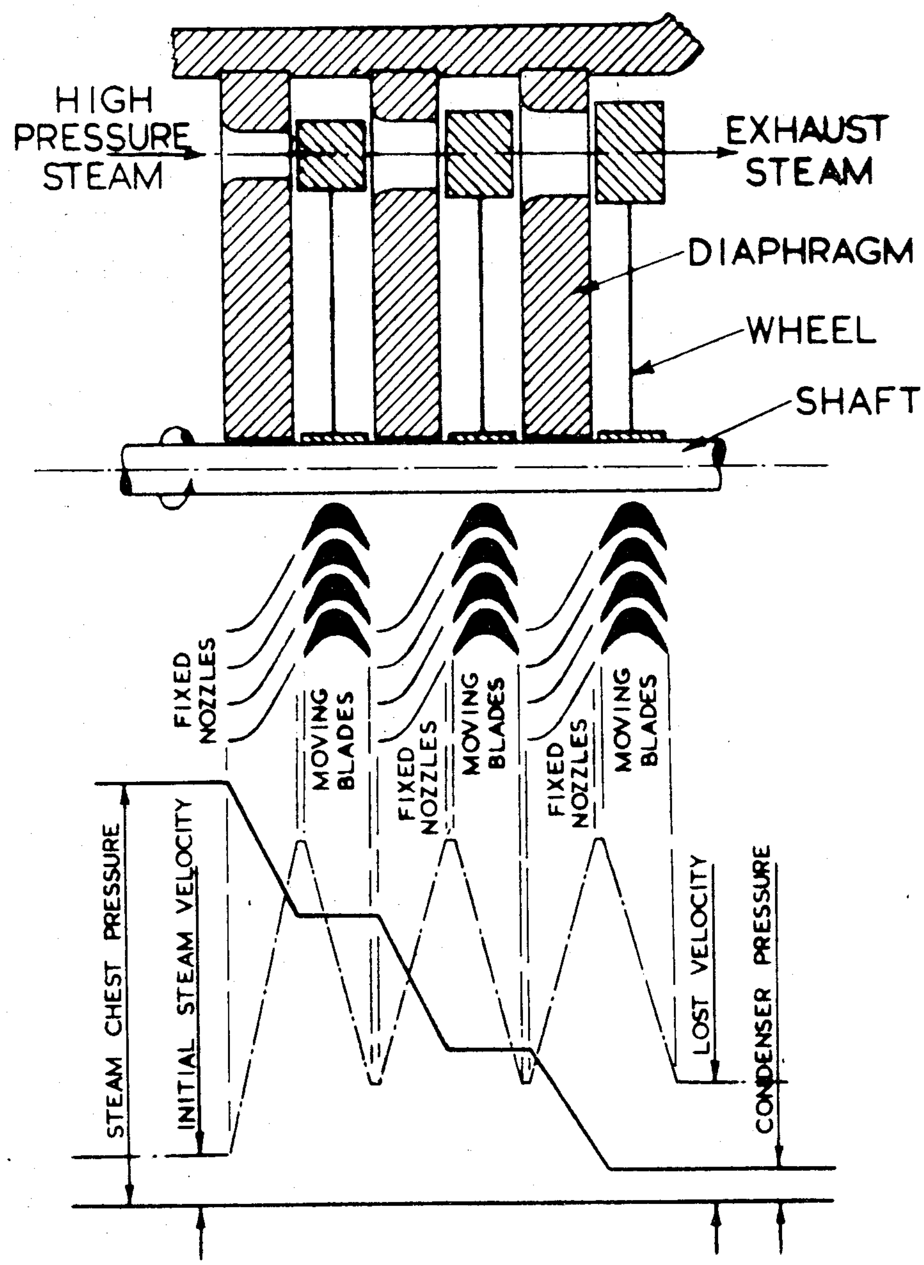


Figure 5 Compounding for Pressure.

the first row of nozzles, in which it is only partially expanded. It then passes over the first row of moving blades where most of its velocity is absorbed. From this row it exhausts into the next row of fixed nozzles and is again partially expanded, thus absorbing a further portion of its total pressure drop. The velocity obtained from the second row of nozzles is absorbed by the next row of moving blades. This process is repeated in the remaining rows until the whole of the pressure drop has been absorbed. Since no pressure drop takes place across the moving blades as illustrated in figure 5, it represents an impulse turbine.

Compounding for Velocity (using Curtis Stage)

Rows of moving blades separated by rows of fixed blades are keyed in series on the turbine shaft as shown in figure 6. It illustrates a small turbine which consists of only one curtis stage. The steam is expanded through nozzles from boiler pressure to the condenser pressure all in one big step as shown in the pressure-velocity diagram. The resulting high velocity steam jet is passed over the first row of moving blades. Only a portion of this high velocity is absorbed by this row of blades, the remainder being exhausted onto the next row of fixed blades. The fixed blades are shaped so that they have about the same cross-sectional area from inlet to outlet and therefore do not produce any reduction in pressure, although there is a slight reduction in velocity due to friction losses. These fixed blades are therefore called guide blades because they merely change the direction of the jet without appreciably altering its velocity. The jet then passes on to the next row of moving blades and a further portion of the velocity is absorbed by them. This process is then repeated as the steam flows over the remaining pairs of blades until practically all the velocity has been absorbed.

This is also an impulse type of turbine. The curtis stage is nowadays only used as the first high pressure stage of a turbine, where a large pressure drop is required for the first row of nozzles. A large drop in pressure also results in a large drop in temperature so that using the curtis stage as the first stage in the turbine protects the casing and rotor from higher pressure and temperature conditions.

A velocity compounded or curtis stage uses approximately the same heat drop as four normal impulse stages. Therefore it is used to provide a shorter and cheaper turbine. There is, however, some sacrifice in efficiency.

Curtis stages have been built with as many as four rotating rows of blades. However, it can be shown that the ratio of work obtained from each row is in the proportion 7:5:3:1. In other words the last row does only 1/16 of the total work of the stage. Two-row Curtis stages are common.

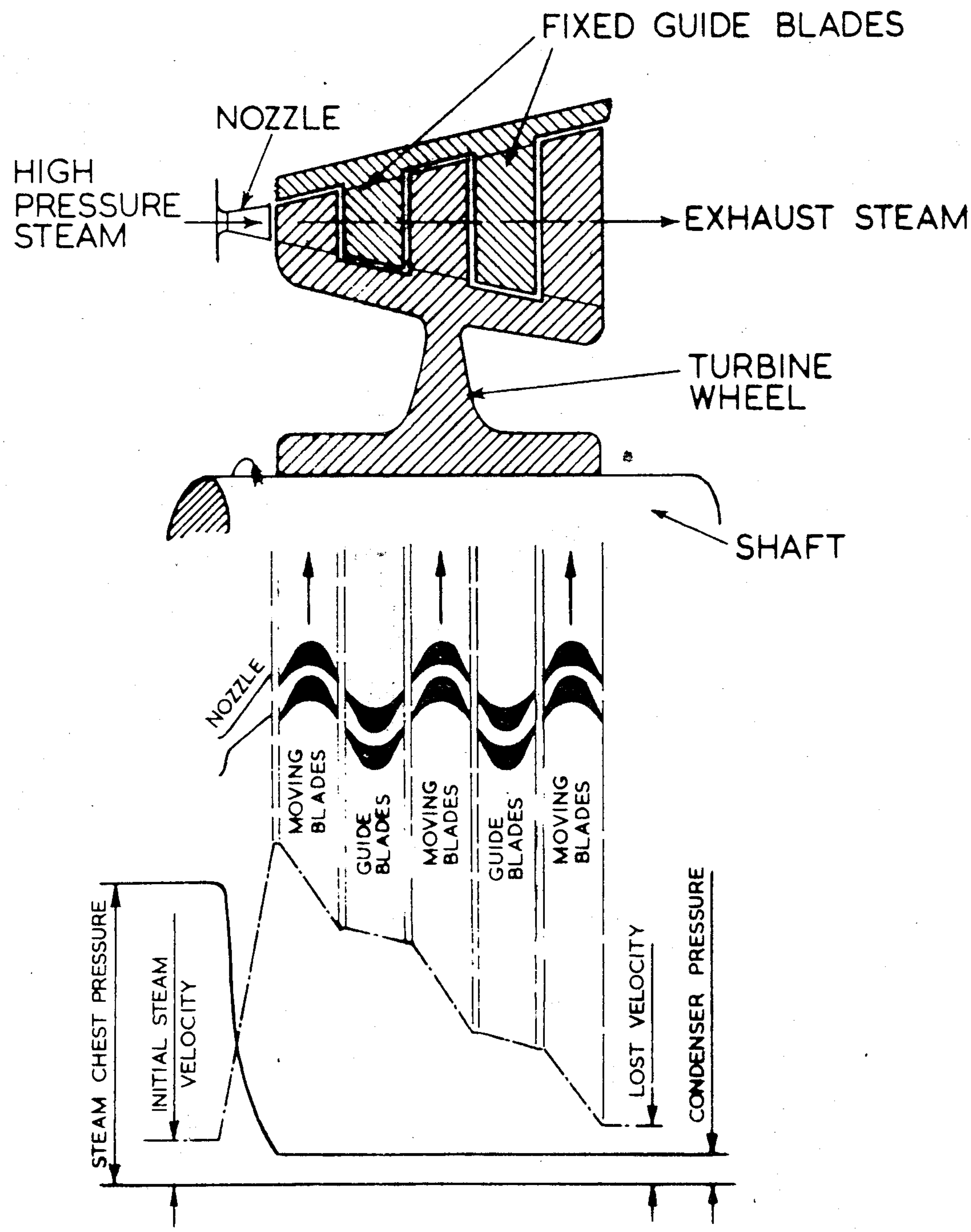


Figure 6 Compounding for velocity.
(Curtis Stage)

Notice that the rateau stage, figure 5 uses convergent nozzles, whereas the curtis stage uses convergent-divergent nozzles. This implies that for the rateau stage the pressure ratio is above critical and hence the velocity is either sonic or less than sonic. The convergent-divergent nozzles imply pressure ratios below critical with supersonic velocities.

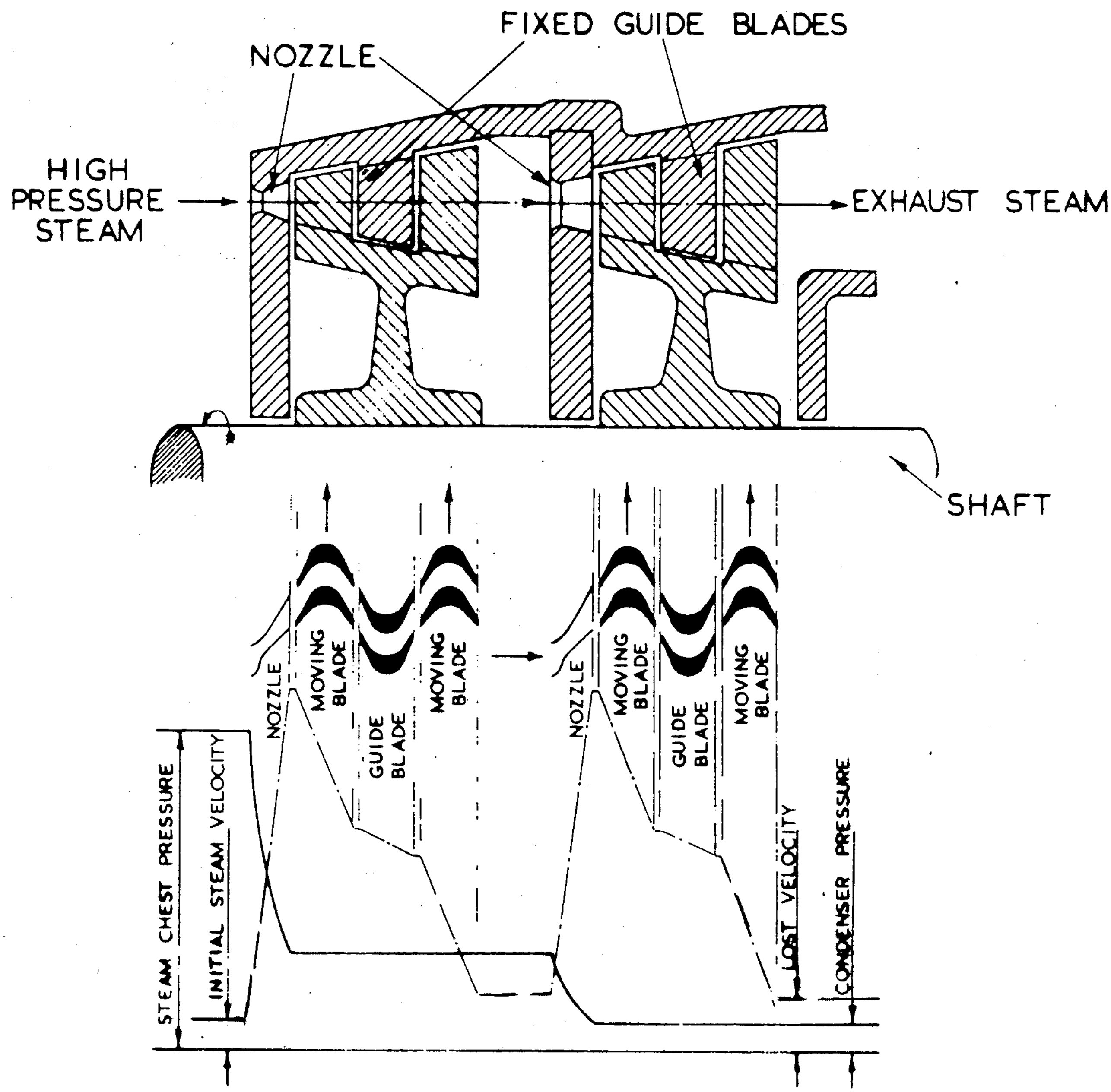


Figure 7. Compounding for pressure & velocity.

Pressure-Velocity Compounding

Pressure-Velocity Compounding is a combination of both the previous methods and has the advantage of allowing a bigger pressure drop in each stage and so less stages are necessary. Hence, for a given pressure drop the turbine will be shorter. The curves of pressure and velocity for this type of turbine are shown in Figure 7.

It will be noticed that the diameter of the turbine is increased at each stage to allow for the increasing volume of steam. A row of nozzles is fitted at the beginning of each stage but the pressure is constant over any one stage and the turbine is thus an Impulse Turbine.

Reaction Turbine

The reaction turbine was designed and developed by Sir Charles Parsons from England. As described previously in this type of machine the pressure drop takes place gradually over the fixed and moving blades and since the pressure falls over the moving blades, the turbine is a reaction turbine.

Figure 8 shows the pressure and velocity diagrams for a reaction turbine. The pressure line falls almost continuously as the steam passes over the rows of blades. The steam never reaches a high velocity as it is continuously expanding at a slow rate and at the same time the velocity is being absorbed by the moving blades. The blade velocity is consequently relatively low.

In one reaction stage the enthalpy drop across the row of moving blades is about the same as across the row of stationary blades--i.e. the stage uses equal degrees of impulse and reaction. When this occurs the stage is called a 50% reaction stage. This results in the fixed and moving blades having identical shape.

At this point we want to define the term velocity ratio which plays a significant role in blade design:

$$\text{Velocity ratio, } \rho = \frac{\text{blade velocity , U}}{\text{steam velocity , V}}$$

We said for the impulse turbine maximum efficiency occurred when the steam velocity was about twice the blade velocity. This means that the velocity ratio would be about 0.5 or to be exact it should be 0.48 for an impulse turbine. The velocity ratio for maximum efficiency in a reaction turbine is quite a bit higher--being around 0.80 to 0.85. This means that blade velocity is almost equal to steam velocity.

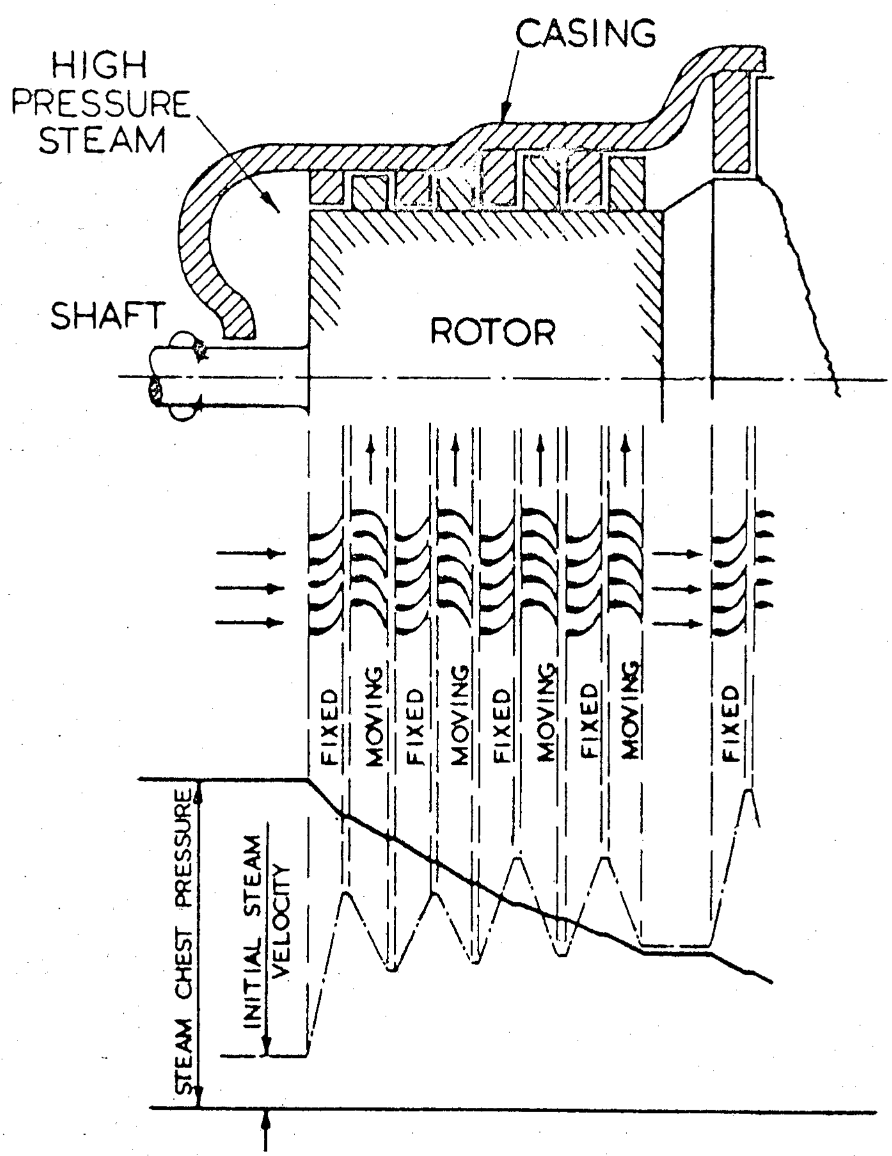


Figure 8 Reaction Turbine

Owing to the comparatively high velocity ratio for a reaction turbine, only a small heat drop can be accommodated per blade row. (If steam pressure drop were large resulting in high steam velocity then blade speed might exceed its limit.) This means that a larger number of stages are required for a reaction turbine than for an impulse turbine. Even though pressure drop across each row of blades is low, it is necessary that the tip clearances be small, to prevent excessive leakage losses. This has been achieved in small turbines by a feature referred to as "end-tightening." For this the axial clearance can be set to any desired value with the machine cold, by moving the shaft longitudinally and, if desired, adjusted while the machine is running by means of an adjusting gear fitted to the thrust block. A supplementary seal is provided by a radial fin which is machined complete with the shroud. With solid couplings and double-flow cylinders, end-tightening is not feasible, and reliance has to be placed on fine radial clearances.

As mentioned previously the dummy piston is used on the reaction turbine only, to balance steam thrust on the shaft.

Impulse-Blade Efficiencies

The efficiencies used by turbine designers are determined by test and a detailed discussion of them is beyond this course. However briefly the factors that must be considered are:

(1) Nozzle height. A high nozzle is usually more efficient than a low one.

(2) Over-and underexpansion.

(3) Steam displacement. This factor must be considered when there is partial admission, i.e. when nozzles are used around only part of the nozzle ring. As the blades travel through the arc where there are no nozzles, they carry stagnant steam with them. Therefore, when a jet of steam enters the blades, it must displace the stagnant steam before imparting energy to the blades.

(4) Mean diameter. It has been determined by tests that a large diameter wheel will be slightly more efficient than a small one.

(5) Moisture. Moisture in the steam will decrease efficiency because:

(a) the moisture moves slower than the steam and will strike the back of the blade and retard its forward motion.

(b) the moisture reduces the quantity of steam actually doing work.

Also since moisture in steam has an erosive effect on the blade edges, it is generally not allowed to be greater than about 13%. The moisture is estimated empirically to reduce the efficiency ½% for each 1% moisture present in steam.

Reaction-blade Efficiencies

The factors involved in efficiency of reaction stages are similar to those for impulse stages. These are:

(1) Blade leakage

(2) Blade height

(3) Blade width

(4) Trailing edge thickness

(5) Moisture in steam. The moisture correction for reaction blades may be higher than for impulse blades--perhaps 1¼% for each 1% moisture in steam after isentropic expansion.

Comparison

Figure 9 shows an efficiency comparison of the curtis rateau and reaction stages for various velocity ratios. This shows that the reaction stage has a higher efficiency than either of the other stages but its velocity ratio is quite a bit higher for maximum efficiency.

Another interesting comparison of the efficiencies of curtis, rateau and reaction stages is obtained by plotting the efficiencies against enthalpy drop instead of velocity ratio for an assumed wheel velocity. This is shown in figure 10. Here you can see for maximum efficiency the reaction stage can have an enthalpy drop of only around 10 Btu/lb., while for the curtis stage enthalpy drops of 160 Btu/lb. can be allowed for and still maintain relatively high efficiency.

Therefore, as we said before, a reaction turbine is more efficient than an impulse turbine but it requires a greater number of stages for the same output. This results in a longer turbine.

Twisted Blades

When blade height becomes a significant part of the total stage diameter, then blade speed will vary considerably from root to tip. Thus the ratio of blade speed to steam speed changes over the length of the blade. Then a twisted blade must be used as shown in figure 11.

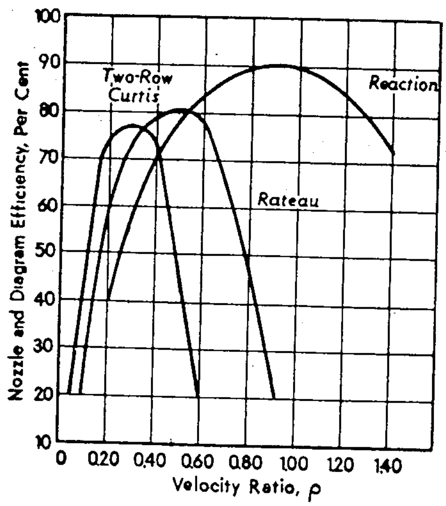


Figure 9 Nozzle and blade efficiencies.

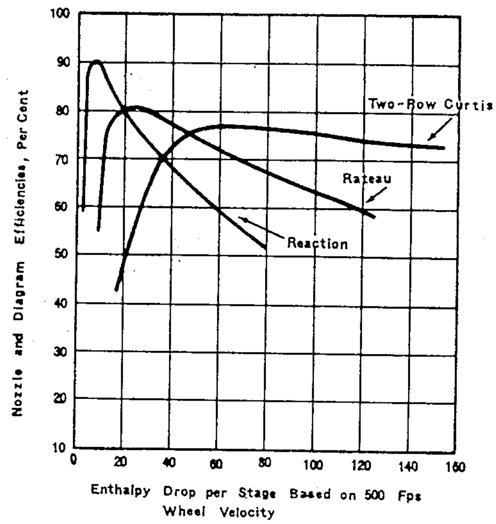


Figure 10 Nozzle efficiency vs. enthalpy drop.

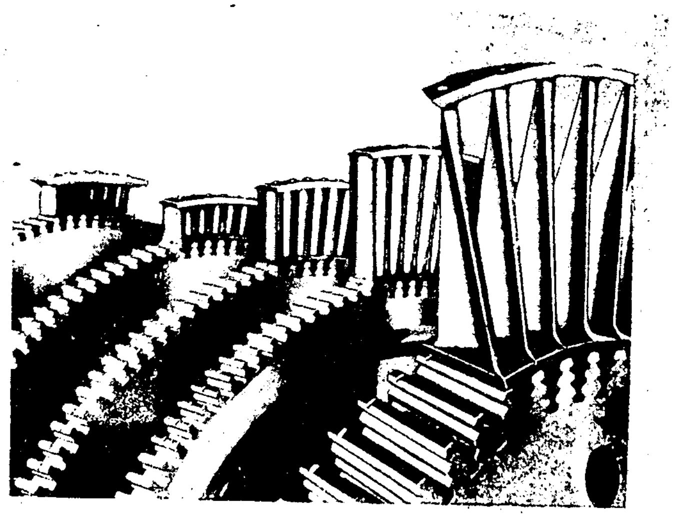


Figure 11. Shows shape of twisted blades; side entry blades dovetail into rotor discs to transmit force to turbine shaft. Shrouds tie groups of blades together and help prevent steam leakage.

Blades are twisted for two reasons:

(1) To accommodate changing velocity ratio. Large turbine impulse-type stages generally have some reaction which varies along the height of the blade. It is only at the root section where there are pure impulse conditions and at the tip of a low pressure blade there maybe as much as 50% reaction. Thus the object of radial variation of blade curvature is to match the nozzle exit steam speed to peripheral blade speed, at all radii, and in this way maintain efficient velocity ratios.

(2) To confine the steam to the circular annulus of the blades. Due to the centrifugal effect there would be a tendency for the steam to move radially outward from the rotor. By making the top part of the blade to produce reaction and the bottom part an impulse effect, there is a radial pressure difference which tends to cause the steam to move radially inward. The radially outward and inward tendencies thus balance each other, in the case of the twisted blade, and steam flows in a more or less straight line, parallel to the shaft, towards the exhaust end.

Shrouding

The strip of metal joining the tips of the blades in figure 11 is called a shroud. It serves to tie all the blades together at the tips and keep them equally spaced. This adds strength to the blades. It also helps to confine the steam within the space of the blades and thus reduces leakage.

Blade Root Fixing

The part of the blade attached to the rotor disc is called a root. Figure 11 shows one type of blade root known as a side entry dovetail root. This is a strong method of fixing blades to rotor discs. Figure 12 shows other types of roots commonly used. The diagram is self-explanatory. The axial fir tree root is another name for the side-entry dovetail root.

Lacing

To dampen vibration long blades may be laced together in batches, as shown in figure 13, so that those undergoing flow impingement are tied to others passing between steam jets. The lacing holes are however, a source of weakness and the wires disturb the steam path.

Stellite Strips

In the low pressure part of the turbine condensation droplets may erode the leading edges of the moving blades and therefore "Stellite" protection strips are often brazed to the leading edges as shown in figure 13.

Stellite is a very hard, wear resistant metal. However, because it is hard it is also brittle and therefore it is not advisable to make the whole blade of stellite.

Baumann Exhaust (multi-exhaust stage)

A special type of exhaust blading invented by a Dr. Baumann, used by one manufacturer to achieve greater exhaust area without undue lengthening of blades of the last moving row is shown in figure 14. It uses two-tier blades for the second last stage. Steam from the outer tier exhausts directly to the condenser.. The net result is that, without lengthening the blades of the last row, but by lengthening those of the previous row the leaving loss can be appreciably reduced.

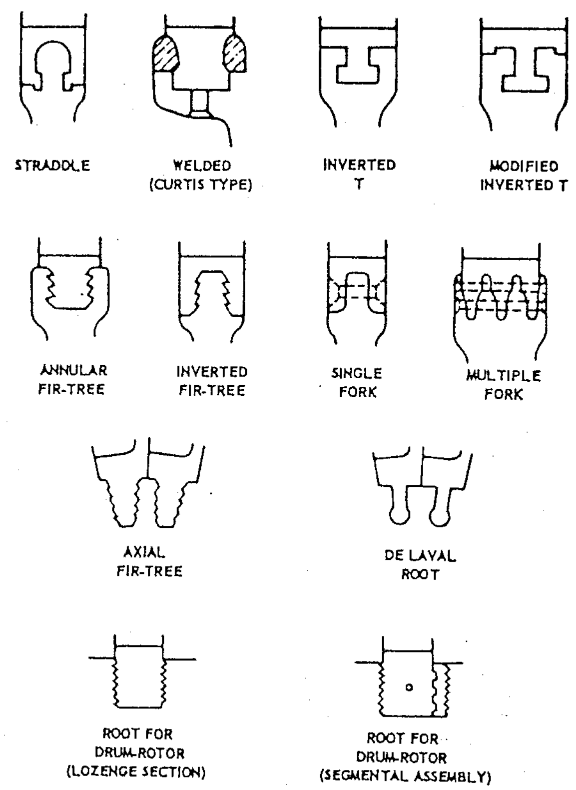


Figure 12 Types of blade root.

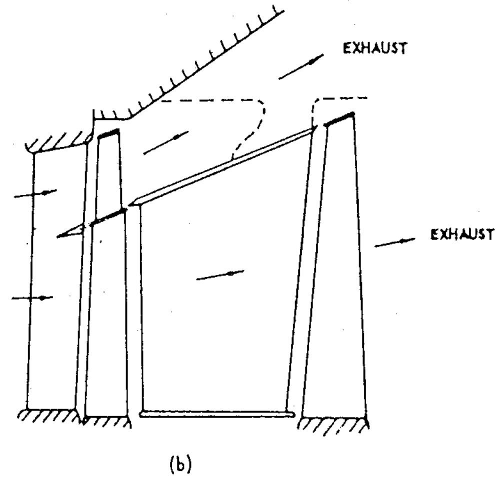


Figure 14 Baumann Exhaust

This type of arrangement is also referred to as a "multi-exhaust stage".

Figure 13 Low pressure turbine blading

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-5 - Turbine Blading

A - Assignment

1. Name the two factors which determine the speed of a turbine blade.
2. What are the two basic types of turbine stages? What is the basic difference between them?
3. Explain why the diaphragms have to be securely anchored to the turbine casing.
4. What is meant by the term "compounding" in turbines?
5. What is meant by the term "curtis stage" in a steam turbine?
6. Draw the pressure-velocity diagrams when:

   (a) compounding for pressure in an impulse turbine.
   (b) " " velocity " " " " .
7. Draw the pressure-velocity diagram for a reaction turbine.
8. Define veolcity ratio.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment and System Principles - T.T.2

4 - Turbine, Generator and Auxiliaries

-6 - Turbine Governing

0.0 INTRODUCTION

Previously it has been mentioned that the function of the turbine governor is to regulate automatically the speed and power output and to enable changes in these to be made when required. The speed and output are dependent on the amount of steam that flows through the turbine. Steam flow is determined by how many governing valves are open and how far they are open. The governor, then regulates turbine speed and output by controlling the position of the governing valves.

In the lesson on "Lubrication" T.T.4 level, it was said that the lubrication oil pump delivered high pressure oil and that some of this oil was used as power oil and relay oil. Any movement of the governor valvesis accomplished hydraulically by employing this power oil and relay oil. By varying the oil pressure in a hydraulic cylinder called a servomotor the valve stem is made to move either up or down and this causes the valve to move in an opening or closing direction. The oil pressure in the servomotor is dependent on the amount of high pressure oil that flows into or out of it. Therefore, we can say that the governor controls the position of the governing valves by regulating the amount of high pressure oil flow to or from the valve servomotors and hence controls the speed or output of the turbine.

No two manufacturers have exactly the same type of governing systems. So, what is described in this lesson is representative of what governing systems are like and how they function, but they may not necessarily be exactly the same as what the reader might be famliar with.

1.0 INFORMATION

To regulate the speed a device is necessary which will sense the magnitude of the speed within a specified range, and produce a corresponding displacement which can be used as a corrective control. Three methods of doing this are in use; namely hydraulic, electrical and mechanical.

A hydraulic governor for a turbine consists of a centrifugal pump driven from the main shaft the pressurized oil from it being led into a cylinder containing a spring loaded piston. The pressure is proportional to the square of the speed so that the position of the piston is a function of the speed. This type is employed in the U.S.A. but is not common in Canada.

The electrical governor is a more recent innovation, made practicable by the development of robust servo-mechanisms and circuit components. A small a.c. generator driven from the turbine shaft provides an electric signal of frequency proportional to the speed of the shaft. A frequency-sensitive circuit produces a voltage proportional to this frequency, which, after amplification by a magnetic amplifier, is fed to a torque motor which produces a proportional displacement. This method is used in some variable speed turbines and may well be used for large turbine generators in the future.

The majority of turbines are controlled by a mechanical centrifugal governor which is driven from the main shaft through gearing.

Centrifugal Flyball Governor

The heart of a mechanical governing system is the centrifugal flyball governor. An example of this type of governor is illustrated in figure 1. It consists of two flyweights or flyballs attached to the outer end of what are called two bell crank levers. The inner ends of the lever are in contact with a sliding sleeve. In this particular case the flyweights and bell crank levers are rotated at high speed by a worm gear which is connected directly to the turbine shaft. The bell crank levers are pivoted at A.

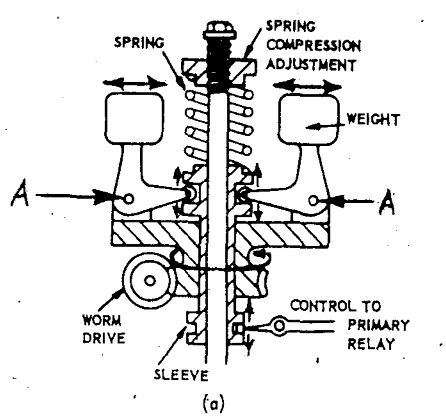


Figure 1. **Centrifugal flyball governor.**

If the turbine shaft speed increases the flyweights will move outwards due to increased centrifugal force. This will cause the sliding sleeve to move upwards causing the lever to the primary relay to move. If the turbine shaft speed decreases all the movements will take place in the opposite direction. In this way the centrifugal governor produces a displacement which is proportional to the turbine shaft speed.

Resistance to outward movement of the flyweights is provided by a tension spring, which opposes the centrifugal force caused by rotation of the governor assembly. The equilibrium position of the flyweight will change as the speed is varied in a manner described above.

To ensure quick response to changes of centrifugal force it is desirable that the radial inertia and friction forces of the governor assembly should be small. Hence, a good governor might be designed with small weights, the speed of revolution being increased by different sized gears to compensate for the loss of flyweight output force. To relieve the governor of work in moving the control mechanism, the flyweight output force may be amplified in two steps by means of a primary relay. More of this will be said a little later on.

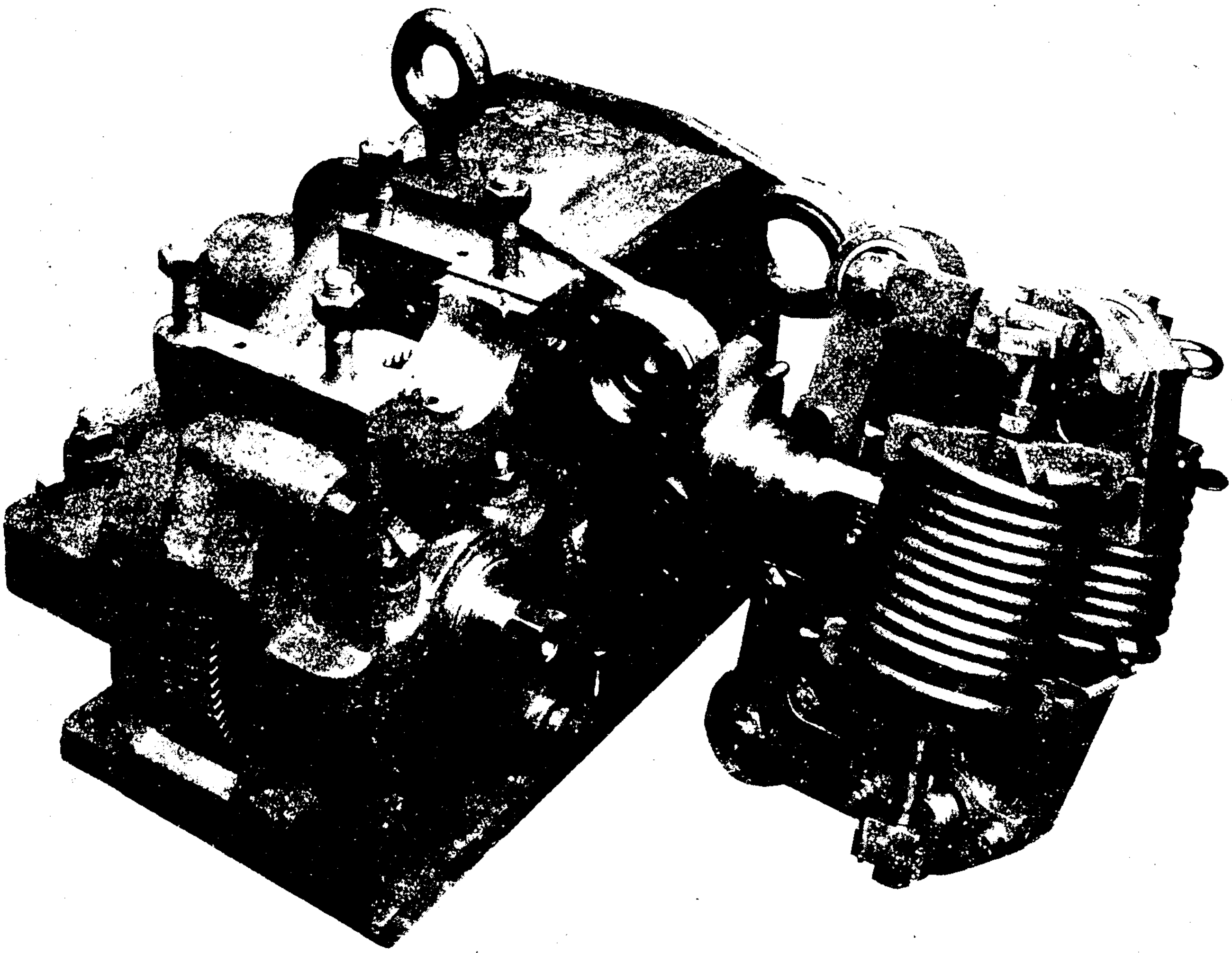


Figure 2. Centrifugal flyball governor with its geared drive.

Figure 2 shows another version of a centrifugal flyball governor. It has two springs, is horizontal instead of vertical, and is driven by the turbine shaft through helical reduction gears, rather than a worm gear. However, the principle of operation is the same.

Figure 3 summarizes in block diagram the basic sequence of steam turbine regulation. The centrifugal device senses that the shaft speed is too low, say. It then signals the throttle valves to open further, causing more steam to flow. This increases the turbine driving torque and results in greater generator output. The shaft has increased inertia and its rotational speed is increased. The centrifugal device senses this increase in speed and stops its signal to the throttle valve.

Before we proceed with describing the present day governing systems, let us look at some simpler ones from which modern systems have evolved.

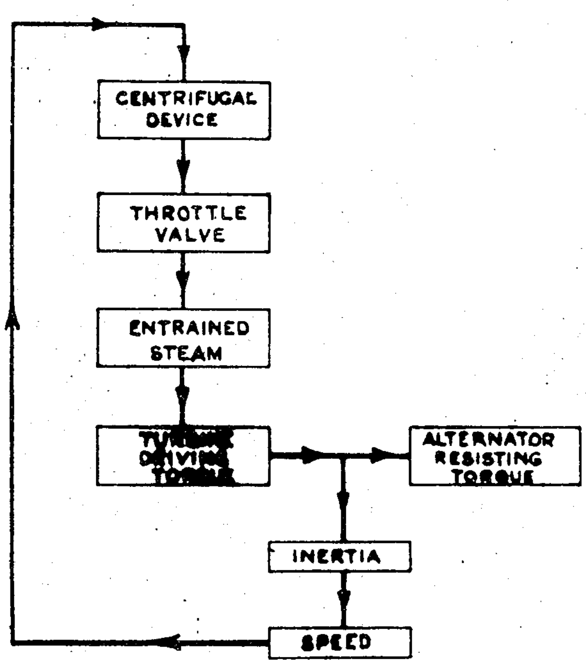


Figure 3. Simplified block diagram showing basic sequence of steam turbine regulation.

Direct Acting Control Gear.

The simplest form of control gear is shown in figure 4 where the centrifugal governor supplies the power, for direct operation of the throttle valve. This arrangement has been satisfactory for small turbines operating with low steam conditions, where the unbalanced forces to be overcome are relatively small. The flyweights move inward for a reduction in speed causing the sliding sleeve to move downward. This moves the throttle valve upward allowing greater steam flow, etc.

Mechanical Oil Relay Gear.

For large turbines operating under modern steam conditions, considerable power is required to move the E.S.V. and governing valves and it is general practice to interpose a displacement transmission mechanism with force amplifications as shown in figure 5. This consists of a centrifugal governor 'A' coupled to an oil distribution valve 'C', a servomotor

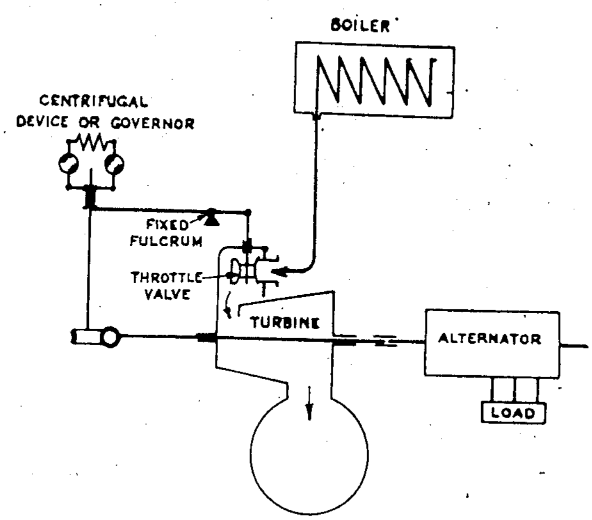


Figure 4. Direct-acting control gear.

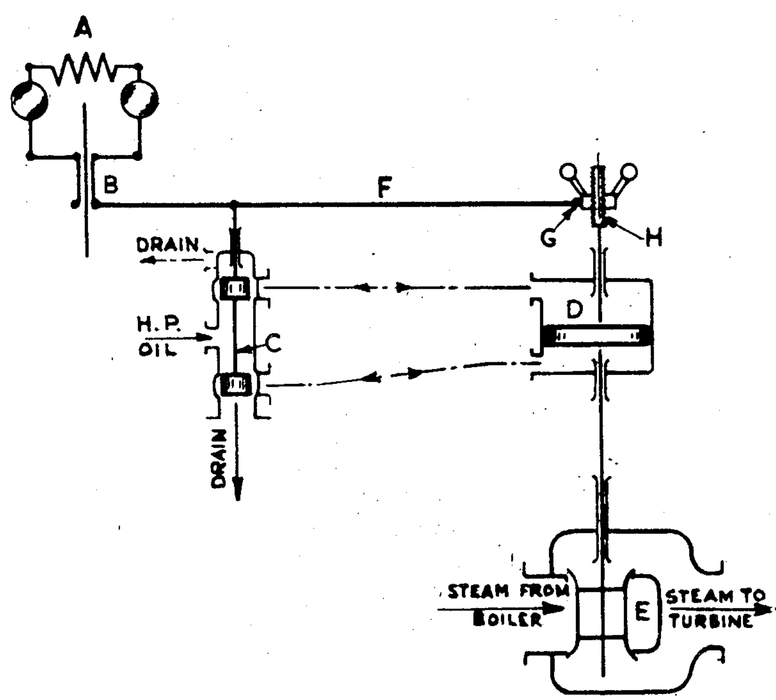


Figure 5. Mechanical oil relay control gear.

or relay piston 'D' and a compensating or feed back lever 'F'. On change of speed, movement of governor 'A' displaces the valve 'C' which applies oil pressure to one side of the piston 'D' and simultaneously opens the other side of 'D' to drain.
Under steady conditions the valve 'C' must be in its central position and thus the compensating lever 'F' causes the movement of the steam valve to be proportioned to the speed change.

The movement of valve 'E' can also be accomplished by manually rotating screw 'H'. Point 'B' in this case acts as a fulcrum because for steady speed it stays in one location. Downward movement of 'H' then means a downward movement of valve 'C'. In this case H.P. oil can flow to the bottom of piston 'D'. Due to the position of 'C' the top of piston 'D' is now open to drain. Thus an upward movement of piston 'D' takes place. This means that valve 'E' will open further allowing more steam to flow into the turbine.

This simple arrangement has two main disadvantages. Firstly, physical limitations are imposed by the compensating lever since it makes it necessary for the governor and servomotor to be close together. Secondly, the power required from the governor to operate the mechanism is still unduly large and this necessitates a centrifugal device of heavy construction rotating at a comparatively slow speed; such a construction has a relatively slow response and consequently reduces the rapidity of action of the entire control gear. This gear has proved satisfactory for intermediate steam conditions. However, with the advent of elevate steam conditions together with increased steam quantity the unbalance forces on the throttling valves is considerably greater and it requires a larger force to open the valves. Also a faster response is demanded by the lower inertias of the rotors. Therefore an improved type of control gear is required.

Typical Modern Control System.

Figure 6 shows a diagrammatic arrangement of the improved control gear. The centrifugal governor 'A' is driven direct from the turbine shaft by helical gears and is mechanically connected to the oil governing valve 'D' operating in the

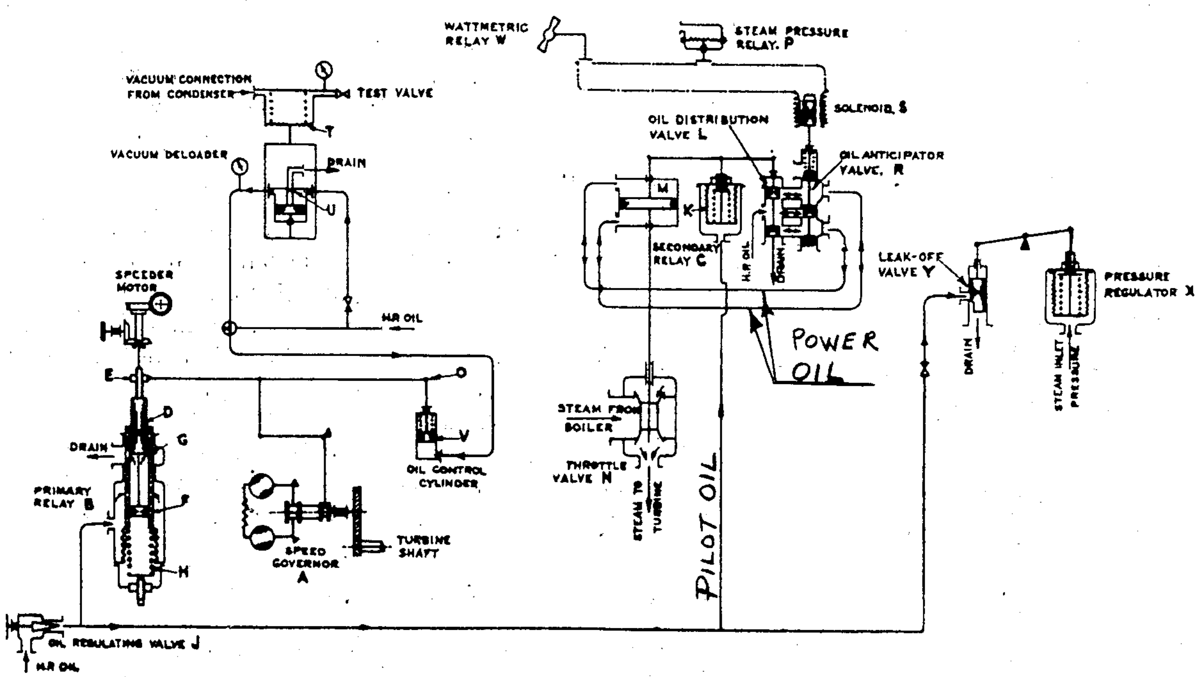


Figure 6. Diagrammatic arrangement of modern control system.

primary relay 'B', the latter being hydraulically connected with the secondary relay 'C'. The centrifugal governor runs at high speed and its degree of sensitivity is conferred on the control gear as a whole by virtue of the subsequent two stages of amplification.

Displacement of the centrifugal governor 'A' figure 6 is transmitted to the oil governing valve 'D' which momentarily moves in relation to the bellows sleeve 'F'. The position of the latter is controlled by the hydraulic pressure imposed on the unbalanced annular area and the opposing spring 'H' which are in equilibrium under steady-state conditions. This condition coupled with the controlling exhaust restriction between 'D' and 'F' and 'G' produces an oil pressure which decreases as the speed increases and vice versa. The flow of oil is determined by the pressure and the regulating valve 'J'. Changes in pressure are rapidly transmitted by the hydraulic system to the secondary relay 'C', where the spring waded bellows 'K' transforms the pressure into a displacement. The latter operates the main throttle valve 'N' by the combination of the oil distributing valve 'L' and the servomotor 'M'.

Nowadays servomotors of the type 'M' are made to "fail safe" That is, instead of the oil pressure on top of the piston there is a strong spring. When oil pressure is applied at the bottom of the piston, it not only opens the valve but also compresses the spring above. If for any reason the oil pressure fails, the spring will instantly shut the throttle valve.

The secondary relay 'C' together with oil distribution valve 'L' is often referred to as a pilot valve. The oil causing actuation of the pilot valve is referred to as pilot oil. The oil causing movement of the servomotor for the governing valves and emergency stop valves is referred to as power oil.

The whole control system illustrated in figure 6 often referred to as a sensitive oil system.

This improved control gear has many advantages, the most important ones of which are listed below:

1. The introduction of a high-speed centrifugal governor allows a direct drive through helical gears and eliminates the possible troubles arising from the use of worm gears.

2. Since it is followed by two stages of amplification the centrifugal governor is small. Hence a light design of low inertia can be employed with consequent improvement in sensitivity and rapidity of response.

3. The hydraulic transmission confers complete freedom in the position of the steam chests.

4. Adjustments to various controls are relatively simple.

5. The compact centrifugal governor and primary element are completely housed in and protected by the steam-end pedestal, but at the same time are easily accessible.

6. It is readily adaptable for multi-valve operation due to the flexibility of the hydraulic system. It can easily be connected to four throttle valves 'N' each with its own relay valve 'C' and servomotor 'M'.

Often pilot valves and their sleeves and the servomotor pistons will stick due to friction. To overcome this friction an artificial "dither" is introduced by means of small pulsations in the high pressure oil.

The above description has covered automatic control when the turbine is operating normally. However, for starting up and manually changing the load the speeder gear is employed, which is shown in figure 6. As mentioned previously, the speeder gear can be operated by electric motor in which case it can be done remotely from the control room or it can be done by turning a handwheel at the turbine. When the turbine is shut down valve 'D' would be all the way up so that if there were any high pressure oil supply it would all drain away through the drain ports 'G' of the primary relay 'B'. Now, if by means of the speeder gear valve 'D' is moved downward it will reduce the flow area of the drain ports 'G', causing oil pressure to build-up on the downstream side of regulating valve 'J'. An increase in oil pressure results in an opening movement of throttle valve 'N' and this will cause the turbine to rotate. The governing oil pressure is gradually increased by means of the speeder gear till valve 'N' passes enough steam to rotate the turbine at approximately 4% of generator synchronous speed, at which time the centrifugal governor is rotating fast enough to take over the control of oil pressure and from then on the governing controls operate automatically. With spring loaded governing valves the turbine can be instantly shut down or "tripped" by opening the high pressure oil to drain.

Sequential Opening of Governing Valves.

For the method of throttle control of steam flow all governing valves open at the same time and at the same rate and therefore the governing system can be relatively simple. However, in the case of nozzle governing, the valves open in sequence and this requires special design in the governing system. It can be done in two ways. 1) by varying the power oil pressure or 2) a combination of varying oil pressure and control rod.

Let us say we have a system that can produce a maximum power oil pressure of 120 psi. and that the governor valves are spring loaded. For sequential opening of valves by varying the pressure only, the tension on the springs would be adjusted so that at around 60 psi. power oil pressure, two governings valves would start to open. At 80 psi. they would be fully open and the secondary valve would start opening. At 100 psi. this third valve would be fully open and the tertiary valve would start opening till at 120 psi all four valves would be fully open.

In method No. 2 all the oil relays are connected to the same control rod. The relays are placed in such a position that the control rod displaces each pilot valve from the neutral position by progressively increasing amounts. Thus while the first servomotor is opening the first governing valve the second servomotor remains drained until its pilot valve passes the neutral position and high pressure oil is admitted. Valves of nozzle governed turbines are sometimes also operated from a camshaft rotated by a single power servo of the rotory type. The position of the cams will then determine the sequence of valve opening as the camshaft is rotated.

When a turbine is on load - i.e. the generator is synchronized with the transmission line and producing power, the shaft speed is dependent upon the system frequency. Under these conditions, using the speeder gear to increase steam flow to the turbine, will increase torque output. This in turn increases generator resisting torque resulting in greater generator power output. The opposite effect will occur if steam flow to the turbine is reduced. The speed for a 60 cycle frequency, will, however, remain very nearly constant at 3600 rpm. or 1800 rpm. depending upon whether the generator is a two pole or four pole machine.

Now lets look at it another way. Ontario Hydro has an electric power transmission grid in Ontario. This grid is being fed by a number of generators operating in parallel with each other. There is a load demand on the grid due to operation of various electrical equipment. When there is an increase in load demand the generators will attempt to meet this demand. This will result in a momentary reduction in shaft speed because the turbine can't meet the increase in required torque. The centrifugal governor senses this reduction in speed and sends a signal to open the governing valves further thus providing more torque output to the turbine shaft, and thus restoring the shaft back to synchronous speed again. When there is a decrease in load demand the opposite effect will occur.

Entrained Steam.

The steam which is located at any instant between the control valve and the last stage of the turbine (entrained steam) introduces a severe time delay into the regulation. This is very apparent if a sudden drop in a load occurs and the synchronizing tie is broken, when, even if the control valve closes instantaneously the ensuing expansion of the entrained steam will cause appreciable overspeeds. Overspeeds attained in practice are due to this effect, coupled with the inherent time delay of the control gear.

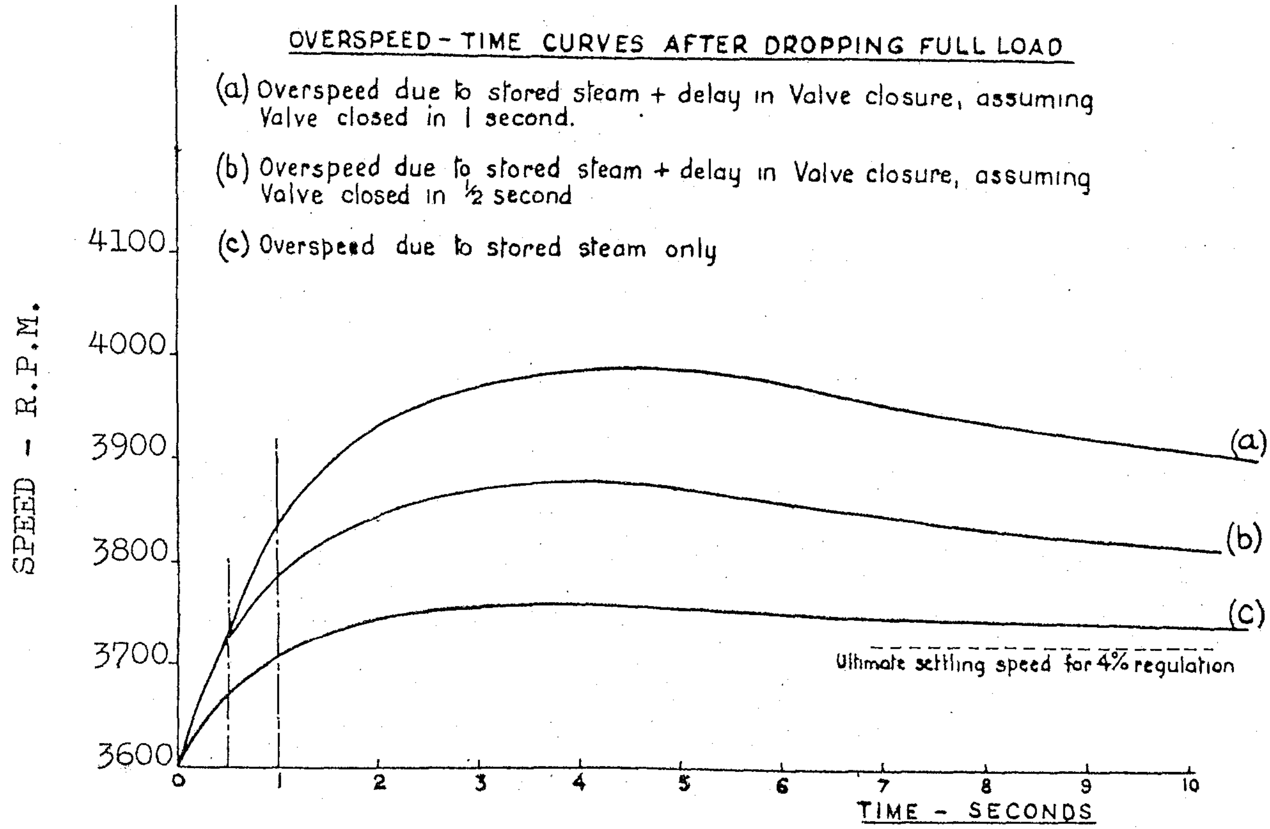


Figure 7. Typical speed curves, showing the effect of entrained steam.

The major effect of entrained steam on speed rise is produced while the valves are closing, and is shown on the typical speed curves in figure 7, from which the importance of valve closing times can be appreciated. It follows, therefore, that rapid response of the control geat is essential to reduce speed rise as far as possible, since this is a factor which has been made more difficult by the increase in output and steam conditions.

Governor Anticipators.

If the control of the steam valves is left to the centrifugal governor, they will not close, even in the ideal case of instantaneous response, until the speed has risen by the amount of permanent speed variation (approximately 4% for full load). Therefore on loss of full load the total overspeed will be the sum of the permanent speed variation plus the additional speed rise due to stored steam. To avoid excessive overspeed it is essential to close the steam valves in the shortest possible time by the use of a device which will anticipate the governor operation and close the steam valves when the shaft is at practically normal speed.

Referring back to figure 6 an arrangement to meet this requirement is shown in this illustration. The energizing circuit for the solenoid 'S' is completed by the contacts of a wattmetric relay 'W' and a steam pressure relay 'P' under predetermined conditions of operation.

The wattmetric relay is arranged so that its contacts are closed below some suitable low value of generator load, and are open for valves above this setting. The steam pressure relay contacts are closed only when the steam pressure is above a value corresponding to about 50% of full load. Thus the energizing circuit will be completed only when there is a combination of high steam pressure and low load, such as would occur on a sudden load reduction from above the value corresponding to the steam pressure relay setting to a value below the wattmetric relay setting. With these conditions prevailing, the solenoid displaces a plunger causing the oil valve 'R' to operate the servomotor piston 'M', which immediately closes the steam valves. For all other conditions either one or both sets of contacts will be open to make the gear inoperative and permit normal loading or de-loading of the machine. Other similar anticipatory devices are available from different manufacturers.

Auxiliary Centrifugal Governor.

In the case of reheat turbines, two centrifugal governors are employed, both driven in the way described in this lesson. Both governors can operate all the controlling valves by varying the oil pressure of a common pilot oil system. The governors are so adjusted that the main speed governor exercises control when the turbine is running at normal speed. The auxiliary governor starts to share control of the system if the machine should reach an overspeed of about 1%. Above this speed, both governors control the governor valves, intercept valves, and release valves. The two governors operating together cause the valves to move very rapidly in the event of a sudden loss of load. In the unlikely event of either governor sticking, the other will control the system independently.

In the event of the machine suddenly losing its load (i.e. the generator trips), the control system will operate in the following sequence:- as the turbine speed starts to rise, due to the entrained steam in the turbine, the main speed governor will start to close the governor valves. At about 1% overspeed, the auxiliary governor will operate and share control of the governing system. At about 2 1/2% overspeed, the intercept valves will start to close and they will be fully closed at about 4% overspeed. At this point the release valves will open and release the steam in the reheater and connecting pipes to the condenser.

When the disturbance caused by the load throw-off has passed, the turbine will continue running at a constant speed slightly above normal under the control of the main governor valves. The release valves will have automatically re-closed and the intercept valves re-opened fully. In order to re-synchronize the machine, the speed must be brought back to normal by means of the main governor control.

Protective Devices.

When considering the design of any type of mechanical device, the possibility and consequence of its failure to operate must be borne in mind, and apparatus must be fitted to reduce the severity of such failures.

The more likely dangers to which a turbine is subject during operation may be summarized as follows: overspeeding, oil failure, thrust bearing failure, vacuum failure, boiler priming, excessive vibration and excessive temperature.

The last two items are monitored by means of supervisory gear, details of which are found in the next lesson on "Supervisory Equipment." The remainder are catered for by a series of automatic hydrailic devices known as trip gear and unloading gear. Trip gear, by releasing the pressure from the power oil system closes the E.S.V.'s, causing the immediate shutdown of the turbine- generator. Unloading gear exerts an over-riding control on the turbine governing gear, and limits the output of the turbine when inlet and exhaust conditions are abnormal.

1. Emergency Stop Valve Operating Gear

The stop valve is held open against a powerful spring, which stores energy necessary to close it. It may be opened either by hand or hydraulic power, but in each case loss of power oil pressure causes it to close.

Figure 8 shows two schemes in common use. In illustration (a) the hydraulic plunger 'A' is raised and sealed by operating the handwheel 'B' turning the turnbuckle.'C'. It is maintained in this position by power oil pressure in 'D'. The handwheel and turnbuckle are now turned in the opposite direction, thus raising the valve against the plunger. Loss of pressure in 'D' will cause the valve to close. In illustration (b) the valve is operated by spring loaded servomotor similar to that described for the governor valve, and is closed by a loss of power oil pressure.

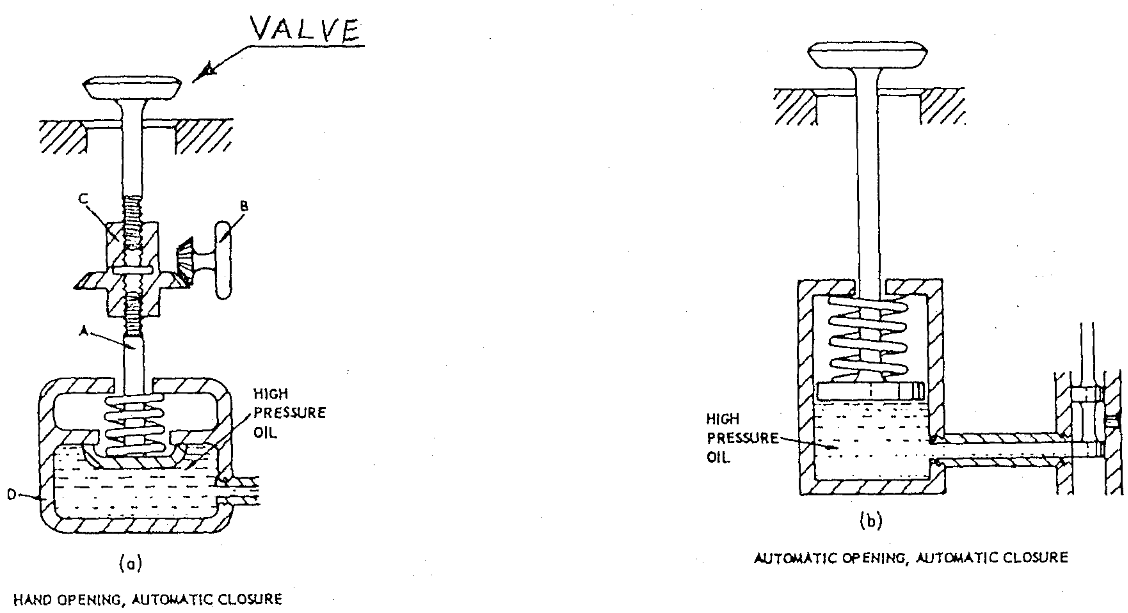


Figure 8. Emergency stop valve control gear.

2. Tripping Devices

A typical emergency trip valve as shown in figure 9 consists of a ported sleeve, a plunger and a spindle in the main body. Oil from the discharge of the main oil pump is supplied to a chamber in the upper portion of the valve body. When the valve is in the running (reset) position as shown, oil is distributed to the various areas of the governing system as shown.

The oil from the main oil pump passes from the upper chamber through ports in the sleeve around the plunger to two additional chambers in the valve body to which the relay piping is connected.

The plunger has a relieved centre section and fits into the ported sleeve making a sliding fit at its upper and lower diameters. The upper diameter is larger than the bottom so that when relay oil at pump discharge pressure surrounds the centre portion of the plunger there is an out-of-balance force tending to move the plunger upwards. The plunger is held down in the reset position by a projection at the top of the spindle. The spindle is held down by the latching mechanism at the bottom. The latch is released by the manual trip lever, overspeed trip rings, or emergency trip solenoid. When the latch is released on a trip or a test, the spindle and plunger move upwards, cutting off

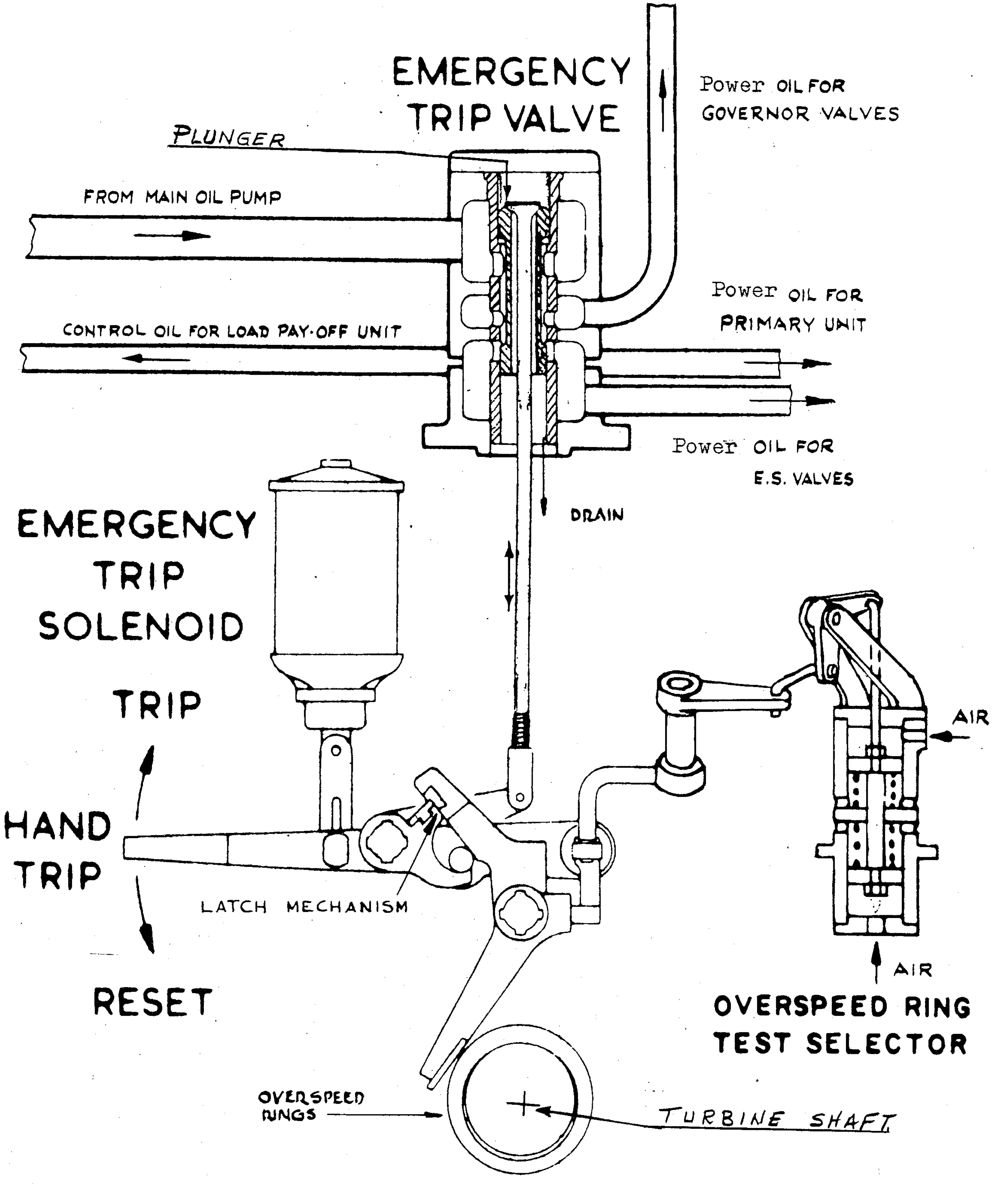


FIGURE 9. EMERGENCY GOVERNOR

the supply of oil from the upper to lower chambers, and opening the bottom chamber to drain. The Emergency Stop Valves will immediately trip closed, because of loss of power oil pressure.

This emergency trip valve can be actuated in three ways:

(a) by hand at the turbine

(b) by solenoid using a switch in the control room, or as will be explained later, automatically when an electric fault occurs.

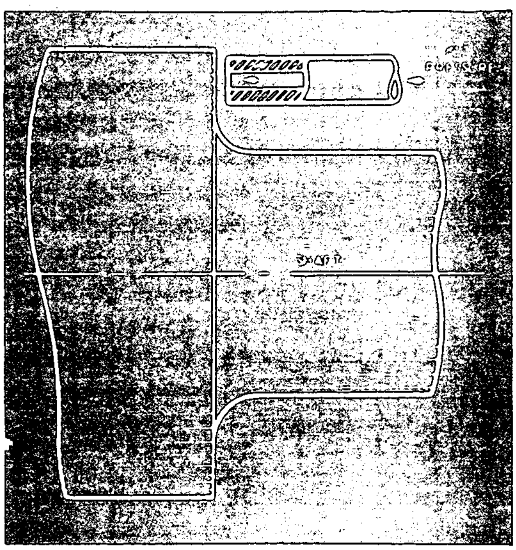
(c) by overspeed rings.

3. Faults that Trip Turbine.

(a) Overspeeding. The overspeed trip consists of either a spring loaded bolt, or overspeed rings incorporated in the shaft at the governor and as shown in figure 9. The ring is placed so that its centre of gravity is a short distance from the axis of rotation. The out-of-balance force is arranged to overcome the spring compression at 10% overspeed causing the bolt or rings to knock the trip lever which releases the oil pressure from the stop valves. This trip may be tested without actually overspeeding or shutting down the unit.

(b) Lubricating oil failure. Loss of lubrication would quickly lead to wiping of the journal and thrust bearings with consequent damage to the shaft. If the main oil pump auxiliary a.c. pump and d.c. flushing pump all fail to deliver sufficient oil then a pressure switch would electrically actuate the solenoid in the trip gear.

(c) Thrust bearing trip. Owing to variation of the thrust load, the thrust bearing is more susceptible to wear than the journal bearings. To prevent damage to blades and glands, should the wear become excessive, a tripping device is sometimes fitted as illustrated in figure 10.



A fine jet of high pressure oil plays on the face of a flange of the shaft (generally on the thrust collar) the nozzle being in close proximity to the flange. Thus only a small quantity of oil flows under normal conditions, since the clearance is small and the pressure behind the nozzle is maintained. Should the clearance increase owing to axial shaft movement, more oil will leak away, the pressure will fall, and the emergency stop valves will trip.

Figure 10. Thrust wear trip

(d) Low vacuum trip. In case the vacuum unloading gear fails to stop the falling vacuum, a vacuum trip device is incorporated independently, which trips the turbine generator off load when the vacuum falls to 20 ins. mercury. It consists, usually of a spring-loaded bellows, operating an oil pressure release valve in the high pressure oil circuit, or a switch in series with the solenoid of the solenoid-operated valve.

(e) Generator fault. The solenoid in figure 9 can also be made to actuate a turbine trip automatically should generator faults occur such as: reverse power or generator motoring, high phase-to phase or phase-to-ground current, phase current unbalance, high stator ground and loss of field current, and sealing oil failure. The generator circuit breaker, of course, would also trip under these conditions.

4. Faults that Unload Turbine.

(a) Low vacuum unloading gear. Should the vacuum fall to a low value the temperature of the exhaust steam will increase, and may possibly damage the last rows of blading and the condenser tubes. To avoid this a device is used which, below a certain vacuum, progressively decreases the steam flow as the exhaust pressure rises, thus tending to restore the vacuum. It does not close the governing valves completely, as this would cause mo--toring. The vacuum is measured by some such device as spring-loaded bellows, which control the pilot valve of a relay. The output of the relay is arranged to override the output of the governor primary relay, thus controlling the governing valves as shown in Figure 11.

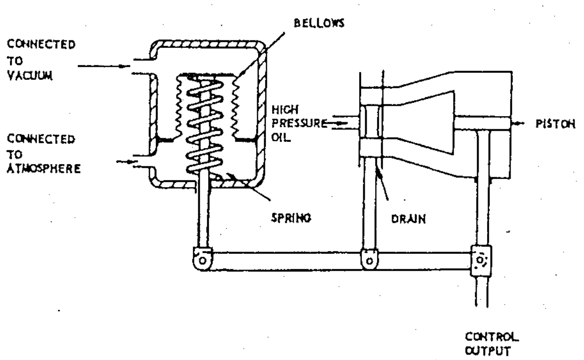


Figure 11. Vacuum unload--ing gear.

The gear is designed so that it cannot reload the turbo-generator automatically when the vacuum is restored, as this might result in 'load snatching' which would disturb the electrical system. The condenser also has an atmospheric relief valve, although with an unloading gear it only has to handle 10% of full load flow.

(b) Low steam pressure unloading. The correct functioning of the turbine control gear depends on the maintenance at the stop valve of the design steam conditions. If the boiler failed to maintain the steam pressure, the turbine would demand more steam, which would cause the pressure to fall still further. Furthermore, if the fall in pressure were sudden, due to a fault in reactor control, the sudden demand for more steam might cause water to be carried into the turbine, causing serious damage. Protection against this may be obtained by reducing the load on the turbine, should the inlet steam pressure fall below a certain value.

One type of gear for performing this function is shown in figure 6, where it is labelled as a pressure regulator. It relies on a bellow to detect the fall in pressure. A small line leading from the main steam line connects to the unloading device. An orifice is located in the line to reduce the steam pressure to a proportional but lower value. With a drop in steam pressure the spring will push the bellows downward causing the leak-off valve 'Y' to open allowing some of the power oil to flow to drain; this causes a closing movement on the governing valves thus reducing load.

5. Anti-Motoring Switch

If the generator remains connected to the bars after the steam supply to the turbine has been cut off, it will act as a motor. The turbine rotor will warm up owing to disc friction and churning of stagnant steam and become dangerously hot. This is prevented by means of an oil operated trip switch, which energises the operating coil of the main breaker when the oil pressure drops.

Load Limiting Meter.

This type of device can be installed with the unit so that it overrides the centrifugal governor to hold the maximum load at any point the operator wants. The limiter controls steam flow during startup by setting governor valves as needed.

Testing Protective Devices.

The absolute reliability of the entire control system is so vital to the safety of the turbine-generator that it is now becoming general practice to make provision for testing all the essential components of the control system while the machine is in operation. All the control valves may generally be tested while the unit is on load. For this purpose, an isolating valve is fitted in the oil supply line to each of the H.P. emergency stop valves, governor valves, intercept valves, reheat emergency stop valves and relief valves. When any one of these isolating valves is closed, the steam valve affected will operate. Arrangements can be made so that valves will reset automatically when the oil supply is restored.

Conclusion

This lesson has described the governing system in a general way. For more detailed information the reader should consult the design manuals for a particular thermal station.

D. Dueck.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment and System Principles - T.T.2

4 - Turbine Generator and Auxiliaries

-6 - Turbine Governing

A - Assignment

1. In a sentence or two describe the function a steam turbine-governor performs.
2. Draw a diagram of and describe a centrifugal flyball governor.
3. Why is direct acting control gear not used on large turbines?
4. Give 5 advantages of a sensitive oil governing system?
5. What is meant by throttle control? nozzle governing?
6. Briefly, why does a turbine overspeed when there is a sudden loss in load?
7. What type of dangers is a turbine subject to?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-7 - Supervisory Equipment

0.0 INTRODUCTION

Instrumentation is provided on modern high pressure and temperature turbines to indicate and record differential expansions between steam rotors and casings, rotor eccentricity, steam/metal differential temperatures, bearing vibration, speed and load. This equipment which is necessary to ensure safe and satisfactory operation of the plant is known as Supervisory equipment (sometimes also referred to as turbovisory equipment).

1.0 INFORMATION

Correct clearances between rotating and stationary parts of the turbine must be maintained under all conditions of operation, even though temperatures may range from cold, up to the maximum, and overall expansion of the order of one inch in length may be experienced. Figure 1 lists some typical clearances between moving and stationary blades in a turbine. One can readily see that casing and rotor have to expand very nearly at the same rate otherwise there will be rubbing of metal surfaces.

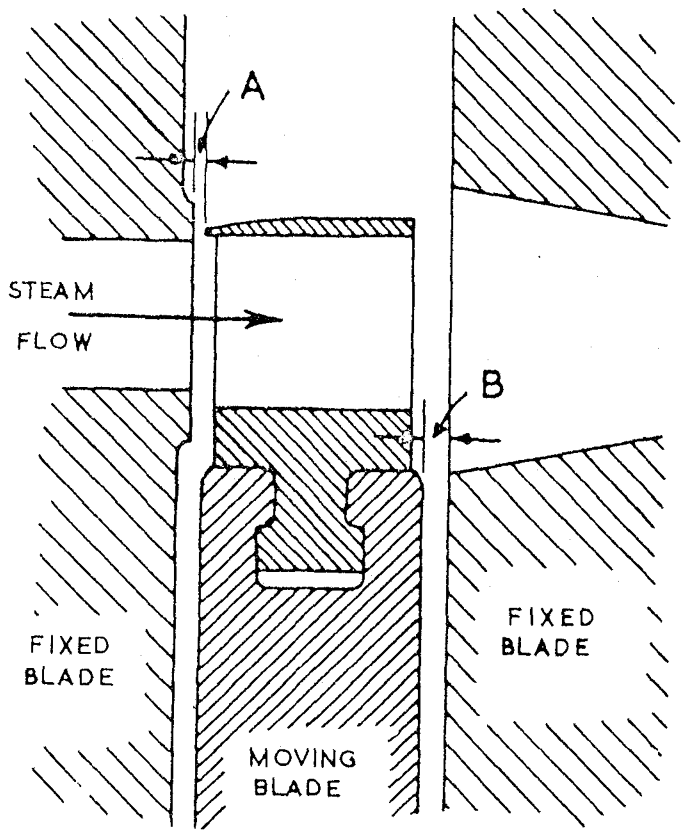
Types of Supervisory Measurments

Special instrumentation is necessary to indicate and, where required, record the following:

1. Indication and recording of vertical eccentricity.
2. Indication and recording of horizontal eccentricity.
3. Indication and recording of differences of expansion between rotating and stationary parts known as differential expansion.
4. Indication and recording of shaft speed.
5. Indication of electrical output.
6. Indication of turbine casing temperatures.
7. Indication of vibration amplitudes at bearings.

As will be explained later on, most detectors are electromagnets arranged in matched pairs. These electromagnets form arms of a wheatstone bridge. It is customary to use low voltage d.c.

across opposite 'corners' of these bridge networks to observe 'an out-of-balance' direct current on an instrument across the other 'corners'. However in this case d.c. cannot be used, since variation in inductance has to be measured. The reading of the instrument would be affected by other disturbances caused by shaft rotation, for example, a 60 cycle 'wobble' may be super-imposed. The voltage supply to the bridge would therefore be alternating current of a frequency sufficiently high to avoid other 60 cycle effects and distortions caused by the higher harmonics of 2-pole flux. A frequency of 500 or 1000 cycles per second is normally chosen for the supply to the bridge , and also for the 'carrier' wave to the electronic rectification circuitry. This means a separate high frequency motor-generator set for the supervisory gear which is generally located in the supervisory gear cubicle.



STAGE NO.	INLET EDGE CLEARANCE A	OUTLET EDGE CLEARANCE B
1	0·030 INCH	0·238 INCH
2	0·032 "	0·239 "
3	0·028 "	0·226 "
4	0·032 "	0·228 "
5	0·032 "	0·236 "
6	0·061 "	0·231 "
7	0·059 "	0·225 "
8	0·061 "	0·228 "
9	0·054 "	0·230 "
10	0·059 "	0·235 "
11	0·054 "	

INLET - CLEARANCE IS CRITICAL WHEN ROTOR CONTRACTING
OUTLET - CLEARANCE IS CRITICAL WHEN ROTOR EXPANDING

Figure 1. H.P. cylinder axial clearances.

Since different makers have different circuit arrangements, the conversion from detectors to instruments is illustrated only by means of 'block' diagrams as shown in figure 2.

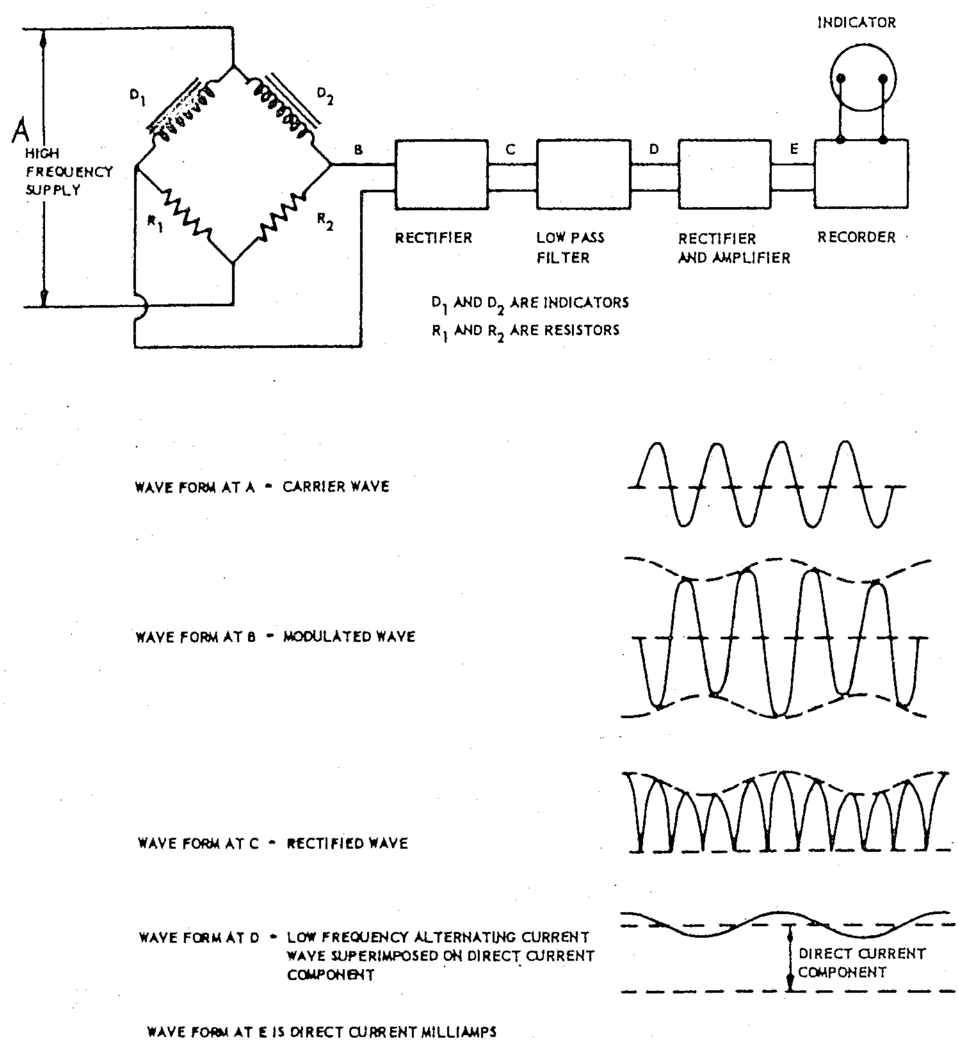
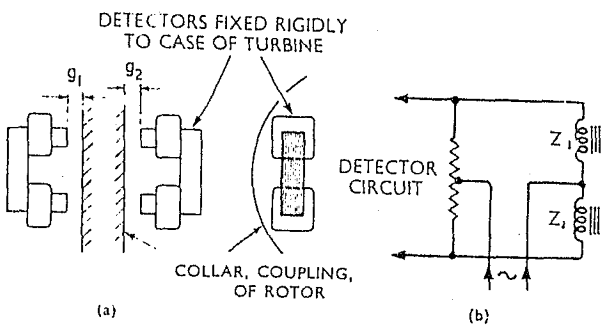


Figure 2. Electronic circuits for conversion of detector signals to instrument readings.

Differential Axial Expansion

Detection of axial clearances between moving and stationary parts inside a turbine is a most valuable operational aid. The rotor, as mentioned in previous lessons, is located axially relative to the casing by means of a thrust collar, which is usually adjacent to the portions exposed to the higher temperatures. During the running up of the unit, the rate of heating of the rotor, because of its smaller mass, frequently differs from that of the casings. Consequently the rotor system and the casings expand at different rates, and all clearances are affected in some degree according to the longitudinal temperature distribution, the distance from the thrust collar, and the different and changing expansion of parts close together. When required, provision is made for an independent control of the rate of casing expansion, by means of a separate supply of steam to the horizontal joint. Although the axial clearances most affected are those usually generous, at the end farthest from the thrust collar, change in the relatively finer clearances, which are usually associated with the highest temperatures, may be the critical factor. Measurement of relative axial movement is therefore desirable at convenient points along the rotor as well as for the turbine as a whole; the measurement also serves to guide the operator when adjusting the flow of joint warming steam.

The measurement of differential axial expansion is simple: it involves the use of a variable-inductance bridge in the form of a pair of detector electromagnets connected in series and shunted by a centre-tapped resistor. The detector magnets are rigidly fixed to the turbine casing with their poles opposed, on each side of a collar, coupling, or other suitable part of the rotor, as in figure 3(a). A difference in movement of rotor and casing increases one air gap and decreases the other by a similar amount. The bridge is excited by a high-frequency generator, and the air gap is so proportioned that the required range is covered by the central, and therefore linear, part of the bridge characteristic, as referred to in figure 3(b).



(a) Position of detectors ($g_1 + g_2$ = constant).
(b) Circuit diagram.
Over linear portion open-circuit voltage $\propto Z_1 - Z_2$

Detector circuit current $\propto \dfrac{\text{open-circuit voltage}}{\text{equivalent impedance (constant)}}$

Figure 3. Differential axial expansion.

In practice, the detector circuit is biased so that zero differential expansion corresponds to a finite output from the bridge. The arrangement is such that rotor expansion relative to the casing circulates an increased current in the detector circuit, and rotor contraction decreases the current.

Shaft Eccentricity

The control of rotor deflection is probably the most significant problem to be faced during starting. Accordingly, the principal requirement for supervisory equipment is to give the operator adequate warming of distortion before it is aggravated to an appreciable degree. This is particularly important. Continued increase of speed with a shaft out of true can lead to a situation in which correction is beyond control by the time vibration can be felt by the operator, and extensive damage to the machine may occur.

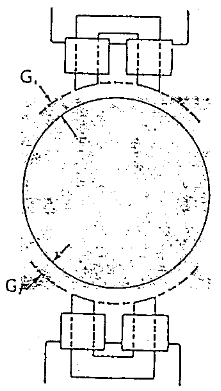


Figure 4. Position of eccentricity dectors in relation to the shaft.

The measurement and recording of shaft eccentricity at a suitable position on the turbine is carried out by using, as a detector, a pair of electromagnets excited by the same high-frequency generator as employed for differential axial expansion. These electromagnets are diametrically opposed across the shaft, and are rididly fastened to a bearing pedestal, as shown in figure 4. As it is important to know the truth of the shaft at low speed, as well as at higher speeds, the amplifying equipment following the detectors is arranged for two methods of operation. The first, known as the 'turning gear' operation, is used during the barring of the turbine at low speeds: the second method, known as the 'steam' operation, caters for speeds above this level.

When on turning gear, a slowly alternating trace is given by the recorder pen, and the width of this "band" or "envelope" is the eccentiricity. It is useful to remember this when attempting to straighten a shaft or checking for bending or deformation.

During normal operation eccentricity is recorded as a single line.

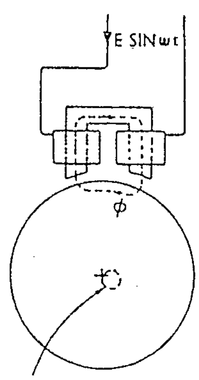


Figure 5

Figure 5. Single detector in relation to an eccentric shaft.

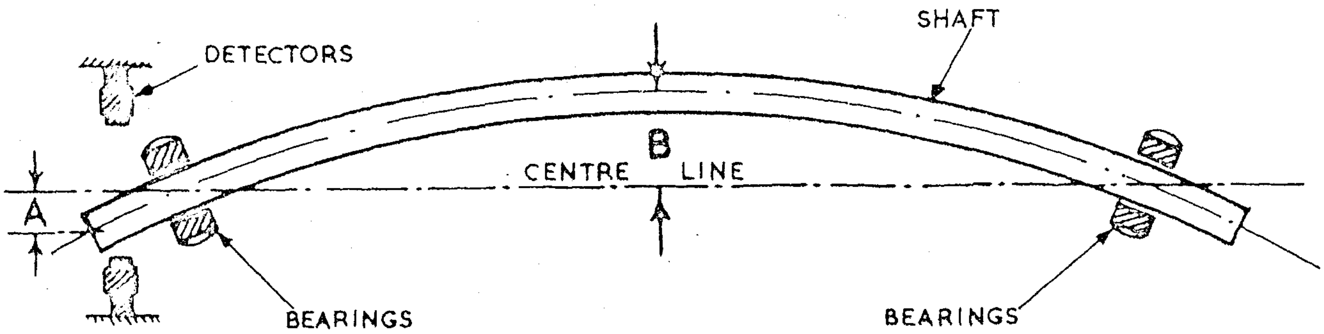


Figure 6. Hogged shaft (exaggerated for clearness) and arrangement of eccentricity detector coils.

Figure 5 shows a single eccentricity detector electromagnet in relation to an eccentric shaft, the centre of which describes an eccentric circle of radius 'r'. The coils are energized by the high frequency generator, and each forms an impedance in which the magnet path includes, in series, the core, the turbine shaft, and the two air gaps. As the shaft rotates, the air gaps vary with the transverse displacement, so that the carrier current flowing through the coils is modulated as a function of shaft eccentricity and speed as shown in figure 2. Figure 6 shows a badly bent or "hogged" shaft where the air gap to the detector is greater on top than at the bottom. This type of situation would show up on the eccentricity recorder chart.

Shaft Speed and Load

The speed indicator is usually supplied from a tacho-generator direct-coupled to the shaft. The a.c. output of this small generator, which has a permanent magnet field is linear with speed and is of sufficient magnitude to work the indicating and recording instruments. It is considered useful to have an automatic range switch to give two ranges, one 0 - 3600 rpm, and the other say 3350 - 4100 rpm.

The measurement of load is obtained from a wattmetric device.

Overall Expansion

The axial movement relative to the foundation of the bearing pedestal remote from the anchor point, can be transmitted by rack and pinion, or other suitable means, to the driving shaft of a toroidal potentiometer included in a simple bridge circuit. The bridge circuit is energized from a constant d.c. supply: it is so arranged as to give a signal proportional to the position of the potentiometer shaft, which itself depends on the degree of overall expansion of the turbine.

Bearing Pedestal Vibration

Vibration detectors are mounted transversely on the bearing pedestals. Each detector consists of a mass of laminated soft iron which is suspended from flat springs between the pole faces of a U-shaped laminated core, so that freedom of motion is permitted in the transverse plane only. A permanent magnet is fixed to the detector base and the magnetic circuit is completed through the soft iron mass and core around which coils are wound. When mounted on a vibrating pedestal, the soft iron mass remains in one position so long as the resonant frequency of the mass and suspension springs is not approached. Accordingly, with a pedestal vibrating, there is a relative motion between the soft iron mass and the core so that a signal proportional to the amplitude of vibration is generated in the coils. After rectification, this signal is transmitted to a recorder, which is sequentially connected by automatic switching to the detectors.

It is metimes possible for a turbine shaft to run with appreciable eccentricity without it being known. Vibration measurements, while giving a good guide to the general running of a turbine, may not give a true interpretation of the rotor condition. Cases have been known when sets have run without causing undue vibration although several blades were missing from several wheels.

For 60 cycle generation, smooth running is experienced when vibration amplitudes at bearing pedestals do not exceed 0.005 ins. An acceptable condition for reasonably good operation is an amplitude of between 0.001 ins. and 0.002 ins., but rough running is soon reached above the latter figure.

Use of Supervisory Equipment

If supervisory equipment were not available during initial stages of starting-up positive differential expansion would take place unknown as shown in Figure 8. The introduction of gland steam and flange steam alternatively may keep the turbine in a smooth running state, depending on the skill and experience of the turbine operator. Eccentricity may, however, ultimately develop if the temperature gradients become excessive, because of these 'swings' in expansion between wheel and diaphragm. When supervisory equipment is available to guide the turbine operator, the machine can be run up without the 'swings' of differential expansion as indeed is shown in Figure 9, and it is possible to keep eccentricity below 0.005 ins.

Figure 7 illustrates the relative positions of supervisory detectors on a turbine.

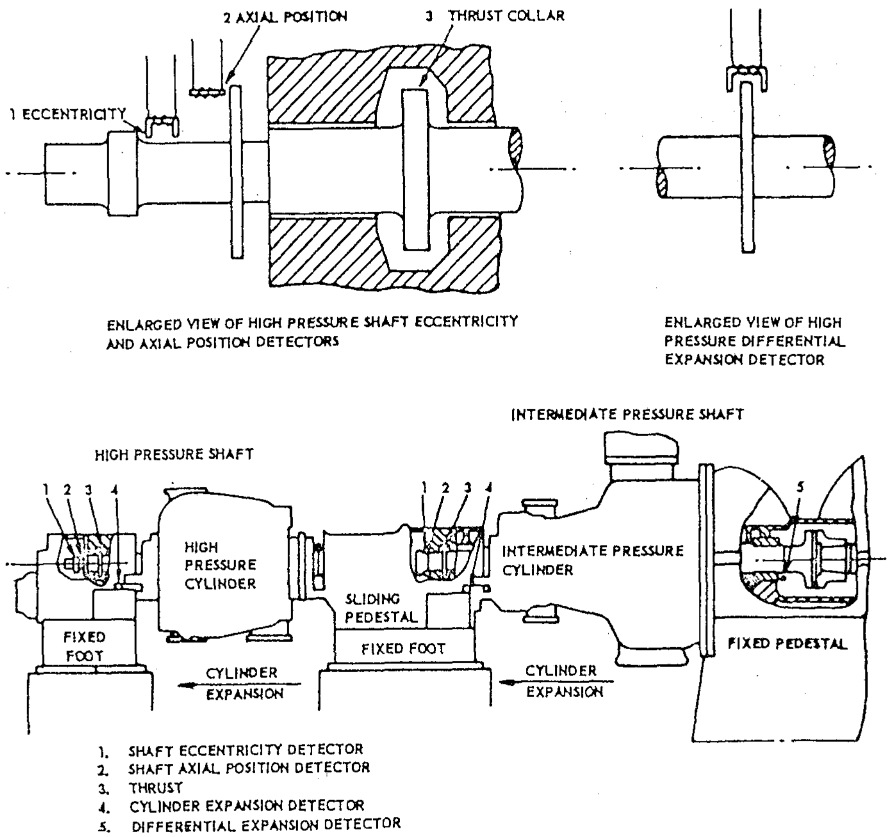


Figure 7. Relative positions of detectors on turbine.

Figure 10 illustrates the starting-up of a 60 MW turbine after a five day shut-down. This record shows the effect of a coincidence of contracting cylinder and a bending rotor. The bending was apparently caused by rapid heat flow into the rotor in the region of the gland. The eccentricity was reduced by lowering the temperature of the steam in the glands by reducing steam flow to them. The irregularities in the differential expansion were found to be caused by restraint on the movement of the high pressure pedestal.

It should be pointed out that this represents unusual conditions for, as can be seen, it was found necessary to use the barring gear during the run, as a shaft-straightening operation.

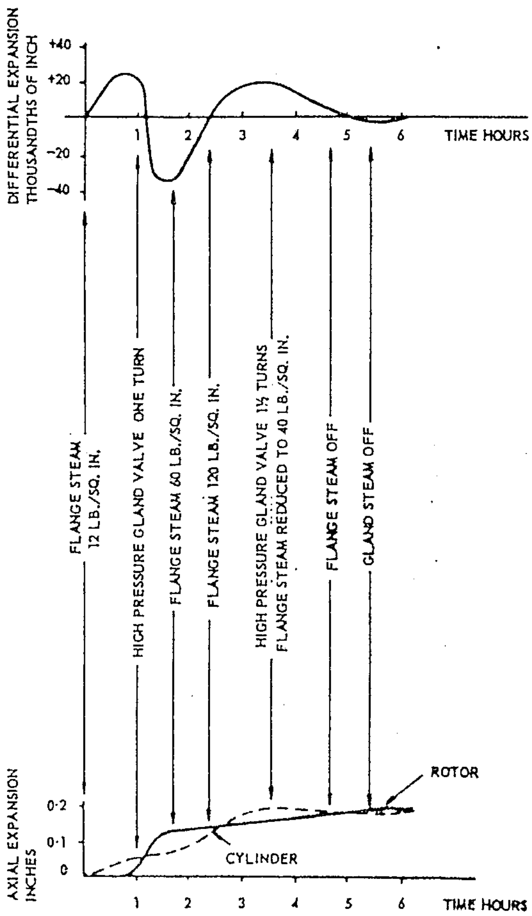


Figure 8

Figure 8. Differential and axial expansion during a bad start of a turbine (without supervisory equipment.).

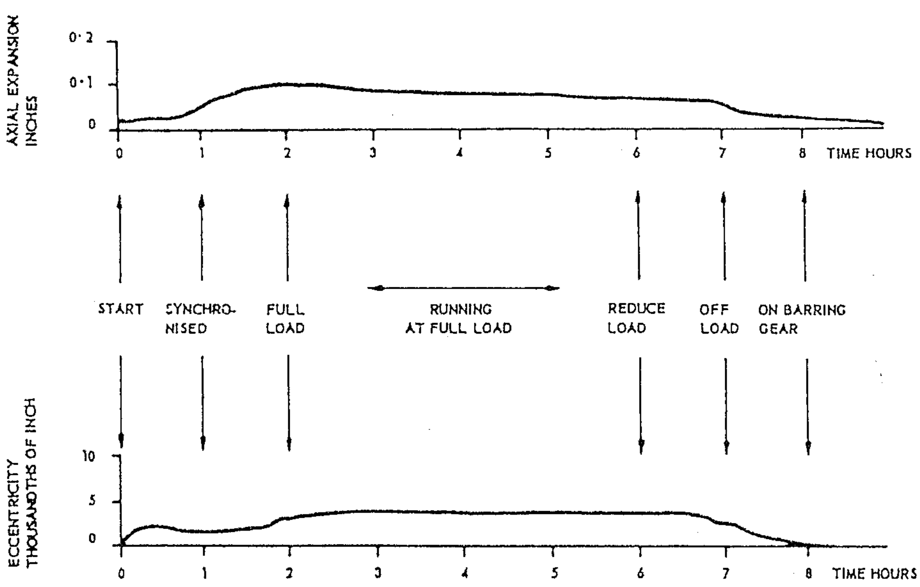
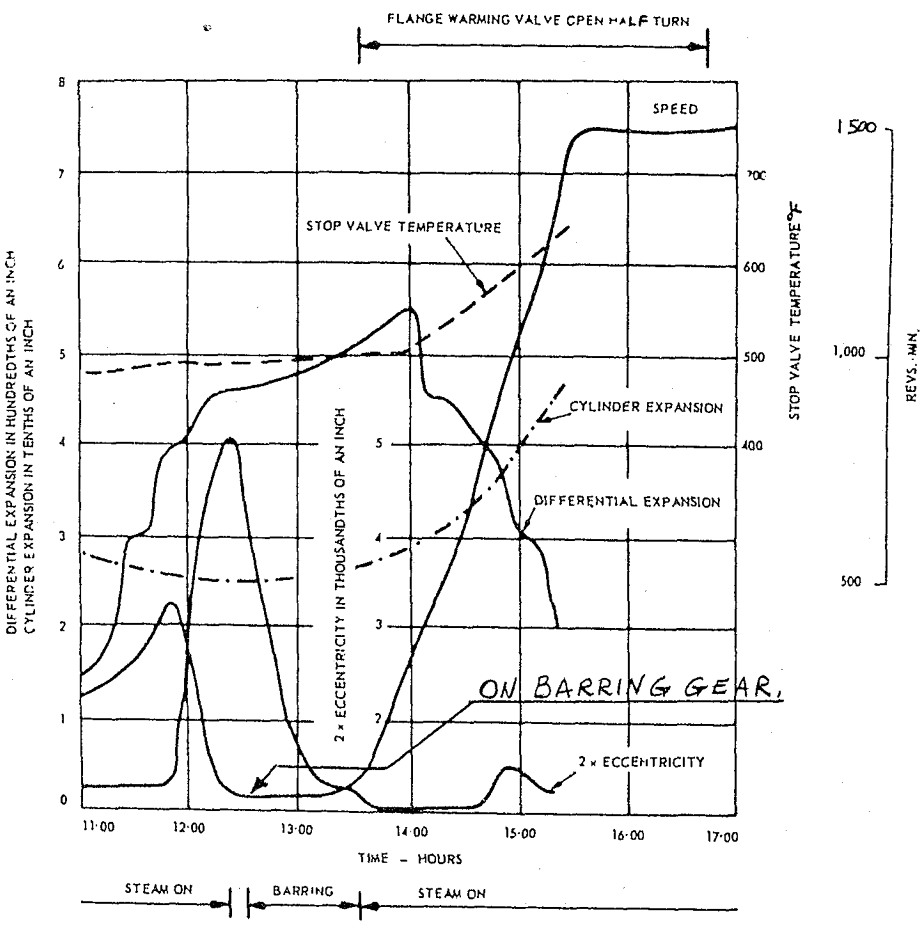


Figure 9. Axial expansion and eccentricity during a start-up, loading and shut-down of a large turbine (with supervisory equipment.).



<u>Figure 10.</u> Start of 60 MW turbine after five-day shut-down.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & Systems Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-7 - Supervisory Equipment

A - Assignment

1. Name 5 types of supervisory measurements that are indicated or recorded for a high temperature-pressure turbine.

2. Why is a high frequency generator used for the supervisory equipment?

3. Why is it important for the operator to know what the difference in expansion is between the rotor and turbine casings?

4. If the eccentricity meter records a high reading, what does this indicate?

5. What do the vibration detectors tell the operator about the operation of the shaft?

6. In figure 10 of this lesson the diagram shows that flange warming steam was on from 13:30 hrs. to 16:45 hrs. What was the purpose of this?

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-8 - Generator Protection

0.0 INTRODUCTION

Faults affecting A.C. generators are fortunately rare, but when they do occur they must be detected immediately. This is done by protective devices which can do either of two things:

(a) they can disconnect or "trip" the machine if the fault is internal, or if external conditions are so abnormal that continued operation would result in damage, or

(b) they can lessen the severity of a fault by taking protective action.

Modern methods of constructing large generators, coupled with gas or liquid cooling, enable much more efficient use of material to be made with a consequent reduction in size and weight per kva. This is very desirable, but makes it all the more necessary to study protective requirements carefully, as these have to take into account a wider range of thermal characteristics and in most cases a greater sensitivity to unbalanced loading conditions.

It is not possible to measure all hazards or dangerous conditions of operation with one device or protective system and a number are usually employed, which together guard against the rises associated with a particular application. This lesson will discuss some of the faults and hazards that may at times be experienced by the stator or rotor of a large generator and against which protection is required.

1.0 INFORMATION

Stator Faults

Breakdown of winding insulation results in either a fault to earth, a short circuit between turns or a short circuit between phases. Other faults, originating from defective joints, or inadequate or defective bracing of end-turns or terminals, will if undetected, reach the stage where there is a breakdown of insulation.

The above-mentioned failures are usually caused by overvoltage and/or deterioration of the insulation. Overvoltage is caused by system transients, lighting, switching surges or sudden loss of load, while insulation deteriorates because of foreign matter moisture, corona discharge and vibration. If solid insulation hardens then it may loosen some stator bars. It is necessary to reduce to a minimum the clearance time for any fault, so that the laminations are not damaged, the repair being effected by replacing the faulty stator bar. A delayed clearance, may damage the laminations so that re-stacking may be necessary and also there is a risk of fire.

A fault to earth is most likely to occur in the slot portion of a winding, with consequent arcing to the core at the point of the fault. In addition to conductor damage, burning and welding of the laminations occur, and while replacement of a conductor bar is serious enough, extensive damage to the laminations may result in having to rebuild the core down to the fault. To limit the damage as much as possible it is almost universal practice to connect an impedance, or earthing resistance between the generator winding neutral point and earth. Practice varies somewhat in the value of impedance used, ranging from the passage of rated current on the one hand to a low value on the other with a full phase-to-earth short circuit.

Phase to phase or three-phase short-circuit currents are not limited by the earthing impedance. Because of the severity of the shock both to machine and transmission system, extra care in maintaining the quality of the insulation at the end windings, terminals, and associated copper work is more than justified.

Unbalanced Loading

A three-phase balanced load produces a reaction field which to a first approximation is constant and rotates synchronously with the rotor field system. Any unbalanced condition can be resolved into positive, negative and zero sequence components.

The positive sequence component is similar to the normal balanced load. The negative sequence component is similar, except that the resulting reaction field rotates counter to the d.c. field system and produces a flux, which cuts the rotor at twice the rotational velocity. Double-frequency currents are thus induced in the field system and in the rotor body. The resulting eddy currents are verylarge and so cause severe heating of the rotors. So severe is this effect that a single-phase load equal to the normal three-phase rated current can quickly heat the rotor slot wedges to the softening point, resulting in their extrusion by centrifugal force to the point where they may strike the stator iron. Concentration of heating also occurs on portions of end-bells and surface fusion has been known to occur.

Rotor Faults

Electrical faults on generator rotors are usually winding faults to earth, faults between turns or sections of windings, and open circuits.

1. Rotor Earth Faults. Generator field circuits are not deliberately earthed and, hence, a single earth fault caused by weakening insulation is not serious. An alarm relay circuit is provided on larger machines. When an alarm is sounded it is recommended that the machine be taken out of service as soon as possible for examination. The presence of a single earth fault on a rotor obviously raises the potential on other parts of the winding. This is not usually dangerous in itself since the rotor voltage is only a few hundred volts. However unbalanced faults on the system induce high voltages in the rotor windings and it is these voltages under system disturbance conditions which may cause a second fault to earth on a rotor which already has one earth fault. A second earth fault will cause far more serious faults between turns or sections of the rotor windings, and may cause serious vibration owing to magnetic unbalance.

2. Rotor Turns of Section Faults. Any fault between turns on a rotor winding will distort the balance of the magnetic field. This may cause arcing at the point or points of faults, resulting in heating of local spots on the rotor, thermal unbalance and eccentricity. This will be accompanied by mechanical vibration of the rotor, and if this is serious, mechanical damage will follow. There is no type of protective gear suitable to deal with this sort of fault. The use of an over current relay is not possible because high over currents are induced in a rotor winding under external fault conditions. The initial warning of possible trouble is given by the earth fault alarm and in the event of rotor vibration, shorted rotor turns should be suspected. Rotor faults are often difficult to locate since they frequently disappear when the machine is at rest. This is because of the absence of centrifugal stresses when the rotor is not running at speed.

3. Rotor Open Circuits. Open circuits on rotor windings sometimes occur and they are shown by fluctuations in the rotor current, or even in complete loss of excitation and, therefore, of synchronism. In such cases it is usual for the machine to be switched-out manually.

Loss of Excitation

If a synchronous generator is completely deprived of its D.C. excitation it operates as an induction generator at a speed a little higher than the synchronous speed. It draws its excitation in the form of reactive power from the system. The current conditions on the machine are steady although they are of a high value. The system is thus deprived of the reactive power being supplied from the generator and is called upon to supply reactive power to the machined and a sudden and sometimes a large change in the system load current will take place. In a large, closely interconnected system, no serious consequences are likely since the automatic voltage regulators on the remaining generators in the system provide for rapid re-adjustment of the machines to their correct excitation level. However, the rotor of the affected machine will heat rapidly because of induced currents flowing in the iron, particularly at the rotor ends. Similarly, the stator windings overheat owing to the excessive currents.

Protection is not usually provided and the machine will continue to deliver its power to the system and the speed rise will be very small. The operator has a few minutes in which to restore the field before dangerous overheating occurs. If the field cannot be restored within this time the machine must be taken off load and disconnected from the system.

On some systems, however, where transmission feeders are long and plant is widely scattered, loss of field on one machine could easily and rapidly precipitate instability elsewhere. In such cases a directional type of protective gear can be used, which measures the impedance of the machine to the busbars. Loss of field causes a substantial drop in this impedance (from 200% to between 30% and 40% of its rating), and the consequent relay operation trips the machine.

Instability or Out of Step Operation

A generator may lose synchronism with the system if there is insufficient excitation or this may follow a serious fault on the system. The machine or machines, will slip poles and fluctuating currents will flow through the stator and rotor windings. This will be accompanied by voltage fluctuations and possibly loss of the use of station auxiliaries.

The first reaction of the operator is to disconnect the machine from the system. A quick remedy however is to boost the field current as well as reducing load where the machine is operating above 75% full load.

One of the functions of a voltage regulator is to prevent a machine becoming unstable during normal operation. This condition may arise following a fault on the system, the clearance of which has been delayed or which leaves the generating plant coupled through a high reactance. Loss of synchronism protection is not normally provided on turbo-generators although it is often essential on frequency converters.

Overload

A generator operating on a large system under continuous supervision is not in much danger from overloading. The power which can be generated is limited by the steam production, and so cannot rise unnoticed and be maintained for any appreciable period above the set programme. It is not usual practice to provide overload protection to supervised machines. In this sense overload or over-power must not be confused with overcurrent protection.

Motoring

If the turbine power source fails, a synchronized generator will continue to run at synchronous speed, drawing power from the system. This is referred to as motoring. With certain types of prime mover this situation could lead to mechanical danger but in general a supervised turbine-generator in a main power station will not pass into this state without the fact being obvious to the control staff.

The risks of a turbo-generator motoring are not to the generator but mainly to the turbine and secondarily to the system. Overheating of the steam turbine will occur if there is insufficient steam flowing through it to carry away the heat caused by windage if the vacuum is not maintained. The length of time before serious overheating takes place varies with the type of turbine from half a minute to half an hour. The conditions vary so much that no specific guidance can be given but the designer normally makes recommendations.

Special protection apart from normal indicators is rarely provided. Where motoring would be dangerous, however, sensitive directional power relays are used these are fitted with a time delay and sound an alarm. Tripping is usually initiated by the operator. The time lag is necessary to prevent alarms being given when synchronizing or during system disturbances.

Tripping and Alarms

It is modern practice because of its vital importance to have a d.c. battery supply at 125 volts or 250 volts for operating tripping and alarm relays. The main protective devices trip hand-reset master tripping relays which in turn carry out the many tripping operations.

In modern practice the main protective devices which operate to trip the master relay are:

(1) percentage differential relays

(2) high voltage earth fault

(3) low voltage earth fault

(4) standby earth fault

(5) generator transformer oil pressure surge

(6) generator transformer winding temperature

(7) emergency trip button

When the master relay is tripped it in turn trips:

(a) the main circuit breaker or circuit breakers

(b) neutral circuit breaker (if installed.)

(c) unit transformer circuit breaker

(d) field switch

(e) emergency stop valves on the turbine

It may also sound a trip alarm.

Summary

This lesson has discussed some of the faults and hazards experienced by large generators, and some corrective action that can be taken. However, for more detail regarding electrical protective equipment refer to the course on "Instrumentation and Control-General" T.T.2 level.

Dick Dueck

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3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-8 - Generator Protection

A - Assignment

1. What type of faults can occur when there is a breakdown of stator winding insulation?
2. What is usually the cause of stator insulation breakdown?
3. What undesirable affects are produced by unbalanced loading?
4. Name 3 types of electrical faults that can occur in the rotor.
5. What undesirable affects are produced if the generator continues to operate when there has been a loss of excitation?
6. What is meant by the term "motoring"? Why is it undesirable?
7. Can a generator fault trip the turbine, and if so, how?

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-9 - Feedheaters

0.0 INTRODUCTION

This lesson will describe a few details about design and construction of feedheaters and some heater protective devices that are generally employed in a steam power station.

1.0 INFORMATION

The design and construction of feedheaters can become quite involved and a manufacturer has to draw on many years of experience in order to do a competent job in this field. As has been mentioned previously the feed system is usually divided into low-pressure and high-pressure portions. Therefore, when considering the design of feedheaters it is convenient to divide them in low-pressure and high-pressure groups as the two types have many differences.

However, before proceeding any further the term "terminal temperature difference" (abbreviated as T.T.D.) should be defined.

Terminal Temperature Difference of a feedheater is the difference between the saturated temperature of the extraction steam supplied to the heater, and the temperature of the feedwater outlet from the heater.
The T.T.D. is illustrated in figure 3.

It is desirable to have the T.T.D. as low as possible - i.e. the feedwater outlet temperature from a heater should be as nearly equal to the saturated temperature of the heating as practicable. It is found that the T.T.D. is directly related to the surface area of the heater tubes. The greater the heater surface area, the lower the T.T.D. will be. However, the greater the heater surface area the greater the cost will be also and therefore the decision as to what T.T.D. to use in designing a heater has to be based on economic considerations.

Low Pressure Heaters

The design of L.P. heaters is in many respects a relatively straight forward task compared with those of the high-pressure group, although some factors of detail construction discussed in the latter part of this lesson also apply. The heat transferred to the feedwater can be regarded as a straight forward process of condensation - i.e. the steam gives up its latent heat of vaporization to the feedwater.

From economic considerations a low-pressure heater is usually designed to have a terminal temperature difference between outlet feed and saturated steam temperature of 10°F or less. It will be found that in practice it is extremely difficult to approach a terminal temperature difference less than 2°F. In addition to the provision of adequate surface, the design of the heater on the steam side is concerned only with baffling adequate to give proper tube support, to permit steam flow to all parts of the nest, and with appropriate venting arrangements to avoid the possibility of residence in the heater of pockets of incondensables which would both blanket otherwise useful condensing surface and cause corrosion. Figure 1 illustrates longitudinal and plan view cross-sections of a 'U' type L.P. heater. The 'U' tube arrangement permits differential expansion between tubes and shell. At the steam inlet a baffle is generally fitted to prevent impingement erosion of the tubes at this point. The tubes are fixed in the tubeplate by roller expanding and the structure of the tubeplate, header and bolting is relatively light.

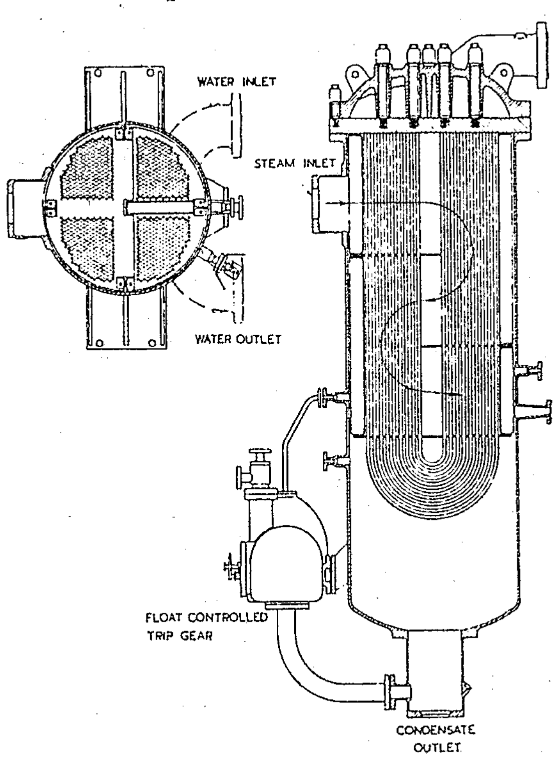


Figure 1. Longitudinal & plan view cross-sections of a 'U' type L.P. Heater.

Drain Flash Condenser

At the T.T.3 level we have explained how heater drains or "heater drips" are handled as they cascade towards the condenser. We also described the function of heater flashboxes in this regard. The water draining from the flash-boxes still contains a fair quantity of residual heat. This residual heat from accumulated drains of a number of flashboxes can be transferred to the feed-water if economically justifiable, by allowing such drains to be taken to a drain flash condenser. (D.F.C.) before they pass to the main condenser. This D.F.C. is similar in construction to a low pressure heater, and may in fact be incorporated with it on a common header as shown in figure 2. Extraction steam flows into the top of the L.P. heater and condenses. The condensate drains out the bottom of the L.P. heater, flashes into steam and flows into the bottom of the D.F.C. where it condenses again. The feedwater enters the double header, flows through the drain flash condenser tubes and from there through the L.P. heater tubes.

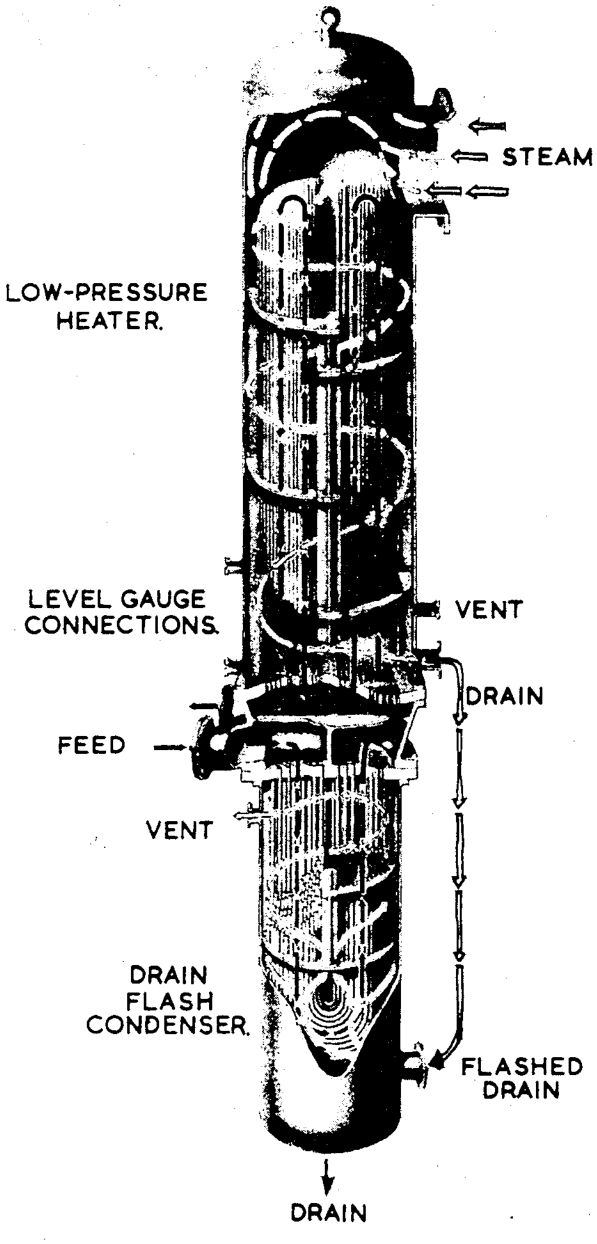


Figure 2. Sectionalized view of combined low-pressure feed heater and drain flash condenser.

High Pressure Heaters

The rapid increase in the size of turbine-generator units throughout the world and the use of ever higher boiler conditions in the search for improved thermal efficiency have brought many new problems to the design of high-pressure feedheaters. With the low pressure section of the feed train the change has been largely one of increase in size and relatively slight increase in operating pressures due to increased numbers of units in the feed train and the raising of deaerator operating pressures and temperatures. But with high-pressure heaters, especially if connected to the boiler feed-pump discharge, the operating pressures have increased swiftly, as have the operating temperatures of the heater due to higher conditions in the extraction steam.

With the advent of extraction steam of high temperature it has become economically necessary to take advantage of the often considerable amounts of superheat, which this steam contains and to decrease the terminal temperature difference. Consequently the conception of heat transmission on the steam side of the heater has to cater for such circumstances.

With the steam appreciably superheated as it enters the heater, desuperheating has to take place before condensation occurs. The steam, while desuperheating behaves as a gas and heat transfer to the surface of the tube takes place by convection, while, with wet steam heat transfer is conductive from the liquid condensate on the outside of the tube to the feedwater on the inside of the tube. The arrangement of heater surface and baffling must therefore take account of this, and that portion of the heater nest which the steam first traverses must be designed on the basis of convective heat transfer. Thus for heat-transfer considerations the two portions of the nest the desuperheating and the condensing can be treated separately.

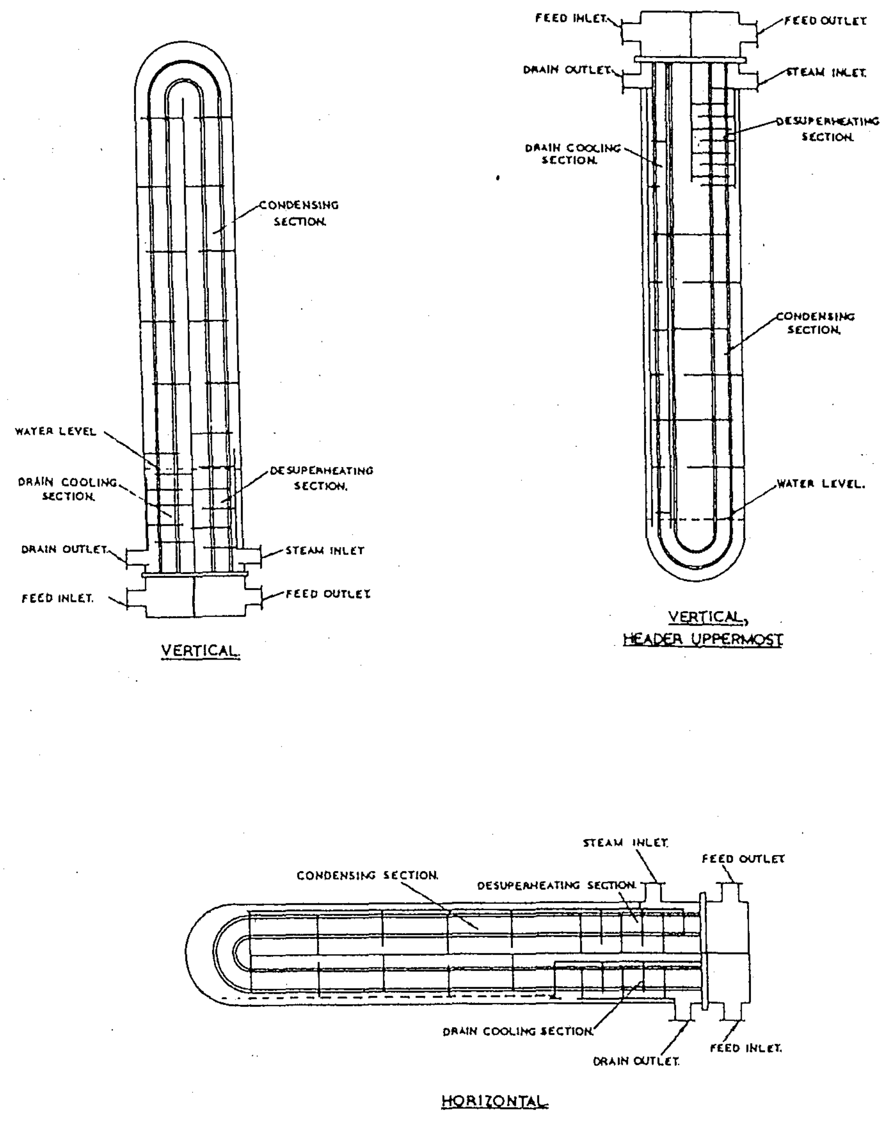


Figure 3. Representative methods of arranging desuperheating, condensing and drain cooling sections in high-pressure feed heaters.

Figure 3 illustrates arrangements for desuperheating, condensing and drain cooling sections in high-pressure heaters. Notice that the desuperheating section has baffling arranged so that the superheated steam flows in baffled cross-flow generally. counter-current to the flow of feedwater in the tubes.

The number and size of heaters associated with large turbine-generators demands that their configuration shall be relatively flexible to enable them to be fitted in neatly with the general plant arrangement. Horizontal heaters were often used on early designs, but later vertical heaters with headers at the top became standard practice.

On large machines, however, this brought in its train certain disadvantages of feed-pipe layout, and also problems, in larger sizes of heaters, in lifting the heavy nest from the shell and in threading it back on replacement.

Current practice recognizes the advantages of the vertical heater with its header at the bottom. Feed-pipe arrangement is usually cased and the nest can be exposed by removal of the relatively light shell, which may be in several sections to reduce crane height. Header design can also be arranged so that both the header cover plate and the nest can be removed without breaking the high-pressure feed-pipe connections to the header.

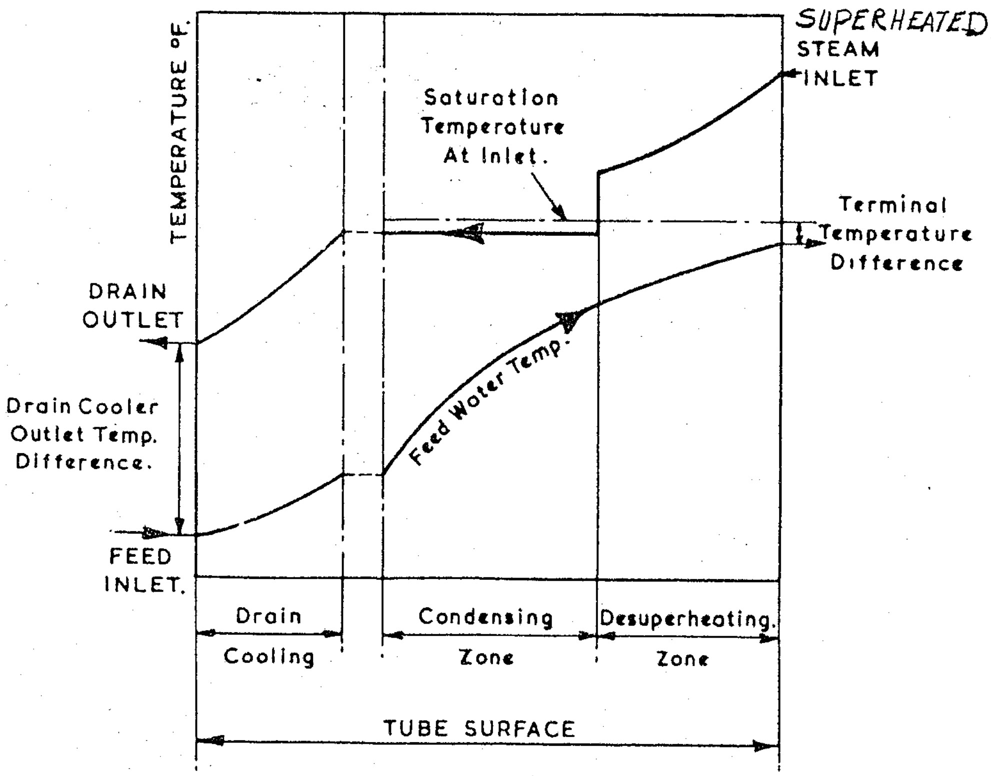


Figure 4. Temperature diagram for desuperheating type feed heater.

Figure 4 illustrates the pattern of temperature difference involved between steam and feedwater for a desuperheating-type feedheater. To the left of the diagram is a portion showing the temperatures in a drain cooler associated with the heater. The provision of drain-cooling surface will have been determined by the initial consideration of economics of the feedheating cycle, and such surface may be included in the feedheater proper or in a separate vessel.

The superheated steam reduces considerably in temperature in the desuperheating zone and remains at constant temperature during condensation. The difference between the solid horizontal steam temperature line and the dotted line is shown to represent depression of the saturated steam temperature caused by pressure drop of the steam through the tube nest.

The desuperheating zone in a high-pressure heater enables attainment of a terminal temperature difference of less than 2°F. From figure 10 one can also see that for a certain design it is possible (depending on the degree of superheat) to provide a feedwater outlet temperature which is above the saturation steam temperature. This is usually called "negative terminal temperature difference."

Radial-Pitch Feedheater

In heater design consideration must also be given to the effect of high-temperature extraction steam on the components,

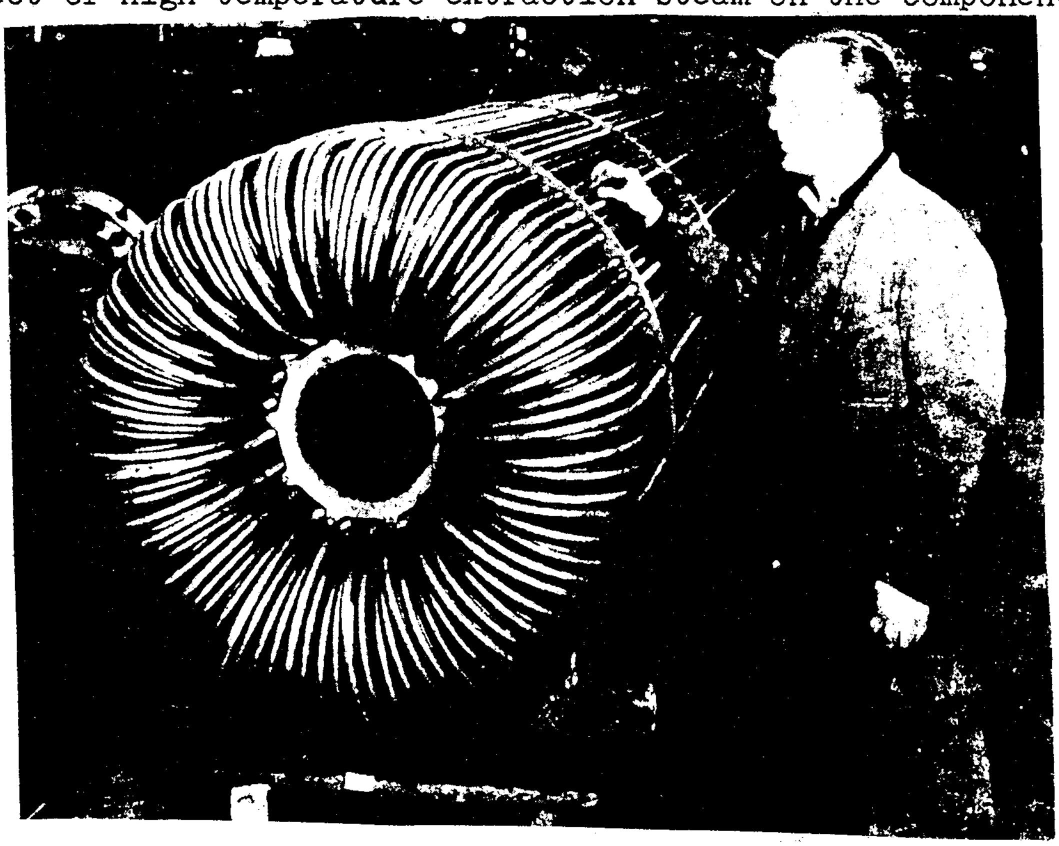


Figure 5. End view of radial-pitch feedheater nest.

because of non-uniform heat distribution, particularly if the steam is introduced to one side only of the heater. Figure 6 shows the nest layout of a heater designed to deal with this situation.

The central inlet disperses the hot steam equally round the nest, and also ensures that it is desuperheated before coming into contact with the shell. Thus the shell will not be distorted and overstressed due to uneven heating. Figure 5 shows an end-view of this type of heater. The arrangement of tubes, to permit this central inlet, is radial, as can be seen from figures 5 and 6. Therefore, this is known as a radial-pitch feedheater.

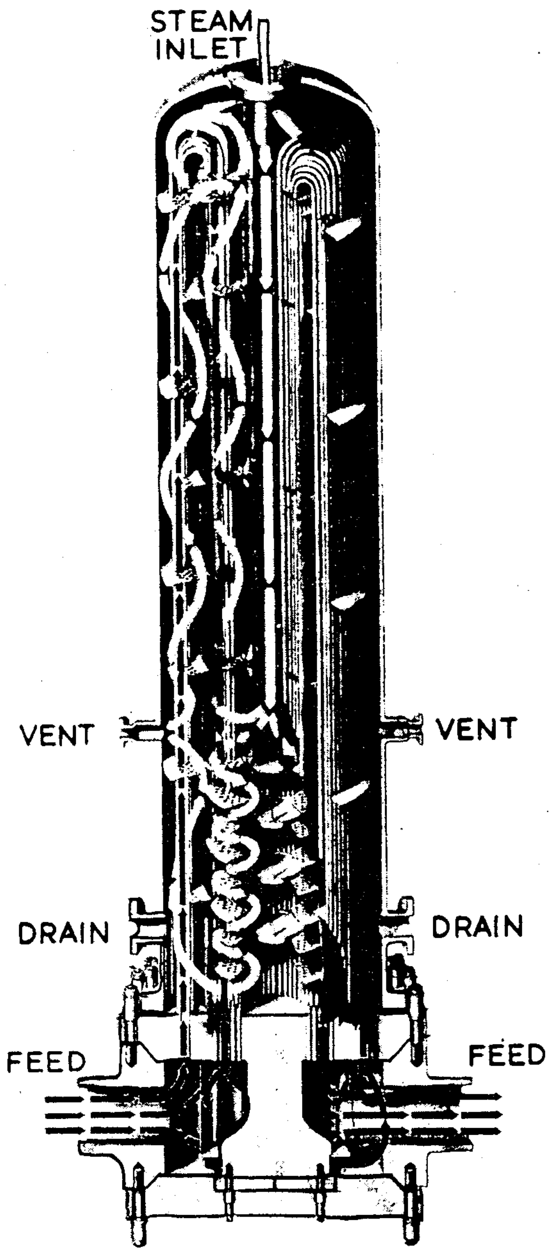


Figure 6. Sectionalized view of radial-pitch feedheater.

Heater Headers.

Design of the heater headers involves three problems:-

1) to provide sufficient metal in tension or shear to affix the header components to one another.
2) to provide a joint that will not leak.
3) to provide access to the internals of the header, preferably without disturbing the feedwater connections.

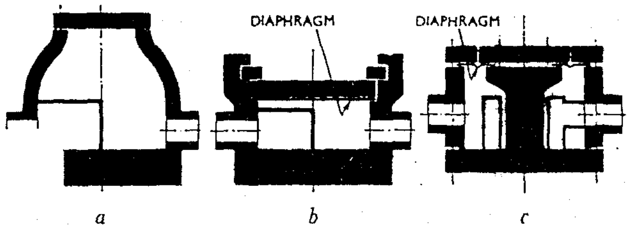


Figure 7. Elements of construction of three types of high-pressure feed-heater headers.
a) Bottle type
b) Ring-in-shear type
c) Radial-pitch tube nest.

Figure 7 shows three types of headers that have been used in heater design. The bottle type provides a strong seal but access to the header internals is difficult during manufacturing and maintenance.

The ring-in-shear type has been used successfully up to the highest operating pressures yet built. It has the advantage of reducing the number of leak-tight joints to one.

The radial-pitch type header shown in figure 3 c) is the one that has been used for the radial-pitch heater shown in figure 6. It has a mushroom-shaped centre post to which the cover plate is bolted as well as to the outer rim of the header. This makes the header quite strong and still reasonably accessible. It is a common type nowadays for vertical heaters with the header at the bottom.

Tube fixing.

For L.P. heaters the conventional method of making tube-to-tubeplate joints is the well known roller expanding process, similar to the method used in fixing condenser tubes. During this process the tubes, and to some extent the tubeplates are permanently deformed.

For H.P. heaters, the practice nowadays generally is to weld the tubes to the tubeplates. This can be done by either welding the tube to the tubeplate on the feedwater face of the tubeplate, or by welding the tube to a stub on the steam face of the tubeplate.

Tube Material and Strength

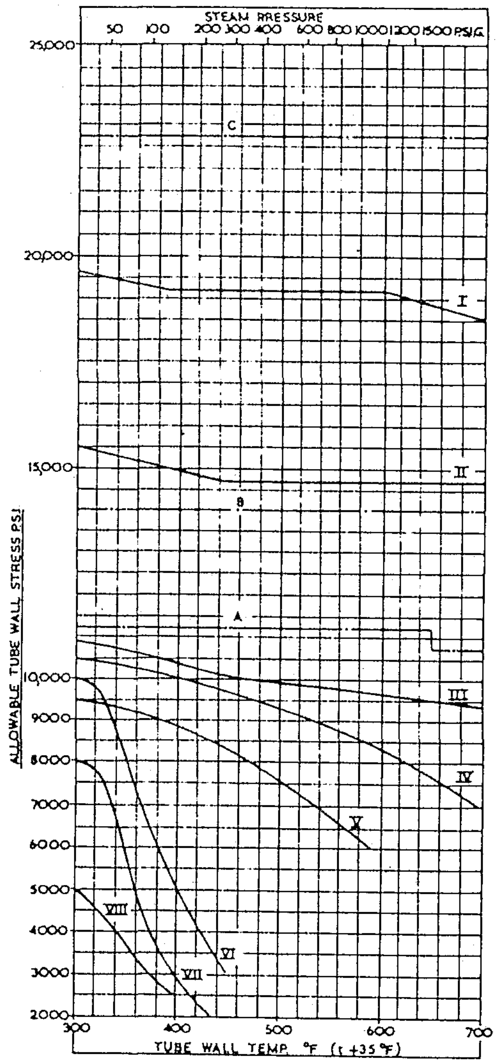
Heater tube material has to be selected on the basis of the following characteristics:

1) It has to be strong enough to withstand working temperature and pressures.

2) It has to have high thermal conductivity.

3) It has to be resistant to corrosion.

For many years it has been the practice to use copper or brass (70% copper, 30% zinc) for L.P. heaters, air ejectors, low-pressure drain coolers etc. In some cases aluminum is now employed. The tubes for H.P. heaters of course have to withstand much higher stresses and in this case 70/30 and 90/10 cupro-nickel alloys have been widely used.



TUBE MATERIAL.

NON FERROUS.

I MONEL METAL STRESS RELIEVED.
II MONEL METAL ANNEALED.
III 70/30 COPPER-NICKEL.
IV 80/20 COPPER-NICKEL.
V 90/10 COPPER-NICKEL.
VI ADMIRALTY BRASS.
VII RED BRASS.
VIII COPPER.

FERROUS.

A. MILD STEEL B.S. 1508/151 & 171.
B. ½% MOLYB. 1508/240.
C. D.T.D. 740 LOW ALLOY STEEL (MOLYB/BORON.)

Figure 8. Curves of maximum allowable tube-wall stress for usual materials.

For very high pressures monel metal is being used in some cases. It consists of 30% copper, 67% nickel, the remainder being made up of manganese, silicon etc. Monel metal is very expensive and therefore not used often. The alternative is to use steel. However, its heat conductivity is not as good as for brass and cupro-nickel alloys and therefore the heater has to be built with a larger surface area.

Figure 8 gives a comparison of maximum allowable tube-wall stress for various materials.

For Nuclear Stations, the pressure of feedwater is not very high and the pressures for the H.P. heaters are no higher than for L.P. heaters in a modern conventional station. For this reason copper or brass are satisfactory tube materials for both the L.P. and H.P. heaters in a nuclear power station.

Feedheater Protective Devices

On large turbine installations the volume of steam stored in the feedheaters and extraction steam pipes is necessarily large, and, particularly in the H.P. heaters, the energy of this steam is considerable. When a turbine trips this steam could flow back into the turbine again and contribute considerably towards the overspeed level attained.

To prevent this, the extraction steam lines generally are provided with special forced closing non-return valves fitted as close as possible to the turbine. These valves are arranged to close on a signal from the auxiliary governor and also by the turbine trip gear.

There is danger of feedheaters flooding in case a tube should burst or if a condensate drain should get plugged. The feedheater and piping could completely fill up and water could spill into the turbine. This would result in turbine blade damage. For this reason groups of heaters are generally provided with piping and valve arrangements such that heaters may be automatically by-passed on the feedwater side. This stops condensation of steam in the heater and avoids the possibility of water passing back along the extraction steam pipe.

Heaters are generally provided with some kind of condensate level indication and a float switch. When the level becomes too high the float switch actuates an alarm and automatically operates appropriate valves to by-pass the heater.

D. Dueck.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment and System Principles - T.T.2

4 - Turbine, Generator and Auxiliaries

-9 - Feedheaters

A - Assignment

1. Define terminal temperature difference.
2. What is the purpose of baffles in a heater.
3. What is meant by a desuperheating feedheater?
4. What is a negative terminal temperature difference?
5. What is the advantage of a radial-pitch feedheater?
6. What hazard is involved when a heater starts flooding?

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3 - Equipment & System Principles - T.T.2

4 - Turbine, Generator & Auxiliaries

-10 - Steam Cycles--Nuclear

0.0 INTRODUCTION

In nuclear generating stations up to the present time the temperature and pressure conditions of steam coming from the boiler are relatively low as compared to conventional coal-or oil-fired generating stations. This is because the materials used for the reactor vessels are not able to withstand high temperatures and pressures. As a result the steam at the emergency stop valves is generally at or near saturation conditions. This means that as soon as some steam expansion takes place, the pressure will have dropped below the saturation line and condensation will occur already in the second or third stage in the turbine. The formation of moisture in the turbine at such an early stage presents special problems.

Power production by means of nuclear generating stations is in too early a stage of development for us to refer to anything as typical, as yet. However, this lesson will deal with the problem presented by moisture in a steam turbine and one way this problem can be minimized. It will also briefly describe a cycle proposed for a 500 MW unit.

1.0 INFORMATION

Erosion

When blade erosion occurs in a steam turbine it is a serious problem because of the following reasons:

1.) Damage due to flying pieces of metal; chips of metal may break off from badly eroded blades which then strike and damage blading in a lower pressure stage. If these chips are carried through the last stage they may strike and puncture condenser tubes.

2.) Reduction in efficiency; badly eroded blades operate less efficiently, resulting in higher operating costs.

3.) Higher maintenance costs. Eroded blades have to be replaced every so often, and this is an expensive operation.

Two main factors producing blade erosion are wetness of steam and the velocity of water drops on the blade. Erosion of turbine blading can also be caused by solid matter being carried over from the boiler although this happens very rarely.

In a nuclear power station it is not likely that erosion will occur in the first or second stage. Although water particles may be formed in these stages they appear as a fine mist and their velocity will not differ much from the carrying steam. When, however, the mist particles eventually coalesce, the inertia and size of the resulting larger drops cause their velocity to fall below that of the steam. It is under these conditions that erosion will occur.

Experience has shown that erosion is serious with wetness exceeding about 10% where present day blade tip speeds of 1400 ft./sec. to 1500 ft./sec. are in use. Figure 1 shows several photographs of the diaphragms of a 20 MW impulse type steam turbine which has been operating in a nuclear station for a little over two years. Its inlet steam conditions are 400 psig., 450°F. This is dry saturated steam. Shaft speed of rotation is 3600 rpm.

Normally, the worst erosion occurs on the moving blades near the exhaust end. However, in this particular case the moving blades were all in reasonably good shape while the diaphragms in the high pressure end of the turbine were badly eroded.

You will recall that in an impulse turbine, a pressure drop takes place only across the diaphragms, which means that steam gains in velocity as it passes through the nozzle. In figure 1(a) you can see that the nozzles are worn away badly at the outlet end. The steam (together with the moisture) not only gains in velocity as it passes through the nozzles, it also has to change its direction of flow. Condensate, being relatively dense, offers a resistance to this change of direction and consequently will bear heavily against the nozzle wall, causing the wall to be eroded. Figure 1(b) is a view looking at the outlet side of the diaphragm. Notice the paths eroded on the shrouding due to leakage steam flowing between the diaphragm and casing. This leakage steam is undergoing a throttling process because of the drop in pressure across the diaphragm. This is sometimes also referred to as wiredrawing.

Turbine blading is designed so that there will be a smooth path of flow from nozzles to moving blades, and moving blades to nozzles etc., so that steam does not strike the leading edge of the blading. However, figure 1(d) shows that condensate has been striking the leading edges of the nozzles. No doubt the designer did not expect the high percentage of moisture that obviously was there and hence designed the angle of moving blades more for dry steam flow than wet steam flow. Water droplets move more slowly

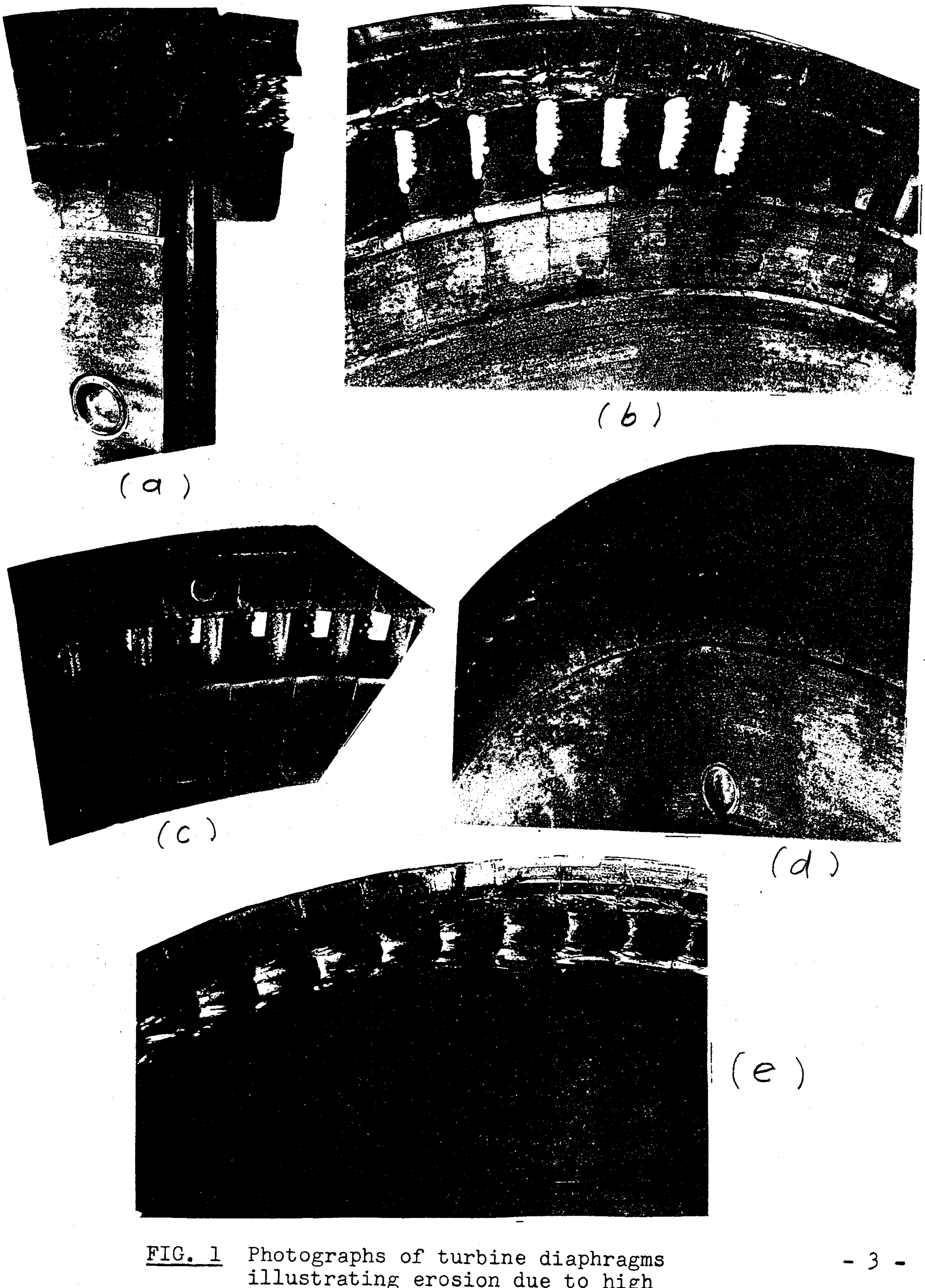


FIG. 1 Photographs of turbine diaphragms illustrating erosion due to high percentage moisture in steam.

than the steam due to their weight, hence what is resulting is that the moving blades are directing the water droplets in the steam right at the leading edges of the nozzles of the diaphragms.

Now that we have discussed the effect that moisture can have on turbine blading, we can describe ways in which attempts can be made to minimize blade erosion. These are as follows:

1.) Reduction of wetness of steam in the turbine exhaust.
2.) Improvement in blade materials and design to resist erosion.
3.) Arrangements for extracting suspended moisture from the steam during its passage through the stages of the turbine.

Under item 1 the wetness of steam can be reduced by using higher initial steam temperatures. However, since this is still very difficult in nuclear stations for the reasons mentioned previously, this is not practicable. However, reheating will reduce wetness also and this method can be used in nuclear stations.

Under item 2, the blades can be improved by making the leading edges of low pressure blading of stelite, a method which has already been described in the lesson on "Turbine Blading" T.T.2 level. Or else a better grade of steel can also be used, which may be case-hardened by special process.

To provide for item 3, modern turbines invariably have special arrangements for the removal of moisture in stages where wet steam is expected. One method has already been described in figure 7, of the lesson on "Turbine Construction", T.T.2 level. That is, grooves are provided in the casing between diaphragms or rows of blades, and connections are provided for the water to drain into the condenser. Up to 50% of the entrained moisture can be removed in this way.

<u>Moisture Separator</u>

Another vital method of moisture removal has been introduced into the steam cycle with the advent of nuclear generating stations. Steam, after it has expanded in the high-pressure part of the turbine, and contains about 11% moisture by weight, is led through a moisture separator where moisture content is reduced down to between 5% to 0%.

There are various designs for moisture separators; for example the scrubbers in a boiler drum separate moisture from steam before it flows into the main steam line, or it may be of the type where wet steam enters the separator tangentially; the moisture drops downward and steam rises upward. Figure 2 shows a type where flow is horizontal from inlet to outlet. However, each one of these separators employs a centrifugal action for separating the moisture.

In the separator shown in figure 2 the working elements are the helicoid tuyere and the vane. "Helecoid" refers to a spiral or helix type of shape and "tuyere" refers to a pipe or nozzle through which steam is forced. In this case the helecoid tuyere consists of long twisted blades held together in the shape of a cone. The spaces between blades are shaped like long thin slits or very high turbine nozzles. Steam at fairly high velocity enters from the left-hand end and as it flows through the helicoid tuyere it attains a spiral path. The centrifugal action thus established causes the denser moisture to move outwards due to centrifugal force and thus separates from the steam. The vane produces a similar effect. The moisture drips to the bottom and collects in the two drains as shown. Steam leaves the moisture separator with around 5% to 0% moisture.

Steam Reheater

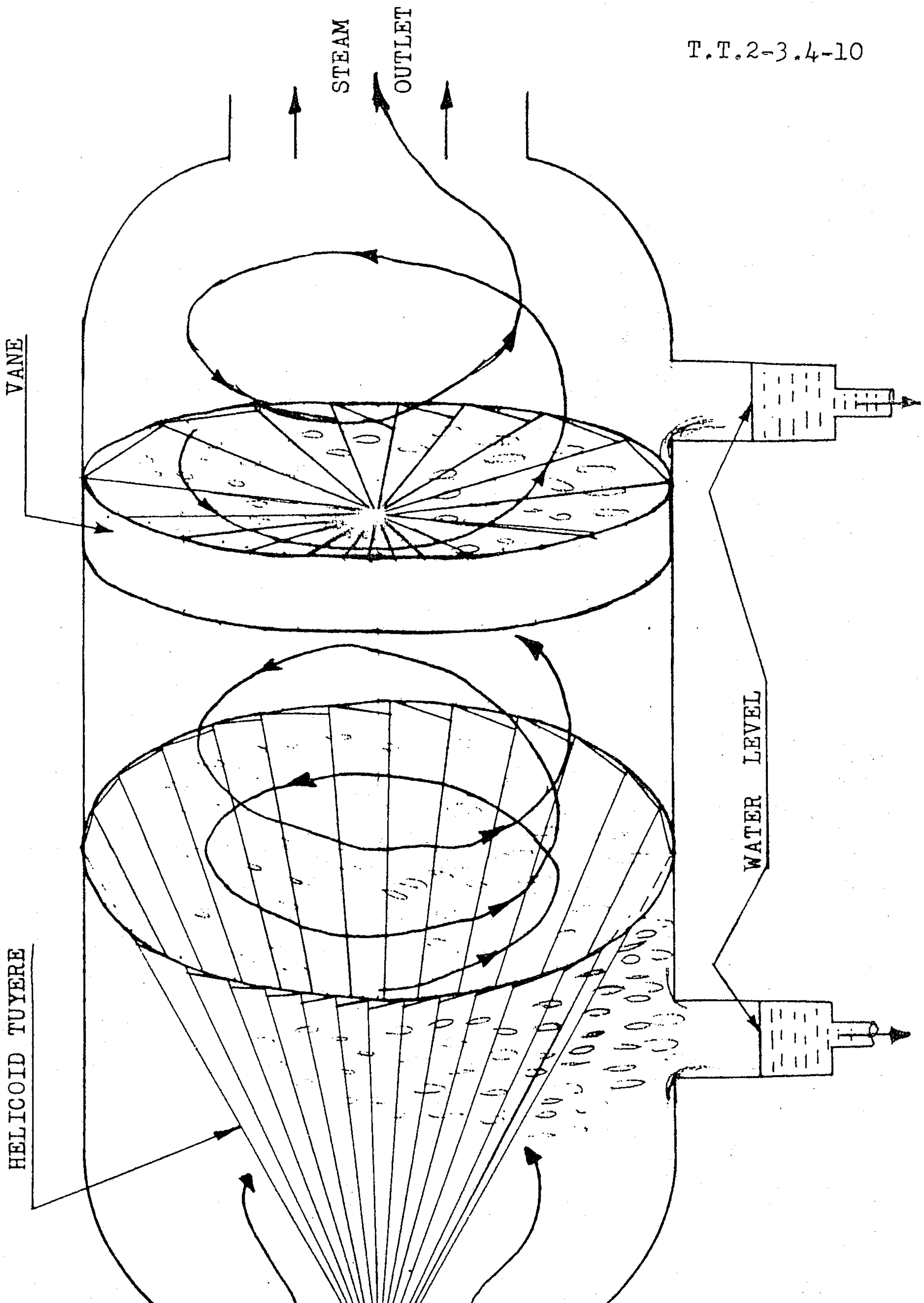
In a conventional station when there is a reheater, it is an integral part of the boiler and steam has to return to the boiler to be reheated. However, in nuclear stations the boilers are not made to accommodate reheaters. Also, since the pressure in nuclear steam cycles is already low to start with, the pressure losses involved in leading steam back to the boiler would mean too much of a loss in efficiency. Therefore the reheater when there is one in a nuclear steam cycle, is located near the turbine and downstream from the moisture separator. Live steam from the boiler is used as a source of heat as shown in figure 3. An example of the type of conditions one could expect for wet steam coming from the H.P. turbine is shown entering the reheater on the right-hand side. In the reheater the turbine steam increases in temperature by $430 - 305.3 = 124.7^{\circ}F$. Looking on the Mollier diagram, it will be noticed that this brings the turbine steam into the superheat region at the reheater outlet. The steam from the boiler condenses as it gives up its heat. This condensate is introduced back into the feedwater system at a suitable place.

The reheater as can be seen from figure 3 is a type of heat exchanger with live steam flowing through the heater tubes and wet steam from the turbine flowing around the outside of the tubes.

Steam Cycles

Figures 4 and 5 illustrate two studies of the type of cycles that could be used for a 500 MW unit in a nuclear power station.

FIGURE 2 DIAGRAM ILLUSTRATING MOISTURE SEPARATOR



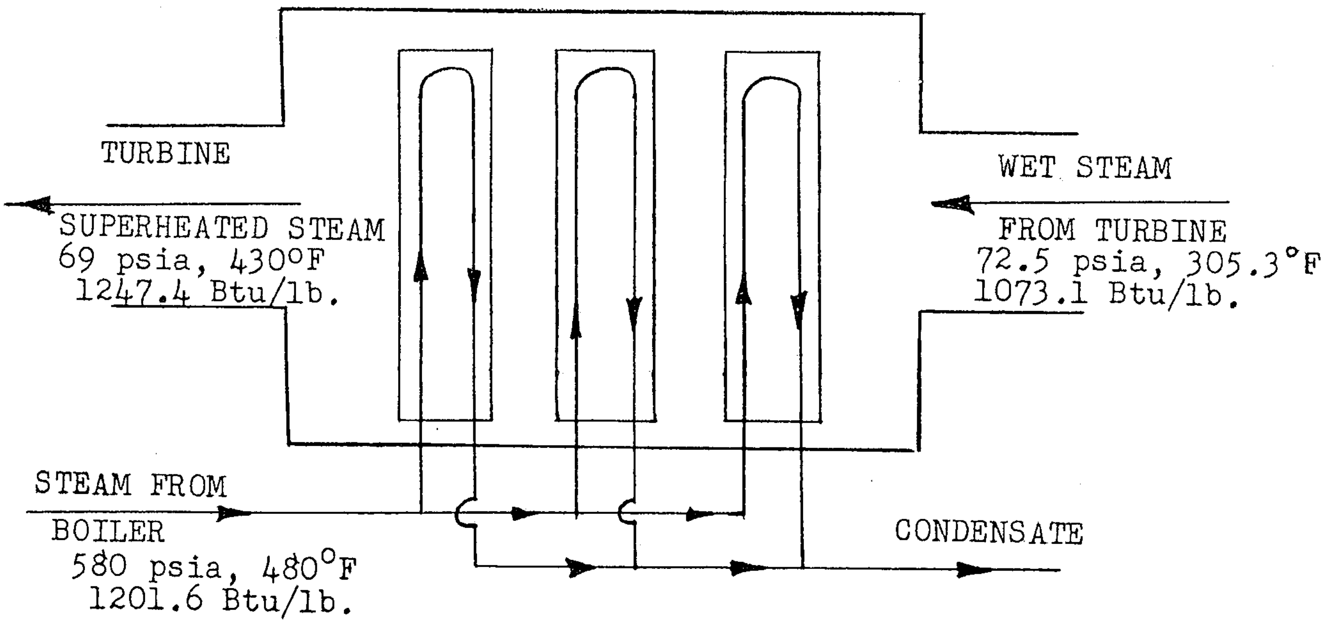


FIG. 3. Diagram illustrating a reheater for steam conditions normal for a nuclear power station.

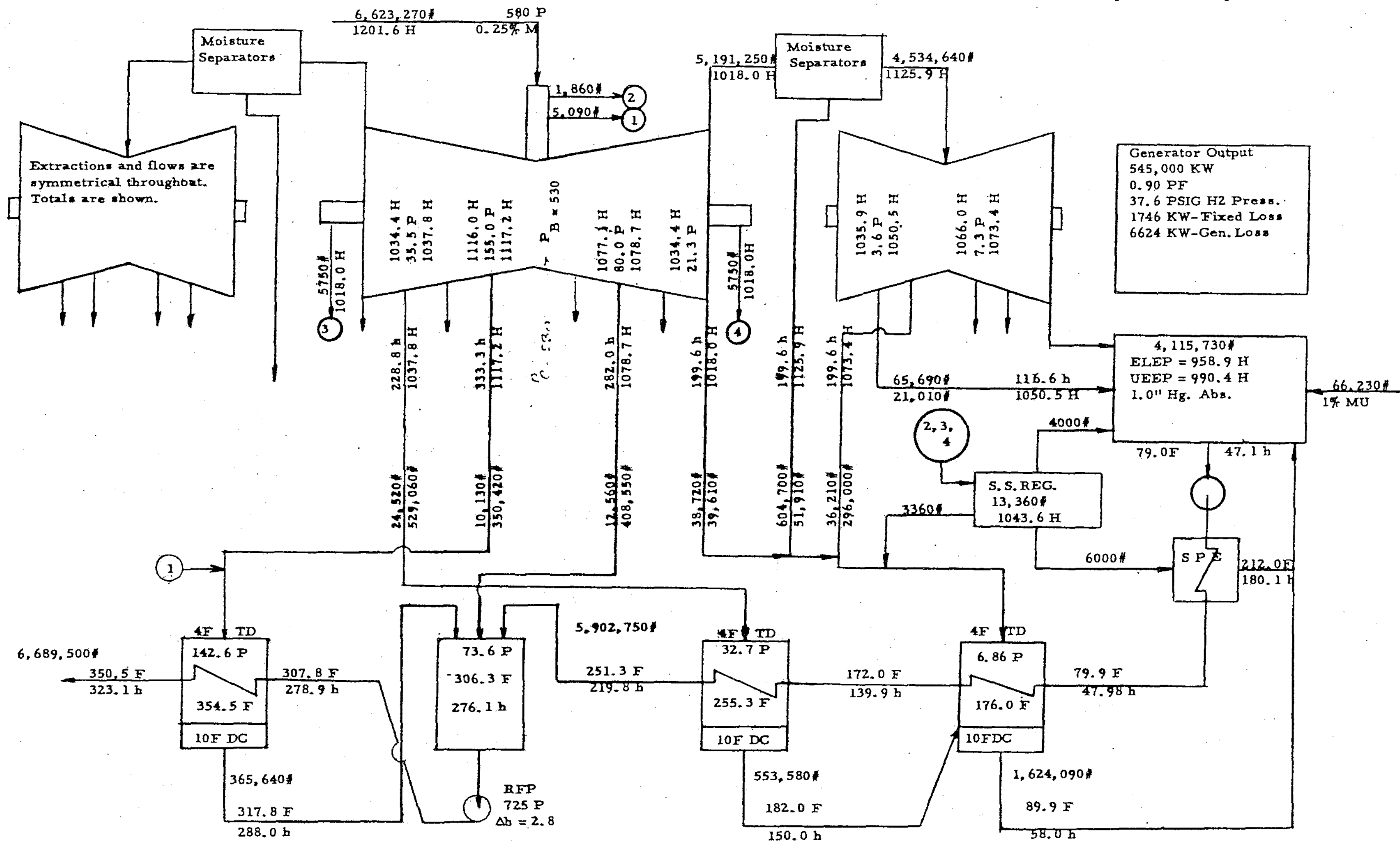
In both illustrations the cylinders are of the double flow (or opposed flow) type. Figure 4 uses two separators but without reheat while figure 5 uses one moisture separator with a single stage reheater. Reheating brings the steam conditions in the superheat region. Notice that figure 4 has a quadruple-flow exhaust section, while figure 5 has a six-flow exhaust section. The steam in the H.P. cylinder of figure 4 undergoes a much greater expansion than steam in the H.P. cylinder of figure 5. Greater expansion means more stages and a longer turbine, hence a more expensive H.P. turbine. However, which of the cycles is the more desirable depends upon an economic study of the two.

The pressure, temperature, enthalpy and flow conditions as given are self-explanatory. Some of the abbreviations used are explained below:

T.D. = heater terminal temperature difference.
D.C. = drain cooler approach difference. This is the difference between the heater drain temperature and the temperature of the feedwater entering the heater.
M = moisture.
E.L.E.P. = expansion-line end-point
MU = make-up water.

PRELIMINARY PRESSURE DISTRIBUTION

LEGEND:
H, h = Enthalpy, Btu/Lb.
# = Flow, Lb/Hr.
P = Pressure, PSIA
F = Temperature, F degrees



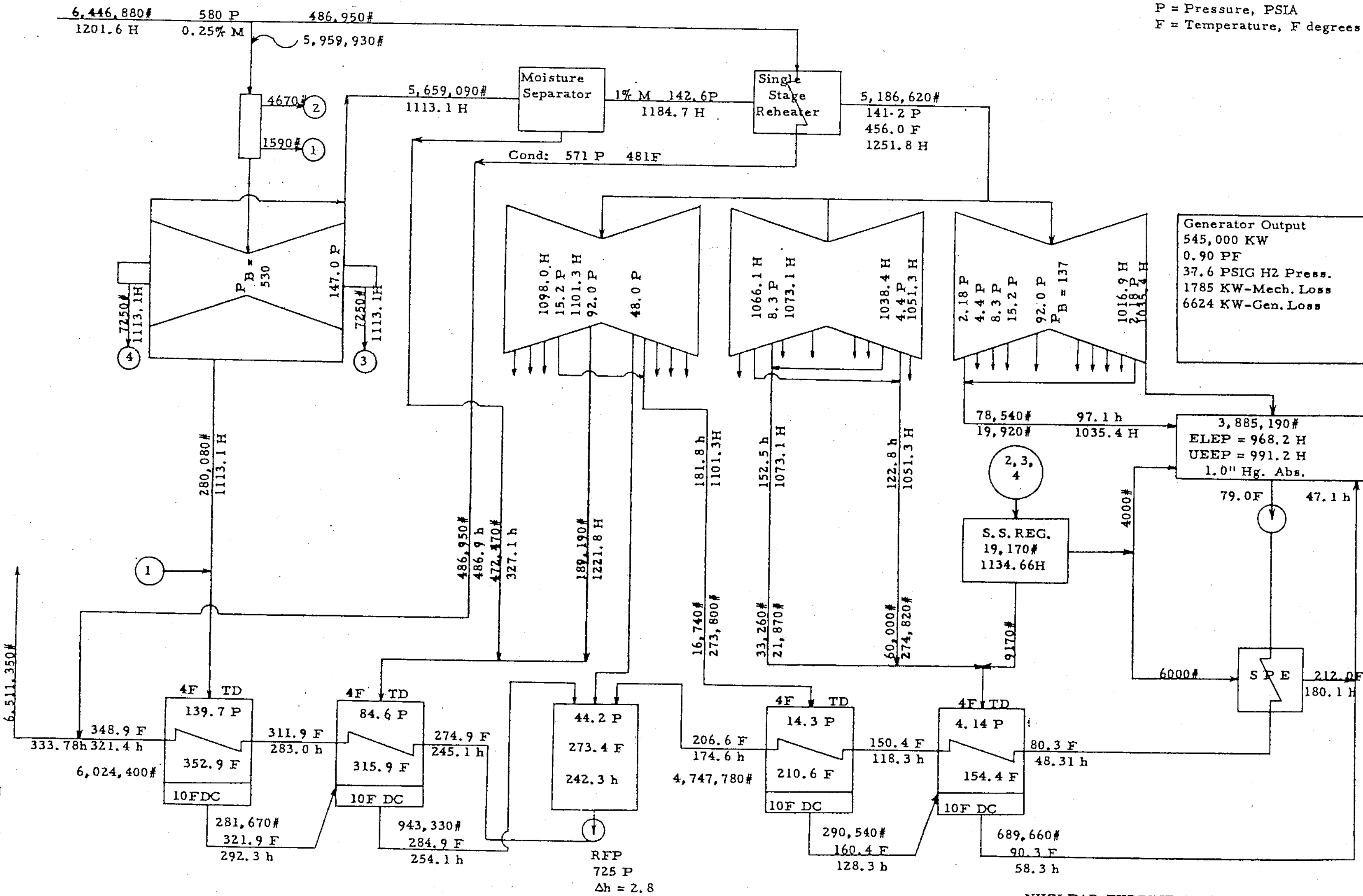
NUCLEAR TURBINE-- NON REHEAT
545,000 KW @ 1.0" Hg. Abs. & 1% MU
~~TC6F-38" LSB~~ 1800 RPM TC4F 43" LSB
580 P 1201.6 H 0.25% M
GEN: 600 MVA @ 0.90 PF & 45 PSIG H2 Press. (LIQ)

GROSS HEAT RATE = $\frac{6,623,270\ (1201.6 - 323.1)}{545,000}$ = 10,676 BTU/KW-HR

FIGURE 4

PRELIMINARY PRESSURE DISTRIBUTION

LEGEND
H, h = Enthalpy, Btu/Lb.
# = Flow, Lb/Hr.
P = Pressure, PSIA
F = Temperature, F degrees



NUCLEAR TURBINE-SINGLE STAGE REHEAT
545,000 KW @ 1.0" Hg. Abs. & 1% MU
TC6F-38" LSB 1800 RPM
580 P 1201.6 H 0.25% M
GEN: 600 MVA @ 0.90 PF & 45 PSIG H2 Press. (LIQ)

$$\text{GROSS HEAT RATE} = \frac{6,446,880\ (1201.6 - 333.78)}{545,000} = 10,266 \text{ BTU/KW-HR}$$

FIGURE 5

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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-10 - Steam Cycles--Nuclear

A - Assignment

1. What is the objection to turbine blade erosion?
2. Give 3 ways in which blade erosion can be minimized.
3. Give 2 methods that are used in a nuclear station to remove moisture from the turbine steam.
4. Describe how moisture is removed from wet steam in a moisture separator.
5. Explain how it is possible to change saturated steam to superheated steam in a heat exchanger, by using as the heat source, steam, which is itself at the saturation point as shown in figure 3 of this lesson.