NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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4 - Turbine, Generator and Auxiliaries

-1 - How Steam Turbines Work

0.0 INTRODUCTION

At the T.T.4 level we have introduced the steam turbine in a general way. We have said that it is one of the major components in a steam power station and that it is used to drive an a-c generator to produce electricity. We can now proceed to discuss the turbine in a bit more detail. This lesson will describe in elementary terms how a steam turbine works.

1.0 INFORMATION

Steam turbines have two main working elements: 1.) nozzles and 2.) blades or buckets.

In simplest terms a nozzle is a hole in a wall separating a region of high pressure and a region of low pressure. The steam, of course, will flow through the hole or nozzle from the higher-to the lower-pressure region. It flows through the nozzle at a velocity depending on the pressure difference; the greater the difference, the higher the velocity.

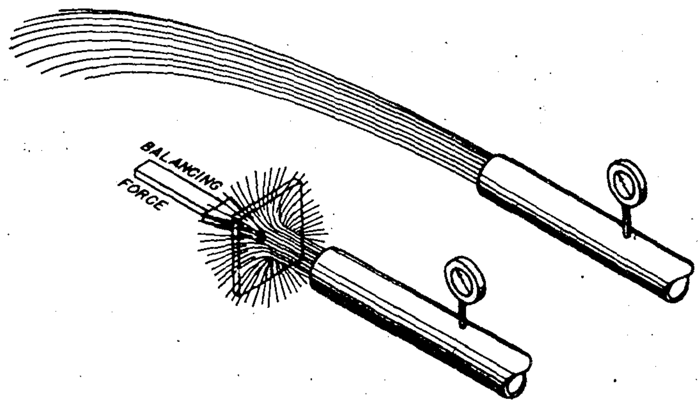


FIGURE 1. Energy in the water makes it shoot out of the hose nozzle; the higher the pressure, the farther the water travels. The water stream exerts a definite pressure on a plate held in its path.

No doubt you have had experience with water hoses--figure 1, right. The more you open the supply valve, the higher the pressure before the nozzle and the farther the water shoots into the air (lower pressure region). You may have experimented by holding a flat plate in the water stream, (left) and found that it needed a balancing force to hold it in place against the force exerted by splashing water. If the plate was released, the water's velocity would push it some distance from the nozzle. A stream of compressed air or steam would act exactly the same way.

To use these elementary actions we could construct a crude steam turbine as shown in figure 2. We could fix a nozzle in place, which gets its steam from a boiler. The nozzle would be aimed so that the high-velocity steam jet hits blades mounted on the edge of a wheel. The force exerted by the jet on the blades would turn the wheel on its shaft. Through a coupling the shaft can then turn some machine and do work. This arrangement transforms energy in the steam to mechanical shaft energy. This turbine is obviously very inefficient. The steam splashes off the blades with considerable velocity and wastes a lot of kinetic energy. To see what kind of refinements we could make, we will have to study nozzle and blade actions a little more closely.

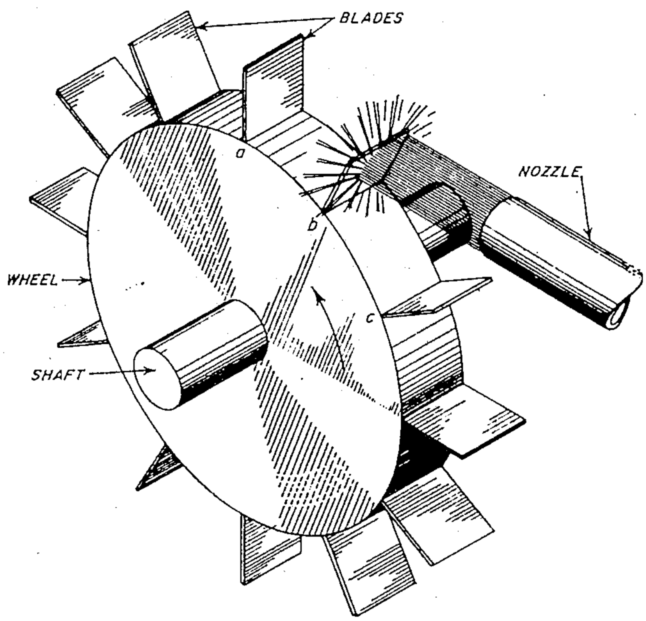


FIGURE 2. Crude steam turbine has flat blades on wheel perimeter. Steam flowing from nozzle exerts force on blades, making wheel turn.

Nozzles

For steam 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. Figure 3 shows two types of nozzles (or holes) that form effective jets. 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.

The pounds/sec. flow also depends on the cross-sectional area of the nozzle. The larger the area, the greater the weight of flow. The two factors are directly related; doubling area doubles flow, tripling area triples flow, etc. More will be said about nozzles in a later lesson, but this will do for now.

Blades or Buckets

When we investigated the rig in figure 2, we found that the steam jet splashed wastefully off the moving blades. To improve this setup, let us experiment with various blade or bucket shapes,

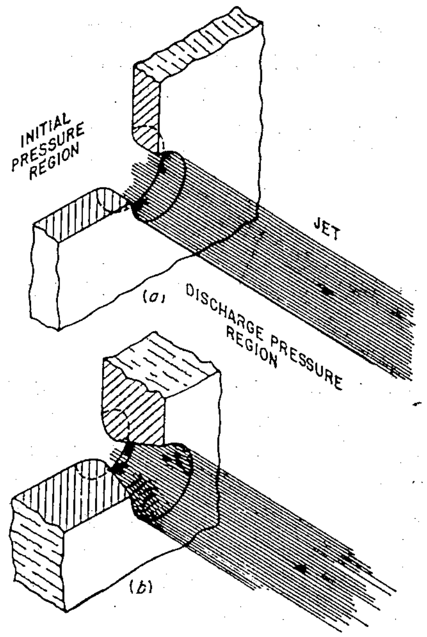


FIGURE 3

Figure 3 - Cross section of nozzle needed depends on pressure drop of steam passing through. a) Parallel wall nozzle used when exhaust pressure is more than half the initial pressure.
b) Diverging nozzle used when exhaust pressure is less than half the initial pressure.

as in figure 4. Here we set up various shapes of blades on some sort of weigh scale that measures the force developed when the blade turns the steam jet out of its original path of travel. We keep the speed of the jet coming out of the nozzle constant. The blades are stationary.

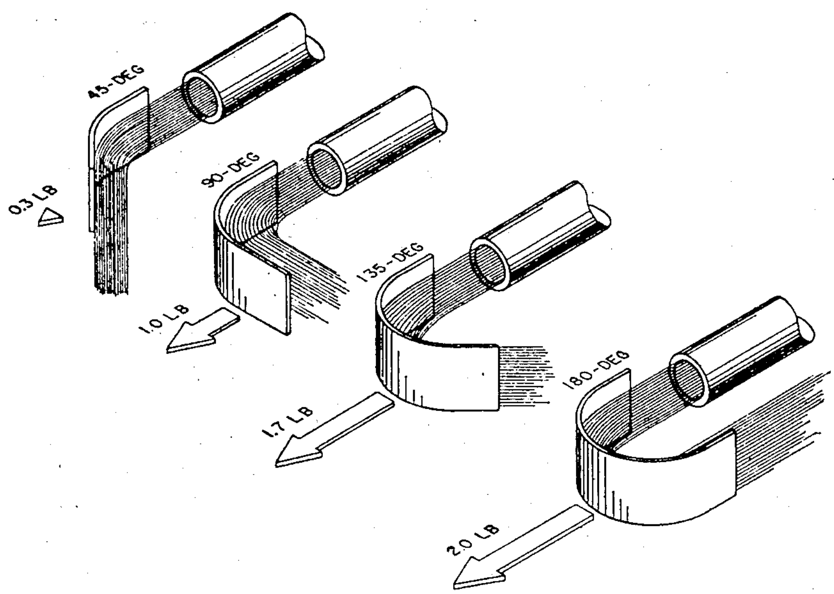


FIGURE 4. Force developed on a blade changing the direction of a steam jet, increases with the amount of angular change, steam jet flow staying constant and blade remaining stationary.

For the blade turning the jet through an angle of 45° we find the scale shows a force of 0.3 lbs. developed acting in the original direction of the jet. When the blade is bent to turn the jet by 90° the force jumps up to 1 lb. This condition is equivalent to that of the flat blades in figure 2. Our curved blade, though, changes the direction of the jet much more smoothly and with no splashing.

Now let us bend the blade so that the jet turns through an angle of 135°. The force rises to 1.7 lb. We get the maximum effect of 2.0 lb. by bending the blade so that the jet turns back in the direction it came from; that is the jet turns through an angle of 180°. The force developed is just double that of turning the jet through 90°.

The analogy we have used is for blades standing still. The steam jet leaves the blade with the same speed it enters assuming no friction between jet and blade surface.

In studies of objects in motion we find that any object that has mass and speed has energy because of this motion. This is called <u>kinetic energy</u>. The steam has kinetic energy because it travels at a certain speed. If it leaves the blade at the same speed as it enters, then it hasn't given up any of its kinetic energy. To transfer its kinetic energy to the blade, it must keep the blade moving. The blade should do this so that the jet leaves the blade at zero ground speed. At zero speed the steam has given up all its kinetic energy.

To realize this, all we need to do is let the 180° blade in figure 4 move in the direction of the steam jet. What happens as blade speed varies and jet speed out of the nozzle stays constant? For zero blade speed we already have the answer from figure 4. As the blade starts to move we will find the force starts to drop below the 2.0 lb. level. But we will also find that the speed of the jet leaving the blade (referred to the earth or stationary nozzle) has dropped below that at which it enters the blade. This means the jet has given up some of its kinetic energy. The blade has acquired this energy by being moved by the jet passing over its surface.

As we let the blade speed up, we will find the jet velocity leaving the blade drops more and more, indicating transfer of kinetic energy to the blade. At a certain blade velocity we will find the jet leaves with zero speed. The effect is that the blade just appears to drop the steam as it travels along, very much as a sky-writing airplane leaves its smoke in the air. Figure 5 shows an analogy demonstrating the velocity relations quite clearly. Zero steam-jet speed means that the jet has given up all its kinetic energy to the blade. This is the condition we need for maximum energy transformation. We will find the force developed on the moving blade is now only 1.0 lb., just half of the force at zero blade speed. <u>At this condition the blade speed is just half of the initial jet velocity</u>.

If we let the blade speed up some more the steam will now leave the jet with some velocity in the direction of blade movement. Thus, the jet again has not given up all its kinetic energy.

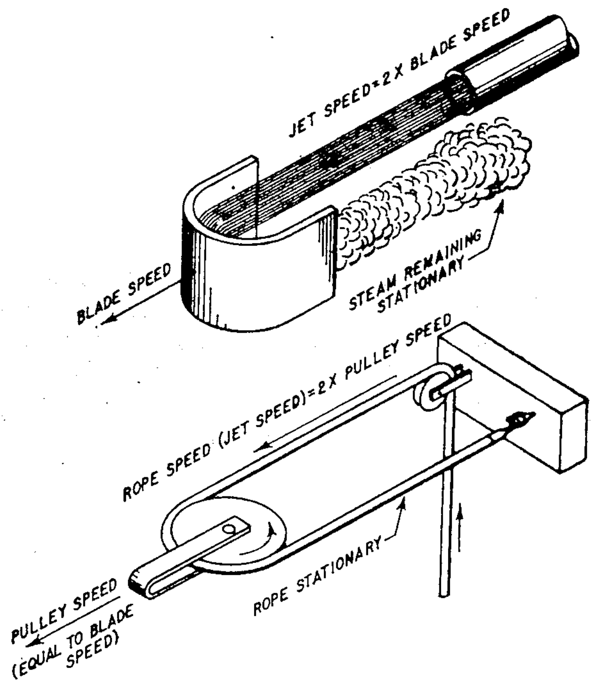


Figure 5 - Steam jet entering blade at a velocity just twice blade speed will leave blade with zero velocity. Rope and pulley analogy shows relation.

FIGURE 5

The exit velocity keeps increasing in this direction until blade speed equals jet speed. Then the jet cannot reach the blade and, of course, the developed force drops to zero.

Blades on a Turbine Wheel

While we would like to use the 180° blade in the turbine, no practical method has been found to mount them on a wheel. The nearest approach we can make to the ideal is an arrangement as shown in Figure 6. Here blades with about a 150° turn are mounted on the perimeter of a wheel carried by a shaft. The nozzle supplying the steam is mounted to one side of the blades, so that the steam jet can enter spaces between the blades.

Notice that this nozzle is rectangular in cross-section as compared to the nozzles shown in figure 3 which are circular. In a large steam turbine the nozzles are rectangular shaped in cross-section, but there would be a great number of them spaced around the wheel at the entrance of the turbine. However, for now we are considering one nozzle discharging against one wheel.

Figure 4 shows how the steam would flow if the wheel were locked so it could not rotate. For this condition the jet would develop maximum turning force, but the wheel would not be doing any work and the jet would not give up its kinetic energy.

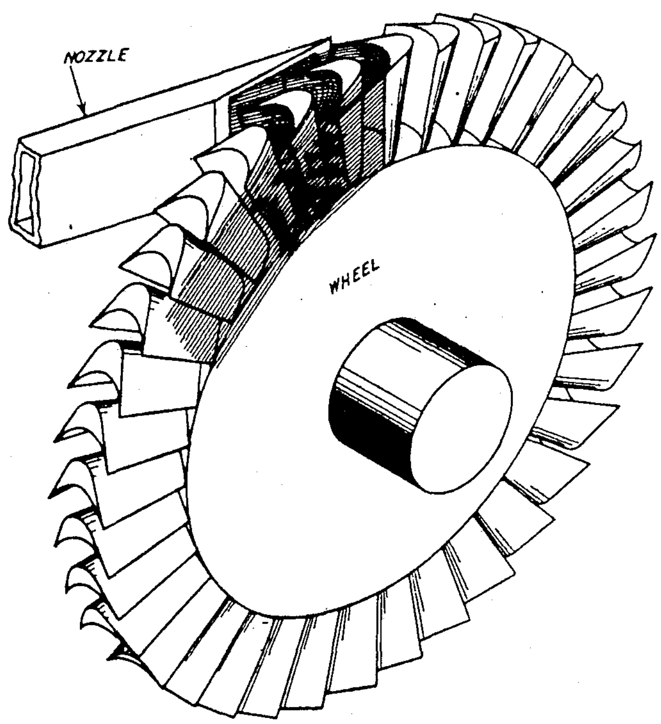


Figure 6 - When single-stage wheel is locked, the blades simply change direction of steam jet, but jet exerts maximum turning force on wheel.

FIGURE 6

When the blades move at about half the steam-jet speed, the steam jet follows a path that leaves the blades at right angles as shown in figure 7. This is usually the most efficient condition.

Figure 7 shows the internal working elements for a small single-stage turbine. There is an upper limit to the speed at which the turbine wheel with its blades can be made to rotate. Above a certain rpm., the wheel would be torn apart by centrifugal force. On the other hand it is known that we can make more efficient use of steam energy by generating it at high pressure and temperature and exhausting it at low back pressure. This gives us a high jet velocity. Since blade speed should be about half of jet velocity, this means that for a single wheel, as in figure 7, the rpm would be very high.

To get high efficiency from high-pressure, high-temperature steam and stay within strength limits of the turbine wheel, we have to use pressure staging or pressure compounding. In effect this means taking the exhaust steam leaving the blades of the first wheel and passing it through a second set of nozzles. These nozzles drop the steam pressure to a lower level and direct the resulting steam jet into the blades on a second wheel. In other words, the total pressure drop across the whole turbine is divided into smaller pressure drops across individual nozzles. This reduces the steam speed leaving the nozzles, in turn reducing the needed blade speed and wheel rpm.

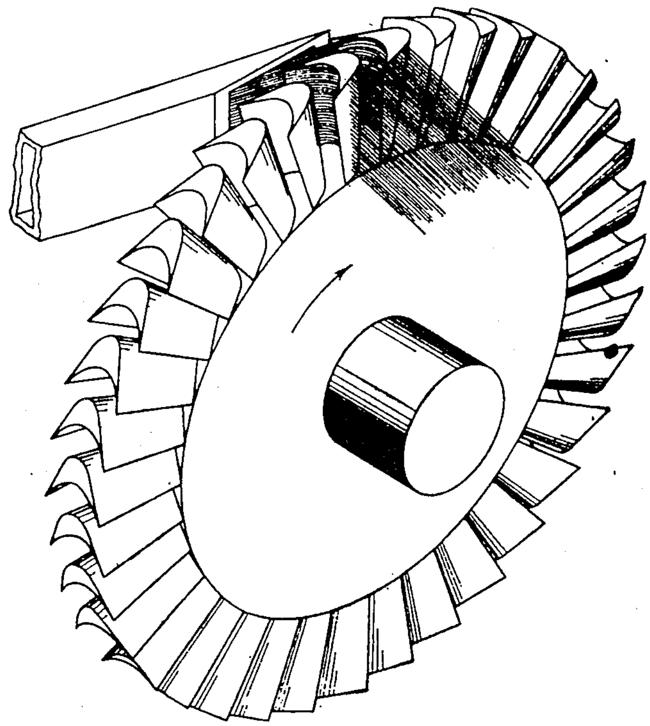


Figure 7 - When single-stage wheel turns at most efficient speed, the steam jet leaves blades in a direction parallel to shaft.

FIGURE 7

This type of a turbine is shown in figure 8. The steam flow is from left to right. It would be coming from the main steam line and we said previously that the maximum velocity of flow in the main steam line is around 11 ft./sec. Therefore, the steam enters the first stage nozzle block at relatively low velocity. As it passes through the first stage nozzle block the velocity of the steam is greatly increased due to the difference in pressure and now has high kinetic energy, which is given up to the first row of moving blades. At the exit of the moving blades the steam has a lower velocity. It then flows into the second set of nozzles or stationary blades where the flow path is redirected so that the steam will strike the second row of moving blades at the proper angle.

The steam loses some of its velocity as it passes through each row of moving blades and by doing so gives up its kinetic energy which is used to turn the wheel and hence the turbine shaft. Or in short, kinetic energy is changed to mechanical energy.

Figure 9 shows the flow path that steam takes as it passes through a turbine. Imagine that each row of nozzles and each row of blades is uncurled to rest on a flat plane. The diagram depicts a portion of what you would see if you were looking at the top of the flattened out wheels.

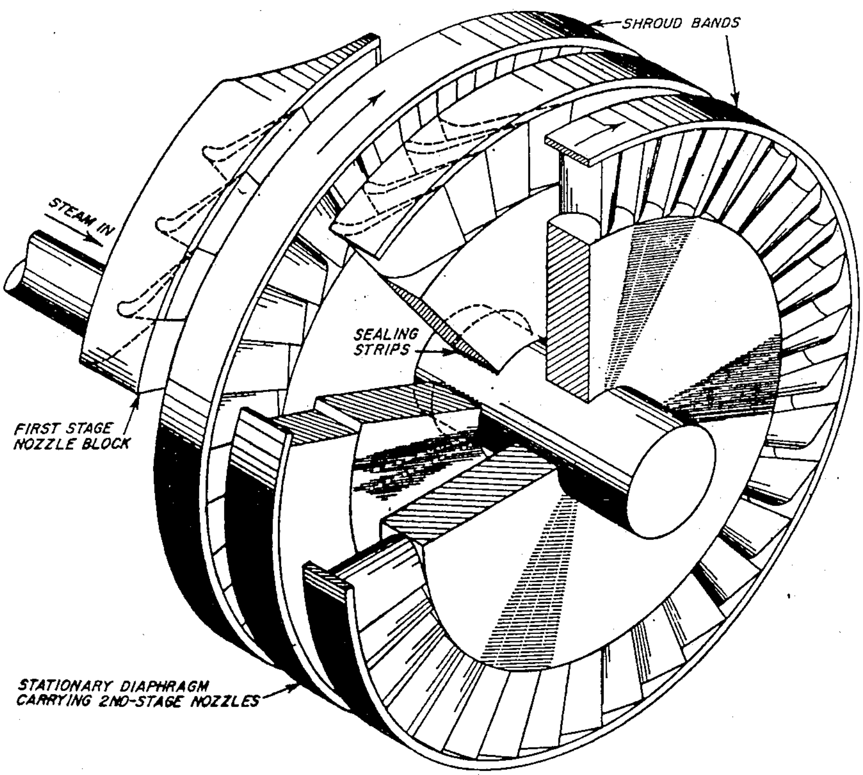


FIGURE 8

Figure 8 - Basic arrangement of a pressure-compounded turbine. Casing to which first stage nozzle block and second-stage nozzles are fixed is not shown.

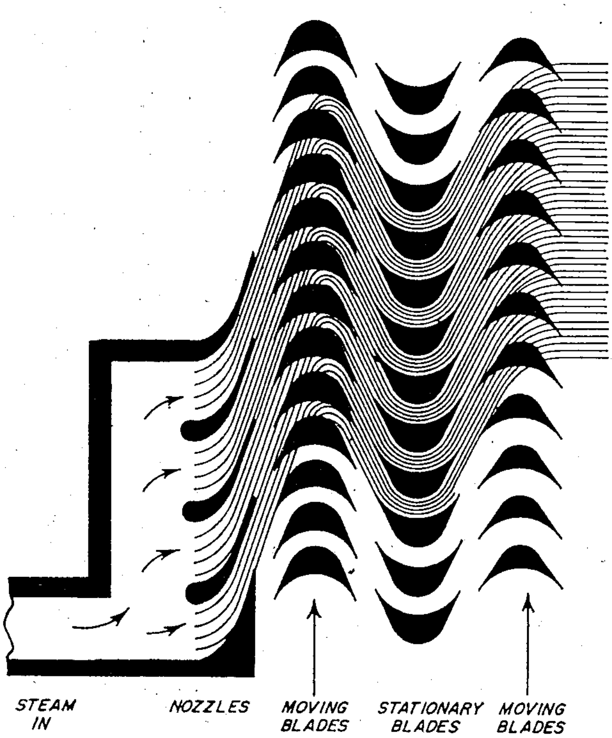


FIGURE 9

Figure 9 - Diagram showing the flow path of steam through a pressure compounded turbine.

One row of stationary or fixed blades and one row of moving blades is called a stage. This is derived from the fact that steam is expanded in stages in the turbine.

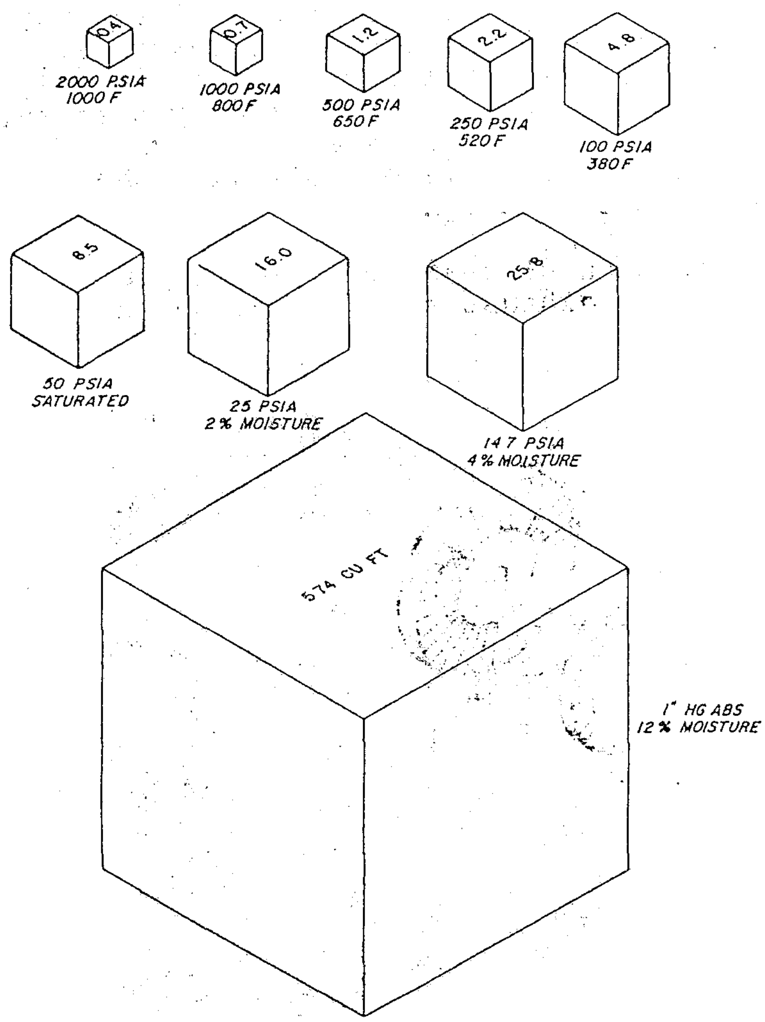


Figure 10 - Volume occupied by 1 lb. of steam initially at 2,000 psia and 1000°F increases enormously as it flows through a turbine to a low pressure exhaust. Turbine-nozzle and blade sizes must be proportioned accordingly.

FIGURE 10

Expansion of Steam

We have said previously that initially the steam entering a turbine must be at a very high temperature and pressure. As the steam expands in the turbine to a low pressure exhaust the steam increases enormously in volume. Figure 10 shows the increase in volume of 1 lb. of steam as it expands from a pressure of 2,000 psia (and 1000°F) to a pressure 1" Hg abs. (mercury absolute.) The increase in volume is:

$$\frac{574}{0.4} = 1430 \text{ times.}$$

This large increase in volume of steam has to be allowed for when designing the nozzles and blades. Succeeding stages of blades will have to be longer and longer in order to provide a bigger and bigger flow path area. Figure 11 shows several selected stages of diaphragm from a steam turbine. In cases where steam flow in lb./hr. is not very high the first stage might not have nozzles all the way around. So in the early stages volume flow increase is taken care of by increasing the number of nozzle openings per stage as in (a), (b), and (c). Once the entire circumference is occupied by nozzles, the only way to increase the opening is to increase the height of the nozzles as in (d) and (e). The moving blades following the nozzles are correspondingly increased in height.

The above discussion on expansion of steam explains why a turbine is always larger towards the exhaust as compared to the inlet.

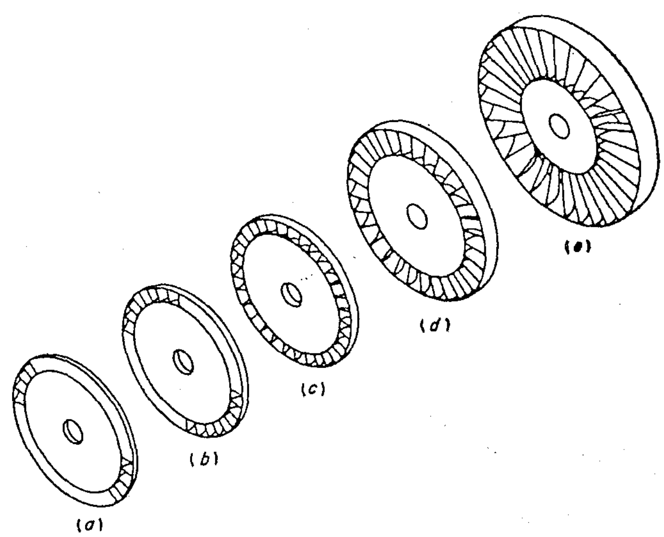


FIGURE 11. Increased area needed by steam flowing through turbine to low back pressures, shown by the number and height of nozzles in succeeding diaphragms.

Figure 12 (next page) shows a sectional view of how the working components appear in a small regenerative condensing steam turbine.

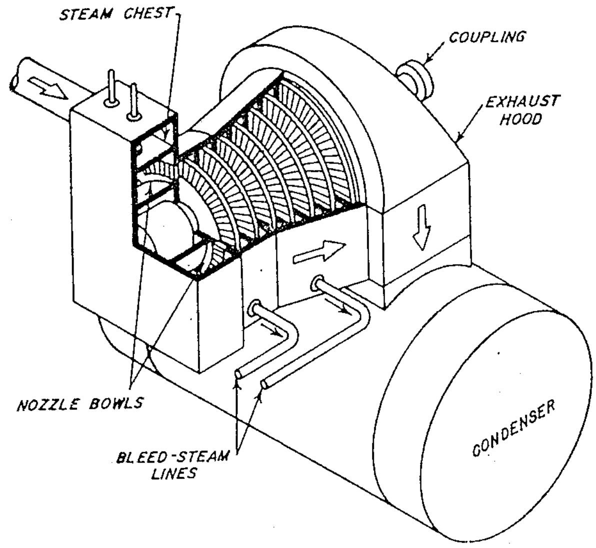


FIGURE 12. Sectional view of regenerative-condensing steam turbine with steam extracted from intermediate stages for feedwater heating.

In this lesson we have tried to show step by step how a steam turbine uses energy in high-pressure high-temperature steam and converts it to mechanical shaft work. The description has been fairly elementary, and dealt mainly with small turbines. However, this lesson gives us a good basis from which to describe the steam turbine in greater detail.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-1 - How Steam Turbines Work

A - Assignment

1. In simple terms what is a nozzle? Why are nozzles necessary in a steam turbine?
2. What is the purpose of blades or buckets in a turbine wheel?
3. Why is it better to expand the steam in a series of stages in a turbine rather than to expand it all in one stage?
4. Why does the area of the flow path in a steam turbine have to be designed bigger and bigger as it approaches the exhaust end of the turbine?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine Generator and Auxiliaries

- 2 - Turbine Types and Shaft Arrangements

0.0 INTRODUCTION

Up to this point we have discussed only one turbine which is a single flow, single casing type as was shown in figure 12, of the previous lesson. In industry there are actually a great variety of turbine types and arrangements. This lesson will describe a number of types and arrangements which one might expect to find in an electric power station.

1.0 INFORMATION

Basic turbine types, as shown in figure 1 can be divided into two main classes: 1.) condensing units exhausting steam at less than atmospheric pressure. Exhaust steam flows directly into a condenser. 2.) Noncondensing backpressure units exhausting at higher than atmospheric pressure. The exhaust steam is used for some other process, and therefore is called process steam. For example in pulp and paper mills the exhaust steam could be used in the paper making process; a public utility could be using the process steam as a source of heat for a central heating system etc.

The two classes may be further sorted according to steam flow in the turbine:

1. Straight flow and double flow.
2. Reheat and double reheat.
3. Automatic extraction.
4. Non-automatic extraction.

In figure 1 the first two turbines shown are examples of a <u>straight flow</u> machine. Figure 1 (c) is an example of a double flow turbine; the steam flow divides into two separate paths and then flows throught a couple of stages before it exhausts to the condenser.

Fifteen basic turbine types: condensing and noncondensing

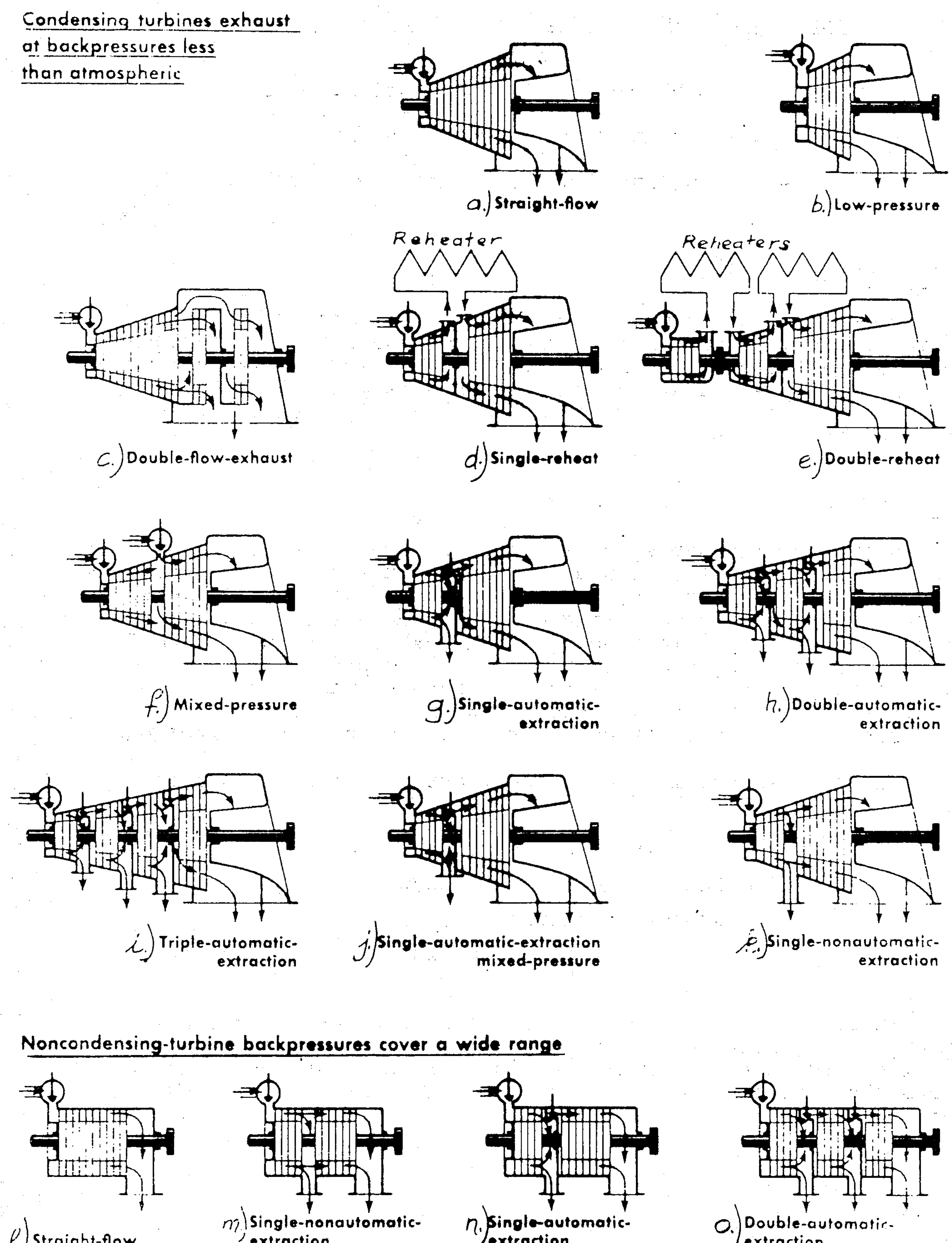


FIGURE 1

Figure 1 (d) is a single reheat turbine. That is the steam comes from the boiler, flows through several stages in the turbine and decreases considerably in pressure and flows back to the boiler to be reheated again. Then it expands further in succeeding stages till it enters the condenser. In figure 1 (e) is a double reheat turbine where the steam is reheated twice. Reheat turbine cycles are more efficient than straight flow turbines. Double reheat turbine cycles are still more efficient. However because of the extra equipment required in reheat turbine cycles it requires careful economic studies to determine whether a double reheat cycle is economically advisable. Some double reheat turbines are in operation but triple reheat are not economically feasible at present.

Figure 1 (f) depicts what is called a mixed pressure turbine. Stop valve pressure enters the front of the turbine via a governor valve. After several stages steam at stop valve pressure enters the turbine via an overload valve. It is called overload valve because it can produce roughly 115% to 120% of full load, depending on the design. It enters the turbine casing by means of an overload belt which is an annulus which girdles the turbine casing. Up to 100% load it is generally so arranged that the overload valve is closed. Thereafter it starts opening. Sometimes turbines are designed so that the overload belt starts opening at 75% or 80% of full load. In this case the turbine would be designed for most efficient operation at 80% full load. On overload the turbine doesn't operate at maximum efficiency but at maximum power output.

In automatic extraction a valve regulates automatically the amount of steam which flows through the exhaust and thereby automatically regulates the amount of steam which will flow through the extraction belt. These can be made as single, double, or triple automatic extraction turbines as shown in figure 1 (g) 1 (h) 1 (i) In a non-automatic extraction turbine as in figure 1 (k) there is no valve regulating the extraction flow.

Up to this point we have discussed condensing type turbines only --i.e. turbines which exhaust to a condenser. Non-condensing turbines are called back-pressure turbines because they exhaust at a relatively high pressure as compared to condensing turbines. These can be made as straight flow, single-nonautomatic extraction, single automatic extraction or double automatic extraction as shown om figure 1 (l) to figure 1 (o).

Shaft Arrangements

So far we have mentioned turbines with single casings only. (a turbine casing is often referred to as a cyclinder.) However with the modern trend towards larger and larger units at higher and higher pressures it was found that 30 or more stages were required to fully expand the steam and it was found that to put all these stages into one cylinder was impractical. The castings couldn't be made big enough. So to get around this problem several cylinders were built on one shaft. Then it was found that the turbine could now be built to a fairly large output, (200 MW or more) but that generators could not be built big enough because of limitations imposed on the generator rotor. The solution was to build two generators for one unit. However putting both generators together with several turbine cylinders on one shaft was impracticable because of its length so cylinders and generators were put on two separate shafts to make up one turbine generator.

A turbine with cylinders on two separate shafts is called a cross-compound turbine. A turbine with cylinders on one shaft only is called a tandem compound turbine because they are intandem. (Compound meaning more than 1 cylinder)

The piping leading from the discharge of one cylinder to the inlet of the next cylinder is referred to as cross-over piping no matter what the shaft arrangement may be.

The first cylinder that the steam enters in a turbine is called a high pressure (H.P.) cylinder because of the high steam pressure. The cylinders which exhaust to a condenser are referred to as low pressure (L.P.) cylinders. In reference to steam pressure cylinders between the H.P. and L.P. cylinders are referred to as intermediate pressure (I.P.) cylinders.

Up to around mid 1950's units above 100 MW were generally cross-compound turbines. Then advancements were made in generator cooling so that units with an output of 300 MW could be built as tandem compound units. Today tandem-compound units can be built with an output up to 500 MW. With outputs above this, cross-compound units have to be used. The largest cross-compound unit on order today anywhere in the world has a 1000 MW. capacity. To make this possible, great advancements have been made in generator cooling techniques, which will be covered later on.

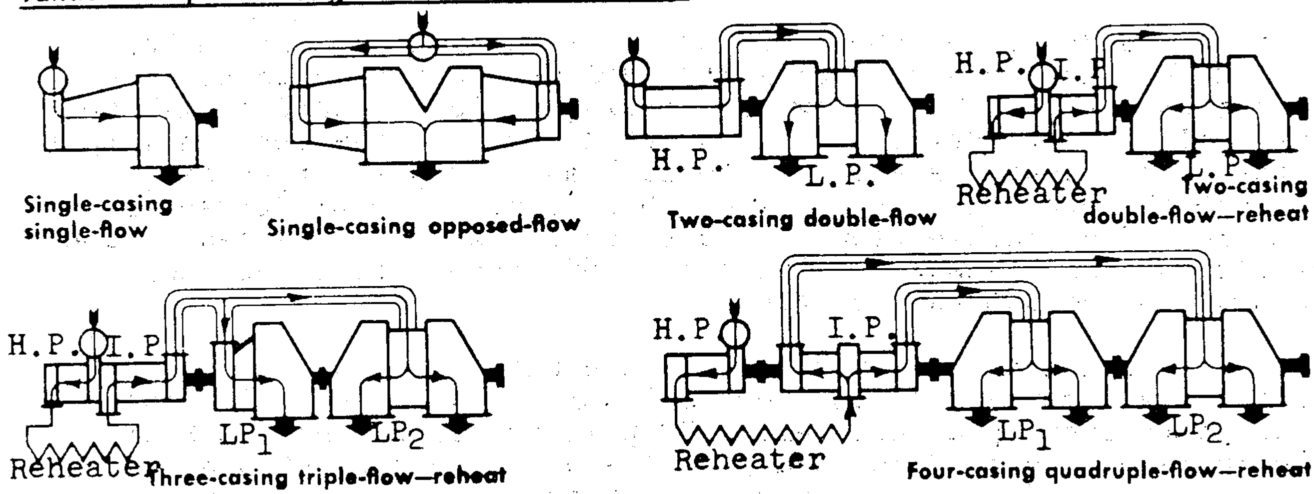
In the case of cross-compound units, it is often the case that one shaft will rotate at 3600 rpm, while the second shaft will rotate at 1800 rpm.

Figure 2 depicts several shaft arrangements for tandem compound and cross-compound turbines. The single-casing (or cylinder) single flow turbine we have covered previously. The opposed flow is used to neutralize thrust on the shaft. This type of turbine can handle twice the steam flow of a single flow turbine and therefore has twice the output.

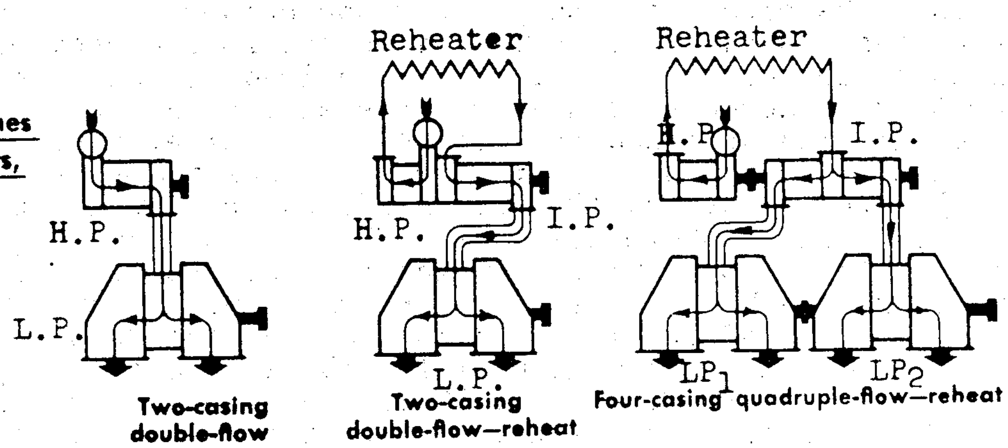
You will notice that the two-casing double flow turbine has the L.P. casing with steam flow in opposite directions. This again

Casing and shaft arrangements depend on capacity, steam conditions

1. **Tandem-compound designs work on a single shaft**



2. **Cross-compound turbines usually have two shafts, sometimes three**



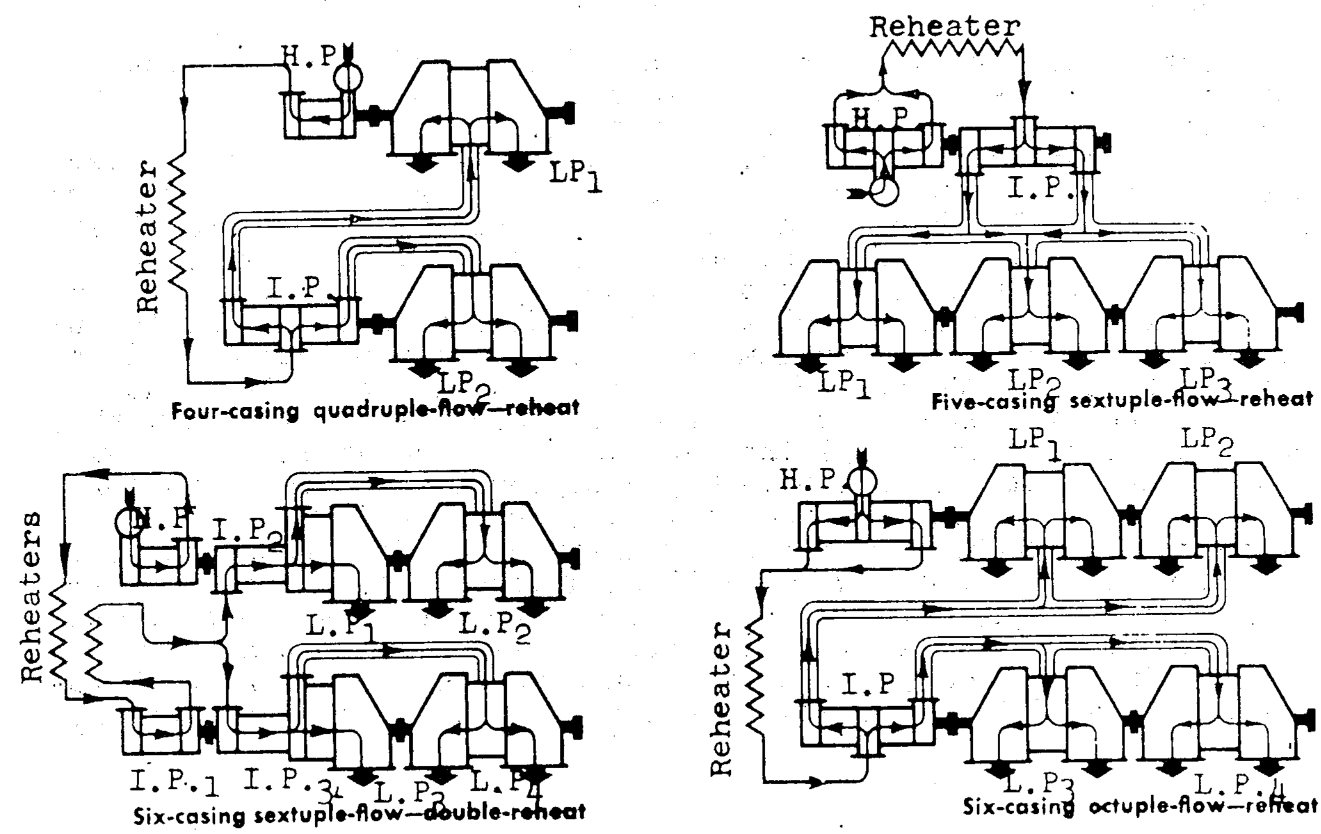


Figure 2

neutralizes thrust on the L.P. shaft. Also after the steam has passed through the H.P. casing it's volume has increased to such an extent that a single flow L.P. casing could not handle all the steam flow. For larger units with even higher pressures the larger quantity of steam has expanded to such an extent after passing through the H.P. and I.P. casings that it requires two three or sometimes four double flow L.P. casings to handle the great volume of flow at low pressures. These type of arrangements are shown towards the bottom part of figure 2.

You will notice that reheaters are used in conjunction with these arrangements. A reheater is generally used where initial stop valve steam pressures are 1600 psi. or more. Ontario Hydro has in recent years purchased 300 MW and 500 MW conventional steam turbines using 2400 psig. stop valve pressure. Of course the higher the pressure, the greater the number of stages required for the steam to expand to almost absolute zero pressure. Hence the greater the number of cylinders required per turbine. The shaft arrangment depends to a very large extent on the turbine output capacity required and the pressure and temperature of steam supplied at the turbine stop valves. The max.temperature is fairly well fixed at around 1000 to 1100°F in present day turbines because metals we have nowadays will not withstand higher temperatures for the pressures at which they are required to operate. However, the higher the steam pressure and temperature the lower the quantity of steam flow required and vice versa. Take for example a 200 MW turbine with inlet conditions of 565 psig and 482°F requires a steam flow of 2,500,000 lb/hr. Compare this with a 200 MW turbine with steam inlet conditions of 2350 psig and 1050°F. It requires a steam flow of 1,270,000 lb/hr.

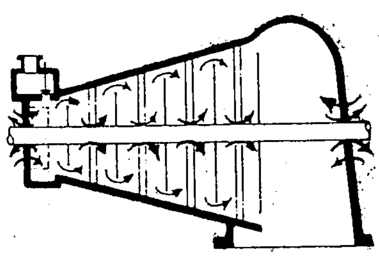
Both these turbines have the same output capacity and the number of cylinders required for each are the same. However, the size of cylinders for the turbine with lower steam conditions would be considerably greater. The volume of steam for the lower pressure turbine would be much greater and therefore would require a much larger flow area. This means that the blades for the last several stages in the turbine would have to be much longer. We mentioned previously that if the blade tip speed exceeded a certain value that the turbine wheel would disintegrate due to centrifugal force. Therefore, the maximum allowable blade tip speed is in the neighborhood of 1800 ft/sec. For a 3600 rpm shaft, this means that the turbine wheel must not exceed a diameter of 10 ft. with a maximum blade length of 31 inches. If this size of wheel cannot handle the steam flow which is the case for the turbine with low steam inlet conditions mentioned above, then the only alternative is to go to a machine with a lower shaft rpm. For reasons associated with the generator which will be discussed later on in order to produce 60 cycle, a-c, the next lower rpm that can be selected is 1800 rpm. Hence large units with low inlet steam conditions, such as in a nuclear power station must rotate at 1800 rpm. Turbine blades will be discussed in greater detail in a later lesson.

Seals and Glands

Through the clearance of rotating and stationary parts of a turbine, steam leakage may occur and, under certain conditions, air infiltration also. To reduce such leakage and infiltration to a minimum, various forms of seals or glands are installed. The diagram in figure 3 shows the location of leakage and infiltration points. At the high pressure end, steam tends to leak out around the shaft. During starting, however, pressure in the first stage of condensing turbines may be low enough to cause an inflow of air. At the low pressure end of a condensing turbine normal tendency is toward air leakage into the machine because the pressure in the exhaust end of the L.P. cylinder is below atmospheric. In non-condensing turbines the tendency is toward steam leakage outward.

Steam leakage between stages may occur where the shaft passes through the diaphragms; leakage may also occur through the clearance between the turbine wheel and casing.

POINTS IN A TURBINE WHERE LEAKAGE OCCURS



Where shaft passes through the casing, at high- and low-pressure ends, air may leak in or steam may leak out. Steam may also leak between stages at diaphragm openings.

Fig. 3

Figure 4 shows the glands placed in such a way as to stop leakage past the turbine wheels at the casing. The glands consist of thin circular metallic strips or serrations. In figure 4 the glands are shown as attached to the shaft and casing. Often they are attached directly to the shrouding of the turbine wheels and the diaphragms.

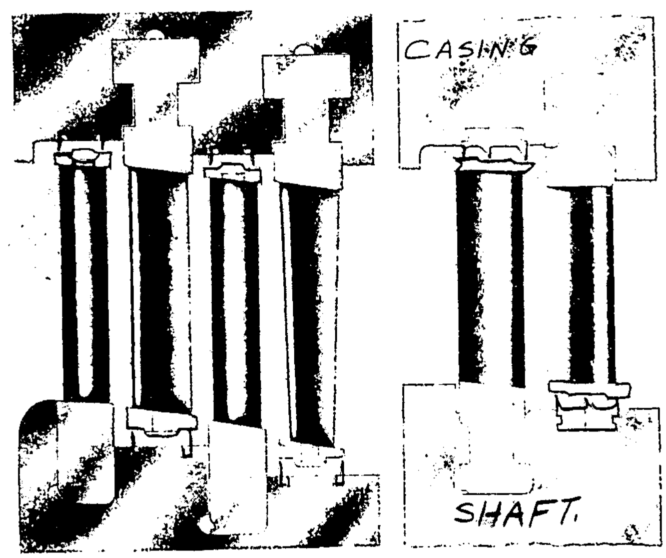


Fig. 4 Diagram showing glands between diaphragms and shaft and turbine wheels and casing.

Steam leakage from the turbine reduces power output and increases steam consumption per unit of output. Infiltration of air throws extra load on the condenser air removal equipment and tends to raise backpressure which results in lower efficiency.

For efficient operation, both forms of leakage must be kept to an absolute minimum. There are four types of seals or glands used in turbine practice:

1. stuffing boxes
2. carbon-ring packings
3. labyrinth seals
4. water glands

A stuffing box is fitted with soft metallic packing rings. This arrangement is used on some small machines for shaft sealing at the low-pressure end. Carbon rings are used quite often on large turbines. However labyrinths and water glands or combinations of these are the most common ones used on large turbines.

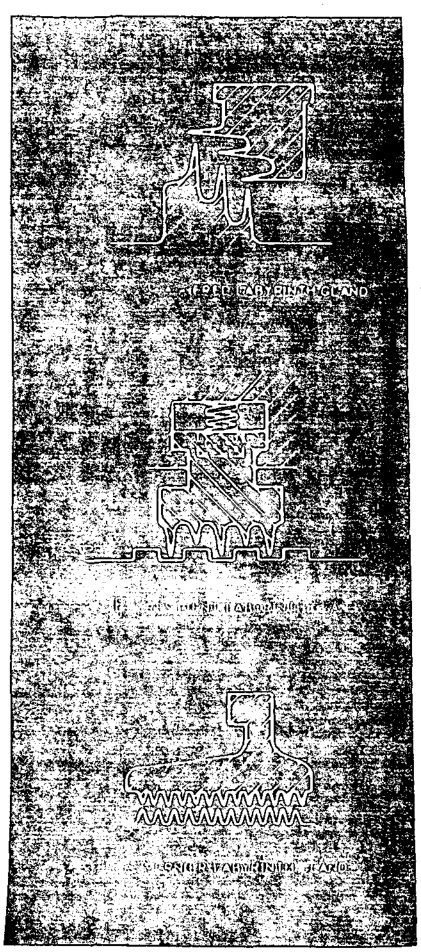
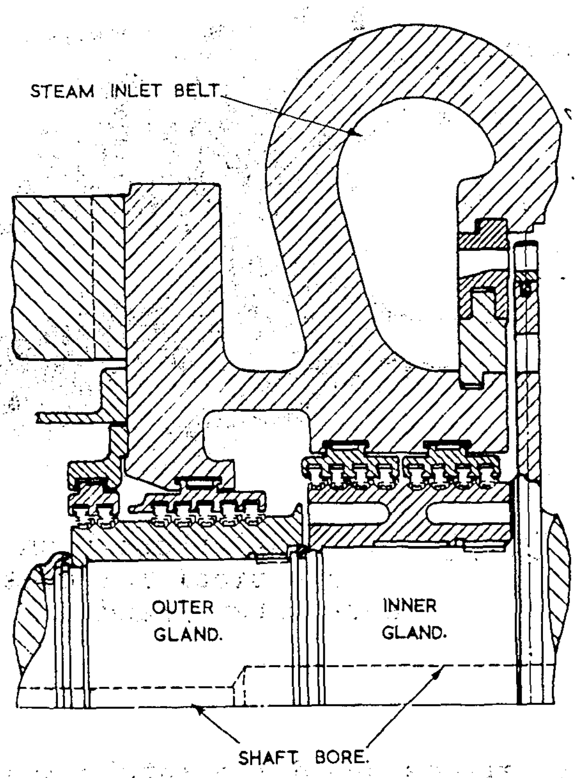


Fig. 4 shows three types of labyrinth gland designs: a) staggered, b) resilient and c) vernier. The lower part in each case is part of the shaft.

Labyrinth seals consist essentially of a number of thin circular strips or serrations fastened to the casing or a member supported from the casing and positioned so that the clearance between the shaft and edges of the strip is small. The resistance offered by this series of obstructions to steam flow is enough to hold leakage to a minimum. Labyrinths are sometimes used alone and frequently in combination with water glands. Figure 5 shows a sectional view of a typical high pressure labyrinth type gland. The cross hatched area depicts the turbine casing. At the right hand side of fig. 5 is the first turbine wheel.

Figure 5 Labyrinth gland designs.



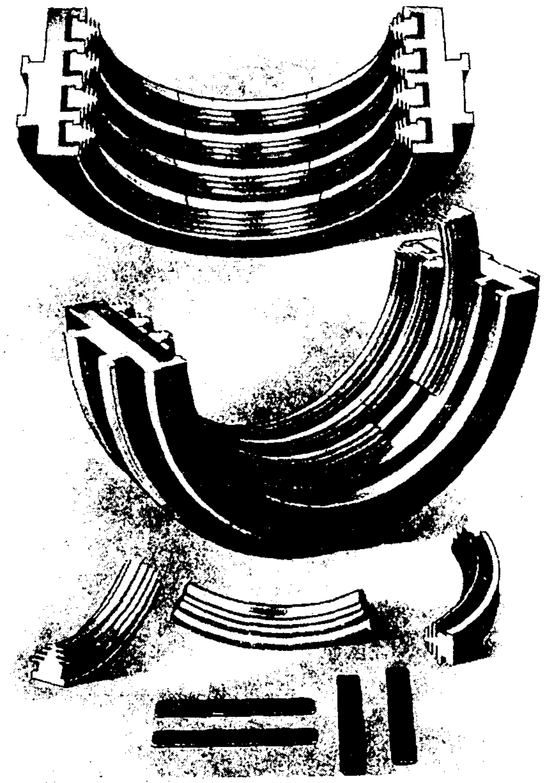


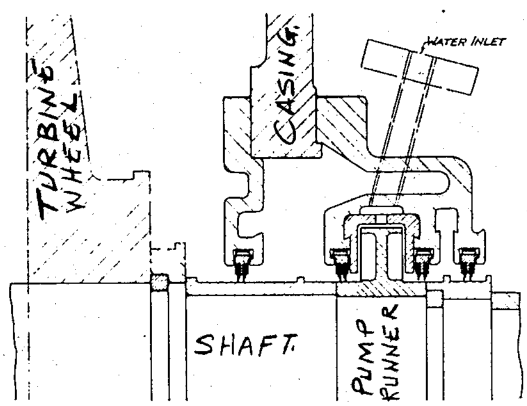
Fig. 6 Sectional arrangement of typical high pressure labyrinth gland.

Fig. 7 Composite parts of high pressure labyrinth turbine gland.

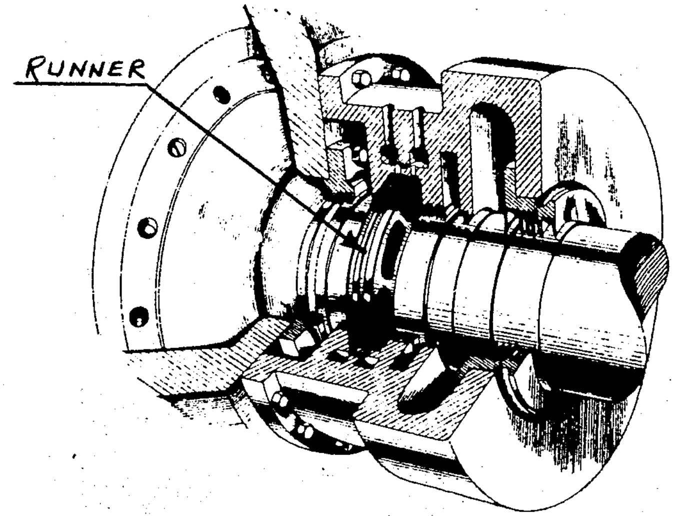
Figure 6 shows an enlarged view of two halves of a staggered type of labyrinth gland. These glands are fitted into the casing and held in place by leaf springs.

A water gland is merely a centrifugal pump runner rotating with the turbine shaft and confined in a housing attached to the casing. Commonly supplied with condensate, the runner holds the water in a ring at its periphery, forming a positive seal. Figures 8 and 9 depict a typical water gland.

In most cases, water glands are used in combination with some other form of seal, labyrinths usually, because the water gland is not effective until the turbine reaches operating speed. The additional seal also reduces the casing side steam pressure on the water gland when it is used at the high pressure end.



<u>Fig. 8</u> Section of typical low pressure water gland.



<u>Fig. 9</u> Sectional photograph of a typical low pressure water gland.

The circular metallic strips in glands are deliberately made thin so that in case they rub against the shaft they will quickly melt and avoid local hot spots on the shaft. A hot spot can weaken a shaft.

However, it is desirable that the glands remain in place, because as we said before in-leakage of air means reduced back pressure and thus reduced turbine efficiency. Since the clearances are very small it becomes apparent that the shaft and casing must expand at the same rate, otherwise these glands will become damaged.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles T.T.3

4 - Turbine Generator and Auxiliaries

-2 - Turbine Types and Shaft Arrangements

A - Assignment

1. What are the two main classes of steam turbines?
2. How does a reheat turbine differ from a straight flow turbine?
3. What is a cross-compound turbine? A tandem compound turbine?
4. Why is it that large turbines may require two, three, or even four double flow L.P. casings?
5. Name 4 places where leakage may occur in a turbine.
6. Name 4 different types of glands used for steam turbines.
7. Briefly describe labyrinth seals and draw one type.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-3 - Surface Condenser Details

0.0 INTRODUCTION

At the T.T.4 level we briefly dealt with a single pass surface condenser. This lesson will continue to describe in greater detail how the surface condenser is put together and will also describe the means used to extract air and gases from the condenser. This lesson assumes that the student has taken the course on Heat and Thermodynamics at the T.T.3 level.

1.0 INFORMATION

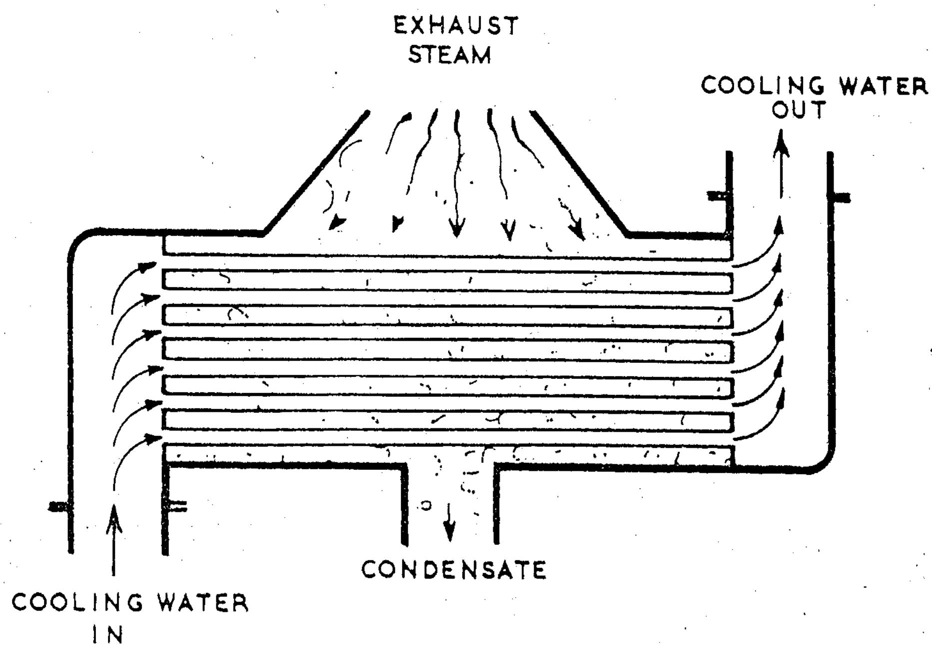


FIGURE 1. Single pass condenser.

Figure 1 shows the single pass type of surface condenser. That is the circulating water passes through the condenser once and then is discharged back to where it came from. As we said previously in this course this is the type of condenser to be used when there is a plentiful supply of cooling water. Compare this with the double pass condenser shown in figure 2. Here the inlet waterbox and outlet waterbox are at the same end of the condenser. Notice the dividing plate placed in the waterbox on the left hand end. The circulating water enters the waterbox from below, flows through the bottom bank of tubes to the other end of the condenser.

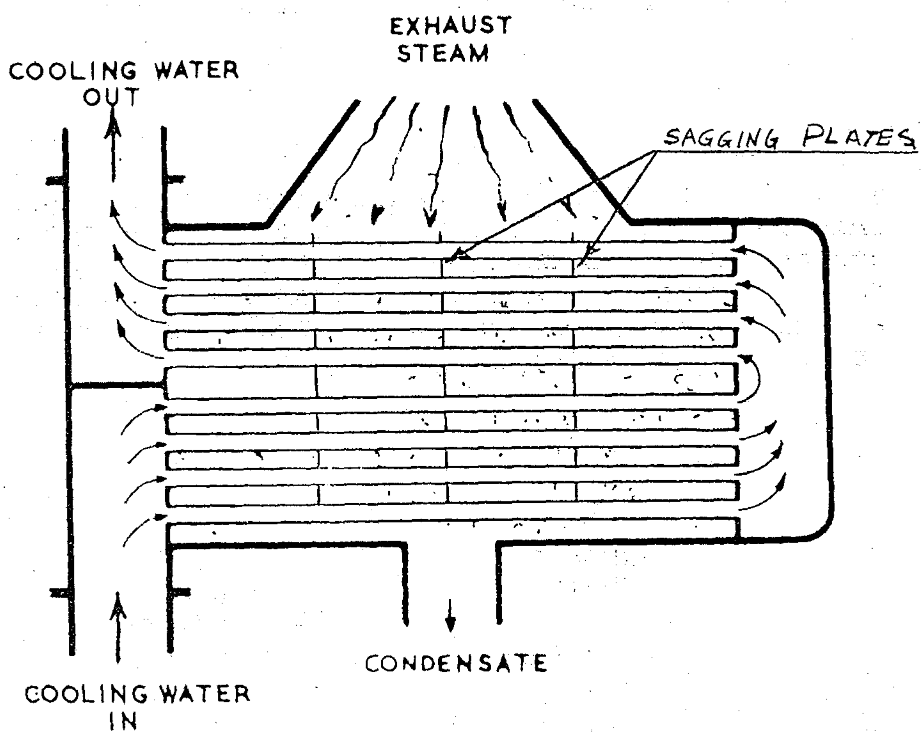


FIGURE 2. A double pass condenser.

At this point the water can enter the top bank of tubes and pass across the condenser once more. It then discharges through the top of the outlet waterbox. Sometimes triple pass condensers are used also.

The latter two types of condensers are generally used when there is a shortage of water supply which means that a high temperature rise (say 30 to 40°F) of the cooling water is allowed for. In a single pass condenser the temperature rise of cooling water from inlet to outlet is approximately 20°F.

The condensers we have been discussing so far are single shell type. Often they are designed with the circulating water side divided into two halves. Each side has an independant C.W. inlet and an independant outlet. Figure 3 shows a photograph of this type of condenser. The waterboxes have a door for each half which can be swung open when unbolted. This type of arrangement allows maintenance work to be done on one half of the condenser while the unit can operate on near half-load with the other half of the condenser in service. Notice that the condenser in figure 3 is also

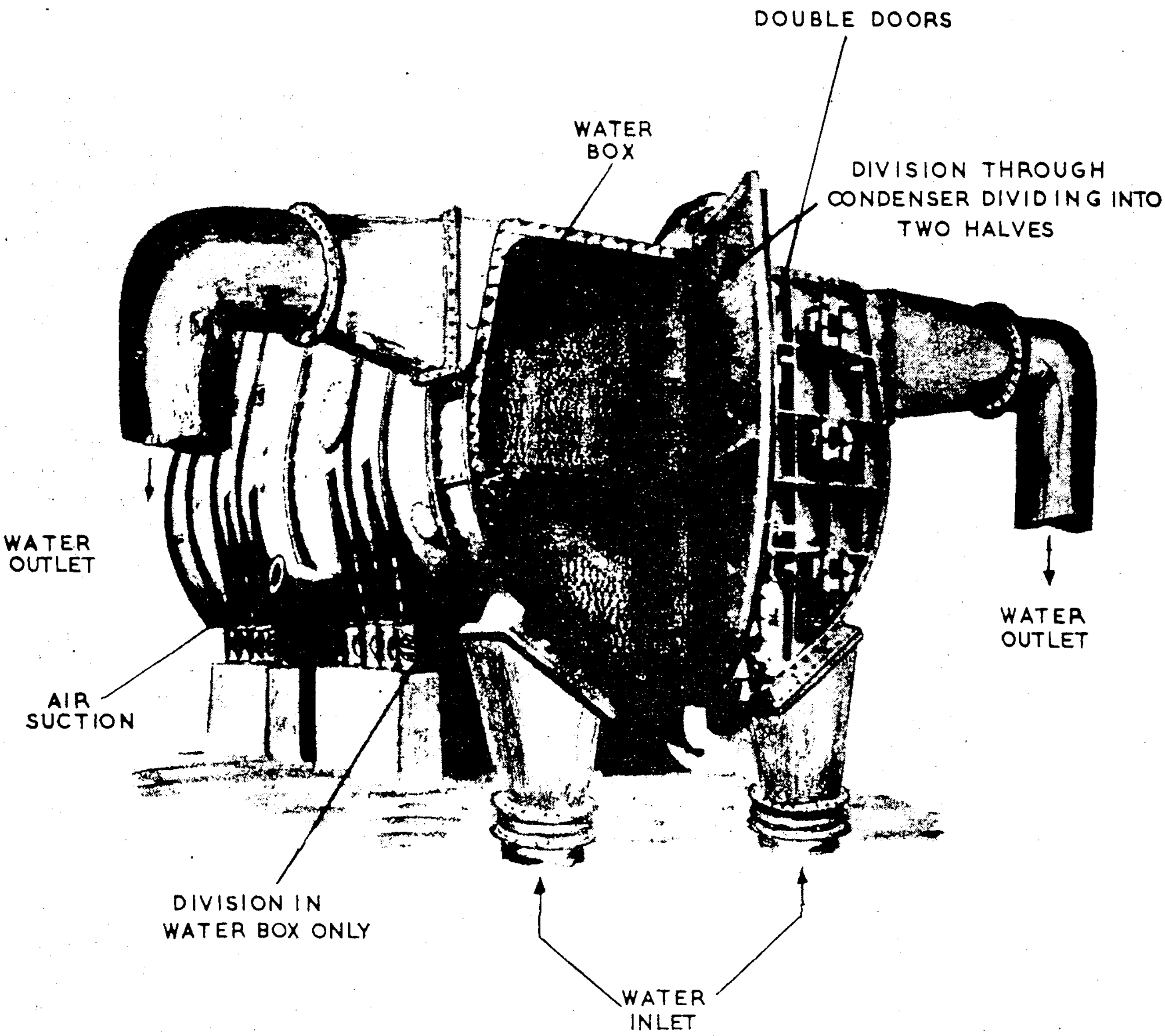


FIGURE 3. Single shell condenser with circulating water side in two halves; double pass condenser.

a double pass condenser because the C.W. outlet is at the same end as the inlet.

Large units sometimes have two shells, such as in figure 3 standing underneath the turbine side by side in which case they are called twin shell condensers. Triple or quadruple shells can also be used. This provides one shell for each L.P. exhaust.

Backwashing

From experience it is found that various objects such as leaves, fish, algae, weeds, mussels or clams etc. can be washed into the waterboxes by the circulating water. Since the size of condenser tubes is from 1/2" to 1" in diameter, they tend to become clogged. Often this material can be flushed out of the condenser by letting the water flow in the reverse direction from normal. This is called backwashing. It can be done when the original design provides baffles appropriately placed in the waterboxes. If this fails to solve the problem then the tubes have to be cleaned by other means.

Tubeplates

Condenser tubes have to be held in position and a means has to be provided for sealing to prevent circulating water from mixing with the steam and condensate. This function is performed by what are called tube plates. There is one tube plate at each end of the condenser where the condenser tubes terminate. Since condensers are normally 20 to 30 ft. long, the tubes also need plates in between the tube plates to prevent the tubes from sagging. These plates are called sagging plates. They have to be spaced in such a way that the frequency of vibration of the tubes does not amplify the frequency of vibration of the turbine blades, otherwise there could be damage to the condenser and turbine due to vibration over a long period of time.

The tubeplates are generally made of non-ferrous metals such as Admiralty Brass (alloy of copper and zinc.)

Since the inside of the condenser is at near perfect vacuum there is a great force due to atmospheric pressure trying to crush the condenser shell. For this reason the shell has to be strongly reinforced from the inside. This is doubly true for the tubeplates because the C.W. water pressure is acting on them as well. Therefore the tubeplates have to be firmly held in position by staybolts running from one end of the condenser to the other. They prevent any end stress from being applied on the tubes.

Leakages

There are two leakage problems associated with condensers--leakage of air and leakage of circulating water into the steam space.

We have mentioned air leakage previously but will summarize its undesirable effects here. Practically all air entering the condenser does so through leakages into the turbine spaces which are under vacuum and can have one or more of the following ill effects on operation:

1. Increase in turbine back pressure. This raises the saturation temperature which means more heat is going to waste.

2. Air blankets the outside of the tubes and prevents the transfer of heat to the cooling water. This reduces the effectiveness of the condenser.

In the course on Heat and Thermodynamics we said that the lower the pressure, the lower the temperature at which steam will condense. The lower the temperature at which steam will condense, the more heat energy contained in the steam can be converted to useful mechanical work. Hence, the lower the back pressure the greater the amount of useful work that can be obtained from the steam and the greater the efficiency of the turbine will be. Since air leakage into the condenser raises the back pressure of the turbine it produces effects opposite to the ones desired and therefore must be avoided.

Tube fixing

The possibility of water leakage is dependent to a large extent on the type of tube fixing to the tubeplates and therefore the relative merits of the various types are reviewed below. The action of steam can cause tubes to rotate and the method of fixing must prevent this rotation, in addition to holding the tubes tight.

Some types of tube fixing employ some form of packing and these can be divided into two classes: metallic and non-metallic. The non-metallic packing, is, however, subject to serious damage when acid washing is used to remove scale from the internal surface of the tubes. If packings are used on modern plant then metallic packings are favoured in preference to non-metallic packings for this reason.

Figure 4 shows the inlet and outlet tubeplates with various types of tube fixings. Figure 4 (a) shows a tube secured by ferrules at both ends. Experience has shown that this type of fixing prevents leakage but does not prevent tube rotation and tubes have failed because of wear against the support plates.

This type of fixing was therefore superseded by figure 4 (b) where the ferrules are bell-mouthed and tapered. As the ferrules are screwed inwards they press against the tubes and hold them tight to prevent tube rotation. However, these type of ferruled fittings are complicated and because they are not flush with the tubeplates, cause turbulence and thus resistance to flow.

The inlet and outlet fixings of figure 4 (c) which consist of a rolled-in and bellmouthed inlet and a full box packing at the outlet and is a better arrangement. It is simple, gives reasonably good flow and has proved very reliable in preventing leakage.

The tube can move relative to the tubeplate on the righthand side and thereby avoids stress on the tubes when they expand relative to each other.

In recent years tubes have been rolled in at both ends as in figure 4 (d) with successful service. This arrangement demands that the relative expansion of tubes be allowed for and this is usually done by having an expansion piece fabricated into the condensed shell.

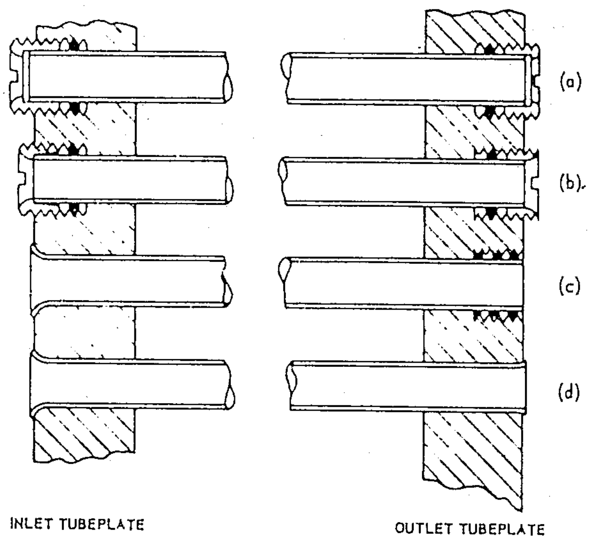


FIGURE 4
Condenser tube fixings

Tube material

Tube materials fall roughly into three categories:

1. Brasses	2. Cupro-nickels	3. Bronzes
Commercial brass - 70% copper; 0.4% arsenic; Remainder zinc.	Cupro-Nickel - 30% nickel; 2% iron; 2% manganese; Remainder copper.	Tin Bronze - 12% tin; Remainder copper.
Admiraltry Mixture - 70% copper; 0.4% arsenic; 1.25% tin; Remainder zinc.	Cupro-Nickel - 30% nickel; 1% iron; 1% manganese; Remainder copper.	Aluminum Bronze - 7% aluminum; Remainder copper.
Aluminum Brass - 76% copper; 0.4% arsenic; 2% aluminum; Remainder zinc.	Cupro-Nickel - 20% nickel; 1% iron; 1% manganese; Remainder copper.	

1. Brasses	2. Cupro-nickels	3. Bronzes
	Cupro-Nickel - 10% nickel; 1% iron; 1% manganese; Remainder copper. Cupro-Nickel - 5% nickel; 1% iron; 1% manganese; Remainder copper.	

The first two brasses are not very resistant to corrossive or erosive action and their rise is confined to stations which draw water from fresh water streams or operate with cooling towers. Their heat transfer properties are good. The aluminum brass has very good corrossive resistance and can therefore be used where salt water is used as cooling water if it doesn't contain too much suspended matter.

In general the cupro-nickel tubes have high resistance to corrossion and erosion but they are expensive and have a higher resistance to heat transfer due to the nickel content. This means a a large cooling surface is required and a reduction in condenser performance can mean, under certain cooling water conditions, heavier scaling. These type of materials are therefore used only for extreme water conditions such as sea water.

The bronzes are not very common. The first one is expensive due to high cost of tin. It has however high resistance to sand erosion. The second one has good resistance to corrossion but has been replaced by aluminum brass.

Volume of Cooling Water

When steam leaves the turbine it still contains its latent heat of vaporization which amounts to about 970 Btu/lb. of steam. Before this lb. of steam will completely condense, this amount of heat has to be removed. We said previously that the cooling water temperature rise was about 20°F for a single pass condenser. Since it requires 1 Btu to raise 1 lb. of water 1°F, it means that to raise 1 lb. of water 20°F takes only 20 Btu. Therefore to condense 1 lb. of steam we require 970/2- or approximately 50 lbs. of cooling water.

For a 220 MW unit, about 2,000,000 lb. steam/hr. flow into the condenser. This means we have to have a cooling water flow of 2,000,000 x 50 = 100,000,000 lb./hr. or roughly 10 million gallons per hour. One can see that an enormous amount of cooling water is required for a steam power station.

The condenser tubes are generally around 3/4" in diameter and the maximum velocity of flow is limited to around 7 ft./second.

Varying C.W. Temperatures

The variations of C.W. inlet temperature from summer to winter where sea or river water is used may be as much as 30°F and such climatic variations have a big effect on back pressure. This is illustrated by the following example in which the rise in temperature of the cooling water through the condenser is assumed constant:

	Summer	Winter
C.W. inlet temp.	65°F	35°F
Rise through condenser	20°F	20°F
C.W. outlet temp.	85°F	55°F
Terminal difference (Exhaust steam--C.W. outlet)	10°F	10°F
Saturation steam temp.	95°F	65°F
Back pressure	1.66 inches Hg.	0.62 inches Hg.

Operating conditions are thus more favorable in winter than in summer. It is found that less cooling water is required in winter than in summer.

Undercooling

From the above example for winter operation the saturation temperature of steam was 65°F. This temperature is dependent on the back pressure. If the backpressure doesn't change, the saturation temperature doesn't change.

In winter the situation often arises that if there is too much cooling water flowing through the condenser, the condensate as it drips downward, is cooled a number of degrees below saturation temperature by the lower tubes. This is called undercooling and represents an unnecessary waste. For one thing it uses more pumping power than is actually required and also the additional heat which has been removed to bring the condensate below the saturation temperature has to be added later on in the feedheating system.

The ideal situation is where the condensate leaves the condenser at saturation temperature.

Condenser Location

Up to recent times the condenser (or condensers) were usually located at right angles to the turbine generator shaft. However, with the advent of 500 MW units, a new type of condenser has been introduced by some manufacturers. It is situated parallel to the turbine generator shaft and consists of one shell only. For this size unit there are 3 double flow L.P. cylinders (6 exhausts) and the one condenser shell has to reach across all 6 of these exhausts. This means that the condenser has to be about 70 ft. long. Figure 5 shows a sectional view of such a condenser. Notice that

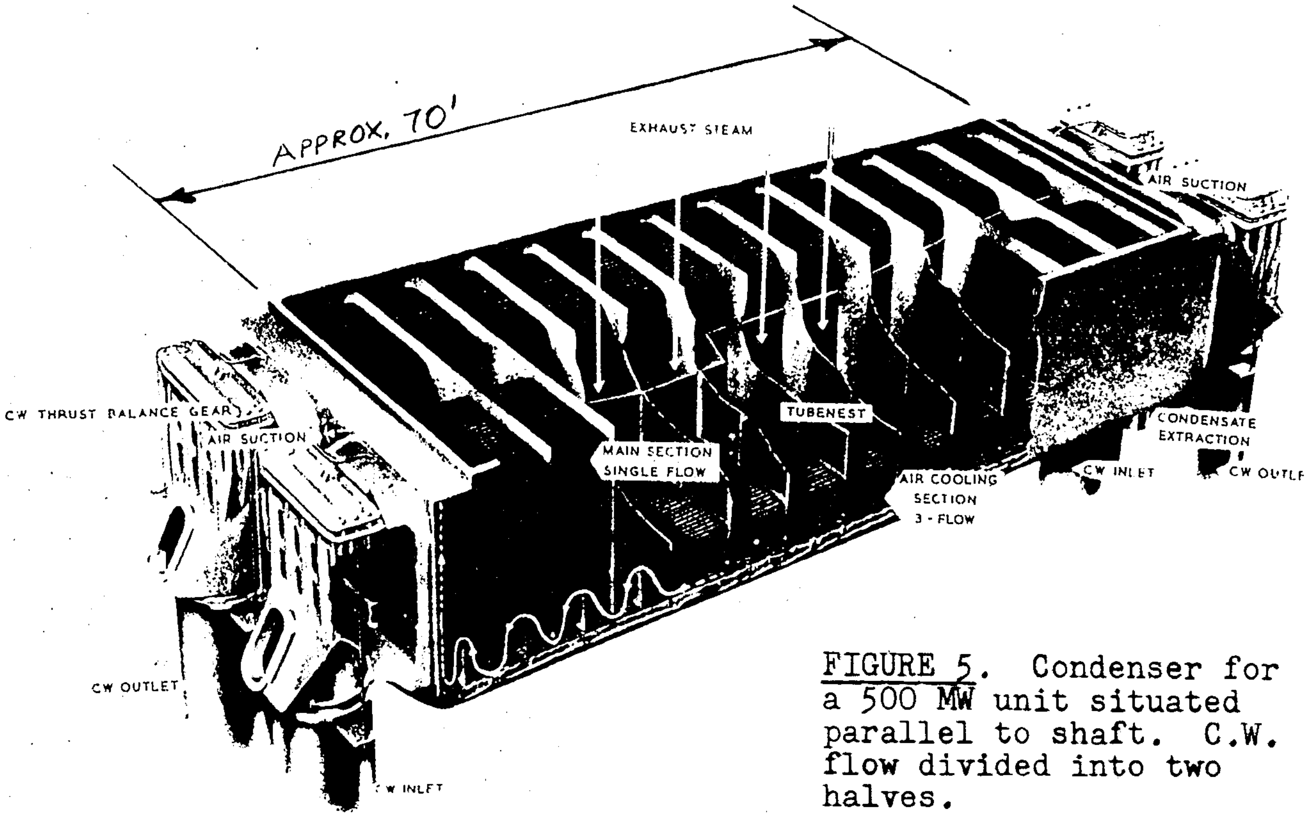


FIGURE 5. Condenser for a 500 MW unit situated parallel to shaft. C.W. flow divided into two halves.

the C.W. flow and the waterboxes are divided into two separate halves. Water on one side flows in one direction and water on the other side flows in the opposite direction. If the C.W. flowed in the same direction on both sides of the condenser then the C.W. outlet end of the condenser would be operating at a temperature of about 20°F higher than the inlet end. This could cause undesirable steam currents from the high to low temperature end. Also some turbine cylinders would be operating at a higher temperature than others, causing thermal distortion of the shaft. These two could combine to make for a rough running shaft (high vibration).

Methods of Air Extraction

The air extraction equipment has to be capable of dealing both with the conditions when vacuum is being raised and also with normal operating conditions. When raising vacuum the air extraction equipment has to remove not only the air which has filled the condenser during shutdown but also the air which has filled the spaces in the turbine including L.P., I.P., and H.P. cylinders, crossover pipes and piping right up to the emergency stop valves. This involves a large volume of air. Under normal operating conditions the quantity of air to be handled is reduced considerably.

Air and other gases are generally removed from a condenser by what are called air jet ejectors or by rotary air pumps. Figure 6 shows a sectional view of a steam-jet air ejector. Steam at about 300 psi. enters the nozzle which changes pressure energy to velocity energy. When the steam leaves the nozzle it has a very high velocity. This steam jet is directed to pass into the diffuser. The diffuser inlet has a much larger cross-sectional area than that of the high-speed jet entering. This excess area is open to the low-pressure region of the condenser which is to be evacuated, that is, the air-vapor offtake of the condenser. Since a gas in any enclosed container always completely fills the volume, the non-condensable gases in the condenser together with some vapor that gets carried along, will of their own accord tend to fill the air-vapor outlet connections right up to the inlet of the diffuser. At this point, then, we have high-speed steam that can go in only one direction (into the diffuser) surrounded by low-pressure mixture. Here we have the lowest pressure in the whole condensing system, in fact, in the whole steam plant cycle.

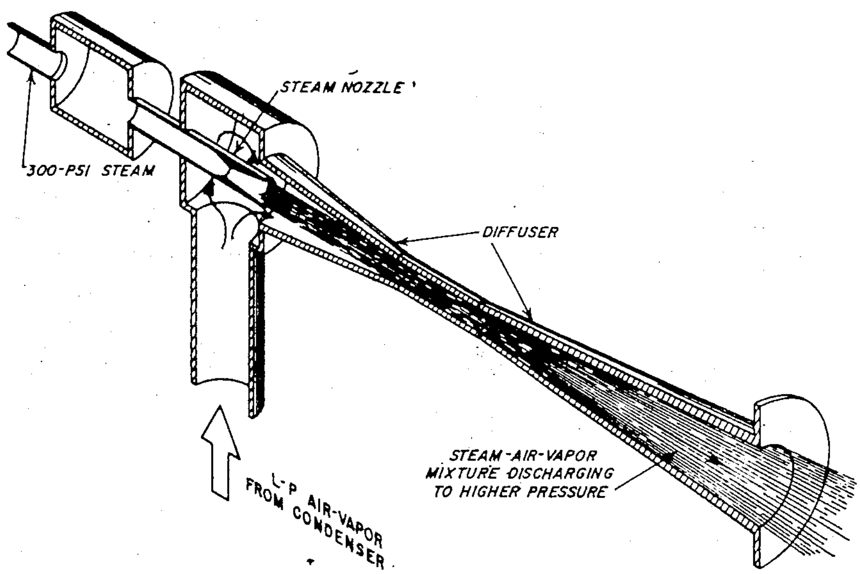


FIGURE 6. Sectional view of steam-jet air ejector which removes gas-vapor mixture from low-pressure region by entrainment in high-speed steam jet.

The low-pressure air-vapor mixture coming into physical contact with the high-speed steam jet is "batted" along by the steam to enter the diffuser also and is discharged along with the steam to a higher-pressure region. This constant removal of low-pressure mixture at the diffuser inlet encourages more gas vapor to flow into the diffuser and so establishes a continuous flow from condenser into ejector.

For start-up purposes, a large ejector is used which can handle such a large volume that pressure in the condenser is reduced to 10" Hg. absolute in not more than 10 minutes. This is called a hogging ejector. It uses a large quantity of steam.

During normal operation a much smaller quantity or air is required to be extracted and we look for a more economic way of extracting than with a hogging ejector. An ejector works best (with minimum steam use) over certain pressure-increase ranges and it is more economical to put two air ejectors in series as shown in figure 7. The first raises the pressure of the gas-vapor mixture from about 1 to about 5 in. Hg. abs. (2.5 psia). The second ejector then raises the pressure to slightly above atmospheric to discharge it into the surrounding air. Before the discharge from the first ejector enters the second one, the steam-gas mixture goes through a small condenser, called an intercondenser, to remove most of the steam as condensate. The cooling water normally is the main condensate from the main condenser. In this way the heat in the steam is recovered to warm the main condensate. Similarly, after the steam-gas mixture emerges from the second ejector, it passes through another small condenser, called an aftercondenser, so that primarily only the noncondensibles are discharged to atmosphere. The inlet and outlet connections for the cooling medium are not shown in figure 7.

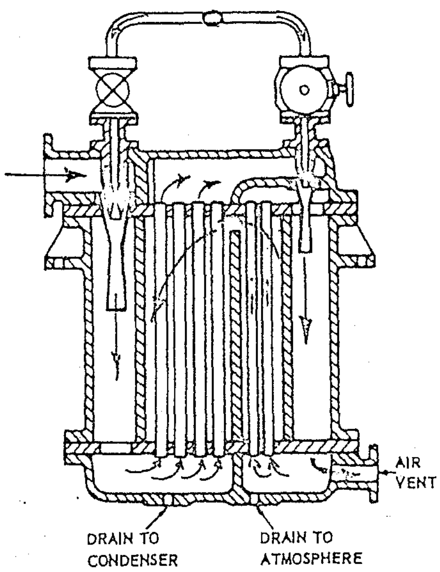


FIGURE 7
Two-stage ejector.

The steam for the air-jet ejector is normally taken from the main steam line which could be 1,000, 1,600 or even 2,400 psi. The ejector only requires it at around 300 psi., which means that it has to be desuperheated. This results in a big loss. The bigger the turbine and condenser the greater the flow of ejector steam required and above a certain size of unit it is no longer economical to use air-jet ejectors. Then the alternative is to use rotary air pumps, as shown in figure 8. It is generally electrically driven.

It uses water instead of steam as the ejecting medium. The impeller flings "slugs" of water at the throat of the diffuser. These successive slugs trap pockets of air. The air is thus carried along by the momentum of the water. The diffuser discharges to a tank where the air and liquid are separated.

Rotary air pumps are generally used for large units rather than air jet ejectors.

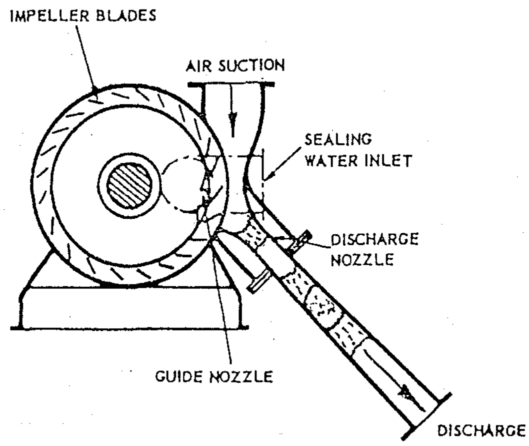


FIGURE 8. Rotary air pump

In this lesson we have only discussed air extraction from the steam side of the condenser. There is another air extraction system related to the condenser but it extracts air from the C.W. side of the condenser and these two systems should not be confused. It is assumed here that the second air extraction system will be described in the course on process systems.

Condenser Exhaust Valve

Steam flows into the condenser at a fast rate and if the C.W. supply should fail the backpressure could rise very rapidly in the condenser. Since the condenser is not made to withstand pressure from the inside it could easily burst. To avoid this a connection to atmosphere with an exhaust valve in the line is connected to the condenser. It acts as a relief valve and opens as soon as pressure in the condenser is above atmospheric pressure. In addition to this the L.P. cylinder often have bursting discs built in which burst when the pressure is a certain amount above atmospheric. This then provides an opening to relieve the pressure.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-3 - Surface Condenser Details

A - Assignment

1. In what way does a single pass condenser vary from a double pass condenser?
2. What is the purpose of backwashing a condenser?
3. What undesirable effects does air produce in the condenser?
4. Why is it important to have the condenser tubes fixed tightly?
5. Explain why such a large volume of cooling water is required for the condenser.
6. What is undercooling? Why is it undesirable?
7. Is the turbine backpressure better in summer or winter? How does this effect the cycle efficiency?
8. Name two methods of air extraction from the condenser.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-4 - Regenerative Feedheating System

0.0 INTRODUCTION

At the T.T.4 level we have discussed L.P. heaters, deaerators and H.P. heaters to a certain extent. This lesson will describe some more of the equipment used in the regenerative feedheating system and how this equipment is related and connected together.

The reader should bear in mind that regenerative feedheating systems vary from manufacturer to manufacturer and even from designer to designer. The examples in this lesson in some cases portray feedheating systems actually installed in a particular steam station. The student can expect to find in any steam power station equipment similar to what is described here, but should not expect it, or the arrangement to be exactly the same from station to station.

Again it is assumed that the student has taken the course on Heat and Thermodynamics at the T.T.3 level.

1.0 INFORMATION

In figure 1 is shown a diagramatic arrangement of a typical tandem-compound reheat unit with six-stage feedwater heating. The diagram has been simplified considerably for clarity because there are many more connections than this normally.

As a review of the lesson called "Closed Feed Cycle" T.T.4 level, we will go over the cycle again. Condensate extraction pumps draw condensate from the condenser hotwells and pump it through No. 1 and No. 2 L.P. heaters to the deaerator.

Normally boiler feed pumps (B.F.P.) would draw the water from the deaerator, but in this particular example the feedwater is taken from the deaerator by lift pumps (or booster pumps) and pumped through Nos. 4, 5 and 6 H.P. heaters. From there boiler feed pumps take the water and pump it into the boiler at the required pressure. Steam is formed in the boiler and passes from the steam drum through the superheater to the H.P. cylinder of the turbine. The discharge from the H.P. cylinder passes back to the boiler to be reheated and from there passes through the I.P. and L.P. cylinders to the condenser.

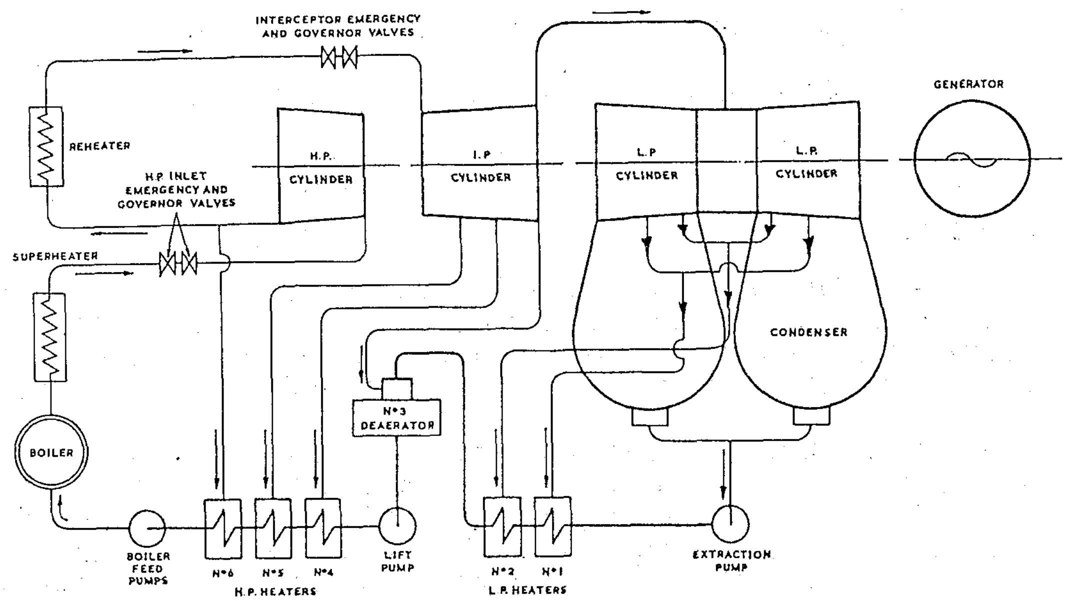


Fig. 1 Diagrammatic arrangement of a typical tandem-compound reheat unit with six-stage feedwater heating.

Extraction Steam Pressure

As we said previously the heat required to raise the temperature of feedwater in feedheaters comes from steam which is extracted from various points in the turbine. The pressure and temperature of the steam differs a great deal from L.P. heaters to H.P. heaters. Take for example the extraction steam for L.P. heater No. 1. It comes from a place in the L.P. turbine near the last couple of stages. As such the steam is probably at a pressure of around 5 - 10 psia. (i.e. below atmospheric). This means that the steam in the L.P. shell is under vacuum conditions. On the other hand the extraction steam for H.P. heater no. 6 in figure 1 is taken from the discharge of the H.P. cylinder. Steam at this point may have a pressure of approximately 400-500 psia. which means that the heater shell is subjected to this pressure. This of course refers to a conventional type of steam station. Steam pressures in Nuclear stations are much lower than in a conventional station.

Feedwater Pressures.

Figure 2 shows typical full load conditions of feedwater for a 60 MW turbine. This feedheater train differs somewhat from the one shown in figure 1. The flow is from right to left.

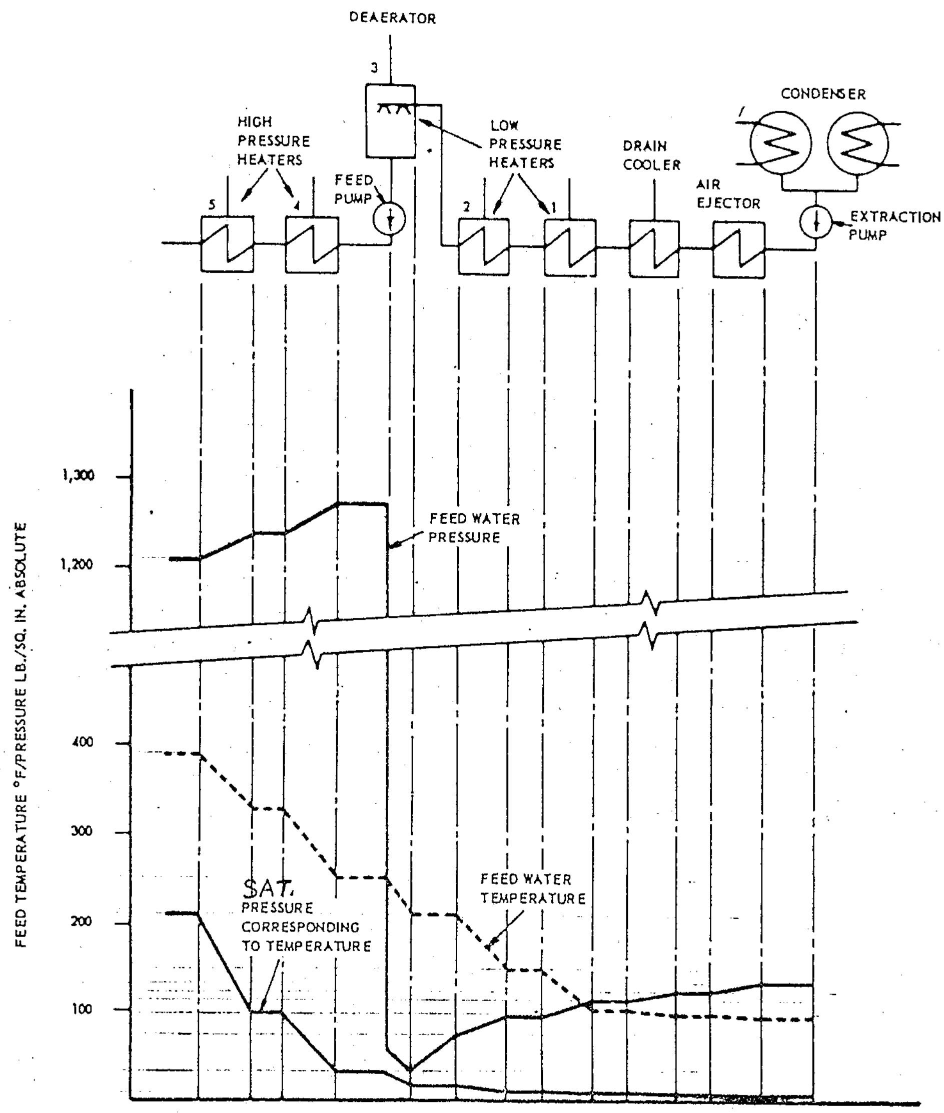


Fig. 2 Typical full-load conditions of feed-water for a 60 MW turbine.

It is intended that this example will give the student an idea as to the pressures and temperatures encountered in a regenerative feedheating system.

The extraction pumps raise the water pressure up to around 130 psi. The feedwater temperature at this point is roughly 95°F. As the flow proceeds through air ejector coolers drain cooler No. 1 & 2 L.P. heaters, and No. 3 heater (or deaerator), the F.W. temperature increases steadily step by step, but the pressure drops due to friction losses through the heater tubes. In the deaerator the pressure is 31 lbs/sq. in. for this cycle, and at this point the pressure is very near saturation -- i.e. the water is near boiling point, which is most desirable because this drives off any dissolved gases. From the deaerator the water drops into the reserve tank and the pressure at the bottom of the reserve tank is higher than in the deaerator because of the height of the water in the tank.

The boiler feed pumps then raise the feedwater pressure up to 1265 psia. The inside of the feedheater tubes are subject to these pressures. In this particular case the extraction steam would not be as high as in the previous example. The outside of the heater tubes (or the condenser shell) in this example would be subject to an extraction steam pressure of 110 psia for the H.P. No. 4 heater and 250 psia. for H.P. No. 5 heater. The final feed temperature arrives at around 390°F prior to passing into the boiler.

From a practical point of view the final feed temperature must be kept considerably below the boiler saturation temperature (roughly 100°F) to guard against evaporation in the feed lines in the event of a reduction in feed pressure.

It should be noted that all temperatures and pressures referred to above relate to full load 60 MW conditions. Figures for lighter loads are lower. For example at 24 MW the feed water is heated from 79°F at the extraction pump discharge to 319°F at the final H.P. heater outlet.

Heater Drains (or Drips)

The water resulting from condensation of the extraction steam falls to the bottom of the heater body and must be removed at the rate that it is formed. Sensible heat in the drain water from the H.P. heater represents about 28% of the total extracted from the turbine at this point. Obviously this water must be conserved and the fullest use made of its heat content.

The present day methods of achieving this have been evolved in the light of past experience, and certain recommendations are now generally applied. It will be appreciated that the drain water leaving the heater is almost at boiling point and that even a slight reduction in its pressure will cause it to boil. In a typical modern system, the drain water leaving the highest pressure feed heater enters a container called a flashbox, which is subject to the pressure of the next lower pressure feed-heater. Therefore the drain water will boil and some of it will evaporate to form steam. In engineering terms we say the water is flashing into steam.

The drain water is not led directly to the next lower pressure feedheater because when flashing takes place it can have an erosive effect on the metal with which it comes in contact. A heater designed to prevent erosion due to flashing becomes quite complicated. It is easier and more economical to provide a separate flashbox in which the flashing can take place.

Flashbox

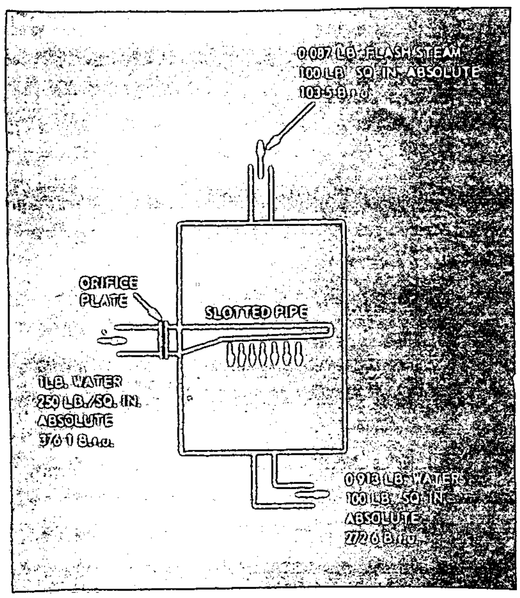


Fig. 3 Flashbox

Fig. 3 shows a diagrammatic arrangement of a typical flashbox. It consists of a steel container with a connection at the top through which vapor can escape. A drain water connection is located at the base. Entering water drains are connected to the side or at the base. The entering drain water flows through an orifice plate mounted close to the casing of the flashbox to ensure that flashing will take place inside the box and not in the pipework. Erosion often occurs immediately after an orifice in a pipe and if the orifice was located some distance from the flashbox the pipe would soon be worn through. Each drain connection to a flash box has its own orifice plate, the size of the aperture of the plate is designed to limit the flow of drain water from the heater at the same rate as the steam is condensing in the heater. This ensures that there is always some water in the heater, otherwise live steam could flow through the drain at the bottom of the heater and into the flashbox.

The size of the flashbox depends on the volume of vapour it has to accommodate. One pound of dry saturated steam at 100 psia. occupies 4.4 cu. ft. while at a condenser pressure of say 29 ins. Hg. the volume is 643 cu. ft. In the higher pressure boxes, a pipe extends from the orifice plate into the flashbox, as is shown in figure 3. The underside of this pipe is slotted to give a downwards direction to the incoming water in order to minimise entrainment of water by the flash steam passing upwards.

In figure 3 the drain water coming from a heater is at 250 psia. At this pressure 1 lb water contains 376.1 Btu. The top of the flashbox is connected to the next lower pressure heater which is at 100 psia. At this pressure 1 lb. water can contain 298.5 Btu only. Therefore 376.1 - 298.5 = 77.6 Btu of heat is given up by the water. In order for water to evaporate it must absorb latent heat of vaporization which at 100 psia, is 889.7 Btu. The heat released by 1 lb. of water as it reduces in pressure from 250 psia, to 100 psia, will evaporate:

$$\frac{77.6}{889.7} = 0.087 \text{ lbs. of steam}$$

The water left in the flashbox is 1 - 0.087 = 0.913 lbs. which will drain out the bottom to the next lower pressure flashbox. This will become apparent in the feedwater flow diagrams shown in the next figure.

Drain Cooler

A drain cooler is a heat exchanger used to increase the temperature of the feedwater but its only source of heat is from drain water from feedheaters. It does not receive extraction steam from the turbine. Figure 4 shows how three drain coolers have been fit into a regenerative feedheating system.

Regenerative Feedheating Systems

Figure 4 shows a diagram of a 200 MW regenerative feedheating cycle. All valves and flow control devices have been omitted to simplify the diagram. One turbine cylinder only is shown for the same reason. Actually the steam would be extracted from a number of cylinders as shown in figure 1. The weight of steam W lb/hr extracted in each line, as well as the pressure P, and temperature of the steam is for full load conditions.

If you add all the extraction steam flows together with the condenser flow you get 1,269,000 lb/hr which must equal the flow into the turbine. Of this 399,570 lb/hr or roughly 33% is extraction steam.

Notice the related connections and locations of the flashboxes and drain coolers in the cycle. The first flashbox on the left hand side is called the # 6 drain cooler flashbox. It receives its drain water from No. 6 H.P. heater but is controlled by No. 6 drain cooler because the flashbox will be at whatever pressure the drain cooler is at and this pressure will determine how much of the water in the flashbox will form into steam. Similarly, the next flashbox is called # 5 H.P. heater flashbox because it has a steam connection to No. 5 H.P. heater and is thus controlled by this heater. Orifices in the drain water lines determine the amount of flow taking place in these lines.

As you can see from the diagram the drain water is cascading from one flashbox to the next--from lower pressure to lower pressure till finally it arrives in the condenser. This type of arrangement is often called a <u>cascading cycle</u>.

Observe also that in the cycle in figure 4 there is a No. 5 & 6 desuperheater, a No. 5 & 6 H.P. heater and a No. 5 & 6 drain cooler. However, this is called a 6 stage feedheater train because it extracts from only 6 points in the turbine.

You will notice that of the No. 5 & 6 heaters only the desuperheaters (D.S.) receive extraction steam directly from the turbine. This extraction steam is superheated and it has to be desuperheated (i.e. lowered to its saturation temperature) before it can be condensed. No condensation takes place in No. 5 & 6 desuperheaters. The steam is only lowered to its saturation temperature and from there passes on to No. 5 & 6 H.P. heaters where it condenses, and drains to the flashboxes. Flash steam from here heats the drain coolers.

You will notice that the extraction steam for No. 5 D.S. is at 799°F, whereas the extraction steam for No. 6 D.S. is at 655°F even though the latter comes from a place in the turbine where you would expect higher steam conditions than for the former. It is not shown in the diagram but this is a reheat cycle and extraction steam for No. 6 D.S. comes from the discharge of the H.P. turbine cylinder. The main steam is then reheated in the boiler to 1000°F and passed on to the I.P. turbine cylinder. The extraction steam for No. 5 D.S. comes from the I.P. cylinder and thus can be at a higher temperature. If you look up the enthalpies you will find that steam entering No. 5 D.S. has 1424 Btu/lb while steam for No. 6 D.S. has 1332 Btu/lb. This explains why No. 5 D.S. is located later in the cycle than No. 6 D.S.

200 MW feed water flow diagram

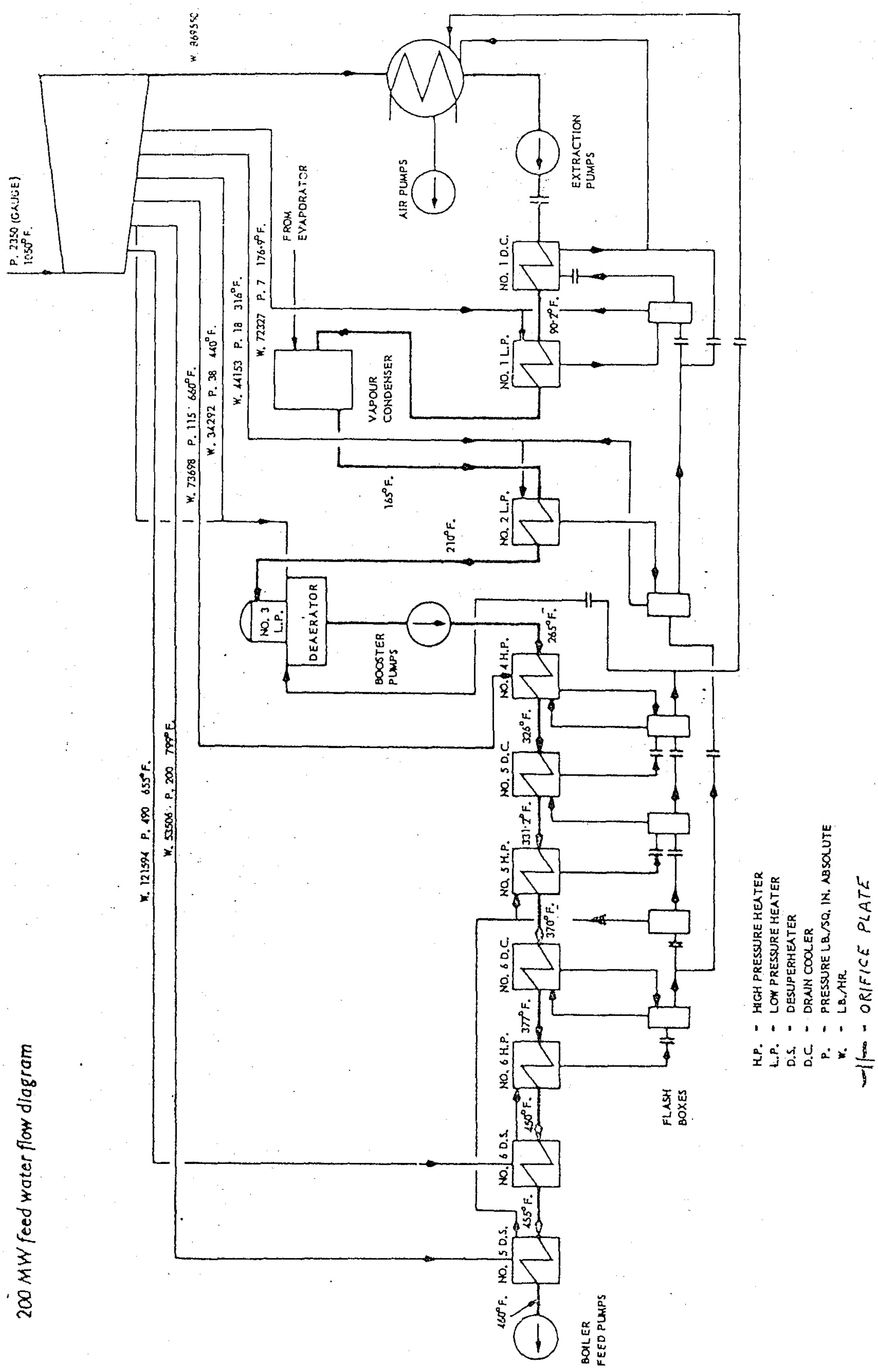


FIGURE 4

The pressure of feedwater for the system in figure 4 is raised in three steps--by extraction pumps (usually three 50% capacity pumps), by the booster pumps (possibly three 50% capacity pumps) and by the boiler feed pumps (possibly three 50% capacity pumps). The boiler feed pumps for this cycle would have a discharge pressure of roughly 2500 to 2600 psig. If the feedheaters had to withstand this pressure they would have to be built strong accordingly and would thus be quite expensive. For this reason booster pumps are used as an " in-between stepping stone" for large units.

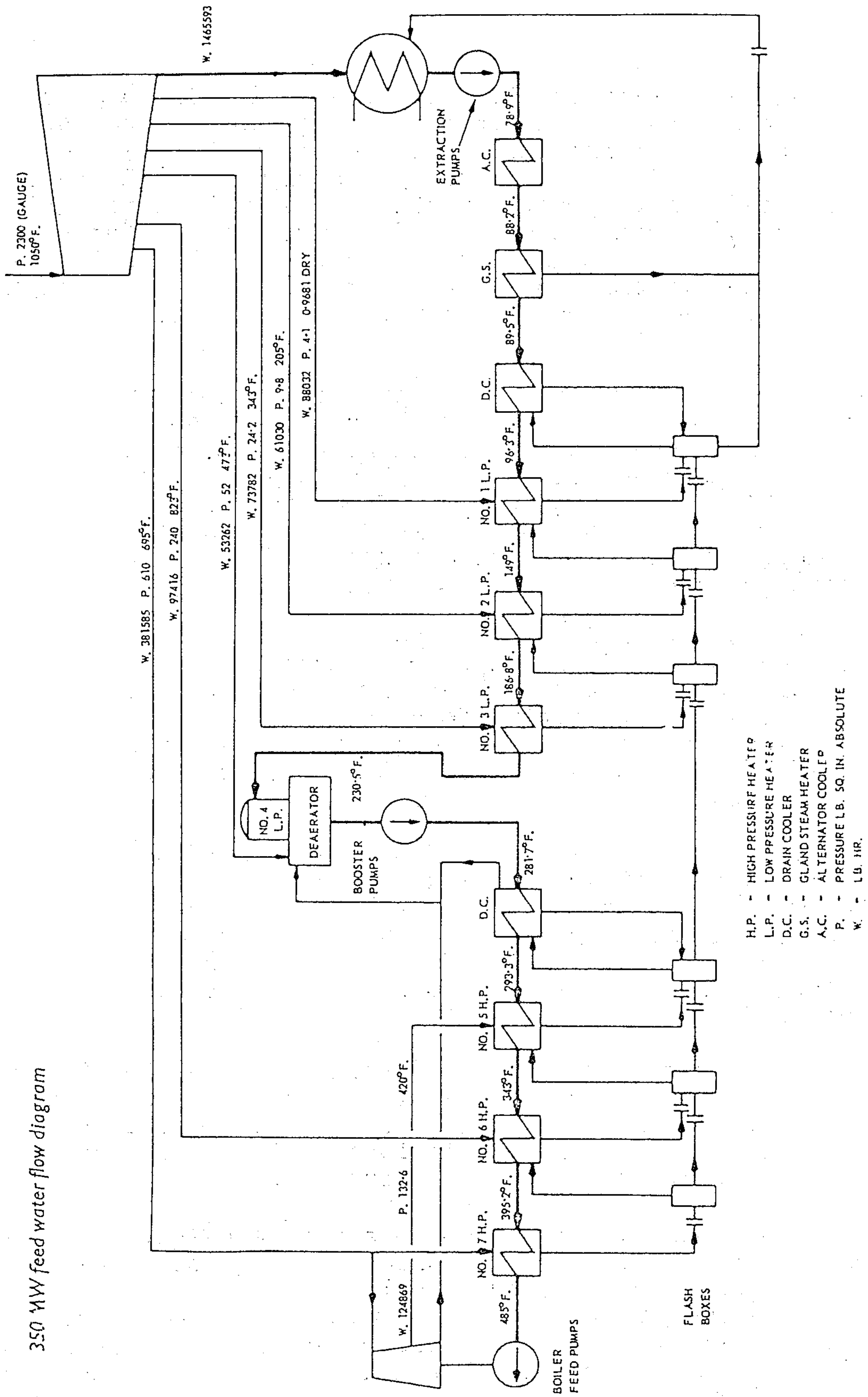
Notice in figure 4 there is a vapour condenser. Sometimes when raw water or make-up water for the system is too unclean it is evaporated in an evaporator (to remove the impurities) and then condensed in a vapour condenser. The heat used for evaporation comes from turbine extraction steam. The vapor thus formed from the raw water gives up its latent heat of vaporization in the vapor condenser. This heat is used for raising the temperature of the feedwater as can be seen in the diagram.

Figure 5 shows a diagram of a 350 MW feedwater flow diagram. In this case the boiler feed pumps are driven by a steam turbine which is incorporated in the regenerative feedheating system. This turbine receives its motive force from extraction steam from the H.P. turbine cyclinder. The cycle efficiency can be improved by about 0.5% for this size unit by using a steam turbine rather than electric motor to drive the boiler feed pump.

Figure 5 also includes a gland steam heater (G.S.) which receives steam leaking off from the turbine glands and it is used to heat feedwater. There is also an alternator (or generator) cooler (A.C.). The heat produced in the generator due to electrical resistance of the copper windings, is transported to the generator cooler where it is used to heat feedwater and raise its temperature 9.3°F.

Figure 6 portrays a regenerative feedheating system for a 550 MW unit with seven stages of heating. The No. 7 stage consists of two separate heater shells, half of the feedwater flowing through one and half through the other. Each shell combines a desuperheating part and an H.P. heater part.

Why the two shells? In a non-reheating regenerative cycle optimum efficiency is attained approximately when the sensible heat rises across all heaters are equal. However, a thermodynamical feature of a reheat-regenerative cycle (in which the steam for the last heater is bled before the reheater), is that optimum efficiency requires the sensible heat rise in the last heater to be greater than the average sensible heat rise in all other heaters. This feature renders the last heater surface very much



350 MW feed water flow diagram

FIGURE 5

greater than that of the preceding heaters, and in figure 6 the last heater is made in two shells to limit the dimensions and weights. These two shells work in parallel, and on account of the large quantity of steam bled to this heater, a high pressure drain cooler is provided to utilize a proportion of the sensible heat in the last heater drain at the highest possible temperature level.

From the above example you can see that every effort is made in regenerative feedheating systems that as little heat as possible is wasted.

Heater Vents

Steam may contain some non-condensible gases. If these get trapped in the heaters they would blanket the tubes and hinder the transfer of heat to the feedwater. To ensure that this does not happen each heater is vented to the condenser by means of small bore tubing. The tubing has a small bore so that as little steam as possible will be wasted to the condenser.

Drain Pump

The cascading type of cycle is simple and needs little attention. However, there is another method which eliminates the flashboxes. The heaters drain water is pumped directly into the feedwater line. The advantage is that the heater drain water temperature is generally a little higher than the feedwater coming out of the heater and pumping the drain water into the feed line will give a higher feed temperature.

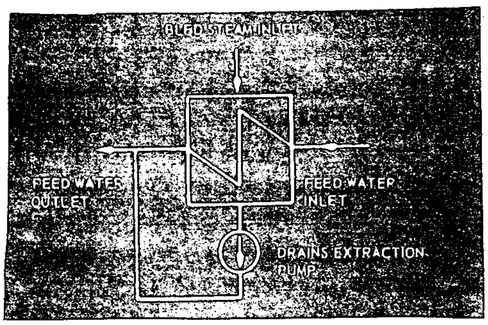
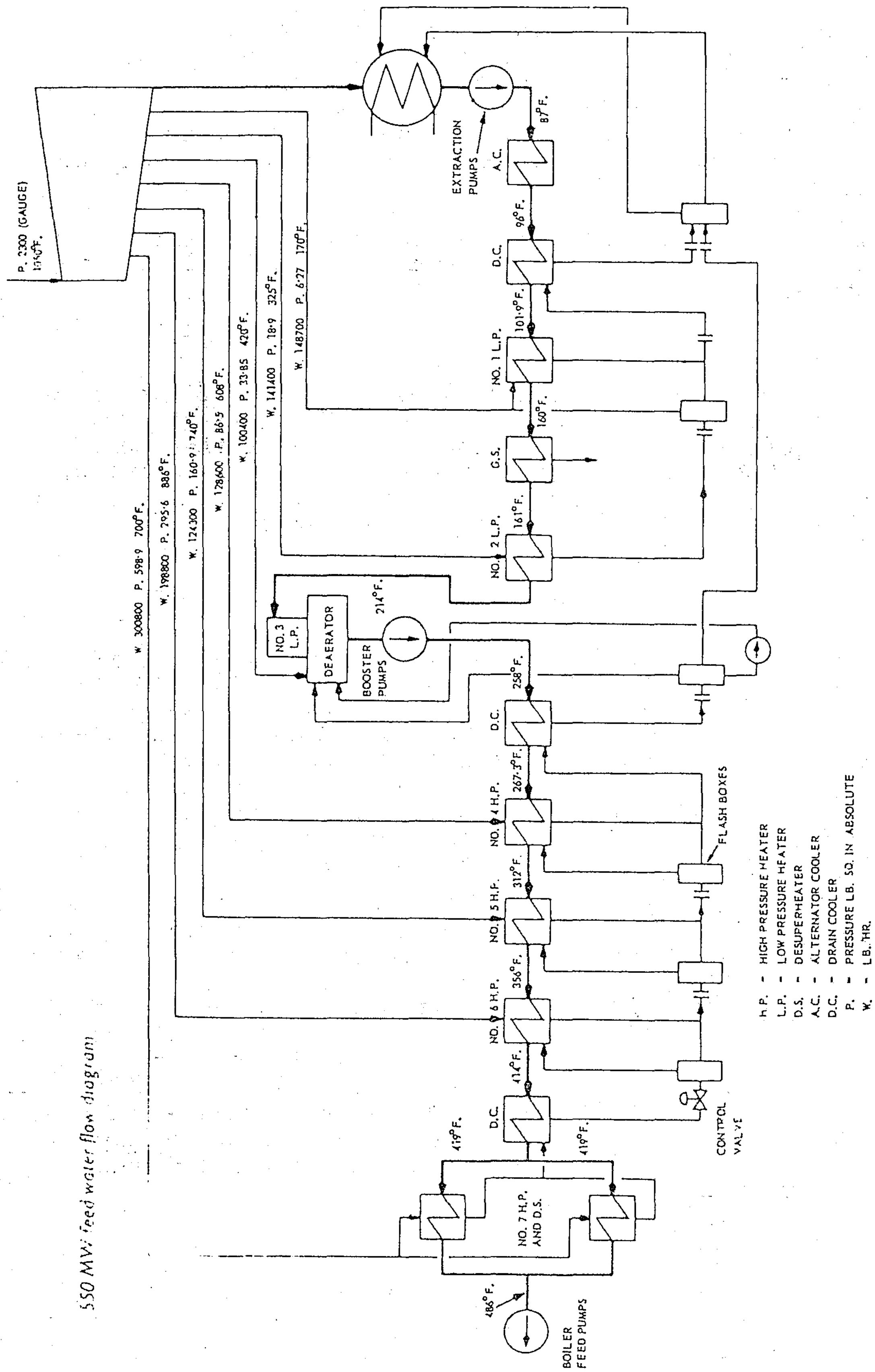


Figure 7 Heater Drain Pump

Against this advantage there is the danger of introducing dissolved gases into the feed water by leakage through the pump glands or from pockets of air in the heater body. Also the cost and complexity of drain pumps discourage the use of them for surface type heaters. You will notice that figure 6 shows a drain pump taking water from one of the flashboxes and pumping it into the deaerator.



550 MW: feed water flow diagram

FIGURE 6

Dick Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-4 - Regenerative Feedheating System

A - Assignment

1. Explain how it is that extraction steam pressure varies over a wide range.
2. What do you understand by the term "heater drains (or heater drips)"?
3. Draw simplified regenerative feedheating system for a typical unit in a steam power station.
4. Explain how flash steam is formed.
5. In what way does a drain cooler differ from an ordinary heater?
6. What do we mean by a cascading cycle?
7. Why are feedheaters vented?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-5 - Generator Construction

0.0 INTRODUCTION

We have briefly introduced the electric generator at the T.T.3 level. This lesson will continue to describe the generator construction from a mechanical point of view and will deal more with how things are put together and why rather than describe how a generator produces electricity. It is assumed that electric generation has been covered in the "Electrical" course.

1.0 INFORMATION

The detailed design of generators is left to the manufacturer who must produce a machine that meets the requirements laid down in the customer's specifications.

Stator Construction

Modern stator frames, which are circular, are made of welded steel plate designed to form a rigid structure. Inside this outer shell is mounted a cage as shown in figure 1. The compartments of the cage, which form part of the cooling system, provide ample space for the circulation of a cooling medium. The stator bars, which are seen in the figure, lie axially along the inside of the cage and hold the steel stamplings of the stator core. A certain amount of flexibility is generally allowed in the support between the outer shell and cage to assist in damping out generator vibrations occuring at 120 cycles/sec. These vibrations tend to be set up by the magnetic pull of the rotor which pulls on and releases sections of the core each time a north or south pole face passes by.

Stator Core

The stator core experiences losses due to what is called hysteresis and eddy currents. In order to minimize these losses it is necessary to adopt a laminated construction for the core. This lamination, one small section of which is shown in figure 2, is stamped out of thin sheet steel of high magnetic permeability, the usual thickness being about 0.014 ins. One side of each sheet is treated with a suitable insulating material, such as paper, enamel, or varnish, to prevent the flow of eddy currents between laminations, and it is fitted with duct spacers as shown. The stampings are built up in packets to form the core with duct spacers put in to form ventilation spaces at regular intervals. A section of the core is shown in figure 3. The number of stampings required to make a stator core could well run into the hundreds of thousands.

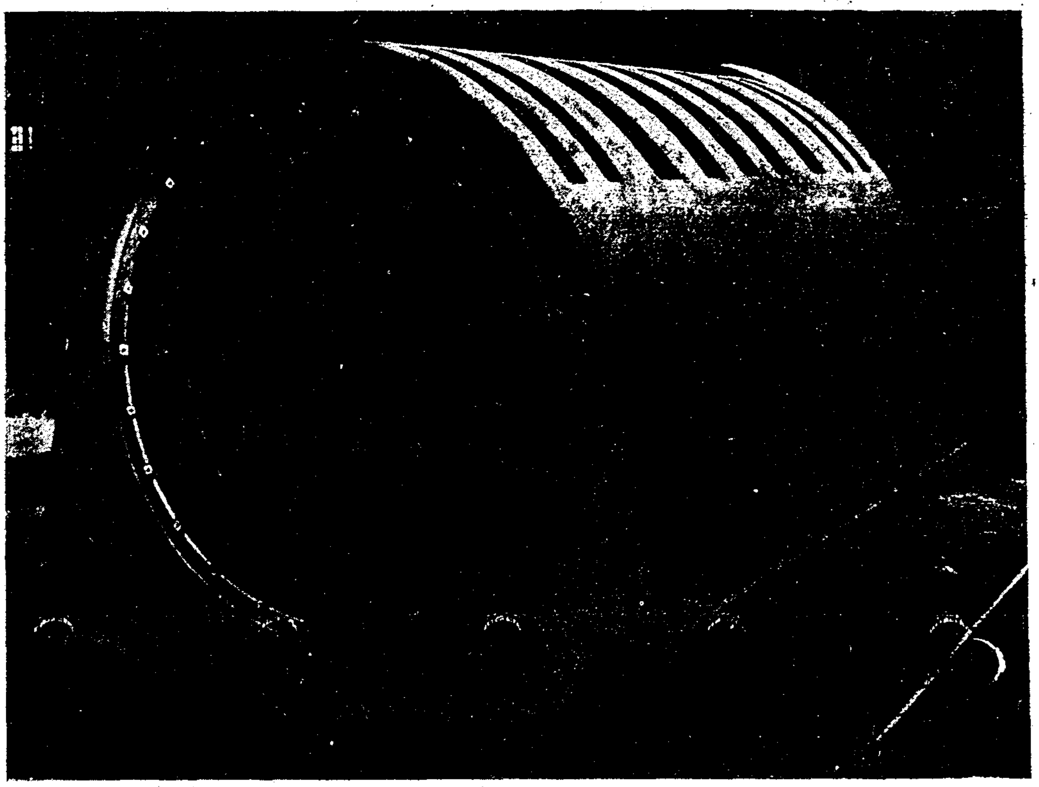


FIGURE 1. Stator frame of 37,500 KVA. generator with stator bars in position.

You will notice that the stamping in figure 2 has a little niche on each side at the top. Adjacent stator bars shown in figure 1 fit into these niches and act as keys to secure the stampings inside the stator.

Figure 4 shows an end view of a stator frame with laminated core in position ready for winding. Notice the slots into which the windings fit. When completed the core is clamped between heavy end-plates of non-magnetic material.

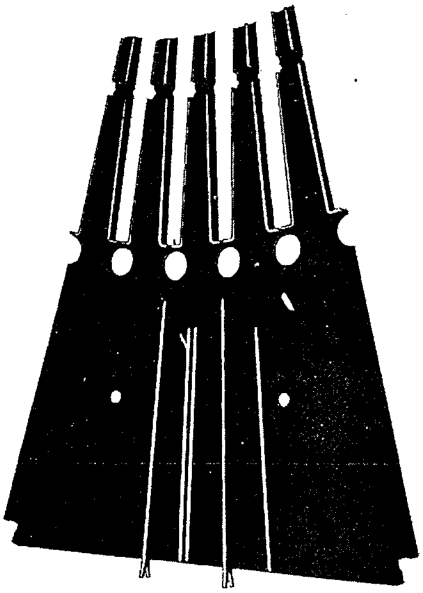


FIGURE 2

Figure 2 - Typical lamination stamping with rolled steel duct spacers.

Figure 3 - Section of a stator core. Duct spacers for ventilation welded flush to the sides of a heavy punching give winding slots with smooth unbroken sides.

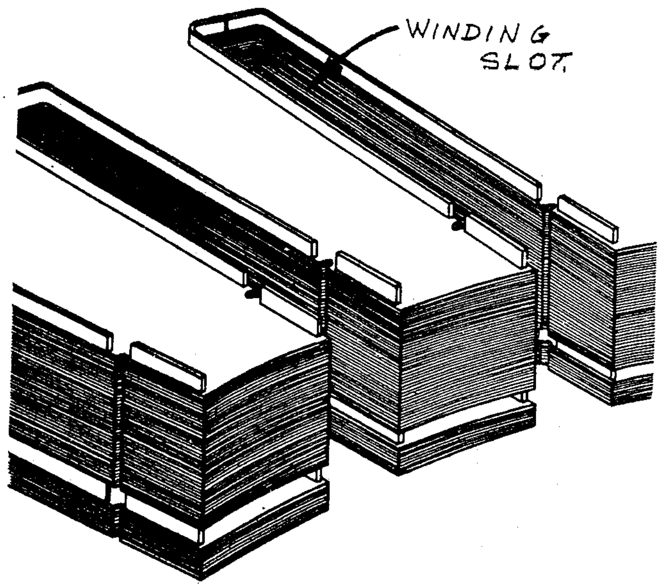


FIGURE 3

In large electric generators it is an advantage to have a large air gap between the rotor and stator. The size of air gaps will be between 2 or 3 inches, so that the diameter of the stator will be the rotor diameter plus twice the air gap. The diameter and length of the rotor will depend on the critical speed of the shaft and the peripheral speed. (dritical speed will be explained in a later lesson and the limiting peripheral speed will be discussed later on in this lesson.) Therefore the stator length and inside diameter is determined from the rotor design.

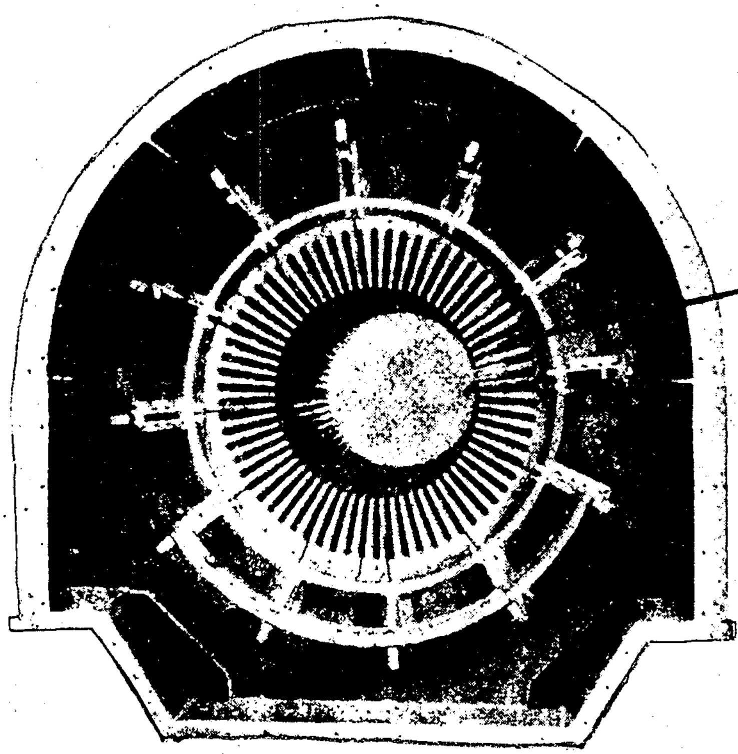


FIGURE 4. Stator Frame with core in position ready for winding.

Stator Windings

Because of skin effects with alternating currents, the stator windings have to be divided into a number of thin conductors each wrapped with insulating material in order to obtain an even distribution of current in the conductors. The conductors, on the whole, are quite massive in large machines. Because of the use of deep slots in the core, there is a considerable flux leakage, with the result that the voltage induced in the laminated conductors is not identical. This difference in voltage will cause circulating or eddy currents to flow in the laminated conductors which are joined together at the terminals of the machine. To minimize this eddy current loss the conductors are usually transposed in the slots in a manner shown in figure 5 so that each conductor consecutively snakes its way from top to bottom and then again from bottom to the top of the winding. This method of transposition is known as the "Roebel" stranded bar.

The individual strips of copper are insulated with mica splittings attached to a backing of glass fabric, nylon or other suitable material by means of shellac or bitumastic varnish. The strips are then tightly bound together by the further application of mica tape and suitable varnish to form a bar of the required dimensions. This is pressed together tightly in a steam heated press and consolidated into a rigid conductor. The bars are then subjected to vacuum drying and impregnation in order to eliminate moisture and to prevent voids forming in the insulation. Figure 6 shows a cross section of the end product of two such windings placed one on top of the other in a stator core slot.

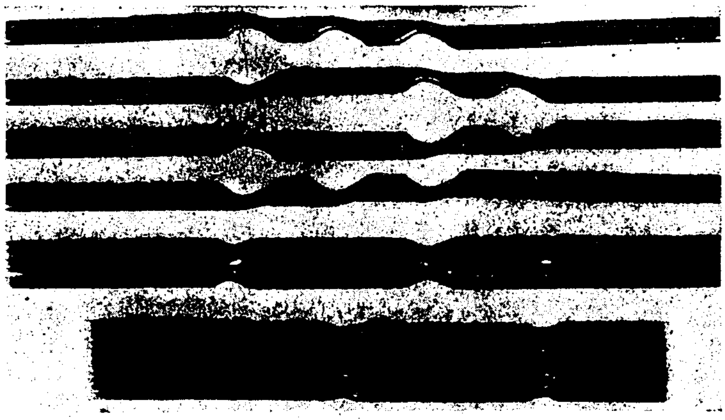


FIGURE 5. Roebel stranded bar showing method of transposing stator conductors to reduce eddy current losses.

As you are no doubt aware we are talking about a 3-phase generator, good electrical design requires that there be between 9 and 12 slots per phase per pole for large modern machines. These generators are two pole machines, therefore there would be 18 to 24 slots per phase. So for a 3-phase machine there would be from 54 to 72 slots depending on the designer.

Since there are two windings in each slot that means 36 to 48 windings per phase. These windings are silver soldered or brazed together end to end to form one continuous conductor per phase as shown diagrammatically in figure 9. The points where the windings are brazed together are called <u>end turns</u>.

The end turns of the windings, when completed are very strongly braced to the frame as shown in figure 7. This forms a very rigid structure which can withstand the heavy stresses set up under short circuit conditions. The windings are secured in the slot by driving

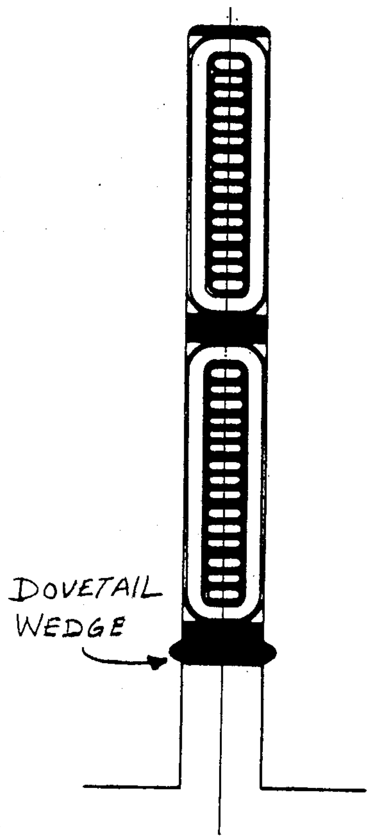


FIGURE 6. Typical stator slot assemblies showing windings, one on top of the other.

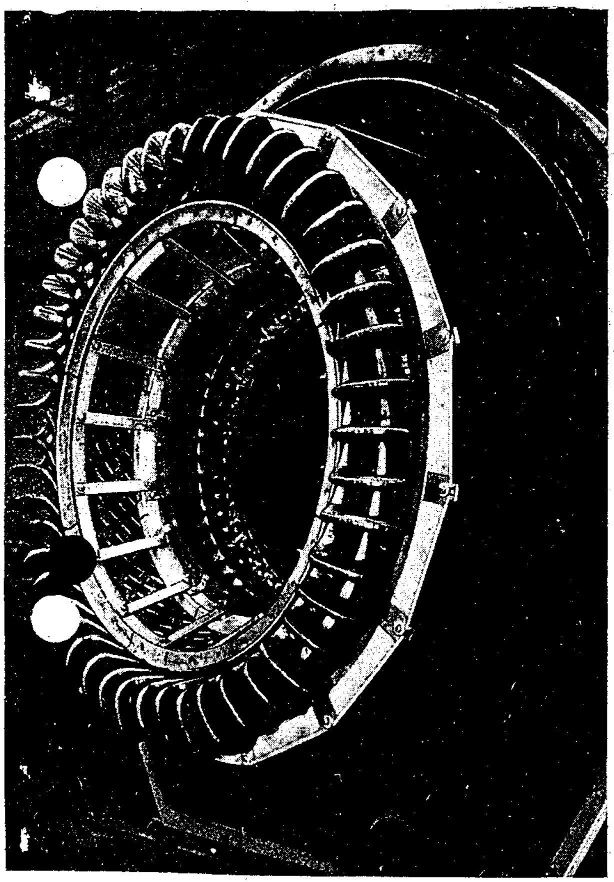


FIGURE 7. The end windings of a large generator stator showing supports and clamps.

in dovetail wedges of insulating material as shown in figure 6.

In figure 8 you can see the terminal leads of the stator windings brought out at the bottom of the frame.

The end windings are protected by specially designed covers which also act as guides for the flow of the cooling medium. These are usually made of non-magnetic material, such as aluminum alloy, because it is necessary to avoid having magnetic material in the vicinity of end winding leakage flux, since this would cause heavy eddy current losses. In some instances covers are made of special bakelised board or other suitable material, in preference to metal.

Rotor Construction

One of the most difficult mechanical engineering problems a manufacturer has to face is the satisfactory design of the rotor. The outside diameter is limited by the maximum peripheral speed which is usually accepted, using the existing materials as being 550 ft./sec. Therefore 35 ins. is normally the maximum diameter permitted for 3600 rpm machines. The length between bearings must also be kept within limits because of the critical speeds of the shaft.

When a shaft is rotating at its critical speed it will experience extreme vibrations which can damage the machine. Above or below the critical speed the shaft will run smoothly. A shaft may have two or three critical speeds, but it should never operate at a speed near to critical. The distance between bearings and weight of the shaft will govern what the critical speeds are and therefore we said the distance between bearings must be kept within certain limits. The first critical speed can fall between, say, 1250 and 1400 rpm. and the second critical speed not less than 4320 rpm. (since operational speed is 3600 rpm.)

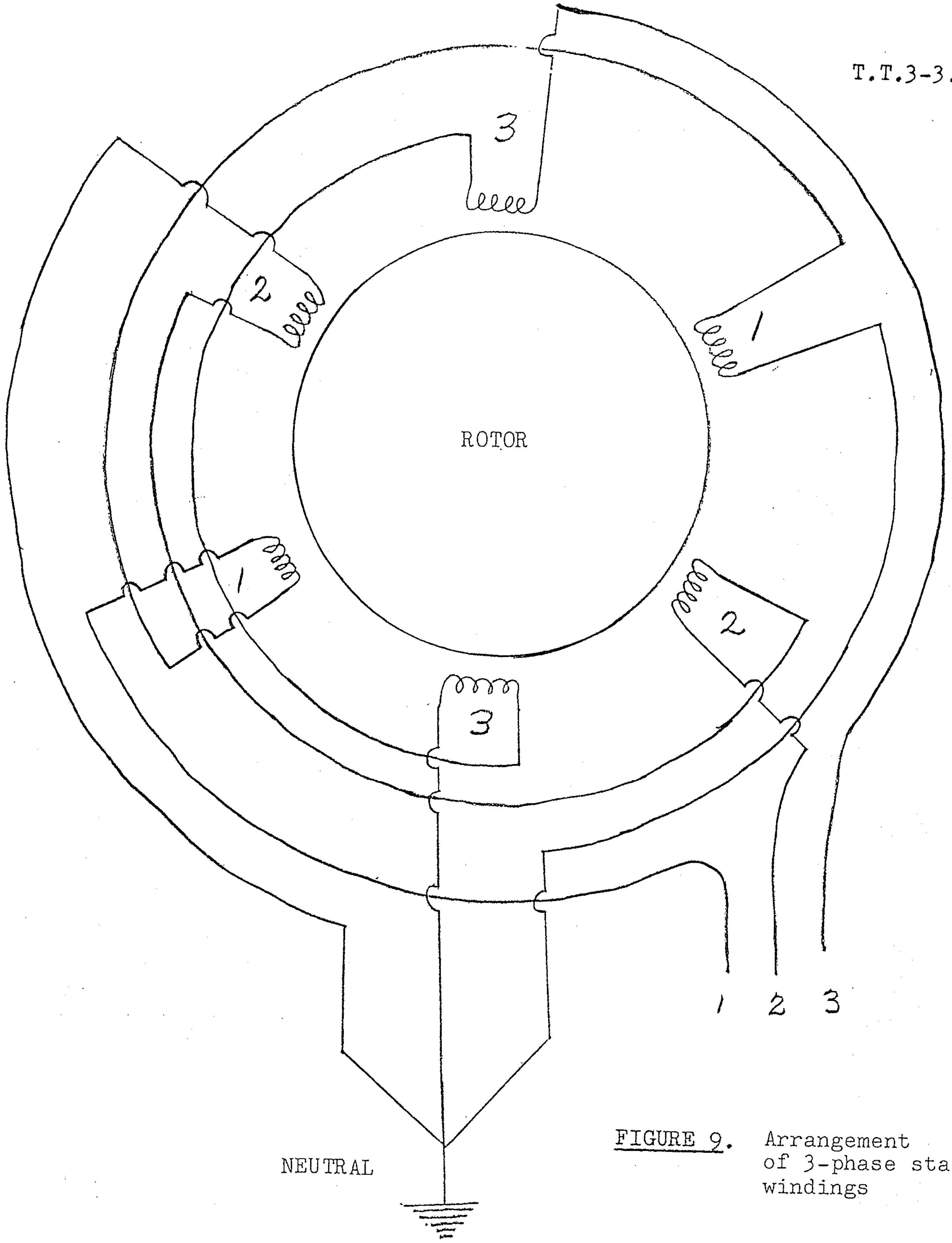


FIGURE 9. Arrangement of 3-phase stator windings

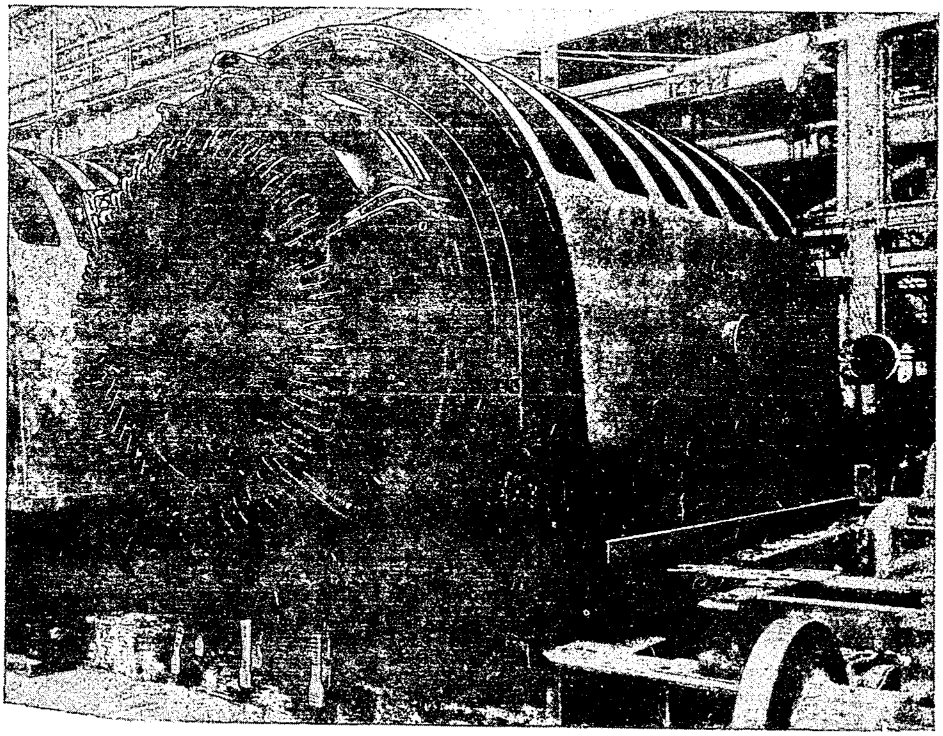


FIGURE 8. The terminal leads of the stator winding brought out at bottom of frame.

These two limiting factors (peripheral speed and distance between bearings) determine the maximum size (and output) to which a generator can be built. It will determine whether two generators have to be built per unit and hence whether the turbine will have to be cross compound or tandem compound.

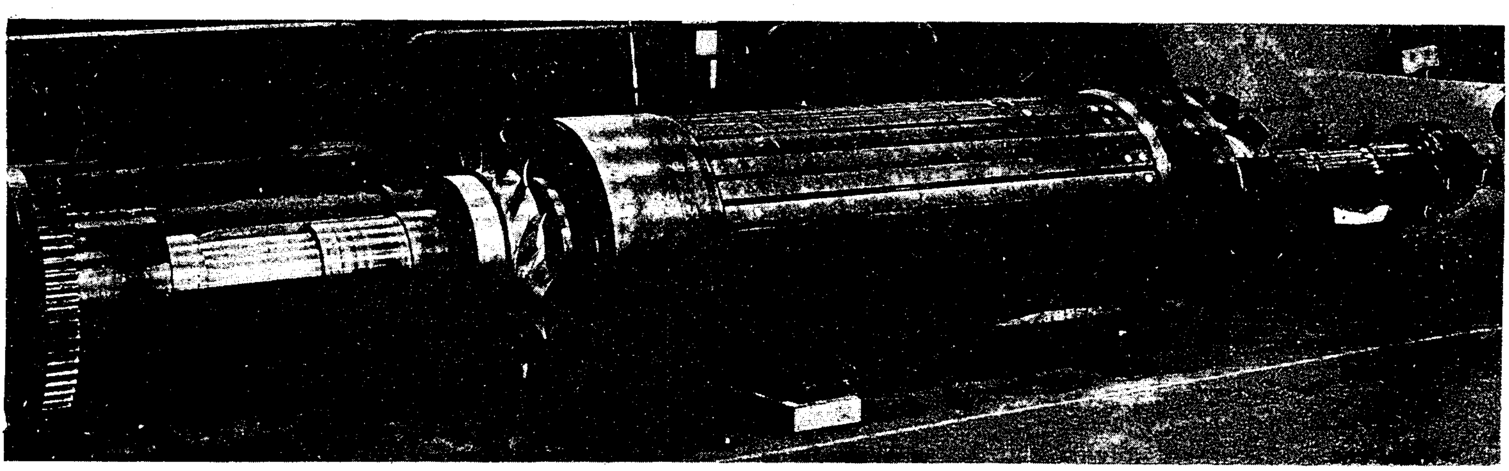


FIGURE 10. A 23,530 KVA. rotor ready for installation. Note the fans on each end and collector rings at the right hand end.

In figure 10 we have a typical rotor completely assembled. The rotor has to have a number of deep slots machined in it as shown in figure 11 to accommodate the field windings which will be carrying high currents. For a 120 MW unit the current carried would be in the region of 1300 to 1800 amperes d-c. The percentage of the rotor surface which is slotted, varies with the design but will usually fall between 68% and 74%. This slotting tends to weaken the rotor body mechanically. By careful design, however, the amount of copper required voltage and counteract armature reaction can be accommodated in the rotor without weakening it substantially.

FIGURE 11. Rotor showing slots for windings. Ventilation slots are below the main slots.

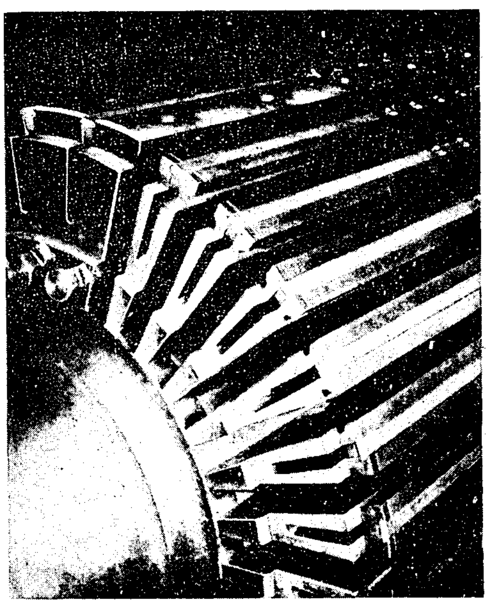


FIGURE 12. Rotor showing ventilating slots in the teeth between the main slots.

Rotor Body

Rotor bodies for large machines of 120 MW and over are normally made from a solid steel forging of high grade nickel chrome molybdenum steel which has a yield point of about 68,000 psi. The weight of the forging for a 120 MW machine will be about 60-70 tons, while the completed weight of the rotor when wound will be between 30-40 tons.

When determining the position of the slots, for the windings it is necessary to allow ample passages in the rotor through which the cooling medium can move freely, since correct cooling is one of the major factors in design. Two methods by which this can be done are considered. In figure 11 the ventilating slots are provided below the coils at the bottom of the slot while in figure 12, the ventilating slots are machined in the teeth. Along the length of the rotor a number of holes are drilled in the teeth to allow the cooling medium to flow out of the slots into the air gap. The cutting of slots in the teeth improves cooling of the iron to such an extent that it results in a 10% increase in output for the same temperature rise.

Winding of Rotor

The copper conductors for winding the two field coils for two-pole cylindrical rotors used on modern 3600 rpm. machines are usually made up of continuous copper strip formed into a number of oblong frames of suitable dimensions. These are placed in the various slots of the rotor, each frame forming a turn of the field coil. The rotor slots are lined with a trough made of insulating material. It consists of mica bonded to asbestos, glass fabric or other suitable material. As each layer of field coil is pressed into the appropriate slot in the rotor, micanite strip is fed in to insulate the layers from each other.

After the rotor is wound the coils are clamped down hard in the slots by heavy steel clamping rings and baked at a high temperature. When the rotor has cooled off, further insulation is inserted on top of the conductors. The conductors are then held down firmly in the slot by non-magnetic dovetail keep which are driven home.

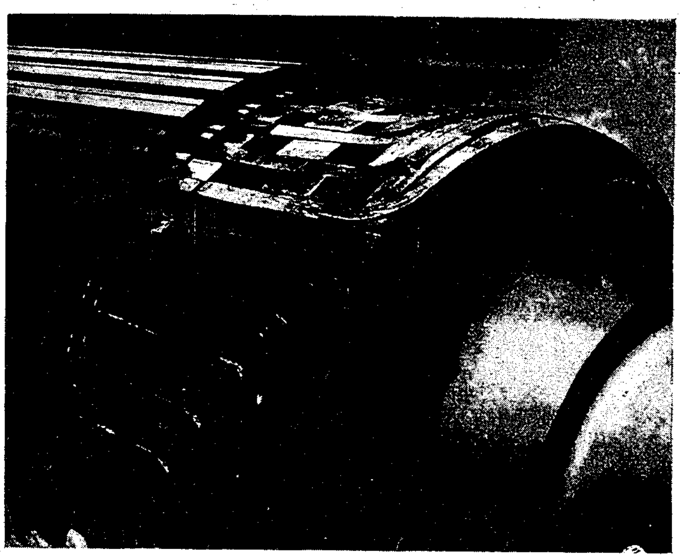


Figure 13 - View of rotor after windings installed. Treated asbestos spacers prevent movement of rotor end turns.

FIGURE 13

So that the end-turns are unable to move, treated asbestos spacers are wedged in between them as shown in figure 13. Over the end-turns is fitted a squirrel cage damper ring as shown in figure 14, which prevents all risk of arcing between end bells and rotor body in the event of short-circuits or unbalanced loading.

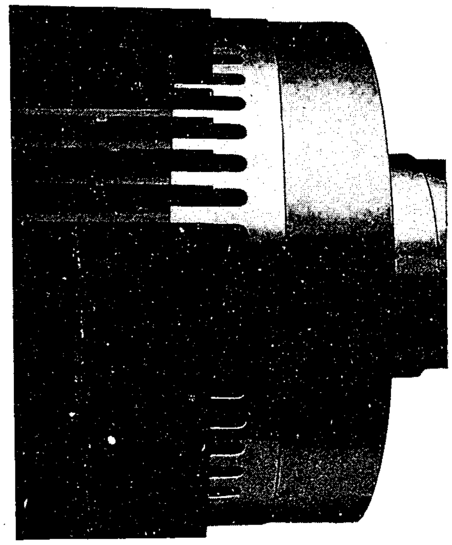


FIGURE 14
Damping ring.

Retaining Rings (or End Bells)

The coil retaining rings or sometimes also referred to as end bells serve to hold the rotor end turns securely in position. They prevent outward movement of the end-turns under the action of centrifugal force. Figure 15 shows a photograph of a typical end-turn retaining ring. Figure 16 shows a cross-section of rotor retaining ring and part of the shaft. The retaining ring is shrunk on to recesses turned in the ends of the rotor body. They are of massive construction and are usually of non-magnetic steel.

FIGURE 15. Typical end-turn retaining ring.

To avoid local concentrations of high stress, the rings are continuous: there are no sharp edges, and there are no holes for ventilating or locating purposes.

Coil retaining rings are the most highly stressed components of a rotor and it is because of these rings that the peripheral speed is not allowed to be above 550 ft./sec. They are usually made of nickel-chrome manganese steel. The non-magnetic properties of the rings reduce end winding flux leakage and this helps to keep the stray flux losses to a minimum.

Slip Rings

Each of the two ends of the loop making up the copper windings for the rotor is connected to a slip ring on the shaft. Figure 10 shows two slip rings at the right-hand end of the shaft. Slip rings are shrunk on to an insulated sleave of micanite attached to a steel bush pressed into the shaft. They are usually made of high grade steel.

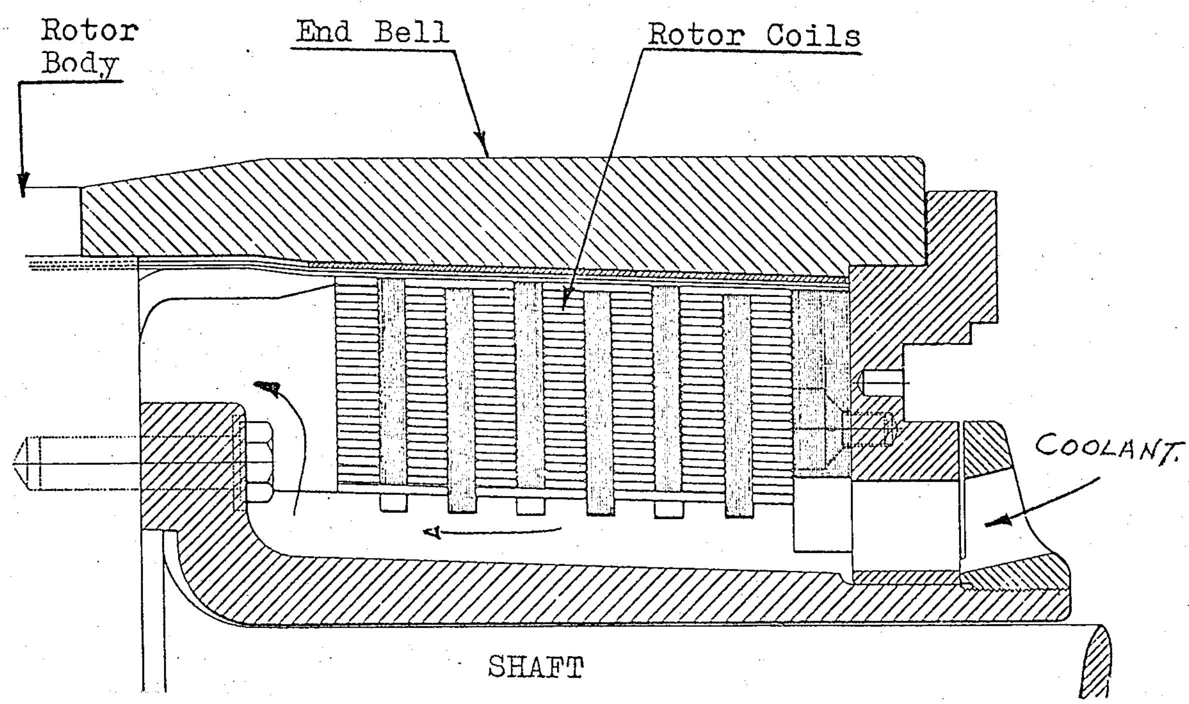


FIGURE 16. Cross-section of rotor retaining rings holding end-turns in place.

Carbon brushes rubbing on the slip rings as they rotate provide a connection for the d-c circuit from the exciter to the rotor windings.

Insulating Generator Bearing

Because of a certain amount of magnetic out-of-balance in the machine, a small voltage is induced in the rotor body which would be sufficient to cause a heavy current to flow in the circuit formed by the rotor journals, bearings and bedplate if they were allowed to be a closed metallic loop (a continuous circuit.).

The current would cause heavy pitting and corrosion of the bearings and journal surfaces. It is therefore prevented from flowing by insulating the outboard bearing and its oil pipes from the bed-plate.

Cooling

So far we have not mentioned anything about cooling the generator. This will be covered in detail in a later lesson.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-5 - Generator Construction

A - Assignment

1. Why is the stator core made of laminated sheets?
2. What property must the core steel have?
3. Describe the Roebel stranded bar. Of what advantage is it?
4. Draw a sketch showing the arrangement of windings for a three-phase stator.
5. What two things limit the size of a generator rotor?
6. What is the rotor body made of?
7. What is the purpose of rotor end-turn retaining rings? What limitations do they impose on the generator?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator and Auxiliaries

-6 - Generator cooling

0.0 INTRODUCTION

We have previously discussed to some extent how a generator is constructed. Now we are ready to describe the methods used to cool generators and this lesson will deal with this subject.

1.0 INFORMATION

The present day electric generator for a steam power station is a very efficient machine - approximately 98% efficient. 2% of the input from the turbine consists of losses which are wasted one way or another and appear as heat in various places of the generator. This does not appear to be very much until you consider that 2% of a 200,000 KW machine is equal to 4,000 KW. Since all of this 4,000 KW is converted to heat it is like putting a heater of this size right into the generator. You can well imagine that this would give off a tremendous amount of heat. (4000 KW x 3413 = 13,652,000 Btu/hr.)

Losses.

The total losses of 4000 KW in the generator mentioned above are made up roughly as follows:

Windage and friction losses	40%
Rotor losses	12%
Stator I^2R losses	12%
Stator iron losses	26%
Stray losses	10%
	100%

The above terms may not be familiar so they will be explained as follows:

Windage - can be considered air friction losses. The rotor is rotating at very high speed and air or gas offers resistance to this rotation.

Rotor losses are due to the resistance to electric current flowing along the copper windings. (I^2R losses)

Stator I^2R losses. The power dissipated in the form of heat in the copper conductors of the generator is equal to the square of the current I times the resistance R.

Stator Iron losses. In generator magnetic circuits the flux steadily alternates in direction. Iron seems to be magnetized by pointing many molecules of the core in the same direction. The continuous changing of position of these molecules seems to cause internal friction and heating. This form of iron loss is called Hysteresis. Another iron loss is in the form of eddy currents. As the flux changes in an iron core, it sweeps across the iron causing a relative movement. This induces an emf in the metallic parts (not the windings) and causes a current to circulate through the iron; this is an eddy current.

Since all the above losses contribute a tremendous amount of heat in the generator the designer has to think of a way of removing this heat. The design of the generator will depend to a very large extent on the method of cooling used. There are three methods of cooling a generator. These are listed as follows:

1. Air cooling for both rotor and stator.
   (a) in open machines
   (b) in enclosed or ducted machines

2. Hydrogen cooling for rotor and stator

3. Hydrogen cooled rotor and water cooled stator.

Air Cooling

Small generators are generally open type of machines. That is, fans on the rotor draw air from the surrounding atmosphere circulate it through the generator and discharge it again to the surrounding atmosphere.

It has the disadvantage that dirt and oil can coat the windings. This causes the insulation to deteriorate and also hinders conduction of heat through the insulation. This produces a danger of overheating and could result in fire. This type of machine is also quite noisy.

The diagram in figure 1 shows a simple closed air ventilation system. A fan on each end of the rotor draws cool air from the cooler into the generator, circulates it through the stator core and over the rotor windings, where it picks up heat. The air is then forced via a duct through a cooler below the generator. The same air is used over and over again.

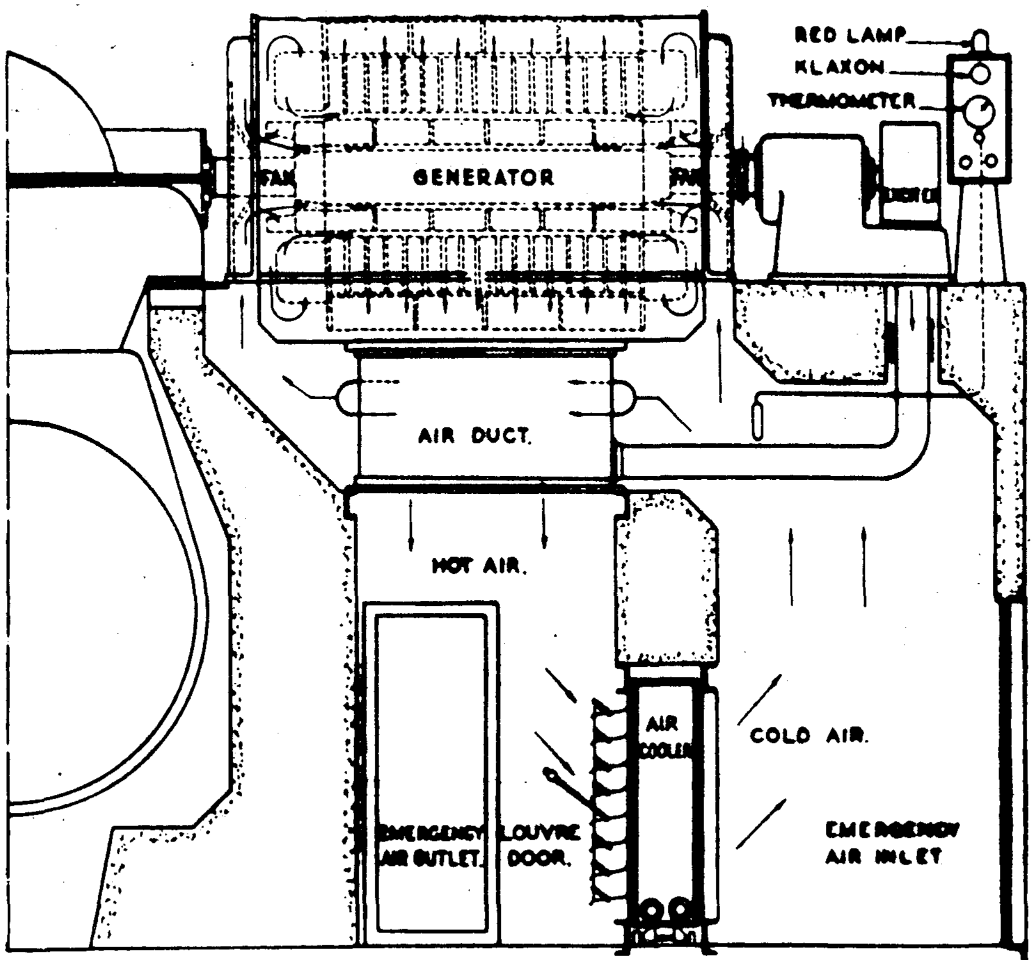


Figure 1.
Simple closed air ventilation system.

The advantages of this system are:

1. It keeps the air and windings clean.
2. It reduces the fire risk.
3. It cuts down the noise level.

This type of arrangement is used for generators with outputs up to 20 or 30 MW. Above this output hydrogen cooling is generally used nowadays

Hydrogen cooling

As we mentioned previously, one of the main factors contributing to the heating of generators is the windage loss. If a suitable cooling medium which has low density is used, the total heat generated can be reduced. Hydrogen is such a medium and it has been used for years. The characteristics of hydrogen compared to air as a cooling medium are shown in Table I. The windage losses of a rotor operating in hydrogen is only 10% of that in air. Hydrogen conducts heat approximately seven times more readily than air as seen from table 1. This allows the generator to develop about 25% more output for the same physical size.

TABLE I

Characteristics	Air	Pure Hydrogen Pressure in psi 0.5	30	45
Density	1	0.07	0.14	0.22
Thermal Conductivity	1	6.7	6.7	6.7
Heat Transfer Coefficient	1	1.55	2.7	3.6

Also an increase in hydrogen pressure (and hence an increase in density) means an increase in capacity to absorb heat. The following general rule applies in this respect:

(a) In the range of 0.5 to 15 psig - for every 1 psi, increase in H_2 pressure an increase of 1% in generator output can be expected.

(b) In the range of 15 to 30 psig - for every 1 psi. increase in H_2 pressure an increase of 1/2% in generator output can be expected.

Therefore comparing the same physical size a generator operating at 30 psig hydrogen pressure would have a 22% greater output than the generator operating at 0.5 psig hydrogen.

We can further deduce that for the same physical size a generator cooled with hydrogen at 30 psi would have 25% + 22% = 47% greater output than a generator cooled with air.

From the above discussion one can readily see that it is a great advantage to use hydrogen as a coolant in a generator. However, because of the extra manufacturing cost and additional equipment required for the hydrogen cooling it is not economical for units with output less than 30 MW.

Rotor Cooling with Hydrogen

When the size of unit exceeds 100 MW. use has to be made of the advantage of higher gas pressure in order to limit the size of machine. To take full advantage of hydrogen cooling, it is necessary to bring the cooling medium into direct contact with the copper conductor. This can be done in several ways using a special shape of conductor. Figure 2 shows a section of a typical direct cooled rotor. Notice the two hollow channels in each conductor. Fans located on each end of the rotor force the hydrogen through the conductors; with holes located in appropriate places the gas follows the paths indicated by the arrows. Figure 3 shows a section through two rotor winding slots and intermediate ventilation slot in the rotor tooth. The black spots are the ventilation holes. This arrangement allows direct cooling of both the copper windings and the rotor body. This means that heat does not have to flow through the insulation. Direct cooling as shown in figure 2 increases the kva that can be carried by the field ampere turns by about 1 1/2 to 2 times. The first units with this type of arrangement started operating around 1952. Most machines of 100 MW and over are designed with some form of direct cooling in the rotor.

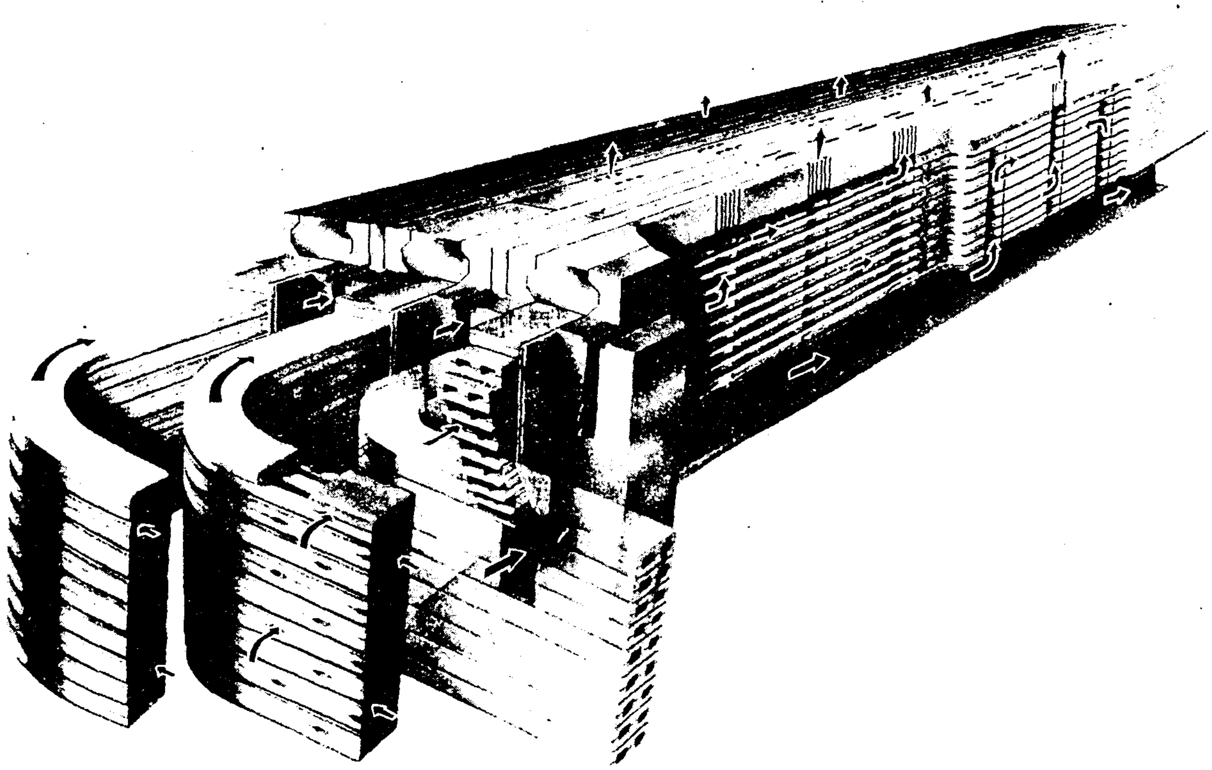


Figure 2. Direct hydrogen cooling system for the rotor showing network of interconnecting radial and axial ducts.

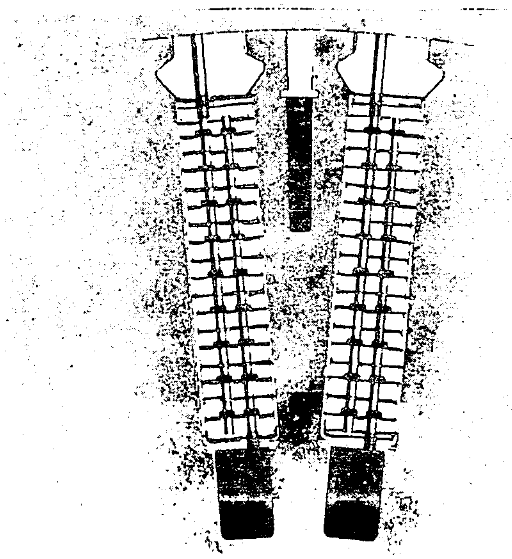


Figure 3. Section through two rotor winding slots and intermediate ventila--tion slot in the rotor tooth.

The hydrogen pressure is normally about 30 psig although some manufacturers recommend pressures up to 45 psig for the largest units.

Stator Cooling with Hydrogen

Since there is a relatively large gap between rotor and stator the two can be treated as separate parts of the machine as far as cooling is concerned. Stator cooling is also improved by higher H_2 pressure. Figure 4 shows the flow paths of the gas as it sneaks through the duct spaces between laminations of the stator core. Stator windings can also be cooled from the inside or direct cooling method by building up the windings with a very lightly insulated tube in the centre.

After the hydrogen has picked up heat it is passed through a cooler where its temperature is lowered and is then ready to be used over again. These coolers are located right in the generator casing rather than below the generator as is the case for air cooling.

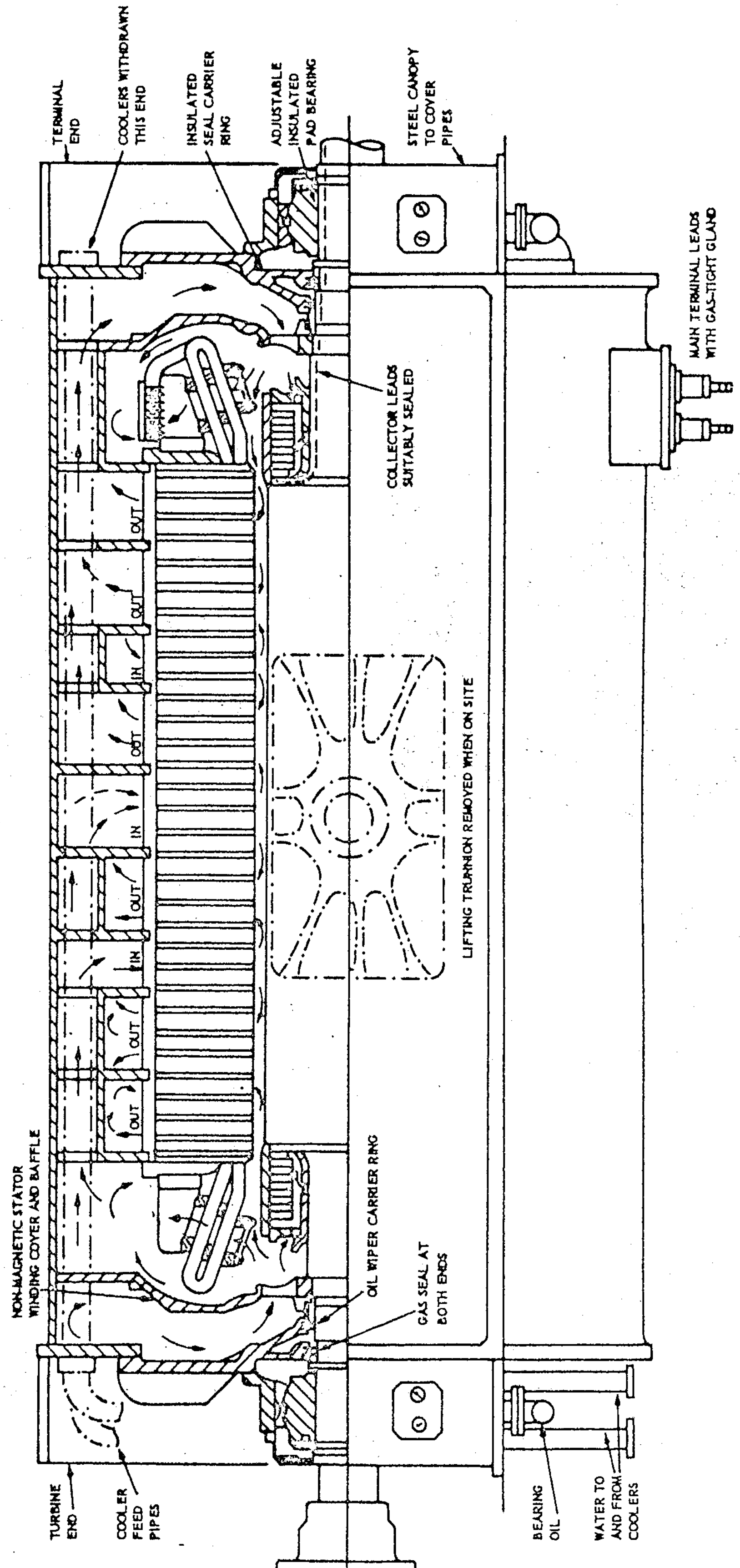


Figure 4. Section arrangement of a typical hydrogen - cooled machine

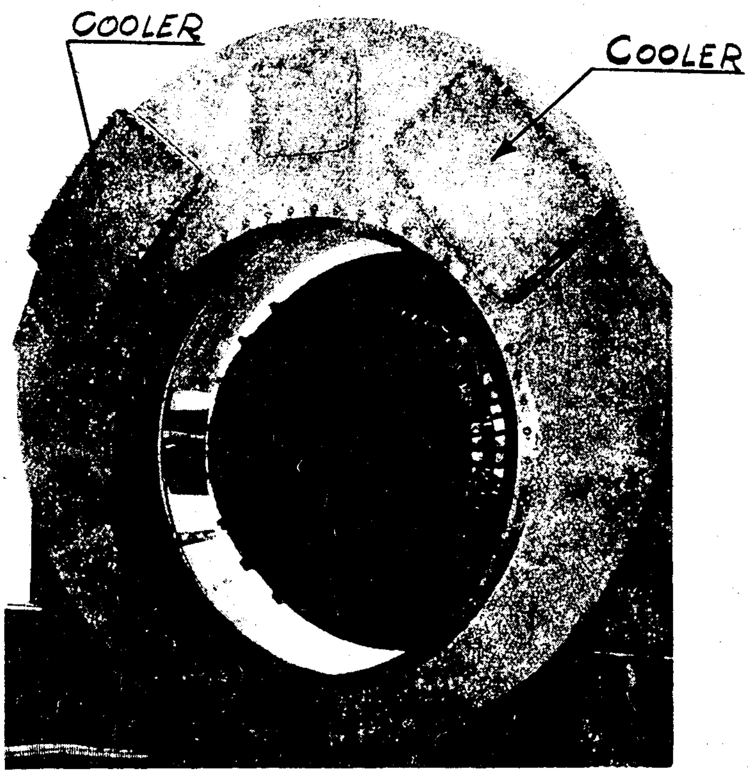


Figure 5. End view of a 60 MW generator stator showing two horizontal coolers in the top of the casing.

Figure 5 shows an end view of a 60 MW generator stator with two coolers at the top of the casing. These type of coolers are horizontal and run the full length of the generator. Generally demineralized water is circulated through them as a cooling medium.

Often four horizontal coolers are located in the top half of the casing rather than two. Another arrangement used nowadays is to have two vertical coolers built into the casing on either side of the generator. Or for larger units there may be two vertical coolers on either side of the generator.

We have now covered two of the three methods of cooling generators. Before we proceed to deal with liquid cooling of the stator windings let us summarize the advantages and disadvantages of using hydrogen as a cooling medium in generators:

Advantages.

1. Lower windage losses. Windage losses of rotor in H_2 is only 10% of those in air.
2. Higher thermal conductivity. This allows the generator to develop 25% more output for the same physical size as compared with air.
3. Longer life span of insulation, because of absence of oxygen and lower temperature gradient across the insulation.
4. Lower noise level.
5. Reduced fire hazard because hydrogen does not support combustion.

Disadvantages

1. Hydrogen can be explosive when mixed with air. Between 4.1% to 74.2% mixture by volume of air hydrogen is explosive. It is most explosive when there is 30% of hydrogen in air by volume. For this reason generator casings are made explosion proof so that they will be able to withstand 500 to 600 psig. which would be the internal pressure produced should an explosion occur. The purity of hydrogen in a generator should always be kept between 95 to 98%.
2. Danger of hydrogen leaking out to the atmosphere. This is minimized by the use of oil gland seals on the rotor shaft.

Water Cooling of Stator Conductors

Around 1955 to 1957 manufacturers started producing large units (200 MW and over) with direct liquid cooling of the stator windings.

Liquids cooling is the most efficient method of cooling. This is especially so when water is used. The pumping power required is only about 1/8th of the fan power used for hydrogen cooling. The relative advantages of water, oil and hydrogen cooling for stators are shown in figure 6.

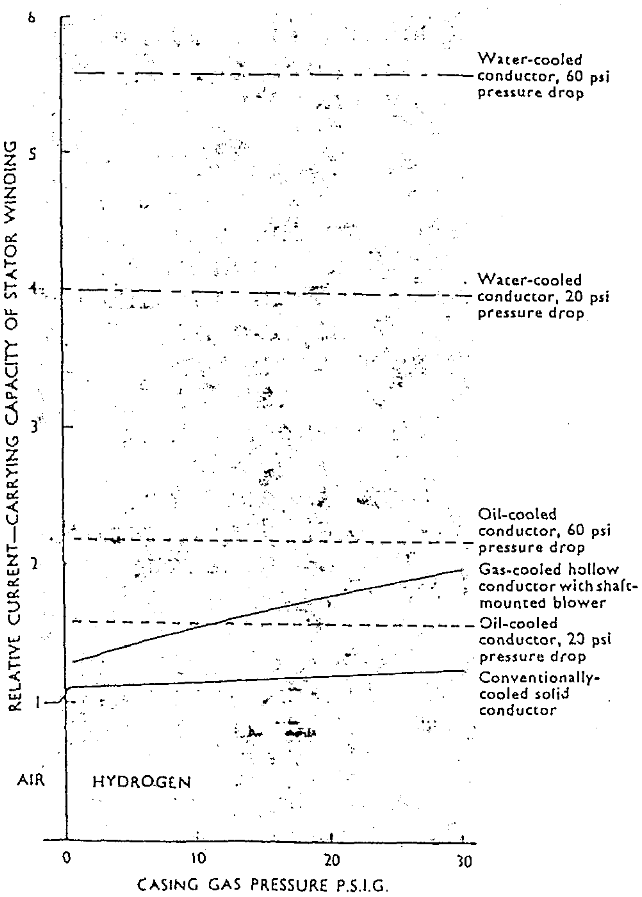


Figure 6. Relative current carrying capacities of stator windings occupying the same space and employing different methods of cooling. The conventional and gas-cooled hollow conductors are based upon a maximum actual temperature of 266°F and the liquid-cooled only 140°F.

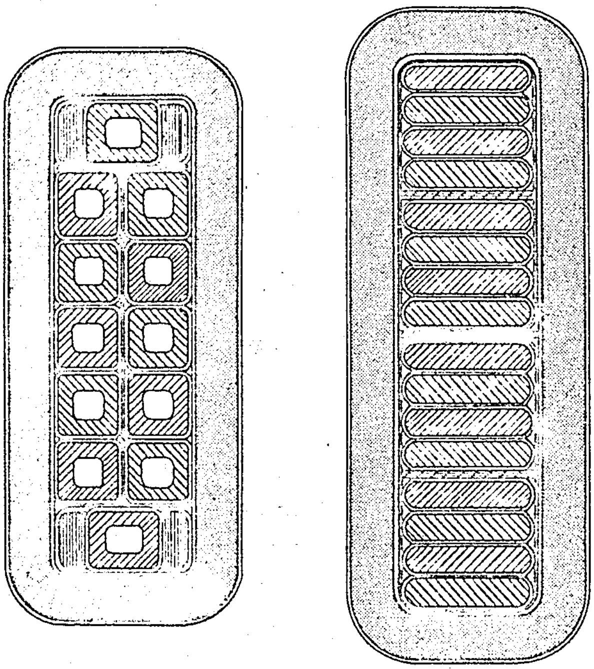


Figure 7. Cross-section of generator stator conductors with liquid cooling (left) and conventional cooling (right).

It can readily be seen that water cooling has a decided advantage over any of the other methods of cooling. Notice the difference in size of generator stator conductors (including insulation) for liquid cooling and conventional cooling. The liquid cooled conductors are hollow inside and the water runs through them.

With this improvement in cooling methods it has become possible to use much higher voltage and amperage for the same size winding without overheating. A voltage of 13.8 KV used to be considered high up to around 1957. Nowadays 18 KV is considered acceptable and there is talk of using 20 KV. A higher KVA rating is of course synonymous with greater generator output. It has been found that a frame normally used for a 120 MW generator can be made to have an output of 200 MW when using direct water cooling in the stator windings and direct hydrogen cooling in the rotor windings. This improvement in cooling methods in recent years has made it possible to build generators of 500 MW output. This means that a 500 MW turbine generator can be built as a single-shaft tandem compound unit.

Figure 8 shows a simplified diagram of a water circuit of a generator with a water-cooled stator. A distilled water pump, pumps the water through a cooler, then to a manifold within the generator casing.

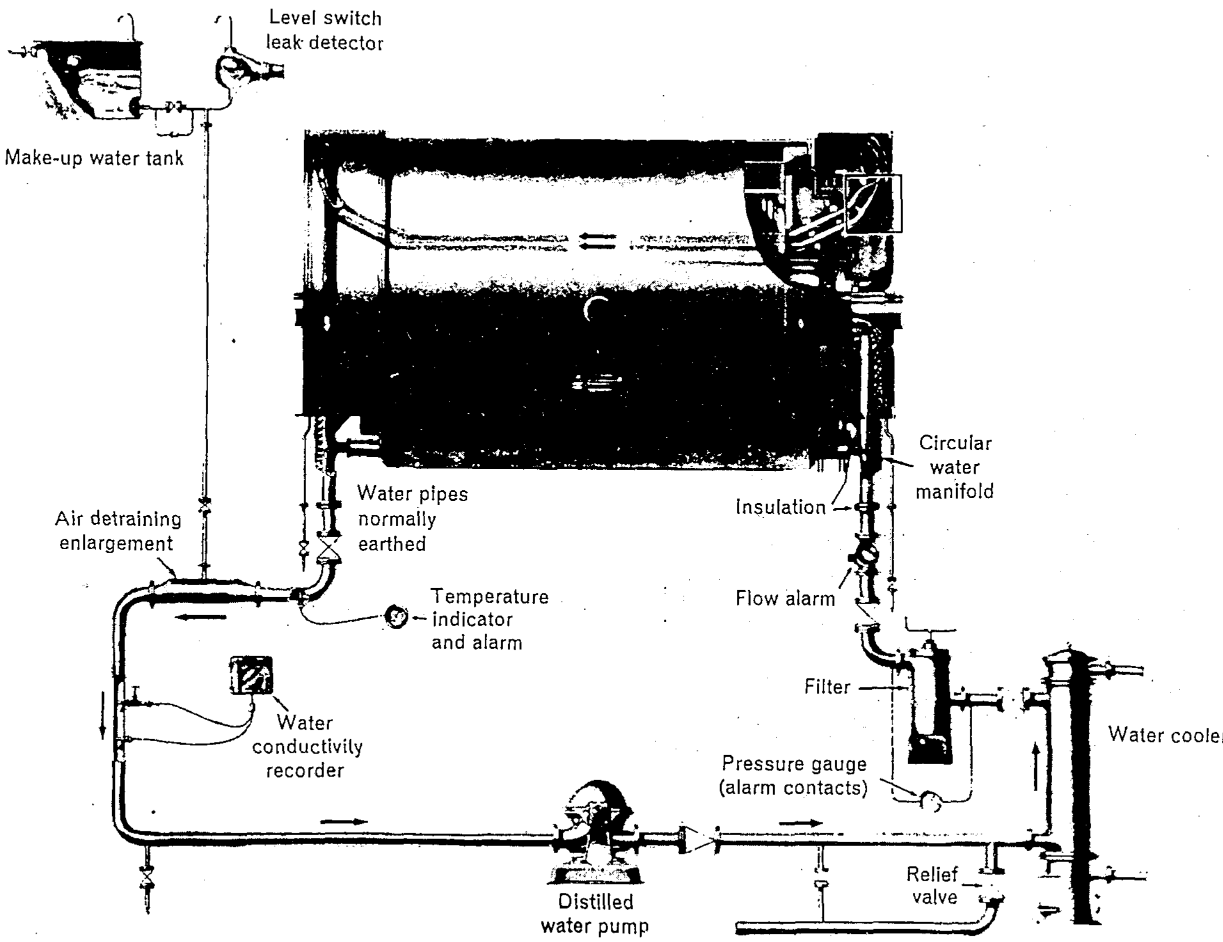
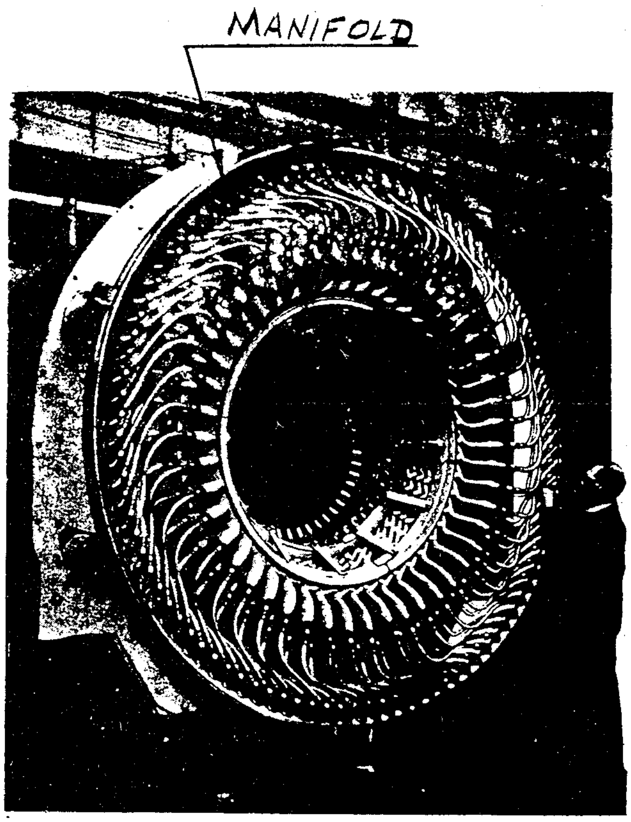


Figure 8. Simplified diagram of the water circuit of a generator with a water-cooled stator.

From the manifold the water is led via flexible hose to the hollow copper stator windings. After flowing through the windings as the arrows indicate, it collects in another manifold at the other end. From there it returns to the suction side of the pump. There is generally a standy pump with a reliable power supply, and a standby cooler in this type of a system.

Figure 9 shows an end view of a generator with manifold and hose assemblies in position. This one is made by a particular manufacturer. Other manufacturers have slightly differing arrangements.

In figure 10 is shown a section of an end connection between the liquid-cooled stator coil and the hose assembly which leads to the manifold.



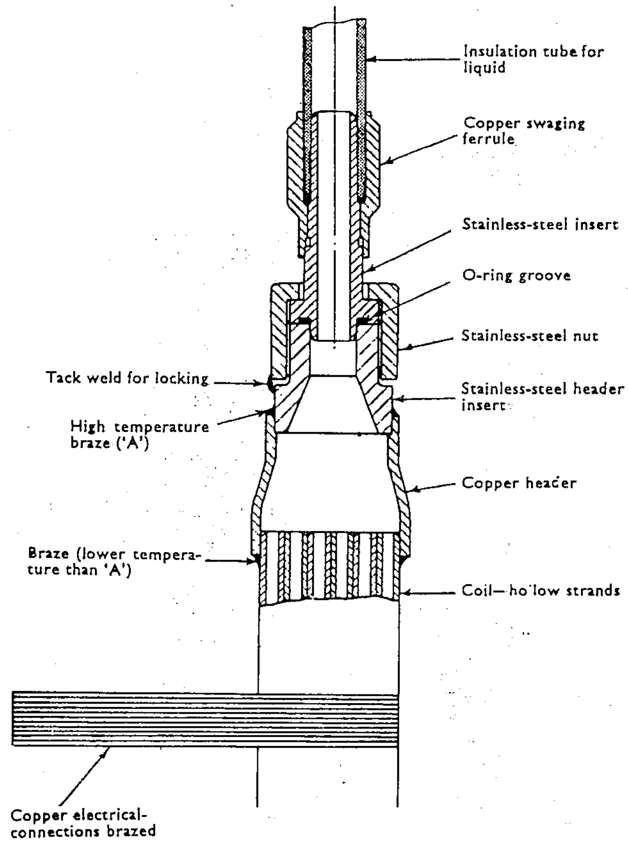


Fig. 9. End view of generator showing manifold and hose assemblies in position

Fig. 10. Section showing end connections of the liquid-cooled stator coil to the hose assembly which leads to the manifold.

Figure 11 depicts a cross-section through two water cooled stator coils. All the insulating material used for the coils. is labelled.

Water is normally not considered a good insulator. However, the water used for this purpose is condensate from the closed feed cycle and as such has been demineralized. Thus it is a very poor conductor of electricity so that there is very little loss due to electrical conduction along the water columns to ground.

As you can well imagine the hose assemblies connecting the manifold to the individual windings must be of a special type. They must be impervious to water, chemically inert and good electrical insulators. They must also be able to withstand the temperatures likely to be experienced under the most severe conditions of operation. The material that one manufacturer uses for the hoses is polytetrafluoroethylene (P.T.F.E.).

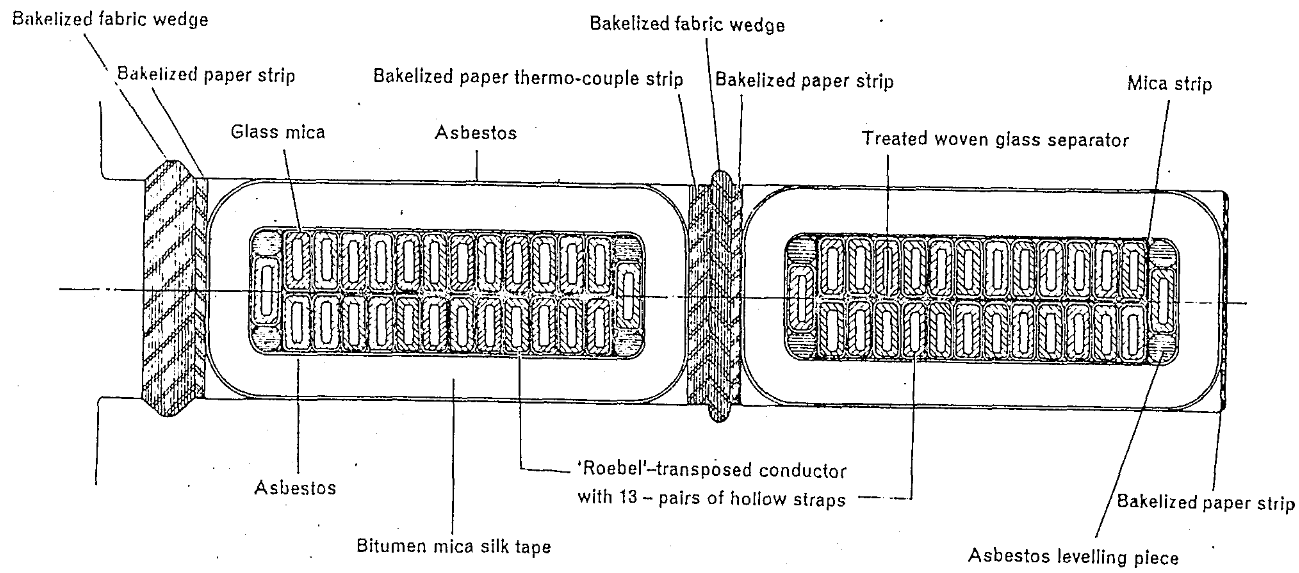


Figure 11. Cross section through two water cooled stator coils. Notice also the material used.

General Shaft Seals

We have mentioned previously that hydrogen pressure inside the generator may be as much as 30 to 45 psig. This means that there has to be a very good seal between the generator shaft and frame, otherwise hydrogen could leak out and form an explosive mixture with air.

Figure 12 shows two different types of shaft seals that can be used. Notice the location of the journal bearing in each case. It is generally built as an integral part of the generator and cannot be seen when the generator end covers are on. The shaft seal is between the rotor and the bearing. Seal oil is supplied to a middle chamber in which are also located the sealing rings which float on the shaft. The seal oil which has to be at a pressure higher than the generator hydrogen pressure, seeps between the shaft and sealing rings and flows to both the right and left thus filling the only clearance gap between generator frame and shaft. A chamber on each side of the middle chamber collects the oil and allows it to drain away at the bottom. The oil flowing to the hydrogen side will absorb some hydrogen. Therefore the drainage from this side has to be led to a detraining tank where hydrogen is separated from the oil and after this it is safe to be drained back to the oil tank.

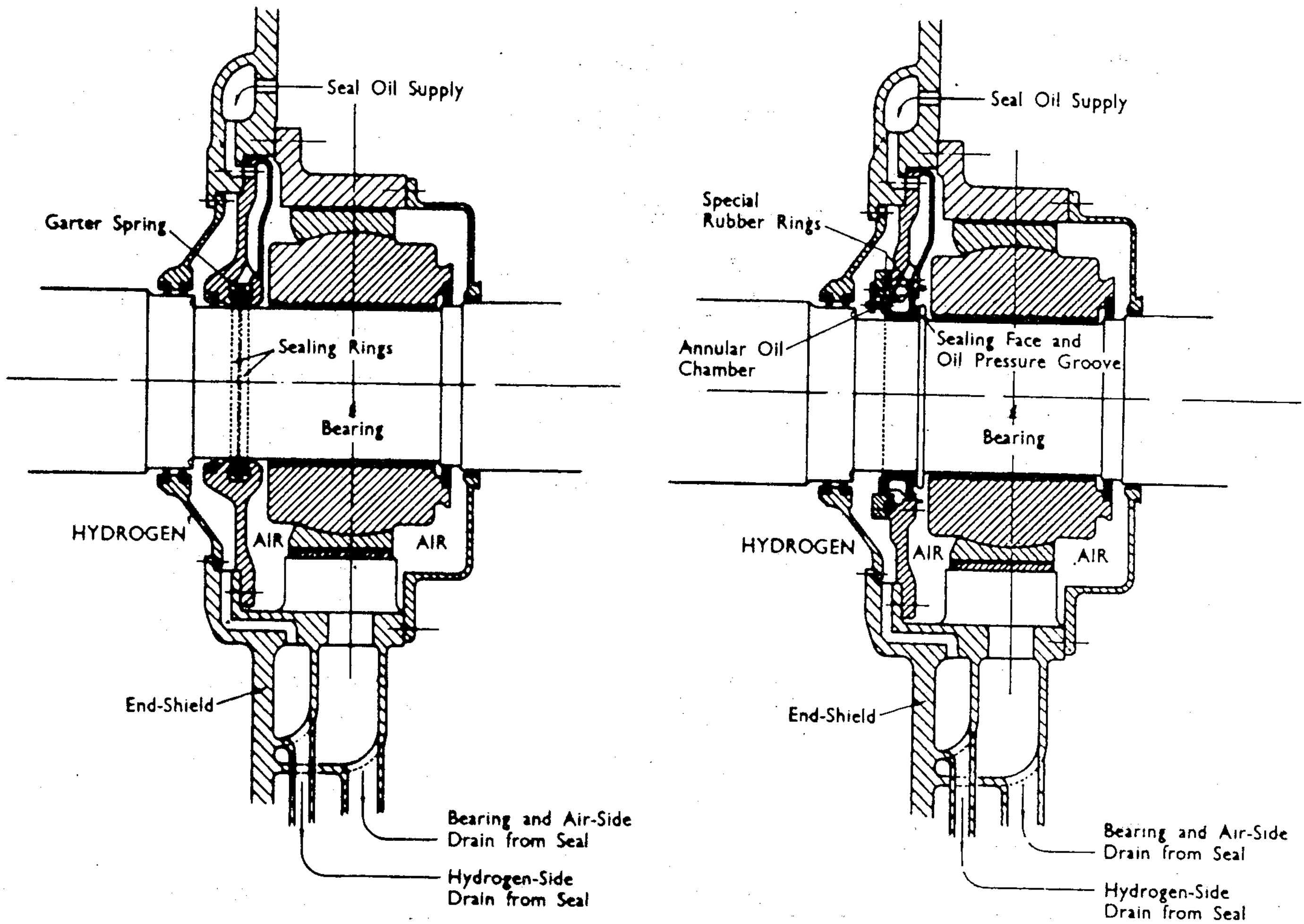


Figure 12. Typical generator shaft sealing arrangements.

The drainage from the air-side can safely be led back directly to the oil tank.

The oil supply in some cases comes directly from the main lubricating oil system, while in other machines the sealing oil supply comes from a completely separate system. However, this sealing oil has to be supplied as long as there is hydrogen in the generator even when the unit is shut down. Therefore a unit that uses sealing oil from the main lubricating oil system during normal operation would still have to have an independent sealing pump and cooler to supply sealing oil during shutdown conditions.

In cases where the oil from the main lubricating system is used the main oil tank generally has a ventilating fan installed on top to keep the tank at a slight vacuum. This fan discharges to the outside of the station. This prevents any hydrogen pockets from forming anywhere in the oil system.

During operation of the generator it is of utmost importance that sealing oil is supplied at all times. If there should be a sealing oil failure, the sealing rings would quickly wear away because of lack of lubrication and cooling, leaving a gap with dangerous possibilities of hydrogen leakage. In such a case the hydrogen would have to be purged out of the generator as quickly as possible.

Temperature Measurement

Thermocouples are generally placed in between the windings and also the core to measure metal and gas temperatures that exist at various places in the generator. The temperature of the rotor of course cannot be measured in this way so what is done is to measure the temperature of the gas emitted from the ventilation holes. Knowing the conductivity of various materials in the rotor a rough estimate can be made as to copper winding and rotor temperature.

D. Dueck.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-6 - Generator Cooling

A - Assignment

1. Name 5 kinds of losses one can expect in a generator.
2. Name the methods of generator cooling that can be used.
3. What is the advantage of increasing the hydrogen pressure in a generator?
4. What is meant by direct cooling?
5. List at least four advantages of using hydrogen cooling in a generator.
6. Under what type of conditions is hydrogen dangerous?
7. Of what advantage is it to use direct water cooling in the stator windings?
8. Why is the loss of generator sealing oil supply a very serious situation?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-7 - Pumps

0.0 INTRODUCTION

In our discussions about the regenerative feedheating system we have mentioned condensate extraction pumps, booster pumps and boiler feed pumps several times. This lesson will describe these pumps in greater detail. It is assumed here that the student has studied the lesson on "Pumps--Centrifugal and Rotary" in the Mechanical course T.T.4 level.

1.0 INFORMATION

Condensate Extraction Pumps

We have said previously that the condensate extraction pumps take water from the hotwell of the condenser and discharge it through the L.P. feedheaters to the deaerator. We have also said that the condenser is at near perfect vacuum on the steam side having approximately 1" Hg. absolute pressure.

Condensate extraction pumps are specially designed to handle condensate at the boiling point corresponding to the high vacuum with only a small positive pressure on the suction side of the pump. They are generally of the centrifugal type in the hotwell and located near the condenser 6 to 8 ft. below the level of condensate.

In modern steam power stations either two 100% duty pumps or three 50% duty pumps are provided per unit and operate in parallel. These can be two-or three-stage pumps.

The most important factor in extraction pump design is the avoidance of the entry of oxygen into the condensate system. This can be prevented in a number of ways and these are discussed below.

A two-stage design is shown in figure 1 and it will be noted that the gland at the left-hand side, adjacent to the first impeller, is under vacuum when the pump is in service. The gland is therefore, sealed by leading a condensate supply to it from the second-stage discharge pipe. If one of the pumps is acting as a standby, it will be standing idle but its piping is also connected to the condenser, and therefore air could get in through the standby pump glands.

Therefore, sealing water must also be supplied to the glands of the standby pump. It will be appreciated that this results in a complicated small bore piping system together with the necessary operational requirements of changing over the sealing connections when the standby pump is brought into service.

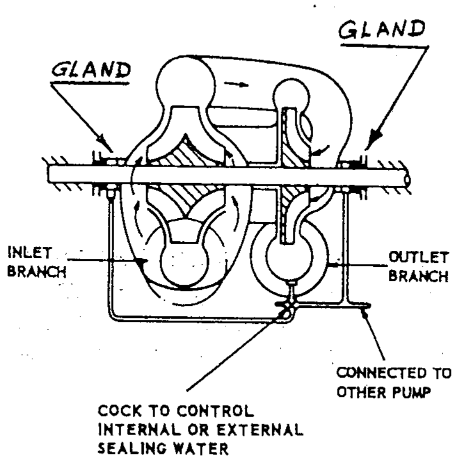


FIGURE 1. Two-stage extraction pump.

This method of gland sealing is not ideal because the sealing water, being at a relatively high pressure, is drawn into the pump and can under certain conditions, draw in air, against the flow of water leaking from the gland. This appears to be caused by the leakage water being rotated by the action of the shaft when the pump is in use, thus forming a helix. The air then leaks into the pump between the water streams forming the helix.

A partial solution is to fit an internal bearing at the suction end of the pump as illustrated in figure 2. This bearing must be water lubricated and may, therefore be damaged by suspended matter. Most damage will occur on a vertical spindle arrangement, after the overhaul period. Some manufacturers fit a bearing drain connection so that the bearing can be flushed out, but this feature merely minimizes, rather than avoids the possibility of damage. Probably the best arrangement, where the hydraulic characteristics permit it, is to mount the first stage impeller on the shaft between two impellers, running in parrallel on a two-stage-design as shown in figure 3, or in series for the three-stage-design. This arrangement is slightly more expensive but it does ensure positive pressure on the pump glands.

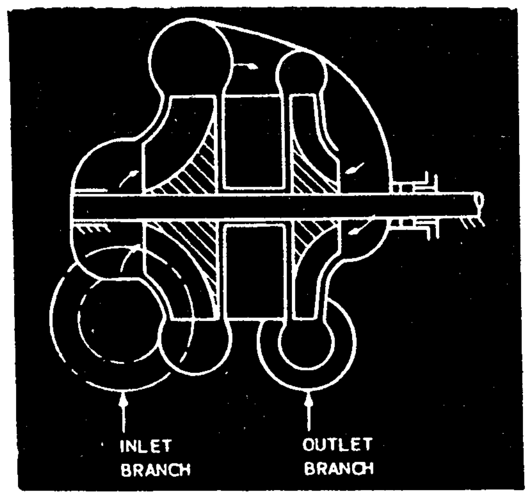


FIGURE 2. Arrangement of two-stage extraction pump with internal bearing.

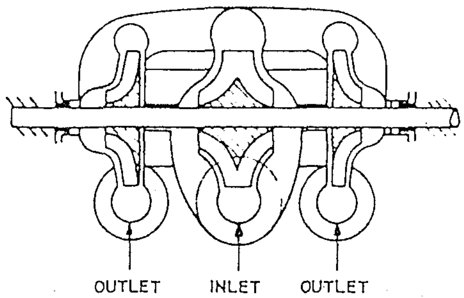


FIGURE 3

Figure 3 - Arrangement of two-stage extraction pump.

Boiler Feed Pumps (B.F.P.)

The feed pump is the most important component of the feed system from the point of view of plant safety. It must be reliable because modern boiler has only a small water capacity in relation to the rate of evaporation. The complete stoppage of feed water supply rapidly reduces the drum level and a period of two minutes is typical for the interval between sounding of the boiler low water-level-alarm and the boiler tubes starting to be empty of water. This results in overheating of the tubes and perhaps ultimate failure. (In nuclear stations boilers hold even less water, however, the temperatures are not as high.) It is necessary therefore to provide standby pumping plant and at the same time desirable to achieve optimum economy of capital and running costs. The method and extend to which this is done depends upon a combination of factors which vary in value from one layout to another and upon the economic factor.

In order to provide standby feed pump capacity there are different arrangements that can be used. Either three 50% capacity pumps or two 100% capacity pumps can be used. One pump is thus always on standby. However, boiler feed pumps have become fairly reliable and in some designs it is now the practice to have two 50% capacity pumps with one 10% capacity pump as standby and for use during starting up and shutting down.

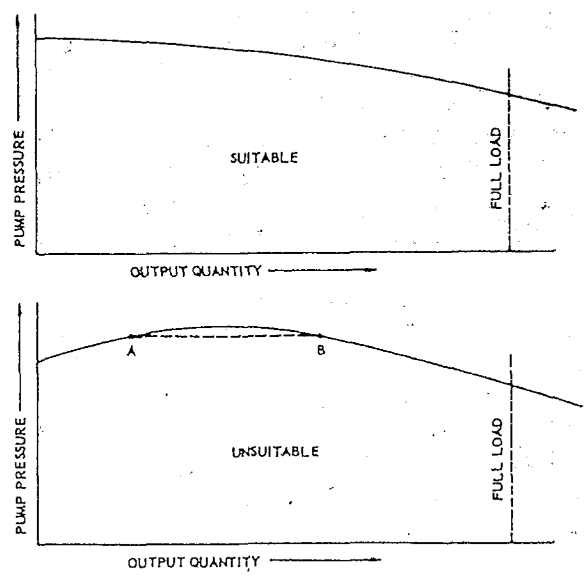


FIGURE 4. Feed pump pressure/ output characteristics.

Parallel Operation

To enable pumps to run in parallel without surging each must have a pressure characteristic which falls continuously from no load to full load. Figure 4 shows both suitable and unsuitable pressure/output characteristic curves. The lower curve shows two discharge rates A and B, at which the pressure is the same. If this is the required pressure and the boiler requires an intermediate quantity, the pump may surge between A and B with subsequent damage to the pump and piping. For stable operation of pumps in parallel as would be the case for two 50% capacity pumps, the steadily falling characteristic shown by the upper curve is necessary. If the pumps have similar characteristic curves they will share the load equally.

Position of Pump in Feed Train

For highest pumping efficiency the pump should be placed where it can receive water at the lowest possible temperature to take advantage of the greater density. A pump is a volumetric displacement machine; the higher the temperature, the greater the volume and hence the more work the pump will have to do. Unfortunately, where the feed pump operates at a low temperature, more of the feed pump operates at a low temperature, more of the feedheating system is subject to high feed pressure and this is expensive in both initial cost and maintenance cost. Economically it is necessary to adopt a compromise position for the pump. The modern requirement of high level deaeration fixes the pump position immediately after the deaerator, where the temperature at full load is 220 to 275°F. As we said before, there is a need to keep the feed pressure in excess of the saturation pressure at all points of the system so that there is no boiling and consequent vapor lock.

The use of two-stage pumping with heaters in between as we mentioned in the lesson on "Regenerative Feedheating System", removes the need for heating surface at high feed pressures. Below 900 psi it is not worthwhile to use the booster pump (or two-stage) arrangement, however, above this pressure it can be advantageous. Figure 5 shows an arrangement of booster and main feed pumps, each with a 100% duty standby. This system is for a 120 MW unit operating at 1500 psi, and 1000°F.

On the latest large modern units the main 100% duty feed pump is driven from the generator shaft through a gear box and fluid coupling. This effects a considerable financial saving in the case of motors, electrical wiring and electrical operating costs. For start-up and standby, however, conventional pumps are necessary. A typical layout of this is shown in figure 6.

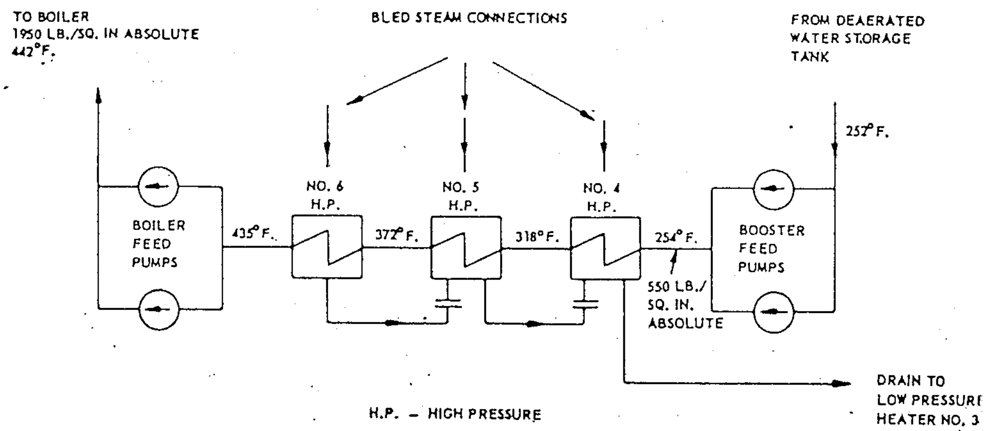


FIGURE 5. Arrangement of booster and boiler feed pumps.

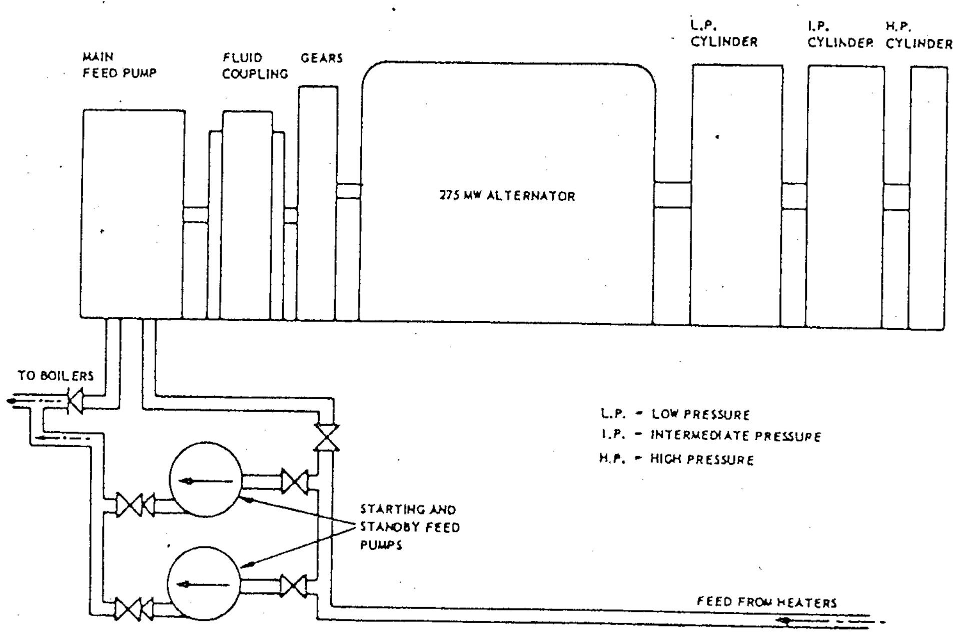


FIGURE 6. Unit feed pump arrangement.

Suction Conditions

If the pump is supplied with water near boiling point, then the pump may become vapour-locked and its efficiency will be impaired, because of pumping both vapor and liquid. To avoid this, it is necessary to ensure that the water reaching the pump suction is at a sufficiently high pressure in relation to its temperature. The full flow deaerator/heater must be raised to a suitable height and pressure to achieve this, as we mentioned in a previous lesson. An automatic valve controlling water from the storage tank, is provided both to bypass the deaerator storage tank and to deliver water direct to the feed pump suction. The piping between deaerator and pump suction should be large and direct to assist in avoiding suction troubles under changing load conditions. A typical suction pressure is between 36 and 40 psi.

Capacity

100% pumping duty for the feedwater must exceed the guaranteed turbine continuous maximum rating steam rate to provide cover for poor vacuum, make-up, sudden boiler demands and pump deterioration. It is a characteristic of centrifugal pumps that reduced pumping efficiency owing to erosion and wear, has a greater effect on the quantity delivered than on the discharge pressure. It is uneconomical to provide extra capacity for sustained boiler blow-off.

An excess capacity of 20% above turbine continuous maximum rating steam rate is regarded as adequate.

The horsepower required to drive these pumps can become quite enormous for large units. Take for example for a 200 MW unit, using 565 psi. steam at turbine inlet, the horsepower required to drive a 100% duty feedpump is 2300 H.P.; for a 500 MW unit using 2400 psi. inlet steam it is 16,500 H.P.; for a 500 MW unit using 3500 psi. it is 23,000 H.P. When water is raised to such high pressures it becomes hotter due to compression. Some of this power then goes into heating the feedwater and its temperature may be raised as much as 50 to 70°F while going through the B.F.P.

As we have mentioned previously this power can either be supplied by electric motor, auxiliary steam turbine or directly from the main turbine shaft.

As far as discharge pressure is concerned it would be very desirable to have it directly related to the turbine stop-valve pressure. However, each plant requires individual design particularly in respect of the boiler pressure, the height of the boiler drum above the feed pump, and the friction of piping and valves. All these valves could add up to around 300 psi. depending on the arrangement so if the stop valve pressure at the turbine is required to be 2400 psi. then the B.F.P. would have to deliver feedwater at 2700 psi.

Gland Sealing

On modern high pressure feed pump it is essential to have an effective gland seal to prevent ingress of air to and egress of water from the pump via the shaft. This is normally effected by some form of water-cooled stuffing box, sealed by water at booster pump discharge pressure. This bypasses the high pressure heater train, and is therefore at a higher pressure than feed pumpsuction. On the discharge side of the pump high internal pressure is suitably reduced by fine clearance sleaves, a balance valve and a leak-off is passed to some suitable point in the system, such as the dearator storage tank, and the drain water is returned to the condenser via the clean drains system.

Operation

At low loads the water in the B.F.P. may just be churning around without going anywhere and this could result in overheating of the pump. To avoid this an arrangement is generally made so that if the pump is operating below 20% of capacity the feedwater is automatically recirculated back to the deaerator. This then establishes a minimum quantity of flow through the pump, sufficient to keep it from overheating.

In larger units the normal operating temperature of the feedwater at the B.F.P. may be as much as 485°F. The pump supports which rest on a concrete foundation would naturally also be at this temperature. In order to aboid weakening of the concrete due to this temperature, the B.F.P. foundation often has cooling tubes through which process water is circulated during operation.

In the case of large pumps operating with very small clearances, there is always danger of bearings overheating. Cooling of these bearings is handled by the lubricating oil which is passed through a cooler located at the B.F.P.

A suitable speed control from low load to full load can result in quite a saving in operating the B.F.P. This is generally done in one of two ways: 1) variable speed motor or 2) fluid coupling.

B.F.P.--Types of Design

Boiler feed pumps are multi-stage machines and can be classified into the following categories:

1) Axially split casing.

2) Radially split double casing.

Figure 7 shows a five stage axially split casing pump--i.e. a single casing with a horizontal flange. Figure 8 shows the impellers for this type of pump. The advantage of this arrangement is that the inlet and outlet connections need not be disturbed when opening up the pump. This pump can be used up to 1600 psi. Above this there is danger of the flange leaking.

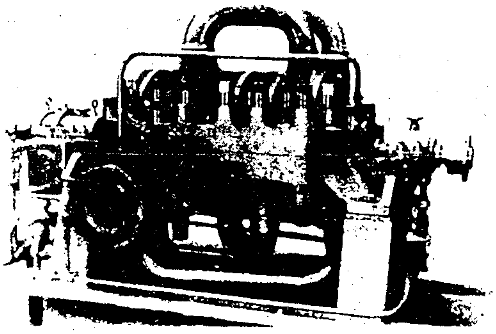


FIGURE 7. Axially split easing volute pump.

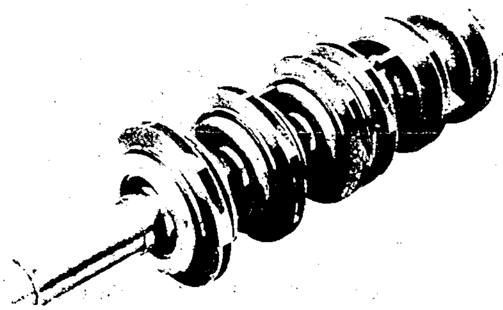


FIGURE 8. Rotor of six-stage opposed-impeller axially split casing pump. Stage-pieces are assembled on the shaft between the impellers.

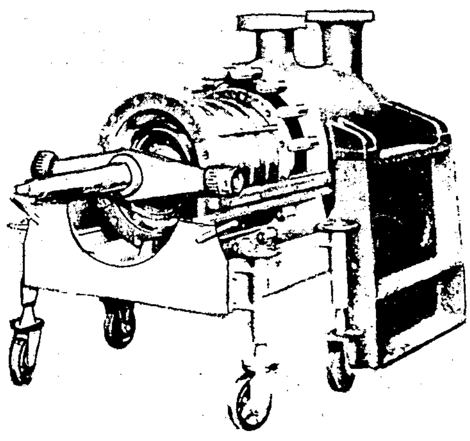


FIGURE 9. Rotor assembly of radially split double-casing pump being inserted into its outer casing.

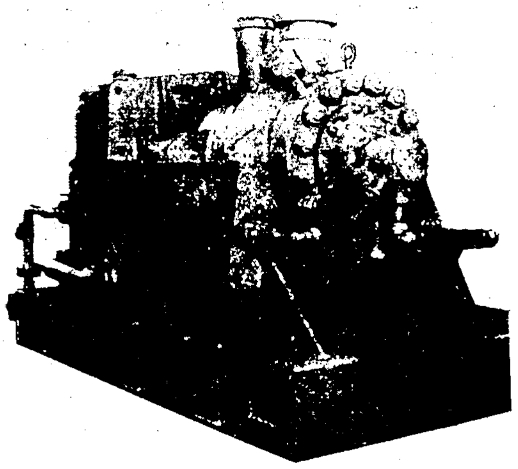


FIGURE 10. Multistage radially split double-casing pump.

Figures 9 and 10 show a radially split pump with an inner and outer casing. The inner casing contains the shaft, impellers and volutes which can be slipped out of the outer casing by undoing the flange bolts on one end. In this way maintenance can be done without having to cut off the inlet and outlet connections. This type of boiler feed pump can be used up to and above 3,500 psi.

The radially split double-casing design can also be made with an inner casing which is axially split as shown in figure 11.

Of course normally in a power station these pumps would be covered with a few inches of insulation so it would be fairly difficult to tell from the outside as to which type you are looking at.

Balancing for Thrust

For high pressure pumps axial thrust becomes quite an important consideration as far as design is concerned. There are several different arrangements which can be used to balance the axial thrust.

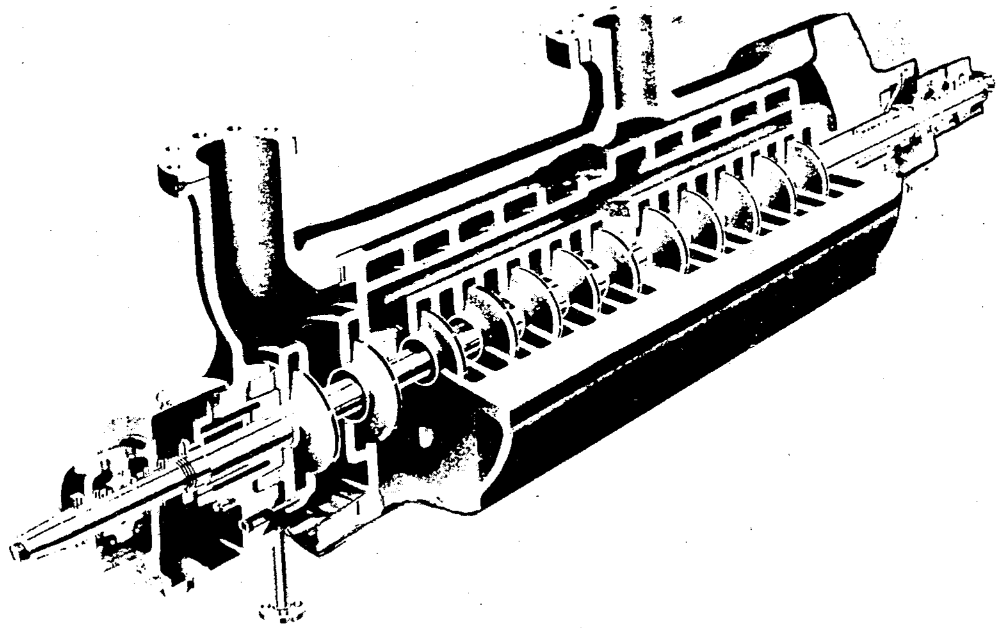


FIGURE 11. Double-casing multistage pump with axially split inner casing.

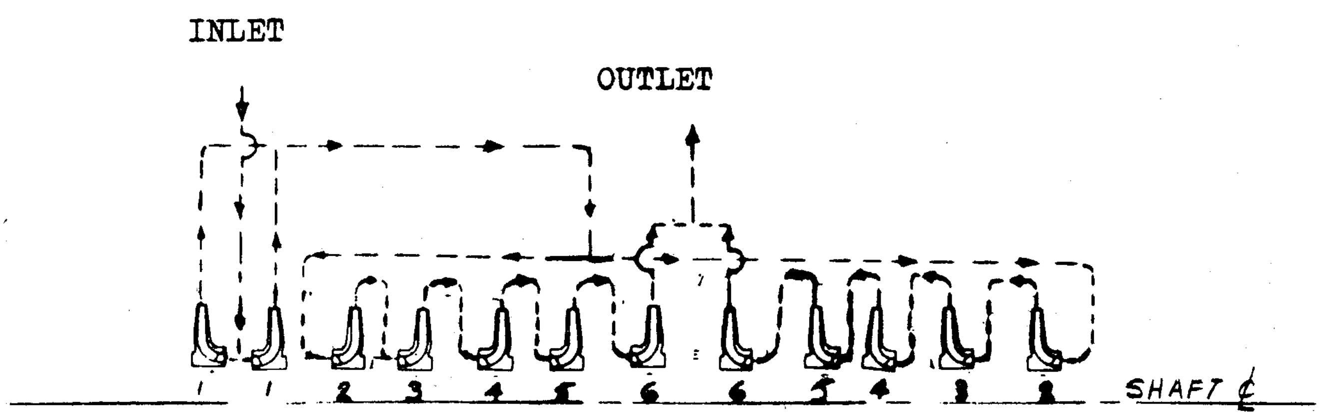


FIGURE 12. Diagram showing the impeller arrangement and flow resulting in axial balancing of thrust for the pump shown in figure 11.

These arrangements are:

1. Impellers with an even number of stages may be divided into two opposing groups.

2. Individual stages may be balanced by a balancing chamber on the back of each stage.

3. Double-suction impellers may be used on all stages.

4. A balancing device, such as the automatic balancing disc and a balancing drum, may be used on the discharge end of the pump.

Summary

This then is a general description of the condensate extraction and boiler feed pumps--a very vital part of the regenerative feed-heating system.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

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

4 - Turbine, Generator & Auxiliaries

-7 - Pumps

A - Assignment

1. What is the most important factor in extraction pump design?
2. Why do condensate extraction pumps have to be of special design?
3. Why is the boiler feed pump such an important component as far as plant safety is concerned?
4. Why is the boiler feed pump sometimes placed after the H.P. heaters in the feedheating cycle for large units?
5. Name 3 ways a B.F.P. can be driven.
6. What is meant by recirculation as far as the B.F.P. is concerned?

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4 - Turbine, Generator and Auxiliaries

-8 - Feedwater Treatment

0.0 INTRODUCTION

In the course on Common Processes and Services the lesson on "Make up Water", T.T.4 level describes the treatment of raw water taken from a river or lake, to be used in various systems in a steam power station including the feedwater heating system. It mentions that harmful impurities such as magnesium clacium, bicarbonate, carbonate, chloride and silicate are removed from the water. If further mentions that the method used for removing these impurities is by chemical treatment, coagulation, filtration and deionozation. After the raw water has been treated it is stored in huge reserve tanks. When make-up water is required in the feedwater system it is bled from these reserve tanks into the condenser or into the feedwater piping somewhere between the L.P. heaters and the boiler feed pumps.

However, the water that is already in the feedwater system picks up various kinds of impurities as it circulates within the cycle and these impurities are rendered harmless by appropriate treatment. The latter type of feedwater treatment will be described in this lesson.

1.0 INFORMATION

The increase in operating pressures and temperatures in recent years has increased the significance of corrosion of feed line metals, including both the steam and water sides of heaters, feed pump components, and in particular, high pressure heaters. The nature of the attack has varied with the plant design the metals used for construction, the purity of feed water (with particular reference to dissolved gases and hydrogen ion concentration value), and working temperatures and pressure. In all probability corrosion of feed line metals is taking place slowly in all existing stations and will continue to take place in the future. The corrosion is probably general in distribution and is occurring at such a slow rate that no serious consequences will materialize within the lifetime of the plant.

The dissolution of metals into feed water under power station operating conditions is known to be influenced, amongst other things, by dissolved oxygen, carbon dioxide, gaseous oxides of sulphur, and ammonia, although the precise significance of combinations of some or all of these is not yet known, however, that at proposed working conditions in the future only extremely small concentrations of dissolved oxygen and carbon dioxide will be tolerable if corrosion of metals is to be kept within reasonable limits. The results of feed line corrosion may be placed in two categories:

(1) The effect of metal wastage upon the feed line components.

(2) The effect of corrosion products including metals and metal oxides, when transported into the boiler plant.

The metal content of feed waters at progressive points in the line has been measured at many stations, and from this research it is clear that the largest significant pick-up of metal on the water side takes place in high pressure heaters. On the steam side of heaters experiences vary considerably, depending upon the operating conditions, the design of drain and venting systems, and the materials of construction used. When metals dissolve in feed water they are precipitated from solution on meeting conditioning chemicals in the boiler. They may be deposited as metal oxides either at selective points, or in a random manner. In either case they may become potential danger points, either by interfering with heat transfer locally and thus causing overheating of tube metal or by taking some part in corrosive processes in the boiler. In many cases copper is found deposited in metallic form and it is thus free to act as a cathodic area should corrosive processes be at work. Whilst there is little or no conclusive evidence that copper, thus deposited, is the sole instigator of corrosive attack, both copper and its oxide may well play secondary roles by assisting or accelerating corrosion already under way. To avoid troubles from both the effects it is desirable to reduce the pick-up of metals by feed water to a minimum. In recent years committees on Feed Water Standards have considered the setting of standards for feed and boiler water for units operating at 1,500 and 2,350 lb/sq. in. The reports of these committees have stressed the need for minimising metal pick-up by feed water, and has recommended the continuance of research in this field. The report also recommended that the concentration of iron and copper in feed water should, together, not exceed 0.01 parts per million.

In the future the problem may be approached in two ways:

(1) By developing a metal or alloy that is not subject to attack, or by coating metals with a resistant film so that the feed water and metal are not in direct contact.

(2) By chemical treatment of feed water in such a manner as to render it innocuous to metals.

Feed Line Materials

The ideal material to be used for construction would be a metal or alloy which possesses the necessary characteristics for working, in manufacture and assembly, together with a high degree of corrosion resistance and physical strength at the operating temperature and including qualities which give it maximum resistance to metallic "creep". In the absence of this ideal, progress towards this end is being made, partly based upon the results of practical experience derived from observations made on operating plant and partly by metallurgical research. This work is essentially a protracted process which is complicated in practice by the number of variables involved, for example, by concentrations of dissolved carbon dioxide and oxygen and other gases, and also by the purely analytical difficulties encountered in measuring extremely small quantities of impurities found in feed waters. Research can only be carried out by covering systematically a wide range of stations, types of plant, metals, and waters. In work of this kind it is very easy to draw incorrect conclusions through the neglect or ignorance of additional factors.

Materials for new plant are selected in the light of the latest knowledge of their behaviour. The actual choice has to be tempered by cost, and naturally the cheapest metal or alloy which is believed to be capable of satisfactory service, is used. Thus, in low pressure heaters cast iron and copper or brass are used for the shell and the tubes respectively. In high pressure heaters, where conditions enhance corrosion rates and creep becomes more significant, it is necessary to use steel shells and, for tubes cupro-nickel alloys which, though more expensive, have more appropriate physical properties. The introduction of 0.5 to 2.0 per cent iron into 70/30 cupro-nickel has been found to improve its properties. Higher iron content gives maximum erosion resistance, and lower iron content gives maximum corrosion resistance. Mild and stainless steels have also been tried and thought has also been given to the use of bimetaltubes. The latter, though expensive, could be made to combine the best physical properties of an outer tube with a more corrosion-resistant sleeve of a second metal or alloy.

Attack by cavitation, erosion, and corrosion is also minimised in pumps and pipework by the judicious selection of materials in accordance with the chemical and mechanical conditions prevailing.

Chemical Treatment

Because of their volatility, gaseous impurities may either circulate through the system and produce effects in the dissolved state on the water side of feed heaters, or pass over from a boiler and enter the steam side of the heaters with bled steam from the turbine. In the latter case, what happens to the gases next depends upon the physical design of the heater, the contact time with the condensed steam, the drains and venting arrangments, and the distribution of the gas between the liquid and vapour phases in the heater concerned. This is goverened by the "distribution coefficient" of the gas concerned. As a result of this, in practice, different gases tend to concentrate at a particular point in the system, for example, it is well-known that ammonia concentrates in ejector drains, and sulphur oxides have been known to concentrate in the high pressure heater steam side. The number of variables involved makes accurate knowledge extremely difficult to obtain, though certain basic facts are known, namely:

(1) The rate of corrosive attack in general is increased with increasing temperature.

(2) Metal corrosion proceeds more rapidly with waters of low pH value than with those of higher alkalinity.

(3) A reduction in metal pick-up by feed water can be achieved by reducing the concentration of dissolved gases by either mechanical or chemical means.

Ideally, a feed water should contain no dissolved gases and should be alkaline. In practice, this can only be achieved mechanically, by extremely efficient venting of gases from condensers and drain boxes, and by ensuring that feed pump glands and other possible points of oxygen ingress are designed so that no dissolved gases are admitted. It is also necessary to include in the feed line a direct contact heater/de-aerator through which all water supplied to the feed pumps must pass. The effects of the last traces of dissolved gases may than be countered, and the pH value of the water elevated, by the addition of a chemical or chemicals.

In plant operating at low and medium pressure it is possible to use solid reagents such as caustic soda for pH elevation, and sodium sulphite for removing oxygen traces. With modern boilers, however, the high rates of evaporation would cause such a rapid concentration of solids in the boiler water that excessive blowdown would be required. Also, at modern operating temperatures, sodium sulphite decomposes, giving gaseous compounds of sulphur. The use of solid reagents under modern conditions is, therefore ruled out and attention is being turned to the use of volatile chemicals which will remain in the system without increasing the dissolved solid matter. Amongst these are hydrazine, morpholine, cyclohexylamine, octadecylamine and ammonia, and some brief comments on these are warranted.

Hydrazine

This compound reacts with dissolved oxygen in alkaline solutions. Any excess of the substance decomposes in the boiler and gives ammonia, which is not considered detrimental in the complete absence of dissolved oxygen. The use of hydrazine has been found to produce a considerable reduction in metal pick-up by feed water. This chemical is also used in many instances to afford protection to standing boilers by injecting into the boiler water during shutdown.

Cyclohexylamine and Morpholine

These substances are volatile amines and have been found to reduce the pick-up of metals and elevate the pH value of the water. They have the advantage of being more stable than hydrazine and, therefore, passing over in vapour form with the steam to the turbine, thus affording protection throughout the cycle. The effects of these and many other volatile amines are being investigated at present, and ultimately it is hoped that an amine may be developed, which will combine the qualities of being sufficiently volatile to circulate through the steam system without causing excessive losses of the material to occur at the ejectors and vents, yet which is sufficiently soluble to dissolve quickly in condensate and sufficiently stable enough to avoid excessive breakdown when exposed to superheat temperatures. As mentioned earlier gaseous impurities tend to concentrate at points in the system, and it may be possible in the future to select a corrosion preventative which has similar concentrating tendencies, so that the protective chemical is concentrated at the danger point.

Ammonia

The injection of ammonia into the feed line has been practised in many European stations for some time with beneficial results. It has been satisfactorily demonstrated in the United Kingdom that ammonia is not detrimental in the absence of dissolved oxygen and carbon dioxide. At one station a considerable reduction in metal pick-up was effected by reducing the dissolved oxygen and carbon dioxide, the ammonia in the feed line remaining constant. When more information about the mechanism of metal dissolution has been obtained, it should be possible to say whether the use of ammonia for pH elevation can be justified in normal operation.

Octadeylamine

One other approach to the prevention of corrosion of feed line metals has been investigated. This is the use of octadecylamine, a compound which has the property of adhering to metal surfaces, thus preventing their wetting by water. Consequently, this and similar compounds are known as filming amines since in effect they form a protective film between water and metals. The use of such compounds is limited, but good results in reducing metal pick-up have been obtained in some experiments.

Future Trends

The future aim will be to eliminate completely all trace of dissolved gases from feed water by mechanical means. Alloys and metals will also be developed, which have the highest degree of corrosion resistance. If, however, this should not be possible, the use of protective chemical treatment will inevitably have to be improved and extended.

D. Dueck

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

4 - Turbine, Generator and Auxiliaries

-8 - Feedwater Treatment

A - Assignment

1. Name 4 gases that may be dissolved in feedwater.
2. Name two undesirable affects that corrosion may have in a closed cycle.
3. Explain in a sentence or two the reasons for injecting each one of the following volatile chemicals into the feed system:

   (1) Hydrazine

   (2) Cyclohexylamine & morpholine

   (3) Ammonia

   (4) Octadecylamine

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-9 - Fire Protection

0.0 INTRODUCTION

There are a number of areas around the turbine and generator where fires could occur and for this reason a fire protection system is required. This lesson will describe in a general way some of the fire hazards and the methods used for protection. It is assumed that a more detailed description of the fire protection system will be given in the courses describing the systems for each particular nuclear station.

1.0 INFORMATION

Turbine Protection

Because of the large quantities of oil pipes around the turbine which supply bearing lubricating oil, power oil and governing oil there is always the danger of oil leakage. If this oil leaks on to high temperature steam piping it could very easily ignite and therefore there is always danger of fire around a turbine.

Hence the turbine oil system for each unit is normally protected by a wet pipe automatic sprinkler system equiped with an alarm check valve. The sprinkler system is so designed as to be able to extinguish fires originating around the equipment listed below and on the floors, platforms and steel support columns in the immediate spill areas:

main oil reservoir.
main oil pump.
bearing oil piping.
relay oil piping.
seal oil piping.
governor.
governor valves.
emergency stop valves.
interupt valves.
reheat emergency stop valves.
hydrogen control panel.
hydrogen detraining tank.
emergency seal oil pump (dc).
seal oil filter.
seal oil cooler.
seal oil pump (ac).
main oil coolers.
auxiliary oil pump.
emergency flushing oil pump (dc).
oil heater.
oil purifier.
jacking oil pump.
oil booster pump.

Sprinkler spacing is generally such as to provide unobstructed water impingement on all hazards and spill areas. A common high level tier of sprinklers is normally provided over the entire hazard area and additional tiers of sprinklers at lower elevations is provided to sprinkle hazards which are shielded from high level sprinkler discharge due to obstructions. In general a horizontal sprinkler spacing of from 8 to 10 feet is desirable with vertical spacing of tiers dependent upon the elevation of obstructions.

The sprinkler heads are generally thermostatically controlled. The sprinkler heads are set to open at temperatures of around 135°F to 165°F. However, under machine platforms and for other high ambient temperature areas 280°F sprinkler heads may be used.

Sprinkler heads are normally of standard 1/2" diameter orifice type and arranged so that the spray is emitted at a 120° angle.

Generator Protection

For an air cooled generator there is generally a CO_2 fire protection system provided. The system is installed to give fast automatic protection to the generator and exciter and to restrict damage to the faulted area.

Carbon dioxide is used since water on the generator windings presents the following problems:

a) drying-out after repairs is extensive if water is used whereas with CO_2 no clean-up is involved.

b) a complex hi-potting test must be carried out.

c) water is corrosive.

d) water spray cannot be made automatic because of the danger of accidental tripping. CO_2 will cause the least interference to normal operation of the electrical equipment.

For a hydrogen cooled generator the only fire protection provided is the wet pipe automatic sprinkler system on the outside in the vicinity of the bearings, and seal oil system. No CO_2 protection is provided inside the generator because hydrogen does not support combustion and therefore the inside of the generator is not considered a serious enough fire hazard to warrant this type of protection.

D. Dueck

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-9 - Fire Protection

A - Assignment

1. What is the main reason for fire hazard in the area of a turbine generator?

2. What type of a system is normally used for turbine fire protection?

3. Why is CO_2 used as fire protection in an air cooled generator?