Turbine, Generator & Auxiliaries - Course 234

TURBINE CONSTRUCTION

The last twenty years have witnessed a rather remarkable increase in the size of saturated steam nuclear turbines. This increase has been accomplished without any major technological breakthroughs. The ability to progress from a 25 MW turbine generator to an 800 MW turbine has been accomplished not through the invention of anything comparable to the jet engine or the transistor but rather through a natural extension of existing practice and knowledge. This fact greatly simplifies the study of turbine construction because the fundamental principles underlying the early nuclear steam turbine are virtually identical to those underlying the units being erected today. For anyone with a detailed understanding of the construction of any of the Ontario Hydro nuclear turbines, it requires very little insight to understand the construction of the other units. This lesson will concentrate on the construction of the large turbine generator units which are installed at Pickering NGS and Bruce NGS as they are typical of the current generation of saturated steam turbines.

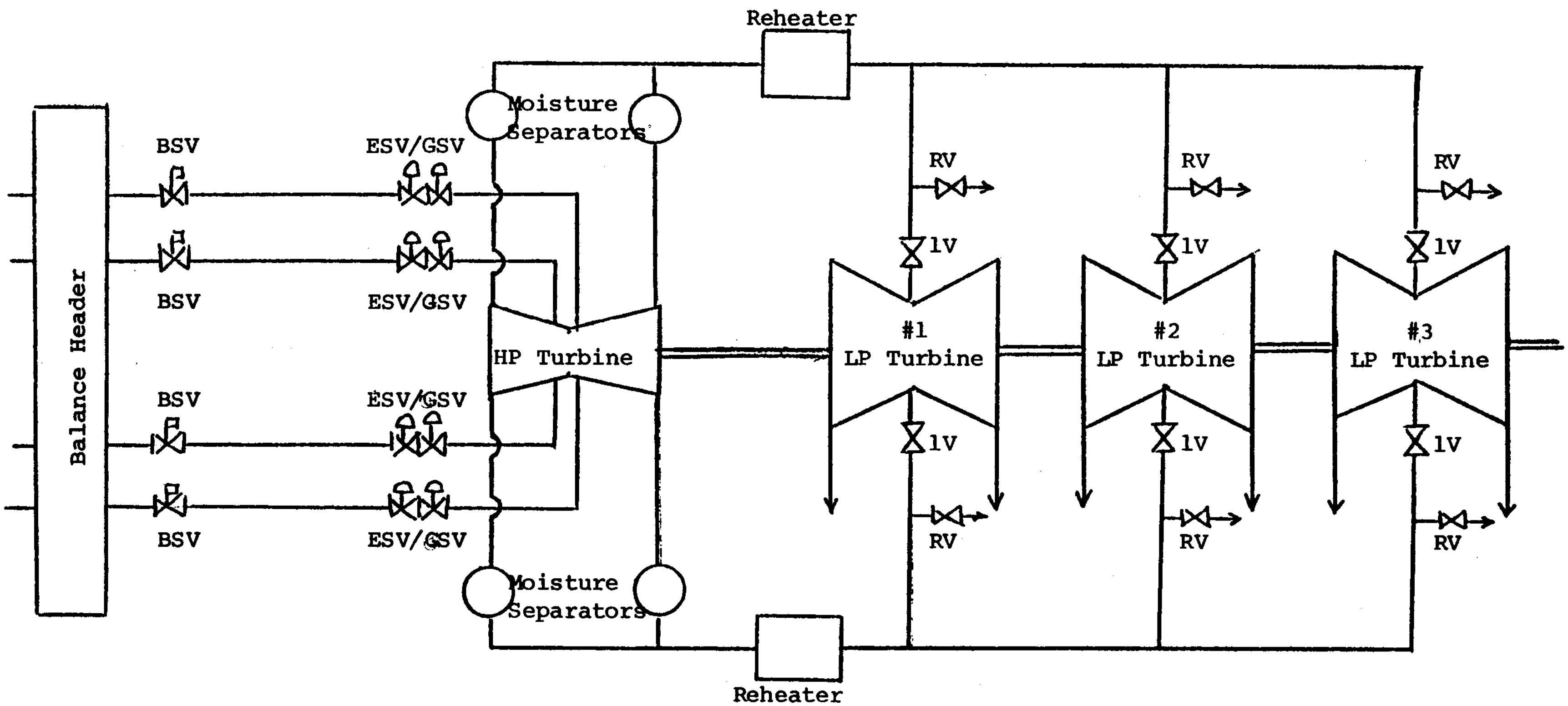
Figure 1.1 shows the steam system and turbine unit typical of a large nuclear generating station. Steam passes through a double flow high pressure turbine and exhausts through four moisture separators and two reheaters to three low pressure turbines in parallel. The four turbines are tandem compounded (all the turbines and the generator are on a common shaft). Steam admission to the high pressure turbine is through four steam lines each containing an isolating valve, emergency stop valve and governor steam valve. Each low pressure turbine receives steam from two low pressure steam lines. Each line contains an intercept valve which shuts off steam to the low pressure turbine in the event of a turbine trip.

The basic design of the turbine unit and, for that matter, all steam turbine units is determined by how the particular unit handles the fundamental problems listed below:

1. Moisture content of the steam;

2. Expansion of steam from inlet of HP turbine to outlet of LP turbine;

3. Axial thrust produced from the pressure drop through the turbine;



Typical Large Turbine Unit

Figure 1.1

4. Expansion of turbine casing and rotor during heat up and cool down;

5. Thermal stresses in casing and rotor during heat up and cool down.

We will examine each of these problems in detail and discuss the various design features which are intended to eliminate or minimize them.

Moisture Content of Steam

The steam which is produced by the steam generators of a CANDU nuclear generating station is saturated steam (steam containing no moisture but still at the boiling temperature). The steam cannot be superheated above the boiling temperature because metallurgical limits within the reactor will not allow a heat transport system temperature appreciably hotter than that required to produce saturated steam at a boiler pressure of 4000 kPa.

Because saturated steam is at the boiling point, as soon as heat energy is extracted from the steam in the high pressure turbine some of the steam condenses into water droplets which are carried along with the high velocity steam. This water strikes the fixed and moving blading with an effect quite similar to sand blasting. If the moisture content of the wet steam is much above 10%, the effect of the erosion is so severe that the blades are seriously damaged in too short a time to be economically attractive. In addition to erosion, the condensation of steam within the turbines generates large quantities of water which must be drained from the casing. If this were not done the water would collect in the bottom of the casing causing erosion, corrosion and rubbing.

The presence of moisture in the steam passing through a turbine has a pronounced effect on turbine efficiency. Since the water droplets cannot move as fast as either the steam or the blades, the back side of the moving blades are continuously running into the water droplets. This results in a retarding force being exerted against the blades and decreases turbine efficiency. Typically turbine efficiency is decreased by 1% for each 1% average moisture in the steam passing through the turbine.

In order to prevent excessive erosion and maintain acceptable turbine efficiency, the maximum moisture content of the steam in a turbine must be limited to about 10 - 12% and the average stage moisture content to no more than about 5%.

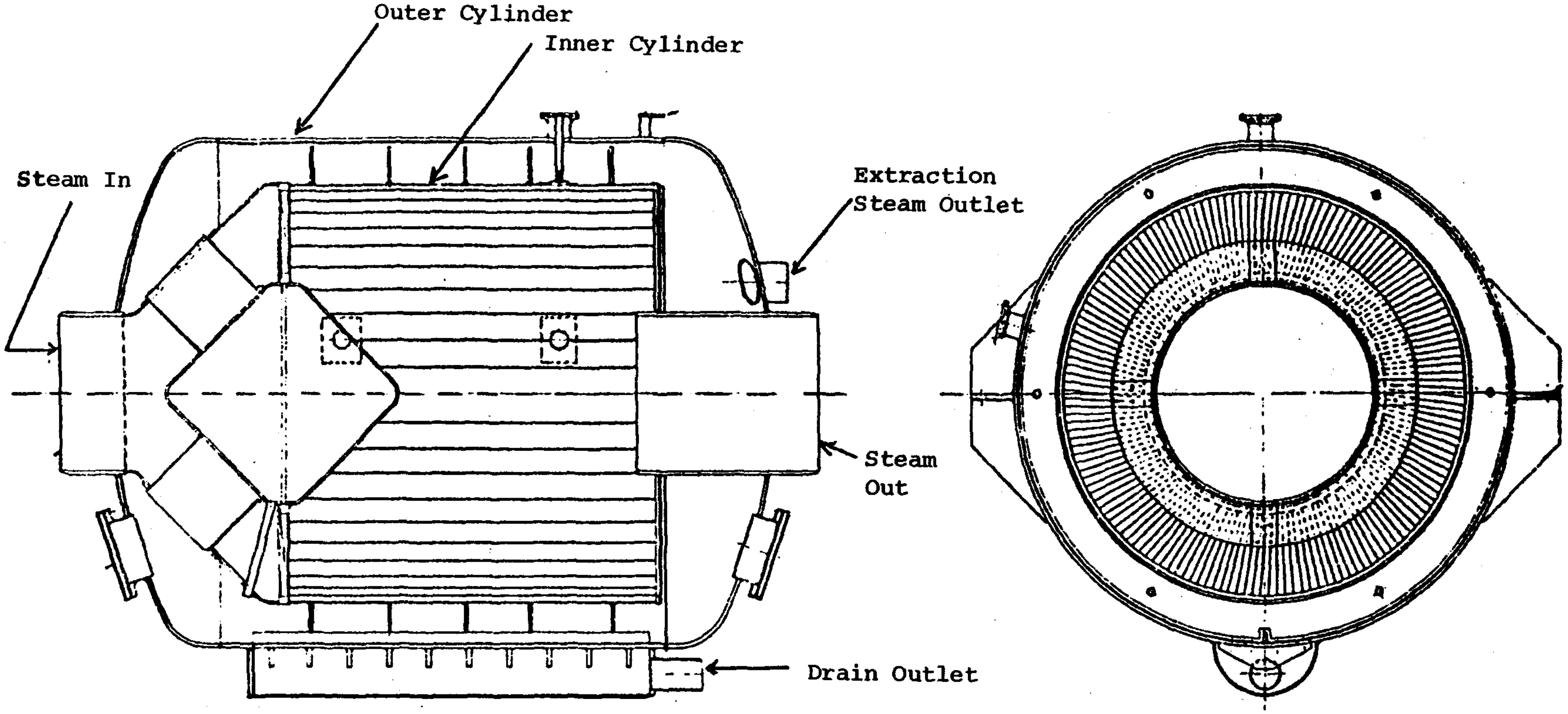
The design features which are used to limit either the moisture content of the steam or the effects of this moisture are:

(a) moisture separator,
(b) reheater,
(c) extraction steam,
(d) auxiliary moisture separators,
(e) water extraction grooves,
(f) turbine stage drains,
(g) steam piping design, and
(h) blade design and type.

With saturated steam at 250°C and 4000 kPa(a) entering the high pressure turbine, the moisture content of the steam reaches 10% long before the useful heat energy of the steam has been removed. The common solution to this problem is to exhaust the steam from the HP turbine, pass it through a moisture separator where the moisture content is reduced to near zero (.4% - .5% is typical) and then utilize the steam in a low pressure turbine. Figure 1.2 shows a typical moisture separator.

Wet steam is passed over a stationary ring in a conical diffuser, which imparts a swirling motion to the steam. The heavier water droplets separate and pass through longitudinal slots in the inner cylinder. The water runs down the space between the inner and outer cylinder and is collected in a trough at the bottom. The use of moisture separators enables the moisture content in the low pressure turbine to reach a maximum of only 10% at the exhaust. Without the moisture separators, the exhaust moisture would be closer to 20%.

Although it is not possible to produce superheated steam at the inlet to the high pressure turbine, it is possible to superheat the steam entering the low pressure turbine. This is done with a tube in shell heat exchanger called a reheater. Live steam at 250°C and 4000 kPa(a) is passed through the tubes and is used to heat the 500 kPa(a) steam exhausting from the moisture separators from approximately 150°C to 225°C. This superheating of the steam entering the LP turbine is doubly beneficial. First, it increases the energy content of the steam entering the LP turbine, allowing the turbine to do more work. Second, since the steam is heated well above its boiling point, condensation does not occur until a considerable amount of heat energy is given up. This means that no moisture will appear until several stages into the low pressure turbine and the moisture in each subsequent stage will be lower than it would be without the superheating.



Moisture Separator

Figure 1.2

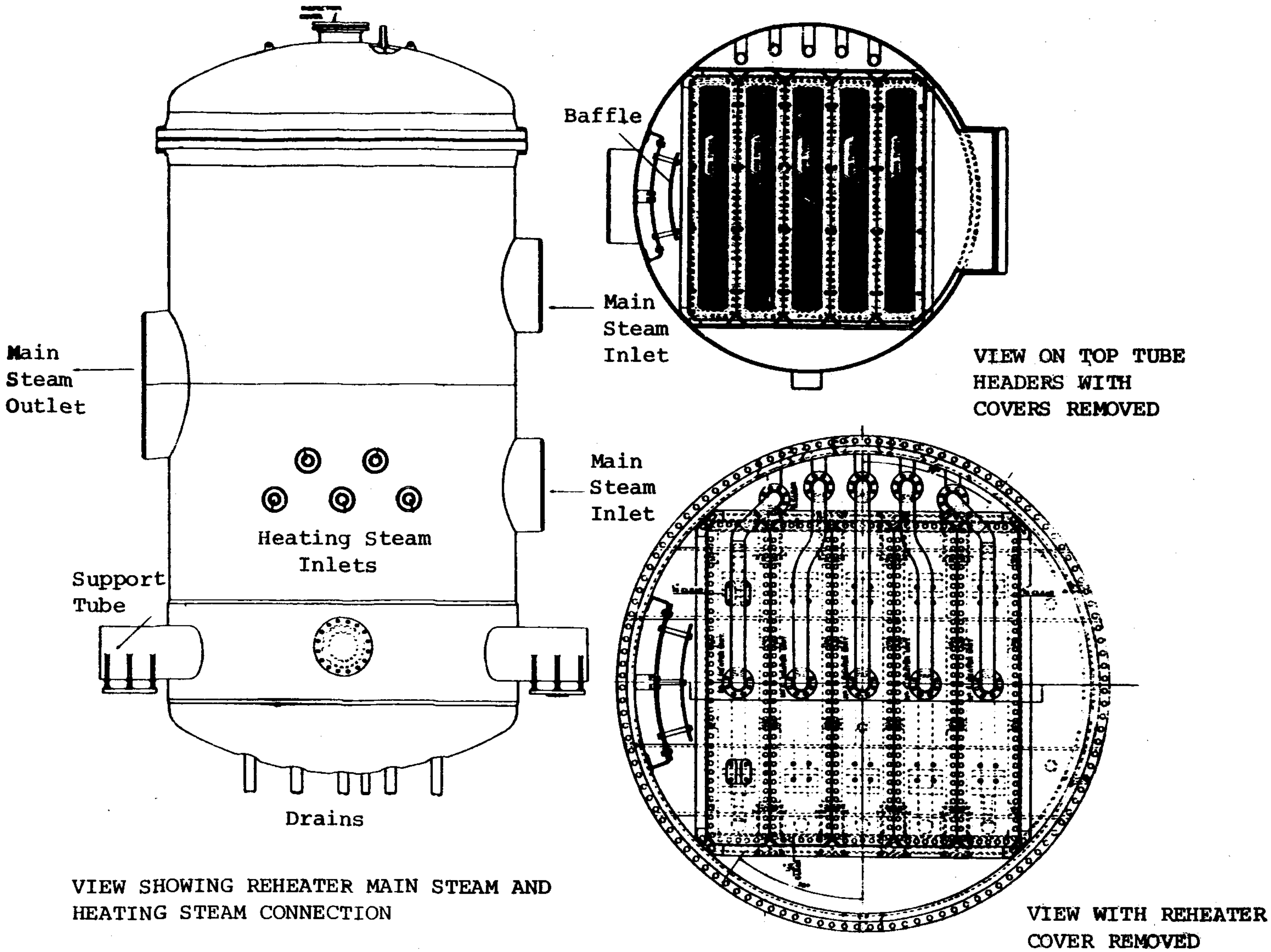
The erosion of the LP blading will be considerably less and, since the average stage moisture content is less, the turbine efficiency will be greater. Figure 1.3 shows a typical reheater. Steam exhausting from the moisture separators passes over several banks of tubes before leaving the reheater. The live steam which condenses in the reheater tubes drains to a tank and is then pumped directly back to the steam generators.

The removal of extraction steam for feedheating from the turbine casing is an effective method of moisture removal. Since the water droplets are much more dense than the steam, they tend to be centerfuged to the outer edge of the moving blades. If extraction steam leaves at the trailing edge of the blade tips, the water has a great tendency to leave with the extraction steam. Normally the percentage of water which leaves with the extraction steam is much higher than the percent moisture in the stage from which the steam has extracted. The extraction of steam from the turbine casing is a major method of moisture removal and if extraction steam is not removed from a turbine (such as the HP turbine at Bruce NGS) other provisions for moisture removal must be made.

An effective alternative to moisture removal by bleeding extraction steam is the auxiliary moisture separator. In this system, which is shown in Figure 1.4, steam is extracted from one side of a double flow HP turbine, the moisture removed and the steam returned to the same place on the opposite side of the turbine. Approximately 5% of the total steam flow through the HP turbine passes through the auxiliary separators. Although this method of moisture removal is slightly less effective than extraction steam bleeding, it has the benefit of allowing further use of the steam in the turbine.

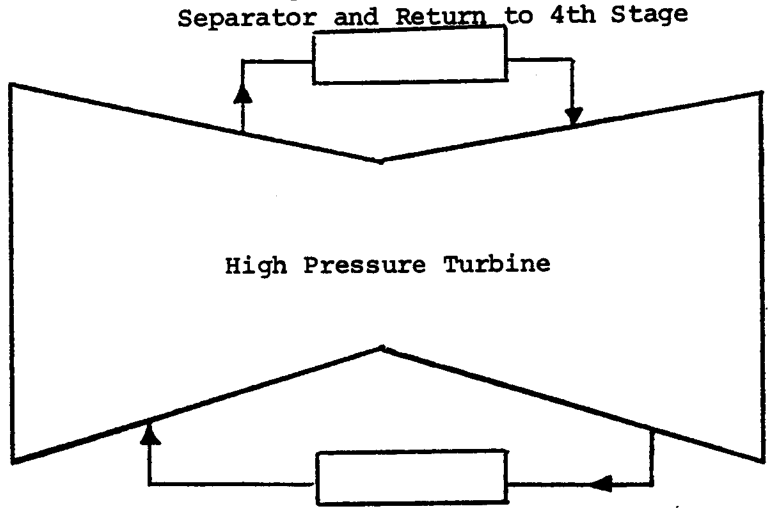
Another method of moisture removal which utilizes the centerfuging of water droplets to the outside of the moving blades is the placement of water extraction grooves after certain stages. These consist of annular grooves cut in the casing wall at the low pressure side of the moving blades in these stages. The water which is thrown off the blades passes through the grooves and is directed either to the inlet of an auxiliary separator or to the exhaust of the turbine. This method may be used in either a high or low pressure turbine.

In addition to removal of water entrained in the steam, provisions must be made for removal of water which collects in the lower part of the casing. The presence of standing water can cause corrosion, erosion, rubbing and fretting. This standing water may be removed by the water extraction grooves or by stage drains located in the bottom of the lower casing. These drains are sized to allow only a small flow of steam when the casing is dry.



Reheater

Figure 1.3



Auxiliary Moisture Separator

Figure 1.4

The piping associated with the turbine unit is equipped with steam traps to drain off water from low points in the system and thus prevent water entry into the turbine. These traps work automatically to pass water but block the passage of steam. Since they are designed to pass only the small amount of water present while operating, they are fitted with bypass valves to allow rapid draining of the relatively large quantities of water associated with startup.

To prevent backflow of drain water from entering the turbine, the lines are normally sloped downward from the turbine. Since backflow of water from a flooded or ruptured HP feedheater or from a flooded deaerator could have disastrous results, these lines are normally fitted with assisted non-return valves to prevent backflow of water. The low pressure extraction lines may be fitted with non-return valves as well, but these are for prevention of steam backflow.

The drop in turbine efficiency due to the mechanical force of water droplets on the blading is typically of the order of 1% loss in efficiency for each 1% average moisture content. However, this approximation depends on the type of staging used. Reaction turbines are more sensitive to the effects of water droplets decreasing efficiency by impact with the moving blades. Typically a 1/2 - 3/4% reduction in stage efficiency for each 1% moisture is encountered in an impulse stage. This effect is on the order of 1 - 1 1/4% for each 1% moisture in a reaction stage. In those turbines which encounter wet steam conditions such as the high pressure turbine in a nuclear unit, this fact has an influence on turbine design. One alternative is to make the HP turbine an impulse turbine; if, however, the HP turbine is a reaction turbine the need to keep the moisture content low can be readily appreciated.

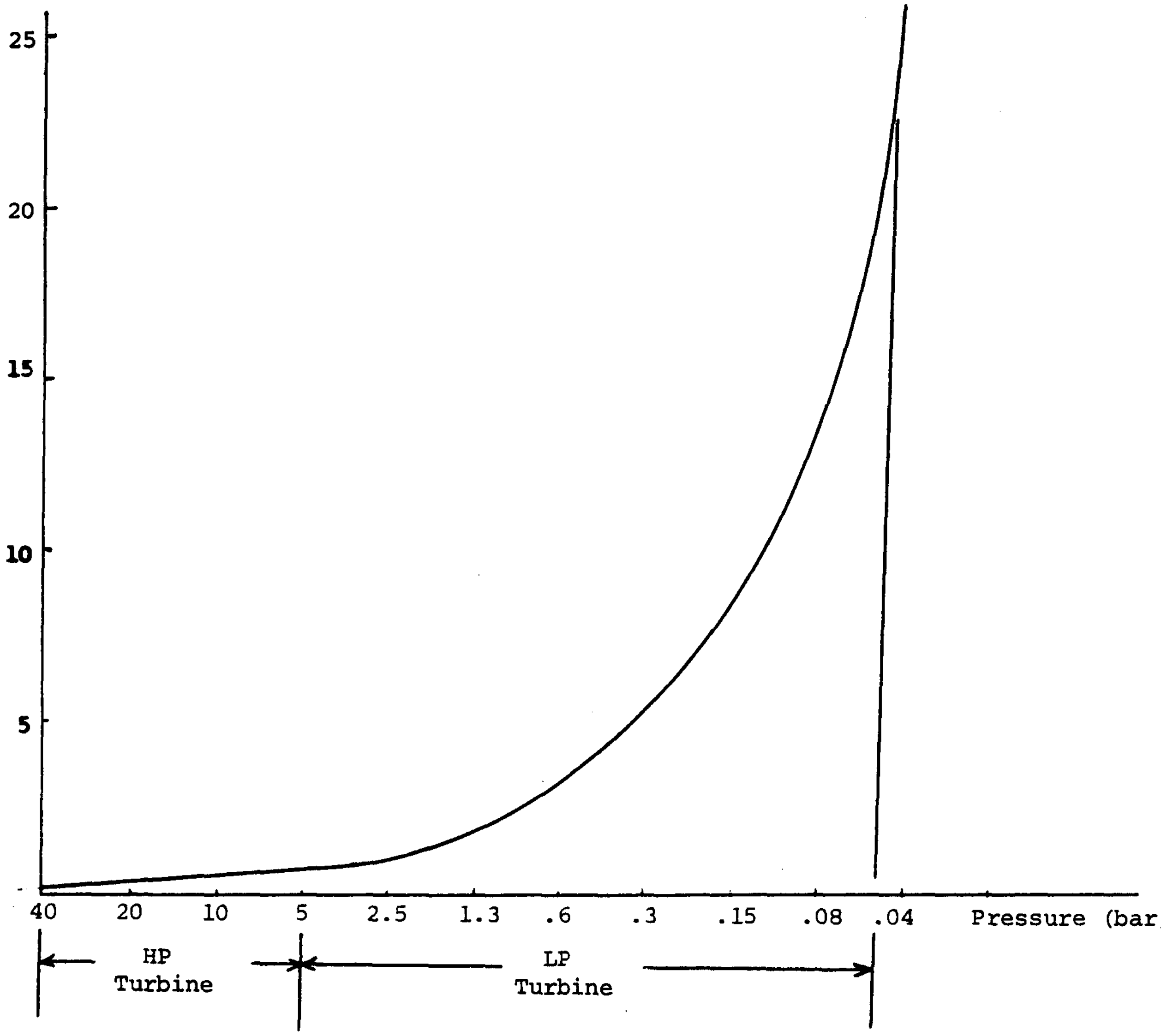
The common practice of placing shroud bands over the tip of moving blading to prevent blade tip leakage can have a major impact on water erosion of the blading. The shroud bands interfere with the centerfuging of water off the blades and their presence can cause blade deterioration near the tip of the blade. The current practice in modern saturated steam HP turbines is to delete the shroud bands. While this design increases tip leakage, it is superior in water removal from the blades.

In the latter stages of the low pressure turbine, the effect of erosion near the outside of the blades can be particularly severe. This is due to the combined effects of high moisture content and high peripheral blade speed. To strengthen the blade tips it is normal to braze erosion shields stellite or tungsten-chrome tool steel, on the leading edge of the blades. This hard, erosion resistant insert helps to minimize blade deterioration.

Steam Expansion

At the HP turbine inlet one kilogram of steam occupies .05 cubic meters. By the time this kilogram has expanded to the conditions at the LP turbine exhaust, it occupies 28 cubic meters or a 560-fold increase. To put this another way, if 5.4 cubic meters of steam per/second entered the HP turbine and no steam was removed, 3025 cubic meters per/second would leave the LP turbine exhaust. Unless some provision is made to handle this expansion, the steam will "choke-off" in the LP section of the unit and the HP section will never be able to attain its design steam flow and power level.

The graph in Figure 1.5 shows the expansion of steam from the steam generator to the exhaust of the LP turbine. The overwhelming majority of the expansion occurs in the latter stages of the LP turbine. The two most obvious ways of dealing with this expansion is increasing the area of the stages from HP inlet to LP exhaust and by having a larger number of LP turbines than HP turbines. Figures 1.6 and 1.7 show typical HP and LP turbines. You will notice the rapid increase in blade height in the LP turbine. This becomes more dramatic when it is recognized that there are three of the LP turbines.

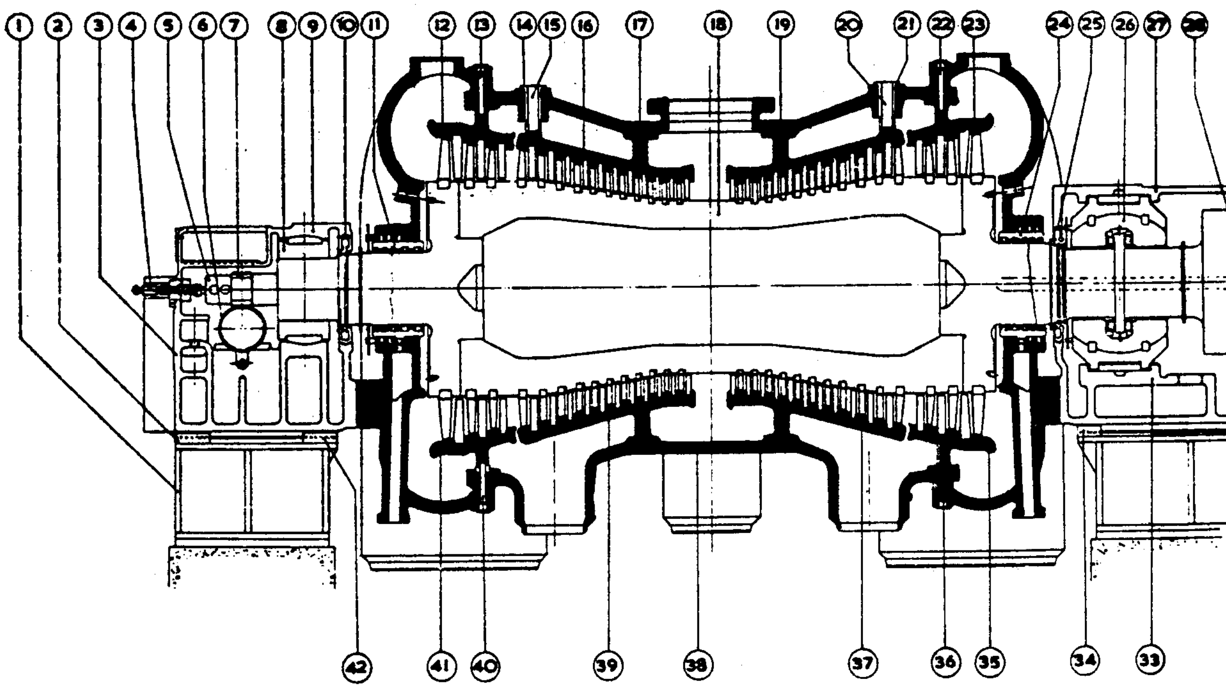


Steam Expansion Through a Turbine

Figure 1.5

HP TURBINE

Nº	NAME OF PART	Nº	NAME OF PART
1	Nº 1 BEARING BLOCK PEDESTAL	21	ECCENTRIC BUSH
2	LONGITUDINAL KEY	22	DOWEL
3	Nº 1 BEARING BLOCK	23	INNER CYLINDER 2ND BLADE CARRIER TOP HALF (RIGHT HAND)
4	OVERSPEED TRIP TEST PLUNGER	24	Nº 2 GLAND
5	OVERSPEED GOVERNOR	25	Nº 2 OIL BAFFLE
6	MAIN WORMWHEEL	26	COMBINED Nº 2 BEARING AND THRUST
7	MAIN WORM	27	Nº 2 BEARING KEEP
8	Nº 1 BEARING	28	COUPLING SPACER
9	Nº 1 BEARING BLOCK KEEP	29	
10	Nº 1 OIL BAFFLE	30	
11	Nº 1 GLAND	31	
12	INNER CYLINDER 2ND BLADE CARRIER TOP HALF (LEFT HAND)	32	
13	DOWEL	33	Nº 2 BEARING BLOCK
14	ECCENTRIC BUSH	34	LONGITUDINAL KEY
15	LOCATING KEY	35	INNER CYLINDER 2ND BLADE CARRIER BOTTOM HALF (RIGHT HAND)
16	INNER CYLINDER 1ST BLADE CARRIER TOP HALF (LEFT HAND)	36	DOWEL
17	OUTER CYLINDER TOP HALF	37	INNER CYLINDER 1ST BLADE CARRIER BOTTOM HALF (RIGHT HAND)
18	HP TURBINE SHAFT	38	OUTER CYLINDER BOTTOM HALF
19	INNER CYLINDER 1ST BLADE CARRIER TOP HALF (RIGHT HAND)	39	INNER CYLINDER 1ST BLADE CARRIER BOTTOM HALF (LEFT HAND)
20	LOCATING KEY	40	DOWEL
		41	INNER CYLINDER 2ND BLADE CARRIER BOTTOM HALF (LEFT HAND)
		42	LONGITUDINAL KEY



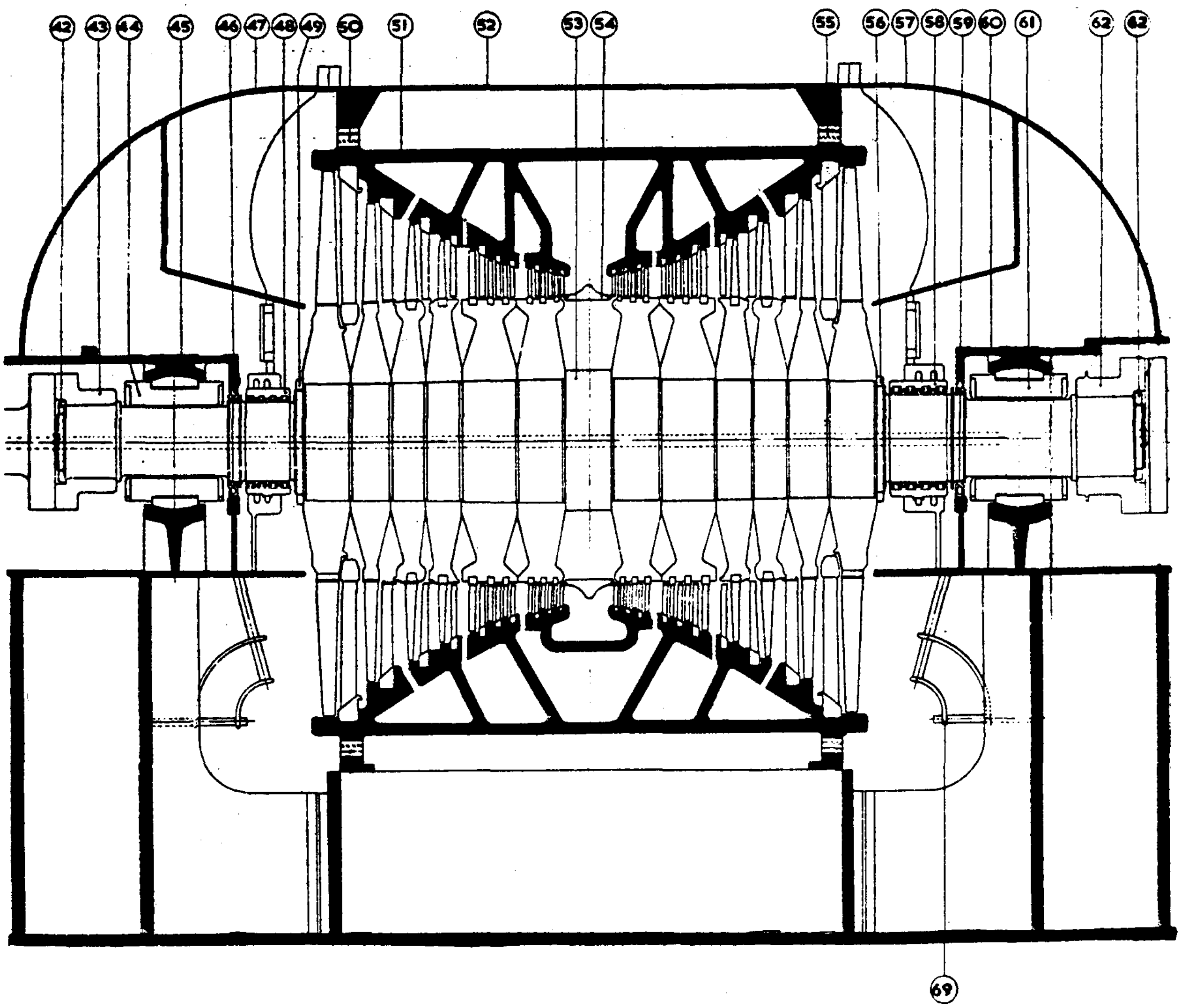
Typical HP Turbine with Extraction Steam after Stage 11

Figure 1.6

LP TURBINES

Nº	NAME OF PART
42	LP SHAFT COUPLING NUT
43	COUPLING ADAPTOR
44	Nº 3 BEARING
45	Nº 3 BEARING KEEP
46	Nº 3 OIL BAFFLE
47	LP CYLINDER EXHAUST COVER (END SECTION)
48	Nº 3 GLAND
49	LP SHAFT DISC RETAINING NUT
50	GUIDE KEY
51	LP CYLINDER CENTRE (TOP HALF)
52	EXHAUST COVER CENTRE SECTION
53	LP TURBINE SHAFT
54	DEFLECTOR RING (IN HALVES)

55	GUIDE KEY
56	LP SHAFT DISC NUT
57	LP CYLINDER EXHAUST COVER (END SECTION)
58	Nº 4 GLAND
59	Nº 4 OIL BAFFLE
60	Nº 4 BEARING KEEP
61	Nº 4 BEARING
62	COUPLING ADAPTOR
63	LP SHAFT COUPLING NUT
66	LAY SHAFT GUARD
67	GUIDE BRACKET
69	DOUBLE VANE DIFFUSER



Typical LP Turbine

Figure 1.7

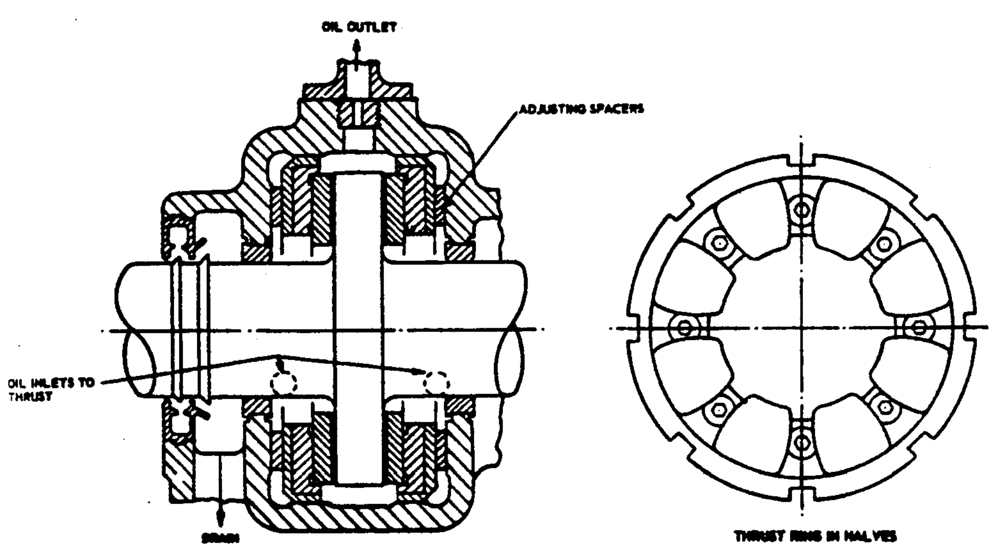
Another method of dealing with steam expansion is the removal of extraction steam for feedheating. This reduces the quantity of steam which must flow through the latter stages of the LP turbine. For example, in the low pressure turbines at Bruce NGS, only 75% of the steam which enters the turbine passes through the entire turbine. The remaining 25% is either extracted as steam or drained off as condensed water.

Axial Thrust

Whenever there is a pressure drop across a turbine component such as a diaphragm or blade wheel there is a net force which pushes the component from the high pressure side to the low pressure side.

The net force is proportional to the pressure difference and the area of the component. The larger the pressure drop and the larger the area, the greater will be the thrust.

Since the diaphragms which support the fixed blade nozzles are rigidly supported by the casing, they can be easily constructed to absorb this force. The blade wheels, however, are not rigidly mounted and the thrust must either be eliminated or accommodated.

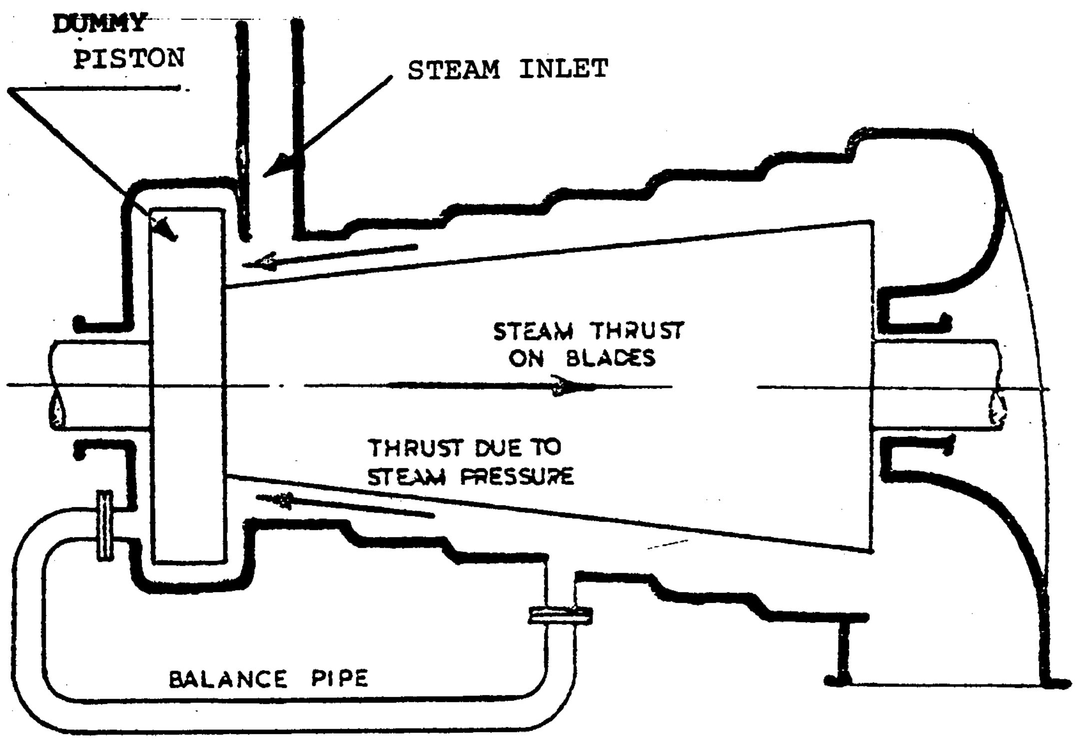


Tilting Shoe Thrust Bearing

Figure 1.8

The most common method of dealing with axial thrust is a thrust bearing. A simplified thrust bearing is shown in Figure 1.8. The thrust collar rotates with the shaft and stationary shoes on either side of the collar absorb the axial thrust. Oil is supplied between the shoes and thrust collar to lubricate and cool the bearing. While thrust bearings are used on all turbines, they are limited as to how much thrust they can absorb. Except for very small turbines a thrust bearing alone is not sufficient to deal with axial thrust.

A second method of dealing with axial thrust is the double flow turbine. In this type of turbine (Figures 1.6 and 1.7) steam enters at the center of the turbine and flows to the exhaust in both directions. Whatever axial thrust is produced on one end is compensated by an equal force of opposite direction on the other end. In a double flow turbine the net axial thrust on the shaft is very close to zero.



Single Flow Reaction Turbine with Dummy Piston

Figure 1.9

A method of compensating for axial thrust which is not used on any turbine in the nuclear generation division but is seen on some single flow conventional turbines is the dummy piston, shown in Figure 1.9. The combined axial thrust on the blade wheels is balanced by the opposite thrust on the dummy piston.

The final method of dealing with axial thrust is to eliminate it by using moving blades which have no pressure drop across them. This type of blading is impulse blading. Since the impulse stage has no pressure drop across the moving blades, there is no net axial force acting on the blade wheels. Single flow HP turbines frequently have impulse blades to eliminate axial thrust. Because the pressure drop per stage is much less in an LP turbine (50 kPa/stage) than in an HP turbine (250 kPa/stage), the need to eliminate the pressure drop across the moving blades in an LP turbine is much less. In moderate sized single flow turbine units, it is quite common to see an impulse stage HP turbine and a reaction stage LP turbine. The thrust is eliminated in the HP turbine but can be handled with a thrust bearing in the LP turbine.

To summarize, all turbine units have a thrust bearing. In order to maintain the forces on this bearing within acceptable limits the following are utilized:

1. Impulse staging in single flow HP turbines,

2. Dummy piston in some single flow HP turbines if reaction blading is used,

3. Double flow turbines in HP turbines with reaction blading and in large LP turbines with reaction blading.

Expansion of Turbine Rotor and Casing

During startup, shutdown and load changes, the temperature of the steam passing through the turbine changes. This causes the metal to heat up or cool down and provisions must be made to allow for the net movement of the casing and rotor as they expand and contract. Each casing has a fixed anchor point about which expansion and contraction occur. The casings are keyed remote from the anchor points to allow movement relative to the anchor point. Since the temperature change of the HP turbine casing from cold to hot is much greater than for the LP turbines, the HP turbine foundation must allow for more expansion than is necessary in the LP turbines. These expansion keys generally must be lubricated at regular intervals and the inability of the keys to move freely can result in high stresses in the casings and large forces on the keys (particularly the HP casing where expansion is greatest).

The rotor is fixed at the thrust bearing and expands by movement through the journal bearings along its length. The rotors in each of the turbines and the generator are solidly coupled to each other and expansion is cumulative.

The combined rotors of a large turbine generator are over sixty meters long and since the individual casings are much shorter the rotor expands more as it heats up than do the casings through which the rotor passes. In addition, since the rotor and casing are of different mass and shape, they do not heat up or cool down at the same rate, which results in differential expansion during the steam temperature changes, which accompany startup, shutdown and load changes.

The phenomenon of differential thermal expansion of the rotor and casings is known as axial differential expansion. This differential expansion is of two forms:

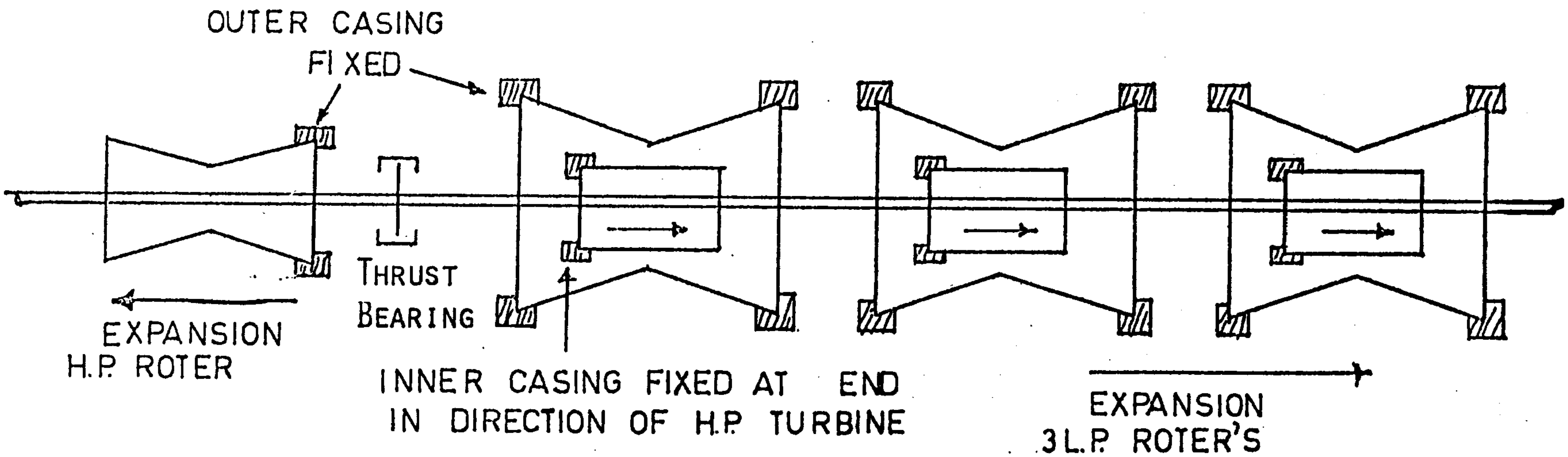
1. Transient axial differential expansion due to differing metal masses and rates of heatup and cooldown, and

2. Permanent axial differential expansion due to the difference in rotor and casing length and equilibrium operating temperatures.

Because the rotor is usually of a smaller mass than the casing and because it is surrounded by the steam, the rotor tends to heat up and expand faster than the casing. The net effect is that not only does the rotor expand relative to the casings, from steady state cold to steady state hot, but during turbine warmup it tends to expand faster than the casing simply because it is heating up faster.

The method of coping with the permanent axial expansion of the rotor relative to the casings (from cold to hot) is to allow sufficient clearance between the fixed and moving blades to accommodate this expansion. Figure 1.10 shows the arrangement of a typical large turbine unit. The clearances between fixed and moving blades must increase, the further the point is from the fixed point of the shaft (thrust bearing). Typically the clearances are in the order of 5mm in the HP turbine, 12 mm in the #1 LP turbine, 19 mm in the #2 turbine, and 26 mm in the #3 LP turbine.

Expansion of the low pressure turbines' outer casings is relatively small compared to shaft expansion and, therefore, the differential expansion detectors are essentially measuring shaft position. Since shaft growth is cumulative as one moves away from the thrust bearing, an expansion in the first LP turbine results in a similar increase occurring in the second and third LP turbines. If this does not occur something is wrong with the indication of differential expansion for the #1 LP turbine.



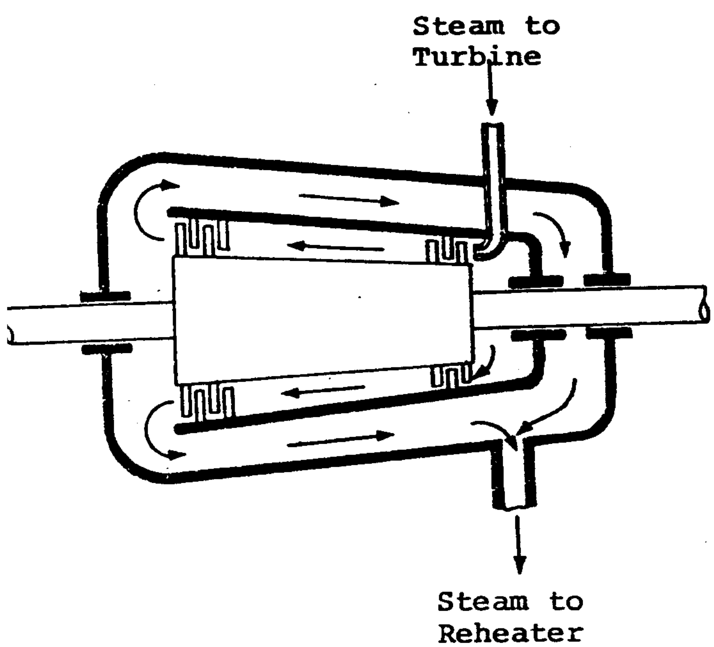
Fixed Points of Shaft and Casings

Figure 1.10

However, the converse is not true. An increase in the #2 or #3 LP turbine would not necessarily be reflected in the turbines closer to the thrust bearing. A change in the differential expansion of only one LP turbine could occur due to large changes in extraction steam flow from only one turbine or changes in the position of intercept/reheat emergency stop valves.

Since increased clearances reduce turbine efficiency, this method of compensating for differential expansion cannot be used indiscriminately. The use of clearances to compensate for differential axial expansion is usually restricted to that necessary to handle the steady state cold to steady state hot expansion. The transient differential expansion caused by differential heatup rates are usually handled by some other construction or operational technique.

The use of double casings is a common method of equalizing the heatup rate between the rotor and casing. The fixed blades are supported by the inner/casing. Steam is allowed to flow around the outside of the inner casing where it heats the inner casing from both sides and thus enables the heatup of the casing to proceed more rapidly and thus remain closer to the heatup of the rotor.

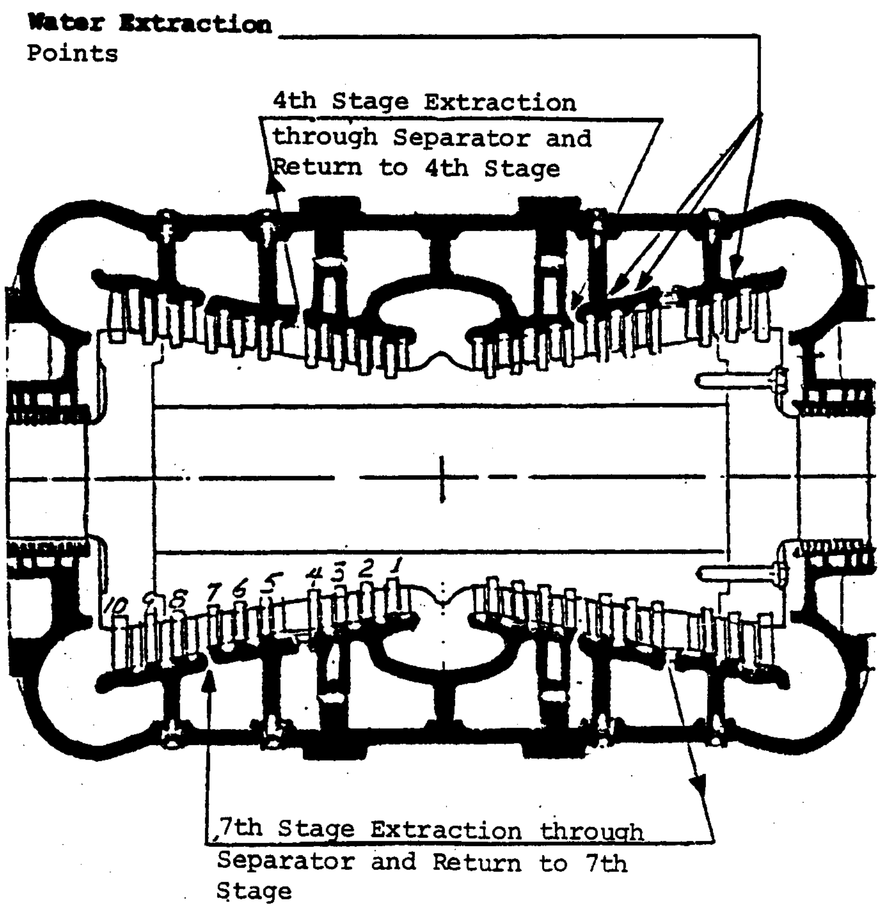


Simple Double Casing

Figure 1.11

Figure 1.11 shows the basic principle of a double casing on an HP turbine. The exhaust steam from the turbine is returned over the inner casing to help increase the rate at which the inner casing heats up and expands.

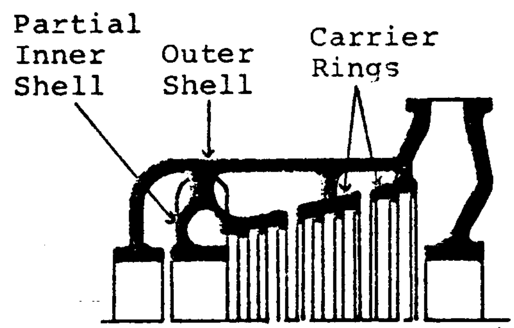
In practice the construction is not quite so simple. In Figure 1.6 you can see that the warming of the inner casing is achieved by a combination of exhaust steam, extraction steam and inlet steam flowing around the outside of the inner casing. In turbines with internal moisture separators, the steam flow to and from the separator may also be used to provide heating of the outside of the inner casing (Figure 1.12).



HP Turbine With Auxiliary Separators

Figure 1.12

A similar method of increasing the heatup rate of the fixed parts of the turbine is the use of carrier rings to support the fixed blade diaphragms. Figure 1.13 shows the general arrangement of carrier rings. The steam flowing through the turbine is allowed to flow behind the carrier ring and heat it from the back side, thereby improving the rate of heatup.



Single Flow Turbine with Carrier Rings

Figure 1.13

You will notice the similarity in appearance between a series of carrier rings and the double cylinder. Generally the distinction between the two is not clear cut and the portion of the inner/casing which carries the diaphragms is generally called a carrier ring.

Another method of equalizing transient expansion between the rotor and casing is the use of a hollow rotor or drum rotor as it is frequently called. This method of rotor construction is frequently used in large HP turbine rotors and can be seen in Figures 1.6 and 1.12. The drum rotor allows for a more uniform rate of heatup between the rotor and casing and thus minimizes transient differential expansion.

Despite all these construction techniques, a given turbine can only accommodate a given amount of transient axial expansion. The allowable rate of heatup is set to keep, amongst other things, the differential axial expansion within acceptable limits.

Thermal Stresses in Turbine and Rotor

If a large structure such as a turbine casing or rotor is heated, the part in contact with the heat source will heat up first and the temperature of the more remote parts will lag behind as heat is transferred through the structure. This means some parts will be hotter than others and will, therefore, expand more. Since the structure is rigid, the expansion of the hotter parts tends to stretch the colder parts. This stretching sets up tensile stresses in the metal, which, over the short or long term (depending on the magnitude of the stress), can cause cracking, deformation or rupture of the metal.

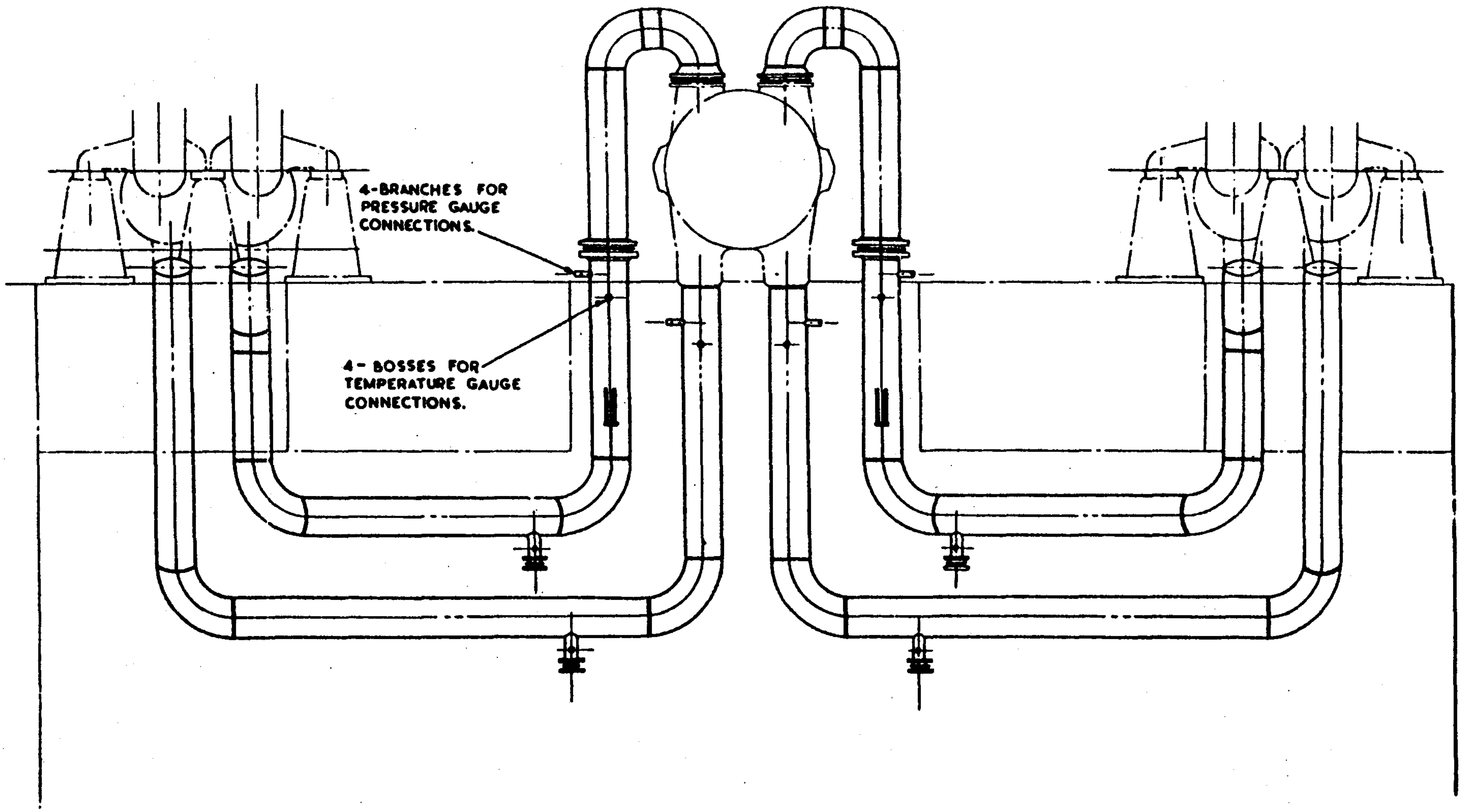
You have no doubt already recognized that the design features intended to minimize differential axial expansion such as the double casing, carrier rings and the drum rotor, also serve to minimize thermal stresses, since they allow for a more even heatup of the casing and rotor.

Lagging (thermal insulation) of the casing and steam piping also serves to minimize thermal stresses because it prevents high thermal gradients associated with high heat losses. Wet lagging, however, can cause "quenching" of a part of the hot metal which results in high local thermal stresses.

The admission or extraction of steam from the casing can cause uneven heating and thermal stressing of both the rotor and casing unless the entry or exit of steam is uniform around the casing. Figure 1.14 shows how steam is admitted to the inlet of the HP turbine by four admission lines which tend to equalize the thermal gradient around the casing. In addition, this arrangement tends to minimize radial thrust on the rotor. Extraction steam is generally removed by 360° entry into an extraction belt which circles the inner cylinder. This also tends to minimize the temperature gradient and the radial thrust associated with steam removal.

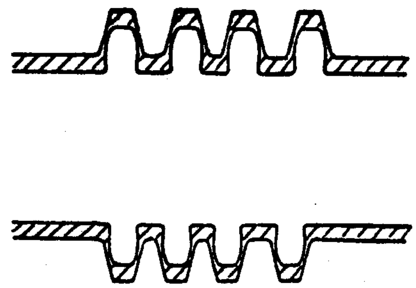
In Figure 1.14 the loops in the steam admission lines are placed there to prevent the creation of stresses in pipework when the piping expands while heating up. These thermal expansion loops allow for expansion of the piping. If the piping was not allowed to expand, the force developed on heatup would be sufficient to buckle the piping and possibly push the turbine off its foundation.

Another method of allowing for steam line expansion is the expansion joint shown in Figure 1.15. These expansion joints allow room for the piping to expand. Figure 1.16 shows the thermal expansion loops and expansion joints associated with the inlet to the LP turbines.



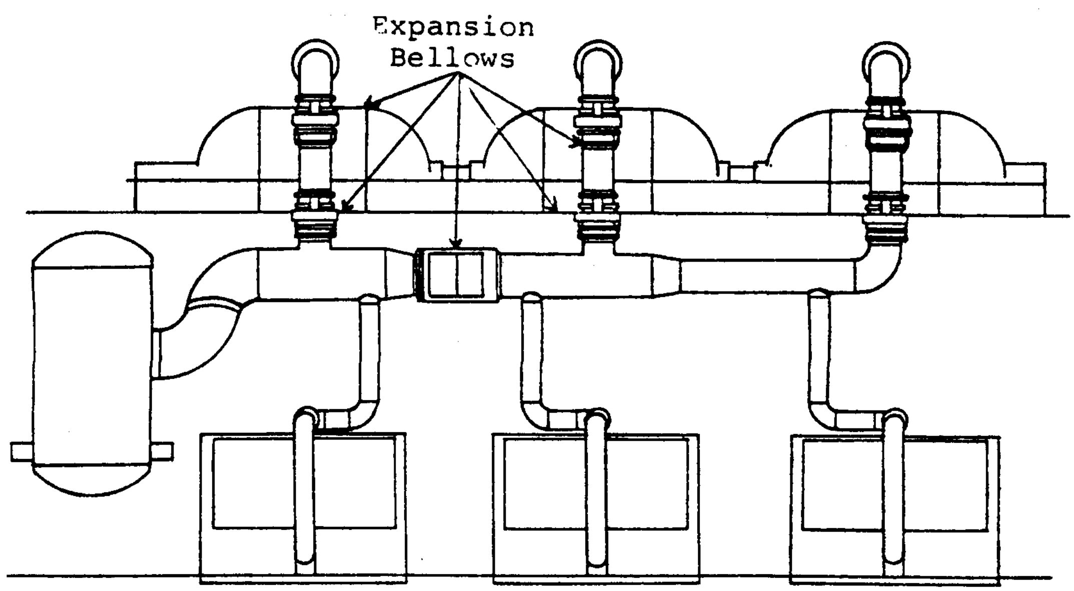
ARRANGEMENT OF H.P. INTERCONNECTING PIPES

Figure 1.14



Expansion Bellows

Figure 1.15



Expansion Joints on LP Turbine Inlet

Figure 1.16

ASSIGNMENT

1. Explain how the turbine unit at your station handles the following:

   (a) Moisture,

   (b) Steam expansion from HP turbine inlet to LP turbine exhaust,

   (c) Axial thrust,

   (d) Rotor and casing expansion and differential expansion, and

   (e) Thermal stresses in casing and rotor.

2. What are the problems associated with moisture in the steam passing through a turbine? What design features are intended to minimize this moisture or its effect?

3. What is differential axial expansion? What methods are used to compensate for this problem?

4. A common method of fixing blade wheels on an LP turbine shaft is to build wheels with a hole in the center through which the shaft is passed. In early large LP turbines, the wheels tended to come loose from the shaft when heating up or increasing load. Why do you think this happened?

5. In Figure 1.7 what methods are used for compensating for differential axial expansion, casing stresses and moisture?

R.O. Schuelke