TUTORIAL ON STEAM TURBINE DRIVERS FOR FOSSIL AND NUCLEAR FEED PUMP APPLICATIONS

by
Jerry DiOrio
Senior Service Engineer
Siemens Demag Delaval
Hamilton, New Jersey

Jerry DiOrio is a Senior Service Engineer for Siemens Demag Delaval, in Hamilton, New Jersey. His 34 years of experience in turbomachinery has involved all aspects, including thermodynamic performance, mechanical design, manufacturing issues, and field troubleshooting.

Mr. DiOrio is a graduate of the University of Dayton with a B.S. degree (Mechanical Engineering Technology).

ABSTRACT

This tutorial will present an overview of the operation and design of steam turbines used for driving reactor and boiler feed pumps, called a boiler feed pump turbine (BFPT). Overall, these turbines are designed in much the same way as other mechanical drive turbines; however, there are characteristics to these units that make them unique to this application. A feed pump turbine must be integrated and completely compatible with the pump and the main turbine generator (TG) unit because these pieces of equipment are so interrelated.

One of the most unique features of these turbines is their ability to operate from two separate and very different steam supplies. To accommodate these steam sources, two separate inlets must be used, each with its own special characteristics. The high pressure (HP) inlet has a separate steam chest designed to accept the boiler pressure. This inlet will feed a nozzle block that has an arc of admission of 25 percent or less. The low pressure (LP) inlet generally utilizes approximately 50 percent of the arc and has a much larger flow passing area.

This type of turbine varies in output across the operating range of the plant. There are two basic operating cycles. The two pump operating cycle usually runs between 40 percent and 110 percent main unit load (MUL), this operation is achieved by using all LP steam. The one pump operating cycle, startup to approximately 65 percent MUL, uses both steam sources. The percent of each is determined by the load required, as well as the LP steam conditions. As the load increases on the TG main unit, the LP pressure supply to the BFPT increases, the HP steam is used to start up the BFPT unit and supplemented by the LP steam source to achieve the required load.

There are numerous other factors beside modes of operation to consider when designing a boiler feed pump turbine. These include the speed and power requirements of the pump, steam characteristics from the main unit and boiler, and exhaust pressure.

From a mechanical perspective there are a number of design features available on this style turbine to consider for maintaining reliability. The most obvious design feature is the single flow versus double flow design.

The rotating component of the turbine, the rotor, is the key for maintaining reliability. It consists of an integrally forged shaft, blades with various styles of fastening, and shrouds or blade covers. The blades have either an axial or radial entry design root to secure them to the disk portion of the shaft. Shrouding also has multiple variations. It can be riveted, integral, integral with wire, or an integral interlocking design.

Stationary elements consist of the diaphragms and labyrinth seals. The diaphragms direct the flow between the rotating blade rows.

All boiler feed pump turbines have two journal bearings and a thrust bearing. In most cases tilting pad journal bearings are used because of their inherent stability and damping characteristics. Thrust bearings are mostly the standard double acting type.

Controlling the steam flow to the turbine is done with the valve operating gear assembly. This assembly consists of a simple linkage arrangement that controls the lift of the diffuser valves feeding the low pressure chest and the high pressure valve. The entire assembly is raised and lowered by an oil controlled servo motor.

There are various other peripherals such as turning gears, probes, and grounding brushes that may or may not be used, which will also be discussed.

INTRODUCTION

This paper is to inform those unfamiliar with the design and operation of steam turbines used to drive reactor and boiler feed pumps. A basic understanding will be presented of how the turbine design is dictated by both the pump and main turbine generator. The steam sources come from the boiler and crossover steam from the turbine generator unit. The operating parameters, speed and horsepower, are set by the pump requirements.

It will describe the various configurations used and how two unique and very different steam sources can be inputted into the same machine. How these sources are used and controlled will be explained. The main components, the rotor and diaphragms, will be described in detail. Other components such as the journal bearings, thrust bearing, turning gear, and other miscellaneous peripherals will also be described.

Readers should be left with a good understanding of how a steam turbine is designed and how it operates in conjunction with the pump and main turbine generator.

STEAM SOURCES

Boiler feed pump drive turbines (BFPT) are basically designed the same as a mechanical drive steam turbine. However, there are features unique to this application with regard to their design and operation. It must be completely integrated and compatible with the pump and the main unit turbine generator (TG). When designing a BFPT there are three main factors that need to be considered: steam sources, operating points, and startup requirements.

BFPT accept steam from two separate and unique sources. These sources can include the boiler, the cold reheat line, the main unit crossover piping, or extraction points from the TG. The most common sources used are low pressure steam from the main unit crossover line between the high pressure (HP) and intermediate pressure (IP) turbine and high pressure steam from the boiler. The low pressure steam from this source will normally range from 75 psig to 250 psig and will vary across the load range as the main unit load varies. High pressure steam taken from the boiler is normally 2400 psig. On supercritical plants this pressure can be as high as 3500 psig (Figure 1).

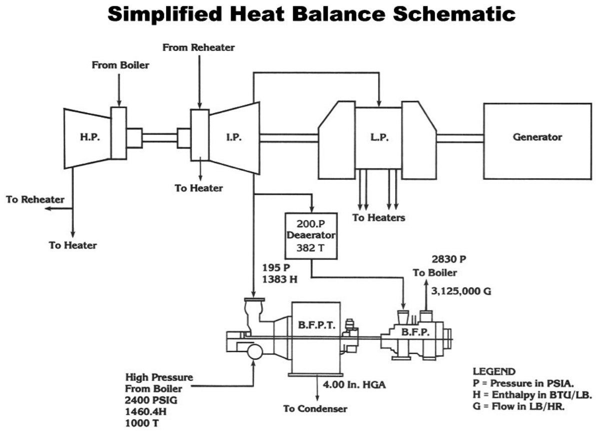

Figure 1. Typical Heat Balance.

This is the optimum combination for most modern plant cycles. The crossover steam is easily accessible and meets acceptable pressure levels for this type turbine. Steam pressure from the boiler allows the operators to start the feed pump turbines without using an auxiliary steam source. This high pressure steam can also be used to produce additional power from the BFPT when operating at single pump runout conditions.

The one very unique feature of this style turbine is its ability to utilize two very different steam sources. This is accomplished by using two separate inlets, each with its own special characteristics. The high pressure inlet has a separate rugged steam chest insert using heavy cast walls to accommodate pressures up to 2400 psig. There is a nozzle block attached to this inner chest that normally utilizes 25 percent or less of the active arc of admission. This area is small because its flow passing capability is determined by the requirements for full pressure operation. Because of the high pressure, the steam flow to this inlet is normally controlled by a single bayonet type valve. The low pressure (LP) inlet is a fabricated part of the main case and will utilize approximately 50 percent of the active arc. This inlet has areas much larger than the high pressure inlet to accept the higher volumetric flow from the main unit crossover, which is up to 250 psig. The steam entering this inlet is controlled by a series of four to five diffuser type valves, each feeding a separate chamber in the steam chest (Figure 2).

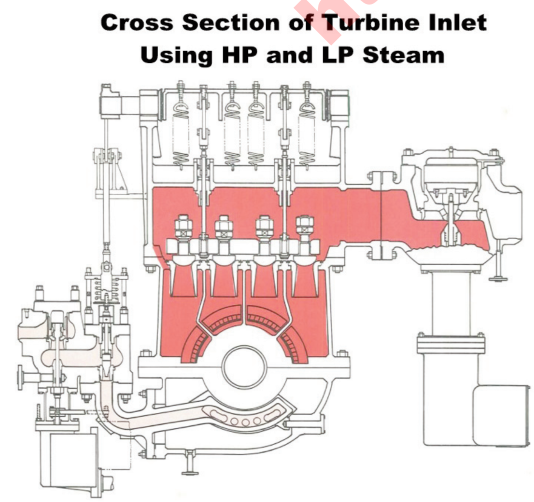

Figure 2. Inlet Cross-Section.

OPERATING POINTS AND OTHER FACTORS THAT INFLUENCE DESIGN

Due to the nature of its operation, a BFPT varies in output across the operating range of the plant. The steam characteristics are determined by the external steam sources supplying the steam, while the required power and speed are determined by the requirements of the pump. The medium for coordinating and transmitting the operational requirements of the main unit across the load range is the heat balance. In most cases, a main unit manufacturer will specify a guarantee flow called the rated flow; this defines the plant-rated point at a given set of conditions. In many cases architect-engineers or plant owners may refer to this main unit-rated point in various terms when establishing their own heat balances. This point may be referred to as the guarantee rating, 100 percent maximum continuous rating, or just 100 percent main unit load (MUL). No matter what terminology is used to define this point, it should be used as the basis for defining the other load conditions in terms of “percent of main unit load” (Figure 3).

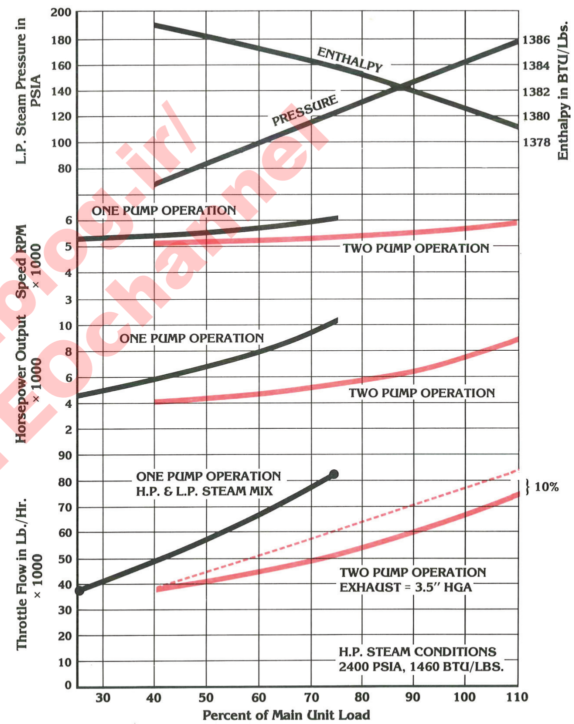

Figure 3. Constant Pressure Operation Performance Chart.

When designing the BFPT for a typical 2400 psig plant utilizing constant pressure operation and two 50 percent capacity pumps, it is necessary to determine the plant operating conditions at a minimum of three operating points. These points include the 110 percent main unit load, which is sometimes referred to as: valves wide open (VWO) or VWO plus 5 percent over pressure, which defines the maximum capacity of the low pressure steam; 40 percent MUL, which sets the minimum output of the LP steam on two pump operation; and the run out point, which falls around 65 percent MUL and is the maximum output required with one pump. From these three points the flow areas of the LP and HP nozzles can be determined. On most applications the 40 percent MUL point will set the maximum area of the LP nozzles due to the low crossover pressure and temperature of the main unit. The HP nozzle sizing is determined by the startup requirements or the one pump run out point.

To achieve optimum operation across the load range, it is beneficial to evaluate data at various intermediate points. For two pump operation, this includes 105 percent MUL also known as VWO, the 100 percent guaranteed load, and the 75 percent MUL.

One pump requires two additional points, 50 percent MUL and 25 percent MUL. When using a constant pressure cycle, the two pump operating points between approximately 40 percent load and 110 percent load are achieved using all LP steam.

The one pump operating conditions are somewhat more difficult to design for since they can be a combination of the two steam sources. The percent of each is determined by the load required, as well as the LP steam conditions. As the load increases on the main unit, the LP pressure supply to the feed pump turbine increases. This mixture of steam for one pump operation is normally used up to approximately 65 percent MUL.

A variation of this constant pressure cycle is to use a variable or sliding pressure boiler. This mode of operation normally occurs below 75 percent MUL. When using a sliding pressure boiler, less load is required to maintain the lower boiler pressure. This results in reducing the horsepower and speed requirements of the BFPT. This effect enables the turbine to go down as low as 20 percent MUL on LP steam with two pumps operating (Figure 4).

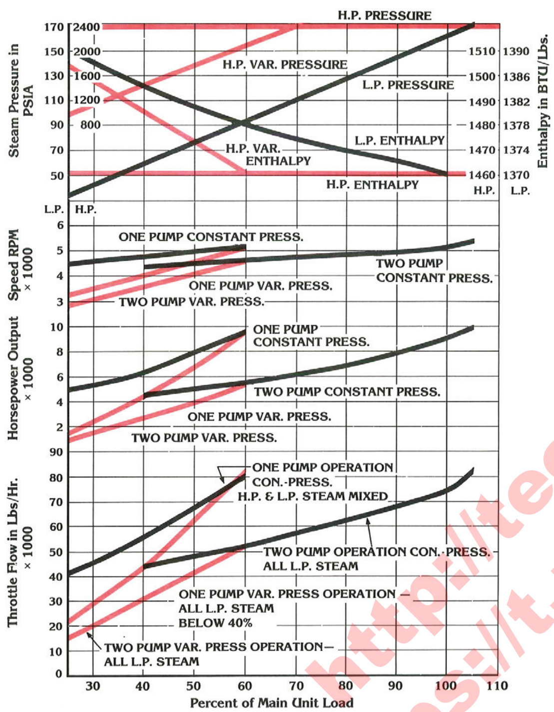

Figure 4. Variable Pressure Operation Performance Chart.

These various modes of operation are only one factor to consider in selecting a BFPT. Other factors such as the speed and power requirements of the pump, as well as the characteristics of the steam used to supply the energy, have a profound effect in establishing the size of the turbine. A good example of this is as the inlet pressure and temperature increase, the specific volume of the steam decreases. As the specific volume decreases it results in increased flow passing capability for a given area, ultimately leading to a decrease in nozzle area and possibly inlet size.

The exhaust pressure affects the turbine in a similar way. As the exhaust vacuum increases, the specific volume of the exiting steam increases. This increase in specific volume causes an increase in volumetric flow, which affects the sizing, as well as the performance, of the exhaust end. In this case, to maintain the performance of the exhaust end, an increase in the exhaust area is required. In many cases this increase in area is accomplished by using a double flow exhaust. Speed is another factor to be considered when discussing a system approach. The turbine driver must be coordinated with the pump and in doing so care should be taken not to optimize one piece of equipment at the expense of the other (Figures 5, 6, 7).

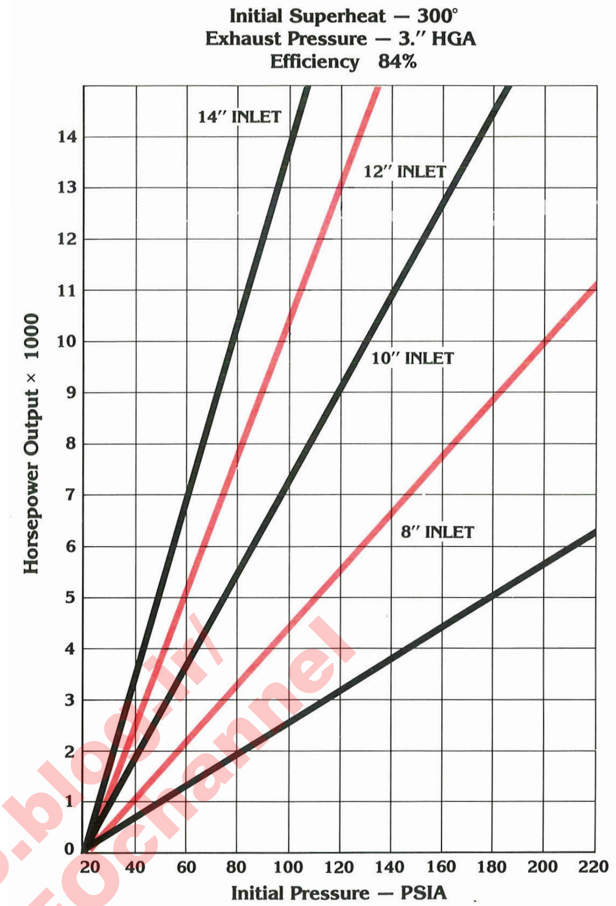

Figure 5. Inlet Pressure Versus Horsepower.

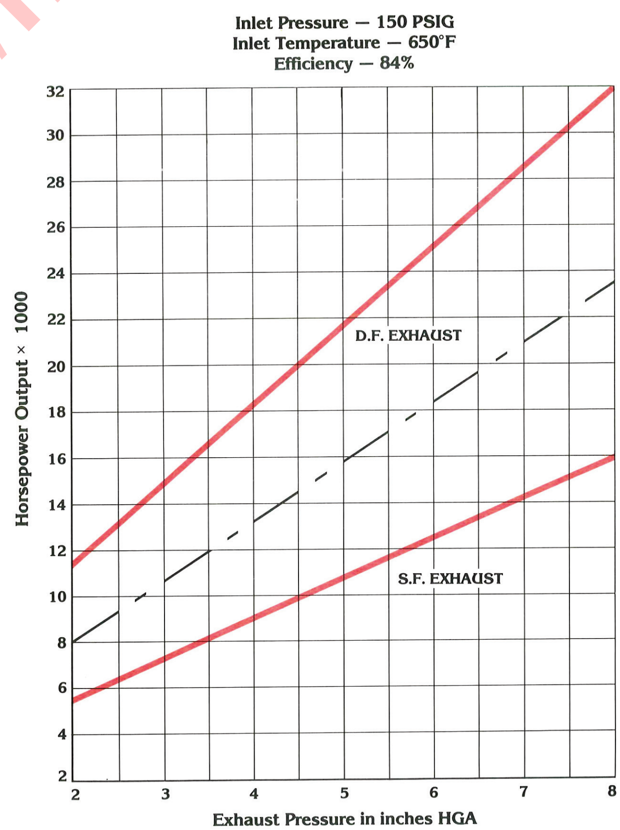

Figure 6. Exhaust Pressure Versus Horsepower.

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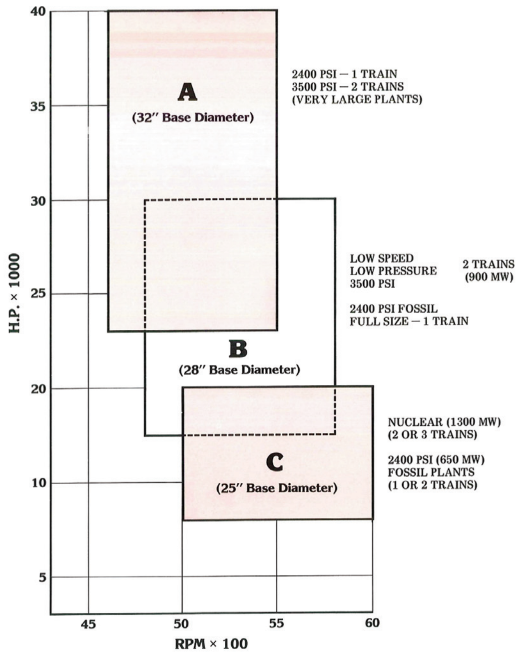

Figure 7. Turbine Selection Chart.

The concepts discussed above are somewhat generalized for clarity and brevity. It is practically impossible to look at one factor alone to evaluate these turbines. The information given above represents some broad guidelines to give an awareness of how these factors can affect the design of the turbine.

STARTUP

The method of starting up a feedwater system is something that is often overlooked until late in the design stage. However, this should be considered and decided upon early, along with the operating conditions since it is dependent upon the available pressure levels.

Startup can be achieved by using either the LP or HP inlet. This is normally dependent on the pressure level of the desired steam source. Steam pressure that does not exceed 50 psig over the LP design pressure, up to a maximum of approximately 250 psig, is introduced into the LP inlet. Pressures above this level will normally be fed into the HP inlet. As noted earlier, the flow passing capability of the HP inlet is established by the design pressure of the boiler. Therefore, at low pressures the flow passing capability and available horsepower output may not be sufficient to meet starting requirements without careful coordination early in the design phase. An alternate method, when the pressure is too low for the HP inlet, is to throttle down the boiler steam pressure to levels acceptable for the LP inlet.

MECHANICAL DESIGN FEATURES

From a mechanical perspective there are numerous design features available on this style turbine to consider for maintaining reliability.

The most obvious design feature is the single flow versus double flow design. As explained above, the double flow design is used when the volumetric flow of the steam exceeds the available flow area of the single flow design. By using the double flow feature the designer is able to maintain the same base diameter of the rotor for optimizing turbine efficiency while doubling the available flow area. The turbine efficiency is directly related to optimizing the stage velocity ratio, which is related to the base diameter of the turbine rotor. Since speed is governed by the pump manufacturer, the ability to control or maintain base diameters by use of double flowing the turbine is important. Typically, the higher the speed the smaller the base diameter. Most BFPT base diameters range from 25 to 32 inches (Figures 8 and 9).

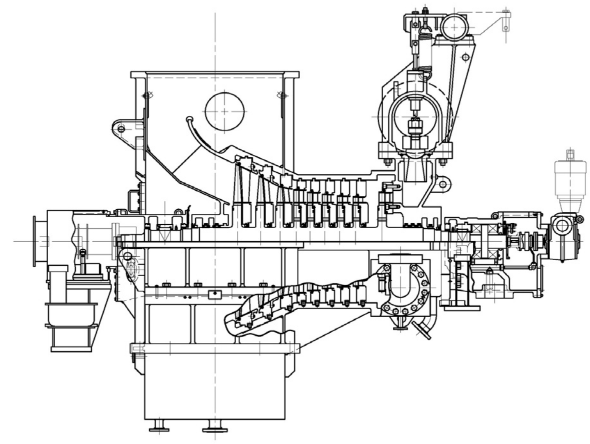

Figure 8. Single Flow Cross-Section.

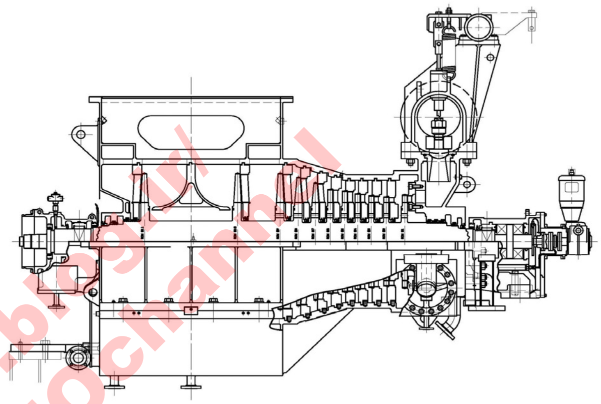

Figure 9. Double Flow Cross-Section.

SHAFTS

The rotating component of the turbine plays a major role in maintaining reliability. This component, the rotor, is primarily composed of a shaft, disks, and blades. When specifying a shaft, there are two items to consider: material and the method of disk attachment to the shaft. A good material for most applications is chrome moly vanadium steel (ASTM A470 Class 7). There are some nuclear applications where the LP inlet steam is extremely wet; this can cause excessive erosion in the turbine. In this case a stainless steel material (ASTM A768 Grade 1, Class 1) can be used for the rotor. However, although this material helps prevent erosion, it leads to other concerns. A babbited bearing running on the stainless steel will lead to bearing failures commonly referred to as wire wooling. This can be resolved by overlaying the stainless steel in the bearing areas with a carbon steel weld.

There are two primary methods of disk attachment: integrally forged and keyed with a shrink fit. The majority of rotors in operation today are of the integrally forged variety, which can maximize reliability.

BLADE MATERIAL

The turbine blading is a subject that lends itself to its own tutorial, and numerous papers have been written on this subject. For purposes of this application the subject matter will be kept to the basics. Turbine blades, sometimes referred to as turbine buckets, have a multitude of variations. These variations include material, erosion protection, fastener design, shroud design, and airfoil design.

The material and erosion protection are somewhat related. In most cases the Rateau section of the turbine, blading before the exhaust end, does not have provisions for erosion protection. They are typically made from a 403 stainless steel (ASTM A276 Type 403) or a 422 stainless steel, (ASTM A565 Grade 616). The 422 is used for control stages and highly stressed stages since it has a higher yield and tensile strength than the 403 and is good for temperatures as high as 1000°F.

The low pressure end blading, blades in the exhaust case, are predominately made of the 422 material. However, they can also be made from titanium (ASTM AMS 4928) or special 403 material. These are the blades where erosion protection is required in many applications. The most common methods of erosion protection are flame hardening or using stellite on the inlet edge of the blade.

The most common method is stelliting because it is much more effective than flame hardening. Up until approximately 1998 this stellite was applied by brazing a stellite strip to the blade and hand working it to the contour of the airfoil. Since then, new processes have been developed to apply a welded stellite. The main reason for the late development of this weld process was that it could not be applied to the 422 material. Also, metallurgical advances in the treatment of the 403 material developed processes to raise its mechanical properties to those of the 422 where needed. Titanium is used only in special cases where taller blades and higher speeds are required. Blades made of titanium are expensive and require more care at assembly; however, this material does have good resistance to erosion.

BLADE FASTENERS

The fastener is also an important part of the blade; its function is to secure the airfoil to the disk. There are two basic types: axial entry and radial entry. Each of these two types has numerous variations. For this application there are four common type fasteners used. The T root and radial fir are radial entry fasteners, and the axial fir and bulb and shank are axial entry fasteners. As the names imply radial entry fasteners are loaded on the disk in a radial direction, and axial entry fasteners are loaded axially (Figures 10 and 11). Radial entry fasteners require wider disks than axial entry fasteners to provide enough disk material to maintain acceptable stress levels, typically these fasteners require at least a quarter inch more than the width of the airfoil, as the width of the airfoil increases, this additional allowance increases. On the other hand, an axial entry fastener is basically the same width of the airfoil; this allows the designer to use wider blades in the same axial space.

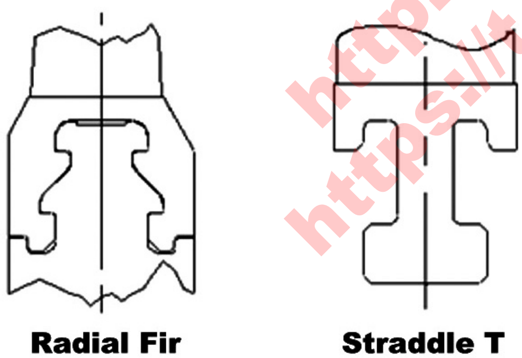

Figure 10. Radial Entry Fasteners.

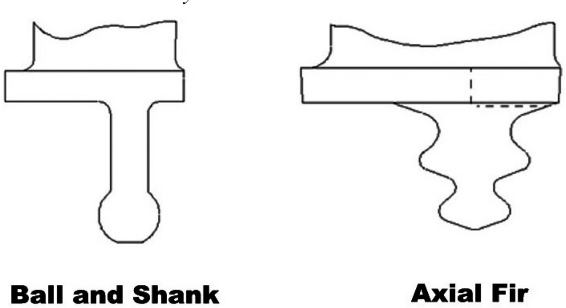

Figure 11. Axial Entry Fasteners.

The radial entry design requires a closing blade or closing piece. Basically the last blade assembled is required to have a simplified straight configuration to enable it to be inserted rather than slid into position (Figure 12). Once this blade is fitted, pins are drilled and reamed axially through the disk and closing piece to secure this last blade. Depending upon the operating speed and weight of the blade, a closing piece may be required; this is a closing blade with the airfoil removed to reduce the centrifugal stress. When a closing piece is required, it is necessary to insert a balance piece 180 degrees away from the closer to achieve good balance. When using axial entry fasteners, there is always a full 360 degrees of blading.

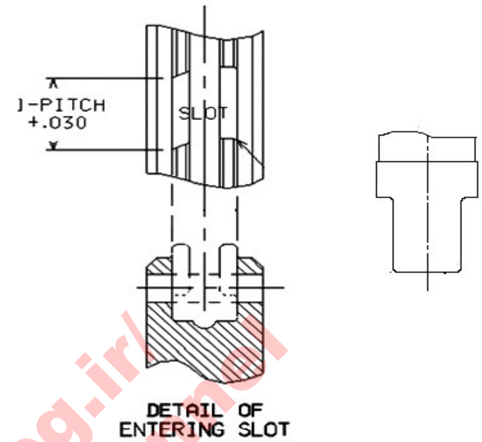

Figure 12. Closing Blade Geometry.

BLADE AIRFOIL SECTIONS

Moving up the blade, the next section would be the airfoil. This is where the steam is directed through to turn the rotor and generate the power. There are two basic types of airfoils used: straight sided and taper twisted. The straight sided airfoils are used through the Rateau section of the turbine. The taper twisted sections are used on the taller blades in the exhaust or low pressure end of the turbine. On a straight section the airfoil shape is constant from the base to the tip. The taper twisted start out with a larger section at the base that is blended to a smaller section at the tip, and as the size of the section tapers down the blade twists changing the inlet angle of the section. These are used to maximize efficiency by better distribution of the steam because of the low pressures and increasing volumetric flow (Figure 13).

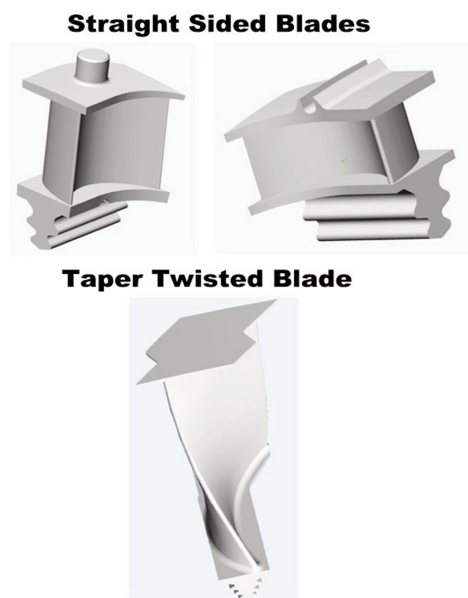

Figure 13. Tapered and Straight Blades.

BLADE SHROUDS

The final portion of the blade is the shroud. Again, there are a variety of selections to choose from. Shrouds can be plain riveted, integral plus riveted, integral with rolled in wire, integral tongue and groove, and integral Z-lock. The oldest of these designs is the plain riveted; in this design a tenon is machined protruding from the top of the airfoil section, a metal band commonly referred to as a shroud is fitted over these tenons to form the blades into groups. The tenons are then peened to attach the shroud to the blade. The size of the blade groups is used by the designer to help control the harmonics and excitation index of the blades. If blade harmonics are not controlled, they can excite blade frequencies, which will result in failures. In the past, these tenons took many shapes. Some were kidney shaped, some matched the contour of the airfoil, and others were round. Most modern blades that still utilize tenons use the round shape. The round tenon is preferred because it is much easier to peen, either by machine or by hand. This style is used mainly on the Rateau stages with the exception of the control stage and in some older units it has been used across the entire steam path (Figure 14).

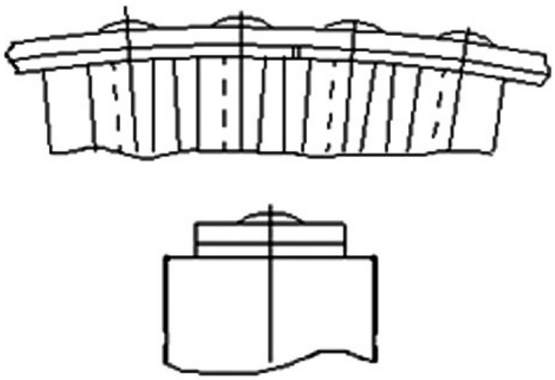

Figure 14. Riveted Design Shroud.

The second configuration consists of an integral shroud on top of the airfoil with a tenon on top of the integral shroud for attaching additional loose shrouds. An integral shroud is a cover that is machined from the same piece of bar stock and is an integral part of each blade. When using this design, one or more additional shroud bands may be attached. These additional shroud bands are used to create a full 360 degree locking of the blades, which provides additional damping of the blade frequencies. This design is extremely rugged and is found mostly on the first stage, which is called the control stage. This first stage experiences the most severe duty due to partial arc affect of the multiple steam inlet ports (Figure 15).

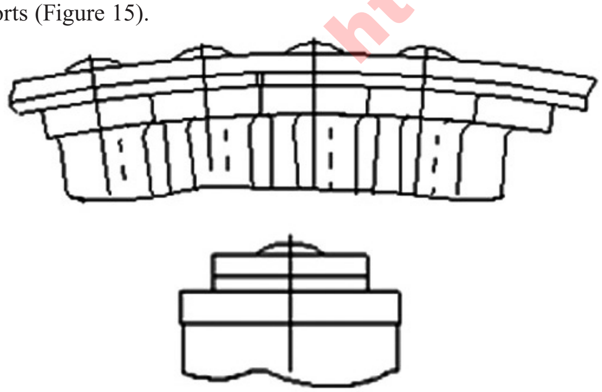

Figure 15. Integral Plus Riveted Design Shroud.

Option three, the integral shroud with rolled in tie wire is much like the second configuration. However, on this design the integral shrouds should be in hard contact with the adjoining blade. After they are assembled in the rotor, a horn-like configuration is machined into the top of the integral shroud to accept the wire. The wire is then put into position on the top of the shroud and is held in position by rolling the horns over the wire. Again this design offers the 360 degree blading with additional damping provided by the wire. The tie wire design has been applied from row two to the end of the turbine. In a few cases it has even been used on lightly loaded first stage blades (Figure 16).

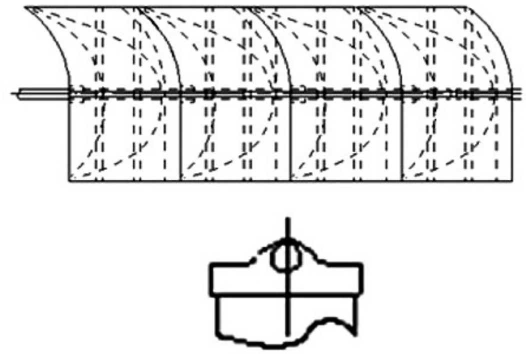

Figure 16. Integral Plus Wire Design Shroud.

The last two types, integral tongue and groove and integral Z-lock, are normally found on the low pressure end blades. The integral tongue and groove is just as the name implies; one end of the shroud has a protruding tab that fits into the opposite end of the adjoining blade. The Z-lock has Z shaped interlocking surfaces on both ends of the integral shroud. Both designs provide for excellent damping on the taller taper twisted blades. Although both designs are assembled to provide a tight contact between blade shrouds, these designs allow for additional contact as these taper twisted blades get up to speed and twist or unwind. The main difference between these two designs, other than the shape, is that the tongue and groove blades can be assembled on an individual basis while the Z-lock have to be assembled as a 360 degree group. This is accomplished on the tongue and groove by utilizing straight ends on the last two blades to be assembled and locking them with a welded pin. Since there is no additional machining required on the Z-lock design it can be used for titanium as well as steel blades (Figure 17).

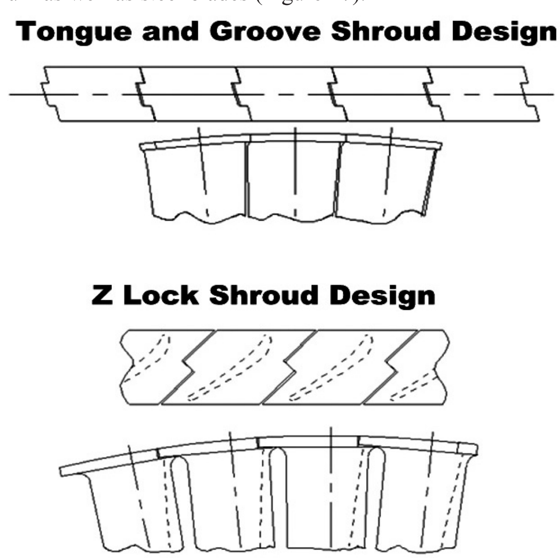

Figure 17. Tongue and Groove and Z-lock Shroud Design.

STATIONARY ELEMENTS—DIAPHRAGMS

The stationary elements of an impulse design turbine are the diaphragms. These diaphragms contain the airfoils, also referred to as vanes, that guide the steam from row to row and may contain the stage seals or moisture separation rings as required. A diaphragm is fabricated from three main components: an inner ring, a vane assembly, and an outer ring. The outer ring positions the diaphragm in the case and includes provisions for attaching outer seal rings or tip seals if applicable. In some cases where moisture levels are high a moisture separation ring may also be attached to this outer ring.

The vane assembly is composed of two spacers and the vanes. The spacers are strips with the shape of the vane machined at the proper setting angle and pitch. The vanes are then welded into these spacers. The final portion, the inner ring, is used for attaching the shaft seals, and any root seals where applicable. These three components are welded together to form a diaphragm (Figure 18).

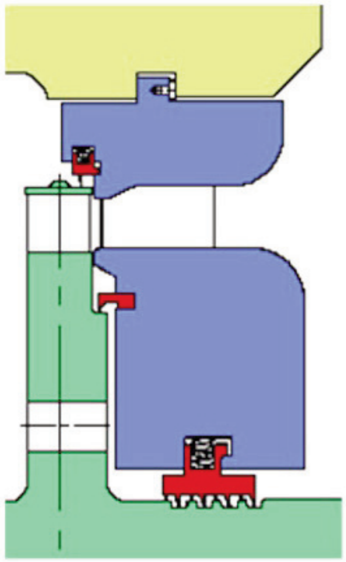

Figure 18. Stage Detail.

There are various materials that can be used in the construction of this component. The inner and outer rings are always made from the same material. The material used is determined in most cases by the stage temperature. For temperatures up to 800°F a carbon steel (ASTM A36) can be used, from 800 to 875°F a carbon moly (ASTM A204 Grade B) is used, from 875 to 950°F 1.25 Chrome 1 moly (ASTM A387 Grade 11) is used, and in cases where temperatures exceed 950°F up to 1000°F 2.25 chrome 1 moly (ASTM A182 Grade 22) is used. The vanes and spacing strips are made from a 405 stainless steel (ASTM A240 Type 405).

The thickness of the inner and outer rings will vary in any given machine. The first diaphragm in the machine is normally the widest and the thicknesses will gradually decrease as they approach the last stage. The stage operating conditions, pressure, temperature, and pressure differential will decrease from the inlet to the exhaust end of the unit. These conditions along with blade quantity, required area, and steam angles will determine the proper vane section for strength and optimized performance.

Moisture separation rings are used in the wet region of the turbine, usually the LP end section. These rings are set up in such a way that centrifugal force will drive the water particles up between the rings and settle in the groove where the water is channeled to an area with a drain.

SEALS

Seals are used to minimize the steam leakage across the stage. Any steam not going through the stationary and rotating components equates to lost power. There are various types of seals used on these machines. A basic stage can include as many as three different types of seals. Tip seals, which seal across the shroud portion of the blade, root seals, which are found below the blade riding on the disk, and labyrinth or root seals, which seal across the shaft. Tip and root seals can be spring backed or staked. The spring back style is more forgiving to high vibrations caused by operational upsets; however, a designer may not be able to use this design due to space limitations. Labyrinth seals have even more variations; they can be straight or stepped, conventional spring back or retractable. Straight seals are used when the stage pressure drops are small thermal growth is not excessive, and reducing the stage leakage is not as critical. Stepped labyrinth seals are more effective since they create a more difficult path by using high and low teeth to prevent the steam from leaking across the stage. In many applications the stepped seals may be used through the Rateau section, while straight seals are utilized in the low pressure section. Conventional labyrinth seals are the standard in most cases for BFPT. They are designed with radial springs that maintain them in the design position. However, like the spring backed tip seals the springs will allow them to retract during an upset with minimal damage in most cases. The main advantage to the retractable is that during startup and low loads when the rotor may be going through some high vibration modes, they are in an open or raised position. As the turbine reaches load they close down to the design clearance. With the variable operation of this type turbine and the relatively low stage pressure drops, they are not very practical due to their high cost. They can be very cost effective for the main unit.

The spring backed tip seals and labyrinth seals are made of the same material in most cases. Up to 850°F a leaded nickel brass material (ASTM B271 Alloy 976) is used, above 850°F a ductile niresist iron material (ASTM A439 Type D2C) can be used. Staked in seals can utilize the leaded nickel brass. However, above 850°F a 416 stainless (ASTM A582 Type 416) is used since the niresist material does not lend itself to the staking process (Figures 19 and 20).

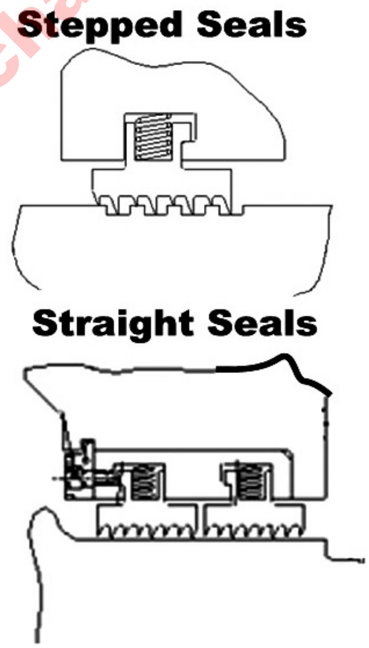

Figure 19. Straight and Stepped Seals.

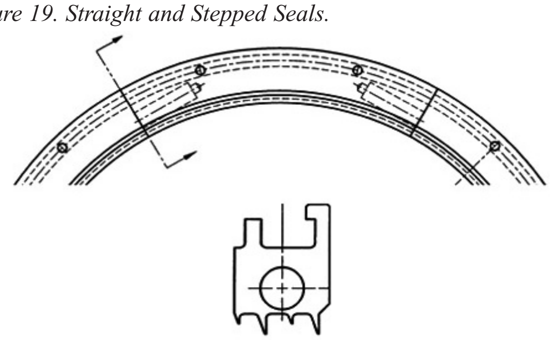

Figure 20. Conventional Retractable Seals.

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BEARINGS

Like the pump the turbine has three bearings, two journal bearings and one thrust bearing. There are two basic styles of journal bearings: sleeve and tilt pad. Sleeve bearings are found mostly on older design machines since tilt pad bearings offer better rotor stability and are more tolerant to impurities in the oil. Tilt pad bearings can be load on pad (LOP) or load between pad (LBP). The orientation chosen is based entirely on the rotor lateral analysis. There is no inherent advantage of one over the other. The one obvious difference in the two is that the LOP usually has two oil feeds, while the LBP generally has three (Figure 21). Both styles can have variations in the pivot design; this is a function of the bearing manufacturer. The two most common designs are the self aligning pivot and the spherical pivot (Figure 22).

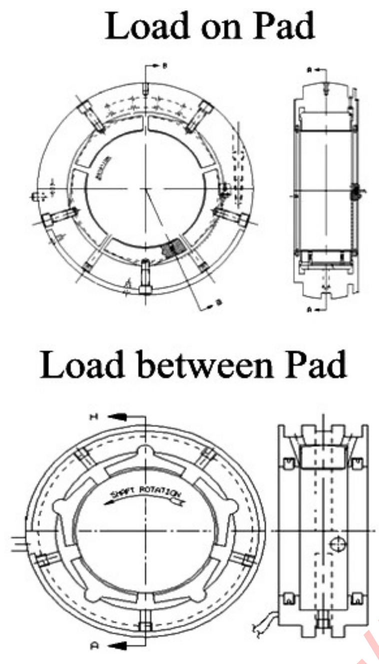

Figure 21. Load on Pad and Load Between Pad.

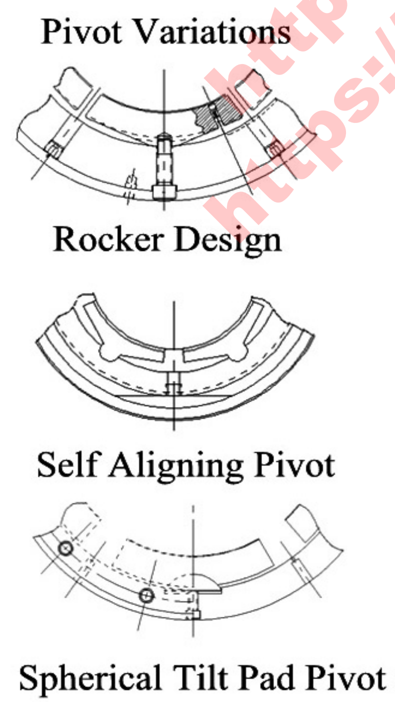

Figure 22. Journal Pivot Variations.

Thrust bearings also have two distinct styles: the standard flooded design and the leading edge groove (LEG). Most BFPT in current operation utilize the standard flooded type. The LEG was a later addition to the market. Unlike the standard thrust bearing that runs flooded, the LEG feeds the oil through slots located in the ends of the pads putting the oil directly on the pads. It requires a larger drain area since the theory is to evacuate the oil and uses offset pivot pads. The advantages of the LEG are less horsepower loss and a higher load capability (Figure23). Most bearings today are equipped with temperature sensor devices. These can be either resistance temperature detectors (RTDs) or thermocouples that are embedded in or near the babbit layer of the bearing. These devices can be inserted through the babbit layer with a babbited tip, through the bottom of the shoe just below the babbit area, and either secured with a spring and clip or epoxied in (Figure 24). These temperature devices need to be specified by the end user to ensure they are compatible with the sites electronics.

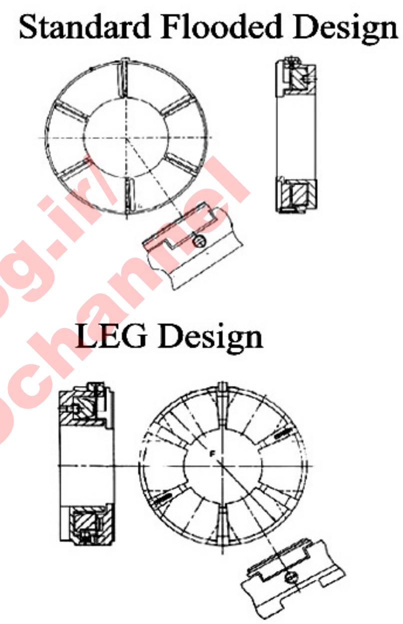

Figure 23. Flooded and LEG Thrust.

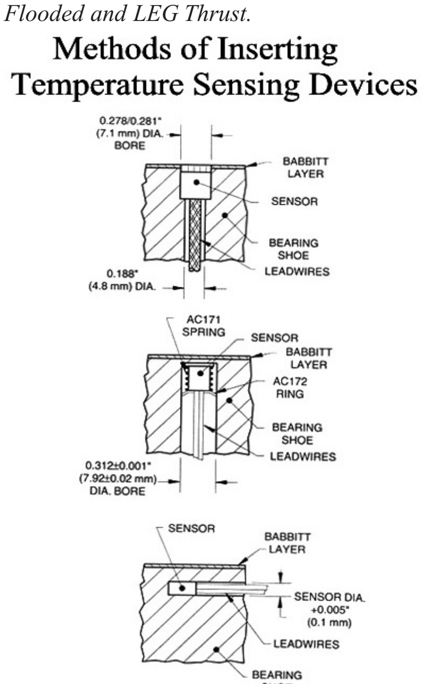

Figure 24. Temperature Instrumentation.

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VALVE OPERATING GEAR

The valve operating gear is the mechanism by which the steam flow is controlled into the turbine. Most valve operating gear assemblies are relatively simple. A series of diffuser valves, usually four to five for the low pressure inlet, is mounted on a bar. Each valve has an adjusting nut that is used to set the valves to lift at different heights. This assembly is connected with two spindles, which penetrate the steam chest cover to a lever arm that raises the valve bar parallel to the valve seats. This lever arm is then connected to a servo motor, which rotates the lever arm raising or lowering the bar to allow the valves to admit enough steam to meet the required load. Large springs are used to facilitate the timely closing of the valves, and to help overcome the force from the steam trying to open the valves. The high pressure inlet has a single bayonet type valve attached to a rod, which is connected to the lever assembly and begins lifting after all the low pressure valves are fully open.

Although the actual operating procedure is simple, the design and material selection of many of the components have to be well thought out to ensure reliable operation. The valves are made of a 422 stainless steel material (ASTM A565 Grade 616) with a stellited contact surface and the seats are made of a chromium-molybdenum steel (ASTM A182 Grade22) with stellited contact surfaces. The stellited contact surfaces are much more resistant to steam cutting and damage from hard contact of the seat to the valve. The lifting bar is also made from the chromium-molybdenum steel and nitrided to help prevent wear from the steel-on-steel contact of the valves. Spindles are constructed from a nitrided 422 stainless steel; the nitriding helps prevent wear from the bushings where the spindle penetrates the steam chest. The only other part requiring special detail to material is the bushings in the steam chest cover. This need to be made from a nitrided 403 stainless steel (ASTM A276 Type 403) to prevent wear and maintain clearances. The material requirements for the remaining parts not in the steam environment are not as critical. Areas of wear in these areas are accommodated by various types of bearings or bushings.

Another consideration the designer must take into account is thermal expansion. The spindle nearest the inlet flange, where it penetrates the lifting bar, must have a tighter clearance than the opposite end. This allows the bar to grow without destroying the parallelism of the spindles (Figures 25, 26, and 27).

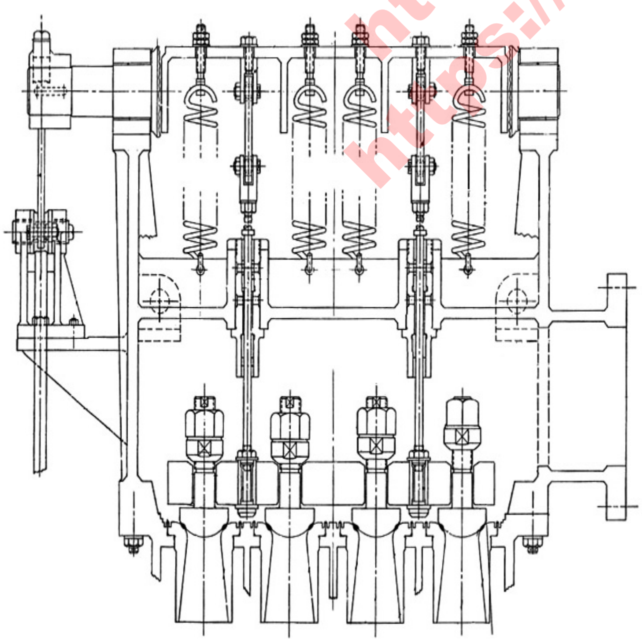

Figure 25. Valve Operating Gear—Front View.

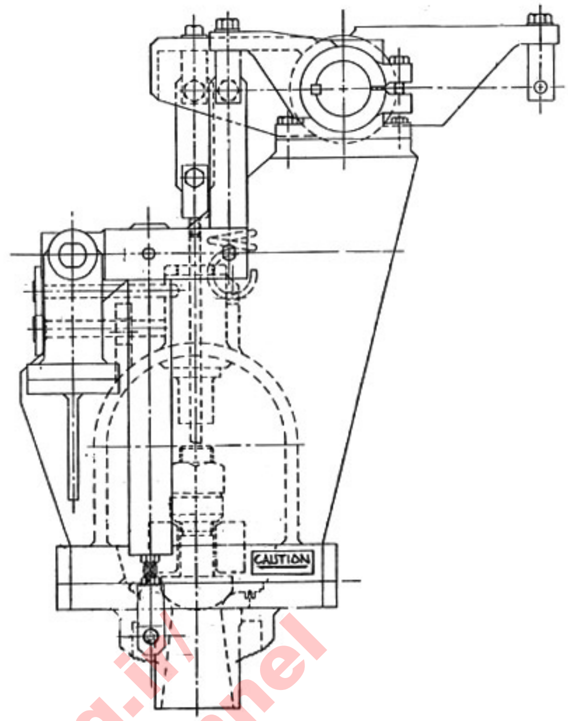

Figure 26. Valve Operating Gear—Side View.

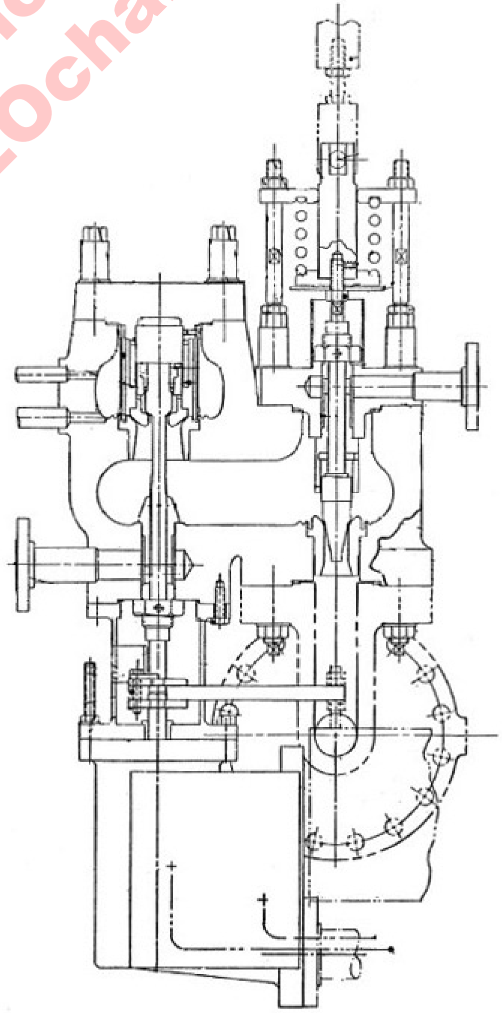

Figure 27. High Pressure Control Valve.

OTHER PERIPHERALS

There are other peripheral devices associated with a BFPT. These include such items as turning gears, probes, and grounding brushes. Turning gears are used to rotate the rotor at low speeds during startups and shutdowns. This allows the rotor to thoroughly cool or heat up while rotating, preventing the rotor from taking a bow due to rapid changes in temperature. Although it is generally agreed that this is required for the main unit, some designers believe that because of the shorter bearing span and stout rotor design in many cases, turning gears are not necessary for the BFPT.

Instrumentation has become a common practice on turbines today. Even many older machines not originally equipped with these devices have been retrofitted. These devices include speed pickups, axial pickups, keyphasers, and vibration probes. The speed pickups and keyphaser are used along with a 30 or 60 tooth gear for electronically controlling the machine. The axial probe monitors the axial movement of the rotor and can be used as an indicator of thrust wear. These three type probes are typically mounted on a bracket under the bearing cap on either end of the turbine. Vibration probes, on the other hand, are mounted externally through the bearing caps. These probes come in pairs mounted 90 degrees apart at the same longitudinal location along the shaft near the journal bearings. These probes continually monitor the radial vibration. The monitoring equipment is normally set up to have a warning and trip point so that the turbine can be shut down before any type of catastrophic failure can occur. As with the temperature monitoring devices these probes need to be specified by the end user to ensure that they are compatible with the station monitoring equipment.

Finally, grounding brushes may be used to remove stray electrical currents from the turbine shaft. If these stray electrical currents are present and are not removed, they can lead to spark erosion of the bearings, seals, or other components. Grounding brushes come in a variety of styles and mounting configurations. They can be a toothbrush type that rides radially on the shaft or a plunger type that can be mounted either radially or axially. In most cases these brushes would be installed to ride on the shaft end in the proximity of the bearing.