STEAM TURBINES

The steam turbine is an engine designed to convert the thermal potential energy of steam to rotational kinetic energy on a rotor which is free to rotate when supported by a series of axially aligned bearings. This rotational energy is then used to drive some mechanism or machine, such as an electric generator, pump, compressor, or ship propeller. Energy conversion is achieved by allowing for the controlled expansion of the high-energy (pressure and temperature) steam in a number of series arranged stages, each stage allowing a portion of the total thermal energy to be released and then converted to high kinetic energy in a fixed blade row. The resulting jet of high-velocity steam is then directed into a following row of blades which are attached to a central spindle and caused to rotate, thereby driving an attached mechanical or electrical device. There can then also be a further energy release or reaction in the rotating row.

The concept of expanding steam through some form of nozzle and employing the resulting kinetic energy to produce shaft rotational energy has been recognized for centuries, by Hero in 120 B.C. and later by Giovanni de Brance in 1629. However, it was not until the late nineteenth century that the principle was applied to some practical application—first by Sir Charles A. Parsons, who designed a 10 horsepower unit in 1894 and who in 1890 built four 75 kW units for electrical lighting. Later other designs followed: in 1889 by de Laval and 1896 by Curtis.

The units resulting from these early designs formed the basis of the modern high-capacity units, which provide much of the energy conversion capability throughout the world. Unit sizes have increased so that today there are operating units rated at 850,000 kW in a single line (tandem compound), driving a single electrical generator, and units rated up to 1,300,000 kW in a two-line (cross compound) application driving two separate but electrically interconnected generators. While this increase in unit size has continued to grow, so have the conditions of the steam which is admitted to the units. There have also been advances in the manner in which the equipment of the total cycle is configured to increase the efficiency of the conversion of the thermal energy to electric or mechanical output from the turbine units.

The physical properties of steam, pressure, absolute temperature, and specific volume are related by the "General Gas Equation." This equation provides a relationship capable of predicting the steam properties throughout the expansion passage or steam path. In examining these properties, possibly the most significant characteristic in terms of affecting the arrangement and design of the steam turbine is that as the steam expands from inlet to exhaust conditions, most often in a subatmospheric condenser, there is a dramatic increase in the volumetric flow. This increase in volumetric flow, along with the limitation placed on axial velocities throughout the unit, requires that in the lower-pressure sections either multiflow or parallel-flow stages are required. This multiflowing of stages allows increases in the quantity of steam which can be admitted to a unit and therefore allows an increase in the output which can be achieved.

Since the introduction of the steam turbine as the primary power generating unit, there have been advances in unit output. With advancing steam conditions and improvement in cycle arrangement there have also been improvements in unit and cycle efficiency.

TURBINE TYPES

Despite their apparent similarities, there are a variety of types of turbine, and they can be classified in a number of ways, the most apparent being their type of application—that is, electrical power generation, mechanical drive, or marine. They can also be classified by size, arrangement, and output. A factor having considerable influence on the details of any turbine is the type of thermal cycle in which it is to be employed, the energy source, and the steam conditions used in that cycle.

A significant advance in cycle arrangement was the use of "regenerative feed heating." Feed heating is an arrangement by which a portion of the steam entering the unit is removed from the expansion, at a number of suitable steam conditions, and used to preheat the condensate returning from the condenser to the boiler. This feed heating is applied to raise the temperature of the condensate to close to the saturation temperature of the water entering the boiler. The boiler has then only to supply the latent heat and superheat to the working fluid.

A straight condensing non-reheat turbine is one in which the steam enters the unit and expands in a series of stages to a defined exhaust pressure. As discussed above, the lower-pressure stages can be paralleled, but this has no effect on the non-reheat character of the unit. There is also the reheat-type arrangement, in which the steam, after expanding to a suitable pressure, is removed from the turbine, returned to the boiler or other reheating device, has its temperature level raised, and is then returned to the turbine to continue its expansion. Some advanced cycles have been arranged to incorporate two stages of reheating.

The reheat cycle provides considerable advantage in terms of cycle efficiency, but suffers to a small degree from the additional technical complexity associated with the addition of more equipment which can be subject to failure. However, with modern design and manufacturing practices, these potential problems are capable of being overcome.

There are a variety of turbine design configurations in terms of the steam flow. The most common form is for the steam to enter the unit and flow axially from inlet to exhaust, and while the direction of axial flow may be reversed in certain sections, this is normally done only to help balance the axial thrust developed within the unit. There are units with a steam path designed for steam to flow radially outwards from the inlet, to flow radially in and then to exhaust axially, and even to flow tangentially. However, the majority of units are of the axial flow type.

TURBINE PERFORMANCE AND EFFICIENCY

There are a number of losses which occur within the steam turbine, and these losses are associated with a number of the mechanical components comprising the unit. These losses include, in addition to the more obvious steam path, those in bearings, couplings, and valves. There are also losses associated with steam leakage, heat radiation and conduction, and pressure drops in lines from one section of the unit to another. These losses are predictable, and in terms of mechanical losses they remain relatively constant throughout the operating life of the unit. However, those losses experienced within the steam path are a function of a number of complex factors both mechanical and aerodynamic, and the efficiency within the blade rows tends to decrease with time. A portion of these steam path losses are recoverable when the unit is opened for inspection and refurbishment, but it is normally not possible to recover the full losses which have been incurred and associated with operation, except at very high costs, and such improvement can normally only be sustained for relatively short periods.

Steam path efficiency, η_{sl}, is defined as the ratio of useful energy converted to power in the unit or section to energy which is available for conversion (1). This expansion in the steam path is shown as in Fig. 1. The steam enters the stage

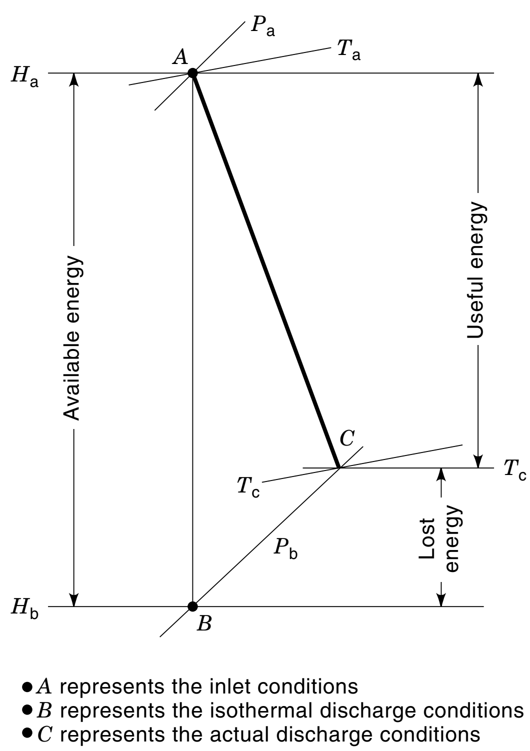

Figure 1. The steam expansion line of the turbine stage, as represented on the Mollier enthalpy–entropy diagram.

with conditions H_a, P_a, and T_a. The steam discharges from the stage with steam conditions H_c, P_c, and T_c. Had the expansion been isothermal, the conditions would have been H_b, P_b, and T_b. Therefore, the state line efficiency can be defined as

$$\eta_{sl} = \frac{\text{Useful energy converted to power per pound of steam}}{\text{Available energy per pound of steam}} = \frac{H_a - H_c}{H_a - H_b} \quad (1)$$

where

H_a is the enthalpy at inlet to the turbine section or stage at pressure P_a
H_b is the enthalpy at discharge for an isothermal expansion from P_b
H_c is the actual discharge enthalpy at pressure P_b

If account is taken of the velocity (kinetic energy) of the steam leaving the final stage in a turbine section, then this is considered irrecoverable and is known as the "leaving loss." In other than the low-pressure section, a portion of this energy can be recovered, as the kinetic energy of the steam entering the section. This leaving loss is of little significance, except in the final or exhaust stage in low-pressure sections, where the leaving velocity is carried to the condenser and represents energy lost to the unit.

The steam path losses can be considered to be influenced by several diverse factors including (a) the design philosophy used to define the various components comprising the steam path, (b) the manner and quality with which these components are manufactured to meet the design specification and then installed in the unit, and (c) the manner in which the unit is operated and maintained.

The actual losses can be divided into four principal groups, which are as follows:

Aerodynamic Losses. The steam expands through the steam path at high velocities; and it flows through the blade elements, causing a frictional interaction with the surfaces with which it is in contact. As the unit ages, there is a deterioration of this surface due to a number of phenomena which cannot be entirely avoided. These include the passage of solid particles of magnetite (Fe_3O_4) through the steam path, these particles being carried into the unit from the boiler and other portions of the steam cycle. Also the steam carried into the steam path can have chemical compounds suspended in the steam, which, as the pressure decays, come out of solution and attach themselves to the steam path surfaces, thereby increasing roughness and therefore increasing frictional losses. Another influence of these chemical "carry-ins" is that they may form corrosive compounds when deposited on the steam path component surfaces. Under certain circumstances, these compounds can cause corrosive pits to form, which cause flow disruption and therefore losses.

In addition to these surface losses, there can be losses due to high velocities above Mach 1 where supersonic losses are induced. There are also losses associated with components included in the steam path from structural considerations. Such components include tie or lacing wires, as well as notch blocks or missing blades at row closing windows. These components cannot be avoided, but modify the flow and cause disturbances in the steam path.

Steam Leakage Losses. The steam path is comprised of a series of stationary and rotating blade rows which have pressure drops produced across them and which operate with relatively small clearances, even at the highest blade tangential velocities. Steam which bypasses the blade elements does no work. Also, this leakage steam can reenter the steam path, disturbing the orderly flow from one blade row to the following, thereby inducing further losses.

In the case of shaft end packing, where the rotor passes through the stationary outer casings, the steam leaks from the internal portions of the unit to the lower pressure surrounding the casing, or to some lower pressure, where it can be returned to the steam path or cycle, but at a lower thermal energy level. Again this leakage represents a loss to the steam path.

The most common seal used in the steam turbine is the labyrinth type which consists of a series of "fins," forming a number of chambers into which the steam expands converting a portion of its thermal energy to kinetic in the chambers, this kinetic being successively destroyed by the tortuous path of expansion. The quantity of steam leaking past any series of labyrinth seals can be determined by the use of "Martin's equation" (3). A simple multitooth labyrinth seal is shown in Fig. 2.

$$m = k \cdot A_e \sqrt{\frac{P_1\left(1 - \frac{P_2}{P_1}\right)^2}{V_{s1}\left(N + \ln\frac{P_1}{P_2}\right)}} \quad (2)$$

where

m is the quantity of steam leaking in unit time
k is a constant, including a flow coefficient
A_e is the circumferential leakage area, $= D \times C$
D is the mean diameter of the seal
C is the running clearance
P_1 is the upstream steam pressure
V_{s1} is the upstream steam specific volume
P_2 is the downstream steam pressure
N is the number of series strips

As a unit operates there are a number of phenomena, including starting and stopping, load changes, and other transients which cause the rotor to vibrate to the extent the knife edge of the seal strips rub and therefore wear. This action causes the clearance to open, with a resulting increase in the leakage quantities. Therefore as a unit ages, there will be an increase in leakage flow and a resulting reduction in output.

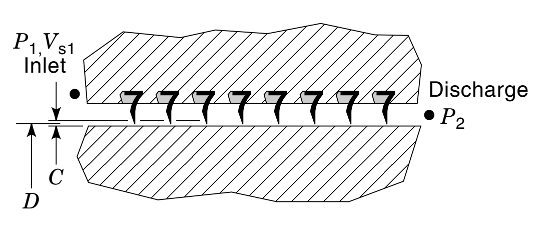

Figure 2. Parameters of steam and geometry defining the labyrinth seal.

Moisture Losses. As the steam expands from its inlet conditions, there is a gradual reduction of its energy level, associated with the conversion to rotational energy in the rotating portion of the unit of energy. This reduction of steam energy continues until ultimately moisture is formed in the steam, which is then required to transport these moisture particles through the steam path.

Associated with this moisture, there are two forms of energy loss within the steam path. First is the kinetic effect. This consists of the energy the steam particles must expend on changing the direction of the water as it passes through the blade rows. Second, are the losses associated with the collision of these water particles on the rotating blade rows and their effect of retarding or "braking" the rotational energy of the blades.

While these losses are not high, they do increase with the increase in the moisture level of the steam (4).

Leaving Losses. As the steam exhausts from a turbine section, it contains velocity energy. In stages other than the last, in any section this velocity energy is carried into the following stationary blade row where a portion of it can be utilized. In any section of a unit, other than the low-pressure sections, this does not represent a significant loss. However, in the last or exhaust stages, velocities are high and these losses are significant.

In fact the turbine exhaust loss comprises three distinct components which contribute to the total loss. The first is the component velocity, which is proportional to the axial leaving velocity squared. The second is the frictional loss within the exhaust hood, and the third is a loss which occurs only at low exhaust velocities and is associated with the steam being unable to clear the exhaust annulus and causing a pressure buildup just downstream of the last stage blades.

Once an exhaust configuration is selected by the design process, the magnitude of the losses cannot be changed except by replacing the exhaust blades with one providing a greater annulus area and therefore lowering the axial exhaust velocity. This is obviously not an easy process, but in some cases it is justified by the savings in fuel cost.

Power Cycle Efficiency. The expression Eq. (1) for the steam path efficiency does not take account of the mechanical losses which occur within the various components of the unit such as bearings, couplings, and so on. Therefore, a quantifiable characteristic was required which would allow the total efficiency of energy conversion of the system to be defined. Two such characteristics are available to indicate the performance level or potential of the turbine generator or turbine-driven unit. These are (a) the "heat rate," which is a measure of the quantity of thermal energy required to generate a specific electrical output for a given period of time (5), and (b) the "steam rate," which provides a measure of the pounds of steam operating between specific thermal conditions required to generate a defined amount of electric or mechanical output for a specified time.

These are defined as

$$\begin{aligned}\text{Heat rate (HR)} &= \frac{\text{Thermal input}}{\text{Electrical output}}\\ \text{Steam rate} &= \frac{\text{Quantity of steam entering the unit}}{\text{Electrical or mechanical output}}\end{aligned} \tag{3}$$

In the basic reheat cycle shown in Fig. 3, the heat rate can be defined by the following expression:

$$\text{Heat rate (HR)} = \frac{m_1(H_1 - h_c) + m_r(H_{hr} - H_{cr})}{kW} \tag{4}$$

In this expression, the term $m_1(H_1(- h_c)$ is the heat added to the feed water in the boiler, and the term $m_r(H_{hr} - H_{cr})$ is the heat added to the steam in the boiler reheater section. The output is given as kW, being the electric generator output. This term can be adjusted to reflect the power consumed by the boiler feed pump and other auxiliaries carried by the plant. There are therefore a number of definitions of heat rate in use. This term also takes account of the generator efficiency because it includes the actual output from the generator.

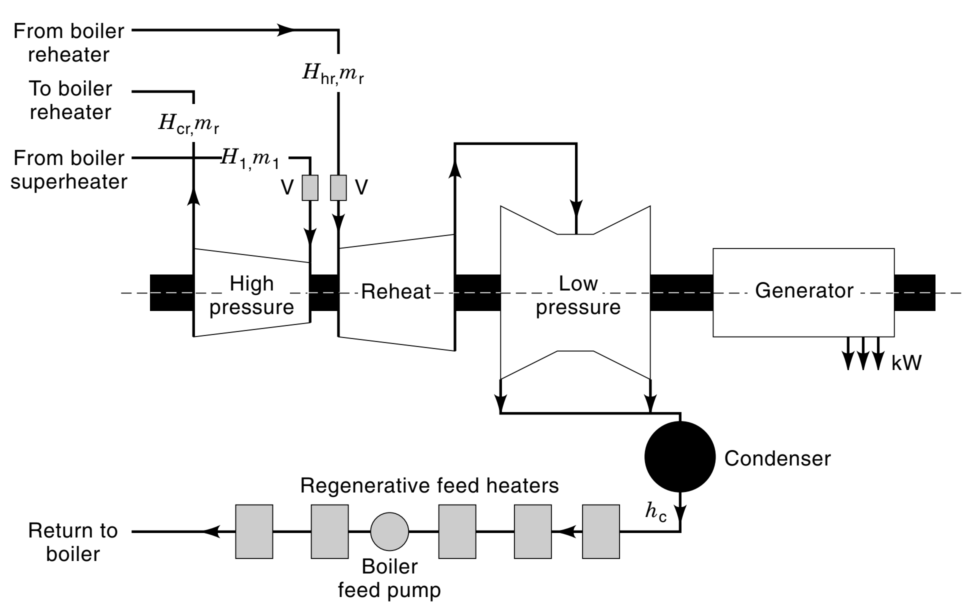

Figure 3. The steam and flow conditions of a reheat steam turbine, showing steam H enthalpies, water h enthalpies, and flow quantities "m" at various locations throughout the cycle. The unit electric output "kW" is also shown.

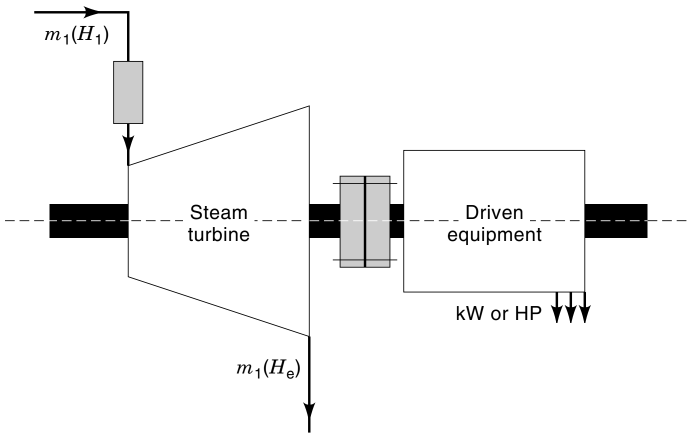

Figure 4. A smaller industrial or marine type unit, showing steam enthalpies H and flow quantities m together with the output in kilowatts "kW" or horsepower "HP."

A typical smaller installation, as required for industrial or marine use, is shown in Fig. 4. Here the steam rate would be defined as

$$\text{Steam rate (SR)} = \frac{m}{\text{kW or HP}} \tag{5}$$

In this definition, the input, in thermal energy, is measured as the steam flow m required for a specified time between steam properties H_1 and H_e to generate an electric output kW or mechanical output HP for a specified time period.

THE TURBINE STAGE

The basic turbine stage consists of two blade rows. The first is stationary, consisting of a number of vanes which are arranged to produce passages between adjacent elements. These vanes are attached to the stationary portions of the unit; they both expand the steam by allowing for a controlled pressure drop, and they are oriented so as to direct the resulting jet into a following row of rotating blade elements. These rotating elements provide a passage which deflects the steam through a turning angle, and they discharge it to enter the stationary blade row of the following stage, or to the condenser in the case of the final or exhaust stage. In changing the steam direction from inlet to near axial discharge, these rotating blade elements have a thrust developed on them, which is transmitted to the central spindle or shaft, causing it to rotate. Therefore, in its simplest form, the stationary blades convert the steam's thermal potential to kinetic energy, and the rotating blades convert the jet kinetic energy to rotational kinetic energy, which is available to drive any machine or mechanism connected to the shaft.

The description of the blade rows represents a simplified explanation of the function of the two blade rows, and it neglects the effect of reaction or pressure drop which is designed to occur in the rotating blade element. In the practical turbine stage, there is in fact a pressure drop of some magnitude in the majority of rotating rows.

The turbine stage consists of a series of stationary and rotating components, through which the working fluid expands. This steam, at entry to the stage, has a pressure which drops in the stationary row and has a further drop in the rotating. Because a pressure differential exists across row blade rows, there is a tendency for the steam to leak or bypass the blade rows if an alternate path of less resistance can be found. The occurrence of leaking steam therefore represents a condition in which the steam gives up its thermal energy and does no work in its expansion (a throttling or nonexpansive expansion). The extent to which this leakage occurs must be controlled to maintain stage efficiency. This individual turbine blade rows are therefore normally arranged to provide systems which minimize this leakage and thereby maximize the work done in the stage.

The thermal energy of the steam must be released in discrete steps, selected to maximize the efficiency of expansion. This control of pressure and enthalpy drop is achieved by ensuring that the discharge area from each blade row is controlled within close tolerances and is a function of the quantity of steam flow and the steam properties at inlet to the stage. The discharge area required on any blade row is determined from the equation

$$m = AC_{\mathrm{d}}\sqrt{2g\frac{\gamma}{\gamma-1}P_1V_{\mathrm{s1}}[R^{2/\gamma} - R^{(\gamma+1)/\gamma}]} \tag{6}$$

where

m is the quantity of steam flowing in unit row
A is the row discharge area
C_{d} is the discharge coefficient
g is the gravitational constant
P_1 is the row inlet pressure
P_2 is the row discharge pressure
V_{s1} is the row inlet specific volume
γ is the ratio of specific heats of steam

In the practical turbine this area is achieved by defining the form of the individual blade vanes forming the expansion passage and then controlling their spatial relationship relative to each other in the final assembly.

Impulse and Reaction Designs

There are two distinct philosophies concerning the manner of releasing the thermal energy within the turbine stages. The first can be defined as the pure impulse stage, in which the total available thermal energy assigned to the stage is released in the stationary blade row. This energy conversion produces a high-velocity jet which discharges from the row and is directed into the rotating elements. The rotating blade row is then designed to convert this kinetic energy to thrust on the individual blade elements by changing the momentum of the jet. This change of momentum produces a tangential thrust on the individual blades, which in turn drive the rotor to which they are attached. In the pure impulse stage the pressure at discharge from the rotating blade row is the same as at entry.

The second form of energy release is within the reaction stage. In this design a portion of the total thermal energy is released in the stationary row, in precisely the same manner as in the impulse stage; the remaining energy is released in the rotating row, as a reaction on the rotating elements. A stage in which half the thermal energy is converted to kinetic in the stationary row, with the remainder converted in the rotating row, is called a 50% reaction stage.

In practice the reaction stage is normally designed so that 50% of the total energy is released in each of the stationary and rotating blade rows. Also, in actual designs of modern units the true impulse stage is only rarely used, and the majority of stages have some small degree of reaction sufficient to ensure a positive pressure gradient throughout the steam path. The degree of reaction used is dependent upon the philosophy of the individual designer of the turbine.

Another stage design, often used at inlet to high-pressure turbine sections, are those in which a single pressure drop occurs in a stationary blade row, followed by a rotating blade row. Then an exhaust from this first rotating row there is a direction changing stationary blade row, which is followed by a second rotating blade row. Because these rows tend to have poor efficiency, they are not used extensively in modern turbines.

Turbine Stage Velocity Ratio and Efficiency

There is a well-defined relationship between individual stage efficiency and the ratio of blade tangential velocity to steam jet theoretical or isentropic velocity (5). This relationship is shown in Fig. 5 for a pure impulse, a 50% reaction, and the two-rotating-blade row design of stage. From these curves it can be seen that for the pure impulse stage, that a ratio of 0.50 provides the optimum efficiency, whereas for a stage having a 50% reaction design, that a ratio of 0.707 is optimum. The two-stage design is optimum at 0.25.

The implication of these curves is clear. It can be seen that for any blade tangential velocity, which is a function of the stage diameter and rotational velocity, the pure impulse stage will require a larger steam isentropic enthalpy for any specified blade tangential velocity. Therefore, the reaction stage will utilize a smaller enthalpy drop than the pure impulse stage, which will require a steam velocity which is twice the

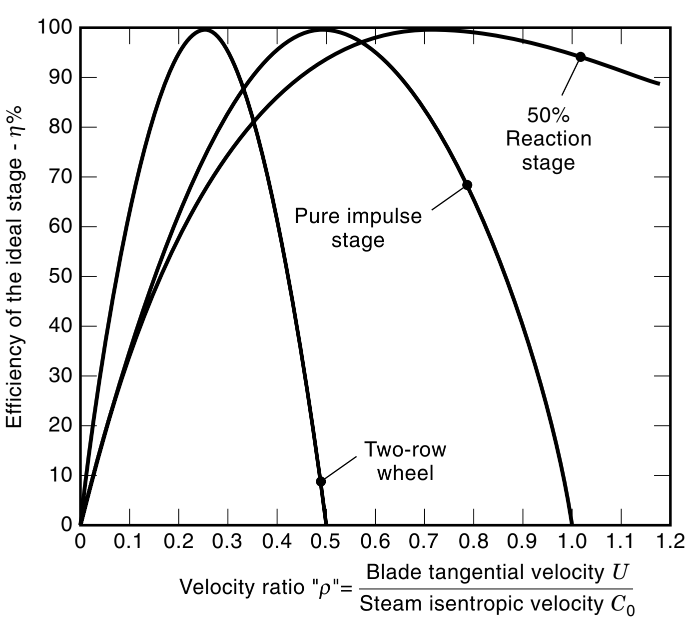

Figure 5. The variation of ideal stage efficiency as a function of the stage velocity ratio ρ for various stage configurations.

blade tangential velocity, whereas in the 50% reaction stage the velocity of the steam for maximum efficiency will be smaller. In fact the 50% reaction stage will require twice the number of impulse stages to accommodate the same thermal energy range. There are, however, few pure impulse stages built in modern units, and a realistic reaction for the "impulse stage" provides for a velocity ratio above 0.5; as a result, the number of impulse stages are somewhat larger, and the ratio of reaction to impulse stages is about 60 to 65%.

The geometry of the individual blade rows is influenced by the magnitude of the pressure drop in the stage. The manner in which this drop is divided between the stationary and rotating blade rows, together with a consideration of the number of stages between any bearing span, has a direct effect on the efficiency.

Consider the two stages in Fig. 6. Figure 6(a) shows the basic arrangement of the impulse stage which comprises the wheel and diaphragm construction; in this design the stationary blade row is carried in a diaphragm, which forms its stationary stage seal on the central portion of the rotor. There are also seals above the rotating blade row, even though the pressure drop is relatively small. In Fig. 6(b) the typical arrangement of the reaction stage is shown. Here, because of the larger number of stages, it is necessary to seal the stationary blade row at a considerably larger diameter; but because the pressure drop is smaller, there is a smaller leakage quantity which bypasses the blade row. Again, a seal is used above the rotating blade, which in this case has a somewhat higher pressure drop. The total leakage losses in the impulse and reaction designs for any specified energy range are at about the same level.

In both stage designs, it can be seen that above the rotating blade row, there is a circumferential band of material attached to the outer surface of the blade vane. These cover or shroud bands are included in the design to limit any flow which would occur through the tip section of the vane. These bands are also used to provide seals or a platform upon which a radial seal can be produced.

A stage can also include one, two, or, in certain older designs, three lacing or tie wires. These wires provide a continuous or semicontinuous coupling between adjacent blade elements, and they are included to transmit vibratory forces developed in one element to others to which it is connected. These ties tend to be limited to the blade rows with longer vanes, because these blade elements have a series of lower natural frequencies and are more easily excited by alternating loads which are produced in the steam path. These wires are included in the design to achieve structural integrity in those stages where the possibility of vibratory loads could induce some form of fatigue failure. These wires also include efficiency losses, due to drag, into those stages which employ them.

The individual blades are attached to the rotor by the use of specially designed fastening devices which are produced integral with the blade vane. There are a variety of forms for these fastenings. Each turbine manufacturer has a series of designs available which can include geometries which allow the blades to be assembled to the rotor entering it in the axial, radial, or tangential directions. There are advantages of security, assembly, and cost associated with each design, and the turbine manufacturer selects the best suited design for each row, after careful evaluation.

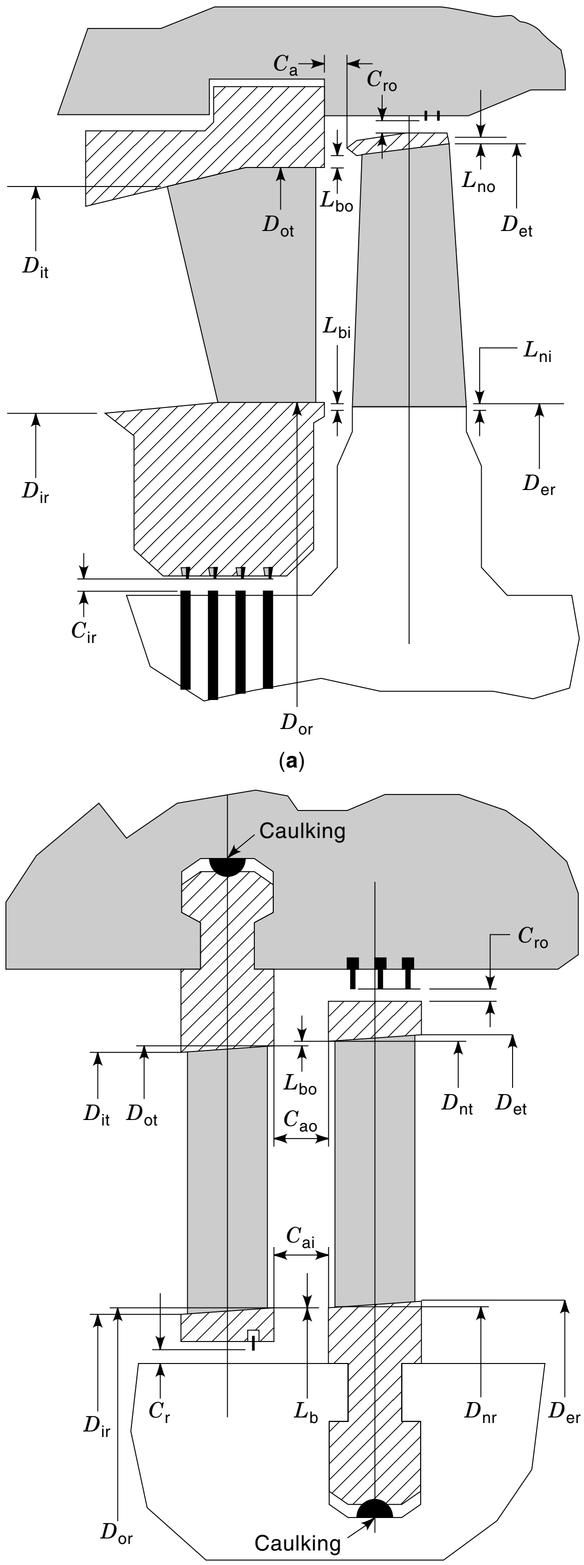

Figure 6. (a) The basic impulse stage showing component arrangement and major dimensions. (b) The basic reaction stage showing component arrangement and major dimensions.

Vector Diagrams for the Turbine Stage

Stationary and rotating blade vanes are designed to produce a profile which will expand the steam by providing a passage through which it can flow, change direction, and exhaust at the correct pressure and velocity to enter the following blade row. The design of profiles is one of the more complex aspects of turbine engineering. This is because it requires that profiles be defined which can be produced at realistic costs with existing machining capabilities and which will induce a minimum of energy losses into the stage. These blades must also have sufficient mechanical strength so they will last for the predicted life of the turbine unit.

As a preliminary step to defining the requirements of a blade vane, it is necessary to establish the required profile inlet and discharge angles (8). These angles are normally determined during the design process by the construction of "vector or velocity diagrams." These diagrams establish the angles which are required of the profile to accept the inflowing steam with a minimum of incidence loss, and they also provide a controllable exhaust which will ensure that the steam discharge area and direction from the row are correct.

Shown in Fig. 7 are velocity diagrams for both a pure impulse and a 50% reaction stage. In these diagrams the blade tangential velocity U is the same on both designs as are the stationary and rotating blade discharge angles α_1 and β_2. Figure 7(b) shows the vane profiles required by these two velocity diagrams to allow for incidence free entry and expansion through the resulting blade expansion passage.

Form of the Blade Vanes

The two blade rows comprising each stage are designed to accept the inflowing steam, deflect it through the desired angle sufficient to allow the required degree of energy conversion, and then deflect it into the following blade row, either rotating or stationary. These functions are achieved by ensuring that the profiles form the correct shaped passage between the elements, and also form the correct area at discharge from each row.

In an actual unit, the blades must be manufactured to achieve both the required vane profile and also ensure that as they are mounted to the major components which give them location within the unit, so their spatial relationship both to elements within their own row and rows both up- and downstream is correct.

TURBINE CONTROL

The turbine control system is designed to control both unit rotational speed and output. For large utility units, connected to large power grids, it is necessary for the governing system to control speed to ensure that system frequency is maintained. The control system must also react in response to system load demands, to adjust control valve settings and admit the required quantity of steam.

In non-reheat units, there are valves located at the inlet to the steam turbine unit; these valves open and close to control the admission of steam to the high-pressure section. In reheat units, there are large quantities of steam contained between the cold outlet and the hot reheat return to the unit. Therefore, it becomes necessary to incorporate valves between the

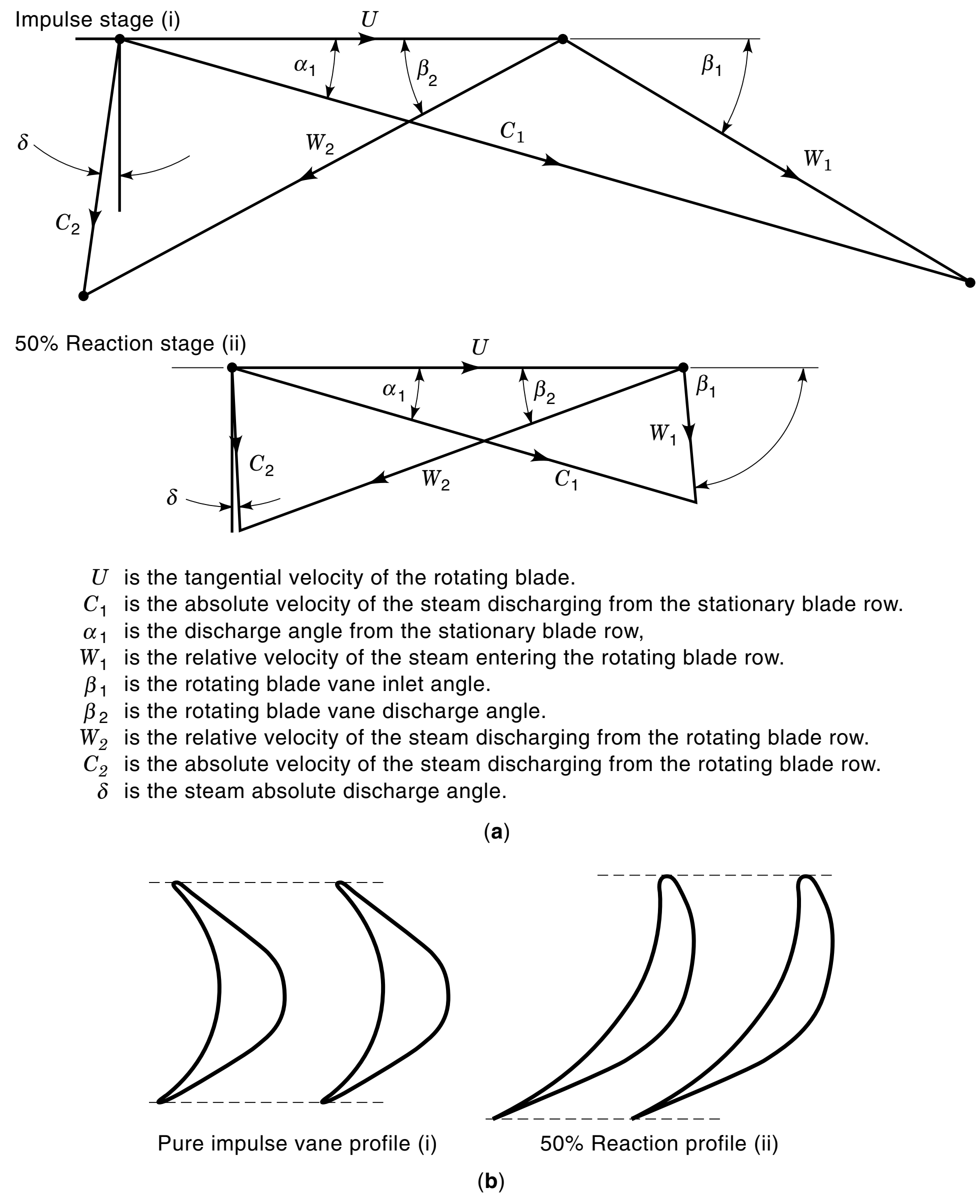

Figure 7. (a) The velocity vector diagrams for the basic impulse stage (i) and 50% reaction stage (ii). (b) Vane profiles for (i) the pure impulse stage with a constant expansion passage width and (ii) the 50% reaction stage with a converging passage.

boiler superheater and reheat section inlet, so that in the event of the unit being disconnected from the power grid or other driven device, this quantity of steam cannot reenter the turbine and contribute to its overspeed. These intermediate valves incorporate both stop and control functions.

A second major function of the control system is the protection of the turbine-generator unit against both internal disruptions and system disturbances. Therefore, the control system is designed to react to disturbances, by disconnecting the unit from the grid. When disconnection occurs,the control system is designed to limit overspeed transients (to levels which will allow the unit to continue to carry station auxiliaries) and then to resynchronized and connected to the grid as the disturbance is cleared. Alternatively, if the overspeed is sufficiently high, the control system must regulate the overspeed and bring the unit to a low turning speed, so the unit can then be restarted and re-connected to the grid after the disturbing condition has been corrected.

There are two basic control systems in use, which are described in the following section.

Mechanical Hydraulic System. The mechanical hydraulic control (MHC) system consists basically of a governor system, connected to and driven from the main turbine rotor, and is therefore sensitive to its rotational speed. In large utility units connected to power grids, the turbine rotational speed is governed by system frequency, and therefore the governor becomes in effect a load control device, which actuates control valve position through an oil-pressure-sensitive system, in which the oil pressure varies in response to unit speed. This governing system controls the positioning of the valves (through servomotors), which open and close the control valves to vary the quantity of steam admitted to the unit and thus vary the load which can be generated.

The control system oil, which is controlled by rotor speed, will, upon loss of electric load and in response to speed increases, cause the governor system to close the main and intermediate control valves. This is a fast-acting system and will, on a well-designed unit, cause the valves to go rapidly to a "closed" position.

Electro-Hydraulic System. The electrohydraulic control (EHC) system employs solid-state technology together with a high-pressure hydraulic system. This faster-acting system is more responsive than the MHC system and provides not only a faster response, but a more reliable system. The EHC system is also more flexible, allowing interconnection with other external systems and functions such as "overspeed anticipatory devices," which cause main and intermediate stop and control valves to close if the initial rate of rotor acceleration, on loss of load, indicates an emergency overspeed will result. This EHC system also allows remote control and remote diagnostic monitoring and is capable of being computer-controlled from the control room.

The high-pressure oil system used on the EHC design has strict requirements for cleanliness and fluid stability. Because of the possible leakage problems associated with a high-pressure oil, it is necessary to use a fire-resistant fluid in this system.

A well-designed EHC system is capable of taking the main and intermediate stop and control valves from a fully open to a fully closed position in about 0.2 s.

Stop and Control Valves

The steam turbine is furnished with a number of valves, each with various functions and operating modes. The largest and most significant valves in terms of unit operation are the main and intermediate stop and control elements.

Under emergency conditions, when load is rejected, if the unit overspeed does not exceed 8 to 10% normal operating speed, the stop valves will remain open. However, in the event the unit goes into an emergency overspeed, in excess of about 110% normal, then these valves will close and remain closed until the unit is restarted. These main stop valves normally have a small bypass function, a valve which opens to admit prewarming steam as the unit is started. This bypass valve normally has the capability of bringing the unit to speed and synchronizing on full arc admission before the main stop valves are opened.

The control valves are located behind the main stop valves, often within the same valve body, to reduce frictional losses. These valves have a function of controlling unit speed and also open and close in response to load demands.

Full and Partial Arc Admission

There are two modes for admitting steam to the turbine unit at part load; these are full and partial arc admission. At full load these two admission systems operate in essentially the same manner, with the valves in their "wide open" position. At this valve setting, steam is flowing through a complete inlet arc. However, at part load the entire valve flow area is not required, and the inlet area of the valves must be reduced to admit less steam. In the unit designed for full arc admission, each of the control valves partially close, thereby throttling the steam and thus destroying a portion of the available thermal energy. With the partial arc admission system, only the top (or last to open) valve closes, leaving the remaining elements to operate wide open and therefore not suffering from the energy degrading effects of throttling.

As load demands reduce further, the full arc admission unit again closes all valves by some amount, to limit the steam admission. In the partial arc design, the closure occurs on only one valve (or two, if sufficient area reduction cannot be achieved with the closure of just one).

Therefore, sequential closing of the valves offers advantages in terms of the unit performance at part loads. However, such a partial admission unit will suffer problems associated with partial arc admission where the control (first) stage blades must pass alternately through arcs of high-energy steam jets and dead bands. This effect causes high-impulse loads to be placed on these blades and can result in damage.

Sliding Pressure Control

An alternate method of steam admission into the control stage is "sliding pressure operation" in which the initial pressure of the steam produced in the boiler is reduced with reducing load to a condition that the control valves can be left wide open at partial loads. This method of steam admission has the effect of requiring less energy to be expended in the boiler, to raise the conditions of the steam to one suitable for admission to the turbine unit, and does not suffer from the disadvantage of having to throttle the steam at inlet and thereby sacrifice a considerable energy penalty.

There is a requirement for close control of the boiler and turbine so that both respond to the demands of each other, both accurately and in a timely manner.

BEARINGS AND LUBRICATION SYSTEM

The turbine, irrespective of its driven load, must be supported on a system of fluid film bearings, which in multishaft arrangements must be aligned both vertically and transversely to minimize stresses introduced in the shaft. This will also help eliminate the possibility of introducing vibrations into the rotor. These bearings must maintain alignment during transient conditions, at startup and shutdown, when the unit passes through critical speeds and will exhibit higher than normal levels of vibration.

The Axial Bearings

The axial bearings must carry and support the rotor weights without excessive deflection. The bearings are sized so that an oil film developed between the journal and bearing pads prevent metal-to-metal contact, thereby causing no wear on the journal or bearing pads themselves.

Bearings can be subject to various forms of damage; but provided that the oil is maintained at its design defined level of cleanliness, the bearings are aligned, and oil supply is maintained, they will operate with a minimum of maintenance.

The Thrust Bearings

The thrust bearings are used to carry the axial thrust developed in the unit steam path and are also used to provide axial location of the rotor, which is necessary to allow axial clearances to be maintained at design values. A thrust bearing is supplied for each rotor; or, in the case of solidly coupled elements, one bearing which is sized to carry the total axial thrust is supplied. The thrust collar produced on the rotor is aligned to a series of pads carried in a separate housing which is located in the stationary portion of the unit, and therefore provides positive position of the rotating and stationary portions of the unit.

The Lubrication System

A reliable lubricating oil system is required to ensure that the bearings are able to operate in a manner which maintains alignment and minimizes the frictional losses which can occur. This system supplies oil to each of the axial bearings, the thrust bearing, and any other location or piece of equipment which requires it.

The major component of the lubricating system is the oil tank, which is a reservoir holding sufficient oil to charge the system at all loads and under conditions of emergency operation. In many applications there are oil pumps which are driven directly from the main rotor. In addition, there are backup systems: Normally, there is one which operates on an ac supply driven by power produced in the turbine, and there is also a final backup system which utilizes batteries. This dc backup system is only used in emergencies when the ac supply is cut off or when the unit is totally disconnected.

The oil tank contains a filtration system which receives the oil returning from the unit, removes any particulate matter, and returns the oil to the tank where it is reused. It is normal practice for the condition of the oil to be monitored, both for purity and temperature rise within the various bearings. Higher oil temperature rises can be an indicator of pending bearing problems.

The steam turbine comprises two major groups of elements. These are the stationary and rotating portions of the unit. Both sets of components are precision engineered and are designed for operation up to 200,000 h (30 years) without the need for major replacement. However, it is recognized by both designers and operators that some components are consumable and will require replacement during the life of the unit.

The Rotor

The main rotating component of the steam turbine is the rotor. The rotor comprises a central shaft, along with blades which are attached to it, and these blades may carry stage hardware such as cover bands and tie wires. Rotors are constructed from monobloc forgings (a central spindle with wheels shrunk onto this spindle), or they can be built from individual forgings which are weld connected to form a single central shaft onto which the various components are mounted.

The rotor is subjected to a number of stresses; these are either (a) steady, due to rotation, or (b) alternating, due to steam forces developed with the steam flow path. There are also thermal stresses set up in the unit, which are most severe during thermal transients and can ultimately initiate cracks at regions of high stress concentration. The rotating components of a unit tend to be more susceptible to damage than the stationary portions.

The Casings

The casings are the most important stationary components and are designed to achieve various functions. First they must be of sufficient strength so they can withstand the pressure differentials which are produced across their walls. In the case of the high and intermediate or reheat sections, there is a high pressure within the unit. In the case of low-pressure sections, the inner sections of the casing can be subjected to both a positive and a negative pressure gradient from inside to outside the unit. For double-casing low-pressure designs, the outer casings must be able to resist the vacuum force developed across their outer walls.

The high- and intermediate-pressure casings are designed to be able to contain a rotor burst and therefore prevent the rotor entering the power house. In the case of a low-pressure section the probability of rotor containment is low.

The Diaphragms

In the case of wheel and diaphragm stages (impulse type) and in many low-pressure sections, the stationary blade rows are located in and are carried by a diaphragm, which is itself located in the casings, either outer or inner. These diaphragm elements can have large pressure drops developed across them and can have an inner web which extends down from the blade row to the rotor, at which interface point it is necessary for the diaphragm to carry seals to prevent excess steam leakage from one stage to the next.

OPERATING PROBLEMS

During operation, the steam turbine can be subjected to a number of phenomena which have the potential to introduce conditions and situations resulting in damage. These conditions, if not corrected, could require that the unit be removed from service for extensive maintenance, or could even cause mechanical failure to the extent that components will require expensive replacement to correct the condition. These conditions include those associated with the steam environment, the high stress levels which are developed within the various components themselves, and the possible ingress of chemical ions which have the potential to form aggressive compounds capable of causing corrosive-type damage. In addition to these influences which are associated with the turbine-generator unit, its design, and its operating environment, there can also be system disturbances which can introduce a number of problems. During the design phase, the design process attempts to anticipate and allow for these phenomena in selecting and dimensioning the individual parts of the unit, but it is always difficult to predict both the severity and frequency of these conditions.

The following factors are among the more common conditions causing unit damage which can require component replacement or repair, possibly resulting in a forced outage situation or the extension of a normal maintenance outage.

Steam Environment Effects

The steam environment in many units introduces conditions in which the temperature reduces the mechanical properties of the materials of construction, and the pressure results in high-pressure drop loads which must be carried by the components themselves, resulting in high stresses.

At the lower-pressure conditions, moisture forms in, and is transported by, the steam and is deposited on the various steam path components. This deposited moisture has the potential to cause various forms of erosion such as "worming," "washing," and "impact." Each of these mechanisms is capable of removing material from the surface of the various components of the steam path and casing, introducing losses in efficiency and also introducing the possibility of mechanical failure when sufficient material has been removed.

Stress Level Effects

Stresses are induced in the various components of the steam turbine due to a number of different phenomena. The most obvious are those induced by the effects of rotation. There are, however, other phenomena, particularly in the steam path, where high loads are induced by the flowing steam and its change of momentum in flowing through the blade rows. There are also temperature change considerations, where the rapid increase or decrease of temperature will induce thermal stresses into many components which cannot change temperature throughout their total volume to accommodate the changes in expansion rates.

These various stresses will combine and are often amplified by vibratory effects to induce levels of stress in many components which can ultimately fail. In the higher-temperature sections of any unit, the presence of both high temperature and high stresses has the potential to induce creep into various components, ultimately leading to mechanical rupture.

Speed and Load Transients

During normal operation there are often conditions introduced into the total system which will cause the unit to trip and enter an "overspeed" transient. While the main and intermediate stop and control valves will respond to this condition, it is normal for the unit to reach an overspeed condition, which induces centrifugal stresses into various components, while they are still at operating temperature. This condition can consume a portion of their remaining life. The initial design of these units anticipates the occurrence of, and allows for, these emergency conditions. However, there is a limit to the number of such incidents which can be tolerated.

Load changes will normally introduce a change of temperature distribution throughout the steam path, resulting in temperature variation within the various components. There are limits set by the manufacturer for the rate of change which is acceptable, because these affect the thermal stresses within both stationary and rotating components.

Ingestion of External Matter

The steam power cycle is a complex arrangement of a number of mechanical components, many operating above, and a number operating below, atmospheric pressures. These pressures above and below atmospheric cause a continuous leakage, both into and out of the working fluid cycle.

The leakage out of the cycle must be made up with replacement fluid, which must be purged of any contaminants it may contain. This purging or washing operation does have the potential, if not continuously controlled, of introducing compounds into the working fluid with a potential to form or promote the formation of corrosive compounds. Similarly, those elements which leak into the system are most commonly air (which contains oxygen) and water at condenser tube leaks.

Water which leaks into the condenser, normally termed "raw water," contains a number of impurities, particularly if the condenser is cooled by "brackish or salt" water.

Once a corrodent has entered the turbine steam path, it will eventually come out of solution in the steam path, possibly forming aggressive compounds which will concentrate at various locations and then has the potential to cause corrosive pitting, or will induce stress corrosion cracking or contribute to corrosion fatigue.

Water ingestion can be another cause for damage within the unit. The many lines carrying steam, water, and a steam–water mix which are connected to the turbine unit can, under certain conditions, cause water to flow back into the steam path. This water has both "quenching" effects, which cause high thermal stresses in both stationary and rotating components, the rotating being the most sensitive. Reentering water can also produce excessive impact loads on the rotating blade elements, capable of their total destruction.

Many lines which are connected to the turbine body proper and have access to the steam path are equipped with "nonreturn" valves. These valves are included to prevent the reentry of water which could collect in these lines under certain load conditions or on unit "shutdown," when condensate will drain to the lowest points to which they have access.

Vibratory Loading

The turbine-generator unit is a high-speed machine, operating at high direct stress levels and subject to a number of vibratory loads introduced into the unit. These loads can be a consequence of variable flow in the steam path and can be caused by rotating with unbalanced loads or by misalignment of the various portions. These vibratory loads can transmit themselves back into the unit and induce failure.

Many of these phenomena are predictable and are taken into account during the design of the unit. However, it is entirely possible for stray and unpredictable exciting forces to exist, which even at low amplitude have the capability of inducing failure if they are coincident with the fundamental or lower harmonics of the various components of the unit.

Unit Alignment

A multisection turbine unit comprises a number of individual sections, each comprising an outer and possibly an inner casing containing the rotor. These casings are subjected to an internal pressure which in other than the low-pressure sections is above atmospheric.

In the event that there is a vertical or transverse shift of the casing, this will not affect the rotor alignment, unless the bearings are located within the casing, a design feature of many low-pressure sections. However, such casing movement can introduce "rubs" between rotating and stationary portions, causing uneven heating, and therefore it has the potential to introduce problems associated with alignment.

In the case of bearing movement, whether this bearing is located in the casings or more likely is carried by a separate pedestal, there will be misalignment between the rotor sections. This movement will result in additional bending stresses being set up in the central portion of the rotor. This can introduce vibratory problems and will result in additional stresses and shear forces introduced into the coupling flanges.

External System Disturbances

External system disturbances are not a common cause of unit failure or damage. However, in the event of a generator short circuit, causing high-amplitude oscillations, the turbine rotor could be damaged. Similarly, a distribution line disturbance, grounding, or lightning strike remote from the station can be transmitted back to the turbine which will possibly reject load and enter into an overspeed transient. This can induce high levels of shaft torque which has the potential to damage the turbine unit.

STARTING AND STOPPING STEAM TURBINES

The starting and stopping of steam turbines can be a complex operation. The complexity is introduced by the effect of temperature and the design requirement of not exceeding temperature ramp and cooling rates which introduce excessive thermal stresses into a number of components. For this reason the manufacturer will normally specify both heating and cooling rates, which, if exceeded, will consume a portion of the life of these major components.

In addition to the concerns for thermal stress, the turbine unit will normally pass through a number of critical speeds both during “startup” and “shutdown.” It is normal practice for the operator to approach these speed levels, then to accelerate through them as rapidly as possible.

MATERIALS FOR TURBINE CONSTRUCTION

The steam turbine is designed to operate for periods of years between being opened for inspection and the undertaking of any maintenance refurbishment which may be required. During these operating periods, the components are subject to a number of complex stresses, which are induced by rotational speed, high temperatures, and pressure differentials. Therefore, the material from which the various components are produced must be of high quality and must meet design requirements in every detail.

The design engineer specifies the materials to be used for each particular application. For the major components, the specification will indicate chemical composition, mechanical properties, the method of manufacture, and the cycles and form of heat treatment to which the material is to be subjected to meet both short- and long-term requirements.

The most common and consistently used materials are the alloy steels, the majority of which are 2.25% to 12 to 13% chromium, and are used for all major stationary and rotating components. While many of these steels conform very closely to national standards, they are usually customized by the turbine manufacturers to suit their individual requirements, the actual materials being based on extensive research into the long-term characteristics when operating in environments similar to those encountered in the turbine unit.

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STEEL INDUSTRY. See METALS INDUSTRY.

STEEPEST DESCENT ALGORITHM. See ADAPTIVE FILTERS.