NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

INDEX

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

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3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-1 - Definitions

0.0 INTRODUCTION

This lesson will define several terms that are commonly used in referrence to steam power plants.

1.0 INFORMATION

Willan's Line

We have mentioned previously that the turbine output varies with steam flow. In the early days of turbine design it was discovered that if a graph was plotted with steam flow, lb/hr. and generator load, k.w. as co-ordinates resulting curve for a throttle governed turbine was a straight line. This line has been given the name "Willan's Line," figure 1. You are no doubt aware that even at no-load the turbine consumes some steam. The point at which the Willan's Line intersects the y-axis gives the no-load steam consumption.

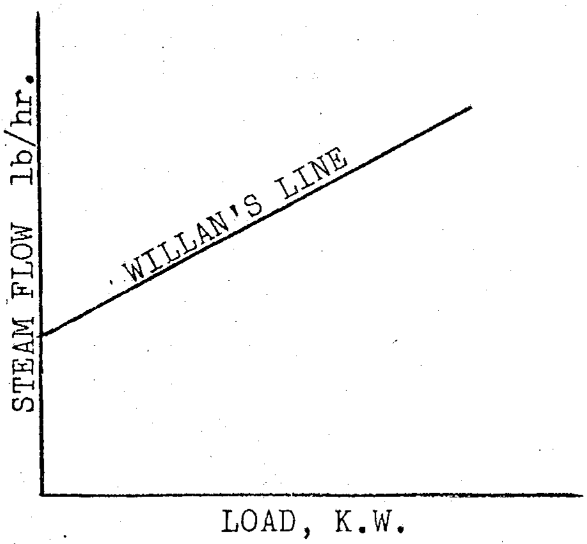

Figure 1 Willan's Line

Figure 2 illustrates different Willan's Lines. The solid black line shows the results for a turbine with an infinite number of governing valves. The straight broken line depicts the situation for a turbine employing throttle governing. The crooked broken line shows what takes place when nozzle governing is employed. The throttling losses are not as great in nozzle governing and therefore the steam flow approaches more closely the ideal situation for an infinite number of governing valves. The point where all three lines meet indicates the situation when all governing valves are fully open.

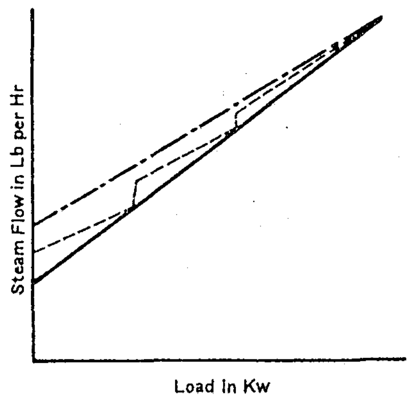

Figure 2 Willan Lines

Heat Rate

When referring to the performance of steam cycles the term "thermal efficiency" is of practically no importance to either the designer or the operator. Instead a term called heat rate is used to guage plant performance, and may be defined as follows:

Heat Rate is the energy input to a plant in Btu divided by the energy output, in k.w. hr.

Plant performance can be expressed as either gross station heat rate or net station heat rate.

Gross station heat rate =

$$\frac{\text{total heat produced by fuel burn-up Btu.}}{\text{total power produced by generator, k.w. hr.}}$$

The power output in this case is measured at the generator terminals, and does not take into account the power required to operate auxiliaries such as C.W. pumps, condensate pumps, B.F. pumps, fans etc. At rated load on the generator, these auxiliaries require from 4.5 to 6% of the generator output for a large modern conventional station and about 10% of generator output for Canadian type nuclear stations. At lower loads the percentage is greater. Net station heat rate is based on the output of the station after deducting the power consumed by the auxiliaries.

Net station heat rate =

$$\frac{\text{total heat produced by fuel burn-up, Btu.}}{\text{total heat produced, k.w. hr. - k.w. hr. used in plant}}$$

It is desirable to keep the heat rate of a unit as low as possible. From the above equations it is evident that the net station heat rate is always greater than the gross station heat rate.

Net station heat rates have decreased steadily during past years. In the early 1920's , a net station heat rate of about 45,000 Btu/k.w.hr. was considered staisfactory; modern units have heat rates of about 9000 Btu/k.w.hr. for conventional stations and about 11,000 Btu/k.w.hr. for nuclear stations.

Steam Rate

Steam rate refers to the lbs. steam required to produce one k.w.hr. or abbreviated: steam rate = lbs./k.w.hr.

Thermal Efficiency

Thermal efficiency has been defined in the course "Heat and Thermodynamics" as output divided by input. Therefore:

Thermal efficiency =

$$\frac{\text{total power produced - k.w.hr. used in plant}}{\text{total heat produced by fuel burn-up, Btu.}}$$

The units in the denominator and numerator should be the same so in order to change k.w.hr. to Btu. we have to multiply by the conversion factor 3412. Notice also that the right-hand side of the equation is the reciprocal of the net station heat rate. Therefore we can write the equation as follows:

$$\text{Thermal efficiency : } e = \frac{3412.}{\text{heat rate}}$$

Peak Load

Peak load is the maximum load consumed or produced by a unit or group of units in a stated period of time. It may be the maximum instantaneous load or the maximum average load over a designated interval of time. Maximum average load is ordinarily used. In commercial trans-actions involving peak load (peak power), maximum average load is taken as the average load (power) during a time interval of specified duration occurring within a given period of time say a month. The time interval is selected during the period in which the average power is greatest.

Base Load

A unit operating on base load is used to supply that part of the load of a system which stays fairly constant during the day, week, or month. A base-load unit is operating at 100% full power all the time, providing the system demands it.

Spinning Reserve

Spinning reserve is that reserve generating capacity connected to the bus and ready to supply power.

Capacity Factor

Capacity factor is the ratio of the average load on a machine or equipment, for the period of time considered, to the rating of the machine or equipment. When applied to a station, this factor is called station factor or station capacity factor. If the capacity factor is applied to a station for a period of a year, the plant is charged for energy that could have been generated during that period, including for example time taken for annual overhaul, inspection or breakdowns.

Let's say we're considering a unit rated at 200 MW for a period of one year or 8760 hrs. And let's say that the total k.w.hrs. sent out during this period is actually 1,401,600,000. Then:

$$\text{Capacity Factor} = \frac{1,401,600,000}{200,000 \times 8760} = 80\%$$

Capacity factor is sometimes also referred to as load factor, however load factor also has another meaning and therefore capacity factor is used here to avoid confusion.

Mills/k.w.hr.

For purposes of comparing the economics of one unit with another, or one station with another, the cost per kilowatt-hour sent out is generally expressed in mills. One mill is one thousandth or 0.001 of a dollar. Hydraulic power stations produce power at around 3 mills/k.w.hr. This of course varies with the type of site and capacity. Large, modern, high-pressure high-temperature coal-fired units can produce electric power at around 4 mills/k.w. hr. The present day cost estimate for 500 MW Candu type nuclear unit is around 4 mills/k.w. hr.

When evaluating the cost of a power station a number of factors have to be taken into account. These are:

(1) Capital cost-- that is the initial cost of the site, buildings and structures and all equipment associated with the station, engineering and construction costs, etc.

(2) Finance charges for the capital cost.

(3) Depreciation-- in estimating cost it is generally allowed that the life of a station is 25 to 30 years.

(4) Operating cost-- covers items such as fuel, maintenance, staff and any other expenses related to operating the station.

As was mentioned the above costs are expressed in mills/k.w. hr. sent out and the sum of the total k.w. hrs. sent out over the life of the station is based on an estimated capacity factor. The capacity factor may vary from 30% to upwards of 80% depending on the particular sitaution. The cost estimates for Ontario Hydro's nuclear power stations are based on a 30 year life and an 80% capacity factor.

Dick Dueck

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3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-1 - Definitions

A - Assignment

1. Describe what is understood by Willan's Line.
2. Define heat rate.
3. Define peak load.
4. Explain what is meant by capacity factor.
5. Explain the term mills/k.w.hr.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment and System Principles - T.T.1

4 - Turbine, Generator and Auxiliaries

-2 - Turbine Theory

0.0 INTRODUCTION

This course on "Turbine, Generator and Auxiliaries" at T.T.4, T.T.3, and T.T.2 levels has explained the closed cycle, in general terms how a turbine works, how nozzles and blades or buckets behave and something about the regenerative feed-heating system. With this background we can now proceed in this lesson to develop some theory which is the basis for turbine blade design.

This lesson will assume that the reader is familiar with the properties of steam and has studied the course on "Heat and Thermodynamics", T.T.1 level or at least is familiar with the Carnot and improved Rankine cycles, and the Mollier Chart. It will also assume familiarity with trigonometric functions. The Carnot and the Rankine cycles are briefly reviewed here as well as the following two definitions:

An adiabatic process has been defined as a process in which no heat is transferred to or from the working fluid. Because a steam turbine is well insulated there is relatively little transfer of heat from the steam to the outside of the casing and for all intents and purposes the flow of steam through a turbine can be considered an adiabatic process.

An isentropic process has been defined as a reversible adiabatic process during which entropy remains constant.

1.0 INFORMATION

In the course on "Heat and Thermodynamics" it has been stated that the most efficient of all cycles is the Carnot cycle The efficiency of the Carnot cycle has been given as:

$$\text{efficiency } e = \frac{T_1 - T_2}{T_1}$$

where: T_1 = absolute temperature at which heat is supplied

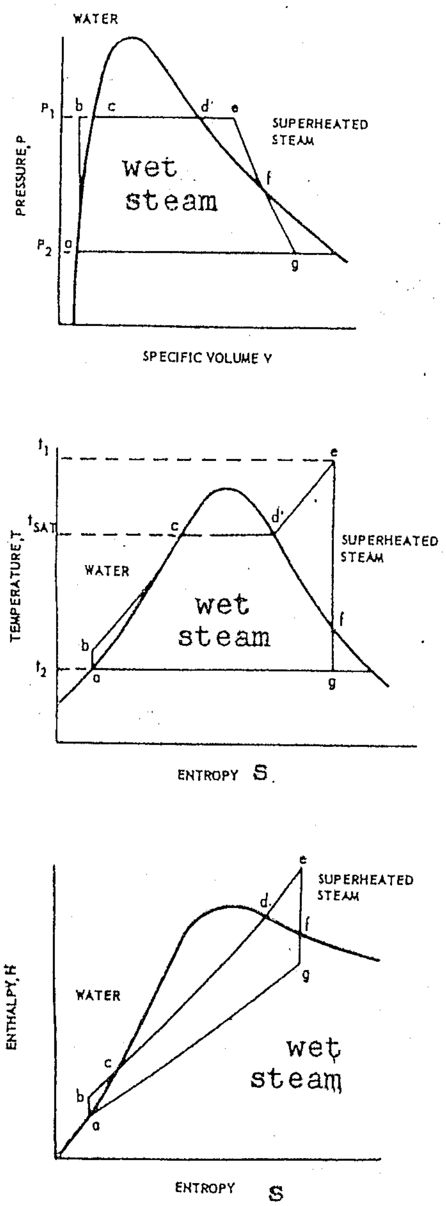
T_2 = absolute temperature at which heat is rejected.

From the equation it can be seen that for maximum efficiency, the supply temperature should be as high as possible and the rejection temperature should be as low as possible.

The steam power station uses the Rankine cycle, modified to include superheating, regenerative feedheating and possibly reheating. Figure 1 illustrates this cycle with superheating only, in terms of the pressure-volume, temperature-entropy, and enthalpy-entropy (Mollier chart) diagrams. By examining the temperature-entropy diagram, the effect of varying steam conditions on the cycle efficiency may be seen, the area enclosed representing the work done. It should be realized in practice, however, that in general the temperature-entropy diagram exaggerates the benefits of pressure and understates the benefits of temperature. For this reason the enthalpy-entropy diagram or Mollier chart is more commonly used in representing the Rankine cycle for steam turbines.

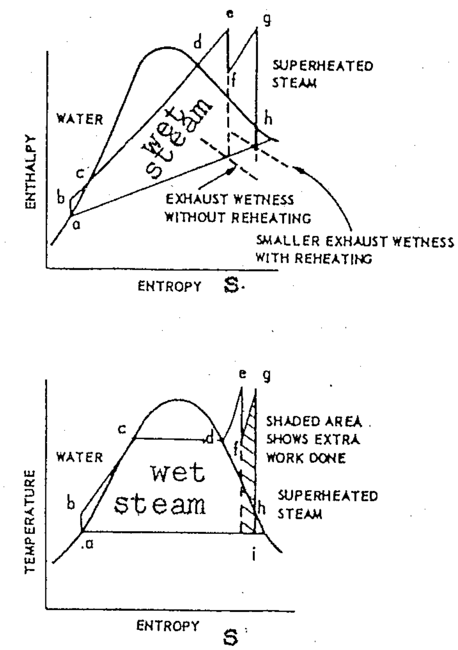
A higher pressure in the boiler raises the temperature at which latent heat is added, which in turn increases the cycle efficiency, but also increases the moisture content at the turbine exhaust, which is detrimental. The maximum exhaust wetness which can be tolerated, without excessive blading erosion, is about 11 to 13%, and this, therefore determines the maximum pressure which can be used with a given steam temperature. A higher steam temperature at the superheater outlet raises the mean temperature at which the superheat is added, thus increasing the cycle efficiency. Moreover, if the pressure of the steam remains unaltered, the effect on the turbine will cause a smaller part of the expansion to be in the wet steam region, giving an improvement in turbine efficiency. Thus superheat is doubly beneficial; the final steam temperature is, however, limited by the properties of the materials from which the superheater, piping, valves, steam chest and inlet blades are constructed. For feritic materials the present limit is about 1050°F. Using this temperature and 12% exhaust wetness, the corresponding initial steam pressure would be about 1800 psig for a non-reheating cycle.

To obtain full benefit from high steam conditions it is necessary to use large turbines. This is partly because the extra cost of high temperature components is offset by a saving in the number of components per megawatt and partly because losses become proportionately smaller. Further, unless high-density, high condition steam is used at a high rate of flow, the high-pressure blade heights become very small and inefficient. A lower exhaust pressure lowers the temperature at which heat is rejected, thus increasing the cycle efficiency. For condensing turbines the vacuum that can be obtained is determined primarily by the temperature of the cooling water at the site chosen. Any possible improvement in vacuum is very effective in increasing the work done, since a narrow



b.c.d.e. - HEATING AT CONSTANT PRESSURE
e.f.g. - IDEAL EXPANSION AT CONSTANT ENTROPY
g.a. - EXTRACTION OF LATENT HEAT IN CONDENSER
a.b. - IDEAL PRESSURE INCREASE AT CONSTANT ENTROPY IN FEED PUMP

Figure 1. Basic Rankine Cycle with superheating (omitting feedheating)



AVERAGE TEMPERATURE OF b.c.d.e. AND f.g. HAS INCREASED, DUE TO REHEATING

b.c.d.e. - HEATING AT CONSTANT PRESSURE
e.f. - IDEAL EXPANSION AT CONSTANT ENTROPY BEFORE REHEATING
f.g. - REHEATING AT CONSTANT PRESSURE
g.h.i. - IDEAL EXPANSION AT CONSTANT ENTROPY AFTER REHEATING
i.a. - EXTRACTION OF LATENT HEAT IN CONDENSER
a.b. - IDEAL PRESSURE INCREASE AT CONSTANT ENTROPY IN FEED PUMP

Figure 2. Effect of reheating (omitting feedheating)

but large addition is made to the temperature-entropy area shown in figure 1.

As has been mentioned previously, on large turbines of 100 MW and over it is economic to increase the cycle efficiency by using external reheating. This is one way of partially overcoming temperature limitations. By returning partially expanded steam to a reheater, the average temperature at which heat is added is increased. Also by expanding this reheated steam through the remaining stages of the turbine, the exhaust wetness is considerably less than it would otherwise be, as shown in figure 2. Conversely, if the maximum tolerable wetness is allowed, then the initial pressure of the steam can be appreciably increased.

Stage Efficiency

Now that we have reviewed the Rankine cycle in a general way let us look at an individual stage in a turbine. As we have said in previous lessons, when steam is allowed to expand through a nozzle, it assumes kinetic energy at the expense of its enthalpy or total heat. This expansion is an isentropic process as portrayed by line 1-2' in the enthalpy-entropy diagram, -figure 3. However, as we mentioned previously, when steam at high velocity flows through turbine nozzles and blades it experiences friction. This results in some of the kinetic energy bering converted back into heat energy and tends to reheat the steam. Hence, the expansion in a nozzle, rather than being strictly isentropic, takes place from 1 to 2, and the difference in enthalpy between h_2 and h_2l is called the frictional reheat. Although thie results in a greater enthalpy of steam at the nozzle outlet than one would theoretically expect, there is also an increase in entropy, and as we know an increase in entropy represents a loss in availability of energy.

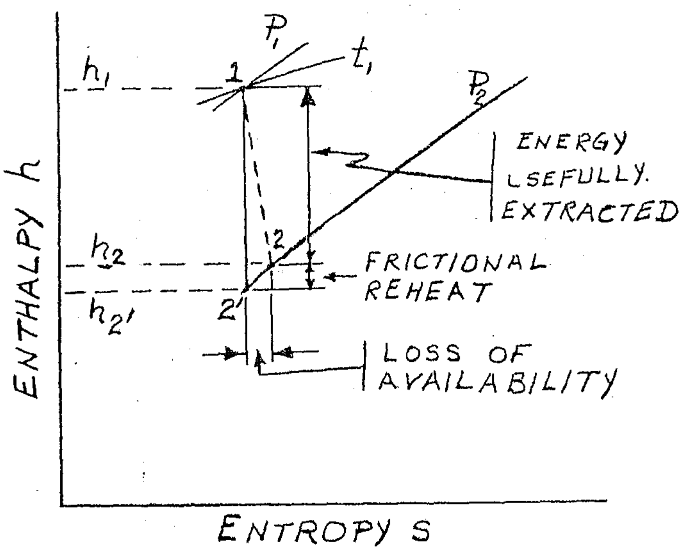


Figure 3. Useful extraction of kinetic energy.

However, friction represents only one of the losses. Below are listed other losses that can be expected in a turbine stage, which if all multiplied together results in the actual efficiency of a turbine stage. (Item no. 1 repersents the efficiency related to friction losses described above).

1) Expansion efficiency = $\frac{\text{kinetic energy produced/lb.steam}}{\text{enthalpy supplied/lb. of steam.}}$

2) Diagram efficiency = $\frac{\text{work done on rotor/lb steam}}{\text{kinetic energy produced/lb. steam}}$

3) Leakage factor for the fixed blades.

4) Leakage factor for the moving blades

5) Dryness fraction (in the wet region it is found in practice that for each additional 1% moisture, there is about 1/2% to 1 1/4% loss of efficiency. Hence, the dryness fraction is included in the product).

The efficiency of a well-designed stage in a modern turbine is between 85 and 90% and is calculated from the equation:

$$\text{Stage efficiency} = \frac{h_1 - h_3}{h_1 - h_2}$$

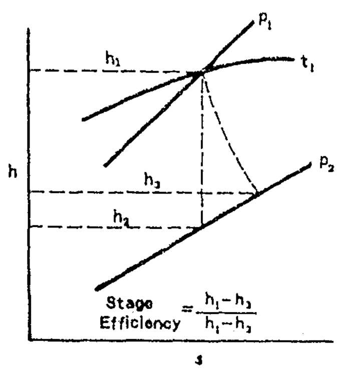


Figure 4. Stage efficiency shown on Mollier diagram.

as can be seen in figure 4. Of the 15 % of the available energy lost, some is dissipated as heat friction as already mentioned and some rejected in the form of kinetic energy. That is the steam still has some velocity as it leaves the moving blades. The latter may be partially or wholly reclaimed by the nozzles of a subsequent similar stage if carefully designed. This is known as carry-over,

The kinetic energy leaving the last stage in the turbine cannot be reclaimed and is termed the leaving loss or exhaust loss. To minimize this loss it is important that the velocity of the steam leaving the last wheel should be small and for this reason the annular area of the last row of blading is made as large as is economically practicable.

The Condition Line

Steam entering a turbine must travel through the emergency stop valve, a strained steam chest and governor valve; all of these cause a pressure drop. Friction loss through this equipment would be about 4% of the initial pressure for a multivalve governor and 10% for a single-valve turbine. This pressure drop is a throttling process represented by constant enthalpy from the entrance conditions p_1, t_1, h_1, and s_1, to point 2,

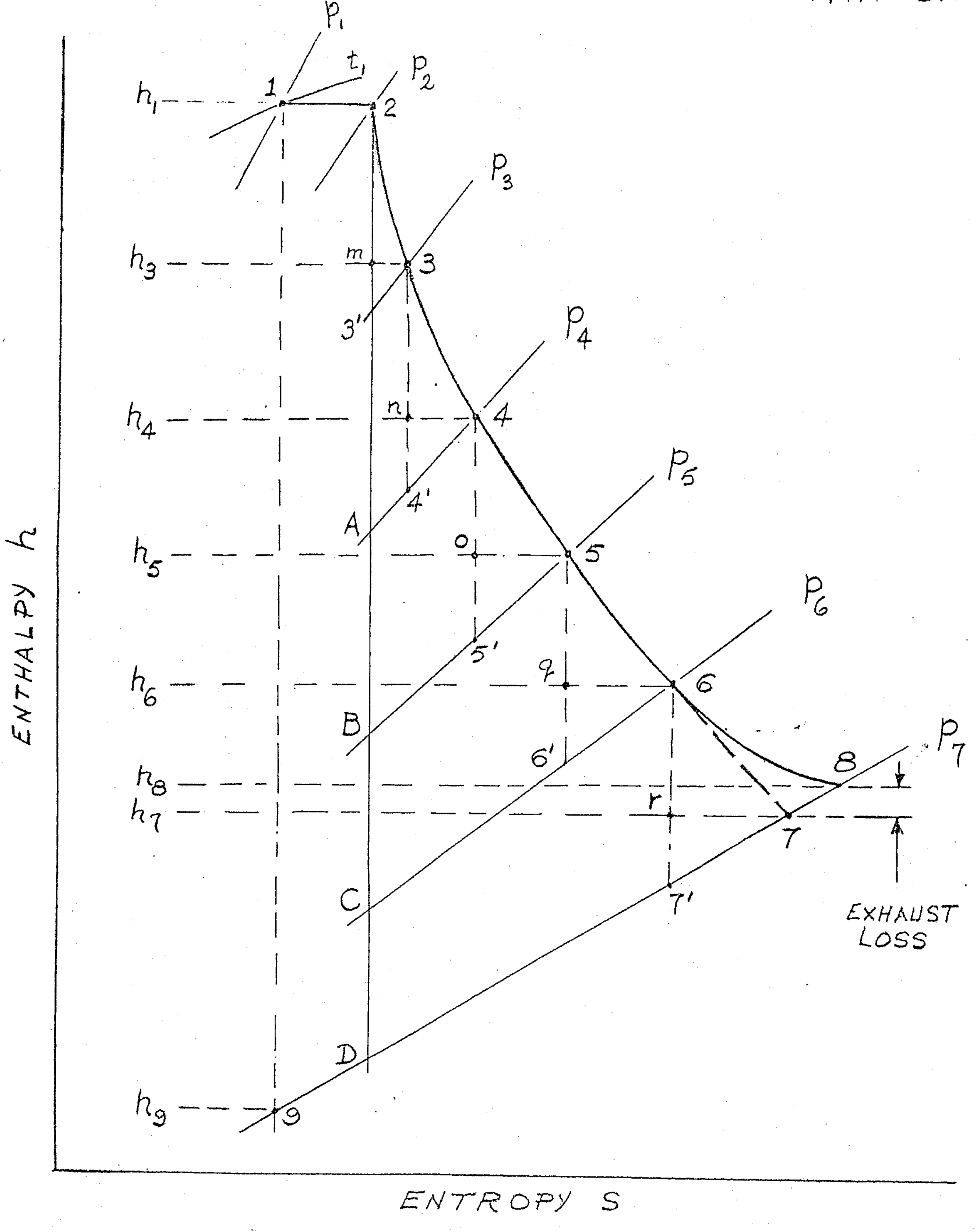


FIGURE 5. TURBINE CONDITION LINE.

figure 5. Figure 5 depicts a portion of the Mollier diagram for steam expansion in a turbine and point 2 represents the entrance to the first stage of the turbine. In figure 1 this portion is represented as the line e.f.g. on the enthalpy-entropy diagram. (In the course on "Heat and Thermodynamics" it was mentioned that the throttling process is one in which a substance drops from a high pressure to a lower pressure and that enthalpy remains constant during this process. That is why line 1 to 2 is horizontal.)

For purposes of discussion, let the isentropic expansion from p_2 to p_7, the distance 2-D, figure 5, be divided into five equal parts, such that the same work will be done in each stage of the ideal turbine. However, the steam actually leaves the first stage in some condition 3, instead of 3^1, the second stage in some condition 4 instead of 4^1 and so on. If we let e_s = stage efficiency and if the isentropic drop of enthalpy is taken as $h_2 - h_3{}^1$ for the first stage then $(1 - e_s)(h_2 - h_3{}^1)$ is the amount of reheat and is represented to some scale by the vertical distance 3^1 - m, figure 5. The point 3, representing the true condition of the steam as it leaves the first stage, is therefore found by the intersection of the horizontal line m - 3 and the pressure line p_3. Similarly, if P_4 is the exit pressure from the second stage, the actual condition of the steam at this point is determined by the amount of reheat, 4^1 - n, which determines the location of 4. For a reheat of 5^1 - 0 in the third stage, the condition of the discharge is represented by point 5. The process will continue in this manner until steam emerges from the last stage at enthalpy h_7. In as much as this is the last stage of the turbine, there is no possibility of obtaining useful work from the kinetic energy of the steam going into the condenser where the steam will come to rest. Thus, the kinetic energy discharging from the last stage, will be changed back to enthalpy again so that steam at turbine discharge will actually be at enthalpy of point 8. As mentioned previously, enthalpy difference represented by $h_8 - h_7$ is the exhaust loss. The energy transferred to the condenser circulating water is represented by the difference between h_8 and the enthalpy of saturated liquid at pressure p_7.

A curved line drawn through points 1, 2, 3, 4, 5, 6, and 8 indicating the condition of the steam throughout the turbine at points where it is possible to extract and measure the steam properties, is known as the <u>condition line</u>.

Because the constant pressure lines on a Mollier diagram diverge in the upward right-hand direction, the distance 3 - 4^1 is greater than the distance 3^1 - A and 4 - 5^1 is greater than A - B etc. In other words, the ideal work possible in the second stage is increased because of the friction of the first stage. Of course there is no net gain, since the loss in the first stage more than offsets the gain of ideal work in the second stage.

However, from the paragraph above, it follows that:

$$(h_2 - h_3{}^1)+(h_3 - h_4{}^1)+(h_4 - h_5{}^1)+(h_5 - h_6{}^1)+(h_6 - h_7{}^1)$$

is greater than $(h_2 - h_D)$. The ratio of these two quantities is known as the <u>reheat factor</u>. Thus:

$$\text{Reheat Factor} = \frac{(h_2 - h_3{}^1)+(h_3 - h_4{}^1)+(h_4 - h_5{}^1)+(h_5 - h_6{}^1)+(h_6 - h_7{}^1)}{h_2 - h_D}$$

The reheat factor is obviously always greater than one. If the constant pressure lines were parallel on the mollier diagram, the reheat factor would be one. Now if the isentropic enthalpy drop $h_2 - h_D$ is divided into equal divisions corresponding to the number of pressure stages and if these divisions are used to determine the intermediate pressures between stages, then the work will not be the same in each stage, of the real turbine. This effect too is explained by the divergence of the constant pressure lines. To get the same work in each stage of the actual turbine, which is desirable, we must use a cut-and-try approach.

Since in a Nuclear Power station the steam conditions of the stop valve may only be at saturation pressure and temperature the condition line will appear for the most part below the saturation line as illustrated in figure 6. Steam is expanded in the turbine from p_A to p_B at which point moisture content is approximately 11 to 12%. The steam is then directed to flow out of the turbine into a moisture separator where most or all of the moisture is removed and only saturated steam remains. Pressure will drop from p_B to p_C during this process due to friction. If provision is made for reheating the steam then there is a possibility for it to become superheated so that downstream of the moisture separator and reheater steam conditions will be at point C. This steam is then further expanded in an L.P. turbine to point 'D' which represents the condenser pressure.

The Principle of the Steam Turbine

We have mentioned previously that there are two fundamental transformations of energy in a turbine. First a portion of the energy of the steam is converted into kinetic energy of a moving jet in nozzles. Then a portion of this kinetic energy, through the action of the jet, imparts an impulse to a row of moving blades, which is then converted to shaft work. An impulse is equal to what is known as a change in momentum. Impulse is the product of a force times the time during which the force acts. If the force acts continuously we may use the impulse for any convenient length of time, say one second. The principle of impulse and momentum then may be expressed in the form:

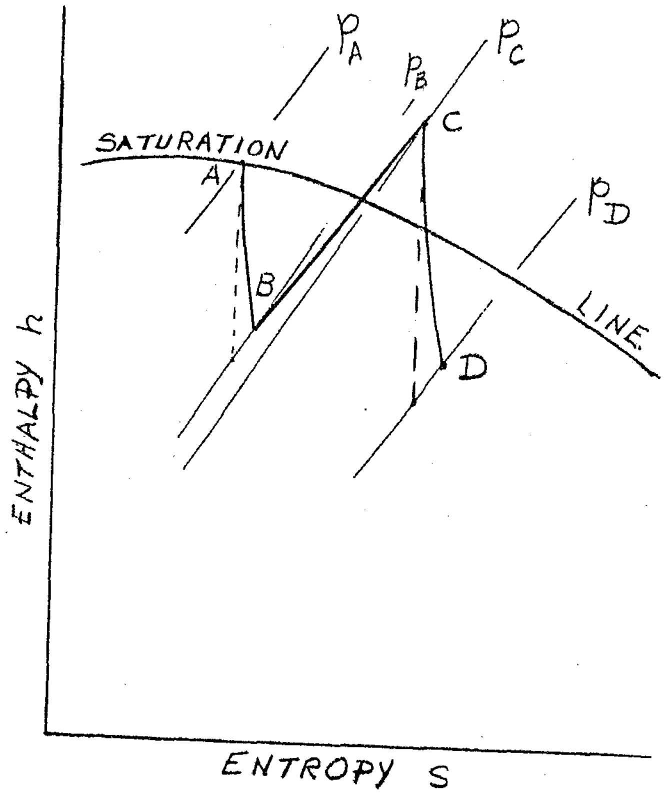


Figure 6. Condition line for steam conditions experienced in a nuclear power station.

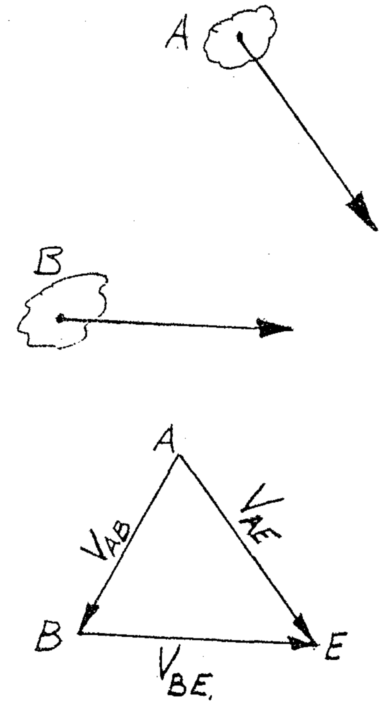


Figure 7. Diagram illustrating relative velocities vectors.

$$F_x.t = \frac{w}{g}.V_{x_2} - \frac{w}{g}.V_{x_1} = \frac{w}{g}(V_{x_2} - V_{x_1})$$

where: F_x = force acting in "x" direction

t = time in seconds

$\frac{w}{g} = \frac{\text{weight in lbs.}}{32.2 \text{ ft/sec/sec.}}$ = mass, in slugs.

V_{x_2}, V_{x_1} = velocities in "x" directions ft/sec.

As you can see from the above equation the linear momentum of any body of mass w/g is the product of the mass and the velocity of the body (all points in the body are assumed to be moving with the same velocity at any instant.) The force F_x (a vector quantity) acting in a particular direction (x-direction) during some time 't' produces the change of momentum in the x-direction in which the force is acting. This direction is not necessarily in the direction of the absolute velocity. The subscript x in the velocity symbols, indicates that the vectors for these velocities are parallel to the vector for F_x, and that, for example V_{x_1} may be the x- component of some absolute velocity V_1.

Another principle that should be stated at this time is the principle of relative velocities. The velocity of a body relative to the earth is termed its absolute velocity. But in as much as the earth is moving, it is evident that velocity is a relative matter. Thus the question arises: What is the velocity of body A relative to body B as shown in figure 7, when both A and B have some velocity relative to the earth E? The answer is: The absolute velocity of A is equal to the absolute velocity of B plus (vectorially) the relative velocity of A to B; that is:

$$V_{AE} = V_{AB} \overset{\rightarrow}{+} V_{BE} \quad ------ \quad (1)$$

By transposing the term V_{BE} to the left side of the equation, we see that the relative velocity of A with respect to B (both of which are moving) is equal to the vector difference of the absolute velocities A and B, thus:

$$V_{AB} = V_{AE} - V_{BE} \quad ------ \quad (2)$$

Velocity Diagrams for Single-Stage Impulse Turbine

A vector diagram or velocity diagram as shown in figure 8 can be used to derive an expression for shaft work output from a turbine. For simplification let's assume there is no loss due to friction. The velocity of the steam which we sill call V_1, as it leaves the nozzle is found from the expression given in the lesson on nozzles, T.T.2 level, as follows:

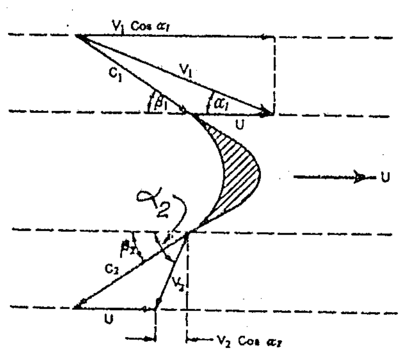


Figure 8. Theoretical velocity diagram for an impulse blade.

$$V_1 = 223.7\sqrt{h_o - h_1}$$

where h_o = enthalpy at nozzle inlet Btu./lb.
h_1 = enthalpy at nozzle outlet Btu./lb.

As seen in figure 8, the steam with velocity V_1, is directed at some angle α_1, called the nozzle angle, with the plane of the wheel. The tangential speed of the blades is "U". Choosing some convenient scale we lay out the vector V_1 at the angle α_1, with the direction of motion of the blades. Now we subtract the vector U, and get the resultant C_1, which is the velocity of the steam relative to the moving blades. The velocities of C_1 and C_2 are spoken as the relative velocities at entrance and at exit, respectively. Now if the vector C_1, represents the direction of the steam relative to the blades, then the blades should be so shaped at entrance that the sides are tangent to C_1 in order that the steam shall enter with the least shock.

Impulse blading is often symmetric or nearly so. Therefore we next draw 'C_2' at the same angle α with the horizontal as C_1, but on the opposite side as shown. The problem now is to find the absolute velocity of the steam as it leaves the blades. If there is no frictional loss through the blades the velocity of steam relative to the blades does not change; that is $C_1 = C_2$. We now have the relative velocity of steam at exit from the blade and the absolute velocity of the blades. From equation (1) we can see that the vector sum of C_2 and U will be equal to the absolute velocity of steam V_2.

A consideration of the energy quantities reveals that the steam enters and leaves the blades with the same enthalpy, since no expansion occurs in the moving impulse blade and there is no loss due to turbulence under theoretical consideration. Consequently the only change in the energy of the steam is the kinetic energy, measured by the absolute velocities V_1 and V_2. This change of energy is delivered to the shaft as work. Thus the work for these ideal blades is:

$$W = K.E._1 - K.E._2 = \frac{V_1^2 - V_2^2}{2g} \text{ ft.lb./lb.steam.}$$

Normally during turbine calculations the blade is omitted from velocity diagrams, different methods of showing these type of diagrams are shown in figure 9. Parts a) and b) are both drawn for on impulse blade with no allowance for friction. The type of diagram used in (a) indicates the velocities by the vector length, but the directions are incorrect. The methods of parts (b) and (d) have the advantage of showing both the magnitude and direction of the vectors. Figure 9 (c) is drawn the same as (a) but indicates that frictions has been taken into consideration (since C_2 is smaller than C_1).

Let us next consider a general case of steam flow past the blades with friction and with or without an expansion during this passage. If there is friction and no expansion (no pressure drop) 1/2 is less than V_1. If there is friction and an expansion too, V_2 may be smaller or larger than V_1,

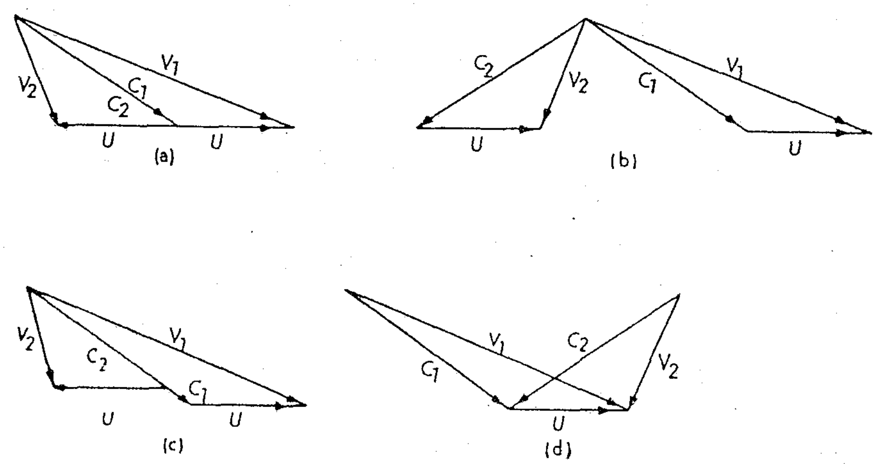


Figure 9. Combined velocity diagrams for no friction loss at (a) (b) and (d); with friction loss at (c).

depending on relative magnitude of the frictional effect and the expansive effect. In any case the friction tends to retard the steam and expansion tends to speed it up. It does not matter which occurs as far as the subsequent equation (4) is concerned. Applying the general energy equation as described in "Heat and Thermodynamics", T.T.2 level, to a row of turbine blades, we have, per pound of steam (or of any substance, for that matter):

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

$$W = \frac{V_1^2 - V_2^2}{2gJ} + (h_1 - h_2) \text{ Btu/lb.} \quad - - - - (3)$$

where: W = shaft work
h_1 = enthalpy at stage inlet, Btu/lb.
h_2 = enthalpy at stage outlet, Btu/lb.
V_1 = absolute velocity at blade inlet ft/sec.
V_2 = absolute velocity at blade outlet ft/sec.
g = acceleration due to gravity, 32 ft/sec/sec.
J = 778 ft. lbs. = 1 Btu.

In general, then, the work is equal to the decrease in kinetic energy, as measured by the absolute velocities, algebraically plus any decrease of enthalpy during passage. The change of enthalpy during passage can be measured by energy quantities relative to the blades. The enthalpies are the same whether referred to the blades or to the earth, since this energy depends only on the state of the substance itself. However, kinetic energies relative to the blades depend upon the relative velocities. Thus taking the blades as the reference body, we have:

$$h_1 + \frac{C_1^2}{2gJ} = h_2 + \frac{C_2^2}{2gJ}$$

$$\text{or} \quad h_1 - h_2 = \frac{C_2^2 - C_1^2}{2gJ}$$

Using this value of $h_1 - h_2$ in equation (3) we get:

$$W = \frac{V_1^2 - V_2^2 - (C_1^2 - C_2^2)}{2gJ} \quad \text{Btu/lb.} \text{ - - - - - - } (4)$$

The omission of J of course changes the units to ft.lb. instead of Btu. To match the units of $g = 32.2$ all velocities should be in ft/sec. We observe that the net work is equal to the loss of kinetic energy relative to the ground minus the loss (or plus the gain) of kinetic energy in the blades. There is a relationship between equation (4) and figure 10.

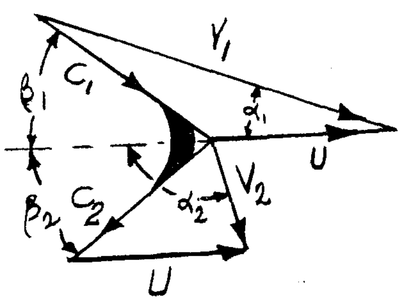


Figure 10. Velocity diagram single stage, with friction.

The energy chargeable to the blades or rotor of an impulse turbine is taken as the kinetic energy of the entering jet $V_1^2/(2g)$ ft. lbs. The blade efficiency for the impulse turbines is therefore:

$$e = \frac{V_1^2 - V_2^2 - (C_1^2 - C_2^2)}{V_1^2} \text{ - - - } (5)$$

Certain remarks of a general nature about figure 10 are now in order. The angle β_1 is not necessarily equal to angle β_2 in impulse blades. An inspection of this diagram shows that the magnitude and direction of V_2 and therefore the efficiency of the blades is affected by the value of β_2. A discussion leading to an evaluation of the optimum value of β_2 is beyond the scope of this course. In the most advantageous arrangement, if it were practically possible, angle α_1 would be zero, the blades would absorb all the energy of the jet, $V_1^2/(2g)$ and the exit velocity V_2 would be zero. This arrangement would result in a blade efficiency of 100% The nozzle angle α_1 evidently then affects the efficiency: the smaller the angle, the better in general the efficiency. If it is too small, however, the blades will interfere with the entering jet. A typical value of α_1 is 20°. Next, we observe that the relative magnitude of the jet velocity V_1 and the blade velocity U affect the magnitude and direction of each of the other vectors in the diagram, and consequently affects the blade efficiency. Specialized works on steam turbines will give proof of the proper relation between V_1, α_1, and U. Suffice it to say here that for maximum blade efficiency, the blade speed U should be roughly half the jet velocity in impulse turbines. That is the velocity ratio $U/V_1 = 1/2$. In a frictionless turbine, the blade speed should be such that $\alpha_2 = 90°$. If $\alpha_2 = 90°$, then V_2 is a minimum and the loss due to this inevitable residual velocity is a minimum. The loss of relative velocity from C_1 to C_2 is due to friction which cannot be entirely eliminated. The value of C_2 depends on the magnitude of the relative velocity, but is of the order of 10% less (90% blade efficiency) than C_1, in the impulse turbine.

Driving Force on Buckets

The driving force on buckets and the work done may be found from the principle of impulse and momentum. Referring to figure 11 which is similar to figure 10 we see that the component of the original velocity in the direction of motion of the blades is:

$$V_1 \cos\alpha_1$$

The steam leaves the blades with a component velocity in the direction of motion of the blades of:

$$V_2\ (-\cos\alpha_2)$$

The change of momentum is the (mass) x (change of velocity), where the change of velocity accounts for a possible reversal of direction as well as a change in magnitude. Thus:

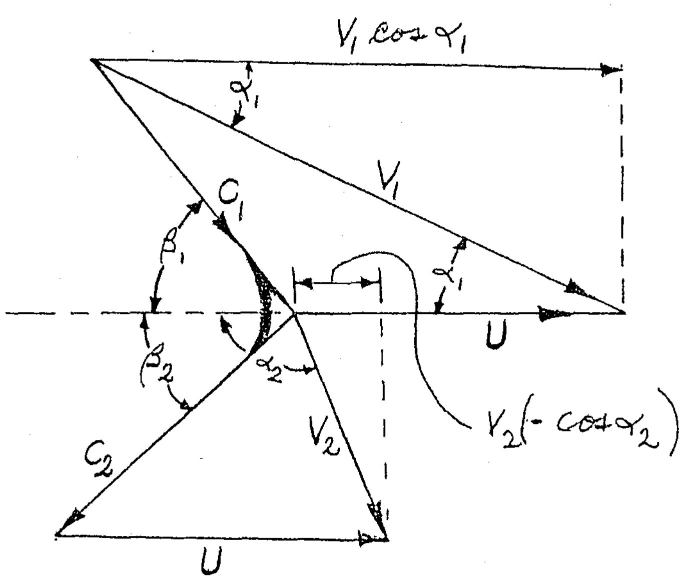


Figure 11. Components of vectors which act in the direction of rotation of the turbine blades are $V_1 \cos\alpha$ and V_2 $(-\cos\alpha_2)$ in this example.

$$Ft = \frac{w}{g}\left[V_1 \cos \alpha_1 + V_2 (-\cos \alpha_2)\right]$$

or

$$Ft = \frac{w}{g} (V_1 \cos \alpha_1 - V_2 \cos \alpha_2)$$

If the time is taken as 1 second t = 1 and w is the rate of flow in lb/sec., this equation then becomes

$$F = \frac{w}{g} (V_1 \cos\alpha_1 - V_2 \cos\alpha_2) \text{ lbs} \quad - - - \quad (6)$$

"F" is the tangential force on the blades. Observe that this force may be said to be an impulsive force, since it is derived from the rate of change of momentum of the steam; hence the name of this type of turbine. If the blades are moving 'U' ft/sec. then F.U is the rate of work Power in ft.lb/sec. Therefore

$$\text{Power} = F.U = \frac{w}{g}(V_1 \cos\alpha_1 - V_2 \cos\alpha_2).U \text{ ft.lb/sec.} \quad - - \quad (7)$$

If the result from equation (4) is multiplied by 'w' lb/sec. of steam delivered to the blades, the answer should be the same as in equation (7) except that the units would be in Btu/sec. The horsepower developed can of course be found from equation (7) by dividing by 550. (Since 1 H.P. = 550 ft.lbs/sec.)

In finding $\cos\alpha_2$, we must remember that the cosine of an angle greater than 90° is a negative number and the negative sign must be carried into equations (6) and (7).

Theory of Reaction Blades

Both the underlying principle and the construction of reaction turbines differ materially from those of the impulse unit.. One row of stationary and one row of rotating blades constitute a reaction stage. The blades are unsymmetrical, with the outlet angle much larger than the inlet angle, so that the available area for the steam jet is in the shape of a converging nozzle. There is a drop in pressure in both the stationary and rotating rows with a corresponding increase in specific volume. The absolute velocity V_1 increases in the stationary blades and the relative velocity C_1 increases in the moving blades, but the absolute velocity V_2 of the steam leaving the moving blades is less than the absolute velocity V_1 at which the steam entered the moving blades. Since the turbine runs full of steam, the volume flow at any section is:

Flow = (free area A x velocity V) cu.ft./sec.

However, as we have mentioned before, the volume increases as the steam progresses through the turbine, either the area or velocity or both, must also increase. In order to keep the blade lengths within reasonable limits, the area and the velocity are increased toward the exhaust of the turbine.

All turbine blades may be classed according to their degree of reaction, which is defined as follows:

$$\text{Degree of Reaction} = \frac{\text{enthalpy drop in moving blades}}{\text{enthalpy drop in the stage (stationary + moving blades)}}$$

The degree of reaction may vary from 0 to 100%, zero reaction being pure impulse. It is not at all uncommon for impulse blades to have a small amount of reaction (say 5%) to improve their efficiency.

Parsons, who invented the reaction turbine, referred to the blading he developed as impulse - reaction blading. Since his blades were 50% reaction, they were also 50% impulse. However, blades containing an appreciable percentage of reaction effect, such as 50% reaction, are known by the shortened term reaction blades.

Since there is an equal enthalpy drop in the stationary and moving rows of 50% reaction blades, the pressure drop across the moving and stationary rows is equal; also the blade passages have the same shape. This represents a manufacturing advantage. It also follows that the velocity diagrams for both stationary and rotating rows are identical as in figure 12.

Space does not permit a detailed discussion of all varieties of blades, and, therefore this lesson will only develop the equations for 50% reaction blades. In this case:

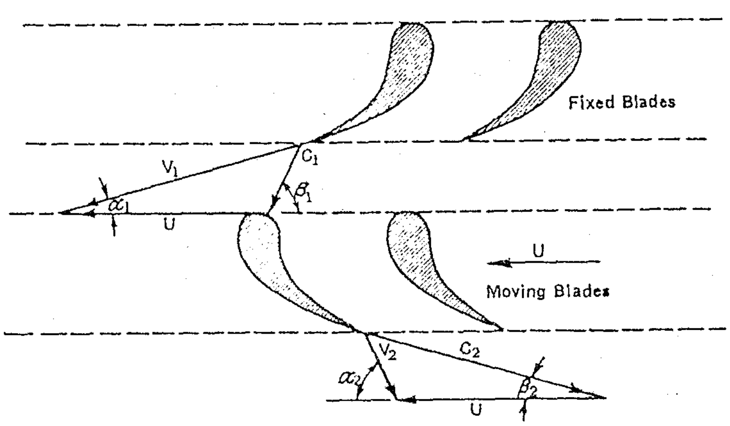


Figure 12. Velocity diagrams for reaction blades.

$$V_1 = C_2$$

$$C_1 = V_2$$

$$\alpha_1 = \beta_2$$

$$\beta_1 = \alpha_2$$

(The exit velocity V_2 both in reaction blading and impulse blading is the carry-over velocity which can contribute work in the succeeding stage.)

To develop equations for reaction blading we proceed in a manner similar to that for impulse blading. The force acting on the moving blades is (referring to figure 12)

$$F = \frac{w}{g} (V_1 \cos \alpha_1 + V_2 \cos \alpha_2) \quad - - - - - - (8)$$

Since $V_1 = C_2$, $C_1 = V_2$, $\alpha_1 = \beta_2$ and $\beta_1 = \alpha_2$ it can be shown that:

$$\frac{w}{g}(_1 \cos \alpha_1 + V_2 \cos \alpha_2) = \frac{w}{g}(C_1 \cos \beta_1 + C_2 \cos \beta_2)$$

$$\therefore F = \frac{w}{g} (C_1 \cos \beta_1 + C_2 \cos \beta_2) \quad - - - - - - (9)$$

From the velocity diagram it can be seen that the vector:

$$V_2 \cos \alpha_2 = C_2 \cos \beta_2 - U$$

but:

$$C_2 \cos \beta_2 = V_1 \cos \alpha_1$$

Therefore:

$$V_2 \cos \alpha_2 = V_1 \cos \alpha_1 - U$$

Substituting this into equation (8) we get

$$F = \frac{w}{g} \left[V_1 \cos \alpha_1 + (V_1 \cos \alpha_1 - U) \right]$$

$$= \frac{w}{g} (2V_1 \cos \alpha_1 - U)$$

Let ρ = velocity ratio $= \frac{U}{V_1}$; $U = \rho V_1$

Then: $F = \frac{w}{g} (2V_1 \cos \alpha_1 - \rho V_1)$

But rate of work (or power) = F x U, so that:

$$F = \frac{w}{g} (2UV_1 \cos \alpha_1 - U^2)$$

$$= \frac{w}{g} (2\rho V_1{}^2 \cos \alpha_1 - \rho^2 V_1{}^2)$$

$$\therefore \text{rate of work } W = \frac{w}{g} V_1{}^2 (2\rho \cos \alpha_1 - \rho^2) \text{ ft.lbs/sec} \quad - - - (10)$$

where:

F = force, lbs.
w = steam flow, rate, lbs/sec.
g = acceleration due to gravity 32.2 ft/sec/sec.
U = blade velocity, ft/sec.
V_1, V_2 = steam inlet and outlet absolute velocities, ft/sec.

C_1, C_2 = steam inlet and outlet relative velocities, ft/sec.

ρ = velocity ratio U/V_1.

D. Dueck.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment and System Principles - T.T.1

4 - Turbine, Generator and Auxiliaries

-2 - Turbine Theory

A - Assignment

1. Draw a typical Rankine cycle enthalpy-entropy diagram:
   a) without reheating
   b) with reheating.

2. What are the advantages of reheating steam in a Rankine cycle?

3. Explain what is meant by frictional reheat?

4. What is meant by carry-over in regard to steam turbine blading?

5. Briefly explain what is meant by the "turbine condition line".

6. What is the reason as to why the reheat factor is always greater than one?

7. Steam at a pressure of 250 psia, and 400°F expands isentropically in a turbine stage to a pressure of 100 psia. If the stage efficiency is 90% calculate the actual enthalpy of steam at the exit of the moving blade. Use a mollier chart to find your answer.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-3 - Improving Turbine Performance

0.0 INTRODUCTION

We have said previously that the amount of energy steam has available for conversion into shaft work depends on its initial temperature and pressure and on the way it expands to the lower pressure in the turbine. A Mollier chart tells how much energy steam has available at each state in terms of enthalpy and how much energy is available for a given expansion.

This information is used in this lesson to describe varying turbine performance for different steam conditions.

1.0 INFORMATION

In the course on "Heat and Thermodynamics" T.T.1 level an equation was given for thermal efficiency of the complete steam cycle. However, if we're considering the turbine only the equation for efficiency is a little different, although efficiency is still output divided by input.

The efficiency of one stage in a turbine has been expressed as:

$$\text{Stage efficiency} = \frac{\text{actual enthalpy drop}}{\text{isentropic enthalpy drop}}$$

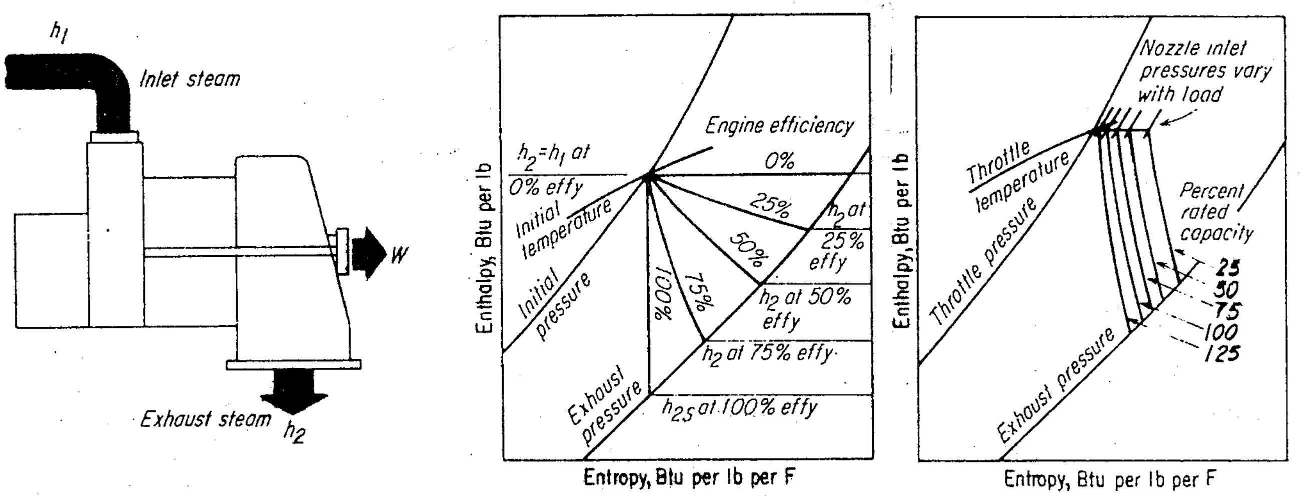
When this ratio is applied to a group of blades or to an entire turbine, it is referred to as the Rankine Cycle Ratio (R.C.R.) or the internal efficiency of the turbine. This expression does not include such mechanical losses as bearing loss, oil pump power, etc.

Referring to figure 1, we can develop the relationship for efficiency as follows:

$$\text{Available energy} = h_1 - h_{2S}$$

$$\text{Energy doing useful work} = h_1 - h_2$$

$$\text{and Rankine Cycle Ratio} = \frac{h_1 - h_2}{h_1 - h_{2S}}$$



a) Turbine converts internal energy of steam into useful shaft-work output

b) Engine efficiency measures available energy changed to shaft-work output

c) Turbine condition line shifts as the governing valve throttles steam flow

Figure 1

When the mechanical and generator losses are taken into account in this equation, it is referred to as engine efficiency which can be expressed as follows:

$$\text{Engine Efficiency} = \text{R.C.R.} \times e_{mg}$$

$$= \left(\frac{h_1 - h_2}{h_1 - h_{2S}}\right) e_{mg}$$

where: R.C.R. = rankine cycle ratio.
h_1 = enthalpy of steam at inlet, Btu/lb.
h_2 = actual enthalpy of steam at exhaust Btu/lb.
h_{2S} = ideal enthalpy of steam at exhaust, Btu/lb.
e_{mg} = mechanical and generator efficiency.

An ideal turbine would convert all the available energy to work and therefore h_2 would be equal to h_{2S}, making the engine efficiency equal to 100%. The 100% engine efficiency curve is shown as a straight line in figure 1 (b). Turbines of course never operate at 100% efficiency and figure 1 (b) also depicts the condition for 75, 50, 25 and 0% efficiency. In the case of 0% the steam is only throttled but does not give up any energy.

Turbine design generally aims for highest engine efficiency compatible with the cost of energy. Engine efficiency also varies with the load it's carrying. Figure 1 (c) shows the condition line for a single-valve turbine at different loads. As the governor valve throttles steam flow with dropping load, steam pressure at the inlet nozzles drops. This reduces the available energy because of the fact that the enthalpy rises at the exhaust with dropping load as can be seen by comparing exhaust enthalpy for 125% capacicity with 25% capacity.

Noncondensing Turbines

Figure 2 (a) shows the variation in available energy in steam used by noncondensing turbines. To find the actual work developed, a value taken from the graph would have to be multiplied by the engine efficiency. Actual exhaust moisture will be less than the ideal moisture shown in figure 2 (a).

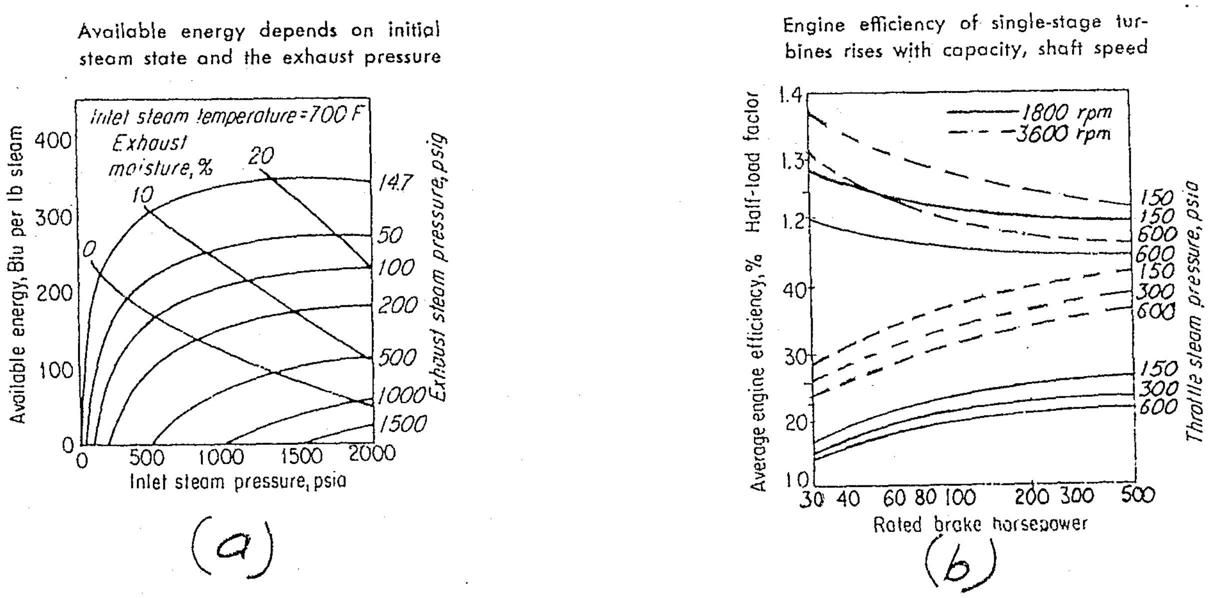


Figure 2. Inlet and exhaust steam conditions fix shaft power generated by a turbine.

Figure 2 (b) shows typical engine efficiencies at rated capacity for single-stage turbines ranging from 30 to 500 shaft H.P. The solid curves show efficiencies at 1800 rpm and the broken curves show efficiencies at 3600 rpm. From the graph it is apparent that efficiencies rise with an increase in capacity and with an increase in shaft speed. However, since this is a single-stage turbine it cannot handle high pressures efficiently. Therefore this turbine is less efficient at 600 psia. throttle pressure, than at 150 psia. throttle pressure. To find engine efficiencies at half capacity, multiply by the half-load factors given at the top of figure 2 (b).

Condensing Turbines

Figure 3 shows the available energy in steam used by condensing turbines with 1 in. Hg backpressure (solid lines) and 3 in. Hg backpressure (broken lines). Notice that the energy rises with initial steam temperature and pressure on the left-hand side of the diagram. However, up around the critical point at 3206 psia. and 705°F, available energy tends to decrease for a rise in pressure. Also beyond 5000 psia. available energy decreases for a rise in initial steam pressure. This can also be observed

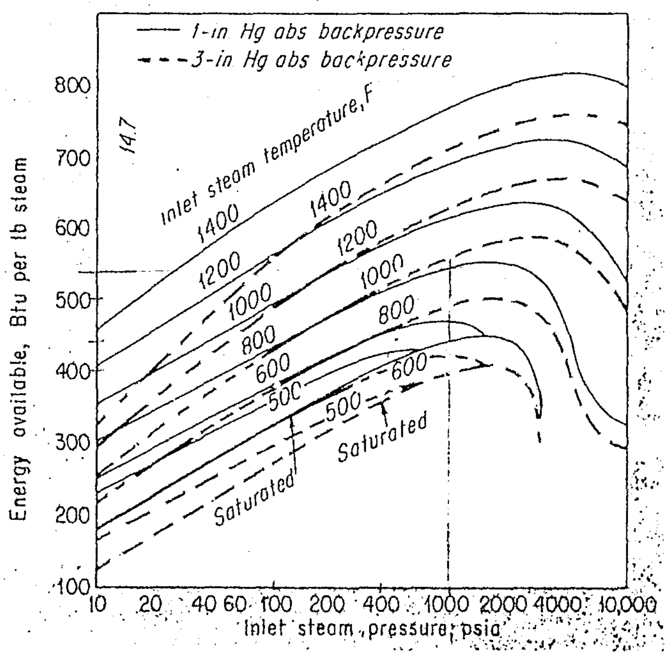


Figure 3

Figure 3. Available energy for condensing turbines depends on inlet steam pressure and temperature as well as backpressure.

by looking in steam tables such as "Thermodynamic Properties of Steam" by Keenan and Keys or a Mollier Chart which show that along a constant temperature column or line, the enthalpy of initial steam conditions decreases as pressure increases for superheated steam, even below the critical point.

The dotted lines indicate that higher backpressure reduces available energy at all inlet-steam levels. In figure 3 we also see the reason for intensive development of high-pressure high-temperature turbines in the past decade: higher available energy makes better use of expensive materials.

In figure 4 is plotted the theoretical heat rate in Btu/kw hr. against inlet steam temperature. These curves correspond to the ones given in figure 3. You will notice that theoretical heat rate drops with increasing inlet-steam temperature and pressure and as we mentioned before, a drop in heat rate is desirable in a steam turbine cycle. Again, exceptions to this general trend occur around the critical-point area. The broken lines again indicate that a higher backpressure will result in a higher heat rate.

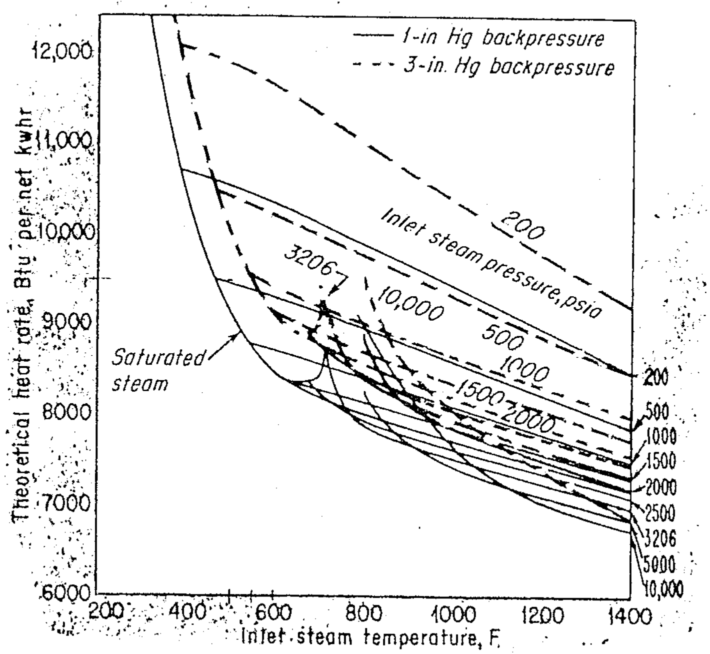


Figure 4. Theoretical heat rate of condensing turbines generally improves with rising throttle-steam pressure and temperature.

Estimating Turbine Performance

Given: 5000 kw turbine, inlet steam at 800°F, 1000 psia.; exhaust at 1 in. Hg abs.; engine efficiency of 74%.

Find: Actual heat rate.
Thermal efficiency.
Full-load steam rate of turbine.

Solution: From figure 4:

Theoretical heat rate = 8400 Btu/kw hr.

Actual heat rate = 8400/0.74 = 11,350 Btu/kw hr.

$$\text{Thermal efficiency} = \frac{3412}{11,350} = 0.301 \text{ or } 30.1\%.$$

From figure 3:

Available energy = 545 Btu/lb.

Energy released = 545 x 0.74 = 403 Btu/lb.

Full-load output = 5000 x 3413 = 17,060,000 Btu/hr.

$$\text{Steam flow} = \frac{17,060,000}{403} = 42,300 \text{ lb./hr.}$$

$$\text{Full-load steam rate of turbine} = \frac{42,300}{5,000} = 8.46 \text{ lb./kw hr.}$$

Reheat Turbines

We've mentioned previously that in the reheat cycle steam is expanded partially in the H.P. cylinder and then all of the steam is passed through a reheater and from there it passes through the remainder of the turbine and into the condenser. One of the things the designer has to determine, is to what extent expansion should take place in the H.P. cylinder before reheating. Figure 5 shows the reheat cycle thermal efficiency plotted against the ratio of reheat pressure to throttle pressure. The curves plotted are for several different turbine throttle pressures. Initial temperature is 1000°F and final reheat temperature is 1000°F. For a straight-condensing turbine, the theoretical efficiency is given at the 1.0 ratio of reheat to throttle pressure, which would result in the same efficiency as for the straight Rankine cycle. This is labelled on the right-hand side of the diagram.

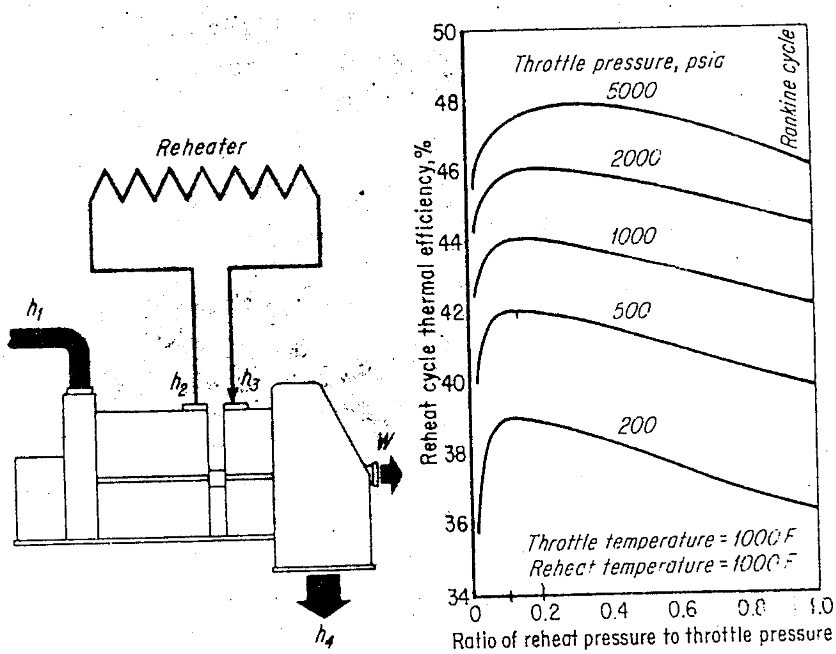


Figure 5. Diagram illustrating that maximum gain in thermal efficiency is made with reheats at low pressures.

From figure 5 it can be seen that for a diminishing steam pressure at the reheater that is as the ratio of reheat pressure to throttle pressure decreases, the thermal efficiency rises. This holds true until the pressure ratio is about 0.3 at 5000 psia., or 0.1 at 200 psia., etc. Any further reheat-pressure drop rapidly depresses thermal efficiency. In general, reheating gains about 2% in thermal efficiency. When actual engine efficiencies are considered, relative gain can be larger, about 4 to 5%. This may not sound like very much but for a 200,000 K.W. unit, a gain of 4% can mean a gain of 1,680 x 10^6 kw hrs. based on 80% capacity factor over a plant life of 30 years.

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NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-3 - Improving Turbine Performance

A - Assignment

1. A condensing steam turbine has steam entering at 550 psia., 500°F and its exhaust pressure is 1" Hg absolute. The actual enthalpy of exhaust steam is found to be 900 Btu/lb. Calculate the Rankine Cycle Ratio and engine efficiency for the turbine. Assume a mechanical efficiency of 95% and a generator efficiency of 98%. Use the Mollier Chart given in the lessons on "Heat & Thermodynamics" to obtain your solution.

2. A 200 rated brake horsepower, single-stage turbine receives steam at 150 psia. Use figure 2 in this lesson to determine the efficiency of this turbine at half load when: (a) operating at 1800 rpm. (b) at 3600 rpm.

3. Under the heading for "condensing turbines" we have said that enthalpy of initial steam conditions decreases as pressure increases and yet figure 3 shows that available energy rises up to the critical pressure point. Explain.

4. The inlet conditions of a 20,000 kw turbine are 550 psia., and 500°F; exhaust pressure is 1" Hg abs.; engine efficiency is 75%. From curves given in this lesson find the (1) actual heat rate, (2) thermal efficiency and (3) full-load steam rate of the turbine.

5. If we consider a reheat turbine with throttle steam at 500 psia., 1000°F, to what pressure should the steam be expanded in the H.P. cylinder before it is reheated to 1000°F, in order to obtain the best thermal efficiency. Use the information available in this lesson to obtain your solution.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & Systems Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-4 - Turbine Operational Problems

0.0 INTRODUCTION

This lesson will describe some steam turbine operational considerations and problems in a general way.

1.0 INFORMATION

1.1 Pre-Commissioning Period

This period includes the final stages of erection. Operationally the most important aspect is ensuring adequate internal cleanliness of the plant. Considerable operational troubles of steam turbines arise from the effects of dirt, debris, scale and silica being carried into blading, glands and bearings from the boiler and pipe systems.

Internal cleaning of plant is usually carried out in two stages.

(A) Cleaning in the Final Erection Stage

All accessible parts should be wire brushed or other mechanical methods should be used and all extraneous loose material should be removed.

Subsequent precautions to be taken are as follows:

(1) The lubricating and control oil systems should be thoroughly cleaned and then closed.

(2) The feed heating and condensing system should be thoroughly hosed and flushed to waste and the system should then be closed.

(3) The boiler should be thoroughly hosed down, starting at the top.

(B) Pre-service Cleaning After Erection

With modern plant a higher standard than that produced by the methods described in (A) above is essential. Some dirt will escape the most careful manual cleaning and in particular mill scale and foundry sand are difficult to remove. These materials are prolific sources of silica, the effects of which are dealt with later.

It has been found that only chemical cleaning will produce the required standard of cleanliness. Because of differences in types of plant, chemical cleaning cannot be dealt with in detail and general principles only are given. It is convenient to treat separately the boiler and the feed-heating condensing system.

(1) Boiler. The boiler is normally 'boiled-out' with a solution of trisodium phosphate (Na_3PO_4) and caustic soda ($NaOH$). This is done at the requisite pressure so that circulation is established. The unit is then emptied. This action is known as "being blown down-under pressure from all points".

This is followed by acid cleaning, a weak solution of either citric acid or a mixture of hydrochloric and hydroflouric acids being used. In both cases suitable inhibitors and wetting agents must be added. Hydrochloric acid must not be used if austenitic steel is used in the plant because of the risk of surface etching which causes local increases in stress.

One example of accomplishing pre-service cleaning is described below:

A circulation circuit including drum, water walls, downcomers, headers and risers is established. The acid solution is circulated for at least six hours at a rate of at least 10 percent of the boiler maximum continuous rating. The solution should be at approximately 200°F.

When there is no longer any appreciable increase in iron content in the circulating liquid, the acid should be drained away as rapidly as possible and the plant rinsed with softened water. The unit is then filled with acidulated softened water which is neutralised with ammonia or hydrazine. This solution should be circulated for not less than six hours at a temperature of 120°F. After this stage, the unit is drained, opened up, hosed down and examined.

The unit should then be filled with demineralised water containing 1,000 parts per million of hydrazine and allowed to soak for six to eight hours. The water is then drained so as to reach working level before pressure is raised and a thorough circulation is established for a period of 72 hours.

Finally the unit is under about 100 lb./sq. in. pressure from all available points, cooled and inspected. If any loose material is discovered during the inspections it is removed.

(2) Feed-heating and Condensing System. The first operation is an alkaline cleaning and degreasing of the steam side of condensers and feed-heaters. Each item may have to be treated separately but circulation circuits are preferred. The cleaning fluid would be a solution of about 200 parts per million of caustic soda and 100 parts per million of trisodium phosphate in deionised water initially at 210°F. Circulation or static soak would be for approximately four hours, followed by a hot deionised water flush.

The remainder of the system on the inlet to the boiler including all feed pipework from condenser extraction pump discharge to boiler inlet, the water side of feed heaters, the deaerator and storage vessel, but excluding pumps, is cleaned with acid. The acid used is a 3 percent solution of citric acid in deionised water which contains inhibitors and wetting agents.

A complete circulation should be maintained for a minimum of four hours. The deaerator storage vessel heaters are used to raise the temperature to approximately 200°F. Circulation and cleaning continue until samples of the acid show no appreciable increase in iron content.

When cleaning is complete the system is completely drained of acid as rapidly as possible. The system is then completely refilled with water, circulation started and ammonia added to give a hydrogen ion concentration of 7.5 to 8.0, the temperature being raised as before. The process is continued for eight hours.

During the circulation, hydrazine is added to the extent of 1,000 parts per million to condition the metal surfaces. When the circulation is stopped, the system is again emptied as rapidly as possible and is then inspected at selected points.

1.2 Initial Commissioning

The operation of steam turbines requires a different approach, in the period of initial commissioning, from that required during normal service operation. The turbine will probably be incomplete to facilitate the detection of leaks; joints may be unlagged. Also, certain instruments may not be provided.

A new machine requires careful supervision and high standards of instrumentation, protection and controls are required before any preliminary runs are attempted. Also, safe access to all parts of the machine is essential. As a further safeguard against dirt and debris, fine strainers should be fitted in the steam chest and in oil supply lines.

All auxilliaries should be tested as soon as electrical and steam supplies become available. A trial of vacuum raising, with glands sealed, may be made before the turbine is first run, during which the system may be checked for leaks and the operation of ejectors proved. The turbine must never be started up without adequate lagging on high pressure steam pipes and on high pressure cylinders, otherwise distortion may occur.

1.3 Loading

The optimum loading of several turbines is discussed later on but the loading of a single machine is considered below.

(A) Initial Loading after Start-Up

Loading may be of a routine nature, at an arbitrary rate ordered by the System Control Centre, or it may be standard practice with high merit plant to increase to full load as quickly as possible.

In the latter case, loading after start up is linked with quick starting and this is dealt with in detail in the section by that name. Particular attention must be given to the rate of loading and its effect on the rate of increase of temperature, the expansion of the rotor and casing and the extent of clearances as indicated by the supervisory equipment. Loading rates for cold turbines are difficult to give as so much depends upon the machine and conditions. With hot machines, loading rates of one MW per minute are common. Rates of two MW per minute have been achieved but have not been generally adopted.

During a loading operation it may be found that the clearances, as indicated by the differential expansion, are approaching the limits. An assessment of the rate of increase of differential expansion should be made and the load reduced to a point where the rotor expansion is stopped. The machine should be held at this load until there is an increase in the clearances and then the load may be increased cautiously until the movement of the rotor is stabilised.

(B) Routine Load Changes

With routine load changes, in steps, there is time for warming up and thermal stability to be reached. During any loading change there are several points to which attention must be given.

(1) Changes in the thrust position with load will be known but a careful watch must be kept, particularly on large machines with flexible couplings. Thrust difficulties have arisen owing to locked couplings. This cannot be tolerated indefinitely and a maker's modification is essential. A sudden load change may free the lock and thereby provide temporary relief. The modern tendency towards solid couplings should remove this difficulty.

(2) Glands and gland sealing must be carefully supervised both under manual control and when the operation is taken over by automatic regulators. Both oversealing and undersealing are detrimental, the former by wasting heat and the latter by allowing air ingress via the glands with subsequent mal-effects in the condenser and also because it may cause local cooling effects at glands and possible gland damage or shaft vibration.

1.4 Unloading

If normal load reduction is part of the programme then little is required beyond the precautions given above. Unloading prior to shut-down may be considered as:

(1) Normal. Preference is sometimes given to rapid unloading prior to shut-down, in order to conserve heat in the turbine for readiness for subsequent startup. The merit of this has not been established; tests have shown that there is little importance in the rate of unloading in respect of temperature after a shut-down period.

However, on some machines the rate of unloading affects the machine clearances. The rotor is particularly sensitive to the reduced flow of steam and contracts so rapidly that there may be a danger of blade contact on the back edge of the moving blades. Such machines are exceptional and require special tests to establish a safe procedure.

(2) Emergency. Emergency unloading may be manual, using governor control or by the emergency trip button, or automatic by the use of the protective devices. In such cases, normal control of the turbine is subordinated to the necessity for rapid shut-down. The most severe condition occurs when the main switches are open on full load. It is then that the governor and overspeed devices must be completely effective to avoid a wrecked machine.

1.5 Supervision On Load

If a turbine, after commissioning and after experience has been gained, is running satisfactorily, on-load operation becomes largely routine. A high standard of supervision is required because changes usually takes place slowly and are not easily detected. If a sudden change occurs, it is likely to be of an emergency nature as discussed later on.

Normal supervision will be concerned with:

(1) Mechanical conditions:- principally the lubrication of bearings but also checking on vibrations and unusual noises.

(2) Thermal conditions:- which means the establishment of optimum cycle conditions. Once established there is little required of operators provided initial steam conditions are correct, with the one important exception of the back pressure at the turbine exhaust. This is discussed in detail in the next section.

1.6 The Effect of Back Pressure on the Economy of a Turbine

The back pressure is affected by variables such as load, and the temperature and the quantity of the condenser cooling water. For a given load, control is exercised by the temperature and the quantity.

For the condenser, only a certain quantity of circulating water at a given temperature, sufficient to remove the latent heat, is required. If more is used the condensate is overcooled and this, combined with the excess pump power, results in a loss of thermal efficiency. Adjustment to the correct quantity of circulating water is essential but it does not follow that with the condenser operating at its most efficient level the same conditions apply at the turbine.

In any turbine, steam leaves the final wheel with a residual velocity and the kinetic energy corresponding to this velocity,

given by $\frac{V^2}{2g \quad J}$ Btu/lb., is almost wholly lost.

Where V = absolute velocity of steam from last stage wheel in ft./second.

Where J = Joules equivalent =
= 778 ft. lb./Btu

This loss is called the leaving or exhaust loss and may account for between 1 to 3 percent of the heat available.

For a given turbine, with constant steam flow, the leaving loss will be governed by the back pressure since this affects the specific volume of the steam in cu. ft./lb. because, if

w = steam flow in lb./second
A = annular exhaust area in sq. ft.
v = specific volume of steam in cu. ft./lb.
V = absolute velocity of steam from last stage wheel in ft./second

then $$w = \frac{AV}{v}$$

There are then two effects. Consider, for example, the lowering of exhaust pressure towards optimum with constant steam flow.

(1) The available energy per pound of steam increases, as also does the leaving loss. There will be an increase in output.

(2) The lower condensate temperature requires additional bled steam for the low pressure feed heater causing a reduction in steam flow through later stages of the turbine with a resultant loss in generation.

The optimum exhaust pressure occurs when the above two effects are equal. If exhaust pressure is below optimum then increase in output owing to (1) above, is less than the decrease owing to (2). There will be a net reduction in output and an increase in heat rate as the exhaust pressure is lowered further. At exhaust pressures above optimum, a lowering of exhaust pressure will cause an increase from (1) greater than the decrease caused by (2) and will result in a net increase in output and a reduction in heat rate.

It must be appreciated that this applies only to optimum conditions for the turbine. For an overall optimum economy, the net reduction in heat rate caused by lowering the back pressure must be weighed against the cost of increase in circulating water pumping power.

1.7 Availability

For the greatest economic benefit to the operation of the Commission's supply system, the availability of the most efficient and modern plant must be high.

Once a breakdown has occurred then it is largely the responsibility of maintenance staffs and possibly the manufacturers, to return the plant to service as soon as possible. Turbines, however, are complex machines and there may be many reasons for delay. Apart from the severity of damage there may be other reasons. For example, spares may not be stocked, shop facilities or skilled men may be committed elsewhere.

This lesson does not refer to failures resulting from design defects, or faulty erection or materials, but to failures resulting from mal-operation. Apart from such faults as inadequate drainage and insufficient attention to lubrication, the main operational causes of outages are:

(1) Attempting to carry out operations such as starting up, loading and changing load too quickly. Excessive temperature gradients are produced and result in high stresses across metal walls and flanged joints. This creates the possibility of metal failure or at least joint failure, if not immediately, then in the future. A transient disturbance such as a sudden increase or decrease in steam flow or steam temperature may cause a rotor temporarily to alter its position in relation to the casing and result in a high speed blade rub or gland or thrust damage.

To overload a turbine means that initially there must be a sufficient opening of the control valve or throttle valve available to enable the additional quantity of steam to pass. This means that normally full load can be obtained with a first stage nozzle box or steam chest pressure below full stop valve pressure.

This additional steam flow may produce, with persistent overloading:

(a) Excessive expansion, and the upsetting of clearances, particularly at the high pressure inlet.

(b) Excessive stresses at blade roots and incipient failure.

(c) Excessive thrust forces and possible damage.

(2) Overloading. Under certain conditions there is a possible economic case for overloading turbines where boiler plant is available. This idea is only tentative and, in general, overload running of turbines should not be permitted without very careful investigation.

1.8 Emergency Action

Under this heading it is difficult to be precise, but it is desirable that any engineer concerned with turbine operation should be completely sure of the action required under emergency conditions.

Emergency action may be subdivided into two categories:

(1) That taken by automatic devices independent of manual controls.

(2) That taken manually before any automatic device operates.

Under the first category the action envisaged covers certain predictable occurrences and the devices installed to deal with these have been dealt with previously, but a selection is given below:

(a) Overspeed trip--for sudden loss of load.

(b) Emergency stop valve--for loss of load and speed rise.

(c) Governor and control valve gear--for excessive speed rise.

(d) Vacuum unloader--for loss of vacuum.

(e) Alarms and auto-regulator for low oil pressure.

(f) Alarms and possibly trip for excessive shaft eccentricity.

It is essential that all these devices are kept completely serviceable and are tested, preferably under full working conditions, at frequent intervals.

Emergencies in the second category requiring manual emergency action may be of the type referred to in (1) but for which no automatic device is fitted. Other occurences could be bearing overheating, serious joint failure, fire and unusual noises or behaviour.

A considerable responsibility devolves upon the operating engineer when there is clearly some abnormality and the cause is not known. It may be necessary to shut the turbine down at once to avoid serious damage. In other circumstances, however, an immediate shut-down will give no opportunity to locate the trouble and it may be necessary to start up again for this purpose. A considerable degree of judgement is called for.

1.9 Quick Starting Technique

A steam turbine is primarily designed to carry its load at a rated speed and under specified pressure and temperature conditions. A machine will normally do this satisfactorily once stable conditions have been established. The stable conditions required are: correct thermal gradients from inlet to exhaust, all clearances must be normal, expansions must be freely completed, and alignment of the shafts must be within permissible limits.

Quick starting technique aims at running a turbine up to speed and loading as rapidly as possible and at the same time controlling the rate at which heating to working temperature takes place. The rate of heating depends upon the difference between the temperature of the steam supply, the temperature within the turbine, and upon the quantity of steam flowing. For proper control of the heating rates, it is therefore necessary to have estimates of the average high pressure and intermediate pressure cylinder casing temperatures. The estimates are required for various time intervals after shut-down from, say, a period of full load operation. In addition, the corresponding suitable steam temperatures for starting are required. Figure 1 shows cooling curves for a 60 MW turbine.

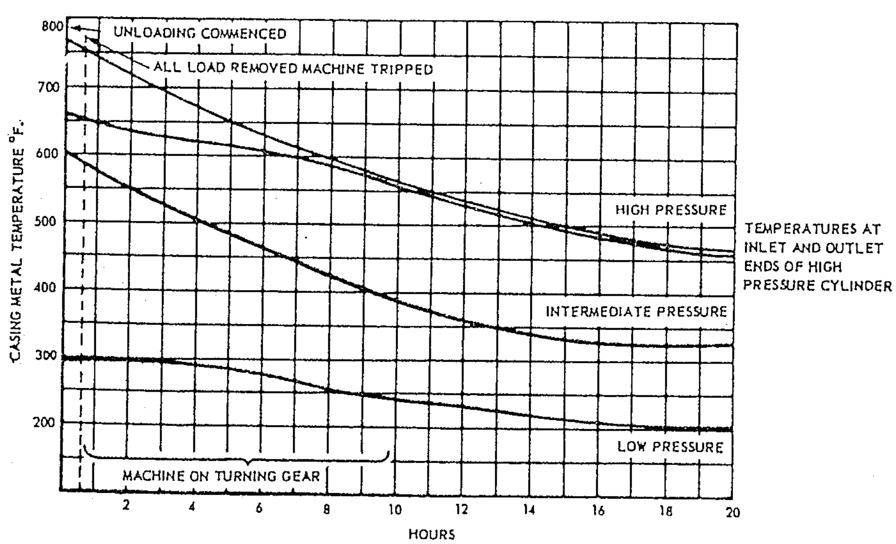


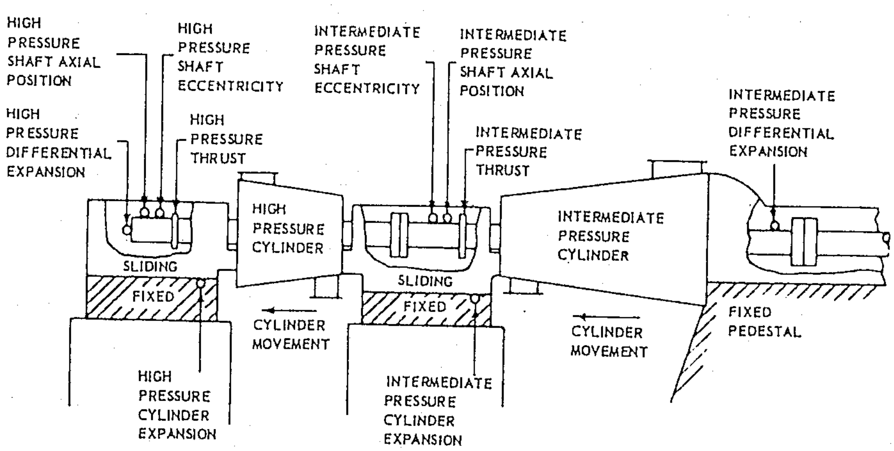
Figure 1. Turbine casing cooling curves (600 psi., 800°F, 60 MW turbine)

Starts from cold require different treatment from the starts from hot condition. The operator should, therefore, know the expected turbine stop valve and intermediate pressure inlet steam temperatures for various steam flows obtained as the running up proceeds. After the start the steam temperature will rise. The rate of rise must be limited to an appropriate value so that the required casing temperatures are not exceeded. This point is discussed further under thermal considerations.

Assuming that required steam temperatures are available for starting and subsequent loading, it should be possible, after experience, to predict times for these operations. The first application of load after a start from cold is usually critical and machines often have to be run at low loads to advance the heating process. Excessive heating of the exhaust end may limit the period during which this is permitted.

Achievement of the desired control of steam temperature, particularly at small steam flows, is a problem associated with the boiler plant. It is often necessary to take more steam from the boiler than is actually required by the turbine. This point is referred to later when discussing by-passing at start-up.

Precise knowledge of the mechanical condition of the turbine is called for during any quick starting operation. Figure 2 shows the position on the turbine of instruments which not only measures the factors involved but also assist the operator.



<u>Figure 2.</u> Arrangement of supervisory instruments for quick starting.

1.10 Factors to be Considered in Quick Starting

The various factors to be considered in any quick starting technique and their significance to the operator are discussed below. Instrumentation may be provided on high pressure turbines only, or on both high pressure and intermediate pressure as required.

(1) Cylinder Expansion. Cylinders are rigidly anchored to the bed plate at one end and expansion is permitted axially towards the steam inlet end. The cylinder expansion is measured relative to the fixed bed plate and the expansion is thus a total linear expansion and may be up to 1.0 in.

(2) Shaft Axial Position. The shaft or rotor is located axially by a thrust bearing and the shaft axial position indicator will show the position of the shaft relative to the thrust bearing. Under normal conditions the indication is a function of the load.

(3) Differential (Shaft/Cylinder) Expansion. This is the most significant of all the measurements made. The differential expansion indicator gives the expansion of the shaft relative to the cylinder. Limits are given within which safety is assured. If the differential expansion is allowed to increase or decrease outside the safety limits then there is risk of a rub occurring between the fixed and moving blades or glands.

(4) Shaft Eccentricity. There are several reasons why a shaft may become bent, for example, unbalanced heating or cooling. The shaft is inaccessible except at the bearings and it is here that indicators are fitted to show eccentricity resulting from transient shaft bending.

Eccentricity caused in these ways may be removed by turning the rotor at low speed under steam or by barring. However, eccentricity caused by more permanent faults, such as mechanical out-of-balance or bearing instability, may persist and may increase with speed.

(5) Bearing Pedestal Vibration. Instruments to measure vibration are not always installed as a permanent feature. A portable type of vibrometer is all that is required. The eccentricity measurement is considered sufficient to give an indication of the dynamic balance of the turbine.

(6) Speed. In order to facilitate the interpretation of data on quick starting a measurement of turbine speed is always made.

1.11 Thermal Considerations

The use of high temperature steam in modern turbines introduced the need for the information given above, but, apart from the variables referred to, there are other considerations.

(1) Thermal Stresses in the Casing. When a difference of temperature exists between the inside and outside of a thick walled cylinder, the inner surface becomes heated first and there is a relative expansion between the inner and outer surfaces which produces stresses. The temperature differences should not be greater than 100°F if the elastic limit of the metal is not to be exceeded.

The normal method of measurement of this temperature difference is by the thermo-couples in the outside of the casing. Estimates are then made or tests are carried out to measure the temperature gradient across the metal thickness. The steam temperature at the stop valve is then allowed to exceed the outer metal temperature by between 100°F and 200°F.

(2) Differential Expansion Stresses. There are a number of places where differential expansion stresses may be important. In particular, under starting conditions severe temperature differentials may exist between the surfaces of the horizontal joint flange and the bolts and result in greatly increased stresses. Such conditions could cause crushing of the joint faces and overstressing of the bolts. The controlled use of flange heating steam reduces these effects. Other areas liable to overstressing are steam chest joints and nozzle box joints.

(3) Thermal Distortion of the Casing. Even in the simplest examples, a turbine casing is a complex structure. Uneven metal thicknesses when combined with uneven heating may result in distortion. Such distortion has no permanent effect on the life of the cylinder, but it may destroy some small radial clearances and cause a rub, especially at the glands.

A severe rub would cause local over-heating and possibly shaft bending, followed by heavy vibration and further damage. The avoidance of distortion is primarily a problem of design and requires simplicity and symmetry of the casing design and both minimum and uniform metal thicknesses.

1.12 Technique

A large number of quick starting tests have been carried out and a considerable amount of information collected. Although certain principles can be laid down, experience must be the final guide for any given machine.

(A) Conditions at Start-Up

The severity of stresses and magnitude of expansions will depend on the condition of the turbine on start-up. It is convenient to sub-divide these conditions as follows:

(1) After a shut down of from 6 to 8 hours (for example, overnight.)

(2) After a shut down of from 12 to 14 hours, (for minor maintenance work, or at weekends).

(3) After a shut down exceeding 48 hours, (for major work or overhaul).

Tests show that little difficulty is experienced with a hot start from condition (1), varying degrees of difficulty from condition (2) and maximum difficulties with a start from condition (3). This progressive increase in difficulty is due to the fact that under condition (1) the differences from normal working will be at a minimum, while at (3) they will be at a maximum.

(B) Control of Steam Temperature

The avoidance of thermal stresses and uncontrollable expansions requires the precise temperature control of steam admitted to the turbine. There will be less control of steam temperature with turbines supplied from a pipe range than with plant arranged on a unit basis.

Where steam is supplied from a pipe range the steam will be at full temperature and, depending upon start-up conditions, will be in excess of that required for the turbine. Under these conditions thermal stresses may be excessive. Also, the rotor, being of a considerably lighter mass than the casing, will expand at a faster rate than the casing and the available differential may disappear, thus causing a rub. Flange heating will reduce both these possibilities by supplying heat to the casing to reduce the expansion lag. Obviously, when starting up with full temperature steam from a pipe range, there will be less difficulty with a hot start than with a cold one.

With the unit arrangement the steam temperature, after the initial warming up of steam pipes, will depend on the heating rate of the boiler. Difficulties are again dependent upon the temperature and pressure conditions in the turbine at start up. For a cold start, little difficulty need be anticipated and the turbine and boiler can be brought up together; the matching of steam and metal temperatures being relatively easy.

With a hot start, however, it is likely that the steam temperature to the turbine will be lower than the temperature within the turbine and some cooling effects are likely. The effects noticeable immediately could be either joint failures or, more likely, a sharp axial contraction of the rotor and the possibility of a rub on the back of the moving blades.

The problem is to raise the temperature of the steam to a rate which will maintain the required differential between steam and metal temperatures. Difficulties in boiler operation may occur owing to the combination of low steam flow and relatively high steam temperature demands. It is possible to overcome these difficulties, however, by the provision of a turbine bypass to the condenser. This bypass must be capable of handling up to 25 percent of the maximum boiler output. This enables safe boiler heating rates to be used to raise the steam temperature to the required amount above the metal temperature before rolling is started. By this means, steam and metal temperatures may be readily matched after an 8 hour shut-down. Typical records of starting up a 60 MW machine after 2-day and 12-day shut-downs respectively are shown in figures 3 and 4. They are for a 1500 rpm machine but would be similar for an 1800 rpm machine.

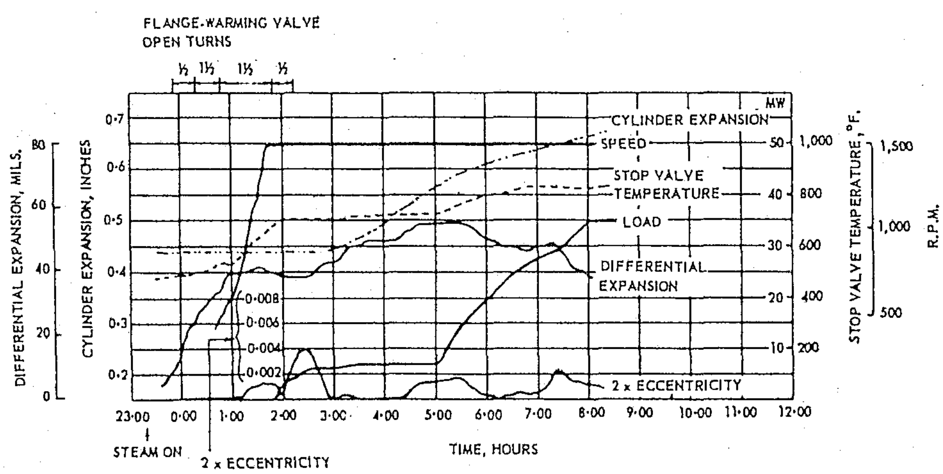


Figure 3. Start of 60 MW turbine after two day shut-down.

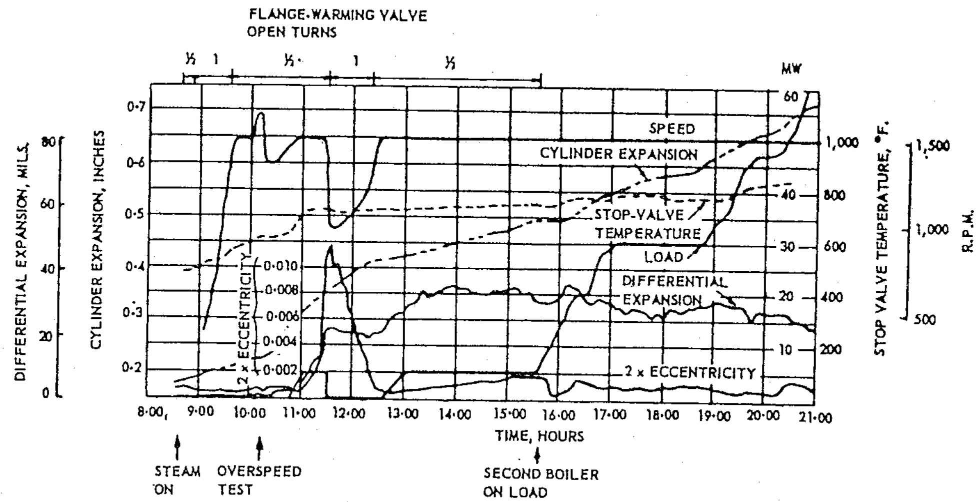


Figure 4. Start of 60 MW turbine after twelve day shut-down.

1.13 Efficiency Considerations and Testing

Considerations which affect the efficiency of steam turbines fall, roughly, into two categories, namely those connected with day-to-day operation and those that are associated with the internal conditions of the machine.

1:13:1 Day-to-Day Operation - Terminal Conditions

The terminal conditions, namely the steam pressure and temperature at the stop valve, the vacuum at exhaust flanges and the final feed temperatures, all have a significant effect on steam turbine efficiency. These variables have to be maintained within close limits of the specified conditions.

(1) Stop Valve Pressure. The effects of pressure are generally small and automatic control equipment of the required standard is usually installed to maintain this variable within the desired limits.

(2) Stop Valve Steam Temperature. An average change of heat consumption of a turbine of ± 1 percent can occur for a ± 40°F change in steam temperature. Since the normal tolerance allowed may result in an inherent inaccuracy of ± 20°F in 1,000°F, it will be seen that there may be 0.5 percent loss in the turbine's heat consumption from this cause. Until temperature measuring equipment of improved accuracy is made available, frequent calibration of steam temperature thermometers

should be made. Thermometric devices of a high standard of accuracy should be used for this.

(3) Back Pressure. The back pressure is the most significant of all the terminal conditions. As a rough guide a change of 0.25 in. of mercury has an effect of 1 percent on the heat consumption of the turbine. For exact knowledge of the variation of heat consumption with vacuum at various loads, reliance is placed on the maker's vacuum correction curves, a typical example of which is given in Figure 5. These are not always correct, especially where a better vacuum than specified is experienced. The effects of back pressure variations have been discussed previously. Experience indicates the need for special tests to check the makers' correction curves at the time of the official acceptance tests. Because of the importance of the back pressure on the assessment of the performance of a turbine, an accurate measurement is essential. The Kenotometer is used for vaccum measurement.

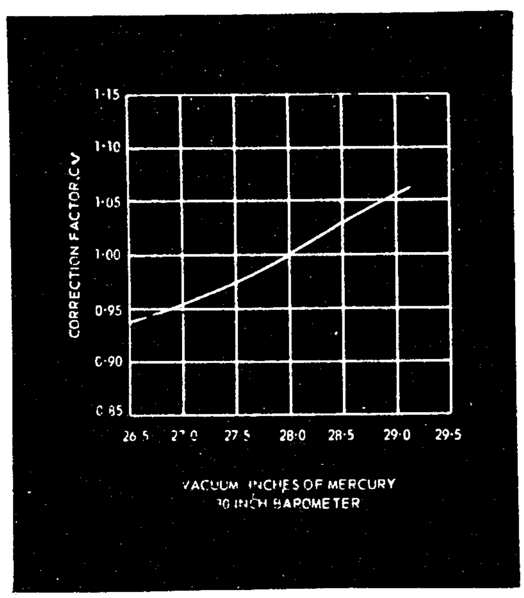


Figure 5. Vacuum correction factors.

On modern turbines, the exhaust area is very large and, although the design of exhaust passages should ideally give a uniform steam flow, in practice there will be variations in pressure across the exhaust area. Thus, to obtain an accurate overall vacuum measurement, it is desirable to take the average of a number of readings over the exhaust area.

For normal operation, two or more Kenotometers connected to distribution pipes across the exhaust flanges are sufficient. For testing work a number of individual measurements should be made using special manometers and the readings averaged.

(4) Final Feed Temperature. This variable is not dependent, to any extent, upon day-to-day operation but rather upon the condition of the feed heating plant.

1:13:2 Internal Condition of Turbine

The efficiency of a steam turbine is very much a function of its internal condition and in particular of:

(1) Condition of blading in respect of fouling, corrosion and erosion.

(2) Maintenance of the design clearances between fixed and moving parts. Considerations under (1) are dealt with fully in the section under blading.

Deterioration of clearances is difficult to detect without inspecting the turbine internally. It has a very bad effect on steam and, therefore, on heat consumption and as such is a particularly difficult problem from the efficiency point of view. Day-to-day deterioration takes place with no means of making accurate assessment of the amount on individual machines. One solution to this problem would be to carry out frequent full-scale consumption tests. This is, however, impracticable because of the large numbers of trained testing staff and the quantity of equipment needed.

Any system of monitoring the day-to-day heat consumption of the turbines depends upon the measurement of steam flow. Condensate metering is not reliable for this purpose because of the effect on the measurement by possible changes in hot well levels, bypassing and recirculation. A steam flow meter of the required accuracy does not at present exist. If, however, such a meter were developed and the accuracy of the associated panel instruments correspondingly improved, then monitoring the heat consumption would be a simple matter.

1:13:3 Testing

It will be seen from the above that the provision of accurate and reliable data concerning the variables involved is essential for the efficient operation of plant, particularly turbines.

Various tests may be carried out, and include:

(1) Full scale heat tests under acceptance test conditions.

(2) Routine testing using instruments provided.

(3) Restricted tests for special purposes.

Full scale tests are normally carried out as acceptance tests, as soon as possible after commissioning. Elaborate instrumentation and careful calibration are needed. The responsibility rests mainly with the contractor and special precautions are taken to ensure that the turbine and associated plant are in a condition for testing and that the turbine under test is completely isolated from adjacent turbines. Calculation of results would be in accordance with the method set out in the terms of contract.

1.14 Logging of Readings and Indication of Trends

The practice of logging instrument readings varies widely in different power stations. Whatever the method, the reasons are the same:-

(1) To ensure that all instruments are observed and their readings logged at regular fixed intervals to aid the detection and correction, where possible, of short term deviations.

(2) To provide a record of operation in numerical form over a given period, which can be analysed for turbine performance and from which long term trends can be detected.

The intervals between which instruments are read are selected in accordance with the conditions under which particular plant is operated. This may be either base load or two-shift. Attention has also to be paid to the nature of the readings. One hour intervals may suffice for base load plant where conditions remain relatively steady for long periods. Half-hourly or quarter-hourly intervals may be necessary for plant on two-shift or other operations of a variable nature. Certain instruments may have to be observed and their readings logged at much shorter intervals than the others because of the variable nature of the readings or for some other special purpose.

In practice operating results are based, over a period of time, on arithmetical or weighted averages and the interval between which readings are logged will depend to some extent upon the accuracy required. To be reliable, logged readings depend on the integrity of the operator in reading the instruments at the intervals laid down and in logging the actual reading. The operator's knowledge of particular instruments which require zeroing or balancing is also important, for example temperature indicating equipment on the bridge principle and the Kenotometer.

The short term deviations referred to in (1) are usually associated with normal adjustments required for load changes. Corrections will be made as required without waiting for the set time for log readings. Such deviations are, in general, unlikely to be serious. If, however, the safety of the plant is affected emergency action will have to be taken. Long term trends can only be detected by a trained engineer who has a number of aids to assist him.

The processing of the data, after the completion of the log sheet, is an important step. Some of the readings require a simple arithmetical averaging, while others may need weighting with some other variable; for example steam temperatures may need weighting with steam quantity or load. In all cases the processing should be carried out as soon as possible so that any necessary action may be initiated. In this connection it is desirable to avoid unnecessary work, such as working out averages to a higher degree of accuracy than the original readings.

Considerable reduction in the time required to process the data on a log sheet can be effected by the use of desk calculating machines.

Generally, trend indications appear only if there exists some basis for comparison. Two general forms of comparisons are used:

(a) Comparisons with past experience or behaviour under certain conditions, for example increasing or decreasing load, running up or shutting down.

(b) Comparison with standard readings taken when the turbine is in a known condition.

Past experience will be based on routines established by tests, as in the case of loading schedules and starting-up programmes. For these there will exist curves or tables. For standard conditions the turbine will be in a known condition immediately after acceptance tests, or after a major overhaul. Of these two, the former is to be preferred as the machine will then be in full commission and in, or near, its optimum condition.

At certain loads, say 20 percent, 40 percent, 60 percent and 100 percent of full load and with set terminal conditions of steam and vacuum, sets of readings should be taken of certain selected items. Repeat runs may be made to ensure that the readings are reproducible. The selected readings would include stop valve steam pressure, stop valve steam temperature, stage pressures, vacuum and turbovisory instrument readings.

There is some advantage in taking complete sets of readings, which could be used as standards and printed at the head of the log sheet olumns for the information and guidance of the operators. With these standards recorded, some, or all, of the loads, with the set terminal conditions, are reproduced at periods of, say, one month and a comparison of the results obtained is made with the standards. Some form of curve or string chart could be used to assist in detection of trends.

The importance of this careful and systematic analysis cannot be over emphasised. Shift operators, by the nature of their duties, are not in a position to detect long term trends but they eventually become aware of the developments, especially when they adversely affect operation.

Some illustrations of the type of trend referred to are:

(i) Blade fouling which is detected by stage conditions changing.

(ii) Bearing 'hot spots' which, in the early stages, may only slightly affect oil temperature.

(iii) Thrust deterioration detected by changes in turbovisory equipment readings.

In the future, the practices of logging may be modified by the increasing use of recorders and integrating meters. It must be appreciated, however, that these instruments are only of use after an incident and they are no substitute for the trained human observer able to take action on his observations.

1.15 Blading

If bearing, gland and other problems are disregarded it is axiomatic that the performance of a steam turbine is only as satisfactory as the condition of its blading allows. Apart from troubles associated with vibration of blades and mechanical damage caused by slugs of water or debris, the cumulative troubles associated with operation are almost entirely caused by the formation of deposits or blade erosion.

1:16:1 Deposits

Fouling of the blades of high pressure turbines is caused by deposits of material carried over in the steam from the boiler. The material carried over may consist of a variety of substances both soluble and insoluble in water, but the two chemicals which mainly cause deposit formation on turbine blades are sodium hydroxide (caustic soda) and silica.

The prevention of carry-over is both a chemical and a mechanical problem. Chemistry demands that the composition of the boiler water should not cause foaming, whilst mechanically, the steam separators and drum internals should be correctly designed and installed and properly maintained. Silica vaporizes at pressures upwards of 600 lb./sq. in. and this, therefore becomes a separate problem, as prevention of carry-over and attention to the design of drum internals will not prevent volatilised silica from leaving the boiler with the steam.

The mechanism of formation of deposits on turbine blades is not completely understood as yet, but information is available which broadly indicates two classes:

(A) Deposits Soluble in Water

Analyses of deposits found on various turbine blades show that deposits of this type are associated with sodium hydroxide. The melting point of sodium hydroxide is just over 600°F; therefore, at temperatures below this the caustic soda is solidifying and, together with any water it may contain, forms a sticky viscous material which readily adheres to the turbine blades. This also forms a base to which other salts, such as sulphates and phosphates, will become cemented. Deposits of this type may be expected in the parts of turbine where the temperature falls below 600°F and where the steam is not sufficiently wet to exercise a continuous washing action. For example, on a 3-cylinder turbine with steam initially at 900°F, the greater part of the intermediate pressure cylinder may be affected. These deposits are generally found on the concave surfaces of the blades, possibly owing to the inertia of the sodium hydroxide particles which prevents them from following the curved steam path, and thus they are carried into contact with the metal.

(B) Deposits Insoluble in Water

Analyses have shown that deposits of this type are invariably associated with silica ($Si\,O_2$). The point in the turbine where the silica begins to form deposits on the blades is rather difficult to establish, except by experience, because the crystallizing of the silica out of solution from the steam depends upon its vapour pressure, the temperature and, possibly, other variables which are not yet fully understood. Silica may be found adhering to the soluble deposits on the blades.

1:16:2 Effects of Deposits

Deposits on turbine blades decrease both the efficiency and the reliability of operation and also limit the capacity of the turbine. Figure 6 shows the effects of blade fouling on stage pressures.

(A) Decrease in Efficiency

Deposits on turbine blades cause partial choking of the area available for steam flow. This will result in an increased pressure difference across the groups of stages affected. The turbine log sheets will provide all the necessary evidence for the detection of blade fouling.

It can be shown that if:

P_1 = steam pressure before nozzles in first stage, lb./sq. in. absolute.

v_1 = volume per lb. of steam at p_1

w = steam flow through turbine in lb./hr.

then $w \dfrac{v_1}{P_1} = C$ (a constant for a given turbine)

C can be calculated for clean conditions and its value will then diminish as the blading becomes fouled. This method can be used for any section of the turbine and also at partial loads, provided reliable values of w and p_1 are known.

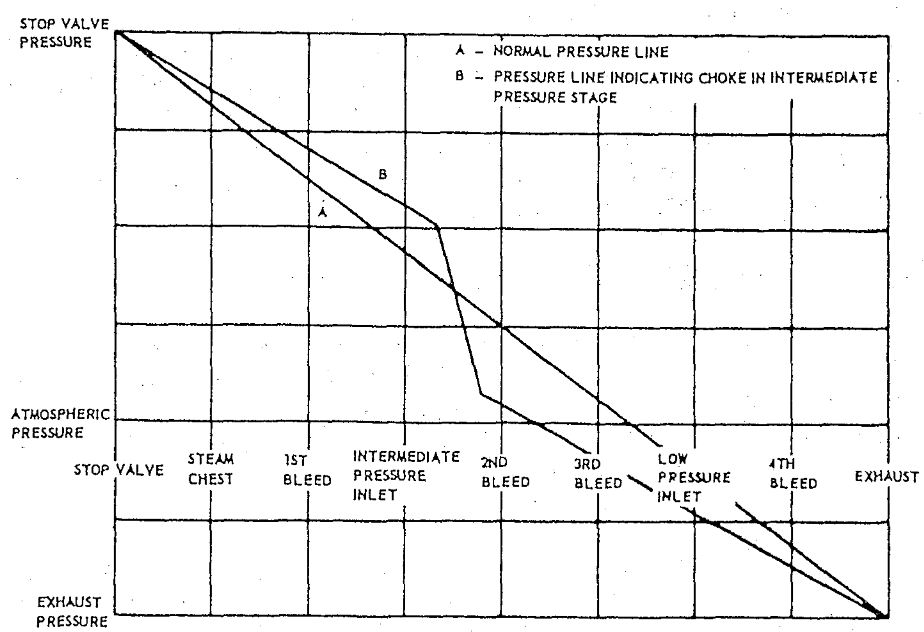


Figure 6. Turbine blade fouling indicated by stage pressures.

A simpler, although perhaps less sensitive method, would be to plot periodic readings of the stage pressures as given by the panel instruments at a given load. Any falling off in, say, bled steam pressures or intermediate pressure inlet steam would indicate fouling.

A quantitative estimate of the decrease in efficiency may be made from changes in internal efficiency in those groups of stages in which deposit formation occurs. Although such control is simple it is against accepted practice to have excessive temperature measuring facilities in turbines after the strainer, owing to the danger of damage from broken pockets.

In general, at the commencement of deposit formation there is a high rate of decrease of efficiency. The rate of decrease falls off as the deposits increase.

(B) Decrease in Reliability of Operation

This arises from the necessity to remove the deposits, and these operations, discussed later, usually mean loss of availability. The decision to take a turbine out of service to remove deposits will depend upon the cost of replacement plant. With modern high merit plant this cost is normally considerable and if outages become frequent, then the remedy is to investigate the cause by checking the condition of boiler water and separator maintenance.

If the blade fouling is allowed to persist, a condition may eventually be reached when the magnitude and direction of the thrusts are affected. The extent of the change would depend upon the type of machine, but if the trouble is not dealt with, thrust bearing failures and blade stripping may result. The possibility of mechanical damage will be increased if the deposits are formed in a non-uniform manner.

(C) Limitations to the Capacity of the Turbine

The progressive choking of the blade passages causes a reduction in total steam flow through the turbine for a given pressure before the first stage nozzles. This pressure can be taken as a basis for estimating the capacity limitation as it is usually a function of the number of control valves open, or the opening of the main throttle valve.

The upper limit to the first stage nozzle pressure occurs when it equals the stop valve pressure. It follows, therefore, that the upper limit of steam flow through the turbine occurs when no further increase of first stage nozzle pressure is possible. This also limits the capacity of the turbine. As blade fouling increases, the steam flow and, hence, the load, can be maintained up to the limit of pressure given above. If the turbine is to be kept in service under these conditions, then a load limitation must be accepted as well as a decrease in efficiency.

It is unusual for turbines to be maintained in service until deposits on blading have reached such proportions because of the danger of mechanical damage referred to above. The decision to remove these deposits is based on economics.

1:16:3 Methods of Removing Deposits from Turbine Blading

Certain types of deposits can be removed while the plant is in operation but outage of plant may be required before some types can be removed. This latter type of deposit may be either soluble or insoluble in water.

(A) Deposit Removal in Operation

Normally the presence of water in a steam turbine is to be avoided. There are, however, occasions, for example during starting up, when the steam is either saturated or becomes wet early in its expansion, when deposits will be dissolved by the moisture entrained in the steam. The effectiveness of this method depends upon the temperatures in the turbine and the temperature of the steam. The degree of washing will be at a maximum with a cold turbine containing steam below full temperature and will be at a minimum with a hot turbine after only a few hours shutdown containing steam at, or near, full temperature. A warning should be given here that on no account should saturated or wet steam be admitted to a hot turbine, owing to the possibility of sudden local cooling and high thermal stresses.

Insoluble silica deposits can be removed to some extent if there is a considerable change in temperature in the stages affected. Owing to the different co-efficients of expansion of the blade metal and the deposits, cracks are formed in the deposits, the deposits then fall off the blade surface and are carried away with the steam. Such changes in temperature would be caused by shutting down, starting up or by reasonably rapid load changes.

(B) Deposit Removal Requiring Outage

The withdrawal of a turbine from service for blade cleaning usually means that the degree of washing by other methods is insufficient. When a turbine is out of service, cooling may be either natural or artificial. To be effective, two or three days are normally required for natural cooling and such periods of outage are not always possible. One method of artificial cooling is to arrange a portable fan to blow air into the low pressure end of the turbine. The air then passes back through the turbine and out to atmosphere at the high pressure end. This method reduces the period required for cooling to between 12 to 24 hours. The use of air eliminates the possibility of local overcooling which might occur if saturated steam is used.

When the turbine has cooled sufficiently the temporary arrangements are removed, blanks replaced and the turbine started up. The condensate must be run to waste, sampled and tested periodically until the test indicates a high degree of purity. Following this the turbine may be brought up to speed. During the washing process the speed should not be allowed to exceed 25 percent of full speed and all casing drains must be open.

There are alternative methods of achieving the same object. One method is to deliberately reduce the steam temperature at a rate not exceeding 100°F per hour, while the machine is on load. When the lowest limit is reached, the machine should then be unloaded and its speed reduced to approximately 25 percent of its full speed by regulation of the stop valve, thus keeping the governor throttle valve or control valves fully open. Keeping to the above rate of change of temperature, water should then be injected to make the steam wet. The washing procedure is then similar to that described earlier.

After completion of washing the steam temperature may be raised to normal, at the above rate, following which the speed and load may be raised. In all these washing processes careful watch must be kept for signs of rubbing or other unusual occurrences.

With insoluble deposits, a loss of availability is inevitable where any considerable degree of cleaning is necessary. Some success has been obtained in the United States of America with a hot caustic soda wash. Generally, the procedure would be the same as for the wet steam and water wash for soluble deposits. The essential differences are:

(1) Vacuum is lowered to about 5 to 10 in. mercury absolute.

(2) Caustic is injected until it appears in the condensate.

(3) The turbine is shut down for about 15 minutes to allow caustic to soak in.

(4) The start-up to about 25 to 30 percent full speed and inject again.

(5) Repeat shut-down and soak cycle until negligible silica appears in condensate.

(6) Thoroughly wash the turbine with wet steam until condensate shows neither caustic nor silica.

All soft packings exposed to caustic must be replaced and valves with special alloy bushings and seatings cleaned and freed. Deposits can also be removed mechanically by opening up the turbine. Whichever method is chosen, it must be one which will not damage the blades or remove any of the parent metal.

A suitable method is to use a fly ash blast, using a suitably soft abrasive such as screened fly ash from pulverised fuel boilers. Care must be taken to ensure that deposit removal is uniform and this will mean removal of fixed diaphragms from the turbine. A disadvantage of the method is that an enclosure is required for the blasting operation. Yet a further method of silica removal is by immersing the elements in a bath of inhibited acid (hydrofluoric) but this requires skilled attention and few stations have the requisite facilities.

Extreme care is necessary during construction and overhaul to prevent the entry of dust and grit into the boiler internals, so that the silica concentration can be kept below three parts per million for 1,500 lb./sq. in. plant, and less than 1.5 parts per million for 2,300 lb./sq. in. plant.

1.17 Maintenance

Maintenance may be either routine day-to-day maintenance or maintenance which involves dismantling the machine. It is the latter which will be considered in this Section and is usually referred to as overhaul.

1:17:1 Frequency of Overhaul

Some power companies overhaul turbines every three years. It is, of course, sometimes necessary to dismantle a turbine for examination after periods of less than three years, but this would be because of unusual circumstances or known defects. Conversely, the dismantling of a set after three years may not, in fact, be necessary.

Except if major defects, which are known or suspected, occur, modern high merit plant should be kept in service for as long as possible. The tendency now is to keep plant in service until its performance deteriorates to a point where the cost of outage for overhaul is out-weighed by the cost of continuing to run the plant at a low efficiency.

Other factors affecting the decision to overhaul are the over-riding need for high availability and the outage of turbine and boiler at the same time in unit plant.

1:17:2 Preparation

It is an important part of the maintenance of steam turbines that the fullest attention be paid to trend indications gained from operating data and, also, that the fullest investigation be carried out, into reported abnormalities or defects in operation. With this information maintenance can be concentrated most effectively and provision for spares and shop facilities at makers' works can be made in advance.

Before a turbine is taken out of service for overhaul, all materials and spares required should be to hand. In some cases scaffolding and lifting gear can be arranged for special jobs. The sheet metal covers and some lagging can be removed. The important parts of the turbine must be exposed as soon as possible so that the maximum time will be available for correcting defects. A complete schedule of items requiring attention should be prepared and the work planned in proper sequences to avoid interference and delays.

1:17:3 Major Items Requiring Attention During Overhaul

The major parts of a turbine requiring maintenance under overhaul are detailed below.

(A) Blading

The condition of the blading can be gauged from operational data which may be supplemented, possibly, by consumption tests or stage pressure readings. Blade fouling has been dealt with but it is important to stress general cleaning, if necessary, which may entail removal of the lower half diaphragms as well as the rotor and top half diaphragms. The blading is inspected for evidence of erosion, corrosion or rubbing. Blades are 'dressed' as necessary; badly damaged sections are replaced.

In low pressure stages, lacing wires are replaced or rebrazed as necessary. Shroud bands in high pressure and intermediate pressure stages are inspected for excessive rubbing and are dressed or replaced. In many instances the remedial work may be of a temporary nature, but provision is made for future replacement as material becomes available.

Turbine blades are inspected for cracks in the blade or at the root, particularly in the low pressure stages. This test is usually carried out using magnetic fluid and a permanent magnet. A trained observer should carry out this work. Most repair work on blading requires a specially trained operative.

(B.) Glands

Included under this heading are diaphragm shaft packings and dummy piston packings in reaction turbines which are all usually of the labyrinth type. Indication of the condition of the casing glands would be given by the operators, if excess steam were needed for sealing. Packings are cleaned, straightened where necessary and adjustments made to give correct clearances . Spring loaded sections, if fitted, are adjusted. Badly worn or damaged sections are usually replaced.

Some attention should be given to the axial and radial shaft landings. If badly damaged owing to a heavy rub or foreign matter, the makers should be consulted. Low pressure shafts may be re-built by one of the metal spraying processes. Alternatively, landings have been turned down to give a smooth shaft and packings adjusted to the new diameter. If shafts are sleeved instead of having integral glands then replacement will be necessary. Gland steam pipes, vent pipes and drainage holes are examined for cleanliness.

(C) Diaphragms (including nozzles)

Diaphragms are inspected for cracks by the magnetic particle method, or, possibly, by hammer test. Checks are also made for distortion, proper fit in casing grooves and erosion of the landings. Evidence is also sought of rubbing. The diaphragm halves are often removed from the casing grooves so that the latter may be cleaned, and seizing because of corrosion, at some future date prevented.

It is important that diaphragms should be a good axial fit in the grooves and also that there is provision for radial clearance in the grooves. The nozzles are examined for deposits and are cleaned if necessary. The edges of nozzles may be 'nicked' by foreign matter. These can be straightened by a light hammer and steel block.

(D) Alignment

Because of the speed of the rotating masses and the large out-of-balance forces which can appear as vibration, the alignment of a large modern turbine is very carefully carried out when erected.

The general principle of alignment is that, assuming the coupling faces to be true with the shafts, the shafts are aligned in such a way that a continuous curve is formed, with their natural deflections, from governor to exciter. This point is

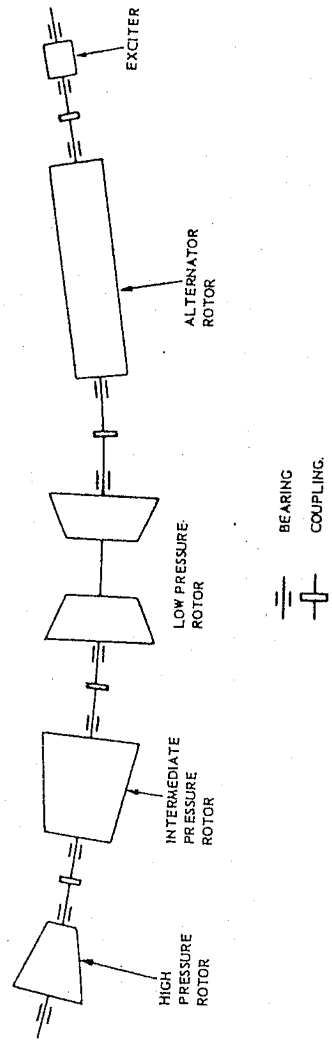


Figure 7. Arrangement curve for turbine-generator.

Figure 7

illustrated in Figure 7. The shafts will retain their natural deflection at any speed other than the critical speeds. Adjustment to give correct alignment is carried out by the adjustment of bearing positions to match the static deflection of the shaft.

It is not necessary to know the shaft deflection curve as correct alignment is obtained by accurate measurements between coupling faces and over the coupling periphery. When equal measurements are obtained by clock gauge or feelers at four points 90° apart round the coupling periphery at locations x and y, as shown in Figure 8, then correct alignment can be assumed, provided that the coupling faces and periphery are 'true' with the shaft.

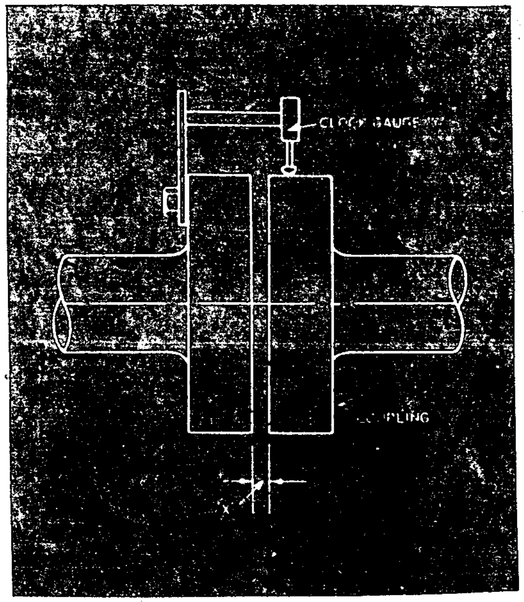


Figure 8. Alignment measurements at coupling.

Two general cases of misalignment occur:

(1) The axes of the two shafts may meet but may not be in a straight line.

(2) The axes may be parallel but may not be in line.

These two cases are as shown in Figures 9 and 10.

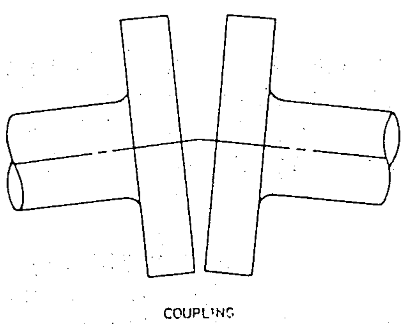


Figure 9. Shafts meeting but out of line.

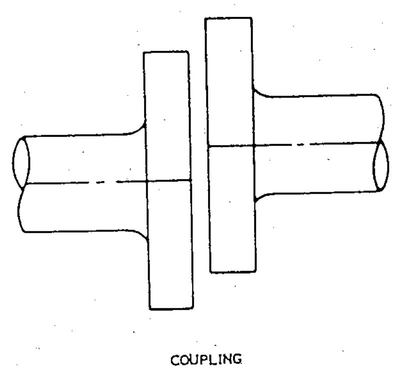


Figure 10. Shafts parallel but out of line.

Most makers supply an alignment gauge for a particular machine which consists of a plate with a gap to cross the coupling and which has two true edges accurately aligned as illustrated in Figure 11. When applied across a coupling, if both edges are wholly in contact with the shafts on each side, the correct alignment is established. Misalignment of the type indicated in (1) and (2) on the previous page will be revealed in the manner shown in Figure 11. The gauge can also be laid on the horizontal joint to check horizontal alignment.

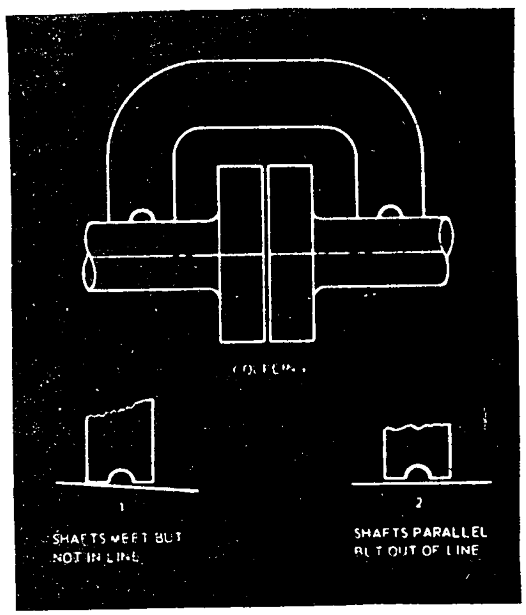


Figure 11. Use of alignment gauge.

(E) Clearances

The efficient operation of a turbine depends to a large extent on the maintenance of the correct clearances between fixed and moving elements. Excessive clearances result in increased steam consumption while reduced clearances may result in blade rubbing.

When a turbine is erected the clearances are carefully set and a record is kept at the station. When the top halves of the casing are removed the clearances should be checked against the record. Care must be taken to ensure that the rotors are in the running position when taking measurements. Provision is usually made to move the rotor axially to a position for lifting and casing.

Particular care is necessary with the clearances at the velocity stages now frequently fitted to the high pressure end of impulse machines. A thorough check of clearances is essential if any replacement blades, nozzles or packing rings have been fitted.

(F) Bearings

A thorough examination is made of bearings for wear, grooving of the bearing metal and shaft, loose bearing metal, correct contact surface and possible evidence of electrolysis. Modern bearings are of the spherically seated type and the fit in the housing is checked for tightness and alignment, adjustments are made if required.

The condition of oil orifices, including the area for high pressure jacking oil, oil throwers, baffles and the cleanliness of all oil and water passages are checked. It is usual to measure and record bearing clearances. For this purpose a bridge gauge is used and the measurement is compared with previous records. Variations will indicate bearing wear or settlement. A typical permissible clearance is 0.001 inch to 0.002 inch per inch diameter of journal.

Main thrust bearings are of the usual Michell type and little wear is experienced, normally. The pads, however, should be checked for freedom of movement.

D. Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-4 - Turbine Operational Problems

A - Assignment

1. Why is it important to thoroughly clean all turbine, heater & piping internals during the pre-commissioning period?
2. Why must the turbine never be started up without adequate lagging on high pressure steam pipes, and cylinders?
3. What is the purpose of limiting the rate of loading of a turbine-generator?
4. What dangers may exist when a turbine is persistently overloaded?
5. Why does starting up after a 48 hr. shutdown result in a more severe condition than starting up after a 6 to 8 hr. shutdown?
6. Describe the type of deposits that can be expected on turbine blading.
7. Would you expect any deposits due to sodium hydroxide on turbine blading in a present-day Canadian nuclear power station?
8. Name at least five turbine components that normally require maintenance during an overhaul.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator, & Auxiliaries

-5 - Gains Due to Regenerative Feedheating

0.0 INTRODUCTION

The regenerative cycle involving the extraction of steam from turbines for the purpose of heating the feedwater was invented by Ferranti in about 1905 but was not commercially applied until some time later. As we've mentioned previously the advantages of this cycle are:

(1) the reduction of temperature stresses in the boiler by introducing hot feedwater rather than cold water.

(2) the increased economy, because of the lower heat rate. Feedheating makes use of latent heat of vaporization during condensation of steam which would otherwise be lost in the condenser.

In order to evaluate the advantages of a regenerative heating system, calculations referred to as a heat balance are done. Since the Law of Conservation of energy states that energy cannot be created or destroyed we know that energy which enters a system must eventually also come out. Therefore when doing a heat balance all the energy flowing into the system should equal all the energy flowing out.

The importance of the heat balance in the design of steam power plants cannot be overestimated. Once the capacity of the station is determined, the proper size of the boilers, heaters, condenser, pumps, piping--in fact practically all the mechanical equipment-- can be determined solely by a heat balance. The weights and sizes of most of the mechanical equipment to be used for determining the sizes of structural member will depend on the heat-balance requirements. Likewise the electrical system for many of the station auxiliaries, such as pumps will depend on heat-balance determinations. Such economic factors as the advisability of including certain equipment in the cycle or the cost of the electrical energy generated can be decided only after sufficient heat balances have been made.

Heat balance calculations can become quite involed and a full treatment of this subject is beyond the level of this course. Therefore, this lesson will illustrate the advantages of regenerative feedheating, by means of various curves which have been obtained from heat balance calculations, but we will not show how these curves were arrived at.

1.0 INFORMATION

Regenerative Cycles Without Reheating

Referring to figure 1 raising feedwater temperature to highest level at h_f means less fuel is needed to evaporate each lb. to the throttle-steam state. Amount of heating that the extracted steam

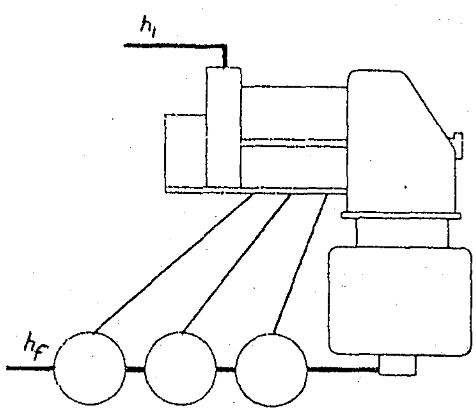


Fig. 1 Regenerative heating cycle extracts steam to heat returning feedwater.

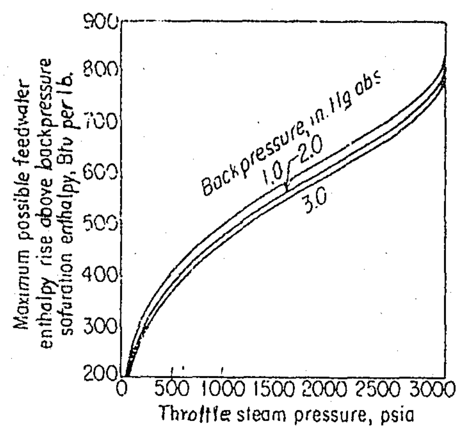


Fig. 2 Feedwater can be heated through higher ranges as steam pressure rises.

can do depends on throttle pressure of the cycle. This is illustrated in figure 2 where maximum enthalpy rise above condensate hotwell enthalpy is plotted against throttle steam pressure for three different backpressures. The higher the throttle pressure the greater the enthalpy rise of the feedwater.

The boiler has to add $(h_1 - h_f)$ Btu. to each lb. of steam. In the straight-condensing cycle, h_f is saturated-liquid enthalpy at the condenser pressure. In the regenerative cycle h_f ideally can be raised to the saturation enthalpy of the throttle pressure (if it's less than the critical pressure of steam, 3206.2 psia.)

Figure 3 shows reduction in straight-condensing-cycle heat rates for a range of throttle pressures and temperatures when

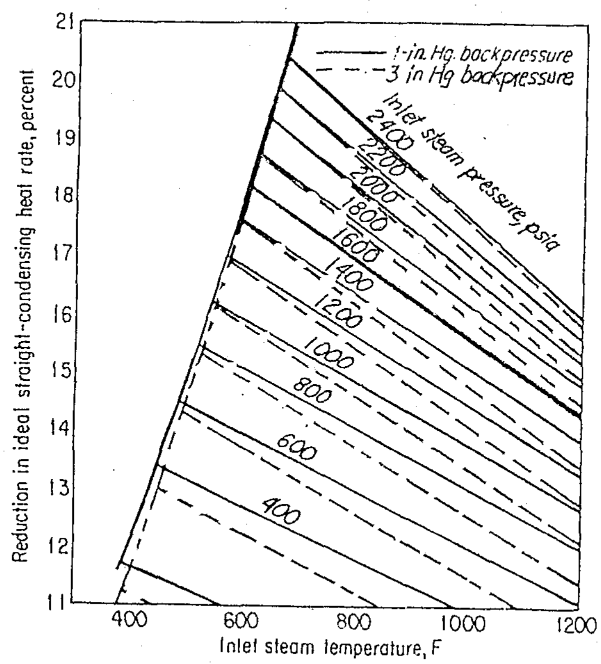


Fig. 3 Rising steam pressure improves gain in regenerative-cycle heat rate, but that's offset by rising inlet-steam temperature.

feedwater is heated regeneratively to the maximum possible. But these curves assume that heating is done reversibly in an infinite number of heaters. Actual regenerative cycles must be limited to relatively few heaters, so we cannot achieve ideal performance. Figure 4 shows the ratio of actual to theoretical reduction in straight-condensing heat rate plotted against % of maximum possible rise in feedwater enthalpy for various numbers of heaters in the cycle. It shows that the greater the number of feed-heaters, the closer the actual reduction in heat rate will be to the ideal reduction in heat rate.

For example, let's assume the 5 KW turbine mentioned in the lesson on "Improving Turbine Performance", T.T.1 level, has three feedwater heaters. The inlet steam conditions were 800°F and 1000 psia. Let's estimate actual heat rate with 1" Hg. abs. backpressure in the condenser and feedwater heated to 70 % of maximum possible rise.

For the straight-condensing example given in the lesson on "Improving Turbine Performance" the heat rate was 11,350 Btu/kw hr. From figure 2, maximum possible rise in feedwater enthalpy above backpressure saturation enthalpy is 495 Btu/lb. Then the actual enthalpy rise due to feedheating is:

Actual enthalpy rise = 495 x 70% = 346 Btu/lb.

From figure 3 the theoretical reduction of heat rate in straight condensing turbine = 14.8%, and from figure 4, for 3 heaters the ratio of:

$$\frac{\text{Actual reduction of heat rate}}{\text{Theoretical reduction of heat rate}} = 0.71$$

Then the correction factor for the actual regenerative cycle as compared to straight condensing is:

$$\text{Correction factor} = 1 - 0.148 \times 0.71 = 0.895$$

Thus the full-load regenerative-cycle heat rate =

$$11,350 \times 0.895 = 10,160 \text{ Btu/kw hr.}$$

This is 11,350 - 10,160 = 1,190 Btu/k.w.hr. better than for a unit without regenerative feedheating.

For any number of heaters there is a certain enthalpy rise at which economy gain becomes a maximum, as figure 4 shows. (The symbol for infinity is ∞, and therefore the top curve is representative of an infinite number of heaters in the regenerative system). For higher or lower enthalpy rises the gain will be less. For example, if one heater takes steam from the main steam line to produce 100% feedwater enthalpy rise, there is no economy gain

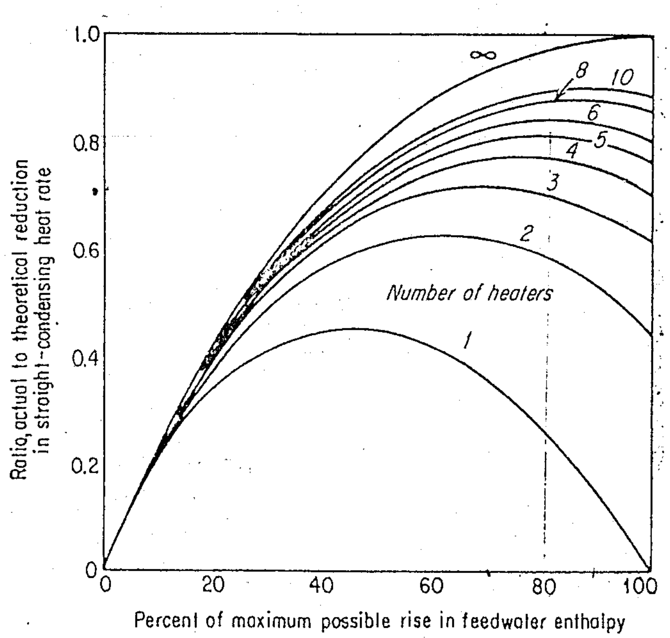


Fig. 4 More extraction points, properly placed in an actual turbine regenerative cycle, improve the heat rate.

because the heating steam has done no work in the turbine. At the other extreme, a heater tied to the exhaust stage cann't do any heating because steam and condensate have the same temperature.

Let's extend our study a little more by finding the heat rate of our 5 MW turbine when it uses ten heaters for feedwater heating to 90% of maximum possible rise. From figure 4, the heat-rate reduction for number of heaters and actual enthalpy rise is 0.89 . Then the correction factor for the actual regenerative cycle becomes 1-0.148 x 0.89 = 0.868. Full-load 10-heater regenerative cycle heat rate becomes 11,350 x 0.868 = 9,850 Btu/k.w.hr. Adding (10 - 3) = 7 heaters reduces the heat rate by (10,160 - 9,850) = 310 Btu/k.w.hr. Compare this with a gain of 1,190 Btu/k.w.hr. for the first 3 heaters.

To decide on number of heaters, we would have to compare charges on the extra heaters, piping and valves to the reduction in annual costs. If saving is greater than carrying charges, the larger number of heaters would probably be justified.

Regenerative Cycles With Reheating

This type of cycle combines steam-reheating and regenerative-feedwater-heating principles to achieve overall thermal efficiencies. Coupling high efficiencies with high throttle pressures and temperatures, these modern turbine cycles appear to be trending to the ultimate in overall thermal performance.

Figure 5 shows full-load heat rates that can be achieved by turbines with feedheating ranging in capacity from 100 to 500 MW, with throttle pressures ranging from 1000 to 6000 psig. Initial and reheat temperatures are kept at 1000°F for all units and backpressure is constant at 1.5 in. Hg. absolute. All units use six stages of regenerative feedwater heating. Their gross heat rates are figured on generator-terminal outputs.

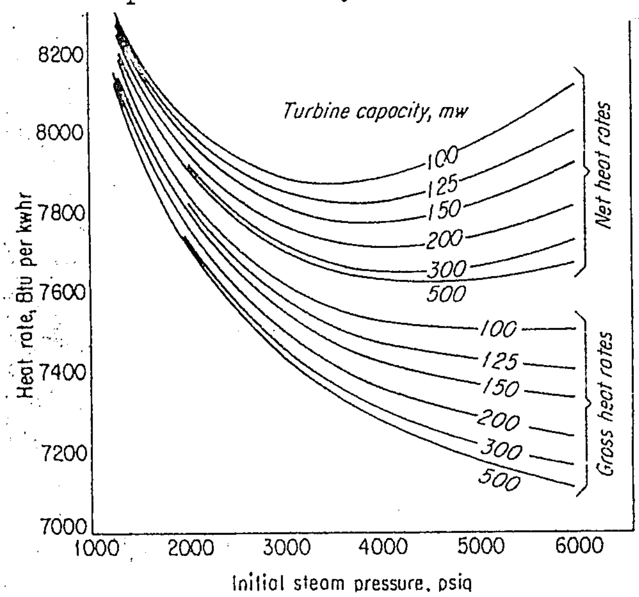


Fig. 5 Full-load heat rates predicted for turbines with six heaters, 1000/1000°F steam 1.5 in.Hg.abs. exhaust pressure.

Deducting feedwater-pump energy input from generator output gives net heat rates in this particular case.

The difference between the net heat rate curves and the gross heat rate curves shows the large amount of work input the feedwater pump requires at high throttle pressures. Taking this into account we find that the optimum net heat rate for the 100 MW unit comes at about 3400 psig throttle pressure. Best pressure for the 500 MW unit is about 4400 psig. The spread of curves as the pressure increases show the performance gain that can be made by going to larger capacity units at higher throttle steam pressures. Capacities, steam conditions, cycles, investment and fuel costs--all must be carefully coordinated to arrive at best choice for a central station generating unit.

Typical large condensing turbines with regenerative feedheating vary in their heat rate performance with changing load and different backpressures, figures 6 to 9. As units become larger and steam conditions higher, heat rates over the range of outputs become lower and lower. This is partly because of larger size, partly because higher steam conditions can be justified for the larger units, fig. 5. This gives them an inherent thermodynamic advantage.

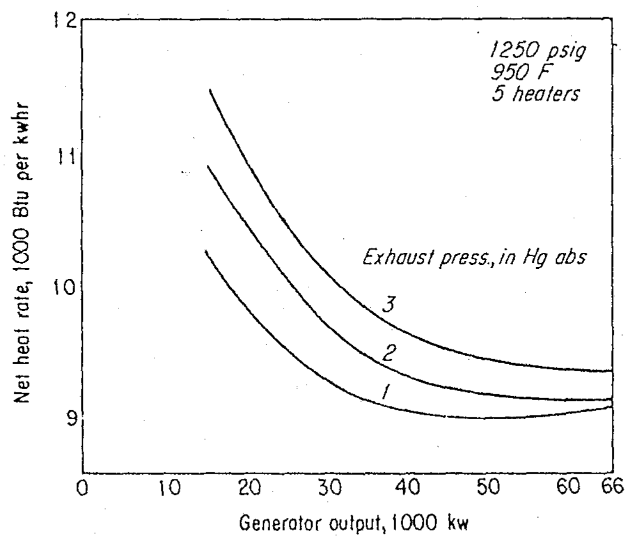


Fig. 6 Typical heat rates of 66 MW tandem-compound double-flow 3600 rpm unit vary with load and backpressure.

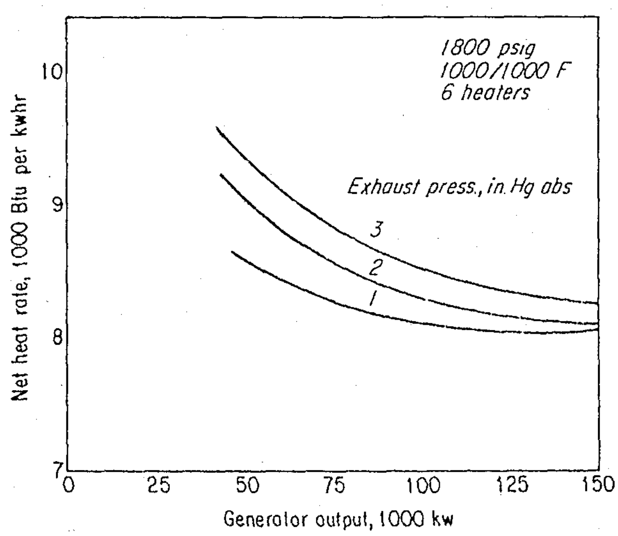
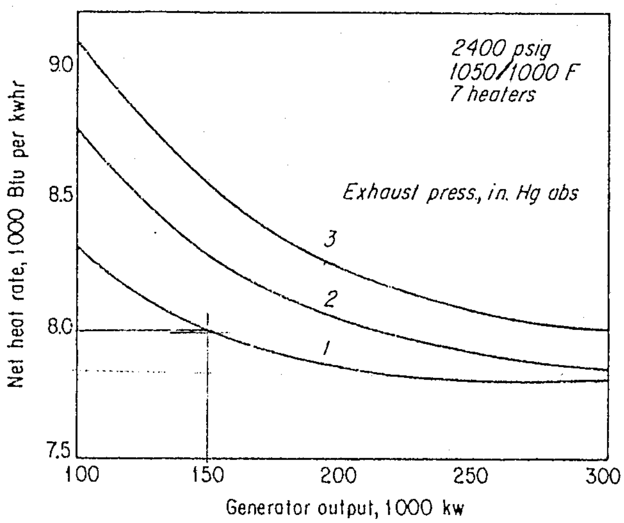


Fig. 7 A 150 MW tandem-compound double-flow turbine with higher steam conditions works at lower overall heat rates.

In figure 6 to 9 heat rates are usually minimum near maximum capacity. This follows because throttling isn't needed to control steam flow at full load.



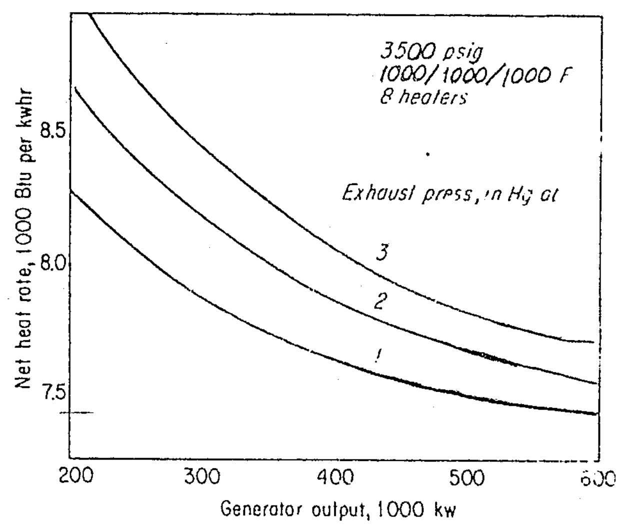


Fig. 8 A 300 MW tandem-compound four-flow turbine with still higher inlet steam shows more heat-rate improvement.

Fig. 9 A 600 MW cross-compound four-flow turbine with 3600/1800 rpm. shafts shows best performance of the four.

Dick Dueck

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator, & Auxiliaries

-5 - Gains Due to Regenerative Feedheating

A - Assignment

1. What is meant by the term "Heat Balance" in respect to a regenerative feedheating cycle?

2. In figure 2 of this lesson, why is the maximum possible feedwater enthalpy rise greater with a 1" Hg. abs. backpressure than with a 3" Hg. abs. backpressure?

3. A 20,000 k.w. turbine receives throttle steam at 600 psia and 600°F and exhausts to 1" Hg. abs. It has 5 feedheaters and feedwater is heated to 80% of maximum possible. Calculate the full load heat rate for this cycle, assuming a straight condensing heat rate of 10,500. Use information available in this lesson.

4. Compare the net heat rate for a 600 MW double reheat unit, throttle pressure of 3500 psig with a 300 MW unit, throttle pressure of 2400 psig. Assume a backpressure of 1" Hg. abs. Use information available in this lesson.

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-6 - Governing Theory

0.0 INTRODUCTION

In the lesson on governing at the T.T.2 level we have described various components of the governing system. In this lesson we will describe some theory which is the basis for designing governing systems.

1.0 INFORMATION

Before we can discuss the effect of governing and load and blade types on the shape of the condition curve it is necessary to understand the variation of the pressure distribution within the turbine for changes in the steam flow. It can be proved that the flow is approximately proportional to the inlet pressure. Since the flow through a stage of a turbine is essentially the same as the flow through the inlet nozzles the increase or decrease of the steam flow will cause a corresponding increase or decrease in the pressure differential for each stage and therefore for the entire turbine. However the exhaust pressure for most turbines remains nearly constant for changes in steam flow, so that the initial pressure for a group of stages will increase when the flow is increased. Although this straight-line relationship is not absolutely correct, it is so close to the more complicated but more accurate solutions that it is used extensively in power-plant design. Since no turbine operates with zero exhaust pressure the stage-pressure curve will bend at very low flows, as indicated by the solid line in figure 1.

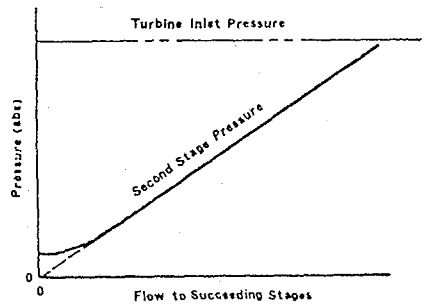


Fig. 1 Variation of stage pressure with flow.

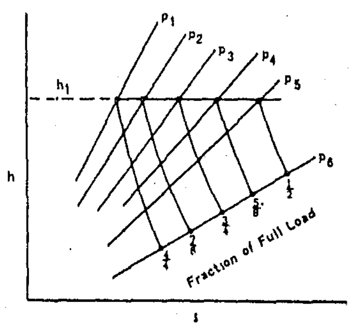
As we've mentioned previously governing of a turbine may be accomplished in two basic ways:

(1) Throttle governing--reducing flow by decreasing the inlet pressure.
(2) Nozzle governing--reducing the steam flow with constant inlet pressure by reducing the area available to the steam jet.

We also mentioned that throttle governing inefficient for a turbine to be used mostly for peak loading but it is desirable for a turbine to be used for base loading because it involves a simpler governing system. Nozzle governing is employed for a turbine used for peak loading.

<u>Throttle Governing</u>

The effect of throttle governing on the condition line is shown in figure 2. When no throttling takes place--i.e. the governing valve is fully open, then turbine inlet pressure is equal to stop valve pressure p_1 and the turbine produces 4/4 of full load. If the governing valves throttle the steam to say p_3 then the turbine would produce say 3/4 of full load and so on. Since a throttling process is one of constant enthalpy, the thermal energy of the steam entering the turbine is the same for all loads; but because of the increase in entropy, there is less available energy during the expansion, and because of the decreased initial turbine inlet pressure, the flow is also less for fractional loads.



<u>Fig. 2</u> Effect on the condition line of throttle governing.

<u>Nozzle Governing</u>

Nozzle governing involves the use of several valves instead of one and as we've mentioned previously these valves are located in the steam chest of an impulse turbine or impulse-reaction turbine and each admits steam to a different portion of the nozzle ring. The valves may open sequentially or in pairs, but in any event the throttling effect will be reduced because only a portion of the total steam flow will be throttled. For any given group of valves open at one time the pressure drop across the first impulse row will be the difference between the constant inlet pressure and the pressure curve for the second stage as determined by the method previously outlined in figure 1. Therefore, as the flow through the turbine decreases, the pressure drop and the thermal head for the first stage will increase or the percentage of the turbine load that is developed by the first stage will increase.

Since the reverse of this is also true when increasing load, it can be seen that a point will be reached where the pressure differential across the first stage will become so small that it will be impossible to pass sufficient steam through the first stage to develop the required load by the remaining stages of the turbine will operate under unfavourable conditions with a resulting poor efficiency. For this reason the first stage is by-passed only when the turbine is operating under heavy load and efficiency becomes of minor importance also, by-pass governing subjects the shell to abnormally high temperatures and the seals to abnormally high pressures.

When by-passing any stages of a turbine some small quantity of the steam must be allowed to pass through the by-passed stages to provide adequate cooling. The windage effect of the blades in the dense atmosphere of steam can easily be great enough to increase the blade temperature and to reduce the strength of the metal. Figure 3 shows the effect of multivalve governing on the condition line. Note that the first stage develops a larger percentage of the turbine load as the flow is reduced.

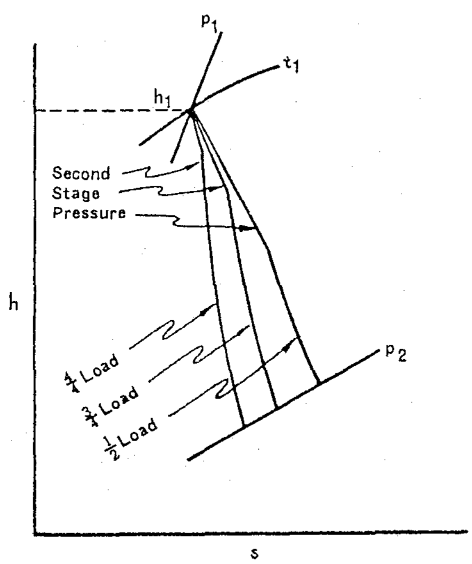


Fig. 3 Effect on the condition line of multi-valve governing.

Curves of Steam Rate

We have mentioned the effect of throttle and nozzle governing on the condition curve and it was observed that in both cases the enthalpy differential per pound of steam for the turbine was decreased as the load decreased. This would indicate that the steam rate (lb/kwh.) increases as the load decreases, as shown by the solid line, figure 4, for a turbine that has an infinite number of governor valves, and that the nozzle area decreases in infinitely small increments as the load decreases. Under such conditions, no wasteful throttling action would occur at any time, and the steam-rate curve would be at a minimum for all loads.

The decrease in efficiency as represented by the increase in steam rate for loads above and below most efficient load, would then be due to the decrease in blade efficiency and to leakage, windage and friction.

Since some steam would be required to operate the turbine generator at rated speed but with no load, the steam-rate curve would reach infinity at zero load.

With a finite number of governor valves, the steam-rate curve would have a number of humps as shown in figure 4, and part of the steam would be throttled as each valve gradually closed. The

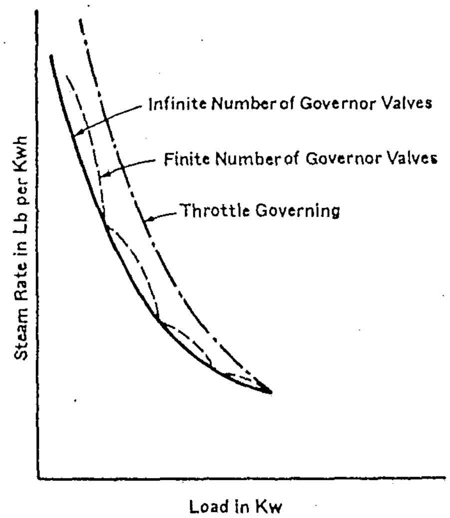


Fig. 4 Steam rate curves

envelope of this curve, i.e. a curve through points representing complete closure of one or more valves, would be the curve for an infinite number of valves. As the number of governor valves is decreased, the steam rate at low load will be further removed from the optimum until the limit is reached when only one valve (throttle governing) is used.

Speeder Gear

Before we go on todiscuss governor characterisitcs let us review what we have previously said about the speeder gear. We mentioned that governors are fitted with stops which limit the speed range, above and below the running speed, over which they operate. Within this range the speed may be controlled by an operator, who makes an adjustment to the relationship between the radius of the governor weights and the position of the governing valves of the turbine. The adjusting gear, known as speeder gear, may be operated by hand or remotely by electrical control of a motor. Using a worm drive followed by a screw, the adjustment may be affected in a number of different ways, for example by controlling the tension of the governor spring if of the axial type or by controlling the position of a sliding sleeve, containing ports surrounding the pilot valve of the primary relay.

When starting, the governor weights are against their inner stop and the governing valves are opened by using a starting handwheel to move the pilot bobbin of the valve relay, thus providing a course adjustment in the linkage. When the turbine runs into the governor speed range the governor weights begin to open, and the speeder gear may then be used for synchromizing the machine with the transmission grid. Thereafter the gear is used for controlling the power output, the speed being fixed by the system frequency.

How does a governor behave? As a first step to understanding we can remove the speeder spring in figure 5, and find out how flyweights act by themselves. First let us hold the weights in their innermost position--weights in--with the speeder rod, and turn the assembly at its rated 100% speed. We'll exert a downward resisting force 'F' on the rod while weights turn through a circle with radius 'R'.

If the weights move through a circle with larger radius, we find that the force on the speeder rod is larger, though the speed is still 100%. This is shown by the relationship:

$$\text{Force } F = MR(\text{rpm})^2$$

where F = force, lbs.

M = mass of flyweights

R = radius of flyweight motion

rpm = revolutions per minute

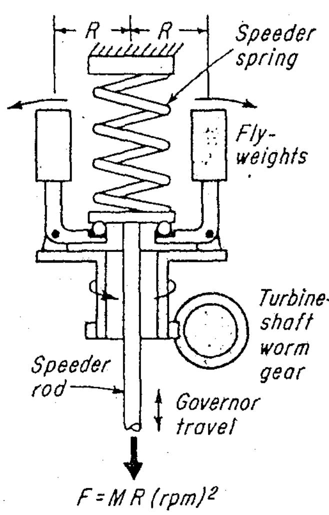


Fig. 5. Flyweight governor balances the force of spring and weights, and moves the speeder rod.

Using this equation and plotting force 'F' against governor travel or speeder-rod position for various speeds from 98 to 102%, we get the family of straight-line curves in figure 6, which ideally converge at a common point off to the lower left of the chart. Slope of each speed curve has a scale reading in lb. force per inch of governor travel.

A speeder spring with a scale to match the 100% curve develops the same resisting force 'F' for a given lb. per in. of compression or governor travel. Spring scales set at the middle of governor travel at 100% speed would be fine as long as all factors stayed constant. But as soon as load on the unit decreased, the shaft would speed up. Then the weights would develop more force than the spring over the range of governor travel. Weights would fly to their weights-out limit shutting the steam valve.

As steam flow stopped, the unit would slow down. Slightly below 100% speed the springs' mechanical force would overbalance the weights' centrifugal force and slam them to weights-in, completely opening the steam valve. So if weight and spring scales are equal they're continually fighting, producing wild hunting from fully shut to wide open throttle valve, without hope of reaching a balance. Obviously this is no way to govern a turbine.

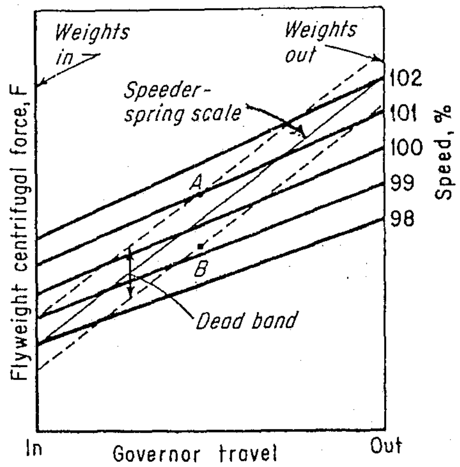


Fig. 6 Speeder-spring force must grow faster than the weight force as speed rises.

Speed Droop

To get out of the above mentioned dilemma, the spring force must grow faster than weight force as speed rises. The spring scale should be steeper (large) than the flyweight scale. This is represented by the thin solid line in figure 6, which rises more steeply than the heavier lines.

Suppose the turbine is running at 100% speed, where the speeder-spring curve crosses the 100% speed-curve. When the shaft load drops, shaft speed rises. Then increased force of the flyweights momentarily overbalances the spring force, raising the speeder rod to decrease throttle-valve opening. Decreased steam flow limits shaft speed rise, so spring force again balances the higher flyweight force at the new higher speed. The governor can't travel to its weights-out limit because the spring exerts greater force than the weights beyond the new point of balance.

If the turbine is running at 100% speed and load rises, the shaft slows down. Then the flyweights lesser force lets the over-balancing spring force lower the speeder rod to open the throttle valve wider. Increased steam limits the speed drop so the diminishing spring force matches weight force at the lower speed. The governor won't travel to its weights-in limit because the weights exert more force than the spring below the new balance point.

At each point in governor travel the turbine runs at a definite speed, slower at full load and faster at no load. For the spring scale in figure 6 speed varies by 4% from full to no load. This is the governor's regulation or speed droop. By definition then:

Speed Droop = the percent of speed change of a turbine-generator when changing from full load to no load. This can be expressed in equation form as:

$$S = \frac{N_n - N_f}{N_f}$$

where S = governor regulation or speed droop,%

N_n = no-load speed, rpm.

N_f = full-load speed, rpm.

Frictional Effects

Frictional effects ignored in the ideal model, play a big part in design of a working governor. Suppose the unit is running at 98% speed in figure 6 at the weights in position and then speeds up. Weights force rises, but because of friction in the linkage and glands the governor dosn't move immediately . By the time the turbine reaches 99% speed, force is large enough to overcome friction and the governor travels along the upper dotted curve. Now let's bring the speed up to 101% at "A" by reducing turbine load. Next we'll start to load it again; the turbine slows down to 99.4% speed at "B" before the governor responds by moving towards its weights-in limit along the lower dotted line. Vertical distance between upper and lower dotted lines measures the governor's dead band, which in turn defines sensitivity: speed change needed to produce a corrective movement in governor travel.

Speed-output Curve

Figure 7 shows how a governor varies shaft speed with load. A given spring scale A produces 100% speed at full load. As turbine is unloaded the shaft speed rises until it reaches 104% speed at no load. However, in order to keep the speed constant at all loads, the speed changer can be used to put additional spring force on the lever controlling the primaryrelay.

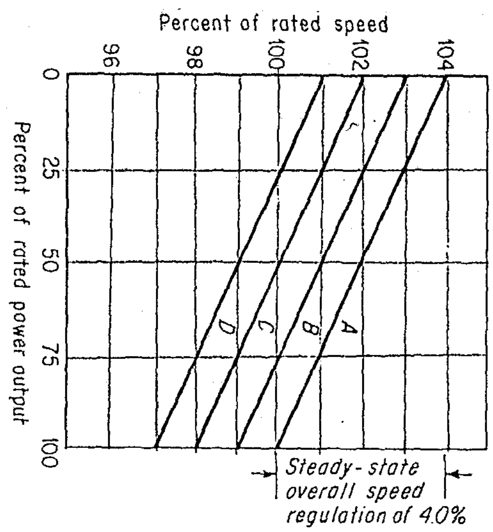


Fig. 7 Speed-power curves show how a speed changer shifts droop characteristics.

Suppose we have a 100% load at 100% speed and bring load down to 75% in figure 7. Speed rises to 101%. We would reduce this to 100% by loosening the speed-changer spring thus changing the the governor characteristic to that portrayed by line "B". Figure 7 shows characteristics for speeder spring adjusted to hold 100% speed at 25,50,75 and 100% loads.

Speed Regulation for a Single Unit

The above discussion has been about a single turbine-generator which is not operating in parallel with any other turbine-generator. This discussion is continued for a single unit below. In figure 8, line "A" is the speed characteristic of a single unit.

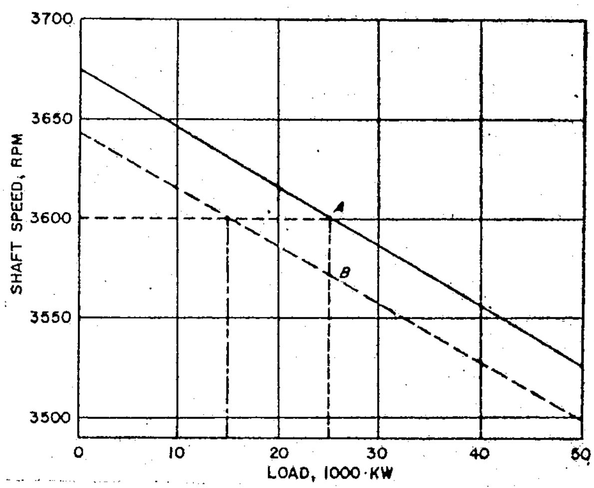


Fig. 8 Speed-regulating characteristic of turbine governor.

It shows that when the unit carries no load, it runs at a speed of 3674 rpm., but as it is loaded, its speed drops until it is only about 3526 rpm. at 50,000 KW (full load). The line here is straight to show the general trend. An actual unit will have a line that is wavy and bends, but the general trend will be as shown here.

Now for a 60 cycle system the shaft speed should be kept exactly at 3600 rpm. Time "A" shows that only at at 25,00 KW load does the speed stand at that level. How can the speed be kept constant at all the other loads? That's where the speed changer comes into play. As we said before by tightening or loosening the spring the speed characteristic can be raised or lowered. Thus by properly adjusting the speed changer the unit speed characteristic can be lowered to the line "B". Here the speed is 3600 rpm. at a load of 15,000 KW. So while the governing system will automatically open and close steam valves to accommodate load changes, supplementary adjustments have to be made to the speed changer either manually or by separate automatic means to keep the speed exactly constant.

With changing loads the speed cannot be kept exactly constant; it keeps swinging above and below the desired level. But over a period of time the average speed will be very close to the standard needed.

Speed Regulation for Units Operating in Parallel.

When generators are connected in parallel the load division between generators depends on the governor droop characteristics of the turbines that drive the generators. Paralleled synchronous generators run at the same speed, just as if they were connected mechanically.

The following example will illustrate how two generators with different speed-regulating characteristics (droop) behave when operating in parallel. Figure 9 shows the speed characteristic of a 100,000 kw unit "A" and a 50,000 kw unit "B". When carrying a combined load of 80,000 kw both units run at exactly 3600 rpm., with "B" carrying 30,000 kw and A, 50,000 kw. Now suppose the load

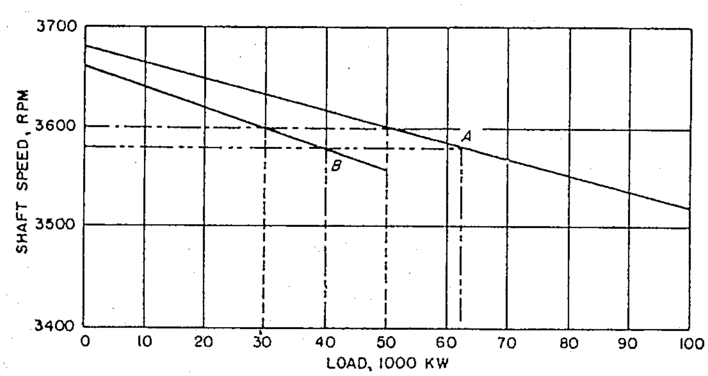


Fig. 9 Parallel operation of two turbines with different speed-regulating characteristics.

rises from 80,000 to 102,000 kw. What happens? Being tied together electrically, both units must run at the same speed. So they divide the total load so that each has the same speed and each satisfies its own governor speed characteristic. This means that both units slow down to 3,580 rpm, with B carrying 40,000 kw and A, 62,000 kw. To bring the system speed back to 3600 rpm, the speed changers on both units must be adjusted to raise their speed characteristics an equal amount. In general, the unit with least droop will always take the largest share of the total load.

The speed changers can be used to redivide the total load of 80,000 kw between units A and B by simultaneously lowering B's and raising A's characteristics. Thus B will carry less load and A more while the system speed will stay constant at 3600 rpm. Reversing the changes in speed-changer settings will make B take more of the load and A less.

When we are considering a transmission grid, to which numerous generators are synchronized in parallel, then speed changes for changes in load are extremely small, but the fact that the governor of each unit tries to conform to this falling characteristic while the unit is connected to the grid enables the power output to be controlled, using the speeder gear. The speed droop for most turbine generators can be adjustable from 2 to 6%.

Sample Problem

A system contains two units: Unit A is 40,000 kw. and has a 3% speed droop, and unit B is 50,000 kw. and has a speed droop of 4%. Both units are electrically connected and are generating 50,000 kw. total (30,000 kw for Unit A) at 3600 rpm 60 cycles. If the load reduces to 45,000 kw what will be the system frequency, and what will be the load on each unit if no adjustment is made on the governor speed changer?

Solution: Let "a" equal the decrease in load for unit "A" and "b" equal the decrease in load for unit "B". The percentage decrease in speed for each unit will be the same and will be:

$$\% \text{ increase in speed for A} = \frac{a}{40,000} \times 3\%$$

$$\text{and } \% \text{ increase in speed for B} = \frac{b}{50,000} \times 4\%$$

$$\text{Thus: } \frac{3a}{40,000} = \frac{4b}{50,000} \quad - \quad - \quad - \qquad (1)$$

Load reduction = 50,000 - 45,000 = 5,000 kw

$$\text{and: } a + b = 5,000 \quad - \quad - \quad - \quad - \quad - \qquad (2)$$

Solving equations (1) and (2) simultaneously produces:

$a = 2,580$ and $b = 2,420$

or the loads for each unit are:

Unit "A" = 30,000 - 2,580 = 27,420 kw
Unit "B" = 20,000 - 2,420 = 17,580

The speed increase will be (2,580/40,000) x 3 = 0.1935%
or the speed will be 3600 x 1.001935 = 3607 rpm. and the frequency will be: 60 x 1.001935 = 60.116 cycles.

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NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-6 - Governing Theory

A - Assignment

1. Discuss how the first stage in a turbine is affected when load is changed from full load to low load.

2. What is the reason for using a by-pass around the first stage for certain turbines? Name two disadvantages of such a by-pass.

3. Define speed droop.

4. Calculate the % speed droop for the governor of a turbine-generator when no-load speed is 3674 rpm and full load speed is 3526 rpm.

5. Define governor sensitivity.

6. A system contains two units: Unit A is 80,000 kw and has a 2% speed droop and unit B is 40,000 kw and has a speed droop of 4%. Both units are electrically connected and are generating 60,000 kw (40,000 kw for unit A) at 3600 rpm, 60 cycles. If the load reduces to 40,000 kw, what will be the system frequency, and what will be the load on each unit, if no adjustment is made on the governor speed changer?

NUCLEAR ELECTRIC G.S. TECHNICAL TRAINING COURSE

3 - Equipment & System Principles - T.T.1

4 - Turbine, Generator & Auxiliaries

-7 - References

The following texts have been used as references for writing this course and for further information these texts may be consulted.

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by Central Electricity Generating Board.

(2) Electric Generation -- Steam Stations
by Skrotzki; McGraw Hill.

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by Philip J. Potter; The Ronald Press Co.

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