NUCLEAR TRAINING CENTRE

COURSE 134

NUCLEAR TRAINING COURSE

COURSE 134

1 - Level
3 - Equipment & System Principles
4 - TURBINE, GENERATOR & AUXILIARIES

Index

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Turbine, Generator & Auxiliaries - Course 134

OBJECTIVES

At the end of this course you will be able to:

Courses 434, 334 and 234

1. Meet the objectives for the Courses 434, 334 and 234.

134.00-1 Turbine Theory

1. State a working definition of:

    (a) entropy
    (b) enthalpy
    (c) percent moisture
    (d) quality.

2. Sketch and label a Mollier Diagram showing:

    (a) the saturation line
    (b) constant pressure lines
    (c) constant temperature lines
    (d) constant percent moisture lines
    (e) constant degree of superheat lines.

3. On a sketch of a Mollier Diagram, plot the condition line for the steam system in your plant showing:

    (a) outlet of steam generator
    (b) inlet to HP turbine
    (c) outlet of HP turbine
    (d) inlet to moisture separator
    (e) outlet of moisture separator
    (f) inlet to reheater
    (g) outlet of reheater
    (h) inlet to LP turbines
    (i) outlet of LP turbines.

    (Personnel not at a generating station will use Pickering NGS as it is typical of large units.)

4. Explain what is meant by Rankine cycle and Carnot cycle.

5. Calculate Carnot Cycle Efficiency and explain it's significance.

6. Explain the advantages of superheated steam and why superheated cannot be produced in our nuclear steam generators.

7. Explain using an enthalpy-entropy diagram the extraction of useful energy from the steam passing through a turbine stage including:

   (a) initial temperature, pressure and enthalpy
   (b) useful energy extracted
   (c) loss of entropy
   (d) frictional reheat
   (e) exhaust pressure
   (f) actual exhaust enthalpy
   (g) isentropic exhaust enthalpy.

8. Define and explain the significance of:

   (a) stage efficiency
   (b) expansion efficiency
   (c) diagram efficiency
   (d) fixed blade leakage factor
   (e) moving blade leakage factor
   (f) dryness factor.

9. State and explain the factors affecting stage efficiency including:

   (a) expansion efficiency
   (b) diagram efficiency
   (c) fixed blade leakage factor
   (d) moving blade leakage factor
   (e) steam moisture percentage.

10. Explain the significance of carryover from a turbine stage and the significance of carryover from the final turbine stage (exhaust loss).

11. Draw a typical condition line for a multi-stage turbine and indicate and explain:

   (a) initial pressure, temperature and enthalpy
   (b) stage pressures
   (c) pressure drop across throttle valve
   (d) isentropic enthalpy drop for each stage
   (e) actual enthalpy drop for each stage
   (f) exhaust pressure
   (g) exhaust loss.

12. Explain the following:

   (a) Curtiss Stage
   (b) Rateau Stage
   (c) Reaction Stage
   (d) Impulse Stage.

13. Explain the factors influencing the choice of turbine blading including:

    (a) maximum diagram efficiency
    (b) enthalpy drop per stage
    (c) velocity ratio
    (d) steam pressure drop across the stage
    (e) axial thrust
    (f) moisture effects.

14. Explain what is meant by "nozzle governing" and "throttle governing and the advantages and disadvantages of each.

15. Explain how each of the following affects turbine efficiency:

    (a) superheating
    (b) moisture
    (c) moisture separator
    (d) feedheating
    (e) pressure drop in piping and valves.

<u>134.00-2 Turbine Operational Performance</u>

1. Define:

    (a) Station Heat Rate
    (b) Turbine Heat Rate
    (c) Derating.

2. Explain why station heat rate and turbine heat rate are not equal.

3. Explain the effects of each of the following on turbine heat rate:

    (a) condenser vacuum
    (b) moisture in steam passing through a turbine
    (c) pressure drop through inlet valves
    (d) boiler pressure
    (e) final feedwater temperature
    (f) blade tip leakage
    (g) air inleakage to condenser
    (h) faulty gland seals or gland seal steam operation
    (i) faulty air extraction system operation.

4. Given a design heat balance, compute a Design Turbine Heat Rate for your station.

5. Explain which plant components, operating parameters and flow rates have a major effect on heat rate.

6. Develop a systematic approach to improving a degraded heat rate.

7. Discuss the factors which could cause derating of a turbine-generator unit.

8. List the major factors which could cause a decrease in condenser vacuum and explain how you would differentiate between them.

9. List the major factors which could decrease the efficiency of the feedheating system and how you would differentiate between them.

134.00-3 Turbine Operational Problems

1. Discuss the factors affecting the severity of the following operational problems, the possible consequences and the design and operational considerations which minimize their frequency or effect:

   (a) overspeed
   (b) motoring
   (c) low condenser vacuum
   (d) water induction
   (e) condenser tube leak
   (f) blade failure
   (g) expansion bellows failure
   (h) bearing failure or deterioration
   (i) low cycle fatigue cracking.

2. Explain the advantages of using FRF as a hydraulic fluid for turbine control.

3. Explain the precautions which must be exercised with FRF and an electrical-hydraulic control system.

134.00-4 Turbine Startup

1. Describe the sequence of events on a unit startup including:

   (a) generator seal oil
   (b) turbine lubricating oil system
   (c) jacking oil pump
   (d) turning gear
   (e) position of governor steam valves, intercept valves and steam release valves
   (f) position of speeder gear
   (g) position of emergency stop valve

1. (Continued)

   (h) temperature in deaerator
   (i) condensate extraction pumps
   (j) boiler feed pumps
   (k) air extraction system
   (l) gland seal system
   (m) condenser cooling water system
   (n) stator cooling system
   (o) hydrogen cooling system
   (p) boiler stop valve position
   (q) condenser vacuum
   (r) lube oil temperature
   (s) runup to operating speed
   (t) synchronizing
   (u) loading of generator.

2. Explain the reason for each of the following in the startup sequence:

   (a) gland sealing system
   (b) air extraction system
   (c) condenser circulating water system
   (d) main lube oil system
   (e) control oil system
   (f) seal oil system
   (g) generator cooling systems
   (h) turning gear.

134.00-5 Factors Affecting Startup and Rates of Loading

1. Explain the reasons for each of the following:

   (a) COLD, WARM and HOT startup procedures
   (b) block load on synchronizing
   (c) limitation on rates of loading
   (d) HOLD and TRIP turbine supervisory parameters.

2. Discuss the factors which limit the rate at which a large steam turbine may be started up and loaded including:

   (a) steam pressure
   (b) draining steam piping and turbine
   (c) condenser vacuum
   (d) thermal stresses in casing and rotor
   (e) differential expansion between casing and rotor
   (f) lube oil temperature
   (g) generator rotor temperature
   (h) shaft eccentricity
   (i) vibration
   (j) critical speeds.

134.00-6 Reliability and Testing Requirements

1. Explain the hazards of an unterminated turbine overspeed.
2. Discuss the two factors which determine control valve unavailability: valve unavailability and tripping channel unavailability.
3. Discuss the effect of testing frequency on tripping circuit unavailability.

134.00-7 Maintenance

1. Outline a program of preparations prior to shutting down a turbine generator unit prior to overhaul.
2. Discuss items which should be examined during overhaul including:

   (a) blading
   (b) glands
   (c) diaphragms and nozzles
   (d) alignment
   (e) thrust bearing
   (f) radial bearings
   (g) casing
   (h) rotor
   (i) casing drains
   (j) evidence of presence of water
   (k) clearances between fixed and moving blades
   (l) shroud clearances
   (m) turbine flange faces.

3. Outline the basic factors to be considered in turbine maintenance.
4. Outline the factors which determine when a major turbine overhaul is scheduled.

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Turbine, Generator & Auxiliaries - Course 134

TURBINE THEORY

The subject of turbine theory lends itself to a rapid digression into a maze of esoteric scholarship which is of use only to the design engineers. On the other hand as turbine units become larger and push further toward the limit of existing knowledge, the need for operating and supervisory personnel to understand the reasons for the limitations placed on the unit becomes a part of daily existence. It seems unlikely we can ever return to the halcyon days of judging turbine unit performance by the "rumble of the engine and the smoke from the exhaust". The purpose of this lesson is to discuss the basic theory of turbine and steam cycle operation from the standpoint of understanding why turbines are constructed in a certain manner. It is hoped that this approach will give the reader an appreciation for the design features of a typical large nuclear turbine unit without the need to resort to a detailed mathematical treatment. Those who desire a more rigorous treatment are referred to the large number of existing textbooks on power plant theory and applied thermodynamics.

THERMODYNAMICS

The second law of thermodynamics tells us that it is impossible to construct a system operating in a cycle which can convert all the heat energy input from a heat source to useable work. It further defines the maximum efficiency of any cycle can by derived from the equation:

$$\eta = \frac{T_1 - T_2}{T_1} \qquad (1.1)$$

where: T_1 is the absolute temperature at which heat is supplied
T_2 is the absolute temperature at which heat is rejected

A CANDU nuclear generating station supplies heat in the steam generators at approximately 250°C and rejects heat to the circulating water in the condenser at approximately 33°C. Using equation 1.1 between these two temperatures we get:

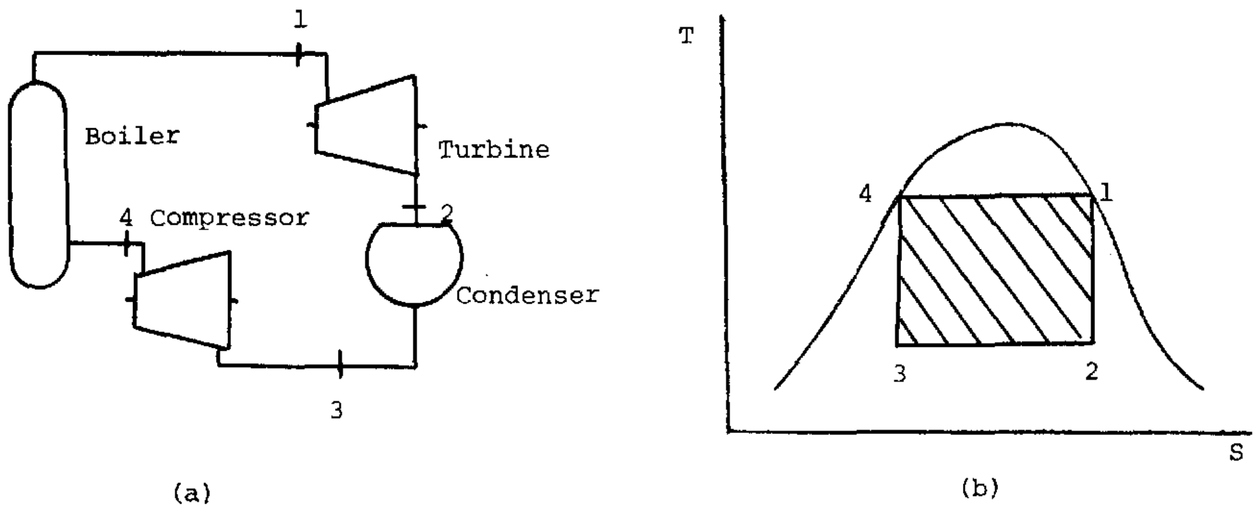
$$\eta = \frac{523°K - 306°K}{523°K}$$

$$= \frac{217°K}{523°K}$$

$$= 42\%$$

In a system operating between 250°C and 33°C a maximum of only 42% of the heat supplied can be converted to work. It is obvious that this maximum efficiency can be increased by increasing the temperature at which heat is supplied or by lowering the temperature at which heat is rejected. In a CANDU nuclear power plant, however, these temperatures cannot be varied substantially in a direction which will improve efficiency. The upper limit is imposed by material limits within the fuel elements, while the lower limit is imposed by the available temperature of condenser cooling water from the lake or river and absolutely limited by the freezing temperature of water at 0°C.

It is well to remember that this 42% represents an upper limit on the efficiency of a CANDU generating station. As long as a CANDU system is used to convert heat energy to electrical energy the cycle cannot be more efficient than 42%.



THE CARNOT CYCLE

Figure 1.1

Figure 1.1(a) shows a system which has an ideal efficiency as described by equation 1.1. This system is described as a Carnot Cycle. Heat is added in the boiler at 250°C, work is extracted in the turbine, heat is rejected in the condenser at 33°C until about 80% of the steam is condensed and then the wet steam is compressed to saturated water at the pressure in the boiler. While this cycle would have a theoretical efficiency equal to the maximum of 42% it has several practical draw backs:

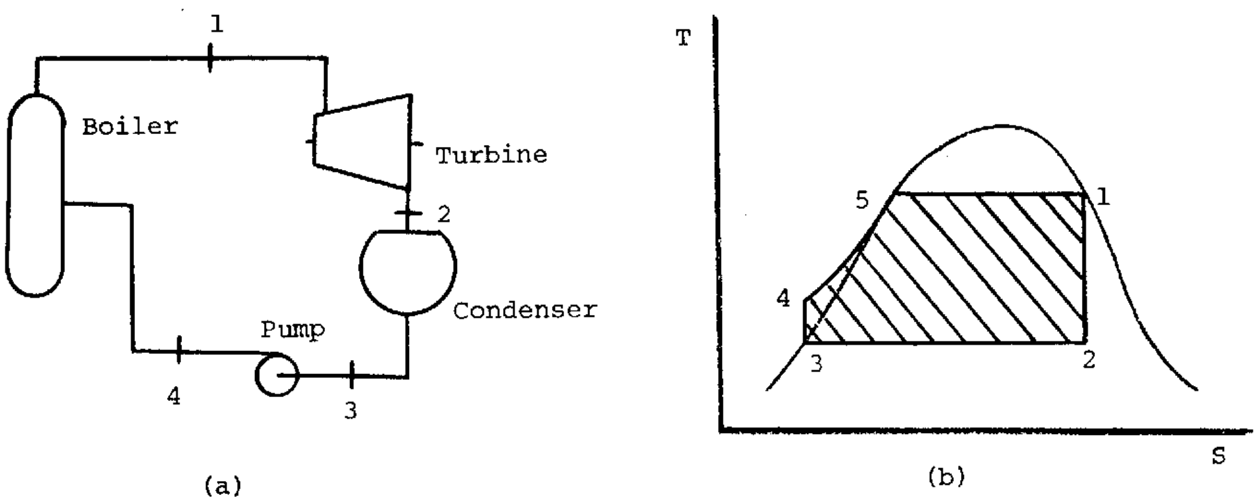
(a) it is difficult to stop the condensing process short of complete condensation to water,

(b) the compressor must handle a low quality wet steam which tends to separate into its component phases forcing the compressor to deal with a non-homogeneous mixture.

(c) the volume of fluid handled by the compressor is high and the compressor must be comparable in size and cost to the turbine,

(d) because the compressor consumes a large percentage of the turbine output power, this cycle is very sensitive to irreversibilities. While this cycle is ideally 42% efficient, if the compressor and turbine are only 80% efficient, the cycle efficiency drops to about 28%. If the compressor and turbine efficiency drop to about 50%, the cycle becomes a net consumer of energy.

Most of the practical problems of the Carnot Cycle can be avoided by allowing the steam to completely condense and then compressing the liquid to boiler pressure with a small feed pump. The resulting cycle, shown in Figure 1.2, is known as a Rankine Cycle.



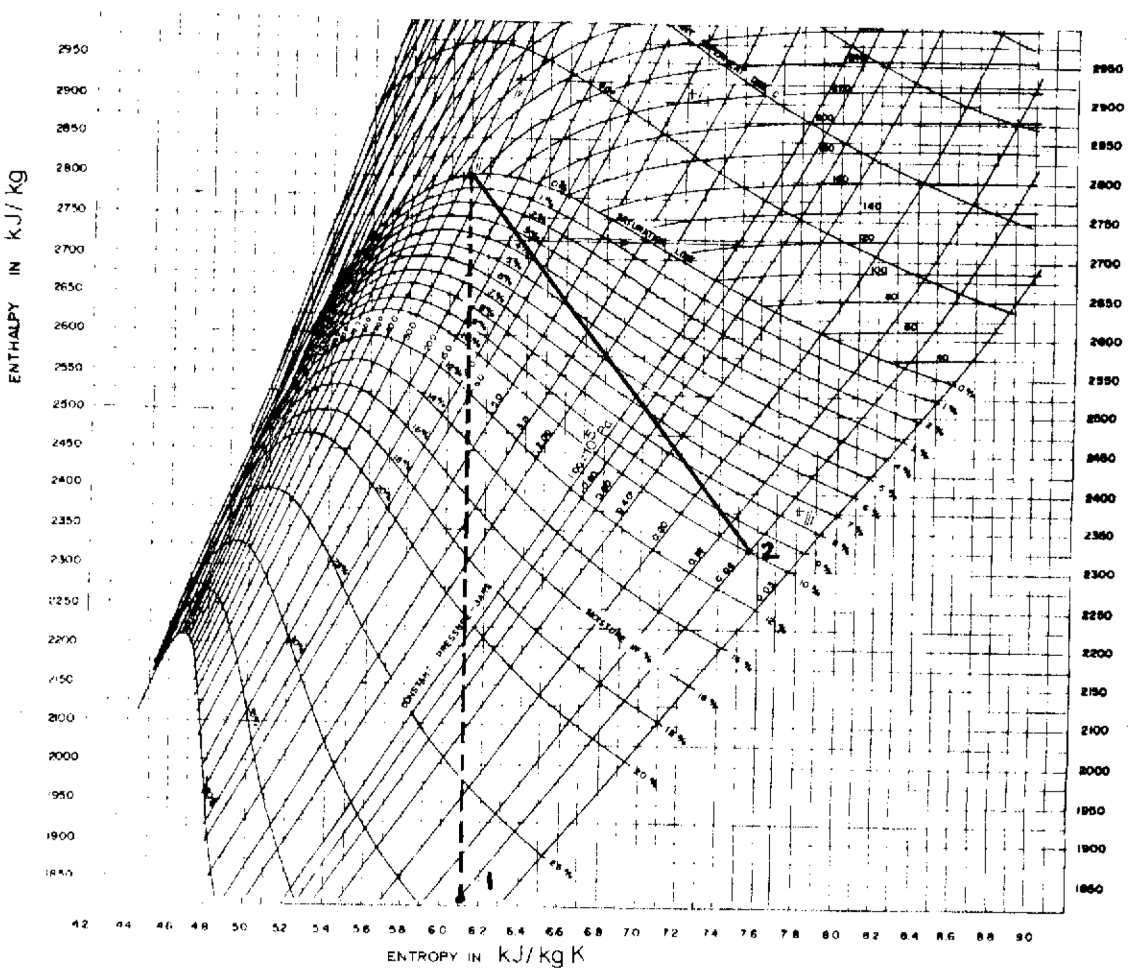
THE RANKINE CYCLE

Figure 1.2

It is evident without calculation that the efficiency of this cycle will be less than that of the Carnot Cycle operating between the same temperatures, because all the heat supplied is not transferred at the upper temperature. Some heat is added while the temperature of the liquid is increasing from T_4 to T_5. By comparing the work output per kilogram of steam (the shaded area of the T-s diagram), it is apparent that the steam consumption is less in the Rankine Cycle. In addition since the power requirements of the pump is a small percentage of the turbine output, the effect of irreversibilities is significantly less than with the Carnot Cycle. While the Rankine Cycle has a lower ideal efficiency than the Carnot Cycle, the practically attainable efficiency is not much different and the plant is certainly smaller and less costly.

In the Rankine Cycle shown in Figure 1.2, the steam exhausting from the turbine has a moisture content of 28%. This is much too high for any economic turbine. The water droplets which are carried in the wet steam cannot move as rapidly as the steam and as the water passes through the moving blades, the back of the blades continually strike the slower moving moving droplets. This exerts a retarding effect on the moving blades which decreases efficiency. On the order of 1% of <u>turbine stage efficiency</u> is lost for each 1% average moisture in the stage. In addition the erosion effect of the water droplets for moisture percents much above 13-14% would shorten the blade life to an economically unattractive point.

The effect this has on the design efficiency of a turbine unit can be seen on the Mollier diagram in Figure 1.3.



SINGLE HP TURBINE

Figure 1.3

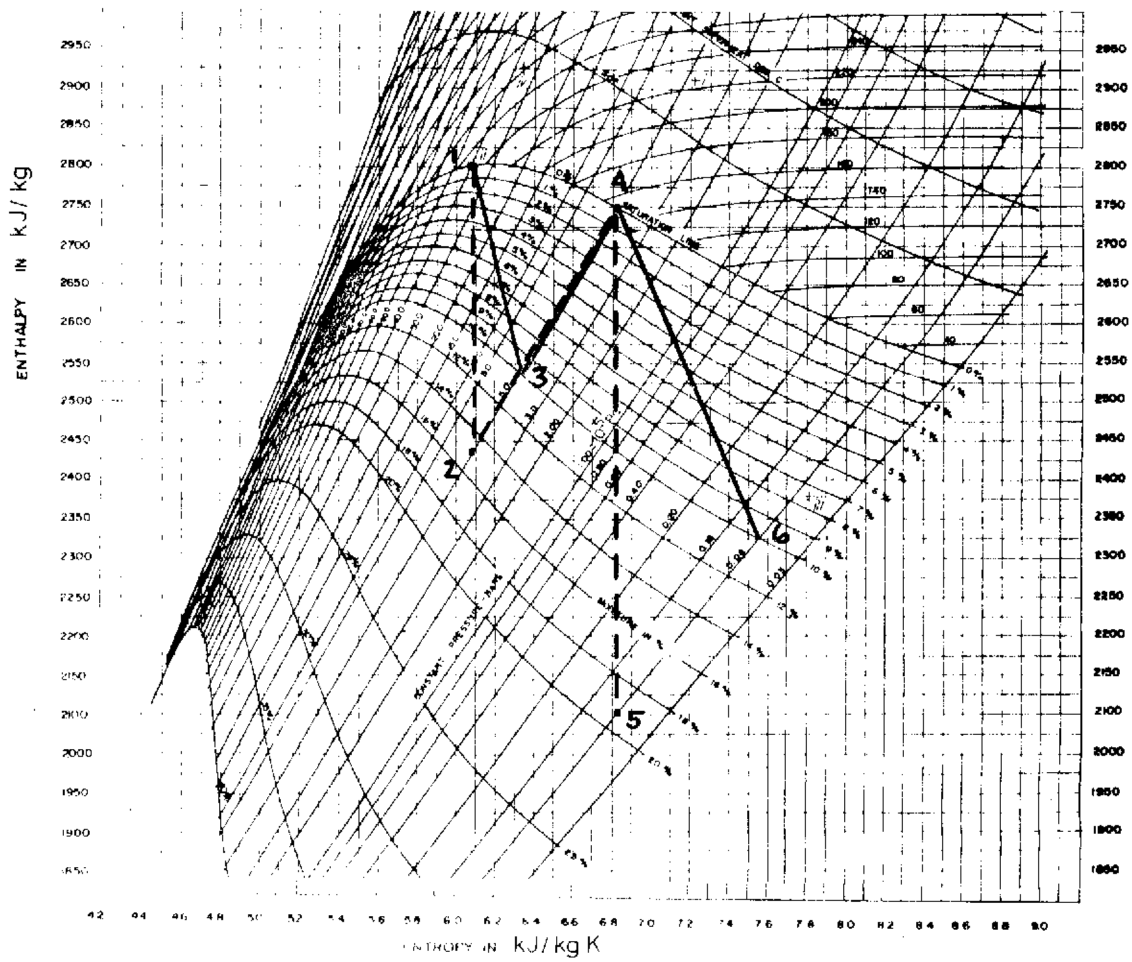
If the exhaust moisture must be limited to around 10%, as it is on most large turbine units, then the turbine must exhaust at point 2 (10% moisture, 33°C) rather than at point 1. That is, we cannot design this turbine to be isentropic because we cannot handle the increased moisture of a completely reversible expension.

To decrease the exhaust moisture to acceptible values we must accept a considerable increase in entropy which implies a loss of available energy and a decrease in efficiency. In this case the turbine efficiency must be limited to only about 50% of the ideal efficiency because we cannot cope with the moisture content a higher efficiency would imply. While the turbine unit shown in Figure 1.2 has an ideal efficiency of 35%, the problem of exhaust moisture alone limits the practical efficiency to about 17%.

MOISTURE SEPARATION

To improve the efficiency of the cycle above that possible with a single turbine, it is common to remove the steam from the turbine at 10% moisture, separate the water from the steam, and then utilize the steam in a second turbine. While the exact pressure to remove the steam for moisture separation depends on a number of factors, plant efficiency is generally optimized at a pressure in the 500-700 KPa (g) range. Figure 1.4 shows the effect of such moisture separation.

The dashed line (1245) shows the ideal isentropic process. While this process still results in a moisture content above the maximum acceptible, the real turbine process (1346) is much closer to the ideal than was possible in the single turbine. The isentropic efficiency of this process is slightly over 35%, a small improvement over the isentropic efficiency without moisture separation. However, the realistically allowable process is almost 25% efficient which is a considerable improvement over the 17% for the process without moisture separation.



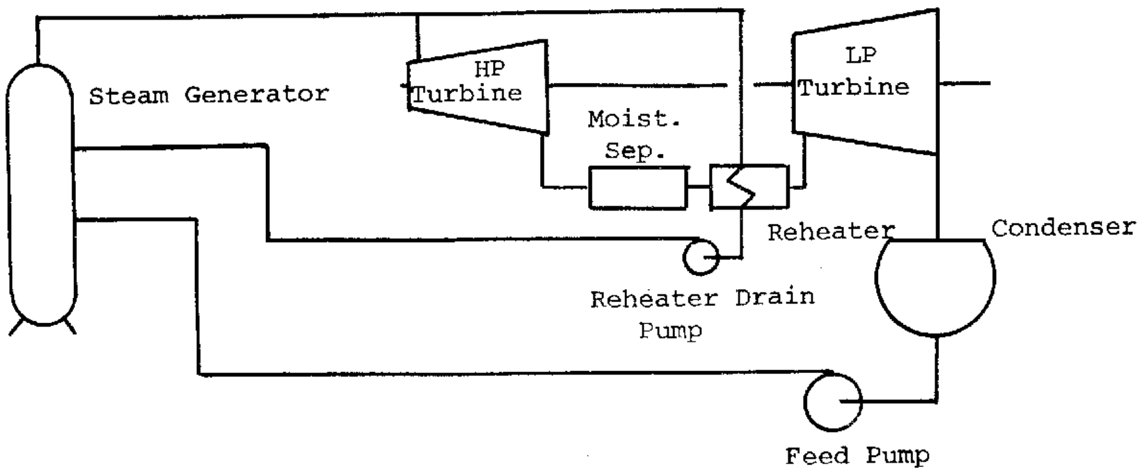
MOISTURE SEPARATOR

Figure 1.4

REHEATING

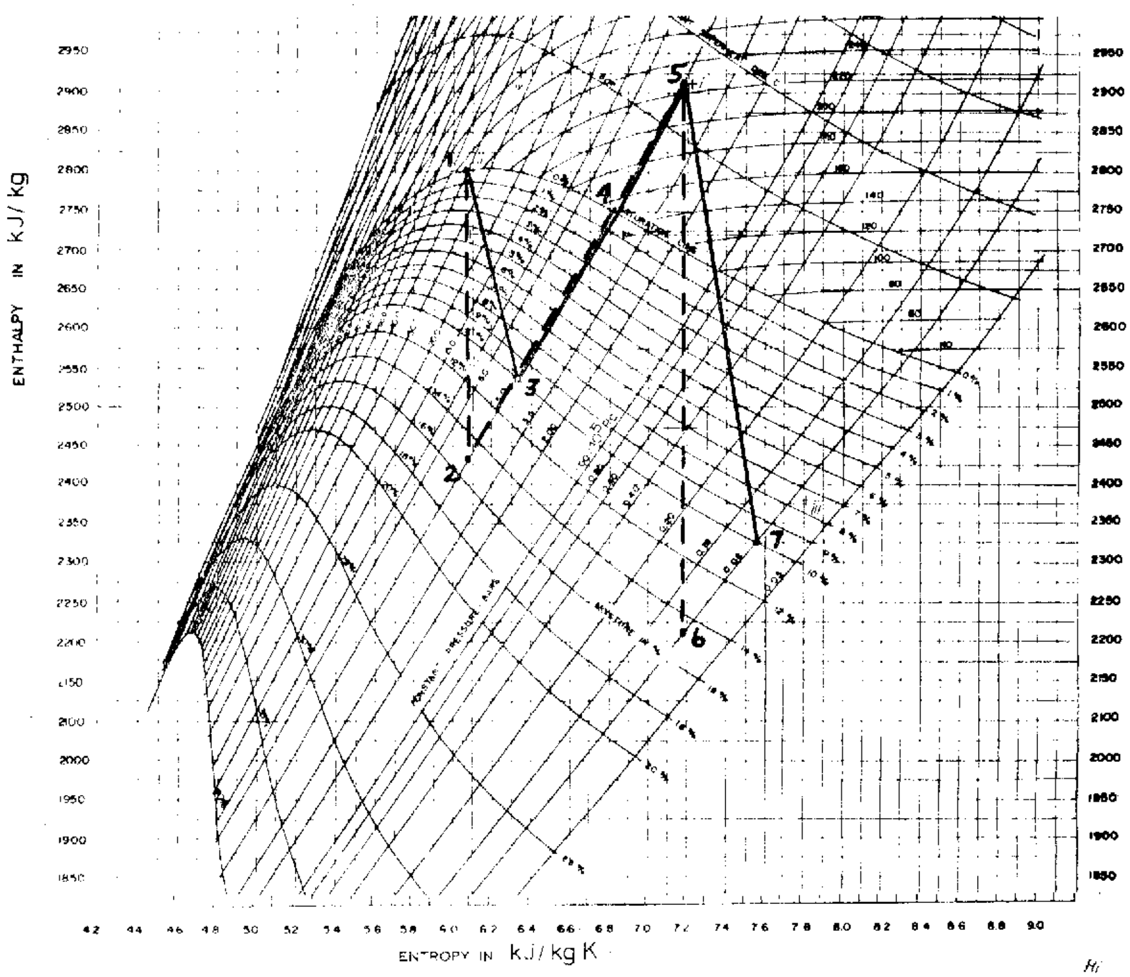
Reheating is often used to further improve the cycle efficiency. Figure 1.5 shows a typical nuclear turbine system with reheating and moisture separation.

A percentage of the steam produced in the boiler is lead to a reheater where it is used to superheat the steam exhausting from the moisture separator.



SIMPLIFIED NUCLEAR STEAM CYCLE

Figure 1.5



REHEATER

Figure 1.6

The dashed line (12456) in Figure 1.6 shows the ideal isentropic process which has an efficiency of 38% which is still not significantly above the 35% attainable by a single turbine without moisture separation or reheating. However, the realistically allowable process (13457) is almost 30% efficient. It should be noted how much more closely the allowable condition process follows the ideal process in Figure 1.6 than occurred in Figure 1.3.

In addition it should be noted that the average moisture content in the low pressure turbine with moisture separation alone is about 5% when the exhaust moisture is held to 10%. However, with reheating the average moisture content is no more than 1% with the same 10% limit on exhaust moisture. Not only does this decrease erosion in the low pressure turbine, but the decrease in efficiency due to water droplet impingement on the moving blades is substantially reduced.

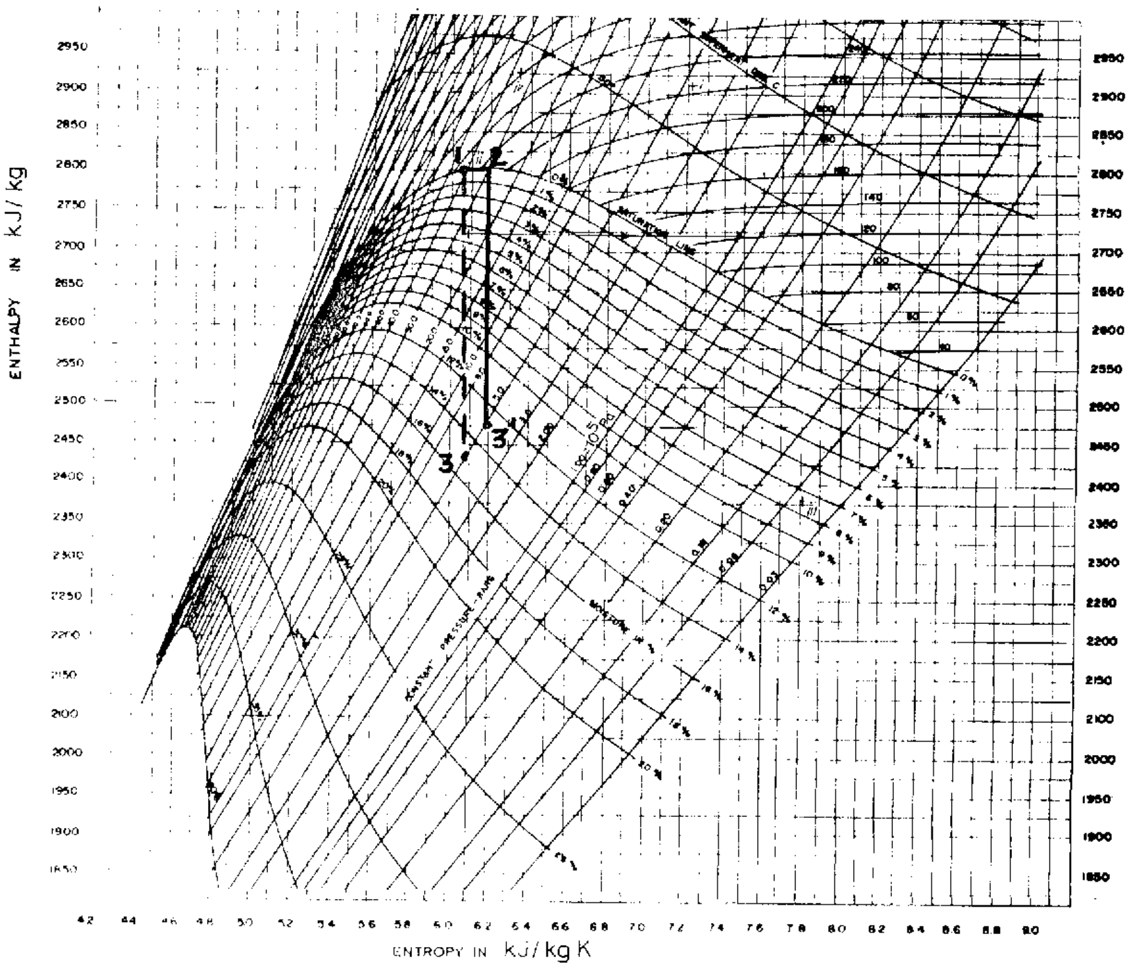
SUPERHEATING

Almost without exception, conventionally fuelled power plants superheat steam before sending it to the high pressure turbine. Not only does this give the high pressure turbine the same benefits that reheating gives to the low pressure turbine but in addition the raising of steam temperature above saturation temperature increases the average temperature at which heat is extracted from the heat source and thus increases the Carnot efficiency. Unfortunately, we are unable to add any appreciable amount of superheat to the 4000 KPa(g) saturated steam produced in our steam generators. The same metallurgical limitations which restrict steam generator temperature to 250°C, restrict the temperature in a hypothetical superheater to about 250°C; that is, no superheat.

While a CANDU reactor could produce superheated steam at a pressure lower than 4000 KPa(g) this is unattractive not only from the standpoint of a lower saturation temperature in the boiler and, therefore, a lower Carnot efficiency but also from the lower steam density which would require larger piping and components for the same power output.

PRESSURE DROPS IN PIPING AND VALVES

The pressure drops which occur as steam passes down the main steam piping and through valves can be considered a throttling process. Throttling is a constant enthalpy process; that is, heat content of the steam does not change, even though the pressure and temperature decrease. A throttling process can be shown as a horizontal line (constant enthalpy) on a Mollier diagram. These pressure drops have to be held to a minimum because the entropy of the steam increases and, therefore, the availability of energy decreases.



INLET VALVE PRESSURE DROP

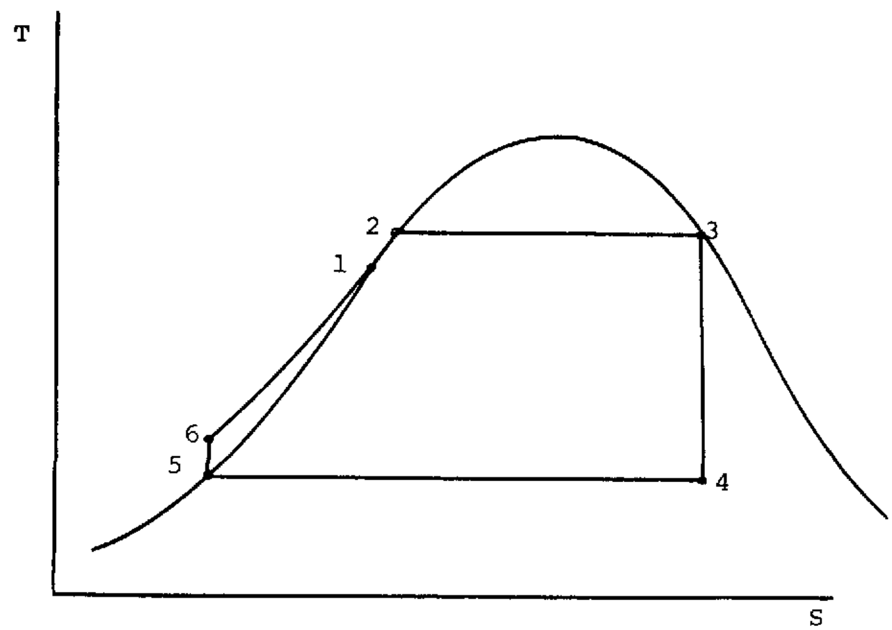
Figure 1.7

Figure 1.7 shows the pressure drop across the inlet valves to the high pressure turbine. If the exhaust pressure does not change, then the pressure drop results in less available energy. In this case the 25% pressure drop (12) results in only 87% (13'/13) as much energy available in the high pressure turbine. Typically the pressure drop between the steam generators and inlet to the high pressure is held to no more than 5%. The effect of a pressure drop across the moisture separator, reheater and valves between the HP and LP turbines is similar and this pressure drop is likewise held to a maximum of about 5%.

FEEDHEATING

The theoretical aspects of regenerative feedheating is fully discussed in the 225 Heat and Thermodynamics course and does not require a complete rediscussion in this lesson.

The advantages of extracting steam from the high and low pressure turbine for use in heating feedwater is fairly obvious. With turbine exhaust wetness limited to 10%, only about 10% of the latent heat of vaporization can be utilized by passing the steam through the remaining stages of the turbine. However, if the steam is extracted from the turbine and used to heat feedwater all of the latent heat of vaporization can be utilized. Of course, we are in a sense robbing Peter (turbine output) to pay Paul (heating feedwater) so there is a point of diminishing returns but the initial effect is quite pronounced in favor of increasing cycle efficiency. In addition the extraction of steam from the low pressure turbine helps to reduce the vast volumes of steam which the latter stages of the low pressure turbine must handle. The effect of a higher final feedwater temperature can be seen on the T-s diagram in Figure 1.8.



EFFECT OF FEEDHEATING

Figure 1.8

Feedheating has raised the feedwater temperature from T_6 to T_1 so the boiler must only increase the temperature from T_1 to T_2 before steam production begins. This raises the average temperature at which the boiler adds heat energy and therefore increases the Carnot efficiency. Feedwater typically enters a nuclear steam generated heated to near 175°C. This results in the steam generator adding heat energy at an average temperature 35°C hotter than without feedheating.

It is worth noting that in plants such as Bruce N.G.S. where the preheater is located external to the boiler, it is the temperature of feedwater entering the preheater which effects efficiency. Thermodynamically the preheater is not a feedheater but rather an extension of the steam generator.

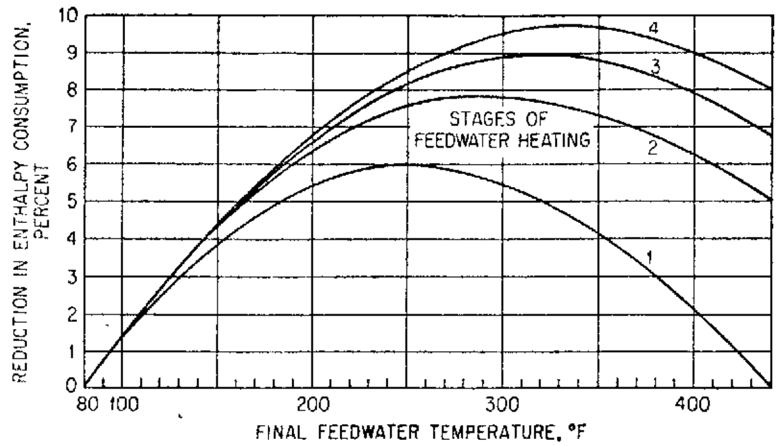


Figure 1.9

Figure 1.9 shows the typical effect of feedheating on cycle efficiency. Since the temperature difference between the extraction steam entering a feedheater and the feedwater leaving a feedheater is typically 5°C or less, the final feedwater temperature closely approximates the temperature of the highest temperature extraction steam.

Examination of the curves in Figure 1.9 reveals the following:

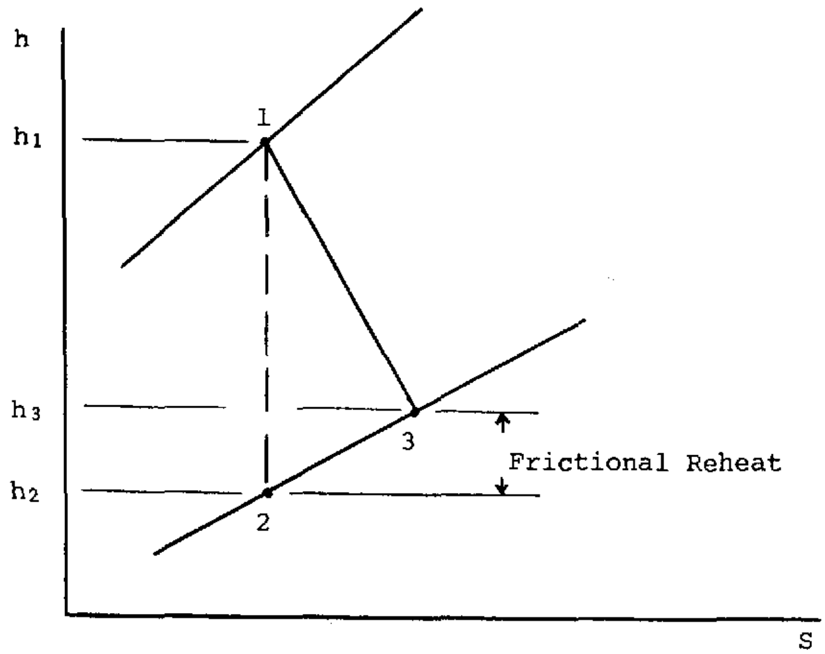
(a) as the number of feedheaters increases, the optimum temperature and, therefore, pressure of the extraction steam to the last feedheater increases,

(b) there is little advantage to be gained in going beyond six to eight stages of feedheating, and

(c) since the curves are relatively flat on top the extraction steam pressures can vary substantially from the optimum without much effect on efficiency.

TURBINE STAGE EFFICIENCY

Thus far we have discussed the effects of various cycle components on ideal efficiency whether isentropic or that realistically imposed by exhaust moisture. Turbines cannot always be designed to work as we would like them to work and we must accept efficiencies lower than theoretically possible.



STAGE EFFICIENCY

Figure 1.10

Figure 1.10 shows the condition line for a turbine stage operating on wet steam between pressure P_1 and P_2. The dashed line (12) represents the ideal isentropic path between these two pressures and $h_1 - h_2$ represents the ideal work done by a kilogram of steam passing through the stage. In a real turbine we would find this much work was not done and the actual path through the stage (13) would result in less heat energy being extracted from the steam. The ratio

$$\frac{h_1 - h_3}{h_1 - h_2}$$

is known as stage efficiency and for a well designed stage is typically between 75% and 90% depending on the type of stage, the enthalpy drop across the stage and the moisture content of the steam.

There are a number of reasons why stage efficiency is not 100%, but a significant source of inefficiency is friction between the steam and blading. This friction adds heat energy back into the steam and results in a leaving enthalpy higher than ideal. Because of the significance of this friction heating, the amount of isentropic enthalpy drop not utilized in a stage is known as frictional reheat even though friction is not the only cause of inefficiencies. Although frictional reheat results in a greater enthalpy of the steam at the outlet of the stage than one would theoretically expect, there is also an increase in entropy which represents a loss in availability of energy.

Stage efficiency is a product of five factors as described below:

$$\text{Stage Efficiency} = \left(\begin{matrix}\text{Expansion}\\\text{Efficiency}\end{matrix}\right)\left(\begin{matrix}\text{Diagram}\\\text{Efficiency}\end{matrix}\right)\left(\begin{matrix}\text{Fixed}\\\text{Blade}\\\text{Leakage}\\\text{Factor}\end{matrix}\right)\left(\begin{matrix}\text{Moving}\\\text{Blade}\\\text{Leakage}\\\text{Factor}\end{matrix}\right)\left(\begin{matrix}\text{Dryness}\\\text{Factor}\end{matrix}\right)$$

where: $\text{Expansion Efficiency} = \dfrac{\text{Steam Kinetic Energy Produced}}{\text{Steam Enthalpy Supplied}}$

$\text{Diagram Efficiency} = \dfrac{\text{Work Done On Rotor}}{\text{Steam Kinetic Energy Produced}}$

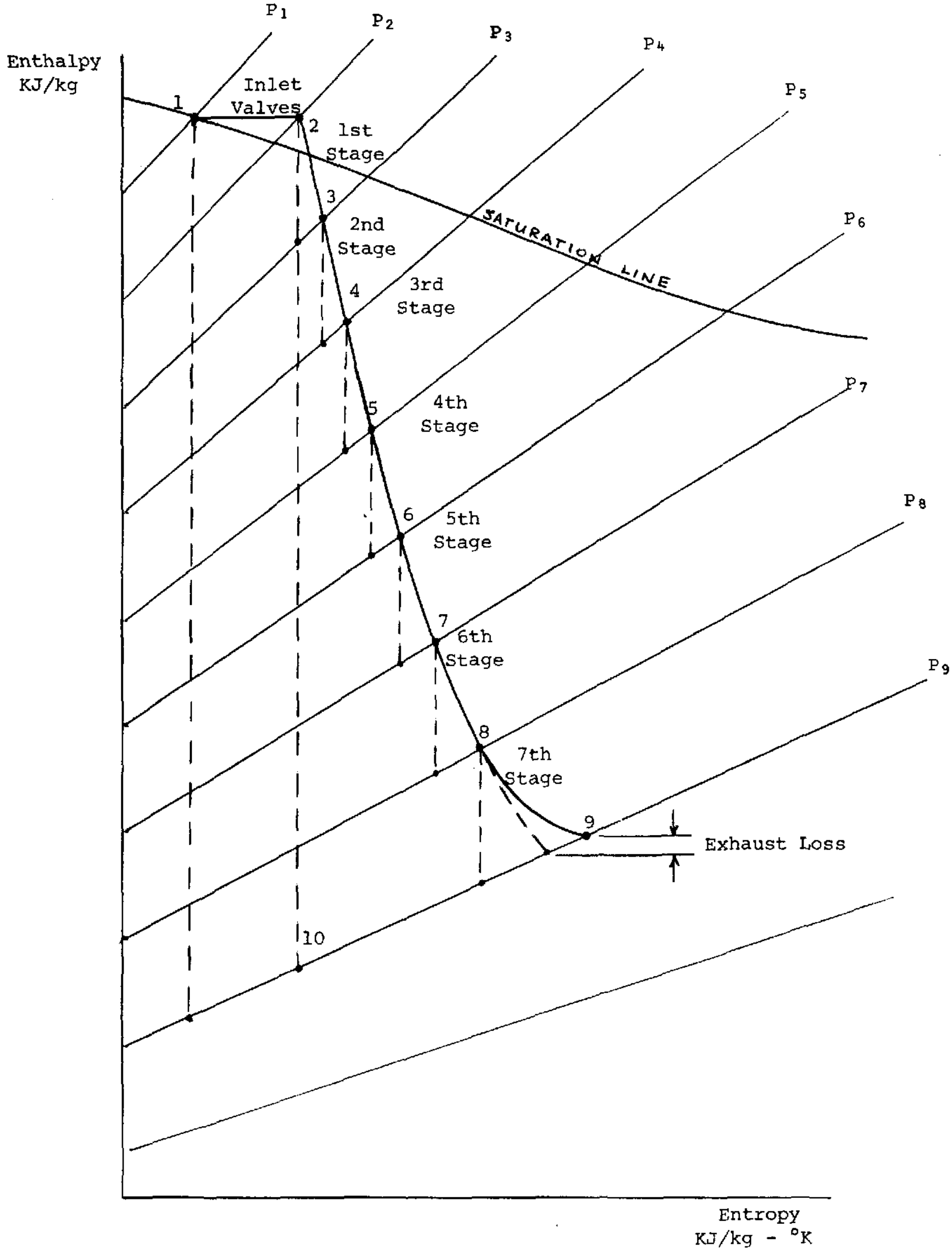
Dryness factor accounts for the decrease in efficiency due to moisture impingement on the moving blades.

Of practical significance in turbine design is the efficiency of the conversion of steam kinetic energy to work. If this diagram efficiency of the turbine is not 100%, then some steam kinetic energy is lost as steam leaves the moving blades with some velocity. This loss of kinetic energy is known as carry over. If the subsequent stages are well designed, this carry over can be partially or fully recovered; however, the carry over from the last stage represents an unrecoverable loss of energy. After the steam leaves the last stage, this kinetic energy is converted to heat and appears on the Mollier diagram as an unexpected increase in exhaust enthalpy. This leaving loss or exhaust loss as it is called must be minimized by insuring the velocity of the steam leaving the last stage is as small as possible. For this reason, the annular area of the last row of blading is made as large as economically possible.

THE TURBINE CONDITION LINE

Figure 1.11 shows a typical turbine condition line for a seven stage saturated steam turbine.

You will note the pressure drop across the inlet valves and steam strainer. At the normal operating load of the turbine, the designer attempts to achieve an equal enthalpy drop in each stage so the work produced in each stage is approximately equal. The abrupt increase in enthalpy at the turbine exhaust is the appearance of the exhaust loss as heat energy. The turbine efficiency would be expressed by $\dfrac{h_2 - h_9}{h_2 - h_{10}}$.



P1
P2
P3
P4
P5
P6
P7
P8
P9
Enthalpy
KJ/kg
1
Inlet
Valves
2
1st
Stage
3
2nd
Stage
4
3rd
Stage
5
4th
Stage
6
5th
Stage
7
6th
Stage
8
7th
Stage
9
Exhaust Loss
10
SATURATION LINE
Entropy
KJ/kg - °K

Figure 1.11

As a representative value the efficiency of the Pickering N.G.S. high pressure turbine is approximately 75% while the efficiency of the low pressure turbine is approximately 85%. If exhaust losses were eliminated completely in the LP turbine, the efficiency would be nearly 89%. The greater efficiency of the low pressure turbine is a combination of the effects of reheating and lower average moisture. Figure 1.12 shows the condition line for a typical large turbine unit with reheaters and moisture separators. Points 1 through 6 show the extraction points for feedheating.

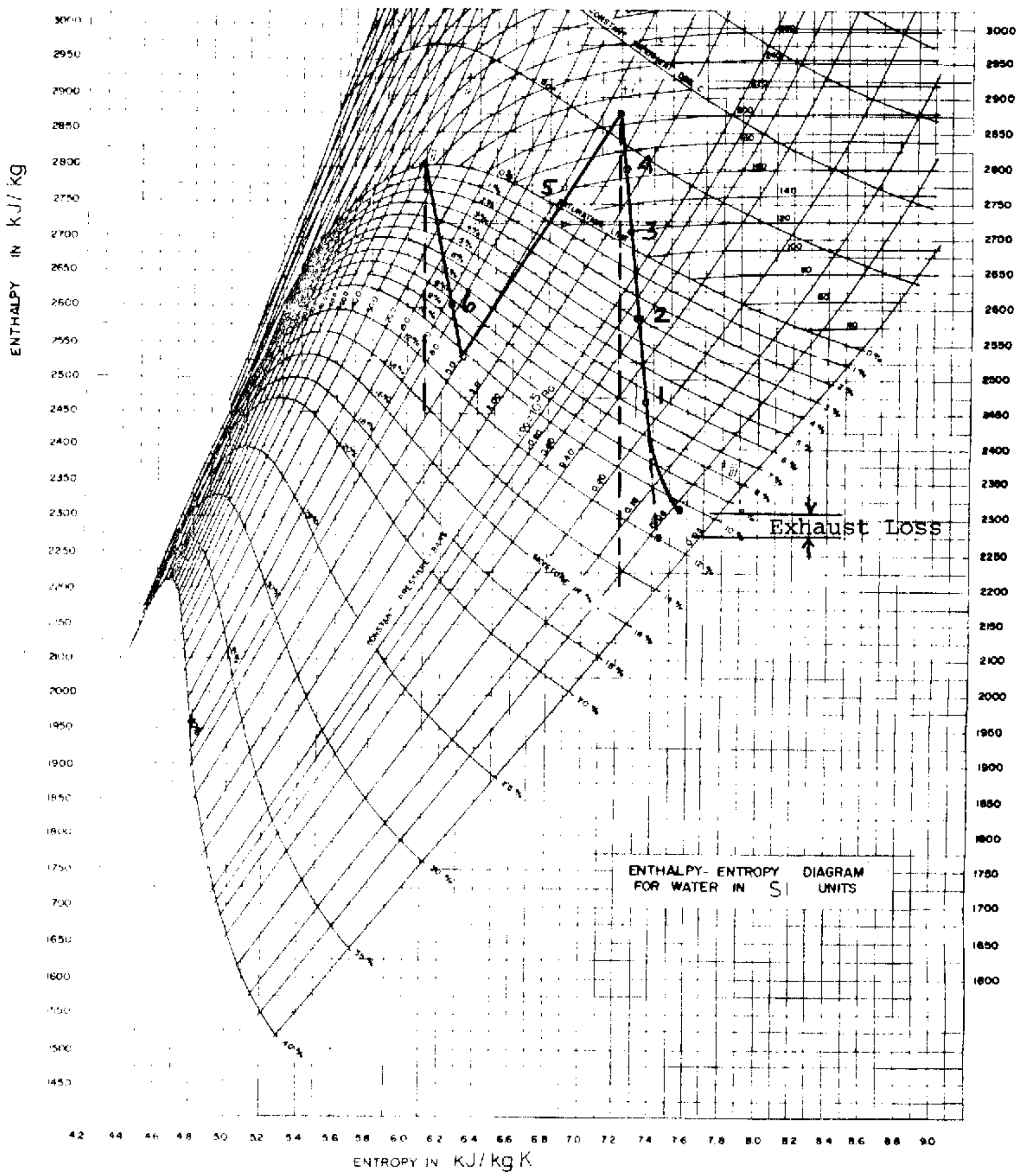


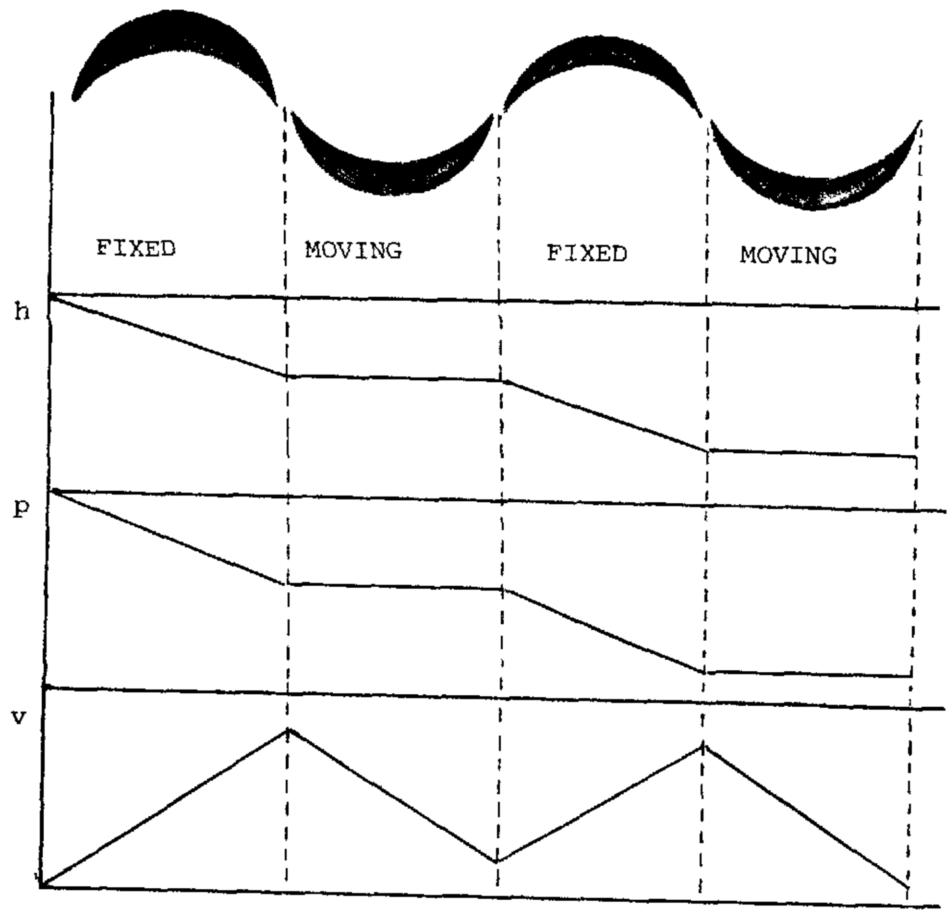
Figure 1.12

TURBINE STAGE TYPES

There are two basic types of turbine stages: the reaction stage and the impulse stage. The fundamental difference between the two types of staging is the part of the stage in which heat energy is converted to steam kinetic energy. In the impulse stage, this conversion takes place only in the fixed blades; in the reaction stage, this conversion takes place in both the fixed and moving blades.

THE IMPULSE STAGE

As the steam passes through the fixed blade nozzles of an impulse stage, the enthalpy or heat energy of the steam is reduced and the velocity is greatly increased. This high velocity steam is then directed by the fixed blades into the moving blades. The steam changes direction in the moving blades and imparts an impulse (force x time) to the moving blades. Figure 1.13 shows the pressure, velocity and enthalpy change across two impulse stages.

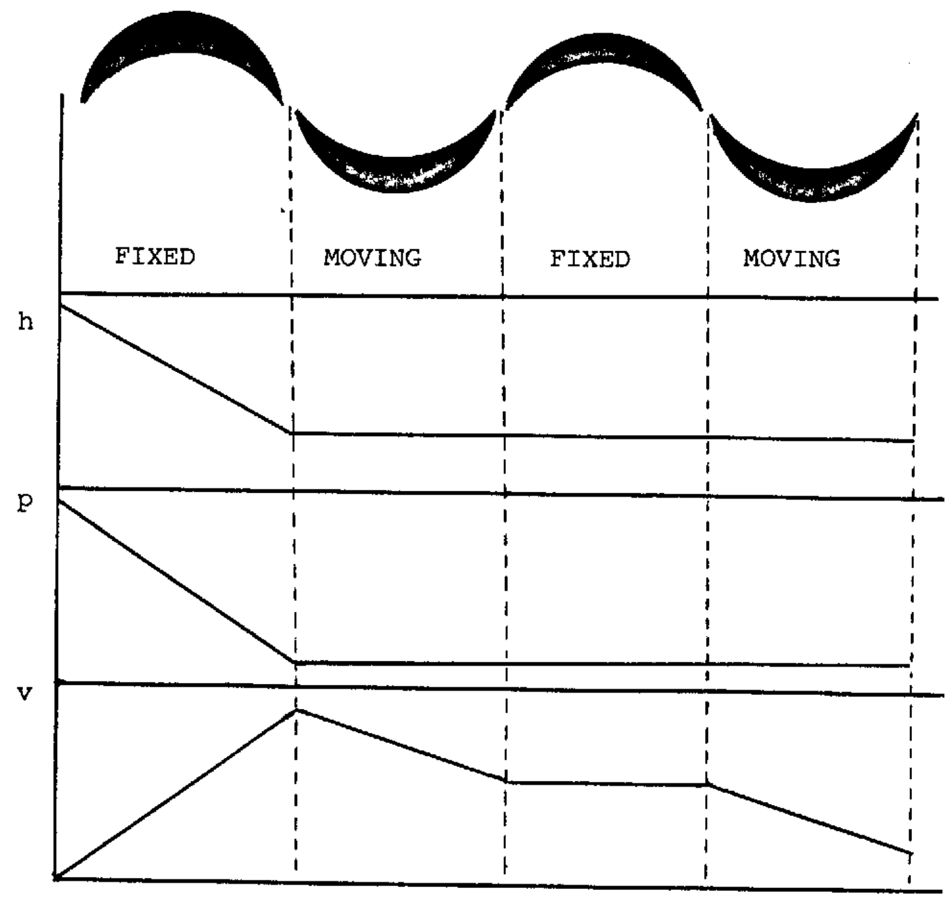


RATEAU STAGES

Figure 1.13

This type of impulse stage (fixed nozzle, moving blade) is known as a Rateau Stage. The pressure and enthalpy decrease across the nozzle as heat energy is converted to steam kinetic energy (velocity). Across the moving blades, the steam velocity decreases as kinetic energy is transferred to the moving blades. You will note the absence of a pressure drop across the moving blade. The turbine shown in Figure 1.13 would be referred to as a two stage Rateau turbine.

Figure 1.14 shows another type of impulse stage arrangement known as a Curtiss Wheel.



TWO STAGE CURTISS WHEEL

Figure 1.14

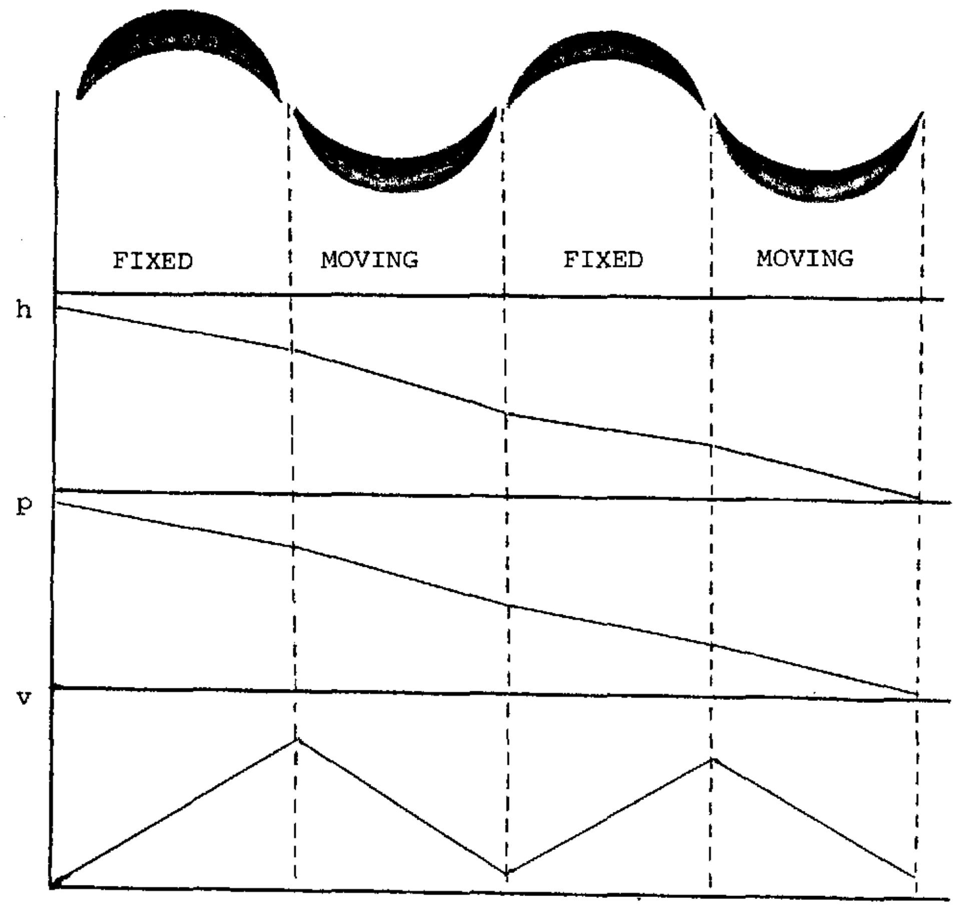
The second set of fixed blades are not nozzles and only serve to redirect the steam into the second set of moving blades. Because the second fixed blades only redirect the steam there is no change in steam velocity across these blades. The turbine shown in Figure 1.14 is a two stage turbine. These two stages are collectively called a Curtiss

Wheel. You will note that in the Curtiss Wheel, as in the Rateau stage, there is no pressure drop across the moving blades.

THE REACTION STAGE

The fixed blade nozzles in a reaction turbine convert heat energy to steam kinetic energy in the same manner as the Rateau stage. The high velocity steam imparts an impulse to the moving blades.

The reaction stage differs from the Rateau stage in that the moving blades are shaped like a nozzle so that heat energy is converted to kinetic energy in the moving as well as the fixed blades. This conversion forces the moving blades away from the expanding steam in a reaction effect similar to a rocket reacting to the escaping exhaust gasses.



REACTION STAGES

Figure 1.15

Figure 1.15 shows the pressure, enthalpy and velocity change across a two stage reaction turbine. Heat energy is converted to kinetic energy in both the fixed and moving blades. The moving blades move in response to both an impulse and a reaction effect. You will note that a pressure drop occurs across the moving blades.

The distinction between impulse and reaction stages is more clear cut in theory than in practice. Turbine stages are classified by their degree of reaction or the ratio of the enthalpy drop across the moving blades to the enthalpy drop across the entire stage. The degree of reaction may vary from 0% to 100%; zero reaction being pure impulse. It is not at all uncommon for impulse stages to have a small amount of reaction to improve their efficiency. Generally if the degree of reaction is no more than 5-10%, the stage is called an impulse stage, otherwise it is called a reaction stage.

CHOICE OF TURBINE STAGE

The decision of which type of stage to use in a turbine is never clear cut. Each type of stage has its particular advantages and disadvantages and an application in which it is the superior choice.

AXIAL THRUST

Reaction turbines have a pressure drop across the moving blades. Because of this, the force on the high pressure side of the blade wheel is greater than the counteracting force on the low pressure side. This force difference means there is a tendency of the wheel to move in the direction of decreasing pressure. In a single flow, high pressure reaction turbine, the cumulative force can be very large and the thrust bearing necessary to handle this force would be extremely large and costly. Although there are methods (for example a dummy piston) of compensating for this thrust in a single flow high pressure reaction turbine, the least complex method of handling axial thrust in a single flow turbine is to use impulse staging. Since the impulse stage has no pressure drop across the moving blades, it produces no axial thrust.

In a low pressure turbine, the pressure drop across the moving blades of a reaction turbine is much less. For a typical 50% reaction nuclear steam turbine, the pressure drop across the moving blades of the HP turbine would be 200 KPa per stage while the pressure drop across the moving blades of the LP turbine would be 25 KPa per stage. It is possible to economically construct a thrust bearing which will handle the thrust of a single flow low pressure reaction turbine. The result is that while most single flow HP turbines have impulse blading, many single flow LP turbines have reaction blading.

In large turbine units with large diameter low pressure blade wheels, even the small pressure drops across the moving blades of a low pressure reaction turbine produce a large axial thrust. In such units the low pressure turbines are typically double flow to compensate for this thrust.

EFFICIENCY AND ENTHALPY DROP

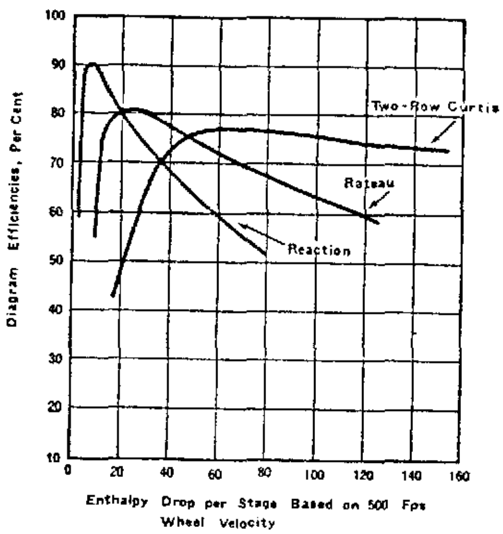


Figure 1.16

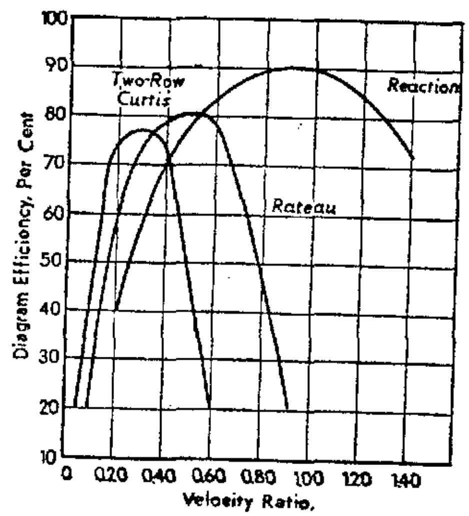


Figure 1.17

Figure 1.16 shows how the diagram efficiency for the three major types of turbine stages changes as the enthalpy drop per stage varies. If the enthalpy drop per stage is kept small the reaction turbine is attractive due to its higher maximum efficiency. Typically, the number of stages in a reaction turbine is high to keep the enthalpy drop low. In some instances it is difficult to keep the enthalpy drop across a stage within the proper range for a reaction turbine. In these cases, the Rateau stage and occasionally even the Curtiss Wheel is used to keep the efficiency up. The use of an impulse stage in the first stage of a nozzle governed high pressure turbine is widespread. In addition, under certain conditions of reheating, the enthalpy drop across the first stage of the low pressure turbine may be quite large and require an impulse stage.

VELOCITY RATIO

Velocity ratio is the ratio of blade tangential speed to steam velocity and each stage type has a different velocity ratio at which it runs most efficiently. Figure 1.17 shows the relationship between velocity ratio and diagram efficiency. As blade wheels become larger to accomodate the high volumes of steam in modern turbine units, the blade tangential velocity increases and the velocity ratio increases. As a result the reaction turbine become more attractive as the velocity ratio increases. Large turbines and particularly large low pressure turbines are commonly reaction turbines.

MOISTURE EFFECTS

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

BLADE LEAKAGE

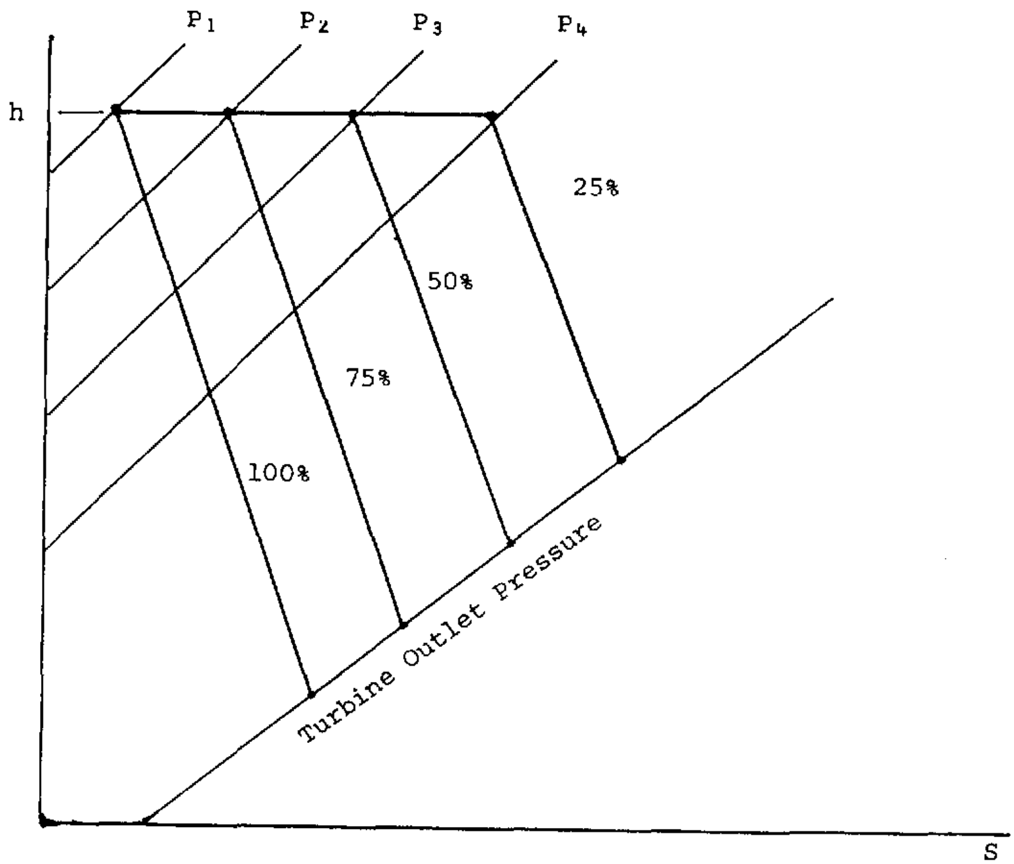
Since the reaction turbine produces a pressure drop across the moving blades, there is a tendency in the reaction turbine for the steam to crawl over the end of the moving blades. This effect can be quite pronounced in a high pressure turbine where the pressure drop per stage can be fairly high. This type of leakage is less of a problem in the impulse stage which makes its use in HP turbines attractive in minimizing the moving blade leakage factor. In HP reaction turbines, a higher blade tip leakage must be expected and usually an increased number of stages is required to keep the pressure drop per stage reasonably low.

TYPES OF GOVERNOR VALVES

There are two basic types of governor valves in widespread use: nozzle governor valves and throttle governor valves. Not only is the type of governor valve indicative of the service the unit was designed to see, but in addition is a determiner of the construction of the high pressure turbine.

THROTTLE GOVERNORS

In throttle governor valves, the steam flow to the turbine passes through a governor valve which controls steam flow to the turbine by throttling the steam and thereby controlling the steam pressure at the inlet to the high pressure turbine. Whether there is a single governor valve or several valves in parallel, the governor valves throttle the steam flow equally. At 25% turbine full power all the governor valves are passing 25% of their design flow. At 50% of full power all are passing 50% of their design flow and so on. The advantage of throttle governing is the simplicity of control and construction. This is particularly true of the steam inlet to the first stage nozzles since all of the nozzles are used at all times with the governor valves regulating steam flow through the first stage.



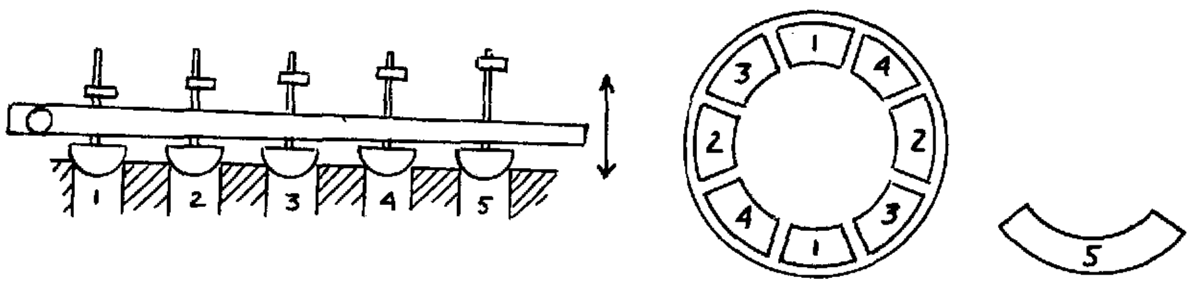
EFFECT OF THROTTLE GOVERNING

Figure 1.18

Figure 1.18 shows the condition lines for a throttle governed turbine at various power levels. Inspection of this diagram readily shows the disadvantages of throttle governing. At power levels below 100%, there is a large pressure drop across the throttle governor valves. This results in a large increase in entropy and a corresponding decrease in available energy. Since each kilogram of steam does less work at low power, the low power steam consumption per kilowatt-hour is much greater than at high power. This is clearly inefficient. The result is that throttle governing is seldom used on turbines that are used for variable load service. A throttle governed turbine is designed to operate at a nearly constant high power level.

NOZZLE GOVERNING

Nozzle governor valves are arranged as shown in Figure 1.19 and are opened sequentially either singly or in pairs.



Steam Chest With Bar Lift Nozzle Governing

First Stage Nozzles

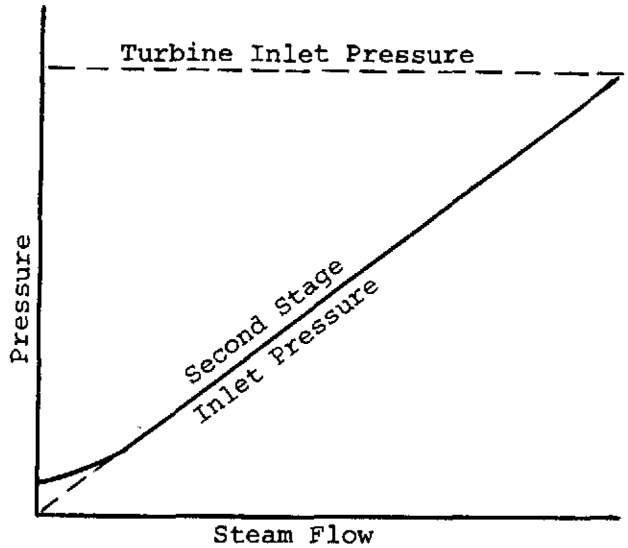
Bypass To Second Stage

NOZZLE GOVERNING

Figure 1.19

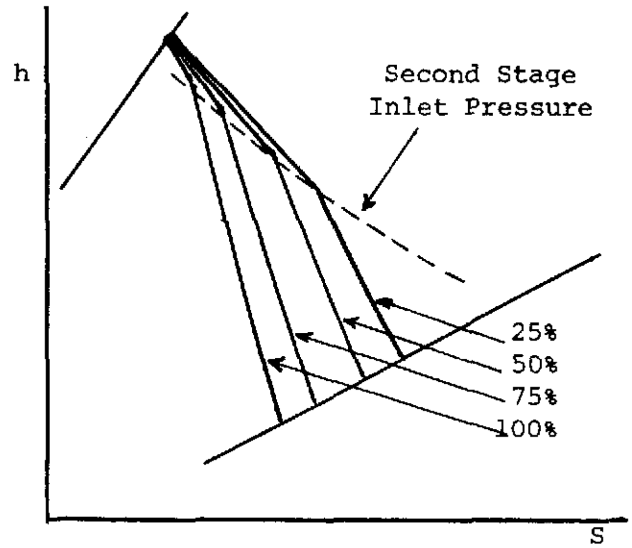
As the nozzle block moves up the valves open sequentially and as a result only one valve is throttling steam flow at any one time. This results in a nearly constant inlet pressure to the first stage and eliminates much of the adverse pressure drop associated with throttle governing at low power levels. To prevent the effect of the one valve which is throttling from effecting the pressure at the inlet of the first stage it is necessary to resort to each valve admitting steam to a different arc on the first stage nozzle ring as is shown in Figure 1.19. This complicates the structure of the inlet to the first stage nozzles and makes the use of a single flow turbine virtually manditory with nozzle governing.

To appreciate one of the other design features of a nozzle governed turbine, it is necessary to understand that the flow of steam through a turbine stage is roughly proportional to the pressure drop across the stage. This means as the flow through a stage increases, the pressure drop across the stage must increase.



VARIATION OF STAGE
PRESSURE WITH FLOW

Figure 1.20

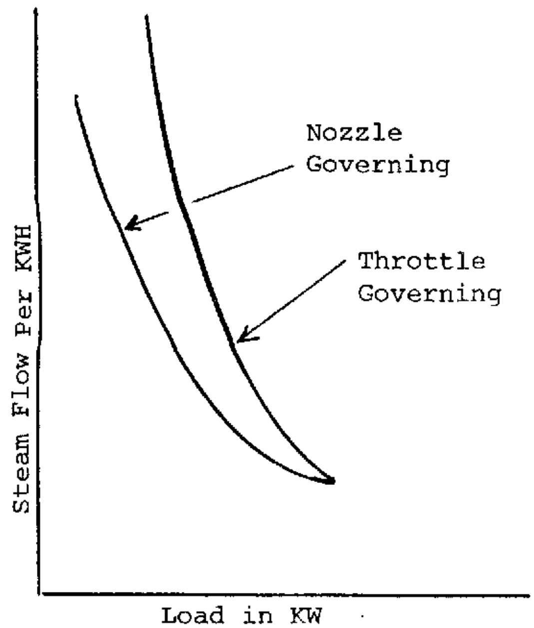


EFFECT OF NOZZLE
GOVERNING

Figure 1.21

Since the outlet pressure of the turbine is relatively constant, the inlet pressure to each stage increases as the power level increases. However, the inlet pressure to the first stage is at steam generator pressure and does not change. Figure 1.20 shows the inlet pressure to the first and second stages of a nozzle governed turbine. At low power level the pressure drop across the first stage of the turbine is very large and the first stage does a large percentage of the total work of the turbine. For this reason the first stage must be efficient with a large enthalpy drop and is usually an impulse stage. As the power level increases the pressure drop across the first stage decreases until it may be so small that the first stage becomes unable to pass enough steam to develop the required turbine power. To increase the maximum power of the turbine when efficiency is of a secondary importance, the first stage may be bypassed by some of the governor valves to send steam directly to the second stage.

Figure 1.21 shows the condition lines for a nozzle governed turbine at various power levels. Even though the nozzle governed turbine has a higher steam consumption (kg/kw-hr) at low power than at high power, it is more efficient at low power levels than a throttle governed turbine.



COMPARISON OF NOZZLE
AND THROTTLING GOVERNING

Figure 1.22

Figure 1.22 is a comparison of steam rates for nozzle and throttle governing. The nozzle governed turbine consumes less steam per kilowatt-hour for all power levels up to the maximum. The nozzle governed turbine is more efficient as a variable load turbine; however, a base load turbine which seldom runs below full power can obtain good efficiency with a much simpler governing system.

ASSIGNMENT

1. Explain the effects that each of the following have on the efficiency of a turbine steam cycle:

   (a) excessive moisture in the turbine.

   (b) pressure drop across the inlet valves.

   (c) moisture separator.

   (d) live steam reheater.

   (e) superheating.

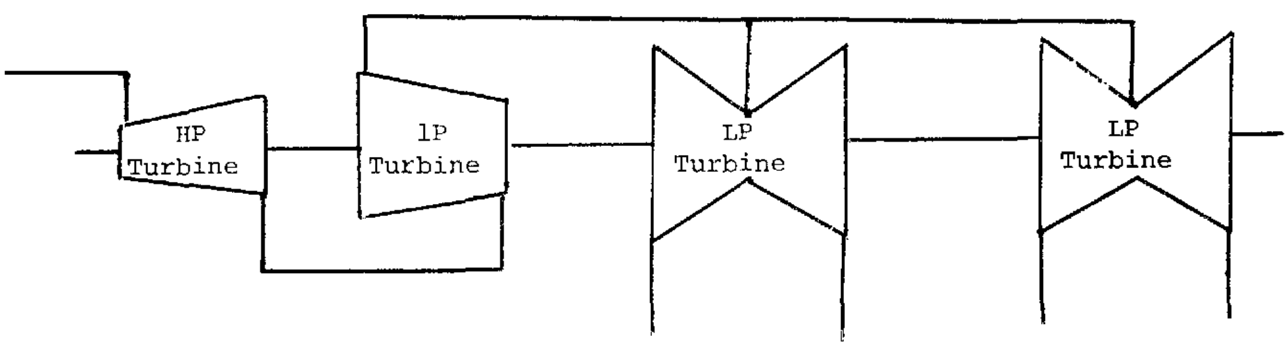
   (f) regenerative feedheating.

2. Draw and explain a condition line for a typical large CANDU turbine unit having one HP turbine and three LP turbines.

3. What are the problems involved in building a generating station utilizing a Carnot cycle.

4. Explain why we utilize feedheating in our generating stations.

5.



The diagram above is of a four casing turbine unit used at a conventionally fired, superheated steam generating station. Discuss possible reasons for:

(a) single flow high pressure turbine.

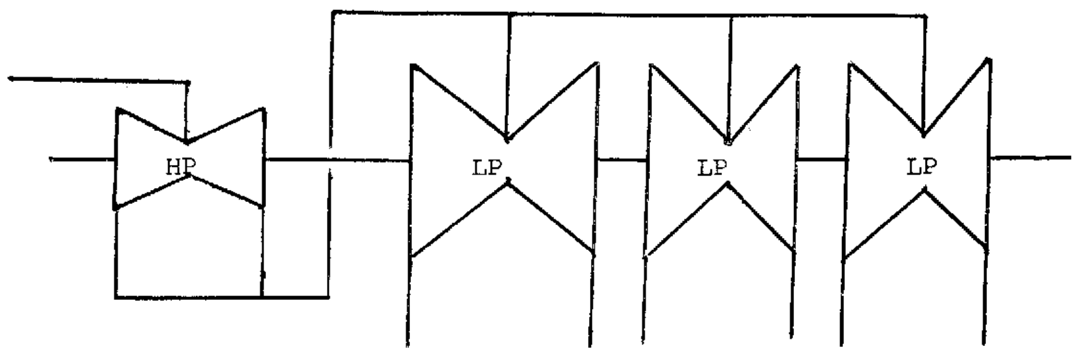
(b) single flow intermediate pressure turbine.

(c) double flow low pressure turbine.

(d) Curtiss Wheel as first two stages of HP turbine.

(e) Reaction stages in LP turbine.

6.



The diagram above is of a four casing turbine unit used at a nuclear fuel, saturated steam generating station. Discuss possible reasons for:

(a) double flow high pressure turbine.

(b) double flow low pressure turbine.

(c) reaction stages in LP turbine.

(d) reaction stages in HP turbine.

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 134

TURBINE OPERATIONAL PERFORMANCE

The purpose of any steam power plant is to convert heat energy released by the fuel into generator electrical output. Under the best circumstances this process is rather inefficient. In a typical CANDU generating station something on the order of 70% of the heat generated in the reactor is lost through the production of waste heat and through internal electrical power requirements. The ability to produce higher temperature superheated steam, allows conventional fossil fuel steam plants to be somewhat more efficient but even in these plants only about 40% of the heat energy is converted to electrical output from the generator.

Because the conversion of heat energy to electrical energy is at best rather inefficient and because the conversion process involves a great potential for a degrading of even this efficiency, steam plant operators have long been concerned with assessing the amount of heat energy required to produce a kilowatt-hour of electrical output. The amount of heat energy which must be produced in fuel burnup to produce a kilowatt-hour of electrical output is known as the station heat rate and the operator who saw an upward trend in this station heat rate was concerned not only because of wasted fuel dollars but also because it indicated something unpleasant was happening to his plant.

Theoretically the computation of a station heat rate for a nuclear generating station is simpler than for a conventional plant. The reason is that no conventional plant operator knows as much information about his heat source as a nuclear operator does. For reactor control and regulation the precise power level in the reactor must be known at all times. For a reactor operating at a constant power level the number of kilojoules of heat energy produced in an hour is simply

$$\frac{KJ}{hr.} = (\text{Thermal MW power})(3.6 \times 10^{6}\ \text{KJ/MW-hr.}) \qquad 2.1$$

The station heat rate, SR, can, therefore be easily computed by the following formula:

$$SR = \frac{KJ}{KW\text{-}hr}\ \text{output} \qquad 2.2$$

If a nuclear generating plant produces 1743.5 MW of thermal power in the reactor and 543 MW of electrical output, then the station heat rate can easily be computed as:

$$SR = \frac{(\text{Thermal MW power})(3.6 \times 10^6 \text{ KJ/MW-hr})}{\text{KW output}}$$

$$= \frac{(1743.5 \text{ MW})(3.6 \times 10^6 \text{ KJ/MW-hr})}{(543,000 \text{ KW})}$$

$$= 11559.1 \text{ KJ/KW-hr}$$

It does not require any particular insight to realize that if this value increases to 11,609.1 KJ/KW-hr, or by about .4%, then the plant is operating less efficiently, fuel is being wasted, and plant components are being taxed unnecessarily.

The problem comes in the fantastically inexpensive cost of nuclear fuel. The 1976 cost of CANDU nuclear fuel is only about $ .07 to $ .12 per 1,000,000 KJ. This should be compared to $ .60 to $ .90 for coal and $1.80 to $2.20 for oil. Although these values will undoubtedly change rapidly, the relative standing should remain nearly constant. Thus over a year, the inefficiency described above would cost the oil fired station about $450,000, the coal fired station about $170,000 and the nuclear station about $20,000. It should come as no surprise that you can get a lot more excited about a .4% increase in station heat rate at a conventional generating station than at a nuclear generating station.

The obvious economic conclusion of this is that based on fuel costs you cannot justify the expense of vast sums of money tracking down inefficiencies in a nuclear generating station. On the other hand there is another implication to an increasing heat rate. If the station heat rate is increasing then something is not working as it should. As the efficiency of the plant decreases due to a gradual deterioration of component capabilities, the probability of a forced outage increases. In the event the plant must be shutdown for required maintenance, the cost of alternate energy sources to replace the megawatt hours of lost electrical output can be staggering. At the time of this writing the estimated cost for alternate energy for a Pickering size nuclear generating station is on the order of $5,000 per hour. This figure depends on the cost differential between conventional and nuclear fuel but it is doubtful it will decrease over the foreseeable future.

The typical nuclear station is thus faced with a delemma, the horns of which are a very marginal rate of return on dollars invested for improvement in plant efficiency and a truly tremendous cost for an unplanned plant outage. It should be reasonably obvious that the tradeoff in dollars between these two extremes is a major challenge.

The practice of assessing steam plant operational performance as a method of determining the condition of the unit is fundamental to lengthing the time between major overhauls while avoiding a forced outage. However there are several factors which complicate the assessment of the condition of an operating turbine unit which deserve some attention.

Accuracy and Adequacy of Instrumentation

Once the operator has detected a trend of increasing station heat rate he is faced with the dual problems of first determining precisely where the problem lies and second deciding whether the problem justifies the expenditure of down time and maintenance dollars to correct it. The ability to trace a deteriorating heat rate to its ultimate cause requires the ability to determine accurately the actual conditions at various locations. This is often difficult and occasionally impossible using existing instrumentation. For example, to assess the condition of the turbine unit it is desirable to determine the heat energy delivered to the unit per kilowatt-hour of electrical output. The Turbine Heat Rate (THR) can be determined by the formula:

$$THR = \frac{M_1(h_s - h_{fw}) + M_2(h_2 - h_1)}{KW} \qquad 2.3$$

where:
- M_1 = steam flow through the stop valve (KJ/hr)
- h_s = steam enthalpy at the stop valve (KJ/Kg)
- h_{fw} = enthalpy of final feedwater (KJ/Kg)
- M_2 = steam flow through reheater (Kg/hr)
- h_2 = steam enthalpy from reheater (KJ/Kg)
- h_1 = steam enthalpy to reheater (KJ/Kg)
- KW = total electrical output in KW

In analyzing this theoretically rather simple formula it becomes obvious that the accurate determination of turbine heat rate using this formula requires the steam flow at two points as well as four steam enthalpies three of which require steam quality determinations. Before applying this formula these quantities must be known with sufficient accuracy to yield an accurate turbine heat rate. If the steam flows can only be determined within ± 5%, then the probability of correctly detecting changing unit efficiency is rather doubtful. While necessary parameters can often be estimated or derived from other parameters, this is at best often difficult and the installation of permanent or portable accurate instrumentation may well be the only way to assess unit performance properly.

Reproducibility of Initial Conditions

Although a variety of conclusions can be drawn from an increasing net station heat rate, one has to be careful not to be chasing a will-o'-the-wisp. A large number of factors can effect the net station heat rate which have little or nothing to do with a deterioration of the turbine or steam/feedwater system performance. Whenever a heat rate is conducted it is absolutely essential that uniform initial conditions be utilized on which to base comparisons. Normal variations in condenser vacuum, makeup water flow, steam pressure, steam generator level, adjuster rod motion, xenon inventory, generator hydrogen purity and heat transport temperature can result in hours of searching for nonexistent problems.

The method of turbine unit analysis which can yield the most productive results is based on initial conditions which are reasonably easy to reproduce. Full power with the steam and feedwater system in a "normal lineup" will probably result in the fewest induced errors. In addition these conditions should be held constant for some time interval - say five or ten minutes - prior to taking the heat rate data.

The following is a partial list of conditions which should be met prior to conducting a heat rate determination.

(1) generator producing approximatlly 100% of rated gross output

(2) steam flow from steam generators equal

(3) feed flow to steam generators equal

(4) xenon at 100% equilibrium

(5) steam generator levels constant

(6) condenser vacuum at some reference level

(7) reactor reactivity at equilibrium condition

(8) generator hydrogen purity at some reference level

(9) heat transport temperature constant

(10) steam pressure constant

(11) no steam generator blowdown in progress

(12) no steam flow to bulk steam plant or through reject system

The precise initial conditions and acceptable range of parameters should be specified for a net station heat rate. In addition, if these conditions cannot be met the quality of the heat rate will be downgraded.

<u>Frequency of Heat Rate Determination</u>

A detailed determination of heat rates on the various components of a generating station is the most accurate method of measuring turbine unit performance and assessing the proper operation of the turbine and its auxiliaries. However, this can be expensive in both time and manpower. For this reason it is generally considered good practice to carry out such tests only once per year and during pre- and post-overhaul tests. On the other hand, when heat rates are computed infrequently it is likely that comparability of results suffers. What probably represents the best compromise is frequent calculalions of net station heat rate using equation 3.1 to insure reproducibility of results and to detect any major changes, and to conduct a detailed heat rate at annual intervals. The net station heat rate conducted at the time of the detailed analysis could then be used to check reproducibility.

Turbine Heat Rate

Once a change has occurred in station heat rate, it becomes necessary to make a determination of the implication of the change. Generally this implication can fall into the following four categories:

(a) insignificant

(b) correct the problem without shutting down the turbine

(c) correct the problem including shutting down the turbine

(d) correct the problem during the next scheduled turbine outage.

The classification into one of these categories can logically occur only by knowing what is causing the change in heat rate. Since station heat rate is effected by a wide variety of things from the reactor to the generator, the problem becomes one of localization of cause.

Turbine heat rate is defined as the number of Kilojoules per hour delivered to the turbine unit per Kilowatt of generator electrical output. As such it is sensitive only to changes occuring in the steam/feedwater system and is independant of problems associated with the reactor, heat transport system and steam generators.

Turbine heat rate is computed from equation 2.3.

$$THR = \frac{M_1(h_s - h_{fw}) + M_2(h_2 - h_1)}{KW}$$

where: M_1 = steam flow through the stop valve (Kg/hr)
h_s = steam enthalpy at the stop valve (KJ/Kg)
h_{fw} = enthalpy of final feedwater (KJ/Kg)
M_2 = steam flow through reheater (Kg/hr)
h_2 = steam enthalpy from reheater (KJ/Kg)
h_1 = steam enthalpy to reheater (KJ/Kg)
KW = total electrical output in KW

Referring to the diagram 2.1 you will notice that Turbine Heat Rate would be computed as:

$$THR = \frac{(108.0)(3600)(2793.39 - 726.23) + (88.352)(3600)(2793.39 - 1103.98)}{790699}$$

$$= \frac{8.0645 \times 10^9 + .5373 \times 10^9}{790699}$$

$$= \frac{8.6018 \times 10^9}{790699}$$

= 10878.7 KJ/KW-hr

In lieu of equation 2.3 turbine heat rate can be computed from the following:

$$THR = \frac{M_3 h_s - M_4 h_{fw} - M_5 h_{rhd}}{KW} \qquad 2.4$$

where: M_3 = steam flow from steam generator (Kg/hr)
h_s = steam enthalpy from steam generator (KJ/Kg)
M_4 = feedwater flow (Kg/hr)
h_{fw} = enthalpy of final feedwater (KJ/kg)
M_5 = reheater drain flow (Kg/hr)
h_{rhd} = enthalpy of reheater drains (KJ/Kg)
KW = total elecrical output in KW.

The choice of equation 2.3 or 2.4 will depend largely on the accuracy to which the parameters used in the equations can be determined. In either case, the accurate determination of flow rates is probably the most limiting factor. From an operational standpoint equation 2.4 is probably the most convenient and can be further simplified by making some assumptions.

Since liquid flow can generally be determined more accurately than vapor flow it is often easier to approximate M_3 as the sum of M_4 and M_5. In addition since the enthalpy of saturated steam at steam generator pressure, it can often be treated as a constant Equation 2.4, therefore, becomes:

$$THR = \frac{M_4(h_s - h_{fw}) + M_5(h_s - h_{rhd})}{KW} \qquad 2.5$$

where: h_s = steam enthalpy of saturated steam at the design steam generator pressure. (KJ/Kg).

Since the majority of the heat energy loss (roughly 90%) occurs in the steam/feedwater system, a change in turbine heat rate is reflected virtually one to one in station heat rate. This means if turbine heat rate increases by .5% we would expect an increase of about .5% in station heat rate.

Once the cause for increasing station heat rate has been tracked to the steam/feedwater system, the problem is "simply" one of tracking down the particular offending component.

Condenser Backpressure

If the condenser back pressure increases, the turbine output will decrease and turbine heat rate will increase. Figure 2.2 shows this effect. The result has such a great impact on turbine heat rate, that two otherwise comparable turbine heat rate determinations with differing condenser vacuums will yield widely different results.

At a constant power level an increase in condenser back-pressure can be caused by only four things:

(a) increase in the average temperature of the condenser cooling water in the tubes

(b) flooding of the condenser tube surfaces due to high hotwell level

(c) air leakage into the shell of the condenser

(d) a decrease in the overall heat transfer coefficient of the tubes.

The cooling water temperature will vary considerably between summer and winter. Under normal conditions, however, the temperature rise across the condenser is reasonably constant. This can be seen from the formula expressing the heat rejected to the condenser cooling water:

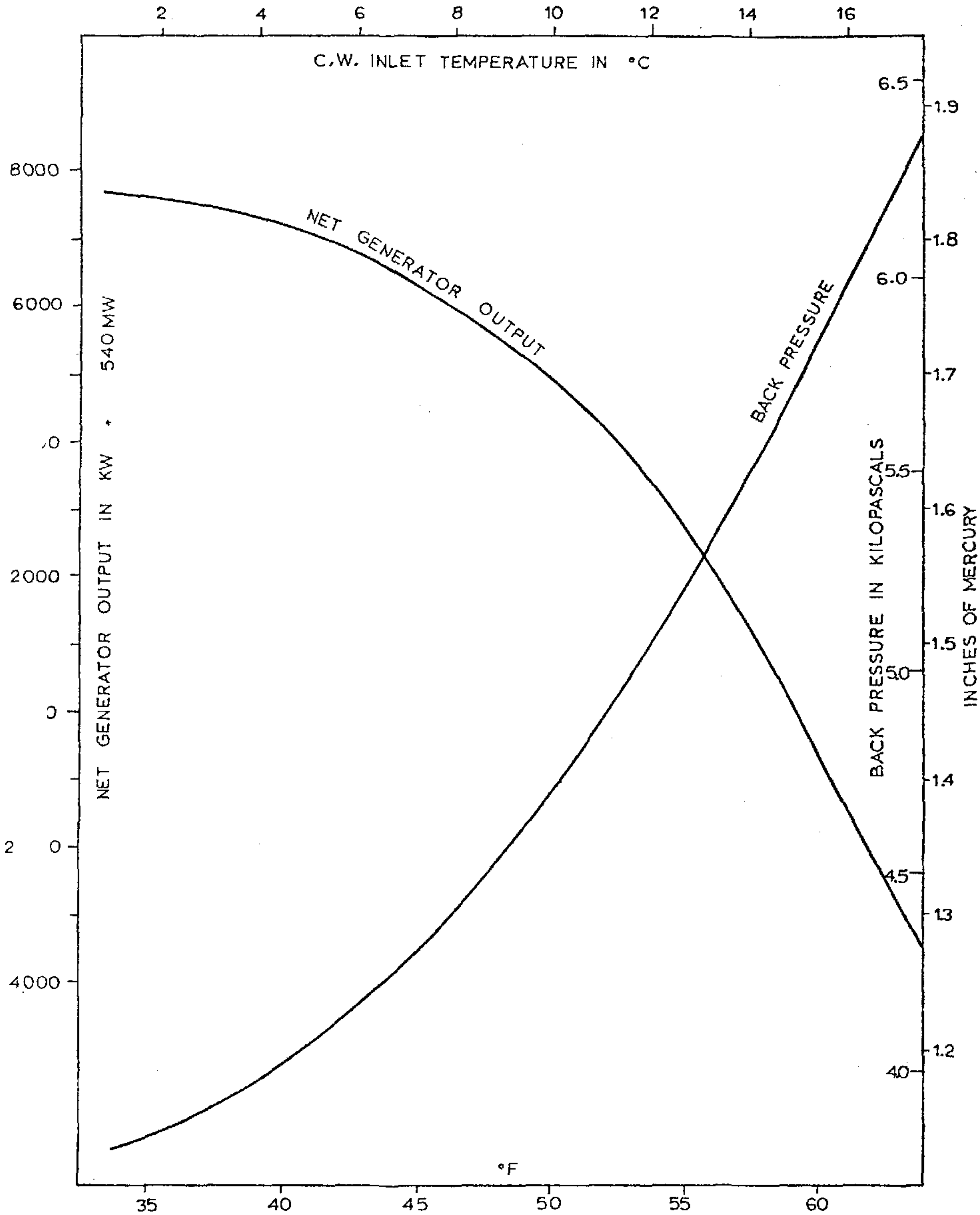
$$\overset{\circ}{Q} = m\, Cp\, \Delta T \qquad 2.5$$

where: $\overset{\circ}{Q}$ = the heat rejected to the ccw (KJ/sec)

m = ccw flow (Kg/sec)

Cp = the constant pressure specific heat capacity of water (KJ/Kg°C)

ΔT = the temperature rise in ccw across the condenser (°C)



Effect of Cooling Water Temperature on Back
Pressure and Net Output of Pickering G.S.

If the rate at which heat is rejected and the ccw flow rate remain constant, the ccw ΔT will remain constant. At a constant generator power, the rate at which heat is rejected to the ccw is reasonably constant for small changes in condenser vacuum. This implies that if vacuum is decreasing and condenser ΔT is remaining constant, then the cause is possibly an increase in the temperature of water entering the ccw system from the lake. This can be easily checked by monitoring ccw inlet temperature.

A change in lake water temperature usually occurs due to seasonal or diurnal fluctuations. Occasionally, however, a high wind has produced a sufficient current to return the water at the condenser outlet back into the ccw intake with a resulting rapid increase in ccw temperature.

Other factors which increase the average temperature of the condenser cooling water in the tubes are usually associated with a decrease in ccw flow from such causes as cavitation or air binding of the ccw pumps, marine growth in the condenser tubes, sand in the condenser tubes, blockage of the screens or tubes with debris, and air binding of the tubes or water boxes.

Returning to equation 2.5 you will notice that if the ccw flow rate decreases while a nearly constant rate of heat rejection is maintained then the ccw ΔT will increase.

$$\overset{\rightarrow}{\overset{\circ}{Q}} = \overset{\downarrow}{m} \, Cp \, \overset{\uparrow}{\Delta T}$$

The implication is that if vacuum is decreasing and ccw ΔT is increasing then there is a decrease in ccw flow. A decrease in ccw flow can be particularly troublesome if it results in a partial blockage of some tubes. This will increase the ccw flow through the remaining clear tubes with possible tube cavitation and failure resulting.

Air inleakage to the condenser will lower the condenser vacuum (increase the backpressure). However, since the increase in backpressure is attributable to air pressure as opposed to steam pressure, the temperature of the condensing steam (condensate temperature) will not rise appreciably as the backpressure increases.

A good measure of the tightness of the condenser and associated subatmospheric systems is the dissolved O_2 level in the condensate leaving the hotwell. If this level is rising then regardless of what is happening to condenser vacuum air is getting into the system. This additional air ingress can occur either from increased leaks or decreased air extraction capability. In either case the cause of the problem must be found and corrected not only to eliminate the inefficiencies caused by increased backpressure but because the long term effects of general and localized corrosion on the condensate, feedwater and steam generator will eventually produce significant problems.

If the overall heat transfer coefficient of the tubes decreases due to corrosion, scaling or fouling of the tube surfaces, the effect is quite similar to air ingress. Condenser Cooling Water ΔT remains constant and backpressure increases. However, since the increase in backpressure is attributable to steam pressure as opposed to air pressure, the condensate temperature will rise corresponding to the saturation temperature for the condenser backpressure. In addition, the condensate dissolved O_2 will not change if the cause of the vacuum decrease is not air.

The below table summarizes the type of parameter changes for various conditions affecting vacuum:

CAUSES OF POOR PERFORMANCE OF CONDENSER

SITUATION	CCW INLET °C	CCW OUTLET °C	CCW ΔT °C	BACK-PRESSURE KPa(a)	SATURATION TEMPERATURE CORRESPONDING TO BACK-PRESSURE °C	CONDEN-SATE TEMP. °C
Normal Operation	15.0	25.0	10.0	4.50	31.0	31.0
Decrease in ccw Flow	15.0	30.0	15.0	5.17	33.5	33.5
Increase in ccw Inlet Temp.	20.0	30.0	10.0	5.94	36.0	36.0
Air Leakage	15.0	25.0	10.0	6.27	37.0	31.5
Tube Surface Fouling	15.0	25.0	10.0	5.17	33.5	33.5

Feedwater Heating System

The feedwater heating system can be the source of a significant increase in turbine heat rate. Turbine heat rate is particularly sensitive to final feedwater temperature and a 3°C change in final feedwater temperature can affect turbine heat rate by as much as .2%. Generally the performance of the feedheating system can be monitored by watching:

(a) the final temperature of feedwater leaving each feedheater, and

(b) the terminal temperature difference between extraction steam and feedwater leaving the heater.

If these parameters remain close to those of the design heat balance then there cannot be too much wrong with the system.

When one feedheater does not raise the feedwater at the outlet to the design value, then the next feedheater will require more extraction steam if this temperature loss is to be regained. Because this additional extraction steam is taken from a point nearer the steam generator, energy which is normally utilized in the turbine is bled off as extraction steam. For a constant load output under these conditions, additional steam to the high pressure turbine is required and turbine heat rate increases.

The following problems are likely to encompass the majority of feedheating system deficiencies:

(a) Long term contamination of feedheating surfaces. This can occur due to oil ingress from the turbine or buildup of corrosion products.

(b) Extraction steam valves not fully open.

(c) Insufficient venting of the feedheater shell. This can be caused by fully or partially shut valves in the vent lines.

(d) Increased level in the feedheater shell. This floods out some of the tubes and reduces the heat transfer area.

(e) Tube blockage due to foreign material in the feedlines.

(f) Changes in extraction steam pressure or quality due to problems in the turbines. If the enthalpy of the extraction steam decreases then the flow of extraction steam will increase.

By carefully analyzing the feedwater ΔT and Δp, the terminal temperature difference, the feedheater shell pressure, and the shell drain temperature and comparing these parameters with design values, the cause of the deficiency in feedheater performance can be localized.

Turbine Internal Efficiency

The primary causes of a reduction in internal efficiency are:

(a) chemical deposition on turbine blades,

(b) increase in blade tip clearances due to erosion or physical contact between fixed and moving parts,

(c) changes in blade surface.

Because of wet steam conditions in the high pressure turbine, low steam generator pressure (and, therefore, temperature), and the shift to volatile steam generator chemistry, chemical deposition on turbine blading is not frequently a cause of loss of turbine efficiency on nuclear steam turbines. There have been cases of chemical corrosion of blading due to steam generator carryover in plants using solid steam generator chemistry and, therefore, the effects of possible chemical attack cannot be completely ignored.

Tip rubbing can be minimized by careful adherence to run-up and loading procedures and avoidance of conditions likely to produce excessive differential axial and radial expansion between the casing and rotor. Control of excessive blade erosion due to wet steam or standing water conditions is largely a design problem related to adequate moisture removal from each stage. However, errors in design such as improper sizing of stage drains and inadequate ability to remove moisture from moving blades can cause significant decreases in efficiency due to erosion.

With the wet steam conditions which exist in a nuclear steam turbine, the blades will gradually be eroded due to moisture impengement. This damage is usually most severe in the high pressure turbine and latter stages of the low pressure turbine and usually first affects the trailing edge of the front side of fixed blades and the leading edge of the back side of moving blades. The result of this erosion is that the profile and surface of the blade will change with time. If the wear becomes extensive, the blades may change to the extent that stage efficiency is reduced. It is very difficult to detect such blade wear without shutting down the turbine and examining it internally. However, careful observation of the pressure drops across a stage or group of stages may make the change apparent.

It is more likely in practice that if the blade wear is such that there is a noticeable increase in steam consumption, it will show up in the form of excessive vibration due to the out-of-balance of the blade wheels.

Steam Generator - Water Chemistry

Removal of impurities in the steam generators can have an effect on station heat rate because hot water lost through blowdown must be replaced through cold makeup water. If the amount of blowdown is significant, there may be a noticeable effect on station heat rate, however, this effect is far out weighed by the consequences of running with out-of-specification steam generator chemistry.

The long term effect on heat rate through tube fouling, turbine blade deposits or derating far exceeds the advantage gained by minimizing blowdown. Each out-of-specification condition is significant and the cause should be rapidly corrected to avoid both the short and long term effects.

Gland Steam Consumption

Problems in the gland seal system are usually not sufficient to cause a noticeable increase in turbine heat rate. Since only about.08% of the steam flow from the steam generators is used to seal the turbine glands, the effect on heat rate is minimal. However, steam flow to the glands is a good indicator of the basic condition of the gland and for this reason can be valuable in diagnosis of gland problems.

Deterioration of turbine labyrinth glands usually occurs from thermal bending of the shaft or radial rubbing in the glands during startup. These problems can cause significant vibrations on startup but tend to become self limiting as the unit speed is increased above the critical speed and the unit warms up. This is particularly true of radial rubbing and as a result gland deterioration can occur without the operator being fully aware of the problem.

The problems can be largely eliminated by:

(a) correct operator interpretation of vibration on startup,

(b) correct steam to rotor and steam to casing differential temperatures,

(c) avoidance of low bearing oil temperature,

(d) avoidance of prolonged low speed operation, and

(e) proper alignment of rotor and casing during overhaul.

Providing the turbine unit is reasonably free of air leaks, the level of dissolved oxygen in the condensate, the length of time to draw a vacuum, and the length of time to lose vacuum when the air extraction system is shutdown are good measures of the condition of the gland.

Derating

Derating of a generating station is the process of restricting generator output below full power because of some abnormality in the system. Because the generating station is forced to run at less than its design capability, derating can have a significant effect on station heat rate.

The ultimate derating occurs when the heat source system is no longer capable of producing safely the number of kilojoules per hour required to produce the generator design output. In the case of a reactor there is an absolute upper limit to power output. Although the generating station is designed to allow some increase in station heat rate before reaching this upper limit, a decreasing station efficiency will eventually reach the point where the reactor plant reaches its limit before the generator gets to 100% of its design output.

When this occurs there are only two possible alternatives:

(a) increase the design capability of the reactor, and thereby allow the reactor to produce more power by lowering the safety margin. This, of course, would require consultation with the reactor designer and with the AECB to obtain consensus that the reactor plant was overdesigned in the first place.

(b) find and correct the cause of the increasing station heat rate so that the reactor can once more produce design generator output within design reactor specifications.

Deratings of a more temporary nature may occur when the generating station cannot be safely operated at 100% of design generator output. While the problems in the conventional end of a nuclear station which may result in derating are almost endless, the following have been occasional sources of deratings.

Condenser Circulating Water ΔT

To limit algae growth in the vicinity of the outflow, environmental authorities have imposed a limitation of 10°C on the temperature differential of the CCW across the condenser. Inability to meet this limit at full power necessitates a derating until the temperature rise is within the limit. In the case of some older stations, the limit was imposed after the station was built and has resulted in what amounts to permanent derating.

Generator Hydrogen Pressure

Loss of hydrogen from the generator results in a decrease in effective generator cooling and an increase in generator temperatures. The generator manufacturer specified maximum generator ratings for various values of hydrogen purity and pressure. In addition there are limits on stator and rotor temperatures which could be exceeded under full generator load with less than design hydrogen pressure. For these reasons if it is impossible to maintain hydrogen pressure at the design value for the rated load, the unit would have to be derated.

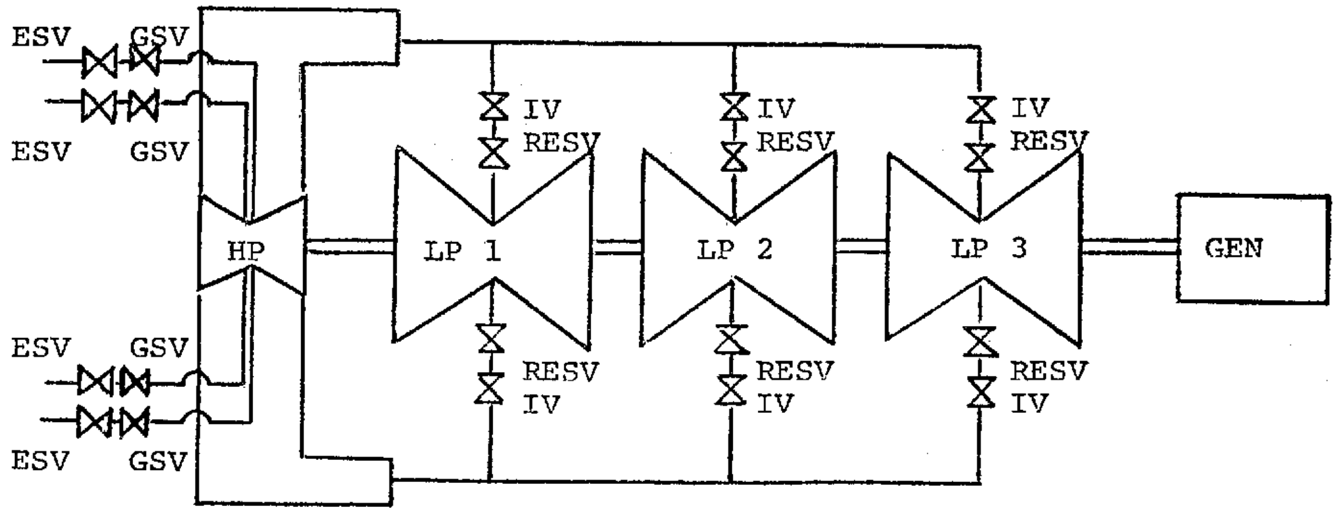
Feedheaters in Service

In addition to improving the efficiency of the steam cycle as described in the last lesson, the feedheating system is necessary to protect the steam generators from several hazards. If the temperature difference between the incoming feedwater and the water in the steam generator is excessive, there may be significant thermal stresses set up in the preheater, tubes, tubesheet and other structural members of the steam generator. These stresses can cause a shortening of steam generator life through fatigue, or if the stresses are severe enough, cause tension failure of the stressed members. For this reason, the feedheating system must heat the feedwater to a sufficient temperature to lower this ΔT and, therefore, the resulting stresses to within design limits.

If some feedheaters are out of service, the remaining feedheaters may not be able to raise the final feedwater temperature to a sufficiently high value to allow an acceptible ΔT between the feedwater and the generators. In this case the feedwater flow would have to be reduced to allow the available feedheaters to raise the final feedwater temperature to within an acceptible ΔT. This problem has been minimized by having two banks of feedheaters either one of which can supply full feedheating capability. However, during "poison prevent" operation when extraction steam is unavailable and the deaerator provides the only effective feedheating, the requirement to heat feedwater above a specified lower limit imposes a very real limit on plant operation.

Turbine Steam Control Valve Operation

Figure 2.3 shows a typical large turbine unit and the associated control valves. These valves are designed to control the steam supply to the turbine under a variety of normal and casualty conditions. If one of these valves was inoperative the unit might not be able to supply safely 100% of rated load.



ESV = Emergency Stop Valve
GSV = Governor Steam Valve

IV = Intercept Valve
RESV = Reheat Emergency Stop Valve

CONTROL VALVES

Figure 2.3

For example, if an intercept valve would not operate, then the limiting of overspeed on a load rejection would be impaired as some steam would enter the LP turbine through the malfunctioning intercept valve. Under this condition, if continued unit operation was necessary the steam flow to the turbine might have to be reduced so that an overspeed would be limited to acceptable values. Similarly, if a reject valve were unable to open, the pressure in the piping from the HP to LP turbine might rise to unacceptable values on a load rejection. In this case the turbine might have to be derated to allow continued safe operation.

ASSIGNMENT

1. Define Station Heat Rate.

2. Define Turbine Heat Rate.

3. Why are station heat rate and turbine heat rate not equal?

4. What design factors can effect heat rate?

5. What operating factors can effect heat rate?

6. For your station how would you answer the question:

   "If half the feedheaters in a feedheating system were inoperative, why might it be necessary to derate the turbine?"

7. What factors might cause derating of a station?

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 134

TURBINE OPERATIONAL PROBLEMS

As steam turbine units increase in size and complexity the operational problems also increase in magnitude. Not only has construction and control become more complex but materials have been pushed closer to their operating limits. As structures have become more massive, thermal gradients and pressure stresses have become more complex. In addition, with the increase in size it is no longer possible to build operational models and test them exhaustively before putting them into commercial operation. Today's turbine units go directly from the drawing board to on-site erection and commissioning.

Problems due to design errors are reasonably rare, but they do occur and because these occurrences are generally catastrophic it is becoming practice to build mathematical models of the system. Computer programs are then used to simulate normal, abnormal and casualty operations so that an assessment of in-serive performance can be obtained prior to commissioning. By this method it is often possible to establish operating limitations before the design leaves the drawing board.

Because of the large capital investment in any modern generating station, reliability of the unit becomes a significant concern. This is particularly true of nuclear stations where the cost of alternate electrical energy sources can be truly phenominal. For this reason it is becoming standard practice to assign a dollar value to estimated unreliability and factor this into the initial cost of the unit. This practice hopes to avoid an initial low price which turns out to be no bargain in service.

Despite these precautions, problems do occur particularly in the first year or so following commissioning. Not only do problems more frequently occur with a new plant but the operating and maintenance personnel require some time to familiarize themselves with the station.

The problems discussed in this lesson are derived from significant event reports and operating experience in nuclear and non-nuclear stations. Particular problems are included either because they occur with some frequency or because they represent a significant hazard to the turbine unit. The comments in this lesson are only of a general nature and are not intended as a substitute for design or operating manuals which constitute the manufacturer's specific recommendations on the operation of a specific turbine unit.

Overspeed

The hazards of an unterminated overspeed generally fall into one of three categories:

(a) speed will rise to a level where the centrifugal forces on the largest diameter wheels will cause tensile failure (rupture) of the wheel,

(b) speed will rise into a critical speed region and remain there long enough for the resulting amplification of vibration to cause failure, or,

(c) speed will rise to a level where the added stress due to centrifugal force will fail a component which has been weakened through fatigue, erosion or some other long-term phenomena.

The potential for an actual overspeed of the turbine unit occurs from two principle conditions: load rejection and testing of the overspeed trip mechanism. The response of both the mechanical-hydraulic and electrical-hydraulic governing systems to an overspeed following load rejection is discussed in the level 2 course and lesson six of this course.

The periodic testing of the operation of the overspeed bolts to trip the unit on an actual overspeed condition places the unit in a condition which can easily result in damage. Because the operation of the overspeed bolts is the last protective feature which functions to limit overspeed, the testing of this trip requires either the disabling of protective features which operate at lower overspeeds or raising the setpoint of these features above the trip point of the overspeed bolts. If the protective features fail to operate properly, the unit speed can be raised to dangerous levels. The testing of overspeed tripping devices is always a hazardous evolution and requires a detailed operating procedure. At least, two independent methods of monitoring turbine speed should be used and personnel conducting the test should be in continuous communication with each other. There should be no question under what conditions the test will be terminated. The raising of speed to the trip point should be smooth and rapid enough to limit the time above operating speed to that required to allow monitors to follow the speed of the unit. Personnel conducting the test should constantly ask themselves if the unit is safe, even if <u>none</u> of the trips function as expected. It should be borne in mind that the vast majority of turbine casualties involving overspeed occur during this type of testing.

Motoring

When the reactor heat production is lost through a reactor trip, the governor steam valves will shut to prevent the turbine steam consumption from lowering heat transport system temperature and pressure. If the generator output breaker is left shut, the turbine generator unit will motor with the turbine being driven by the generator acting as a synchronous motor. There are certain advantages to maintaining the turbine unit motoring during a reactor trip. Keeping the unit at operating speed shortens the time from steam admission to generator loading on the subsequent startup. This enables a faster recovery: first to avoid xenon poison-out and second to return the generator capacity to the grid.

During motoring the turbine blading is turning through dead steam and the friction between the steam and the blading rapidly overheats the long turbine blades at the exhaust end of the low pressure turbine. The problem is made more severe if the vacuum decreases and the blading encounters higher than design steam densities. The problem can be partially alleviated by an exhaust spray system and a cooling steam system as shown in Figure 3.1.

The exhaust spray system uses water off the discharge of the condensate extraction pump which is sprayed into the exhaust annulus of the turbine. This spray helps cool the dead steam as it is circulated by the rotation of the final low pressure turbine stages. To aid this system, steam is taken from the high pressure steam line ahead of the governor valves and routed to the inlet to the LP turbine. This "cooling steam" keeps a positive direction of steam flow through the LP turbine stages which helps to remove the windage heat.

Even with both cooling steam and exhaust sprays in operation, the final stage LP blading will overheat in something like an hour. This will require stopping the motoring of the unit. However, since the reactor will poison-out in about the same time frame, there would be little advantage in extending this limit.

Low Condenser Vacuum

When condenser vacuum decreases below design values, the turbine unit is subjected to a variety of unusual stimuli. Heat rate increases as less work is extracted from each kilogram of steam; the turbine internal pressure profile changes; extraction steam pressure and temperature change; the distribution of work between the high and low pressure turbine changes. However, the most immediate problem associated with vacuum decreasing below design is that the condenser will eventually not be able to condense all the steam being exhausted

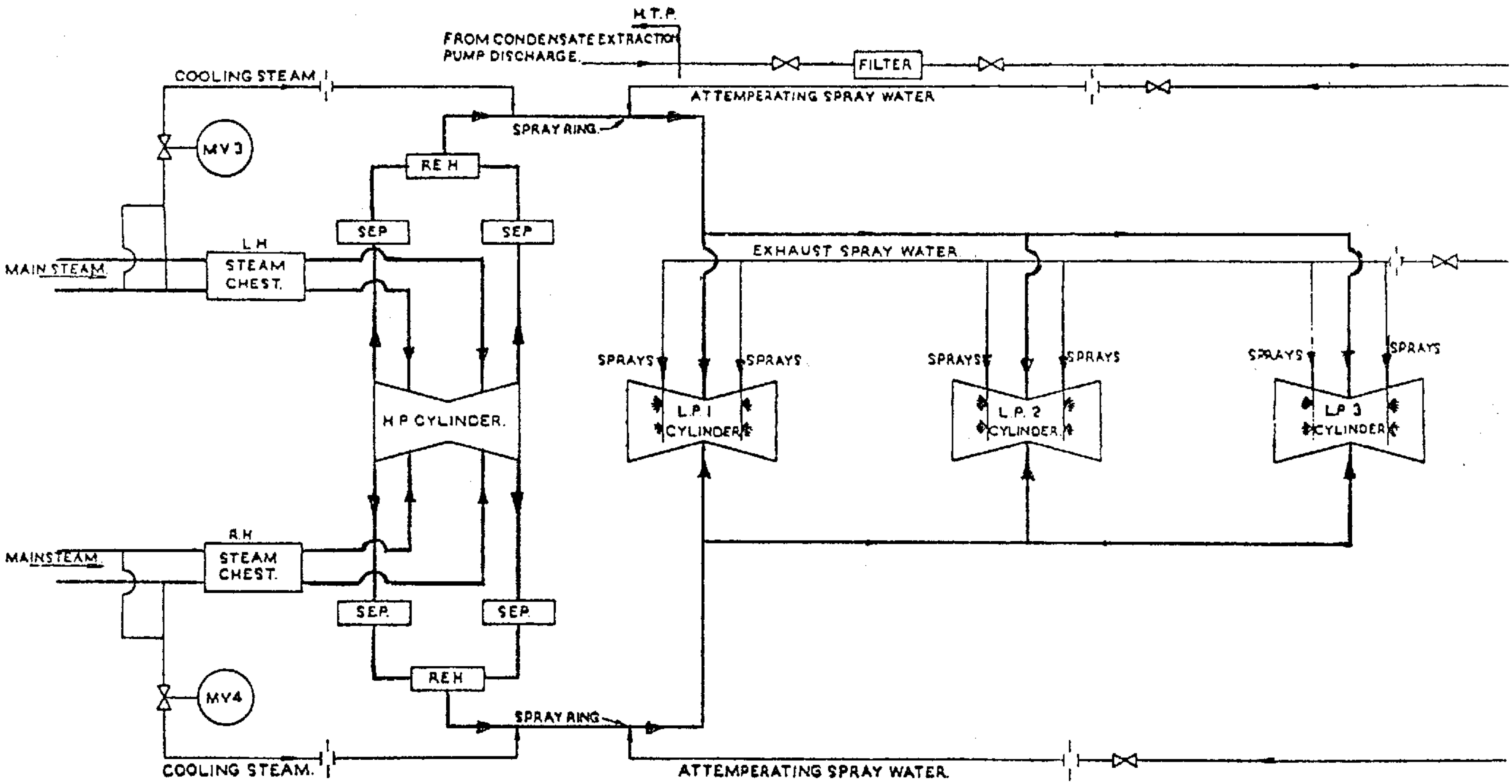


Figure 3.1

SCHEMATIC ARRANGEMENT OF L.P. CYLINDER EXHAUST SPRAY COOLING SYSTEM AND COOLING STEAM SYSTEM.

to it. In order to restore equilibrium to the condenser, the amount of steam rejected to it must be decreased. For this reason when vacuum has fallen below the minimum at which full power can be handled, the turbine will automatically begin to unload to a power level where the condenser can again reach equilibrium. With a design vacuum of 720 mm of Hg [5 kPa(a)], unloading will start at around 705 mm of Hg and will continue until either vacuum stops decreasing or 10% power is reached at 575 mm of Hg. If the vacuum decreases further the emergency stop valve will trip shut at 560 mm of Hg.

The combination of a low power level and low condenser vacuum imposes particularly severe conditions on the low pressure turbine blading. Not only do the long blades have to pass through a high density steam-air mixture but the absence of adequate steam flow through the turbine decreases the rate of heat removal. The adverse effect of low vacuum, low steam flow is the reason for terminating vacuum unloading before the governor steam valve fully shuts off steam. This effect also explains why, on a startup, vacuum should be the best obtainable before rolling the turbine with steam. It is desirable to maintain condenser vacuum on a shutdown until the turbine speed has decreased to about 50-60% of synchronous, to avoid a no steam flow, low vacuum condition.

Water Induction

Water damage to modern saturated steam turbines can be roughly divided into two categories: long term erosion by wet steam and catastrophic damage due to ingress of large quantities of water. The former cause of turbine damage is covered in the level 2 course and will not be covered here.

Slugs of water can enter the turbine through a number of places, however, the two most common sources of turbine damage are due to water induction through the governor steam valve and through the extraction steam lines. Water induction causes damage in three principle ways:

(a) direct impact damage on turbine components such as blading, diaphragms and blade wheels,

(b) excessive thrust caused by water impingement leading to thrust bearing failure or hard rubbing between components, and

(c) thermal damage to components due to quenching by water which may result in excessive thermal stresses, thermal distortion, or permanent warping. This is particularly true in the superheated section of the low pressure turbine.

Slugs of water which enter a turbine at high velocity will take the shortest path through the turbine, possibly clearing out both fixed and moving blades in the process. Because of the greater fluid density and the resulting impact on the rotor, induction of water from the steam generator may result in thrust loads much higher than design values. A failure of the thrust bearing can result in excessive axial travel of the rotor and subsequent severe rubbing damage to blading, blade wheels, diaphragms, glands and other components. Because of its high heat capacity, water contacting hot turbine parts can cause severe thermal stresses and distortion. This distortion can cause secondary damage if a turbine is restarted before the distortion has dissipated. While thermal distortion is not particularly severe in saturated steam portions of the turbine, it can be a significant cause of damage in the superheated sections.

Prevention of water induction requires both proper operation of protective features and careful avoidance of operating errors. The induction of water from the main steam line is minimized by the steam generator level control system, the high level alarm and by closure of the governor steam valve on high water level. However, improper or inadequate draining of steam lines during startup and subsequent loading can result in slugs of water being accelerated down the steam lines and into the turbine.

Water induction into the turbine can have particularly severe consequences on startup. While running under load, the steam flow can be of some benefit in absorbing water and minimizing thermal distortion, particularly in superheated sections of the turbine. Moreover, damage from rubbing can be increased when rotor speed is in the critical speed range.

If high vibration or other serious problems necessitate shutting down the turbine, the unit should not be restarted until all the water has been drained from the unit and the cause of water entry found and corrected. In addition, sufficient time should be allowed for relief of thermal distortion of the casing and rotor. Experience has shown that the most serious damage from water induction often occurs considerably after the first indication of water induction and attempting to restart may result in extensive damage due to rubbing between fixed and moving parts.

Condenser Tube Leaks

The consequences of impurities entering the feedheating system and steam generators through condenser tube leaks is covered in considerable detail in the level 2 Chemistry course. The adverse consequences of feedwater contamination with raw lake water fall into three general categories:

(1) Introduction of ionic and non-ionic impurities which may cause or accelerate localized corrosion of feed and steam generator system components particularly the steam generator tubes.

(2) Introduction of impurities which can lead to formation of boiler scale on steam generator tubes which decreases the overall heat transfer coefficient of the tubes, and

(3) Introduction of impurities which upsets normal system chemistry which can result in increased general corrosion of components. This can be deleterious in its own right and can cause release of the oxide film from the feed system which results in an accumulation of these oxides in the steam generator.

Historically the well being of the steam generators following a condenser tube leak has been maintained through blowdown, pH control and the use of phosphates (PO_4^{-3} and HPO_4^{-2}) to minimize boiler scale. While a steam generator which is well laced with phosphates is protected against boiler scale by the phosphates (which can also be used to maintain a buffered pH in the steam generator), the phosphates can cause considerable problems:

(1) Over the long term the phosphate salts (Na_3PO_4 and Na_2HPO_4) tend to come out of solution and deposit in various locations throughout the steam generators. The deposition of literally hundreds of pounds of steam generator chemicals and corrosion products in the steam generator not only reduces the heat transfer area and alters the design flow distribution around the tube bundle but also acts as a collection point for all manner of corrosive ions which promote localized corrosion, and

(2) Phosphates which leave the steam generator via blowdown act as a water pollutant through the stimulation of growth of marine life.

To reduce the deposition of boiler chemicals, there has been a gradual decrease in the specified level of phosphates in the steam generators. This has culminated in adaption of "zero solids" or "volatile" boiler chemistry in which phosphates have been eliminated during normal operation and are injected only during tube leaks. Recently, the effects of phosphates on water pollution has raised serious questions about even this use.

There are few problems in a nuclear generating station which present the potential for long term headaches that improper care of the steam generator does. Several nuclear generating stations in the United States have experienced long shutdowns while retubing steam generators. On the other hand the short term effects of steam generator abuse are reasonably undramatic and in the early years of operation there may be a tendency to treat the steam generators as if they will go on forever.

The impurities introduced by condenser tube leakage must be rapidly isolated and the steam generators quickly returned to in specification chemistry. In addition conditions which lead to accelerated erosion or corrosion of condenser tubes must be eliminated. These factors include sand in the circulating water which tends to erode tubes, high circulating water flow which may cause tube cavitation and erosion, tube fretting due to incorrect installation, high steam exhaust velocity or improper steam baffle operation which causes erosion from the steam side, and indiscriminant dumping of live steam to the condenser which accelerates steam side erosion.

Blade Failure

If there is a complete failure of a turbine blade in operation the effects may be disastrous as sections of blades get stuck between rows of fixed and moving blades and can strip the blade wheel. The resulting vibration can completely wreck the turbine. This type of failure due to metal failure is extremely rare due to advanced metallurgical developments and methods of blade fixing. However, because of the high stresses imposed on rotating blades and shroud bands, even minor errors during installation or replacement of blading may lead to early blade vibration, cracking and ultimate failure.

Probably the most significant source of blade failure is damage induced by water impact and erosion. Not only is the quality of steam entering the turbine important but in addition the ability of the blade to shed water can influence blade life. Use of cantilevered blades without shrouds is becoming reasonably widespread in nuclear steam HP turbines as the shroud tends to restrict the centrifuging of water droplets off of the blade. There have also been cases of blade tip and shroud band erosion and failure due to inadequately sized stage drains which resulted in standing water in the turbine casing.

Water erosion in the exhaust end of the HP and LP turbines has caused failure of lacing wires and damage to the leading edges of the blading. The erosion of blading causes pieces of metal to break off which may cause damage to fixed and moving blades in subsequent stages. Defects of this kind are minimized by having a very hard stellite or chrome steel insert welded to the leading edge of LP turbine blades. In cases of extreme water erosion, however, these inserts may become undercut and themselves break loose to become a source of impact damage.

In operation, centrifugal stresses, bending stresses and thermal stresses may ultimately cause fatigue cracking of the blade roots. These cracks can only be detected during shutdown by non-destructive testing. Any evidence of blade cracking should be treated with caution as it is not only indicative of an abnormality within the turbine but also can lead to catastrophic blade failure.

Expansion Bellows Failure

Expansion bellows are used extensively in large turbines on LP pipework and between LP turbines and the condenser when the main condenser is being used as a reject or dump condenser.

In practice the bellows develop hairline cracks due mainly to thermal cycling as a result of load changes. Failure may also be caused by overload, for example, if an expansion bellows is fitted between the LP turbine and the condenser, the bellows may become strained if the condenser is over-filled without supporting jacks in position to take the weight.

Bearing Failure or Deterioration

Recent experience indicates that approximately half of all major turbine problems involved the bearings and lubricating oil system. With only a few exceptions most bearing problems can be traced directly to malfunctioning or maloperation of the lube oil system. Provided the lube oil system performs its primary function of supplying clean lube oil at the proper temperature and pressure to the bearings at all times when the turbine/generator shaft is rotating, there is usually little problem with the bearings.

Since even a brief failure of the lube oil flow to the bearings can result in considerable damage to the unit, the system is designed to provide continuous oil flow under a variety of pump shutdowns and power failures. The automatic features which provide the backup lube oil supply must be tested frequently to insure satisfactory operation. In particular the pressure switches which indicate low lube oil pressure should be tested not only for proper annunciation but also to insure that they are capable of starting the appropriate backup pump. In addition the response time of backup pumps should be tested to insure that continuity of lube oil flow is maintained. Testing should be conducted with consideration given to the consequences of a failure of the system to operate as designed. For example, if the starting of the dc emergency lube oil pump is tested by turning off the auxiliary oil pump, with the unit on the turning gear, the shaft will be left rotating with no oil flow if the dc pump fails to start.

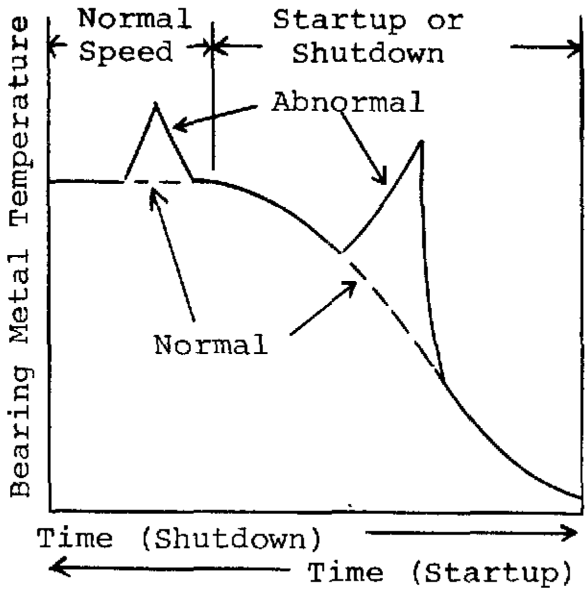
Of almost equal importance to bearing well-being is the cleanliness of the lube oil. Contamination of the turbine lube oil with water, fibers, particulates, dirt, rust and sludge can not only destroy the lubricating properties of the oil but also can cause accelerated bearing wear due to deposition of grit between the bearing and shaft journal. The lubricating oil should be sampled frequently. The results can be used to assess the quality of the oil and the efficiency of the purification system. Sample points should be chosen to insure samples represent not only the oil in the sump but also the oil going to the bearings. Metal particles in either the lube oil samples or the strainers should receive particular attention as they may indicate bearing, journal or pump deterioration.

One of the most effective ways to monitor proper bearing performance is through bearing metal temperature. A gradual increase in metal temperature over a period of several weeks or months can indicate a gradual deterioration of the bearing. Bearing metal temperature is influenced primarily by load, shaft speed and the type of bearing. Of a lesser importance under normal conditions are bearing journal surface, alignment, oil flow and inlet oil temperature. With the shaft at rated speed and oil flow and temperature normal, an upward trend in bearing metal temperature indicates a change in bearing load, alignment or journal surface condition. Temperature spikes of the type shown in Figure 3.2 can be excellent indicators of bearing deterioration. High spots on the journal or bearing can cause metal to metal rubbing until wear has eliminated the contact. This is particularly true on shutdown or startup when the oil film in the bearing is thinner and, therefore, there is more susceptibility to scoring.

Bearings should be inspected for wear and alignment at least each time the turbine unit undergoes a mojor overhaul. Journals should be checked for smoothness and uniform roundness and diameter from one end to the other. Journals should be inspected for scoring or an uneven surface which occurs from scoring and self-lapping over an extended period. A bearing metal wear pattern such as shown in Figure 3.3 is indicative of journal to bearing misalignment.

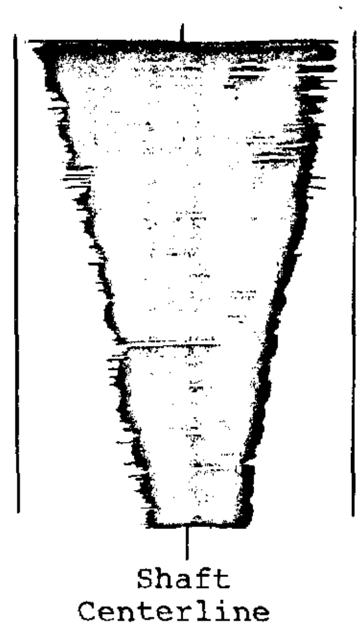
There is a popular and untrue notion that spherical seated journal bearings are self aligning during operation. Bearings must be properly aligned to prevent deterioration.

Additionally, the ball seats must be tight to prevent wear of the ball seats from causing vibration of the bearing.



Abnormal Bearing Metal Temperature

Figure 3.2



Misaligned Bearing Wear Pattern

Figure 3.3

Electrical-Hydraulic Governing Systems

Failure of the turbine governing system has been a continuing source of problems. As turbine unit size increases there has been a continuing need for faster acting and more reliable governing systems. This trend has resulted in high pressure, fire resistant fluid (FRF) electrical-hydraulic governing systems replacing more conventional mechanical-hydraulic governors using lubricating oil on many large turbine generators.

While the need for periodic testing of governing systems remains undiminished, the need for system cleanliness and hydraulic fluid purity in high pressure, FRF governing systems is considerably more critical. Removal of impurities which could foul and eventually score the electrical-hydraulic control valves is essential in any high pressure hydraulic system. Control valve clearances are extremely small and the valves are particularly susceptible to sticking, scoring and eventual internal leakage. In addition, electrically actuated valves generally have little reserve power to free galled or sticking stems. The requirement for periodic testing of an electrical-hydraulic governing system to insure freedom of valve movement is doubly beneficial in that fluid flow through the control valves keeps the valves clean. There is a great amount of practical experience which indicates that if a hydraulic control valve is not exercised for several months it will probably not operate.

All oil and particularly the synthetic, phosphate esters used in FRF systems have a tendency to pick up impurities notably chlorides and water which tend to accelerate system corrosion and eventual failure. The use of water or chlorine based cleaning solvents around these systems is an invitation to problems.

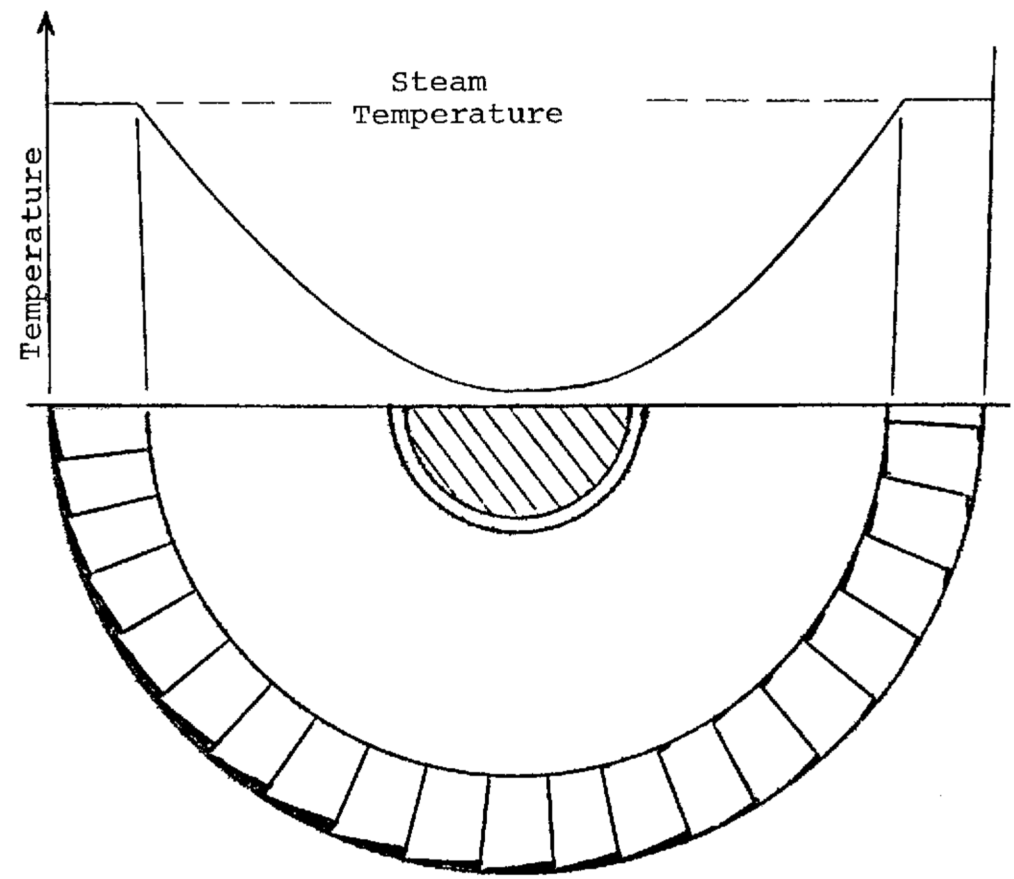
The fire resistant fluid used in most electrical-hydraulic governing systems is a synthetic phosphate ester hydraulic fluid. The fluid looks like and feels like a light mineral oil. It has good lubricating properties and excellent stability. The FRF used by Ontario Hydro has an excellent combination of chemical and physical properties: low particle count, low chlorine content, high electrical resistivity and negligible corrosion of most metals. This makes it a good fluid for use in the close tolerance valves, limit switches and overrides which are used in electrical-hydraulic governing systems. Above all, FRF virtually eliminates the fire hazard associated with conventional petroleum oils leaking onto hot steam lines.

FRF is reasonably non-toxic to the skin and exposure through soiled clothing presents a minimal hazard although FRF entering the eyes can cause a burning sensation and cause subsequent irritation. Phosphate esters can cause fatal poisoning, however, if inhaled in large quantities or if ingested in even moderate amounts. Under usual station operating conditions, inhalation of the vapor is almost impossible due to the low volatility of the fluid. However, when heated to decomposition, the phosphate ester can emit highly toxic fumes of phosphorous oxides. For a personal safety standpoint FRF can be harmful and special care should be taken to prevent ingestion, inhalation or absorption through the skin by persons who handle it. However, it can be handled safety if certain precautions are taken. These include no smoking or eating while handling the fluid, use of rubber gloves and safety glasses and availability of an eye wash fountain.

From an environmental standpoint, FRF can present a potential problem due to its toxicity and relative stability. For For this reason disposal procedures for this fluid should be carefully adhered to.

Low Cycle Fatigue Cracking

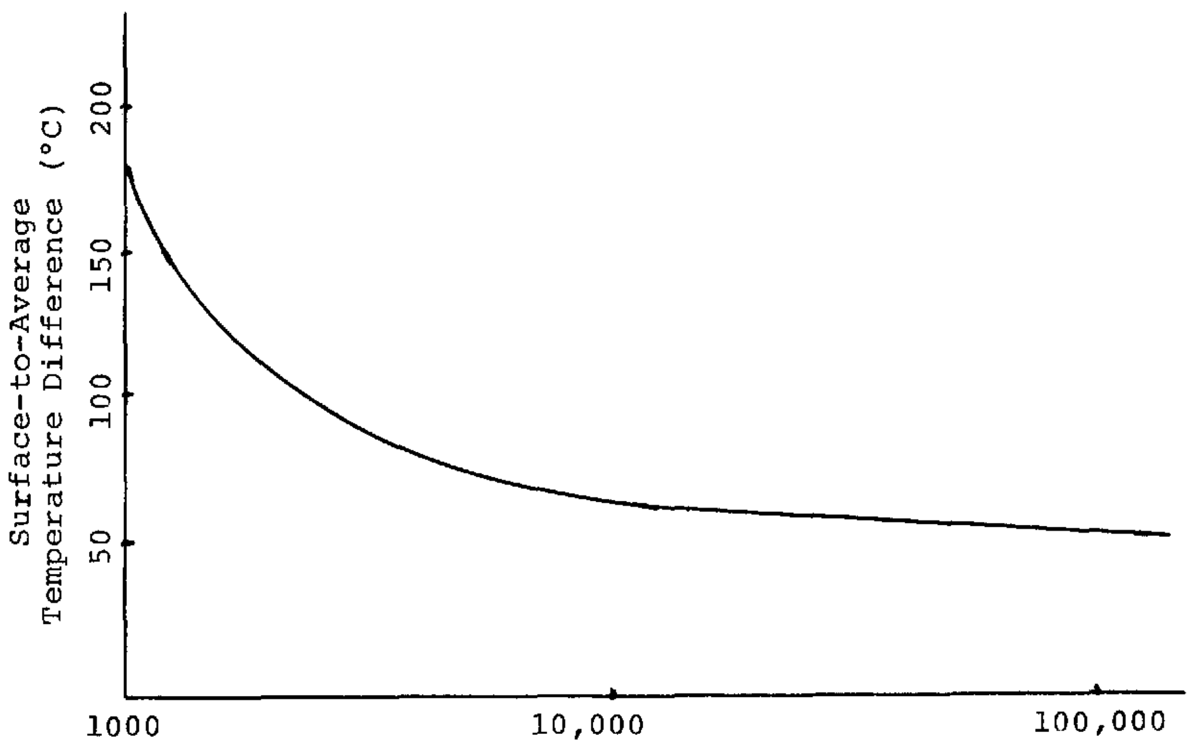
In large conventional and nuclear steam turbines the structural components, particularly the rotor, are subjected to extremely high thermal stresses during startup and subsequent loading. These stresses result from differential expansion within the casing and rotor as they are heated to equilibrium operating temperature. Figure 3.4 shows a typical temperature profile across the rotor during startup or loading when the steam temperature rises above the equilibrium temperature of the rotor.



Temperature Profile Across Rotor During Heat-Up

Figure 3.4

The hot, outer surface of the rotor wants to expand relative to the colder shaft centerline. This produces a compressive stress at the surface of the rotor which is constrained from expanding as much as it would like to expand. On the other hand, the inner portion of the rotor is forced to expand more than it would like to and is there fore under tensile stress. As the average rotor temperature rises to equilibrium, the stresses relax. In being forced to expand at other than the desired rate the metal may be stressed (either in tension or compression) beyond the elastic limit and permanent deformation may occur. Even if the elastic limit is not exceeded, repeated thermal cycling may result in fatigue cracking. As the level of stress rises the number of cycles necessary to induce fatigue cracking decreases. Since it is the temperature gradient across the rotor which produces this stress, the higher this gradient, the higher the stress and the lower the number of cycles to failure.



Cycles to Produce Cracking

Figure 3.5

Figure 3.5 shows the effect that increasing the difference between surface temperature and average rotor temperature has on fatigue cracking.

In addition to the formation of rotor surface cracks, the thermal cycling of the rotor bore can result in cracking which may lead to catastrophic failure of the rotor under the combined effects of centrifugal and thermal stresses.

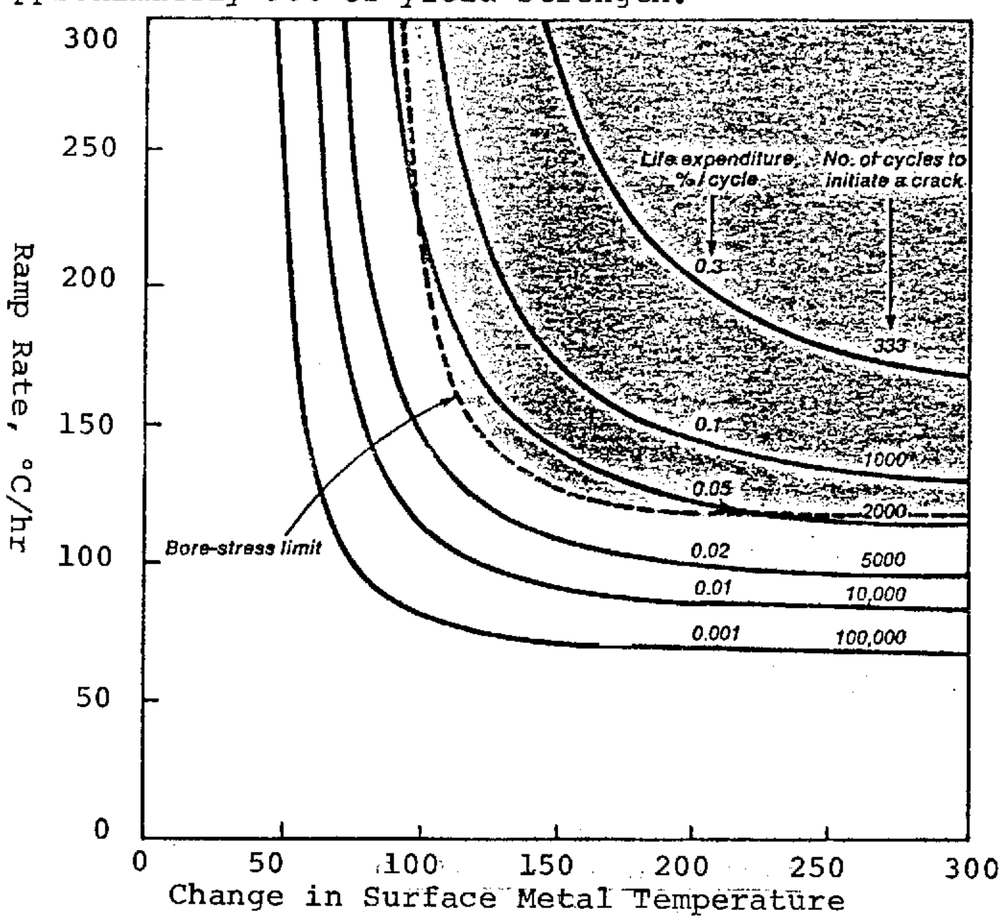
The phenomena of brittle fracture is dealt with in some detail in the 128 materials course. As it relates to the turbine rotor, catastrophic failure (bursting) of the rotor requires three preconditions:

(a) a tensile stress,

(b) a temperature of the metal below the nil-ductility transition temperature (NDTT), and

(c) a crack of critical length to initiate a brittle crack.

The bore stress limit imposed by designers is based on limiting the rotor bore stress to a level such that a crack of a size that could be conceivably missed during non-destructive testing would not grow to a critical size during the rotor lifetime. It has been verified that to prevent such a crack from growing to critical size in a reasonable number of cycles, the peak bore stress (centrifugal plus thermal) must be limited to approximately 90% of yield strength.



Cycle-Life Expenditure

Figure 3.6

Figure 3.6 shows a typical cycle-life-expenditure curve for a large turbine. Although the curves vary from unit to unit depending on size, geometry and material properties the general shape is uniform.

It should be noted how significantly the number of cycles to failure is reduced for only moderate increases in the heat-up rate. For a 200°C heatup, it requires 100,000 cycles to initiate cracking with a 70°C/hr heatup rate. If the heatup rate is increased to 115°C/hr, it requires only 2,000 cycles to initiate cracking.

The area which lies to the right of the Bore-stress Limit should be avoided as the thermal stresses are sufficiently high that when they are combined with centrifugal stresses, catastrophic failure of the rotor may result.

ASSIGNMENT

1. Discuss the factors affecting the severity of the following operational problems. Include in your discussion the possible consequences and the design and operational considerations which minimize their frequency or effect.

   (a) overspeed
   (b) motoring
   (c) low condenser vacuum
   (d) water induction
   (e) condenser tube leaks
   (f) blade failure
   (g) expansion bellows failure
   (h) bearing failure
   (i) low cycle fatigue failure

2. What are the advantages of using FRF as a hydraulic fluid for turbine control? What are the precautions which must be exercised?

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Turbine, Generator and Auxiliaries - Course 134

TURBINE START-UP

This lesson will discuss the general operation of a steam turbine unit during start-up from a cold condition. It should be appreciated that no two turbines have identical operating characteristics, even when they are of identical design. This is due to differences in material, clearances, auxiliary systems, age and general condition. Thus while it is possible to outline a starting procedure in general terms, this must be adapted in detail to suit the particular machine under consideration. In all cases, the manufacturer's instructions should be followed with variations introduced as a result of experience where necessary.

Figure 4.1 shows graphically the major steps required for the start-up of a large turbine unit.

CHECK STATE OF PLANT

The plant should be checked both visually and administratively to ensure that all system maintenance is complete and the plant is physically ready for start-up. During the last few days prior to a start-up, the operator should continually review the status of systems to ensure no surprises are encountered when each system needs to be started up. A review should be made of all tests which will be called up for this start-up and necessary equipment and personnel made available. Following system maintenance, the actual position of valves should be checked against the flow diagram. This valve lineup verification may involve only a small portion of a single system or, in the case of an extended shutdown, all plant systems. The fact that a valve is supposed to be open and is shown as open on a flow diagram does not mean the valve is open.

SYSTEM START-UP PRIOR TO PLACING TURBINE ON TURNING GEAR

Prior to placing the unit on the turning gear the following auxiliary systems will be placed in operation:

(1) all electrical supplies
(2) low pressure service water
(3) low pressure instrument air
(4) high pressure instrument air
(5) condenser cooling water.

The inventory of makeup water should be checked and the water treatment plant placed in service if necessary. The temper-

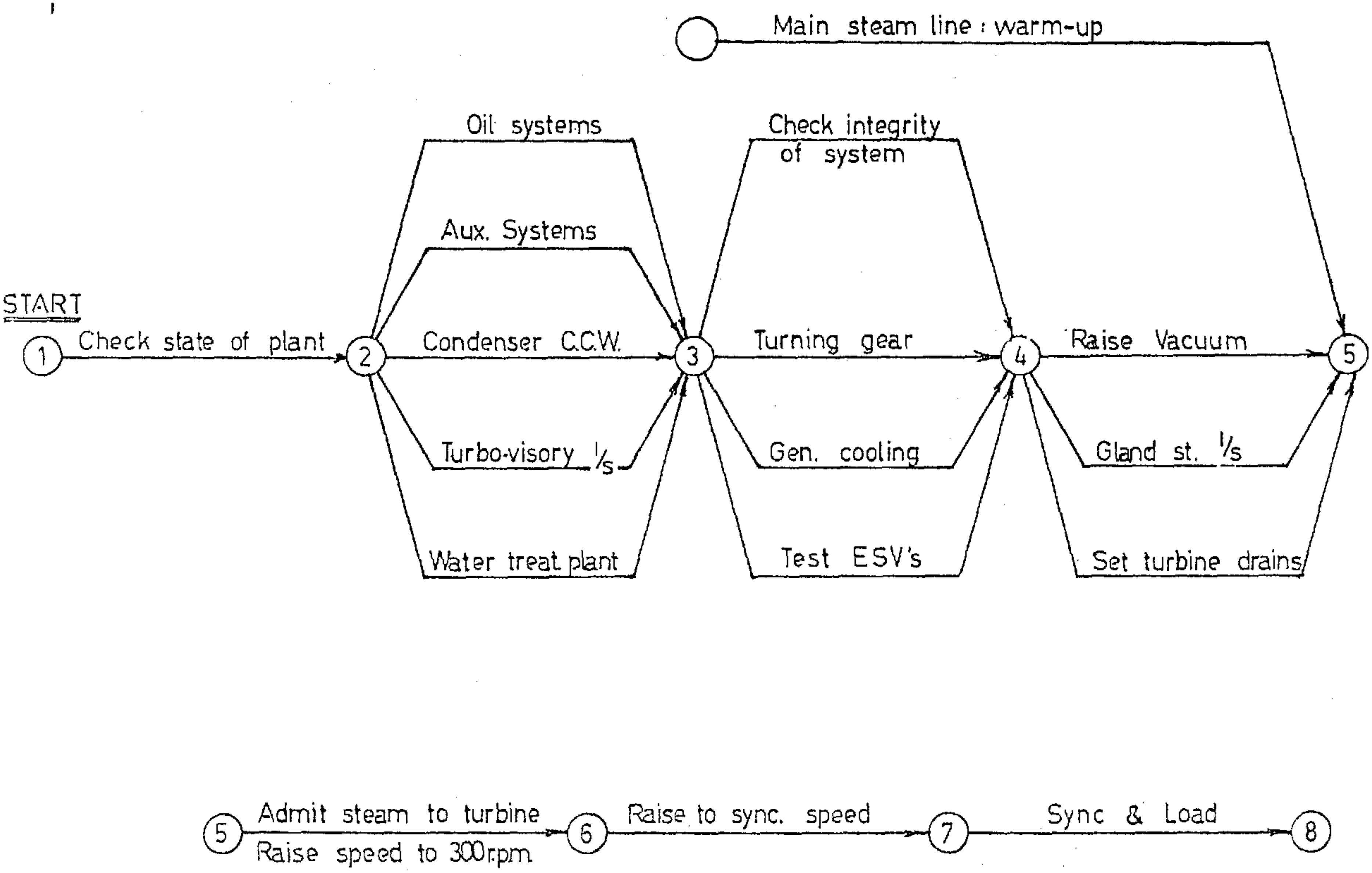


Figure 4.1

Turbine Unit Start-Up

ature of the deaerator should be sufficiently high to allow the addition of water to the steam generators. The warmup of the deaerator takes approximately 36 hours.

The generator seal oil system should already be in operation if the generator is pressurized with hydrogen. However, even if the generator is not pressurized with hydrogen, seal oil must be supplied to the seals prior to placing the unit on the turning gear as the oil lubricates and cools the seals.

The lubricating oil system is started and tested. The turning gear lube oil pump or auxiliary lube oil pump will be supplying oil to the bearings and the proper pressure should be checked. The jacking oil pump is placed in service and its discharge pressure to the bearings is checked. The differential pressure across the lube oil filters should be checked. Following maintenance, the lube oil flow to each bearing should be checked prior to starting the turning gear.

The turbovisory system is energized to provide the following indication:

(a) shaft eccentricity
(b) differential expansion
(c) casing expansion
(d) shaft position
(e) shaft rotational speed
(f) emergency stop valve position
(g) governor steam valve position
(h) bearing vibration
(i) thrust bearing temperature
(j) journal bearing temperatures
(k) lube oil temperatures
(l) generator hydrogen gas pressure and temperature
(m) generator hydrogen purity
(n) generator hydrogen seal oil differential pressure
(o) generator hydrogen-to-stator water differential pressure
(p) generator stator coolant pressure, temperature and conductivity.

TURNING GEAR OPERATION

The turbine should be placed on the turning gear some time prior to steam admission. Twenty-four hours would be typical. This allows time to roll out minor shaft eccentricities, warm up the lube oil and flush any impurities from the lube oil system. Since the lube oil purifier will be operating during turning gear operation, this time will allow for removal of impurities in the oil. During this turning gear operation prior to rolling with steam, shaft eccentricity and lube oil filter and strainer differential pressure should be checked.

The generator cooling system is put in operation and the hydrogen purity and pressure checked. Hydrogen pressure must be maintained above stator cooling water pressure to prevent leakage of water into the generator. The stator cooling system is put in operation and the conductivity of the water checked. Prior to rolling the turbine with steam, service water is supplied to both the hydrogen and stator water coolers.

If it is not already running the auxiliary oil pump would be started to supply oil to the control oil system. In the case of an electrical-hydraulic governing system using FRF, the FRF system would be placed in operation.

At this time the emergency stop valves would be tested to insure they functioned properly. The boiler stop valves will be shut; the governor steam valves will be fully open with the speeder gear wound to the lower stop which corresponds to about 1650 rpm. This insures that during the start-up, the turbine will come on the governor at as low a speed as possible. After the emergency stop valves are tested, they are fully shut. At this point the governing and control systems are ready for operation.

VACUUM RAISING

The reactor is brought to criticality and the temperature of the heat transport system is raised. As this occurs, steam generator and balance header pressure increases. Reactor power is established to increase heat transport system temperature at some rate (usually about 2-3°C per minute). As steam generator temperature and pressure increases, the steam reject valves (atmospheric steam reject vales), Figure 4.2, would open if pressure rose above the pressure determined by the reactor heat up ramp. During this heatup the steam generator and steam header pressure would be determined by the heat transport system temperature as it was raised by reactor power. The main steam header drains would be open to allow condensed water to drain from the lines as the steam lines are brought up to operating temperature.

The condensate system, Figure 4.3, is placed in an operating condition with one condensate extraction pump running, one pump in automatic (to start automatically at 60% power) and one pump in standby. The condensate system is lined up to deliver condensate through the low pressure feedheaters to the deaerator. The level in the deaerator is now being maintained by the level control valves which regulate the flow of condensate

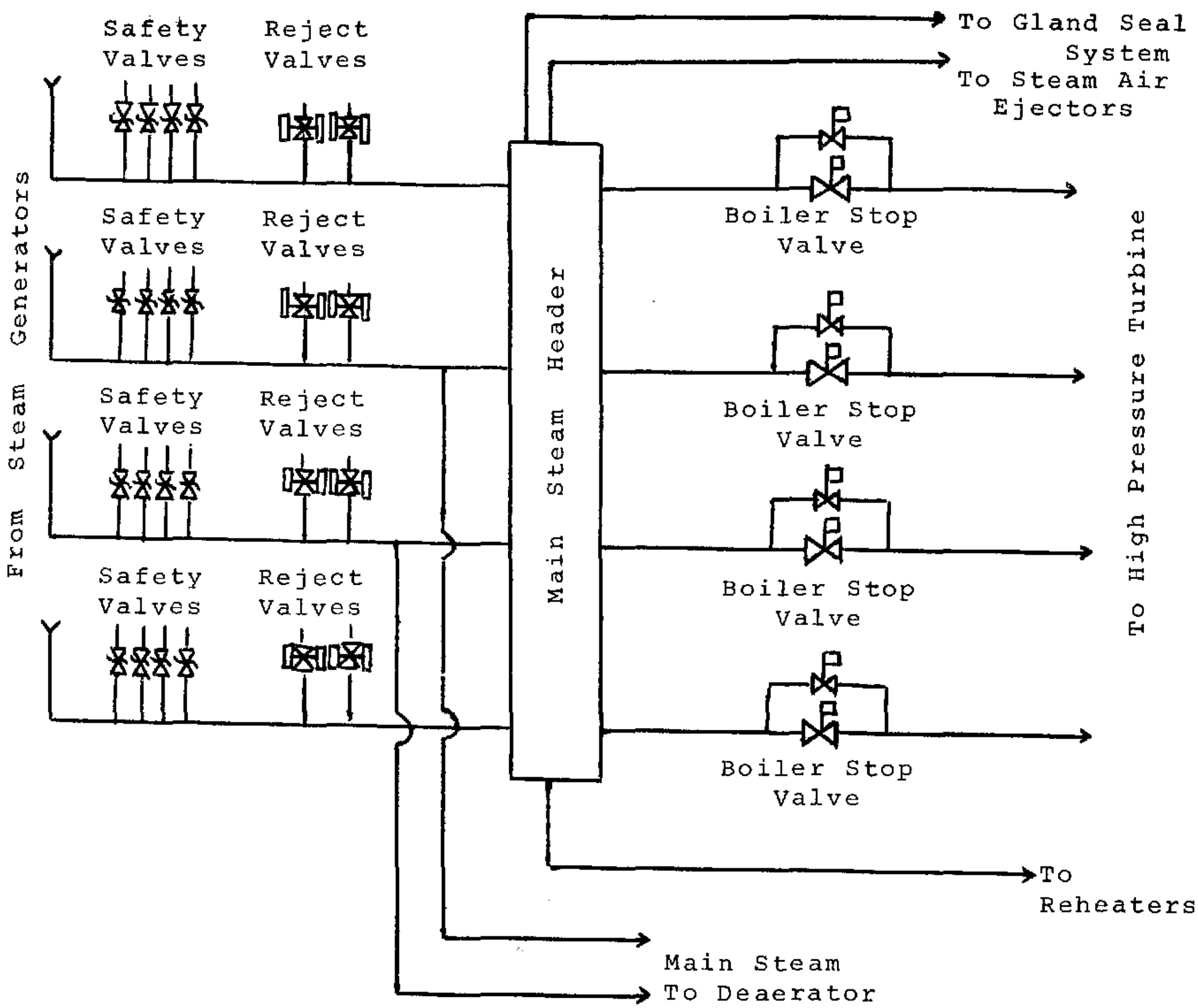


Figure 4.2

Steam System

to the deaerator. When little or no condensate is being passed to the deaerator, the condensate extraction pump re-circulation line opens to allow sufficient flow through the condensate system to:

(1) insure adequate cooling to the gland exhaust condenser and, if applicable, the air ejector condenser (Bruce NGS, DPNGS) and stator water cooling heat exchanger (DPNGS);

(2) insure adequate flow exists through the condensate pump to keep it from overheating.

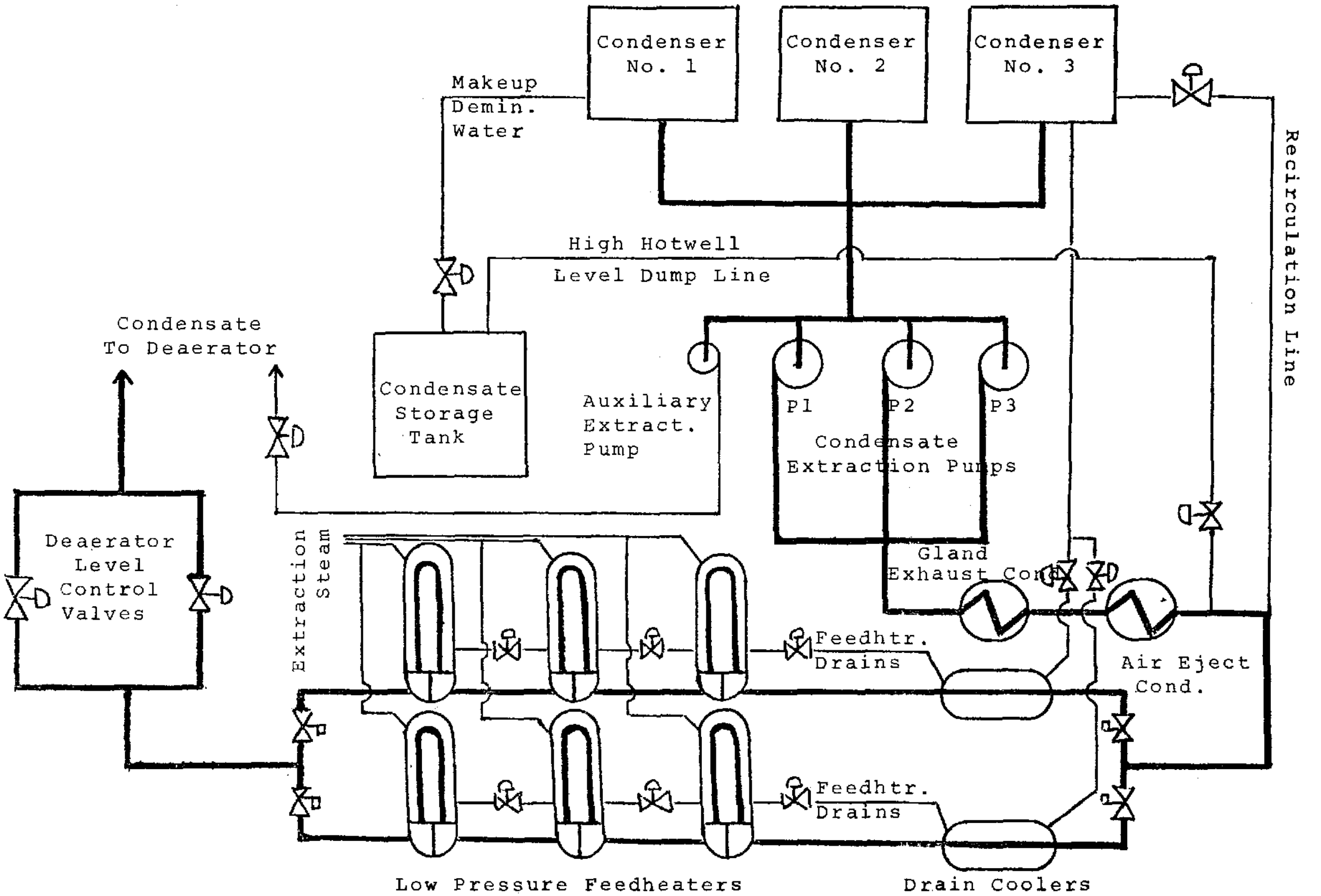


Figure 4.3

Condensate System

When the main steam header steam pressure is sufficiently high, the control valve which emits main steam to the deaerator will open to allow main steam to enter the deaerator for heating and deaerating incoming condensate (this valve will close automatically when extraction steam of sufficient pressure is available from the low pressure turbine). It should be noted that before steam is admitted to the turbine, no extraction steam is available for the low pressure feedheaters and condensate is entering the deaerator at essentially the same temperature it left the condenser hotwell. As part of the start-up of the condensate system the chemical feed system is placed in service.

After the condensate system is in operation, the feed system, Figure 4.4, is started up. The deaerator must be warm enough to prevent thermal shock to the steam generator before a feed pump is started. Typically this means the deaerator must be within 130° to 135°C of the steam generator temperature. One boiler feed pump is started, one pump is in automatic (to start automatically at about 50% power) and one pump is in standby. The feed system is lined up through the low pressure feedheaters to the steam generators. The levels in the steam generators are now being maintained by the feedwater regulating valves which regulate the flow of feedwater to the steam generators. To prevent the boiler feed pumps from overheating at low flows, each individual pump is fitted with a recirculation line.

At power levels below 25-50% of full power, the extraction steam available for the high pressure feedheaters may not be of sufficient temperature to provide any feedheating. For this reason, extraction steam which would normally go to the HP feedheaters is led to the condenser until sufficient power is reached to provide extraction steam heating in the HP feedheaters. The HP feedheaters are normally receiving extraction steam by 50% of full power. Until the turbine is above 25% of full power, the deaerator is providing virtually all of the feedheating.

The main steam pressure is allowed to rise as the heat transport system heats up. At approximately 1000 kPa(g) the gland seal system is placed in operation and the air extraction system is started to evacuate the condenser and turbine unit. The turbine and extraction steam drains are opened and then placed in automatic. By this time the condenser cooling water system, generator seal oil system, stator cooling system and generator hydrogen system should be in fully operating condition. As the gland seal steam heats up the rotor, differential expansion between the turbine rotors and casings is checked.

The main steam isolating valves (boiler stop valves) are opened to raise pressure in the steam lines up to the emergency stop valves. The reheater is placed in an operating condition.

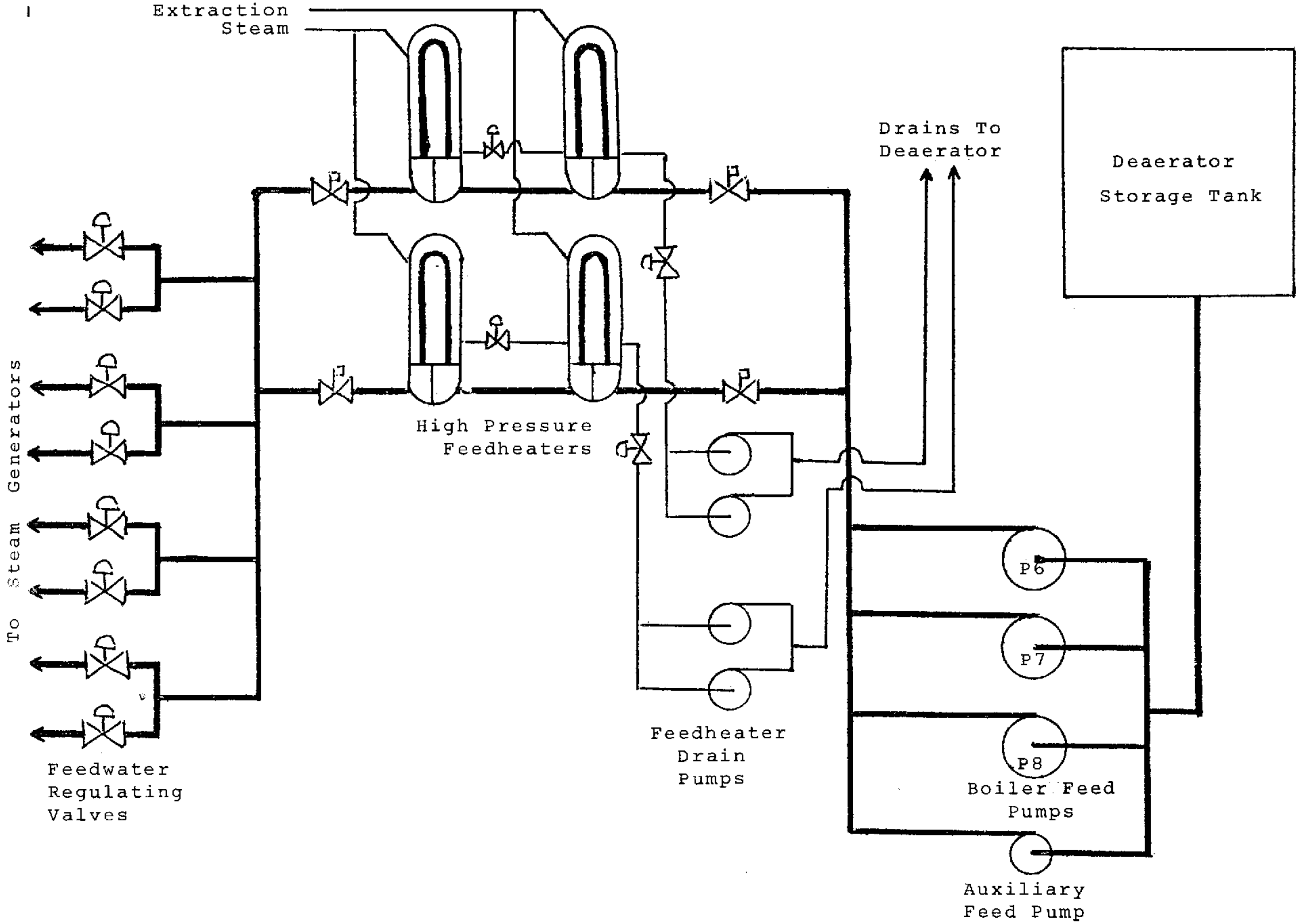


Figure 4.4

Feed System

STEAM ADMISSION TO THE TURBINE

Prior to admitting steam to the turbine the shaft eccentricity is checked within limits. The generator field is energized and the steam pressure at the emergency stop valve is determined to be satisfactory. Provided condenser vacuum is greater than approximately 380 mm mercury [50 kPa(a)], steam is admitted to the turbine by cracking open the emergency stop valves, Figure 4.5. Of the four emergency stop valves, generally only two valves - master ESV's - are used for turbine runup. The other two ESV's - slave ESV's - are not utilized until after the unit has attained operating speed. The slave ESV's will begin to open automatically after the master ESV's are 50% open. The turbine is brought up to 300 rpm. The turning gear will disengage automatically as speed increases above turning gear speed. The turning gear and jacking oil pump will both shut down at some preset speed typically between 100 and 250 rpm. Turbovisory parameters are checked within the limits of Figure 4.6.and speed is either held constant or the unit tripped depending on the values.

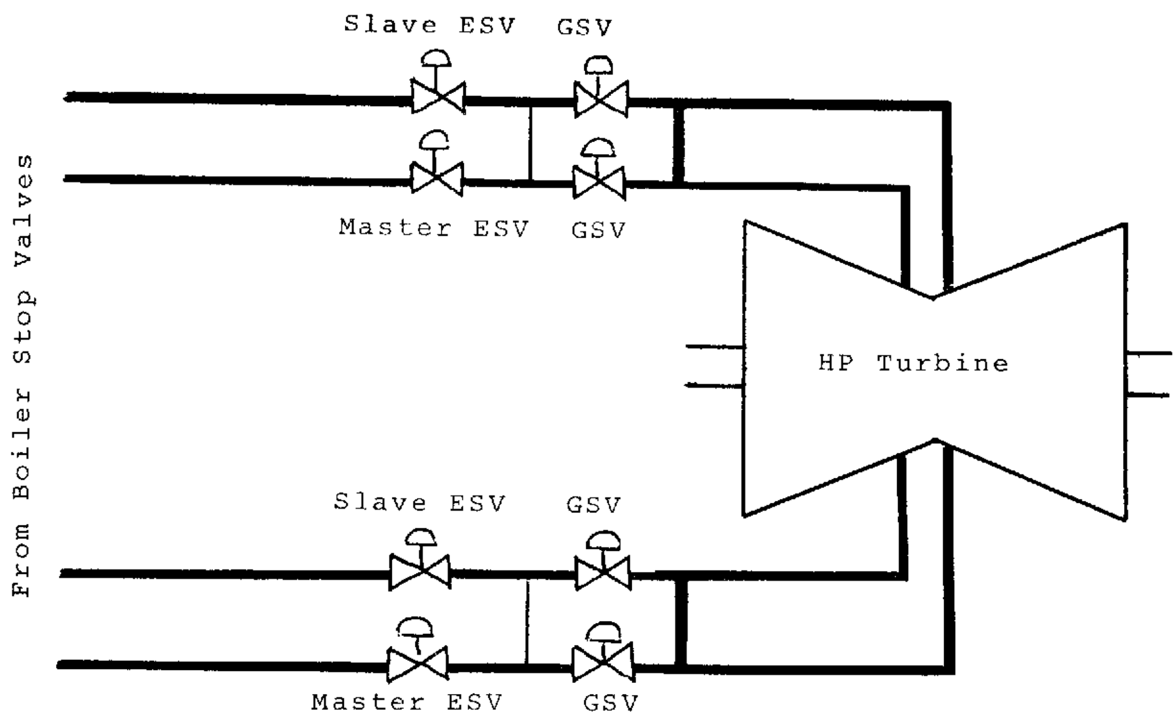


Figure 4.5

Steam Control

Parameter	Parameter Value	Action
Bearing Vibrations	A = 0.003" B = 0.002"	TRIP at any speed. If speed >300 <600 reduce speed and hold at 300 rpm.
H.P. Differential Expansion	>+0.100" <-0.100" >+0.050" <-0.050"	TRIP HOLD
L.P. 1 Differential Expansion	>+0.150" <-0.300" >+0.100" <-0.250"	TRIP HOLD
L.P. 2 Differential Expansion	>+0.150" <-0.600" >+0.100" <-0.050"	TRIP HOLD
L.P. 3 Differential Expansion	>+0.150" <-0.900" >+0.100" <-0.850"	TRIP HOLD
H.P. Shaft Eccentricity	0.008" 0.006"	TRIP HOLD
Lub. oil temp. Gen. rotor temp.	<39°C <20°C	Reduce & hold speed at 1200 rpm Reduce & hold speed at 1200 rpm

Figure 4.6

Turbovisory Parameters For Start-Up

If condenser pressure is below 33 kPa(a) and turbovisory parameters within specified limits, speed is increased to 600 rmp. Steam-to-metal differential temperature is checked to insure steam temperature remains above metal temperature, but that metal temperature does not increase more than about 250°C/ hour. In addition, the trend of turbovisory parameters is checked. At this point the unit speed is held constant, reduced, or the unit tripped depending on the value and the trend of these parameters. The speed would be held constant at 600 rpm until:

(a) all turbovisory parameters are below the HOLD value and have an acceptable trend, and

(b) condenser pressure is below 18 kPa(a).

Speed would then be increased to 900 rpm. This speed is held until lube oil is above 39°C, generator rotor temperature is above 20°C, condenser pressure is below 9 kPa(a), and turbovisory parameters are acceptable. Speed is then increased to 1200 rpm and conditions checked again. At this point, the low condenser vacuum trip is placed in operation. (Up until this point the low vacuum trip has been disabled to allow opening of the ESV's at below the minimum acceptable operating vacuum.)

Turbine speed is now ready to be increased through the critical speed range of the turbine and generator. In this speed range (typically 1225 to 1400 for the turbine; 1550 to 1650 for the generator) the rotational frequency of the unit, matches the natural frequency of the rotor. Prolonged operation in this region could destroy the turbine and/or generator through amplification of vibration. For this reason the speed is brought up rapidly through the critical speed range to about 1700 rpm. If the turbovisory parameters develop HOLD values while in the critical speed range, the speed is lowered to 1200 rpm. By the time speed reaches 1700 rpm the auxiliary oil pump should have shut down (main oil pump providing sufficient discharge pressure) and the governor should have taken control of the turbine. The ESV's can now be fully opened.

SYNCHRONIZING AND LOADING

The turbine speed is now raised to 1800 rpm with the speeder gear controlling the position of the governor steam valves. Excitation is adjusted to provide a terminal voltage equal to grid voltage and the generator is synchronized with the grid. The unit is now ready for loading.

Up until the time that loading of the generator commences the steam reject valves (atmospheric steam reject valves) have been controlling steam generator pressure. When loading of the generator is to begin, steam generator pressure control is transferred to the governor speeder gear. In this mode of pressure control, the speeder gear is under computer control and will open the governor steam valves to reduce steam generator pressure to the desired setpoint. Loading is thereby accomplished by raising reactor power which in turn raises heat transport system temperature and steam generator pressure. This causes the speeder gear to open the governor steam valves and thereby return steam generator pressure to the desired setpoint. If the turbine is unable to accept the additional steam (HOLD turbovisory parameter, exceeds maximum loading rate, etc), the reject system will operate to maintain the desired steam pressure.

The value and trend of turbovisory parameters are checked within limits as the turbine increases load and warms up to operating temperatures. By this time the turbine and extraction steam drains should be shut. The maximum rates of loading specified in Figure 4.7 are followed.

As turbine/generator power increases the operator will check to ensure that:

(1) the deaerator transfers from main steam to extraction steam

(2) the extraction steam flow to the HP feedheaters is established

(3) AUTO feed pump starts

(4) AUTO condensate extraction pump starts.

The unit is now in an operating state and loading would continue to 100% power or whatever value is desired.

ASSIGNMENT

1. Using Figure 4.1 as a guide, discuss the major steps involved in starting up a large turbine unit.

R.O. Schuelke

TYPE OF START	COLD TURBINE	WARM TURBINE		HOT TURBINE	
Period of shut down Hrs.	-	36	12	6	1
Estimated metal temperature °C	21	116	188	216	243
Maximum Condenser Back Pressure kPa(a)	50	30	13.5	8.5	6.5
Time from turning gear to full speed Mins.	40	20	10	5	3
Block load on synchronizing. MW	-	-	50	100	140
Load 0-50 MW @	2½ MW/MIN.	5 MW/MIN.	-	-	-
50-150 MW @	10 MW/MIN.	12½ MW/MIN.	20 MW/MIN.	50 MW/MIN.	100 MW/MIN
150-300 MW @	15 MW/MIN.	20 MW/MIN.	25 MW/MIN.	50 MW/MIN.	100 MW/MIN
300-540 MW @	15 MW/MIN.	20 MW/MIN.	35 MW/MIN.	35 MW/MIN.	100 MW/MIN
Time from synchronizing to full load. Mins.	56	32	18	11	4

Figure 4.7

Rates Of Loading

Turbine, Generators and Auxiliaries - Course 134

FACTORS LIMITING STARTUP AND RATES OF LOADING

Modern large steam turbines are exceedingly complex machines. Since they require a large capital investment the basic reliability of the unit is of considerable importance. In addition, the cost of alternate power sources makes the reliability of a nuclear steam turbine/generator particularly important.

Turbine/generator units are designed so that when they are installed, operated and maintained properly they will provide reliable power for essentially the life of the station without serious difficulties. Once a unit is running in a steady state condition, there is relatively little chance that a major problem due to maloperation will occur. However, during startup, warmup or load changes the conditions imposed on the unit are much more severe. Under these transient conditions a great potential exists for considerably shortening the useful life of the turbine and generator. When one considers the infrequency of startups on large nuclear steam turbines the time saved by a less than optimum warmup is insignificant when compared to the potential for long and short term problems.

The major factors which limit the rate at which a large turbine can be started up and loaded fall into the following categories:

(a) low cycle fatigue damage,
(b) high stress in turbine rotor,
(c) low turbine or generator rotor toughness at low temperatures,
(d) excessive vibration,
(e) water induction,
(f) excessive axial expansion,
(g) high stresses in turbine blading, and
(h) low oil temperature.

The avoidance of these conditions are the major determiners of startup procedure and failure to appreciate the consequences of these conditions can lead to premature turbine/generator aging and failure.

LOW CYCLE FATIGUE DAMAGE AND HIGH ROTOR STRESS

Steam temperature changes associated with warmup and loading of a turbine unit can impose significant temperature gradients across the rotor, casing and associated steam piping. The greater the average metal-to-steam temperature differential, the greater will be the imposed thermal stresses. The phenomenon of fatigue cracking is dependent on the magnitude of the peak stress imposed: as the peak stress increases, the number of cycles to produce cracking decreases. This is shown graphically in Figure 5.1. In addition, if the stress imposed is great enough the yield strength of the metal will be exceeded and permanent plastic deformation will occur. The effect of an excessive temperature differential between the average metal temperature and the steam is shown in Figure 5.2. Although the yield point can be exceeded in any turbine component, the combined effects of centrifugal stress and thermal stress make the rotor the most critical component. Exceeding the yield strength of the rotor metal at the bore can have immediate and disasterous effects on the unit since the rotor may well rupture.

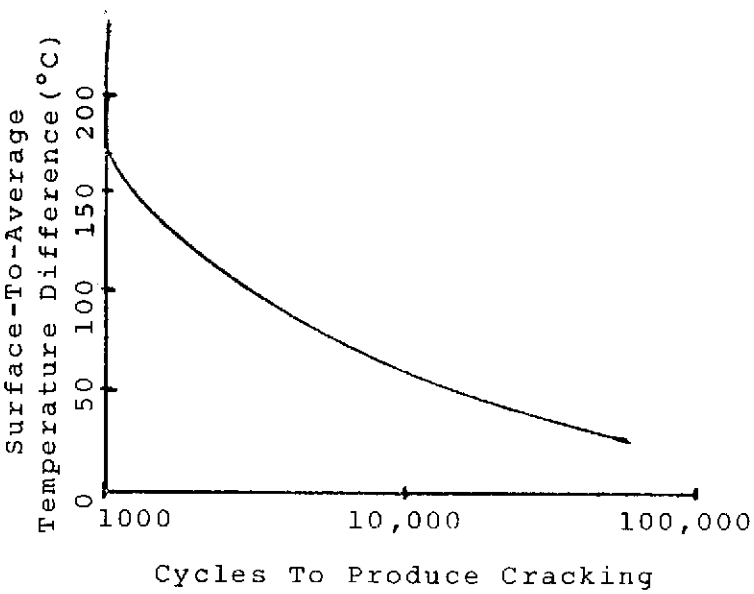


Figure 5.1

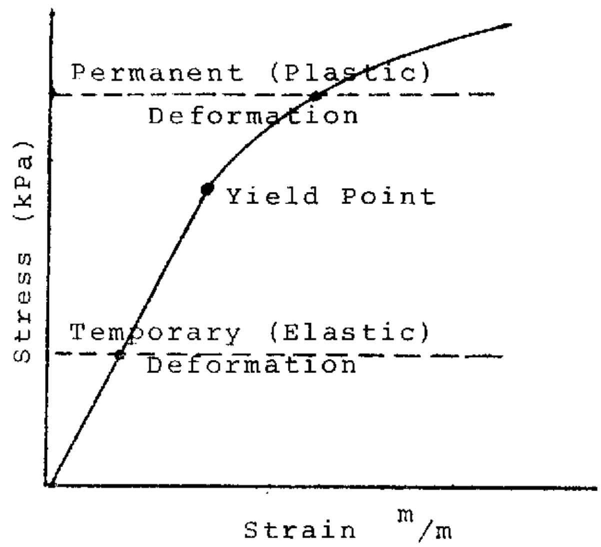
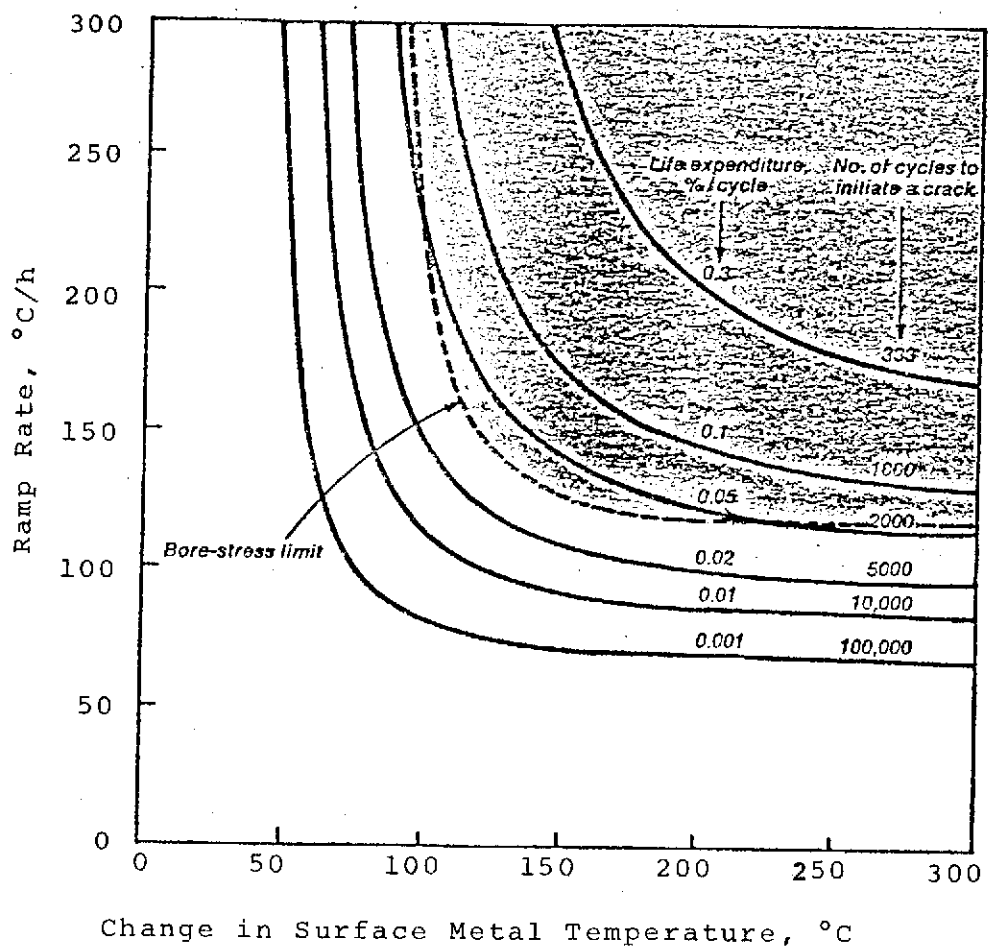


Figure 5.2

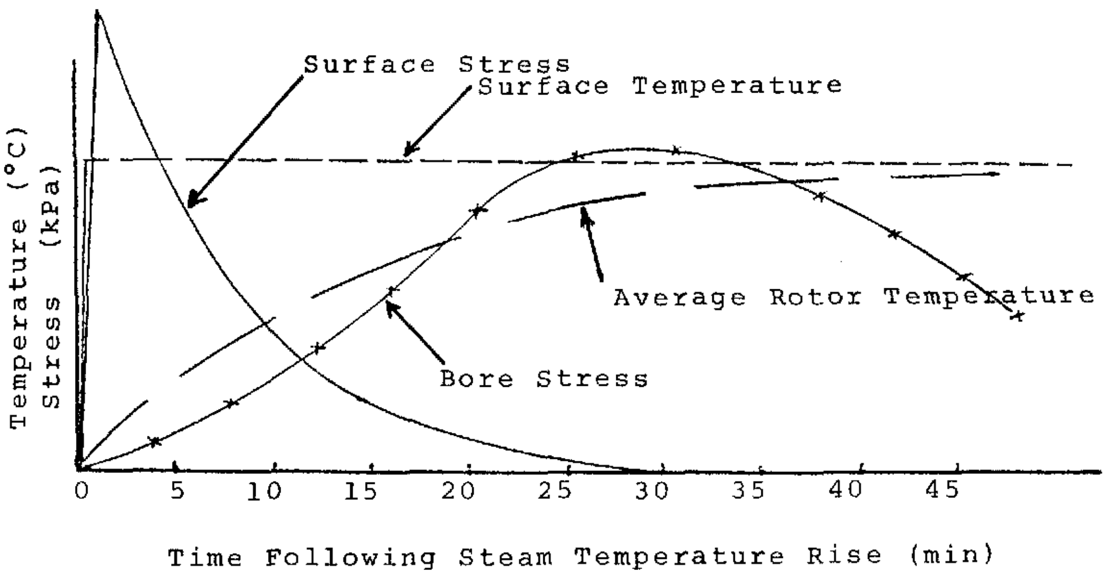
Figure 5.3 shows the effect which thermal cycles have on turbine life. Since the effects of thermal cycling are cumulative, the life expenditure per cycle is additive. For a 200°C warmup, a heatup rate of 100°C per hour (life expenditure of .02% per cycle) is the equivalent of 20 startups with a heatup rate of 70°C per hour (life expenditure of .001% per cycle).



Cycle-Life Expenditure

Figure 5.3

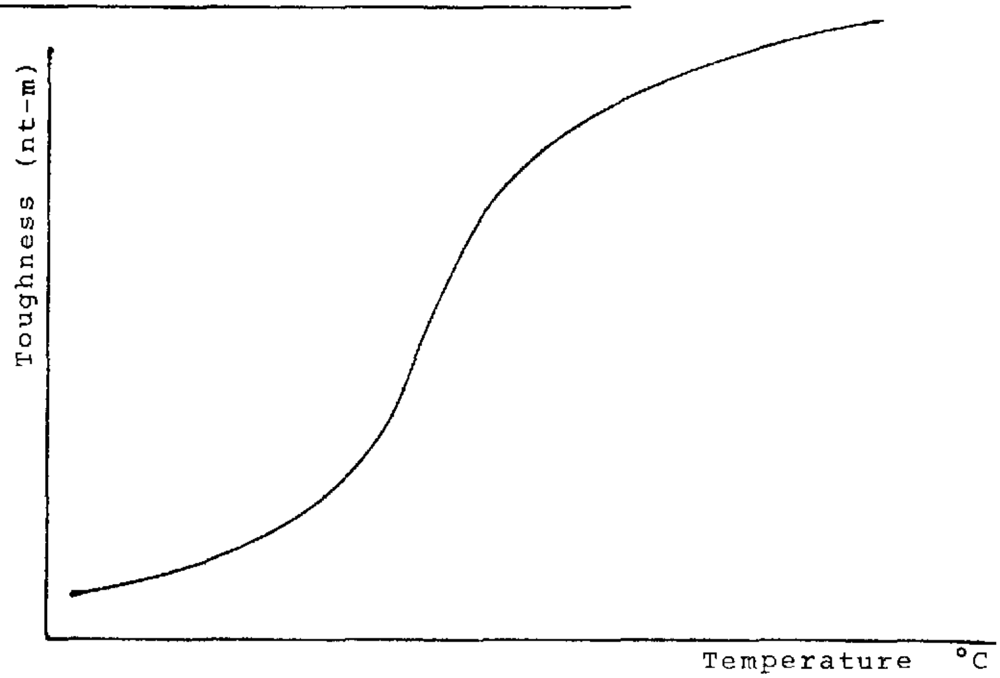
The maximum thermal stress at the bore of the rotor depends on the difference between the average rotor temperature and the bore temperature. On a load change the average rotor temperature lags the steam temperature as the rotor warms up. Thus the maximum bore stress is dependent on the rate at which the rotor heats up (average rotor metal temperature) and may not reach a maximum until 20 to 30 minutes following the load change. Figure 5.4 shows this effect. While the peak surface stress (surface fatigue cracking) occurs almost immediately, the peak bore stress is delayed. The conclusion is obvious: the effect of an excessive heatup rate does not immediately disappear but persists for some time. If the recommended heatup rate is exceeded, an additional soak must be imposed to allow the bore stress to dissipate.



Effect Of Temperature Change

Figure 5.4

LOW ROTOR TOUGHNESS AT LOW TEMPERATURES



Effect Of Metal Temperature On Toughness

Figure 5.5

Figure 5.5 shows the effect which temperature has on metal toughness (toughness is the ability of a metal to absorb energy in both elastic and plastic deformation). The warmer a metal is the less brittle it becomes and the greater the stress it can withstand without cracking. At low temperatures most metals will fail with very little stress in a sudden and catastrophic manner (brittle failure). Because of this phenomenon it is necessary to insure that the turbine and generator rotors are warm before imposing a high stress on them. The temperature at which toughness of a metal rapidly increases is known as the nil-ductility transition temperature because below this temperature the metal exhibits "nil-ductility". Typically the nil-ductility transition temperature for a saturated steam HP turbine rotor is in the range of 50° to 100°C and for generator rotors in the range of 10° to 30°C.

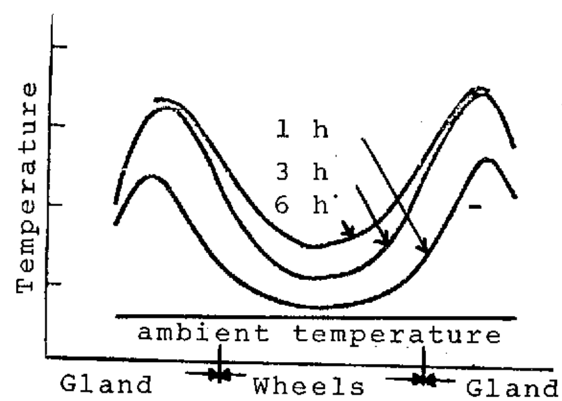
There are several methods of increasing the rotor temperature prior to imposing the high rotor stresses associated with high speeds and loads:

(a) generators normally have electric heaters installed;

(b) generator fields can be energized to provide I^2R heating of the rotor;

(c) turbines may be run at low speed and low steam flow for specified soak times to allow the rotor to come up in temperature while still at low rotor stress levels;

(d) the gland seal system can be operated for several hours while on the turning gear prior to steam admission to the turbine. As shown in Figure 5.6, this raises rotor temperatures. This increase in rotor temperature not only makes the rotor more resistant to cracking but aids in rolling out minor shaft eccentricities on the turning gear.



Effect Of Steel Steam Operation On Rotor Temperature

Figure 5.6

(e) the HP turbine casing can be pressurized with steam at 400-500 kPa(g) for several hours of turning gear operation. This procedure has been recommended by at least one turbine manufacturer (General Electric) as a method of raising HP rotor temperatures. In this method, the intercept valves are shut and steam is admitted to the HP turbine to pressurize it. In a four hour period the steam can raise the average rotor temperature above 150°C. This essentially turns a "cold" startup into a "warm" startup.

The factors of low cycle fatigue failure, rotor bore stresses and rotor transition temperature are the major determiners of the rate at which a turbine can be brought up to operating temperature (heatup rate) and load (loading rate).

The factors which must be considered in assessing the impact of a startup on the health of the turbine are:

(a) the initial metal temperature;

(b) the magnitude of the change between initial and final metal temperature;

(c) the rate of change of metal temperature, and

(d) the mechanical and centrifugal stress (as distinguished from thermal stress) in the rotor material.

Generally a heatup of short duration, at a low heatup rate, on an already hot turbine with low rotor mechanical stresses results in the minimum effect on the turbine. If one of these factors is less than optimum, the others will have to be carefully controlled to prevent unnecessarily shortening turbine life.

In assigning maximum rates of heatup and loading, the initial condition of the turbine is divided into three basic categories COLD, WARM and HOT depending on how long the unit has been shut down. As shown in Figure 5.7 the hotter the turbine, the shorter the time to full load and the greater the allowable loading rate.

Because the loading rate is dependent upon initial temperature, there is a necessity to insure the metal temperature does not decrease after loading commences. This could occur if the unit were brought on line at a very low load and the subsequent loading carried out at a low rate. In this situation the turbine could well cool down as the heat input at low load is less than the losses to ambient temperature from a hot

TYPE OF START	COLD TURBINE	WARM TURBINE		HOT TURBINE	
Period of shut down Hrs.	-	36	12	6	1
Estimated metal temperature °C	21	116	188	216	243
Maximum Condenser Back Pressure kPa(a)	50	30	13.5	8.5	6.5
Time from turning gear to full speed Mins.	40	20	10	5	3
Block load on synchronizing. MW	-	-	50	100	140
Load 0-50 MW @	2½ MW/MIN.	5 MW/MIN.	-	-	-
50-150 MW @	10 MW/MIN.	12½ MW/MIN.	20 MW/MIN.	50 MW/MIN.	100 MW/MIN
150-300 MW @	15 MW/MIN.	20 MW/MIN.	25 MW/MIN.	50 MW/MIN.	100 MW/MIN
300-540 MW @	15 MW/MIN.	20 MW/MIN.	35 MW/MIN.	35 MW/MIN.	100 MW/MIN
Time from synchronizing to full load. Mins.	56	32	18	11	4

Rates of Loading

Figure 5.7

turbine. To prevent this from occurring, a "block load" is specified for a warm or hot turbine. This block load which is brought on the generator at the time of synchronizing draws sufficient steam to keep the unit at the pre-startup temperature.

VIBRATION

The causes of vibration are legion but the effects are usually divisible into four general categories:

(a) fatigue failure due to cyclical loading,
(b) rubbing damage due to component travel beyond design limits,
(c) impact damage due to the pounding effect of bad vibration, and
(d) noise.

The general subject of vibration is treated in detail in the mechanics courses and will be treated here only as it relates to turbine runup.

As can be seen in Figure 5.8, the effect of vibration is generally dependent on the amplitude of the vibration and the frequency of vibration. The higher the speed, the lower the amplitude required to produce unacceptable vibration. With very few exceptions the causes of vibration only intensify as speed increases. There simply is no truth to the popular belief that vibration can be "smoothed out" by raising speed to a high enough value.

Because vibration is the end result of so many turbine/generator problems, excessive vibration is the most obvious warning of a poor runup technique. Figure 5.9 shows some of the major causes of excessive vibration in turbines, the characteristic frequency and probable solutions.

Permanent imbalance to a turbine which has previously run smoothly is generally caused by shaft eccentricity (sag or hog) or loss of material from the rotor (failed blading, shrouds or lacing wire). A rather insignificant change in the center of mass away from the center of rotation can develop large forces. The force created by a one kilogram imbalance located one meter from the center of rotation of a 1800 rpm turbine rotor would be on the order of 10% of the weight of the rotor. The shaft eccentricity necessary to produce sufficient vibration to rapidly destroy the turbine is only on the order of 1-2 mm.

Proper operation of the turning gear on shutdown and while shut down will prevent most eccentricity problems. In addition, there is a limited ability to "roll out" minor eccentricities by operating the shaft on the turning gear for some time prior

GENERAL MACHINERY VIBRATION SEVERITY CHART

For use as a GUIDE in judging vibration as a warning of impending trouble.

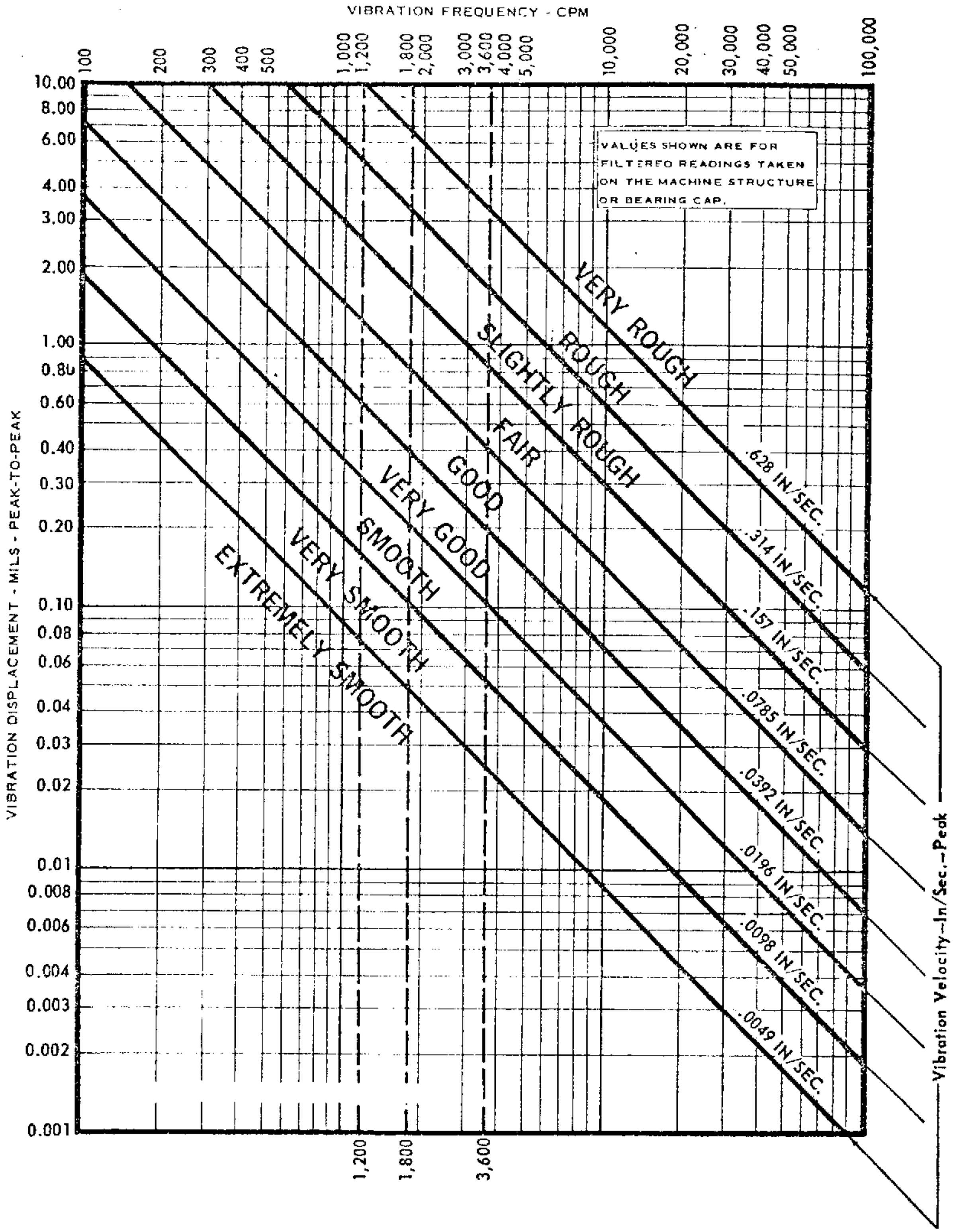


Figure 5.8

Figure 5.9

Possible Causes Of Turbine Vibration

Cause Of Vibration	Identifying Frequency	Possible Solution
Permanent Unbalance	Running Speed Frequency	Repair or Balance Rotor
Temporary Unbalance	Running Speed Frequency	Rotor balancing may be necessary; however, other causes of vibration such as component distortion and rubbing may also require correction.
Impacting	Subharmonic: 1/2, 1/3, or 1/4 of running speed frequency.	Eliminate contact between fixed and moving parts.
Oil Whip	Less than 1/2 of running speed frequency.	Improve bearing parameters (1) increase loading, (2) increase temperatures, (3) design new bearing.
Out of Round Journals	Harmonic: 2, 3 or 4 times running speed frequency.	Machine Journals Round
Rocking Journal Bearings	Harmonic: 2, 3 or 4 times running speed frequency.	Prevent rocking by better fixing of bearing.

to steam admission. The possibility of inducing shaft bending is particularly good if the shaft is left stationary with gland seal steam applied. This results in uneven heating and thermal expansion which can induce permanent distortion.

While severe bending can occasionally be rolled out it is usually uncertain and always time consuming. Failing this, the rotor would have to be sent to the manufacturer for straightening.

Vibration caused by temporary imbalance of the rotor is normally due to some form of thermal distortion. The most frequent causes of this type of uneven heating are:

(a) gland rubbing which produces localized hot spots,

(b) water quenching of the shaft in the area of water glands,

(c) water entry into the turbine.

This latter cause can produce particularly severe vibration if the source of water is in only one of the steam admission lines to the turbine. This can result in a chilling of only part of the rotor and produces an imbalance due to differential contraction of the rotor.

Contact between fixed and moving parts can cause two types of vibration:

(a) Direct contact vibration. Each time the fixed and moving parts contact each other a displacement occurs, and

(b) by acting as a force on the rotor which excites the rotor to vibrate.

This second type of vibration which is similar to striking a rotating piano string sets up a vibration in the rotor which beats against the fundamental frequency of rotation, alternately constructively and destructively interfering with the fundamental frequency. The net effect on the stationary parts of the turbine (what we detect as vibration) is a sum of the natural frequency of the shaft and the rotational frequency of the turbine. While the net frequency of vibration may be at almost any subharmonic of the rotational frequency (1/2, 1/3, 1/4, etc) the usual dominant frequency is at 1/2 the rotational frequency.

These subharmonic vibrations do exist and apart from contact between fixed and moving parts there are a number of causes which have been theorized including oil whip, non-uniform radial blade clearances and non-uniform radial bearing clerances. It should be borne in mind that the cause of excitation forces listed above are only theories and have not been proven to be the only forces present.

Vibration can increase significantly as the turbine speed coincides with the shaft "critical speed". The critical speed of a shaft is that speed at which the shaft is most sensitive to bending or deflecting. This means the internal damping of the shaft is at a minimum as shown in Figure 5.10.

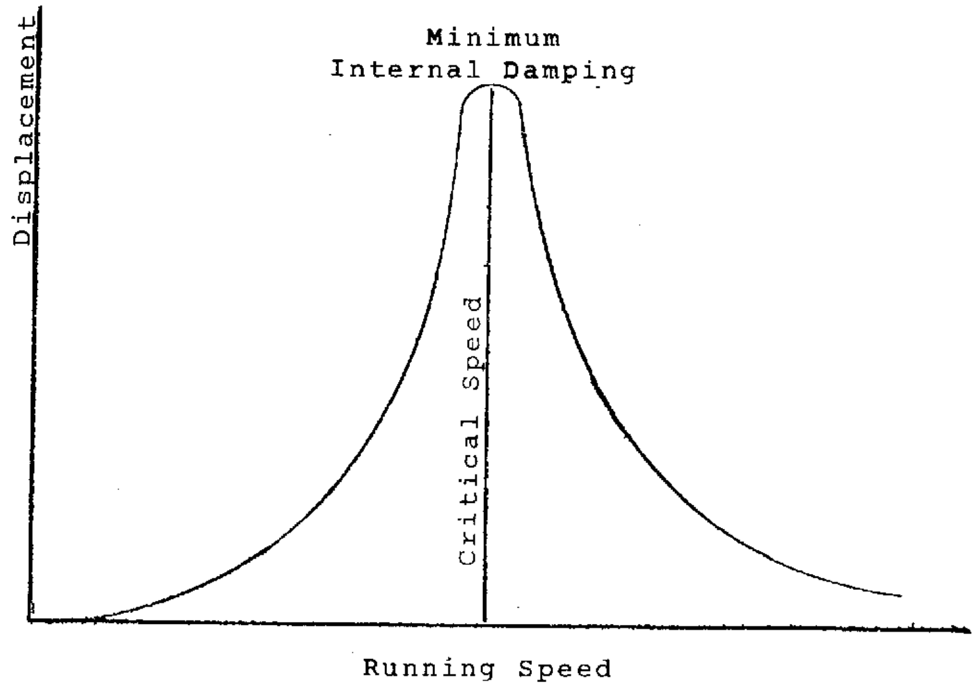


Figure 5.10

It so happens that this critical speed or frequency coincides with the frequency at which the shaft would oscillate if struck. At speeds below 70% of the first critical speed the shaft can be considered as rigid while at speeds above 70% of the first critical speed the shaft is flexible. When the rotational frequency of the turbine coincides with the natural frequency of the shaft (critical speed) the shaft vibrates sympathetically. The greater the vibration, the greater the excitation of the shaft. This resonant condition can rapidly increase the amplitude of vibration until the machine is seriously damaged. For this reason, the turbine should be carefully checked for abnormal vibration prior to passing through the critical speed. In addition, the machine should be accelerated through the critical speed zone as expeditiously as possible.

WATER INDUCTION

The possibility of water induction into the turbine is particularly great during startup. The steam which has condensed in the steam lines and turbine must be removed before any appreciable quantity of steam is directed to the unit. Not only can inadequate draining result in direct impact damage on the turbine blading but steam and water hammering in main and extraction steam lines can cause significant shock loading of piping and valves.

Water lying in steam piping may only move after a sufficient steam flow is established. In saturated steam systems the ability of the steam to "absorb" moisture is rather limited and water left in steam lines during startup may remain there for a considerable period of time until the steam flow is sufficient to move the water. These "water slugs" can cause considerable damage in the turbine. Even if they don't reach the turbine the impact at piping bends or valves can be great enough to deform or even rupture the piping.

Excessive moisture can also enter the turbine through improper steam generator level which decreases moisture separation efficiency. At low steam flows the steam generator level control system tends to be less sensitive to changes than at high flows. Even if the level control system is capable of maintaining level at extremely low flows, the ability to handle rapid steam flow transients is usually diminished. In some plants, the steam generator level control system receives no input from steam flow or feed flow at low power levels and, therefore, functions as a single element controller on steam generator level. For better or worse this makes the system less responsive to changes. For this reason, steam generator levels must be carefully watched while changing load and particularly at low power levels.

Leakage of feedwater regulating valves at low steam flows can make it extremely difficult to maintain boiler level. At low power levels if the leakage through a feedwater regulating valve exceeds the steam flow out of the associated steam generators, the level will increase. This problem may not be apparent while operating as the operating steam flow will be well in excess of even fairly troublesome low power leakage.

Another condition which can lead to high steam generator levels and possible moisture carryover is the effect of adjusting the "level set point" of the steam generator level control system. The level set point is the level which the controller will maintain at minimum power and represents the set point upon which the steam generator level control system determines the desired steam generator level. Basically the level controller adds to level set a gain based on steam generator power to obtained desired level (the level the controller tries to maintain)

$$\begin{matrix}\text{Desired}\\ \text{Level}\end{matrix}\ (\text{meters}) = \begin{matrix}\text{Level}\\ \text{Set}\end{matrix}\ (\text{meters}) + \text{Gain}\left(\frac{\text{meters}}{\%\ \text{power}}\right)\left(\%\ \text{power}\right)$$

If level set is adjusted high at low power levels the level which the controller seeks to achieve at high power level will be above the design value and may exceed the alarm set point. If the level set is adjusted away from the design value, the operator must be careful to compensate for this on power changes or he may find himself with abnormally high or low steam generator levels.

AXIAL DIFFERENTIAL EXPANSION

Axial differential expansion between the rotor and the casing is a major limitation on the rate at which the turbine unit can be warmed up and loaded. As the turbine warms up the casing and rotor heat up at different rates and therefore expand at different rates. The differential rates at which the two parts expand can cause metal-to-metal rubbing between fixed and moving parts particularly in the glands and blading.

Each turbine unit usually develops a characteristic differential axial expansion pattern in response to normal transients on startup and loading. This pattern should be familiar to the operator because the amount and rate of differential expansion can provide an early diagnosis of problems even before limits are exceeded.

In addition to excessive heat up rates, abnormal differential expansion can be caused by water induction, thrust bearing failure, binding of components (constrained from expanding), and quenching of casings from wet lagging. By understanding the differential expansion pattern for normal conditions, these abnormal conditions can be spotted prior to becoming major casualties.

There are basically three methods of maintaining axial expansion within acceptable limits:

(a) limit the heatup rate so that temperature gradients across the casing and rotor are small enough to keep differential expansion to an allowable value;

(b) design the rotor and casing to limit the temperature differential between fixed and moving parts to a small enough value to force reasonably equal expansion of both;

(c) allow sufficient clearances between fixed and moving parts to accommodate the differential expansion between rotor and casing from cold to hot conditions.

The latter method must be used to accommodate the fact that the rotor expands from its fixed point at the thrust bearing through the entire length of three low pressure turbines while each casing is fixed at one end. This arrangement, which is shown in Figure 5.11, results in the rotor expanding in the order of 25 mm (one inch) more than the casing in the last LP turbine. The only satisfactory method of handling this is to allow sufficient room in the LP turbines to accommodate this growth. Thus the clearance between fixed and moving blades increases in the LP turbines the further one is from the thrust bearing. Typically the clearances are in the order of 5 mm in the HP turbine, 12 mm in the #1 LP turbine, 19 mm in the #2 LP turbine, and 26 mm in the #3 LP turbine.

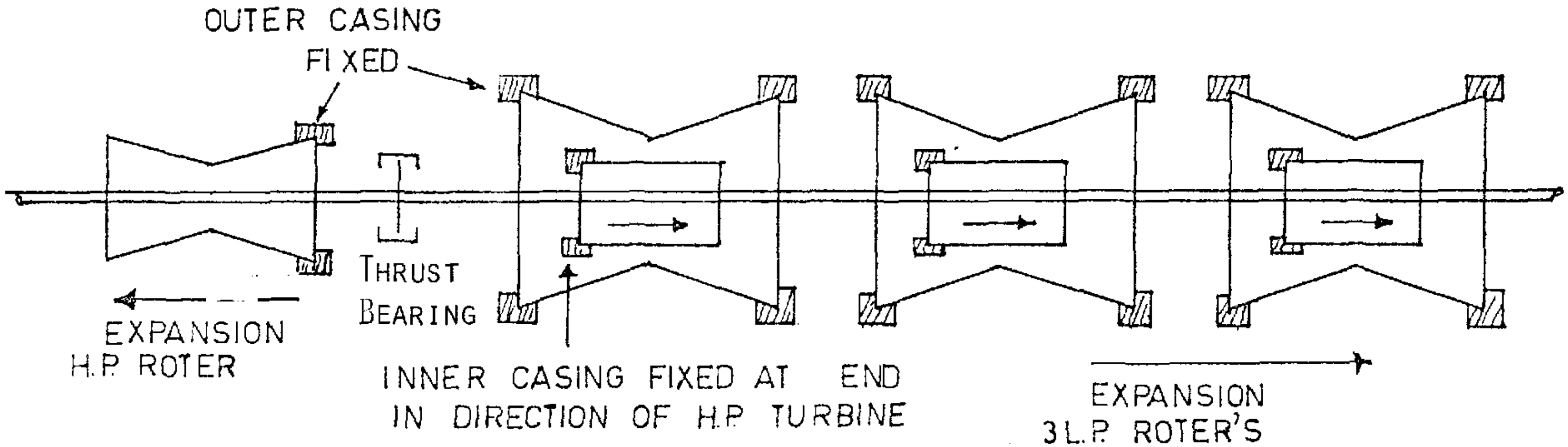
Expansion of the low pressure turbines' outer casings is relatively small compared to shaft expansion and, therefore, the differential expansion detectors are essentially measuring shaft position. Since shaft growth is cumulative as one moves away from the thrust bearing, an expansion in the first LP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

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

Since increased clearances reduce turbine efficiency, this method of compensating for differential expansion cannot be used indiscriminately. Such construction techniques as carrier rings, double casings and drum rotors which are discussed in the level 2 course are utilized to equalize the heatup rates of fixed and moving parts.

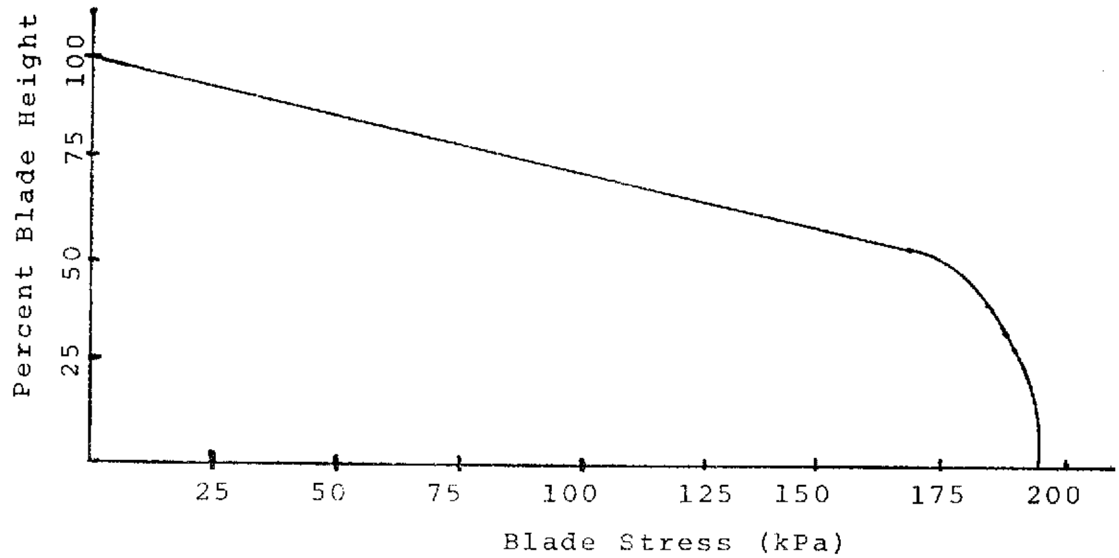
STRESSES IN TURBINE BLADING

In the long blades of the latter stages of the low pressure turbines the tensile stresses set up in the blades due to centrifugal force are quite impressive. At 1800 rpm the stress in the root of a blade 1 meter long with an overall last stage diameter of 3.6 meters is in excess of 175,000 kPa (25,000 psi). Figure 5.12 shows the distribution of stress along the length of the blade.



Fixed Points Of Shaft And Casings

Figure 5.11

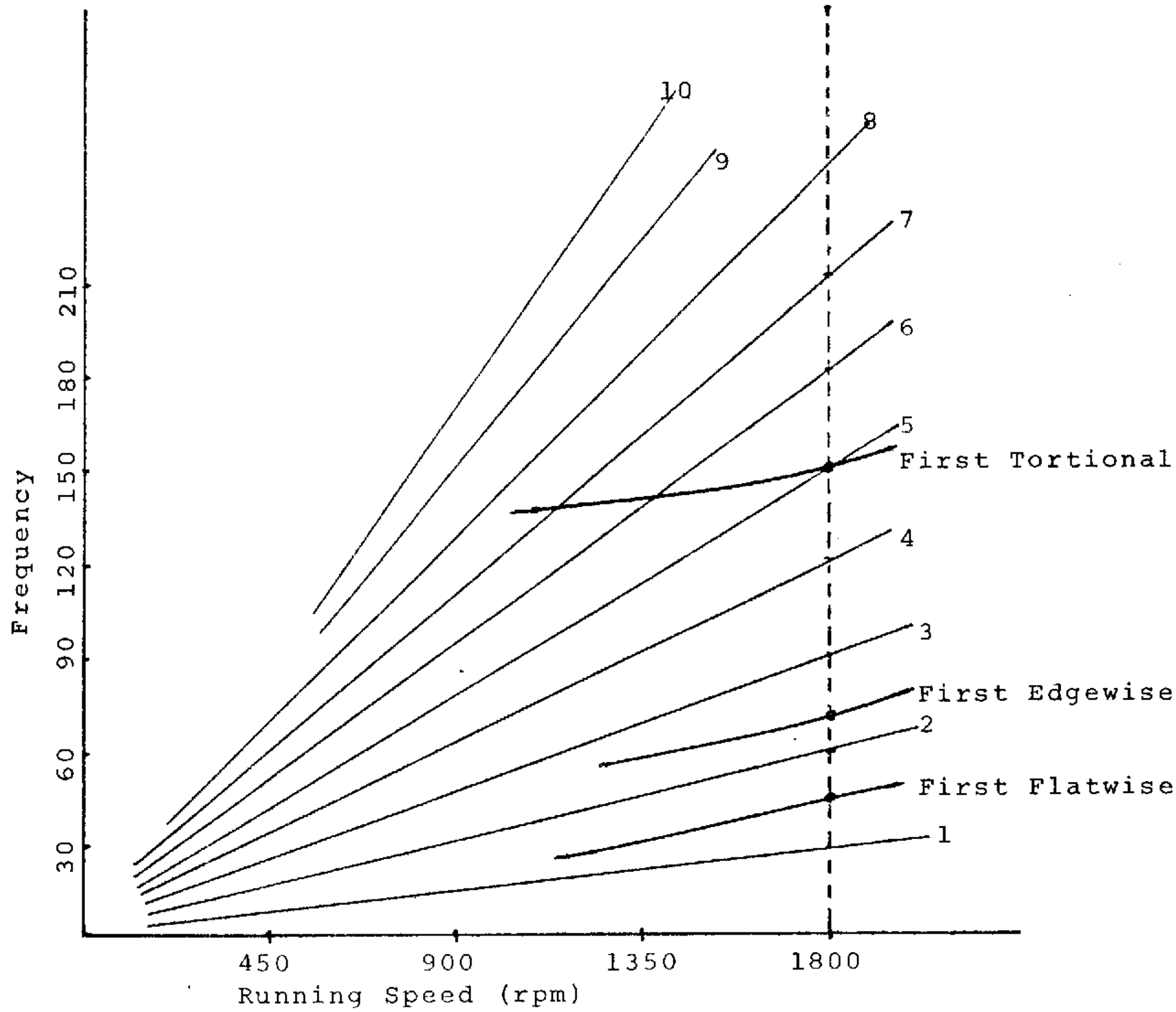


Stress Profile In Long Turbine Blade

Figure 5.12

In addition to designing blades which must withstand these stresses for years of operation, the designer must ensure the blades do not come into resonance at the operating speed or any harmonics of this speed. Using computers and advanced experimental facilities the designer develops a "Campbell Diagram" for each stage to insure that the blades do not resonate at operating speed. A typical Campbell Diagram is shown in Figure 5.13. The numbered diagonal lines are harmonics of the fundamental rotational frequency of the turbine. For example, the condition of the first flatwise resonant frequency lying midway between the fundamental rotational frequency (30 Hz) and the 2nd harmonic (60 Hz) is optimal. The coincidence of the first torsional resonant frequency and the 5th harmonic is undesirable and would necessitate stiffening of the blades to prevent undesirable fatigue stresses in the blading. In fact, the coincidence of any harmonic with a resonant frequency of the blade is undesirable.

The point of this discussion is that the blades have a variety of rather complex stimuli with which to deal and further complicating their environment only serves to shorten blade life. Apart from erosion of blading due to poor steam quality, probably the greatest factor in shortening turbine blade life is condenser vacuum. Transient heating of blading under a combination of high speed, low steam flow and low vacuum conditions can appreciably shorten blade life. For this reason vacuum on startup should be the best obtainable and, as turbine speed increases, the minimum acceptable vacuum



Campbell Diagram

Figure 5.13

should increase. In addition, exhaust sprays may be necessary on startup to prevent overheating of the latter stages of the LP turbine.

OIL TEMPERATURE

Generally speaking, the viscosity of oil is a direct function of temperature; the higher the temperature the lower the viscosity. Thus at low temperatures oil does not flow nearly as well as at high. In fact, at low oil temperatures the flow of oil to the bearings may be inadequate to keep the bearings cool and elevated bearing metal temperatures may develop. Even near operating oil temperatures, the effect of decreasing oil temperature out of the oil coolers may be to increase rather than decrease bearing temperatures.

Another phenomenon of low oil temperature is bearing instability caused by oil whip or oil whirl. This oil whip is shown in Figure 5.14 and results from high oil viscosity at low oil temperatures. The wedge of oil which by design should support the shaft becomes unstable and moves around the journal and tends to drive the journal within the bearing at slightly less than half the running speed. The resulting vibration can become quite serious.

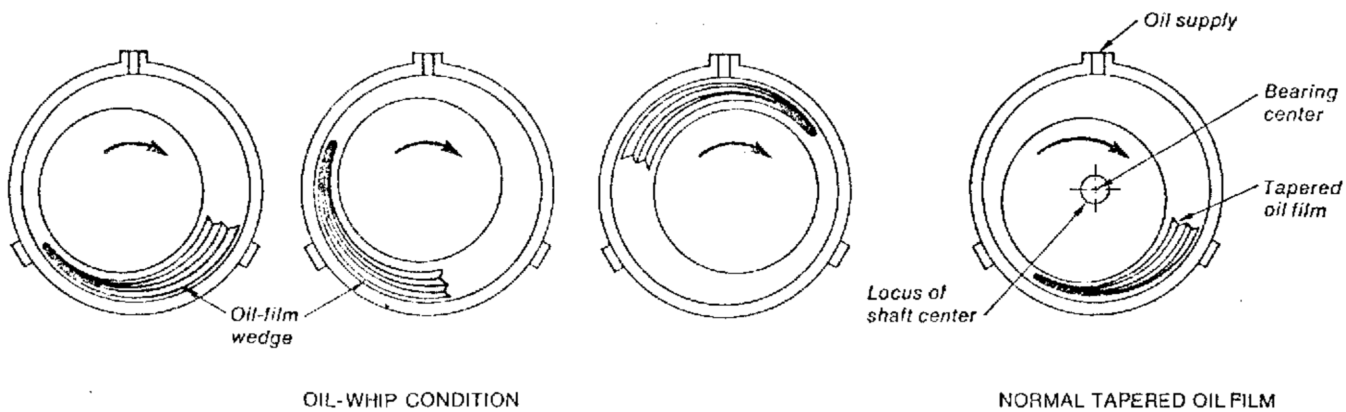


Figure 5.14

There are a number of factors which influence this phenomenon but for a well-designed bearing which has exhibited no instability in the past, the likely cause is an increased oil viscosity caused by low oil temperature. However, the phenomenon of oil whip is intensified by low loading of the shaft in the bearing; the more lightly loaded the bearing the more likely the shaft is to wander about in it. For this reason bearing vibration at normal oil temperature may be an indication of shaft misalignment within the bearings.

ASSIGNMENT

1. Explain the reasons for the following requirements on startup.

   (a) Placing the turbine on the turning gear 24 hours prior to steam admission.

   (b) Having the turbine on the turning gear prior to applying sealing steam to the glands.

   (c) Energizing the generator field prior to steam admission to the HP turbine.

(d) Increasing the minimum acceptable vacuum as turbine speed increases.

(e) Draining steam lines, turbine and extraction steam lines.

(f) Checking vibration.

(g) Checking axial expansion.

(h) Holding speed at 1200 rpm until lube oil temperature is above 39°C and rotor temperature is above 20°C.

(i) Bring speed up quickly through the critical speed range.

(j) Returning to 1200 rpm if HOLD parameters develop in the critical speed range.

(k) Block loads on synchronizing.

(l) COLD, WARM and HOT loading rates.

2. Why is vibration undesirable?

3. What is a "Campbell Diagram" and why is it used by turbine designers?

4. What is oil whirl (oil whip)?

5. You are conducting a startup with a specified loading rate of 15 MW per minute and find you have gone from 150 MW to 300 MW in 7 minutes.

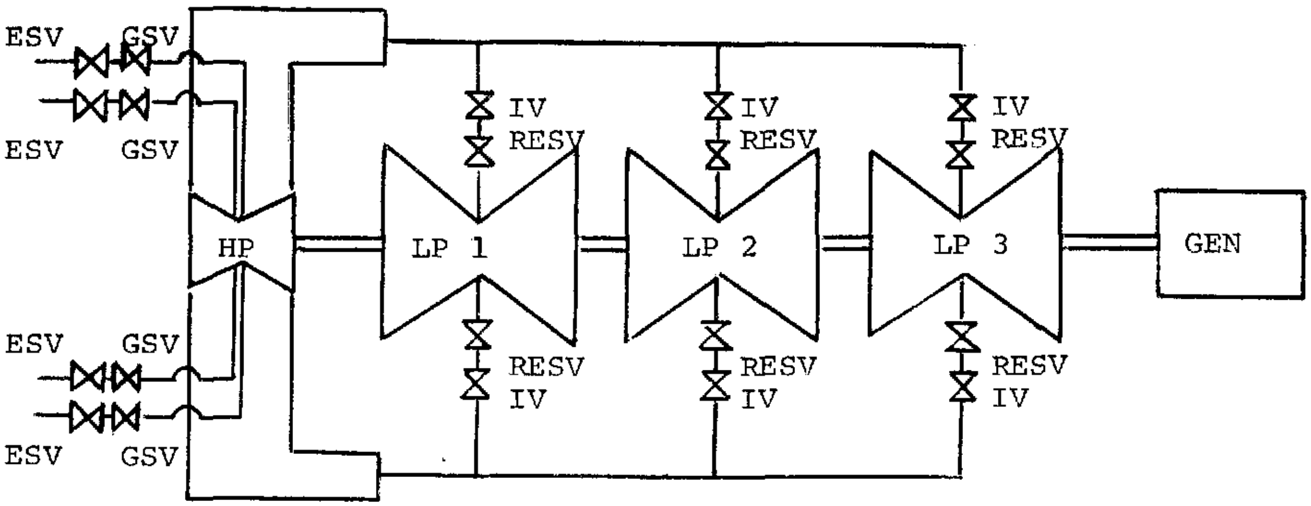
   (a) What problems does this present?

   (b) What action would you take?

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 134

RELIABILITY AND TESTING REQUIREMENTS



ESV = Emergency Stop Valve
GSV = Governor Steam Valve
IV = Intercept Valve
RESV = Reheat Emergency Stop Valve

CONTROL VALVES

Figure 6.1

Figure 6.1 shows the layout of the turbines and steam valves associated with a large generating station. The ability of these valves to quickly and reliably shutdown the turbine unit on a fault is important to the safety of not only the turbine but also the personnel and equipment surrounding the turbine. This is of particular significance in the case of a turbine overspeed because of the possibility of blade wheel failure which may result in pieces of the wheel being thrown through the casing. In the casualties of this type which have occurred around the world the following has been typical:

(a) significant personnel injury in a majority of the cases,

(b) fatalities in several cases,

(c) blade wheel fragments being thrown through the casing and turbine hall wall, with some pieces travelling up to a quarter of a mile, and

(d) lubricating oil and generator hydrogen fires.

In addition, in nuclear stations there is a growing concern with the possibility of an unterminated overspeed casualty causing missiles which could cause failure of components in the nuclear steam supply system. This latter consideration is of sufficient public safety concern to warrant the AECB setting a target for the maximum allowable frequency of unterminated turbine overspeed incidents of 10^{-4} events per year. That is, that a turbine unit must experience an unterminated overspeed incident no more frequently than once in 10,000 years. Experience has shown that potential overspeed incidents (for example a loss of output line) occur approximately once per year. In order to meet the AECB target for unterminated overspeed incidents, it is necessary to prove that the unavailability of the system to arrest a potential overspeed is less than 10^{-4}.

This lesson will discuss the general approach which must be taken in establishing a testing frequency for the turbine components associated with overspeed protection in order to meet this target frequency.

THE PROBLEM

If the generator load is lost through the opening of the output breaker, the counter torque which the load current exerts on the generator rotor is lost. Unless the steam supply to both the high pressure and low pressure turbine are rapidly shut off, the turbine speed rapidly increases and in a matter of seconds reaches a point between 175% and 200% of operating speed where the stress on the largest wheels in the low pressure turbines exceeds the ultimate tensile strength of the metal. At this point the blade wheel ruptures into several large fragments (60° to 120°) and many smaller ones. These pieces may be thrown through the casing severing steam lines and lube oil lines. At this point the overspeed will be terminated and the unit will begin to slow down.

OPERATION OF THE OVERSPEED TRIP

The response to an overspeed condition varies from plant to plant depending on the type of governing system (mechanical-hydraulic or electrical-hydraulic) and the fluid which operates the control valves (lubricating oil, fire resistant fluid or air). However, the large nuclear steam turbine units operated by Ontario Hydro have the following common characteristics:

(a) High pressure steam is admitted to the turbine through four main steam lines. Each of these lines has an emergency stop valve and a governor steam valve in series.

(b) Low pressure steam exits from the high pressure turbines and, after passing through the moisture separator and reheater, enters the three low pressure turbines. Each low pressure turbine has two steam input lines and each of these six low pressure steam lines has an intercept valve. (In the case of the Bruce Nuclear Generating Station which is shown in Figure 6.1, there is a reheat emergency stop valve in series with each intercept valve).

(c) Regardless of the action taken by the overspeed protection devices, if the turbine speed rises to between 110% and 112% of operational speed, a tripping device, using a spring loaded, centrifugally operated overspeed bolt, operates and shuts all of the valves.

(d) Because there is more than one steam line associated with steam admission to each turbine, any single line can be closed at power without appreciably effecting the unit output. This permits on-load testing of each valve.

ANALYSIS

The unavailability of a particular valve can be related to its failure rate and the interval between tests by the formula

$$Q_i = \frac{\lambda_i \, T_i}{2} \qquad 6.1$$

where: Q_i = Unavailability of Valve i

λ_i = Failure Rate (per annum) of Valve i

T_i = Interval Between Tests (in years) of Valve i

This relationship assumes that on the average a failed valve has been in that condition for half the test interval before being found and corrected on the next test. It is obvious from the equation that as the failure rate of the valve decreases or the interval between testing decreases, the unavailability of the valve decreases. That is, frequent testing and a reliable valve will minimize the probability of a valve failing to operate when called upon (Unavailability).

VALVE UNAVAILABILITY

Referring to Figure 6.1, if steam fails to be shut off to the high pressure turbine in a particular inlet line then both the governor steam valve (GSV) and the emergency stop valve (ESV) in that line must be unavailable. So the unavailability associated with a particular high pressure inlet line is:

$$Q_{LINE} = Q_{GSV} \, Q_{ESV} \qquad 6.2$$

Similarly, if steam fails to be shutoff to the low pressure turbine in a particular inlet line then both the intercept valve (IV) and the reheat emergency stop valve (RESV) in that line must be unavailable. So the unavailability associated with a particular low pressure inlet line is

$$Q_{LINE} = Q_{IV} \, Q_{RESV} \tag{6.3}$$

It can be seen that by having both an intercept valve and a reheat emergency stop valve in the inlet lines to the low pressure turbine the unavailability associated with each line is considerably reduced.

For a successful turbine trip, all ten of these lines must shut. So the unavailability of the valving for proper shutdown of the unit is given by the expression

$$Q_{VALVING} = 4 \, Q_{GSV} \, Q_{ESV} + 6 \, Q_{IV} \, Q_{RESV} \tag{6.4}$$

Table 6.1 summarizes valve failure rate data based on past testing and the unavailability of the steam valve system for various test intervals.

TABLE 6.1

STEAM ADMISSION VALVE UNAVAILABILITY

TEST INTERVAL	PREDICTED UNAVAILABILITY
1 Week	2.3×10^{-6}
2 Weeks	9.2×10^{-6}
1 Month	4.3×10^{-5}
3 Months	3.8×10^{-4}

$$Q_{VALVING} = 4 \, Q_{GV} \, Q_{ESV} + 6 \, Q_{IV} \, Q_{RESV}$$

COMPONENT	PREDICTED FAILURE RATE
Governor Steam Valve	.07/annum
Emergency Stop Valve	.07/annum
Intercept Valve	.04/annum
Reheat Emergency Stop Valve	.03/annum

TRIP LOGIC UNAVAILABILITY

In the turbine unit shown in Figure 6.1, all of the valves are held open against spring tension by a fire resistant fluid (FRF) system. On a overspeed condition, the

FRF pressure is dumped from the underside of the valve operating pistons and spring tension drives the valve shut. The tripping circuit is composed of two independent channels either one of which is capable of dumping the FRF and shutting all 20 of the valves. The two trip channels are interlocked to prevent both channels being tested at the same time.

Figure 6.2 shows a simplified arrangement of the turbine tripping system. Each channel consists of:

(a) an overspeed sensor in the form of a spring loaded, centrifugally operated overspeed bolt mounted on the HP turbine rotor,

(b) a spring loaded pilot trip plunger,

(c) a mechanical linkage between the overspeed bolt and the pilot trip plunger,

(d) a spring loaded main trip plunger,

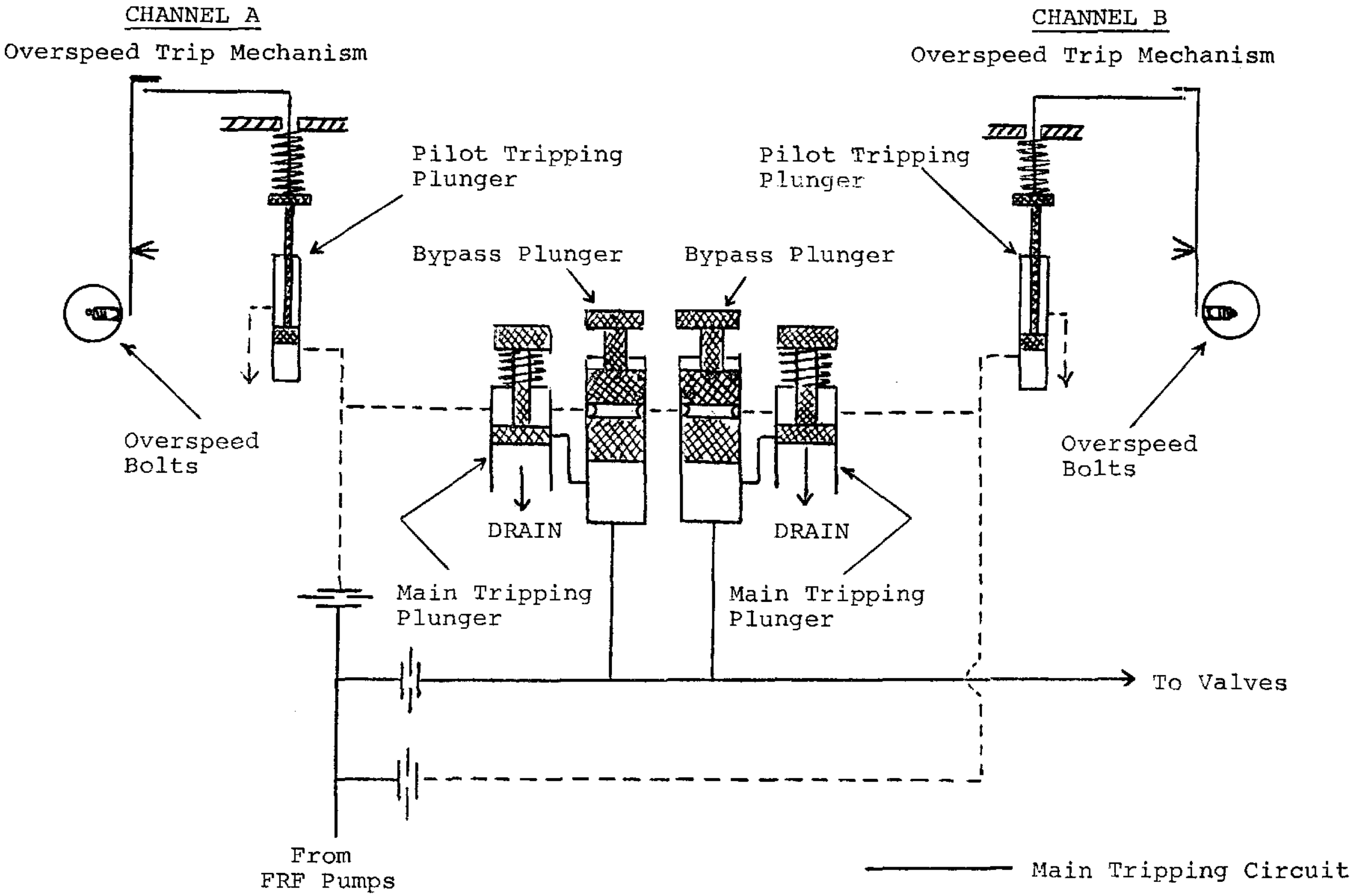
(e) a spring loaded bypass plunger, used to gag a channel for test, and

(f) an FRF circuit called the pilot tripping circuit.

The drawing shows all equipment in its normal operating state. The pilot trip plunger is held up against spring tension by a mechanical linkage shown on the drawing as the "overspeed trip mechanism". In the up position the pilot trip plunger isolates the pilot tripping circuit (dashed line) from the drain, thereby keeping the pilot circuit pressure up. The pilot tripping circuit pressure forces the main trip plunger down against its spring. In this position, the main trip plunger isolates the main tripping circuit from the drain on the underside of the main trip plunger.

On an overspeed, centrifugal force overcomes the spring tension on the overspeed bolts and forces them out to eventually contact the "overspeed trip mechanism". The overspeed trip mechanism releases the pilot trip plunger and allows it to spring down and release the pilot tripping pressure to drain. As the pilot tripping pressure is reduced, the main trip plunger springs up opening the main tripping circuit to drain. The falling pressure in the main tripping circuit trips the steam admission valves.

Because the pilot tripping circuit for Channel A and B are connected to both main trip plungers, a trip on either channel will trip both main tripping plungers to dump. This design makes the system less suseptible to main trip plunger failure since either trip plunger can receive a trip signal from either overspeed bolt.



OVERSPEED TRIPPING CIRCUIT

Figure 6.2

Bypass plungers are provided to allow on-power testing of the tripping circuit. Depressing the bypass plunger for a channel:

(a) gags the channel by isolating the main tripping plunger for that channel from the main tripping circuit, and

(b) isolates the pilot tripping plunger of the channel to be tested from the main tripping plunger of the other channel.

It is then possible to test the entire trip channel without tripping the turbine and without totally disabling the turbine trip system. Routine on line tests of the overspeed trip is accomplished by forcing the bolts out under oil pressure. The oil pressure required to trip the channel under test is then correlated to an operating speed.

It can be seen that an overspeed trip channel can be rendered unavailable by any of the following:

(a) Overspeed Bolt (OSB) fault

(b) Trip Linkage (TL) fault

(c) Pilot Trip Plunger (PTP) fault

(d) Main Trip Plunger (MTP) fault

(e) Bypass Plunger (BP) fault

(f) Testing (T)

Thus, the unavailability of either overspeed Channel, A or B, can be expressed as

$$Q_A = Q_B = Q_{OSB} + Q_{TL} + Q_{PTP} + Q_{MTP} + Q_{BP} + Q_T \qquad 6.5$$

Since either overspeed trip channel is capable of effecting a successful turbine trip, the unavailability of both channels is approximately

$$Q_Z = (Q_{OSB} + Q_{TL} + Q_{PTP} + Q_{MTP} + Q_{BP} + Q_T)^2 \qquad 6.6$$

Equation 6.6 makes two symplifying, and conservative, assumptions:

(a) that both channels can be in test simultaneously (they cannot)

(b) that the overspeed bolt on one channel cannot trip the main tripping plunger on the other channel (they can)

TABLE 6.2

TRIP SYSTEM UNAVAILABILITY

TEST INTERVAL	PREDICTED UNAVAILABILITY One Channel	PREDICTED UNAVAILABILITY Both Channels
1 Week	7.4×10^{-3}	5.5×10^{-5}
2 Weeks	5.7×10^{-3}	3.3×10^{-5}
1 Month	7.3×10^{-3}	5.3×10^{-5}
3 Months	1.8×10^{-2}	3.2×10^{-4}

$$Q_A = Q_B = Q_{OSB} + Q_{TL} + Q_{PTP} + Q_{MTP} + Q_{BP} + Q_T$$

$$Q_Z = (Q_{OSB} + Q_{TL} + Q_{PTP} + Q_{MTP} + Q_{BP} + Q_T)^2$$

COMPONENT	PREDICTED FAILURE RATE
Overspeed Bolt	.04/annum
Trip Linkage	.01/annum
Pilot Trip Plunger	.04/annum
Main Trip Plunger	.04/annum
Bypass Plunger	.01/annum
Testing	1 hour/test/channel

Table 6.2 summarizes the failure rate of tripping circuit components based on past testing and the unavailability of the tripping system for various test intervals. From this table it can be seen that monthly testing of the overspeed trip circuit is quite satisfactory. More frequent testing can effect, at most, only a modest decrease in unavailability and can actually cause an increase in unavailability due to the fact that a channel must be gagged to test it. Testing at three month intervals is unacceptible since it results in a trip circuit unavailability which alone (not including valve unavailability) exceeds the specified target unavailability of 10^{-4}.

SHUTDOWN SYSTEM UNAVAILABILITY

Overall overspeed trip unavailability is simply the sum of the unavailability contributions from the valves and the tripping circuit. Table 6.3 summarizes the system unavailability for various testing intervals and indicates monthly testing is the minimum frequency which will meet the target unavailability.

TABLE 6.3

OVERSPEED TRIP UNAVAILABILITY

TEST INTERVAL	PREDICTED UNAVAILABILITY
1 Week	5.7×10^{-5}
2 Weeks	4.2×10^{-5}
1 Month	9.6×10^{-5}
3 Months	7.0×10^{-4}

$Q_T = Q_Z + Q_{VALVING}$

ASSIGNMENT

1. Explain how equation 6.4 would be altered if the governor steam valves could not, by design, operate fast enough to prevent a potential overspeed from reaching an unacceptible high speed?

2. What would be the effect on the unavailability of the total overspeed protection system (valves and tripping circuit) if a 3 channel tripping circuit were used? a 6 channel circuit? an infinite channel circuit? (Calculations are not necessary)

3. What is the practical consequences of having a large number of tripping circuit channels?

4. Explain how equation 6.4 would be altered if a successful turbine trip was achieved if only five of the six low pressure inlet lines were to shut.

R. Schuelke

Turbine, Generator & Auxiliaries - Level 134

MAINTENANCE

Knowing when to schedule turbine maintenance is one of the most difficult decisions facing generating station personnel. If planned outages are too close together, considerable generating time can be lost; if too far apart, the probability of forced outage is increased significantly.

Some utilities overhaul turbines every three years while others lengthen the time between major overhauls to twice this period or even longer. Within broad limits, the time between major overhauls does not in and of itself determine the reliability of the turbine, generator and associated auxiliary systems. Frequent disassembly, inspection and reassembly of a turbine not only results in loss of maintenance dollars and generating capacity but may result in needlessly disturbing a properly operating unit.

As the cost of maintenance and alternate power sources increases there has been a general increase in the time between major overhauls. This can be accomplished without increasing the risk of forced outage provided the need for overhaul is assessed periodically and the decision as to when to overhaul is made upon engineering judgement rather than a "gut feeling" that the unit will run for another year without an overhaul.

The decision regarding the scheduling of major maintenance is based on several factors:

(a) progressive deterioration of the turbine heat rate to the point where the cost of outage for overhaul is outweighed by the cost of continuing to run the plant at low efficiency;

(b) major defects such as increasing bearing temperature, vibration, contamination of lube oil with bearing metal, shaft eccentricity or alignment and control valve maloperation;

(c) planned outage of the reactor system or electrical distribution system;

(d) availability of alternate power sources and demand for power from the hydro grid;

(e) availability of maintenance personnel, and

(f) availability of repair parts and repair facilities.

Regardless of what system is used for planning major plant maintenance, the scheduling of a major overhaul of all four turbines at a station in January of the same year represents a complete loss of touch with reality.

There are basically three approaches to scheduling maintenance for a particular piece of equipment: breakdown, preventive maintenance and engineered maintenance.

Breakdown maintenance allows equipment to operate to failure before it is repaired or replaced. It requires little planning but manpower utilization may be inefficient and downtime is usually unplanned and excessive. There are almost no major pieces of plant equipment which can be operated exclusively under this system. However, a number of more minor components are maintained by repair or replacement upon breakdown: steam traps, reducers, most manual valves and a wide variety of minor piping and electrical systems.

Preventive maintenance makes use of scheduled inspections and periodic equipment overhaul or replacement as a means to prevent breakdowns. Although preventive maintenance is often effective, it is generally expensive and has little value in predicting future performance. In certain systems, particularly those in which deteriorating performance cannot be readily measured, preventive maintenance is necessary to prevent forced outage. A number of non-destructive tests performed periodically on piping systems falls into this category as do routine pump and valve overhauls and certain statutory inspection requirements.

Engineered maintenance tempers the other two maintenance philosophies by using diagnostic and periodic testing and past experience to determine the frequency of overhaul, replacement or repair.

In any maintenance system each piece of equipment is assigned to one of these three categories. When left alone, equipment will always assign itself to the first category.

PERIODIC TESTING

All systems which are essential to the safe and reliable operation of the generating station probably require some form of periodic testing. These tests range from the critical tests of steam admission valves, safety valves, and turbine supervisory and tripping systems to the periodic calibration of pressure and temperature gauges and cycling of infrequently operated valves. Regardless what system is involved and what tests are required the following should be a requirement:

(a) a list should exist of the required tests including the frequency of testing, procedure to be used for testing and the equipment necessary for the testing;

(b) a method should exist for scheduling the required tests which includes a method of indicating when a test has been completed;

(c) for each test there should be indicated under what circumstances the test can be rescheduled or deleted, how this will be accomplished, who may authorize rescheduling a test and who will be informed if a test is not conducted;

(d) the schedule should state precisely who is responsible for insuring the test is conducted; and

(e) the schedule should state what action should be taken if the test results are abnormal.

DIAGNOSTIC TESTING

Diagnostic checks should focus on parameters which are indicative of unit performance including:

(a) changes in efficiency,

(b) vibration,

(c) oil and water purity,

(d) changes in flow, pressure and temperature,

(e) variation in control system response, and

(f) changes in operating environment.

It is insufficient to use alarm setpoints as the indicators of abnormal performance. Alarm conditions are operating limits not diagnostic tools. In addition, trends indicating a need for corrective action may develop over weeks or even months and may not be obvious to operating personnel. If the alarm point for bearing metal on a particular bearing is 95°C, and the bearing is currently at 75°C, the operator may not recall it was 55°C a year ago.

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, if necessary, shop facilities at the manufacturer's plant can be made in advance.

TURBINE OVERHAUL

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.

The current operating climate in today's generating stations poses some difficult problems:

(a) because of the high cost of an outage on a nuclear steam turbine, the overhaul must be completed as quickly as possible. This means there will be little excess time for casual inspections, training unfamiliar personnel and doing work over;

(b) as the length between overhaul increases the probability of having a large number of personnel who have hard core turbine overhaul experience decreases. Many maintenance personnel may have never seen the inside of a turbine;

(c) with lengthening periods between overhauls the need for detailed inspections increases. It's been a long time since the turbine was last opened and will be a long time before it is opened again;

(d) if trained personnel are to be available for the next overhaul, they will need to gain experience during this overhaul.

The sequence of events of a typical major turbine overhaul is as follows:

1. Remove the relevant pipework and the upper half casings.

2. Remove bearing covers and shaft coupling bolts. The thrust bearing is left undisturbed as a fixed point until clearance measurements are taken.

3. Measure blade and gland clearances between the fixed and moving parts.

4. Remove the thrust bearing and the rotors for detailed examination of fixed and moving blades, blade wheels, diaphragms, glands, casing, bearings and other internal parts.

5. Repair glands and restore all radial clearances. Bearing measurements are taken and clearances checked.

6. Replace the rotors and measure clearances of blades and glands.

7. Check alignment of the shaft.

8. Refit the upper cylinders.

9. After remaking the horizontal casing joints and refitting all heavy parts, the final coupling alignment readings are taken and adjustments made if necessary.

10. Refit the thrust bearing and journal bearing covers.

The major parts of a turbine requiring attention during overhaul are detailed below.

Moving Blading and Blade Wheels

While the condition of the blading can be occasionally gauged from operational data such as steam consumption, stage pressure drop and vibration, inspection is the most reliable method of assessing blade condition. The most frequent signs of blade and blade wheel deterioration are moisture erosion, cracking, rubbing, lacing wire erosion and shroud rubbing and erosion. Erosion from moisture generally shows up first on the leading edge of the back side of moving blades where the blades impact with the slower moving droplets of water moving through the blading. On low pressure blading where stellite or chrome steel inserts have been fixed to the leading edges of blading, erosion can be seen as undercutting of the softer steel around the insert. "Fretting" of the blade tips or shroud bands may indicate the presence of standing water within the lower casing.

The blading and shrouds should be inspected for rubbing both between fixed and moving blades and between the rotating elements and the casing. Clearances should be taken between all fixed and moving parts. Rubbing of any kind is abnormal and may indicate misalignment, excessive thrust or journal bearing clearances, past turbine startups or loading in excess of design limits, water induction or component distortion.

Blade or wheel cracking is generally detected at the blade roots (particularly in the LP turbine blades) or at the wheel to shaft junction. These cracks are usually found by liquid dye penetrant test or magnetic particle test. Any cracks should be ground out and either filled or dressed prior to closing out the turbine. Any grinding on the rotor or blade wheels should be done only by experienced personnel and then only to the manufacturer's limit. If the grinding proceeds too deeply the rotor can be subjected to high stresses and further cracking. Lacing wires occasionally fail due to erosion or fatigue cracking and these should be inspected with particular attention given to the area where the lacing wire joins the blades.

Any unusual condition within the turbine casing should be discussed with the turbine vendor as they are indicative of deeper problems. Additionally, many problems which appear hopeless upon discovery can be corrected at least temporarily by skilled turbine maintenance personnel.

Nozzles and Diaphragms

Most of the comments regarding moving blades are equally applicable to the fixed blading. Moisture damage frequently shows up first on the trailing edge of the inside of the blades as an erosion or loss of material. Diaphragms should be checked for cracks. Checks should also be made for distortion and proper fit in the casing grooves. The seal between the diaphragm and shaft should be examined for evidence of rubbing and erosion. The diaphragm halves are often removed from the casing grooves so that the latter may be cleaned. The diaphragms must be allowed to accommodate thermal expansion in the casing grooves and corrosion products or misalignment can result in seizing and distortion.

Casing

The horizontal flange between the upper and lower casing halves should be carefully inspected for evidence of steam cutting. The casing and bolts should be inspected for cracking by magnetic particle or dye penetrant test. Cracking is most likely to occur at the threads of the bolts, at the outside surface of the casing and where the diaphragms and carrier rings join the casing. Particular attention should be given to edges and notches in the casing which can act as stress raisers.

The inner casing should be checked for evidence of erosion and corrosion particularly in and around the horizontal casing joint and where extraction steam or steam to the auxiliary separators leaves the casing. The lower part of the casing should be checked for evidence of standing water which could indicate blocked or inadequately sized drains.

Glands

Indication of the condition of the casing glands may be judged by excessive gland seal steam consumption or by the pressure necessary to seal the glands. Dissolved oxygen level in the feedwater or excessive air extraction requirements may also indicate faulty glands.

When the labyrinth glands are opened, they should be cleaned, straightened if necessary and adjusted to correct clearances. Badly worn or damaged sections should be replaced.

Some attention should be given to the axial and radial shaft landings. If there is heavy damage due to rubbing or foreign matter, the turbine manufacturer should be consulted.

Bearings

A thorough examination should be made of bearings for wear, grooving of the bearing metal and shaft, loose bearing metal, correct contact surface and possible evidence of electrolysis. Although electrolysis is not a frequent problem, it can occur if the shaft grounding device is not making good contact. The problem is accentuated in saturated steam turbines where the wet steam promotes static charge formation on the rotor. The dissipation of this static charge is normally accomplished by the shaft grounding devices but if forced to the voltage will dissipate through the journal bearings causing pitting or the thrust bearing causing bearing material loss at the trailing end of the shoe.

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 1 mm per meter diameter of journal. 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.

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

Emergency Stop Valves and Governor Valves

The emergency stop valves should be periodically disassembled and checked for proper clearance between the valve spindle and bushing. Scaling, galled areas and scoring should be eliminated. The seat and disc should be examined for wear damage and cracking. The strainer should be inspected for deposits and damage and should be cleaned, repaired or replaced. The sealing surfaces should be inspected for steam cutting.

The oil piston should be inspected for contamination, rust, wear and freedom of movement.

<u>Studs and Bolts</u>

The bolts and studs used to form steam tight joints at high temperatures are made of creep-resistant material. The practical requirement for these bolts is that after extended number of hours of service the initial strain must not relax to the point where they exert insufficient stress to keep the joint tight. On the other hand, the material must have sufficient ductility to be retightened a number of times without cracking.

These bolts must be treated with reasonable care if they are to adequately perform their function while surviving in a rather hostile mechanical and thermal environment. The strain on the bolts is of particular importance. To enable creep-resisting, high tensile strength bolts to be accurately strained one of three possible methods of tightening is employed:

(1) By means of a torque wrench. This method is generally used on small bolts and requires that the mating threads are reasonably free to move and do not produce a significant torque of their own. The torque wrench used for these operations should be calibrated before use and checked afterwards.

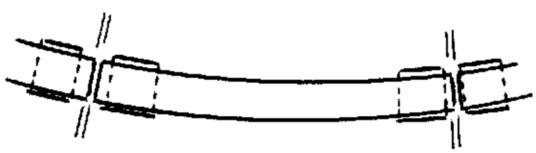
(2) By heating the bolts and turning the nut a specified number of degrees or number of "flats". The heating of bolts is a common method of tightening the HP turbine casing bolts (LP casing bolts are generally tightened with a torque wrench). High pressure bolting materials generally have low impact strength and, in consequence, are prone to failure by cracking, even at room temperature, if subjected to any form of impact or hammering. Therefore, the most convenient and practical way of producing the necessary bolt loading is by tightening the nut when the bolt has been expanded by heat. The material used for turbine bolts normally has a lower coefficient of linear expansion than the flange material, so once the

bolt is properly tightened, the "squeeze" on the flange will increase as the joint assembly is heated.

(c) By hydraulic stretching of the bolts to a given strain and then hand tightening the nut. This method is accurate and usually quicker than heating but the hydraulic jacks necessary for large diameter bolts are bulky and difficult to handle.

Shaft Alignment

Figure 7.1 shows the general arrangement of a large saturated steam turbine with one HP cylinder and three LP cylinders tandem compounded. The rotor in each cylinder is supported by one bearing at the end of each rotor and the coupling between each rotor occurs between adjacent turbine bearings. This allows the flexibility of each rotor to be independent of the other rotors and enables independent balancing and removal of each rotating element. Since each shaft has an elastic deformation due to gravity the entire line of shafting must lie on a curve and the bearings are lined out to suit this static deflection or "catenary" as it is called. Although it should be obvious, it is worth mentioning that the center of rotation for the unit is the center of this catenary rather than a horizontal line. The couplings between individual shafts are therefore aligned to join the shafts in this catenary. Figure 7.2 shows the type of bearing and coupling alignment used to reflect this catenary and compares it to the alignment which would exist if the shafts were mounted horizontally.


With Catenary

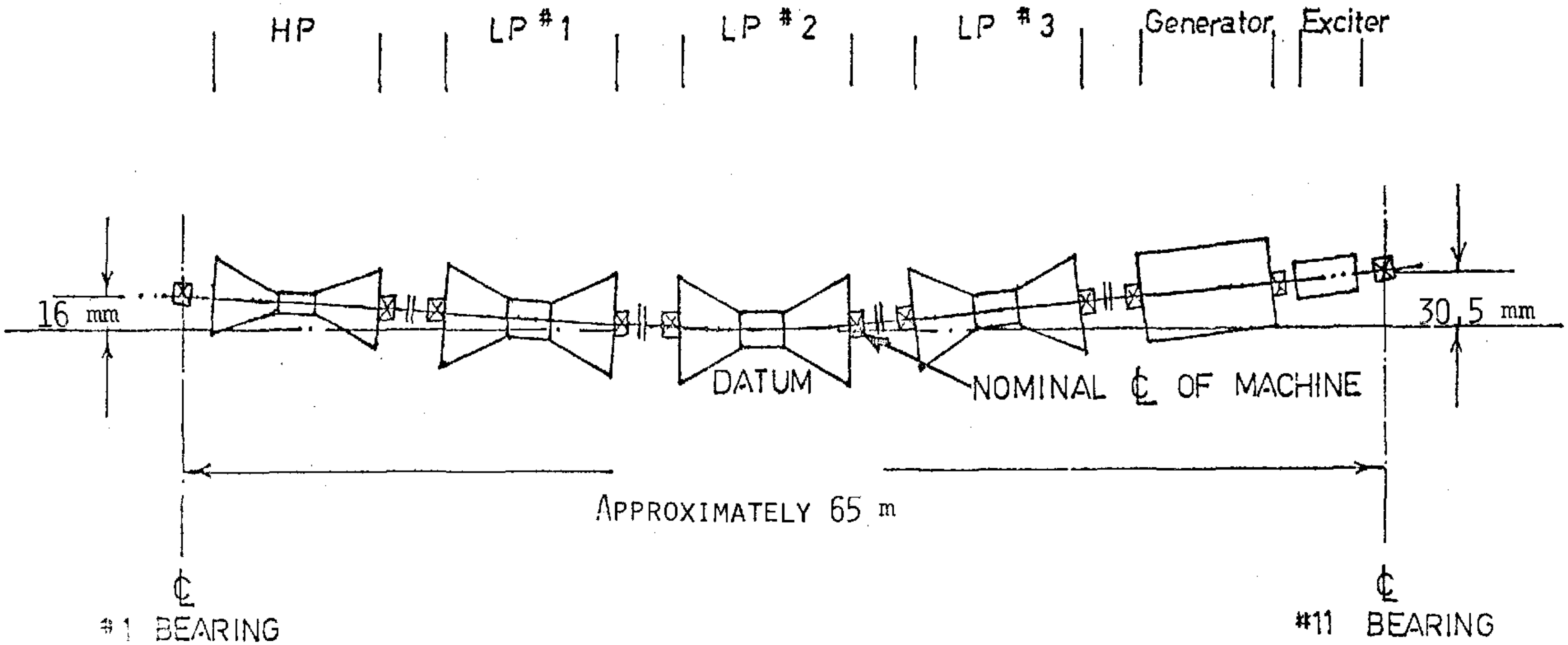

Horizontal Bearings

Effect of Shaft Catenary

On Bearings And Couplings

Figure 7.2

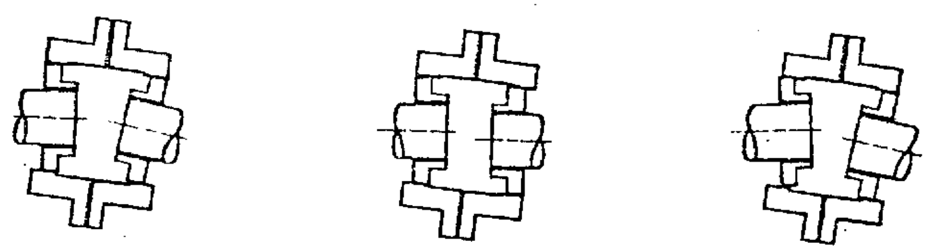
There are two basic types of misalignment which can exist in this type of shaft: misalignment of the shafts to each other (shaft coupling misalignment) and misalignment of the shafts to the bearings (bearing misalignment).



<u>Typical Shaft Catenary</u>

<u>Figure 7.1</u>

There are two types of shaft coupling misalignment: angular, where the center lines of the two shafts meet at an angle (Figure 7.3(a)); and offset, where the center lines are parallel but offset to one another (Figure 7.3(b)). Figure 7.3(c) shows a combination of the two.



Angular Misalignment (a)

Offset Misalignment (b)

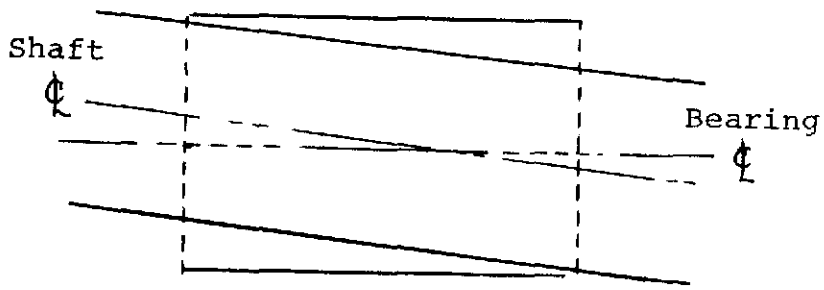
Angular With Offset Misalignment (c)

Coupling Misalignment

Figure 7.3

Coupling misalignment, even with flexible couplings, results in axial and radial forces which produce axial and radial vibration. This is true even when the misalignment is within the limits of "flexibility" of the coupling. The size of the forces and therefore the amount of vibration generated will increase with increased misalignment. The significant characteristic of vibration due to misalignment is that it will be in both the radial and axial direction. Axial vibration is the best indicator of misalignment. When the axial vibration is equal to or greater than one half of the radial vibration then misalignment should be suspected. It is worth noting that the symptoms of a bent shaft are almost identical to angular misalignment.

Bearing misalignment is shown in Figure 7.4. No vibration will result from a misaligned journal bearing unless an unbalance of the shaft exists. The reaction of the misaligned bearing to the unbalance will produce vibration in both the axial and radial direction. In this case the misalignment results in an axial component which is generally small when compared to the radial component.



Bearing Misalignment

Figure 7.4

The approximate alignment settings are obtained during erection by means of a taut piano wire passed through the bottom half of the turbine casings, pedestals and stator core. Then by means of an internal micrometer, the horizontal distance between the wire and casing or bearing can be measured from both sides of the horizontal joint faces and the casing and pedestal positions adjusted to centralize the wire within the machine. Similar measurements are taken on the vertical axis to the bottom of the casings and by accounting for the sag of the wire, it is possible to adjust the casings to the static catenary of the unit. Alternately a precision telescope or, more recently, a laser may be used for this purpose. The rotors are then used to obtain final settings by adjusting the bearings to give concentricity and parallelism of coupling faces.

The permissible error of coupling alignment readings is extremely small and a total error of .025 mm is usually accepted as being the limit.

After installation of the rotors and their couplings, dial gauge readings are taken on the shafts at each coupling as the rotor is revolved through 360°.

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.

Maintenance Records

It is essential that good records be kept of all turbine maintenance: casualties, significant events, general chronology of each overhaul, alterations, replacement of components and all numerical clearances and measurements. Experience has proved that the more detailed these records are, the more valuable they become. There simply is no substitute for knowing exactly what was seen, done and thought the last time the unit was worked on.

This is particularly important when few people exist who were involved in the last overhaul, but even when such personnel exist they may not remember everything they saw and did.

ASSIGNMENT

1. Outline the basic factors to be considered in turbine maintenance.

2. What factors influence the decision as to when to schedule a major turbine overhaul?

3. Outline a program of preparations prior to shutting down a turbine generator unit prior to overhaul.

4. Discuss the items which should be examined during overhaul including:

   (a) blading
   (b) glands
   (c) diaphragms and nozzles
   (d) alignment
   (e) thrust bearings
   (f) radial bearings
   (g) casing
   (h) casing drains
   (i) rotor
   (j) evidence of presence of water
   (k) clearances
   (l) turbine flange faces

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