NUCLEAR TRAINING DEPARTMENT
COURSE PI34

NUCLEAR TRAINING COURSE # PI34

TIMS # PI3004

NUCLEAR TRAINING COURSE

COURSE PI-34

TURBINE & AUXILIARIES

INDEX

Turbine & Auxiliaries - Course PI 34

OBJECTIVES

Following the completion of this course the trainee will be able to:

334.00-1 The Steam Turbine

1. Identify and state the functions of the turbine:

   a) casing;
   b) rotor;
   c) shaft;
   d) blade wheel;
   e) diaphragm;
   f) nozzle;
   g) fixed blades;
   h) moving blades.

2. State the definition of a turbine stage.

3. Explain the difference between an impulse and a reaction turbine.

4. Explain the difference between double flow turbines and single flow turbines and explain the advantages of a double flow turbine.

5. Draw the pressure and velocity changes across the fixed and moving blades of a reaction and an impulse turbine.

6. State the purpose of a nozzle.

7. Explain why turbines usually have several stages.

8. Explain the relative advantages and disadvantages of the two types of turbine blading with respect to:

   a) stage efficiency;
   b) velocity ratio;
   c) moisture effects;
   d) axial thrust.

9. Explain the effect of steam flow on the pressure gradient through a large turbine.

234.00-1 Turbine Construction

1. Explain how moisture is removed from the steam in the steam system and turbine by each of the following:

   a) moisture separator;
   b) reheater;
   c) extraction steam;
   d) auxiliary moisture separators;
   e) water extraction grooves;
   f) turbine stage drains;
   g) steam piping design.

2. Explain the adverse effects of moisture in the steam passing through a turbine.

3. Explain how the following are utilized to reduce the adverse effects of moisture:

   a) stage type;
   b) shroudless moving blades;
   c) erosion shields.

4. Explain how steam expansion is compensated for in a large turbine unit.

5. Explain how axial thrust is handled or minimized.

6. Explain how the following compensate for axial differential expansion:

   a) blade clearance;
   b) double casing;
   c) carrier ring;
   d) drum rotor;
   e) heatup rate.

7. Explain how the following minimize thermal stresses:

   a) double casing;
   b) carrier rings;
   c) drum rotor;
   d) expansion loops;
   e) expansion joints.

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

   a) moisture;
   b) axial thrust;
   c) steam expansion;
   d) differential axial expansion;
   e) thermal stresses in casing, rotor and steam piping.

334.00-3 The Steam System

1. Given a schematic diagram of a typical steam system label the following components:

    a) steam generator;
    b) safety valves;
    c) steam reject valves;
    d) balance header;
    e) steam isolating valve (boiler stop valve);
    f) emergency stop valve;
    g) governor steam valve;
    h) high pressure turbine;
    i) moisture separator;
    j) reheater;
    k) intercept valves;
    l) release valves;
    m) low pressure turbines
    n) reheat emergency stop valve (if applicable);
    o) exhausts to the condenser;
    p) main steam strainers.

2. List the major functions of each of the components in 1 and explain the basic construction of each component.

3. Explain how turbine speed is controlled after connecting the generator to the grid.

4. Explain the consequences of a failure of each valve listed below (consider for each valve two cases: the valve fails to the open position and will not shut; the valve fails to the shut position and will not open):

    a) steam generator safety valve;
    b) steam reject valve;
    c) emergency stop valve;
    d) governor steam valve;
    e) intercept valve (with/without) RESV;
    f) release valve.

5. Explain the essential difference between a safety valve and relief valve.

234.00-2 Steam Control To The Turbine

1. Explain how the steam control valves respond the the following:

    a) reactor trip;
    b) turbine trip;
    c) load rejection;
    d) normal startup and loading.

2. Explain how steam generator pressure is related to heat transport system temperature and power level.

3. Explain "poison prevent" operation including:

   a) when is it used?
   b) why is it necessary?
   c) where does the steam go?
   d) how long can it be maintained?

4. Explain how the Emergency Stop Valve and Governor Steam Valve are used to control steam to the turbine on a unit start-up.

5. Explain why the closure of the Emergency Stop Valve alone cannot guarantee the safety of the turbine on a fault condition.

6. Explain the difference between the following:

   a) steam release valve;
   b) reheater blow-off valve;
   c) reheater bursting disc.

7. Explain the advantages of dumping steam to atmosphere, a reject condenser or to the main condenser when the turbine is not available.

8. Discuss in writing the functions of extraction steam check valves including:

   a) why they are air motor assisted;
   b) why they are put on HP extraction lines;
   c) why they are put on LP extraction lines;
   d) why they are installed on the extraction steam lines to the deaerator;
   e) why some LP extraction lines don't have them.

334.00-7 The Condenser

1. State the major functions of the main condenser.

2. Identify and state the function of the following condenser components:

   a) inlet and outlet water box;
   b) air ejector suction;
   c) steam heating lanes;
   d) turbine exhaust;
   e) hotwell;
   f) sagging plates;
   g) tubes;
   h) shell;
   i) tube sheets;
   j) stay bars.

3. Explain why modern condensers have an expansion piece fabricated into the condenser shell.

4. Explain why air must be removed from the condenser.

5. Explain the operation of a steam jet air ejector.

6. Explain what a hogging ejector is and when it is used.

7. Explain the operation of an air extraction vacuum pump.

8. State the function and operation of the following condenser cooling water system components:

   a) course screen;
   b) band screen;
   c) chlorine injection.

9. State the possible causes of a loss or decrease of condenser vacuum.

10. Explain the effect of the season of the year on condenser vacuum.

11. State what parts of the turbine and condenser must be evacuated prior to startup.

12. Explain the method of drawing and maintaining a vacuum in the condenser.

13. For the CCW system, state:

   a) the maximum outflow temperature and _T allowed;
   b) the reason for these limits.

PI 25-6 The Mollier Diagram and the Turbine Processes

1. Sketch a Mollier diagram from memory. Label the following on your sketch:

   (a) constant enthalpy lines
   (b) constant entropy lines
   (c) saturation line
   (d) constant temperature lines
   (e) constant pressure lines
   (f) constant moisture content lines
   (g) constant degree of superheat lines

2. On the sketch of a Mollier diagram that you have drawn, illustrate these turbine processes:

   (a) expansion in the high pressure turbine
   (b) moisture separation
   (c) reheat
   (d) expansion in the low pressure turbine

3. Explain how moisture separation and reheat each:

   (a) increase the enthalpy of the steam at the LP turbine inlet
   (b) reduce the moisture content of the steam at the LP turbine outlet

4. Define throttling and, using a Mollier diagram, explain how throttling of the steam supplied to the turbine affects:

   (a) the pressure, temperature and moisture content of the steam at the turbine inlet
   (b) the amount of heat which can be converted into mechanical energy by the turbine.

334.00-8 Feedwater Heating Systemm - I

1. State the reasons why feedheating is necessary.

2. Given a sectional diagram of a feedheater, identify and give the function of the:

   a) extraction steam inlet;
   b) vent;
   c) tubes;
   d) baffles;
   e) drain outlet;
   f) feedwater inlet;
   g) feedwater outlet;
   h) shell.

3. Explain how the deaerator operates to heat feedwater and remove air.

4. Explain why feedheaters are vented to the condenser.

5. Explain why the deaerator has electric heaters.

334.00-9 Feedwater Heating System - II

1. List the sources of heating in a typical feedwater system and discuss in which components these sources are used.

2. For each feedheating system heat exchanger listed below, explain the source of heat energy:

   a) stator water cooler;
   b) gland exhaust condenser;
   c) air ejector condenser;
   d) drain cooler;
   e) low pressure feedheater;
   f) deaerator - storage tank;
   g) high pressure feedheater;
   h) preheater.

3. Explain why heat rejected to the generator hydrogen or stator water cooling systems can not be used to heat feedwater.

4. Explain why the deaerator - storage tank has three sources of heat energy.

234.00-7 Feedwater Control and Operation

1. Explain the operation and control of the following valves:

   a) feedwater regulating valve;
   b) deaerator storage tank level control valves;
   c) normal makeup valve;
   d) emergency makeup valve;
   e) condensate extraction pump recirculation control valve;
   f) boiler feed pump recirculation valve.

2. State and explain the sequence of events which occur in the feedwater system on an increase or decrease in power level including:

   a) level in steam generators;
   b) feedwater regulating valve;
   c) deaerator storage tank valve;
   d) deaerator storage tank level control valves;
   e) hotwell level;
   f) makeup valves;
   g) pump requirements;
   h) extraction steam flow;

3. Given a line diagram of a typical feedwater system label the following:

   a) condenser hotwell(s);
   b) condensate extraction pumps;
   c) condensate recirculation line;
   d) makeup water connections;
   e) gland exhaust condenser;
   f) air ejector condenser (if applicable);
   g) stator water cooling heat exchanger (if applicable);
   h) drain coolers;

i) LP feedheaters;
j) deaerator storage tank control valves;
k) deaerator;
l) boiler feedpump;
m) feedpump recirculation line;
n) HP feedheaters;
o) preheater (if applicable);
p) steam generators;
q) feedwater control valves;
r) extraction steam connections;
s) main steam connections;
t) feedheater drains;

4. Explain what is meant by a "cascading feedheating system".

5. Explain th reason for parallel arrangement of feedheaters and duplication of pumps and control valves.

6. Explain how extraction steam flow to a feedheater is self-regulating.

7. Explain the need for and method of level control in a feedheater.

8. List and briefly decsribe three effects of scale forming substances entering the secondary side of a steam generator.

9. List three main sources of scale forming substances in the secondary side of a steam generator. Briefly describe how each is a contributor to scale.

10. State the reason the feed system pH is kept at 8.8 - 9.2 which is lower than optimum for steel.

11. Briefly describe the harmful effects of Chloride; Oxygen; Ammonia; Carbon Dioxide in the feed system.

12. Define carryover, briefly describe two types.

13. State where and why the following would be added to a secondary heat transport system

    a) Morpholine
    b) Hydrazine (2 reasons)
    c) Cyclohexylamine

14. Briefly describe the purpose of blowdown for the steam generator.

15. Briefly describe using either written explanation or diagram, the difference between intermittent and continuous blowdown.

PI 25-7 Efficiency of the CANDU Turbine Cycle

1. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by raising boiler pressure.

   (b) State the main limitation on the improvement in (a).

2. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by lowering condenser pressure.

   (b) State two limitations on the improvement in (a).

3. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by superheating in the boiler.

   (b) State the main limitation on the improvement in (a).

4. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by:

      (i) reheating between the high and low pressure turbines

      (ii) using extraction steam for feedheating.

   (b) State the main limitation on each improvement in (a).

   (c) State two practical benefits of each improvement in (a).

5. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by moisture separation.

   (b) State the practical benefit of moisture separation.

234.00-3 Steam Valve Hydraulic Control

1. Define "dead time" and "reservoir effect".

2. Explain the advantages of Fire Resistant Fluid over turbine lubricating oil as a hydraulic fluid for steam operators.

3. Explain why a hydraulic fluid is used for governor steam valve actuation.

4. Explain the problems associated with an FRF control system.

5. Discuss the need for fluid purity and cleanliness in an FRF Control System.

234.00-4 Governor Operation

1. Explain the functions of the turbine governor.

2 Explain the control of turbine speed:

   a) on startup below operating speed;
   b) on startup at operating speed but not connected to the hydro grid;
   c) when connected to the hydro grid.

3. Explain what is meant by "speed droop", why it is built into a governor system and how it aids the stability of a generator operating in parallel with other generators.

4. Explain how the active power (real power) out of a generator is increased, for a large turbine/generator supplying the Ontario Grid System.

5. Explain how the speed of a turbine is varied when the generator is not connected to the grid.

234.00-5 Turbine Governors

1. Explain how the governor and steam control valves on the turbine at your station respond to a load rejection.

2. For a mechanical hydraulic governor, explain the sequence of events on an overspeed following a load rejection including the operation of the:

   a) electric anticipator;
   b) main governor;
   c) auxiliary gear;
   d) speeder gear;
   e) emergency trip plunger.

3. Explain the advantages on an electrical hydraulic governor over a mechanical hydraulic governor.

4. Explain the function of the load limiter.

334.00-10 The Lubricating Oil System

1. Explain the purpose of and state when the following pumps and components are used:

   a) main lube oil pump;
   b) auxiliary lube oil pump;
   c) dc emergency lube oil pump;
   d) turning gear lube oil pump;
   e) jacking oil pump;
   f) the lube oil purifier.

2. Explain why control oil is at a higher pressure than bearing oil.

3. Discuss the lube oil system including the purpose of:

   a) pumps;
   b) strainers;
   c) filter;
   d) purifier;
   e) coolers;
   f) pressure reducing valve.

334.00-11 The Turning Gear

1. Define "shaft hog" and "shaft sag". Explain how and why each of these occur and what can be done to avoid these conditions.

2. State and explain the sequence of events prior to starting the turning gear motor.

3. Explain why the turning gear operates at 10-30 rpm rather than some lower speed (say 3-4rpm).

4. Explain how the turning gear and jacking oil pump are controlled on a unit startup and shutdown.

234.00-6 Turbine Operational Problems

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) moisture carryover;
   f) blade failure;
   g) expansion bellows failure;
   h) bearing failure.

234.00-8 Factors Limiting Startup and Rates of Loading

1. Explain the reasons for each of the following:

   a) COLD, WARM and HOT start up 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 starter 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 motor temperature;
   h) shaft eccentricity;
   i) vibration;
   j) critical speeds.

334.00-13 Ontario Hydro Nuclear Turbine Units

1. Classify a cross-sectional view of a turbine as to:

   a) number of HP and LP turbines;
   b) single or double flow turbines;
   c) type of compounding;
   d) moisture separators;
   e) reheaters.

2. Classify the turbine unit for your station using the twelve point classification system. You will be able to explain what is meant by each item. (Those who are not assigned to a generating station will be able to classify the Pickering Unit as it is more typical of large units).

Turbine, Generator & Auxiliaries - Course 334

THE STEAM TURBINE

The basic principle on which all steam turbines operate is that the heat energy which is possessed by the steam can be converted to kinetic energy of the steam. The heat energy of the steam is reduced and the velocity of the steam is greatly increased. If this high velocity steam is directed against a blade, the force exerted by the steam can be used to move the blade. The device used to convert the heat energy of the steam to kinetic energy is called a nozzle. Figure 1.1 shows how the high velocity steam leaving fixed nozzles can be used to move a series of blades. The steam can then be passed through a second set of fixed nozzles.

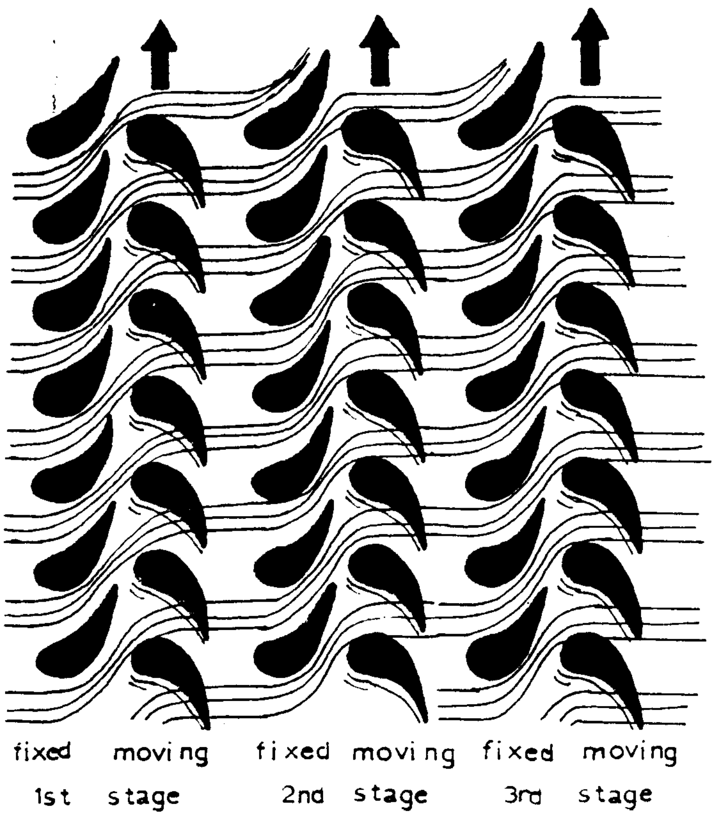


Figure 1.1

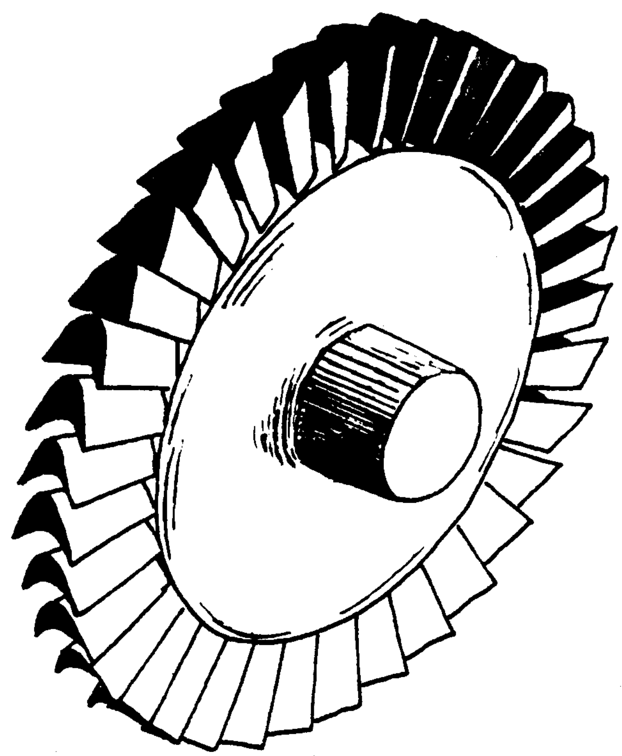


Figure 1.2

If the blades were arranged on a blade wheel such as shown in Figure 1.2, the high velocity steam leaving the nozzle could be used to turn the wheel. A set of fixed blade nozzles and moving blades is known as a stage and it is common to utilize a number of stages in a turbine to extract the useful heat energy in the steam.

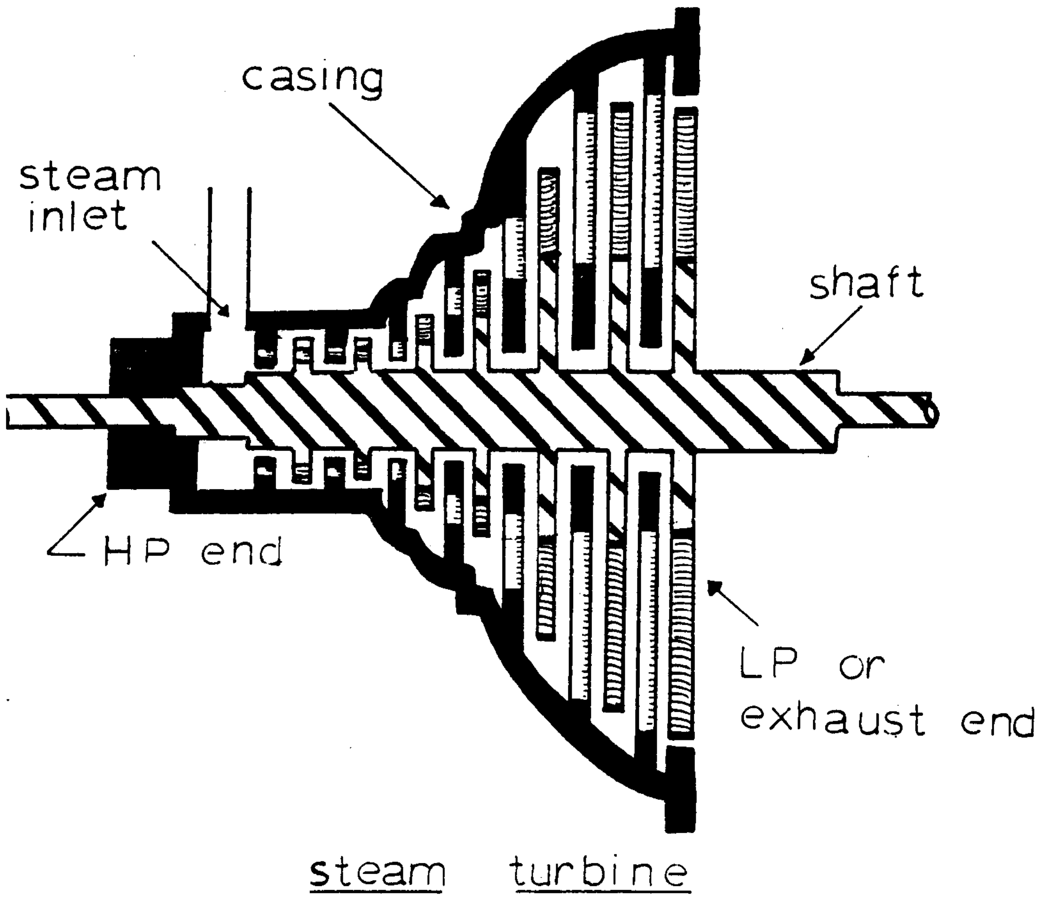


Figure 1.3

Figure 1.3 is a simplified section view of a turbine. The steam turbine consists of a single rotating shaft which has a number of blade wheels attached to it. The steam passing through the turbine is contained within a casing. The casing is usually split into an upper and lower half which are bolted together. This enables the upper half to be raised for maintenance. Attached to the casing are diaphragms which support the fixed blade nozzles. Figure 1.4 shows the construction of a typical diaphragm. The upper half is attached to the upper casing and the lower half to the lower casing. You will note that the turbine casing gets progressively larger as the steam goes from the high pressure end to the low pressure end. This is due to the expansion of the steam as the pressure is reduced. Steam entering the high pressure end of a modern nuclear turbine unit is typically around 250°C and 4000 kilopascals. At this temperature and pressure, one kilogram of steam occupies .05 cubic meters. The steam leaving the turbine unit and entering the condenser

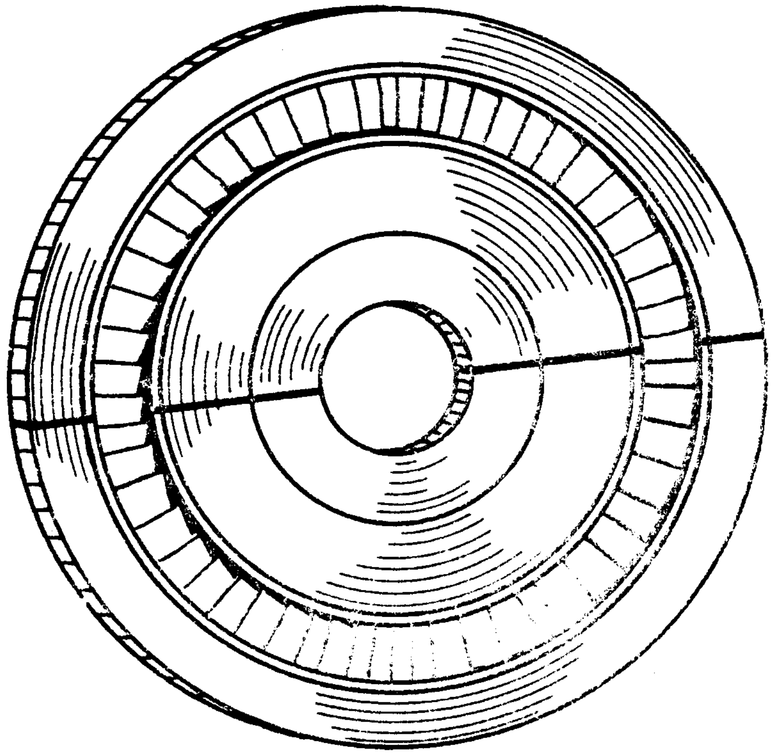


Figure 1.4

is typically around 35°C and 5.6 kilopascals. At this temperature and pressure, one kilogram of steam occupies 25.2 cubic meters. The steam expands roughly five hundred times from the inlet conditions to the exhaust conditions and provisions, such as increasing the casing size, must be made to handle this increased volume.

As the steam passes through the turbine, heat energy is converted to kinetic energy and the steam gives up a portion of its latent heat of vapourization. As this occurs, a portion of the steam condenses into minute water droplets which are carried along with the steam. When these droplets strike the moving blading, the effect is similar to sandblasting and the high velocity water droplets erode the turbine blading.

In addition, the presence of moisture in the steam passing through a turbine has a pronounced effect on turbine efficiency. Since the water droplets are not moving as fast as either the steam or the blades, the back side of the blades are continuously running into the water droplets. This results in a retarding force being exerted against the blades and decreases turbine efficiency.

Although modern turbines incorporate a number of features to remove the moisture within the turbine and to harden blading against moisture erosion, wet steam with greater than approximately 10% moisture cannot be tolerated. When the percent moisture reaches about 10%, the wet steam must be exhausted from the turbine. Unfortunately, with saturated steam at the inlet to the turbine, the steam reaches 10% moisture content before all of the useful heat energy is extracted from the steam. Because of this it is common to exhaust the steam from one turbine, pass the steam through a moisture separator where the moisture content of the steam is reduced to near zero, and then direct the steam into a second turbine. This second turbine usually exhausts to the condenser. While the steam is removed from the turbine for moisture separation, it is common to pass it through a reheater. This reheater superheats the steam going to the low pressure turbine by using main steam through a heat exchanger. This reheating of high pressure turbine exhaust steam improves the efficiency of the turbine unit. Figure 1.5 shows a section view of a turbine unit consisting of two turbines with a moisture separator and reheater between them.

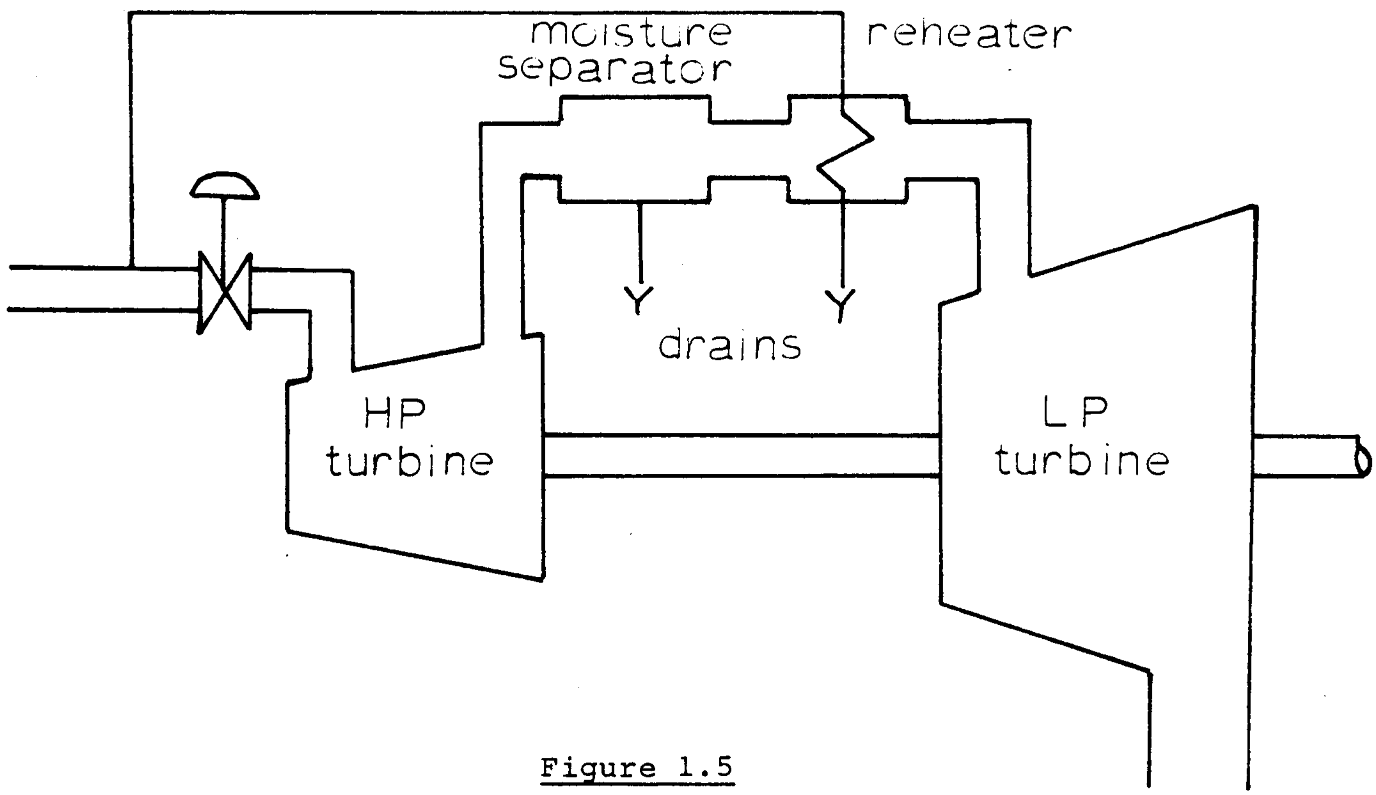


Figure 1.5

By convention, the turbine which receives steam directly from the steam generator is called a high pressure turbine or HP turbine. The turbine which exhausts to the condenser is called a low pressure turbine or LP turbine. The turbine unit shown in Figure 1.5 would be described as containing one high pressure turbine, a moisture separator, a reheater and one low pressure turbine.

Thus far, we have talked about turbines in which the steam enters at one end, expands through the turbine and exhausts at the opposite end. Such turbines are called single flow turbines because the steam flow is in one direction. Modern turbines may contain blading which allows substantial pressure drops across the moving blading. These large pressure drops tend to push the blade wheels from the high pressure side to the low pressure side. As steam flows become large and pressure drops across moving blading become significant, the force exerted on the rotor from the high pressure end may become difficult to handle. To account for this force, it is common to build double flow turbines such as shown in Figure 1.6.

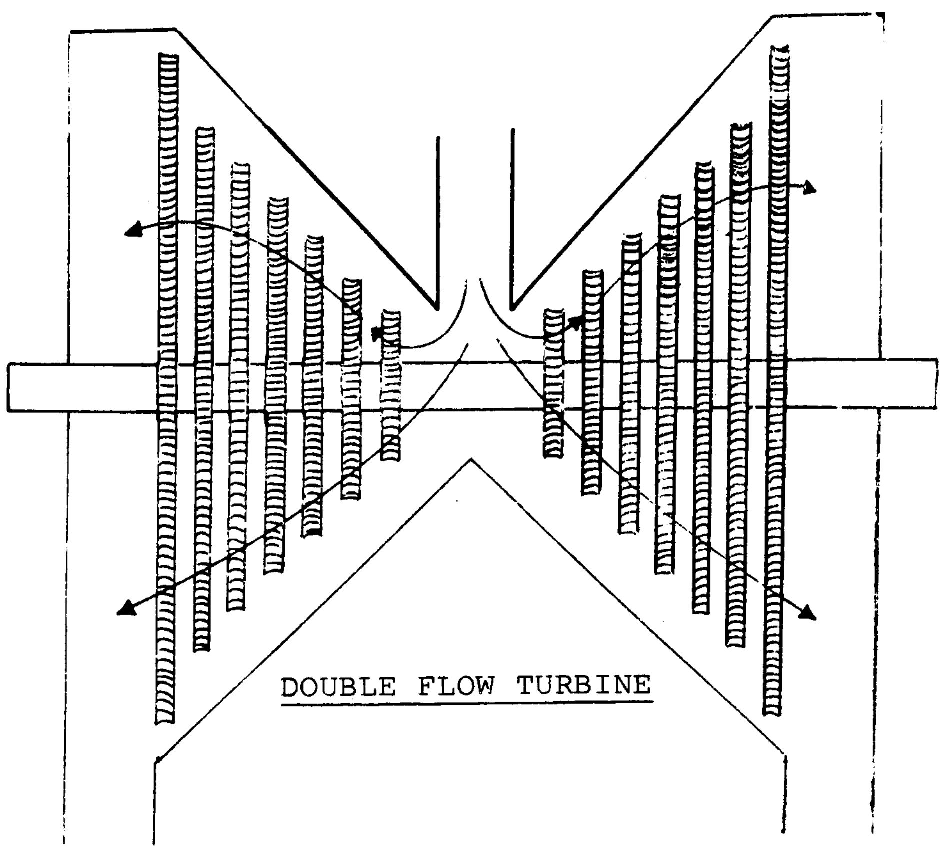


Figure 1.6

Steam enters the turbine in the middle of the casing and expands outward in both directions before exhausting at the ends of the turbine. In a double flow turbine, the force exerted by the pressure drop on one end of the rotor is exactly balanced by the force in the opposite direction exerted on the other end of the rotor. Double flow turbines are found in the majority of large turbine units.

Earlier, we discussed the extremely large increase in the volume of steam as it expanded through the turbine. In large turbines it is usually not possible to accommodate all of the steam in only one low pressure turbine. Normally one high pressure turbine will exhaust to two or more low pressure turbines. Figure 1.7 shows a turbine unit typical of those installed at Pickering or Bruce Generating Stations. In these turbine units, one double flow high pressure turbine supplies three double flow low pressure turbines. The arrangement of all the turbines in a turbine unit on a common shaft is known as a tandem compounded turbine unit. All turbine units in the Nuclear Generation Division of Ontario Hydro are tandem compounded.

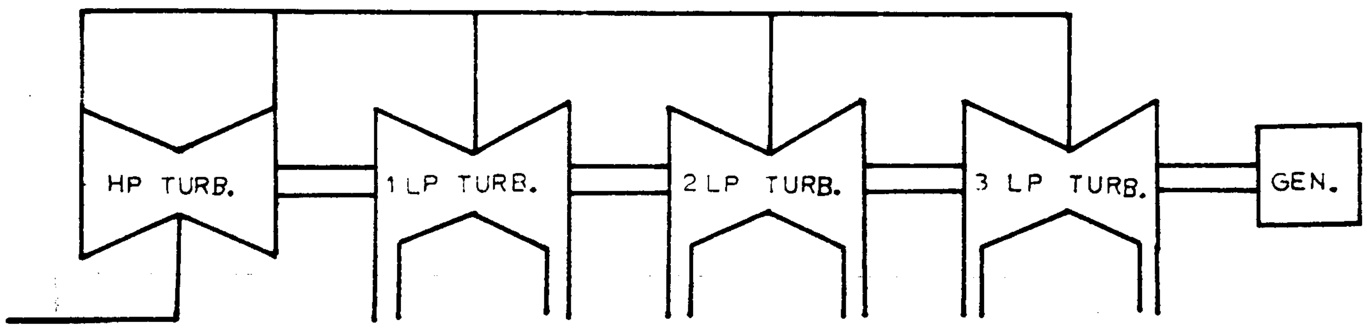
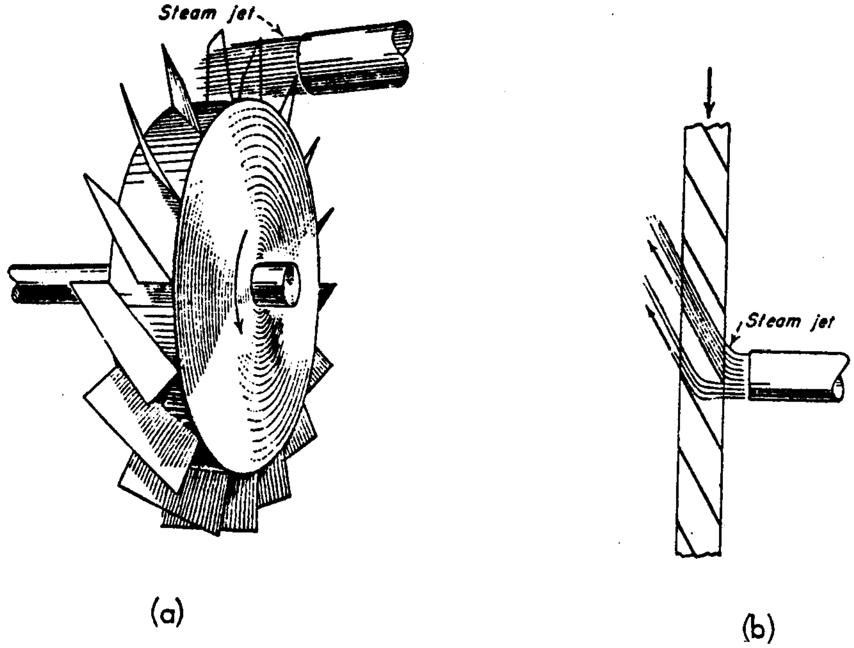


Figure 1.7

As mentioned earlier, the basic principle upon which all steam turbines operate is that the heat energy which is possessed by the steam can be converted to kinetic energy of the steam. The heat energy is reduced and the velocity is greatly increased. This high velocity steam can be used to turn a turbine wheel and create shaft mechanical power. Steam turbines can be divided into two principle types: reaction turbines and impulse turbines. The basic difference between the two types is in which part of the stage steam heat energy is converted to steam kinetic energy.

THE IMPULSE TURBINE

The first commercial steam turbine was manufactured in 1882 by Gustaf De Laval as a prime mover for a cream separator. The prime mover for De Laval's turbine was a single stage turbine with only one nozzle as is shown in Figure 1.8.



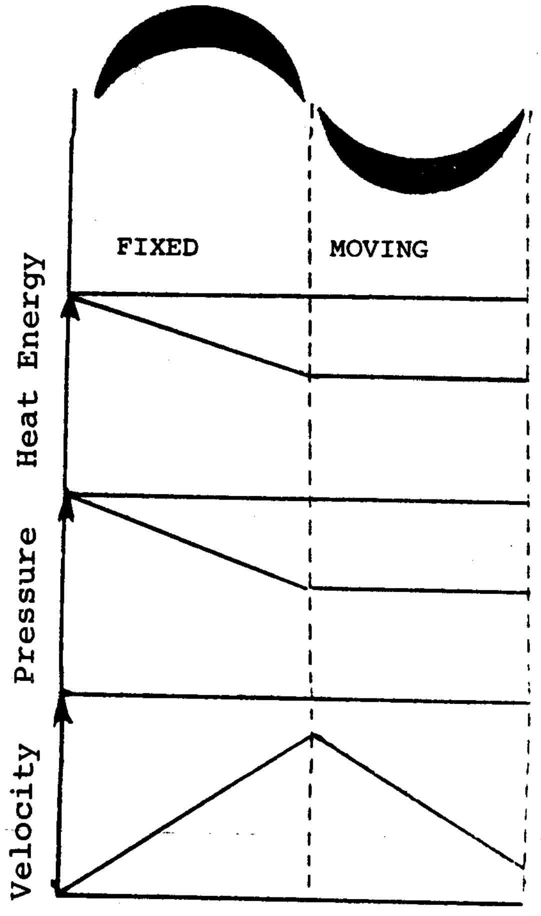
De Laval's Simple Impulse Turbine

Figure 1.8

The nozzle is shaped such that the heat energy of the steam is converted to kinetic energy. The high velocity steam is directed against the blades on the outer rim of a moving wheel. As the steam strikes the blades, they are moved out of the jet of steam. Changing the momentum of the steam delivers an impulse to the blade and the blade moves as a result of this impulse. Turbines which use this principle for converting steam kinetic energy to mechanical energy are called impulse turbines. Impulse stages of the type we are discussing are often called Rateau stages after their inventor.

What physically happens as the steam passes through one stage of an impulse turbine? One stage of an impulse turbine is shown in Figure 1.9. In the nozzle, the heat energy of the steam decreases and the steam velocity increases. Since pressure is one of the factors influencing the heat energy of the steam, the pressure decreases across the nozzle as the heat energy decreases.

In the moving blades, the high velocity of the steam is reduced as steam kinetic energy is converted to blade kinetic energy. The moving blade is shaped only to change the steam direction and so no conversion of heat energy to kinetic energy takes place in the moving blades. As a result, heat energy and pressure are constant across the moving blades of an impulse turbine.

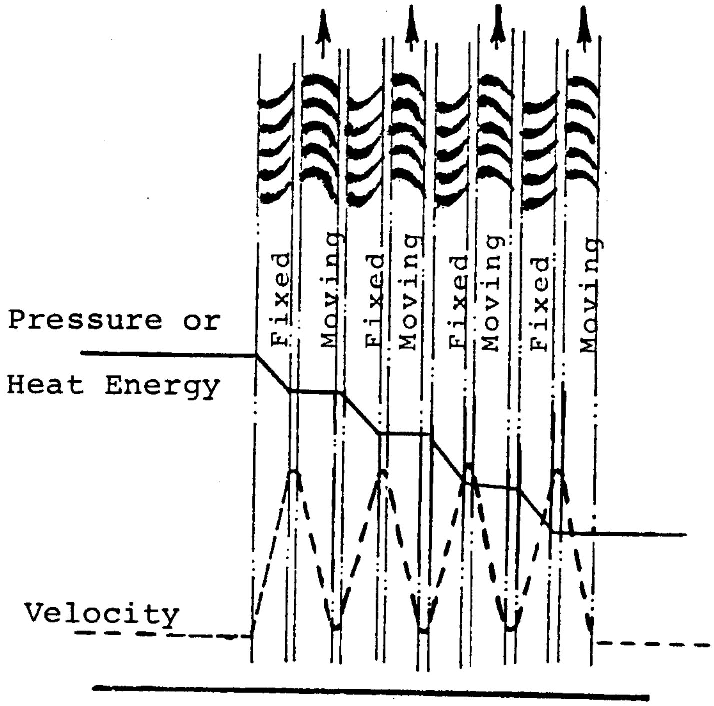


Impulse Stage

Figure 1.9

Although single stage turbines of the type De Laval manufactured are not uncommon, they are rather inefficient and as a result, are usually small. Turbines for use in large generating stations are constructed with many stages (ten to fifteen stages being typical). In addition, nozzles are usually mounted around 360° of the blade wheel so that power is delivered to the moving blades throughout the entire revolution.

Figure 1.10 shows the pressure, velocity and heat energy changes across a four stage impulse turbine. Note that since heat energy is converted to steam kinetic energy only in the nozzles, the heat energy and steam pressure decrease only in the nozzles.



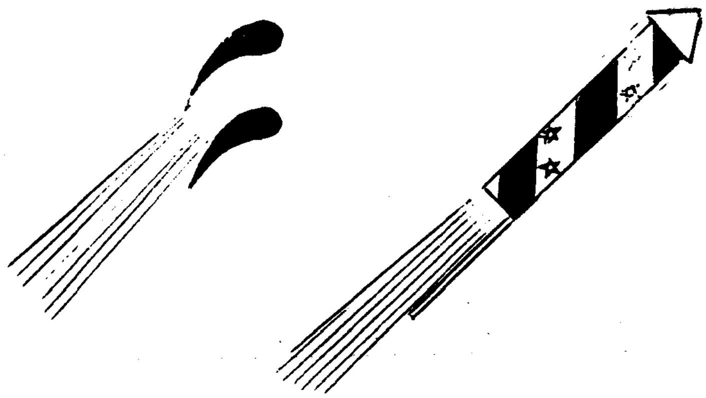
Four Stage Impulse Turbine

Figure 1.10

THE REACTION TURBINE

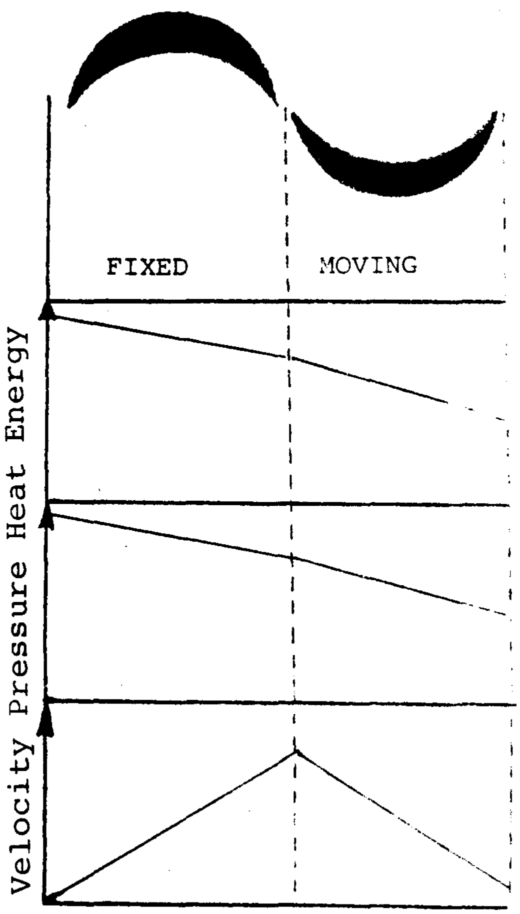
A few years after De Laval's development of the impulse turbine, Sir Charles A. Parsons developed the first reaction turbine. In the reaction turbine, the moving blades are shaped to allow the steam to expand as it leaves the moving blades. The blade moves away from the steam as a reaction to the steam expansion. The effect is exactly the same as a rocket moving in reaction to the expansion of gases.

In a practical reaction turbine, the moving blade is driven by a combination of this reaction effect and the impulse effect we have just examined. Since virtually all turbines utilize the impulse effect of high velocity steam to produce at least some of the force on the moving blades, the division between impulse and reaction blades is never very clear cut. Generally, if any force is produced by reaction, the stage is called a reaction stage. If no force is produced by reaction, the stage is called an impulse stage.



The Reaction Effect

Figure 1.11

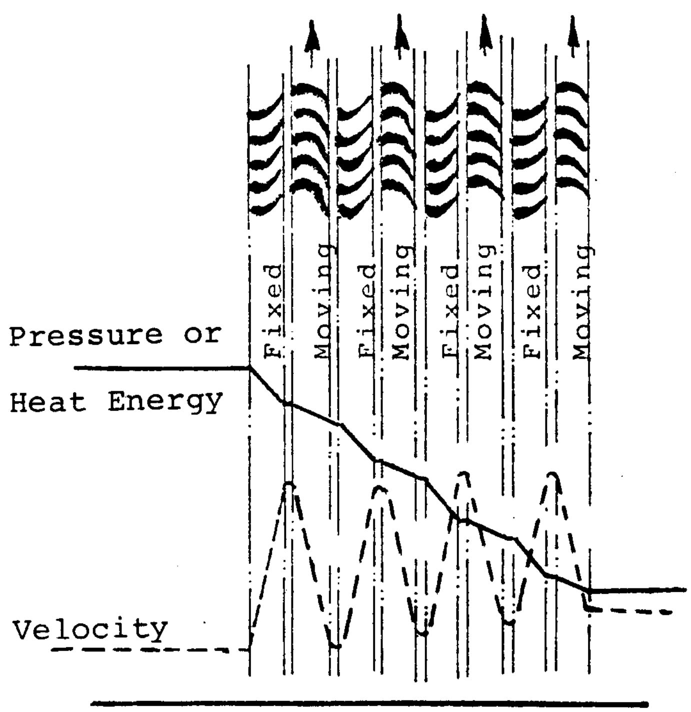


Reaction Stage

Figure 1.12

What physically happens as the steam passes through one stage of a reaction turbine? One stage of a reaction turbine is shown in Figure 1.12. The fixed blades are nozzles which work exactly as they do in the impulse stage. The heat energy and pressure of the steam decrease and the steam velocity increases. In the moving blades, the high velocity steam produces an impulse on the moving blades.

However, the shape of the moving blade allows steam expansion as the steam passes through this blading. This steam expansion lowers steam pressure and heat energy in the moving blade while the steam expansion produces a reaction effect and the blades move away from the expanding steam. In the reaction stage, steam heat energy is converted to steam kinetic energy in <u>both</u> the fixed and moving blades. In the impulse stage, steam heat energy is converted to steam kinetic energy in <u>only</u> the fixed blades.



<u>Four Stage Reaction Turbine</u>

<u>Figure 1.13</u>

Figure 1.13 shows the pressure, velocity and heat energy changes across a four stage reaction turbine. Note that since heat energy is converted to steam kinetic energy in both the fixed nozzles and moving blades, the heat energy and pressure decrease in both fixed and moving blades.

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.

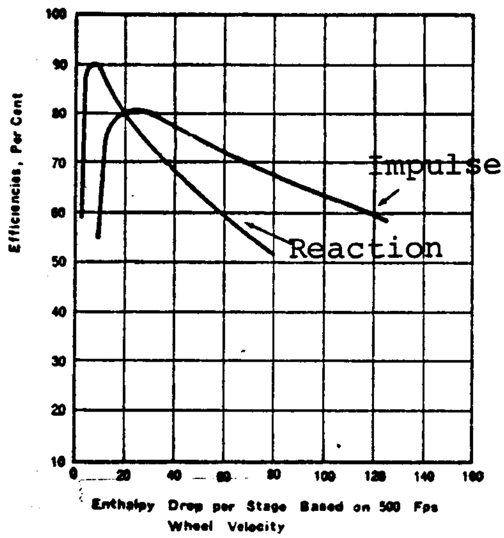


Figure 1.14

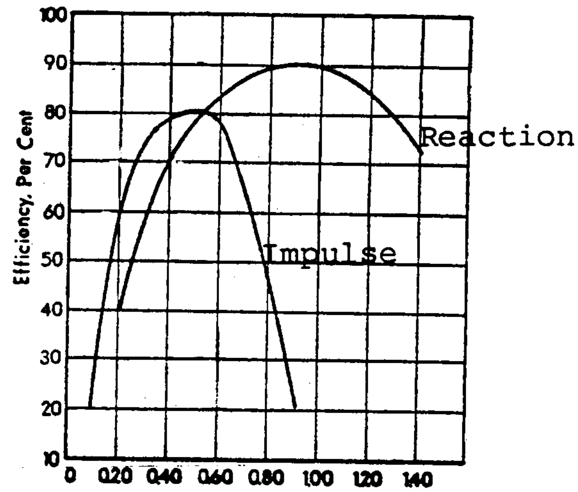


Figure 1.15

Efficiency and Energy Drop Per Stage: Figure 1.14 shows how the efficiency of a reaction turbine and an impulse turbine vary as the heat energy (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 per stage low. In some instances, it is difficult to keep the enthalpy drop across a stage within the proper range for a reaction stage to give good efficiency. In these cases, an impulse stage is used to keep high efficiency with a large enthalpy drop.

Velocity Ratio: Velocity ratio is the ratio of how fast the blade is moving to how fast the steam is moving. Each type of stage has a different velocity ratio at which it runs most efficiently. Figure 1.15 shows the relationship between efficiency and velocity ratio. The impulse stage is most efficient when blade velocity is about one half of steam velocity. On the other hand, the reaction stage is most efficient when blade velocity is only slightly less than steam velocity.

As blade wheels become larger to accommodate the high volume of steam in modern large turbine units, the blade tangential velocity increases and the velocity ratio increases. As a result, the reaction turbine becomes more attractive since it develops its maximum efficiency at higher velocity ratios. Large turbines and particularly large low pressure turbines, are commonly reaction turbines.

Moisture Effects: Reaction turbines are more sensitive to the effect of water droplets decreasing efficiency by impact with the moving blades. Typically for a given percent moisture in the steam passing through a stage, the reaction turbine will suffer a loss in efficiency almost twice as great as an impulse turbine. 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 reactor turbine, the need to keep the moisture content low can be readily appreciated.

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 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 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.

PRESSURE GRADIENT THROUGH A TURBINE UNIT

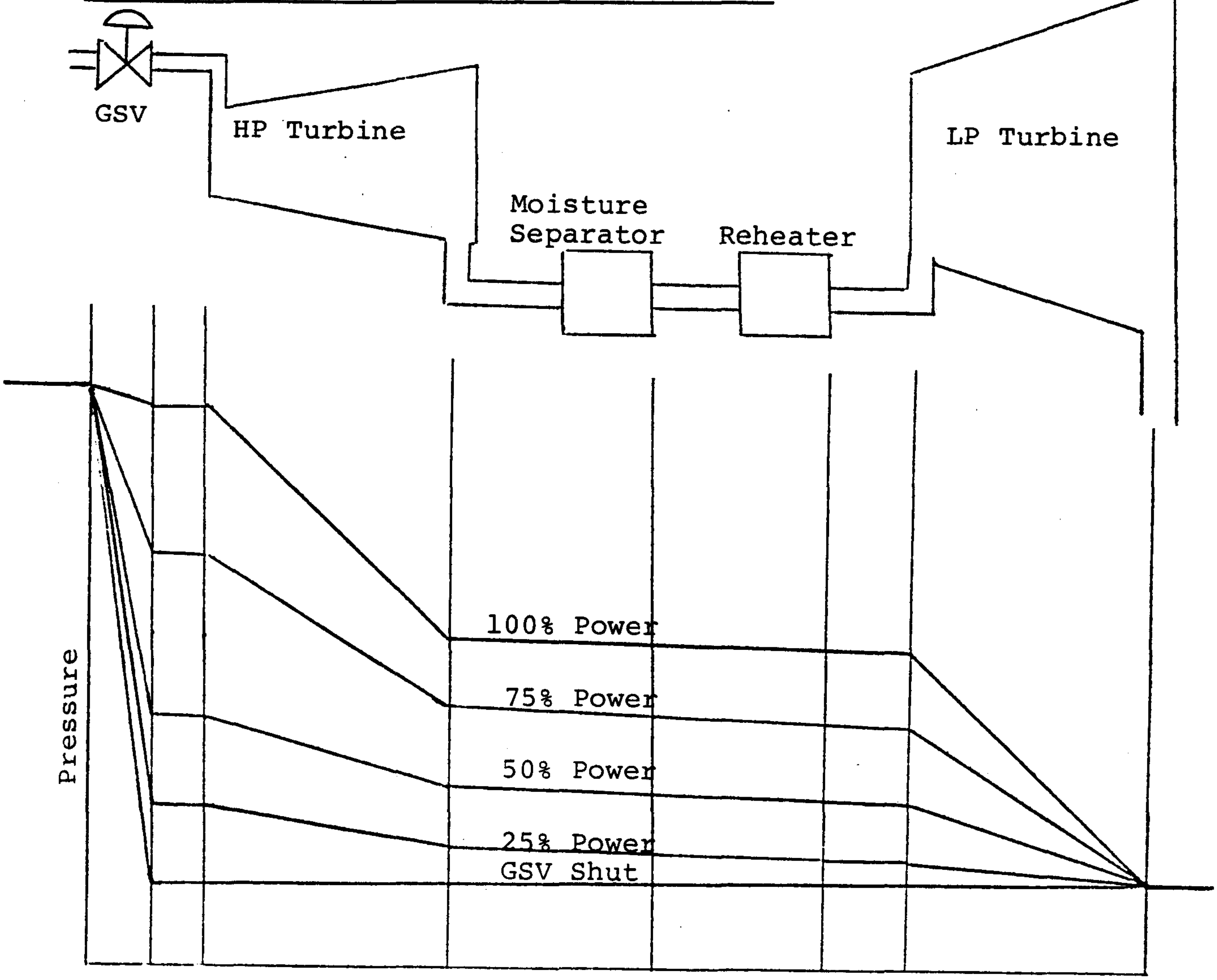


Figure 1.16

The turbine unit sits between two large vessels (the steam generator and the condenser) in which the temperatures and pressures are reasonably fixed. Between the steam generator and the condenser the temperature must drop from 250°C to 35°C and the pressure must drop from 4000 kPa(a) to 5.6 kPa(a).

Consider the case of a turbine unit such as shown in Figure 1.16. The turbine is shutdown and the condenser is being evacuated with the air extraction. Since there are no shut valves between the condenser and the steam admission valve, the turbine will be evacuated along with the condenser. With the steam admission valve shut, this valve has steam generator conditions on one side and the condenser vacuum on the other.

If the steam admission valve is cracked open to begin rolling the turbine, steam will begin to flow through the turbine. The pressure drop across a stage of a turbine is roughly proportional to the square of the steam flow through the stage. Thus at low steam flow rates, the pressure drop through the turbine will be very small and the majority of the pressure drop between steam generator and condenser will occur across the throttled steam admission valve.

As the turbine is brought up to operating speed, synchronized with the grid and loaded, the steam flow will increase. As the steam flow increases, the pressure drop through the turbine will increase. Since the pressure at any point in the turbine is equal to condenser vacuum plus the pressure drop from that point to the condenser, as steam flow increses the pressure throughout the turbine will increase. At the same time the pressure drop across the steam admission valve will decrease as the valve continues to be opened. The increase in the HP turbine first stage pressure is directly proportional to the increase in steam flow and as such, is an excellent indicator of turbine power. Typical pressure gradients for a large CANDU turbine unit are shown in Figure 1.16.

ASSIGNMENT

1. Identify and state the functions of the following turbine components:

   (a) casing
   (b) rotor
   (c) shaft
   (d) blade wheel
   (e) diaphragm
   (f) nozzle
   (g) fixed blade
   (h) moving blade

2. What is meant by the term "turbine stage"?

3. State the two principles by which steam kinetic energy is converted to blade kinetic energy and explain these principles.

4. Why are large power turbines multi-staged rather than having only one large stage?

5. Draw the pressure and velocity changes across a four stage impulse turbine.

6. Draw the pressure and velocity changes across a four stage reaction turbine.

7. Explain the difference between double flow and single flow turbines and explain the advantage of a double flow turbine.

8. What is the purpose of a steam nozzle?

9. The NPD turbine is a 25 MW turbine unit with a single flow HP turbine and a single flow LP turbine. Explain:

   (a) Why the HP turbine is an impulse turbine?

   (b) Why the LP turbine is a reaction turbine?

   (c) How the LP turbine can be single flow?

10. The Pickering NGS-A turbine is a 540 MW turbine unit with a double flow HP turbine and three double flow LP turbines. Explain:

    (a) Why the HP turbine has reaction blades?

    (b) Why the LP turbine has reaction blades?

    (c) Why the LP turbine is double flow?

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 234

TURBINE CONSTRUCTION

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

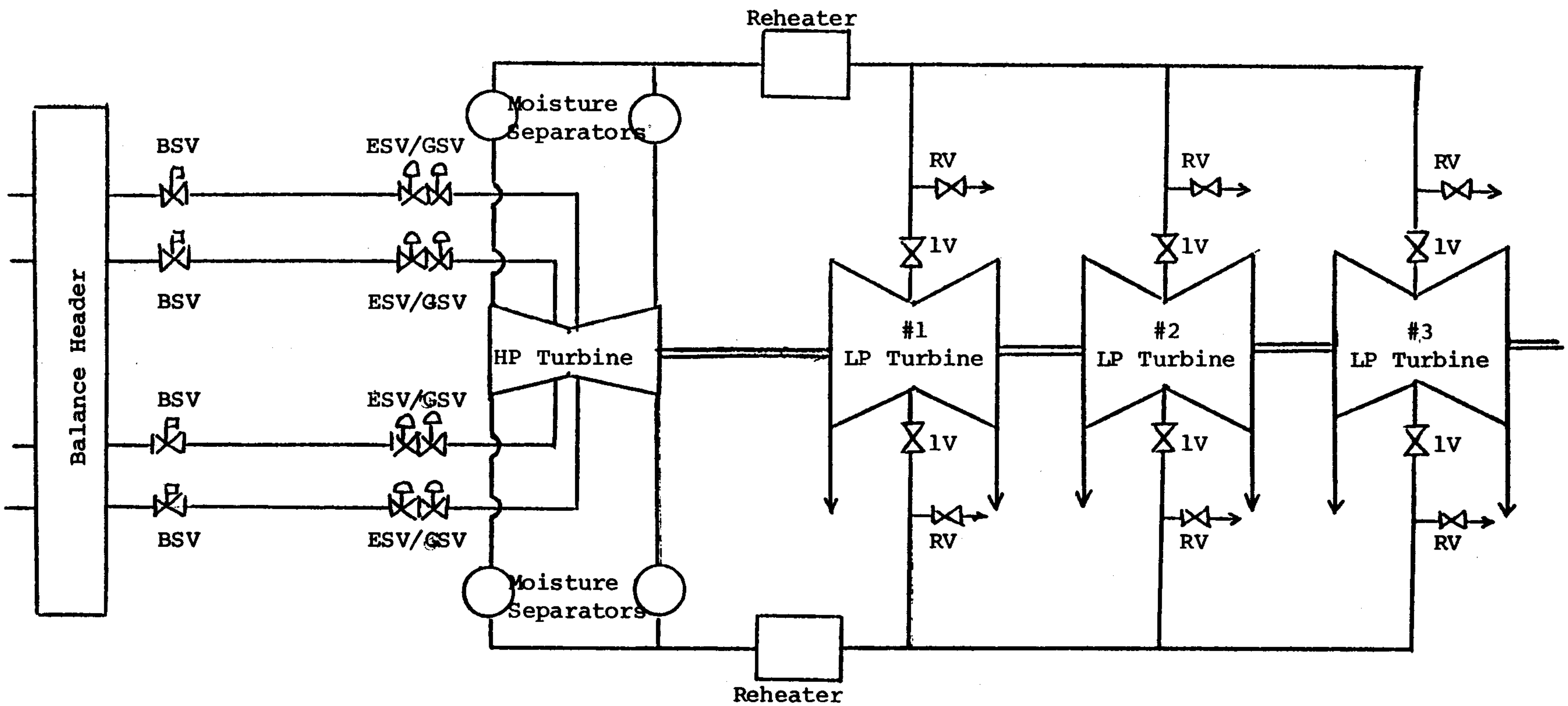
Figure 1.1 shows the steam system and turbine unit typical of a large nuclear generating station. Steam passes through a double flow high pressure turbine and exhausts through four moisture separators and two reheaters to three low pressure turbines in parallel. The four turbines are tandem compounded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

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

1. Moisture content of the steam;

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

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



<u>Typical Large Turbine Unit</u>

<u>Figure 1.1</u>

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

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

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

Moisture Content of Steam

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

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

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

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

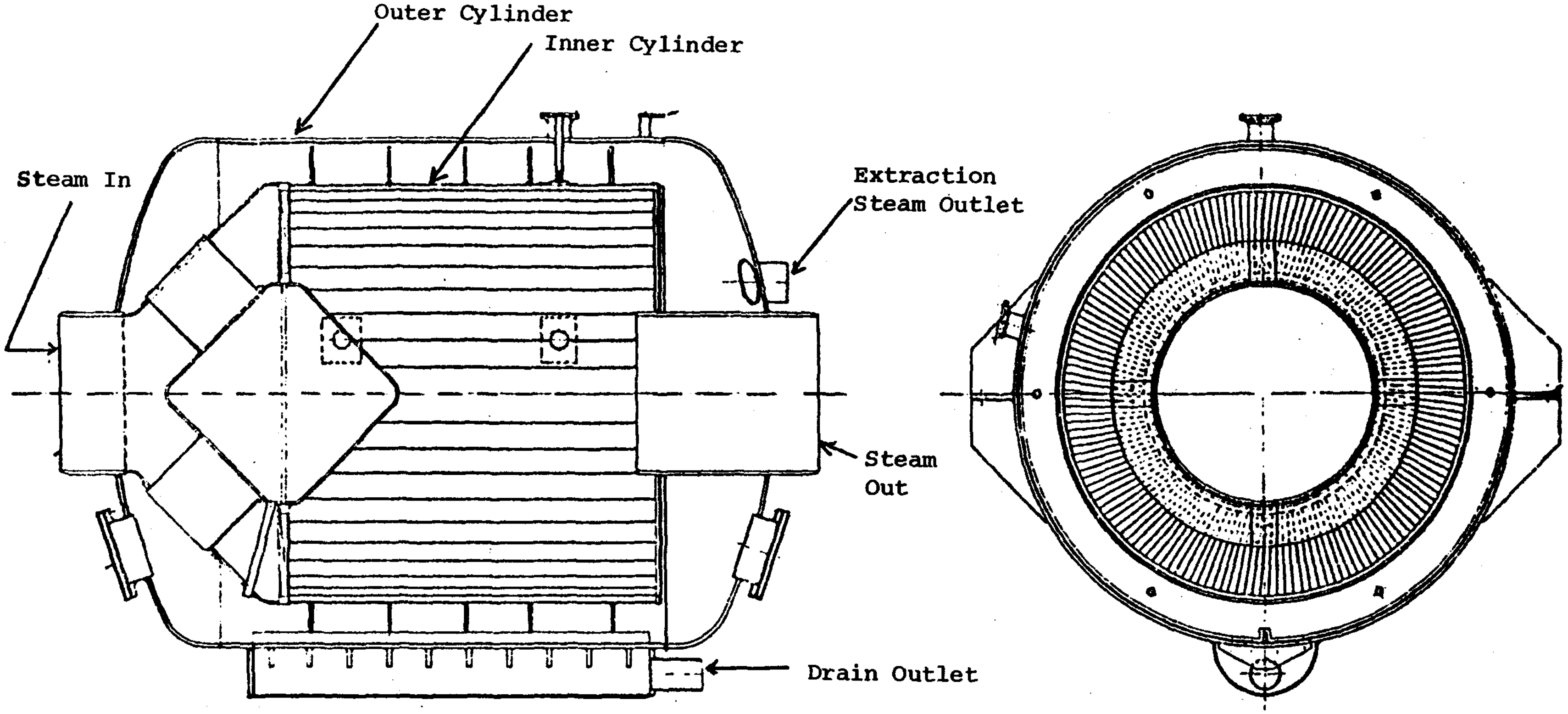
The design features which are used to limit either the moisture content of the steam or the effects of this moisture are:

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

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

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

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



Moisture Separator

Figure 1.2

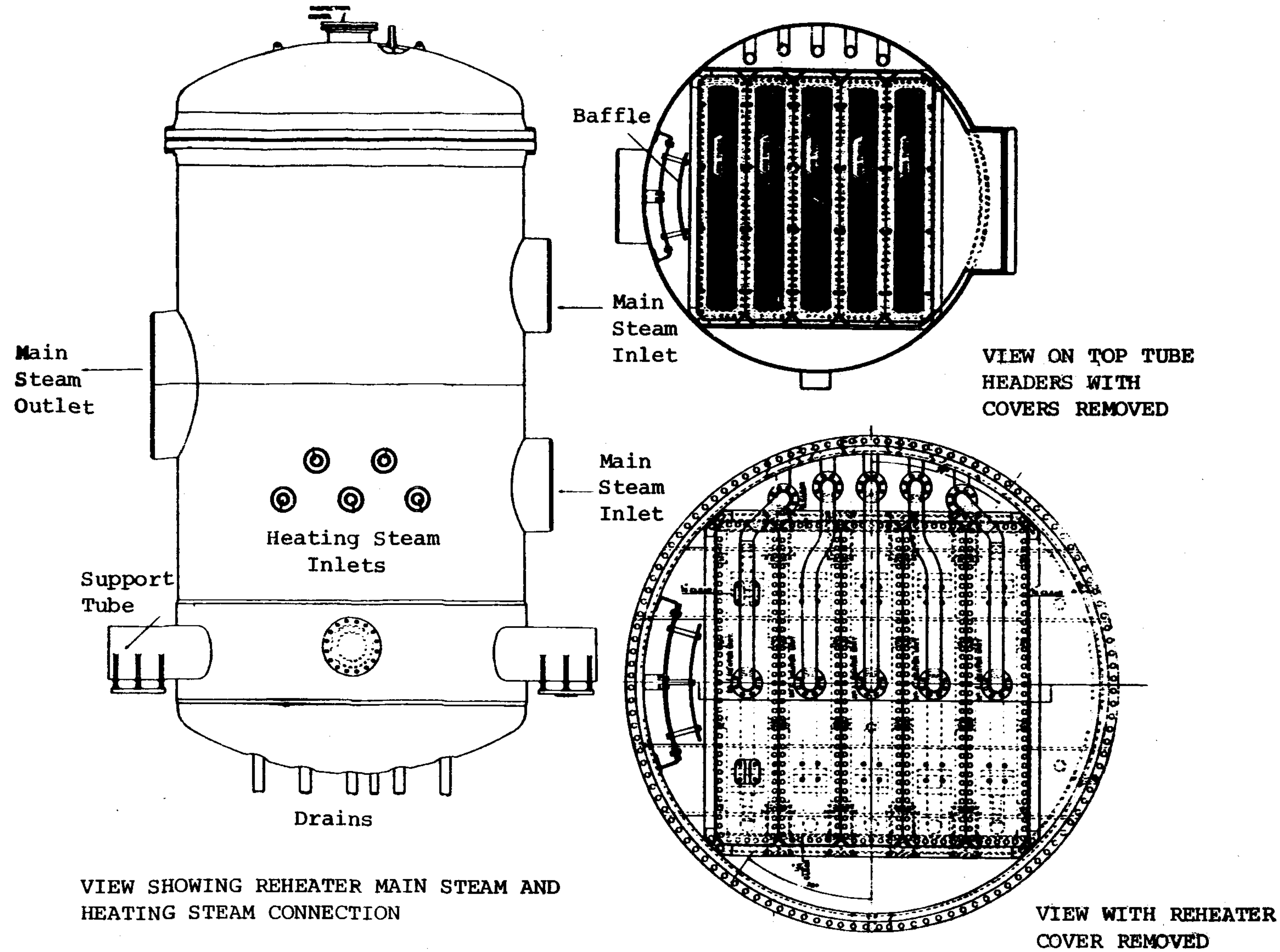
The erosion of the LP blading will be considerably less and, since the average stage moisture content is less, the turbine efficiency will be greater. Figure 1.3 shows a typical reheater. Steam exhausting from the moisture separators passes over several banks of tubes before leaving the reheater. The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

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

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

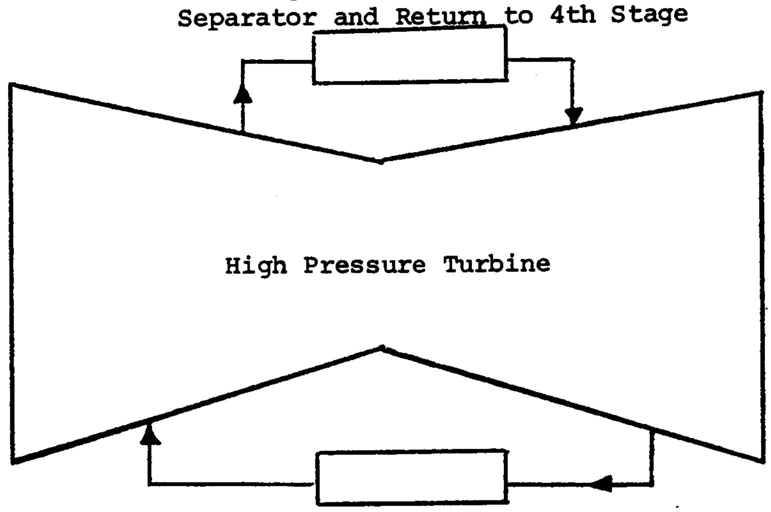
Another method of moisture removal which utilizes the centerfuging of water droplets to the outside of the moving blades is the placement of water extraction grooves after certain stages. These consist of annular grooves cut in the casing wall at the low pressure side of the moving blades in these stages.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

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



<u>Reheater</u>

<u>Figure 1.3</u>



Auxiliary Moisture Separator

Figure 1.4

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

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

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

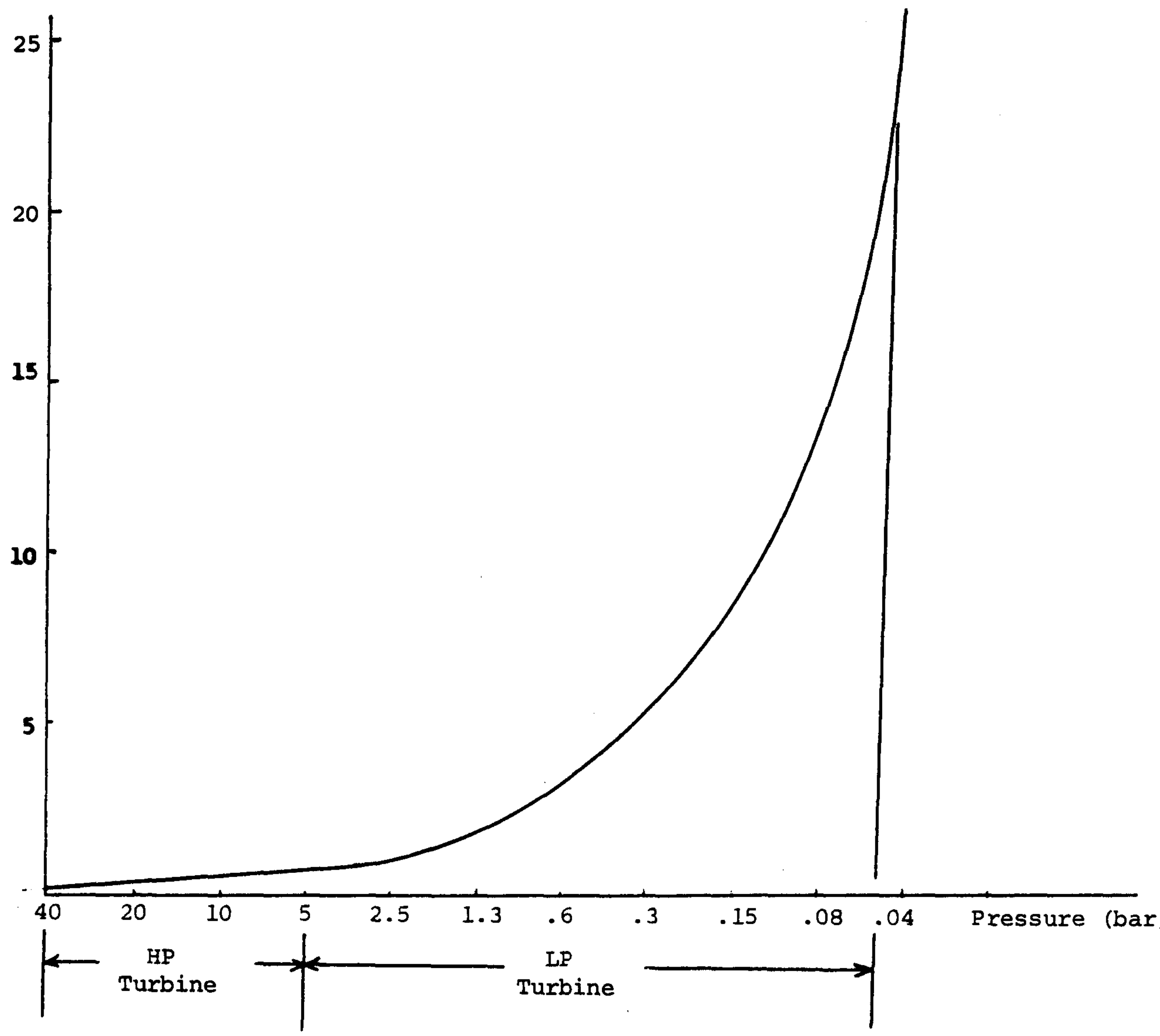
The common practice of placing shroud bands over the tip of moving blading to prevent blade tip leakage can have a major impact on water erosion of the blading. The shroud bands interfere with the centerfuging of water off the blades and their presence can cause blade deterioration near the tip of the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

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

Steam Expansion

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

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

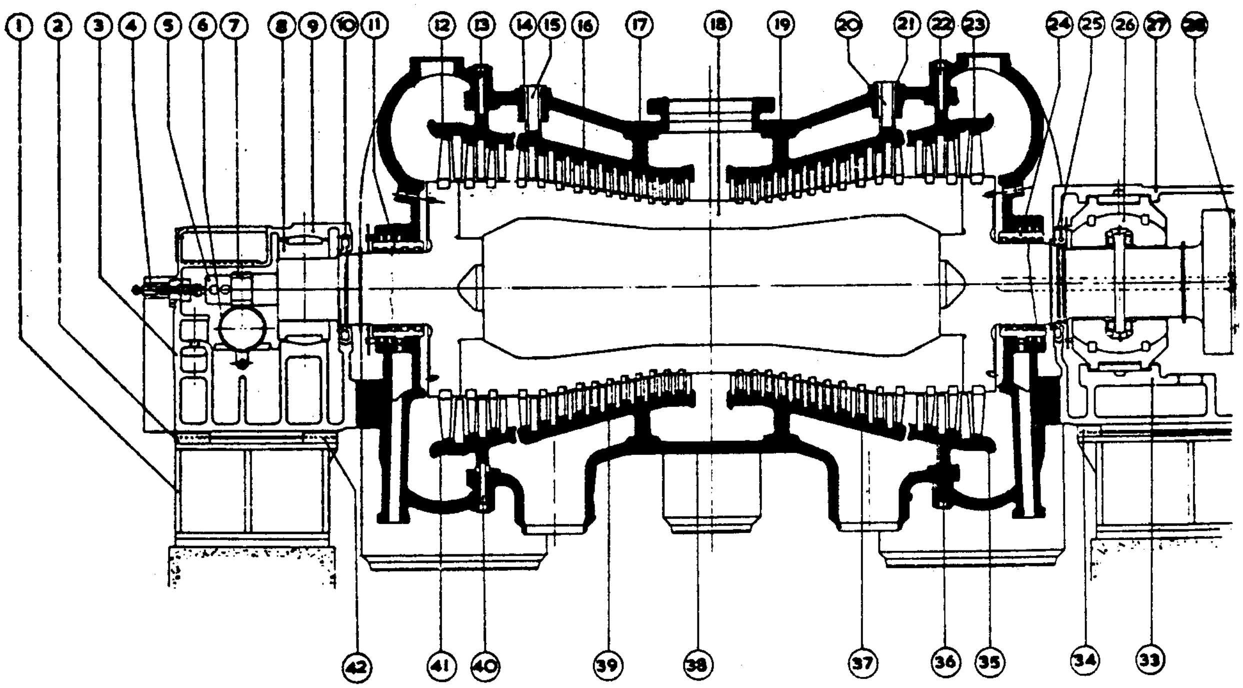


Steam Expansion Through a Turbine

Figure 1.5

HP TURBINE

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

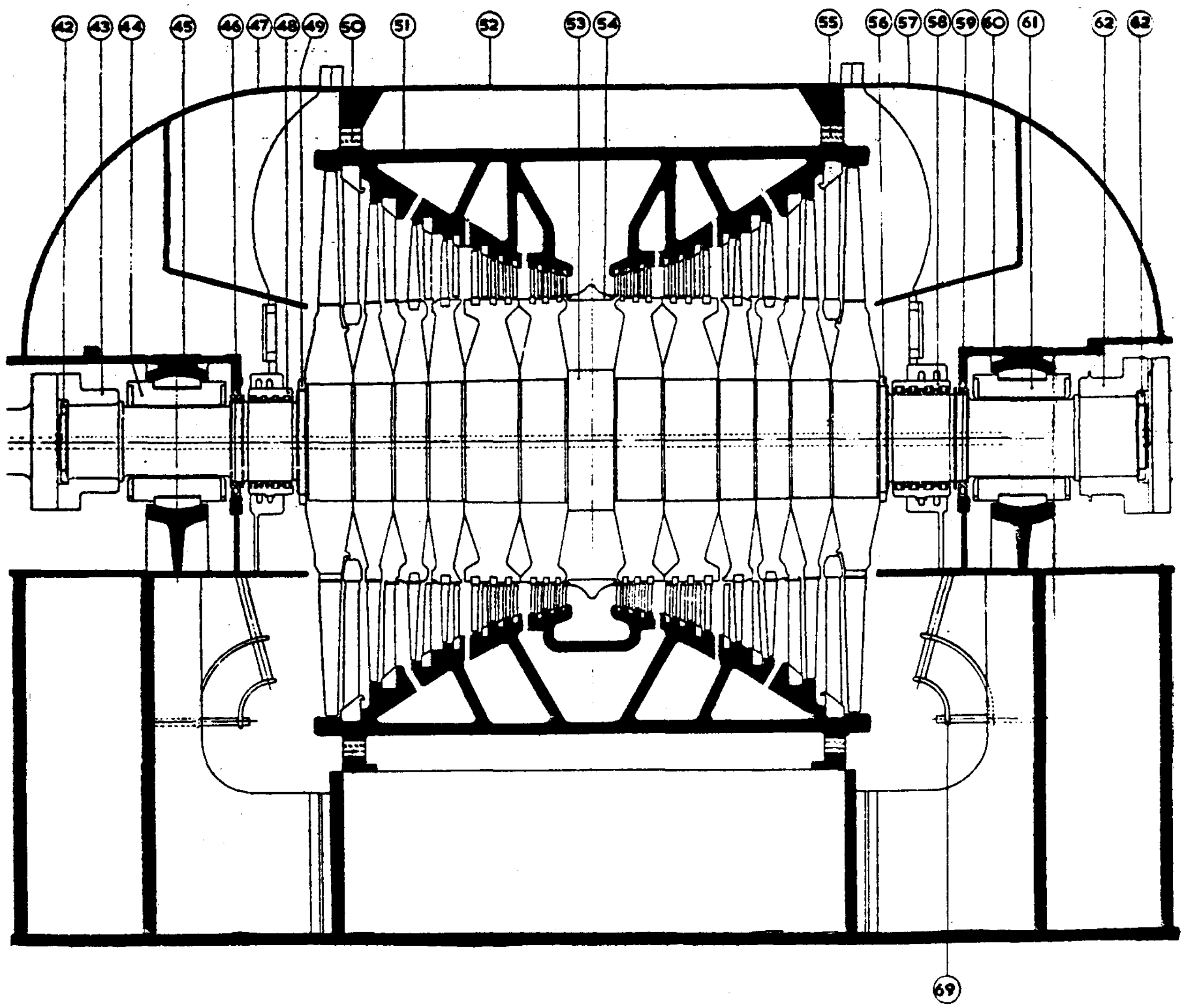


Typical HP Turbine with Extraction Steam after Stage 11

Figure 1.6

LP TURBINES

Nº	NAME OF PART	Nº	NAME OF PART
42	LP SHAFT COUPLING NUT	55	GUIDE KEY
43	COUPLING ADAPTOR	56	LP SHAFT DISC NUT
44	Nº 3 BEARING	57	LP CYLINDER EXHAUST COVER (END SECTION)
45	Nº 3 BEARING KEEP	58	Nº 4 GLAND
46	Nº 3 OIL BAFFLE	59	Nº 4 OIL BAFFLE
47	LP CYLINDER EXHAUST COVER (END SECTION)	60	Nº 4 BEARING KEEP
48	Nº 3 GLAND	61	Nº 4 BEARING
49	LP SHAFT DISC RETAINING NUT	62	COUPLING ADAPTOR
50	GUIDE KEY	63	LP SHAFT COUPLING NUT
51	LP CYLINDER CENTRE (TOP HALF)		
52	EXHAUST COVER CENTRE SECTION		
53	LP TURBINE SHAFT	66	LAY SHAFT GUARD
54	DEFLECTOR RING (IN HALVES)	67	GUIDE BRACKET
		69	DOUBLE VANE DIFFUSER



Typical LP Turbine

Figure 1.7

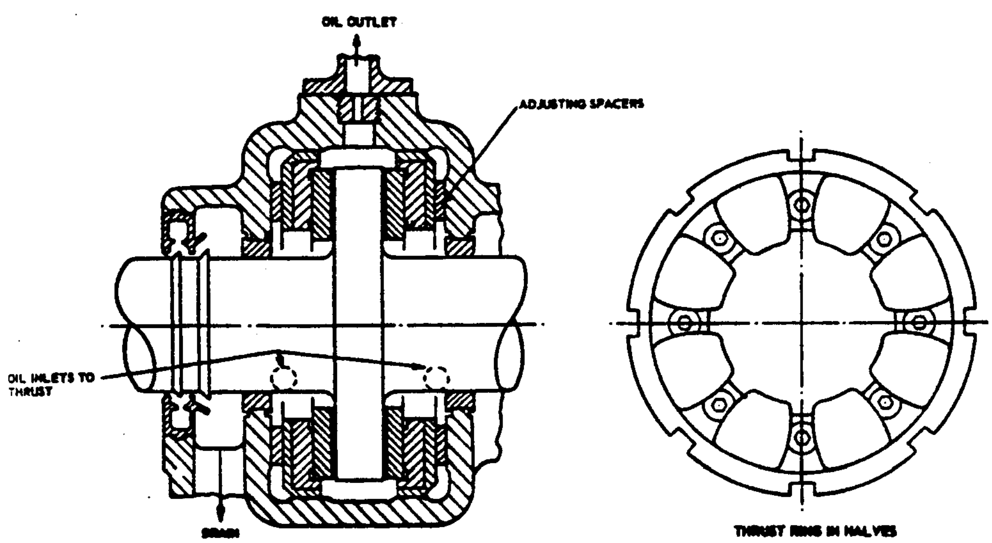
Another method of dealing with steam expansion is the removal of extraction steam for feedheating. This reduces the quantity of steam which must flow through the latter stages of the LP turbine. For example, in the low pressure turbines at Bruce NGS, only 75% of the steam which enters the turbine passes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

Axial Thrust

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

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

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

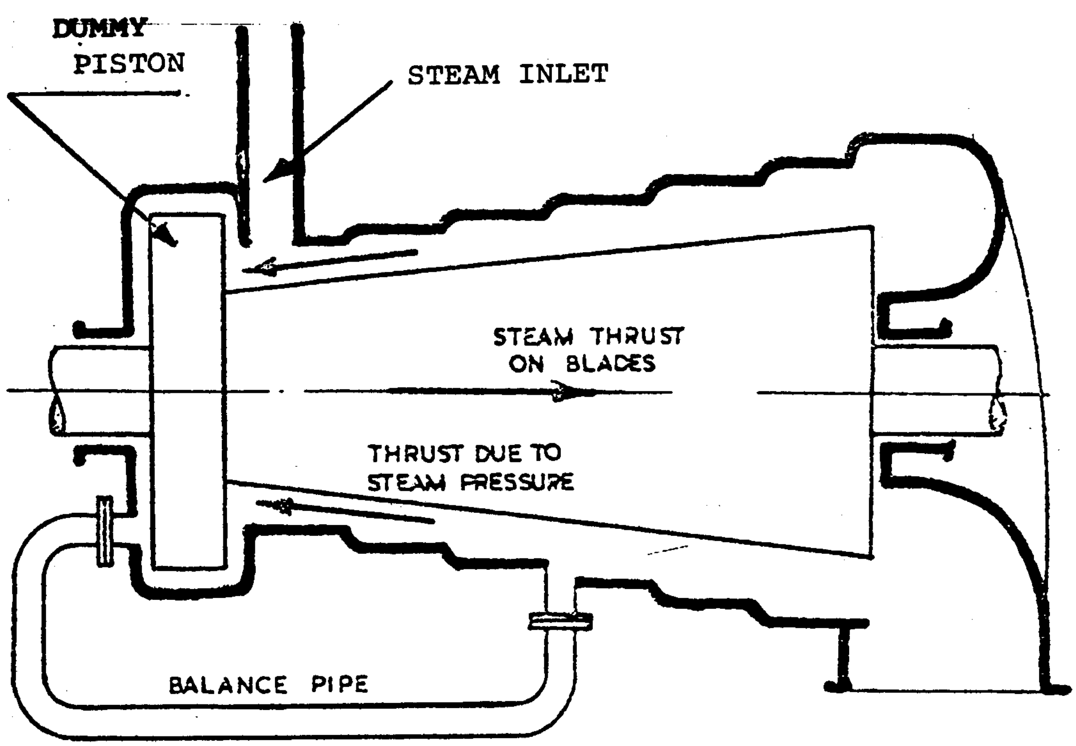


Tilting Shoe Thrust Bearing

Figure 1.8

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

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



Single Flow Reaction Turbine with Dummy Piston

Figure 1.9

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

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

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

1. Impulse staging in single flow HP turbines,

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

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

Expansion of Turbine Rotor and Casing

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

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

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

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

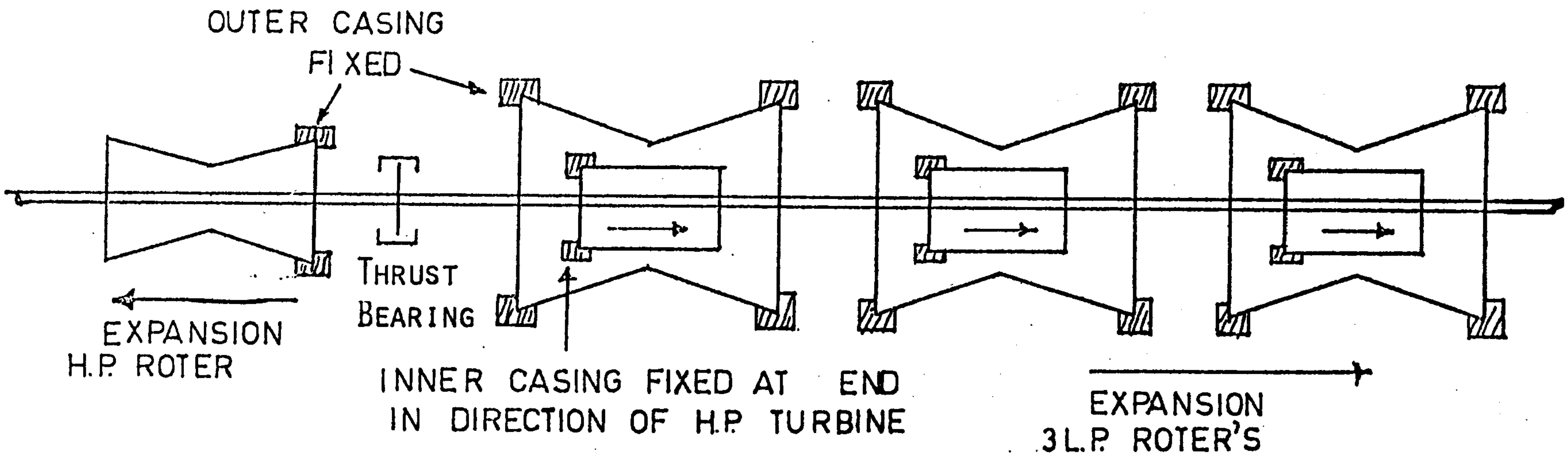
1. Transient axial differential expansion due to differing metal masses and rates of heatup and cooldown, and

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

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

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

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



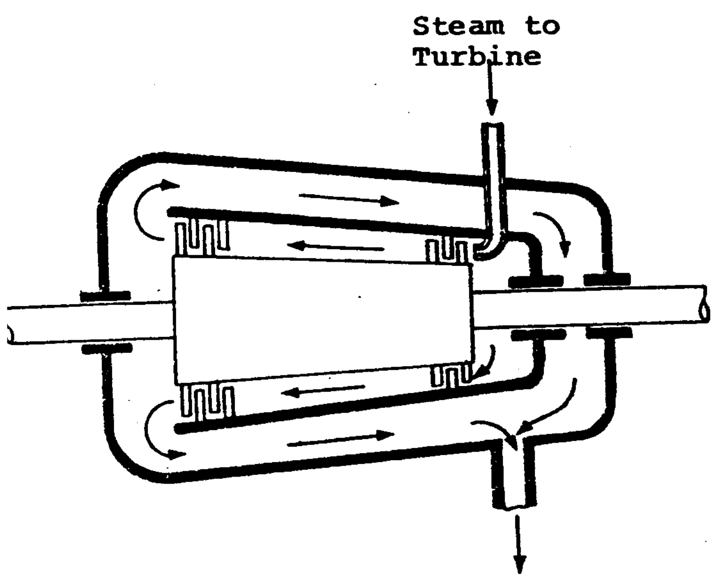
Fixed Points of Shaft and Casings

Figure 1.10

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

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

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

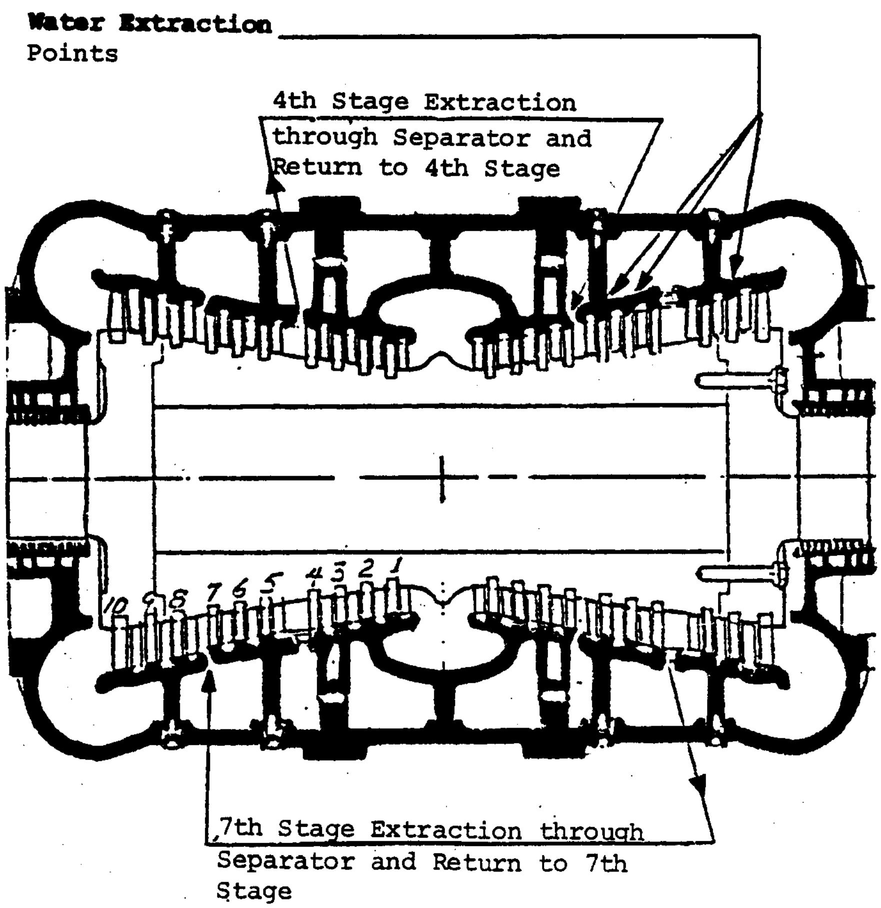


Simple Double Casing

Figure 1.11

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

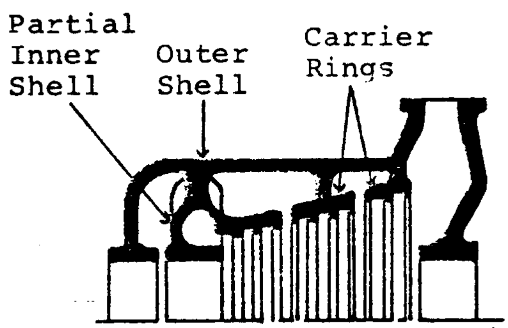
In practice the construction is not quite so simple. In Figure 1.6 you can see that the warming of the inner casing is achieved by a combination of exhaust steam, extraction steam and inlet steam flowing around the outside of the inner casing. In turbines with internal moisture separators, the steam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



HP Turbine With Auxiliary Separators

Figure 1.12

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



Single Flow Turbine with Carrier Rings

Figure 1.13

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

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

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

Thermal Stresses in Turbine and Rotor

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

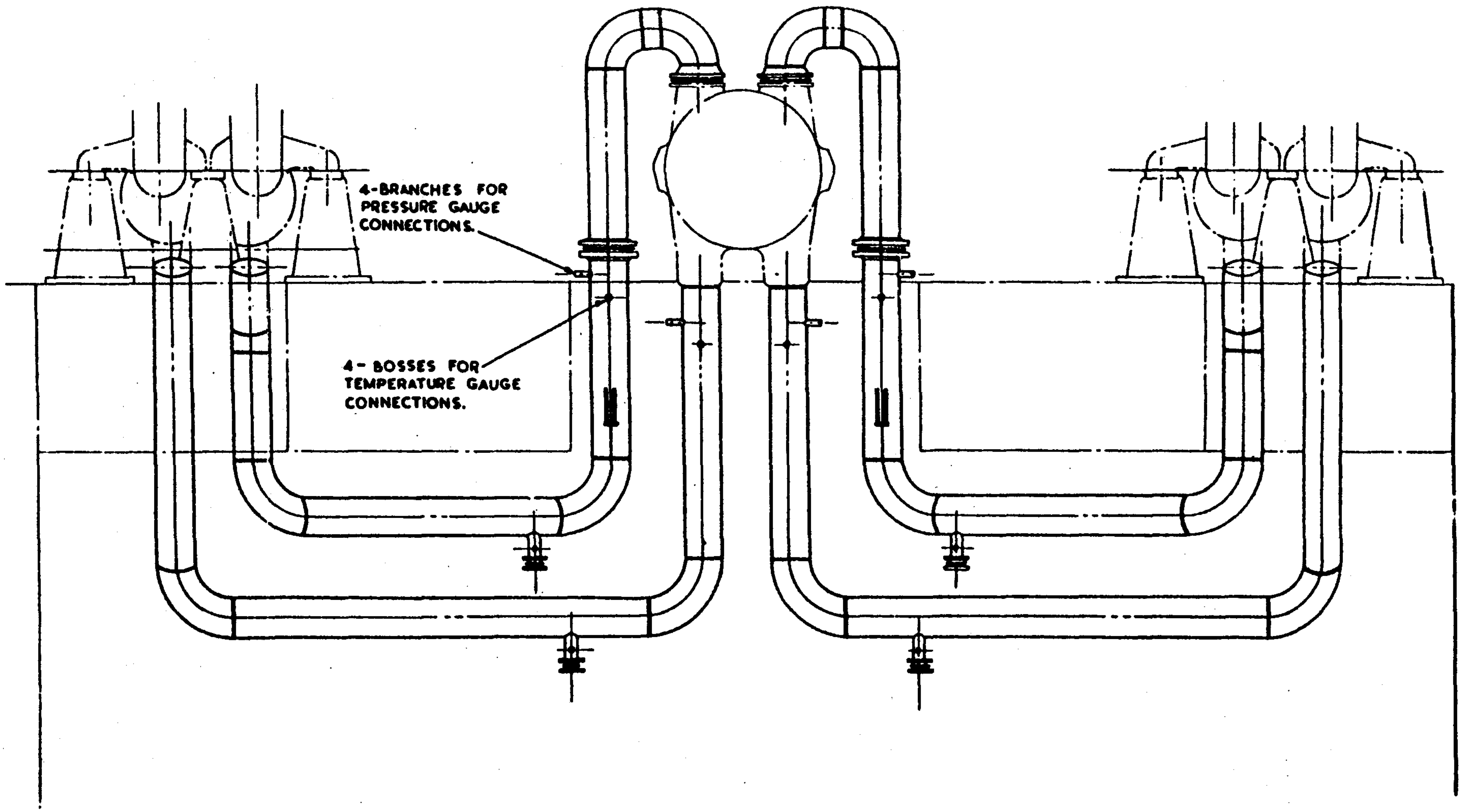
You have no doubt already recognized that the design features intended to minimize differential axial expansion such as the double casing, carrier rings and the drum rotor, also serve to minimize thermal stresses, since they allow for a more even heatup of the casing and rotor.

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

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

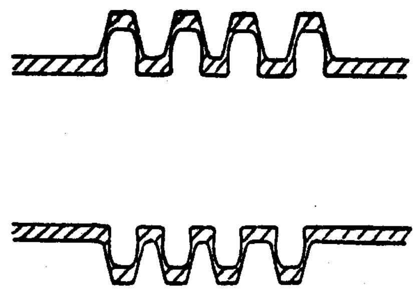
In Figure 1.14 the loops in the steam admission lines are placed there to prevent the creation of stresses in pipework when the piping expands while heating up. These thermal expansion loops allow for expansion of the piping. If the piping was not allowed to expand, the force developed on heatup would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

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



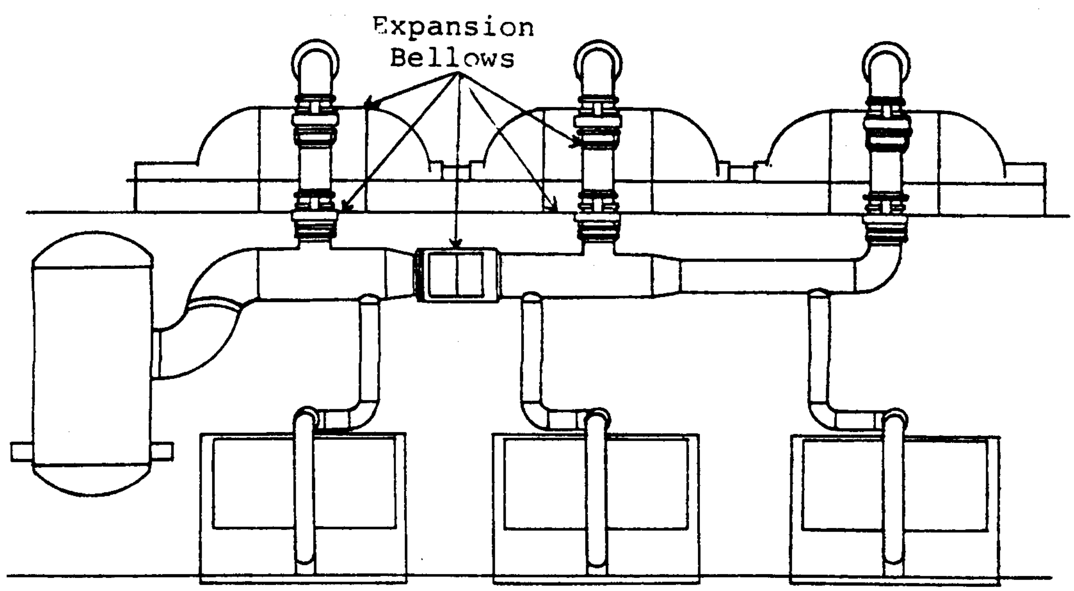
ARRANGEMENT OF H.P. INTERCONNECTING PIPES

Figure 1.14



Expansion Bellows

Figure 1.15



Expansion Joints on LP Turbine Inlet

Figure 1.16

ASSIGNMENT

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

   (a) Moisture,

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

   (c) Axial thrust,

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

   (e) Thermal stresses in casing and rotor.

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

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

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

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

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 334

THE STEAM SYSTEM

Figure 3.1 shows the steam piping and valves associated with a typical large CANDU turbine unit. The valves shown are not only the largest and most obvious valves in the steam system, they are also the most significant from the standpoint of turbine control and system safety.

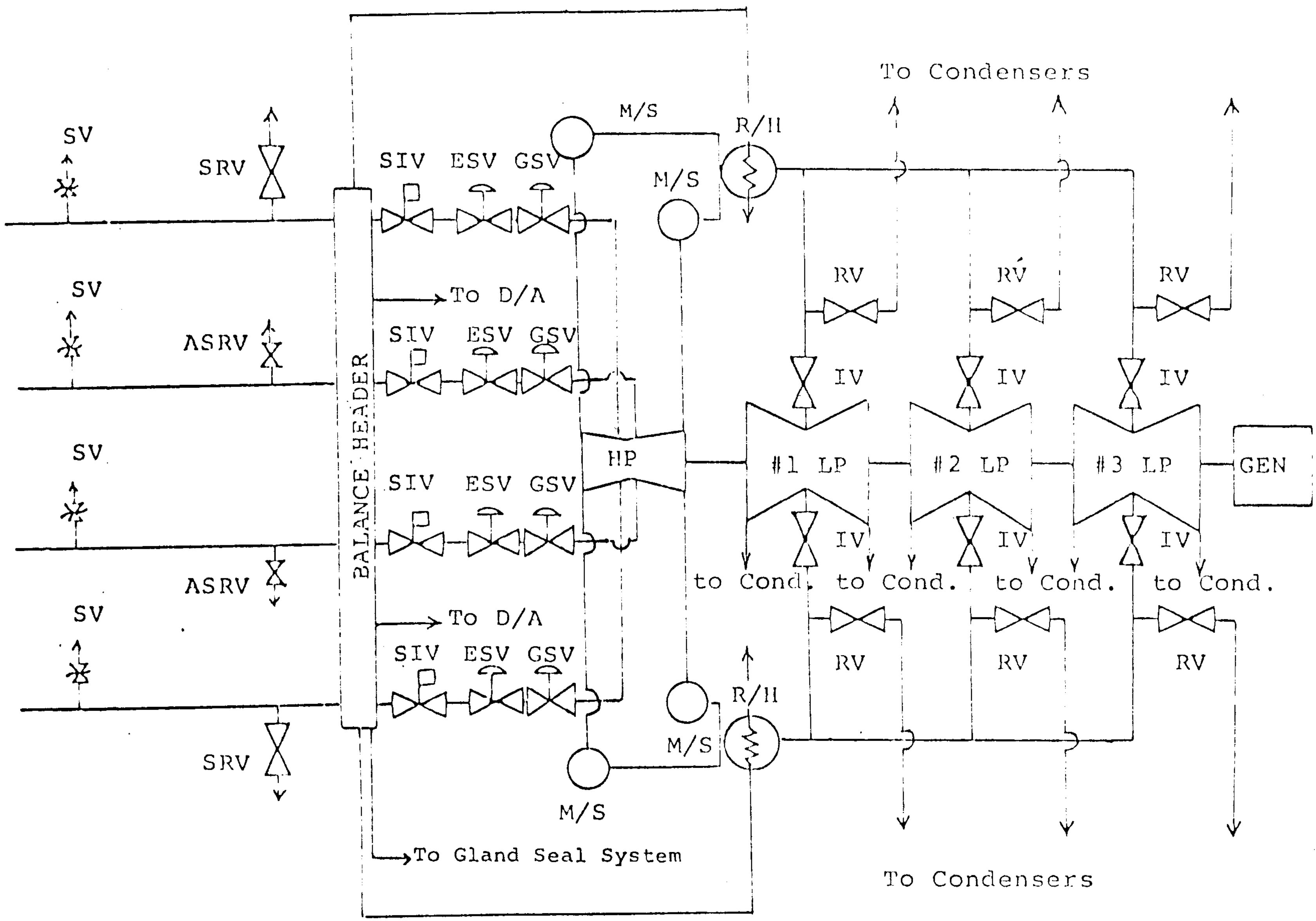
The Steam Generator Safety Valves

The steam generator safety valves are installed either on the main steam piping or on the steam generators. There should not be an isolation valve between the safety valve and the system it protects. The function of these valves is to prevent the steam generators and steam piping from being overpressurized to the extent that the strength of the pipework could be exceeded. The steam generator safety valves are required by law and must be capable of relieving the full design steam flow of all the steam generators without the steam pressure rising above 110% of design working pressure. If a steam generator safety valve is inoperative, the maximum steam flow from the steam generators must be reduced so that the operable safety valves can relieve the entire flow.

Figure 3.2 shows a typical steam generator safety valve. When the steam pressure under the disc reaches the set pressure, the upward force overcomes the spring force, the valve pops open and steam escapes to atmosphere. A not infrequently encountered refinement in safety valves is to assist the opening of the valve with either an air operator or an electric solenoid which can be energized on receipt of a signal from the control room automatic control equipment or from the control room operator.

Safety valves are set to open at about 6% above the maximum working pressure. Once the valve has opened or lifted, it will remain open until the pressure is reduced significantly below the opening pressure. The percentage difference between lifting and reseating pressure is known as reseating <u>blowdown</u> and is typically set between 3% and 10% of the pressure at which the valve opens. If the safety valve had no blowdown, the valve would rapidly open and shut at its lift pressure as the steam pressure below the disc fluctuated due to flow through the valve. This "chattering" would rapidly destroy the valve seat and disc. On the other hand, if the blowdown was too large, there would be an unacceptable loss in pressure and water mass from the steam generators when the valve lifted. The safety valve blowdown must be

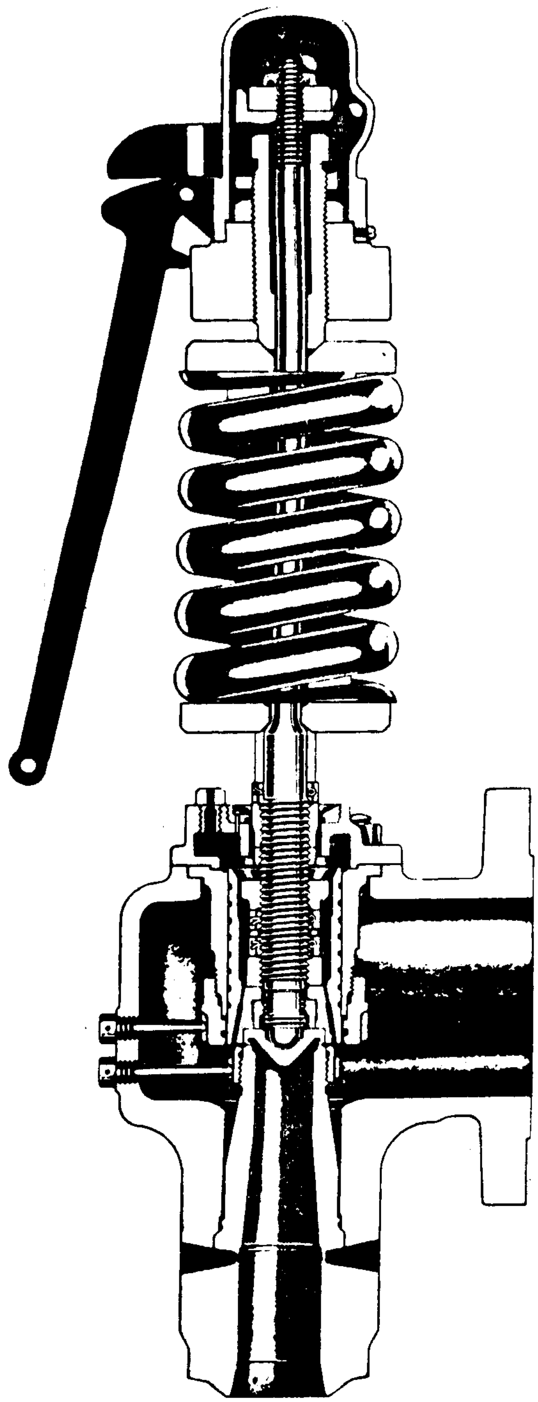
From Steam Generators



SV = Steam Generator Safety Valves
SRV = Large Steam Reject Valves
ASRV = Small Steam Reject Valves
ESV = Emergency Stop Valves
GSV = Governor Steam Valves
RV = Steam Release Valves
IV = Intercept Valves
SIV = Steam Isolating Valves

Figure 3.1

large enough to ensure an adequate reduction in pressure and yet small enough to limit the loss of fluid and fluid pressure to acceptable values.



Steam Generator Safety Valve

Figure 3.2

The number of safety valves protecting a large steam system is generally between 10 and 20. The lift pressures for the valves are generally staggered so that all valves do not lift at the same time. For a steam system with a maximum operating pressure of 4000 kPa(a) and say sixteen safety valves, the lift pressure might typically be 4 safety valves at 4240 kPa(a) (106% of working pressure), 4 safety valves at 4280 kPa(a) (107%), 4 safety valves at 4320 kPa(a) (108%) and 4 safety valves at 4360 kPa(a) (109%). The purpose of this scheme of staggered lift pressures is to ensure that:

1. only the required number of safety systems to handle the overpressure are called upon to operate

2. the rapid increase in steam flow when the safety valves lift is held to values commensurate with steam generator and reactor plant design limits, and

3. the mechanical and thermal shock to the system is held to acceptable values.

<u>Steam Reject Valves (SRV)</u>

The steam generator safety valves are required to protect the steam generators from an overpressure condition which could have catastrophic results. They must be extremely reliable; the simpler the construction the better. For both legal and practical reasons, we cannot design safety valves to perform too many complex functions or they may not be able to always perform their intended primary function. It is desirable however, to be able to exercise precise control of steam pressure by varying the rate at which steam is released from the steam system. This requirement is fulfilled by the <u>steam reject valves</u>. There are three basic requirements to release steam from the steam system:

1. for minor variations in pressure above the desired pressure due to temporary mismatches between reactor power and turbine power

2. for large boiler pressure transients which may result from a turbine trip with the unit at power, and

3. to release approximately 60% of design steam flow to prevent xenon poison out when the turbine is unavailable to remove reactor power.

Large CANDU generating stations have two sets of reject valves:

1. small valves (three inches in size) which are used for minor pressure transients, and

2. large valves (ten inches and larger in size) which are used for large pressure transients and to maintain steam flow above 60% during a turbine outage, to prevent xenon poison out.

The reject valves are pneumatic valves operated to open on receipt of an electrical signal from the control computers. The valves are held shut by steam pressure within the steam system and will not open unless the actuators receive the proper electrical signal.

If the boiler pressure rises above the desired programmed valve, the small reject valves open. If pressure continues to rise, indicating the small valves are passing insufficient flow to control boiler pressure, the large reject valves open. If the large reject valves open, a limit switch is operated which initiates a reactor setback.

The Steam Balance Header

The steam balance header, or main steam header as it is also called, is a large cylindrical steel vessel. It receives the steam from all the steam generators and equalizes the steam pressure before the steam goes to the turbine. The balance header also absorbs the expansion forces set up by the thermal expansion of the steam piping as it heats up.

The steam balance header and the steam piping adjacent to it supply steam to the following:

1. the turbine
2. the reheaters
3. the turbine gland seal system
4. the steam air ejectors (if the station uses this type of air extraction system)
5. the main steam which supplies the deaerator during poison prevent and startup when extraction steam from the turbine is available, and
6. steam to the heavy water plant (if applicable to the station).

The Steam Isolating Valves (Boiler Stop Valves)

The function of the steam isolating valves, or boiler stop valves as they are frequently called, is to isolate the turbines from the steam generators. This will allow maintenance to be performed on the turbine while the steam generators are hot and the steam piping pressurized with steam.You will notice that the steam generators cannot be isolated from one another on the steam side.

The steam isolating valve is normally a parallel slide valve as shown in Figure 3.3. This type of valve presents little restriction to flow when fully open and gives virtually complete isolation when shut. When the valve is shut and under pressure from the steam generator side, the disc on the turbine side is held in tight contact with the seat by steam pressure. When the valve has little pressure difference between the inlet and outlet sides, the discs are kept in contact with the seats by the spring. This overcomes the chief disadvantage of a normal single disc gate valve in that such a valve has a tendency to leak at low differential pressures. The steam isolating valve is normally operated by an electric motor.

The discs of the steam isolating valve may be as much as .67 meters in diameter. With full steam generator pressure on one side and atmospheric pressure on the other side, the force holding the disc against the seat may be as large as 500,000 newtons. This force may exceed the capacity of the motor to open the valve and even if it did not, the force may be sufficient to damage the valve if it is opened.For this reason, the steam isolating valve is normally fitted with a small bypass valve to allow pressure equalization before the main valve is opened. The bypass is operated by the same switch which energizes the main valve motor and opens fully before the main valve opens. The bypass valve does not shut until after the main valve is fully shut.

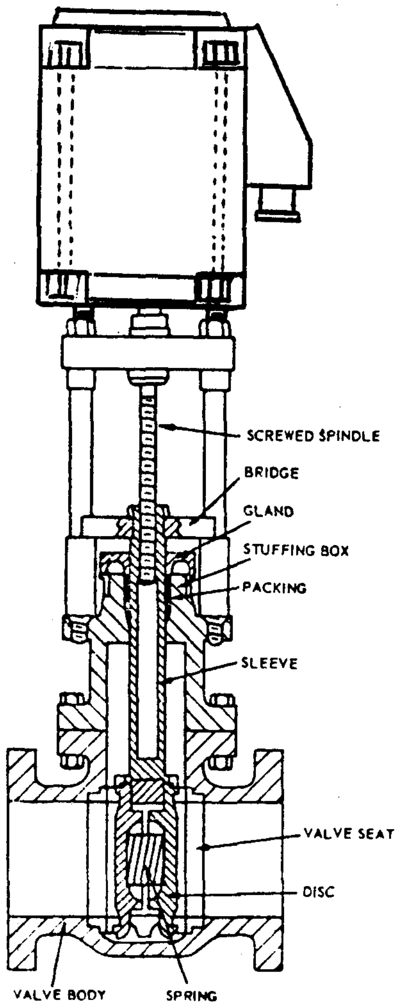
Steam Strainers

Immediately before the emergency stop valves, the steam passes through steam strainers. These strainers are designed to prevent solid impurities which may have contaminated the steam from entering the steam admission valves where these impurities could foul the valve seats or cause valve sticking by fouling the valve stems. The strainers also prevent impurities from entering the turbine where considerable damage could be done to the blading.

The Emergency Stop Valve (ESV)

The primary function of this valve, which is shown in Figure 3.4, is to immediately stop the flow of steam to the turbine in any emergency situation which, if allowed to continue, could damage the turbine.

The emergency stop valves are held open by high pressure hydraulic fluid (either the turbine lubricating oil or a separate hydraulic fluid system). The fluid pressure operating on the underside of the operating piston compresses a spring above the piston. On a turbine or generator fault which requires the valve to shut, the oil pressure is released from the underside of the operating piston and the spring slams the valve shut.



Steam Isolating Valve (Boiler Stop Valve)

Figure 3.3

The emergency stop valves are held open by high pressure hydraulic fluid (either the turbine lubricating oil or a separate hydraulic fluid system). The fluid pressure operating on the underside of the operating piston compresses a spring above the piston. On a turbine or generator fault which requires the valve to shut, the oil pressure is released from the underside of the operating piston and the spring slams the valve shut.

The Governor Steam Valves

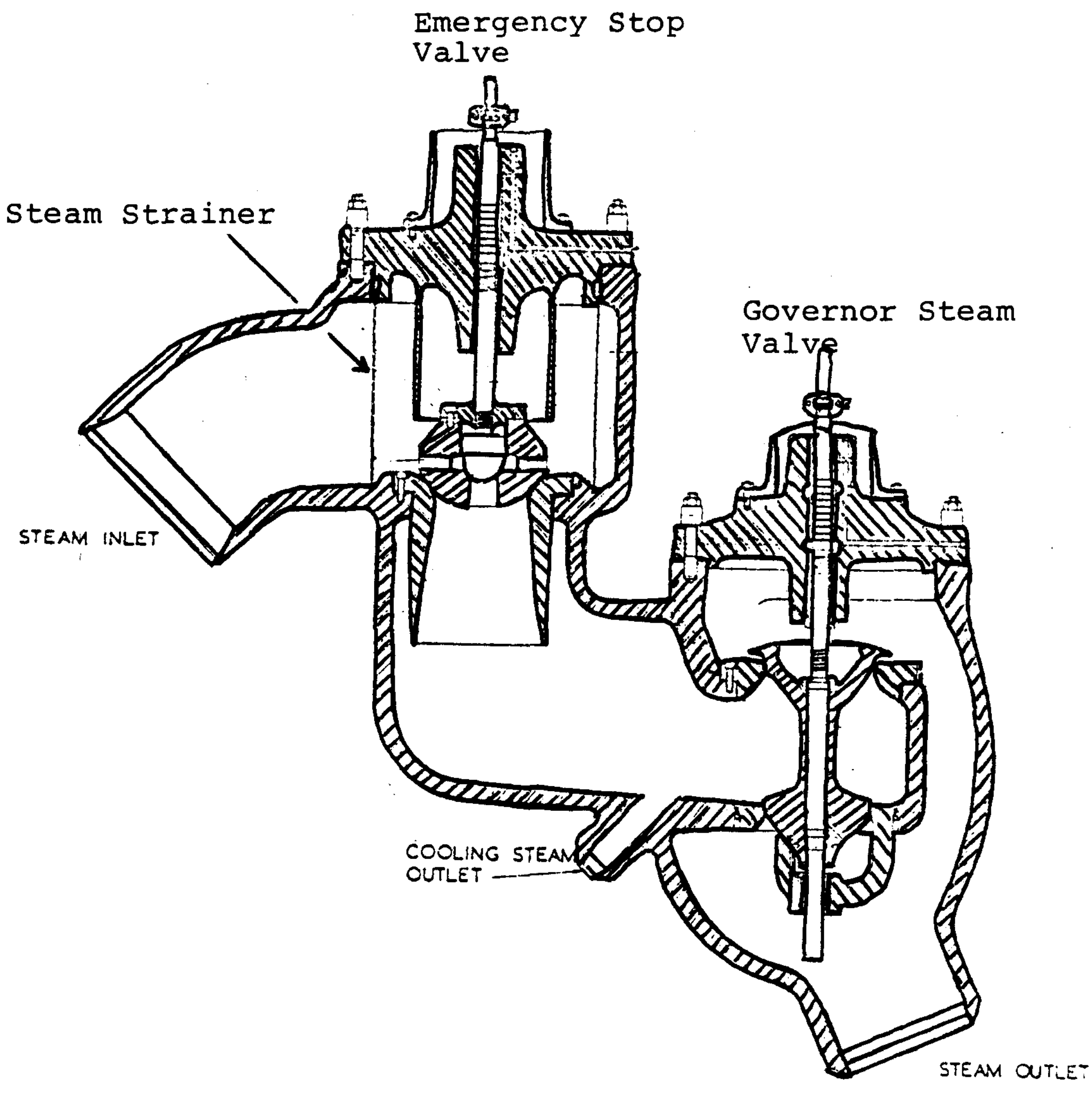
The function of the governor steam valves, also shown in Figure 3.4, is to control the quantity of steam flowing to the turbine so the electrical output of the generator can be varied.

The valves are operated by a combination of oil pressure and spring force. If the oil pressure under the piston is increased, the valve opens against spring tension. If the oil pressure is reduced, the spring tension forces the valve in the shut direction until the spring force is balanced by the oil pressure. The position of the governor steam flow to the turbine corresponds to the desired electrical load on the generator.

The governor steam valves are designed to control steam flow to the turbine over a rather limited speed range near the operating speed of the turbine. When the turbine is being started up, the governor steam valves are wide open and cannot properly control the steam flow to the turbine. For this reason, the steam flow to the turbine is controlled by gradually opening the emergency stop valve until the turbine is near normal operating speed. At this time, the governor steam valves can control the steam flow and the emergency stop valve is fully opened.

Thus, when the turbine is being brought up to operating speed, it is the emergency stop valve which is used to control the speed of the turbine. When the unit approaches operating speed, the governor becomes able to control the position of the governor steam valve to maintain the unit speed at a set value. The operator can vary this set value and thus has control of the turbine speed through the governor steam valve. However, once the output breaker has been shut and the generator is connected or synchronized to the Ontario electrical grid, the generator and therefore, the turbine, can only run at one speed which is determined by the grid frequency of 60 Hertz. The operator no longer can vary the speed of the turbine above or below its operating speed. If the operator now opens the governor steam valve and admits more steam to the turbine, the turbine and generator will not speed up. As more steam and therefore thermal power is admitted to the turbine, more electrical power will flow from the generator to the grid. In short, while the governor steam valve always is used to vary the steam supply to the turbine, the effect of this is to change the turbine's speed when the generator is not connected to the grid and to change the electrical power output when the generator is connected to the grid.

The governor steam valves on large CANDU generating stations all operate simultaneously. At 50% of maximum steam flow, all of the governor steam valves are passing 50% of their design maximum flow. At 100% of maximum steam flow, all the valves are passing 100% of design maximum flow. Such a governing system is known as throttle governing since the flow of steam to the turbine is controlled by throttling the steam with all the governor steam valves.



Steam Chest Assembly

Figure 3.4

The Intercept Valve

In the event of a turbine or generator fault which required the steam flow to the turbine to shut off, the emergency stop valve would shut. On large multicylinder steam turbines, the volume of steam which has already passed the emergency stop valves is sufficient to continue driving the turbine by expansion through the low pressure turbine. This "entrained steam", which is contained in the HP turbine, moisture separators, reheaters and low pressure piping, must be prevented from entering the low pressure turbine. An intercept valve, located on each inlet line to each low pressure turbine, performs this function.

The intercept valves are shut at the same time that steam is shut off to the high pressure turbine. The intercept valves are usually butterfly valves which provide good isolation when shut, low flow resistance when open and can be shut against a high flow rate through the valve. The valves are normally operated by either the same hydraulic fluid which operates the governor steam valves and emergency stop valves or by air operators.

The Steam Release Valves

When the intercept valves shut, the entrained steam must be removed from the turbine unit. This may be done either by steam release valves, which exhaust the steam to the condenser, or by reheater blow-off valves which exhaust the steam to atmosphere.

These valves will also function to protect the cross over piping against overpressure, such as might occur if an intercept valve shut while at power or a heater tube failed. The excess pressure would cause a signal to be sent to the release valve, opening it.

ASSIGNMENT

1. Draw a schematic diagram of the steam system for a large CANDU generating station having one double flow HP turbine and three double flow LP turbines, tandem compounded. Include in your diagram:

   (a) steam generators
   (b) steam generator safety valves
   (c) steam reject valves
   (d) balance header
   (e) steam isolating valves
   (f) emergency stop valves
   (g) governor steam valves
   (h) moisture separators
   (i) reheaters
   (j) intercept valves
   (k) release valves
   (l) connections for steam to the deaerator, air ejectors, gland seal system and reheaters.

2. What is the function of the steam generator safety valves? What is "blowdown" as it relates to a steam generator safety valve? What are the consequences of too large or too small blowdown?

3. Why are steam generator safety valve lifting pressure staggered so that all safety valves do not lift at the same time?

4. What are the functions of the steam reject valves? Why are most stations equipped with both steam reject valves and steam generator safety valves?

5. What are the functions of the emergency stop valves?

6. What is the purpose of the steam isolating valves?

7. What is the purpose of the governor steam valves?

8. What are the functions of the intercept valves and steam release valves?

9. What is meant by "throttle governing"?

10. How is turbine speed controlled:

    (a) on a startup below operating speed?
    (b) at operating speed when the generator is not yet synchronized with the grid?
    (c) at operating speed after the generator is synchronized with the grid?

11. Consider each of the following casualties and explain what would happen:

    (a) a governor steam valve fails and goes shut at 100% power
    (b) a governor steam valve stuck in the 100% power position
    (c) an intercept valve failed and goes shut at 100% power
    (d) an intercept valve stuck and did not shut on a loss of generator load from 100% power
    (e) two emergency stop valves stuck open on a loss of generator load from 100% power
    (f) a release valve failed and came open at 100% power.

R.O. Schuelke.

Turbine, Generator & Auxiliaries - Course 234

STEAM CONTROL TO THE TURBINE

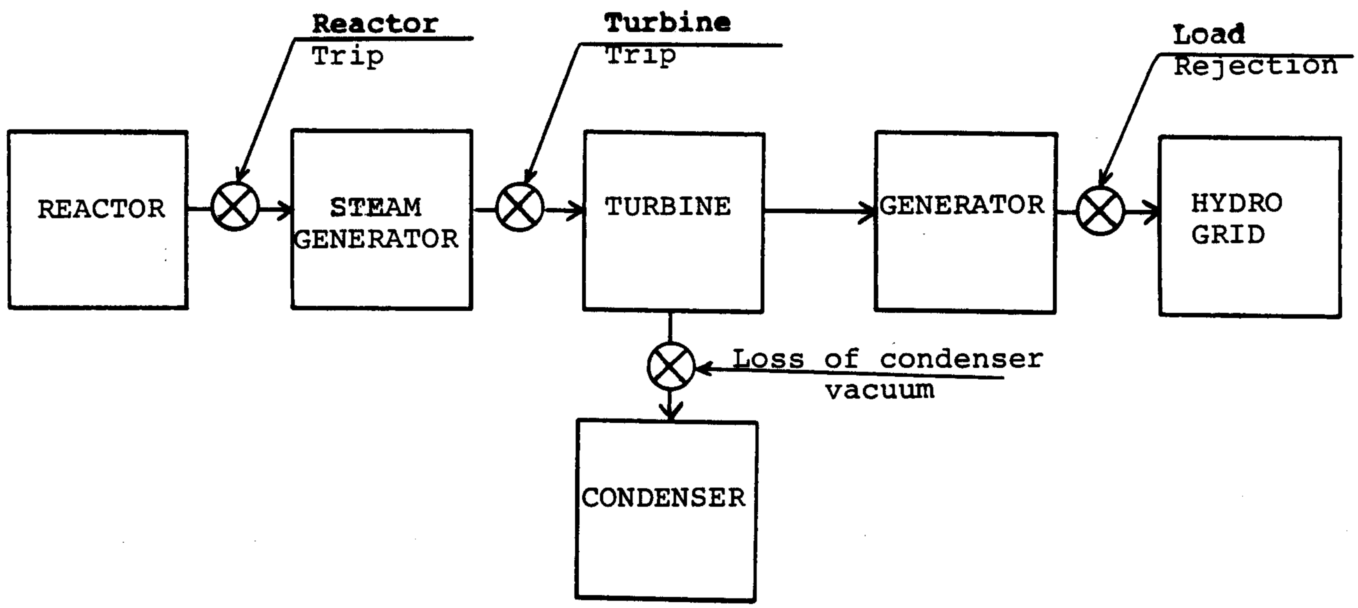


Figure 2.1

The steam generators of a large nuclear generating station produce on the order of 4,000,000 kilograms of steam per hour. The function of this steam is to transmit thermal power from the steam generators to the turbine. During normal, full power operation, the thermal power added to the steam in the steam generators is removed in the turbine (33%) and the condenser (67%). The power delivered by the steam is converted first to mechanical shaft power by the turbine and then to electrical power in the generator. This electrical power is then fed to the Hydro grid. The flow of power from the heat transport system to the Hydro grid is shown in Figure 2.1. Also shown are the points at which this flow can be interrupted. Since the capability of the system to store energy is rather limited, the interruption of power flow at some point imposes an urgent need to restore a balance between power input from the heat transport system and power output. The method of achieving this control is the subject of this lesson.

Boiler Pressure Control

In order to appreciate the control of steam in a CANDU generating station, it is necessary to understand how steam pressure in the steam generators is maintained. As a starting point consider the thermal power transferred across the tubes of the steam generators. This power may be expressed by the equation

$$\dot{Q} = UA\,\Delta T \qquad (2.1)$$

where: $\dot{Q}$ = the thermal power conducted from the heat transport system to the water in the steam generator.

U = the overall heat transfer coefficient of the tubes.

A = the total tube area.

ΔT = the temperature difference between the average heat transport system temperature in the tubes and the temperature of the water in the steam generator riser.

As power level is increased the left hand side of this equation obviously increases. If the equality is to be maintained, the right side of the equation must likewise increase. But the tube area (A) doesn't change and the overall heat transfer coefficient (U) changes only slightly. This means if the right hand side of the equation is to increase, then ΔT must increase.

$$\overset{\uparrow}{\dot{Q}} = \overrightarrow{U}\overrightarrow{A}\ \overset{\uparrow}{\Delta T}$$

So for the type of steam generators we have in our CANDU generating stations, as power level goes up, then the difference between heat transport system temperature and steam generator temperature increases.

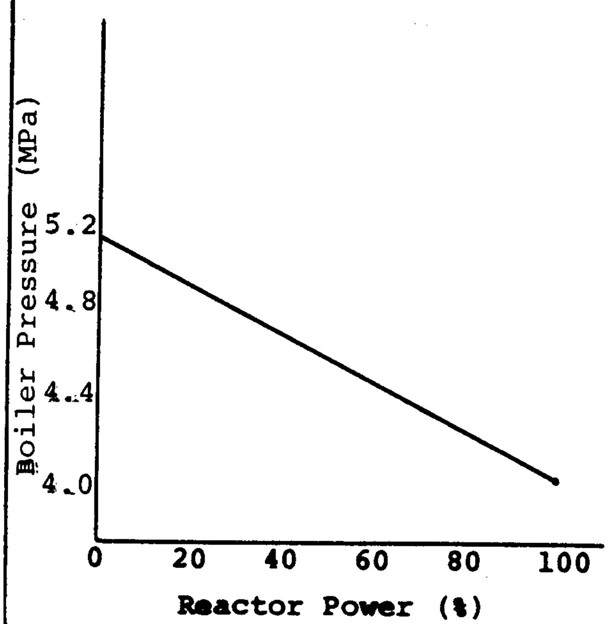
The temperature difference ΔT is :

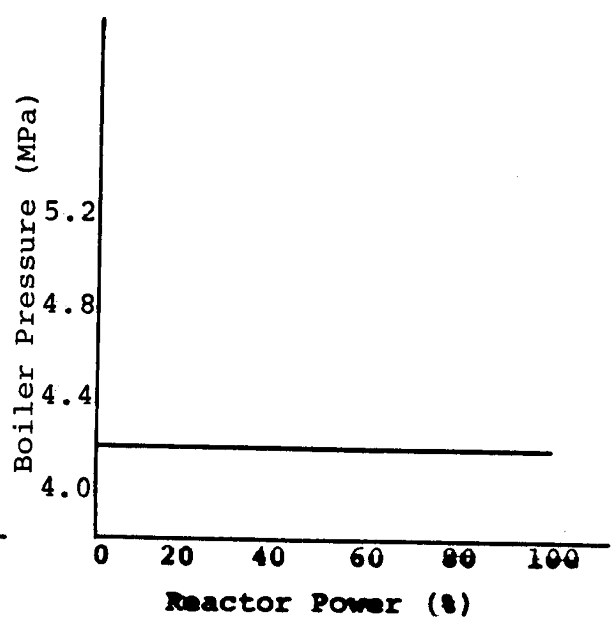
$$(T_{\substack{\text{average}\\ \text{HTS}}} - T_{\substack{\text{steam}\\ \text{generator}}})$$

For ΔT to increase the average heat transport system temperature must go up or the steam generator temperature must go down or both must occur simultaneously.

In some plants (Pickering NGS A for example) it is desired to hold the heat transport system temperature constant to minimize the swell and shrink associated with the volumetric expansion and contraction of the heat transport system. If this is done then as power level increases, the steam generator temperature must decrease. If steam generator temperature decreases, then since the steam generator is a saturated system, steam generator pressure must decrease. Figure 2.2 shows this effect for Pickering NGS A.

In other plants (Bruce NGS A for example) there is a pressurizer which can accommodate the volume changes in the heat transport system. In such systems, the steam generator temperature is held constant and the heat transport system temperature is allowed to increase as power level increases. This results in a constant steam generator pressure as is shown in Figure 2.3.





Pickering NGS

Bruce NGS

Steam Pressure Vs Power

Figure 2.2

Figure 2.3

The shape of the boiler pressure curve (Figure 2.2 or 2.3), for the particular plant, determines the desired boiler pressure for a given power level. Once the shape of the boiler pressure curve is established, then for each power level, all the other parameters are adjusted to maintain the correct boiler pressure, corresponding to that power level.

The boiler pressure control system has basically two means of restoring boiler pressure to the correct programmed value:

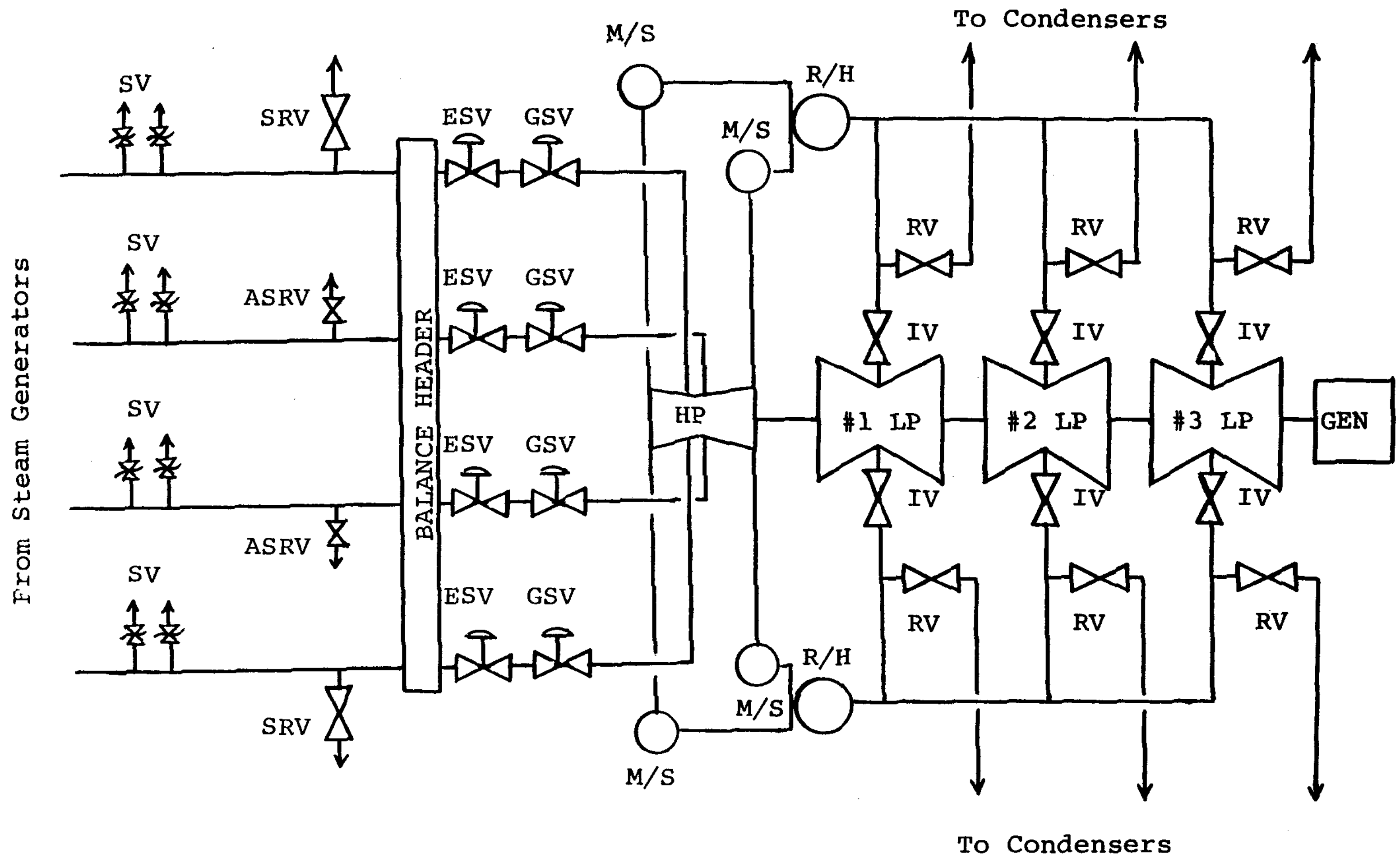
1. adjust the flow of steam out of the steam generators,
2. adjust the flow of heat into the boilers by changing reactor power.

Loss of Heat Source

If the heat source is either partially or fully lost, through a reactor trip, stepback or setback, then thermal power is leaving the steam generators at a faster rate than it is being added by the heat transport system. In this case, steam generator pressure will begin to fall and with it, heat transport temperature and ultimately heat transport pressure. The turbine speeder gear is runback, to avoid lowering heat transport system temperature. The runback will generally be terminated when boiler pressure stabilizes at the programmed level, indicating reactor power is again matched with steam power.

If reactor power is lowered far enough, the speeder gear will unload the turbine completely and the turbine unit will motor. To avoid poisoning out following such a reactor trip, the reactor power must be raised to above about 60-70% of full power, in approximately 40 minutes. The time limits for the recovery would be on the order of:

Diagnose and clear trip:	25 minutes
Regain criticality:	11 minutes
Raise power to 60-70% F.P.:	4 minutes
	40 minutes.



SV = Steam Generator Safety Valves
SRV = Large Steam Reject Valves
ASRV = Small Steam Reject Valves
ESV = Emergency Stop Valves

GSV = Governor Steam Valves
RV = Steam Release Valves
IV = Intercept Valves

Figure 2.4

Turbine Trip

Figure 2.4 shows the control valves associated with a large CANDU generating station. On a turbine trip these valves must function to accomplish three goals:

1. shutdown the turbine,

2. exhaust the steam which was formerly going to the turbine, to prevent a rapid rise in steam generator pressure, and

3. keep the reactor at sufficiently high power to prevent a poison out before the turbine can be brought back into service.

When the turbine trips, the governor steam valves and emergency stop valves shut as a result of governor power oil being dumped. At the same time the intercept valves shut and the release valves open. The generator output breaker opens and the turbine and generator begin to slow down.

Steam flow to the turbine stops much more rapidly than reactor power can be reduced and pressure in the steam generator increases. This excess steam pressure is relieved by exhausting the steam to atmosphere, through the steam generator safety valves or through reject valves, depending on the design of the particular station.

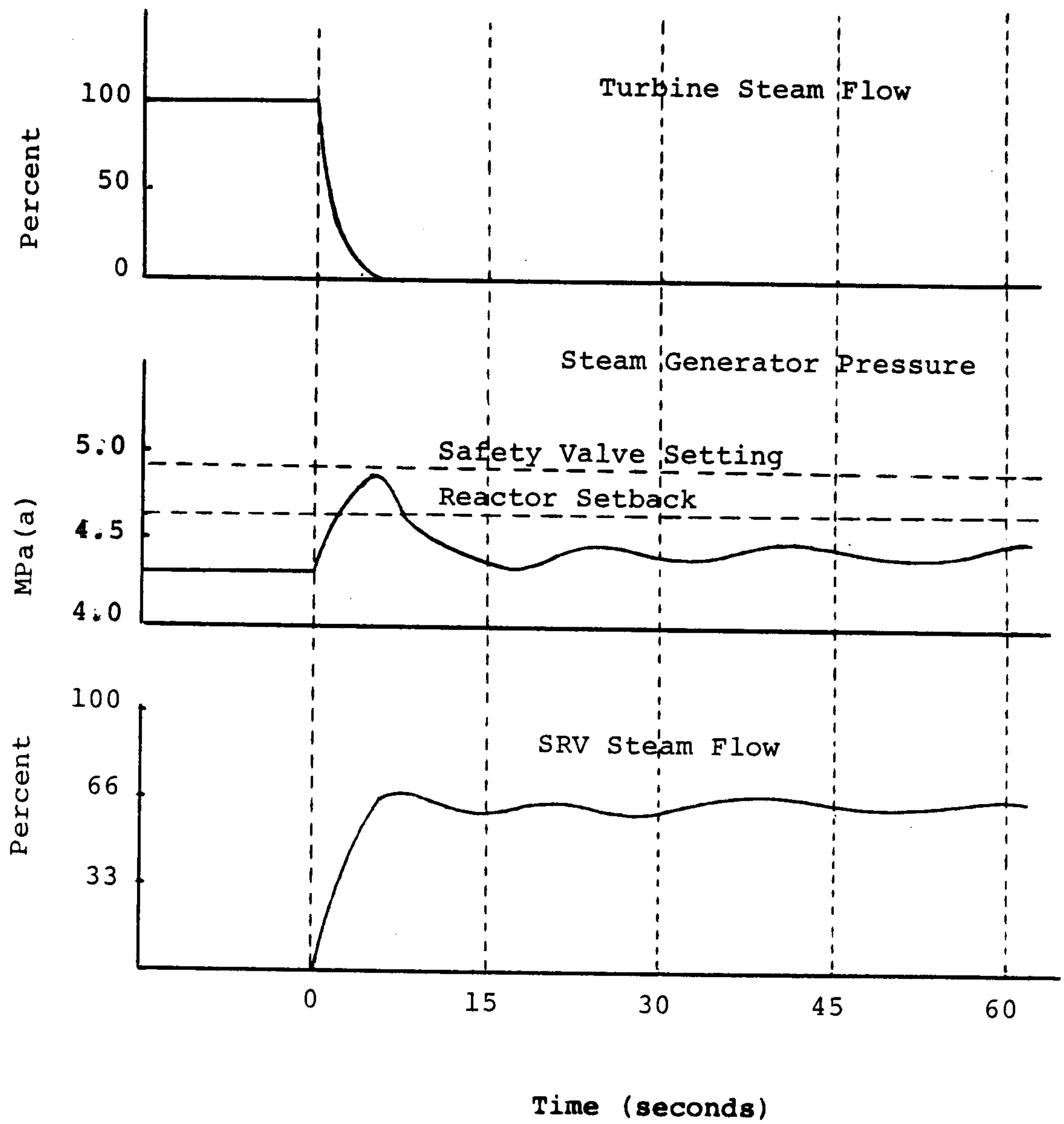
Large CANDU generating stations have two sets of reject valves:

1. small valves (three inches in size) which are used for minor pressure transients

2. large valves (ten inches in size) which are used for large pressure transients and to maintain steam flow above 60% during a turbine outage, to prevent xenon poison out.

On a turbine trip at 100% full power, steam generator pressure increases, as is shown in Figure 2.5. The small reject valves open, then the large reject valves open, and finally the reactor is set back on high boiler pressure. This action is sufficient to limit steam generator pressure and prevent the steam generator safety valves from lifting.

If reactor power was allowed to drop to zero (to match turbine power), the reactor would poison out in about 40 minutes. To prevent this from occurring, the reject valves are used to remove about 60% of full power steam flow. This allows the reactor to remain at a sufficiently high power level, to prevent poison out.



Turbine Trip

Figure 2.5

There are three basic types of reject systems used for poison prevent following a turbine trip:

(a) reject to atmosphere,
(b) reject to main condenser, and
(c) reject to a reject condenser.

(a) Reject to Atmosphere is the simplest form of reject system. In this system both the small and large reject valves discharge to atmosphere. The steam generators produce steam equivalent to about 70% of full reactor power of which 60% is rejected to atmosphere, via the reject system and 10% is sent to the deaerator for feedheating. The rejection of 60% of full power steam to atmosphere, results in a tremendous loss of water, which must be made up through the condensate makeup system. Since makeup water is being consumed faster than it can be produced by the water treatment plant, the reject system can only operate for four to six hours before the reactor must be shutdown due to low water inventory.

(b) Reject to the Main Condenser avoids this problem by condensing the reject steam in the main condenser and pumping it back to the steam generators, via the normal condensate and feedwater systems. The small reject valves go to atmosphere but are not used during poison prevent, so no water is lost during reject system operation. While rejecting to the main condenser greatly increases the length of time during which poison prevent operation may be conducted, it requires that the main condenser be available and under normal vacuum for the system to operate.

(c) Reject to a Reject Condenser enables the reject system to operate independently of the main condenser and still not deplete the station water inventory. However, this system is more costly than either of the other two systems and requires either that the system be available at all times, or that some other reject system (atmospheric reject or safety valves) be available while the reject condenser is being put into operation.

Load Rejection

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.

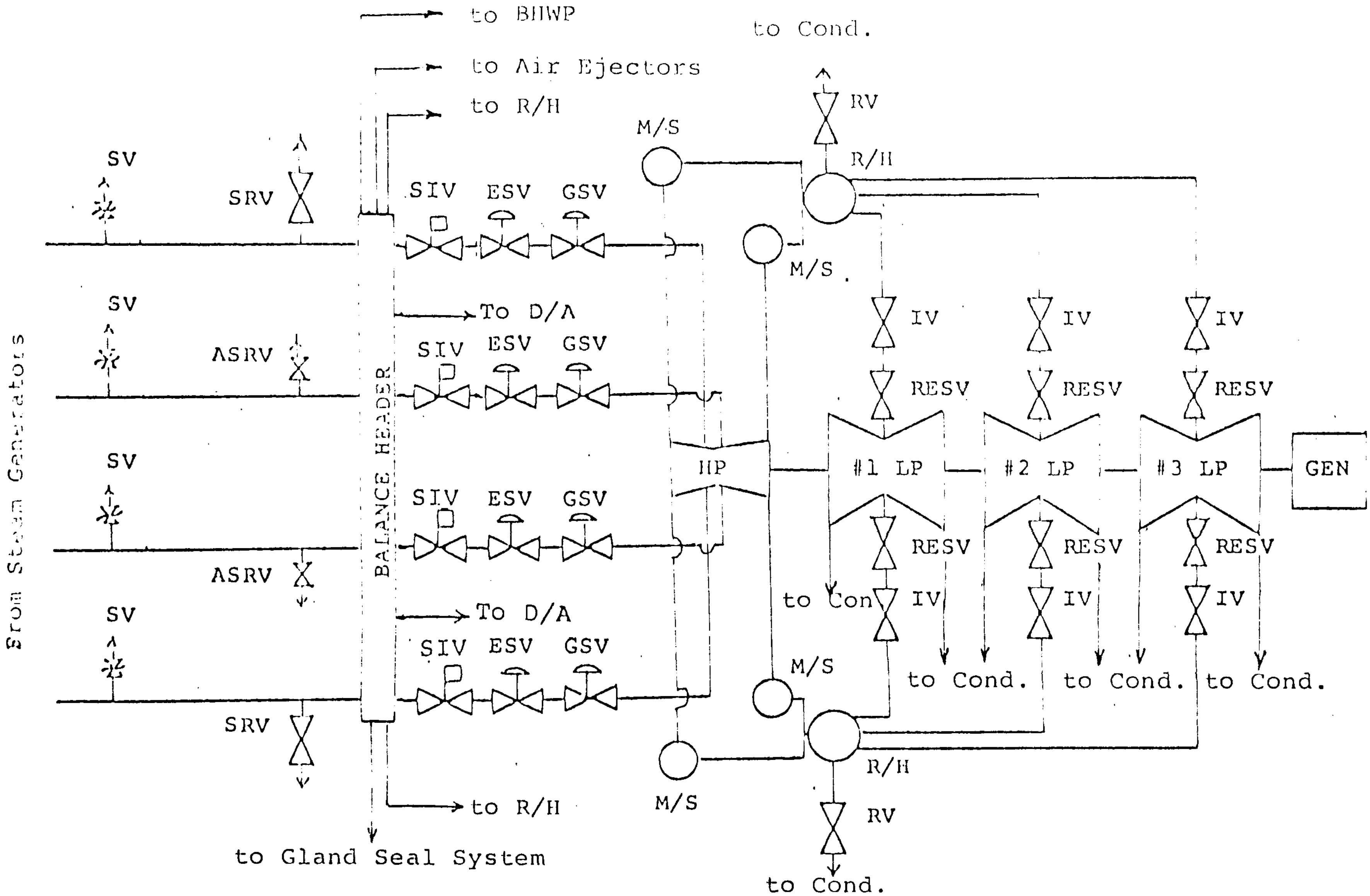
The prevention of such an unterminated overspeed is dependent on the speed with which steam is shutoff. The governor will sense the speed increase and close the governor steam valves. The governor may not be able to operate fast enough to limit the overspeed to acceptable levels. In turbines with such slower acting governors, the emergency stop valve is designed to shut for a short time period, (about five seconds), to rapidly reduce steam flow to the HP turbine, until the governor is capable of handling the overspeed. After sufficient time has elapsed for the governor steam valve to shut, the emergency stop valve will reopen.

On large, multicylinder steam turbines, the volume of steam, which has already passed the governor steam valves, is sufficient to overspeed the turbine by expansion through the low pressure turbine. This "entrained steam", which is contained in the HP turbine, moisture separator, reheater and LP piping, must be prevented from entering the LP turbine. An intercept valve, located on each inlet line, to each LP turbine, performs this funtion. These valves shut at the same time that steam is shutoff to the HP turbine. When the intercept valves shut, the entrained steam must be removed from the turbine unit. This may be done by release valves, which exhaust the steam to the condenser or reheater blow-off valves which exhaust the steam to atmosphere.

In Figure 2.4 there is one release valve for each intercept valve. On some stations there are only two release valves, one for each reheater. While this arrangement, which is shown in Figure 2.6, saves on initial valve cost and maintenance cost, the reduced release capacity requires two modifications to the system:

1. reheater bursting discs are installed to prevent overpressure of the LP pipework in the event of a reheater tube rupture (six release valves will relieve this overpressure, two will not);

2. reheat emergency stop valves are placed in series with each intercept valve. The reduced release valve capacity means that entrained steam is released less rapidly and the consequences of an intercept valve sticking open are more severe. Therefore, two valves are used on each LP turbine steam admission line.



SV = Steam Generator Safety Valves
SRV = Large Steam Reject Valves
ASRV = Small Steam Reject Valves
ESV = Emergency Stop Valves
SIV = Steam Isolating Valves

GSV = Governor Steam Valves
RV = Steam Release Valves
IV = Intercept Valves
RESV = Reheat Emergency Stop Valv

Figure 2.6

Another source of steam, which may continue to drive the unit by expansion in the LP turbine, is the extraction steam contained in the lines to the feedheaters and deaerator. On a turbine trip, this steam tends to flow back to the turbine as the pressure in the turbine decreases. In addition, water in the feedheaters may flash to steam, on decreasing turbine pressure, following a trip. To prevent this from occurring, air motor assisted check valves are placed in the extraction steam lines. These valves close on reverse steam flow and prevent the extraction steam from returning to the low pressure turbine. There are generally no check valves in the extraction steam lines to the first two low pressure feedheaters as the steam pressure is too low to do any real work by backflow and expansion in the remaining LP stages.

Normal Startup

When the turbine is at low speed on the turning gear, the governor senses a speed well below the minimum at which it can control the turbine. The governor speeder gear typically, cannot be set below about 1650 rpm, for an 1800 rpm machine. In this condition the governor steam valves are fully open (calling for maximum steam flow to the turbine) and, therefore, cannot be used to control steam flow to the unit on a startup, while it is below operating speed. The emergency stop valves are used for this purpose.

The turbine is brought up to near operating speed by throttling steam through the emergency stop valves. When the turbine speed rises above about 1650 rpm, the governor becomes able to control the speed of the unit through the governor steam valves. Synchronizing is controlled through the action of the speeder gear and governor controlling the governor steam valves.

Once the generator is synchronized to the grid, it can only run at the speed determined by the grid frequency (1800 rpm for a four pole, 60 Hz generator). The admission of more steam, by further opening the governor steam valves, increases the electrical load supplied to the grid.

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. Through this system turbine/generator power will follow reactor power up to the desired operating load.

ASSIGNMENT

1. Explain how the steam control valves in your station respond to the following:

   (a) reactor trip
   (b) turbine trip
   (c) load rejection
   (d) normal startup and loading.

2. What would be the consequences of failure for each valve(s) listed below? (Consider the three cases: the valve fails open, the valve fails shut, the valve fails in its normal operating position.)

   (a) Steam generator safety valve
   (b) Steam reject valve
   (c) Emergency stop valve
   (d) Governor steam valve
   (e) Intercept valve (plant without reheat emergency stop valves)
   (f) Intercept valve (plant with reheat emergency stop valves)
   (g) Release valve.

3. Explain how steam generator pressure is related to heat transport system temperature. Include in your discussion the variation of steam generator pressure with power level and the effect on the type of heat transport system pressurizing system used.

4. Explain "poison prevent" operation including:

   (a) under what conditions is it used?
   (b) why is it necessary?
   (c) where does the steam produced go?
   (d) how long can it be maintained?

5. Explain "motoring" including:

   (a) under what conditions is it used?
   (b) why is it necessary?
   (c) what are the problems and how are they moderated?
   (d) how long can it be maintained?

6. Why are two sizes of reject valves used on most plants?

R. Schuelke

Turbine, Generator & Auxiliaries - Course 334

THE CONDENSER

In the section on the steam turbine, we discussed how the turbine converted the heat energy of the steam passing through it to the mechanical energy of the rotating shaft. It is obviously to our advantage to extract as much work as possible from the saturated steam that is generated in the steam generator. In the simplest terms, the lower the temperature and pressure at the outlet of the low pressure turbine, the greater will be the amount of energy which can be extracted from the steam. In fact, if the exhaust of the turbine is near a perfect vacuum rather that at atmospheric pressure, roughly 35% more energy can be extracted from the steam passing through the turbine. It is the condenser which provides the means of maintaining this low absolute pressure at the exhaust of the low pressure turbine. The way in which the condenser provides this low pressure is through condensation of the steam into water.

At the outlet of the low pressure turbine, a kilogram of wet steam occupies 28 cubic meters. When this steam condenses to water, the one kilogram of water occupies .001 cubic meters (about one quart). Thus as one kilogram of steam condenses, almost 38 cubic meters of empty space (which was previously filled with steam) is left behind. It is this creation of empty space through the condensing of steam which provides the vacuum in the condenser.

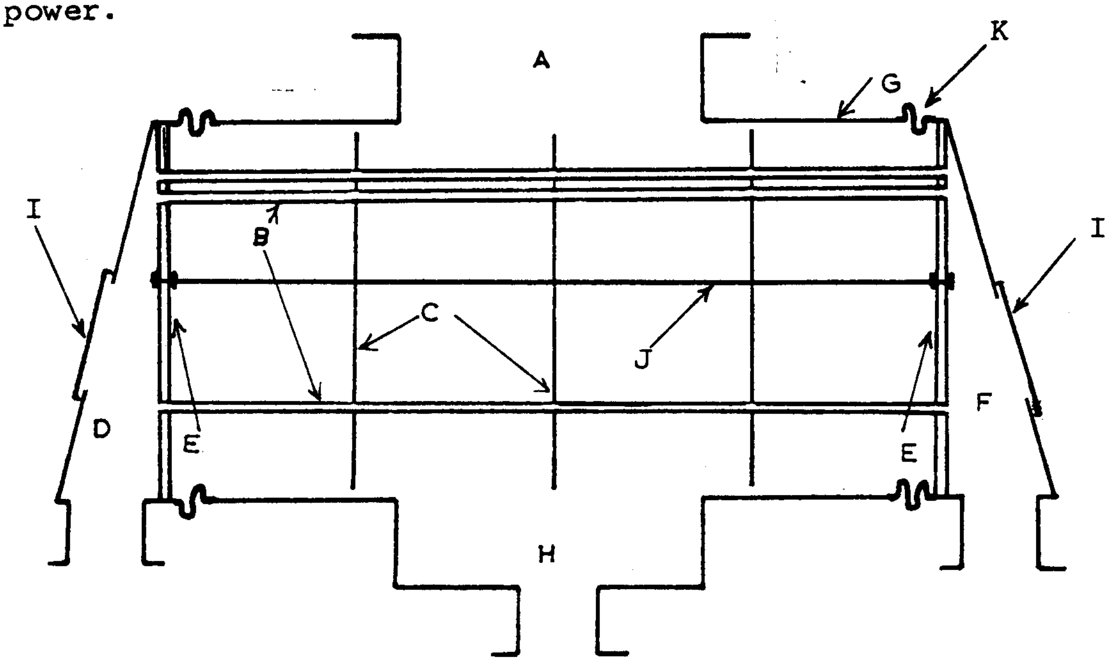
Figure 7.1 shows a sectional view of a condenser. The cooling water which removes the latent heat of vapourization and causes the steam to condense flows through the tubes. This condenser cooling water is taken from the lake or river, pumped through the condenser tubes, and then flows back to the lake or river. The condenser cooling water inlet to the condenser is known as the <u>inlet water box</u> and the condenser cooling water outlet as the <u>outlet water box</u>.

The type of condenser shown in Figure 7.1 is known as a single pass condenser since the cooling water passes through the condenser only once. Double pass condensers are also found in the Nuclear Generating Division but only on the smaller units. The tubes are supported at either end by <u>tube sheets</u> and along their length by <u>sagging plates</u>.

In addition to giving support to the condenser tubes, the sagging plates are spaced to ensure that the tubes are damped against vibration induced either by resonance with the turbine running frequencies or by steam passing over the tubes. The sagging plates are bolted or welded to the condenser shell and they contribute to the condenser shell strength. When the condenser is under full vacuum, the shell must withstand an external pressure of almost 100 kPa. The sagging plates help counteract this pressure force which is tending to collapse the condenser shell. Axial support is given to the tube sheets by <u>staybars</u> which run parallel to the tubes. These staybars tend to prevent any significant axial stresses from being carried by the tubes.

On the steam side of the condenser, the steam leaves the low pressure turbine through an <u>exhaust trunk</u> and enters the <u>shell</u> of the condenser where it passes around the tubes. The <u>condensed water</u> or <u>condensate</u> falls into the bottom of the condenser and is collected in a <u>hotwell</u>.

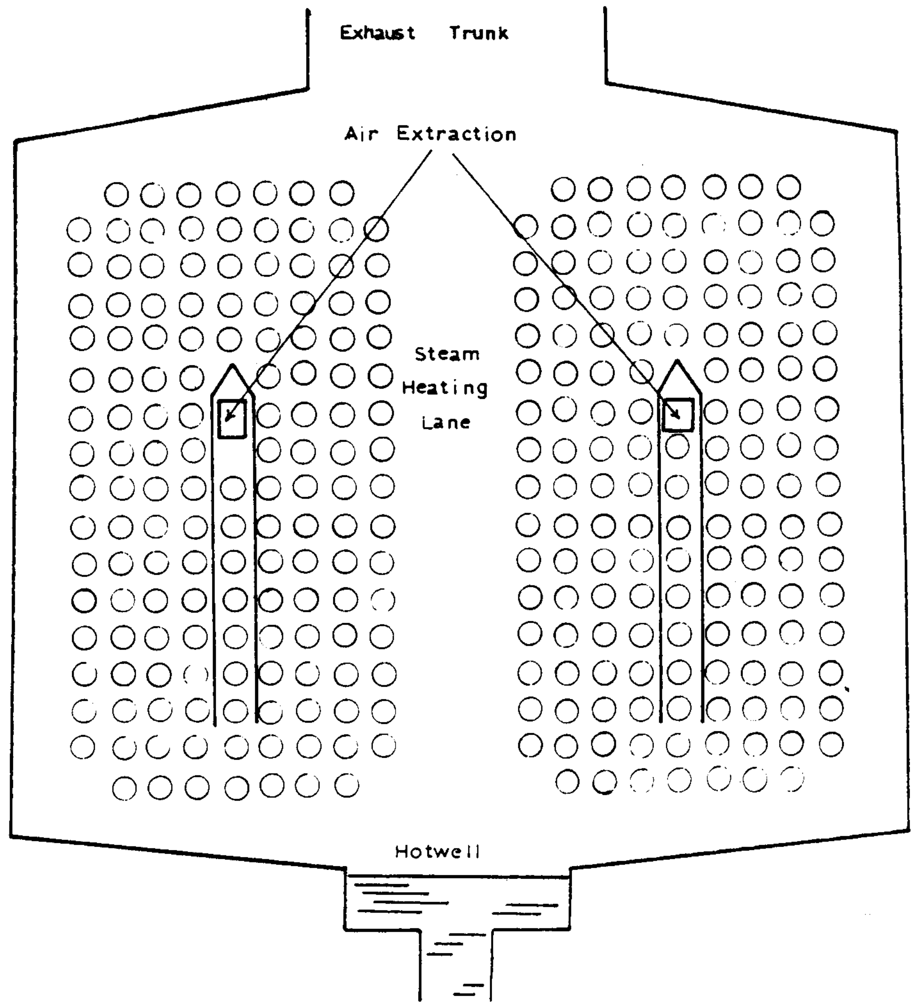
A large turbine unit generally has one condenser for each low pressure turbine. However, to prevent variations in the back pressures of each condenser, the shells of the condensers are joined by a large (say 2 metre diameter) <u>balance pipe</u>. This balance pipe also allows a condenser half to be shutdown for cleaning while the turbine unit remains at power.



A. Exhaust Trunk
B. Tubes
C. Sagging Plates
D. Inlet Water Box
E. Tube Sheet
F. Outlet Water Box
G. Shell
H. Hotwell
I. Inspection Doors
J. Staybars
K. Expansion Joints

<u>Single Pass Condenser</u>

<u>Figure 7.1</u>



Condenser Cross Section

Figure 7.2

Figure 7.2 shows a cross section of a typical single pass condenser. The exhaust trunk may be bolted or welded to the bottom of the LP turbine casing. Alternately it may be supported independently and connected to the LP turbine by a flexible bellows. The latter method is more typical of large turbine units since it allows the condenser to expand independent of the LP turbine.

There is a large steam heating lane or steam excess lane down the centre of the condenser shell which contains no cooling water tubes. The purpose of this lane is to ensure that a portion of the exhaust steam goes to the bottom of the condenser. By allowing a significant percentage of the steam to reach the bottom of the condenser, the following is achieved:

1. The water in the hotwell is kept at saturation temperature. If the water in the hotwell was allowed to become subcooled, it would be an unnecessary loss of heat from the condensate. This unnecessary loss of heat energy would make the steam/feedwater cycle less efficient as this lost heat would have to be put back into the condensate by either the feedheating system or the steam generator.

2. The lower condenser tubes are forced to transfer as much steam out of the condenser as the upper tubes. This ensures the entire tube surface is used to transfer heat.

3. The thermal expansion of the tubes is equalized. If the lower tubes were cooler than the upper tubes, it would produce a bending force on the condenser shell and tubesheets.

Condenser Air Extraction

The steam will continue to condense and maintain a good vacuum as long as four conditions are fulfilled:

1. No air enters the shell (steam) side of the condenser.

2. The tubes carry a normal flow of relatively cool lake water.

3. The tubes remain exposed to the steam.

4. The tube surfaces are not fouled by corrosion products or other materials.

Any change in the system which invalidates one of these statements will result in a decrease in the vacuum in the condenser. Apart from a decrease in turbine efficiency, a decreasing vacuum will result in overheating of the low pressure turbine blading as the blades must pass through higher density steam than they were designed to encounter.

If air enters the condenser, most of it will remain there because it cannot condense with the steam. The air will gradually fill up the empty space left by the condensing steam until the vacuum in the condenser is destroyed. In addition, some of the air which leaks into the condenser will dissolve in the condensing water. The oxygen in the air can cause considerable corrosion problems in the condensate and feedwater systems. Since the condenser operates below atmospheric pressure, there is a tendency for air to leak into the condenser and this air must be removed.

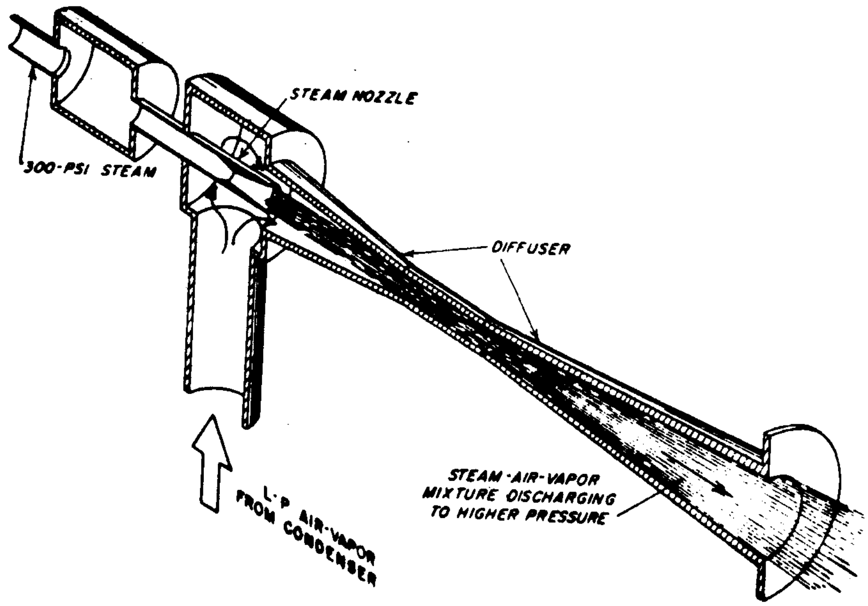
The normal method of starting up a turbine unit is to evacuate the condenser shell and turbine to a partial vacuum and then to evacuate the remaining air after rolling the turbine with steam. Once the condenser is at normal operating vacuum [about 5 kPa(a)], the air extraction system must remove only the small amount of air which leaks into the condenser.

The air extraction system has to be capable of dealing both with vacuum raising and normal maintenance of vacuum. When raising vacuum, the air extraction system has to remove not only the air which fills the condenser shell during shutdown, but also the air in the HP and LP turbines and pipework back to the emergency stop valves. The volume of air which must be removed is typically about 6500 cubic meters. To prevent excessive startup time being expended in drawing a vacuum, this air must be removed in something on the order of an hour. On the other hand, the maximum amount of air leakage into an operating unit is something like one cubic meter per minute. This requires nowhere near the air removal capacity that initial evacuation requires. Whatever the method of air extraction, there will typically be a large air removal capacity for initial evacuation and a smaller capacity for maintaining vacuum.

The air extraction system removes air and other non-condensable gases from the condenser by either a steam air ejector or vacuum pumps. Whatever the method of air removal, the system creates area of lower pressure than condenser vacuum. Air flows from the condenser to this low pressure area where it can be removed. In Figure 6.2, you can see the air extraction points in the condenser. They are located within the tube bundles to ensure that gases moving toward these points must pass over many tubes prior to removal from the condenser. The air extraction system will remove any gas and would just as soon suck out steam as air.However, in passing over the tubes, the steam is condensed and the air extraction system only removes non-condensable gases.

Steam Air Ejectors

Figure 7.3 shows a sectional view of a typical steam air ejector. Steam at about 2000 kPa(g) enters the nozzle where steam heat energy is converted to velocity. The high velocity steam jet is directed into a diffuser. The diffuser inlet has a much larger cross sectional area than that of the high speed jet entering it. This creates an extremely low pressure at the inlet to the diffuser. This low pressure area is connected to the condenser. The low pressure air-vapour mixture from the condenser comes into physical contact with the



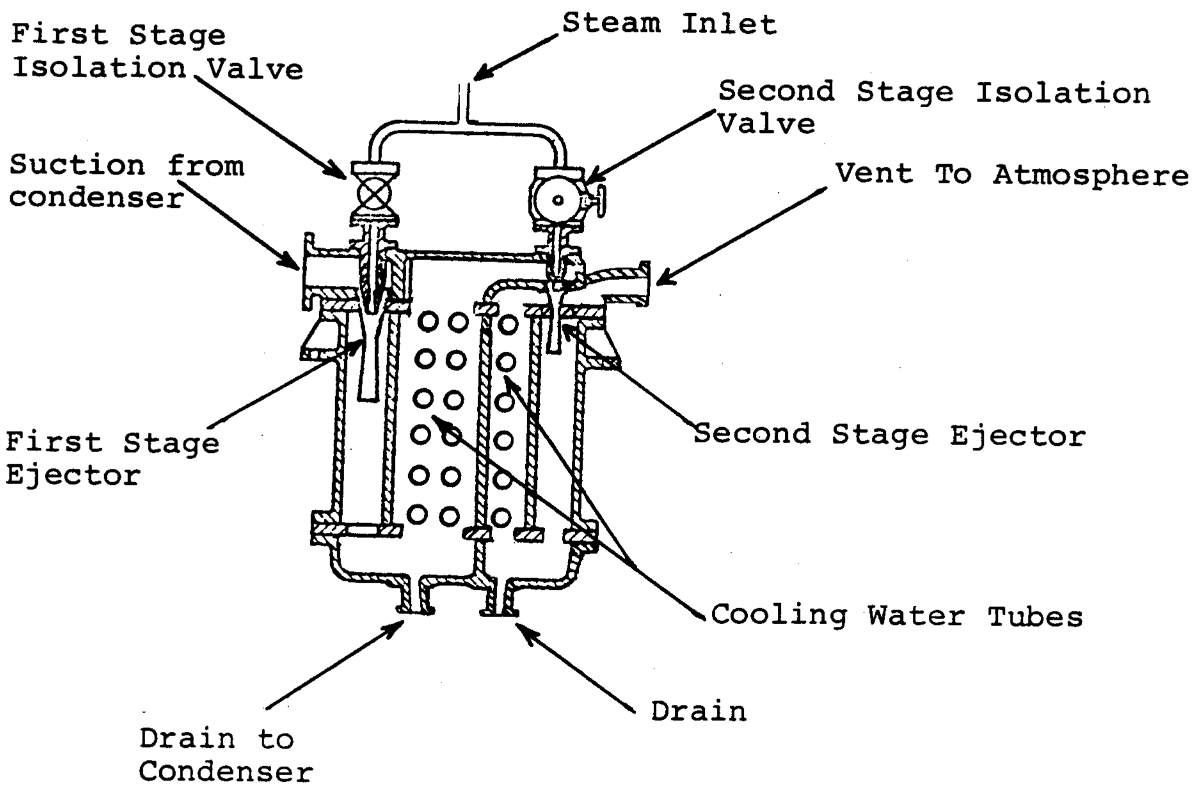
Steam Air Ejector

Figure 7.3

high speed jet and is "batted" along by the steam entering the diffuser. The air travels through the diffuser with the steam and is discharged to a higher pressure region. This constant removal of low pressure airvapour mixture at the diffuser inlet encourages more air to flow into the diffuser. This establishes a continuous flow from condenser into ejector.

For startup purposes, when large volumes of air must be removed, a high capacity steam ejector called a hogging ejector is used. This high capacity ejector can evacuate the condenser and turbine unit to normal vacuum in something like 45 minutes. The hogging ejector uses large quantities of steam. After passing through the ejector, the steam is released to atmosphere. While this improves the capacity of the hogging ejector, it makes the ejector wasteful of steam. Thus while the hogging ejector is desirable for rapid vacuum raising, it is far too inefficient for normal operation.

During normal operation, a much smaller steam ejector is used. This holding ejector as it may be called, is normally a two stage steam ejector as shown in Figure 7.4. When the air ejector is working properly, the air from the main condenser is drawn into the first stage. The steam and air mixture from the first stage pass into an intercondenser where most of the steam is condensed. The air is then drawn



Typical Two-Stage Condensing Air Ejector

Figure 7.4

into a second stage. The steam and air mixture from the second stage pass into an aftercondenser where the steam is condensed and the air released to atmosphere. The cooling water for both the intercondenser and aftercondenser is main condensate taken from the condensate extraction pump prior to the low pressure feedheaters. The advantage of having two stages in the air ejector is that the air is raised from condenser vacuum to atmospheric pressure in two steps rather than one. The first stage raises pressures from 5 kPa(a) to about 35 kPa(a), while the second stage raises pressure from 35 kPa(a) to normal atmospheric pressure - 101.3 kPa(a).

Vacuum Pumps

An alternative method of air extraction is use of vacuum pumps. These pumps are of a variety of designs although they are normally rotating positive displacement pumps. The usual configuration is to have more than one pump. The number of pumps running depends on the vacuum conditions. Pickering NGS-A, for example, has three screw type positive displacement vacuum pumps for each condenser. All three pumps are used for initial vacuum raising. During normal operation, only one pump is required. However, if backpressure rises to 8 kPa(a), a second pump will start; if the backpressure rises to 12 kPa(a), the third pump will start.

Another possible combination is to use vacuum pumps for normal vacuum maintenance and a hogging ejector for initial vacuum raising.

There is really no clearcut advantage of one air extraction method over the other. Vacuum pumps allow easier on-off control and tend to increase plant efficiency since they don't use steam. However, in practice, they require more maintenance and are generally less reliable.

Condenser Cooling Water

The condenser cooling water system, or CCW system as it is most often called, is used to remove the latent heat of vapourization from the exhaust steam entering the condenser. The rise in CCW temperature across the condenser is limited to about 10°C for three reasons:

1. The higher the temperature rise of CCW across the condenser, the higher condenser pressure and therefore, the lower cycle efficiency.

2. The higher the temperature rise, the greater the tendency for air to come out of solution and collect in the high points of the CCW system.

3. If the water leaves the condenser much warmer than it enters, there is a rapid growth in marine life in the warm outflow. The prevention of this biologic polution has resulted in legal limits on condenser temperature rise.

The heat energy which must be removed from each kilogram of exhaust steam is about 2450 KJ. In increasing its temperature by 10°C, a kilogram of CCW water can remove about 42 KJ. Thus it requires about 58 kilograms of CCW water (2450 KJ/42 KJ) to condense one kilogram of exhaust steam. It requires about 60 cubic meters of CCW flow per second for a single large turbine unit. This extremely large flow rate presents two problems:

1. an extremely large amount of pumping power must be consumed, and

2. there can be little placed in the CCW system for water treatment and purification which impedes flow.

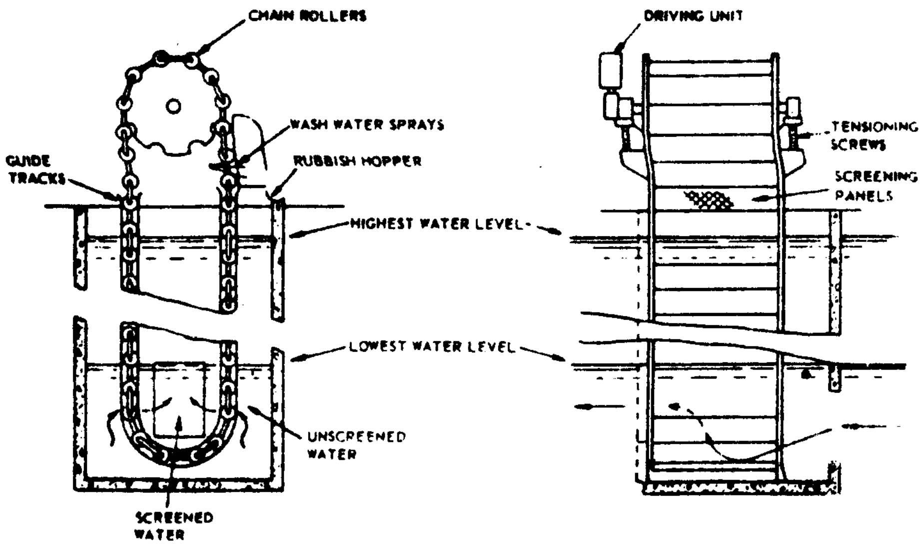
For a large CANDU generating station, the required CCW pumping power is on the order of 4 MW per unit. To keep this pumping power within economic limits, the pumps are generally of high capacity and low discharge head. Since the condenser is normally located above the level of the lake or river, appreciable pumping power may be expended to raise the water up to the condenser. Much of this power requirement can be eliminated by operating the CCW system as a siphon. This means the pump is only required to overcome the friction flow losses.

When a CCW system is operated as a siphon, the CCW side of the condenser operates below atmospheric pressure. As water passes through the condenser, air which has dissolved in the CCW water comes out of solution. These gases tend to collect in the condenser tubes and water boxes blocking flow and destroying the siphon. The CCW system is protected against the air binding of the tube side of the condensers by an air removal system. This vacuum priming system as it is called, removes air from the condenser water boxes. It is worth mentioning that this system is completely separate from the air removal system for the shell side of the condenser.

Because of the extremely large flow rate of CCW water, there is little water treatment except intermittent chlorination and trash removal. Water first passes through a coarse screen. The spacing between bars must be small enough to prevent logs, boats, people or ice from being swept into the CCW system. On the other hand, the spacing must be wide enough to prevent a large quantity of small fish or seaweed from rapidly plugging the screen. The coarse screen spacing is generally 2 to 15 cm, with exact spacing largely determined by the water conditions near the station: the "dirtier" the water, the finer the mesh of the coarse screen.

The CCW flow next passes through a band screen or travelling screen as it is alternatively called. This screen has a fine mesh (typically 1 cm or less) and is driven by a motor. When the screen accumulates sufficient trash to raise the differential pressure, the motor is turned on and the screen slowly rotates to a new position. The screen is washed with high pressure water jets near the top of its travel. The purpose of the screen is to block any trash large enough to block the tubes.

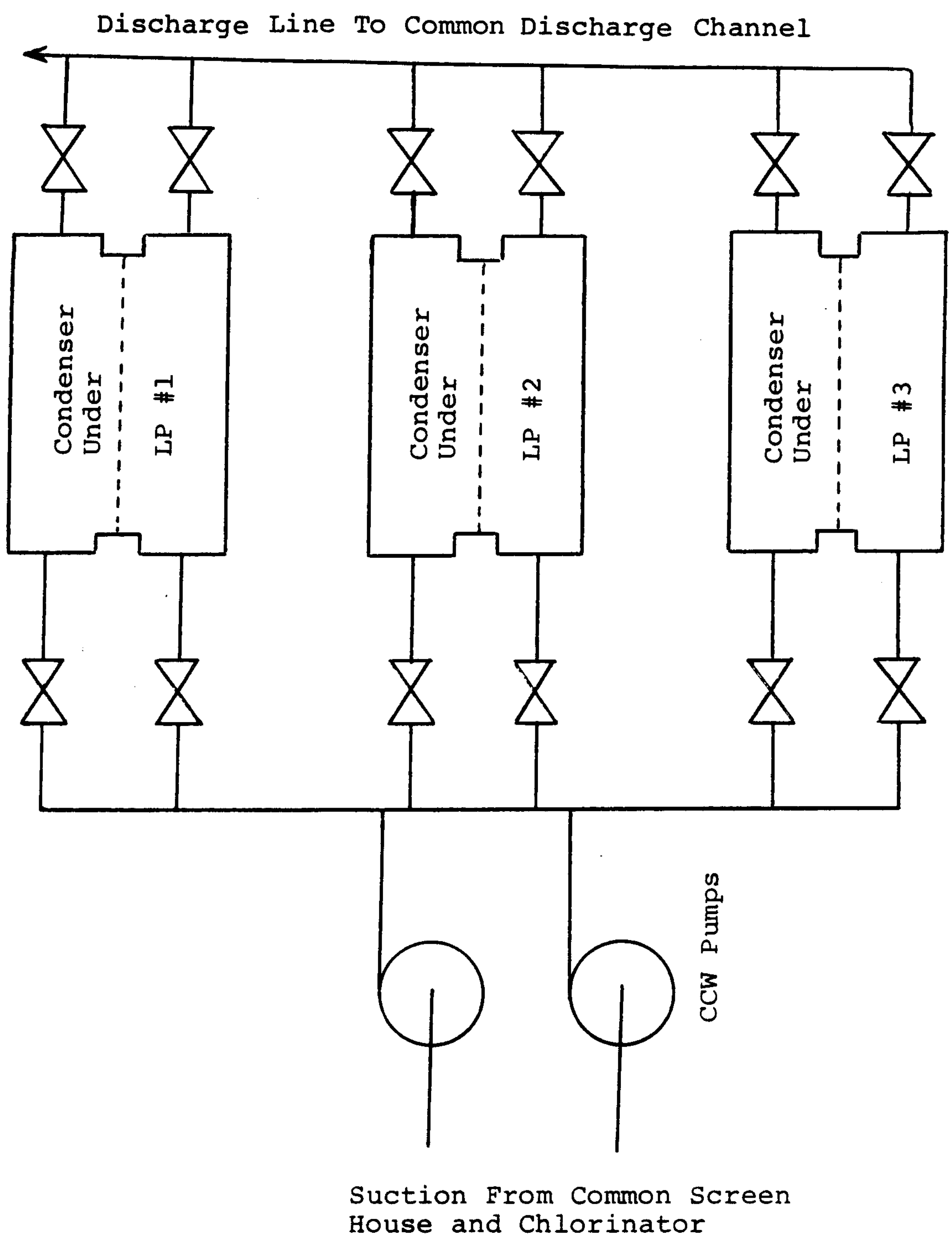
Some marine life may pass both the coarse and travelling screens and accumulate on the condenser tube sheets and tube surfaces. The warm conditions and constant renewing of water result in rapid growth of this marine life. This growth can rapidly result in tube plugging. To restrict the growth rate chlorine is intermittently injected into the CCW intake. The chlorine injection rate must be low enough to prevent poisoning of the marine life in the CCW outflow.



Travelling Screen

Figure 7.5

Regardless of what methods are used to restrict tube and tube sheet fouling, eventually the tube sheets (particularly the inlet tube sheet) must be cleaned. To facilitate cleaning of condenser CCW sides while operating, the circulating water system of large stations supply individual condenser halves which can be isolated from each other. The arrangement shown in Figure 7.6 allows half of one condenser to be isolated and opened for cleaning. The balance piping between condensers ensures the remaining five condenser halves are sufficient to keep the unit at power.



CCW System For Single Turbine
Unit with Three LP Turbines

Figure 7.6

ASSIGNMENT

1. Why does the LP turbine exhaust to a vacuum rather than to atmosphere?

   How is this vacuum maintained?

2. Draw a cross section of a typical condenser showing:

   (a) exhaust trunk
   (b) tubes
   (c) tube sheets
   (d) inlet and outlet water boxes
   (e) sagging plates
   (f) staybars
   (g) shell
   (h) hotwell
   (i) expansion joints
   (j) waterbox inspection doors

3. Why are turbine units with more than one condenser fitted with balance pipes between condensers?

4. Why are the air extraction points for the condenser shell surrounded by tubes?

5. What is the purpose of a steam heating lane.

6. Why must air be extracted from a condenser shell? From condenser waterboxes?

7. What is a hogging ejector?

8. How does a steam air ejector work?

9. What provisions are made to prevent blockage of condenser tubes?

R.O. Schuelke

<u>PI 25-6</u>

Turbine and Auxiliaries - Course PI 34

THE MOLLIER DIAGRAM AND THE TURBINE PROCESSES

<u>Objectives</u>

1. Sketch a Mollier diagram from memory. Label the following on your sketch:

   (a) constant enthalpy lines
   (b) constant entropy lines
   (c) saturation line
   (d) constant temperature lines
   (e) constant pressure lines
   (f) constant moisture content lines
   (g) constant degree of superheat lines

2. On the sketch of a Mollier diagram that you have drawn, illustrate these turbine processes:

   (a) expansion in the high pressure turbine
   (b) moisture separation
   (c) reheat
   (d) expansion in the low pressure turbine

3. Explain how moisture separation and reheat each:

   (a) increase the enthalpy of the steam at the LP turbine inlet
   (b) reduce the moisture content of the steam at the LP turbine outlet

4. Define <u>throttling</u> and, using a Mollier diagram, explain how throttling of the steam supplied to the turbine affects:

   (a) the pressure, temperature and moisture content of the steam at the turbine inlet
   (b) the amount of heat which can be converted into mechanical energy by the turbine.

Mollier Diagrams and The Turbine Set

This module deals with the turbine process. When you have finished the module, you should be aware of the major equipment of the turbine set and you should be able to show the turbine process on a Mollier diagram.

Mollier Diagrams

The Mollier diagram is a very useful tool. It can be used to depict the various processes associated with the turbine set. It may also be used quantitatively for various calculations. Your consideration of Mollier diagrams in this course will be limited to a qualitative look at the turbine set.

A Mollier diagram for water is shown in Figure 6.1. The axes of the diagram are enthalpy and entropy. Remember that enthalpy is the heat content of water above 0°C reference point.

Entropy is harder to explain since it has no physical significance. For our purposes, entropy can be considered as an indicator of the availability of heat energy to do work. Say we have two substances that have the same heat content, but one substance is at lower temperature than the other. The amount of work that can be done by the lower temperature substance is less than the work that can be done by the higher temperature substance. The entropy of the lower temperature substance is greater than that of the other substance. Thus, the greater the entropy, the less work is available from a given amount of heat.

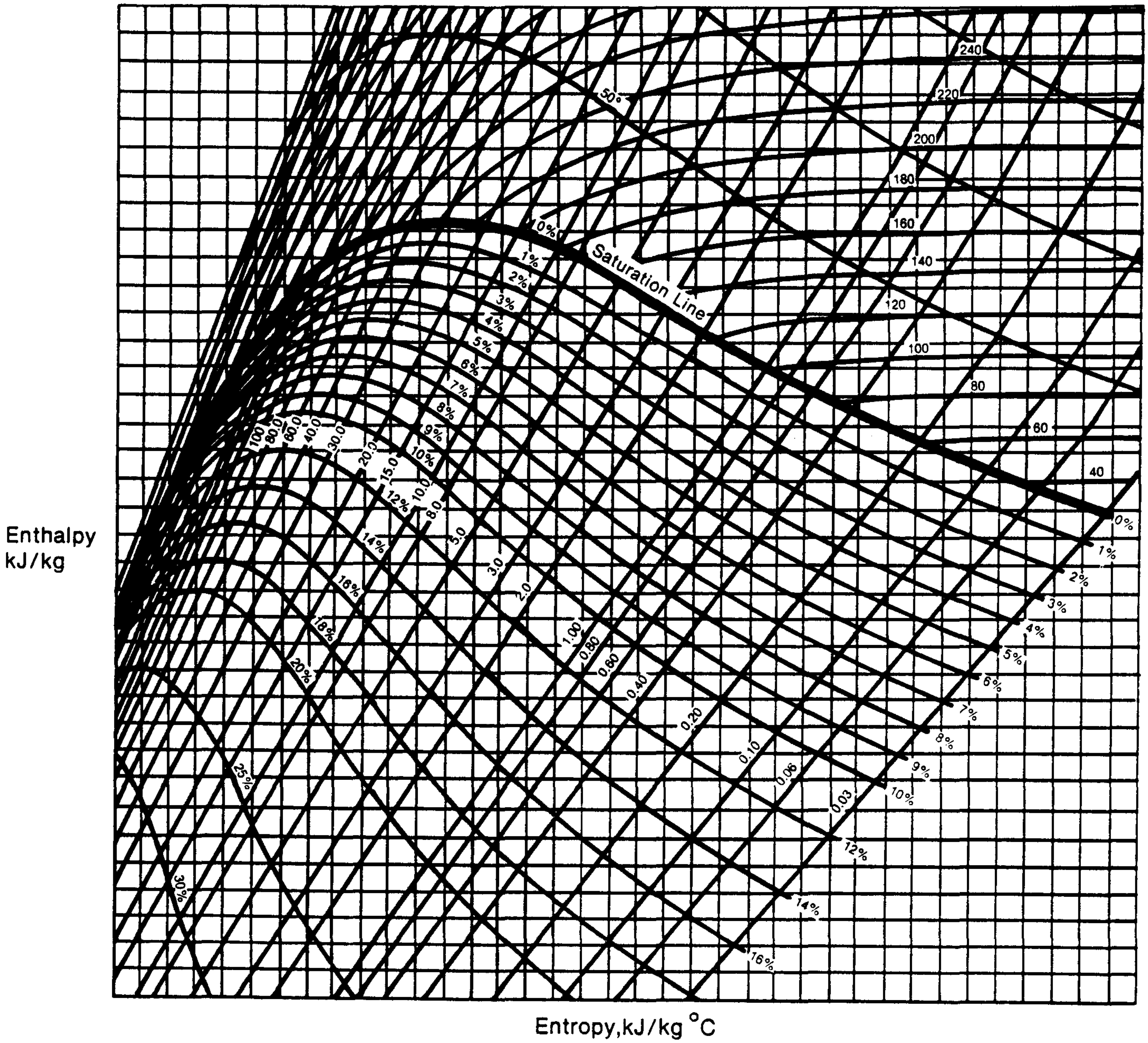

Enthalpy kJ/kg
Entropy,kJ/kg °C
Saturation Line
50°
240
220
200
180
160
140
120
100
80
60
40
0%
1%
2%
3%
4%
5%
6%
7%
8%
9%
10%
12%
14%
16%
18%
20%
25%
30%
100
80.0
60.0
40.0
30.0
20.0
15.0
10.0
8.0
5.0
3.0
2.0
1.00
0.80
0.60
0.40
0.20
0.10
0.06
0.03

Figure 6.1

The Mollier diagram for water deals with three states of water: saturated steam, wet steam, and superheated steam. On Figure 6.2, the darker line marked saturation line represents saturated steam at different conditions of temperature and pressure. Superheated steam is represented above the saturation line. Wet steam is represented below the saturation line.

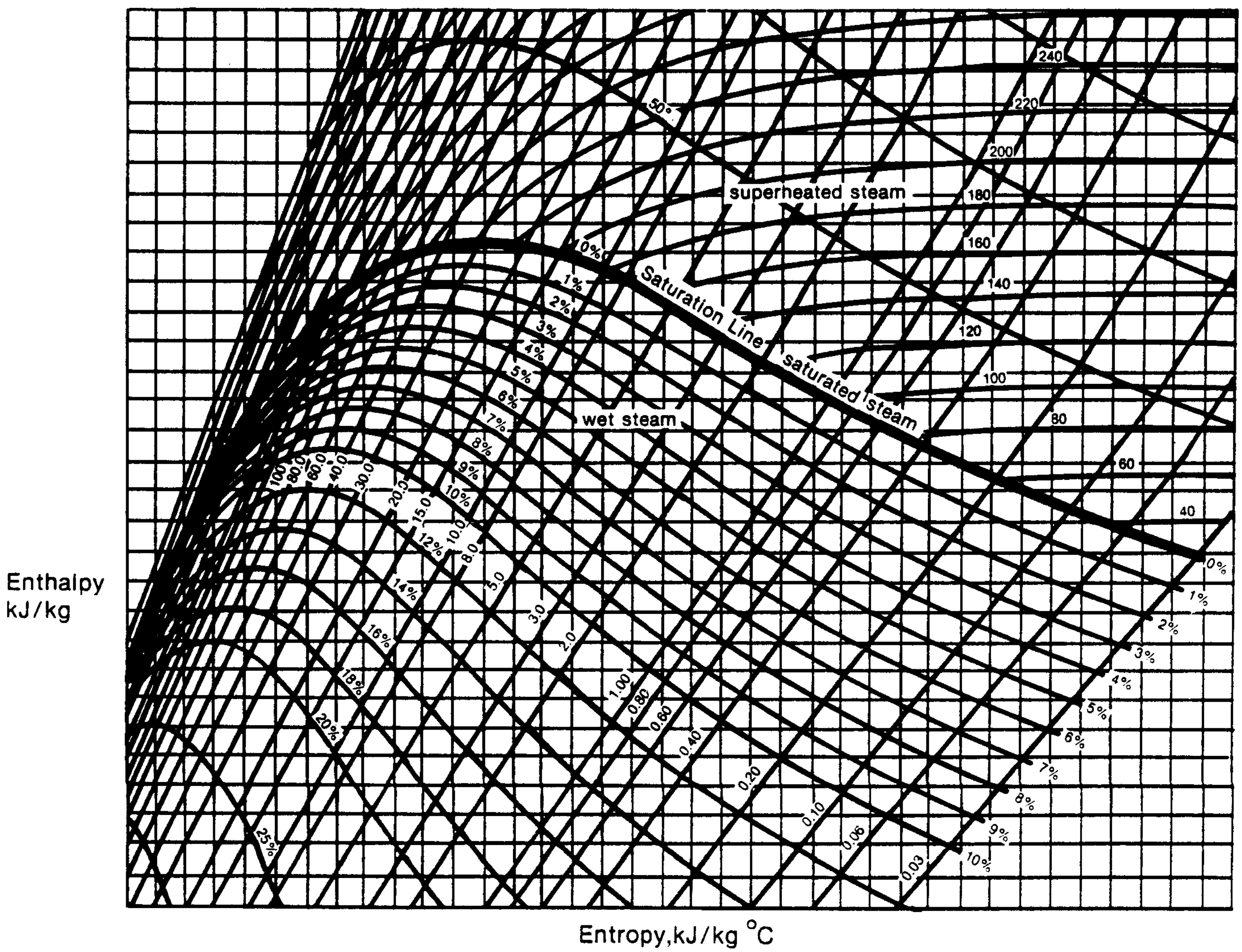


Figure 6.2

Besides enthalpy and entropy, there are other variables shown on a Mollier diagram.

Figure 6.3 shows a Mollier diagram with constant temperature lines indicated.

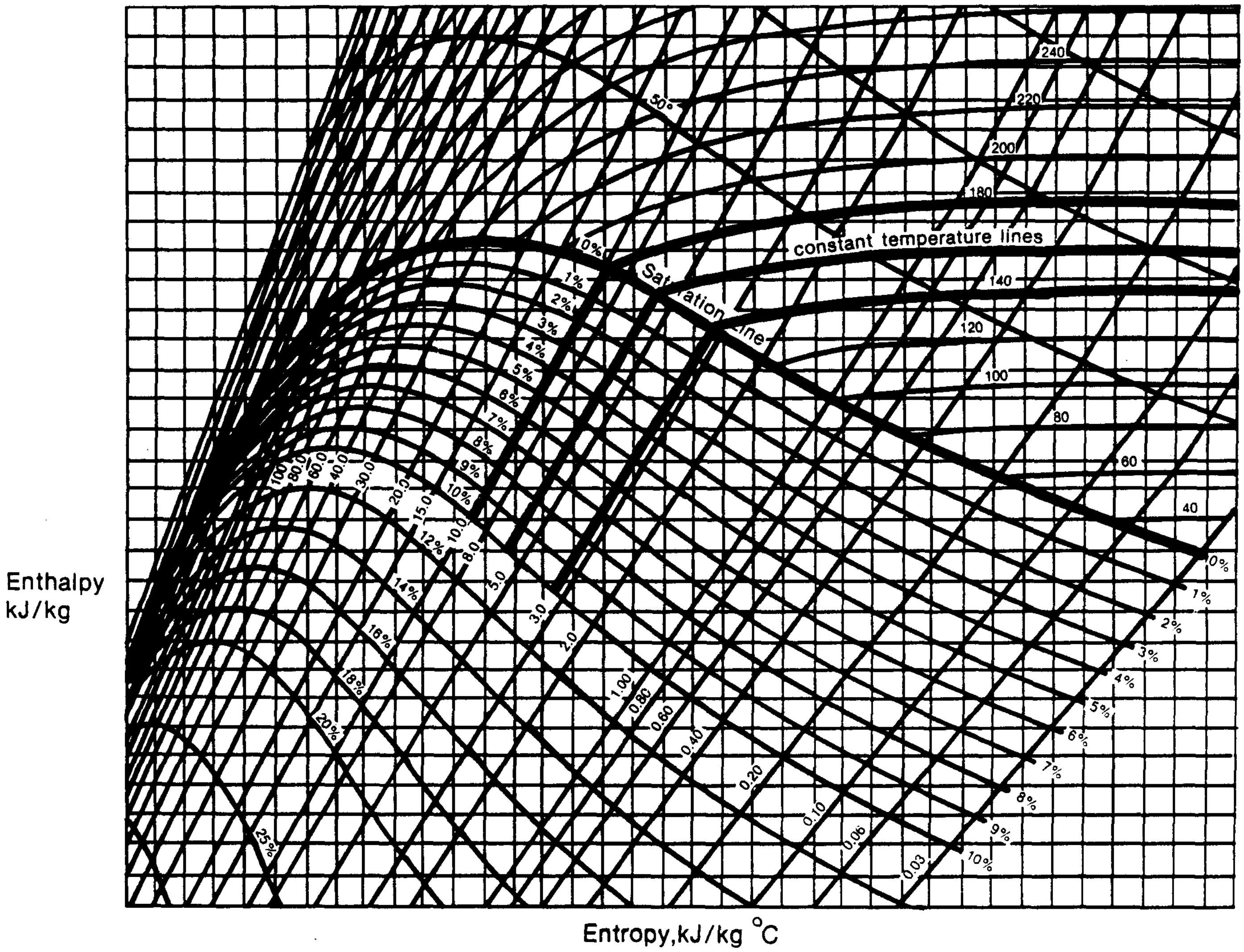


Figure 6.3

Figure 6.4 shows constant pressure lines. Note that the constant pressure lines and constant temperature lines are parallel in the wet steam region, but not in the superheated steam region. This is because for saturated conditions (e.g. wet steam) the temperature remains constant at constant pressure as heat is added. For conditions which are not at saturation (e.g. superheated steam) the temperature does not remain constant as heat is added at constant pressure.

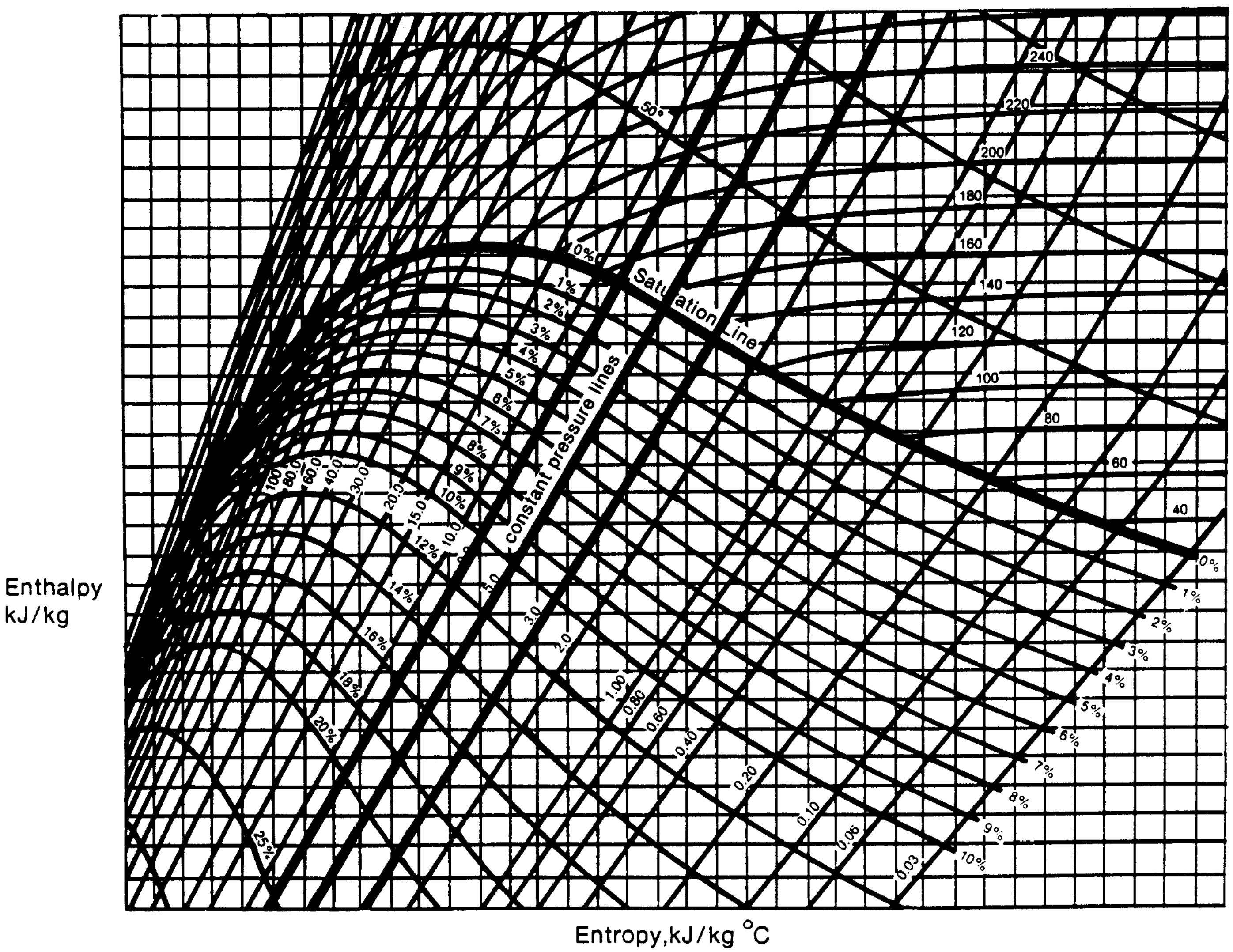


Figure 6.4

Figure 6.5 shows constant moisture content lines. These show wet steam at different temperatures and pressures that have the same percent moisture.

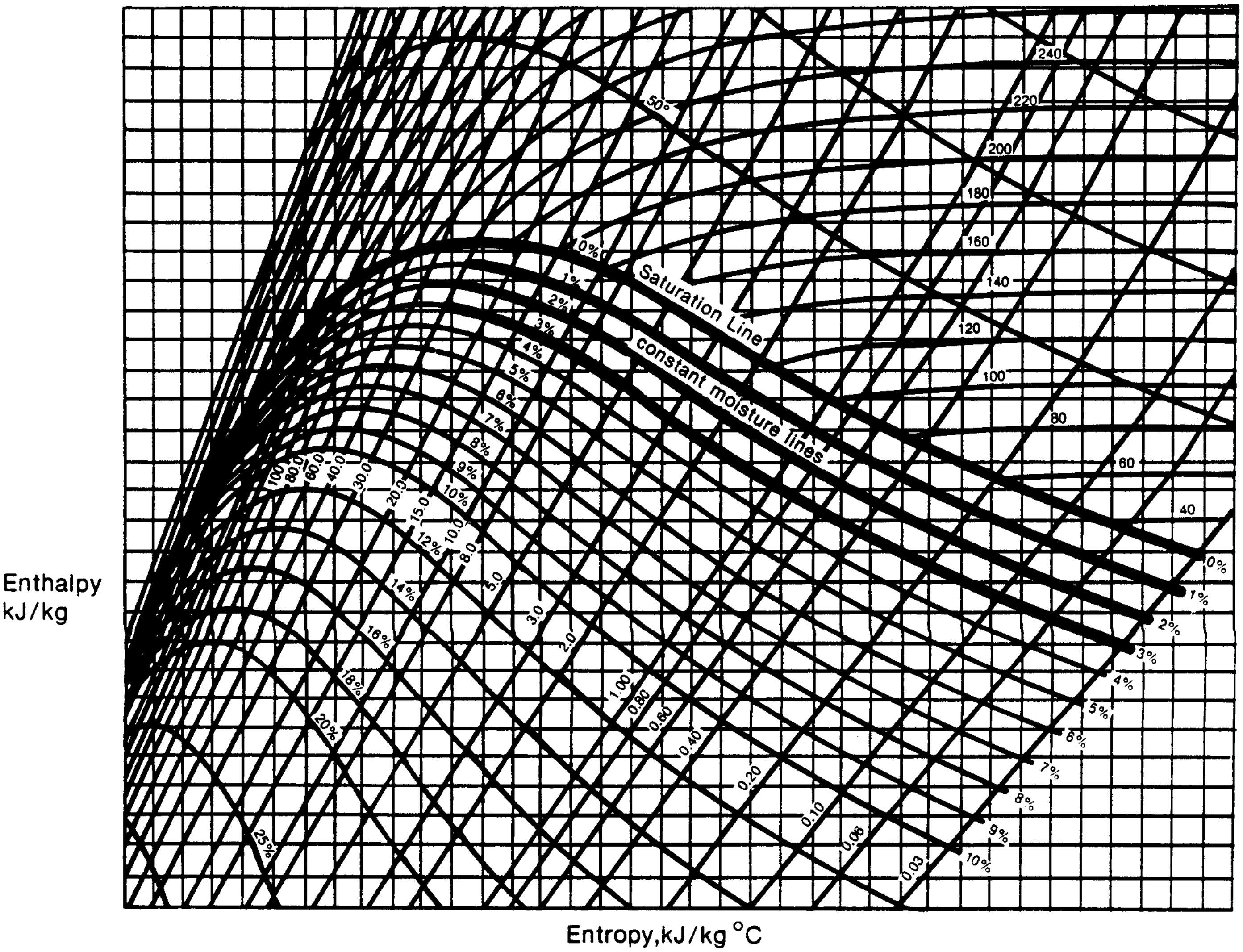


Figure 6.5

Figure 6.6 shows constant degree of superheat lines. These show superheated steam at different temperatures and pressures that are at constant temperature difference above the saturation temperature.

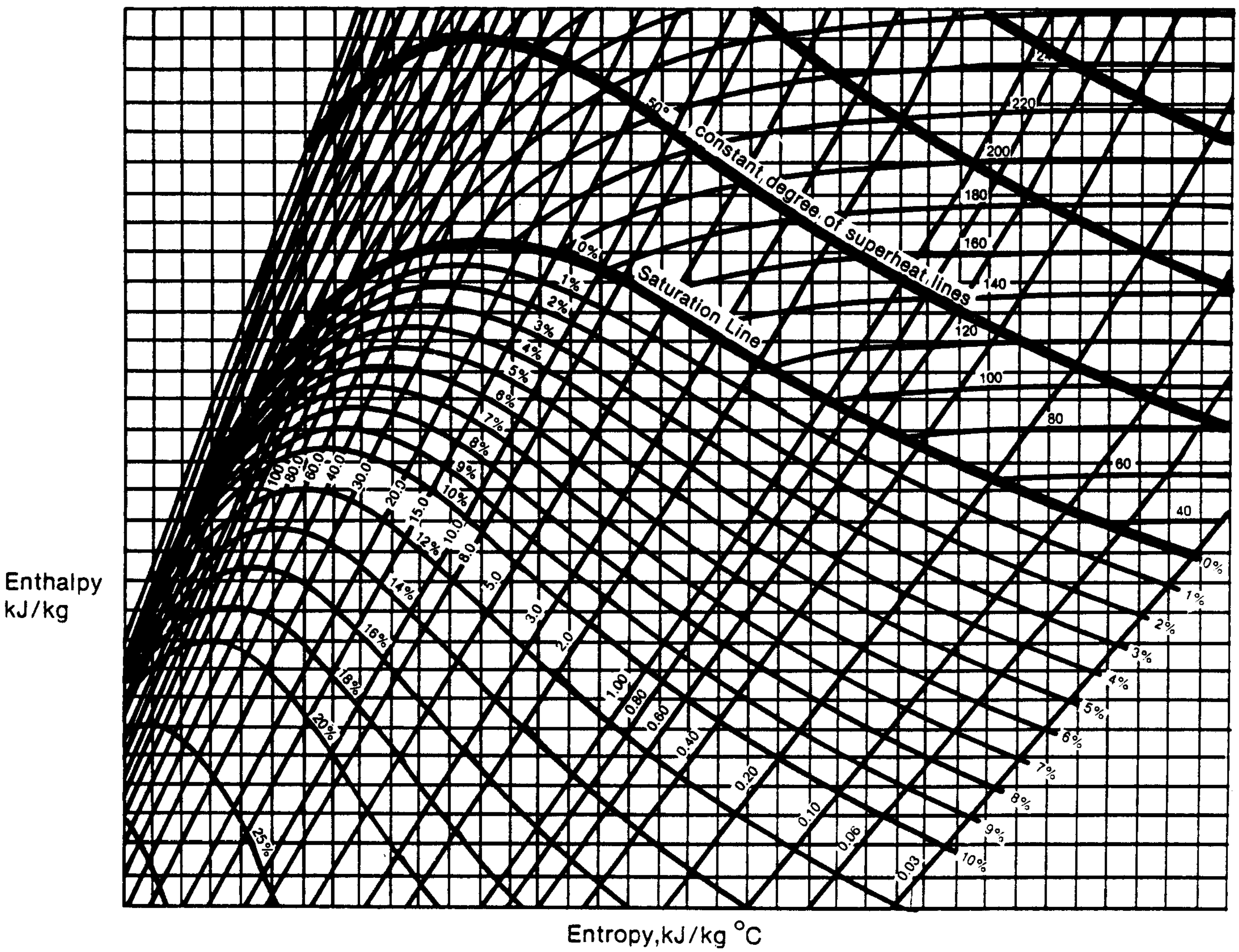


Figure 6.6

→ Answer the following question in the space provided, then check your answer with the "TEXT ANSWERS".

6.1) From memory, sketch and label a Mollier diagram for water. Your labels should include the following:

(a) constant enthalpy lines
(b) constant entropy lines
(c) saturation line
(d) constant temperature lines
(e) constant pressure lines
(f) constant moisture content lines
(g) constant degree of superheat lines

The Turbine Process

The turbine set of a large CANDU unit is shown in Figure 6.7. Note that this diagram is somewhat simplified for clarity.

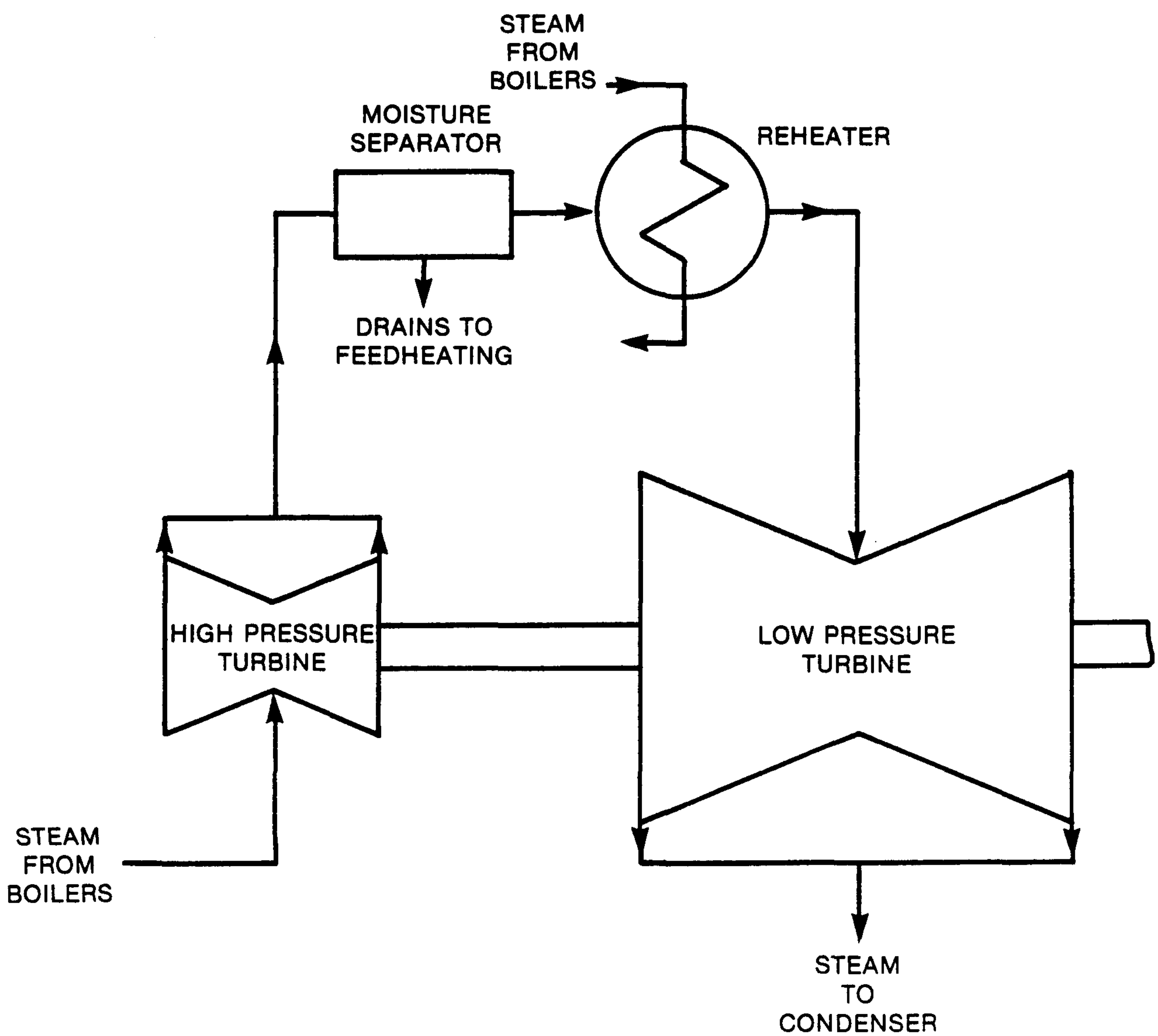


Figure 6.7

Saturated steam at about 250°C and 4 MPa(a) comes from the boilers to the high pressure (HP) turbine. As the steam flows through the turbine, its pressure and temperature drop. The effect of these changes is to turn the turbine shaft - i.e. some of the heat energy of the steam is converted into shaft mechanical energy.

As the steam supplied to the turbine is saturated, the heat which is extracted from the steam is a portion of the latent heat of vaporization. The effect of this is that the steam starts to condense and as it flows through the turbine more and more moisture is produced. Finally, at the outlet of the HP turbine, the steam typically has a moisture content of about 10% and pressure of about 800 kPa(a) with a corresponding saturation temperature in the order of 170°C. This part of the overall process is shown (from A to B) on Figure 6.8

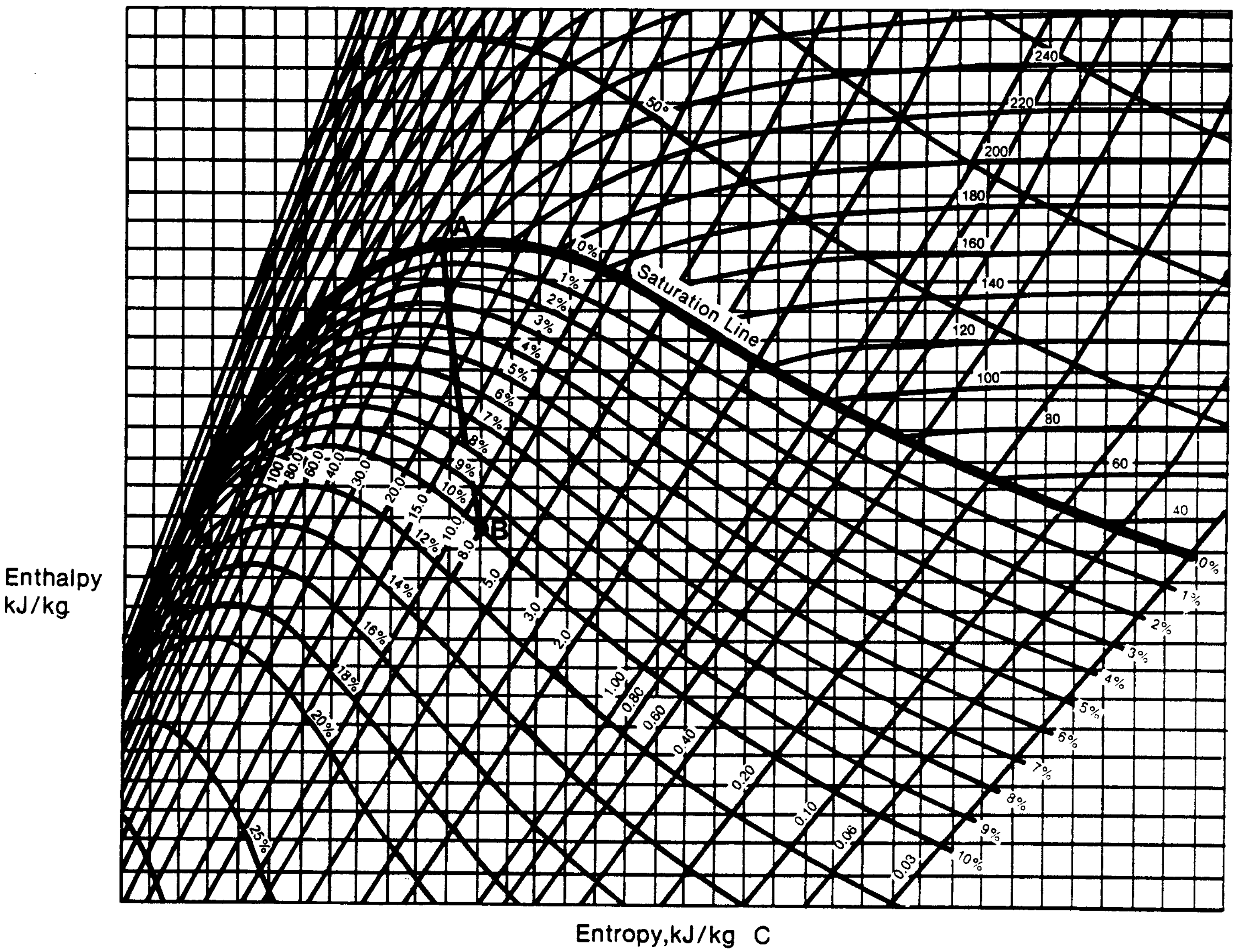


Figure 6.8

Why is the steam taken out of the turbine at this point? The reason lies in the moisture content. At moisture levels above about 10%, erosion caused by the impingement of the liquid droplets on moving parts becomes unacceptable. The steam must have the moisture removed before it can be used to produce more power. This is accomplished in the moisture separator. Liquid is physically removed from vapor in the separator. The steam at the moisture separator outlet is essentially saturated at about 170°C and 800 kPa(a). The separation process is shown (from B to C) on Figure 6.9.

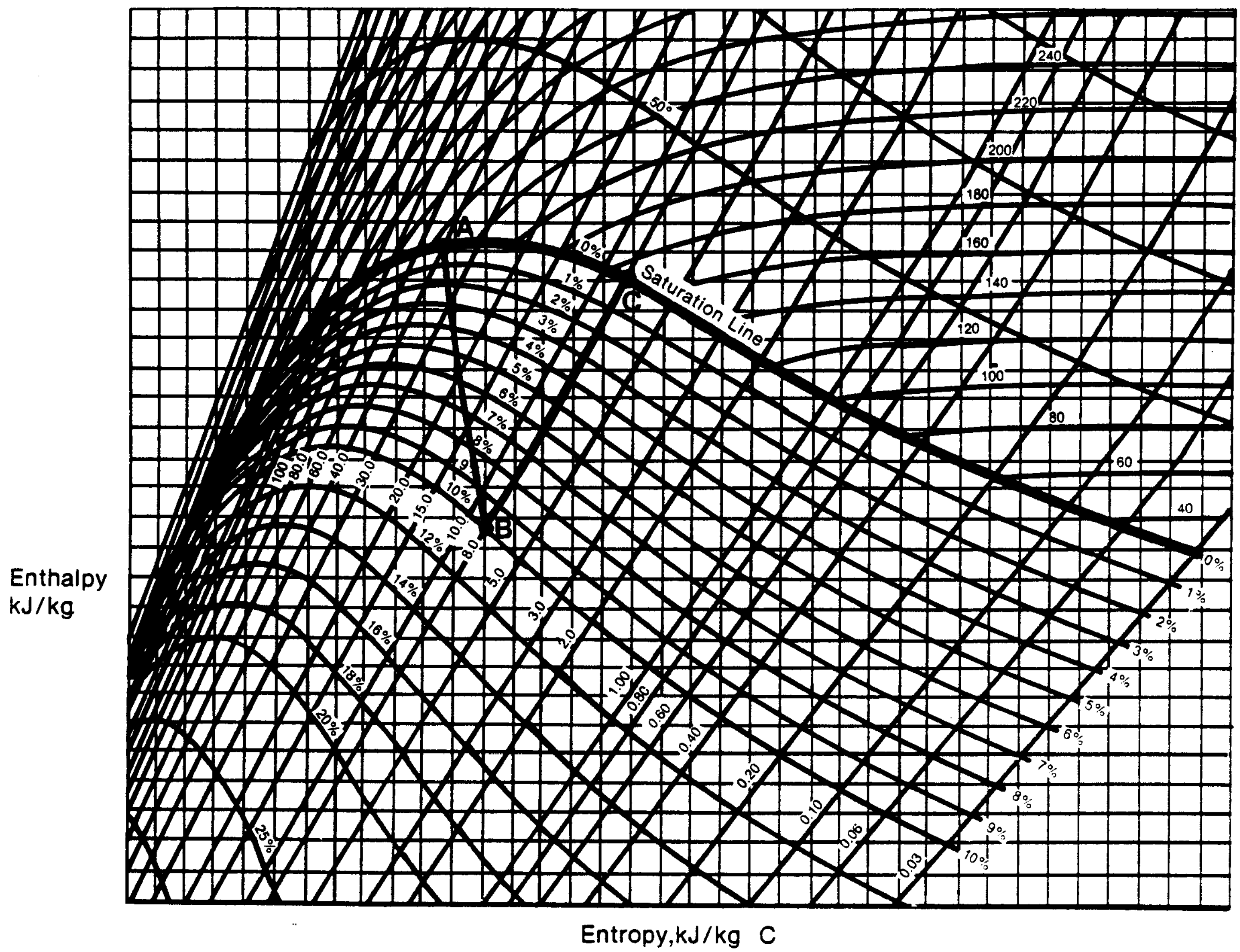


Figure 6.9

Note that the steam enthalpy has increased. How can this be true? Has any heat been added to the steam to increase its enthalpy? The answer is no, no heat has been added. The enthalpy has gone up because the moisture has been removed. Remember that enthalpy is heat content per kg of water above 0°C. The enthalpy of wet steam is a weighted average of saturated liquid enthalpy and saturated steam enthalpy. If we remove the liquid from the wet steam, the heat content per kg of the fluid that remains will be higher. However, there will be fewer kilograms of fluid after separation than before. Thus, the total heat contained in the steam leaving the moisture separator is certainly smaller than that in the steam entering, since some heat is contained in the water separated by the moisture separator.

→ Answer 6.2 before you proceed. Check your answer with the one in the "TEXT ANSWERS".

6.2) Explain how moisture separation increases the steam enthalpy of the steam exiting the HP turbine.

The steam could now enter the LP turbine and generate more mechanical energy. The problem with this is that condensation would start immediately, and before the steam could reach the design exit conditions (i.e. 30°C and 4 kPa(a)) its moisture content would be completely unacceptable. In fact, if you look at Figure 6.10 and follow the dotted line from C (moisture separator exit conditions) to D' (30°C and 4 kPa(a)), you will see that the moisture content will be about 15%. This value, however, would be much smaller than the moisture content of the steam at the LP turbine exhaust if no moisture separator were used so that steam could flow from the HP turbine directly to the LP turbine. Such a process is shown on Figure 6.10 as Line B-D" and you can see that the moisture content would be about 21%.

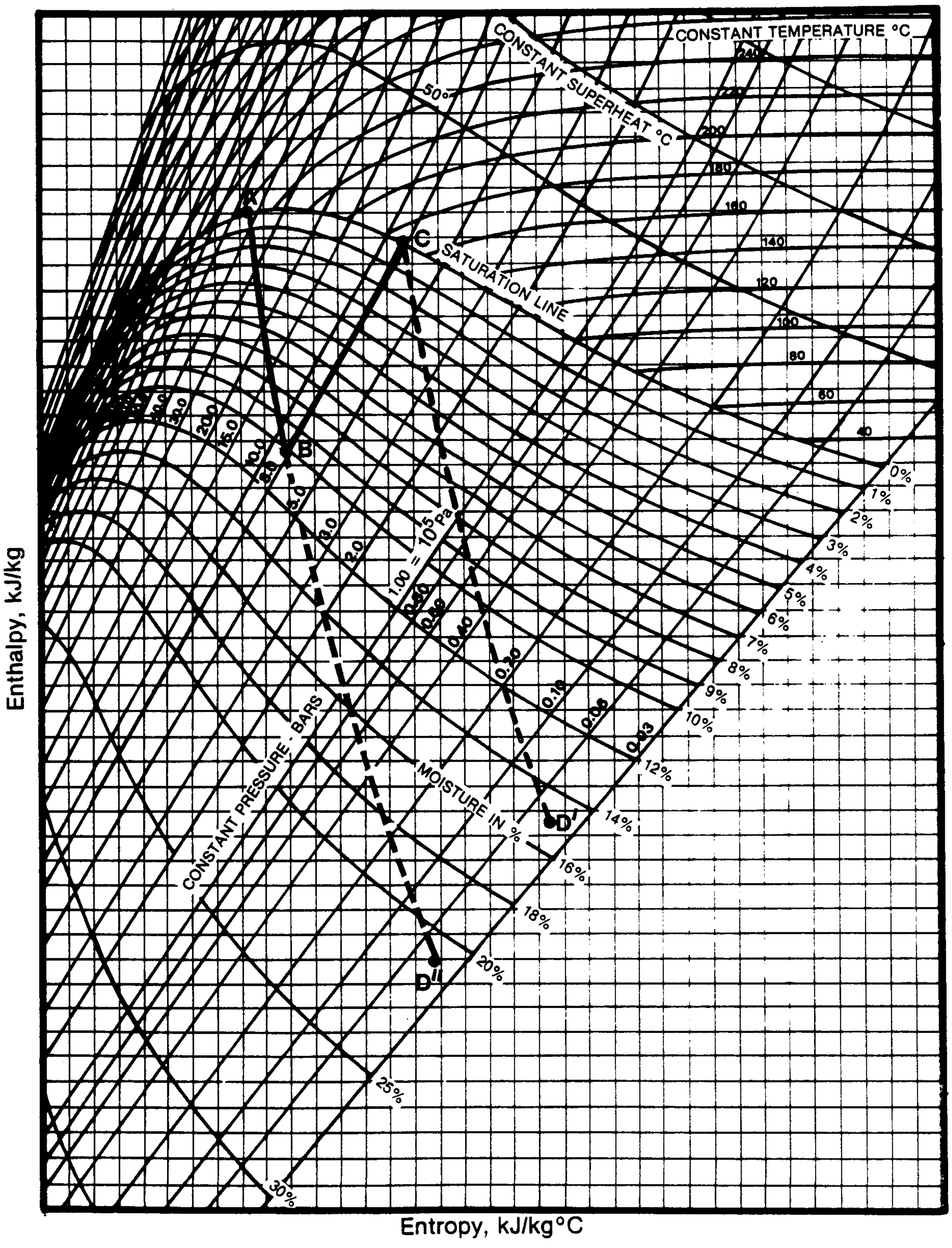

CONSTANT TEMPERATURE °C
CONSTANT SUPERHEAT °C
SATURATION LINE
CONSTANT PRESSURE - BARS
MOISTURE IN %
Enthalpy, kJ/kg
Entropy, kJ/kg°C
A
B
C
D'
D''
1.00 = 10^5 Pa
0%
1%
2%
3%
4%
5%
6%
7%
8%
9%
10%
12%
14%
16%
18%
20%
25%
30%

Figure 6.10

The steam leaving the moisture separator is then put through a shell and tube heat exchanger called the reheater in order to avoid moisture problems. The reheater uses live steam (taken from the main steam line before the HP turbine) to heat the steam coming from the moisture separator. Since this steam enters the reheater saturated at 170°C, as it is heated it becomes superheated steam. The exit temperature of the steam is about 235°C and the pressure is about 800 kPa(a). The reheating process is shown in Figure 6.11 from C to D.

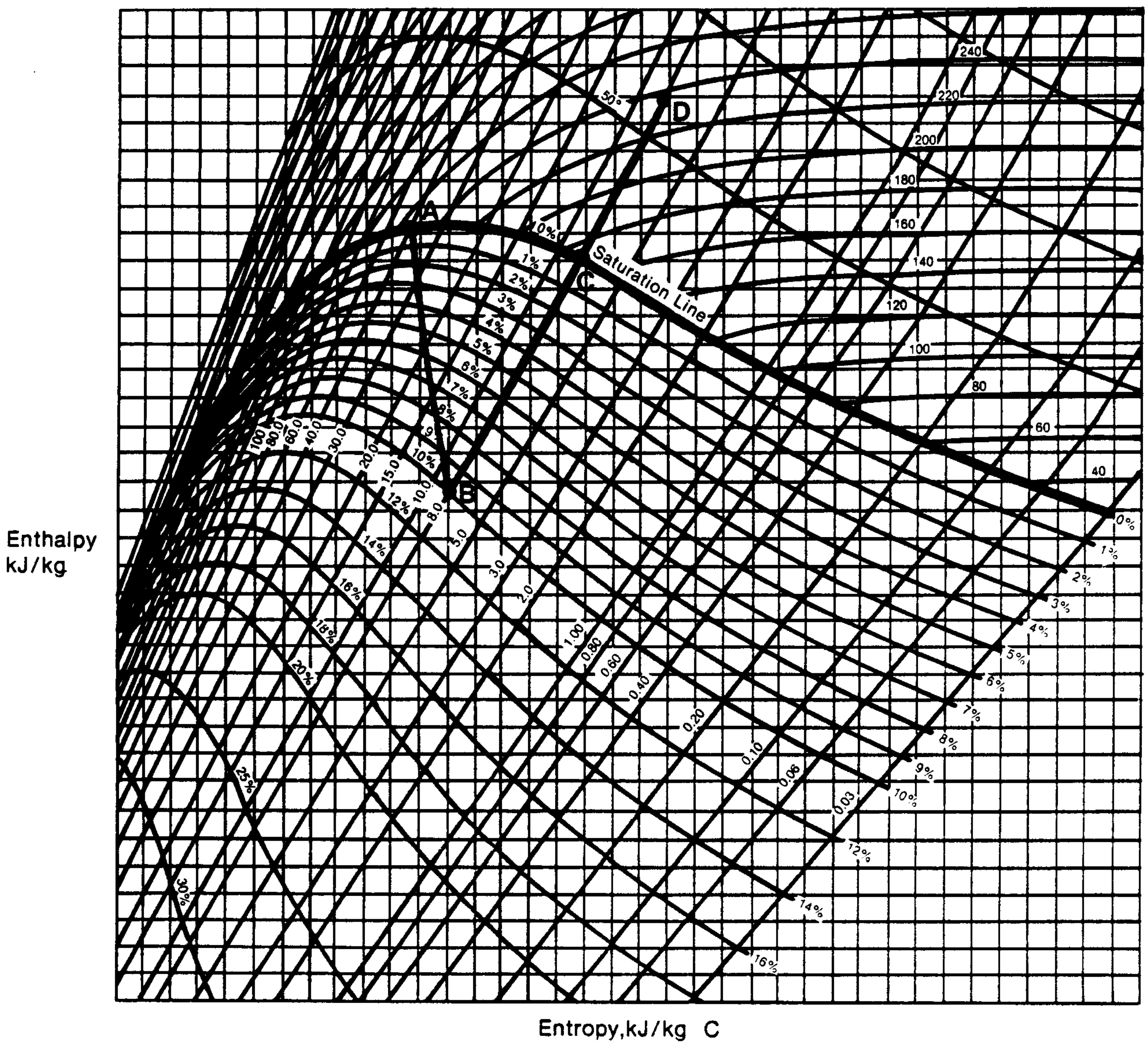


Figure 6.11

→ Answer question 6.3 in the space provided, then check your answer with the "TEXT ANSWERS".

6.3) Explain how reheat increases the enthalpy of the steam coming from the moisture separator.

The steam now enters the low pressure (LP) turbine. It expands, cools, and condenses, generating shaft mechanical energy as it flows. At the LP turbine exit the steam is typically about 10% wet at 30°C and 4 kPa(a). This part of the process is shown in Figure 6.12 from D to E.

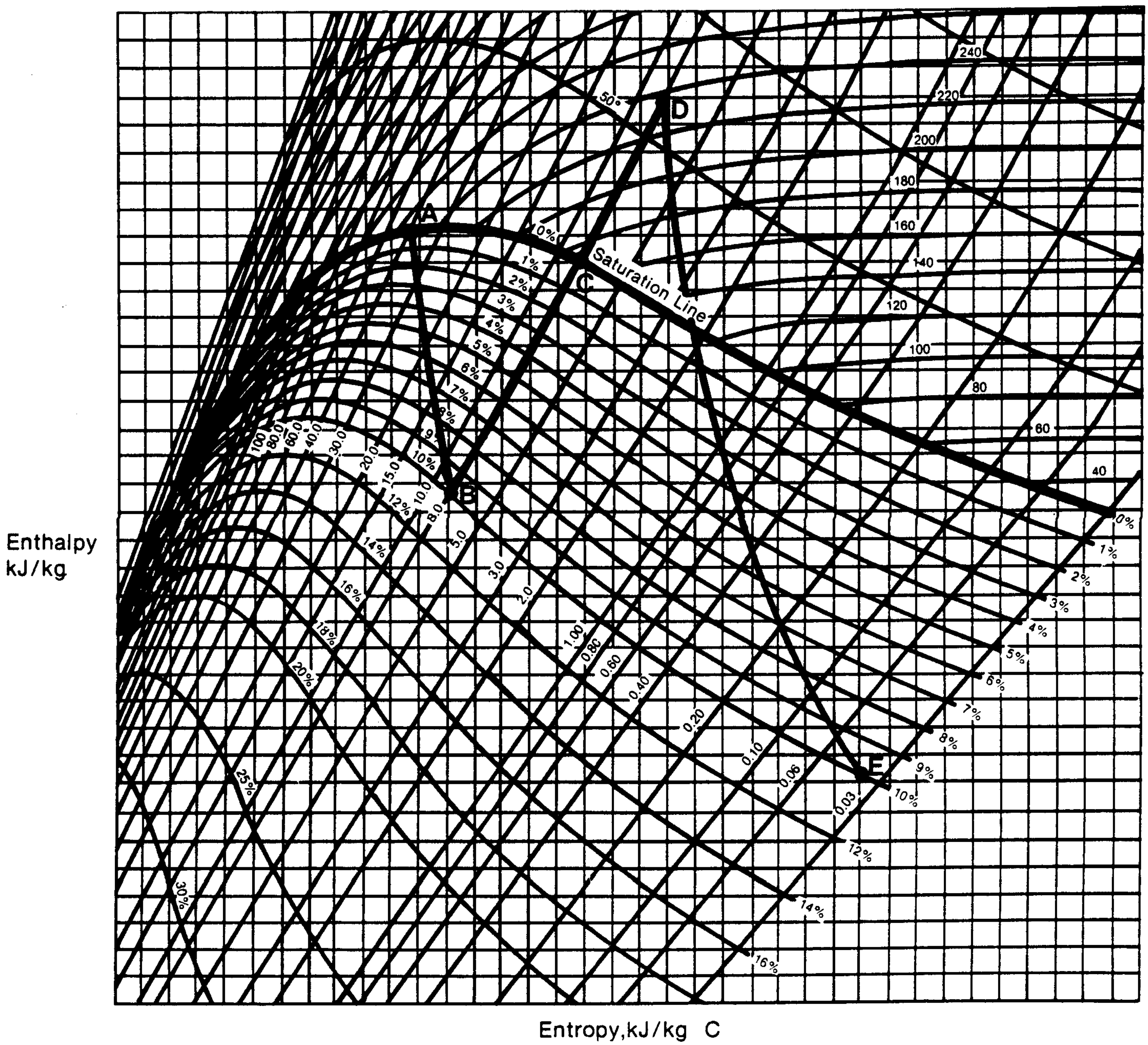


Figure 6.12

→ Answer the following question in the space provided, then check your answer with the one in the "TEXT ANSWERS".

6.4) Sketch a Mollier diagram from memory. On it, show the overall turbine process, including:

(a) high pressure turbine
(b) moisture separator
(c) reheater
(d) low pressure turbine

6.5 On the Mollier diagram you have already sketched, show how the moisture separator and reheater reduce the moisture content of the steam at the LP turbine outlet.

Throttling

This is a process where a compressible fluid expands from one pressure to a lower pressure but no mechanical work is done. Although, throttling occurs to a certain extent in any pipeline (especially if it is long), partially open valves are one place where the process is most noticeable.

When throttling takes place, the enthalpy of the fluid remains constant. This is true because:

(a) no mechanical work is done by the fluid,
(b) there is no heat loss because the flow occurs at high speed and there is no time for heat to pass through the valve casing or pipe walls. Often they are lagged with thermal insulation which makes heat transfer even more difficult.

Since the enthalpy of a throttled fluid does not change, throttling can be shown on a Mollier diagram as a horizontal line. Figure 6.13 shows throttling of wet steam. As you can see, the steam pressure and temperature are reduced. The moisture content of the steam is reduced as well and, if throttling is sufficiently large, superheated steam can be produced.

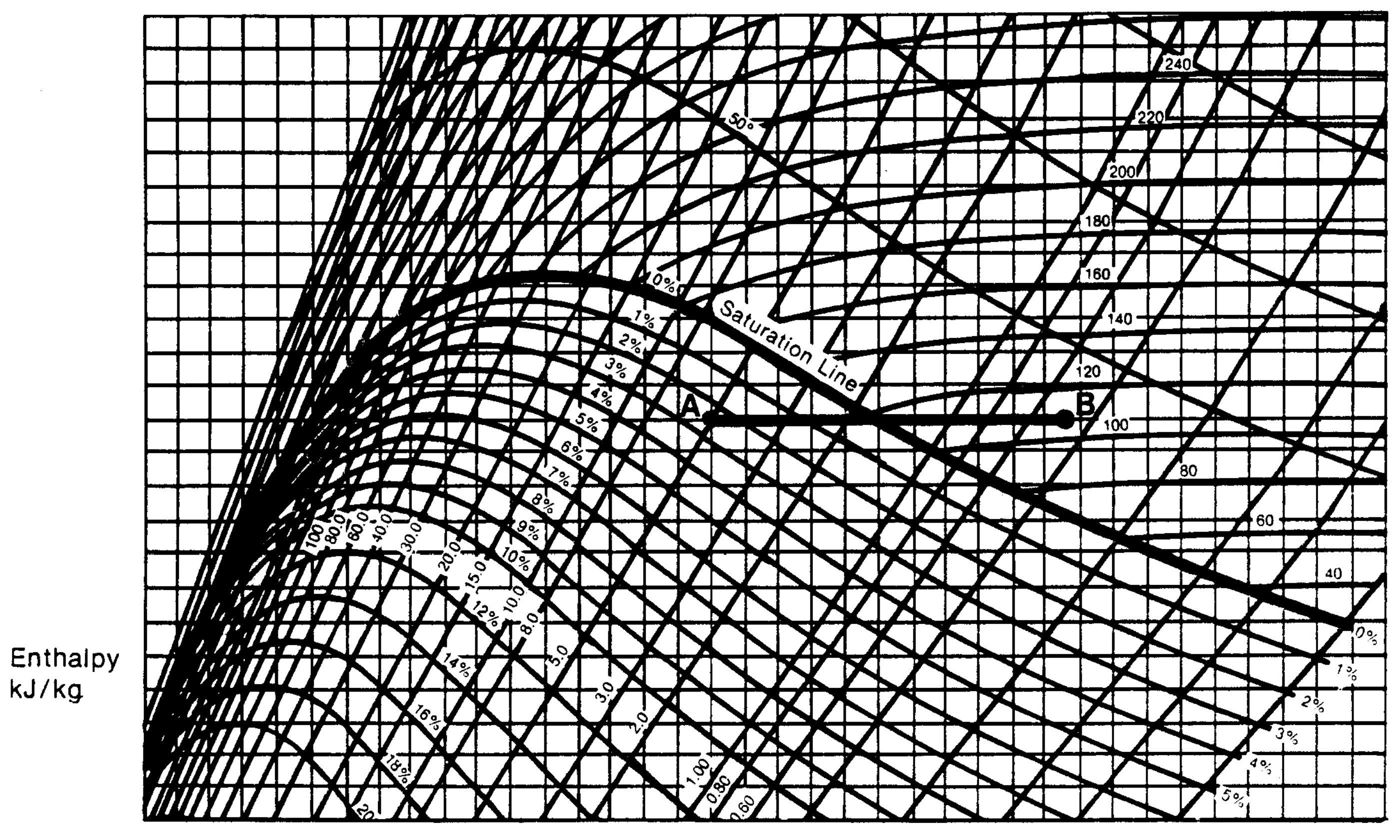


Figure 6.13

In CANDU stations, a typical place where throttling occurs is in the governor steam valves located at the HP turbine inlet. These valves are used to control steam flow to the turbine and they are partially closed at partial load conditions. The smaller the percentage opening, the greater the throttling effect. Less steam at a lower pressure and temperature gets into the turbine and thus its power is reduced.

Throttling of steam by the governor steam valves is illustrated on Figure 6.14. As you can see, the temperature and pressure of the steam drop and it becomes superheated.

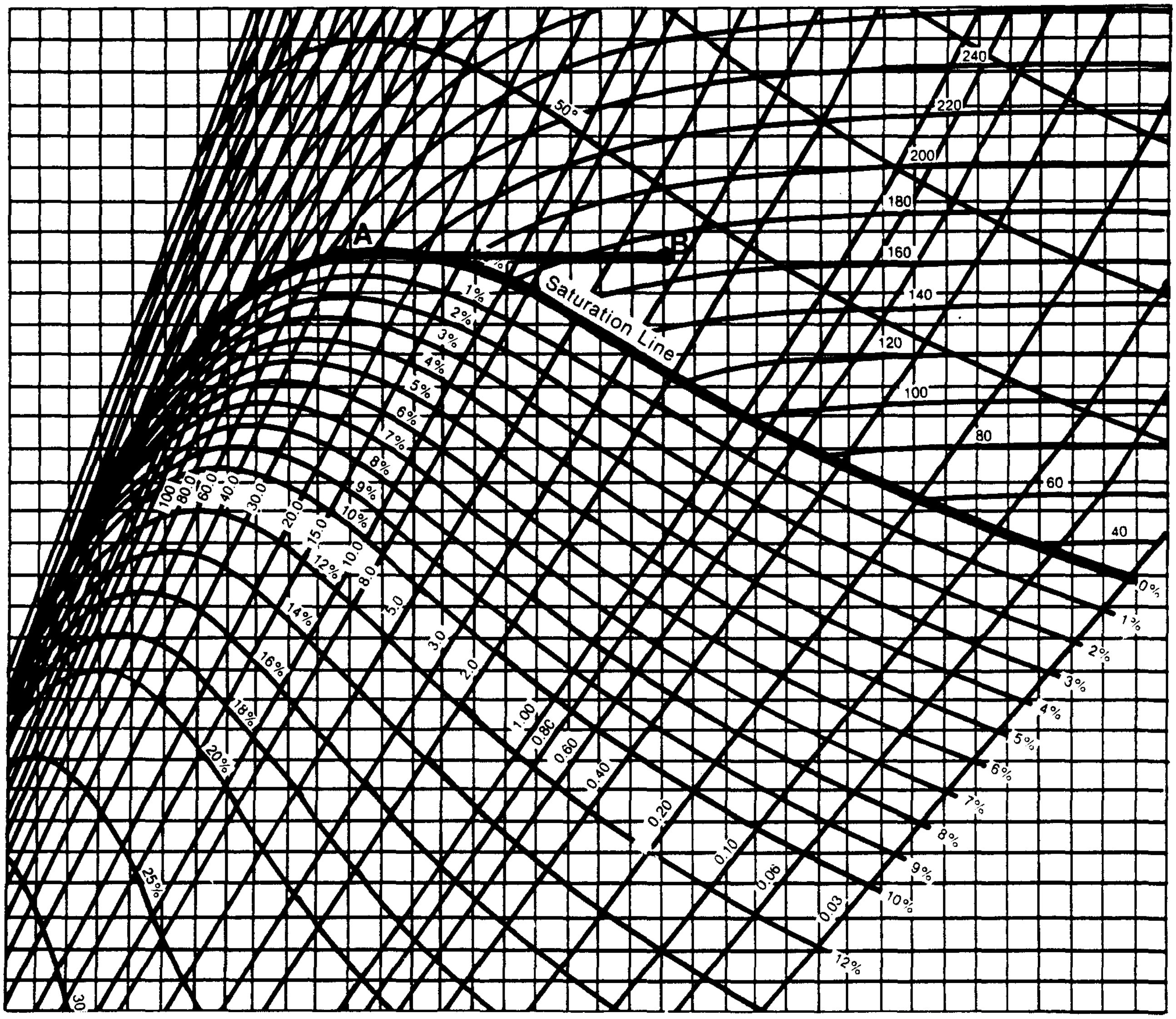


Figure 6.14

This looks like a convenient and simple method of producing superheated steam and thus avoiding, or at least reducing, the problems caused by moisture in the steam. So, why is this method not used in our CANDU stations?

To answer the question, recall what entrogy is and note that during throttling the entropy of the fluid increases. This means that less work is available from the heat contained in the steam. This is shown on Figure 6.15 which illustrates the simplest possible case of an ideal turbine (no friction losses) without a moisture separator and reheater. Line A-B illustrates the turbine process in the case when the governor steam valves are fully open so that throttling of the steam is negligible. Line C-D shows the turbine process when the valves are partially open so that the steam is throttled *before* entering the turbine. In the latter case, much less heat energy can be extracted from the steam and converted into mechanical work. This reduction in work is the reason why throttling is not used to produce superheated steam at the turbine inlet.

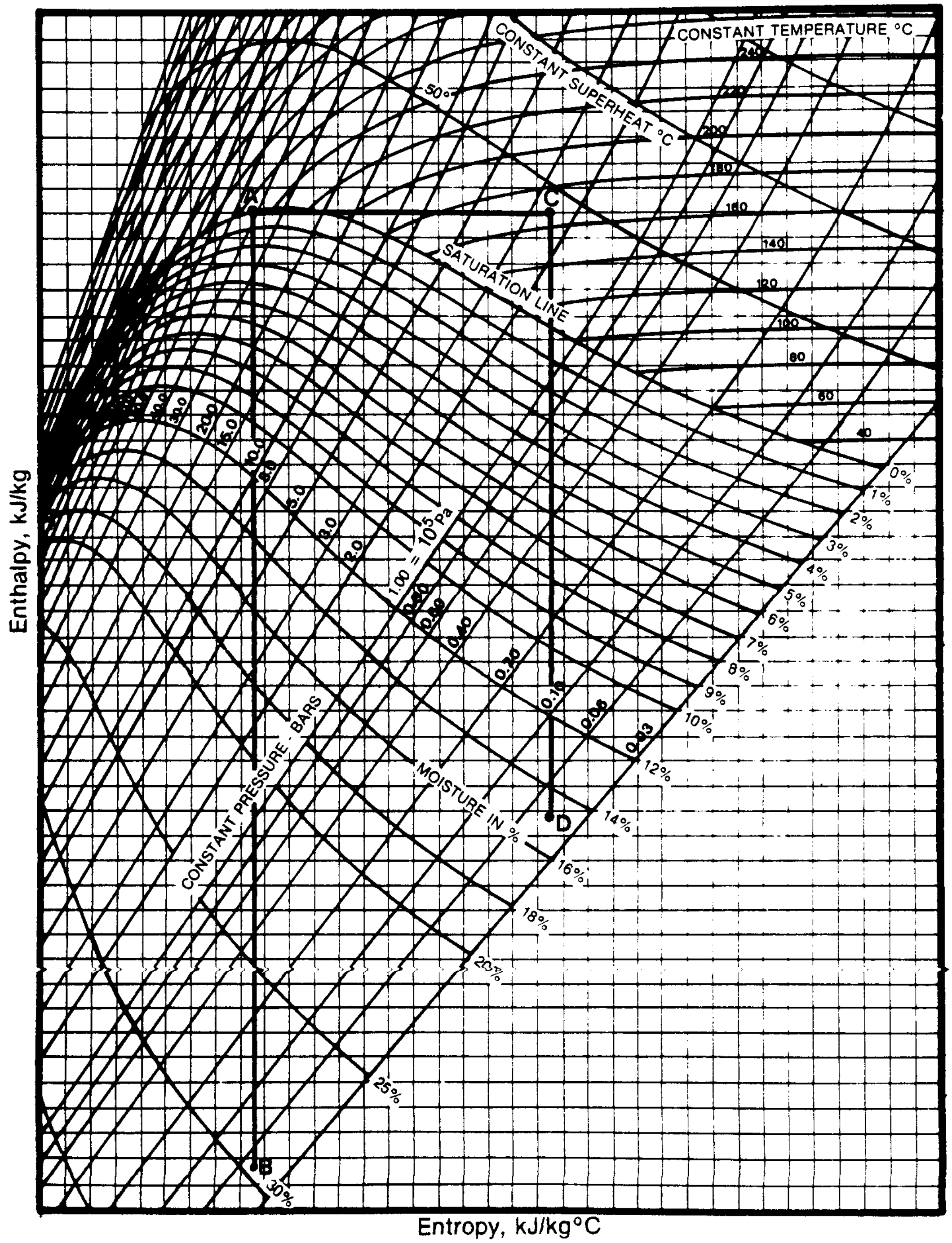

CONSTANT TEMPERATURE °C
CONSTANT SUPERHEAT °C
50°
240
220
200
180
160
140
120
100
80
60
40
A
C
SATURATION LINE
0%
1%
2%
3%
4%
5%
6%
7%
8%
9%
10%
12%
14%
16%
18%
20%
25%
30%
10.0
8.0
5.0
3.0
2.0
1.00 = 10^5 Pa
0.80
0.60
0.40
0.20
0.10
0.06
0.03
CONSTANT PRESSURE, BARS
MOISTURE IN %
D
B
Enthalpy, kJ/kg
Entropy, kJ/kg°C

Figure 6.15

6.6 Define throttling and, using a Mollier diagram, explain how throttling of the steam supplied to the turbine affects its:

(a) pressure
(b) temperature
(c) moisture content

6.7 Using a Mollier diagram, explain how throttling of the steam supplied to the turbine affects the amount of heat which can be converted into mechanical work by the turbine.

→ You have now completed module 6. If you are confident you can answer the objectives, obtain a criterion test and answer it. If you feel you need more practice, consult with the course manager.

PI 25-6 TEXT ANSWERS

6.1) Your diagram should have the same shape and labels as Figure 6.16:

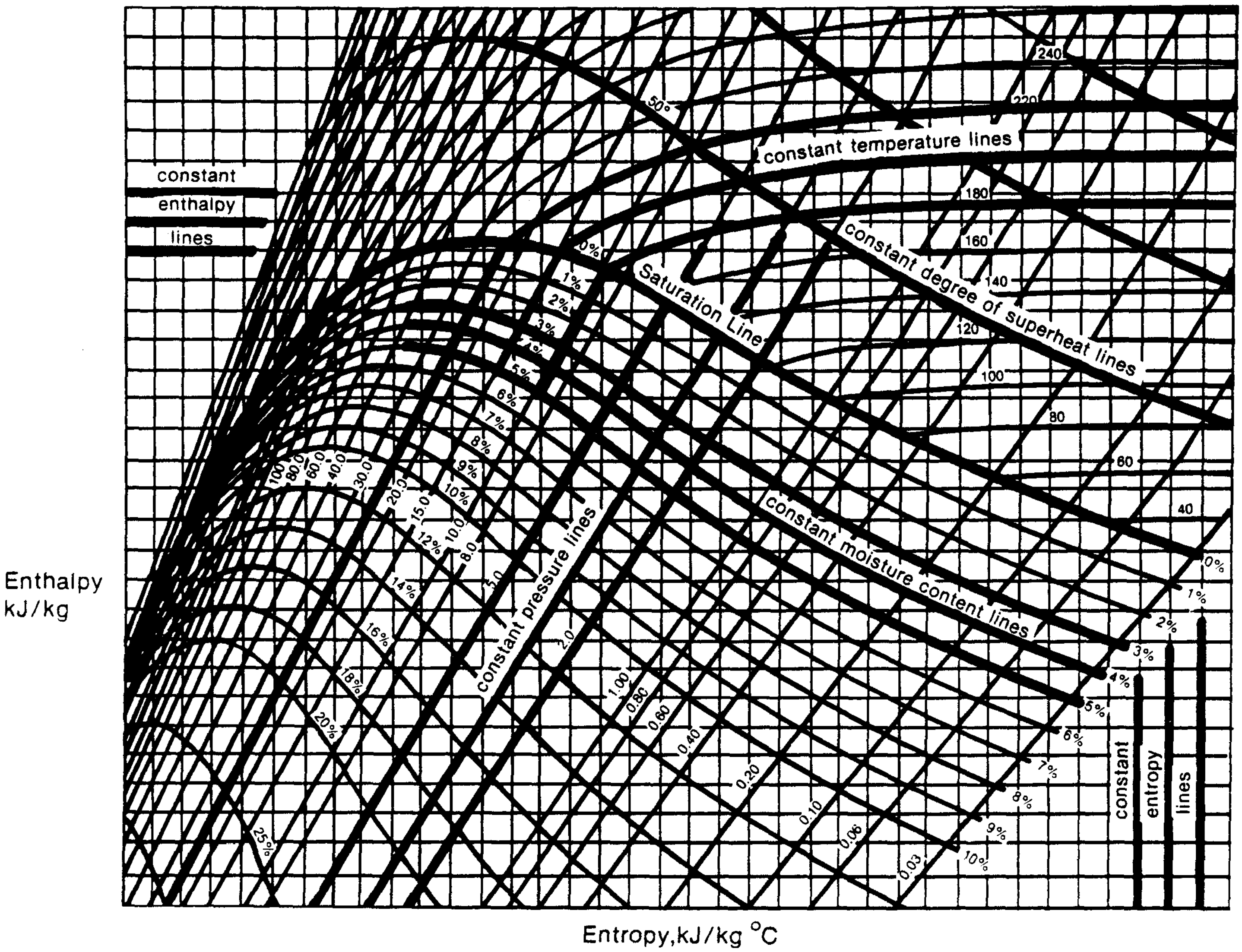


Figure 6.16

6.2) Moisture separation increases the enthalpy of the steam coming from the hp turbine by removing the liquid portion of the wet steam. The fluid that remains after the separation is essentially saturated steam, and the heat content per kg of fluid remaining will be higher. The mass flow rate of fluid, however, drops by the amount of moisture removed. Thus the enthalpy of the steam has increased, while the amount of flow has decreased.

6.3) Reheating increases the enthalpy of the steam coming from the moisture separator using live steam (at 250°C) from the main steam line before the hp turbine. The live steam condenses in the reheater, giving up enough heat to heat the saturated steam at 170°C and 800 kPa(a) to produce superheated steam at about 235°C and 800 kPa(a).

6.4) Your answer should look like Figure 6.17.

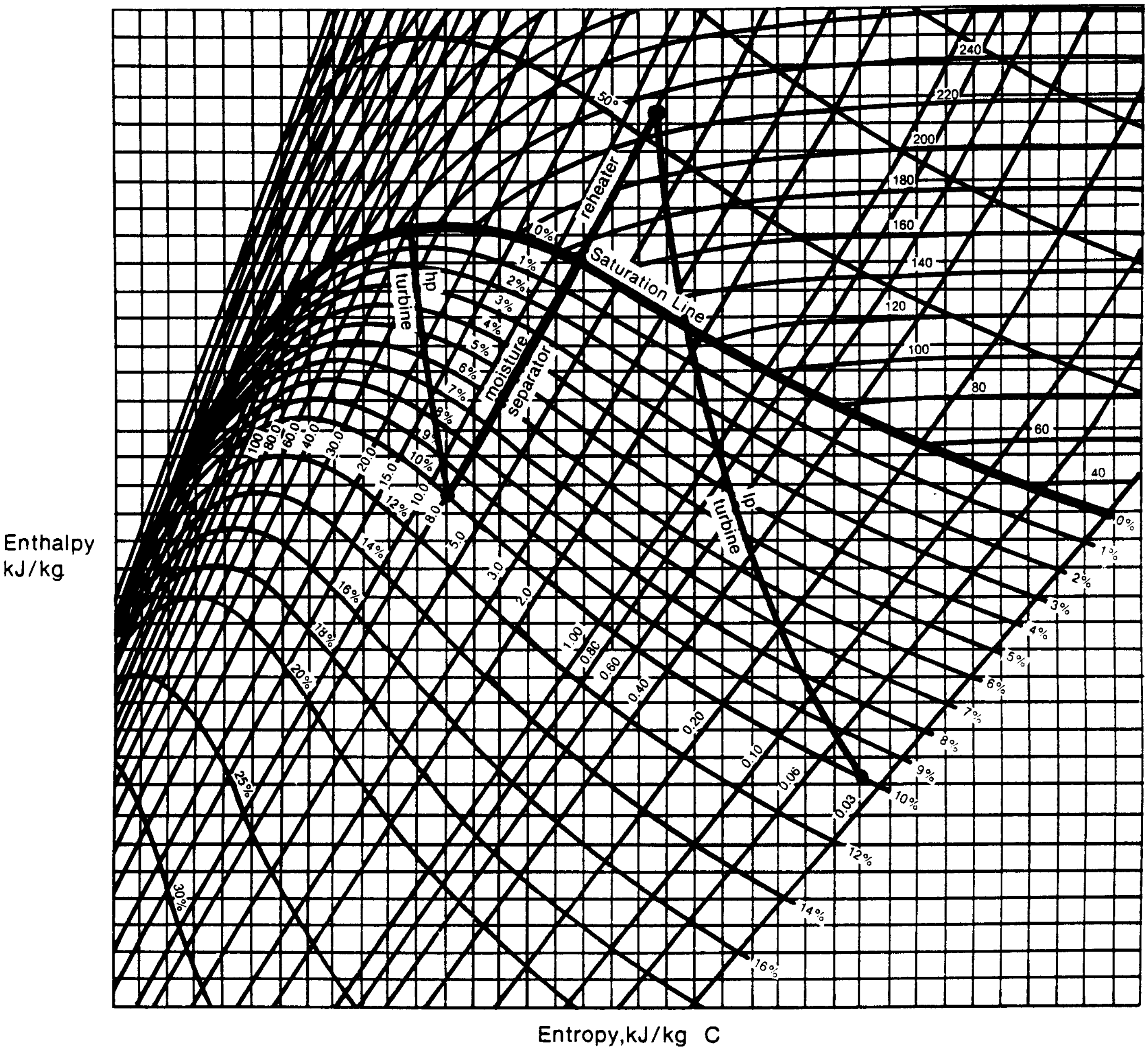


Figure 6.17

6.5 Your answer should look like Figure 6.18:

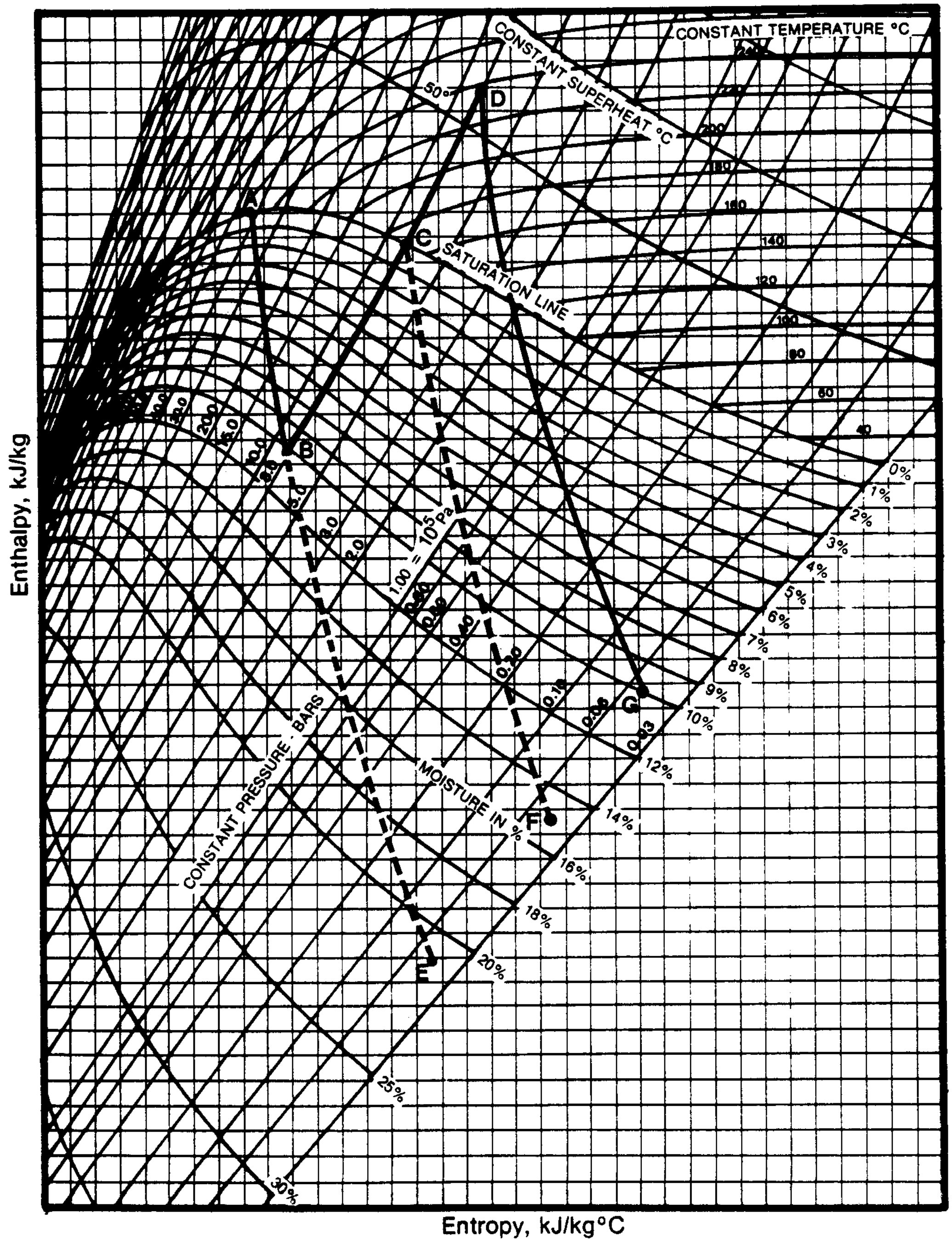


Figure 6.18

As the diagram shows, using a moisture separator allows the moisture content of the steam at the LP turbine exhaust to be reduced from about 21% to about 15%. Using a moisture separator followed by a reheater allows a further reduction of the moisture content to about 10%.

6.6 Throttling is a process which occurs when a compressible fluid expands from one pressure to a lower pressure, and no mechanical work is done. During this process the enthaply of the fluid remains constant.

Throttling of the steam supplied to the turbine is shown on Figure 6.19.

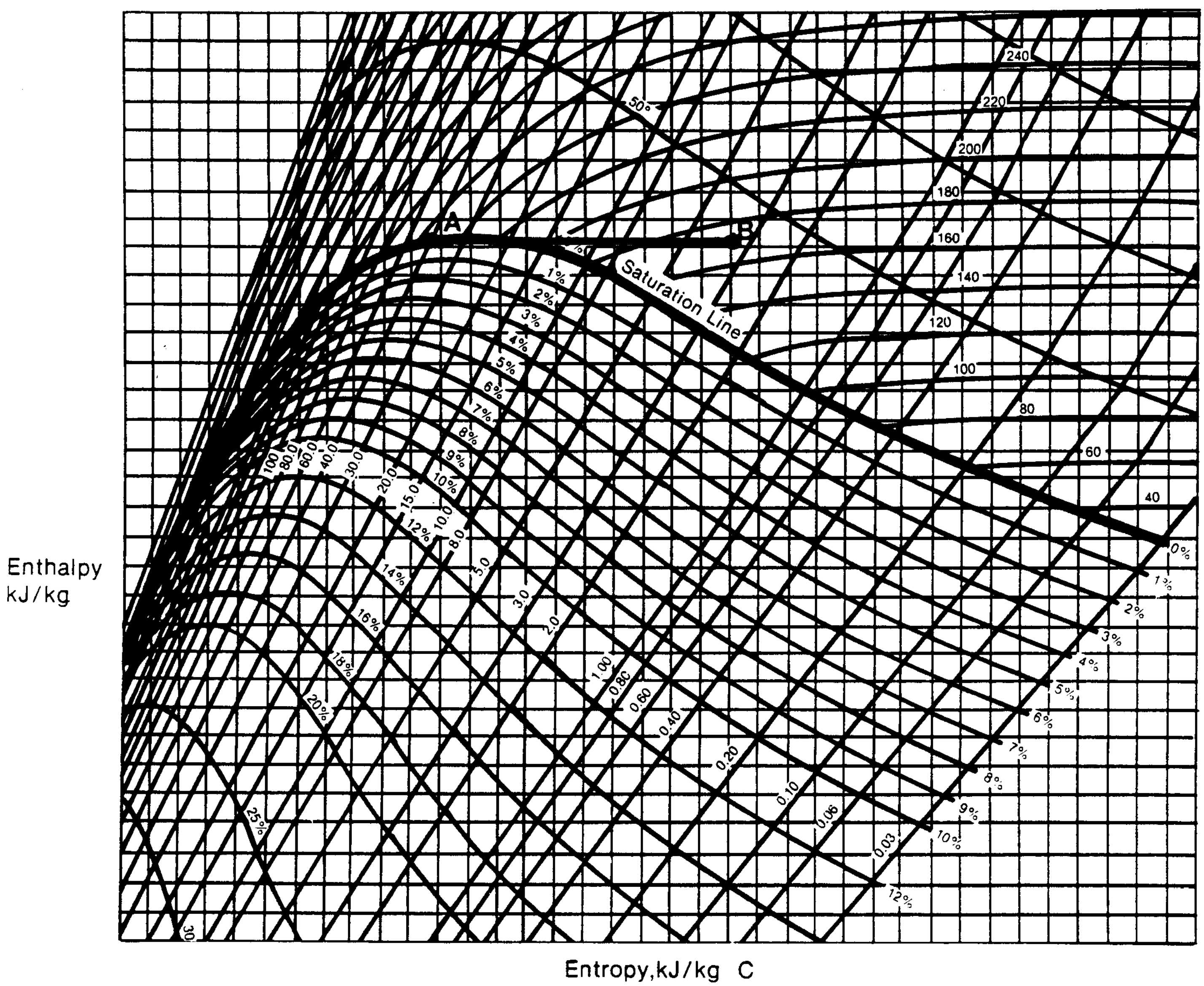


Figure 6.19

As the diagram shows, both the pressure and temperature of the steam are reduced, and it becomes superheated (its moisture content is certainly zero).

6.7 Your answer should look like Figure 6.20

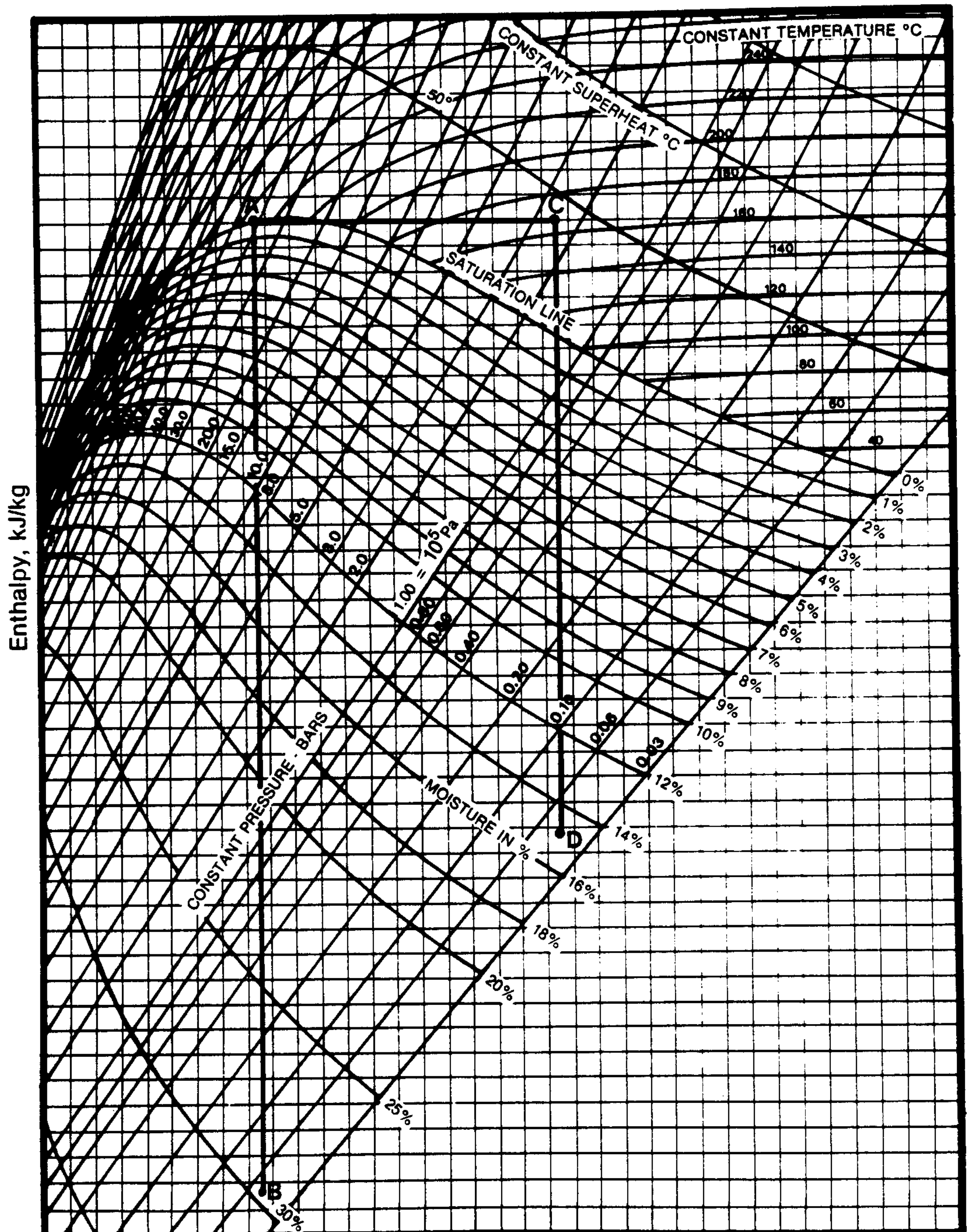
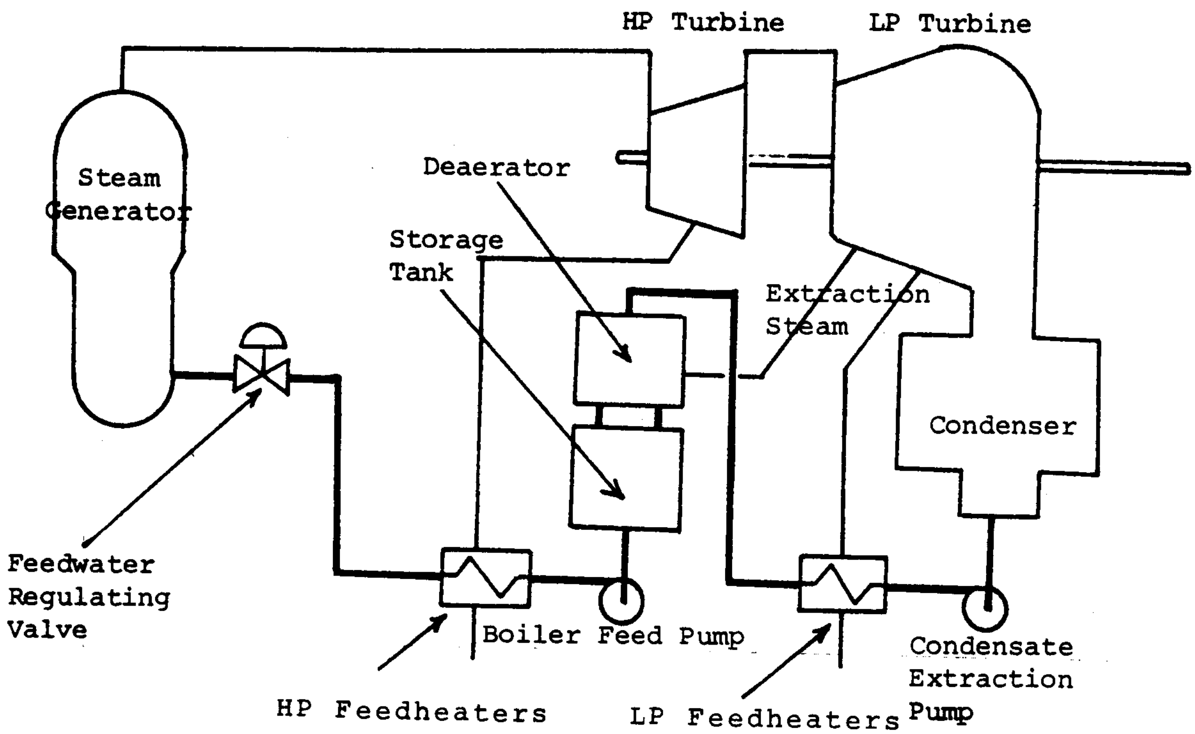


Figure 6.20

As the diagram shows, throttling of the steam supplied to the turbine reduces the amount of heat which can be converted into mechanical work by the turbine ($h_A - h_B > h_C - h_D$).

Turbine, Generator & Auxiliaries - Course 334

FEEDWATER HEATING SYSTEM - I



The Feedwater Heating System

Figure 8.1

The water which is collected in the hotwell is at a temperature of 33°C and an absolute pressure of only 5 kilopascals. Since the steam generators are at a pressure of roughly 4 megapascals, the pressure of this water must be greatly increased as it is pumped from the hotwell to the steam generator. In addition, the feedwater must be heated to approximately 175°C prior to entry to the steam generator for two reasons:

1. this greatly increases the efficiency of the steam/feedwater cycle, and

2. it lessens the thermal shock of the relatively cold feedwater entering the 250°C steam generator.

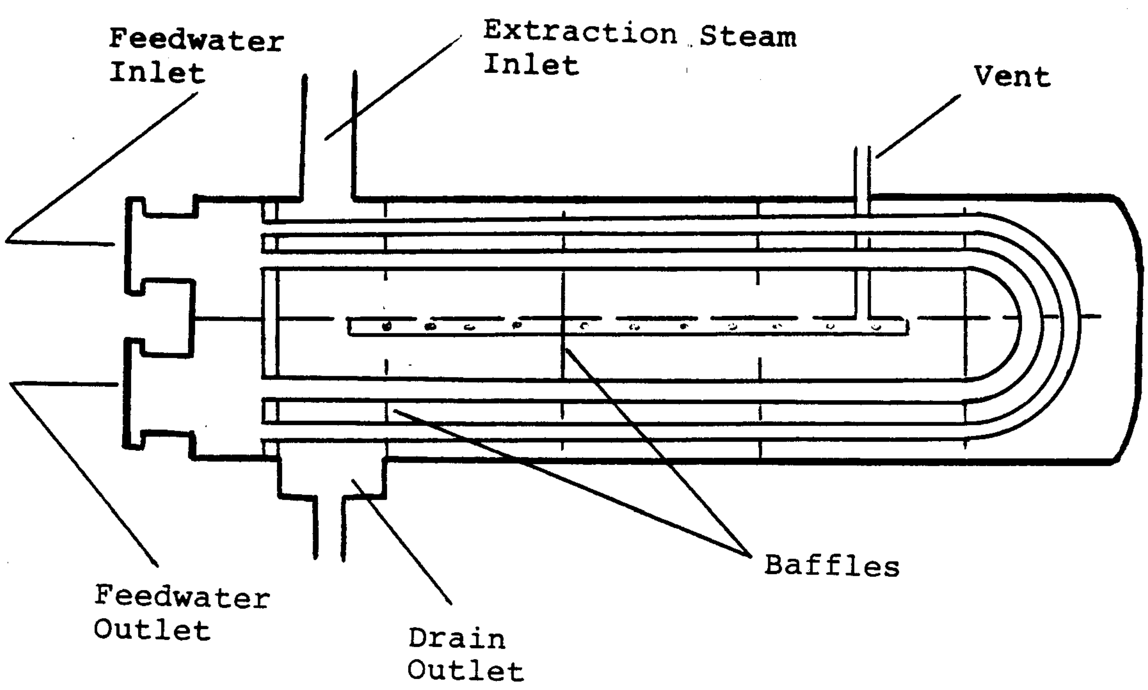
In order to increase the temperature and pressure of the feedwater, the feedheating system has two sets of pumps and a number of feedheaters. The feedheating system is generally divided into three parts:

1. a low pressure feedheating system,
2. a deaerator and storage tank, and
3. a high pressure feedheating system.

Low Pressure Feedheating System

The water which leaves the condenser hotwell is sent to a condensate extraction pump. This pump raises the absolute pressure of the feedwater to approximately 700 to 1400 kilopascals. The low pressure feedheating system gets its name from the fact that this feedwater is still at low pressure compared to the 4000 kilopascals in the steam generator.

The steam which is used to heat the feedwater in the low pressure feedheaters is extracted from the turbine (typically the low pressure turbine) and is known as extraction steam. The extraction steam condenses in the shell of the feedheater and the water is sent back to the condenser.



Feedheater

Figure 8.2

Figure 8.2 shows a typical low pressure feedheater of the type used in most large generating stations. Feedwater flows through the tubes. This type of feedheater is known as a double pass feedheater since the feedwater passes through the heater twice in going from the inlet to the outlet. Extraction steam enters the shell of the feedheater and is condensed as it passes over the outside of the tubes.The baffles in the shell of the feedheater not only give support to the tubes but also ensure the steam flows over the entire tube heating surface. The air vent on the heater shell is connected to the condenser to remove air and other non-condensible gases from the shell of the heater. If the air was not removed, it would form an air blanket around the tubes and reduce the heat transfer.

The feedwater is typically heated in more than one low pressure feedheater and leaves the last low pressure feedheater at approximately 80°C to 100°C.

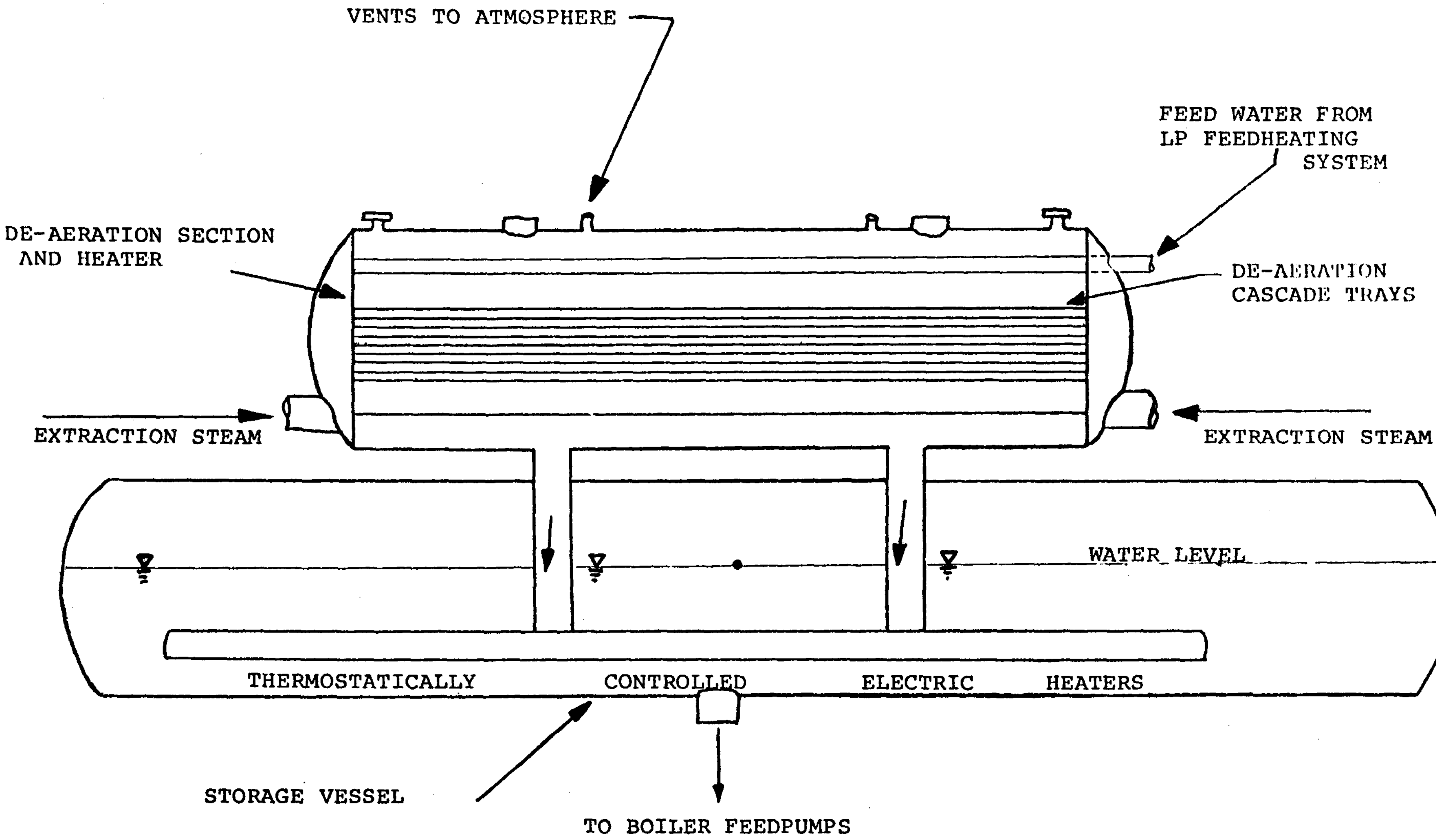
The Deaerator and Storage Tank

The deaerator, as the name implies, removes air from the feedwater. If this were not done, the corrosion rate of the metals in the high pressure feedheating system and steam generator would be significantly increased. In addition to removing air, the deaerator acts as a feedheater.

Figure 8.3 shows a typical deaerator and its associated storage tank. Extraction steam from the low pressure turbine enters the deaerator near the bottom and passes upward.The incoming feedwater enters the deaerator near the top and is sprayed downward over cascade trays. The steam condenses and in doing so, heats the feedwater. As the feedwater is heated to near the boiling point, the air is released from the water droplets. In addition, the steam passing over the water droplets "scrubs" the air off the surface of the droplets. The air is vented to atmosphere off the top of the deaerator. While the steam to the deaerator normally comes from the low pressure turbine, there is an alternate source of steam which comes directly from the main steam line. This source is used to provide necessary feedheating during unit startup and at other times when steam is shut off to the turbine and extraction steam is, therefore, unavailable.

The feedwater and condensed steam from the deaerator drain into a storage tank. This storage tank has three basic functions:

1. to maintain a positive suction head for the boiler feed pump,



Deaerator/Storage Tank

Figure 8.3

2. to hold, together with the condenser hotwell, sufficient water in storage to maintain a full load for a period of ten minutes, and

3. during shutdown and low load periods, to maintain the water in the storage tank at 125°C with electric heaters in the storage tank. This is to prevent air absorption in the water and to minimize thermal shock to the steam generators during startup. These heaters are much too small to provide adequate heating of the feedwater at high power levels when steam provides the heating.

High Pressure Feedheating System

The boiler feed pumps take a suction on the deaerator storage tank and raise the absolute pressure to between 4 and 7 megapascals. This high pressure feedwater is then sent through high pressure feedheaters. The high pressure feedheaters are identical to the low pressure feedheaters except:

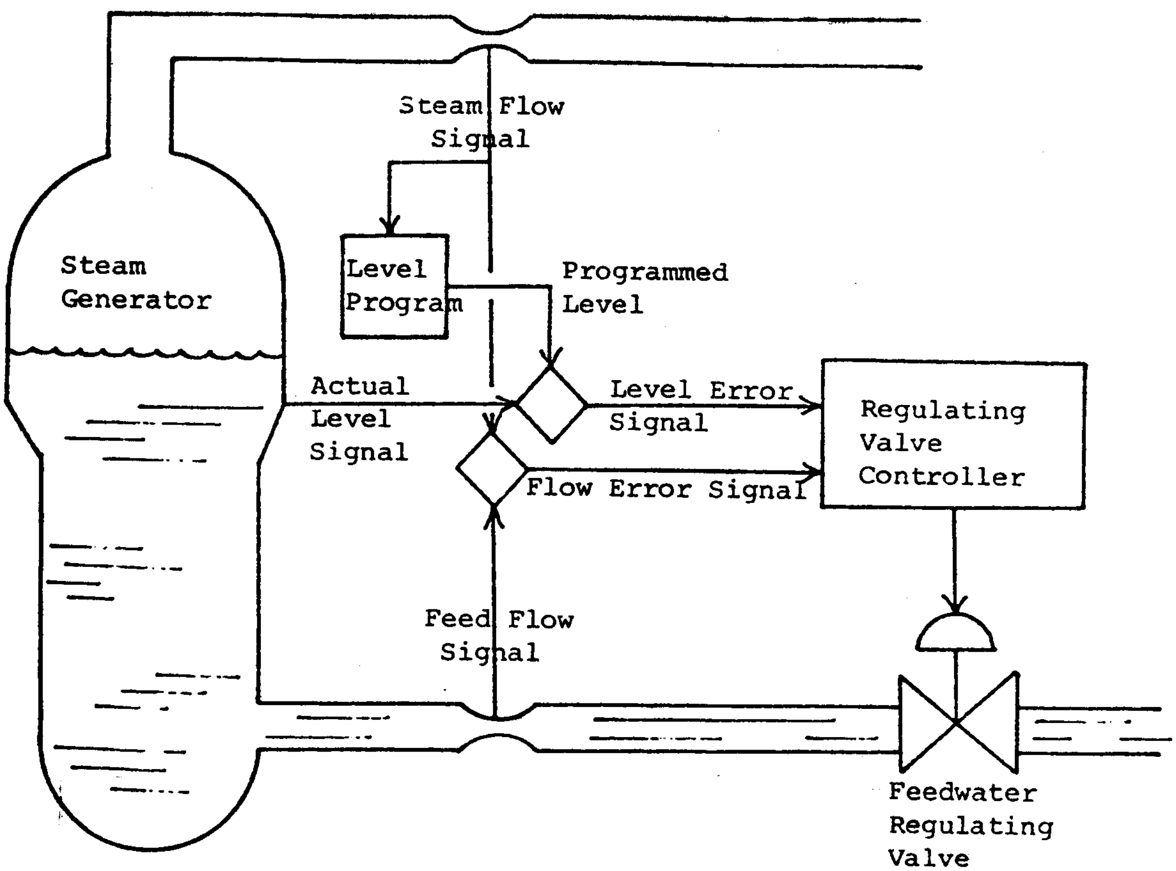
1. they are designed to operate at higher pressure and temperature,

2. the extraction steam comes from the high pressure turbine and/or moisture separator, and

3. the drains from the condensed extraction steam are pumped back to the deaerator.

Before the feedwater enters the steam generator, it passes through a feedwater regulating valve. The function of this valve is to allow sufficient feedwater to enter the steam generator to match the steam flow out and maintain the level in the steam generator. To do this, the controller for the valve compares steam flow out of the steam generator with feedwater flow into the steam generator and moves the valve to make the two equal. In addition, it compares the actual steam generator water level with a predetermined programmed level and positions the valve to make these two equal.

It is critical that water level be maintained properly.If the steam generator is either too high or too low, the cyclone separators and scrubbers in the steam generator will not operate properly. The wet steam which results could cause severe damage to the turbine blading.



Feedwater Regulating Valve Control

Figure 8.4

ASSIGNMENT

1. What is the function of the feedwater heating system?

2. What is the function of the deaerator storage tank?

3. What is the function of the low pressure feedwater heating system?

4. In a feedwater heater, why is it necessary to have a vent on the heater shell and where do the air vents go?

5. The baffles in a feedheater serve two purposes, what are they?

6. Describe how the deaerator removes dissolved gases from the feedwater.

7. How is the feedwater flow controlled to the boiler to ensure that the water level is constant?

8. What are the effects of a very high water level in the boiler?

9. Why are electric heaters fitted in the deaerator storage vessel?

10. Draw a line diagram of a feedheating system showing:

    (a) Hotwell
    (b) Condensate Extraction Pump
    (c) Feedwater Regulating Valve
    (d) Boiler Feed Pump
    (e) Deaerator
    (f) LP Feedheater
    (g) HP Feedheater
    (h) Extraction Steam Lines
    (i) Steam Generator

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 334

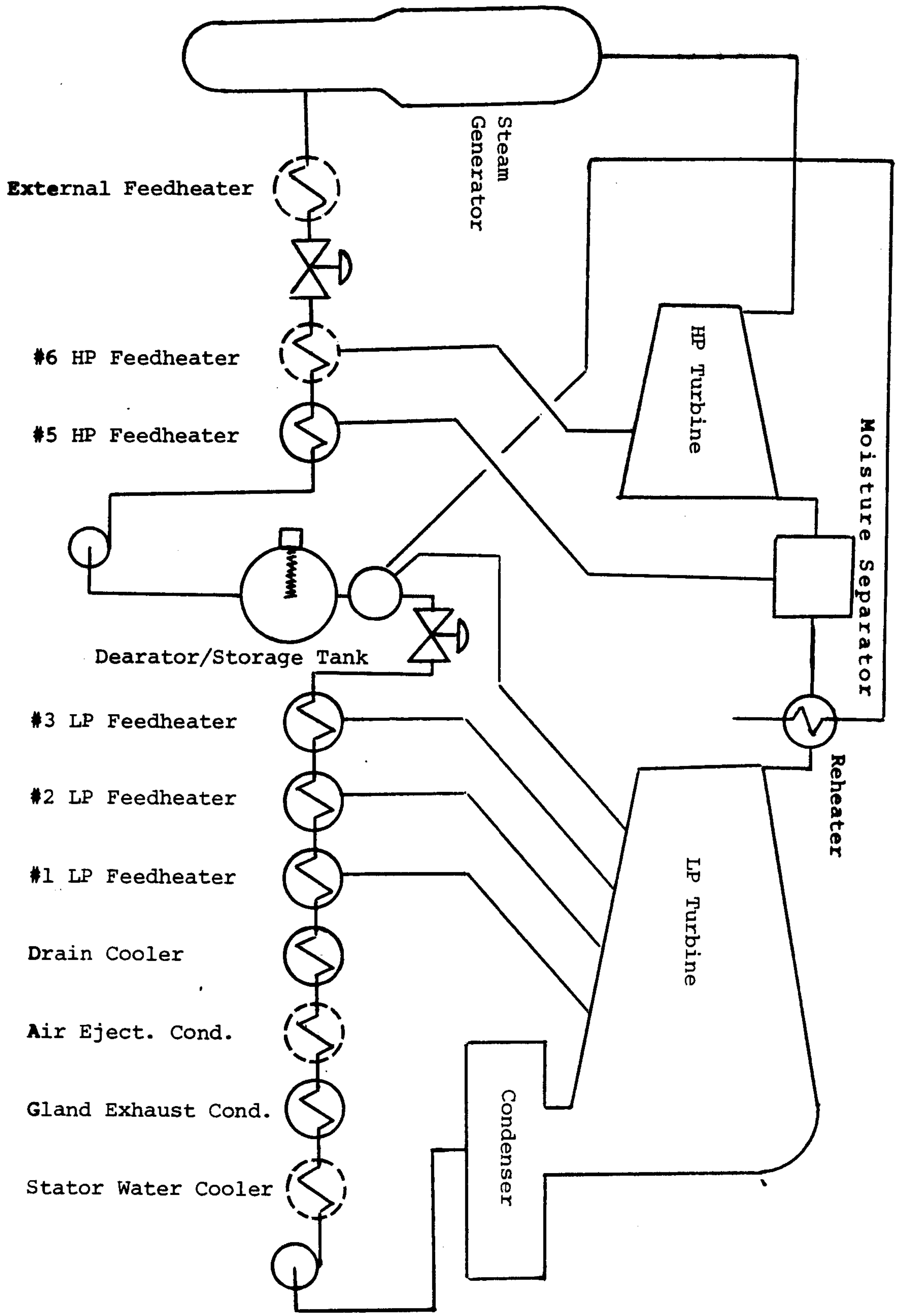
FEEDWATER HEATING SYSTEM - II

Figure 9.1 shows the feedheaters of a typical CANDU generating station. Depending on plant configuration and temperatures, the exact number of feedheaters and the sources of heat energy varies from station to station. The purpose of this lesson is to examine the sources of heat energy available for feedheating and discuss why they are used or not used in particular stations.

There are a number of sources of heat available for the feedheating system. In the general order they would appear in an <u>ideal</u> feedheating system, they are:

1. Heat from the generator hydrogen cooling system;
2. Heat from the generator stator water cooling system;
3. Heat from the steam pipe and turbine drains;
4. Exhaust steam from turbine glands;
5. Exhaust steam from steam control valve glands;
6. Exhaust steam from steam air ejectors;
7. Extraction steam from the low pressure turbine;
8. Main steam;
9. Electric heaters;
10. Drains and extraction steam from the main moisture separators;
11. Extraction steam from the high pressure turbine;
12. Reheater drains;
13. Heat transport system.

We will consider each of these heat sources and discuss where they are used in a typical feedheating system.



Typical Feedheating Chain

Figure 9.1

The Generator Hydrogen Cooling System

This system removes heat from the generator by circulating hydrogen over the rotor and through slots in the stator. Although the system rejects a reasonably large amount of heat (about 3 MW for a large unit), the temperature is quite low (35-40°C). With a condenser pressure of 4.5 to 5.0 kPa(a), the hotwell temperature is 31-33°C. This means that there is insufficient temperature difference between the generator hydrogen and the condensate to allow for efficient heat transfer.

Although generator hydrogen is being used for feedheating at various generating stations, it is not used at any NGD stations. The hydrogen is cooled in low pressure service water coolers.

The Generator Stator Water Cooling System

This system removes heat from the generator stator windings by pumping extremely pure water through the stator conductors. The stator water removes about 6 MW from the generator. The stator water temperature is about 50-60°C to the cooler and about 30-40°C from the cooler. For smaller units (DP NGS for example) this results in a high enough differential temperature for use to be made of the stator water cooling system in heating feedwater. On larger CANDU generating units, the differential temperature is too small when condensate temperature is near the upper end of its normal range. In these stations, heat is removed from the stator water cooling system in low pressure service water coolers.

Steam Pipe and Turbine Drains

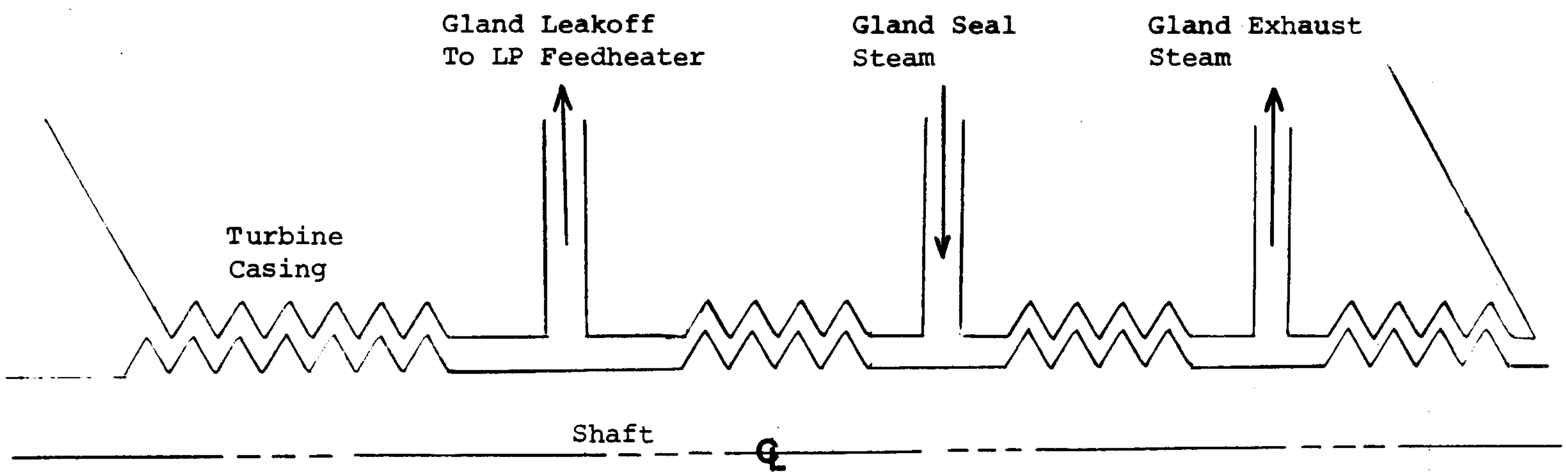
During system warmup and, to a lesser extent, during operation, condensation occurs in the steam piping and turbine. This water is removed by steam traps which allow water to escape but not steam. The normal operation of the trap is for water to collect until a certain amount accumulates. The water is then discharged from the trap.

While this water is hot enough for effective feedheating, the quantity during normal operation is relatively small and in almost all turbine units, this water is passed directly to the condenser.

Gland Exhaust Steam

The normal leakoff of steam and air from the turbine and control valve glands is exhausted to a gland exhaust condenser. The steam condenses and the air is removed by a gland exhaust fan. The gland exhaust condenser is cooled by condensate. With the exception of those station with stator water heating of feedwater, the gland exhaust condenser is normally the first heater in the feedwater system.

It was mentioned in the lesson on turbine construction that the high pressure turbine glands are self sealing at power. The outward leakage of steam is more than sufficient to seal the gland. This excess steam is supplied to a low pressure feedheater. The general arrangement of the HP turbine gland is shown in Figure 9.2.



HP Turbine Gland

Figure 9.2

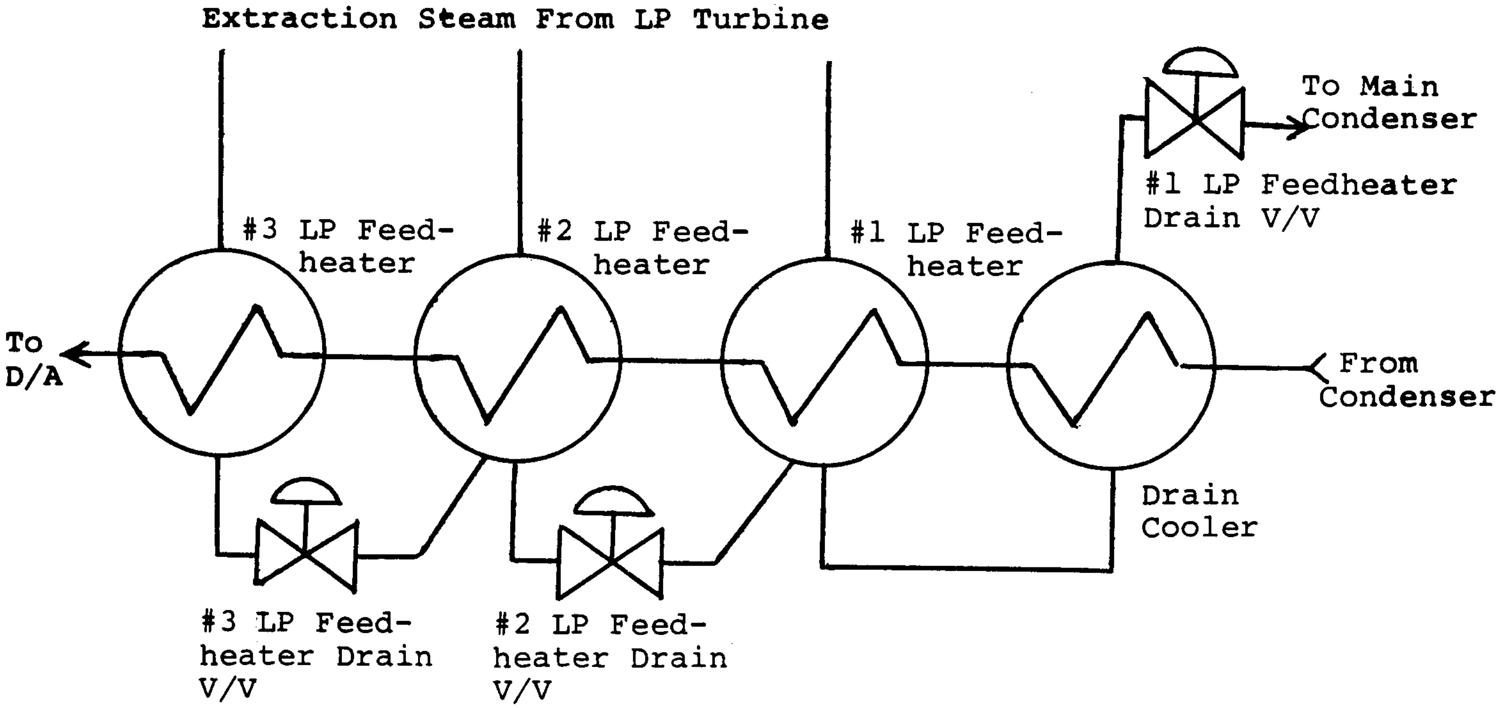
Steam Air Ejector Exhaust

The air extraction system is used to remove air and other non-condensible gases from the main condenser. On those plants with steam air ejectors, the steam exhausting from the ejectors is cooled in an intercondenser and an aftercondenser. The inter and after condensers are cooled by feedwater.

Low Pressure Turbine Extraction Steam

The majority of the feedheating in the low pressure feedheating system and deaerator is done with extraction steam. For a large CANDU generating station about 20% of the steam entering the low pressure turbine is bled off as extraction steam. Roughly 1/3 of this steam goes to the deaerator; the remaining 2/3 goes to the three LP feedheaters.

The low pressure feedheater drains are cascaded from higher to lower temperature feedheaters. The drains from the number one low pressure feedheater is passed through a separate drain cooler prior to flowing back to the condens-er. The arrangement of low pressure feedheaters is shown in Figure 9.3. The feedheating in the low pressure feedheating system is done by a combination of extraction steam from the turbine and cascaded drains from the next higher low pressure feedheater.



Cascade LP Feedheaters

Figure 9.3

The contribution of extraction steam to feedheating can be seen by the temperature rise through the low pressure feedheating system. The feedheating prior to the drain cooler raises the feedwater temperature from 29°C to about 31°C. The drain cooler and three low pressure feedheaters raise the temperature from 31°C to about 120°C.

The low pressure extraction steam to the deaerator accomplishes two functions:

1. heat the feedwater by direct contact with the incoming feedwater, and

2. deaerate the incoming feedwater.

The extraction steam raises the temperature of the incoming feedwater to the range of 125-150°C depending on the individual unit design. This temperature corresponds to the boiling temperature at the deaerator pressure of 250-450 kPa(a).

The deaeration of feedwater must take place above atmospheric pressure [101.3 kPa(a)] to prevent air re-entering the system after deaeration. On the other hand, the deaerator cannot be operated at too high a pressure since the cost and size of a large, high pressure vessel would be prohibitive.

Since the deaerator is a direct contact feedheater, it is more efficient than closed feedheaters where the steam and feedwater do not mix. The closed feedheaters, however, do not require a separate pump after each feedheater.

Main Steam

By far the majority of feedheating prior to the steam generator (or more precisely prior to the preheater) is done with extraction steam. During normal operation, roughly 99% of the feedheating comes from turbine extraction steam. When the turbine extraction steam is at low temperature (startup) or unavailable (turbine trip, poison prevent), there is essentially no normal feedheating capability. However, to prevent excessive thermal shock to the steam generators, the feedwater must be heated to within about 125°C of boiler temperature. This necessary feedheating is accomplished by taking main steam from the boiler outlet and dumping it through a control valve into the deaerator.

The amount of main steam required is considerably less during startup (when feedwater flow is relatively low) than during poison prevent (when feedwater flow is roughly 70% of full power). For this reason, there are normally two control valves: a large one for poison prevent, and a smaller one for start up.

As the turbine becomes available to provide extraction steam to the deaerator, the supply of main steam is automatically shut off.

Electric Heaters

In large turbine units thermostatically controlled electric heaters are used in the deaerator storage tank to maintain the temperature during off load and low load periods. The heaters are also used to warm up the deaerator prior to startup if the deaerator has been cooled off for maintenance.

It is desirable to keep the deaerator warm during shutdown periods to minimize the amount of air entering the system. In addition, when shutdown or at low power levels, the supply of hot water in the deaerator can be used for adding feedwater to the steam generator. The electric heaters are of relatively small capacity and can do no appreciable feedheating when the turbine is at power. As an example of this, it requires about 36 hours for the heaters to bring the 240,000 kg of water in a deaerator from room temperature to 125°C. At 100% power, this mass of water passes through a typical deaerator every seven minutes.

Main Moisture Separator Extraction Steam and Drains

The drains from the main moisture separator are sent to the high pressure feedheater immediately after the deaerator. Since the quantity and heat content of these drains are insufficient to provide the required amount of feedheating, they are supplemented by extraction steam from the main moisture separator.

High Pressure Turbine Extraction Steam

The feedwater enters the preheater at about 170-175°C. If the crossover between the HP and LP turbine is at a high enough pressure, the high pressure feedheater using moisture separator extraction steam will be the last feedheater. If the crossover pressure is lower, there will be a second high pressure feedheater which uses extraction steam from the HP turbine.

If there is more than one HP feedheater, the drains are cascaded in a manner similar to the LP feedheaters. However, because of the small pressure differential and large elevation difference between the first HP feedheater and deaerator, the HP feedheater drains are pumped back to the deaerator.

Reheater Drains

Having given up only the latent heat of vapourization, the reheater drains are at the same temperature as the steam generator. They are too hot to be efficiently used in a feedheater and are pumped directly back to the steam generator.

Heat Transport System

The feedwater leaves the last stage of feedheating at about 170-175°C. The heat transport system is used to convert this subcooled water into saturated steam at 250°C. Roughly 80% of the heat energy required to raise condenser water to saturated steam comes directly from the heat transport system in the preheaters and steam generators.

The function of the preheater is prevention of thermal shock by raising the temperature of the feedwater to steam generator temperature before releasing it into the steam generator shell. Whether the preheater is located internal or external to the steam generator, it is really just an extension of the steam generator. From the standpoint of the feedheating system, it makes little difference whether the preheater is internal or external to the steam generator. At Bruce NGS-A, the location of the preheater external to the steam generator was done for heat transport system considerations. It was desired to create cooler heat transport fluid for the central fuel channels. This was achieved by cooling some of the heat transport fluid with feedwater in an external preheater.

Figures 9.4 and 9.5 show the arrangement of a typical feedheating system for a large CANDU generating station. The drawing is a composite of the various stations and does not contain the exact arrangement of any one station.

ASSIGNMENT

1. List the available sources of heat for a feedheating system. For each source, list the feedsystem components in which the particular source is used.

2. The deaerator - storage tank has three sources of heat available. Under what circumstances is each used?

3. Why might some stations not use a high pressure feedheater using HP turbine extraction steam?

4. On large generating units, why is service water rather than feedwater used to cool the generator?

5. Why is the deaerator located near the center of the feedheating chain?

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 234

FEEDWATER CONTROL AND OPERATION

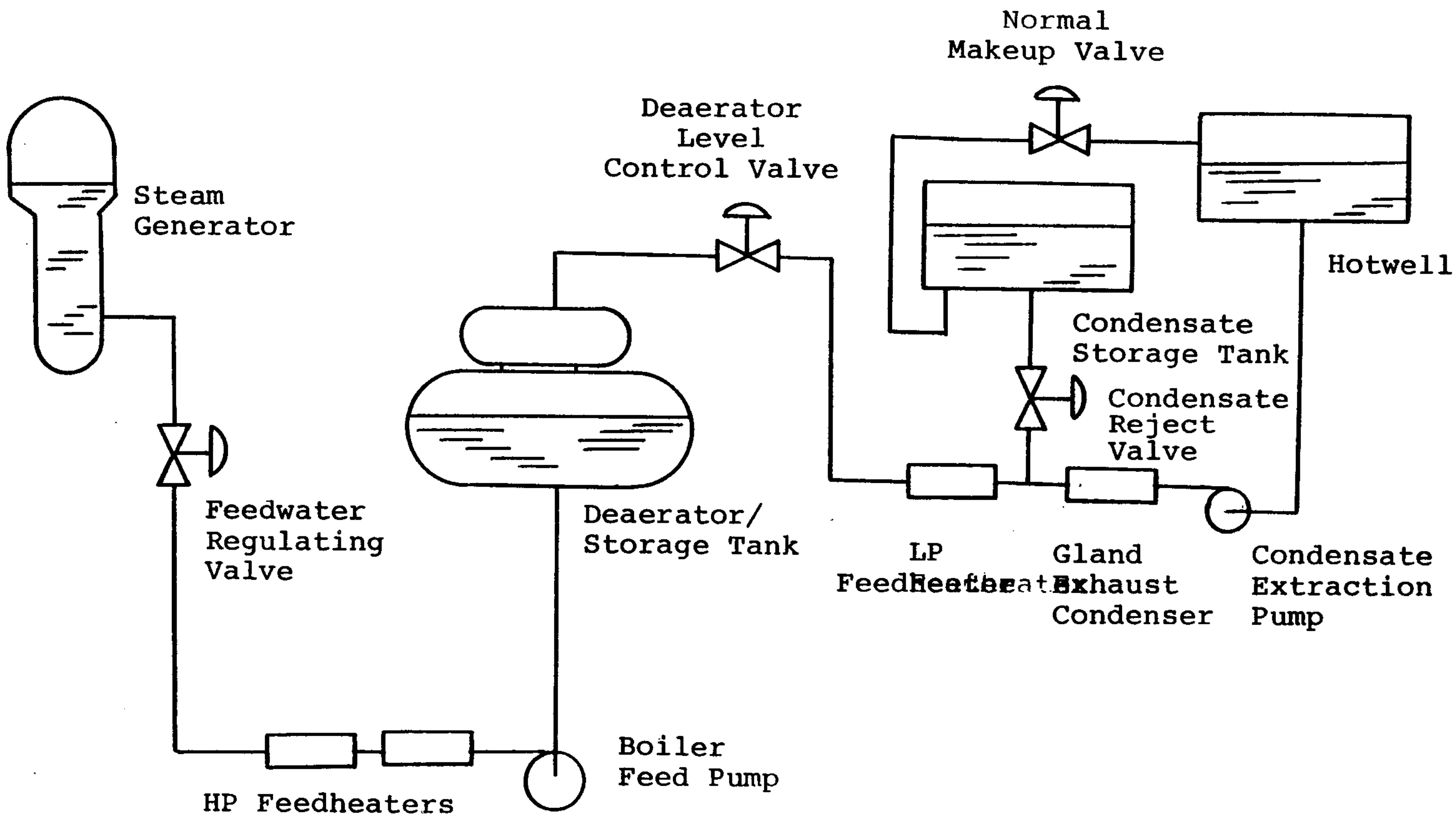
The feedwater system must be capable of delivering to the steam generator the proper quantity of heated feedwater for all power levels. Since the quantity of water stored within the system (primarily in the steam generators, deaerator and hotwells) is rather limited, there is a necessity for the system to maintain a strictly controlled water inventory and to be able to rapidly respond to changes in feedwater demand.

The water inventory in the feedwater system is maintained by controlling the level in the steam generators, deaerator storage tank and hotwells. Figure 7.1 shows the level control valves associated with the feedwater system of a large nuclear generating station. As a starting point we will discuss level control in each of the three vessels and then discuss the overall operation of the system.

Steam Generator Level Control

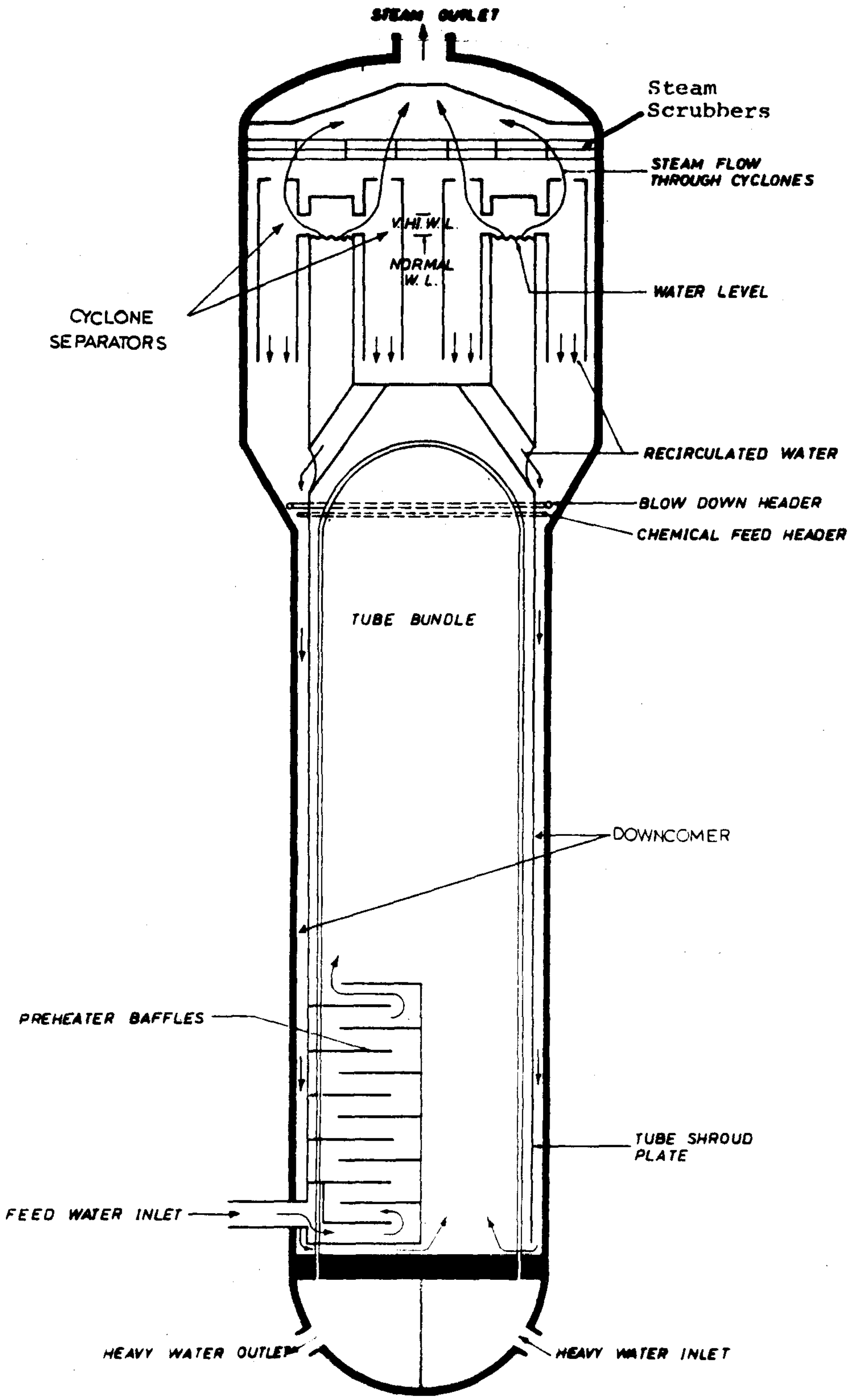
It is impossible to discuss steam generator level control without including a discussion of swell and shrink as it relates to the steam generators. Swell is a transient increase in boiler level which accompanies a rapid increase in steam flow; shrink is a transient decrease in boiler level which accompanies a rapid decrease in steam flow. If the steam flow from a steam generator (Figure 7.2) increases, the pressure in the steam generator drops. This results in flashing of water to steam in the tube bundle (riser) section of the steam generator. This increase in volume in the tube bundle forces water to back up the downcomer and, since downcomer level is what is measured as steam generator level, this results in an upswelling of boiler level which is called swell. On a down power transient, the pressure in the steam generator increases. This results in a collapsing of steam bubbles in the tube bundle. This decrease in volume in the tube bundle causes a drop in downcomer level which is called shrink.

The magnitude of swell and shrink depends on the size and rate of the power change: the greater the size and rate of change of steam flow, the greater the resultant swell or shrink. If the change is slow enough there may be no apparent swell or shrink. Figure 7.3 shows the effect of swell and shrink from a rapid power change of 50% of the maximum continuous rating for a generator with a constant level program (level which feedwater regulating valves attempt to maintain does not change).



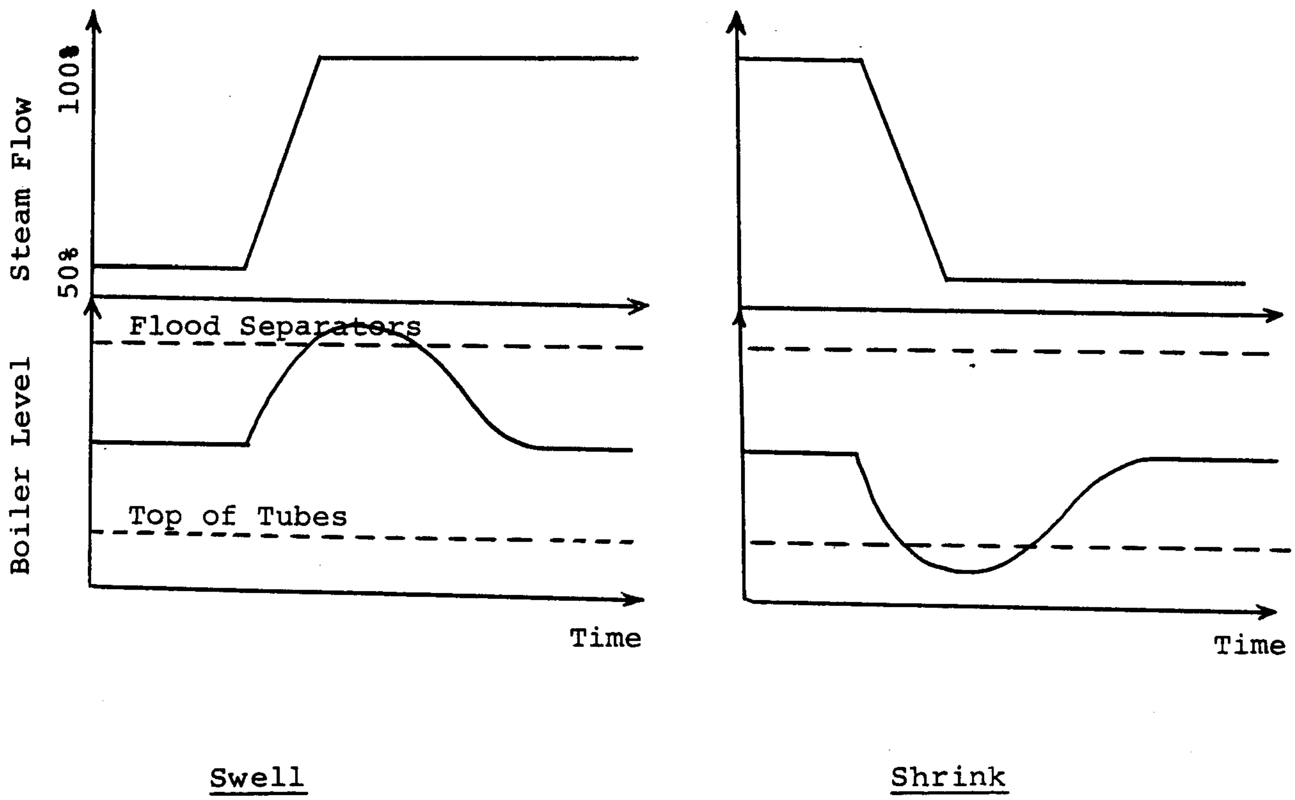
Feed System Control Valves

Figure 7.1



Steam Generator

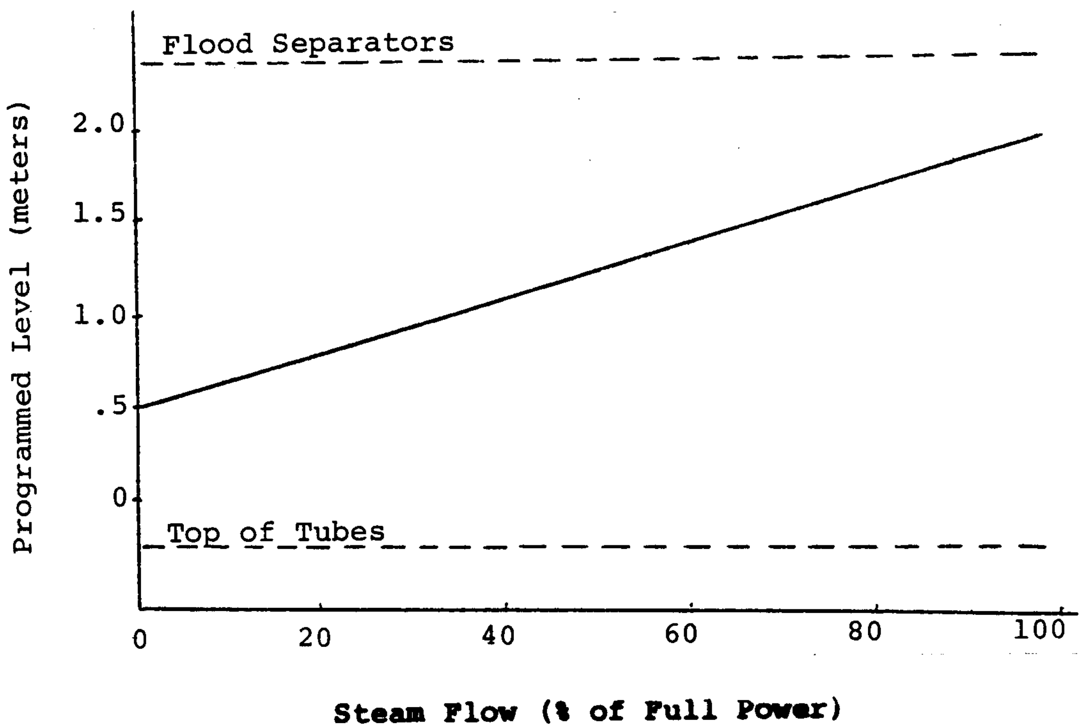
Figure 7.2



Swell and Shrink (Constant Level Program)

Figure 7.3

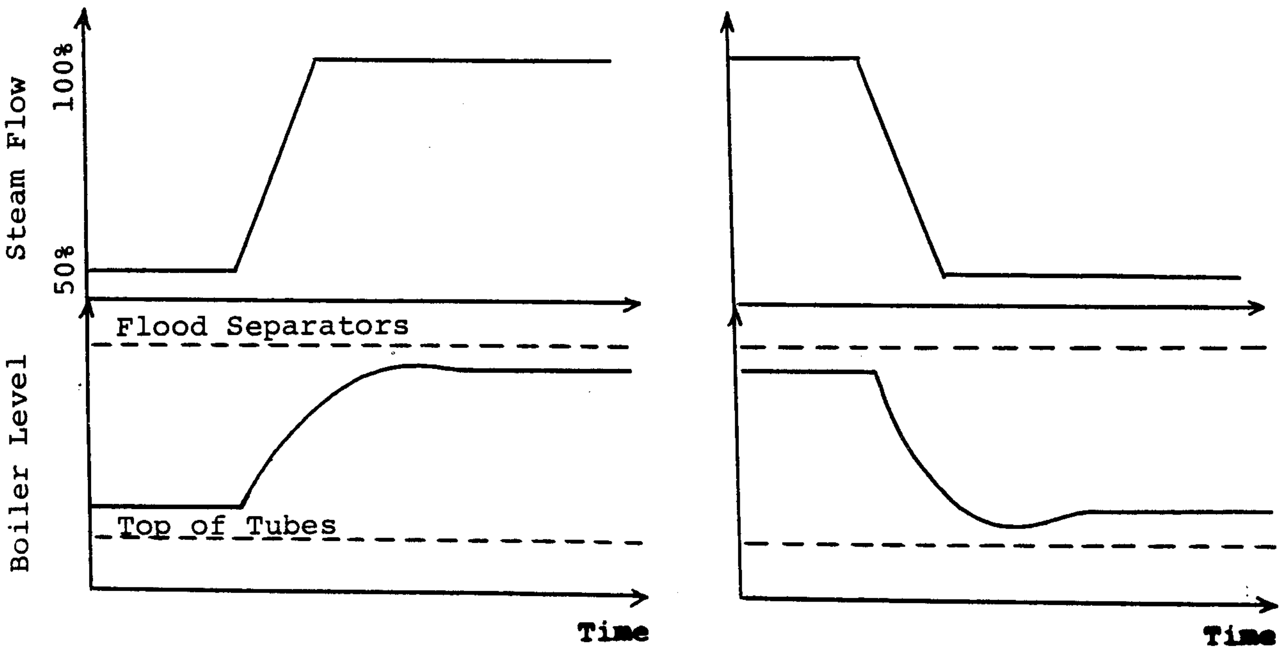
Steam generator level must be held within fairly close limits to keep from flooding the separators on the high end and to keep from uncovering the tubes on the low end. To prevent swell or shrink from causing an out of specification boiler level, the operating level is programmed to accommodate swell and shrink. The programmed level is low at low power levels to allow room for swell in the event of a fast power increase (safety valves open, steam reject valves open). At high power level, the programmed level is high to accommodate shrink in the event of a fast power decrease (turbine trip, load rejection).



Programmed Boiler Level

Figure 7.4

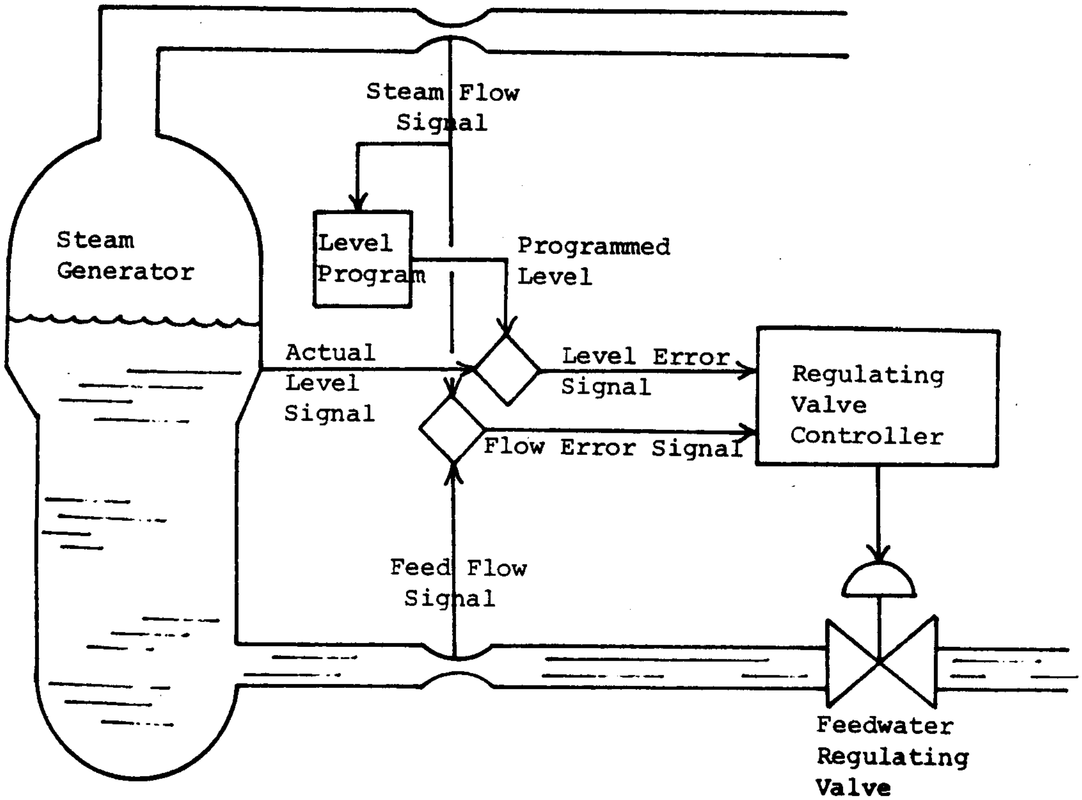
Figure 7.4 shows the level program for a typical large CANDU generating station. As steam flow increases from 0% to 100% of full power, the level program increases from .5 meters to 1.9 meters above the reference point (the location of the zero level reference point is somewhat arbitrary but corresponds approximately to the top of the tube bundle). Figure 7.5 shows the effect of swell and shrink in a steam generator with this type of level program. By comparing this with Figure 7.3 you can see the benefit of this type of level programming.



Swell and Shrink (Ramped Level Program)

Figure 7.5

The water level in the steam generators is regulated by the feedwater regulating valves. These valves are controlled by three variable parameters: steam generator level, steam flow and feedwater flow. Figure 7.6 shows a block diagram of feedwater regulating valve control.



Feedwater Regulating Valve Control

Figure 7.6

Steam flow is used to derive the programmed level in the steam generator (programmed level is proportional to power level). The actual steam generator level is compared with the programmed level and a "level error" signal is developed: actual level greater than programmed level shuts the feedwater regulating valve; actual level less than programmed level opens the feedwater regulating valve. At the same time steam flow is compared to feed flow and a "flow error" signal is developed: steam flow greater than feed flow opens the feedwater regulating valve; feed flow greater than steam flow shuts the feedwater regulating valve. It is the algebraic sum of the level error and flow error signals which determine the direction and magnitude of feedwater regulating valve movement.

The operation of the feedwater regulating valve on an increase in power level (steam flow) is shown in Figure 7.7. During normal power level changes the rate is slow enough that swell and shrink effects are not seen.

(a) Governor steam valves open;

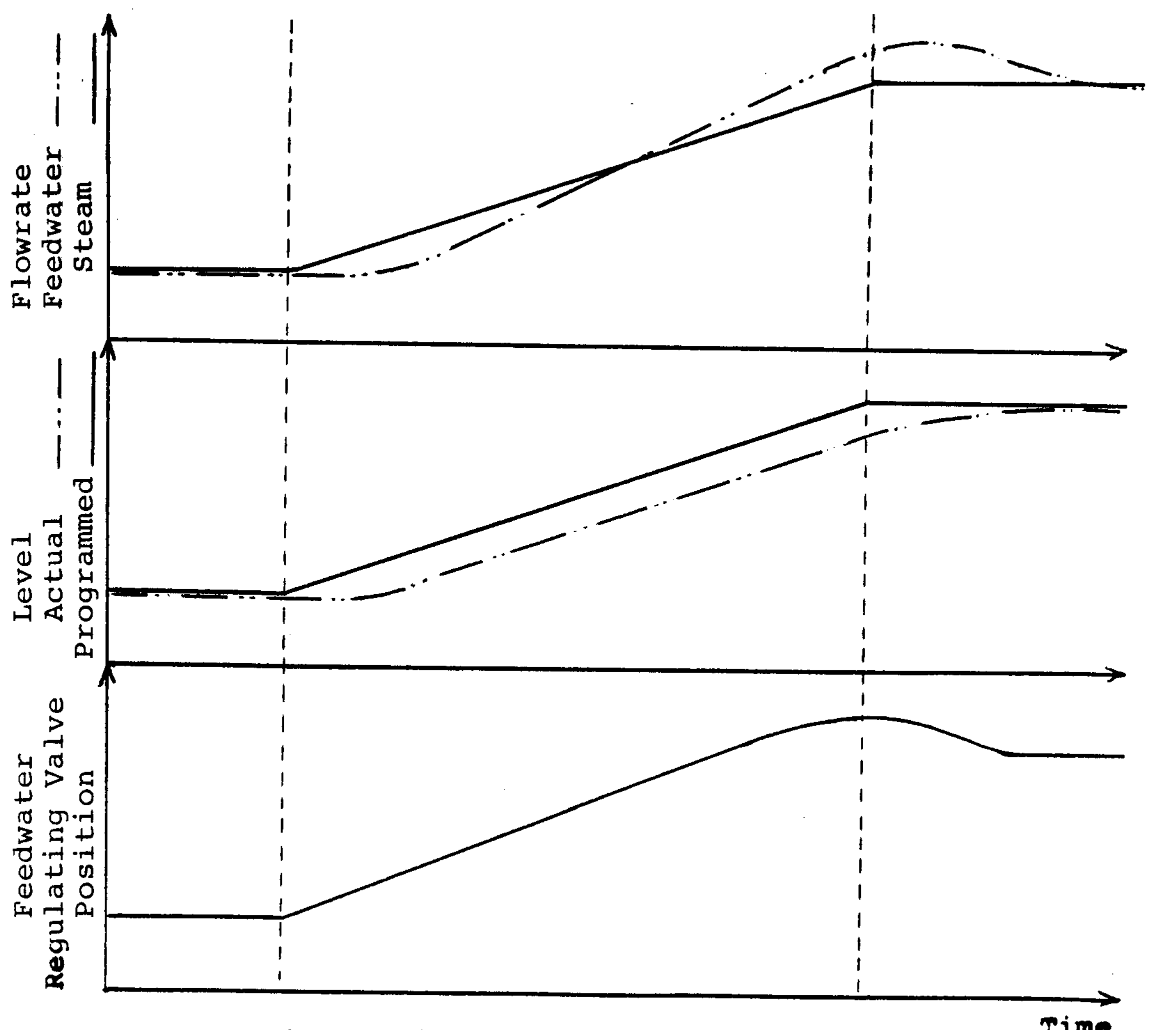
(b) Steam flow increases (steam flow greater than feed flow) and develops flow error signal;

(c) Programmed level increases (actual level less than programmed level) and develops level error signal;

(d) Both (b) and (c) act to open the feedwater regulating valve admitting more feedwater to the boiler;

(e) Feed flow rises above steam flow and level in boiler starts to increase;

(f) When actual level equals programmed level, the feedwater regulating valve closes partially so that feed flow matches steam flow.



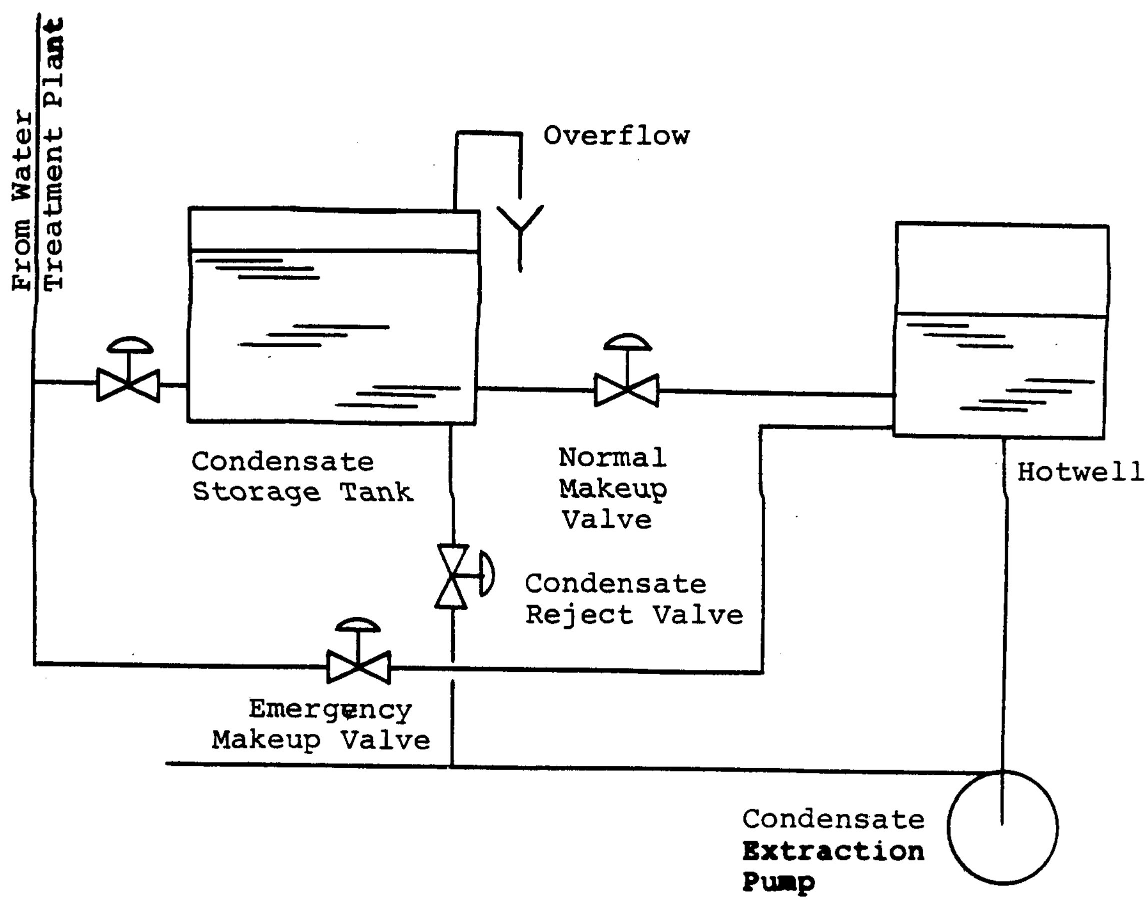
Increasing Power Transient

Figure 7.7

Deaerator Storage Tank Level Control

The deaerator storage tank is the primary inventory of feedwater within the feed system. The level in the deaerator storage tank is controlled between 2.75 meters and 3.00 meters. The condensate flow to the deaerator is regulated by the deaerator level control valves (usually three 50% valves or two 100% valves). When deaerator storage tank level falls, the control valves open up to admit more water from the condensate extraction pumps. The reverse occurs when deaerator level rises.

Condenser Hotwell Level



Makeup and Reject System

Figure 7.8

The level in the condenser hotwell is used to indicate the need for increasing or decreasing the inventory of water in the feedwater system. If hotwell level is too high, water is rejected from the system; if hotwell level is too low, makeup water is added to the system. Figure 7.8 shows the condensate makeup and reject system which is used to maintain hotwell level. If hotwell level drops below .5 meters, the normal makeup valve opens and water is "vacuum dragged" from the condensate storage tank into the hotwell. If the hotwell level rises above .8 meters, the condensate reject valve opens and the excess condensate is pumped back to the condensate storage tank. If the level in the condensate storage tank gets too low, water is added from the water treatment plant.

If, in an emergency, hotwell level drops below .35 meters, the large emergency makeup valve opens to admit water directly from the water treatment plant storage tanks into the hotwell.

These three level control systems (steam generator, deaerator and hotwell) work in conjunction to maintain a proper feed system water inventory. The response to a gradual increase in power would be as follows:

(a) steam flow increases;

(b) feedwater regulating valve opens to supply additional feedwater to match steam flow and to raise boiler level to higher programmed level;

(c) level drops in deaerator storage tank;

(d) deaerator level control valves open to restore storage tank level;

(e) level begins to drop in hotwell. By this time the increased steam flow to the turbine which started the transient is beginning to enter the condenser as LP turbine exhaust. However, since the programmed boiler level increased as steam flow increased, there will be a net loss of water from the hotwell (water that was in the hotwell is now in the boiler). This will probably result in a need for some makeup water.

Since there always exists some steam and feedwater leaks and since some water is lost through blowdowns and sampling, there is a continuous loss of water from the system. Even at constant power level, the normal makeup valve will periodically open on low hotwell level to makeup for these losses.

Recirculation Lines

There is a minimum flow of condensate which must exist through the condensate extraction pumps, boiler feed pumps, air ejector condenser and gland exhaust condenser to keep them from overheating. This minimum flow is typically on the order of 10% of full power flow. Recirculation lines are fitted on the discharge of each boiler feed pump to recirculate water back to the deaerator, when the feedwater regulating valves are nearly closed. This keeps boiler feed pump flow above the minimum required. Another recirculation line is fitted downsteam of the air ejector condenser. When the deaerator level control valves are nearly closed, this recirculation valve opens. The water is recirculated back to the hotwell to maintain an adequate flow through the condensate extraction pump, gland exhaust condenser and air ejector condenser.

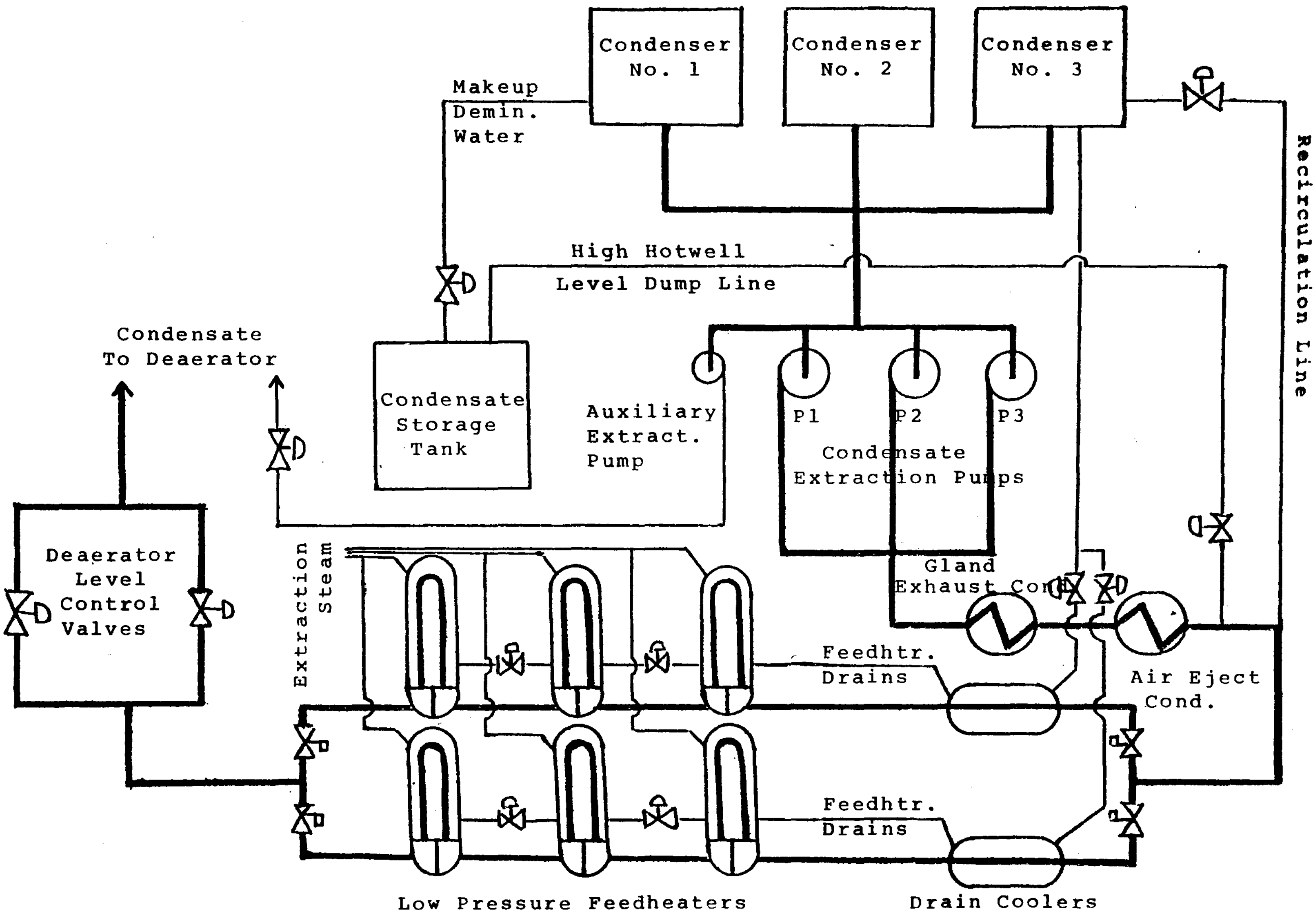
Feed System Reliability

Figures 7.9 and 7.10 show the feedwater system of a typical large CANDU generating station. Reliability is achieved by having two banks of high pressure and low pressure feedheaters. Although both banks are normally in operation, either bank is capable of handling 100% of full power flow. This allows isolation of one bank for maintenance or tube leak isolation while maintaining the plant at 100% power.

The feedwater system normally has sufficient excess pumping capacity so that one boiler feed pump and one condensate extraction pump can be shutdown and still enable 100% full power operation. The system shown in Figures 7.9 and 7.10 has three 50% extraction pumps and three 50% boiler feed pumps. Other combinations frequently encountered are two 100% pumps or two 50% and one 100% pumps.

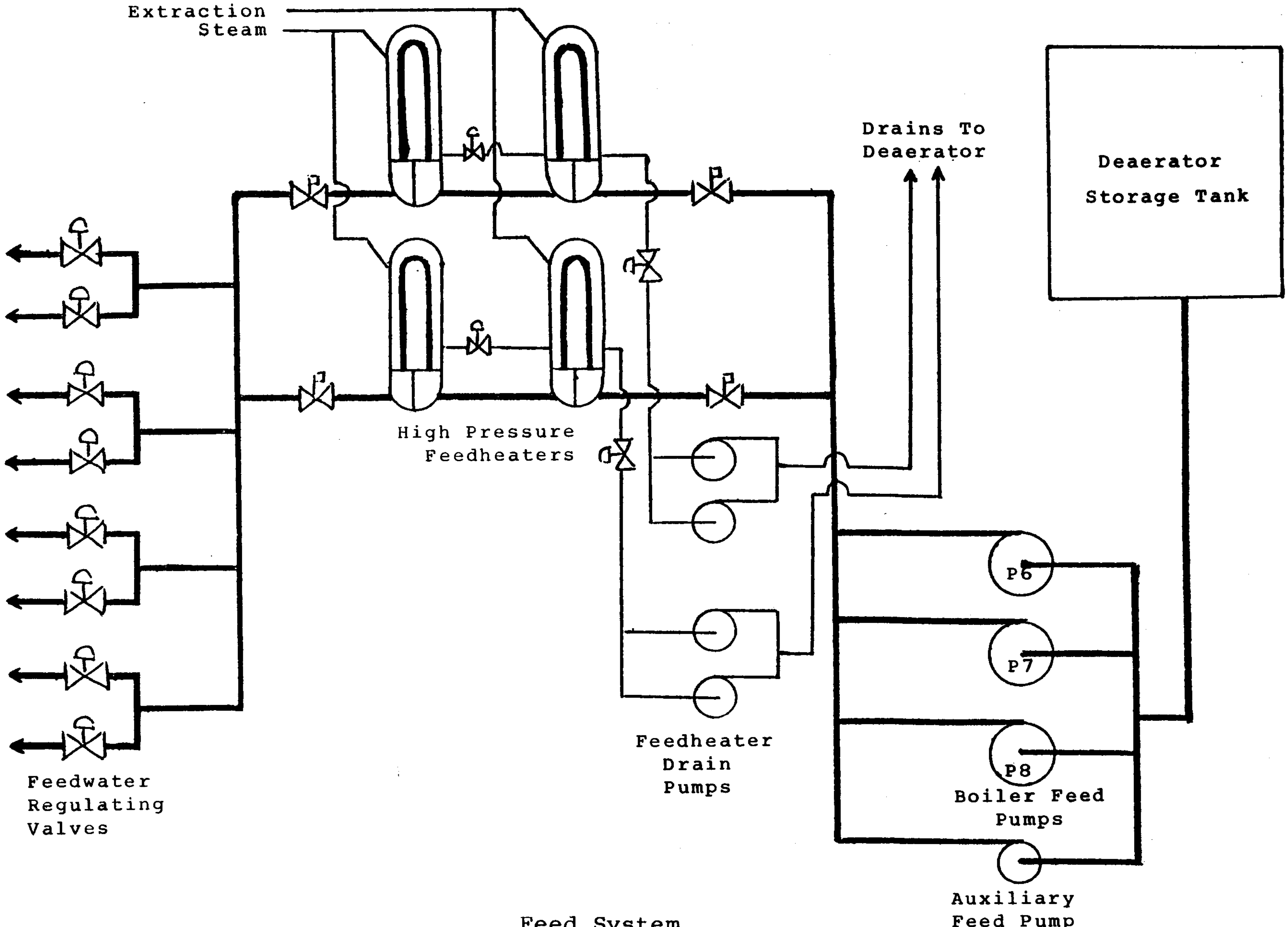
A 5% auxiliary condensate pump powered from class III power is available for maintaining deaerator level when shutdown and during reactor cooldown. A class III powered 3% auxiliary boiler feedpump is provided for use under the same conditions.

In order to recover the maximum benefit from the extraction steam to the feedheaters, the feedheater drains are cascaded. That is, the drains from one feedheater are sent to the next lower pressure feedheater. Some of the heat energy of the drain water may be recovered by allowing partial flashing of the drains back to steam in the lower pressure feedheater. In the LP feedheaters, the #3 LP feedheater drains cascade to the #2 LP feedheater, the #2 LP to the #1 LP feedheater and the #1 LP feedheater drains pass through a drain cooler and then back to the condenser.



<u>Condensate System</u>

<u>Figure 7.9</u>



<u>Feed System</u>

<u>Figure 7.10</u>

In the HP feedheaters, #6 HP feedheater cascades to #5 HP feedheater and the drains from #5 HP feedheater are pumped back to the deaerator. This cascading of feedheater drains results in significantly more heat energy being removed from the extraction steam. If the drains were sent directly to the condenser, the sensible heat of the drains would be lost.

Feedheating Temperature Control

The flow of turbine extraction steam to the feedheaters is dependent on the differential pressure between the turbine extraction belt and the feedheater shell. Just as the pressure in the main condenser is a function of inlet lake water temperature and ccw flow, so the feedheater shell pressure is a function of inlet feedwater temperature and feedwater flow. Turbine extraction belt pressure is a function of turbine steam flow (power level). At low power virtually all of the turbine will be under a vacuum. As power increases, turbine pressure will rise, until at 100% power only the final four or five stages of the LP turbine operate below atmospheric pressure.

As power level increases, the flow of feedwater through the feedheaters increases and lowers the pressure in the feedheater shell. This increases the flow of extraction steam to the feedheater and raises the outlet temperature of the feedwater. As more extraction steam is condensed in the feedheater shell, the water level in the shell begins to rise. An automatic drain valve opens to allow more drain water to cascade to the next feedheater.

At low power levels, the low pressure in the turbine unit results in low extraction steam temperatures. This means that at low power levels the extraction steam to the deaerator and HP feedheaters is at too low a temperature to be of any value in feedheating. At low power levels, the extraction steam which would normally go to the HP feedheaters is sent directly to the condenser. Main steam is admitted to the deaerator to provide feedheating.

As turbine power increases, turbine extraction steam pressure and temperature increase. Between 25% and 50% of full power, extraction steam is automatically cut in to the HP feedheaters. In addition as extraction steam pressure to the deaerator increases, the main steam is gradually cut off, until extraction steam provides all of the feedheating.

During poison prevent operation when the turbine is shutdown or motoring there is no extraction steam available for feedheating. In this case, main steam is supplied to the deaerator to provide all of the feedheating.

Feed System Chemistry

The purpose of the various chemical treatments is to provide low conductivity, low total solids, low dissolved oxygen, and optimum alkaline conditions in our feed system and steam generators. 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 four 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.

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.

4. Introduction of impurities which can lead to increased moisture carryover from the steam generators.

Maintenance of feed system and boiler chemistry involves:

(a) Minimized Dissolved Oxygen

The presence of dissolved oxygen greatly increases the general corrosion rate of feed system and boiler materials. In addition a wide variety of localized corrosions are caused or accelerated by the presence of dissolved oxygen. Oxygen and other non-condensable gases are removed by the condenser air extraction system and the deaerator. In addition hydrazine (N_2H_4) is injected into the feed system after the deaerator. The hydrazine reacts with the oxygen and removes it.

$$N_2H_4 + O_2 \rightarrow N_2 + H_2O$$

Hydrazine decomposes fairly rapidly at boiler temperatures and must be continuously added to the feed system.

(b) Maintain Basic PH

Feed system and steam generator pH is maintained between 7.5 and 10.5 by addition of morpholine (diethylenimide oxide) or similar "amine" compound. A slightly basic pH minimizes the corrosion rate of feed system and boiler metals. In addition the release of metal oxides into the system is minimized. Compounds such as Na_2HPO_4 and Na_3PO_4 are gradually disappearing as methods of pH control. These compounds tend to cause long term buildup of insoluable deposits in the boiler. In addition, they contribute to boiler dissolved solids which tend to increase carryover.

(c) Maintain Low Conductivity

Conductivity of steam generator is an excellent measure of water purity. If conductivity increases, it is indicative of condenser leakage or poor makeup water quality. In addition high conductivity indicates high dissolved solids which may contribute to carryover. When conductivity reaches 100 μmho/cm a blowdown must be initiated, and corrective action (locate source of impurities and restore low conductivity) should be initiated as soon as possible.

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.

Figure 7.11 shows the typical chemistry specifications for a large CANDU generating station.

Chemical Control Specifications

USI 36000
R-1 April 1973

System: Boilers

Variable	Sample Point or Indication	Sample Frequency	Range/Limit	Typical or Desired Value	Remarks
Conductivity	BOILER	(1) 2 boilers/shift	(6) 100 micromhos/cm	(6) 50 micromhos/cm	(1) 2 boilers per shift in each unit on rotating basis.
pH	WATER	(1) 2 boilers/shift	7.5 - 10.5	8.7	(2) Dependent on pH requirement.
SiO_2	SAMPLING	once/week	(6) <10 mg/kg	<5mg/kg	(3) Tritium must be measured in all boilers if routine analysis of main steam upon resampling has shown an increase by factor of 2 or more from the previous results.
N_2H_4	STATION IN REACTOR	(1) 2 boilers/shift	20-500 μg/kg	(2) 100 μg/kg	
Tritium	BUILDING.	(3) (4)	(3) (4)	(≃3 micro Ci/kg)	(4) Marked increase might indicate boiler leakage followed up by other tests, eg, gamma scan.
Magnesium	EACH BOILER	(1) 2 boilers/shift	<1 mg/kg (5) (6)	<0.100 mg/kg	
Sodium	IS SAMPLED	(1) 2 boilers/shift	N.A. (5)	2 mg/kg	(5) High concentration may indicate condenser leakage or bad make up water.
Copper	INDIVIDUALLY	once/week	N.A.	<100 μg/kg	(6) Boiler must be blown down immediately if its conductivity ≥100 micromhos but it is a good practice to blow down already at 50 micromhos. Blowdown to be also initiated if either SiO_2>10mg/kg, Mg>1mg/kg or Cl->10 mg/kg.
Iron		once/week	N.A.	<100 μg/kg	
Chlorides		once/week	N.A. (6)	<10 mg/kg	

Figure 7.11

ASSIGNMENT

1. Draw a schematic diagram of a typical feedheating system showing:

   (a) HP turbine extraction steam
   (b) LP turbine extraction steam
   (c) LP feedheaters
   (d) HP feedheaters
   (e) condensate extraction pump
   (f) boiler feed pump
   (g) deaerator
   (h) steam generators
   (i) feedwater regulating valves
   (j) deaerator storage tank level control valves
   (k) condensate storage tank
   (l) drain cooler
   (m) air ejector condenser
   (n) gland exhaust condenser
   (o) recirculation lines.

2. What is the function of the:

   (a) condensate recirculation line?
   (b) boiler feed pump recirculation line?

3. Explain how level is maintained in each of the following:

   (a) steam generator
   (b) deaerator
   (c) condenser hotwell.

4. Explain the sequence of events for an increase in turbine power from 10% to 100%. Include in your discussion:

   (a) steam to deaerator
   (b) steam generator level
   (c) deaerator level
   (d) hotwell level
   (e) extraction steam to HP feedheaters
   (f) condensate extraction pumps
   (g) boiler feed pumps
   (h) increase of extraction steam to feedheaters.

5. Why is makeup necessary in the feedwater system?

6. How is the reliability of the feedwater system guaranteed?

7. How and why are each of the following maintained within specifications?

   (a) pH
   (b) O_2
   (c) Conductivity.

R.O. Schuelke

Chemistry - PI 24

SECONDARY HEAT TRANSPORT SYSTEM - IMPURITIES

Reference: "Corrosion in Action" (a film on video-tape).

The Secondary Heat Transport System can be thought of as a heat sink for the Primary Heat Transport System via the steam generator.

Consider the following questions:

1. What would be the effect of a layer of scale on the secondary side of the steam generator tubes?
2. What would be the source of such a scale?

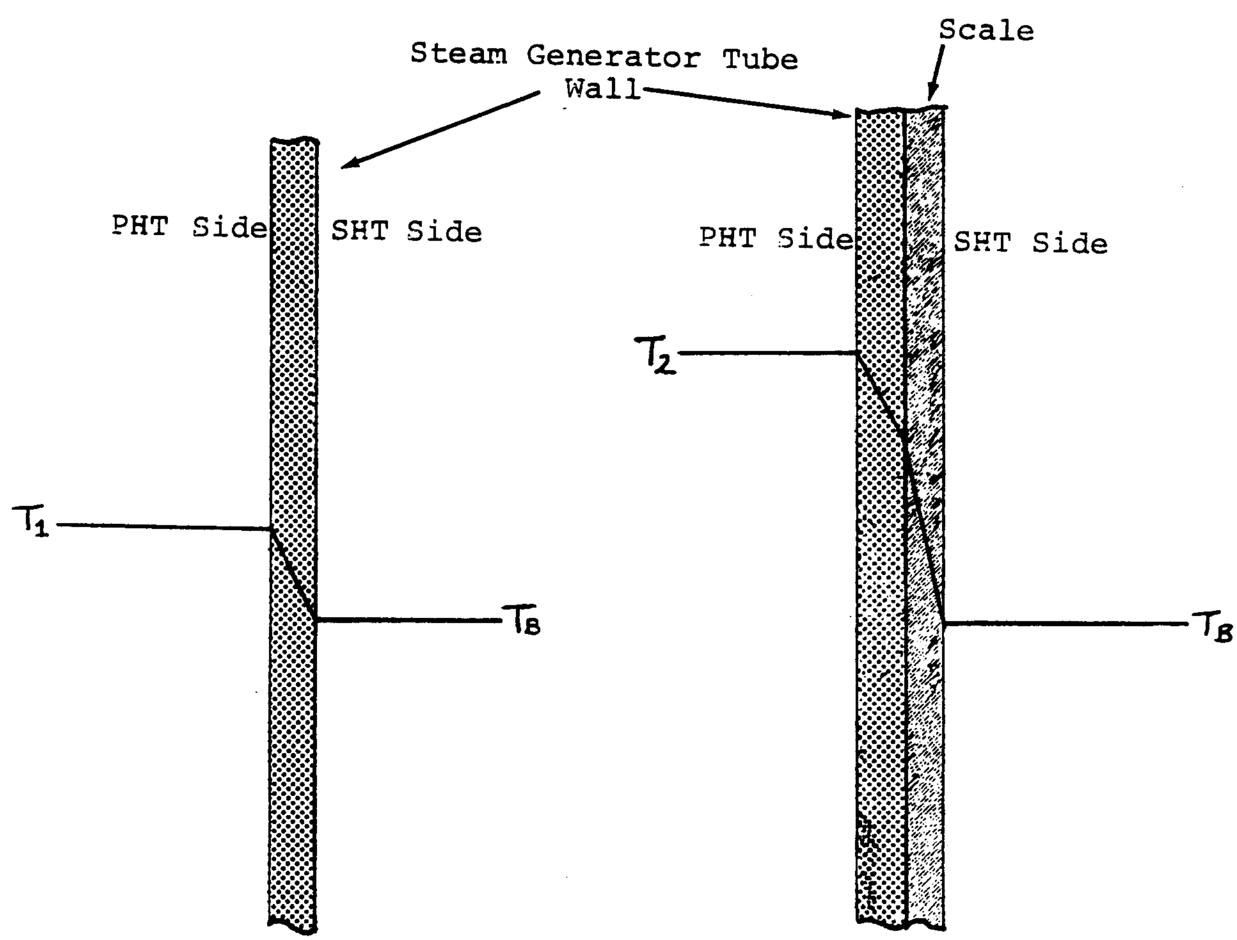
Effects of Scale

i) Loss of Heat Transfer

While exact calculations require data on scale porosity and thickness, it is sufficient to realize that scale will, due to its low thermal conductivity, act as an insulating barrier for heat transfer from the PHT system to the SHT system. This loss in heat transfer results in:

ii) Higher Tube Metal Temperature

This effect can be shown as follows:



Section A

Section B

Figure 1

Figure 2

Effect of Scale on Boiler Heating Surfaces

Figure 1 is a drawing of a **clean** steam generator tube showing the normal temperature profile between the PHT system and SHT system with a temperature gradient through the tube. Note that T_B, the secondary side temperature is fixed by the boiling temperature of water at the steam generator secondary side pressure.

Figure 2 shows the same tube with a layer of scale. T_B remains the same since it is fixed; however, due to the insulating effect of the scale layer, the temperature profile is modified and the primary side tube wall temperature rises to T_2.

If T_2 is high enough, reactor trip sequence will be initiated.

iii) Sludge Pile on Lower Tubesheet

Particulate matter or loose scale will settle in a steam generator and form a pile of sludge on the lower tubesheet. Actual measurements in Ontario Hydro plants have found sludge piles up to 10 cm.

iv) Cracking of Tubes

Given that a sludge pile is porous and that "boiling" does occur in it, any impurities in the boiler water will concentrate in the sludge pile as there is no washing effect from a high water flow at the site of boiling. If these impurities are high in something like chlorides then high chloride concentrations will develop adjacent to the steam generator tubes leading to Stress Corrosion Cracking of the tubes.

v) Denting

If scale forms between a tube a tube support plate and the concentration mechanism outlined above takes place for a chemical species which is corrosive to the tube support plate, then the resultant corrosion products formed on the plate, being higher volume than the original material will exert enough pressure on the tube to crimp it, or **dent** it as this phenomenon is known in the literature.

Source of Scale

There are 3 main sources of scale forming substances. In descending order of probability:

- feedwater system corrosion products.
- condenser leaks.
- Water Treatment Plant malfunction.

If oxygen, and/or carbon dioxide is present in the feedwater, the mild steel piping and other ferrous components will be attacked resulting in the formation of iron oxide. These iron oxides usually break off the surfaces where they are formed and travel with the feedwater as suspended solids or colloids (smaller than suspended solids). They will end up in the steam generator and stay there as the steam generator may be thought of as a distillation unit in some senses. Similarly, oxygen with the aid of ammonia or carbon dioxide may attack brass (Cu/Zn) or cupro-nickel components to form small particles of oxides of copper, nickel and zinc. You should note that the sludge found in steam generators consists mainly of iron oxide with some copper oxide and quite small amounts of nickel and zinc oxides.

A further problem with copper oxides is that if they deposit on steel tubes, a galvanic cell is set up plating out metallic copper at the expense of iron oxide formation from the steel (pitting).

If a condenser tube has even a pin-hole leak, raw water will be sucked into the condenser due to the difference in pressure of raw water in the tubes and the near total vacuum of the turbine exhaust steam in the condenser shell. This raw water will contain mineral salts and may contain suspended solids and/or organics. The big offenders of this group are calcium and magnesium bicarbonates (hardness) and silica (sand). The hardness salts can bake onto the steam generator tubes just as they do in your tea-kettle at home.

Silica is a dual threat. Not only will it form silicates which form a dense tenacious scale, but at the high temperature of the steam generator, silica has the potential to volatalize and travel with the steam only to deposit out on steam system components and turbine blades when the steam temperature drops.

(Silica may appear in raw water as a suspended solid, and/or a colloid, and/or a dissolved species.)

Although a remote possibility when one considers the automatic trip feature provided, Water Treatment Plant (which provides make-up to the system) malfunction could introduce anything between slightly "off-spec" product to regenerant wastes extremely high in mineral content to the feed system.

On the opposite end of the spectrum, from deposition or scaling which cause loss of heat transfer and flow restriction are **corrosion** and **erosion** which cause metal wastage and possible failure.

"What impurities or conditions will cause erosion or corrosion in the Secondary Heat Transport System?"

Erosion

Erosion is caused by suspended particles in a fluid stream doing **physical** damage to metal components by impingement. The sources of these suspended solids are, in order of probability:

- feed system corrosion products.
- metal filings and scraps left from improper clean-up after construction.
- condenser leaks.

Corrosion

Corrosion is metal wastage by **chemical** action.

i) pH - feed system and steam generator are kept at an optimum pH. By optimum for steel systems, we mean about 9.2 - 9.6 which will be high enough to promote formation of a protective magnetite layer and low enough to prevent caustic embrittlement. This pH is of course, high enough to prevent acid attack.

For systems containing brasses, a somewhat lower pH is favoured to prevent caustic dissolution of the metal. However, systems containing brasses have steel components as well and a compromise pH of 8.8 - 9.2 is usually used. (Main condenser and low pressure feedheater tubes are usually made of brass except for Bruce-B and later stations which are all ferritic).

ii) Chlorides - If the chloride ion is present in the steam generator system, the components made of austenitic stainless steels will be damaged due to stress corrosion cracking. In fact, great care must be taken at all stages in the life of stainless systems to prevent their being exposed to chlorides, even initial fills for hydrostatic testing must use demineralized water.

Chlorides are present in raw water and will enter the system whenever it is exposed to a source of raw water (condenser leak; Water Treatment Plant malfunction; carelessness).

iii) Oxygen - If air leaks into the system, oxygen will be present which will corrode steel parts just the way the automobile fender disappears after a few years.

iv) Ammonia, Carbon Dioxide - Ammonia from hydrazine (see module 21-2) break-down will attack copper containing components (brasses) as will CO_2. Steel components are attacked by Carbon Dioxide.

Carryover

Carryover is the entrainment of boiler water in the steam. Chemically, the problem of carryover centers on the solids contained in boiler water which will lead to deposit formation on steam system components, valve gear, and turbine blading. Mechanical problems associated with carryover are covered in the Turbines course. In general, carryover is caused by two phenomena; foaming and priming.

Foaming is the condition resulting from the formation of bubbles on the surface of the boiler water. The foam may entirely fill the boiler steam space or may be of relatively minor depth. In either case, this foaming condition leads to appreciable entrainment of boiler water in the steam.

Priming is a more violent action, resulting in the throwing of "slugs" of boiler water over with the steam. This action may be similar to the "bumping" experienced when water is boiled in an open beaker or it may be the bulk rise of boiler water level leading to massive carryover.

The chemical causes of carryover are related to the impurities in the boiler water. In general high concentrations of solids in the boiler water, whether dissolved or suspended will lead to high carryover.

The presence of oil in boiler water is highly undesirable from the standpoint of carryover. In combination with the alkaline boiler water, oils have a strong tendency to foam. (Alkalis and oils generally form soaps.)

Boiler water impurities are controlled by "blow down" which will be covered in module 21-2.

Practice Exercises:

Consider each of the six objectives at the front of this module as if it were a question. Write out your answers and have them checked by the course manager or a colleague.

If you have any difficulty picturing the mechanisms in this module be sure to consult the course manager before proceeding.

P. Dodgson

Chemistry - PI 24

CHEMICAL ADDITIVES AND BLOWDOWN

In Module 21-1, we discussed the sources and effects of various impurities which may be present in the secondary system. Recall that the major problems centered on:

- deposition.
- corrosion.
- erosion.

In order to minimize these effects **chemical additives** may be injected into the system.

"What chemicals are added to the secondary system, where, and why?" (If you want to take a quick look at a secondary system schematic, see Module 70-0).

Morpholine

Morpholine is a volatile organic chemical which is in-
je
pholine is a pH controller and is injected, in a modern station, automatically and continually as required on a signal from a pH monitor.

Hydrazine

Hydrazine is injected just after the deaerator. It is an oxygen scavenger and is used to pick up any oxygen which passes the deaerator.

$$N_2H_4 + O_2 \longrightarrow 2H_2O + N_2$$

Secondarily, hydrazine helps raise the feed system pH to optimum level.

Both morpholine and hydrazine are volatile liquids and do not contribute to system solids. Furthermore, they travel with the steam and help protect the steam system.

Cyclohexylamine

For those plants where only ferrous components are present (eg, Bruce B and Darlington), the use of cyclohexylamine which is a slightly stronger base than morpholine may be advantageous to achieve the higher pH required at a lower injection rate. Cyclohexylamine would be injected at the Condensate Extraction Pump discharge just the same as morpholine. Note that cyclohexylamine is also a volatile organic liquid.

Phosphates

Until quite recently, it was the practice to inject one form or other of Sodium Phosphate into the steam generator. The benefits were that phosphate would stay in the water of the steam generator and provide optimum pH conditions. Furthermore, any calcium or magnesium salts entering due to condenser leakage, would be precipitated into a soft non adherent sludge by the phosphate. (You have seen this when detergent powder consisting mainly of phosphates is added to laundry.)

Recent data (1975-1980) now points to the use of phosphates as a contributory factor in the thinning of Inconel tubes. Therefore, the use of phosphates in NGD has been discontinued. Note that for conventional ferrous boilers, "phosphate" is the treatment of choice.

Blowdown

Throughout our discussion of Secondary side impurities, it has become evident that solids will accumulate in the steam generator. These solids are: corrosion products, slight contamination of feedwater, "dirt" in systems. For the most part, these solids are not volatile and if allowed to accumulate in the steam generator, will contribute to deposition and carryover.

Another point to remember is that silica, for which there is no effective chemical treatment, accumulates in the steam generator and if its concentration is allowed to rise, it will ""boil off" and travel with the steam. Modern power plant practice sets the limit of 0.02 parts per million of silica in steam. This limit can be achieved by limiting silica in the steam generator.

"How are "inert" solids and silica controlled in the steam generator?"

Blowdown is the method used to remove part of the boiler water and replace it with makeup water. The makeup water contains a much lower impurity concentration than the boiler water and so the amount of impurity in the boiler is reduced.In addition, the boiler water is removed from the bottom of the boiler where the solids have collected. When the valves are opened, the water surges out under the high boiler pressure and the solids are carried along in slurry form.

The purpose of blowdown depends on the condition which has made it necessary. Blowdown is used to avoid scale formation due to excessive amounts of suspended solids and to prevent carryover of solids in the steam. It is also used to control high dissolved solids concentrations. For high pressure boilers (> 4000 kPa), blowdown is based upon limiting the silica content of the boiler water so as to minimize the amount of silica that goes over with the steam. Contamination of steam with silica is usually not a big problem where boiler pressures are maintained below 2800 kPa.

Types of Blowdown

a) Intermittent Manual Blowdown

There are two types of boiler water blowdown - intermittent manual blowdown and continuous blowdown. Manual blowdown (or sludge blowdown) is necessary for operation of a boiler regardless as to whether or not continuous blowdown is also used. The blowdown takeoff is usually located in the lowest part of the boiler so that in addition to lowering the dissolved solids concentration of the boiler water, it will also remove a portion of the sludge which tends to concentrate in the lower part of the boiler.

When continuous blowdown is also installed, the primary purpose of manual blowdown is to aid in the removal of suspended solids which may have accumulatd in the lower boiler. However, in instances where the boiler feedwater is exceptionally pure, blowdown can be employed infrequently (eg, at Pickering).

When continuous blowdown is not installed, it is necessary to use manual blowdown to control concentrations in the boiler water (eg, at NPD). Whenever the dissolved solids or suspended solids approach or exceed predetermined limits, it is necessary to use manual blowdown to lower these concentrations. In practice, the valve for bottom blowdown is opened periodically in accordance with an operating schedule and/or chemical control tests. From the standpoint of control and results, frequent short blows are preferred to infrequent lengthy blows. This is particularly true when the suspended solids content of the water is appreciable. Examine the records of short, frequent blows versus long, infrequent blows in Figure 1. The curves show that with the use of frequent, short blows, a more uniform concentration of solids is maintained and in general, a smaller quantity of water is blown down because of better control. However, from an operating standpoint, it is sometimes undesirable to use manual blowdown frequently enough to properly control concentrations - particularly where the feedwater solids content is high.

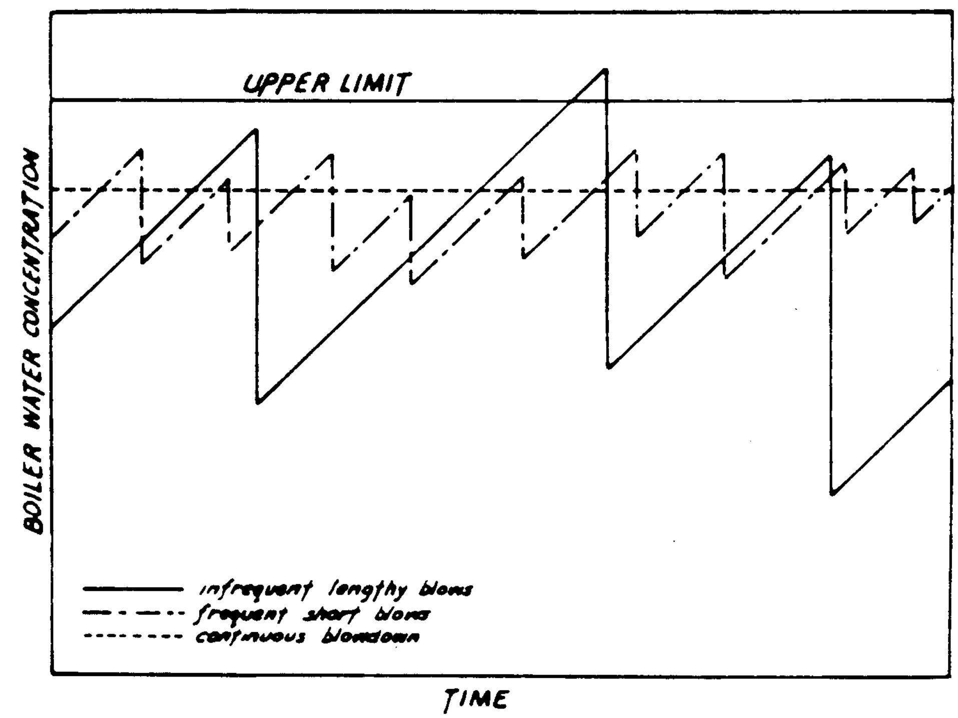


Figure 1

Effect of Different Types of Blowdown on Boiler Water Concentration

b) Continuous Blowdown

Continuous blowdown may be viewed as an extension of the practice of frequent, short blows. Blowdown water is withdrawn from the boiler continuously. Periodically, adjustments are made to the rate of withdrawal in accordance with control test results.

Continuous blowdown has two advantages. First, it is possible to maintain close control of boiler water concentrations at all times. Second, the heat content of the blowdown water can be recovered. If an efficient heat exchanger is used, the only heat loss is the terminal difference between the incoming feedwater and the blowdown water to the sewer; usually about 10°C.

Further savings can be realized by flashing a portion of the boiler water into steam at a lower pressure (a process similar to the use of dashboxes on the extraction steam lines from the turbines) and using the resulting low pressure steam for process or feedwater heating. The use of a flash tank in conjunction with a heat exchanger makes it possible to use a smaller heat exchanger as a large portion of the recoverable blowdown water heat is contained in the steam. Thus a savings can be made on capital equipment cost.

Practice Exercises

1. Fill in the following table:

Item	Added at:	for the purpose of:
Morpholine		
Hydrazine		
Cyclohexylamine		

2. Cyclohexylamine would be used if -----.

3. The purpose of blowdown is to -----.

4. Write a brief description or draw a diagram to explain the difference between continuous and intermittent blowdown.

   Check your answers to these with a colleague.

Putting Things Together:

The AECB has a question which they are fond of posing which covers the material in modules 21-1 and 21-2. Perhaps you would like to draft out an answer, then compare it to the sample answer provided.

Question:

With respect to boiler feed water discuss:

(a) pH
(b) conductivity
(c) dissolved gas content
(d) organic matter

In terms of: (a) Why they are controlled.
(b) How they are controlled.
(c) Consequences of not controlling.

Answer:

(a) pH is controlled to prevent corrosion of steels and brasses in the system.

It is controlled at 8.6 - 9.2 by adding MORPHOLINE, a pH controller at the Condensate Extraction Pump discharge. MORPHOLINE is a volatile (thus will not form deposits) amine and the addition of a small amount will raise the pH to the required range. At the deaerator outlet, HYDRAZINE is added to scavenge OXYGEN. It should be noted that hyrazine is also a volatile amine and a slight excess will help maintain the pH in the desired range.

If pH is not controlled, too low a pH, ie, below 8.6 will lead to acidic attack on the mild steel in the system leading to loose corrosion products which will be transported to the steam drum. (Not to mention the damage to system components as well). Too high a pH will result in two problems:

(i) Brasses in the systems, eg, L.P. Heater tubes, will tend to dissolve.

(ii) Any protective magnetite layer on steels will tend to be removed.

The range of 8.6 - 9.2 is chosen in the first place as a compromise between the optimum pH of 10.0 for steels and a neutral pH (7.0 - 8.0) for brasses.

(b) Conductivity is a measure of dissolved ionized substances. If not controlled, these substances may accumulate as scale in the steam generator; may be carried over with steam to form turbine deposits, or may be corrosive or agents leading to stress corrosion cracking (eg, Cl^-).

Low conductivity is maintaned by adding only pure make-up water, and by ensuring integrity of the unit main condenser and by blowdown. If conductivity is not controlled, the potential hazards are:

- scaling in the steam generator
- boiler solids carryover
- corrosion of system components
- if chlorides are involed stress corrosion cracking of stainless steels and zircalloys may occur.

(c) The dissolved gases that are normally considered are

Nitrogen and Oxygen from air

Ammonia from breakdown of hydrazine and morpholine

Carbon Dioxide from breakdown of morpholine and tramp organics

Oxygen and CO_2 are corrosive to steels. Ammonia, and CO_2 are corrosive to brasses and N_2 is a nuisance as it may lead to carryover in the steam generator and adds to the non-condensibles loading of the condenser.

Gases are removed mechanically by the deaerator and by the air ejector from the air removal section of the condenser.

More specific treatments of a particular gas are:

Oxygen: Residual oxygen after the deaerator is removed by the addition of Hydrazine

$$N_2H_4 + O_2 \longrightarrow 2H_2O + N_2$$

Ammonia: Air ejector drains may be run to sewer as this is where the majority of ammonia collects.

Carbon Dioxide: Most likely neutralized at pH of 8.6 - 9.2.

The consequenes of not controlling these gases are:

- corrosion by OXIDATION (O_2)
- gouging of steels by CO_2
- gouging of brasses by Ammonia and somewhat by CO_2
- bumping and priming in the steam generator
- large non condensibles therefore low condenser vacuum.

(d) Organic matter if not controlled may lead to foaming in the steam drum, asphaltic deposits and the generation of CO_2 in the system.

There is no true internal treatment for organics other than blowdown. Externally, organic matter is removed from make-up water in the clarifier and in the carbon filters.

The consequences of not controlling organic matter are:

- foaming in the steam drum
- asphaltic deposits in the steam drum
- generation of CO_2.

P. Dodgson

Turbine and Auxiliaries - Course PI 34

EFFICIENCY OF THE CANDU TURBINE CYCLE

Objectives

1. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by raising boiler pressure.

   (b) State the main limitation on the improvement in (a).

2. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by lowering condenser pressure.

   (b) State two limitations on the improvement in (a).

3. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by superheating in the boiler. .

   (b) State the main limitation on the improvement in (a).

4. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by:

   (i) reheating between the high and low pressure turbines

   (ii) using extraction steam for feedheating.

   (b) State the main limitation on each improvement in (a).

   (c) State two practical benefits of each improvement in (a).

5. (a) Explain how the thermal efficiency of the CANDU turbine cycle can be improved by moisture separation.

   (b) State the practical benefit of moisture separation.

Module Seven deals with efficiency. After definitions and simple calculations are dealt with, the rest of the module concerns the thermal efficiency of the CANDU cycle. The intent of this module is to make you aware of various considerations that have been taken into account in order to make CANDU generating stations as efficient as possible.

Efficiency can be defined as output divided by input, often expressed as a percentage value. This definition can apply to many things: the efficiency of heat transfer in the boilers, the efficiency of a pump, the efficiency of a turbine, etc. So that you can consider the thermodynamic cycle that represents a CANDU unit, we will define a particular type of efficiency - thermal efficiency.

The thermal efficiency of a system is defined to be the net work output of the system divided by the total heat input to the system, often expressed as a percentage. The net work output is the work produced by the system minus the work put in to the system in order to make it operate.

→ Answer questions 7.1 and 7.2 in the space provided before you proceed, then check your answers with the "TEXT ANSWERS".

7.1) Define the following:

(a) Efficiency: ______________________________

____

(b) Thermal efficiency: ______________________

____

7.2) 2390 MW of heat are added in the boilers of a CANDU unit. If the unit produces 788 MW of electricity and if 5.5 MW are put in to pump the feedwater from the condenser to the boilers, what is the thermal efficiency of the cycle?

→ At this point you should be able to do the first two objectives for this module. If you feel you need more practice, consult with the course manager.

CANDU Cycle Thermal Efficiency

The secondary heat transport system is the thermodynamic cycle of a CANDU unit. It is this system that the heat produced by nuclear fission is used to produce shaft mechanical energy. Needless to say, it is important that the thermal efficiency of this cycle be as high as can be reasonably achieved. This section deals with the cycle thermal efficiency of a CANDU unit - how it can be maximized and what practical limits are imposed on it.

Boiler Pressure

The pressure (and with it the temperature) of steam produced in the boiler should be as high as possible. Overall, as the saturation temperature and pressure of the steam leaving the boiler are raised, the ratio of net work output to heat input is increased. In other words, the higher the temperature and pressure of the steam, the higher the proportion of heat energy made available to produce work.

There is an upper limit on steam temperature (and thus pressure) in the CANDU system. This upper limit is imposed because of considerations involving the fuel and fuel sheath. The hotter the steam to be generated in the boilers, the hotter the PHT D_2O must be. This results in an increase in the surface temperature of the fuel in the reactor. The fuel is uranium dioxide (UO_2) manufactured in cylindrical pellets. As UO_2 is a ceramic material, its thermal conductivity is very low and the pellet core temperature is therefore much higher than the surface temperature. The temperature profile of a fuel pellet, including the sheathing, is shown in Figure 7.1.

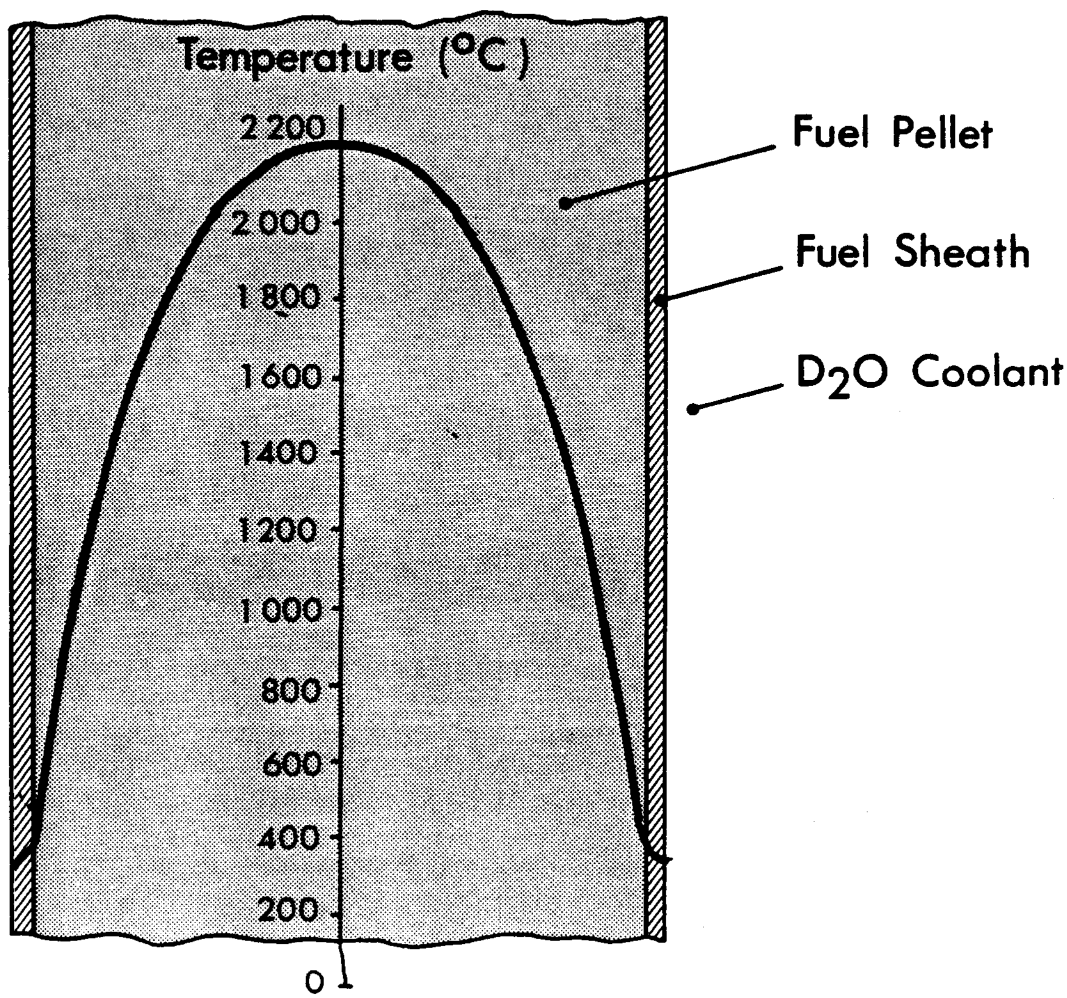


Figure 7.1

Note that temperatures in the order of 2000-2200°C occur in the centre of the fuel in order to produce 250-260°C steam. If higher steam temperatures were used, the pellet temperature would have to increase, and the danger of release of fission products due to fuel sheath failure would significantly increase. Thus the steam temperature is limited to a maximum of about 250-260°C.

→ Answer the following questions before you proceed. Check your answers with those in the "TEXT ANSWERS".

7.3 Explain how the thermal efficiency of the CANDU cycle can be improved by raising boiler pressure.

7.4) State the limitation on the improvement in thermal efficiency due to raising boiler pressure.

Condenser Pressure

The pressure (and temperature) of steam entering the condenser from the LP turbine should be as low as possible. As the pressure and temperature at the LP turbine outlet decrease, the work that has been extracted from the steam increases. The ratio of net work output to heat input increases significantly because of the extra work made available. Thus the thermal efficiency increases.

There are two main limitations on lowering the condenser pressure:

(a) The temperature of the condenser cooling water (CCW) is the main limitation. The water used to condense the steam is lake water. In summer the temperature of the CCW at the condenser inlet is about 20°C. Remember from Module PI 25-2 that this leads to steam temperature about 30°C in the condenser. It is interesting to note that as the CCW temperature drops with the approach of winter, the CANDU cycle efficiency increases.

(b) The lower the condenser pressure and temperature, the higher the moisture content of the steam leaving the LP turbine. Remember that moisture content above about 10% is undesirable. Without the use of moisture separation, reheating, feedheating, and some other features (which you will consider in PI 34), the moisture content of steam entering the condenser at 30°C would be 21%. Moisture content would have to be reduced in order to have a low condenser pressure.

→ Answer the following questions in the space provided, then check your answers with those in the "TEXT ANSWERS".

7.5) Explain how the thermal efficiency of the CANDU cycle can be improved by lowering condenser pressure.

7.6) State two limitations on the improvement in question 7.5.

Superheating

Superheating steam in the boilers (ie, heating the steam to a temperature higher than the saturation temperature for the steam pressure) will increase cycle thermal efficiency. Here is an extra heat input required to superheat the steam but the ratio of available work from this steam to the total heat input required is increased. Thus the overall cycle thermal efficiency is increased.

The main limitation on superheating is the limit of 250-260°C imposed by the fuel and fuel sheath considerations mentioned in raising boiler pressure. Superheated steam at 250-260°C could be produced, but it would be at a lower pressure than saturated steam at the same temperature. The ratio of available work to heat input would be less because of the lower pressure.

Another option would be to use a fossil-fuelled superheater. The price of such fuel is, however, the main reason why fossil-fuelled superheaters are not used.

→ Answer questions 7.7 and 7.8 in the space provided, then check your answers with those in the "TEXT ANSWERS".

7.7) Explain how the thermal efficiency of the CANDU cycle can be improved by superheating in the boiler.

7.8) State the main limitation of the improvement due to superheating.

Reheating

Remember that reheating between the HP turbine and the LP turbine produces superheated steam entering the LP turbine. As a result, the moisture content of the steam in the LP turbine is reduced and the turbine efficiency is therefore, increased. Thus, the ratio of net work output to heat input, ie, the cycle thermal efficiency, is increased.

The main limitation on reheating is due to the temperature of the live steam used to heat the main steam. This live steam is taken from the steam leaving the boiler, so it is limited to about 250-260°C.

Although reheating to higher temperatures could be achieved by means of fossil-fuelled reheaters, economical considerations (fuel price) make this option unattractive.

→ Answer the following questions before you proceed. Check your answers with those in the "TEXT ANSWERS".

7.9) Explain how the thermal efficiency of the CANDU cycle can be improved by reheating between the high pressure and low pressure turbines.

7.10) State the main limitation on the improvement due to reheating.

Reheating provides practical benefits besides an increase in cycle thermal efficiency; these benefits are the main reason reheating is done. The first benefit is a reduction in moisture content in the steam as it goes through the LP turbine. The result of having drier steam is that the steam can be utilized to a lower temperature and pressure. This allows a lower condenser pressure to be used and this reduction itself also improves the cycle thermal efficiency.

The second benefit of reheating is that the steam flow is smaller to produce a certain power. Remember that reheating increases the enthalpy of the steam before it enters the LP turbine. The steam can produce more work than steam that has not been reheated, and thus less flow is necessary to produce a certain power.

→ Answer the following questions in the space provided, then check your answer with the one in the "TEXT ANSWERS".

7.11) State two practical benefits of reheating.

Feedheating

The water that is returned from the condenser to the boilers is heated from about 30°C to about 175°C. There are two reasons for this feedheating:

(1) If the water is returned to the boilers (which operated at about 250°C) at 30°C, large thermal stresses will occur in the boilers. To reduce these stresses, the water must be preheated.

(2) Less fuel is burned to produce a given amount of power, since most of the sensible heat needed to change the temperature of the water from condenser conditions to boiler conditions comes from feedheating.

Generally, steam is extracted from the turbine to heat the feedwater in the feedheaters. Using the extraction steam increases CANDU cycle thermal efficiency. The increase is due to the use of the heat energy of the steam.

If the extraction steam is not extracted, but allowed to continue through the turbine, it would produce some work and then enter the condenser. In the condenser about 90% of the latent heat of the steam is rejected to the lake as the steam is condensed.

When the extraction steam goes to the feedheater, the work that it could produce in the turbine is lost. However, the steam condenses in the feedheater and its heat is given up to the feedwater. Thus, at the expense of the loss of some work production, a significant amount of heat is conserved within the cycle. This conservation reduces the heat input in the boilers and the ratio of net work output to heat input, ie, the cycle thermal efficiency, is increased.

The increase in thermal efficiency depends on the number of feedheaters used and the pressures at which the steam is extracted from the turbine. The larger the number of feedheaters, the larger the increase in efficiency. However, the gain in efficiency due to installing each successive feedheater gets smaller and smaller. For a given number of feedheaters, it is possible to calculate optimal pressures for the extraction steam so that the increase in the thermal efficiency will be maximum.

The main limitation on the improvement due to using extraction steam for feedheating is economic. Using a larger number of feedheaters makes it possible to achieve a larger increase in the cycle thermal efficiency, however, it increases the capital costs. At some point there is a balance between the benefit from efficiency increase and costs incurred. In a CANDU station this occurs when 5-6 feedheaters are used and the feedwater is heated to about 175°C.

→ Answer the following questions, then check your answers with those in the "TEST ANSWERS".

7.12) Explain how using extraction steam for feedheating can improve the thermal efficiency of the CANDU cycle.

7.13) State the main limitation on the improvement in efficiency due to using extraction steam for feedheating.

Using the extraction steam for feedheating also provides practical benefits. The first is a reduction in moisture content of the steam in the turbine. When wet steam flows through the turbine, it has a very swirling motion. Centrifugal forces are exerted both on vapor and water droplets but due to the difference between their densities, the liquid is centrifuged outwards, ie, towards the turbine casing. As extraction steam is removed from the casing, its moisture content is much larger than the average moisture content of the steam flowing through the turbine. The result of this is a reduction in moisture content in the turbine.

The second benefit has to do with the amount of steam extracted. Up to 30% of the steam flow is removed from the turbine set for feedheating. This reduction in steam flow through the low pressure turbine enables a smaller, and therefore less costly, low pressure turbine to be used.

→ Answer questions 7.14, then check your answer with the one in the "TEXT ANSWERS".

7.14) State two practical benefits of using extraction steam for feedheating.

Moisture Separation

Moisture separation causes a significant increase in cycle thermal efficiency. The main effect of moisture in the steam is a tendency to retard the motion of the turbine blades; this causes a loss in work production in the turbine. Reducing the moisture content allows the turbine to produce more work for the same heat input. Thus the cycle thermal efficiency is improved.

The practical benefit of moisture separation is the same as the first benefit listed in reheating: moisture separation reduces the moisture content of the steam going through the LP turbine, which allows a lower condenser pressure to be used without exceeding the maximum acceptable moisture content of the steam at the turbine exhaust (about 10-12%).

→ Answer the following questions before you proceed. Check your answers with those in the "TEXT ANSWERS".

7.15) Explain how moisture separation can increase the thermal efficiency of the CANDU cycle.

7.16) State the practical benefit of moisture separation.

→ You have now completed the last module of PI 25. If you feel you can answer the objectives for this module, obtain a criterion test and answer it. If you are not confident at this point, please consult with the course manager.

PI 25-7 TEXT ANSWERS

7.1) (a) Efficiency - output divided by input, often expressed as a percentage.

(b) Thermal Efficiency - net work output of a system divided by total heat input to the system, often expressed as a percentage.

7.2) The heat input in this question is 2390 MW.

The net work output is 788 - 5.5 = 782.5 MW.

The thermal efficiency is (782.5 ÷ 2390) x 100 = 32.7%

7.3) As boiler pressure is raised, the steam temperature and pressure both increase. As they increase, the ratio of work available to heat input increases. This will increase the ratio of net work output to heat input, ie, the thermal efficiency of the cycle.

7.4) The main limitation is a maximum achievable steam temperature of about 250 -260°C, imposed by fuel and fuel sheath considerations.

7.5) As the steam pressure and temperature in the condenser are lowered, the work produced in the turbine increases. The ratio of net work output to heat input, ie, the thermal efficiency, increases.

7.6) The increase in efficiency due to lowering condenser pressure is limited by the temperature of the lake water used as the condenser coolant and by the maximum allowable moisture content in the LP turbine.

7.7) If superheating is done in the boiler, the steam will have a higher amount of work available to heat input ratio. The thermal efficiency of the cycle can thus be increased.

7.8) The main limitation on superheating in the boiler is 250-260°C temperature limit due to the material considerations of the fuel and fuel sheath.

7.9) Reheating between HP and LP turbines increases cycle thermal efficiency by providing superheated steam to the LP turbine. As the moisture content of the steam in the turbine is reduced, the turbine efficiency increases, and therefore the cycle thermal efficiency is increased.

7.10) The main limitation on improving cycle thermal efficiency by reheating is the live steam temperature. The live steam is taken from the main steam flow from the boilers; its temperature is limited to 250-260°C.

7.11) Reheating reduces moisture content in the lp turbine, which allows a low condenser pressure to be maintained. Also, the steam flow is less with reheating to provide a given power than the flow without reheating - thus the whole secondary system size is smaller and less costly.

7.12) Using extraction steam for feedheating increases CANDU cycle thermal efficiency because, at the expense of some loss of work production, a large amount of heat is conserved with the cycle. This heat is conserved because the steam is condensed by feedwater rather than by lake water. The ratio of net work output to heat input is increased because much less heat input is required.

7.13) The limitation on the efficiency improvement due to the use of extraction steam for feedheating is economic. It is the balance between the increasing cost of the equipment and the increasing gain in efficiency as the number of feedheaters increases.

7.14) The first benefit of using extraction steam for feedheating is that moisture content in the LP turbine is reduced. The second benefit is that the size of the turbine set is reduced due to smaller flow through the outlet portion of the turbine.

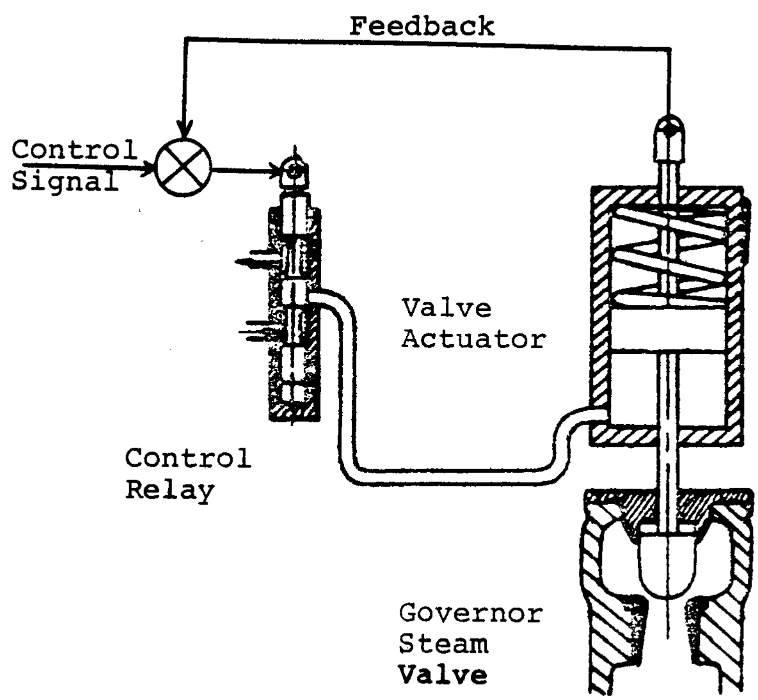
7.15) The effect of moisture separation is that less work is lost in the turbine (the work lost being due to retardation of the moving blades because of liquid impingment). The LP turbine thus has a higher work output for the same heat input, and the cycle thermal efficiency increases.

7.16) The practical benefit of moisture separation is that a low condenser pressure can be maintained without exceeding the maximum acceptable moisture content of the steam at the LP turbine exhaust (about 10-12%).

Turbine, Generator & Auxiliaries - Course 234

STEAM VALVE HYDRAULIC CONTROL

The movement of large control valves to regulate the steam supply to modern steam turbines, requires amplification of the control signals in both force and displacement to provide sufficient force to actuate the valves. Whether the control signal from the governor is mechanical or electrical, the signal is converted to a hydraulic fluid pressure for movement of the governor steam valves. The hydraulic relay and actuator has no competitor as a force amplifier for governor steam valve actuation. No other form of mechanical or electrical amplifier can develop the very high force demanded by the speed at which the governor steam valves, of modern turbines, must operate.



Steam Valve Actuation

Figure 3.1

Figure 3.1 shows the basic hydraulic relay and actuator associated with a governor steam valve. The governor steam valve is opened by increasing the hydraulic pressure on the underside of the actuator piston and compressing the spring; the governor steam valve is shut by draining oil from the underside of the actuator piston and allowing the spring to expand and force the valve shut. The control relay is positioned either mechanically or electrically to supply fluid to, or drain fluid from, the actuator. Mechanical or electrical feedback is provided to position the control relay to neutral and stop the flow of fluid when the governor steam valve has moved to the desired position.

There are two hydraulic fluids which are used for governor steam valve actuation:

(a) turbine lubricating oil, and
(b) a phosphate ester, fire resistant fluid.

Turbine lubricating oil has been used for many years as the hydraulic fluid for governor steam valve actuation. Lubricating oil is readily available and does not require a separate pumping system nor a separate purifying system, when used as a control fluid. In addition, if the lubricating oil system should fail, the governor steam valves will shut since it is lubricating oil which is holding them open. Lubricating oil has several disadvantages, although they were not serious enough to preclude its use as a control fluid:

1. The control fluid has to be compatible with bearing lubrication. That is, the fluid is chosen not for its properties as a control fluid but rather for its properties as a lubricant.

2. The lubricating oil is subject to all manner of contamination which tends to degrade its purity. These include: fibres, water, chlorides, dirt, rust and sludge.

The principle disadvantage of using lubricating oil, as a control fluid, did not become apparent until turbines became fairly large.

The larger the steam flow handled by turbines, the larger the governor steam valves had to be and the greater the force necessary to move the valves at the required speeds. Since Force = (Pressure) (Area), the larger force could be gained in one of two ways:

1. increase the pressure of the control oil, or

2. make the actuators larger.

The upper limit of control oil pressure is about 1,000 kPa(a). This is determined by the fire hazard involved in lubricating oil leaking from the system and soaking thermal insulation in contact with hot steam pipes. Since it is virtually impossible to prevent oil leaks in high pressure hydraulic systems, the only effective way to eliminate this fire hazard is to keep the lubricating oil under low pressure. In addition, since the flow through the control oil system is a very small percentage of the flow to the bearings, the use of high pressure control oil requires needlessly pressurizing a large volume of oil intended for the bearings.

The alternative method of developing a large force, that of making the actuator larger, also has its limitation. As the actuators get larger, the volume of oil which must be moved to effect a response becomes greater and greater. This "reservoir effect", as it is called, results in significant time delays between the initiation of a control signal and the response, as large quantities of oil are moved about the control oil system. The time between the initiation of signal and its execution is known as the dead time or time delay and in large turbine governing systems the combined effects of mechanical inertia and oil reservoir effect can result in significant dead time.

The governing system on the 540 MW Pickering NGSA turbine represents nearly the limit to which lubricating oil control systems can be pushed. Even in this system the exclusive use of lubricating oil for control valve actuators would have resulted in unacceptable dead times. As a result, while the governor steam valves and emergency stop valves have hydraulic actuators, the intercept valves and steam release valves are air operated. This eliminates the long runs of control oil piping which would be necessary to supply these valves with control oil. In addition, the control oil system at Pickering NGSA is equipped with anticipator devices, to decrease the response time of the control oil system. While the use of a lubricating oil control oil system, at Pickering NGSA has proved entirely acceptable the use of lubricating oil on larger units has not been possible.

The dead time associated with large turbines using lubricating oil control oil systems has been virtually eliminated by use of high pressure, hydraulic systems, using a fire resistant fluid. 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.

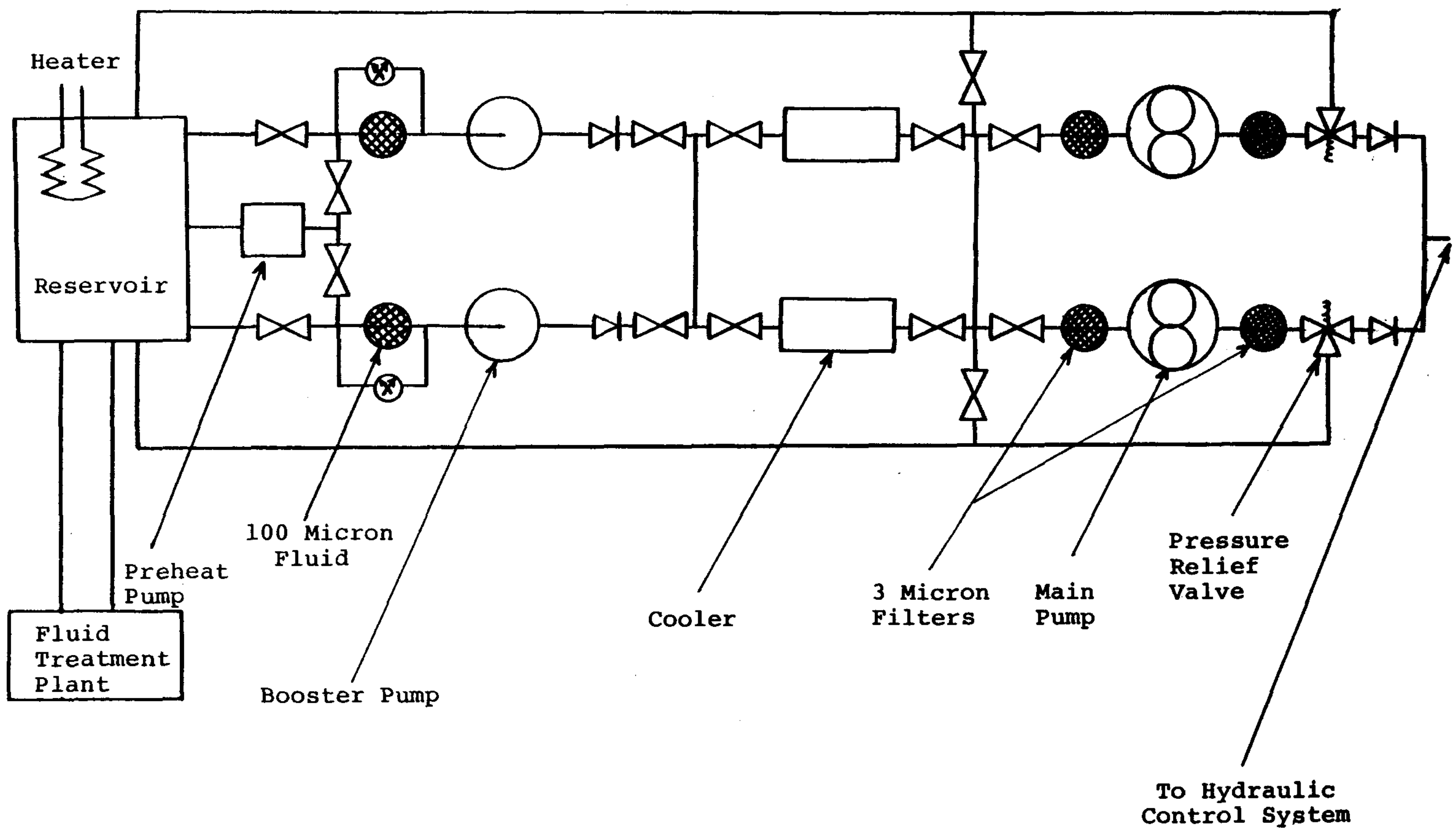
Figure 3.2 shows a typical FRF hydraulic power plant. The pumping system consists of two 100% duty centrifugal booster pumps which supply fluid at 680 kPa(g) to either of two 100% duty coolers and hence to either of two 100% duty positive displacement screw type main pumps which boost the pressure to 6800 kPa(g). Isolation valves permit maximum plant availability during on-load maintenance.

All oil, and particularly the synthetic, phosphate esters used in FRF systems, have a tendency to pick up impurities, notably grit, sludge, water and chlorides which tend to accelerate system corrosion and wear. Because of the use of small, high pressure components, with close tolerances, the cleanliness of FRF systems is particularly critical.

A 100 micron suction strainer is located in each centrifugal pump suction line. Full flow three micron absolute disposable element filters, fitted with differential pressure switches and internal by-passes, are located upstream and downstream of each main pump. This arrangement ensures all the fluid is filtered before entering the main pump and before entering the main hydraulic system. In addition, the fluid in the reservoir is continuously purified of moisture and acid via a fluid treatment plant which includes a vacuum dehydrator and a fuller's earth filtration plant.

Immersion heaters within the fluid reservoir are used to warm the fluid during a cold start. A low capacity, low pressure pump circulates the fluid through the coolers and returns it to the reservoir. When the fluid temperature has reached a safe minimum the centrifugal and main pumps may be started.

By using a separate control hydraulic system it is possible to maintain a much higher standard of fluid control, regarding temperature, viscosity and purity. It is not subject to variation in temperature to suit bearing requirements or seal oil requirements, and it is not subject to metallic pick up from bearings or moisture pick up from steam glands. However, 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



FRF Power Plant

Figure 3.2

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.

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. From 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 safely 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 this reason disposal procedures for this fluid should be carefully adhered to.

ASSIGNMENT

1. Define "dead time" and "reservoir effect".
2. Explain why there is a limit to the size to which turbines with lubricating oil fluid control systems can be built.
3. Why is hydraulic oil used for governor steam valve actuation?
4. What are the advantages of an FRF fluid control system over a lubricating oil control system?
5. What are the problems associated with an FRF control system?
6. Discuss the need for fluid purity and cleanliness in an FRF control system.

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 234

GOVERNOR OPERATION

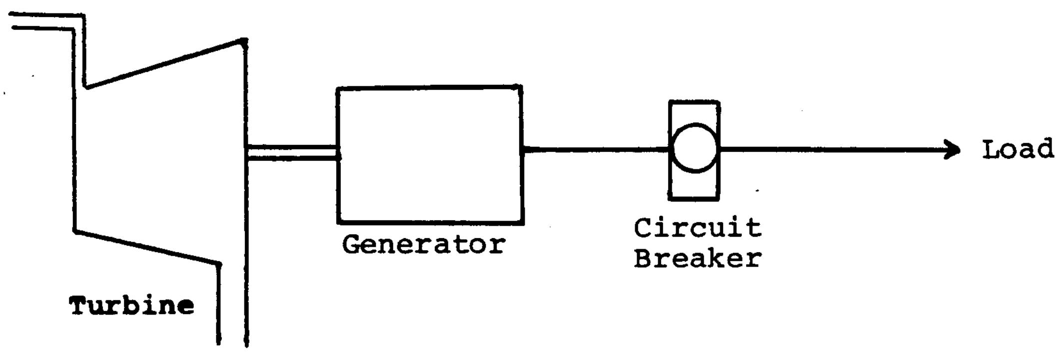


Figure 4.1

Figure 4.1 shows a single turbine driving an ac generator which is supplying a load. Heat energy in the steam passing through the turbine is converted into shaft mechanical power. In the generator this shaft mechanical power is converted to electrical power which then flows to the load where this power is consumed. The turbine is producing a mechanical torque on the shaft. If something were not producing a countertorque of equal magnitude, the rotational speed on the shaft would be increasing. As it turns out the generator, in the process of generating electrical power, exerts this countertorque on the shaft. The countertorque developed by the generator is directly proportional to two quantities:

1. the magnetic field flux of the generator (ϕ), and

2. the load current flowing out of the generator to the load (I),

$$\text{or } T = K\phi I$$

where K = constant of proportionality.

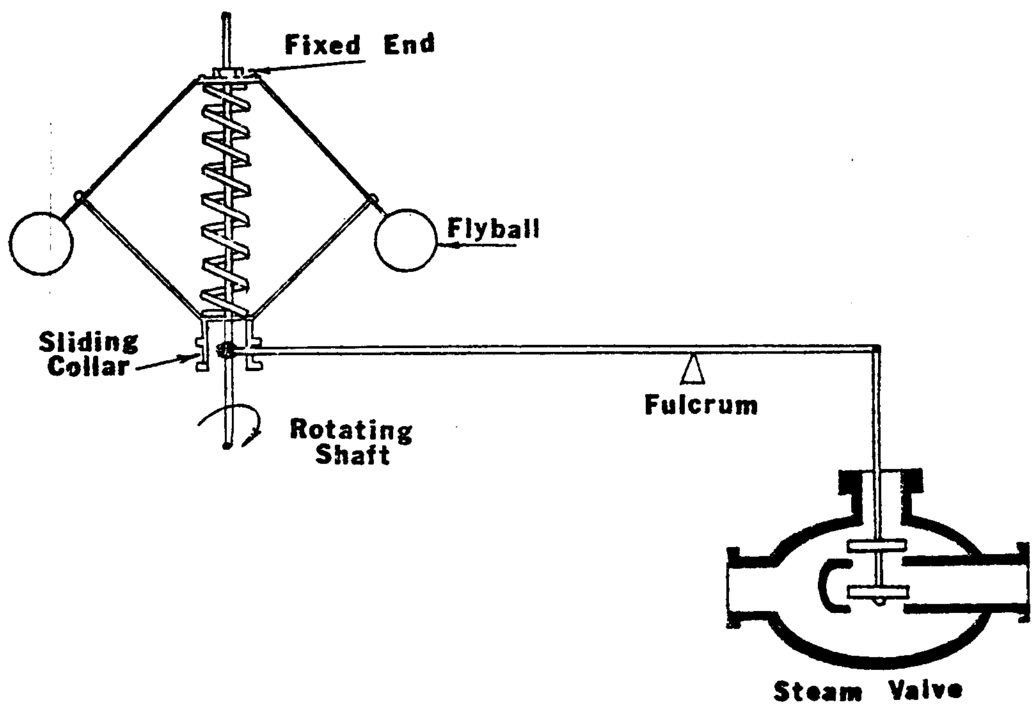
At a constant speed, the mechanical torque produced by steam flowing through the turbine is exactly balanced by this countertorque of the generator.

If the load current flowing from the generator is increased because additional load equipment is energized, the countertorque obviously increases.

$$\overset{\uparrow}{T} = K\overset{\rightarrow}{\phi}\overset{\uparrow}{I}$$

This creates an imbalance between turbine torque and generator countertorque (countertorque is now greater than turbine torque) and the turbine begins to slow down. Clearly, if the turbine is to return to its original speed the turbine torque must be increased by increasing the steam flow to the turbine.

This problem is partially alleviated through the use of the simple governor, shown in Figure 4.2.



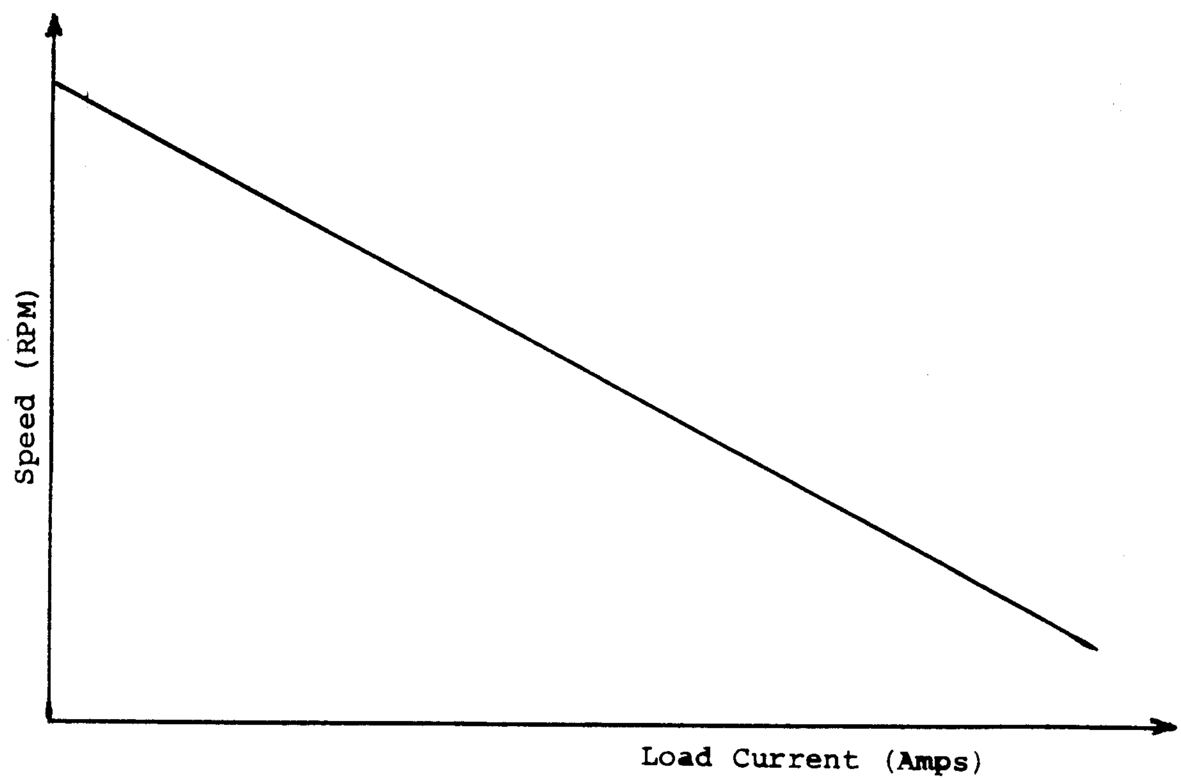
Simple Flyball Governor

Figure 4.2

The flyballs in the governor are driven off the turbine shaft. As turbine speed increases, the centrifugal force acting on the flyballs throws them out against spring tension. As turbine speed decreases, the centrifugal force decreases and the spring pulls the flyballs together. The flyballs are connected to a collar which is free to move. As the flyballs move out (indicating the turbine is speeding up), the collar moves to the left and, acting through the lever, shuts the governor valve. As the turbine speed decreases, the flyballs are pulled in, the collar moves to the right and the governor valve opens, admitting more steam.

With this type of simple flyball governor, the turbine can partially compensate for an increase in generator load. If the load current increases, generator torque ($T = K\phi I$) increases and the turbine slows down. As this happens, the flyballs move in, open the governor steam valve and admit more steam. This increases turbine torque until it is equal to generator counter-torque. At this point speed stabilizes and the flyballs assume a constant position (spring tension balanced by centrifugal force).

The disadvantage of this simple flyball governor is that there is only one speed (flyball position) corresponding to a particular governor valve position. The result is that to increase the steam flow to the turbine, the speed must decrease. If the speed returned to the initial value, the flyballs would return to their initial position. If this occurred, the governor valve would return to its initial position and the steam flow would return to the initial value. Thus, with a simple flyball governor, a particular load can only be achieved at a particular speed. This is shown in Figure 4.3.



Load Versus Speed for a Simple Flyball Governor

Figure 4.3

Since generator output frequency is directly proportional to turbine/generator RPM, this characteristic causes several problems. If the output frequency is not held constant, motors will run at non-constant speeds, load impedances will change and load currents and voltages will vary. For any reliable, practical load distribution system, a constant frequency is essential.

It is possible to construct a constant speed governor which, although somewhat more complicated than the simple flyball governor, overcomes the disadvantage of having an output frequency which varies with generator load current. With the constant speed or isochronous governor the response to a load increase is as follows:

1. load current increases,
2. generator countertorque increases,
3. turbine/generator slows down,
4. governor senses speed drop (flyballs move in),
5. governor opens upon the governor steam valve,
6. more steam is admitted to the turbine,
7. speed returns to the initial speed with turbine power equal to the new generator power.

With a decrease in load, the governor responds in the opposite way:

1. load current decreases,
2. generator countertorque decreases,
3. turbine/generator speeds up,
4. governor senses speed increase (flyballs move out),
5. governor shuts down on the governor steam valve,
6. less steam is admitted to the turbine,
7. speed returns to the initial speed with turbine power equal to the new generator power.

Of course, the Ontario Hydro electrical grid is not individual turbine/generators supplying their own loads, but rather a large number of generators operating in parallel with each other. The operation of many turbine/generators operating in parallel makes the governing somewhat more complex.

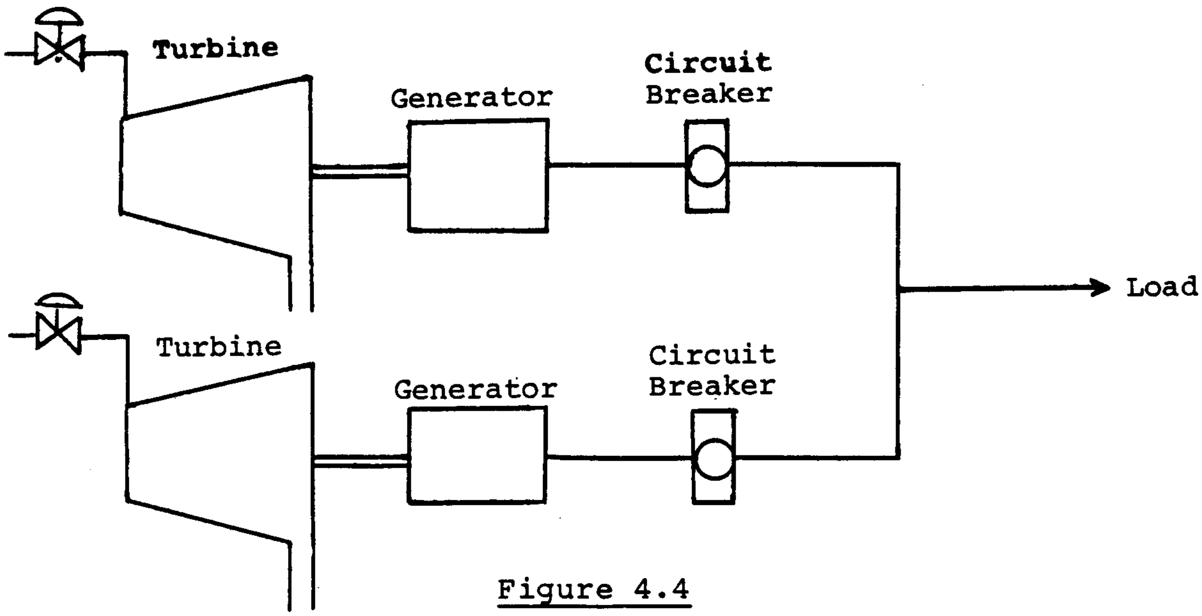


Figure 4.4

Figure 4.4 shows two ac generators operating in parallel supplying a load.

When two or more generators are electrically connected they must operate at the same frequency. If one speeds up, the others must speed up with it; they cannot do otherwise as long as they remain electrically connected. Under normal circumstances, two generators in parallel can no more operate at different frequencies than the wheels of a car can turn at different speeds. In Figure 4.4, the two generators are operating at the same frequency supplying the load. They share the real electrical load in proportion to the turbine mechanical shaft power of each turbine.

If the load increases, the countertorque ($T = K\phi I$) of both generators increases and both generators slow down. Since they are operating in parallel, the two generators must maintain equal frequencies and both slow down equally. Both governors will sense the speed decrease and begin opening up on their respective governor steam valves to return the speed to its initial value. However, if one governor is faster acting it will increase the steam supply to its turbine faster than the other governor will.

Since the generator frequencies must be equal, as the faster-acting governor returns its turbine to the initial speed, its generator will literally "pull" the speed of the other generator up with it. In addition, the turbine with the faster acting governor will pick up a disproportionate share of the load increase. In fact, when two turbine/

generators with constant speed governors are operating in parallel, there is no guarantee that one generator will not pick up all of the load increase. If the total electrical load decreases, the reverse will happen and the faster acting governor will shed its load faster. This is clearly an unstable situation. Although the generators will always return to their initial speed (both have constant speed governors), there will be a constant shifting of real electrical loads between the generators. If many generators were operating in parallel, this constant shifting of real loads between generators would be comical if it were not so hazardous to the stability of the grid.

To prevent this from happening, governors are not designed to be constant speed from no load to full load. Rather each governor is designed to have a small speed droop. That is, as the load is increased, the governor will not return the turbine speed to its initial value but rather to a speed slightly below the initial value. This speed droop is a design characteristic of the governor and is typically between 1% and 5% of full load speed.

$$\text{Speed Droop} = \frac{N_{FL} - N_{NL}}{N_{FL}}$$

where: N_{FL} = full load speed

N_{NL} = no load speed

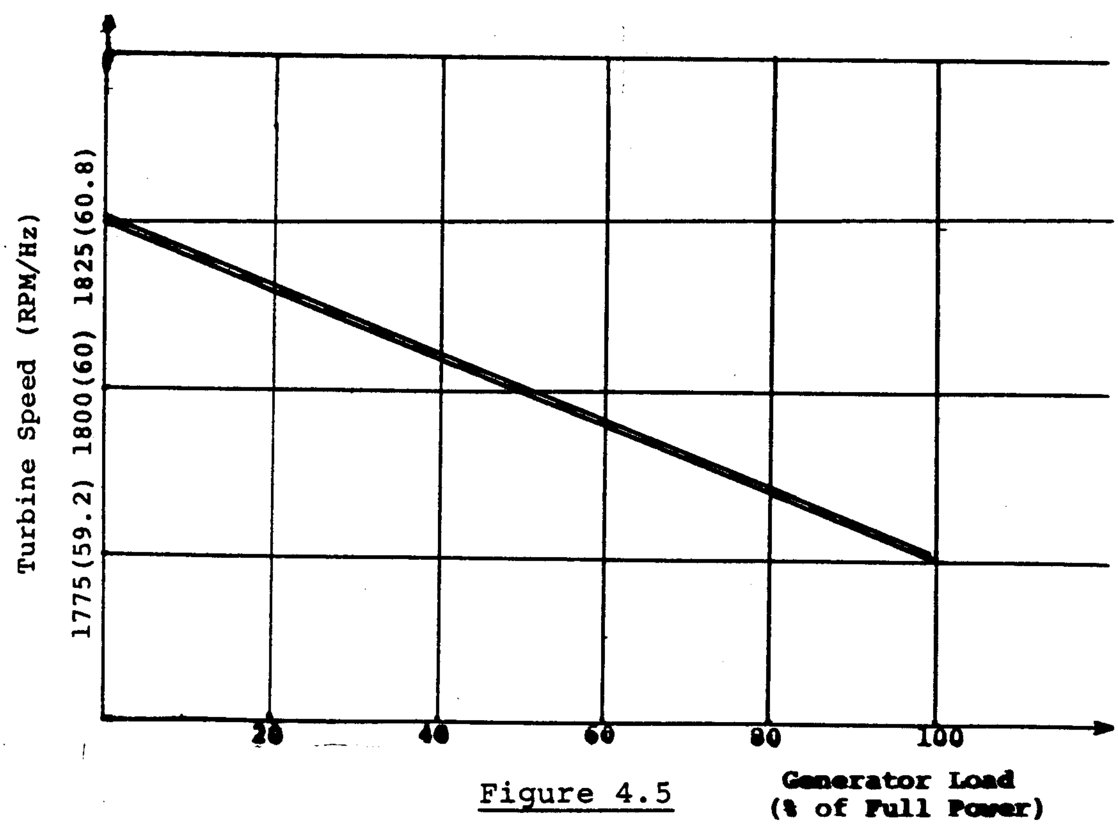


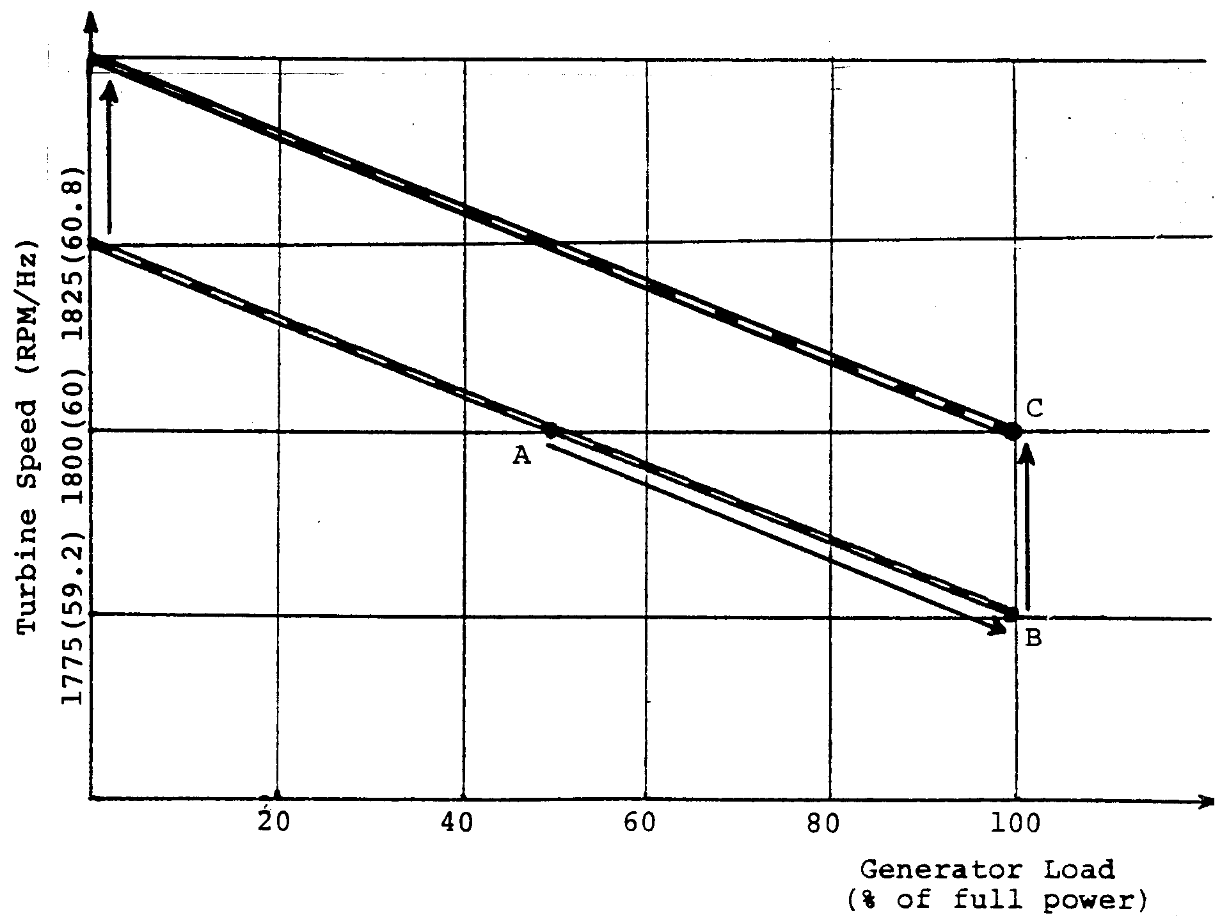
Figure 4.5

Figure 4.5 shows a typical speed versus load curve for a turbine with a governor having speed droop. Consider such a turbine individually supplying a load. If the load increases, the governor will increase the steam supply to meet the additional power demand; however, when speed is stabilized, it will be at a slightly lower speed than it was initially. It would appear at first glance that we are back to the problem of the simple flyball governor. However, the governor is constructed to allow the raising or lowering of the speed droop curve through a device called a <u>speeder gear</u>. The effect of a load increase is shown in Figure 4.6 as follows:

(a) load increases from 50% to 100% of full load,

(b) the governor restores the speed to that dictated by the speed droop curve or 1775 RPM (A to B),

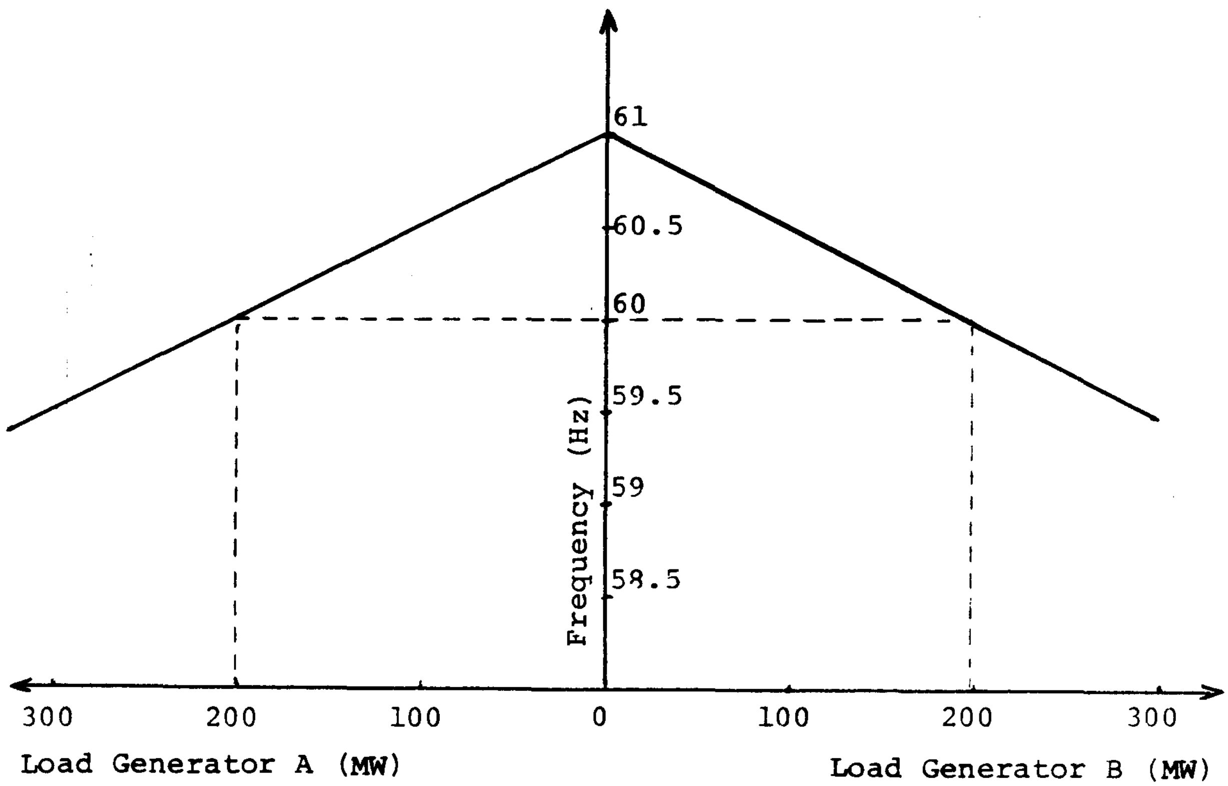
(c) through the speeder gear the speed droop curve is raised (either by the operator or automatically) so that the turbine supplies 100% of full load at 1800 RPM (B to C).



Effect of Load Change

Figure 4.6

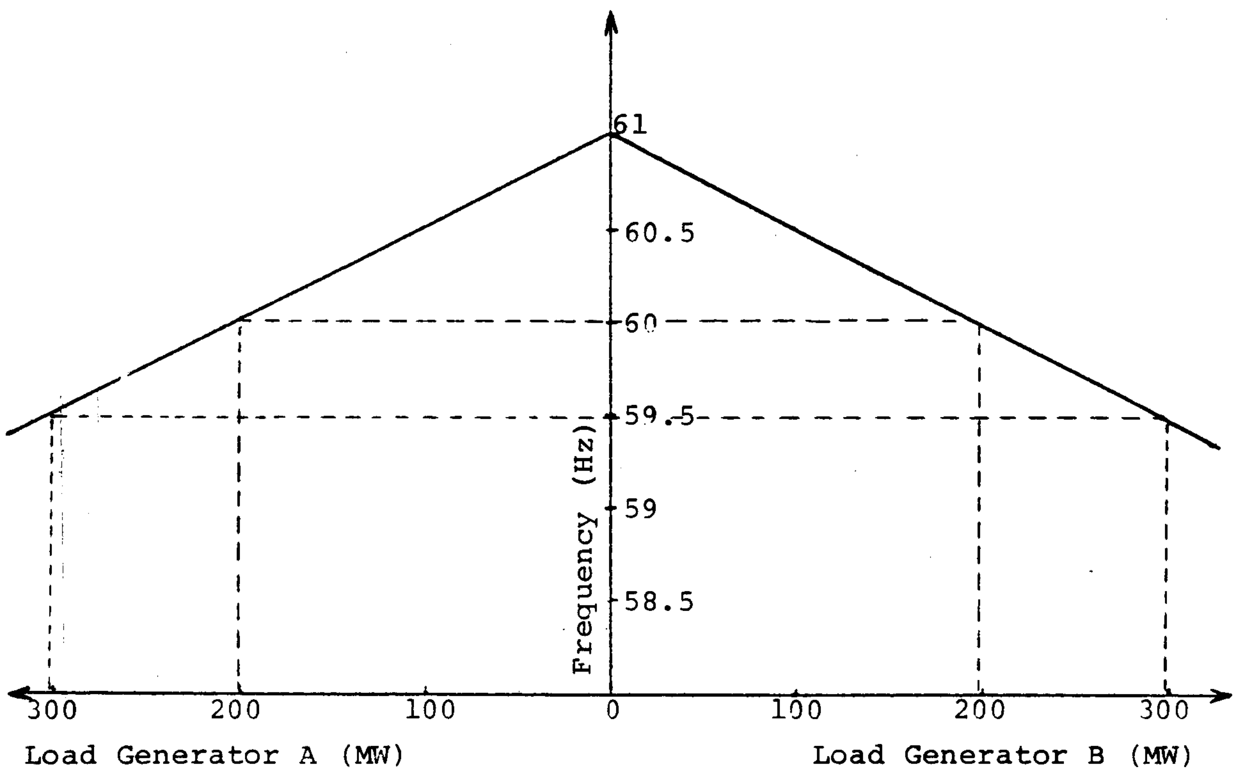
It should be noted that to change the operating point from B to C, what was done in effect was raise the no-load speed from 1825 to 1850 rpm (no-load frequency from 60.8 to 61.6 Hz). Although the governor senses and controls turbine speed, since turbine speed is directly proportional to frequency, it is common to visualize the governor as controlling frequency. When generators are operating in parallel the speed droop of their respective turbine governors eliminates the instability caused by the constant speed governor. Figure 4.7 shows the speed droop curves for two turbine generators equally sharing a 400 MW load at 60 Hz.



Two Turbine/Generator Sharing a 400 MW Load

Figure 4.7

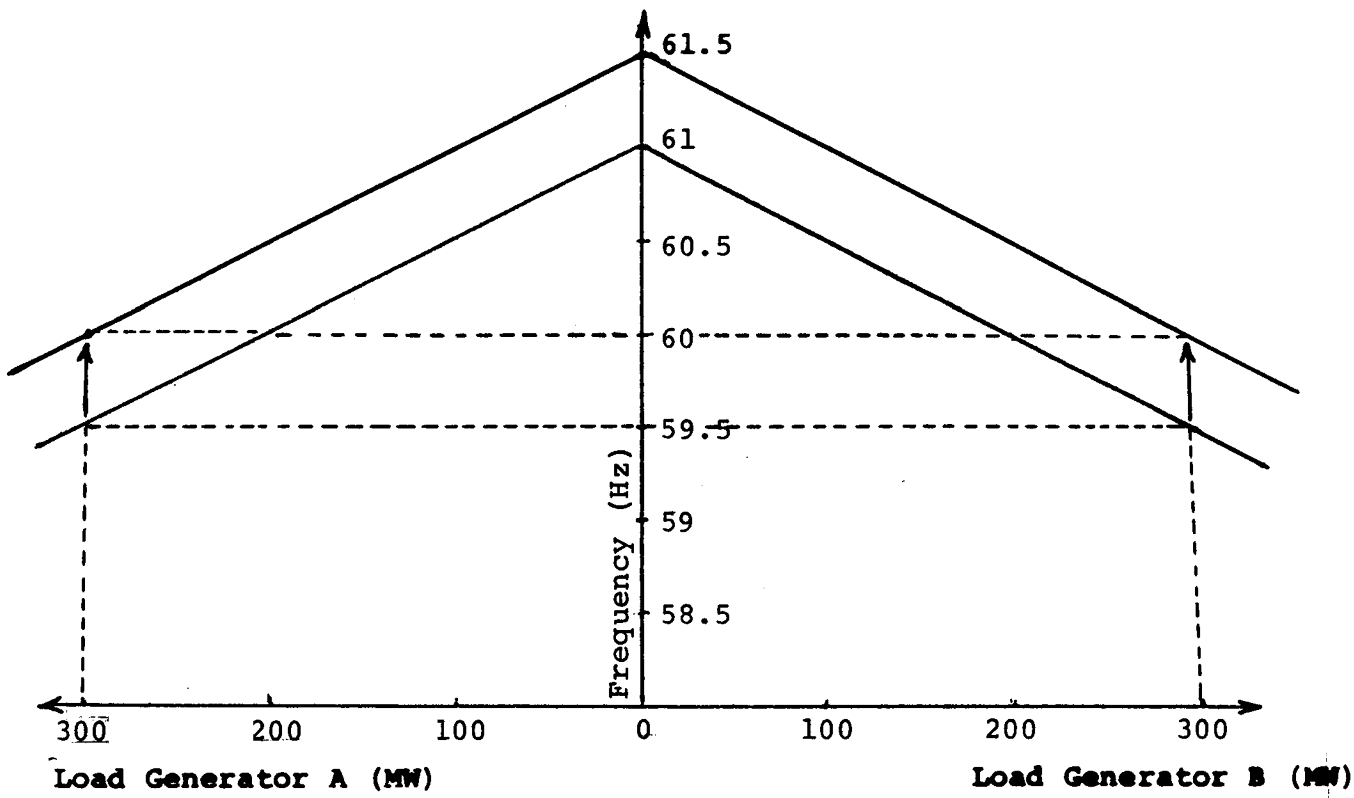
The total load is increased to 600 MW by energizing more loads. Recalling that the two generators must remain at the same frequency, you will note that there is only one point at which these two generators can share 600 MW and that is at a frequency of 59.5 Hz. Furthermore at 59.5 Hz the generators must be equally sharing the load.



Load Increase from 400 to 600 MW

Figure 4.8

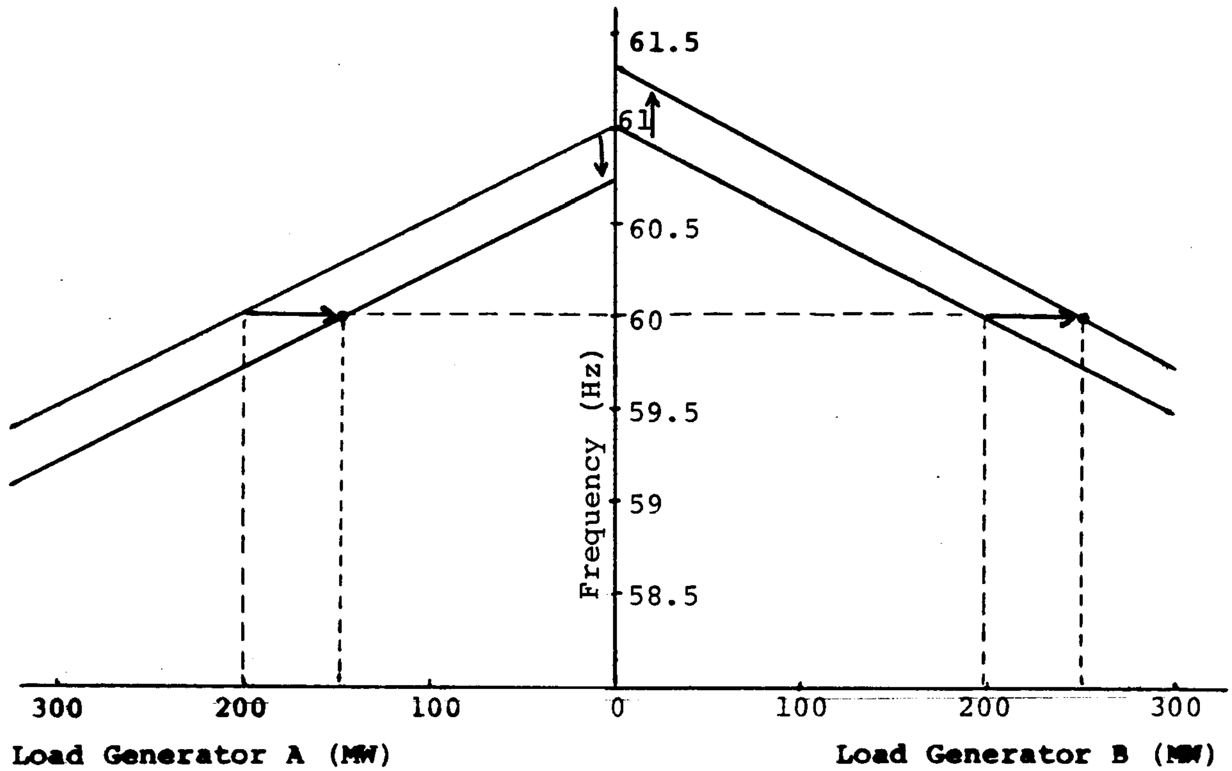
To return the output frequency to 60 Hz the speeder gear would be used to raise both speed droop curves so that the two generators can supply 300 MW each at 60 Hz. To accomplish this the speeder gear increased the no-load speed from 61 Hz to 61.5 Hz (Figure 4.9).



Restoration of Frequency Following a Load Change

Figure 4.9

With the two units operating in parallel equally sharing a 400 MW load at 60 Hz, suppose it is desired to change the load distribution between the two generators. The load of generator A is to be lowered from 200 MW to 150 MW, while the load of generator B is to be increased from 200 MW to 250 MW. Figure 4.10 shows how this is accomplished. The speed droop curve of generator A is lowered, which means for a given frequency (60 Hz) the turbine will call for less steam and the generator load will drop. Simultaneously, the speed droop curve of generator B is raised, which implies that the turbine will call for more steam and generator load will increase. Placing the speeder gear of generator A in the "lower speed" position lowered the no-load speed and lowered the real generator output. Placing the speeder gear of generator B in the "raise speed" position raised the no-load speed and raised the real generator output. However, since both generators were in parallel they could not operate at different frequencies and, provided the operator is careful when he redistributes the load, the common frequency will be 60 Hz.



Varying Load Distribution

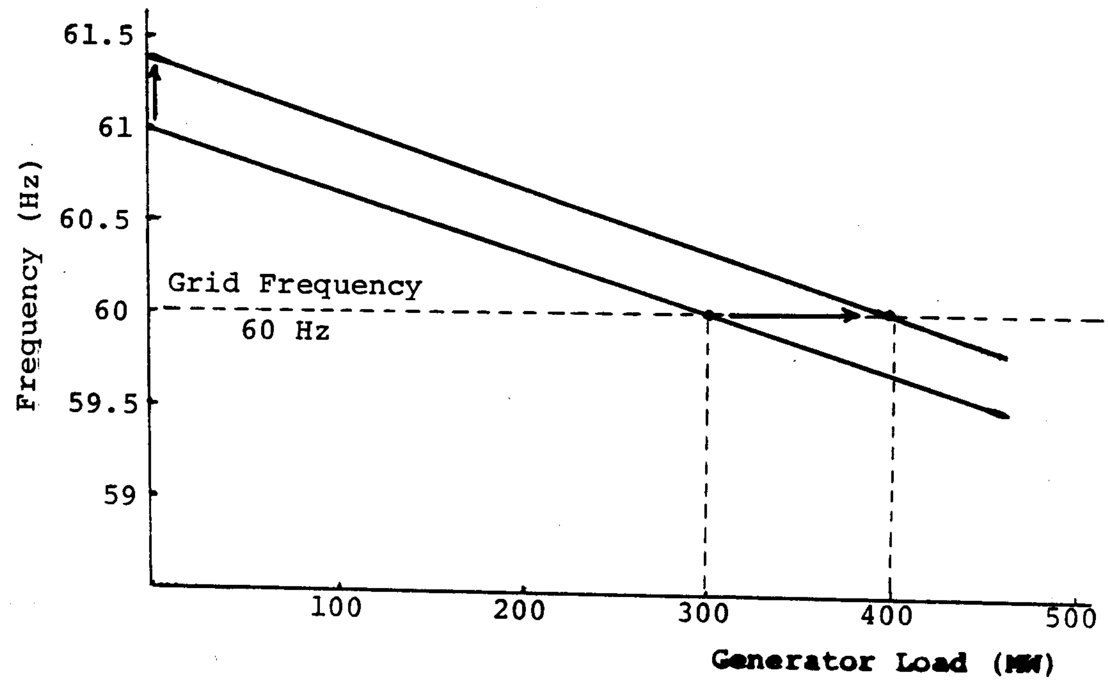
Figure 4.10

The operation of a turbine generator in parallel with the hydro grid is quite similar to the preceding example with a couple of notable exceptions:

1. The hydro grid is large enough that during normal operation the frequency of the grid is constant for all practical purposes. That is when the grid is functioning normally, the frequency cannot be changed by a single generator.

2. The decrease in frequency of the grid for an additional load being energized is much too small to be seen by a governor. That is when the little old lady in Maynooth turns on her stove, the drop in grid frequency (say from 60 Hz to 59.9999999 Hz) is too small to be seen by the governors in the generating stations and the generators cannot automatically pick up this load.

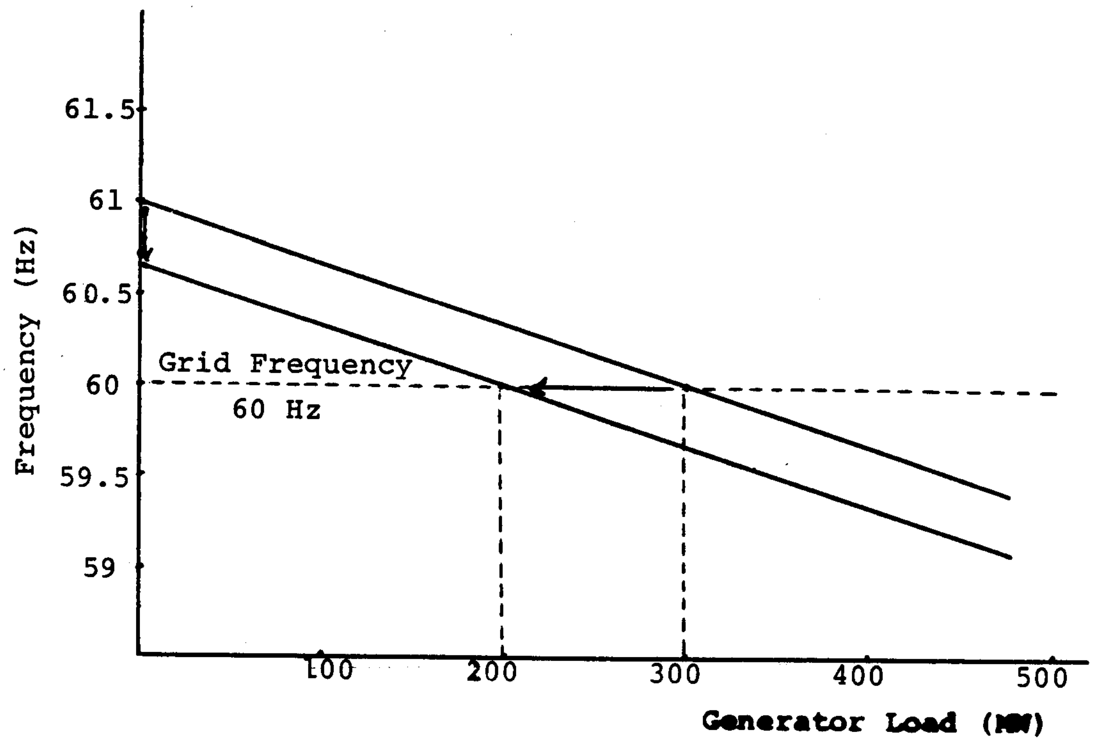
A generator supplying power to the Hydro grid can neither change the grid frequency to any appreciable extent nor sense minor changes in grid frequency. A control centre which can detect small frequency changes must be used to direct turbine generator loading and unloading.



Increasing Generator Load

Figure 4.11

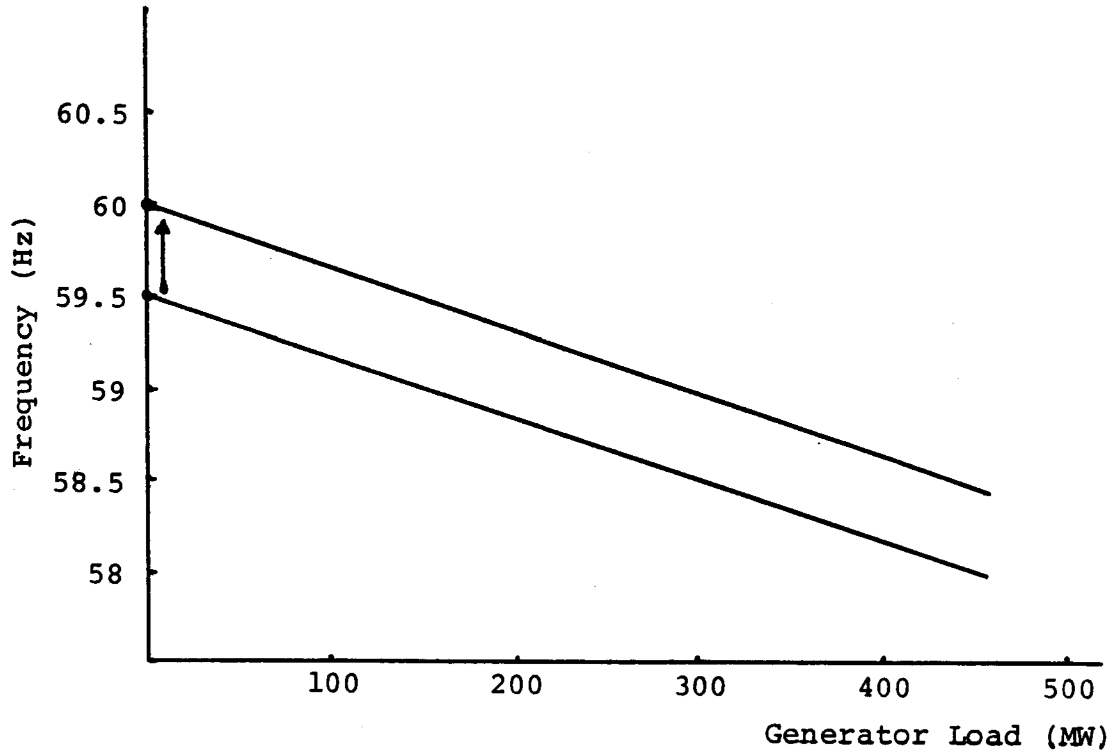
To increase the output of a turbine generator, the speeder gear is moved in the "raise speed" direction, the speed droop curve is raised and the load increases while the frequency remains at 60 Hz (Figure 4.11). To decrease the output of a turbine generator, the speeder gear is moved in the "lower speed" direction, the speed droop curve is lowered and the load decreases while the frequency remains at 60 Hz (Figure 4.12).



Decreasing Generator Load

Figure 4.12

When the turbine generator is not connected to the grid and is supplying no load current, there is no generator countertorque ($T = K\phi I$) to balance the torque produced by the steam in the turbine. The only force acting against the turbine torque is the friction and windage of the turbine and generator. If the speeder gear is moved in the "raise speed" direction when the generator is not synchronized to the grid, the speed droop curve will be shifted upward and the turbine will respond to match the speed determined by the new curve (Figure 4.13). Since the generator is carrying no load, the new, higher speed will be the no-load speed for the new speed droop curve. This higher speed will require a greater steam flow since the friction and windage losses are increased. It should be appreciated, however, that the friction and windage losses are quite small when compared with generator maximum load, and the amount of steam required to raise the turbine speed when the unit is unloaded and unsynchronized is quite small.



Increasing Speed on Generator
Which is not Synchronized to Grid

Figure 4.13

Under normal conditions, the function of the governor can be summarized as being to keep the turbine/generator at the proper load and frequency as determined by the speed droop curve. The operator may change the operative point by moving the speed droop curve. It is then the role of the governor to insure the unit stays there.

As an example of what has been said in this lesson, we will consider the action of the governor during startup, loading and a load rejection.

Startup

You will recall that when the generator is not connected to the grid, the governor will adjust the steam flow to maintain the speed corresponding to the no-load point on the speed droop curve. However, the speeder gear by design is unable to lower the speed droop curve below a no-load speed of about 1650 - 1700 rpm for an 1800 rpm machine. When the speed is below 1650 rpm the governor is trying to bring the speed back up to 1650 rpm, thus the governor steam valves are fully open.

The turbine, therefore, is brought up to 1650 rpm by throttling steam flow with the emergency stop valves. As the turbine speed approaches 1650 rpm, the governor steam valves gradually close until at 1650 rpm they should be passing only enough steam to control the speed at that value. When this happens we speak of the turbine as "coming on the governor" because the governor is capable of controlling the speed at the no-load speed of the speed droop curve. At this point the emergency stop valves can be fully opened.

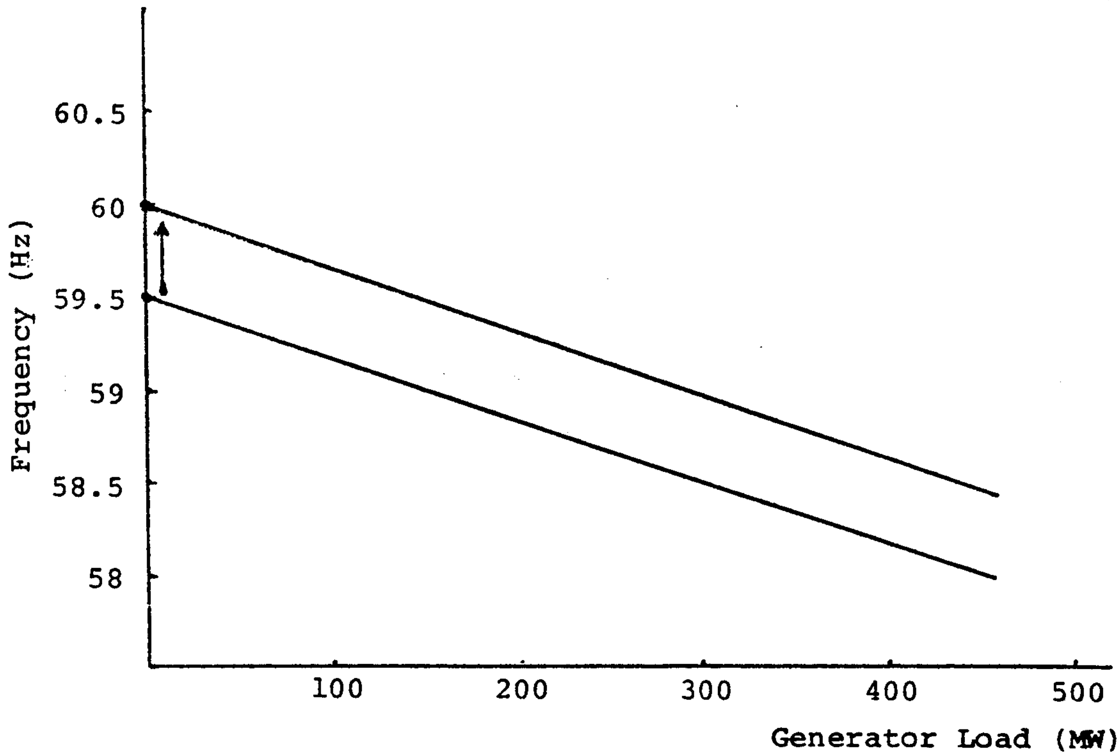
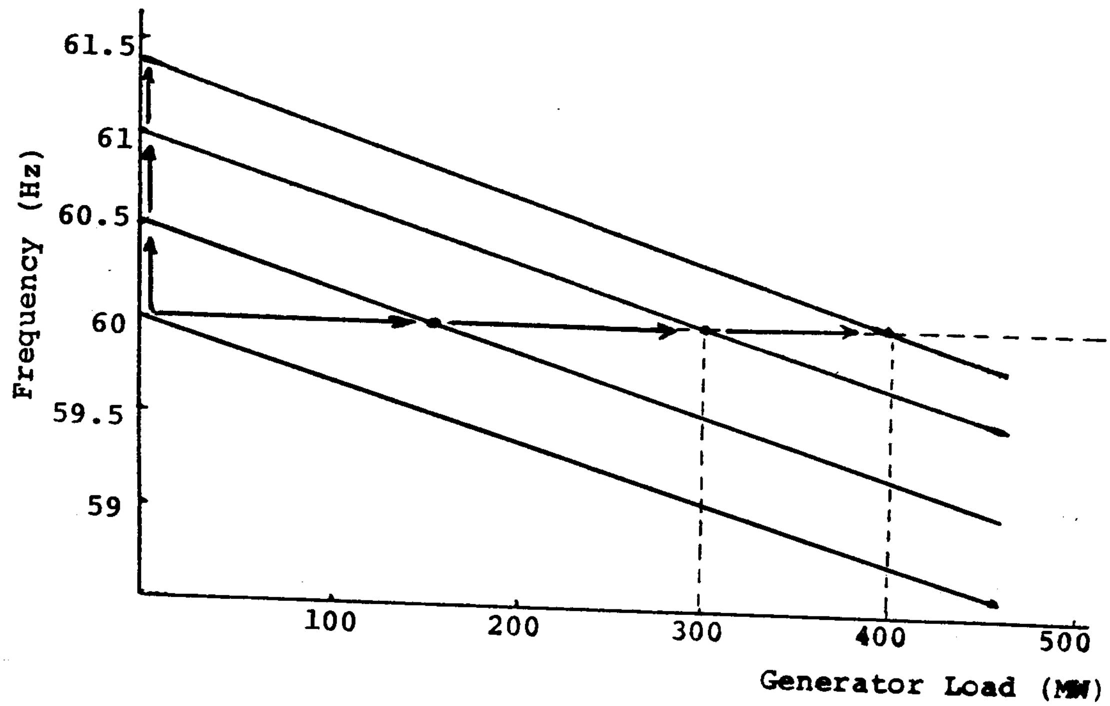


Figure 4.14

The operator can bring the turbine up to 1800 rpm (60 Hz) by moving the speeder gear in the "raise speed" direction. This raises the speed droop curve and the governor admits more steam to the turbine to keep turbine speed equal to the no-load speed of the speed droop curve (Figure 4.14).

Loading

At 1800 rpm, the generator is synchronized to the Hydro grid. Once this has been done, the generator can run at no speed other than 60 Hz (1800 rpm). Regardless of what the operator does with the speeder gear (and therefore the speed droop curve) the turbine and generator must run at 60 Hz. Therefore, as the speeder gear is moved in the "raise speed" direction the speed droop curve will move up but the net effect will be an increase in steam flow and, therefore, an increase in load (Figure 4.15).



Increasing Load on a Synchronized Turbine/Generator

Figure 4.15

Load Rejection

If the generator output breaker trips open due to an electrical fault, the speed will start to rise and the governor will start to close the governor steam valves. However, because of the speed droop of the governor, the governor will attempt to control speed at the no-load value which may be 5% above the normal operating speed. This means the governor has a built-in bias which works against holding speed down on a load rejection. Even though the speeder gear is driven back to the position corresponding to a no-load speed of 1800 rpm, it cannot move fast enough to eliminate the effect of speed droop. In later lessons we will discuss how this problem is handled.

ASSIGNMENT

1. What is the disadvantage of a simply flyball governor?

2. What is the disadvantage of an isochronous (constant speed) governor?

3. What is speed droop? Why are governors designed with speed droop?

4. How can a governor with a 5% speed droop allow a turbine generator to go from 0% to 100% of full load at 60 Hz?

5. A single turbine is supplying a 60 MW load. An additional load of 10 MW is energized. Explain how the turbine responds to supply this load.

6. Why can't the Pickering NGS turbines respond to an additional 10 MW load being energized from the grid?

7. How is the speed increased on a 540 MW turbine/generator which is unconnected from the grid?

8. How is the output load of a 750 MW turbine/generator which is supplying the Hydro grid increased?

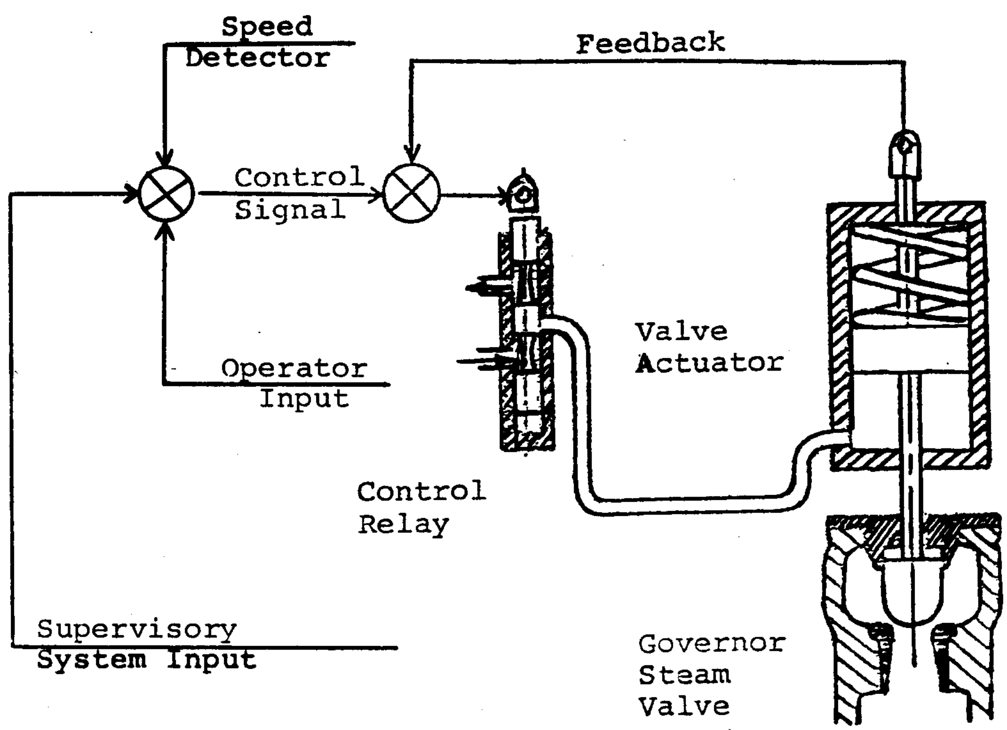
9. Why is the governor biased in the wrong direction to handle an overspeed following a load rejection?

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 234

TURBINE GOVERNORS

We have seen in the preceding lesson how a basic governing system must function to control the steam supply to a turbine. This lesson will concentrate on the method by which two governing systems achieve this control. The two governing systems we will examine are the mechanical governor (used on all NGD turbines up through Pickering NGS-A) and the electrical governor (used on Bruce NGS-A and subsequent units).



Basic Governing System

Figure 5.1

Whatever the type of governor, it must meet certain basic requirement:

(a) There must be a method for the operator to vary the "no-load speed" of the turbine (this enables the operator to shift the position of the governor speed droop curve);

(b) There must be a speed sensor;

(c) There must be a control relay to admit hydraulic fluid to the governor steam valve power pistons;

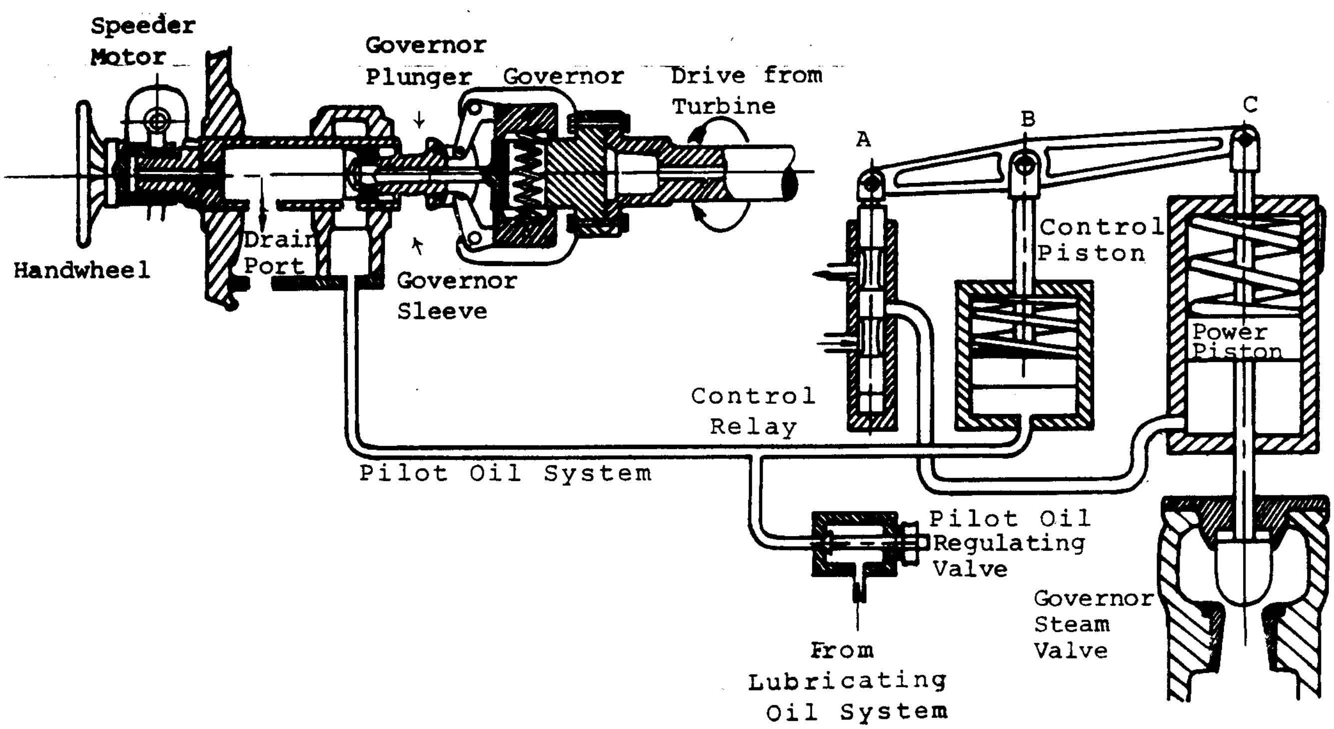
(d) There must be a method of feedback from the governor steam valve position to the control relay;

(e) There must be a method of rapidly closing the governor steam valves to shut off steam in the event the generator load is lost;

(f) There must be a method of shutting the governor steam valves during certain casualties.

The relationship of these components is shown in Figure 5.1.

Mechanical Governor



Basic Mechanical - Hydraulic Governing System

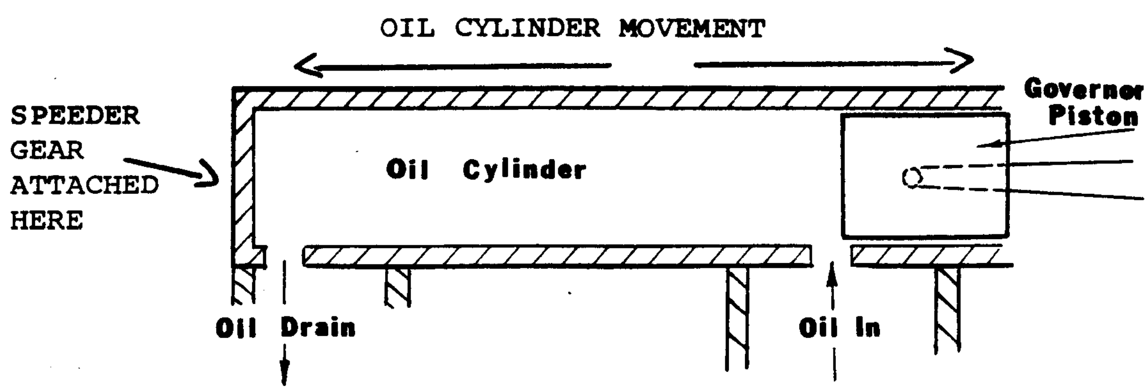
Figure 5.2

Basic Mechanical Hydraulic Governor

Figure 5.2 shows a basic mechanical governing system. Control oil is supplied at a constant rate through a Pilot Oil Regulating Valve and discharged at a variable rate through the Governor Oil Cylinder. The pressure in this Pilot Oil System is thus a function of the rate at which pilot oil is drained from the system. Depending on the opening in the governor oil cylinder, the pilot oil pressure may vary from about 200 KPa(g) to about 600 KPa(g). It is the value of this pilot oil pressure which is the control signal for governor steam valve operation.

If pilot oil pressure increases, oil pressure forces the Control Piston up against spring tension. The lever pivots on the top of the power piston (C) and pulls up the spool of the control relay. This admits Power Oil to the underside of the power piston and forces it up against spring tension. The lever now pivots on the top of the control piston (B) and pushes the spool of the control relay back to the neutral position, thus, shutting off power oil to the power piston. Therefore, the system regains equilibrium, with the governor steam valve more fully open and passing more steam.

The reverse occurs when pilot oil pressure is reduced. As oil pressure under the control piston decreases, spring tension pushes the piston down. The level pivots on the top on the power piston (C) and pushes down the spool of the control relay. This allows power oil to drain from the underside of the power piston and the power piston moves down. The lever now pivots on the top of the control piston (B) and pulls the spool of the control relay back to the neutral position, thus, shutting off the drain of oil from the power piston. Therefore, the system regains equilibrium, with the governor steam valve less fully open and passing less steam.



Governor Piston and Cylinder

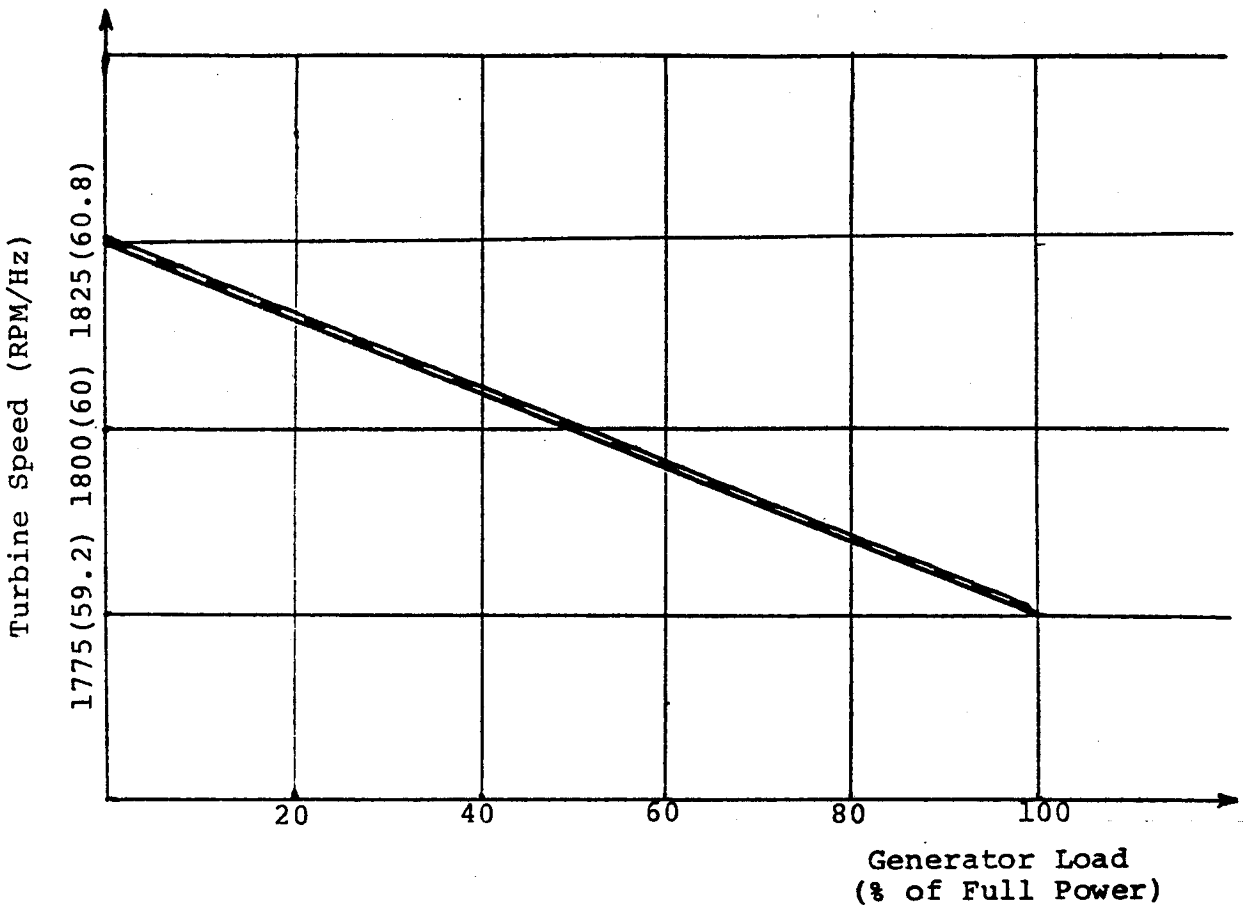
Figure 5.3

We will now turn our attention to how the governor varies pilot oil pressure. The method by which the mechanical governor senses turbine speed is by the position of the Flyballs. The flyballs are connected to the Governor Piston. As the governor piston moves, it varies the size of the opening from the pilot oil system to the governor oil cylinder (Figure 5.3). If the flyballs move out (speed increasing), the piston is withdrawn, more oil drains out, the pilot oil pressure decreases, and the governor steam valve moves in the shut direction. If the flyballs move in (speed decreasing), the piston is inserted further into the cylinder, less oil drains out, the pilot oil pressure increases, and the governor steam valve moves in the open direction.

In addition to the effect of speed on pilot oil pressure (via the flyballs and governor piston), the operator can vary pilot oil pressure through the position of the governor sleeve. The governor sleeve is moved by the Speeder Gear which is powered by an electric motor. A control signal (initiated by the operator or the computer) drives the motor to move the governor sleeve. This increases or decreases the size of the drain port, thereby varying pilot oil pressure.

It is important to keep in mind that all the governor can do is vary pilot oil pressure, and that all varying pilot oil pressure can do is open or shut the governor steam valve. The effect produced by varying governor steam valve position (varying steam flow) is dependent on whether the generator is synchronized or not synchronized to the grid, and has nothing to do with the design of the governor. For example, raising pilot oil pressure will open up on the governor steam valves and admit more steam to the turbine. Whether this increases speed or increases load has nothing to do with the governor but is a function of the external operating conditions.

You will recall from our discussion of governor operation in the last lesson that speed droop is a function of the design of the governor. Figure 5.4 shows a speed droop curve for a typical large turbine generator.



Speed Droop Curve

Figure 5.4

Consider the case of a turbine generator with this speed droop curve supplying a load. As the load is increased, the hydro-mechanical governor will respond as follows:

(a) load increases (current through generator armature increases);

(b) generator countertorque ($T = K\phi I$) increases;

(c) turbine/generator slows down;

(d) the flyballs move in (centrifugal force less than spring tension);

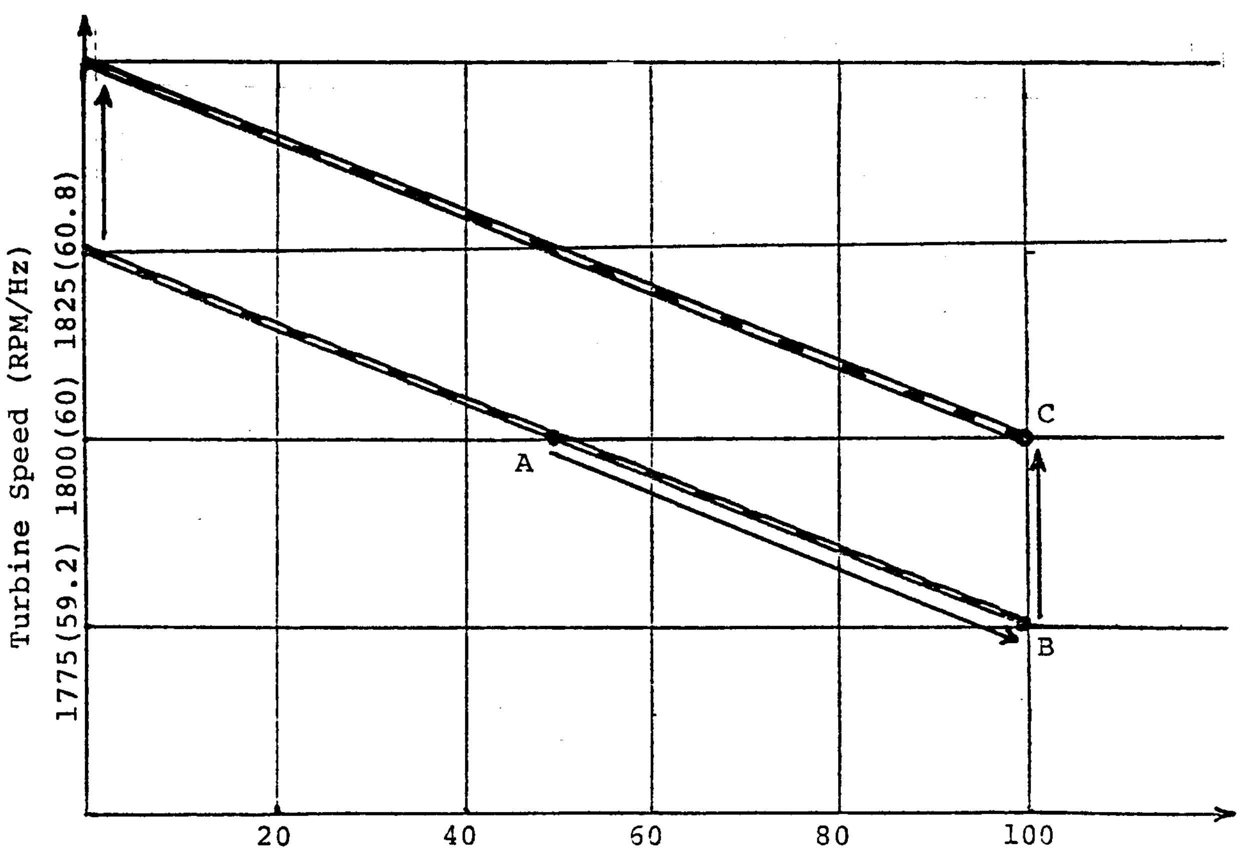
(e) the governor plunger is inserted into the governor sleeve, reducing the rate of drain from the pilot oil system;

(f) pilot oil pressure increases;

(g) the governor steam valve opens further, admitting more steam.

The turbine is now supplying the increased load. However, the speed does not return to the original value rather some lower value. This is, of course, speed droop and is a function of relative value of the spring force and flyball centrifugal force.

It is, however, desirable to keep the turbine/generator operating at a constant frequency. In order to restore the speed of the turbine (frequency) to its original value, the speeder gear is moved in the "raise speed" direction. This moves the governor sleeve to close off the drain port, increasing pilot oil pressure and opening up on the governor steam valves to supply the load at the original speed (frequency). This is shown in Figure 5.5.



Effect of Load Change

Figure 5.5

If a turbine generator with this speed droop is supplying the Ontario grid system, the grid frequency will not vary substantially from 60 Hz. This means that, under normal conditions, the speed of the turbine/generator will remain substantially at 1800 rpm for a 4 pole generator.

In this case (generator synchronized to an infinite bus), the effect of increasing steam flow is to increase the power out of the generator. Since speed cannot change, if more steam power is put in, more must be supplied by the generator.

Load Rejection

If the generator output breaker trips open due to an electrical fault, the speed of the turbine will start to increase. The flyballs will be thrown outward, withdrawing the governor piston from the governor oil cylinder. This dumps pilot oil to drain. As pilot oil pressure is reduced, the control piston is pushed down under spring tension. This lowers the spool of the control relay and dumps power oil to drain. This shuts the governor steam valve.

The mechanical hydraulic governor has two inherent weaknesses which decreases its ability to handle an overspeed following load rejection:

1. The dead time associated with the reservoir effect of low pressure lubricating oil. The time necessary to move the large volume of pilot oil and power oil to shut the governor steam valves, results in excessive overspeeds.

2. When the governor finally gets control of turbine speed, it will attempt to control speed at the no load speed determined by the speed droop curve. This no load speed may be as much as 5% above the normal operating speed. This means the governor has a built-in bias which works against holding speed down on a load rejection. Even though the speeder gear is driven back to the position corresponding to a no-load speed of 1800 rpm, it cannot move fast enough to eliminate the effect of speed droop.

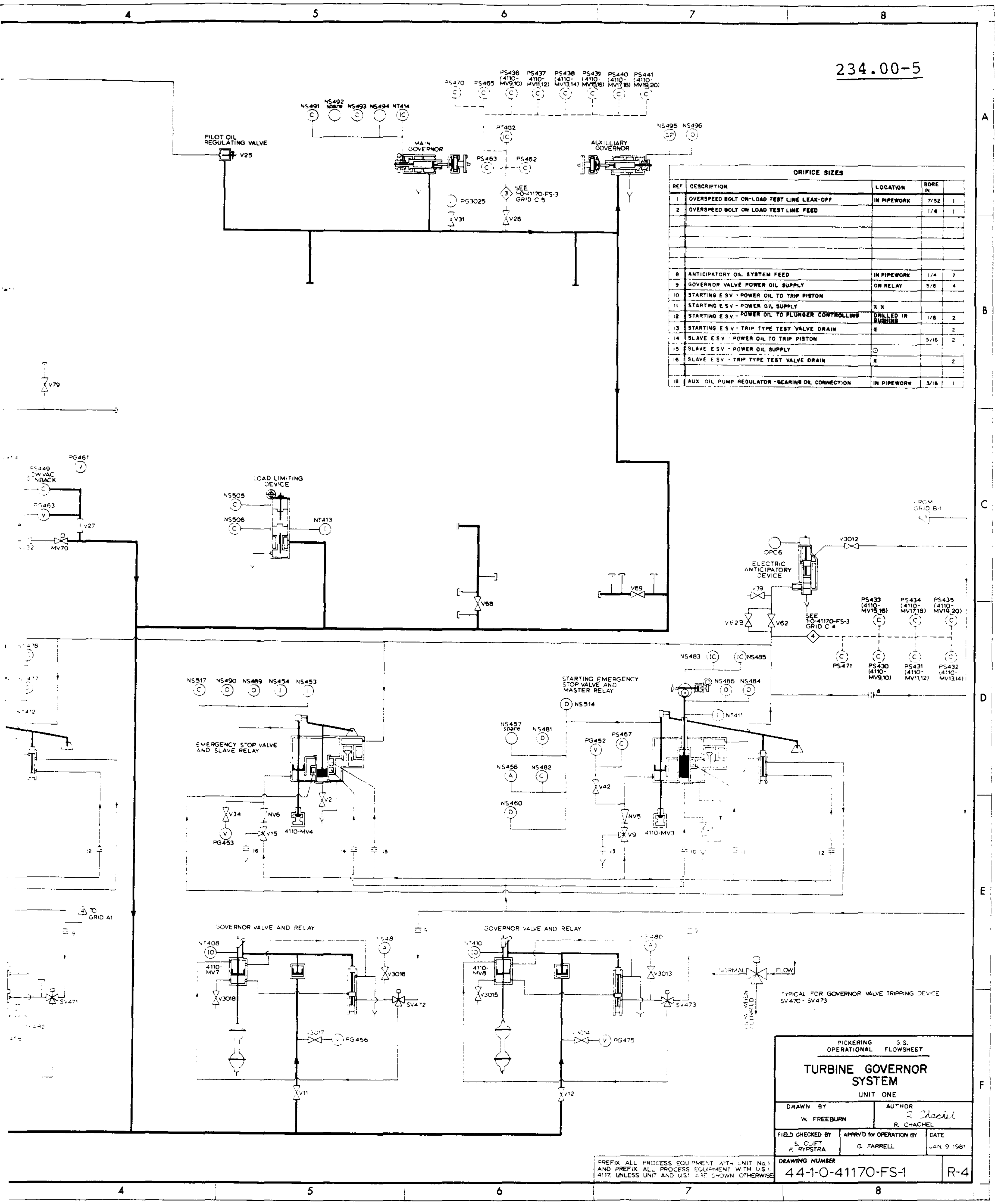
One method of attempting to eliminate both of these problems is the use of an auxiliary governor which operates in parallel with the main governor. This auxiliary governor has no speed droop and is set to spill oil at a constant 1% above normal operating speed. On an overspeed following a load rejection, the auxiliary governor begins to spill oil at 1% above operating speed (1818 rpm for an 1800 rpm turbine). The auxiliary governor not only aids the main governor in dropping pilot oil pressure but also will attempt to control speed at 1% overspeed until the speeder gear has run back to main governor. This effectively eliminates the speed droop bias of the main governor on an overspeed following load rejection.

In the upper, right hand corner of Figure 5.6 you can see the main and auxiliary governor of the Pickering NGS turbine. You can also trace the pilot oil system from the pilot oil regulating valve to the control relays of each of the four governor steam valves.

The unfortunate fact about the mechanical hydraulic governor is that on large turbines, the dead time produced by reservoir effect in the governing system is unacceptably long. The main and auxiliary governors together are not capable of limiting the overspeed following a load rejection to acceptable levels. To limit the amount of overspeed, the governing system must be aided by an <u>electric anticipator</u>.

The governor system shown in Figure 5.6 contains two electric anticipators which operate in parallel. If the generator output breaker opens under load, the electric anticipators are tripped by auxiliary contacts on the breaker. The electric anticipator dumps power oil to drain which shuts the emergency stop valves. At the same time, the low power oil pressure, trips pressure switches which shut the intercept valves and open the release valves.

After five seconds the main and auxiliary governors have regained control of turbine speed. At this time, the electric anticipators reset, and the emergency stop valves and intercept valves reopen and the release valves shut. The turbine will thus end up with the steam control valves (ESV, IV and RV) in their normal position, with the auxiliary governor controlling turbine speed. As the speeder gear is run back, the main governor will eventually gain control of the governor steam valves and lower speed from 101% (auxiliary governor) to 100% of operating speed. Figure 5.7 shows the response to a load rejection of a typical mechanical hydraulic governing system.



ORIFICE SIZES				
REF	DESCRIPTION	LOCATION	BORE IN	
1	OVERSPEED BOLT ON-LOAD TEST LINE LEAK-OFF	IN PIPEWORK	7/32	1
2	OVERSPEED BOLT ON LOAD TEST LINE FEED		1/4	1
8	ANTICIPATORY OIL SYSTEM FEED	IN PIPEWORK	1/4	2
9	GOVERNOR VALVE POWER OIL SUPPLY	ON RELAY	5/8	4
10	STARTING E.S.V. - POWER OIL TO TRIP PISTON			
11	STARTING E.S.V. - POWER OIL SUPPLY	X X		
12	STARTING E.S.V. - POWER OIL TO PLUNGER CONTROLLING	DRILLED IN BUSHING	1/8	2
13	STARTING E.S.V. - TRIP TYPE TEST VALVE DRAIN	*		2
14	SLAVE E.S.V. - POWER OIL TO TRIP PISTON		5/16	2
15	SLAVE E.S.V. - POWER OIL SUPPLY	⊙		
16	SLAVE E.S.V. - TRIP TYPE TEST VALVE DRAIN	*		2
18	AUX. OIL PUMP REGULATOR - BEARING OIL CONNECTION	IN PIPEWORK	3/16	1

PREFIX ALL PROCESS EQUIPMENT WITH UNIT No.1 AND PREFIX ALL PROCESS EQUIPMENT WITH U.S.I. 4117, UNLESS UNIT AND U.S.I. ARE SHOWN OTHERWISE

PICKERING G.S.
OPERATIONAL FLOWSHEET

TURBINE GOVERNOR SYSTEM

UNIT ONE

DRAWN BY	AUTHOR	
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FIELD CHECKED BY	APPR'V'D for OPERATION BY	DATE
S. CLIFT F. RYPSTRA	G. FARRELL	JAN. 9, 1981

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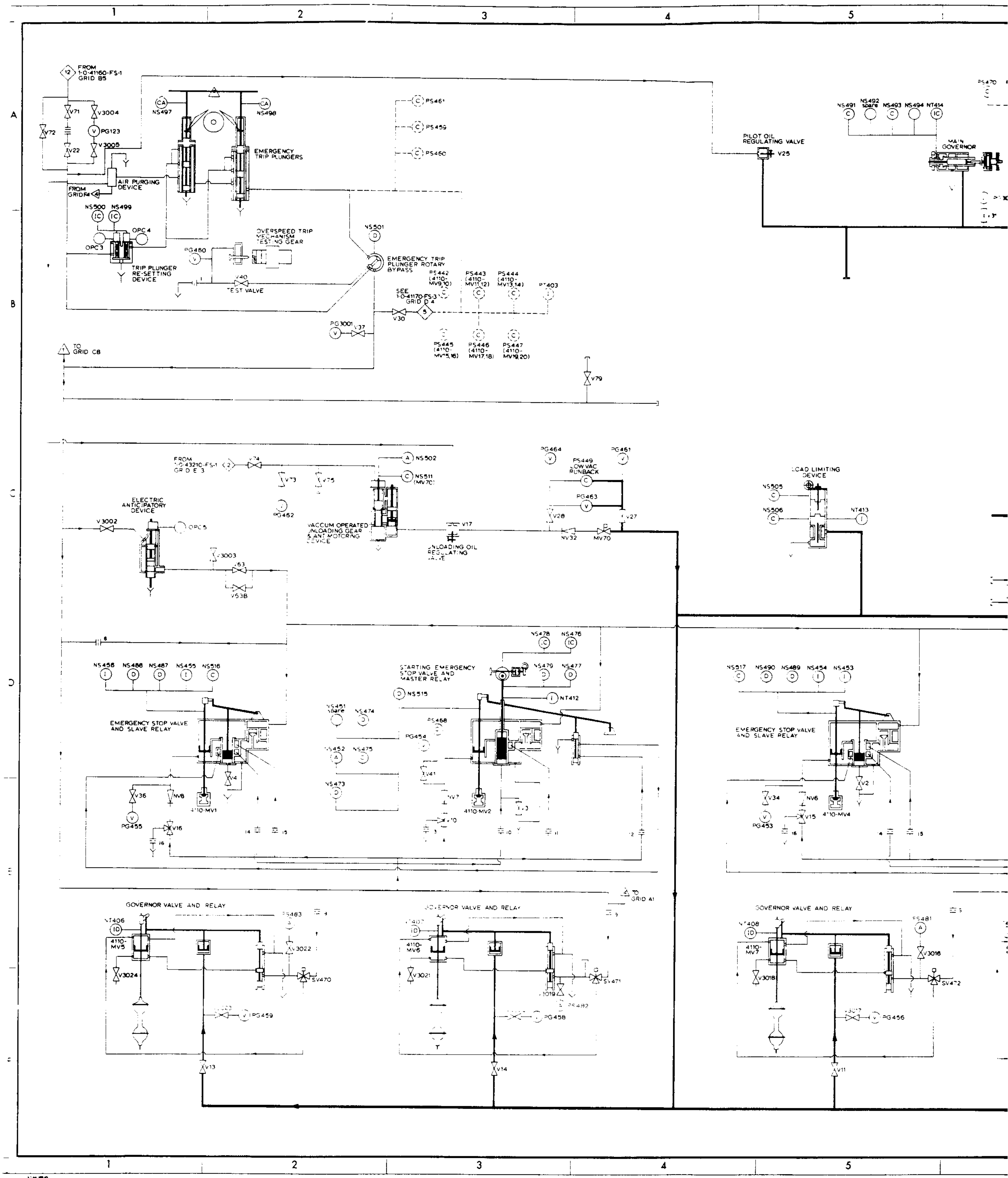
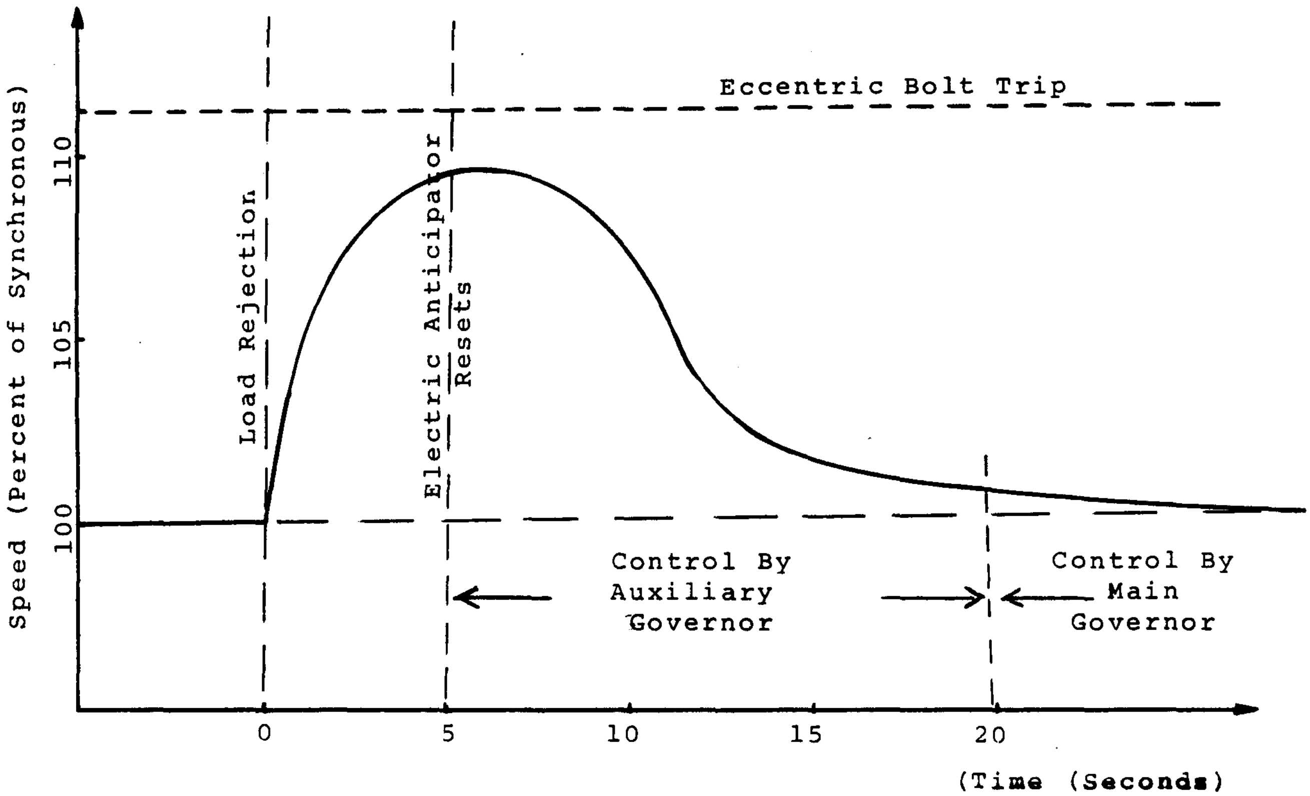


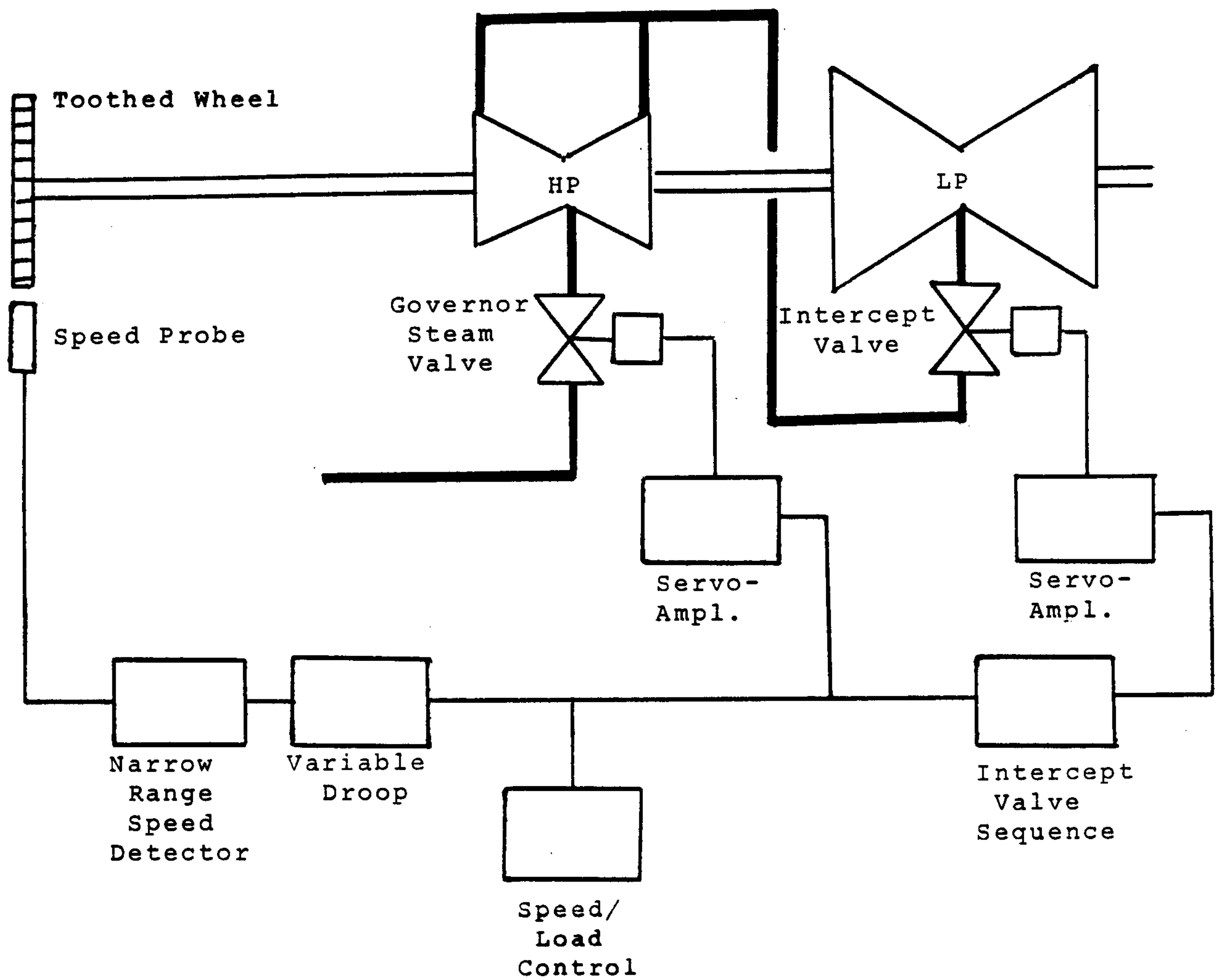
FROM 1-0-41160-FS-1 GRID B5
EMERGENCY TRIP PLUNGERS
AIR PURGING DEVICE
FROM GRID F4
TRIP PLUNGER RE-SETTING DEVICE
OVERSPEED TRIP MECHANISM TESTING GEAR
TEST VALVE
EMERGENCY TRIP PLUNGER ROTARY BYPASS
SEE 1-0-41170-FS-3 GRID D4
TO GRID C8
PILOT OIL REGULATING VALVE
MAIN GOVERNOR
FROM 1-0-43210-FS-1 GRID E3
ELECTRIC ANTICIPATORY DEVICE
VACCUM OPERATED UNLOADING GEAR & ANT MOTORING DEVICE
UNLOADING OIL REGULATING VALVE
PS449 LOW VAC RUNBACK
LOAD LIMITING DEVICE
EMERGENCY STOP VALVE AND SLAVE RELAY
STARTING EMERGENCY STOP VALVE AND MASTER RELAY
EMERGENCY STOP VALVE AND SLAVE RELAY
GOVERNOR VALVE AND RELAY
GOVERNOR VALVE AND RELAY
GOVERNOR VALVE AND RELAY
TO GRID A1
4110-MV1
4110-MV2
4110-MV4
4110-MV5
4110-MV6
4110-MV7



Mechanical Hydraulic Governing System on Load Rejection

Figure 5.7

Electrical Governor



Basic Electrical-Hydraulic Governing System

Figure 5.8

Basic Electrical - Hydraulic Governor

Figure 5.8 shows a basic electrical governing system. The turbine speed is sensed through a probe which counts the teeth passing it on a toothed wheel attached to the shaft. By calculating the rate at which the teeth pass the probe, the speed of the turbine can be calculated. The output of the narrow range speed sensor is fed through the speed droop control to the servo-amplifiers for the governor steam valves and intercept valves. A typical servo control system is shown in Figure 5.9. The intercept valves are sequenced so as to be fully open any time the governor steam valves are more than 25% open, and to be 50% open when the governor steam valves are fully shut.

The opening of the governor steam valves and intercept valves is accomplished through the speed/load control which is the electrical governing system's equivalent of the speeder gear. As with the mechanical governor, the input to the speed/load control may be either manual, computer loading or runback.

In the mechanical hydraulic governor the speed droop is a function of the design of the governor, and as such is virtually a constant. In the electrical governor, however, it is reasonably easy to vary the speed droop to obtain the most desirable droop setting for a particular operating condition. The droop setting is variable between 1% and 12%. Basically, the droop is high (12%) at operating speed when not connected to the grid. The droop is moderate (4%) when at operating speed and synchronized to the grid. The droop is low (1%) on a load rejection.

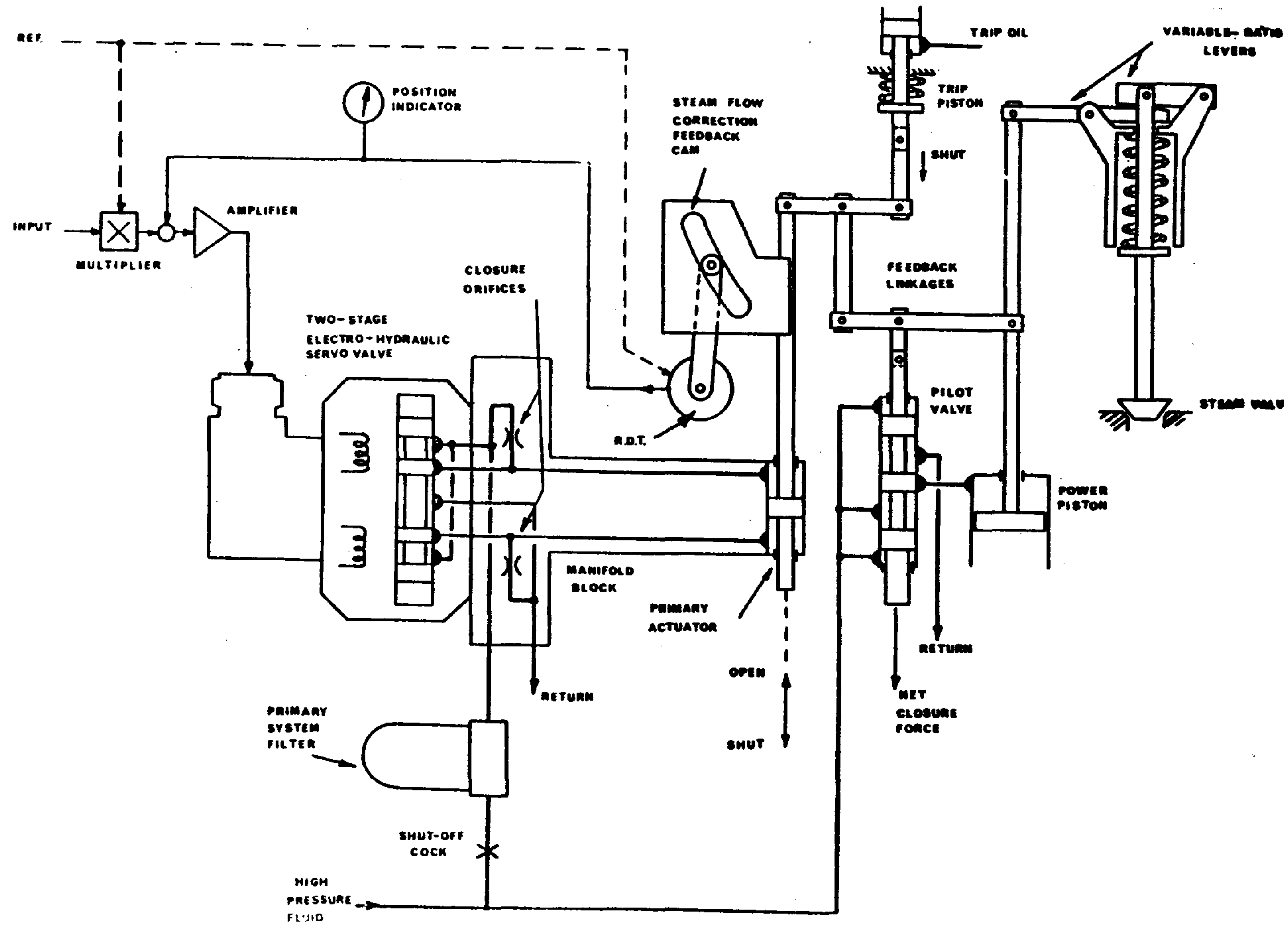
Figure 5.10 shows a schematic block diagram of the electrical hydraulic governing system. The wide range speed detector is used to control the emergency stop valves during run up to operating speed. Once at operating speed, the wide range speed detector plays no role in normal or abnormal speed control.

Load Rejection

The basic difference between the response of the electrical hydraulic governor and the mechanical hydraulic governor on a load rejection, is that the electrical hydraulic governor is so much faster. Because of the use of electrical signals and high pressure FRF, dead time is virtually eliminated. During a full load rejection, the governor can control the entire overspeed without operation of the emergency stop valves.

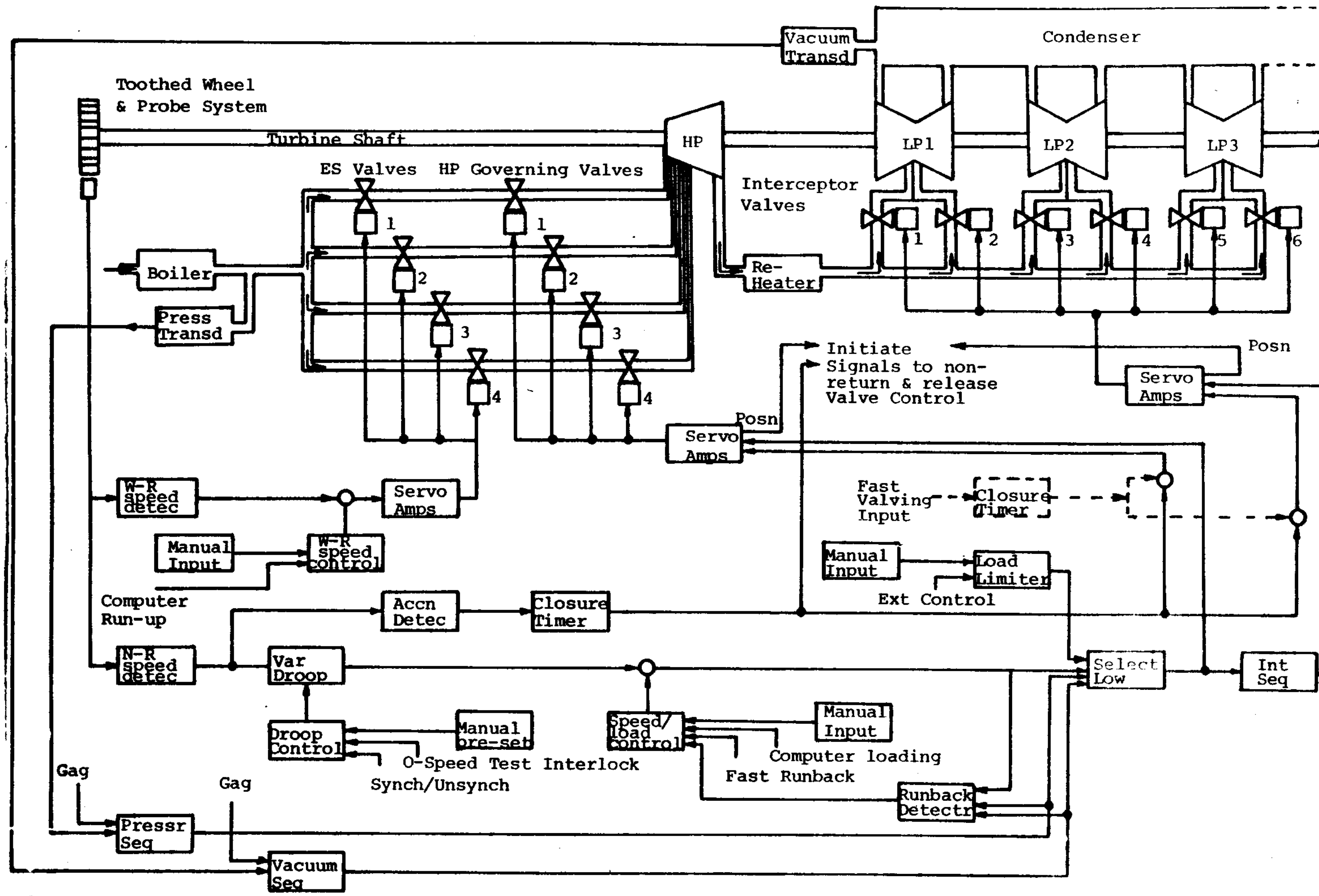
The electrical governing system has two features which come into operation during a load rejection:

1. An acceleration detector senses the rapid increase in speed from the narrow range speed detector. This rapid acceleration initiates a fast valving input which feeds an overriding signal to the governor steam valves and intercept valves. Regardless of what other mode the



Governor Steam Valve Control System

Figure 5.9



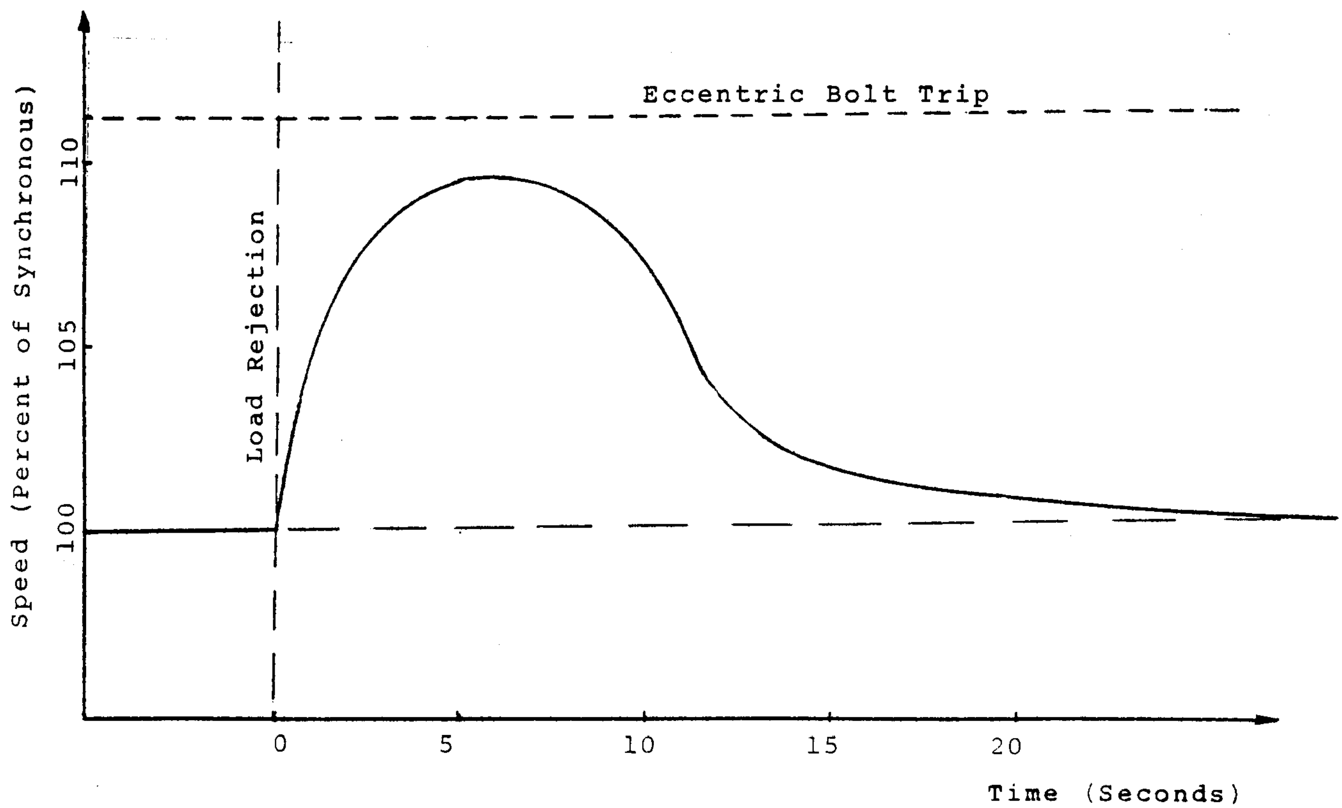
Block Diagram of Electrical-Hydraulic Governing System

Figure 5.10

valves may be in, this signal fully closes the valves at the fastest possible rate. The closure of the intercept valves and governor steam valves initiates the signal to close the extraction steam non-return valves and open the release valves.

2. When the generator becomes unsynchronized by a load rejection, the droop control shifts the speed droop to 1%. This serves the same function as the auxiliary governor in the mechanical hydraulic governing system. The shift in speed droop will insure the turbine returns to a no-load speed of only 1% overspeed until the speed/load control drives the no-load speed back to 1800 rpm.

Figure 5.11 shows the response to a load rejection of a typical electrical-hydraulic governing system.



Electrical - Hydraulic Governing System on Load Rejection

Figure 5.11

Eccentric Bolt Emergency Trip Plunger

Regardless of whether the turbine is equipped with an electrical or mechanical governing system, it will have an eccentric bolt emergency tripping device. This device will shutdown the turbine on overspeed and is set high enough (110%-112% of operating speed) to be required to operate only if the other overspeed devices don't control the speed. That is, if the normal overspeed devices operate properly, the eccentric bolt emergency tripping device should never have to operate.

The tripping mechanism of the eccentric bolt tripping device is shown in Figure 5.12. Spring tension holds the bolt in against centrifugal force. At the trip speed, centrifugal force overcomes spring tension and the bolt is thrown out, contacting a tripping latch. This action operates a plunger which directly or indirectly dumps the fluid holding open the steam admission valves.

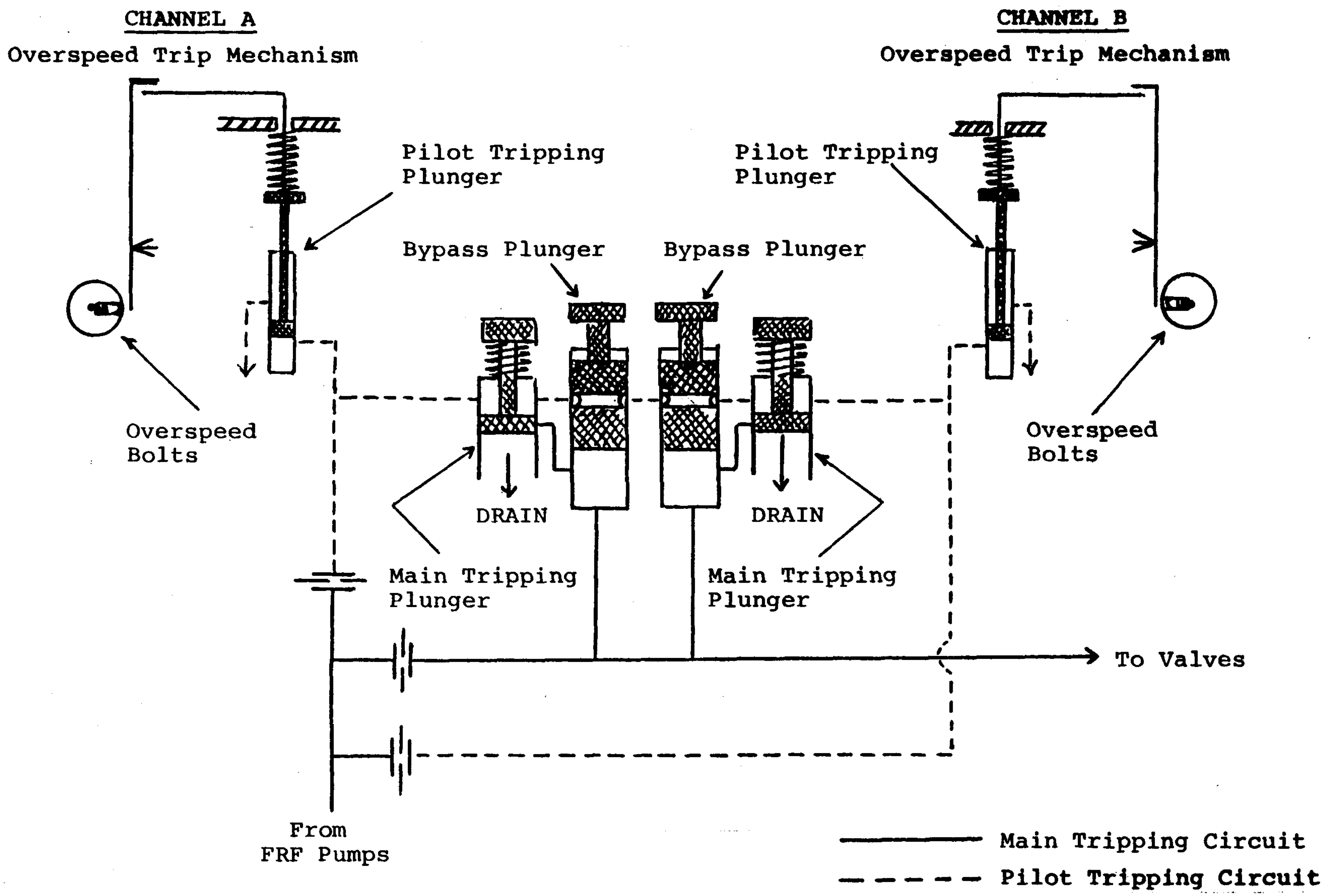
In the case of the mechanical hydraulic governor (Figure 5.6), the eccentric bolt tripping device dumps power oil which closes the emergency stop valves and governor steam valves. The decreasing power oil pressure in turn trips pressure switches which initiates air signals to shut the intercept valves and open the release valves.

In the case of the electrical hydraulic governor, the eccentric bolt tripping device dumps FRF which closes the emergency stop valves, governor steam valves, intercept valves and reheat emergency stop valves. The shutting of the GSVs and IVs initiates a signal to open the steam release valves and shut the extraction steam non-return valves.

Load Limiter

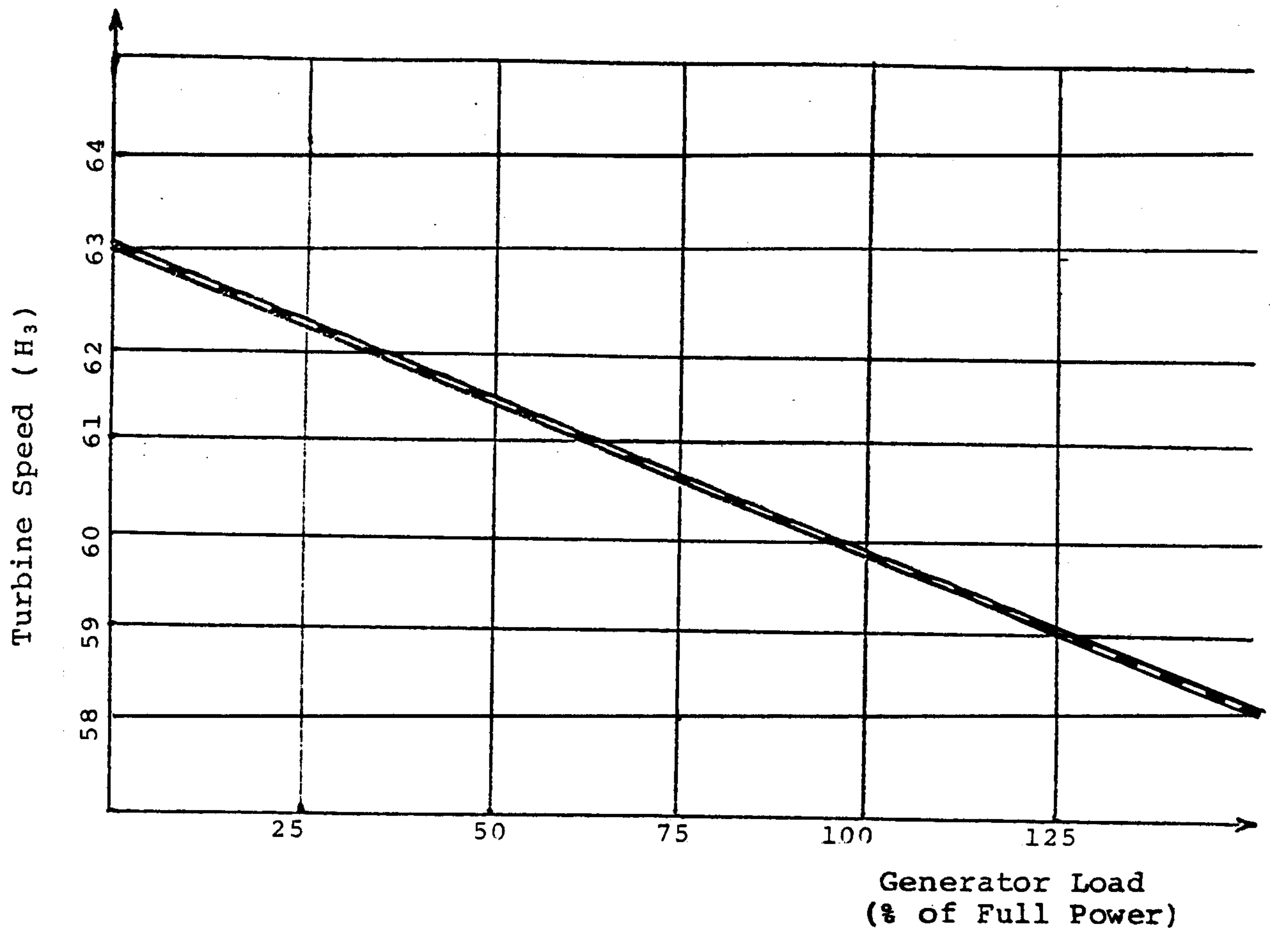
Figure 5.13 shows a typical speed droop curve. If grid frequency were to fall due to a system disturbance, the governor steam valves would open further. This overpowering of the turbine might cause physical damage to the turbine. It may also be necessary to limit load to some value below 100% maximum continuous rating. The load limiter can be adjusted to provide an upper limit to steam flow to the turbine by limiting the opening of the governor steam valves.

In the mechanical hydraulic governor this is done by restricting pilot oil pressure to a maximum value. In the electrical hydraulic governor this is done by a select low feature which limits the governor steam valve opening signal to the lower of that determined by the speed droop or the load limiter.



Overspeed Tripping Circuit

Figure 5.12



Speed Droop Curve

Figure 5.13

Anti-Motoring Device

Small turbine generators usually have a reverse power trip associated with their output breaker which opens the breaker on reverse power to prevent motoring. On large turbine generators, motor is permitted and such reverse power trips do not exist.

There must exist some feature, however, to prevent trying to start the turbine from the generator end by closing the output breaker when the turbine is shutdown. The generator is not designed to withstand the large overcurrents associated with starting the unit with the generator acting as an induction motor. The anti-motoring device prevents closing the output breaker if the emergency stop valves are shut.

ASSIGNMENT

1. For a mechanical hydraulic governing system, explain the part the following play in limiting an overspeed following load rejection:

   (a) electric anticipator,
   (b) main governor,
   (c) auxiliary governor,
   (d) speeder gear,
   (e) eccentric bolt emergency trip device.

2. For an electrical hydraulic governing system, explain the part the following play in limiting an overspeed following load rejection:

   (a) narrow range speed detector,
   (b) acceleration detector,
   (c) variable droop control,
   (d) speed/load control,
   (e) emergency trip plunger.

3. How is active load varied:

   (a) with a mechanical hydraulic governor?
   (b) with an electrical hydraulic governor?

4. Explain the advantages of an electrical hydraulic governor system using FRF over a mechanical hydraulic governor using lubricating oil.

5. What is the function of the load limiter?

6. What is the function of the anti-motoring device?

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 334

THE LUBRICATING OIL SYSTEM

The modern steam turbine and generator are carefully designed pieces of equipment constructed of well selected materials. Its satisfactory performance and useful life in service depend, among other things, on the maintenance of proper lubrication. This is one of the best insurances against turbine outage. We have already examined the support of the turbine and generator rotors by journal bearings. In this lesson we shall discuss how a typical lubricating oil system delivers oil to these bearings.

The lubricating oil system performs three basic functions:

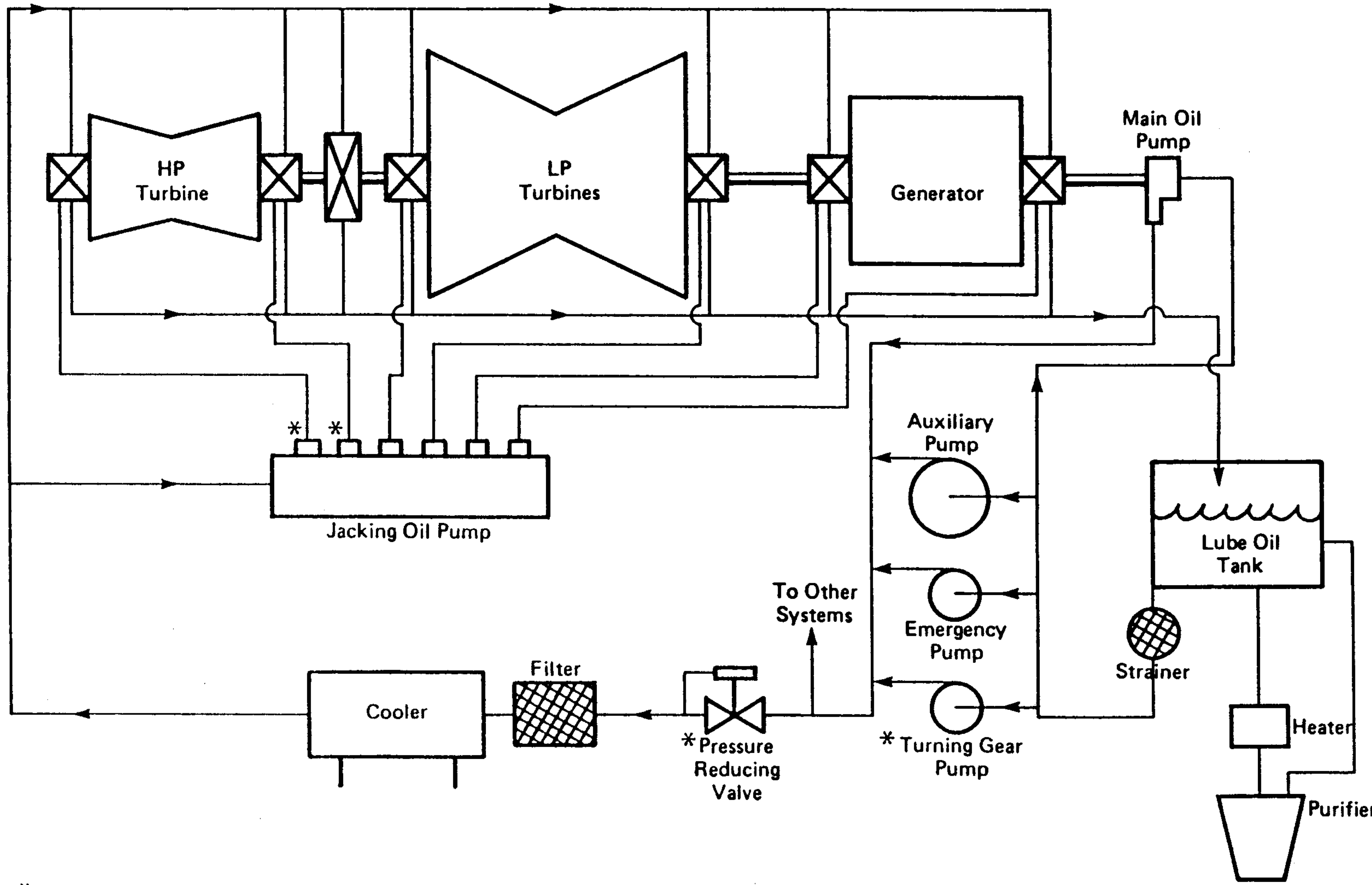
1. It reduces friction between rotating and fixed elements of the turbine and generator such as occur in the journal bearings and thrust bearings. This reduces wear, reduces heat and improves efficiency.

2. It removes heat from the bearings. This heat may either be generated by friction within the bearing or by conduction along the shaft from the turbines.

3. In mechanical hydraulic governing systems, it is used as a hydraulic pressure fluid. In these governing systems, lubricating oil is used for both the pilot oil and power oil systems.

Figure 10.1 is a schematic diagram of a typical lubricating oil system. The lubricating oil system is basically very simple. The majority of the lubricating oil is held in a lube oil tank or lube oil sump. The tank has a capacity of between 4 and 10 cubic meters depending on the size of the unit. The lube oil pumps take suction from this tank and discharge the oil at high pressure (typically at a gauge pressure of from 500 to 1000 kPa). The oil may or may not pass through a strainer prior to entering the pump.

From the lube oil pump discharge piping, the majority of the lube oil is sent to the bearings. A small portion of the oil is sent to the control oil system where it is used for pilot oil and power oil. The control oil is at lube oil pump discharge pressure so that the required force to operate the control valves can be achieved with as small a piston as possible [Force = (Pressure) x (Area)]. The larger the pressure, the smaller is the required area of the piston. With a small piston area, the volume of oil in the operators is



* Only at some stations

Lubricating Oil System

Figure 10.1

small. This allows for faster response time as it requires less time for oil to move in and out of the operators of the valves.

The required oil pressure to the bearings is much lower than that required in the control oil system [100 to 275 kPa(g) is typical]. A higher pressure is not only unnecessary but would require more complicated bearing seals to keep the oil from leaking out of the bearings. For this reason, the lube oil is passed through a pressure reducing valve prior to going to the bearings. On those turbine units with separate hydraulic systems for steam control valve operation, the lube oil pump discharge pressure is only as high as that required by the bearings.

The temperature of the oil flowing to the bearings must be carefully controlled. Coolers are installed in the line to keep the bearing oil at a sufficiently low temperature to properly cool the bearing. However, if the oil is too cool, its viscosity will be high enough to prevent proper oil flow to the bearings. The oil temperature out of the coolers is adjusted to be within a fairly narrow band (43° to 48°C is typical).

The oil flows through full flow filters prior to distribution to the bearings. These filters remove any solid contaminants in the oil which might score the bearings. After passing through the bearing, the oil flows through a drain header back to the lube oil tank.

Main Oil Pump

The main oil pump is the one that delivers all the oil requirements for the turbine-generator at high pressure during normal operation. It is direct-driven from the turbine shaft and may be located at either the turbine or generator end of the shaft. For small units, the main oil pump is generally a gear pump. For large units, a gear pump may not be able to deliver the required quantity of oil at the desired pressure and in this case, a centrifugal type pump is used but it is still driven by the turbine shaft. Because the centrifugal pump is not self priming, large units with centrifugal main oil pumps are equipped with a lube oil driven booster pump to keep the main oil pump primed.

Because the main lube oil pump is an attached pump, it runs at turbine shaft speed. During startup and shutdown, the main oil pump is not turning fast enough to deliver the required pressure or flow. The speed at which the main oil pump becomes able to handle the lube oil requirements is generally around 90 - 95% of operating speed.

Auxiliary Oil Pump

The auxiliary oil pump has two functions:

1. It operates during startup and shutdown when the turbine shaft is not rotating fast enough for the main oil pump to deliver the required pressure and flow.

2. It acts as a standby lube oil pump in the event of a main oil pump failure.

The auxiliary lube oil pump is a full flow and full pressure pump which will completely meet the turbine and generator oil needs. In the event of a main lube oil pump failure, the auxiliary oil pump is capable of supplying all bearing and control oil needs to allow continued full power operation.

The auxiliary lube oil pump is started and stopped automatically by a pressure switch in the discharge piping of the main oil pump. The auxiliary oil pump is normally driven by a Class IV ac electric motor.

Emergency Oil Pump

The time required for a turbine to run down from operating speed to a stop is typically from 20 to 45 minutes. If the bearings did not receive lubrication during this period, they would rapidly overheat and be destroyed. A turbine trip coupled with a loss of ac power would result in bearing damage during turbine coast down unless the turbine is provided with a source of bearing lubrication. There is no way a modern large turbine generator is protected against a complete loss of lubricating oil at operating speed. The only method of protection is to provide a sufficient number of alternate power supply lube oil pumps to insure all lube oil flow is not lost.

The auxiliary lube oil pump is normally backed up with a Class I dc emergency lube oil pump. This pump is automatically started from a pressure switch in the lube oil supply header to the bearings.

The emergency lube oil pump is not a full flow or full pressure pump and will not supply the lube oil requirements for continued power operation. It is essentially a coast down pump. The emergency lube oil pump generally puts out insufficient pressure to keep the emergency stop valves and governor steam valves open. Thus a loss of the main and auxiliary lube oil pumps will result in a turbine trip.

Turning Gear Oil Pump

Some generating stations have a reduced capacity Class III ac motor driven lube oil pump for use when the turbine is on the turning gear. In those stations without this turning gear oil pump, the emergency oil pump is used to provide the reduced flow required when the turbine is on the turning gear.

Jacking Oil Pump

This pump is generally used only for large turbine-generators, and then only during the period when the shaft is rotated by the turning gear.

When the heavy turbine-generator shaft is at rest, it will squeeze the oil film from under the shaft at the bearings. If the shaft is then rotated, there will be metal-to-metal rubbing until the oil can work its way underneath. When the unit is rotated on the turning gear, the shaft is not rotating fast enough to keep a good "oil wedge" between the shaft and the bearing surface. This condition can also lead to metal-to-metal contact.

To avoid this situation, the jacking oil pump injects oil at high pressure (around 6.9 megapascals) into the bearing at the bottom of the shaft. This tends to lift or jack the shaft a few hundredths of a millimeter off the bearing so that there will be no metal-to-metal contact. Once the turbine speed has increased above turning gear speed, the rotation of the shaft will keep an oil wedge under the shaft and the jacking oil pump is no longer required. The jacking oil pump will normally run whenever the turning gear is running and will shutdown when the turning gear is shutdown.

The jacking oil pump may take a suction either from the oil line supplying the bearings, as shown in Figure 10.1, or directly from the oil tank. In either case, however, a lubricating oil pump is started. This is because the jacking oil pump does not supply all the bearings, notably the exciter bearings and thrust bearings, and these bearings would receive no oil flow without the auxiliary oil pump. In addition, since the flow from the jacking oil pump is quite small, it may not provide sufficient oil flow to cool the bearings it does supply. The jacking oil pump is generally a reciprocating plunger type pump and supplies the bearings through individual lines from the pump to each bearing.

Oil Purification System

During operation, the lubricating oil becomes contaminated with a variety of undesirable impurities:

1. Water which most likely enters the system during shut-down from humidity in the air.

2. Fibres which come from the gasket material used to seal joints in the system.

3. Sludge which results from the breakdown of the oil into longer chain molecules and results in a thickening of the oil.

4. Organic compounds which result from a slow reaction between the oil, oxygen and the metal piping.

5. Metal fragments which come from wear products in the bearings and lube oil pumps.

Under normal operating conditions, none of these contaminants builds up rapidly in the oil, but the amount of contamination required to render lube oil totally unsuitable for use is quite small. Not only do these contaminants destroy the lubricating properties of the oil and accelerate corrosion, but they can act as a grit within the bearings to cause bearing wear and unevenness. The insoluble impurities can be removed with filters, but the soluable impurities are only removed by centrifuging.

The majority of the oil purification is accomplished with a centrifugal purifier. The purifier normally operates at about 15,000 rpm. Since the sludge, water, fibres and other impurities are more dense than the oil, they are thrown to the outside and removed from the oil. The impurities normally are collected in the bottom of the purifier and must be removed at regular intervals.

The oil may be heated before it enters the purifier since this lowers the viscosity and improves the separation of impurities from the oil. Also water may be deliberately added to the oil before the purifier, to dissolve impurities. All the water, of course, is centrifuged from the oil before it is returned to the system.

Centrifuges of sufficient capacity to purify all the oil in the system once every 6 - 10 hours, are normally installed. While the turbine is operating, the purifier is run virtually continuously to purify the oil. The purifier is also normally run prior to startup to purify the oil and warm it to near operating temperature.

ASSIGNMENT

1. Name the three jobs done by lubricating oil for a turbine-generator.

2. Why is it important to have sufficient oil flowing through the bearings at all times?

3. Briefly, what is the purpose of each of the following pumps?

   (a) Main Oil Pump
   (b) Auxiliary Oil Pump
   (c) Emergency Oil Pump
   (d) Jacking Oil Pump

4. Briefly, what function does the oil purifier perform?

5. Explain why the order of equipment operation on a start-up is:

   (a) Auxiliary Oil Pump
   (b) Jacking Oil Pump
   (c) Turning Gear.

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THE TURNING GEAR

In the time it takes for a turbine to come to a complete stop following shutdown, the casing and rotor hardly cool down at all. One hour after steam is shut off, the turbine is essentially at the same temperature as it was when operating. As the turbine slowly cools, cool steam and air tend to settle to the bottom of the casing and warm steam and air rise to the top of the casing. This temperature gradient from the top to the bottom of the casing causes the bottom of the rotor to cool faster than the top of the rotor. This results in a greater contraction of the bottom of the rotor than the top.

The differential contraction results in an upward bowing of the rotor called shaft hog. For a large turbine, the temperature gradient necessary to hog the shaft beyond acceptable limits in only 3 - 4°C.

There are few phenomena as reliable as shaft log. If a hot turbine rotor is allowed to come to a complete stop prior to cooling, the shaft will, almost without exception, develop a permanent bow. Although very minor hogging can occasionally be removed by rotating the shaft within the casing, this is generally an uncertain cure at best. Failing in this, the rotor must be removed from the turbine and sent to the manufacturer for heat treatment or replacement.

To overcome this problem on shutdown, the shaft is kept slowly turning until cool. The turning gear assembly consists of an electric motor which drives the shaft through a reduction gear. Depending on the station, the turbine shaft is rotated at a speed between 10 and 30 rpm. The turbine is normally kept on the turning gear for 24 hours or until the turbine is cooled below 100°C, whichever takes longer.

If the turbine is to be shutdown for maintenance, the unit must be kept in a condition which will allow operation of the turning gear for about 24 hours. After this time, the turbine may be stopped and disassembled.

Another condition which can cause permanent bending of the shaft is warming of the shaft by applying gland sealing steam when the shaft is stationary. Whenever gland seal steam is applied, the shaft must be rotated either by the turning gear or with steam, or distortion is likely to occur.

When cold, the shaft can sag under its own weight if left stationary for a prolonged period of time. The requirements for unacceptable shaft sag are not so clear cut as for shaft hog. Rotating the shaft on the turning gear for one hour per day is usually more than sufficient to prevent sagging. Longer periods of non-rotation may occasionally be necessary, particularly during major overhaul when the lubricating oil system, turning gear and bearings may be disassembled. When the turning gear is disassembled, the shaft can be hand barred or rotated with a crane. In these cases, sound engineering judgement on the part of senior station personnel must be used to determine the maximum period the shaft may safely remain stationary.

On selecting the turning gear motor to start, the following sequence of events should occur:

1. Lube oil should be established at the bearings.

2. The jacking-oil pump should start.

3. Seal oil should be established.

On completion of these items, the turning-gear motor should start and engage shaft drive.

A speed of approximately 3 rpm would be enough to prevent shaft distortion, but speeds up to 30 rpm are generally used to create turbulence within the cylinder and thus cool the cylinder and shaft uniformly. Also running at this speed ensures that an oil film is maintained in the joined bearings.

On initial engagement of the turning gear, a close watch should be kept on the eccentricity and vibration indicators. The shaft should be run in this mode until the eccentricity and vibration are within acceptable limits.

On startup, after running on turning gear as outlined above, steam may be admitted to the turbine. As the speed increases above turning gear speed, the gears will disengage, the turning gear motor will continue to run until the turbine speed is greater than 100 rpm to ensure that if a run back is necessary, that the drive will be taken up immediately by the turning gear as soon as the shaft speed is less than 30 rpm.

It is now becoming general practice on shutdown that as soon as the turbine speed falls to 100 rpm, the turning gear motor will start and as soon as the turbine speed falls below 30 rpm, the turning gear drive will automatically engage on the turbine shaft. At the same time as the turning gear motor starts, the jacking oil pump starts. Both shutdown at the same time as well.

ASSIGNMENT

1. Why does a turbine shaft hog?

2. How is hogging prevented?

3. What action would you take if the turning gear failed to start on shutdown?

4. Why is it necessary to run the turning gear before admitting steam to the turbine?

5. What is shaft sag, and when is it likely to occur?

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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-service 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 four catagories:

(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,

(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, or,

(d) speed will rise to a level where the centrifugal forces on the generator rotor will rupture the rotor, or will loosen rotating parts which can then contact stationary parts.

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 lesson five 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.

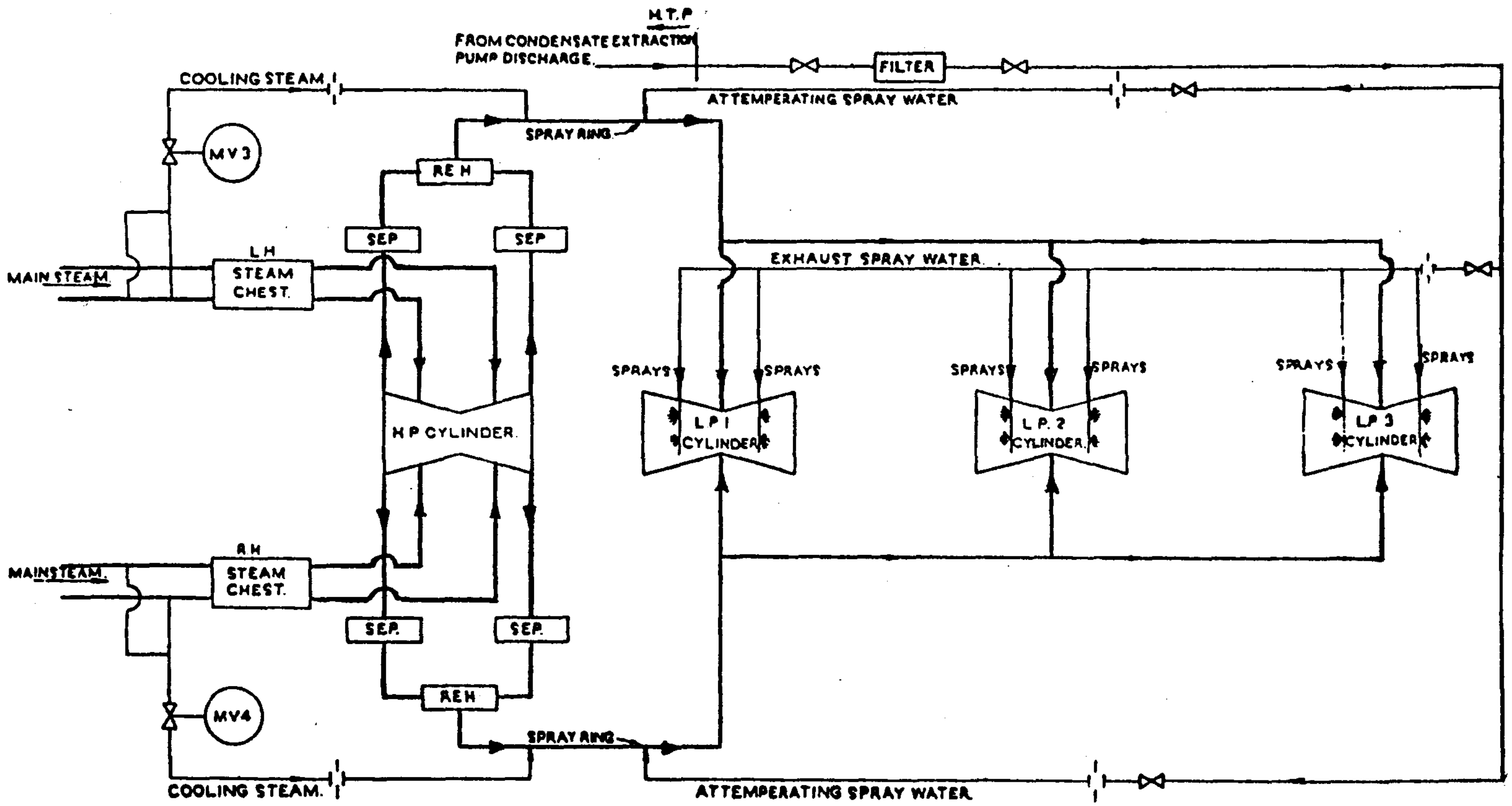
During motoring the turbine blading is turning through dead steam and the friction between the steam and the blading rapidly overheats the long turbine blades at the exhaust end of the low pressure turbine. The problem is made more severe if the vacuum decreases and the blading encounters higher than design steam densities. The problem can be partially alleviated by an exhaust spray system and a cooling steam system as shown in Figure 6.1.

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



Schematic Arrangement of LP Cylinder Exhaust Spray Cooling System and Cooling Steam System

Figure 6.1

eventually not be able to condense all the steam being exhausted 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 lesson 1 and elsewhere in this lesson.

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.

Moisture Carryover

Carryover is the continuous entrainment of liquid boiler water in the steam leaving the steam generators. The cyclone separators and steam scrubbers in the steam generators are designed to remove virtually all of the liquid water and under normal conditions, the steam leaving the steam generators is less than .2% liquid water (moisture). Both design and operating engineers are very sloppy in their use of the term "carryover". This is because these people are concerned not only with the quantity of liquid boiler water leaving the steam generators but also with the quantity of chemicals which is carried along in the water droplets. A water droplet is a mini-sample of boiler water and its makeup is more or less representative of boiler chemistry. Depending on boiler chemistry, the moisture entrained in the steam may contain SiO_2, Cl^-, Na^+, soluable and insoluable calcium and magnesium salts, OH^- and a wide variety of other chemicals. The water which leaves a steam generator can thus cause damage in three ways:

(a) moisture erosion which would occur even if the water were pure,

(b) chemical corrosion from active ions carried with the water, and

(c) chemical deposits on valve seats and stems and on turbine blades.

Depending on which of these effects we are concerned about, one may talk about carryover as the amount of moisture or as the amount of dissolved and undissolved solids in this moisture. You can see that if the amount of solids in the boiler water increases, more solids will leave the boiler, even if no more water leaves the boiler. The point of all this is that when one discusses the causes of increased carryover you have to know whether he is talking about the increase in the quantity of water leaving the steam generator or the increase in chemicals leaving in that water. It is not the purpose of this lesson to solve the problems of the world. Suffice it to say that in this lesson carryover will be defined as the amount of <u>water</u> leaving with the steam. The reader is cautioned, however, that this definition is not universal.

Carryover can be increased by two methods: mechanical and chemical.

Mechanical methods which increase carryover are generally those which decrease the effectiveness of the cyclone separators and/or steam scrubbers. <u>High boiler level</u> can physically flood the separators and decrease their effectiveness. Under certain conditions, <u>low boiler level</u> can cause carryover. If boiler level drops below the bottom of the separator columns, level oscillations can cause overloading of some separators with resultant carryover. <u>Rapid power increases</u> can increase carryover not only through swell flooding the separators but through temporarily overloading the separators as steam rushes from the steam generators. If steam flow is above design either from one steam generator or all steam generators, increased carryover can result. The moisture separators are intended to produce dry steam at some maximum power level. If the steam generator is forced to supply more than this design maximum, steam quality will suffer. This method of inducing carryover can be a particular problem if one steam generator is isolated, possibly forcing the others to operate at higher power levels.

The turbine can be protected against mechanical carryover by:

(a) monitoring boiler level,

(b) adherence to specified loading rates, and

(c) high boiler level closure of the steam admission valves.

Chemical induced carryover results when the chemicals in the boiler water break the surface tension of the water or allow foaming to occur. The presence of oil in steam generators causes foaming to occur on the water surface. This foaming can cause severe carryover and oil in the boilers is an extremely serious problem. Because of this, however, designers have made it virtually impossible for oil to enter the steam/feedwater system. Although it requires a certain creative incompetance to get oil into the boilers, it is certainly possible. One not unlikely way is through leaking or standing oil being sucked into sub-atmospheric piping in the condensate or makeup water system.

Both high undissolved solids and high dissolved solids, particularly the former, can promote carryover through breaking the surface tension of the water. This promotes liquid water being carried off with the steam. In addition, such high solids will be carried over with the moisture and may foul blading and cause control valve sticking and leakage.

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.

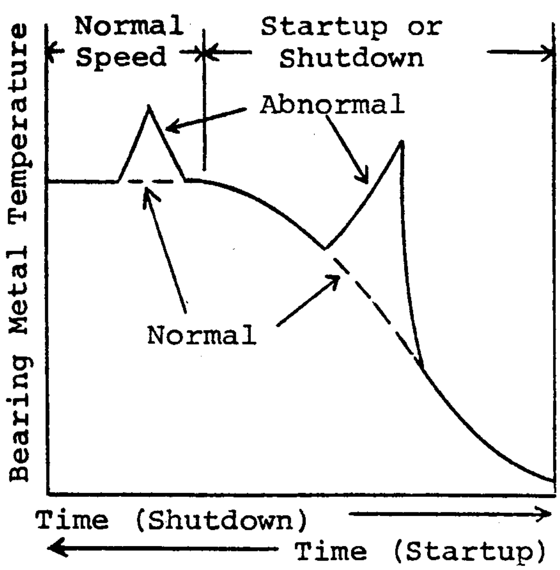
Since even a brief failure of the lube oil flow to the bearings can result in considerable damage to the unit, the system is designed to provide continuous oil flow under a variety of pump shutdowns and power failures. The automatic features which provide the backup lube oil supply must be tested frequently 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 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 6.2 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.

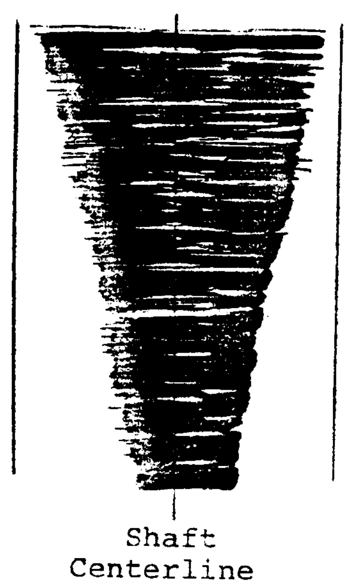
Bearings should be inspected for wear and alignment at least each time the turbine unit undergoes a major overhaul. Journals should be checked for smoothness and uniform roundness and diameter from one end to the other. Journals should be inspected for scoring or an uneven surface which occurs from scoring and self-lapping over an extended period. A bearing metal wear pattern such as shown in Figure 6.3 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 6.2



Misaligned Bearing
Wear Pattern

Figure 6.3

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) moisture carryover
   (f) blade failure
   (g) expansion bellows failure
   (h) bearing failure

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Turbine, Generator & Auxiliaries - Course 234

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) high stress in turbine rotor and casing,
(b) low cycle fatigue damage,
(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.

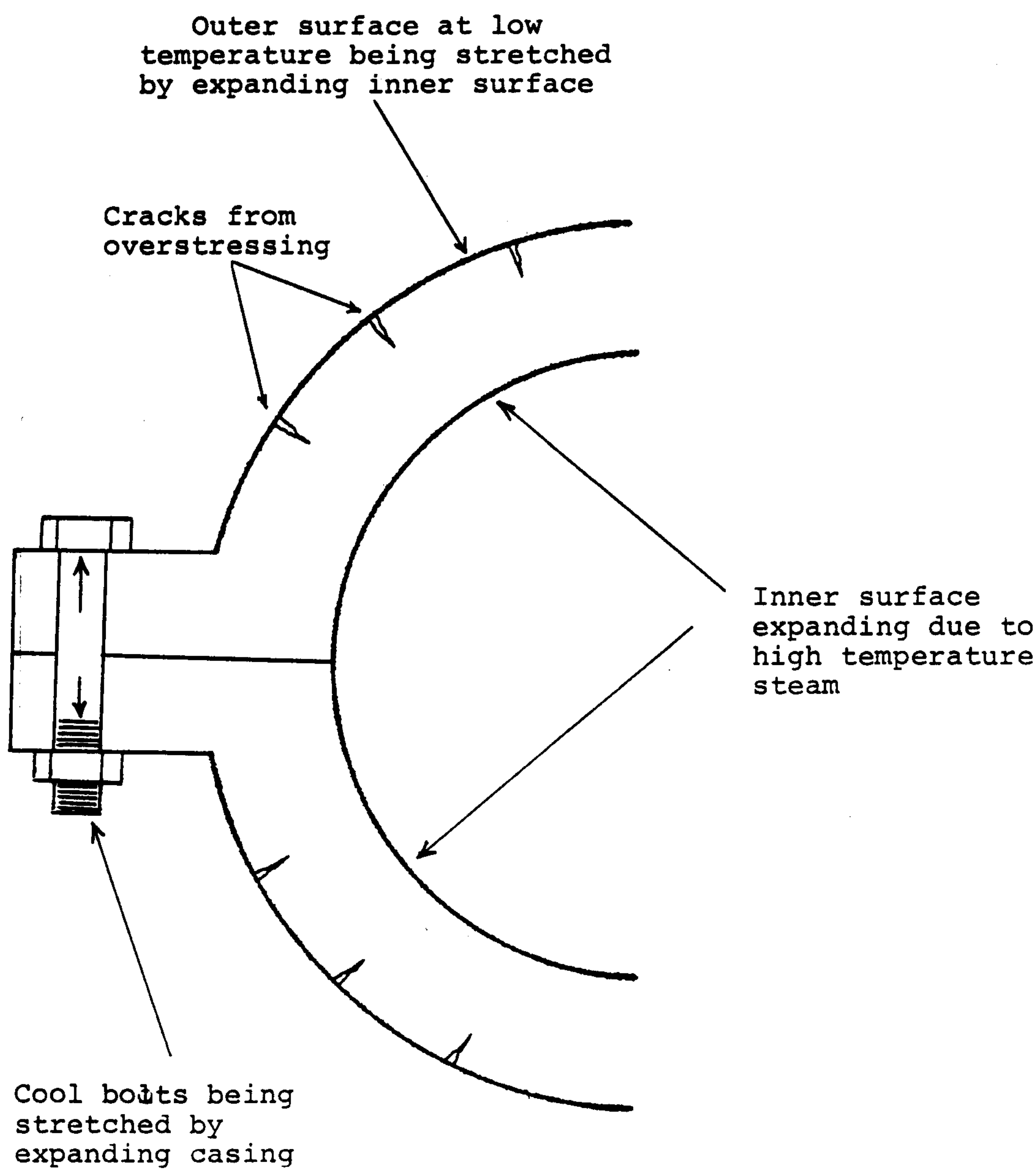
High Stress in Turbine Rotor and Casing

In the lesson on Turbine Construction we discussed the effects of thermal gradients on large structures such as the turbine casing and rotor. The part of the structure in contact with the heat source will heat up first while the temperature of the more remote parts will lag behind. The hotter parts will try to expand relative to the cold parts. Since they are parts of the same structure and have only a limited ability to expand at different rates, the hotter parts are compressed (not allowed to expand as much as they would like to) and the colder parts are stretched (forced to expand more than they want to). The stress imposed by these temperature gradients depends on the temperature difference between the hot and cold parts, the rate of temperature change, and the coefficient of linear expansion of the metal. Now a given piece of metal can only take just so much abuse and still be expected to go back to its original shape. If the stress exceeds the yield strength of the metal, the metal will plastically deform. After plastic deformation has occurred, the metal will no longer go back to its original shape. This plastic deformation may result in cracking, distortion or rupture of the metal.

Figure 8.1 shows the effect of an excessive thermal gradient on a turbine casing during startup. The relatively greater expansion of the inner surface tends to stretch the outer surface. If the temperature difference between the inner and outer surface is great enough, the outer surface can be stretched to the point of cracking. The bolts which hold the upper and lower casings together are particularly susceptible to high thermal gradients on startup and loading. The bolts must have a high strength and creep resistance and, therefore, do not have a great deal of ductility.

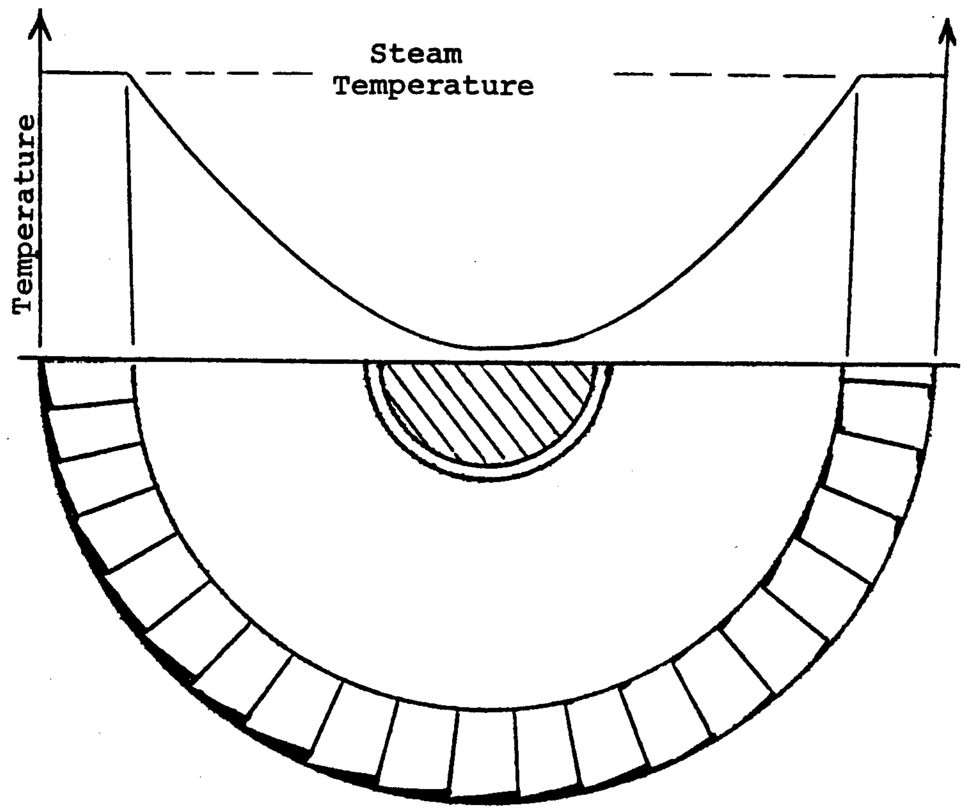
If the casing is heated up at an excessive rate, the casing expands faster than the bolts. This stretches the bolts. If the thermal gradient is great enough, the stress at the bottom of the threads may be high enough to cause cracking.

Figure 8.2 shows a typical temperature profile across the rotor during startup or loading, when the steam temperature rises above the equilibrium temperature of the rotor. 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 therefore under tensile stress.



Effect of Excessive Heatup on Turbine Casing

Figure 8.1

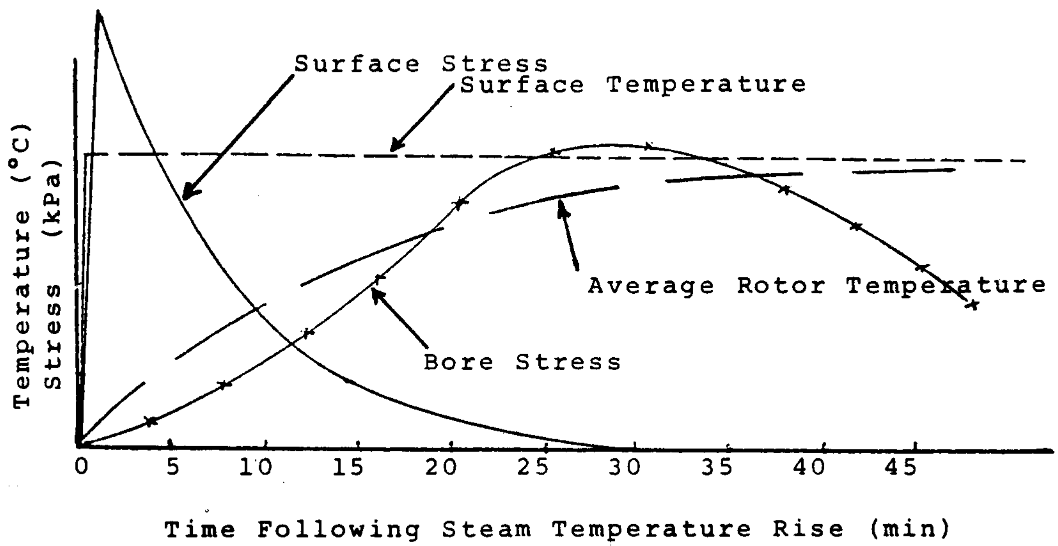


Temperature Profile Across
Rotor During Heatup

Figure 8.2

As the average rotor temperature rises to equilbrium, 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.

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 mximum 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 8.3 shows this effect. While the peak surface stress (surface 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 8.3

Low Cycle Fatigue Damage

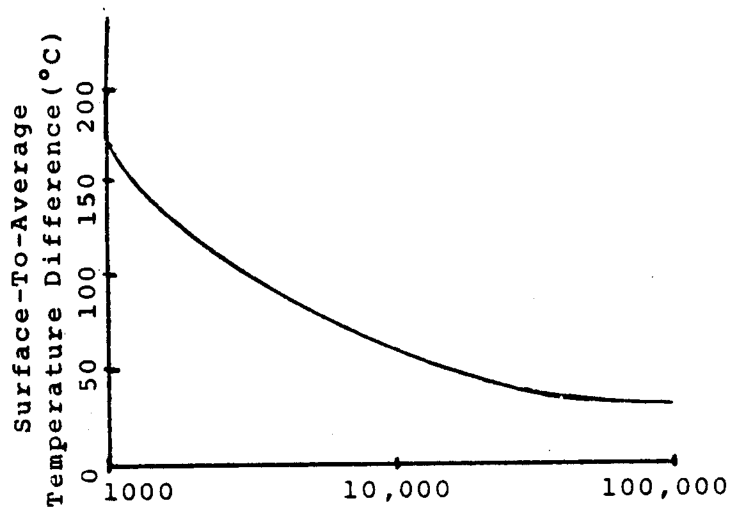
Even if the thermal gradient is not severe enough to produce immediate metal failure, the result of repeated excess stress may eventually lead to failure through fatigue cracking. Metal fatigue is a common phenomenon. It is the method by which most metals can be broken if bent back and forth a sufficient number of times. Fatigue cracking is dependent on three factors:

(a) there must be a high stress imposed,

(b) the high stress must be relaxed,

(c) the alternate stressing and relaxing must occur a sufficient number of times.

Thermal stresses in a casing or rotor produce all of the requirements for fatigue cracking. Startup and loading produce high thermal stresses which relax during steady state operation. This cycle is repeated each time the unit is started up or loaded. Figure 8.4 shows the relationship between the peak stress and the number of cycles to produce fatigue cracking. The greater the average metal to steam temperature differential, the greater will be the imposed thermal stress. As the peak stress (temperature difference) increases, the number of cycles to produce cracking decreases.



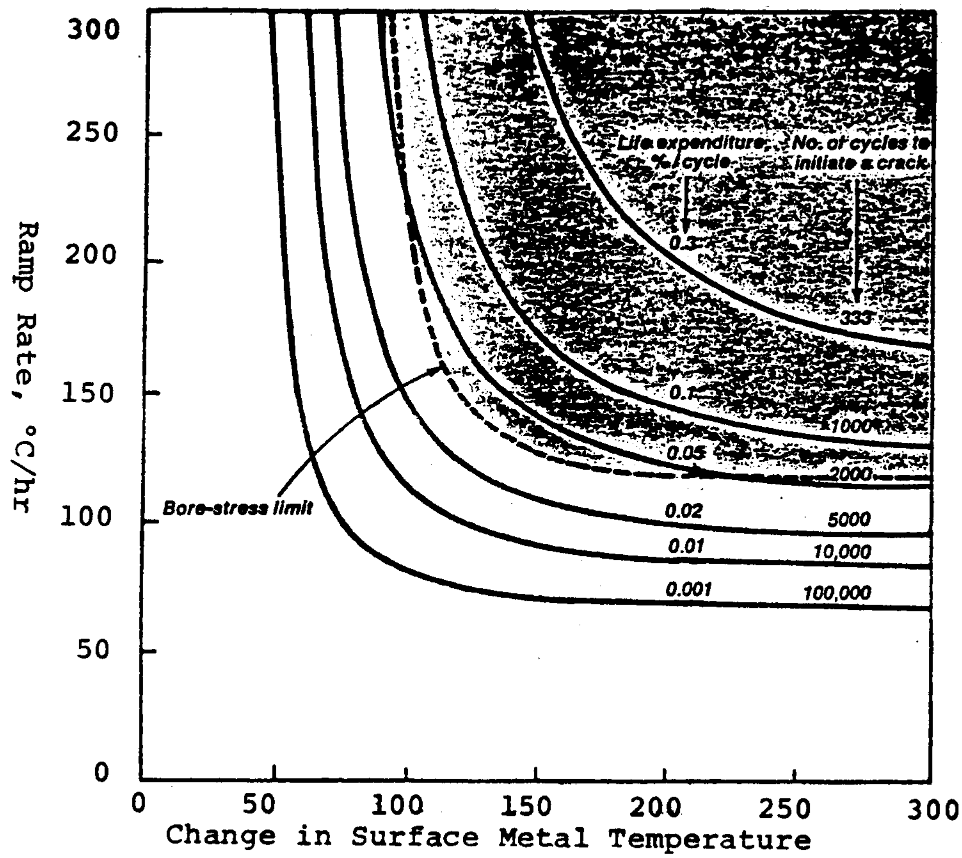
Cycles to Produce Cracking

Figure 8.4

Figure 8.5 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 heatup rate. For a 200°C heatup, it requires 100,000 cycles to initiate cracking with a 70°C/h heatup rate. If the heatup rate is increased to 115°C/h, 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.



Cycle-Life Expenditure

Figure 8.5

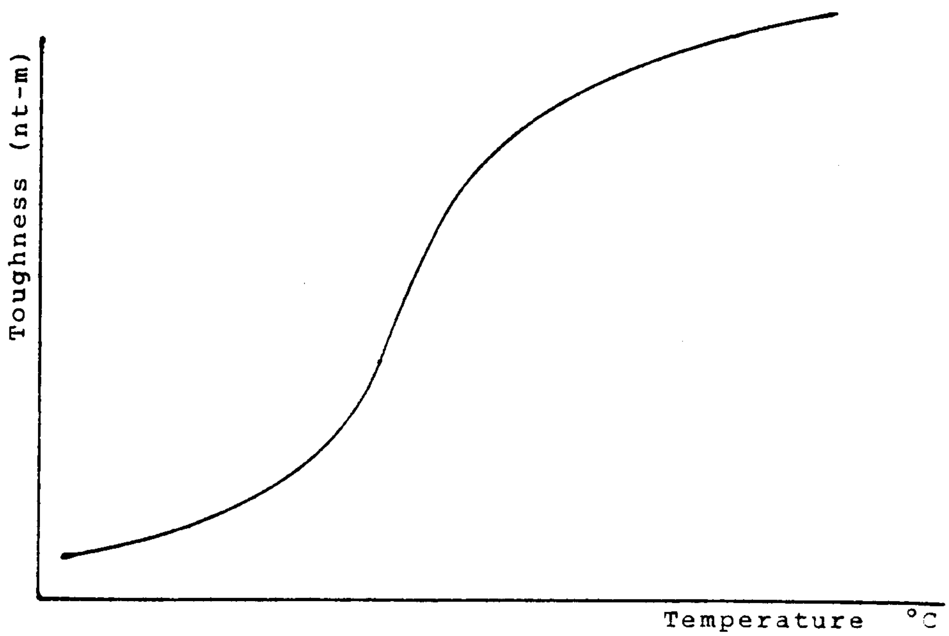
It is fair to ask why 2,000 cycles is any more significant than 100,000 cycles. On a base load nuclear station, the turbine may be only started up and loaded three or four times a year. For a forty year turbine life, that is only 100 to 125 cycles. The question can be answered in three ways.

(a) We don't know how the turbine will be operated ten, twenty or thirty years in the future. New fuels and reactivity mechanisms may allow nuclear stations to run as variable load stations. This might result in imposing hundreds of cycles a year on the turbine.

(b) Any unnecessary abuse when the turbine is young will drastically reduce its ability to absorb a severe transient later in 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 twenty startups with a heatup rate of 70°C per hour (life expenditure of .001% per cycle).

(c) Turbines have experienced rotor failure by fatigue after only a few years of operation. While these have usually been traced to design or material flaws, the cracking frequently occurred as a result of excessive thermal transients. We really never know what deficiencies exist in our turbines which could drastically alter the cycle-life expenditure curve. The stress the designer thought would allow 100,000 cycles to failure, may in fact cause failure in 1,000 cycles.

Low Rotor Toughness at Low Temperatures



Effect of Metal Temperature on Toughness

Figure 8.6

Figure 8.6 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 the range of 50° to 100°C and for generator rotors in the range of 10° to 30°C.

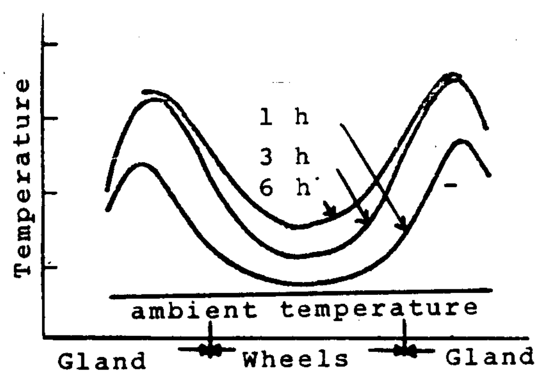
There are several methods of increasing the rotor temperature prior to imposing the high rotor stresses associated with high speeds and loads:

(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 8.7, 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 8.7

(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 8.8 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 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 prestartup temperature.

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 @	2 1/2 MW/MIN.	5 MW/MIN.	–	–	–
50-150 MW @	10 MW/MIN.	12 1/2 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 8.8

Vibration

The causes of vibration are legion but the effects are usually divisible into five 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,
(d) loosening of nuts, wedges and shims, and
(e) 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 8.9, 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 8.10 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 centre of mass away from the centre of rotation can develop large forces. The force created by a one kilogram imbalance located one meter from the centre 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 shutdown 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 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.

GENERAL MACHINERY VIBRATION SEVERITY CHART

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

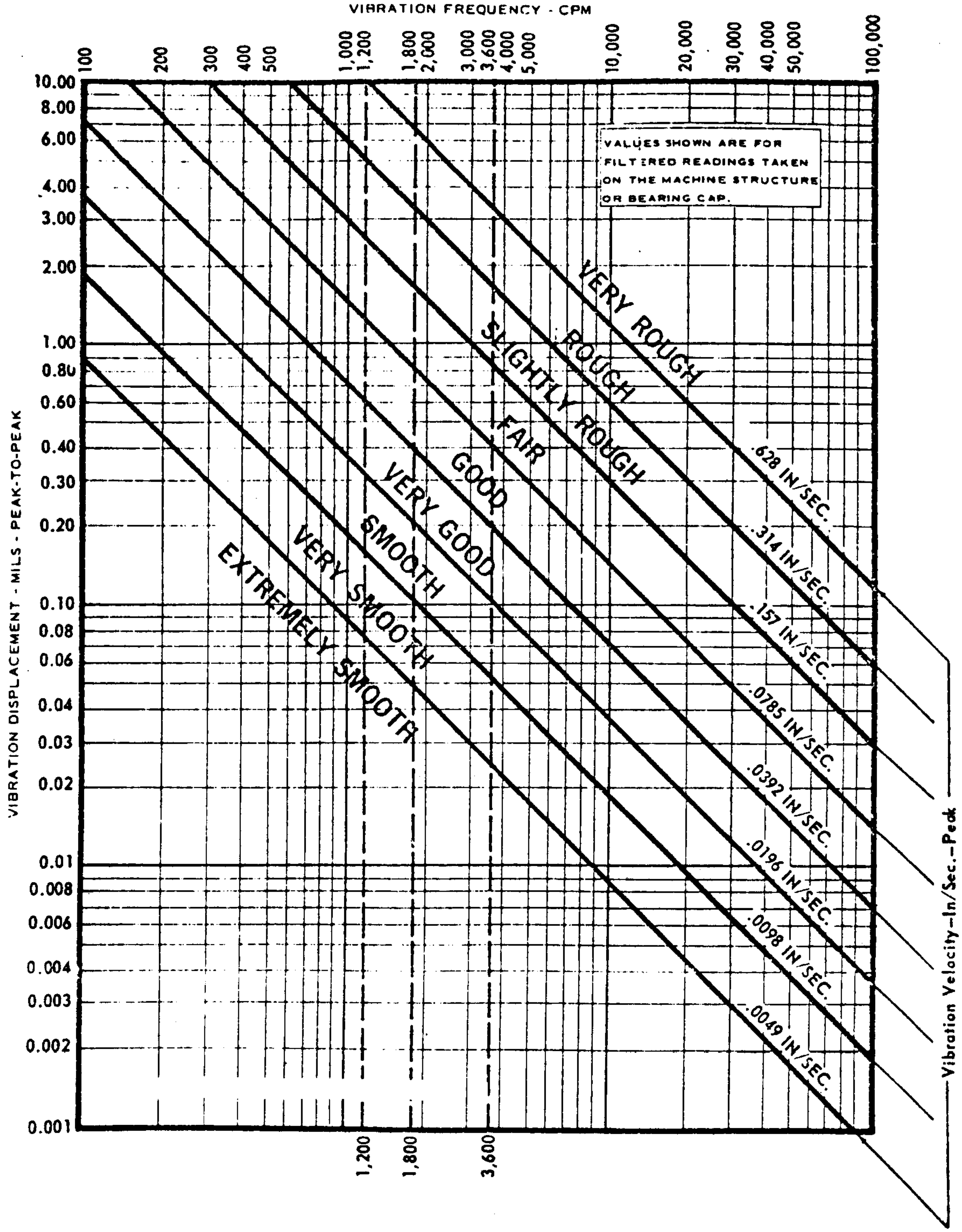


Figure 8.9

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.

Figure 8.10

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 <u>subharmonic</u> 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 clearances. 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 more sensitive to bending or deflecting. This means the internal damping of the shaft is at a minimum as shown in Figure 8.11.

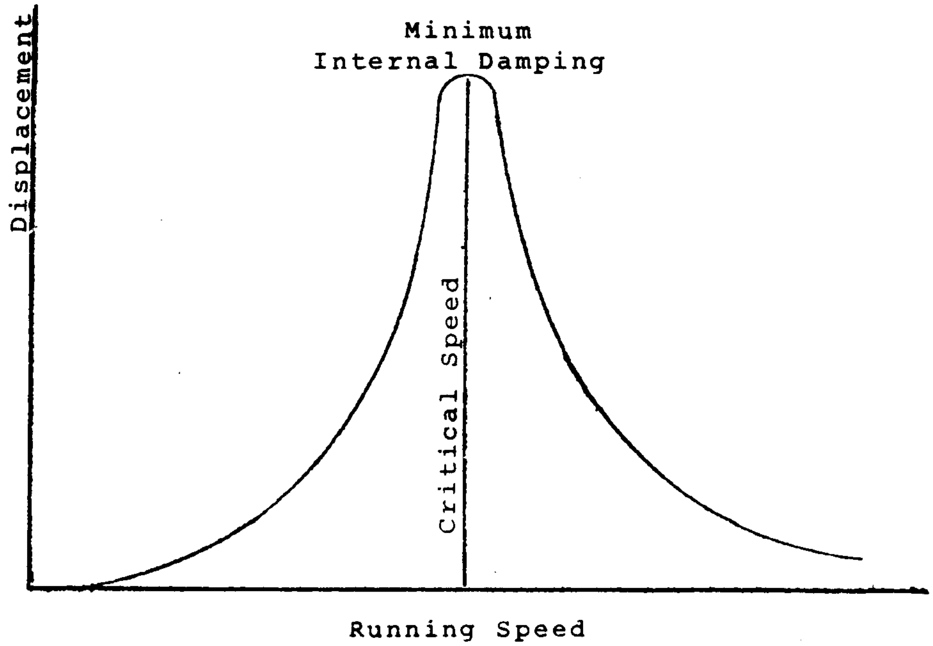


Figure 8.11

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% or 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 particularaly 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).

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

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.

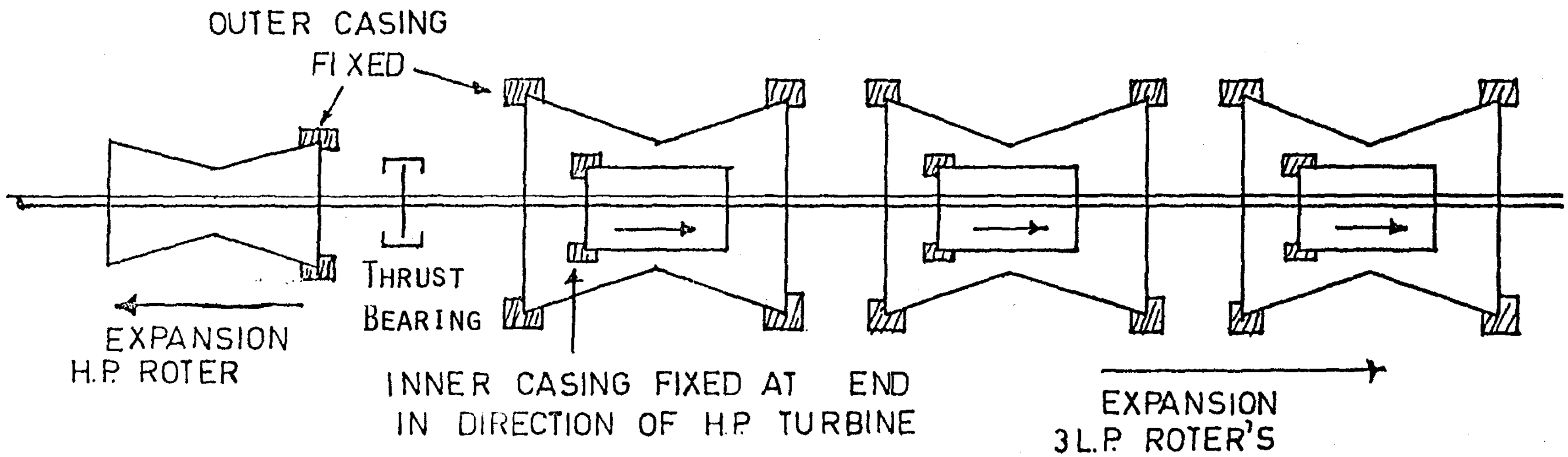
In addition to excessive heat up rates, abnormal differential expansion can be caused by water induction, thrust bearing failure, binding of components (constrained from expanding), and quenching of casings from wet lagging. By understanding the differential expansion pattern for normal conditions, these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



Fixed Points of Shaft and Casings

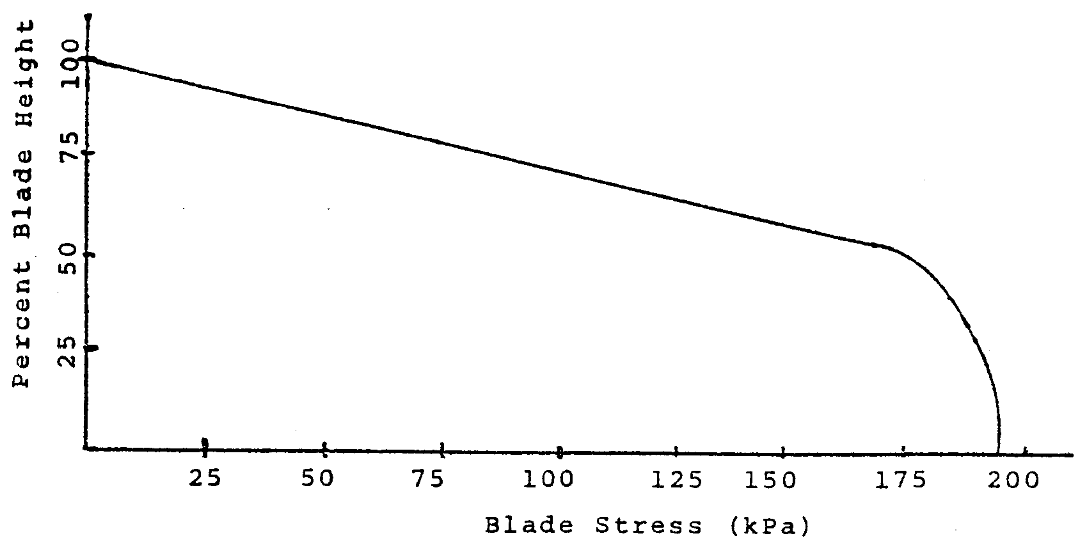
Figure 8.12

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 8.12, 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.

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 one meter long with an overall last stage diameter of 3.6 meters is in excess of 175,000 kPa (25,000 psi). Figure 8.13 shows the distribution of stress along the length of the blade.



Stress Profile in Long Turbine Blade

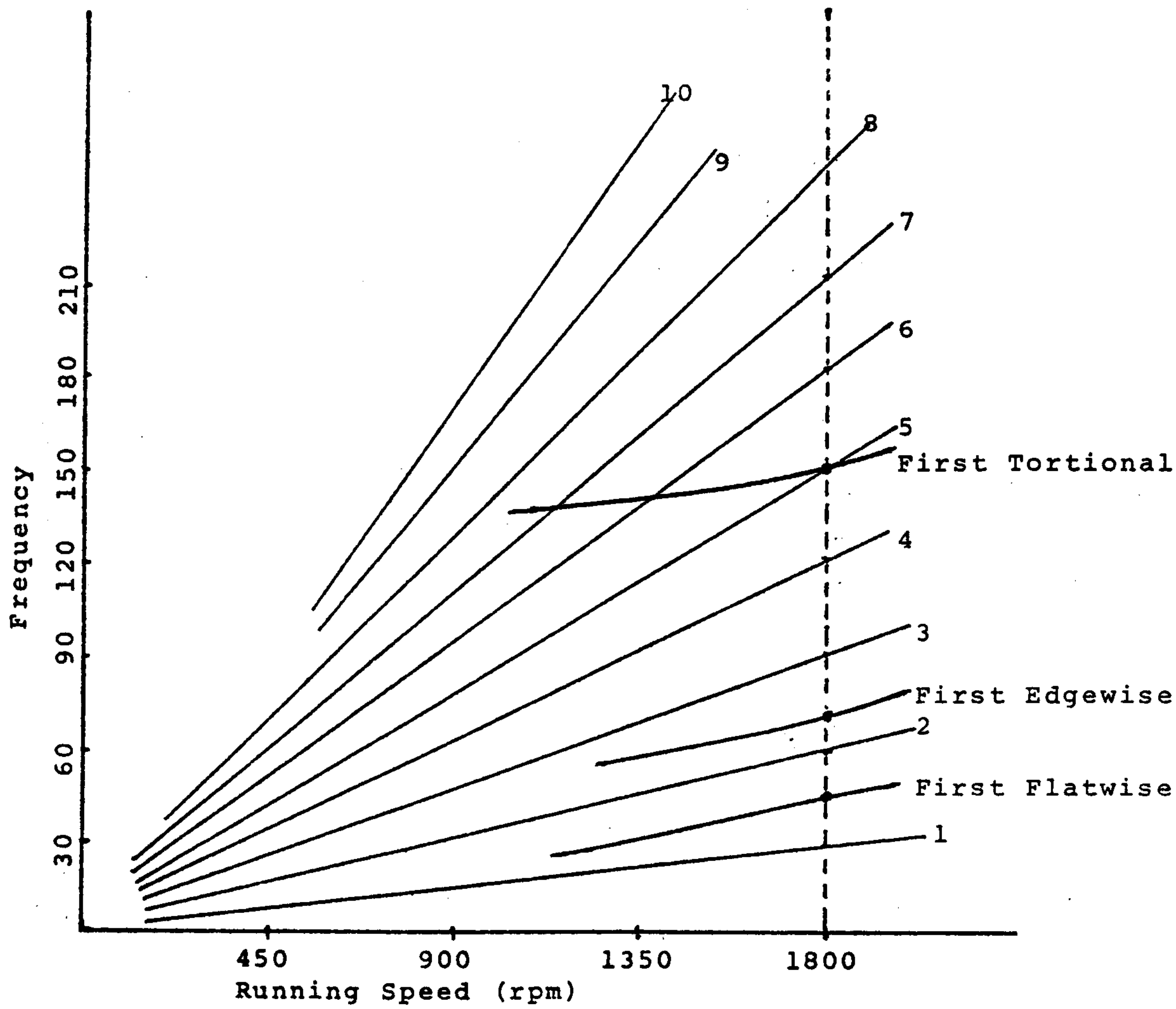
Figure 8.13

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 8.14. 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 second harmonic (60 Hz) is optimal. The coincidence of the first torsional resonant frequency and the fifth 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 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.



Campbell Diagram

Figure 8.14

Another phenomenon of low oil temperature is bearing instability caused by oil whip or oil whirl. This oil whip is shown in Figure 8.15 and results from high oil viscosity at low oil temperature. 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.

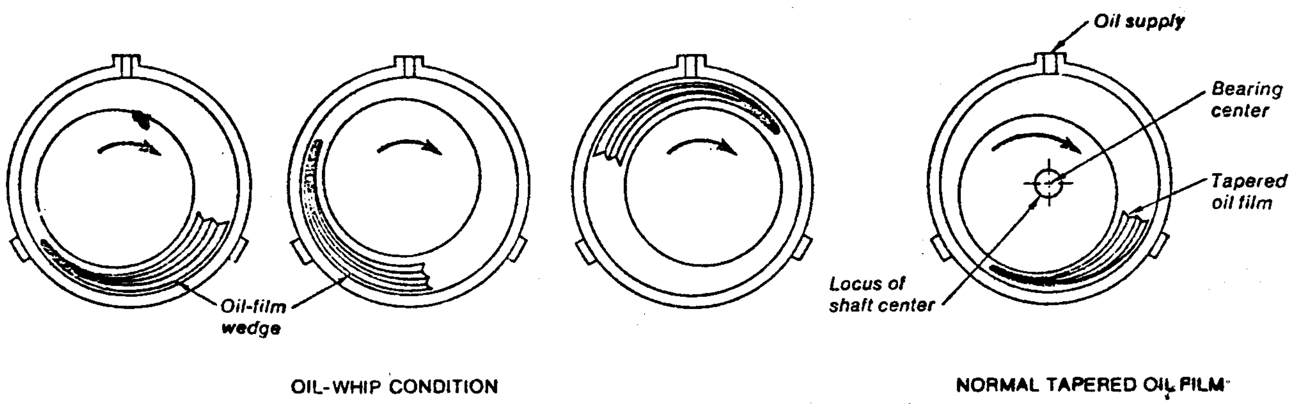


Figure 8.15

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 seven minutes.

   (a) What problems does this present?

   (b) What action would you take?

6. A turbine is started up and loaded at a rate in excess of that specified by the Operating Manual. What problems may this cause:

   (a) for a single startup?

   (b) if continued for a number of startups?

R.O. Schuelke

Turbine, Generator & Auxiliaries - Course 334

ONTARIO HYDRO NUCLEAR TURBINE TYPES

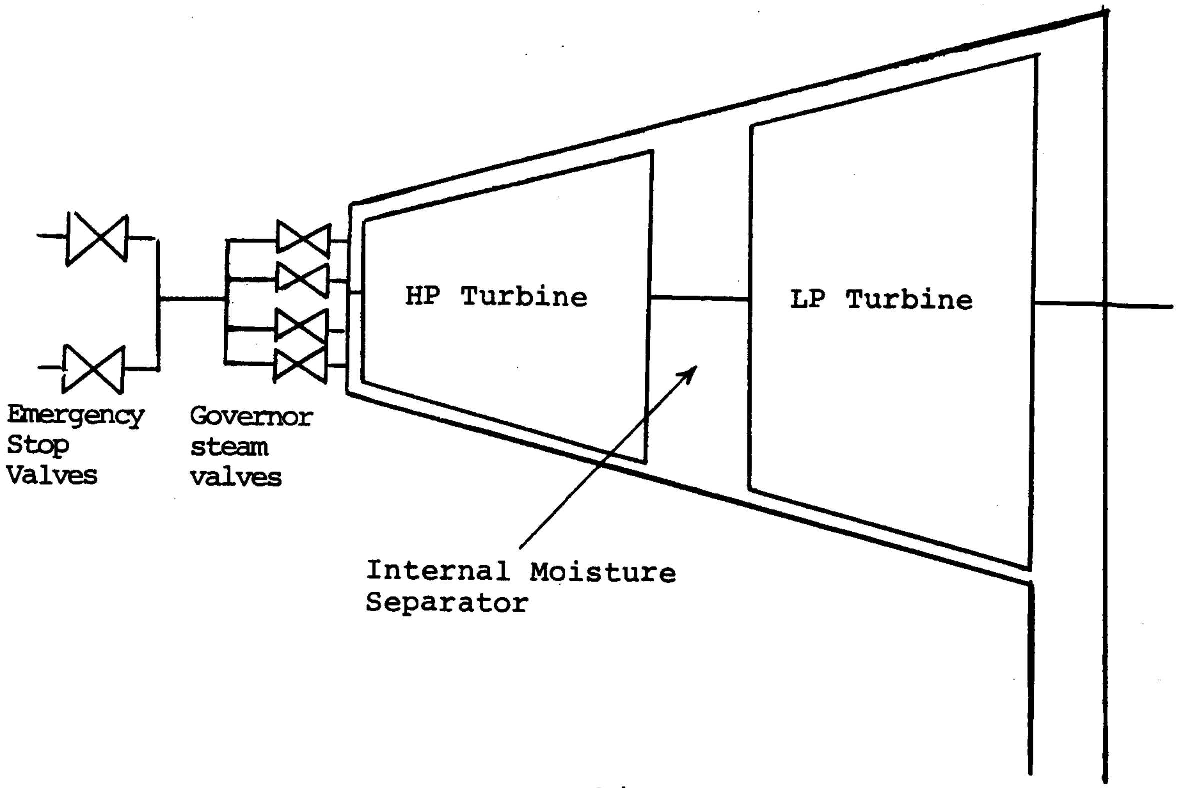
As a review of all we have covered in the course, we will examine each of the generating stations within the nuclear generation division. For each turbine unit, we will look at the topics covered in the course and see how the particular unit is constructed. The topics we will examine are:

1. number of HP and LP turbines,
2. type of steam to HP and LP turbines,
3. type and number of moisture separators,
4. type and number of reheaters,
5. type of turbine blading,
6. type of gland seals,
7. type of governor valves,
8. type of speed governing,
9. type of generator cooling system,
10. type of vacuum raising and maintaining plant,
11. type of feedheating system,
12. steam pressure, steam temperature, speed, generator rated output and generated voltage.

NPD Nuclear Generating Station

1. One single flow high pressure turbine and one single flow low pressure turbine. Both turbines are contained within a common casing.
2. Dry saturated steam enters the high pressure turbine. The HP exhaust is wet steam which passes through an internal moisture separator where the majority of the moisture is removed. The essentially dry steam then enters the LP turbine and exhausts as wet steam to the condenser.

3. One centrifugal type water separator is located within the common casing between the HP and LP turbine.



NPD NGS Turbine

Figure 13.1

4. Steam reheaters are not used.

5. The high pressure turbine has impulse blading. The low pressure turbine has reaction blading.

6. The high pressure turbine has a steam sealed labyrinth gland. The low pressure turbine has a centrifugal water gland.

7. The turbine has four nozzle type governor steam valves.

8. The speed governor is centrifugal fly-ball.

9. The generator is air cooled.

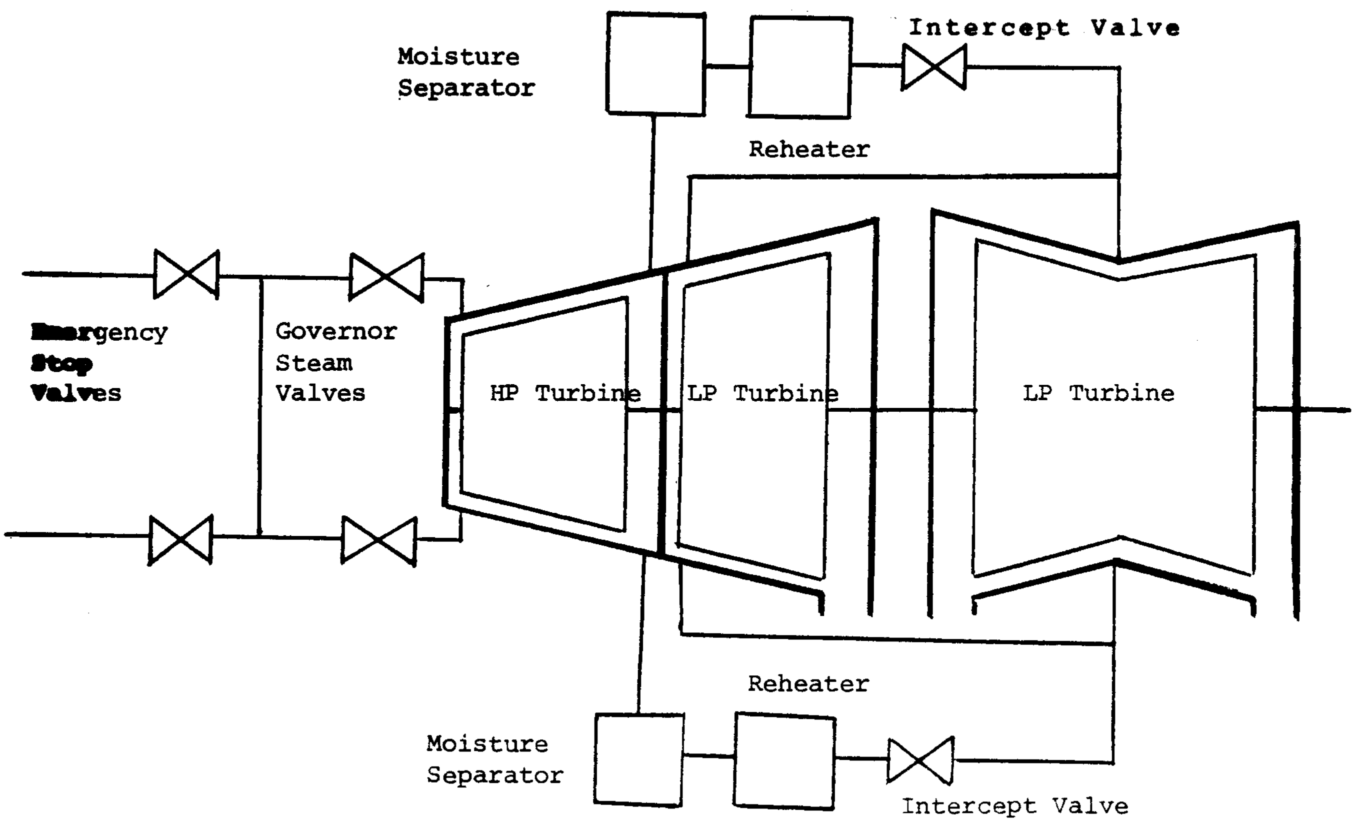
10. A steam air ejector is used for initial vacuum raising. The vacuum is maintained by motor driven vacuum pumps.

11. A regenerative feedheating system is used with the following feedheaters:

    feedheater drain flash condenser
    one LP feedheater
    gland exhaust condenser
    two HP feedheaters
    preheater internal to the steam generator

12. HP turbine inlet: 2760 kPa(a) and 230°C
    HP turbine exhaust: 280 kPa(a), 131°C and 11% moisture
    LP turbine inlet: 270 kPa(a), 131°C and .5% moisture
    LP turbine exhaust: 5 kPa(a) and 33°C
    Speed: 3600 rpm
    Output: 22 MW(e)
    Voltage: 13,800 volts

Douglas Point Nuclear Generating Station



Douglas Point NGS Turbine

Figure 13.2

1. One single flow high pressure turbine, one single flow low pressure turbine and one double flow low pressure turbine.

2. Dry saturated steam enters the high pressure turbine. The HP exhaust is wet steam which passes to moisture separators where the majority of the moisture is removed. The essentially dry steam then enters a live steam reheater where it is heated approximately 70°C above the saturation temperature. This superheated steam then enters the LP turbine and exhausts as wet steam to the condenser.

3. Two centrifugal type water separators. (One separator located in each of the two exhaust lines from the HP turbine.)

4. Two tube in shell live steam reheaters (one reheater in each of the two HP turbine exhaust lines downstream of the moisture separators).

5. The high pressure turbine has impulse blading. The low pressure turbines have reaction blading.

6. The HP turbine gland is a combination of steam labyrinth gland and water gland. The LP turbine glands are water glands.

7. The turbine has two throttle type governor steam valves.

8. The speed governor is centrifugal flyball.

9. The generator rotor is hydrogen cooled; the generator stator is water cooled.

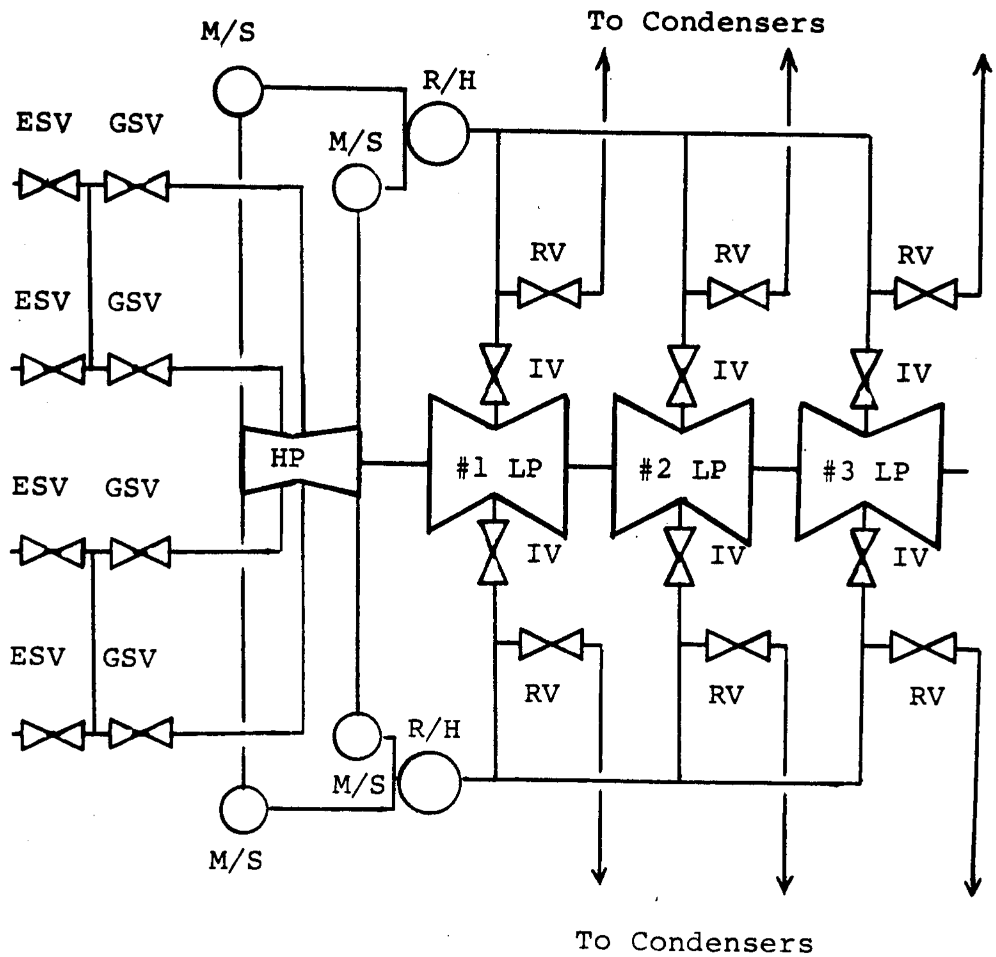
10. A steam hogging ejector is used for initial vacuum raising. The vacuum is maintained by steam maintaining ejectors.

11. A regenerative feedheating system is used with the following feedheaters:

    stator water coolers
    air ejector condenser
    gland exhaust condenser
    drain cooler
    three LP feedheaters
    deaerator
    two HP feedheaters
    preheater internal to the steam generator

12. HP turbine inlet: 3880 kPa(a) and 248°C
HP turbine exhaust: 410 kPa(a), 144°C and 12% moisture
LP turbine inlet: 400 kPa(a) and 221°C
LP turbine exhaust: 3.5 kPa(a), 25°C and 12% moisture
Speed: 1800 rpm
Output: 220 MW(e)
Voltage: 18,000 volts

Pickering Nuclear Generating Station A



ESV = Emergency Stop Valve
GSV = Governor Steam Valve
M/S = Moisture Separator
IV = Intercept Valve
RV = Release Valve
R/H = Reheater

Pickering NGS-A Turbine

Figure 13.3

1. One double flow high pressure turbine and three double flow low pressure turbines.

2. Dry saturated steam enters the high pressure turbine. The HP exhaust is wet steam which passes to moisture separators where the majority of the moisture is removed. The essentially dry steam then enters a live steam reheater where it is heated approximately 75°C above the saturation temperature. This superheated steam then enters the LP turbine and exhausts as wet steam to one of the condensers located under each LP turbine.

3. Four centrifugal type water separators (one separator located in each of the four exhaust lines from the HP turbine).

4. Two tube in shell live steam reheaters (one reheater on each side of the turbine receives the combined exhaust from two moisture separators).

5. The high pressure turbine has reaction blading. The low pressure turbines have reaction blading.

6. The HP and LP turbine glands are steam sealed labyrinth glands.

7. Four throttle valves are used for governor steam valves.

8. The speed governor is centrifugal flyball.

9. The generator rotor is hydrogen cooled; the generator stator is water cooled.

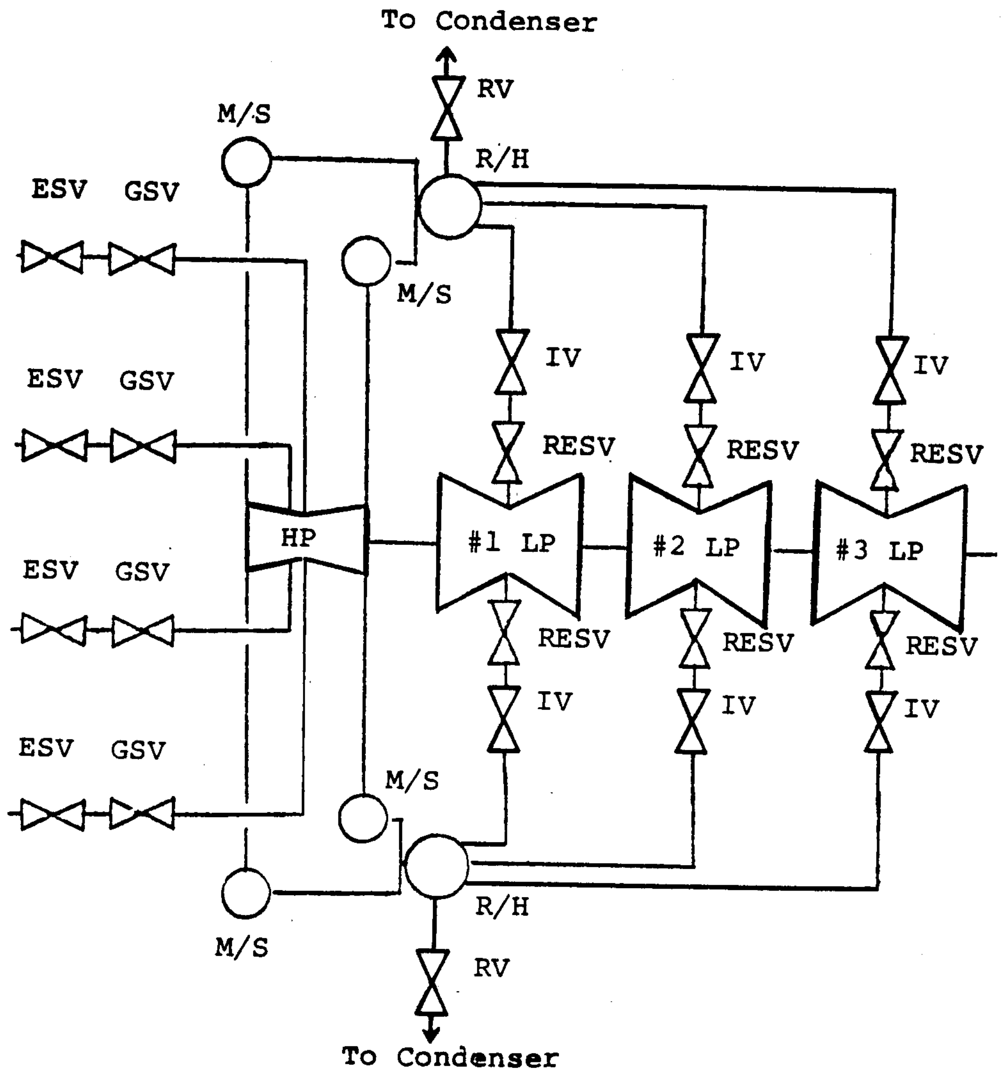
10. Three motor driven screw type vacuum pumps are used for vacuum raising; one of these pumps is used for maintaining vacuum with the other two in standby.

11. A regenerative feedheating system is used with the following feedheaters:

    gland exhaust condenser
    drain cooler
    three LP feedheaters
    deaerator
    two HP feedheaters
    preheater internal to steam generator

12. HP turbine inlet: 4030 kPa(a) and 251°C
    HP turbine exhaust: 508 kPa(a), 150°C and 11% moisture
    LP turbine inlet: 442 kPa(a) and 224°C
    LP turbine exhaust: 5 kPa(a), 33°C and 9% moisture
    Speed: 1800 rpm
    Output: 540 MW(e)
    Voltage: 24,000 volts

Bruce Nuclear Generating Station A



ESV = Emergency Stop Valve
GSV = Governor Steam Valve
M/S = Moisture Separator
R/H = Reheater

IV = Intercept Valve
RV = Release Valve
RESV = Reheat Emergency Stop Valve

Bruce NGS-A Turbine

Figure 13.4

1. One double flow high pressure turbine and three double flow low pressure turbines.

2. Dry saturated steam enters the high pressure turbine. The HP exhaust is wet steam which passes to moisture separators where the majority of the moisture is removed. The essentially dry steam then enters a live steam reheater where it is heated approximately 65°C above the saturation temperature. This superheated steam then enters the LP turbine and exhausts as wet steam to one of the condensers located under each LP turbine.

3. Four centrifugal type water separators (one separator located in each of the four exhaust lines from the HP turbine). Two auxiliary separators on the HP turbine.

4. Two tube in shell live steam reheaters (one reheater on each side of the turbine receives the combined exhaust from two main moisture separators).

5. The high pressure turbine has reaction blading. The low pressure turbines have reaction blading in all stages except the first stage which is an impulse stage.

6. The HP and LP turbine glands are steam sealed labyrinth glands.

7. Four throttle valves are used for governor steam valves.

8. The speed governor is electric using a toothed wheel and proble for speed sensing.

9. The generator rotor is hydrogen cooled; the generator stator is water cooled.

10. A steam hogging ejector is used for initial vacuum raising. The vacuum is maintained by steam maintaining ejectors.

11. A regenerative feedheating system is used with the following feedheaters:

    gland exhaust condenser
    air ejector condenser
    drain cooler
    three LP feedheaters
    deaerator
    one HP feedheater
    preheater external to steam generator

12. HP turbine inlet: 4240 kPa(a) and 254°C
HP turbine exhaust: 923 kPa(a), 176°C and 9% moisture
LP turbine inlet: 835 kPa(a) and 238°C
LP turbine exhaust: 4.2 kPa(a), 30°C and 8% moisture
Speed: 1800 rpm
Output: 788 MW(e)
Voltage: 18,500 volts

ASSIGNMENT

1. For the turbine at your station (those not assigned to a generator station should use Pickering NGS-A) describe the unit using the twelve points in this lesson.

R.O. Schuelke