Turbine, Generator & Auxiliaries - Course 134

TURBINE THEORY

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

THERMODYNAMICS

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

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

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

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

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

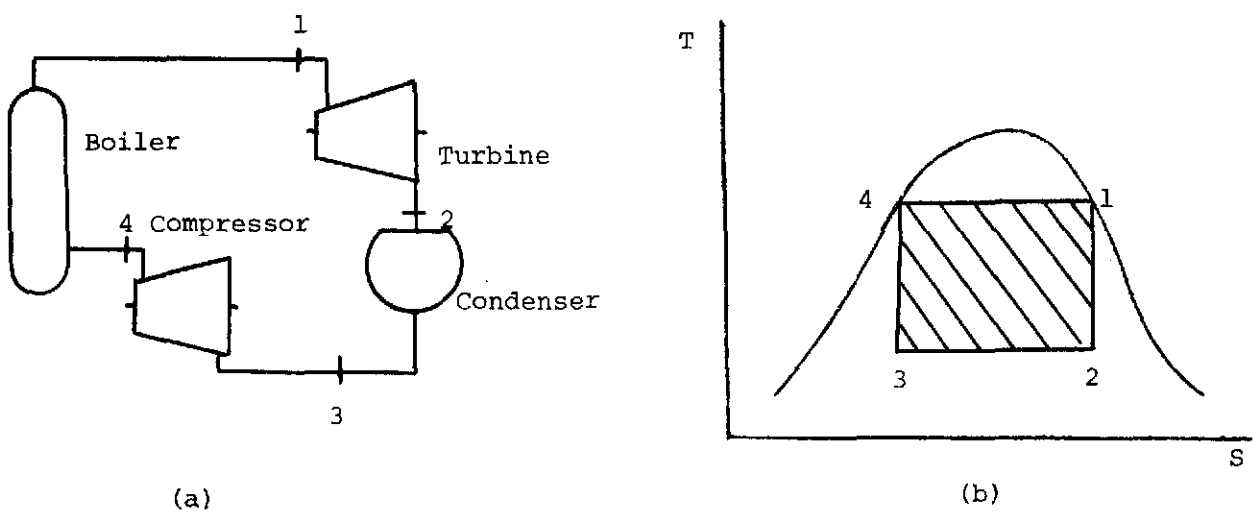
$$= \frac{217°K}{523°K}$$

$$= 42\%$$

September 1976

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

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



THE CARNOT CYCLE

Figure 1.1

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

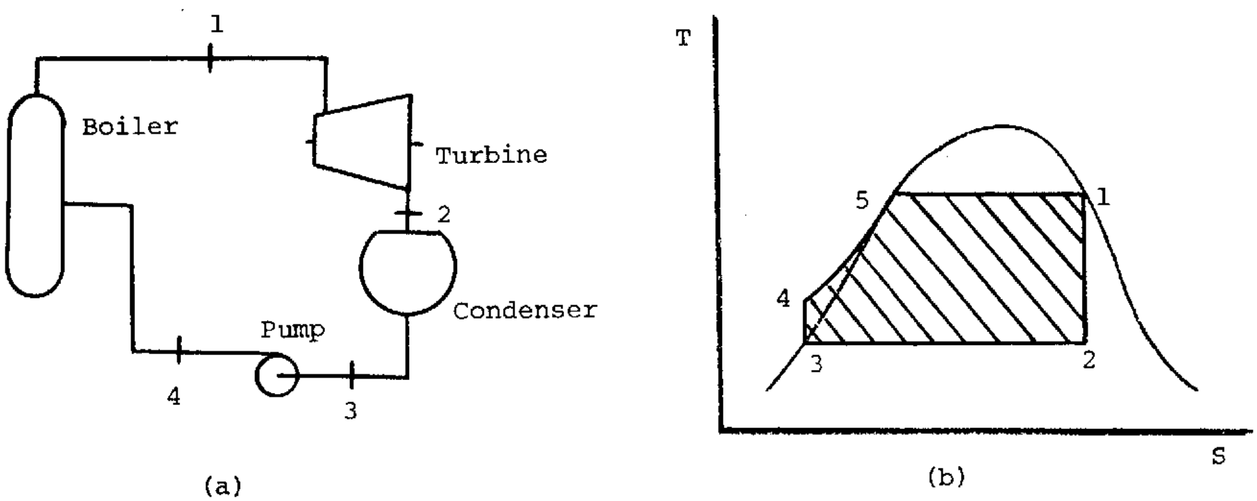
(a) it is difficult to stop the condensing process short of complete condensation to water,

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

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

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

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



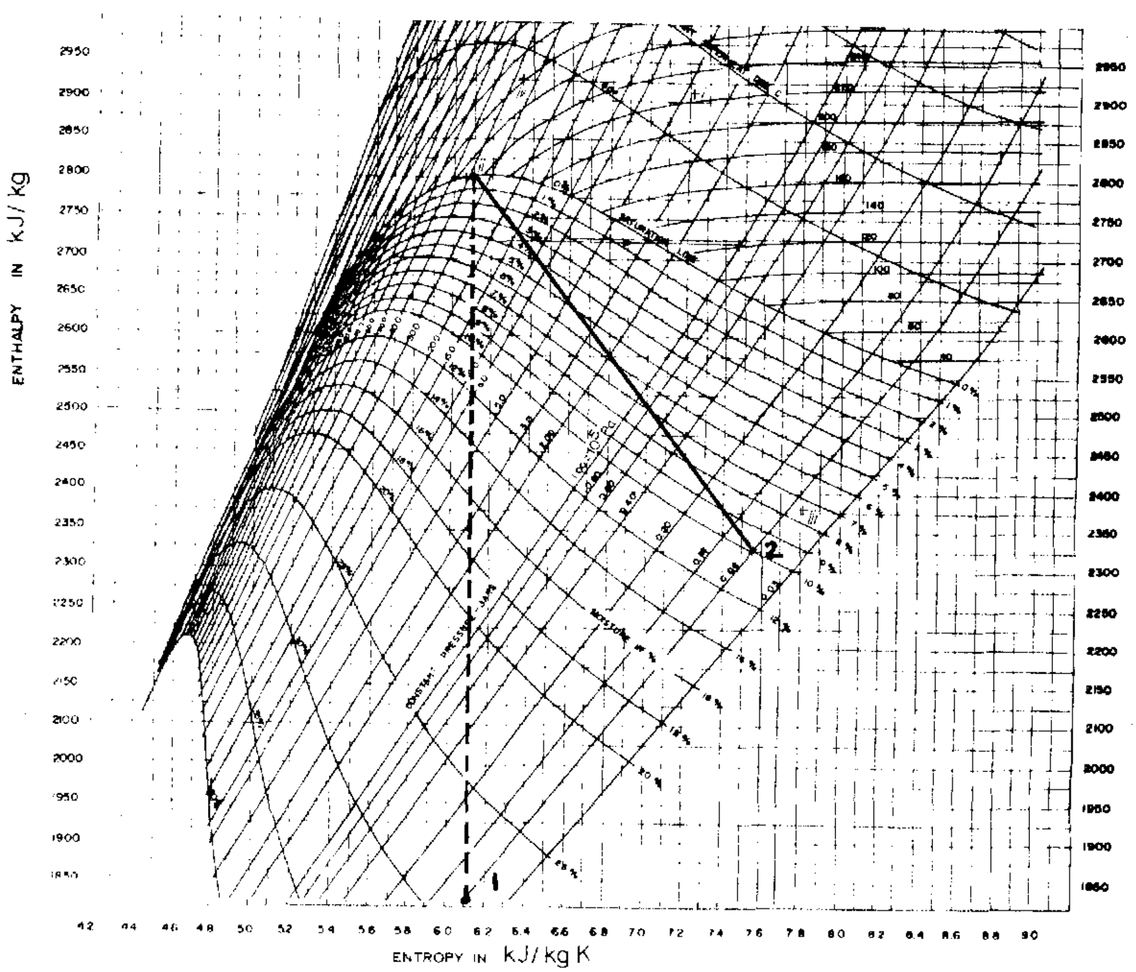
THE RANKINE CYCLE

Figure 1.2

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

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

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



<u>SINGLE HP TURBINE</u>

<u>Figure 1.3</u>

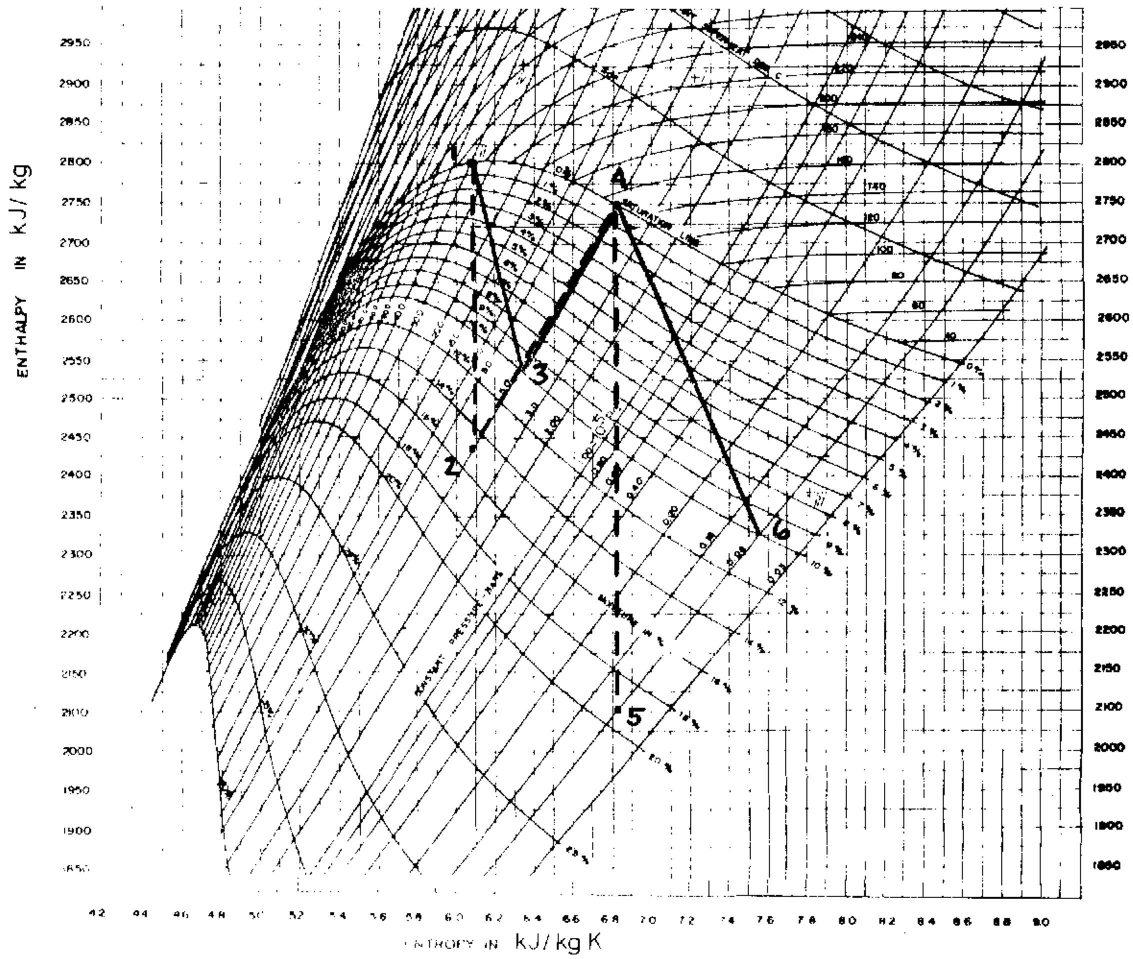
If the exhaust moisture must be limited to around 10%, as it is on most large turbine units, then the turbine must exhaust at point 2 (10% moisture, 33°C) rather than at point 1. That is, we cannot design this turbine to be isentropic because we cannot handle the increased moisture of a completely reversible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

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

MOISTURE SEPARATION

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

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



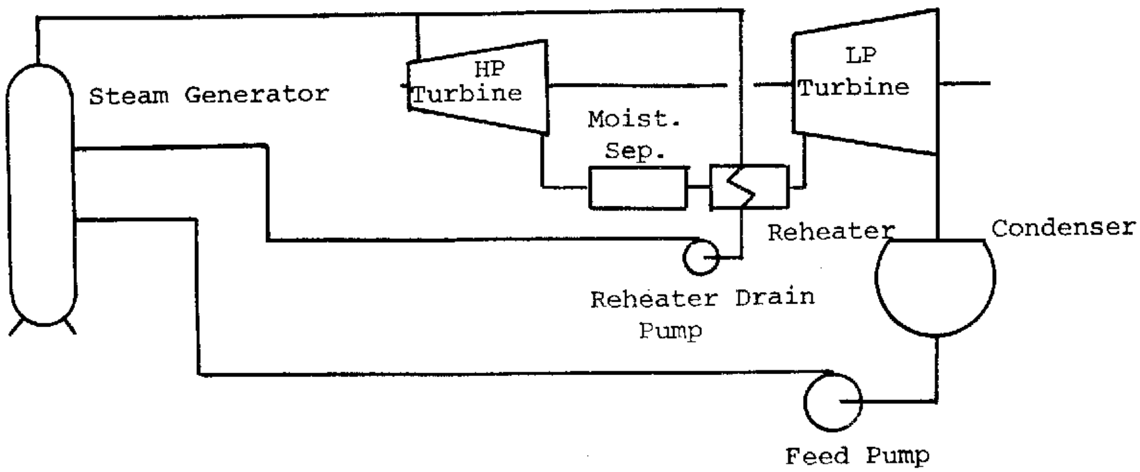
MOISTURE SEPARATOR

Figure 1.4

REHEATING

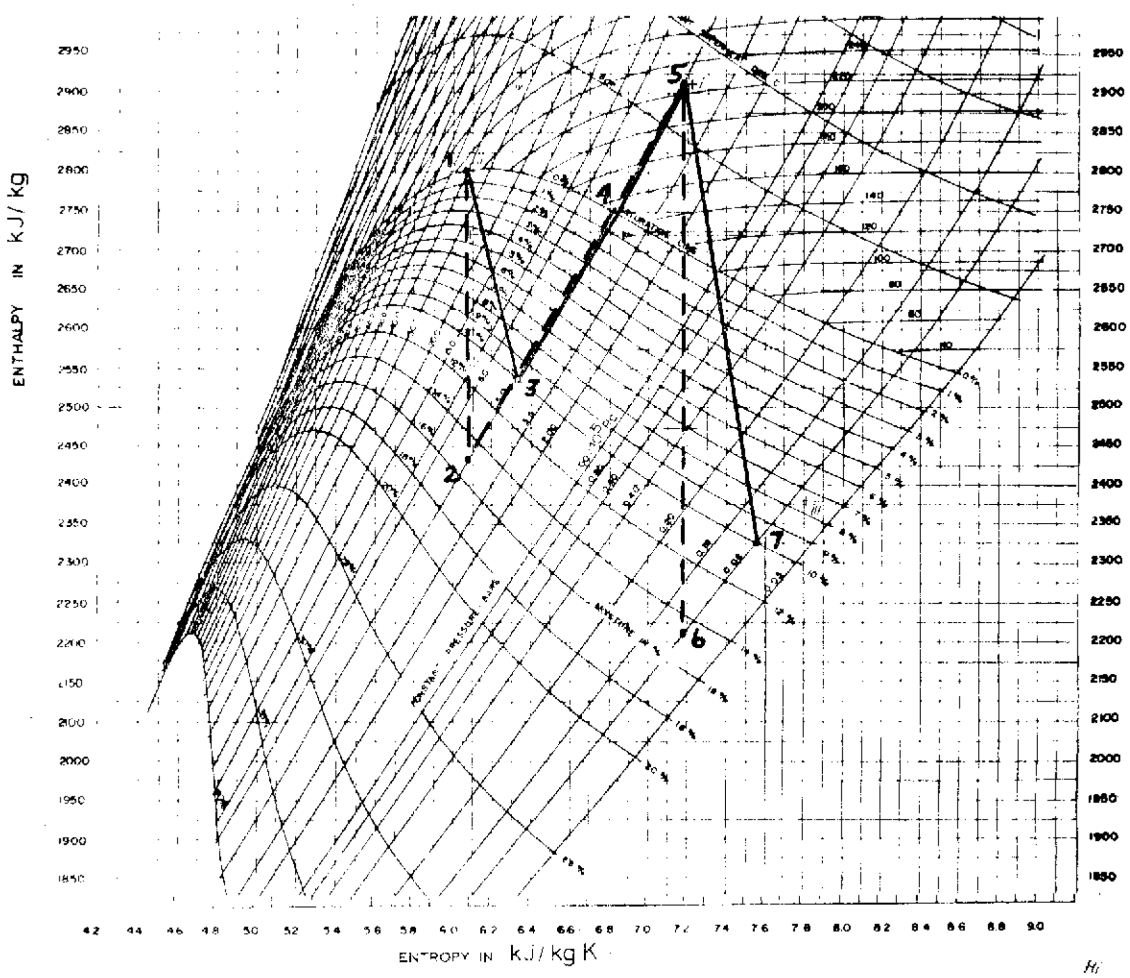
Reheating is often used to further improve the cycle efficiency. Figure 1.5 shows a typical nuclear turbine system with reheating and moisture separation.

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



SIMPLIFIED NUCLEAR STEAM CYCLE

Figure 1.5



REHEATER

Figure 1.6

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

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

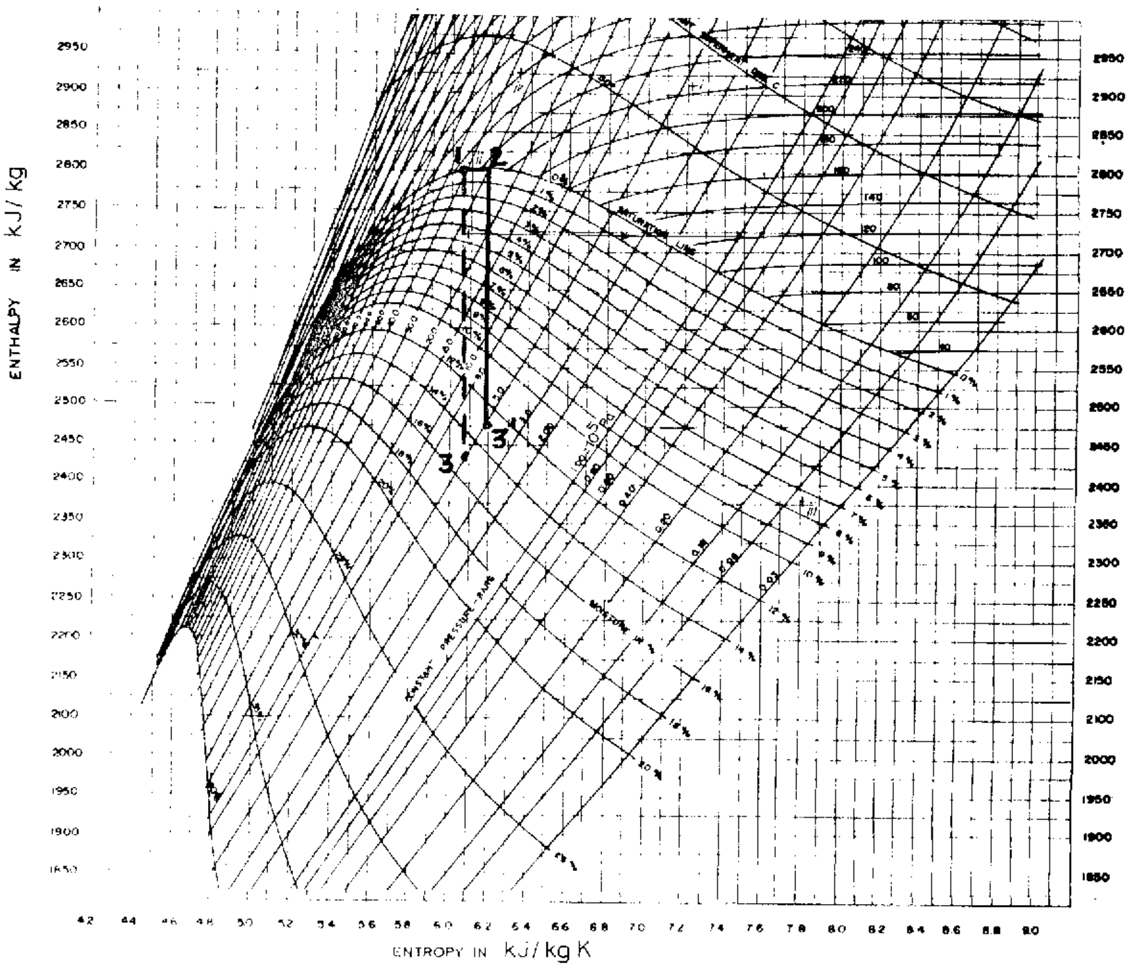
SUPERHEATING

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

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

PRESSURE DROPS IN PIPING AND VALVES

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



INLET VALVE PRESSURE DROP

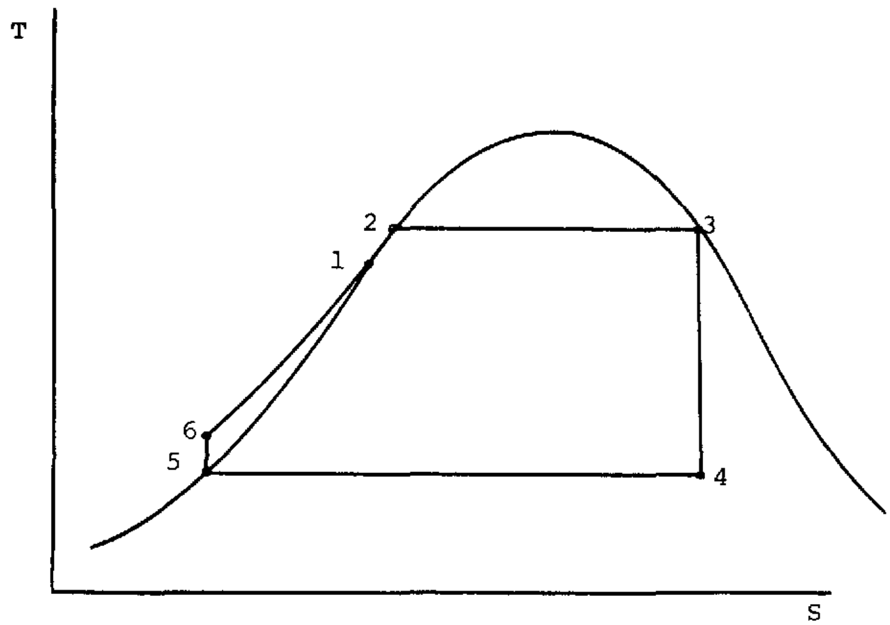
Figure 1.7

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

FEEDHEATING

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

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



EFFECT OF FEEDHEATING

Figure 1.8

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

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

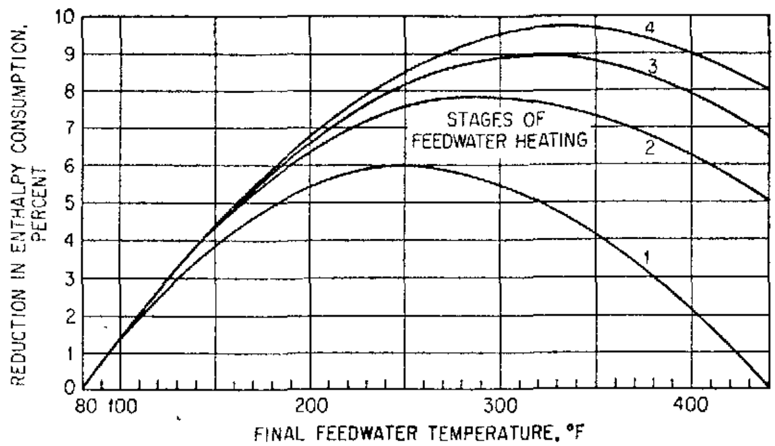


Figure 1.9

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

Examination of the curves in Figure 1.9 reveals the following:

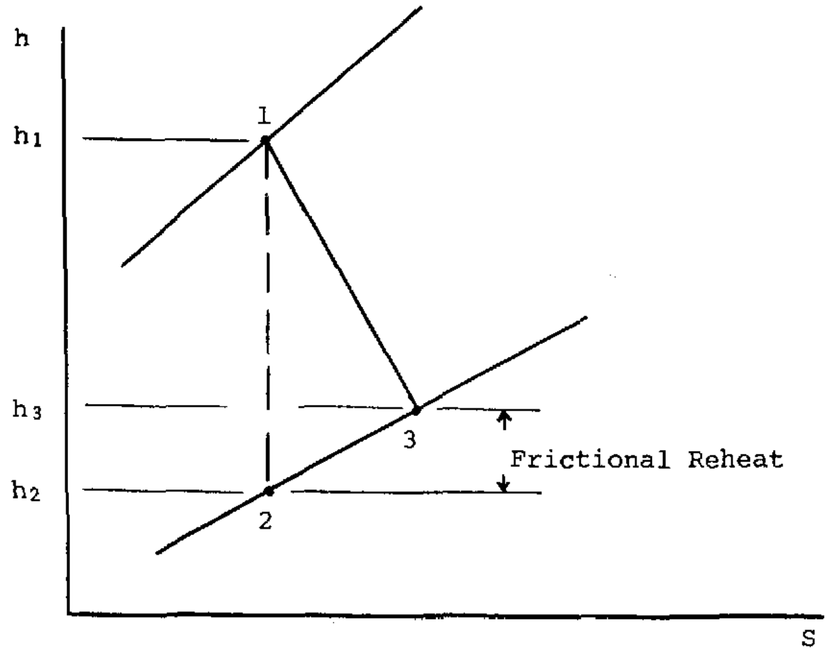
(a) as the number of feedheaters increases, the optimum temperature and, therefore, pressure of the extraction steam to the last feedheater increases,

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

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

TURBINE STAGE EFFICIENCY

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



STAGE EFFICIENCY

Figure 1.10

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

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

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

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

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

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

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

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

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

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

THE TURBINE CONDITION LINE

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

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

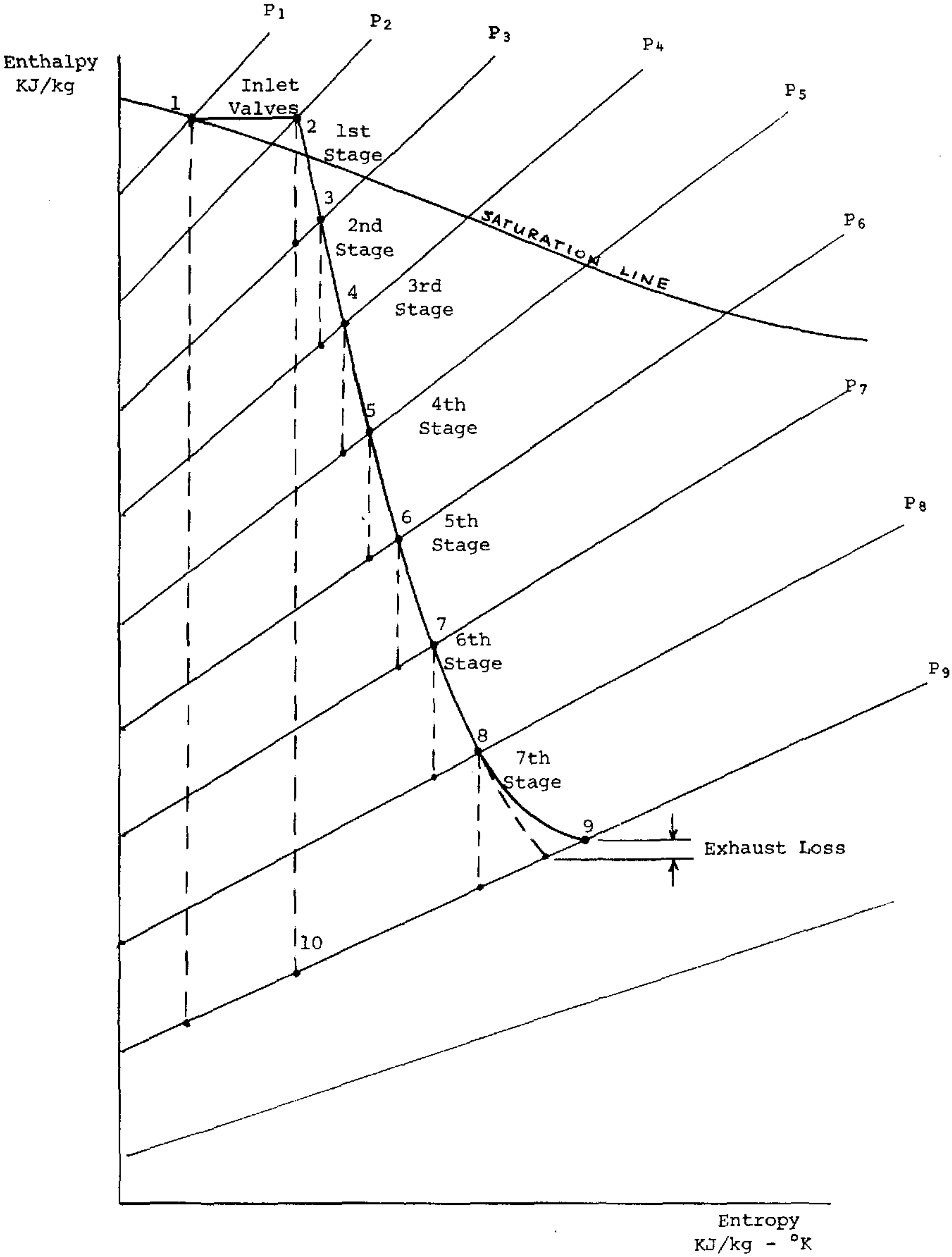

Enthalpy
KJ/kg
P1
P2
P3
P4
P5
P6
P7
P8
P9
1
Inlet
Valves
2
1st
Stage
3
2nd
Stage
4
3rd
Stage
5
4th
Stage
6
5th
Stage
7
6th
Stage
8
7th
Stage
9
10
SATURATION LINE
Exhaust Loss
Entropy
KJ/kg - °K

Figure 1.11

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

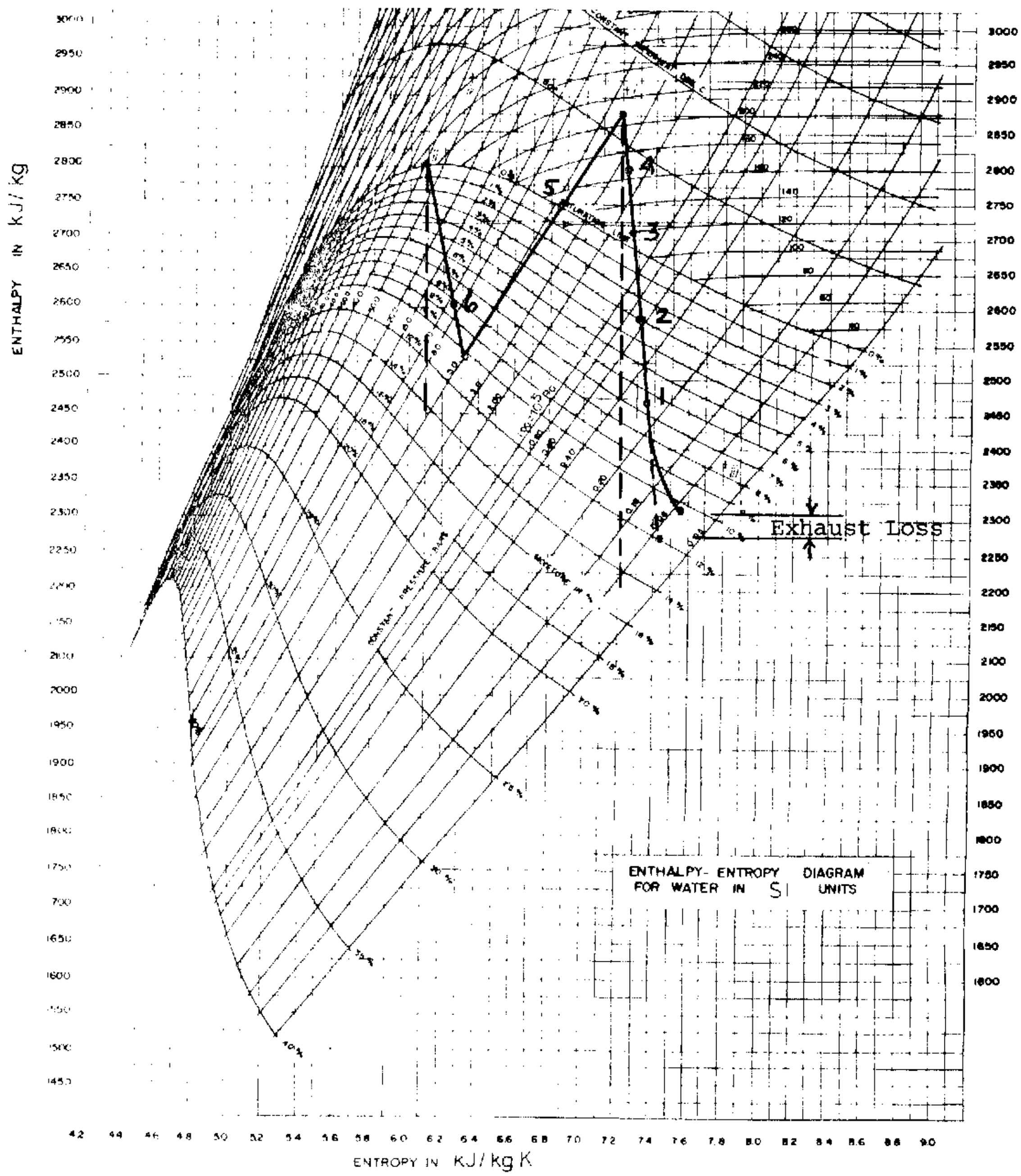


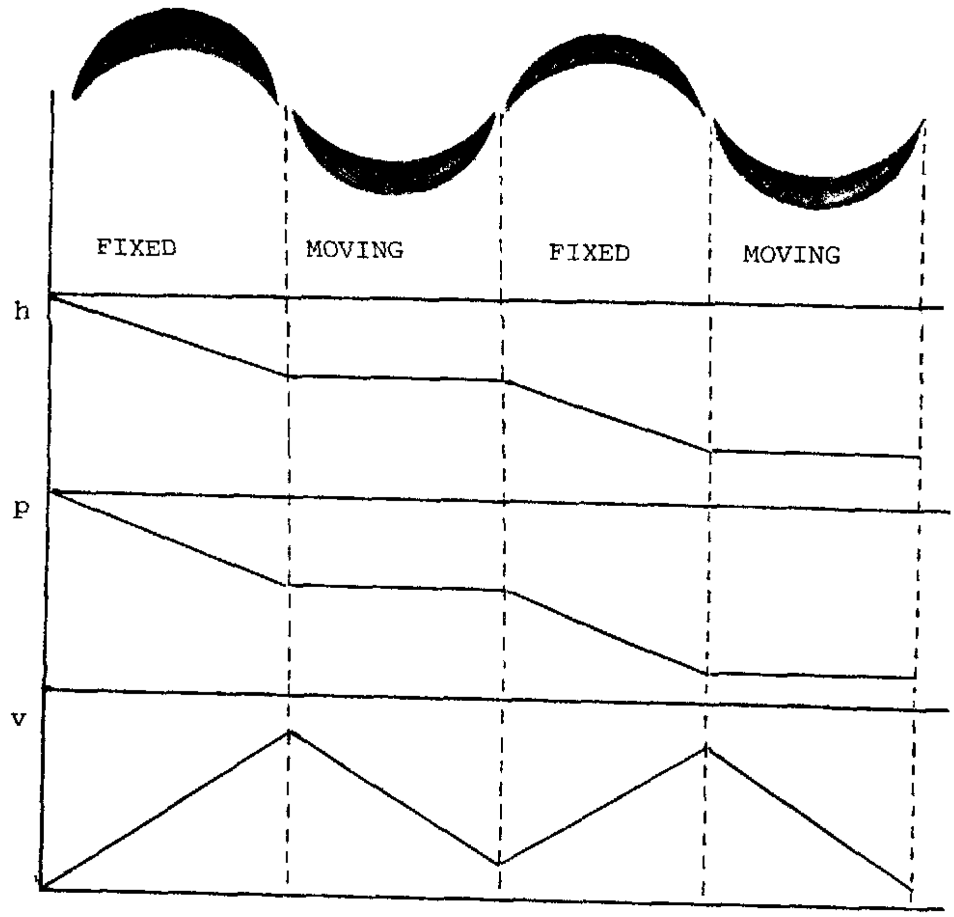
Figure 1.12

TURBINE STAGE TYPES

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

THE IMPULSE STAGE

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

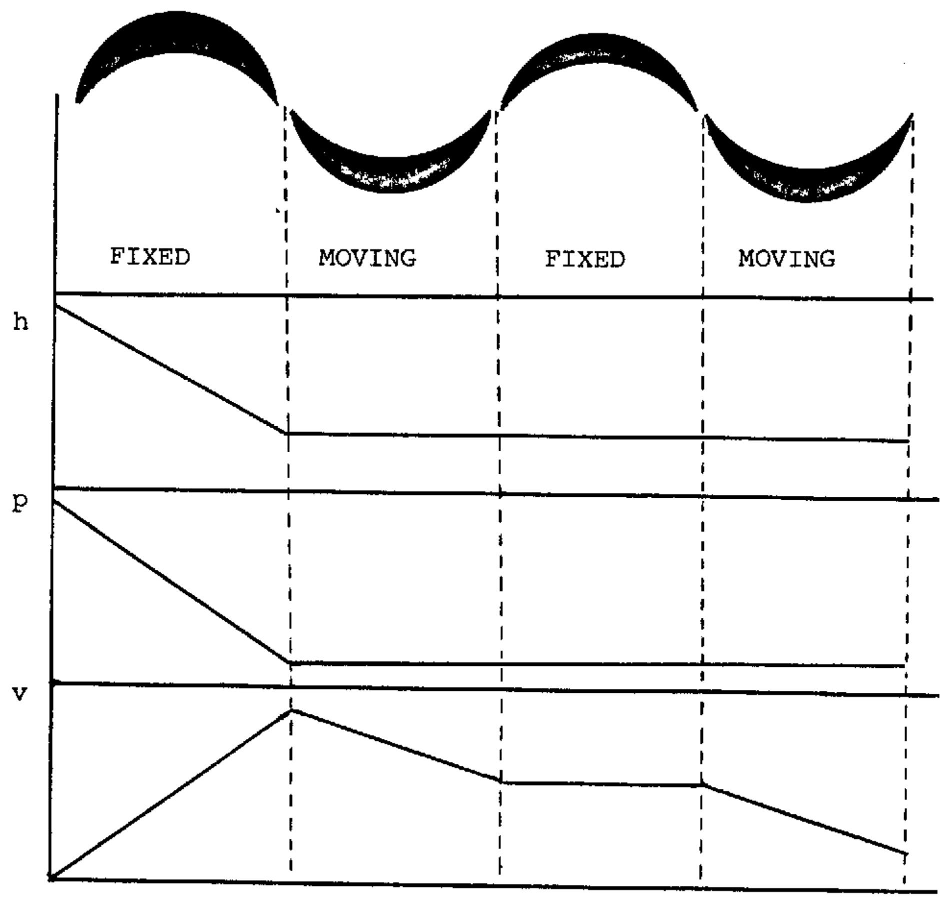


RATEAU STAGES

Figure 1.13

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

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



TWO STAGE CURTISS WHEEL

Figure 1.14

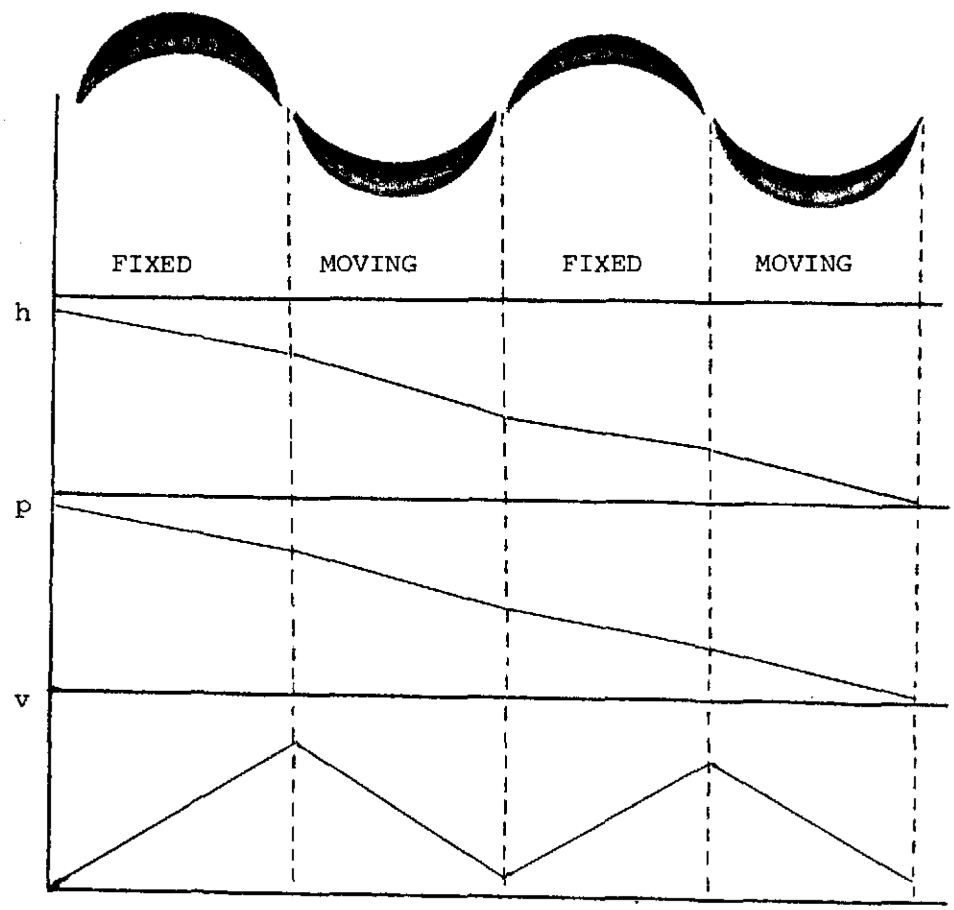
The second set of fixed blades are not nozzles and only serve to redirect the steam into the second set of moving blades. Because the second fixed blades only redirect the steam there is no change in steam velocity across these blades. The turbine shown in Figure 1.14 is a two stage turbine. These two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

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

THE REACTION STAGE

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

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



REACTION STAGES

Figure 1.15

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

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

CHOICE OF TURBINE STAGE

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

AXIAL THRUST

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

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

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

EFFICIENCY AND ENTHALPY DROP

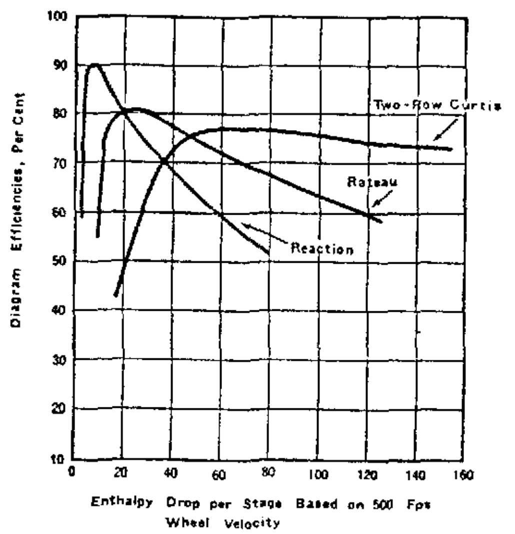


Figure 1.16

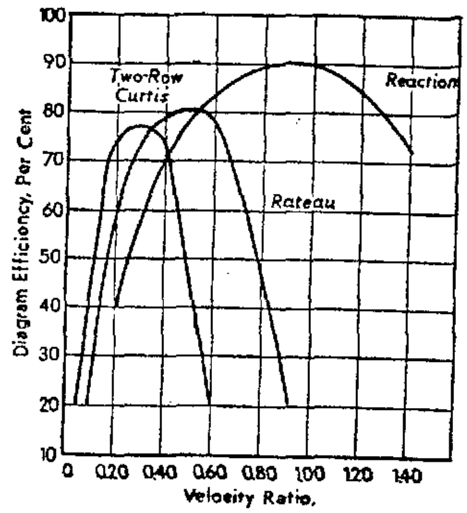


Figure 1.17

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

VELOCITY RATIO

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

MOISTURE EFFECTS

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

BLADE LEAKAGE

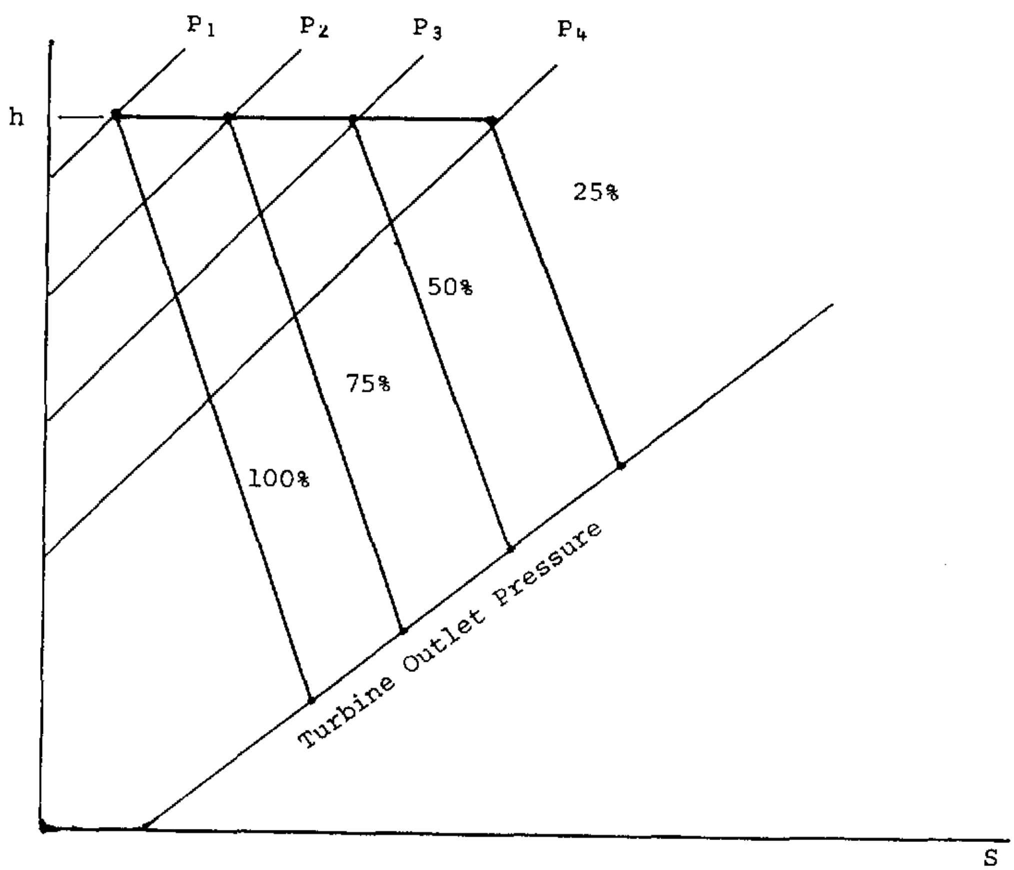
Since the reaction turbine produces a pressure drop across the moving blades, there is a tendency in the reaction turbine for the steam to crawl over the end of the moving blades. This effect can be quite pronounced in a high pressure turbine where the pressure drop per stage can be fairly high. This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

TYPES OF GOVERNOR VALVES

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

THROTTLE GOVERNORS

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



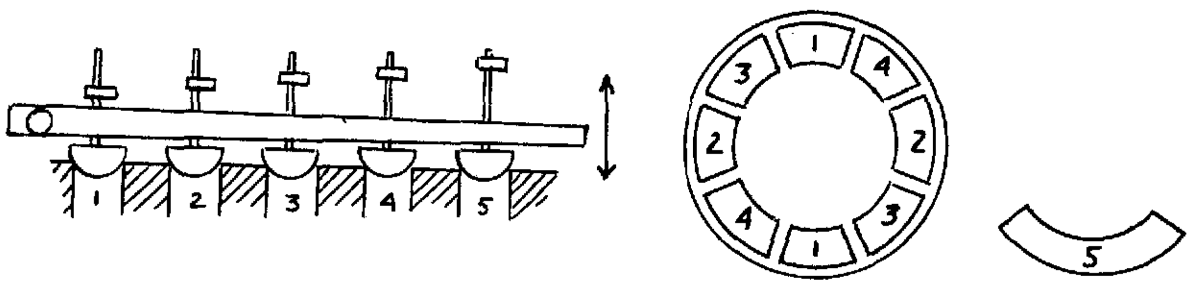
EFFECT OF THROTTLE GOVERNING

Figure 1.18

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

NOZZLE GOVERNING

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



Steam Chest With Bar Lift Nozzle Governing

First Stage Nozzles

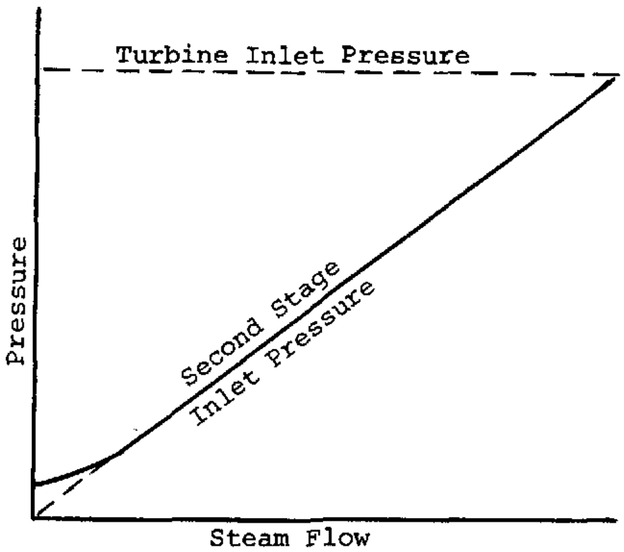
Bypass To Second Stage

NOZZLE GOVERNING

Figure 1.19

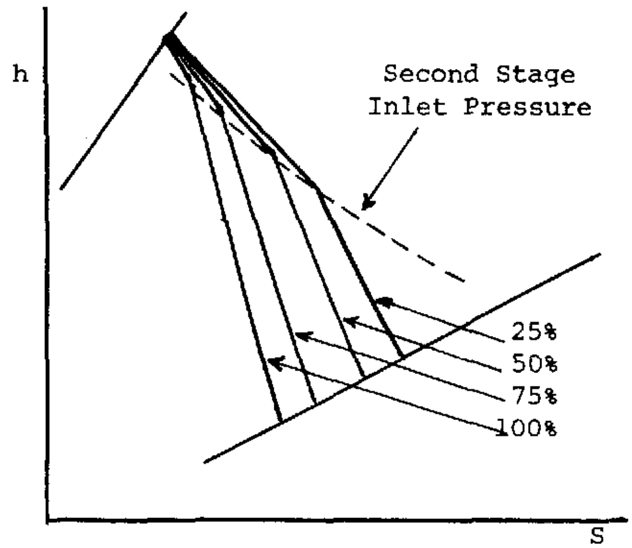
As the nozzle block moves up the valves open sequentially and as a result only one valve is throttling steam flow at any one time. This results in a nearly constant inlet pressure to the first stage and eliminates much of the adverse pressure drop associated with throttle governing at low power levels.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

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



VARIATION OF STAGE PRESSURE WITH FLOW

Figure 1.20

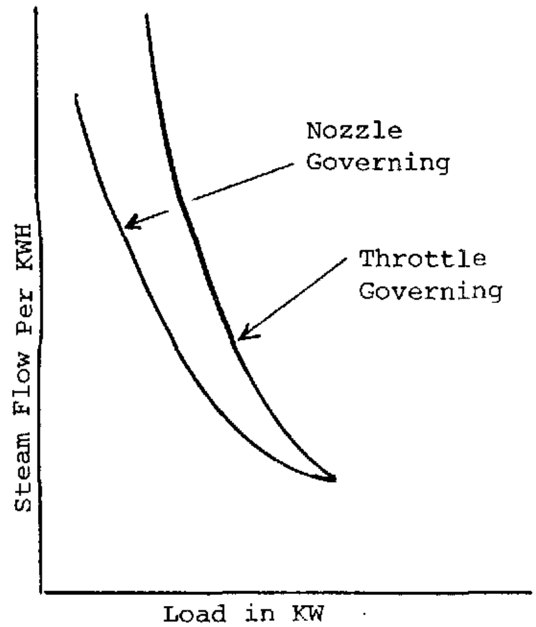


EFFECT OF NOZZLE GOVERNING

Figure 1.21

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

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



COMPARISON OF NOZZLE
AND THROTTLING GOVERNING

Figure 1.22

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

ASSIGNMENT

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

   (a) excessive moisture in the turbine.

   (b) pressure drop across the inlet valves.

   (c) moisture separator.

   (d) live steam reheater.

   (e) superheating.

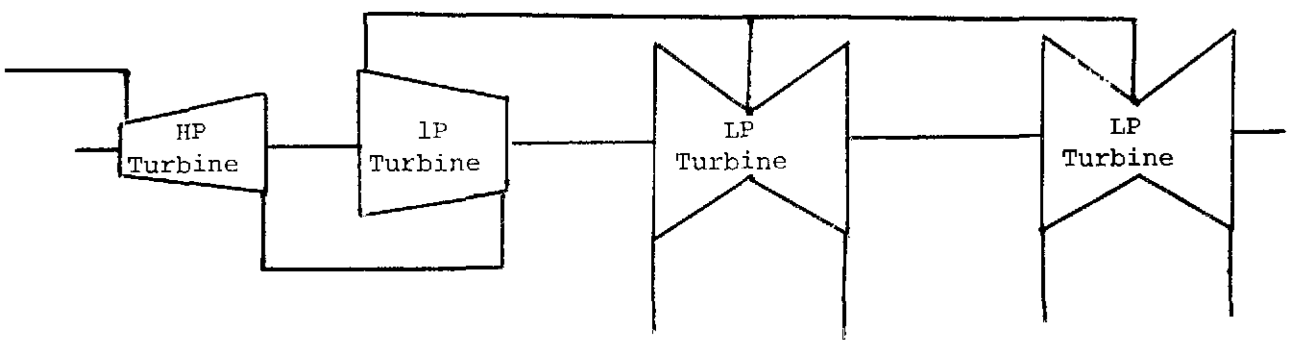
   (f) regenerative feedheating.

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

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

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

5.



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

   (a) single flow high pressure turbine.

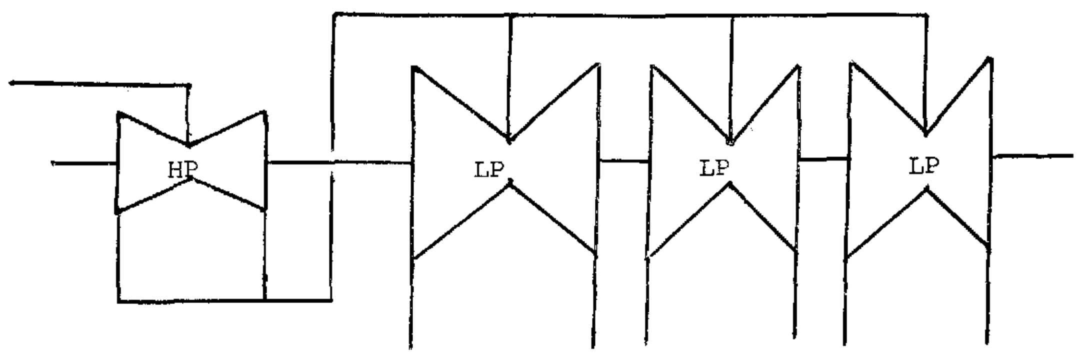
   (b) single flow intermediate pressure turbine.

(c) double flow low pressure turbine.

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

(e) Reaction stages in LP turbine.

6.



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

(a) double flow high pressure turbine.

(b) double flow low pressure turbine.

(c) reaction stages in LP turbine.

(d) reaction stages in HP turbine.

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