Applied Thermodynamics for Engineers
Dipankar N. Basu
Department of Mechanical Engineering
Indian Institute of Technology – Guwahati

Lecture – 21
Improvements and Modifications in Rankine Cycle

Good morning everyone, welcome back for the second time in the module number 7 where we are talking about the vapour power cycles. In the previous lecture you were introduced to the concept of the ideal Rankine cycle which you can say is the most popular and universally adopted cycle for steam power generation. In any kind of steam power plant whether that is thermal based or nuclear based, but Rankine cycle or modified and adopted version of the Rankine vapour power cycle is the one that is universally followed.

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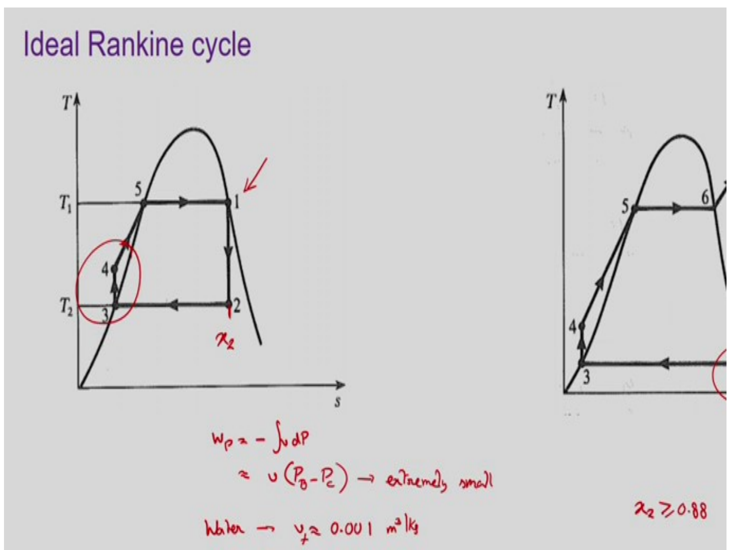

Now in the previous lecture you are introduced to this particular concept of Rankine cycle where we have seen that Rankine cycle is a vapour power cycle, that is it involves the change of phase from liquid to vapour phase and vapour to liquid phase both. That is, it inverses both evaporation and condensation processes. And generally, the cycle comprises of four processes, the ideal cycle at least, where two of them are isobaric in nature isobaric heat transfer processes one is heat addition that is evaporation and there is heat rejection that is condensation. And it also involves two isentropic processes one isentropic expansion in the turbine and one isentropic compression in the pump. And the major difference that you must have observed by now of the Rankine cycle with the Brayton cycle is in this particular process. Of course, both Rankine cycle and Brayton cycle involves two isobaric heat transfer processes and two isentropic processes. However, there are two major difference one in case of Brayton cycle our working medium is gas that is for all the processes the temperature level remains well above the critical point of the corresponding substance. And therefore, it is always treated as a gas we can treat this an ideal gas or maybe real gas whatever but that is gaseous in nature. However, for a Rankine cycle though we have the same four processes but here the temperature level is below the critical point and therefore we have the phase change involved.

And the second big difference which actually is associated with the first one is in this particular process, here the compression part is done entirely with the liquid state. Now we know that the work input requirement for the compression process is proportional to the specific volume in the Brayton cycle, as the working medium is gas with very high specific volumes. So, we have to go for something like intercooling and multistage compressions etc in order to reduce the compression work by maintaining a constant temperature or low temperature.

However here the working medium is liquid, at the end of condensation invariably the state of the working substance is that of saturated liquid that is the state point 3. And then we are doing the entire pump operation in the sub-cooled or compressed liquid zone and the liquid stage or liquid phase being incompressible in nature and the corresponding specific volume being very low. Therefore, the work input requirement for the pump in general is extremely low.

We have already seen that the work input requirement for the pump can often be calculated as:

$$W_P = -\int v dP$$

which invariably reduces to:

$$\approx v(P_B - P_C)$$

because there is hardly any change in the specific volume from state point 3 to 4. And when you are working with water, for water this v is invariably of the order of 0.001 m^3/kg, as long as we are dealing with normal pressure levels.

Unless we are going to something very close to the critical pressure, specific volume for liquid water will remain in that zone only, say:

$$\approx v_f(P_B - P_C)$$

Therefore, whatever may be the pressure difference between the boiler pressure and turbine pressure this pump work remains extremely small. So, though the level of work produced by the turbine may be quite similar between the Rankine cycle and the Brayton cycle however the pump work that is the back work is extremely small thereby giving very high net work output and work ratio and very small, negligibly small back work ratio.

Now the ideal Rankine cycle of course can have two variations as shown here. We have already discussed about both of them. In one case, in this one we call it the Rankine cycle with wet stream because at the inlet to the turbine in the state of the working medium is that of vapour and entire expansion in the turbine is done within the vapour dome, that is starting from saturated vapour to a saturated liquid vapour mixture.

In the second case, at the entry to the turbine in the state of the working medium is that of superheated vapour. So, the entire expansion state can be done on entirely on the vapour side that is like as shown in this particular diagram starting from superheated vapour to saturated vapour or sometimes we may have this expanded a bit more into a mixture zone coming somewhere here that major part done in the superheated vapour side and only a small portion in the mixture side.

And the major advantage of this super heating is in two-fold. One we are able to increase the maximum temperature of heat addition i.e., the maximum cycle temperature which we shall see be seeing the advantage of that one shortly. And the second advantage is here, like in case of wet steam the quality at this point number two your x_2 can be quite low which is much higher with superheated vapour.

And in practical operation we would always like to keep:

$$x_2 \geq 0.88$$

in order to avoid the erosion related issues. So, super heating definitely helps in that point of view. Now both the cycles are ideal cycles that is where we are not considering the irreversibilities which can be present in the compressor and pump and also the pressure drop which the fluid may experience while passing through the boiler or the condenser.

And the effect of isentropic efficiency we have seen through a numerical example. Let us see the effect of all such possible practical difficulties in another one. We are solving several numerical problems in this module in order to show you the magnitude of the effect that we are talking about.

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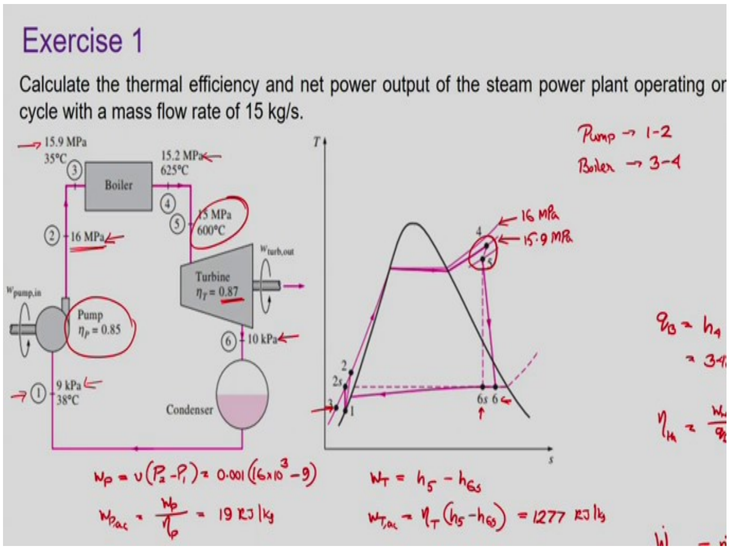

Here we are looking to identify the thermal efficiency and net power output from a steam power plant operating on a Rankine cycle with a mass flow rate of 15 kg/s and this is the cycle. Now look at this if we start from point number 1 which is here, here at the inlet to the pump the pressure is 9 kPa, at the exit it is it has been increased to 16 MPa, but the pump has an isentropic efficiency of 0.85 so there are irreversibilities. Secondly look at this 16 MPa is the pressure at the outlet of the pump, but when the liquid water gets transferred from the pump to the boiler through the pipeline because of the frictional effects there is some loss in pressure. And it enters the boiler with the pressure 15.9 MPa and 35 ^{0}C.

Remember the temperature which is not given, temperature at the exit of the pump may not be 35 ^{0}C may not be 38 ^{0}C also it can be anything else we do not know that information. Now in ideal cycle like shown in this diagram, in the ideal cycle both the heat addition in the boiler and heat rejection in the condenser are done at constant pressure. But look what we are having here at the inlet to the boiler the pressure is 15.9 MPa at the exit the pressure is 15.2 MPa.

So that is 0.7 MPa pressure loss while the fluid passes to the boiler. But it comes out to the temperature of 625 ^{0}C and proceeds to the turbine. Let me get the *Ts* diagram into picture also. Look at the diagram here point number 1 is the state before the pumping operation starts and had the pump been a perfectly reversible one, so it should have reached point number 2*s*, but because of the presence of the irreversibilities that is that 0.85 isentropic efficiencies it reaches point 2. Up to that part is fine because we have already encountered this.

But when the fluid passes from pump exit to the boiler that is further pressure loss, because of which it comes down to this particular point 3. So, 2 is the exit state of the pump that is your pump operation is associated with the process 1-2. However, the boiler operation is associated with not 2 but it starts from point 3 and proceeds to point 4 and this is your point number 4. Had the fluid entering the boiler with the same pressure of 16 MPa that is there is no pressure loss then it would have ended up somewhere here.

But this particular line is the 16 MPa line and this line is that 15.9 MPa line. So, there is a change in the location of point 4 and once it comes out of the boiler then it again has to pass through some pipelines to reach the turbine. So, there is a further drop from this point 4 to point 5. And the turbine operation does not start from point 4, it starts from point 5 which has this particular reading of 15 MPa that is there is a 0.2 MPa drop.

So, the fluid exits the pump at a pressure of 16 MPa and enters a turbine in a pressure of 15 MPa. So, there is one full mega Pascal pressure drop from pump exit to the turbine inlet, which is definitely huge loss a significant loss in exergy or work potential. Now the pumping operation proceeds from 5 to 6. Had the turbine expansion been isentropic it should have reached point 6*s* but we are reaching point number 6 because the turbine has an isentropic efficiency of 0.87. Then generally turbine exit is directly connected to the condenser, so there is no significant pressure has to be considered. So, the steam or mixture is able to enter the condenser with the same pressure of 10 kPa then it proceeds to the condensation process and reaches this point number 1.

And look at this 10 kPa is the pressure of the condenser exit and it enters the pump it enters with 9 kPa, so 1 kPa pressure loss is also present in the condenser side. So, let us try to see the effect of this irreversibilities. If we want to see the effect of this in ideal scenario let us try to work out. In ideal scenario the pump work should have been:

$$W_P = v(P_2 - P_1) = 0.001(16 \times 10^3 - 9)$$

you can calculate the exact specific volume of corresponding to this 9 kPa and 38 ^{0}C. But that will be coming in that order only. So this is the work output that we should have provided to the pump had that been perfectly reversible. But because of the presence of irreversibilities the pump work requirement in actual will be:

$$W_{P,actual} = \frac{W_P}{\eta_P}$$

and if you put the numbers this will be coming approximately 19 kJ/kg. So, this is the pump work requirement.

Similarly, the turbine work that we should have got in this case turbine work per unit mass flow rate that should have been:

$$W_T = h_5 - h_{6s}$$

but because of the presence of irreversibilities the actual work that the turbine is producing is:

$$W_{T,actual} = \eta_T(h_5 - h_{6s})$$

Now to calculate h_5 the state is given as 15 MPa and 600 ^{0}C. So, you can replicate the value of h_5 and then you can also get the value of entropy corresponding to this and using that you can identify the location of 6s. So, from there you can calculate the turbine work. See this 10 kPa is provided as a information about the point number 6 but no information is given about the quality.

So, we cannot get the value of h_6 directly we have to use the thermal efficiency for the turbine. So, this value in this particular case, I have worked out to say some time 1277 kJ/kg. Now the heat input requirement in the boiler what will be your q_B? So the exit state of the boiler is h_4 unless state is a h_3 and all information are given. Point number 4 corresponds to 15.2 MPa and 625 ^{0}C, point number 3 corresponds to 15.9 MPa and 35 ^{0}C.

$$q_B = h_4 - h_3$$

So, if you put the numbers you are going to get

$$= 3487.5\ kJ/kg$$

So, the thermal efficiency should be equal to:

$$\eta_{th} = \frac{W_{net}}{q_B} = \frac{W_{T,actual} - W_{P,actual}}{q_B} = 36.1\ \%$$

We have not calculated the condensation heat in this case but you can easily calculate also using the information from point 6 and point 1 or there is an alternative way also.

We have got the efficiency and you know that the thermal efficiency can also be written as:

$$\eta_{th} = 1 - \frac{q_C}{q_B}$$

Now you know thermal efficiency you know q_B from there also you can easily calculate total amount of heat rejected in the condenser. And the second information you have to calculate the net power output now the mass flow rate is given as 15 kg/s. So,

$$\dot{W}_{net} = \dot{m} w_{net} = \dot{m}\left(W_{T,actual} - W_{P,actual}\right) = 18.9 MW$$

So, once there are so many kinds of irreversibilities and irregularities present in the system this is an approach by which we can tackle this. But we need much more information than the ideal cycle, we need to have proper information about each of the points. So, had there been an ideal cycle this 16 MPa pressure should have been prevailed at all these points 2, 3, 4, 5 etc we do not need to have all other three pressure information. But here as the pressure is varying so you need to know pressure at all these three points.

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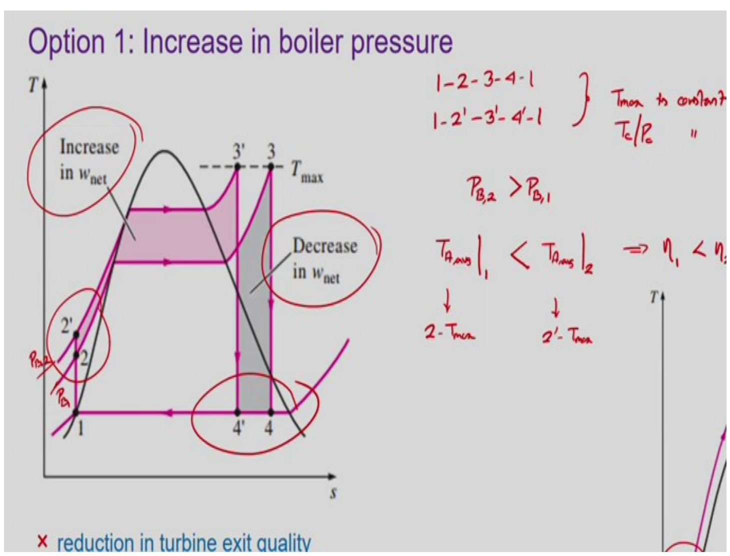

Now let us move on to check out what are the ways you can improve the efficiency of Rankine cycle. This is the standard diagram for a Rankine cycle without any irreversibilities or pressure losses. Now look at this diagram, it is a *Ts* diagram so the during the heat addition process, the temperature of the fluid starts from temperature *T2* and moves on to temperature

T_3. Whereas during the heat rejection process in ideal cycle the temperature remains more or less uniform which is T_4 and or T_1 both of them are equal.

So, the temperature of heat rejection is a constant but the temperature varies during heat addition. And therefore, the average temperature corresponding to heat addition is significantly lesser than the maximum temperature *T3*, maximum cycle temperature. Now if we compare this one with a Carnot cycle, we know that for a Carnot cycle the thermal efficiency is given as:

$$\eta_{th,Carnot} = 1 - \frac{T_R}{T_A}$$

And if we assume that the temperature is varying during a cycle like in case of a Rankine cycle, from here we can draw an implication that the thermal efficiency for a Rankine cycle should increase or even if we want to increase the efficiency we either have to increase the average temperature corresponding to heat addition or we have to reduce the average temperature corresponding to heat rejection.

Now in this particular cycle, the average temperature corresponding heat rejection is a constant because temperature is not varying. But somehow if we can reduce this temperature even further, we should have got an increase in thermal efficiency. And on the heat addition side, there is a lot of scope of improvement because there is significant temperature variation from T_2 to T_3. And therefore, we should have tried to put some effort by which we can increase the average temperature of this heat addition. Then only we can achieve some increase in the efficiency or work output of the Rankine cycle. So, let us explore a few options. The first option is to try to increase the boiler pressure. Look at this, two Rankine cycles are shown. One cycle is this 1-2-3-4-1 and the second cycle is 1-2'-3'-4'-1 and in both the cases the maximum temperature is constant. The maximum cycle temperature is constant, the condensation temperature or condenser side pressure that is also constant.

Only thing that deviates between these two cycles are the pressure on the boiler side. Let us say this particular pressure we term as P_{B1} and the second one we term as P_{B2}. So, clearly your:

$$P_{B,2} > P_{B,1}$$

then look at what we are getting from diagram point of view. the portion that is shaded in gray that corresponds to only the cycle 1 and the portion shaded in pink that corresponds to only to the cycle 2. Now the area enclosed by this cycle in *Ts* plane corresponds to the net heat interaction that is cyclic integral of δq. And as per the first law of thermodynamics for a cycle we know that this also has to be equal to:

$$\oint \delta q = \oint \delta W = W_{net}$$

Therefore area enclosed by this curve way on the *Ts* plane or area enclosed by this cycle on the *Ts* plane is also an indication of the network output that we could have got. And now once we are moving from the first cycle to the second cycle that is you are moving to a higher boiler pressure, the one shaded in grey that is the portion that you are losing now. So, that amount of work output will decrease, but the one shaded in pink that is a new addition so that is an increase in the network output. And though it is quite difficult to understand from this diagram but it can be shown that the one shaded in pink that area is at least even maybe by a smaller amount but is still higher than the area enclosed or shaded in the gray.

There is another way of showing this. If we compare these cycles the temperature corresponding heat rejection remains the same. Now which cycle has a higher average temperature for heat addition? If say T_{Aavg} for cycle 1 and T_{Aavg} for cycle 2, which one is higher? Just from the first one and this particular one varies from 2 to 3. This second one varies from 2' to 3' or 3 because 3 and 3' both are same 3'.

So, we can say that these varies from 2 to T_{max} and this one varies from 2' to again to T_{max}. So, the maximum temperature is same. What about the minimum temperature? $T_{2'}$ is definitely higher than T_2, therefore the average temperature of heat addition correspond to the second cycle definitely is greater than the same corresponding to the first cycle.

And so we can say that the efficiency corresponding to the second cycle or at least the work output that we are going to get from the second cycle is definitely will be higher. Efficiency for this cycle 1 also we will be lesser compared to the efficiency for the cycle 2. So, the first option of increasing the thermal efficiency for Rankine cycle is to increase the boiler pressure. Of course, one shortcoming that we are also getting if we look at the turbine in exit position. As you are increasing the boiler pressure maintaining the condenser pressure the same, the point 4 is shifting inwards that is towards lower quality side. So, the turbine exit quality is reducing and therefore there will be added problem with erosion.

There is another practical difficulty also as you are going to higher pressure levels then the tubings and shims that you are going to use for your plant may have to withstand much higher pressure and so you may need much costlier materials.

But the bigger problem is the reduction in the turbine exit quality. So, the increase in boiler pressure gives you the advantage of an increase in the efficiency and network output, but that causes a reduction in a turbine exit quality which is a disadvantage. In the modern times in an effort to increase the boiler pressure we have also been able to reach the supercritical pressure level where the boiler pressure is actually greater than the critical pressure of the corresponding substance.

And therefore, we are actually not going for any exclusive phase change we are just moving from point 2 to point 3 almost like a single-phase heat addition which has is some advantage of its own, but of course we have to deal with much larger pressure. Like the critical pressure for the for water is 22.1 MPa so if we want to operate the supercritical pressure level, we have to go to something like 250 bar or even higher, which is a significant pressure to deal with.

Still the supercritical power station is definitely on and is a very active area of research and has also been implemented in certain locations.

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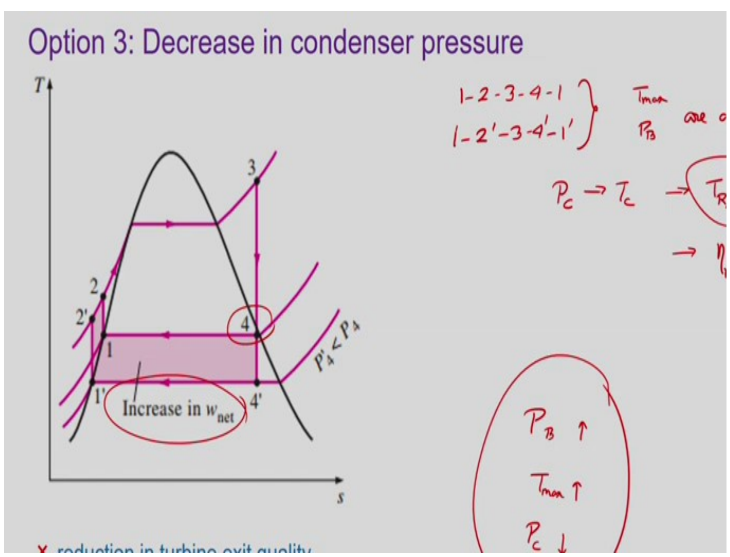

Now option number 2: increase in the maximum temperature. Of course, if we increase the maximum temperature our direct gain will be in the average temperature of heat addition like shown in the diagram. Here again we have two cycles 1-2-3-4-1 and 1-2-3'-4'-1, and between these two cycles boiler pressure and condenser pressure are constant. And now as the pressure remains constant the effect of increase in the maximum temperature is a direct increase in the area enclosed by the *Ts* curve and therefore an increase in the work output.

So, the average temperature of heat addition corresponding to the cycle 2 is definitely greater than the average temperature corresponding to the average temperature of heat addition corresponding to cycle 1. Because in both cases we are starting in addition at point 2 but we are finishing at different points the second cycle finishing at a higher temperature. And accordingly, the thermal efficiency for the second cycle will definitely be greater than the first cycle.

So, increase in the maximum temperature definitely is a potent option. Also look what is happening on the turbine exit. The exit point is getting shifted towards right, that is the exit quality is also increasing it is approaching the superheated vapour. Therefore, here we are gaining from both point of view, we are getting higher pressure and also, we are getting higher turbine exit quality both of which are highly desirable.

So, compared to the increase in the boiler pressure level probably this can be a better option because we are gaining in two fronts. But definite problem is that the material has to deal with higher temperature levels and therefore there will be certain kind of metallurgical limits. The third option is on the condenser side. Here we are maintaining, again we are going for two cycles 1-2-3-4-1 and the second cycle is 1-2'-3-4'-1'.

Between these two cycles the thing that is changing is the condenser pressure and accordingly the temperature corresponding to condensation. But the maximum temperature and the boiler pressure, they are constant. Now once we are expanding in the turbine to a lower pressure level then definitely the exit state is getting shifted to a lower point and there is a significant increase in the area that has been enclosed by the curve.

Also, as a condensation temperature is decreasing so though there is no change in the average temperature corresponding to heat addition are hardly any change. We have the average temperature corresponding to heat rejection that is decreasing. As for this diagram you can see that the heat addition process is also starting from a lower point $T_{2'}$ instead of T_2. So, the average temperature corresponding to heat addition is also decreasing, but this decrease may not be significant compared to this particular decrease leading to an overall increase in the thermal efficiency.

So, there are three ways of increasing the thermal efficiency of a Rankine cycle. First is the boiler pressure, where we have the disadvantage of a loss or reduction in the turbine exit quality. Then an increase in the maximum temperature where the exit quality is also increasing and the third is a deduction in the condenser pressure. But what is happening to the exit quality? The exit point is getting shifted towards left or towards lower quality side.

So, again here we are having a low-quality mixer of the turbine exit so erosion problem will be there. So, this particular option of decreasing the condenser pressure comes with a problem of reduction in turbine exit quality and also a practical difficulty that is we are generally the pressure in maintained inside the condenser is well below the atmospheric pressure. Because we select the condensation pressure based upon the atmospheric temperature.

Now you know that, for water the corresponding atmospheric pressure the saturation temperature is 100 ^{0}C. Then if we want to attain condensation say at a temperature of 20 ^{0}C we have to reduce the pressure significantly below that atmospheric pressure. So, the condenser is already operating under vacuum condition and there is every possibility of air leaking from surrounding into the condenser and thereby hampering the heat transfer performance.

And now if we reduce the condenser pressure even more so higher condensation vacuum means new required which will put additional burden from maintenance and when the fabrication point of view. But still all three options can be exercised. And generally we like to summarize then that we always like to have a higher boiler pressure as much as your material can withstand and also as long as the turbine exit quality is manageable.

We want a higher maximum temperature as long as your metallurgical limits are adhered with and we want a lower condensation pressure as long as you can maintain the vacuum inside the condenser and again the turbine exit quality is manageable. So, all three or the combination of all three or any two of them or quite often used in industries.

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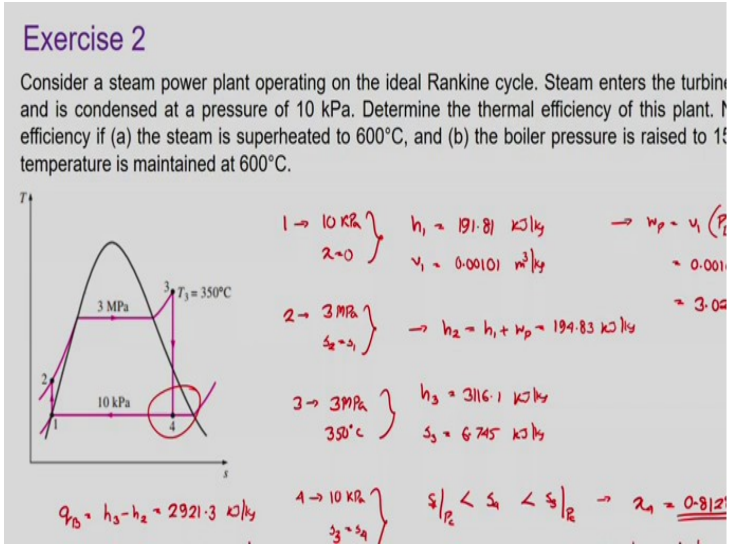

We should would like to show the effect of each of them through this numerical example. Here we are talking about a steam power plant operating on an ideal Rankine cycle. Steam enters the turbine at 3 MPa and 350 ^{0}C and is condensed at a pressure of 10 kPa. So, this is a cycle very standard cycle, shown here. Let us try to calculate the thermal efficiency for this plant.

Super heating is there but none of the options that we discussed about has been explored here. So, let us see, point number 1 corresponds to a pressure of 10 kPa and a quality of 0 because it is saturated liquid. So, corresponding to this if we check from the table your h_1 is 191.81 kJ/kg and specific volume at this particular point is 0.00101 m^3/kg. Look at this magnitude, this is I am taking directly from the table which I have which is kept beside me and the value is that 0.001 that we used earlier also.

Now point number 2 corresponds to a pressure of 3 MPa and no other information are given but assuming the compression process to be isentropic then we know that:

$$s_1 = s_2$$

But this information we generally do not need to make use of because you can directly calculate the pump work as:

$$W_P = v_1(P_B - P_C) = 0.00101(3 \times 10^3 - 10)$$

So, the turbine work is coming a meagre value of 3.02 kJ/kg.

And from there we can calculate h_2 to be equal to:

$$h_2 = h_1 + W_P = 194.83\ kJ/kg$$

So, points 1 and 2 have been identified now move to point 3. Point 3 clearly given as superheated and it is 3 MPa and 350 ^{0}C.

So, from the superheated table we have:

$$h_3 = 3116.1\ kJ/kg$$

and entropy we need here, we need entropy which is:

$$s_3 = 6.745\ kJ/kgK$$

And point number four we know it to be at a pressure of 10 kPa and s_3 to be equal to s_4. And now if we compare the values for this, s_4, is coming to be greater than the s_f corresponding the condenser pressure but less than s_g corresponding to the condenser pressure, which indicates this s_4 is in the mixture dome. Though the diagram is indicative, but you always need to check this way as this s_4 is coming between a s_f for s_g or not. If it is greater than s_g then what will happen? It is on the superheated side. If it is less than s_f, then it is on the sub-cooled side, which hardly happens. But if it is in between the two it is in the mixture side. So, if you put the numbers, then you will be getting:

$$x_4 = 0.8128$$

which is a significant number we need to make a note of this number.

Turbine exit quality for this configuration is 0.81. And from there we can compute h_4 to be:

$$h_4 = h_f|_{P_C} + x_4 h_{fg}|_{P_C} = 2136.1\ kJ/kg$$

So, we can easily calculate the remaining numbers, the q_B and we have to also calculate thermal efficiency. So, your q_B will be equal to:

$$q_B = h_3 - h_2 = 2921.3\ kJ/kg$$

And condenser heat rejection or heat rejection is coming to be equal to:

$$q_C = h_4 - h_1 = 1944.3\ kJ/kg$$

which gives you a thermal efficiency of :

$$\eta_{th} = 1 - \frac{q_C}{q_B} = 33.4\ \%$$

So, this is about the first case where we are talking about an ideal cycle. Now look at this, here the steam is superheated to 600 ^{0}C rest remains same.

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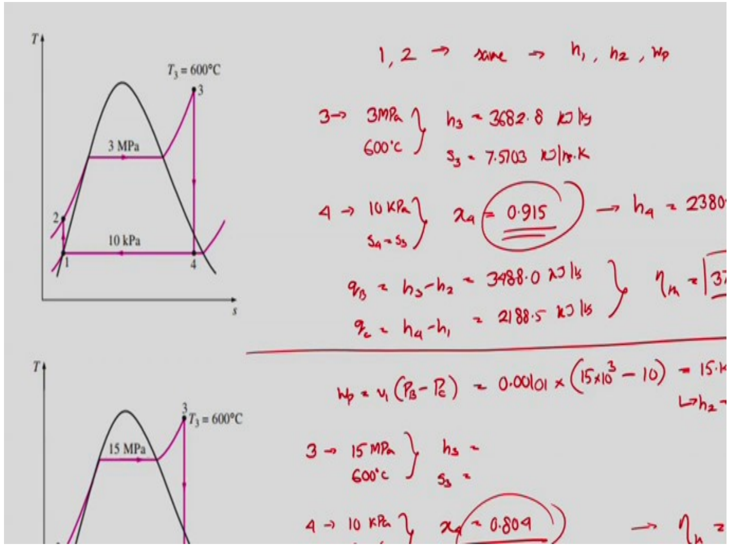

So, this is the situation here point 1 and 2 are same. Accordingly, there is no change in the calculations of h_1, h_2 and the pump work. Of course, we have not calculated the turbine work in this problem if you want the turbine work you can also calculate to be as $h_3 - h_4$. That is not required in our case for our demonstration so I have not shown here but you can also calculate this number. So, 1 and 2 remains same but 3 there is a change.

So, point number 3 what is happening? Here the condition is that of 3 MPa and 600 ^{0}C. So, correspondingly,

$$h_3 = 3682.8\ kJ/kg$$

and

$$s_3 = 7.5103\ kJ/kgK$$

For point number four the pressure is same which is 10 kPa and:

$$s_4 = s_3$$

putting this information, you will be getting here,

$$x_4 = 0.915$$

using which we shall be getting,

$$h_4 = 2380.3\ kJ/kg$$

So correspondingly we have; I am just directly noting the numbers to save some time.

$$q_B = h_3 - h_2 = 3488.0\ kJ/kg$$

$$q_C = h_4 - h_1 = 2188.5\ kJ/kg$$

combining this the thermal efficiency will be coming as:

$$\eta_{th} = 37.3\ \%$$

A few things to compare, here we have just increased the maximum temperature without affecting the boiler and condenser pressure. Then first note this, in the previous case our turbine exit quality was 81% here it has gone to 91%. So, there is a significant increase in the turbine exit quality. And what about thermal efficiency? It was 33% now it has gone to 37%. If you compare the boiler heat addition and condenser heat rejection, boiler heat addition has increased quite a bit, condenser heat rejection is also increased but to a lesser amount degree giving you a higher increasing the thermal efficiency. So, we are gaining in both fronts, thermal efficiency has increased and the turbine exit quality also has increased.

Now the second part, the boiler pressure is raised to 15 MPa and turbine inlet temperature is maintained at 600 ^{0}C. So, boiler pressure now goes from 3 MPa to 15 MPa. So, point number 1 again remains the same, but others have changed. Pump work in this case will be equal to:

$$W_P = v_1(P_B - P_C) = 0.00101(15 \times 10^3 - 10) = 15.14\ kJ/kg$$

So, there is a small increase in the pump work accordingly we can calculate h_2 to be equal to:

$$h_2 = h_1 + W_P$$

I am not calculating the numbers and just showing you the way because by now you know how to get this. Then, point 3 now corresponds to 15 MPa and 600 ^{0}C, using this you can get the value of h_3 and s_3 from the table.

Point number 4 corresponds the same pressure of 10 kPa and

$$s_4 = s_3$$

if you combine that you are going to get in this case to be:

$$x_4 = 0.809$$

So, boiler pressure has increased maximum temperature remains same and just come down from this 0.915 to 0.804 because of the increase in the boiler pressure. Maximum temperature and condensation pressure remain the same.

And if you calculate q_B and q_C, your thermal efficiency will be coming to be equal to 43% approximately. So, though there is a reduction in the turbine exit quality there is a significant increase in the thermal efficiency. So, you may have to go for some kind of trade-off. I would request you to solve the same problem by maintaining the maximum temperature as 600 ^{0}C like in this problem, boiler pressure to be 15 MPa and condenser pressure of 1 kPa and calculate the thermal efficiency from this to see the effect of the condensation pressure. So this is the way we can increase the performance or efficiency for a ideal Rankine cycle.

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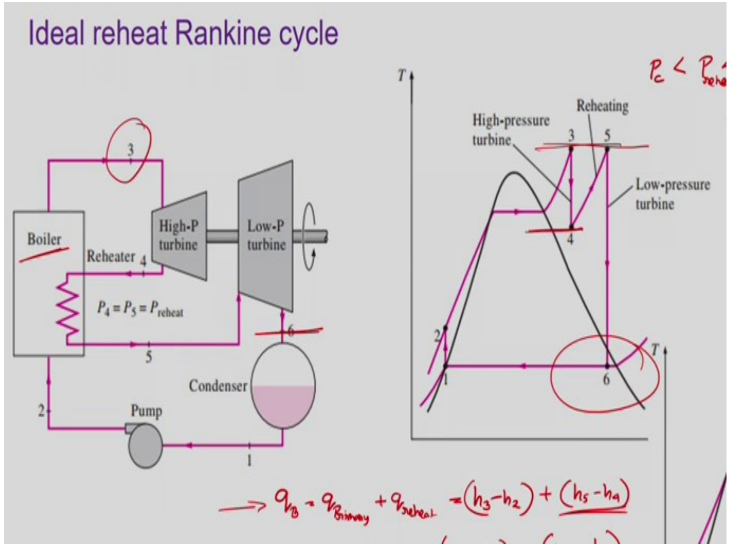

But as we have seen we can get higher efficiency by changing all these three options: boiler pressure, condenser pressure and maximum temperature. But if we want to increase both the efficiency and the turbine exit quality our option is only increase in maximum temperature. Now maximum temperature, theoretically we can keep on increasing but as I have mentioned there are metallurgical limitations and so we may not be able to go to very high temperature.

This 600 ^{0}C which we have used here is a quite decent practical limit. In fact, in India most of the power plant uses a maximum limit of 540 ^{0}C for metallurgical considerations. So, there is a second option, something that we have used in case of Brayton cycle also where we go for multistage expansion and inclusion of reheating in between the multistage expansion.

Just like what we have done in the Brayton cycle, here the steam that is coming from the boiler is initially expanded in a high-pressure turbine up to some intermediate pressure level as shown by this pressure here, the pressure P_4. So, here this pressure P_4 is less than the boiler pressure but it is greater than the condenser pressure somewhere intermediate which is often called the reheat pressure also.

And then at this point 4, at this intermediate pressure the steam is taken back to the boiler to reheat back to certain temperature. Generally, it is heated up to the same temperature but the temperature at point number 5 that is T_5 should be equal to T_3 or less than T_3 in no situation generally goes beyond T_3.

And then it is taken to the low-pressure turbine where it is expanded to the condensation pressure up to this particular point. So, here we are expanding the steam in two stages of turbine: one is a high-pressure turbine and other is a low-pressure turbine. The concept of reheating with multi-stage expansion was introduced in early 1920's but because of practical issues it was abandoned for about 15-20 years.

Then in early forties it started to operate again, where have people started to use one stage of reheating. Later after 1950's people started using two stages of reheating that is instead of having two we can have three turbines also. The first one is the high-pressure turbine ,where the expansion takes place from boiler pressure to first level of intermediate pressure. Then we have an intermediate pressure turbine where the expansion goes from intermediate pressure 1 to intermediate pressure 2.

Then we have the LP turbine where expansion goes from the second intermediate pressure to the condenser pressure. If we keep on increasing the number of reheating, of course we can almost maintain a near constant average reheat temperature or rather I should say the reheat temperature remains almost constant to the maximum cycle temperature, thereby I should say as the number of reheating increases it tries to approach the maximum possible efficiency.

But it is practically not at all feasible because increasing the amount of gain in efficiency that we get by adding more than two reheat stages that cannot be justified corresponding to the cost that is incurred in this. So, hardly any cycle you will see which uses more than two reheat stage. Now when you are using a cycle with a single stage reheat like this then total amount of heat that has been added, the q_B that we are using there that has now two components. One is the primary one that we are using always plus the reheat part now we have, i.e.,

$$q_B = q_{primary} + q_{reheat}$$

$$= (h_3 - h_2) + (h_5 - h_4)$$

so, this is the reheating part of power where the additional heat input will be required. And how much is your turbine work? Your turbine work now corresponds to the work produced by the HP turbine plus the work produced by the LP turbine. So,

$$W_T = W_{HPT} + W_{LPT} = (h_3 - h_4) + (h_5 - h_6)$$

So, there will be an increase in the total work output but also there is increase in the total heat input requirement. So, it is difficult to say whether that or there will be any gain at all in the efficiency. In fact, because of reheating we may get some increase in the efficiency, we may get some reduction in the efficiency as well. And the change in efficiency with the adoption of reheating generally is not more than 1 or 2%.

But the biggest gain is here, the turbine exit quality will be significantly higher compared to a single reheat case or I should say a single stage turbine expansion. Also, generally the heat that is produced the boiler if we are not using reheating a significant part of that heat produced because of the fuel combustion may get wasted so we are also able to make better utilization of our resources by the reheating.

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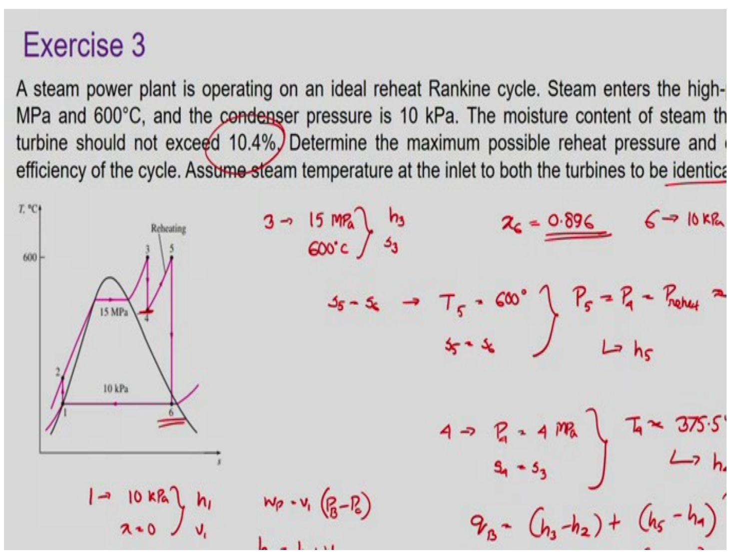

One numerical example again can be considered just to demonstrate the effect of reheating. Here we are talking about a steam power plant operating an ideal reheat Rankine cycle. Steam enters the high-pressure turbine at 15 MPa and 600 ^{0}C. So, here the point stage number 3 corresponds to 15 MPa which is the boiler pressure and 600 ^{0}C. So, from there we can easily calculate h_3 and S_3, I shall not be putting the numbers I am leaving it up to you.

And the condenser pressure is given as 10 kPa and the moisture content of steam at the exit of the low-pressure turbine should not a see 10.4%. That is your point number 6 where:

$$x_6 \geq 0.896$$

and in no case it should fall below this. So, what can be your maximum possible reheat pressure? That is you have to calculate the pressure at this particular point 4 or you have to calculate pressure P_4 or P_5 which are same. Let us put,

$$x_6 = 0.896$$

then for point number 6 what we can say? We know that the pressure is 10 kPa and we also know the value of $x6$, so using this you can calculate the value of h_6 and s_6. Now corresponding to the LP turbine expansion s_5 should be equal to s_6 and 1 for point number 5 you also know that the temperature at the inlet to both the turbines to be identical. So, we also know that

$$T_5 = 600^0C$$

and

$$s_5 = s_6$$

if you combine these two we can calculate P at point number 5 which is also P equal to point number 4 that is your reheat pressure by using suitable use of the tables we can identify this to be:

$$P_5 = P_4 = P_{reheat} \approx 4\ MPa$$

So, once we have this corresponding to this, we can calculate your $h5$.

And now what to do that with the rest part? Point number 3 you have already identified. So, let us come to the pump side for point number 1. We know it is 10 kPa and quality equal to 0, so from there you can get h_1. Pump work will be equal to:

$$W_P = v_1(P_B - P_C)$$

and

$$h_2 = h_1 + W_P$$

$$q_{primary} = (h_3 - h_2)$$

So, you can get the $q_{primary}$ because h_3 you have identified. And for point number 4, now we know that

$$P_4 = 4\ MPa$$

and

$$s_4 = s_3$$

from there we have to identify you can get:

$$T_4 \approx 375.5^0C$$

from there you can calculate h_4. So,

$$q_B = (h_3 - h_2) + (h_5 - h_4)$$

and

$$W_{net} = (h_3 - h_4) + (h_5 - h_6)$$

combining this your thermal efficiency will be coming in the range of 45%, i.e.,

$$\eta_{th} = 45\ \%$$

But more importantly we are able to maintain a significantly high turbine exit quality. So, that is a demonstration of the reheat cycle where with a single reheat we are able to get a decent thermal efficiency and also able to maintain a significantly high turbine exit quality. But as I mentioned we may not get any gain in the efficiency by adoption of the reheating. So, for that purpose in order to ensure the gain in the thermal efficiency we have to use something that again we have used in a Brayton cycle, which is the regeneration.

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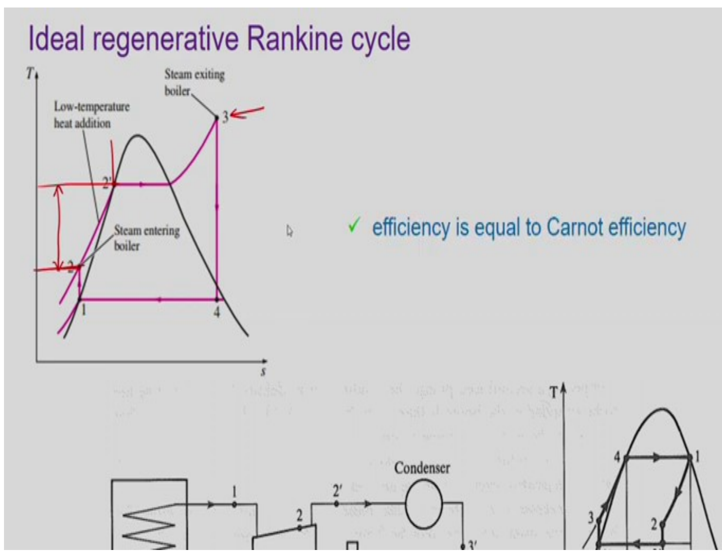

Details considering the regeneration I shall be explaining in the next lecture. But just to show you something, the point 1, the heat addition is starting this is cycle shown without reheating, a standard Rankine cycle with superheat, we are starting heat addition at point 2 finishing heat addition point 3. So, if we are supposed to heat this one, we are using a gas which is maintaining a constant temperature somewhat like this.

Then at this point the temperature difference is extremely high whereas at this point the temperature difference is quite low. And heat addition with high temperature difference means lots of irreversibilites and lots of losses. Instead of using something like this, there are two ways we can do. If we can use a gas temperature profile something like this, that is in all cases the temperature difference between the combustion gas and the steam is not very large then probably we can reduce the thermal efficiency or I should say reduce the irreversibilities. And secondly, the heat that you are adding to the saturated liquid that is between point 2 to 2', if somehow we can manage this such that the steam or liquid water enters the boiler only at point 2' then also the net temperature or average temperature corresponding heat addition will correspond only to find point 2' to point 3.

So, there will be a significant increase in the average temperature of heat addition and currency consequently we can expect gaining the thermal efficiency as well. So, the idea of regeneration is to add up some means where we can raise the temperature of water from 2 to 2' before it enters the boiler. So, we would like to add up some means where we can gain this much of change in the temperature of water before the steam enters the boiler and that is what we refer as the regeneration, because there we shall be using certain means that I shall be explaining in the next class. This is one means that is the first one but I would like to go into the detail in the next class only where I shall be explaining in detail about the process of regeneration, different kinds of regenerative heaters that we use and consequently how much gain we can gain your thermal efficiency.

So, the advantage of having regeneration is efficiency can theoretically be equal to the Carnot efficiency but other two problems I will come back to this again in the next class.

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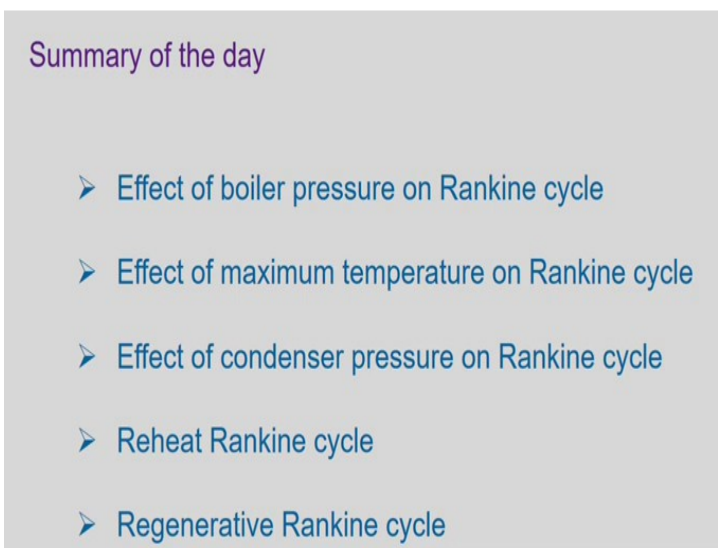

So, today we have discussed about the effect of boiler pressure, maximum temperature and condenser pressure on the performance of the Rankine cycle. The concept of reheat Rankine cycle was introduced and discussed in detail and we have very briefly touched upon the concept of regeneration in a Rankine cycle. This is the point from where I would like to start my next lecture.

So, please wait for that and try to solve a few numerical problems so that you gain more exposure on the use of the reheat cycle. So, thanks for your attention we shall be meeting again very soon. Thank you.