Applied Thermodynamic for Engineers
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Lecture – 23
Binary Vapour Power Cycle

Hello friends. We meet for the fourth time this week for you looking to wrap this particular chapter on vapor power cycle. We have already discussed a lot about the vapour power cycle which we keep on using in industries for commercial power production, which is the Rankine cycle. And also, we have discussed about different ways of improving the performance in terms of both efficiency and work output from the Rankine cycle.

Like the options of modulating the maximum pressure and maximum temperature or maybe the condenser pressure. We have also discussed about the option of adopting reheat: single stage reheat or two stage reheat to the Rankine cycle. And in the previous lecture we have talked about the option of adding regeneration through both are open or closed feedwater heaters.

So now we know that a practical power station will always have a superheater so that the temperature of the steam can be increased even beyond its critical temperature. Then it may have one or two reheaters, so that we can keep on adding heat after completing the expansion in some intermediate temperature, so that to entire expansion in the turbine can be broken into multiple stages. And then we shall be having several feedwater heaters to have the operation of regeneration.

Generally, stations have five to seven number of closed feedwater heaters and one open feedwater heater, which is generally placed in between, just at the central of all these feedwater heaters. The closed feedwater heaters provide interior to heat transfer but they are much more compact. Whereas open feedwater heaters are large and bulky but they provide the great heat transfer because that facilitates a direct mixing of the condensate and the bled steam. And also, the open feedwater heater, often called the deaerator also allows us to remove the non-condensable gases, thereby deaerating the condenser and so helping to reduce the corrosion of the tubes.

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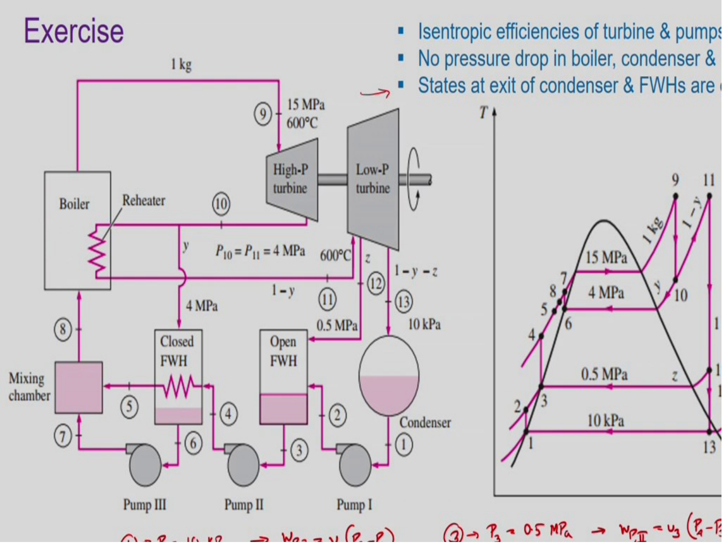

Exercise

- Isentropic efficiencies of turbine & pumps
- No pressure drop in boiler, condenser &
- States at exit of condenser & FWHs are

Now in the previous lecture, I finished by giving you one exercise where we had this particular diagram to consider with. I hope you tried to solve this one. If you have not, then please hold this lecture here and try to solve this on your own, and they only come back to see the way we can solve this. I am actually going to solve this one only partially without giving all the data. You have to solve this on your own. So that you can have full control about which process to model and how.

So, in this station or in this plant we have two turbines: a high-pressure turbine where we are having steam entering at a pressure of 15 MPa and 600 ^{0}C and it is expanded to an intermediate pressure of 4 MPa. Now when this steam coming out of the high-pressure turbine is going to the reheater, in between we are putting a tapping to extract y quantity of bled steam, which is applied to the closed feedwater heater. Then remaining $(1 - y)$ fraction is going back to the low-pressure turbine after being reheated back to the 600 ^{0}C the same temperature received by the high-pressure turbine as well. In the low-pressure turbine it is expanded to the condenser pressure of 10 kPa, but not the entire quantity, rather at some intermediate pressure of 0.5 MPa z fraction of steam has been extracted and supplied to the open feedwater heater, and the remaining $(1 - y) - z$ fraction reaches the final condenser level pressure. The condensate coming out of the condenser pressure is the pressurised to the pressure of 0.5 MPa by this condenser extraction pump. And there it mixes with the bled steam coming from the low-pressure turbine to give this state 3.

Now this condensate is pressurized to the boiler pressure by the boiler feed pump, which is the pump 2 here, and passes through the closed feedwater heater where it mixes with the bled steam coming from the reheater line. And the condensate that is collected the bottom of the closed feedwater heater, it is not left back to the lower side rather it is pumped up back to the boiler pressure level and they mix at this mixing chamber to give the final state of 8. If you plot the entire thing in *Ts* diagram it will be looking something like this. Check the diagram carefully as I remove all these lines from this. So here the boiler operation starts from this particular point 8 and continues up to point 9. 9 to 10 is the expansion in the high-pressure turbine then it is reheated back to the same temperature to point 11.

However, it is only $(1 - y)$ fraction remaining y fraction has been kept back for the closed feedwater heater. From 11 to 13 we have expansion in the turbine and in between at point number 2, z fraction has been taken out to be used in the open feedwater heater. Now 1 to 2 is the operation done in where? Pump 1, the condenser extraction pump.

Then the condenser coming with state number 2 and the bled steam coming with state number 2, they mix to give you the state 3. These are ideal cycles, so, we have the assumption that the exit state from the condenser from all feedwater heaters are that of saturation corresponding to the temperature. 3 to 4 is the process that is done in pump number 2, then we have the closed feedwater heater. And in the closed feedwater heater, the condensate is heated up to point number 5. Whereas the bled steam coming from the reheater line is condensed to point number 6, so this is found back to point number 7 and there we have mixing chamber where we have the mixing of 7 and 5 to give the final point 8. Now like the previous problem that you have solved, if your conditions are to be considered which corresponds to that ideal cycle: here we are considering the isentropic efficiency of turbine and pumps to be 1, no pressure drops in the boiler turbine and feedwater heaters and also, the states at the exit of the condenser and feedwater heaters are of saturated liquid. Now one important consideration that we should consider here is exit state from the closed feedwater heater.

From the condenser the exit state is that of saturated liquid, it is point 1. Similarly, from the open feedwater heater there is only single exit state which corresponds to this point number 3. Both are saturated liquids at their corresponding temperatures. But from the closed feedwater heater there two exit states. One is 5 for the condensate and the other is 6 corresponding the bled steam. Now that 6 correspond to the saturation condition at the pressure at which it has been extracted.

What about 5, that is something where we have to go for some kind of approximation and we shall we sticking to this particular approximation only by assuming that the temperature of 5 and 6 equal to each other. So, though point number 5 correspond to a saturated liquid but here assuming,

$$T_5 = T_6$$

so that the state of 0.5 is that of the boiler pressure which is 15 MPa, i.e.,

$$P_5 = 15\ MPa$$

and

$$T_5 = T_6$$

which is the saturation temperature corresponding to that 4 MPa. This is one assumption that we have to consider, you have to incorporate while solving this. Otherwise we need to complete information about the state 5, at least about is proper temperature. So now, with this consideration what are the points that known to us? Point number 1 is definitely known.

Point 1:

$$P_1 = 10\ kPa$$

and

$$x_1 = 0$$

because it is a saturated liquid. This state is completely defined. Using this you can calculate the pump up corresponding to the pump number 1, which of course will be:

$$W_{P,I} = v_1(P_2 - P_1)$$

and once we have this we can get:

$$h_2 = h_1 + W_{P,I}$$

where from this we can calculate h_1 and v_1 and that we subsequently used to calculate the pump work and the value of h_2.

Then state number 3, what are the things known we know that,

Point 3:

$$P_3 = 0.5\ MPa$$

and

$$x_3 = 0$$

because it is also saturated liquid. So, if we consider this, from there we can calculate or we can directly from the table the value of h_3 and v_3 because v_3 that of saturated liquid at pressure of 0.5 MPa, same for h_3. So, using this we can calculate the pump work for the second pump which will be equal to:

$$W_{P,II} = v_3(P_4 - P_3)$$

and once we add this, we have:

$$h_4 = h_3 + W_{P,II}$$

So, point 1, 2, 3 and 4 are known. What about point 6? For point number 6 again, this is of saturated liquid. So,

Point 6:

$$P_6 = 4\ MPa$$

and

$$x_6 = 0$$

Combining this we can get the value of h_6 and v_6, the output from the third pump will be:

$$W_{P,III} = v_6(P_7 - P_6)$$

$$P_7 = boiler\ pressure = 15\ MPa$$

and using this we can get:

$$h_7 = h_6 + W_{P,III}$$

third for the third pump. So you already know about points 1, 2, 3, 4, 6 and 7. Identify the other points from this.

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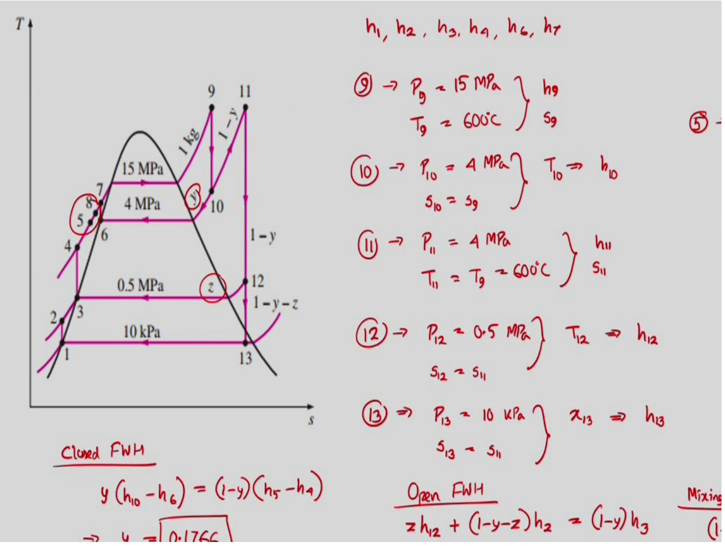

$h_1, h_2, h_3, h_4, h_6, h_7$

⑨ → $\left.\begin{array}{l} P_9 = 15\ MPa \\ T_9 = 600^\circ C \end{array}\right\}$ h_9, s_9

⑩ → $\left.\begin{array}{l} P_{10} = 4\ MPa \\ s_{10} = s_9 \end{array}\right\}$ $T_{10} \Rightarrow h_{10}$

⑪ → $\left.\begin{array}{l} P_{11} = 4\ MPa \\ T_{11} = T_9 = 600^\circ C \end{array}\right\}$ h_{11}, s_{11}

⑫ → $\left.\begin{array}{l} P_{12} = 0.5\ MPa \\ s_{12} = s_{11} \end{array}\right\}$ $T_{12} \Rightarrow h_{12}$

⑬ ⇒ $\left.\begin{array}{l} P_{13} = 10\ kPa \\ s_{13} = s_{11} \end{array}\right\}$ $x_{13} \Rightarrow h_{13}$

⑤ -

Closed FWH

$$y(h_{10} - h_6) = (1-y)(h_5 - h_4)$$

$$\Rightarrow y = \boxed{0.1766}$$

Open FWH

$$z h_{12} + (1-y-z) h_2 = (1-y) h_3$$

Mixing

(1

So, let us go to the other side, so, we are already aware about h_1, h_2 and then h_3, h_4, h_6 and h_7, but not 5 and 8. Now what about point number 9? For point number 9, we know that P_9 is equal to given the previous diagram 15 MPa and 600 ^{0}C. So

$$P_9 = 15\ MPa$$

$$T_9 = 600℃$$

Combining you can get the value of $h9$ and $S9$. Point number 10 it is expanded in the high-pressure turbine up to the intermediate pressure of 4 MPa. So,

$$P_{10} = 4\ MPa$$

you know that isentropic efficiency equal to 1 and

$$s_{10} = s_9$$

So, combining this we can identify the location of T_{10} by using the superheated vapour table because here it is in the superheated side and from the T_{10} you are getting h_{10}.

So, point number 10 is identified. Point number 11, this is the entry point to the low-pressure turbine, here the pressure is:

$$P_{11} = 4\ MPa$$

$$T_{11} = T_9 = 600℃$$

Combining we can easily get h_{11} and s_{11}. So, what should be the values for point 12 and 13? 12 state is known that is:

$$P_{12} = 0.5\ MPa$$

at which the stream has been extracted to be used in the open feedwater heater. And you also know that:

$$s_{12} = s_{11}$$

because it is isentropic expansion seeing this and again from the table you can identify T_{12}. And using T_{12} we can get h_{12}. Finally state number 13, it is the entry to the condenser. So here the pressure is that of condenser pressure that is:

$$P_3 = 10\ kPa$$

and again, as it is isentropic expansion so we can calculate:

$$s_{13} = s_{11}$$

Combining in this s_{13} will be coming in the vapor done because the value of s_{13} will be less than s_G, corresponding the condenser pressure. So, from there we can calculate the value of x_{13} and using s_{13}, we can calculate h_{13}. So, you know the enthalpy at this particular point also. Now point number 5 and 8 are the two that are remaining. Now 8, I shall be coming later on, but for 5 we have an assumption to be considered.

As I have mentioned that for point number 5,

$$P_5 = boiler\ pressure = 15\ MPa$$

So that is 15 MPa and we have assumed that:

$$T_5 = T_6 = T_{sat}\ (corresponding\ to\ 4\ MPa)$$

So, using this we can calculate h_5 either using the compressed liquid table or quite often,

$$h_5 \approx h_f|_{T_6} = h_6$$

So now, you know the values all the points apart from 8. So how much is the energy required for the boiler? Can you calculate now? Before that let us try to calculate this fraction y and z. To calculate the value of y we have to perform the energy balance of across the closed feedwater heater. For the closed feedwater heater, just going back to the previous diagram. Look at this. We have y fraction of bled steam is coming with state 10 and getting converted to state 6 where is condensate is coming with $(1 - y)$ fraction and that is entering at 4 and leaving at 5. So, we can write:

Closed FWH

$$y\ (h_{10} - h_6) = (1 - y)(h_5 - h_4)$$

As you know all the enthalpies here particularly after assuming the value of h_5, then we can easily calculate the value of y and if you calculate properly,

$$y = 0.1766$$

that is the first result that you are looking for.

Now go to the open feedwater heater. For the open feedwater heater what we have? Here the bled steam is entering which state 12 and condensate is entering state 2 and then that is they are combining to form the mixture state 3. So,

Open FWH

$$zh_{12} + (1 - y - z)h_2 = (1 - y)\ h_3$$

So, we have already calculated the value of y all the enthalpies also known putting this,

$$z = 0.1306$$

So, we know both the extracted fractions. Then finally I have to calculate the efficiency, but before that you have to identify the location of point 8. How can you identify h_8? we have to look at the mixture. Look at the mixture, here the condensate is entering state 5 and the condensate stored accumulates at the bottom of the closed feedwater heater, it is entering is 7 and they are forming a mixture of 8. So, combining this in the mixing chamber now, then we have:

Mixing chamber

$$(1-y)h_5 + yh_7 = h_8$$

We have again got the value of y, h_5 has been assumed and h_7 we know, putting this you will get,

$$h_8 = 1089.8\ kJ/kg$$

So, we can now calculate the heat load for the boiler and condenser. Your q_B will be equal to then:

$$q_B = (h_9 - h_8) + (1-y)(h_{11} - h_{10})$$

Where the first term of the above equation is the primary heat supplied in the boiler. And the second term is adding in the reheater we $(1 - y)$ fraction mass fraction multiplied by $(h_{11} - h_{10})$ which is the total boiler heat load.

And heat load for the condenser, q_C will be:

$$q_C = (1-y-z)(h_{13} - h_1)$$

So, from there you can calculate the value of this combining is the thermal efficiency will be coming as:

$$\eta_{th} = 1 - \frac{q_C}{q_B} = 49.2\ \%$$

So, with this similar data one problem was solved earlier, where there was no regeneration. I guess the efficiency that we calculate there was something on 45 to 46%, so the effect of regeneration is to cause an increase in efficiency by over 3 to 4%. And if your interest is to calculate the total turbine work then how can I do this?

Then total turbine work should be equal to:

$$W_T = W_{HPT} + W_{LPT}$$

where

W_{HPT} is the work produced by the high-pressure turbine

W_{LPT} is the work produced by the low-pressure turbine.

For the high-pressure turbine how much you are getting? The work output from high-pressure turbine will be:

$$= (h_9 - h_{10})$$

that is from the high-pressure turbine and from the low-pressure turbine there are two different mass fractions. So, $(1 - y)$ fraction has been extended from 11 to 12 and after the second extraction $(1 - y) - z$ fraction extended from 12 to 13. That is:

$$= (h_9 - h_{10}) + (1 - y)(h_{11} - h_{12}) + (1 - y - z)(h_{12} - h_{13})$$

So, this will be your condenser work output or I should say that the turbine work output. Pump work has already been calculated and putting these values again you can confirm the efficiency from the same as well. So, this way we can easily keep track of the reheat regeneration process. And this is generally the practical cycle that you will encounter in any power stations.

But of course, not with just two feed water heaters, there may be several feedwater heaters commonly 7 to 8, with one of them at least one of them being open feedwater heater. So that is the basic discussion on Rankine cycle that we should always talk about. There are some additional factor that I would like to quickly go through today.

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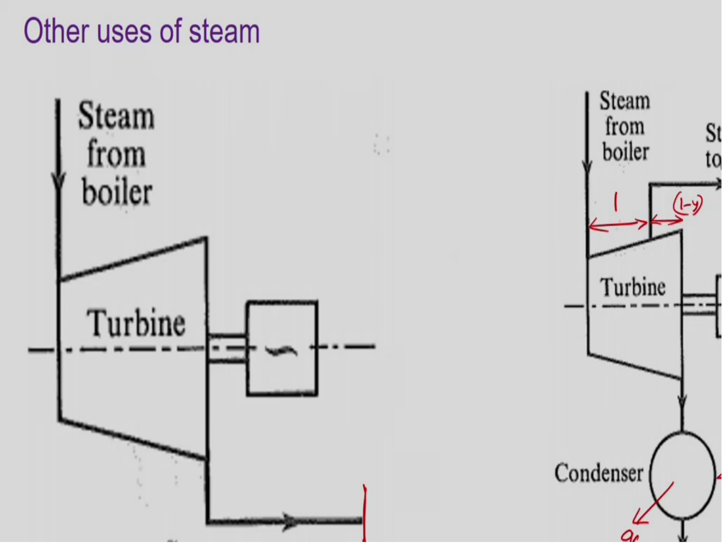

Other uses of steam

There are sometimes the Rankine kind of cycles are also used for other kind of application of steam, apart from power generation. Like couple of examples are showing here one is this called the back-pressure turbine, where the output from the turbine does not go to the condenser whereas the steam is supplied to certain process. And it may come back the circuit following the process or may not but there is no condenser. And then the exit state of the turbine is governed by the pressure required for this particular process. That is why it is called back pressure turbine because the rating of the turbine or the exit pressure of the turbine will be governed not by the condenser pressure rather by the process pressure.

This is another example which is called a pass out turbine where we have a condenser, we have the normal turbine condenser combination. And here some extraction has also been taken, but that extracted stream not going to any feedwater heater rather that is going to certain kind of process. So, we call the pass out turbine because some portion of steam has been passed out. Now at which pressure we should take this extraction? That again will depend upon the process for which you need this steam. This pass out turbine and back pressure turbine both of them have their work output governed by the process requirement. Like suppose for this process, we need y fraction of steam, then again in a turbine, we know that from this initial pressure to this intermediate pressure total expansion is taking place over the entire mass. But from here after up to the final pressure expansion is taking place only for $(1 - y)$ fraction. As a condenser pressure normal is fixed and also, the pressure steam coming from the boiler is having a known pressure. Then the work output will definitely we depend on this fraction y.

If y increases, total output will be lower, if y decreases, total output will be higher. Similarly, for the back-pressure turbine, as the pressure required for the process increases total expansion in the turbine will be lesser so we shall be getting less work output. Similarly, in the process pressure requirement decreases we can get more power output from the turbine. So, this way we can have quite of few other kinds of configurations also in the Rankine cycle mode.

Now here I would like to talk about another term that in back pressure turbine you are saying that the outlet pressure of the turbine depends on the process pressure. Now when we have the condenser like in case of the pass out turbine or normal Rankine cycle. On what you exit pressure should depend on? Of course, the condenser pressure. But on what does the condenser pressure depend on? See in the condenser what is objective? Our objective is to reject this q_C amount of heat which we must reject to adhere to the second law of thermodynamics.

Now, this heat is been rejected to what? This heat is been rejected to some kind of cooling media and that cooling medium normally can be either the liquid water steam coming from certain cooling tower or more commonly coming from some local source of water, large source something like a lake or river. And the temperature of that source of water, that coolant normally is at atmospheric temperature.

Then what should be a condenser pressure? Then your condenser pressure should be equal to the saturation temperature corresponding to that coolant or I should express this in other way possible. Here the coolant temperature is equal to the atmospheric temperature. Then the condenser pressure that we can have that at most can be the saturation pressure corresponding to the atmospheric temperature or slight variation of that.

So that is atmospheric temperature that determines what can be your condenser temperature or I should say the condenser pressure. Of course, I will always prefer to have lower condenser pressure. But atmospheric condition is something that we cannot control in what country something like in India we generally have much higher atmospheric pressure compare to countries which are in same in Canada or in northern part of Europe.

There can be about 10 to 15 ^{0}C difference between the average atmospheric temperatures. So, the condenser pressure also will be lower in colder countries and will be slightly higher in another country, according the efficiency of this Rankine cycle. Also, it will be affected because we have already seen in the second lecture that lower condenser pressure gives you higher work output and it gives higher efficiency.

Therefore, it is always preferable to have a lower atmospheric temperature, but that is something that may not been in our control.

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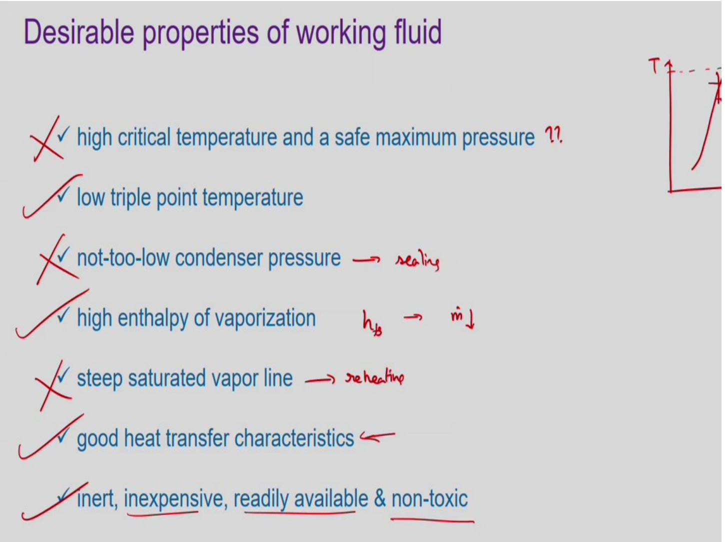

Desirable properties of working fluid

- ✓ high critical temperature and a safe maximum pressure ??
- ✓ low triple point temperature
- ✓ not-too-low condenser pressure → sealing
- ✓ high enthalpy of vaporization $h_{fg} \rightarrow \dot{m}\downarrow$
- ✓ steep saturated vapor line → reheating
- ✓ good heat transfer characteristics ←
- ✓ inert, inexpensive, readily available & non-toxic

Finally, I would like to briefly talk about the desirable properties a working fluid that we are using in Rankine cycle. Water is universally accepted working fluid for commercial power generation, but in certain limited examples we may have some other fluids. Now the desirable properties can be high critical temperature at a safe maximum pressure. If the critical temperature is high then we can perform the phase change operation to a much higher temperature level.

Let us just draw a brief *Ts* diagram. So, normal *Ts* diagram will be something like this, this will be the normal critical temperature. So, this is the maximum temperature up to you can sustain phase change operation is this, the critical temperature. If your critical temperature is high, then we can go for phase change in much higher temperature levels and you do not have to go for too much of superheating. Similarly, we would always prefer this critical temperature to be higher than the metallurgical limit that is your tube can adhere with, which is commonly around 600 to 650 ^{0}C. So, if our working fluid has a critical temperature higher than that 650 ^{0}C it is even better, because than the entire operation entire heat in the boiler can be achieved through the phase change process. We also expect the maximum pressure to be is to be a safer for one, like for water, the maximum pressure can be quite high.

Say, if you want to achieve the phase change operation at temperature of 350 ^{0}C then for water, the corresponding pressure will be something around 16.5 MPa. The saturation pressure corresponding 350 ^{0}C will be around 16.5 MPa, which is really very high pressure to deal with. So if the maximum temperature at which we would like to operate, the saturation temperature corresponding to that or I should say the maximum evaporation temperature at which operating if the saturation pressure corresponding that is lower that is better from material safety point of view. Second one is a low triple point temperature. If the triple point temperature is low, then we can avoid the problem of any kind of freezing. Particularly, if the triple point temperature is lower than the coolant temperature then there will be no possibility of freezing or solidification in your pipe lines and so that is always better.

So, we always want the triple point temperature to be lower than the atmospheric temperature, so that we can avoid any kind of solidification, not too low condenser pressure. Now condenser pressure as I have just mentioned, generally depends on the atmospheric temperature as it is the saturation temperature corresponding to the atmospheric pressure. Now if this condenser pressure is too low, if it is too much below the atmospheric pressure, then there will be every chance of atmospheric air increasing or leaking into the condenser and leading to the aeration of the mixture or also subsequent corrosion. We have to go for deaeration etc, so we always prefer this value to be higher. It should not be to low compared to the atmospheric pressure. Then high enthalpy of vaporization which is of course, is always desirable if your h_{fg}. If the enthalpy of vaporization of the latent heat is high, then of course we can have a lower mass flow rate for a given power output and also, we can maintain a near isothermal profile. Again, because through the phase change only, as the phases changing from the saturated liquid to the saturated vapor state, we can supply significant amount of energy to the working fluid. Then we want a steep saturated vapour line. I am talking about this particular line. If this line is very steep, then, what advantage we can get during expansion? During expansion in the turbine this line or the exit state of the turbine will have a very high quality. So, there will not be significant problem of erosion, that can be achieved only the saturated vapor line is steep one. Good heat transfer characteristics and high thermal conductivity helps in energy transfer to the working medium.

Also, several practical factors, like inert medium it should not be chemically too reactive, inexpensive, readily available, non-toxic etc. These all properties also should be considered. Now water is definitely something that satisfies all these criteria inexpensive, readily available, non-toxic, has a decent heat transfer characteristic etc, and also the enthalpy vaporization for water is also quite good.

So, water may not be inert, because it reacts with most of the metals but we can easily identify materials which are not too much reactive with water. So, what are the working medium scores here, it also scores well here, its enthalpy of vaporization is also quite high and we know the triple point temperature water is 0.0 ^{0}C. Which is generally lower than most of the atmospheric temperature and so there is no problem of freezing. But these are the 3 points, this one this one and this one where water suffers as a working media. Because the condenser pressure for water, we know is very low we always have to go to sub-atmospheric pressures often. In the atmospheric condition atmospheric pressure, the saturation temperature for water is 100 ^{0}C. So, if you want to achieve condensation at 25 ^{0}C, we have to lower the condenser pressure substantially below the atmospheric pressure level.

Similarly, the saturated vapor line is quite relaxed for water on Ts plane. But there are options like, in order to avoid the air increase in the condenser we generally have to provide proper ceiling to the condenser, in order to avoid any kind of leakage and modern designs can of course 100 % leak proof is not possible.

But modern designs allow you a very little amount of leakage. And this particular problem can be taken care of by reheating, so that the quality at the end of expansion, turbine expansion remains quite high. But this is something which is not in our control. Because critical temperature is the property of the water and critical temperature for water is only about 374 ^{0}C.

So, we cannot sustain two phase operation with water beyond 350 ^{0}C. We have to go to superheating stage. So, this is something that often forces us to look form some another alternate medium. But again, there are several advantages for water as we have discussed and also the to problems like the not too low condenser pressure and not steep saturated vapour line that can also be taken care of in practice.

So, quite often instead of dealing with a separate medium all together, then we often go for something known as a binary vapour cycle, where not one rather you to use two working fluids.

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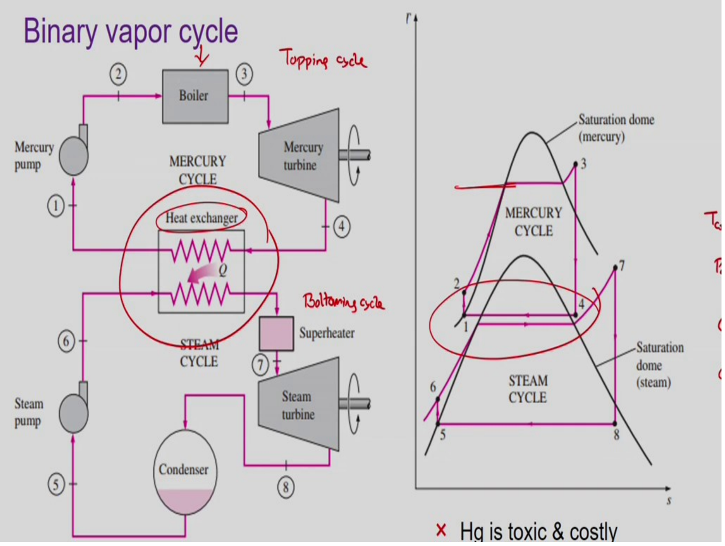

Binary vapor cycle

× Hg is toxic & costly

The one that is shown here with mercury as a working medium is often known as the topping cycle and the other one is using water as a working medium is called the bottoming cycle. Here, we have two Rankine cycles, the topping cycle is a one that is working with a much higher temperature level, here the heat rejection temperature is also significantly high.

Whereas bottoming cycle is the one that is working at low temperature level and it is receiving the heat rejected by the topping cycle in its boiler. So, we are not having any kind of fuel heating or fuel compression in the boiler for the bottoming cycle, rather the heat rejection from the topping cycle is acting as the heat addition to the boiler. Therefore, the condenser of the topping cycle is a boiler for the bottoming cycle, of course we have a normal boiler for the topping cycle as well.

Mercury is shown as the working medium for the topping cycle where we can have a few other metals also like sodium, potassium and sodium potassium mixtures. They can also be working medium. Mercury particularly has several advantages as a working medium. This is a common cycle that we may talk about. Here the topping cycle is working at much higher temperature level where the bottoming cycle which is receiving heat during this process is working at a low temperature level. And as this heat addition process is happening almost isothermally there is very small temperature difference between the mercury and water in the heat exchanger. So, it acts almost like a Carnot cycle and we can have much higher thermal efficiencies. That is why in early 20th century several efforts are done with such kind of binary vapour cycle with mercury as topping cycle, but it has its own problems as well.

Problem like mercury is toxic and it is also much costlier which increases the overall cost of the plant and also much more complicated than the normal Rankine cycle. The enthalpy of vaporization of mercury is quite low. So, we have to go for super heating with the mercury as well. Before that I should also mention the mercury as a work medium has its own advantage. Like the critical temperature for mercury is about 898 ^{0}C which is well above the metallurgical limits. So we do not have to go for too much of superheating with mercury rather we can speak only to the two phase operation. And also, the critical pressure of mercury is approximately 18 MPa which is lower than water. So we can go to very high temperature levels without the need of going to very high pressures.

Like, if you want to use the highest temperature in mercury around 600 ^{0}C corresponding may be significantly lower only around 10 to 12 MPa for this. But one problem with mercury is the low enthalpy of vaporization, here the heat added in the two-phase operation may be quite small, necessitating the superheating. And also, mercury cycle, may not be economically feasible because of the cost of mercury and also the complexity associated with this.

Mercury, this combined operation is quite good because if you want to use mercury as just as a single cycle, then the condenser pressure will be too low. Like at 32 ^{0}C the saturation pressure for mercury is only 0.7 MPa, which is an extremely low temperature on that you are talking about which is impossible to achieve in practice. Whereas if you want to achieve this at reasonable pressure level, then something around the 237 ^{0}C, the saturation pressure of mercury around 7 kPa which normal condensers can work with. And therefore, we can have this heat exchanger working around this 237 ^{0}C there by adding significant amount of heat to the water, for the bottoming cycle. But as mentioned there are problems in the toxicity and cost of mercury and the cycle is not economically feasible.

And also, the gas vapor combined power cycles which are coming later on, they are much superior performance while adopting a similar topping cycle and bottoming cycle. So, these kinds of cycles are often called combined cycle. So, we having topping cycle and bottoming cycle we are getting power output from both of them. But the heat rejection for the topping cycle is used for adding heat to the bottoming cycle, such cycles are called combine cycles.

And though the example that I am showing here is using Rankine cycle as both topping and bottoming one, but it is much more common to have gas vapor combined power cycle, where the topping cycle can be a Brayton cycle and the bottoming cycle can be a conventional Rankine cycle. This is something known as a combined power cycle which is the topic of discussion for the next week.

So binary vapour power cycles, though the concept is very interesting, but may not be very useful in practice, particularly considering the gas vapor combined power cycles. We shall be talking about the combined power cycles in detail from the next lecture onwards.

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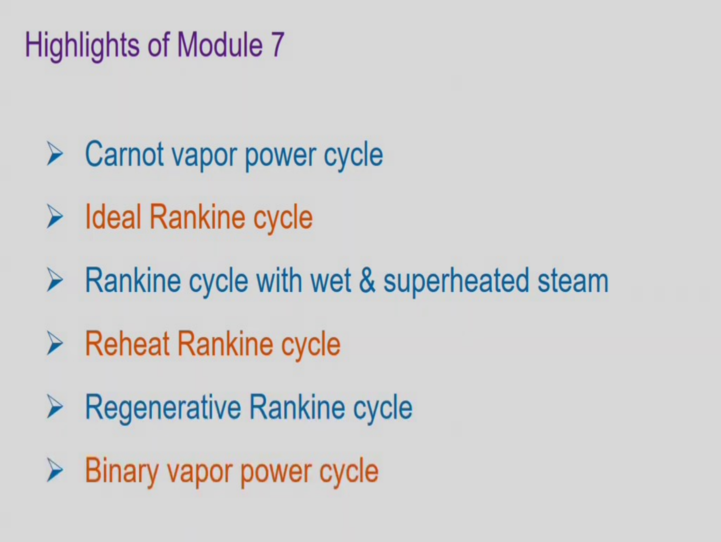

Highlights of Module 7

- Carnot vapor power cycle
- Ideal Rankine cycle
- Rankine cycle with wet & superheated steam
- Reheat Rankine cycle
- Regenerative Rankine cycle
- Binary vapor power cycle

So, what we have learnt in this week? We have learnt about the Carnot vapour power cycle and you have seen why it is not economically feasible of practically feasible. You need to talk about the ideal Rankine cycle, its operation with both wet and superheated steam, then increasing its efficiency in the form of reheating and regeneration and finally finished up with binary vapour power cycle.

That finishes our discussion on the vapour power cycles. Try to solve a few more problems, the problems given in the assignment and if you have any queries, please write back to me and also go through the book. We shall be meeting again in the next week with the combined power cycles. Thank you.