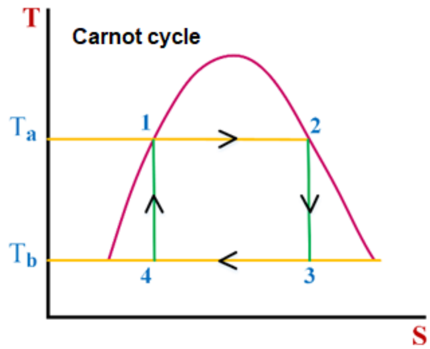
POWER PLANT SYSTEM ENGINEERING

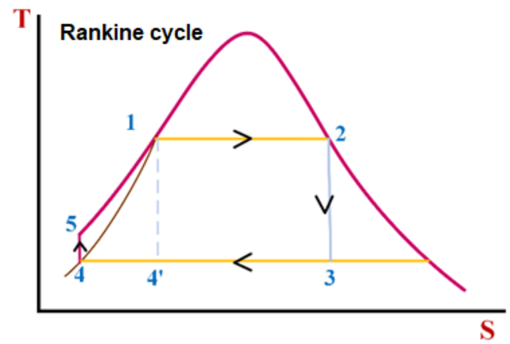
Lec 4: Rankine Cycle

Dear learners, greetings from IIT, Guwahati. Welcome to this course Power Plant System Engineering module 2 Vapor Power System part 1. So, in this lecture we are going to study the most important cycle that are used for steam power systems that is Rankine cycle. In this Rankine cycle we are going to study its theoretical modeling, thermodynamic cycle analysis and we also need to study the different operating parameters for this Rankine cycle. Subsequently we are going to discuss about the effect of irreversibility, losses, process inefficiency and how they are going to affect the cycle. For that we also need to see the effect of condenser and boiler pressures and how the changes in this pressures is going to have a net effect in the Rankine cycles.

So, these are the summary of today's lecture. Now, let me start the first topic that is modeling of Rankine cycles. In our previous lectures, I emphasized about the Carnot cycles and there we have summarized that Carnot cycle is not possible for its implementations in the practical aspects. So, cycle needs to be modified so that its practical utility is ensured.

To summarize some of the important inferences, the first thing I need to emphasize is that, the cycle efficiency for Carnot cycle is almost impractical, but we can do some logical modifications through a Rankine cycle. And in fact, it meets many practical needs, demands to improve the effectiveness. But at the cost of this modifications, the net effect of Rankine cycle is that its efficiency will be smaller than that of the Carnot cycle. So, let us understand why it is smaller than Carnot cycles, because if you look at the T-s diagram the two temperature limits T_a and T_b, $T_a$is the upper temperature $T_b$is the lower temperature. And on this cycle if a Carnot cycle is to operate then we can find out its efficiency.





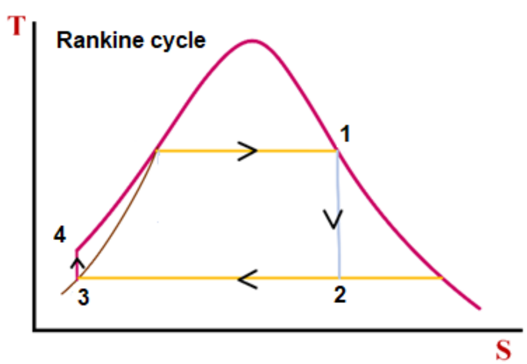
$$\eta = \frac{W_{net}}{Q_{in}};\ \ Q_{5-2} > Q_{1-2} \Rightarrow \eta_R < \eta_C \qquad \oint T\,ds = \oint dQ = |W| \ \Rightarrow W_R > W_C$$

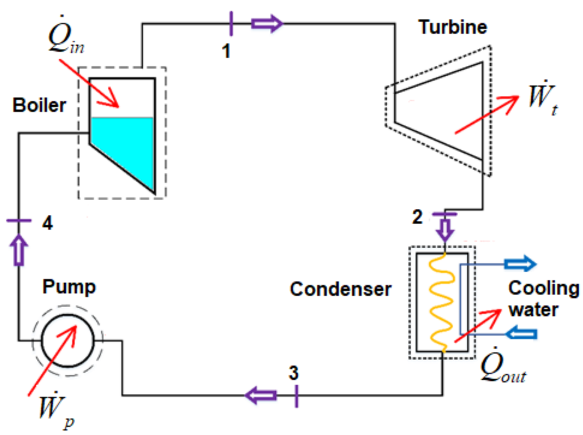
So, since it is a T-s diagram, area under this cycle will give you the net heat transfer. So, the heat addition process in this Carnot cycle is this Q_{1-2}. Now, subsequently when you move on to Rankine cycle, the cycle gets modified as 1-2-3-4- 5. Why? I will come back to this point later and through these things what we will see is that the area under the curve that is heat addition process from 5-1- 2 if you calculate for this Rankine cycle it is always higher. So, comparing this net heat input, when it is higher the thermal efficiency will also be higher, but the advantage that we get is in the form of work ratio or work output.

If you make a comparison of work output, since it is a cyclic process so, net heat transfer is also equal to net work. So, for this Rankine cycle, this area is higher. So, obviously, this work output for Rankine cycle will be higher as compared to Carnot cycle. So, this is the basic advantage that you get just by modifying little bit in this Rankine cycle. Now, what we did in this modification is that, if you compare this Carnot cycle that is 1-2-3- 4 and Rankine cycle that is 1-2- 3-4-5-1, so, if you compare this steam and if I just drop a vertical from point 1 it meets at 4’. So, effectively if you compare in this cycle, Carnot cycle is 1-2-3- 4’. So, what we simply did is that point 4’ was shifted to point 4. What it means in terms of practical utility is that, at point 4’ the state of working fluid is in the liquid vapor region. So, when you move to point 4 it is now saturated liquid and handling this saturated liquid is much easier than the handling of liquid vapor thing. And for that also pumping process will be easier. So, because of these reasons the Carnot cycle was shifted to Rankine cycle.

In addition to that we also have to see the fact that, the heat addition process in the Rankine cycle is increased because if you look at the process 4-5-1, in this heat addition process, what happens is that at constant pressure that is in a T-s diagram and under constant boiler pressure the heat addition process goes from 4 to 5 and from 4 to 5 it is essentially we are adding heat to bring the state of liquid to the state of saturated vapor at point 5. And from 5 to 1 it is again inside the dome. So, it is a liquid vapor region. So, this is the additional process.

Now looking at its implication in terms of its uses, what we think of is that this entire vapor power plant consists of four principal components, that means to make this work in an actual power plant we need to think about four principal components and in fact, this is one of the basic requirement for a vapor power plant. So, the components are turbine, condenser, pump and boiler. These are the four major components and it is presumed that all these components operate in a steady state manner. Other assumptions we make is that, changes in the kinetic energy, potential energy and wherever there is a unavoidable heat losses, they are neglected and energy transfer is either interpreted as work or heat and they are considered component wise and once we consider the component wise then we can bring out the total information about the overall systems. Now, let us understand the each component one by one, first one is the turbine.





So, looking at the cycle let's understand what happens in a turbine. In a turbine, in ideal Rankine cycle, saturated steam at point 1 enters in the turbine and after expansion the steam loses all its energy. So, it comes to the state point 2 that means, 1 - 2 process is the isentropic expansion in a turbine. Then the state of 2 is in the liquid vapor region, and from 2 to 3 the steam loses heat in a condenser and this condenser process is a constant pressure process. Finally, the state of the working fluid is at 3 which is saturated liquid or saturated water.

So, the process in the condenser and that means, condensate leaves at state 3 then it enters to the working feed pump. When it enters to the pump it is in the saturated liquid state. So, its pressure is increased. How much it is increased? It is increased to boiler pressures. So, point 3 to 4 it is a process in the pump and this process is also model as isentropic.

Then from 4 to 5 and subsequently to 1 it is a heat addition process. So, from 4 to 5 the heat addition process takes place to bring the liquid water to the saturated vapor and from 4 to 5 the liquid water goes to saturated liquid at 5. Then from 5 to 1 saturated liquid goes from state 5 to saturated vapor at state 1. So, 4-5-1 process, the entire process happens in the boiler and it is essentially what we call as $to\ Q_{in}$ and again the cycle continues when the steam enters in the turbine.

So, heat addition process that comes from 4- 5- 1, heat rejection process that happens in the condenser that is $\dot{Q}_{out}$ and your turbine work comes as $\dot{W}_t$in the process 1-2 and pump work which is input required is $\dot{W}_p$So, if you want to make a steady state energy balance equations component wise, then we can bring down these expressions

For Condenser: $\frac{\dot{Q}_{out}}{\dot{m}} = h_2 - h_3$, that is done for unit mass basis.

Now for Turbine: $\dot{Q}_{cv} - \dot{W}_t + \dot{m}\left[(h_1 - h_2) + \left(\frac{V_1^2 - V_2^2}{2}\right) + g(z_1 - z_2)\right] = 0; \Rightarrow \frac{\dot{W}_t}{\dot{m}} = h_1 - h_2$

Then for Pump: $\frac{\dot{W}_p}{\dot{m}} = h_4 - h_3,$

& for Boiler : $\frac{\dot{Q}_{in}}{\dot{m}} = h_1 - h_4$

And in fact, many a times when the pump feeds the water to the boiler, we call this as a feed pump and also whatever working fluid enters into the boiler, we call this as a boiler feed water- BFW. Then we need to find out the performance indicator this is the performance indicator for this Rankine cycle, we can go through the cycle 1-2-3-4-5 then 1-2-3-4-5-1.

But now if you want to calculate one by one, thermal efficiency in fact, all the performance parameters I have already mentioned earlier, they are thermal efficiency, turbine work, pump work, net work, work ratio, back work ratio, specific steam consumption and heat rate. These are some of the performance indicator for a steam power plant. So, these things can be calculated by using Rankine cycle. So, first one

Thermal efficiency: $\eta = \dfrac{\left(\dfrac{\dot{W}_t}{\dot{m}}\right) - \left(\dfrac{\dot{W}_p}{\dot{m}}\right)}{\left(\dfrac{\dot{Q}_{in}}{\dot{m}}\right)} = \dfrac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)}$

Thermal efficiency: $\eta = \dfrac{W_{net}}{Q_{in}} = 1 - \dfrac{Q_{out}}{Q_{in}} = 1 - \dfrac{(h_2 - h_3)}{(h_1 - h_4)} = \dfrac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)}$

Work ratio: WR $= \dfrac{\dot{W}_{net}}{\dot{W}_t} = \dfrac{\dot{W}_t - \dot{W}_p}{\dot{W}_t} = \dfrac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_2)}$;

Back work ratio: BWR $= \dfrac{\left(\dfrac{\dot{W}_p}{\dot{m}}\right)}{\left(\dfrac{\dot{W}_t}{\dot{m}}\right)} = \dfrac{(h_4 - h_3)}{(h_1 - h_2)}$

The other interpretation of work ratio is the back work ratio many books interpret as a back work that is pump work to the turbine work. So, both have same meaning. So, ideally speaking our work ratio should be higher or back work ratio should be as small as possible.

The other important aspect is specific steam consumption: SSC $= \frac{1}{W_{net}}\left(\frac{\text{kg}}{\text{kWs}}\right) = \frac{3600}{W_{net}}\left(\frac{\text{kg}}{\text{kWh}}\right)$;

And again heat rate; $HR = \frac{Q_{in}(\text{kJ})}{1(\text{kWh})} \Rightarrow HR\ \alpha\ \frac{1}{\eta_t}$

Then we will try to see what ideal Rankine cycle is. For the ideal Rankine cycle already I have done the thermodynamic analysis, but some of the other reference is that the system is idealized with a viewpoint that there is no irreversibility in the cycle.

So, by irreversibility I mean, if I say boiler pressure is 30 bar, so there is no fluctuation in this boiler pressure and if I say the condenser pressure is let’s say atmospheric, that is also not fluctuating but there are irreversibility, losses in the piping system, that is introduced practically, that is one aspects. Other aspect is that when this turbine and pump they operate they may not operate at its full capacity or full efficiency. So, we introduced these things as an internal effect which drops down the overall performance. So, for that irreversibility comes into pictures.

So, when I say ideal Rankine cycle, the cycle is free from all these uncertainties in terms of irreversibilities. So, in that case what we really say that for an ideal Rankine cycles, for which we have analyze its thermodynamic property or behavior and cycle analysis, what we have assumed is that, it is an isentropic expansion in the turbine, then the heat transfer of working fluid in the boiler as well as in the condenser; they occur at constant pressures, then we have isentropic process in the pump, in the compressed liquid regions, then the heat transfer to the working fluid at constant pressure through the boiler. So, all these things, heat addition process or heat rejection process takes place at constant pressure and the pump and turbine process, they are treated as isentropic.

But there are some situations where an ideal Rankine cycle can be also thought of, instead of saturated steam at state 1 we can add some degree of superheat. So, the state of steam can lie in the superheated region that is 1' and there the turbine process can starts. So, ideally speaking 1-2 or 1'-2', they are the process in the turbine, the very basic difference that we get that, steam at state 1 is heated further before being expanded in the turbines. So, the advantage we get is that, length of this vertical line 1'-2' is higher. So, number-1 advantage; we can extract more work. Number 2 is that, there is a chance that we are going towards the saturated vapor region that means, 2 moves to 2'. So, through this process we can effectively handle the quality of the steam. We will give more details in the subsequent lectures, where we will introduce the term superheat. But anyway now moving further, this is all about the ideal Rankine cycle.

Other process of idealization is interpreted in terms of pump work. Now, if you recall our thermodynamic viewpoint, we say that pump is idealized to operate without any irreversibilities. And in fact, when you say pump it handles liquid which in general is incompressible that means, its density or specific volume do not change. So, if we can idealize this pump process which we call as an internally reversible adiabatic process which means, internally isentropic process, then we can frame this equation or the thermodynamic equation gets modified. So, if you recall

$Tds = dh - vdp$; As it's an isentropic process; entropy is constant, so $ds = 0 \Rightarrow dh = vdp$;

Now, if we apply this equations for the pump, the state points for the pump is 3 and 4. $\Rightarrow h_4 - h_3 = \int_3^4 vdp$; This is nothing, but input work per kg of fluid or water.

So, this becomes

$$\left(\frac{\dot{W}_p}{\dot{m}}\right)_s = h_4 - h_3 = \int_3^4 vdp \approx v_3(p_4 - p_1)$$

Here the assumption made is that since we are handling liquids in the pump the specific volume do not change it comes out of the integrals.

So, this pump equation can be effectively used because normally we know the operating pressures that is$(p_4 - p_1)$. So, this is what the idealization of pump work.

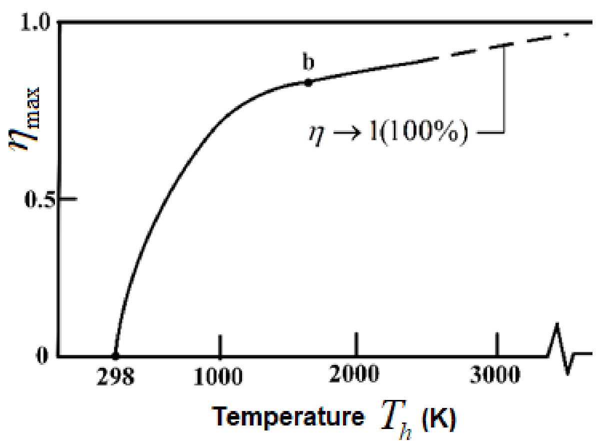
Then let us see how this Rankine cycle is gets compared with respect to Carnot cycles in terms of performance calculations. So, first thing that we need to see is that, work ratio or work output for Rankine cycle is higher, but the thermal efficiency is lower for Rankine cycle in comparison with the Carnot cycle.

But the other side of the story is that, if you really need to see the difference between Rankine cycle and Carnot cycle; the Carnot cycle does not have much flexibility. So, this means that, when you say this Carnot cycle in a T-s diagram as 1-2-3'-4', the expansion work and this compression work they are more or less effectively of same order. So, it has no meaning as per its practical utility is concerned. And main reason being that point 3' is shifted to point 3 and when it is shifted to point 3 you can deal with it easily as we can increase the pressure of the working fluid using the pump. The other thing is that now from process 4 -4', we are adding heat in the Rankine cycle externally so that liquid at state 4 becomes saturated liquid at state 4'. And in fact, in reality these heat addition is normally done through the waste heat or some of the combustion products that goes out of the systems, that is effectively utilized in this heat addition process. But this flexibility is not there in the Carnot cycle. So, that is the main issue; why Carnot cycle is not practically feasible.

Another option is that the Carnot cycle does not have a flexibility to use single phase system which means state 3' in a Carnot cycle is in liquid vapor regions. We need to establish the quality of the state of the working fluid at state 3', but this issue can be resolved by moving the 3' to 3 because information at 3 is known as saturated liquid. So, this allows that we need to condense the vapor completely, handle only liquid as it is done in the Rankine cycle. Pumping from 3 to 4 in liquid phase and heating without work from 4 to 4' are the processes that can be easily achieved in practice. So, 4 to 4' can be achieved by external heating process, through the waste heat or through some other mechanisms which effectively does not come into the performance calculations. So, this is the benefit we get out of this Rankine cycle.

Now let us move on to the thermodynamic angle of this Carnot cycle. Now in a Carnot cycle, it is very difficult to play with the operating pressure because if you look at the efficiency term,

The efficiency of the Carnot cycle is, $\eta_{max} = 1 - \frac{T_c}{T_h}$; T_c stands for the lower temperature, T_h is the higher temperatures. So, efficiency can be increased by two ways- 1) by increasing the T_h 2) by reducing T_c and reducing T_c is not possible because the lower limit is fixed with respect to atmospheric conditions, but increasing T_h is possible.



So, if you plot the efficiency versus T_h for a given T_c , then we can get a curve like this which says that, with increase in the T_h that means, with increase in the heat addition, Carnot cycle efficiency goes up. Now we are going to apply same logic for the Rankine cycles. So, for that purpose we need two important observations because the Rankine cycle consists of components like boiler which operates at higher pressure and condenser which operates at lower pressure.

Let us see what is the effect of increase or decrease of boiler pressure or condenser pressure. So, there are two possibilities here. So, first thing that to get a better efficiency or higher efficiency by reducing T_c or increasing T_h. Now in this T_h for us is nothing, but its correlation is with respect to boiler. So, this T_h or higher temperature of heat addition is dependent on boiler operating pressure and T_c is linked to condenser pressure.

So, our job is to reduce T_c or increase T_h . So, in one case what we have seen is that let us fix the condenser pressure that means, T_c is fixed, we are exploring the how this Rankine cycle looks like.

So, ideal Rankine cycle would be 1-2-3-4-1 that is normal working cycle. Now if I am increasing the boiler pressure. So, the cycle now becomes 1'-2'-3'-4'. Now in this process what did you do? Your mean temperature of heat addition gets increased. So, as a result Q_{in} also increased when Q_{in} increases, your efficiency also increases. So, it means increasing the boiler pressure for an ideal cycle tends to increase the thermal efficiency. The other side of the story is that when we say actual Rankine cycle for a fixed boiler pressure if I drop this condenser pressure down which means that in normal circumstances the water should come down at P_{atm}, but intentionally I keep this condenser pressure below atmospheric so that 2 becomes 2". So, initial cycle which is 1-2-3-4-1 now becomes 1-2"-3"-4". So, when this condenser pressure reduces it tends to increase the thermal efficiency.

So, that is the reason all steam power plant always operates at the most possible or the lowest possible pressure which is the condenser pressures and this is close to maybe 0.008 bar or 0.04 bar or something in that range. And this is generally done to improve the cycle efficiency.

So, whatever I have explained if I just summarize that the lowest feasible condenser pressure is the saturation pressure corresponding to ambient temperature and that is the lowest temperature heat rejection from the turbine. But ideally liquid water at atmospheric pressure could be done directly by boiler or pump or steam that can be discharged to atmosphere by the turbine exit. By including a condenser in the steam side and operating below atmospheric pressure, it will lead to discharge of the steam turbine at lower pressure region thereby improving the thermal efficiency. Another part is that addition of condenser allows to operate in a closed loop and it permits less corrosive than the tap water. So, because of these reasons the thermodynamic aspects always says that higher boiler pressure or lower condenser pressure is the essential motive for the thermodynamic analysis of a steam power system, operating in Rankine cycle.

Apart from that now we are going to look into other possibilities and that other possibilities we categorize as irreversibility and losses. When I say irreversibility and losses it is categorized in two effects, one is internal effects other is external effects.

So, let us understand what this external effects is. So, normally high temperature region that is the boiler pressure and condenser pressure is low pressure region and ideally we say that they have to operate at constant pressure, but due to the frictional losses frictional effects and the piping systems-losses in the piping systems, there is a drop in the pressure. And these are categorized as external effects and the tendency of this external effect is to drop the net power or net work output and they also turn to drop the thermal efficiency. And the internal effect is mainly interpreted in terms of processing efficiency which is introduced in turbines and pump. That means, we say that turbine operates in isentropic manner. Now what happens if they operate in non-isentropic manner? Similarly, pump operates in isentropic manner what happens if it does not operate in isentropic manner.

So, for that reasons we introduced the term called as isentropic efficiency and through this isentropic efficiency we can easily calculate or easily quantify the thermodynamic effects of Rankine cycle by inclusion of irreversibility and losses. So, to do that, for the first instance, we say that as if both the things are operating in irreversibility comes only through this component efficiency of turbine and pump. So, in that case what may happen is that in a turbine process if the process goes in isentropic manner so, it is 1-2s which is isentropic and if it is non isentropic the process goes as a dotted line as shown from 1-2. The same effect comes that means, instead of the saturated liquid regions, the discharge may happen at further drop in pressure.

So, it comes at 3. So, it is a sub cooled region which is 3. Now, from 3- 4 if the process has to happen, for a pump in an isentropic manner, so, we say 3- 4s is isentropic and 3- 4 is non isentropic. So, net effect what we will see here is that, in a non isentropic turbine process your work output will be less, but in a non isentropic pump process your input will be more.

Now we are going to introduce this through the expression which is called isentropic efficiency for turbine which is a net work output per unit mass in an actual process divided by network in output for an isentropic process.

Turbine Efficiency: $\eta_t = \dfrac{\left(\frac{\dot{W}_t}{\dot{m}}\right)}{\left(\frac{\dot{W}_t}{\dot{m}}\right)_s} = \dfrac{h_1 - h_2}{h_1 - h_{2s}}$

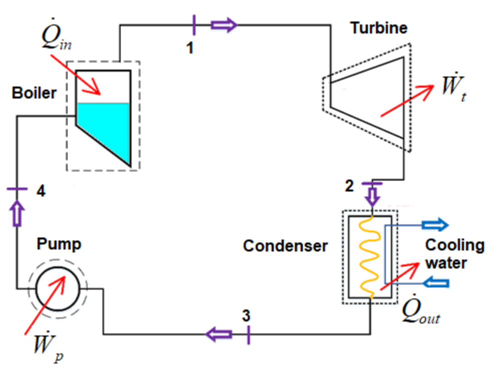
Now, similar way we can write down the pump efficiency because in case of pump your net work input will be higher, which means that your isentropic work has to be lower.

$$\eta_p = \frac{\left(\frac{\dot{W}_p}{\dot{m}}\right)_s}{\left(\frac{\dot{W}_p}{\dot{m}}\right)} = \frac{h_{4s} - h_3}{h_4 - h_3} = \frac{v_3(p_4 - p_3)}{h_4 - h_3}$$

So, this is all about the thermodynamic cycle analysis for a Rankine cycle and how you are going to introduce the irreversibility and losses, component wise. So, with this basic background let us try to solve some numerical problem.

Q1. In a Rankine cycle, the saturated vapor enters the turbine at 8MPa and saturated liquid leaves the condenser at 0.008MPa. The net power output of the cycle is 85 MW. Determine the following parameters for the cycle: (a) thermal efficiency; (b) WR & BWR; (c) mass flow rate of steam; (d) the rate heat transfer into the working fluid in the boiler and condenser; (e) mass flow rate of cooling water in the condenser.

So, the first problem is again an ideal Rankine cycle which says that saturated vapor enters the turbine at 8MPa.

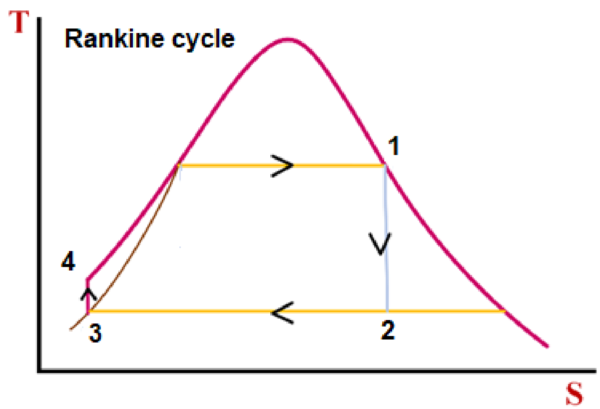


This is state 1. So, if you recall this ideal Rankine cycle the circuit diagrams saturated steam enters at 8MPa and saturated liquid leaves the condenser at 0.008MPa.

So, at the state 3, that is categorized at 0.008MPa. Net power from this entire cycle which is nothing, but W_{net} = turbine work - pump work= 85 MW. So, we need to find out the thermal efficiency, work ratio, back work ratio, mass flow rate of the steams and all. So, I have already given the expressions for these parameters.

To solve this kind of problem, we have to first rely on the steam table. So, you must understand how you are going to find out the state points by looking at the steam tables, from the data given in this problem.

To know the exact state, the ideal choice that we should draw is this T-s diagram first.



So, in this T-s diagram we are going to locate all these points. So, process 1-2 is turbine, 2-3 is condenser, 3-4 is pump and 4-1 is the boiler and vapor pressure in turbine process 1-2 is at $p_1 =$ 8MPa and in condenser process 2-3, it is at p_3 =0.008MPa.

We all know 1-2 is isentropic, 2 -3 is constant pressure, 3-4 is again isentropic, 4 - 1 is again constant pressure.

So, from this data given, first we need to know the use of steam table and try to find out the state points definition in this T-s diagrams.

So, we say saturated steam is at 8 MPa. So, this you can refer as saturated steam table based on pressure. So, at this point we can calculate,

h_1= 58 kJ/kg

Since $s_2 = s_1$ so, we need to find out what is s_1 at 8 MPa.

s_1 = 5.7432 kJ/kg.K

So, at state 2 we can find out $s_2 = s_1$ and

$s_2 = s_f + x_2(s_{fg})$; Where $s_{fg} = s_g - s_f$

$$\Rightarrow \; x_2 = \frac{s_2 - s_f}{s_g - s_f}$$

At 0.008 MPa, $s_f = 0.5926$; $s_g - s_f = 7.6361$; $h_f = 173.88$ kJ/kg ; $h_{fg} = 2403.1$ kJ/kg

Dryness fraction at state 2; $x_2 = \frac{s_2 - s_f}{s_g - s_f} = 0.6745$

$h_2 = h_f + x_2(h_{fg})$ = 1794.8 kJ/kg

Now having said this we have state 1, state 2 information. State 3 information is already built in h_f and s_f. So, from this, we can find out

h_3 = 173.88 kJ/kg; $h_4 = h_3 + v_3(p_1 - p_3)$ = 181.94 kJ/kg

So, we have all the information which is required to calculate these parameters. So, let us go one by one.

a) "Thermal efficiency": $\eta_t = \frac{W_{net}}{Q_{in}} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_4)} = 37\%$

b) "Work ratio" : WR $= \frac{\dot{W}_{net}}{\dot{W}_t} = \frac{(h_1 - h_2) - (h_4 - h_3)}{(h_1 - h_2)} = 99\%$

"Back work ratio: BWR" $= \frac{\left(\frac{\dot{W}_p}{\dot{m}}\right)}{\left(\frac{\dot{W}_t}{\dot{m}}\right)} = \frac{(h_4 - h_3)}{(h_1 - h_2)} = 0.83\%$

.So, we can see this number is too less. That means, this is the essential necessity that back work ratio should be as low as possible

c) Mass flow rate of steam, $\dot{m}$,

We can say $\dot{W}_{cycle} = \dot{m}\, \dot{W}_{net} \Rightarrow \dot{m} = \frac{\dot{W}_{cycle}}{\dot{W}_{net}} = \frac{85 \times 10^3}{955} = 89$ kg/s

d) $Q_{in} = \dot{m}\,(h_1 - h_4)$ = 229.3 MW

$Q_{out} = \dot{m}\,(h_2 - h_3)$ = 144.3 MW

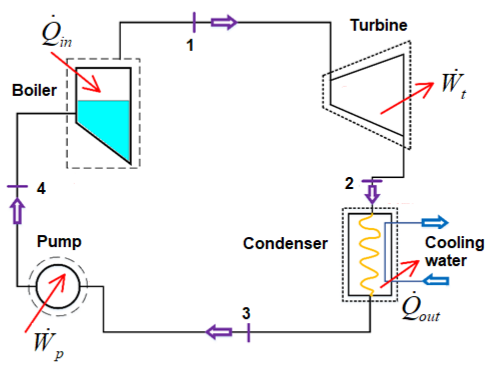
e) Last one is Mass flow rate of cooling water.

So, here what happens this Q_{out} which goes out of the condenser it is taken away by this cooling water. So, we can make an assumption that at some temperature the cooling water enters and some temperature it leaves. A tentative practical inlet temperature could be 20℃, outlet temperature of this cooling water will be let's say 38℃. Now, with these things we require what is the cooling water requirement.

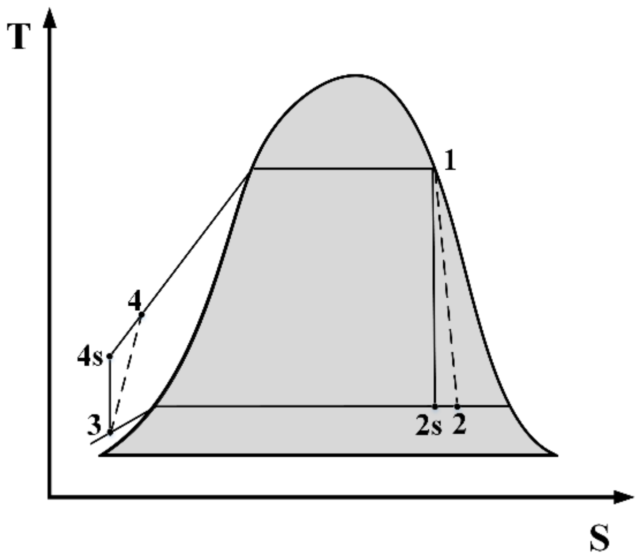
$$\text{So, } \dot{m}_w = \frac{\dot{Q}_{out}}{c_{pw}\,(T_{out} - T_{in})} = \frac{144300}{4.18(38-20)} = 1918 \text{ kg/s}$$

So, what we see is that steam flow rate or working fluid flow rate should be 89 kg/s & for this 89 kg we require the cooling water flow rate 1918 kg/s Of course, this is a huge power plant producing 85MW.

Q2. In a Rankine cycle, the saturated vapor enters the turbine at 8MPa and saturated liquid leaves the condenser at 0.008MPa. The net power output of the cycle is 85 MW. Determine the following parameters for the cycle: (a) thermal efficiency; (b) WR & BWR; (c) mass flow rate of steam; (d) the rate heat transfer into the working fluid in the boiler and condenser; (e) mass flow rate of cooling water in the condenser. Assume, isentropic efficiency of turbine and pump as 88%.



Now, next problem which imposes the component efficiency. So, the component efficiency when it is introduced, we say the turbine and pump operate at 88 percent efficiency. Now, for that process how the cycle is going to be modified we can redraw the cycle in a manner that we can introduce the component efficiency.



So, instead of 1-2s we say, it is 12. Now, instead of 3-4s we say it is 3-4. So, rest of the things remain same. So, we can extract all the data

h_1 = 2758 kJ/kg
h_{2s} = 1795 kJ/kg.
h_3 = 174 kJ/kg
h_{4s} = 182 kJ/kg
In addition to that, the cooling water enters at T_{in} =20 ℃ & leaves at T_{out} = 38℃ Net work output from the cycle, $\dot{W}_{cycle}$ = 85 MW

For compressor, η_c = pump efficiency and turbine efficiency, η_c = 0.88.

Now, with these information let's try to solve. So, first thing we have to use this expression

$$\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}} = 0.88$$

From this data given we can get, h_2 =1910.6

Similarly, pump efficiency

$$\eta_p = \frac{v_3(p_4 - p_3)}{\dot{W}_p / \dot{m}} = 0.88$$

From this data given we can get, ${\dot{W}_p}/{\dot{m}}$ =9.1 kJ/kg

$$h_4 = h_3 + {\dot{W}_p}/{\dot{m}} = 183.1 \text{ kJ/kg}$$

Now in the same method we can find out all the asked parameters one by one

a) Thermal efficiency": $\eta_t = \frac{\dot{W}_t - \dot{W}_p}{Q_{in}} = \frac{(2758-1910.1)-9.1}{(2758-183.1)} = 32.6\%$

b) "Work ratio" : WR $= \frac{\dot{W}_{net}/\dot{m}}{\dot{W}_t/\dot{m}} = \frac{838.3}{(h_1-h_2)} = 98\%$

So, if you recall our previous data a problem data and this data work ratio drops by only 1% . So, from 99% to it becomes 98%

"Back work ratio: BWR" $= \frac{9.1}{847.4} = \frac{(h_4-h_3)}{(h_1-h_2)} = 1\%$

So, even if you introduce the component efficiency, it does not have a significant implications on back work ratio.

c) Then mass flow rate of steam, $\dot{m} = \frac{\dot{W}_{cycle}}{(h_1-h_2)-(h_4-h_3)} = \frac{85\times10^3}{838.8}$

This number we got from the previous example. So, $\dot{m} = 101.4$ kg/s

In earlier problem, it was 89 kg/s. So, by introducing component efficiency your steam consumption gets higher.

d) And similar way we can find out

$\dot{Q}_{in} = 261$ MW in boiler & $\dot{Q}_{out} = 176$ MW in condenser

e) And cooling water requirement $\dot{m}_w = 2339$ kg/s

So, this number is also higher.

So, we have seen that, first of all the efficiency drops, work ratio is not much affected, steam consumption rate is higher and cooling water requirement is also higher.

Why? Because we have introduced irreversibility into the systems through the component efficiency of turbine and pump. So, that is again the basic lead or analysis or design of a steam power plant what it says is that under no circumstances it is advisable to drop work ratio because it will not justify the utility of power. So, with this I conclude. Thank you for your attention.