Steam Power Engineering

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Lecture -03
Performance Estimation of Steam Power Cycles

Welcome to the class. In last class we had seen that the Rankine cycle how does it operate? And then what are the components of Rankine cycle? What are the processes which comprise Rankine cycle and which are the components which execute those processes? Which are the thing which should be known to us in the different corners of the Rankine cycle? Those are also seen and if we want to execute heat and energy interact heat and work interaction for the Rankine cycle; How to proceed with?

Then we had also seen that how to look into the steam table and find out different enthalpies or entropies whatsoever they are required for us. Having those things in that known we have also seen what are the different performance parameters of the Rankine cycle which we are like different efficiencies. Now, we are going to go ahead and see what is the; what are the other factors which would be dealing with the steam power cycles?

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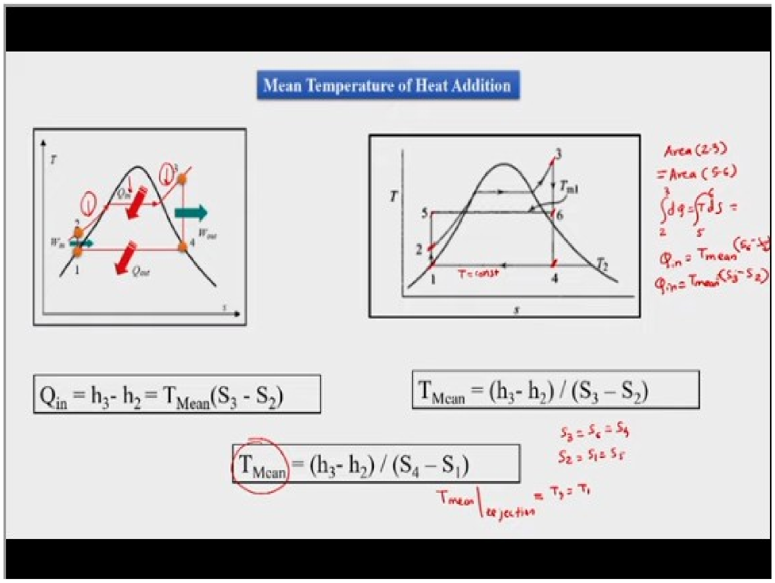

Mean Temperature of Heat Addition

Area (2-3)
= Area (5-6)

$\int_2^3 dq = \int_5^6 T\,ds =$

$Q_{in} = T_{mean}(S_6 - S_5)$

$Q_{in} = T_{mean}(S_3 - S_2)$

$Q_{in} = h_3 - h_2 = T_{Mean}(S_3 - S_2)$

$T_{Mean} = (h_3 - h_2) / (S_3 - S_2)$

$S_3 = S_6 = S_4$

$S_2 = S_1 = S_5$

$T_{Mean} = (h_3 - h_2) / (S_4 - S_1)$

$T_{mean}\big|_{rejection} = T_2 = T_1$

So, the first thing what we would be discussing today is mean temperature of heat addition. Here, we know that our Rankine cycle is comprised of 4 processes. In which process 2 to 3 is heat addition process at constant pressure but the process 2 to 3 has pressure at to be constant but temperature is expected to vary in this in this manner as this heat addition. As we have seen is done in economizer this heat addition is done in latent phase latent heat is added in the evaporator and the super heat is added into the superheater.

So, the sensible heat part which deals with the 2 components which are economizer and superheater they lead to a change in temperature. But we would like to find out what is the mean temperature of heat addition or may be mean temperature of heat rejection as well. So, in this case we would do this thing we can reformulate the same cycle with a new sketch where process 2 to 3 is replaced by process 5 to 6 such that area under the curve 2 to 3.

So, area under the curve 2 to 3 process is equal to area under the curve 5 to 6; but the peculiarity of process 5 to 6 is it is a constant temperature process. However, we don’t say here the process 5 to 6 is a isobaric heat addition but we are interested in finding out what is the mean temperature of heat addition in case of the boiler. So, having known this we know now $Q_{¿}$ is amount of heat added into the boiler which is $Q_{¿} = h_3 - h_2$.

But $h_3 - h_2 = T_{mean}(S_3 - S_2)$. Basically, this is the constant temperature we know that $TdS = dh - v\,dP$ we know from second law$dQ = TdS$. So, if we integrate from 2 to 3 so but area under the curve is same so this integration can be done between 5 to 6 $\int_2^3 dQ = \int_5^6 TdS$ .So, 5 to 6 is a constant temperature process this is$Q_{¿}$, so 5 to 6 is a constant temperature process.

So, which we call it as $T_{mean}$and then it happens to be $T_6 - T_5$but T_6 is equal to $T_6 = T_4$ or T_3 and $T_2 = T_1$ so sorry $S_6 = S_4$or S_3 and $S_2 = S_1$or S_5. So we can comfortably write here that $S_6 - S_5$ so $S_6 = S_3$ and S_2 is as it is $S_5 = S_2$. So, heat addition can be found out by $h_3 - h_2$; but $h_3 - h_2 = T_{mean}(S_3 - S_2)$. Then we have $T_{mean} = \frac{h_3 - h_2}{S_3 - S_2}$ which further would lead to $T_{mean} = \frac{h_3 - h_2}{S_4 - S_1}$.

Since, $S_3 = S_6 = S_4$ and $S_2 = S_1 = S_5$ which we can take any of the entropies. So, this is what the method of calculation of T_{mean} for the isobaric process of heat addition. Parallely, if I am interested in finding out what is the mean temperature of heat rejection but then finding out that is not that difficult since in the process 4 to 1. Although we say that it's an isobaric process but this process is a phase change process so there temperature is also constant.

So, we actually have temperature constant in this process so mean temperature of heat rejection $T_{mean \vee Rejection} = T_4 = T_1$ so that is not that difficult to find out. If this process would have variation of temperature then we would have evaluated it by the same method as what we did for the temperature addition process, heat addition process.

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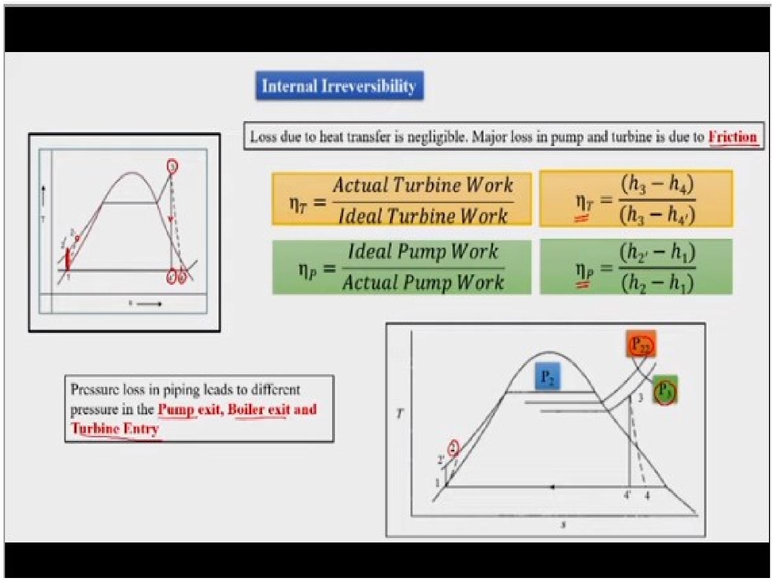

Internal Irreversibility

Loss due to heat transfer is negligible. Major loss in pump and turbine is due to **Friction**

$$\eta_T = \frac{Actual\ Turbine\ Work}{Ideal\ Turbine\ Work} \qquad \eta_T = \frac{(h_3 - h_4)}{(h_3 - h_{4'})}$$

$$\eta_P = \frac{Ideal\ Pump\ Work}{Actual\ Pump\ Work} \qquad \eta_P = \frac{(h_{2'} - h_1)}{(h_2 - h_1)}$$

Pressure loss in piping leads to different pressure in the **Pump exit, Boiler exit and Turbine Entry**

Then we are talking about now till time we were discussing about Rankine cycle which is an ideal cycle where all the components behaved as the ideal components where there are no losses. But now we might encounter with the losses in the different components of Rankine cycle. Here, let us consider the irreversibility which is of internal kind; we mean internally irreversibility mean the flow which is passing through the different components, encounters, the loss while flowing into the circuit of the Rankine cycle.

So, loses due to heat transfer are negligible into the pump and turbine because the process which is getting executed in pump and turbine is isentropic expansion in the turbine and isentropic compression in the pump. The process being isentropic it is very fast it is so it is adiabatic it is as well reversible that is what our expectation is. The loss in this process which is 1 to $2'$ or 3 to 4 or in the pump or in the turbine takes places so rapidly such that there is no much chance for heat to be getting lost or heat interaction with the surrounding for the working medium.

So, we except that the losses due to heat transfer are negligible in the pumps and in the turbines. But the major loss is loss due to friction; so frictional losses are important into the turbine and pump. So, we have to find out what is the efficiency of turbine? So turbine efficiency(η_T) or isentropic efficiency of a turbine is defined as actual work output of the turbine divided by ideal work output of the turbine.

We know that turbine is a work producing machine so actually it would produce lesser work than the ideal case or than the required. Hence, in the case of the Rankine cycle over here we would have to come isentropically down from 3 to $4'$when we are talking about ideal process of expansion the isentropic expansion. But instead of coming straight in the process we would come by an unknown path but reached to point 4 and that path being unknown and irreversible we are showing it by a dotted line.

So, ideally we have got a expansion from 3 to 4 ideally we had got a expansion from 3 to $4'$. So, efficiency of the turbine is actual work which is $h_3 - h_4$divided by ideal work which is $h_3 - h_{4'}$. Then we have pump work; in case of pump as well we except towards straight up as what excepted from 1 to $2'$. But, we encounter frictional loss in the pump and then the process in reality would lead to point 2 instead of $2'$.

So, here we know that pump is our absorbing machine it being an absorbing machine ideally it would absorb less work but actually it would absorb more work. So, pump efficiency(η_P) is defined as ideal work input divided by actual pump work input. Therefore, in the present case we have $\eta_P = \frac{h_{2'} - h_1}{h_2 - h_1}$.So, these two efficiencies are very much required when we are dealing with turbines or pump or may be compressors when they are not ideal.

Further there will be loss due to the piping system where pressure loss is taken place due in the pump exit, boiler exit and turbine entry. We except the process 2 to 3 to be a isobaric process but in the case where we having different piping and further there is frictional loss, the pressure in the circuit gets decreased. So, we expect that there is two pressures at the entry to the boiler or steam generator which is at the exit to the pump but we get different pressure at the exit of the boiler and then further we get different pressure at the entry to the turbine.

So, the three entities which expected to have pump exit, boiler completely and turbine inlet which was expected to have same pressure due to pressure loss in friction or due to piping the encounter loss in head.

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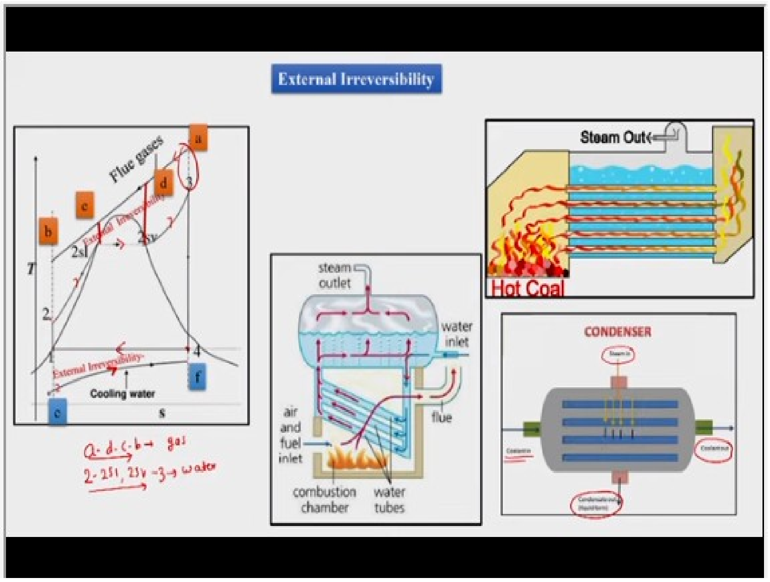
External Irreversibility

But then there is external irreversibility also along with internal irreversibility. When we say that that there is external irreversibility we mean that we are talking about the surrounding which is supplying the heat or which is losing the heat in to the working medium which is steam. Parallely, if we see the steam generator then typical circuit in the steam generator is like this we which we are going to see, elaborate in the class of the steam generator.

But here we have water getting heated where the combustion chamber where having combustion products generated due to the process of combustion maybe we can use oil or we can use coal those gases of combustion would pass over the tubes which are having the water. So, this way we are having heat transfer between the flue gases and the working medium which is water. Otherwise also we would have water stored in a tank and then the hot coal would get burnt and then the flue gases would get transferred through the tubes.

And in this process they would lose their heat to the water which would get converted into the steam and further steam would be passed to the turbine this is the case for the boiler. Further in case of condenser we expect that steam would enter into the condenser then there will be coolant in which is a coolant; generally as I said it is natural resources which are used for the process of condensation.

So, water from river or ocean would be used here and then that water would take the heat from the steam which is entering into the condenser water will get heated in this process. So, this is the coolant which is hot now at the exit of the condenser but steam which is entering into the condenser would get condensed to the liquid at the outlet. So, there is heat transfer between coolant water and steam.

There is heat transfer between combustion products and liquid water or water in principle in 2 phase. So, we expect that the process 2 to 3 is heat addition process into the steam generator or boiler. But for that heat addition we expect that there is the flue gas which is passing on the $T-S$ diagram like this in a process. If d, c, b process; process a, d, c, b is for gases and process 2, 2s1, 2sv and 3 is for water. So, water will get heated in this phase and then flue gases will get cold or become colder in this process.

So, the temperature difference between these 2 entities flue gas and the steam is leading to irreversibilities and these are called as external irreversibilities. Further, there would be external irreversibility in the condenser also we have the condensation process between 4 to 1; 4 is the entry to the condenser for the steam and 1 is the exit to the condenser for the water. This condensation would take place with liquid water from the natural resource which will get heated from e to f. But the major point remembered over here that 1 entity which is steam it changing its phase from 4 to 1.

So, whether the flow of water is in the direction or parallel to the steam or water is not going to affect the amount of heat transfer or the irreversibility which are incurred in the process of condensation. So, this has this part which is the condenser has less amount of improvisation chance for the external irreversibilities. But this part which is the external irreversibility dominant part which is a boiler part has imp lot of scope for improvisation to remove or reduce the external irreversibilities.

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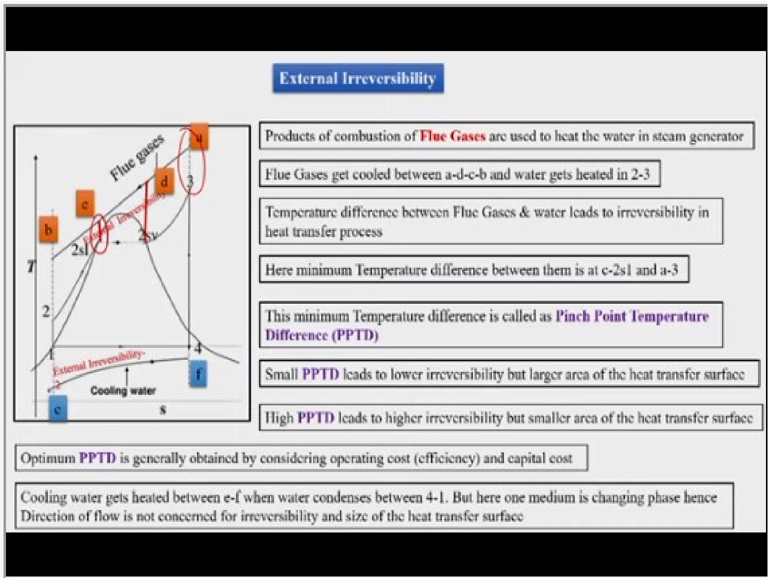

External Irreversibility

Products of combustion of **Flue Gases** are used to heat the water in steam generator

Flue Gases get cooled between a-d-c-b and water gets heated in 2-3

Temperature difference between Flue Gases & water leads to irreversibility in heat transfer process

Here minimum Temperature difference between them is at c-2s1 and a-3

This minimum Temperature difference is called as **Pinch Point Temperature Difference (PPTD)**

Small **PPTD** leads to lower irreversibility but larger area of the heat transfer surface

High **PPTD** leads to higher irreversibility but smaller area of the heat transfer surface

Optimum **PPTD** is generally obtained by considering operating cost (efficiency) and capital cost

Cooling water gets heated between e-f when water condenses between 4-1. But here one medium is changing phase hence Direction of flow is not concerned for irreversibility and size of the heat transfer surface

Then as we said that we have flue gases which are going to warm or which are going heat the water into the steam generator. Then these flue gases would get cold or become colder which will get cooled in the process a, d, c, b and water would get heated in the process 2 to 3. Temperature difference between flue gases and water leads to irreversibility in the heat transfer process as what we have discussed earlier.

We know that any process would be reversible when there are less and less amount of gradient or rather practically when there are gradients are 0, processes are reversible. But if this big temperature gradient exist between system and surrounding then process would be irreversible.

So, the minimum temperature difference is noted at 2 points. This point is minimum temperature difference point and this point is also a second minimum temperature difference point and these points were minimum temperature difference between the flue gas and steam is encountered is called as pinch point temperature difference, pinch point temperature difference.

So, it is a minimum difference between the working medium and the flue gases. Pinch point difference actually should be as minimum as possible such that we can reduce the irreversibilities which are there in the process of heat transfer. If we want to reduce the irreversibilities we have to reduce the pinch point temperature difference. But we know that heat transfer is dependent upon temperature difference but along with that it is also dependent upon surface area.

But we want certain amount of heat to be transferred from the flue gas to the water. So, we expect $h_3 - h_2$ is amount of heat which needs to be transferred from flue gas to the water. But for that amount of heat to be transferred we need that strong temperature gradient or we need very large surface area. So, in order to have pinch point difference lower we have to increase surface area. So, here optimum pinch point temperature difference it based upon the cost effectiveness of the user of the plant or designer.

So, cooling water case as we discussed it is not going to alter any change in irreversibilities since one of the materials is go undergoing the phase change. So, the temperature difference between the condenser gas or condenser steam or water and the water which is the coolant is not going to decide the irreversibility which is encountered in the process of condenser. So, at gross there are external irreversibilities like internaly irreversibilities and we have to work towards reduction of internal and external irreversibilities.

Internal irreversibilities as we have discussed there are mainly due to friction and making the passages very smooth we can reduce the internal irreversibility. Similarly external irreversibility can be reduced if we can reduce the temperature difference between the flue gases and the steam which is passing through the steam generator.

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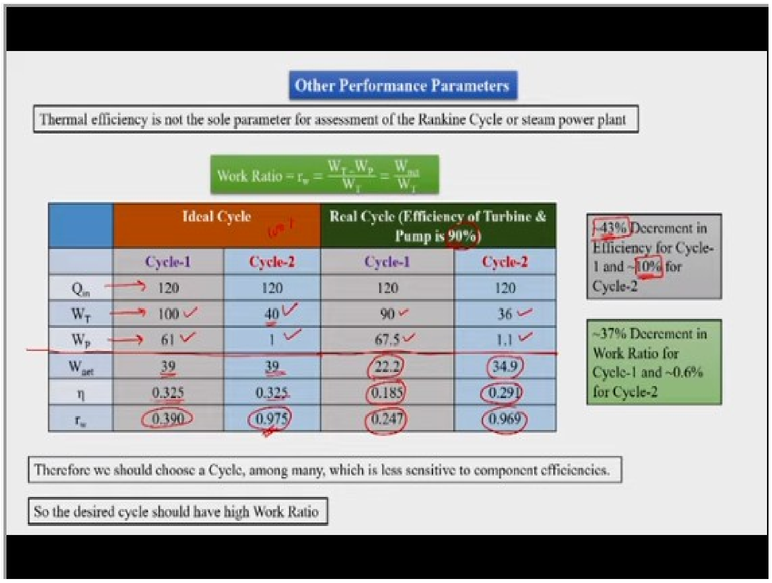

Other Performance Parameters

Thermal efficiency is not the sole parameter for assessment of the Rankine Cycle or steam power plant

Work Ratio = $r_w = \frac{W_T - W_P}{W_T} = \frac{W_{net}}{W_T}$

	Ideal Cycle		Real Cycle (Efficiency of Turbine & Pump is 90%)	
	Cycle-1	Cycle-2	Cycle-1	Cycle-2
Q_{in}	120	120	120	120
W_T	100	40	90	36
W_P	61	1	67.5	1.1
W_{net}	39	39	22.2	34.9
η	0.325	0.325	0.185	0.291
r_w	0.390	0.975	0.247	0.969

~43% Decrement in Efficiency for Cycle-1 and ~10% for Cycle-2

~37% Decrement in Work Ratio for Cycle-1 and ~0.6% for Cycle-2

Therefore we should choose a Cycle, among many, which is less sensitive to component efficiencies.

So the desired cycle should have high Work Ratio

Then there are other performances parameters for the steam power plant. Thermal efficiency is basically not the only performance parameter upon which steam power plant is judged. So, the others performance parameters is called as Work ratio(τ_w). Work ratio here is defined $\tau_w = \frac{W_T - W_P}{W_T}$. So, $W_T - W_P$is a net work and W_T is the turbine work. So, net work upon turbine work is the work ratio. Now, let us see significance of this term which is called as work ratio. Consider that there are 2 cycles; cycle 1 and cycle 2.

In case of cycle 1 and cycle 2 initially let us consider that both the cycles are ideal. So, cycle 1 can be one ideal cycle and cycle 2 can be other ideal cycle. Here, in the cycle $Q_{¿}$ is amount of heat added and which is same for both the cycles. But W_T in cycle 1 we get 100 unit of turbine work but in other cycle we get 40 units of turbine work. Let us consider that the pump work for cycle 1 is 61 but for the other cycle is further small and it is 1.

So, till this point we have assumed; we assumed $Q_{¿}$, we have assumed W_T; we have assumed W_P so $W_T - W_P$ is net work. So, we see that net work is same in cycle 1 and in cycle 2. So, we can see the deficiencies also same where efficiencies defined as net work upon $Q_{¿}$ so $Q_{¿}$ is same network is same so efficiency is same. But, now let us find out which is called as work ratio?

This work ratio is defined as $\frac{W_{net}}{W_T}$. we have seen that $W_{net} = 39, W_T = 61$.

So, $\frac{39}{61}=0.390$ work ratio for cycle-1 but very high work ratio for cycle 2 which is 0.975 but this was all for the ideal cycle. Let us consider the same 2 cycles; cycle 1 and cycle 2 with 90% efficiency of the components and major components as what we say turbine and pump. So, $Q_{¿}$ is same but W_T was 100 but now efficiency is 90% so turbine work is 90 and turbine work from 40 would get released to 36. Pump work is 61, $\frac{61}{0.9}$ so we get 67.5 as pump work in cycle-1 and we get $\frac{1}{0.9}=1.1$.

So, this is now again known thing. Here onwards W_T-W_P is net work which is here and we have net work as this much in cycle 2. Then $\frac{W_{net}}{Q_{¿}}$ is efficiency of cycle 1, this is efficiency of cycle 2.But then we can as well calculate work ratio. The major points to be noted over here we get 43 percent decrement in efficiency for cycle 1 and 10 percent for cycle 2. We can see that without having any change in the Qin, we got higher work ratio initially for cycle 2.

But that high work ratio has indicated one thing which we are understanding now that if work ratio is high then the efficiency change due to component efficiency is less and in present case it is only 10 percent for cycle 2. But in case of cycle 1 there is lot of change in efficiency and it is 43 percent. So, if work ratio is low then component efficiency would lead to lot of change in the thermal efficiency of the cycle.

But if work ratio is high then we don't have to worry too much that would not alter the efficiency of the cycle. There is 37 percent decrement in work ratio but 0.6 percent decrement in work ratio for cycle 2. So, this is for the indication what work ratio gives us therefore one should choose a cycle thermodynamic cycle which has less sensitive to component efficiencies and desired value of work ratio is as high as possible.

We have seen that just we made component efficiency from 100 percent to 90 percent, then that small change in the efficiency of the components lead to large change in turbine work and so it has lead to large change in W_{net} and it has large change in thermal efficiency for cycle 1 and cycle 2. So, what we expect is to use a thermodynamic cycle which has higher work ratio. We use any cycle which has higher thermal efficiency since it can utilize the available $Q_{¿}$ effectively and produces maximum network but this is not the sole method to compare one cycle with the other.

In that we have seen that work ratio is an other parameter which is very important. Since it tells us that what is the life cycle of the component or till what time we have to continue with the existing component. So, work ratio gives a light upon effectiveness or how much would be the change in efficiency if we have certain amount of change in the component efficiencies. So, if system is more reliable upon component efficiency then in that case we have to use higher work ratio cycle.

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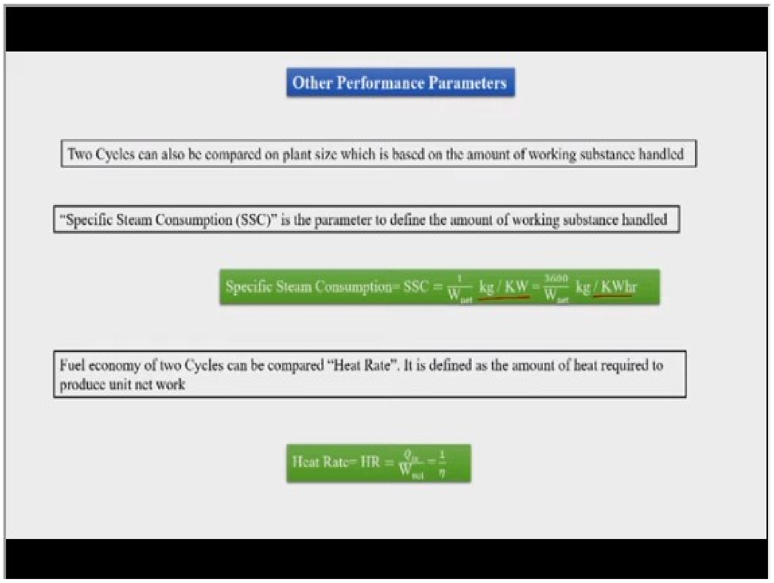

Other Performance Parameters

Two Cycles can also be compared on plant size which is based on the amount of working substance handled

"Specific Steam Consumption (SSC)" is the parameter to define the amount of working substance handled

Specific Steam Consumption= $SSC = \frac{1}{W_{net}}$ kg / KW = $\frac{3600}{W_{net}}$ kg / KWhr

Fuel economy of two Cycles can be compared "Heat Rate". It is defined as the amount of heat required to produce unit net work

Heat Rate= $HR = \frac{Q_{in}}{W_{net}} = \frac{1}{\eta}$

So, there is other performance parameter if there are two cycles which are getting compared with each other. Then we have following thing which are important to be used to compare two cycles of different power plants and one such is called as specific steam consumption or steam rate. So, this steam rate is defined as the $\frac{1}{W_{net}}$ amount of steam required to produce unit amount of work. So, if we multiply that unit by 3600 then it turns out to be kg per kilo watt hour.

So, $\frac{1}{W_{net}}$ is specific steam consumption and $\frac{3600}{W_{net}}$is again specific steam consumption but with different unit which is kg per kilo watt hour and in one case you need is kg per kilo watt. So, this is the other performance parameter which is called as specific steam consumption or steam rate. The other performance parameter is called as heat rate. It is defined as amount of heat required to produce unit amount of net work.

So, amount of heat input divided by net work is practically $\frac{1}{\eta}$. Specific steam consumption actually tells us about how much steam is required to produce the given work output. And if specific steam consumption is high yes we need a bigger boiler, specific steam consumption is low yes we need a smaller boiler for storage or generating the steam. Similarly, if heat rate is higher that means commotion on the fuel is of not good quality.

And then heat rate means we are having amount of heat to be higher if we need we are getting same amount of work from others. So, this is how much amount of coal, oil to be burnt to generate certain amount of power that is what heat rate is. What is the amount of steam which needs to be supplied to the turbine to produce work is called a specific steam consumption.

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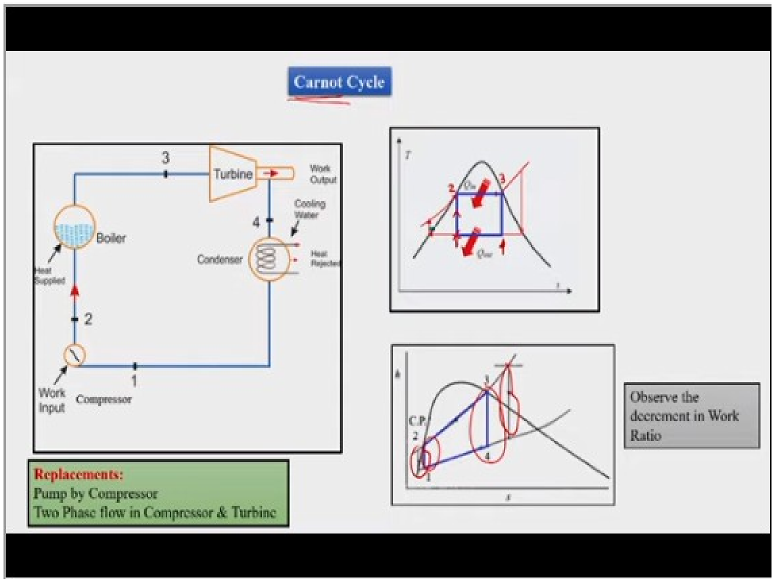

Carnot Cycle

Replacements:
Pump by Compressor
Two Phase flow in Compressor & Turbine

Observe the decrement in Work Ratio

Now we have seen what do we mean by Carnot cycle, we had already seen that there is Rankine cycle, and in Rankine cycle there are four processes. Now we are going to see that in the steam cycle which can as well consider Carnot cycle. Here the major thing to be noted is that pump here pump in the Rankine cycle is getting replaced by compressor. Since, as what we can see the steam which is at the entry to the steam generator is wet steam.

Since this steam is wet we cannot use pump to pump the liquid from 1 to 2 point. So, therefore since we cannot use pump or it is unlikely to use pump since we have to handle 2 phase mixture so process 2 to 3 will be governed in generally by compressor. So, we have compressor instead of pump then we have heat addition, then we have expansion and then we have condenser rest of the processes are as it is. This is a $h-s$ diagram for same Carnot cycle here we can observe a thing that the turbine work was here huge as compared to the pump work.

So, turbine work is large, pump work is very less in case of our Rankine cycle. But, when we are working with Carnot cycle as what we are saying here that we are working with Carnot cycle so since our objective is to study for Carnot cycle in this slide. In this slide we can say that we can see that this is the turbine work and this is the compressor work. So, the compressor work and turbine work they are having comparable amounts. But in case of Rankine cycle we have large turbine work a small compressor work or pump work.

So, this is the composition of Carnot cycle which again comprised of compressor, boiler, turbine and condenser. Process 1 to 2 is in the compressor, process 2 to 3 is in the boiler, process 3 to 4 is in the turbine, so this typical Carnot cycle.

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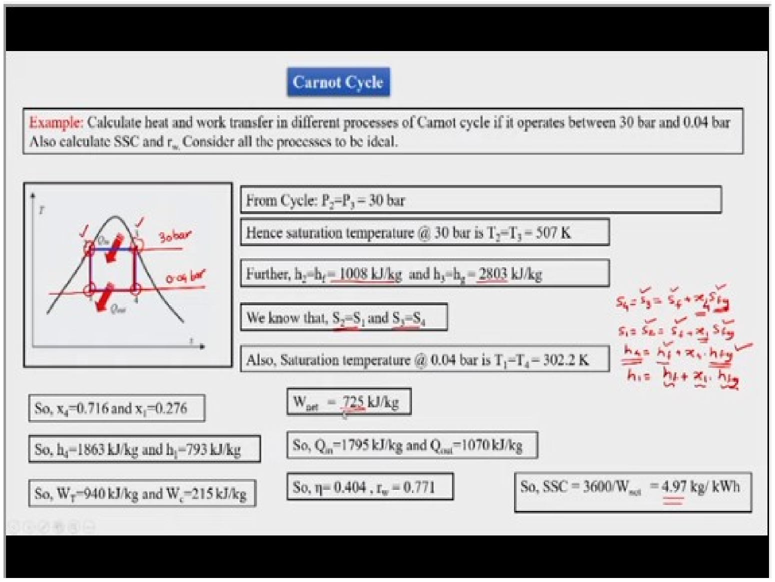

Carnot Cycle

Example: Calculate heat and work transfer in different processes of Carnot cycle if it operates between 30 bar and 0.04 bar Also calculate SSC and r_w. Consider all the processes to be ideal.

From Cycle: $P_2=P_3=30$ bar

Hence saturation temperature @ 30 bar is $T_2=T_3=507$ K

Further, $h_2=h_f=1008$ kJ/kg and $h_3=h_g=2803$ kJ/kg

We know that, $S_2=S_1$ and $S_3=S_4$

Also, Saturation temperature @ 0.04 bar is $T_1=T_4=302.2$ K

$s_4 = s_3 = s_f + x_4 s_{fg}$
$s_1 = s_2 = s_f + x_1 s_{fg}$
$h_4 = h_f + x_4 \cdot h_{fg}$
$h_1 = h_f + x_1 \cdot h_{fg}$

So, $x_4=0.716$ and $x_1=0.276$

So, $h_4=1863$ kJ/kg and $h_1=793$ kJ/kg

So, $W_T=940$ kJ/kg and $W_c=215$ kJ/kg

$W_{net} = 725$ kJ/kg

So, $Q_{in}=1795$ kJ/kg and $Q_{out}=1070$ kJ/kg

So, $\eta = 0.404$, $r_w = 0.771$

So, SSC = $3600/W_{net}$ = 4.97 kg/ kWh

Now, we will solve an example of Carnot cycle; here example states that calculate heat and work transfer in different processes of Carnot cycle if it operates between 30 bar and 0.04 bar pressures okay. Also calculate specific steam consumption and work ratio consider all the processes to be ideal. So, here what is the given to us we have said that the pressure of the boiler is 30 bar we have said the pressure of the condenser is 0.04 bar.

So, these two pressures are given to us so ultimately for us it is given that $P_2=P_3=30\bar{\iota}$ and then we can make use of steam table over here. Since we know that we are working with Carnot cycle and that's why Carnot cycle has four processes which are the processes as isentropic compression, constant temperature heat addition, constant isentropic expansion and constant temperature heat rejection.

These are the four processes of Carnot cycle and to execute those four process which are there in the Carnot cycle we cannot come out of the steam dome. We have to remain inside the dome to execute the processes which are comprising the Carnot cycle. So, here we have point 2 and this point 2 has to be on the saturation line similarly point 3 has to be on the saturation line; then only process 2 to 3 will become isothermal process.

Otherwise, as what we have seen in case of Rankine cycle, process 2 to 3 is constant pressure process but in which there is a chance that temperature would change that's why point 2 is known to us, point 3 is also known to us. So, point 2 and point 3 temperatures are found out from steam table as saturation temperature and that turns out to be corresponding to 30 bar it is 507 kelvin.

Then we can equally find out from the steam table what is the enthalpy at point 2 which is the liquid saturation enthalpy and that is for 30 bar it is 1008 kilo joule per kg. Similarly, we can find out what is the enthalpy at point 3 and it is 2803 which is a saturation enthalpy of steam. Knowing these enthalpies we actually need to find what is the enthalpy at 1 and what is the enthalpy at 4. We do not know those points.

But we know one thing that they are somewhere on this line and we know further that they are exactly the intersection with two vertical lines. So $S_2 = S_1$ is known to us as and as well know $S_3 = S_4$. Since process 1 to 2 and process 3 to 4 are isentropic processes. So, for that all sake what we can do is we can go to the steam table and find out what is entropy corresponding to liquid saturation point and what is the entropy corresponding steam saturation point dry saturation point of the steam.

So, once those entropies are known we will go back to the steam table corresponding to the pressure which is 0.04 bar and for which temperature is 302.2 kelvin. Here we get an answer that $x_4 = 0.716$and $x_1 = 0.276$; one thing we have to remember over here that we know now S_4 we want to find out**.** Basically, enthalpy at 4, for that we have to first find out dryness fraction at point 4 so for that we know $S_4 = S_3$.

But what is S_4*?* $S_4 = S_f$ which is liquid saturation enthalpy corresponding to 0.04 bar + $x S_{fg}$ which is the latent entropy change corresponding to 0.04 bar and this x_4 is the dryness fraction at point 4, so S_3 is known to us, S_f is known to us from the steam table, S_{fg} is known to us from the steam table so S_4 can be found out. Similarly, we can find out x_1 where we know $S_1 = S_2 = S_f + x_1 . S_{fg}$.

Here as well as S_{fg} is known, S_f is known, S_2 is known we can find out x_1 and these x_1 and x_4 are mentioned here as 0.716 and 0.276. So, $h_4$can be found out then; then we know $h_4 = h_f + x_4 h_{fg}$ here h_{fg} is given in the steam table, h_f is given in the steam table where $h_f$is the liquid saturation enthalpy corresponding to we can find out h_4 by writing $h_f + x_4 h_{fg}$ where h_f is the liquid saturation enthalpy corresponding to 0.04 bar condenser pressure and h_{fg} is the latent enthalpy or latent heat corresponding to 0.04 bar saturation pressure then we can know h_4.

Similarly, we can write $h_1 = h_f + x_1 h_{fg}$where as well h_{fg} and h_f are known; x_1 has been calculated so we know h_1. So, h_1 and h_4 are thus calculated. Knowing those we can find out turbine work we know turbine work is $h_3 - h_4$ and that is 940 kilo joule per kg. We know compressor work is $h_2 - h_1$ that is 215 kilo joule per kg then we can find out $W_{net}$and $W_{net} = W_T - W_c$ turbine work minus compressor work and it turns out to be 725 kilo joule per kg.

So, further we can find out $Q_{¿}$ we know that $Q_{¿} = h_3 - h_2$this is $Q_{¿}$ and we know h_3, we know h_2 and then we can find out $h_3 - h_2$and then that is 1795 kilo joule per kg. Similarly, we can find out Q_{out} where $Q_{out} = h_4 - h_1$, h4 is known to us, h1 is known to us so it turns out to be $Q_{out} = 1070\, kJ/kg$. Then we have efficiency and that efficiency is $\frac{W_{net}}{Q_{¿}}$and then $\frac{W_{net}}{Q_{¿}}$turns out to be 40.4 percent for the present Carnot cycle and work ratio is 0.771 this is the example for the Carnot cycle.

Here further we were expected to calculate specific steam consumption and as per the formula specific steam consumption it is $\frac{3600}{W_{net}}$. So it turns out to be 4.97 kilo joule per kilo watt hour. This much steam would be required to produce our net work of 725 kilo joule per kg. This is how we can make use of steam table to find out how to evaluate an example which is which would to be given to us and this was the illustration for finding out the performance parameters or work and heat interaction parameters for the Carnot cycle. We will see rest of the things in the next class. Thank you.