POWER PLANT SYSTEM ENGINEERING

Prof. Niranjan Sahoo
Department of Mechanical Engineering
Indian Institute of Technology, Guwahati
Module 3

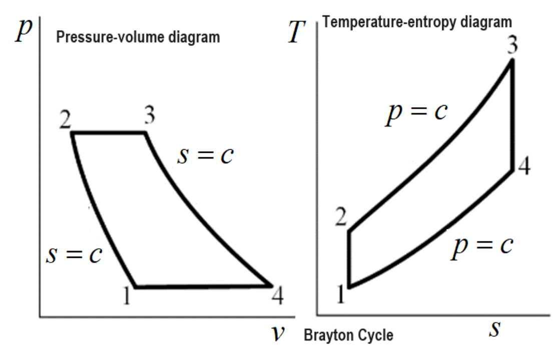
Lec 2: Modifications of Brayton cycle

Dear learners, greetings from IIT, Guwahati. We are in the MOOCs course Power Plant System Engineering module 3, Gas Turbines and Combined Power System. So, in this second lecture we are going to look at the Brayton cycles and moreover in the first lecture we have discussed about the thermodynamic analysis of Brayton cycles and how Brayton cycles becomes the very vital thermodynamic cycles, by which the gas turbines are being operated. And also it was emphasized in that lecture that we can extract the work output as well as thermal efficiency in a conventional Brayton cycle.

But then what happens certain modification needs to be required in the conventional Brayton cycles. These modifications are essentially or mainly due to the fact that many a times, we need to maximize the power output as well as we also expect the system or cycle to operate at its best possible efficiency. Also due to irreversibility, there are losses and they are also quantified in terms of efficiency of various components. Considering this, an optimum Brayton cycle needs to have certain additional requirement.

So, that is the reason we emphasize the non-ideal cycle. This non-ideal cycle is mainly associated, whenever there is friction and also when there is losses in the steady flow components of the gas turbine units and these are unavoidable. So, that is the reasons we need to analyze the non-ideal cycles. Now, the sources of non-idealness means, the cycle needs to be operated three through regeneration mode, there has to be intercooling which is required at the compressor end. We also expect that gases are expanded at much higher temperatures. So, that is the reason, we require turbine reheat. Another way of improvement of cycle efficiency and work output is through water injection. So, when all these components are introduced so, ideal Brayton cycle becomes a complicated device and for which each component does not behave in an ideal manner. So, main objective of this lecture is to look into the necessary modifications that are done in the Brayton cycles with a viewpoint of improving or maximizing power output and efficiency.

So, to start with the Brayton cycle normally consists of two reversible isentropic processes and two constant pressure processes as shown in this pressure volume and temperature entropy diagrams.



So, the cyclic processes that are process 1-2, in which the gas is compressed in an isentropic manner and process 2- 3 , which is constant pressure heat addition, process 3-4, an isentropic expansion in the turbine and process 4-1, a heat rejection process. So, here in the compressor we give work input W_C and in the turbine we get the work output W_T and Q_{in} also is added in the process 2-3, Q_{out} is heat rejected from process 4-1. In the T-s diagrams we also have isentropic compression 1-2, constant pressure heat addition 2-3, isentropic expansion in the turbine 3-4 and constant pressure heat rejection process 4-1. So, here we have net work output denoted by difference in the turbine work and compressor work and heat addition that is taken place from the fuel in the combustion chamber that is Q_{in} and Q_{out} is heat rejected in a heat exchanger.

So, this is the entire thermodynamic analysis we have made in the last class. Now, let us try to understand the non-idealness. So, the non-idealness is mainly associated with the fact that isentropic compression and expansion are relaxed in the compressor. So, that we can introduce the term, efficiency which is called as polytropic efficiency or adiabatic efficiency or isentropic efficiency for turbines as well as compressors. We also introduce the pressure drop which is accounted in, during heat addition or heat rejection processes.

In a typical non ideal cycle or in a workable mode what we expect is that isentropic efficiency is in the range of 85 -90% where, the other non-ideality due to friction that can happen in components like heat exchanger, combustion chamber, gas inlet and outlet, regeneration and mechanical losses. Now, let's understand the effect of this non idealness for example, in the polytrophic efficiency. If you look at the pressure volume or temperature entropy diagram so, had the process been isentropic, the compression would have been 1-2s, but due to this non-idealness, we are binding to give higher work input. Similarly, in the turbine process, if the process had been isentropic then it would have reached 3 - 4 s, but due to non-idealness, it is reaching 4. So, this cycle which is shown as 1-2s-3-4s-1, it is a complete closed cycle and it does not account any kind of non-idealness whereas, actual cycle becomes 1- 2- 3- 4.

So, our main intention is that by introducing these terms, we need to quantify the net work and cycle efficiency. So, to start the thermodynamic analysis by introducing the term turbine efficiency η_T and compression efficiency η_C, we can write down the net work rate for cycle in which actually the term turbine work is reduced by efficiency that is $\eta_T \dot{W}_T$ & the compressor work is also increased by compression efficiency that is to ${}^{\dot{W}_C}/_{\eta_C}$. Thereby the net work output also is decreased. Then even for thermal efficiency we can do the calculation by calculating the net work and the heat added in the cycle. So, with these all these numbers we can actually find out the thermal efficiency of the cycles. Net work rate for cycle, $\dot{W}_n = \eta_T \dot{W}_T - \frac{\dot{W}_C}{\eta_C}$; Thermal efficiency of the cycle, $\eta_{th} = \frac{\dot{W}_n}{\dot{Q}_A}$

$$\Rightarrow \dot{W}_n = \dot{m} c_p \left[(T_3 - T_{4s})\eta_T - \frac{T_{2s} - T_1}{\eta_C} \right]$$

$$= \dot{m} c_p T_1 \left[\left(\eta_T \frac{T_3}{T_1} - \frac{r_p^{\left(\frac{k-1}{k}\right)} - 1}{\eta_C} \right) \left(1 - \frac{1}{r_p^{\left(\frac{k-1}{k}\right)}} \right) \right]$$

Heat added to the cycle, $\dot{Q}_A = \dot{m} c_p (T_3 - T_2) = \dot{m} c_p \left[(T_3 - T_1) - \left(T_1 \frac{r_p^{\left(\frac{k-1}{k}\right)} - 1}{\eta_C} \right) \right]$

Now, how do you define this polytrophic efficiency of or isentropic efficiency for compressor and turbine? We mentioned that it is nothing, but the isentropic enthalpy drop to the actual drop that is for compressor, but for turbine actual drop to the isentropic enthalpy drop. Since we write this gas as an ideal gas we introduce $h = C_p T$ as enthalpy is the function of temperature. Normally when we deal with similar kind of efficiencies for steam turbines, we used to get these values directly from the steam table. But since in this gas turbine cycle, the working fluid is purely gas, so we take ideal gas assumption and accordingly enthalpy is represented as $h = C_p T$. So, compressor and turbine efficiencies are introduced in terms of temperature differences.

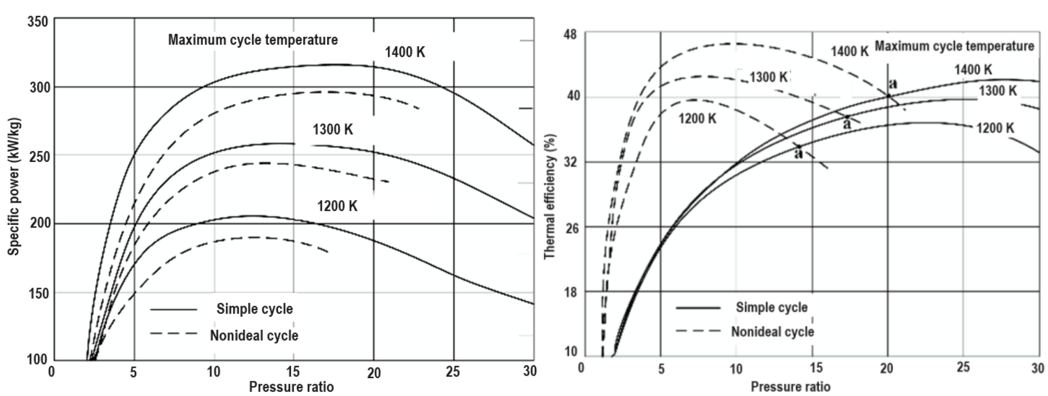
$$\text{Polytropic efficiency(compressor)}, \eta_C = \frac{h_{2s} - h_1}{h_2 - h_1} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$\text{Polytropic efficiency(turbine)}, \eta_T = \frac{h_3 - h_4}{h_3 - h_{4s}} = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

So, after having analyzed all these things, what we can expect from a non-ideal cycle is that both cycle efficiency and specific work strongly dependent on optimum compression ratio and maximum cycle temperature. So, in a typical gas turbine plant, which is normally a single shaft direct cycle open air combustions with 16 stage compression and 3 stage turbine and operate with a diesel fuel, can produce 35 MW of power with a cycle efficiency of 30%. So, this system can operate at 500 rpm with an overall dimension of $(38 \times 11 \times 8)$m. So, basically this particular number essentially says that what kind of range or power requirement that a gas turbine cycle can achieve. Now, when I specify all these numbers, we expect that our main intention is to calculate the power that is derived from the cycle and the cycle efficiency. And to achieve this, what is the maximum temperature that we are going to handle is seen at the turbine inlet point. So, with these three vital parameters we are going to look at the relations among the specific power & pressure ratio and thermal efficiency & pressure ratio. And in this graph we have shown a simple cycle which is a conventional ideal Brayton cycle and a non-ideal cycle.

So, we have also emphasized the working expressions for work network and cycle efficiency and if you put some realistic number and try to calculate we can plot them. So, trend of these plots are shown in this figure. We can observe that with increase in the turbine inlet temperature obviously, your specific work output also goes out and of course, cycle efficiency also goes out. So, we cannot keep on increasing the turbine inlet temperatures, there is a limitation in terms of fuel and also there is a limitation in terms of materials for the turbine blade which are going to be implemented. So, with that restrictions the upper limit normally does not go beyond 1500 K. So, the analysis was normally done at the three temperatures typically between1200 to 1400 and we are trying to plot them.

So, once that temperature is fixed, corresponding numbers also gets fixed for example, maximum specific power and also the cycle efficiency get fixed. But one interesting observation is that for an ideal cycle, these curves are continuous and only dependent on the pressure ratio, but for non-ideal cycles various terms or efficiency terms comes into pictures.

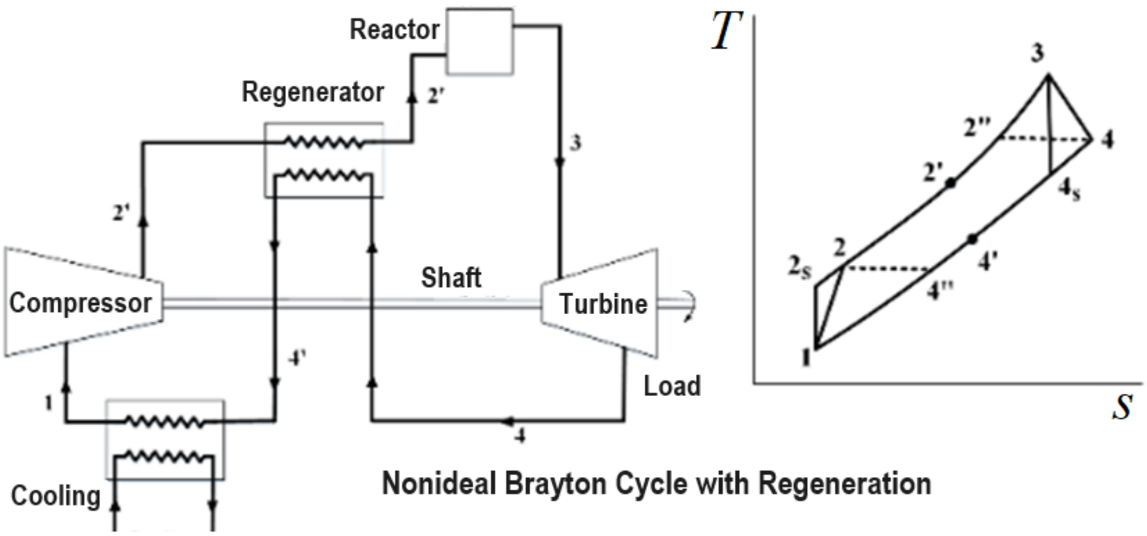


So, due to these reasons, the continuous or monotonous increase of the thermal efficiency or work output is not seen. Beyond some point or beyond certain pressure ratio the non-idealness effect come into picture. And at certain pressure ratio, non ideal cycle efficiency suddenly drops, that you can see in the graph. But one interesting thing, you can see is that for the non ideal cycles, there is a sharp rise. For example, at 1400 K, the thermal efficiency shoots up immediately then retains it for a while and then drops at 2200 K. It suddenly drops that means, there is no range, and it suddenly drops down. So, because of these reasons, it is always advisable to go for very high temperature and side by side we also look at corresponding specific power or work. So, this is one important observation. The second observation is that although it rises sharply the non-ideal cycles are not operated at a very high pressure ratio. That means, with increase in the pressure ratio, after one particular point the efficiency suddenly and also work output suddenly drops.

So, we need to find out for maximum work what is the optimum pressure ratio. And optimum pressure ratio shifts or gets reduced when we are increasing the cycle temperature. And that is the reason the optimum pressure ratio shifts towards the left. That means, at higher pressure ratio, we cannot operate non-ideal cycles for a longer time or in other words we need to find out what is the range of optimum pressure at which the cycle needs to be operated for generating maximum work.

So, this is the essential motive of this non ideal cycles and we also have to see that non ideal cycles have a better approach in terms of improving the cycle efficiency and work output. So the first category of non-idealness we see in terms of regeneration. Basically there are four modifications that are done for a conventional Brayton cycles- first is regeneration, second is compressor intercooling, third is turbine reheat and fourth is water injection. Out of these four we mainly concentrate on regeneration, compressor intercooling and turbine reheat to the most possible extent. There is some mechanism in which water injection is also used for improving the work output and that is mainly done for aircraft engines.

So, what does it mean by regeneration? So, regeneration in terms of steam cycle is viewed as the internal heat exchange within the cycle. But for this conventional or classical Brayton cycle, the turbine exhaust temperature is often higher than the compressor outlet temperature.



Nonideal Brayton Cycle with Regeneration

So, that means, we have not utilized the efficiency of the fuel in a complete manner for which the exhaust temperature seems to be much higher than the compressor outlet temperature. So, how effectively we can manage it? So, to do that what we normally do is that instead of releasing the turbine exhaust to the atmosphere, we use a regenerator. The purpose of this regenerator is that, if you look at this particular cycle there are two fluids. One is high pressure compressor air enters to the regenerator, but that is not at very high temperature. But what is done is that, at state 4, the exhaust from the turbine is getting utilized. In this cycle we have this combustion products and in the other one we have air. So, air is getting heated in the regenerator. Here in the regenerator, the heat exchange takes place from the combustion products to the air. So, it is a kind of preheating of air before it enters to the reactor or combustion chamber. So, in that way what we have shown is that, we are pushing this outlet temperature by taking the heat from this combustion products, from 2 to 2' or 2' and we are also reducing the exhaust temperature which was supposed to be at T_4, it has now come back to $T_4{'}$ or $T_4{''}$. So, that way point 4 shifts to the down point 2 shifts towards the up and a particular balance is achieved by which, heat exchange can be possible to maximum extent. So, in this particular graph we can see the maximum possible temperature change and that is nothing, but $T_4 - T_2$, which is the difference between turbine exit temperature and the compressor outlet temperature. So, that is the maximum possible temperature change, but in the regenerator the temperature due to this heat exchange, what is the best possible way that heat exchange takes place we define the term what is called as effectiveness.

Regenerator effectiveness, $\varepsilon = \dfrac{T_{2'} - T_2}{T_4 - T_2}$;

$(T_4 - T_2)$:Maximum possible temperature change

$(T_{2'} - T_2)$:Actual rise in temperature in the regenerator

Regenerator effectiveness is nothing, ratio of the actual temperature rise to the maximum possible temperature rise and that way we call this as a regenerator efficiency. In a typical gas turbine cycle, we expect that regenerator should operate with 75% effectiveness. So, by using this regenerator, if you correlate this temperature entropy diagram with the thermal efficiency of the cycle then we can see that there is some locations at point ‘a’ for each particular temperature. For example, if you look at this particular graph thermal efficiency versus pressure graph, initially your thermal efficiency shoots up and the efficiency becomes maximum at this point, which means that we are expecting the maximum possible work from the turbine through this process. And if you include a regenerator in this circuit then possibility is that heat exchange can take place at point ‘a’. So, point ‘a’ is nothing, but the intersection point for a simple cycle and non-ideal cycle. So, beyond this point ‘a’ it is expected that the gas from the compressor outlet will no longer be heated because we are crossing the maximum possible temperature rise. So, beyond point ‘a’ it is not possible to operate the regenerator.

So, through this process what extra advantage we get is that for similar output we can go for higher pressure ratio for operating the cycles. So, that is the extra advantage we get through this regeneration process. So, it has been emphasized here that when the efficiency curve of a regenerator cycle cross the simple cycle the effect of regeneration on efficiency is negative. So, these points indicates the pressure ratio at which exhaust gas from the turbine are cooler than that of compression. So, another effect is that although the regenerator has role in improving the cycle efficiency, but it does not have much effect on the specific power. Hence the regenerative cycles are more efficient than simple cycles because there will be a reduction in the fuel consumptions by 3 %. That means, when air is preheated there is a reduction in the fuel consumption and it can go up to 30%.

So, through regeneration we get improvement in the cycle efficiency without loss of any power. This is the first achievement, & second achievement is that we can go for reduction in the fuel consumption. So, these two points are very vital, because of these, we go for a non-ideal cycle that can operate in regeneration mode.

Now, regeneration does not improve the work output. So, what is the mechanism that work output can be increased? So, if you look at your expression, net work is the difference between the turbine work and compressor work. Now, for maximum or more net work output there are two possibilities, either you increase turbine work or reduce the compressor work. So, through these process your net work can be increased.

So, the first type of change that we are looking at is to reduce the compressor work. So, to achieve this we introduce the term what is called as intercooling. Now, how do you do that? So, for that if you look at a conventional flow system like a piston- cylinder mechanism the flow work can be expressed as

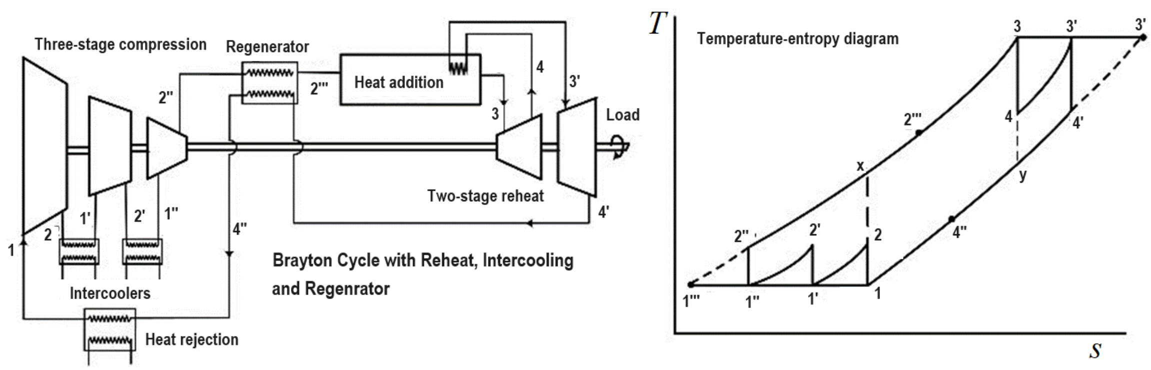
Work in a flow system, $W = -\int_1^2 V dp = -\int_1^2 mRT \frac{dp}{p}$

Here V stands for compression that means, work is input and we can rewrite this workflow system in this manner.

Work for one compressor, $W = m\frac{nR(T_2 - T_1)}{1-n}$; Ideal gas: $pV = mRT$

So, one important point that you need to observe here that, work in the flow systems can also be minimized if your temperature is minimized. So, typically for a constant quantity, if you do the compression at lower temperature, then it is possible that we can reduce the compression work. So, technologically that is the essential requirement, we achieve through this intercooling processes.

So, what we normally do? We do this kind of intercooling in stages. In this particular figure we can see the Brayton cycle which is integrated with intercooling. So, intercooling is normally done at compressor stage. So, instead of single stage compressions, we have introduced three stage compression A, B, & C and in between we have intercoolers. So, basically there are two intercoolers. Now, through this process the changes we made here is that instead of going for compression process from 1 to x we are doing that compression process in stages like going from 1 to 2, then bringing back again from 2 to the initial temperature which is at 1 that is 1' then do this compression again to 2' then come back to 1''.



So, through this process we are doing this compression at lower temperature. And you can imagine this process to be in infinite number of stages, which is also not possible in a realistic way. But here three stage/ four stage are conventionally achieved. So, through this process we are actually reducing the compression work and the cycle approaches to an Ericsson cycle. Like there are these two constant pressure lines, let us say p_2 & p_1. So, to reach p_2, instead of going through increase in the temperature, we are actually going in a constant temperature line to the best possible extent, so the process tries to be more towards the isothermal process. So, that way we are getting much benefit.

So, how much reduction in compression work we can achieve? This we can find out for each stage from the enthalpy difference. And it can be noted that a single stage compression work is always higher than the three stage compression if done at a lower temperature.

Reduction of compression work: $(H_2 - H_1) + (H_{2'} - H_{1'}) + (H_{2''} - H_{1''}) < (H_x - H_1)$

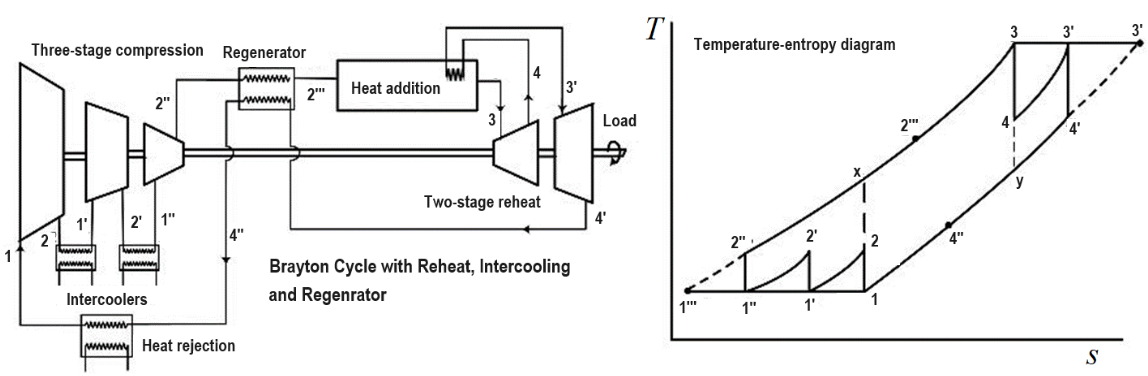
Another important point to be emphasized that in between we have intercooler, so of course, we are increasing the pressure, but not at the cost of temperature. So, an expression can be found out for the pressure ratio per stage. So, if the overall pressure ratio is r_p that means, that much pressure ratio we want to achieve through this compression process then pressure ratio per stage is calculated as stated below.

Overall pressure ratio, $r_p = \left(\frac{T_2}{T_1}\right)^{\frac{n}{n-1}}$; n:Polytropic compression index

Pressure ratio per stage, $r_{p,s} = \sqrt[N_c]{r_p}$; N_c:Number of compressor section

So, that is the idea for how to achieve pressure ratio per stage. So, the improvement in the cycle is the increased work and efficiency. The increased cyclic work is mainly the reduction of the compression work because the operation is done at lower temperature. And through this process of course, heat addition is increased, but this increase in the cycle work offsets the heat addition and that results in the improvement in the efficiency. So, this is the summary what we expect after the compression intercooling.

And similar concept we can introduce in the turbine side and we call this as turbine reheat. As I mentioned earlier, to improve the net work, you need to reduce the compression work through intercooling that we have seen or we can also increase the turbine work. So, that means, instead of expansion in the turbine in a single stage from 3 to y, we come back in stages 3 to 4 then reheat 4 to 3', again come back to 4' and keep on reheating to the 3'. So, this reheating process that means, we are coming back from higher pressure p_2 to p_1 through isothermal manner in this reheating process. And reheating normally is done for the maximum temperature in the turbine.



Brayton Cycle with Reheat, Intercooling and Regenrator

So, this particular things says that we have a two stage reheat. So, after first stage of expansion again it is feed to the heat addition systems reheated again come back to for the second stage turbine. So, that way we can increase both work and efficiency. So, reheat is normally done to increase both work as well as efficiency. So, the achievement of these two intercooling and reheat is that, when cooling and heating are done at constant temperature with rest of the cycle ideal in nature, it would represent an ideal Ericsson cycle. And this Ericsson cycle has efficiency same as that of Carnot cycle operating between the same temperature limits. So, that is the advantage of reheating and intercooling.

And at the end of this thing reheating and intercooling what we expect is that if you include reheat, intercooling in number of stages, then the general expression of non-idealness with all non-idealities can be found out, that is for the specific power and heat addition. And through these expressions, we can model the fact that net work can be improved and cycle efficiency can be increased. So greater the number of reheat and intercooling stages, higher is the efficiency, but at the cost of capital investment and size of the plant.

Specific power, $\frac{\dot{W}_n}{\dot{m}c_p} = T_3\eta_T(n_T+1)\left(1-\frac{1}{r_{pT}^{\left(\frac{k-1}{k}\right)}}\right) - T_1\left(\frac{n_C+1}{\eta_C}\right)\left(r_{pc}^{\left(\frac{k-1}{k}\right)}-1\right)$

Heat addition, $\frac{\dot{Q}_A}{\dot{m}c_p}$

$$= T_3\left\{(n_T+1)-(n_T+\varepsilon_R)\left[1-\eta_C\left(1-\frac{1}{r_{pc}^{\left(\frac{k-1}{k}\right)}}\right)\right]\right\} - T_1(1-\varepsilon_R)\left[1+\frac{1}{\eta_C}\left(r_{pc}^{\left(\frac{k-1}{k}\right)}-1\right)\right]$$

Net power, $\dot{W}_n = \dot{W}_T - \dot{W}_C$; n_T&n_C:Number of reheat & inter cooling stages

η_C&η_T:Adiabatic efficiency for compressor & turbine; ε_R:Regenerator effectiveness

r_{pT}&r_{pC}:Overall pressure ratios for turbine & compressor

There is another method where the non-idealness also comes into picture. Normally we introduce the term that is water injection and this method is normally used for gas turbine cycle in aircraft engines and it is mainly intended for thrust generation. And just to quantify the fact that, normally in gas turbine, blades are always exposed to very high temperature in a continuous manner, so there is a thermal endurance. And to reduce this to some extent, one aspect is the turbine material has to be chosen, but even if the material is also chosen, while implementing we can use this concept of water injection to cool this turbine materials. So, the thereby we can partially improve this net work as well as we can increase the life of the turbine blades. So, water injection is a method by which power output can be increased with marginal improvement in the cycle efficiency. So, the increased work of the cycle is mainly due to the optimum pressure ratio after the injection. Now, what do we do in this? The heat of vaporization reduces the compressed air temperature and this produces the similar effect as that of intercooling. So, for gas turbine cycles, it is more beneficial if water is injected between compressor and regenerator. So, this particular cycle says that through water injection how the regenerator cycle gets modified.

So, this is all about the non-ideal cycles and their role in the power generation processes. So, with our this basic understanding let me solve a numerical problem that completely highlights the non-idealness behavior for a gas turbine cycle which involves reheat intercooling and regeneration.

Q1. A gas turbine plant consists of two-stage compressor with intercooling and single stage turbine with regenerator. Air enters the compressor at 1 bar & 20°C. The maximum cycle temperature is 900°C and pressure ratio is 6. The rate of air flow through the plant is 210 kg/s and the fuel calorific value is 40.8 MJ/kg with combustion efficiency 0.95. The isentropic efficiencies for compressor and turbine are 0.82 & 0.92, respectively. Take, mechanical and generator efficiencies as 96%. Use, c_p = 1.005 kJ/kg.K and k = 1.4 for air and c_p = 1.08 kJ/kg.K and k = 1.33 for gaseous combustion products. Estimate, (a) air-fuel ratio; (b) cycle efficiency; (c) power delivered by the plant; (d) Specific fuel consumption and fuel consumption per hour.

So, there is a big problem statement which I can simplify. It is a two-stage compression process, single stage turbine process. And two-stage compression is achieved through intercooling. We also have the maximum cycle temperature is 900℃ and pressure ratio is given. We also include a combustion chamber, in which fuel is injected with a given calorific value and combustion efficiency. To introduce the non-idealness we will define the term isentropic efficiency for turbines and compressor. Also we have regenerator which is introduced in the cycle and this has efficiency of 0.75. I think that data is missing.

So, to solve this problem first thing that we need to draw is its thermal circuit. So, we say that we have a compressor C_1 and a second compressor C_2. It is coupled with turbine which is a single stage and after this compressor we have a regenerator or heat exchanger. That means, exhaust from the turbine is getting utilized in the heat exchanger and this is nothing, but our regenerator. So, air comes from the atmosphere, turbine exhaust goes out and side by side it produces power when it is coupled with generator. So, in between the first stage and second stage compression there is an intercooler.

So, we have this compressor, two stage turbine, generator and this heat exchanger and we also have here combustion chamber where fuel is added. And we also have an intercooler. So, let me put the notations, state 1. The compressed air enters to the intercooler. So, I will say state i and after intercooling is done it enters at same pressure to the second stage compression. After the compression it is your state 2, then we have state 3, state 4, then we have state 5, state 6 turbine outlet goes at state 7 and finally, exhaust goes to atmosphere at state 8 and the thermodynamic cycle in T-s diagram should look like this.

So, minimum temperature is T_1, maximum temperature is T_6. So, in between we have three pressure lines mainly. So, from state 1-2s and actual process is 2, then from 3-4s, but actual process is 3-4. Then stage 5 is in the heat exchanger and finally, after this combustion chamber this is 6. So, 6- 7 is your expansion. So, this is T_6 maximum temperature. And non-isentropic expansion will lead you 7s and 7 and somewhere we will have 8 and come back to the state 1. So, with this notations we have two stage intercooling and then single stage reheat. So, with the data we have to go step by step first we start with compressor.

Compressor; $p_1 = 1 \text{ bar}, p_2 = 20℃ = 293 \text{ K}$

So, we can find out what is intermediate pressure.

$p_i = \sqrt{p_1 p_2} = 2.45$ bar

Then for first stage compression;

$$\text{For air}, k = 1.4; c_p = 1.005\,\frac{\text{kJ}}{\text{kg. K}}$$

$\frac{T_{2s} - T_1}{T_2 - T_1} = \eta_c = 0.82$; We also have isentropic relation, $\left(\frac{T_{2s}}{T_1}\right) = \left(\frac{p_i}{p_1}\right)^{\frac{k-1}{k}} \Rightarrow T_{2s} = 378.6\text{K}$

Now , $\frac{T_{2s} - T_1}{T_2 - T_1} = 0.82 \Rightarrow T_2 = 397.4$ K

In similar manner we can also find out the second stage compression.

$\left(\frac{T_{4s}}{T_3}\right) = \left(\frac{p_2}{p_i}\right)^{\frac{k-1}{k}} = \left(\frac{6}{2.45}\right)^{\frac{0.4}{1.4}} \Rightarrow T_{4s} = 378.5\text{K}; \quad T_3 = T_1 = 293$ K

Then, $\frac{T_{4s} - T_3}{T_4 - T_3} = \eta_c = 0.82 \Rightarrow T_4 = 397.3$ K

Then in similar way we do this turbine analysis. So, peak temperature that we achieve in the turbine is T_6.

$\left(\frac{T_6}{T_{7s}}\right) = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}$; Given data, $\frac{p_2}{p_1} = 6$

Here I need to emphasize for turbines we use combustion products. So, it is a gaseous combustion products and for which appropriate value of $k = 1.33$.

So, this the basic difference when you use this value in the expression and we also have from this data $T_6 = 900℃ = 1173$ K. O

$$\left(\frac{T_6}{T_{7s}}\right) = \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}} \Rightarrow T_{7s} = 752.2 \text{ K}$$

Now, Turbine Efficiency, $\eta_T = \dfrac{T_6 - T_7}{T_6 - T_{7s}} = 0.92 \Rightarrow T_7 = 785.8 \text{ K}$

Then we move on to regenerator. So, the data is missing that regenerator efficiency $\epsilon =$ 0.7.

For Regenerator, $\epsilon = \dfrac{T_5 - T_4}{T_7 - T_4} = 0.7 \Rightarrow T_5 = 669.3 \text{ K}$

Then combustion chamber, in which fuel enters gas and we get this heat.

Data given, Calorific value, $CV = 40800 \dfrac{\text{kJ}}{\text{kg}}$; Combustion efficiency, $\eta_{comb} = 95\%$

So, for combustion chamber we write this expression like below.

$$(\dot{m}_f)(CV) \times \eta_{comb} = (\dot{m}_a + \dot{m}_f) c_p (T_6 - T_5)$$

$$\Rightarrow \frac{\dot{m}_a}{\dot{m}_f} = 75.56 \text{ or we can express as } \frac{1}{\dot{m}_f} = {}^{75.56}/_{\text{kg air}}$$

Then turbine work, $\left(\dfrac{\dot{W}_T}{\dot{m}_a}\right) = \left(1 + \dfrac{\dot{m}_f}{\dot{m}_a}\right) c_{pg} (T_6 - T_7)$;

Here for gaseous products, $c_{pg} = 1.08 \dfrac{\text{kJ}}{\text{kg. K}}$

$$\Rightarrow \left(\frac{\dot{W}_T}{\dot{m}_a}\right) = 423.7 \ {}^{\text{kJ}}/_{\text{kg air}}$$

Then compressor work, $\left(\dfrac{\dot{W}_C}{\dot{m}_a}\right) = c_p[(T_2 - T_1) + (T_4 - T_3)]$;

As compressor handles only air, $c_p = 1.005 \dfrac{\text{kJ}}{\text{kg. K}}$

$$\Rightarrow \left(\frac{\dot{W}_C}{\dot{m}_a}\right) = 209.7 \ {}^{\text{kJ}}/_{\text{kg air}}$$

$$\text{Then, Net work, } \dot{W}_n = \frac{\dot{W}_T - \dot{W}_C}{\dot{m}_a} = 214 \ {}^{\text{kJ}}/_{\text{kg air}}$$

We have all these things, but we have to find power delivered by the plant; (d) Specific fuel consumption and fuel consumption per hour. So, for that the data we have is flow rate of air 210 kg/s. So, this net work, we obtain for per kg of air.

$$\text{So what we know, mass flow rate of air, } \dot{m}_a = 210 \text{ kg/s}$$

Now, we can find out what is power output.

$$P = \dot{W}_n \times \dot{m}_a \times \text{mechanical efficiency} \times \text{generator efficiency}$$
$$= (214 \times 210) \times 0.96 \times 0.96$$

$$\Rightarrow P = 40972 \text{ kW} \approx 41 \text{ MW}$$

$$\text{Fuel consumption per hour} = 210 \times 3600 \times \frac{1}{75.56} = 10005.3 \text{ kg/hr}$$

$$\text{Where, Air fuel ratio} = \frac{1}{75.56}$$

Then specific fuel consumption we say it is SFC.

$$\text{SFC} = \frac{\dot{m}_f}{P} = \frac{1005.3}{40972} = 0.245 \ {}^{\text{kg}}/_{\text{kWH}}$$

$$\text{Then Cycle efficiency} = \eta = \frac{W_{net}}{Q_{in}}$$

$$W_{net} = \dot{W}_n = 214 \ {}^{\text{kJ}}/_{\text{kg air}}; \quad Q_{in} = \frac{(CV) \times \eta_{comb}}{\dot{m}_f} = \frac{40800 \times 0.95}{75.56} = 513 \ {}^{\text{kg}}/_{\text{kg air}}$$

$$\text{Cycle efficiency} = \eta = \frac{W_{net}}{Q_{in}} = \frac{214}{513} = 0.417 \approx 42\%$$

So, for a non-ideal cycle by introducing all these components, the efficiency turns out to be 42%. So, typically conventional gas turbine cycle or ideal cycle has efficiency in the range of 30 to 35%, but through this component introduction the efficiency was improved to 42%.

So, this is how I emphasize the need of non-idealness in the gas turbine cycles by introducing the concept of regenerator, intercooling and turbine reheat. So, with this I conclude. Thank you for your attention.