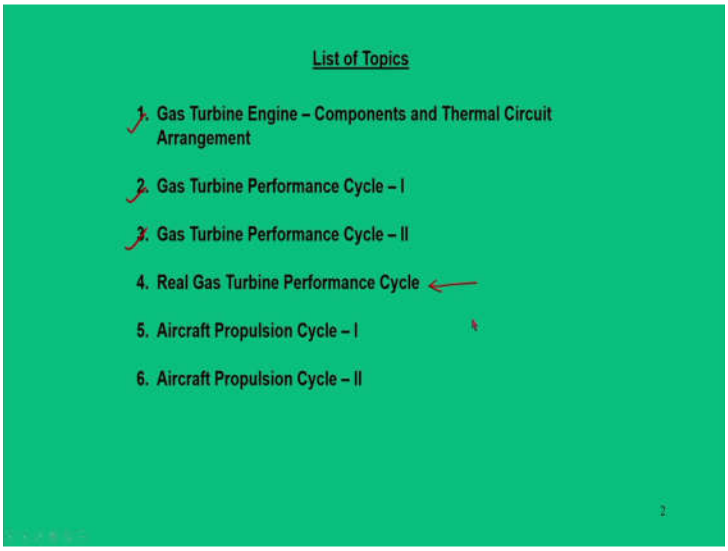
Applied Thermodynamics
Prof. Niranjan Sahoo
Department of Mechanical Engineering
Indian Institute of Technology, Guwahati

Module - IV
Gas Turbine Engines
Lecture - 34
Real Gas Turbine Performance Cycle

Dear learners, greetings from IIT, Guwahati. We are in the MOOCs course Applied Thermodynamics, module 4 Gas Turbine Engines.

(Refer Slide Time: 00:57)



So, previously on this module, we have discussed 3 lectures that is gas turbine engines components thermal circuit, gas turbine performance part 1, gas turbine performance part 2 and just to emphasize the fact that when we are talking about this gas turbine performance, we mainly concentrated on power generation through gas turbine engines. But, apart from that the other application for this gas turbine engines is in the air craft propulsion.

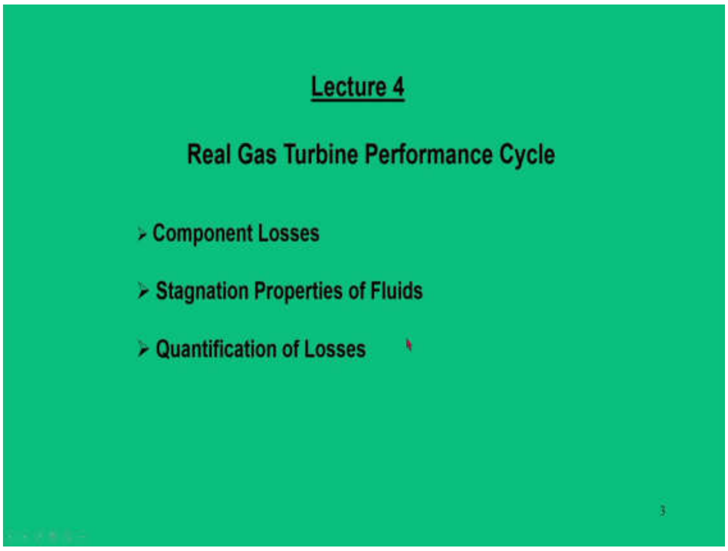
So, we will move to aircraft propulsion cycle in the subsequent class, but before you do that we must understand some other concepts that how the real gas turbine performance cycle differs from the original cycle. A original cycle I mean that the very basic cycle for gas turbine engines is the Brayton cycles and on top of that we introduced the modified cycles in terms of inter cooling, reheating, reheat and inter cooling with a combined performance.

But when you move to aircraft propulsion cycles, our main intention is thrust generations not power. Now, in the process of thrust generations we expect the working fluid has to circulate within the circuit. And those circuits involved the components like compressors, turbines, combustion chamber and these compressor and turbines are called as turbo machines.

And when they circulate they have substantial velocities while crossing those components. One very good assumption could be that the energy content due to the kinetic energies of the fluid can be neglected as compared to turbine work or compression work, but when you do propulsion cycle this velocity has a significant value one cannot ignore.

So, this is the very basic difference that when a gas turbine engines runs in a thrust mode where the velocity components are significant in terms of total energy. So, in our particular topic today that is real gas turbine performance cycle which will be discussing we will debate on those aspects that how a real gas turbine cycle is different from its air standard cycle.

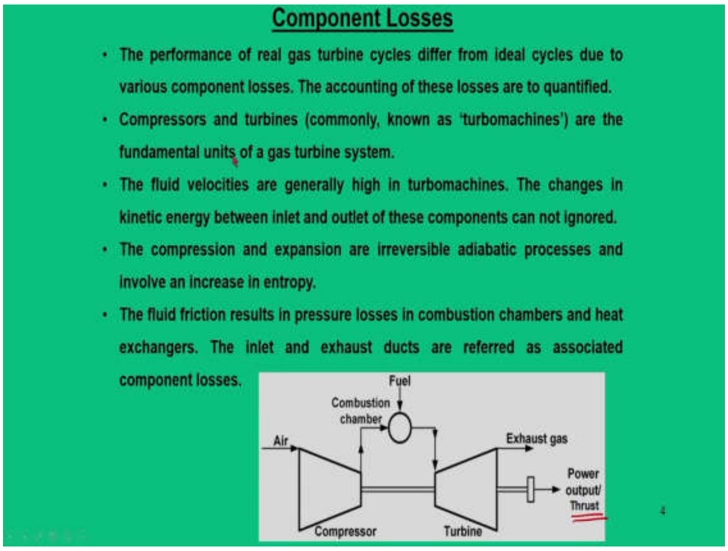
(Refer Slide Time: 03:57)



So, on this lecture the following topics will be covered. So, we will introduce the component losses and since mainly you will be focusing is in subsequent class on aircraft propulsions where we will be looking at some properties which we call as stagnation properties of the fluids.

So, these things we will discuss that how the stagnation properties of the fluid will help us in quantifying the losses at different components. So, these are the 3 significant headings of today's discussion.

(Refer Slide Time: 04:38)



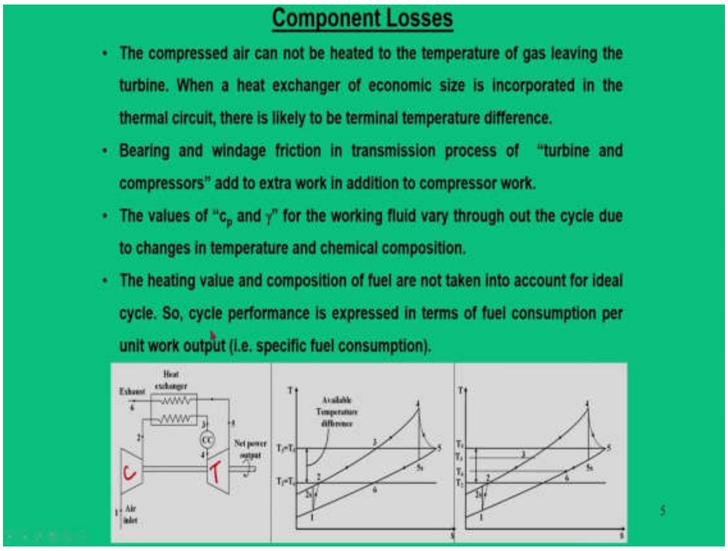
So, let me start. First thing is that component losses. So, the very basic cycle that gas turbine has compressor, turbine joined together with a common shaft in a sense that turbine drives the compressor and fuel gets added and out of this after expansion we get power output or thrust. So, here your main intention is thrust.

So, this is what the simple gas turbine operate, but when you are looking at the component losses what are the different sources of losses. So, let me start one by one. The performance of real gas turbine cycles differs from the ideal cycles due to various component losses.

And accounting of these losses needs to be quantified and since the compressor and turbines are commonly known as turbo machines and they are fundamental units of gas turbine system. And as I mentioned when you deal with mainly propulsion system the fluid velocities are generally very high when they pass through compressor and turbine.

So, because of this the changes in the kinetic energy between inlet and outlet of these components cannot be ignored. So, the compression and expansion process in this compressor and turbines they are mainly irreversible adiabatic and of course, they involve increase in the entropy. And of course, the fluid friction results the pressure losses in the combustion chamber and heat exchanger. In fact, when heat exchanger is incorporated it involves pressure losses.

(Refer Slide Time: 06:21)



Another issue is that the entire idea of this heat exchanger is that we have to preheat the compressed air coming out from the compressor to a upper limit which the turbine exhaust gas has. So, in a sense that looking at the T-S diagram, the available temperature difference is T5 - T6, but that is the maximum available temperature difference, but unfortunately due to pressure losses in the heat exchanger we are not in a position to capture this total heat content with this available temperatures.

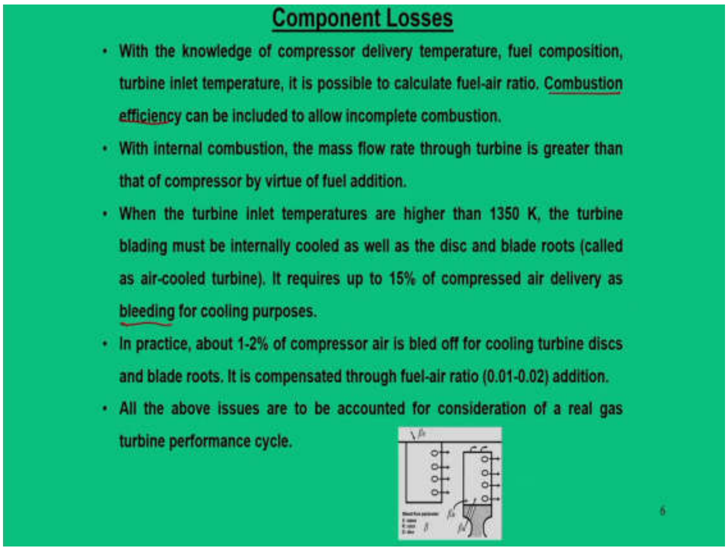
So, in the sense that the compressed air cannot be reached to T5 temperature rather it terminates at T3 whereas, turbine exhaust temperature cannot reach to T2 rather it remains at T6. So, this is what the heat exchanger effectiveness comes into picture. So, there is likely to a terminal temperature difference. Apart from that there is bearing and windage friction in the transmission process as you see the turbine drives the compressor.

So, it means that during the transmission of power from turbine to compressor there is other type of losses that is bearing and windage, this needs to be accounted and apart from that there are values of cp and gamma. So, we in last class also we mentioned that the cp and gamma, when you take the hot gases in the turbine its value is different because they involve chemical compositions and high temperature changes.

So, because of this reason we have to take the variation of cp and gamma. And of course, in some sense cp and gamma also varies with temperatures and in the entire gas turbine units, the temperature of the cycle routinely changes. Another issue could be that heating value and composition of the fuel are not taken into account in the ideal cycles.

There what we us simply said that we just added heat in terms of fuel, but fuel has no role except it just adds heat to the system, but when you actually take the quantification of how much fuel is added then we should be worried about specific fuel consumptions that means, how much fuel you require per unit work output. So, all these things is to be accounted.

(Refer Slide Time: 09:25)



Apart from that we also to quantify this specific fuel consumptions, we also have to assume that there is some combustion efficiency that means, fuel has certain energy which we called it as a calorific value. Entire calorific value of the fuel is not recovered. Almost 98 or 99 percent of the fuel gets combusted. So, by assuming a combustion efficiency we allow incomplete combustion. So, this is also a part of component losses.

And another issue could be that when during this combustion process we add fuel into the air. So that means, mass flow rate in the turbine is different because the compressed air comes from the compressor. Till that point of time there was no fuel and in the combustion chamber fuel gets added. So, be by virtue of this the mass flow that enters into turbine is different and than that of compressor.

So, this virtue of the fuel addition the mass flow rate in the turbine is greater than that of compressors. Another important issue is that, in the gas turbine engines, the turbine blades are typically exposed to the highest temperature, that temperature could be 1350 K or more than that.

So, it means that in a continuous operation the blades are likely to experience very high temperature and we call this kind of phenomena is thermal endurance and with time the blade quality degrades. So, to have a longer lifespan of the blade we must cool the blades continuously.

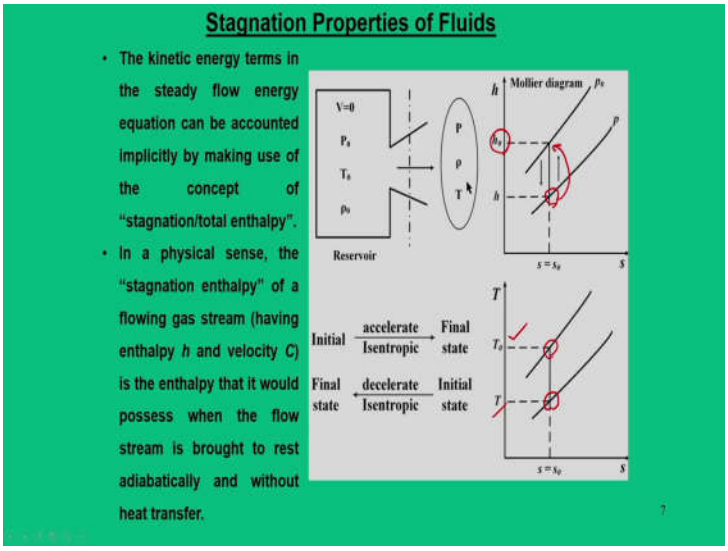
So that means, internal cooling is allowed through the blade roots. So, we call this as a air cool blades. Now, from where this air will come from? This air normally comes from the compressors; the combination of the stator, rotor and disc parameter from the compressor. Some of the compressed air is bypassed to the turbines just for the cooling purposes.

So, that means, a total about 15 percent of the compressed air delivery goes as a bleeding and this phenomena you called as a bleed flow or bleeding. And this bleed flow why do we require? Just to cool the turbine blades. So, in a sense that we must allow this fact for a continuous operations and for that reasons we have to add some extra mass in terms of fuel.

So, this fuel air ratio is in the range of 0.01 to 0.02 gets compensated or gets added to counter the amount for this kind of bleed flow. So, whatever I have discussed all these issues need to be accounted and we call them as a component losses and that has to be incorporated in a real gas turbine cycles.

However, some of these losses we can do it mathematically or do it by incorporating by simple changes and some of the things we get the information through continuous experimental data. So, based on those observations the losses are quantified.

(Refer Slide Time: 13:15)



Now, the next section that we are going to discuss is the stagnation properties of the fluid. So, why we introduce the stagnation properties of the fluid? Because of the fact that the first important information that aircraft propulsion system has it uses turbo machines and it has like compressors and turbines.

So, entry and exit state of the components turbines and compressors, they have essentially very high velocities. And when you say high velocity they also has certain kind of energy and we call this is a kinetic energy. And. in fact, this kinetic energy is a significant contribution as far as the total energy is concerned.

Now, to account for this fact, so when you actually say the temperature, the temperature is whatever we measure that is due to the static temperatures, but that temperature does not account due to this kinetic energy or velocity associated with the flow.

So, for that reason we implicitly define a properties which hypothetically we call as stagnation properties and they implicitly use these velocity components. How? So, let us explain that. So, the kinetic energy in terms of steady flow energy equation can be accounted implicitly by making use of concept of stagnation or total enthalpy.

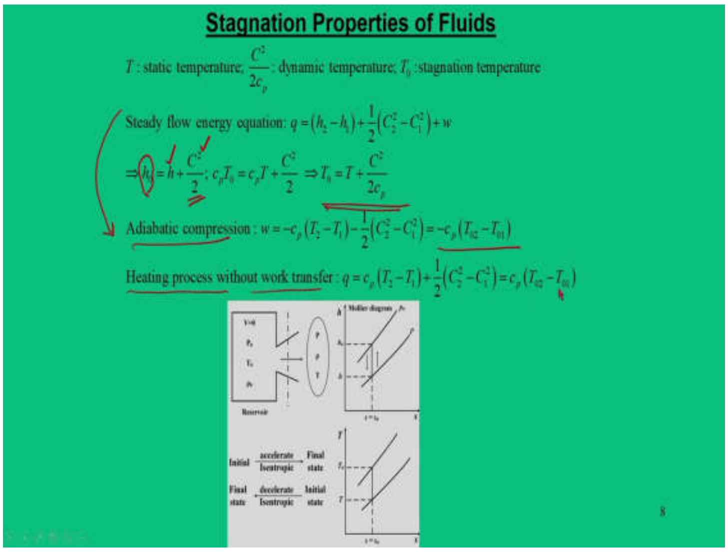
So, in a physical sense stagnation enthalpy of a flowing fluid stream is the enthalpy that it to possess when the flow is brought to rest adiabatically without heat transfer. For example, if you look at this particular figure.

We have a reservoir it contains total energy and that energy we say in terms of stagnation temperature, stagnation pressure or stagnation density. And as this reservoir has total energy content, by appropriating suitable diverging section we can accelerate the flow.

Or in other sense is that when the flow is already in accelerating stage that means, we can decelerate in such a way that we can bring it adiabatically and without heat transfer to rest. So, for a given arbitrary systems if you have initial state we can accelerate isentropically to a final state and when you are in the initial state and we can decelerate the flow isentropically to the final state.

So, now I am putting this stagnation enthalpy in the reversed definitions that already the flow has certain velocity, certain pressure and certain temperature. We are decelerating the flow so that its velocity becomes 0. Now, in those case if you want to refer in this enthalpy diagram and if the flow has certain enthalpy at a certain pressure; so we can define another parameter what is called as a stagnation enthalpy which is much higher than this. That means, by introducing a velocity component for a given flow at an arbitrary state, for that particular state another property can be defined that is represented as stagnation state. So, in a sense that at any arbitrary state if your enthalpy is h, its stagnation enthalpy could be defined as h_0 by introducing these velocity components. Similarly, the static temperature is T, corresponding stagnation temperature would be T_0.

(Refer Slide Time: 17:16)



Now, how do you do this mathematically? Let us see here. Now, you recall the steady flow energy equations $q=(h_2-h_1)+\frac{1}{2}(C_2^2-C_1^2)+w$. So, basically heat transfer q, w is the work transfer, change in enthalpy and change in the kinetic energy.

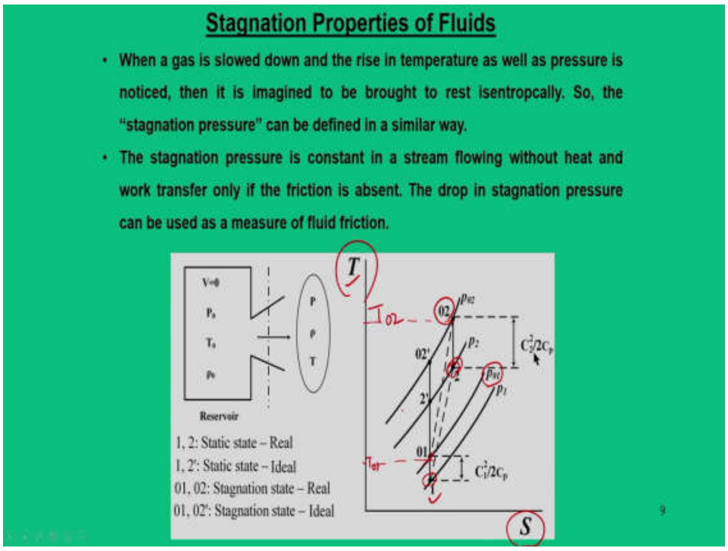
Now, if you apply this equation at two different states, one at stagnation state other at arbitrary states and without any work and heat transfer these equations can be represented as $h_0=h+\frac{C^2}{2}$. So, without heat transfer and without work transfer enthalpy can be represented as $h_0=h+\frac{C^2}{2}$.

So, implicitly if you define as h_0 it accounts for static enthalpy, it accounts for the enthalpy associated with the kinetic energy and then we can simply express enthalpy in terms of $c_pT_0=c_pT+\frac{C^2}{2}$. So, the dynamic temperature becomes $\frac{C^2}{2c_p}$. So, from this enthalpy expression we can define as the stagnation temperatures.

Now, going for another situation that from this equations another way of representing the steady flow equations is adiabatic compression situation and heating process without work transfer. What we can say is that another way of representation of the steady flow equations can be in the form of adiabatic compressions and for that process instead of static temperatures and we can also represent in terms of stagnation temperature.

So, in earlier compression system we used c_p in a compression is $c_p\left(T_2-T_1\right)$, but here the way of representation would be $c_p\left(T_{02}-T_{01}\right)$. So, in similar sense we can do the heating without heat transfer could be q is equal to $c_p\left(T_{02}-T_{01}\right)$.

(Refer Slide Time: 19:33)

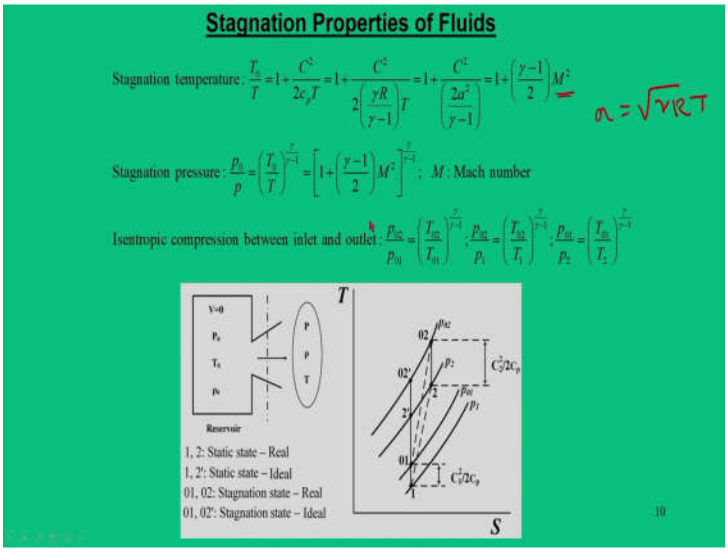


So, now, by defining this stagnation state we can just revisit the temperature entropy diagram when there are two states 1 and 2. So, the there are two practical states; that means, a compression system compression process from state 1 to 2. Now, for the state 1 the hypothetical way of representing its stagnation temperature as T_{01} and for that stagnation temperature the stagnation pressure would be p_{01}.

Now, when the system is at 2 the corresponding stagnation pressure would be p_{02} and corresponding stagnation temperature would be T_{02}. So, in a sense that for any arbitrary states we can define its corresponding stagnation states by accounting the velocity factor.

So, in a first case when it was state in a state 1 its velocity factor was $\frac{C_1^2}{2c_p}$ and in second case the velocity fact came factor is $\frac{C_2^2}{2c_p}$.

(Refer Slide Time: 21:05)

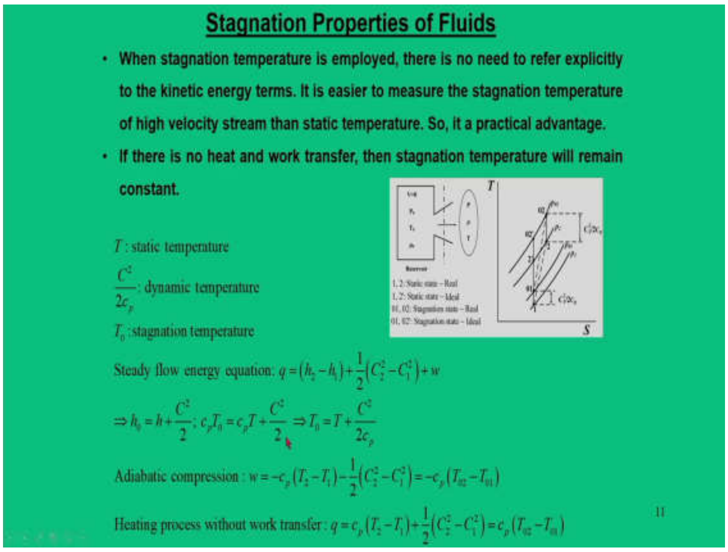


And by doing this another expressions that we can derive that by defining the stagnation temperature $\frac{T_0}{T}=1+\frac{C^2}{2c_pT}$. So, here you use this c_p as is equal to $\frac{\gamma R}{\gamma-1}$. Now, here we can write this $a=\sqrt{\gamma RT}$ that is nothing but speed of sound.

So, speed of sound is normally defined by $\sqrt{\gamma RT}$ and by taking the ratio of velocity of the fluid to the speed of sound we now represent this stagnation temperature in a parameter which is M and we call this M as Mach number. So, this Mach number of the fluid is very vital factor in deciding or in calculating the stagnation temperatures and pressures.

Now, having said all these things when the system goes from state 1 to 2 different isentropic relations can be derived from $\frac{p_{02}}{p_{01}}=\left(\frac{T_{02}}{T_{01}}\right)^{\frac{\gamma}{\gamma-1}}$. Another important factor I would like to explain that from the beginning we assume that the process happens to be adiabatic. Now, what happens in such situations? If the changes are also associated with change in the pressure then such a process can be considered as an isentropic process by defining this stagnation properties.

(Refer Slide Time: 22:50)

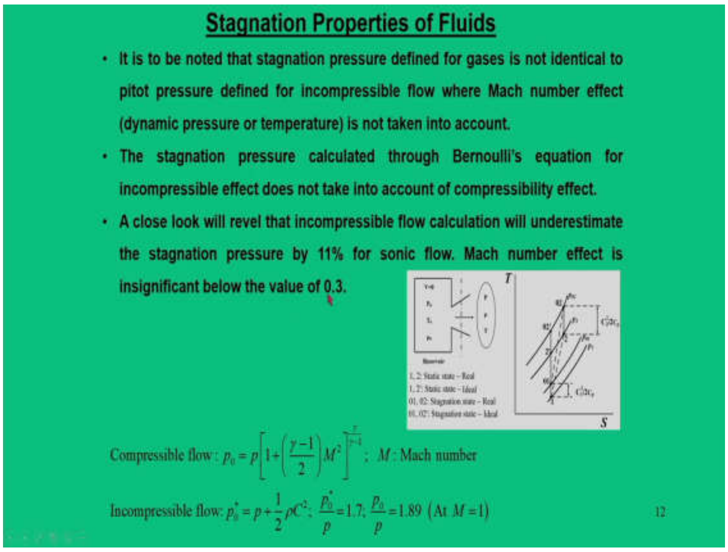


Having said this when you say these stagnation properties, when the stagnation temperature is employed there is no need to refer explicitly kinetic energy terms because it is easier to measure stagnation temperature of high velocity stream than that of the static temperature. So, it is a practical advantage.

And in a process if there is no heat and work transfer then stagnation temperature will remain constant. So, as you can see here that from the steady flow equations when you derive this adiabatic compression if there is no work transfer then you will see that in the process total $T_{02} = T_{01}$.

But when total temperature remains same, the static temperature or speed could vary, but the total temperature remains same. Similarly, for an heating process if there is no heat transfer total temperature remains same, but static temperature and speed could vary.

(Refer Slide Time: 24:00)



Another aspect that I just want to explain is that prior to define this stagnation properties, we also come to know similar term when you deal with the pitot. That point of time when you deal with in compressible flow we also say that stagnation pressure of a flowing fluid can be defined by static pressure plus the dynamic pressure that is $p_0^* = p + \frac{1}{2}\rho C^2$.

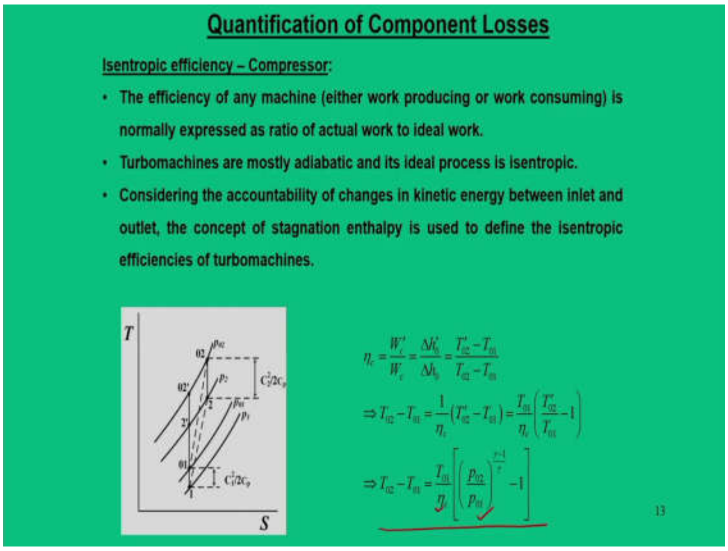
But there main assumption was that fluid was incompressible and that point of time we say p is a static pressure and $\frac{1}{2}\rho C^2$ is your dynamic pressure. But, that point of time the effect of Mach number was not taken into account as when we say incompressible, role of density is insignificant.

But when you deal with the compressible flow, the density changes are so significant that it will also change the stagnation pressure value. So, a typical quantification could be that if a stagnation pressure is defined for a compressible flow and for an incompressible flow for a some situations that when you have Mach number of 1, we will find that from the Bernoulli's equations we will find this ratio $\frac{p_0^*}{p} = 1.7; \frac{p_0}{p} = 1.89$.

So, there will be about 11 percent difference or deviations. So that means because the Mach number effect was not taken into account when you deal with Bernoulli's equations. So, in a sense that one should not confuse about definition of stagnation pressure when you deal with the pitot pressure or based on the Bernoulli's equation calculation and the stagnation pressure which is calculated from the isentropic relations.

So, in fact, when you look for the jet propulsion cycle we normally think about the flow is compressible for which the Mach number is always more than 0.3 and effect of Mach number becomes insignificant below the value of 0.3.

(Refer Slide Time: 26:35)



Now, let us go one by one how to quantify the component losses. First type of loss that we are going to introduce is the isentropic efficiency and that is for the compressors. So, previously we also defined this isentropic efficiency in our previous lectures, but that was defined in a context of static temperature or pressure.

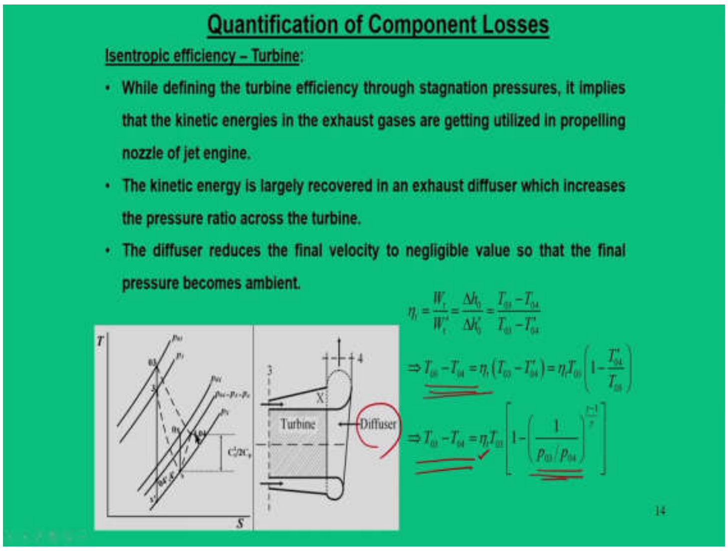
Because when you deal with the gas turbine engines with a power generation motive the terminal velocities at the components particularly compressor and turbines, they do not offer significant contribution to the total energy. So, in more or less we simply neglect those numbers.

But, when you deal with the turbo machines involved in the gas turbine engines for air propulsion methodology then we have to represent those efficiency, but in terms of stagnation temperatures or pressures. So, this is how we did it very basic difference that instead of representing the efficiency in terms of static value we are going to write them in a stagnation temperature difference.

So, first thing when you say work consuming device the isentropic efficiency can be defined as the isentropic work to the actual work. So, isentropic work will be less. So, compression efficiency is defined. So, based on that we can calculate in a compressor the total temperature difference in terms of component efficiency that is compressor efficiency $T_{02} - T_{01} = \frac{T_{01}}{\eta_c}\left[\left(\frac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$.

And a since it is easy to represent the temperature in terms of pressure, one can represent the total temperature difference across the compressor can be represented in terms of component isentropic efficiency of the compressor and the stagnation pressure ratio.

(Refer Slide Time: 28:50)



In a similar sense when you deal with turbine one can write the turbine efficiency as the actual work to the isentropic work. So, actual work for a turbine can be represented in terms of stagnation pressure that is $\frac{c_p\left(T_{03} - T_{04}\right)}{c_p\left(T_{03} - T'_{04}\right)}$. So, c_p will gets cancelled and finally, we get the turbine efficiency.

In a similar sense where the temperature difference across the turbine that is total temperature difference in terms of turbine efficiency and in terms of pressure ratio as well. So, in both way that pressure and temperature are incorporated in the calculations.

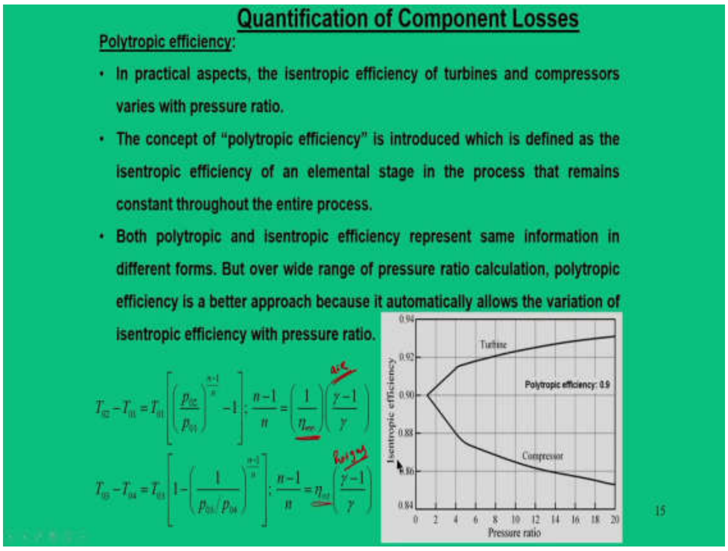
$$T_{03} - T_{04} = \eta_t T_{03}\left[1 - \left(\frac{1}{p_{03}/p_{04}}\right)^{\frac{\gamma-1}{\gamma}}\right]$$

So, this is how you deal with isentropic efficiency for turbines, but what happens is that in a real sense when the turbines are incorporated in jet engines the main intention is to recover thrust. So, this turbine which is used in the jet engine is integrated with a nozzle kind of assembly followed by a diffuser.

So, since we have to extract as much as energy from the exhaust flow when dealing expansion process, so, typically we get a higher pressure ratio across the turbine and because you are incorporating them through a nozzle shaft arrangement. Second thing we also expect that when the exhaust gas goes out of the atmosphere its pressure is almost close to ambient. So that means, we incorporate a component what we call as a diffuser.

So, what does a diffuser do? It brings the final velocity to a negligible value so that pressure becomes ambient. But, we have to just to remember that while dealing with the component efficiency, isentropic efficiency turbine we still look into the expansion process from T_{03} to T_{04}. So, this is the actual expansion process in a turbine.

(Refer Slide Time: 31:11)



Now, apart from this there is another way of representing this isentropic efficiency what we call as a polytropic efficiency. Let us see that we also have isentropic efficiency then what is the necessity of defining a polytropic efficiency. What has been observed is that if when you say isentropic efficiency, it has a fixed value, but when you say fixed value it has to operate at certain pressure ratio.

If you look a wide range of pressure ratio, say from 2 to 20, it can be seen that the isentropic efficiency of the compressor varies from maybe from 0.85 to 0.9 and similarly when you go for higher pressure ratio turbine efficiency increases. So, in a sense that we say that even though we define isentropic efficiency, but it is a function of pressure ratio. So, it means that in a wide range of pressure ratio the isentropic efficiency also changes.

So, to counteract this argument if you have very wide range of pressure ratio in comparison, people used to define a concept called as a polytropic efficiency. That means, if you recall the same expression that we used in terms of stagnation temperature and stagnation pressures for compressors and this is for turbine that is $T_{03} - T_{04} = T_{03}\left[1 - \left(\frac{1}{p_{03}/p_{04}}\right)^{\frac{n-1}{n}}\right]$.

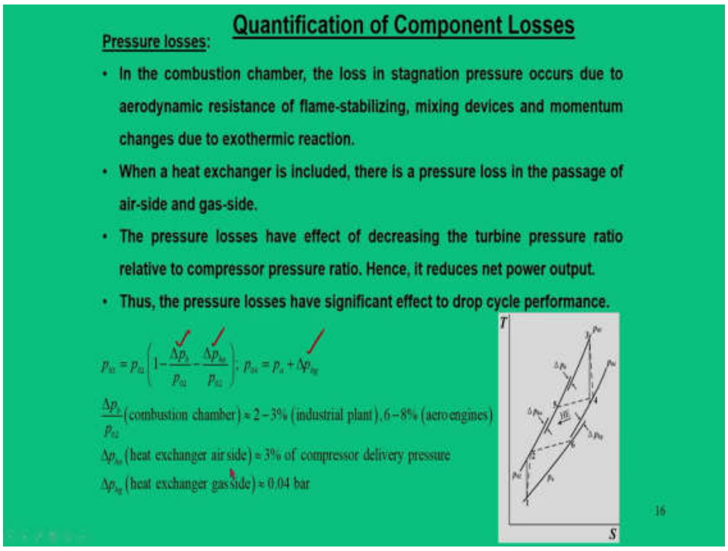
So, if you look at these expressions only difference that lies is that instead of γ we are representing that exponential parameter as $\frac{n-1}{n}$ and that $\frac{n-1}{n}$ is defined in terms of polytropic efficiency $\eta_{\infty c}$. $\eta_{\infty c}$ means, it is a polytropic efficiency, c stands for compressor. Similarly, $\eta_{\infty t}$ that is polytropic efficiency for turbines.

And of course, we can say that when you say compressor it is for air and when you say turbine it is for hot gas. So, this gamma is also going to change. So, in a sense that by keeping polytropic efficiency same, and as gamma has a realistic value for air and hot gases, we can have a value of n defined.

$$\frac{n-1}{n} = \left(\frac{1}{\eta_{\infty c}}\right)\left(\frac{\gamma-1}{\gamma}\right); \frac{n-1}{n} = \eta_{\infty t}\left(\frac{\gamma-1}{\gamma}\right)$$

And for this reason a expression of polytropic efficiency has a better advantage when we compare the isentropic efficiency in a wide range of temperature and pressures for compressors and turbine, but more or less the physical meaning of isentropic efficiency and polytropic efficiency is same.

(Refer Slide Time: 34:17)



Now, let us see that how you are going to calculate the component losses and it is a pressure loss, so, we have talked about compressor and turbine. Let us see what happens to pressure losses in heat exchanger. In fact, heat exchanger is also an important component and these heat exchanger component losses are taken into account through pressure losses.

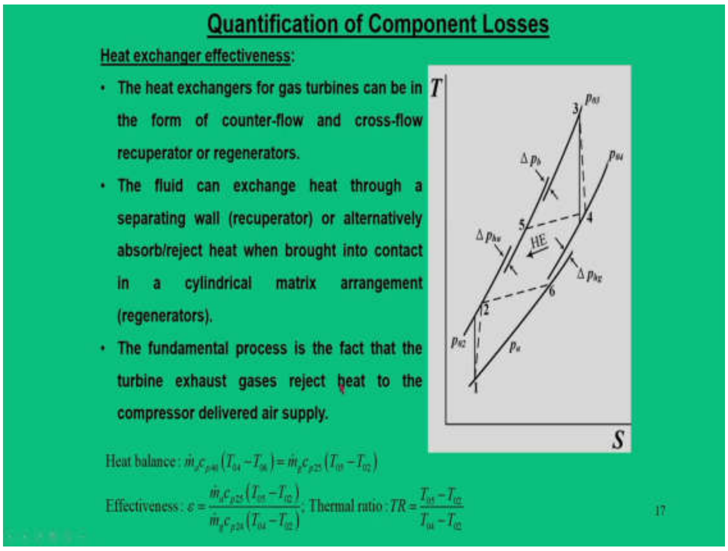
So, what happens when the heat exchanger is normally incorporated. The main intention is that to tap the heat from the exhaust of the turbine and they has to preheat. So, in the process of the pressure losses in the passage of air side and gas side there is a drop in the pressure and we say Δp_b. And at the same time we have the pressure loss that is accounted due to air side and due to gas side.

So, in this process we take the pressure losses into account and the T-S diagram slightly changes by taking these pressure losses. So, typical estimate that can come from the total pressure $p_{03} = p_{02}\left(1 - \frac{\Delta p_b}{p_{02}} - \frac{\Delta p_{ha}}{p_{02}}\right)$; $p_{04} = p_a + \Delta p_{hg}$.

So, the pressure losses in air side and gas side is taken into account and p_b that happens in the combustion chamber that is also taken into account. And over wide range of experiments a typical number that has been generated these losses are quantified $\frac{\Delta p_b}{p_{02}}(\text{combustion chamber}) \approx 2-3\%\ (\text{industrial plant}), 6-8\%\ (\text{aero engines})$.

And $\Delta p_{ha}\ (\text{heat exchanger air side}) \approx 3\%$ of compressor delivery pressure whereas, in the gas side because of the low pressures, we quantify typically 0.04 bar. So, these numbers we can use widely for our calculations.

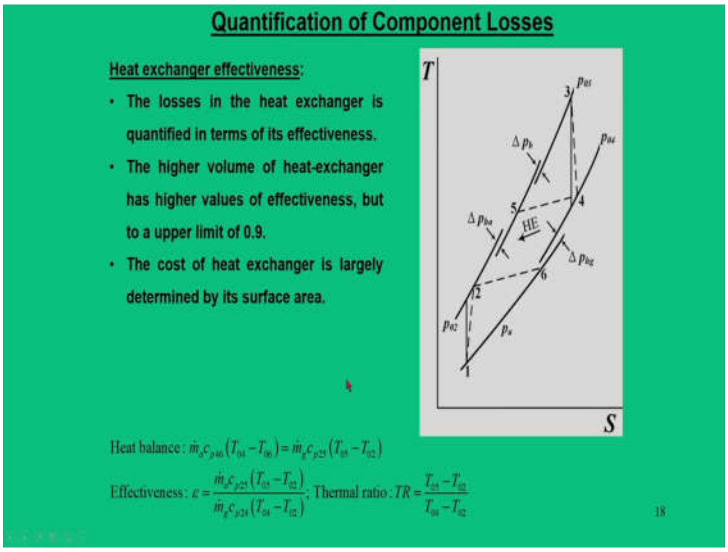
(Refer Slide Time: 36:42)



And again moving from the heat exchanger we also defined the heat exchangers effectiveness to quantify in the losses in our previous study. And there we express the losses in the heat exchanger as a function of effectiveness, or in the thermal ratio.

Now, why do this effectiveness is required? Because the heat exchangers are typically in the form of counter flow or cross flow recuperator or regenerator. And the main fundamental process here is that they have to tap the heat from the turbine exhaust.

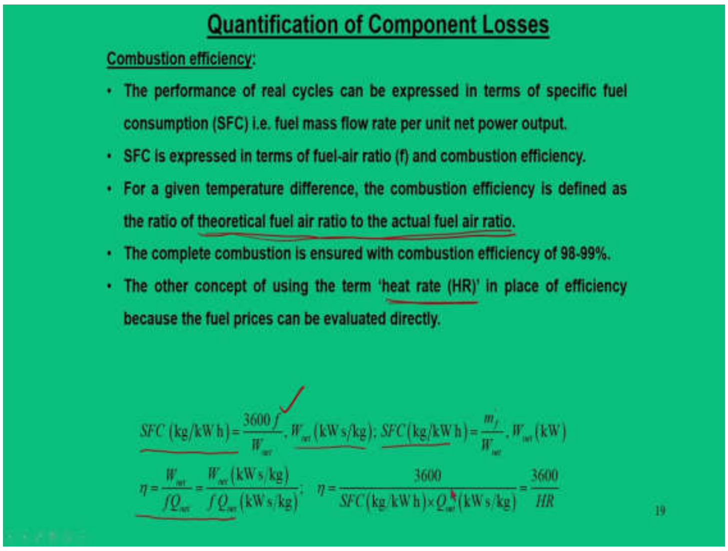
(Refer Slide Time: 37:32)



And by incorporating all the pressure losses they must perform in a economic way. And another aspect is that ideally if you use higher volume heat exchanger it will have higher effectiveness, but then the cost will be high because cost is mainly determined by its surface area and volume.

So, the cost will be high. So, in order to have economic size heat exchanger we must assume some effectiveness value and that is limited in the range of 0.9. So, this is another way of quantifying the effectiveness of a heat exchanger component.

(Refer Slide Time: 38:20)



Then we will move to the combustion efficiency. So, ideally speaking while dealing with the combustion efficiency we need some important parameters. First thing we ensure that the combustion process is not complete and typically by assuming a combustion efficiency of 98 to 99 percent we say that it is a incomplete combustions. Now, when it is incomplete combustion then we should not combust the complete fuel.

So, for which we have to find a ratio that is known as fuel air ratio that is theoretical fuel air ratio for complete combustion to the actual fuel air ratio and that fuel air ratio represent as *f.* Whereas, in the IC engine we call this as air fuel air ratio or fuel air ratio, but in the gas turbine engines we mainly focus on fuel air ratio because fuel is the most vital part particularly in the aircraft engines because it is costly.

So, based on that one can define this combustion efficiency and in terms of net work that is produced and how much fuel we are going to consume in terms of fuel air ratio from its calorific value. So, the total cycle efficiency can be also represented in this manner.

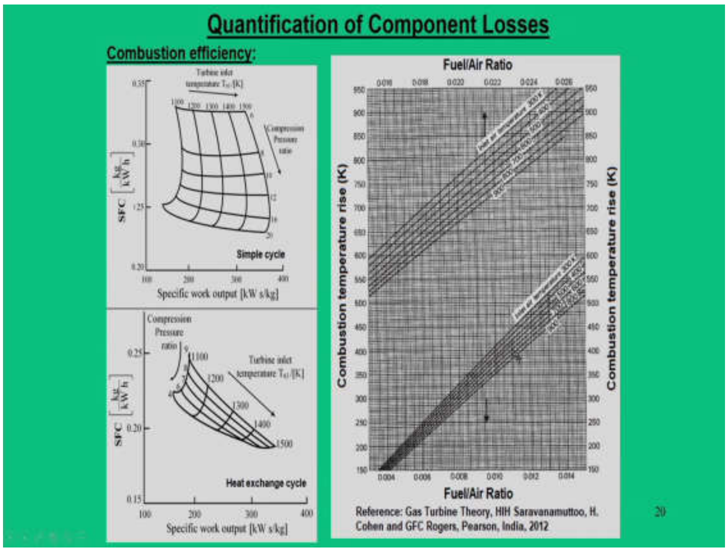
$$\eta = \frac{W_{net}}{fQ_{net}}$$

Apart from that there are some standard way of representing specific fuel consumption normally represented as $\mathrm{kg/kW\,h}$, where W that is equal to $\frac{3600\,f}{W_{net}}$, where

$$W_{net}\left(\mathrm{kW\,s/kg}\right);\ SFC\left(\mathrm{kg/kW\,h}\right) = \frac{m_f}{W_{net}},\ W_{net}\left(\mathrm{kW}\right).$$

In gas turbine engines the other concept of instead of efficiency they call at heat rate. So, we can relate between heat rate and efficiency that is $\eta = \frac{3600}{SFC\left(\mathrm{kg/kW\,h}\right)\times Q_{net}\left(\mathrm{kW\,s/kg}\right)} = \frac{3600}{HR}$. So, that is what the standard way of representation of combustion efficiency.

(Refer Slide Time: 41:02)



So, this particular plot shows that in a real gas turbine engines how specific fuel varies with specific work output for a simple cycle and for a heat exchange cycle. If you see here other parameter that was introduced is turbine inlet temperature.

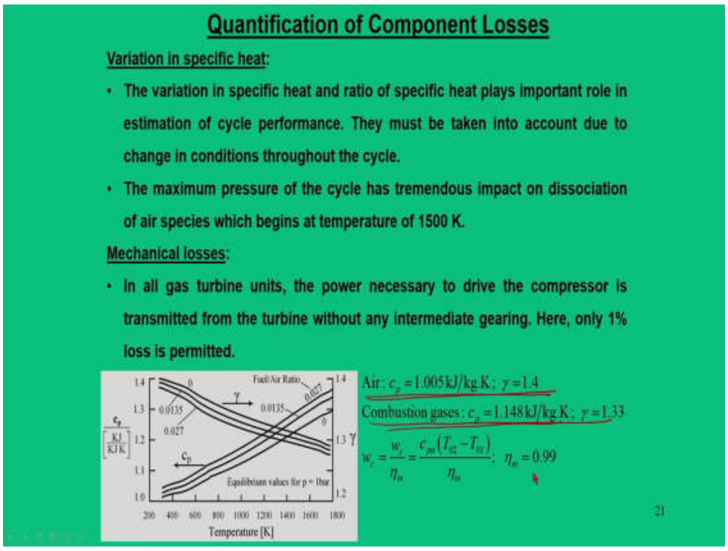
And with increase in the pressure ratio you can see this specific fuel consumption drops. And that means, if you operate a very simple cycle we require very wide range of specific fuel consumption. But, when you go for heat exchange cycle for same power output, fuel efficiency has a very steady value and of course, we can play with wide range of turbine inlet temperatures and compression pressure ratio.

So, these are the some characteristic curves that defines the specific fuel consumption variation with respect to specific work output. And of course, there is another table that talks about how you are going to calculate the fuel air ratio with respect to combustion temperature rise. So, I have taken it from the reference gas turbine theory from Saravanamuttoo and Cohen and Rogers books.

And what it says is that in a given problem you are suppose to find a fuel air ratio, but what you will be knowing is the terminal temperatures. So, this terminal temperatures are in particular in the combustion are represented as a combustion temperature rise. So, in a combustion chamber the temperature rise is defined in the y axis and for a given inlet air temperature that comes out from the compressor is drawn in this line.

That means, if you know this inlet temperature of air and if you know the combustion temperature rise then you can easily plot what is the fuel air ratio. So, these curves talks about that how you are going to get the fuel air ratio graphically. And over wide range of operating temperatures these graphs has been plotted. So, graphically it is possible to quantify.

(Refer Slide Time: 43:35)



Now, apart from this other type of loss which is are some simple losses. We can introduce the variation in the specific heats. As I mentioned these temperature plays very vital role and in the thermal always the cycle temperature at different location changes continuously. Maximum cycle temperature could be as high as 1500 K.

That point of time dissociation of air species plays a very vital role. So, because of this reason we say that the specific value from the compression side we take for air and the specific heat value for turbine side we call this as a combustion gases.

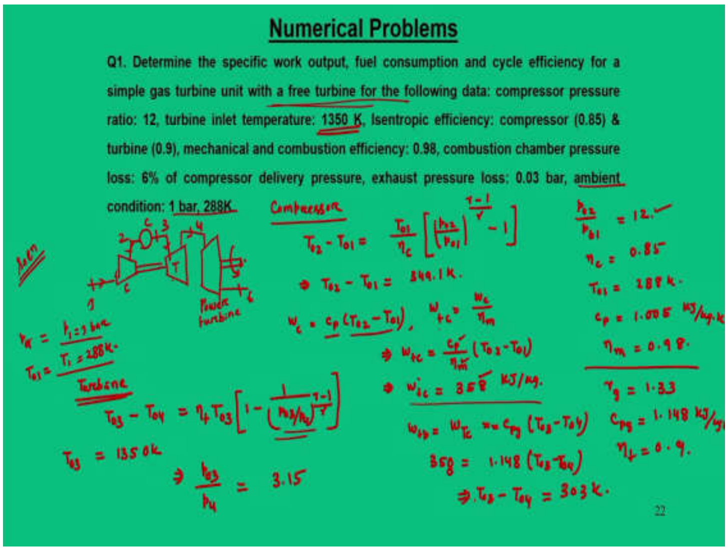
So, c_p for air you can write as 1.005 kJ/kg-K and that for gamma is 1.4 and for combustion gases we say c_p as 1.148 kJ/kg-K and gamma is equal to 1.33. So, this particular plot says that how with temperature the c_p value changes and how the gamma value also drops.

So, roughly that means, with temperature the gamma value drops. At some point down the line you take the entire variation of c_p as a function of temperature as well as gamma as function of temperature or you take a consolidated average value that make suitable justifications.

So, in our analysis we will not say that variation with temperature rather we will take some two fixed values, one is compressor side, other is the turbine side and last part is the mechanical losses. As I mentioned always the compressor is driven by the turbine. So, there is a transmission losses in terms of bearing and windage. So, here only 1 percent loss is permitted.

So, for that reason the compressor power and turbine power is equal, but in between there is a term what we call as a mechanical efficiency and this mechanical efficiencies assume to be 98 to 99 percent.

(Refer Slide Time: 46:02)



So, with this I have introduced all kind of component losses and how they are going to quantified. Now, with this particular concepts let us see one numerical problem. So the problems are similar to what we have done in the earlier lectures, but here we are going to introduce the component efficiency and how they are going to be introduced and in particular in terms of stagnation temperatures.

So, this is how the philosophy is all about. So, for a gas turbine engine we need to find out specific work output, fuel consumption and cycle efficiency and it is with the following data. Compression pressure ratio is 12, turbine inlet temperature is 1050 K and we are given with isentropic efficiency for compressor 0.85, turbine 0.9, mechanical and combustion efficiency are given as 0.98, combustion chamber pressure losses as 6 percent of compressor delivery pressure, exhaust pressure loss is 0.03 bar and ambient condition is 1 bar, 288 K.

So, for that let us draw the diagram first that what we are going to calculate. So, simple gas turbine circuit we can have a compressor and that is driven by a turbine and we have a combustion chamber and of course, another thing is that with a free turbine. That means, there is a free turbine that means, the exhaust from the primary turbine is further expanded in another turbine and we call this as a free turbine that is for power generation.

So, we can put the state point as 1, exit from the compressor as 2, combustion chamber entry as2 and combustion chamber exit at 3 and first turbine exit at 4 and this is free turbine or we say power turbine and its exit is at 6 and this shaft is running, power turbine drives the generator. So, we start with 1 and ends with 6.

So, the condition that ambient condition is 1 bar and 288 K we can write p_1 as 1 bar T_1 as 288 K and to start these things let us go one by one first compressor. So, for compressor we recall the working formula that is in terms of stagnation temperatures.

We can write $T_{02} - T_{01} = \dfrac{T_{01}}{\eta_c}\left[\left(\dfrac{p_{02}}{p_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]$. So, we have to say this is T_1 as a T_{01} because this is ambient condition. And here the static and temperature are same because ideally when it is ambient condition. So, it is at zero velocity. Similarly, we can say p_{01} is also p_1. So, by putting this value we can find out.

$$T_{02} - T_{01} = 349.1K$$

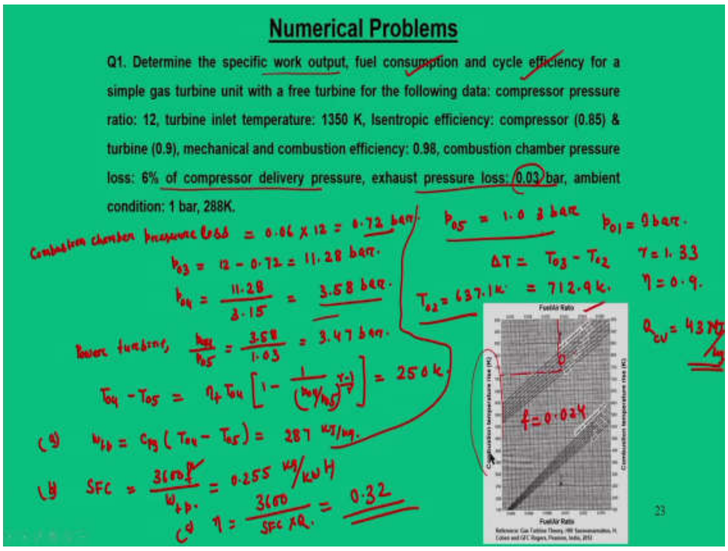
$$w_c = c_p\left(T_{02} - T_{01}\right); w_{tc} = \frac{w_c}{\eta_m} = \frac{c_p}{\eta_m}\left(T_{02} - T_{01}\right) = 358\text{kJ/kg}$$

Then you come back to turbine side because you also have to know other pressure ratio and temperatures. So, for turbine side we can write

$$\because w_{tp}=w_{tc}=c_{pg}\left(T_{03}-T_{04}\right)\Rightarrow 358=1.148\left(T_{03}-T_{04}\right)\Rightarrow T_{03}-T_{04}=303K$$

$$T_{03}-T_{04}=\eta_t T_{03}\left[1-\left(\frac{1}{p_{03}/p_{04}}\right)^{\frac{\gamma_g-1}{\gamma_g}}\right]\Rightarrow p_{03}/p_{04}=3.15$$

(Refer Slide Time: 55:17)



Then we have to find the component efficiency that means, you have to go to combustion chamber pressure loss. That is nothing but 6 percent of compressor delivery pressure that means,

Combustion chamber pressure loss = $0.06 \times 12 = 0.72$ bar

$$p_{03}=12-0.72=11.28\text{bar}$$

$$p_{04}=\frac{11.28}{3.15}=3.58\text{bar}$$

$$p_{05}=1+0.03=1.03\text{bar}$$

For power turbine: $\dfrac{p_{04}}{p_{05}}=\dfrac{3.58}{1.03}=3.47$

$$T_{04}-T_{05}=\eta_t T_{04}\left[1-\left(\frac{1}{p_{04}/p_{05}}\right)^{\frac{\gamma_g-1}{\gamma_g}}\right]\Rightarrow T_{04}-T_{05}=250K$$

Specific work output $w_{tp}=c_{pg}\left(T_{04}-T_{05}\right)=287\text{kJ/kg}$

Second part specific fuel consumption SFC = $\frac{3600f}{w_{tp}}$. Now to calculate f we require this particular graph, we require this f. So, for that we require the combustion temperature rise. So, in our case the combustion temperature rise $\Delta T = T_{03} - T_{02} = 712.9K$. So, corresponding to T_{02} of 637.1K and temperature rise of 712K, we can find out f from the graph as 0.024. So, putting this we can calculate SFC as 0.255 kg/kWh. So, we got the fuel consumption, the last part is cycle efficiency.

$$\eta = \frac{3600}{\text{SFC} \times Q}; Q_{\text{CV}} = 43\text{MJ/kg} \Rightarrow \eta = 0.32$$

So, this particular big problem helps you to get the complete understanding about a real gas turbine engine performance and that involves some of the parameters needs to be referred in a graphical manner.

And it will help you that how in a real gas turbine engines the component efficiencies are taken into account. So, this will help in the designing of a real gas turbine power plant. With this I conclude and thank you for your attention.