Applied Thermodynamics
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Module - 03
Internal Combustion Engines
Lecture - 25
Thermodynamics Analysis of Air Standard Cycles

Welcome to this course Applied Thermodynamics Internal Combustion Engine module 3.

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So, here are the list of topics, we are now in the lecture number 4, that is Thermodynamic Analysis of Air Standard Cycles.

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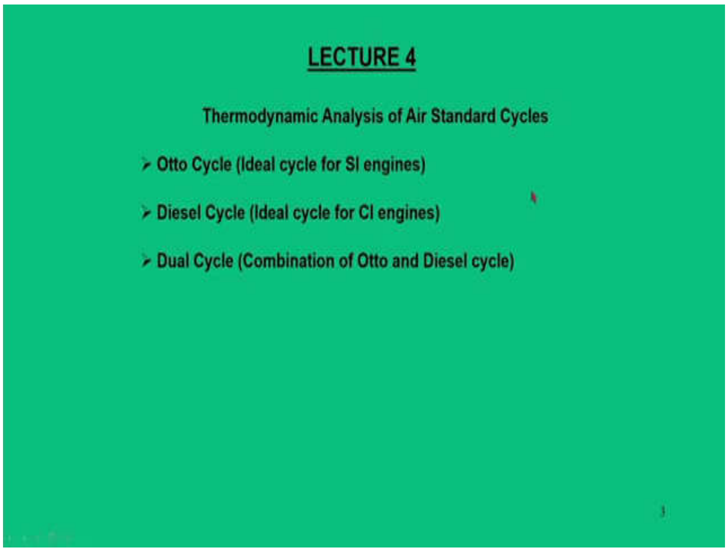

So, in this lecture we are going to discuss about three important cycles; one is Otto cycle, which is an ideal cycle for SI engines, diesel cycle which is an ideal cycle for CI engines, dual cycle which is a combination of both Otto and diesel cycle as far as the combustion mechanism is concerned.

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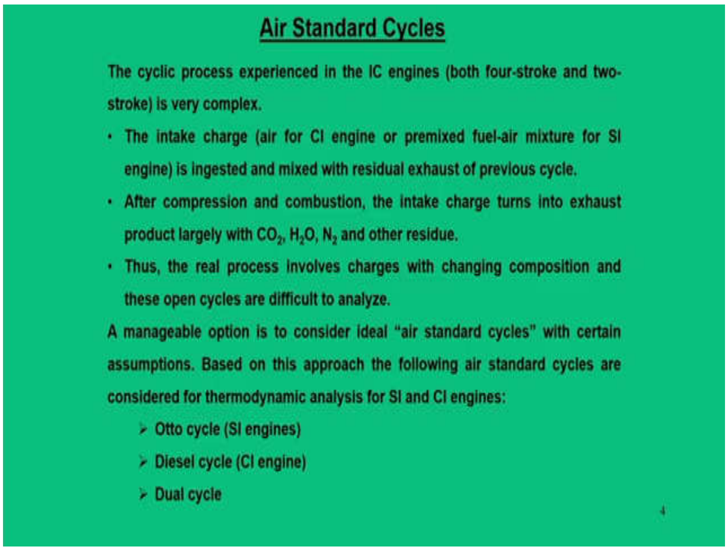

So, let me give you some brief introduction about what the air standard cycle means. Ideally, when you look at Carnot cycle, we need a working fluid to run the engines. In a similar sense, when you discuss about SI engine or CI engines, there are working fluids which are very specific to those engines. For example, in SI engines petrol is the base working fluid, for CI engines diesel is the base working fluids.

But, before you go to those analysis, the first estimate one should know about the fact, that this would give some estimate as far as thermodynamics analysis is concerned. To simplify this analysis the base working fluids like, that is petrol or diesel they are replaced with air.

So, when we are replacing working fluid with air, there are certain assumptions that we are going to make. So, to study all these things the importance of air standard cycles is realized. So, whatever I have explained that the cyclic process in IC engines is very complex either it is a four stroke or two stroke, the intake charge that is air in CI engine and premixed fuel air mixer for SI engine is ingested and mixed with the residual exhaust of the previous cycles, since it is a cyclic process.

After the compression and combustion the intake charge turns into the exhaust product which is largely in the form of CO_2, H_2O, N_2 and other residues. Hence, the real process involves charges with changing composition and these open cycles are difficult to analyze.

So, to simplify this analysis a manageable option is to consider “air standard cycles” with certain assumptions. So, based on this approach the air standard cycle is considered as the thermodynamic analysis for CI and SI engines. So, for SI engines it is the Otto cycle, for CI engine it is the diesel cycle. And, there is another cycle which is also gives a good estimate when you combine the combustion process of Otto and diesel cycles in a dual cycle.

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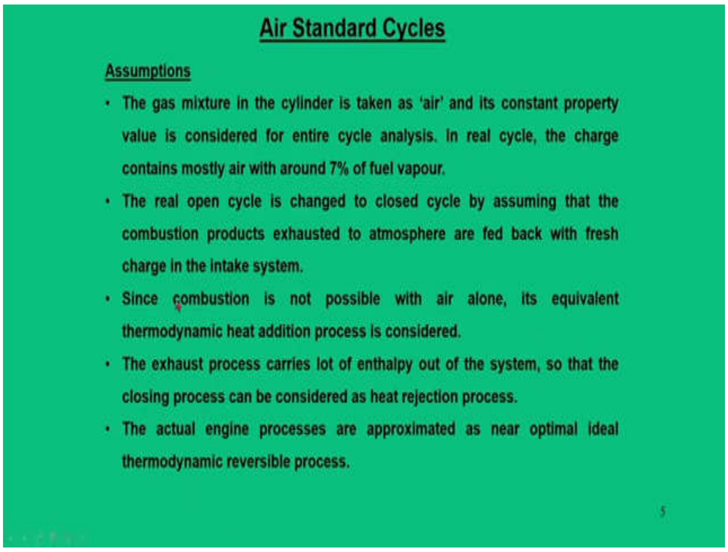

So, we will discuss it one by one. So, before you discuss them, let us see that what are the assumptions involved? So, when you tell that we are going to replace air as working fluid, there are certain logics, means that why we should go for air as a working fluid? So, it has been seen that when you see actually the fresh charge which is a mixture of fuel and air, the major constituents is air, because roughly the ratio between air and fuel is about 15.

So, that means, 15 times by mass of air is used to prepare the fresh charge involving fuel and air. So, predominantly the contents are air. Second thing, IC engines is a real open cycle and it is because the fresh charge is inducted every time, but in our assumption of air standard cycle, it is assumed that the combustion products are exhausted to atmosphere, and they are fed back with the fresh charge in the intake systems.

So, in one way that combustion product goes out, side by side we are feeding the fresh charge. Third assumption is that since the combustion is not possible with air alone. So, it is equivalent to consider the thermodynamic heat addition process appropriately means that, physically the cycle requires the heat addition processes through fuel.

But, here it is not possible to consider that, rather we say that whatever the amount of heat that is required, they are being fed into the cycle. Next assumption is that the exhaust process carries a lot of enthalpy out of the system, so that the closing process can be considered as a heat rejection process. Moreover the actual engine processes are approximated as near optimal to ideal thermodynamic processes.

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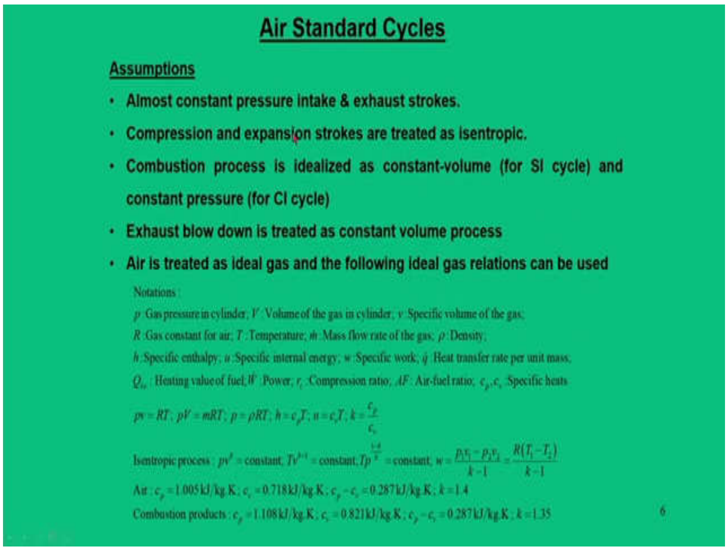

Now, we will consider about the assumptions that are involved in our calculations. We also have to give a thermodynamic meaning about the processes that are encountered in the IC engine. So, first assumption in that line is that, the intake and exhaust strokes are constant pressure processes.

Compression and expansion strokes are isentropic processes. Combustion process can be idealized either by a constant-volume processes, which is used in SI engine cycle or a constant pressure processes, which is used in the CI engine cycle. These exhaust blow down process is treated as a constant volume process. And, for all the study working fluid is treated as air and it is considered as a ideal gas.

So, here I have written down the some basic equations that an ideal gas follows. $pv = RT;\ pV = mRT;\ p = \rho RT;\ h = c_p T;\ u = c_v T;\ k = \dfrac{c_p}{c_v}$.

So, ideally for air $c_p = 1.005\,\text{kJ/kg.K}$; $c_v = 0.718\,\text{kJ/kg.K}$ $R = c_p - c_v = 0.287\,\text{kJ/kg.K}$.

And, here we consider this specific heat ratio k as 1.4, but to bring more clarity or more closeness to our analysis, we are going to use the c_p value in terms of combustion products. They are slightly different that is we are going to use the value of k as 1.35 and corresponding $c_p = 1.108\,\text{kJ/kg.K}$; $c_v = 0.821\,\text{kJ/kg.K}$ in our analysis.

But, interestingly you can see that $R = c_p - c_v$ for air as well as combustion products, they match very well. So, in some sense where R is considered, whether you take combustion product or air, this remains same in the calculations. So, with this logics, it makes some sense, that air can be considered as working fluid in air standard cycles for SI engines, CI engines and dual cycles.

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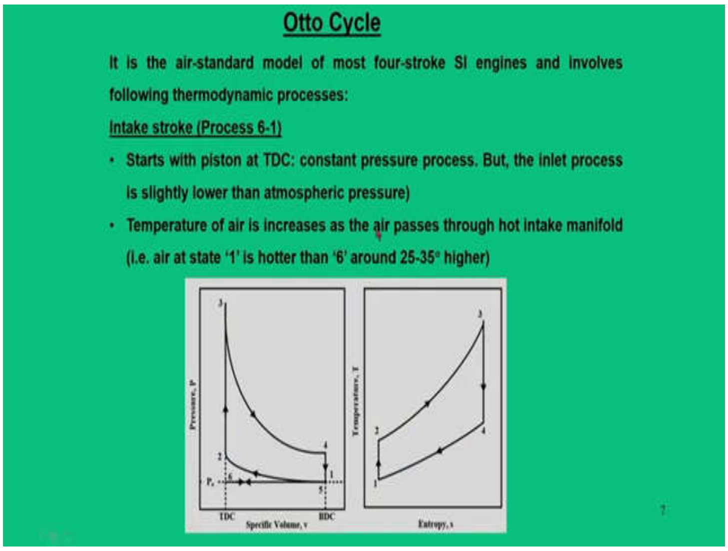

So, let us start with the first cycle that is Otto cycle. The air standard model for a four-stroke SI engines is considered for Otto cycle. So, it starts with an intake stroke as you see in this figure there are two types of diagram, which is represented; one is pressure volume diagram, other is temperature entropy diagram.

So, you start with the process 6-1, which is we called as a intake stroke. So, it starts with piston at TDC; so, it is a constant pressure process as you can see atmospheric condition is getting into this cylinder, so it is a intake stroke, that is from 6 to 1. But, ideally the inlet process is slightly lower than atmospheric. So, that flow can enter, because you need a pressure differential. So, at that time when we are in the state 1, the temperature of air increases as the air passes through hot manifold.

So, normally whatever atmospheric temperature we are considered that is slightly higher, because when it passes through the intake manifold, temperature of air increases. And, close to 25 to 35°C higher than the atmospherics so; that means, normally state point 1 is higher than atmospheric.

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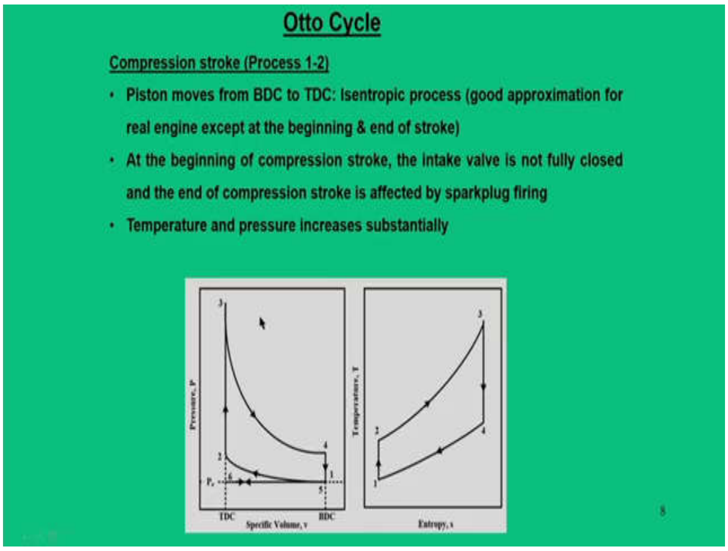

Then, we have compression stroke. So, the compression stroke starts when with piston moves from BDC to TDC. So, in this diagram you can see that when piston moves from BDC to TDC, this process 1-2 happens. So, and this process is isentropic in nature as you can see in the T-S diagram. So, in the beginning of the compression stroke the intake valve is not fully closed, and the end of the compression is affected by sparkplug firing. So, normally after the compression stroke you need to start the combustion.

So, sparkplug needs to be fired somewhere close to point 2. So, although we see a very sharp rise, but ideally it is not so, but in our air standard assumption that consideration is not taken seriously. So, now in the summary is that in the compression stroke the both pressure and temperature increases.

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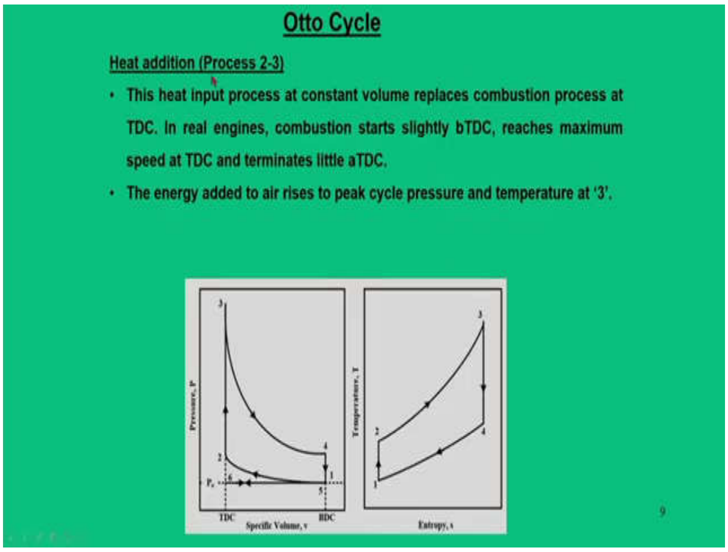

Next is the heat addition process. So, we have already explained that heat addition in a SI engine process can be best approximated, when it is considered as a constant volume processes. So, that means, point 2 to 3, we can regard it as a constant volume processes, when the piston is exactly at TDC, but in reality you will find that the combustion starts slightly before TDC and it reaches maximum speed at TDC and then terminates little after TDC.

So, somewhere in this zone we can see in reality you will find that the sharp peak what you are seeing in the Otto cycle is not so. But, the air standard cycle assumption does not take into account of these facts. So, the energy added in the air rises the peak pressure and temperature at point 3.

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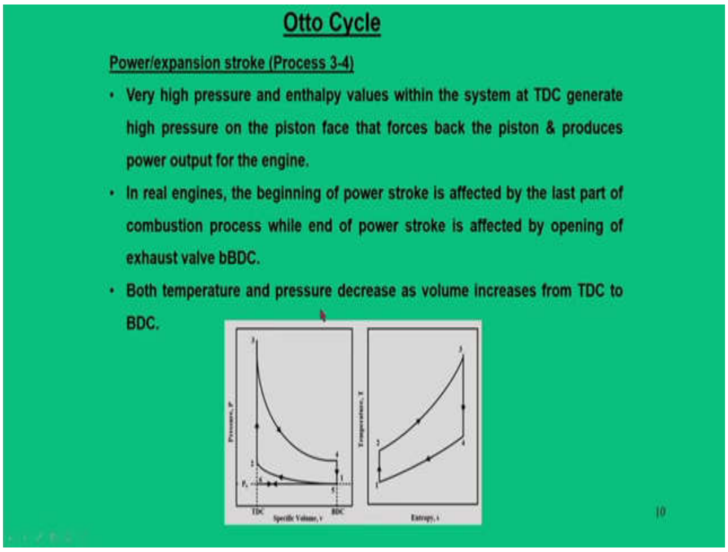

Then, we have power strokes. So, after the combustion process is over. So, pressure and temperature gets increased. So, the piston phase sees a force acting on it. So, in this process the piston starts moving again from TDC to BDC. So, we call it as a power strokes.

But in real engines the power stroke is affected by last part for the combustion processes, because still that point of time, the combustion is not complete, but the power stroke has also started. And, also this power stroke is also affected by the opening of exhaust valve that is before BDC.

Now, in the process of this expansion we can see 3 to 4 in an isentropic process. And, in this process of expansion, the pressure and temp temperature decreases, as the volume increases from TDC to BDC.

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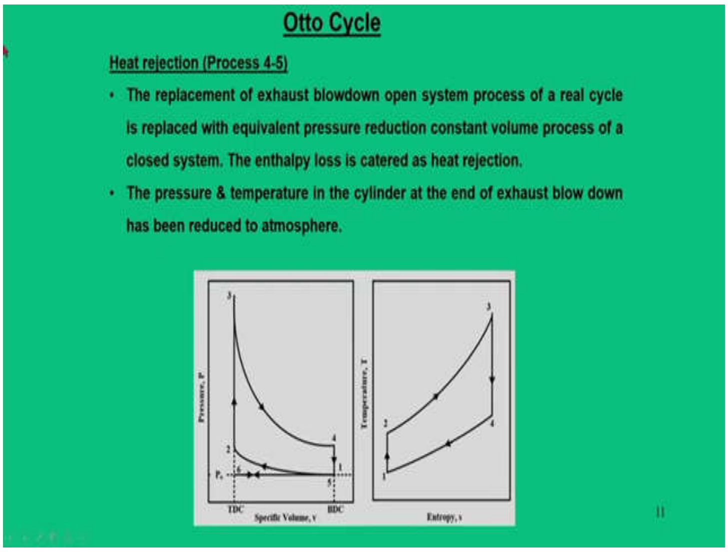

Then, we have heat rejection process. So, that is regarded as a constant volume processes from 4 to 1, and this happens close to BDC or mostly at BDC. And, this 4 to 1 is represented in the temperature entropy diagram like this. So, it is a constant volume process.

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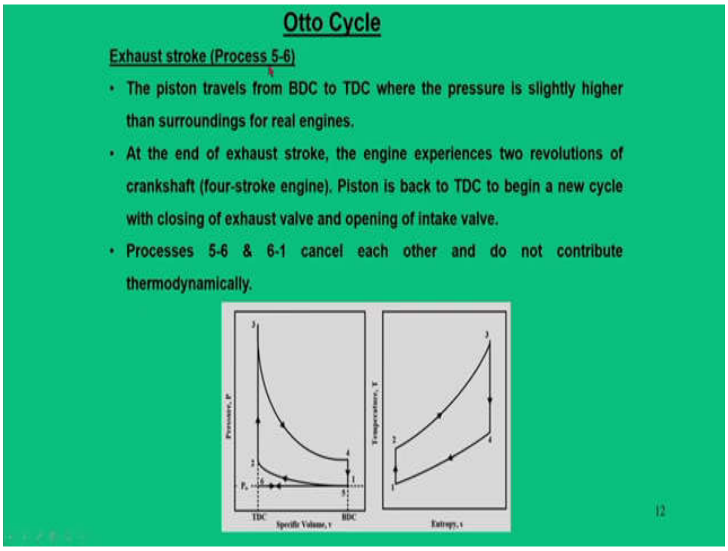

Then, the last stroke which you called as a exhaust stroke that is from 5 to 6. So, in a real engine the exhaust product goes out; that means, when the piston is at BDC the exhaust blow down starts, the exhaust valve opens. Then because of this reason, at the end of exhaust stroke the engine experiences two revolutions of crankshaft; that means, piston is back to TDC to begin a new cycle closing the exhaust valve and opening the intake valve. So, this happens at the end of exhaust stroke.

And, as you can see from this pressure volume diagram, when you started the intake stroke it goes as 6 to 1, and when you come as exhaust stroke it goes as 5 to 6, hypothetically or ideally the 5 and 1 points are very close. So, in our air standard cycle the state conditions at this point 5 and 1 are almost same.

So, when you do the work calculation since it is a constant straight line and arrow for each straight line are opposite in nature, so the work in that P-v diagram will get cancelled. So, that means, the intake and exhaust stroke work does not come into the thermodynamic analysis of Otto cycle.

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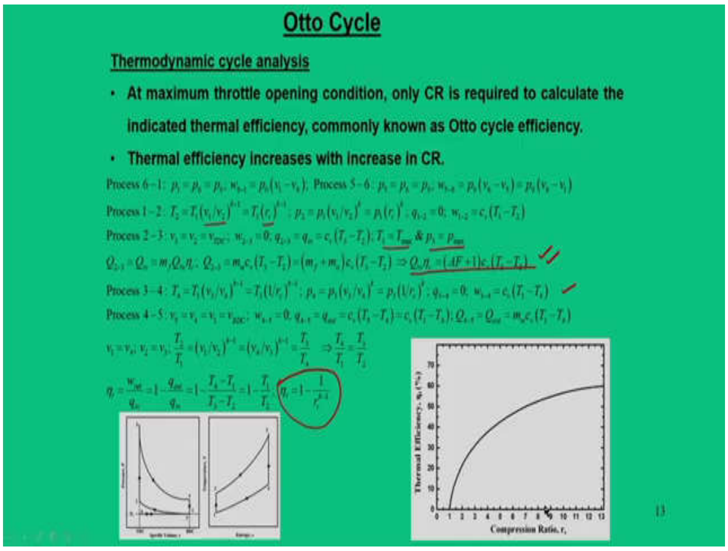

So, we will see this one by one now after explaining all these thermodynamic process, let us see that, how we can represent them in the form of equations? So, first thing is that when you look at the cycles, we basically look at the compression ratio, because of the piston location between 1 and 2 that is between BDC and TDC. So, we represent the compression ratio as $r_c = v_1/v_2$, that is one important parameter which we are going to use.

Now, let us see the process wise. So, let us start with the intake process that is 6 to 1. So, it is a constant pressure process $p_1 = p_6 = p_0$. And, the specific work for the process 6 to 1, we can write is $p_0\left(v_1 - v_6\right)$.

Similarly, for process 5-6 another expression can be written where $p_5 = p_6 = p_0;\ w_{5-6} = p_0\left(v_6 - v_5\right)$. So, if you add this equation work transfer for 6-1 and 5-6, it will be 0 because they will cancel each other. Then, we move to the compression process that is for 1 to 2.

So, we can write the equation as isentropic process. So, isentropic equation can be used for state points between 1 and 2, we can write $T_2 = T_1\left(v_1/v_2\right)^{k-1} = T_1\left(r_c\right)^{k-1}$.

Similarly, we can write pressure in terms of compression ratio, $p_2 = p_1\left(v_1/v_2\right)^k = p_1\left(r_c\right)^k;\ q_{1-2} = 0;\ w_{1-2} = c_v\left(T_1 - T_2\right)$. Then for process 2 to 3 that is constant volume heat addition $v_3 = v_2 = v_{TDC};\ w_{2-3} = 0;\ q_{2-3} = q_{in} = c_v\left(T_3 - T_2\right)$.

So, in that process you have the maximum pressure in the cycle that is $T_3 = T_{\max}\ \&\ p_3 = p_{\max}$ and that is nothing but we denote it as maximum pressure in the cycle, as p_{max} and T_{max}. Now, in this situation we are trying to make some correlation that, how we can make this q_{in} calculations in our analysis.

So, ideally instead of representing this q_{2-3}, where there is no mass term is involved; but in some extent in reality the fuel takes a role. So, in that sense what we can say $Q_{2-3} = Q_{in} = m_f Q_{hv}\eta_c$.

When we multiply these three things we get the total Q not in terms of kg. So, we get a total Q that is nothing but $\left(m_f + m_a\right)c_v\left(T_3 - T_2\right)$. So, these two equation gives rise to a another important equations that is $Q_{hv}\eta_c = \left(AF + 1\right)c_v\left(T_3 - T_2\right)$.

So, this particular equations is very vital in bringing the air fuel concept into the thermodynamic calculation of Otto cycles. So, rest of the things are similar to what we have already discussed that is Process 3-4, that is expansion process which is almost opposite to that of compression process.

So, we can write similar isentropic equations for expansion process. Process 4-5 is a constant volume process, because it is a blow down process. So, that point of time $v_5 = v_4 = v_1 = v_{BDC}$. And, in that process $w_{4-5} = 0$; $q_{4-5} = q_{out} = c_v \left(T_1 - T_4 \right)$.

So, from this also we can find out the total heat. So, instead of calculating heat per unit kg we can calculate the total heat as q_{out} that is bringing total mass into picture. So, from all this analysis, we can calculate the net work which is nothing, but difference in the expansion work and compression work, and q_{in} which is heat being supplied by the fuel from this equations.

Then, we can find out the thermal efficiency of the Otto cycle as $\eta_t = \frac{w_{net}}{q_{in}}$. And, ultimately when you bring out these isentropic relations, final expression we can represent this Otto cycle thermal efficiency as $\eta_t = 1 - \frac{1}{r_c^{k-1}}$. It is a simple expression, but, what we can see that r_c is a compression ratio; that means, indicated thermal efficiency is a function of compression ratio.

When you plot this equation it shows a rise; that means, with when you increase the compression ratio the efficiency rises. So, it can be plotted in the range of compression ratio in each engines are operated.

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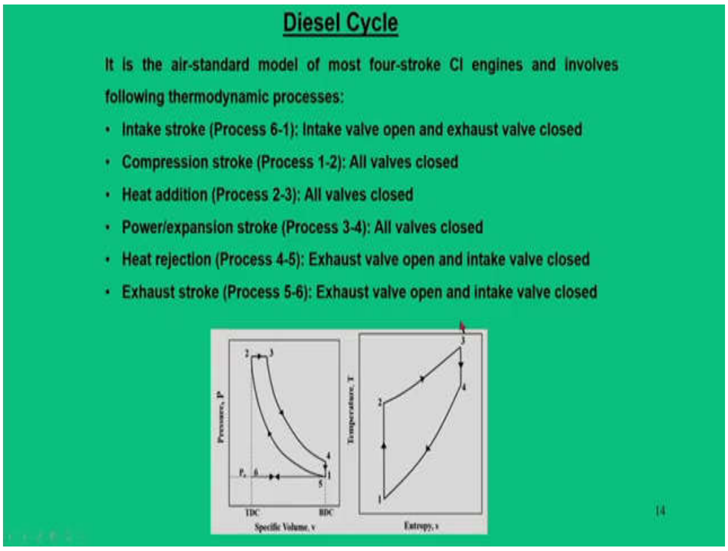

The another model for air standard model for 4 stroke CI engine is a diesel cycle. So, here I am not going to explain them in the details same processes are involved, that is except the certain change, that I will explain. The first one intake stroke process 6-1, compression stroke process 1-2, heat addition process that is process 2-3, here it is the change, because here it is a constant pressure processes, when the both the valves are closed. Then, you have power stroke or expansion stroke, heat rejection process and exhaust process.

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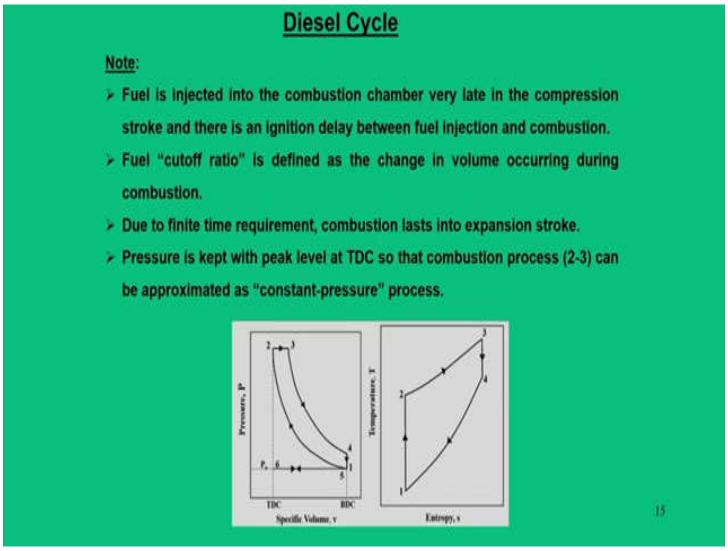

But, some important analysis here we are going to make here, that the assumption of constant pressure process, because fuel is added into the air. So, that it is not possible to maintain the constant volume process when fuel is added.

So, best approximation is that we should go for a constant pressure process. Since, the fuel is continuously injected towards the end of compression stroke and we can define a ratio which is called as fuel cut of ratio, that is based on what time the fuel injection starts and what time the fuel injection stops? So, at 2 it starts and at 3 it stops.

So, based on the ratio v_3 / v_2 a fuel "cutoff ratio" can be defined. So, rest of the processes are simple and now we will revisit the thermodynamic analysis.

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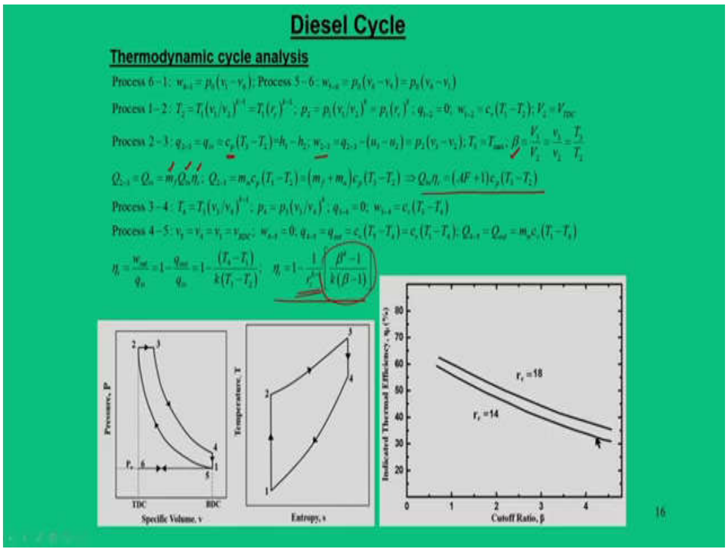

So, here we can see that all equations follow the similar one as we do in the Otto cycle, but only difference that we can see is the heat addition process. That is while calculating q_{2-3} the heat input we are using c_p here instead of c_v. And, since it is a constant pressure process, it also incurs the work transfer w_{2-3} and also we define a parameter fuel cut off ratio that is $\beta = \frac{V_3}{V_2}$.

And, from this analysis we can say $\frac{v_3}{v_2} = \frac{T_3}{T_2}$, but ideally when we are bringing the calorific value and combustion efficiency of the fuel, then instead of calculating the specific values of q, we can represent it in the total value, that is total Q. And, that can be represented by bringing the fuel calorific value, mass product of the fuel and combustion efficiency $Q_{in} = m_f Q_{hv} \eta_c$.

So, based on this equation we can frame this particular equation, which is $Q_{cv}\eta_c = (AF+1)c_p(T_3 - T_2)$. Rest of the things process 3-4 is a expansion process and expansion process is isentropic in nature, you can use the corresponding equation.

Process 4-5 is your exhaust blow down. So, it is a constant volume processes, where we have work transfer as well as where we have heat transfer equations are involved. So, from this one can calculate the thermal efficiency, $\eta_t = \frac{w_{net}}{q_{in}}$ and $\eta_t = 1 - \frac{1}{r_c^{k-1}} \left[\frac{\beta^k - 1}{k(\beta - 1)} \right]$.

As you can see the efficiency calculation, apart from this particular ratio as we see in the Otto cycle, there is another term that gets added. And, this particular term involves the fuel cut off ratio β and the specific heat ratio for the combustion products. When we are going to plot thermal efficiency versus fuel cut off ratio for different compression ratio we can plot it. So, as you can see when the compression ratio increases the thermal efficiency increases, when the fuel cut off ratio increases the thermal efficiency drops.

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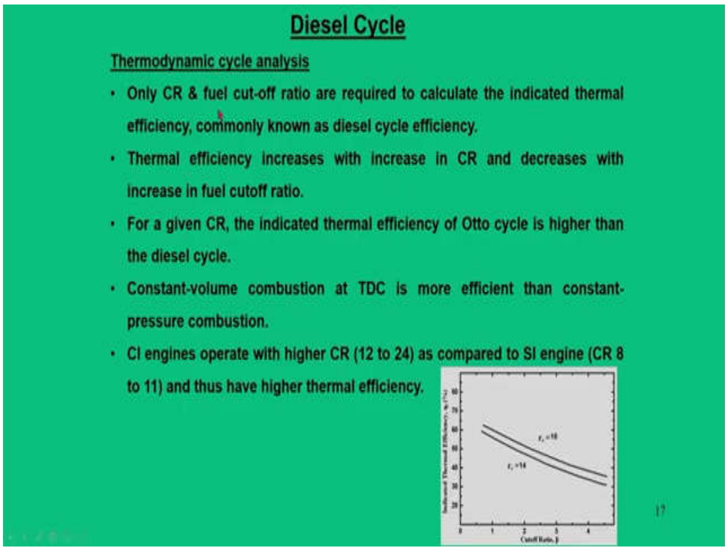

So, this is the some summary is that when you do this diesel cycle analysis, we require the compression ratio and fuel cut off ratio to calculate the diesel cycle efficiency. Thermal efficiency increases with compression ratio and decreases with increase in the fuel cut off ratio.

For a given CR compression ratio, indicated thermal efficiency of Otto cycle is higher than the diesel cycle. Thus the constant volume at combustion at TDC is more efficient than the constant pressure combustion process. CI engines operate with higher compression ratio as compared to SI engines. So, they have higher thermal efficiency.

Because, in general these CI engines operate higher compression ratio, because of this reason they have higher thermal efficiency.

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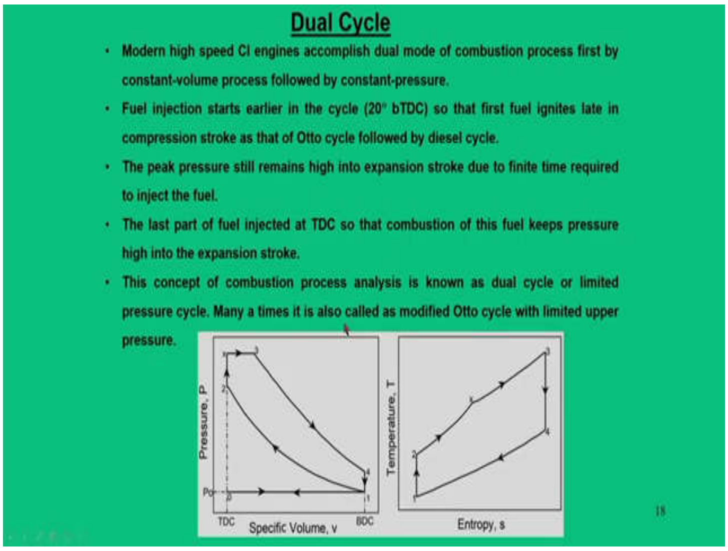

Then, we will move onto another cycle which is dual cycles. So, people think of that how we can improve the efficiency of Otto cycle at the same time; we can approach the efficiency of diesel cycle, just by doing some alteration in the heat addition process.

So, by doing so, what we can say is that instead of having constant volume processes, where there is no work output and if you replace with a constant pressure processes that means, there can be certain work output. So, in that process we will get your work output increases and thereby you can enhance the cycle efficiency of Otto cycle closer to diesel.

So, with those concept the modern high speed engines accomplish dual mode of combustion process first by a constant-volume, it is then it is followed by a constant pressure process. So, in this case the fuel injection starts earlier in the cycle; that means, 20° bTDC. So, that first fuel ignites late in the compression stroke as that of Otto cycle followed by the diesel cycle.

So, the peak pressure remains high into the expansion stroke due to finite time required to inject the fuel. And, last part of the fuel is injected at TDC so, that combustion of the fuel keeps the pressure high in the expansion stroke. That means, when the expansion stroke starts this combustion keeps the expansion pressure high.

So, we will have a larger domain of expansion stroke. So, this particular concept of combustion process is known as dual cycle, or people call it as a limited pressure cycle, many a times it is also referred as modified Otto cycles with limited upper pressure.

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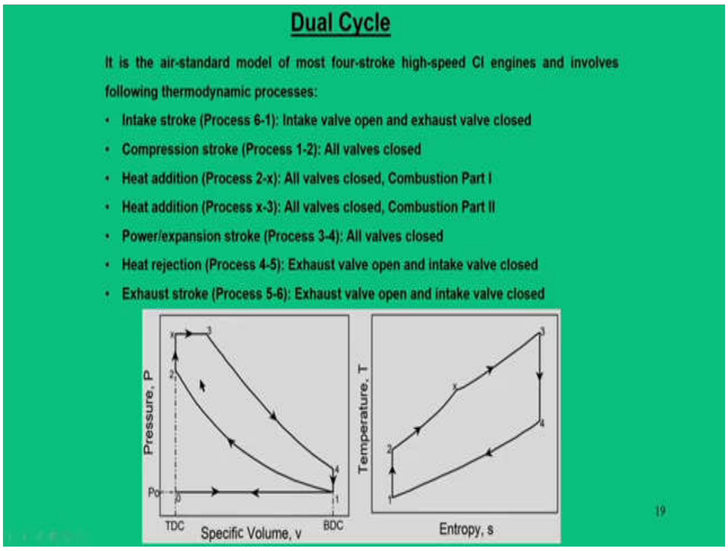

So, as you can see in this figure that in a dual cycle the change that happens is heat addition process first from 2 to x that is at a constant volume process and from x to 3 that is at constant pressure process. So, here corresponding temperature diagrams also drawn for the similar and concepts. Now, this heat addition process does with all the valves closed and we call this as a combustion Part I, combustion Part II; that means, combustion takes place in the 2 parts.

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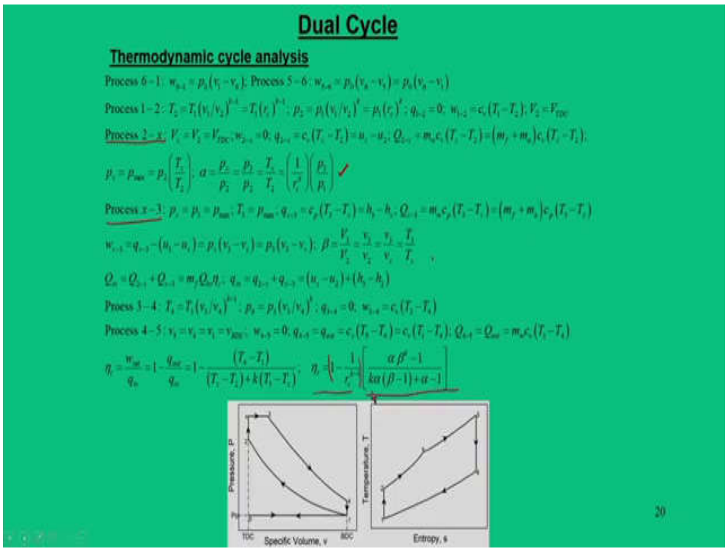

So, this here also the analysis remains simple, I do not want to go deep into this and here what we can see is that, the change that happens is work transfer that is between process x-3 and process 2-x. So, these are the two processes that is going to alter.

So, here if you can see for process x-3 there is heat transfer and also there is work transfer. Process 2-x there is only heat transfer, but there is no work transfer. So, in that process we get an additional work output through same amount of heat, being supplied in 2 modes.

So, thereby we can calculate the thermal efficiency as a function of three factors; first one is the compression ratio, other one is the fuel cutoff ratio, that is $\beta = \frac{V_3}{V_2}$ and that is also a parameter $\alpha = \frac{p_x}{p_2} = \frac{p_3}{p_2} = \frac{T_x}{T_2} = \left(\frac{1}{r_c^k}\right)\left(\frac{p_3}{p_1}\right)$. So, the thermal efficiency is calculated as $\eta_t = 1 - \frac{1}{r_c^{k-1}}\left[\frac{\alpha\beta^k - 1}{k\alpha(\beta - 1) + \alpha - 1}\right]$.

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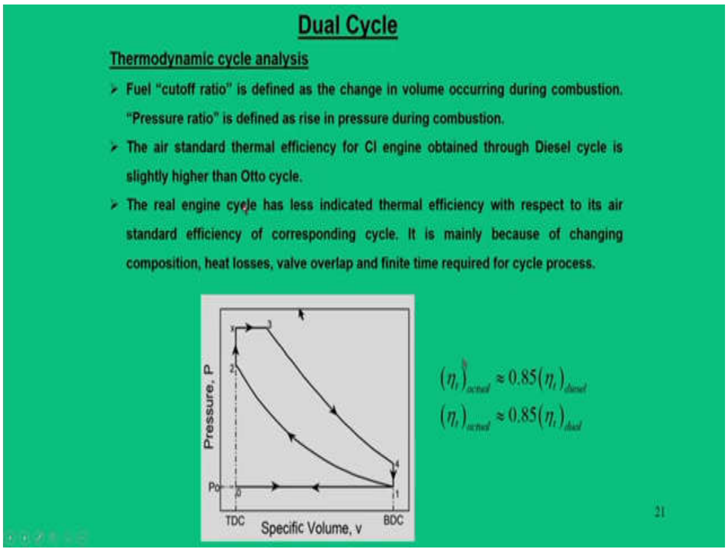

So, whatever I have explained this is shown here, one is the fuel "cutoff ratio", other is the "pressure ratio", that takes part into the calculation of thermodynamic cycle efficiency. The air standard cycle of CI engines is obtained through diesel cycle is higher than that of Otto cycle, but here we must remember the diesel cycle operate at higher compression ratio. Whereas, Otto cycles operate at, lower compression ratio; so, this is the reality.

The real engine has less indicated thermal efficiency with respect to it is air standard efficiency of corresponding cycle. So, it is mainly because of the fact that changing composition, heat losses, valve overlap, finite time required for cycle process. So, at this stage we can say that an actual cycle operating in a diesel cycle is having efficiency about 85 percent that of diesel cycle.

For example, if you calculate the cycle efficiency of a diesel cycle and take actual number because of changing composition, heat losses, valve overlap, finite time; so, the actual efficiency will be less than that, which is a which is roughly we can say about 85 percent of the corresponding diesel cycle.

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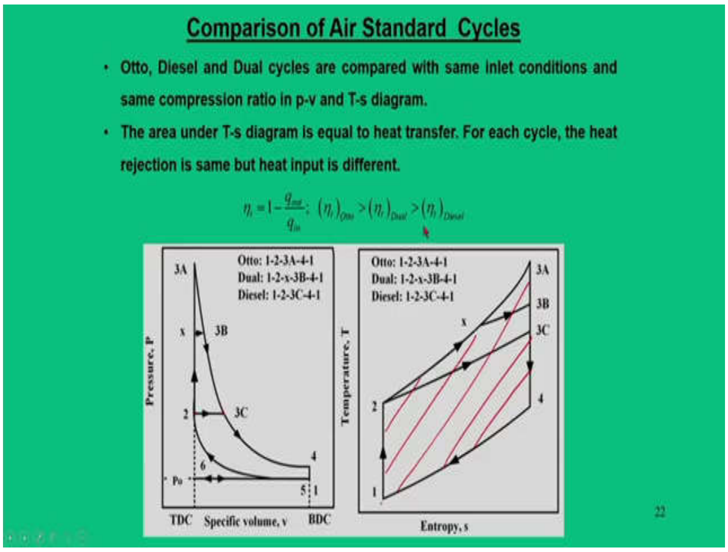

Now, next approach is to compare the air standard cycles. One way of comparing them is that for same inlet conditions and same compression ratio. So, we have pressure volume diagrams and temperature entropy diagrams. So, what we are making constant is for same inlet conditions.

So, the condition one is same for all the cycles. And, also compression ratio is same; that means, point 1 and 2 locations are same. So, as you can see here, that for same condition 1 and 2 compression ratio and same inlet conditions, first I need to draw an Otto cycle.

So, when I draw an Otto cycle for same compression ratio the Otto cycle is represented as 1-2-3-A-4. Now, with same compression ratio if I want to draw a diesel cycle, then what should I do? Then from point 2, I have to draw a constant pressure line? So, it is intersects the expansion curve 3A and 4 at 3C. So, this represents the starting point of the heat addition for diesel cycle point 2 and heat addition stops at point 3C. So, 2-3C process is heat addition process.

And, somewhere down the line if you want to represent a dual cycle, we can cut off at any point on the line 2-3A. So, here it is represented as x. So, the dual cycle is 1-2-x-3B-4-1. In similar sense we can also represent the temperature entropy diagrams.

Now, when we are actually comparing the thermal efficiency we can represent is $\eta_t = 1 - \frac{q_{out}}{q_{in}}$. So, as you can see for each cycle the heat rejection is same; that means, q_{out} is same, but heat input is different. So, heat input is different why because we can get it from the temperature entropy curve.

And, we can see for Otto cycle it is the highest, because the area under the diagram for Otto cycle is the highest. So, that is nothing, but the cycle 1-2-3A-4. So, by putting this logic or concept we can say the thermal efficiency of Otto cycle is highest. $(\eta_t)_{Otto} > (\eta_t)_{Dual} > (\eta_t)_{Diesel}$

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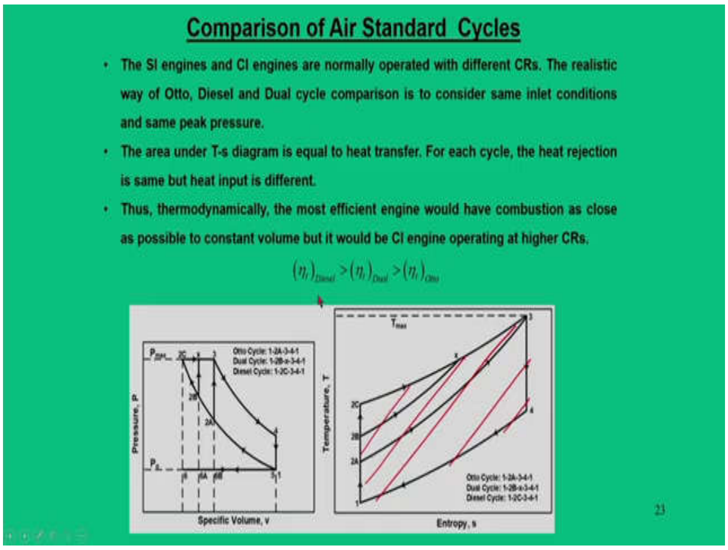

The another way of comparison is that is that based on the same inlet condition and same peak pressure and temperature, what does this mean is that normally the engines are operated in different compression ratio. So, the realistic approach for them is to consider the Otto, diesels and dual cycles for same inlet conditions and same peak pressure and temperature.

In doing so, what we are going to do is that same logic; we can first say is that we have to consider same inlet condition and same peak pressure. So, peak pressure remaining the same so, obviously it will be a diesel cycle first. So, we can first draw a diesel cycle that is 1-2C-3-4 and from this diesel cycle, we can derive the Otto cycle as 1-2A-3-4 by maintaining same peak pressure and same inlet conditions. And, somewhere we have to make a cut off to draw the dual cycle, that is 1-2B-x-3-4-1.

So, in same sense we can also calculate the heat addition in the diesel cycle is the highest that is q_{in} is highest for this particular cycle. So, based on that equation we can find for this particular conditions; that means, at different compression ratio, but same inlet condition and peak pressure diesel cycle is the most efficient one. $(\eta_t)_{Diesel} > (\eta_t)_{Dual} > (\eta_t)_{Otto}$

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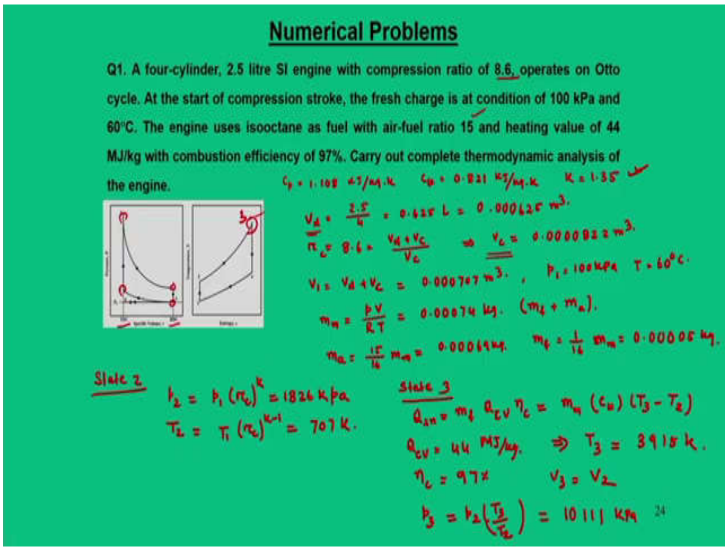

So, with this we conclude the end of the lecture. Now, let us solve some numerical problems based on these study what we have done so far. So, the first problem which I am going to discuss is consider a four-cylinder 2.5L SI engines with compression ratio 8.6 and it operates on Otto cycle. So, basically this is an Otto cycle analysis. So, you we have to use all thermodynamic equations of Otto cycles.

So, the fresh charge condition which is at the inlet condition is 100 kPa and 60°C, we also have given with air fuel ratio of 15 and the fuel calorific value of 44 MJ/kg, with combustion efficiency of 97%. So, we are asks with to carry out the complete thermodynamic analysis.

So, first thing to start this problem let us take that what c_p, c_v and γ or k value we should take. So, beginning analysis we have represented the c_p value as 1.108 kJ/kg-K, c_v we can treat it as 0.821 kJ/kg-K and k as 1.35. So, this is in contrast with air, because air has this number little bit different. And, why we are we going to use this, these are values used for combustion products and with this value if you do the analysis, we will get a number which is close to the actual cycle.

So, to start the problem first thing, for all the state point we have to find out the pressure temperature and volume. So, for the volume calculations we have to start with the BDC and TDC.

$$V_d = \frac{2.5}{4} = 0.625\text{L} = 0.000625\text{m}^3$$
$$r_c = 8.6 = \frac{V_d + V_c}{V_c} \Rightarrow V_c = 0.0000822\text{m}^3$$
$$V_1 = V_d + V_c = 0.000707\text{m}^3, p_1 = 100\text{kPa}, T_1 = 60°\text{C}=333\text{K}$$
$$m_m = \frac{pV}{RT} = 0.00074kg = m_f + m_a, AF = 15 \Rightarrow m_a = \frac{15}{16}m_m = 0.00069kg; m_f = \frac{1}{16}m_m = 0.00005kg$$

So, for state 2, compression process $p_2 = p_1\left(r_c\right)^k = 1826\text{kPa}$, $T_2 = T_1\left(r_c\right)^{k-1} = 707K$. Then for process 3 $Q_{in} = m_f Q_{hv} \eta_c = m_m c_p \left(T_3 - T_2\right) \Rightarrow T_3 = 3915K$

$$v_3 = v_2 \Rightarrow p_3 = p_2\left(\frac{T_3}{T_2}\right) = 10111\text{kPa}$$

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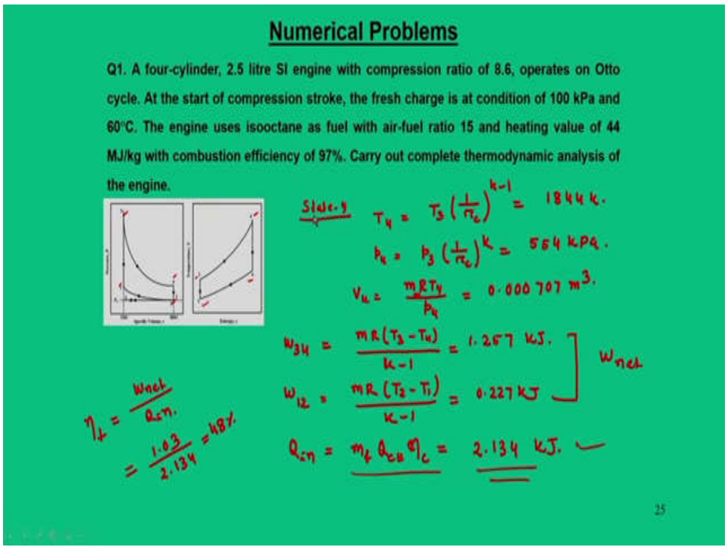

For state 4 $T_4 = T_3 \left(\frac{1}{r_c}\right)^{k-1} = 1844K; p_4 = p_3 \left(\frac{1}{r_c}\right)^{k} = 554\text{kPa}; V_4 = \frac{m_m R T_4}{p_4} = 0.000707\text{m}^3$

So, with this we calculated all the state points, properties values at all the states points. Now, we have to revisit work and heat calculations.

$$w_{34} = \frac{mR(T_3 - T_4)}{k-1} = 1.257\text{kJ}; w_{12} = \frac{mR(T_2 - T_1)}{k-1} = 0.227\text{kJ}$$

$$Q_{in} = m_f Q_{hv} \eta_c = 2.134\text{kJ};\ \eta_t = \frac{w_{net}}{q_{in}} = \frac{w_{34} - w_{12}}{q_{in}} = 48\%$$

This is all about the thermodynamic analysis of an SI engine.

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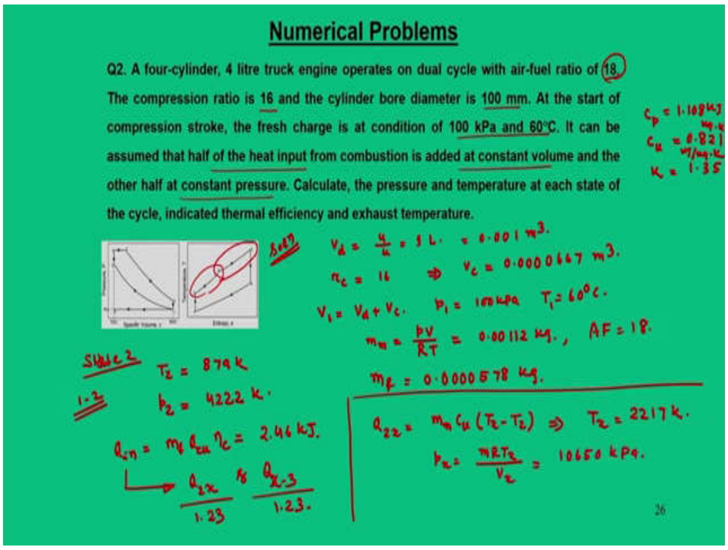

And, in the similar logic we have another problem that is four-cylinder, 4 L truck engine that operates in a dual cycle here the air fuel ratio is 18, compression ratio is 16, and bore diameter is 100 mm, in intake condition is 100 kPa and 60 °C, but the entire idea is that heat addition process. So, total heat addition process is divided into two halves. So, the first half is at constant volume and second half is at constant pressure and both amount is equal.

So, we need to calculate the pressure temperature and at each states. So, the solution of the problem we can start with similar way as we did it.

$$V_d = \frac{4}{4} = 1\text{L} = 0.001\text{m}^3, r_c = 16 \Rightarrow V_c = 0.0000667\text{m}^3$$

$$V_1 = V_d + V_c; p_1 = 100\text{kPa}, T_1 = 60°\text{C}$$

$$m_m = \frac{pV}{RT} = 0.00112kg, AF = 15 \Rightarrow m_f = 0.0000578kg$$

State-2:- $T_2 = 879\text{K}; p_2 = 4222\text{kPa}$

$$Q_{in} = m_f Q_{hv} \eta_c = 2.46\text{kJ; } \text{Q}_{2x} = Q_{x3} = 2.46/2 = 1.23\text{kJ}$$

$$Q_{2-x} = m_m c_v \left(T_x - T_2\right) \Rightarrow T_x = 2217\text{K}; p_x = \frac{mRT_x}{V_x} = 10650\text{kPa}$$

$$Q_{x-3} = m_m c_p \left(T_3 - T_x\right) \Rightarrow T_3 = 3208\text{K}; V_3 = \frac{mRT_3}{p_3} = 0.000097\text{m}^3$$

State-4:- $v_4 = v_1 = 0.0010667\text{m}^3; T_4 = T_3 \left(v_3/v_4\right)^{k-1} = 1386\text{K}; p_4 = p_3 \left(v_3/v_4\right)^k = 418\text{kPa}$

State-5:- $T_5 = T_4 \left(p_5/p_4\right)^{\frac{k-1}{k}} = 957\text{K}$

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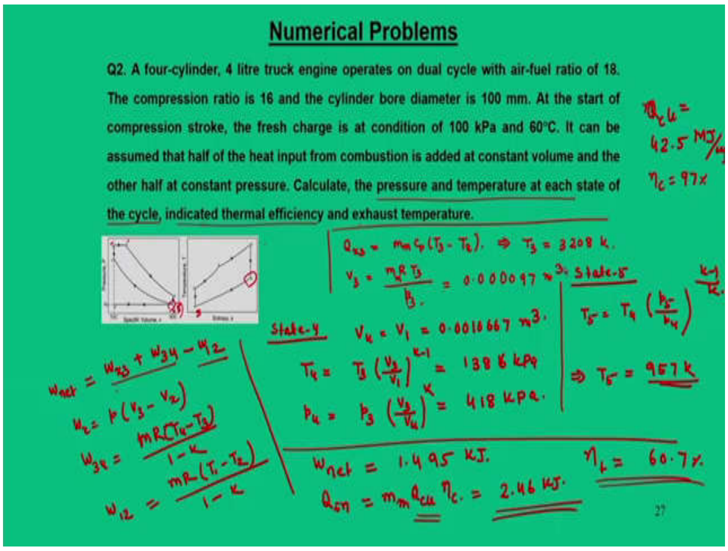

So, this is all about the thermodynamic state pressure and temperature of the cycle.

So, now main job is to calculate indicated thermal efficiency.

$$w_{net} = w_{x3} + w_{34} - w_{12} = p_x\left(v_3 - v_x\right) + \frac{mR\left(T_4 - T_3\right)}{1-k} + \frac{mR\left(T_1 - T_2\right)}{1-k} = 1.495\text{kJ}$$

$$\text{Q}_{in} = m_f Q_{hv} \eta_c = m_f \times 42.5 \times 10^6 \times 0.97 = 2.46\text{kJ}$$

$$\eta_t = \frac{w_{net}}{q_{in}} = 60.7\%$$

So, the analysis is very lengthy, but due to time constraints, I have explained it in very briefly. I hope you can understand this problem and for all these equations you can refer the slides in which this equations are given elaborately. So, with this I conclude this talk for today.

Thank you for your attention.