Applied Thermodynamics for Engineers

Dipankar N. Basu
Department of Mechanical Engineering
Indian Institute of Technology – Guwahati

Lecture – 14
Fuel-air cycle

Good morning everyone into the week number 5 where our topic of discussion is real cycles for reciprocating engines. In the previous week we talked about the gas power cycles or gas standard cycles, which of course cycles can be device for conceptualize both for reciprocating and rotary engines. But we primarily focus on the reciprocating engines or to be more precise automobile devices or cycles suitable for automobile engine.

The cycle that you have discussed among them the Carnot cycle or Ericsson cycle etc can be conceptualized for both closed and open systems and also for both rotary and reciprocating kind of applications. But the two cycles which are primary focus of a discussion i.e., Otto and diesel cycles have their application exclusively in the field of reciprocating engines. And therefore we can safely say that in the previous week we have talked about the ideal cycle suitable for reciprocating engines.

Now, we have seen that the Otto cycle which is a cycle suitable for SI engines or Spark ignition engines that conceptualizes or visualises the combustion process in the form of constant volume heat addition. Whereas the Diesel cycle, which is the ideal cycle for compression ignition engine or diesel engines that visualises combustion as a constant pressure heat addition process.

That particular heat addition process is the only difference between the schematic representation of this cycles that is one with heated at constant volume other at constant pressure. And accordingly, we have done their analysis. We have drawn cycle diagrams on *Pv* and *Ts* planes to see or to directly compare their nature. We have developed expression for their efficiencies, work output and other performance parameters and we also have compared them in terms of various parameters.

And from there what is evolved is that the most important parameter governing the engine performance is a compression ratio. Generally, with the increase in compression ratio efficiency of both Otto and Diesel cycle increases quite sharply. Of course, that settles down after a bit in general with increasing compression ratio it increases. And therefore it is always logical to have higher compression ratio as much as we can considering the practical limit.

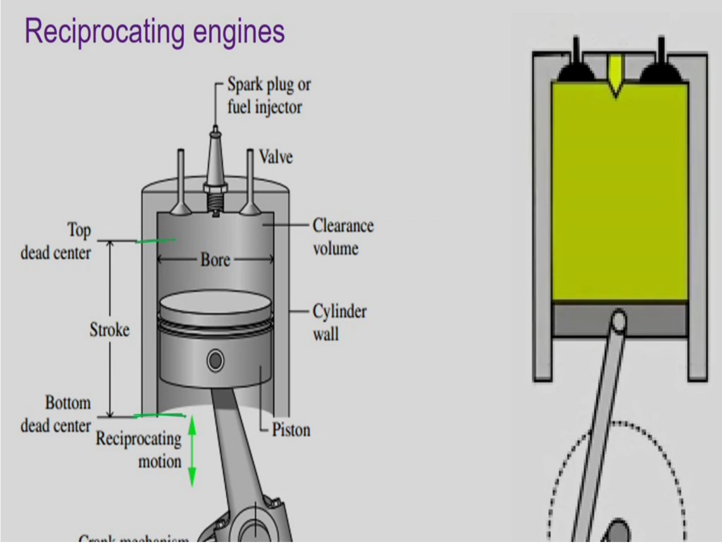
Practical limit can be the self ignition temperature for Otto cycle and the practicable space and weight constant corresponding to the diesel cycle. But as we have mentioned there also that these cycles are only idealisation i.e., they provide some kind of upper limit for the operation of the systems. But there are several kinds of practical losses that is possible in those cycles which reduces the actual output from the cycles and therefore can significantly affect the thermal efficiency.

It is very much possible that suppose I give you SI engine and using the performance parameters of that you do an analysis assuming and ideal Otto cycle and you are finding the corresponding thermal efficiency to be around 70%. However, if we run the actual engine in experimental condition, you are getting only efficiency of 30 to 35%. So, there is a huge difference in the theoretical prediction and practically obtained values.

So, while the ideal cycle state of some kind of upper limit of operation but if you are looking to get a realistic prediction about the performance of you looking to design such an engine then those cycle can give you far of results. And therefore, it is required to take into consideration all the practical losses that can be present in the engine operation so that we can get a more realistic estimation of the performance.

And that is what you are going to focus in this particular week. Of course, you are going to talk about the cycles, but we shall be talking more about the different kind of losses and other effects that can be present in the cycle and accordingly how we can incorporate them to get a more realistic estimation about the cycle performance. So, we are going to talk about the real cycles or factors affecting the performance of the ideal cycle in case of reciprocating engines primarily the Otto and Diesel cycle.

(Refer Slide Time: 05:01)



These are some diagrams which have shown in the previous week itself. This is standard reciprocating engine for we know that the piston reciprocates within the cylinder. Generally, the top most position is referred as the top dead centre and the bottom most position refers to the bottom dead centre. This is the top dead centre position and this is the bottom dead centre position and the distance between the dead centres is called the stroke volume or displacement volume.

The stroke volume is probably the first parameter that is always specified in terms of any engine specification. Like whenever you are talk about say two wheelers, we talk about 150 cc, 125 cc, or very heavy two wheelers we talk about 220 cc. The cc is nothing but the stroke volume or displacement volume. Similarly, for smaller cars we talk about 1000 cc engines for sedans we talk about 1.4 litre, 1.5 litre engine more than 2 litre engines for SUV's they are stroke volume and nothing else.

The volume left on top of the top dead centre is the clearance volume. So, volume and clearance volume together give the total volume of the cylinder. And the ratio of total volume to the clearance volume is the compression ratio, which we have already identified as a most important parameter even in case of ideal cycles, and the same is true for really engine as well.

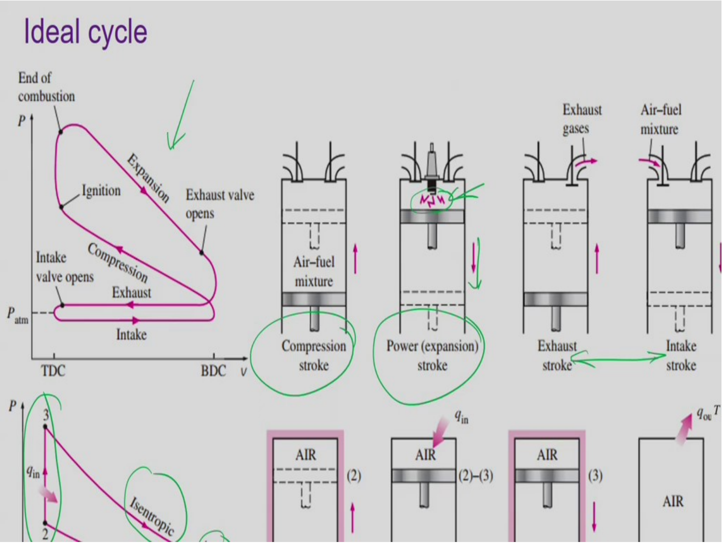
We have the crankshaft here and the crank mechanism. The crankshaft rotates and the corresponding rotary mechanism is converted to the reciprocating motion of the piston by the crank assembly of crank mechanism. 180 ^{0} rotation of the crankshaft response to 1 complete stroke of the piston. And when the piston is operating this is the situation that is generally get **(Video start: 06:45)** and this animation show in the previous slide itself but somehow it malfunctioned.

Just look at this point it start this is suction, now is compression, this is the power stroke and now it is the exhaust. In suction look how the inlet valve is opening. Now both valves are closed in combustion exhaust function and now it is opening again during exhaust stroke. If, I run it again, so, this is compression, this expansion and exhaust valve is open to exhaust stroke. As soon as the exhaust valve closes the inlet valve open again to facilitate the inlet or suction process.

So, this way the cycle keeps on operating generally it operates at a very high speed is very common to have cycles operating at something like 3000 rpm. 3000 rpm means you are talking about 3000 full revolution of the crankshaft in one minute. **(Video Ends: 07:35)** So, 3000 revolutions per minute is equal to 3000/60 revolutions per second. So, that means you are talking about 50 revolutions per second. That means in one second the crankshaft is having 50 revolution and if you are talking about a four-stroke engine, we know that two revolutions of the crank shaft or full rotation of the crankshaft corresponds to 4 stroke that is one full cycle.

So, 50 revolutions per second leads to 50/2 that is 25 cycles per second, so that is a very high speed that you are talking about. And this is only the way all 4 stroke engines work.

(Refer Slide Time: 08:31)


Ideal cycle

This is the ideal cycle again a slide repeated from the previous week's work. This is the actual operation where we have the compression stroke to start with, then we have the power stroke during which the piston is moving down from TDC to BDC following a spark which is produced here. This a 4 stroke spark ignition engine, then we have the final exhaust stroke. During the exhaust stroke the exhaust valve is open and at the end of exhaust stroke as soon as all the exhaust gases are gone out, we have the suction stroke starting again during the piston is moving down making more space for the new fuel air mixture to come in. And this is the corresponding representation on a *Pv* plane. However, this is an actual cycle and this conceptualized in the form of an ideal cycle, the Otto cycle just like this, where we have assumed several air standard assumptions and following which we get this representation.

In the Otto cycle we have two isentropic processes and two constant volume process. The heat addition is a constant volume similarly the heat reaction is also at constant volume in between we have an isentropic compression process and an isentropic expansion process. So, this isentropic compression process is represented of this compression stroke. This isentropic expansion process, the representative of this power or expansion stroke, the spark which is happening here that is being conceptualized in the form of this constant volume heat addition.

And the exhaust and inlet process together has been coupled into this one constant volume heat rejection process. This is as per the air standard assumption. This is the corresponding representation *Ts* plane. It is generally ideally suited to represent each of circles on both *Pv* and *Ts* plane because area enclosed by the diagram and the *Pv* plane is a direct representation of the total work done. Whereas the area enclosed by the cycle on *Ts* plane is a direct representation of the total heat introduction or net heat supplied to the system. And as we know that, as per the first law of thermodynamics, for a closed system undergoing a cycle we know that:

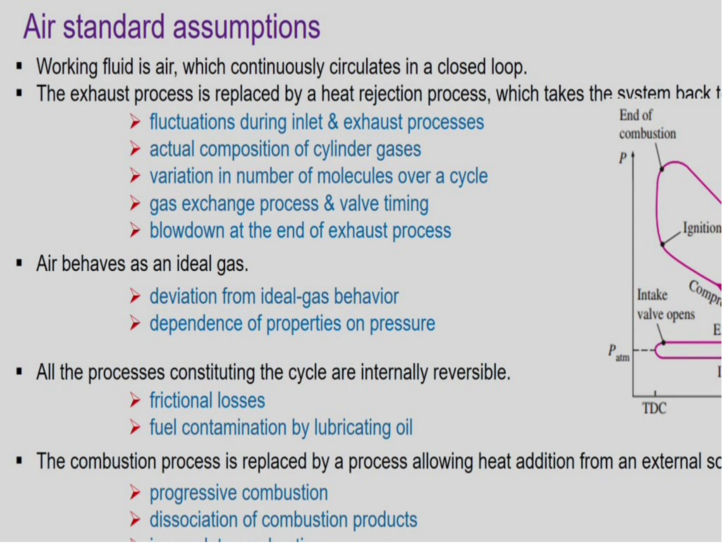
$$\oint \delta q = \oint \delta w$$

from where we can easily say that the net heat supplied to the system has to be equal to net work output that you have obtained from the system.

So, any of the diagrams theoretical is sufficient to provide you on the performance parameters in the form of work output or efficiency but generally is a logical to represent both so that we also get ideas about the maximum pressure temperature etc for the cycle.

Now this is an ideal cycle, the Otto cycle similarly can also draw the Diesel cycle as we have already discussed in the previous week. Now let us quickly look at the air standard assumptions using which we have actually plotted the ideal cycle.

(Refer Slide Time: 11:27)



Air standard assumptions

- Working fluid is air, which continuously circulates in a closed loop.
- The exhaust process is replaced by a heat rejection process, which takes the system back t
  - fluctuations during inlet & exhaust processes
  - actual composition of cylinder gases
  - variation in number of molecules over a cycle
  - gas exchange process & valve timing
  - blowdown at the end of exhaust process
- Air behaves as an ideal gas.
  - deviation from ideal-gas behavior
  - dependence of properties on pressure
- All the processes constituting the cycle are internally reversible.
  - frictional losses
  - fuel contamination by lubricating oil
- The combustion process is replaced by a process allowing heat addition from an external sc
  - progressive combustion
  - dissociation of combustion products

So, this is the first air standard assumption. If you remember I ensure that you remember the air standard assumption. I am writing the same assumptions here, is a copy of the same slide, but I have reordered these assumptions and to suite some requirements. So, one assumption was the first assumption, the working medium is air and we are assuming it to be a closed system, so that we can assume the same quantity of air is continuously circulating with closed loop.

And we are also assuming that there is no inlet and exhaust process that these two processes together are replaced by a heat rejection process generally constant volume with reaction in case of both Otto and Diesel cycles or even in case of dual cycles. But that is different in case of the Stirling and Ericsson cycles because they generally have heat interactions in the form of isothermal processes. So, in all the cycles that you have studied starting from Carnot cycle upto Ericsson cycle in all the cases, we are assuming that there are no inlet process and exhaust process weather those two has been clubbed into single heat rejection process, which is at constant volume in Otto, Diesel and dual combustion cycle. Whereas that is isothermal in case of Carnot, Stirling and Ericsson cycles.

So, the objective of the heat rejection process is to take the system back to the initial state means, to take the system back to the state at the point of commencement of the compression process. So, these two are the idealizations. But what are the things that you are losing because of this? What are the practical effect or the practical factors that actually do not obey this particular thing during real operation?

One thing that you are neglecting is the fluctuation during the inlet and exhaust processes. As the fuel air mixture goes in, during the inlet process, the pressure inside the system that keeps on fluctuating. In fact, if you just think about the wall generally outside you have atmospheric pressure or slightly higher than atmospheric pressure. Whereas in the inside the cylinder as piston starts to go down in the inlet process the pressure falls below atmospheric thereby creating a slight vacuum.

So, that the fuel air mixture can flow easily into the cylinder, but as soon as it starts flowing into this, the pressure inside the cylinder starts to increase and they by causing some kind of fluctuations to the valve. The same is applicable during the exhaust process as well. So, both pressure and temperature that keeps fluctuating may not be by a very significant margin, but that is very much possible which can lead to certain issues related to the design of the valves.

A bigger problem or bigger issue that you are completely eliminating is the actual composition of cylinder gases. As we are assuming the working medium to be air and it is continuous to be air during all the four processes in the ideal cycle. But if you think about the practical cycle, for an SI engine operation, it is a fuel air mixture that you supply. So, during the inlet process and also during the compression process your working medium is actually a mixture of hydrocarbon fuel and air. So, it is mixtures of several gasses like in air predominantly we have oxygen and nitrogen.

Whereas in the fuel we can have several kinds of hydrocarbons both, we can have methane and similar kind of things or we can have heavy hydrocarbons as well in the group of ethylene or even going to even heavier hydrocarbons in the group of benzene. So, we are completely neglecting the actual composition. And if you think about the post combustion scenario because of combustion the hydrocarbon gets converted to a mixture of carbon dioxide and water vapour.

And in case of incomplete combustion we can also have carbon monoxide. Sometimes just something we call soot, that is dry carbon only minute carbon particles, if the fuels content something like sulphur; we may have oxides of sulphur as well. So is again a mixture of gases and which is actually different from what we had during the inlet and compression processes. This composition of cylinder gases that has been completely eliminated during the ideal cycle. The variation in the number of molecules over a cycle that also we are neglecting.

The implication of this shall be discussing shortly, but as the number of molecules changes the pressure inside the cylinder also changes proportionately and number of molecules definitely changed because of the chemical reactions. That is also there we are neglecting we are completely eliminating the chemical reaction. You are considering it to be a fixed mass of air. Then the gas exchange process and the valve timing. Now, what do you mean by gas exchange process?

I am just asking you the same diagram for the actual cycle. Here some marks are shown. As you can see though the exhaust process supposed to start with the piston reaches the bottom dead centre someone here, the exhaust valve has opened at this particular point only. Similarly, the inlet valves opens somewhere here. And the ignition process spark were ideally we are supposed to provide the spark once the piston reaches the top dead centre some here, but spark has been provided a bit below that.

This way these things are called the valve timings i.e., exactly when to open the inlet valve and when to close it. And when to open the exhaust valve and when to close it and when to provide the spark in case of spark ignition engine. When to spray the fuel in case of CI engines these are timings are very important. We shall we discuss in detail about the valve timing diagram in the next lecture. But for the moment you can take that though ideally you should open and close all the valves at the dead centres, but practical speaking considering the finite time requirement for this valves opening and a few other factors, we generally open this was a bit earlier were there ideally should open and also close them a bit later than ideally it should close. The exhaust valve should open at this particular location, but it is opening earlier than this. Similarly, it should close when the piston reaches the top dead centre but sometimes, we close it well beyond may be somewhere here. So, we are making the opening of the valve earlier than ideally required and closing them much delayed than ideally required.

These things related to the valve timing and because of that we can have a gas exchange process. Just focus on this portion. If the inlet valve open as it is shown and the exhaust valve opens at this particular position, then there is a small period of time or small window of time during this both valves are open. So, the phase charge that comes in it is possible for that to get mixed with the exhaust gases and also some phase charge may get lost. This is sometimes referred the gas exchange process. I shall we mentioning this bit more in the next lecture and this is a kind of loss if we cannot control them properly.

Similarly blow down at the end of the exhaust process, to open the valve generally all these valves in commonly open inward. So, whenever we are trying to open the valve, we have opened the valve against the pressure of the exhaust. What I am trying to say that, if this is a cylinder and this is exhaust valve now when the piston starts to come up in this direction it is actually forcing the gases against exhaust valve.

And therefore, if you want to open the exhaust valve download then we have open it against this increasing pressure of the exhaust gases. So that will require some additional amount of work input. To avoid that what we do, before the piston reaches this bottom dead centre position and going back to the diagram this position. We will open the valves from here. So, as the piston is going down, there is no additional pressure acting on the valve on the inner surface of the valve was easier to open.

But as soon as we open the exhaust valve that is virtually the end of the expansion process. So, this particular thing is referred as the blow down or exhaust valve blow down i.e., again a kind of loss. These actual losses we shall be talking more in the next lecture. So, these all are losses that is happening because of the first two assumptions, all these factors we are neglecting.

Similarly, we are all assuming air to be an ideal gas. But practically, you know that it can deviate from ideal gas behaviour under certain temperature conditions, so that can lead to some loss of information. Also, assuming air to be ideal gas we are considering properties like internal energy or enthalpy or specific heat to be function of temperature alone. But certain situations particularly when you are talking on very high pressure these properties can significantly vary with the pressure as well.

So, that pressure dependence we are taking out of the picture and therefore we may again lose some significant information. We are assuming all the process constituting the cycle to internally irreversible so we are neglecting the frictional losses. But friction can always be very significant factor and this is a loss we can never avoid infact it can be significant portion of the difference that can exist between ideal cycle efficiency and actual cycle efficiency. In practical engines in order to avoid to reduce the friction we add lubricating oil and this lubricating oil can get mixed up with the fuel by changing a property of the fuel itself, that contamination part or contamination effect also we are neglecting. Then we are neglecting the combustion process altogether. The combustion process replaced by process allowing the heat addition from an external source. Like in case of Otto cycle we are assuming combustion to take place at constant volume heat addition in case of diesel cycle at constant pressure heat addition.

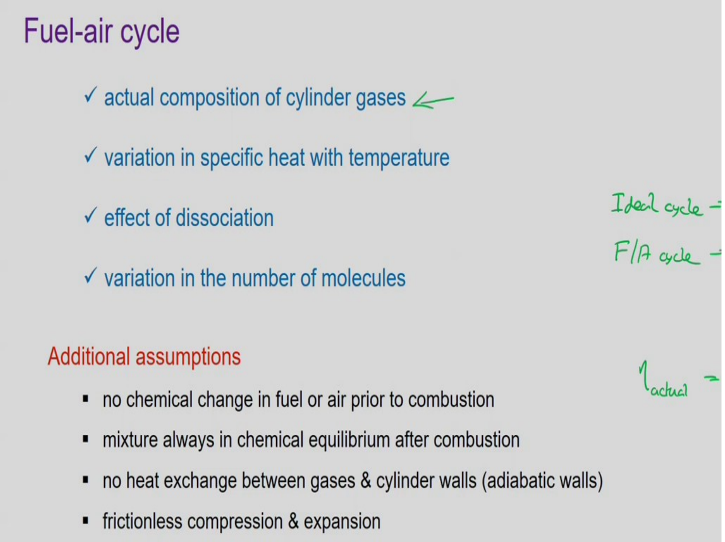
Whereas in Stirling and Ericsson cycles we are assuming this to be isothermal heat addition. And this combustion process, therefore, we are completely eliminating but that definitely has a significant impact in the actual cycle operation. Because like in case SI engine combustion to be instantaneous. Practically it requires the finite amount of time and as the combustion progress there is a gradual change in the composition of the gases. And the dissociation of combustion products when it goes to very high temperature the products of combustion and disassociate back to the original products or original reactants that is called disassociation.

That can have significant impact we shall we talking about this one also shortly. It is very much possible to have incomplete combustion particle in case of CI engines. The hydrocarbon that you are supplying instead of burning complete to form carbon dioxide, that can form carbon monoxide of some intermittent hydrocarbons leading to incomplete combustion and which can affect the thermal efficiency as well. So, unless we properly simulate the combustion process, we cannot take care of all these factors.

And also heat transfer to the cylinder walls can be significant factor in reducing the temperature level of the gases. Though we can assume always a cylinder walls to be adiabatic or whether there is no heat loss to the cylinder walls, but such kind of heat leakage particularly considering that the temperature in such cases can be in the range of 1500 to 2000 K or even higher such leakages are unavoidable.

And finally we are assuming the specific heat to be constant that is probably at least from theoretical that is probably the biggest source of error that we can encounter. Because specific heat, a strong function of temperature increases rapidly with increase in temperature. So, because of each of the air standard assumptions we are losing something I should say we are eliminating some other external factors which lead to the deviation between the actual cycle prediction or actual cycle performance and ideal cycle prediction. And therefore, to add if not possible for with all at least some of them can still be taken care of theoretically and for that we consider something as the fuel-air cycle.

(Refer Slide Time: 24:00)



Fuel-air cycle

- ✓ actual composition of cylinder gases ←
- ✓ variation in specific heat with temperature
- ✓ effect of dissociation
- ✓ variation in the number of molecules

Ideal cycle –

F/A cycle –

Additional assumptions

- no chemical change in fuel or air prior to combustion
- mixture always in chemical equilibrium after combustion
- no heat exchange between gases & cylinder walls (adiabatic walls)
- frictionless compression & expansion

η_{actual} =

Fuel-air cycle is a modification of the ideal cycles where among all the factors that I just listed in the previous slide not all but four of the factors we are considering or we are adding to the actual cycle. Factor number 1, the actual composition of cylinder gases: in fuel-air cycle just I repeat we are adding these factors to the ideal cycle which we have already discussed. Like the Otto and Diesel cycle that we have discussed in the previous week.

On these ideal cycles we are adding four factors to have the fuel-air cycle and the first factor is the actual composition of cylinder gases. So, we are going to consider the true composition of the mixture during the suction and compression stroke of a SI engine. Whereas we shall be talking about the mixture of combustion products after combustion process. In case of CI engines however you do not need to consider the mixture. Because in CI engine, we know that it is only air which is supplied only suction and compression process that is only air.

So, there will be no change in case of CI engine or diesel cycle during those two processes. However, during the post combustion processes we need to consider the composition of combustion gases. Then the second factor that we consider is the variation of specific heat temperature. Heat probably has the most significant effect on cycle performance, then the effect of dissociation i.e., the reversible reaction or I should say the reverse reaction that can take place at a very high temperature that will be taken care of.

And the variation in the number of molecules that can also have significant effect and we shall be seeing shortly. So, these are the four factors that you are going to consider in case of fuel-air cycles. These are the four cycles you are going to add on the ideal cycles to get the fuel-air cycle. Of course, in fuel-air cycle assuming several factors like no chemical change in fuel or air composition prior to combustion. So chemical reaction will start only at the point of combustion but in true engine, we may have some pre combustion reactions with change in temperature, those are neglected here. We are talking about any change in chemical composition just from the point where combustion is initiated. Then mixture always in chemical equilibrium after combustion that is an important assumption. Then we are assuming no heat exchange between the gases and cylinder walls. We have seen that this can have significant effect those kinds of heat leakages that we are neglecting assuming adiabatic walls.

Then friction is also neglected, frictionless compression and expansion processes are assumed. Then negligible fluid motion inside the cylinder. The fluid itself can move inside the cylinder because of the because of such fast motion resistance, but that fluid motion corresponding kinetic energy of the fluid that you are neglecting. We are assuming the fuel to be completely vaporized and perfectly mixed with air during a supply process.

So, as we are supplying fuel, we are assuming that the liquid fuel that is gasoline has completely vaporized and we are successful in preparing a perfect mixture of this fuel and air and then only is supplied to this. Similarly, you are also assuming combustion to instantaneous that is at constant volume. Of course, these last two points are applicable only for SI engine-based cycles. Because in CI engine-based cycles we have combustion at constant pressure and also supply only air.

So, this last two points are not applicable for CI engines only for SI engines. So now let us consider all these four factors one after the other. But before that I would just like to mention that in case of air standard cycles are ideal cycles, which we have discussed there. What is the most important parameter that you have identified that can affect the ideal cycle performance? That is a compression ratio and that is probably the only factor that we can consider. In case of CI engines we can add the cut-off ratio to this.

The fuel-air cycle is the first effort where we are considering the actual composition of gases. That is the nature of the fuel, the nature of fuel air mixture that comes into play. And therefore, we can analyse the second most important factor in actual engine performance that is the fuel air ratio. Fuel air ratio is:

$$^{F}/_{A}\, ratio = \frac{\dot{m}_f}{\dot{m}_a}$$

which nothing but the mass of fuel divided by mass of air in a mixture or we can say the mass flow rate of fuel by mass flow rate of air. This mass of fuel and air present in a mixture is represented by this fuel air ratio.

Sometimes we talk about the other thing also that is air fuel ratio. Air fuel ratio is nothing, but just the opposite of this mass of air by mass of fuel present in a mixture.

$$^{A}/_{F}\, ratio = \frac{\dot{m}_a}{\dot{m}_f}$$

So, after compression ratio, this is probably the second most important factor, but as we are assuming the air to be the working medium in case of the ideal cycle the fuel air mixture or fuel air ratio never comes into consideration in case ofiIdeal cycles, but that we can analyse in this fuel-air cycles.

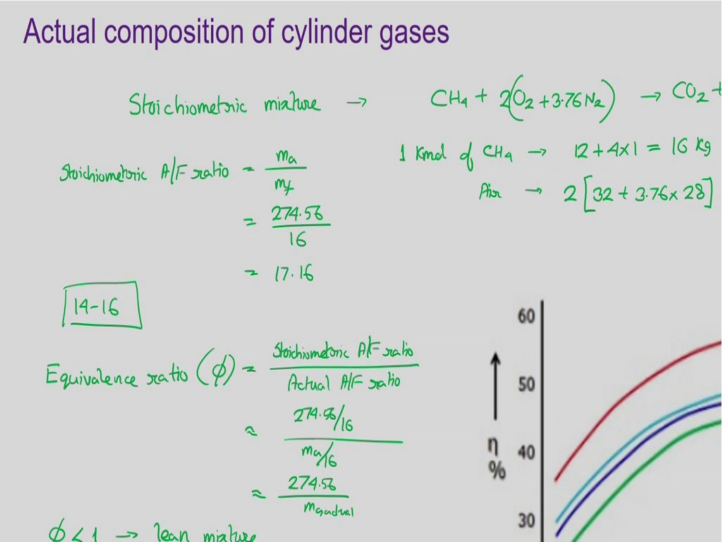
And before moving any further I should mention why you are talking so much for the fuel air cycle because as a just mentioned that efficiency predicted by the ideal cycle can be far off from the actual cycle efficiency. However commonly we have seen that the efficiency of the actual cycle can be quite close to the efficiency predicted by the fuel-air cycle. It can well be in the range of 85 to 90% of the efficiency created by the fuel-air cycle. And therefore, it is very important to have a complete understanding of the fuel-air cycle.

And of course, you are not going to develop an expression for efficiency rather we shall be focusing on identifying the effect of all these four factors. And once we know that we can properly draw the corresponding modified cycle. Then we can calculate the efficiency from there and that efficiency be quite close to what we can get in practice. Why there is still some difference? That is because this assumptions like we are neglecting any kind of frictional losses, we are neglecting any kind of heat leakage we are neglecting the chemical non-equilibrium that can be present in the system.

All this can have some 10-15 % influence which reduces actual cycle efficiency, even further to the fuel air efficiency. Also using the fuel air assumption, we can get a correct estimate of the maximum pressure and temperature that can exist in a cycle. The pressure temperature ranges created by ideal cycle again can be far off than the real operation. But fuel-air cycle gives a quite reasonable estimate about the maximum pressure and temperature of the cycles and also a quite reasonable estimate about the role played by different parameters on the cycle performance.

And therefore, for design and construction point of view, fuel-air cycle probably is more important. But of course, that cannot undermine the ideal cycle, because it is ideal cycle only which allowed us to conceptualization diagram. We are creating a fuel-air cycle by adding these four factors on the ideal cycle. So, the ideal cycle comes first then we are adding this factor to get a more realistic estimation in the form of fuel-air cycle, and on that once we can add the factors that friction etc then we get the actual cycle operation.

(Refer Slide Time: 31:43)



Actual composition of cylinder gases

Stoichiometric mixture → $CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 +$

1 Kmol of CH_4 → $12 + 4 \times 1 = 16$ kg

Air → $2[32 + 3.76 \times 28]$

$$\text{Stoichiometric A/F ratio} = \frac{m_a}{m_f} = \frac{274.56}{16} = 17.16$$

$\boxed{14\text{-}16}$

$$\text{Equivalence ratio } (\phi) = \frac{\text{Stoichiometric A/F ratio}}{\text{Actual A/F ratio}} \approx \frac{274.56/16}{m_a/16} \approx \frac{274.56}{m_{a,actual}}$$

$\phi < 1$ → lean mixture

So, the first factor is the actual composition of cylinder gases. The actual composition of mixture as we have already mention that can change significantly. And as that changes, corresponding property of the mixture can also change. Like whatever may be the property of air let us say, if we talk about the specific heat, whatever may be the specific heat of air, mixture of carbon dioxide and water vapour may have a very much different value of the specific heat. So, we need to know the proper composition of the cylinder gases.

And in this conjunction, I would like to introduce one term that is known as the stoichiometric ratio or stoichiometric mixture. The stoichiometric mixture refers to a mixture of fuel and air which contains exactly the same amount of air that is required for its complete combustion. I repeat, a stoichiometric mixture refers to a mixture of fuel and air which contains exactly the same amount of air precisely the same amount of air required to have complete combustion of that quantity of fuel.

Like if we consider simple chemical reaction using methane as a fuel. So if we have:

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

it is a complete chemical reaction. And it is complete in all sense or not in all sense. Because in practical engine to have the combustion of methane we cannot supply pure oxygen supply air and air can be visualised to be a combination of oxygen and nitrogen. And generally, by volume nitrogen is 3.76 times of oxygen. So, whenever you are supplying any 1 mole of oxygen, 3.76 moles of nitrogen also come with that. And therefore, to complete this reaction we also need to have this 3.76 moles of nitrogen on the product side.

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + (2 \times 3.76)N_2$$

It is an inert kind of gas. The weight comes in it just goes out the same way. Now it is a complete chemical reaction then to have combustion complete combustion of 1 mole of methane how much mole of air we require? We can have a very simple calculation. So 1 mole of methane, if we take rough values then we can calculate the mass of 1 mole of methane to be; let me say 1 kilo mole I should write, this is a common SI symbol so it is carbon is 12 plus hydrogen is we can just approximate values you want so it is equal to 16 kgs 1 kilo mole is 16 kgs. Then corresponding air so we have 2 into 1 mole of Oxygen that is approximately 32, we are taking approximate number just the neglecting the fractions plus 3.76 in nitrogen is approximately 28 so, that comes roughly 274.56 kg the stoichiometric air fuel ratio. The chemical equation is as follows:

$$1 \text{ kmol of } CH_4 \rightarrow 12+4\times1 = 16 \text{ kg}$$

$$\text{Air} \rightarrow 2[32 + 3.76 \times 28] = 274.5$$

So, from here we get that, to have complete combustion of 1 mole of methane that is about 16 kg of methane we have to supply 274.356 kg of air. And their corresponding stoichiometric ratio will be: mass of air divided by mass of fuel and mass of air in this case 274.56 divided by 16 give your number of approximately 17.16. So the stoichiometric air fuel ratio of methane is approximately 17.16.

$$Stoichiometric\ {}^{A}\!/_{F}\, ratio = \frac{274.56}{16} = 17.16$$

For practical hydrocarbon fuels is generally varies in the range of 14 to 16. The stoichiometric air fuel ratio, for practical gasoline or diesel group of fuels it is in the range of 14 to 16. So, stoichiometric air fuel ratio gives an idea exactly or I should say the minimum quantity of air required to ensure complete combustion. Truly speaking in practice if you provide exactly this amount of air we may not see the complete combustion because completeness of combustion depends upon each molecule of fuel is able to come in contact with oxygen or not.

But theoretically speaking this exactly this amount of air should ensure complete combustion of methane and in the product, we shall be having only carbon dioxide and water vapour, no methane left out, no carbon monoxide kind of intermittent product. And another parameter I have to define here, that is called the equivalence ratio. During practical operation it is not possible to precisely supply the stoichiometric air fuel ratio. We generally supply additional amount of air or less amount of air. And that factor is measured in terms of this equivalence ratio.

Equivalence ratio commonly denoted by ϕ it is defined as:

$$Equivalence\ ratio\ (\phi) = \frac{Stoichiometric\ {}^{A}\!/_{F}\, ratio}{Actual\ {}^{A}\!/_{F}\, ratio}$$

Say if we are talking about the stoichiometric air fuel ratio to actual air fuel ratio, this ratio we are calling as the equivalence ratio. Like if we talk about a given, if you just go back with example of methane, we trying to get complete combustion of 16 kg or 1 kmol of methane, then this one can be written as the stoichiometric air fuel ratio will be:

$$= \frac{274.56/_{16}}{m_a/_{16}}$$

So, it gives us:

$$= \frac{274.56}{m_{actual}}$$

So, when we are talking about a fixed quantity of fuel this one can we represent in terms of the corresponding quantity of air. When this ϕ is less than 1, what does that mean? When ϕ equal to 1 means what? That is talking about the stoichiometric mixture only, actual stoichiometric air fuel are equal to each other and you are supplied the stoichiometric quantity of air only.

When ϕ is less than one that means we have supplied more amount of air than required for complete combustion and that mixture is known as a lean mixture. That particular air fuel mixture is known as a lean mixture, lean means it contains more air then required. We have supplied air more than what is required for stoichiometric consideration. Whereas ϕ is greater than 1. What does that mean? Then we have supplied less amount of your than required we call that rich mixture. Rich mixture means we have to supply more air or whatever minimum quantity of air is required we have not supplied, that we have supplied less. Or from other point of view, we can say that the given quantity of air we have supplied more amount of fuel that is why this lean and rich terms comes in.

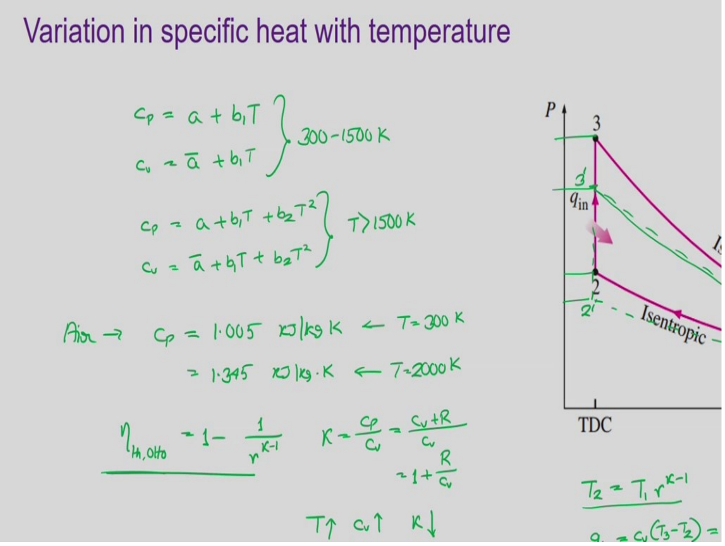
Now, this composition when we are talking about this composition of cylinder gases, then this stoichiometric air fuel ratio and the equivalence ratio comes into picture. Because this strongly can affect the performance of the cylinder gases. This is one diagram where we have the compression ratio on the horizontal axis and the thermal efficiency on the vertical axis. This is the air standard efficiency, as the compression ratio, air fuel increases.

Now, look at this actual diagram, now when you are talking about 100%, we actually talking about the stoichiometric mixture. You can see there is a significant difference between the actual prediction and the air standard prediction. Or you can say this difference is the red one corresponds to the ideal cycle air standard cycles and the purple line corresponds to fuel-air cycle working with stoichiometric mixture. And now as the composition changes, if we make the mixture leaner that is and more amount of air you can see the efficiency is actually increased when you gone to 120%.

Whereas when we are talking about a richer mixture equivalence ratio is greater than 1 in this case, but mixture is rich. Or you can say we have supplied 80% air than required to have a stoichiometric mixture. Then we are having a reduction in the efficiency. So, in practical engines lean mixtures give you better performance. Of course, there is some limit, it does not mean that you can keep on adding more and more air.

But generally, a leaner mixture particular in the range of 120 to 130% or equivalence ratio in the range of 0.8 to 0.9 we get a very high range of efficiency. So, composition of cylinder gases definitely can affect the performance. To get the theoretical prediction about these values, we can use different combustion charts and now a days with digital computers, we can easily go for detail combustion analysis.

(Refer Slide Time: 42:20)



Variation in specific heat with temperature

$$\left.\begin{aligned} c_p &= a + b_1 T \\ c_v &= \bar{a} + b_1 T \end{aligned}\right\} 300-1500\,K$$

$$\left.\begin{aligned} c_p &= a + b_1 T + b_2 T^2 \\ c_v &= \bar{a} + b_1 T + b_2 T^2 \end{aligned}\right\} T > 1500\,K$$

$$\text{Air} \rightarrow \quad c_p = 1.005\ kJ/kg\,K \leftarrow T = 300\,K$$

$$= 1.345\ kJ/kg \cdot K \leftarrow T = 2000\,K$$

$$\underline{\eta_{th,Otto} = 1 - \frac{1}{r^{K-1}}} \qquad K = \frac{c_p}{c_v} = \frac{c_v + R}{c_v} = 1 + \frac{R}{c_v}$$

$$T\uparrow \quad c_v\uparrow \quad K\downarrow$$

$$\underline{T_2 = T_1 r^{K-1}}$$

$$q_{in} = c_v(T_3 - T_2) =$$

The next factor i.e., the variation in specific heat with temperature. This is something that we can directly represent on the cycles. Specific heats are strong functions of temperature. When we are talking about lower temperature say something in the range of 300 to 1500 K, it is generally:

$$C_P = a + b_1 T$$

and similarly

$$C_v = \bar{a} + b_1 T$$

this is in general in the range of 300 to 1500 Kelvin. And what is the relation between these two:

$$C_P - C_v = a - \bar{a} = R$$

R is the gas constant. So, the difference C_P and C_v always remains constant. And as in this case, C_P and C_v almost follows a linear relationship over this temperature range. And now we go to higher temperature then C_P becomes quadratic function, something in the form of:

$$C_P = a + b_1T + b_2T^2$$

and similarly

$$C_v = \bar{a} + b_1T + b_2T^2$$

this is when the temperature is greater than 1500 K, which is quite common in case of IC engines. So, you can see as a temperature increases the specific heat keeps on increasing very, very rapidly.

If I can give you one simple example, if we talk about air C_P for air at normal atmospheric temperature like 300 K is about 1.005 kJ/kgK this is at T = 300 K whereas when go to T= 2000 K this becomes 1.345 kJ/kgK. This is a significant change in the value of the specific heat. A corresponding change the C_v can also happen, means the same way just you substitute R for air, the value of C_v can also be obtained.

And the change in C_P and C_v is directly reflected in terms of the K. And if you think about the Otto cycle efficiency, we know that the efficiency is predicted as ideal thermal efficiency for an Otto cycle is:

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

Now this r here is a design parameter because the compression ratio depends on the location of top dead centre and bottom dead centre. So, for given engine, compression ratio is fixed. But k is a property of the gas mixture that you are talking about and k is a ratio of C_P and C_v.

$$k = \frac{C_p}{C_v}$$

So, if you just use this particular relation so we can write it as:

$$= \frac{C_v + R}{C_v}$$

i.e.,

$$k = 1 + \frac{R}{C_v}$$

Now as the temperature increases, we can see that C_v increases then what will happen to k? As C_v increases k will decrease and therefore it is directly going to affect the thermal efficiency and all other similar calculations. Let us check the standard Otto cycle diagram. This is the standard Otto cycle diagram where we are starting the compression process from point number 1 and falling isentropic process going to point number 2.

Now, we know that

$$T_2 = r^{k-1}$$

So, as the compression process goes on the temperature also keeps on increasing and as the temperature increases corresponding specific heats change. And as the specific heat change, corresponding k will decrease. Then what will be the effect of this? As a specific heat is changing then if we modify the diagram, your actual compression line will start deviating from this because the change in k.

Because the temperature that is at a particular point been credited with this relation that is no longer valid as temperature is increasing k is continuously decreasing and at the end of compression, we may reach up at point somewhere here is 2'. This T_2' can be significantly lesser than the T_2. Remember the volume is same corresponding to both 2 and 2' because they are talking about both TDC location, but it is only because of the change in specific heat with temperature or corresponding reduction in the value of k with temperature, the point 2 come down to 2'. So this way we have a reduction in the location of point 2'.

And now if we talk about the constant volume heat addition. Instead of starting from point 2 and here starting from point 2'. Now, if you are giving a fixed amount of energy, in the ideal process:

$$q_{in} = c_v(T_3 - T_2)$$

In the actual process what is happening already we have down to point 2', q_{in} remains same but here:

$$= c_v(T_3 - T_2')$$

we are already down to point 2' which is at what temperature and during heat addition C_V will increase even further with temperature thereby causing further reduction in this location of 3' and the heat addition process may go on and end up only somewhere here 3'.

So, 2' is the point signifying the end of compression process and 3' will be the end of combustion process. This difference between the 2 and 2' has been manifested further between 3 and 3' because the 3' has come lower or has a come down from 3, because of the lower positioning of this 2' plus even further reduction in C_v with increase in temperature. During the heat addition process temperature increases rapidly, accordingly the C_v increases also very rapidly.

And if q_{in} remains constant then C_v increases corresponding temperature is always smaller which is signified here. And this is a constant volume process, so as the final temperature reduces final pressure also will reduce. So in ideal situation you should have:

$$\frac{P_3}{T_3} = \frac{P_2}{T_2}$$

This is the ideal situation. What we are getting here? Here are getting:

$$\frac{P_3{}'}{T_3{}'} = \frac{P_2{}'}{T_2{}'}$$

as 3' and 2' all has come to a lower location so P_3' also will be lower than actual P_3.

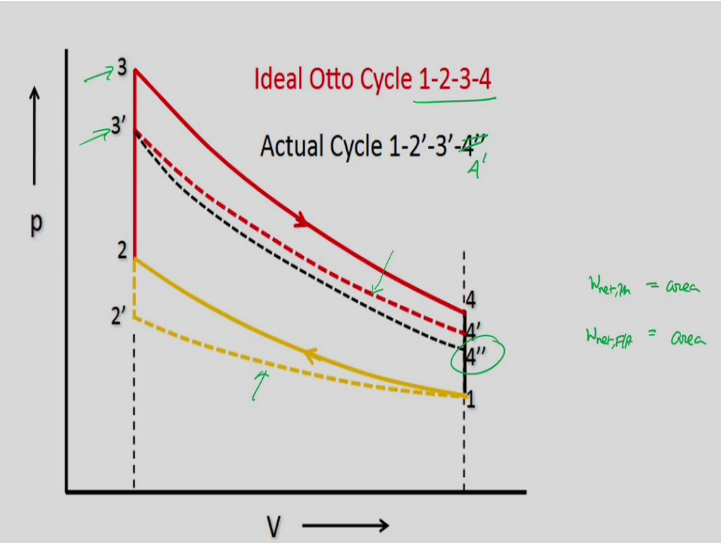
Similarly, here we shall be having T_3' lower than theoretical value of P_3. So, this is significant modification in the expression of cycle for this. Now what about the expansion process, during the expansion process if the specific it remains constant, then we shall be having a line somewhat like, let me draw properly, your line will be something like this ending of somewhere here, to give you the point 4' and 4'- 1 is the exhaust process. But actually this expansion is happening, what is happening to the temperature?

As the expansion is happening temperature is decreasing now when temperature is decreasing C_v also decreases accordingly k increases. Now as k is increasing in this case the line will now start moving in the higher direction, now as k is increasing temperature will increase this green line green confirmation that are drawn from 3'to 4' that actually does not take into consideration the changes specific heat. But if you Consider the changes specific heat with increase in temperature then your line will be somewhat like this and finishes of somewhere here.

So, we shall be having a point 4'' which signifies the end of the cycle. So, during process 1-2 we are having a lowering in the temperature level and so the final temperature at the end of compression both temperature and pressure lower than ideal cycle. During constant volume heat addition again specific is changing, because of that the final temperature and pressure at the end of constant volume heat addition is significantly lower compared to the ideal cycle.

And similarly, the temperature and pressure at the end of expansion is also lower than the idea cycle prediction. However, there is some kind of recovery because of the reduction in temperature and corresponding increasing the value of *k*.

(Refer Slide Time: 51:46)



So, if we combine all the factors, this is the diagram that you have. Here 1-2-3-4 is the ideal Otto cycle. This dotted yellow line represents the compression process, actual compression process using the point 2 to come down to 2'. Then 2' to 3 is the actual combustion process. Then you can see there are two lines drawn on the black line does not; the dotted black line does not take into consideration the changes specific during expansion. Whereas the dotted red line that is I am talking about this particular line that takes into consideration the actual cycle for change in specific heat during the expansion process because of the reduction in temperature.

So, this is not written properly this should be 4'. Here this 4'' this point has been identified neglecting the changes specific during expansion. However, 3' to 4' is a true expansion process which takes into consideration the reduction in temperature and corresponding increase in *k* in the expansion process only. So, now look at the two diagrams you have the theoretical work production from the cycle. So,

$$W_{net,th} = area\ 1-2-3-4-1$$

whereas the net work output from this fuel-air cycle considering only one effect of course everything is wrong considering.

$$W_{net,F/A} = area\ 1-2'-3'-4''-1$$

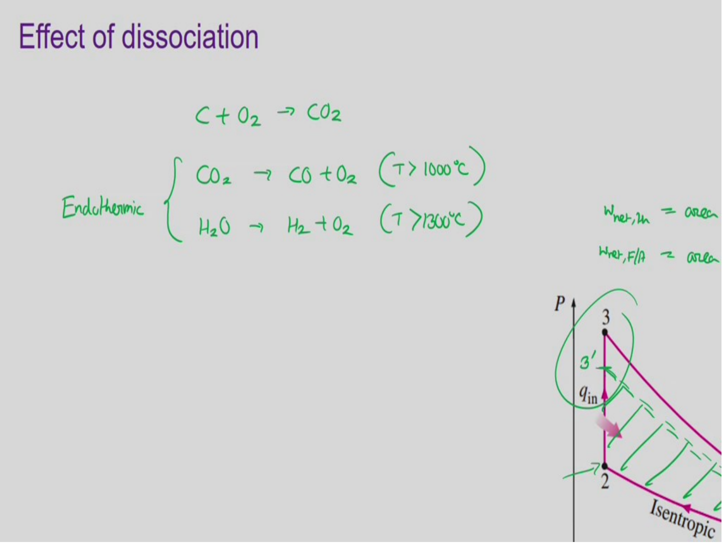
Only the variation is specifically the temperature what we are getting here corresponding area is starts from 1 then goes to 2' then we have 3' then come down to 4'' and then 1. And which area is smaller definitely the second area is a bit smaller. Particularly the expansion process is starting from a much lower pressure level which always signifies a lower workout from this.

Similarly, the final temperature level is also lower. This 3' corresponds to peak pressure and temperature as I have mentioned earlier. The peak pressure and temperature predicted by the ideal cycle which corresponds to point 3 is quite far off then what is been predicted by this fuel-air cycle considering the effect of specific heat given by point 3'. So, this is the effect of the variation in specific heat with temperature.

But the question is why is specifically should vary with temperature? A rough answer can be given by considering the amount of energy absorbed by the atoms. Specific corresponds to the energy of the molecules, but inside molecules can have atoms in case of multi atomic molecules and a portion of the energy also goes to the atoms. And as the energy level and temperature level increases, molecules keep on vibrating and atoms and molecules also give some vibrating and their vibration level also keeps on increasing.

Therefore, as we move on to higher temperature, more and more energy will be required towards this added vibration enhance vibration of atoms. So, more amount of energy will be absorbed by the molecule itself for a given amount of temperature rise. That is what can be a rough explanation about this increasing specific heat for this and effect of that you can clearly see from here.

(Refer Slide Time: 55:22)



Effect of dissociation

$C + O_2 \rightarrow CO_2$

Endothermic $\begin{cases} CO_2 \rightarrow CO + O_2 & (T > 1000\,^\circ C) \\ H_2O \rightarrow H_2 + O_2 & (T > 1300\,^\circ C) \end{cases}$

$W_{net,th} = area$

$W_{net,F/A} = area$

Next is the effect of dissociation. Effect of dissociation talks about the reverse chemical reaction that can happen only at very high temperature. Like we know that

$$C + O_2 \rightarrow CO_2$$

but that is it normal condition. When you go to very high temperature then go to the

$$CO_2 \rightarrow C + O_2$$

this happens when the temperature is generally beyond 1000 ^{0}C. Similarly, the water vapour can also break to hydrogen and oxygen when the temperature crosses something like 1300 ^{0}C.

And both these reactions are endothermic in nature. That is when these reactions happen, they will absorb energy from the surrounding for these reactions to happen. And therefore, as the temperature increases the rate of disassociation keeps on increasing and the corresponding amount of energy absorbed from the surrounding which we have, that also keeps on increasing thereby lowering the temperature of the total system.

Accordingly, it can have a significant effect on the system performance. Look at this, during the compression process generally the temperature is quite low. End of compression this temperature also significantly lower to cause any kind of disassociation. Dissociation comes into picture during the combustion or heat addition process. And as we move from point 2- 3 maybe a point somewhere here, if the disassociation starts to get initiated that will restrict the total temperature rise.

Because whatever heat is being added that from the surrounding source or the amount of heat produced because of combustion that will again get absorbed because of this endothermic dissociation reaction. And so, the final temperature will get restricted to some 3'. So, again, the effect of dissociation is to reduce the temperature. Remember here there is no change in the position of this point 2. The disassociation is not associated compression process, that is primarily associated with this combustion process.

And because of this now your expansion again will have something like this following a line to move to a point 4'. But the opposite effect, similar to the previous case we can have the opposite effect here as well. As the temperatures has to come down in the expansion process, again this product of dissociation that is CO NO_2 recombine back to form CO_2 there. And that reaction is exothermic some add additional energy is produced. So, this line may deviate a bit. And again, it will end up at some point 4'.

But truly speaking this recombination of dissociation products that generally happens towards the end of the expansion process and that time the pressure level is too low to have any significant effect on the cycle performance. So, again if we talk about the net work output from the theoretical cycle that corresponds to the area 1-2-3-4-1.

$$W_{net,th} = area\ 1-2-3-4-1$$

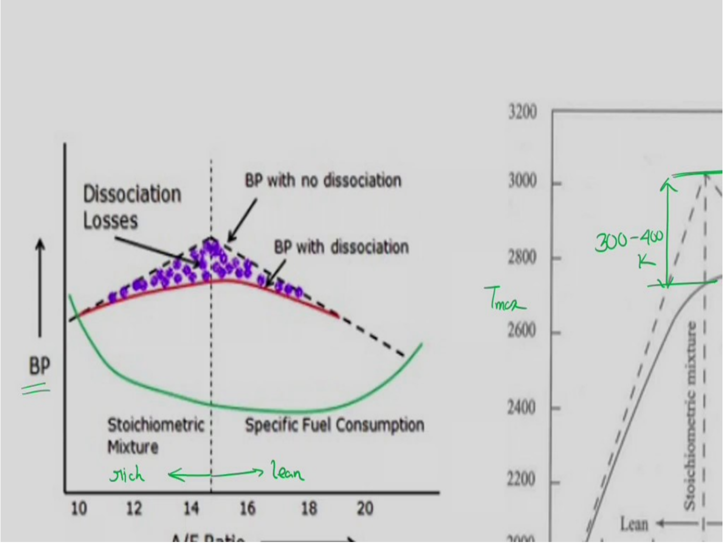
whereas the network output predicted by the fuel-air cycle considering the effect of dissociation only. Then it will be the area correspond to 1 and 2 both are not changing 3 has come down to 3' 4 has moved to 4'' and 1.

$$W_{net,F/A} = area\ 1-2-3'-4''-1$$

And this area definitely is significantly smaller. If we have this out your hatched area for the actual fuel-air cycle or fuel-air cycle considering the effect of dissociation is this much. There is still some amount of area left, so, there is significant reduction in the total work output. So, the effect of dissociation is to reduce the total work output and also to reduce the maximum pressure and temperature of the cycle.

We can also think about what we mentioned in conjunction with Carnot cycle. To increase the thermal efficiency, we need to have the temperature of heat addition to be higher. And as this maximum cycle temperature is reducing so the mean temperature of heat addition is coming down thereby lowering the thermal efficiency.

(Refer Slide Time: 59:51)



The effect of dissociation actually is strongest corresponding the stoichiometric mixture. Here we have plotted the big power or the power output corresponding the air fuel ratio which is in the range of 14 to 16 for common hydrocarbons. So we can see the effect of disassociation is a strongest at for the stoichiometric mixture. This black dotted line is I am talking about this line, this line corresponds to the brake power output variation with air fuel ratio, and we get the maximum output corresponding to stoichiometric mixture vertical dotted line.

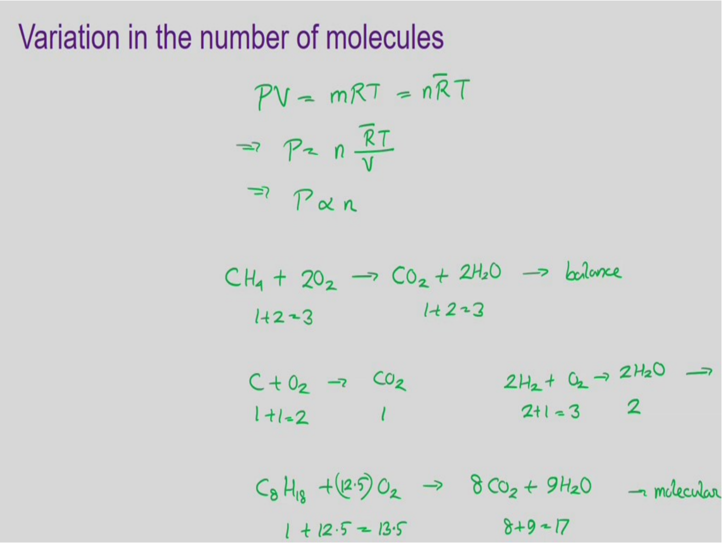
And because of the presence of dissociation that is shown by the red line. You can see the effect of disassociation the strongest for the stoichiometric mixture simply because stoichiometric mixture produces the highest temperature. So, in this direction, air fuel ratio is increasing. So, we are talking about a lean mixture here and this lean mixture air is more fuel is less so the total energy released during combustion is less. So, the maximum temperature that can be produced that is also there so disassociation effect is lesser. Similarly, when air fuel ratio reduces that is your moving to the rich mixtures, in this case fuel is more but air is less, so here there is more probability of having incomplete combustion.

Incomplete combustion means the combustion product you already having the terms like CO and O_2 present. And the presence of these products reduces the rate of disassociation or probability of dissociation. So again, the effect of dissociation is lesser for rich mixtures. So, the dissociation is a strongest corresponding to stoichiometric mixture. Similarly, we can also see the maximum temperature. This is your maximum temperature and this thing can be called the degree of richness.

Here zero corresponds to the stoichiometric mixture and anything greater than this is rich mixer, less than this is the lean mixture. So, you can also say that this is somewhat similar to the equivalence ratio. Because in case of equivalent ratio the $\phi = 1$ corresponds to the stoichiometric mixture, $\phi > 1$ we talked about rich mixture. So, it is something like $\phi - 1$ approximately. So, you can again see when there is no dissociation, this is the highest temperature. However, in the presence of dissociation there is a change of about in the range of something like 300 to 400 K reduction in the final cycle temperature.

Corresponding relation will be in the cycle pressure as well and therefore we can lose in the cycle efficiency and output point of view.

(Refer Slide Time: 1:02:57)



Variation in the number of molecules

$$PV = mRT = n\bar{R}T$$

$$\Rightarrow P = n\frac{\bar{R}T}{V}$$

$$\Rightarrow P \propto n$$

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O \rightarrow \text{balance}$$

$$1+2=3 \qquad 1+2=3$$

$$C + O_2 \rightarrow CO_2$$

$$1+1=2 \qquad 1$$

$$2H_2 + O_2 \rightarrow 2H_2O \rightarrow$$

$$2+1=3 \qquad 2$$

$$C_8H_{18} + (12.5)\,O_2 \rightarrow 8CO_2 + 9H_2O \rightarrow \text{molecular}$$

$$1 + 12.5 = 13.5 \qquad 8+9=17$$

Final effect is the variation in the number of molecules. Now, you may be wondering what can be the effect of the number of molecules. Now the effects comes from the ideal gas equation of state. Think about the ideal gas equation states that

$$PV = mRT = n\bar{R}T$$

where

n is the number of molecules

R bar is the universal gas constant

Therefore, we can write:

$$P = \frac{n\bar{R}T}{V}$$

Now for a given volume and temperature then *R bar* is universal constant it is directly proportional to the number of molecules that present there and the number of moles present there.

And therefore, if the number of moles increase the pressure will increase number of moles decreases the pressure will decrease thereby affecting the work output. Think about standard chemical reaction, like if we talk about say,

$$CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O$$

the one that we just considered earlier. Now on the reactant side how many number of moles you have? 1 + 2 equal to 3 moles, on the product side 1 + 2 equal to 3 moles. So, because of the reaction there is no change in the number of molecules, i.e., they are in balance.

Now another even simple reaction:

$$C + O_2 \rightarrow CO_2$$

Now how many number of moles you can have on the reactant side? 1+1 =2 whereas we have only one on this side. And another common example

$$2H_2 + O_2 \rightarrow 2H_2O$$

On the reactant side we have 2 + 1 that is 3 and on the product side you have 2. So because of this chemical what is happening? Number of moles on the product side that is actually reducing and as the number of moles are reducing this thing is called molecular contraction. That is because of the chemical reaction number of moles are reducing or total volume of total mass you can say is getting contracted or I should not say mass in mole sense that is getting contracted.

And as it is reducing, number of moles, so pressure inside the cylinder will also reduce. A typical example with from other way around with:

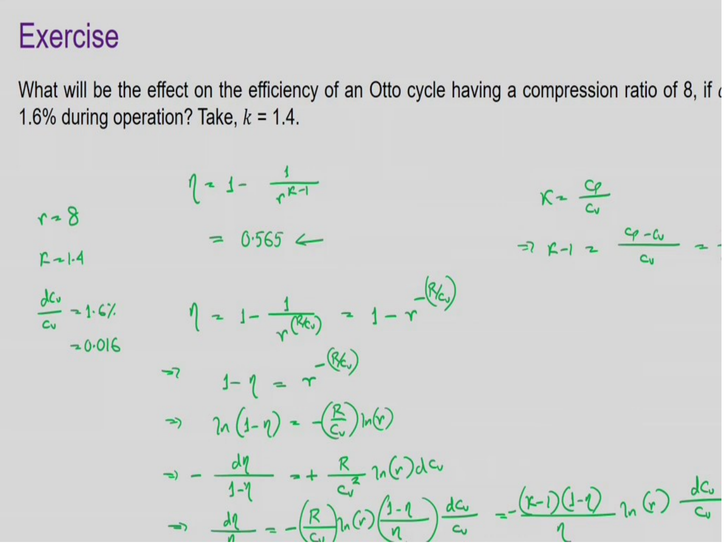
$$C_8H_{18} + O_2 \rightarrow 8CO_2 + 9\ H_2O$$

so how many moles of oxygen will be required? Then how many moles we have on the reactant side? We are 1 + 12.5 that is 13.5 moles. Then how many moles we have on the product side? We are 8 + 9 that is 17 moles. So, number of moles are increasing in this case. Here more number of moles produced because of the chemical reaction and this is an example of molecular expansion. Number of moles increased accordingly pressure inside the cylinder is also is going to increase.

So, because of this addition in the number of molecules the system pressure can directly change affecting the total output. So these are the four factors that can primarily affect the performance of engines ideal cycles, and these are the four factors that you take into account in case of fuel-air cycles. If you combine their effect primarily effect of if you just take a summarised effect, we have of course discussed each of them separately.

But the effect of dissociation and effect of addition is specifically temperature is more or less the same. To reduce the peak pressure and temperature of the cycle and thereby to reduce the net work output of the cycle, the effect of variation in the number of molecules is to affect the system pressure. Whereas the change in the gas composition can affect the chemical properties of this. I would like to end lecture today with small numerical exercise. Here just read the question.

(Refer Slide Time: 1:07:28)



Exercise

What will be the effect on the efficiency of an Otto cycle having a compression ratio of 8, if
1.6% during operation? Take, k = 1.4.

$r = 8$

$k = 1.4$

$\frac{dc_v}{c_v} = 1.6\%$
$= 0.016$

$\eta = 1 - \frac{1}{r^{k-1}}$
$= 0.565 \leftarrow$

$k = \frac{c_p}{c_v}$

$\Rightarrow k-1 = \frac{c_p - c_v}{c_v} =$

$\eta = 1 - \frac{1}{r^{(R/c_v)}} = 1 - r^{-(R/c_v)}$

$\Rightarrow 1-\eta = r^{-(R/c_v)}$

$\Rightarrow \ln(1-\eta) = -\left(\frac{R}{c_v}\right)\ln(r)$

$\Rightarrow -\frac{d\eta}{1-\eta} = +\frac{R}{c_v^2}\ln(r)\,dc_v$

$\Rightarrow \frac{d\eta}{\eta} = -\left(\frac{R}{c_v}\right)\ln(r)\left(\frac{1-\eta}{\eta}\right)\frac{dc_v}{c_v} = -\frac{(k-1)(1-\eta)}{\eta}\ln(r)\frac{dc_v}{c_v}$

What is the effect on the efficiency of Otto cycle having a compression ratio of 8, c_v increases by 1.6 % during operation k equal to 1.4 that is given. Only the variation of specific heat that we have to consider or variation of k with c_v that you consider to get the cycle efficiency. Now for Otto cycle we know that,

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

it is given that

$R = 8$,

$k = 1.4$.

So, if you put it there the efficiency under ideal condition is there is no variation specification this should be 0.565 or 56.5 % but there is a variation in c_v.

So how to take care of that we know that

$$k = \frac{c_p}{c_v}$$

that is

$$k - 1 = \frac{c_p - c_v}{c_v} = \frac{R}{c_v}$$

So

$$\eta = 1 - \frac{1}{r^{\left(\frac{R}{c_v}\right)}} = 1 - r^{-\left(\frac{R}{c_v}\right)}$$

we are trying to relate this η with c_v because these are information given and that is why we modified k in terms of c_v and know R is a constant, r is also a constant. So,

$$1 - \eta = r^{-\left(\frac{R}{c_v}\right)}$$

or we are taking a log on both sides to simplify this:

$$ln(1 - \eta) = -\left(\frac{R}{c_v}\right) \ln (r)$$

now we are differentiating both side then what you are going to get:

$$-\frac{d\eta}{(1 - \eta)} = +\left(\frac{R}{{c_v}^2}\right) \ln (r) \quad dc_v$$

And in this side which term is variable c_v is only because both R and r are constants. Taking everything on this side what we can have is:

$$\frac{d\eta}{\eta} = -\left(\frac{R}{c_v}\right) \ln (r) + \left(\frac{1 - \eta}{\eta}\right) \frac{dc_v}{c_v}$$

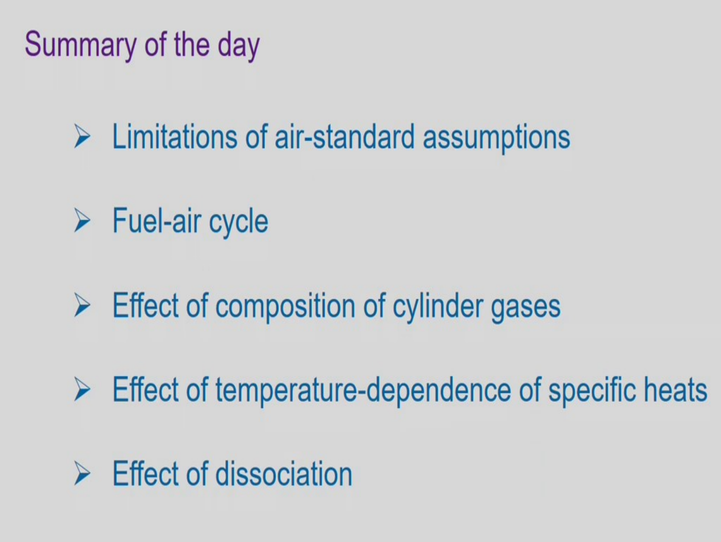
So,

$$= -\frac{(k - 1)(1 - \eta)}{\eta} \ln (r) \frac{dc_v}{c_v}$$

It is given that c_v increases by 1.6 % and I have minus sign also ok. So, this c_v increases by 1.6 % this dc_v / c_v is this 1.6 % or you can say 0.016. We already have k we have r and η is calculated 0.565. If you put everything then you are going to get this to be −1.025 % that is only 1.6 % variation in the specific heat during operation causes more than 1 % reduction in the final thermal efficiency.

This is again only considering the specific heat, a very simple analysis just show how the variation is specific heat can affect the cycle performance. We shall be solving couple more similar problems in our next lecture and then we shall we talking about those losses which we have not considered fuel-air cycle and also the valve timing diagram.

(Refer Slide Time: 1:11:58)



Summary of the day

- Limitations of air-standard assumptions
- Fuel-air cycle
- Effect of composition of cylinder gases
- Effect of temperature-dependence of specific heats
- Effect of dissociation

So, just to summarise today's observation. We have discussed about the limitations of air standard functions, practical factors, which you are excluding air. Then the fuel-air cycle was introduced and four factors, which comprises of various cycles: effect of composition of cylinder gas, effect of temperature dependence of specific heat, effect of dissociation and the effect of variation in number of molecules have been discussed.

So, on this fuel-air cycle we shall be adding the actual cycle effects and also the valve timing diagram in the next lecture. Till then you please revise this lecture and if you have any query, please write to me, Thank you.