Applied Thermodynamics for Engineers
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Lecture - 12
Otto, Diesel and Dual Combustion Cycles

Morning friends, welcome to the second lecture of our week number 4 where we are talking about the air standard cycles or so-called gas power cycles. Now, in the previous lecture, we have seen that thermodynamic power producing cycles can broadly be classified into two categories depending on the state of the working medium, the gas power cycles where the working medium always remains in the gaseous phase. Because the working temperature levels are well above the critical temperature of the substance. Whereas the other category can be the vapor power cycles where we have exclusive liquid to vapor or vapor to liquid phase change and operation invariably is limited below the critical temperature of the substance, but in this particular week, we are restricting ourselves to the gas power cycles which are also called the air standard cycles because of the assumptions that we have discussed in the last lecture.

Vapor power cycles will be discussed in one of the subsequent modules. Now, here we have seen that we are primarily focusing on the reciprocating engines or the automobile engines and trying to develop ideal cycles which can somewhat simulate the operation of those reciprocating engines. We have discussed about the different types of reciprocating cycles or reciprocating engines that are possible like two-stroke cycles and four-stroke cycles.

We can have SI engines or CI engines and accordingly we can have different types of cycles governing the operation of either SI engine or CI engine. We have also discussed about the difference between the ideal cycle and actual cycles. We have seen that while Carnot cycle is the most ideal cycle that we can have because that is a totally reversible cycle. They are providing the maximum possible efficiency between two temperature limits.

Ideal cycles have generally efficiency lower than the Carnot cycle, because ideal cycles are internally reversible but not externally reversible. Like in Carnot cycle that being totally reversible we can have heat transfer only in the mode of infinitesimally small temperature difference between the system and the surrounding, which requires either an infinite surface area or infinite amount of time, both of which are practically impossible.

Therefore, in ideal cycles, we remove the external reversibility consideration thereby making the cycle externally reversible and hence allowing heat transfer with finite temperature difference. However, the cycles are internally reversible i.e., there is no friction present and therefore their efficiencies generally are lower than the Carnot cycle efficiency. However, this ideal cycle is also a kind of assumptions because it neglects the internal irreversibilities and also several complexities that maybe present in the actual cycle. Thereby, they provide somewhat upper limit of operation of the actual cycles, which each of the actual engines try to reach.

We have also discussed briefly about the Otto cycle which are the ideal cycle for SI engines. However, because of the time constraint, I was trying to go through very quickly through the mathematical derivation of the efficiency of Otto cycle for which I made some mistake.

Of course, I corrected that but I am not at all happy. So, to have a proper description of the same thing, I would like to repeat that portion and also some subsequent discussion on the Otto cycle and then moving on to the next cycle in our course.

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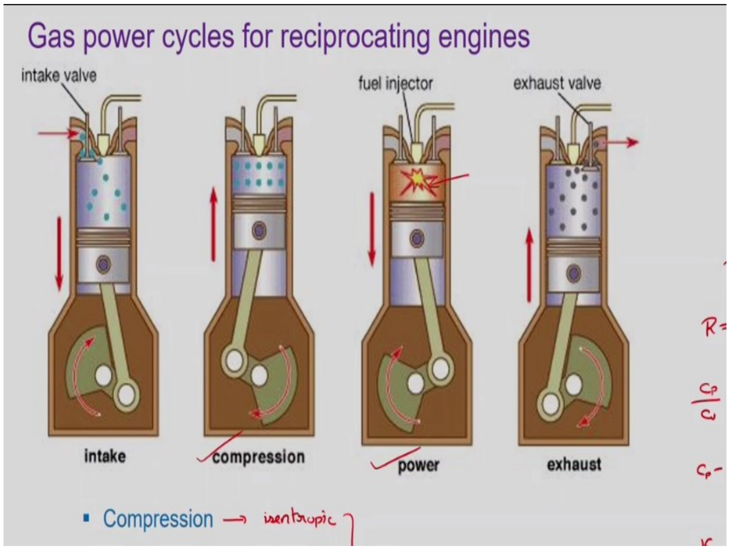

Gas power cycles for reciprocating engines

- Compression → isentropic

Now, this is a brief summary of what we have seen. In reciprocating engines, we have seen that in a 4-stroke cycle generally there are four strokes to comprise one particular cycle, the intake stroke, compression stroke, power stroke and the exhaust stroke and that is true for both SI engines and CI engines. We have seen that once we apply the air standard assumptions on them, we generally can identify four processes to resemble this 4-stroke.

The first process is compression, which actually is a representative of this particular stroke. Second process is heat addition, heat addition substitutes the combustion process because here we neglect any kind of chemical reaction in the ideal cycle rather considering the amount of heat produced by combustion to be supplied from some external source by the means of some kind of heat addition.

So, heat addition is synonymous to the combustion process that is going on inside the system something like this one. Then, we have the expansion or power stroke, which is representative of this one. Finally, we have the heat rejection process, which you can visualize to be opposite of the heat addition process and as in air standard cycles as per our assumption, we are considering it to be a closed system thereby completely eliminating any inlet and exhaust processes.

So, the heat rejection process actually allows it to be considered as a closed system, and get the system at the end of expansion stroke back to the initial point of the compression stroke. So, these are the four processes that primarily constitute any air standard cycle. But different air standard cycles can be conceptualized depending upon the individual nature of all these four processes.

Firstly, heat addition and heat rejection that is where primarily the difference lies, but before that we have the compression and expansion processes. Now, compression and expansion processes as we are assuming the systems to be internally reversible, so they are always frictionless along with that, if we assume these two processes to be reversible as well, then that is a best possible performance that we can get.

Like what we are saying is that as all these processes are already internally reversible plus if we assume them to be adiabatic, then we get this compression process to be isentropic. And we know that when a process is isentropic or reversible adiabatic, then compression is a process which requires work input from outside, then, it is possible to have the smallest possible work input.

Similarly, if we have the expansion process also to be isentropic, then it is possible to have maximum possible work output from this cycle. And that is why this compression and expansion processes are always assumed to be isentropic. But heat addition and heat rejection can vary. Like when this heat addition and heat rejection processes are, if we are assuming the heat addition process to be isothermal or I should say reversible isothermal and similarly the heat rejection as well to be reversible isothermal.

Then, what we get this, isentropic compression and expansion, reversible isothermal heat addition and heat rejection i.e., of course our most ideal cycle which is the Carnot cycle, where the heat addition and heat rejection processes are also externally reversible as you are considering reversible isothermal heat transfer with infinitesimally small temperature difference between system and surrounding.

And in case of ideal cycles, we cannot have isothermal heat addition and heat rejection. We need to have something else. So, when heat addition and heat rejection both are considered to be isochoric, means happening at constant volume, then the cycle that we get, that cycle is called the Otto cycle. In Otto cycle, both heat addition and heat rejection are isochoric or happening at constant volume. This is the ideal cycle for SI engine. Whereas when the heat addition process is assumed to be isobaric that is happening at constant pressure whereas heat reaction remains isochoric, then the cycle that we get, that cycle we call the Diesel cycle. This is the ideal cycle for the CI engines.

Similarly, it is also possible to conceptualize another cycle where heat addition and heat rejection both are isobaric. That particular cycle is another air standard cycle known as the Brayton cycle or Joule cycle. But this is generally not using reciprocating engines rather it is use more in gas turbine kind of rotary devices. This one we shall be discussing in the corresponding week, but here we are primarily focusing on the Otto and Diesel cycle. So, in case of Otto cycle we have the heat addition process to be isochoric, whereas in case of Diesel cycle we have the heat addition process to be isobaric. In both cases, heat rejection is isobaric in nature.

So, now you want to discuss about the Otto cycle first and then moving on to the Diesel cycle, and then a combination of these two in the form of a dual combustion cycle. But before that I would like to quickly mention about one of the assumptions under the air standard assumptions which is we are assuming the working medium to be air which is assumed to be an ideal gas.

That is a very good assumption as that simplifies our analysis quite a bit. As we are assuming air to be an ideal gas, so we can always represent it by this equation of state

$$Pv = RT$$

where

v is the specific volume,

P is the pressure,

T is the absolute temperature and

R is the gas constant, typical value for R is 0.287 kilo joule per kg Kelvin. So, this is a constant, this is always known.

And another assumption that we have considered that is the specific heats to be constant. We have assumed the specific heat C_p and C_v to be constant. Now C_p/C_v, this ratio is given by K which I already used in Otto cycle. So, as we already know the value of R which is a constant for air, then truly speaking while solving a numerical problem, we need to know just one among these three: C_p, C_v and K. Because what is the relation of C_p with R, we know that:

$$C_p – C_v = R$$

and now we have

$$C_p /Cv=K$$

Therefore, if we know any one of these three that is K, C_p and C_v, if we know just any one among the three plus R which is already there, then we can calculate the other two as well. So, though in several textbooks you may find some given value of C_p, C_v etc, but we shall be following this methodology where we are going to take R as a constant. And we are going to specify one of three either K or C_p or C_v and remaining two needs to be calculated from the given situation.

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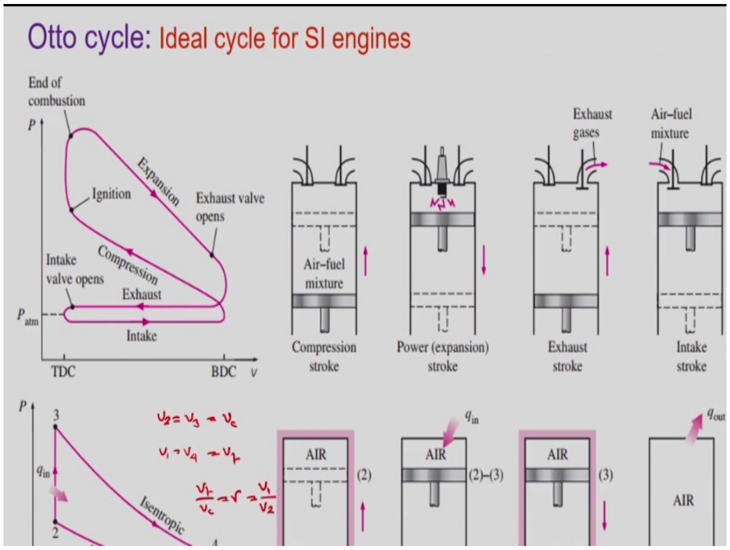

Otto cycle: Ideal cycle for SI engines

$v_2 = v_3 = v_c$

$v_1 = v_4 = v_T$

$\frac{v_T}{v_c} = r = \frac{v_1}{v_2}$

Now, we move to the Otto cycle. This slide I have shown in the previous picture itself, Nikolaus Otto is a Frenchman who was a first to conceptually develop this cycle. The one that is shown here that is what goes in practice, but this is the corresponding air standard version of this, where we have the isentropic compression process as 1 to 2, so we have 1 to 2 as isentropic compression.

Then, we have 2 to 3 as isochoric heat addition. Then, we have 3 to 4, what is that 3 to 4? 3 to 4 is again another isentropic process but during when the volume is increasing, pressure is decreasing so it is an isentropic expansion process. And 4 to 1 is isochoric, just we have discussed in a previous slide, it is an isochoric heat rejection process during which the volume remains constant.

So, here the smallest possible volume the system can have is this v_2 where

$$v_2 = v_3$$

which corresponds to the top dead center. Here, this volume that is

$$v_2 = v_3$$

is nothing but the clearance volume because the piston is at the top dead center corresponding to this location. Whereas v_1 and v_4, they are also equal because 4 to 1 is an isochoric process and this corresponds to the bottom dead center which is having the total volume.

And as per our definition,

$$\frac{V_t}{V_c} = r$$

where

r is the compression ratio.

So, this compression ratio here can be represented as v_1/v_2 as well or we can write in terms of v_3 and v_4 as well.

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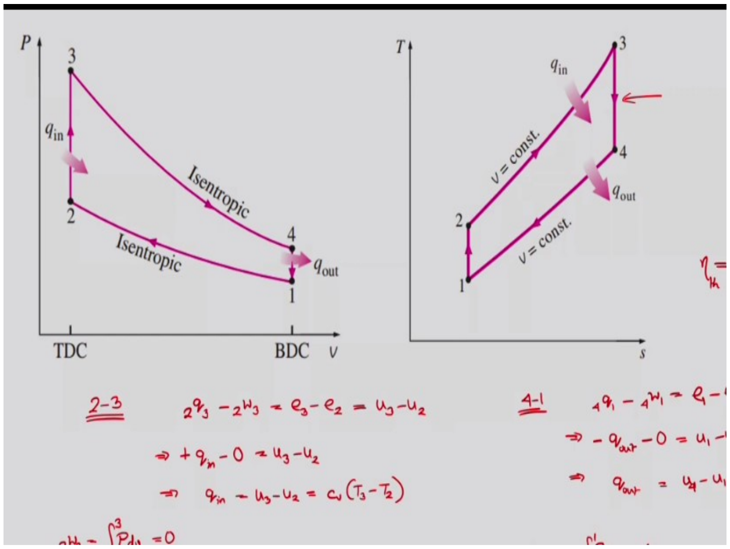

2-3 $\quad {}_2q_3 - {}_2w_3 = e_3 - e_2 = u_3 - u_2$

$\Rightarrow +q_{in} - 0 = u_3 - u_2$

$\Rightarrow q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$

${}_2w_3 = \int^3 Pdv = 0$

4-1 $\quad {}_4q_1 - {}_4w_1 = e_1 -$

$\Rightarrow -q_{out} - 0 = u_1 -$

$\Rightarrow q_{out} = u_4 - u_1$

So, this is the corresponding *Pv* diagram and *Ts* diagram. Look at this 1 to 2 and 3 to 4 are isentropic processes, so they are given by vertical lines. No heat transfer along them, only changes in the value of temperature, but entropy remains constant. Whereas during 2 to 3, temperature and entropy both are increasing as entropy is increasing. So, it signifies that is a heat addition process.

Because you have to remember that it is an internally reversible cycle. Therefore during 2 to 3 there is no entropy generation and so entropy of the system can only increase because of heat addition or entropy transfer with heat addition. Similarly, 4 to 1 entropy is decreasing which signifies this is a heat rejection process, because entropy can decrease during a reversible process only when heat is being rejected by the system.

So, we have to perform the mathematical analysis of this one quickly, something that we have done in the previous lecture but I would like to repeat that one because the mistake that I did there I do not want to do it once again in this. So, here we know that just what we have done earlier, the amount of heat transfer can be used to get the expression for the efficiency, sometimes called the thermal efficiency or first law efficiency which is:

$$\eta_{th} = \frac{w_{net}}{q_{in}}$$

Here, we are using w and small q to denote it is per unit mass. And using the first law expression we can write this as:

$$= 1 - \frac{q_{out}}{q_{in}}$$

So, we have to get q_{out} and q_{in}. Now, during process 2 to 3, we have heat addition; during 4 to 1, we have heat rejection whereas during 1 to 2 and 3 to 4, there is no heat transfer. So, we have to analyze each of the process individually to get this q_{out} and q_{in}.

As there is no heat transfer involved with two of the processes, let us keep them for the moment. We are just going to analyze 2 to 3 first. So, for 2 to 3 if we write the first law of thermodynamics, then we can have:

$${}_2q_3 - {}_2w_3 = e_3 - e_2$$

where

$e_3 - e_2$ is the rate of change of energy of the system

And as we are assuming generally assuming this one to be a stationary system, so this one can be written as:

$$= u_3 - u_2$$

as the changes in kinetic and potential energies of the system is negligible.

Now ${}_2q_3$ during this process, heat is being added so ${}_2q_3$ is typical:

$${}_2q_3 = q_{in}$$

${}_2w_3$can be considered as:

$${}_2w_3 = \int_2^3 Pdv = 0$$

now volume remains constant, so that becomes equal to 0. So, the expression becomes:

$$+ q_{in} - 0 = u_3 - u_2$$

which gives us:

$$q_{in} = u_3 - u_2$$

and as we are dealing with an ideal gas during where the change in internal energy can directly be related to the change in temperature in terms of the specific heat which is again assumed to be constant.

So, we can write this one to be:

$$= C_v (T_3 - T_2)$$

You can refer to our second week discussion where we developed expression for du. If you put the ideal gas equation, I think we did one exercise also. If you put the ideal gas equation of state there, you will find that du will be:

$$du = C_v \, dT$$

So, you get q_{in}. Instead of writing this way, you could have converted everything in terms of rate equation as well, that is our wish.

And now you have to analyze 4 to 1, if we want to write in rate equation, you have to put a dot on top of everything. There is *q dot* and *w dot* like this. For 4 to 1 what we can write is the same way:

$${}_4q_1 - {}_1w_4 = e_1 - e_4 = u_1 - u_4$$

Now, during this process heat is being rejected, so ${}_4q_1$as per our convention is:

$${}_4q_1 = -q_{out}$$

because we are taking heat rejected by the system to be negative.

Now ${}_1w_4$ refers to:

$${}_4w_1 = \int_4^1 P dv = 0$$

Again it is an isochoric process, so this is equal to 0.

So,

$$-q_{out} - 0 = u_1 - u_4$$

which gives us:

$$q_{out} = u_4 - u_1 = C_v (T_4 - T_1)$$

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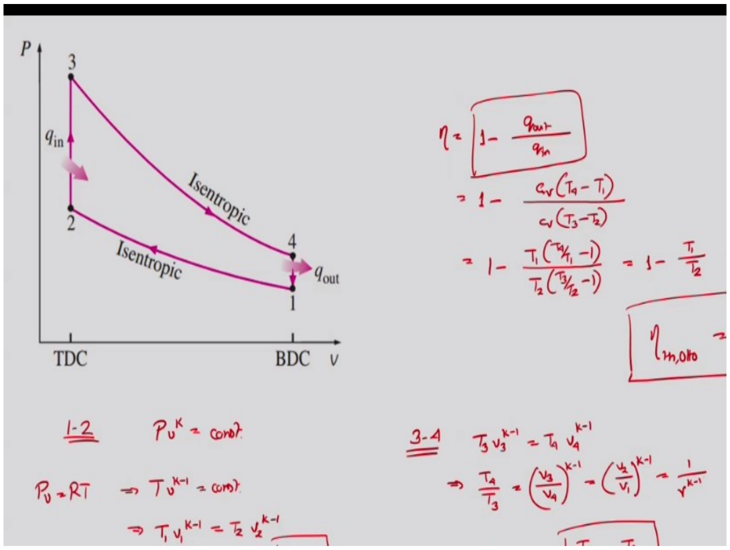

$$\eta = \boxed{1 - \frac{q_{out}}{q_{in}}}$$
$$= 1 - \frac{c_v(T_4 - T_1)}{c_v(T_3 - T_2)}$$
$$= 1 - \frac{T_1\left(T_4/T_1 - 1\right)}{T_2\left(T_3/T_2 - 1\right)} = 1 - \frac{T_1}{T_2}$$
$$\eta_{th,otto} =$$

1-2 $\quad Pv^k = const.$

$Pv = RT \quad \Rightarrow Tv^{k-1} = const.$

$\Rightarrow T_1 v_1^{k-1} = T_2 v_2^{k-1}$

3-4 $\quad T_3 v_3^{k-1} = T_4 v_4^{k-1}$

$\Rightarrow \frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1} = \left(\frac{v_2}{v_1}\right)^{k-1} = \frac{1}{r^{k-1}}$

Now, we go back to our expression for efficiency which is:

$$\eta = 1 - \frac{q_{out}}{q_{in}}$$

that is

$$= 1 - \frac{c_v(T_1 - T_4)}{c_v(T_3 - T_2)}$$

so C_v cancels out and by taking T_1 common from the numerator and T_2 from the denominator, we can write this one as:

$$= 1 - \frac{T_1\left({}^{T_4}/_{T_1} - 1\right)}{T_2\left({}^{T_3}/_{T_2} - 1\right)}$$

Now, to get the values for this temperature, we have to look to the process itself, other two processes rather these two isentropic processes.

Now, during 1 to 2 it is being an isentropic process, the process equation can often be represented as:

$$Pv^k = constant$$

where

k is the ratio of specific heat

P is the pressure

And from ideal gas equation of state, we know that

$$Pv = RT$$

If we impose it here, we can write this to be:

$$Tv^{k-1} = constant$$

So, if we use this now for the process 1 to 2, I would like to request you to check this from Pv^k, how we are getting Tv^{k-1} by using $Pv = RT$. R becomes a part of this constant now. We just have to replace P, so if we put this now:

$$Tv^{k-1} = constant$$

for process 1 to 2, we can write:

$$T_1 v_1^{k-1} = T_2 v_2^{k-1}$$

which gives us

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1}$$

And as you have seen from the previous slide:

$$\frac{v_2}{v_1} = r$$

so that gives us:

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \frac{1}{r^{k-1}}$$

Now, check out process 3 to 4, 3 to 4 is another isentropic process, so during this particular process, we have:

$$T_3 v_3^{k-1} = T_4 v_4^{k-1}$$

which gives us:

and considering the two isentropic processes, we know that:

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1} = \left(\frac{v_2}{v_1}\right)^{k-1} = \frac{1}{r^{k-1}}$$

We can write:

$$\frac{T_1}{T_2} = \frac{T_4}{T_3}$$

or if we reorient this we can write:

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}$$

So, if we take it back to the expression for η, then we are left with:

$$\eta = 1 - \frac{T_1}{T_2} = \frac{1}{r^{k-1}}$$

So, the thermal efficiency for the Otto cycle is a sole function of the compression ratio and the k for the substance.

Maybe we can write this way:

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

So it depends on only two things, one is the compression ratio and other is the k for the substance. For a given substance, k can be taken as a constant. Then, it depends only on the compression ratio. And like I have mentioned in the previous lecture, compression ratio is the first most important parameter that can govern the performance of system. So, that is what you have here. The efficiency is depending only on the compression ratio for a given substance. Now, this way you can calculate the efficiency and if your objective is to get some other parameters like if you want to calculate the work output, how you can get it? The net work output, w_{net} can easily be obtained as:

$$w_{net} = \eta_{th}\, q_{in}$$

So, this way we can calculate the work output. If your objective is to calculate the individual heat transfer values, then actually in the next lecture we shall be solving some numerical problems. But one important suggestion there I can give here that do not try to remember the expressions for these efficiencies. For Otto cycle, it is very simple, so probably you can remember this.

But the subsequent to that we shall be doing today that only makes it very difficult to remember those expressions. And there is every chance of making some mistake if you are trying to solve some numerical problem by remembering those equations. So, always try to come from the first principle i.e., just try to calculate the heat transfers. Say how much will be your q_{in}, the amount of heat transfer?

In this case, q_{in} corresponds like we have derived in the previous slide:

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

Similarly, your q_{out} will be:

$$q_{out} = c_v(T_4 - T_1)$$

So, if you know this term, know the temperatures involved in the cycle, then you can easily calculate q_{in} and q_{out}. Now,

$$\oint \delta q = q_{in} - q_{out}$$

and as per the first law of cycle that has to be:

$$= w_{net}$$

And once you know q_{in} and q_{out}, again from here you can easily calculate the efficiency without trying to remember this. So, this way several things can be calculated from the cycle, just from the knowledge of this q_{in} and q_{out}.

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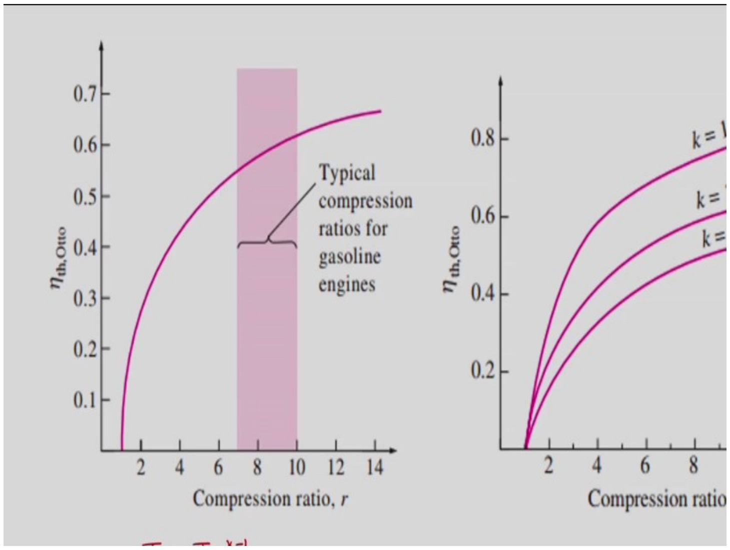

Now, if we plot the expression for η_{th} with R, then we can get a curve like this. You can see with increasing compression ratio efficiency keeps on increasing sharply or at least for lower values of compression ratio, the slope is very steep like in this portion. However, it settles down quite a bit in here. So, as we go to higher compression ratio, the incremental increase in this thermal efficiency is lower. But you never reach a point of inflection, it keeps on increasing continuously.

Similarly, the effect of k is something like this, smaller the value of k higher will be the efficiency for a given compression ratio. So, if we have two identical SI engines, one working on say helium which is a monatomic gas and other working on say air which can be visualized to be a diatomic gas. Then, for the helium one is going to give you higher efficiency for the same compression ratio because it has a lower value of k. This corresponds to roughly monatomic gases. This is approximately for diatomic gases; this is approximately for triatomic gases. So k keeps on, as k keeps on increasing, the thermal efficiency keeps on reducing.

In practical engines, we generally have heavier molecules like carbon dioxide or carbon monoxide or sometimes methane etc which have much lower value of k in the range of 1.2 or even lower. So, their efficiency will definitely be much lower compared to these ideal cycles. Now, you can see here, I have given a range of typical compression ratios for gasoline engine. If we can get higher efficiency by working at a higher compression ratio, then why you do not need to, why we are not going for such very high compression ratio? Why you are restricting it to such a narrow band something like 7 to 10 as shown here.

Practical SI engines hardly have compression ratio greater than 10 or 12. That is because of the temperature consideration. You can see from the previous slides that as the compression ratio increases, the temperature also increases because we have just seen that:

$$T_2 = T_1 r^{k-1}$$

So, as r increases T_2 keeps on increasing, so higher the compression ratio temperature of the gas or air at the end of compression that also keeps on increasing.

Now, in ideal case, your working medium is air, so there is no issue but in actual cycle we have some hydrocarbon fuel. And every hydrocarbon fuel has autoignition or self-ignition temperature. Now, if during the compression process, we said the compression ratio such that the T_2 becomes higher than the self-ignition temperature. Then, before the compression process finishes, the mixture or your fuel will start to self-ignite, which is highly undesirable process. And that is why we want to restrict our compression ratio such that this autoignition can be avoided. We said compression ratio such that T_2 should always be greater than the autoignition temperature for the fuel that you are dealing with, so that the autoignition can be avoided.

If the system or if the fuel auto ignites, then that leads to something known as the engine knocking. There is one number called Octane number, I cannot discuss this here because that is a topic that is beyond our scope. But Octane number generally is a criteria that is related to this autoignition temperature and that identifies whether the fuel is going to auto ignite or not for a given compression ratio range for this.

So, but in a nutshell, keeping the autoignition temperature in mind, we have to keep the compression ratio limited. Several attempts have been done to improve this Octane number and thereby to increase the autoignition temperature. Like since 1920's, people started adding lead to the gasoline fuels which is a quite inexpensive option of increasing the Octane number.

Addition of lead has shown to increase this autoignition temperature quite markedly. However, when we came to 1970's, it has been observed that because of the presence of lead during the combustion process lead participates in the chemical reaction and produces several very harmful gases, which are harmful to our health and of course leads to several environmental pollutions. And that is why since 1975 gradually the lead has been phased out from the gasoline because of the international protocol.

In fact, whatever automobile vehicles that were fabricated after 1975, they are all suitable for unleaded gasoline and so we are back to lower compression ratios. Nowadays, people use several kinds of gasoline blends etc, which has been able to increase the autoignition temperature a bit and thereby causing or affecting some small amount of change in the effective range of compression ratios but still some way to give.

This is a very active area of research and quite open area of research where we are looking to increase the autoignition temperature of the fuel by making some blend or by adding something to the raw gasoline and so that we can cause some increase in the compression ratio. Even one level increase in the compression ratio can cause some 2 - 3% change in this Otto cycle efficiency which actually can lead to huge amount of fuel saving, that is why it is extremely important.

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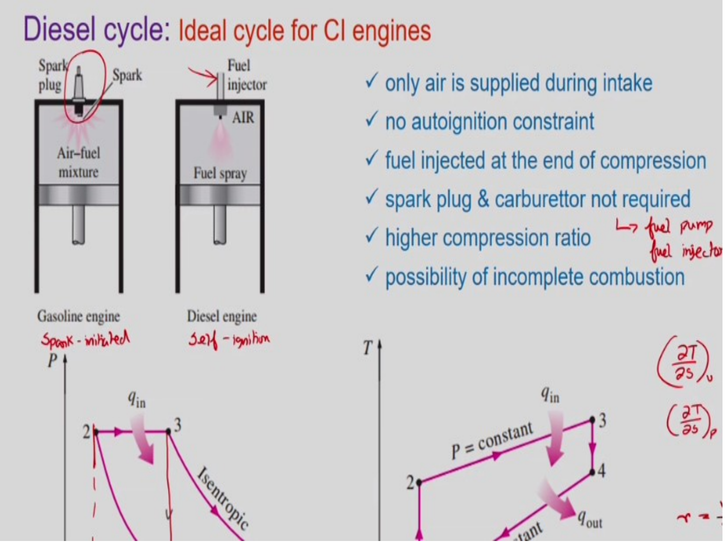

Diesel cycle: Ideal cycle for CI engines

- ✓ only air is supplied during intake
- ✓ no autoignition constraint
- ✓ fuel injected at the end of compression
- ✓ spark plug & carburettor not required
- ✓ higher compression ratio ↳ fuel pump fuel injector
- ✓ possibility of incomplete combustion

Now, we move to the second one which is the Diesel cycle called the ideal cycle for CI engines. Where in fact we can go beyond this autoignition temperature limit and go to much higher compression ratios. As the name suggests, it is the engine which uses diesel group of fuels as the working fuel. Rudolf Diesel, the picture shown here, he was the German scientist was the first inventor of the CI engine concept.

The most important difference between SI engine and CI engine are in this ignition process. In case of SI engine, we have a fuel air mixture which is generally prepared inside a carburetor that is supplied during the intake process. So, during an intake process if fresh fuel air mixture is supplied to the cylinder, now during the compression process, the fuel air mixture is compressed to a high pressure and temperature limit or temperature level.

However, we have to limit the temperature below the autoignition temperature, then we produce a spark so that the combustion is initiated. In case of Diesel engines, during the intake process, we supply only air. We supply only air, no fuel during the intake process. So, it is only air which is supplied quite similar to the ideal one. It is air which is supplied during the intake process; it is air which goes through the compression process. As there is no fuel inside, so we can go to any temperature limit during the compression process. There is no autoignition temperature limit, so we can easily heat air to very high temperature or I should say we can compress air to very high-pressure levels, so that the temperature can be well above the temperature of the corresponding autoignition temperature of the fuel.

And now at the end of this compression process, there is a fuel injector. In case of SI engine, we had this spark plug. Here, we have the fuel injector. Here, liquid fuel that is diesel which is sprayed inside the combustion chamber at the end of the compression process. As the liquid fuel comes inside, the fuel droplets are immediately exposed to an environment which is maintained at a temperature well above its self-ignition temperature, so they auto ignite, so self-ignites.

Therefore, self-ignition is actually desirable in case of diesel engines. So, in case of gasoline engines, the ignition is initiated by a spark plug, so we have spark-initiated combustion or spark ignition. Whereas we have self-ignition here in case of Diesel engine. In order to have this self-ignition only, we supply only air during the intake process and compress that air to a very high temperature and then the fuel is sprayed inside.

That is a major difference in CI engines. So, it is only air which is supplied during the intake process, air is also compressed because there is no fuel inside, so we do not have to be bothered about the autoignition temperature limit. We can compress air to very high-pressure levels and therefore we can go to much higher compression ratios in case of CI engines. Fuel is injected at the end of compression process as the fuel comes in, invariably the temperature is well above its autoignition temperature and therefore the fuel auto ignites.

Then, spark plug and carburetors are not required because we do not need to produce any spark, so spark plug is not required and carburetor is required to prepare a fuel air mixture. Now, as we are not preparing fuel air mixture, air is supplied at the beginning of the inlet process and fuel is sprayed after the compression process. So, carburetor is also not required but of course we have a fuel injector and we generally also need to have a fuel pump.

Because the pressure of this fuel also needs to be higher than the pressure of air for it to be sprayed inside this. So, we need to have a fuel pump and a fuel injector or spray gun which are in fact much costlier and heavier compared to the spark plugs and carburetors and higher compression ratio as I mentioned. But one drawback here is there is possibility of incomplete combustion. Because in case of SI engine, the fuel and air are uniformly mixed in the carburetor before supplying to the cylinder. So, every fuel molecule generally has sufficient number of air molecules around it. But here in Diesel engine as a fuel is being sprayed inside, so it is very much possible that some of the fuel droplets or fuel molecules may get sufficient amount of oxygen in a neighborhood and some of the droplets may not get any.

So, it is possible some amount of fuel may remain unburned leading to incomplete combustion, loss of fuel and subsequent pollution problems which is not that much significant for SI engines. So, spark plug and carburetors are not there, instead of that we have fuel pump and fuel injector, which are costlier and heavier compared to spark plugs and carburetors.

So, here the combustion as I have mentioned earlier, here the heat addition in ideal cycle is visualized in the mode of constant pressure. Because in case of SI engine as soon as we are initiating the spark, the combustion happens. And it happens almost instantaneously before the piston starts to move again that is why we assume it to be a constant volume combustion. However, in case of CI engines as we are spraying the fuel that requires a substantial amount of time to supply a fixed mass of fuel.

And by that time, the piston definitely keeps on moving. So, we generally try to spray the fuel such that the pressure inside remains constant thereby achieving this combustion process at constant pressure. However, there is change in volume. Like this is your bottom dead center location and this is the top dead center location. If I draw like this, this is your BDC and this is your TDC.

And this combustion process actually happens over this particular volume, so there is a good amount of change in the volume of this. And the total volume available for the expansion process is only this much. Whereas the entire change from clearance volume to total volume is available for expansion in SI engine, in CI engine that is lesser. We of course want to reduce this one to a smaller level, but that depends upon how much mass of fuel we have to inject and also the design of the injector.

This is the corresponding *Ts* diagram. One important consideration that we must do here, that is the slope of this constant pressure line. Now, from our earlier discussion, we know that the slope on a *Ts* diagram, the slope of a constant volume line is generally given by:

$$\left(\frac{\partial T}{\partial s}\right)_v = \frac{1}{c_v}$$

whereas the slope of a constant pressure line is given as:

$$\left(\frac{\partial T}{\partial s}\right)_P = \frac{1}{c_p}$$

Now, we know that

$$C_p > C_v$$

so

$$1/ C_p < C_v$$

Therefore, the slope of a constant pressure line is less than the slope of a constant volume line. So, from point 2, we have to draw a constant pressure line which is shown here for this. And here to characterize along with the compression ratio, we have to define another parameter. So, the compression ratio here remains the same i.e., the ratio of volume before compression to volume after compression:

$$r = \frac{v_1}{v_2}$$

Here, we define a new parameter called the cutoff ratio. Cutoff ratio given by rc which is given by volume after combustion divided by volume before combustion i.e.,

$$r_c = \frac{v_3}{v_2}$$

or volume at the end of combustion to volume at the beginning of combustion. This is a new parameter that is required for Diesel engines.

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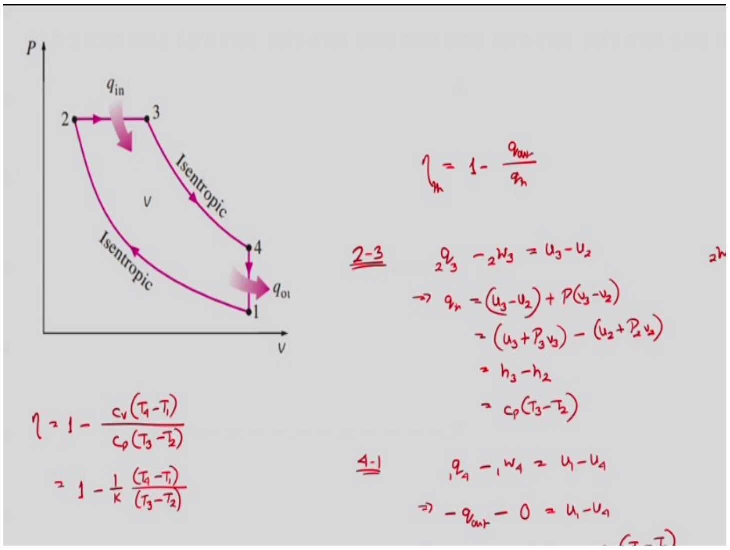

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

2-3 $\quad {}_2q_3 - {}_2w_3 = u_3 - u_2$

$$\Rightarrow q_{in} = (u_3 - u_2) + P(v_3 - v_2)$$
$$= (u_3 + P_3v_3) - (u_2 + P_2v_2)$$
$$= h_3 - h_2$$
$$= c_p(T_3 - T_2)$$

4-1 $\quad {}_1q_4 - {}_1w_4 = u_1 - u_4$

$$\Rightarrow -q_{out} - 0 = u_1 - u_4$$

$$\eta = 1 - \frac{c_v(T_4 - T_1)}{c_p(T_3 - T_2)}$$
$$= 1 - \frac{1}{k}\frac{(T_4 - T_1)}{(T_3 - T_2)}$$

So, if we analyze it again, so we shall be trying to do it quickly. So, we have the thermal efficiency is again:

$$\eta = 1 - \frac{q_{out}}{q_{in}}$$

for process 2 to 3,

$${}_2q_3 - \quad {}_2w_3 = u_3 - u_2$$

as earlier.

$${}_2q_3 = q_{in}$$

$${}_2w_3 \neq 0$$

in this case because 0 is not a constant volume process.

Now

$${}_2w_3 = \int_2^3 Pdv$$

As the pressure remains constant during this process, we can write this to be equal to:

$$= P(v_3 - v_2)$$

so,

$$q_{in} = (u_3 - u_2) + P(v_3 - v_2)$$

That is we can write this to be equal to:

$$= (u_3 + P_3v_3) - (u_2 + P_2v_2)$$

as we have

$$P_2 = P_3$$

what is $u + Pv$? That is enthalpy. So, it is:

$$= h_3 - h_2$$

and for an ideal gas we know this is:

$$= c_p(T_3 - T_2)$$

Similarly, for process 4 to 1, we can write the same way whatever we have done earlier:

$${}_1q_4 - \quad {}_1w_4 = u_1 - u_4$$

Now

$${}_4q_1 = -q_{out}$$

It is a constant volume process. So, it is equal to 0

So,

$$q_{out} = (u_4 - u_1) = c_v(T_4 - T_1)$$

If we want to analyze this, then:

$$\eta = 1 - \frac{c_v(T_1 - T_4)}{c_p(T_3 - T_2)}$$

that is C_p / C_v is k, so we can write it to be $1/k$ into the same way. If we take T_1 common from here okay, there is no need of doing this for the moment now. Let us just write it as the ratio of temperatures i.e.,

$$= 1 - \frac{1}{k}\frac{(T_1 - T_4)}{(T_3 - T_2)}$$

Now, we have to evaluate each of these temperatures.

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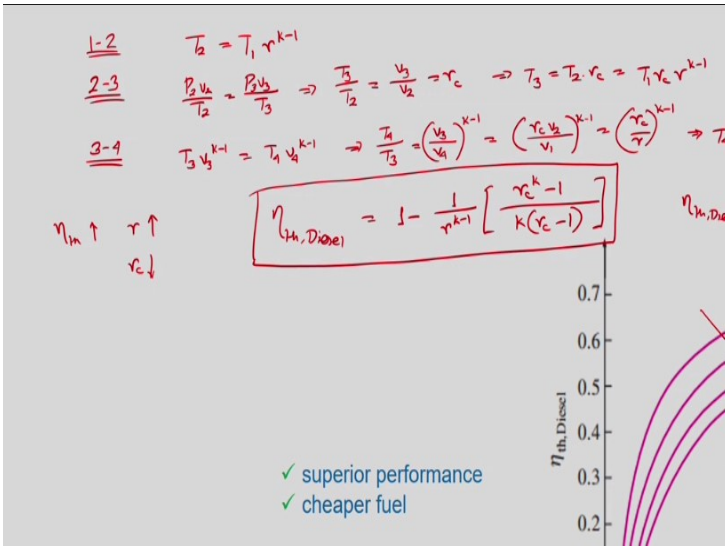

1-2 $\quad T_2 = T_1 r^{k-1}$

2-3 $\quad \frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3} \Rightarrow \frac{T_3}{T_2} = \frac{V_3}{V_2} = r_c \Rightarrow T_3 = T_2 \cdot r_c = T_1 r_c r^{k-1}$

3-4 $\quad T_3 V_3^{k-1} = T_4 V_4^{k-1} \Rightarrow \frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} = \left(\frac{r_c V_2}{V_1}\right)^{k-1} = \left(\frac{r_c}{r}\right)^{k-1} \Rightarrow T$

$\eta_{th} \uparrow \quad r \uparrow$

$r_c \downarrow$

$$\boxed{\eta_{th,Diesel} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right]}$$

✓ superior performance

✓ cheaper fuel

So, let us try to analyze each of the process one upon one. Now 1 to 2 is an isentropic process. So, for an isentropic process, we have already seen that we can write:

$$T_2 = T_1 r^{k-1}$$

2 to 3 is a constant pressure process. So, for a constant pressure process, we can write using the ideal gas equation of state, we can write:

$$\frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3}$$

That is, P_2 and P_3 are equal to each other because it is a constant pressure process. So,

$$\frac{T_3}{T_2} = \frac{v_3}{v_2} = r_c$$

which is equal to r_c which we have just defined. So,

$$T_3 = T_2 r_c \quad = T_1 r_c^{k-1}$$

that is for process 2 to 3. Similarly, for process 3 to 4, look at the diagram, 3 to 4 is an isentropic process. So, for an isentropic process just like we have written earlier:

$$T_3 v_3^{k-1} = T_4 v_4^{k-1}$$

that is,

$$\frac{T_4}{T_3} = \left(\frac{v_3}{v_4}\right)^{k-1}$$

And what is your v_3?

$$v_3 = r_c v_2$$

and v_4, what is your v_4? Look at this diagram, $_4v_1$ and $_1v_4$ is a constant volume process. So v_4 and v_1 are equal. So,

$$\frac{T_4}{T_3} = \left(\frac{r_c v_2}{v_1}\right)^{k-1} = \left(\frac{r_c}{r}\right)^{k-1}$$

from where you can actually put back the T_3 expression in terms of T_1. Now, if you combine all of them, then I am directly writing the expression for the efficiency for a Diesel cycle. You just have to take these three expressions for T_2, T_3 and T_4 and put it here. And if you do it, you are going to get the thermal efficiency for an ideal Diesel engine to be:

$$\eta_{th,Diesel} = 1 - \frac{1}{r^{k-1}}\left[\frac{{r_c}^{k-1}}{k(r_c - 1)}\right]$$

This is the expression for the ideal Diesel cycle. If we plot it in terms of compression ratio, again you will be seeing that for a given value of r_c, r keeps on increasing.

Typical compression ratio for diesel engines is much larger, you can see the range is 12 to 23, 24 in that range which is well above the same for SI engines. CI engines are used in heavy vehicles where you need larger powers like in case of buses, trucks, ships, etc where you go for CI engines. Whereas SI engines are commonly used in two-wheelers, smaller passenger cars, etc. Electric trains can also be an example of CI engines.

Whereas SI engines we generally are restricted to low power applications, because the compression ratio is lower. One important or interesting thing that you can see, if we put r_c = 1 of course this expression will lead to a 0/0 situation. But r_c = 1 actually refers to an Otto cycle and for r_c = 1, you get the highest efficiency for a given value of compression ratio.

In fact, with increase in cutoff ratio, the efficiency decreases. Of course, it increases with compression ratio, so your efficiency, it increases with as r increases but r_c has to decrease for this and $r_c = 1$ refers to the Otto cycle. You can see from the expression also, the term in this square bracket, here r_c is always greater than 1, k is also greater than 1.

So, the term in the square bracket will actually will always be greater than 1. If the term is greater than 1, then from here we can easily see for the given value of our:

$$\eta_{th,Diesel} < \eta_{th,Otto}$$

But, practically speaking the Diesel cycles or diesel engines always operate with much higher compression ratio than Otto cycle based engines and therefore they offer higher compression ratio.

So, the advantage of CI engines or superior performance because they work at higher compression ratio, they give higher work output, much better efficiency and that is why they found application in bigger commercial vehicles. Another commercial reason is the fuel is cheaper. As we are spraying the fuel inside, so we can use a heavier fuel and also less refined fuel whereas in SI engines, we need very fine fuel.

That is why we have to go for petrol or gasoline which is a better grade or superior grade of petroleum. But the disadvantages of the CI engines are for the given compression ratio, we can see that efficiency is lower, there heavier and costlier like as they work in much higher compression ratio, so the pressure range inside the engines are higher. So, the engine structure has to be much stronger.

Components like fuel pumps and injectors etc, also heavier and much costlier compared to spark plug etc. They are pollution prone because of the incomplete combustion and also because of the higher-pressure range, heavier structure, vibration is another problem. That is one of the big issues of not applying CI engines to two wheelers or smaller cars, the vibration related problems etc.

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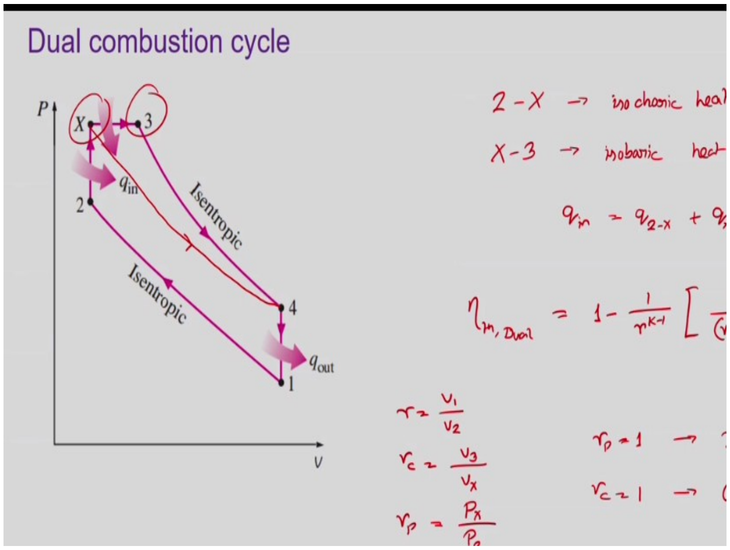

Dual combustion cycle

2 - X → isochoric hea

X - 3 → isobaric heat

$q_{in} = q_{2-x} + q$

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

$r = \frac{v_1}{v_2}$

$r_c = \frac{v_3}{v_x}$

$r_p = \frac{P_x}{P_2}$

$r_p = 1$ →

$r_c = 1$ →

Now, we have seen that in SI engines or sorry in Otto cycle we considered the combustion to happen in constant volume. Whereas in Diesel cycle we have seen to be happening at constant pressure. Practically, both of them are not correct. Practically, combustion happens partially in constant volume mode and partially in constant pressure mode. So, somewhat intermediate cycle is defined in the form of the dual combustion cycle where it is visualized that combustion is partly happening at constant volume and partly at constant pressure.

Here, this 2 - *x* refers to an isochoric heat addition part whereas this *x* - 3 refers to an isobaric heat addition part. So, total heat added will be heat added during the process:

$$q_{in} = q_{2-x} + q_{x-3}$$

The heat rejection of course remains the same that is at constant volume. The same way you can perform analysis of this one. I am leaving this to you as we have already done the analysis for Otto cycle and Diesel cycle step by step.

You will surely be able to do this. I am just giving you the expression for this dual cycle, the expression for efficiency. It will be:

$$\eta_{th,Dual} = 1 - \frac{1}{r^{k-1}} \left[\frac{r_p {r_c}^{k-1}}{(r_p - 1) + k r_p (r_c - 1)} \right]$$

Here, this *r* remains the same, that is volume before compression/volume after compression $v_1/v_2 > 1$.

$$r = \frac{v_1}{v_2}$$

r_c is the cut off ratio which corresponds to the isobaric part of heat addition. It refers to volume after constant pressure heat addition/volume before constant pressure heat addition which is v_x.

$$r_c = \frac{v_3}{v_x}$$

r_p is called the pressure ratio, it refers to the constant volume heat addition part, it refers to the pressure after constant volume heat addition/pressure before constant volume heat addition.

$$r_p = \frac{p_x}{p_2}$$

you can see all of them are actually greater than 1 and this expression for dual cycle is a good amalgamation of both Otto and Diesel cycle.

Like if we put r_p = 1 in this, just put r_p = 1, see what you are getting. When you are putting r_p = 1, it reduces to the Diesel cycle, r_p = 1 means this point x collapses to point 2, so your 2 to 3 line will be something like this which is just what we have in case of Diesel cycle. Whereas if we put r_c = 1, it collapses to Otto cycle, r_c = 1 means point x and point 3, they collapse on each other, so the expansion start from this particular point onwards only.

So, this will be your expansion stroke in that case. So, you can see from the mathematical expression also. It is a good combination of both the cycles. And efficiency point of view for a given compression ratio, we can always show that thermal efficiency for the Otto cycle is greater than the efficiency for the dual cycle and that is greater than efficiency for the Diesel cycle.

Now, we shall quickly be trying to compare the Otto and Diesel cycle in terms of a few parameters.

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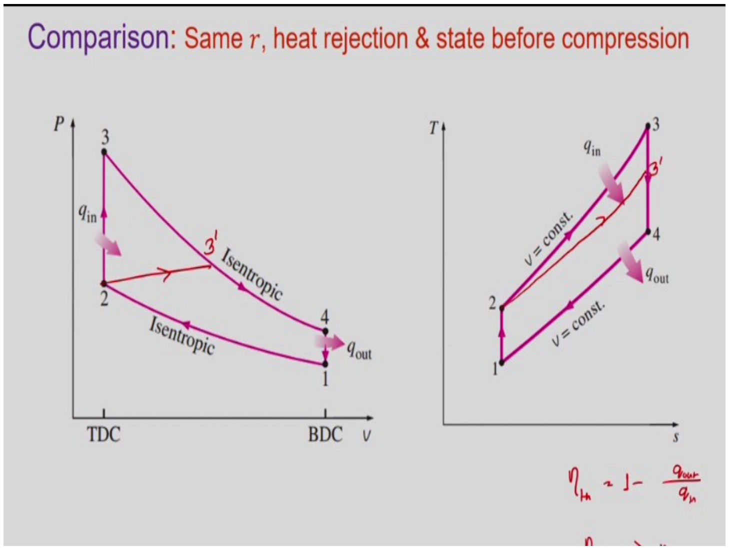

Comparison: Same r, heat rejection & state before compression

$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$

First set of comparison where we have the same compression ratio, same amount of heat rejection and same state before compression. The Otto cycle diagrams are shown, you have to plot the Diesel cycle on top of this. We are talking about same compression ratio and same state before compression, same state means point 1 is same, same compression ratio. So point 2 is also same means 1-2, the process remains same, same amount of heat rejection.

Same amount of heat rejection means as per the *Ts* diagram, the area under this particular curve 4-1 has to be the same, so process 4- 1 is also same. That is 1- 2 and 4-1 are same that is point 1-4 they are same between Otto and Diesel cycle, but point 3 can be different. So, we have to see how we can show this, how we can do this. So, Otto cycle is shown here.

If we want to retain point 1, 2 and 4 the same, then how we can have a Diesel cycle? Your Diesel cycle will be given by this particular line, this point 3' or if we draw it on the *Ts* plane from point 2, we have to draw a constant pressure line. Constant volume line is already shown. If we draw a constant pressure line and see where it cuts this 3-4 line, this is your 3' which is the Diesel cycle.

Now, for this particular set of comparison which cycle is having higher efficiency, just check out. We generally would like to do it from heat transfer point of view.

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

Now, here q_{out} is same for both the cycles, which one is having higher q_{in}, which area under the heat addition curve is greater for the Otto cycle because there the heat addition curve is 2 to 3, area under 2 to 3 is definitely greater than the area under 2 to 3'.

So, q_{in} is greater for this and from there we can easily say that the Otto cycle efficiency will be greater than the Diesel cycle efficiency in terms of this set of comparison.

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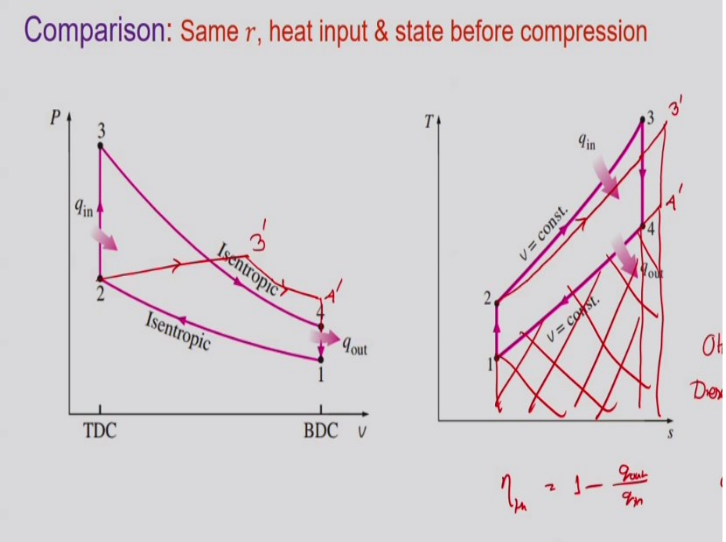

Comparison: Same r, heat input & state before compression

$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$

Whereas if we draw another set of comparison where we again have the same compression ratio, same state before compression that is point 1 and 2 or 1-2, they are same but different amount of heat input. Process 2-3 same amount of heat input, but heat rejection may be different, so how we can plot this? Otto cycle is shown same magnitude of heat input we have to provide. Now, the area under the curve 2-3 shows the heat added to the Otto cycle. Same amount of heat we have to add to the Diesel cycle as well. So, this is the constant pressure line. If the constant pressure line finishes here, the area under this curve is lesser.

So, we have to extend this such that the area under this curve becomes similar to the area under this 2-3 curve. So, this becomes your new 3' point, so like this way, we have to plot your 4' also get shifted here. So, your Otto cycle diagram is 1-2-3-4-1. Diesel cycle diagram now becomes 1-2-3'-4'-1. If we want to draw the same thing on incase of the Pv diagram, then we have to draw it somewhat like this, extend up to some point somewhere here 3' which goes to something like this sound like this and then we have 4' somewhere here. So 1-2-3'-4'-1 which corresponds to same compression ratio, same state before compression and same amount of heat input but different amount of heat rejection. Now, again we compare the efficiencies:

$$\eta_{th} = 1 - \frac{q_{out}}{q_{in}}$$

Here, q_{in} is same, which one is having larger q_{out}. Look at the diagram q_{out} refers to area under the heat rejection curve.

For the Otto cycle, heat rejection is 1-4 or I should say 4-1. So, this area is the heat rejection corresponds to the Otto cycle. How much is for the Diesel cycle? For Diesel cycle, your area is this much, so q_{out} for Diesel is greater than q_{out} for Otto. So, here again as it is having larger heat rejection but same amount of heat input, so

$$\eta_{Otto} > \eta_{Diesel},$$

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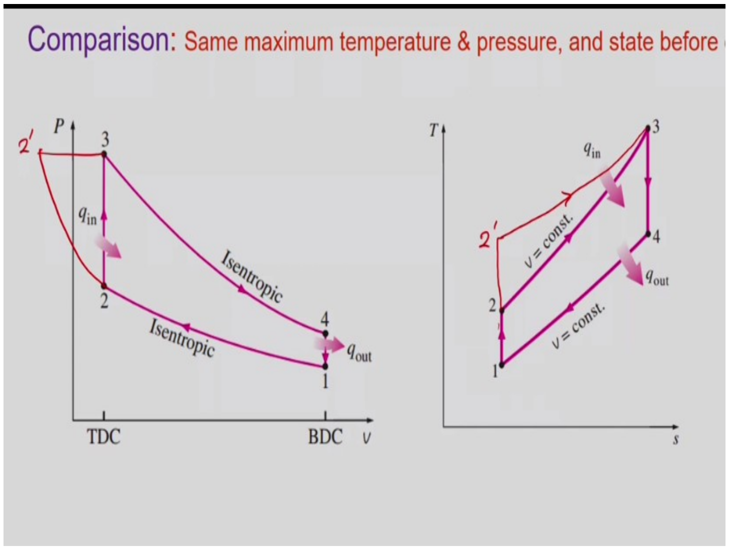

Comparison: Same maximum temperature & pressure, and state before

And one final comparison I would like to do which is very important from practical point of view where compression ratios are not same, state before compression is same which is this point 1 and the maximum pressure and temperature remains the same. Now, which point in the cycle has the same maximum pressure and temperature? Point 3, this is the one corresponds to maximum pressure and temperature. So 1 and 3 are same but other points are different.

So, how we can compare them? Again, the Otto cycle is shown to plot the Diesel cycle here point 1 remains the same. Let us extend this line vertically this line goes like this. And through point 3 constant volume line is already there which leads to point 2. We backtrack a constant pressure line, so a constant pressure line will be somewhat like this. This leads to your point 2'; this is your 2', 3 and 4.

Similarly, here actually it is going out of the scale. If you want to plot it here, you have to continue up to this and then somewhat like this. So, this will become your point 2' and 3 remains the same, 4 actually also remains the same. So, which one is having higher amount of heat rejection in this case? Both are having same heat rejection. What heat input, q_{in} for the diesel and q_{in} for Otto?

Which one is higher? q_{in} for the diesel is definitely much higher in this particular case because the area under the curve 2'-3 is much larger than the area under the curve 2-3. So, efficiency in this case for the diesel will be greater than the efficiency for the Otto cycle for this particular case. These are several other comparisons; also we can think of, you can also try to think of a few other comparisons.

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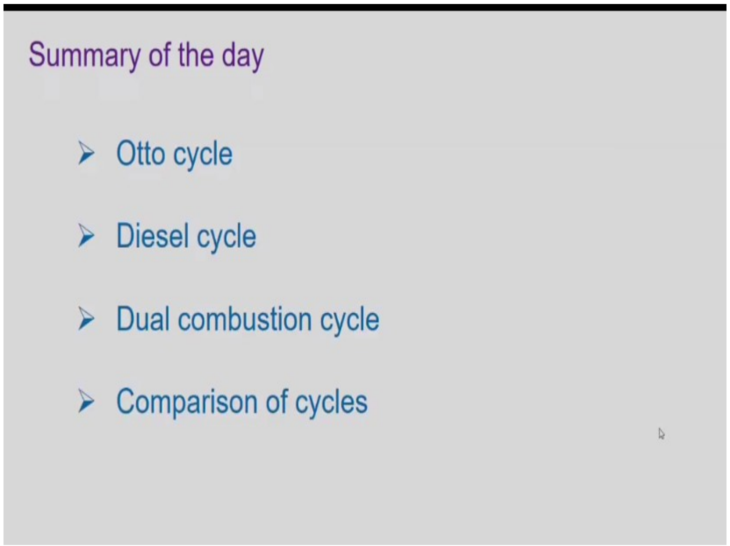

Summary of the day

- Otto cycle
- Diesel cycle
- Dual combustion cycle
- Comparison of cycles

That takes us to the end of today's lecture where we have discussed about the Otto, Diesel and dual combustion cycles and also compare these cycles in terms of three different criterion. So, in the next lecture, I would like to first solve a few numerical problems to show the application of these cycles and how to approach the solution without trying to remember the mathematical formulation. And then we shall be discussing about two more cycles, the Stirling and Ericsson cycles. Till then, goodbye.