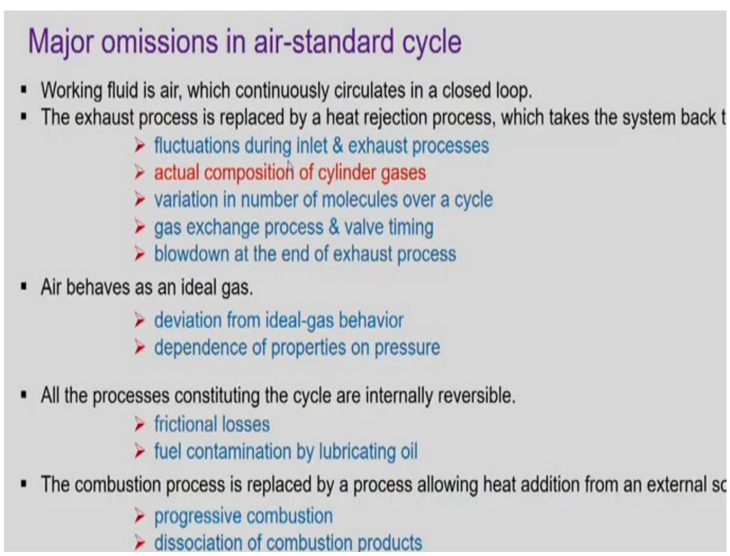
Applied Thermodynamics for Engineers
Dipankar N. Basu
Department of Mechanical Engineering
Indian Institute of Technology – Guwahati

Lecture – 16
Losses in Actual Cycle and Valve-Timing Diagram

Morning friends, so we are meeting for the third time this week where we are talking about the real or actual cycles for reciprocating engines. Previous week we have discussed about the air standard cycles, we have talked about the ideal cycles for SI engines. And there we have considered several assumptions, making it a highly ideal kind of cycle. Now in the first week of this lecture we have talked about the fuel-air cycle which is an improvement over those ideal cycles or air standard cycles, where we relaxed quite a few certain assumptions they are way allowing us to take care of several practical considerations.

And therefore, what predictions that we get from fuel-air cycle that is much closer to what we get in real practice. But still there are quite a few other factors that is not considered even in fuel-air cycle as well. And that creates some kind of difference between the fuel-air cycle prediction and the actual cycle performance. And in today's lecture we shall be looking primarily to those factors which are not considered in fuel air cycle as well.

(Refer Slide Time: 01:30)



So, this is one slide that I have shown earlier also in the very first lecture of this week actually in the second lecture we just focused on the numerical examples of using the fuel-air concept in real cycles. Now there you can see, these are the assumptions the ones written in black are the ones that are considered in air standard cycles and the ones written in blue are several limitations corresponding to those assumptions.

Now in case of fuel-air cycle, quite a few of them actually four of the assumptions we have realized as the most important factors in influencing the engine performance and also it is possible to take care of them through theoretical analysis. The first one is the actual composition of cylinder gases. As we have seen in the numerical examples that you have discussed in fuel-air cycle we can consider real chemical reactions, we can talk about the fuel-air ratio, the fuel air mixture before the combustion process, the products of combustion after the combustion process, though we have not dealt with it but it is also possible to take care of the actual properties of such kind of mixtures. And so, we can deal with the real gas composition or gas environment that persists inside a real engine. Then we can also take care of the variation in the number of molecules over a cycle.

Like we have seen the examples it is possible that during a chemical reaction we can have an increase in the number of molecules on the product side compared to the reactant side or a decrease in the number of molecules leading to molecular contraction or expansion. And at a given pressure and volume or I should say at a given temperature and volume, pressure being directly proportional to the number of moles, pressure keeps on changing along with such kind of molecular contraction or expansion, which can have a direct effect on the pressure magnitude at a particular state point. Such examples also we have solved in the previous lecture where we dealt with different numerical problems. A third factor that was considered is the dissociation of combustion products here. So, the dissociation of combustion products though we have not solved in numerical problems, but at very high temperature we know the products of combustion can participate in reverse reaction like carbon dioxide can get broken into carbon monoxide and oxygen, or water vapour can get broken into hydrogen and oxygen, which are endothermic reactions thereby causing a reduction in the final temperature of the combustion products, and thereby the pressure of the combustion products.

This one can be taken care of by fuel-air cycle and a fourth one, one of the most important assumptions for the air standard cycle i.e., the constant specific heat that was relaxed that the temperature dependence nature of specific heats can be considered in fuel-air cycle. And we have done numerical examples there also. But apart from these four considerations there are still quite a few factors left out.

And some of them can really influence the engine performance quite a bit. Therefore, as we have mentioned the thermal efficiency for an actual cycle quite often is found to be within the range of 80 to 90 % of whatever is predicted by the fuel-air cycle. And therefore, there is a 10 to 20% difference in the actual performance compared to the one predicted by fuel-air cycle.

Remember, similar to a standard cycle fuel-air cycle is also an ideal cycle or a theoretical cycle. And one the theoretical cycle whatever performance prediction that we will get from the theoretical cycle, we have to add a few more factors which will cause a 10 to 20% reduction in the final efficiency or true efficiency that we are getting. And that true efficiency or that difference in the efficiency that comes because of the factors which are still left in blue on this slide.

And out of them, of course all of them are not of equal importance but some of them can really dominate this 10 to 20 % difference. One of the most dominating factors can be this progressive combustion and incomplete combustion. A combustion reaction we are assuming ideal cycle and also in fuel-air cycle to be instantaneous. Like in case of SI engines, we assume the inter combustion to takes place at constant volume, but that is not true in practice.

Because the mixing of fuel and air mixture and also the different chemical reactions that takes place during combustion, they require some amount of finite time. Whatever small it may be but considering the speed at which each of the cycle operates, that time can be a significant fraction of the suction or compression stroke or I should say the compression and expansion stroke.

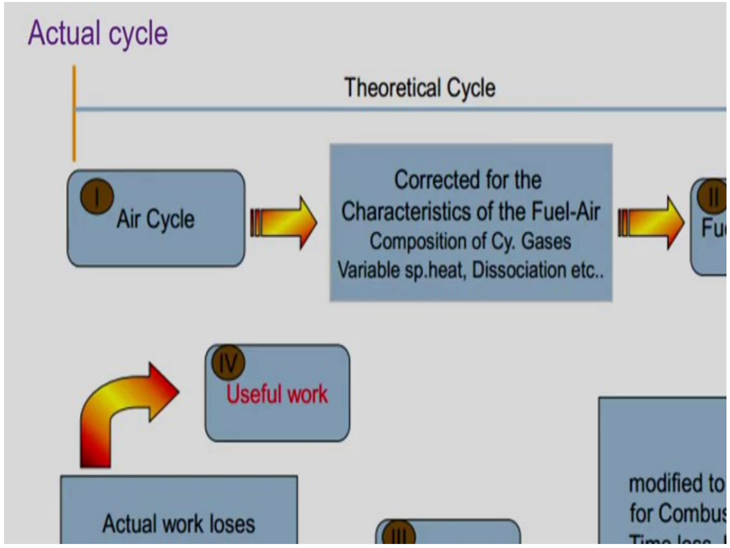
And also, the incomplete combustion can lead to release of lesser amount of energy than what we get in theoretically and so can also affect the final pressure and temperature. So, these two are very important factors and this progressive combustion and incomplete combustion together they are generally called time loss. Or I should say, the loss in efficiency and work output because of this progressive combustion and incomplete combustion are referred to as the time loss.

This is one of the most important factor, I should identify this as number one important factor is the difference between the actual cycle performance and fuel air cycle prediction. Another one very important factor is this one, the heat transfer to the cylinder walls which can be identified the number 2. The heat transfer from the combustion gases to the surrounding air can never be stopped, whatever insulation you may put in but there will be some amount of thermal energy leakage which can directly affect the total energy content of the gases thereby directly affecting its temperature. So, this is another important factor. And the third factor quite often considered is a combination of these two. The gas exchange during the process and the blow down at the end of exhaust process. These two together can be identified as number 3. We can still have other factors which are left out like fluctuations during inlet and exhaust processes, deviation from ideal gas behaviour and property dependence of pressure, fuel contamination by lubricating oil etc. But they are generally much lesser in terms of their effects compared to these three. And of course, there is a fourth one which is the frictional loss. Now in actual cycle efficiency point of view we do not consider friction there. Friction will come in what is the final brake power that we are going to get.

So, this 1, 2, 3 are the three, once we add this or I should say once we add this factor 1, 2 and 3 to the fuel-air cycle what we get that is referred to as the actual cycle. And then finally when we add the frictional losses to that then we get the actual brake power output from your engine. Generally, it has been observed that if we assume the thermal efficiency predicted by fuel-air cycle minus the thermal efficiency that we get from actual cycle.

If we assume the difference to be in the range of something like 20 %, then general is timeless factor contributes about 6 % of that, this heat transfer contributes about 12% of that and remain in 2 to 3% comes from this exhaust blow down and gas exchange, others having a very minor effect.

(Refer Slide Time: 08:54)

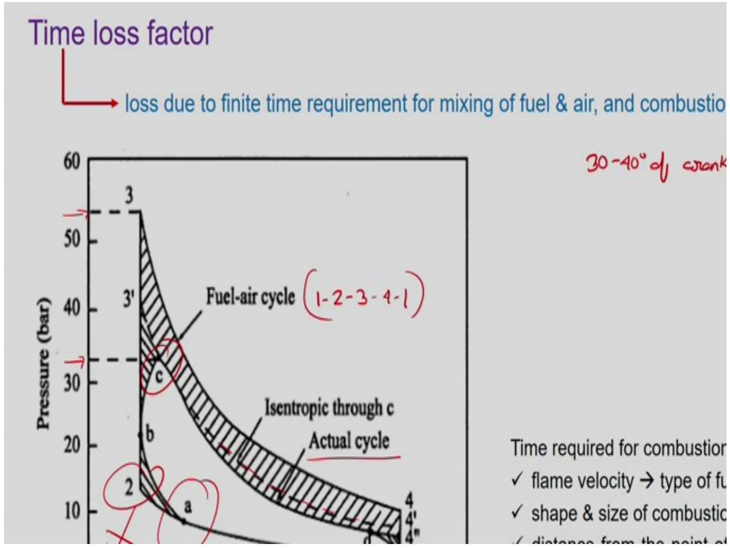


So, if we represent in cyclic form, then this air standard cycle is the one that we have discussed in the previous week the Otto and Diesel cycles. Once we add four factors i.e., the composition of cylinder gases factor number one, the variable effective temperature dependence on specific heat, the effect of dissociation and also the change in a number of moles, then what we get that is a fuel-air cycle which we have also discussed.

And after solving all those numerical problems I am sure you know what we are doing in fuel-air cycle or how we are considering. But still something is left out so one to the fuel-air cycle once we add these 3 factors the time losses the heat losses and the blow down losses. These 3 once we add to the fuel-air cycle then what we get that is called the actual cycle here. This combustion loss that is mentioned that is generally considered as a part of this time loss only so the time loss heat loss and exhaust blow down loss once we add them to the fuel-air cycle what we get that is called the actual cycle.

And from the actual cycle once we subtract the frictional losses this is the final useful work that we get. So, while calculating the efficiency of the actual cycle we do not consider friction because we generally consider the gross output from the engine. And once we subtract the frictional losses from gross output what we get that is the net output which is this useful work. So, let us quickly talk about these three losses.

(Refer Slide Time: 10:25)



The first one is the time loss factor as I have mentioned time loss refers to the loss due to the finite time requirement for mixing of fuel and air and the combustion reaction. Just look at this diagram before looking the complicated diagram just take a look at the points or the cycle formed by points 1 2 3 4 1. So, this is 1 2 3 4 and back to 1 this one is the fuel air cycle, this 1 2 3 4 1 refers to the fuel air cycle.

And as we can see in case of fuel-air cycle also we consider the combustion process to me instantaneous that is a spark is being provided at point number 2 and before the piston starts to move the entire combustion is completed that is instantaneously the temperature and pressure rises to high value. This is the highest pressure corresponding to point 3. But practically because of the finite time requirement for the combustion process, if there is a sufficient amount of piston movement or I should say crank movement involved with the piston or what is a combustion process. It has generally observed that the entire combustion reaction for it to get completed starting from the instant of initiating the spark it requires about 30 to 40^0 of crank rotation. So, 30 to 40^0 of crank rotation and therefore corresponding amount of piston movement. You have to remember that 180^0 crank rotation corresponds to one stroke so, this is not a small fraction. Surely here we are talking about one fifth to one sixth of a stroke which can be a quite significant fraction of piston movement. And because of result of that the appearance of minimum volume and maximum pressure they are not corresponding to each other. Like as you can see if the spark is provided at point number a then the piston is still moving towards the smallest volume towards the top dead center and also combustion also continuing along that so the combustion process starts at point number *a*. It continues and finishes at point number *c,* finishes here. For this entire duration both pressure and temperature keep on increasing. Now, at point number *c*, the system reaches its maximum temperature i.e., the entire amount of energy that could have been released from combustion has been released and is available for useful work or production.

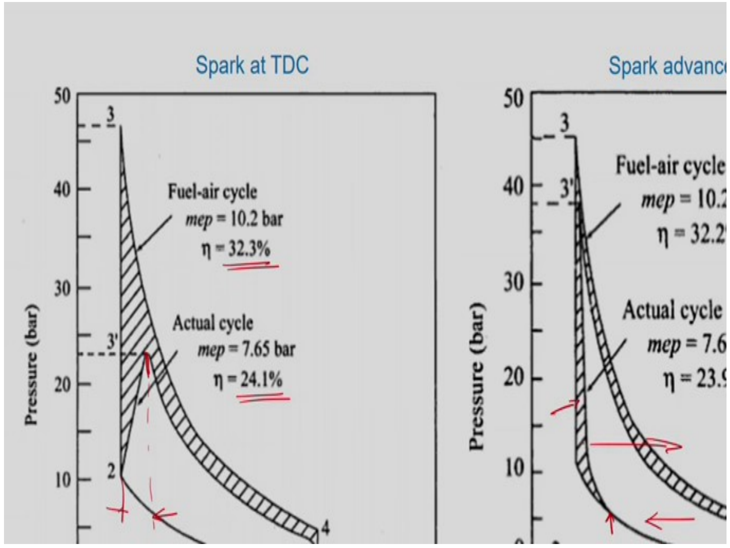
But now look at this point *c*, the pressure at point *c* is way below what we could have got had been working with the fuel-air cycle. There is a significant reduction in pressure, and as the pressure reduces, so the corresponding expansion work also will be much smaller. If we draw an isentropic line through the *c*', then would have got this particular dotted line. But there are further more losses which we shall be adding which leads to there are some more losses, giving us this actual cycle. So, the pressure at point *c* is lower, and also the volume at point *c* is not the smallest possible volume, rather it is a bit higher. Because this one is the smallest possible area and your points may be somewhere here. That means we are getting this amount of downward piston movement, before the final or maximum pressure appears. So, the there is a less scope available to the maximum pressure point to give some work output. And therefore, we are not going to get the maximum amount of work output possible.

So this is the primary reason, the maximum pressure and smallest volume are not corresponding to each other because of the finite time requirement for this combustion process. Time required for combustion can depend on several factors, some of the most important factors are the flame velocity that is the speed with which the flame propagates or flame front propagates within the fresh fuel air mixture. That again is a function of the nature of the fuel that you are using, the fuel air ratio or air fuel ratio or you can say the equivalence ratio, it depends on the shape and size of the combustion chamber that you are dealing with. And also, it depends on the distance from the point of ignition that is a point at which you are providing the spark to the other end of the combustion chamber.

There is no need to go into the detail of any of these explanations but we can understand that the time requirement for combustion can depend on several factors. And therefore, to control this time requirement it becomes very difficult, as we have to deal with several factors simultaneously. So, instead of trying to reduce the time requirement for combustion, the better option that we have is to play around with the instant where we are providing the spark.

Instead of providing the spark at point 2, we generally prefer to provide it a bit earlier but how much early that also requires some kind of optimization.

(Refer Slide Time: 15:31)



Like one example is shown here, here the spark has been provided at the top dead center that is at point 2. So, there is a continuous increase in final pressure. But the final maximum pressure that is reached at this particular volume which is very a larger compared to the minimum possible volume that you can have. This much of difference in volume you can see.

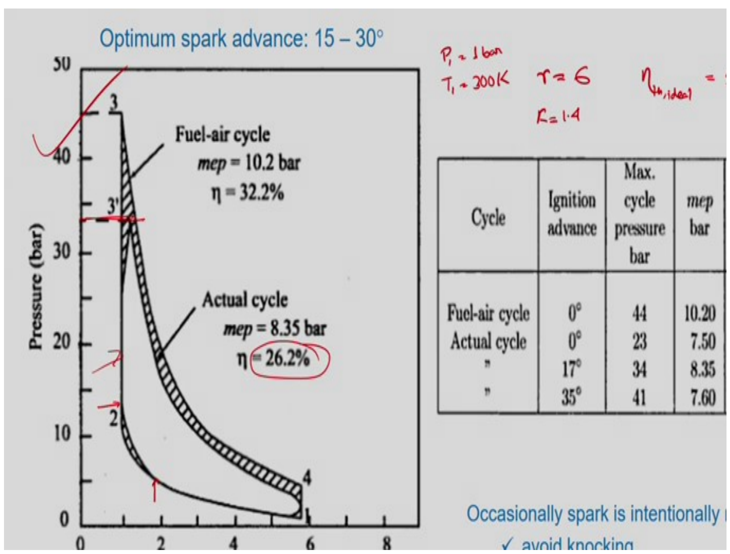
Some numbers are provided in the figure, I am taking these figures from the book of Ganesan, Internal Combustion Engines. You can refer to this book, you will be having the same picture there. If you are dealing with a fuel-air cycle to perform a fuel air analysis using this set of parameters, then you would have got an efficiency of 32.3 % and a mean effective pressure of 10.2 bar. Whereas, because of the spark provided at the top dead center and finite time requirement for combustion which is giving you the maximum pressure at point 3' only you are getting an efficiency of 24.1 % i.e., about 8% loss in the efficiency. This is about $1/4^{th}$ of what we are getting from fuel-air cycle, this is a significant amount of loss. If we provide the spark early, here the spark has been provided 35^0 in advance. Now look at the diagram. Here we are providing the spark much earlier, so that the point of maximum pressure appears at the top dead center i.e., at the point of smallest volume.

And we are quite successful in getting that, the maximum pressure is also quite higher. But look at what we are doing in this part. As the combustion starts at this point, we now have high pressure temperature gases available and these gases tries to expand and tries to move the piston in this direction. But the compression process is still on, so actually piston is moving in this direction. So, during the later part of the compression process the piston actually has to do work against this high temperature high pressure gases which is trying to expand in the opposite direction.

Therefore, we need to spend some more work doing this compression process leading to some further loss in the work output or efficiency. Look at the numbers that are shown on the figure it is the same cycle like in the previous case, fuel-air cycle predicts a thermal efficiency of 32.2 %, but we are actually getting an efficiency of 23.9 %, which is even lower compared to the previous one.

So, that shows that we need to do some kind of optimization regarding at which instant we have to provide the spark. Providing spark at top dead center is never a good option, similarly advancing the spark by large amount that is also not a good option. Because we have we shall be losing some amount of work because of the higher amount of work requirement by the compressor.

(Refer Slide Time: 18:17)



Optimum spark is advanced generally in the range of 15 to 30^0 maybe in the range of 15 to 25 and this is the kind of diagram that you may expect. Here the spark has been provided at this particular point piston reaches top dead center here. At this point the pressure is increasing but still quite low compared to that previous case of 35^0 spark advance. And this is the maximum pressure that we are getting. This is definitely lower than the case when the spark was provided at 35^0.

But significantly higher compared to the situation who has purchased over the top dead center. And the cycle efficiency that we are getting here is 26.2 %. Because eventhough the maximum pressure is lower compared to a 35^0 spark advance, we are also not losing any work in this portion. And therefore, we are finally getting a higher amount of efficiency.

One case study is shown here this is for an SI engine having a compression ratio of 6. Then what will be your thermal efficiency predicted by the air standard cycle or the ideal cycle? We know for an Otto cycle efficiency is:

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

Let us say k = 1.4 if we are taking then if you put the numbers:

$$\eta = 1 - \frac{1}{6^{1.4-1}} = 1 - \frac{1}{6^{0.4}} \approx 51.2\ \%$$

So, that is the thermal efficiency created by your air standard cycle. Now when you are doing the fuel-air cycle using the same condition, then your efficiency actual is coming to 32.2 %.

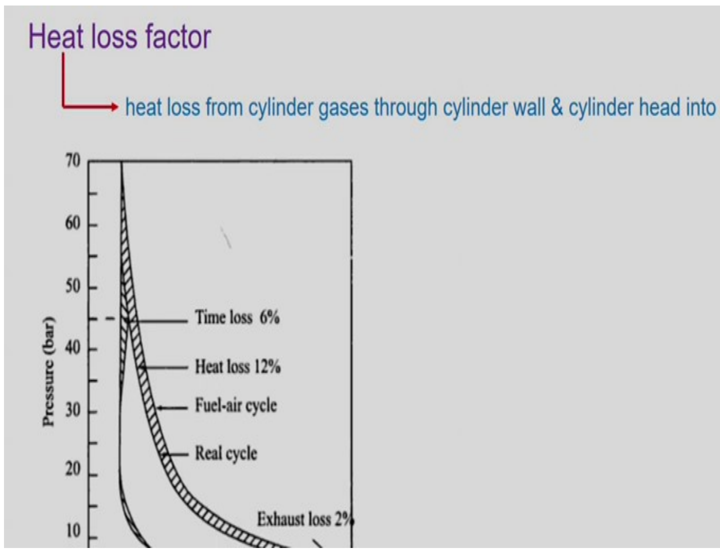
So, there is about 20 % reduction in the fuel air efficiency compared to the ideal cycle efficiency. And this actual cycle efficiency we know that will be quite close to the fuel-air efficiency. Therefore, what we are getting the prediction of air standard efficiency that is extremely wrong, that is far off from what we can get in reality. The maximum pressure and temperature values are also given or maximum pressure rather 44 bar and 10.2 mean effective pressure.

You can try to do a calculation where you take the pressure at the beginning of compression to be equal to 1 bar and temperature equal to 300 K, compression ratio is also given. From there try to predict what can be the maximum cycle pressure and also the mean effective pressure. Now, the actual cycle where we have the option of providing the spark earlier. In fuel air cycle there is no advance, the spark has to be provided when the piston is at the top dead center.

In actual cycle if we provide the spark at top dead center then efficiency is 24.1 %, that we have just seen. Whereas when it is advanced by 35^0, efficiency is even lower 23.9 %. But a 17^0 advance which is this particular case is giving you the result of 26.3, which is about 80 % of our fuel-air efficiency, still 81%. So, this is what we always try to achieve, we try to optimize a point where you should provide the spark and accordingly try to get higher output from the actual cycle.

Occasionally spark is intentionally retarded by providing at the optimal spark advance in order to avoid a knocking kind of undesirable phenomenon, to reduce the exhaust smoke and also to reduce the emission of hydrocarbons and carbon monoxide.

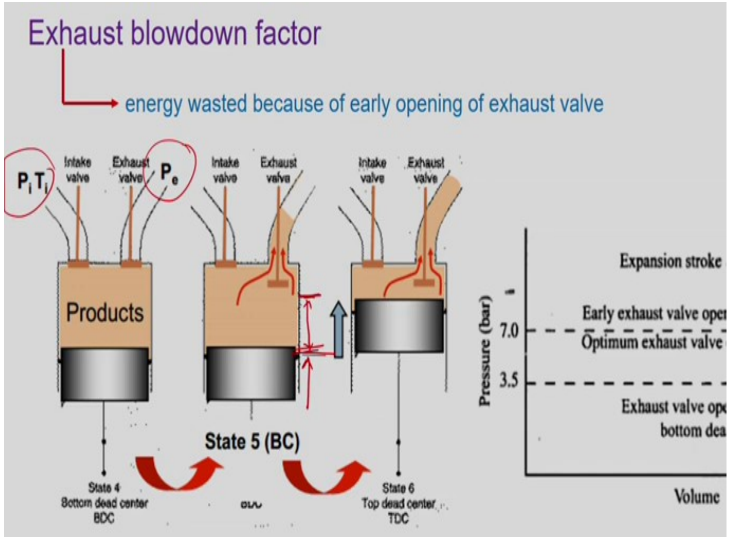
(Refer Slide Time: 21:57)



So, this is our time loss let us move to the heat loss factor. Heat loss very straightforwardly first refers to the loss of cylinder gases through the cylinder wall and cylinder head into the cooling medium. And this can be the effect as I mentioned, even when we are providing a 17^0 of spark advance, then your time loss is this portion and this portion is the heat loss as given by this 12 %, and exhaust blowdown which I shall be coming in the next slide which is about 2 %.

So, the actual cycle, efficiency is 20 % lower then what we are going to get from the fuel-air prediction. The direct effect of heat loss is to cause a reduction in the gas temperature and therefore cause a reduction in the pressure, so, directly can affect the final work output and the efficiency of the engine.

(Refer Slide Time: 22:50)



And then exhaust blowdown loss, it is referred to as the energy waste because of early opening of the exhaust valve. Theoretically the exhaust valve should open when the piston reaches the top dead center at the end of expansion process. But truly speaking at the end of expansion stroke, the pressure of gas inside the cylinder is quite higher than atmospheric. It is generally in the range of 5 to 7 bar, where outside pressure is 1 bar. Now if we want to open the exhaust valve against such pressure, we have to lose some amount of work.

And therefore, often the exhaust valve is opened before the piston reaches the top dead center. As the piston is still going through its expansion stroke the valve is opened a few degrees before top dead center thereby allowing a smoother opening of the valve and less amount of work loss associated the movement of the valve. But at the same time, we have to understand that as soon as we are opening the exhaust valve then the expansion work will stop. Because there is one valve through which the gas can immediately go out, the valve is open and so there is no way you can get expansion work. So, again some kind of optimization required when to open so that we can maximize the work output, but also we can keep this blowdown factor to a smaller one. This is a pictorial representation, when the piston is moving upwards both inlet and exhaust valves are closed.

Look at this, this is the inlet pressure and temperature, this is the exit side pressure. Now, it is a combination of two steps first step is a blowdown. Blowdown means, we open the exhaust valve before the piston reaches the top dead center. This may be your top dead center position, but still the piston is a reasonable degree below the top dead center. That time itself we have opened the exhaust valve which is referred to as a blowdown.

And after that it is a plain displacement because as the piston is moving upward, so that will force all the gaseous to go out, thereby allowing a less amount of work requirement or I should say thereby allowing a more natural process of scavenging the engine cylinder, thereby we can get rid of all the burnt gases. But, as we are opening the exhaust valve at this position, the expansion stroke also stops here expansion is spend only up to this point instead of till the top dead center location.

Now if we plot a pressure vs. volume graph representation, then in case of ideal cycle the exhaust valve should open here. But if the exhaust valve is open after the piston reaches the bottom dead center, then this dotted line is the corresponding work output, because here the pressure at this point is still higher and you have to lose a significant amount of work to overcome this amount of pressure difference between the cylinder interior and the outside.

If we are opening the exhaust valve too early, then that representation is given by this particular line. Here expansion stops at this point itself and so we are losing a significant fraction of the exhaust work. And the optimum opening is somewhere here, we are of course losing some amount of expansion work but at the same time we are keeping the blowdown loss to smallest thereby getting some kind of trade-off.

So, this is what we refer to as an exhaust blowdown loss as we have mentioned earlier the exhaust blowdown can be about 2 to 3% of fuel air efficiency.

(Refer Slide Time: 26:36)

Item			Load
			Full load
(a)	*Air standard cycle efficiency, %*		60.0
1.	Losses due to variation of specific heat and chemical equilibrium, %		13.3
2.	Burning time loss, %	6.5%	4.0
3.	Incomplete combustion loss, %		2.5
4.	Direct heat loss, %		3.5
5.	Exhaust blowdown loss, %	0.9%	0.5
6.	Pumping loss, %	$P_i = 1$ bar, $P_e = 5$-7 bar	0.4
7.	Rubbing friction loss, %		3.2
(b)	*Fuel-air cycle efficiency* $= \eta_{\text{air standard}} - (1)$		$\boxed{46.7}$

So, this is a chart in some kind of tabular form. We are talking about an air standard cycle having efficiency of 60 %, because of the variation in thermo specific heat and chemical composition there is 13.3 % loss. So, we get a fuel efficiency of this 60 – 13.3 to be equal to 46.7 because fuel-air cycle is the is the theoretical cycle which takes care of that temperature dependence of specific heat and the dissociation reactions.

So, we have the fuel-air efficiency of 46.7 % which is 13.3 % lower than that predicted by the theoretical cycle. Now with that fuel-air cycle we have to add this burning time loss and incomplete combustion loss these two together can be referred as a time loss of about 6.5 % in this case. Then there is a direct heat loss of 3.5 and exhaust blowdown loss of 0.55 and pumping loss of 0.4. So, these two together are often referred as the exhaust blowdown loss which is 0.9 % in this case.

So, what will be your gross indicated thermal efficiency? That will be what we are getting from your fuel air cycle which is 46.7 − the losses the time loss in this case is 6.5 the heat loss = 3.5 + the exhaust blowdown loss 0.9 giving you 35.8 % as the actual cycle thermal efficiency or I should say that gross thermal efficiency of the actual cycle. From there once we subtract this rubbing friction loss of 3.2 %, we get the actual brake thermal efficiency.

Here there is a typing error, it is not 6 it is just gross indicate a thermal efficiency or thermal horsepower minus the rubbing friction loss only. And then we can also talk about an actual cycle operation, that is operation with half load when the cylinder is not fully loaded. Now you can see there is no difference in the prediction of the actual cycle and the fuel-air cycle because actual cycle takes into consideration the effect of compression ratio only, fuel-air cycle considers compression ratio and air fuel ratio but not the amount of load. And the effect of the amount of load will be coming primarily on this heat loss factor which increases at lesser load and also the pumping losses which also increases quite rapidly and definitely friction will become doubled. Friction comes because of the movement of piston or rubbing of piston with cylinder walls, whatever load that may be the friction loss will all will remain the same and hence the friction loss.

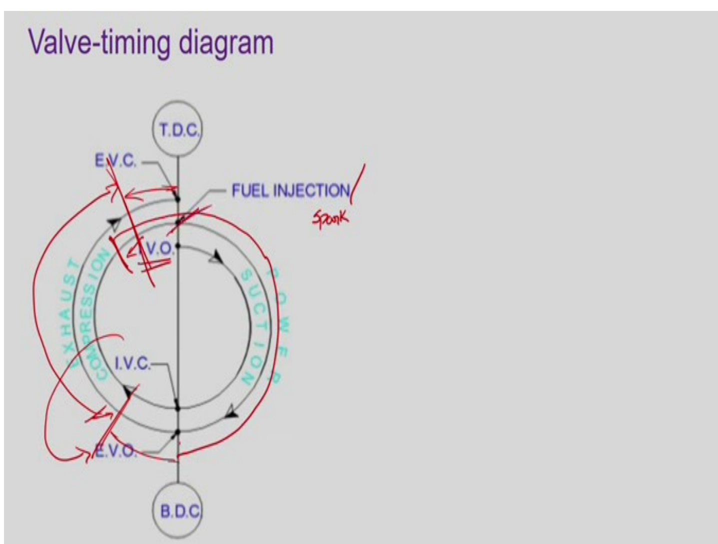
So, for the same amount of load, you can say that the friction loss has increased to double. Here this pumping loss this one actually refers to the loss due to the pressure difference between the inlet side and exhaust side. Because our suction side the fuel air mixture is generally supplied at a pressure of around 1 bar. Whereas on the exhaust side say, I have mentioned the pressure typically ranges from 5 to 7 bar.

So, we can visualize from outside someone who is standing outside IC engine, he or she may say that we are supplying air at a pressure of 1 bar or some gas at a pressure of 1 bar, that is coming out of the pressure of 5 to 7 bar. So, we are actually pumping air from low pressure of 1 bar at the suction side to higher pressure of 5 to 7 bar on the exhaust side. And that loss or that what requirement is the one that referred at the pumping loss.

So pumping loss and exhaust blowdown loss are quite often considered together or separately. But their dependence on loader are different like exhaust blowdown loss remains same however pumping loss that keeps on increasing with reduction in the load. So, at half load we are having a 6.3 % reduction in the actual brake thermal efficiency because of the changes in the heating load, the pumping loss and the rubbing friction loss.

Now we know that there are several kinds of losses present in the actual cycle and to take care of that we may have to play around with the opening and closing of valves and also timing of spark. Like the time loss can be minimized only by suitable adjustment of the position in a spark. Similarly, the exhaust blowdown loss has a direct relation with when the exhaust valve should open.

(Refer Slide Time: 31:16)



And therefore, we often talk about something known as a valve timing diagram which determines at which point, we should open which valve or we should close which valve.

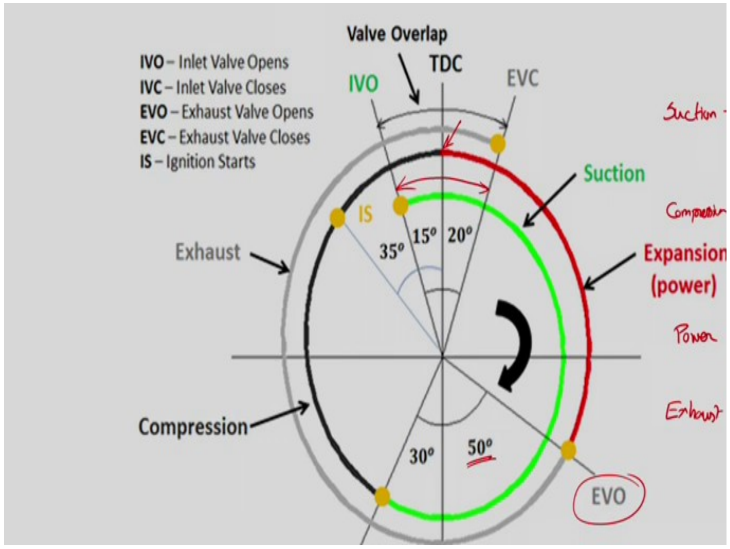
Look at this, this is the ideal scenario for both ideal cycle and the fuel-air cycle. When the piston is at top dead center the inlet valve opens, IVO refers to an inlet valve opening. So, we have the suction stroke going on along this. And at this particular point inlet valve closes, so over 180^0 we have the suction stroke. Then we have the compression going on, here both the valves are closed and the compression continues till the piston goes back to the top dead center. This is the point where we can have fuel injection in case of a SI engine or we can have spark or combustion initiation in case of a SI engine, fuel injection in case of CI engine.

So, we have combustion and in idea cycle we assume combustion to be instantaneous. So, combustion gets completed when the piston is at TDC only. Then we have this power or expansion stroke over 180^0 and when the piston reaches the bottom dead center then we open the exhaust valve and it kept open over half revolution. So, this is where the exhaust valve closes.

So, this is the theoretical cycle but in reality, we have to do something else. The fuel air mixture that we would like to supply, if there is some additional amount of time it is always better so that we can supply more and more amount of fuel. And therefore, instead of opening it at the top dead center, we prefer opening the inlet valve we want to prepone this and we generally come somewhere here, in the range of 25 to 30^0 before that top dead center. And it remains open for suction starting from here up to this and it does not close here also because at this point, the piston is at the farthest from the suction side. And because of the high vacuum that is created inside the cylinder, fuel-air mixture keeps on flowing very rapidly into this. And in order to use the momentum of this mixture we keep the inlet valve open and the inlet valve is allowed to close maybe somewhere here. So, this inlet valve closing comes somewhere here, which is delayed and compression can start only from this point onwards. So, the compression is continued and the spark thing as we have just discussed you need to advance the spark optimally in the range of 20 to 25^0, so the spark is probably somewhere here. So, the compression stroke is spanning over this. In fact, over this portion also we can have compression but here it is not compression of fresh fuel air mixture but it is compression of high-temperature combustion gases.

Then expansion stroke starts from this point, it continues till we open the exhaust valve. And as we have just mentioned you know to reduce the exhaust blowdown loss, instead of opening the exhaust valve top dead center, the bottom dead center we open it somewhere here thereby causing this amount of energy loss in angular sense. This is what we referred as the valve timing diagram.

(Refer Slide Time: 34:32)



This is a typical valve timing diagram that is shown for a 4-stroke SI engine or petrol engine. You can see the inlet valve opening has been preponed by 15^0 and is closing has been postponed by 30^0. Similarly, the exhaust valve opens 50^0 before the bottom dead center and closes 20^0 after the top dead center. That is, for both the valves we have opened them earlier than where they should open and you have closed them later than where they should close.

So, effectively your suction stroke starts from this particular point and continues till this particular point. Which is, in ideal case therefore, suction should continue over 180^0 of crank rotation. But what we are having here is 180^0 plus 15^0 to take care of the early opening of the inlet valve plus 30^0 to take care of the late closing of the inlet valve thereby giving you a total of 225^0 of suction process.

Next is compression, compression like others you should cover 180^0 as for a theoretical cycle. But here compression can start only when you have closed the inlet valve. This is the point where inlet valve closes and then only the compression can start. Because for the compression to happen both inlet and exhaust valve has to be closed then only the pressure can keep on increasing with reduction in volume.

So, compression starts at this point and it continues till the piston reaches the top dead center. So, instead of having a compression over 180^0 it is 180^0 minus the delay in inlet valve closing which is in this diagram to be equal to 30^0. So, compression is taking place over 150^0 of crank rotation. Next is the spark, ideally, we should provide spark once the piston reaches the top dead center, but as we have already discussed you have to provide some time to the combustion process.

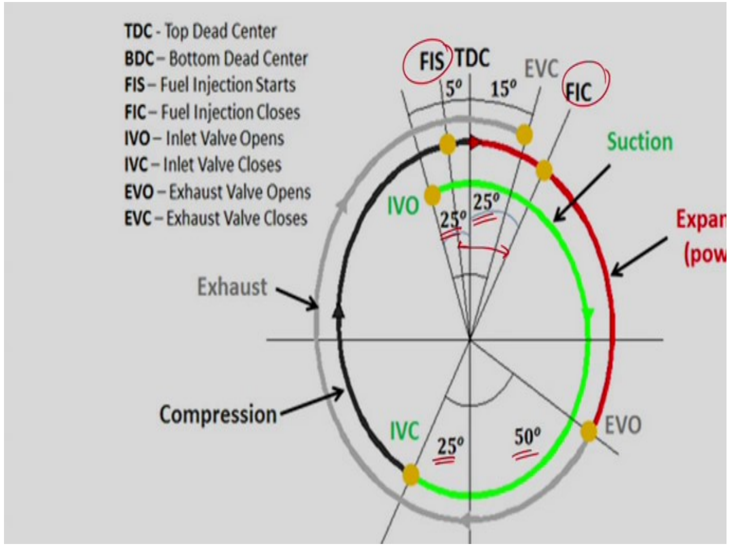
Here we are operating the spark somewhere here which is 35^0 in advance as per this diagram, ideally in the range of 20 to 30^0. So, the spark is provided 35^0 before the top dead center, and therefore over this particular span the piston is compressing not the gas mixture or fresh fuel air mixture rather it is the mixture of combustion gases.

Expansion stroke starts once the piston reaches the top dead center and from there the piston starts to move towards the bottom dead center, so that is the expansion so continues till we open the exhaust valve. The exhaust valve is ideally opened about 50^0 before for the reason that we have just mentioned in order to use the momentum of the upcoming piston, so that we can easily open it and also we can force the exhaust gasses to go out.

So, the power stroke or expansion stroke in ideal cycle it should have occupied 180^0 but, in this case, it is occupying 180^0 minus the earlier and there is more degree of early opening of the exhaust valve which is 50^0 in this diagram, so, only about 130^0. And finally exhaust, again similar to the other three processes ideally should have been at should have covered 180^0. But similar to the suction process it will cover 180^0 plus the amount of early opening which is 50^0 and also this is the early closing of the exhaust valve which is 20^0. So, it remains open over a span of 250^0. Look at this, here we have a span this one over which both inlet and exhaust valves are open. So, while fresh fuel air mixture is coming in through the inlet valve the burnt gases are also going out to the exhaust valve and therefore if we are not careful there is very much a possibility that fresh mixture can also go out through the exhaust valve.

Therefore, during the design stage, it is kept into consideration that the fuel-air mixture coming through the inlet valve should not be able to reach the exhaust valve directly. So, this period is referred to as the valve overlap which is about 30 to 35^0.

(Refer Slide Time: 26:36)

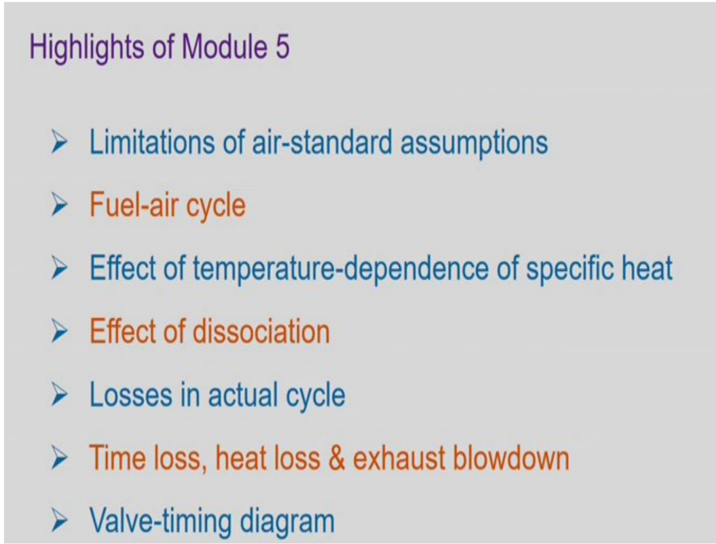


This is the same diagram, but for a 4-stroke CI engine. Here also you can see the inlet valve has been opened 25^0 before and closes 25^0 afterwards, whereas the exhaust valve opens 50^0 before and closes 25^0 afterwards. And fuel injection starts at this particular point and fuel injection closes at this particular point thereby giving a span of this much for fuel injection.

Of course, the numbers all what we are showing here is for 4-stroke a SI engine, they are all tentative, they are more representative of the corresponding ranges. So, by suitable timing of valve opening and closing and also providing the spark in case of a SI engine or fuel injection of CI engine we can significantly reduce the actual cycle losses like the time losses or exhaust blowdown losses.

But in still it is never possible for the actual cycle to give you efficiency comparable to a fuel air cycle.

(Refer Slide Time: 40:14)



That takes us towards the end of our module number 5, where we have discussed about the idea cycle for reciprocating engines. We initially identified the limitations of the air standard assumptions, based on that we developed a fuel air cycle which takes into consideration four of the limitations of air standard assumptions. Like it considers the temperature dependence of specific heat and the dissociation of combustion gases and the real composition of cylinder gases and the change in number of moles because of the combustion reaction. Then, today we have talked about the losses in actual cycle like the time losses, exhaust blowdown losses and heat transfer losses. And we can see that because of the presence of these losses the efficiency predicted by fuel-air cycle is not available in actual cycle rather that will be about 15 to 20% lower than the fuel-air cycle.

So, these are the losses that we have discussed separately and finally we have discussed about the valve timing diagram which is an effort to reduce the losses particularly the time loss and the exhaust blowdown loss. So, this is the end of our week number 5. You please go through this lecture, solve your assignments, go through books, also to clarify any doubt you have and if you still have doubts, do not hesitate to write back to me, Thank you.