Course Name: Engine System and Performance
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Lec 39: Introduction to Variable Geometry Turbine

I welcome you all to the session on engine system and performance. Today, we shall discuss variable geometry turbines. In the last class, we discussed turbochargers, and pertaining to their operation, we covered two different types of turbines. The first category is known as a fixed geometry turbine, and the second one is a variable geometry turbine. We discussed the problematic issues associated with the operation of fixed geometry turbines in the context of turbochargers or turbocharging.

Now, if we recall what we discussed regarding fixed geometry turbines, from the name itself, we can understand the geometry of the compressor, as well as the turbine, is fixed. So, fixed geometry turbine. A turbocharger is essentially a unit in which there is a small turbine. Depending on the requirement, we may need to incorporate an aftercooler or intercooler after the compressor.

But the turbine and compressor, these two rotating devices, are connected to a common shaft. So, if we recall, we discussed fixed geometry turbines (FGT). We discussed that fixed geometry turbines are associated with a few problematic issues in the operation of a turbocharging unit or turbocharger. What are those?

We know that using a fixed geometry turbine, if we need to boost the pressure ratio for a wide range of engine speeds, turbochargers need to be operated at high pressure ratios and at high rotational speeds for a wide range of flow rates. While we are trying to operate turbochargers, two important issues should be taken into account because these two issues are very detrimental for the operation of a turbocharger and these two problematic issues are: One is overspeeding and Number two is overheating. We have discussed these two detrimental effects in the previous class. And let me just discuss these two problematic issues once more. Overspeeding, is associated with the high speed of both the compressor and the turbine.

So, this is basically a fixed geometry turbine. If we need to have a high-pressure ratio produced by the compressor, the compressor speed should be high. To have a higher speed of the compressor, the turbine speed also should be high. That is the concept we have discussed. Now, at high or higher speeds of the compressor and turbine, and rather high compressor and turbine speeds when subjected to high temperatures of exhaust gases. Basically, exhaust gases are allowed to pass through the turbine. When turbine blades are exposed to high-temperature exhaust gases, the turbine is designed to produce a high-pressure ratio. Where, the compressor is designed to produce a high-pressure ratio, the turbine speed will be high.

Revolving components of both these units will be destroyed because of the speed. So, the high speed of both the compressor and turbine when subjected to high temperatures of exhaust gases can destroy the revolving components of the turbocharger. So, this is known as overspeeding, which we have discussed. What is overheating? The sole purpose of having this unit, whether it is a supercharger or turbocharger, is to increase the inlet pressure. Now, when the inlet pressure should be higher because of compressing the intake or incoming air, the pressure of that air or the charge will be further compressed inside the cylinder during the compression stroke. The pressure will be even higher at the end of the compression stroke, which is accompanied by higher temperature as well. So, if the intake or inlet air pressure is higher because of the higher speed of the turbine or compressor, in the subsequent stages of operation, the temperature of the charge or incoming air will be more, and that high temperature may become vulnerable to several parts of the engine. So, this is known as overheating. Essentially, you can understand that a turbine or fixed-geometry turbine, which is used in a turbocharger, cannot be operated at high speed; otherwise, these two problematic issues will arise, and rotating parts or several parts of the engine will become nonfunctional. So, what we need to do is reduce the speed. Again, we have discussed this in the context of a supercharger.

In fact, by using the performance map, we saw that if we reduce the turbine speed, the compressor speed will certainly be reduced because these two elements are connected to a common shaft. So, when the compressor speed is reduced, the compressor operation is limited between two different lines. If you recall, the line on the far-right side of the performance map is known as the choke line. So, the compressor cannot be allowed to operate beyond that line; otherwise, the compressor is said to be choked. Now, similar to what we have studied in the context of a nozzle, when the compressor is operating at a low speed, its operation can reach the surge line. Even when the compressor operation approaches the surge line, it is not desirable at all because, this situation is undesirable because the compressor operation will be highly unstable, and flow reversal will occur.

This means that with a fixed-geometry turbine, we can only operate at high speeds; otherwise, these two problematic issues will arise. If we allow the turbine to run at a low speed, the compressor operation can become unstable as it may approach the surge line. So, accounting for these two issues, the concept of variable-geometry turbines is typically developed.

To circumvent the problems associated with fixed-geometry turbines, variable-geometry turbines are developed. Now, let us examine this particular type, and we can see that this is the variable-geometry turbine. Such turbines are attached to turbochargers to provide greater flexibility in turbocharger operation. Let us now discuss this particular turbine. From the schematic diagram, you can see that the extreme right component is the pneumatic actuator.

So, this is a pneumatic actuator, and if we look at the relatively larger component of this schematic, which is the cross-sectional view of the turbine, so you can see that there are a few rotating vanes. That is why it is known as rotating vanes pivoting. So, these vanes are pivoted on the ring. These are basically rotating vanes and let discuss the hardware of this particular turbine. Then, we will discuss the operational procedure. So, these vanes, which are rotating—rather, vanes can rotate. These vanes are pivoted on the adjustable ring. Because you can see that these are the rings. So, this ring is adjustable. And we can tune the operation of the ring by this pneumatic actuator through this mechanical lever. So, this is a mechanical lever. These blades are the fixed blades or vanes. We can control the passage between two consecutive rotating vanes through this, adjustable ring, using this pneumatic actuator. So, the pressure inside this pneumatic actuator is controlled by the engine control module. Depending on the speed, this pneumatic actuator is operated. This pneumatic actuator controls the passage between two consecutive rotating blades using this mechanical lever.

So, if there is a requirement or if the engine speed is high, when engine is suddenly accelerated or if we need to develop more power from the engine, engine speed will increase. Then this pneumatic actuator will give some engine control module, will operate pneumatic actuator. So, inside this pneumatic actuator, that air pressure is controlled by this engine control module. This pneumatic actuator controls the gap between two consecutive blades which are mounted on the adjustable ring using this mechanical lever. And through this, exhaust gas or exhaust gases that is expelled from the engine cylinder is, allowed to go into the turbine fixed blade through this passage between two consecutive rotating blades. So, if we can control the gap between two consecutive blades, we can control the flow rate of exhaust gas or exhaust gases to be allowed to go into the inner part of the turbine and if we can really have control over the flow rate of the exhaust gases, then we can also control the thrust that will be produced or the rotation of the turbine inner blades, which in turn will control the rotation of the compressor. So, that way we can have greater degree of controllability of turbocharger operation using variable geometry turbine. And if we use this type of turbine, then problem related to the over speeding and overheating, which we have discussed in the context of fixed geometry turbine can be eliminated.

So, this is the fixed geometry turbine. This is also known as a pivoting vanes turbocharger, which are equipped with a variable geometry turbine. And this type of variable geometry turbine or pivoting, is used in the most common type of diesel engine. So, these turbochargers with variable geometry turbines are very common for vehicle diesel engines.

This is something which we can have geometrical modification of the turbine itself. Now, even after having all these modifications, sometimes we may need or require a fixed geometry turbine to be attached to the turbocharger unit. Now, in such a case, what we can do is we can really control the speed of the turbine and have some mechanism, and that mechanism is known as overspeeding and overheating pertaining to fixed geometry turbine. Even knowing these two different problems are there with this turbine, if we need to use this particular turbine in a turbocharging unit, then we need to have some mechanism to control these two to the extent possible. And one possible idea is to bypass a portion of the exhaust gases that are expelled from the engine cylinder.

Knowing these two different problems are inherent to this type of turbine. Sometimes we may need to use a fixed geometry turbine in a turbocharging unit. In such a scenario, what we need to do is have a certain mechanism to control these two problems to the extent possible. One possible solution could be to bypass a portion of the exhaust gases expelled from the engine cylinder to shape the turbocharging unit.

So, to safeguard the turbocharger from these two detrimental effects when the turbocharger unit has a fixed geometry turbine, a portion of the exhaust gases expelled from the engine cylinder is deliberately bypassed around the turbine. So, essentially, what is done, bypassing the exhaust gases—not all, just a portion of them. So, this is what is needed. Now, if we need to bypass it, we have seen that the turbine, whether it is a fixed geometry turbine or a variable geometry turbine, is installed at the exhaust manifold because exhaust gases are released by the engine through the exhaust manifold into the atmosphere.

So, certainly, to exploit or employ or utilize the maximum kinetic energy of the exhaust gases, the turbine is used to rotate the turbine. The turbine is placed at a location close to the exhaust manifold. Now, if we need to bypass, what is done? A portion of the exhaust gases is deliberately bypassed around the turbine housing. So, this is what is done. So, it is made to bypass the turbine housing.

That means, instead of allowing exhaust gases to go into the turbine, exhaust gases are allowed to go to the ambience through the exhaust manifold. So, if we go to the next slide, that means exhaust gases are directed toward the exhaust manifold. So, the exhaust gases—a portion—are not allowed to go into the turbine. And for that, the exhaust gas bypass passage is opened.

So, if we draw the engine cylinder—say, for example, if it is the exhaust manifold, if it is the exhaust valve—then if the turbine is placed over here, what is done? There will be one valve. So, that valve will allow exhaust gases to go into the ambience. So, this is the valve. And this is the turbine. So, we can understand from the schematic depiction is that depending on the requirement, this valve is typically a poppet valve or any flap type valve. So, the valve is operated by a diaphragm actuator and the diaphragm actuator is controlled by the exhaust gas pressure, that means we can see essentially this valve is placed in the exhaust manifold when we need to bypass deliberately a certain portion of the exhaust gases and instead of allowing the exhaust gas to go into the turbine, certain portion of the exhaust gas is deliberately made to bypass to the ambience directly.

And this bypass passage opening, which is also known as waste gate. That is the bypassage opening, because if we control this valve, this will open, and this waste gate will eventually control certain portion of exhaust gas to leave through the exhaust manifold into the ambience, so as to turbine speed can be controlled. So, that is what we wanted to discuss. That means, if we need to use fixed-vehicle turbine.

For any particular requirement, knowing fully that particular turbine, a fixed geometry turbine will have two different problematic issues. Rather than having one fixed geometry turbine in a turbocharging unit, there will be two problematic issues, and those are overspeeding and overheating. We can have this arrangement to prevent the turbocharging unit, to the extent possible, from these two undesirable phenomena. So, with this now, let us solve one numerical problem on superchargers, and then by solving this numerical problem, we shall try to understand or illustrate the concept that we have learned on this particular unit, which is the supercharger. So, if we go to the problem statement then we shall start solving this problem.

Problem 1: An un-supercharged petrol engine develops 740 kW of brake power with a fuel-air ratio of 0.080. The brake-specific fuel consumption is 0.335 kg/kWhr and mechanical efficiency is 0.78. The inlet pressure is 680 mm of mercury absolute, and the mixture temperature is 320 K. The engine is supercharged to a pressure ratio of 1.8 by a centrifugal supercharger of isentropic efficiency 75% and mechanical efficiency is 86%. Inlet to the compressor is same as those for the un-supercharged engine inlet. Assume that the fuel-air ratio remains unchanged and that IHP is proportional to the inlet density. Assume further that the volumetric efficiency does not change due to supercharging. Calculate the BHP of the supercharged engine and its break specific fuel consumption.

It is given: an unsupercharged petrol engine—so it is not supercharged—this is a naturally aspirated engine. This is the brake horsepower, and the fuel-air ratio is 0.080. So, this information we can see from the problem statement, this is the BSFC. So, basically, the inlet to the engine—pressure of air and temperature of the air-fuel mixture—is given as 320, as this is the SI engine. So, typically for an SI engine, during the intake stroke, the charge is drawn into the engine cylinder. That is air-fuel mixture. As I told you, even for the modern CI engines, it is not only pure air that is drawn into the engine cylinder during intake stroke. Rather, there is a provision of supplying fuel in the intake manifold as well because nowadays, multiport fuel injection system is there. And hence, it is not the pure air that is consumed during intake stroke. Rather, it is air-fuel mixture or charge. So, the charge condition is given, pressure and temperature. The engine is supercharged. So next part of the problem is, the engine is supercharged to a pressure ratio of 1.8 by a centrifugal supercharger of isentropic efficiency 75%. So initially it is unsupercharged engine. Next it is said that the engine is supercharged to a pressure ratio of 1.8 by a centrifugal supercharger. So, by a centrifugal supercharger of isentropic efficiency 75%. We know in a supercharging unit only compressor is there because instead of rotating a small turbine, compressor is driven by taking certain amount of energy from the output shaft of the engine. So, the failure ratio remains the same. And that IHP is proportional to the inlet density. So, this is again a clue. The volumetric efficiency also remains the same for these two cases. Calculate the BHP of the supercharged engine and its brake-specific fuel consumption. So, these two we have to calculate.

So, from this problem statement, we have seen so many things. We shall discuss all these things again during the solution process. But what we need to calculate is the brake horsepower and brake-specific fuel consumption during the supercharged condition. So, let us solve this problem first.

Solution:

So, if we start solving this problem, what we can do first is calculate the mass flow rate of fuel. Because if we can calculate the mass flow rate of fuel, we know the air-fuel ratio, then we can calculate the mass flow rate of air. So, to calculate the mass flow rate of fuel, we can use the information given. So, the first case is naturally aspirated which is when there is no supercharger.

$$\dot{m}_f = bsfc \times BHP = 0.335 \times 740 = 0.0688 \text{ kg/s}$$

This is the mass flow rate. By calculating the mass flow rate of fuel, we can calculate the mass flow rate of air,

$$\dot{m}_a = \frac{0.0688}{0.080} = 0.8607 \text{ kg/s}$$

For the first case, that is the naturally aspirated case. Now, for the naturally aspirated engine, we can easily calculate IHP.

Why? Because mechanical efficiency is given, it is 0.78. So, mechanical efficiency is nothing but

$$\eta_m = \frac{\text{BHP}}{\text{IHP}}$$

because brake horsepower is less than indicated horsepower. So, IHP should be equal to, that is,

$$\text{IHP} = \frac{\text{BHP}}{\eta_m} = \frac{740}{0.78} = 948.71 \text{ kW}$$

So, this is the indicated horsepower for the naturally aspirated engine. So, having calculated IHP and we know BHP, then we can calculate frictional horsepower because IHP, that is the power produced inside the cylinder, and BHP, that is the power available at the shaft. So, a certain amount of developed power will be used to overcome the frictional losses, and that is FHP. So, if we calculate FHP for the naturally aspirated case,

$$\text{FHP} = \text{IHP} - \text{BHP} = 948.71 - 740 = 208.71\ \text{kW}$$

We will now consider supercharging. We have discussed the basic need for supercharging. Essentially, now the ambient air will be compressed using a small compressor, and the pressure will be higher, and the density will be higher. So, if we write for the case of supercharging for the compressor, we can have this isentropic relation, that is T_{1s} by T_i. T_{1s} is the temperature at the exit of the compressor following isentropic compression, where the subscript 'i' is used to denote the inlet quantity or quantities at the inlet of the compressor. So, this equal

$$\frac{T_{1s}}{T_i} = \left(\frac{P_{1s}}{P_i}\right)^{\frac{\gamma-1}{\gamma}}$$

$$T_{1s} = T_i\left(\frac{P_{1s}}{P_i}\right)^{\frac{\gamma-1}{\gamma}} = 320(1.8)^{\frac{1.35-1}{1.35}} = 372.6\ \text{K}$$

The actual pressure and actual temperature will not be the isentropic values. Having all these, if we calculate the actual temperature at the inlet of the engine, — If we go back to the previous slide, this T_{1s} is not the actual temperature. This is the temperature at the exit of the compressor following isentropic compression. But the actual temperature at the exit of the compressor will be something different. And for that, we can use the isentropic efficiency of the compressor. So, if we go to the next slide and write the isentropic efficiency of the compressor,

$$\eta_{com,ise} = \frac{T_{1s} - T_i}{T_{1a} - T_i}$$

Here, T_{1a} is the actual temperature at the exit of the compressor. That we have discussed, drawing all the processes in the *h*-*s* plane, if you can recall correctly. So, let me draw this just for the sake of completeness. If we draw the compression process, so this is *h*, this is *s*. So, this is, P_i, and this is P_1. So, this is *i* to 1*s*. But the actual compression process would be like this, and this is 1*a*.

So, assuming the charge to be an ideal gas, or more precisely, a calorically perfect gas, if you assume that C_p is constant— C_p is not a function of time, but a function of temperature—we can derive this. So, using this expression, we can easily calculate T_{1a}, which is nothing but

$$T_{1a} = T_i + \frac{T_{1s} - T_i}{\eta_{com,ise}} = 320 + \frac{372.6 - 320}{0.75} = 390.15 \text{ K}$$

So, this is the actual temperature. Try to understand: the temperature of the air at the exit of the compressor following isentropic compression was 372, whereas the actual temperature is 390.15 Kelvin—slightly higher. Now, if we go to the next slide, it is given that According to the problem statement, this IHP is proportional to rho, which is density.

If we go to the problem statement, it is given that IHP is proportional to the inlet density. So that means this is rho i. So, from there, we can use some expressions. So, we can use the following expressions. And these expressions are IHP_{SC} for the supercharged. So, suffix SC now refers to supercharged condition while IHP_{NA} that is naturally aspirated. This is equal to

$$\frac{IHP_{SC}}{IHP_{NA}} = \frac{\rho_{SC}}{\rho_{NA}} = \frac{P_1}{P_i} \times \frac{T_i}{T_{1a}}$$

So, that inlet pressure is remaining same for both the cases. So, for the supercharged, so this is basically the pressure ratio. R remains the same, while this SC is equal to P_1, so actually, because we are using a compressor. So, that is

$$\rho_{SC} = \frac{P_1}{RT_1}$$

So, R will get canceled. So, we are getting this.

$$\frac{IHP_{SC}}{IHP_{NA}} = 1.6 \times \frac{320}{390.15} = 1.312$$

$$IHP_{SC} = IHP_{NA} \times 1.312 = 948.71 \times 1.312 = 1245 \text{ kW}$$

That is the case, so if we go to the problem statement just let me read a few lines it is stipulated in the problem statement that volumetric efficiency does not change even after supercharging. Then mass flow rate of air remains constant for both conditions.

So, volumetric efficiency does not change, which means the mass flow rate of air to the engines will be the same for both cases, number one. Number two is, if we consider this. So, if we write here quickly, it is given that eta volumetric does not change due to supercharging. So, this means the mass flow rate of air remains constant for both the conditions. So, with this consideration, we can calculate the work input to the compressor, and that is

$$\dot{W}_{com} = \dot{m}_a \times C_{p,a} \times \frac{T_{1a} - T_i}{\eta_{com}}$$

So, that is the work input to the compressor. In fact, we need to have, so what is compressor efficiency, actual work input to the compressor should be always higher than the isentropic compression. So, isentropic compression to work done following or rather actual work to be supplied to the compressor is the compressor efficiency. So, the work to be supplied to the compressor for the isentropic compression to the actual work to be delivered to the compressor. The ratio of these two quantities, compressor efficiency. So, this is essentially actual work, the work done.

Now, it is coming as,

$$\dot{W}_{com} = 0.8607 \times 1.005 \times \frac{390.15 - 320}{0.86} = 70.55 \text{ kW}$$

So, if it is the case, the net BHP of the supercharged engine. So, following this, then we can calculate the net BHP that is brake horsepower of the supercharged engine. That is

$$BHP_{SC} = IHP_{SC} - FHP - \dot{W}_{com}$$

Because I said that for a supercharger, this is the amount of energy which is borrowed from the output shaft of the engine to run the compressor. Whereas for the turbocharger, we have no need to borrow energy from the output shaft of the engine; rather, the kinetic energy of the exhaust gases is used to run a turbine. So, if we calculate it,

$$BHP_{SC} = 1245 - 208.71 - 70.55 = 965.74 \text{ kW}$$

So, this is the answer to the first part of this question. And if we calculate BSFC for the supercharged condition, that is

$$BSFC_{SC} = \frac{\dot{m}_f}{BHP} = \frac{0.0688 \times 3600}{965.74} = 0.2564 \ \frac{\text{kg}}{\text{kW hr}}$$

So, this is the answer to the second part of the question. Let me go back to the problem statement once again. So, we had to calculate the BHP of the supercharged engine and its brake-specific fuel consumption, that is, BSFC. And if we go to the last slide, then we can see These two, that is, the BHP of the supercharged engine and the brake-specific fuel consumption.

So, what we have done in today's class is discuss variable geometry turbines, which are used to bring a greater degree of flexibility in the turbocharging unit. Thereafter, we saw that if we need to use fixed geometry turbines in some cases or situations, knowing fully that fixed geometry turbines are associated with detrimental or undesirable phenomena, then we have one possible way to remove these two detrimental effects to the extent possible, and we have discussed this part as well. Finally, we solved a numerical problem on superchargers, and by solving this problem, we tried to illustrate the conceptual part of this particular unit, the supercharger.

So, with this, I will stop here today, and we shall continue our discussion in the next class. Thank you.