Fluid Dynamics And Turbo Machines.
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Part B.
Module-2.
Lecture-6.
Real Fluid Flow and Efficiency of Turbo Machine.

Good afternoon, I welcome you all for today's discussion on real fluid flow and efficiency of Turbo machines. In the last 2 lectures when we talked about the principles of Turbo machine, we talked about the velocity triangles, we talked about energy transfer, namely we talked about Euler's energy equation and degree of reaction. While doing all these derivations, one thing we have kept in common, what is that, it is the concept of vane congruent flow, or the flow does not vary inside the vane passage from passage to passage, from blade to blade, from shroud to shroud.

However, in reality, such a flow does not exist. Why? So in the first part of today's lecture we will explore the reasons, I will not go into the details of this real fluid flow, I will talk about what the different effects of real fluid flow that can be seen in the velocity triangles. And whenever we have a departure from an ideal flow we will also get a less performance from the idealised condition and hence we will quantify this performance of a real Turbo machine in terms of its efficiency. So let us look at what is a real fluid flow and what are the causes.

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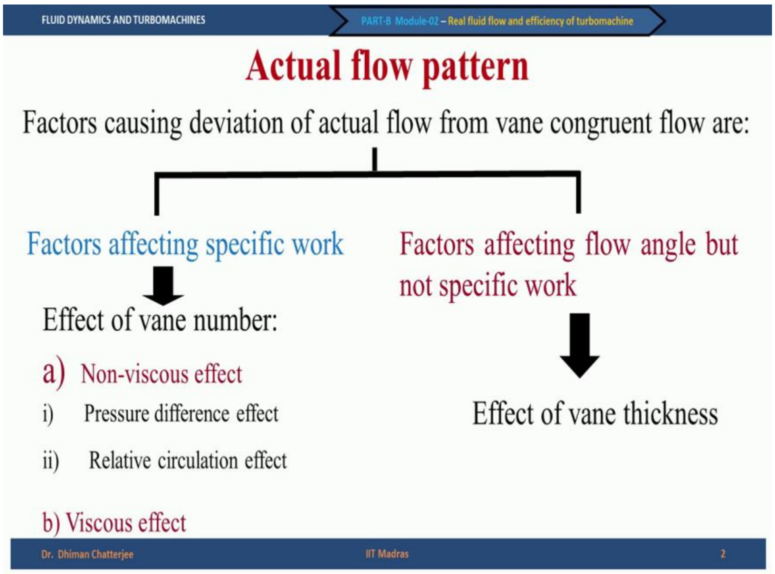

So we are talking about an actual flow pattern. And this actual flow pattern are caused by factors which make the vane congruent flow an unrealistic or not a practical one. So if I look at it, we are talking about factors affecting specific work and factors affecting the flow angle but not the specific work. Let us not worry about these terminologies, what is more important is what are the actual reasons behind these changes. And we can say that we are talking about effect of the vane number. See one of the assumptions of vane congruent flow was that there are infinite number of vanes.

So we have to relax that assumption, they have to say in the real world infinite number of vanes are not possible, the vanes numbers are always finite. And hence we can say that this vanes numbers, of talking about the effect of vane numbers, there are 2 effects we can see, one considering that the fluid is viscous and the 2nd, the fluid is not viscous or inviscid. So this is the first effect of the finite number of vanes. What was the 2nd assumption we had? The 2nd assumption that we had was the vanes are of negligible thickness.

We talked about in vane congruent flow infinite number of vanes with infinitesimally small thickness. So in reality because of stress considerations or manufacturing difficulties we cannot think of vanes which are having negligible or infinitesimally small thickness. So we have to talk about the effects of finite vane thickness. We will start with, without relaxing the effect of viscosity, that is we will first start with there is no viscous effect, just the number of vanes being made finite instead of the infinite vane assumption. So let us look at it.

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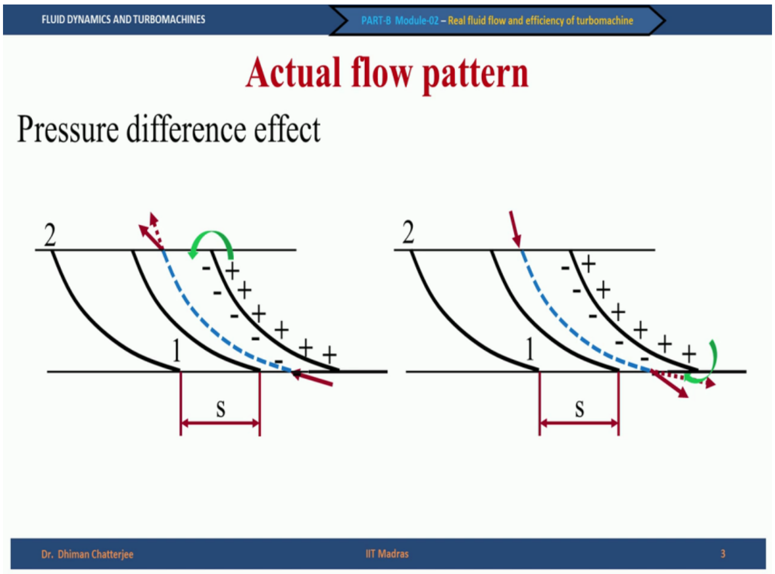

We can talk about the pressure difference effect. When we talk about a finite number of blades we will find that the pressure difference between the pressure side given by the + sign in this drawing and the - sign which is a suction side is going to increase. Smaller the number of blades, the higher will be this pressure difference which is often called the blade loading. And what happens whenever there is a pressure difference the flow tries to establish from the high-pressure side to the low-pressure side.

So now let us look at a scenario when we are talking about the suction side as 1, pressure side as 2 as we have done and we can say that the fluid enters here in case of a pump or compressor from the suction side an ideal trajectory would have been there dashed arrow shown here. But because of the pressure difference there is a deviation, there is a departure and we see that the fluid flow actually leaves the pressure side at an angle which is different from the tangential direction. You may say that the pressure is different all throughout the blade length.

It is true but whenever we are doing this analysis we are lumping the effect of the entire pressure difference on the final or the ultimate plane when the flow leaves the impeller. So what we see, we see that the flow has an adverse pressure gradient and on top of it there you have the pressure which is acting in a way that there is a flow which deviates. In case of turbines when the flow comes from the top, here also we will see there is a pressure difference. But in case of a turbine it is the velocity at the exit of the turbine impeller is much higher compared to the inlet.

In case of pumps or compressors the velocity is low and you have a deviation. So the resultant velocity that comes out at the pump impeller at 2 is actually going to have more deviation from the vane congruent flow direction, the tangential direction in than in case of turbine.

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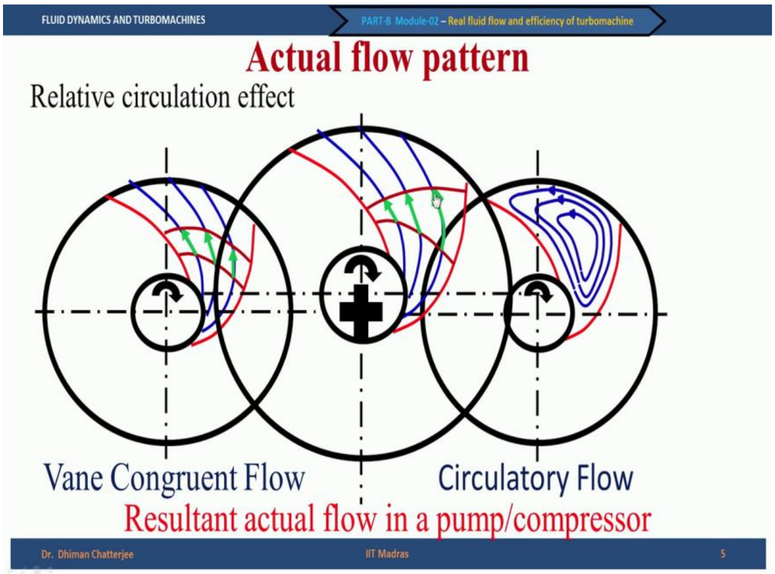

The 2nd effect is called the relative circulation effect. So let us look at what it means. You know that from the vane congruent flow assumption that from blade to blade there is no difference in the velocity at any radial location. In the relative circulation it is said that the flow is apparently having a motion which is a pure circulation in which the arrows are shown here and you have a recirculation. How to visualize it? You can imagine that you have closed the ends and you are simply making the flow to rotate.

So now if you have the real flow, the relative circulation effect shows you that because of the circulation effect + the vane congruent flow, there will be a difference in case of the flow inside the vane passage. This is again you have to remember there is no effect of viscosity. This effect, the relative circulation effect can be thought of in a way, in a similar way as we talked about the pressure difference effect. Wherever there is a higher pressure, we have shown there will be a flow leakage to the lower side. And hence the velocity if you see, the velocity is less here and the velocity is more here.

Thus what we get, we get that along the azimuthal direction at a given radius there is a difference between the velocities at different locations. So we get a nonuniform velocity which is not considered in the vane congruent flow.

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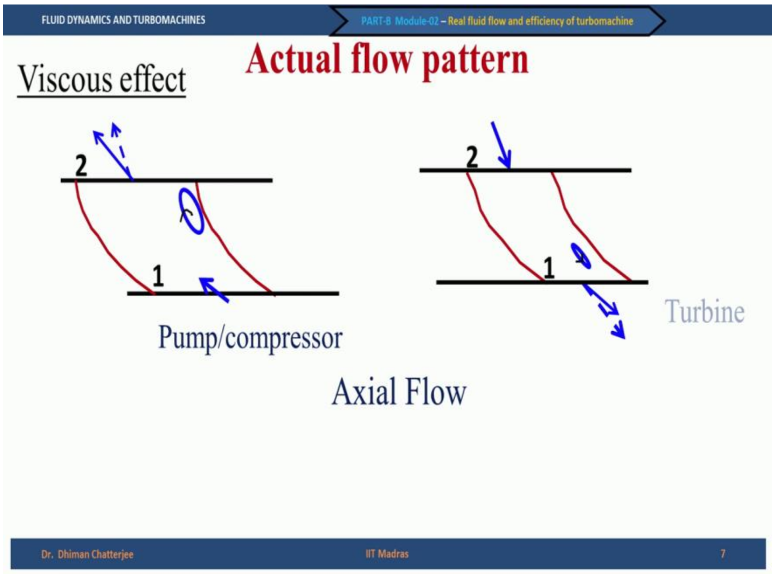

So far we have not considered the viscous effect. Now we, if we consider the viscous effect, we know that in case of pump or compressor there is an adverse pressure gradient and in fluid mechanics you have studied that that is a necessary condition for flow separation. It is not sufficient but it is necessary. So what we see is that in particular line in the off design conditions, there are possibilities that**,** large portion of the flow passage, the vane passage you may have flow separations.

And as a result there will be a very significant effect on the flow direction as shown here. The dashed line arrow is actually corresponding to an idealised case. And the solid arrow is a effect of the viscosity. In case of turbines, what happens is the flow from 2 to 1 and the flow is accelerating. If the flow is accelerating then there is a less chance of flow to separate because we have a favourable pressure gradient. So it can only happen if there is a problem in the geometry or we are talking about a very much deviation from the design condition.

Whatever be the case the effect of this flow separation even if it is present in a minor amount is not going to be significant in case of turbine. So this is an axial flow scenario, we could have made it the same for the radial flow scenario.

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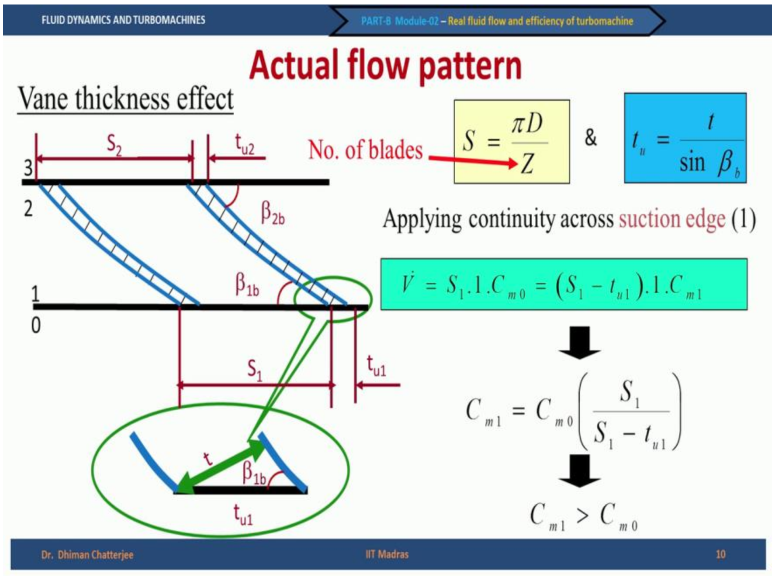

The next effect is the vane thickness effect. When we talk about the vane thickness effect, let us look at the vanes more carefully. So, so far we have shown you only one line representing the vane but here you see the vane has a finite thickness shown by the hatched lines, S1 is the spacing, we will talk about it and this T is the thickness. So if we zoom this portion of the blade, we can show that the blade has a thickness T and of course the T need not be uniform from the inlet to the outlet, from the suction side to the pressure side and the projection of T along the tangential direction is given in our consistent notation of TU.

So at the suction side we will say it is TU1, at the pressure side we will say TU2. Now you imagine the first case when there is no blade thickness, that means we are considering only this line and the corresponding line here. The entire flow area was available for the fluid to flow. Now when you have the vanes of thickness T1 at the in, at 1 and T2 at 2, what happens is a portion of the area represented by this line TU 1, this is not available because this is occupied by the blades. So what will happen?

If the flow area changes but your volume flow rate has not changed, then you have to accommodate it by change of velocity. So let us look at what is the effect. If I say that the spacing is S, then I know that S is given by pie D by Z. How do you get it, you say that the total circumference is pie D at any radius, let us say D1 or D2 and there are Z number of blades, so the spacing between the blades is pie D by Z. And this TU is nothing but T by sin beta as I have shown in this small triangle here.

So now if we use it and apply continuity equation across the suction edge, that is we consider the flow at 0 and 1, 1 being an inside the blade passage which in which the blade angle is beta 1B and 0 is just outside the impeller inlet edge in case of a pump or compressor, then we can write that V dot is equal to S1 multiplied by 1 is the depth and multiplied by CM0 and inside the blade passage you have the blade thickness occupying a portion of the passage area and we write that S1 - T1 whole multiplied by CM 1.

As a result it is very obvious that CM1 is going to be more than CM 0. And this bracketed factor is called the blockage factor which is given by many times in the percentage. We can say that the blade thickness is such that it occupies 3 percent blockage.

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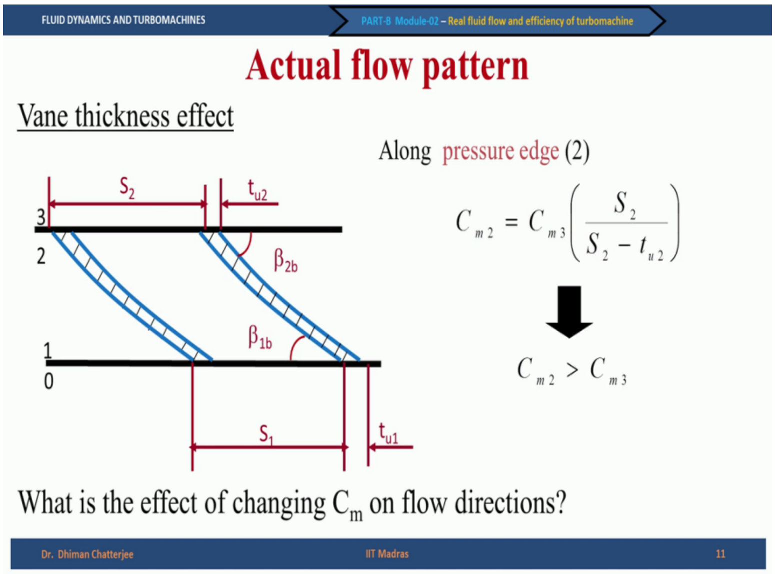

At the outside edge or the exit at 2 and 3, between 2 and 3 we can write that CM2 is equal to CM3 is equal to but multiplied by S2 whole divided by S2 - TU2, if we follow the same practice. 2 is inside the plate passage, 3 is just what leaves. And that gives me that CM2 is greater than CM3. So fine, we get the blade velocities are different. But how does it affect our velocity triangle? To know that we need to know what is the effect on the directions, what are the effects on the blade angles and the flow angles, are they remaining same?

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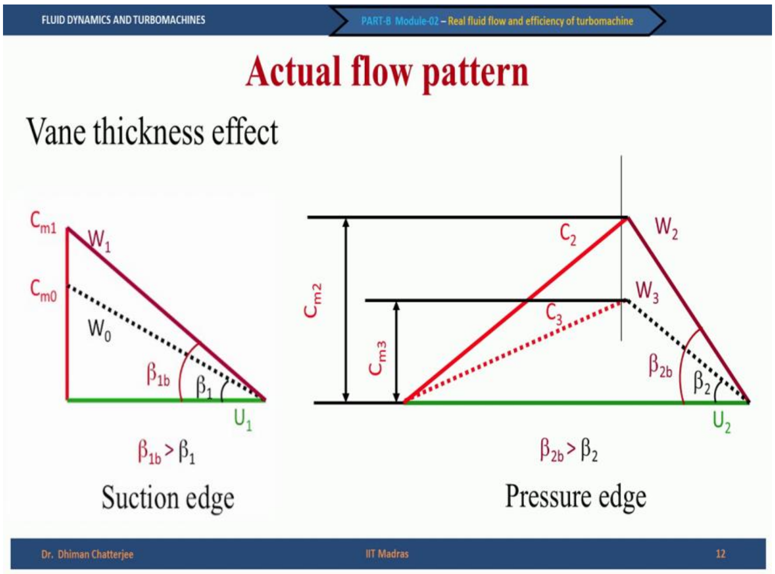

To investigate it, we will look into the velocity triangles. We assume without any problem that like we have been doing consistently that the exit whirl or the inlet whirl, in case of turbine and respectively, that is CU1 or C1U, whatever notation you follow is equal to 0. That is C1 is equal to CM and if we write it, then we show that CM0 is the velocity just outside the blade and CM1 is the velocity just inside.

And we see that there is a change, there is a change in the direction, that is we are talking about this as the absolute velocity and we are talking about the relative velocity being changed from W0 to W1 and there is a change in the angle beta 1 from beta 1B. So we see that there is a deviation in the blade angle, just like we have talked about the deviation in the blade angle because of finite number of blades. In the pressure side also we can similarly talk about that C2 is more and C3 is less and as a result beta 2B is different from beta 2 once again.

So what do we learn? If you summarise whatever you have learned from the actual flow pattern so far, we understood that even in the absence of real fluid properties like viscosity, we can have deviations in from the blade angles in case of the flow is because of the number 1 the blade thickness and number 2 the number of blades, or I should put it in the other way the number of blades and the blade thickness in the way of our discussion. So these have to be kept in mind.

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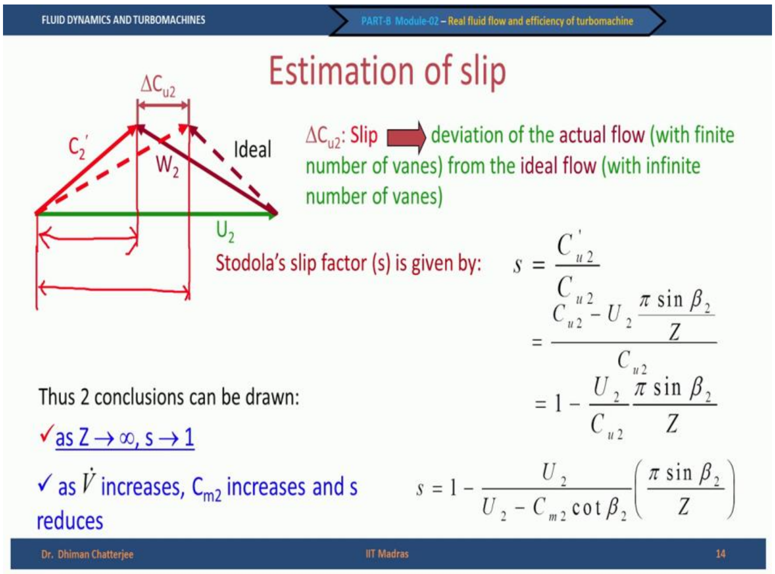

So another concept which comes to our mind when we discuss Turbo machines is the concept of slip. Even in an inviscid flow there is a difference in the actual flow and the vane congruent flow because of this factor called slip which arises in a finite number of blades. So let us look at it. We have this dashed arrow which is an idealised condition of the velocities and the solid arrows which gives you the velocities which is reality, that is C2 prime is real and that is real in the sense it considers the slip but it does not consider the viscosity, I must repeat and the dashed is C2.

So what happens here is that I can say that this is the portion which contributes to the C2 prime, C2U prime and what we have here is the other one which talks about the CU 2 which is an idealised condition of vane congruent flow. And as a result what we see is the difference between the 2 is given by Delta CU 2. And hence we see that Delta CU 2 is a slip which is a deviation of the actual flow with finite number of vanes from the ideal flow with infinite number of vanes and I repeat, I stress that we are not considering any viscous effects.

That is there is no real fluid effect but there is an effect of finite number of vanes. Then there are different slip factors or expressions possible, the one I am just going for an example to bring out the effect of the number of blades is given by Stodola, it is called Stodola's slip factor which is given by the ratio of CU 2 prime by CU 2 where CU corresponds to the vane congruent flow and CU2 prime corresponds to the actual flow in case of finite number of vanes.

And this is the correlation given by Stodola which is given by CU2 - U 2 pie sin beta 2 by Z divided by CU 2 and it can be simplified further dividing throughout by CU 2 we get 1 - U 2 buy CU 2 pie sine beta 2 by Z. Z is the number of blades as we have discussed. And hence we get the relationship CU2 from the velocity triangle is nothing but, this is CU 2 which is that thing but U 2 - CM2 cot beta 2. And we get S is equal to 1 - U 2 by U 2 - CM2 cot beta 2 multiplied by pie sine beta 2 by Z. Thus 2 conclusions can be drawn.

The first one is as Z tends to infinity, this term drops off, this entire term drops off and S tends to 1 which means CU 2 prime becomes CU 2. That was our starting point. If the Turbo machine has infinite number of blades, a very large number of blades, then the guidance is proper, there is no large deviation and we can say that the slip factor in 1 and CU 2 prime is equal to CU 2. The other is if the volume flow rate is more, if volume flow rate increases, then CM2 increases and hence S reduces.

So you see that these are the 2 parameters which are going to determine how much is the slip. Of course I have assumed that beta 2 does not change in this discussion. We will talk about the effect of beta 2 later on when we talk about the pumps in the next week. So we have talked about the actual flows and the causes which make the flow to deviate from the vane congruent flow. Whenever we have a deviation from the idealised world, idealised conditions, we can expect that there will be a deterioration in the performance and this is given as losses in Turbo machines.

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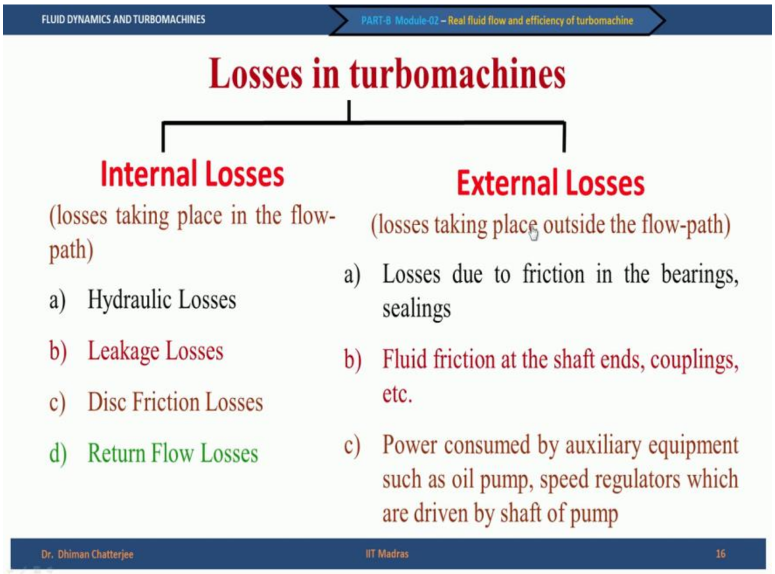

So let us look at what are the different types of losses that we come across and how to account for it and finally how this leads to efficiency. So we can say that there are 2 types of losses, the internal losses or the losses taking place in the flow path and the external losses or the losses taking place outside the flow path. So first we will talk about the internal losses. These are the hydraulic losses, leakage losses, disc friction losses and return flow losses. I will explain each one of these briefly soon.

And the other one is external losses which is losses taking place outside the flow path. What is meant by outside the flow path, that means the losses which are in the locations which are not associated with the fluid flow. For example losses due to friction in the bearings, sealings, etc., fluid friction at the shaft end, power consumed by any auxiliary equipment which we may need, all those. So in this lecture today we will touch upon these internal losses in some detail, the external losses all will be lumped as some loss term.

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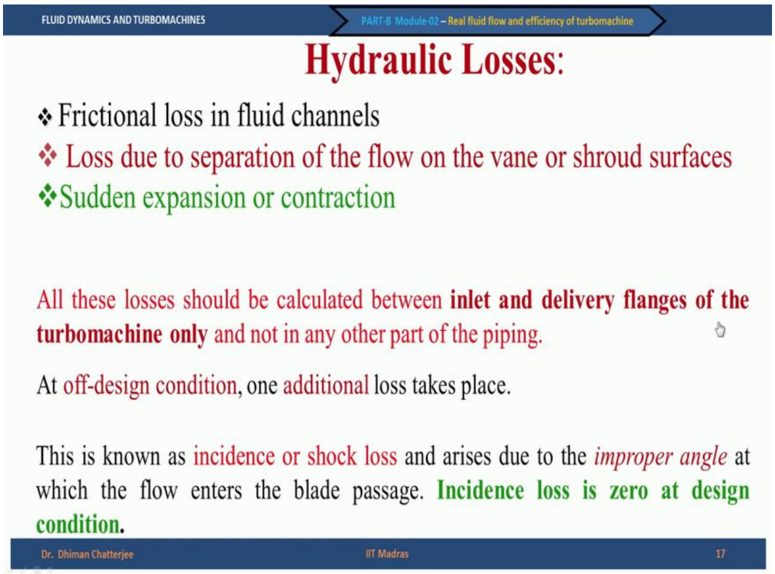

So let us look at hydraulic losses. The hydraulic losses arise because of frictional losses in the fluid channels, that the skin friction pressure drop, the loss can be because of the separation of the flow on the vanes or shroud surfaces because we have talked about already the viscous effect, there could be sudden expansions and contractions because of some geometrical constraints in the design and all this gives rise to the hydraulic losses. But one thing we have to keep in mind.

Let us say we are talking about the pump, this pump is connected between a sump and an overhead reservoir. Water flows from the sump to the overhead reservoir through the pump and pipes. When we talk about the losses, you may tend to think that I have to include the losses in the pipes also but these hydraulic losses that we are discussing today in connection with Turbo machines do not include the losses in the pipe. That is we are talking about the losses that take place inside the Turbo machines, that is between the inlet and delivery flanges of the Turbo machine only.

Please note that we will take care of the piping losses when we talk about the pumps later on. But for the losses, hydraulic losses in the Turbo machines, we are talking about the inlet and delivery flanges and whatever is in between, the losses that take place in this flow passage only. There is also another type of loss, hydraulic loss which can come up, particularly at off design condition, that is called the shock or incidence loss. I have already talked about this, there is a fact that when you are taking the fluid to approach at a proper angle, which means the blade angle is equal to the flow angle, it happens only at the design point.

At any off design condition, and a Turbo machine is expected to work over a range of operate operating flow rate and not just other design conditions and hence the shock or incidence loss has to be accounted for in such off design conditions.

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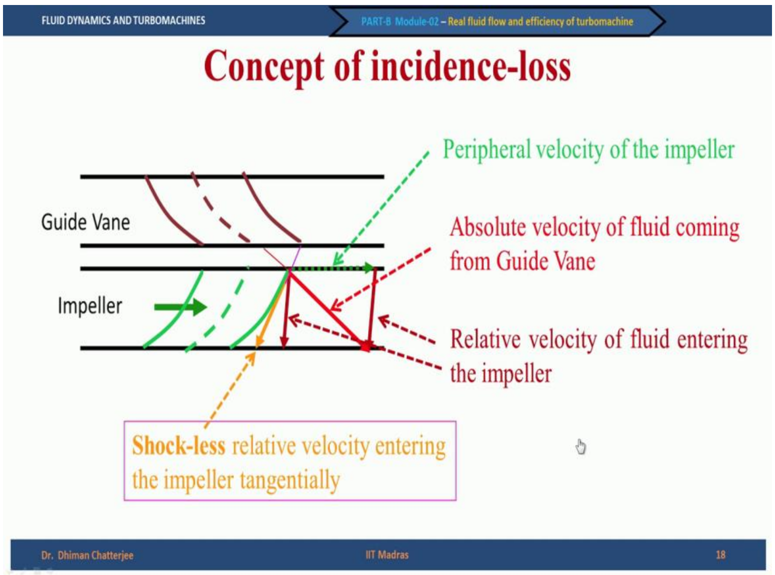

So to understand the concept of incidence loss, let us say that we have a guide vane and the flow leaves the tangentially to the guide vane. So which flow leaves tangentially to the guide vane, it is the absolute velocity. Because guide vanes are fixed, they are not rotating, so whatever leaves the fixed component tangentially must be the absolute velocity and if the condition had been ideal, then the flow should have entered the impeller at tangentially at this point. However we see that it does not happen in such a scenario.

We find that this is a blade peripheral velocity which is fixed in this case were considering and this is the absolute velocity or the fluid coming from the guide vanes, tangential to the guide vanes at the exit of the guide vanes and this is the relative velocity which is now shown here. And if you can zoom this portion, you can imagine that we are assuming this portion, you will see that the blade is here and this orange arrow is tangential to this green blades but this line is not.

And hence there is a difference between the arrows shown here which is a real scenario and the arrows shown here which is the idealised case of the flow, relative velocity entering the impeller tangentially. And this difference in the projection along the tangential direction is what is related with the shock or incidence loss. We will not go into that in this course, but it suffices if I say that the shock or incidence loss will be zero at the design point and is nonzero at any other condition as shown here.

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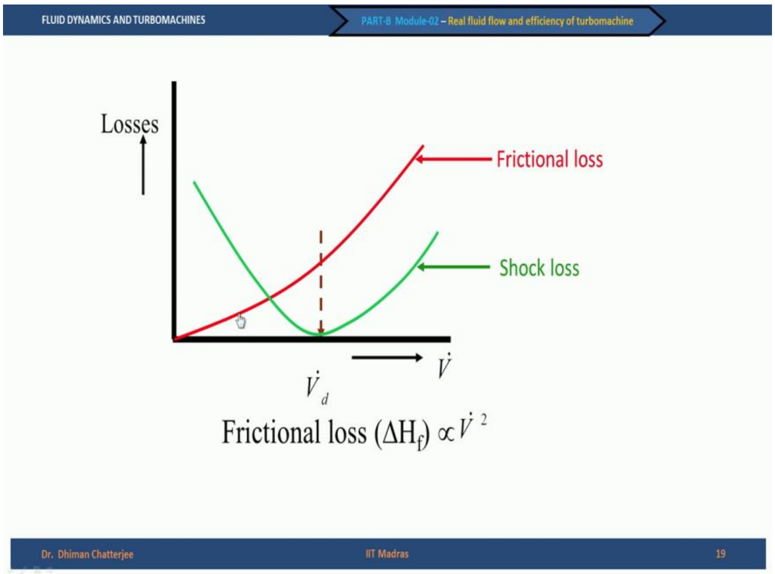

We are talking about the different losses, we are talking about frictional losses, we know that from the pipe flow examples that you have done in fluid dynamics, that the frictional loss is proportional to the volume flow rate square, V dot square and it increases here, it is from zero at V dot equal to 0 in the fashion shown here and we are talking about the shock loss which is zero at the design condition and nonzero and increases on either side of the design condition.

So when we talk about different hydraulic losses, we have to talk about the summation of the loss due to the shock loss or incidence loss and the frictional losses. So we will talk about both the losses together.

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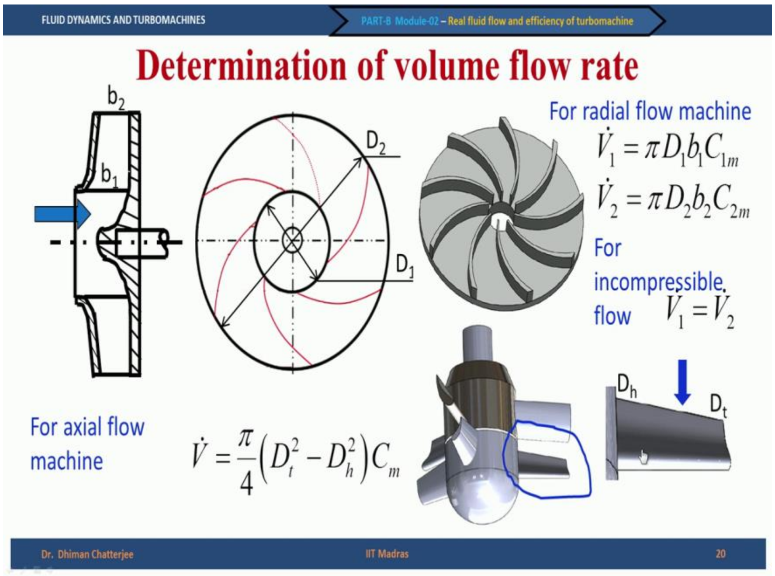

Then we are talking about another important, what we need to know is determination of volume flow rate because that is related with the power. So we come back to the same old impeller we have been discussing so far or the schematic we have shown earlier. And we can define the different dimensions, D1 at the smaller diameter, D2 at the pressure surface, the corresponding blade heights are B1 and B2 so that if I look at the radial flow machine and if I am talking about this blade passage, then I can say that V dot 1 is nothing but pie D1B1 multiplied by C1M and V dot 2 is nothing but pie D2 B2 multiplied by C2M.

And hence for incompressible flow we know that V dot 1 equal to V dot 2 and we can relate D1, B1, C1M with D2, B2, C2 M. For an axial flow machine, it is slightly tricky. So let us look at an axial flow machine and let us focus our attention on the rotor blade. And if I zoom it and put it separately, we can say the portion of the rotor blade attached to the hub is having a diameter BH, this is called the hub diameter and the portion which is far away at the open end, which is called the tip of the blade is called the tip diameter DT and the flow takes place.

And when we talk about it, we are basically considering this as a, blade height as a passage, so we can consider the annulus area which is pie by 4 DT square - DH square and we get that the corresponding volume flow rate is pie by 4 DT square - DH square multiplied by CM. Another point you should note is this expression of CM in case of an axial flow machine is an axial flow direction. In case of a radial flow machine it is a radial flow direction because this will be visible only in the meridional view. So these are the expressions that we need when we talk about the power soon and you will also need these expressions to solve some of the problems given in the tutorial for this week and the next week.

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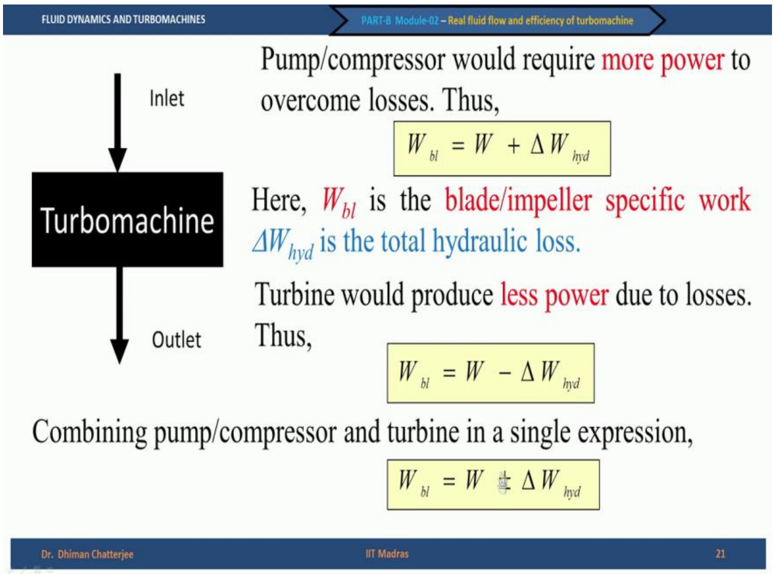

So we are not talking about the Turbo machine at the inlet and outlet, then we are coming back to kind of a blackbox but we know more about that what happens inside the blackbox called Turbo machine. So we say that pump and compressor would require more power to overcome losses, that we have also obtained from thermodynamics if you recollect, we said that in order to run a compressor we need a power which must be greater than equal to some minimum power and we say that in case of even the blade specific work.

So blade specific work W BL is nothing but W which is a specific work or the useful energy difference per unit mass flow rate across the Turbo machine + Delta W hydraulic. And here W BL is note that the blade or the impeller specific work and Delta W hydraulic is the total hydraulic loss that we have discussed so far, that is frictional losses because of the fluid viscosity or viscous effects as well as the incidence loss. And the turbine we know from thermodynamics would produce less power due to the losses and there also we can say that W BL equal to W - Delta W hydraulic.

So if you want to express in an in a single expression, we can say that W BL is equal to W + - Delta W hydraulic. This + sign refers to the pump and the - sign refers to the turbine.

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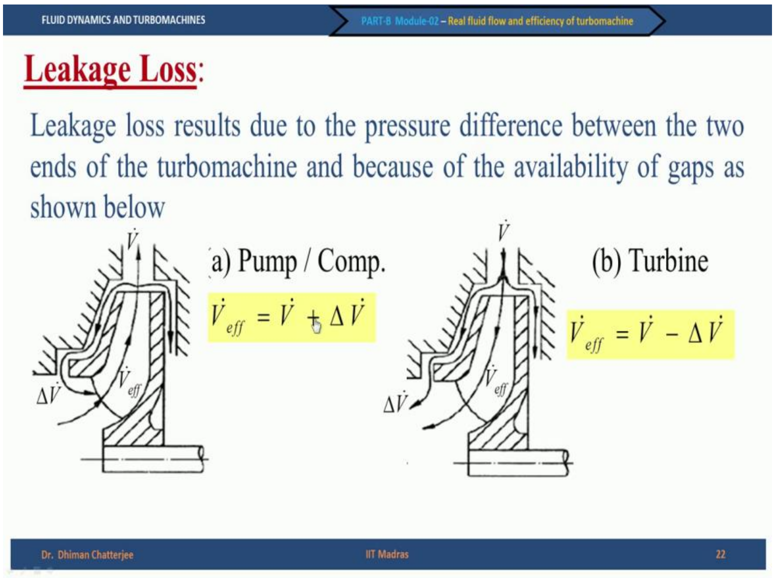

Next is the leakage loss. When we talk about leakage loss, we are talking about the volume flow that is taking place through the leakage because of the pressure difference between the 2 ends of the Turbo machine. Recollect what we have discussed in the very beginning of the Turbo machine lectures when we said the reciprocating pump has physical barriers and hence does not suffer from leakage, whereas Turbo machine pumps let us say centrifugal pump actually has a problem of leakage. So we revisit this problem now.

Let us say that we have an impeller and it has a casing. Now a rotating component cannot be in contact with the stationary component, so there must be some space, some gap between the 2. And this is the schematic of the gap. Now flow is taking place from the smaller diameter to the larger diameter and the flow that should come out and we have measured, let us say we are doing experiments is V dot. Now if you know that there is a pressure difference existing between the pressure side and the suction side and there is a leakage, so what you can expect is a portion of the fluid will come back and rejoin the impeller.

So effectively the volume of fluid that is handled by the pump or compressor is not V dot but V dot + Delta V dot because of the leakage flow. And in case of turbine it is reverse because in case of turbine the fluid is entering at a high pressure and leaving at a low-pressure, so now for the fluid there are 2 options, one is to go through the blade passage and another one is to go through the gaps. So V dot effective in this case of turbine is V dot - Delta V dot. Please note that the signs are becoming consistent for the pumps as positive and for the turbine as negative.

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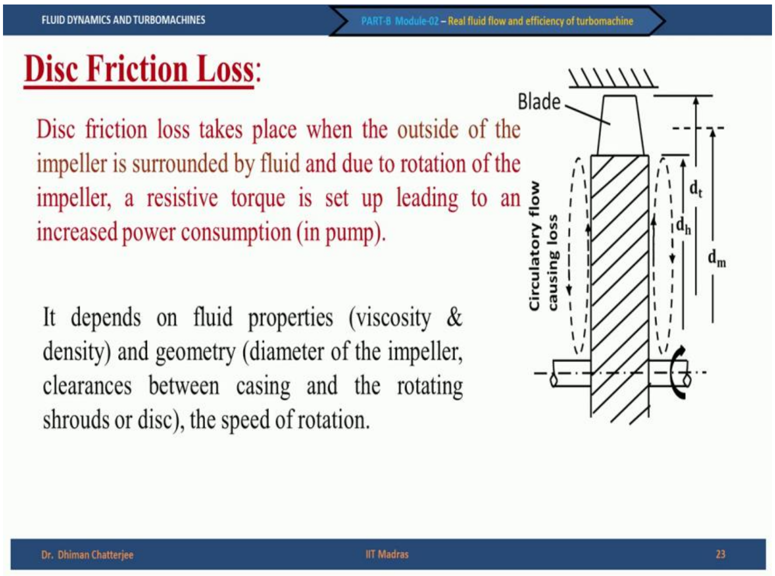

So we have talked about the 2nd, the 3rd is the disc friction loss. And disc fiction loss can be understood again in an analogical way. Let us say I give you an eggbeater or a stirrer and I tell you, I give you 3 fluids, one is air, one is water, another one is glyceryl. I ask you to rotate it, where do you find it most difficult or stirrer the fluid? Of course it is glyceryl. Why, because you have viscosity. And if you have the same fluid, if you have to rotate it at a higher speed, you will have a more energy requirement from there.

Why, because you have to overcome the losses. So disc friction loss takes place when the outside of the impeller is surrounded by fluid and due to rotation of the impeller, a resistive torque is setup leading to an increased power consumption in pumps. Let us look at this picture, this is a blade impeller and we have an axial flow machine, I have taken axial flow machine as an example. This is the hub and the flow takes place. But this blade, this portion of the hub is actually in contact with the fluid and this entire hub is rotating, which means that it will drag along with this because of the fluid velocity the fluid along it and it will have a circulatory flow.

So what it means? It means that that there will be an energy drainage which goes, which is not productive, which goes into just making this motion which is necessary and possible. So what we get it is that actual power output in case of turbine will be lowered further, in case of pumps or compressors, this has to be overcome and hence the power requirement will increase. This disc friction loss depends on fluid properties like viscosity and density and geometry and the speed of rotation.

For example if I take Pelton turbine which rotates in the air or the fan, in the ceiling fan in your home which rotates in air, this effect can be neglected. But if it is in water or any other fluid, this effect can be significant.

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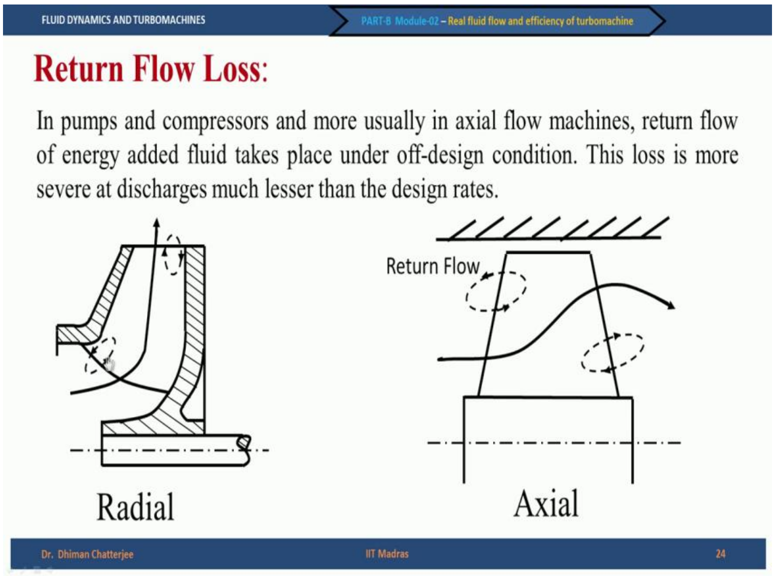

So the last of the losses is the return flow loss. Return flow loss is more commonly seen in pumps and compressors and in the off design conditions. How do you understand return flow loss? Let us say that we have a pump which is either radial or axial and the flow takes place from a low-pressure side to the high-pressure side, you have to overcome the adverse pressure gradient. And now at off design conditions, particularly at a discharge much lower than the designed conditions, the fluid will not have enough energy to overcome it.

So you will find that the flow recirculates and goes back. So what it means, it has entered the blade passage as you can see in the 2 cases and then flows out. So the blade has done some work on this flow which does not produce productively. But please remember this happens at a condition which is far away from the design condition at a very much lower flow rate.

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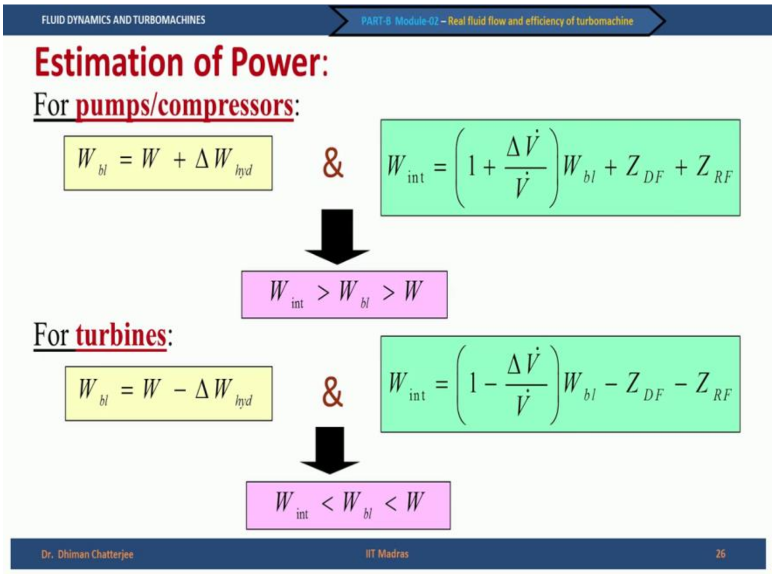

So when we are talking about estimation of power, we can say that the ideal power is P ideal, in this case there are no losses and hence the blade specific work will be the same as the specific work across the turbine, turbomachine and we get the P ideal is rho V dot W but the actual internal power will be P internal is rho V dot + - Delta V dot. Please note that + - with the proper sign convention for pump is positive and for turbine it is negative multiplied by W BL. You have to also relate W BL with W BL infinity with the help of slip term, + - PDF + - P RF.

Please note that this letter PDF which has a + - sign and the PRF, return flow loss which has a + - sign, the + refers to the pump and - refers to the turbine. We can say that equivalent internal specific work, W internal will be one + - Delta V dot by V dot W BL + - Z DF, that is just so we have divided PDF by rho V dot of + - Z RF.

The sign convention remains the same and we can talk about the estimation of power, we say that W BL is W + Delta hydraulic and W internal is 1+ V dot by V dot W BL + Z DF + Z RF, whatever you have done, I am just now separating out for pumps and compressors, hopefully that will also help you in keeping in mind that W internal is greater than W BL is greater than W. Okay. You see here that W internal is greater than W BL and W BL is greater than W.

In case of turbine it is reverse and we see that W internal is less than W BL is less than W. That means a portion of the energy in case of turbine is drained to overcome the losses, in case of pumps and compressors, it has to be supplied with more energy.

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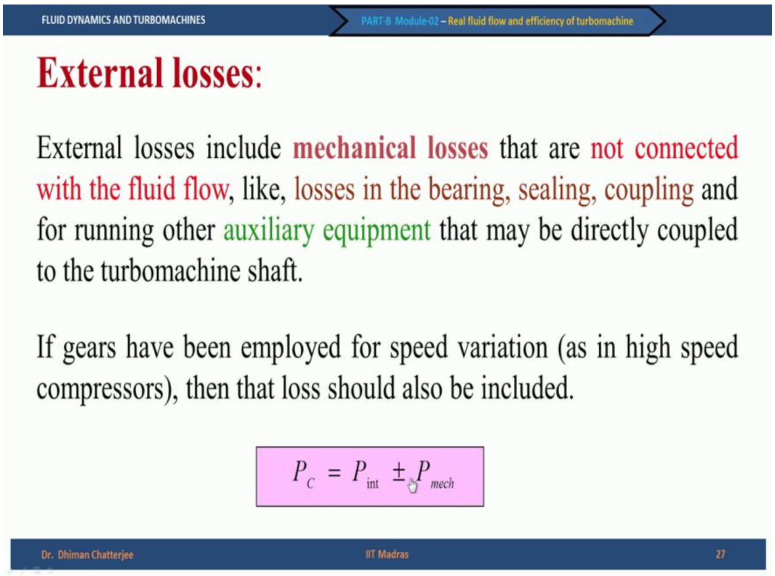

And then we can talk about the external losses, that are the mechanical losses. Without going into details of different types of mechanical losses we can simply say that P coupling is equal to P internal + - P mechanical. This + sign again come for pumps and - sign comes for the turbine. What it means that the coupling power that has to be provided by the motor in case of pump should be more than the internal power because some portion of the power however less maybe will be used to overcome the losses in the mechanical.

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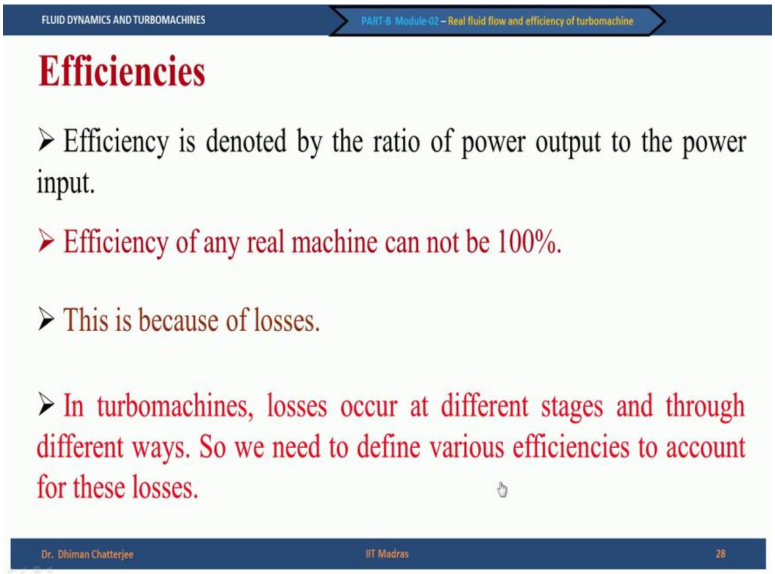

Then we can talk about the corresponding efficiencies as the ratio of power output to the power input and we can say that efficiency of any real machine cannot be 100 percent. Which means this is because of the losses. In Turbo machines losses occur at different stages and through different ways, so we need to define various efficiencies to account for these losses.

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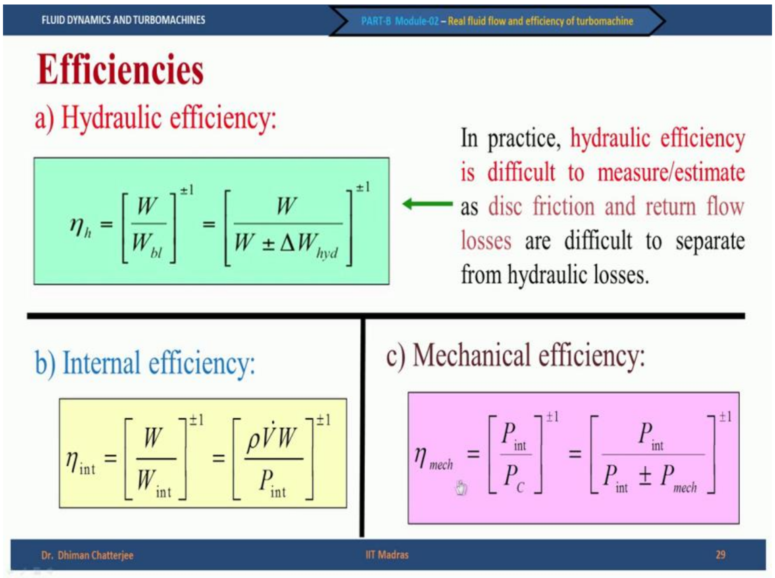

And we can say the hydraulic efficiency as Eta H is equal to W by W BL to the power + -1. In practice hydraulic efficiency is difficult to measure estimate as disc friction and return flow losses are difficult to separate. And hence many times what is done is we talk about internal efficiency as rho V dot W by P internal + - 1, I will talk about what sign you are talking about, it is essentially the + and - for pumps and turbines. And we can say mechanical efficiency is P internal by PC.

Already you know that in case of pumps or compressors, the coupling power from the motor should be more and hence you know the sign convention as I have talked about other cases also and you get P internal by PC.

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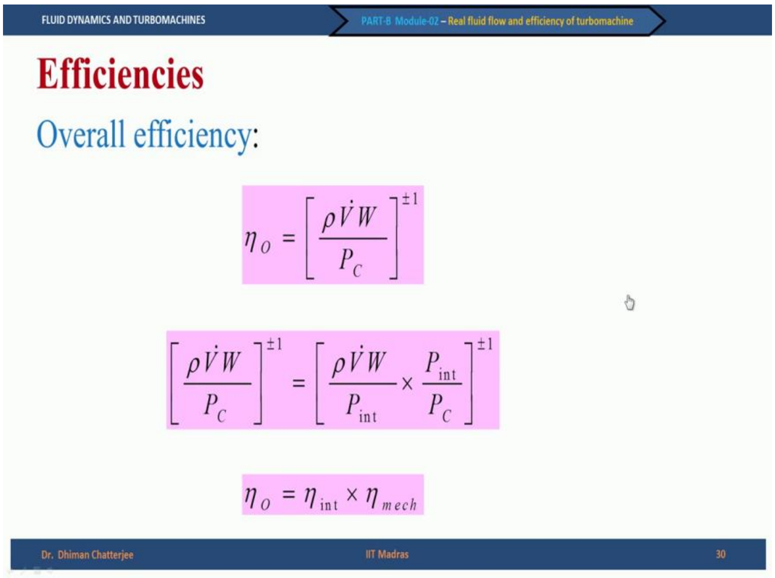

And overall efficiency is the net change in the useful energy across the Turbo machine and the coupling power, the ratio of the 2 to the power + -1. In case of pump the coupling power has to be more and you know which is the + sign, in case of turbine, the actually the fluid has more power and then this will be reversed, we will get a - sign here. So in case of turbine you will write that the PC by rho V dot W, in case of pump you will write it as rho V dot W by PC. And you can show that overall efficiency is nothing but the product of the internal efficiency and the mechanical efficiency.

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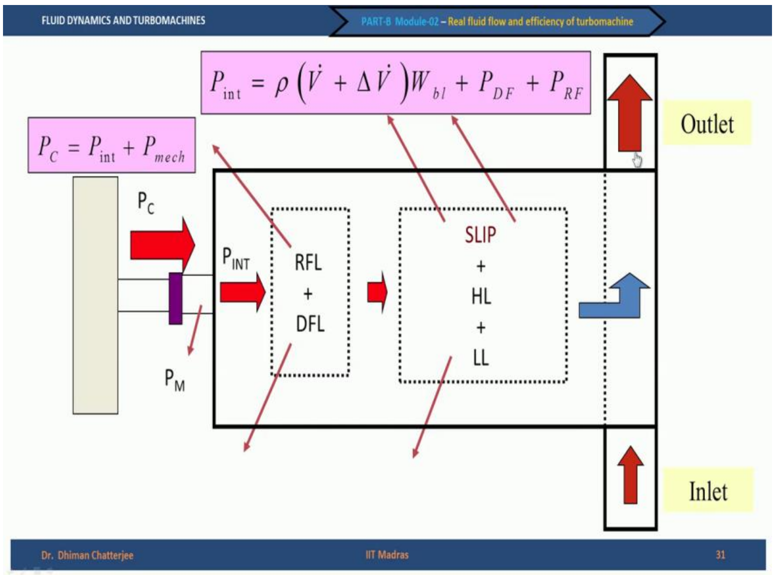

To bring out this flow which is common in a graphical way or with the help of cartoon, let us look at this. This is a motor which is connected to the pump and there is a coupling power which is provided by the motor to the pump and this is my symbol of a coupling between the 2 shafts and there is a mechanical loss. So what happens, the mechanical loss has to be subtracted from the coupling power and what goes into the pump is an internal power P int.

And now there are return flow and disc friction losses which take away some portion of the energy and hence the energy that is available to the fluid to flow is less, we say that PC is P internal + P mechanical and we say that a less amount of energy goes after overcoming the return flow and the disc friction losses. And then we have the slip, the hydraulic losses and the leakage losses, that take away some portion of the energy and we get the P internal and expressed already we have done rho V dot + Delta V dot multiplied by W BL.

If you have W BL infinity from the idealised conditions, you can use slip to get W BL + PDF + PRF and what happens to the fluid? So there is an inlet, a fluid comes in and the pump supplies some energy to the fluid and as a result at the outlet you see this arrow has become bigger, this is a symbolic way of saying that energy is added to the fluid.

However we also should keep in mind that in case of pump the motor gives an energy which is PC which is much more compared to what is actually delivered to the fluid. And hence you can define efficiency accordingly.

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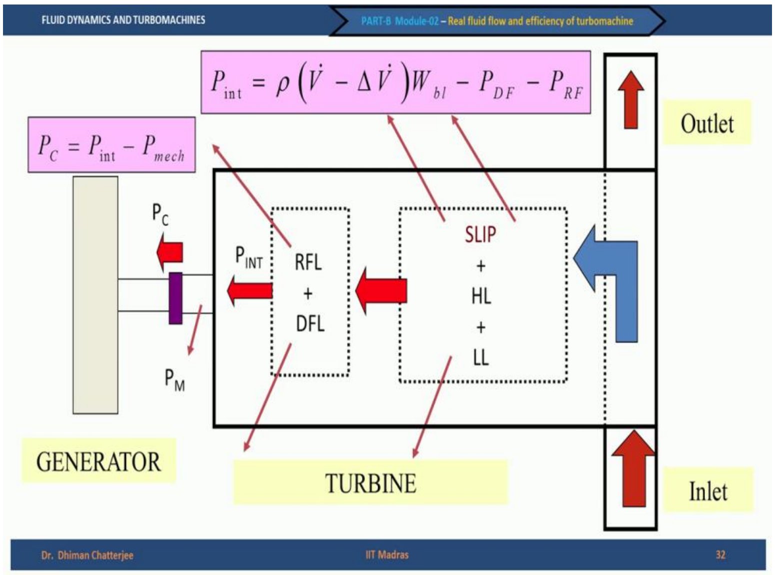

In case of turbine, we have the turbine and the flow takes place in the inlet and the fluid energy is taken by the turbine, it has to overcome the slip, the hydraulic losses and the leakage loss. Then it has to overcome the return flow and the disc friction losses and we get the internal power, what comes out after subtraction of the mechanical losses, we get the coupling power in the generator. And hence this is the output what we get. But if you see that the fluid has actually given much more energy and what goes out in the form of coupling power is much less because it has to overcome the losses.

And we are talking about this as a turbine, so the outlet, what happens, outlet if you see has less energy compared to this inlet because of the fluid, has been taken away some energy by the machine and hence the outlet energy available in the fluid has been schematically shown by a thinner arrow. So this is a turbine and we have the generator.

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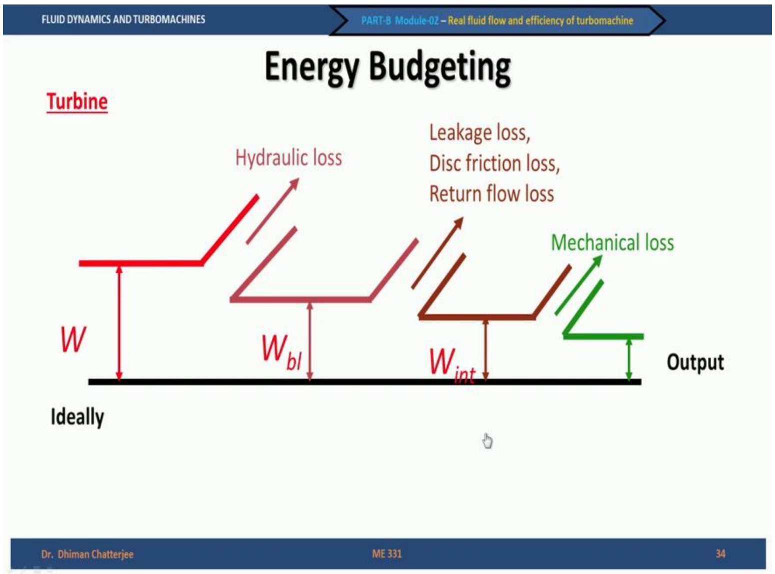

I thought this is useful but if this appears a little complicated, you can think about is simple energy budgeting like this. This is the input, input means which comes from the motor, a portion of the energy is drained away by the mechanical losses, what comes in is the W internal and then you have to overcome the leakage loss, disc friction and return flow loss, you get W BL and ultimately you have to overcome a hydraulic losses. So what is added to the fluid is W which you measure across the pump. In case of turbine there is a reverse scenario.

This is what is available with the fluid which you have measured across the turbine, you assume that all will go into useful work but it does not because a portion goes for hydraulic loss, then we have the blade and then we can say that the portion of the energy that is available in W BL also has to go because of the leakage loss, the disc friction loss and the return flow loss. What comes out of the turbine is even getting reduced by the mechanical losses, what goes, comes out of the generator will be less because of the generator efficiency but we are not considering it here. So what goes from the turbine to the generator is less than what was available with the fluid.

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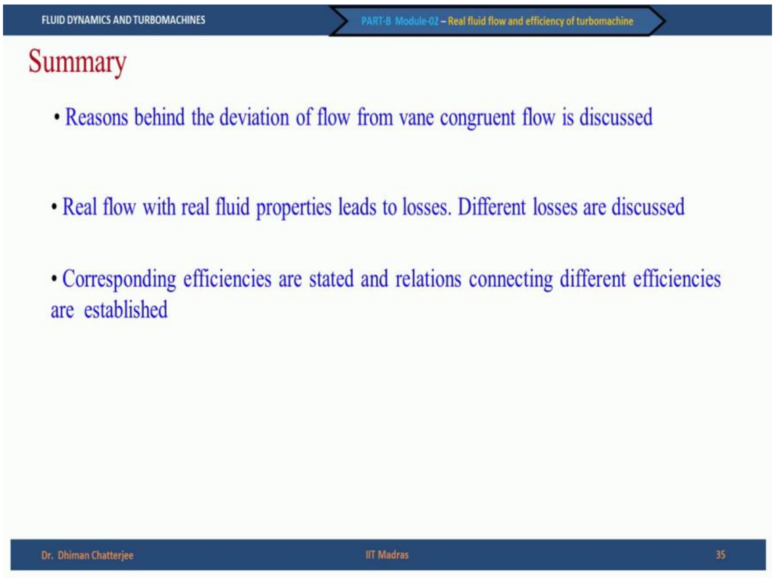

So to summarise we have talked about the reasons behind the deviation of flow from vane congruent flow. We have talked about the real flow with real fluid properties leads to losses and different losses are discussed. And we have talked about the efficiencies which are related with these losses and the relation between the different efficiencies. In the next lecture we will take up some problems which are based on this week's discussion. We will do the step-by-step calculations and also we will find the tutorials which will help you to connect these efficiencies and the losses and the performance of these turbo machines.

In the coming week we will talk about in more detail some of these aspects of efficiencies, etc. in connection with pumps and hydraulic turbines. Thank you.