Fluid Dynamics And Turbo Machines.
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Part C.
Module-2.
Lecture-6.
Pumps.

Good afternoon, I welcome you all for week 7 lecture in fluid dynamics and Turbo machines. In the last 2 weeks we have discussed about various aspects of Turbo machines, mainly the fundamentals, we have derived Euler's energy equation which is the heart of these Turbo machines energy transfer which talks about the energy transfer between the fluid and the blade or either from the fluid to the machine or from the machine to the fluid. But though we have referred to different Turbo machines, we have not gone into details of each of these.

From today onwards, that is this week and the following week we will take up specific Turbo machine applications and we will talk about in details about each of these machines. Today I will discuss pumps. And we already know from our earlier discussion on the classification of pumps based on specific speed.

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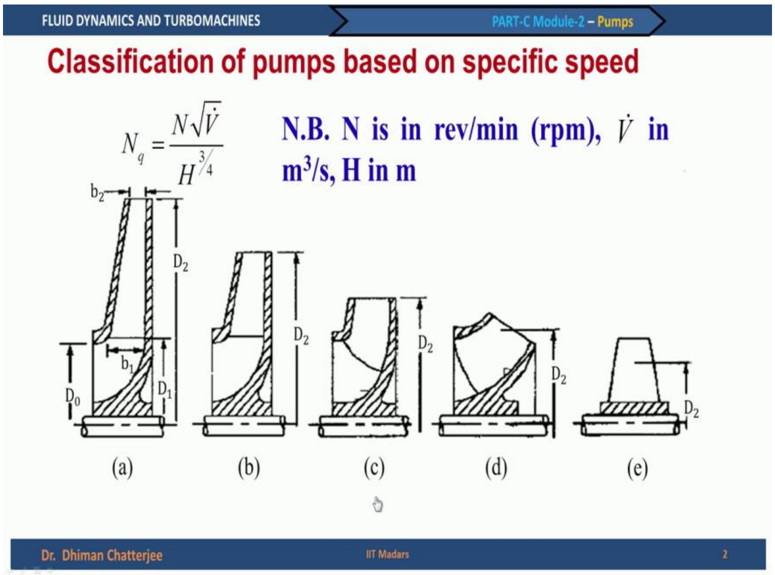

Just recollect that though there are different ways of classifying pumps, we have talked about the use of shape number, we have also talked about the use of specific speed in which the dimensional terms come into picture. Here we will use the definition of specific speed as we have defined earlier which is used mostly in the pump industries. So we say that N is in revolutions per minute rpm, V dot in metre cube per second and H in meters. I was again want to reaffirm that if we use a definition of NQ as I am given the specific speed of pumps, we need to be very careful about the units because NQ has a unit which is rpm in this case.

And we have also seen that as specific speed increases like we see from A to E, the shape of the impeller changes. At very low specific speeds, I will give the values once again when we talk in the next slide, at very low specific speeds in case of A we see that D 2 by D1, that is the outlet diameter to the inlet diameter is large. When specific speed increases, what do we find? We find that the outlet diameter reduces. And you may say that we will still like to have a radial flow pump. So in B we see a radial flow pump with moderate values of D 2 by D1 ratios.

And in C, we find that D2 by D1 has reduced further. One more aspect you will see which is important is that the inlet edge which was straight in case of A or in B has now been curved. This is because as you reduce the diameter D2, then what happens, the blade length which is given in this simple radial flow case is this length will also reduce. If the blades length reduces, then what happens, the guidance to the flow will be improper. And hence in order to extend the guidance further, in order to increase the length of the blade little more, what is done is the impeller has been drawn almost to the eye of the impeller and the edge is made curved so that the blade length can be somewhat compensated, though the D2 by D1 has reduced.

With further increase in the specific speed, such an adjustment is also not possible. And what we get is a mixed flow impeller and you can see that the flow direction inside the impeller makes an angle Theta and that data is less than 90 degrees and more than zero. So it is somewhere in between, the axial and radial. If you increase the specific speed further, then what happens, D1 becomes equal to D2 and the flow becomes purely axial. So if you are given the task of designing a pump, you, we should first ask ourselves what is the specific speed.

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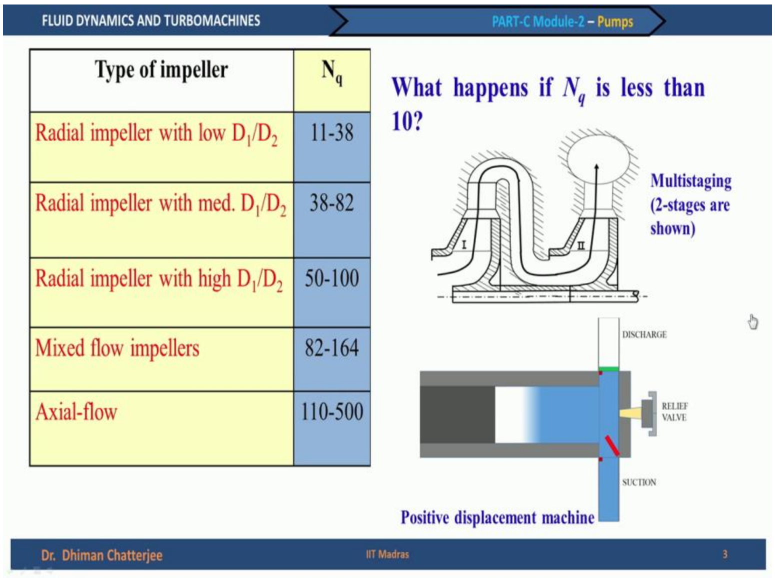

We need to have some charts to know what range the specific speed falls and what should be the impeller shape A, B, C, D or E and this is the guideline which is given for the type of the impellers that we can choose for different values of specific speed. And we see that between 11 and 38 we have radial impellers with load D1 by D2, note that while discussing the last slide I talked in terms of D2 by D1, here it is given in terms of D1 by D2 and hence low D1 by D2 is being mentioned.

Then we have radial impellers with medium D1 by D2 followed by radial impellers with high D1 by D2 and then we have the mixed flow impellers and finally the axial flow impellers. Also you may be inquisitive that why do not we have any values less than 10. What am I supposed to do if I get NQ for some application less than 10. Well the answers are like this. The first thing that we can try is used, we can use multi-staging, that is we can connect 2 impellers in series and stick to the same shaft.

The fluid from one impeller gets energy from this impeller and then it goes to the 2nd one and finally it is collected and taken to the downstreams of the system. So this is possible but sometimes this does not become economical and in these cases recollect what we have discussed about positive displacement pumps that positive displacement pumps are useful when there is a large pressure rise accompanied with a small volume flow rate. And hence you can think of NQ, that is specific speed to below when head is large because head is in the denominator and hence you can also use a positive displacement pump.

I have given a reciprocating pump here just as an example but what I mean is this is a positive displacement pump. Thus when we have NQ less than 10, it is many times advisable, economically desirable to have a positive displacement machine and not a Turbo machines. So in this course we will restrict ourselves to NQ greater than 10.

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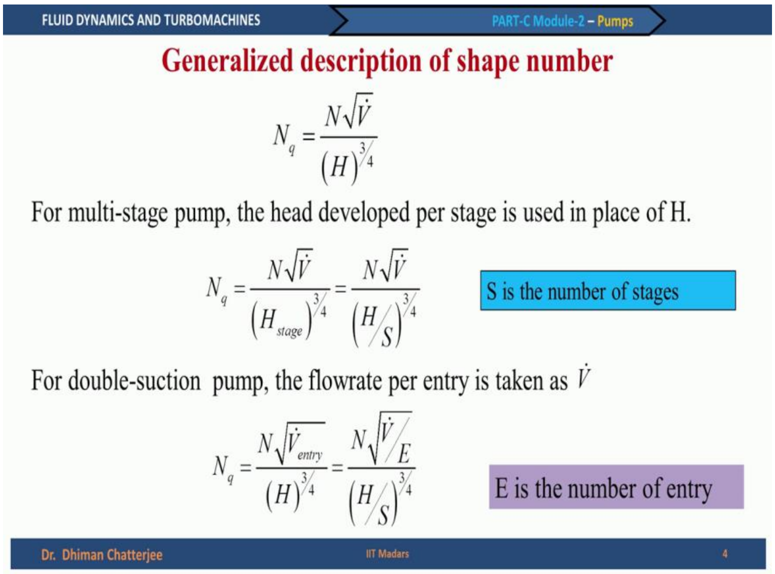

Next we have talk about generalised description of shape number or specific speed. What is the necessity, this is because we know that the pump, just now as we have discussed can be multi-staged. We have shown 2 stages, it can have more than 2 stages. How do I then define the total head developed by the pump and the head that is developed by an individual impeller. Or we can have a volume the flow requirement which is higher, in that case as you can recall we have discussed earlier we can have a double suction pump.

I have already discussed the double suction pump, so how do I describe the specific speed or the shape number in these 2 cases. One, multistage pump, 2nd the double suction pump. Because our definition of NQ given so far does not tell me what is H, what is V dot, it says H is the head developed by the pump and V dot is a flow rate through the pump. So we need to refine this definition of specific speed and shape number, I will give an examplefrom specific speed to tell how this can be done.

Many times what happens is that in a multistage pump we make identical impellers and if the total head requirement is H, head to be developed is H, then what we do is, we divide the head development requirement per stage and have that many number of stages. And hence we should write the definition of specific speed as N V dot by H by S to the power three fourth.

Or in symbolically we can write it as N multiplied by square root of V dot divided by H by S to the power three fourth.

Here H by S refers to the head developed per stage, where H is the total head developed by the pump and S is the number of stages. If we have a double suction pump, of course that has to be considered in the volume flow rate. And hence we can say that we have to talk about the volume flow rate per entry. So what we can do is, we can now define the generalised, the most generalised form of the specific speed or the shape number, I am talking about specific speed particularly as NQ is equal to N times square root of V dot by E where E is the number of entry whole divided by H by S to the power three fourths.

If you have a particular case when it is a single entry, single end suction pump let us say and we have a single stage, then E and S becomes 1 and we get back to the formula given at the top of the slide, that is NQ equal to N V dot, square root of V dot whole divided by H the power three fourth. So now you understand that as the case can be, we have to take care of the number of stages and the number of entries and as the specific speed changes, not only the impeller size changes, we can also talk about the other way the impeller can be different.

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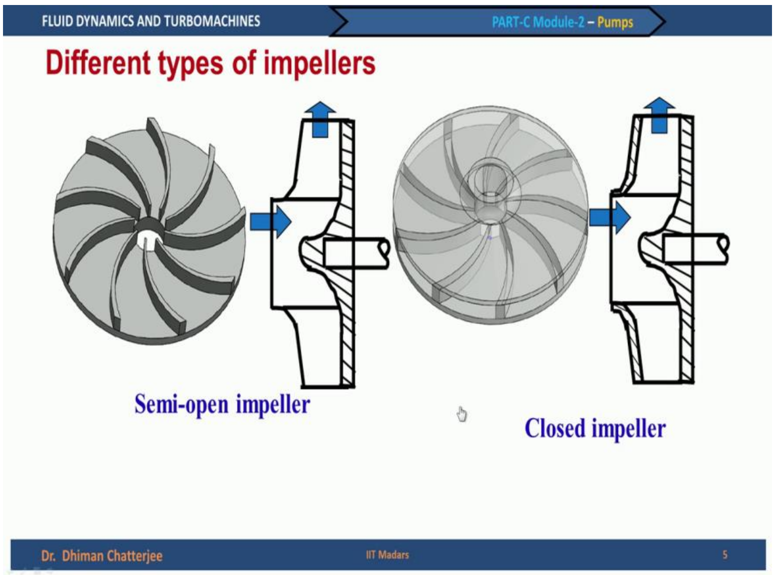

As I have already discussed, let us just recollect what we have discussed in the earlier lectures. We have talked about a semiopen and a closed impeller. And you can also see that time we had not introduced the meridional views, so you can see in the semiopen impeller, the front shroud is missing and this is also described in the meridional view. You can contrast the semiopen impeller with the closed impeller and you can see in this case of closed impeller, we have both the front shroud and the rear shroud and both the shrouds are hatched here. Whereas in the case of semiopen impeller, only the rear shroud is visible, because the front shroud is not present.

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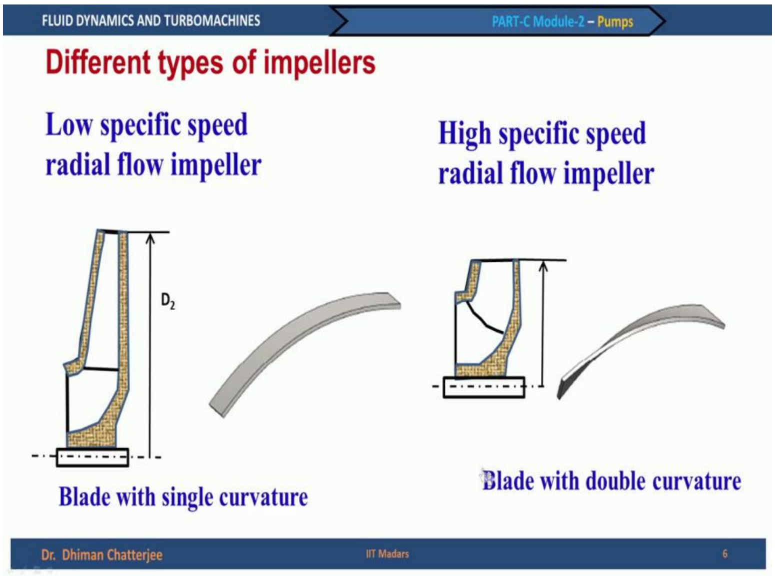

But that is not the only difference as far as the impellers are concerned. That is, impellers can be long or short, radial or mixed or axial, can be semiopen or closed but we can also talk in terms of the individual blade shapes. Let us take the end of low specific speed where we have a radio flow impeller and D2 by D1 is very large or let me say it, in the terms of D2 by D1, D1 is the impeller inlet diameter. In these cases the blades are found to be single curve. Single curvature means, imagine that you have a paper in your hand and you bend the paper and look at it.

So this is a blade, individual blade shown which you can relate to the plate that I have shown here (Refer Slide Time: 11:28). Imagine that I have just taken one of these blades and showing here. So you see that it has one curvature which is bend here, whereas the higher specific speed radial flow impellers, you will see played by double curvature or that which will be visible and you can see the picture given here. How to identify it, you can see that when you look from the top, here we will see only this edge. In this case you can see a part of the surface is visible on the other side and the twist is clear. So we called these blades as played by double curvature.

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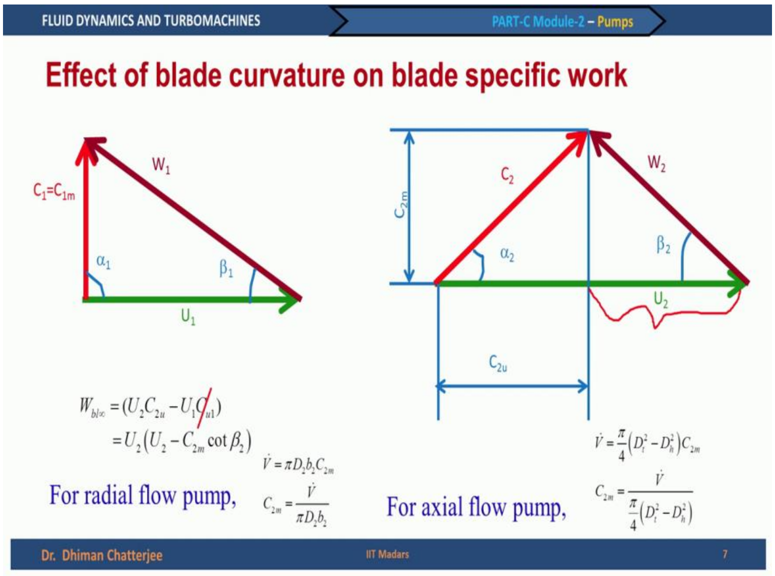

Now this curvature of blades is very important. It plays a very important role in blade specific work, and pressure rise inside the pump and hence in the degree of reaction. You can also appreciate the plate curvature is very important from our earlier discussions because we know that W BL infinity W BL has a term which depends on C 2U. So let us look at that. Let us assume as I have said in many cases earlier that day and let whirl or the pre-whirl centrifugal pump can be neglected. We can assume that the flow is entering in such a way that C1 equal to C1N and C1U equal to 0.

And this is the outlet velocity triangles. U1, W1 and C1 corresponds to the let U2, W2 and C2 correspondence the outlet. I once again remind all of you that this subscript 1 refers to the low-pressure side in case of pump 1 refers to the low-pressure side which is the inlet, the subscript 2 refers to the high pressure side, in case of pump the high pressure side is the outlet. And hence we can write that W BL infinity at this point, I am not worried about the slip and the losses that we have talked about.

We will say we have an idealised vane congruent flow and you can recall that we have used the notation BL for blade and infinity for vane congruent flow and hence W BL infinity refers to the blade specific work under the vane congruent flow, that is idealised assumptions. We will relax these assumptions and we will take into consideration actual specific work later on today. So when we write W BL infinity, we write U2 C2 U and - U1 C1 U and hence by our assumption, C1 equal to C1M and hence there is no whirl component which goes off.

Now look at the outer velocity triangles. This is the C2 U and this is U2, this is C2M, hence what happens is this distance is nothing but U 2 - C2M cot beta 2. And hence once we replace that U C2U by U2 - C2M cot beta 2. We get in terms of U2 and C2 M. Now C2 M or for that matter CM is related with the volume flow rate. For a radial flow pump as I have shown in the last lecture that V dot is nothing but pie D2 B2 times C2M or C2M can be written in terms of volume flow rate divided by the effective flow area letter pie D2 B2.

Since it is a vane congruent flow, I do not have to worry about the blockage because of the blade thickness. If in real life you have a blade thickness, then that can also be accommodated suitably as we have discussed in the last lecture. And for an axial flow, we can write that V dot is pie by 4 tip diameter square - half diameter square multiplied by C2M and we get C2M is V dot by pie by 4 DT square - DH square. What am I trying to say from this is that CM is directly proportional to V dot for a given pump.

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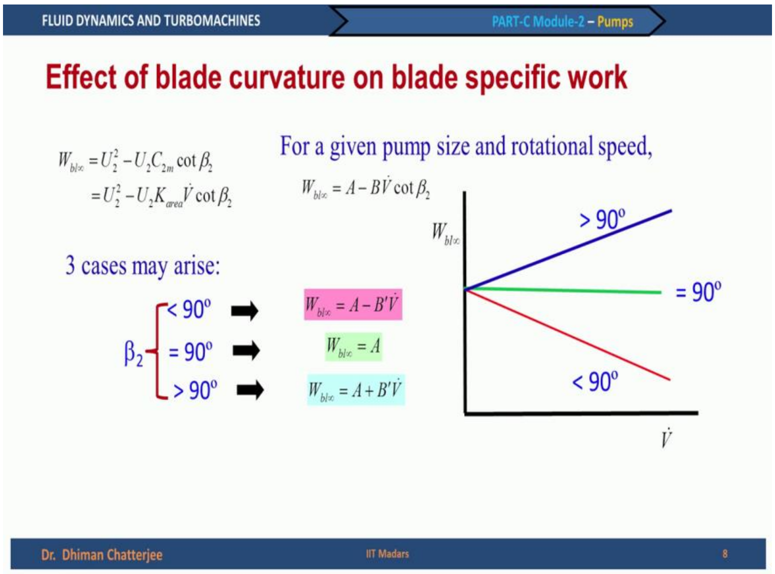

I can write that C2N or C2M is nothing but some proportionality constant which depends on the area times the volume flow rate. So let us see if I write C2M in terms of V dot, what happens. And if we write W BL infinity equal to U2 square - U2 C2 M cot beta 2. Essentially what we have done is we have removed the bracket. So we can write that now U2 square - U2 K area times V dot cot beta 2 as we have just discussed that the C2M is related with the volume flow rate with some area related constant, which is constant for a given pump.

And hence if I say I have a pump of a given size which means my K area gets fixed and if I also say the pump runs at a constant rotational speed, then I should be able to write W BL infinity as A - BV dot cot beta 2. Note here A is the function of the sides as well as the rotational speed, B is a function of the size and the rotational speed, size comes in 2 ways, from through U2 as well as from K area. And hence we get W BL infinity equal to A - BV dot cot beta 2. And 3 cases may arise depending on the values of beta 2.

I may say that I have a value of beta 2 less than 90 degrees, I may say I will have it equal to 90 degrees or greater than 90 degrees. If it is less than 90 degrees, then I get that W BL infinity is A - B prime B dot, why am I writing B prime? Because I am taking B cot beta 2 because in this case beta 2 is fixed. The blade has a definite curvature. Beta 2 recall is the impeller blade angle at the exit. Beta is the blade angle, 2 is the exit or the impeller outlet in case of pump. So beta 2 is the blade angle at the impeller exit as we have discussed in the last class.

So we say that it is less than 90 degree means that whatever be the value, it is fixed. So I can write B cot beta 2 as B prime Times B dot. And we can say that I get a plot is V dot increase as, then W BL infinity decreases linearly. I can also have beta 2 equal to 90 degree. If beta 2 is equal to 90 degree, then this term does not contribute and you get W BL infinity as A and you see that this is the W BL infinity equal to A or it does depend only on the speed and the size of the pump.

We can also have beta 2 greater than 90 degree which gives you, now greater than 90 degree cot beta 2 is negative, we have the - sign here, so - B cot beta 2 can be written as + B prime V dot. And hence we see that with increase of volume flow rate W BL infinity increases linearly. Thus we may be tempted to say that I would like to have a beta 2 greater than 90 degrees because it gives me higher head for a given volume flow rate. That is if I draw a vertical line through here, we will find that at a given volume flow rate W BL infinity is more for greater than 90 degree case.

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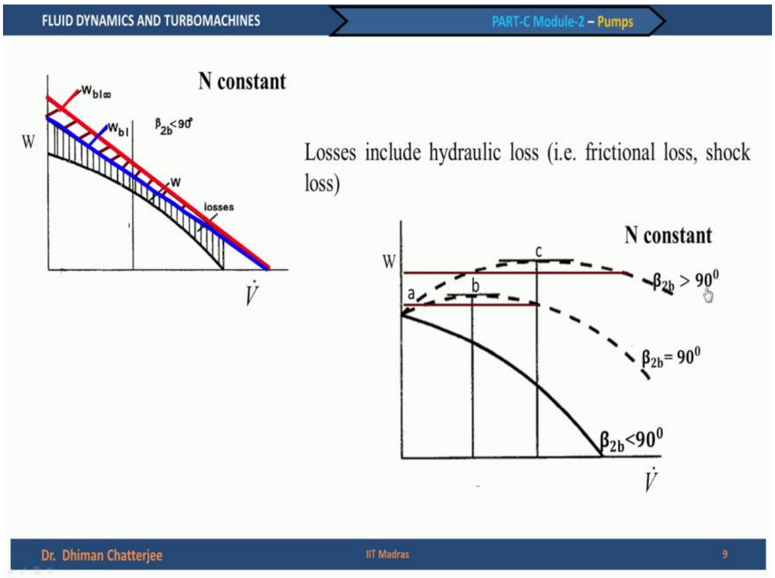

Of course I have assumed that A is same in all the 3 cases but is it really true. This is what we would like to investigate now. We will do it soon but let us look at the actual scenario in which the blade specific work W BL infinity we have said, how does it lead to the specific work of the pump W. So this Red Line shows the linear curve for speed constant and beta 2 less than 90 degrees. I have taken a case where beta 2 is less than 90 degrees. Then we have to also consider that the pump will not have infinite number of blades, vane congruent assumption breaks down, we will have slip.

And once we have the slip, we get the W BL, which is related with W BL infinity if you recollect and we get this blue line. Then we have not still considered the losses. And once we consider the losses we can say that the loss will be minimum at the design point. And at off design point, the loss will increase, why will it increase, it will increase because not only the other losses increase but also the new contribution comes from the shock or the incidence loss as we have discussed.

The shock or the incidence loss is minimum at the design point, nominally it can be taken to be zero at the design point and it increases on either side of the design point. That is at flow rates higher than the design flow rate and at flow rates lower than the design flow rate. And hence we get a point where you have the minimum losses, this hatched, vertical hatched lines in black means the loss and you see the loss is minimum at this point and increases both ways.

And this loss includes both frictional loss as well as the shock loss as we have discussed. This was shown in detail for one blade angle which is less than 90 degrees. However if we have a blade which is more than 90 degree or equal to 90 degree, we have to draw different curves. I want to reemphasise that the speed has been kept constant. If you remember in the earlier derivation we said for a given rotational speed, for a given pump and hence whenever you do pump performance test, these are called pump performance, we have to mention that the speed is constant and the value of the speed alright.

So now what we see is that this curve I am now reproducing here, so for beta 2B less than 90 degree we get a curve as shown, beta 2B greater than 90 degree or beta 2B equal to 90 degree can give you a situation which I will take up little later. It is an interesting situation but not so happy situation for a pump user, so called the unstable curve. I will talk about that characteristics little later. And we have the beta 2B greater than 90 degree as well.

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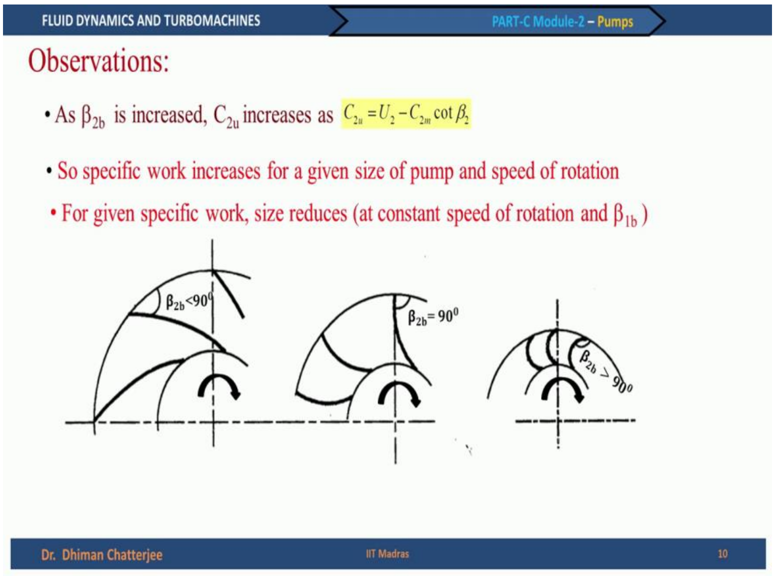

So let us make some of the observations, whatever we have learned so far. As beta 2B is increased, C2U increases as C2U is U2 - C2M cot beta 2 and we know that cot beta 2 value within nothing but cos beta 2 by sin beta 2 will now have a value going reducing and hence this contribution goes down and we will have C2U value increases. Of course beyond 90 degree this becomes negative and we get increase further because of the sign change. So specific work increases, just now we saw for a given size of pump and speed of rotation.

We have not yet explored whether that is good and desirable, that we will take up soon. For a given specific work, let us say I am interested in a given specific work, you may say that why should I keep the pump size constant, I do not need to do that. So in that case what I can say I will keep the specific work constant and reduce the size while keeping the speed of rotation constant and of course in all this assumptions we have made that the inlet blade angle is not changed. So let us consider the case that if we have U 2 C2U which is my W BL infinity, I want to keep the product of U2 C2U constant.

And if the specific speed is, specific work is constant, then what happens, then we see that if C2U increases, U2 has to reduce. A speed of rotation is kept constant, it leads to the diameter has to reduce and we get very compact blade. So you see this is the relative diagram where we can show that the blade, as the angle is less than 90 degree, the size is bigger, this radial tip blade, I will talk about it when beta 2B 90 degree and finally beta 2B is greater than 90 degree we see that the for the same work, specific work, the size has reduced.

That is simply because, I again repeat the mathematical relationship also will tell you that W BL infinity is nothing but U2 C2U and C2U changes and hence to keep W BL infinity constant U2 has to reduce.

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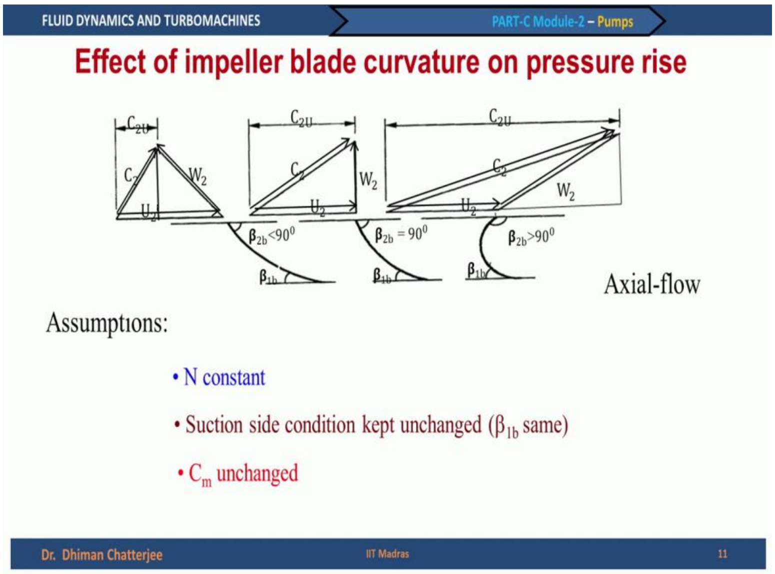

So let us look at this for an axial flow machine. Axial flow pumps we have already discussed and the flow takes place from 1 to 2 and what we see here is that we see that from left to right the blade angle increases. In the left side we have beta 2B less than 90 degree, in the right one we have beta 2B greater than 90 degrees. And we can say as the blade angle becomes greater than 90 degree, what has happened is the C2U component has increased tremendously. C2 has increased, C2U has increased tremendously.

You look at the leftmost velocity triangle and the rightmost one. You see that C2U is much smaller here compared to the C2U over on the right-hand side. So what we are assuming right now is that beta 1B will be kept constant, the speed will be kept constant as we have discussed already, these are very important because if we start changing these, the entire derivation becomes more difficult. Nevertheless we have the same relationships, the same effect can be observed. We are trying to keep it same, particularly the beta 1B because we do not want to understand the effect being compounded by the beta 1B.

And then CM is unchanged. So in all these cases, we are keeping the meridional component of the velocity, absolute velocity to be constant.

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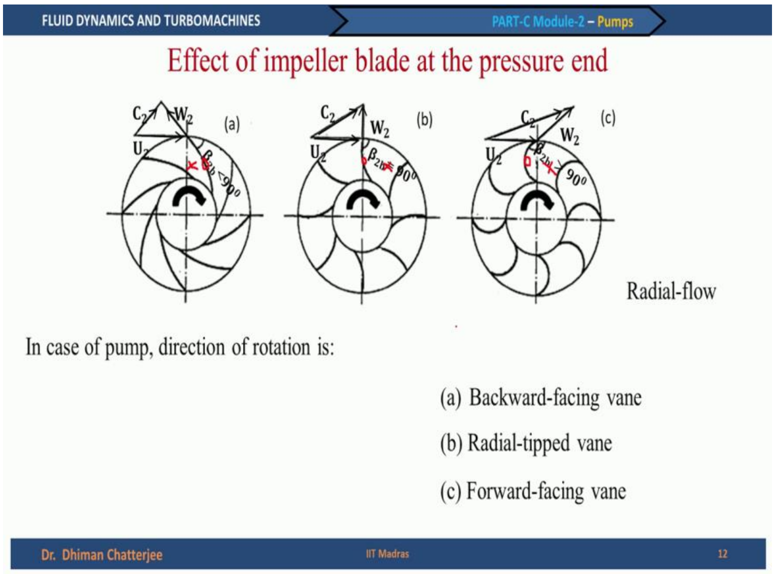

Similarly we can talk about in case of the radial flow pumps. In the case of radial flow pumps, it is slightly more inward, let us try to understand that. We have discussed direction of rotation in case of axial flow pumps. How did we define it, we said the direction of rotation in case of pumps is dictated by the blade, the blade makes the fluid rotate and hence the direction of rotation in an axial flow pump which is basically the blade shape of having an aerofoil from the suction surface towards the pressure surface.

In case of pumps I have the direction of rotation let us say fixed as shown in each of these figures. I can have a beta 2B which is less than 90 degree, I can show you that the arrows are also shown so the flow leaves the blades at this point, we have W2 C2U 2 in each case and we have, the names are given for these blades as beta 2B less than 90 degree, we say the blade is backward facing vane or backward facing blade. Why do we have such a name for these blades? That is because you imagine that you are standing on these vanes and where is the centre of curvature of this vane?

It is somewhere here. That is if I say that I have a vane where you are standing here, you are here and the centre of curvature is here, then what happens, you are standing here looking at the centre of curvature, you are looking against the direction of rotation. You are looking backward to see the centre of the curvature of the blades. Whereas you take this one, in this case you are again sitting on the vanes but the centre of curvature is somewhere near which means you are looking in the direction as the direction of rotation of the blades.

So this one will be considered as the forward facing vane. If you look at it more attentively, the 2nd one is also the same. In this case you have the observer here and the centre of curvature here which is again in the direction of rotation. However for the case B, we give it a special name to ensure the fact is kept in our mind that beta 2B is 90 degree and hence we say that beta 2B is 90 degree is going to be the case of radial tipped vane. So you understand that broadly speaking I should have classified it as backward facing and the forward facing vane.

But though radial tipped vane is essentially a forward facing vane, we call it a radial tipped vane to make us feel comfortable or aware that this beta 2B is 90 degrees. Okay, so there are 3 possibilities, both in axial as well as radial flow pumps. Which one should we go for, which one is normally the practice, let us try to understand that? Before giving the answer, let us do a little more of the work.

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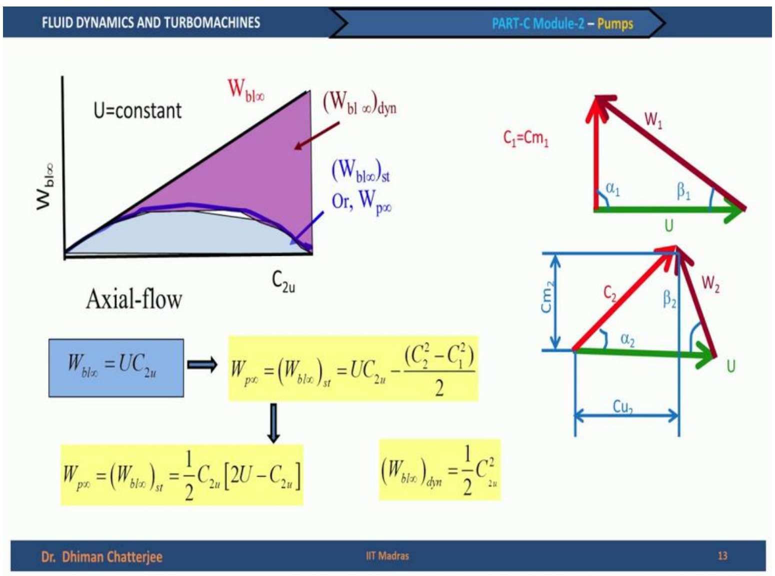

So let us talk about the velocity triangles and like I have discussed, I have taken C1 equal to CM 1 and I have taken the C1U which means is zero, so for an axial flow machine, please note that the derivation that I am trying to show you to bring out the effect of the blade angle need not be restricted to axial flow. You can do the same derivation for radial flow pump as well but the calculations will be more cumbersome, so I feel that to bring out the physics, to bring out the significance of the blade curvature, it suffices if we take the axial flow pump.

And also the nice assumption of C1U equal to 0. So in this case I know that U2 equal to U1 and hence I can write that W BL infinity is U times C2U. And which we have already done, we can say that we have already discussed while discussing the degree of reaction that a portion of the energy is used to increase the pressure in pump and the other part of course goes for the dynamic head. So we can say that WP infinity, this is my way of saying that we are talking about the component of energy transfer which goes into the pressure, P here signifies the pressure.

And infinity to remind us again that we are restricting ourselves to vane congruent flow, ideal flow, lossless flow and we can also say like some books of will say that W BL infinity ST for static. So whatever we notation we follow, we are essentially talking about the static pressure change. And we know the static pressure change is nothing but the total work transfer W BL infinity - C2 - square - C1 square the whole divided by 2. So let us write that, we write the WP infinity is equal to U C2U - C2 square - C1 square by 2.

Which gives us that WP Infinity is equal to half C2U multiplied by Twice U - C2U. This result is important. What it says, if C2U is 0 which is a trivial case because if C1U is zero and C2U is zero, there is no power transfer, there is no energy transfer. So that is a trivial case which cannot be accepted. But anyhow, mathematically if C2U is zero, then there is no pressure rise but C2U if it is equal to twice the U, then also you get no pressure rise. What does it mean? When there is no pressure rise we are talking about an impulse stage when there is no pressure change.

So what will happen is in that case we will see that WP infinity is zero at C2U equal to 0 and at C2U 2U. If we subtract the static pressure change from the total, we get what is known as W BL infinity dynamic, DYN stands for dynamic which is half C2 square. So what will happen, W BL infinity goes with C2U as linear and this blue shaded region, this one gives me the limit of W BL infinity or the static pressure change, you can see that this area is zero at C2U equal to 0 and C2U equal to 2U which is the impulse case.

And W BL infinity dynamic is maximum at this point where C2U equal to twice U. So what we see, if we increase blade angle beta 2, then what we increase is C2U and if we increase C2U further and further, a point will come where there is no pressure change. Is it good, is it desirable, though your W BL infinity has increased? The answer is no. So why it has no, we will explore now.

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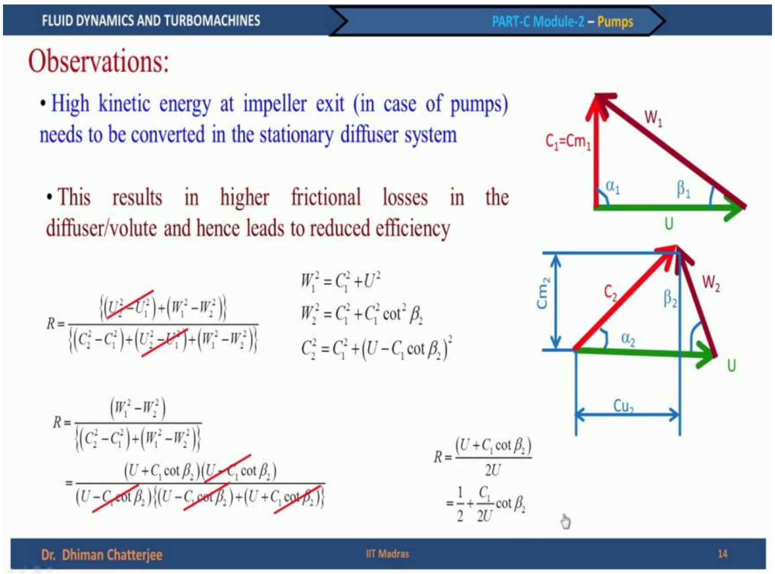

To go into that we scale that high kinetic energy at the impeller exit, in case of pumps needs to be converted in the stationary diffuser system. What is the objective of a pump? To lift water, to raise water, let us say from the ground level to the top of a multistoried building, a high-rise building. So which means that the fluid should have enough pressure which can take it to the top of the building. Now we have a situation when the impeller exit, there is the extreme limit, in the extreme limit when C2U equal to 2U there is no static pressure, it has a very high content of kinetic energy.

So if we have to convert this into pressure then we need to have diffuse the flow and in the stationary diffuser if we know from fluid dynamics as you have studied in the first module of this course, you have studied that there will be losses. If the kinetic energy is high, the velocities are high, then the losses are more and hence when we try to convert this pressure,into pressure rise, we will get less energy conversion. And hence because of these high frictional losses the efficiency will be low. So if we take this example, it is not just for pumps, it is also valid for fans, blowers, etc.

So let us take where we can say this is useful, the high blade angle, is it at all useful in any case or not? Let us take 2 examples. One of a fan or a blower which is used in mines or any such heavy industries. So now we need a high volume flow rate, we also need in order to pay less money for this electricity consumptions, the pump to be more efficient. So in such a scenario, industrial scenarios, it is prudent to use the beta 2 value which is less than 90 degree or in other words backward facing vanes.

However imagine that you are in one of these shopping malls where the shops are actually paying for the space occupied, the per square feet area or per square metre area as the case you want has to be paid. So in that case we really cannot afford to have a large space being occupied by fans. There are wind curtains where you see as you enter you get the pleasant wind. So in such cases what is required is the space should be small, the size of the fan should be small and we have already seen that for the same work output the size can be small if we take beta 2 to be large.

And so it may be possible in some cases to use a fan with a forward facing vane or a pump with a forward facing vane, you if you come out with some example but it is generally not the chosen practice. In practice, we use a backward facing vane for the pumps and the fans. In the case of pumps, most of the times we use a backward facing vane and not a forward facing vane. Now the next thing is the degree of reaction, I told you intuitively with the pressure rise that WP infinity, the pressure rise component goes to 0. Let us look at it from the concept of the degree of reaction.

This is the degree of reaction as you have studied in the last class that U2 square - U1 square +, note that it is W1 square - W2 square whole divided by the total energy transfer which is W BL infinity. And if, since we have assumed that it is an axial flow pump, so what happens, U2 equal to U1 and this term drops off. And we can write the W1 square from the velocity triangle is nothing but C1 square + U square, W2 square is nothing but C1 square because I have assume that C1 equal to CM1 equal to CM 2, so it becomes C M square + WU square which is nothing but C1 square cot beta 2, this angle is beta 2 and this is CM1 equal to C1.

And we have C2 square equal to C1 square + U - U1 cot beta 2. That is I have assumed that CM remains constant and that value is given as C1. So when we derive this, we can say that it becomes W1 square - W2 square whole divided by C2 square - C1 square + W1 square - W2 square. I can substitute these relationships which we have obtained here and get the relationship in terms of U, C1 and beta 2. And we get that R becomes equal to U + C1 cot beta 2 by 2U.

You can verify it yourself that these terms, C1 cot beta 2 gets cancelled and you have to you in the denominator and you have U + C1 cot beta 2. Are we can simplify it and that in terms of half + C1 by 2U cot beta 2.

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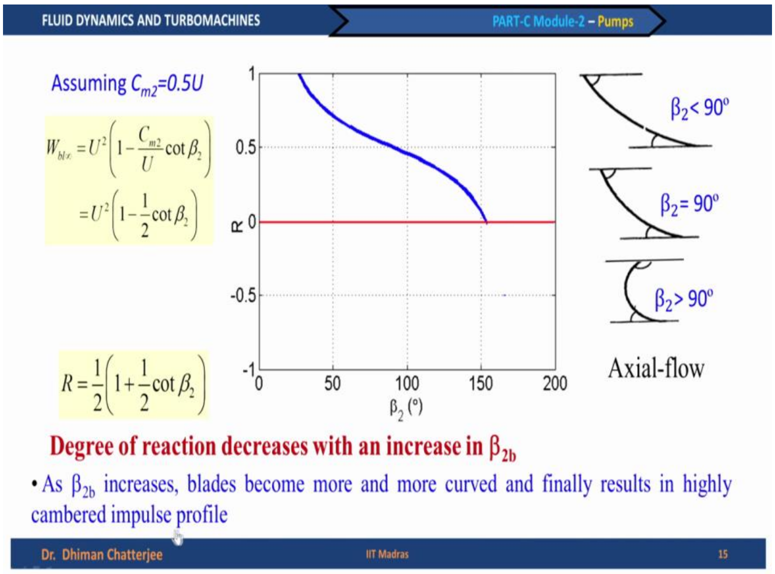

And now if I had assumed a special case, this is not required but just to do a small calculation and see how the degree of reaction varies, I can say that CM2 equal to 0.5U and hence I can write the W BL infinity is U square whole multiplied by one - half cot beta 2 and R is nothing but one + half cot beta 2. There is no necessity again I am repeating that this has to be correct but I chooses this as an example to show how the degree of reaction varies. I have plotted that the degree of racial actually goes to 0 at an angle slightly more than 150 degrees.

And you see the degree of reaction reduces. So corresponding to beta 2 value increases. So this is not desirable and hence we will try to avoid it and we try to give the outlet angles smaller than 50 degrees or so or close to that (())(42:57) also. So then we can say that as beta 2B increases, blades become more and more curved and final results in highly cambered Impulse profile.

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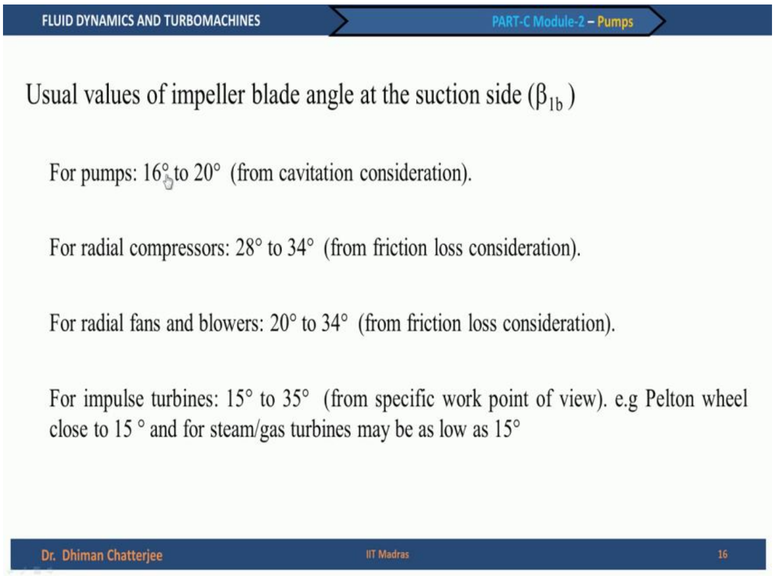

Usual values of impeller blade angles at the suction side beta 1 B is given as, for pumps it is between 16 to 20 degrees, we have to also consider something called cavitation. I will talk about cavitation and cavitation in pumps towards the end of this week. And for radial compressors we can have it between 28 to 34 degrees, we do not need it at this stage and for fans and blowers. So these are the typical values we normally use, I give it in one place so that you can have a comparison. So typically we are talking about for pumps is between 16 to 20 degrees.

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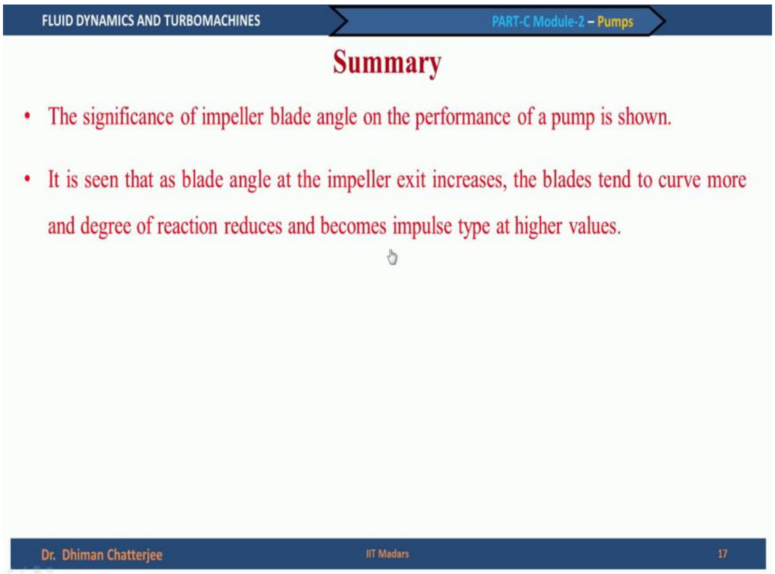

We have discussed so far the effects of the blade angles on the pressure rise inside the pump impeller in terms of degree of reaction and we have also discussed the blade specific work that can be obtained and that how that varies with blade angle. So let us look at the significance of the impeller blades on the performance of the pump. Let us review this point, we said that as the blade angle increases, the impeller becomes more and more curved, the degree of reaction reduces and in fact we have shown that at higher values the pump can become Impulse type.

But as we have discussed in the last lecture the Impulse type of pump is not possible, is not feasible, does not make sense because we want to increase the pressure rate and hence this is not a very good practice. Hence in all practical applications, the impeller blade a angle at the exit is not greater than 90 degrees. Or in other words we talk about the backward facing vane as a desirable choice of blade. Thank you.