Fluid Machines.
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Lecture-21.
Characteristics of a Centrifugal Pump.

Good morning and welcome you all to this session of the course on fluid machines. Now in the last class we discussed the velocity triangles of a typical impeller blade of a centrifugal pump and we derived the expression for the head imparted to the fluid other centrifugal pump or the impeller of the pump and the head developed and the difference between these was recognised as the losses due to fluid viscosity and the ratio of these 2, that is head developed by the pump to that given to the fluid by the impeller was defined as manometric efficiency.

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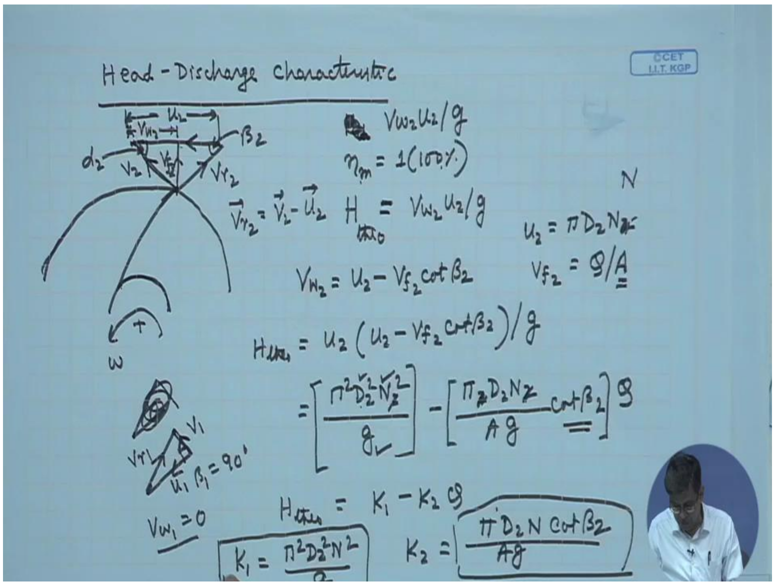

So today we will discuss the most important feature of a centrifugal pump, the head discharge characteristic. Let us write then, head discharge characteristic, head discharge characteristic. Let us consider an impeller as we already shown earlier and part of an impeller, I am not trying the full one, this is the centre of rotation, let the impeller rotate like this at an angular velocity Omega. This is the eye of the impeller, part of the eye and this is the impeller outlet and we consider an impeller blade like this as we discussed earlier.

Now here the outlet velocity triangle remains like this, this is the outlet velocity triangle. This is VR2, the relative velocity at the outlet which was already discussed earlier, this coincides with the impeller blade angle of the outlet which we defined as beta 2. This is the absolute velocity direction, this angle is defined as alpha 2 by the nomenclature, this is the flow velocity at the outlet VF 2. So this is the tangential component of velocity at the outlet and this is obviously the rotor velocity at the outlet, so the rotor velocity at the outlet.

And this is like this, how to draw this triangle, VR2 as I have told V2 - U2. That means VR2 + U2 is V2. You see VR2 and U2 are in the same direction and V 2 represent the other side of the triangle but in the opposite direction. This way we have to draw the velocity triangle, the inlet velocity triangle I draw it here is like this where we do not, sorry. We do not want a, we do not want any swirling component of velocity. So it is the radial direction, this is the tangential direction, so this is VR1, this is U1 and this gives the V1 this is the beta 1 which is 90 degree because VW 1 U1 is 0, this was discussed earlier.

So now we come like this, theoretical head, that is the head, let us consider 1st not theoretical head, as we discussed that the head, that is energy per unit weight imparted to the fluid or given to the fluid by rotor is VW2 U2 by G because VW 1 is 0. Now if we consider the manometric efficiency Eta M to be 1, that is 100 percent, means if we neglect the viscous effects or the losses due to viscous, viscous dissipation, neglect that, then we can tell the manometric efficiency 100 percent, the head developed by the pump equals to VW2 U2 by G, manometric efficiency 100 percent and this is termed as theoretical head.

So therefore theoretical head is VW2 U2 by G. Now from this velocity triangles you see that VW2 is U2 - this part which is VF 2 cot beta-2. Now again we see that U2 is pie D 2N. And what is VF 2, VF 2 is can be written in terms of flow rate Q divided by the cross-sectional area or the area normal to the flow velocity, which is area normal to this velocity, this is flow velocity, which is pie D2 times the width of the impeller at the outlet. So this is area. For given impeller, this is fixed.

So now we can write H theoretical is equal to U2 into U2 - VF 2 cot beta-2 divided by G. Now if we put these values of U2 and the VF 2, we get that U2 square, that is pie square D2 square, N2 square by G - U2 VF 2, that means pie 2 D2 N 2 divided by AG cot beta-2 into Q. These 2 terms I make the bracket. Now for a given impeller of a centrifugal pump, D2 is fixed, moving with this N2 is N actually, pie D2 N, this is N, we have written unnecessarily N 2, there is no suffix required because the impeller rotates at a constant speed N.

So therefore pie D2 N for a given rotational speed, this is local gravity. For a given impeller, the outlet angle is also fixed, so therefore these 2 are constants, so therefore H theoretical can be written as K1 - K 2 Q where K1 is equal to pie square D2 square N square by G and K2 is equal to pie, sorry, sorry pie, not pie square. Pie D2 N by AG, pie D2 N by AG, okay into cot beta-2. So this is true that this is given by K1 by K2 pie D2 N cot beta-2 by AG. So this is K1 and this is K2, constants.

So therefore we see under the theoretical conditions, neglecting the viscous effect, we can express and this trend will be this, that means H theoretical is decreasing with Q and under theoretical condition it is a linear variation, linearly decreasing with Q given by this equation. The head developed is equal to K1 - K2Q, K1, K2 are given by this.

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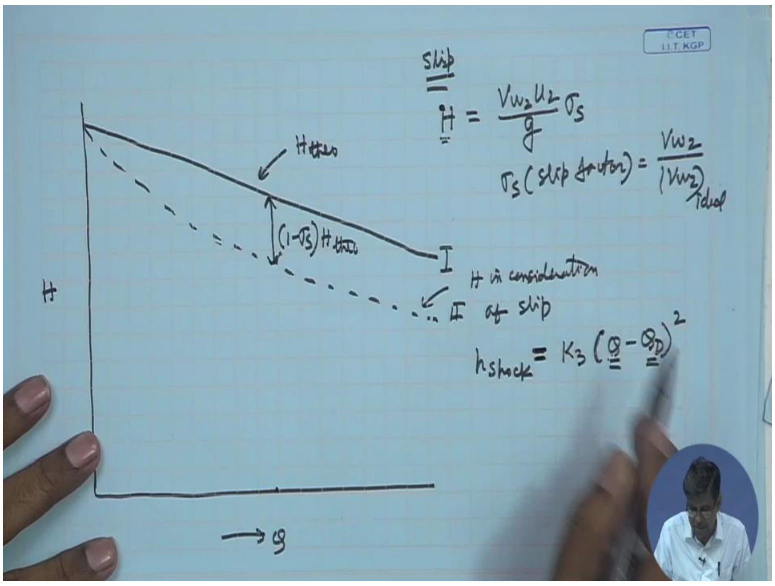

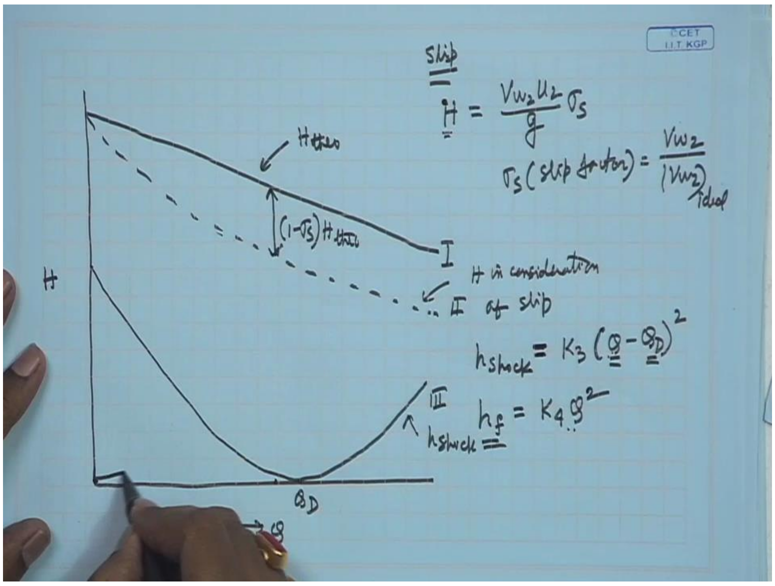

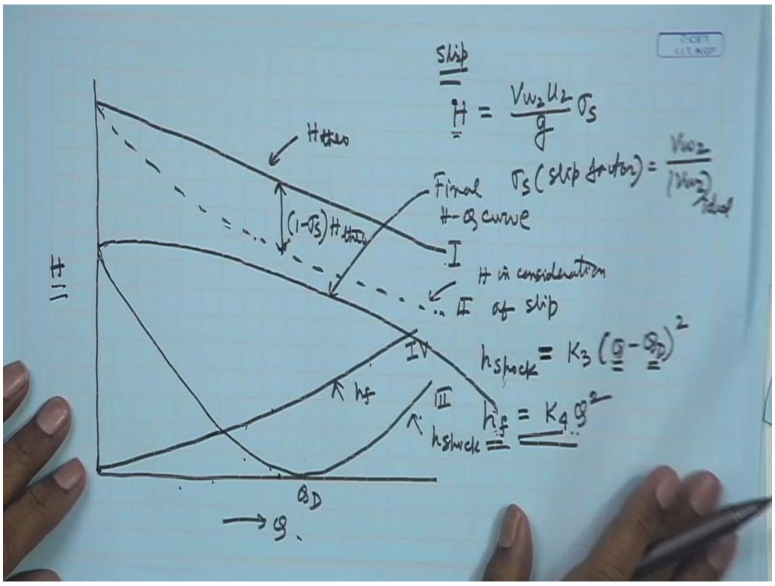

Now if we draw this in an HQ plane, H is the head, Q here. Then what this thing, this is the theoretical, this is curve 1 which we tell as theoretical head. Now if we recognise one by one all the things happening in practice or all the practical features we take into consideration. 1st we take slip, as we define slip as the phenomena that happens because of the pressure difference at the leading and the trailing sides of the blade for which there is a circulation and nonuniformity of velocity in the blade passage which makes the, which reduces rather reduces the tangential component of flow velocity at the outlet and reduces the head developed or the energy imparted to the fluid by the pump.

And this slip phenomena is independent to that of viscosity and it is purely the influence of curvature of the blade and this is 0 when the flow is 0 and at low flow, this is low. This is a monotonically increasing function of the flow. So therefore in consideration of the slip we know that we can write the developed, actual head developed is equal to VW2 U2 by G into Sigma S. Where Sigma S is the slip factor, Sigma S is the slip factor which we defined earlier as the actual VW by the ideal VW, okay.

Now with this, if you draw, this is increasing with flow rate, the curve is diverging, take care of that and draw this H, then this is the curve 2 which is H in consideration of, in consideration of slip. And at any point, at any point means any value of Q, any point in the abscissa, this is 1 - Sigma S into H theoretical, this gap, each and every point. That means this is, this is shown by the dotted line, this curve represents H in consideration of this head after consideration of the slip.

Now I will consider 2 other losses, one is the shock loss. What is the shock loss? As I have told earlier, that the blades of any Turbo machines or fluid machines is designed in such a way that at the design condition, the fluid at inlet and outlet should match the angle of the blades or vane at inlet and outlet respectively so that it can glide the blade at inlet and outlet. But at off design conditions where the flow is less or more than the design flow rate, what happens, this condition is not met.

So what happens that the velocity relative to the fluid is not gliding the blade at inlet or outlet. That means the angle of the relative velocity differs from that of the blade angles at inlet and outlet. Physically what does it imply, the fluid obliquely enters the blade and it impinges the blade because of which there are eddies created and which reduces the mechanical energy through which the, a part of mechanical energy is distributed to inter molecular energy.

This we see as a loss and this loss in mechanical energy is known as shock losses. And therefore these shock loss is proportional to the square of the velocity. As it happens always that any mechanical energy lost due to frictional effect of the fluid like the viscous effect is usually proportional to the square of the velocity game turbulent flow region. And this is square of the flow rate, since the flow velocity is proportional to the flow rate and this can be expressed in terms of some constant K3 into Q - QD whole square why, where Q is off design and this is design flow rate, so this always gives a high-value whether this Q is less than QD or Q is more than QD and this is a scalar quantity, this is the energy.

So another, apart from the shock loss, there is another loss of energy which is the frictional loss in the pump. That means in the impeller and in the diffuser and in the volute casing, volute casing itself is the diffuser, is the fictional loss which can also be expressed in the turbulent region as square of the flow velocity or square of the volumetric flow rate. So if everything is expressed in terms of the fluid which is common and same through the entire pump, then we can show this 1st one, the shock loss is like this, that means this is 0 at the design flow rate and this attends a very high-value by the flow rate tends to 0.

While this one, HA, the frictional losses which is 0 at the 0, this is curve 3, this is H shock and the frictional losses, the curve is like that. That means this is this equation, this equation, H is this K 4 Q square this is curve 4, this is frictional loss due to viscosity. Shock loss is also due to viscosity but the phenomena is different. Now if you take care of these 2 from this head, that means if you subtract these energy losses from this head developed in consideration of the slip, that means geometrically if we sum matter any abscissa, sum the ordinates of curve 4 and 3 and deduct that from curve 2, we get the final hate discharge characteristic which is like this.

And this is the final HQ curve, final HQ curve. This you get by subtracting these 2 from this head at each and every flow rate. So finally this is the HQ characteristics which comes from a theoretically linear one in consideration of slip, shock and the friction in steps so that one can draw and this way you will see that the relationship of HQ looks like this in a centrifugal pump.

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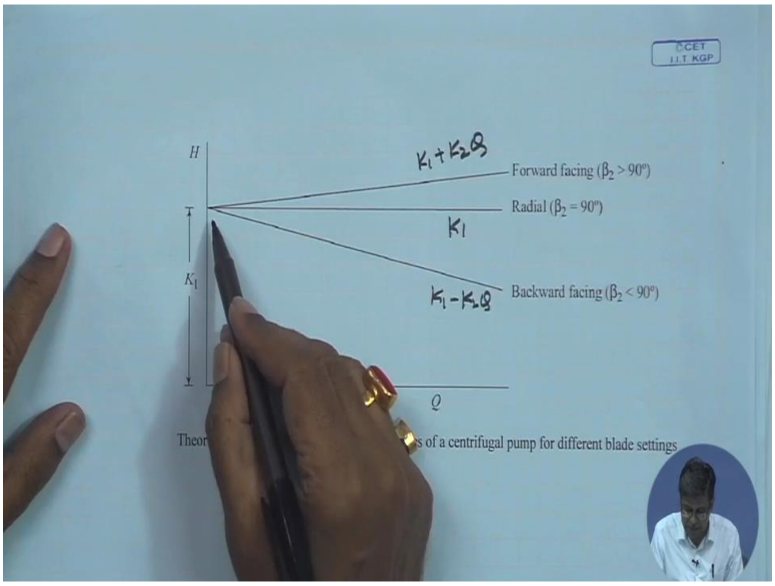

Now we will consider the influence of or effect of which is another important thing, outlet blade angle on this HQ curve. Effect of outlet blade angle on this HQ curve. Already I have written it for your convenience, now there are 3 types of blades which are being defined, which can be used. One is this type where the curvature of the blade is in the direction of the rotation which is known as forward facing vanes. And here the velocity triangle will be like this, you see this is the VR2 matching the blade angle at the outlet.

This is under ideal conditions, that means design conditions, so this is matching with the outlet impeller angle. This is the radial blade, plane, there is no curvature by the blade outlet angle is 90 degree and the relative velocity, here also the relative velocity matches this angle, the relative velocity is 90 degree, that means 90 degree to the tangent matching the outlet blade angle. Here the blade is in this shape were the curvature is in the opposite direction of the rotation, this side is the curvature, that is known as backward facing or backward path blade.

These type of blades are usually preferred, that I will come afterwards, I will tell you and here the velocity triangle is like this, which we deduced earlier. We took this type of impeller blade and the velocity triangle is like that. Now you see from these 3 types of blades, forward facing, radial and backward facing and their velocity triangles, the main difference is that in this case the value of beta 2 here, this one is obtuse, you see this is the beta-2, this is the beta-2, this is obtuse, greater than 90 degree.

Where the value of beta 2 is 90 degree and here the value of beta 2 is less than 90 degrees. So if you remember this formula which we derived earlier that theoretical head is K1 - K 2 Q is given by this relation where you just recollect that K1 - K 2 where K1 is this and K2 is this, so depending upon the sign of cot beta-2, this will be K1 - K2 Q or K1 + K2 Q. Now you will see in case of, if you see it here, sorry this is difficult. I am telling you in case of forward facing blade, K2 is less than 0 because beta-2 is less than, cot beta-2 is obtuse, so cot beta-2 is negative.

So K2 is less than 0, that means K2 is negative. Similarly for radial vanes, beta-2 is 90, cot beta-2 is 0. So therefore here, cot beta-2 is 0 means K2 is 0. So H theoretical is given by K1. Whereas in case of this type of blade, that is the here we deduced that is the backward facing blade, here it is done in the opposite way and the direction of motion, rotation is also changed accordingly. Beta-2 is less than 90 degree, that means cot beta-2 is positive, this is an acute angle.

So here it is in the form of K1 - K2 Q. So if you consider a theoretical as K1 - K2 Q, then we can tell that K2 is less than 0 in case of forward facing vanes, K2 is 0 for a radial blade and K2 is greater than 0 for a backward facing blade. So if you consider this, then we can tell for a forward facing vane, the relationship is like this where K2 is less than 0 that means it will be K1 + K2 Q type, that means head increases with the increase in the flow rate. Okay, or the discharge. In case of radial blade cot beta-2 is 0, so therefore K2 is 0, so it is independent of the flow rate, constant head.

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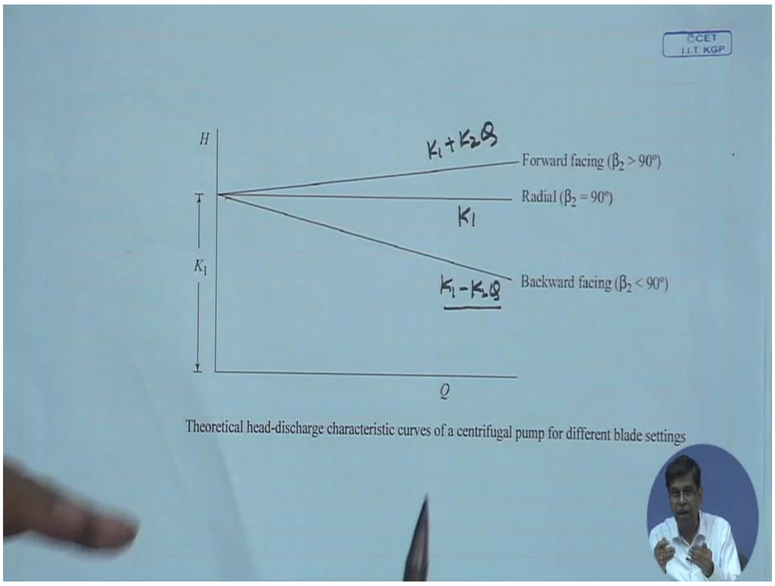

It delivers a constant head at all flow rates, whereas this is the case of backward facing blade, that is the blade whose curvature is in the opposite direction of the rotation which gives an inverse relationship, decreases with Q in a linear fashion. So if now this theoretical head discharge relationship under each case is being converted to the actual one by taking into consideration slip, step-by-step, slip, shock loss and the frictional loss as we did for a backward facing impeller blade.

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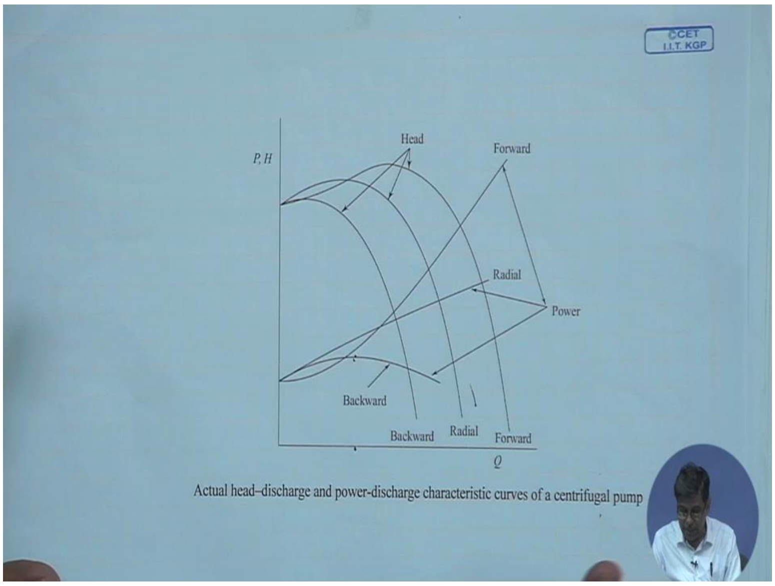

If we do it for both radial and forward facing impeller blade, the curve will look like this. Let me show you the curve, okay. The curve will look like this. This is finally the curve which will come. Now for a backward facing blade, already we got this curve earlier, this looks like this, then this will be the radial one and this shape is for forward facing blade. But we are not discussing much about these 2 since we use mostly this backward facing impeller blades in practice.

Reason for this I will tell afterwards. And accord, along with these curves, the power versus discharge also is shown like this. This is for a backward facing blade which is very important and as a maximum at a particular flow rate which is typically the design flow rate and for other 2, this radial and the forward facing blade, the power monotonically increases with Q. So it is very important to explain this power and the head discharge, power discharge and head discharge characteristics to know that why the backward facing impeller blade is useful and important in practice, that I will discuss afterwards, so today in this lecture it is up to this only. Thank you.