Principle of Hydraulic Machines and System Design
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Lecture – 03
Velocity triangles of pumps, effect of inlet swirl on velocity triangles

So, we will continue our discussion on Principle of Hydraulic Machines and System Design ah.

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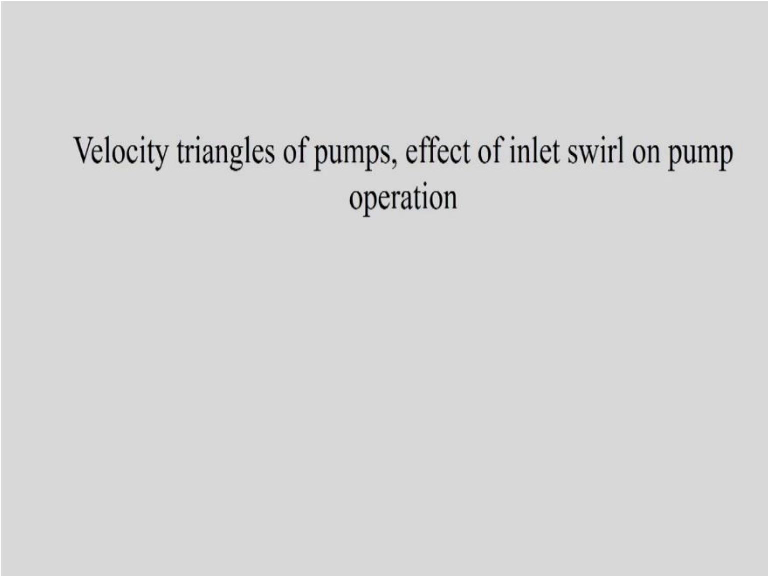

Today's topic is Velocity triangles of pumps effect of inlet swirl on the pump operation and pump performance. Today again we will draw rather we will revisit the velocity triangles for I mean radial flow pump as well as axial flow pump.

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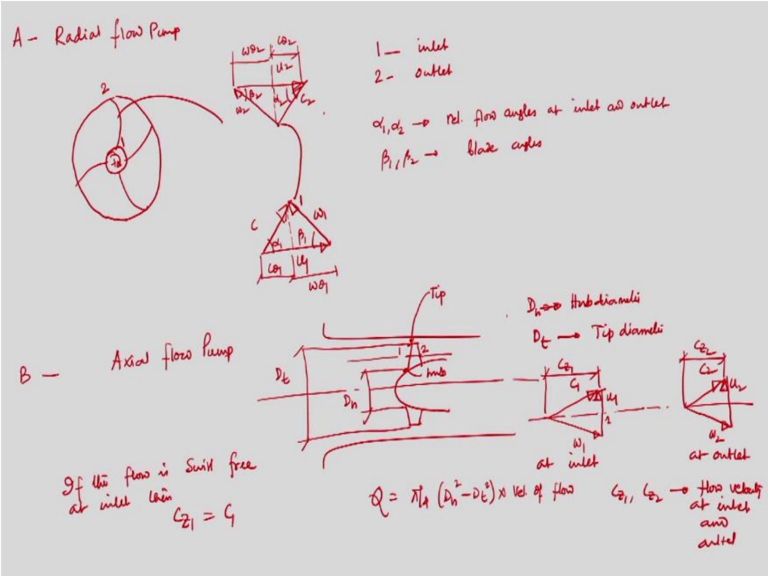

We will initially draw again an impeller of a radial flow pump; we will draw the impeller of a radial flow pump. So, if I take out this blade and if I draw you know enlarged view this is 1, this is 2; 1 is inlet and 2 is outlet. And if I draw the velocity triangles at the inlet, we have seen that there are 3 components; one is tangential velocity u 1, another is relative velocity w 1 and there will be a resultant velocity according to Lami’s theorem that is we call it absolute velocity c 1.

Now, similarly we will get the velocity triangles at the outlet; that means tangential velocity u 2, this is absolute relative velocity w 2 and resultant will be the actual velocity c 2. And we have seen that component of absolute velocity and relative velocities in the tangential directions are c theta $C_{\theta 1}$ and $W_{\theta 1}$ at the inlet. Similarly component of you know absolute velocity and relative velocity in the tangential direction at the outlet are $C_{\theta 2}$ and $W_{\theta 2}$.

So, this is $C_{\theta 1}$ and this is $W_{\theta 1}$; $C_{\theta 1}$ is sometimes known as the swirl component of velocity. Similarly, we are having $C_{\theta 2}$ and $W_{\theta 2}$ and this is β_2 is a blade angle at the outlet; this is α_1 this is flow angle this is α_2 and this is β_1. α_1 , α_2 is the relative flow angle; flow angles at inlet and outlet where β_1 and β_2 are the blade angles. Similarly, if I draw this is essentially for a radial flow pump; this is for radial flow pump.

Similarly, we can draw the velocity triangle for the axial flow machines or axial flow pump; if I draw. So, this is the axis of the impeller, this is impeller we are having blade this is point 2 this is point 1; 1 is the inlet and 2 is the outlet. Now this is known as suppose we are having similar blade at and this is known as hub diameter D_h, this is hub this point is known as hub of the impeller and this point is known as tip; tip and this point is known as hub.

So, Dh is the hub diameter similarly you are having tip diameter this is Dt tip diameter. So, Dh is the hub diameter and Dt is the tip diameter. So, we are having hub diameter and tip diameter; if I draw the velocity triangles at the inlet and outlet. So, I will draw separately; so velocity triangle at the inlet is we are having 3 different components again, we have relative velocity w1 this is point 1, this is the u1 and the resultant is the absolute velocity c1.

Similarly, this is at inlet and we will have velocity triangles at the outlet is w2, u2 and c2; so, that outlet. So, this is a velocity triangle at the inlet and this is at the outlet and we have hub diameter and tip diameter.

Q= $\frac{\pi}{4}(Dh^2 - Dt^2)$* velocity of flow

So, this is c z 1 we will call it Cz1 and this is Cz2 are flow velocity at the inlet and outlet. So, Cz1,Cz2 are the flow velocity at inlet and outlet. So, we can find out flow rate by knowing the Cz1 and Cz2; if the flow swirl free the special case is, if the flow is swirl free at the inlet if the flow is swirl free at inlet then Cz1 = C1 that is there will be no component of swirl at the inlet.

So, this is all about the velocity triangles at the inlet and outlet of the radial flow pump and the axial flow pump. Now we will see that what effect does it have if we make $C_{\theta 1}$; that is the swirl component of flow velocity at the inlet at outlet 0 and negative on the head development characteristics of the pump.

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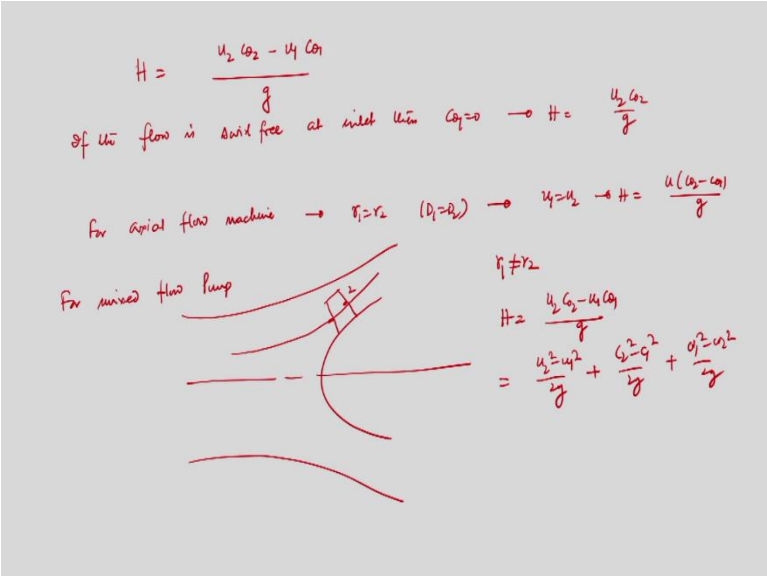

Next I will discuss that in the last lecture we have derived the head developed by the pump H from Euler equation of turbo machines that is $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$. So, the head developed the pump can be expressed in terms of the blade velocities at the inlet and outlet as well as the component of absolute velocity in the tangential direction at the inlet and outlet that is swirl component of velocity at the inlet and outlet.

We have discussed we have also discussed that if the flow is swirl free at the inlet. So, if the flow is swirl free at inlet, then $C_{\theta 1} = 0$ and the head developed by the machines will be simply $u2\ C_{\theta 2}/g$. Now we have also discussed that by making $C_{\theta 1} = 0$ we can have is $u2\ C_{\theta 2}/g$, but if $C_{\theta 1}$ becomes positive; that means, if the flow at the inlet is swirl I mean if there are swirl at the inlet of the flow then probably head developed by the pump will be lesser.

But somehow by making a negative component of swirl velocity we can have higher head development by the pump. And we have also discussed that we can make c theta 1 negative, we can discuss $C_{\theta 1}$ negative by you know making you know direction of the a pump impeller and the flow in the different direction ok. So, we have also discussed that the head developed at the pump that is coming from Euler equation for (Refer Time: 08:53) can be you know expressed in a bit different from that for the axial flow machine we have last lecture we have discussed that can be simply expressed in terms of the relative component of velocity.

So, now, I will discuss that for axial flow machines since $r_1 = r_2$ that is $D_1 = D_2$; then u_1 will be equal to u_2. It is simply head developed by the pump will be u*$(C_{\theta 2} - C_{\theta 1})$/g. But in case of a mixed flow pump, but for the mixed flow pump since if you draw the impeller again since the diameters r1, r2 are not equal. So, this is point 1 this is point 2 and we have like this since r1 not equal to r2. So, head developed by the pump can be written in terms of $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$.

And from the velocity triangles at the inlet outlet using cosine rule we have expressed this head development H in terms of 3 different component of flow velocities like

H = $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$

H = $\frac{{u_2}^2 - {u_1}^2}{2g} + \frac{{c_2}^2 - {c_1}^2}{2g} + \frac{{w_2}^2 - {w_1}^2}{2g}$

So, have seen that the head developed by the pump can be expressed in terms of 3 components of velocities that is, tangential velocity, axial velocity and the relative velocity that is velocity of flow related to the blade ok.

Now, I will try to express this quantity H for a radial flow pump in a bit different form that is in terms of outlet you know parameters at the outlet of the pump; that is the blade angle and also the velocity at the outlet of the pump. So, if I draw the; you know again impeller of a radial flow pump and this is a radial flow pump, and this is the hub and this is the blade.

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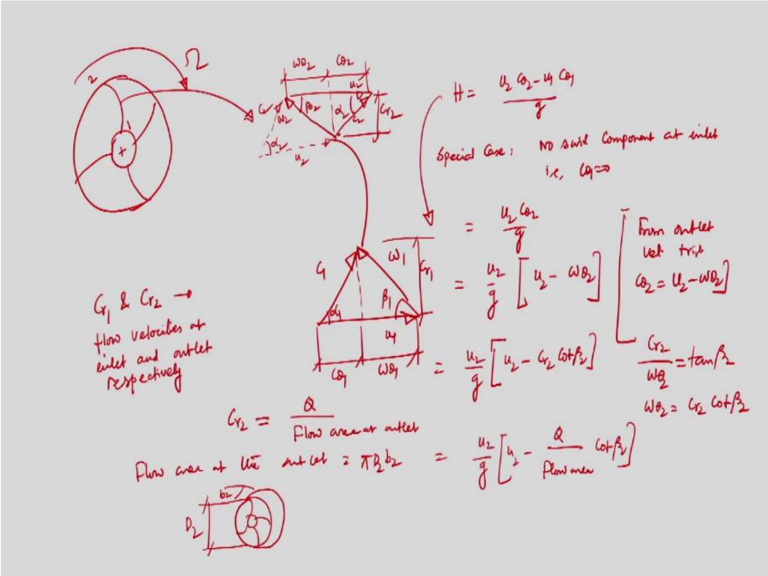

So, suppose pump is rotating in the clockwise direction at an angular velocity omega. And if I draw the blade 1 2; then if I take out the blade and if I take this blade out and if I draw the velocity triangles at the outlet, that this is 2. So, this is u2 this is c2; similarly we can have velocity triangles at the inlet. So, this is w1 this is resultant velocity is absolute velocity we call it absolutely c1 and this is u1, this angle is alpha 1 flow angle, this is the blade angle at the inlet this is $W_{\theta 1}$ and this is $C_{\theta 1}$.

Similarly, we have $W_{\theta 2}$ $C_{\theta 2}$; this is c2, this is alpha 2 this is blade velocity at outlet and this is w 2 and this is blade angle beta 2. So, we can have this kind of things this is u 2 and this is c 2; this is alpha 2. Now we know that the head developed by the pump is H = $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$; irrespective of what kind of pump it is it is whether it is radial flow pump or axial flow pump or mixed flow pump. Now for the radial flow pump and if a special case is that flow is radially inlet; I mean purely radial inlet there is no swirl component at the inlet. So, if special case if we take a special case that no swirl component at the inlet no swirl component at inlet.

That is $C_{\theta 1} = 0$, then this head developed by the pump will be can be written simply by u2 $C_{\theta 2}/g$. And where u2 $C_{\theta 2}$ are the blade velocity and component of absolute velocity in the tangential direction at the outlet (Refer Time: 13:34); as I said that I would like to now express this quantity H in terms of outlet parameters. I mean parameters at the outlet of the pump that is the blade velocity and different other component like $C_{\theta 2}$ and all those things.

H= $\frac{u2\, C_{\theta 2}}{g} = \frac{u2}{g}\ [u2 - W_{\theta 2}]$ $\{ \frac{C_{r2}}{W_{\theta 2}} = tan\beta_2 \}$

$$= \frac{u2}{g}\ [u2 - C_{r2}\ cot\beta_2]$$

So, now question is c r 2 that is velocity of flow at outlet is essentially Q by flow area. So, if I know the discharge from the pump because that is the specification how much you know quantities will be you know discharged by the pump. And if I know the diameter and width of the impeller at the outlet because flow area at the outlet flow area at the outlet.

Flow area at the outlet = $\pi\ D_2\ b_2$

So, if I draw you know what is b? So, this quantity is this quantity is b2 and this quantity is D2. So, this is the impeller diameter at the outlet and b2 is the width at the outlet. So, if I know D2 b2 this quantity I can obtain flow at the outlet. So; that means, now the head developed by the pump that is

$$H = \frac{u2}{g} \left[u2 - \frac{Q}{flow\ area}\ cot\beta_2 \right]$$

So, now, I have expressed the head developed by the radial flow pump head developed by radial flow pump where there is no swirl component of velocity at the inlet in terms of components; that is which can be calculated from the outlet I mean outlet parameters right velocity of absolute velocity sorry a tangential velocity at the outlet. Discharge if I know then I can calculate velocity of flow at the outlet because I know the diameter at the outlet and width of the outlet and the blade angle at outlet.

Now this is very important quantity because from this quantity; I can express that deafening upon the magnitude of beta 2 that is the blade angle at the outlet what could be the head development characteristics by radial flow pump; that we will discuss now. So, we should remember that the head developed by radial flow from where there is no swirl component velocity at the inlet can be expressed in terms of quantities; that is at the outlet that is velocity tangential velocity or blade velocity at the outlet and the blade angle and also the flow area.

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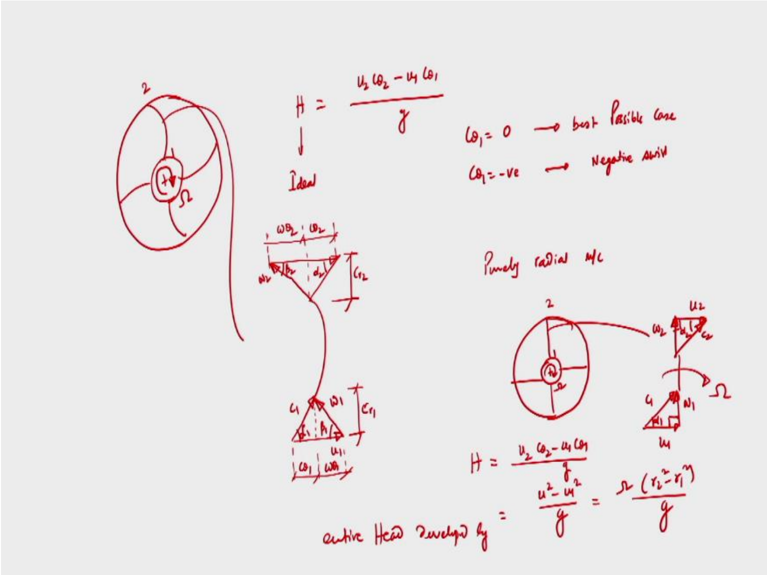

So, we have seen that for a flow of a radial flow pump if I draw the schematic of a radial flow impeller pump of a radial flow impeller. And then we have seen that the head developed by the pump which is predicted by Euler's equation is $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$.

So, this is the head predicted Euler's equation this of course, the ideal head because while you are calculating head by this equation; we do not take into account the frictional losses and also the recirculation losses in the suction side and the separation losses. And now this is I mean, if I draw the blade at a particular blade.

So, suppose this impeller is rotating at an angular velocity omega and if I take out a particular blade 1 2 and if I draw you know enlarged view and if I draw the velocity triangles at the inlet and the outlet. So, this component is relative velocity that is velocity related to the blade; this is the blade velocity u 1 and the resultant velocity is the absolute velocity. And we have seen that the component of you know relative velocity and absolute velocity in the tangential directions are $W_{\theta 1}$ and $C_{\theta 1}$ at the inlet.

Similarly, if we draw the velocity triangles at the outlet we will get like this is the blade velocity at the outlet, this is the relative velocity at the outlet, this is blade angle beta 2 and this is the flow angle alpha 2 and this is known as flow velocity Cr2 at the outlet and this is the flow velocity at the inlet Cr1. And again, if I take the component of absolute velocity and relative velocity in the tangential direction; these are $C_{\theta 2}$ and $W_{\theta 2}$ at the outlet. So, this is inlet flow angle alpha 1 and this is beta 1 inlet blade angle.

And the head developed by the Euler's equation is $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$ and we have see that this u1 $C_{\theta 1}$ is always trying to reduce the head being developed by the pump. So, we have discussed a few cases that $C_{\theta 1}$ may be 0 which is the best possible case this is a best possible case. And we may also have $C_{\theta 1}$ is equal to negative that is negative swirl that we have discussed that is highly possible when the incoming fluid that is fluid entering to the impeller will have a rotation which is in opposite direction of the rotation of the opposite to the rotation of the impeller. I mean incoming fluid will have a different direction of rotation with respect to the impeller rotation; so, this is negative swirl.

If we have a negative swirl; we have seen that the head developed by the pump will increase, but we have discussed that this negative swirl although we may increase head, but it is inviting I mean the negative components swirl maybe; we can develop we can have higher head, but at the same time we are inviting another problem of having cavitation in the pump and this is not a desirable one.

So, that is why $C_{\theta 1} = 0$ that is no swirl at the inlet that is purely radial inlet as the is the most is the best possible case. So, from these 2 we have seen that by how changing the swirl component of velocity at the inlet, we can change the head being developed by the pump or a radial flow machines not only radial flow machines by a pump of course, and it may be true for the axial flow machines also.

But now question is that head can be developed by in many ways I mean by changing the impeller diameter and so all those things. So now, if I take an example say, if we take a purely radial machines; purely radial machine and if I draw the impeller again and if I draw the velocity triangles at the inlet and outlet, so, this is again rotating at an angular velocity capital omega and blades at the straight. So, if I draw this way this is 1 and this is 2. So if I draw this blade and if I draw the velocity triangles; so this is purely radial. So, this is relative velocity this is the absolute velocity c1 and this is the blade velocity u1.

Similarly, I we can have; so this is α_1 and this is perpendicular similarly we have beta this is w 2 and this is c 2 and this is alpha this is u 2. So, this is c 2 and this is w 2 and this is α_2 and β_2 again is perpendicular; in that case ah, so pump is rotating at an angular velocity omega. In that case head developed by the pump will be equal to $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$. Here $C_{\theta 2}$ and $C_{\theta 1}$ itself are the $W_{\theta 1}$ and $W_{\theta 2}$ I mean u2 and u1. So, H= $\frac{u_2{}^2 - u_1{}^2}{g} = \frac{(r_2{}^2 - r_1{}^2)\omega}{g}$

This is purely radial machine; so, here head developed the entire head developed by the centrifugal force or entire head entire head developed by the centrifugal force.; now question is if we now increase r2 of course, we can have higher head developed this is for a purely radial machine. On the other hand, if we have seen that if we can if we change the swirl at the inlet by changing from positive to 0 and negative we can slowly increase head, but negative swirl is not an ideal case because it is always trying to increase head that is true, but it will leads to another problem this is not desirable at all that is it will create pump cavitation.

Now we will see how we can develop head by changing another parameter let us say of course, from these 2 cases we have seen by changing the diameter of the impeller at the outlet we can increase head that is that is always true. But it is not always possible to have that a big diameter because it is the difficult if you have big diameter then again we need to for at we need to run the pump we have we need to put higher power input and also phase is an another important problem. So, we will now see what the effect of you is known blade angle to the contribution of net head being developed by the pump.

So, now we will exercise one another aspect of these pumps.

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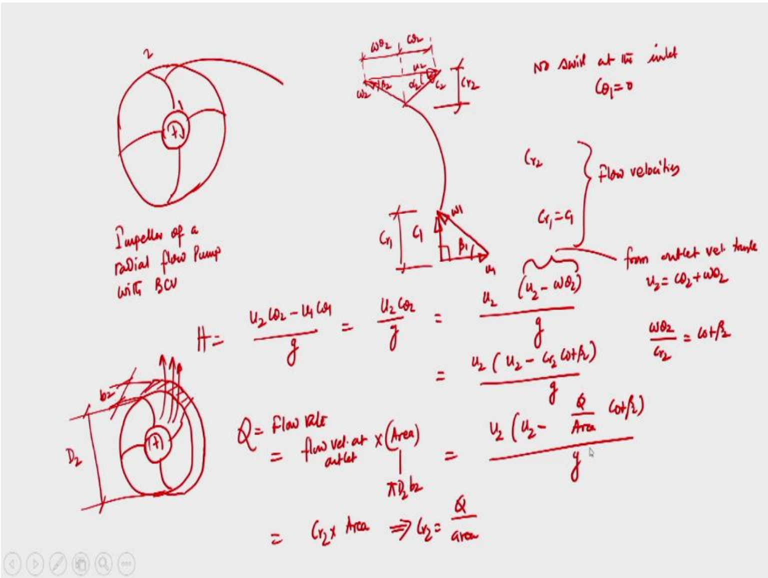

Let us say again we are considering a radial flow pump we are considering a radial flow impeller of a radial flow pump. So, this is impeller of a radial flow pump and this is rotating an at angular velocity ω and if I draw the pump you know blades. So, these are you know this is impeller placed with or equipped with the backward curved vane. So, this is impeller of a; radial flow pump radial flow pump with backward curved vane. So, we will write what is the backward curved vane. So, we will write what is the backward curved vane what is the forward curved vane fine. Now if I again take out a particular blade and if I draw the velocity triangles; let us say this is 1 and I am assuming there is no swirl at the inlet. So, we have seen that $C_{\theta 1} = 0$ is the best possible case, we are assuming no swirl at the inlet at the inlet that is $C_{\theta 1} = 0$.

If $C_{\theta 1} = 0$; then it flow angle will be 90 degree. So, this is relative velocity w1, this is inlet blade angle β_1, this is u1 and this is c1 that is the absolute velocity. So, this angle is perpendicular; now this is velocity triangles at the outlet. So, this is blade speed at the outlet, this is you know relative velocity at the outlet w2, this is β_2 and this angle is α_2 and this is c2 this angle is α_2 ; so, this angle is α_2. So, if I now again write the component of relative velocity and absolute velocity in the tangential direction; we obtained $W_{\theta 2}$ and $C_{\theta 2}$and this is u 2 and this is the flow velocity at the outlet Cr2 and here flow velocity at the inlet is equal to Cr1.

So, Cr1 = c1 and Cr2 and these are the flow velocities flow velocities. So, if I calculate; so now, for this particular case for an inlet swirl is 0 then

H= $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$ $\{C_{\theta 1} = 0\}$

H= $(u2\ C_{\theta 2})/g$

H= $\dfrac{u2\ (u2 - W_{\theta 2})}{g}$

So, now I can express $W_{\theta 2}$, but actually if I draw the impeller in a 3-dimensional view; so maybe impeller looks like this. So, this is the impeller; so maybe it is equipped with a few backward curved vanes and this is the width of the impeller. So, this is the width of the impeller; so this is b2 and this is D2. So, diameter of the impeller outlet is D2 and width is b2.

And so this is the flow area through which liquid is going out fluid is going out. So, I can express this head developed by the pump, this quantity in a bit different form. Now from the outlet triangle I can write

$$H = \frac{u2\ (u2 - C_{r2}\ cot\beta_2)}{g}$$

Q = flow velocity * Area

Q= Cr2* $\pi\ D2\ b2$

$$H = \frac{u2\ (u2 - \frac{Q}{\pi\ D2\ b2}\ cot\beta_2)}{g}$$

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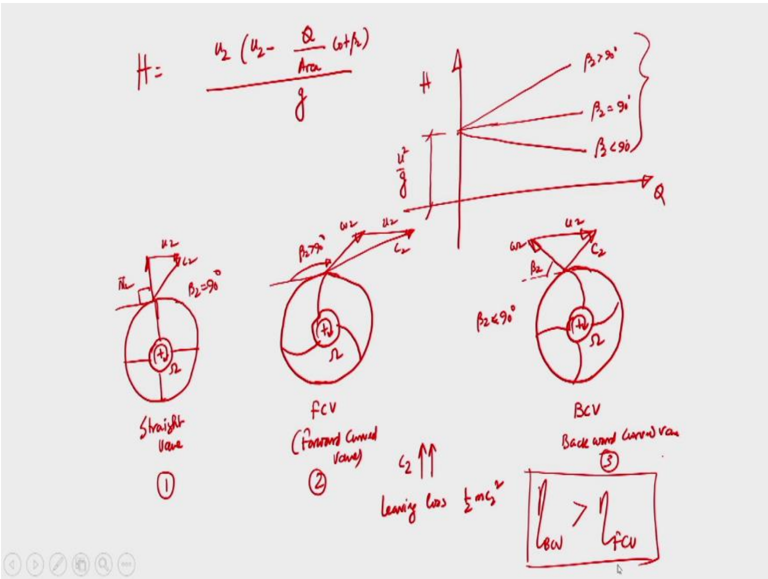

Now from this expression I can see that by changing the blade angle at the outlet; we also can vary the head being developed by the pumps. So, now, we will see a few different cases let us say if I consider H versus Q curve Q versus H that is H, then if I plot when $\beta_2 = 90$ degree then it is simplest $u_2^{2/}$g.

Now, if $\beta_2 > 90$ degree from this expression it is seen that the head developed by the pump will increase. So, this will be for $\beta_2 > 90$ degree that is true because head developed by the pump will increase if we take $\beta_2 > 90$. This is quite obvious because one can have a look at this you know one can have a look at the velocity triangle, this expression and can obtain and if $\beta_2 < 90$ then head developed will be decreased; so this is for $\beta_2 < 90$.

So, by changing the blade angle at the inlet keeping blade angle at the by changing the blade angle at the outlet keeping you know blade angle at the inlet fixed I can vary the head being developed by the pump. So, from this exercise it is you know suggestive of having you know blade angle which will be greater than 90 degree. If blade angle is greater than 90 degree then what will be?

Suppose if I draw now 3 different cases; so, whenever $\beta_2 = 90$. So, it is a statement that is what is I draw in the last lecture last slide that $\beta_2 = 90$ that is suppose this impeller is rotating at an angular velocity omega and this is as I said that all the angles are measured with the tangential direction. So, you know this is w2; so, 90 degree. So, this is maybe c2 and this is u2.

So, this is straight vane straight vane $\beta_2 = 90$ if I consider $\beta_2 > 90$ So, it means like this if we consider an impeller of a radial flow pump and this is; so, beta 2 greater than 90 degree means blade should be like this. So, since blades angles are measured with a tangential direction. So, this is β_2 ; so this is w2 and this will be the absolute velocity and this is the relative velocity. So, this will be the absolute velocity and this is the blade velocity u2 and this is c2 and this is $\beta_2 > 90$, so here $\beta_2 = 90$.

Now, this is a case where we can see that head developed by the pump will be higher; note that here the vanes are not exactly the backward curved vane we will discuss what is. So, this is a called a forward curved vane, so this is called forward curved vanes vane because in all the cases rotation of the pump in the clockwise direction that is pump is rotating in the clockwise direction at a certain speed at an angular velocity omega.

Now if $\beta_2 < 90$ then head developed the pump will reduce; so, if I draw the again one another case. Let us say this is the impeller of a radial flow pump and the pump is rotating at the same angular speed, at the same direction and this is now the case where blades are having you know backward curved vane. So, here this is the absolute velocity this is the blade velocity and this is c2 this is w2 and this angle is β_2.

So, $\beta_2 < 90$ and this is u2; so, this is known as this is a case where it is called backward curved vane. So, this is backward curved vane backward curved vane. So, what we can see from this expression that that the backward curved vanes if we have a $\beta_2 < 90$ that is in all the cases pump is rotating in a clockwise direction at an angular velocity omega, but for the backward curved vane head developed by the pump will be less as compared to 2 other cases, where $\beta_2 = 90$ and $\beta_2 > 90$.

So, maybe from this analysis it is quite you know cleared that someone should use whenever pump is designed a designer should use backward you know forward curved vane only to have a higher head rise or head developed by the pump will be higher. But this is not the case because efficiency of the backward curved vane is always higher than the forward curved vane. So, if we if we have a close look if we have a closer look at these 3 cases I mean case 1, case 2, case 3; what we can see from these that if it is case 1 and if it is case 2 and if it is case 3; then we can see that in case 2 if c 2 is higher.

So, if this case c 2 is higher much higher than the as compared to 2 different other cases. And that is why whenever we are having forward curved vane I mean maybe $\beta_2 > 90$ head rise will be head rise will be high, but the leaving loss which is known as leaving loss that is ½ m c_2^2 this is high. So, from the velocity triangle itself we can see we can clearly see that maybe if we have a forward curved vane when $\beta_2 > 90$; we may have higher head generation that is quite true from the expression whatever I have written above that head developed the pump will increase.

But at the same time the leaving loss from the pump will be high that is ½ m c_2^2 will be high and there is a very competition maybe head rise will be high at the same time leaving loss will be high. But relative to the rise in head the leaving loss the relative loss of you know leaving loss and I mean relative rise of head I mean head will increase at the same time leaving loss will be high. So, the, but the relatively leaving loss will be higher than the rise of the head that is why efficiency of the backward curved vane; efficiency of the backward curved vane is always higher than the efficiency of the forward curved vane this is true because. So, from this discussion we can conclude that maybe if we use a forward curved vane pumps or radial flow pumps or impeller equipped with forward curved vanes; where $\beta_2 > 90$ head rise will be high.

So, forward curved vanes will not only give a higher head rise in that case the leaving loss will be high, but the relative increment of leaving loss will be higher than the relative rise of head as compared to the backward curved vanes and straight vanes; that is why efficiency of the backward curved vane is always preferred than forward curved vanes. And in fact, for $\beta_2 = 90$ also the leaving loss will be higher than the backward curved vane. So, that is why efficiency of the backward curved vanes is always higher than the forward curved vanes, but still there are some cases or there are some places where we need to have forward curved vanes.

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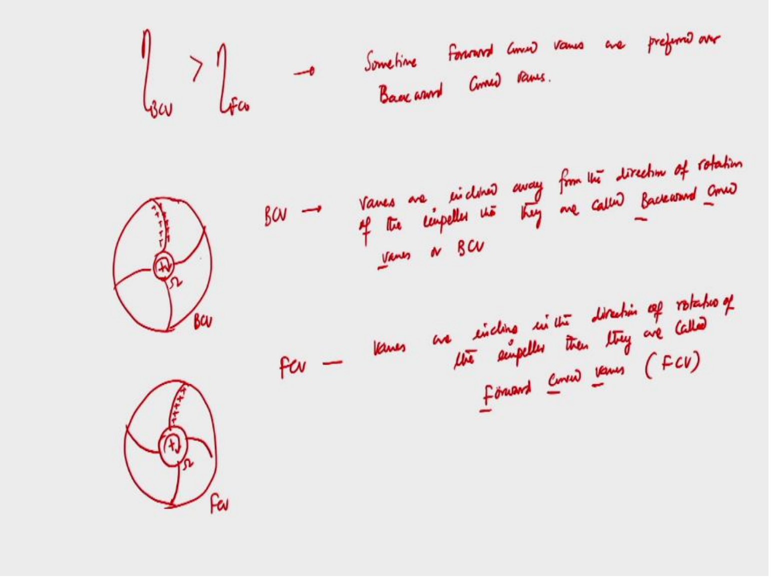

So, I will now discuss another important aspect that from you know these discussions, we have understood that efficiency of the backward curved vane is always greater than the efficiency of the forward curved vane that is true. But sometimes; sometimes forward curved vanes sometimes forward curved vanes are preferred over backward curved vane.

So, we have understood that the efficiency of the forward curved vane will be less than the backward curved vane. But still there are a few cases or there are few you know situation where you need to have forward curved vane that is we have we will be having impeller; that impeller should be equipped with a few forward curved vanes otherwise there will be a problem, why? So, now, one reason is that of course, forward curved vane efficiency will be higher, but as I said you that the relative increment of you know head raise that will lead to that will leads to higher efficiency that is true.

But the relative increment of efficiency because of the higher head rise will be you know less as compared to the leaving loss that is why forward curved vanes are not basically you know higher efficient. But there are few cases if I draw an again, one 2 impellers one is equipped with the backward curved vanes and another is equipped with forward curved vanes suppose this is impeller and impeller is rotating with an angular velocity omega this is backward curved vane; so, this is backward curved vane.

So, this is backward curved vane why backward curved vanes? Because this; these are known as backward curved vane because vanes are vanes or blades are inclined away from the direction of you know ah; away from the direction of the rotation of the impeller, away from the direction of rotation of the impeller. So, when vanes are inclined away from the direction of the rotation of the impeller then we call it backward curved vane that is like this case.

And again, if you draw another impeller which will be having a forward curved vane, so, this is an impeller. If we draw the schematic and suppose, this is rotating the clockwise direction with the same angular velocity and blades are forward curved ; blades are forward curved. So, here this is known as backward curved vane and this is forward curved vane. So, what are forward curved vanes? When vanes or blades are inclined in the direction of in the direction of rotation of the impeller, then they are called; then they are called forward curved; Forward Curved Vane or FCV; then they are called Backward Curved Vanes or BCV; BCV.

So, now forward curved and backward curved vanes are clear because in one case it is inclined towards the direction of rotation of the impeller forward curved vanes, but in other case backward curved vanes this is not inclined in the direction in the direction of the rotation of the impeller. But now question is all the efficiency of the backward curved vanes are preferred over forward curved vanes, but sometimes we all the efficiency of the backward curved vanes are higher than the forward curved vanes, but still sometimes we prefer forward curved vanes in places.

In particular there are industries like say jute industries and paper industries where there is huge amount of dust. So, there is a probability of deposition of dust particle on the top surface of the blade on the blade surface. So, in the in that in that case in that case, but if the dust particles are depositing in like this. So, in backward curved vanes these particles are remain try to stick over there and it will try to have or it will try to start corrosion it will try to erode some blade material.

On the other hand, because of the geometrical or constructional shape of the forward curved vane even you know dust particles are depositing over here like this. Then because of the direction of the rotation of the forward curved vanes the dust particle will always try to remove or away from they no longer will try to remain attached to the blade itself. So, the probability of you know having a corrosion or the blade erosion is not there.

And that is why that there are situations, there are instances, there are places particularly jute industries and paper industries where there is a probability of having deposition of dust particle over the blade surface and if dust particles try to you know deposits dust particle deposit over the blade surface. In case of a backward curved vane, they will try to remain stick over there and they will try to start corrosion and erosion of the blade material.

But because of the constructional geometrical shape of the blade itself; even though the dust particle are depositing on a blade surface for a forward curved vanes, but their shape itself will try to allow the dust particle to be removed from the blade and the probability of you know corrosion and erosion will be no longer there.

Now, from this discussion it is clear that why forward curved vanes are preferred over backward curved vanes in some cases; although the efficiency of the backward curved vanes is higher than the forward curved vane. So, now we work out one example that the problem is suppose there is a centrifugal pump and I will one problem.

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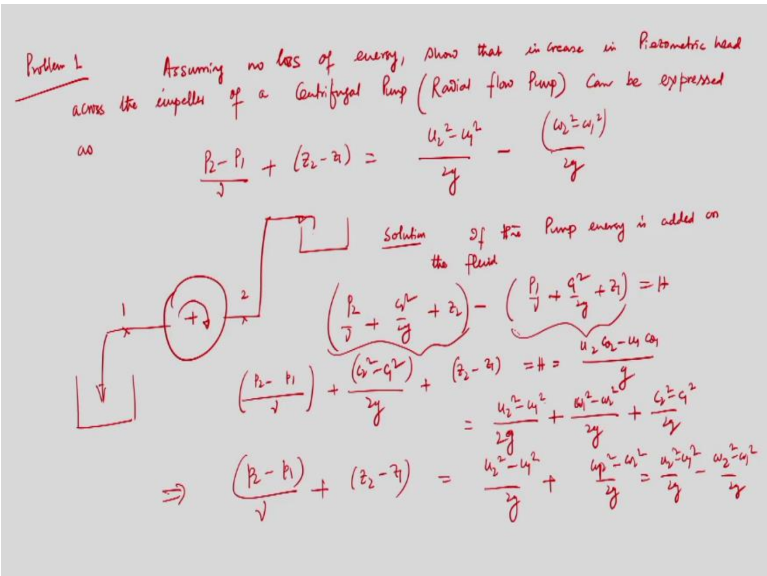

This is not a numerical problem, but it is interesting problem that if we consider a radial flow pump. Centrifugal pump sometimes we call it I mean for a we sometimes call radial flow pump as centrifugal pump because of the presence of centrifugal force.

As I said in the first lecture that centrifugal pump is (Refer Time: 44:38) because if we need to call a radial flow form as centrifugal pump; I mean to be a centrifugal pump because of the presence of centrifugal force ah; then in a mixed flow pump to some extent centrifugal force is there then again we then we have to call the mixed flow pump as a centrifugal pump.

So, I always prefer to call it a radial flow pump. So, one problem is say assuming no loss I am writing the problem assuming no loss of energy so that increase in piezometric head. So, that increase in piezometric head across the impeller of a centrifugal pump, of a centrifugal pump. I am writing, but again I am telling I prefer to call it radial flow pump. So, I am writing a radial flow pump radial flow pump. So, assuming no loss of energy so that increase in piezometric head across the impeller of a centrifugal pump rather radial flow pump I will call it; can be expressed as

$$\frac{P_2 - P_1}{\gamma} + Z_2 - Z_1 = \frac{{u_2}^2 - {u_1}^2}{2g} - \frac{{w_2}^2 - {w_1}^2}{2g}$$

So, I am discussing this problem because if we consider there are no loss of energy can you show that the piezometric head rise or piezometric head across the impeller of a radial flow pump can be written in terms of the blade velocities and the absolute velocities. We cannot; we can express only in terms of these 2 components of velocities. So, we need to solve this problem I mean we need to because is a kind of things; we have to check it because how can we express this without as you know writing in terms of the absolute velocity.

So, suppose if you consider radial flow pump like this and pump is rotating I will schematically I will write. So, pump is rotating like this and pump is discharging water in another place two. So, this is let us say 1 and this is 2; so, it is discharging water in some other place and drawing water from a some. So, this pump is working between these 2 ah; so, this is a radial flow pump which is drawing water from one some and discharging water to other place you know and if it is there is a an if there are no losses I mean there is no loss of energy; how we can show that the piezometric head across the pump impeller only the impeller of a radial flow pump can be written in terms of absolute blade velocity and the radial flow velocity.

So, if the pump now question is if we solve this problem. So, whenever pump is discharging rather pump is you know discharging water in a place; rather pump is drawing water from one place and discharging on other place definitely pump is adding some energy to the working fluid. So, if the pump energy is added on the fluid definitely now question is if the pump energy added on the fluid; that means, in a pump; whenever it is drawing water and discharging in some that water in other place. So, it is definitely pump is adding some energy on the working fluid because pump we need to run the pump either electric motor or diesel engine.

So, if the pump energy is added on the working fluid then what suppose an whenever it is discharging water at some place; then the function of pump is to develop a head. So, can I write that

$$\left(\frac{P_2}{\gamma} + \frac{{c_2}^2}{2g} + Z_2\right) - \left(\frac{P_1}{\gamma} + \frac{{c_1}^2}{2g} + Z_1\right) = H$$

Because suppose whenever pump is drawing water from (Refer Time: 49:33) and discharging at 2 then it has to overcome so, many other losses frictional losses in the pipeline, frictional losses in the vanes and also it has to overcome some static head.

So, considering all those we need to put some energy on the fluid and that is added by the pump itself. So, if pump energy is added on the working fluid; then I can write this equation because whatever will be the pressure at the point 2 and pressure the point 2, the difference that is being calculated that is added by the pump. So, and that is the head developed.

So, what you can said because a point 2 has to have more energy; point 2 has to have more energy otherwise there is no how we can lift water how we can discharge water in some other place. So,

$$\frac{P_2 - P_1}{\gamma} + \frac{{c_2}^2 - {c_1}^2}{2g} + Z_2 - Z_1 = H = \frac{u_2 C_{\theta 2} - u_1 C_{\theta 1}}{g}$$

$$= \frac{{c_2}^2 - {c_1}^2}{2g} + \frac{{u_2}^2 - {u_1}^2}{2g} + \frac{{w_2}^2 - {w_1}^2}{2g}$$

Now we can see that the head developed by pump to precise by a radial flow pump can be written in terms of 3 component of velocities. And we can have what will be the head developed the pump the pump is installed in a place, where we need to develop a head which is no determined by these 2 quantities.

$$\frac{P_2 - P_1}{\gamma} + Z_2 - Z_1 = \frac{{u_2}^2 - {u_1}^2}{2g} - \frac{{w_2}^2 - {w_1}^2}{2g}$$

So, I can express these quantities whatever that is there I mean piezometric head across the pump impeller of a radial flow pump when there is no loss of energy can be expressed in terms of this. Because whenever we have written equation when you did not consider an equation that is on any loss due to fictional, I mean any fiction head loss we have considered.

So, without assuming any loss of energy I can express the piezometric head develop by the pump ; in terms of piezometric head across the pump in parallel of a radial flow pump in terms of the blade velocity and the relative velocity at the inlet and outlet of the pump impeller ok.

. So, next we will see that that is important that is the; you know effect of swirl at the inlet on the pump of operation that is very important ah. So, we will again consider a radial flow pump; so, we are considering that a radial flow pump.

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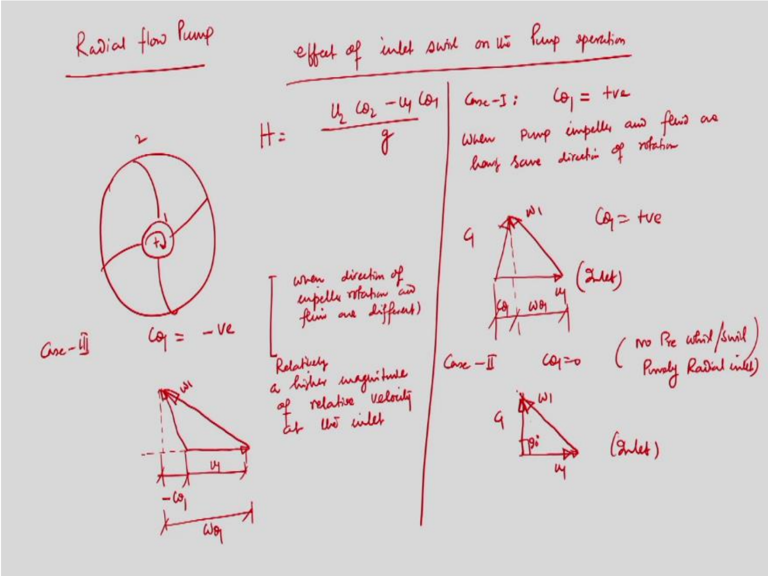

We are considering a backward curve vane because we have seen that the backward curve vanes are you know more efficient than the forward curve vanes. And if this is point 1 and point 2 and if I draw the velocity triangles at the inlet we will discuss; now the effect of inlet swirl on the pump operation; effect of inlet swirl pump operation.

So, we know that the head developed by the pump can be you know head developed by the pump can be expressed from Euler equation of (Refer Time: 55:42) machines; that u is equal H = $(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$ and we have express this quantity in different other form where in terms of only the pure only of the outlet quantities that is outlet velocities that is absolute velocities relative velocities and the tangential velocities of at the outlet of the pump.

So now whenever we are what will be effect of inlet swirl on the pump operation. So, I will discuss different cases; so, if I discuss first case let say case 1 that is $C_{\theta 1}$ is equal to positive we have discussed that $C_{\theta 1}$ becomes positive when pump impeller and the fluid are having are rotating the same directions are having same direction of rotation. Then $C_{\theta 1}$ will be positive and if $C_{\theta 1}$ is positive I will; I will discuss that if $C_{\theta 1}$ is positive then the inlet velocity triangles we will discuss only at the inlet velocity triangles.

So, what will be the velocity triangles? So, what will be the velocity triangles; so, if I draw the velocity triangle at the inlet $C_{\theta 1}$ is positive. So, this is u1 this is; this is c1, this is u1, this is w1 and this is $W_{\theta 1}$, this is $C_{\theta 1}$. So, here $C_{\theta 1}$ is positive; $C_{\theta 1}$ us positive quantity. So, the net head develop by the pump will be always lesser I mean because this quantity will be subtracted from this quantity for a given value of u1 and u2.

So, head develop by the pump will be always lesser ; now if I discuss case 2 that $C_{\theta 1} = 0$ that is no pre whirl or swirl that is swirl free flow then this is called purely radial inlet, purely radial inlet; that is u1 will be same because u1 is the blade velocity at the inlet, it depends upon the diameter at the inlet and the speed.

Since the speed and diameter (Refer Time: 58:47) same. So, the u1 will be same; so, the u1 will remain same and if u1 is remaining same to make $C_{\theta 1} = 0$ suppose u1 is remaining same and to make $C_{\theta 1} = 0$. So, this will be $C_{\theta 1}$ it will be c1 this will be w1 and this will be u1. So, mind it here $C_{\theta 1} = 0$ that is this is 90 degree purely radial inlet purely radial inlet this is 90 degree, but u1 will remain same for all the cases.

Because u1 depends essentially on the value of D1 and ω the pump impeller speed angular velocity of the impeller. So, to keep u1 fixed its we need to make c1 perpendicular that is no swirl component we must have w1 here. So, here head development will be relatively better than the previous case ah, but since w1 becomes higher; it will create another problem that will discuss may be in the next case ah. For the no swirl free I mean no pre whirl no swirl at the swirl free flow this is fine.

But again, if we discuss case 3; if we discuss case 3 that is $C_{\theta 1}$ is negative. Because a negative $C_{\theta 1}$ it is very important it is very clear from this expression that if we can make $C_{\theta 1}$ negative, we can have higher head development characteristic. Because for the same pump for a rotating at a same velocity we still can have higher head development.

If we can make $C_{\theta 1}$ negative that is when this is the case, we can obtain when direction of the impeller direction of the impeller rotation and fluid are different. So, when direction of the pump impeller on the fluid are defined then we can have negative $C_{\theta 1}$; if we can have negative $C_{\theta 1}$ then from this expression we can see that we can have higher head development ah; despite the fact that the pump is running at the same speed at impeller we are not changing anything.

But this negative $C_{\theta 1}$ will create another problem that we will discuss how can have how if we draw the velocity triangle at the inlet that that would be clear. So, if I draw the velocity triangle at the inlet suppose negative $C_{\theta 1}$. So, if I draw the inlet will all these are inlet velocity triangle. So, this is inlet this is at the inlet; so, if I draw the velocity triangles at the inlet.

Suppose and I said that u1 will remain same because u1 depends on the omega as well as the diameter of the inlet see both are remaining same; so, u1 will remain same. Now to make $C_{\theta 1}$ negative that is ; so, u1 will remain same, this is c1 and to make $C_{\theta 1}$ negative. So, so this is the negative component of c theta; so, this is negative component of $C_{\theta 1}$ So, this is the negative component of c theta 1 and this is total is $W_{\theta 1}$; this is $W_{\theta 1}$ and this is u 1.

So, here to keep u1 fixed that is u1 will remain same; if we make $C_{\theta 1}$ negative that is when direction of the pump impeller and fluid are different, then we will have a relatively higher magnitude of relative velocity at the inlet. So, in this case; we will have higher magnitude of relative velocity at the inlet; so we will have relatively.

So, if we make $C_{\theta 1}$ negative of course, we can see from the expression that our head development will be higher for a given other conditions I mean remaining same because ah, but we will have. So, may be by making $C_{\theta 1}$ negative we can have higher head development, but at the same time you are inviting another problem what is the problem?

Because in that in this case we will have a relatively higher magnitude of relative velocity at inlet; so, if relative velocity at the inlet increases pressure may pressure will fall. And if the velocity increases pressure will fall and pressure falls below the vapor pressure at the corresponding temperature, then local carbonation might start.

So; that means, and it will create an undeserved phenomenon that is non cavitation. So, may be; so, what I am telling? By creating or by making $C_{\theta 1}$ negative we may have a relatively higher head development by the same form as compared to 2 different other cases; that is $C_{\theta 1}$ positive and 0, but we are inviting another problem that is lead it will give us a relatively higher magnitude of relative velocity.

And that will that will create another problem because for a higher relative velocity pressure might fall and the pressure falls below the you know vapor pressure at that temperature; then local boiling will take place and it will lead to a you know undesirable phenomena, which is known as cavitation and this is not desirable at all as far as the pump operation is concerned ok.

So, we will stop here today, and we will discuss we will continue to next class.

Thank you.