Principle of Hydraulic Machines and System Design
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Lecture – 05
H-Q curve, System Resistance Curve

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

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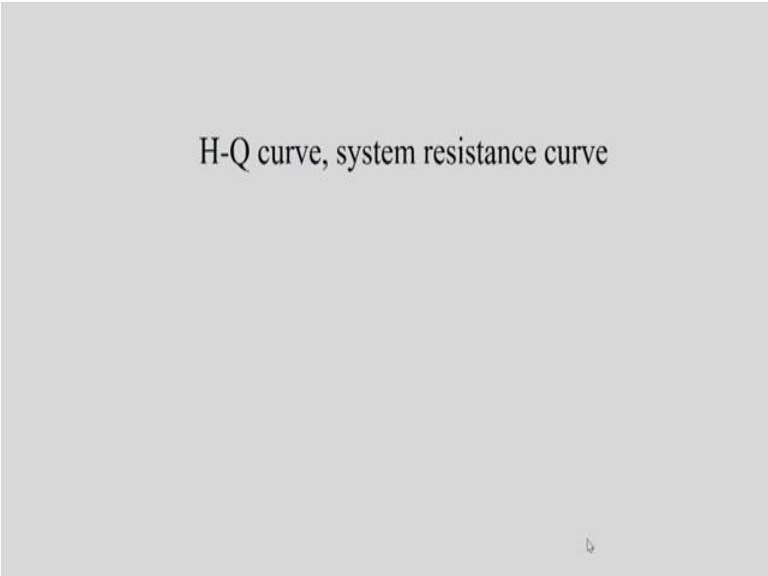

Today I will discuss the H-Q curve and system resistance curve, I mean H-Q curve for the pump and of course, whenever pump is installed in a particular station, then what will be the system resistance?

Because it is important to know the system resistance because the total head, I mean total discharge all these things are you know based on the total head the pump will develop a particular head. But depending upon the system resistance what would be what would what would be the you know discharge that depends; so, we will discuss a few cases today.

Before I go to discuss about the H-Q curve and the system resistance curve are you recapitulate a few things whatever I have discussed in the last lecture. In the last lecture I have discuss about numerical problem when the problem was on the radial flow machine.

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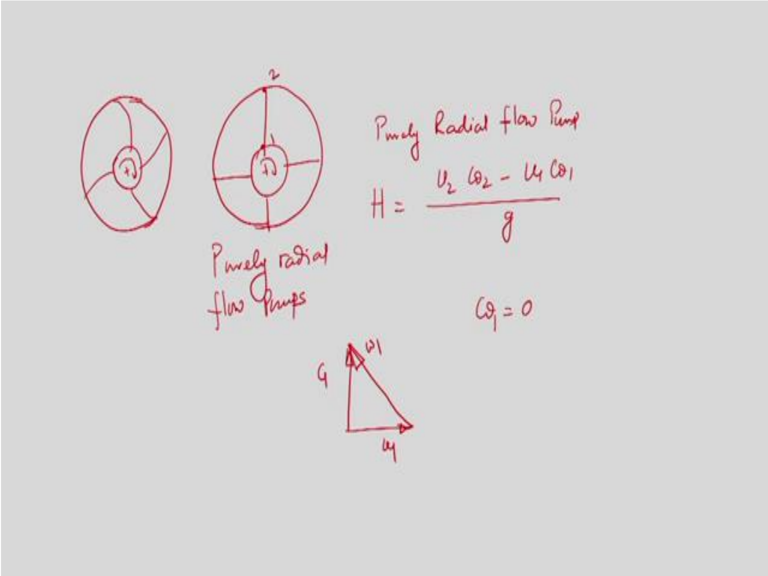

If I draw the radial flow machine again I mean if I draw the impered of a radial flow pump or radial flow machine. So, this is the pump impeller it is rotating in a clockwise direction and if we have back word spacing vane backward curve vane. And if it is a backward curve vane the impeller look like this; we have also discussed that if it is strait vane that is the if I draw an impeller again and if it is a straight vane, then blades are straight.

And we have discussed problem on radial flow pump where it is you know straight vanes. So, purely radial flow machine; purely radial flow pump I can say in that case if I write the head developer the pump using Euler equation for pumps that is H =$(u2\ C_{\theta 2} - u1\ C_{\theta 1})/g$; this is the point 1 and this is the point 2 that is inlet and outlet respectively. So, again if I draw the inlet and outlet velocity triangles particularly for this case that is purely radial flow machines; purely radial flow pumps and then, we can if I draw the velocity triangles at the inlet then this is the velocity triangles; this is u 1 this is blade speed u1 and this is relative velocity w1. So, here $C_{\theta 1} = 0$; similarly that is no solve at the inlet. And if I draw the velocity triangles at the outlet, then again it will have $C_{\theta 2} = 0$ and we have seen that in that case head is develop purely by curly force.

Ah Now I will discuss that even if there is a you know machines it may not be radial flow, but still we can have that we can ignore the soil component at the inlet and outlet that is what I now discuss that without any swirl at the inlet and even if there is a swirl then how can I write this component.

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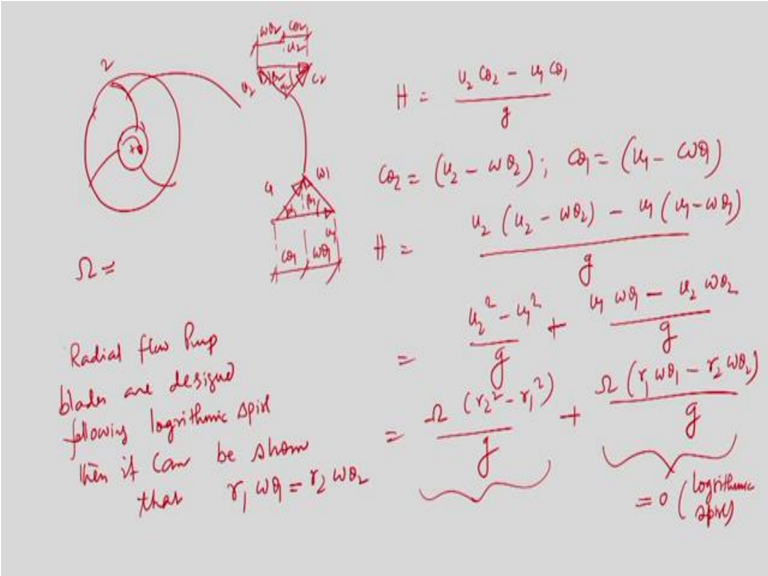

So, I am drawing an impeller; again, I am drawing are impeller of a radial flow pump and it is let us say backward curved are backward spacing or backward vane curve. So, this is rotating in clockwise direction if I take out a particular blade from this and if I draw the inlet and outlet velocity triangles. And it is no swirl at the inlet even if there is a swirl at the inlet and if I draw the velocity triangles velocity triangles is like this; this is u 1 this is w 1 and this is c 1 the absolute velocity. Now, again head develop by the pump can be written in terms of tangential velocity and the swirl component of velocity u1 $C_{\theta 1}$/ g.

Now from inlet and outlet velocity triangles I can write

$C_{\theta 2} = u2 - W_{\theta 2}$

$C_{\theta 1} = u1 - W_{\theta 1}$

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

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

H = $\frac{{u_2}^2 - {u_1}^2}{g} + \frac{u1\ W_{\theta 1} - u2\ W_{\theta 2}}{g}$

H = $\frac{\omega^2({r_2}^2 - {r_1}^2)}{g} + \frac{\omega(r1\ W_{\theta 1} - r2\ W_{\theta 2})}{g}$

Now question if blades whenever blades are designed and for centrifugal pump for a radial flow pump rather I can write radial flow pump; if I for radial flow pump if blades are you know blades are designed following logarithmic spiral, then it can be shown, then it can be shown that $r1\ W_{\theta 1} = r2\ W_{\theta 2}$.

So; that means, when particularly radial flow pumps normally blades are designed following a logarithmic spiral and if it is design following logarithmic spiral, then it can be shown that $r1\ W_{\theta 1} = r2\ W_{\theta 2}$ in that case the second term vanishes. So, the second term will be equal to 0 for the blades design using following logarithmic spiral.

So, essentially the head develop by the pump depends upon the first component that is essentially the co release force. So, this is very important; so, here develop only by the co release force component that is $\frac{\omega({r_2}^2 - {r_1}^2)}{g}$. So, that is what I would like to emphasize here that even if there are swirl at the inlet and outlet, but still for radial flow pump since the blades; if blades are designed following logarithmic spiral, then it can be shown that the head develop by the pump is only because of the coriolis force.

And that is what the problem that we have discussed in the last class that even for you know radial flow machines you know purely radial flow machines. There is no axial component of velocity and you know this is no mixed flow pump or axial flow pump. If the blades are designed following a logarithmic spiral, then I can have we can have the head developed by the pump is essentially the omega into $\frac{({r_2}^2)}{g}$ that is the coriolis component of forces ok.

So, with this now I will move to see what are the; you know different ah. In fact, also I have discussed last class I mean in last lecture the one efficiency known as hydraulic efficiency.

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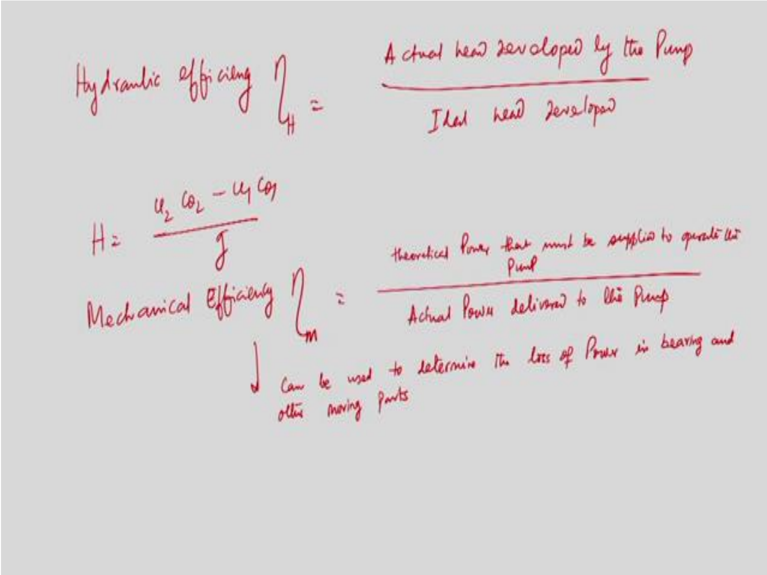

So, hydraulic efficiency of the pump we have seen that hydraulic efficiency of the pump; eta hydraulic can be defined as the ratio of actual head develop by the pump to the theoretical head. Because expression H =$(u2\ C_{\theta2} - u1\ C_{\theta1})/g$; this is the you know ideal head. Because whenever we are obtaining head using this expression we did not consider any kind of losses that is there in you know actual case.

So, whenever fluid is flowing through a impeller passes through an impeller of course, we cannot ignore the frictional losses. So, if we consider all those losses up to an there are the losses like you know pump you know entry and outlet stock loss separation losses. So, if that if we consider all those losses probably the head developed by the pumps not with you know ideal head. So, we will have actual head.

So, this is defined as by the ratio of actual head developed by the pump to the ideal had ideal head developed. So, the expression which we obtain from Euler equation for pumps is giving the idea about the ideal head being developed by the pump, but a actual head developed the pump will be lesser than that. So, this is the hydraulic efficiency that we have discussed.

$$\eta_H = \frac{Actual\ head\ developed\ by\ pump}{Ideal\ head\ developed}$$

Now, we will discuss three efficiencies rather another a few efficiencies one is known as mechanical efficiency this is very important. Mechanical efficiency of a pump mechanical efficiency η_m is equal to this is again an important because you know whenever we are supplying power to the pump ; whenever we are supplying power either by using electric motor because as I said many times that we need to run the impeller we need to rotate impeller.

So, whether we will be using a diesel engine or whether we will be using electric motor that depends upon you know situation to situation. Because in many cases we if we do not have electricity, we need to go for diesel engine to operate the pump. But nowadays it is not the case; so, whatever in the case it is I mean whether we are operating pump using electric motor or diesel engine we need to run impeller. So, of course, we need to provide some energy.

So, now, question is whatever power we are supplying that we need to supply powers whatever power we are supplying to operate the pump, but that amount of power is not supply at the shaft of the power. Because there are so, many other losses I mean. So, there is efficiency which is not the mechanical efficiency which is which is defined again as the ratio of; I am writing it is defined again as a ratio of theoretical head theoretical, you know power that must be supplied to operate the pump to operate the pump to the actual power to the actual power actual power delivered to the pump.

$$\eta_m = \frac{Theoritical\ power\ must\ be\ supplied\ to\ operate\ the\ pump}{Actual\ power\ delivered\ to\ the\ pump}$$

So, as I said that whenever we are connecting pump with electric motor or diesel engine; probably we know that whatever power is coming from the motor electric motor at the outlet of the I mean what is the power output from the electric motor that power output may not be the input power to the pump ; actual in theoretically that should be the input power to the pump that is power available at the shaft of the impeller, but in actual case it is not the case.

So, power output from the motor is not the power input to the shaft of the pump there is ah; I mean loss and to take that loss into account we define one efficiency which is known as mechanical efficiency. That means, whatever power we are supplying whenever we are connecting the shaft or the impeller axis, I mean shaft of the pump with the shaft with the motor.

Then probably output power from the motor is not actually transferred to this power at the pump set; there are I mean there is there is a loss and to take that affect into account we will get little bit lesser amount of power input to the pump. And that is why we are defining one efficiency which is known as the mechanical efficiency. So, the loss is only because bearing loss because whenever we are shaft is you know there is bearing because shaft is placed inside the bearing.

So, there will be bearing losses I mean frictional losses. So, we need to work on that losses and for that we need to you know somewhat amount power will be consumed to overcome that loss and eventually we will get little bit lesser amount of power at the shaft of the pump. So, we are defining mechanical efficiency.

So, from here I can say this mechanical efficiency can be used. So, this is very important this mechanical efficiency can be used to determine to determine the laws of power in bearing and other moving parts. So, of course, bearing is there because shaft is you know in housed in a bearing inside a bearing. So, I mean whenever we are having bearing; we cannot trivial ignore the losses I mean loss and also there are some other losses might be there are so, many other if there are other moving parts.

So, as I said that power which is coming from electric motors somehow a portion of that power will be consumed to overcome the losses in bearing and other moving parts. So, we will get whatever I mean little bit lesser amount of power input to the shaft and that is why I am differing mechanical efficiency.

We can define another efficiency. So, we have defined so, far hydraulic efficiency and the mechanical efficiency; we will now define another efficiency which is known as volumetric efficiency of the pump.

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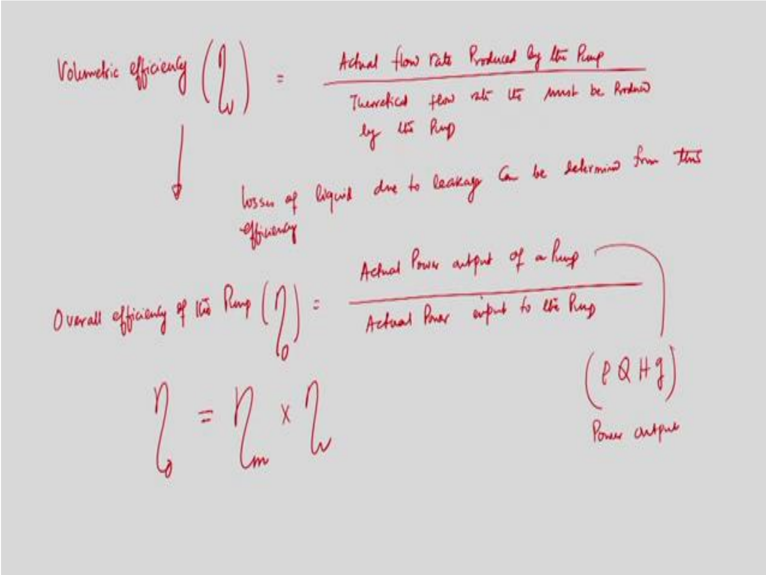

So, this is known as volumetric efficiency of the pump; volumetric efficiency η_V. Again, this is defined as the ratio of some quantities; so, volumetric efficiency defines actual flow rate, actual flow rate produced by the pump; actual flow rate produced by the pump to the theoretical flow rate that must be produced by the pump.

$$\eta_V = \frac{Actual\ flow\ rate\ produced\ \ developed\ by\ pump}{Theoritical\ flow\ rate\ must\ be\ produced\ by\ the\ pump}$$

So, from the definition it is clear that actual flow rate protruded by the pump and theoretical flow that must be produced by the pump. So, whatever flow rate produced by the pump you know theoretical whether ideal that may not be the actual flow rate. So, that is why you are defining volumetric flow rate; that means, whenever pump is a pump is being a pump is installed to handle water.

So, whenever pump is discharging water in certain place I mean in a particular industry; then probably water theoretical flow rate I mean actual in ideal discharge is not the actual discharge. So, maybe we are giving that that pump is designed such a way whenever we designing a pump we designed; I will say that whenever there are three important quantities are you know are there whenever you are designing a pump one is head another is discharge another is the specific speed.

So, whenever you are designing a particular pump, we should keep in mind that pump should be installed in that place. And there whenever I mean I know that assistant engineer or design engineer whenever he is procuring a particular pump he should supply three this three important data to the pump manufacturer; one is head the pump should we install there you know what should be the actual head pump need to develop? And against that particular head what should be the discharge; what should be the you know flow rate that pump need to supply?

So, this 2 are very important of course, specific speed. So, whenever a particular pump is designed, we know that the pump should ideally discharge this amount ah, but in reality, this is not the case I mean there are some losses there are some leakage. So, to take that leakage into account whenever we are installing a particular pump in a system, I mean maybe it is radial flow pump or axial flow pump.

So, we need to know that of course, we should not get I mean whenever design is pump manufacture of supplying data that fine if you install this particular pump in a system probably if the system resistance is like this; in that case pump will be able to deliver let us say Q 1 amount of flow rate.

But if we if we install that particular pump in that system and even pump to a pump and is install that system resistance is like this, but we may not get that Q 1 amount of flow rate I because of some losses; that means, there will be some leakage of flow rate. So, to take that leakage into account we are defining one efficiency which is volumetric efficiency. So, actual it is defined as the ratio of actual flow rate produced by the pump to the theoretical flow rate that the pump must produce.

So, from this ratio from this efficiency; we can have an idea about that losses of liquid, losses of liquid due to leakage mainly due to leakage can be determined from this efficiency from this efficiency. So, volumetric efficiency gives us an idea about the leakage of liquid I mean losses of liquid due to leakage mainly due to leakage. So, although we have designed pump to supply certain amount of flow rate, but if we install that pump in the system; we may not get that amount because of this leakage. So, that is why you are defining one efficiency is volumetric efficiency.

Last one is very important that is known as overall efficiency of the pump this call as overall efficiency of the pump; eta overall that is defined very important. So, this is defined actual power you know output of a pump and actual to the ratio of actual power input to the pump.

So, it is clear from this from the definition of the overall efficiency that whenever again I am repeating that if I run a particular pump using in electric motor. So, if I use an electric motor to operate a particular pump again it may be a radial flow form or axial flow pump or a mixed flow pump. So, actual power supplied to the pump is somewhat something that is there is power output from the motor.

$$\eta_O = \frac{Actual\ power\ output\ from\ the\ pump}{Actual\ power\ input\ to\ the\ the\ pump}$$

So, from this efficiency we can calculate that is overall efficiency can be written in terms of mechanical efficiency into volumetric efficiency. Because when you are considering overall efficiency, we need to concern mechanical efficiency also the volumetric efficiency. Because mechanical efficiency gives an idea about the power input to the shaft may be we are giving; let us say p amount of power to the pump, but that p amount of power is not consumed by the pump itself because of the bearing an bearing losses and losses in other moving parts.

$$\eta_O = \eta_V * \eta_m$$

So, that that is consume by the pumping system so, but we are supplying let us say some amount of power to the pump. And what is the power output of the pump? Because we know that pump in the pump is installed to operate under head of Q under head of H and discharge Q; then what could be the power output from the pump? So, this power output of the pump can be expressed in terms of let us say if pump is installed in a plus wire it delivers Q 1 amount of discharge against head of H then the rho Q into g this is the power output from the pump.

Power P = $\rho\ Q\ g\ H$

So, this is the power output by the pump and to operate that pump we had to supply certain amount of you know power because to run the electric motor; so this gives an overall efficiency of the pump, so these are the efficiencies. So, we have discussed so far hydraulic efficiency, mechanical efficiency, volumetric efficiency and the overall efficiency. So, with this, we next proceed to see about the proceed to discuss about H-Q curve and system resistance curve.

So, now we will discuss about the system resistance curve and the H-Q curve for the particular pump. And we will discuss that we will draw as pumping system when pump is suppose we place the particular pump radial flow pump; again if I can if I consider radial flow pump.

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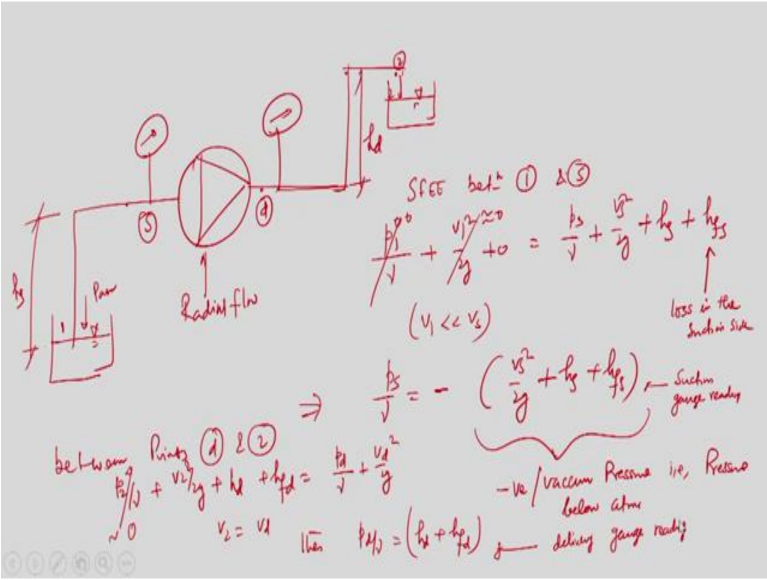

And I am assuming this radial flow pump is installed in a place and it delivers water and if pump is sucking water from a sump and it is delivering water to a place. So, this is let us say this is suction gauge and this is delivery gauge delivery pressure gauge.

So, this is point 1 and this is point 2, this is d, this is s suction this is delivery this is suction side this is delivery side see this is a radial flow pump or it is maybe a mixed flow pump. So, now and pump is installed so, that it will suck in a liquid form a some and it will discharge liquid to another place. If I apply steady flow energy equation between 1 and the point s; so, you can see from the schematic that the head developed by the pump will be the delivery gauge reading minus suction gauge reading that is the head developed by the pump.

. So, the total head developed the by the pump will be delivery gauge reading minus the suction gauge reading. So, now, if I apply steady flow (Refer Time: 23:45) between point 1 and s and between point 2 d and 2 then what would be the expression that I will write and from there I will have an idea about.

So, if I apply steady flow energy equation between point 1 and s, then what can I write?

$$\frac{P_1}{\gamma} + \frac{{V_1}^2}{2g} + 0 = \frac{P_s}{\gamma} + \frac{{V_s}^2}{2g} + h_s + h_{fs}$$

So, this sump is opened atmosphere so, and of course, the area of this particular sump is very large compared to the pipe. So, we can ignore this term because v 1 is much less than vs and this is open to atmosphere; so, gauge pressure;

V1<<Vs

$$\frac{P_s}{\gamma} = -\left(\frac{{V_s}^2}{2g} + h_s + h_{fs}\right)$$

So, this is we can see this is the negative pressure or vacuum pressure, this is negative pressure or vacuum pressure that is pressure below atmospheric.

That is quite obvious because only here the atmospheric pressure is acting Patm. So, of course, here at suction side pressure has to be less than less than atmospheric otherwise water cannot be sucked. this is suction gauge reading. So, this will be obtained by the suction gauge that is placed. So, if I again apply that steady flow (Refer Time: 26:11) between point d and 2.

So, if I apply again between points d and 2 then I can write that I can write that

$$\frac{P_2}{\gamma} + \frac{{V_2}^2}{2g} + h_d + h_{fd} = \frac{P_d}{\gamma} + \frac{{V_d}^2}{2g}$$

V2=Vd

$$h_d + h_{fd} = \frac{P_d}{\gamma}$$

So, what is the pressure rise across the pump impeller? So, if I write the pressure rise across the pump impeller.

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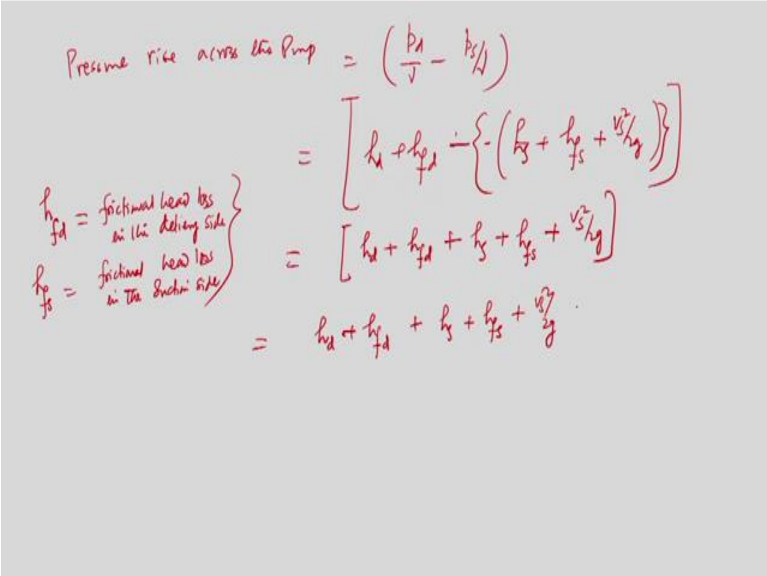

Pressure rise across the impeller = $\frac{P_d}{\gamma} - \frac{P_s}{\gamma}$ $= h_d + h_{fd} + h_s + h_{fs} + \frac{{V_s}^2}{2g}$

Head developed by the pump = $\left(\frac{P_d}{\gamma} + \frac{{V_d}^2}{2g}\right) - \left(\frac{P_s}{\gamma} + \frac{{V_s}^2}{2g}\right)$

$$= \left(\frac{P_d}{\gamma} - \frac{P_s}{\gamma}\right) + \left(\frac{{V_d}^2}{2g} - \frac{{V_s}^2}{2g}\right)$$

$$= \left(h_d + h_{fd}\right) + \left(h_s + h_{fs} + \frac{{V_s}^2}{2g}\right) + \left(\frac{{V_d}^2}{2g} - \frac{{V_s}^2}{2g}\right)$$

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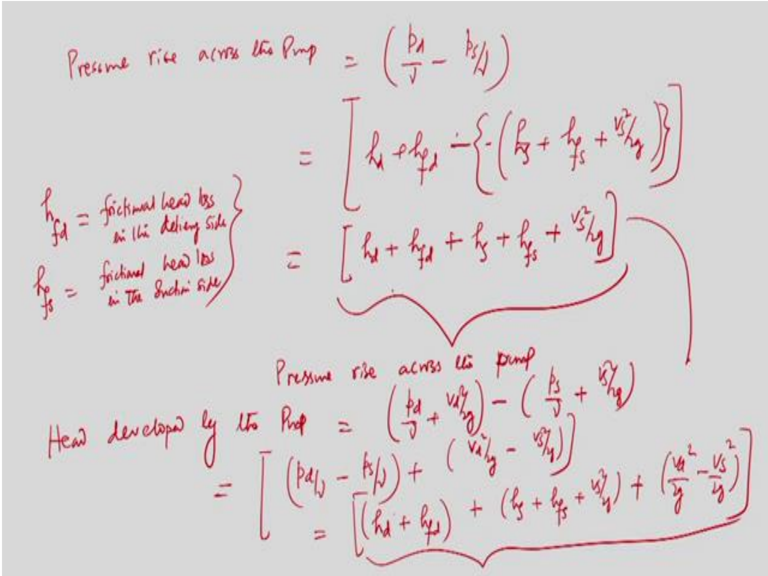

So, this is the expression of this is the expression of pressure rise across the impeller of the pump or across the pump.

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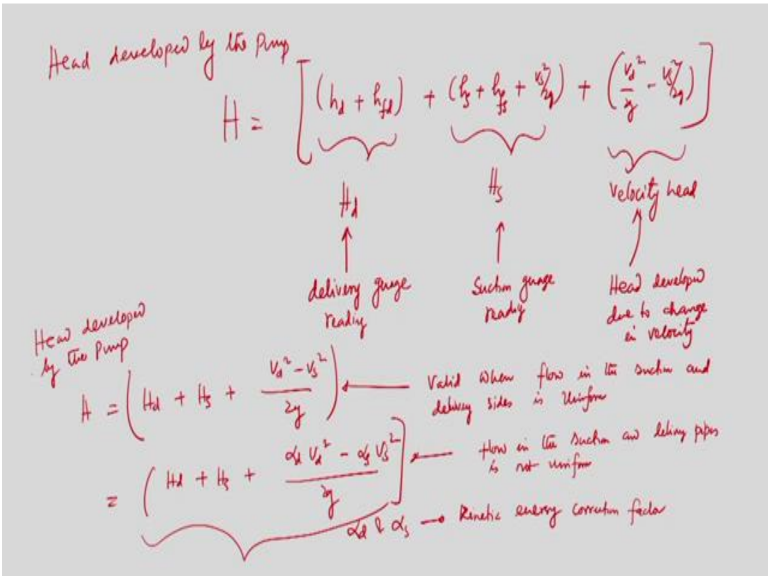

So, I am writing in three components. So, head develop the pump can be written in this form where there are three distinct terms; this is known as H I am writing this is capital H d this is capital H s and this is the velocity head right.

So, the total head developed by the pump is the expression which dictates the total develops by the pump. So, this H d is the delivery gauge reading, H s is the suction gauge reading and this is the component because this component is important because from this I can write that from this component I can write that head developed due to change in velocity.

So, there are three distinct terms; one is the delivery gauge reading, another is suction gauge reading and last one is essentially the head developed due to change in velocity you know suction in delivery side. So, this is the total expression of head develop by the pump; now I know that pump has if I install a particular pump in a station then knowing the static height in the delivery side, frictional losses at the delivery side, static height in the suction side frictional losses. Of course, by knowing the velocity at the delivery and suction side I cannot obtain the total head develop by the pump; that means, how much the head will be developed by the pump.

So, now if I draw the H-Q curve for the pump because this is very important H, what should be the H-Q curve for the pump? So, we have seen that the head develop by the pump can be written H = $H_d + H_s + \left(\frac{{V_d}^2}{2g} - \frac{{V_s}^2}{2g}\right)$. So, this is the head develop by the pump head developed by the pump; now this is valid (Refer Time: 35:54) very important I should discuss here this expression is valid when you know the flow at the suction and delivery side is uniform very importantly this is very important.

So, this expression is valid when flow in the suction and delivery sides is uniform right. So, we have identified the head developed by the pump is having three components; I mean head total had develop by contributed by three different components delivery gauge reading, suction gauge reading, and head developed due to changing velocity.

So, this expression is valid when flow at the suction delivery sides are uniform is uniform. But normally this is not the case that this is seldom true I mean this is not the case that whenever suction side and delivery side flow should be uniform because flow is taking place from a pipe. So, whenever flow is flowing over a solid surface, we know that boundary layer will definitely pumps it is not possible to have any uniform flow.

So, these expressions is not valid the for the cases when we are we are having non uniform velocities at the you know non uniform flow velocity at the suction delivery side. So, this expression can be modified

$$H = H_d + H_s + \left(\frac{\alpha_d V_d{}^2}{2g} - \frac{\alpha_s V_s{}^2}{2g}\right)$$

The last expression is valid when the flow in the flow in the suction and delivery pipes is not uniform rather non uniform and which is the case in which is actual case ; so, flow in the suction and delivery side is always non uniform because of the presence of solid surface and formation of boundary layer.

So, it is seldom true that flow in the suction and delivery side will be uniform. So, considering that the head develop by the pump should be modified by these kinetic energy correction factors. So, this is the actual expression of head being developed by the pump; I mean we need to take into account α_d and α_s which of the kinetic energy correction factor the inlet and outlet I mean suction delivery side.

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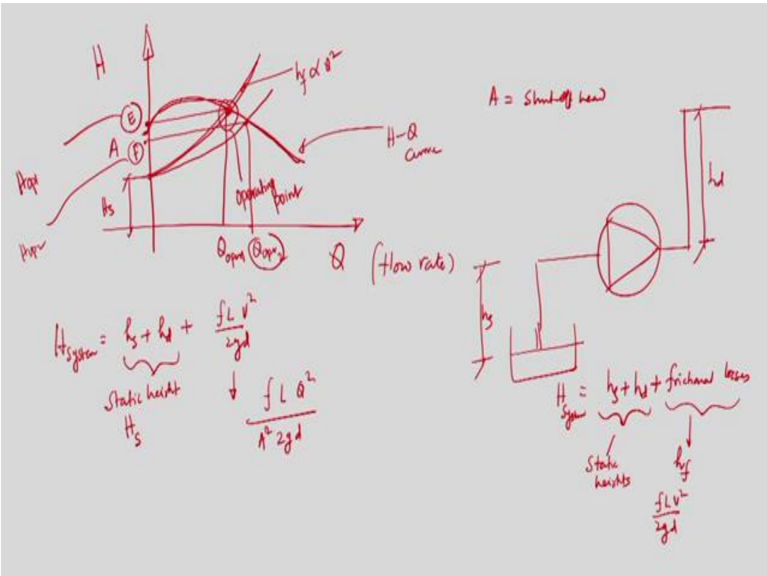

If I now draw the H-Q curve; so, if I draw this is important that H-Q curve. So, what should be this is flow rate and this is H. So, we know that if we keep on suppose the whenever pump supply supplying a particular pump this would provide; I mean they will provide the H-Q curve of the particular pump.

We need to know what the resistance curve system resistance curve is because we know where we need to install a install the pump. So, depending upon the place we need to calculate the system resistance and probably that is the information that we need to provide, but pump manufacturers should provide the H-Q curve.

Normally what is seen that if you keep on increasing Q; of course, frictional losses will be high. So, head developed by the pump will be reduced; so because with increasing Q as I said that frictional losses in the delivery side will increase I mean and also suction side. So, the total head develop by the pump will reduce; so, the H-Q curve is like this. So, this point this point is if it is A; this A point is known as shut off head the head corresponding to the point a is known as shut off head.

So, this is the H-Q curve of the pump right why this is happening? Because I can say that this is not a desirable for the pump operation, I will show you and if the pump H-Q curves looks like. It is very you know problematic because if I draw the system resistance curves; I do not know what the system resistance curve will be. So, suppose I am installing a pump in a place, where suppose this is the note you know symbol of the pump that I am using.

And it delivers water to a place where the total static height is hd and pump is drawing water from a sump where we need to overcome I mean pump needs to take you know total suction side and total head in the suction side is hs. So, pump should be able to develop a head so, that the static height in the suction side and the static height (Refer Time: 41:11) the delivery side should be taken into account.

Now the total head develop by the pump is hs + hd + frictional losses in the suction and delivery side that that is very important that is frictional losses. So, h s plus h d are the static heights these are the static heights in the delivery and suction delivery side plus frictional losses we need to take into account while we are calculating total system resistance. Because from H-Q characteristic the pump will be supplied by the you know manufacturer that is true, but in if we look if I look at the pump H-Q characteristic at the particular pump; then you can see that depending upon the you know nature of the system resistance, we may have different operating point.

But that is why we need to know the actual system resistance where pump should you know should be installed. So, suppose pump is installed in a place where we are, we need to we have static height in the suction side h s and static height in the delivery side h d plus some frictional losses. So, now, if I draw the system resistance then suppose this is the frictional losses it depends I mean we know that from our knowledge in fluid mechanics that what would be the frictional losses in the suction side and delivery side that we can calculate.

So, this is the system resistance H systems. So, this is a system resistance H system, and this is having the system resistance is having static height in the suction side static height in the delivery side plus frictional losses. So, these frictional losses hf is essentially we know from fluid mechanics that is governed by

$h_f = \frac{fLV^2}{2gd}$ {f = 64/Re}

in most of the cases you know in industrial application flow is not laminar because normally recommended velocity in the pipe is normally 1.5 to 2 or rather 2.5 meter per second. So, we need to calculate this f from you know Moody's diagram or there are some if the flow is turbulent then we have some other correlation into calculate. So, that is different issue all together. So, now knowing the value of friction factor we know I know the length of the pipe. So, if I know the discharge cube then I can calculate what will the value velocity and then I can calculate frictional losses.

H = $h_d + h_s + \frac{fLV^2}{2gd} = h_d + h_s + \frac{fLQ^2}{2A^2gd}$

So, why I am writing in terms of Q? Because this is the pump H-Q curve which will be supplied by the pump in manufacturer. And if we keep one increasing flow rate up suppose pump should be able to supply flow rate from its operation 25 percent to 125 percents of its capacity.

So, I cannot have geo discharge from the pump because the shut off head is there. So, we whenever pump is operating in a range of a flow rate then of course, if I keep on increasing the flow rate since frictional losses will increase; so head development by the pump will decrease. So, say for a for a given pump if I would like to have Q 1 discharge, then if I get H one amount of head.

And instead of that suppose again for the from the same pump if I would like to draw Q 2 amount of discharge where Q 2 is greater than Q 1 of course, I may not get h1 because in that case head developed by the pump I mean we should be s 2 and s 2 should be lesser than h 1 and that is why it is having continuously dripping characteristics and that is H-Q curve.

Now, if I superimpose you know system resistance curve over here then how can I obtain the system resistance? Because I know the system resistance is having 2 components one is the static height hs. So, this is static height hs; so, this is static height hs plus it is having some dynamic head loss that is that varies you know quadratically with u; so I will I will having a parabolic in nature. So, I have a parabolic in nature, so this is basically dynamic head loss and this is where hf varies as Q square.

So, of course, pump needs to overcome the static height that is H s plus there are there is a dynamic head loss that varies quadratically with Q. So, if I keep on increasing Q the (Refer Time: 46:09) system resistance will keep on increasing. So, now this is the pump H-Q curve that will be supplied by the pump manufacturer and this is a system resistance curve where we are having static height plus dynamic head loss and these you know point of intersection is the operating point. So, this is point intersection point; so this is Q operating and this is let us say H operating. So, if this point is e then we are getting e point is H operating.

So, this is the H operating. So, whenever system the point at which system resistance cuts the pump H-Q curve; H-Q curve we get the operating point. So, this is the operating point, this is the operating point and Q when H corresponding to the operating points are known as the operating discharge and operating head; so, we obtain.

Now, suppose the same pump is installed in different places where probably you know suction static height remaining same, but, but we would like to draw let us say relatively higher amount of discharge in that case if I keep on increasing discharge; then what will happen? So, our target is to you know have higher discharge from that particular pump.

So, maybe we suppose we need to the system resistance curve is like this. So, in that case we will have this is Q operating 1 and this is Q operating 2. So, if I would like to have higher discharge from that particular pump then that head developed by the pump will be reduced that is what I was discussing; that if I would like to have higher amount of Q then of course, we have to compromise that the head developed by the pump in that case H operating suppose this is the point f and they are H operating 1 and this is H operating 2 will be lesser than H operating 1.

So, this is the case; so by tuning the system resistance I mean depending upon the system resistance I mean static height; we cannot have any you know control because that depends in the situation. Suppose pump needs to supply water in a multistoried building or in industry where pump needs to supply water at say certain height; then we cannot you know you know compromise the static height hd or hs and if the pump draws water from underground sump; then of course, pump has to you know take into account as static height in the suction side ah.

In this context, I will discuss that normally pumps are installed in a flooded suction more, this is not a negative suction because the sump level is the pump axis the impeller axis is the pump axis. So, impeller axis is higher than the sum level. So, this is called negative suction operation, but this is not a you know this is not a good design because always it is advisable to go for a pump installation where pump should be always the pump axis should be always below the water level of the sump this is known as flooded suction.

Because for this case there is a probability of having some undesirable phenomenon that I have discussed in the last class that is cavitation. So, to avoid to avoid I mean to get in an effort to get relief from that kind of undesirable phenomenon as far as the pump operation is concerned; it is always advisable to go for the flooded suction where impeller that is a pump axis the datum of the pump that are impeller axis should be always below the water level of the sump.

Any way that is a different case that I will discuss when I will discuss the system design in detail, but now what you have seen that I can play this static section height also I can play the frictional losses. But if I now question is this is very important that suppose if I would like to have higher discharge from the pump. So, Q will be higher; so, whenever Q is higher we can see that the dynamic head will keep on increasing.

So, maybe I am getting higher Q from this particular pump and for that we are having let us say lesser amount of head developed, but when I increasing the Q probably because of that increasing increment in Q that frictional head loss the dynamic head loss will keep on increasing. So, at his curve may be shifted towards again top.

So, maybe I may not get exactly this amount. So, I may need to compromise maybe Q operating 2 is the desirable, but because if I have increased the volume flow rate from the pump. So, if I increase Q that dynamic head loss will keep on increasing and as a result of which the system resistance again the might be stiffer and it go off. As a result of which I may not get exactly Q optimum 2 rather Q operating 2 I may get another Q ; so, that we need to take into account ok. So, this is the system resistance curve and H-Q curve. So, you know that one important aspect of a pump is the head discharge characteristics.

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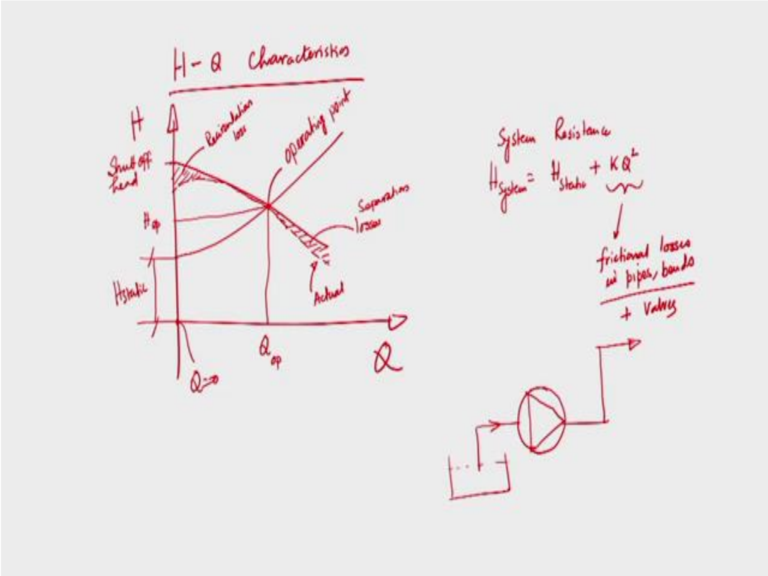

So, head discharge characteristics is very important for the pump selection other pump operation.

So, now sometimes to obtain the H-Q characteristics of a pump this is that this is a characteristic that should be supplied by the pump manufacturer and it is obtained from a test for pump testing. So, if I now try to plot the H-Q curve H versus Q that is the head developed by the pump versus the product to the pump and this curve is obtained from the pump testing.

And the pump curve is if I plot you know data obtain from cements the curve looks like this. So, this is the pump characteristics that is H-Q curve and of course, I know that if we have a system resistance and there will be a 2 components 2 parts of a system resistance; so, this system resistance.

Curve system resistance it is having 2 different parts one is static height H static plus another is dynamic head loss that is sometimes we write it K Q^2. So, this is the total system resistance H system that is static height that permits to overcome as well as the dynamic head loss this part takes care of the frictional losses. So, this part takes care of the frictional losses in pipes and bends etcetera. Also there are some other losses you know because of the presence of because whenever you are designing a pumping system you need to provide a few valves to control the flow rate.

So, there are a few valves because of the presence of valve again we need to take into you know we need to we need to consider losses. So, frictional losses in pipe bends plus some losses because of the; you know presence of valves etcetera. So, that is the dynamic head loss. So, if I plot the system resistance curve that is H static this is fixed because it depends upon the you know a level of pump house as well as the height (Refer Time: 53:28) we would like to discharge the required water. So, this is the H static and this part the dynamic head loss which is essentially a parabolic in nature; so, this is K Q^2.

So, this is the operating point and the head and discharge corresponding to this operating point is the you know this. Suppose this is Q you know operating and this is H operating; now you know the head corresponds to Q is equal to 0. So, this is the point where Q is equal to 0 there is no discharge this is known as shut off head; this is shut of head. So, depending upon the you know system resistance we have an operating point. So, this point is the operating point and head and discharge corresponding to these operating points are the you know; rated rather actual head and discharge that we that we can expect from this pumping system whether pump operation.

Now, this curve is not the actual because whenever fluid start flowing through the pump suction line. If I draw now a schematic suppose I have a pump maybe I am considering a radial flow pump. And it is used to cater water in a place which is let us say certain height away from the pump axis and it discharge water to a place. Now it also draws water from a sump and then we need to consider the total static head that pumps to overcome as well as the fictional losses that we have discussed.

Now question is whenever fluid is going through the pump section line and of course, when you discharge through the deliver line to the desired points; then there will be some other losses. And considering all those losses this shut off head reduces to another point and the actual curve will be like this actual; curve will be like this. So, this is the actual curve actual H Q curve that you may expect from that pumping system and there will be some losses this is the losses in the pump section side and this is also separation losses in the pump delivery side.

So, these are the losses the separation losses and this is also losses in the pump suction side and there maybe recirculation losses; recirculation loss. Although if we carry out

(Refer Time: 56:27) in for a pump to obtain its H-Q characteristic that pump manufacture should supply to the pumping system you know designer.

Because over is designing the pumping system here see needs to know the H-Q curve because depending upon the system resistance and after having this H cube curve he might select the operating point; whether that operating point is good enough to supply the required amount of water or required amount of fluid to that particular case that will depend upon the pump H-Q characteristics.

Whatever data we obtain from a cement; if we plot it will be having up H-Q curve which is continuously drooping in nature that is continuously decreasing. If you if you like to have higher discharge of course, we need to compromise the head that is true because as I increase head then if I see that that upon increasing discharge. So, if I increase discharge that if I would like to obtain higher amount of loaded through the pumping system; then of course, fictional losses will be high. So, I may not expect that the head that I will get will be equal when flow rate was lower.

So, that you need to compromise depending upon the requirement of the system; if we need a higher flow rate of course, we had to have we will have relatively lesser head whatever it is it is continuous drooping in nature ah. So, with increasing discharge the head that will be developed by head being developed by the pump will be decreasing. This continuously drooping characteristic of the H-Q curve of the pump is not the actual one because whenever fluid is flowing rather fluid starts flowing through the suction line and whenever pump is delivering water through the delivery pipe to the desire points; then there will be losses recirculation losses.

Recirculation losses are very important because whenever fluid is fluid through a you know pipe that is flow through a closed conduit; you know boundary layer it start develop. And because of the presence of solid surface you know there will be non-uniformity in the velocity distribution and because of that there will be recirculation losses in the form suction line and also there be separation losses in the delivery line. So, considering these 2 losses actual curve will be like this that whatever I have plotted here. And now question is if I am an assistant designer if I am given responsible to design a pumping system; I will always prefer to have a H-Q curve which should be continuously drooping in nature, why? That I will discuss. So, this is the H-Q curve actual H-Q curve.

Now, what I said now that if I am a if I have given responsible to design a pumping system suppose I am asking for manufacturer to supply a pump maybe it radial flow pump then I will definitely demand that H-Q curve of that pumps should have a continuously drooping characteristics why? That I will discuss now that is if I now discuss that issue.

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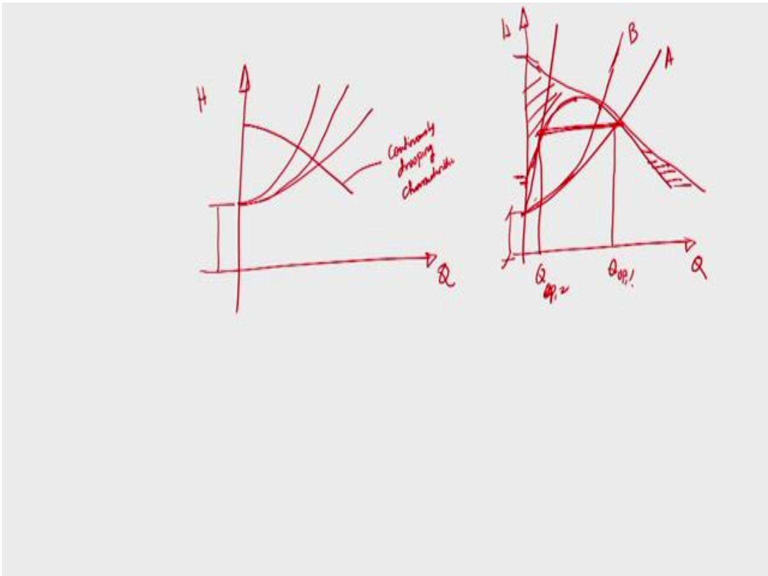

So, this is H-Q curve this is head developed by the pump and this is Q. So, as I said that of course, if I draw the in experimental experimentally; obtain data then I may have a continuously drooping characteristics like this.

And if I have a system resistance curve like this then no point now because of some reason operator or malfunctioning of the operator; if operator tries to you know close the delivery valve placed in a delivery line then of course, this resistance distance will be much more stiffer. So, system is static although static height will remain same, but the dynamic head loss will keep you know keep on increasing as a result of which the system resistance curve would be stiffer and the curve will move towards up.

So, there might be a situation because of this there might be a different reasons, I mean there might be different reason because of malfunctioning of the operator or because of closing of the delivery valve; this curve will try to sit up to up. Now if it is drooping characteristics there will be no problem as far as the pumping operation is concerned.

Now if I talk about the actual H-Q curve that is expected and that is really you know the happening whenever you know pump starts its operation; that is there will be a losses recirculation loss in the suction side and also there are losses in the delivery side ; instead of having a this is this is continuously drooping characteristics. So, this is continuously drooping characteristics; now if I talk about now if I consider the instead of a continuously drooping characteristics, rather what is the actual curve that is if the pump curve is like this that ah.

And so if the ideal curve is like this; so maybe this is the shut off head is reducing because of the recirculation losses. So, form this is a actual shut off head now this is the losses. My question is if I have a pumping you know pump operation to the pump characteristic curve like instead of a drooping; characteristics I have a curve like this; so, and these are the losses.

Now if my system this is the new shut off head now my system resistance curve is like this suppose this is my static height and system resistance curve is like this. So, this is my operating point this is the operating point and you know this is Q this is H; so, this is the Q operating. Now pump is always try to experience a relatively lesser system resistance that is what is expected.

So, as I said you because of some malfunctioning of the operator pump operator or if pump operator closes the delivery valve; gradually because of the because of some you know because of controlling the flow rate in the delivery lines, then it will system curve will be stiffer. And then system curve if suppose system curve becomes stiffer and it there might be a situation when system curve might take a step like this and then whenever it closest to the; whenever it appears to the peak point here then it will suddenly come to the shut off head otherwise.

So, pumps will (Refer Time: 63:11); on the other hand if I consider the first case let us say this is the system distance curve A, when you know operated they didn’t close the valve; then somehow because of as I said that if form operator start closing the valve gradually, then curve will be stripper and then curve will shift over here and there might be situation when pump system resistance curve will be here and then again it will come to the here to provide a relatively lesser amount of flow rate; let us say Q 2 operating this is Q operating 1, this is Q operating 2.

So, system resistance curve will now shift to the another point and as if the pump is trying to operate between these 2 points. So, as I said you the pump will allow try to experience a less resistance less system resistance. So, instead of providing the desired flow rate because of malfunctioning of the operator or because of closing the gradually because of closing the valve at the delivery line gradually system resistance might be stiffer system resistance will be stiffer and there might be a situation when system resistance curve sit to another point.

And as if the pump is fluctuating between 2 operating point and it will be like this that just like a pulsating kind of flow that it will need some times Q operating one and sometimes Q operated is not an desirable phenomenon. This is not a desirable phenomenon for the pumping operation at all as well. So, keeping this in mind whenever I; is I will design or someone will design a pumping system he or she should always look for a pump H-Q curve which will have a continuously drooping characteristics.

So, that the problem because problem related to this I mean this might happened. So, this kind of problem can be avoided that is what is very important I mean someone should take into account whenever he or she is selecting the pump and designing the system ok. So, I stop here today I will discuss in the next lecture.