Principles and Practices of Process Equipment and Plant Design
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Module - 04
Lecture - 56
Plant Hydraulics (Contd.)

Good day to you all. Today we are going to have the 3rd lecture on the Plant Hydraulics, which is a continuation of the topic which requires in order to design a plant; you need to have a fair idea of the plant hydraulics, because it is an integral component of any plant design.

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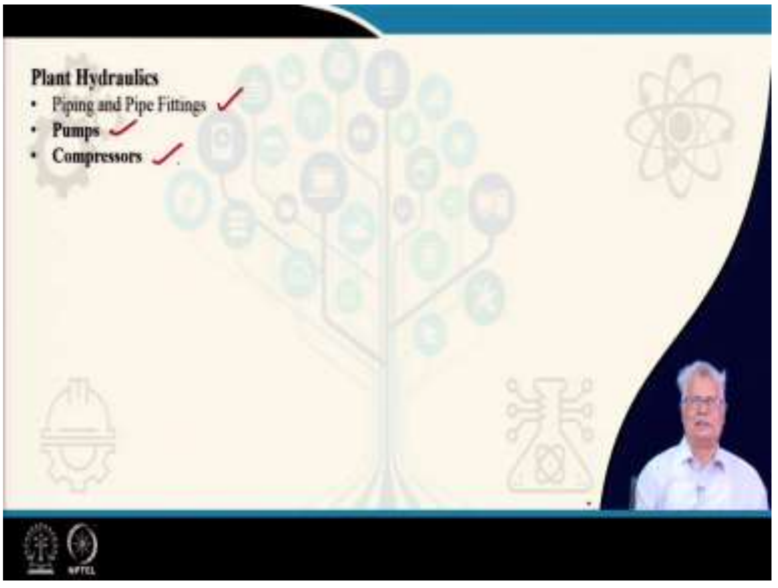

If you look at the plant hydraulics, there are basically three components here and these components are the piping and the pipe fittings, which we have covered already. We have also talked about pumps the different types of pumps; we will be concluding the topic on pumps today and we will complete the compressors as well.

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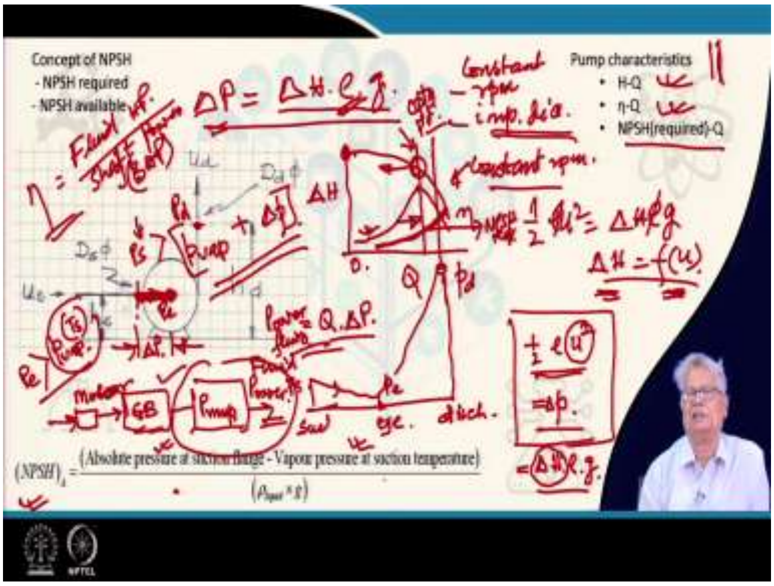

To start with let us look at a few concepts which relate to the pump and what we start here today first is about the pump characteristics. We know in a hydraulic circuit, the purpose of a pump is to supply the energy; the energy which is required to overcome the flow resistances. The flow resistances includes a frictional head that is to be overcome as well as a static head difference.

What do we have here is a schematic of a centrifugal pump and there are certain parameters which are located here. We know that in a pump, in a centrifugal pump particularly; that is what we have been talking about in the last class, you have a pressure here which is a pressure at the suction, you have the pressure at the i and you have the discharge pressure.

So, if I call this P d and if I put it in a scale here, this is my eye and this is my discharge point or discharge flange and this is my suction flange. In that case, the suction the pressure here will be P s, the eye will be at a minimum temperature, which a minimum pressure which is P i and your discharge pressure is after the fully thing the pressure has been developed.

So, this will be your p discharge. The pressure difference between the suction and the eye, basically is responsible for the flow in this part; that means from the suction flange to the eye of the impeller. Then what happens? After it has entered the eye, it turns around with the impeller with a peripheral velocity of the impeller being something u; if it is u, then rho half rho into u square is basically the energy content of the fluid at the tip of the impeller with which it goes out.

And quite naturally this gets converted to the pressure energy and if we see this, this is going to be your p, which is basically the delta p which is the difference of energy at the inlet and the discharge point. I could also write this as instead of writing delta p, I could write this as the head developed into rho into g. So, quite naturally what we have here is, my delta H will be related directly to your u square.

And if you can see very well that, if I write half rho u square is equal to delta H into rho into g; rho rho cancels off and what basically I get here is delta H is function of u. That means, the head developed by the pump is a function of u and it is not a function of rho, this is quite interesting. You will also see one more thing. If you look at the delta P which is getting developed here; it is what, it is basically delta H into rho into g.

Now, your delta H is a function of u. What is u? U basically is a function of the impeller diameter and the impeller rpm. So, quite naturally if you are talking about a pump centrifugal pump characteristics, we will be talking about constant rpm and naturally constant impeller diameter, that is too obvious in fact; that means if I am talking about a pump, its impeller diameter is also fixed.

Now, let us look at what the characteristics should look like. You will also notice here that H is basically a function of u's and I am going to draw this as a relationship between Q and the delta H, which is a head developed here. In case of a centrifugal pump, this is for obviously a constant rpm and certainly impeller diameter is also the same for the pump. So, I am not writing it again, but this relationship usually will look like this.

Now, corresponding to Q is equal to 0, naturally the head is maximum. So, what you have? Here in this case no flow; that means your Q is 0. So, what you have here is, this is the shutoff head; the shutoff head being delta H shut off, the delta the shutoff pressure is going to be simply that delta H multiplied by rho into g. This is what you see that if you close the discharge of a centrifugal pump fully; it shows that the pressure discharge gauge goes to its maxima and shows you the shut off pressure.

Do not run any centrifugal pump throttle below 30 percent of its capacity; why, we will just see that now. Now, if I am talking about a pump; I will be having a motor possibly, which will be running the shaft. Now, you may have a gearbox in between; you may not have that also, in case the motor is driving the pump directly. Then what you have is the pump. What does a pump deliver?

The pump delivers fluid power. And what is the fluid power? It is basically the volumetric flow rate Q multiplied by delta P that is your power, which is delivered to the fluid. So, what is the definition of the efficiency for the pump? If you just look at the pump, your efficiency is going to be fluid horsepower divided by the input power to the shaft, which is also called the BHP, sometimes it is called the SHP also.

So, let us look at how this efficiency should vary. If I have one more axis here, which is eta; you will find that the efficiency would go up and certainly beyond a particular value, I will start dropping down again. So, this is the axis for delta H and this is the axis for efficiency eta; that means the efficiency will peak at a particular value. Quite naturally you would like to have minimum of input power for your maximum amount of the fluid horsepower.

So, you would prefer to have your operating point somewhere here; this is the operating point, which is a desirable one, very close to your maximized efficiency. Now, let us have a look at something very interesting regarding the efficiency plot. You will find here that the efficiency initially is very small, very low; why? Because you are not sending out much of Q, your delta P is high.

So, what happens is, inside your liquid, there is lot of churning and this leads to heat build-up or heating up of the liquid inside your casing. If the heat build-up is sufficient, it is also possible that it will start vaporizing and what you will be having here inside is a mixture of liquid and vapour. The moment you have a mixture of liquid and vapour, your rho falls drastically; your delta H is not a function of rho, but your delta P is, it is a directly direct function of rho, it is proportional to rho in fact.

So, the moment the vaporization starts inside and the chance of this is very high when you have very little flow, because the efficiency is pretty low because of this churning. What you find that, your pressure shown by your discharge pressure gauge would fall suddenly.

So, that is the case of cavitation what you have. And if you throttle your centrifugal pump below 30 percent, you will definitely lead to a problem of cavitation. So, we have an idea that what exactly is a headquart H Q relationship and what exactly is efficient Q relationship. And we definitely have realized that, we do not want any vaporization inside the casing, let us look at it.

We know inside the casing, the minimum point of pressure is the eye of the impeller. We do not have any way of measuring the pressure at the eye of the impeller; but we can have an idea of the pressure at the suction flange, definitely at the discharge also, we have a pressure gauge. And we also know that the pressure at the suction plant has to be higher than the pressure at the eye; otherwise there is no question, otherwise how will the liquid flow from this point to the final point.

So, what you do here is something, instead of; that means in order to avoid vaporization, my P e has to be greater than the vapour pressure. Vapour pressure at which condition? At the suction temperature, because the temperature here and the temperature here are practically same.

So, we since you cannot specify these P e, you will always specify the minimum limit of P s, which you can have, which will still avoid vaporization of a liquid inside. That means, the P s has to be higher than your p vap; your P s has to be higher than your vaporization pressure plus the delta p.

Which delta p? This delta P is the pressure difference between the suction point and the eye of the impeller. And on what it depends? It depends on what exactly is the geometry here; what exactly is the arrangement inside the. And who knows it best? It is best known by the manufacturer.

So, the manufacturer says, yes you definitely have to ensure that your pressure at the suction point has to be above the vapour pressure of the liquid which it is pumping by a definite amount and this is called that NPSH. The NPSH is specified again in terms of head; because the head capacity is a curve and why we are dealing with the head that also we have seen.

So, they by definition, the manufacturer says that you have to have the absolute pressure at the suction flange, the absolute pressure at the suction flange minus the vapour pressure at the suction temperature divided by the rho g of the liquid; this is the amount of NPSH which has to be there, definitely you have to provide at or this is required in order to avoid cavitation in this specific pump.

So, what you have is, usually one more chart is also there, we will have a look at it. What you find here is something interesting. You will also find the NPSH with respect to Q; if I have this as NPSH required, increases with Q. The question is, why should it increase with Q? What is this NPSH required? It is the excess head that you must have at the suction point to avoid cavitation at the eye.

Now, how much of excess? The excesses due to the pressure drop here and this pressure drop increases with flow rate. So, quite naturally at a higher Q, you must have a higher NPSH requirement.

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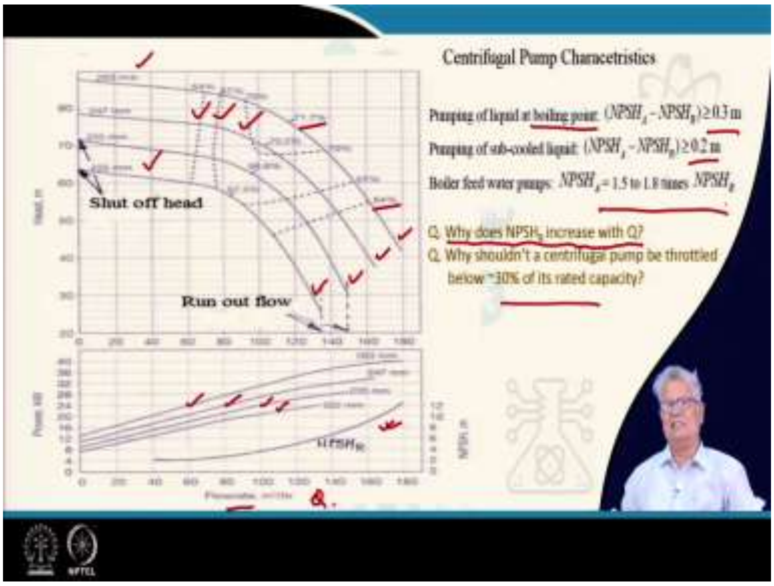

Let us look at a typical industrial pump characteristics, it is a typical centrifugal pump here. Now, what we have here is, you find that there are one, two, three, four impellers which are fitted; typically large industrial centrifugal pumps will have options of fitting a set of impeller in it. The impeller sizes will be slightly different in diameter; in this case it varies from 260 is a maximum and 222 is the minimum.

The efficiency instead of showing it as an independent line; the iso efficiency lines have been drawn here. And what is the range of efficiency here? The minimum shown is 65 or 64 and a maximum it has gone up to about 72. And you will see the contour here and if you really draw, you will have a peak and it is close to the peak value normally what you go for it, I mean you set your operating point there.

There is a something term, another term which is called run out flow; that means a centrifugal pump if you have practically no resistance as at its discharge, that means the head develop will be very low, but its flow rate will be very high. So, that is the end, the limit up to which you can operate your pump is called the runout flow. Quite naturally it is a maximum flow, but you should never go up to the runout flow; because even at runoff flow, your pressure your efficiency is going to be very poor and your head developed is going to fall.

Now, if you look at the power, how it values with a flow rate; quite naturally the power is what, it is basically a product of the delta P of the pump, which is delta H multiplied by rho multiplied by g and the volumetric flow rate Q. So, the we find that the head capacity or H delta H Q plots are different for different impeller sizes. So, naturally you will also have different power lines or different power variations with the flow rate Q.

You will have a single NPSH required line here, irrespective of the impellers. So, that is a question I have. The first question I have already answered, why does NPSH required increase with you; it simply increases, because it is a pressure drop which it has to account for to avoid cavitation and it is a pressure drop means, it is a pressure difference between the suction flange and the eye of the impeller.

So, with Q increasing, naturally the pressure drop would increase with roughly Q to the power about 1.8 and this confirms to that. So, with Q increasing, your NPSH R increases that we know already. Why should not a centrifugal pump be throttled below 30 percent of its rated capacity?

This we have said, because below 30 percent the efficiency falls very largely, very to a very significant extent; there is heating up of the liquid which keeps on recirculating within the casing and it leads to a vapour lock of the pump or cavitation of the pump whatever you may call.

Now, in your installation therefore, you must have a particular value of NPSH depending on your suction pressure. And this has to be higher than the required specified by the manufacturer. There could be different conditions of the liquid, like a liquid which is boiling. In that case, the available should be in excess of the required NPSH by a minimum of about 0.3 meters; for sub cool liquid it is 0.2, yes we can go closer to that and in case of boiler feed what pumps its more and we have to keep a large difference.

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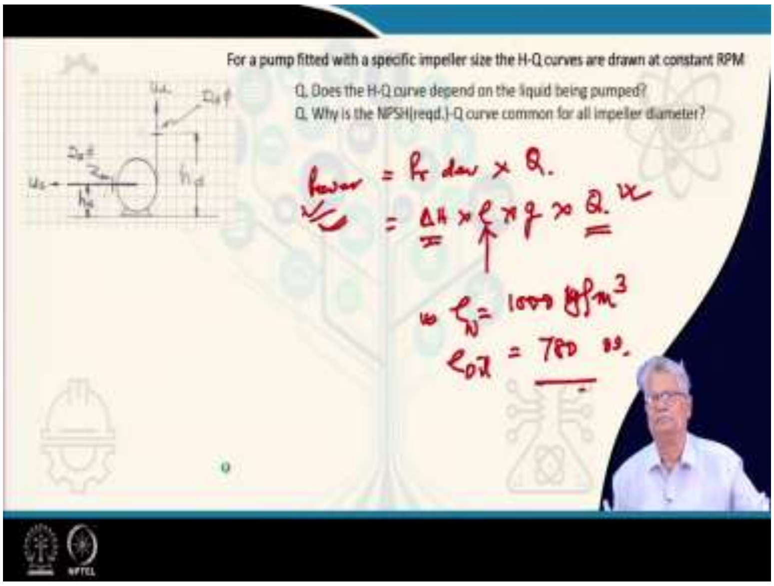

I have the same questions repeated here. Does the head capacity curve depend on the liquid being pumped? Obviously, no; because we have shown earlier that it is independent of the row. So, one thing is very clear, the supplier will normally be supplying you with the pump these characteristics and these characteristics though are usually experimentally generated using water, the head capacity relationship is irrespective of whether it is for water or it is the same head which is developed for your any oil or anything heavier than water which is being pumped.

That means, the head capacity characteristics is not a function of the liquid, which is being pumped. This gives a very interesting point also. You will find the pressure, the power of the pump is equal to what; is equal to pressure developed multiplied by flow rate Q. What is your pressure developed? It is delta H into density into g multiplied by the Q is going to be your power of the pump. This divided by the efficiency of the pump is the input of power which is going to be your pump shaft.

Now, irrespective of your liquid, your delta H is same; that means if you have the same flow rate, your head developed is the same. But your power will be different depending on the value of Q. For example, when you are pumping water; you are having a flow rate, you are having row of 1000 kg per meter cube.

When it is oil, it is maybe 780 kg per meter cube. So, quite naturally for the same flow rate of Q when you are pumping water, your power will be higher and very often it happens if you have a oil service pump, which you are priming with water and if you open your discharge high, so that your Q goes up, it draws too much of power and the pump trips.

So, be careful before you try your pumps out; what your pumping should have a density close to what it is designed for or what it is installed for. And if you have higher than that higher than this specified flow rate or higher than the Q what it is supposed to deliver; your power would go up and normally the electrical drive will trip, this is a very common thing which happens at practice.

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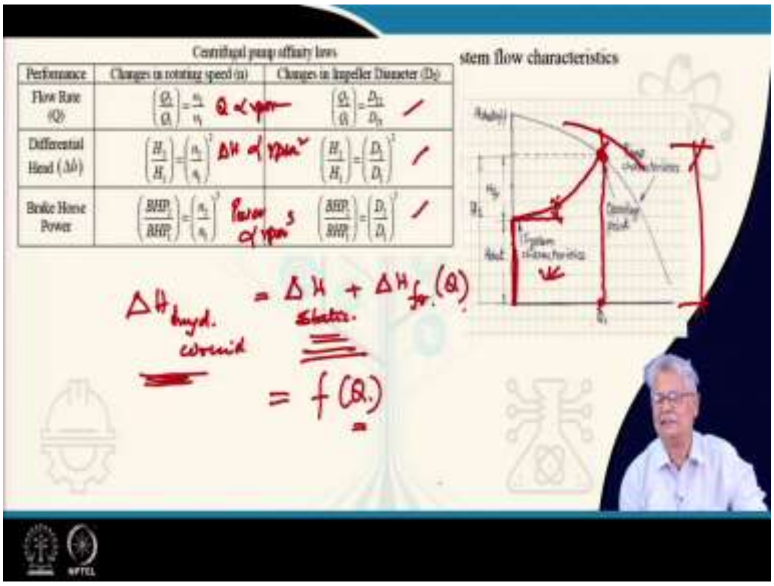

Now, we have certain relationship between the flow rate, the differential head and the brake horse power, which is basically the shaft power, which are called the pump affinity laws. The first one is, the Q is proportional to the rpm m; the second is the head or the rather the delta H is proportional to rpm square. Here what it says; the power is proportional to rpm cube, exactly the same way it is also similarly related to the diameter.

I have talked about the pumps and its characteristics. When I was talking about the system; that means a hydraulic system or a hydraulic circuit which has got a pressure drop. We also have seen that the pressure drop in a circuit has got two components. The total head drop in the hydraulic circuit is equal to the head drop due to the static level difference plus the delta H due to friction.

So, and it is definitely a function of the Q; quite naturally this is not a function of Q, but this one is. So, if we have also seen earlier that, how to draw the relationship between this delta H and Q. So, this is what we have seen that, in my circuit this is my static difference and as my flow goes up, there is additional amount of frictional drop here in the due to the head and the actual variation of head required in order to have a flow of Q takes this particular shape. This is called a system characteristics.

Now, you have a pump in the circuit, we will see a little bit of details of this after this. Now, in order that this head is overcome at a particular Q 1, the quad the amount of head that is required to be developed by the pump has to match and it has to be on the pump characteristics line. Which line? The head capacity line.

So, what you normally have to do is, you have to draw the system characteristics on the pump characteristics line; at the point of intersection, you know what exactly is going to be the flow rate through your Q. Your selection of pump also comes from this. We have an idea that this is going to be your head required for a particular flow rate. So, what you do in that particular case is, you have to have; you know the flow rate and you know this particular operating point.

So, you have to choose a particular pump, which develops a head more than this. So, that I can throttle my pump a little bit; I can throttle my pump a little bit, have the additional pressure drop or head drop across my throttling valve and have my delivery of the Q as it is desired.

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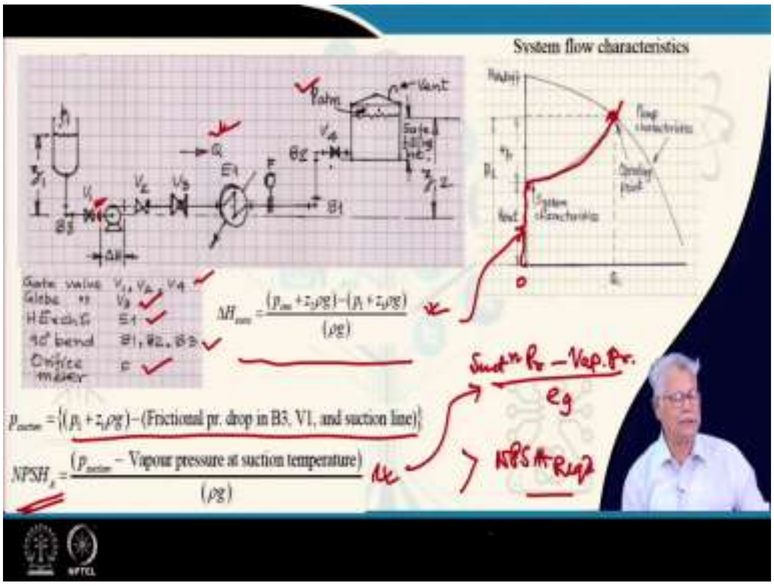

Here is an example. The example that we have here is a simple one; I have a column with some liquid level here and all my reference levels are with respect to the eye of the pump or the suction level of my pump itself. You can see z 1 is the elevation of the liquid level inside my tower, where the pressure inside at the column bottom is p 1.

The what are the fittings here? The liquid comes, there is an isolation valve for my centrifugal pump here; there is a discharge valve, there is a globe valve which is for regulating the flow, then I have got a heat exchanger, which possibly will be cooling the column bottom.

Then I have got an orifice meter here or a flow meter of any type; then I have got 290 degree bends, again another isolation valve for my tank and it goes through the tank and the tank is having a vent, so the pressure inside is p atmospheric. Tanks normally will have a maximum level of filling, which is called the safe filling height.

Now, you will also notice that I have specifically mentioned z 1 and z 2; these are the two levels with respect to the same base, which is the suction or the eye level of your pump. The components which are present in my hydraulic circuit are the gate valve V 1, V 2, V 4; the globe valve V 3, the heat exchanger E 1 90 degree bends are

B 1, B 2, B 3 and you have an orifice meter F. Typically the heat exchanger will have a pressure drop of around 0.7 kg per cm square.

And you already know that, when we are talking of pressure drop; there is a way of finding out the pressure drop for a particular flow and if my flow rate is Q meter cube per hour corresponding to that and if I if my since I assume right now that my circuit is known, maybe it is a 4 inch line or so. Corresponding to that, I it is possible for me to estimate the frictional pressure drop as well as the static head difference which is z 2 minus z 1.

Now, how do I estimate the static head or rather the constant pressure difference which does not depend on the flow? It is basically the pressure at this point, I bring at the static head; I pressure at this point, I add the static head here. And the difference between these two is something which is constant and is not a function of the flow rate. So, quite naturally this is your static head load.

As the flow increases from 0, the additional frictional head keeps up and we have it here as my operating point, that we already have discussed. And here what we do? We just make one more additional calculation. This is finding out the NPSH available, the manufacturer has told us the NPSH required what it should be. How do I calculate by NPSH available? It is nothing, but suction pressure minus the vapour pressure divided by rho g is equal to this.

What is my suction pressure? I have a suction pressure here and how did I find, sorry I had a suction pressure at this particular point. And how did I arrive at this? It is basically this p 1 plus the static head minus the frictional pressure drop.

So, this is how we arrived at the suction pressure and we found out the NPSH here. And once we have done this, we just check this one is above the NPSH required. And we already, we can always follow the guideline which is there in the previous slide.

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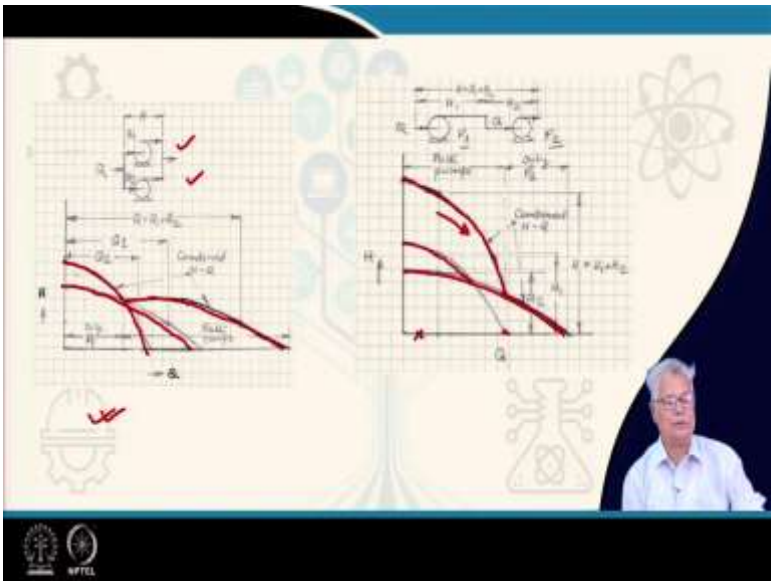

It is possible for us to have requirement of putting two pumps in series. When I have two pumps in series, naturally their heads will get added up. If I have here the characteristics of my pump 2 here and the characteristics of my pump 1 here, one thing is bound to happen. When my flow rate is less; that means I have throttled it much, both the heads get added here. But beyond a threshold of flow beyond a threshold of flow, which is this, there is no nothing to add here.

So, quite naturally the combined head will have a nature like this; this is how it changes with Q. That means if I really have to add or increase the total head required for overcoming the friction or overcoming the resistance to flow, I will be adding pumps in series.

And here is another case, where I really do not need to increase the pressure drop much, but I need to have higher quantity of flow. In that case, I have one characteristics here; the second pump characteristics here, which are pump 1 and pump 2 and a combined characteristic will be looking something like this.

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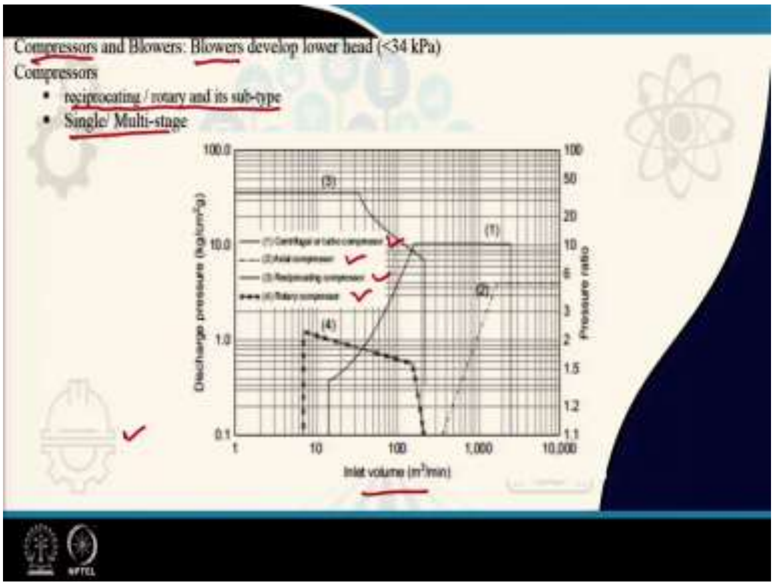

I think with this, we have a fair idea of my pumps and we have talked about pump selections; what we will be doing is, we just introduce you today to the compressors and blowers with the slide that I have here with me on the selection and normally this is a relationship between, this is a relationship between the discharge pressure requirement and the inlet volume.

Possibly you will be interested in knowing that, what we use two things for compressing or increasing the pressure in case of compressible fluids; one is a blower, the other is a compressor. Blowers are normally for high flow rates and lower head; typically below about one third atmosphere, which is around say 34 kilo pascal's or so.

Compressors whether it is a rotary type or a reciprocating type or that could be various different types of rotary compressors; could be single stage or it could be multi stage also. Here we have a typical chart which is used in industry, to select the type of compressor which is shown here; centrifugal or turbo compressor, axial compressors, axial flow compressor, reciprocating compressor and a typical rotary compressor.

Now, what I would like to say here is something else; whenever we talk about blower or heat compressors, quite naturally the volume of flow rate will be different at the suction at the discharge. In case of a compressor, it is always the flow rate at the suction condition which is referred to as the capacity; that means the capacity of a compressor if I say it is say 150 meter cube per hour, it means the suction volumetric flow rate is 150 meter cube at the suction conditions.

With this I think we have an idea that, the in industry this particular chart or other graph is going to give us an idea that what type of compressor you choose from and we will do the calculations and other ways of finding related to the compressible flows in a next class.

Thank you.