Oil Hydraulics and Pneumatics
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Part 03: Pump controlled actuator system: Variable-displacement, Pump-controlled rotary actuator system
Lecture - 82
Pump-Controlled Hydraulic Systems

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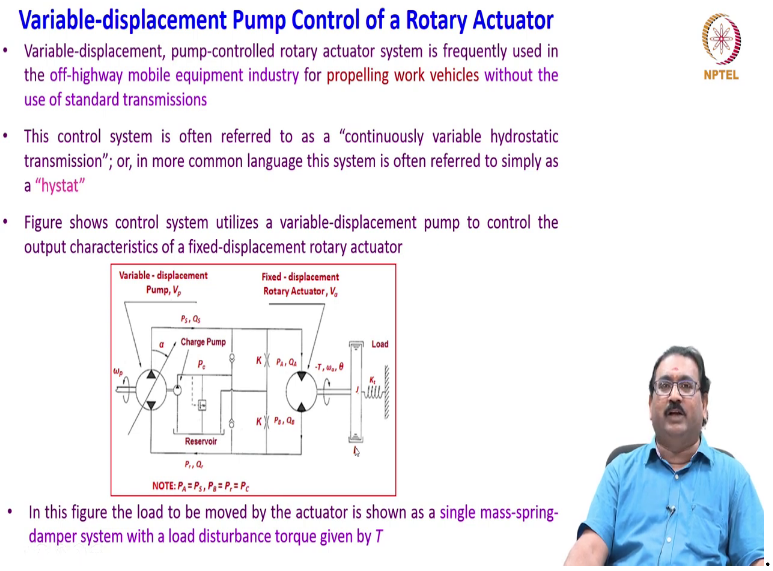

My name is Somashekhar course faculty for this course. Now, we will move on to the Variable-Displacement Pump Control of a Rotary Actuator. Variable-displacement pump-controlled rotary actuator system as I have told you frequently used in off-highway mobile equipment industry for profiling the work vehicles without the use of standard transmissions.

This control system is often referred to as a continuously variable hydrostatic transmission, or in more common language this system is often referred to as a hystat. Figure shows a control system utilizes a variable-displacement pump to control the output characteristics of a fixed-displacement rotary actuator.

Now, we will see here friends I have shown you the many components here. Here you will see now pump is a variable-displacement pump. The angular velocity is omega p. Here you will see it is a the fixed-displacement rotary actuator. Again here I have connected the load arrangement which is shown as J, b damping and a spring k t. Let us we will see now here. Also you will see a charge pump circuit I have shown here which is coupled to the main pump.

Let us will discuss this thing in detail. In this figure, the load to be moved by the actuator is shown as a single mass-spring damping system with a load disturbance torque T.

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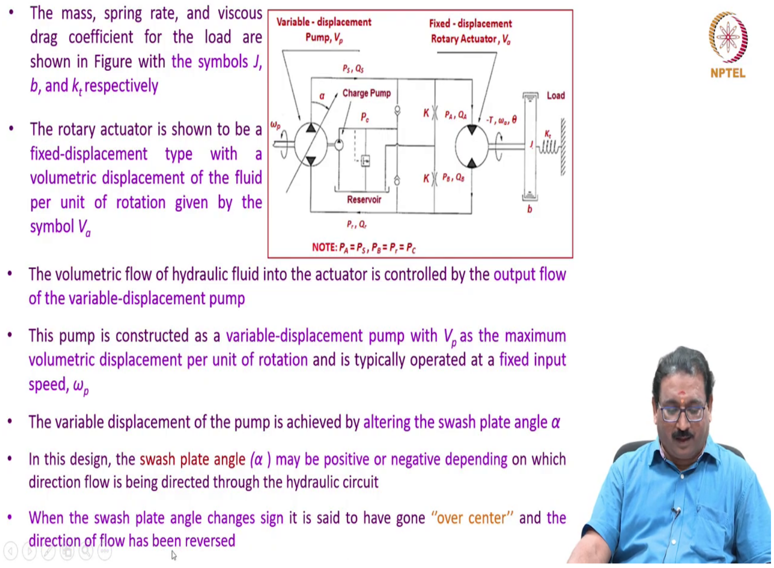

So, the mass, spring rate and a viscous drag coefficient for the load are shown in the figure with the symbols J, b, and k t respectively. The rotary actuator system shown here it is a fixed-displacement type with a volumetric displacement of the fluid per unit rotation is given by the symbol V a here. The volumetric flow of the hydraulic fluid into the actuator is controlled by the output flow of the variable-displacement pump.

This pump is constructed as a variable-displacement pump with a V p as the maximum volumetric displacement per unit rotation and is typically operated at a fixed input speed omega p. The variable-displacement of the pump is achieved by altering the swash plate angle alpha.

In this design, the swash plate angle alpha may be positive or a negative depending on which direction flow is being directed through the hydraulic circuit. When the swash plate angle is changes sign it is said to have gone over center and the direction of flow has been reversed.

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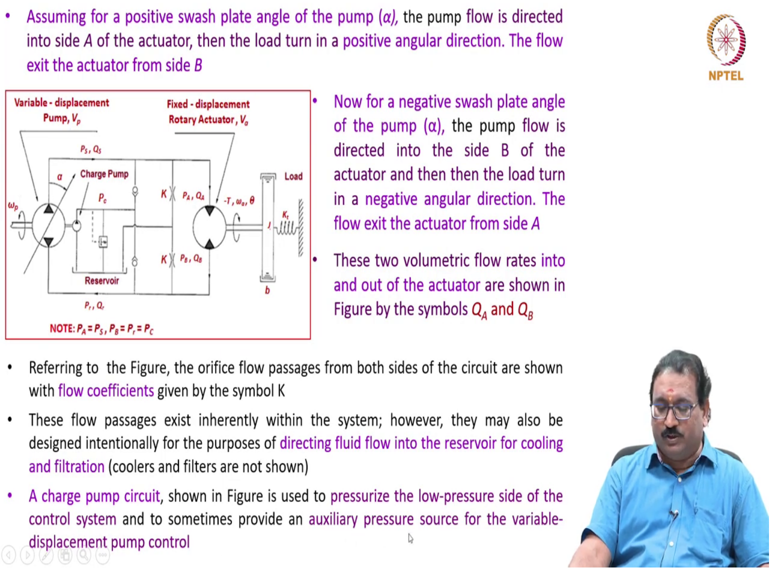

Assuming for a positive swash plate angle of the pump alpha, the pump flow is directed into the side A of the actuator then the load turns in a positive angular direction. The flow exit the actuator from the side B. Now, for a negative swash plate angle alpha, the pump flow is directed into the side B of the actuator and exit the actuator from the A side vice versa here. These two volumetric flow rates into and out of the actuator are shown here as a Q A and Q B.

Referring to the figure, the orifice flow passages from both sides of the circuit are shown with flow coefficients given by the symbol K, low Reynolds number leakage coefficient it is.

These flow passages exist inherently within the system; however, they may also be designed intentionally for the purpose of directing the fluid flow into the reservoir for cooling and filtration which is not shown here.

A charge pump circuit shown in figure is used to pressurize the low pressure side of the control system and to sometimes provide an auxiliary pressure source for the variable-displacement pump controlled system.

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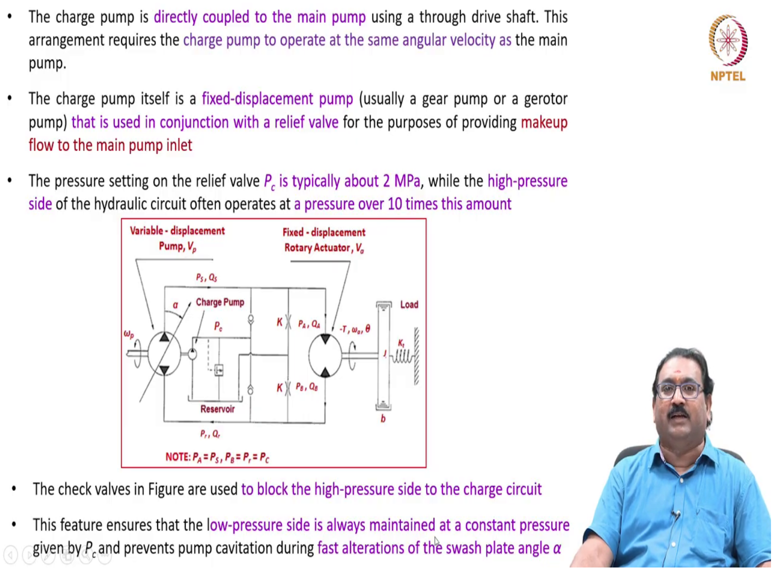

The charge pump is as I have told you directly coupled here, you will see directly coupled to the main pump using the drive shaft. This arrangement requires the charge pump to operate at the same angular velocity of the main pump. The charge pump itself is a fixed-displacement pump usually it is a gear pump or a gerotor pump that is used in conjunct conjunction with the relief valve for the purpose of providing the makeup flow to the main pump inlet.

The pressure setting on the relief valve the P c is typically about 2 MPa, while the high pressure side of the hydraulic circuit often operates at a pressure over 10 times this amount. The check valves shown in figure are used to block the high pressure side to the charge circuit.

This feature ensures that the low pressure side is always maintained at constant pressure given by P c, and prevents a pump cavitation during fast alteration of the swash plate angle alpha. After knowing the constructional details, let us we will move on load pressure analysis.

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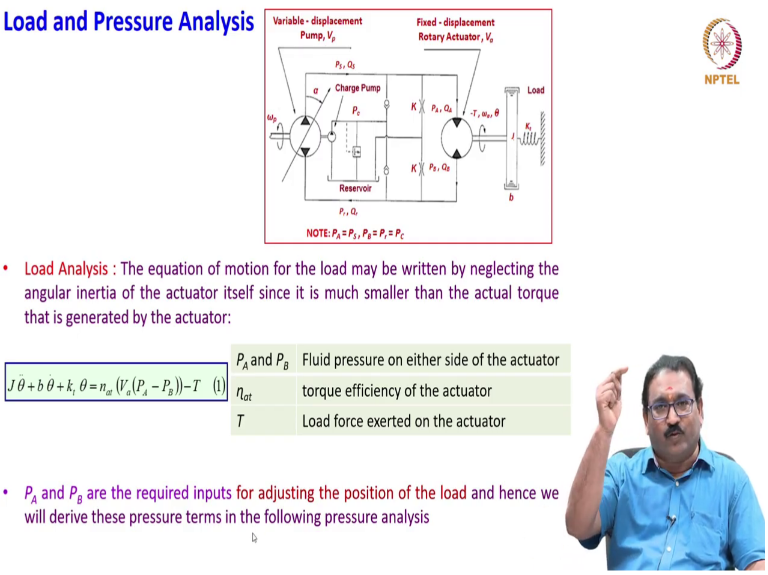

Let us we will see the load analysis. Similar to previous one, the equation of motion for the load may be written by neglecting the angular inertia of the actuator itself, since it is much smaller than the actual torque that is generated by the actuator.

Now, here J theta dot double dot here J theta double dot plus b theta dot plus k t into theta equal to. Now we will see the torque efficiency of the motor nu at multiplied by the pressure multiplied by the volumetric displacement of the actuator and minus T.

Here P A and P B are the required inputs for adjusting the position of the load and hence we will derive these pressure terms in the following pressure analysis as we have done for the linear actuator, similar way we have to do it here.

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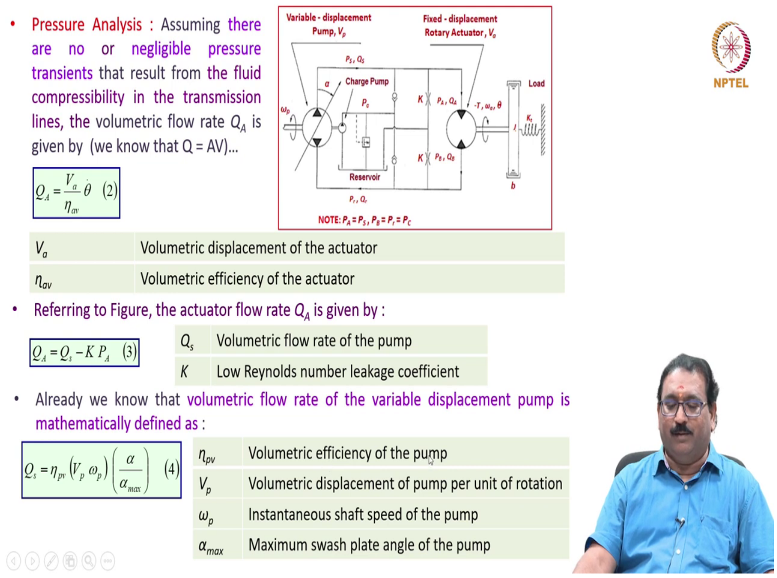

Pressure analysis, assuming there are no or negligible pressure transients that results from the fluid compressibility in the transmission lines. The volumetric flow rate Q A is given by already we know that Q A equal to AV – A into V. So, Q A equal to V a by nu a v into theta dot. Here V a is the volumetric displacement of the actuator; nu a v is the volumetric efficiency of the actuator.

Referring to the figure the actuator flow Q A is given by Q s minus K P A; Q s is a volumetric flow rate of the pump; K is the low Reynolds number leakage coefficient. Already we know that the volumetric flow rate of the variable-displacement pump is mathematically given by nu p v into V p into omega p, now you will see here friends alpha by alpha max.

Where alpha max is the maximum swash plate angle of the pump, omega p is a instantaneous shaft speed, and V p is the volumetric displacement of the pump per unit rotation, n p v is the similarly volumetric efficiency of the pump.

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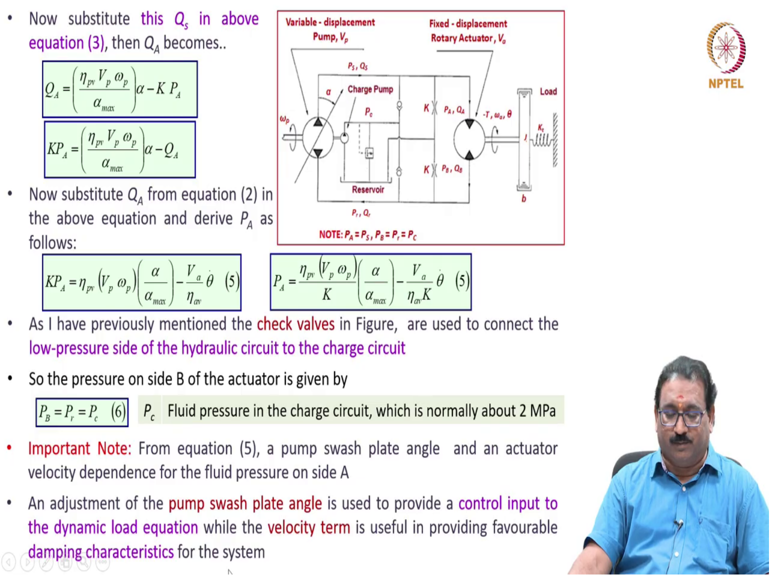

Now, substitute this Q s in the above equation 3, then Q A equal to I am getting this. Now, we will see what we will do, I will transfer minus K P A here because I want to find out P A then Q A will transfer here. Now, what we will do now next? Now, we will substitute the value of Q A from the equation 2 which gives us K P A equal to I am substituting here Q A. Now, what you will do now? P A equal to you will transfer this K here. This is a very important equation we can call it as equation 5.

As I have previously mentioned the check valves in the figure are used to connect the low pressure side of the hydraulic circuit to the charging circuit. So, the pressure on side B of the actuator is given by P B equal to pr that is equal to P c. The P c is a fluid pressure in the charge circuit which is normally 2 MPa.

Important note here, you will see here from the equation 5, the pumps swash plate angle and the actuator velocity depends for the fluid pressure on side A. An adjustment of the pump swash plate angle is used to provide a control input to the dynamic load equation. while the velocity term is useful in providing a favourable damping characteristics of the system.

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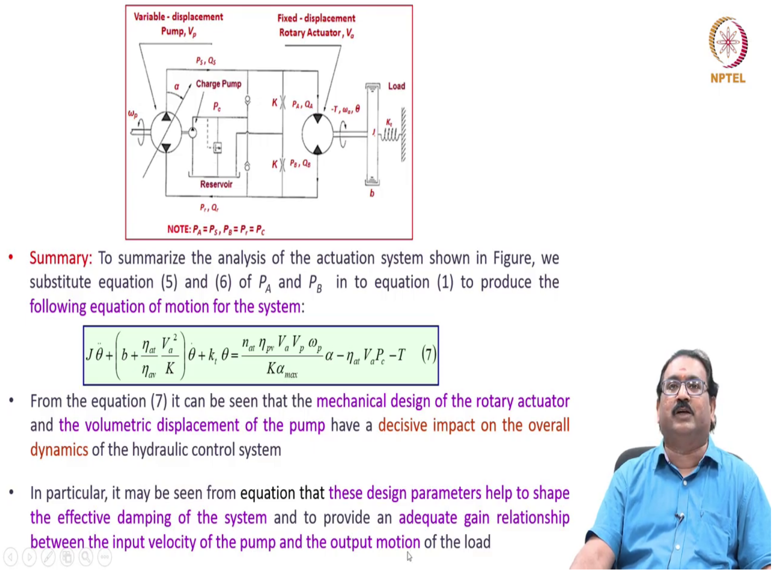

Quickly I will summarize this. To summarize the analysis of the actuation system shown in figure, we substitute the equation 5 and 6 for P A and P B in equation 1 to produce the equation of motion of the system. I am substituting now P A and P B in the equation number 1 which will gives us the complete equation.

From the equation 7, it can be seen that the mechanical design of the rotary actuator and the volumetric displacement of the pump have a decisive impact on the overall dynamics of the hydraulic control system. In particular, it may be seen from the equation that these design parameters help to shape the effective damping of the system and to provide an adequate gain relationship between the input velocity of the pump and the output motion of the load.

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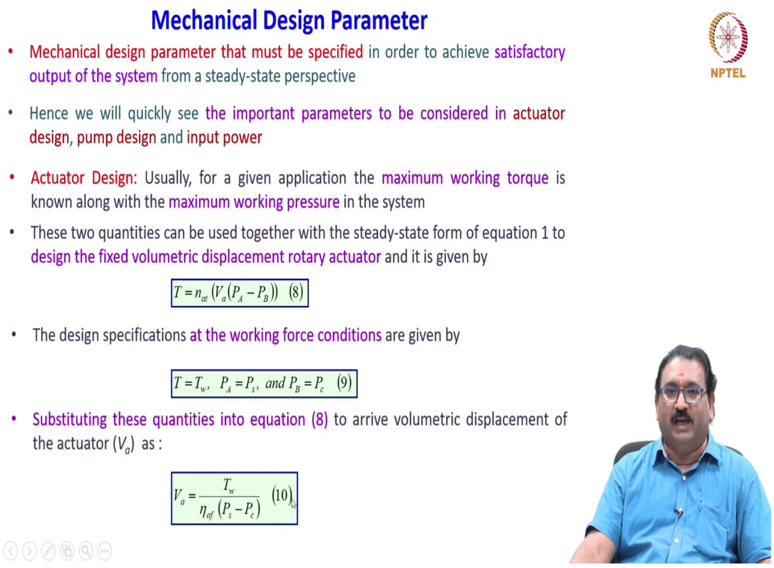

Now, quickly we will see the mechanical design parameters, mechanical design parameter that must be specified in order to achieve the satisfactory output of the system for a steady-state perspective. Hence we will quickly see the important parameter to be considered in actuator design, pump design and input power. Actuator design, usually, for a given application the maximum working torque is known along with the maximum working pressure of the system.

These two quantities can be used together with the steady-state form of equation 1 to design the fixed volumetric displacement rotary actuator. And it is given by the T equal to nu a t into V a P A minus P B. We can call it as equation 8.

The design specifications at the working force conditions are given by T equal to T w, P A equal to P s, and P B equal to p c – call it as equation number 9. Substitute these quantities into equation 8 to arrive the volumetric displacement of the actuator V a as T w by nu a f into P s minus P c, we can call it as equation number 10.

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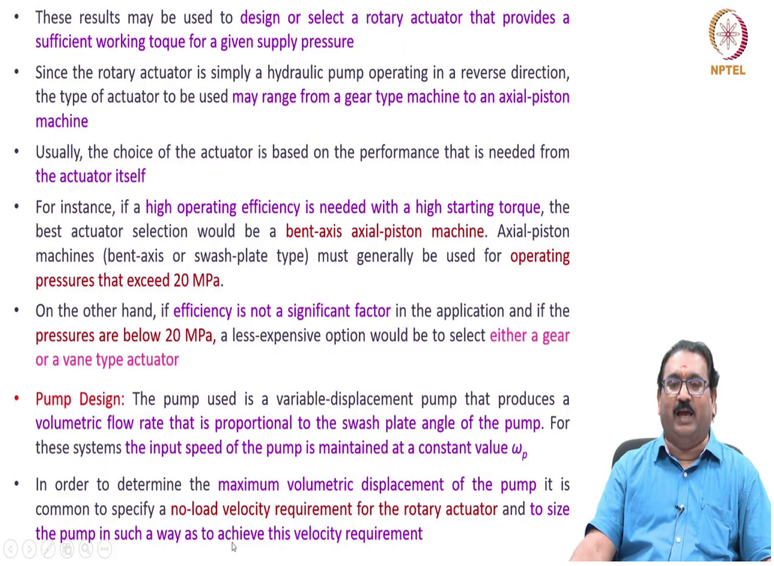

These results may be used to design or select a rotary actuator that provides a sufficient working torque for the given supply pressure. Since the rotary actuator is simply a hydraulic pump operating in a reverse direction, the type of actuator to be used may range from a gear type machine to an axial-piston machine. Usually, the choice of the actuator is based on the performance that is needed from an actuator itself.

For instance, if a high operating efficiency is needed with a high starting torque, the best actuator selection would be a bent-axis axial-piston machine. Axial-piston machine meaning a bent-axis or a swash plate type must generally be used for operating pressures that exceeds 20 Mega Pascal.

On the other hand, if the efficiency is not a significant factor in the application and if the pressures are below 20 Mega Pascal, a less-expensive option would be to select either a gear or a vane type actuator.

Pump design, the pump used is a variable-displacement pump that produces a volumetric flow rate that is proportional to the swash plate angle of the pump. For these system the input speed of the pump is maintained at a constant value omega p. In order to determine the maximum volumetric displacement of the pump, it is commonly to specify a no-load velocity requirement for the rotary actuator and to size the pump in such a way as to achieve this velocity requirement.

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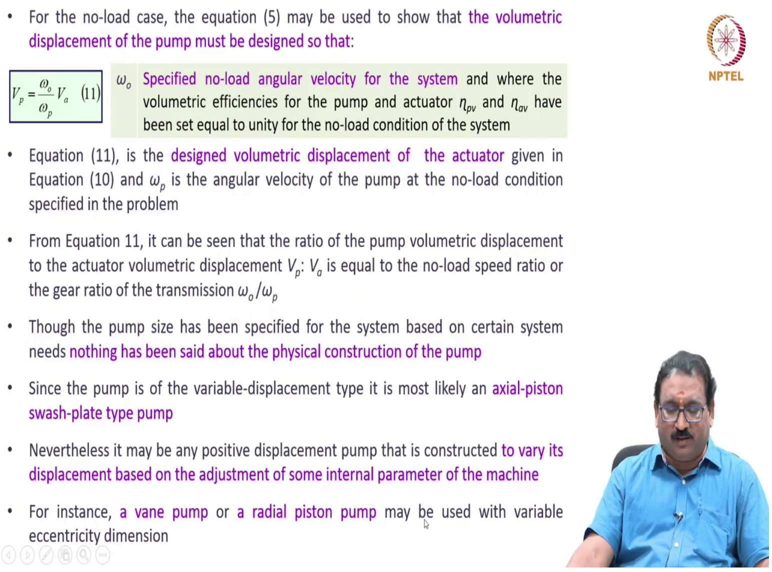

For the no-load case the equation 5 may be used to show that the volumetric displacement of the pump must be designed. So, that V p equal to omega naught by omega p into V a. Omega naught is specified no-load angular velocity of the system. And where the volumetric efficiency for the pump and actuator nu p v and nu a v have been set equal to unity for the no-load condition of the system.

Equation 11 is the designed volumetric displacement of the actuator given in equation 10. And omega p is the angular velocity of the pump at the no-load condition specified in the problem. From the equation 11, it can be seen that the ratio of the pump volumetric displacement to the actuator volumetric displacement that is a V p is to V a is equal to the no-load speed ratio or the gear ratio of the transmission omega naught by omega p.

Though the pump size has been specified for the system based on a certain system needs nothing has been specified about the physical construction of the pump. Since the pump is of the variable-displacement type it is most likely an axial-piston swash-plate type pump.

Nevertheless it may be any positive displacement pump that is constructed to vary its displacement based on the adjustment of some internal parameters of the machine. For instance, a variable pump or a radial piston pump may be used with variable eccentricity dimension.

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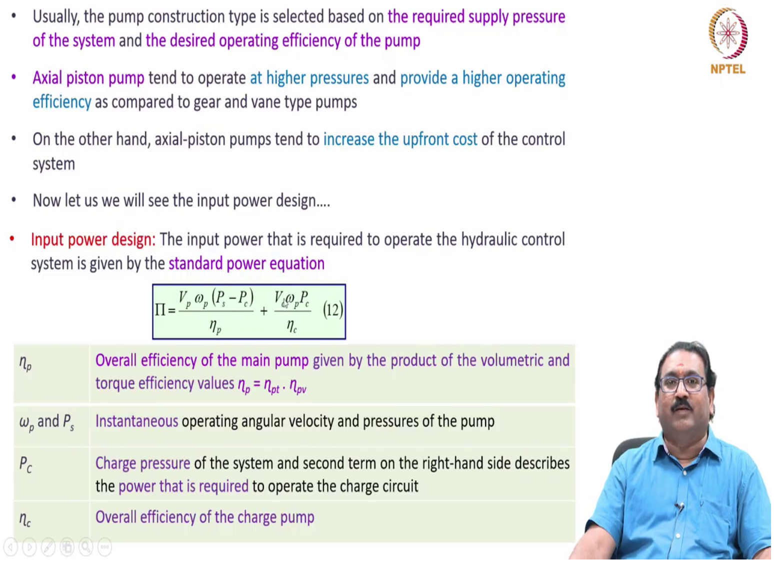

Usually, the pump construction type is selected based on the required supply pressure of the system and the desired operating efficiency of the pump. Axial-piston pump tend to operate at high pressures and provide a higher operating efficiency as compared to gear and a vane type pumps. On the other hand, axial-piston pumps tend to increase the upfront cost of the control system.

Now, let us we will see the input power design. The input power that is required to operate the hydraulic system is given by the standard power equation as pi equal to V p omega p P s minus P c by nu p plus V c omega p P c by nu c. Here nu p is a overall efficiency of the main pump which is given by nu p t into nu p v.

Omega p and P s, the instantaneous operating angular velocity and a pressure of the pump. P c is the charge pressure of the system. And the second term here is there no it is describes the power that is required to operate the charge circuit. Here nu c is the overall efficiency of the charge pump.

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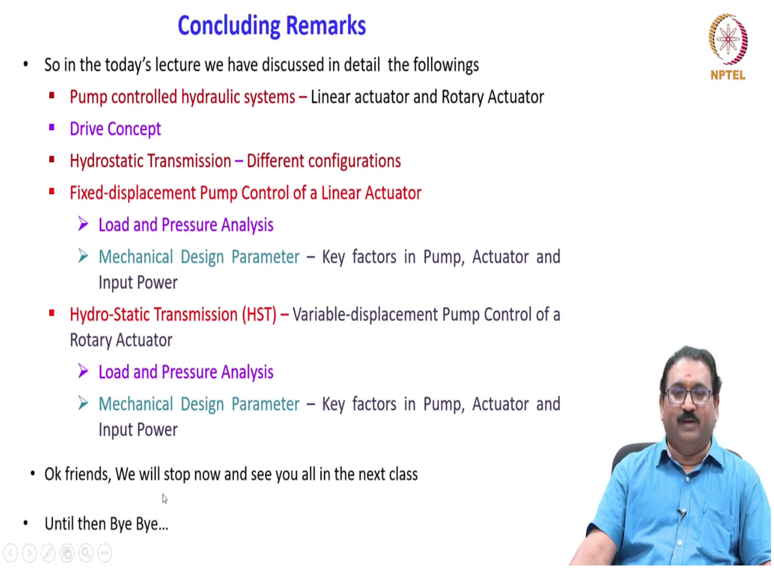

Now, let us we will conclude the today's lecture. So, in today's lecture, we have discussed in detail the followings. Pump controlled hydraulic system – here we have seen the linear actuator as well as rotary actuator. Different drive concepts, hydrostatic transmissions and a different configurations, fixed-displacement pump controlled linear actuator – here we have discussed in detail load and pressure analysis.

Then mechanical design parameter particularly on the pump, actuator and input power. Similarly, the hydrostatic transmission what we can call it is a variable-displacement pump controlled rotary actuator. Here also we will discussed already load and pressure analysis, mechanical design parameters. Ok friends, we will stop now and see you all in the next class. Until then bye bye.

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Thank you one and all for your kind attentions. [FL]