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Hydraulic servo systems

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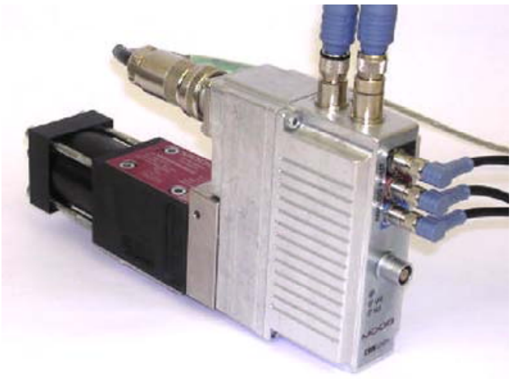
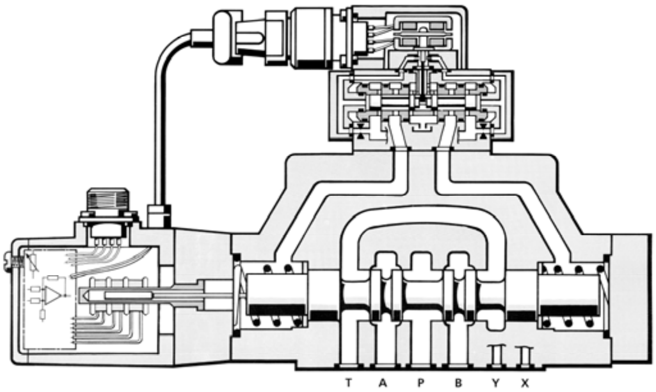

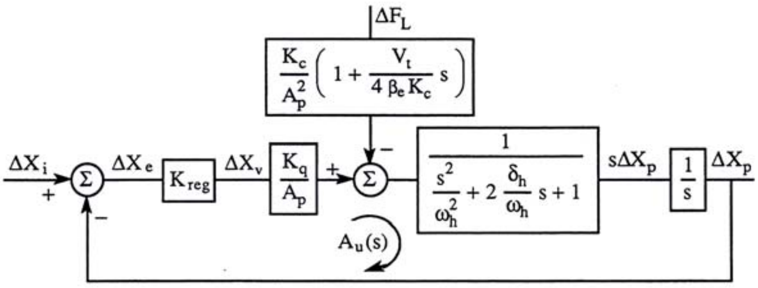
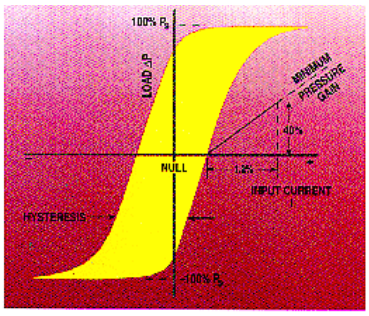


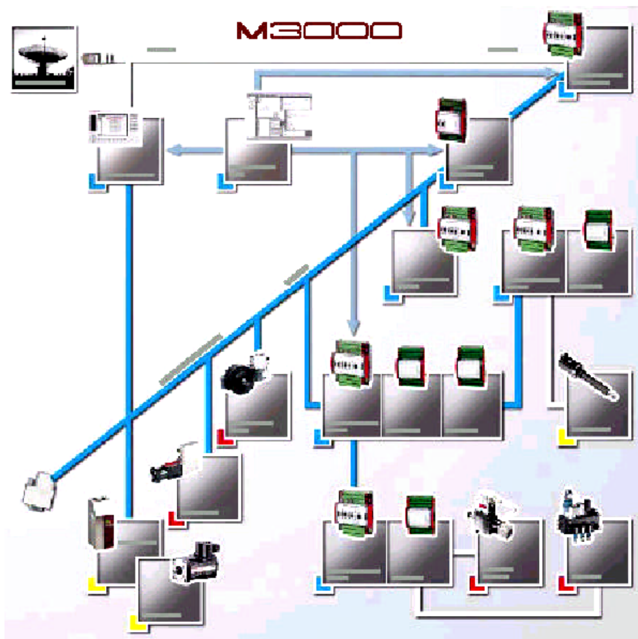

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Hydraulic Servo Systems – Theory and Applications

1. Introduction

When closed-loop hydraulic control systems first began to appear in industry, the applications were generally those in which very high performance was required. While hydraulic servo systems are still heavily used in high-performance applications such as the machine-tool industry, they are beginning to gain wide acceptance in a variety of industries. Examples are material handling, mobile equipment, plastics, steel plants, mining, oil exploration and automotive testing.

Closed loop servo drive technology is increasingly becoming the norm in machine automation, where the operators are demanding greater precision, faster operation and simpler adjustment. There is also an expectation that the price of increasing the level of automation should be contained within acceptable limits.

What is a servo?

In its simplest form a servo or a servomechanism is a control system which measures its own output and forces the output to quickly and accurately follow a command signal, se **Figure 1-1**. In this way, the effect of anomalies in the control device itself and in the load can be minimised as well as the influence of external disturbances. A servomechanism can be designed to control almost any physical quantities, e.g. motion, force, pressure, temperature, electrical voltage or current.

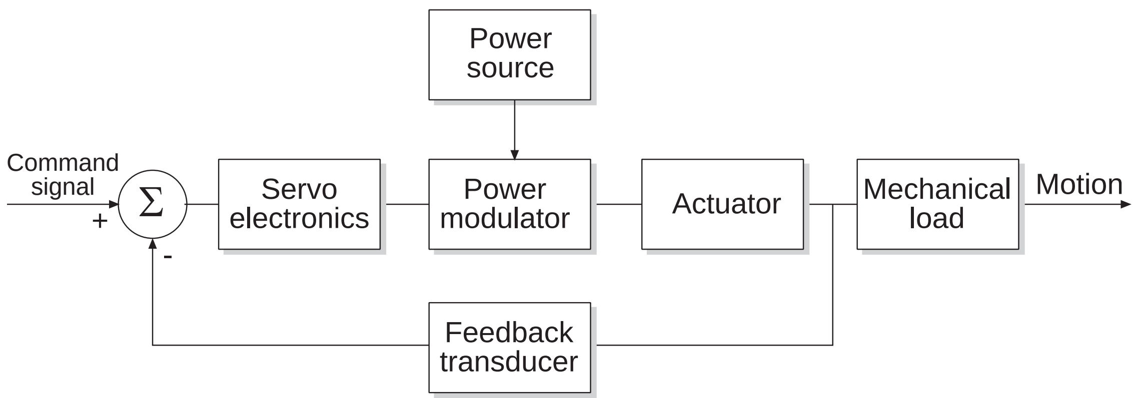

Figure 1-1: Basic servomechanism

Technology comparisons

The potential for alternative technologies should be assessed in the light of the well-known capabilities of electro-pneumatic and electro-mechanical servos. High performance actuation system is characterised by wide bandwidth frequency response, low resolution and high stiffness. Additional requirements may include demanding duty cycles and minimisation of size and weight. The last mentioned requirements are of special interests in aerospace applications. The most important selection criteria can be summarised as follows:

- *Customer performance*
- *Cost*
- *Size and weight*
- *Duty cycle*
- *Environment*: vibration, shock, temperature, etc.

The performance available with electro-hydraulic servos encompasses every industrial and aerospace application. As indicated in **Figure 1-2** electro-hydraulic servos will cover applications with higher performance then electro-mechanical and electro-pneumatic servos. This is easily explained because electro-hydraulic servo systems have been designed and developed to accomplish essentially every task that has appeared.

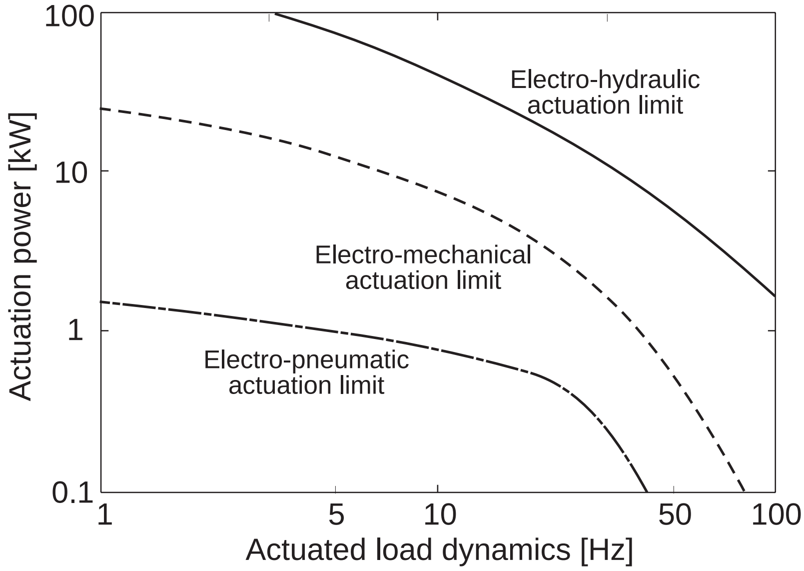

Figure 1-2: Typical performance characteristics for different types of servo actuators

The above figure indicates that applications in the lower range of power and dynamic response may also be satisfied with electro-pneumatic servos. However, the best choice is always determined by considerations, such as those selection criteria discussed above. In most applications the aspect of cost is generally dominant.

Experience indicates that electro-mechanical or electro-pneumatic actuators tends to have lower cost than electro-hydraulic actuators in the low performance range. This cost difference rapidly dissipates for applications that require high power and/or high dynamic response.

In comparing costs, one must be careful to consider the total cost of entire servo-actuation system. The higher cost of an electro-hydraulic servo often results from the power conversion equipment needed to provide high pressure fluid with low contamination level. It is also clear that the relative cost of an alternative actuation system designed for a specific application will depend, primarily, on the actuation power level.

Capabilities of electro-hydraulic servos

When rapid and precise control of sizeable loads is required an electro-hydraulic servo is often the best approach to the problem. Generally speaking, the hydraulic servo actuator provides fast response, high force and short stroke characteristics. The main advantages of hydraulic components are.

- *Easy and accurate control of work table position and velocity*
- *Good stiffness characteristics*
- *Zero backlash*
- *Rapid response to change in speed or direction*
- *Low rate of wear*

There are several significant advantages of hydraulic servo drives over electric motor drives:

- ♦ Hydraulic drives have substantially higher power to weight ratios resulting in higher machine frame resonant frequencies for a given power level.
- ♦ Hydraulic actuators are stiffer than electric drives, resulting in higher loop gain capability, greater accuracy and better frequency response.
- ♦ Hydraulic servos give smoother performance at low speeds and have a wide speed range without special control circuits.
- ♦ Hydraulic systems are to a great extent self-cooling and can be operated in stall condition indefinitely without damage.
- ♦ Both hydraulic and electric drives are very reliable provided that maintenance is followed.
- ♦ Hydraulic servos are usually less expensive for system above several horsepower, especially if the hydraulic power supply is shared between several actuators.

Different electro-hydraulic concepts

In electro-hydraulic applications different concepts will be used in order to meet the actual requirements. One example of a system where the weight is of great importance is an Electro Hydraulic Actuator (EHA) to be used in aircraft applications. A typical EHA-concept is shown in **Figure 1-3**. This EHA consists of an electric motor, a speed controlled pump with low displacement, a cylinder and an accumulator used as a tank. In a real application there is also a need for additional functions, such as by-pass damper and safety facilities, not shown in the figure. In order to save weight no cooler is applied and there will be a risk for too high fluid temperature with failure of the EHA as a consequence. Therefore, the losses and thereby the temperature of the fluid is of great importance in this application.

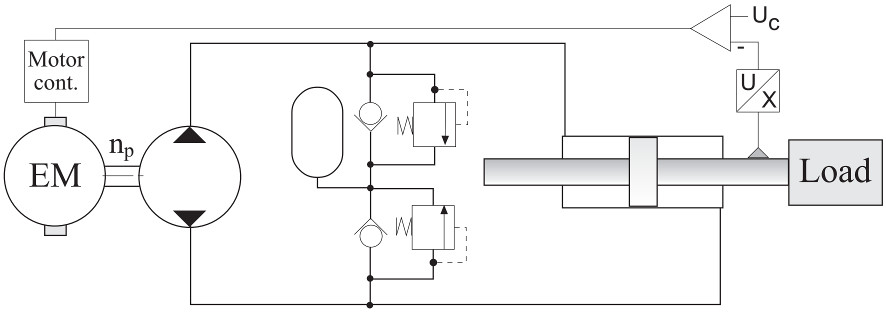

Figure 1-3: Electro Hydraulic Actuator with electric motor control

Other more conventional concepts are a pump or a valve controlled actuator, as shown in **Figure 1-4**. The main difference between those systems is that the pump controlled system is one separate unit supplied by an electric wire to the motor (the same as for the system in Figure 1-3) and the valve controlled actuator is supplied by a constant pressure hydraulic line. In the last case the EHA is feed by a central hydraulic supply unit.

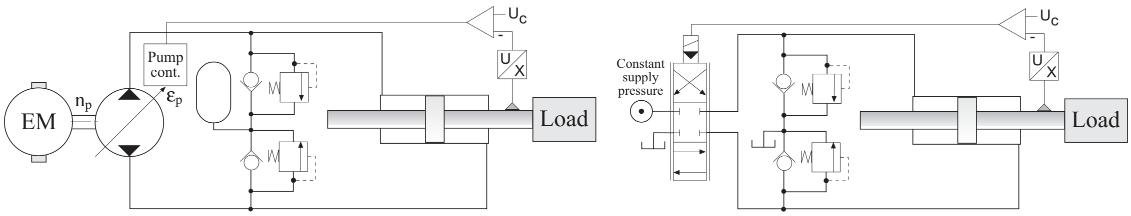

Figure 1-4: Electro Hydraulic Actuators with pump and valve control respectively

Comparing the electric motor controlled EHA in Figure 1-3 and the pump controlled EHA in Figure 1-4, the overall efficiency will be better in the first case, where the pump shaft speed (n_p) is controlled. The efficiency curves for similar systems are shown in **Figure 1-5**. Maximum pump flow is the same in both cases and pressure drop over the directional valve is included as losses.

Looking at the variations in overall efficiency it is clear that speed control has a favour over displacement control, especially in the power range up to 50% of maximum power. However, there are other problems to overcome in the pump speed control concept. For example, the amplitude of the flow pulsations from the pump must be very low at low shaft speeds in order to avoid problems with low frequency vibrations in the system. This, require a special design of the pump. One suitable pump design is the inner gear concept. In such a pump both kinematic and compressibility dependent flow pulsations are extremely low compared to piston pumps.

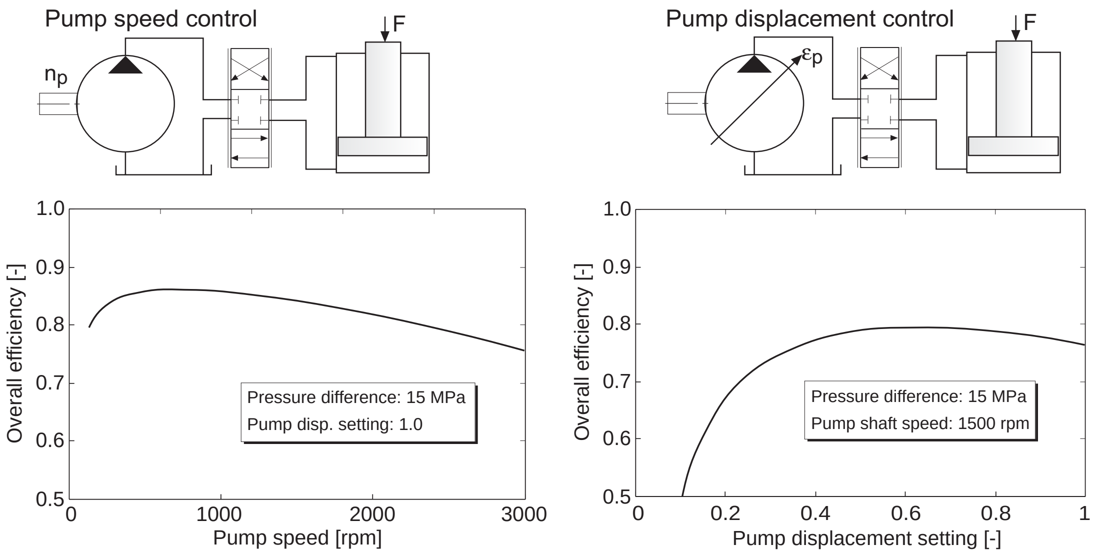

Figure 1-5: Overall efficiencies with pump speed control and displacement setting control

2. Configuration of an electro-hydraulic servo

The basic elements of an electro-hydraulic servo is shown in **Figure 2-1**. The output of the servo is measured with a transducer device to convert it to an electric signal. This feedback signal is compared with the command signal. The resulting error signal is then amplified by the regulator and the electric power amplifier and then used as an input control signal to the servo valve. The servo valve controls the fluid flow to the actuator in proportion to the drive current from the amplifier. The actuator then forces the load to move. Thus, a change in the command signal generates an error signal, which causes the load to move in an attempt to zero the error signal. If the amplifier gain is high, the output will vary rapidly and accurately following the command signal.

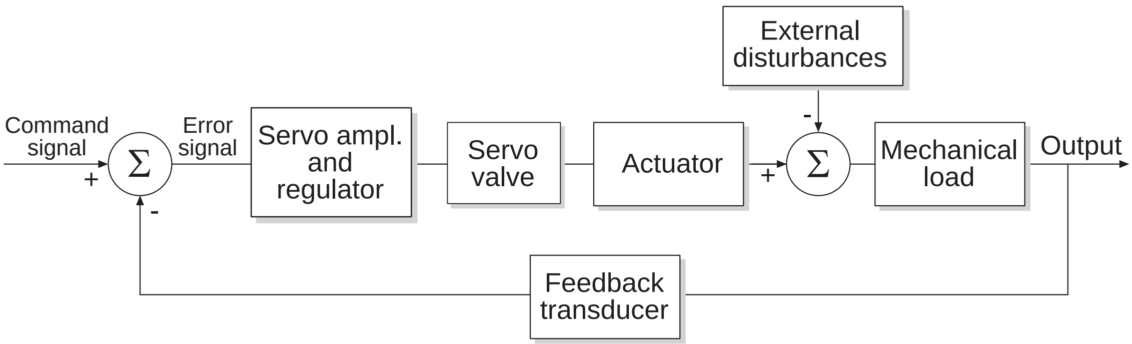

Figure 2-1: Components in an electro-hydraulic servomechanism

External disturbances (forces or torque) can cause the load to move without any changes in the command signal. In order to offset the disturbance input an actuator output is needed in the opposite direction (see Figure 2-1). To provide this opposing output a finite error signal is required. The magnitude of the required error signal is minimised if the amplifier gain is high. Ideally, the amplifier gain would be set high enough that the accuracy of the servo becomes dependent only upon the accuracy of the transducer itself. However, since the control loop gain is proportional to the amplifier gain, this gain is limited by stability considerations. In some applications, stability may be critical enough that the desired performance is not possible to reach.

The three common types of electro-hydraulic servos are:

- *Position servo* (linear or angular)
- *Velocity or speed servo* (linear or angular)
- *Force or torque servo*

Position servo

Probably the most basic closed-loop control system is a position servo. A schematic diagram of a complete position servo is shown in **Figure 2-2**.

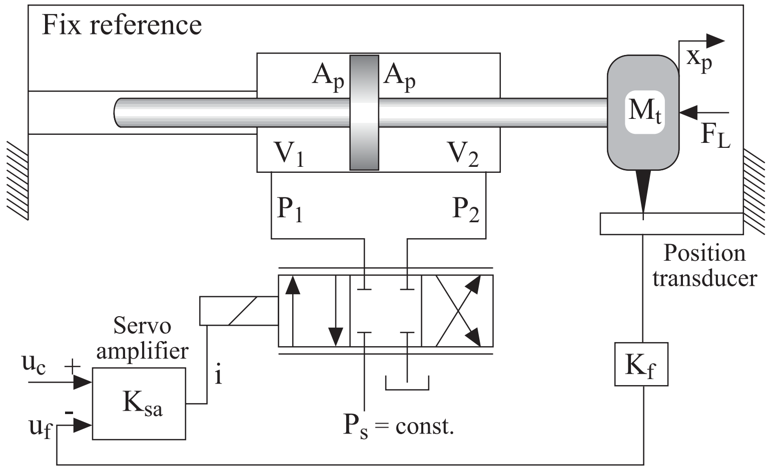

Figure 2-2: Symbol circuit of a position servo

In Figure 2-2 the actuator or load position is measured by a position transducer, which gives an electric signal (u_f) in voltage as an output. The servo amplifier compares the command signal (u_c) in voltage with the feedback signal (u_f). Then, the resulting error signal are gained the with the factor K_{sa}. The output current signal (i) from the amplifier will control the servo valve.

Velocity and force servos

Another common types of closed loop control systems are velocity (speed) and force (torque) servos. The configuration of these systems are identical to the position servo depicted in Figure 2-2, expect that the transducer measures velocity or force instead of position and that the controller may have different characteristics. **Figure 2-3** shows both a speed and a force servo. It is notable that the same type of servo valve can be used in all of these applications. As indicated in Figure 2-3, velocity or speed servos are more commonly used to control the shaft speed of an hydraulic motor than to control linear velocity.

In the velocity servo the servo amplifier is of integrating type, as shown in Figure 2-3. Compared to a position servo the velocity servo has no integration between servo valve displacement and the output velocity. Therefore, the integration in a velocity servo is generally provided electronically in the amplifier. The integration is desirable to minimise static errors and to maintain stability.

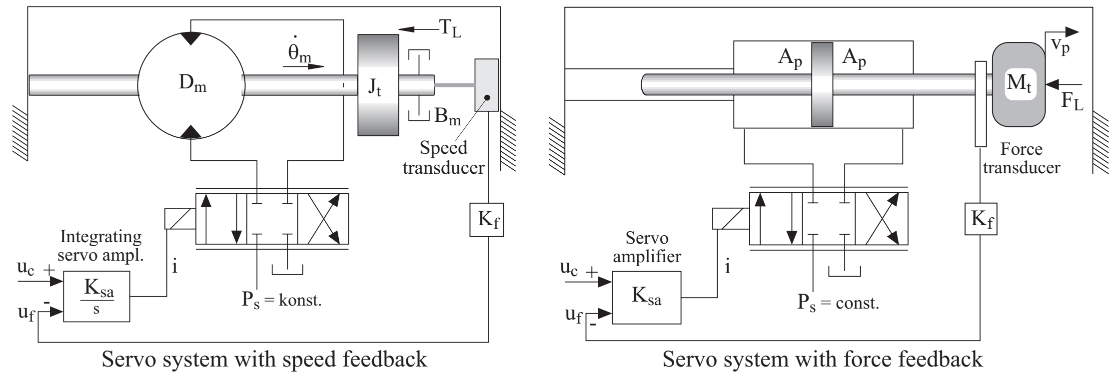

Figure 2-3: Symbol circuit of a speed and a force servo

In a real force servo the transducer measures the output force and this signal is fed back to the amplifier. A more simpler way to implement a force servo is to use the load pressure in the actuator as a feedback signal. This is quite close to a true force servo except from the friction force in the actuator.

3. Servo valves and their characteristics

The heart of the hydraulic servo system is the servo valve and it is essential that its characteristics be thoroughly understood. A servo valve is a component which work as an interface between an electrical (or mechanical) input signal and the hydraulic power represented by the product of flow and pressure. Depending of the application there are different types of servo valves to use.

3.1 Number of lands and ports

The most widely used valve is the sliding valve employing spool type construction. Typical spool valve configurations are shown in Figure 3-1. As explained in the figure, spool valves can be classified by the numbers of ways the flow can enter and leave the valve and the number of lands. Because all valves require a supply, a return and at least one line to the load, valves are either of three-port or four-port type.

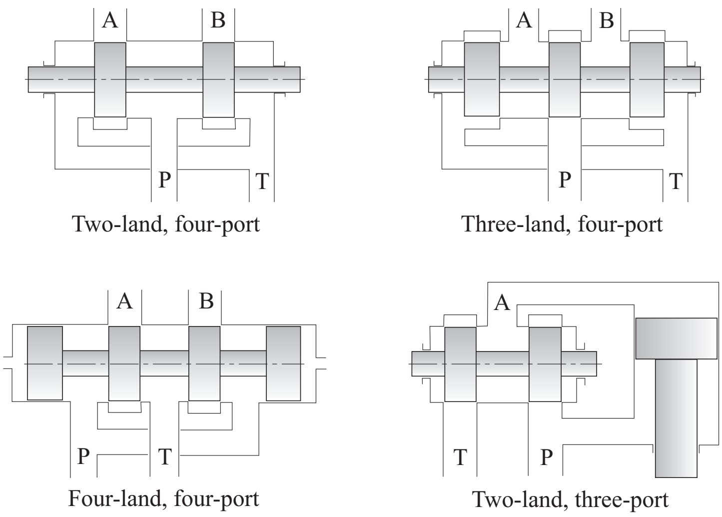

Figure 3-1: Typical confuguration of the main stage of hydraulic servo valves

The four-port valves in Figure 3-1 have two, three and four lands. For mechanical positioning of the main spool, two or three lands can be used. With four (or at least three) lands on the spool it is possible to use hydraulic pressure for positioning of a four-port valve. This is the most common concept for high response servo valves. Special valves may have more than four lands.

A three-port valve, shown down to the right in Figure 3-1, requires a bias pressure acting on one side of an unsymmetrical cylinder for direction reversal. Usually the head-side piston area is twice the rod-side area and supply pressure acts on the smaller area to provide the bias force for reversal.

3.2 Types of valve center

The type of valve center are defined by the width of the land compared to the width of the port in the valve sleeve when the spool is in neutral position. If the width of the land is smaller than the port, the valve is said to have an ***open-center*** or to be ***under-lapped***, as shown in **Figure 3-2**. A ***critical-center*** or ***zero-lapped*** valve has a land width identical to the port width. A valve with a land width greater than the port width is called ***closed-center*** or ***over-lapped***.

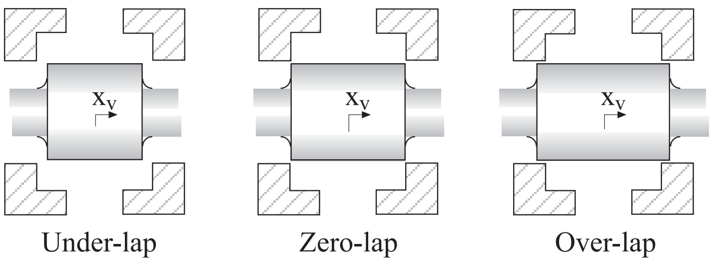

Figure 3-2: Different valve lapping when the spool is in neutral position

The flow characteristics of the valve may be directly related to the type of valve center. Corresponding to Figure 3-2 there are three important flow gain characteristics, with the shape shown in **Figure 3-3**. In fact, it is better to define the type of valve center from the shape of the flow gain near neutral position than from geometrical considerations. A critical center valve may be defined as the geometrical fit required to achieve a linear flow gain in the vicinity of neutral position, which usually necessitates a slight overlap to offset the effect of radial clearance.

A majority of four-way servo valves are manufactured with a critical center because of the emphasis on the linear flow gain. Closed center valves are not desirable because of the dead-band characteristics in the flow gain. With a proportional amplifier the dead-band results in steady state error and can cause backlash which may lead to stability problems in the servo loop. It is possible to compensate for dead-band electronically but it will at least influence the response time of the servo valve.

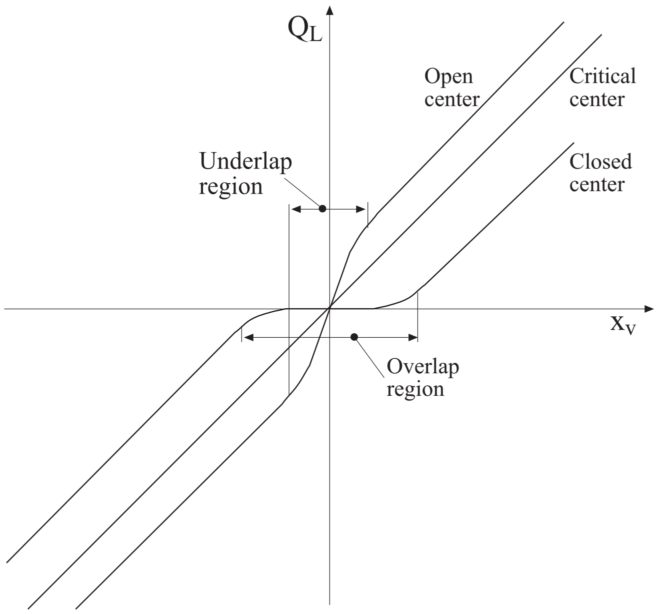

Figure 3-3: Flow gain (load flow, Q_L versus spool stroke, x_v) of different center types

Open center valves are used in applications which require a continuous flow to maintain an acceptable fluid temperature and/or an increase of the hydraulic damping. However, the large power loss in neutral position, the decrease in flow gain outside the under-lap region and the decreased pressure sensitivity of open center valves restrict their use to special applications.

Valve sleeve

Since the flow characteristics of a servo valve is of great importance in a servo system the valve must be manufactured with high precision. That close and matching tolerances for spool lands, ports and radial clearances must also be held as constant as possible during operating conditions. In order to compensate for the influence from pressure and temperature the spool is working in a sleeve (bushing), which is surrounded by the valve housing. The sleeve or bushing, shown in **Figure 3-4**, is pressurised both inside and outside and therefore, the pressure will not influence the radial clearances.

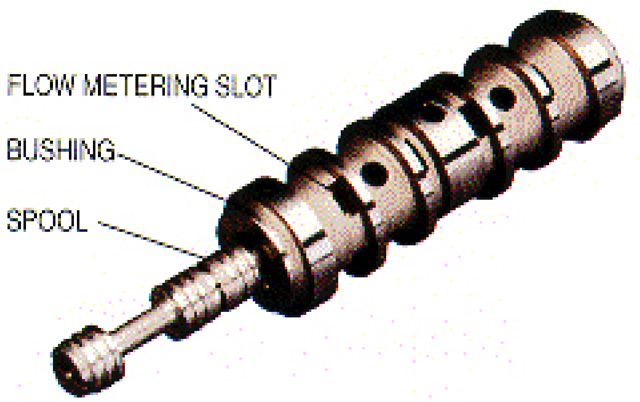

Figure 3-4: Main spool and bushing of a servo valve

In Figure 3-4 it can also be seen that the port in the bushing is formed as a slot (flow metering slot). Each port will include at least two or up to four slots arranged symmetrically around the bushing. This means that the flow metering area, for a spool with the diameter, d_s and the spool displacement, x_v will be calculated as,

$$A(x_v) = f_{sc} \cdot \pi \cdot d_s \cdot x_v$$

where f_{sc} is the fraction of spool circumference that is opened to the slots. Normally, this parameter will be in the following interval: $0.25 \le f_{sc} \le 0.5$.

3.3 Examples of electro-hydraulic servo valves

The electro-hydraulic servo valve connects the electronic and hydro-mechanical portions of a hydraulic system. Such a valve has electric current as input signal. This electric signal is then transformed proportionally, by different types of feedback loops, to a mechanical or a hydraulic signal.

Type of feedback

The shape of the steady state flow-pressure curves of a servo valve are given from the type of feedback used in the valve. Three types of feedback can be identified, which are *spool position*, *load pressure* and *load flow feedback*. In an ordinary flow direction controlled servo valve, which is commonly used in position, velocity and force servos, the main spool position is proportional to the input signal. The position feedback used in this case can be realised in different ways, such as direct mechanical feedback, force feedback or electrical position feedback. In force feedback servo valves the main spool position is converted to a force by a spring and this force is balanced at the torque motor armature against the torque due to the input current. With pressure compensation or pressure feedback the load flow or load pressure can be maintained to varying proportionally to the input signal. However, this types of valves will be best suited for special applications such as constant flow or constant pressure control. The pressure-flow characteristics of the mentioned valves is illustrated in **Figure 3-5**.

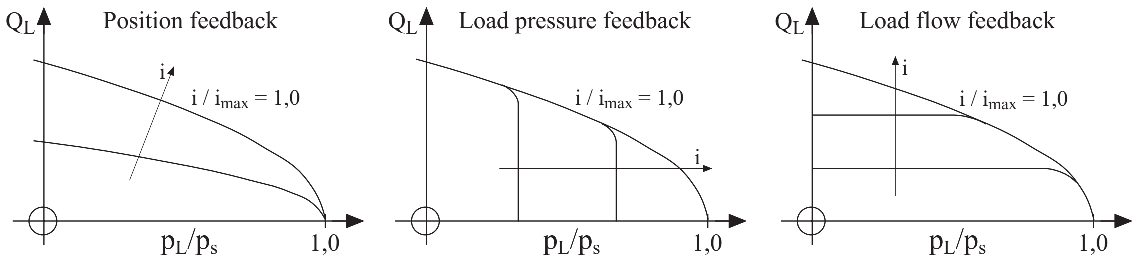

Figure 3-5: Pressure-flow curves of a servo valve using position, load pressure and load flow feedback.

There are also servo valves where different characteristics are combined. One of these valves is called the *dynamic pressure feedback servo valve*. This valve has a characteristic of a position feedback valve at low frequencies and characteristics of a load pressure feedback valve at higher frequencies. This behaviour is useful in some applications to increase the damping of valve-actuator combinations.

Number of stages

Servo valves may also be broadly classified as either *single-stage*, *two-stage* or *three-stage*. Single-stage servo valves consist of a torque motor or a linear force motor directly attached for positioning of the spool. Because torque or force motors have limited power capability, this in turn limits the hydraulic power capacity of single-stage servo valves. In some applications the single-stage concept may also lead to stability problems. This is the case if the flow forces acting on the spool are close to the force produced by the electro-magnetic motor. Flow forces are proportional to the flow and the square root of the valve pressure drop, which gives a limitation in hydraulic power.

Single-stage valves

A single-stage servo valve with a linear force motor is shown in **Figure 3-6**. The valve illustrated in the figure is a valve, which employs just one linear force motor (proportional magnet) to move the spool either side of the central position. The electric signal from a position transducer is then used for closed loop control of the spool position. In addition, the spool has a "power off" position whereby when no power is applied to the magnet, the bias spring pushes the spool fully over to the right side. I "power off" position all ports (A, B, P and T) are closed. In normal operation the spool will operate either side the null position but in the event of a power failure or machine shut-down, the spool will move to the "power off" position. The maximum pressure for this type of valves is normally 350 bar and the maximum flow is less than 80 litre/min.

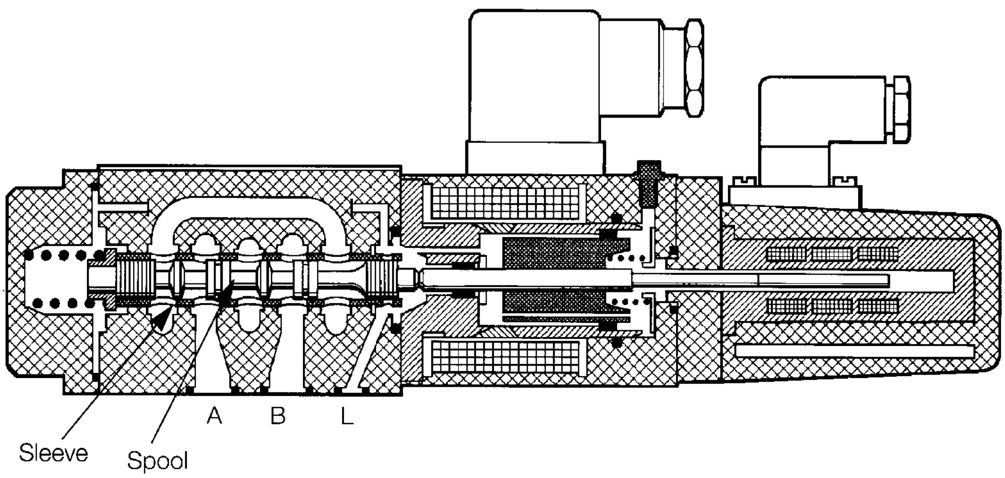

Figure 3-6: Single-stage servo valve with electrical position feedback, Vickers

Figure 3-7 shows a another single stage servo valve with a linear force motor which can actively stroke the spool from its spring centred position in both directions. This is an advantage compared with proportional solenoids with one force direction only, as in figure 3-6. The closed loop spool position electronics and pulse width modulated (PWM) drive electronics are integrated into the valve. This permits control directly from, for example, a machine control without the use of additional interface electronics.

The valve in Figure 3-7 has a quite strong force motor. High spring stiffness and resulting centring force plus external forces (flow forces and friction forces) must be overcome during out-stroking. During backstroking to centre position the spring force adds to the motor force and provides additional spool driving force which makes the drive very less contamination sensitive. The relatively high force from the force motor also means that the influence from flow forces on the spool position control is very small. This is important to avoid reduction of the hydraulic damping caused by dominant flow forces.

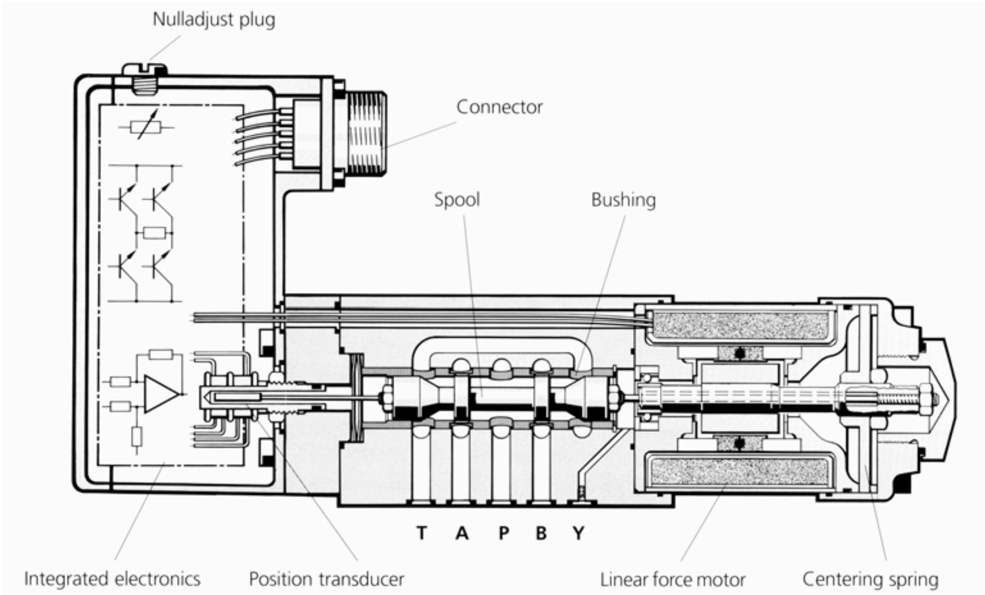

Figure 3-7: Single-stage servo valve with electrical position feedback, MOOG Direct Drive Valve (DDV)

Two-stage valves

One of the most common type of servo valve is the two stages one. The servo valve shown in **Figure 3-8** use an electrical torque motor, a double-nozzle pilot stage and a sliding spool second stage. Electrical current in the torque motor gives proportional displacement of the second stage spool. The flapper in the pilot stage attaches to the centre of the armature and extends down, inside the flexure tube. A nozzle is located on each side of the flapper so that flapper motion varies the nozzle openings. Differential pressures caused by flapper movement between the nozzle are applied to the ends of the valve main spool.

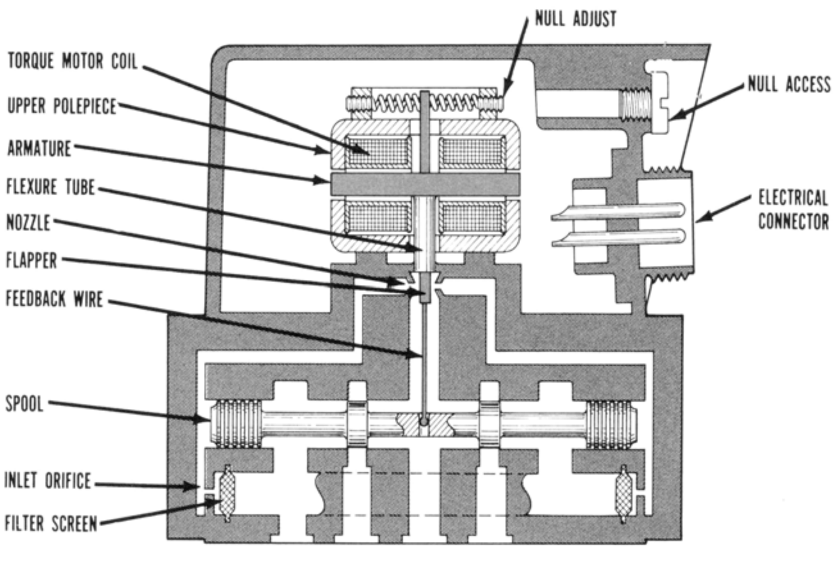

Figure 3-8: Two-stage servo valve (MOOG)

Operation: Electrical current in the torque motor coils causes either clockwise or counter-clockwise torque, as shown in **Figure 3-9**, on the armature. This torque displaces the flapper between the two nozzles. The differential nozzle flow moves the spool to either the right or left. The spool continues to move until the feedback torque counteracts the electromagnetic torque. At this point the armature/flapper is returned to centre, so the spool stops and remains displaced until the electric input changes to a new level.

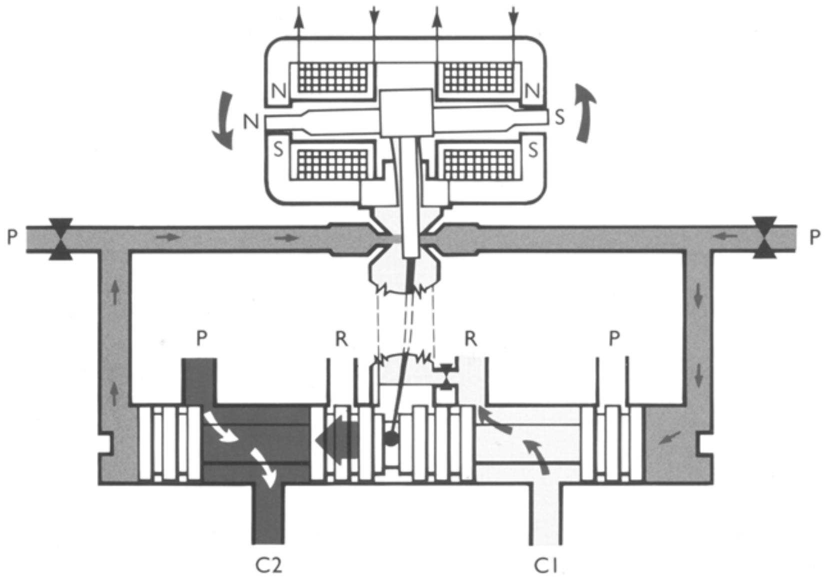

Figure 3-9: Valve responding to change in electrical input for a two-stage servo valve, MOOG

Instead of flapper nozzle pilot stage a jet pipe stage can be used as illustrated in **Figure 3-10**. The "servojet" consists mainly of a torque motor, jet pipe and receiver. A current through the coil displaces the jet pipe from neutral position.

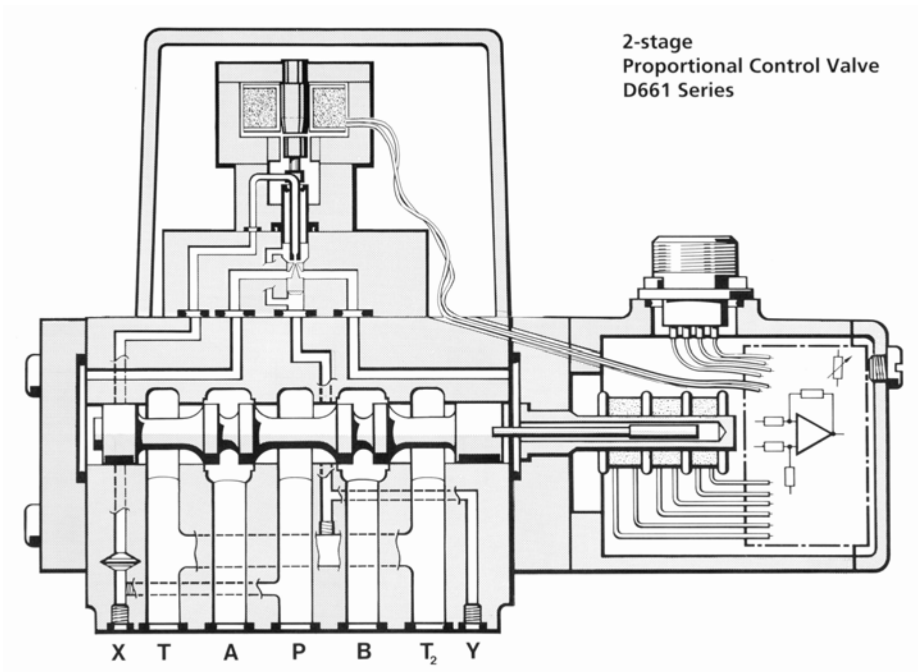

Figure 3-10: Two-stage servo valve with jetpipe pilot stage, MOOG

This displacement combined with the special shape of the nozzle direct focussed fluid jet more into one receiver than the other. The jet now produces a pressure difference in the control ports. This pressure difference results in a pilot flow, which in turn causes a spool displacement. One advantage for the jet pipe pilot valve is the less sensitivity for contamination than for example a flapper nozzle valve.

The position control loop for the main stage spool is closed by the integrated electronics. An electrical command signal is applied to the integrated position controller which drives the valve coil. A position transducer measures the position of the main spool. This signal is then fed back to the controller where it is compared with the command signal. The controller drives the pilot valve until the error between the command and feedback signal is zero. Thus the position of the main spool is proportional to the electrical command signal.

Three-stage valves

For high flow capacity the required power to drive the main spool will be high. In such applications (flow capacity over 150 litre/min and maximum pressure about 350 bar) a three-stage valve will be used. Three-stage means that a two-stage servo valve is used as a pilot valve for the main stage, just as shown in **Figure 3-11**.

This valve have an electrical feedback for the main spool position control. In the controller this signal is compared with the command signal. The controller drives the pilot valve, in this case a two-stage servo valve, until the error between the command and feedback signal is zero. In the same way as for the foregoing valves presented, the position of the main spool will be proportional to the electrical command signal.

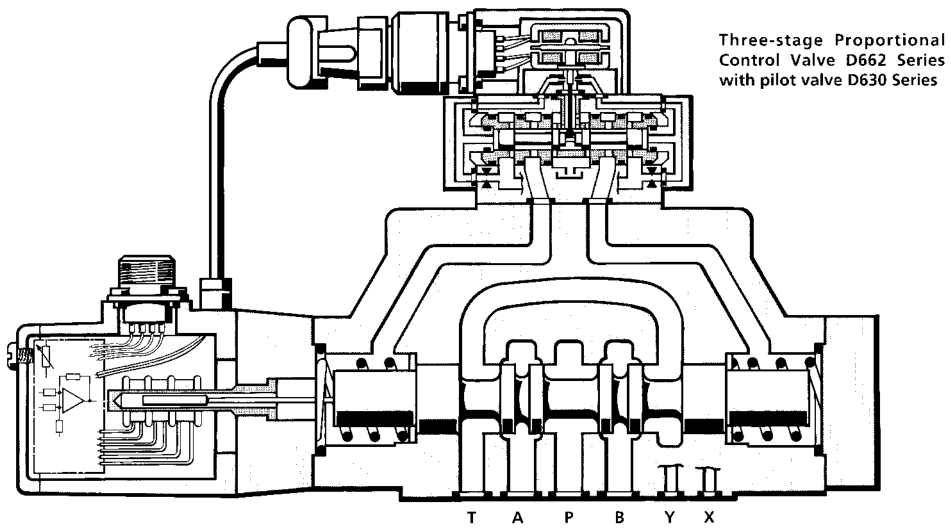

Figure 3-11: Three-stage servo valve with electrical feedback for main spool position control

It is also notable that the valves in the Figures 3-10 and 3-11 have no sleeve for the main spool. Therefore, these valves are often called proportional valves instead of servo valves. Today, the main difference between proportional and servo valves are just the sleeve. A proportional valve is not manufactured by such precision as a servo valve and therefore also less expensive and mainly used for open loop control. However, the valves shown above have a quite linear characteristics through zero position of the main spool and are well suited for closed loop control as ordinary servo valves.

P/Q-valve
In some applications there is a need for both pressure and flow control. Velocity control can be required for a part of the working cycle and pressure control for another part of the cycle. Other requirements can be pressure limiting during velocity control. A valve capable of handling these requirements is shown in **Figure 3-12**.

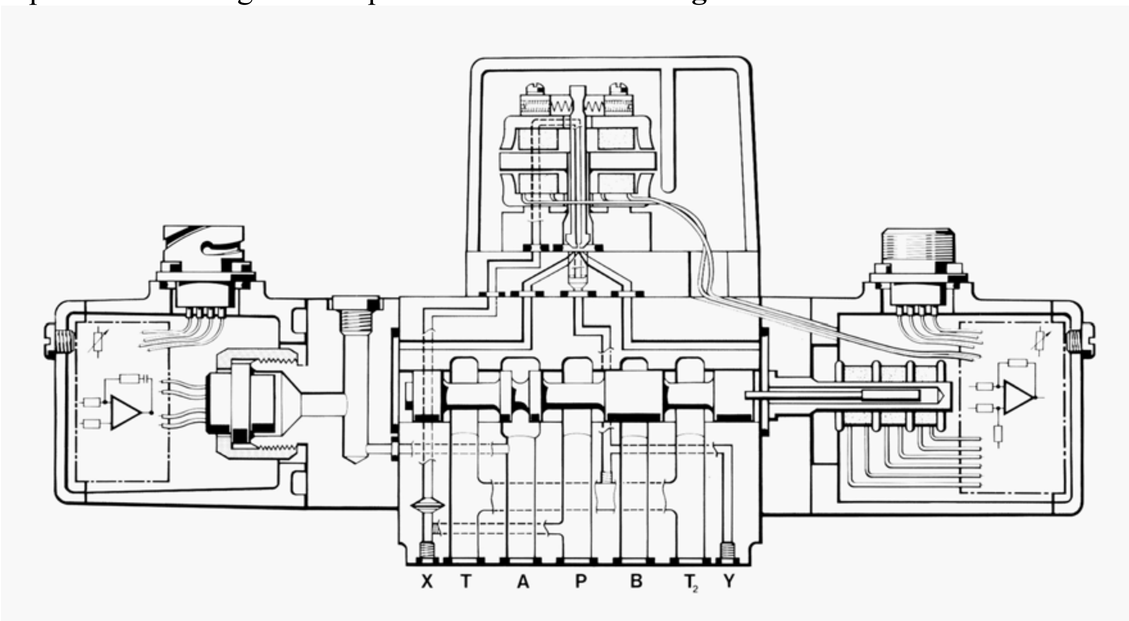

Figure 3-12: MOOG PQ-proportional control valve for pressure and flow control

3.4 General steady state valve characteristics

The design of a servo valve depends upon the requirements on hydraulic power capacity, accuracy and dynamic response. The valve characteristics will also depends on the port lapping and the type of internal feedback control loops in the valve.

As mentioned before, the most common type of servo valve is a four-port critical-center valve with spool position control. The main stage of this type of valve is schematically shown in **Figure 3-13**. This valve is a devices that use mechanical motion to control the hydraulic power from a source to an actuator.

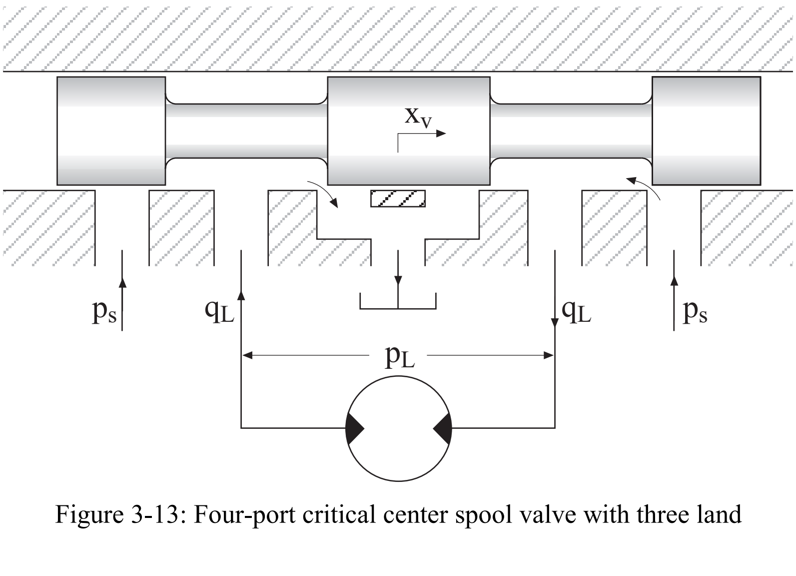

Figure 3-13: Four-port critical center spool valve with three land

The load flow through the valve is expressed by the equation

$$q_L = C_q w x_v \sqrt{\frac{1}{\rho}\left(p_s - \frac{x_v}{|x_v|} p_L\right)} \qquad (3\text{-}1)$$

where, C_q = flow coefficient, [-] $p_L = p_1 - p_2$ = load pressure dif., [Pa]
p_s = supply pressure, [Pa] x_v = spool displacement, [m]
w = area gradient, [m] ρ = fluid density, [kg/m^3]

Assume constant supply pressure (p_s) and let the load pressure vary in the range $-p_s \le p_L \le p_s$ and the spool displacement in the range $-x_{vmax} \le x_v \le x_{vmax}$. With this assumptions the load flow characteristics will be as illustrated in **Figure 3-14**.

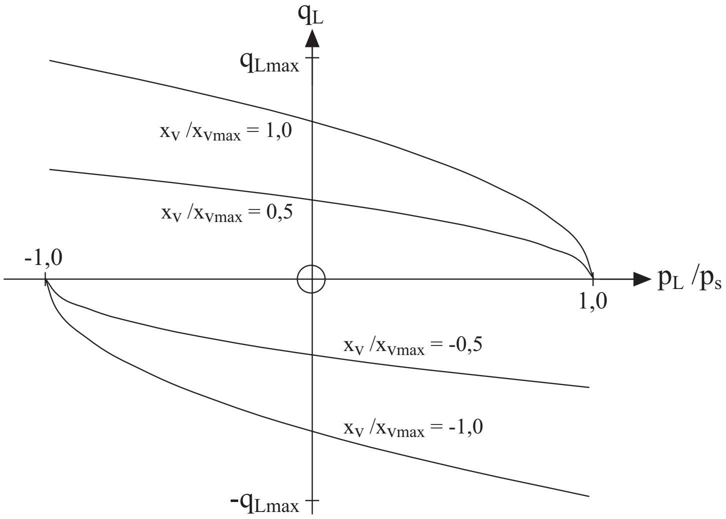

Figure 3-14: Load flow characteristics for a 4-port critical center valve

If the valve has an ***electrical input*** signal (i_v) and the spool displacement x_v is proportional to i_v ($x_v = K_i i_v$), the valve flow equation can be written as

$$q_L = K_i C_q w i_v \sqrt{\frac{1}{\rho}\left(p_s - \frac{x_v}{|x_v|} p_L\right)} \qquad (3\text{-}2)$$

Valve Coefficients

The valve coefficients for an ideal critical center valve can be obtained by differentiation of the equation for the pressure-flow curves (equation 3-1) or graphically from a plot of the curves (Fig. 3.14). These partial derivatives define the most important parameters of the valve. Assuming ***constant supply pressure*** (p_s) they are:

Flow gain $$K_q = \left[\frac{\partial q_L}{\partial x_v} \right]_{p_L = const} \qquad (3\text{-}3)$$

Flow-pressure coefficient $$K_c = \left[-\frac{\partial q_L}{\partial p_L} \right]_{x_v = const} \qquad (3\text{-}4)$$

Pressure sensitivity $$K_p = \left[\frac{\partial p_L}{\partial x_v} \right]_{q_L = const} = \frac{K_q}{K_c} \qquad (3\text{-}5)$$

For an ***electro-hydraulic servo valve*** (electrical input) the valve coefficients will be defined as

$$K_{qi} = \left[\frac{\partial q_L}{\partial i_v} \right]_{p_L = const} , \; K_{ci} = \left[-\frac{\partial q_L}{\partial p_L} \right]_{i_v = const} , \; K_{pi} = \left[\frac{\partial p_L}{\partial i_v} \right]_{q_L = const} = \frac{K_{qi}}{K_c} \qquad (3\text{-}6)$$

Now, the linearized equation of the valve pressure-flow curves becomes

$$\Delta Q_L = K_q \Delta X_v - K_c \Delta P_L \qquad (3\text{-}7)$$

or $$\Delta Q_L = K_{qi} \Delta i_v - K_{ci} \Delta P_L \qquad (3\text{-}8)$$

3.5 Critical center four-way valve

Assuming a symmetrical valve (symmetrical orifices around zero, which means that $A_1(x_v) = A_2(-x_v)$), only positive spool displacement (x_v) can be studied. The valve flow equation can now be simplified as

$$q_L = C_q w x_v \sqrt{\frac{1}{\rho}(p_s - p_L)} \quad \text{or} \quad q_L = K_i C_q w i_v \sqrt{\frac{1}{\rho}(p_s - p_L)} \qquad (3\text{-}9)$$

The valve ***coefficients for an ideal critical center valve*** can be obtained by differentiation of (3-9).

Flow gain $$K_q = C_q w \sqrt{\frac{1}{\rho}(p_s - p_L)} \; , \; K_{qi} = K_i C_q w \sqrt{\frac{1}{\rho}(p_s - p_L)} \qquad (3\text{-}10)$$

Flow-pressure coeff $$K_c = \frac{C_q w x_v}{2\sqrt{\rho(p_s - p_L)}} \; , \; K_{ci} = \frac{K_i C_q w i_v}{2\sqrt{\rho(p_s - p_L)}} \qquad (3\text{-}11)$$

Pressure sensitivity $$K_p = \frac{2(p_s - p_L)}{x_v} \; , \; K_{pi} = \frac{2(p_s - p_L)}{i_v} \qquad (3\text{-}12)$$

As system design parameters, the null operating point is the most important. Evaluation of these coefficients at the point $q_L = p_L = x_v = 0$ gives the ***null coefficients for the ideal critical center valve*** as,

Theoretical null coeff. $K_{qi0} = K_i C_q w \sqrt{\frac{p_s}{\rho}}$, $K_{ci0} = 0$, $K_{pi0} = \infty$ (3-13)

These null coefficients are just ***theoretical***, but the computed null flow gain has been amply verified by tests of practical critical center valves and may be used with confidence. However, the theoretical values for K_{ci0} and K_{pi0} are far from that obtained in tests. It is possible to compute more realistic values for these null coefficients once the leakage characteristics around zero operation for such valve has been investigated.

Practical null coefficients for a critical center valve

It has been shown that the valve coefficients vary a lot with the operating point. The most important operating point is the origin of the flow-pressure curves ($p_L=q_L=i_v=0$) because system operation usually occurs near this region. The null coefficients are of special interest, since they are used as design parameters for the servo system where the servo valve is used. The servo loop gain is proportional to the valve flow gain (K_{qi0}). This null coefficient represents the highest flow gain, which gives the maximum loop gain to obtain stability. Most of the hydraulic damping comes from the valve flow-pressure coefficient (K_{ci0}). The null-coefficient gives the lowest K_c-value and limits the loop gain for stability. The pressure sensitivity (K_{pi0}) determines the stiffness of the servo system. Or in other words, how well load disturbances can de controlled out by the closed loop system.

Leakage characteristics of a practical critical center four-way spool valve

It is the leakage characteristics, which actually differentiate a practical valve from an ideal once. An ideal valve has perfect geometry so that leakage flow are zero. The real valve has radial clearance and perhaps minute under- or overlap. In order to compensate for the radial clearance a critical valve normally has a small overlap of about 5-25 µm. The leakage performance of such valves dominates their behaviour and the associated flow-pressure curves at small valve openings, let say $|x_v| \le 0.02 \cdot x_{vmax}$ (or $|i_v| \le 0.02 \cdot i_{vmax}$). Therefore, the null coefficients, K_{ci0} and K_{pi0} are strongly dependent of the valve leakage performance. On the other hand, for larger valve openings equations (3-11) and (3-12) fits quite well.

Consider a four-way, assumed to have matched and symmetrical orifices, as shown in **Fig 3-15**. With blocked load ports it can be seen that the two gaps for each land from supply port (P) to tank (T) will be equal when the spool is in neutral position, $x_v = 0$. For an out-stroked spool, $x_v > 0$, one gap for each land will be opened up and the leakage flow (q_{le}) is restricted by the other gap.

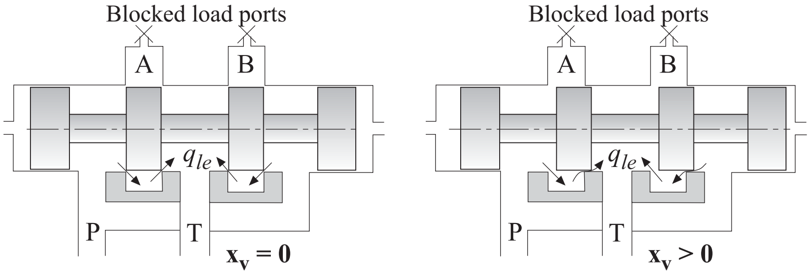

Figure 3-15: Leakage flow in a valve with blocked load ports

Blocked line pressure sensitivity curve

If its true that all the four gaps from P to T are equal at $x_v = 0$, the leakage flow over each land will be equal and also the load pressures p_A and p_B will be equal. In order to introduce a ***load pressure difference, $p_L > 0$*** ($p_L = p_A - p_B$), the spool must be moved from neutral position. By stroking the valve and recording load pressure, p_L versus input current (i_v) the pressure sensitivity can be measured.

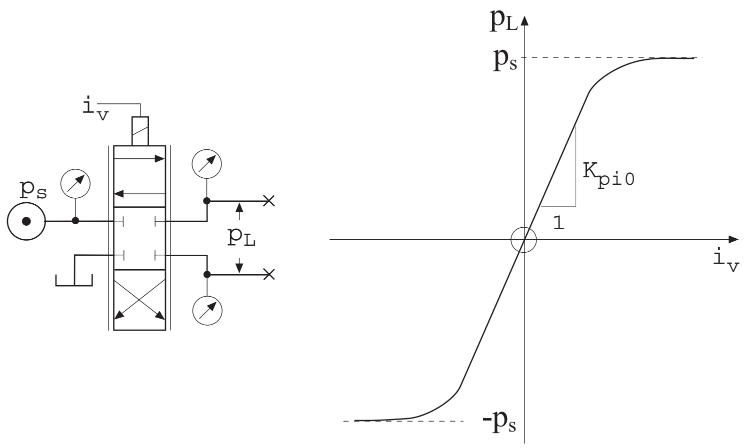

Figure 3-16: Blocked line pressure sensitivity for a critical center valve

The load pressure difference, p_L quickly increases to full supply pressure after a very small increasing of input current. For a good servo valve a ***typical value of pressure sensitivity*** is ***$K_{pi0} = 0.8 \cdot p_s/(0.01 \cdot i_{vmax})$***. As an example p_s = 350 bar and i_{vmax} = 50 mA, gives ***$K_{pi0} = 5.6 \cdot 10^{10}$ Pa/A.*** Pressure sensitivity (gain) is usually specified as the average slope of the of load pressure drop versus input current in the region between ***±40%*** of maximum load pressure drop.

Leakage flow curves

By stroking the valve (i_v) and measuring the supply flow (q_s) for a given supply pressure and with blocked load ports the ***leakage flow curve*** can be plotted, as shown in **Figure 3-17**.

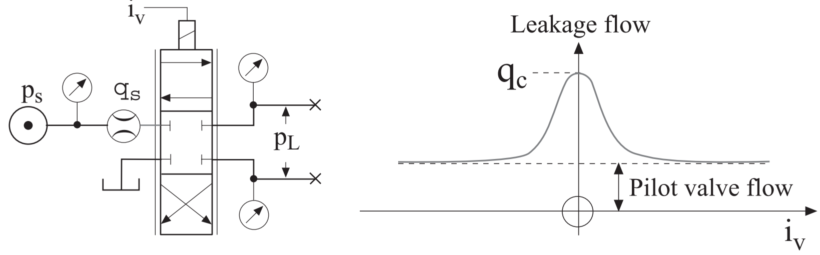

Figure 3-17: Leakage flow characteristic for a two-stage four-way servo valve

The leakage flow is maximum (q_c) at valve neutral ($i_v = 0$) and decreases rapidly with valve stroke because the spool land overlap the return valve orifice, see Fig. 3-15. It can be observed that the flow consumption of the pilot stage of the valve is constant because of the constant supply pressure.

Looking at the supply flow (q_s), it can be shown theoretically that the following expression is valid:

$$\frac{\partial q_s}{\partial p_s} = -\frac{\partial q_L}{\partial p_L} = K_c \qquad (3\text{-}14)$$

From equation (3-14) it is obvious that by measuring the center leakage flow (q_c) versus the supply pressure (p_s) it is possible to calculate the null coefficient K_{ci0}.

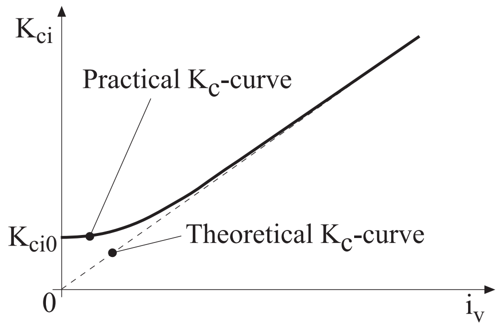

Figure 3-18: Flow-pressure coefficient versus input current

The null coefficient, K_{ci0} is determined by the leakage characteristics only. However, for higher stroking of the valve K_{ci} will follow equation (3-11).

About the null coefficient it is important to note that the K_c-value varies as the valve wears. For a new the center leakage flow can be nearly laminar. For a worn valve the center flow will be more and more turbulent. The center leakage can be expressed in principal as,

$$q_c = cons\tan t \cdot \mu \cdot p_s \quad \text{(laminar flow)} \qquad (3\text{-}15)$$

$$q_c = cons\tan t \cdot \sqrt{p_s} \quad \text{(turbulent flow)} \qquad (3\text{-}16)$$

where μ is the dynamic viscosity of the fluid.

This fact also gives implication on the ***null pressure sensitivity***, K_{pi0}. An approximate expression for this null coefficient of a real critical center valve can be obtained by dividing equation (3-10) by (3-15) and (3-16) respectively,

Laminar center flow: $$K_{pi0} = cons\tan t \cdot \mu \cdot \sqrt{p_s} \qquad (3\text{-}17)$$

Turbulent center flow: $$K_{pi0} = cons\tan t \cdot p_s \qquad (3\text{-}18)$$

It is perhaps worth mentioning that equation (3-17) and (3-17) represents two extremes and for a practical servo valve, measurement of K_{pi0} related to the supply pressure (p_s) will be something between the equation (3-17) and (3-18).

Flow gain

The flow gain of a servo valve with electrical input represents the slope of the control flow (q_L) versus input current (i_v) curve at any specific operation region. For a practical critical center valve the flow gain varies depending on non-linearities and hyteresis, as illustrated in **Figure 3-19**.

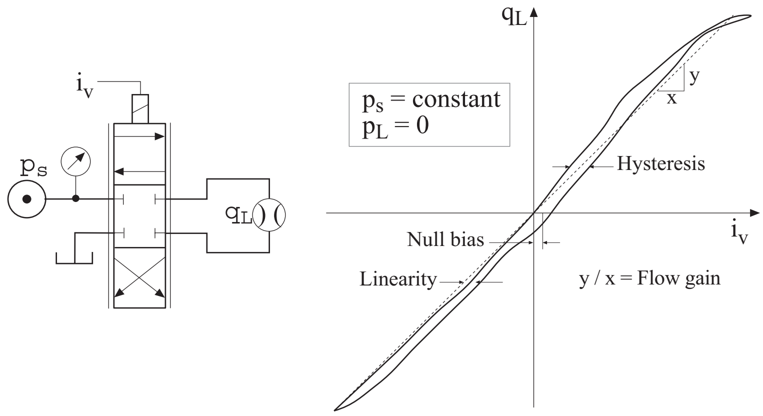

Figure 3-19: Flow gain characteristics for a four-way critical center valve

The straight dotted line in Figure 3-19 drawn from the zero flow point, throughout the range of rated current is called ***Normal Flow Gain***. The operating conditions for the flow curve in the figure above is constant supply pressure and no load pressure ($p_L = 0$). Flow gain as demonstrated in the figure is the normal flow gain with zero-load, which is called ***No-Load Flow Gain***. The no-load flow gain, $\boldsymbol{K_{qi0}}$ will vary with supply pressure as described by equation (3-13).

3.6 Open center spool valve

Consider the four-way spool valve shown in Figure 3-20. When the valve is centred, the underlap of the supply and the return ports are identical with the value U.

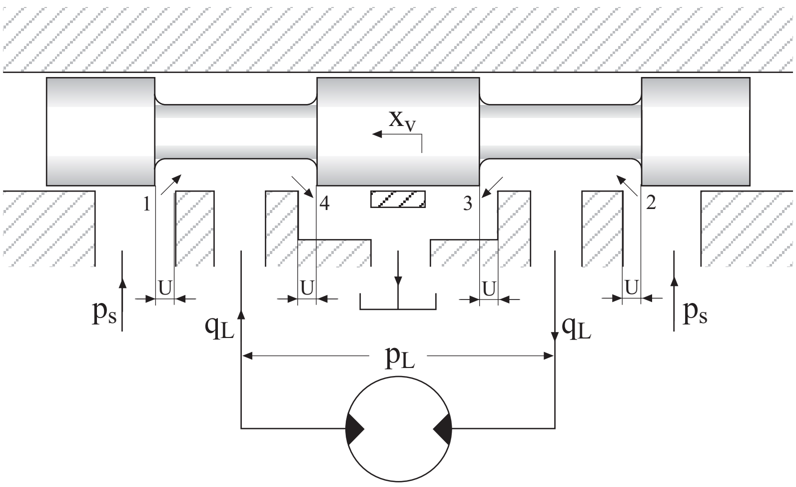

Figure 3-20: Four-way open center spool valve with underlap U

If it is assumed that the valve is matched and symmetrical and if only the underlap region is studied, $|x_v| \le U$, the orifice areas are (observe the direction of x_v in Fig. 3-20),

$$A_1 = w(U + x_v) = A_3 \tag{3-19}$$

$$A_2 = w(U - x_v) = A_4 \tag{3-20}$$

The subscripts of the area refer to the number at the ports in Figure 3-20 and arrows at the ports indicate flow directions. The general flow equation for positive and negative spool stroke ($|x_v| \le U$) and load pressure p_L can be written as,

$$q_L = C_q w(U + x_v)\sqrt{\frac{1}{\rho}(p_s - p_L)} - C_q w(U - x_v)\sqrt{\frac{1}{\rho}(p_s + p_L)} \qquad (3\text{-}21)$$

This is the equation for the ***pressure-flow curves*** of an ***open center*** four-way spool valve for operation within the ***underlap region***. In normalised manner equation (3-21) becomes

$$\frac{q_L}{C_q w U\sqrt{\frac{p_s}{\rho}}} = \left(1 + \frac{x_v}{U}\right)\sqrt{1 - \frac{p_L}{p_s}} - \left(1 - \frac{x_v}{U}\right)\sqrt{1 + \frac{p_L}{p_s}} \qquad (3\text{-}22)$$

Equation (3-21) is plotted in Figure 3-21. These curves are quite linear compared to those for a critical center valve in the null displacement region. However, outside the underlap region the valve acts as a critical center valve, because only two orifices are acting at a time.

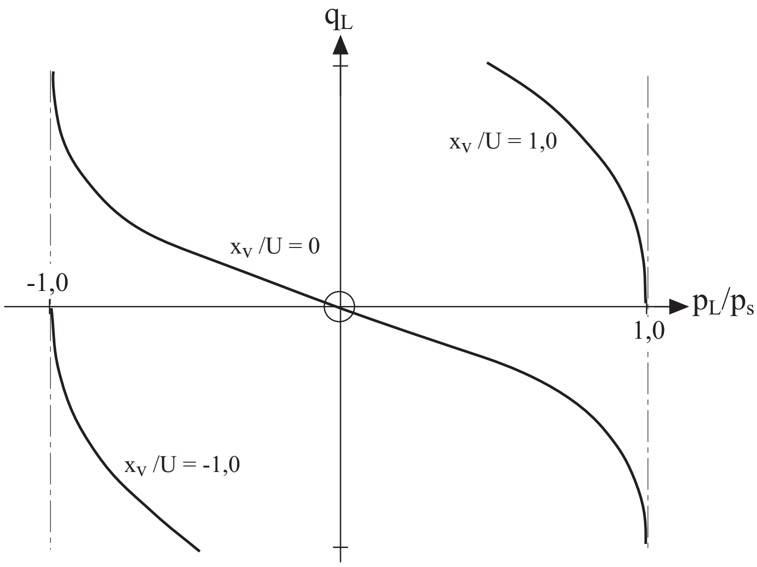

Figure 3-21: Pressure-flow curves for underlap region of an open center four-way spool valve

The valve coefficient can be obtained by differentiating (3-21). Evaluating the derivatives at $q_L = p_L = x_v = 0$ will gives the ***null coefficients***. The coefficients are,

Flow gain	$K_q = C_q w\sqrt{\frac{1}{\rho}}\left[\sqrt{p_s - p_L} + \sqrt{p_s + p_L}\right]$, $K_{q0} = 2C_q w\sqrt{\frac{p_s}{\rho}}$	(3-23)
Flow-pr. coeff.	$K_c = \frac{C_q w x_v}{2\sqrt{\rho}}\left[\frac{U + x_v}{\sqrt{p_s - p_L}} + \frac{U - x_v}{\sqrt{p_s + p_L}}\right]$, $K_{c0} = \frac{C_q w U}{\sqrt{\rho \cdot p_s}}$	(3-24)
Pressure sensitivity	$K_p = \frac{K_q}{K_c}$, $K_{p0} = \frac{2 p_s}{U}$	(3-25)

As discussed before the null coefficients are the most important ones. It can be noted that the flow gain in the underlap region is twice of that for a critical center valve. Further, K_{c0} depends on the area gradient w and the underlap U and K_{p0} is independent of w.

Leakage flow curves similar to those defined for the critical center valve can be made for the open center valve. The total center flow through the valve is useful since it describe the power loss at null operation. At this point, $\boldsymbol{p_L = x_v = 0}$ the orifice areas are $\boldsymbol{A_1 = A_2 = wU}$, which gives the total center flow as

$$q_c = 2C_q wU\sqrt{\frac{p_s}{\rho}} \tag{3-26}$$

3.7 Three-way spool valve analysis

A three-way spool valves must be used together with a differential area piston, to provide direction reversal. The area ratio is normally two, as shown in Figure 3-22.

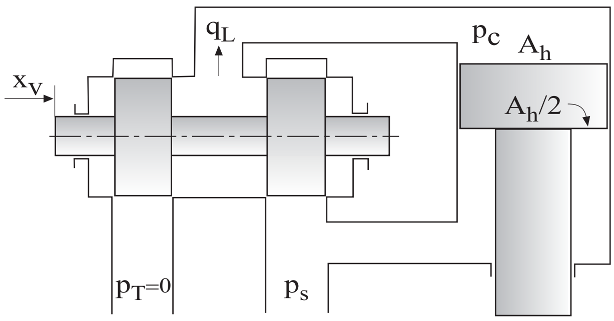

Figure 3-22: Three-way spool valve with differential piston

The rod and the head side areas of the piston is such that the steady state control pressure acting on the head side is about

$$p_{c0} \approx \frac{p_s}{2} \tag{3-27}$$

This design relation allows the control pressure p_c to rise and fall and to provide equal acceleration and deceleration capability. With no loads on the piston this is satisfied by making the head area twice the rod area ($A_h = 2A_r$). This rule is generally used for piston sizing even with load forces. However, this type of valve and piston combination is usually preferred in hydro-mechanical servos where the external load is quite small.

For a ***critical center valve*** the pressure-flow curves can be expressed as

$$q_L = C_q w x_v \sqrt{\frac{2}{\rho}(p_s - p_c)} \quad \text{for} \quad x_v \geq 0$$

$$q_L = C_q w x_v \sqrt{\frac{2}{\rho} p_c} \quad \text{for} \quad x_v < 0 \tag{3-28}$$

The null operating point for a three-way valve is defined by $\boldsymbol{q_L = x_v = 0}$ and $\boldsymbol{p_L = p_s/2}$. Evaluating the derivatives at this point gives the ***null coefficients for a critical center three-way valve*** as

Flow gain	$K_{q0} = C_q w \sqrt{\frac{p_s}{\rho}}$	(3-29)
Flow-pressure coefficient	$K_{c0} = \frac{C_q w x_v}{\sqrt{\rho \cdot p_s}} = 0$	(3-30)
Pressure sensitivity	$K_{p0} = \frac{p_s}{x_v} = \infty$	(3-31)

Comparing the null coefficients, it can be noted that the flow gain is the same but the pressure sensitivity is half of that for a four-way critical center valve. Therefore load forces will cause twice the static error compared with a four-way valve.

For a practical valve the null coefficients become

Practical null coeff.	$K_{q0} = C_q w \sqrt{\frac{p_s}{\rho}}$, $K_{c0} > 0$, $K_{p0} = \frac{K_{q0}}{K_{c0}}$	(3-32)

Three-way valves can also be of ***open center type***. For operation within the underlap region, the pressure-flow curves is given by

$$q_L = C_q w (U + x_v) \sqrt{\frac{2}{\rho}(p_s - p_c)} - C_q w (U - x_v) \sqrt{\frac{2}{\rho} p_c} \qquad (3\text{-}33)$$

Evaluating the derivatives at $\boldsymbol{q_L = x_v = 0}$ and $\boldsymbol{p_L = p_s/2}$ will gives the ***null coefficients***. The ***null coefficients of a three-way open center valve*** are,

Flow gain	$K_{q0} = 2 C_q w \sqrt{\frac{p_s}{\rho}}$	(3-34)
Flow-pressure coefficient	$K_{c0} = \frac{2 C_q w U}{\sqrt{\rho \cdot p_s}}$	(3-35)
Pressure sensitivity	$K_{p0} = \frac{p_s}{U}$	(3-36)

4 Position servos with valve-controlled cylinders

4.1 Asymmetric cylinder

Consider a valve-controlled piston with position feedback as shown in **Figure 4-1**.

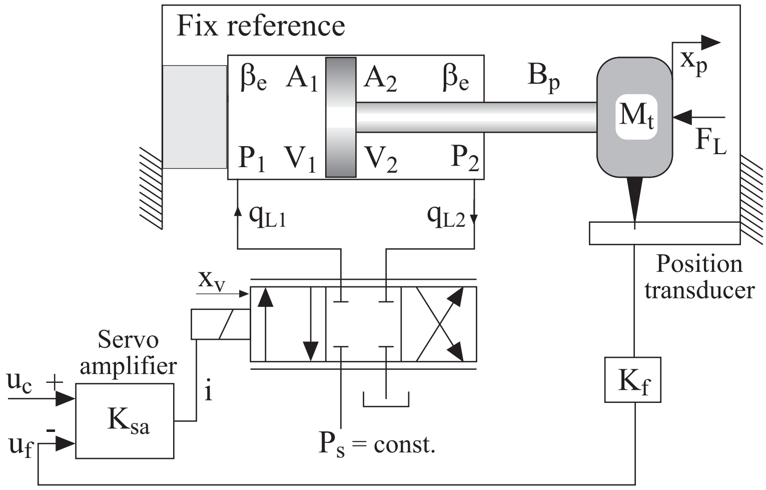

Figure 4-1: Asymmetrical cylinder controlled by four-way critical center spool valve

If it is assumed that the servo valve is matched and symmetrical and the supply pressure (p_s) is constant and the tank pressure is approximately zero.

The general flow equations for ***positive spool stroke*** (***x_v***) can be written as,

$$q_{L1} = C_q w x_v \sqrt{\frac{2}{\rho}(p_s - p_1)} \quad \text{and} \quad q_{L2} = C_q w x_v \sqrt{\frac{2}{\rho} p_2} \qquad (4\text{-}1)$$

The valve coefficient can be obtained by differentiating (4-1). For an operating point 0 the coefficients are,

Flow gain	$K_{q1} = C_q w \sqrt{\frac{2}{\rho}(p_s - p_{10})}$, $K_{q2} = C_q w \sqrt{\frac{2}{\rho} p_{20}}$	(4-2)
Flow-pressure coeff.	$K_{c1} = \frac{C_q w x_{v0}}{\sqrt{2\rho \cdot (p_s - p_{10})}}$, $K_{c2} = \frac{C_q w x_{v0}}{\sqrt{2\rho \cdot p_{20}}}$	(4-3)

The two linearised and laplace-transformed equations describing the valve flow becomes

$$\Delta Q_{L1} = K_{q1}\Delta X_v - K_{c1}\Delta P_1 \qquad (4\text{-}4)$$

$$\Delta Q_{L2} = K_{q2}\Delta X_v + K_{c2}\Delta P_2 \qquad (4\text{-}5)$$

Considering no leakage flow in the cylinder gives the linearised and laplace-transformed continuity equations for the volumes V_1 and V_2 as

$$\Delta Q_{L1} = A_1 s \Delta X_p + \frac{V_1}{\beta_e} s \Delta P_1 \qquad (4\text{-}6)$$

$$-\Delta Q_{L2} = -A_2 s\Delta X_p + \frac{V_2}{\beta_e} s\Delta P_2 \qquad (4\text{-}7)$$

The final equation for the actuator arises from the forces of the piston. The linearised and laplace-transformed force equation will be written as

$$A_1\Delta P_1 - A_2\Delta P_2 = M_t s^2\Delta X_p + B_p s\Delta X_p + \Delta F_L \qquad (4\text{-}8)$$

Combining equation (4-4) to (4-7) gives

$$K_{q1}\Delta X_v = A_1 s\Delta X_p + \left(K_{c1} + \frac{V_1}{\beta_e} s\right)\Delta P_1 \qquad (4\text{-}9)$$

$$-K_{q2}\Delta X_v = -A_2 s\Delta X_p + \left(K_{c2} + \frac{V_2}{\beta_e} s\right)\Delta P_2 \qquad (4\text{-}10)$$

Introducing the position feedback gain K_f, the servo amplifier gain K_{sa} and the transfer function of the servo valve $G_v(s)$ the spool displacement of the valve (x_v) becomes

$$\Delta X_v = \left(U_c - K_f\Delta X_p\right)K_{sa}G_v(s) \qquad (4\text{-}11)$$

By using the equations (4-8) – (4-11) a block diagram of the closed loop system will be as shown in **Figure 4-2**.

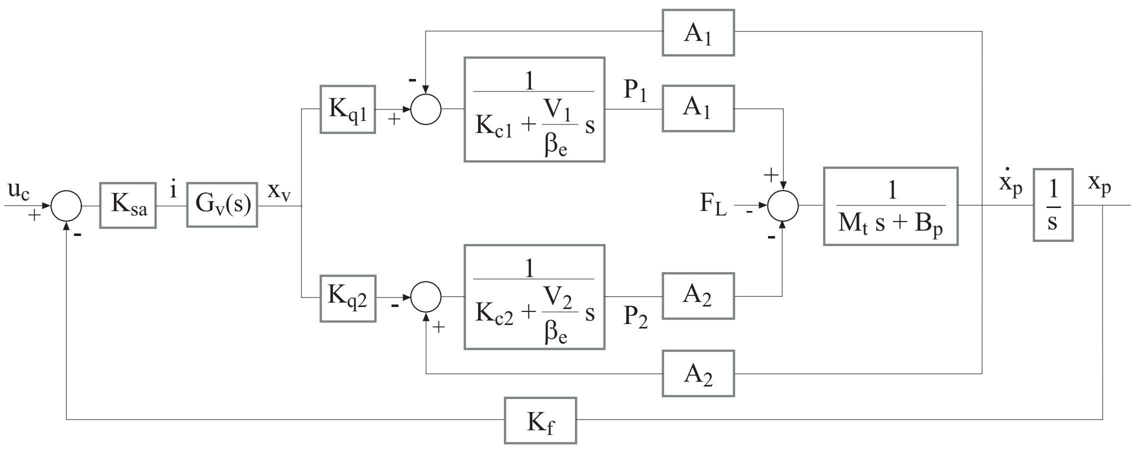

Figure4-2: Block diagram of a position servo with a valve controlled asymmetric cylinder

In the above equations the leakage in the cylinder has been neglected. If leakage is taken into account an extra term will be added to the K_c-values. This can be modelled as $K_{ce1} = K_{c1} + C_{p1}$ and $K_{ce2} = K_{c2} + C_{p2}$.

A problem with the asymmetric cylinder is that the behaviour of the system will not be the same for both stroking directions. When the direction of output velocity is changed it will also be a pressure jump because of the fact that different piston areas are used to produce the actuator force acting on the load.

Example: Variation in resonance frequency for an asymmetric cylinder with line volumes

A mass loaded asymmetric cylinder with line volumes is shown in **Figure 4-3**.

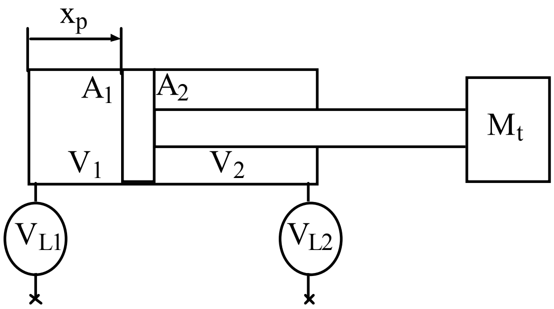

Figure 4-3: A double acting hydraulic cylinder with line volumes

Parameters

Piston displacement: $0 \le x_p \le x_{pmax}$, $x_{pmax} = 1$ m

Piston area: $A_1 = 0.02\ m^2$, $A_2 = 0.75A_1$

Line volumes: $V_{L1} = V_{L2} = k \cdot A_1 \cdot x_{pmax}$, $0 < k < 1$

Bulk modulus: $\beta_e = 1000$ MPa

Resonance frequency: $\omega_h = \sqrt{\dfrac{K_h}{M_t}}$ where $K_h = \beta_e \left[\dfrac{A_1^2}{V_{L1} + A_1 x_p} + \dfrac{A_2^2}{V_{L2} + A_2 (x_{p\max} - x_p)} \right]$

Figure 4-4 shows the relative frequency ω_h/ω_{hmin} versus piston displacement (x_p) for k = 0.05 (the highest curve) and k = 0.10 (the lowest curve). It is notable that the cylinder will be much stiffer when the piston is close to end positions.

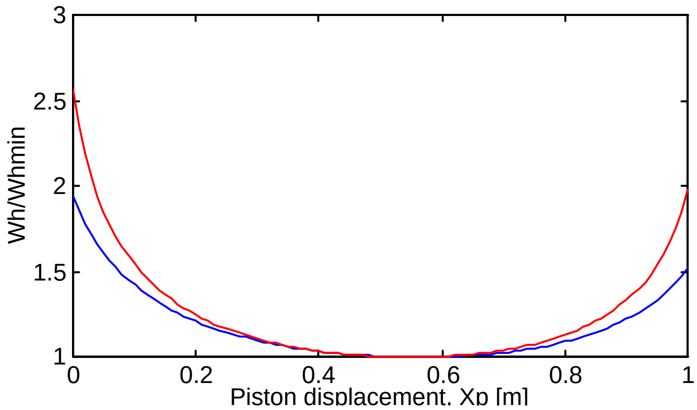

Figure 4-4: Relative resonance frequency versus piston displacement for two different line volumes

4.2 Valve controlled symmetric cylinder

A position servo with a symmetric cylinder is shown in **Figure 4-5**.

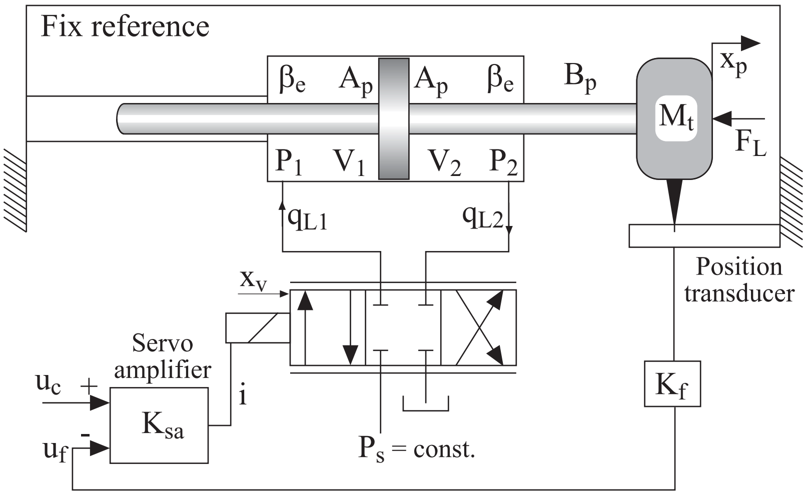

Figure 4-5: Position servo with a valve controlled symmetric cylinder

The general flow equations for the four-way critical center valve can be written as,

$$q_L = C_q w x_v \sqrt{\frac{1}{\rho}(p_s - p_L)} \quad \text{where} \quad p_L = p_1 - p_2 \tag{4-12}$$

From equation (4-12) the valve coefficient can be obtained as

Flow gain	$K_q = C_q w \sqrt{\frac{1}{\rho}(p_s - p_{L0})}$	(4-13)
Flow-pressure coeff.	$K_c = \frac{C_q w x_{v0}}{2\sqrt{\rho \cdot (p_s - p_{L0})}}$	(4-14)

The linearised and laplace-transformed equations describing the valve flow becomes

$$\Delta Q_L = K_q \Delta X_v - K_c \Delta P_L \tag{4-15}$$

Considering no external leakage flow in the cylinder (just across the piston) gives the linearised and laplace-transformed continuity equations for the volumes V_1 and V_2 as

$$\Delta Q_{L1} - C_p(\Delta P_1 - \Delta P_2) = A_p s \Delta X_p + \frac{V_1}{\beta_e} s \Delta P_1 \tag{4-16}$$

$$-\Delta Q_{L2} + C_p(\Delta P_1 - \Delta P_2) = -A_p s \Delta X_p + \frac{V_2}{\beta_e} s \Delta P_2 \tag{4-17}$$

Because of the symmetric cylinder it is possible to calculate the load flow as

$$\Delta Q_L = \frac{\Delta Q_{L1} + \Delta Q_{L2}}{2} \tag{4-18}$$

Assume that the piston is in centred position (x_p=0) and the volumes are $V_1 = V_2 = V_t/2$, where V_t is the total pressurised volume in the cylinder. Combining equation (4-16) to (4-18) using the definition of the load pressure difference $\Delta P_L = \Delta P_1 - \Delta P_2$ gives

$$\Delta Q_L = A_p s \Delta X_p + C_p \Delta P_L + \frac{V_t}{4\beta_e} s \Delta P_L \quad (4\text{-}19)$$

Combining equation (4-15) and (4-19) and with ilectrical input to the servo valve gives

$$K_{qi} \Delta X_v = A_p s \Delta X_p + \left(K_{ce} + \frac{V_t}{4\beta_e} s \right) \Delta P_L \quad (4\text{-}20)$$

where the total flow-pressure coefficient is $K_{ce} = K_c + C_p$.

The final equation for the actuator arises from the forces of the piston. The linearised and laplace-transformed force equation will be written as

$$A_p (\Delta P_1 - \Delta P_2) = A_p \Delta P_L = M_t s^2 \Delta X_p + B_p s \Delta X_p + \Delta F_L \quad (4\text{-}21)$$

Introducing the position feedback gain K_f, the servo amplifier gain K_{sa} and the transfer function of the servo valve $G_v(s)$ the spool displacement of the valve (x_v) becomes

$$\Delta X_v = (U_c - K_f \Delta X_p) K_{sa} G_v(s) \quad (4\text{-}22)$$

By using the equations (4-20) – (4-22) a block diagram of the closed loop system will be as shown in **Figure 4-6**.

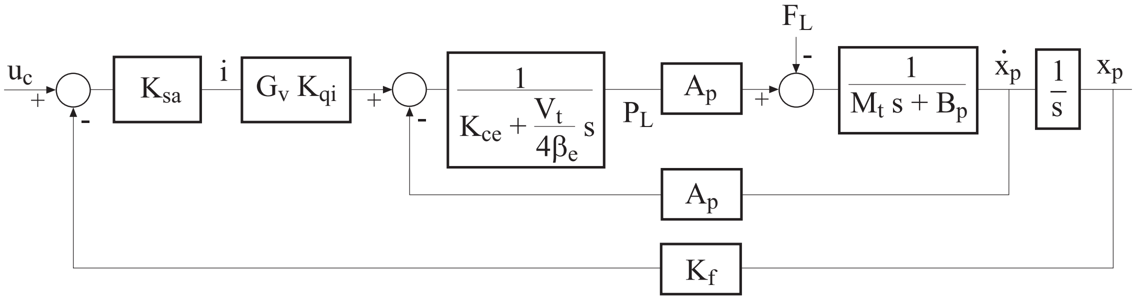

Figure 4-6: Block diagram of a position servo with a valve controlled symmetric cylinder

Consider a first order transfer function for the servo valve, written as

$$G_v(s) = \frac{1}{1 + \frac{s}{\omega_v}} \quad (4\text{-}23)$$

The block-diagram in Figure 4-4 can now be reduced to the following form, shown in **Figure 4-7**.

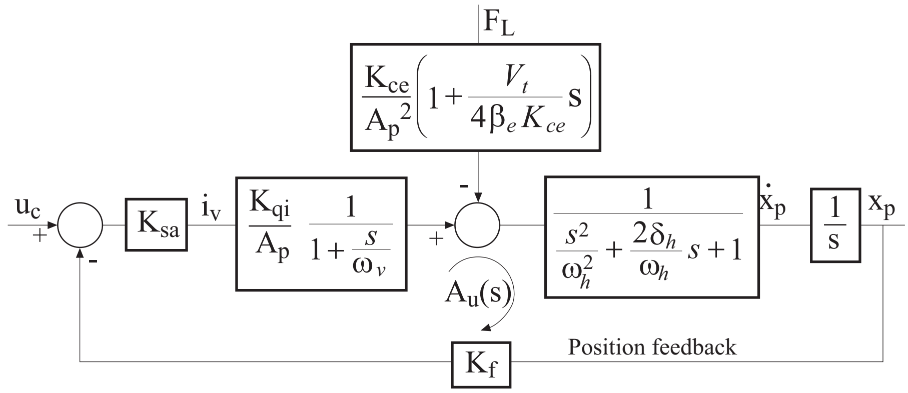

Figure 4-7: Block diagram of a position servo with a valve controlled symmetric cylinder

If the term $B_p K_{ce} / A_p^2$ is smaller than unit the hydraulic resonance frequency ω_h and the hydraulic damping δ_h shown in the block-diagram (Figure 4-5) will be expressed as

$$\omega_h = \sqrt{\frac{\beta_e A_p^2}{M_t}\left(\frac{1}{V_1}+\frac{1}{V_2}\right)} \quad V_1 = V_2 = V_t/2 \text{ gives } \quad \omega_h = \sqrt{\frac{4\beta_e A_p^2}{M_t V_t}}$$

$$\text{and} \quad \delta_h = \frac{K_{ce}}{A_p}\sqrt{\frac{\beta_e M_t}{V_t}} + \frac{B_p}{4A_p}\sqrt{\frac{V_t}{\beta_e M_t}}$$

If the term $B_p K_{ce} / A_p^2$ is included the resonance frequency and damping becomes

$$\omega_h' = \omega_h \sqrt{1+\frac{B_p K_{ce}}{A_p^2}} \quad \text{and} \quad \delta_h' = \frac{\delta_h}{\sqrt{1+\frac{B_p K_{ce}}{A_p^2}}}$$

In order to study the stability of the servo system the ***open loop gain*** $A_u(s)$ must be analysed. Figure 4-5 yields

$$A_u(s) = \frac{K_{sa} K_{qi} K_f / A_p}{\left(1+\frac{s}{\omega_v}\right)\cdot s \cdot \left(\frac{s^2}{\omega_h^2}+\frac{2\delta_h}{\omega_h}s+1\right)} = \frac{K_v}{\left(1+\frac{s}{\omega_v}\right)\cdot s \cdot \left(\frac{s^2}{\omega_h^2}+\frac{2\delta_h}{\omega_h}s+1\right)} \qquad (4\text{-}24)$$

$\boldsymbol{K_v}$ expresses the steady state loop gain and the value of this parameter must be set to a certain level to make sure that the control system will be stable.

The critical parameter in this servo system is the amplitude margin A_m, which is expressed as

$$A_m = -20^{10}\log\left|\frac{K_v}{-2\delta_h \omega_h}\right| \quad [\text{dB}] \qquad (4\text{-}25)$$

In other words, the control system will be stable if the amplitude margin is positive, which gives the stability criteria as

$$K_{v\max} < 2\delta_{h\min}\omega_{h\min} \qquad (4\text{-}26)$$

or for a specified amplitude margin

$$K_{v\max} = 10^{-\frac{A_m}{20}} 2\delta_{h\min}\omega_{h\min} \quad (4\text{-}27)$$

The open loop gain of the position servo with $K_v = \delta_h\omega_h$ (Am = 6 dB) is shown in **Figure 4-8**.

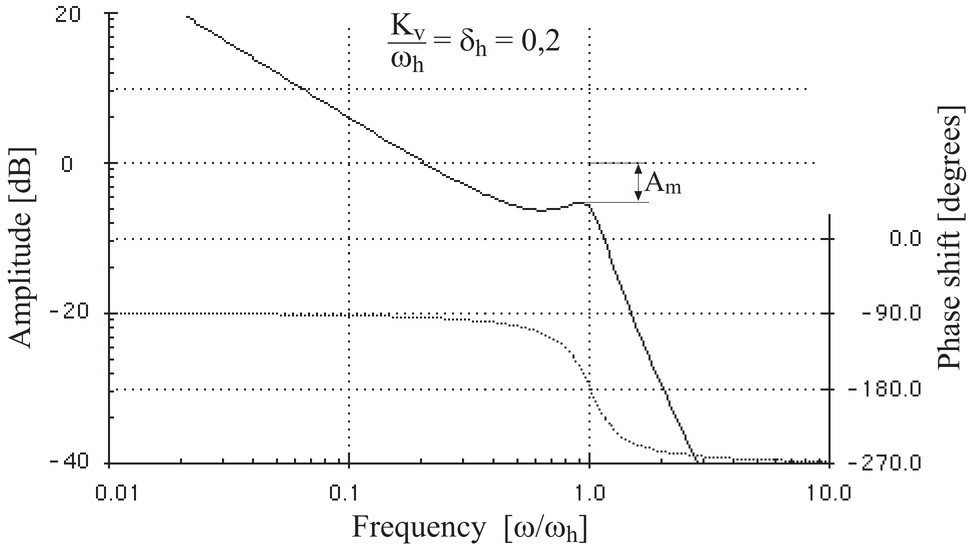

Figure 4-8: Bode-diagram of the open loop gain of the position servo depicted in Figure 4-5

A bode-diagram of the closed loop system is shown in **Figure 4-9**.

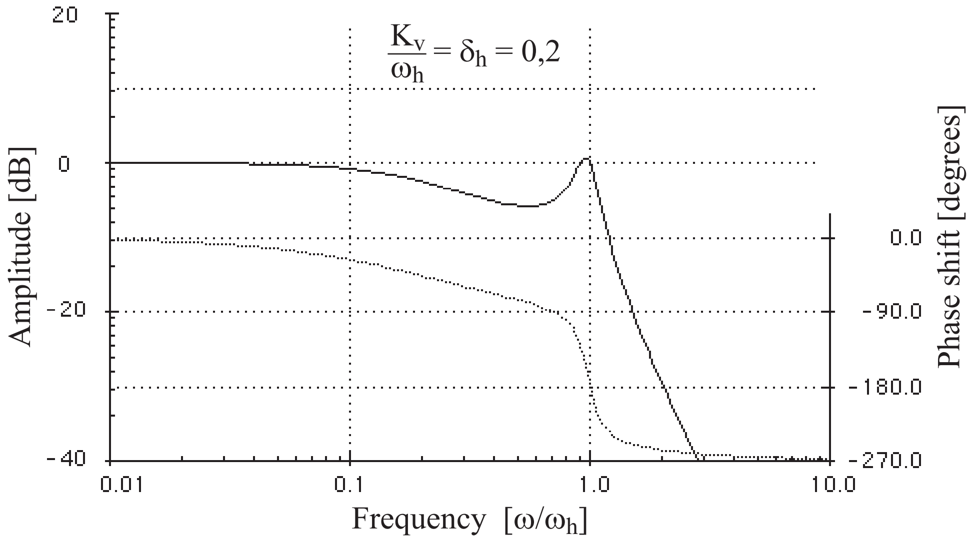

Figure 4-9: Bode-diagram of the closed loop gain of the position servo depicted in Figure 4-5

4.3 Three-way valve controlled cylinder with force feedback

A three-way spool valves must be used together with a differential area piston, to provide direction reversal. The area ratio is normally two, as shown in **Figure 4-10**. In the figure the position feedback is implemented by using a force feedback (spring) on the spool of the control valve. The control valve is equipped with a proportional magnet, which gives a force (F_m) proportional to the input current (i). Then the force balance on the valve spool means that the piston position (x_p) will be proportional to the input current.

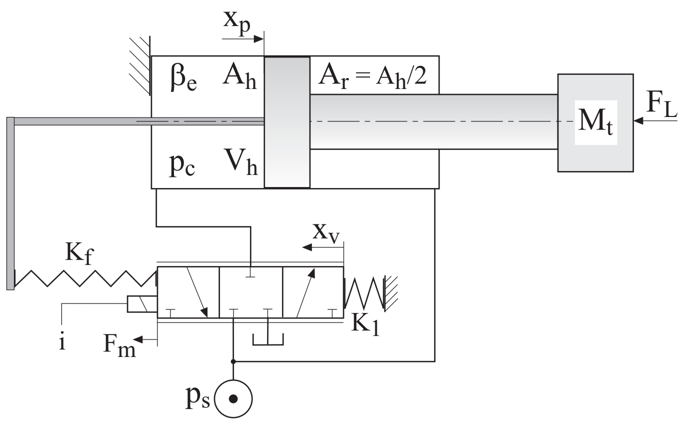

Figure 4-10: Linear actuator with a three-way spool valve, differential piston and force feedback

Neglecting flow forces the forces acting on the valve spool is given by the equation:

$$\Delta F_m - K_f\left(\Delta X_p - \Delta X_v\right) - K_1 \Delta X_v = 0 \;\Rightarrow\; \Delta X_v = \frac{\Delta F_m - K_f \Delta X_p}{K_f + K_1} \qquad (4\text{-}28)$$

Assuming that the magnet force is a linear function of input current, as $\Delta F_m = K_m \Delta i$ the block diagram of the system will become as shown in **Figure 4-11**.

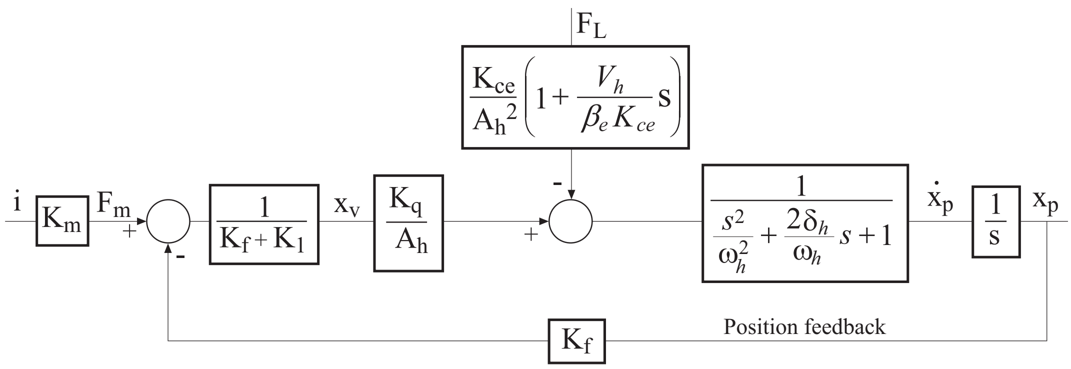

Figure 4-11: Block diagram of an actuator with a three-way spool valve, differential piston and force feedback

In the system above there is only one cylinder volume acting as a spring, namely V_h. If also the viscous friction of the piston is low ($B_p K_{ce} / A_h^2 < 1$) the hydraulic resonance frequency ω_h and the hydraulic damping δ_h shown in the block-diagram (Figure 4-11) will be expressed as

Hydraulic resonance frequency: $\omega_h = \sqrt{\dfrac{\beta_e A_h^2}{M_t V_h}}$

Hydraulic damping: $\delta_h = \dfrac{K_{ce}}{2A_h}\sqrt{\dfrac{\beta_e M_t}{V_h}} + \dfrac{B_p}{2A_h}\sqrt{\dfrac{V_h}{\beta_e M_t}}$

If the term $B_p K_{ce} / A_h^2$ is included the resonance frequency and damping becomes

$$\omega_h' = \omega_h \sqrt{1 + \frac{B_p K_{ce}}{A_h^2}} \quad \text{and} \quad \delta_h' = \frac{\delta_h}{\sqrt{1 + \dfrac{B_p K_{ce}}{A_h^2}}}$$

In order to study the stability of the servo system the ***open loop gain*** $A_u(s)$ must be analysed. Including valve dynamics $\left(G_v = \dfrac{1}{1 + s/\omega_v} \right)$ Figure 4-11 yields

$$A_u(s) = \frac{\dfrac{K_f K_q}{(K_f + K_1)A_p}}{\left(1 + \dfrac{s}{\omega_v}\right) \cdot s \cdot \left(\dfrac{s^2}{\omega_h^2} + \dfrac{2\delta_h}{\omega_h} s + 1\right)} = \frac{K_v}{\left(1 + \dfrac{s}{\omega_v}\right) \cdot s \cdot \left(\dfrac{s^2}{\omega_h^2} + \dfrac{2\delta_h}{\omega_h} s + 1\right)} \qquad (4\text{-}29)$$

$\boldsymbol{K_v}$ expresses the steady state loop gain and the value of this parameter must be set to a certain level, according to equation (4-27), to make sure that the control system will be stable.

This type of actuators, are commonly used for spool control in big valves and for displacement control of variable pumps and motors. Because of the fact that the cylinder pressures are only controlled in one chamber, the stiffness of the actuator will be reduced compared to an actuator with two-chamber control. That means less accuracy in system with heavy external forces.

4.4 Influence from flow forces on valve spools

In an actuator system with a direct controlled servo valve the flow forces acting on the valve spool will influence the spool position. The flow forces act as a positive load pressure feedback, which means reduced hydraulic damping in the system. To reduce this influence the valve spool positioning system must be stiff enough so that the deflexion of the spool, caused by the flow forces, is small.

By introducing a spring between the electric armature (magnet) and the valve spool, the influence from flow forces can be studied. This is illustrated in **Figure 4-12**.

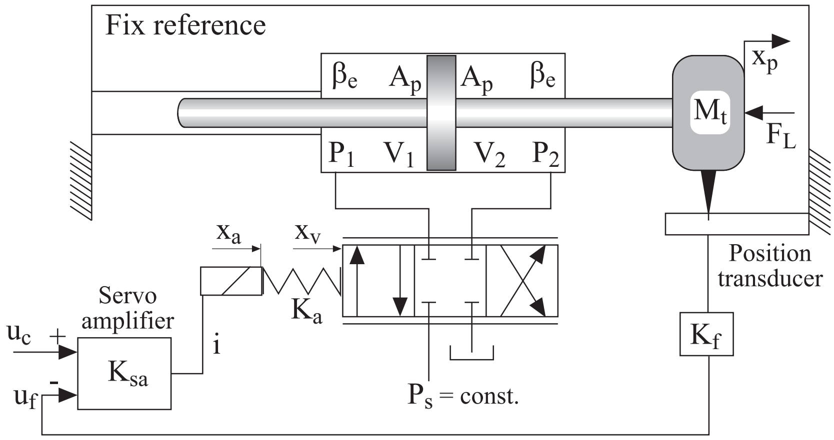

Figure 4-12: Position servo with a weak spool control of the servo valve

Referring to Figure 4-12 the armature position (x_a) can be expressed as $x_a = i \cdot K_m$, where K_m is the force coefficient of the magnet. In the four-port valve the flow forces remains from two orifices, which gives the total steady state flow force as,

$$F_{ft} = 2C_q w \cdot x_v (p_s - p_1)\cos(\varphi) + 2C_q w \cdot x_v \cdot p_2 \cdot \cos(\varphi) = 2C_q w \cdot x_v (p_s - p_L)\cos(\varphi)$$

The force equation for the valve spool can now be expressed as,

$$(i \cdot K_m - x_v)K_a - 2C_q w \cdot x_v (p_s - p_L)\cos(\varphi) = (i \cdot K_m - x_v)K_a - K_{ft} \cdot x_v (p_s - p_L) = 0$$

A linearised expression of the force equation, according to the variable x_v and p_L gives,

$$(\Delta i \cdot K_m - \Delta X_v)K_a - K_{ft}(p_s - p_{L0})\Delta X_v + K_{ft} x_{v0} \Delta P_L = 0$$

In this equation the valve dynamics is ignored. This can be done if the bandwidth of the valve is higher than the dominant resonance frequency of the system. The spool displacement will become as,

$$\Delta X_v = \frac{\Delta i \cdot K_m K_a + K_{ft} x_{v0} \Delta P_L}{K_a + K_{ft}(p_s - p_{L0})} = \frac{\Delta i \cdot K_m K_a + K_{fp} \Delta P_L}{K_a + K_{fx}} \qquad (4\text{-}30)$$

Implementation of equation (4-30) in a block diagram of the servo system gives the result presented in **Figure 4-13**.

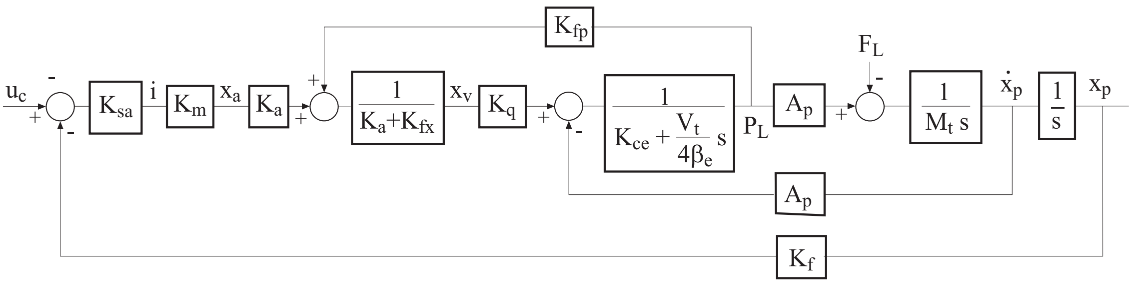

Figure 4-13: Block diagram of a position servo with flow forces acting on the spool control of the servo valve

Figure 4-13 shows that the flow force act as a positive load pressure (p_L) feedback. It is well known that the flow/pressure-coefficient of the valve and cylinder (K_{ce}) act as a negative load pressure feedback. Therefore, it's interesting to combine these effects and see what happens with the resulting K_{ce}-value. This can be done by reduction of the block diagram as shown in Figure 4-14.

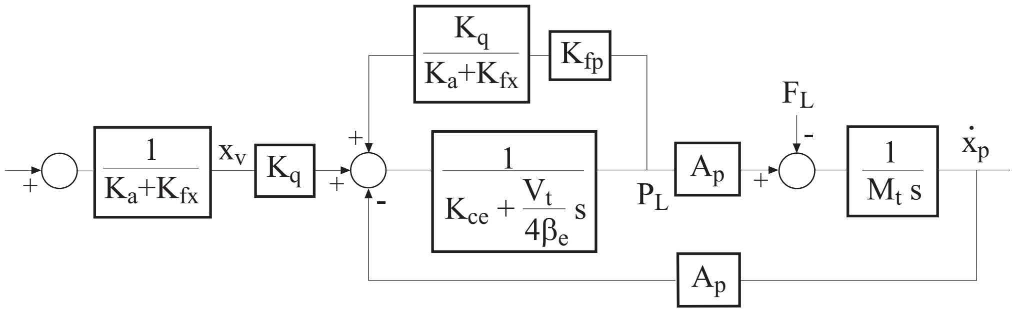

Figure 4-14: Block diagram showing the load pressure feedback from flow forces

If the positive load pressure feedback is co-operated into the main transfer function from flow to load pressure, this will give the resulting K_{ce}-value as,

$$K_{ce}^{*} = K_{ce} - \frac{K_q K_{fp}}{K_a + K_{fx}} \qquad (4\text{-}31)$$

Equation (4-31) shows that $K_{ce}^{*} < K_{ce}$ and since the hydraulic damping is proportional to the K_{ce}-value it is true that flow forces acting on a valve spool will reduce the hydraulic damping. For a good valve it's therefore important to design the armature so that stiffness (in this model K_a) is much higher than the spring coefficient of the flow forces, K_{fx}. With such a design the influence from flow forces will be small.

4.5 Position servo with mechanical springs at connectors

Figure 4-14 shows a linear hydraulic position servo. For the cylinder the weakness of the mechanical part is related to spring systems in the rear mounting end (K_1) and in the piston rod (K_L) as shown in the figure.

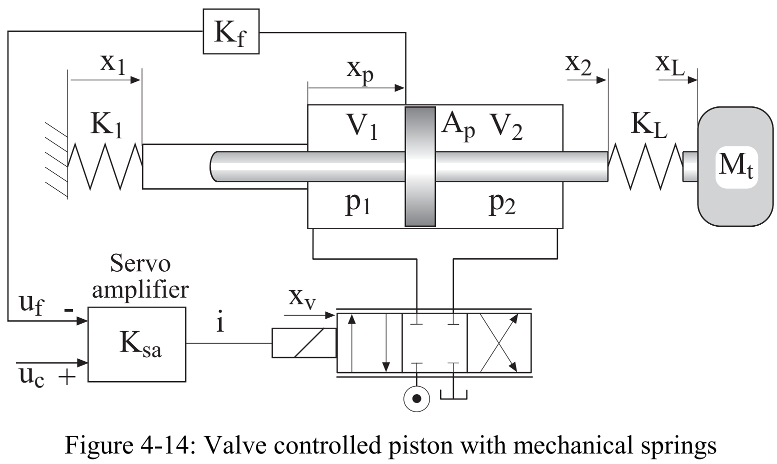

Figure 4-14: Valve controlled piston with mechanical springs

According to Figure 4-14 the block diagram for the system, without position feedback, will be developed as in Figure 4-15.

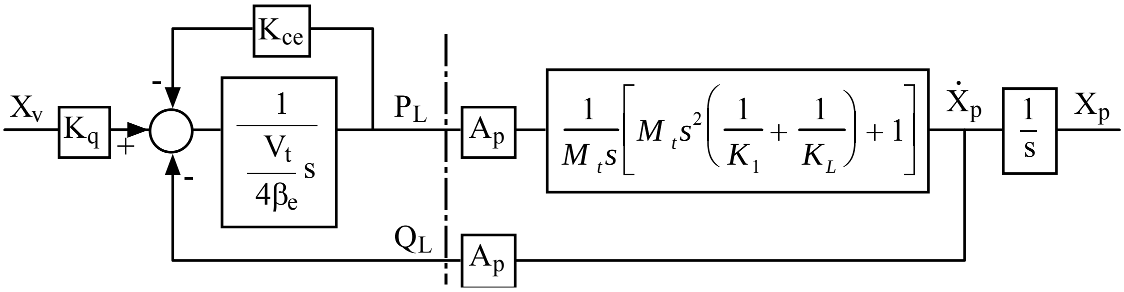

Figure 4-15: Block diagram for a valve controlled cylinder with elastic mountings

Reduction of the block diagram in Figure 4-15 and completing with the transfer function from X_p to X_L and the position feedback loop gives the following diagram.

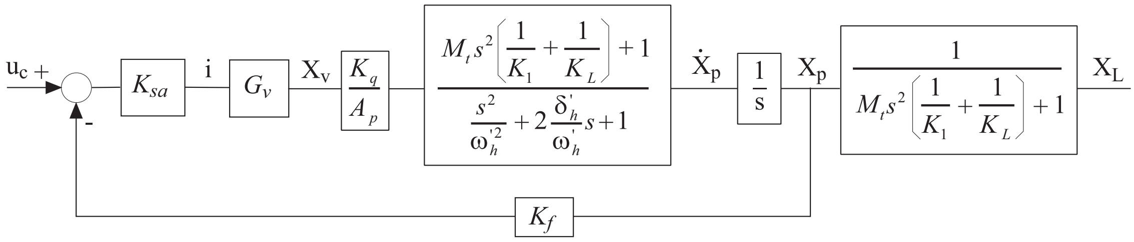

Figure 4-16: Complete block diagram for a valve controlled cylinder servo with elastic mountings

where $\omega_h' = \sqrt{\frac{K_e}{M_t}}$; $\frac{1}{K_e} = \frac{V_t}{4\beta_e A_p^2} + \frac{1}{K_1} + \frac{1}{K_L}$ and $\delta_h' = \frac{K_{ce} M_t}{A_p^2} \cdot \frac{\omega_h'}{2}$

If can be noted that the effective spring gradient K_e is derived from the series connection between the hydraulic spring gradient $\frac{4\beta_e A_p^2}{V_t}$ and the two mechanical springs K_1, K_L.

Simulation of position servo with mechanical springs

The following parameter values is used in the simulation model:

$A_p = 2{,}5 \cdot 10^{-3}$ m^2	$\beta_e = 1{,}0 \cdot 10^9$ Pa
$B_p = 0$	$K_f = 25$ V/m
$K_{ce} = 1{,}0 \cdot 10^{-11}$ m^5/Ns	$K_{qi} = 0{,}02$ m^3/As
$K_{sa} = 0{,}05$ A/V	$M_t = 1500$ kg
$V_t = 1{,}0 \cdot 10^{-3}$ m^3	$\tau_v = 0{,}005$ s
$K_1 = K_L = 5{,}0 \cdot 10^7$ N/m	

The transfer function from valve flow to piston position (X_p) has the form:

$$G_{hxp} = \frac{\frac{s^2}{\omega_L^2} + 2\frac{\delta_L}{\omega_L}s + 1}{\frac{s^2}{\omega_h^2} + 2\frac{\delta_h}{\omega_h}s + 1}$$ and from X_p to X_L we have: $$G_{hxL} = \frac{1}{\frac{s^2}{\omega_L^2} + 2\frac{\delta_L}{\omega_L}s + 1}$$

where ω_h = 91 rad/s, δ_h = 0.11 and ω_L = 123 rad/s, δ_L = 0.01 (δ_L belongs to the viscous friction on the load side).

The valve dynamics is modelled as $G_v = \dfrac{1}{\tau_v s + 1}$

Form the parameter list it can be seen that the steady state loop is $K_v = \dfrac{K_v K_{sa} K_{qi}}{A_p} = 10$, which gives an amplitude margin of about 6 dB.

Simulation of the system with piston position feedback, X_p

A DYMOLA-model of the servo system with position feedback (X_p or X_L) is shown in **Figure 4-17**. By setting the feedback gain to 1 or 0 the feedback loop can be activated or eliminated.

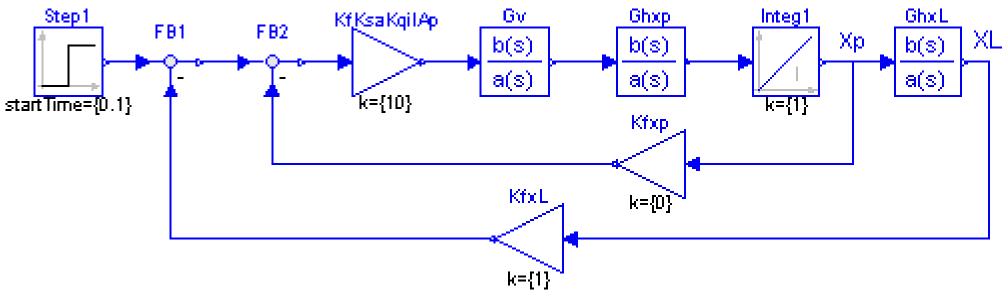

Figure 4-17: DYMOLA-model of the position servo with position feedback of piston or load (X_p or X_L)

Simulation results with piston position feedback (X_p) are shown in Figure 4-18 and 4-19 respectively. It can be noted that the amplitude of oscillation is higher for X_L then for X_p, because of the dynamics between these positions.

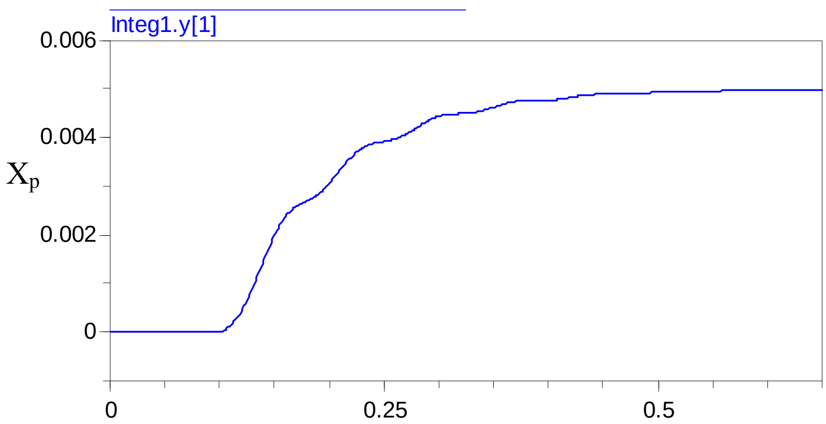

Figure 4-18: Step response (0 - 5 mm) of piston position (X_p) with piston position feedback, X_p

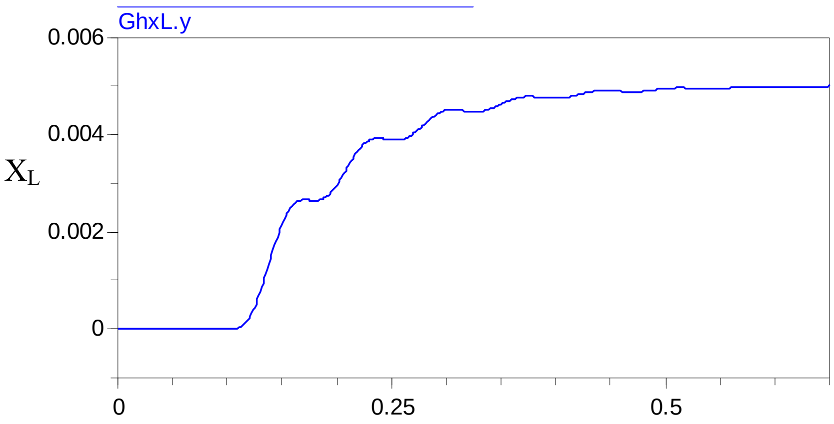

Figure 4-19: Step response (0 - 5 mm) of load position (X_L) with piston position feedback, X_p

Simulation of the system with load position feedback, X_L

Simulation results with load position feedback (X_L) are shown in Figure 4-20 and 4-21 respectively. From the simulation model (Figure 4-17) it is obvious that the introduced load damping (δ_L) will not affect the amplitude of X_L, but the piston position (X_p) will be better damped if the load damping is increased.

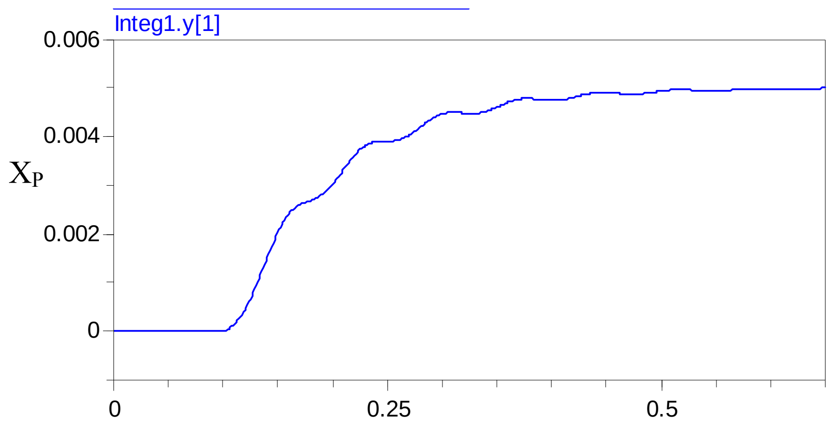

Figure 4-20: Step response (0 - 5 mm) of piston position (X_p) with load position feedback, X_L

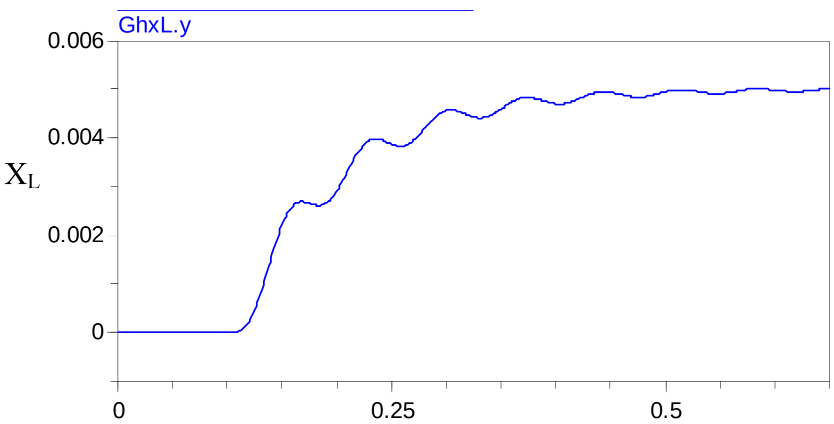

Figure 4-21: Step response (0 - 5 mm) of load position (X_L) with load position feedback, X_L

The results shows that feedback of X_p gives less oscillations amplitude than if X_L is fed back. The reason is that with feedback of X_p acts ω_L as an anti-resonance and reduces the resonance-top from ω_h. A request for the reduction is that ω_L is quite close to ω_h, in this case is $\omega_L = 1.4\ \omega_h$ (observe that ω_L always is greater than ω_h in the given system).

5 Servo systems with valve or pump controlled motors

5.1 Four-way valve controlled motor with position feedback

The hydraulic actuator composed of a valve controlled rotary motor is a widely used combination. All the non-linearities existing in a valve controlled cylinder system will exist even here. One special problem is that the displacement volume in the motor is not constant but varies in a discontinuous fashion with the shaft rotation. By using a motor with a high number of "pumping" elements (pistons) the amplitude of the kinematic displacement variation will be reduced and can be ignored in dynamic calculations. A simplified angular position servo is shown in **Figure 5-1**:

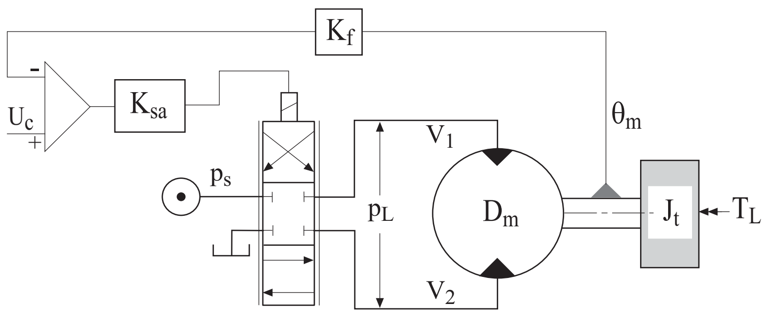

Figure 5-1: Valve controlled angular position servo

The linearised and laplace-transformed equations describing the valve flow becomes

$$\Delta Q_L = K_{qi}\Delta X_v - K_c\Delta P_L \quad (5\text{-}1)$$

Assume that the volumes between valve and motor are equal, $V_1 = V_2$. The total pressurised volume is $V_t = V_1 + V_2$. From the cylinder case, equation (4-16) to (4-18) and the definition of the load pressure difference $\Delta P_L = \Delta P_1 - \Delta P_2$ gives

$$\Delta Q_L = D_m s\Delta\theta_m + C_m\Delta P_L + \frac{V_t}{4\beta_e}s\Delta P_L \quad (5\text{-}2)$$

Combining equation (5-1) and (5-2) gives

$$K_{qi}\Delta X_v = D_m s\Delta\theta_m + \left(K_{ce} + \frac{V_t}{4\beta_e}s\right)\Delta P_L \quad (5\text{-}3)$$

where the total flow-pressure coefficient is $K_{ce} = K_c + C_m$.

The final equation for the actuator arises from the torque of the motor. The linearised and laplace-transformed force equation will be written as

$$D_m(\Delta P_1 - \Delta P_2) = D_m \Delta P_L = J_t s^2 \Delta\theta_m + B_m s \Delta\theta_m + \Delta T_L \qquad (5\text{-}4)$$

Introducing the position feedback gain K_f, the servo amplifier gain K_{sa} and the transfer function of the servo valve G_v(s) the spool displacement of the valve (x_v) becomes

$$\Delta X_v = (U_c - K_f \Delta X_p) K_{sa} G_v(s) \qquad (5\text{-}5)$$

By using the equations (5-3) – (5-5) a block diagram of the closed loop system will be as shown in **Figure 5-2**.

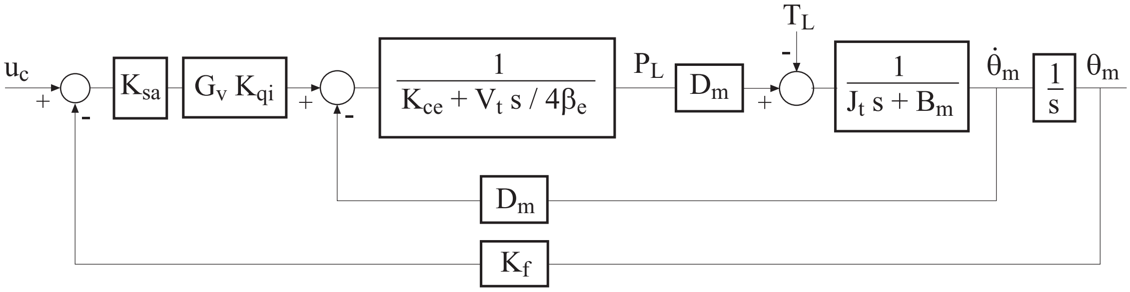

Figure 5-2: Block diagram of a valve controlled angular position servo

The block-diagram in Figure 5-2 can be reduced to the following form, shown in **Figure 5-3**. G_v is a typical low pass filter describing the servo valve dynamics (eq. 4-23).

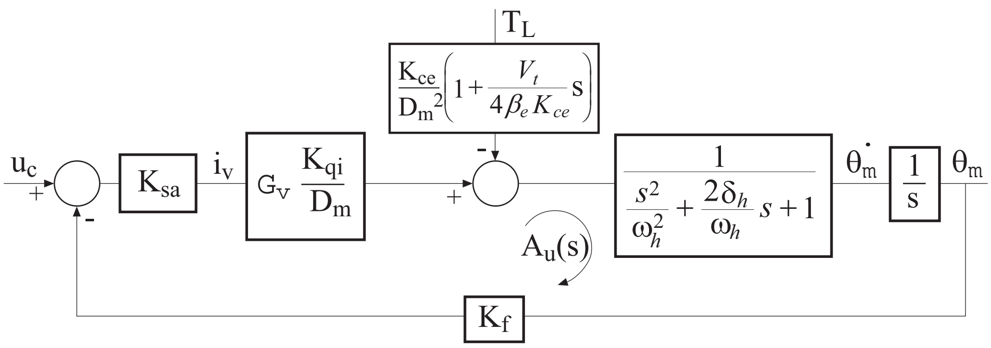

Figure 5-3: Reduced block diagram of a position servo with a valve controlled motor

If the term $B_m K_{ce} / D_m^2$ is smaller than unit the hydraulic resonance frequency ω_h and the hydraulic damping δ_h shown in the block-diagram (Figure 5-3) will be expressed as

$$\omega_h = \sqrt{\frac{\beta_e D_m^2}{J_t}\left(\frac{1}{V_1} + \frac{1}{V_2}\right)} \quad V_1 = V_2 = V_t/2 \text{ gives } \quad \omega_h = \sqrt{\frac{4\beta_e D_m^2}{J_t V_t}}$$

$$\text{and} \quad \delta_h = \frac{K_{ce}}{D_m}\sqrt{\frac{\beta_e J_t}{V_t}} + \frac{B_m}{4D_m}\sqrt{\frac{V_t}{\beta_e J_t}}$$

If the term $B_m K_{ce} / D_m^2$ is included the resonance frequency and damping becomes

$$\omega_h' = \omega_h \sqrt{1 + \frac{B_m K_{ce}}{D_m^2}} \quad \text{and} \quad \delta_h' = \frac{\delta_h}{\sqrt{1 + \frac{B_m K_{ce}}{D_m^2}}}$$

In order to study the stability of the servo system the ***open loop gain*** $A_u(s)$ must be analysed. Figure 5-3 yields

$$A_u(s) = \frac{K_{sa}K_{qi}K_f / D_m}{\left(1+\frac{s}{\omega_v}\right)\cdot s \cdot \left(\frac{s^2}{\omega_h^2}+\frac{2\delta_h}{\omega_h}s+1\right)} = \frac{K_v}{\left(1+\frac{s}{\omega_v}\right)\cdot s \cdot \left(\frac{s^2}{\omega_h^2}+\frac{2\delta_h}{\omega_h}s+1\right)} \quad (5\text{-}6)$$

$\boldsymbol{K_v}$ expresses the steady state loop gain and the value of this parameter must be set to a certain level to make sure that the control system will be stable.

5.2 Valve controlled motor for an angular velocity servo

A valve controlled motor is often used for velocity (shaft speed) control. If an integrating amplifier is used in a velocity servo the loop gain $A_u(s)$ will be in principle the same as for a position servo with proportional control. Such a velocity servo is shown in **Figure 5-4**.

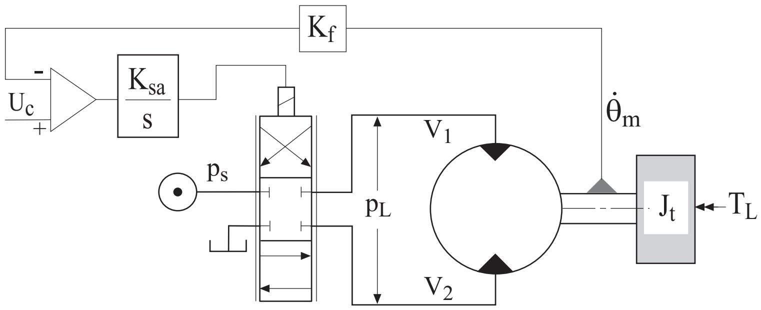

Figure 5-4: Angular velocity servo

A block diagram of the velocity servo is shown in **Figure 5-5.** An integrating amplifier means that the control error will be integrated and the steady state control error becomes zero.

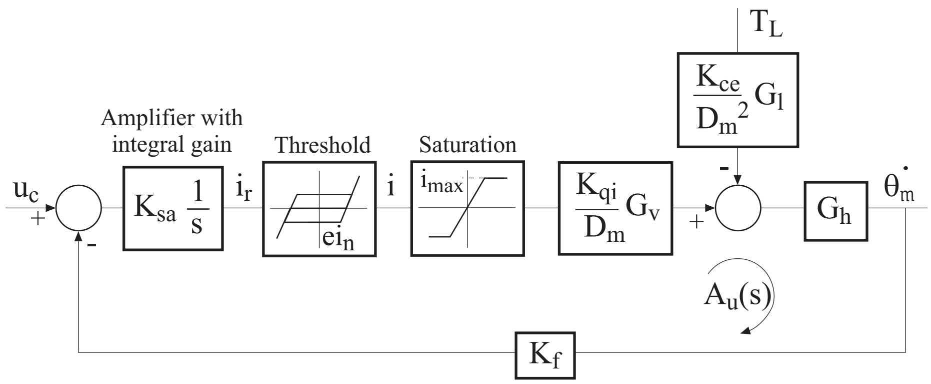

Figure 5-5: Angular velocity servo

The transfer functions in the above block-diagram are:

$$G_v(s) = \frac{1}{\frac{s}{\omega_v}+1} \quad , \quad G_1(s) = 1 + \frac{V_t}{4\beta_e K_{ce}} s \quad , \quad G_h(s) = \frac{1}{\frac{s^2}{\omega_h^2} + \frac{2\delta_h}{\omega_h} s + 1}$$

In order to study the stability of the servo system the ***open loop gain*** $A_u(s)$ must be analysed. Figure 5-5 yields the same open loop gain as for the position servo, shown in equation (5-6). Therefore, the design criteria for stability will be the same as for a position servo with proportional control.

By using an integrating servo amplifier the steady state stiffness will be considerably increased. The integration of the velocity error will cancel out the disturbance. The ***stiffness*** of the closed loop system describes the controlled signal deflection $\Delta\dot{\theta}_m$ due to variations in the disturbance torque ΔT_L. By setting $U_c = 0$ in the block-diagram in Figure 5-5 and if threshold and saturation are neglected the new block-diagram becomes as in **Figure 5-6**.

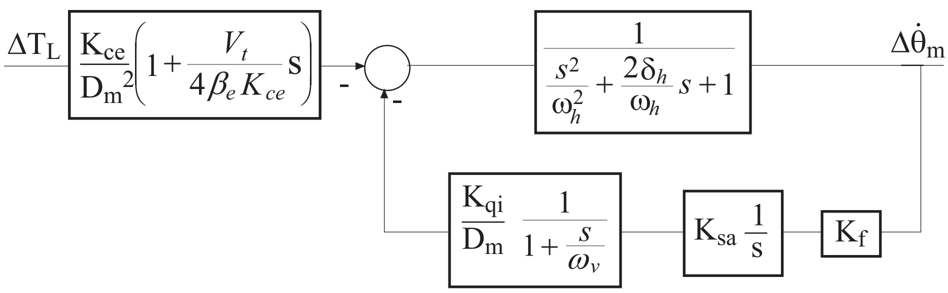

Figure 5-6: Block diagram for stiffness calculation of a velocity servo with integrating controller

The stiffness of the closed loop servo is defined as $S_c = \frac{-\Delta T_L}{\Delta\dot{\theta}_m}$. Neglecting the valve dynamics ($G_v = 1$) the block diagram in Figure 5-6 gives the closed loop stiffness as

$$S_c = K_v \frac{D_m^2}{K_{ce}} \cdot \frac{\frac{s^3}{K_v \omega_h^2} + \frac{2\delta_h}{K_v \omega_h} s^2 + \frac{s}{K_v} + 1}{s \cdot \left(1 + \frac{V_t}{4\beta_e K_{ce}} s\right)} \approx K_v \frac{D_m^2}{K_{ce}} \cdot \frac{\left(\frac{s}{K_v}+1\right) \cdot \left(\frac{s^2}{\omega_h^2} + \frac{2\delta_h}{\omega_h} s + 1\right)}{s \cdot \left(1 + \frac{s}{2\delta_h \omega_h}\right)} \qquad (5\text{-}7)$$

where the steady state loop gain $K_v = K_{sa} K_{qi} K_f / D_m$.

The ***steady state stiffness*** is defined as $\left|S_c\right|_{s\to 0} = \left|\frac{-\Delta T_L}{\Delta\dot{\theta}_m}\right|_{s\to 0}$. Equation (5-7) gives

$$\left|S_c\right|_{s\to 0} = \left|K_v \frac{D_m^2}{K_{ce}} \cdot \frac{1}{s}\right|_{s\to 0} = \infty \qquad (5\text{-}8)$$

From equation (5-8) it can be noted that the steady state stiffness goes to infinity because of the integrating controller. It has to be observed that this is true only at very low frequencies (steady state conditions). In practical applications the steady state stiffness also will be limited by the resolution of the velocity transducer.

5.2 Pump controlled motor

Pump controlled motors are the preferred power element in applications which require considerable horsepower for control purposes. This type of closed hydrostatic transmission gives much higher efficiency compared to a valve controlled actuator, since there are no flow orifices in the main circuit. However, the comparatively slow response of the pump displacement controller, limit their use in high performance systems. In **Figure 5-7** a pump controlled motor used as an angular position servo is shown.

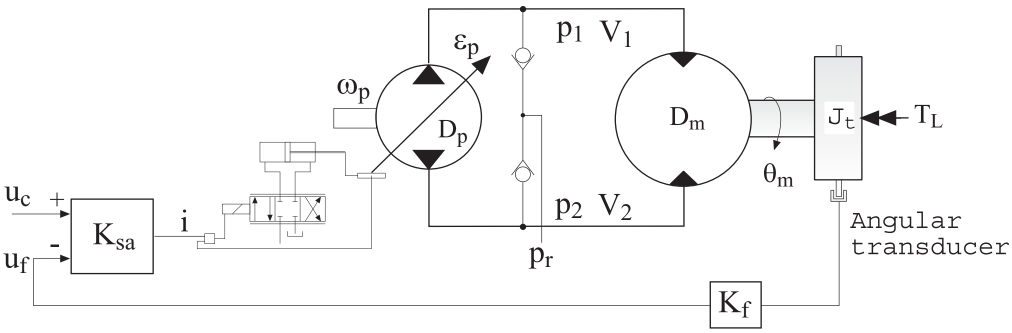

Figure 5-7: Pump controlled motor used as an angular position servo

As shown in the figure the pump displacement setting (ε_p) is controlled by a position servo. During normal operation the pressure in one line between pump and motor will be at replenishing pressure (p_r) and the other pressure will modulate to match the load. The two lines will switch functions if the load dictates a pressure reversal. It is possible for both line pressures to vary simultaneously if transients are rapid and load reversals occur. However, for system modelling it is assumed that only one pressure varies at the same time and that both sides are identical.

If it is assumed that p_1 in Figure 5-7 denotes the high pressure the continuity equation for the volume V_1 (constant volume) is

$$\Delta\varepsilon_p D_p \omega_p - C_{tp}\Delta P_1 - C_{tm}\Delta P_1 - D_m s\Delta\theta_m = \frac{V_1}{\beta_e} s\Delta P_1 \qquad (5\text{-}9)$$

Introducing the total leakage coefficient for pump and motor, $C_t = C_{tp} + C_{tm}$ gives

$$\Delta\varepsilon_p D_p \omega_p - C_t\Delta P_1 - D_m s\Delta\theta_m = \frac{V_1}{\beta_e} s\Delta P_1 \qquad (5\text{-}10)$$

Assuming lumped constants to describe the load, Newton's second law is used to obtain the torque balance equation for the motor shaft. If, the friction torque is described only by a viscous friction coefficient, B_m the torque equation is given as

$$D_m(\Delta P_1 - \Delta P_r) = [P_r = constant] = D_m\Delta P_1 = J_t s^2\Delta\theta_m + B_m s\Delta\theta_m + \Delta T_L \qquad (5\text{-}11)$$

The transfer function for the pump displacement controller from input current (i_v) to displacement setting (ε_p) is

$$\frac{\Delta\varepsilon_p}{\Delta i_{ps}} = K_{ps}G_{ps} = K_{ps}\frac{1}{1+\dfrac{s}{\omega_{ps}}} \qquad (5\text{-}12)$$

where K_{ps} is the proportional gain for the pump servo and G_{ps} is a low-pass filter describing the dynamics of the pump servo.

Combining equation (5-10) to (5-12) gives the block diagram shown in **Figure 5-8**.

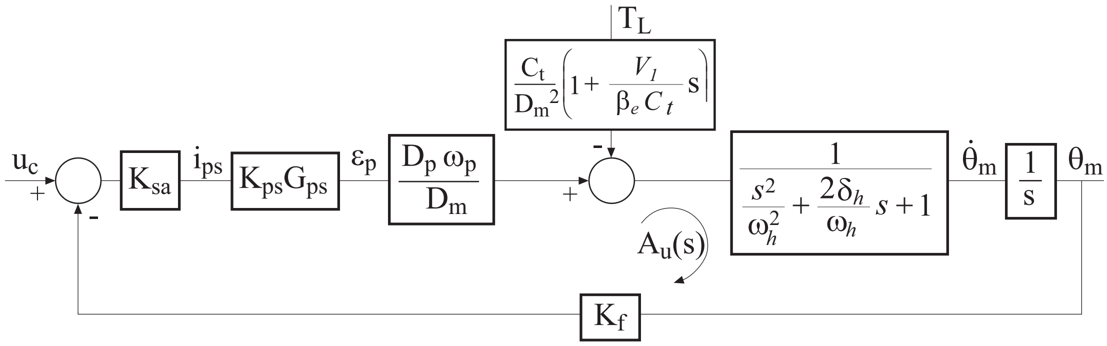

Figure 5-8: Block diagram of a pump controlled motor used as an angular position servo

If the term $B_m C_t / D_m^2$ is smaller than unit the hydraulic resonance frequency ω_h and the hydraulic damping δ_h shown in the block-diagram (Figure 5-8) will be expressed as

$$\omega_h = \sqrt{\frac{\beta_e D_m^2}{J_t}\left(\frac{1}{V_1}\right)}\ . \text{ If } V_1 = V_2 = V_0\ , \quad \omega_h = \sqrt{\frac{\beta_e D_m^2}{J_t V_0}}$$

$$\text{and} \quad \delta_h = \frac{C_t}{2D_m}\sqrt{\frac{\beta_e J_t}{V_0}} + \frac{B_m}{2D_m}\sqrt{\frac{V_0}{\beta_e J_t}}$$

If the term $B_m C_t / D_m^2$ is included the resonance frequency and damping becomes

$$\omega_h' = \omega_h\sqrt{1+\frac{B_m C_t}{D_m^2}} \quad \text{and} \quad \delta_h' = \frac{\delta_h}{\sqrt{1+\dfrac{B_m C_t}{D_m^2}}}$$

In order to study the stability of the servo system the ***open loop gain*** A_u(s) must be analysed. Figure 5-8 yields

$$A_u(s) = \frac{K_{sa}K_{ps}D_p\omega_p K_f / D_m}{\left(1+\dfrac{s}{\omega_{ps}}\right)\cdot s\cdot\left(\dfrac{s^2}{\omega_h^2}+\dfrac{2\delta_h}{\omega_h}s+1\right)} = \frac{K_v}{\left(1+\dfrac{s}{\omega_{ps}}\right)\cdot s\cdot\left(\dfrac{s^2}{\omega_h^2}+\dfrac{2\delta_h}{\omega_h}s+1\right)} \qquad (5\text{-}13)$$

$\boldsymbol{K_v}$ expresses the steady state loop gain and the value of this parameter must be set to a certain level to make sure that the control system will be stable. In this system it will be noticed that both the hydraulic resonance frequency and its damping are reduced because of the fact that the pressure is varying in only one volume between pump and motor.

5.3 Pump controlled symmetric cylinder

It is possible to replace the motor in Figure 5-8 with a symmetric cylinder, without any changes in the rest of the system. In that case we will have a closed hydrostatic transmission with a linear actuator, which is shown in **Figure 5-9**.

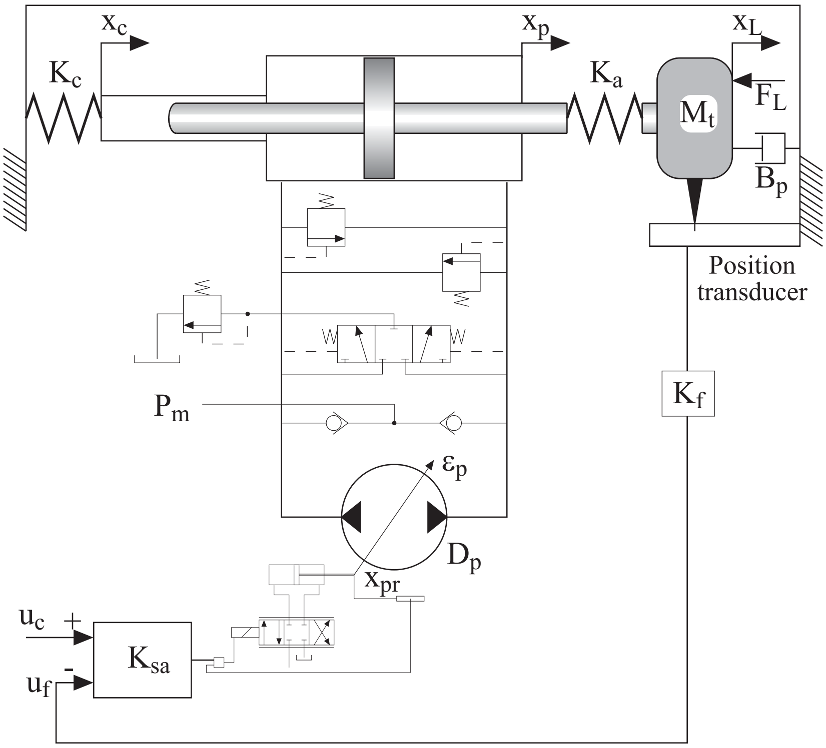

Figure 5-9: Pump controlled cylinder used as a linear position servo

Figure 5-9 also illustrates the valves needed to control maximum level of the high pressure (pressure relief valves) and to maintain constant low pressure (cooling valve). The dynamics of this system will be similar to that described in Figure 5-8.