



Chapter 2

Hydraulic energy control, conductive part

Motion Control in Offshore and Dredging

Hydraulic energy control, conductive part

To get the hydraulic energy generated by the hydraulic pump to the actuator, cylinder or hydraulic motor in a controlled way, more than just pipe work is needed. We call this the conductive part of hydraulic drive system. In this chapter we will discuss the most important basic functions of the functional valves that are applied for this purpose. Detailed information about the valves is available from the often very detailed specification and documentation of the different manufacturers.

2.1 Pressure control valves

Pressure control valves are used to regulate the pressure. As far as the construction is concerned, a distinction is made between a spool and a seat construction.

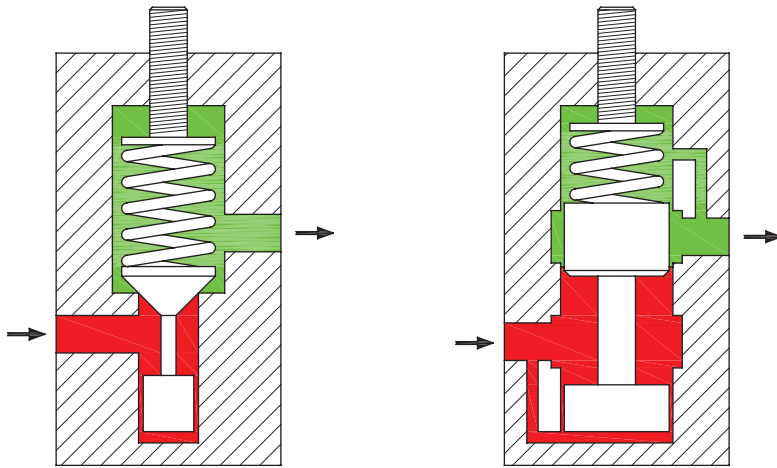


Fig 2.1 Seat type (left) and piston type (right) relief valve

2.1.1 Pressure Relief Valves

A pressure relief valve must limit the system pressure in a hydraulic installation to a maximum value. If the system pressure exceeds this value, then the pressure relief valve opens and lets oil flow from the pressurized inlet side to the return line.

Direct Acting Pressure Relief Valve, see figure 2.1.1.A

The seating valve (3) is pressed into the seat by the spring (2). The inlet pressure P presses underneath the pressure valve against the spring pressure. If the pressure rises above the pressure set in the spring, the valve will open to discharge the oil via the outlet port T .

The pre-tension in the spring can be set progressively. This type of valve is not suitable for large volumes of fluid. This valve is absolutely leak proof if its inlet pressure is lower than the set value.

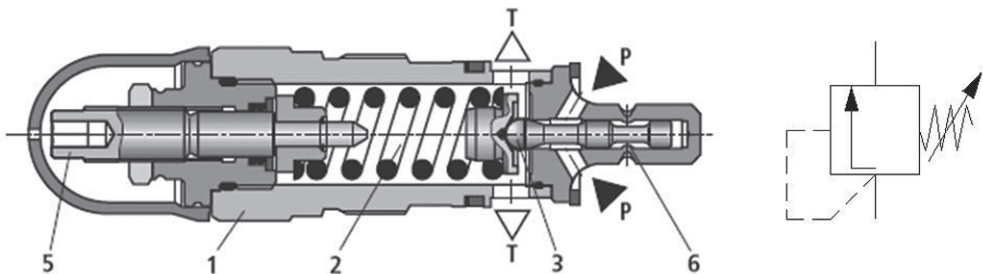


Fig 2.1.1.A Direct acting pressure relief valve with its symbol (Courtesy of Bosch Rexroth)

Pilot Operated Pressure Relief Valve, see figure 2.1.1.B

The pilot operated pressure limiter valve is suitable for large fluid flows, because this valve consists of a main valve and a pilot (indirect) valve.

The main valve consists of the cartridge (3) which is pushed into the seating by a spring. The inlet pressure P works on the underside of the main valve. At the same time it works on the topside of the main valve, but this time via the choke (7). The main valve remains closed, because of the balanced forces working on it. However, when the inlet pressure P rises above the level set in the pilot valve, the pilot valve (8) will open, thus lowering the pressure at the top side of the main valve. The main valve will open because the forces working on it are no longer in balance. Due to the design of the valve, a large flow capacity is possible.

This type of valve is not leak proof due to leakage via the pilot section to the spring chamber and via the outer circumference of the main piston to the return line.

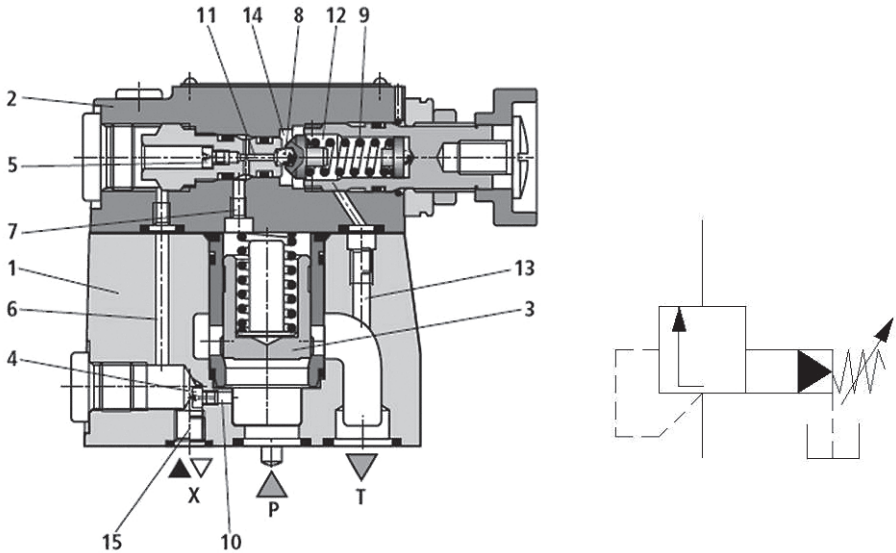


Fig 2.1.1.B Pilot operated pressure control valve with its symbol (Courtesy of Bosch Rexroth)

2.1.2 Pressure reducing valve

A pressure reducing valve, figure 2.1.2, is used to reduce the pressure in (part of) the system on the exhaust side of the valve to a certain, pre-determined, value. Here too we have so-called direct acting and pilot operated valves. As before, the direct acting valves have a limited flow capacity.

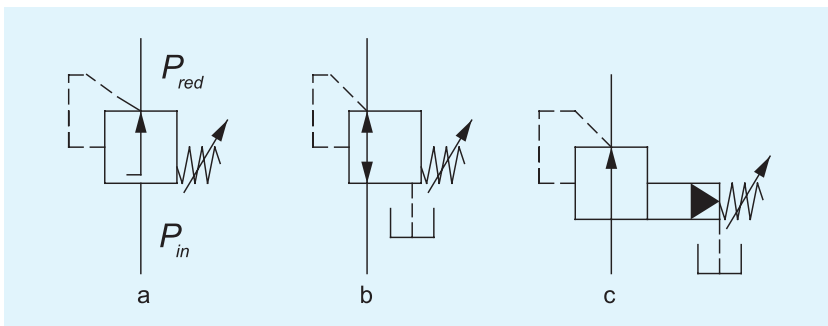


Fig 2.1.2 Pressure reducing valves: a) direct acting pressure reducing valve, b) relieving pressure reducing valve and c) pilot operated pressure reducing valve

A relieving pressure reducing valve is, for example, used when the pressure in a cylinder rises above the set value P_{red} of the pressure reduction valve as the piston is pushed into the cylinder. The higher pressure can be reduced through the valve via the correction outlet (often the T- or Leak oil pipe).

2.1.3 Pressure sequence valve

A pressure sequence valve opens when the inlet pressure rises above the value set for the valve. When this happens, the valve opens completely and connects the inlet side with the exhaust/outlet side, which means that the circuit on the exhaust side will be at the same pressure as the circuit on the inlet side of the valve.

In the example as shown in figure 2.1.3. the left cylinder will first move out and, when the pressure in that cylinder rises above the set value of the sequence valve, the sequence valve will open and the right cylinder will move out.

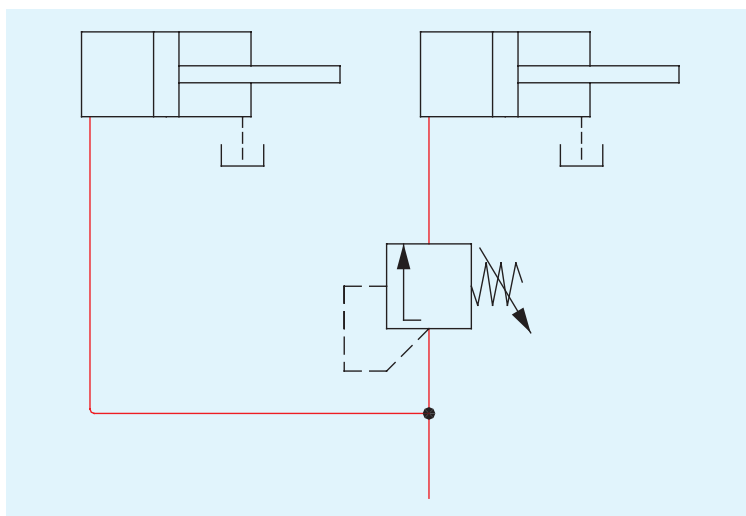


Fig 2.1.3 A pressure sequence valve to control a second cylinder

For these valves too there is a choice between direct working and piloted valves. The flow capacity of the direct working valves is again limited.

2.1.4 Brake (counter balance) valve

A variant on the sequence valve is the brake or counter balance valve. Two names for valves with nearly the same function but with different features. In paragraph 7.1 and 7.2 details are given for the function and features of the counter balance valve and the braking valve. With both valves it is possible to keep an actuator under control where an external load is applied and the actuator is driven by that load, as in the shown situation in figure 2.1.4. These valves have an external pilot control line to open the valve when the actuator has to be driven, in this case the cylinder has to be retracted.

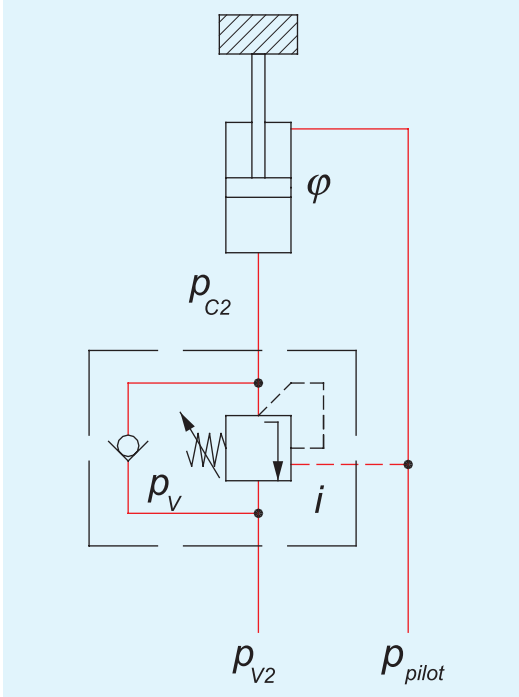


Fig 2.1.4 Function of the counter balance and brake valves with a cylinder.

With the cylinder vertical mounted the weight of the mass initiates a pressure p_L at the bottom area. The counterbalance features a relief function. The setting of this relief valve should be at a value of 130% of the maximum induced pressure p_L . If a pressure is applied at the annular end port two effects will be noticed. At first the pressure at the bottom area will increase due to the force balance on the cylinder piston. The second effect is that the counterbalance valve is gradually opened by this pressure at the annular end via the pilot pressure line.

The equation for the static pressures and dynamic behavior with the use of a counterbalance valve is in detail explained in equations 7.5. The result of the equations for the static behavior is given by:

$$p_{C2} = p_L + (p_V + p_{V2} - p_L) \cdot \frac{\phi}{i + \phi} \quad (2.1)$$

and:

$$p_p = \frac{p_V + p_{V2} - p_L}{i + \phi} \quad (2.2)$$

From both equations we find that the pressure p_{C2} and the pilot pressure p_{pilot} are sensitive for back pressure of the valve p_{V2} . Optional valves are available that are non-sensitive to this back pressure. This is than achieved by venting the chamber of the main relief mechanical spring.

2.2 Directional valves

With directional valves it is possible to regulate the start, stop and direction of the fluid flow.

There are two construction groups, identical to the basic types of pressure regulator valves:

- Seating valves, which are more or less leak free. The necessary force required for switching is large because the valve has to be opened against the pressure in the spring.
- Spool valves, where a spool is controlled in different positions by springs and hydraulic pressure.

Because of the small clearance between the spool and the housing, there is always some leakage from blocked ports to other ports. In the example of the 4/3 valve as shown below leakage occurs from the pressure inlet port P to the working ports A and to B, but also from the inlet port P to the tank line port T and from the working ports A and B to the tank line port T.

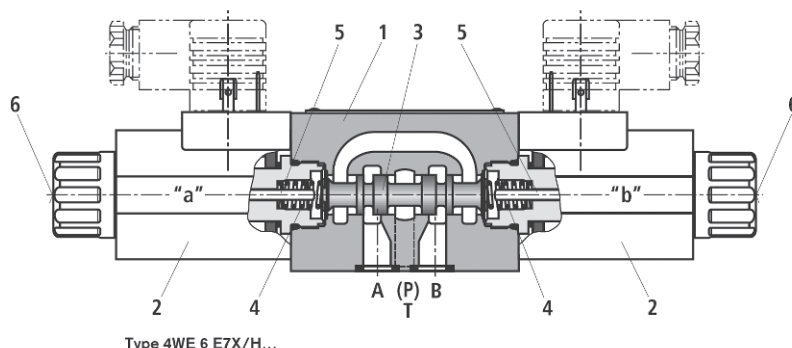


Fig 2.2.A Directly operated directional spool valve (Courtesy of Bosch Rexroth)

In this valve are two magnets a and b that can move the spool to the left or to the right. That way the central connection P is connected with either port A or port B. At the same time port A or B might be connected with port T. The size of the valves is specified according to a CETOP norm in sizes: 03, 05, 07, 08 and 10. The mounting sizes of the valves are also all standardised, which makes it possible to exchange a valve of a certain size and make with one of the same size but a different make.

The naming of the pilot valves is made up of the number of working ports (control ports not included) followed by the number of control positions.

Three examples are given in figure 2.2.B

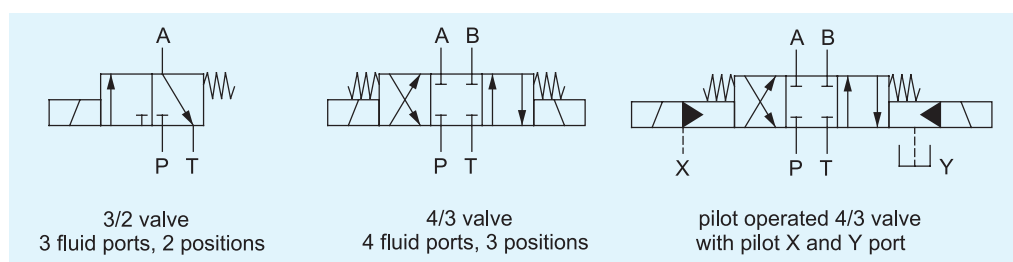


Fig 2.2.B Naming of the directional control valves, depending on number of fluid ports and control positions. P = pressure port, T = tank port, A and B = working ports and X and Y are pilot ports

The idle position is the position of the valve when no controls are applied to it. The fluid ports are indicated in this state. In the above examples a mechanical spring controls the spool to idle position. The working ports A and B and P and T port are designated by capital letters. For a pilot operated directional valves pilot ports X and Y are added. X is the control port with a certain pilot pressure and Y is the port that is always connected to a low pressure line e.g. the tank line or a separate drain line. From the idle position of the valve it is possible to reach several different switch symbols, see figure 2.2.C., representing

the most used basic symbols. The three different symbols can be achieved by mounting different spools in the valve body.

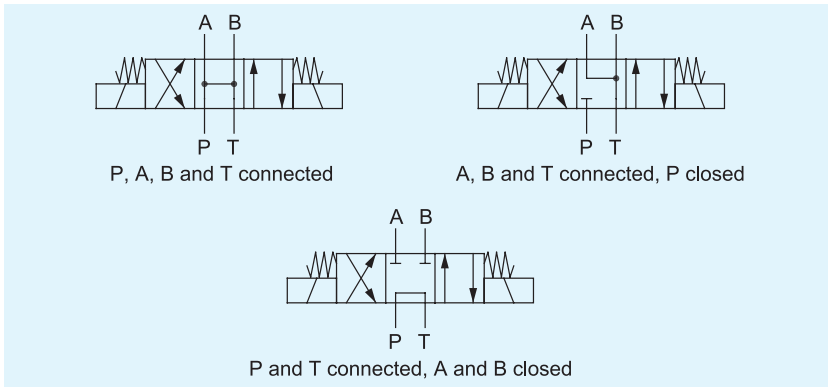


Fig 2.2.C Different flow paths in a directional valve with different spool types

The maximum flow through a valve of a certain size is limited. For a CETOP 3 valve this limit is about 30 lpm. The flow through the valve generates a force on the spool that is opposite to the operating force of the magnet. The flow forces intend to move the spool to the central position. At the same time the pressure drop across the spool increases, for example between P and A, at a rate relative to the square of the flow rate.

The maximum working pressure for currently available directional valves is about 350 bar.

The limitations on the flow rate can be largely overcome by fitting a so-called pilot valve. In that case a directly controlled CETOP 3 valve is mounted directly onto a large main valve. The connections between ports A and B and the pilot valve are used to move the much larger control spool in the main valve from left to right. This significantly increases the maximum flow allowed. For a CETOP 7 valve it goes up to about 150 lpm, for a CETOP 10 valve up to about 800 lpm.

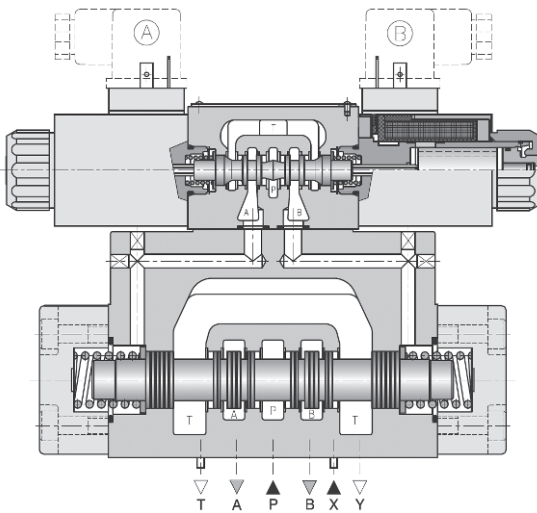


Fig 2.2.D Pilot operated directional control valve (Courtesy of Argo-Hytos)

In the pilot controlled valve the pilot valve uses the pressure from port P or from port X to regulate the position of the main spool. If there isn't pressure on port P in all situations, for example if port P is connected to port T in the idle position, then it will be possible to use the external control pressure from port X to switch the valve.

2.3 Flow valves

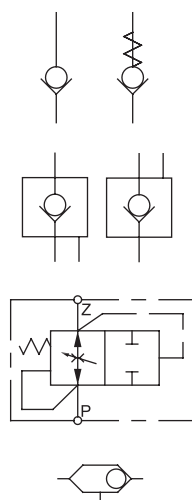
2.3.1 Non-return (check) valves

The function of check valves is to let oil flow through the valve in one direction only. There are unloaded and spring loaded valves.

A controlled check valve can be opened against the flow or closed with the flow by a separate pilot control line. In a number of cases the pilot check valves are fitted with an external oil leak connection.

In a hose burst protection valve, the flow through the valve from Z to P can only take place during a controlled movement of the cylinder. When the flow is too large, the valve closes, stopping the flow.

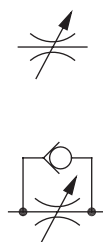
A shuttle valve has two inlet ports and one outlet port. The inlet port with the highest pressure allows flow to the exit port whilst, at the same time, closing the other inlet port. This valve is used to pass on the pressure of two inlet ports to, for example, the controls of a load sense pump.



2.3.2 Throttle valves

In a variable throttle valve the opening area can be varied step less. The volume flow is amongst other parameters dependent on the pressure drop across the valve, see for details paragraph 2.4.5.

To make sure that the speed of an actuator is restricted only in one direction a check valve is fitted. This combination is also known as a speed control valve or throttle and check valve.



Shut-off (isolating) Valve

A shut-off valve is used to close a hydraulic pipe, not to choke it. The symbol shows the valve in open position. When the valve is closed the symbol is drawn solid.



Flow control valve

The volume flow is independent from the pressure difference across the valve, which means that the flow at the outlet point stays constant, even if the pressure at the inlet or outlet points changes. For the valve to work properly a minimum pressure drop of 7-14 bar is necessary across the valve.

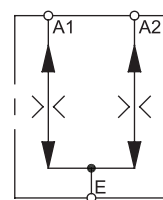


There are also flow control valves with temperature compensation.

Flow distribution valve

A flow distribution valve splits the volume flow into equal proportions. The volume flows stay equal, not dependent on the outlet pressures. The pressure difference across the restrictors is kept the same, which ensures that the flow volumes remain the same too. Both outlet flows are never exactly the same. Differences of 3-10% are normal.

This valve can also work in the opposite direction, in the sense that two equal flows can be accepted.



2.4 Proportional and servo valves

2.4.1 General

The varying of the volume flow to/from a hydraulic cylinder is called proportional flow control. The term control is actually wrong. No feedback of the achieved flow speed takes place. This means that this is not a case of control in line with 'control theory'.

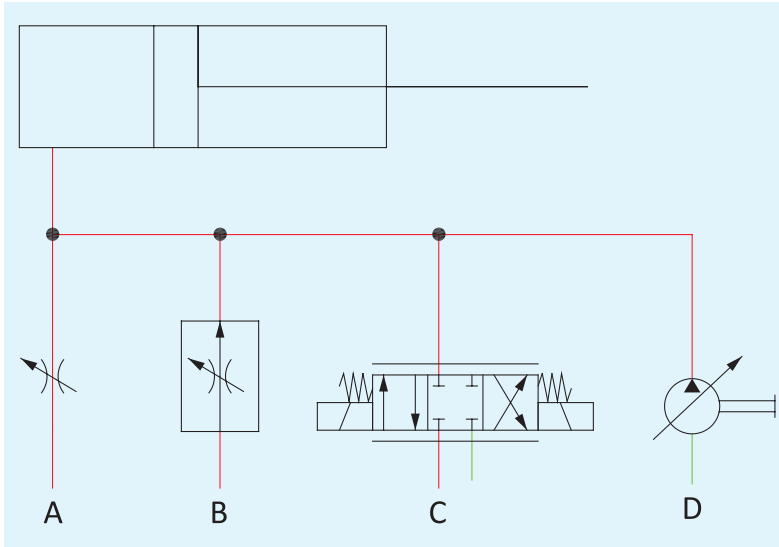


Fig 2.4.1 Proportional flow control in different ways

A proportional volume control can be achieved in several different ways. Figure 2.4.1 displays a number of options. The simplest one is the variable throttle (A), followed by the 2-way flow regulator (B), the proportional valve (C) or the variable pump (D).

2.4.2 Proportional controls

If a proportional valve is used to control a cylinder then two variable choke control valves are actually used (see figure 2.4.2.A). For example, the valve is controlled and a variable choking between inlet port P and outlet port A develops. In that case a variable choke develops from port B to port T where the passage size from P to A is equal to the passage size from B to T.

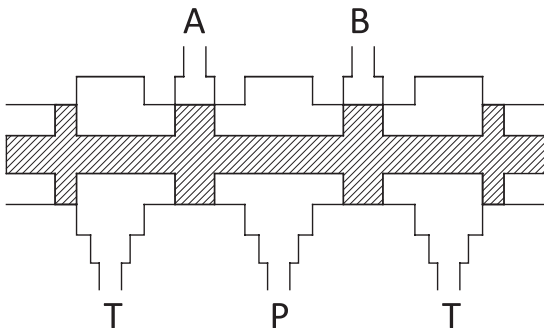


Fig 2.4.2.A 2 Chokes are always active in a proportional valve

In figure 2.4.2.B both the cylinder surface and thus the input and output volume flow are the same. Because the passage size of choke 1 is equal to choke 2 and because the volume flow Q_A is equal to volume flow Q_B the pressure drop ΔP_1 must be the same as the pressure drop ΔP_2 .

* There are also designs with a volume flow ratio of 2:1 for use with a differential cylinder ($q=2$), see paragraph 2.4.5

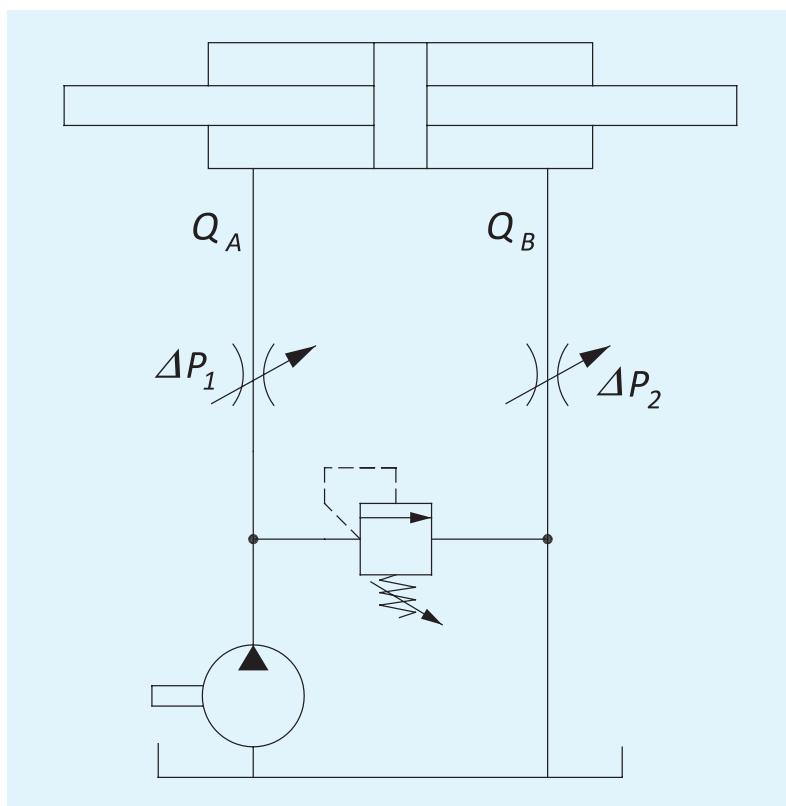


Fig 2.4.2.B A proportional valve is presented by two variable chokes

The pressure drops ΔP_1 or ΔP_2 are dependent on the bores of chokes 1 and 2. Because we are dealing with a proportional valve, the choke bore is variable. The maximum bore size is dependent on the CETOP size of the valve. The volume flow capacity of a valve is expressed in the nominal volume flow Q_n . This is the volume flow in lpm which will occur from P to A or from B to T with a 'standard' pressure drop across a full opened choke of $\Delta P = 5$ bar. In the manufacturer's documentation the pressure drop is defined by a total pressure drop across the valve of 10 bar. In this case however, they take the pressure drop from P to A + the pressure drop from B to T = 5 + 5 = 10 bar.

A proportional valve can also be used to control a drive system. In order to achieve control, a feedback takes place of a position, a speed or a force. In these cases a higher accuracy level is demanded for the valves. In the air and space industries very accurate valves have been developed for this purpose with the special name of 'servo valve'. A servo valve is in effect also proportional. Historically the standard volume flows for a servo valve were and are set at a pressure drop of 35 bar across the choke. This gives a total of 70 bar across the inlet and outlet choke. Accuracy wise and dynamic, a large difference exist between a standard proportional valve and a servo valve. Valves have now been developed which, accuracy wise, fill the gap between the proportional valve and the servo valve.

2.4.3 Higher pressure drop across the valve ports

The pressure drop across the valve can however be many times larger than the '5 bar' or '35 bar' mentioned earlier. In the example shown in figure 2.4.3.A the external load on the cylinder is F . This force requires a pressure drop across the cylinder of, for example, 50 bar. If the pressure in the limiter valve is set to 100 bar then 50 bar is 'left' for the proportional valve. This leaves 25 bar for each of the variable ports of the proportional valve.

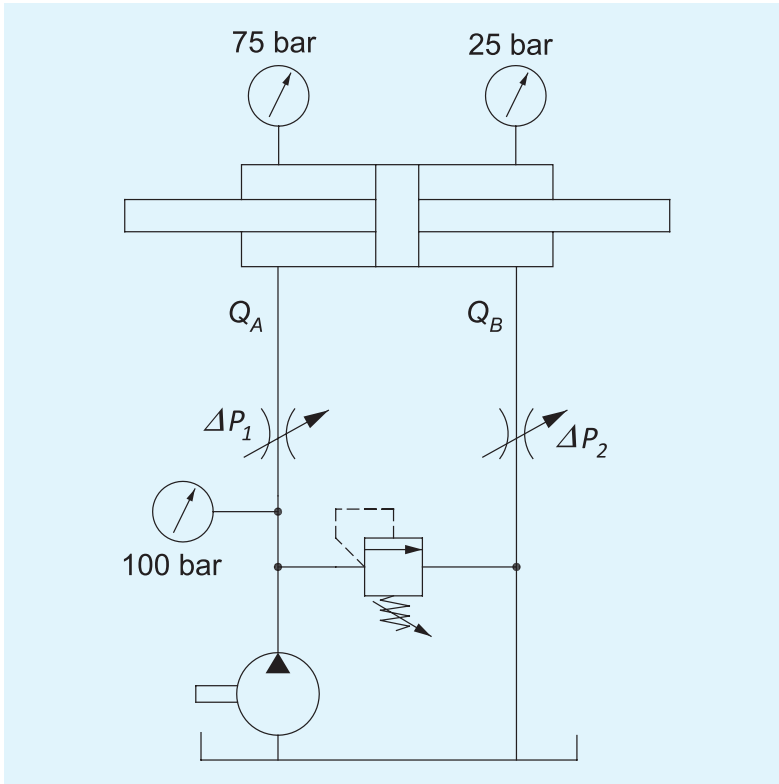


Fig 2.4.3.A Pressures in the circuit when using a proportional valve

The actual volume flow Q_{act} when the valve is fully open is then larger, as per the formula. This means that the pump needs to have sufficient capacity to supply this volume flow.

$$Q_{act} = Q_n \times \sqrt{\frac{\Delta P_{1,2}}{5bar}} \quad (2.3)$$

Where

Q_n = nominal flow

Q_{act} = actual flow.

The flow through the valve for a certain pressure drop relative to the input signal to the valve is called the "volume amplifier". Most manufacturers provide these graphs in their documentation, see figure 2.4.3.B. The diagram on the left is for a proportional valve, where the standard pressure drop of 5 bar across the choke is used. The diagram on the right is for a servo valve, where the standard pressure drop of 35 bar across the choke is used.

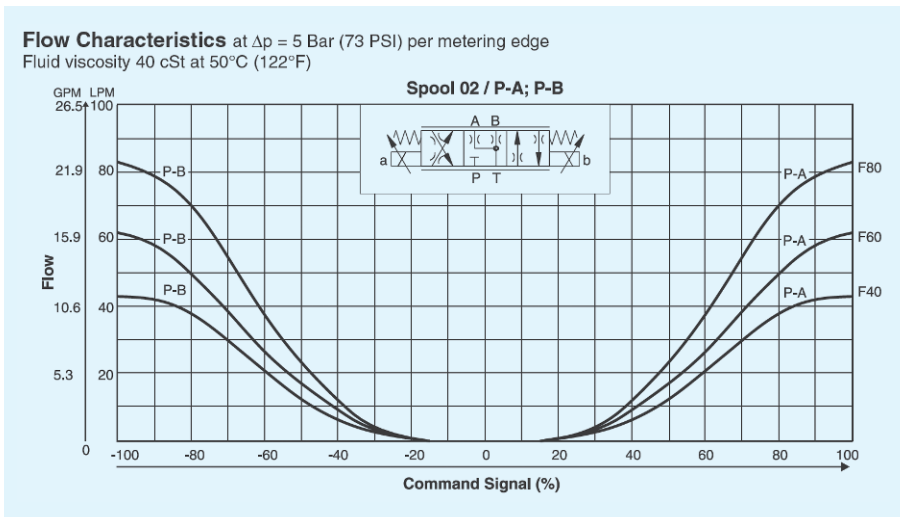


Fig 2.4.3.B Flow of proportional valves as function of command value for different spool types and pressure drops (Courtesy of Parker)

2.4.4 Performance curve for the proportional valve

The flow through a proportional or a servo valve is limited by the force of the magnets that control the valve or, in the case of a pilot valve, the force in the pilot. A large volume stream requires a larger piston force. The manufacturer indicates the volumetric capacity in a so-called “Performance Curve”. Figure 2.4.4 shows an example of this type of graphs. The graphs for type F40, F60 and F80 are for different types of spools in the valve. The valve will be able, for a certain inlet pressure at the P port, to deliver a certain volume flow as long as the point of operation is at the left side of the graph line.

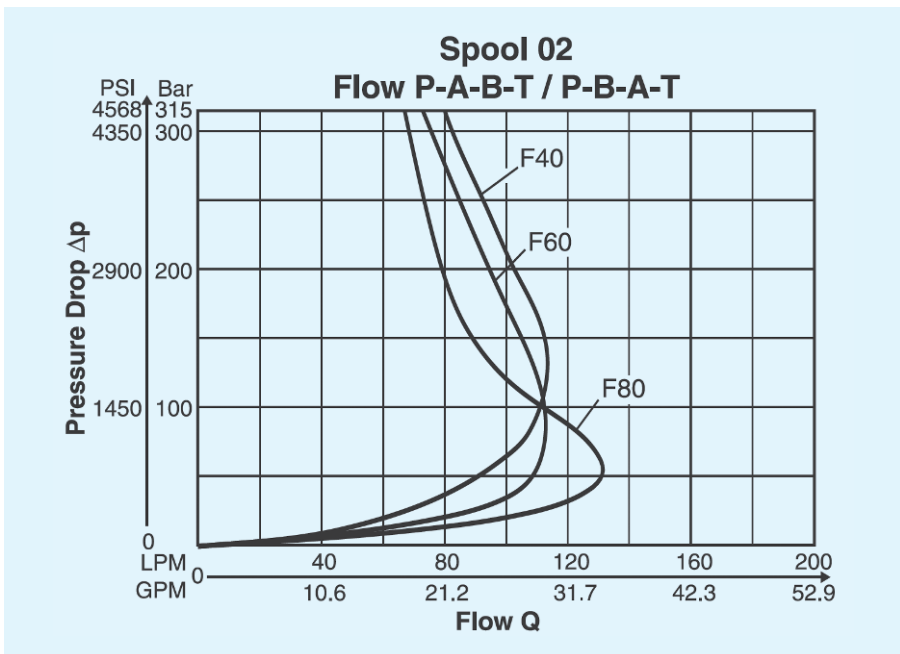


Fig 2.4.4 Performance curve for a proportional valve (Courtesy of Parker)

2.4.5 The asymmetrical spool

The pressure drop across variable chokes of proportional valves mentioned in earlier paragraphs are, to a large extent, determined by the volume flow through a valve. The volume flow and thus also the pressure drop for a cylinder with different piston surfaces can increase considerably.

For large volume flows this large drop in pressure also means a large loss of energy. Energy losses in a hydraulic installation are always converted directly into heat. The temperature of this type of installation can therefore rise quickly unless a sufficiently large oil cooler has been installed.

To get around this problem, a so-called asymmetrical spool is installed. For this type of spool, the bore for port P to A or port A to T is always twice the size of port P to B or port B to T. Such spools are therefore only applied for larger ports (with flow capacity from 80 lpm up to 600 lpm).

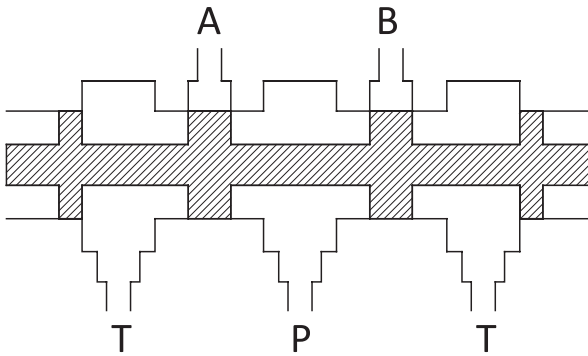


Fig 2.4.5.A Asymmetrical spool with larger port size P to A and A to T

The pressure drop across a choke is given by:

$$Q = c \cdot A \cdot \sqrt{\frac{2 \cdot \Delta P}{\rho}} \quad (2.4)$$

or

$$\Delta P = \frac{\rho}{2} \cdot \left(\frac{Q}{c \cdot A} \right)^2 \quad (2.5)$$

where

A = surface area of the port.

From this last formula it can clearly be seen that the pressure drop across a port increases relative to the square of the flow rate going through it. We can assume that the flow to/from the bottom side of the cylinder is twice that to/from the rod side of the cylinder. In that case, if the surface area of the choke at the bottom end always twice that of the one at the rod end is, then it is clear that the pressure drop across both ports will be the same.

Comment: The surface ratio ϕ for a cylinder is hardly ever exactly 2. For deviations from that ratio it may still be advisable to use an asymmetrical spool. The pressure drop across the ports for an asymmetrical spool with a surface ratio of 2 and a cylinder with a surface ratio of ϕ , the following pressure drops can be calculated across the ports.

$$\frac{\Delta P_2}{\Delta P_1} = \frac{4}{\phi^2} \quad (2.6)$$

Where

ΔP_1 = pressure drop for the bottom side

ΔP_2 = pressure drop for the rod side.

2.4.6 Slowing down of a load

Proportional valves are often used to slowly accelerate and decelerate a load. When a load is being decelerated, be it with a hydraulic motor, or with a cylinder, a significantly different characteristic is important. This is easiest explained through the drive of a hydraulic motor, as shown in the diagram below.

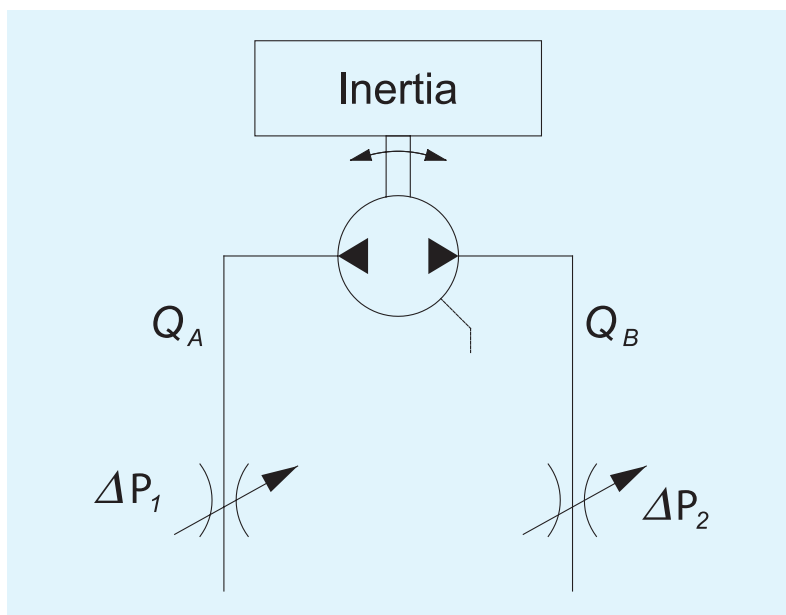


Fig 2.4.6 Deceleration of a hydraulic motor

During braking, a brake torque is required at the hydraulic motor. This can cause a high a high breaking pressure at the outlet side of the hydraulic motor, in this case a high value for ΔP_2 . This is no problem for a proportional valve. The pressure drop across the outlet port can easily be generated by moving the control spool of the valve slowly towards the closed, central position. Do remember though that, apart from a small loss due to leakage, the flow rate through the inlet port of a hydraulic motor is always equals the flow rate through the outlet port. The pressure drop ΔP_1 across the inlet port must therefore always be the same as the pressure drop ΔP_2 over the outlet port. During the deceleration of this hydraulic motor there must be sufficient pressure on the inlet side of the inlet port to achieve the pressure drop ΔP_1 . If the inlet pressure is not sufficient then “negative” pressure will occur at the inlet side, causing cavitation. A possible consequence of cavitation is mechanical damage to the hydraulic motor.

The same possibility of cavitation also occurs when a cylinder is used. There the pressure drop over the inlet port, as a result of the surface ratio of the cylinder, can easily be a factor higher than the pressure drop over the outlet port. In that situation is even more important that there is sufficient feed pressure at the inlet port of the proportional valve during braking.

2.4.7 2-Way and 3-Way pressure compensation, loadsensing

The volume flow through one of the ports of a proportional valve is amongst other things dependent on the pressure drop across the port:

$$Q = c \times A \times \sqrt{\frac{2 \times \Delta P}{\rho}} \quad (2.7)$$

If it is possible to keep the pressure drop across a port constant, then the flow Q is directly proportional to the size of the valve opening A . This constant pressure drop can be achieved by adding a so-called two-way compensator.

Such a compensator consists of a pressure control valve that is brought to a pressure balance by the hydraulic pressure on the inlet side of the proportional valve, by a mechanical spring and by the hydraulic pressure behind the proportional valve. In most cases the spring equals a pressure drop across the valve of 8 bar. A shuttle valve is used to sense the load pressure in the A or the B line. When the pressure balance is disturbed, then the pressure control valve will change position to the point where the balance has automatically restored itself.

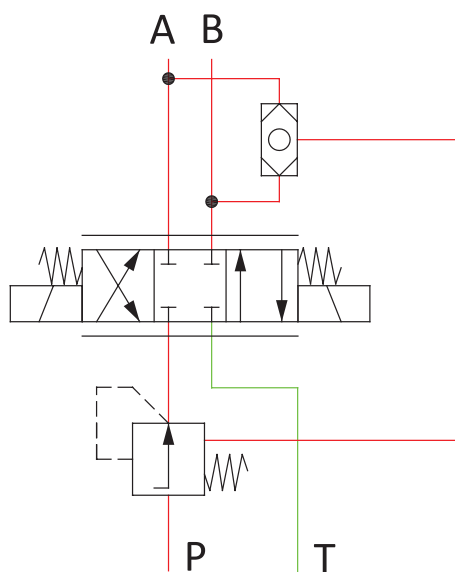


Fig 2.4.7.A A two-way pressure compensator in combination with a proportional valve

With the use of a two way pressure compensator we get:

$$P = P_A + 8 \text{ bar} \quad (2.8)$$

or

$$P = P_B + 8 \text{ bar} \quad (2.9)$$

Imagine that the pressure on the exit A or B port of the variable proportional valve rises due to for example a higher load on a hydraulic motor. This higher pressure immediately guides the pressure control valve in a direction that will open the pressure control valve further, giving a larger opening between the inlet pressure P of the two-way compensator and the inlet side of the variable proportional valve. The mechanical spring of 8 bar determines the ongoing pressure difference between the inlet and outlet side of the variable choke of the proportional valve. The pressure drop across the two-way control valve itself

varies between a minimum of about 8 bar up to almost the maximum value of the feed pressure (often a constant pump pressure). This means that the two-way pressure regulator can also cause large energy losses. Several two-way pressure regulators with proportional valves can be connected in parallel for multiple actuators.

The disadvantage of a two-way pressure compensator is that the drive tends to behave in an unstable way more easily (due to oscillating movements).

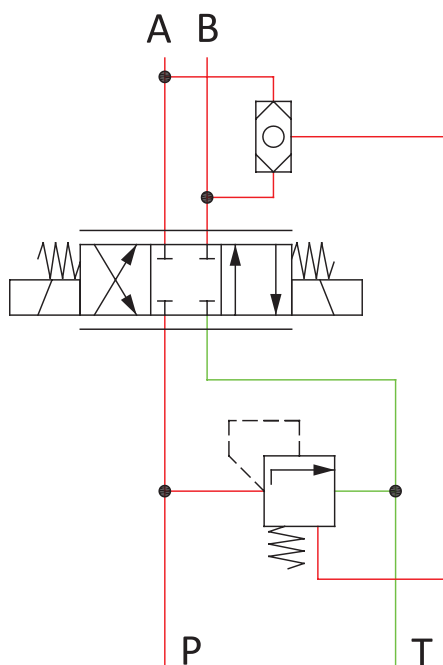


Fig 2.4.7.B A three-way pressure compensator in combination with a proportional valve

An often used variant to the two-way compensator is the so-called three-way compensator. In this case the pressure control valve also regulates the pressure at the inlet side of the valve to such a value that the pressure drop across the inlet port to the A or the B side remains constant. The surplus oil is now directed towards the return line, giving a pressure drop across the three-way pressure control valve equal to the maximum load pressure plus about 8 bar. That way, the three-way pressure regulator causes considerably less energy loss compared with the two-way pressure regulator.

Be careful to note that the most common pressure settings for two- and three-way compensators are 8-10 bar. In the meantime, the nominal flow rate through a proportional valve is specified for a pressure drop of 5 bar across the port. This means that when a two-way compensator is fitted, the volume flow will be a factor of $8/5 = 1,6$ times larger.

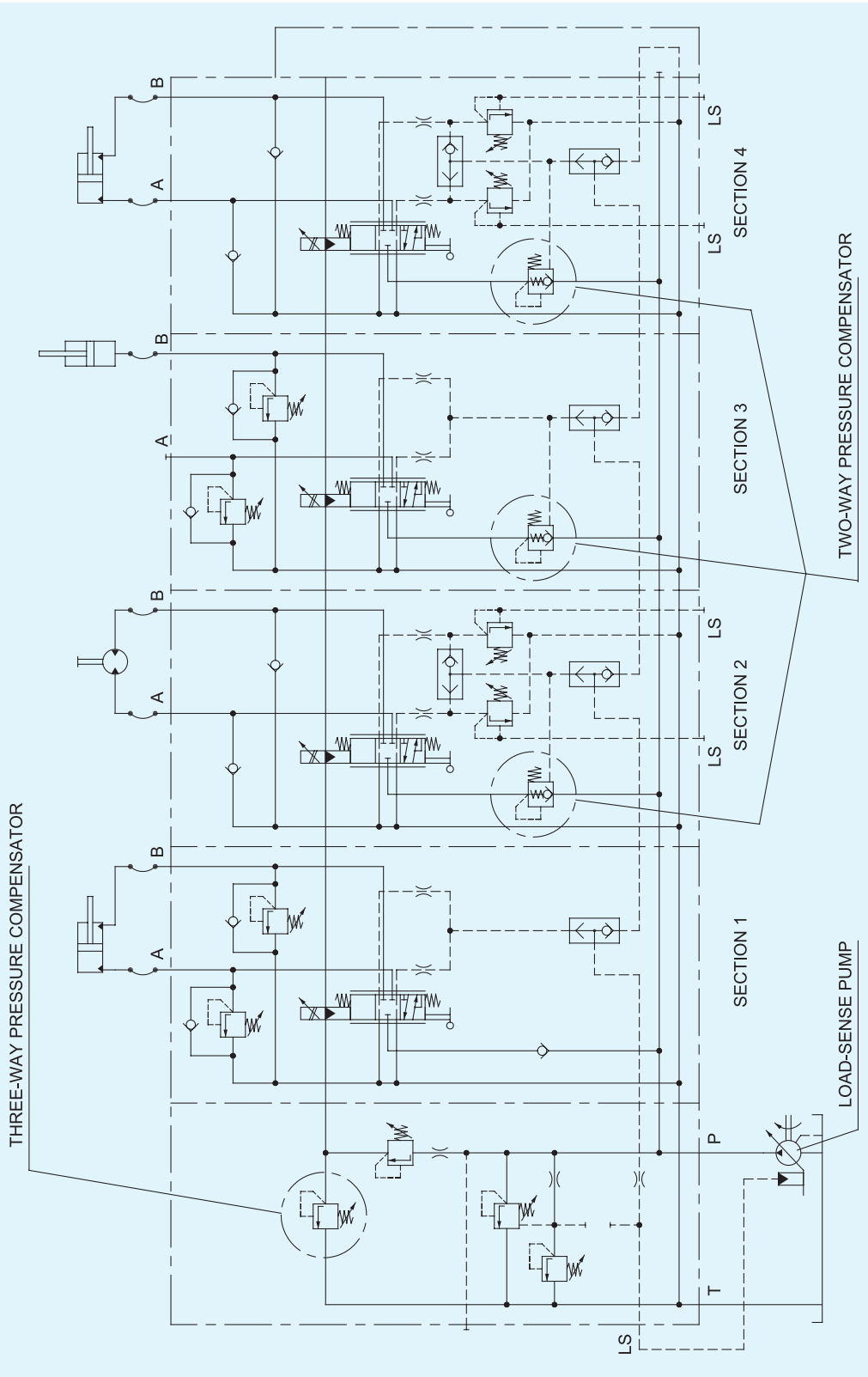


Fig 2.4.7.C A multi proportional valve as used in mobile, shipbuilding and offshore industry

Manufacturers of so-called multi body valves (valves with their sections fitted together) provide excellent examples of the design of a two-way pressure compensator. The diagram in figure 2.4.7.C shows the hydraulic schema for a multi body valve with four different proportional valves. The first section does not have a two-way compensator, the next three sections do. The second and fourth section have additionally been fitted with extra pressure control valves. This way it is possible to set the maximum secondary pressure (= the pressure after the proportional choke). When this pressure is reached, the two-way compensator closes altogether. Then it will no longer be possible for the secondary pressure to increase further. That limits the maximum pressure in the port after the proportional choke.

This example also includes, as standard, a 3-way pressure regulator. This regulator is applied when a pump with constant output is used. The surplus oil is then discharged to the tank against the highest load pressure present.

It is also possible to include a so-called load sensing control in this drawing. The highest load pressure is brought to the adjustable pump via the LS-port. The load-sensing regulator on the pump (you will need to order this specially) delivers a pressure to the P-port which is equal to the highest load pressure present, increased by about 25 bar (this is the minimum pressure drop at which the load sensing regulator can operate).

The advantage of the load sensing regulator is clear. The pump only delivers the minimum amount of energy required for the system. This also means that no discharge of surplus oil, via the pressure relief valve, to the tank takes place.

One possible disadvantage of the load sensing regulator is the risk of unstable pressure, especially with driven loads that have a low natural frequency. Another disadvantage is the relatively high minimum pressure drop of 25 bar that remains. A new development is the application of a fully electronic version of the load sensing regulator. Both the stability of the control circuit and the remaining pressure drop can be improved significantly.

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