Chapter 9

# *COMPLEMENTARY DIRECTIONAL VALVES, REMOTE CONTROL*

Some valves that do not switch the flow between the ports of actuators can be defined as directional valves because they do no control but stop or allow the fluid flow. We are going to refer to them as 'complementary directional valves' in order to distinguish them from the valves explored in chapter 8.

A number of mobile and stationary machines must be controlled or handled away from the main distributors: for this reason, remote control devices send a mechanical, hydraulic or electric signal to the directional valve that then operates the actuator.

## NON-RETURN VALVES

ISO 1219/1 standard sets the characteristics of these valves. It defines simple valves as 'check valves', valves provided with an opposing spring as 'spring-loaded check valves' and valves with an external control as 'pilot-operated check valves; these valves are commonly known as **non-return valves**.

It is important to highlight the internal sealing between the mobile element and the seat of these valves is almost perfect; they can therefore be described as *leak-free*.

### Ball or poppet check valve

The task of this type of valves is to allow a one-way flow and to prevent the opposite direction of flow (Figure 9.1). These valves are basically made up of a fixed part whose ends are connected to the hose; the mobile element (a ball or a conical poppet) slides in the internal seat. If outlet pressure (out) were higher than inlet pressure (in), the valve would not switch even in the free flow direction.

Note that the use of balls should be considered carefully because balls subjected to pressurised fluids tend to vibrate and to transmit this vibration to the whole circuit.

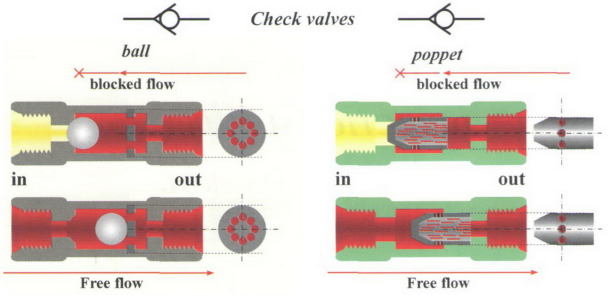



*Figure 9.1*

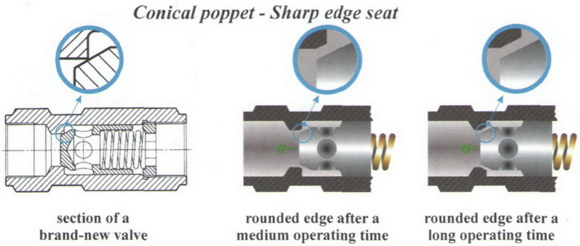



*Figure 9.1 b*

Before analysing the different valves with a conical poppet, it is worth stressing that the poppet seat in most versions does not have the same shape as the poppet, but it has a **sharp edge**, as shown in the drawing in Figure 9.1 b.

This practical measure aims at ensuring a perfect sealing despite the component ageing. As a matter of fact, if the seat was flared and had the same angle as the poppet (lead cone), the seat and the poppet would wear out in an irregular manner resulting in leakages because of the continual friction and the abrasion performed by the particles contained in the contaminated non-perfect oils. On the contrary, a sharp edge always matches the shape of the poppet as the pressurised fluid presses the poppet against it powerfully.

This remark *applies to any seat of a conical poppet*, like for instance the poppets of control valves.

### Spring-loaded ball or poppet check valve

Spring-loaded ball or poppet check valves are similar to the previous type of valve, but they are provided with an opposing spring (Figure 9.2). The pressure of the fluid must be higher than the spring force in the free flow direction. If the outlet pressure (out) were higher than the inlet pressure (in), the valve would not switch even in the free flow direction.

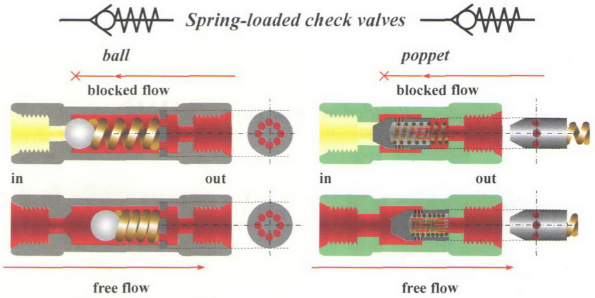



*Figure 9.2*

### Right angle check valve

The inlet/outlet ports of right angle check valves are shaped into an inverted L and they play the same role as spring-loaded ball or poppet check valves.

However, the holes on the conical poppet in this type of valves do not allow the flow from a port to another but ensure sealing between the poppet and the seat. As a matter of fact, with pressure in the blocked direction (port out), the fluid entering the back of the mobile element and the spring push towards the inlet port (in) preventing leakages (if the fluid is not contaminated) (Figure 9.3).

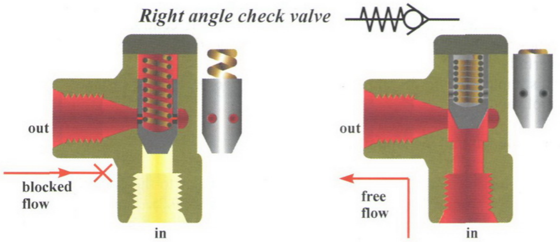



*Figure 9.3*

**Restriction check valve**

In this special type of valve the flow is free in a direction (in to out), while flow is restricted in the other direction (out to in) (Figure 9.4).

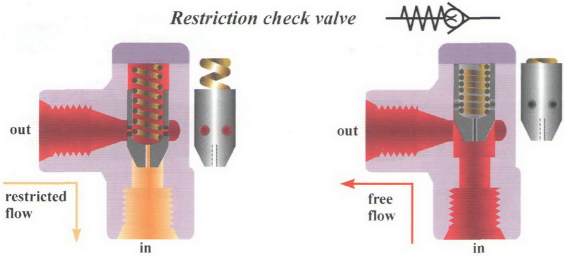



*Figure 9.4*

**Pilot-to-close operated check valve**

These valves have the same design as the previous valves plus a pilot piston. When the hydraulic signal is sent on the piston, the rod connected to it exerts a force on the conical poppet allowing free flow from the previously blocked port to the inlet port (Figure 9.5). No back pressure is allowed on the inlet port in the out-to-in direction.

Many circuits need a check valve whose blocked direction can be unblocked (out to in). For instance, a linear actuator that pushes a load must be blocked in extension (or in any other stroke point).

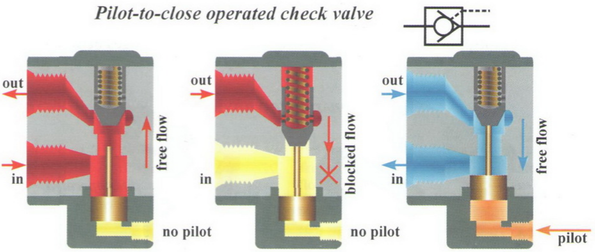



*Figure 9.5*

For the reasons stated in the previous chapter (spool-body leakages), directional valves 4/3, albeit with closed centre, cannot retain a stable position. A simple check valve certainly blocks this position but the return stroke cannot be carried out. Pilot-operated check valves enable the fluid to flow to the outlet port upon the return signal when the hydraulic connection of the pilot is on the opposite hose (Figure 9.6).

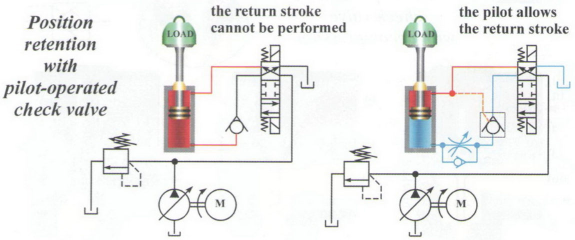



*Figure 9.6*

The surface of the pilot piston depends on the force it is expected to develop on the poppet. As Figure 9.5 (operating principle of pilot-operated valves) shows clearly, the pressurised fluid over only the outlet port (and not over the in port – central drawing of the Figure) flows through the holes of the conical poppet and enters the back of the mobile element, thus pressing towards the inlet port along with the spring (like in right angle check valves).

Pilot and main pressure being equal, the pilot piston surface must be dimensioned so as to exert a higher force than that exerted by the pressurised fluid inside the poppet; in other words, it must be slightly larger than the section of the sharp edge on which the conical poppet lies.

The ratio of the pilot pressure to the working pressure usually ranges between 1:3 and 1:5.

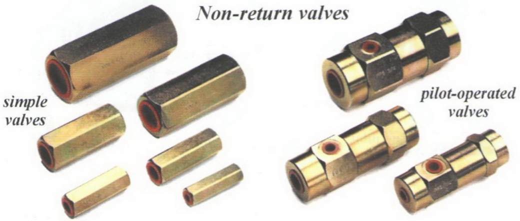



*Figure 9.7*

**Pilot-operated check valve with decompression poppet**

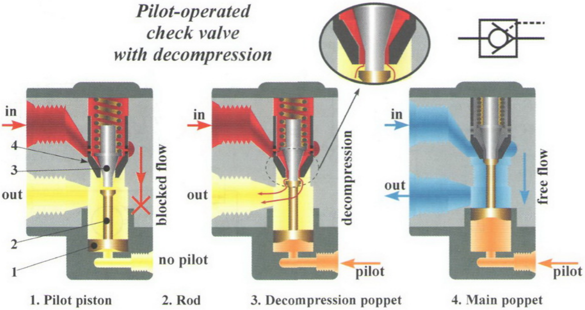



*Figure 9.8*

As far as the previous remark is concerned, systems must often work at low pilot pressures or overpressures and it is thus essential to include a system that releases the poppet. Pilot-operated check valves with decompression poppet are provided with a double conical poppet: the smaller decompression poppet (1) is axially inserted in the main poppet (4). The upper end of the rod (2) of the pilot piston (1) matches the decompression poppet and the whole force delivered (pilot fluid) acts on the tiny section of this poppet (the ratio can be as much as 12:1).

As soon as the pilot pressure is conveyed, the rod first acts on the smaller poppet entailing a decompression in the upper chamber, then it lifts the main poppet thus allowing a quick out-to-in flow (Figure 9.8).

**Pilot-to-open operated check valve**

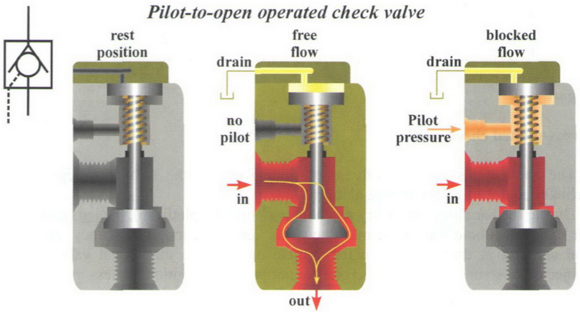



*Figure 9.9*

Pilot-to-open operated check valves are less popular than the previous types of valves. The fluid can flow only in one direction (in to out) in these valves. The opposing spring retains the plate at rest on its seat. When the pilot is off, the pressurised fluid pressing on the plate outweighs the spring force and flows out (out). As soon as the pilot signal is sent, its pressure acts on the upper piston, thus pushing the plate to its seat and blocking the in-to-out flow (Figure 9.9).

The minimum pilot pressure of these valves must be considered carefully: the in-to-out flow can be blocked only if the force exerted by the pilot (pilot pressure on the piston plus spring force) is higher than the force the fluid exerts on the plate. Manufacturers usually specify the minimum pilot pressure, which depends on the surface of the upper piston.

Pilot-to-open operated check valves are widely employed for the unloading of the accumulator that may be found in the circuit. For safety reasons, this must be connected to the tank whenever the pump is switched off.

When the system is on, the pilot directly connected to the pump delivery port keeps the valve closed. As soon as the prime mover stops, the pilot switches off and the pressurised fluid in the accumulator can be unloaded into the tank (Figure 9.10).

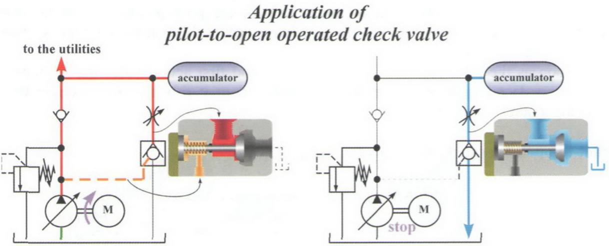



*Figure 9.10*

**Cross-piloted check valve**

The applications of these valves are, among others, the stabilising legs of motor vehicles equipped with a mobile platform.

During the handling (stationary machine with moving platform), the legs stabilised on the earth must hold the up the machine while keeping it motionless; they are retracted and retained in their seat when the vehicle moves. The directional valve, which is subjected to some inevitable leakages, cannot guarantee motionlessness in both phases: the whole flow is conveyed to the cylinders for platform handling when the legs stabilise the vehicle, whereas the pump is usually off when the vehicle is moving. As a result, every leg is equipped with a cross-piloted check valve.

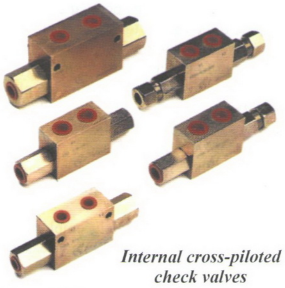



*Figure 9.11*

The drawing of Figure 9.12 shows how simple this system is as it consists of two pilot-to-close operated check valves assembled in a single block. When the directional valve is in the central position γ, the two mobile parts prevent the fluid from flowing and retain the cylinder rod; in position α, the pressurised fluid opens the left part and entails the piloting that opens the right part, thus promoting back pressure unloading. The opposite situation (pressurisation on the right part, piloting on the left part) occurs upon the return stroke (Figure 9.12).

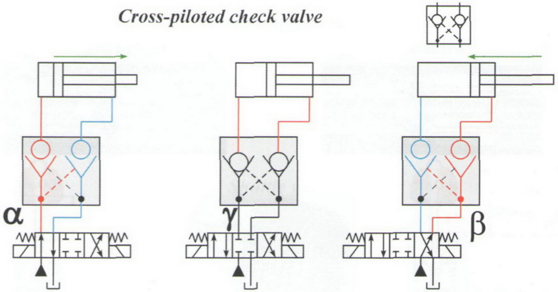



*Figure 9.12*

**Automatic shut-off valve**

Systems whose cylinder holds a suspended load demands special measures so that the suspended object cannot fall due to any malfunction and cause serious damage. Such accidents can be imputed non only to causes that are often not related to the hydraulic system (like faulty ropes or loosened clamps) but also to the burst of the hose connected to the cylinder during load handling or downtime. In this case, the oil leaking quickly makes the load fall abruptly. Non-return valves (also cross-piloted check versions), which are often in series with the directional valve and not directly connected to the cylinder ports, cannot prevent such an accident.

Automatic shut off valves (referred to as 'hose burst valves' by some manufacturers) are mounted directly on the cylinder ports and their operating principle is almost the same as simple check valves.

A block with several side holes is tightened in a sleeve; an opposing spring holds apart the disc from the block (the disc is axial to the block). When the system works properly, the pressurised fluid from the inlet port (in) flows through the side holes, overcomes the disc (the disc has a smaller diameter than the sleeve cavity) and reaches the supply port of the cylinder. In the event that the hose bursts, the fluid abruptly pushed by the falling load in the reverse direction presses against the disc; as pressure drop outweighs the force exerted by the spring, it makes the disc snap shut against the fixed block, thus closing the side ports (Figure 9.13). The spring load must obviously be set so as to sustain the unloading pressure when the system works properly.

If the slow descent of the load is expected instead of its sudden block, the disc must be provided with a tiny hole: the emergency valve does not block the whole fluid subjected to back pressure and the weak flow allowed makes the load descend to the ground gently. However, this option has a drawback as the fluid unloaded into the tank leaks from the broken hose and it is thus released to the environment.

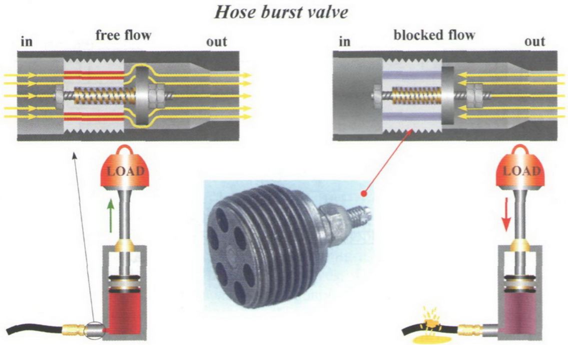



*Figure 9.13*

**Non-return valve with electric control**

The different versions of non-return valves can be controlled by means of a solenoid. Their symbol is that of a non-return valve inside the square of the standard symbol for solenoid valves.

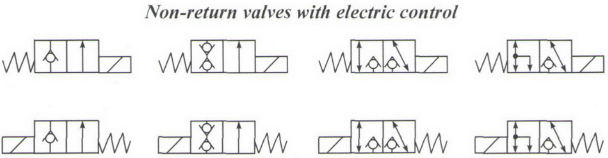



*Figure 9.14*

Manufacturers provide a wide range of combinations and Figure 9.14 shows the most popular ones. The operating principle of these valves is based on the poppet systems mentioned in the previous chapter. They can be employed as both directional valves and non-return valves. They are usually size 03.

## OTHER VALVES

### Shuttle valve

Sometimes the pilot signal of a valve can come from two different circuits. A ball or a conical popper is inside a T-shaped cavity. The fluid flowing to port 'in 1' pushes the ball in the opposite seat ('in 2') and flows through the 'out' port. The same phenomenon occurs if the fluid reaches port 'in 2': the ball moves to the seat 'in 1' allowing the fluid to flow through 'out' (Figure 9.15).

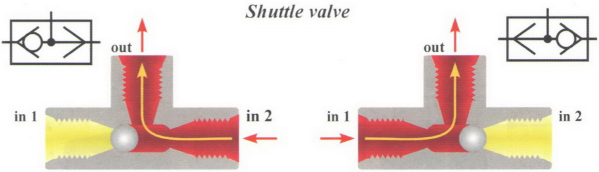



*Figure 9.15*

When the fluid flows in from both 'in 1' and 'in 2' simultaneously, 'out' experiences the higher pressure: for instance, if 'in 1' = 50 bar and 'in 2' = 80 bar, out = 80 bar.

Shuttle valves (or selector valves) are widely employed in Load Sensing pilot systems.

### Deceleration valve

Cylinder cushions reduce the final speed of the rod; however, the forward or return stroke must often be decelerated over a length that the cushion cannot fully cover, which means ensuring whether a long end-stroke deceleration or rod deceleration and stop in any point of the stroke. In these cases it is essential to equip the rod with a cam and to provide the cylinder with a deceleration valve.

This is essentially made up of a conical spool valve operated by an external mechanical device (which is similar to a limit switch) and a non-return valve for the return of the rod under ordinary flow conditions (Figure 9.16).

The valve mechanical pilot, operated by the cone-shaped cam on the rod, moves the spool that closes the holes of the valve body gradually and it reduces flow thus decelerating and stopping the actuator.

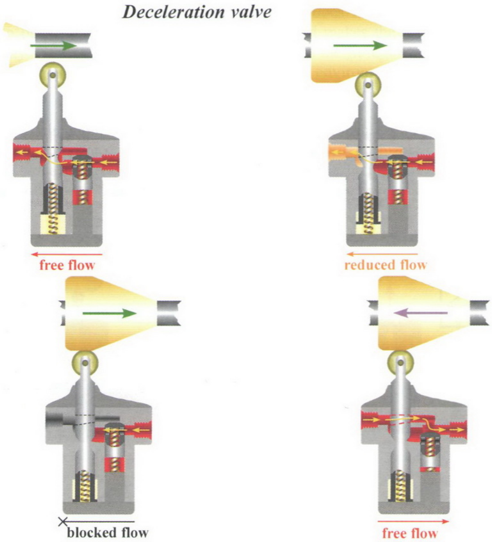



*Figure 9.16*

When the fluid flows in the opposite direction, the non-return valve opens allowing full flow.

**Actuator deceleration**

It is important to remember that the reduced flow of deceleration valves can be controlled only by means of the cam positioned on the rod of the actuating cylinder. Since the actuator deceleration depends on the needs of the process, it is often essential to add more control components along with the deceleration valve.

Common applications are actuators controlling drill chucks or the handling of the slide of some machine tools, which need the speed to be decreased automatically and to be kept constant over a remarkable length (the restriction must be performed in the unloading way like in most cylinders).

An effective and rational solution is electromechanical operation (Figure 9.17): upon the slow position, a switch operates the flow control valve via a NO solenoid valve 2/2; the parallel check valve is useless in the reverse stroke because the quick return is guaranteed by the 2/2 moving back to the rest position. Note that there is no need to provide the rod with as long a cam as the decelerated stroke.

***Actuator deceleration - Electromechanical system***

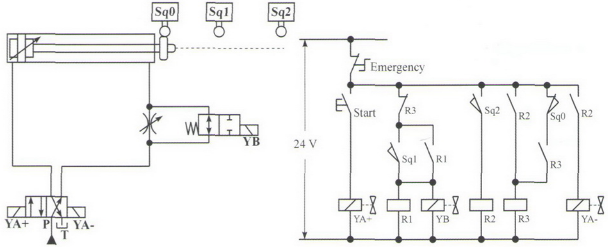



*Figure 9.17*

A single proportional valve interfaced with the closed loop electronic control unit delivers even better results: as a matter of fact, it controls direction, reduces speed and controls the force variation of the process.

If electromechanical or electronic circuits cannot be used, it is possible to resort to a hydraulic system controlled by a deceleration valve, clearly during speed reduction. This valve, operated by the rod cam, can deliver only a roughly reduced flow that cannot be adjusted; the inclusion of a flow control valve in parallel with the deceleration valve enables the operator to adjust the decelerated stroke. The cushion inside the cylinder ensures a soft stop at end of stroke (Figure 9.18).

- ✓ *quick forward stroke:* the fluid unloaded into the tank flows freely through the deceleration valve while the flow control valve is off;
- ✓ *slow forward stroke:* the hole of the deceleration valve closes gradually, which makes the fluid flow through the restrictor; the rode slides slowly, depending on the setting of the turning knob of the flow control valve;
- ✓ *quick return stroke:* after reversing the direction by acting on the directional valve, the pressurised fluid flows through the totally open hole of the deceleration valve (opening of the internal non-return valve); the fluid is unloaded freely through the dedicated hose.

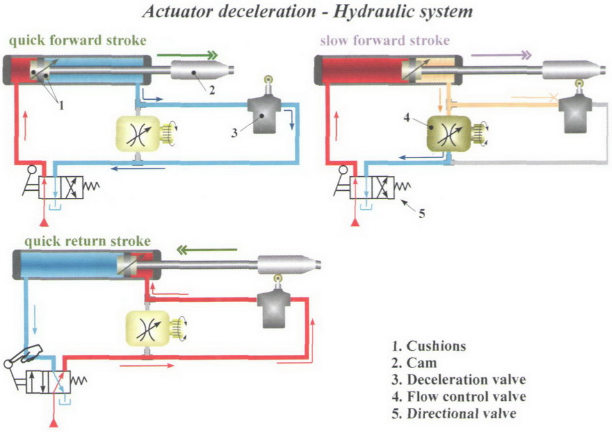



*Figure 9.18*

Figure 9.18 shows clearly that the inclusion of the cam (especially if it has long stroke) entails a remarkable long rod extension. In order to solve this problem, the cam is usually replaced by an additional rod solidly connected to the piston rod and the inclusion of one or more bushes prevents its rotation (Figure 9.19).

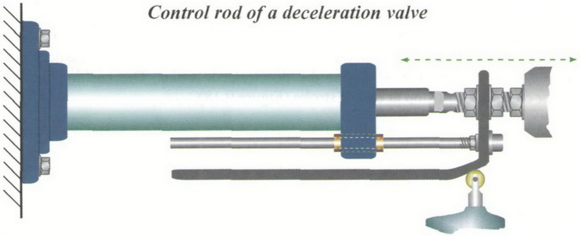



*Figure 9.19*

### Prefill valve

ISO 5598 standard defines a prefill valve as a *'valve which permits full flow from a reservoir to a working cylinder during the advance phase of a cycle, permits the operating pressure to be applied during the working phase and permits free flow from the cylinder to the reservoir during the return phase'*.

Vertical cylinders working downward, having large capacities and demanding a quick advance phase, then a slow high-pressure stroke and finally a quick return stroke need high flow rates available in a short time in the first phase and, to some extent, in the final phase. Prefill valves, designed specifically for flow rates equalling as much as thousands of litres per minute with little pressure drops, enable the fluid to flow from the tank to the cylinder quickly. It is important to highlight that the distance between the tank and the prefill valve must be as short as possible (to be precise, the best situation consist in the valve directly connected to the tank) because any connection hose, albeit with a large diameter, always entails some pressure drops.

A simple prefill valve can basically be a large pilot check valve: the descent of the piston due to gravity triggers the suction of the fluid from the tank through the valve; when it moves upward, the pilot allows the return to the tank by opening the check valve (Figure 9.20).

***Filling with a pilot check valve***

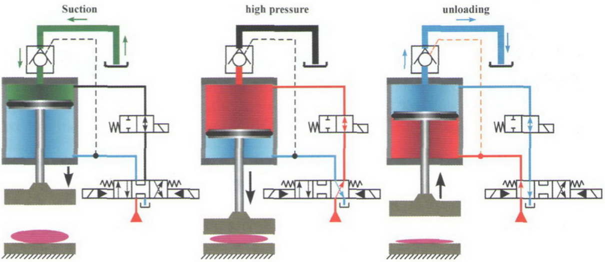



*Figure 9.20*

As the Figure shows, the pilot check valve and the high-pressure hose must be connected to the cylinder cap end separately. In most cases the solution adopted consists in connecting a valve to the cap end of the actuator so that it manages the whole prefilling, high pressure and unloading phases.

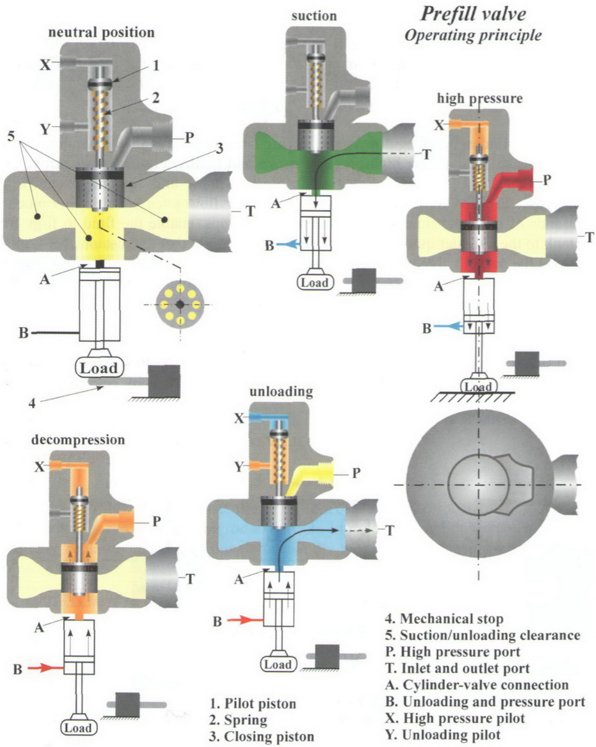



*Figure 9.21*

Prefill valves perform five tasks (see Figure 9.21):

1. *Neutral position:* the springs keeps the closing piston (3) at rest, the oil is in the clearance (5) connected to the tank via the T port; an external device retains the

piston of the actuating cylinder in the initial position. There is no fluid or mechanical movement.

2. *Suction:* as the mechanical stop (4) is removed, the load drags the piston downward, the vacuum developed in the upper chamber of the body sucks oil from the tank through the large valve clearance (5).
3. *High pressure:* the fluid conveyed to port X pushes the piston (3) downward closing the suction clearance (5); the fluid that reaches port P from the pump flows through the holes inside the piston (3) and performs the pressing by entering the cylinder.
4. *Decompression:* a brief decompression is essential because the sudden outflow of high pressure fluid could cause fluid hammers. The fluid flows in the reverse direction and it is unloaded via port P that is now connected to the tank by means of a directional valve (not shown in the Figure).
5. *Unloading:* once decompression is accomplished, Y and X are respectively switched on and off when pressure equals about 15 bar; the closing piston (3) moving back to the rest position enables the fluid to be unloaded via the clearance (5) of the prefill valve directly connected to the tank.

Since prefill valve promote the high flow rates in a short time, they are ideal for the use in one of the most important hydraulic stationary applications, that is to say **presses**.

Presses have been described in general terms so far as they were considered as a large single cylinder from a hydraulic point of view. This is true just for small presses (actually small tonnage presses have a spring-loaded single-acting cylinder) whereas large deep-drawing presses are more complex from a mechanical and hydraulic point of view. The large pressing cylinder is fundamentally a ram cylinder that 'works' hydraulically only at a high pressure during the phase of deformation of the material. In the advance phases of the downward stroke and throughout upward stroke, the rod/piston is dragged by two or four smaller vertical cylinders next to the columns; their cap-ends are solidly connected to the upper structure of the press and the rod tips are fixed to the die-hold slide. The rod of the larger cylinder too is obviously fixed to this slide.

Upon starting, the fluid is conveyed from the pump to the upper end of the side cylinders: their rods move downward dragging the larger cylinder, which is not supplied by the pump for the time being as it sucks the fluid directly from the tank. During pressing, the fluid from the pump is diverted to the cylinder that changes the shape of the material. After pressing, the lateral cylinders (now on the opposite side) are supplied so that they bring the cylinder back to the rest position. As we already stressed, the advance phase needs a high flow rate to reach the upper chamber of the main cylinder at an adequate speed; the unloading speed depends not only on the size of the clearances between the main cylinder and the tank but also on the pressure and the diameter of the smaller cylinders.

This operation needs the presence of some components (Figure 9.22): directional and control valves for the side cylinders, a sequence valve (it allows the inlet of fluid from the pump into the main cylinder as soon as a specific pressure is reached; see chapter

10) and a prefill valve (its design must be slightly different from the prefill valves described above). We just mention the fact that these presses require also electrohydraulic systems (safety interlocks) that respect the standards set by the Machinery Directive, the Pressure Equipment Directive, etcetera; in addition, presses are often equipped with electrohydraulic proportional control systems along with a PLC or a PC.

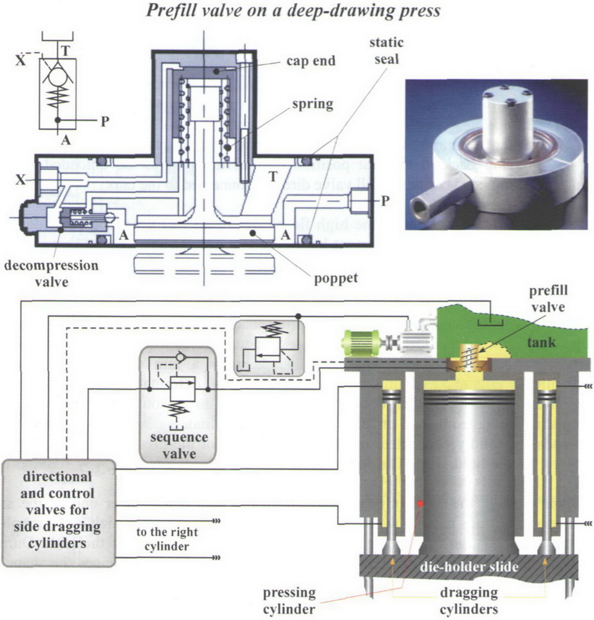



*Figure 9.22*

Since the tank is placed on the pressing cylinder, that is to say on the upper part of the press, the prefill valve is flanged directly between the bottom of the tank and the cap end of the main cylinder (sandwich flange).

At rest, the spring keeps the poppet of the prefill valve (Figure 9.22) closed. When it is started, the solenoid valve makes the side cylinders move downward; the ensuing large cylinder movement entails a vacuum in its chamber that releases the spring in the prefill valve: the poppet slides downward allowing the fluid to flow from the tank to the pressing cylinder. As soon as the slide touches the metal to deep-draw, the rods stop and the poppet closes because there is no more vacuum; as pressure increases in the side cylinders, the sequence valve opens enabling the fluid from the pump to reach the cylinder that deforms the material.

In order to trigger the upward stroke, the direction is reversed by acting on the directional valve connected to the dragging cylinders and meanwhile the pilot X (inactive so far) is pressurised (see the tables provided by the manufacturer of the prefill valve to set pilot pressure properly).

The pilot fluid acting on the plate of the prefill valve pushes the poppet downward: the liquid held in the larger cylinder chamber is thus unloaded directly to the tank. This type of valve too can have an optional decompression device for the beginning of the upward stroke.

## REMOTE CONTROL

Remote control is made possible by a mechanical device, some electric contacts and an auxiliary valve with pneumatic or hydraulic control; this operation can be performed via a lever, a button or a pedal (Figure 9.23).

***Button, pedals and levers for distributor remote control***

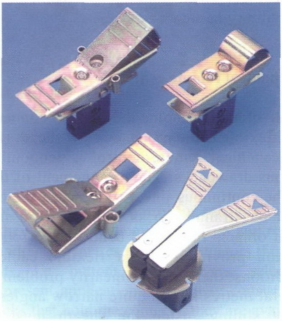


*Figure 9.23*

The distance from the directional valve can range from few centimetres to several metres for mechanical, pneumatic and hydraulic remote control and there are almost no limits for electric remote control. What follows is a list of its advantages (only the first point applies to mechanical remote control).

1) The control can be performed far away from the distributors: all the controls are in the cabin of mobile machines or on the control panel of stationary machines while directional valves are close to the actuators.

2) Large valves can be controlled easily: handy joysticks replace the long levers needed to counterbalance high pressure/flow on spools.

3) The cost for the use of large valves is reduced: if distributors are positioned in the cabin or in the control panel (hence away from the actuators), they need very long hoses with a large diameter, which causes substantial pressure drops. In remote control, directional valves are not close to the actuators and control hydraulic pipes have a much smaller section. As a result, costs are cut and less space is required.

4) Several different control components can be included in a single unit: standard joysticks can be equipped with several hydraulic valves and electric buttons.

5) The whole operation can be performed by means of a single control unit made of different types of controls, even from a remarkable distance from the machine: the operator can manage the systems from a distance avoiding contact with falling or contaminating material, overturning vehicles, vibration, etc.

**Mechanical remote control**

The block that holds the lever is connected to the valve body via a rigid tie rod or a flexible cable protected by a sheath (Figure 9.24).

Mechanical remote control is economical and easy to install but it has rather serious drawbacks. The arrangement of the versions with rigid tie rods is limited by the continuous linearity between the operating lever and the valve; the device must be arranged so as to prevent possible shocks (also minor shocks) against blunt objects that could bend the tie rod thus undermining its functionality. The version provided with a flexible cable covered with a sheath offers better conditions: there are no linearity problems and the cable can slide over as much as several metres and in quite narrow angles along the actuating arms. However, the cable sliding inside the sheath must be cleaned and greased periodically; unfortunately, the lubricating grease makes some corrosive substance enter the sheath, like dust, water, or other liquids, thus causing a very poor sliding, oxidation, corrosion and frays.

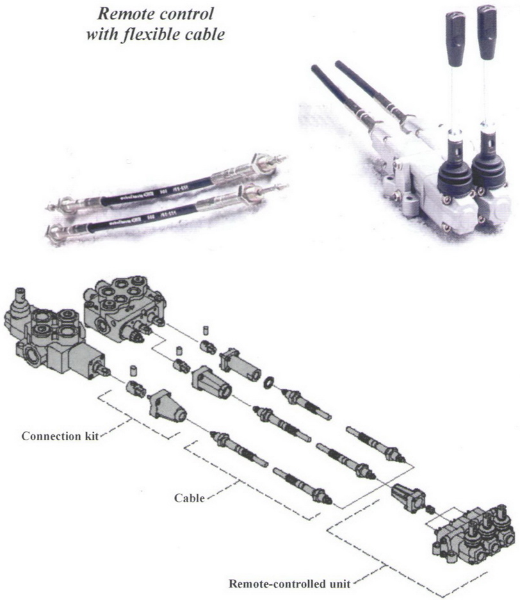



*Figure 9.24*

### Electrical remote control

This is clearly the most versatile and economical type of control for solenoid valves. There are no limits to angles even though they are repeated countless times and remote control can work at a distance of several hundreds of metres from the actuators if cables are dimensioned properly. Control units, i.e. buttons, small levers, selectors, potentiometers, etc., are modular in most present applications since they consist of a mobile unit along with additional small blocks that hold the contacts; they are available in every electric material wholesale business.

Start controls, adjustments, functional tests and manual reset operations have the same configuration also in proportional applications with electronic control unit.

***Remote control joysticks***

*Figure 9.25*

The controls in stationary applications are usually gathered in one or more fixed control panels; self-propelled machines have fixed positions on the command console in front of or next to the driver (Figure 9.26) or mobile remote control units, depending on the applications.

***Electric remote control portable unit***

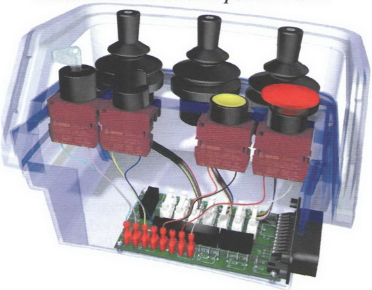


*Figure 9.26*

This notion can sound obvious and trivial but the control arrangement must be studied carefully by taking both the safety standards in force and the system functionality into account. The control panel of broaching machines (spline cutters) or sheet cutting machines must be not only arranged at a safety distance in order to avoid the contact between the operator and the machine, but also fastened to a fixed surface to prevent it from shifting accidentally next to moving parts. Since crane handling on self-propelled machines often requires the driver to keep a distance that is much higher than the jib radius, it is essential the driver can work by means of a cable remote control or a radio control.

During a building demolition with a hydraulic hammer mounted on a crawler vehicle, the worker has to stay in the driver's seat firmly while handling the systems via the console and under no circumstances can the driver leave its position, except for emergency situations in which however the driver must first make sure the vehicle cannot cause damage to third parties. Nonetheless, such an operation may be very dangerous in some situations because the material could collapse or the vehicle runs the risk to overturn if it is on a steep slope; consequently, the vehicle needs an additional remote control for safety reasons. Farm tractors pose different challenges. Pulled machinery needs different controls: a single unit cannot control a weed-killing machine, a mower, a laser land leveller, a four-furrow ploughs, a sprayer and so on altogether. Each system has its own mobile console that has to be applied on the tractor according to the task the machine is expected to perform.

### Pneumatic remote control

The advantages of pneumatic controls have already been dealt with in Chapter 8, § 'Valve Control'. In brief, pneumatic valves send a compressed air control signal directly on the spool of the hydraulic valve. Like in hydraulic systems, the movement of the lever (hence the control signal) is often proportional to flow: as the rod moves on, the spool of the main valve slides gradually (Figure 9.27).

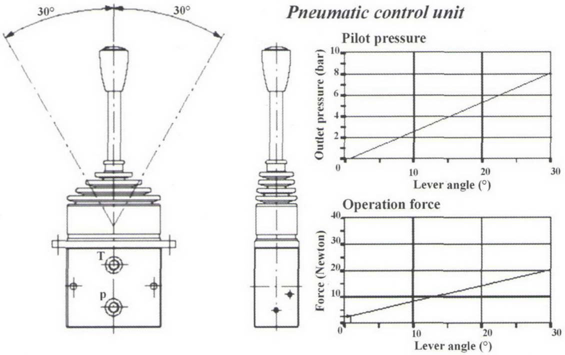



*Figure 9.27*

Since working pressures are quite low in pneumatics, these systems use only small-medium valves: the control on valves subjected to remarkable forces would need a pilot piston with a large diameter.

These valves are very useful in the standard fixed applications described in the chapter and the paragraph mentioned above; they are often unsuitable for mobile applications because machines are often equipped with a compressed air generator.

Pneumatic controls are widely employed in industrial vehicles (most of them have a pneumatic braking systems) for dump bodies, auxiliary cranes, etc; they are also used for medium and powerful farm tractors that need a compressor for trailer braking.

**Hydraulic remote control**

***Hydraulic control units***

*Figure 9.28*

Hydraulic remote control units consist of a part handled by the operator and a valve mechanically connected to it that transmits the signal to the main distributor.

The handling part can be a lever, a pedal, a button or an adjusting handle (Figure 9.28). The setting of handles or the shift angle of the lever, pedal or button determined by the operator makes the spool in the main distributor slide accordingly, thus affecting the actuator speed. Buttons and handles need only one valve sending the signal to a single monostable distributor; levers and pedals can have either two (single valve) or four positions (double valve) for each lever and two positions (one valve) for each pedal. In addition, a wide range of positional and ergonomic solutions are available for lever and pedal controls, like the automatic return to the neutral position, attachment to all the positions and manual return, attachment to a single position and so on.

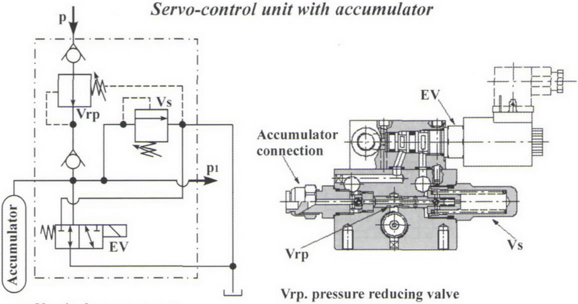
Pilot pressure can result from an auxiliary pump or a servo-control provided with an accumulator. This (Figure 9.29) has some security systems designed to avoid overpressures and to connect the accumulator to the tank as soon as a danger looms.

***Servo-controls with accumulator***

*Figure 9.29*

Servo-controls with accumulator (Figure 9.30) are usually made up of a pressure reducing valve Vrp, a relief valve Vs, an (optional) unloading solenoid valve EV and two check valves.

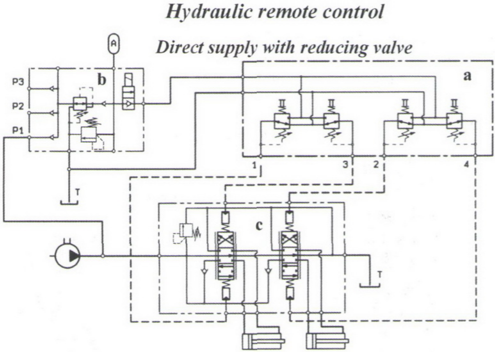
***Servo-control unit with accumulator***




**p. Nominal pressure port**
**p1. Pilot pressure**

**Vrp. pressure reducing valve**
**Vs. Relief valve**
**EV. Unloading solenoid valve**

*Figure 9.30*




**a. Control unit** **b. Pressure reducing valve with accumulator and unloading**
**c. Main distributors** **d. Auxiliary pump**

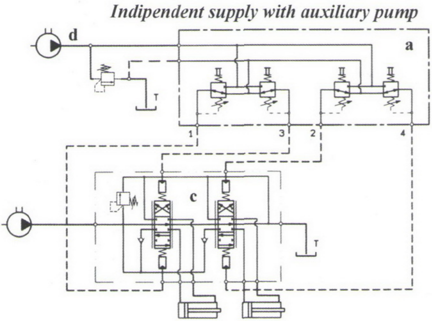



*Figure 9.31*

Vrp reduces the system working pressure to the level needed for piloting; the relief valve Vs protects the circuit from possible overpressures if the pressure reducing valve does not work properly. Manufacturers usually provide these devices with or without the remote-controlled solenoid valve EV that connects only the servo-circuit and the accumulator to the tank. A check valve connected in series between EV and Vrp prevents the fluid (pilot pressure) stored in the accumulator to flow to the control valves. The task of the accumulator is to ensure the pressurisation of the supply circuit also when the system is off; emergency operations can be carried out in such a situation.

***Internal view of a hydraulic control***

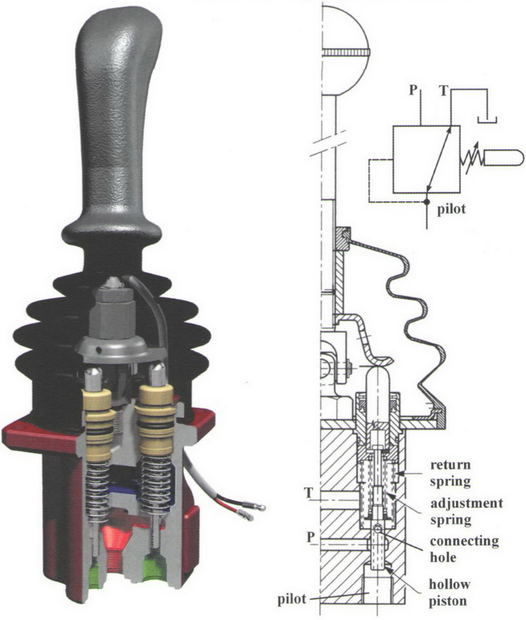



*Figure 9.32*

The hydraulic diagrams in Figure 9.31 show the servo-control systems with independent supply via an auxiliary pump or the servo-control described so far; an electric control on the handle starts the unloading of the solenoid valve. Both diagrams refer to controls with four positions for two distributors. All the control valves are generically drawn as directional valves 3/1 (one position).

Control hydraulic valves are based on the same operating principle as direct-acting relief valves. By acting on the lever (Figure 9.32), the fluid from port P flows through the connecting hole to the pilot port, thus operating the command of the main distributor connected to a dedicated hose. The increase in pilot pressure develops an opposite force inside the hollow piston that outweighs the adjustment spring load thus making the piston move upward. The fluid is then unloaded via the connecting hole to the chamber of the springs and it reaches the tank through port T. When it is released, the return spring moves the lever back to the neutral position.

Manufacturers apply lever, handle, button, pedal or joystick controls to remote control valves with proper mechanical methods.

**Radio control**

Many applications have been equipped with a radio control system for years. This remote control device sends signals (electromagnetic waves or laser signals) to the receiver mounted on the machine (Figure 9.33). Receivers process and amplify inputs transforming them into commands for the valves.

These signals must use special frequencies so as not to interfere with other devices or to be disturbed by them. In order to avoid this problem, digital signals have recently been used instead of modulated waves.

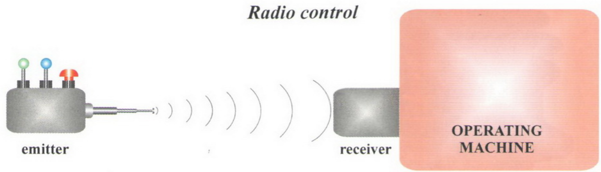



*Figure 9.33*

Chapter **10**

# *PRESSURE CONTROL AND ADJUSTMENT VALVES*

The perfect functionality of oil hydraulic systems is the result of constant controls and adjustments of the speed and the force of the actuator, the final part of the system. The fundamental principles of force and speed (see the chapter devoted to hydrostatics and hydrodynamics) are respectively pressure and flow in fluid power.

Control or adjustment valves rest on two basic prerequisites: the force exerted by the pressurised fluid versus an external reaction and the reduction of the flow cross-section.

The accurate and constant control of parameters is related to many physical and mechanical factors that are specific to the phenomena developed inside the valve and inherent to that very type of system, like temperature changes, viscosity, external mechanical conditions and so on. As a result, more sophisticated methods are needed: they must always guarantee precise processes, easy handling, a good quality of the products obtained with stationary or mobile operating machines; furthermore, they must satisfy both the needs of users and safety standards.

The explanations and descriptions of this chapter refer to valves installed in a traditional manner, i.e. contained in a parallelepiped or another solid provided with connecting ports to rigid pipes or hoses. The operating principles covered apply also to *cartridge valves*, which are dealt with in Chapter 12.

## RELIEF VALVES

**Relief** valves (also known as **pressure relief** valves under ISO 5598 or simply as **safety** valves) are mainly employed to limit the maximum pressure of the system and protect the pump from higher pressures that could damage it seriously.

Relief valves are also widely employed in many devices in which it is crucial to avoid pressure rises; consequently, these devices are equipped with a main relief valve and other valves set to lower pressures and arranged in special points of the system. Pressure relief valves serve as, among others, anti-shock valves for hydraulic motors or hydrostatic power steering, safety valves for accumulators, oil change valves and anti-skid cross valves for hydrostatic drive.

In brief, as soon as pressure exceeds the maximum limit set, the internal parts of relief valves unload some fluid into the tank by opening a clearance. This relieves overpressure at the outset preventing the pressure within the system from soaring excessively.

Its symbol under ISO 1219/1 reflects this notion perfectly (the arrow can be whether with or without the trait next to its end). The clearance (represented as an arrow) inside a spring-loaded mobile part is blocked when the valve is off (nominal pressure circuit); as pressure rises unduly, the hydraulic pilot (broken lines) acts on the flat face of the mobile part and overcomes the force of the spring making it move. This sliding connects port P to port T and the fluid is thus unloaded into the tank. Figure 10.1 (valve symbol and illustrative valve design) reflects this description.

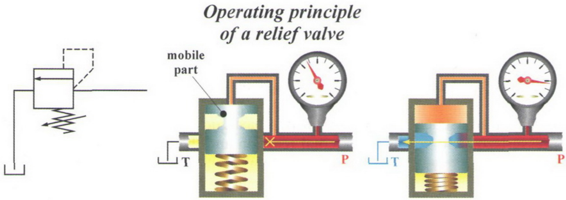



*Figure 10.1*

Hydraulic systems are more effective than electromechanical ones: a fuse or a magnetothermic switch protects electric motors from excessive currents, although overloads break the circuit; relief valves in oil hydraulics simply release the pressure excess without stopping the system.

However, there are some drawbacks too because the *energy resulting from* the pressurised fluid versus the poppet subjected to a spring *is transformed into heat* with ensuing overheating of the valve and the hydraulic liquid.

**Direct-acting relief valve**

Direct-acting relief valves are the simplest and cheapest method to limit the pressure within a system that is not subjected to high flow rates.

Under normal working conditions, that is to say during the movement of the rod subjected to a load (the same conditions apply to hydraulic motors), the pressure within the circuit is less than the relief valve pressure setting. At the end of stroke of the actuator and in any situation in which pressure exceeds the maximum level set (*override pressure*), relief valves unload some flow as soon as pressure soars.

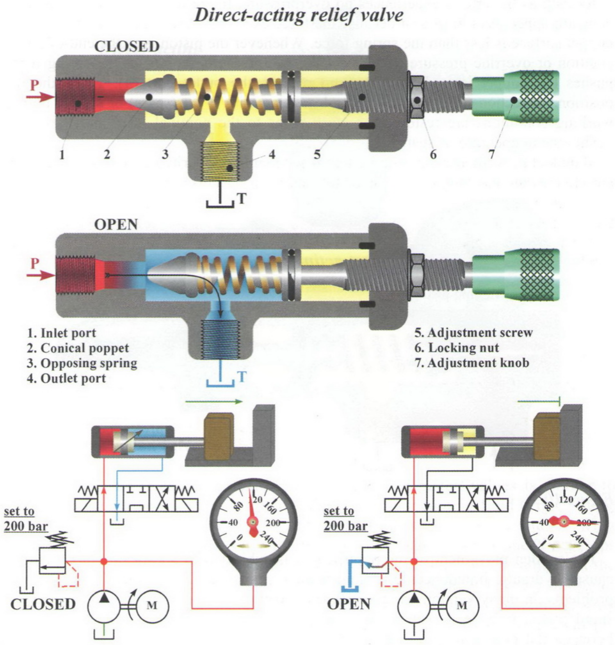



*Figure 10.2*

The pressurised fluid is still found in the inlet port (1) (Figure 10.2) and it presses against the conical poppet (2) (some versions have a ball instead), retained in its position by a spring (3), set to a pressure that is slightly higher than the operating pressure. Pressure is set manually by acting on a turning knob (7), solidly connected to the screw (5) that increases or decreases the loading of the spring. Clockwise revolution makes pressure setting increase.

As long as the circuit experiences no overpressure, the relief valve is closed and the conical poppet stays in its seat because the force exerted by the pressurised fluid on the poppet surface is less than the spring force. Whenever the piston is in the end-of-stroke position or override pressure develops, fluid overpressure overcomes the spring force, pushes the poppet inward and the fluid reaches the outlet port (4). When the rest position ( directional valve in the central position with P-T by-pass) or simply normal working conditions are restored, pressure drops to the nominal level and the spring pushes the poppet into its seat.

Manufacturers often equip turning knobs with simple security devices (cap, key, nut, etc.) to prevent unauthorised personnel to handle the system (Figure 10.3).

***Cushioning piston of direct-acting relief valve***

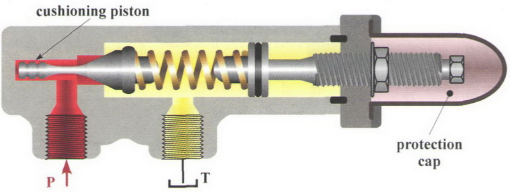



*Figure 10.3*

The sudden movement of the poppet, if it is subjected to abrupt pressure changes, causes hydraulic imbalances like fluid hammer, noise and vibrations. To solve these problems, in many versions (Figure 10.3) the conical poppet is solidly connected to a small piston in a dead seat whose task is to cushion its sliding. The fluid leaking between the piston and its seat entails a weak pressure drop that slows the poppet down.

Nonetheless, these problems occur also under normal working conditions: as a matter of fact, springs can resonate with mechanical vibrations (see the final part of Chapter 3 and the last paragraph of Chapter 16) and the inevitable pulsating flow due to the pump (Chapter 4), thus causing disturbing vibration and often poor results in terms of workpieces machined by precision machine tools.

Besides usual expedients like the use of odd-numbered piston pumps, the inclusion of an accumulator, thru-bulkheads and flanges with elastomer, these simple valves must be replaced by more efficient types, i.e. hydraulically self-balancing valves (see later on).

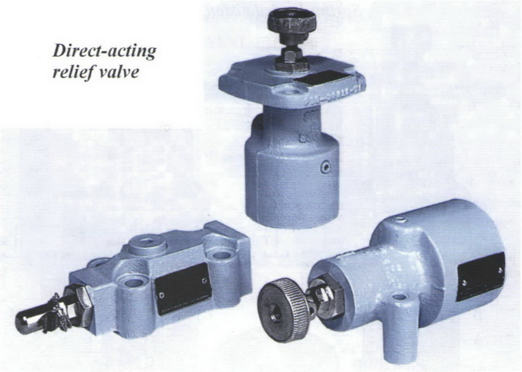



*Figure 10.4*

**Spring-loaded piston valve**

The design and operating principle of spring-loaded piston valves is similar to those of most compensators covered in Chapter 5. The incoming fluid (Figure 10.5) exerts an equal and opposite force on the internal faces of pistons $\alpha$ and $\beta$ having the same bore. As pressure rises unduly, the liquid through the pilot line X develops a higher force than the opposing spring $\gamma$ on the pilot piston $\omega$; the movement of the spool $\alpha/\beta$ connects P to port T. The fluid that fills the chamber between the pilot piston and the lower face of piston $\beta$ is unloaded through the hole $\Phi$ communicating with the tiny channel $\delta$ inside the spool $\alpha/\beta$ (failing it, the fluid leaked into the chamber would block the spool in the return phase) and it is unloaded into the tank via line Y; these lines also guarantee optimum lubrication of sliding mechanical parts.

Spring-loaded piston valves are designed so as to allow their conversion into sequence valves, unloading valves, etc. by making some modifications. The reverse direction of flow results from turning or replacing caps and opening/closing ports.

Like the others valves with the same design described later on, spring-loaded piston valves have hydraulically self-balancing spools (the fluid exerts an equal and opposite force on the internal faces of pistons $\alpha$ and $\beta$). This enables the use of a small piston and consequently opposing springs that exert little force. However, the dynamic forces due to high flow rates could make the spool close again.

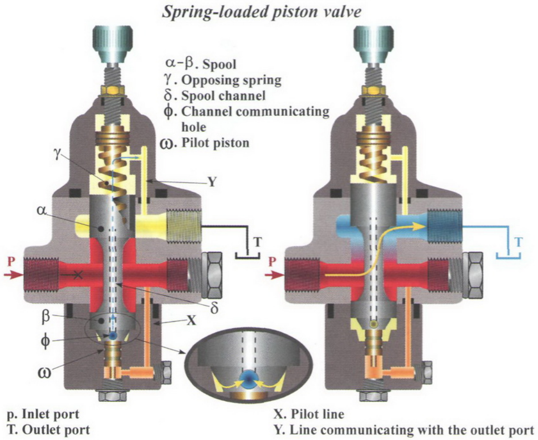



*Figure 10.5*

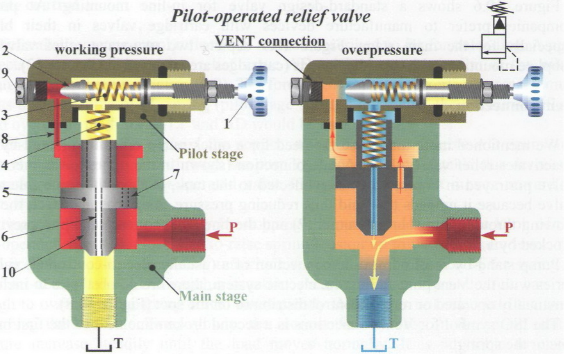
**Pilot-operated relief valve**

Pilot-operated relief valves (also known as **two-stage** valves) are essential in the event of high pressures and flow rates because direct-acting relief valves can hardly cope with them. They consist of the same direct-acting pilot valve as those described previously and a main valve that holds a special piston ending with a poppet that blocks or connects ports P and T.

Under normal working conditions at nominal pressure (Figure 10.6), the incoming fluid flows through the balance fluid channel (7) and retains the piston (5) while the poppet is in its seat; as a matter of fact, the pressure on both faces of the piston is equivalent. Also the conical poppet (2) is motionless because the force exerted by the conical poppet spring (3), manually set by means of a knob (1), is higher than the force of the pressurised fluid from the communicating channel (8).

Whenever pressure rises unduly, the conical poppet (2) opens and unloads the fluid via the tank channel (6). This undermines the force balance on the piston (5): pressure on the upper part drops dramatically and the pressurised fluid from port P pushes the lower face upward. The poppet (10) opens and the fluid reaches port T.

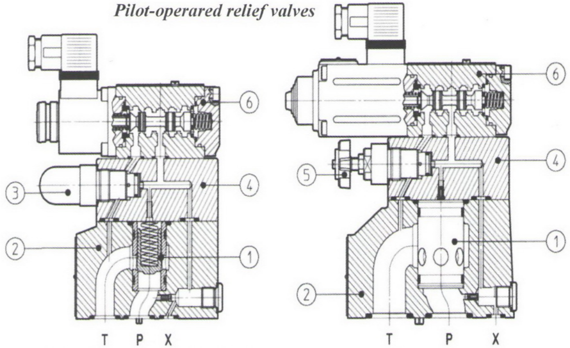
The pressure drop in the system and within the valve restores normal working conditions; the spring (4) pushes the piston (5) downward and the poppet (10) solidly connected to the piston blocks port T preventing the fluid to reach the tank.




1. Adjustment knob 2. Conical poppet 3. Conical poppet spring 4. Main stage spring
5. Piston 6. Tank fluid channel 7. Balanced fluid channel
8. Line connecting the two stages 9. Cap 10. Piston poppet
p. Inlet port T. Outlet port

*Figure 10.6*

*Pilot-operared relief valves*




1. Poppet 2. Main stage 3. Protection cap
4. Pilot stage 5. Turning knob 6. Venting solenoid valve

*Figure 10.7*

Figure 10.6 shows a standard-design valve for in-line mounting. At present, companies prefer to manufacture devices with cartridge valves in their blocks, especially for the main stage. Figure 10.7 depicts two two-stage relief valves for subplate mounting, with cartridge inserts (cartridges are analysed in Chapter 12).

### Vent connection

We mentioned more than once the need for a quick pump unloading (stand-by) that deactivates relief valves. If the Vent connection (shown in the upper left corner of the valve portrayed in Figure 10.6) is connected to the tank, it plays the same role as the valve because it unloads the fluid thus reducing pressure. As a matter of fact, the fluid flowing through the balance channel (7) and the line (8), can reach the port previously blocked by a cap (9).

Pump stand-by results from the connection of a (usually electro-controlled) valve in series with the Vent port; failing the electric system, there are no obstacles to including a manually operated or remote control distributor on the spot (Figure 10.8).

The ISO symbol for Vent connections is a second broken line next to the first broken line or the spring.

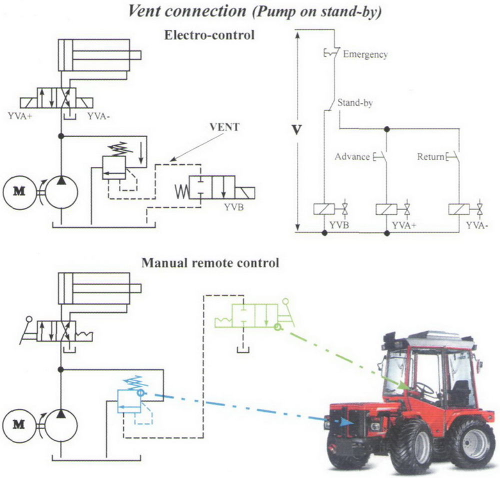



*Figure 10.8*

**Remote control**

Relief valve remote control reflects the method described so far for unloading. A pipe connects a second direct-acting relief valve AD to the vent connection of the pilot-operated relief valve PL (Figure 10.9). The force the AD springs exerts necessarily must be less than the force of PL spring (pilot stage spring); otherwise, the pressurised fluid would overcome the force of LP and AD would be useless.

When pressure mounts unduly, the poppet of the direct-acting valve AD opens and allows the pilot fluid to flow through the port $T_{AD}$ connected to the tank; this decreases the pressure within the pilot-operated valve PL: when the poppet moves upward, it diverts the flow into the tank.

Setting can be performed simply by stopping the pump and turning the knob of the pilot-operated valve PL clockwise to raise spring tension up to the maximum level; the pump is started up after decreasing (anticlockwise rotation) the tension of the spring of the direct-acting valve AD. In such a situation, pressure is quite low and a few bars are enough to overcome the spring force, consequently the fluid is unloaded into the tank; by turning the knob clockwise slowly, the increase in the force on the spring makes pressure increase steadily until the load moves normally. It is advisable to check settings after some operating time because parameters are like to be altered due to fluid overheating. Under constant working conditions, the tension of the PL spring PL has to be reduced (but it must obviously be higher than the other spring) so that it can protect the pump and the actuating system from excessive pressures in the event of a breakdown of the adjustment valve AD.

It is worth fixing the axial screw to the turning knob of the main valve PL in order to prevent tampering.

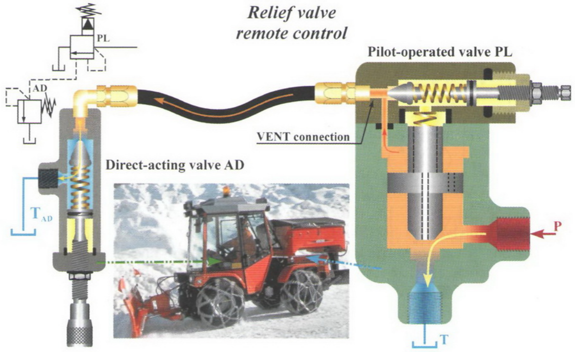



*Figure 10.9*

Remote control is more reliable in electronic **proportional control** relief valves. A solenoid valve with a direct current solenoid that replaces the opposing spring is positioned on two-stage valves; the electronic control system receives adjustment inputs and then determines the tension on the coil by increasing or decreasing the force that counters the pressure on the conical poppet.

### Pressure selection

The last paragraph of Chapter 5 explored the option of multiple maximum pressures in a single circuit, but that situation demanded three different safety valves; on the contrary, most manufacturers enable operators to choose two or three working pressures with a single component.

Relief valves with three pressures (Figure 10.10) are made up of a main valve and three pilot-operated valves controlled by a solenoid valve 4/3. A second solenoid valve 2/2 places the pump on stand-by. The operating principle rests on the fact that in all positions two pilot-operated valves do not work because they are whether connected to the closed port or mutually connected.

- ✓ Central position of the solenoid valve 4/3: p1 and p3 are inactive (their unloading clearances are closed); the pilot fluid reaches the tank via p2.
- ✓ Position with excited YVA: p2 and p3 are inactive (the valve connects them mutually); the pilot fluid reaches the tank via p1.
- ✓ Position with excited YVB: p1 and p2 are inactive (the valve connects them mutually); the pilot fluid reaches the tank via p3.

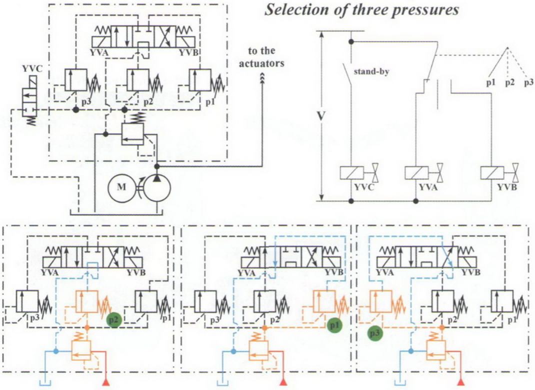



*Figure 10.10*

### Unloading relief valve

Circuits provided with an accumulator need a special relief valve that both connects the pump to the tank whenever the pressure in the accumulator exceeds the maximum level set and reduces pressure to the nominal level by unloading the fluid stored in the accumulator.

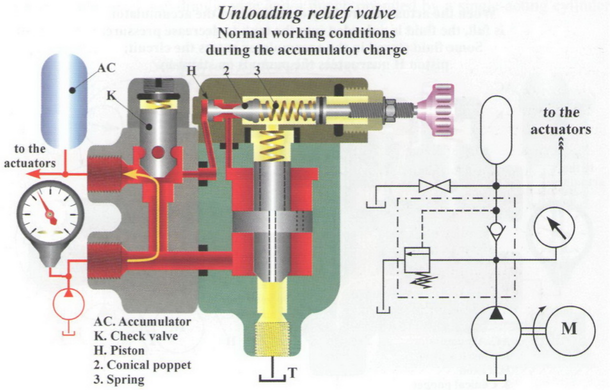



*Figure 10.11*

Unloading relief valves are essentially made up of an ordinary pilot-operated relief valve and a non-return valve. A small piston H is placed between the non-return valve and the relief valve (Figure 10.11); its diameter is usually 15% larger than the surface of the conical poppet 2 of the relief valve in order to develop a higher force than that of the spring 3 countered by poppet 2.

When the accumulator (AC) is being filled (see Chapter 14), the valve works as usual because the spring retains the conical poppet. The fluid pushed by the pump overcomes the non-return valve (K) and reaches the accumulator.

When pressure exceeds the level set (Figure 10.12), in other words as soon as the accumulator is full and the actuators experience override pressure, the relief valve unloads some fluid into the tank and the non-return valve closes. As a result, the fluid cannot reach the accumulator and the non-return valve prevents it from flowing back to the relief valve, which ensures the pump is on stand-by.

The pressure within the circuit acts on the piston (H) and keeps the conical poppet (2) open. The piston (H) is crucial because it enables the accumulator to unload an adequate amount of fluid or to transfer it to the actuators; failing the piston (H), the valve would constantly change its position while the conical poppet would continuously open and close.

As the fluid flows from the accumulator to the actuators, pressure drops; the force exerted by the spring (3) overcomes the piston (H) force and restores the valve initial filling position.

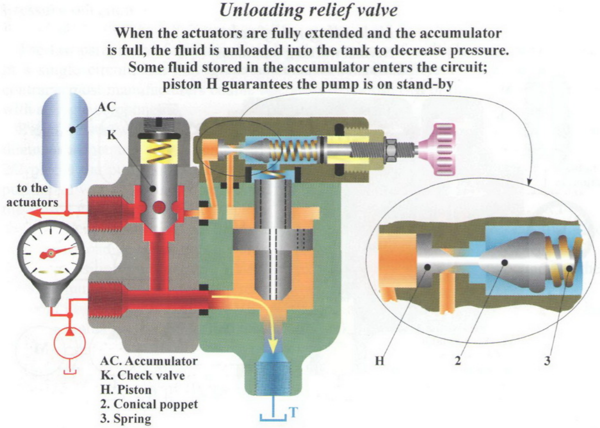



*Figure 10.12*

## PRESSURE REDUCING VALVES

Circuit relief valves control higher pressures in the actuators, but they cannot protect an actuator that needs lower pressure while the others are working at a higher pressure With a few exceptions, a relief valve is suitable for a system with a single actuator, but two or more actuators demanding different pressures need one or more pressure reducing valves for lower pressure circuit. Pressure reducing valves are based on this principle and have special devices to control the downstream pressure constantly (some special measures are taken not to alter the low downstream pressure).

Pressure reducing valves have almost the same design as safety valves, but they rest on the opposite principle.

- *Safety valves*: they prevent pressure surges in the whole circuit. They are closed (which means they do not act) if the circuit is at nominal pressure. The inlet port P ensures piloting. It does not need any drain.

- *Pressure reducing valves*: they reduce pressure in a part of the circuit; they are usually open when the circuit is under normal working conditions. When upstream pressure is higher and exceeds the limits set, downstream pressure is kept constant. The outlet port ensures piloting. Drain is essential.

In order to understand their operating principle better, consider a simple punching machine made up of a hydraulic vice and a punch operated by a single-acting cylinder (Figure 10.13).

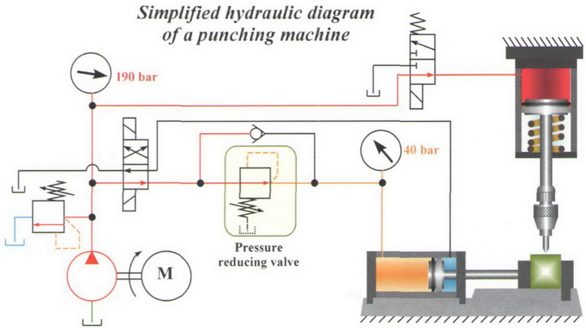



*Figure 10.13*

Since the actuating punch demands the higher pressure, the relief valve must be set accordingly. If the vice is subjected to this maximum pressure, it can deform the object it clamps, consequently it needs a pressure reducing valve.

It can be presumed that, in such a simple system, pressure can be reduced by means of single diaphragm restriction (perforated disc) to the detriment of speed (restrictions reduce the flow rate) in order to simplify the design. Actually, vice closing needs little pressure, but when the vice is holding the workpiece (static phase), the pressure is the same as that of the relief valve (Bernoulli' principle, Chapter 1). The check valve, in parallel with the pressure reducing valve, is essential to restore the vice rest position quickly.

In brief, *restrictions* are a very economical method to reduce pressure during the actuator movements (dynamic phases), but *not during static phases*. *Pressure reducing valves* instead keep a constant downstream pressure *during both static and dynamic phases*, obviously provided that upstream pressure does not drop below the secondary level.

**Direct-acting pressure reducing valve**

Direct-acting pressure reducing valves consist of a spool (a/b) countered by spring (m) that is set by means of a screw/knob (v), a supply port (p1), an outlet port (p2) and a drain port (Y). A channel connects the lower face of the piston (b) to the outlet port (Figure 10.14).

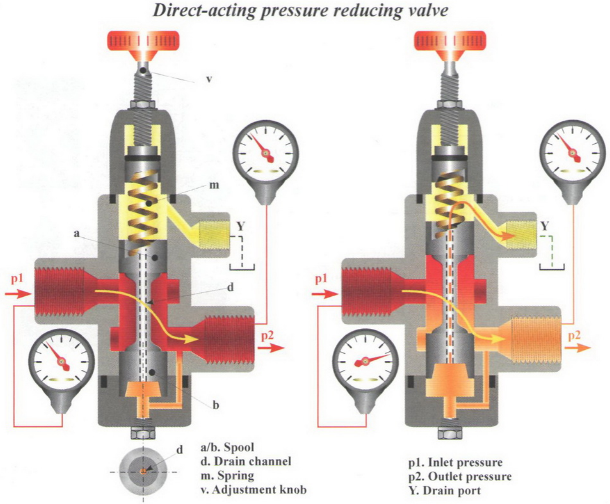



*Figure 10.14*

When pressure does not exceed the maximum level, the spring set manually keeps the spool (a/b) totally open. The pilot fluid acting on the lower face of the piston (b) cannot outweigh the spring (m). As pressure surges, the pilot fluid acting on the piston (b) exceeds the spring force and pushes the spool upward, thus creating a restriction that reduces outlet pressure.

In static phases (for instance when the piston is in the end-of-stroke position), the drain channel (d) inside the spool and connected to the tank via port Y guarantees a low and constant pressure.

As previously said, fixed restrictions cannot reduce pressure in static phases; if the spool was motionless in these phases, it would serve as a fixed restriction with an ensuing pressure rise on the actuator. On the contrary, pilot pressure drops as the fluid flows out from the pilot chamber through the drain and the spring pushes the spool downward increasing the flow section; yet, since there is no flow, it almost closes (the weak drain pressure prevents it from closing). This occurs time and again throughout the static phase of the actuator. Therefore, the balance between the internal drains in the almost-closed position avoids vibrations.

Although all direct-acting pressure reducing valves rely on the same operating principle, manufacturers devise different versions (for example, like in Figure 10.15); as a matter of fact, they design different spools, pilot systems, positions of internal ways and they conceive valves for panel mounting, flange mounting or, like in the Figure, stack valves.

***Direct-acting pressure reducing valve***

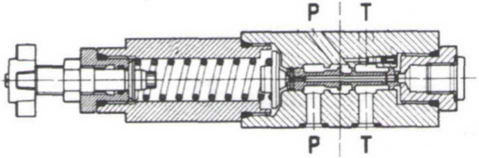



*Figure 10.15*

**Pilot-operated pressure reducing valve**

Pilot-operated pressure reducing valves (Figure 10.16 shows a standard version) have a more precise pilot system than direct-acting pressure reducing valves. They are quite similar to two-stage relief valves, but they have different spool designs and the pilot fluid must be unloaded via an external drain connection placed over the pilot stage.

Like in direct-acting pressure reducing valves, the fluid leaks from (b) to (a) through the spool channel (d) and acts on the conical poppet (c). Unlike direct-acting versions, they cause a balance on the spool itself; the only task of the spring (mr) placed on the spool upper face (a) is to keep the spool at rest during the operating phase with inlet pressure equalling outlet pressure (Figure 10.17). The pressure surge overcomes the force of the pilot spring (m) and the pilot fluid in unloaded via the drain port (Y). This guarantees optimum and constant spool balance, which is undoubtedly more precise than in single-stage direct-acting versions. The amount of drained fluid ranges between 0.4 and 0.8 l/min depending on the versions.

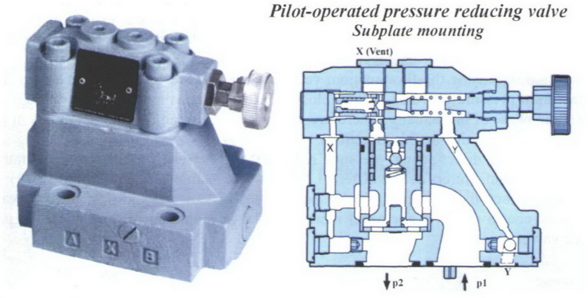



*Figure 10.16*

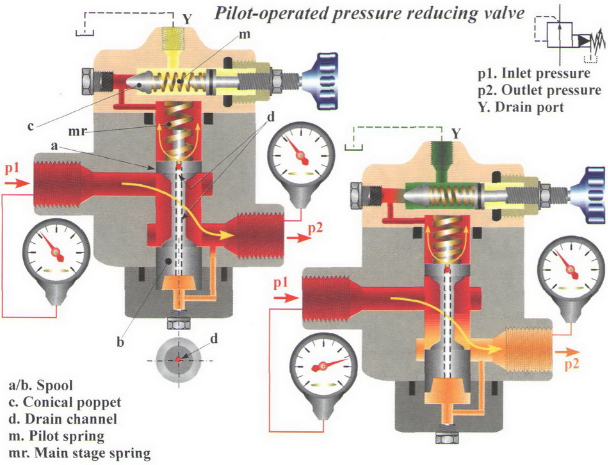



*Figure 10.17*

As Figure 10.17 shows clearly, the pilot stage consists of a relief valve. Stack valves (see the last paragraph of Chapter 8) have two axial stages. The stack pressure reducing valve in Figure 10.18 is equipped with a pilot ball and a turning knob that can be locked with a key; a pressure gauge can be connected to its right end.

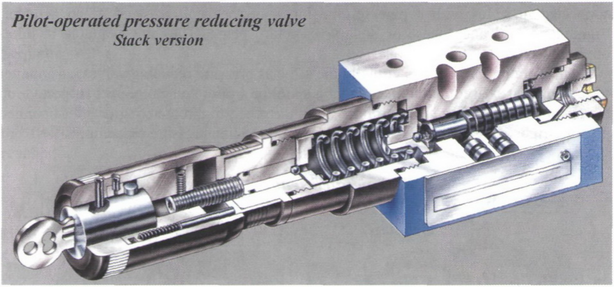



*Figure 10.18*

In the reverse operating phase, for instance if the pressure reducing valve is active on the advance phase of the actuator (see Figure 10.13 – *Simplified hydraulic diagram of a punching machine*), the unloaded fluid flowing in the reverse direction can cause hydraulic imbalances upon the return of the piston in the actuating cylinder.

For this reason it is crucial to connect a by-pass check valve in parallel with the pressure reducing valve (Figure 10.19). Standard pressure reducing valves or versions provided with check valve are available at any wholesale business.

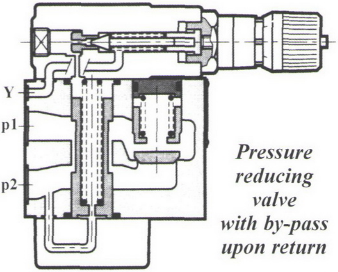
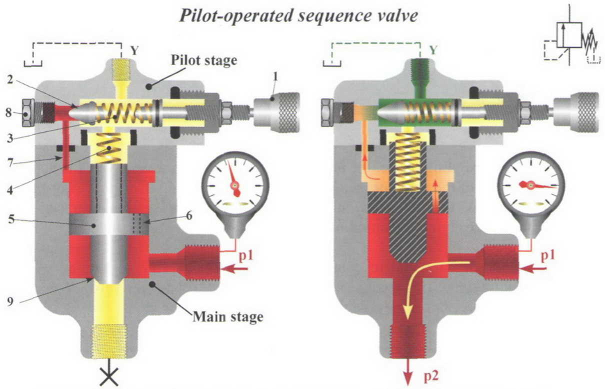



*Figure 10.19*

## PRESSURE CONTROL VALVES

Pressure control valves monitor the pressure acting on a part of the circuit in pre-established moments of the operating process. Except for few versions, they are usually connected in series with a pipe for the thrust or the back pressure of an actuator, triggering its advance phase or braking its return phase. They are available in pilot-operated or direct-acting versions.

Their operating principle is virtually the same as pressure reducing valves; as a matter of fact, they are based on the balance of a spool or a pilot conical poppet subjected, on the one hand, to the force of a spring and, on the other, to direct-acting or pilot-operated pressure (it depends on its design). They need a drain like pressure reducing valves.




**1. Adjustment knob 2. Pilot conical poppet 3. Conical poppet spring 4. Main stage spring 5. Piston 6. Balance fluid channel 7. Line connecting the two stages 8. Vent connection cap 9. Poppet p1. Supply port p2. Secondary port Y. Drain port**

*Figure 10.20*

Their similarity with relief valves is evident because they too are closed in the rest position, but they differ in the type of outlet. While relief valves unload some or all the fluid via the port directly connected to the tank, pressure control valves have a primary port, a secondary port and a third port for pilot fluid drain, like pressure reducing valves.

Figure 10.20 shows the most popular pressure control valve, that is to say a sequence valve (the Figure portrays a two-stage version). Like relief valves, also present sequence valves have a cartridge insert (main stage).

Pilot-operated valves (like that in Figure 10.20), which are ideal for high flow rates/pressures, are less popular and practical than direct-acting valves with reversible heads and spool, which are suitable for several control systems. This text considers only the latter for simplicity reasons.

### Sequence valve

Sequence valves are typically used to control two actuators 'in sequence'; in other words, if there are two cylinders, the second one is set into motion only after the first one has accomplished its sliding or in any case when the set pressure is reached.

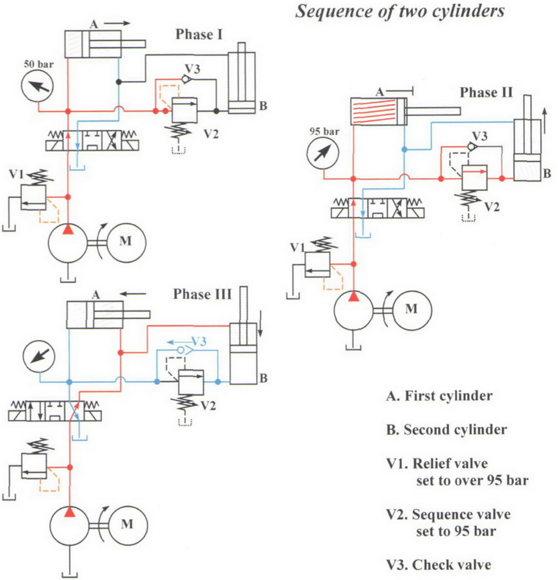



*Figure 10.21*

The circuit in Figure 10.21 reflects this mechanism. The command of the directional valve (phase I) sets cylinder A into motion at a pressure below 95 bar, which is the pressure set on the sequence valve V2. When piston A moves to the end of its stroke (Phase II), pressure rises up to 95 bar and the sequence valve opens so that the fluid can flow, resulting in the rod sliding out of cylinder B.

The opposite position of the check valve makes the rod slide back to the rest position (phase III). The unidirectional valve V3 that cuts off the sequence valve promotes the return of B.

Such a system is clearly perfectly suitable for motors but, if there are two motors, the second one must be started only after stopping the first motor at the maximum or set pressure.

Sequence valves are employed in a wide range of devices and we are now going to describe one of their easiest applications that are a common sight in everyday life. Refuse collection vehicles are essentially equipped with a device for lifting wheelie bins and a compressing system inside the truck container. Packing can start only after waste is dumped into the body of the vehicle. The two lifting forks, usually provided with two parallel cylinders having the same dimensions, grab, lift and flip wheelie bins upside down as rods are in the advance phase. When bins are flipped upside down, lifting cylinders are in the end-of-stroke position; pressure reaches the level set on the sequence valve that opens allowing the pressing cylinder to advance. The lifting actuators in the diagram in Figure 10.21 are cylinder A, while cylinder B performs pressing.

The symbol of sequence valves under ISO 1219/1 looks like that of relief valves, except for the outlet port that is connected to the actuator instead of the tank. Even if the standard does not establish it, it is advisable to add the tank drain line along with the spring.

Figure 10.22 depicts a direct-acting sequence valve with internal and external pilot control (if there is external pilot control, the solenoid valve must be connected to a pressure branch that enables the sequence as soon as a specific level is reached).

Sequence valves can have an internal non-return valve for the unloading phase of the actuator (Figure 10.23). During phase II (circuit in Figure 10.21 – *Sequence of two cylinders*), the pressurised fluid flows through the holes of the non-return valve; when the piston of actuator B returns, i.e. during phase III, the fluid that overcomes the upper part of the adequately-shaped spool (some manufacturers prefer to enlarge the clearance on the valve body) pushes the non-return valve poppet downward, thus flowing to the tank.

Most manufacturers provide also stack versions of spool direct-acting sequence valves respecting the ISO standards on mounting surfaces (Figure 10.24).

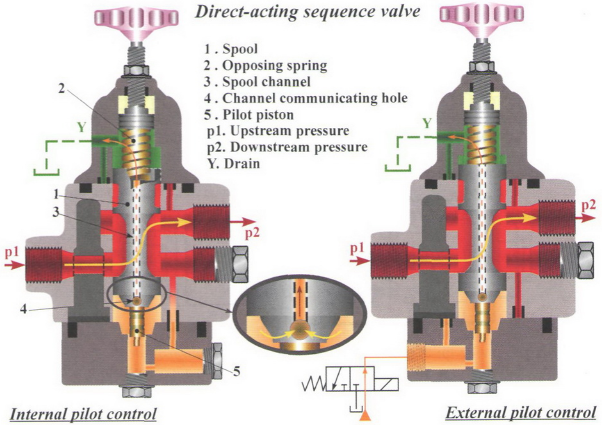



*Figure 10.22*

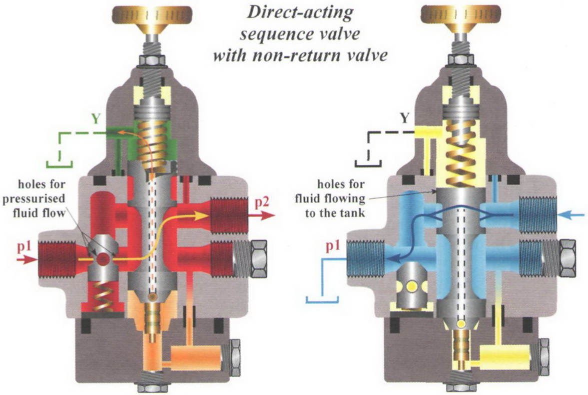



*Figure 10.23*

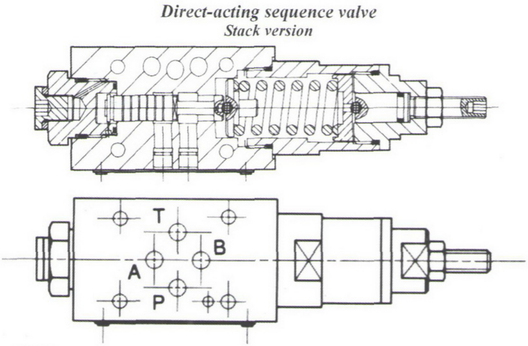



Figure 10.24

**Back pressure valve**

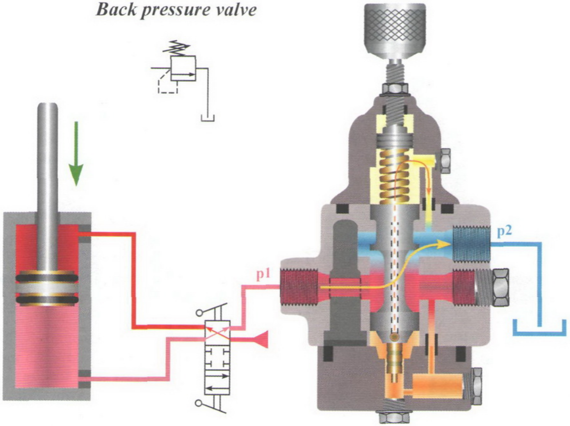



Figure 10.25

Back pressure valves control unloading pressure according to the overpressures that can develop within the system and they are thus used to make the piston move as soon as the pressure set is reached.

Unlike the valves analysed so far, the return pipe from the cylinder is connected to port p1, whereas p2 is directly connected to the tank (Figure 10.25). The valve obviously works in both the forward and return stroke of the actuator.

When the pressure on the piston of the actuating cylinder equals zero, the spring ensures the valve is closed and there is no pilot pressure. When pressure acts on the actuating piston, back pressure develops immediately between the outlet port of the cylinder and the spool that is still closed. The pilot fluid from the out line of the distributor triggers the upward movement of the spool that opens the clearance, thus connecting the cylinder line to the tank.

Pressure changes on the piston affect the part subjected to back pressure, resulting in the spool opening or closing. This means that back pressure increase makes the outlet orifice open more whereas the valve will tend to close if it drops.

Figure 10.25 shows clearly that the operating principle and the symbol of back pressure valves are almost the same as those of direct-acting relief valves: the difference lies in the way they are connected to the circuit.

**Unloading valve for high/low pressure double pump**

Presses equipped with a double fixed displacement pump system with high/low flow (low/high pressure) need a mechanism that disconnects the prefill pump, that is to say high flow and low pressure, when pressing is carried out. This issue has been already covered in Chapter 5, § In-depth Analyses, Double fixed displacement pump for high and low flows, but we are now going to consider it as we explore the device for disconnecting this pump.

Unloading valves are controlled by an external hydraulic signal from the line related to the other high-pressure pump. Upon cycle starting, both pumps deliver the flow needed for a quick advance phase. During pressing, pilot pressure acting on the valve spool makes it open connecting the low-pressure pump to the tank. The disconnection, or rather the stand-by mode of this pump, allows to save much energy on the prime mover and to avoid a useless strain of the hydraulic generator. As the hydraulic diagram of Figure 10.26 shows evidently, the non-return valve cannot be arranged inside the unloading valve.

Another mobile application of high/low flow double pump with unloading valve consists in vehicles equipped with a crane. During the advance and return phases, cranes need high speed and low pressure because there are no objects to move during these phases. When objects are fixed to a crane and then moved, the load exerts a considerable force and the pilot from the line related to the high-pressure pump opens the unloading valve, placing the other pump on stand-by; since only the low-flow pump is working, the movement is adequately slow.

In this manner, it is the machine itself that determines the movement speed. Drivers have to handle only one lever and no dangerous abrupt movements can occur when the load is fixed to the crane. However, these systems are often based on the *regeneration* technique (see Chapter 17).

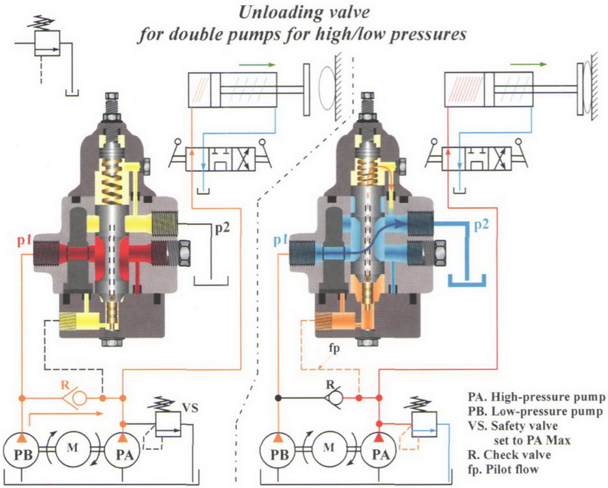



*Figure 10.26*

### Brake valve

Any motor deprived of its energy source is subjected to the inertia of the driven rotating masses. Closed centre valves cannot be blocked instantly because the strong torque on the shaft, due to the driven rotating mass inertia of the load, undoubtedly causes very serious mechanical damage, like bearing wear, overheating, misalignment and even break of the shaft itself.

As a result, braking has to be performed by a valve that gradually restricts the outlet port and a directional valve 3/2 is essential if there is only one revolution direction; two opposite brake valves and a distributor 4/3 with by-pass centre for both rotary directions.

Brake valves (we refer only to the single-revolution direction for simplicity reasons) need two pilots, i.e. an external pilot connected to the pressure line and an internal pilot.

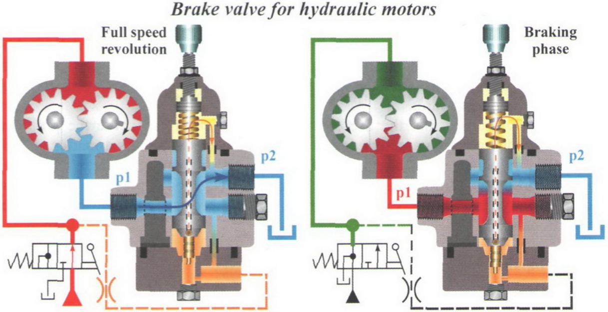


*Figure 10.27*

The pilot fluid from the pressure line lifts the spool during the revolution; the spool then connects p1 to p2 and allows the fluid to be unloaded into the tank through the directional valve (Figure 10.27).

If the distributor is arranged in the opposite position, the pump is in by-pass, the motor keeps rotating due to inertia, there is no external pilot fluid for the time being and the spool moves down blocking port p2. Back pressure develops between the motor outlet port and port p1 of the brake valve and it acts on the internal pilot port; the spool slides up again thus opening a tiny clearance to the tank. Revolutions slow down gradually, back pressure decreases, slowly opening the tiny clearance until the valve closes and the motor stops. The metering orifice on the external pilot line partially but adequately prevents pressure drops of the fluid in the chamber next to the pilot: failing this, the pilot fluid would reach the tank through the directional valve and the brake valve closes immediately.

This too accounts for the popularity of hydraulic motors as opposed to three-phase electric motors. Indeed, although the latter demand less energy, they need heavy and large mechanical devices, unsafe reverse current deceleration circuits or delicate and sophisticated electronic circuits for braking.

## COUNTERBALANCE OR OVERCENTRE VALVES

ISO 5598 defines a counterbalance valve as a 'pressure control valve which maintains pressure to prevent a load from falling or overrunning'. Consequently, they fall into the category of pressure control valves but they differ from the previous type in their specific task and their adaptability to many systems (after making some modifications). These valves are often referred to as **overcentre valves**, **load control valves** or **boom lock valves**, although such terms are not set by ISO standards.

An object in free fall, or rather, countered only by pressure drops in the totally open directional valve and the pipes connected to the tank, speeds up more and more. Furthermore, the high falling speed can require a higher flow rate than that delivered by the pump on the opposite side of the actuator resulting in overheating and cavitation. The load can obviously be arranged in extraction or retraction, in other words it can be faced upward or downward.

It can be presumed that the use of a simple restriction can solve the problem; actually, its operating parameters do not satisfy its needs at all. As a matter of fact, the restrictor connected to the tank develops a non-stop back pressure, even during normal working conditions; as the load falls too quickly, back pressure on the flow reducing valve increases, even though it is not precise enough to guarantee proportionality.

**Counterbalance valve**

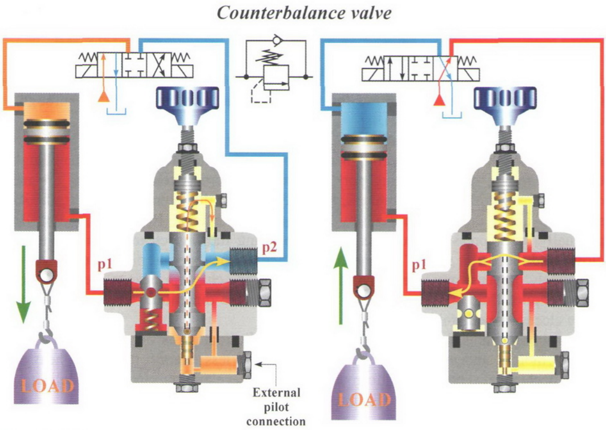



*Figure 10.28*

Counterbalance valves and brake valves fundamentally share almost the same operating principle. The load is faced downward; the pressurised fluid acting on the face of the piston that is opposite the load allows it to descend, the lower port of the cylinder is connected to the port p1 of the counterbalance valve and p2 to the directional valve. Since the spring keeps the valve spool in the rest position in the beginning, back pressure develops between the cylinder and the valve, which lifts the spool via the internal pilot. The fluid flows through the resulting valve opening to the tank. When the load is lifted instead, the check valve inside the counterbalance valve allows the fluid to flow directly during retraction (Figure 10.28).

***Remote control counterbalance valve***

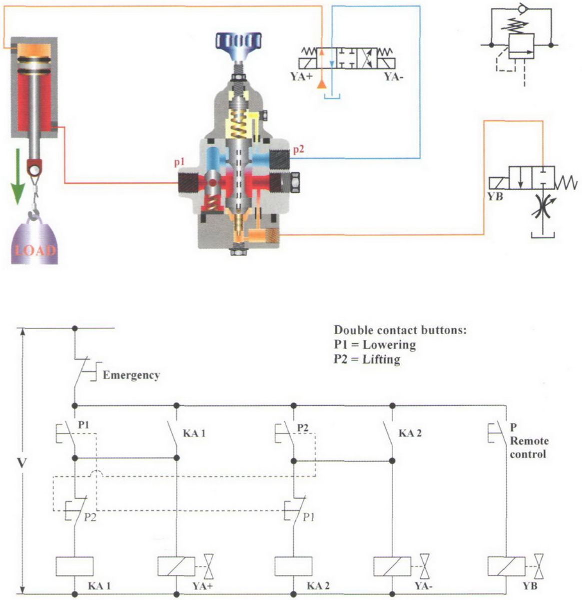



*Figure 10.29*

The unit can be connected to the tank by adding a simple remote control consisting of a solenoid valve 2/2 subjected to its control button (Figure 10.29).

If the load has to be controlled in both directions, two opposite valves are connected.

Chapter 12 ('Cartridge Valves') further explores overcentre valves.

**Motor counterbalance valves**

Hydraulic motors for load lifting (hoists) that may cause the free fall of a body need not only a relief valve (it should not be confused with the brake valves for motors previously analysed, which is suitable only for stopping the rotating mass inertia but also a device that blocks the suspended load instantly in any point of the stroke. Neither one nor more non-return valves can guarantee blocking because leakages inside the motor are unavoidable and the load lowers inevitably.

Motors have to be stopped by means of a mechanical shoe or disc brake that acts on the output shaft directly and this device can be handled with a simple hydraulic cylinder. This single-acting cylinder, which is in the rest position during blocking for safety reasons, is set operated (i.e. it neutralises the friction of brake shoes on the shaft) during the lifting or lowering of the actuator. As a result, the counterbalance valve is provided with a shuttle valve that operates the braking cylinder during load handling (Figure 10.30).

Besides in normal lifting winches, these valves are essential in the motor device for net winches on fishing boats.

*Counterbalance valve for hoists*

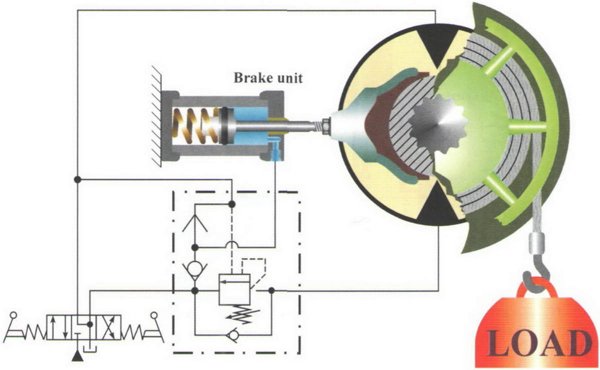



*Figure 10.30*

Chapter 11

# FLOW CONTROL VALVES

Flow control in hydraulic circuits involves a speed change of the actuator. A lower flow rate reduces the speed of a cylinder or diminishes the number of revolutions on the shaft of a motor, whereas a higher flow speeds up every type of cylinder. Like pressure, flow changes too result from a reduction or enlargement of the flow cross-section, yet section reductions cause some problems: besides reducing speed, the restriction in series with the port of a cylinder develops differential pressure, fluid overheating and abnormal movement of the user.

The need for precise and regular movements demands the use of special compensated flow control valves while simple systems require just some simple devices for speed control, like fixed or adjustable restrictors. Hydraulic equipment provided with electronic control delivers the best results also in terms of flow control.

## RESTRICTOR VALVES

Cross-section restrictions are clearly the simplest and cheapest method to reduce the speed of actuators. However, this often leads to poor results because the transfer is irregular and the fluid is subjected to rather detrimental temperature changes.

Movement irregularity is due to a number of causes we are going to explore briefly. As long as the pressure $p_c$ developed by the load is less than the maximum pressure $p_{max}$ set on the relief valve, the speed of the rod or the drive shaft is constant, provided that the differential pressure $\Delta p$ across the restrictor plus $p_c$, does not reach the maximum level $p_{max}$ set on the relief valve. Actually there is no flow reduction during this phase: the whole flow of the pump reaches the actuator that is working at a reduced pressure. To put it simple, as the fixed displacement pump delivers a fixed flow and the relief valve is off, the whole flow cannot but end up to the actuator.

Problems arise when $\Delta p$ plus the pressure developed by the generator, now subjected to a higher load, reach the $p_{max}$ set. The relief valve unloads some fluid into the tank and the actuator diminishes speed. If these parameters did not vary at all, the movement would be uniform, but this is not always the case. Load is hardly constant, therefore pressure $p_c$ changes accordingly; as $p_c$ rises, the differential pressure $\Delta p$ too between the restrictor ends changes. As a result, pressure/flow changes cause asymmetric movements of the piston or irregular motor revolutions.

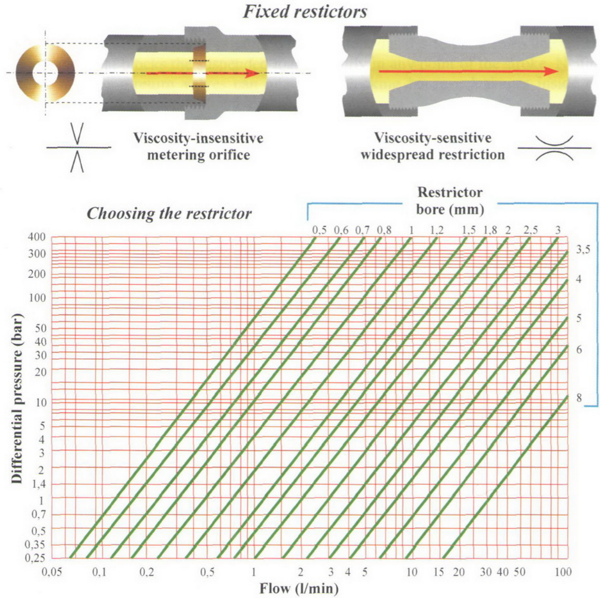



*Figure 11.1*

Nonetheless, restrictors offer some major advantages: they are light, occupy little space and maintenance workers can even build them by themselves in emergency situations. Unfortunately, they are often used inappropriately to reduce pressures or speeds, thus causing oil overheating and ensuing operating anomalies; the actual task of restrictors in a number of simple mobile applications is often just to 'cushion' the end-of-stroke motion of the rod, while a simple cushion would be perfectly suitable for this.

### Restrictors

The reduction of the flow cross-section in diaphragm restrictors (left drawing in Figure 11.1) is the result of the inclusion of a simple disc with some holes in a sleeve; flow reduction depends on the calibrated bore. In diaphragm restrictors, also referred to as thin-walled restrictors, pressure/flow drop is due to the whirlpools that develop downstream of the disc and there is almost no friction between the fluid molecules. For these reasons, restrictors are not affected by viscosity and, what is more, they are not subjected to temperature alterations.

These very adaptable components are often arranged between the pipe and the fitting of the port of an actuator or a directional valve; they are widely employed in many applications, especially in the mobile field.

*Widespread restrictions* (right drawing in Figure 11.1) are made up of a restricted pipe length placed between two pipes via fittings or, like diaphragm restrictors, between the pipe and the component. Since the internal friction between the liquid and the restriction makes pressure drop, *widespread restrictions* are sensitive to viscosity and temperature.

### Adjustable two-way restrictor valves

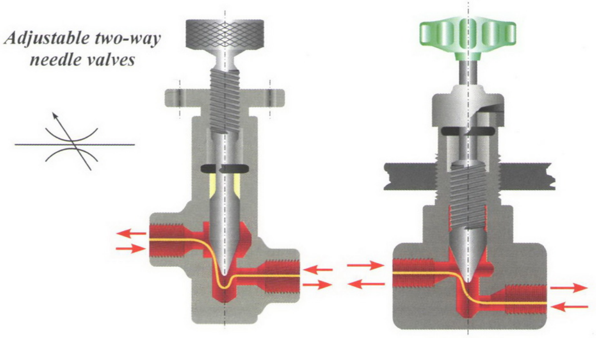



*Figure 11.2*

The flow cross-section can be adjusted by means of a needle-like plunger that is operated by a screw provided with a handwheel or a turning knob (Figure 11.2); pipes are connected to the valve body. By turning the handwheel or the knob clockwise, the flow passage is gradually restricted, thus reducing flow.

Besides needle valves, there are other types of adjustable restrictor valves, which confront systems with the same problems though because they differ just in the design. Figure 11.3 shows some designs of adjustable restrictor valves other than needle valves.

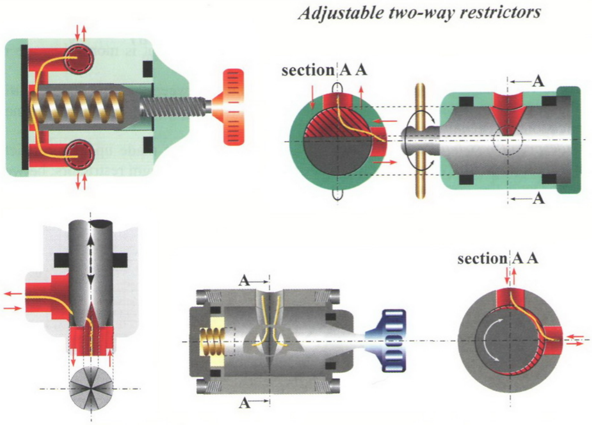



*Figure 11.3*

### Throttle check valve

Throttle check valves, usually held in a single block, are made up of two components: an adjustable two-way restrictor in parallel with a check valve.

In the in-to-out direction (Figure 11.4), the fluid from the inlet port flows to the outlet port through the reduction chamber whose restriction depends on the position of the needle; the check valve blocks the flow. In the reverse direction (out-to-in), the check valve connects the two ports cutting off the chamber.

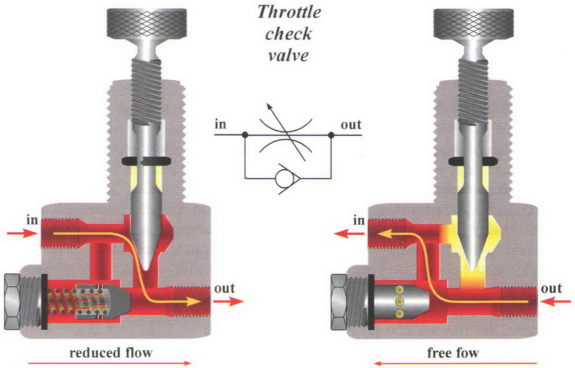



*Figure 11.4*

Despite sharing the same operating principle, one-way **pinch** valves (two-way versions are available as well) have a quite unusual design.

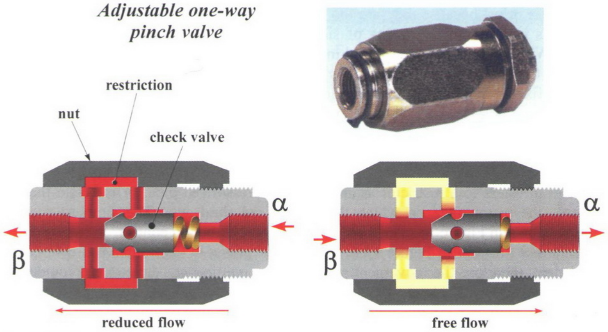



*Figure 11.5*

Pinch valves (Figure 11.5) reduce the flow from α to β; by tightening or loosening the nut, the restriction between it and the valve body is reduced or widened while the spring and the inlet pressure α keep the check valve in its seat. Opening the check valve promotes the full flow from β to α: the pressure in β overcomes the spring force and pushes the poppet allowing the fluid to reach port α.

## SPEED CONTROL WITH RESTRICTORS

Rod sliding speed depends first of all on the cylinder capacity. Pump flow and pressure being equal, the rod of a cylinder with a small bore takes less time reaches the stroke end than a same-stroke cylinder with a higher bore.
For instance, the pumps of two independent circuits delivering the same flow ($Q_1 = Q_2 = 30$ l/min) simultaneously supply their respective cylinders ($c_1$ and $c_2$) having different volumes and the same bore. If $c_1 = 6$ litres and $c_2 = 15$ litres (both loads exert the same force and pressure drops are equal), the rod of the smaller cylinder takes less time to reach the stroke end:

$$t\,c_1 = \frac{c_1}{Q_1} = \frac{6}{30} = 0.2\,\text{min (12 seconds)}$$

$$t\,c_2 = \frac{c_2}{Q_2} = \frac{15}{30} = 0.5\,\text{min (30 seconds)}$$

The time the rod of a cylinder takes to travel the stroke can be modified in two manners: that is to say by using a variable displacement pump or by adding some proper flow control valves to the circuit. As variable displacement pumps have to be ruled out a priori because they are too expensive in uncomplicated applications, flow control is carried out by means of simple restrictors or compensated flow control valves. While the latter are explored in the following paragraph, we are now going to focus on the position of restrictors. Variable restrictors are covered, but the same rules apply to fixed versions, provided the flow cross-section is set accurately.

### Meter-In

Meter-in refers to a restrictor arranged in the pressurised pipe, or rather in the pipe towards the piston face on which the pressurised fluid acts. The simplest meter-in circuit, shown in Figure 11.6, enables extraction and retraction control.

In the left drawing in Figure 11.6, the piston sections in the double rod cylinder are equivalent, therefore the movement speed is the same in both directions, provided the load is constant. Problems arise instead if the meter-in technique is applied to differential cylinders. If the relief valve is set to the maximum pressure during the forward stroke and during the return the rod has to drag the same load, the set pressure is not enough to move the load due to the section difference (rod reduces it). Assuming a force F equals 1000 daN and the sections of a differential piston are $S_1 = 20$ cm$^2$ and $S_2 = 10$ cm$^2$, the pressure needed ($p = F/S$) to move the load is (pressure drops are not taken into account):

$$p_{forward} = 1000 \text{ daN}/20 \text{ cm}^2 = 50 \text{ bar}$$

$$p_{return} = 1000 \text{ daN}/10 \text{ cm}^2 = 100 \text{ bar}$$

Return cannot occur if the relief valve is set to the forward pressure of 50 bar. If the return level is set to 100 bar, the success of the operation is not totally guaranteed: the forward movement is too quick and, if higher abnormal forces develop, the rod continues to travel its stroke all the same and this risks damaging the object to be moved. In addition, in this latter case, in order to reduce the forward stroke speed, the valve should be restricted so that the differential pressure $\Delta p$ equals 50 bar (100 – 50), resulting in energy waste and much higher temperatures than the maximum tolerable level; however, return is no longer possible at a reduced pressure, unless there are very low speeds.

What is more, there is another insurmountable difficulty. As a matter of fact, when the valve is in the central position (hence the pumps should be on stand-by), the restriction entails very high dissipation, leading to energy waste and much heat. For this reason, meter-in must absolutely be ruled out in all real applications under these conditions.

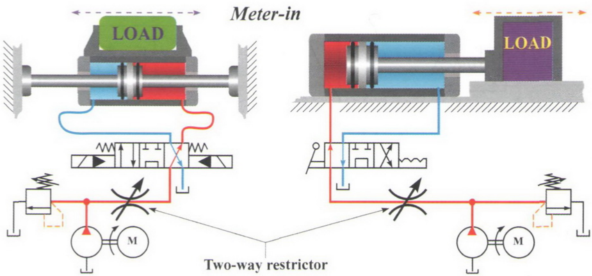



*Figure 11.6*

In order to obviate these problems at least partially, it is advisable to arrange two throttle check valves between the ports of the directional valve and those of the cylinder, like shown in circuit A in Figure 11.7. During the forward stroke, valve $V_1$ reduces the flow while the check valve of $V_2$ allows a free flow to the tank; during the return stroke, $V_2$ entails a restriction whereas $V_1$ enables free flow to the tank.

Circuit B experiences less dissipation because the actuator pressure affects the relief valve directly and continuously. Uncertainties are drastically reduced if the load is moved only in one direction as the previous remarks do not apply to returns without loads.

Even if meter-in could be used for horizontal handling (resistive loads), it is strongly advisable not to use this technique in vertical handling (tractive loads), i.e. if the load experiences free fall. Since the fluid flowing to the tank is not confronted with restrictions (the restrictor is by-passed due to the check valve), it gains speed and gets out of control.

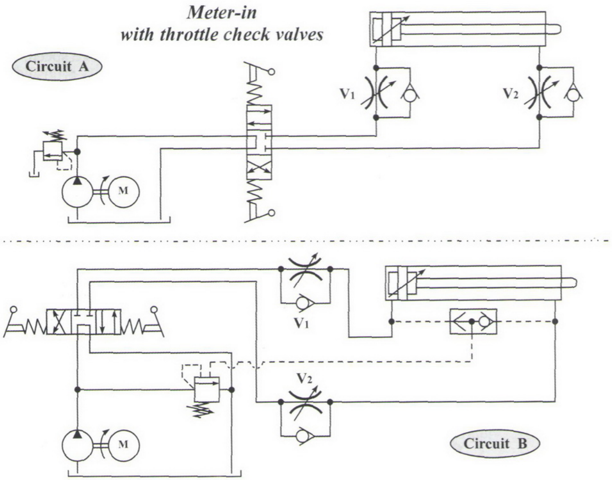



*Figure 11.7*

## Meter-Out

Meter-out is fundamentally the opposite of meter-in because the throttle check valve is downstream of the actuator and it is mainly suitable for tractive loads.

Figure 11.8 shows the options available. In circuits α and β the throttle check valve develops back pressure, which counters the fall of the load thus reducing the lowering speed; in circuit γ the load is controlled in both directions. Many applications of circuits α and β need to limit also the reverse direction by adding an additional throttle check valve, like in circuit γ.

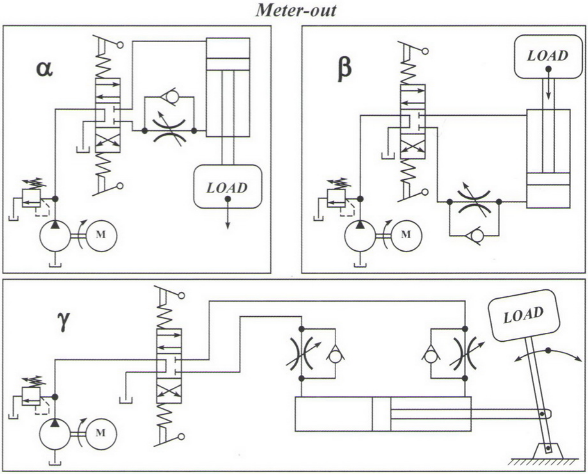



*Figure 11.8*

The meter-out technique is not problem-free. The throttle check valve can develop a rather high back pressure that demands very robust and perfectly sealed pipes and fittings; in addition, the hyper-pressurised fluid generates much heat in the valve restriction.

A simple example can clarify what has just been described. Assuming a cylinder pushes a horizontal load (Figure 11.9) and the balance between the relief valve and the throttle check valve enable the relief valve itself to allow a flow rate of 6 l/min to the tank, the cylinder is filled with 14 litres of fluid per minute in the $V_1$ chamber. Since one chamber is twice as large as the other, the cylinder pushes 7 l/min from $V_2$ to the tank during the forward stroke. If pressure drops are not taken into account and the cylinder is directly connected to the tank (which means there is no throttle check valve), the pressure of the outgoing fluid would equal zero (anyway, under these conditions pressure drops in well-designed systems would develop little back pressure, about 4 – 6 bar).

If a throttle check valve is mounted between the cylinder and the tank, the friction due to the restriction causes a dangerous back pressure. This adds to the working pressure, thus undermining the standard setting of the pressure relief valve.

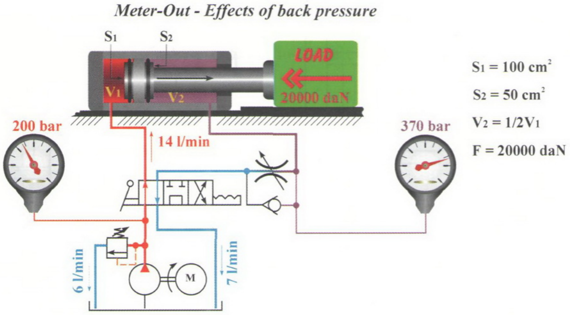



*Figure 11.9*

Another problem too needs addressing: every installer and maintenance worker knows that if throttle check valves are close to the directional valve while the cylinder is quite far away in meter–out circuits, upon starting the rod gains a high speed and then it slows down progressively because of the meter-out control. There is a simple solution for these situations: throttle check valve should be directly arranged on the actuator heads.

Since meter-out is suitable for tractive loads, the fall of the load causes an increase in the force F exerted by the load that further increases back pressure.

### Bleed-off

A third control method, known as *bleed-off*, is based on the diversion of some fluid to the tank directly through restrictors without the check valve (they cannot be referred to as two-way restrictors because the fluid flows in only one direction); restrictors are not in series with the pressurised pipe like in meter-in circuits, but in a T branch to the tank.

The inclusion of a single restrictor between the pump and the directional valve enables flow reduction in both directions of the rod, but forward and return speeds are not equivalent in differential cylinders. Flow can be roughly controlled in both directions by connecting a restrictor to both the ports of the actuator (Figure 11.10). This technique is not suitable for tractive loads.

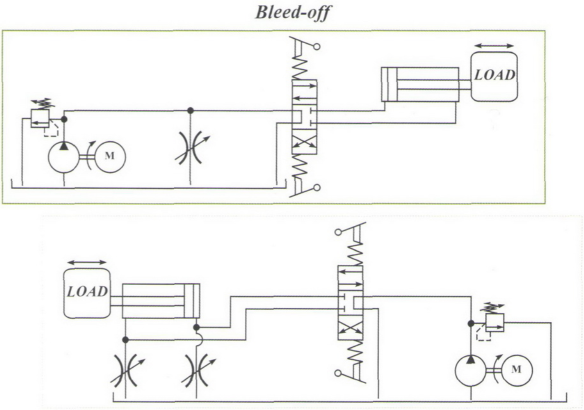



*Figure 11.10*

No high differential pressure across the restriction develops in bleed-off circuits because some fluid is unloaded to the tank and the pressure within the circuit results only from load. Much energy can be saved as the relief valve does not act (its pressure setting is slightly higher than the maximum nominal pressure of the load).

At first it can be presumed the restriction causes a remarkable $\Delta p$ that is proportional to the flow clearance. If the restrictor was totally closed, the fluid would directly flow to the actuator and nominal pressure would depend only on the load. A totally open restrictor would make pressure equal zero, provided its diameter was at least equivalent to the bore of pipes: as the fluid can flow freely through the valve, it would reach the tank. When the restrictor is partially open (and this is what actually occurs), some fluid is diverted into the tank, while the remaining fluid acts on the actuator. For instance (see Figure 11.11), if 3 l/min of fluid are unloaded through the restriction to the tank and the pump delivers a flow of 10 l/min, 7 l/min are conveyed to the cylinder; the load develops some pressure and the rod moves according to the inlet flow rate. The relief valve must be set to the load pressure plus the inevitable pressure drops in unloading pipes.

Unlike meter-in and meter-out circuits, bleed-off circuits do not manage to control the load directly because there is neither differential pressure nor back pressure. Consequently, meter-in and meter-out restrictors control loads roughly, even if much energy is used, because the relief valve is always open; in bleed-off circuits the control is linked to the maximum load only, but energy costs are cut since the relief valve does not intervene.

## COMPENSATED FLOW CONTROL VALVES

The types of restrictors described above provide poor results in terms of flow control reliability, especially if actuators are subjected to load variations, which is the case more often than not.

The die-holder slide of a machine tool must move at a constant speed; however, the tool faces various obstacles, like harder materials, hollow spaces resulting from previous machining or on the contrary small additional shoulders. The bucket of an excavator sustains different types of stress while removing beaten earth consisting of sand, tuff and stones. In brief, the load, i.e. the force that counters the slide or the digging bucket, is unsteady.

Load variations entail a constantly changing differential pressure $\Delta p$ that is proportional to the load itself in systems provided with restrictors. This affects the actuator movement.

The restriction cross-section must be fixed during machining since it has to be handled while the machine is not working. As the flow is roughly proportional to the root of the differential pressure, $Q = \sim \sqrt{\Delta p}$ (this has already been explored in Chapter 5, § In-Depth Analyses), a change in $\Delta p$ causes the flow to vary, which makes speed become unsteady. Consider a simple example: assume a circuit supplied by a fixed displacement pump delivering a flow Q of 40 l/min is equipped with a relief valve set to p = 150 bar, a restrictor, a cylinder with a bore S of 20 $cm^2$. If the load exerts three different forces ($F_1$ = 1000 daN, $F_2$ = 1600 daN, $F_3$ = 2200 daN) during the transfer, the pressures $p_{c1}$, $p_{c2}$, $p_{c3}$ developed by the load are (p = F/S):

$$p_{c1} = 1000/20 = 50 \text{ bar}$$
$$p_{c2} = 1600/20 = 80 \text{ bar}$$
$$p_{c1} = 2200/20 = 110 \text{ bar}$$

If the little approximation $Q = \sim \sqrt{\Delta p}$ is not considered, the flow rate downstream of the restrictor can be calculated with the formula $Q_v = \sqrt{\Delta p}$ if the clearance of the valve is about 1.5 mm. Since differential pressure $\Delta p$ is equivalent to the difference between the pressure set p and the load pressure $p_c$, the less the difference is, the less the flow rate is, resulting in an obvious speed drop:

$$Q_{v1} = \sqrt{p - p_{c1}} = \sqrt{150 - 50} = \sqrt{100} = 10 \text{ l/min}$$
$$Q_{v2} = \sqrt{p - p_{c2}} = \sqrt{150 - 80} = \sqrt{70} = 8.4 \text{ l/min}$$
$$Q_{v3} = \sqrt{p - p_{c3}} = \sqrt{150 - 110} = \sqrt{40} = 6.3 \text{ l/min}$$

Compensated flow control valves guarantee a constant flow on the actuator if differential pressures are unsteady.

The chart in Figure 11.11 highlights the trend of flow according to the pressure jump between upstream and downstream of a standard restrictor (red line) and a compensated flow control valve (green line).

***Flow trend vis-à-vis pressure jump***

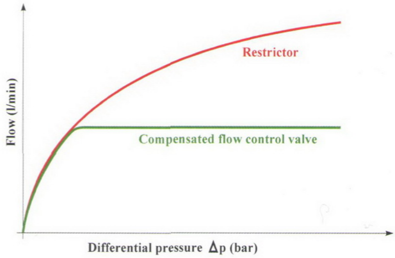



*Figure 11.11*

**Restrictor-type pressure compensated flow control valve**

Restrictor-type pressure compensated flow control valves are basically made up of a pre-settable restrictor and a double piston spool serving as a compensator; the inlet port (in) is connected to the compensation spool whereas the adjustable restriction chamber ends with the outlet port (out).

Under any load condition, the spool (2) (see Figure 11.12) must guarantee a constant differential pressure between upstream and downstream of the restriction (8) previously set by the restrictor (7) by means of a knob (6).

Since the pressure (chamber 3) on the internal faces of the piston is equal and opposite (hence balanced and irrelevant), hydraulic compensation develops on the external faces (chamber 1 and 5), supplied by the channels connected to the chamber respectively upstream and downstream of the restriction. As a result, the pressure $p_1$ within chamber 1 acts on the surface $S_1$ of the upper piston, while the areas $S_2$ of the lower piston is subjected to both the pressure $p_2$ (chamber 5) developed downstream of the restriction and the opposing force $F_m$ of the spring (4); the surfaces S of both pistons are equal. Consequently, the following formula determines *balance conditions*:

$$S_1 \cdot p_1 = S_2 \cdot p_2 + F_m \quad \text{or} \quad F_m = S \cdot (p_1 - p_2) \text{ therefore}$$

$$(p_1 - p_2) \text{ that is } \Delta p = F_m/S = \text{constant}$$

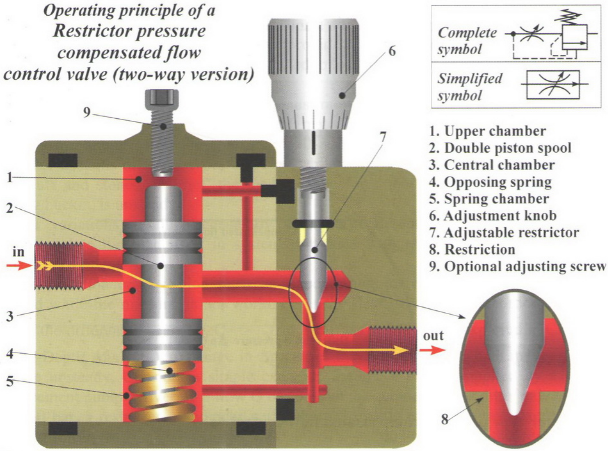



*Figure 11.12*

Load increase or decrease results in a proportional change in $p_2$ that, according to the previous formula, creates a new balance by opening or closing the inlet port (in) with the spool.

The anti-jump adjustment screw (9), which is usually optional, prevents the actuator from setting in motion abruptly. Failing it, if there is no load upon starting the system, the spring keeps the flow control valve totally open; full flow forces the actuator to setting into motion too quickly. To sum up, screw (9) adjustment reduces the inlet port passage in the beginning, in other words it serves as the compensation that is going to develop later on when the actuator is subjected to a load.

Large restrictor pressure compensated flow control valves are designed so as to sustain flow rates up to 300 l/min and a maximum pressure of 320 bar. In general, the minimum operating pressure, which is equivalent to the spring opposing force, is about 7-10 bar; the minimum controllable flow, which depends on internal leakages, valve design and nominal pressure, must be at least 0.5 l/min (0.02 in small valves).

**Pressure and temperature compensated two-way flow control valve**

Pressure and temperature compensated two-way flow control valves can be equipped with a non-return valve for return by-pass and they ensure both pressure and temperature compensation.

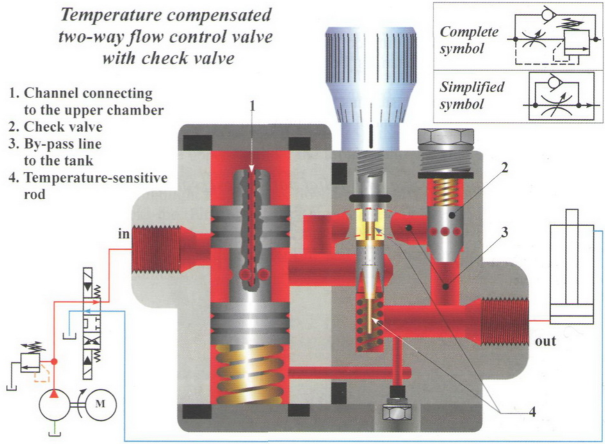



*Figure 11.13*

The spring and the load acting on the internal surface keep the poppet of the non-return valve (2) closed (Figure 11.13) during the advance phase of the cylinder (steady flow). During the return phase, the fluid opens the poppet and flows to the tank through the by-pass line (3).

An increase in fluid temperature would diminish viscosity undermining flow control because, pressure drops being equal, less viscous fluids increase flow. The rod (4) not only connects the adjustment knob to the restrictor, but also compensates temperature changes. As it is made of a heat-sensitive material, it lengthens whenever temperature surges occur narrowing the restriction slightly. The opening is diminished accordingly thus ensuring a constant flow.

Pressure and temperature compensated flow control valves are usually designed so as to sustain tolerances equalling about 3% if temperature rises/drops by 20 °C.

**By-pass type flow control valve with relief port to reservoir**

Unlike two-way versions, compensated three-way flow control valves (or compensated three-way flow control valves) are equipped with a relief valve that unloads the excess fluid to the tank.

When pressure exceeds the maximum level set in two-way flow control valves, the excess fluid is unloaded by the external relief valve regardless of the load on the actuator; the relief valve inside the valve body in three-way valves enables fluid unloading exclusively depending on load pressure. This means saving much energy and curbing temperature changes because fluid overheats less than in two-way valves.

Nonetheless, if the circuit sustains much strain, it is possible to use temperature compensated three-way flow control valves provided with the temperature-sensitive rod described above.

### *By-pass flow control valve with relief port to reservoir (three-way version)*

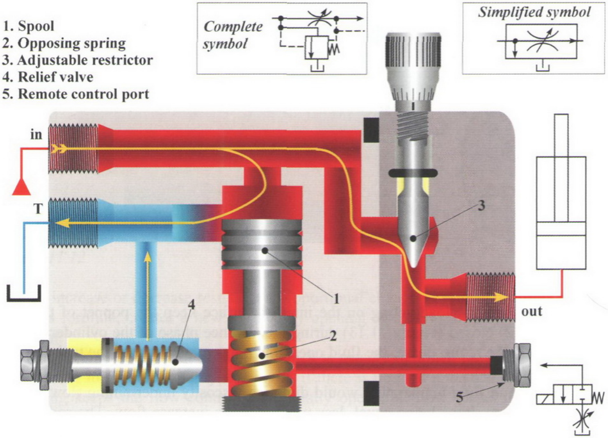



*Figure 11.14*

Like in two-way versions, under normal working conditions spool compensation results from the pressure upstream of the adjustable restrictor (3) that acts on the upper chamber of the piston of the spool (1) and from the spring force plus the pressure downstream of the restrictor that act on the lower chamber (Figure 11.14). Unlike pressure and temperature compensated two-way flow control valves, a partially opened spool makes excess fluid flow to the tank through port T.

If the piston is positioned like in the Figure 11.14 due to the setting; the actuator, which moves at a single preset speed, does not need the whole flow rate delivered by the pump, while the excess fluid is thus unloaded into the tank by means of the flow control valve (instead of the safety valve in two-way versions). Under steady conditions, the slight changes in load, hence in differential pressure, maintain spool compensation: the upward movement causes less unloading and vice versa.

As soon as load increases dramatically, higher outlet pressure further lifts the spool but it also opens the poppet of the relief valve (4) allowing some fluid to reach the tank. This makes pressure drop in the spring chamber; the spool moves down, restores the previous position and keeps the fluid conditions required.

A remote control can be connected to a dedicated port (5) to further reduce flow. If the fluid is unloaded to the tank via the port (5), the force in the lower chamber diminishes, whereas in the upper chamber the force due to working pressure pushes the rod downward, widening the opening to port T.

It is important to highlight that by-pass flow control valves, albeit high-performance, have some drawbacks too. First of all, they are more expensive than two-way versions and their installation costs are higher because they need a third pipe for unloading; secondly, they are not reliable in meter-out and bleed-off circuits and they are not compatible with accumulators.

***Compensated flow control three-way valve with fixed restriction***

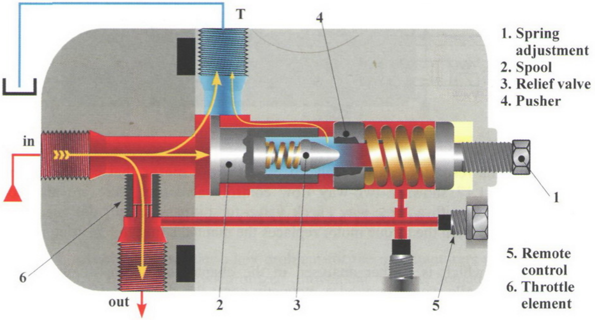



*Figure 11.15*

Figure 11.15 shows a special and compact compensated flow control three-way valve without manual restriction setting. The restriction depends on the interchangeable throttle element (6) while spring tension can be adjusted by loosening or tightening the screw (1). The relief valve (3) is held in the spool (2) and is subjected to the pilot action through the opening in the pusher (4). Remote control can be carried out with the connection to the port (5). By blocking port T (outlet port to the tank), this valve can serve as a pilot-operated relief valve.

**Electrical control of flow control valves**

Some applications need the actuator to perform a reduced or a full speed stroke on different occasions. For this reason, the compensated two-way flow control valve is provided with a solenoid valve 2/2 in parallel with the check valve in a single body. When the solenoid is at rest, the flow control valve reduces the fluid causing a slow stroke; upon applying voltage, the solenoid valve by-passes the flow control valve by supplying the maximum flow rate to the cylinder (Figure 11.16).

***Flow control valve with solenoid valve for maximum speed***

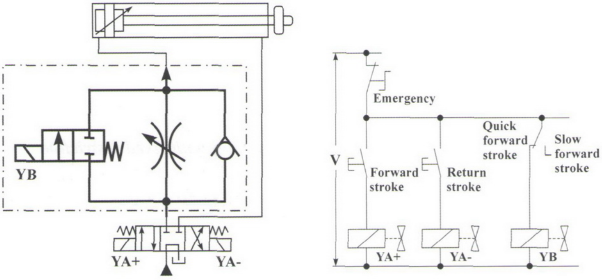



*Figure 11.16*

The adjustable restrictor can be replaced by a **torque motor** that controls a spool valve. Figure 11.17 shows a two-way flow control valve (but there are also some versions equipped with a three-way check valve for by-pass return) designed in the United States in which the torque motor replaces the manual adjustable restrictor.

This device, which is further analysed in the chapter on servo-valves, basically consists of two coils wound around a magnetic core; the differential electric current applied to the coils causes the movement of the core mechanically connected to the spool. The restriction degree depends on how much the holes in the directional valve are open. The current to the torque motor, hence the restriction, is set by turning a knob positioned on a dedicated remote control panel or a panel on the valve.

Such a system is knows as an *'open loop'* type, which means it is a worker that sets the control and only additional manual interventions readjust settings. *'Closed loop'* systems guarantee higher performances because flow is automatically controlled non-stop by means of additional electronic devices supported by feedback signals.
Flow can also be controlled with excellent results by means of solenoid valves with a proportional solenoid (see related chapter) that control both direction and flow.

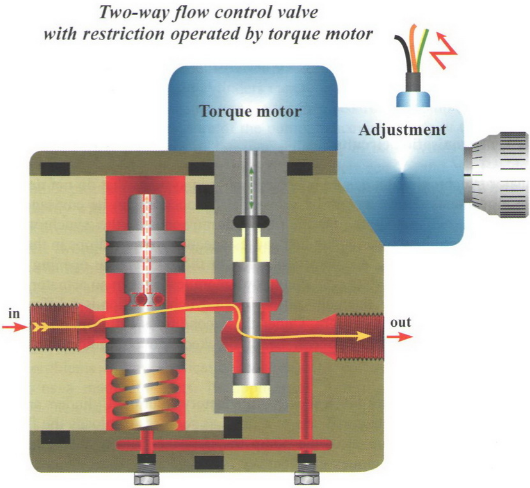



*Figure 11.17*

## SYNCHRONISATION OF TWO OR MORE ACTUATORS

We are now going to consider a key problem for the manufacturers and the repairers of equipment needing the synchronized movement of the actuators. This means actuators must move at the same time and speed for functional reasons and, in most cases, they must also reach the end-of-stoke position simultaneously or separately but at precise intervals.

Some examples of the numerous applications of synchronised actuators are fertiliser spreaders and lifting devices in the mobile industry, as well as press brakes for metal sheet bending in the stationary area. Fertiliser spreaders consist of a trailer or a device directly mounted on tractors that spread granular chemical fertiliser over the field; their distance from the earth depends on the type of fertilisation and workers can replenish their container staying in a comfortable position. Lifting devices are platforms fixed to the floor that can lift loads like cars, large containers for liquids, packs of stone slabs, metal sheets, plastic sheets, grass panes and so on.

The task of synchronised actuators in these machines is evident: if the device, albeit equipped with guides or parallelograms, is lifted or lowered by a single cylinder, its stroke is bound to be blocked because the mass of the large object to lift is not distributed homogeneously. As a result, loads must be lifted uniformly with two cylinders in fertiliser spreaders or press brakes, while lifting devices often need four or more cylinders.

### Simple synchronisation circuits

Actuator synchronisation can be achieved easily in systems that do not demand very precise movements. However, no basic parallel connection can be envisaged, even if there are just two actuators. During no-load movements, the synchronisation of actuators is undermined by anything that causes unequal pressure drops in the actuators, like different-length pipes, a connection piece to the port whose opening is slightly different from the other, a static seal that allows more leakages in an actuator or a worn-out dynamic seal. Anyway, the main cause of poor synchronisation is a difference in the external resistive load on cylinders. Under these conditions lifting platforms not only perform unequal lifting but also risk being blocked between guides as they move obliquely.

The left diagram in Figure 11.18 shows a satisfactory solution. Cylinders are mutually connected in series and actuator B sets in motion as soon as A unloads the fluid. When the directional valve is activated, the pump transfers the fluid to the chamber $V_1$ of cylinder A, which then supplies chamber $V_3$ of cylinder B from $V_2$; chamber $V_4$ is connected to the tank via the directional valve. In the return phase the pump is connected to $V_4$ and the fluid flows from the chamber $V_3$ of B to the chamber $V_2$ of A; $V_1$ is connected to the tank.

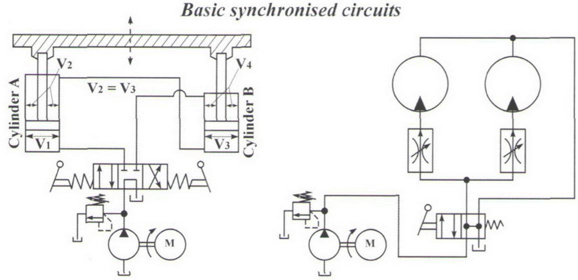



*Figure 11.18*

However, such a circuit demands some special measures. It is equipped with differential cylinders: the larger chamber $V_3$ of B must have the same capacity as the chamber $V_2$ of A, therefore designers have to dimension B carefully; in addition, in order not to undermine the balance between pressures, it must not have a piston with a smaller diameter. Two double rod cylinders ensure equivalence, but these actuators are unsuitable for lifting devices because a space for reverse strokes should be envisaged.

Furthermore, the working pressure of the pump, hence the setting of the relief valve, depends on the differential sum of the two actuators, in other words the differential pressure of cylinder A plus the differential pressure between $V_2$ and $V_3$. Besides this, there is another problem, that is to say fluid leakages between $V_2$ and $V_3$, which can be replenished only with an additional prefill system.

The synchronised movement of two oil hydraulic motors both controlled by a compensated two-or three-way flow control valve (right diagram in Figure 11.18) is even more challenging. The system requirements affect its application also under these circumstances because it is suitable only for low-precision machines.

There are two better solutions: the use of flow dividers, whose success rate equals about 97% under optimum conditions, or the use of closed-loop electrically-controlled proportional valves, whose success rate equals 100%. The use of flow dividers does not raise any problems in any application, whereas the electrohydraulic option is often unfeasible; as a matter of facts, its application in fertiliser spreaders would have disproportionate costs in relation to their needs. On the contrary, it is essential in order to control large lifting devices for fragile materials or objects in precarious balance.

**Flow division**

Flow dividers divide one flow into two or more equal or proportional flows from a single supply channel. The four diagrams in Figure 11.19 show some popular applications of flow dividers; some applications demand a specific flow divider, as we are going to see during the analysis of its different versions. For instance, circuit (d) is compatible only with gear flow dividers with four outlet ports; if the four actuators were cylinders, they would necessarily need phase correction valves (see end of chapter).

(a) *Synchronised stroke of two linear actuators*: two totally uniform flows are essential in lifting devices equipped with two cylinders that must move simultaneously, like loading platforms or scaffolding. The direct connection of the directional valve to the actuators in parallel is very likely to cause different strokes. At the end of the stroke, relief valves, if set properly, remedy the problems that could arise during the normal operation of the flow divider.

(b) *Two actuators supplied by a singe source and subjected to different loads*: it is similar to the previous type but each cylinder has its own directional valve and the flow divider delivers two different flow rates that are mutually proportional.

*Applications of flow dividers*

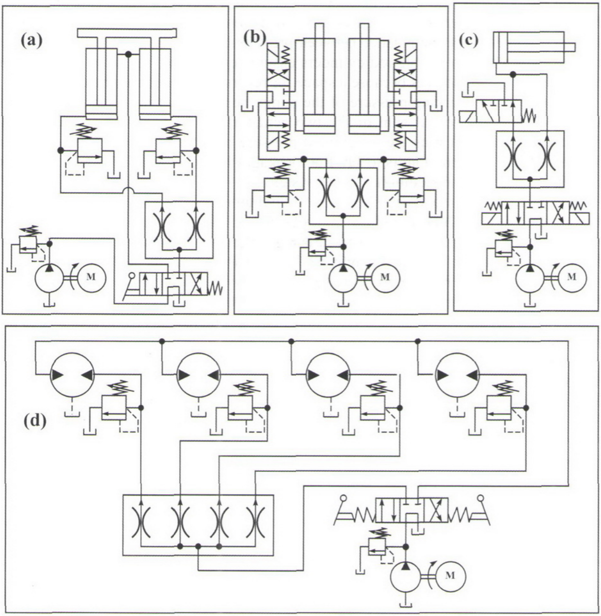



*Figure 11.19*

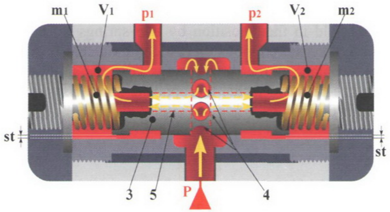
(c) *Pressure increase*: the characteristics of this type are similar to high/low pressure systems. Both outlet pipes are connected to the port of the actuator, but one of them is subjected to the directional valves that operates the connection to the port or connects it to the tank. When the solenoid valve is at rest, the pressure in the cylinder is equivalent to the level set on the relief valve of the pump, apart from pressure drops; when a component of the flow divider is connected to the tank, it becomes a motor that leads to a rise in the pressure of the remaining flow to the cylinder.

(d) *Supply of multiple cylinders*: it obviously needs a flow divider with as many outlet ports as actuators. Flow can be divided equally or proportionally.

It is important to take into consideration the reverse return phase in which flow is combined through the same valve that previously divided it (see remarks on gear flow dividers below). If the flow divider and the control directional valve are suitable for the system, no problems arise: flow is almost linear, with a maximum margin ranging between 3 and 5%. In addition, it is useful to mount flow control valves on the opposite branches for return control, even though relief valves ensure better results.

**Spool flow dividers**

Spool flow dividers provide only two outlet flows from a single hydraulic source. The fluid (Figure 11.20) from the pump flows through port P first and then through the calibrated holes (4) placed in the middle of the spool (3) thus entering the internal channel (5); if flow conditions are perfectly balanced, the spool is retained in a perfectly central position by two springs $m_1$ and $m_2$ that exert equal and opposite forces. The fluid then flows through the restrictions (st) due to the design of the spool and enters ports $p_1$ and $p_2$, which are connected to the actuators.




***Operating principle of a Flow divider***

**p. Inlet port**
**p1, p2. Outlet ports**
**m1, m2. Centring spring**
**V1, V2. Counterbalance chambers**
**3. Spool**
**4. Calibrated holes**
**5. Internal channel**
**st. Restrictions**

*Figure 11.20*

If loads are not balanced (for example, if the pressure developed in chamber $V_1$ is higher than that of chamber $V_2$, resulting in a lower flow rate in $V_1$), the higher pressure in $p_1$ overcomes the opposing spring force, pushing the spool to the right until flow rates are balanced.

This obsolete type of flow divider, also known as 'single-acting' flow divider, is replaced worldwide by a more efficient 'double-acting' version, i.e. a **flow divider with three centring springs**. In single-acting flow dividers, the combination of flow (i.e. the return phase in which two flow rates move through the valve in the opposite direction and are combined in port P, which is now connected to the tank by the directional valve) is not much reliable; on the contrary, flow dividers equipped with three springs deliver excellent results as **flow combiners**.

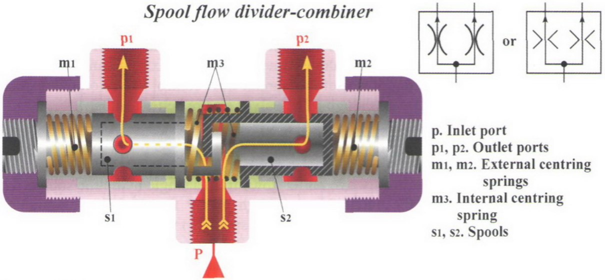



*Figure 11.21*

The operating principle of flow divider-combiners provided with three centring springs (Figure 11.21) is almost the same as that of the previous type of flow divider. They are equipped with two spools held in their position by two springs $m_1$ and $m_2$ pressing on their external ends but, unlike the previous type, a common spring $m_3$ acts on their internal faces. When flow is balanced, the three springs keep the spool in the central position; as flow increases, the pressure inside the opposite spool rises, which makes it move outwards balancing the flow.

**Gear flow dividers**

Gear flow dividers consist of two or more gear motors held in a single housing. All the pairs of gears share the same inlet flow P (Figure 11.22), whereas each pair has its own opposite outlet port ($p_1$, $p_2$, etc.); there are as many outlet ports as motors. Each pair is solidly connected to the others via their drive shafts. A change in the height of teeth of one or more pairs results in different flow rates.

Each pair of gears acts like a motor: the inlet flow acting on the teeth moves gears in opposite directions so that they transfer the fluid to the outlet ports. The volumetric efficiency of motors affects the effectiveness of the system; when in good conditions, devices guarantee accuracy rates of as much as 95 – 97%.

The load cannot be maintained suspended while the spool is in the central position because internal leakages within all flow dividers would add to the leakages of the directional valve. Unlike spool flow dividers, gear flow dividers ensure no satisfactory combination of flow.

For simplicity reasons as well as in order to distinguish them from spool versions, gear flow dividers are represented with the symbol shown in Figure 11.22 in many catalogues. This symbol does not comply with ISO 1219/1, which provides gear flow dividers have the same symbol as spool flow dividers.

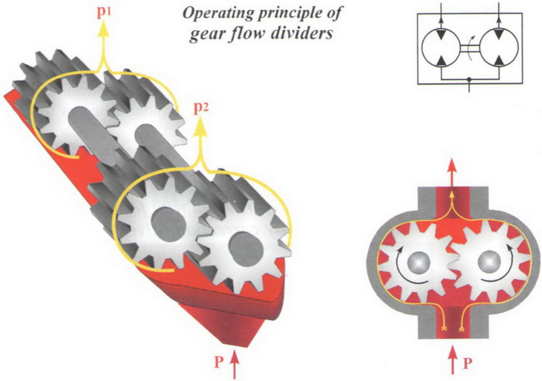



*Figure 11.22*

The type of flow dividers must be chosen depending on the flow rates required. Maximum and minimum speeds for flow dividers are respectively 3500 rpm and 500 rpm, however the optimum range is 1500-2500 rpm. Like any motor, speed (rpm) is related to displacement c and flow Q:

$$\text{rpm} = Q \cdot 1000/c$$

***Internal view of a gear flow divider with three ports***

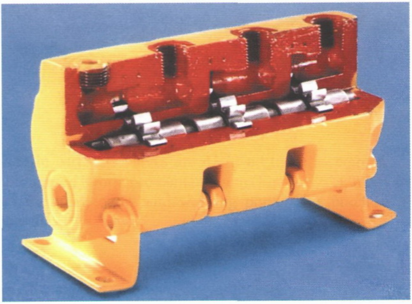


*Figure 11.23*

### Phase correction valves for flow dividers

Despite good performances, sometimes it is not possible to make rods reach the end-of-stroke position simultaneously. Phase correction valves, that is to say a non-return valve and a relief valve adequately assembled according to the actuators, set the zero stroke position thus balancing the movement of one or more actuators that reach the end-of-stroke position after the others.

Correction phase valves are connected independently in spool flow dividers, whereas they are held in a single block in gear versions (Figure 11.24).

***Gear flow divider provided with four ports and phase correction valve***

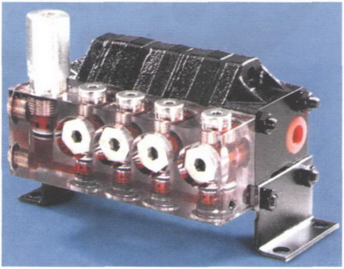


*Figure 11.24*

**Chapter 12**

# *CARTRIDGE VALVES*

Like traditional valves, cartridge valves too satisfy the vast majority of circuit needs; furthermore, they occupy little space and their easy assembly reduces labour costs. There are two types of cartridge valves: screw-in and slip-in (cap) versions. Figure 12.1 shows several cartridge valves: mono and bistable directional valves, pressure or flow control valves, check valves.

***Screw-in cartridge valves***

*Figure 12.1*

## MANIFOLD

Screw-in or slip-in cartridge valves are designed to be assembled in a metal single block properly perforated in order to contain them (Figure 12.2); many internal channels connect them to the inlet, outlet and working ports. This practice is referred to as 'manifold' worldwide.

### Assembly

Figure 12.3 shows a standard assembly: different cartridge valves such as directional valves, auxiliary valves, flow and pressure control valves (in the middle) are placed in a single block (on the left) provided with connexion clearances and valve seats.

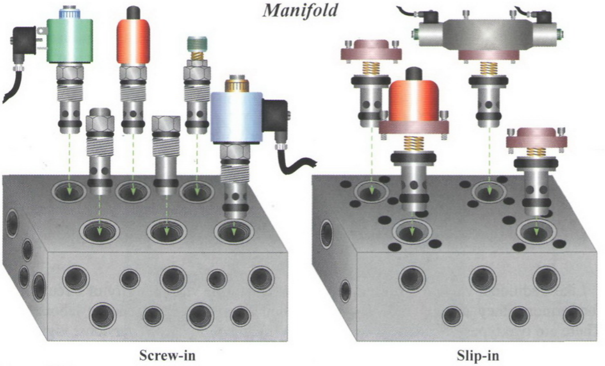



*Figure 12.2*

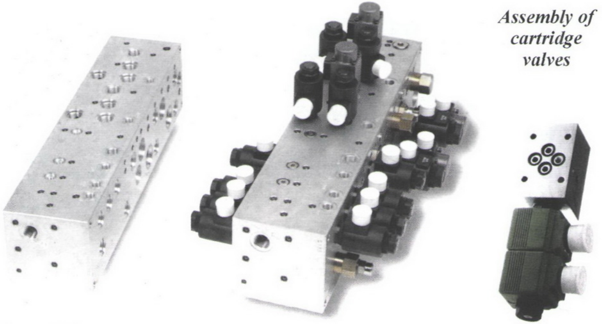
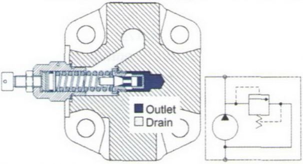



*Figure 12.3*

Cartridge valves are often individually fitted into blocks (Figure 12.3, right lower corner) on a traditional mounting surface according to ISO standards; in these cases components are used as normal single valves or sandwich valves mounted on panels.

Another widespread technique consists in the inclusion of one or more valves in the body of a generator or an actuator. Figure 12.4 shows two examples: the right safety valve controls the maximum pressure the pump can sustain, while the right pump is connected to the tank and equipped with a solenoid.

***Gear pumps equipped with cartridge valves***



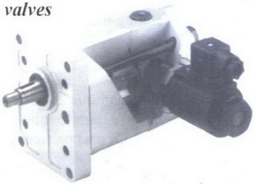


*Figure 12.4*

Another example can be found in Chapter 6 (Additional Components, built-in pilot-operated check valve).

***Single blocks***

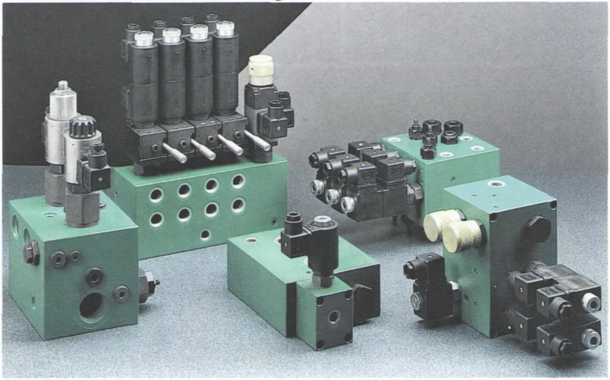


*Figure 12.5*

Manifold cartridge valves with ISO mounting surface or held in single blocks are widely used in mobile and stationary applications; they ensure functional solutions and excellent performances in general presses, snow cats, tractors, sawing machines, shears, lifting jacks, earthmoving equipment, agricultural and forest equipment, construction equipment, plastic processing machines, shoe factories, trucks, street sweepers, iron and steel machine tools, the aerospace and naval industries and so on.

The single block system includes many solutions: simple systems with one or more assembled valves (Figures 12.5, 12.6, 12.7) as well as many groups placed in different zones of the operating machine, according to the specific needs.

**Mono-block structure**

The manifold technique was not very popular in the past because of the manufacturing costs of single blocks. Seat holes required the manual change of many perforation tools and it was particularly hard to calculate the perfect position and length of communicating channels. In addition, the problem of internal pressures further complicated the design phase: in the case of two nearby parallel channels, the pressurised fluid can crack the wall between the two holes thus entering the other channel (the minimum distance between two holes must equal 6-7 mm for aluminium, 5 mm for cast iron and 3-4 mm for steel).

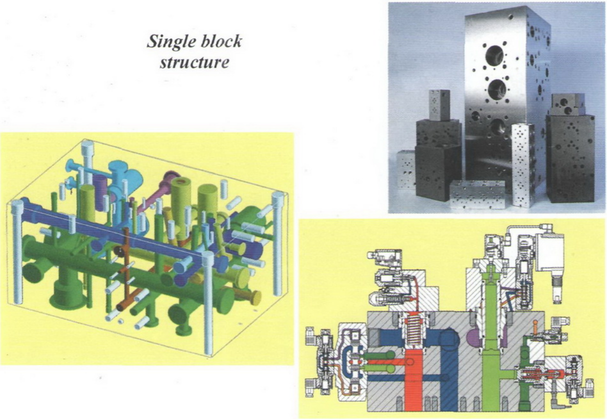



*Figure 12.6*

At present, the design of single blocks, usually made of aluminium or steel, is computer-aided with the help of three-dimensional software: once designers establish the positions of the valves, the ports (P, A, B, T, etc.) and the connection clearances, the program calculates the sizes and the security distance between connection channels and it displays the whole 3D structure so that designers can modify it if necessary.

Figure 12.6 shows a typical 3D view of a computer planning block (left), the internal connections between screw-in cartridge valves and slip-in cartridge valves (lower right corner) and single blocks of different sizes (upper right corner).

The complete program is connected to numerical control machine tools (CAD/CAM) that manufacture cavities, screw threads, clearances, connection ports and screwed holes for the cap fastening in slip-in cartridge valves.

***More manifold combinations***

*Figure 12.7*

**Operational issues**

The great versatility of cartridge valves offers important advantages during not only the design phase but also the assembly process and repairing and maintenance operations, with a considerable costs reduction.

What follows is a list of the most important aspects:

- ✓ Functionality: apart from the easy and handy software employed in the design phase, the manifold is reversible since it is possible to replace a valve with a different type of valve (provided that connections among valves are suitable, otherwise the block is useless).

- ✓ Rationality: manifold elements are held together in one or more groups of blocks that require less space than traditional valves. In most cases, the inclusion of one or more valves in the pump or in the actuator (manufacturers provide them with cartridge valves on demand) facilitates its arrangement since the other valves occupy more space than the body of the generation component or the actuator and their assembly is often very difficult.
- ✓ Fewer external connections are needed, as there are several single block internal clearances; this reduces leakages.
- ✓ Easy maintenance/ repairing: single valves can be removed easily without altering connections and this simplifies and reduces maintenance and repairing times; if the whole manifold has to be overhauled, it is enough to disconnect the external piping and remove locking screws.
- ✓ Comfort: vibrations cause less noise in manifold than in traditional systems; space reduction ensures a more flexible design of a comfortable control cab.

**Insertion cavities**

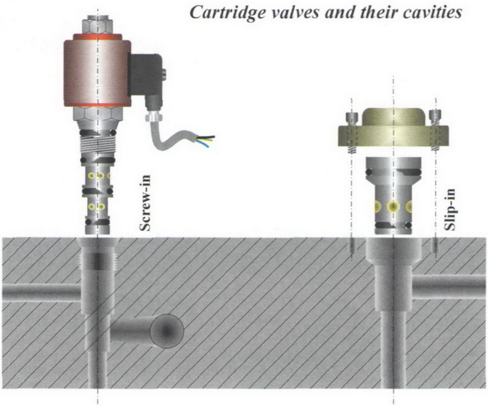



*Figure 12.8*

The term 'cavity' refers to the shape of the hole in the block where the screw-in or slip-in cartridge valve is placed.

These are standard cavities (Figure 12.8) and they differ according to the valve, while holes, connection clearances quantity and position change according to the sizes of the element and they comply with SAE standards and ISO 5783 'Hydraulic Fluid Power - Code for identification of valve mounting surfaces and cartridge valve cavities' and ISO 7368 'Hydraulic Fluid Power - Two-port slip-in cartridge valves – Cavities'.

The following paragraphs provide a general description of cavities for screw-in and slip-in cartridge valves.

## SCREW-IN CARTRIDGE VALVES

Screw-in cartridge valves are essentially made of a body at whose end there is a hexagonal nut solidly connected to the body, a screwed part, many static seal joint rings often coupled with an anti-extrusion seal; a mobile element inside them connects the perpendicular holes of the body to the single block ways.


*Figure 12.9*

Each screw-in valve is directly tightened on the block only through a normal spanner with fixed jaws. The use of washers, additional sealing rings, glue or Teflon is not necessary.

It is advisable to consult manufacturers' catalogues in order to use the right tightening torque so as to avoid damage and cracks. Advice on compatible fluids, viscosity, filtration degrees, minimum and maximum temperatures too should be taken into account.

Most catalogues also list the tools that are necessary to obtain a seat cavity if assemblers want to manufacture single blocks by themselves.

Screw-in cartridge valves serve as pressure and flow control/adjustment valves, directional valves and auxiliary valves according to the mobile element and the internal design. In most cases, their operating principle is similar to the operating principle of traditional valves since the mobile element can be a spool, a ball or a poppet.

Assuming control and adjustment mobile components are spools, spool cartridge valves can have 2, 3 or 4 ways; except for pilot-operated logic valves whose operating principle is the same as cartridge slip-in valves, *ball or poppet* versions have only two or three ways inside the single block.

One-way and two-way restrictor, closed centre directional valves 2/2, 3/2, 4/2, 4/3 or by-pass version with manual, fluid or electric control, relief valves, control valves, flow dividers, simple or pilot-operated check valves and many electronic proportional control valves are easily available among their numerous Italian and international manufactures.

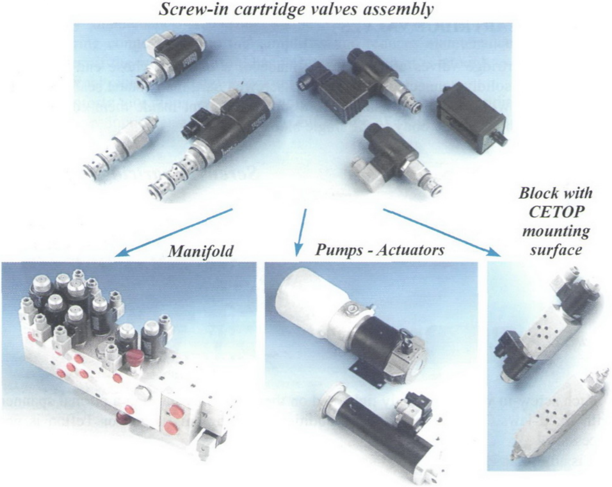



*Figure 12.10*

It is important to highlight that many panel valves available on the market are nothing else than built-in single or multiple cartridge valves.

**Cavities**

Standard cavities or seats, where cartridge valves are tightened, differ in size, number of ways and valve fastening thread; present standards set their size and design. As the cavity design changes only in its dimensioning, which is proportional to the size, Figure 12.11 shows some examples without sizes.

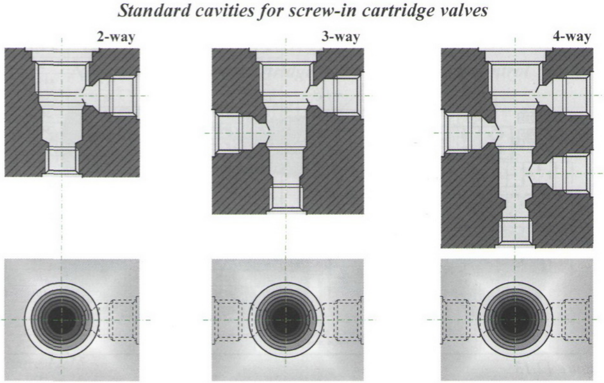



*Figure 12.11*

**Cavities preparation**

Cartridges cavities are manufactured in four stages (Figure 12.12):

1) Cylindrical perforation with a simple spiral drill.
2) Graduated reaming according to the number of ways with a 'notched' tool.
3) Creation of the female thread for cartridge fastening with a tapping tool.
4) Perforation of radial ways.

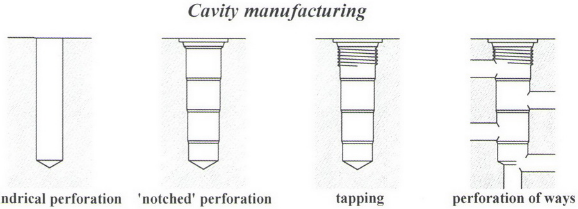
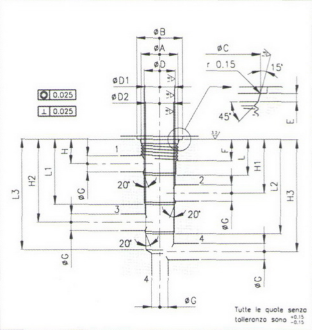



*Figure 12.12*

***Cavity dimensioning (4 ways)***




| | S 20/4 | S 30/4 | S 50/4 | S 70/4 |
|---|---|---|---|---|
| A | M 18x1.5 | M 22x1.5 | M 33x2 | M 42x2 |
| B | 23 | 28 | 40 | 50 |
| C | $19.8^{+0.10}_{-0}$ | $23.8^{+0.10}_{-0}$ | $35.4^{+0.10}_{-0}$ | $44.4^{+0.10}_{-0}$ |
| D | $15^{+0.027}_{-0}$ | $19^{+0.033}_{-0}$ | $28^{+0.033}_{-0}$ | $38^{+0.039}_{-0}$ |
| D1 | $14^{+0.027}_{-0}$ | $18^{+0.027}_{-0}$ | $27^{+0.033}_{-0}$ | $36^{+0.039}_{-0}$ |
| D2 | $13^{+0.027}_{-0}$ | $17^{+0.027}_{-0}$ | $26^{+0.033}_{-0}$ | $34^{+0.039}_{-0}$ |
| E | $2.6^{+0.20}_{-0.20}$ | $2.6^{+0.20}_{-0.20}$ | $3.1^{+0.40}_{-0}$ | $3.3^{+0.20}_{-0}$ |
| F | 11 | 13 | 16 | 19 |
| G | 8 | 10.5 | 15 | 20 |
| H | 13 | 15 | 21 | 26 |
| H1 | 28 | 33 | 46 | 57 |
| H2 | 43 | 51 | 71 | 88 |
| H3 | 58 | 69 | 96 | 119 |
| L | 18.5 | 22 | 32 | 39 |
| L1 | 33.5 | 40 | 56 | 70 |
| L2 | 48.5 | 58 | 80 | 101 |
| L3 | 57 | 68 | 90 | 114 |

*Figure 12.13*

**Auxiliary directional seal valves**

Screw-in auxiliary directional seal valves have a ball or poppet that reduces internal leakages considerably. Balls are inexpensive and particularly suitable for intermittent operation (balls tend to vibrate) while poppets sustain high pressures and forces.

The simplest type of such a valve is the **non-return valve** (Figure 12.14): a poppet or a ball allows the in-to-out flow, whereas the opposite direction is blocked (for further details see Chapter 9, Non-Return Valves).

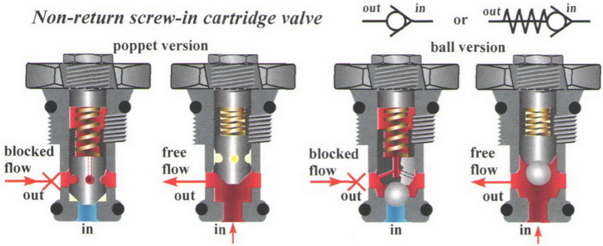



*Figure 12.14*

**Non-return cartridges with manual control**, which is similar to the previous ones, can be operated via a button (monostable version in Figure 12.15) or another device (lever, pedals…).

### *Non-return screw-in cartridge valve with manual control*

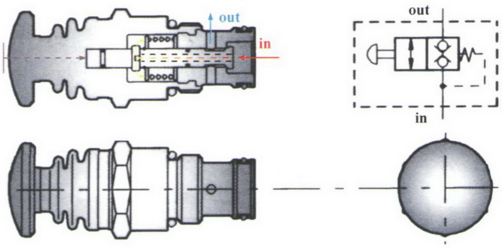



*Figure 12.15*

**Pilot-operated check valve** are similar to the previous ones but they are equipped with a little piston that is supplied by the third lower way (Figure 12.16).

### *Non-return screw-in cartridge valve with control*

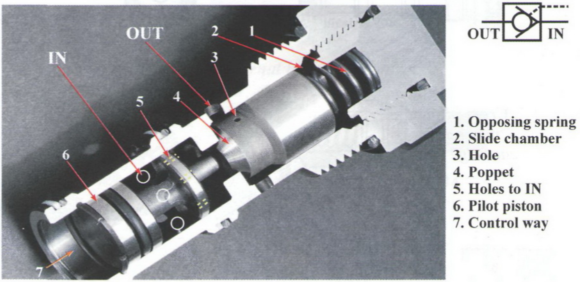



*Figure 12.16*

It is important to consider the value of the pilot pressure $p_{pil}$ according to $p_{out}$ (pressure over the out port, whose section is $S_2$) and the possible $p_{in}$ (pressure over the in port, whose section is $S_1$). Pilot pressure can be determined by means of the following formula:

$$p_{pil} = p_{in} \cdot S_1/S_2$$

Sometimes it is necessary to overcome considerable back pressures acting on the in port. Unlike in traditional check decompression valves with hydraulic control (Chapter 9), a certain amount of fluid is drained in cartridge valves. Consequently, the pilot pressure is kept at a normal level; the valve ensures sealing because the **drain** is separately connected to the tank through the upper way (Figure 12.17).

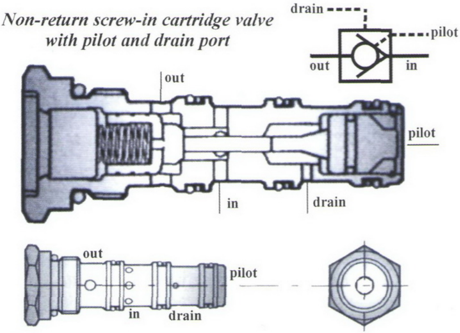



*Figure 12.17*

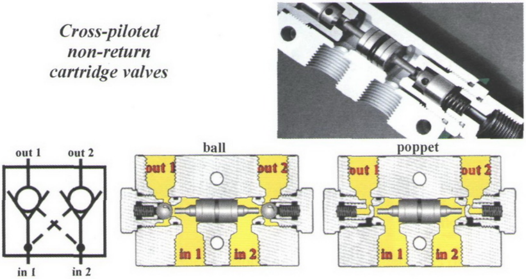



*Figure 12.18*

The inclusion of two pilot-operated check cartridges in a parallelepiped results in **cross-piloted non-return valves** (for more details see Chapter 9). Anyway, it is necessary to bring together the end faces of the two pilot pistons that in this case receive the pilot fluid on opposite surfaces (Figure 12.18).

**Shuttle valves** (Figure 12.19) can receive the signal simultaneously or individually from ports 1 and 3 and transmit it to port 2 (see Chapter 9).

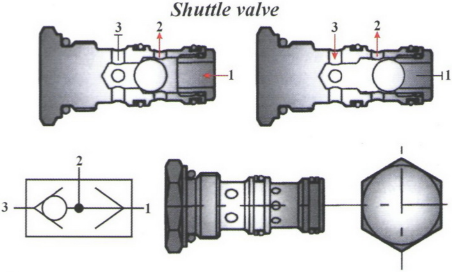



*Figure 12.19*

## Spool directional valves with hydraulic control

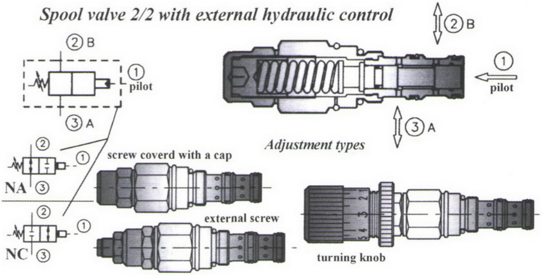



*Figure 12.20*

Directional screw-in cartridges, controlled by an **external** hydraulic input, are generally available in **2/2** (Figure 12.20) and **3/2** versions (Figure 12.21). Some manufacturers produce even four-way versions.

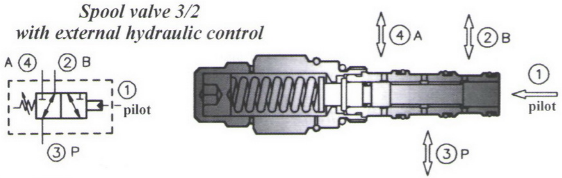



*Figure 12.21*

In **self-controlled** spool cartridges **3/2**, when the inlet pressure increases (port 1), the spool connects port 1 to port 2 (Figure 12.22).

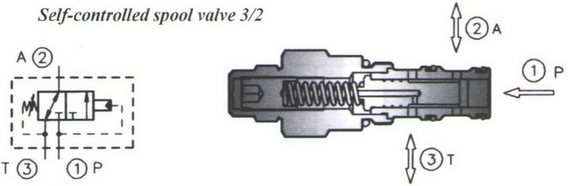



*Figure 12.22*

### Shuttle valve for closed circuits

**Self-controlled** shuttle cartridges **3/3** are usually used in the closed circuits of hoists and hydrostatic transmissions; it replaces shuttle valves in these systems since these are not suitable for considerable flows. Self-controlled shuttle cartridges 3/3, connected to a relief valve, allow the fluid to replenish in the typical closed circuits for engines; they limit pressure in the return line and ensure that the fluid has an appropriate temperature. As they are similar shuttle valve, they are usually referred to as simply '**shuttle valves**' or '**flushing valves**', to be more precise '**hot oil shuttle valves**'.

Cartridges simply send the fluid to the relief valve set to system needs. What follows is a description of their operating principle; further details can be found in paragraphs concerning closed circuits. The spring, that has a force of few daN, plays a double role: traction and compression. It maintains the spool in a central position and prevents the connection between $p_1$ or $p_2$ and the T port that is connected to the relief valve. Pilot pistons correspond to the external faces of the spool; pilot port $X_{p1}$ and pressure port $p_2$

are directly connected to the cavity $X_{p2}$ whereas the fluid flows from $p_2$ to the related pilot surface through $X_{p2}$.

A closed circuit is made of a branch that connects the pump delivery pipe to the motor inlet port and another branch conning the outlet port of the motor to the suction port of the pump; when the system works at full capacity, the pressure in the return branch is high but much less than the delivery pressure. In the reverse direction (variable displacement pumps with flow direction reversal) of the actuator, the functions of branches change: the one that was connected to the delivery and the inlet ports of the motor now plays the return role and vice versa.

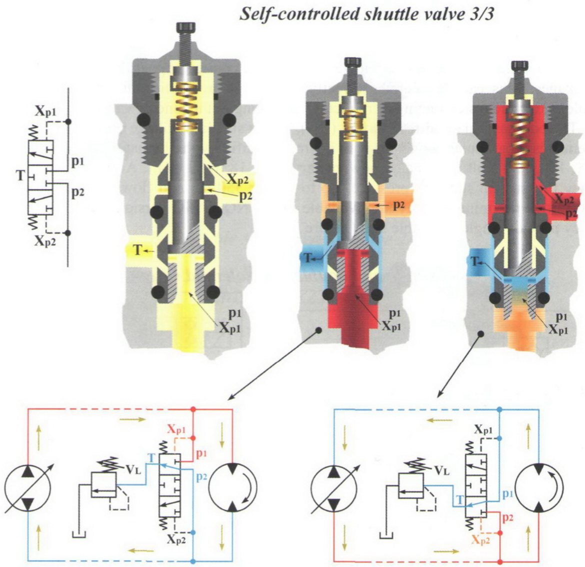



*Figure 12.23*

A specific pressure acts on both control faces of the shuttle cartridge when the system works at full capacity, but the spool is subjected to delivery pressure, which is always higher than the other pressure.

Since overheated oil in closed circuit systems must be conveyed to the tank through the 'outlet-suction' secondary branch (an auxiliary pump replenishes it through another device), the control that presses on the spool sends the opposite fluid to the relief valve with the minimum working pressure in the delivery branch.

For example, assume that the fluid flows clockwise (Figure 12.23); the high pressure $p_1$ overcomes the force of the spring and acts on the spool flat side that connects $p_2$ (low pressure) to the relief valve $V_L$. In the reverse direction, the fluid flows anticlockwise: the pilot port $X_{p2}$ of $p_2$, now at a high pressure, pushes the spool downward, conveying the fluid from $p_2$ to $V_L$.

**Solenoid seal valves**

The mobile element in solenoid seal valves is a conical poppet that prevents internal leakages. This can be operated directly or through hydraulic internal control by the solenoid, which is extracting or retracting according to the kind of valve.

The coil has usually direct current (DC), with a standard voltage of 12, 24, 48 Volt; it demands a particular connector with a circuit rectifier with alternating current (AC) 24, 48, 110, 220 Volt. Dissipated powers vary from 18 to 33 Watt. According to the size, they can sustain standard pressures up to 210 or 350 bar and flows ranging from 1 to 200 litres per minute. They are available in 2/2 NO or NC versions and 3/2 versions; they provide single or double seal (Figure 12.24).

***Single and double seal solenoid valves***

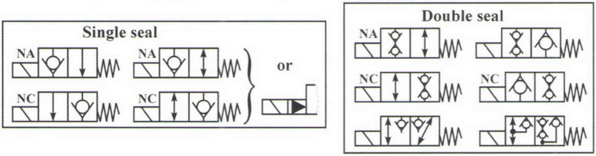



*Figure 12.24*

**NC** solenoid **single seal** poppet valves **2/2** with internal hydraulic control (Figure 12.25) are made up of a main poppet, which contains a pilot pin solidly connected to mobile core.

The poppet is kept closed by the pressurised fluid that acts inside it; with the excitation of the solenoid, the pilot pin lifts and the poppet connects the in/out ports. The solenoid needs oil-bath mechanical parts in order to lubricate them and cool them down.

In the rest position (NC), fluid can flow from 'out' to 'in'; when the valve is on, there is free flow in both directions.

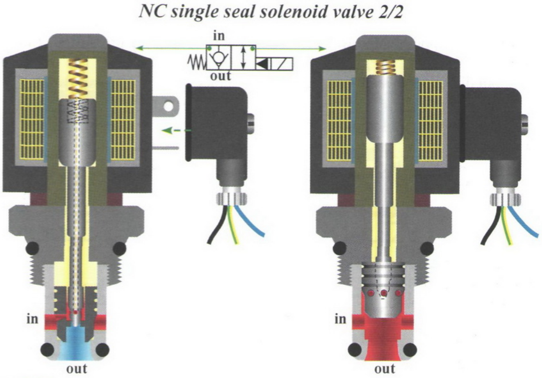



*Figure 12.25*

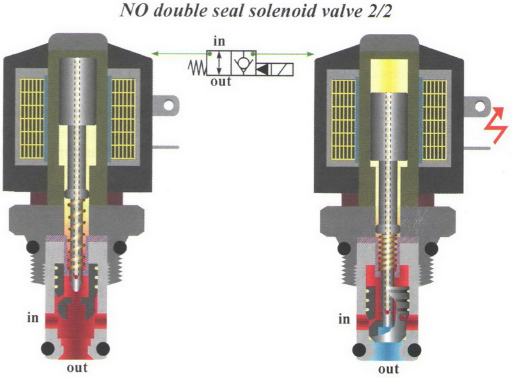



*Figure 12.26*

The excitation system in **NO** solenoid **single seal** spool valves **2/2** (Figure 12.26) with internal hydraulic control is the same as the previous one; in this case it is necessary to keep the pilot pin in a retracted position with the help of the spring, in order to push it downward upon switching. In this way the pilot pin closes the central hole of the main poppet, which was previously open, so that the pressurised fluid on the inlet port pushes against the poppet and blocks the communication. The out-to-in flow is free in both directions also under these circumstances.

Solenoid single seal valves with an always free out-to-in flow can respond to principles other than those described so far.

*2/2 NC and NO single seal solenoid valves*

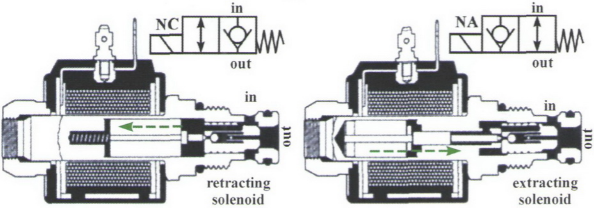



*Figure 12.27*

When NC versions are in the rest positions (Figure 12.27), the flow from 'out' presses on the poppet, overcomes the weak force of the spring (which is over the solenoid) and reaches the other port.

The flow from 'in' enters the tiny hole (in order not to change the poppet position and to lubricate the solenoid); it overcomes the weak power of the spring and moves the pilot pin, but the movement of the ball inside the poppet prevents it from connecting to the 'out' port. With the excitement of the solenoid that performs retraction (which means it moves away from the cartridge), the poppet now connects the inlet port to the outlet port and there is free flow in both directions.

The NO version works in a similar manner and their solenoid performs extraction. In order to avoid that the fluid flowing through the side hole of the poppet presses on the flat side, a second seat where the pilot pin can slide is needed; as a result, the surface of the spool is covered and it cannot be affected by the pressurised fluid.

**Solenoid single seal** valves **2/2** whose flow is free only in one direction according to positions share the same operating principle but they have no internal ball. (Figure 12.28). On the contrary, **solenoid double seal** spool valves **2/2** allow flow where in one or the other direction only in one position. In the rest position, **2/2 NO** versions allow in-to-out flow and vice versa; when they are excited, they block flow in both directions. (Figure12.29)

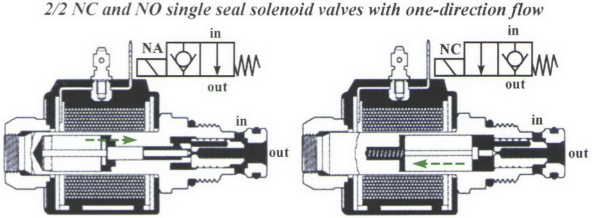



*Figure 12.28*

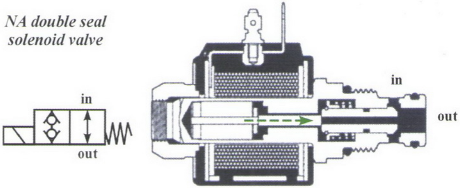



*Figure 12.29*

**NC** solenoid **double seal** valves **2/2** work the opposite manner: the flow in the rest position is blocked in both directions. When the valve is excited, it allows in-to-out or out-to-in flow (Figure 12.30).

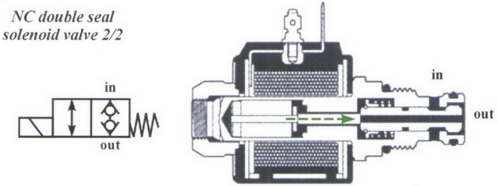



*Figure 12.30*

3/2 versions are obviously arranged in 3-way cavities (Figure 12.31). Their poppet is similar to that of the versions above.

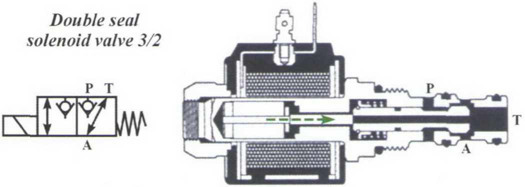
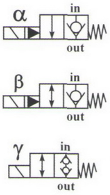



*Figure 12.31*

Like in traditional solenoid valves, if the flow rises, internal pressure drops increase as well (see Chapter 8 – Distinguishing Traits). Figure 12.32 shows the general charts about components with a maximum flow of 50 l/min (environmental temperature equals 20 °C, oil viscosity and internal temperature are respectively 26 cSt and 50 °C).

***Solenoid seal valves - Pressure drops***

***Maximum flow = 50 l/min - Generic data***




| | | α | β | γ |
|---|---|---|---|---|
| not excited | in→out | | | |
| | out→in | B | B | |
| excited | in→out | C | B | A |
| | out→in | | B | A |

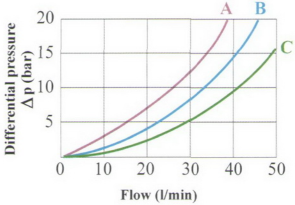



*Figure 12.32*

**Solenoid spool valves**

The operating principle of solenoid spool valves is much similar to the operating principle of panel valves. As far as solenoids are concerned, the remarks on solenoid seal valves apply also to solenoid spool valves. They can sustain pressures ranging from 210 to 350 bar, but unlike solenoid seal valves, there are many different versions of this kind of valve that can sustain flows from 8 to 60 l/min.

2/2, 3/2, and 4/2 versions are usually monostable versions (one solenoid); 3/3 and 4/3 versions are controlled by a double solenoid.

**2/2 NC solenoid spool valves** have the same design as solenoid seal valves except for their mobile element (Figure 12.33).

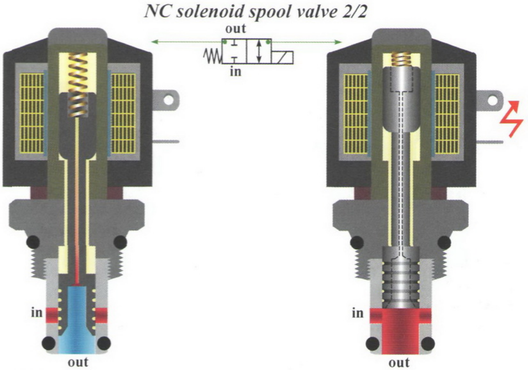



*Figure 12.33*

**NO** solenoid valves **2/2** results from a spool having a different shape (Figure 12.34)

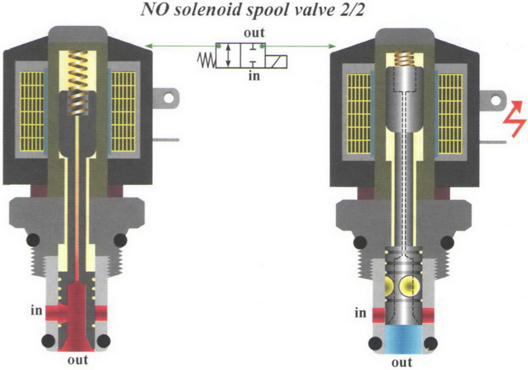



*Figure 12.34*

**3/2** screw-in solenoid **spool** valves are essentially used for the on/off control of single-acting cylinders (Figure 12.35).

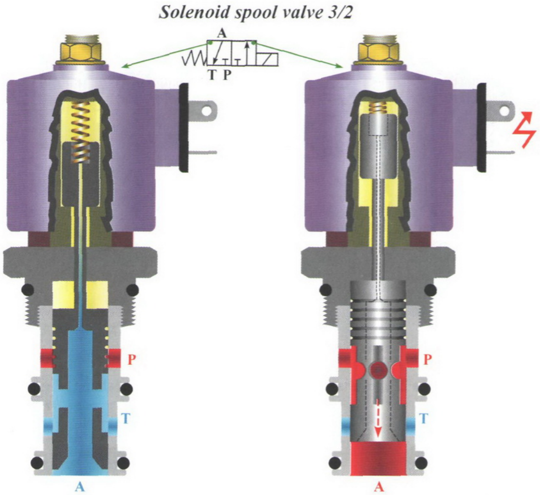



*Figure 12.35*

Specific applications require 3-way valves with three positions (double solenoid), but they are quite rare (Figure 12.36).

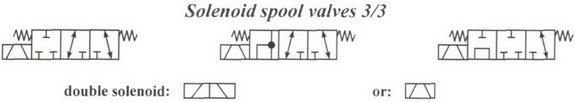



*Figure 12.36*

Screw-in solenoid spool valve 4/2 (Figure 12.37) can be easily found in the monostable version. In order to keep the coil excited, the systems demands an electric circuit with relays, start and stop buttons or (only under specific circumstances provided by standards) a simple bistable switch.

See Chapter 8 for information on the problems about spool lap conditions, insulation class, insertion, protection degree, etc.

***Solenoid spool valve 4/2***

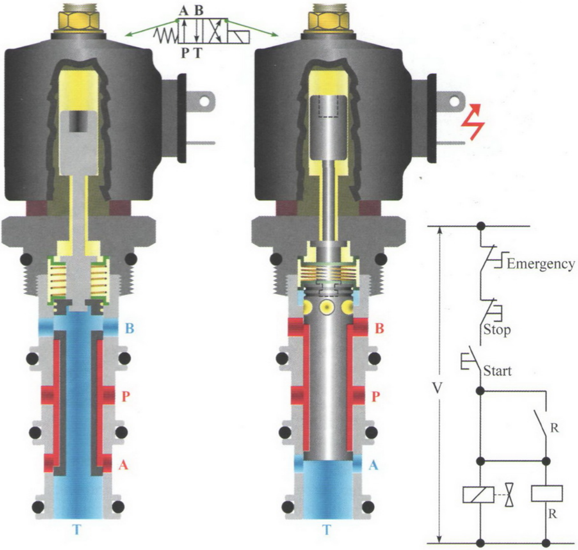



*Figure 12.37*

Other cartridges 4/2 available on the market are shown in Figure 12.38.

***Solenoid spool valves 4/2***

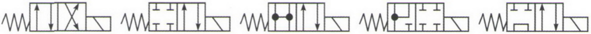


*Figure 12.38*

Spring-centred screw-in solenoid **spool** cartridge valves **4/3** (Figure 12.39) have two solenoids.

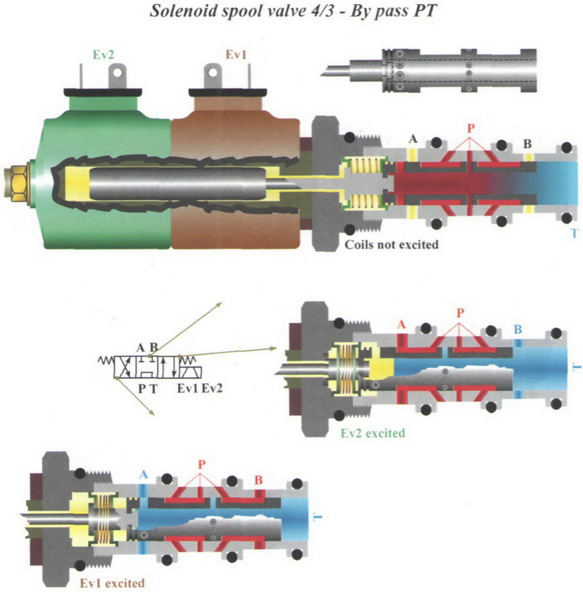



*Figure 12.39*

Figure 12.40 shows the most common spool cartridge valves 4/3.

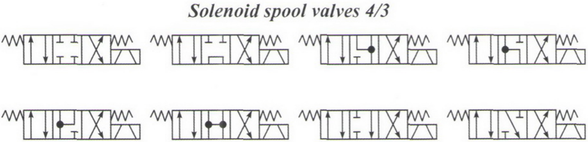



*Figure 12.40*

***Cut-away view of a closed centre cartridge valve 4/3***

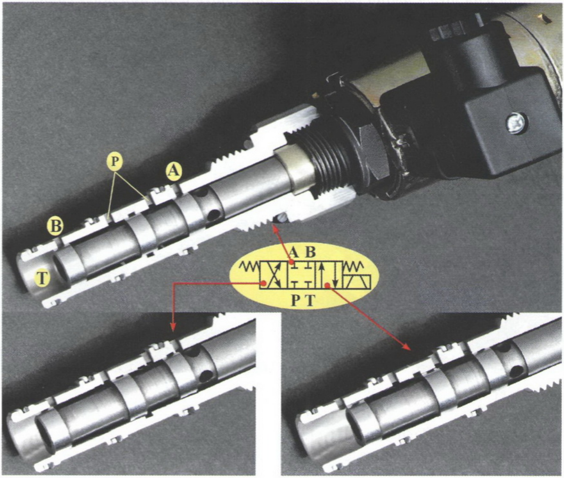



*Figure 12.41*

**Relief and pressure control valves**

Direct-acting or pilot-operated relief and pressure control valves, whether screw-in or cartridge versions, are based on the same operating principle as panel valves (see Chapter 10 for further details). Figure 12.42 shows a direct-acting relief valve with cushioning piston.

***Direct-acting pressure control cartridge valve***

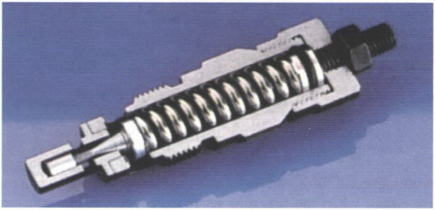


*Figure 12.42*

***Manifold with direct-acting relief valve, non-return valve and other components***

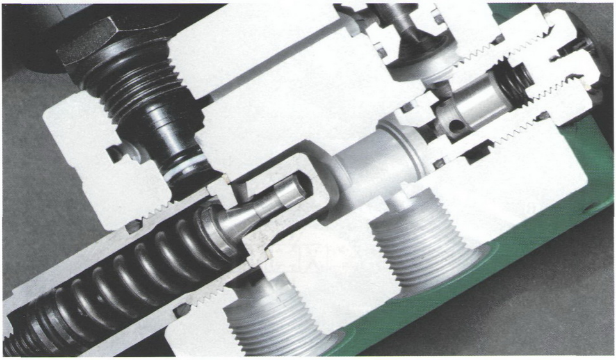


*Figure 12.43*

All types of pilot-operated cartridge relief valves have the same operating principle as panel valves. Their control stage and main stage are axial (Figure 12.44).

***Pilot-operated relief valve***

**1. Adjustment screw**
**2. Pilot spring**
**3. Pilot poppet**
**4. Positioning spring**
**5. Pilot outlet port**
**6. Piston**

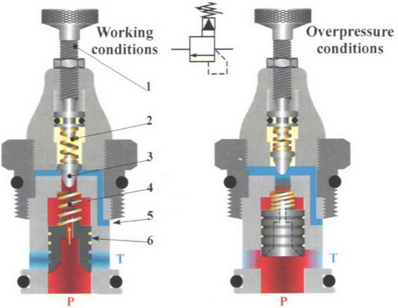



*Figure 12.44*

Under working conditions, the piston (6) is balanced as downstream and upstream pressure are equal; when the pressure set is exceeded (spring adjustment 2), the pilot poppet (3) opens up and allows the fluid to flow to the tank through the clearance (5). As upstream pressure is considerably reduced, the piston is no more balanced (6): the pressure on port P now pushes it upward. For this reason, port T opens up and the fluid is conveyed into the tank, which results in a pressure drop. When normal conditions are restored, the spring (4) brings the piston (6) back to the initial position and the piston then closes the tank port T.

**Direct-acting unloading relief** cartridge valves are usually used in double pump low/high pressure circuits and they have an external pilot (Figure 12.45).

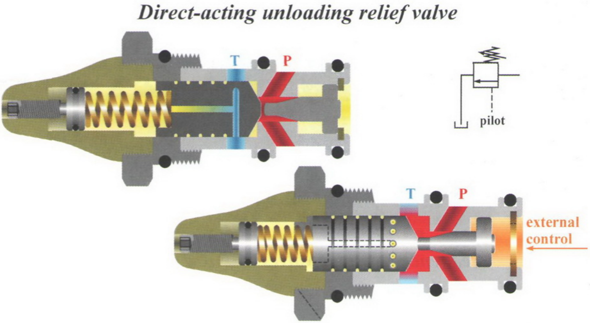



*Figure 12.45*

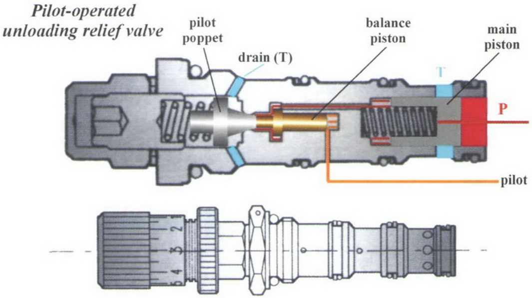



*Figure 12.46*

Systems provided with an accumulator need **pilot-operated unloading relief valves**.
When the pressure set is reached (full accumulator), the valve connects the pump to the tank and the accumulator distributes flow to users (Chapter 10).

As it has already been said in respect of Figure 10.11, a non-return valve needs to be added to the utilities with accumulator to the relief valve connected to the tank of Figure 12.46. Like panel versions, cartridge versions holding both components in a single block are available (Figure 12.47).

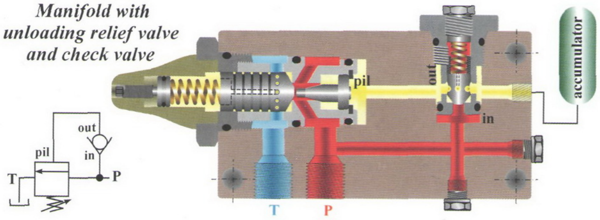



*Figure 12.47*

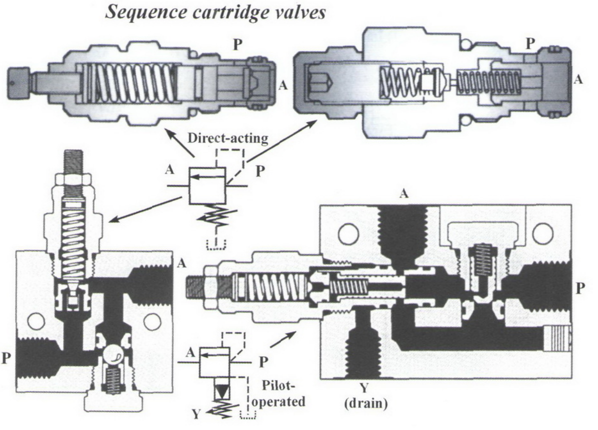



*Figure 12.48*

Cartridge **sequence** valves are available in direct-acting and pilot-operated versions. These valves do not differ from the relief valves previously described, apart from their outlet connection that is not connected to the tank but to the user in this case. Figure 12.48 shows some sequence valves available on the market.

**Anti-shock** blocks are made up of two opposite relief valves (Figure 12.49) that prevent overpressures due to external recoil.

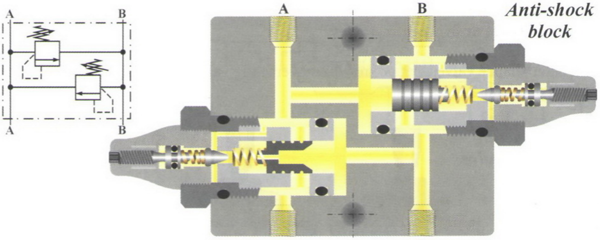



*Figure 12.49*

***Anti-shock and anti-cavitation block***

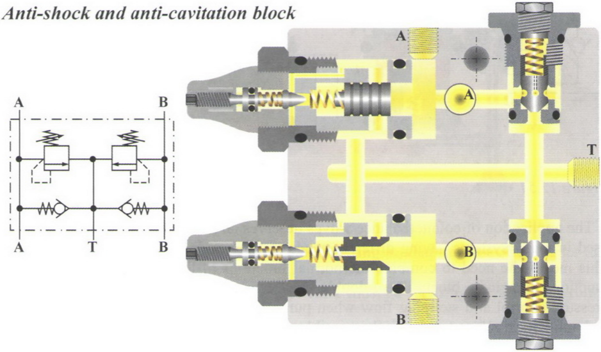



*Figure 12.50*

If the two directions have a different pressure/flow, like in differential cylinders (different piston area), the anti-shock blocks are no longer reliable and it is advisable to employ **anti-shock anti-cavitation** blocks that are equipped with two relief valves and two non-return valves. (Figure 12.50)

In abnormal situations, the vacuum that can develop inside the cylinder or the hydraulic motor leads to cavitation. One-way valves ensure the suction of the fluid from the tank; this compensates the missing volume. The pipe connected to the tank must have an adequate cross-section and be placed in the tank under the level of the fluid.

### Counterbalance or overcentre valves

Counterbalance valves, better known as overcentre valves, are designed to control the free fall of a load.

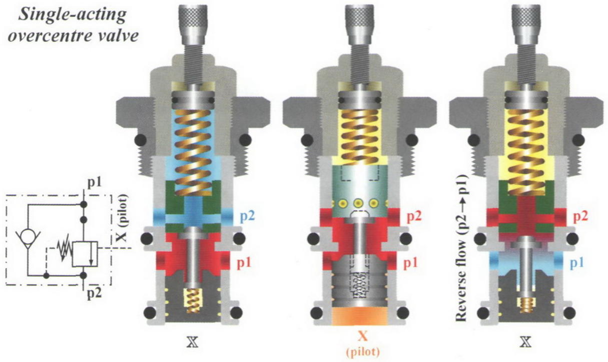



*Figure 12.51*

The application of counterbalance valves deserves an in-depth analysis since they are used in many systems having different needs and demanding suitable compact valves. This means the need to combine the normal functions of the valve with other special functions in a single block, like a double valve working in both directions, performing pressure control and stopping flow when pump is not working, for the security brake and anti-shock valves to sustain the possible recoils due to obstacles hitting the moving load.

Valves must not only sustain the flow/pressure required, but also be small-sized as they have to be arranged in small spaces; they also must allow easy manual adjustment.

The modular system with traditional external panel valves is often unsuitable because the inclusion of many valves occupies too much space especially on the actuators of mobile and stationary machines. In this case, the ideal solution is a little manifold with one or more cartridges near the port aligned with the actuator or even placed on the actuator.

***Single-acting compact overcentre valve with non-return valve***

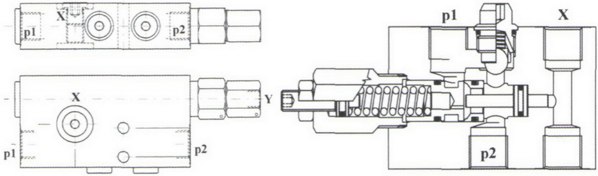



*Figure 12.52*

**Double-acting counterbalance valves**

The combination of two single-acting opposite and cross-piloted valves in a single (Figures 12.51 and 12.52) results in a double-acting counterbalance valve that controls the movement of a load in both directions (Figure 12.53).

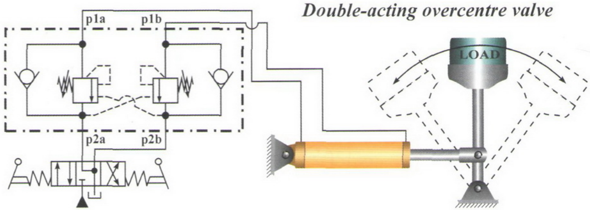



*Figure 12.53*

Every type of control valves needs an initial **setting** $p_{tv}$ that must be 30% higher than the maximum working pressure $p_{max}$.

That is the reason why during the control valve must be set using the following rule:

$$p_{tv} = p_{max} \cdot 1.3$$

With a system working at a maximum pressure of 150 bar, the valve must be set at ($150 \cdot 1.3 =$) 195 bar.

***Cut-away view of a double-acting overcentre valve***

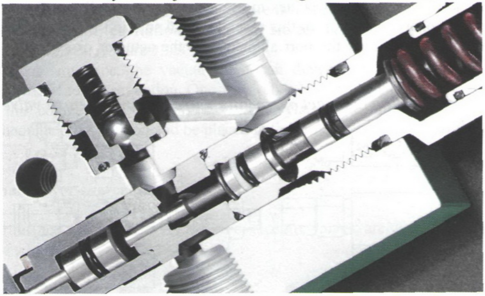


*Figure 12.54*

The pressures that develop inside the valve affect its functioning. The valve spool is conditioned by the pressure that acts on the pilot piston and by the internal pressure $p_{load}$ developed by the free fall load that acts on the surface of the poppet/ piston of the spool itself.

The relation of the pressure set $p_{tv}$, the pressure developed by the load $p_{load}$ and the **pilot ratio** $R_{pil}$, in other words the ratio of the effective area of the control piston to the effective area of the piston on the spool (Figure 12.55), defines the pilot pressure $p_{pilot}$ needed:

$$p_{pilot} = \frac{p_{tv} - p_{load}}{R_{pil}}$$

***Pilot ratio***

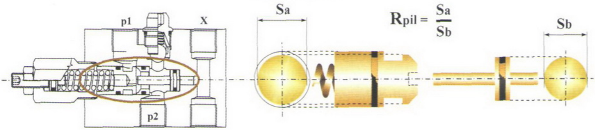



*Figure 12.55*

For example, assume the pressure set $p_{tv}$ equals 300 bar, the pressure produced by the load is 240 bar and the ratio of pilot pistons to the spool equals 4:1, pilot pressure is:

$$P_{pilot} = \frac{300 - 240}{4} = 15 \text{ bar}$$

The pilot ratio influences the functioning of the system. Another ratio (for example 8:1) that refers to a weak pilot pressure ensures energy conservation and high-speed movements; a low ratio (for example 4:1) has a much higher pilot pressure and it is recommended for unstable systems as they enable a high-precision control.

If the distributor is a remote control version, the connection pipe and the fittings to the overcentre valve must be calculated with extreme precision in order to avoid back pressures. Back pressures can undermine valve functioning because they act on the poppet and the pilot piston in the reverse direction. Counterbalance valves with **back pressure compensation** are used instead if the pipe and fittings cannot be determined with precision or if back pressures cannot be avoided for some reason.

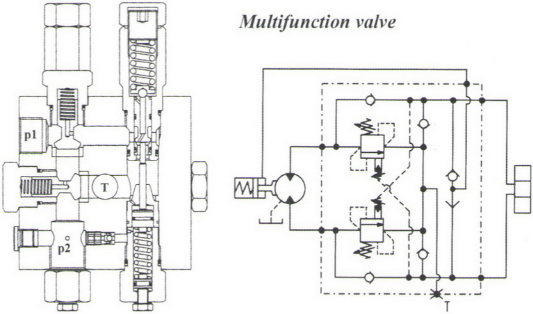



*Figure 12.56*

**Multifunction** valves are suitable for complex systems; they combine double-acting overcentre, anti-shock, anti-cavitation valves and the regeneration system. Figure 12.56 shows a manifold with an additional security block that ensures the perfect stability of the suspended load when the machine is not working. These units often fall into the category known as '**Motion Control**'.

***Single- and double-acting overcentre valves***

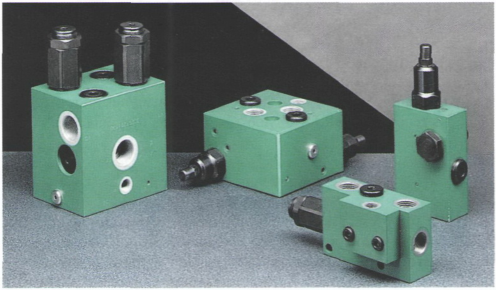
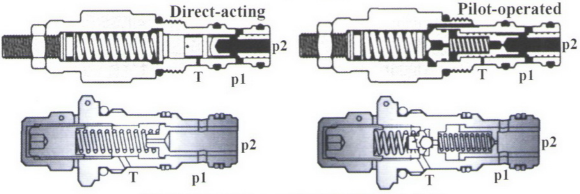


*Figure 12.57*

**Pressure-reducing valves**

Chapter 10 explores the differences between relief valves and pressure-reducing valves and the operational features of the latter. We are now going to analyse screw-in cartridge versions only; it is important to underline that the constant drain connected to the tank through port T is fundamental in pressure-reducing valves. The quantity of drained fluid ranges from 0.4 to 0.6 l/min in the vast majority of these valves.

***Pressure-reducing cartridge valves***
***(Simple type)***




**N.B. Port T which is connected to the tank, is the drain**

*Figure 12.58*

There are two types of pressure-reducing cartridge: **simple** type or with **pressure relief** in case of overload. The latter releases some flow through port T if the external system triggers overpressure without stopping the circuit. Figure 12.58 shows two different types of simple direct-acting and pilot-operated pressure-reducing valves.

In **direct-acting** pressure-reducing cartridge valves (Figure 12.59), when p1 and p2 are equal, the spring keeps the spool in a low position. as pressure rises in the primary branch (p1>p2), the higher force of the fluid counters the spring, lifts the spool and sends part of the liquid to the drain (T), without affecting the pressure in the secondary branch.

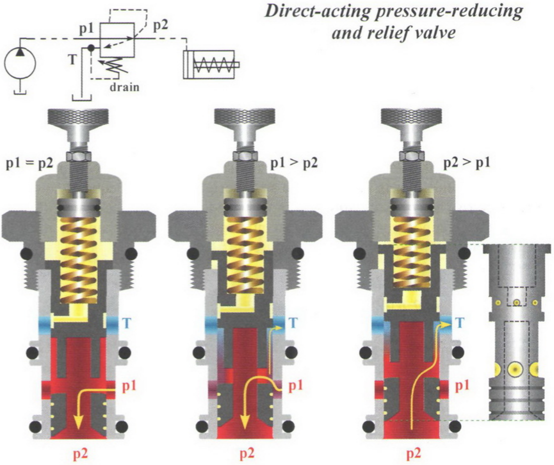



*Figure 12.59*

If overpressure develops in the secondary branch (p2>p1), the valve acts as a relief valve: the push of the fluid p2 on the spool opens port T (the symbol used in the Figure is not provided for by ISO standards).

The operating principle of **pilot-operated** pressure-reducing and relief valves is similar to that of the previous type but they controlled by the pilot stage (Figure 12.60).

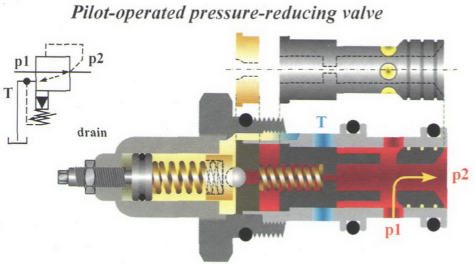



*Figure 12.60*

**Flow control**

Like the vast majority of cartridge valves, flow control valves works almost like traditional valves; the difficulty lies in obtaining the same results even if spaces inside the cartridge are considerably reduced. It is advisable to take into account flow control problems and applications (Chapter 11) so as to design systems adequately.

**Adjustable two-way restrictors**, commonly needle type, are the easiest flow control valves but they cannot guarantee a constant flow if pressure changes; they ensures excellent sealing in the closed position (Figure 12.61).

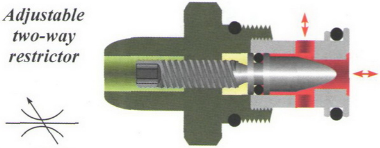



*Figure 12.61*

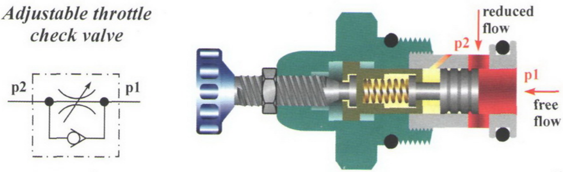



*Figure 12.62*

Apart from needle valves, in which the non-return poppet/spring is placed inside, adjustable **throttle check valves** can have a spool (Figure 12.62); however, they do not ensure good sealing in the closed position.

Unlike the previous type, **compensated flow control** valves guarantee a constant flow (and movement), despite changes in the downstream load. This is not feasible in throttle check valves because the change of external forces results in a chance in the differential pressure over the throttle and this means the actuator is out of control.

Differential pressure compensation keeps the flow constant and it also ensures a constant transfer or rotation of the actuator. The balance is obtained through the opposing force of the spring and the thrust of the pressurised fluid on the internal surface of the spool. Figure 12.63 shows the most popular versions available on the market.

*Compensated flow control cartridge valves*

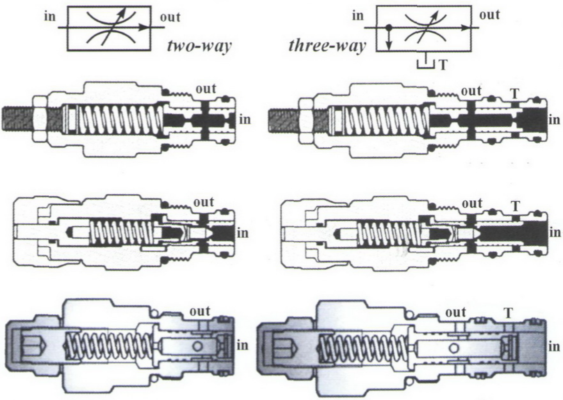



*Figure 12.63*

The relief valve conveys the excess fluid to the tank in **two-way compensated** flow control cartridges; in **three-way compensated** valves, the excess fluid is directly transferred into the tank or transferred to another actuator.

The advantages of three-way versions are clear as the safety valve is not continuously subjected to the outlet force, provided the other parts of the systems are adequately dimensioned.

***Flow divider/combiner cartridge valve***

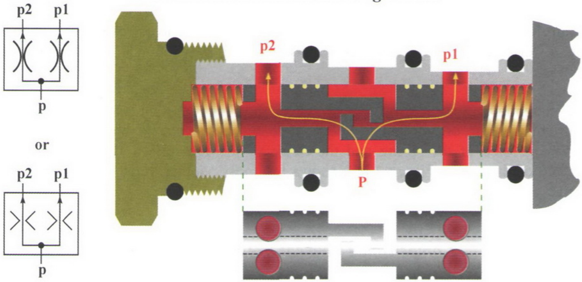



*Figure 12.64*

**Flow divider/combiners** (Chapter 11) divide one flow into two equal or proportional flows (Figure 12.64). They must be placed in a four-way cavity in cartridge versions, with the outer cavity that is opposite the fastening thread and closed by the spring. Cartridge flow dividers deliver different flows (from 6 to 120 l/min) and they generally work at a maximum pressure of 210 bar.

***Manifold flow control valves***

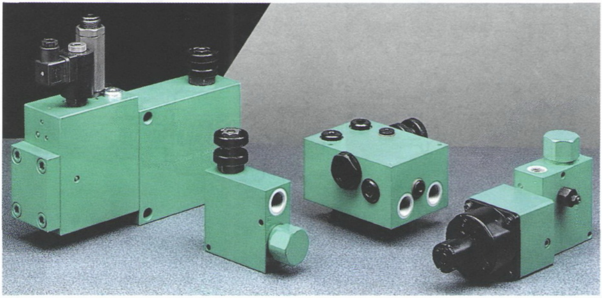


*Figure 12.65*

### Other screw-in cartridges

Screw-in cartridge **logic** valves share the same operating principle as slip-in cartridge valves (see next paragraph). They are available for direction, pressure and flow control, but their use is strictly related to the use of a second pilot valve.

Relief/pressure control and flow control valves with **electronic proportional control** are very important. All types of valve share the same principles (see the chapter on electroproportional control); we are going to mention only the screw-in cartridges normally used in manifold oil hydraulic systems.

Proportional valves for pressure and flow control/adjustment are made up of a cartridge control valve (the same as similar valves in most cases) and a direct current solenoid of 12 or 24 Volt that is controlled by the electronic unit. This unit can be directly connected to the solenoid and contain the control potentiometers; otherwise, it can be placed somewhere else and be connected to the coil through an electric cable (Figure 12.66). Both versions are equipped with output clamps for the connection to the control component (potentiometers, switch or button, and more and more joysticks)

***Proportional cartridge valves control***

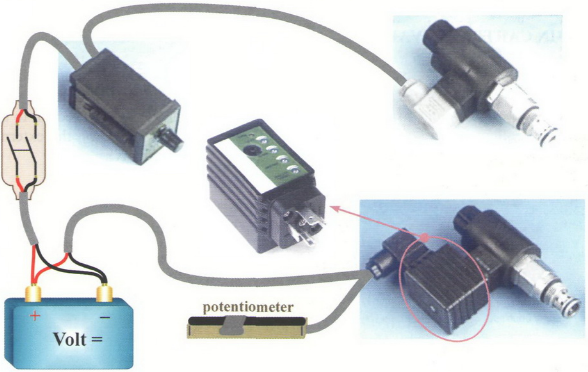



*Figure 12.66*

Figure 12.67 shows the most common electroproportional cartridges.

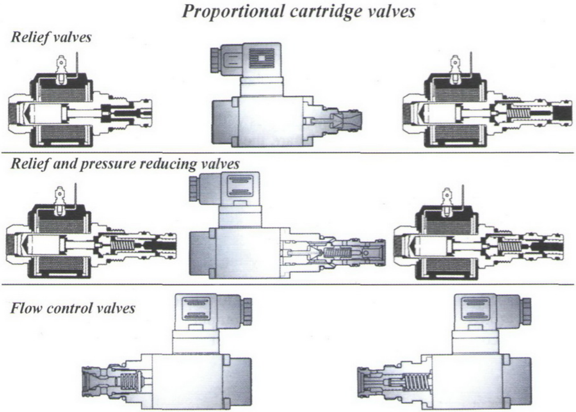



*Figure 12.67*

## SLIP-IN CARTRIDGE VALVES

Slip-in cartridge valves, also known as **logic valves**, have only two ports, no fastening thread and they are arranged in their cavity through a coulisse; they are fastened via a slip-in that obstructs the cartridge and activates the pilot connection or connects it to the upper valve. Static sealing is ensured by some O-rings that are placed on the external side of the cartridge and push against the cylindrical surface of the cavity.


*Figure 12.68*

Besides being arranged in a single hollow body, in many systems the cartridge is made up of two cylindrical hollow components of different size. The diameter of the first component (referred to as 'collar') is higher than the second; it is next to the flange and blocks the head of the cylinder, which has a smaller diameter. The second component, called 'bush', matches cavity channels through the central and side holes. In this case too, an O-ring ensures static sealing between the two cylinders. The poppet inside the bush enables the hydraulic communication between the two cartridge/cavity ports; the poppet is opposed by a spring that pushes against on the one hand the internal seat of the poppet and on the other the upper part of the collar.

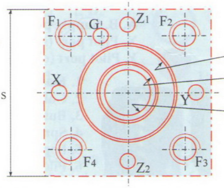
### Size

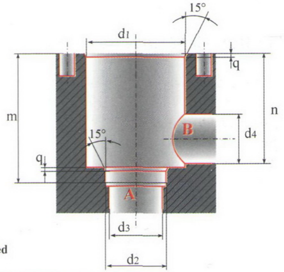
The sizes of slip-in cartridge valves change according to the bush bore. ISO 7368 'Hydraulic fluid power -- Two-port slip-in cartridge valves – Cavities' sets these sizes, whereas manufacturers establish nominal and pilot pressure.

| Sizes of slip-in cartridges | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Port bore (mm) | 16 | 25 | 31,5 | 40 | 50 | 63 | 80 | 100 |
| Size | 06 | 08 | 09 | 10 | 11 | 12 | 13 | 14 |
| Maximum flow *[in general]* (l/min) | 200 | 450 | 700 | 1100 | 1700 | 2800 | 4500 | 7000 |

### Cavities and slip-in

***Cavities and slip-in sizes***




**A, B. Supply ports- X. Pilot, supply port- Y, Pilot, return port- Z1. Additional pilot, supply- Z2. Additional pilot, return- F1, F2, F3, F4. Threaded fastening holes- G. Coupling plug hole-**

| Dimensions (mm) | $d_1$ | $d_2$ | $d_3$ | $d_4$ | X,Y, $Z_1,Z_2,Z_3,Z_4$ | $F_1,F_2,$ $F_3,F_4,$ | G | s | m | n | q |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Size 06 | 32 | 25 | 16 | 16 | 4 | M8 | 4 | 65 | 54 | 42,5 | 2 |
| Size 08 | 45 | 34 | 25 | 25 | 6 | M12 | 6 | 85 | 70 | 57 | 2,5 |
| Size 09 | 60 | 45 | 32 | 31,5 | 8 | M16 | 6 | 102 | 83 | 68,5 | 2,5 |
| Size 10 | 75 | 55 | 40 | 40 | 10 | M20 | 6 | 125 | 102 | 84,5 | 3 |
| Size 11 | 90 | 68 | 50 | 50 | 10 | M20 | 8 | 140 | 117 | 97,5 | 4 |
| Size 12 | 120 | 90 | 63 | 63 | 12 | M30 | 8 | 180 | 150 | 127 | 4 |
| Size 13 | 145 | 110 | 80 | 80 | 16 | M24 | 10 | 250 | 200 | 170,5 | 5 |
| Size 14 | 180 | 135 | 100 | 100 | 20 | M30 | 10 | 300 | 239 | 205,5 | 5 |

NB:13 and 14-sized flanges are round and have 8 same-bored fastening holes (F1...F8)

*Figure 12.69*

The dimensioning of cartridge and flanges of cartridge valves are defined according to their size (Figure 12.69).

**Cartridge**

Slip-in cartridges require a poppet, whereas balls are impossible to use. These cartridges are essentially non-return valves that perform their functions individually or with other external valves through the flange.

Unlike non-return valves, slip-in cartridges have a pilot system that is based on three different areas on the poppet. These areas allow free flow in one or both directions depending on their type and application; when they are coupled with their respective valves on the flange, they act as directional valves, relief/pressure control valves and flow control valves. Slip-in cartridges, which can be considered as zero overlap, ensure rapid responses; this allows high-quality automations and diminishes down-time that is so frequent with positive overlap spool valves thus reducing operating cycles.

Figure 12.70 shows the details of a slip-in manifold with cartridge and block.

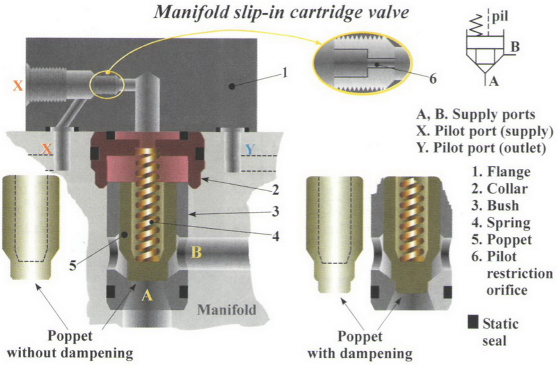



*Figure 12.70*

The poppet without dampening allows a quick start of the actuator while the dampened version promotes a soft start; the latter can be used with the directional function only.

The poppet design determines two or three control areas (Figure 12.71). In the poppet ($\alpha$) with two effective surfaces $A_A$ and $A_p$, since the total area $A_p$ is the sum of the lower seat of the spring and the upper end of the poppet itself, their ratio is 1:1. If the poppet has a different shape ($\beta$, $\gamma$, $\delta$) there are three surfaces; as a matter of fact, the third surface $A_B$ is the result of the difference between $A_A$ and $A_p$. In this case the ratio depends on the space occupied by $A_B$. This ratio usually equals 1:1.07 or 1:2 according to the use and the design. Other ratios such as 1:1.1, 1:1.5, 1:1.6 and 1:1.8 are not rare.

In this manner, the spool can be shifted with a pressure on port B, which allows flow through A.

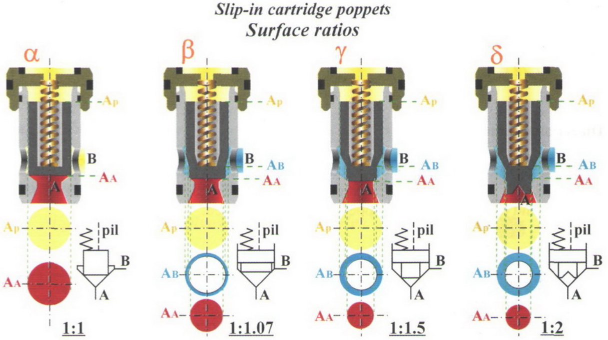



*Figure 12.71*

For instance, in a size 10 poppet with a 1:1.07 ratio, the surface $A_p$ equals 12.56 cm$^2$ and $A_A$ must be 11.74 cm$^2$ (12.56/1.07 = 11.74); the $A_B$ surface is equivalent to their difference, that is 0.82 cm$^2$.

If the size is 08, the ratio is 1:1.5 and the surface $A_p$ equals 4.91 cm$^2$, $A_A$ is equivalent to 3.3 cm$^2$ (4.91/1.5 = 3.3) and $A_B$ to 1.61 cm$^2$ ($A_p - A_A$).

This principle is shown in Figure 12.72. The poppet with two surfaces having the same diameter (2S) blocks flow from B to A, but it allows flow from A to B. With the inclusion of a pilot line on $A_p$ and a force at least equivalent to the force exerted by a pressure on A (the force on $A_p$ is equal to a certain pressure plus the spring force), A-to-B flow too is blocked.

Three-surface poppets (3S) offer many directional options. Failing pilot, the pressurised fluid from A would flow freely to B and vice versa since it would act on the surface $A_B$. With pressure in B and pilot on $A_p$, the flow is blocked; the pilot from the pressurised line A blocks the whole flow. It is crucial to highlight that the 3S position marked with an asterisk (lower right corner), with pilot from the pressurised A, does not guarantee perfect sealing since the fluid can flow from $A_p$ to B through the clearance of the cylindrical fit.

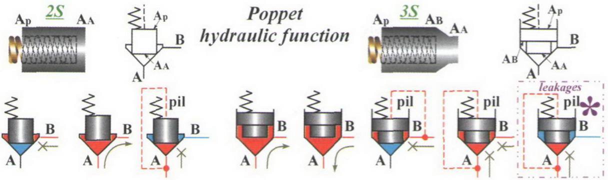



*Figure 12.72*

**Directional control**

Directional control can be obtained not only with the simple methods previously described, but also with the inclusion of a solenoid valve on the covers of slip-in cartridge and (a) three-surface poppets that can be piloted.

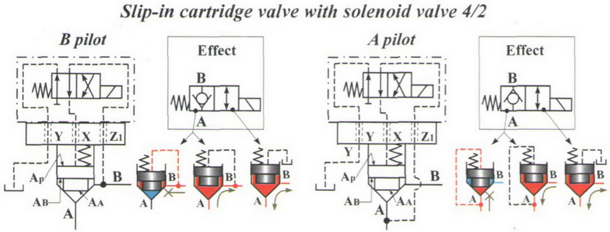
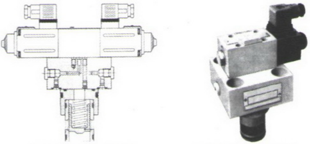



*Figure 12.73*

A directional valve is included in the left diagram (B pilot – Figure 12.73). This means the solenoid is excited and the fluid can flow in both directions; when the solenoid valve is in the rest position, the flow is free from A to B and blocked from B to A, whereas $A_p$ is connected to the tank. In the right diagram (A pilot), the results are the same with the excited solenoid, while A-to-B flow is blocked if the solenoid valve is at rest. Figure 12.74 shows the main connection diagrams of a cartridge/slip-in/solenoid valve 4/2 and 4/3.

***Connection diagrams: cartridge, slip-in valve and solenoid valve***



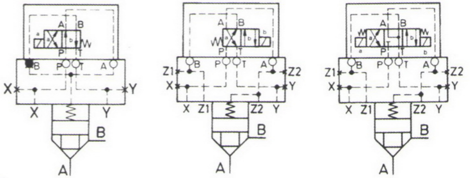



*Figure 12.74*

The inclusion of a **shuttle valve** enables to take the pilot fluid from whether line A or B; the solenoid valve determines the cycle (Figure 12.75).

***Slip-in cartridge valves with shuttle valve***

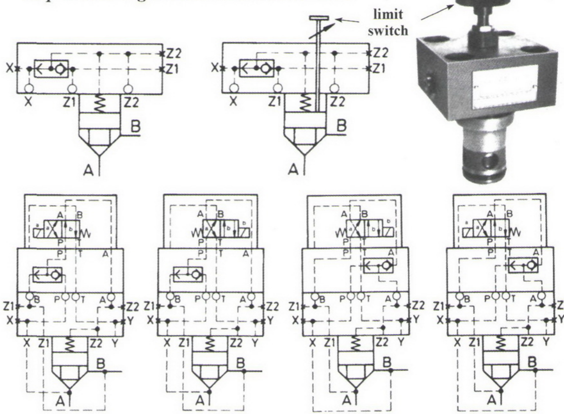



*Figure 12.75*

The inclusion of a **calibrated hole** on the connection X of the slip-in cartridge (Figure 12.70 Manifold slip-in cartridge) slows cartridge movements down thus avoiding vibrations. The diagram in Figure 12.76 shows pressure drops in the hole according to the flow.

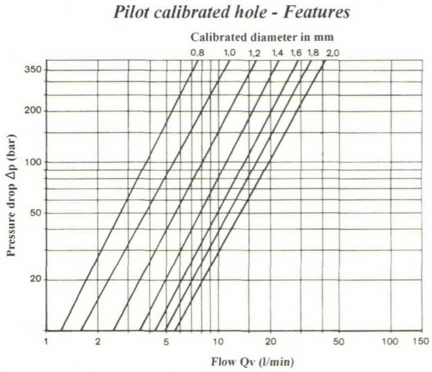



*Figure 12.76*

As a result, an actuator can be operated by means of cartridge valves managed by a distributor or a pilot solenoid valve.

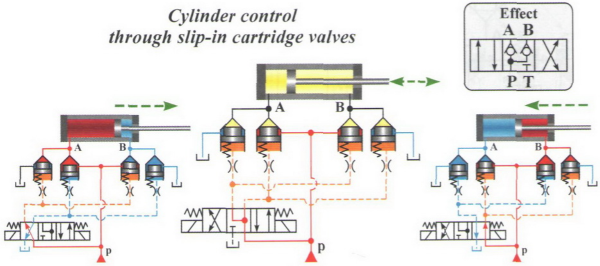



*Figure 12.77*

Unlike the control with two-stage spool valves that are always subjected to leakages, sealing is excellent in every such system (Figure 12.77). Furthermore, unlike two-stage spool versions, costs are considerably reduced in systems provided with medium- and large-sizes valves.

Regeneration circuits too are feasible with manifold slip-in cartridges (Figure 12.78).

***Regeneration circuit through slip-in cartridge valves***

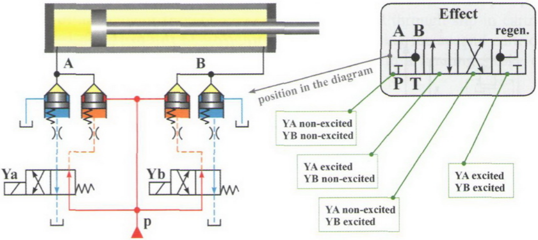



*Figure 12.78*

**Pressure control**

Pressure relief and control with slip-in valves are always pilot-operated type. The cartridge acts as the main stage while the pilot is assembled in the coverage flange. Poppets have two surfaces $A_p$ and $A_A$.

***Pressure relief slip-in cartridge valve (two-stage)***

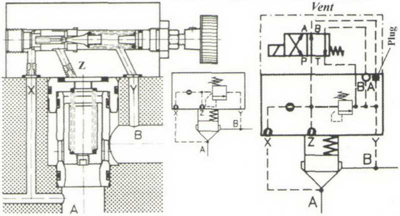



*Figure 12.79*

Pressure **relief** slip-in cartridges have the same operating principle as panel valves. The pilot stage, made up of the traditional spring-poppet relief valve and held inside the flange, is connected through pilot port X to the port A of the cartridge, which is kept closed by the working pressure derived from Z and acting on the $A_p$ area.

The pilot poppet opens when the pressure is higher than the pressure set, allowing the fluid to reach port Y; the pressurised fluid in A lifts the poppet of the cartridge (main stage) flowing from B to the tank (Figure 12.79).

If a non-return valve and an indispensable pilot connection are added, the slip-in cartridge described above can be used as an **unloading** valve for low/high pressure double pump or in circuits equipped with an accumulator (Figure 12.80).

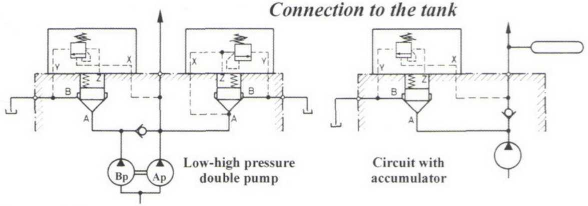



*Figure 12.80*

Pressure-reducing pilot-operated cartridges share the same operating principle as traditional cartridges.

**Flow control**

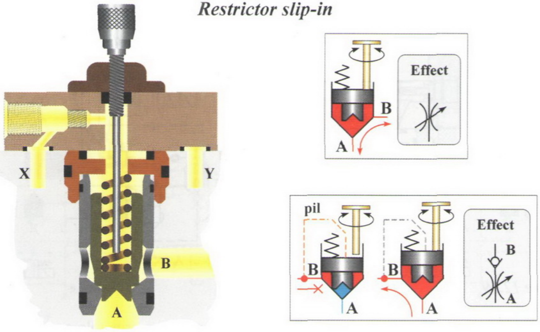



*Figure 12.81*

Slip-in cartridges are useful also for flow control. Non-compensated **adjustable restrictors** require poppets having a ratio of 1:2. Check devices with internal or external control through an auxiliary distributor can be made not only with two-way devices (flow from A to B and vice versa) without pilot, but also with appropriate connections on line X.

The restriction can easily be set with an adjustable screw that limits the stroke of the poppet (Figure12.81). The reduced flow non-return action can be obtained by connecting the pilot port X to the port B of the cartridge.

The speed of an actuator (Figure 12.77, Cylinder control through slip-in cartridges) can be adjusted, for example in meter-in circuits, by replacing the second cartridge on the left with a cartridge having a ratio of 2:1 provided with a limiting stroke device (Figure 12.82).

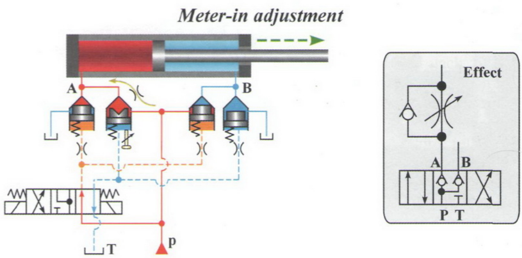



*Figure 12.82*

The slip-in technique for **compensated** flow control requires the use of two components: a three-surface poppet having a ratio of 2:1 and provided with a limiting stroke screw and a 1:1 two-surface poppet (Figure 12.83).

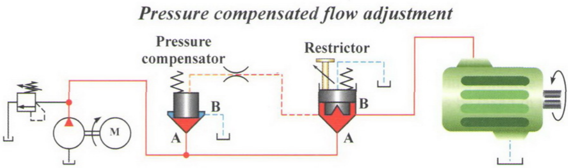



*Figure 12.83*

Slip-in cartridges with a 2:1 poppet reduce flow according to the screw setting, whereas the 1:1 cartridge develops pressure compensation.

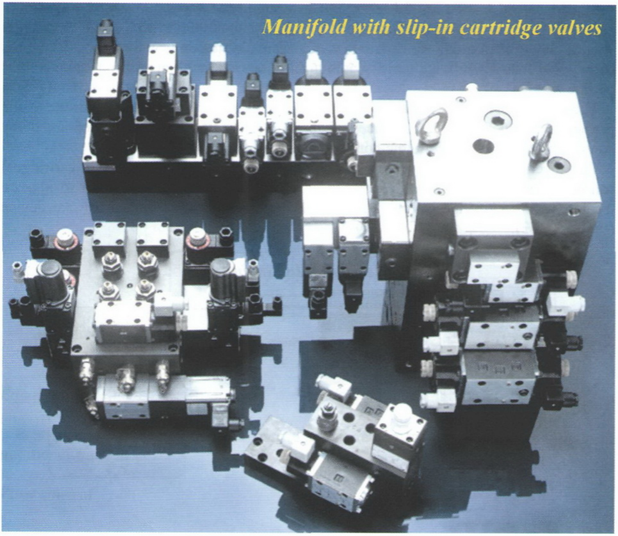



*Figure 12.84*