Chapter 5

OIL HYDRAULIC PUMPS – CONTROLS AND FURTHER IN-DEPTH ANALYSIS

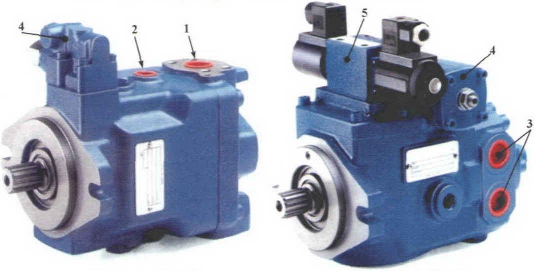
We have already mentioned the need for displacement adjustment in variable displacement pumps in chapter 4. This subject was partially elaborated on when we dealt with mechanical controls in radial piston and vane pumps and the main applications have already been considered; we are now going to carry out an in-depth analysis of high-precision controls, which are essential in systems demanding flexibility and accuracy.

As we already stressed, the pumps suitable for closed circuits (for rotary hydraulic actuators with direction reversal) must allow **flow reversal** while keeping the prime mover in the same direction of rotation. As a result, inlets/outlets have the same bore and a mechanical, hydraulic or electrohydraulic device to reverse the plate angle, normally connected to the displacement controller, is fundamental. In theory, every pump with in-line or inclined cylinder block can be equipped with the flow reversal device; actually, the standard pumps than can be applied to stationary open circuits are one-way pumps whose inlets/outlets have different bores and they also are less expensive because there is no such device.

Moreover, flow reversal pumps do not need inlets/outlets with different bores because they never perform direct suction. An auxiliary pump that is coaxial with the main pump guarantees the replenishing fluid of the primary (closed) circuit in the applications in the mobile industry, like the hydrostatic drive in heavy vehicles (crawler or wheeled excavators, combine harvesters, equipment for asphalt surfacing removal, truck mixers, etc.). The auxiliary pump has an interface circuit and obviously sucks directly from the tank.

It is not unusual in circuits for small/medium drive to find some versions with main/auxiliary pumps in a compact casing, like in the Figure 4.19 'Gerotor flanging' in chapter 4 or, in circuits for powerful drive, coaxial pumps made up of a main in-line axial pump and an auxiliary external gear pump, such as in the Figure 3.14 'Coaxial pumps' in chapter 3.

Axial piston pumps - Variable displacement



for open circuit *for closed circuit*

1. Inlet **2. Outlet** **3. Inlet - Outlet**
4. Displacement controller **5. Flow reversal electromagnetic valve**

Figure 5.1

DISPLACEMENT ADJUSTMENT

Variable displacement pumps can be adjusted with manual or automatic systems. **Manual adjustment** is performed via a worm screw connected to the hydraulic component and ending with a hand wheel (see the Figure 'Variable pump with inclined cylinder block' in chapter 4); yet, manual adjustment is imprecise, slow-responding and obviously it does not allow any automatic control in the event of load variation. We stress the fact that this elementary system must not be mistaken for the mechanical adjustment mentioned in the paragraphs about variable displacement radial piston and vane pumps because in these types of pump the initial manual adjustment sets flow to zero depending on the maximum pressure of the connected circuit. The manual adjustment system was improved by replacing the screw with a double-acting cylinder, but it is not worth analysing it because this device dates back to the prehistory of hydraulics!

The different types of displacement controllers are applied to pumps depending on the needs of the machine on which the pump is installed. A principle all controllers (also known as **compensators**) share is the relation between the pressure in the pump or the actuators and the resistance opposed by a force inside the compensator itself. A mobile component inside compensators sustains on the one hand the force exerted by pressure and on the other the resistance opposed (usually a spring). As this ratio goes up or down, the mobile component moves, thus affecting displacement either mechanically (for instance, if it moves the inclined mobile component of the pump) or hydraulically (when it opens a clearing towards the plate adjustment piston). In addition, the simplest mechanical compensators for vane and radial piston pumps already mentioned do not fall into this category because they do not have any moving component; in any case, these types of pump can be interfaced with the compensators we are now going to analyse.

As we analyse the different adjustment components, it will be evident that they have almost the same flow or pressure adjustment operating principle, while simple details, like the different flexibility of the spring, affect their fields of application. Consequently, they are generally defined as flow or pressure controllers, whereas the specific term mainly refers to the use of the special circuit in which, for example, the final result is zero flow or constant pressure.

Another problem is the choice of the most precise term between 'control' and 'compensator'. Even if the former is suitable for every situation, most of these components actually *compensate* the variation of a parameter by changing another parameter: for instance, the decrease in flow in a constant power regulator is compensated by the increase in pressure; yet, note that a flow stop controller does not compensate very much! As a matter of fact, the flow is constant even at a variable pressure. The word 'compensator' must therefore be used with proper prudence, in those applications where there is an actual balance between pressure and flow.

Constant power regulator

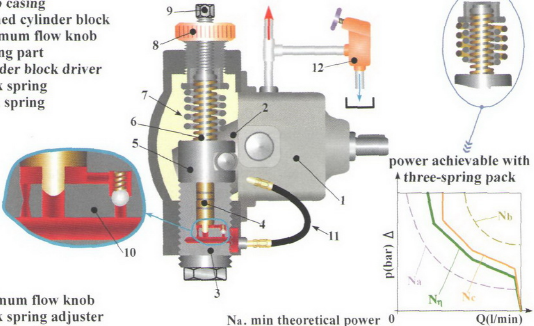
Some systems need to exploit the maximum power allowed and to keep it constant even if speed (flow) varies substantially and load (pressure) soars. Since hydraulic power is the product of pressure and flow ($N = \dfrac{Q \cdot p}{600 \cdot \eta_g}$), the less the load is, the higher the flow rate is and vice versa, which makes the result of the equation stay the same. The compensator, switched on by the pilot pressure Δp obtained from the pressure in the pump, affects displacement depending on the difference between pressure itself and the resistance exerted by the opposing springs (Figure 5.2).

Assume there is a system whose load needs a constantly increasing pressure and whose translation speed does not need to be kept constant (prerequisite for a stable constant pressure). The moving component (4), strained by pressure, moves the driver (5) upward, which lessens the inclination of the cylinder block (2).

At the beginning the unit faces the resistance of the slight force of the Qmax spring that, at low pressure (high Q, low p), diminishes the inclination of the cylinder block by a few degrees. As pressure rises (flow must be further reduced in order to keep the power absorbed by the pump constant), the thrust of the piston moving component make the driver touch the Qmin spring, which is more rigid than the other spring (the simple situation described so far is meant to make understand the process; in real designs, like in the compensator in Figure 5.10, – the Qmin spring too touches the component (5): as pressure goes up, it becomes more rigid than the Qmax spring). From now on, the increasing pressure acts, via the moving piston, against the Qmin spring that is manually adjusted by means of the knob (8).

Constant power regulator

1. Pump casing
2. Inclined cylinder block
3. Maximum flow knob
4. Moving part
5. Cylinder block driver
6. Qmax spring
7. Qmin spring



8. Minimum flow knob
9. Qmax spring adjuster
10. Deceleration valve
11. Pressurised connection
12. Relief valve

Na. min theoretical power
Nb. max theoretical power
Nc. actual power
Nη. real power

Figure 5.2

The relief valve (12) is set according to the maximum pressure required by the load and certainly it is not more than the working pressure specified on the pump plate. The deceleration valve (10) (whose operating principle is the same as for the valves on the actuator cylinders) cushions the quick piston stroke, especially when the pressure drops excessively and suddenly. Under these circumstances, displacement does not soar or plunge, thus avoiding hydraulic recoils on the system and the pump.

The line chart in the lower left corner of Figure 5.2 shows the actual power Nc that can be reached if the controller is equipped with a three-spring pack (hence three different grades of rigidity). Due to volumetric and mechanical efficiency, the real power is Nη, which in any case is almost the same as Nc. The curve trend is not constant because springs respond according to the system requirements: a rise in pressure results in less rigidity with the ensuing inclination angle change.
The maximum Nb and minimum Na pressure is the pressure that can be obtained by replacing again the spring pack of this compensator.

Flow stop controller

Flow stop controllers guarantee a constant and maximum flow throughout the whole pressure range. When the pressure reaches the maximum level set, flow rate is zero, although the oil leaked is replenished (this must not be mistaken with the system described in respect to vane pumps because a rise in pressure in vane pumps makes flow drop down to zero gradually). The operating principle of flow reduction controllers and constant power regulators are similar. Flow reduction controllers are equipped with a single return spring touching, on the one hand, the compression hand wheel and, on the other, the cylinder block driver if there is an inclined cylinder block pump or on the rod/pin solidly connected to the inclined plate (Figure 5.3).

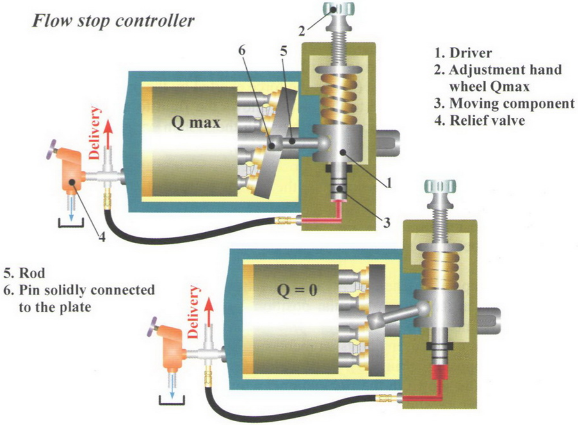


Figure 5.3

The force exerted by the spring opposes the pressure acting on the moving component. As the small piston moves when the spring force yields to the pressure rise, it acts on the driver that puts the swash plate in line with the drive shaft (p max, Q = 0). Maximum pressure is determined by the compression the hand wheel exerts on the spring. In order to avoid excessive pressures that can damage the pump and the circuit, it is important to add a relief valve also in this case.

Other controllers related to flow management

Torque limiting compensators are not suitable for systems demanding frequent standbys at maximum pressure (loaded linear or rotary actuators waiting for a new order like, for instance, a cylinder pressing on a workpiece undergoing strain or a hoist keeping a weight suspended) because the energy wasted by the relief pressure during standbys at maximum pressure is an additional energy cost. As a result, another device is added, i.e. a **hydraulic pressure limiting** controller that keeps the maximum pressure reached and reducing flow to a very low level so as to compensate leakages only.

Flow limiting controllers ensure that flow ranges only between the limits set but they allow pressure variations. Consequently, the translation speed of the actuator cannot exceed the limit required by the manufacturing process despite pressure variations (the higher pressure is, the lower flow is and hence speed, or vice versa).

Pressure limiting compensator

This type of compensator guarantees the constant pressure of the load despite flow variations (Figure 5.4). A clearing next to the outlet leads the pressurised fluid to the piston (5) opposed by the spring (3). As the force exerted by the pressure overcomes the spring force, the clearing close to the piston allows the fluid to act on the inclination piston (8). Oil or any other fluid compatible with the system parts is sent to the spring chamber through a tiny clearance (10) in the rod (6).

In order to understand the adjustment process better, assume an actuator (be it a motor or a cylinder) is directly connected to the pump. Note that the compensator pilot pressure is not the same as the working pressure, but it amounts to about $3 \div 12$ bar, in proportion to the maximum pressure the pump can sustain; in addition, the opposing spring too must be adjusted according to the pressure characteristics of the generator.

When the pump is switched on (hence the actuator does not sustain any strain), the inclined plate is in its extreme angular position, so flow and pressure are respectively maximum and almost zero.

The movement of the user entails an increasing pressure, however it is not enough to outweigh the opposing spring (3). As a result, flow is still maximum while pressure keeps on rising (as shown in the right compensator in the Figure); the spring chamber, the chamber under the slider and the inclination piston (8) are connected to the drainage hole, in other words unloading into the tank. The revolution of the hand wheel (2) determines the tension of the spring (3) establishing the adjustment pressure, i.e. the pressure resulting in the fluid flowing towards the inclination piston (8). When this pressure is reached, the piston (5) moves upward, allowing the pressurised fluid to act on the piston (8) with the ensuing decrease in the angle of the swash plate: flow diminishes in favour of the pressure the actuator needs (left compensator in the Figure).

As a matter of fact, during this phase, as flow diminishes, pressure too drops slightly in the beginning, so as to reverse and immediately stop the movement of the piston (5)/slider (4) unit: the drain connected to the piston (8) is thus clogged and the plate stabilises the new and reduced flow by keeping its own new position. Note that the return stroke of the adjustment unit (4-5-6) must be very short because the drainage hole must not open when the pressure conduit leading to the piston (8) clogs, otherwise the fluid in the inclination piston would be unloaded and the actuating piston (9) would bring the plate back to the maximum displacement.

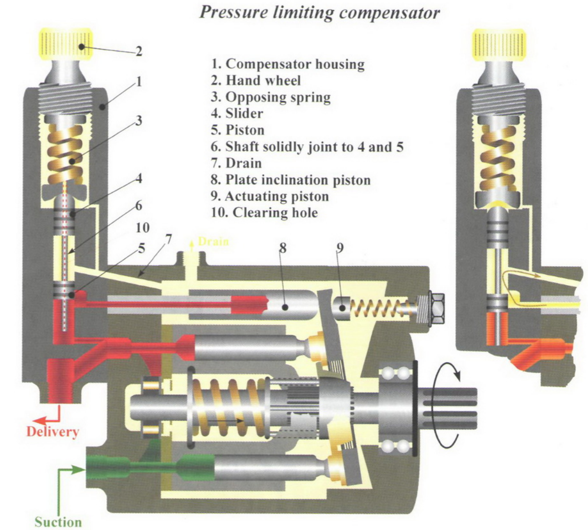


Figure 5.4

If pressure keeps on going up, the piston (8) drives the plate into the zero displacement position, so there is zero flow (or almost zero because leakages are replenished) and maximum pressure maintained. A relief valve on the delivery hose stabilises the maximum strain allowed.

As the load on the actuator diminishes in the final phase of the cycle, the pilot pressure of the controller decreases; the opposing spring (3) pushes the slider downward, the drainage clearance to the piston (8) is blocked, thus opening the path to the drain. The piston (9) brings the plate back into the extreme angle position, resulting in the initial situation (maximum flow, zero pressure).

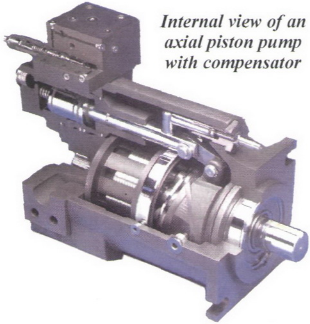


Figure 5.5

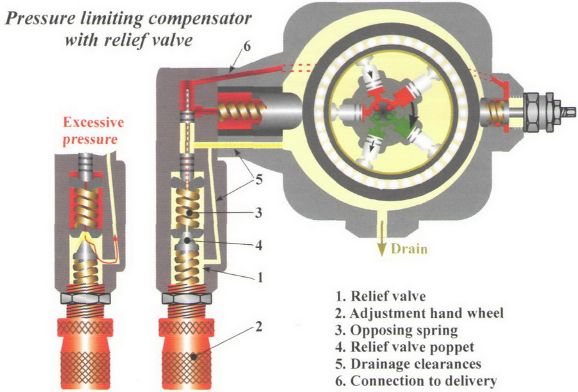


Figure 5.6

Some manufacturers place the relief valve inside the compensator so that it is coaxial to the adjustment hand wheel (Figure 5.6). This valve is very tiny because it does not affect nominal pressure but the pilot pressure acting on the opposing spring via the clearance between the slider and the pin. The force the adjustment hand wheel (2) exerts stabilises both the relief valve setting (1) and the opposing wheel setting (3). If there is an excessive load, the pressure on the spring (3) pushes the poppet back (4), thus allowing the fluid to flow in the drainage conduit (5).

The pressure limiting compensator described so far is applied to a radial piston pump, but it is suitable for any other type of variable displacement generator

Load Sensing pressure limiting compensator

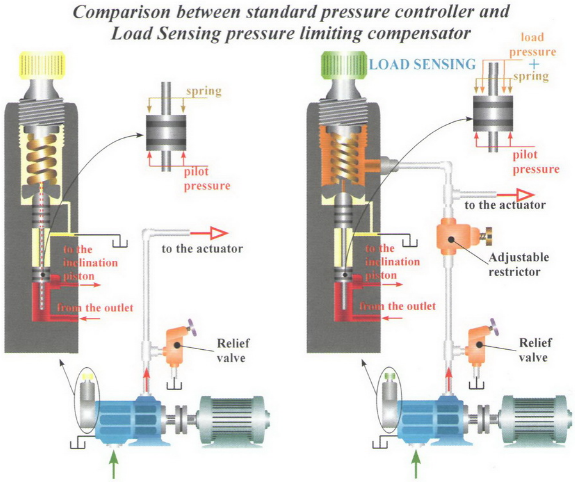


Figure 5.7

If the system is equipped with the device known as **Load Sensing**, displacement variation does not depend on the pressure over the generator outlet, like in the previous compensators, but on the opening of the outlet restrictor. As a result, this controller ensures a constant difference between the pump outlet pressure and the working pressure (usually 14-20 bar). The restrictor is essential because a pump without it would always reach the maximum displacement. Flow does not vary with the load within the working limits. The Load Sensing pressure limiting compensator continuously compares outlet and working pressures and sends the fluid signal to the displacement controller.

In standard controllers, the fluid flowing to the piston (8) (see Figure 5.4 'Pressure limiting compensator') is affected by the relation of the forces acting on the piston (5) and resulting from, on the one side, the pilot pressure due to delivery and, on the other, the opposite mechanical energy the spring exerts on the slider (4) and transmitted via the rod (6). In Load Sensing compensators, the balance on the piston (5) always stems from the outlet pressurisation versus the force of the spring properly adjusted, *plus the pressure on the load.* In order to implement Load Sensing settings, it is fundamental to add a restrictor after the outlet so as to entail a pressure drop between the pump and the actuator.

Figure 5.7 (comparison between a standard controller and a simple Load Sensing compensator, which is in any case found on some applications) displays the operating principle of this new element clearly.

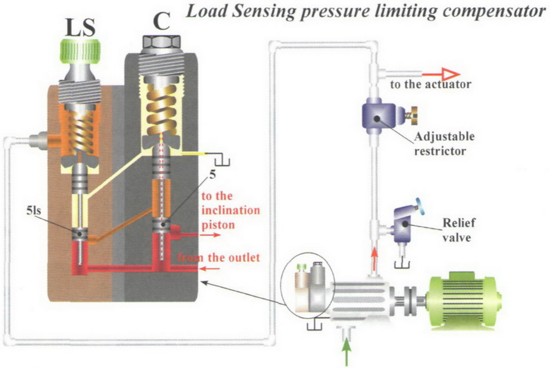


Figure 5.8

An additional controller in series with the main compensator, which is usually prearranged for coupling, can be added to ensure a more precise adjustment (Figure 5.8). In this version, the main compensator C works as a standard controller as the spring is not powered by the load pressure; the second controller LS, powered by the outlet pressure, sustains the load pressure in the spring chamber: the pilot pressure due to the delivery acts on one side of the piston (5ls), the load pressure and the spring on the other.

The fluid flowing out from LS acts on the upper part of the piston (5) of the main compensator C and it opposes the outlet pressure. Hence, the operating principle is similar to the previous one because the ultimate goal is to send the Load Sensing signal on the compensator piston (5) in contrast to the opposing force of the outlet pressure. The flow control valve (adjustable restrictor) permits to set the flow rate that best suits the actuator (operation speed). A relief valve that is coaxial with the adjustment hand wheel can be applied also to Load Sensing compensators (Figure 5.9).

Load Sensing pressure limiting compensator with relief valve

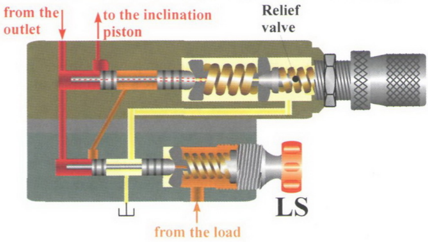


Figure 5.9

Torque or power summation

Some applications, especially in the mobile industry, demand a controller that can manage the maximum power of the prime mover to which two variable displacement pumps are connected (the pumps work either simultaneously or alternately).

In order to better highlight this problem, consider the hydraulic system of a powerful wheeled earthmoving machine. The different operations of vehicle self-propulsion, stability, various movements related to earthmoving and so forth (flattening, pitting, silting up, levelling off embankments, debris removal, gravel loading, etc.) and other applications like heavy percussive hammers are oil hydraulically operated; the pumps needed to perform these and other operations are assembled to the prime mover.

Obviously, it is not convenient to have a pump for each application because too many pumps would be needed, with the ensuing prohibitive costs and excessive need for space. In short, the hydraulic circuits of a powerful standard excavator can be listed as follows:

1. Hydrostatic drive (vehicle self-propulsion) served usually by a variable displacement pump in closed circuit, plus a second small auxiliary fixed displacement pump.
2. Hydrostatic power steering served by a dedicated fixed displacement pump or connected to the pump of another system via a priority valve.

3. Stabilising legs (four cylinders hold up the vehicle in order to prevent it from moving during the excavation) by the pump of another system that are kept in their position by their blocking valves.
4. Digging bucket operated by three or more cylinders and handled on a perpendicular sliding guide through one or two hydraulic motors. Sometimes the bucket is replaced by a percussive hammer for building demolition. The circuit is served by a dedicated pump.
5. Removal bucket (earthmoving or other operations) or levelling blade, operated by two or more pairs of cylinders connected to a pump.

Even if we do not take into consideration the pump for hydrostatic power steering, as well as the *auxiliary* pump for the drive as it needs a negligible amount of energy, the pumps mentioned at points 1, 4 and 5 have a high energy demand, which is even higher than the nominal power of the endothermic motor if they occur simultaneously when there is the ma maximum demand.

Since the closed circuit pump of the hydrostatic drive must be managed autonomously for a series of reasons analyses later on, it is vital to act on the power management of the pumps of points 4 and 5. There is a reason why the displacement type was not specified at points 4 and 5. As a matter of fact, fixed displacement pumps are assembled on some types of excavator (we have already mentioned this in the previous charter about 'intravane' vane pumps), while others manufacturers opt for variable displacement systems, which is what we are going to deal with in respect to torque summation.

The two variable displacement pumps, which are available on the market already arranged in a single casing, are connected to the shaft of the Diesel motor via a gear transmission; the driver of the inclined cylinder block of each one is connected to a single torque summation. The pilot pressure from the outlet to each pump sums uninterruptedly in the controller (Figure 5.10). This permits to exploit the power of the prime mover fully when one of the pumps is not working or when on the contrary it is working within its power limits.

Consider a Diesel motor with a power of 100 kW: if each pump had its own controller and assuming some processes demand the maximum power from both of them simultaneously, the dimensioning of each pump could not exceed 50 kW (efficiencies excluded) because the summation would be equivalent to the power of the prime mover. When circuit α has zero pressure, the power of the motor can be exploited only by half at best because the relief valve of circuit β is set to a maximum pressure/flow of 50 kW, i.e. the highest power its pump can deliver.

If a single torque summation control is added, the pilot signals of α and β are summed in the device that inclines cylinder blocks depending on the overall pressure. The two coaxial pistons of the controller that receive the pilot fluid must have the same cross section in order to avoid thrust imbalances. *Each* pump can be designed to process *the same power as the prime mover*. As a matter of fact, when pump β is not operated, the pressure/flow ratio in pump α can be almost equal to the overall motor power: the

driver, moved by the piston subjected to high pilot pressure, pushes the cylinder block into the minimum displacement position (max pressure, min flow).
When the power of pump α is at 20%, pump β can work up to 80%, if necessary: when the summation of pilot pressures α is 20% and, for example, β is 65%, motor power equals 85 kW, efficiencies aside.

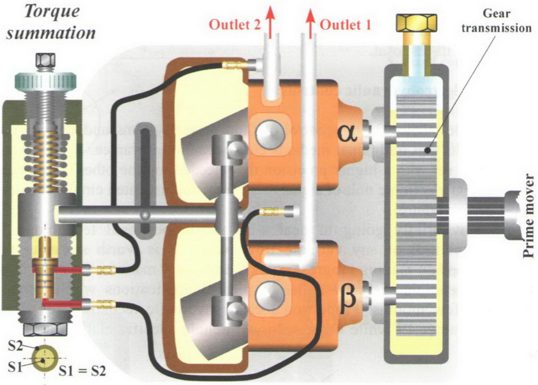


Figure 5.10

These situations are quite usual in real applications. In order to understand how important this is, consider two examples using a machine already mentioned, a wheeled excavator with a prime mover of 100 kW, two identical pumps with 100 kW each and the torque summation control.

I. A quite deep pit is being dug in a very hard ground with a digging bucket; pump power is in the region of 40 kW, the other systems are not working (except for the stabilising legs held in position only by their blocking valves). Sooner or later the teeth of the digging bucket are confronted with a very tough obstacle, like a big rock firmly embedded in the ground. Pressure shoots up in the hydraulic circuit of the bucket, leading the compensator driver to drive the pump displacement up to the minimum level. The force exerted on the bucket, equalling roughly 100 kW, budges the rock, then pressure decreases, flow increases and the obstacle is removed. This operation would have failed with a machine equipped with pumps with a maximum power of 50 kW and single controllers.

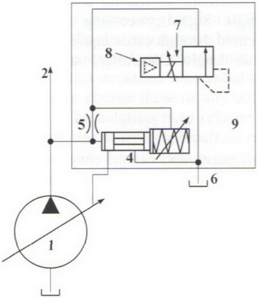
II. In another part of the pit the ground is very uneven and muddy and the stabilising legs cannot ensure enough static quality. Consequently, the excavator driver props up the second bucket, i.e. the removal bucket, against the ground. During the excavation, the digging bucket keeps on bumping into large obstacles and the equipment is subjected to jerks that undermine its already precarious stability. The excavator driver is thus forced to act simultaneously on the two systems: during the excavation he ‘adjusts’ the removal bucket on the ground. The removal bucket absorbs 25% of the overall power; hence the digging bucket has 75 kW at its disposal.

Proportional electrohydraulic controls

The use of electronics for the control of fluid power systems undoubtedly results in high-level results that traditional methods clearly cannot guarantee. If on the one hand controls can be set with a higher precision than needed, on the other the processing in the management electronic unit of signals sent from sophisticated circuit sensors allows an almost perfect control.

We are now briefly going to deal with the proportional technology for the displacement control of any variable flow pump. It is worth stressing that this technology keeps on making progress quickly and every manufacturer can improve it in many respects; that is why a list of present applications would prove to be obsolete in a few months. Proportional electrohydraulic technology is thoroughly analysed in chapter 19 while other sections include specific elaborations on pump controls.

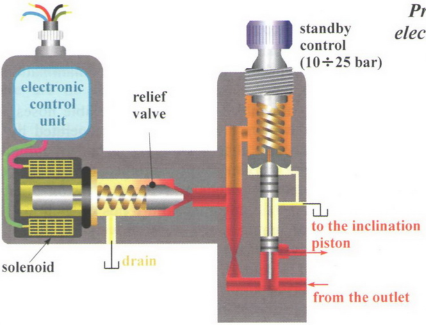
The hydraulic system of the compensator is usually similar to the Load Sensing type, whereas the electronic control is performed on a relief valve provided with an electromagnetic coil. This coil receives the signal of the electronic control unit and governing ‘proportionally’ the pressure of the fluid sent into the spring chamber by the compensator (Figures 5.11 and 5.12). The hydraulic operating principle is the same as for the previous controllers: the pilot signal comes from the outlet, the piston is subjected on the one hand to this pressure and on the other to the sum of the forces exerted by the spring and the fluid proportionally controlled in the chamber of the spring. The fluid is driven into both the spring chamber and the relief valve through a fixed constriction. The relief valve setting determines the pressure in the spring chamber. The electronic control unit receives the input from the circuit, processes it and transmits its command to the solenoid matched with the relief valve. This ‘command’ consists of a tension that varies according to the requirements of the system; the increase in tension causes the valve to open the constriction, while tension drop makes the clearance shrink. As we have already said, the relief valve constriction controls the pressure of the fluid sent into the spring chamber; the fluid surplus is unloaded through the compensator via the pump drain, like in the previous cases.



Relief valve in the hydraulic compensator

1. **Variable displacement pump**
2. **Outlet**
3. **Relief valve**
4. **Compensator**
5. **Fixed constriction**
6. **Drain**
7. **Solenoid**
8. **Electronic control unit**
9. **Compensator casing**

Figure 5.11



Proportional electrohydraulic control

Figure 5.12

The proportionality determined by the electronic unit is thus affected by the input signal(s), whether they are hydraulic or electric. Hydraulic signals, usually derived from Load Sensing, are transformed into electric signals by some special devices we are going to analyse later on; there are different types of exclusively electric inputs, from the simple command of an electromechanical limit switch to feedback detection with sophisticated transducers like encoders, magnetostrictives, LVDTs, etc.

The mechanical stress on the compensator spring usually leads to standby pressure (see chapter 3 – Intermittent operation): when the electronic unit is not working, pump displacement is at the maximum angle. Signal processing and the electronic management of the system can be implemented through cards equipped with a Single chip microprocessor, PLC (Programmable Logic Controller) and PC (Personal Computer).

IN-DEPTH ANALYSES

The limited analyses and remarks of this paragraph cannot conclude the issue of hydraulic generation. Pumps are affected by the peculiarities of each system; the choice of their specific characteristics is made taking into consideration the different parts of the circuit and the working conditions. Many parts of this text refer to this issue, which cannot be analysed at this stage yet because a specific knowledge on control and command, as well as on specific applications, is needed.

What pump?

The criteria followed to choose the generator often depend on a number of analyses that are anything but simple. Is it better a fixed or variable displacement, gear or screw, inclined or in-line cylinder block, radial or axial piston pump?

The type of pump, whether it has fixed or variable displacement, is certainly chosen according to flow/pressure characteristics, compatible rotational speed with the prime mover, viscosity grade, working life, resistance to contamination, heat and corrosion… and, last but not least, its good value for money.

As a small vehicle repair shop assembles two or three flexible hoses a week on average, it can be fully satisfied with a crimping machine operated by a manual pump, which is generally much cheaper than a crimping machine equipped with an electric pump. In a circuit whose maximum pressure is 80 bar, an economical external gear pump perfectly meets the needs and a radial piston pump that can sustain 300 bar is wasted! There are no advantages in terms of duration, efficiency and quality, but only costs, encumbrance and excessive weights. Still, the system must be analysed carefully. If a system has to reach peak pressures equalling about 250 bar, the gear pump is doomed to deteriorate soon and a radial piston pump is not suitable; the preference should instead go to cheaper pumps that can nonetheless sustain those pressures, like for instance an intravane pump.

Consider for instance a circuit like the previous one, but whose peak pressure reaches 110 bar and that is supposed to work in a silent environment. A gear pump *could* not satisfy the last condition and this can depend also on its positioning next to the machine: before choosing another type, it is worth ensuring whether a different positioning or a low-cost sound insulation are viable. In addition, a screw pump with the same performances occupies more space: can it fit in?

Like in any other industry, the quality of the product varies from a manufacturer to another because some focus on mechanical characteristics, while others on suction or pressure efficiency … luckily most manufacturers focus on all of them!

Unfortunately, some companies manufacture unreliable, short-lived products that do not deliver the performances glorified in their information documents. As a matter of fact, the ideal hydraulic pump must have minimum tolerances, bearings and bushes subjected to adequate heat treatments, accurate assembly and tests, practical experiments: all this demands substantial costs small-sized businesses cannot always afford. So, how to choose these or any other hydraulic components in order to avoid poor products? The solution is to rely on experienced manufacturers, to detect and record abnormalities whenever repair or replacement are performed, to compare the details of some parts, like the flutes next to the delivery slots in the distributor plate or those of the bushes in gear pumps, to ensure the pump is equipped with properly dimensioned bearings, an oil retainer made up of a good mixture and anti-extrusion rings coupled with seals; in addition, pumps must comply with ISO standards. Besides large-sized Italian and international companies, many small-sized businesses manufacture very competitive products and most of them are listed in the ASSOFLUID Directory, the directory of the Association of Manufacturing and Trading Companies in Fluid Power Equipment and Components.

Some different versions offer almost the same performances, so what criteria should be taken into consideration in order to choose a pump if even their prices are equivalent? In general, the choice is dictated by experience. Assume a company manufactures a certain machine and records good performances and few abnormalities on its axial piston pump with rounded head; when the compares its product to a similar product by a competitor with a different pump but the same price, there are no relevant differences. Why should the company change the product?

All in all, it all comes down to the same issue at stake with a wide range of commercial products, i.e. subjective choices. Indeed, we prefer by far a certain television to a similar model and shortly after we find out they are manufactured by the same company with the same techniques; we purchase a car by car maker A and not B because we have always been driving that brand; we do not drink Tom's wine but Dick's wine because we believe it is better, yet we are unaware of the fact that they both buy it from the same farm cooperative.

At first sight, a pump can often seem not powerful enough to meet the needs of the system, not only when different versions offer the same performances. The only task of the hydraulic system of a specific tool machine is to block four large vices having the same stroke and clamping force. Work pieces must never be clamped all at once, on the contrary vices are closed one at a time. If each clamping speed is ensured with a flow of 10 l/min, the pump has a displacement of 8 cm^3 and a volumetric efficiency $\eta_v = 0.9$; the pump directly coupled with a three-phase electric motor whose speed is 1400 rpm guarantees a flow of 10 l/min ($Q = c \cdot rpm \cdot \eta_v/1000 = 8 \cdot 1400 \cdot 0.9 /1000$). It would be nonsense to multiply the single flow rate by the number of vices because they are never closed simultaneously. Yet, as they are opened simultaneously and at the same speed as closing, the pump is not up to the task because a flow of 40 l/min would be demanded (the pressure/flow ratio, which varies between phases, is not taken into account in order to simplify the situation). The solution simply consists in adding a hydraulic accumulator that stores energy while the work piece is machined with closed vices, in other words a preset amount of pressurised fluid that is then unloaded into the cylinders of open vices.

The hydrostatic power steering of a not-so-powerful farm tractor is connected to the only pump the vehicle is equipped with via a priority valve (essential if there is no pump for the hydrostatic power steering). The pump powers not only the hydrostatic power steering but also the rear tool lift and it is dimensioned according to it. The priority valve diverts the flow in favour of the circuit of the hydrostatic power steering excluding or at least reducing the flow to the other device. A higher flow pump would have solved this problem without undermining the lift unit, but on the other side as it occupies more space it could not fit on the drive shaft. In addition, even if designers manage to work out this space problem, they must take into considerations the fact that competitors keep on assembling the system equipped with the priority valve in order to present costs from rising.

Notes on fixed displacement pumps

Anyway, many systems can be equipped with either fixed or variable displacement pumps. Flow and pressure in fixed displacement generators are controlled by some external components.

Consider the flow/pressure chart of a circuit with a cycle demanding two different power phases (Figure 5.13 - left). In the first phase, pressure is 50 bar and flow equals 60 l/min, whereas in the second phase they amount respectively to 175 bar and 30 l/min. If the pump overall efficiency is η_g is 0.9, the product of pressures by flow rates results in the power line chart of each phase (Figure 5.13 - right).

Chart 1

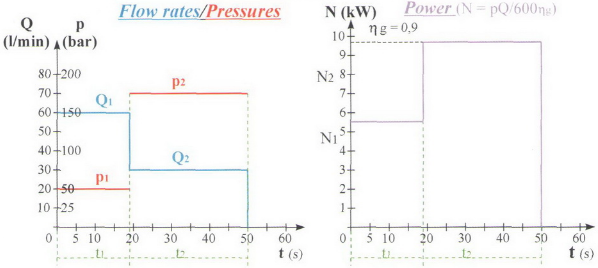


Figure 5.13

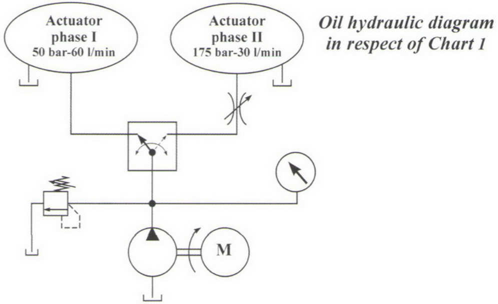


Figure 5.14

If a *fixed* displacement pump provided with a relief valve is installed, it is vital to choose a pump that is suitable for the maximum pressure and flow of the system, in the example p_2 and Q_1. The relief valve is set to p_2 (175 bar) and the prime mover must ensure a maximum power that is not equivalent to the phase N_2 (the maximum power the system is required) but to the maximum flow Q_1 and maximum pressure p_2 : $N_{max} = Q_1 \cdot p_2 / 600 \cdot \eta_g = 60 \cdot 175 / 600 \cdot 0.9 = 19.5$ kW. As a matter of fact, if the pump was suitable only for the second phase (i.e. the system maximum power phase), pressure would be equivalent to the demand, but in this case the pump chosen would have a maximum outlet of 30 l/min and in the previous phase it would not ensure the flow needed (60 l/min).

As the adjustable constrictor (Figure 5.14) in phase II controls both flow and a pressure difference, this system entails a substantial waste of energy and a considerable installation effort because the relief valve must unload the flow difference $Q_1 - Q_2$ into the tank in the second phase. In order to understand this concept better, consider each phase of the cycle by taking into consideration the previous chart and the oil hydraulic diagram in Figure 5.14.

- Phase 1: load pressure is 50 bar and the manometer displays it since there are no constrictions (pressure drops in the hoses are not taken into account), the maximum flow of the pump equals 60 l/min; the relief valve is inactive.
- Phase 2: the manometer displays 175 bar and flow rate is as little as 30 l/min due to the constrictor; the pump is exploited to its maximum potential: the flow rate is always at the maximum level since the pump has a fixed displacement. The relief valve, set to this pressure, unloads the fluid surplus into the tank. Consequently, a prime mover with a power of 19.5 kW is necessary.

Since the output power is 5.6 kW (50 • 60/600 • 0.9) in the first phase and 9.7 kW (175 • 30/600 • 0.9) in the second phase, the prime mover is even twice as powerful as demanded (Figure 5.15)!

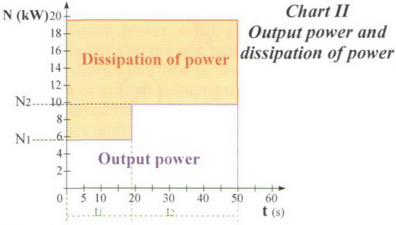


Figure 5.15

In order to curb the dissipation of power, it is not worth improving systems where, despite different flows, pressure in the two phases does not change very much, yet an energy waste like in the example is really exorbitant and must be reduced by changing the control method.

Before carrying out an in-depth analysis of this issue, we have to consider the behaviour of the fluid in the second phase of the cycle.

The pressure of 175 bar the manometer displays is equivalent to the sum of the actuator pressure and the pressure inside the constrictor. The task of the constrictor is to reduce the flow generated by the pump in order to slow down the actuator; the tiny clearance causes a pressure drop with the ensuing differential pressure. For instance, if the manometer displays a pressure p_1 of 175 bar (Figure 5.16) and p_2, which depends on the load, equals 110 bar, the resulting differential pressure at the constrictor ends is 65 bar (pressure drops in the hoses are not taken into account). By reducing the constrictor clearance, flow drops and the differential pressure rises. As we have already said, the relief valve unloads the fluid surplus into the tank. The chart in Figure 5.16 illustrates the output power and the dissipation of power of the second phase of the cycle only.

The diagram of Figure 5.17 shows the solution for circuits demanding two pressures/flow rates. The phases are controlled by their relief valves; this method is known as 'control at discrete pressure levels'. During the first phase of the cycle at the maximum flow rate, the relief valve ensures a proper operation and acts only if pressures exceed 50 bar. In the second phase the second relief valve, positioned after the diverter valve, is set to the actuator pressure (110 bar). The flow surplus (30 l/min) is unloaded at this pressure and the power the prime mover is demanded is very limited: 110 • 60/600 • 0.9 = *12.3 kW*, resulting in saving (19.5 – 12.3) 7.2 kW (the relief valve set to 175 bar and placed after the pump prevents the pump from being damaged by the peak pressures that occurs during the transients of the diverter valve).

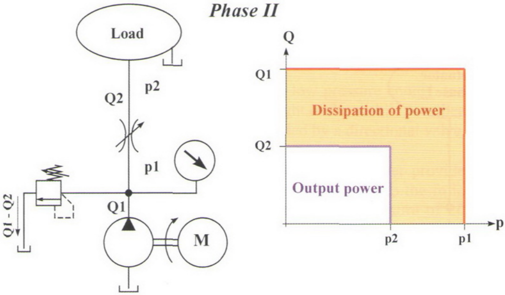


Figure 5.16

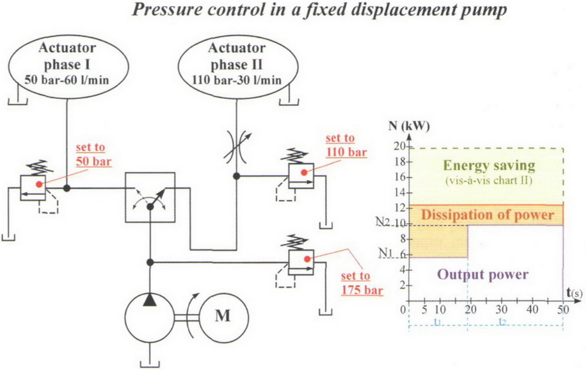


Figure 5.17

Now consider a circuit (Figure 5.17b) managed by a pressure compensator: this systems allows to save more energy than the previous one. We are not going to analyse it because the hydraulic notions and components related to it are dealt with in the following chapters. This circuit can easily be understood after reading such parts.

Fixed displacement pump provided with pressure compensator

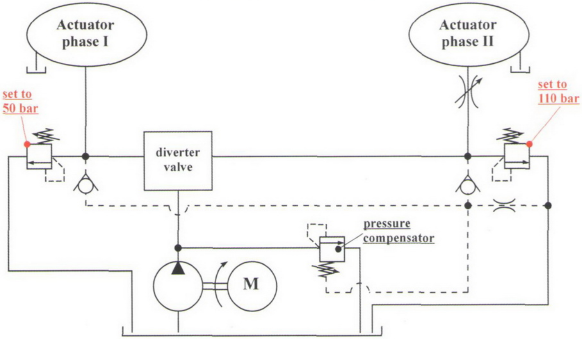


Figure 5.17b

Double fixed displacement pumps for high and low flows

A system with a two-phase cycle in which the two circuits cannot be selected manually like in the previous example demand two coaxial pumps providing respectively a high flow and low pressure and low flow and high pressure. The most popular oil hydraulic application in the manufacturing industry, *the press*, reflects this situation.

Such a machine, especially in its larger versions, needs rather complex control methods and the actuator cylinder must be filled with great accuracy also because of fluid compressibility (see Chapter 2). The working methods of presses are covered later on as while are now going to focus on the operation of the pressing cylinder of a small/medium machine. The double pump is not a peculiarity of presses only, many units need a quick approach to the load condition (high flow – low pressure) and a slow operation (low flow– high pressure) and, in difficult cases, the system requests a two fixed displacement pumps.

The cycle of a press, in respect to only the cylinder for deep drawing, bending or, more in general, deformation (we exclude the retention mechanisms of the mobile crossbeam at rest, its sliding guide during the upward/downward movement, the proportional control of the pressing, safety systems) is defined as follows:

a) Fast downstroke – High flow and low pressure enough to overcome the friction on the sliding guide of the mobile crossbeam. The weight itself moves the mobile part; a substantial flow rate is demanded in order to fill the high-capacity cylinder.
b) Pressing – Low flow and high pressure suitable for the workpiece and its deformation.
c) Fast upstroke – High flow and medium-low pressure enough to overcome the friction on the sliding guide and the weight of the mobile crossbeam.

NB: the downstroke and upstroke are controlled by a directional valve.

The two pumps that are coaxial with the single prime mover provide respectively high flow/low pressure and vice versa. Assuming, in respect to the following Figure, for the first pump a flow of 130 l/min and a maximum pressure of 20 bar (needed especially for the third upstroke phase) and for the second pump a pressure of 250 bar with a flow of 24 l/min, the power each pump demands is (overall efficiency η_g in both pumps is 0.85):

$$N_1 = p_1 \cdot Q_1/600 \cdot \eta_g = 20 \cdot 130/600 \cdot 0.85 = 5 \text{ kW}$$
$$N_2 = p_2 \cdot Q_2/600 \cdot \eta_g = 250 \cdot 24/600 \cdot 0.85 = 11.8 \text{ kW}$$

As either the first or the second pump can be cut out thanks to their valve, the mechanical power on the shaft of the prime mover is equivalent to the maximum power demanded, plus the power the standby pump needs.

Actually, the alternate cut-out system is not much convenient because it demands a more complex circuit and most of all because the high-pressure pump cannot work if there are additional frictions, especially in the upstroke. As a result, a non-return valve β is added in series with the first low-pressure pump in order to prevent the (high-pressure) flow of the second pump from interfering with it (Figure 5.18). Consequently, the two hydraulic generators are operational in this complex cycle:

- Phase I – As the directional valve is switched, the first pump (1) ensures the maximum flow rate, its pressure depends on the limited friction the sliding guide of the mobile crossbeam faces. Also the second pump (2) provides its maximum flow, while pressure is negligible.
- Phase II – The force demanded to deform the workpiece determines the pressure of the second pump; its fluid acts on the non-return valve β: the relief valve of the first pump becomes operational. If the command is delayed (directional valve), the excessive pressure in the circuit of the second pump is reduced by its relief valve.

- Phase III – The reversal of the directional valve causes the first pump to provide the maximum flow and the pressure demanded results from the weight of the mobile crossbeam plus the resistance of the frictions on the sliding guides. This pressure is found on both the pumps.

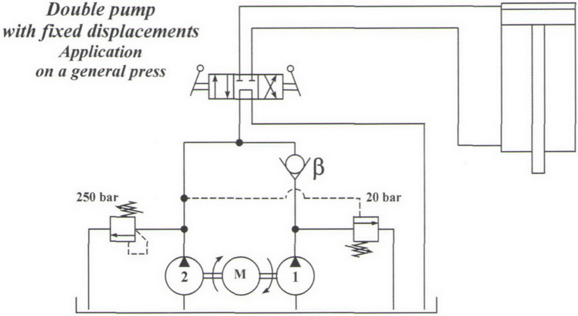


Figure 5.18

As far as relief valves are concerned, it is important to stress the fact that the control of the valve of the low-pressure pump is performed by the line of the second high-pressure pump. Since there is no sensor that determines the beginning of the pressing, it is the increasing pressure on the second pump that causes the first pump to unload. Consequently, the valve of the first pump that in the Figure is set to 20 bar is not an actual relief valve but an *unloading valve* (Chapter 10, § Pressure control valves).

Three or more fixed displacement coaxial pumps

The use of three or more pumps with different flow rates in systems that need multiple flows at a single pressure ensures a wide range of flow rates. The pumps can be coaxial with the single prime mover or they can be operated by multiple motors. The single pump or the two pumps (inactive in some phases) are put on standby via their directional valves 2/2; the non-return valve prevents the flow in the operational pumps from interfering during the standby. Three pumps guarantee seven flow rates, i.e. the three flows of each pump and the sums depending on the position of the valves 2/2. Since the circuit needs a single maximum pressure, a single relief valve is employed (Figure 5.19).

Triple fixed displacement pump

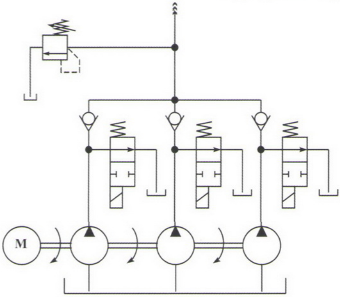


Figure 5.19

Multiple pressure circuit

The circuit often demand different pressures during the various phases of the cycle. In these cases, whether there is a single displacement pump or there are multiple fixed displacement pumps with the same pressure but different flow rates, pressure selection is performed by adding, via the directional valve, one or more relief valves set to the most appropriate level (Figure 5.20).

This method allows to use more preset pressures: it is enough there are as many relief valves (and their directional valves) as there are working pressures. Yet, this makes the system become much more complex and installation costs increase; in addition, if any pressure had to be reset, this operation should be carried out manually by acting on the respective relief valve. Such a structure needs also a system of electric sensors that control the solenoids of the directional valves connected to the various relief valves. Under these conditions, the electromechanical control system is suitable for up to two or three pressures and in any case it is not flexible, with variations depending on the load and possible new pressures.

A single relief valve, this time controlled by an electronic proportional unit, guarantees a wide range of pressures continuously adjusted to the load demands. The electronic unit receives the inputs from the feedback sensors placed on the machine and sends the proportional signal on the control solenoid supplied with the relief valve.

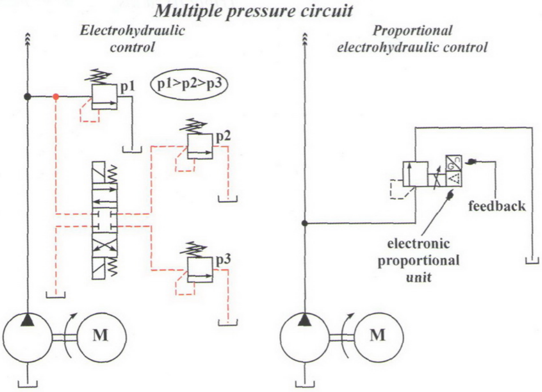


Figure 5.20

Notes on controls for variable displacement pumps

What was previously explained in respect to fixed displacement pump can certainly be applied to variable displacement pumps with better results. Systems requiring multiple pressures and flows are designed without most relief valves with their control components and especially with a single pump. As far as dimensioning is concerned, the circuit is lighter and the higher cost of a variable displacement pump than a fixed displacement version is often counterbalanced and even reduced by the lack of the numerous control components previously analysed.

Nonetheless, also these components of hydraulic generation confront us with some particular issues because flow control is closely linked to the type of the controller. Sometimes the controller cannot ensure the performances needed. It is vital to establish at the very beginning the real requirements of the system, the exact number of the different flow rates and pressures and avoid resorting to complex controllers when the circuit has standard demands. It is wrong to install a Load Sensing pressure limiting compensator in a system that needs only zero flow at a maximum pressure.

As the structure becomes more complex, the compensator must be chosen by taking into consideration its characteristics as well as its drawbacks; in flexible systems it is better to opt for a controller whose standards are higher than those of the employable controller. The choice of the controller is thus affected by the peculiarities of the machine. Proportional changes in flow and pressure need a constant power compensator, whereas a mechanical pressure compensator is suitable for high flow rates with low pressures that alternate with low flows at a single high pressure; finally, cycles with multiple high pressures demand a Load Sensing compensator.

However, all this restricts the system flexibility to a larger or smaller extent, in other words the use of a certain type of controller limits the system performance. The ideal solution is the compensator with closed loop (feedback) electronic proportional control, high or low flow rates, different pressures, constant powers... almost everything is possible, provided the system is equipped with proper sensors. So why are the archaic traditional controllers still taken into consideration if proportional controllers are so efficient? The answer is simple: the cost of their components and, in a number of machines, the unfeasible or inconvenient installation.

Remarks on pressure compensators

A variable displacement pump equipped with a *pressure* controller *mechanically set* via a spring (see the homonymous Figure in the previous paragraph) can deliver the maximum flow at any pressure as long as it does not exceed the level set on the compensator. Once the pressure set on the controller is reached, the flow decreases in proportion to the plate inclination until it equals zero if the pressure keeps on going up. For this reason, low flow rates are affected by the set pressure and they cannot be reduced at a lower pressure. The power absorbed by the prime mover during the operation at maximum flow (pressure below the set value) is equivalent to the power delivered by the load but energy is wasted during the compensation phases.

A difference of pressure Δp arises between the load and the outlet to the pump because of pressure drops in the valves for the control of actuators, hoses, bends, fittings, etc.; this pressure drop is generally depicted as a variable constriction that could be found as an additional speed control in the compensation phase.

The flow rate Q_{max} is constant, although the flow does not diminish very much because of the constriction, until the set pressure p_T before the constriction acts. This pressure arises when the load faces a resistance equal to pressure p_C, so that it generates a constriction difference Δp (p_T - p_C). For instance, if the set pressure is 100 bar (p_T) and the load pressure is 60 bar (p_C), the differential pressure Δp must equal 40 bar so that the controller acts so as to reduce flow (Q_T). Still, since differential pressure depends on the cross-section of the restrictors, it cannot satisfy the compensation condition (the pump maintains the maximum displacement) or exceed it up to the maximum pressure allowed, thus causing the compensator to bring the plate into the zero inclination position. However, it is important to note that the system automatically switches back to lower flows.

The left chart in Figure 5.21 shows that, under compensation conditions, a dissipation of power occurs, as represented by the area Q_T, p_C, p_T.

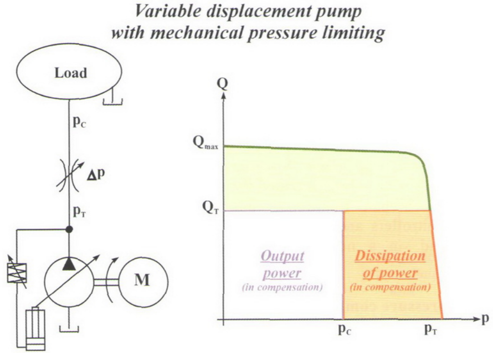


Figure 5.21

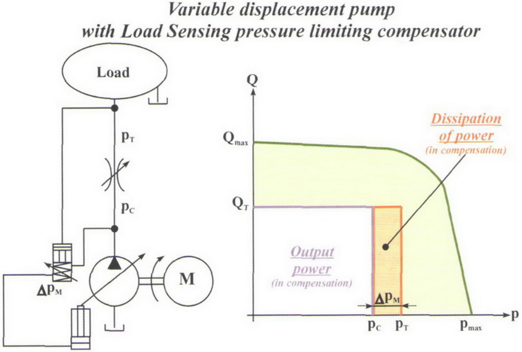


Figure 5.22

The differential pressure Δp, i.e. the difference between the set pressure and the load pressure ($p_T - p_C$), entails a substantial dissipation of power, which is inversely proportional to p_C. In addition, as the flow of a constriction is roughly proportional to the root of the differential pressure, $Q = \sim \sqrt{\Delta p}$, a variation in Δp triggers a flow change that destabilises the system. The irregular trajectory of the curve between Q_{max} and p_T, in other words the area of maximum flow, is due to the pressure drops that occur when pressure increases and approaches the compensation setting.

The use of the *Load Sensing* pressure limiting compensator limits the dissipation of power to a much smaller area. The chart in Figure 5. 22 shows the effectiveness of this device that reduces the dissipation to only the difference Δp_M resulting from the spring.

Combining controls

Further operational and energy advantages result from the combination of controllers with different characteristics.

The combination of the *constant power* compensator *with the Load Sensing pressure limiting compensator* allows to limit the dissipation of energy depending on the load pressure and not, like in standard controls, on the pressure over the outlet to the pump.

Another improvement consists in adding to these components a *proportional valve with electronic control unit* in series with the Load Sensing compensator. In this manner various pressures can be limited in order to respond to the system demands; this device is perfectly suitable for equipment requiring high/low flow phases (deep drawing or scrap presses) in which it is essential to ensure the constant power curve does not vary throughout the cycle.

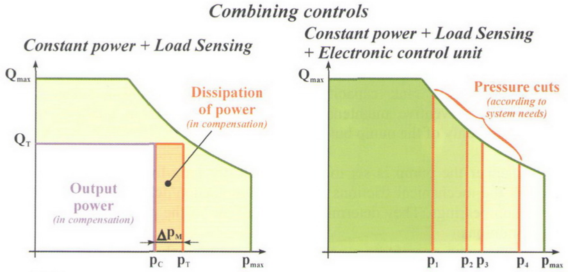


Figure 5.23

Other combinations of control result in special machines, like in the mobile industry the *Load Sensing and torque summation control* or the *Speed Sensing* for the hydrostatic drive of small vehicles.

The latter acts in demanding situations in order to avoid the stalling of the endothermic motor. It is common knowledge that as soon as the maximum power allowed is exceeded a motor tends to slow down and then stops. The setting of the Speed Sensing depends on the maximum rotational speed allowed; this device reduces pump displacement if more power is demanded.

Factors related to pump duration

In theory a pump has an endless working life: the fluid inside it guarantees a constant and abundant lubrication of each component; the film that reaches every moving part prevents a wear and sudden temperature changes. Actually, many factors shorten its life even if the pump works under excellent conditions. And excellent conditions are just a dream more often than not: dirty oil, reduced suction, vibrations, heat, excessive pressures, poor maintenance…

The maximum flow, pressure and rotational speed shorten pump life because pumps must continuously sustain the highest strain they can endure and these premises entail abrupt temperature changes with the ensuing expansion of the components, change in the fluid properties, cavitation or in any case poor suction, viscosity below the limit allowed. Even if these parameters stay within standard limits, it is essential to consider the possible frequent peak pressures, the pressure waves due to fluid hammers and the vibration inside the system.

Also the lubricating, antioxidant properties, as well as the thermal conductivity of the fluid employed, are essential; if liquids other than mineral oil are used, it is worth reaching out to the manufacturer and do not forget that, in comparison to oil, synthetic fluids shorten the working life of component on average by 15%, the water-oil mixtures by 30% and water-glycol mixtures by as much as 50%.

Fluid replenishment, a proper installation, cleaning and replacing filters in compliance with the manufacturer deadlines, suction and return hoses properly dimensioned, suitable tank capacity, inclusion of heat exchanger and vibration dampeners and preventive maintenance, play a central role in guaranteeing along working life not only of the pump but of the whole system.

In this manner the pump is set to be hydraulically long-lived, yet there is still the matter with the mechanical frictions not related to suction/delivery, i.e. the friction of ball and plain bearings. They determine the actual life of the pump, assuming there are optimal conditions.

Even if any pump guarantees a proper lubrication of each sliding part thanks to the internal drainage, ball and plain bearings have an independent life, whose length depends on the design quality and especially on operational conditions.

We have already analysed the remarkable internal hydrostatic balance on most available pumps, but thrusts are transferred onto the casing via the bearings. For this reason, rather high rotational speeds and the axial strains transmitted by the rotary unit needs the dimensioning of the ball (ball and roller) and plain (brass) bearings be established according to the type of pump. For instance, the thrusts on the bearings of an axial piston pump with fixed inclined plate are very different from the thrusts on the same pump with a rotary plate. In the former, the fixed plate is next to the pump casing and the axial stress of pistons is directly transmitted to the casing; the rotary plate needs two massive bearings, usually roller bearings, one inclined and forming a single unit with the plate, the other detached and perpendicular to the drive shaft. The solution for the bent axis pumps is even more complex: most axial and radial thrusts transfer to the shaft and they demand a series of accurately dimensioned bearings.

To sum up, the designer must take into consideration the mechanical laws of the fatigue strain in order to calculate the working life ratio between the hydraulic part and bearings; in this manner, he/she can adopt appropriate measures so that bearings do not wear out too quickly. Since the hydraulic part is not much subjected to wear, the life of a pump is mainly determined by the quality of its bearings.

The illustrative graph in Figure 5.24 shows the ideal operational range of any volumetric pump. The repeated trespassing on other areas results in various strains leading to an early wearing of the moving parts, especially bearings.

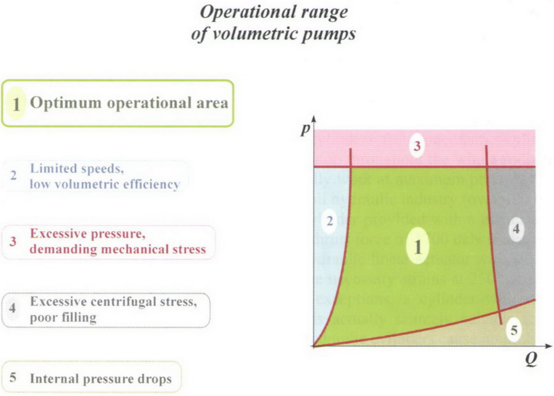


Figure 5.24

Variable displacement piston pump
Electroproportional controls

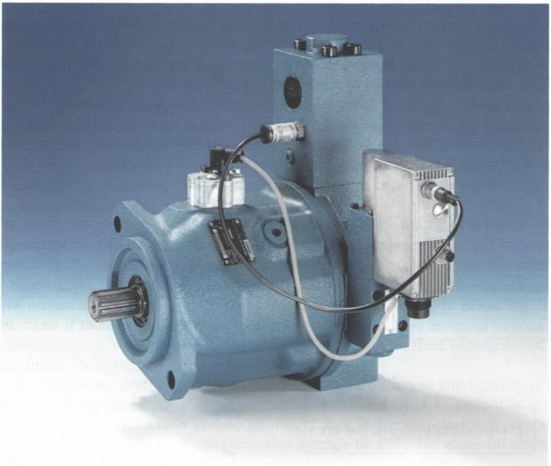


Figure 5.25

Chapter 6

LINEAR AND ROTARY ACTUATORS

The hydraulic energy generated by a pump is transformed into mechanical energy by some parts that have a linear or rotary motion generally referred to as **actuators**. Linear actuators are the different types of **cylinders** (also known as **jacks**), while actuators with a rotary motion are commonly referred to as **motors**.

A third type of special actuators includes parts that do not perform a round angle revolution (less than 360°) and than can have a linear or circular basic operation based on the same principles as the other two types. **Limited rotation** actuators are also known as **rotary** actuators.

This chapter deals with the different versions of cylinders and rotary actuators, while chapter 7 focuses on hydraulic motors.

PRINCIPLES

Two characteristics distinguish oil hydraulic cylinders from pneumatic cylinders: greater forces can be applied and it is possible to stop throughout the whole rod stroke (intermediate positions). Pneumatic cylinders usually work at maximum pressures of 6-7 bar while very high pressures are reached in the oil hydraulic industry (over 400 bar in special applications). For instance, a pneumatic cylinder provided with a piston whose bore Φ is 200 mm at a pressure of 6 bar ensures a thrust force of 1700 daN if losses due to friction account for about 10%, whereas a oil hydraulic linear actuator with the same dimensioning and obviously designed to sustain the necessary strains at 250 bar entails a thrust force of 70600 daN. While, with few exceptions, a cylinder with such a dimensioning in pneumatic power transmission is actually scarcely suitable for the modern applications of this industry, quite larger hydraulic cylinders cause no problems. In pneumatics it is advisable to guarantee on/off handling (i.e. totally extracted or retracted rod) for every type of cylinder, except for special applications we obviously cannot illustrate in this work. Rod intermediate positions are risky and unstable due to the precarious back pressure of the chamber that is not operational. This is a standard process in oil hydraulic cylinders but in this case the fluid is almost incompressible and then it does not enable shifting because of the load. As a matter of fact, speed increases as a body experiences a perpendicular free fall; the back pressure in pneumatics, perhaps accurately calculated under static conditions, cannot be controlled any longer:

the speed increase during the fall inevitably drives the piston to the end-of-travel position. This cannot occur in oil hydraulics because the oil 'blocks' the piston in its position even if actually this principle is not so simple: different kinds of measures must be taken in order to face valve and seal wear, leakages and fluid compressibility, high temperatures and other factors.

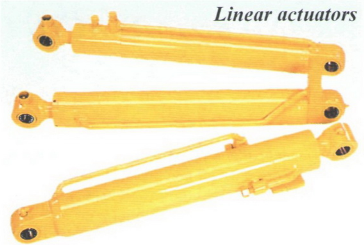


Figure 6.1

Operating principle

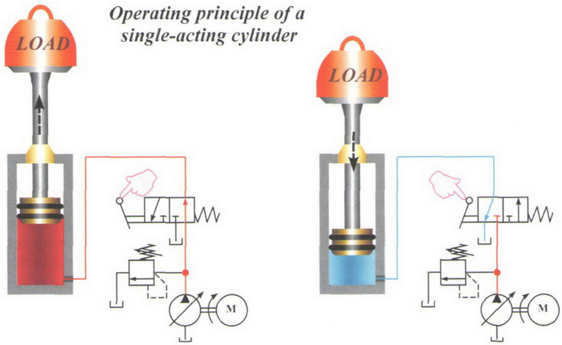


Figure 6.2

A cylinder is fundamentally made up of the followings parts:

- ✓ Body
- ✓ Rod end

- ✓ Cap end
- ✓ Piston
- ✓ Rod
- ✓ Static and dynamic seals

As soon as the fluid touches the flat face of the piston, the resulting pressure overcomes the force opposed by the load and the force of seal internal friction; the piston moves to the end-of-travel position. The rod, solidly connected to the piston, comes out of the rod end, thus performing the translation of the load or the pressing of an object.

The fluid in **single-acting cylinders** (S.A.) flows in and out through the single pipe connected to the end that is opposite to the load; external forces, like load weight or an opposing spring (Figure 6.2), ensure the stop repositioning during extraction or retraction.

The opposite chamber of **double-acting cylinders** (D.A.) needs an outlet for the back pressure developed by the piston stroke that reduces the volume; the fluid is conveyed to the tank through dedicated pipes. The same rules apply to the return to the stop position: pressure on the opposite face of the piston and unloading of the fluid held in the other chamber into the tank (Figure 6.3).

Operating principle of a double-acting cylinder

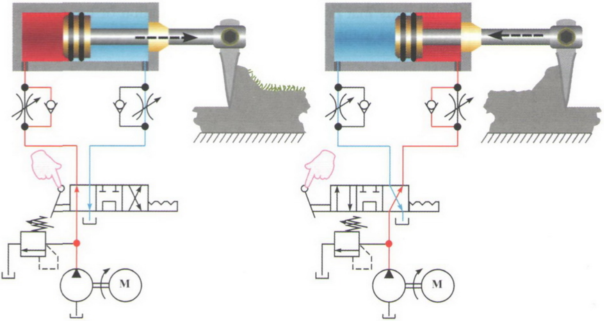


Figure 6.3

Fluid supply and unloading are performed when the dedicated directional valve is operated; the translation speed is controlled by flow control valves or directly by a variable displacement pump equipped with a compensator.

Dynamic seals, assembled on the piston, prevent fluid from leaking from one chamber into the other, while **static seals**, positioned in the interfacing points of the parts (ends-body, ends-fittings) or between the rod-end sliding parts, avoid leakages from the inside to the outside. The function of seals, that is to say preventing leakages, is called **sealing**.

The cylinder **bore** is equivalent to the bore of the body and hence also to the piston diameter plus the play of dynamic seals. In general it refers to the surface of the piston that corresponds to the cylinder bore (piston plus seals).

The ovalisation *tolerances* and the *roughness* inside the body are established by some ISO standards. An H7 ovalisation with a roughness of 0.4 micrometres certainly provides high precision with excellent sliding of sealing parts; an H12 ISO tolerance and a roughness of about 2÷3 micrometres can undermine the actuator efficiency.

The maximum length the rod can travel outside the cylinder is called cylinder **stroke**. The piston surface is measured in mm^2 or cm^2. The bore, the stroke, the diameter of the piston and the rod are measured in millimetres. The dimensioning of a cylinder is usually stated as follows:

S.A. or D.A. Φ… (mm) rod Φ… (mm) stroke… (mm), for instance,

- S.A. (spring) Φ125, rod Φ56, stroke 300 (single-acting cylinder, spring return, body internal bore mm 125, rod diameter 56 mm and rod maximum stroke outside the cylinder mm 300)
- D.A. Φ320, rod Φ144, stroke 1000 (double-acting cylinder, body internal bore mm 320, rod diameter 144 mm and rod maximum stroke outside the cylinder mm 1000)

The different additional parts that promote an optimum operation of the component are analysed in the part devoted to the types of cylinders.

Dimensioning

Since the area of the circle is

$$S = \frac{D^2(\mathrm{mm}) \bullet \pi}{4} (\mathrm{mm}^2) \quad \text{or} \quad S = \frac{D^2(\mathrm{mm}) \bullet \pi}{400} \; (\mathrm{cm}^2),$$

the force exerted in double-acting differential cylinders can be calculated by means of the following formula:

$$\text{Extracting force } p \bullet \frac{D^2 \bullet \pi}{400} \bullet \eta \; (\mathrm{daN})$$

$$\text{Retracting force } \; p \bullet \left(\frac{D^2 \bullet \pi}{400} - \frac{d^2 \bullet \pi}{400}\right) \bullet \eta \; (\mathrm{daN})$$

where D = piston diameter (or bore) d = rod diameter

The retracting formula is valid for both faces in double-acting cylinders whose piston faces have the same area (double rod). What follows is the formula to determine the piston diameter D quickly if its surface S (in cm^2) is known:

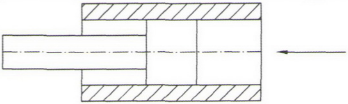
$$D = 11.28 \bullet \sqrt{S} \; (\mathrm{mm})$$

In single-acting cylinders with spring return, the force of the spring at its maximum compression has to be subtracted from the force developed during the positive stroke:

$$\text{S.A. extracting force } p \bullet \frac{D^2 \bullet \pi}{400} \bullet \eta - Fm \quad (\text{daN})$$

where Fm = spring force (daN) at the maximum compression.

Differential cylinders - Theoretical extracting force



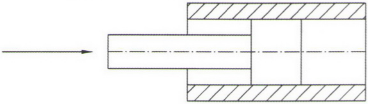
Bore ø (mm)	Piston area (mm²)	Cylinder force in kN						
		10 Bar	40 Bar	63 Bar	100 Bar	125 Bar	160 Bar	210 Bar
25	491	0.5	2.0	3.1	4.9	6.1	7.9	10.3
32	804	0.8	3.2	5.1	8.0	10.1	12.9	16.9
40	1257	1.3	5.0	7.9	12.6	15.7	20.1	26.4
50	1964	2.0	7.9	12.4	19.6	24.6	31.4	41.2
63	3118	3.1	12.5	19.6	31.2	39.0	49.9	65.5
80	5027	5.0	20.1	31.7	50.3	62.8	80.4	105.6
100	7855	7.9	31.4	49.5	78.6	98.2	125.7	165.0
125	12272	12.3	49.1	77.3	122.7	153.4	196.4	257.7
160	20106	20.1	80.4	126.7	201.1	251.3	321.7	422.2
200	31416	31.4	125.7	197.9	314.2	392.7	502.7	659.7

Figure 6.4

Efficiency η varies depending on the diameter Φ of the piston (the larger the diameter is, the more efficiency increases), the type of dynamic seals (O-ring or lip seal) and seal wear. Efficiency η usually ranges between 0.9 and 0.8 (10- 20% loss).

The tables in Figures 6.4 and 6.5 for standard differential cylinders, in compliance with ISO 6020/2, show the approximate extracting and retracting forces (efficiency η is not considered and forces are measured in kN).

Differential cylinders - Theoretical retracting force



Bore ø (mm)	Piston area (mm²)	Cylinder force in kN						
		10 Bar	40 Bar	63 Bar	100 Bar	125 Bar	160 Bar	210 Bar
12	113	0.1	0.5	0.7	1.1	1.4	1.8	2.4
14	154	0.2	0.6	1.0	1.5	1.9	2.5	3.2
18	255	0.3	1.0	1.6	2.6	3.2	4.1	5.4
22	380	0.4	1.5	2.4	3.8	4.8	6.1	8.0
28	616	0.6	2.5	3.9	6.2	7.7	9.9	12.9
36	1018	1.0	4.1	6.4	10.2	12.7	16.3	21.4
45	1591	1.6	6.4	10.0	15.9	19.9	25.5	33.4
56	2463	2.5	9.9	15.6	24.6	30.8	39.4	51.7
70	3849	3.8	15.4	24.2	38.5	48.1	61.6	80.8
90	6363	6.4	25.5	40.1	63.6	79.6	101.8	133.6
110	9505	9.5	38.0	59.9	95.1	118.8	152.1	199.6
140	15396	15.4	61.6	97.0	154.0	192.5	246.3	323.3

Figure 6.5

Rod speed is obtainable from the flow formula:

$$Q = S \cdot v \quad \rightarrow \quad v = Q / S;$$

Since in oil hydraulics flow is measured in dm^3/min, section in cm^2 and speeds in metres per minute, the formula is:

$$v = \frac{dm^3 \cdot 10^{-3} / \min}{cm^2 \cdot 10^{-4}} = m/\min$$

SINGLE-ACTING CYLINDERS

As we have already said, the pressurised fluid in single-acting cylinders acts on only one flat face of the piston. The following paragraphs deals with additional components like rod bearings, rod driver, dust scraper, cushions, etc., which are similar to those of double-acting cylinders.

Ram cylinder

It is the flat face of the rod itself that replaces the piston in this component and it is usually placed vertically or in other positions in which load weight drives it to the stop position (Figure 6.6).

Operating principle of a ram cylinder

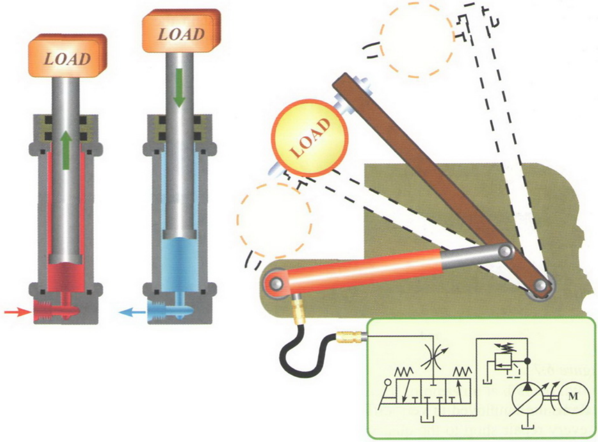


Figure 6.6

In the best versions, the diameter of the lower end of the piston is larger by a few millimetres in order to prevent it from pulling out at the upper limit; still, this causes a hydraulic cushioning in the final part of the stroke promoting the slowdown of the rod and also avoiding harmful recoil on the upper end. Proper seals inside the upper end prevent fluid external leakage.

No dynamic seals can be put on the larger end of the rod that serves as a piston; in order to improve rod stiffness, slide rings are often mounted so that oil flows through some clearances below. However, this makes for an increase in costs because the body bore must have little roughness.

Nevertheless, the manufacturing process of the ram cylinder described so far is expensive and it also forces manufacturers to design most of these components in a simpler manner, that is without the larger rod end; as a matter of fact, it is often replaced by a simple seeger ring with simple emergency functions. An o-ring seal placed in its seat and machined on the rod a few centimetres from the limit, for the reasons stated above not fitting closely to the chamber or even with a slightly smaller external diameter, optimises the actuator efficiency.

The limit stop is ensured by a type of external mechanical stop, like a tie rod, a surface plane or any other detail solidly connected to the machine; in order to prevent the rod from pulling out, it must be fixed, able to sustain the actuator force and to guarantee maximum safety also during maintenance (Figure 6.7).

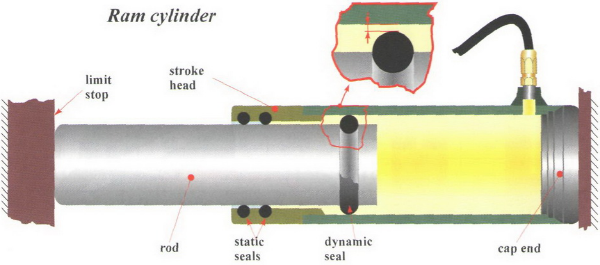


Figure 6.7

Rams are employed in very different industries, ranging from simple car jacks found in every repair shop to the quarrying of dimension stones in marble quarries, from the operation of many actuators in the mobile industry to the lifting of rides in funfairs and amusement parks.

Despite its large diameter, the rod cannot be subjected to strong combined compressive and bending stress (radial loads) because the friction due to tangential actions would undermine its alignment since it rests only on the rod head. Besides jacks for simple applications, there are several ram cylinders whose strokes are quite long, usually swinging on a rear hinge. A special version is equipped with a return spring in order to compensate for the lack of return of the load.

Extending or retracting cylinders

The return of these cylinders is always triggered by the load external force. Unlike ram cylinders, they are provided with a piston with dynamic seals that is dimensioned depending on the load to translate; as a result, they can develop considerable forces.

Even if they are used during extension or retraction, which means they are mainly subjected to axial loads, they can sustain possible radial loads on the rod fairly well; the rod is held by the bush inside the front end and by the piston, which can be provided with a sliding seal.

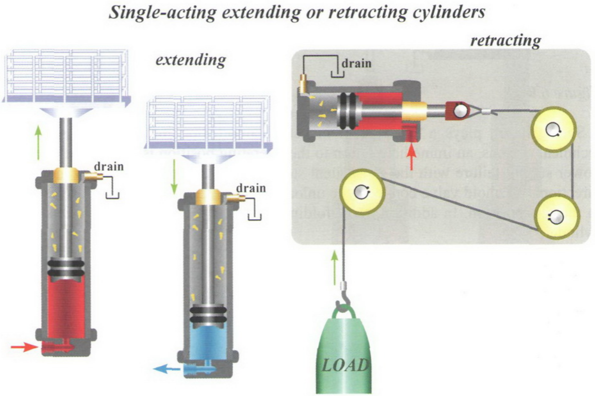


Figure 6.8

The fluid leaked into the chamber that is not subjected to pressure is drained into the tank through a tiny pipe (Figure 6.8). Cylinders can be supplied with additional components, like a cushion, an air bleed and so on.

Spring return cylinders

Spring return cylinders have the same characteristics as the previous types of cylinders, but they are employed when the return stroke cannot be triggered by the load. Springs must be dimensioned with light thrusts in order to avoid energy waste during the work stroke. In addition, it must be stressed that rod stoke is reduced because the retracted string occupies some space in the chamber (Figure 6.9).

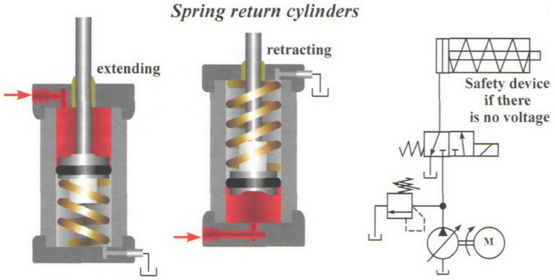


Figure 6.9

Spring return plays a key role in industrial automation in which, for security or technical reasons, an immediate return to the operating position is vital in the event of a power supply failure with the subsequent stop of the prime mover: the 3/3 monostable directional solenoid valve connects the unloading cylinder and the spring drives the rod to the rest position. In addition, most folding presses in repair shops have this type of cylinder.

Very small linear actuators are needed in order to clamp vices, to block slides or rotary tables, for pressing or punching, cutting or bending (Figure 6.10). Single-acting or spring return cylinders with pistons having a bore ranging from 12 to 40 mm and strokes between 2 and 25 mm are available on the market.

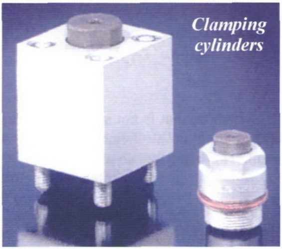


Figure 6.10

Telescopic cylinder

Telescopic cylinders are made up of a system of pipes (two to five depending on the stroke demanded) sliding one within the other; the end pipe can be hollow or solid according to the diameter. An inward margin in the larger body blocks the smaller one, which has an outward margin at the limit stop (like in ram cylinders). The seals placed on the top of each cylinder avoid external leakages (Figure 6.11).

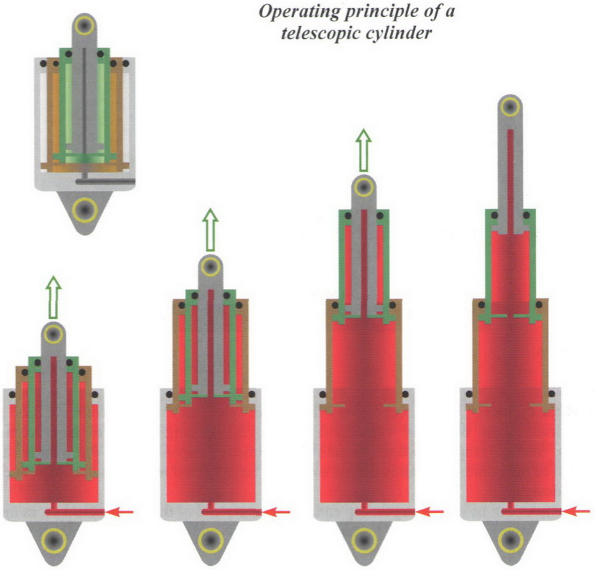


Figure 6.11

When the actuator is totally retracted, the fluid develops a pressure on the whole flat surface of all the sliding parts and then the larger body slides out; when the body is totally extracted, the effective surface of the pistons diminishes and pressure rises in order to overcome the load force: this occurs for all the pipes until the last and smallest rod slides out. If the power supply is constant, speed increases progressively during the sliding because cylinders are arranged in decreasing order of internal capacity. The first rod that retracts is the smallest, progressively followed by the others up to the rod of the cylinder that has the largest diameter

Figure 6.12

These cylinders, which occupy very little space when they are retracted, are used for platforms, dump bodies of vehicles (Figure 6.12), opening/closing of large vertical doors. Strokes range between one and fifteen metres depending on the number and length of the bodies. Actuators cannot evidently be subjected to high radial stress due to their design.

The inlet can be on the largest cap end, at the upper end of the smallest rod or, in most cases, on the swinging pin. The guide rings on the ends of each sliding part ensure precise body sliding; a rod wipes is mounted on every upper ring and some stop rings avoid the impact with the lower cap end when every part is retracted, which could cause burrs and deformations on the cylinder flat surfaces (Figure 6.13).

A fair number of people believe these actuators are widely employed for lifts. Actually, except for few versions, which by the way require complex electroproportional servo-systems, lifts are equipped with very long rams whose bodies and rod are made up of many parts; pulleys for steels ropes are put on the top of the rod in order to double the stroke.

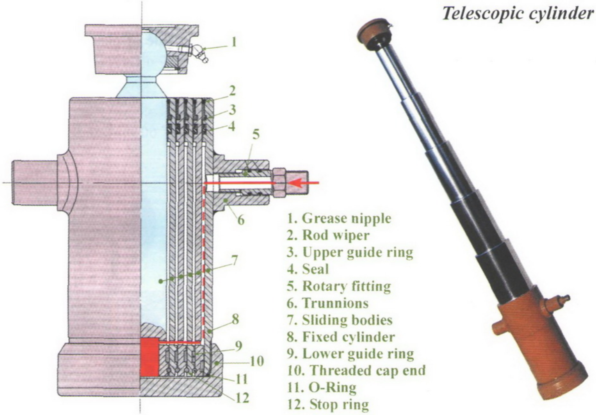


Figure 6.13

Telescopic cylinder can become double-acting with some modifications.

DOUBLE-ACTING CYLINDERS

In the different versions of double-acting actuators, one of or both effective surfaces of the piston cannot be exploited fully in terms of hydraulics. In differential cylinders and the resulting cylinders (tandem), a piston face is solid, while the other is reduced by the presence of the rod; the rod is connected to both piston surfaces in double rod cylinders.

Differential cylinder

Differential cylinders are the most widespread linear actuators. Since they can be controlled in both directions, they ensure high reliability and safety standards. They are called 'differential cylinders' because the active section is limited by the space occupied by the rod itself in the part of the piston axially connected to the rod (Figure 6.14). The parts of a differential cylinder, like rod wiper, seals, etc., are dealt with in the following paragraph.
In differential cylinders, the effective surface S_d of the pulling piston is reduced by the presence of the rod (Figure 6.15).

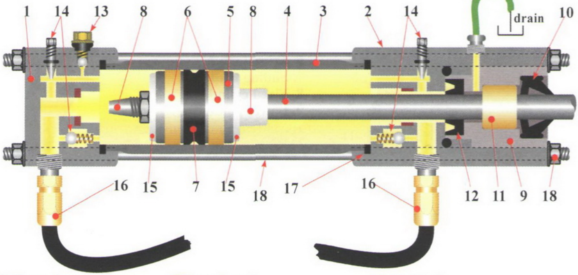
By calculating the differential ratio R_S of the cap end side/rod end side (annular) piston areas, the difference between pressures, speeds and extracting and retracting volumes can be determined easily (upper-case D refers to the cap end side of piston and lower-case d to the rod end side).

$$R_S = S_D / S_d$$

Assuming a cylinder with cap end side or full piston area of 60 cm^2 and whose rod has an annular area of 40 cm^2, its differential ratio is:

$$R_S = 60 / 40 = 1.5$$

Differential cylinder



1. Cap end
2. Rod end
3. Body
4. Rod
5. Piston
6. Piston ring
7.Dynamic seal
8. Braking nose
9. Rod cartridge
10. Rod wiper
11. Rod bearing
12. Sealing ring
13. Air bleed
14. Damper unit
15. Bump stop
16. Inlet
17. Sealing ring
18. Tie rod

Figure 6.14

If the load requires F = 8000 daN during both extraction and retraction, extracting pressure p_D is:

$$p_D = F(daN)/S(cm^2) = 8000/60 = 133 \text{ bar}$$

Retracting pressure p_d is:

$$p_d = p_D \bullet R_S = 133 \bullet 1.5 = 199.5 \text{ bar.}$$

Assuming flow rates are equal and the extracting translation speed v_D is 2 m/min, retracting speed v_d is:

$$v_D \bullet R_S = 2 \bullet 1.5 = 3 \text{ m/min ;}$$

a flow controller is needed if the same translation speed has to be guaranteed both during extraction and retraction.

Given a volume V_D of 4.5 dm^3, the effective capacity during retraction is:

$$V_D / R_S = 4.5 / 1.5 = 3 \text{ litres}$$

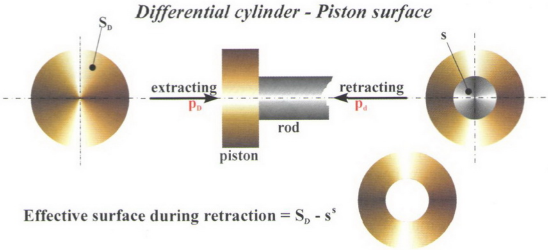


Figure 6.15

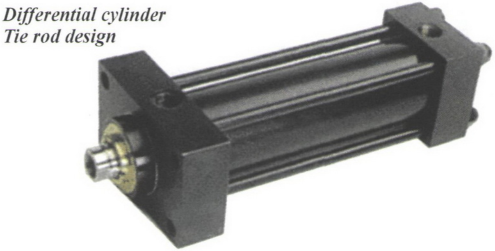


Figure 6.16

Double rod cylinder

Double rod cylinders are similar to differential cylinders, but both their ends have the rod out. Indeed, a rod is solidly connected to both piston faces (Figure 6.17).
Unlike standard double-acting cylinders, double rod cylinders are not differential because active surfaces are the same, so extracting and retracting forces are equivalent, even if rod diameters can be different in some particular cases.

In the first case, this linear actuator can ensure the same extracting and retracting speed without affecting flow control.

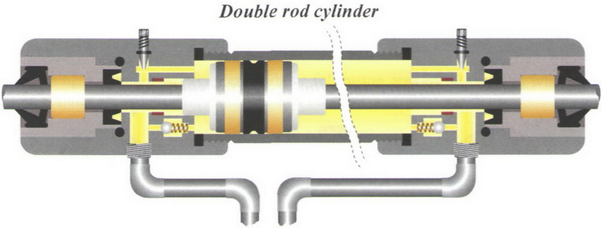


Figure 6.17

Double rods allow the use of high radial loads acting on them. The radial load acting on the single rod bearing of a double acting single rod cylinder (differential) tends to misalign it; the ensuing wear on the margin causes an intolerable rod clearance.

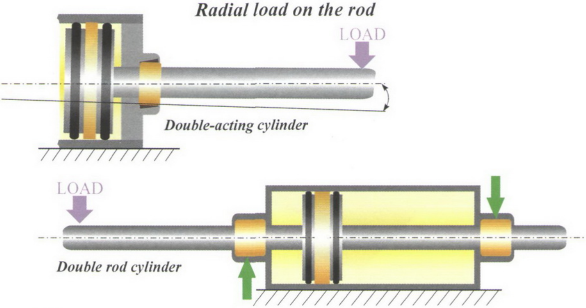


Figure 6.18

The piston itself shifts to an inclined position, quickly wearing out dynamic seals, the piston guide ring and the body. Note that the piston guide ring (or rings) counterbalances the tangential action; double rods are suitable for small cylinders at the most without the piston guide ring or under extreme operating conditions.

Figure 6.18 shows the wear of the rod bearing of a double-acting cylinder above and the correct application of a double rod below.

Double rod cylinders are also employed in automations in which position sensors cannot be placed along the active rod (Figure 6.19).

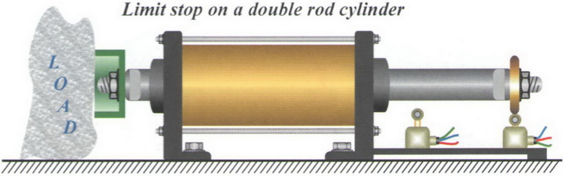


Figure 6.19

Anyway, these actuators challenge designers. Indeed, as one of the rods is used only for supporting or signalling, it can slide out of the operating area, thus becoming a source of danger.

They are widely used as actuators of the hydrostatic power steering in self-propelled machines. The special valve conveys the fluid to the cylinder that is mechanically connected to the steering tyres depending on the position of the steering wheel.

Tandem cylinder

A tandem cylinder gives substantial forces where there is limited radial space for the arrangement of the component. It is practically made up of a chamber divided by a plate inside which two pistons are axially connected to a common rod. In this manner the force developed is almost double and it is equal to the force of a single cylinder whose section is twice as large; this is obviously to the detriment of a longer length (Figure 6.20).

The movement is affected by the fluid flowing in from the two inlets of the pistons; force can be reduced by limiting the pressure of one of them. It is enough to pressurise only one inlet for the return if the tandem cylinder is free or not subjected to high forces.

As the pushing force is double, the external rod is not proportioned to its piston but it must be suitable for the overall work. A variation of the previous tandem consists in making the two **rods independent** (Figure 6.20) so that the device can carry out several tasks:

- ✓ Operation like the previous cylinder.
- ✓ Operation of the cylinder with the external rod only (the other cylinder is inactive).
- ✓ Operation of the cylinder with the internal rod only (the external rod moves because it is dragged; actually it is inactive).
- ✓ Operation of either of them with different pressures (preset pressures so that there is no need to act on pressure control each time) and consequently different speeds.

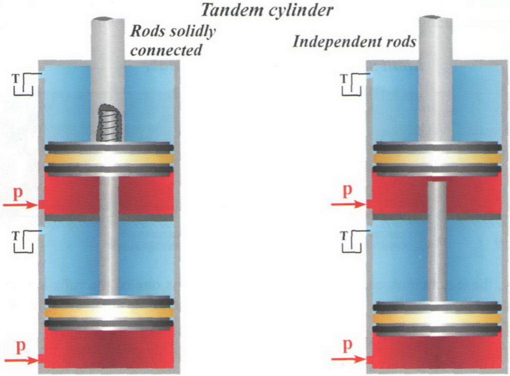


Figure 6.20

Multi-position linear actuators

Three different **positions** can be obtained in the tandem version with independent rods, provided that the stroke of the rods is not the same (Figure 6.21).

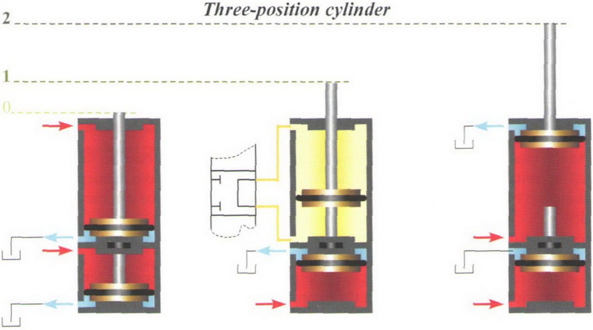


Figure 6.21

In order to have **four precise positions**, two differential cylinders can be firmly added on the cap ends; the rod of one of them is fixed whereas the cylinders must be able to slide freely (Figure 6.22). Equal strokes result in three positions, however the tandem version described above is more suitable for this case; different strokes generate four positions.

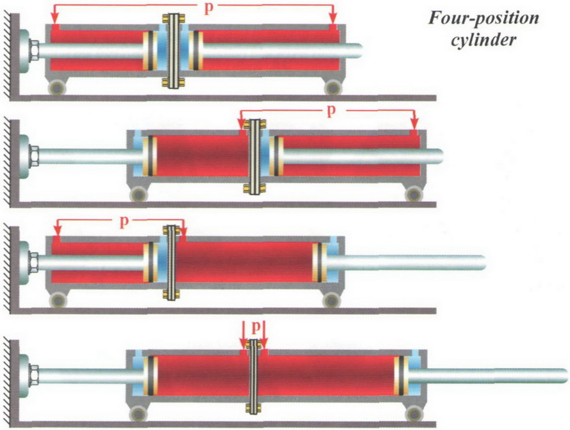


Figure 6.22

As countless strokes can be obtained, it is difficult to find these devices directly on the market: as a matter of fact, after calculating their strokes carefully, they are usually assembled in the business that is going to use them.

Tandem actuators with independent rods, four-position actuators and other devices resulting from the assembly of multiple cylinders are quite unusual for oil hydraulic applications because, unlike pneumatic power transmission, different lengths are possible with a single cylinder by stopping the rod at any point of its stroke. Multi-position actuators are indispensable only if position detectors (feedback, sensors, electromechanical or hydraulic limit switches) cannot be arranged on the machine.

PARTS OF A CYLINDER

The following components and remarks refer to differential cylinders for simplicity reasons, but they apply to most types of cylinders as well.

Materials and assembly

The different parts of cylinders are made of adequate metals, usually cast iron and steel, depending on whether they are used for light or heavy applications. The rod must be chosen by taking into consideration not only the torque and the bending moment, but also the humidity of the environment, acid resistance, etc.; standard rods are made of chromed or stainless steel (Figure 6.23).

Figure 6.23

The assembly between the ends and the body is usually the result of:

- ✓ *welding*: the body extremities are directly welded to the corresponding ends;
- ✓ *tightening*: the body, whose extremities are male threaded, is screwed to the female threaded ends; a sealing o-ring is placed at the internal bottom of the thread of the ends;
- ✓ *tie rods*: four tie rods arranged through the two ends fix them to the body thanks to eight nuts with their lock washers, self-locking nuts or similar components; there is a sealing o-ring like in the previous system.

The table in Figure 6.24 shows the characteristics of the cylinders complying with ISO standard 6020/2.

Double acting cylinders – Design characteristics

REFERENCE STANDARD: *Cylinders in accordance with the ISO standard 6020/2*
MAXIMUM WORKING PRESSURE: *210 bar (21 MPa)*
STANDARD WORKING TEMPERATURE RANGE: *-20°C ÷ +80°C*
SEALS: *easily available seals are used in the following versions for piston and rod sealing:* *-Low friction series consisting of a PTFE ring, for operating speeds up to 1 m/sec, temperature from –20°C to +80°C, compatible with mineral oil and water/glycol mixtures.* *-Series with lip seals consisting of an elastomer and PTFE ring, for operation without leakages under static conditions and the same characteristics of speed, temperature and compatible fluids.* *The series described above can be made of Viton for temperatures up to +150°C, compatible with oils containing few phosphoric esters.*
ASSEMBLY: *cylinders result from the assembly of standard parts, fixed with four tie rods and without welding.*
ROD: *lapped, ground, peeled, cold hot-rolled, steel precision bars with the following characteristics:* *CHROME LAYER: 5/100 of the Ø* *HARDNESS OF THE CHROME LAYER: HV>850 HRC=66-68* *SURFACE ROUGHNESS: 0.10/0.25MY Ra* *TOLERANCE: ISO f7* *STRAIGHTNESS: 1mm : 1000mm* *MATERIAL: 38NCD4 EN 10083/1*
ROD CARTRIDGE: *made of cast iron G25 ISO185*
COMPONENTS (ends, piston,...): *made of steel C40 EN 10083/1*
BODY: *cold-drawn high-precision pipes with the following characteristics:* *INTERNAL SURFACE FINISH: max Ra 0.4my – TOLERANCE: ISO H8* *EXTERNAL SURFACE FINISH: according to the DIN 2391 standard* *STRAIGHTNESS: 1mm : 1000mm* *MATERIAL: St52 DIN 2391*
TIE RODS: *made of steel 39NCD3 EN 10083/1*

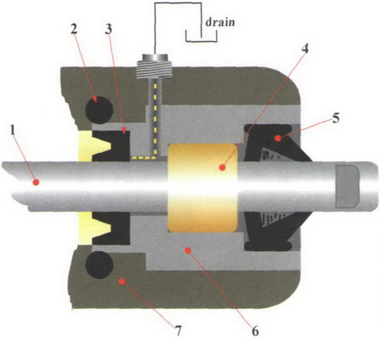
Figure 6.24

Rod cartridge

The **rod bearing**, coupled between the rod end and the rod, prevents friction, hence the wear of these two parts. It is the rod bearing that is subjected to wear because it is made of a softer material than the rod. Another task of the rod bearing is to sustain the transversal stress external forces exert on the rod. In standard versions it is placed in the **rod cartridge**, which is actually an extension of the end that holds its whole length.

The **rod wiper**, positioned after the rod bearing between the end and the rod external part, removes dirt due to the manufacturing process from the rod when it returns. If there was no rod wiper, dirt would not only contaminate the fluid, but also enter the cylinder and reach the piston dynamic seals thus damaging the system seriously. One or more static seals between the end and the rod (rod seals) prevent fluid leakages (Figure 6.25).

Rod cartridge



1. Rod
2. Cartridge-end seal
3. Rod seal
4. Rod bearing
5. Rod wiper
6. Cartridge
7. End

Figure 6.25

As far as these parts are concerned, it is vital to stress that the design described so far does not apply to all systems. In many versions the clearance inside the rod cartridge, properly machined and without the rod bearing, holds the rod directly and the seat for the rod wiper is at its end. The **drain** is between the end and the rod cartridge; it is used if the rod seal does not seal enough.

Air bleed

Air in an oil hydraulic circuit causes not only noise, vibration and poor suction, but also irregular strokes in the actuators due to high compressibility. Air removal must be performed before the system is operated and at regular intervals during its operation according to the system needs.

The air in the cylinders can be removed simply by loosening pipe fittings slightly. Since they are faced downward, this operation is rather unfeasible, consequently they have to be equipped with a manual or automatic air bleed. Manual air bleeds are made up of a ball held with a screw and positioned in a tiny clearance connected to the cylinder chamber (Figure 6.26).

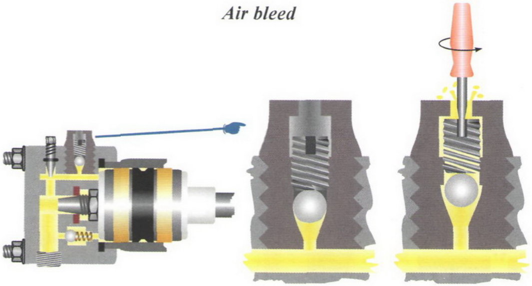


Figure 6.26

Air bleeding is performed as follows:

1) The relief valve setting is reduced to the minimum level.
2) The prime mover is started.
3) The directional valve is positioned so that the piston moves to the limit stop position.
4) The air bleed screw is loosened (one or two turns). It is important to avoid removing it from its seat.
5) The screw is repositioned as soon as the fluid flowing out has no air bubbles.
6) The relief valve is set again to the standard working level.

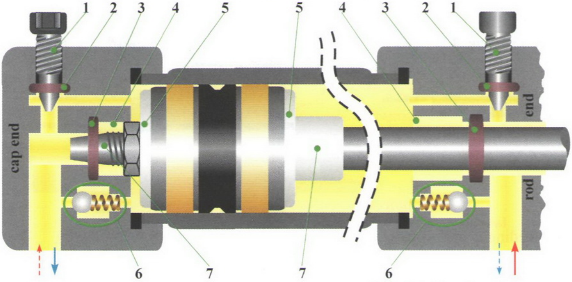
Automatic air bleeds too are available on the market (Figure 6.27); it is essential to check them periodically following the manufacturer instructions.

Figure 6.27

Cushion

The shocks between the end and the piston at the end of each stroke reduce the working life, cause an irritating noise and can damage the workpiece. Cylinders can be equipped with cushions to prevent these shocks (Figure 6.28).

Cushion with needle valve



1. Adjustable needle valve
2. Sealing ring
3. Floating and self-centring bush, made of bronze or other materials
4. Cushioning chamber
5. Bump stop
6. Non-return valve
7. Braking nose

Figure 6.28

At the end of the return stroke, the conical nose at the end of the rod (7), slides into the left cushioning chamber (4), blocking the main outlet. The adjustable restrictor (1) is connected to the cylinder chamber through a conduit. The outgoing fluid, which cannot flow into the main outlet, flows into this tiny channel. As a result, the piston slides towards the end gently.

This occurs also during the forward stroke, but in this case it is the opposite bush that blocks the right channel. When the cushioning chamber is closed, the non-return valve (6), also known as check valve, promotes the quick motion of the piston. As a matter of fact, the fluid would act only on the surface of the conical nose; upon the opening of the valve, pressure can act directly on the piston. A ring (5) known as **bump stop** cushions the shocks of the piston on the end when the cylinder has no cushions or if its restriction is rather limited. Designers conceived various cushions with different operating principles. We are going to analyse the most popular versions.

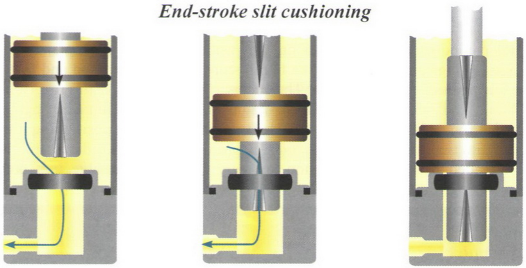


Figure 6.29

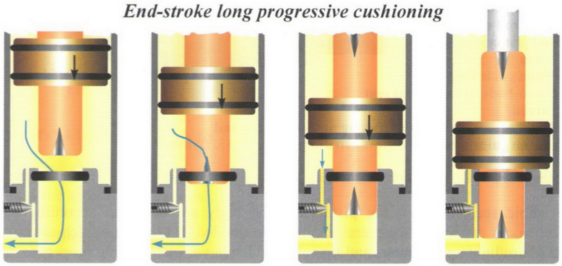


Figure 6.30

In the **slit** cushioning (Figure 6.29), the part of the rod coming out of the piston towards the end built-up with a cylindrical nose, usually made of a softer material than the rod. The rod slowdown cannot be adjusted manually but it is triggered by the nose slit (also the other nose obviously has the slit).

The fluid subjected to back pressure progressively flows through the slit, which has to be dimensioned according to the expected speed and to extend beyond the ring seal in order to avoid leakages when the piston starts shifting.

End-stroke **long progressive** cushioning (Figure 6.30) is a mix of the previous systems. In the first phase the gradual slowdown depends on the slit and then on restrictor valve setting.

These cushions too need the non-return valve.

Piston

Several components are arranged on the piston: single or multiple seals, piston guide ring or rings and sometimes the bump stops already described and that are not always found in standard versions. The rod is rigidly tightened to the piston (male thread on the rod – female thread inside the piston) and fixed with a nut tightened on the threaded rod end (Figure 6.31). In order to promote the seal positioning, the piston is divided into two parts in some versions: a part on the smooth rod end and the other on the threaded part. Rod and piston are very rarely in the same unit or welded together.

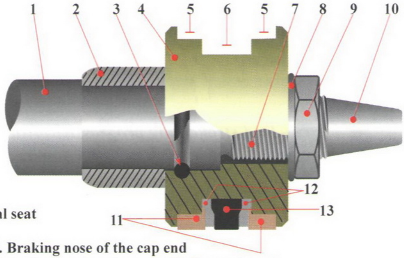


Figure 6.31

Dynamic seals between the piston and the body are designed according to the performances expected. In general they can be divided in **O-rings**, which are suitable for medium/light applications, and **lip seals**, which have better sealing properties due to the larger surface in contact with the body and the tangential forces the pressurised fluid exerts on the opposite wall. Dynamic seals are often backed up by PTFE (Teflon) **back-up** or anti-extrusion rings that ensure maximum stability by pressing the seal between the sides of the piston recess (seal seat).

An elastic ring enhancing the seal pressing on the body wall is arranged inside some seals in the lower part of the piston recess. Some cylinders with a medium/large diameter and subjected to high pressures can have multiple sealing rings on each side known as **packing sets or chevron seals**.

Guide rings or **wear bands** allow the piston to slide on the body perfectly and they also counterbalance the radial thrusts on the rod due to the transverse load (see chapter 18 for more details on seals).

Figure 6.32 shows some examples of assembly of sealing rings and wear bands on the piston.

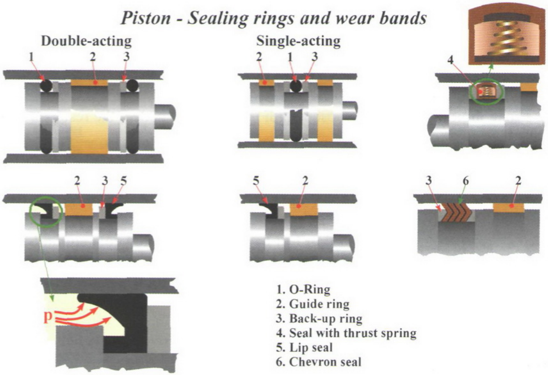


Figure 6.32

ADDITIONAL COMPONENTS

Besides the components described so far, cylinders can have additional parts that are vital in some special applications.

Stroke limiting stop tube

If the forward stroke has to be limited, a bushing with the same length as the expected reduction is arranged on the rod. Stroke limiting is essential in long-stroke linear actuators (many manufacturers recommend using it for medium-sized cylinder over 1000 mm). The rod, whose tip is subjected to a substantial radial stress, transmits and amplifies the strain on the rod bearing held in the rod cartridge and on the wear band (we have already mentioned this in the paragraph on double rod cylinders). This is what typically occurs in a (first class) lever whose fulcrum is on the opposite site of the input. A stroke limiting stop tube makes the fulcrum move toward the centre and it prevents the misalignment and early wear of the rod bearing and the wear band by reducing the amplification of the strain on the rings (Figure 6.33). As a result, the cylinder stroke must be determined by taking into consideration the length that actuation demands plus the tube length.

For instance, if there is no stroke limiting stop tube (Figure 6. 33 – left), the rod travels a stroke (a) of 1 m and the distance between the barycentre of the rings (b) is 20 mm; if the radial force F on the tip equals 10000 N at the end of the stroke, the resultant R is (the rod cartridge length is not taken into account):

$$F : R = b : a \quad \rightarrow \quad R = \frac{F \cdot a}{b} = \frac{10000 \cdot 1000}{20} = 500000\ \text{N}$$

If a stroke limiting stop tube whose length is 200 mm is added, the resultant R is (c = tube length + a):

$$R = \frac{F \cdot a}{c} = \frac{10000 \cdot 1000}{(200 + 20)} = 45455\ \text{N}$$

The overall stroke of the cylinder is:

actuation stroke (a) + tube length= 1000 + 200 = 1200 mm.

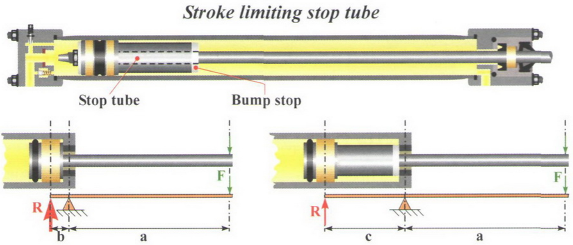


Figure 6.33

Stroke limiting stop valve

The return stroke can be limited by a simple valve made up of a spool acting on the outlet of the cap end; the rod operates a tie rod solidly connected to the spool during the return stroke (Figure 6.34). The spool blocks the clearance gradually, which results first in braking and then in blocking fluid unloading in the final position; this stops the piston depending on the setting of the ring on the rod.

During the forward stroke, the pressurised fluid in the cap end inlet pushes the spool into the opposite direction and so it can finally flow to the piston. The opposing spring simply keeps the valve in a neutral position during the forward/return phases.

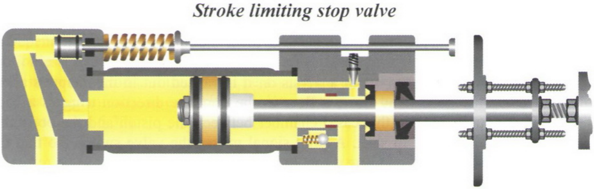


Figure 6.34

Stroke limiting stop valves are employed just in few manually controlled systems for which electricity is not suitable. Actually, electromechanical or electroproportional devices ensure stroke limiting in most systems: reed sensors, inductive sensors or transducers signal stroke limiting.

Pilot-controlled check cartridge valve

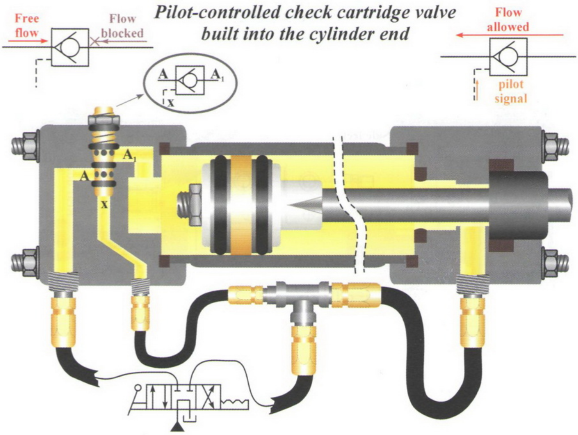


Figure 6.35

The rod must often be blocked in a point of the forward or return stroke for some time. If the directional valve is positioned on closed centres, the inevitable leakages of the valve itself hamper the stability of the rod position. This is what happens in a pressing or soldering system demanding long stops with the pump on stand-by and the rod pressing.

The non-return valve (an in-depth analysis on it is carried out in the chapters about valves) allows only a one-way flow; a flow in the opposite direction triggers a back pressure in the (unloading in this case) chamber between the piston and the body that does not enable the piston to shift. A pilot signal makes the valve switch (so that the fluid can flow also into the opposite direction) as soon as the stroke has to be resumed. This signal in cylinders is usually obtained from the opposite outlet: when the directional valve is switched, the flow acts simultaneously on the valve control (opening) and on the opposite face of the piston.

The controlled non-return valve in the cartridge-type design is positioned in a seat on the cylinder ends (Figure 6.35); if both directions are blocked, both ends have a non-return valve.

Cylinders fastening

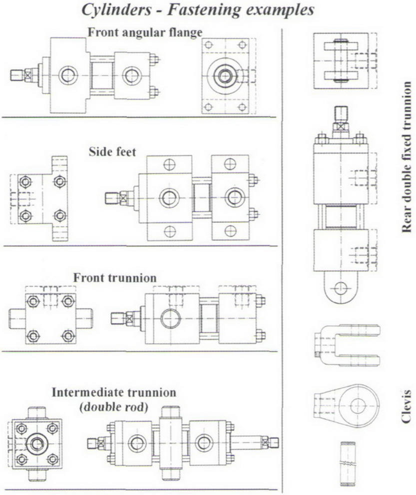


Figure 6.36

The clamps that fasten cylinders to the external fixed or moving parts are dimensioned according to the actuator size and the ideal alignment with it.

Depending on the operation involved, clamps must allow the cylinder to stay motionless, to rotate around its centre or an end, to block the rod in order to make the body move. The rod end thread can be assembled directly to the load or it is possible to connect mechanically an extension, a joint, a clevis or a ball clevis, always on the thread (Figures 6.36 e 6.37).

Cylinder fasteners and joints

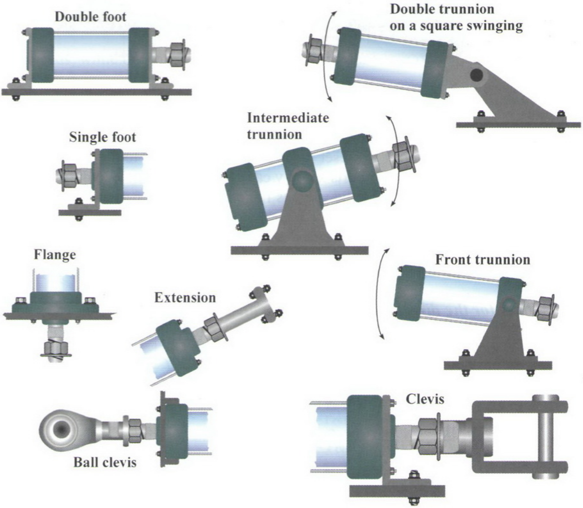


Figure 6.37

Hydraulic fittings

The standard fitting between the directional valve and the cylinder is made up of two rigid pipes or hoses between their ports; other valves like flow control valves, relief valves, non-return valves, etc., if necessary, are added on the valve ports, the cylinder ports or along the piping on a dedicated block according to the accessibility of the cylinder. In many applications the two pipes are attached to a block solidly connected to the cylinder end (Figure 6.38).

Cylinder - Hydraulic fitting on a plate

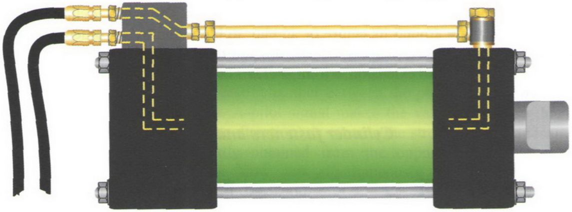


Figure 6.38

The directional valve can even be directly coupled on the cylinder end in some dedicated versions. The block is preset for the direct connection to the pressurised pipe P and unloading pipe T; the two ports x and y can be used for remote hydraulic piloting. The mechanical valve or the solenoid valve is fixed on the upper part of the block.

The drawing in Figure 6.39 shows the plate with the lateral face of connection to the pipes in the upper part (a) and the assembly surface of the directional valve in the lower part (b).

Plate for the direct connection to the directional valve

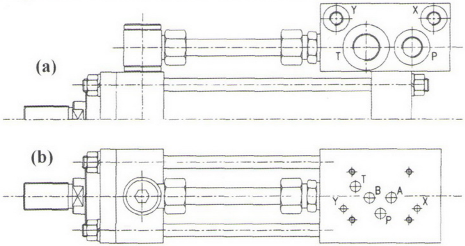


Figure 6.39

CHOOSING THE ROD

We already stressed many times the fact that the rod of any cylinder is subjected to strong strain during its stroke. Such a strain undermines the structure and causes an early wear of bushes and seals with the ensuing increase in fluid leakages.

Combined bending and compressive stress

The composite stress acting on the rod is referred to as 'combined bending and compressive stress'. Manufacturers design rod dimensioning by taking into consideration the highest tolerable combined stress depending on the maximum force the piston exerts at the maximum pressure. Besides the other characteristics of the cylinders, catalogues often include tables or nomograms stating the maximum combined stress each component can sustain. Combined bending and compressive stress load is proportional to the rod stroke: diameters being equal, the longer the stroke is, the less the cylinder can sustain the combined stress.

Another key factor that undermines the cylinder ability to sustain the combined stress is the type of cylinder fastening. The method that causes the lowest strain is an anchor on both ends, while the worst technique is the rear trunnion or only the front flange.

Rod dimensioning

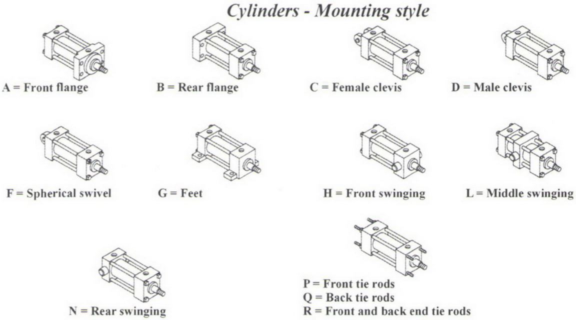


Figure 6.40

The following method, which can be found in many catalogues, allows us to choose the rod diameter quickly without performing complex calculations (Figure 6.40 shows the different types of fastening).

1. Identify mounting style using the codes above.
2. Determine the stroke factor 'Kc' by searching for the characteristics of rod connection to the moving part shown in Figure 6.41.
3. Multiply the Kc factor by the actual stroke. The result is the reference length 'Li'.

4. If the thrust needed is known [p • S_{piston} (in kN)] and depending on the reference length 'Li', the point of intersection can be found on the 'Chart to choose the rod diameter' (Figure 6.42): the rod diameter corresponds to the curve *above* it.

Rod stroke factor (Kc)

Mounting style	Rod connection	Mounting	Stroke factor Kc
A-P-R	Fixed and supported		2
	Fixed and rigidly guided		0.5
	Jointed and rigidly guided		0.7
B-Q	Fixed and supported		4
	Fixed and rigidly guided		1
	Jointed and rigidly guided		1.5
H	Jointed and rigidly guided		1

Mounting style	Rod connection	Mounting	Stroke factor Kc
C-D-F-N	Jointed and supported		4
	Jointed and rigidly guided		2
G	Fixed and supported		2
	Fixed and rigidly guided		0.5
	Jointed and rigidly guided		0.7
L	Jointed and supported		3
	Jointed and rigidly guided		1.5

Figure 6.41

Chart to choose the rod diameter

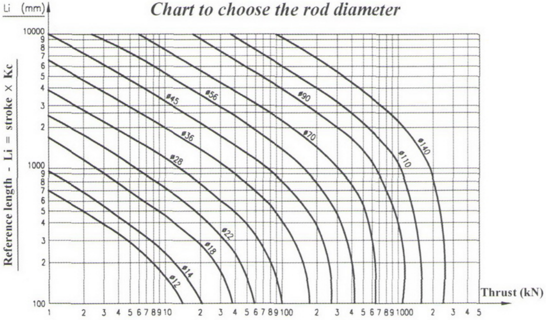


Figure 6.42

This method can be used provided that only few centimetres of the rod with the cylinder on stand-by come out of the front end. Otherwise, the factor Kc has to be multiplied by the stroke plus the length of the rod out.

POSITION DETECTION

By 'position detection' we mean the ability to check whether the load is actually or virtually found on the end stroke, the begin stroke or any other point of the stroke of the actuator. The detection can be:

- ✓ *Visual* – The operator checks the position of the rod or the actuator connected to it in order to acts accordingly. For instance, if the operator handles a platform, he/she stops the directional valve as soon as the expected position is reached.
- ✓ *Automatic, for warning* – The final position, the danger position, etc., is detected by an electric limit switch that starts a warning light, a siren or another device. For instance, a siren sounds an alarm (by means of a limit switch on the actuator) throughout the lifting of a container.
- ✓ *Automatic, without human intervention* – The initial, intermediate or final position is detected by a mechanical, electric or electronic detector (a limit switch or a sensor) transmitting the signal to the control component that performs the appropriate operation. For example, the tool of a punching machine must move back to a neutral position after the sheet is machined: a limit switch placed at the limit stop of the guide rail (or better still, by detecting the piston position directly on the cylinder), sends the switch signal to the coil of the directional solenoid valve.

Automatic position detection is carried out by a/an:

- *Mechanical/hydraulic limit switch* – The mechanical contact part, that is a small ball, a roller, a rod, an antenna or another component, moves the hydraulic valve; this valve usually sends the pilot signal to the main directional valve. The mechanical contact part moves by means of cams, trunnions carefully arranged of the machine.
- *Electromechanical limit switch* – The mechanical contact part, having the same characteristics and movement as the previous limit switch, switches an electric contact that sends the signal to the solenoid valve control unit.
- *Electric sensor* – There are no mechanical contacts between the actuator and the detection mechanism. The sensor detects the position of the piston, the rod or the load inductively or capacitively and transmits the signal to the control unit.
- *Electronic transducer* – It is the same as an electric sensor, but an electronic unit manages it. It is essential in feedback closed loop circuits.

This paragraph deals with electromechanical limit switches and electric sensors. Mechanical/hydraulic limit switches are nothing more than directional valves and electronic transducers are analysed in the chapters devoted to proportional control.

Electromechanical limit switches

The mechanical part, moved by the cam on the machine or on the rod, switches the position of the contacts housed in the container (Figure 6.43).

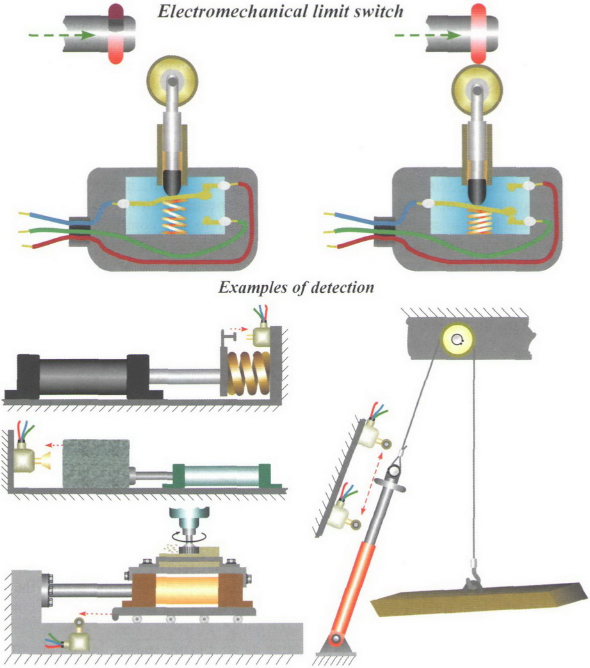


Figure 6.43

This device must comply with the standards of the electric industry (CEI – Italian Electrotechnical Committee) and be totally watertight; it is advisable to use it with low electric potential (12, 24 volt DC or AC). Insulation must be considered carefully in systems demanding electric potentials of 110 or 220 volt AC.

Magnetic Reed sensors

Magnetic sensors detect the *piston* position by means of an electric signal. In order to switch over the internal contact, they must be operated by the magnetic field generated by a permanent magnet placed inside the piston. As a result, the piston translation is signalled by the sensors placed at the initial and final limit stop and also at intermediate positions.

The contact shells inside the sensors are held in a glass envelope (Reed principle). The magnetic field that moves with the piston, next to the sensor, switches the position of its contact (normally open or closed depending on the circuit needs) that switches the solenoid valve once it is conveyed to the control unit (Figure 6.44). A light-emitting diode on the visible part of the Reed sensor signals the accomplishment of the switching.

The magnetic field of the permanent magnet must not be subjected to interferences; consequently, the whole cylinder has to be made of **non-magnetic** materials (aluminium, bronze, stainless steel). These sensors can be applied to every type of single- and double-acting cylinder, but in most cases they have to be prearranged by the manufacturer because it is very difficult to modify standard cylinders.

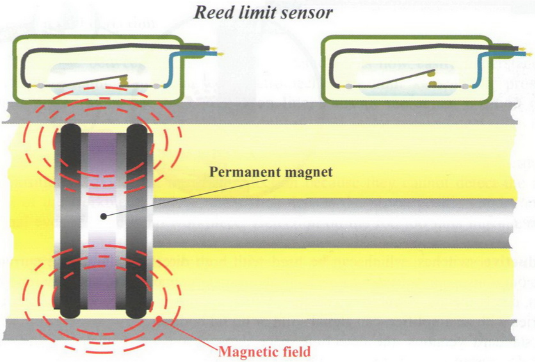


Figure 6.44

Some guides or clamps help to position the sensors on the cylinder body; in addition, the use of reed sensors avoids the inclusion of large limit switches on the rod.

A special static sensor, provided with a control electronic circuit, replaces the mobile contact of a circuit having a very high resistance on stand-by and a low resistance when it is subjected to the magnetic field. The change due to the auxiliary circuit provides the switch signal.

Although they are very simple and convenient, reed sensors are rarely used in oil hydraulics: sometimes the rod, the piston and the body made of non-magnetic metals makes them unsuitable because of the incompatibility of the material (bronze, aluminium, etc.) or for its high cost (stainless steel).

Proximity switches

Proximity switches are on/off transducers without mobile mechanical parts. They are shaped into a parallelepiped (usually for long-distance detection), a smooth cylinder to be arranged in a suitable recess or a threaded cylinder with a double fastening nut.

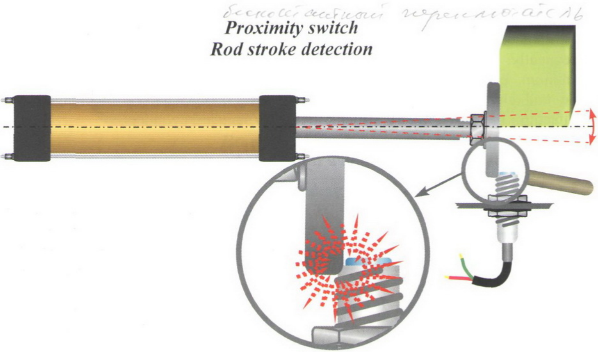


Figure 6.45

Inductive switches, which can be used with both direct or alternating current (the weak outgoing signal in electromechanical must be connected in series with a power relay), detect metals; **capacitive** switches, which are operated only with a direct current in series with an amplifier, can signal solid or liquid bodies.

In standard versions, the detection space depends on the diameter of the external body; for instance, a diameter of Φ 18 mm allows the detection of an object at the maximum distance of 5 – 6 mm.

Note that these switches, albeit very reliable in many applications, are rather unreliable for the direct detection on the rod: the limited tolerance of end-stroke angular deflection demands a precise approach of the switch. In addition, because of the wear of the different mobile parts, the rod itself could damage or incline the switch undermining its detection effectiveness (Figure 6.45). For this reason, switches having a fairly large diameter are needed in order to carry out a detection at least 7 – 8 mm from the rod.

Proximity switches can be positioned in the cylinder heads so as to avoid the detection on the rod or on the machine; the piston position can be detected only at the beginning- and end-of-travel position because switches cannot be arranged on the body (Figure 6.46).

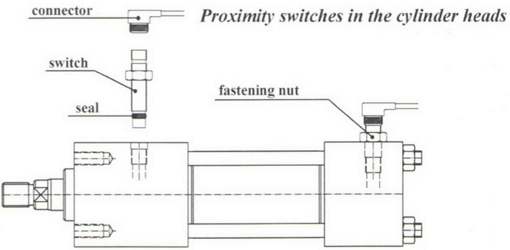


Figure 6.46

Virtual or actual detection

The difference between actual and virtual detection is now easily understandable. Mechanical/hydraulic limit switches, electromechanical limit switches or proximity switches, placed along the load translation line, transmit the actual position of the workpiece and they do not work properly only if the detector breaks down or an object ends up between it and the workpiece accidentally.

Reed sensors or proximity switches, positioned on the cylinder and detecting the piston position, provide just a virtual detection because they cannot detect the actual position of the workpiece. Indeed, the switch detects the piston position and transmits the signal even if there is no workpiece. Detection on the rod too falls into the virtual category.

As no type of detection is totally reliable and safety conditions are strict, machines have to be equipped with external mechanical limit switches to avoid positions beyond the limit allowed.

Servocylinder

By 'servocylinder' we mean a linear actuator provided with a switch that can detect the piston position at any time (Figure 6.47). Except for some obsolete versions with mechanical feedback, most servocylinders have linear transducers managed by an electronic control unit. These transducers (this subject is expanded on in the chapter devoted to electroproportional control) can range from simple linear potentiometers to the LVDTs, from encoders to magnetostrictives.

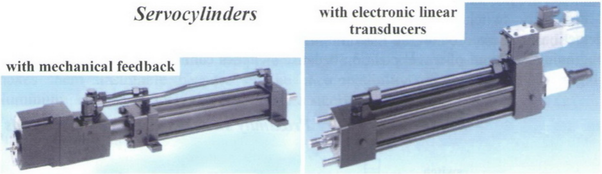


Figure 6.47

Unlike the old versions demanding an external space equal to the rod length, these transducers are placed inside the rod (Figure 6.48).

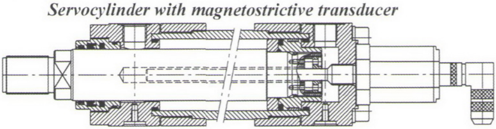


Figure 6.48

The directional solenoid valve controlled by the proportional electronic system is placed on the cylinder head.

Figure 6.49

ISO STANDARDS FOR CYLINDERS

A number of ISO standards for oil hydraulic cylinders establish their dimensioning and mounting style, working pressures, bores, diameter, external thread and rod stroke, guide ring seats, seals and rod wiper, types of joints for the rod end, piston-rod ratio and fitting dimensioning.

In particular, the following standards set the mounting style of the cylinders. The last number preceded by a hyphen refers to the year the standard was issued. The standards are very important for manufacturers, designers and fitters because they set the precise dimensions needed to arrange linear actuators on the machine.

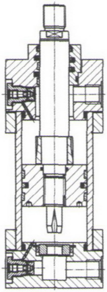
The ISO 6020 standard

The ISO standard 6020 is divided into three parts and it establishes the mounting dimensions for oil hydraulic cylinders, in accordance to the standards on clearance holes for screws (ISO 273), bores (ISO 3320), rod threads and dimensions (ISO 4395), strokes (ISO 4393), bodies (ISO 4394) and nominal pressure (ISO 3322). This standard also defines the word '*mounting*' as the 'device enabling the mounting of the cylinder on the corresponding part'. The nominal pressure of such cylinders is set to 160 bar.

ISO 6020/1: 'Hydraulic fluid power – Mounting dimensions for single rod cylinders, 16 MPa (160 bar) series – Part 1: Medium series'.

Standard bores (mm): 25 – 32 – 40 – 50 – 63 – 80 – 100 – 125 – 160 – 200 – 250 – 320 – 400 – 500.

Mounting style: rectangular or round flanges, single rear fixed or removable ball jointed eyes and clevises, front or rear trunnions, fixed or removable intermediate trunnions.



ISO standard 6020/1 cylinder

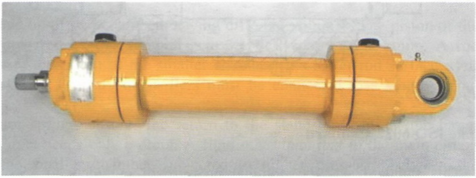
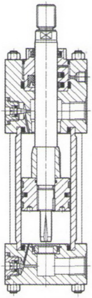


Figure 6.50

ISO 6020/2: 'Hydraulic fluid power – Mounting dimensions for single rod cylinders, 16 MPa (160 bar) series – Part 2: Compact series'.

Mounting style: rectangular heads, single or double fixed ball jointed eyes and clevises, side feet, built-in front or rear trunnions, fixed or removable intermediate trunnions, front or rear outward tie rods or studs.



ISO standard 6020/2 cylinder

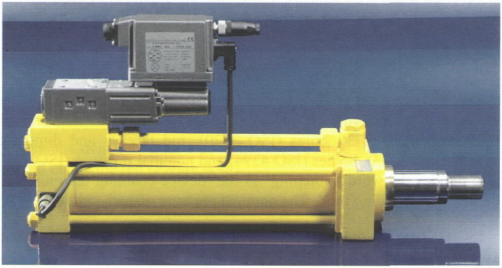


Figure 6.51

ISO 6020/3: 'Hydraulic fluid power -- Mounting dimensions for single rod cylinders, 16 MPa (160 bar) series -- Part 3: Compact series with bores from 250 mm to 500 mm'.

Standard bores (mm): 250 – 320 – 400 – 500.

Outline drawing of an ISO standard 6020/3 cylinder

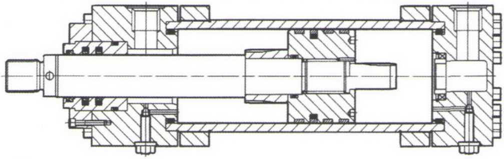


Figure 6.52

The ISO 6022 standard

This standard sets the nominal pressure of cylinders to 250 bar. The reference standards are the same as for ISO 6020.

ISO 6022: 'Hydraulic fluid power - Mounting dimensions for single rod cylinders, 25 MPa (250 bar) series'

Standard bore (mm): 50 – 63 – 80 – 100 – 125 –140 – 160 – 180 –200 – 250 – 320.
Mounting style: front and rear round flanges, single removable rear ball jointed eyes and clevises, fixed or removable intermediate trunnions.

ISO standards 6020/3 and 6022 cylinders

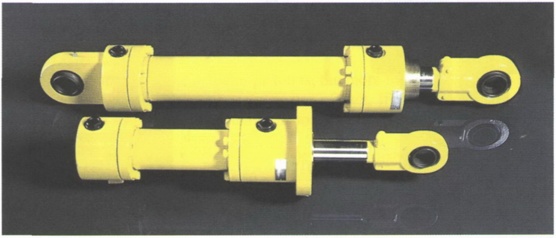


Figure 6.53

SPECIAL ACTUATORS

Some components cannot be classified as simple basic actuators due to their design, yet they are based on the operating principle of cylinders.

Pressure intensifier

Some applications, like high-pressure sealing tests for metal pipes, require a far higher pressure than the maximum pressure the pump can deliver. Pressure intensifiers (Figure 6.54) respond to this need fairly well although stroke reduces as pressure increases.

This device is made up of two chambers having different volumes. The piston of the larger chamber is powered by the nominal pressure delivered by the pump. A rod connects this piston to the piston of the smaller chamber (the lower rod surface itself often serves as a piston). Pressure intensifiers must be connected to a **single-acting actuator cylinder**.

The robust actuator springs ensures the return to the stand-by position; the larger chamber facing the inlet needs only a drain. A fluid other than that in the main circuit can be found in the smaller chamber subjected to higher pressure (between the intensifier and the actuator cylinder). Leakages on the high pressure circuits are replenished by a dedicated replenishing tank.

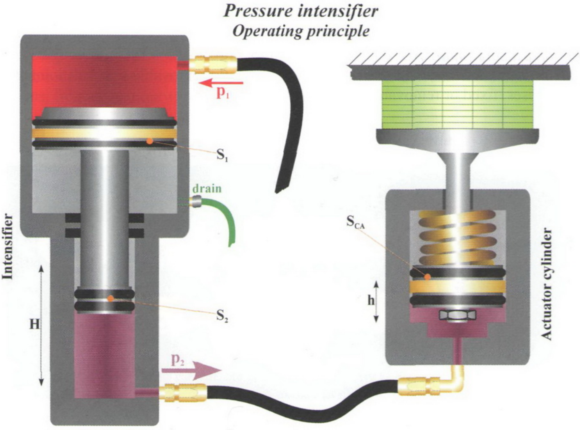


Figure 6.54

The volume difference of the two chambers causes a substantial pressure rise according to Pascal's principle. The following simple calculations show this phenomenon:

Pressure intensifier → Larger piston diameter D = 100 mm; working pressure $p_1 = 60$ bar; smaller piston diameter d = 30 mm; calculate the outgoing pressure p_2.

$$\text{Larger piston surface } S_1 = \frac{D^2 \bullet \pi}{400} = \frac{100^2 \bullet 3.14}{400} = 78.5 \text{ cm}^2$$

$$\text{Smaller piston surface } S_2 = \frac{d^2 \bullet \pi}{400} = \frac{30^2 \bullet 3.14}{400} = 7 \text{ cm}^2$$

$$p_1 \bullet S_1 = p_2 \bullet S_2 \qquad \text{therefore} \qquad p_2 = p_1 \bullet \frac{S_1}{S_2} = 60 \bullet \frac{78.5}{7} = 673 \text{ bar}$$

The force F_1 exerted on the larger piston is:

$$F_1 = p_1 \cdot S_1 = 60 \cdot 78.5 = 4710 \text{ daN}$$

Actuator cylinder → If the surface of the actuator cylinder piston is equivalent to the surface of the smaller piston of the multiplier (7 cm^2), the force F_{CA} is obviously equal to the force of the larger piston: $F_{CA} = p_2 \cdot S_2 = 673 \cdot 7 = 4710$ daN

Assuming an actuator cylinder has a surface S_{CA} of 40 cm^2, the force F_{CA} exerted on it is:

$$F_{CA} = p_2 \cdot S_{CA} = 673 \cdot 40 = 26920 \text{ daN}$$

The larger the diameter of the actuator piston is, the more force is.

As force goes up, stroke shortens; as a matter of fact, if the stroke H of the larger piston is 50 mm, the stroke h of the actuator piston is:

$$h = H \cdot \frac{F_1}{F_2} = 50 \cdot \frac{4710}{26920} = 8.75 \text{ mm}$$

Decelerators

Decelerators are independent parts, they are not powered by the hydraulic circuit and their task is to reduce the speed of the moving mass in the final part of the stroke.

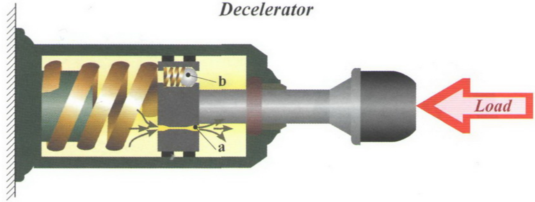


Figure 6.55

The load pushing the rod towards the spring forces the oil held in the left part to flow through the tiny constriction (a); the non-return valve (b), which has a larger cross section than (a), promotes a quick return (Figure 6.55). An elastic component is arranged inside it to counterbalance the different volumes developed during the motion.

Collection devices

This section briefly deals with the numerous actuators operated by one or more single- or double-acting cylinders and whose task is to collect various objects or to block them on the bench in order to machine them.

The collection devices of the mobile industry are orange peel grapples mounted on mobile arms to collect scraps or logs, digging buckets, the powerful crushers of excavators and, last but not least, the huge excavator hammers for building demolition, although they are not involved in this subject. The devices in the stationary industry include holders and clamps to hold objects in their numerous shapes and versions; depending on their design, they can lift heavy and unstable loads, collect large objects, enter frames and round or square holes, hold unstable workpieces, hold and deform resistant materials.

ROTARY ACTUATORS

Rotary actuators are those devices that usually perform revolutions of less than 360° and that has to ensure the stroke in an alternative manner. Since the operation of these actuators with motor standard systems would be too complex, they are often applied to systems derived from the operating principle of linear cylinders.

Rotary actuators can deliver very high torques, even as much as 100000 Nm and in some versions the rotation can exceed 360° for a few revolutions.

Simple vane

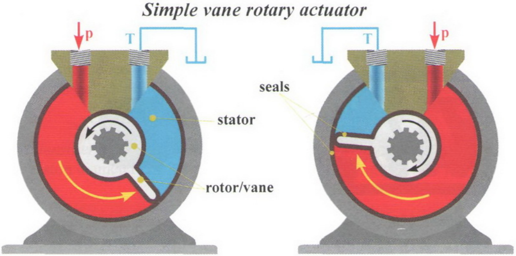


Figure 6.56

This type of actuator can ensure revolutions of about 280°. The vane, connected to the central rotor, is operated by the fluid conveyed to the cylinder seat. In order to guarantee the sealing between the two delivery/unloading chambers arising from the revolution, both the internal part of the stator and the rotor/vane are covered with a dedicated seal (Figure 6.56).

Double vane

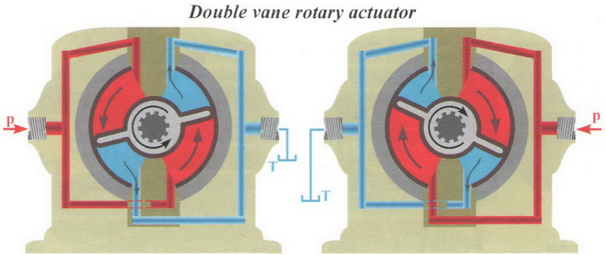


Figure 6.57

Figure 6.58

Since the thrust effective surface is larger in double vane rotary actuators, they can reach higher torques than simple vane rotary actuators; the maximum revolution is less than 180°. Also their rotary vanes and their stator chambers need adequate seals.

Figure 6.57 shows clearly that the fluid flows into and out of two chambers; the flow reversal due to the directional valve obviously results in the opposite revolution.

Rack cylinder

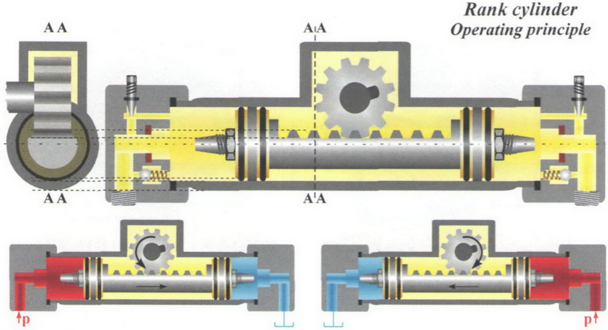


Figure 6.59

Two side pistons are solidly assembled to a rack shaft that is coupled to a gearwheel transmitting the rotary motion via the pinion (Figure 6.59). The use of the inlet of either a piston or the other causes a clockwise or anticlockwise revolution.

Large versions guarantee high torques and they are the best rotary actuator on the market. The number of teeth and the length of the rank determine the revolution angle.

Torque actuator

In torque actuators, the rotating shaft has a spiral slot inside the cylinder (Figure 6.60). The piston can move but it cannot revolve because it is forced to slide on a guide fixed inside the body. Unlike all the other types of cylinders, the rod and the piston are not solidly connected together but simply constrained by a hinge coming out of the internal clearance of the piston and having the same profile as the spiral slot on the rod.

Actually, this device is similar to the classic mechanical ‘screw-nut’ coupling: the piston and the rod serve respectively as a nut and a screw. When the fluid is conveyed, the piston moves making the rod rotate; the rod end outside the cylinder works as a rotary shaft.

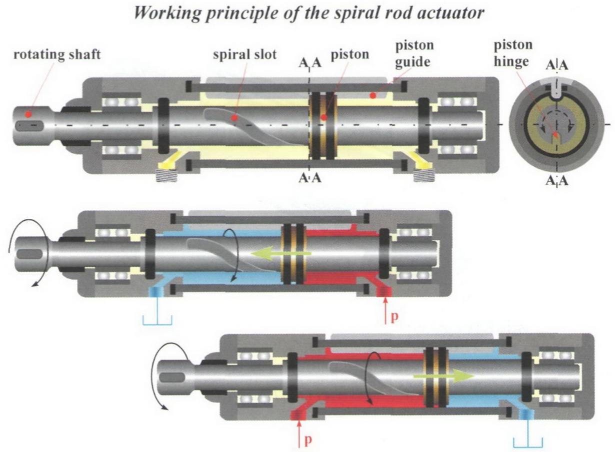
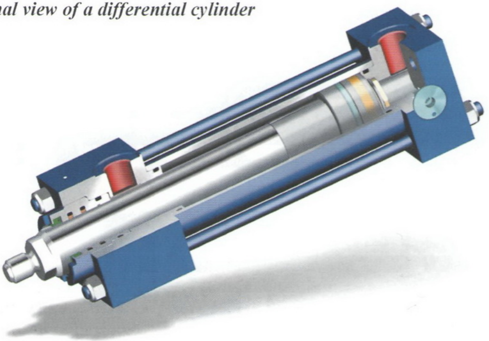

Working principle of the spiral rod actuator
rotating shaft
spiral slot
A A
piston
piston guide
piston hinge
A A
A A
A A
p
p

Figure 6.60

Internal view of a differential cylinder



Servocylinder

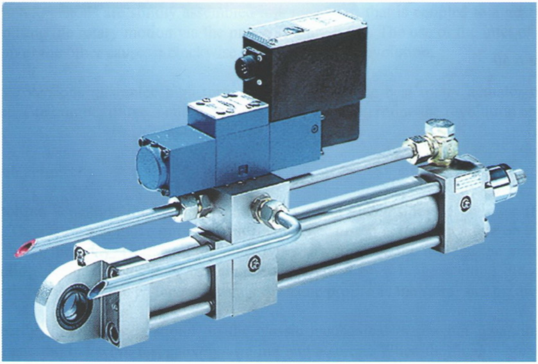


Figure 6.61

Chapter 7

OIL HYDRAULIC MOTORS

Oil hydraulic motors are more and more successful thanks to their excellent performances, their remarkable speed range, the simplicity of direction reversal, their sturdiness (although there are light weights) and the opportunity to arrange them in many positions. They are employed in a wide range of industries, like construction, farming, self-propelled machines drive, lifting and dragging equipment, navigation, aerospace industry and military applications.

BASICS

Motor design is very similar to pump design, even if most motors must allow direction reversal and sustain the vibration and strain transmitted from the load.

Operating principle

Oil hydraulic motors are based on almost the same operating principle as linear actuators. The main difference consists in the fact that cylinders have a limited piston stroke whereas motors have an unlimited stroke.

The piston of the cylinder, the part that transforms hydraulic energy into mechanical energy, is replaced by gears, vanes or pistons in rotary actuators. Nonetheless, the principle is evidently the same: the pressurized fluid exerts a force on the surface of moving parts.

In short, the pressure p on the surface S of the part generates the force F that outweighs the load opposition and makes the actuator rotate; the torque results from the product between this force and the distance between the centre of the rotor (shaft) and the middle of the part subjected to the pressure. This arm spans from the shaft axis to the middle of the tooth or the vane in gear and vane versions, while it depends from the angle of the plate between the piston ball joint and the shaft axis in axial piston motors and the distance of the piston acting on the eccentric shaft in radial piston motors and the shaft axis (Figure 7.1).

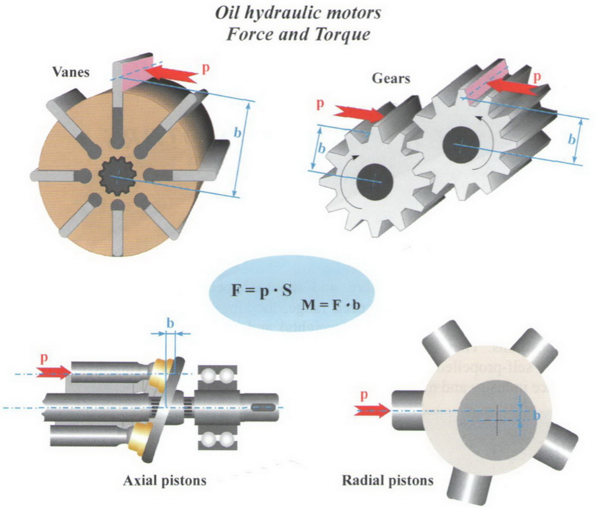


Figure 7.1

Dimensioning

Efficiency

Given a specific flow rate, the volumetric efficiency η_v of an oil hydraulic motor is the ratio of actual rotational speed rpm_e to theoretical rotational speed rpm_t $\rightarrow$ $\eta_v = rpm_e/rpm_t$.

Mechanical or pressure efficiency η_m is the ratio of actual torque M_e to theoretical torque $M_t \rightarrow \eta_m = M_e/M_t$.

Overall efficiency $\boldsymbol{\eta_g}$ is the product of volumetric efficiency η_v by mechanical efficiency $\eta_m \rightarrow \eta_g = \eta_v \bullet \eta_m$.

Flow

$$Q = \frac{c \bullet rpm}{1000 \bullet \eta_v} = dm^3/min = l/min. \text{ Therefore } c = \frac{Q \bullet 1000 \bullet \eta_v}{rpm}$$

Rotational speed

$$\text{rpm} = \frac{Q \cdot 1000}{c} \cdot \eta_v \text{ (rev/min)}$$

Torque

$$M = \frac{c \cdot p \cdot \eta_m}{20 \cdot \pi} \quad (\text{N} \cdot \text{m})$$

Mechanical power absorbed

$$P = \frac{M \cdot \text{rpm}}{9740 \cdot \eta_v} \quad (\text{kW})$$

Hydraulic power

$$N = \frac{2 \cdot \pi \cdot 10^{-3} \cdot M \cdot \text{rpm}}{60} =$$

$$\frac{Q \cdot p \cdot \eta_g}{600} \quad (\text{kW}) \quad \text{or} \quad N = \frac{Q \cdot p \cdot \eta_g}{450} \quad (\text{HP})$$

N = Hydraulic power (kW)
P = Mechanical power (kW)
M = Twisting moment or torque (N • m)
c = Displacement (cm^3/rev)
Q = Flow (dm^3/min o l/min)
p = Pressure (bar)
η_v = Volumetric efficiency
η_m = Mechanical or pressure efficiency
η_g = Overall efficiency
rpm = Rotational speed (rev/min)

General features

	Variation	*Rotational speed*	*Pressure*	*Torque*
Flow	Increase	Higher	Constant	Constant
	Reduction	Lower	Constant	Constant
Displacement	Increase	Lower	Lower	Higher
	Reduction	Higher	Higher	Lower

Satisfactory oil hydraulic motor have a good torque during starting and when it is fully operational, as well as a rotational speed that is suitable for the system needs. Flow increase makes speed rise; pressure increase (at a constant flow rate) marginally reduces speed as it causes more leakages. Its increase in variable displacement versions results in a drop in the number of revolutions and a higher torque, while the opposite phenomena occur if it diminishes. The table above sums up the standard operational features of oil hydraulic motors.

Motor starting needs enough flow rate to counterbalance internal leakages and a pressure suitable for the starting torque; displacement cannot be diminished to zero because speed would tend to infinity. This subject is dealt with in the final part of this chapter ('Pump-Motor Combinations').

Like pumps, hydraulic motors (except for gear motors) can have a variable displacement, but variable displacement axial piston actuators (be them in-line or bent axis versions) are used instead mainly for design and cost reasons.

Oil hydraulic motors are divided into low-speed high-torque motors and high-speed low-torque motors according to their features; orbital motors and radial piston motors fall into the first group, gear, vane and axial piston motors into the second group.

High-speed low-torque motors can guarantee rotational speeds ranging between 30 and 3000 rpm, but they are rather irregular at low speeds; conversely, low-speed high-torque motors are suitable for a maximum of 300 rpm and they can ensure excellent torques even below 1 rev/min and with good revolution regularity.

As a result, it is vital to establish whether the rotational speed is compatible with both low-speed high-torque motors and high-speed low-torque motors. The preference is usually given to low-speed high-torque motors or to a mechanical speed reduction unit applied to high-speed low-torque motors: this is the typical case of medium-large hydrostatic drives since some manufacturers choose radial piston motors (low-speed high-torque motor) while others prefer an axial piston motor provided with a speed reduction unit that is lighter and ensures energy savings despite the inclusion of this mechanical device.

HIGH-SPEED LOW-TORQUE MOTORS

High-speed low-torque motors are divided into two categories: motors that, albeit high speed, have a high torque (axial pistons motors) and cheaper motors with lower torque. Except for orbital motors, gear motors and, at least to some degree, vane motors have a poor starting torque, especially if they are equipped with balancing bearings; as a matter of fact, bearings are excellent for axial and radial hydrostatic balancing but they are an obstacle to the starting (efficiency is very low if there is no pressure).

External gear motors

External gear motors are the cheapest oil hydraulic motors. They are very popular, easy to manufacture and they can work even if the fluid is rather contaminated. They look like external gear pumps, but there is an external difference because they have two openings having the same bore, as well as a difference in terms of overall efficiency (their efficiency is lower by 5 ÷ 10%). For these reasons, they are not suitable for applications needing a fairly good starting torque (sometimes they cannot be started up even with quite low loads); on the contrary, because of their good value for money, lightness and small dimensions, they are the ideal solution for high-speed and no-load starting applications like, for instance, the movement of blades for mowers and garment shredders.

The pressure (p) developed by the contact of the effective area of each tooth of both gears entails the torque needed for the revolution. Only one gear is solidly connected to the drive shaft (Figure 7.2).

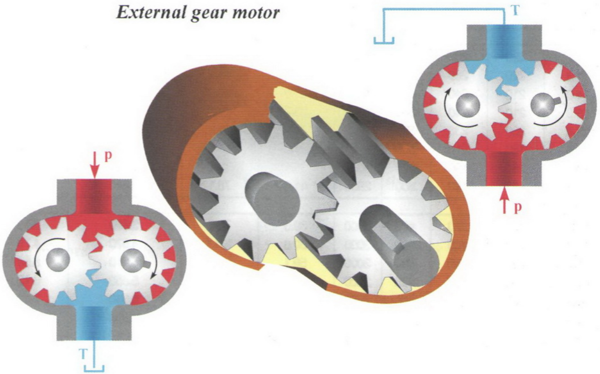


Figure 7.2

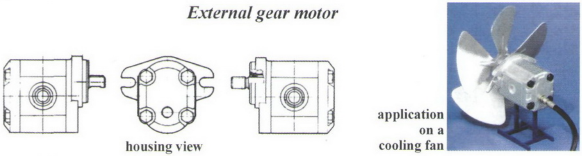


Figure 7.3

Like gear pumps, also motors have radial/axial balancing bearings and leakage clearances.

Standard versions of external gear motors have displacements ranging from 2.5 to 100 cm^3/rev (some special versions approach 200 cm^3/rev) and they are designed to sustain up to 3500 rpm at nominal pressures of about 150 – 200 bar and peak pressures up to 250 bar (Figure 7.4).

Standard versions of oil hydraulic gear motors

DISPLA-CEMENT (cm^3/rev)	MAXIMUM PRESSURE (bar)		SPEED (rev/min)		NOMINAL FLOW		NOMINAL TORQUE	NOMINAL POWER
	WORKING	PEAK	MIN	MAX	at 1500 rev/min	at maximum speed	at 100 bar (daNm)	at 1000 rev/min and 100 bar (kW)
2.7	220	260	500	5000	4.1	13.5	0.370	0.39
4.1	210	250	500	4000	6.2	16.4	0.560	0.59
5.1	210	250	500	4000	7.7	20.4	0.680	0.72
6.1	200	240	500	3800	9.2	23.2	0.850	0.89
9.5	220	260	500	3000	14.3	28.5	1.270	1.33
11.3	220	260	500	4000	17.0	45.2	1.510	1.58
14.0	210	240	500	4000	21.0	56.0	1.870	1.96
15.8	210	240	500	4000	23.7	63.2	2.130	2.23
17.8	200	230	500	3600	26.7	64.1	2.340	2.45
20.8	180	210	500	3200	31.2	66.6	2.810	2.94
23.4	160	190	500	3000	35.1	70.2	3.180	3.33
27.9	150	180	500	2800	41.9	78.1	3.840	4.02
26.4	230	270	400	3000	39.6	79.2	3.590	3.76
33.7	220	260	400	3000	50.6	101.1	4.570	4.79
39.4	220	260	400	3000	59.1	118.2	5.310	5.56
42.7	220	260	400	2800	64.1	119.6	5.850	6.12
51.4	200	240	400	2400	77.1	123.4	6.940	7.27
60.0	180	220	400	2800	90.0	168.0	8.200	8.58
69.6	170	200	400	2500	104.4	174.0	9.610	10.07
77.6	160	190	400	2300	116.4	178.5	10.660	11.16
87.6	140	170	400	2000	131.4	175.2	11.960	12.53
42.7	220	260	400	2800	64.1	119.6	5.850	6.12
51.4	200	240	400	2400	77.1	123.4	6.940	7.27
60.0	180	220	400	2800	90.0	168.0	8.200	8.58
69.6	170	200	400	2500	104.4	174.0	9.610	10.07
77.6	160	190	400	2300	116.4	178.5	10.660	11.16
87.6	140	170	400	2000	131.4	175.2	11.960	12.53

Figure 7.4

Direct drive Gerotor motors

Internal gear motors are divided into Gerotors having two mobile gears and orbital motors, which are dealt with in the following paragraph.

The design and operation of Gerotor motors are similar to those of Gerotor pumps. Gerotor motors are made up of an inner gear solidly connected to the shaft and an outer gear that can revolve on itself (Figure 7.5). The outer gear always has one more tooth than the inner gear, which usually has six teeth; for this reason, each Gerotor tooth meshes with the corresponding tooth of the outer gear only once during every revolution.

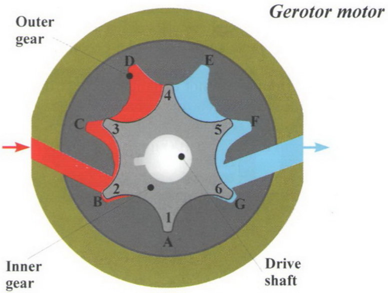


Figure 7.5

Vane motors

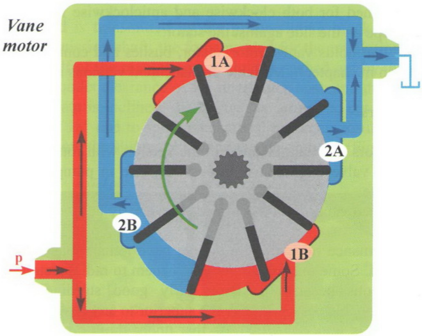


Figure 7.6

Like for vane pumps, the balance of these motors is due to two inlet diametrically opposed and mutually connected and two outlets. In this manner, the transversal forces acting on the delivery chamber are counterbalanced by the forces acting on the other. Consequently, the hydraulic thrust neutralises most of the radial imbalance promoting the dimension reduction of bearings.

The incoming fluid generates the torque needed for the revolution by acting on the vanes that form cavities 1A and 1B; when the rotor is moving, the volume of the fluid compressed between the vanes in the pressurised area is progressively moved towards the outlet and it is pushed out through cavities 2A and 2B. Direction reversal results from switching the directional valve so that the inlet becomes the outlet and vice versa (Figure 7.6).

Unlike vane pumps, vane motors cannot exploit the effect of the centrifugal force to make blades come out; since they have to be already in position when the motor is switched off in order to promote the action of the initial flow, it is essential to equip them with opposing springs in the inner part (Figure 7.7).

Opposing springs on a vane

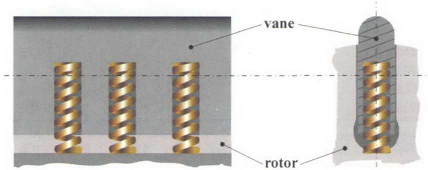


Figure 7.7

As they are designed for both clockwise and anticlockwise revolution, vanes must have a rounded shape on the side against the stator.

The 'cartridge' that holds vanes, rotor, stator, bushes and control heads is similar to the cartridge shown in chapter 4. Variable displacement versions are quite unusual.

In-line piston motors

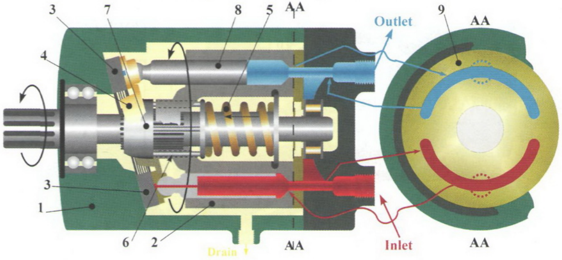
In-line piston motors are manufactured in the version with rotary cylinder block and distributor bushing; valve operation is impossible, like in pumps with fixed cylinder block and swash plate.

Maximum speed can exceed 3000 rpm and it is usually advisable not to work below 50 – 30 rpm.

These high-performance oil hydraulic motors have remarkable torques despite their high rotational speed. Some manufacturers prefer them to radial piston motors for some low-speed applications because of their fairly good sturdiness, weight, limited dimensions and reasonable costs (yet they are superior to any type of external or internal gear motor). Still, it is important to stress that under these circumstances slow-speed rotations must be performed by including a mechanical reduction gear because swash plate motors not only lose most of their torque, but they also have rather irregular revolutions at low speeds.

A hydrostatic sliding block fitted to the plate swings on the ball joint of every piston; the incoming fluid flowing through the clearance of the distributor bush reaches half of the cylinders (fluid is delivered through the other clearance connected to the other half of cylinders in order to get the other revolution direction).

Axial piston motor
Inclined plate - Rotary cylinder block



1. Motor housing
2. Rotary block
3. Fixed inclined plate
4. Sliding block holder
5. Preload spring
6. Spring pin
7. Thrust ring
8. Piston
9. Distributor bush

Figure 7.8

The force developed is transmitted by the piston/sliding block to the plate solidly connected to the transmission shaft; the force needed for the revolution develops on the plate. In the second half of the revolution, the fluid is pushed out by the pistons that are now connected to the outlet clearance of the distributor bush and that are guided by the plate. The more the plate is inclined, the higher the torque is, but the maximum angle must not exceed about 18° due to some mechanical problems resulting from the forces acting between the piston, the sliding block and the swash plate (Figure 7.8).

Axial piston motors - In-line cylinder block

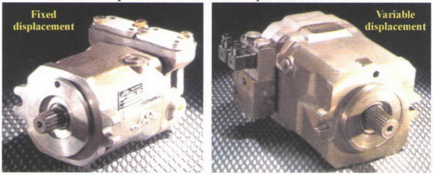


Figure 7.9

Bent axis piston motors

The operation and performances of bent axis piston motors are similar to those of in-line piston motors but they have a higher starting torque. The plate in bent axis piston motors is perpendicular to the drive shaft and the piston unit are usually at an angle of 25°.

Like bent axis pumps, the ball bearings of the transmission shaft must be quite large because the cylinder block that rotates without mechanical bearings transmits the overall strain to them.

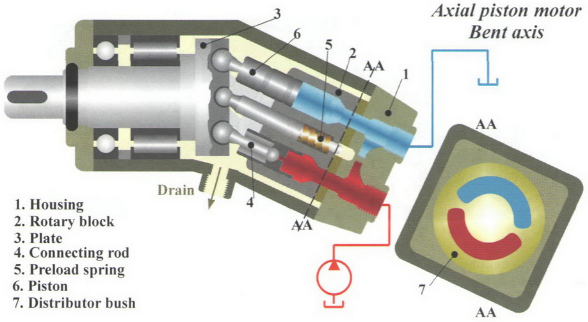


Figure 7.10

Connecting rods made up of a small cylinder with truncated ball ends (the plate ball has a larger diameter than the ball inside the piston) are attached to the plate, solidly connected to the transmission shaft. In order to guarantee the cylinder block is held firmly in place between the plate and the distributor bush, the preload spring pushes a single ball joint against the middle of the plate (Figure 7.10).

Axial piston motors with variable displacement

Both in-line and bent-axis axial piston motors can have a variable displacement. The variation can be performed by means of the manual, mechanical or proportional controllers analysed in chapter 5 by applying the rules on displacement and pressure of oil hydraulic motors.

Figure 7.11 shows a variable displacement in-line piston motor with pressure controller.

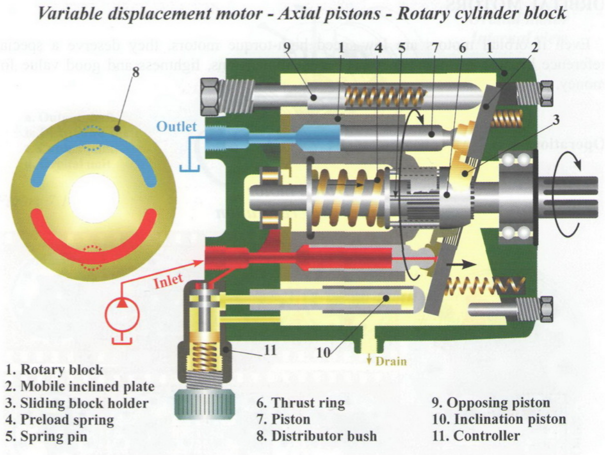


Figure 7.11

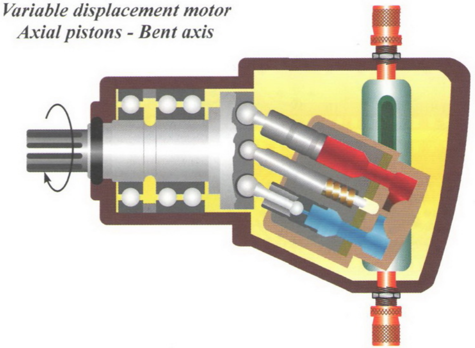


Figure 7.12

ORBITAL MOTORS

Even if orbital motors are low-speed high-torque motors, they deserve a special reference because of their versatility, small dimensions, lightness and good value for money.

Operation

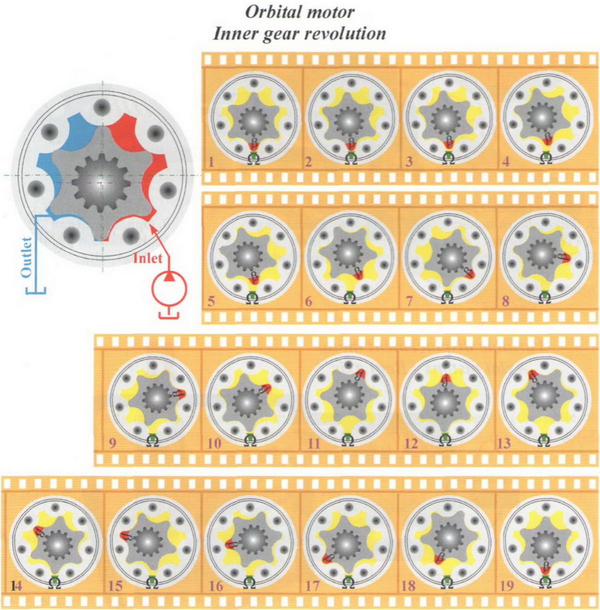
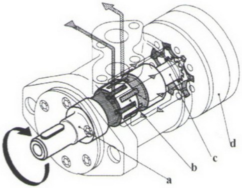


Figure 7.13

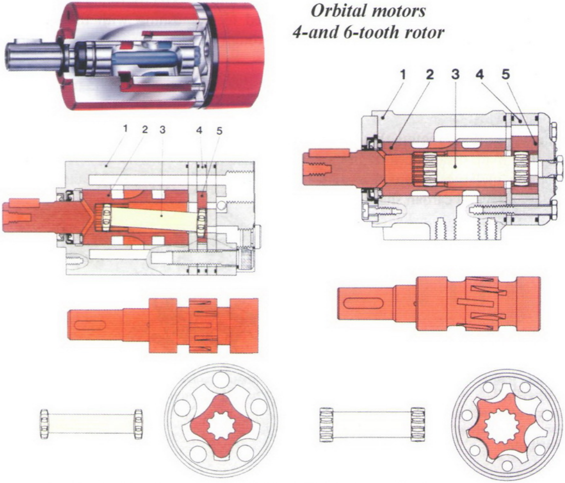
Orbital motor
Internal view



a. Output shaft
b. Distributor shaft
c. Cardan joint
d. Orbital unit

Figure 7.14

Orbital motors
4-and 6-tooth rotor



1. Housing 2. Output/distributor shaft 3. Cardan joint 4. Ring gear 5. Rotor

Figure 7.15

The torque of orbital motors develops in the inner gear, which has 4 to 8 teeth depending on the performance, placed in a gear having inner teeth (like in gerotors, this gears always has one more tooth). Unlike gerotors, the inner gear 'orbits' the axis of the fixed gear, which means it pivots around the centre of the stator. As one rotor tooth always meshes with a stator tooth, pressure and unloading areas are constantly separated.

Figure 7.13 shows the positions of the rotors during a whole revolution (360°); the reference point is the inner gear tooth marked as β and the fixed point Ω of the stator gear.

The distribution of the pressurised and unloading fluid on the orbital unit is guaranteed by a series of channels on the external surface of the distributor shaft (forming a single unit with the output shaft); such channels connect the inlet/outlet on the housing to the corresponding chambers between the orbital rotor and the outer gear (Figure 7.14).

The motion of the rotor is transmitted via a cardan joint inside the distributor/output shaft; as the orbital motion has to be transformed into a perfectly round revolution, the cardan joint ends consist of adequately shaped teeth meshing with the small gears placed inside the shaft and in the middle of the rotor (Figure 7.15).

Roller

Rollers are orbital motors that deliver higher performances than the motors described above. The main differences lie in the gear because its teeth consist in some rollers; the distribution in larger versions is performed by a disc having enough clearances and moved by a second cardan shaft (Figure 7.16).

a. Output shaft
b. Cardan joint
c. Orbital unit
d. Distributor control shaft
e. Non-return valves
f. Distributor

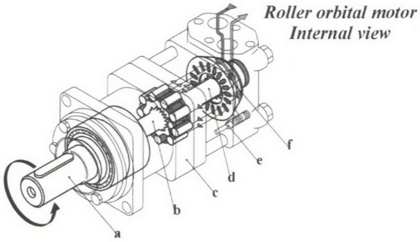
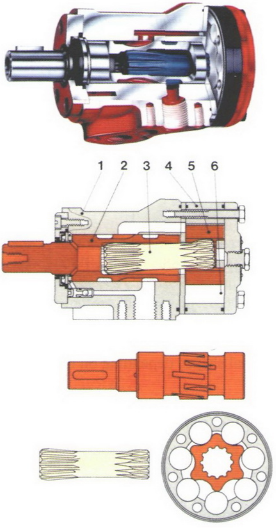


Figure 7.16

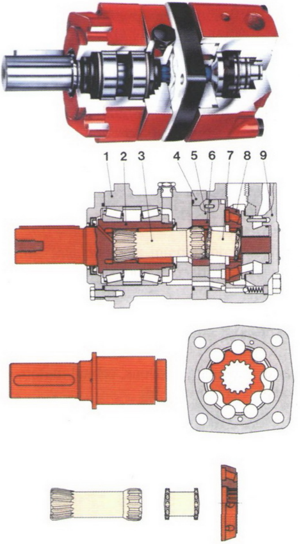
The rollers inside the gear further reduce the friction with the rotor teeth, so they enable more efficiency, higher pressures, frequent direction reversals and the use of low-viscosity oil.

The rotary distributor disc is positioned so as to face the orbital unit and it receives the fluid directly from the inlets (Figure 7.17).

Roller orbital motors



1. Housing 2. Output/distributor shaft
3. Cardan joint 4. Gear 5. Rotor 6. Roller



1. Front housing 2. Output/distributor shaft
3. Cardan joint 4. Gear 5. Roller 6. Rotor
7. Distributor disc cardan joint 8. Distributor disc
9. Rear housing

Figure 7.17

Features

Orbital motors have a high starting torque, regular revolution, constant working torque (related to speed) and they are suitable for open and closed circuits; they can be unequipped with the external drain line and they are usually compatible with synthetic fluids.

The following table shows the detailed features of orbital motors according to their dimensioning.

Orbital motors features						
Type	Displacement *(cm^3/rev)*	Minimum speed *(rpm)*	Maximum speed *(rpm)*	Maximum torque *(daNm)*	Maximum pressure *(bar)*	Power *(kW)*
Gear rim from	8	50	2000	0.7	70	1.1
Gear rim to	160	9	385	30	175	10
Roller, Cardan joint from	50	10	775	10	140	7
Roller, Cardan joint to	250	10	300	61	175	16
Roller, Distributor disc from	80	10	810	20	175	16
Roller, Distributor disc to	800	5	250	188	160	42

Shapes and applications

Besides standard versions, orbital motors with special shapes depending on their use are available (Figure 7.18).

Orbital motors - Special shapes

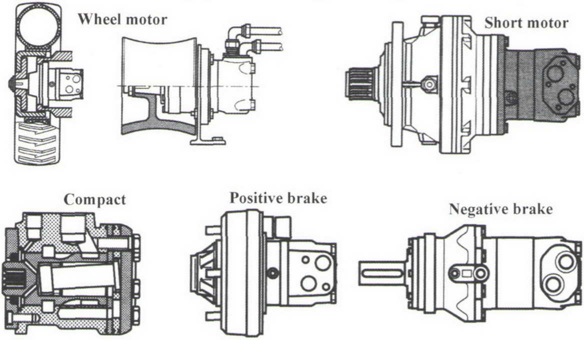


Figure 7.18

Wheel motors are provided with a back round flange that allows both a substantial space reduction for the coupling with the hub of a wheel or with a winch and a better distribution of radial loads between the two bearings of the motor.

The coupling with a reduction gear occurs in *short motors*, unequipped with the output shaft; yet, this demands a reduction gear that absorbs radial and axial loads.

Compact motors are designed for small spaces. Other remarkable applications are the motors provided with a *positive parking brake* or *negative parking brake*. In other words, brake discs are coaxial with their operation cylinder in orbital motors. The cylinder in negative versions keeps the drum in the braking position by means of a spring when it is not on duty and it releases the brake discs only when the fluid flows; The cylinder in positive versions acts in the opposite manner, in other words it acts during the motion. We specify that brakes can be shoe brakes or disc brakes depending on their design.

Some of these special types have a drain line that prevents internal pressure from wearing out the shaft seal. Manufacturers indicate in their catalogues if it is necessary. It is usually necessary to use external drain when motors are in series.

Orbital motors in their different shapes are employed in many applications, ranging from tool machines to construction equipment, platforms, lifting and transport, processing of wood, plastic and rubber.

Figure 7.19

LOW-SPEED HIGH-TORQUE MOTORS

Low-speed high-torque motors are employed in stationary and mobile applications demanding a high starting torque and low speed, like horizontal presses for plastics injection moulding and self-propelled machines hydrostatic drive. Low-speed high-torque motors include not only orbital motors but also the different types of radial piston motors.

Cam-type radial piston motor

In these motors the internal cam box is fixed while it is the rotary cylinder block that transmits motion to the transmission shaft. The number of pistons determines the number of cams: if the piston is totally extended (that is it stretches in the middle of the cam), the corresponding side pistons must touch the external sides of the cam. For instance (Figure 7.20), six cylinders need four cams.

At the ends of the pistons, ball race bearings are fitted in the pistons themselves by half roll on the cams. The tangential action exerted by pistons presses on the cams and consequently it makes the cylinder block rotate on the shaft fixed to the housing; the output shaft is solidly connected to the cylinder block.

The symmetric design of the system ensures the balance of the outstanding mechanical radial loads developed by pistons: the pressurisation of a piston corresponds to the pressurisation of the opposite piston (180°).

Radial piston motor - Fixed cam-type - Rotary cylinder block

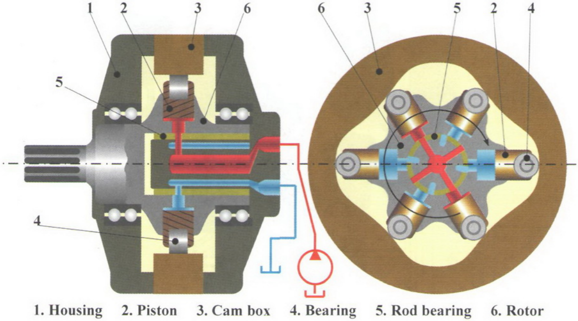


Figure 7.20

The fluid flows in and out the pistons thanks to some conduits inside the cylinder block and it is distributed into these clearances by a solidly revolving distributor bush that is connected to the inlet/outlet flange on the housing.

These actuators are typically employed as wheel motors on mobile vehicles due to their good mechanical balance and their design.

A version with a similar design has a **fixed cylinder block**, attached to the main housing, and a **rotary cam box**. As pistons touch their bearings, they make the box rotate. The box transmits its round motion to the output shaft. The distributor shaft (in this case it is rotary and solidly connected to the box) makes the fluid flow in and out through the conduits connected to the cylinders.

Both versions of these motors have a sturdy design and good overall efficiency; the maximum and minimum rotational speeds are respectively 500 rpm and 2 rpm. They reach torques equalling 10000 Nm.

Crankshaft radial piston motors

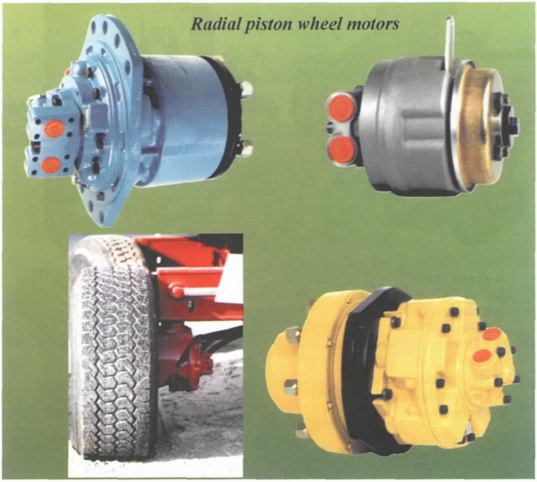


Figure 7.21

Crankshaft radial piston motors, also simply known as **radial** motors, are the low-speed high-torque actuators that deliver the best performances in terms of torque and power.

As these very sturdy motors have limited internal leakages, high displacement, minimum and regular revolutions of 1 rpm, they are vital in slow-speed applications like the hydrostatic drive of large self-propelled vehicles (Figure 7.21), powerful hoists and for the injection of thermoplastics into horizontal presses. However, they are rather heavy and they occupy much space.

The incoming fluid into the cylinders ranged as the spokes of a wheel (Figure 7.22) presses them on the eccentric shaft making it move. As the pistons must follow the movement of the eccentric shaft and be aligned with it, they are equipped with an adequate swinging structure, consisting of a ball joint or a double bearing bush joint on the piston head (Figure 7.22 and Figure 7.23).

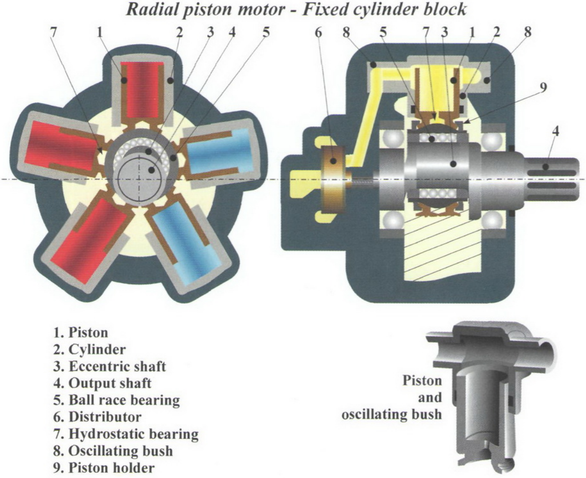


Figure 7.22

A bearing having multiple ball or roller races placed directly on the eccentric shaft transmits the thrust to the transmission shaft. The main task of this bearing is to reduce the sliding between the eccentric shaft and the piston foot, promoting better starting and less wear. A clearance on the foot of the piston allows the fluid to lubricate and to develop a hydrostatic support between the parts.

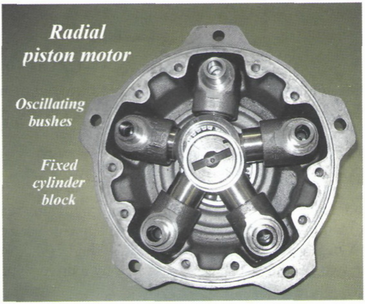


Figure 7.23

The rotary distributor, which is axial to the transmission shaft, delivers or unloads the fluid through the wide clearances inside the oscillating bushes.

Figure 7.24

In order to improve both the revolution regularity and the torque, high-performance versions have **two rows**, in other words two cylinder blocks acting on two out-of-phase eccentric shafts solidly connected to a common output shaft.

Fluid column radial piston motors

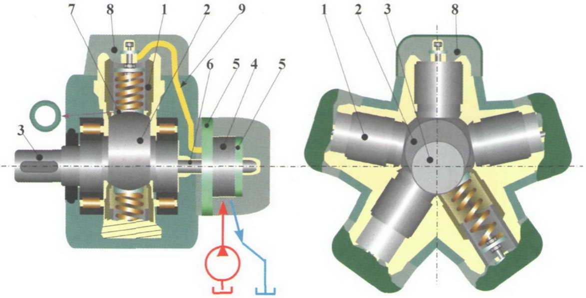
The design of fluid column radial piston motors is similar to that of crankshaft radial piston motors, but they have a different propulsion system. The force on the round eccentric shaft is transmitted only by the fluid column flowing in by means of the rotary distributor and held in the rotary telescopic cylinders. A spring inside the cylinder guarantees the contact between the stator cap and the eccentric shaft (Figure 7.25).

The fluid column prevents most oblique thrusts that cause much friction between the pistons and the eccentric shaft in the previous radial motors, leading to the cylinder ovalisation.

Two metal round rings at the cylinder ends ensure optimum sealing: the pressure inside deforms the margins making them adhere to the opposite surfaces of the eccentric shaft and the stator cap. The incoming and outgoing fluid is controlled by the rotary distributor between two fixed discs and it is operated by a drive shaft solidly connected to the output shaft.

Fluid column radial motors have a single or double row (Figure 7.26 shows a double row motor) in medium versions with displacements of for instance 190 cm^3/rev, starting torque of 90 Nm/bar, nominal pressure of 250 bar and peak pressure of 420 bar, minimum and maximum speed of 1 and 800 rpm, output power of 46 kW; larger versions have displacements of 11000 cm^3/rev, starting torque of 172 Nm/bar, nominal pressure of 200 bar and peak pressure of 350 bar, minimum and maximum speed of 0.5 and 100 rpm, output power of 310 kW. The recommended viscosity range is 30 ÷ 50 mm^2/s.

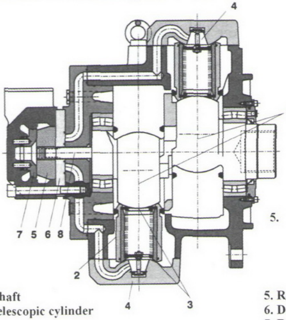
Fluid column radial piston motor



1. Telescopic cylinder 2. Eccentric shaft 3. Output shaft 4. Distributor 5. Fixed disc 6. Distributor drive shaft 7. Spring perforated disc 8. Round cap 9. Inlet/Outlet

Figure 7.25

Fluid column radial piston motor - Double row



1. Eccentric shaft
2. Swinging telescopic cylinder
3. Round cap
4. Cover
5. Rotary distributor
6. Distributor plate
7. Reaction ring
8. Drive shaft

Figure 7.26

All radial piston motors, be them fluid column, double oscillating bush or ball joint versions, demand an external drain line. The hose must be positioned in the dedicated clearance on the upper part of the housing because if it was placed in the lower part it would unload most of the fluid held inside thus hampering the lubrication of mobile parts. In order to avoid the early wear of the sealing ring on the transmission shaft (oil retainer) and an excessive temperature, the maximum pressure inside the motor body (drain) must not exceed 5 bar.

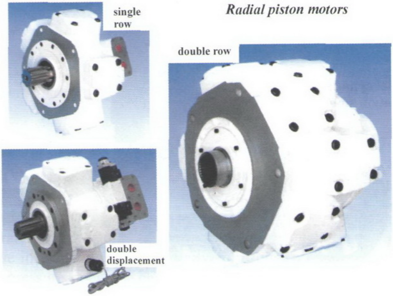


Figure 7.27

Radial piston motors with dual displacement

The displacement of radial piston motors depends on the eccentricity of the cam next to cylinders feet: the more the misalignment between the eccentric shaft and the transmission shaft is, the greater the displacement is; displacement equals zero when the eccentric shaft is in line with the transmission shaft.

A mechanism ensuring the constant variation of displacement would be rather expensive in these actuators. Dual displacement motors can be used if speed has to be increased (maximum displacement reduction) keeping a constant pressure. This type of motors have almost the same design as fixed displacement fluid column or swinging piston motors, but they have a mobile eccentric shaft, or rather, a shaft that can take up two different transverse positions (Figure 7.28 and Figure 7.29).

Two small pistons facing one another and pressing on the clearance of the eccentric shaft are found inside the transmission shaft at the junction with the eccentric shaft. The supply and the keeping of pressurised oil in one of the two cylinders causes the maximum eccentricity, while minimum eccentricity results from the supply to the other cylinder.

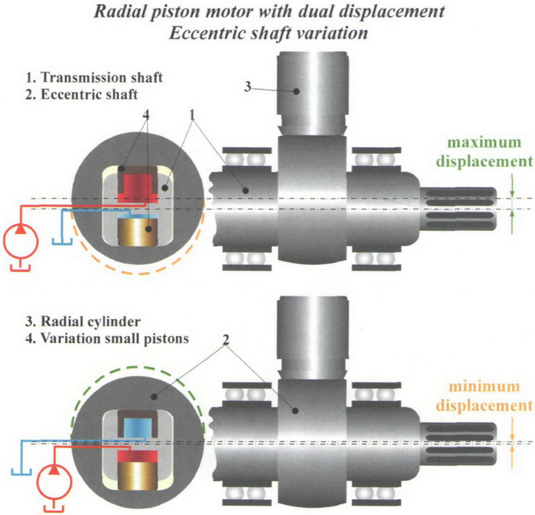
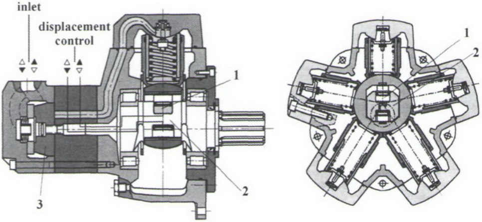


Figure 7.28

Fluid column radial piston motor with dual displacement



1. Variation small piston
2. Drive shaft
3. Distributor

Figure 7.29

Dual displacement piston motors have minimum and maximum displacements from 150-300 cm^3/rev with 5 pistons up to 580-1250 cm^3/rev with 9 pistons or 5400-2700 cm^3/rev with 5 pistons.

Radial piston motor with dual displacement
Pistons swinging on double bushes

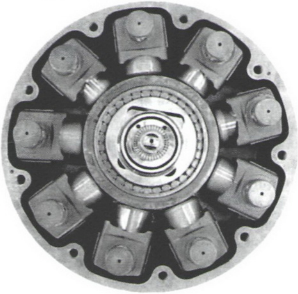


Figure 7.30

PUMP-MOTOR COMBINATIONS

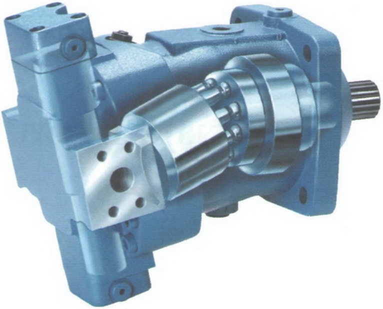
We are now going to go back to what we mentioned at the beginning of this chapter (§ Basics, General features). If the displacement of an oil hydraulic motor stays the same, the increase in the flow the pump delivers results in a higher speed while, loads being equal, the pressure does not change; given a constant inlet, a rise in the motor displacement makes the speed on the shaft decline and, loads being equal again, pressure drops. The opposite conditions cause the opposite results. Even if pressure does not vary, flow reduction makes the shaft slow down; the reduction of the motor displacement entails higher speed and pressure. If flow and displacement are constant, as a variation of the load on the drive shaft makes pressure change (higher force = higher pressure and vice versa), a suitable compensator applied to the pump, to the motor or both guarantees a constant power by varying displacement. Unlike pump compensators, compensators applied to motors make displacement increase (with reduction of the number of revolutions) when pressure goes up.

Like pressure, torque is constant as the power delivered by the pump changes, but it rises when the displacement of the motor increases.

In addition, the displacement of hydraulic motors can be diminished only to a specific limit, usually set to about 1/3 of the maximum effective volume; as a matter of fact, mechanical efficiency is rather poor below this limit and consequently it is impossible to guarantee a tolerable flow/pressure/rotational speed ratio.

The following fundamental points must be taken into account in pump-motor transmissions, in open or closed circuits, in order to establish power/rotational speed ratios:

- ✓ *Fixed displacement pump combined with a fixed displacement motor* – Although it is more economical, this combination delivers poor performances. Once both displacements are set, a constant speed develops on the drive shaft from the minimum to the maximum load value; when the latter is reached, safety valves switch on reducing the number of revolutions quickly.
- ✓ *Variable displacement pump combined with a fixed displacement motor* – Since flow changes do not affect torque and pressure, the displacement variation of the pump causes a rise or drop in speed but it does not affect the force delivered, *as long as the load is constant.* As the displacement of the motor declines too much, the excessive reduction in the flow delivered by the pump makes group efficiency drop dramatically.
- ✓ *Fixed displacement pump combined with a variable displacement motor* – This combination guarantees a constant power on the drive shaft. As a matter of fact, as motor displacement can change if for example the speed of prime mover changes, the product of torque by speed can be constant, resulting in a constant power.
- ✓ *Variable displacement pump combined with a variable displacement motor* – The features of this combination are the sum of the characteristics described so far. Unlike the previous combinations, motor displacement can be reduced considerably: mechanical efficiency is still reasonable if displacement is reduced to 1/6. Yet, it is important to consider whether it is really indispensable because the pump, its compensator and the motor have rather high costs.



Housing and internal details of a bent axis axial piston motor with variable displacement

Figure 7.31

Chapter 8

DIRECTIONAL VALVES

Oil hydraulic systems are basically made up of at least four parts:

- a generator (prime mover - pump - tank)
- a safety valve
- a linear or rotary actuator
- rigid or flexible pipes to connect the parts of the circuit

Fully operational circuits usually have at least one *directional valve* (also known as direction control valve, distribution valve or, generically, distributor), i.e. the control component that sets the actuator in motion, stops it and reverses its direction (Figure 8.1).

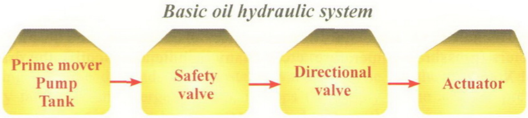


Figure 8.1

DEFINITION OF DIRECTIONAL VALVES

Directional valves have different designs and control systems and they play different roles. However, they all share the following characteristics:

1) Spool
2) Ports
3) Positions

4) Control system
5) Reset system
6) Size

Spool

The spool (or **distributor**) is the mobile element that connects the inlet port (P) to the outlet ports (A, B); the outlets, and in some versions also P, are also connected to the tank (outlet port T).

Ports

Directional valves have some ports that can be open, closed or mutually connected depending on the position of the spool. For instance, two-way valves have an inlet port and an outlet port whereas four-way valves have an inlet port, two outlet ports and a port connected to the tank.

ISO 9461 standard '*Hydraulic fluid power – Identification of valve ports, subplates, control devices and solenoids*' provides that the ports both in drawings and on the external part of the valve body should be marked with different letters. The supply port (pressurised fluid) is marked **P** whereas **T** refers to the outlet port; the ports to the actuator are marked **A** (single port) or **A** and **B** (two ports). External control ports are marked **X** for control A and **Y** for control B.

A special port **V** is dedicated to the pilot device whose action is initiated by venting to a lower pressure. **L** identifies the drain port; an additional port ('M') can be included to take fluid samples for examination and to measure parameters (pressure, temperature, etc.). Many technicians identify the drain of control **X** as **Y**.

Directional valves –Identification of ports– ISO 9461	
Type of port	***Letter of identification***
Supply port (inlet)	P
First working port	A
Second working port	B
Tank port (outlet)	T
First control port	X
Second control port	Y
Pilot lower pressure	V
Drain port	L
Take-off port	M

Positions

The spool can maintain a variable number of stable positions. The most popular valves have two or three positions, but versions with four or more positions too are employed for many reasons. A three-way valve with two positions is referred to as **3/2**, a four-way valve with three positions as **4/3** and so on.

Control system

The task of a control system is to control the spool, which determines the valve position. It can be manual, mechanical, electric, electrohydraulic, pneumatic or provided with a proportional electronic control depending on the type of the force exerted on the spool.

Reset system

Reset systems allow resetting the initial position after the end of the control signal; they can be automatic or result from another control. Automatic reset systems are usually mechanical (spring). Valves having automatic reset are referred to as '**monostable**' valves because they have only one stable position, that is the rest position that can be open or closed (normally open **NO** or normally closed **NC** valve); '**bistable**' valves have two stable positions without external signals, in other words they need an additional control to reposition the valve.

The need for monostable or bistable valves result from both cycle design and security reasons. For instance, a valve provided with an electric control to open and close a clamp must be a bistable valve because it guarantees the closing of the clamp in the event of a power supply failure or an emergency stop; an advance actuator on a working machine demands a monostable valve because, in the event of a power supply failure or an emergency stop, it stops the supply to the actuator preventing movements that could damage the machine and that would be very dangerous for the operator. Bistable valves are used when the position must be maintained also without the control on the pilot, whereas monostable valves are employed if the rest position has to be restored failing inputs. Consequently, a simple input to the pilot causes and maintains the position in bistable valves. Most valves with three or more positions are monostable valves.

Size

Size refers to the dimensions of the valve body for the coupling with the subplate according to ISO standards. Size affects the maximum flow rate allowed and the diameter of the ports of threaded connections.

DISTRIBUTION

Besides the control/reset systems analysed in the following paragraphs, directional valves are essentially made up of the system for flow distribution. A cavity inside the valve body has some clearances properly connected to the external supply/working/outlet/control ports.

The flow of the supply port P is diverted to ports A or B and then it is conveyed to the outlet port T by the spool found inside this cavity.

The spools of automatically managed valves (pneumatic, hydraulic, electric control) in oil hydraulic applications consist of a small shaped cylinder, except for few rotary spools. Manually controlled valves can have spool or seat rotary elements.

Valves with a **poppet** (quite useful in pneumatic power transmission in the 2/2, 3/2, 4/2 versions) are used as non-return and control valves in oil hydraulics; they are rarely employed for direction control only.

Rotary distributor

Rotary distributor valves allow or stop flow by means of the rotation of the mobile part, that is an adequately perforated ball or a small cylinder; the angular rotation is made possible by a manually operated lever.

The simplest rotary valves are two-way **ball valves** (Figure 8.2 and Figure 8.3) with two positions. The ball is set between two pre-compressed seals in the best versions: as pressure rises, seals adhere more to the ball.

Figure 8.2

An adequately perforated ball can result in a directional valve 3/2 or, more rarely, 4/2.

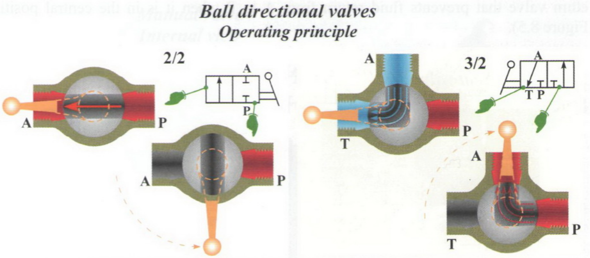


Figure 8.3

Rotary flat slide valves can be 4/3; their control is manual (lever or turning knob). A round plate with proper clearances revolves in its seat connecting the holes on the valve body (Figure 8.4).

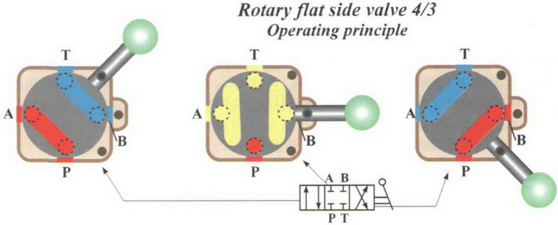


Figure 8.4

Rotary distributor valves (also known as roller valves or piston valves) share the same principle. They consist of a lever that makes a small cylinder rotate; this cylinder has some splines that connect the holes in the valve body.

Seat valve

Some springs keep conical poppets in their seats closed. The cam in the central position blocks the ports; by turning the lever solidly connected to the cam, cam followers push the poppets allowing the fluid to flow. When the lever is in the left position, the supply port P is connected to port A, whereas A is unloaded to the tank if it is in the right position. The poppet with the spring facing upward is actually a non-return valve that prevents fluid return from A to P when it is in the central position (Figure 8.5).

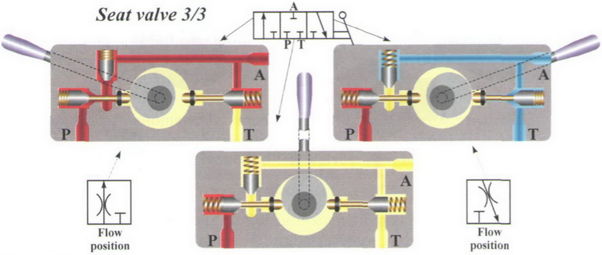


Figure 8.5

Slide valve

Most directional valves in the stationary and mobile fields are slide valves (or spool valves). The word 'slide' in respect of valves derives from the ancient musical organ (Vitruvius, '*De Architectura*', 1st century BC) in which the air/musical pipe connection was operated by an square or rectangular valve 2/2; in other words, its design looked like a furniture drawer sliding on its sliders (Figure 8.6).

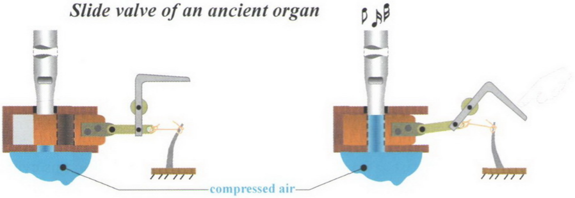


Figure 8.6

Unlike the similar pneumatic valves, oil hydraulic slide valves (Figure 8.7) never have seals between the spool and the seat of the valve body. Sealing is guaranteed by coupling tolerance and fluid viscosity; this implies inevitable leakages between the clearances. If the actuator is blocked and the valve is the central position with closed working ports, it is essential to add a non-return valve. Static seals between the valve seat and the control system are vital too in order to avoid the contamination of the pilot.

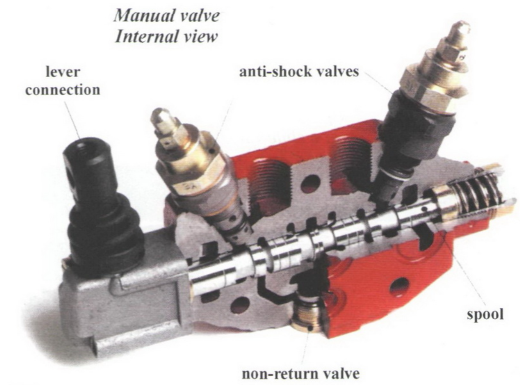


Figure 8.7

Their very simple operating principle is based on the axial movement of a spool on the internal valve body; its movement is the result of the force exerted on the side faces of the distributor by means of the various control systems. Spools have different designs depending on supply, working and outlet ports.

The spool of bistable valves 2/2 (Figure 8.8) ensures the flow from supply port P to working port A in a position, while the other spool position blocks both ports.

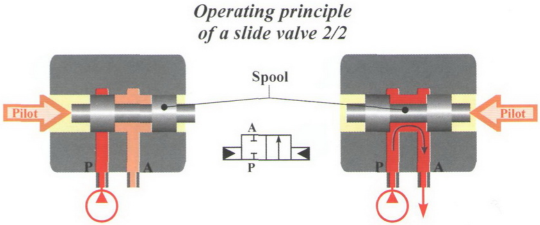


Figure 8.8

The valve 3/2 shown in Figure 8.9, typically used for single-acting cylinder control, is operated by an external (manual, electric, etc.) pilot and by spring return: the movement of the spool connects supply port P with working port A; when the external control switches off, the spring restores the rest position (P is closed and A is connected to T).

In bistable valves both positions result from their respective external controls.

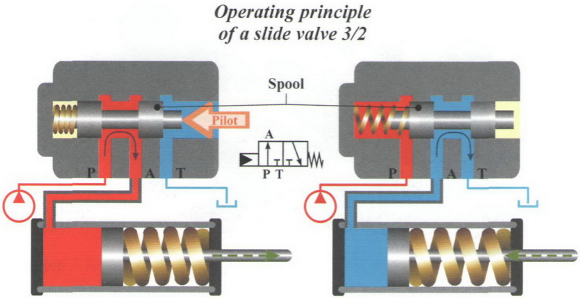


Figure 8.9

The on/off control of differential cylinders (if there are no stops during the stroke) or motors in both directions of revolution demands a valve 4/2 (Figure 8.10). The valve switch connects P to B and A to T or P to A and B to T.

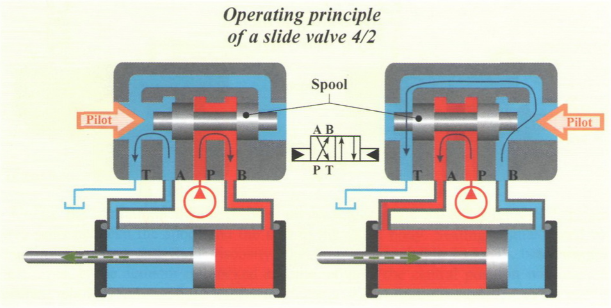


Figure 8.10

Unlike valves 2/2, 3/2 and 4/2 that can usually take up two positions, spool valves 4/3 (Figure 8.11) must perform not only the switch side positions of working ports but also the central position with closed ports (closed centre), or mutually connected ports (open centre or different types of by-pass).

Two opposing springs facing the smaller faces of the spool keep the central position; external controls move the spool in the two opposite directions. The following drawing shows a valve 4/3 with closed centre intermediate position; the next paragraph deals with the spools suitable for different centres.

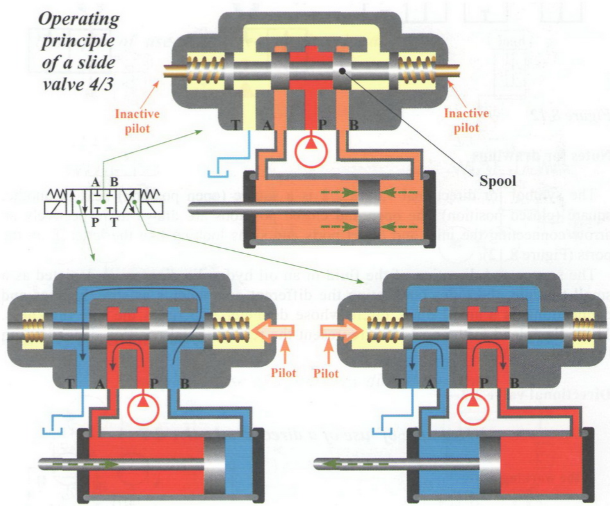


Figure 8.11

PATHS AND POSITIONS IN VALVES

Control parts (switches, buttons, etc.) are drawn only in the rest or initial position in graphic representations of functional electric diagrams. On the contrary, in fluid power diagrams, according to ISO 1219/1 and 1219/2 standards '*Fluid power systems and components -- Graphic symbols and circuit diagrams*', directional valves are drawn in every possible position; this helps to understand circuits better because it is not necessary to know the positions of a wide range of valves by heart (Figure 8.12).

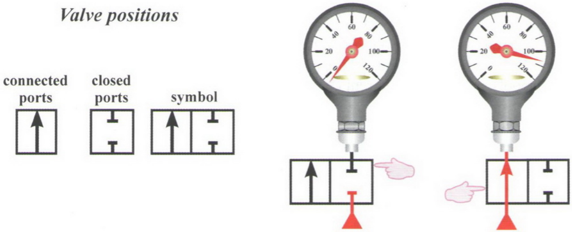


Figure 8.12

Notes for drawings

The symbol for directional valves 2/2 is a square (open position) next to another square (closed position); the open and closed positions are drawn as respectively an arrow connecting the inlet and outlet ports and signs looking like the letter T on the ports (Figure 8.12).

The source and direction of the fluid in an oil hydraulic diagram is depicted as a small triangle, the pipes connecting the different components as straight lines and connections as a small black circle whose diameter is 5 times the line thickness. These lines lead to the ports that represent the rest or initial position depending on the type of valve.

Directional valve 2/2

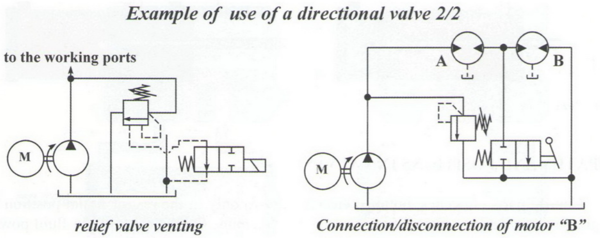


Figure 8.13

Valves 2/2 are employed only for stop purposes. Figure 8.13 shows the typical use of a valve 2/2, in this example with electric control (solenoid) for relief valve venting; the right drawing shows a simple disconnection system of one of the two motors in series via a valve 2/2, in this case with manual control.

Three-way directional valves

Three-way directional valves guarantee the movement of single-acting cylinders. Valves 3/2 are suitable for on/off actuations, i.e. with fully extended or retracted rod; a valve 3/3 is essential if the rod must be stopped in any point of the stroke (Figure 8.14).

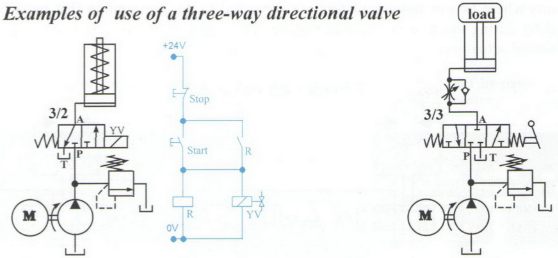


Figure 8.14

Four-way directional valves

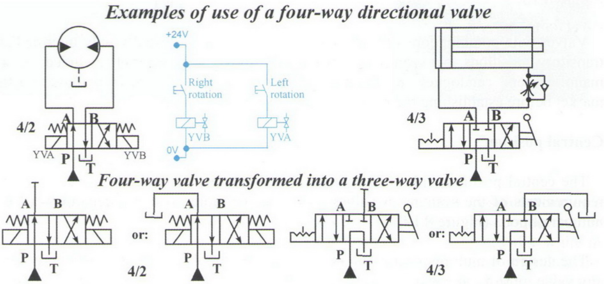


Figure 8.15

Four-way valves with two or three positions (4/2 or 4/3) are used in oil hydraulic circuits for the control of double-acting cylinders and motors (Figure 8.15). The central, open- or closed-centre, by-pass, etc. position depends on the system requirements. A four-way valve can be transformed into a valve 3/2 or 3/3 by blocking only one outlet port.

Valves with a larger number of ways or positions

Some machines often demand directional valves with more than three positions. In addition, the valves of some special applications need more than four ports.

Valves 6/3 (Figure 8.16), in most applications with a lever manual control, are used in circuits where two or more actuators are subjected to the same pump that must be on stand-by during inactive phases (see Chapter 17, § Open Circuits – Series and parallel connected actuators).

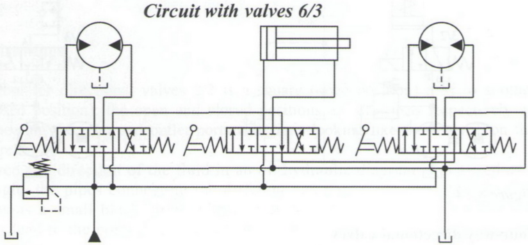


Figure 8.16

Valves with combinations of ports and positions (these are usually only intermediate transitory positions between other positions) above 4/3 are not included in all manufacturers' catalogues and designers should be aware of what is available on the market before establishing the positions and the ports of a valve.

Central position

The central position with open centres, closed centres, by-pass, etc. depends on the requirements of the systems in valves with three or more positions, regardless of the number of ports (Figure 8.17).

The design of multiple centre valves is not much complex: indeed, only the spool of any valve must be adequately shaped, while the valve seat and clearances stay the same. Figure 8.18 outlines the spool design in the central position in the most popular valves 4/3.

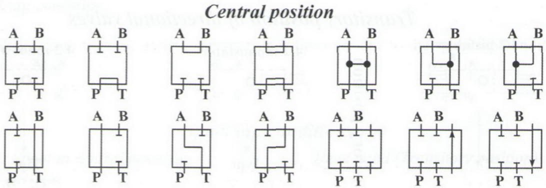


Figure 8.17

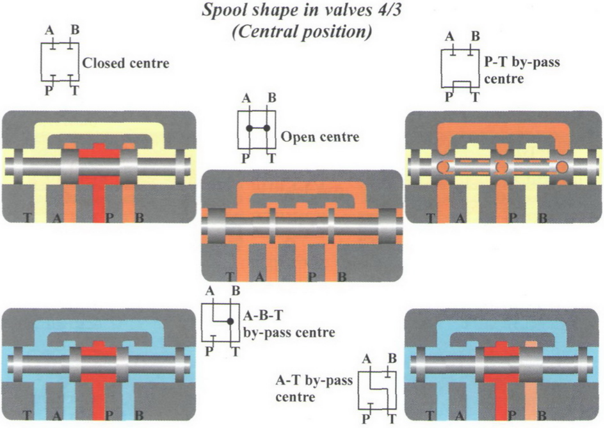


Figure 8.18

Transitory position

According to its design, the spool in valves with two or more positions causes a transitory position when moving from one position to another, thus connecting or blocking some ways. This entails instability for the actuators in many circuits, while in others the functionality of restricted passages in the transitory phase promotes the operator's handling (for instance when handling a suspended load at various speeds); it is therefore important to check transitory intermediate positions in manufacturers' catalogues (Figure 8.19).

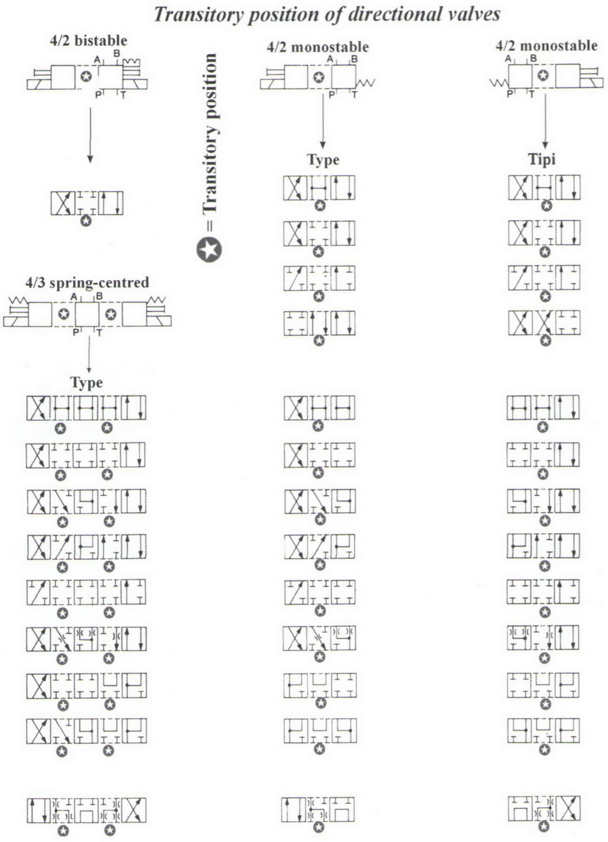

Transitory position of directional valves
4/2 bistable
4/2 monostable
4/2 monostable
= Transitory position
Type
Tipi
4/3 spring-centred
Type
A
B
P
T

Figure 8.19

Spool lap condition

Leakages and, to some extent, transitory positions are due to the spool lap conditions (Figure 8.20).

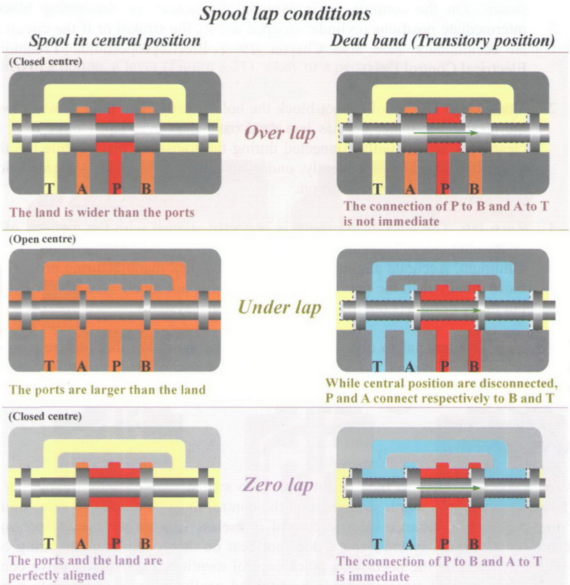


Figure 8.20

By 'spool lap conditions' we mean the type of alignment between the spool and body clearances. While the holes in the valve body stay in the same position, spool lap conditions can have three different styles according to the spool design:

1. **Over lap**. The spool lands are much wider than the holes. This ensures an

optimum sealing but it also causes a long **dead band**, i.e. the stroke the spool travels to open the fluid clearance, with the ensuing intervention delay. Although this is the most popular type of on/off valves, it can cause some problems with the system. For example, the spool in a valve 4/3 with by-pass P-T centre almost blocks the clearances in the dead band, thus forcing the pump to unload via the relief valve and subjecting the circuit to substantial strains. On the contrary, it is essential for actuations demanding blocked intermediate positions (cylinder stopped during the stroke) or if the circuit has an accumulator (see also Chapter 19 § Electroproportional Techniques, Electrical Control Unit).

2. **Under lap**. The spool cannot block the holes totally. The fluid flows between the clearances; under lap has less dead band than over lap. Unlike over lap, clearances are mutually connected during the transitory phase while the fluid keeps on flowing; consequently, under lap cannot be used for systems held in position, accumulators and so on.

3. **Zero lap**. The land of the spool is perfectly aligned with the margins of the inlet/outlet holes. There are almost no leakages; the dead band is reduced to the minimum. Accurate alignments demand a complex manufacturing process and more money than the previous laps; zero lap is not much used in on/off valve but many designers opt for them in proportional systems.

To sum up, it is crucial to choose a valve with the spool lap that best responds to system requirements. If over, under and zero laps play a major role in oil hydraulics, a poor design in on/off actuations will lead to poor performances.

VALVE CONTROL

'Valve control' refers to the control system to switch valves. Before exploring the different systems, it is worth stressing that the control must be quick and safe, suitable for the system (for instance electric control is useless if a simple manual control is suitable for the whole system) and it does not bear on the system manufacturing costs dramatically. Despite the need for a quick control mentioned above, the valve control lever in many manual systems should be handled gently in order to promote a more constricted flow, hence a soft starting.

The spool in small valves can be moved directly by the pilot, that is by a button, a lever or a solenoid. This operation in large valves, dimensioned for high flow rates/pressures, often requires higher forces. The solenoids of solenoid valves will be rather large, demanding much electricity and entailing overheating and vibrations.

For this reason, manually controlled valves are provided with long levers so as to reduce the muscular effort whereas solenoid valves, pneumatically or hydraulically controlled valves are servo-controlled. Both directly-operated and servo-controlled directional valves are controlled by various systems that are always operated by a worker via a simple button, a pedal or a PC (personal computer) or PLC (programmable logic controller).

Manual operation

'Manual operation' refers to the valve direct control exerted by the muscular force applied to a button, a lever (Figure 8.21), a key or a pedal pilot.

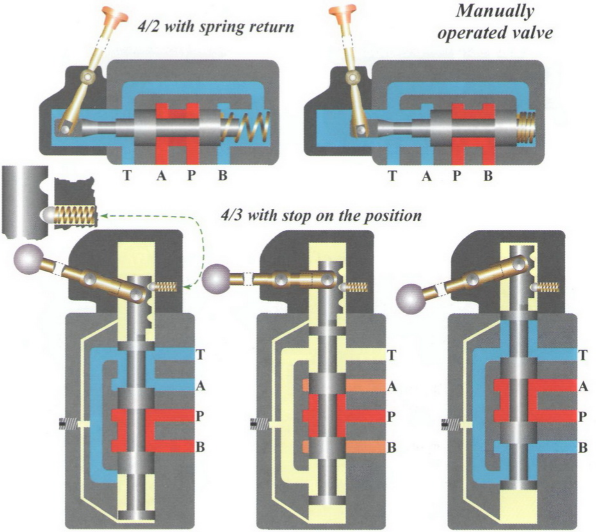


Figure 8.21

In order to promote control compactness (Figure 8.22 and Figure 8.23), all the valves can be held in a monoblock along with the other valves, like relief valves, control valves, non-return valves, etc. (some valves in Figure 8.23 have double manual and electric control).

Monoblock manual valves

Figure 8.22

Overview of monoblock valves

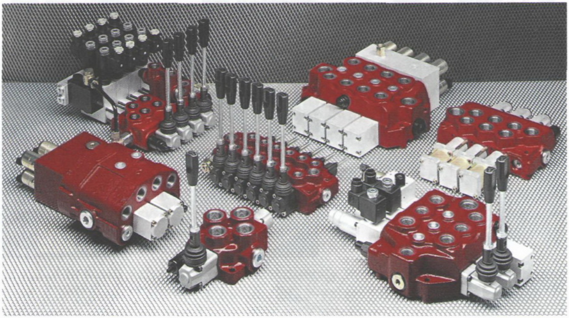
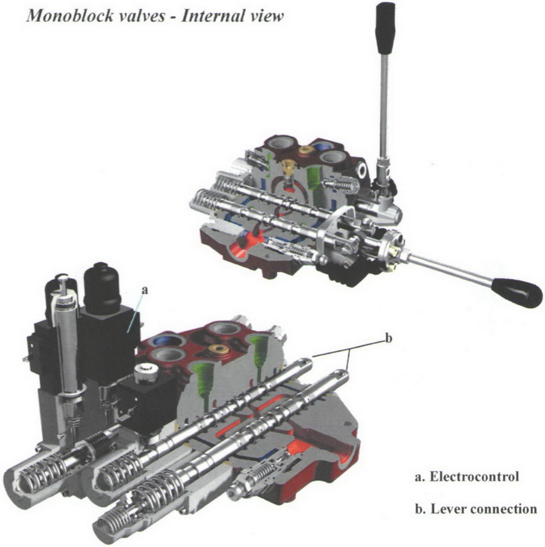


Figure 8.23

Figure 8.24 shows the standard internal design of lever monoblock valves.

Monoblock valves - Internal view



a. Electrocontrol

b. Lever connection

Figure 8.24

Mechanical operation

Mechanical operated valves have small units at one end that control switching in working machines (Figure 8.25). The operation of the pilot mechanical unit is due to cams and hinges well positioned on the machine. The rod of the actuator cylinder is often equipped with an operation cam. Mechanical operated valves are best known as **hydraulic limit switches**.

Mechanically operated valve (limit switch)

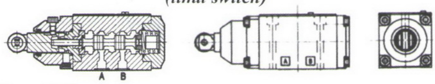


Figure 8.25

Hydraulic limit switches can be arranged in tiny spaces thanks to their compact size; they are precise and quick, but they cannot compare with electric valves, especially if these electric devices are proximity switches, photocells, optoelectronic sensors, optical fibres, magnetostrictives or any other component without mechanical operation.

Mechanical actuators are usually employed for monostable valves 3/2, although it is not unusual for them to be applied to mono and bistable valves 4/2 as well (Figure 8.26). They must be used only as signals controlling the directional valve and they can be operated by a plunger, a ball, and a bi-directional or unidirectional roller lever. Hydraulic limit switches are mainly used to enhance the safety level. As a matter of fact, they compensate for power supply failures, electrocontrols, PLC breakdowns and so on: as a result, when the load is moving (hydraulic energy is still acting), limit switches block the movement.

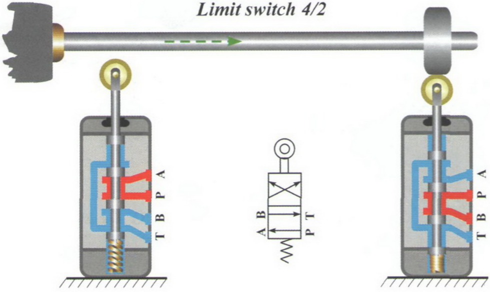


Figure 8.26

Pneumatic operation

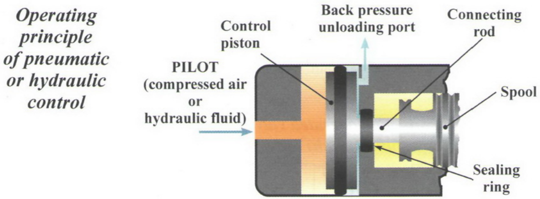


Figure 8.27

Valve control can also be performed by means of compressed air. The pilot air is pushed into the valve against a piston with a suitable area; the piston is larger than the spool and connected to it. As the spool moves, the fluid switches from one port to another; the seals between the spool and the rod prevent fluid leakages to the pneumatic area (Figure 8.27).

For operations demanding much energy, the pneumatic valve controls the main (two-stage) valve by transferring the pressurised fluid to the larger spool ends (Figure 8.28).

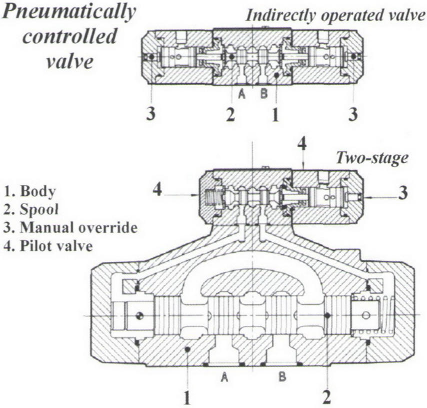


Figure 8.28

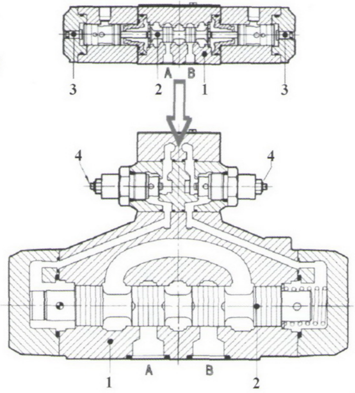
Hydraulic pneumatic valves are very useful in explosive and damp or humid environments where electric controls cannot be used for safety zone reasons.

Pneumatic pilot valves are also employed for lift platforms (for instance car lifts in car repair shops): at a safe distance from the platform, the pneumatic buttons 3/2 (lifting - lowering) are on the control panel and they are obviously connected to the FRL unit (compressed air treatment unit) and the compressor; a gear pump in parallel with an emergency manual pump is usually placed below the control panel. The delivery pipe of the pump and pneumatic control pipes held in a raceway are connected, next to the platform, to the valves that operate the hydraulic cylinders for lifting, shifting and clamping. Possible damage to Rilsan pneumatic pipes due to falling objects, friction, etc. does not endanger operators working next to the equipment; on the contrary, if there were solenoid valves, torn electric conductors (if there is high voltage) would expose them to a serious danger due to the same reasons.

Hydraulic operation

Hydraulically operated valves are similar to two-stage pneumatic valves, but the diameter of their piston is smaller because pilot pressure is higher. They are usually provided with an adjustable restrictor valve that reduces the flow pressing on the main spool by constricting the clearance; this ensures a soft starting (Figure 8.29). Hydraulic pilot signals normally result from the movement of limit switches on the machine.

Two-stage valve with hydraulic control



1. Body
2. Spool
3. Manual override
4. Brakes

Figure 8.29

SOLENOID VALVES

Since the control of on/off directional valves by means of electric coils is the most complex system, it requires an in-depth analysis.

Solenoid control enables the spool movement due to a precise voltage applied to it; therefore, this process comes down to the transformation of an electric signal into a hydraulic signal. In fluid power, the types of transformation of such a signal are divided into *direct operation* (the mobile core operated by the solenoid connects hydraulic directional ways directly) and *indirect operation* (the principle of *two-stage servo-control* is applied).

The solenoid valves analysed in this chapter refer to on/off controls only. In systems with electronic control unit, the solenoid unit and the spool must be designed in a different manner in order to satisfy special needs.

On/off solenoid

In solenoids, electromagnetic energy is transformed into mechanical energy in order to perform the movement of the spool held in the hydraulic valve. Basically, solenoids are made up of a *coil*, i.e. a copper insulated conductor wound in a spiral on a pipe (or *sleeve*) inside which a cylindrical *core* made of ductile iron slides. The excitation resulting from the voltage conveyed to the coil ends trigger a magnetic field that moves the core axially. The element (air or oil) between the core and the coil is called '*gap*'.
This is the general operating principle; however, the following measures are needed to guarantee the device works properly.

Traditional design solenoid

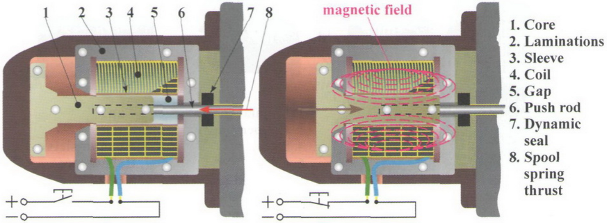


Figure 8.30

Oil hydraulic solenoid valves can have a 'traditional' or 'wet' solenoid (the latter is the most popular, whereas the former is seldom used).

In **traditional** solenoids (Figure 8.30), the coil wound around the sleeve (3) is surrounded by some iron *lamination* stack (2) that reduces leakage current and dissipate the heat it generates. The mobile core (1) sliding in the coils is rigidly and axially connected to the '*push rod*' (6) that presses on the valve spool. When the solenoid is not excited, the spool spring keeps the core in the retracted position; the air fills the gap (5) between the coil, the space inside it and the core. When it is excited, a magnetic field between the coil and the air gap develops and the core is attracted by the coil because the core iron has more magnetic conductivity than air. As a result, the air gap diminishes and the mobile core keeps this position as long as the solenoid is excited.

It is crucial to prevent the hydraulic fluid from leaking into the air gap (between the core and the coil) in traditional solenoids. A dynamic seal between the valve and the solenoid, inside which the push rod slides, is used for this purpose; yet, as it is subjected to wear, it cannot guarantee a long working life of the component.

Modern **wet** solenoids are different because the fluid of the hydraulic circuit (flowing from the clearance of outlet port T) enters the device in order to fill the gap. The fluid flowing lubricates the parts and what is more it enhances cooling; it is clear that dynamic seals are totally useless in oil bath solenoids. These solenoids obviously need designs and material other than traditional solenoids.

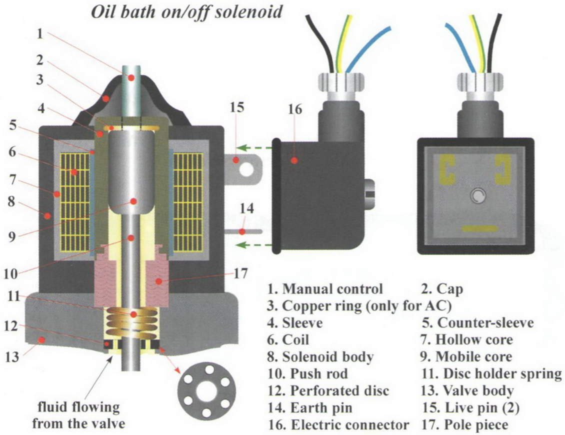


Figure 8.31

The coil (see Figure 8.31) is a copper winding (6) insulated by a very thin and special transparent film; the coil is between the hollow core (7) made of ductile iron and counter-sleeve (5). This unit, held in the body (8) made of insulating material, is next to the sleeve (4). At the lower end of the sleeve there is another tube-shaped component called 'pole piece' (17) on which the core rests during the excitation and whose task is to improve the gap closing.

When the coil (4) fed through a connector (16) and the two pins (14) is excited, the mobile core (9) slides on the sleeve (4) and pushes the spool in the valve body (13) via the push rod (10).

The perforated disc (12), held in position by the opposing spring (11), allows a suitable amount of fluid to flow from the outlet port into the sleeve in order to lubricate and cool the solenoid down.

In emergency situations (power supply failure) or during the adjustment of the system, the solenoid valve can be operated manually by acting on the manual control (1) (not all solenoid valves are provided with it) covered with a boot protected cover (2) and equipped with seals to prevent cooling fluid leakages from the sleeve. This emergency system in some versions is safe from accidental shocks and interventions by unskilled staff: the control (1) fits in the body (8) and it can be operated only by means of a proper tool.

In solenoid valves with alternating current, the mobile core is attracted and released as many times as the positive and negative waves of the sinusoid (50Hz > 100 movements per second of the mobile core). This causes the buzzing of the component and the mobile core subjected to so many shocks wears out quickly.

By adding a copper ring (-3- in Figure 8.31), an induced current develops resulting in a 90° phase shift of the magnetic field; for this reason, the attractive force developed by the two magnetic fields never equals zero.

The electrical coil in most solenoids on the market can be replaced very easily and this operation can be carried out with pressurised circuit because the sleeve is watertight and solidly connected to the valve body (Figure 8.32).

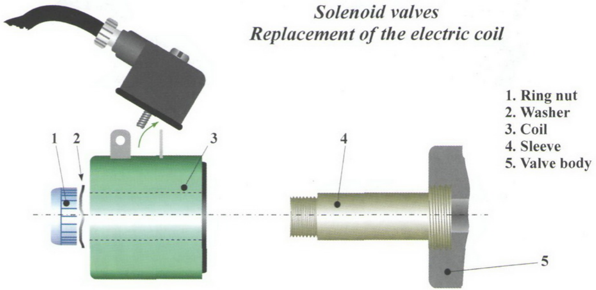


Figure 8.32

Directly operated solenoid valves

The substantial currents, needed to operate an electromagnet subjected to pressurised fluid, confine the use of directly operated solenoid valves only to small and medium versions.

In Figures 8.33 and 8.34, the only task of disc holder springs is to hold the perforated disc in its positions.

Although all manufacturers comply with the standards in force, each of them conceives its own design by taking both its functionality and manufacturing costs into account.

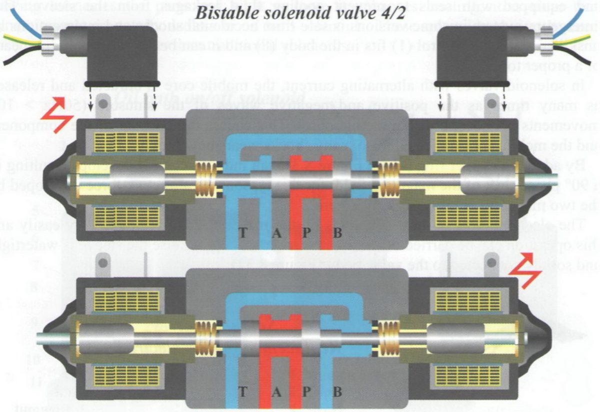


Figure 8.33

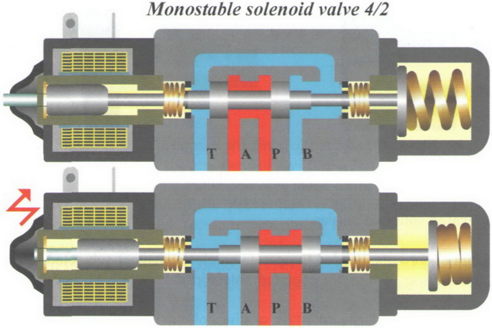


Figure 8.34

In general, as wet solenoids require the outlet/sleeve connection and no valve body changes (the same position of the clearances for any valve with the same number of ports but with different positions), it is worth using spool leans against perforated discs. This means that springs in valves with two positions play only this role, while the springs in valves with three positions (closed centre or in the multiple by-pass connections) also centre the spool.

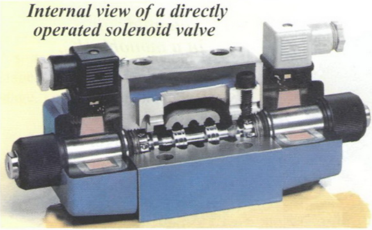


Figure 8.35

In spring-centred valves 4/3 (Figure 8.36), mobile cores are also provided with additional opposing springs that keep them retracted when they are not excited.

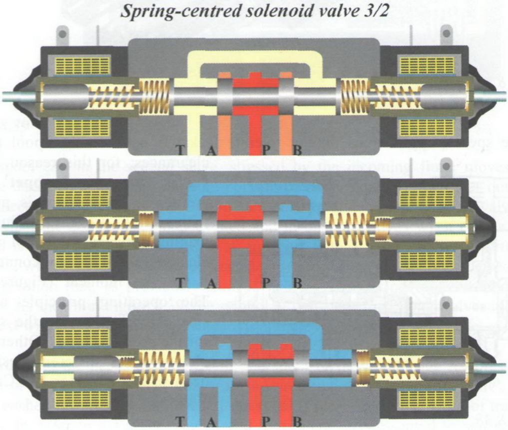


Figure 8.36

An interesting solenoid valve 4/3, combined with a device to neutralise fluid hammers, is analysed in Chapter 16, § Hydraulic Oscillation.

Like lever-operated valves, also some solenoid valves can be held in a monoblock (Figure 8.37):

Figure 8.37

Some special applications do not tolerate leakages between the spool and the clearances; for this reason, special valves known as '**poppet**' valves are used. Their spool is provided with seals to prevent the fluid from leaking between the spool and the clearances that are not connected to it for the moment (Figure 8.38). The operating principle and the sleeve lubrication are the same as for the previous types. Other poppet plugs with a different design are dealt with in Chapter 12, 'Cartridge Valves'.

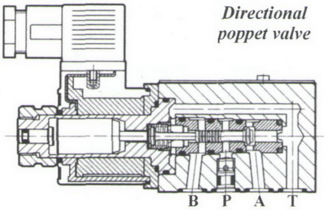


Figure 8.38

Two-stage (indirectly operated) solenoid valves

Indirect operation or *servo-control* should not be mistaken for the homonymous system used in pneumatic power transmission in which the excitation of the solenoid opens the clearance allowing pilot air that presses on the single spool.

In oil hydraulics, except for the control of valves via a pneumatic signal, this method is not effective enough; hydraulic servo-control is made possible by the *two-stage* system. When the solenoid (i.e. the controlled solenoid valve) is excited at the first stage, the corresponding spool opens the clearance so that the fluid can reach the second stage, i.e. the main directional valve (Figure 8.39).

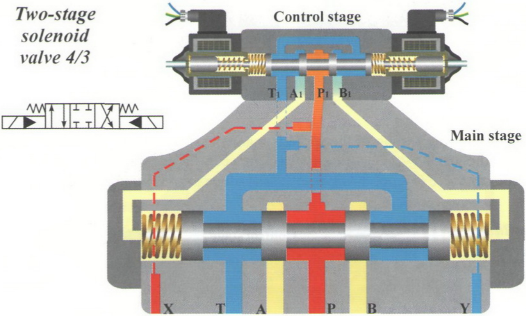


Figure 8.39

The spool end in the second stage, stressed by the incoming fluid, moves in the opposite direction connecting pressure port (P) to outlet port (B); the other outlet port (A) connects to the tank port (T). During the return (operated in monostable valves with spring when the coil is no longer excited and in bistable valves when only a second solenoid is excited) P is connected to A and port B to T. This applies to directional valves 4/2 whereas the operations described above in valves 4/3 are interspersed with a central position that, as previously said, is held by two springs and be in different versions. As shown in Figure 8.39, the symbol of two-stage directional valves is a single valve next within the depictions of solenoids and the hydraulic operation.

According to the needs of the system, the spool of the main stage requires a high, medium or low transfer speed; this process is referred to as '**response time**'. As the fluid is pushed towards the flat faces of the spool (Figure 8.39), the speed of transfer is medium. In order to get a very high-speed movement, it is essential to add two **pilot pistons** inside the valve that match the flat faces of the spool (Figure 8.40). Speed drops because the fluid flowing from the first pilot stage does not act on the whole flat face of the spool any longer but on the tiny surface of the piston. Still, the inclusion of this component demands the right evaluation of pilot pressure: if it was set in proportion to the spool surface, it would not be sufficient for the pilot piston.

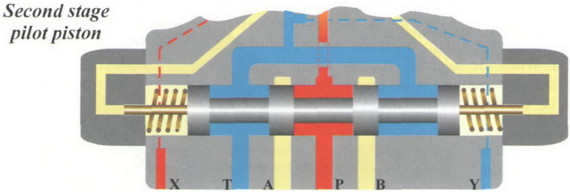


Figure 8.40

The slow adjustable sliding of the spool is made possible by two **restrictors** between the first and the second stage, to be precise between the clearances A_1 and B_1 of the solenoid control valve (Figure 8.41); restrictors reduce pilot flow to the main spool.

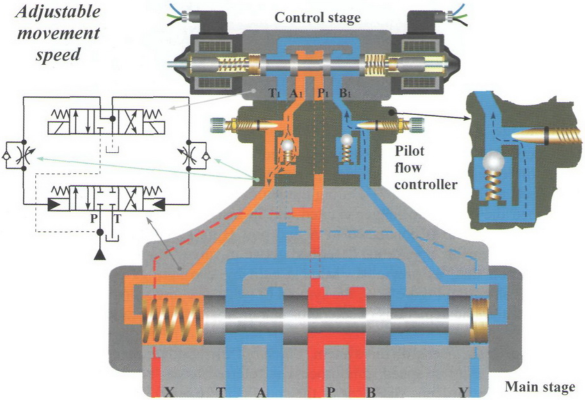


Figure 8.41

The constriction can be simple or with non-return valve depending on the version. In the first type, flow is limited in both clearances A_1 and B_1. By adding a non-return valve, only the outlet way is constricted, developing an outlet back pressure that slows the spool down by reducing pilot flow. Non-return valves are obviously arranged on both pilot channels A_1 and B_1: when P_1 is connected to A_1, the pressurised fluid opens the non-return valve flowing in a fully open clearance , whereas the fluid diverted to B_1 enters the constriction; when the control is switched (P_1 to B_1,) the opposite situation develops (see also 'Auxiliary valves').

The **stroke adjustment** of the main spool determines its maximum movement with the ensuing constriction between supply and working ports (Figure 8.42). In this manner flow to the actuator can be reduced without adding additional flow control valves; this method clearly allows only rather rough settings.

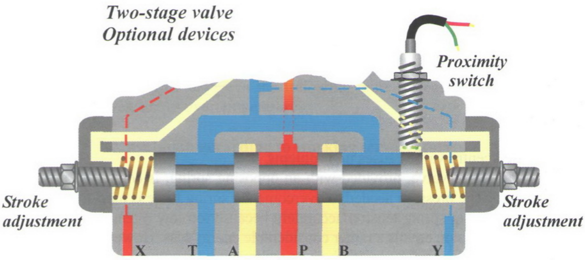


Figure 8.42

Many manufacturers provide an optional proximity switch, usually inductive type (Figure 8.42) next to one end of the (either directly- or indirectly-operated) spool. The switch detects the spool position and sends the positioning end signal to the control unit.

Servo-controlling valves with by-pass centre

Usually ranging from 4 to 10 bar at least , the pilot pressure of directional valves 4/3 with closed centre (a) is ensured in every operating phase because the main supply port P is always at working pressure.

As supply pressure equals almost zero during the central position (P is connected to T) in valves with by-pass P-T centre, the electric command of the solenoid cannot operate the hydraulic control. As a result, a precise pressure must be developed somehow in order to operate the control. Figure 8.43 shows how to solve this problem without pressure on port P of the second stage.

Servo-control with by-pass P-T

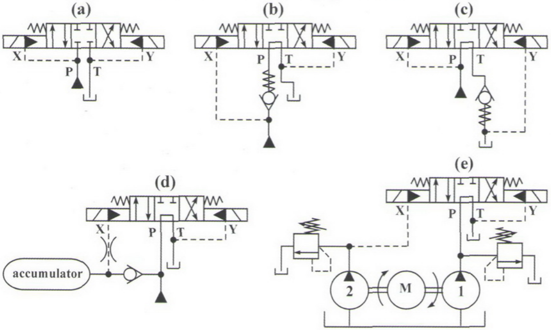


Figure 8.43

a) A closed centre valve causes no problems.

b) The non-return valve, provided with an opposing spring, is connected in series with the pump. The pressure needed to overcome the spring force is used for the control. It is crucial to set spring tension accurately because too low a tension is useless while too high a tension entails a rather considerable energy loss. However, such a system is subjected to a constant dissipation throughout its operational phase. Manufacturers provide an optional arrangement of this device inside the valve body.

c) It is similar to the previous type but it has a spring on the non-return valve in series with the outlet port. The same remarks and optional device as point b) apply to this type.

d) An accumulator neutralises energy loss. This device (see Chapter 14) drives back the fluid stored during the operational phases. As soon as the solenoid operates the spool of the first stage, oil flows from the accumulator to the pilot port. It is essential to include a restrictor because the accumulator pressure must be much higher than pilot pressure. The system does not always yield the expected result: the fluid stored could be insufficient if there are several valves and a single accumulator.

e) This is the most rational solution. Controls are supplied by another low-displacement pump (marked as 2) that delivers a medium pressure (a small external gear pump is more than enough); the additional pump is coaxial with the main pump and equipped with a relief valve set to the pilot pressure. Pressure is always constant and different valve units can be supplied by dimensioning displacement correctly.

Standard voltage and current requirement

The solenoids of directional solenoid valves can need direct current (DC) or alternating current (AC). DC standard versions have voltages of 12V, 24V, 48V, 110V, while the voltages of AC standard versions with AC-50 Hz equal 24V, 48V, 110V, 220V.

The current absorbed depends on nominal voltage and the size of solenoid valves. For example, the current absorbed by a small-sized 03 valve with 12V DC equals 2.5÷3A (depending on its design); the inrush and operating currents are respectively 8.5÷9A and 2.2÷2.5A in 24V AC versions. The current absorbed by a 05 valve with 12V DC is 3.5÷4A, whereas, the inrush and operating currents are respectively 19÷20A and 3.5÷4A in 24V AC versions (see 'Insertion' for inrush currents). The current absorbed by two-stage versions amounts to similar levels because pilot solenoid valves have the same sizes.

Insulation class

CEI standards establish the insulation class according to the working temperature of solenoids. The temperatures shown in the following table for each class code refer to working temperatures, not to overtemperatures.

Insulation according to working temperature
15 – 26 CEI Standard

Insulation class	*Temperature °C*
Y	90
A	105
E	120
B	130
F	155
H	180
200	200
220	220
250	250

Insertion

'Insertion' refers to the total time the solenoid is excited, i.e. from the moment voltage is applied to the coil ends until the contact is switched off.

Solenoid valves need AC in old-style PLC or traditional electromechanical controls whereas DC is almost always employed in present serial systems.

The power the solenoid absorbs is measured in Watt. This is equivalent to the product of voltage V by current I in DC; phase angle φ must be taken into account in AC.

DC $\rightarrow$ P = V • I (Watt) AC $\rightarrow$ P = V • I (VA) V • I • cos φ (Watt)

When dimensioning AC electric systems (especially the diameter of conductors and the size of the optional transformer), it is essential to consider that the inrush current absorbed by the solenoid is much higher than the operating current.

The attractive force depends on the level of inductance of the coil. As it is inversely proportional to the current, with the maximum gap (initial position), the higher the current, the lower inductance. At the beginning of the excitation, the inductive resistance vis-à-vis the initial position of the mobile core is minimum, as a result the movement demands much current. As the mobile core moves, the inductive resistance increases and current drops.

Consequently, as the inrush power must be higher, the identification plate of AC solenoid valves specifies inrush current and operating current. The dimensioning of the electric system (diameter of conductors and power of the optional transformer) should be consistent with the maximum number of coils acting simultaneously.

Continuous rating

Continuous rating is the maximum insertion time solenoids can sustain. The code 'ED 100%' stands for continuous rating and a lower percentage refers to an intermittent rating, that is alternating working and rest time.

$$ED = \frac{\text{Insertion time}}{\text{Insertion time} + \text{Rest time}} \bullet 100$$

$$\text{For instance } ED = \frac{10'\,(\text{insertion time})}{10'\,(\text{insertion time}) + 10'\,(\text{rest time})} \bullet 100 = ED\ 50\%$$

IP Protection Degree

The EN 60529 standard sets the protection degrees of electric devices against accidental contacts, ingress of liquids and foreign objects.

Degrees consist in two digits preceded by 'IP'; the first digit (from 0 to 6) relates to the degree of protection against accidental contacts of the human body and foreign solid objects, the second digit (from 0 to 8) to water protection. For instance, IP 65 devices are protected against the ingress of dust and any other tiny object (6), as well as water jets (5).

DIN and IEC international standards too should be considered during the design phase.

IP Protection Degree EN 60529			
Protection against accidental contacts and solid foreign objects		***Water protection***	
First digit		*Second digit*	
0	Non-protected	0	Non-protected
1	Protected against solid objects up to 50mm; hands	1	Protected against vertically falling drops of water
2	Protected against solid objects up to 12mm, e.g. fingers	2	Protected against dripping water when tilted up to 15°
3	Protected against solid objects over 2.5mm	3	Protected against spraying water
4	Protected against solid objects over 1mm	4	Protected against splashing water
5	Protected against dust limited ingress	5	Protected against water jets
6	Totally protected against dust	6	Protected against heavy seas
		7	Protected against the effects of immersion
		8	Protected against submersion

Special structures

In *explosive environments* the sparks due to anomalies can damage people and equipment very seriously. It is advisable to use pneumatic control valves in simple systems; the components subjected to electricity in complex systems demanding electric/electronic mechanisms must be held in suitable *flameproof boxes* and controls must be *fail-safe*. The last paragraph of Chapter 17 further analyses this subject.

The use of electronic technology in fluid power requires constant up-to-date electronic control components. Solenoid valves, which are more and more employed in PC or PLC serial systems, must be energy-efficient, compact, quick-response and long-lived devices.

DISTINGUISHING TRAITS

One- and two-stage valves and solenoid valves are manufactured in different sizes established by ISO standards. These sizes set the valve dimensions depending on maximum flow/pressure.

Flow/pressure characteristics

Valve size affects pressure and flow, which vary depending on the type of spool, size being equal. Manufacturers' catalogues always specify these data. Figure 8.44 shows some illustrative examples of directly operated 03 and 05 valves, as well as two-stage 07 and 10 valves.

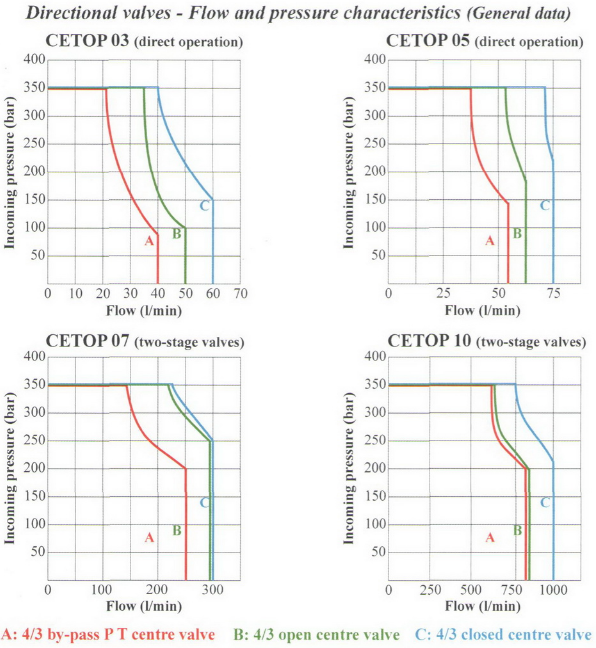


Figure 8.44

Operation ranges are related to the limits of the forces of the solenoids and centring springs. The forces developed by pressure drops on the spools usually tend to close them. As a matter of fact, the solenoid is not powerful enough to maintain the position of the spool, therefore it yields, then opens again when there is no flow and so on, thus causing vibration. On the contrary, for instance in single-solenoid solenoid valves, when there is flow only in one of the P-A o B-T connections (or vice versa), pressure drops can support the solenoid but hamper the centring spring; consequently, the spring force is not sufficient to the centre spool and the load shifts.

Pressure drop

In directional valves the clearances in the body, the spool shape and leakages develop a differential pressure Δp between inlet and outlet ports. These pressure drops are affected by flow in every size version. Figure 8.45 shows some illustrative examples about this phenomenon.

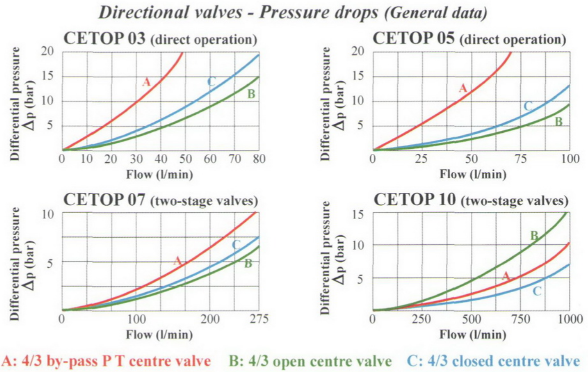


Figure 8.45

It is important to highlight that the data in the charts refer to tests carried out on a fluid with a viscosity of 36 cSt and at a temperature of 50 °C and Δp can seem quite irrelevant in this situation. When temperature (hence viscosity) changes, Δp rises: at low temperatures the pressure drop due to high viscosity is almost negligible (as fluid motion is turbulent, viscosity is not very relevant), while high temperatures make leakages increase dramatically. Even if temperature does not change, the number of valves must be taken into account: each valve entails a pressure drop and overall Δp has to be determined by adding all of them.

Sizes

ISO 5783 standard sets valve size according to the diameter of the main ports (P, T, A, B). Sizes are identified as a number ranging from 00 to 13. For instance, size 02 stands for a small valve while 10 refers to a valve that can transfer substantial flows.

The sizes usually employed in oil hydraulics range between 03 and 10. Most companies manufacture directly-operated 03 and 05 valves and two-stage valves with 05, 07, 08, 10 (pilot 03 or 05) sizes (main stage).

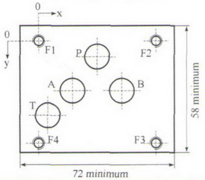
As the European Fluid Power Committee established these dimensions before the ISO, it is still common practice to use the CETOP code followed by the number mentioned above in order to identify valve sizes.

Mounting surface

Valves must be coupled carefully with their subplate on which there are holes for pipe fittings.

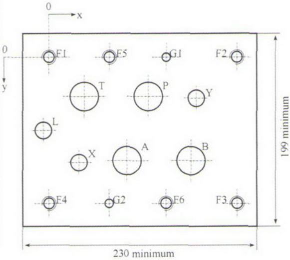
Examples of mounting surfaces according to ISO 4401

Size 05 without pilot ports



Axes	P	A	T	B	F1	F2	F3	F4
	Diameter 11,2 max	Diameter 11,2 max	Diameter 11,2 max	Diameter 11,2 max	M6	M6	M6	M6
x	27	16,7	3,2	37,3	0	54	54	0
y	6,3	21,4	32,5	21,4	0	0	46	46

Size 10 with drain and pilot ports



Axes	P	A	T	B	L	X	Y	G1	G2	F1	F2	F3	F4	F5	F6
	Diameter 32 max	Diameter 32 max	Diameter 32 max	Diameter 32 max	Diameter 11,2 max	Diameter 11,2 max	Diameter 11,2 max	Diameter 11,2 max	Diameter 11,2 max	M20	M20	M20	M20	M20	M20
x	114,3	82,5	41,3	147,6	0	41,3	168,3	138,6	41,3	0	190,5	190,5	0	76,2	114,3
y	35	123,8	35	123,8	79,4	130,2	44,5	0	158,8	0	0	158,8	158,8	0	158,8

Figure 8.46

ISO 4401 sets these standard accurate combinations and ensures **interchangeability**, i.e. the ability to replace a damaged or worn-out component with a same-sized component manufactured by a different company that respects this standard.

The ISO 4401 standard sets the dimensions of the **mounting surface**. Every size has its own surface dimensions, distance and position between holes, width of ports (by the way already established by ISO 5783), threaded holes for fixing screws and cavities for coupling plugs between the valve and the subplate. Besides letters for hydraulic ports

(P, T, A, B, X, Y, L), the letter 'F' always followed by a digit identifies threaded holes for fixing screws and G, followed by a digit if there is more than one hole, stands for the position of the cavities for coupling plugs (Figure 8.46).

ISO 4401 also establishes the tolerances of mounting surfaces, characterised by a maximum roughness of 0.8 μm (according to ISO 4287 and ISO 1302), surface flatness of 0.01 mm per 100 mm (according to ISO 1101), H12 cavity diameter for coupling plugs. The distances of holes vis-à-vis mounting surfaces cannot exceed the tolerances of 0.1 mm for plugs and fixing holes, 0.2 mm for port holes. Since the standard does not spell out working pressure limits, manufacturers are expected to set them freely.

Manufacturers that strictly abide to these standards can publish the following sentence in their commercial documentation: '*Mounting surface dimensions conform to ISO 4401:2005, Hydraulic fluid power – Four-port directional control valves – Mounting surfaces*'.

Anomalies

The problems that may occur in the operation of directional valves are mainly due to spool malfunction. Available valves certainly guarantee high-level performances, but they require accurate installation and maintenance.

Excessive screw tightening on the mounting surface deforms the body, thus blocking the spool (recent standards have provided that screw heads should be deeper, so that tightening force is applied on a lower point than the spool axis); contaminated fluids deposit erosive particles inside the valve body and this damages the spool that keeps on sliding on the body. If the temperature system exceeds 70 °C most of time, this deforms the spool, albeit temporarily, while seals bake and change permanently. In addition, spools are very likely to seize up due to high temperatures because most low-quality valves are made of poor materials.

Stack valves or sandwich valves

Valves or solenoid valves are assembled in a single block in systems requiring various controls. Figures 8.47, 8.48 and 8.49 show some types of assembly, but it is evident the use of sandwich valves depends on the space available and pressure/flow ratios.

Besides the safety valve (at the end of every unit in the previous Figures), circuits often need several different valves, in series with the directional valve. As each of them has a mounting surface complying with ISO standards, they can be interfaced by putting them together in a single compact block.

Stack valves

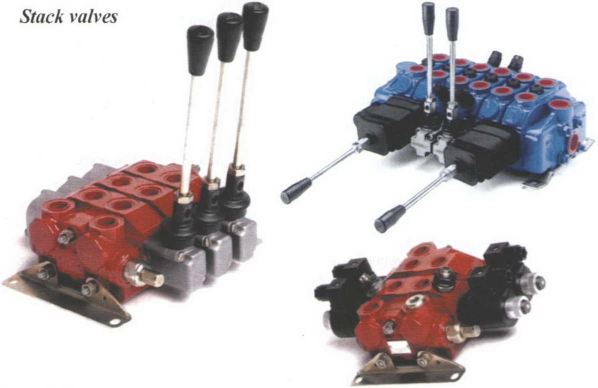


Figure 8.47

More stack valves

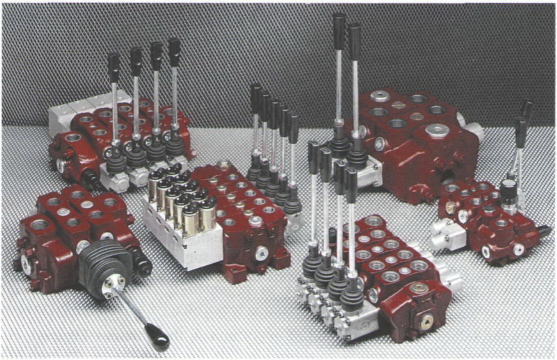


Figure 8.48

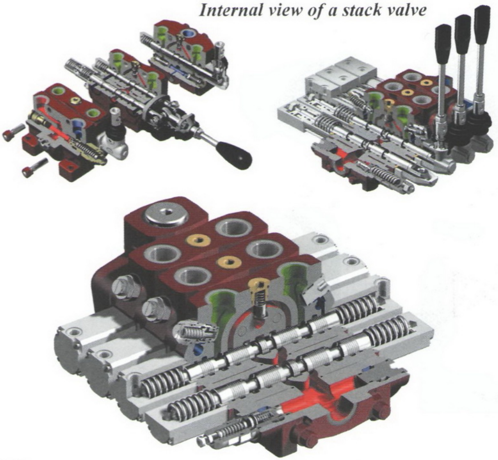


Figure 8.49

Figure 8.50 shows a sandwich valve made up of (from the top to the bottom) an solenoid valve, a double check valve, a double one-way flow control valve, a relief valve, a pressure reducing valve, a double counterbalance valve.

Mounting surfaces conforming to ISO standards are shown in the lower part of Figure 8.51, which depicts different types of sandwich valves.

Sandwich valves

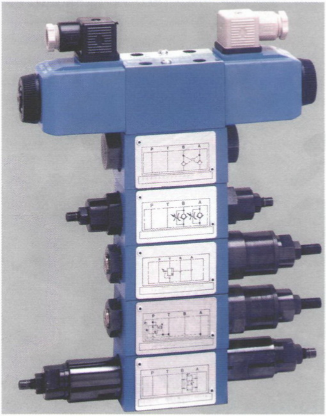


Figure 8.50

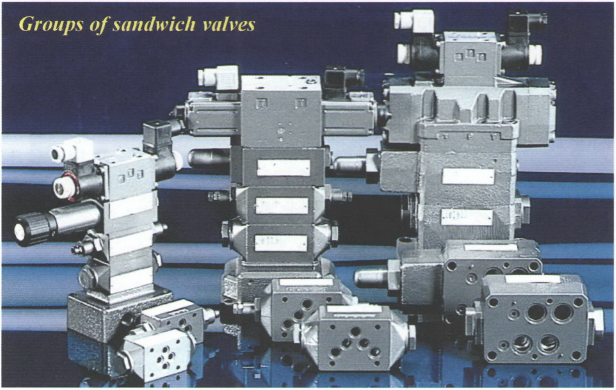


Figure 8.51