Appendix

Articles of interest and index by model code

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Appendix

Powers of attraction

Article of interest

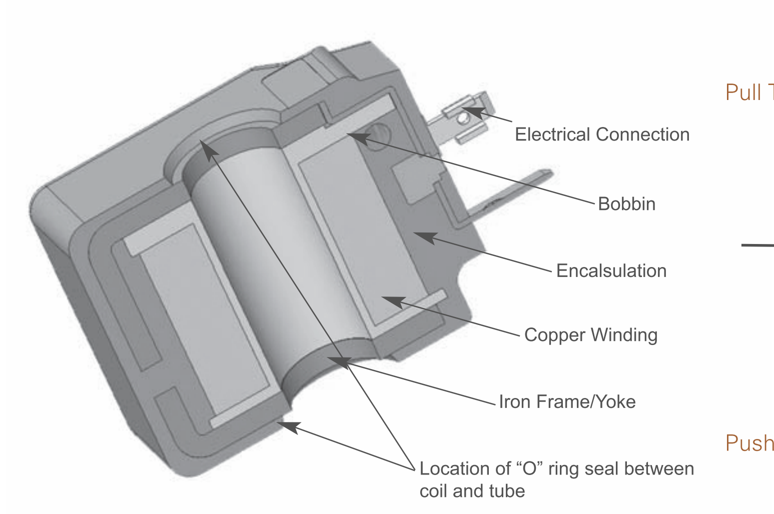

Figure 1. Component parts of a typical solenoid coil

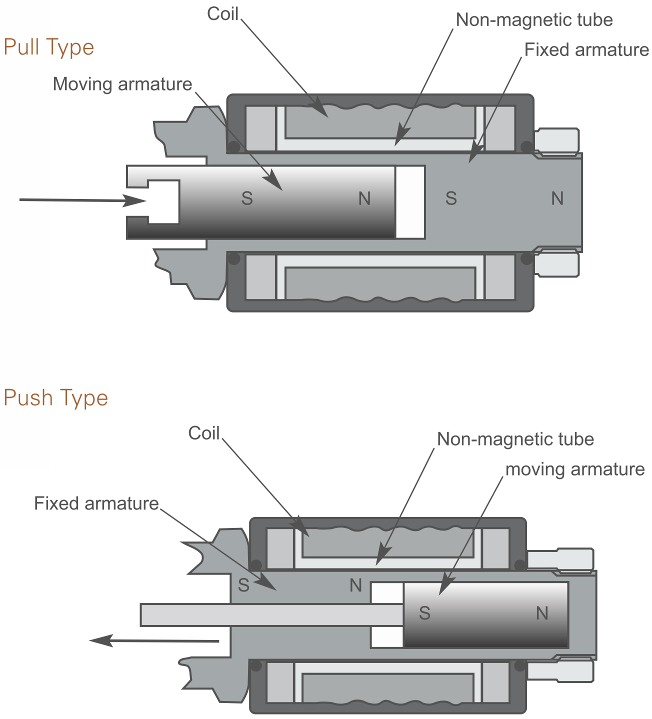

Figure 2. Pull and push type actuators

Electromagnetic force is responsible for practically all phenomena encountered in daily life they maintain the relationship between atoms so every action whether push or pull relies on these invisible forces to create movement.

The use of these forces both magnetic and electrical has a huge impact on our lives. Man has learned to harness them in many ways - generating electricity by movement of magnets or creating magnetism by using electrical current. It is the latter that we rely on the operation of numerous hydraulic valves. We call these solenoid valves where by an electromagnetic actuator provides a force or movement to a hydraulic control element which in turn controls the fluid in a system.

Typically a solenoid valve uses an electromagnetic actuator (see Figure 1), consisting of coils of copper wire wound around a bobbin enclosed in an iron yolk, which is encapsulated in a heat resistant thermosetting plastic.

Various types of connector can be molded into the assembly to give varying degrees of water and dust resistance commonly known as the "IP" rating.

This coil assembly fits over a non-magnetic tube which contains fixed and moving ferrous armatures. When a current is applied to the coil the flux magnetizes the armatures which are attracted together. The level of attraction is dependent on the design and the level of the current.

There are two main types of electromagnetic actuator (Figure 2), the pull version and the push version. Both rely on attraction to work, the only difference being the layout of the fixed and moving armatures and the way they are connected to the part being actuated. (Reversing the polarity of the voltage in a simple coil will not change function of the actuator.)

With clever design of the tube and armature the force exerted by the coil can be made to be roportional to the current applied. This allows us to produce proportional directional, pressure and flow control valves.

The shape of the pole ends between the fixed and moving armatures along with the design of the non-magnetic in-fill allows the tube designer to change the force displacement characteristic produced by the combination of the coil and the tube. You can therefore have tubes designed to give proportional movement and those to give proportional force over a small movement. The latter of these designs is typically used for pressure control and the former for flow regulation or directional control.

The design of the hydraulic section of the valve enables the oil to pass from port to port due to the movement or force created by the effect of the electromagnetic flux on the two armatures and subsequently on the poppet or spool.

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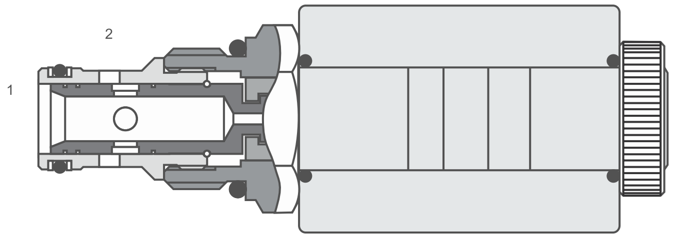

Figure 3. S510A Simple two position, two port directional control valve (using pull type solenoid)

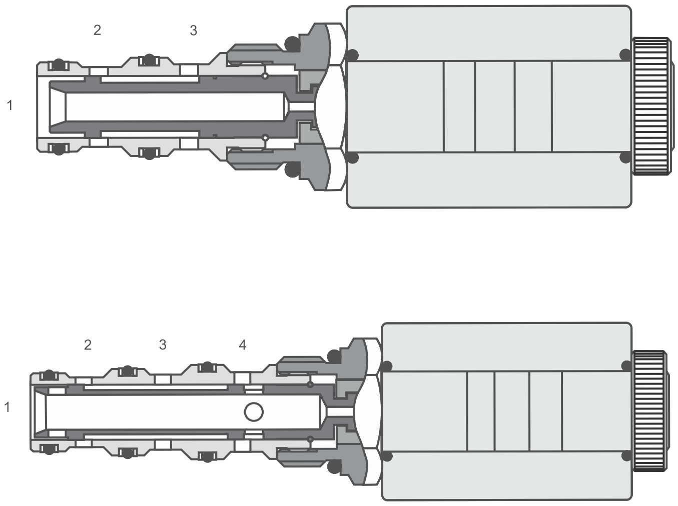

Figure 4. S525 & S542 still two position, but now three and four port directional control valves (again using pull type solenoid)

The simplest device is a two positioned, two ported valve made up of a spool and sleeve connected to the armature (Figure 3). The armature pulls or pushes the spool within the sleeve either closing or opening a ring of holes in the sleeve that connects one port to the other.

More ports and a more complicated sleeve/spool assembly can create two position three or four ported valves (Figure 4).

By using two coils on a common tube a three positioned valve with four ports can be achieved. There are various designs but common layouts use either one fixed and two moving armatures or two fixed and one moving armature, the example below uses the former. (Figure 5)

The poppet valve (Figure 6) consists of a poppet that is forced onto a seat against a spring in the case of a normally open valve and pulled away from a seat against a spring for a normally closed valve. (Poppet valves give minimal internal leakage compared to spool valves.) The balance between the working pressure, the spring force and the magnetic force is very important and in order for the poppet valve to pass higher flows it is necessary to operate the valve in two stages. The armature will force a pilot poppet onto a seat contained within a larger poppet so that when the pilot poppet is opened a flow is created across an orifice allowing the pressure difference to act to open the bigger poppet. By doing this you can control very large flows using a small pilot solenoid valve.

When specifying a solenoid valve there are several important characteristics of the assembly the machine designer must take into account.

All coils will come with a rating that indicates the voltageß and the power consumption in Watts. Establishing the requirement for a machine seems straightforward but there are pitfalls that must be avoided.

Most coils are advertised as continuously rated provided they are working within set parameters. When a solenoid is energized it will generate heat, the amount of heat experienced by the coil will depend on the power being

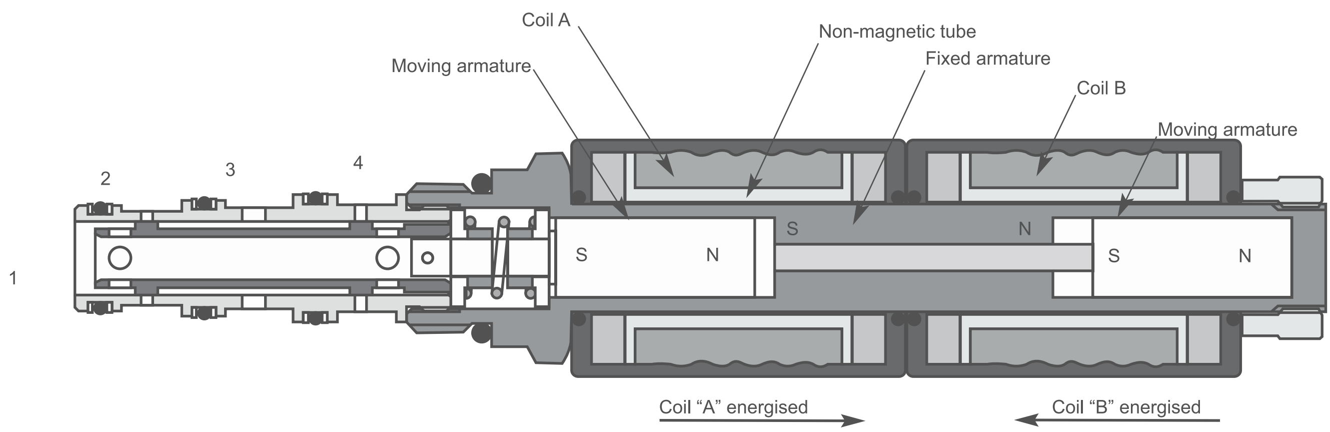

Figure 5. S570 Two solenoids on a common tube to give a three position four port directional control valves

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Powers of attraction

Article of interest

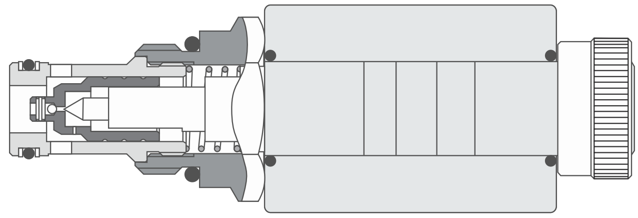


Figure 6. S501 A typical pilot operated poppet valve - good flow characteristics and minimal internal leakage (using push type solenoid)

applied, the design of the coil and any cooling effects. The temperature of the valve is important because it affects the performance of the encapsulating material. The materials used will have a class given to them.

For example Class 'H' guarantees the material up to 180° C while class 'F' up to 155° C. If the ambient temperature plus this increase in coil temperature rises above the rating of the insulation and encapsulation material then failure will occur. Most catalogs will indicate an ambient temperature of between -20° C and +40° C.

Change in temperature also affects the resistance of the coil and hence the force it will exert, but more of that later.

The second most important consideration is the application and the exposure of the coil to the elements. Coils and tube assemblies are designed to withstand different levels of water ingress. This is called the 'IP' rating. The connectors used will carry their own IP rating such as DIN 43650 at IP65, most Deutsch connectors are IP67. The level of water ingress protection increases with the higher number. Details can be found in BS EN 60529. Some manufacturers also seal the coil/tube joint with an "O" ring to prevent corrosion and possible weakening of the tube.

On machinery where the valve is mounted a long way from the power source voltage loss along the wires has to be accounted for. The coil is rated to give a performance to a valve specified with a tolerance on the voltage. Typically +/- 10% of the nominal voltage. If the voltage at the coil drops then full performance from the valve will not be achieved. Proportional valves in particular require the availability of a constant current to operate effectively and predictably. And it should be remembered that as the coil warms up the resistance will change affecting the voltage and so the reaction of the valve to change in input signal. Utilizing Pulse Width Modulation (PWM) will minimize these adverse effects. Hysterisis within the valve can also be reduced by applying PWM to the signal. Feedback loops within the electronic control can also make the valve more accurate.

Most hydraulic companies will specify a performance figure for their valves that is available at a percentage of the maximum power usage. This is to allow good operation when the coil heats up and there is a resultant loss of magnetic force, therefore when testing a machine it is important to verify the valve function at stabilized temperature/most extreme service conditions.

A coil is an inductor – it stores energy and resists change, so when switched off it will generate a brief high voltage signal which potentially can damage other electronics devices on the vehicle or system.

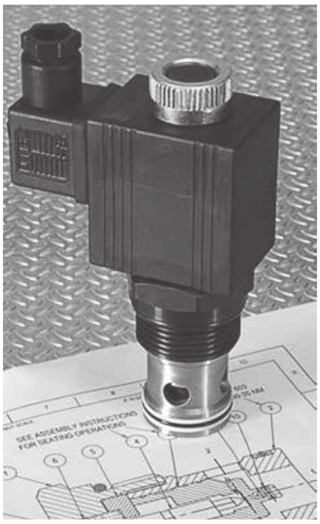
To protect against these, coils or connectors can be fitted with diodes to permit the energy to dissipate safely.

As yet there are not generally recognized standards as to how response times should be measured, some manufacturers will quote figures from the time when the power is switched to the point where the armature reaches the end of its stroke, others from the time the power is switched to the point where the hydraulic fluid reacts. When looking at different manufacturer's products it is important to compare like for like.

Note: Response times are usually different for switching the power on or off.

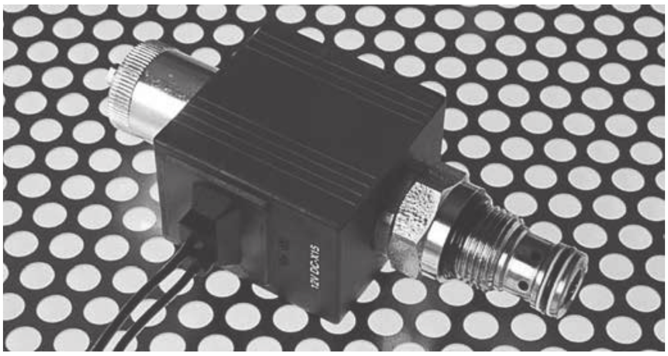
With mechanical valves the performance is often based on the pressure drop across the valve at a set flow. With electrically operated valves it is more likely that the balance between the solenoid force, the internal spring force and the flow forces will dictate the working envelope of the valve.

Force on valve components due to pressure is usually straightforward but forces are also generated by the fluid passing over the surface of the valves components, these flow forces can either act with the coil force to help the valve to stay in its operated



condition or act against the coil force. There have been occasions where they have caused a valve to switch back to its original position as soon as the flow force overcomes the magnetic force causing an actuator to change direction in the middle of its stroke. Performance data often indicates that a valve will allow more flow in one direction than the other. This can be directly attributed to the flow forces within the valve.

In the case of most solenoid valves the pressure limitation will be determined by the tube design and the factor of safety



A simple proportional 2 port restrictive style pressure compensated flow control valve. Type PFR24A employed by the manufacturer. But in some cases over pressurization of the valve may cause the valve to open, as the force created overcomes the magnetic force exerted by the coil.

A typical example of good hydraulic design using electrical operation is shown in Figure 7. This valve is used on a pilot line to provide a pressure compensated flow at all times with the ability to switch from one pressure to another by energizing the solenoid. The armature compresses a spring within the valve that increases the force on the poppet increasing the setting of the valve.

Figure 8 shows a bi-directional poppet valve for flows up to 90 liters/min To give bi-direction to the poppet and so the flow two very small shuttle valves are situated in the poppet. These direct the flow through the opening orifice from the high pressure side of the valve. When the valve is de-energized the poppet is balanced and offset closed by light spring.

Figure 9 shows a proportional pressure reducing valve, where with no current applied, the regulated port is connected to tank. As the current increases the pressure in the regulated port will rise to balance the increasing solenoid force.

Pressure in the regulated port acts on the spool tending to close it against the force of the solenoid. As pressure in the regulated port increases (with a constant current applied to the solenoid) the spool will shift, restricting the inlet, equilibrium is then achieved and a reduced constant outlet pressure held. Varying the current will alter the force applied to the spool and hence the pressure in the regulated line.

Solenoid operated valves are used on most machines and effectively control the required functions. Proportional controls are becoming more common as the cost of the technology is reducing and the advantages of digital control are realized. The variety of solenoid operated valves is wide because of the ingenuity of the design engineers to solve complex and often conflicting demands.

A solenoid valve can give electrical control to almost any hydraulic function. This reduces pipe work, removes most hydraulic lines from the vicinity of the operator, adds flexibility and functionality and ultimately adds to the competitiveness of the machine in which they are used.

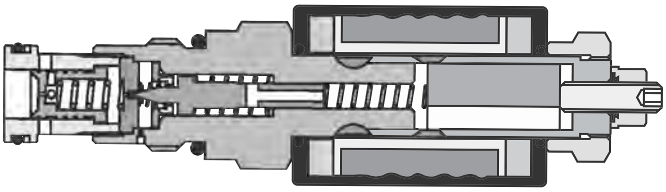


Figure 7. A pressure compensated flow control with pressure switching

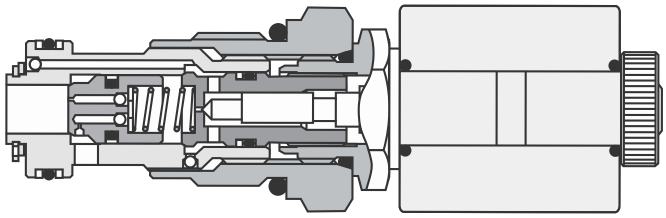


Figure 8. S717 High flow bi-directional poppet valve

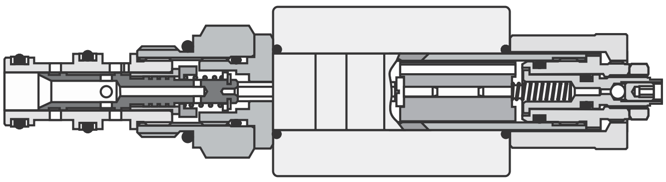


Figure 9. PPD22A Proportional pressure reducing valve

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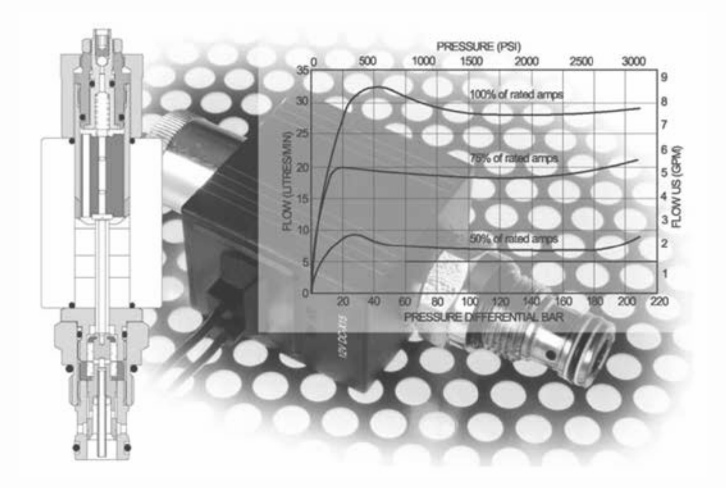
Keeping things in proportion

Article of interest



Electronic control of solenoid operated valves is becoming commonplace in many different types of machines. Traditionally proportional valves have been applied to industrial applications such as machine tools but increasingly mobile machine manufacturers are accepting the improvement in machine performance despite a reticence to adopt black box type technology provided by electronic control cards.

To the layman proportional control can be a little daunting but with the help of electrical and electronic engineers the technology is not so frightening.

For the successful introduction of proportional control technology a new language has to be understood. It is not necessary to understand the underlying electronics but the function, application and benefits of each term will make for a smooth application.

Proportional control can be applied to pressure, flow or directional control valves. By altering the current applied to the coil you can change the force exerted by the electro magnet. This can be done in a linear way so that the change in current produces a proportional change in pressure, flow or position of any valve. Innovative valve design can give smooth linear control with low hysteresis while using a simple standard coil.

Electro-magnetic force is proportional to the current flowing through the windings of copper within the coil so if a constant current is not maintained the performance of the valve will alter. This means that if you are running from a battery and the voltage falls due to loss of battery charge then the current will also fall causing the solenoid force to decrease. If the running or ambient temperature of the coil rises then the resistance will increase thus causing the current to fall subsequently affecting the performance of the valve. It is necessary to try to maintain a constant current irrespective of changes in voltage or resistance. This is why electronic control circuits are important.

The electronic card or controller introduces the ability, not only to control the current but to provide other advantages. The input signal can be from a Joystick or potentiometer (manually controlled), or from a feed back from elsewhere in the system.

This signal can be controlled to introduce such things as Ramp, where there is a controllable linear increase in the signal over a period of time regardless of how violently the operator moves the joystick, this allows smooth operation of the machine without rapid changes in force or speed and Dither, rapid small current oscillation that keeps a spool moving to prevent it sticking due to silt or small contaminant.

As we have said, the force produced by a given coil is determined by the current applied, the current can be quite high so an amplifier card is used to increase a low input signal (governed by the joystick or potentiometer) to that required to operate a hydraulic valve, the amount of increase is called Gain or I-Max and is simply defined as the output signal/input signal. One could, of course, drive a hydraulic valve directly from the joystick but this would require the potentiometers on the joystick to dissipate a lot of heat and as the current available in a machine varied the performance of the hydraulics would vary. An amp card can pass the currents required but it will, again, produce a fair amount of heat. The solution to this is a technique called Pulse Width Modulation (PWM). PWM effectively produces a variable output signal by switching the full current on and off very quickly, typically at 100 to 400Hz. If the current is on for 75% of the time the average current output will be 75% of the full current available, if on for 50% the out put will be 50% and so on. This reduces the heat generated and power consumed. At low frequencies it will naturally produce dither but the level of dither produced will depend on the output current required, the maximum will be at 50% dropping off to zero at 0 and 100% output. These figures may not suit a particular valve so higher frequency PWM and deliberately generated dither may be preferable but, typically the higher the frequency the higher the cost of the card. The input to the amplifier card can be analogue (from joystick or potentiometer) or digital (from a PLC Controller). Using a PLC, functions can be programmed to give automatic sequential control to a machine or system.

There are many different types of Proportional valve from the very sophisticated with on-board electronics complete with positional feed back to the more simple device that work in open loop systems, probably relying on the operator as feed back to the controller through a Joystick. These valves can be inexpensive but at the same time give infinite control to the operator. The electronics can control the input to prevent the machine from going out of control under rapid change of input signal.

Hysteresis can be improved by introducing a low frequency PWM signal producing dither within the moving components of the valve. If the loop is closed electronically then even a crude proportional valve can give accurate performance.

Legislation now dictates that high pressure hydraulic lines are not acceptable in the cab environment so electronic proportional valves enable the machine operator to retain the feel afforded by direct, lever operated valves and to work in a safer environment.

A typical example is a pavement sweeper. As the operator moves around he has to make constant adjustments to both the height and speed of the brushes. To do this he employs proportional pressure reducing valves to raise the brush and proportional flow controls to govern its speed.

A proportional pressure-reducing valve used for this application is shown in Figure 1. The PPD2 2A is a self contained cartridge that can be installed in a manifold

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along side of other control elements. The valve has a maximum rated flow of 20 liters per minute and controls the output pressure relative to the current supplied to the coil. The hysteresis without PWM is around 16% maximum but comes down to 5% with PWM at around 200Hz. The 'dead band' is around 16% of the rated current. This means that the control begins after 16% of the rated current is applied. Adjusting the minimum current, L/min, on the electronic card can accommodate this. The operator will then not have to move the joystick far before he gets a reaction from the valve.

The valve has a maximum inlet pressure rating of 210 bar and an outlet pressure from 0 to 28 bar.

In this application the cylinders controlling the height of the brushes are single acting. The weight of the brush will drive it into the ground so the proportional valve is used to raise the brush to a position most suitable for the ground conditions, during sweeping. The control in the cab is a simple potentiometer.

Figure 2 shows a proportional flow regulator that is used to control the speed of the brushes. This valve has a flow range of 0-28 L/min with a compensation accuracy of around 10% on the set flow with change in regulated pressure from 20-210 bar. The valve is capable of compensating in both directions. The response time is around 300ms to change from one flow to another and the hysteresis with a PWM of around 200Hz is about 4%.The internal leakage prior to the valve opening is 200cc/min with 210 bar difference between the two ports. While this is not the most sophisticated proportional valve design it is very competitively priced and gives accuracy and repeatability that is well within the requirements of most mobile machine applications.

On the same machine this valve is used to control the fan speed providing the suction that draws up the waste into the hopper. It is important that the valve gives reliable operation because excessive speed on this kind of fan can be dangerous due to the inertia generated by the heavy steel fan.

In some cases on other applications any excess flow needs to be diverted to tank to limit heat generation. A by-pass style pressure compensated flow regulator is required. This cartridge can be used in conjunction with a standard compensator and a pilot relief valve to give the complete circuit controlling the regulated flow, bypassing the excess flow to tank at working pressure and supplying a system relief limiting the pressure in the regulated line. Figure 3 shows a typical manifold.

Another area where a simple proportional valve can provide benefits far beyond expectation is by using it as the control element in a much larger cartridge valve.

A simple proportional pilot relief cartridge can be used in conjunction with normally open or normally closed logic

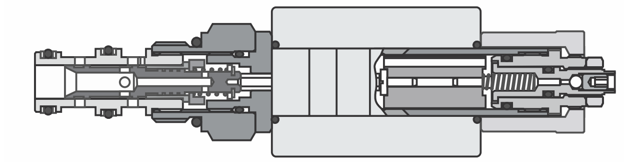

Figure 1. PPD22A Pressure reducing cartridge

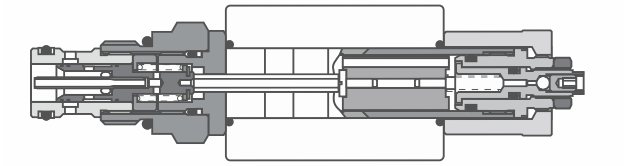

Figure 2. PFR24A Flow control

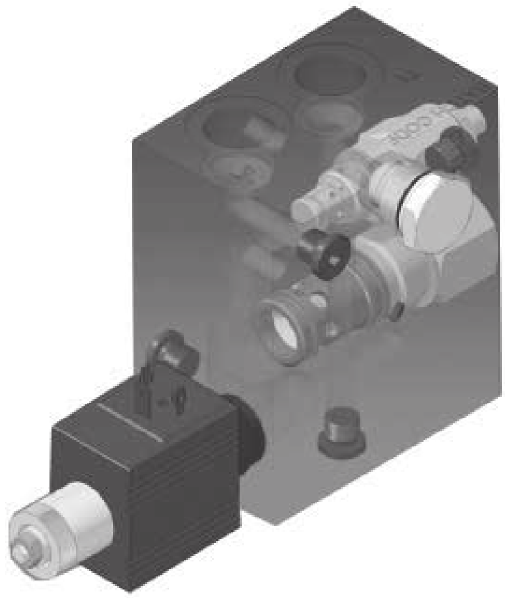

Figure 3. Proportional pressure compensated flow control HIC.

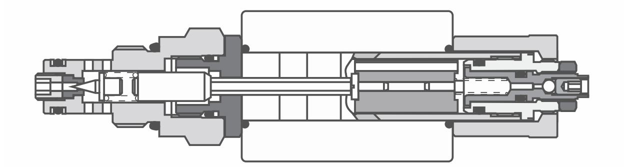

Figure 4. PDR2 pressure relief cartridge

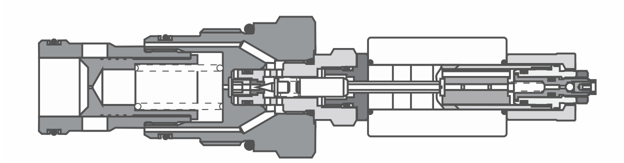

Figure 5. 400 L/min proportional pressure relief valve

Keeping things in proportion

Article of interest

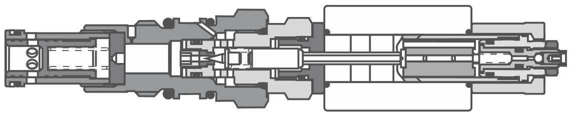

Figure 6. 100 L/min proportional pressure reducing valve

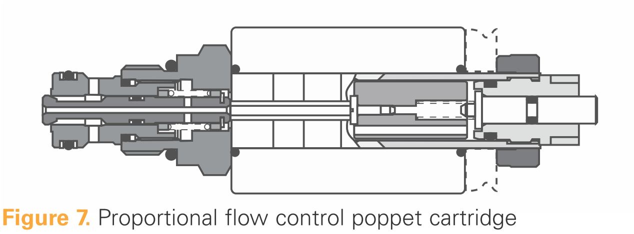

Figure 7. Proportional flow control poppet cartridge

elements to give high flow relief function or pressure reducing functions. Fig 4 shows a typical proportional relief cartridge which has several pressure ranges between 7 bar and 350 bar. The maximum rated flow is 2 L/min; this is adequate for use as a pilot control valve. Hysteresis is around 7% with PWM at 200Hz with a dead band of 7%.

Figure 5 shows the cartridge fitted into the back of a normally closed logic element. The valve now becomes a 400ltr/min pilot operated relief valve proportionally controlled. This valve has been used on large earth moving equipment giving different pressure control for the different functions. Using a PLC this is done automatically. On operating each function a signal is sent to the relief valve to alter the pressure setting.

Figure 6 shows the cartridge fitted into a normally open logic element. Some cooling fan applications demand pressure control to limit the speed as opposed to flow controls. The pilot cartridge can give suitable control to the logic element with flows in excess of 100 L/min. Within the machine a temperature sensor will signal the fan to start running governing the speed depending on the temperature. This signal is received by the pilot relief cartridge automatically adjusting the current feed to adjust the pressure setting of a normally open logic element. The fan speed is proportional to the pressure. The valve will control this pressure irrespective of potential changes in inlet pressure.

Figure 7 shows a proportional valve that would not be considered as the most accurate and sophisticated valve in hydraulics but is very competitive and ideal for the applications it was designed for. The valve shown provides load holding by virtue of its poppet design and a proportional speed control. The valve has been designed so that under heavy load conditions the valve reduces the speed slightly giving a form of compensation. The dead band is around 45% of the maximum rated current due to the poppet design but with electronic adjustment of minimum current the operator never notices this quirk. The feed back on this system is the operator who will drive the machine at a speed that he feels comfortable.

Proportional valve technology need not be expensive and very sophisticated. Often the valves are over designed for the real machine requirement. A hysteresis of 1% compared with 5% will increase the cost of the valve significantly but the operator of most mobile equipment would not notice. A response time of 10ms against 300ms would not be a problem given the excellent response of human reflexes. A good valve is one that is suitable for the application at the correct price. Electronics can improve the performance of any valve making the performance indistinguishable to that of more sophisticated devices.

Even simple electronics can give the operator enough confidence to enjoy the benefits of proportional valve technology. The valves and electronics can be cost effective and give the operator the benefit of smooth reliable control and added security as he goes about his working life.

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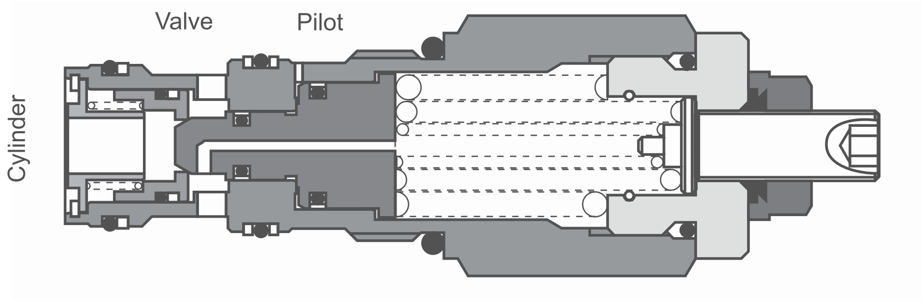
Figure 1: 1CE Standard overcenter valve



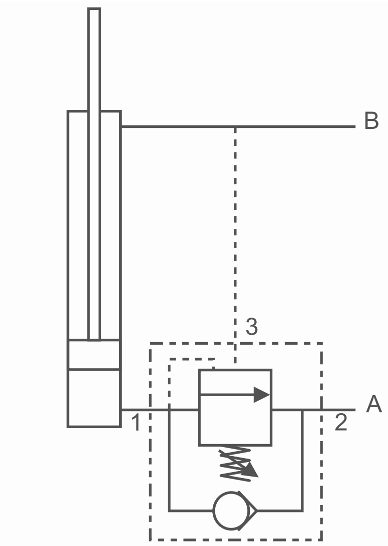


Figure 2

There are two basic designs, each with several variants. The direct acting design, is ideal for flows up to 200 L/min whereas the differential area design, is suitable for flows up to 300 L/min.

There are now many types of overcenter or motion control valves available to the designer of hydraulically operated machines, each one has its own place and specific benefits to the user. The function of these valves can be divided into three basic groups.

1. Load Holding; where the overcenter valve prevents the movement of a load when the directional valve is in the neutral position. Permitting the use of open center directional valves and negating leakage past the spool of closed center directional valves.
2. Load Control; where the overcenter valve prevents the actuator running ahead of the pump due to the load induced energy thereby eliminating cavitation in the actuator and loss of control.
3. Load Safety. In the case of hose failure an overcenter valve mounted onto or into an actuator will prevent uncontrolled movement of the load. When a boom is used as a crane then hose failure protection is vital as the loss of load control could cause damage to people or property.

Each of these functions is applicable to linear or rotary motion.

The standard overcenter valve (fig 1) can be described as a pilot assisted relief valve with an integral free flow check. The difference between this design of valve and a pilot check is that the check valve will open fully as soon as the pilot pressure is sufficient to open the valve because the only resistance to opening is the pressure locked in to the cylinder port. With an overcenter valve the pilot pressure has to overcome the force of the spring which is reduced by load pressure. This ensures a gradual opening and a metering of the flow as it passes the poppet. Integrated Hydraulics overcenter valves consist of a poppet that seals flow from an actuator, a check element, which permits free flow to the actuator and a pilot section that opens the poppet allowing flow from the actuator at a controlled rate. There are two basic designs, each with several variants. The direct acting design, whereby the pressure in the actuator acts on the full area of the nose of the poppet, is ideal for flows up to 200 L/min whereas the differential area design, whereby the pressure acts on an annular area, is suitable for flows up to 300 L/min. Being of poppet type both designs exhibit excellent leakage characteristics with maximum leakage of up to 0.5 ml/min for valves up to 200 L/min capacity and up to 4ml/min for valves with 300 L/min capacity.

The cartridge has three ports, a cylinder port (1), a valve port (2) and a pilot port (3). If pressure, above the setting of the valve is applied to the cylinder port it will open as a relief. When applied to the valve port pressure will open a low pressure check allowing free flow into the cylinder port. Pressure applied to the pilot port acts over a larger area on the poppet than the area referenced to the cylinder port, so the valve will open at a low pressure.

For most applications the relief setting should be approximately 1.3 times higher than the maximum load induced pressure. This ensures that with the maximum load on the actuator the valve will remain closed until pilot pressure is applied. The pilot pressure required to open the valve will depend on the pilot ratio that is the ratio between the relief area and the pilot area. The pilot pressure can be calculated:

$$\text{Pilot pressure} = \frac{\text{Valve Setting - Load Pressure}}{\text{Pilot Ratio}}$$

A typical application would entail mounting the overcenter valve in or on the end cap of a cylinder (fig 2). The cylinder port of the valve being connected to the full bore area of the cylinder, the valve port to the directional control line A and the pilot connected to the annulus inlet, line B and so to the directional control line B. As soon as the pressure rises in the inlet port of the annulus (line B) to retract the rod to a point where it reaches the required pilot pressure the actuator will begin moving at the flow at which the pressure setting was made. If the load causes the flow to

Overcenter valves

Article of interest

increase then the inlet will be starved of oil and the pressure will begin to drop at this port. The reducing pressure will be sensed at the pilot allowing the spring to begin to close the valve preventing load run-away. In this way the valve will continually meter, controlling the load throughout its movement. When the pressure needed to move the load is higher than the pilot pressure needed to fully open the valve the only restriction produced is the pressure drop due to flow in the fully open condition.

With the standard overcenter the spring chamber is vented through the poppet to the valve port which creates a problem if there are varying or high back pressures. Pressure in the valve port increases the effective setting of the valve by a factor equivalent to the pilot ratio plus one. This means that if there is a standing back pressure of 50 bar with a pilot ratio of 5:1 the effective relief setting would be increased by 300 bar. This creates problems if the application demands a closed center directional valve and the utilization of service line reliefs. The relief valves will operate to limit inlet pressure but will not act if there is an external load which needs to be limited. The overcenter will not allow oil past the seat due to the back pressure created by the service line relief valves. To overcome this problem the part balanced 1CER series was created (fig 3).

Any back pressure therefore, does not affect the setting of the valve or the amount of pilot pressure needed.

The 1CER series overcenter valve performs in the same way as the standard valve under most conditions. But the relief section of the valve is not affected by back pressure.

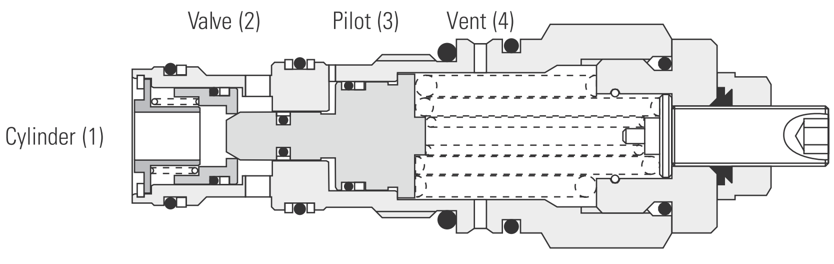
The poppet is designed to balance back pressure over two areas on the poppet. The first is a annular area between the seat (dia a) and the center seal (dia b) on the poppet which acts to open the valve and the second at the spring end of the spool (dia c) acting to close the valve. These areas are the same, the poppet is therefore balanced and so pressure in the valve line will not affect the relief performance of the valve. It must be noted that the pilot pressure required to open the valve is still affected on a one to one ratio by any back pressure.

The advantage of this design is the ability to use the valve on closed center directional valve systems allowing service line relief valves to operate as normal. Most other valves of this type on the market have an atmospheric vent which limits their use in corrosive atmospheres and are prone to leakage.

The 1CER valve does have some draw backs in certain applications. Because the pilot pressure is affected by back pressure the valve can not be used in regenerative circuits on the annular port of the cylinder. Also if used with a meter out proportional system the constantly varying backpressures can cause both the part balanced and the standard valve to go unstable. For this is the reason the fully balanced version, 1CEB series (fig 4) is available. In this case the spring chamber is vented to atmosphere or to a separate drain port.

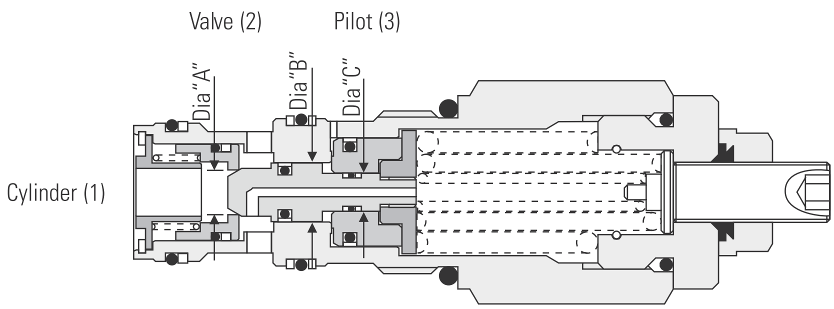
Any back pressure therefore does not affect the setting of the valve or the amount of pilot pressure needed.

For the standard, Part Balanced and Balanced valves there are various pilot ratios available to the system designer, which is best for his circuit? A general rule is that high pilot ratios are suitable for constant, stable loads and low pilot ratios for unstable and varying loads. The pilot ratio does not necessarily affect the working pressure by much given that the normal working pressure of a system is often much higher than the pilot pressure required to fully open the valve. If this is the case then the piloted open pressure drop will determine the systems efficiency.



1CEB Fully Balanced Overcentrevalve

Figure 3



1CER Part Balanced Overcentrevalve

Figure 4

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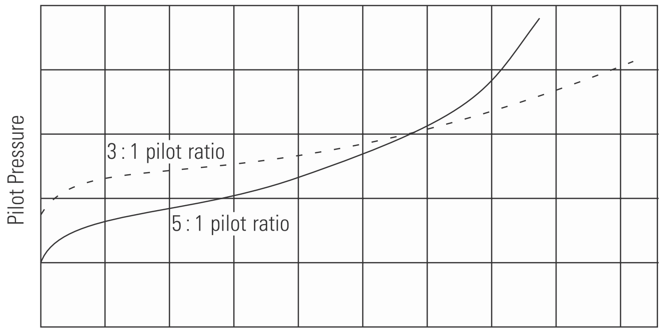


The Effect of Pilot Ration on Flow

Graph 1

Graph 1 shows the pressure drop curves of two valves with different pilot ratios. The higher pilot ratio valve is more restrictive than the low pilot ratio valve. This shows that above a certain pressure the lower pilot ratio valve is more efficient than the higher pilot ratio valve. It is important that the total performance is taken into account before specifying an overcenter valve.

The two stage overcenter valve, 1CEL (Fig 5) has been developed to overcome a problem which has been a continual nuisance to designers of machines incorporating long unstable booms. Instability problems affect many machines, most noticeably those with high capacity cylinders particularly in conjunction with slender booms that are subject varying frictional forces. The best example is the Telescopic Handler that usually has a long cylinder to extend or retract its boom. At the end of its stroke the pressure of the oil within a cylinder rises to the setting of the main relief valve for that part of the system and by its nature, the motion control valve re-seat locks in that pressure (irrespective of any load induced pressure). When the operator lowers the load, this stored energy gives the valve the message that a heavy load is on the cylinder; therefore it takes less pilot pressure to open. As a result, the valve opens very quickly and allows the stored energy to dissipate causing a momentary runaway condition, this causes a rapid acceleration of the load that is then checked by the motion control valve and brought under control. The consequence of this is an initial instability as a boom is retracted; the number of jerks will depend on the stiffness of the system at the time of lowering. This instability can sometimes continue through the whole of the cylinder's stroke, its magnitude, in extreme cases, can cause severe operator insecurity or even the loss of a load.

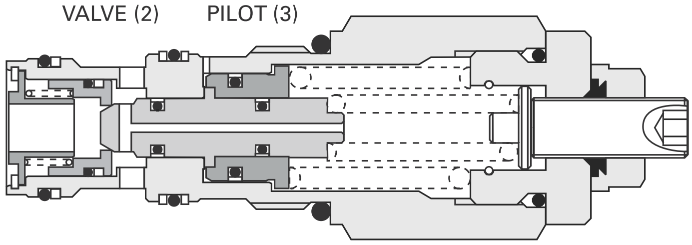
The 1CEL valve uses two springs to control the poppet, only the outer spring being effected by the pilot piston, leaving the inner to generate a counterbalance pressure. The two-stage valve has overcome many instability problems by preventing the total decay of the stored energy in the cylinder and stopping the valve over reacting. It allows the pressure to fall to the counterbalance setting, which can be adjusted dependant upon the severity of the application. This back pressure can also help to stiffen the boom during its movement further through its stroke, for example when wear pads on the box sections of a telescopic boom create changing frictional forces. This works well but with some systems, the backpressure created by this valve causes problems due to the reduction in available force. On certain machines, when for instance a crowd cylinder is bottomed, the oil from a slave cylinder has to be forced across a relief valve; the boom cylinder creates an induced pressure by virtue of its downward force. It is possible that an unloaded boom will not lower due to the counterbalance pressure. Also in the fully piloted open position the valve still generates a backpressure heating the oil and creating inefficiency.

To overcome these problems another variant is available in which the counterbalance pressure is reduced as the pilot pressure increases. This design has a second pilot ratio, which acts to reduce the backpressure applied by the center spring. Indeed the valve can be piloted fully open, eliminating the counterbalance pressure altogether so improving the efficiency of the system. With a primary pilot ratio of 4:1 and a secondary ratio of 0.5:1 the initial unloading of the stored pressure happens at a low pilot pressure

It is important that the total performance is taken into account before specifying an overcenter valve.

followed by a more gentle reduction as the pilot pressure increases. The overall setting of the valve is a combination of the outer and the inner spring forces divided by the seat area.

The practical application of either of these valves involves the establishing a range of acceptable settings. For example, the requirement is for the valve to be set at 200 bar (3000psi) with a counterbalance pressure between 35 and 70 bar (500-1000psi) - there are two springs within the valve, the outer one is fixed and the inner adjustable. For this application the outer spring would be set to give 165 bar (2400psi) and the inner adjustable between 35 and 70 bar (500-1000psi). This would give the valve an adjustable range of 165-235 bar (2400-3400psi). Given a pilot ratio of 6:1 or 4:1 depending on the type this extra pressure setting would have little effect on the pilot pressure needed to open the valve during normal operation.



1CEL COUNTERBALANCED OVERCENTRE VALVE

Figure 5

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Overcenter valves

Article of interest

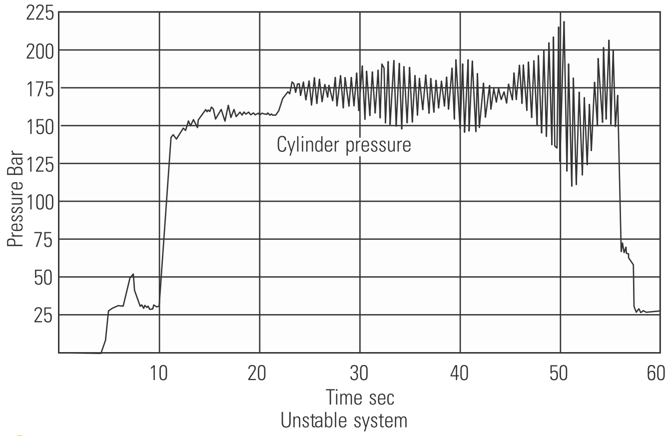
Graph 2 shows a typical recorded instability picking up machine frequencies and getting worse and Graph 3 shows the counterbalanced overcenter valves preventing the problem getting worse, dampening out the initial instability and the counterbalance pressure falling as the pilot pressure increases.

The zero differential range of load control valves 1CPB (fig 6) have been designed with 'BoomLoc' hose rupture valve applications in mind. Typically the valve is piloted open from the hydraulic remote control operating the main directional spool valve. By setting the overcenter to open just after the main valve it will control the flow rate at low speed but as the overcenter opens more rapidly than the directional valve the directional valve will control the flow rate at higher speeds. It is a pilot operated metered poppet valve. The poppet seals against a tapered seat, as the pilot pressure increases the poppet will move off the seat. Flow is dependant upon the axial movement of the poppet which in turn is dependant upon the force exerted by pilot pressure balanced by that exerted by the spring. The poppet is hydraulically balanced so this valve is unaffected by valve line AND cylinder pressure but it will not provide any relief function. If over pressure, shock or thermal relief are required a second relief element is required.

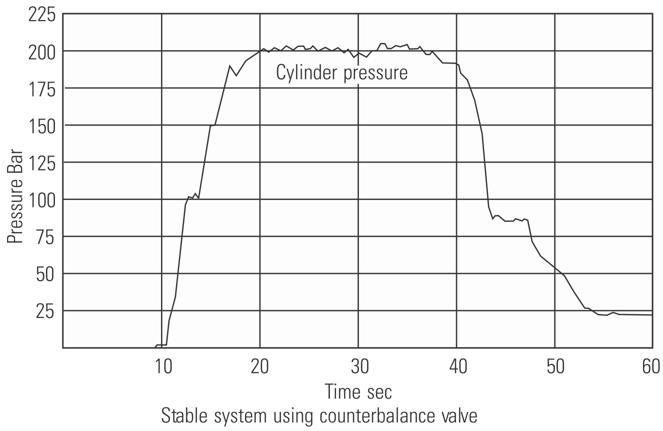
The successful application of motion control valves, particularly in areas that are demanding involves the anticipation and resolution of numerous factors only some of which can be discussed in this article. Motion control valves are adjustable, are available in several pressure ranges with many pilot ratio options. Most of the valves fit in a common cavity (the exception being the fully balanced, 1CEB and zero differential, 1CPB versions when required with an external rather than an atmospheric vent) and are available

in sizes from 30 to 300 L/min. The flexibility of cartridge valve technology can therefore be easily applied to bring stability. The standard range of valves described here can be used to solve the vast majority of motion control problems and we are constantly developing new valves that will further improve stability and load control.

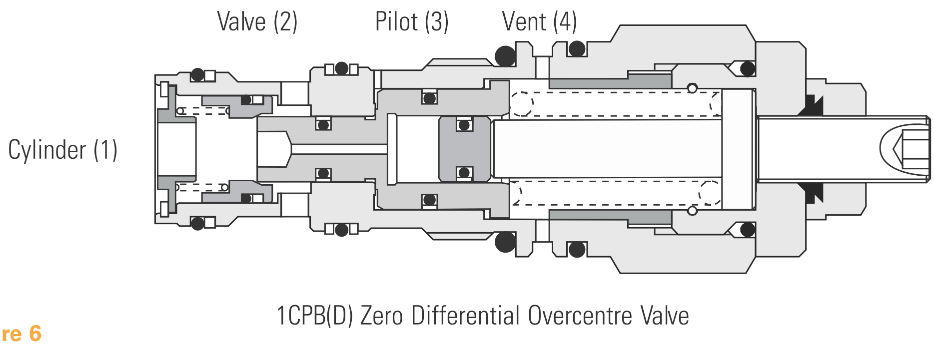
The standard range of valves described here can be used to solve the vast majority of motion control problems.



Graph 2



Graph3



1CPB(D) Zero Differential Overcentre Valve

Figure 6

The requirement for hose rupture protection on mobile plant is enshrined in law in many territories and likely to become so in many others. ISO8643 requires that, in the event of a hose failure while lowering a boom it should not accelerate to more than twice its original speed with the control lever held in the same position. Additionally the valves introduced to achieve this should not unduly effect the operation of the machine to which they are fitted. (The current 100% maximum increase may be reduced to 75% in the future). If a hose were to fail while a boom is lifting or static the load should be held in position. We have developed a range of hose rupture valves, designated "BoomLoc", that are designed to meet the stipulations of ISO8643 and can be applied to numerous different machines.

BoomLoc valves are reliable because they make use of standard off the shelf components that have been field proven, in most cases for many years. All our cartridge valves are manufactured to a high standard with moving parts hardened and precision finished to give a long trouble free service life. Their performance is predictable so set up and development times can be dramatically reduced even when applied to a new system. In service the valves perform as intended with cartridges having been tested to over 1,000,000 cycles in our development department and having been used for many years in the field. When operating under normal circumstances, i.e. with hoses intact, BoomLoc valves offer high efficiency, as the hydraulic fluid is free to pass through the valve to the cylinder with negligible pressure loss. And by selecting the most appropriate package to match the performance of any given directional valve pressure losses in the return direction can be kept to an absolute minimum.

BoomLoc valves can provide a very compact solution. The valve should be mounted on the cylinder, either directly onto its port or connected to the port by rigid tube, so space can be quite restricted. Using Integrated Hydraulics Boomloc valves unique design potential to the full, transfer plates, commonly employed to permit the fitting of more bulky valves can often be eliminated. The block can be designed to suit the customer's installation and can even be done away with if the cartridge valves are incorporated into cylinder end caps thus dramatically reducing the space requirement and the cost of the overall package. Furthermore additional features can be incorporated in the block making the machine more versatile.

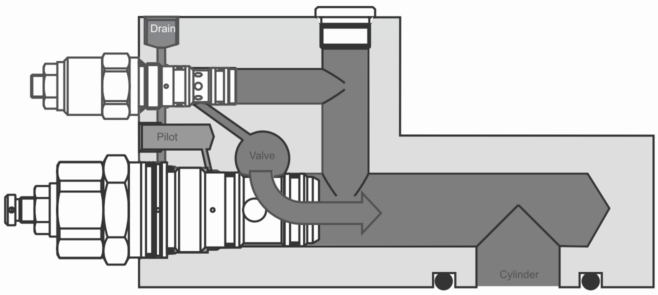
To achieve the desired protection without adversely effecting the operation of the machine it requires the accurate balancing of the hose rupture valve and the main directional valve. In order to work effectively the opening of the hose rupture valve should, ideally, lag behind that of the directional valve and the difference in pressure drop should remain constant throughout the operating flow range. To achieve this the hose rupture valve should be matched to the system in which it is employed, a set up that works on one type of machine would not necessarily work on another.

Due to the unique seat and poppet arrangement in the 1CPB series cartridges used in their "BoomLoc" range and the flexibility of the cartridge valve design "BoomLoc" valves can be tailored to suit most directional valves and so provide exceptionally fine control. A level of control that is particularly desired for levelling and grading, which can reduce (if not eliminate) the "washer board" effect frequently found when other valves are used. This fine control also has the benefits of enabling the operator to accurately position loads during craning operations.

"BloomLoc" valves are designed to meet the stipulations of ISO8643 and can be applied to numerous different machines.

Reduced installation costs can be realised using these "BoomLoc" valves as they use a simple direct SAE mounting to cylinder. In order to minimise stock requirements we have, where possible designed the cylinder port face to be suitable for both SAE3000 and 6000 flanges. A pilot bleed port is available on most valves simplifying installation further. Service costs are also minimal, in the unlikely event of a BoomLoc valve being damaged, repair usually just involves the replacement of one or two self-contained cartridges. What is more there is rarely any need to remove the block from the cylinder when changing the cartridges - reducing the risk of the ingress of contaminants and the down time of the machine. Spares inventory is frequently reduced as the same two, standard, off-the shelf cartridges (albeit set differently) are often used on a wide range of machines.

Operation: Free flow to cylinder



Upon operating the control to raise a boom, pilot pressure from the hydraulic remote control unit operates the appropriate spool in the main directional valve, permitting flow to the cylinder. At the cylinder the flow passes through our Hose Rupture Valve (HRV) check sections with minimal pressure loss and enters the cylinder. (The fluid from the other end of the cylinder flows directly to tank via the main control valve).

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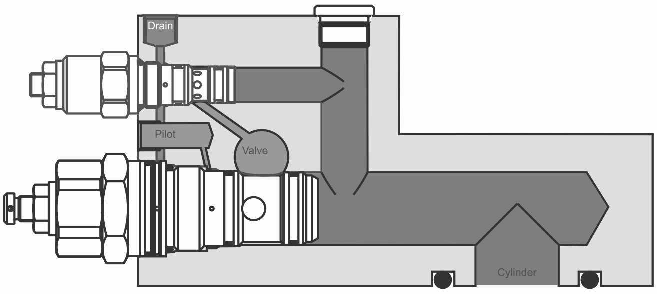
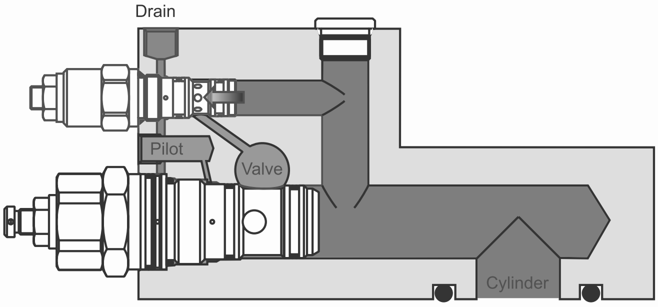
Load Holding



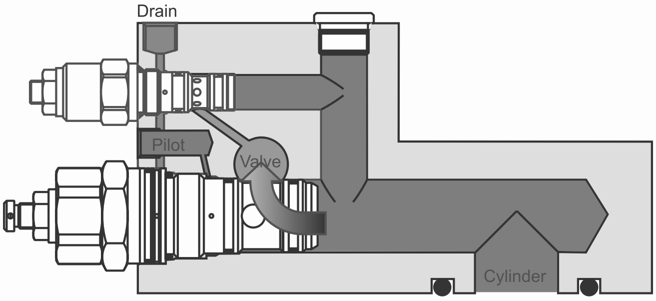
When the control lever is returned to neutral, the check valves in the HRV close and any return flow is blocked, the load is now locked in position.

Relieving



Cavitation and excessive pressure in the boom cylinder, (bottom end circuit), is prevented by an overload relief and make up check, located in the HRV. Flow being returned to tank via the main control valve port relief.

Controlled lowering



When the control lever is pushed forward to the boom lower position, pilot pressure from the hydraulic remote control unit operates the appropriate section of the main directional valve, opening flow from the cylinder return line to tank. At the same time the pilot pressure opens the main poppet of the HRV, thus allowing oil to flow from the bottom end of the cylinder to the return line. The rate at which the boom descends is dependent on the position of the poppet in the HRV and the spool in the main directional valve, So in the event of a total hose failure the HRV will prevent the boom accelerating above twice its original speed. As the flow rate is now dictated by the pressure drop across only the BoomLoc valve. Releasing the control lever will permit the poppet in the BoomLoc HRV to close stopping the boom from further descent.

BoomLock valves set-up procedures

For reasons of safety it is recommended that ALL adjustments to the Hose Rupture Valve be carried out with the bucket rested on the floor.

Unless otherwise requested the cartridges are preset to 350 bar (relief cartridge) and 10 bar (pilot catridge, part number 1CPB** or 1CPBD**). The pilot valve will normally require adjustment, but the relief setting of 350 bar is generally suitable for most applications and ensures the maximum protection of the cylinders. Check the maximum pressure of the system and adjust the relief valve if required. Prior to all adjustment ensure pilot line has been fully bled. This can be achieved by operating the joystick to either roll in or lower the respective cylinder and opening the bleed port on the Hose Rupture Valve. In cases where no "BLEED" port is provided, disconnect the pilot hose from the Hose Rupture Valve and running it to a suitable container should suffice.

Relief Valve setting: The relief setting of 350 bar has been calculated to meet most systems but can be adjusted to individual requirements. It is advised that the setting be between 10 and 20% higher than the main control valve. To increase setting, screw the adjuster clockwise to increase pressure at a rate of approximately 65 bar per full turn. To decrease setting, screw the adjuster anti-clockwise to decrease pressure at the same rate.

Pilot Cartridge setting: To successfully set the pilot cartridge pressure, two 400 bar and one 50 bar gauges need to be used. On the Hose Rupture Valve, connect one 400 bar gauge to the "E" port where provided (or the cylinder port if no “E” port) and the other 400 bar gauge in the valve inlet line “V”, and connect the 50 bar gauge in the pilot line "P".

The procedure can be related to both the Arm cylinder and the Boom cylinders but for the ease of explanation, the following procedure is for Boom cylinders only.

1 Fully swing out Arm cylinder. Raise Boom to full extension and at the end of its stroke record the pressure in the valve and cylinder gauges.

2 To check setting, slowly move joystick to lower Boom. When the gauge in the valve line starts to fall, it is a signal that the main control valve has started to open, at this point note the setting in the pilot line, typically 8 bar.

3 Continue to slowly operate the joystick and note the reading in the pilot line when the gauge in the cylinder line starts to fall. This indicates the setting of the pilot cartridge (1CPB(D), typically 10 bar.

It is recommended that the Pilot Cartridge should dwell between 1.5 and 2 bar behind the Main Control Valve. If the pilot valve is set too low, pressure at "E" falls before "V" - adjust pilot valve clockwise.

If the difference between 2 and 3 above is greater than 2 bar - adjust the pilot valve anti-clockwise.

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Our brave new world of Hydraulics provides technology to industry that is ever improving and more complex. The demand for machines that think for themselves reducing human error have inspired the engineering fraternity to ever greater feats of hydraulic ingenuity.

The simple directional valve has become an electronically controlled mechanism that provides fine control to the movement of machinery. The pump has become more efficient by adding feed back controls in the form of pressure compensation and load sensing, providing stable controlled flow to a pre-determined level to reduce energy losses. Even some actuators have built in transducers to provide position feed back completing the loop.

It is a shame that when using this modern technology the simpler and most important valve in a system can be as crude as a ball on a seat. The humble relief valve takes a back seat to the point where great effort is made not to allow this valve its rightful roll in providing the ultimate system protection. "Don't let it operate because it is noisy" or "we can not guarantee that the pressure control will be consistent". "The valve opens too soon and does not close quickly enough".

From the main system relief to the safety relief there are valves available that are equally advanced in their innovation and technology as the higher profile pumps, directional control valves and actuators. The problem is that many engineers do not understand the reasons for the different designs and their individual applications or how to assess the performance. This article will attempt to throw some light on what is available and where to apply the different designs.

It is true that the simplest relief valve is a ball sitting on a seat with a spring keeping it closed until the pressure over the area of the seat is high enough to allow the valve to open and allow flow to pass. The flow capacity is limited by the size of the seat and the pressure difference across the opening. To get more flow across the valve the ball has to move further back against the spring increasing the force and therefore the required pressure. A basic relief valve curve will look like Graph 1.

Graph 1 is based on a poppet style direct acting relief. The cracking pressure is the point "A" at which the pressure over the area of the seat is the same as the spring force. The initial opening characteristic "B" depends on the cone angle of the poppet, the second section of the curve "C" depends on the relationship between the design of the poppet and its movement which is effected by the rate of the spring, generally the higher the spring rate the steeper the gradient. As more flow passes through the valve the relief curve will meet the orifice curve "D".

The performance of a direct acting relief valve can be altered by innovative poppet design.

By using the flow forces to help open the valve the effect of a high rate spring can be reduced and the gradient be kept relatively flat.

Figure 1 shows a section through a typical relief valve where the poppet design allows for a relatively low pressure rise due to increase in flow. A problem with this type of valve is that too much flow can cause the valve to have a negative pressure rise causing the valve to go unstable with fluctuating pressure.

The re-seat and repeatability of the valve depends upon the hysterisis. Internal seals cause friction against the bore as the valve tries to close. If a seal is under pressure then the hysterisis increases, graph 2.

Graph 1. Basic relief curve

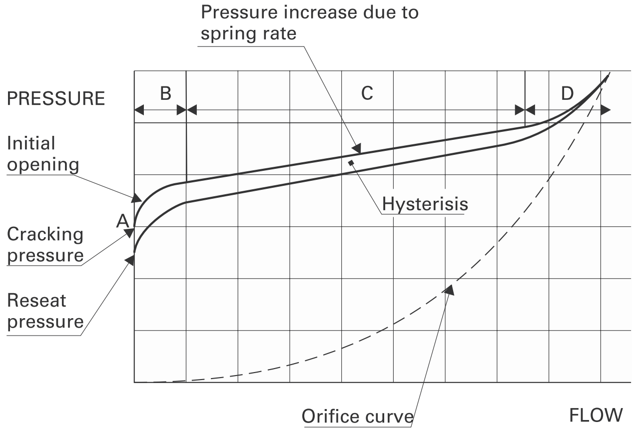
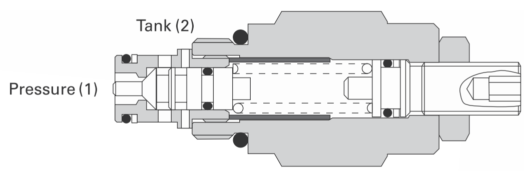


Figure 1. 1DR30 Direct acting poppet



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Relief valves

Article of Interest

Graph 2. Relief Curve showing the effect of hysterisis

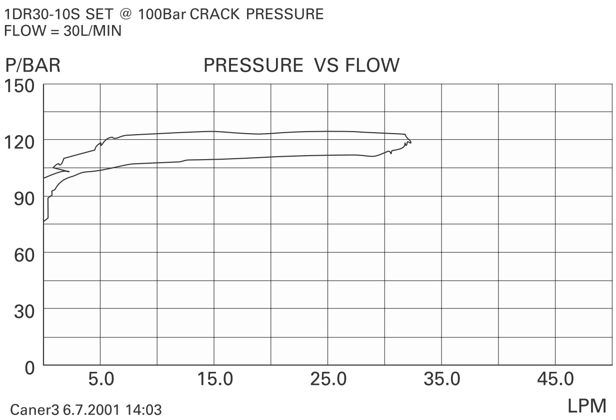
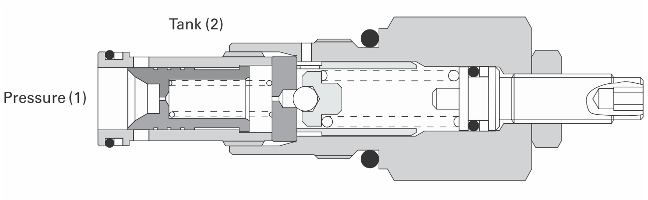


Figure 2. 1AR100 pilot operated spool



A poppet valve should not leak more than 1/3 cc/min up to the cracking pressure allowing it to be placed in a line where low leakage is important, and performing duties such as a service line relief.

A simple relief valve like this will give cost effective relief protection to small systems or where the valve is not the main pressure control but a pressure limiting device.

They are not generally suitable for high flows because the spring would have to be of excessively high rate which would give an unacceptably steep relief curve.

Figure 2 shows a typical pilot operated, spool type relief valve that gives good control over varying flows. This valve, due to its design, allows a high flow to pass with very little rise in inlet pressure. The valve has a good re-seat and good repeatability due to there being no internal seals. A pilot operated relief valve is suitable as a main pressure control but due to the two stage design it is not suitable for safety applications where speed of operation is important. In the case of a rapid increase in inlet pressure the system will be subject to a longer pressure spike than if a direct acting valve where used.

Increasing the flow capacity of a direct acting valve by reducing the area over which the pressure acts is possible, figure 3 shows a differential area poppet type relief valve that has the capability of very fast action and a high flow capacity for its size. The internal seal is subject to inlet pressure so the valve will display relatively poor re-seat characteristics.

Graph 3. Comparison of Pilot and direct acting differential area type opening curves

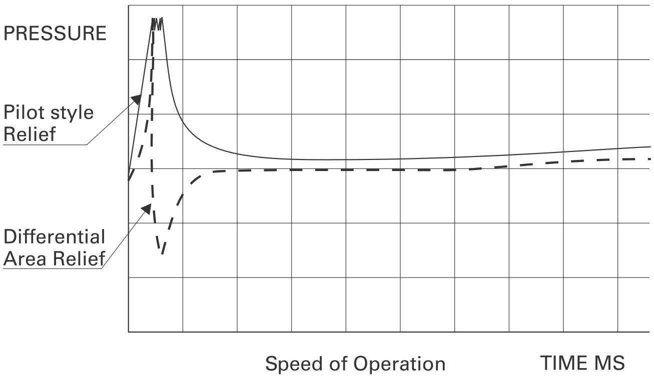
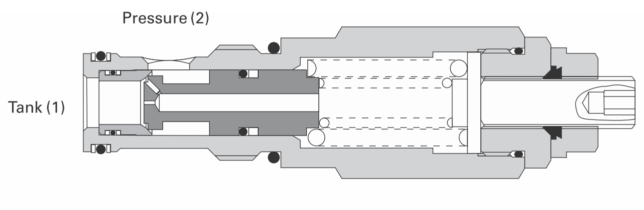


Figure 3. 1LR100 direct acting differential area



Graph 4. Opening characteristics of pilot style relief

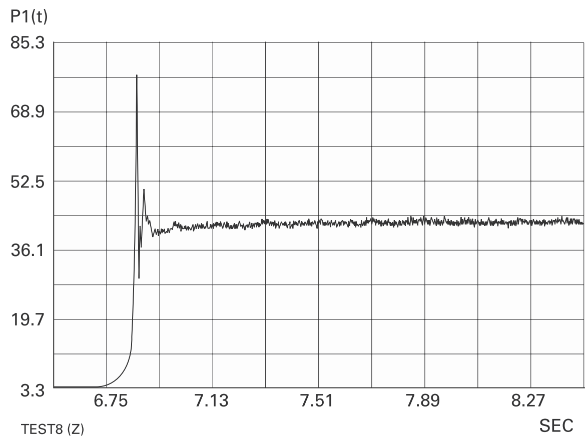
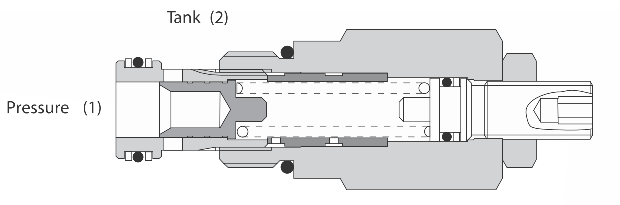


Figure 4. 1GR60 Direct acting spool



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The design of the poppet is such that as the valve begins to open the flow past the poppet draws oil from the spring chamber (by venturi effect over the small holes in the poppet annulus) causing initial over opening. This removes most of the pressure spike. The valve is therefore highly suitable as protection for actuators.

Graph 3 shows a comparison between the typical opening characteristics of a pilot style valve and a differential area direct acting valve with the special poppet described above. The difference between the opening characteristics of a pilot style relief valve and a direct acting spool type relief valve are shown in graph 4 & 5 respectively, graph 4 clearly illustrating the pressure spike permitted by the pilot style valve.

Figure 4 shows a spool type direct acting relief valve. These are suitable for low pressure systems where stable or constant operation is required. They provide quiet operation even with fluctuating pressures. The spool opens up a ring of holes in the sleeve that gives a more gradual increase in flow area than a poppet valve.

Spool valves will give between 50 and 100cc/min leakage before they open.

The four main types of relief valve as detailed above cover most applications but there are many variations on a theme that give flexibility to a systems design.

Ventable relief valves, figure 5, are used to provide an unloading function, presenting an ability to be remotely operated and the possibility of switching between more than one pressure.

Unloading relief valves or 'kick down' valves, figure 6, provide an off load of pressure when the setting is reached, the valve remains fully open until the pressure falls to zero. This removes any force created by an actuator that could cause mechanical damage within a system.

In order to simplify the design of a circuit and reduce its cost system designers frequently require a valve to perform additional functions, two such valves are shown below, figure 7 a relief valve in conjunction with bypass check and figure 8 a cross line relief valve.

When designing hydraulic systems it is important to consider the performance of the minor components such as relief valves. These may be minor in cost but they have a major impact in terms of value. A poor relief valve can effect the efficiency and life of a complete machine

From overall pressure control to actuator protection the relief has to be of the correct type to ensure sound performance and component integrity.

There are also electrically controlled proportional valves available that tie in with electronic systems. That is another subject but they should never be allowed to replace the humble mechanical relief valve, the correct application of which can permit a machine to operate to its optimum performance over a long period.

Graph 5. Opening characteristics of direct acting relief

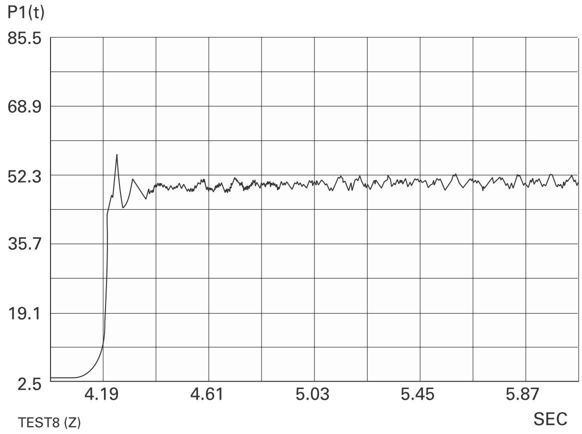


Figure 5. 1VR100 Ventable relief

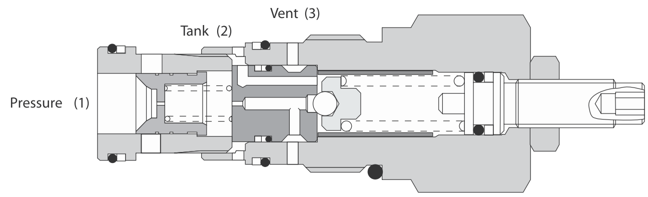


Figure 6. 1UAR100 Unloading relief

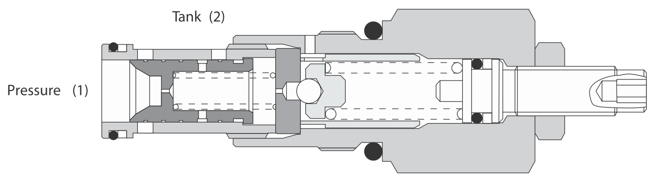


Figure 7. 1ACR100 Relief combined with by-pass check

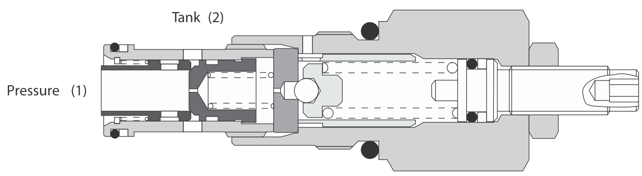
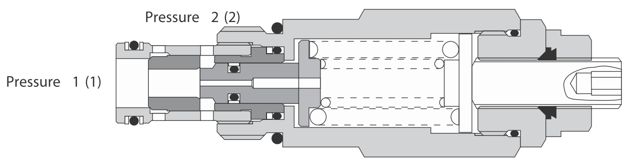


Figure 8. 1CLLR50 Dual relief



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Highway hydraulics

Article of Interest

Anyone who has attempted a long journey in a car will notice that the flow of traffic affects how quickly and safely you arrive at your destination. There are various ways that highway agencies seek to control these two elements. Speed cameras slow drivers down to enhance safety, variable speed limits keep the traffic flow at a constant and diversions limit the weight of traffic on any road. Some of these measures are more effective than others.

It would be great if we could accurately predict the time it takes to get from 'A' to 'B'. Unfortunately due to restrictions and accidents our journeys become more difficult and less efficient. Thankfully we do not have to deal with the random variables caused by human frailties when utilising hydraulics within a machine provided the designer uses the best available flow control valves within his system.

In hydraulics it is important to control the flow of oil to augment the safety and efficiency of a machine. The results will vary depending on the accuracy and repeatability of the valves used. It is important to understand the function and operation of the different types of flow regulator before applying them; from the simple needle valve to the proportional pressure compensated priority device. Each type has its place and there are applications that warrant the differing complexity.

We understand that flow takes place from high pressure to low pressure and the amount of flow is dictated by the pressure difference. By introducing restrictions into the line we can control the flow provided the excess flow is given a means of escape.

With a needle valve the flow will not be controlled until the inlet pressure reaches a point at which an upstream relief valve opens or a pump compensator operates to reduce the inlet flow and maintain a balance. The flow across a needle valve depends on the pressure difference across it so changes in outlet pressure will have an impact on the controlled flow. To overcome this problem compensators of various guises are introduced into the system.

There are three main types; Restrictive style, By-pass style and Priority style. The restrictive style flow control consists of a needle valve and compensator element. The compensator can be situated after the needle valve (See figures 1 & 4). Fluid pressure at the high-pressure side or inlet to the needle valve is sensed on one end of the compensating spool and pressure at the lower pressure side or outlet is sensed by the other end of the same spool. This spool is usually offset to the open position by a spring. The flow through the valve is determined by the position of the needle as the compensator will always ensure that pressure drop across the needle is constant, usually 7 bar. Opening the needle will permit the flow to rise until the pressure drop exceeds the setting of the compensator causing it to shift closing a ring of holes in its sleeve and thus limiting the flow.

Figure 1. Restrictive type flow regulator

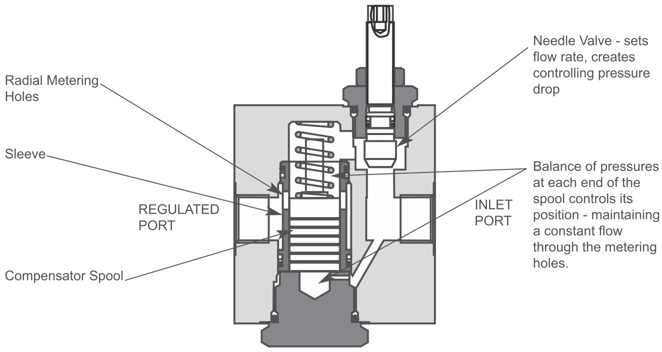


Figure 2. By-pass type flow regulator

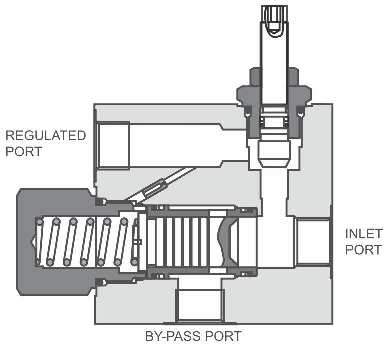
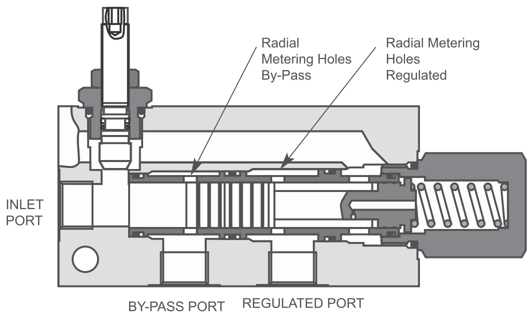


Figure 3. Priority type flow regulator



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As the flow through the valve will be lower than the flow being delivered by the pump the inlet pressure will rise to a point where an upstream relief will open feeding excess flow to tank or a pump compensator backs off reducing the inlet flow to that required to satisfy the flow setting of the flow regulator.

The by-pass style of flow control (Figure 2 & 5.) again consists of a needle valve and compensator element but has a third port to allow the excess flow to pass to tank at working pressure as opposed to maximum relief valve setting in the case of the restrictive style. The compensator spool is biased closed by a spring, inlet pressure acts to open the spool against the action of the spring and pressure downstream of the needle valve acts to close the spool in the direction of the spring. Flow through the compensator spool goes to tank. As the pressure drop increases across the needle valve the pressure difference is sensed across the compensator spool until it moves to open the inlet to the tank port. Increase in outlet pressure will tend to reduce the flow across the needle valve but the co-responding change in pressure drop will cause the compensator to restrict the line between the inlet and the tank line. In this way the spool will meter the bypass flow to maintain a constant controlled flow that relates to the force exerted by the spring and the orifice size created by the needle valve. With this type of flow regulator it is important that the tank line pressure is kept to a minimum as this may increase the flow through the regulated line above that required.

The priority style flow control (Figure 3 & 5.) is similar to the bypass style except that it allows the excess oil to be used for other functions even if the working pressure for this function is higher than the controlled flow pressure.

The flow from the pump enters the inlet port and passes across the needle valve, then on through the sleeve, past the compensating spool and out through the regulated port. The passage of oil across the needle valve creates a pressure difference which is sensed across the compensating spool. When the flow is sufficient to create a 7 bar pressure difference across the needle valve the compensating spool will begin to move uncovering the radial holes in the sleeve and opening up a path to the bypass port. Oil will therefore begin to pass to the bypass line. If the flow tries to increase across the needle valve, and so to the regulated port, there will be an increase in pressure difference sensed by the compensating spool causing it to move further against the spring, open further the line to the bypass port and limiting the flow to the regulated line. If the inlet flow falls below the setting of the valve the pressure difference across the needle valve will drop below the 7 bar needed to keep the bypass line open therefore the priority line is always satisfied before the bypass line opens. Changes in operating pressure on either of the two outlets will alter the inlet pressure to the higher of the two pressures (plus the control pressure which is 7 bar). If the working pressure in the regulated line is higher than that in the bypass line then the tendency would be for the flow to try to take the easy way out and flow down the bypass line. This would detract from the flow passing across the needle valve lowering the pressure difference causing the compensating spool to shift, increasing the restriction to the flow to the by-pass and reducing the restriction of the flow to the regulated line. In this way the compensating spool will maintain the regulated or priority flow at a constant level.

Figure 4. Restrictive type cartridge flow regulator

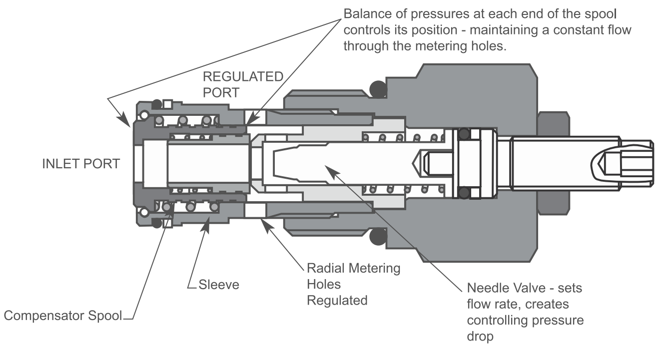


Figure 5. Priority type cartridge flow regulator also used for by-pass operation.

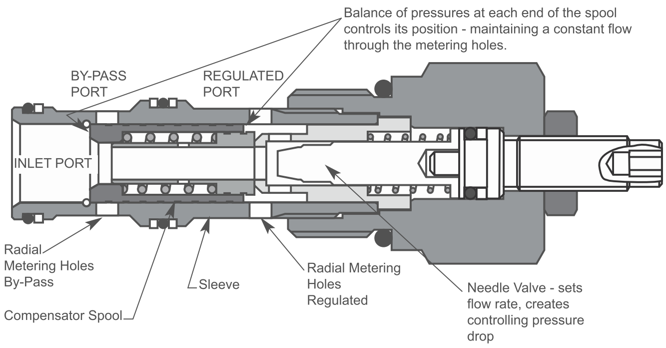
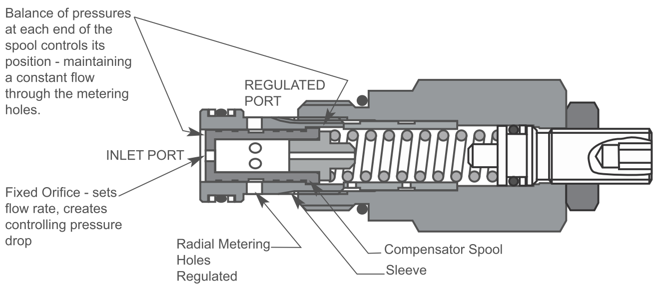


Figure 6. Restrictive type cartridge flow regulator



If the working pressure in the bypass line is higher than that in the regulated line then there would be a tendency for the flow to increase through the regulated line, increasing the flow would increase the pressure difference across the needle valve and cause the compensating spool to meter the regulated line.

If this is not successful then either an overcenter valve around the motor or a sequence valve before or after the actuator is necessary.

In today's environment electronic control requirements are becoming more common. To achieve this, the needle valve can be replaced with an electro-proportional valve. With electronic speed feed back, errors in the compensation can be corrected by adding electronic metering. Typical applications include salt or fertiliser spreaders where a relationship between the vehicle speed and the density of spread is required. For high flows it is sometimes necessary to use an electro-proportional pressure reducing valve to pilot open a separate orifice of some kind (figure 8.). This circuit uses a pilot operated poppet valve to provide the proportionally controlled orifice. Integrated hydraulics' zero differential valves benefit from having a shrouded seat which gives superior flow control as the poppet opens. The poppet is hydraulically balanced so it will open proportionally upon the application of pilot pressure up to 25 bar regardless of system or induced pressure.

Accurate regulation of flow is vital to the safe and efficient operation of hydraulic systems and therefore the machines on which they are used. Each application is different and will demand different control solutions and pressure compensated flow control devises, either mechanically or electronically operated, are designed to offer that accurate, cost effective, reliable and repeatable control.

Figure 7. Priority flow control with sequence valve (and unloading circuit)

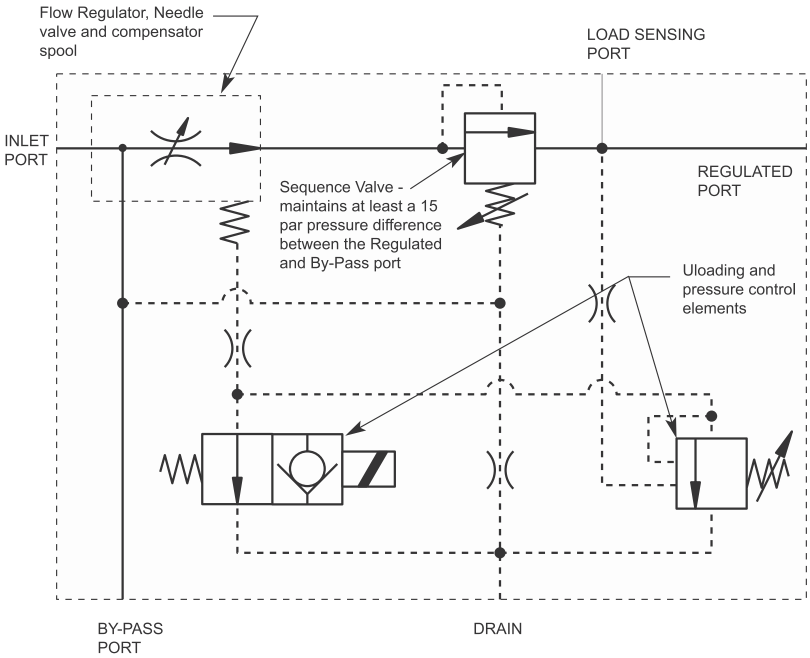
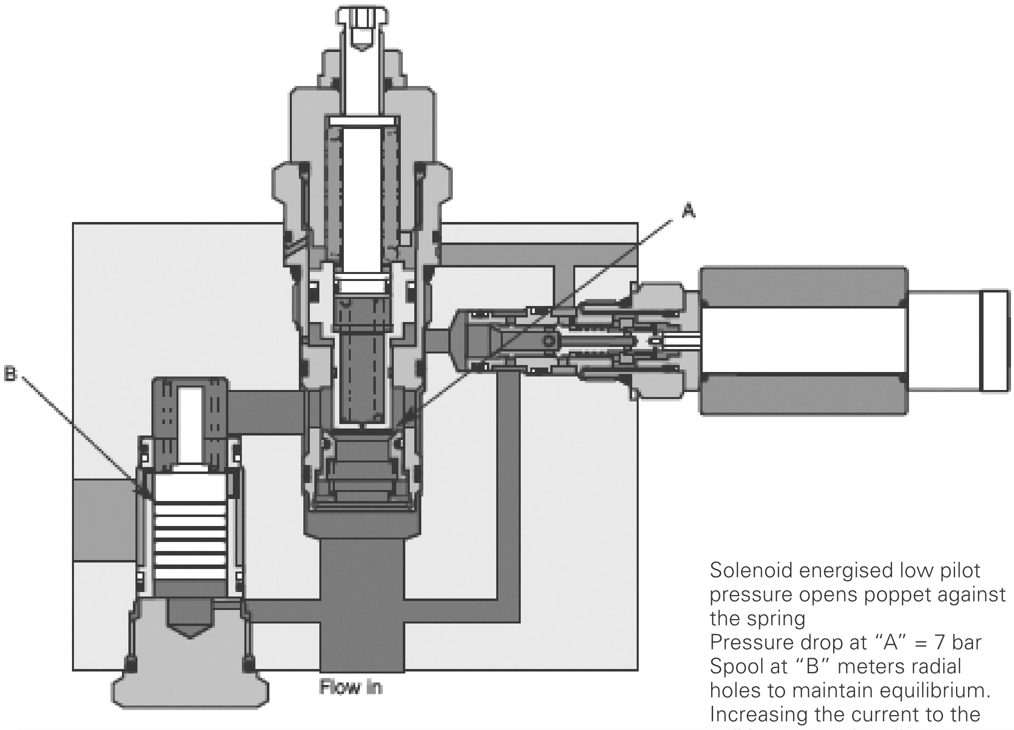


Figure 8. Proportional restrictive type flow control



Pressure Compensated proportionally controlled flow to directional valve

Solenoid energised low pilot pressure opens poppet against the spring
Pressure drop at "A" = 7 bar
Spool at "B" meters radial holes to maintain equilibrium.
Increasing the current to the coil increases the pilot pressure which opens the poppet valve, increasing flow at which 7 bar pressure drop is generated.

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In order for users of mobile plant to operate auxiliary equipment from the hydraulic system of the carrier so avoiding the need for an auxiliary power source accurate control of the flow and pressure is vital. The requirements of various attachments will differ both by type and manufacturer and the carriers operating system will also vary.

The following article charts the progress of several variants that solve the problem of differential pressure requirements whilst at the same time offering additional benefits. The standard Priority Flow Regulator (2FP) is the basis for a range of valves designed to provide priority flow and a bypass flow which can be used at different pressures. The setting is controlled by a simple needle valve with a compensating spool restricting the flow to the port working at the higher pressure thus maintaining the controlled flow from the regulated port. Before describing the variations in design and application it is necessary to understand the workings and performance of the standard priority style flow regulator as the normal operation of the variants (2FPH series) are the same.

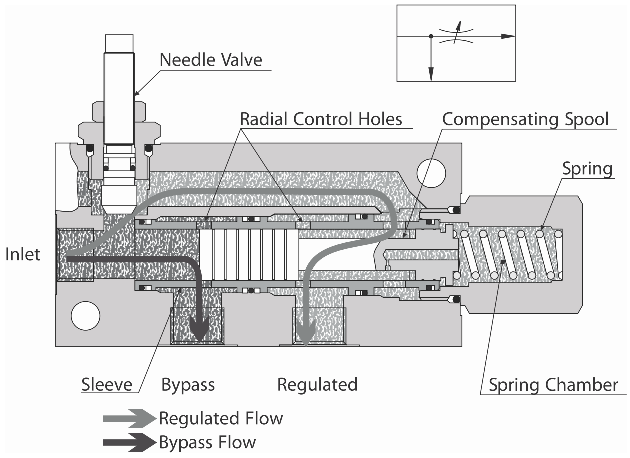
2FP series priority flow regulator

The flow from the pump enters the inlet port and passes across the needle valve, then on through the sleeve, past the compensating spool and out through the regulated port. The passage of oil across the needle valve creates a pressure difference which is sensed across the compensating spool. When the flow is sufficient to create a 7 bar pressure difference across the needle valve the compensating spool will begin to move uncovering the radial holes in the sleeve and

The standard Priority Flow Regulator (2FP) is the basis for a range of valves designed to provide priority flow and a bypass flow which can be used at different pressures.

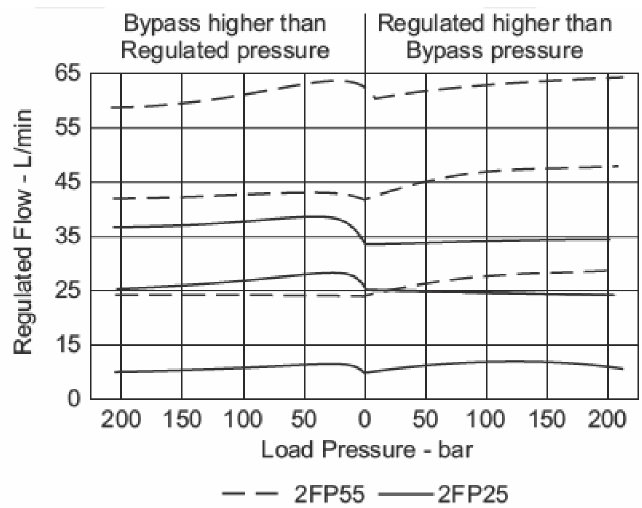
opening up a path to the bypass port. Oil will therefore begin to pass to the bypass line. If the flow tries to increase across the needle valve, and so to the regulated port, there will be an increase in pressure difference sensed by the compensating spool causing it to move further against the spring, open further the line to the bypass port and limiting the flow to the regulated line. If the inlet flow falls below the setting of the valve the pressure difference across the needle valve will drop below the 7 bar needed to keep the bypass line open therefore the priority line is always satisfied before the bypass line opens. Changes in operating pressure on either of the two outlets will alter the inlet pressure to the higher of the two pressures (plus the control pressure which is 7 bar). If the working pressure in the regulated line is higher than that in the bypass line then the tendency would be for the flow to try to take the easy way out and flow down the bypass line. This would detract from the flow passing across the needle valve lowering the pressure difference causing the compensating spool to shift, increasing the restriction to the flow to the by-pass and reducing the restriction of the flow to the regulated line. In this way the compensating spool will maintain the regulated or priority flow at a constant level.

If the working pressure in the bypass line is higher than that in the regulated line then there would be a tendency for the flow to increase through the regulated line, increasing the flow would increase the pressure difference across the needle valve and cause the compensating spool to meter the regulated line.



During the normal operation of any system utilizing this type of valve both the regulated and the bypass pressures will be constantly altering causing the compensating spool to meter. The valve will maintain the priority flow within +/- 10% of its setting throughout its range. The largest movement in flow will occur when pressure differential is transitional, the higher pressure varying between the bypass and regulated ports, this causes the compensating spool to move from metering one ring of holes in the sleeve to the other.

(See graph)



Attachment valves

Article of Interest

2FPH55, 95 & 195

The 2FPH series of flow regulators are based around the standard priority flow controls but with the addition of a pressure control and a solenoid vent both causing all of the flow to pass to the bypass port.

This type of flow regulator lends it's self to attachment circuits where a piece of ancillary equipment needs a controlled flow with a pressure limitation which is lower than the maximum working pressure of the carrier.

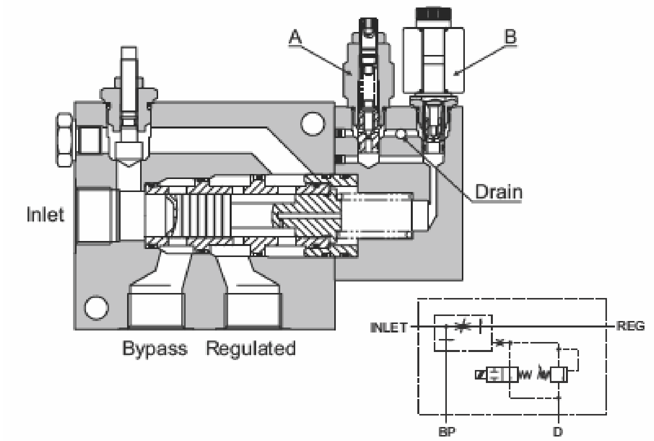


Figure 2. Priority type pressure compensated flow regulator with pressure control and overide- 2FPH55/95/195

They were originally developed for Hammer circuits where the hydraulic hammer requires a constant flow for efficient operation. There was also a pressure limit for safe operation which was well below the normal working pressure of Excavators. It was also necessary to maintain the other functions on the carrier to enable movement of the arm and the pecking action needed to fire the hammer into life. This was achieved by control of the spring chamber pressure in the standard priority flow regulators. The dampening orifice in the spool on these valves enabled the use of pilot valves which were mounted directly into the spring housing giving a very neat, compact solution. In normal operation these valves behave just as the 2FP range but if the pressure in the bypass circuit exceeds the setting of the relief valve ("A" in Fig 2) the compensator spool reacts by cutting ALL flow to the Regulated port, only when the pressure in the by-pass port drops below the setting of the relief will normal flow be resumed. When using this type of valve with a hammer it is an advantage to have the hammer turn off if the operator decides to put too much pressure on down stroke in an attempt to increase the speed of operation. In this way potential damage to the hammer is avoided. Energizing the Solenoid valve ("B" in Fig 2) has the same effect, venting one side of the compensator spool and causing it to block flow to the regulated port.

The range included three sizes which corresponded to the three larger sizes of priority flow regulator with rated regulated flows of 55, 95 and 195 L/min. At the time this covered the majority of the applications.

2FPH250/350

It was not long before the operators of Excavators realized that a power take off provided by this sort of valve enabled the use of more varied attachments without the extra power source and the associated pipe work. The disadvantage with the original valves was that the pressure sense for limiting the regulated pressure was referenced to the inlet. This meant that if the inlet pressure rose higher than the setting of the attachment then the valve would switch all of the flow to the bypass starving the attachment of oil. The valve therefore became limited to hammers and other high pressure attachments. The other alternative was to set the pressure control valve in the flow regulator high and fit an external relief valve on the regulated line.

Developments in Excavator control systems and also in the attachments demanded that a new design of valve was necessary. The two new additions to the range have been designed to accommodate the attachments which may require the bypass pressure to work higher than the regulated pressure while maintaining the function of the attachment. The pressure limit to the regulated line is achieved by using a pilot unloading valve ("A" in Fig 3) which senses pressure from the regulated line only. This allows the bypass pressure to rise above the regulated setting without affecting the operation of the attachment. Again in normal operation these valves behave just as the 2FP range but if the pressure in the regulated line rises above the setting of the unloading valve it will vent the spring chamber causing the compensator spool to shift cutting the flow to the regulated port. The resultant lack of flow will cause the pressure to drop, the unloading valve to close and the compensator to open the regulated port again. In practice this all happens smoothly and balance is maintained. By operating the pilot solenoid valve ("B" in Fig 3) and venting the spring chamber to tank, all of the flow will pass to the bypass port. While the valves will work perfectly adequately with hammers it will also work with generators, compactors, crushers and flail mowers. With this design it was also felt that increases in regulated flow were desirable hence the 250 and 350 L/min regulated flow rating.

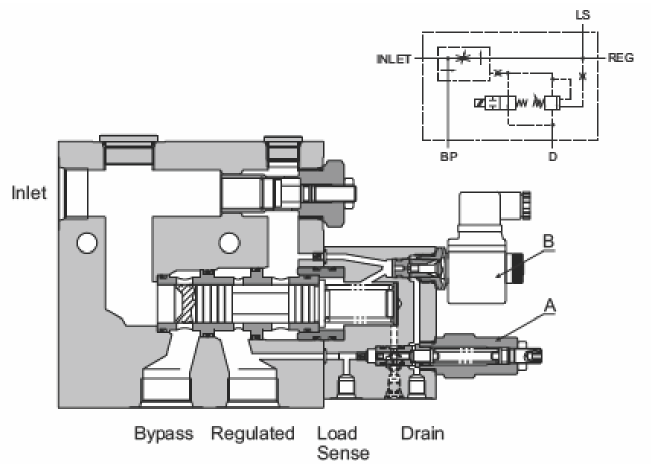


Figure 3. Priority type pressure compensated LS flow regulator with pressure control on the regulated port and overide - 2FPH250/350

Maintaining equilibrium

Article of Interest

In Hydraulics there is a loose rule that oil under pressure will take the easy way out! A little bit like human nature. If given the choice of an easy or difficult job most people will take the easy route. There are ways of encouraging people to be more balanced in their approach by applying a restriction to the easier task thus encouraging an equal work flow both for the difficult and the simple. This is not dissimilar to Hydraulics.

In a hydraulic system flow takes place from high to low pressure, pressure being the result of restriction to the movement of the oil. If one actuator provides less restriction than another then the former will move first. If the pressure to move the first actuator rises due to more restriction caused by the increasing flow then the second actuator may start to move but more slowly.

This can cause problems when two or more independent cylinders are required to move together. If they are unequally loaded then the cylinder that provides the least amount of resistance will move first.

There are many machines that have this problem both in linear and rotary movement including transmission circuits.

There are a number of answers to the problem but one of the simplest and most cost efficient is the humble spool type flow divider. Unfortunately misapplication of these valves can cause more system problems than solutions. In the first place it is important to understand how these valves work. Fig 1 shows a section through a typical unit.

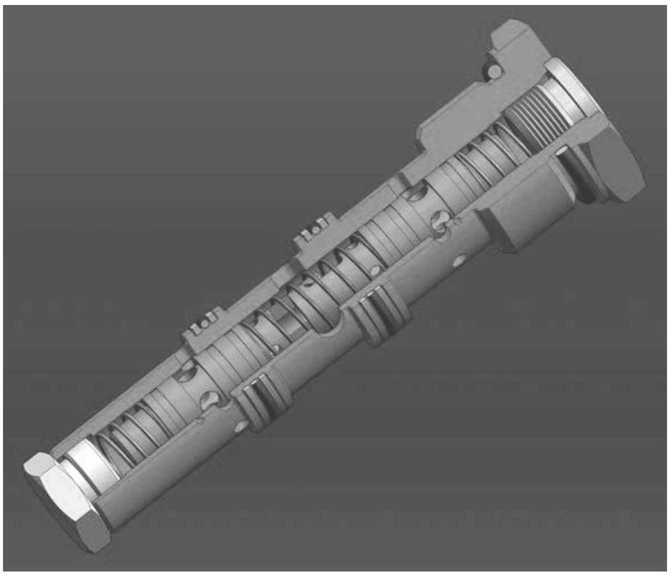
The valve functions as a flow divider and combiner by maintaining equal pressure drops through metering orifices situated in the two spools that are linked in this case by two 'lugs'. In the division mode the oil enters the valve through port '2', passing through both spools and out of ports '1' and '3'. The oil passes through the control orifices in the spools and if the flow is equal the spools remain in the central position as shown. If due to a change in outlet pressure there is a tendency for more oil to pass through one side than the other then the pressure drop rises in that side causing it to drag the other spool across eventually restricting the outlet of the higher flow side while keeping the lower flow side open.

As soon as the pressure drop through the control orifice in both spools is equal the assembly will maintain a metering position keeping the flow from both legs equal. Any change in the outlet pressures will cause the spool to move accommodating the change by metering the oil through the path of least resistance.

When the valve is being used as a combiner the spools will be pushed together as shown in 'Figure 2'. The oil then flows through the same orifices in the other direction combining out of port '2'. When there is a change in equilibrium the spools push each other to restrict the line of least resistance.

The accuracy of the valve depends on the size of the two orifices, the spring force and the leakage across the spools. If the tolerances on these items are kept to a minimum then accuracies of +/- 3% can be achieved.

There are a number of answers to the problem but one of the simplest and most cost efficient is the humble spool type flow divider.



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Maintaining equilibrium

Article of Interest

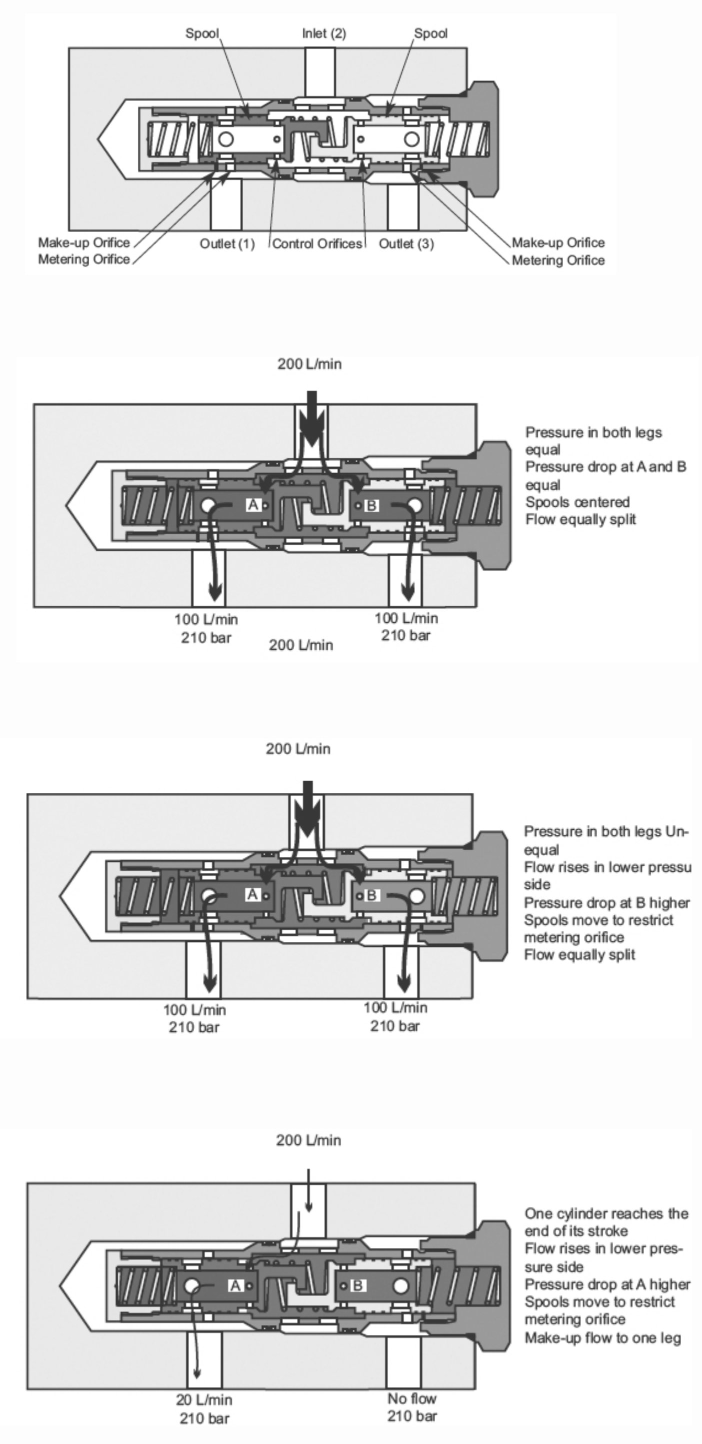

Figure 1. Flow Divider

Most production valves state an accuracy of +/- 10% on inlet flow. In some applications this can cause a problem when the cylinders are not flexible enough to accommodate this inaccuracy.

With this kind of design there is also a minimum flow at which the valve will operate. The relationship between the orifice diameter and the spring force opposing the movement of the spool means that there is a minimum flow before the spool will move and start to compensate.

If for some reason the flow from either leg is restricted then the spools will react to the offset pressure drops causing the spools to move to one end of the cartridge blocking off both outlets. This can be overcome by placing relief valves down stream of the flow divider to allow the flow to continue through a blocked or restricted outlet.

In cylinder applications the cylinders may not reach the end of stroke together. There will be a small make up flow but if relief valves are used the slower leg will catch up at 50% of the inlet flow. There are versions of flow divider that have extra holes to increase the make up flow. These however, are less accurate as the pressure difference between the two legs increases.

It is not necessary to fit flow dividers of this type on the inlet and outlet lines. The flow divider combiner will maintain equal division in both directions but care must be taken to size the flow divider to suit the outlet flow if it is in the full bore side of the cylinder as the flow will be increased by the rod/bore ratio of the cylinder.

It is not practical to cascade these valves to control more than two cylinders because the inaccuracy of the valves will be additive so you could end up with 20 to 30% difference in the flow.

It is also important that the valves are not over flowed. When in the dividing mode the pressure drop through the spools acts directly on the 'lugs', to rip the two spools apart. The normal factor of safety is 4:1 on ultimate tensile strength and as pressure drop through an orifice raise as the square of the increase in flow, so putting twice the rated flow through the valve will produce 4 times the pressure drop and probably break the 'lugs' The normal and most common division ratio is 50/50 but it is possible, by having different diameter orifices in the opposing spools, to produce offset ratios. The ratio of the orifice area in each spool will determine the offset flow ratio.

In spite of these draw backs there are many applications where the performance is good enough and therefore provide a cost effective solution to the problem of providing effective division of flow despite varying pressures in each actuator.

The accuracy of the valve depends on the size of the two orifices, the spring force and the leakage across the spools.

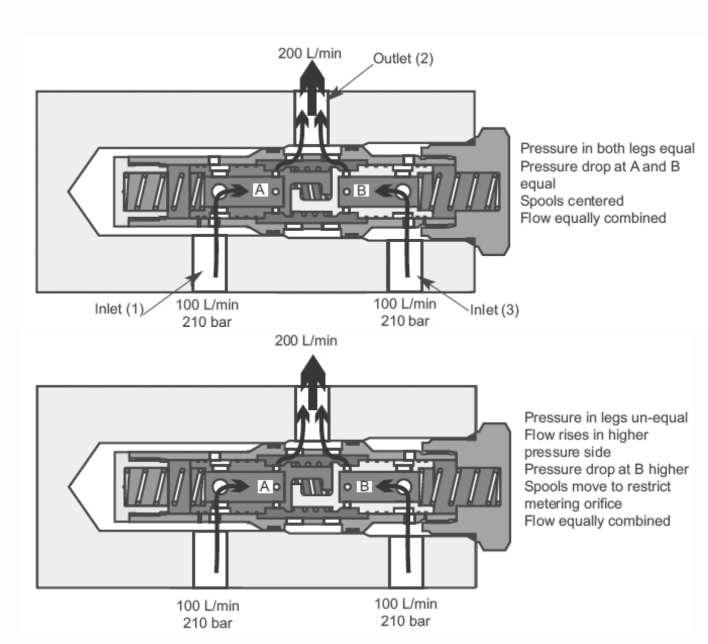

Figure 2. Flow combiner

A typical example is on the arms of a tarpaulin cover for tipping trucks. The arms, either side of the lorry, have to extend together first and then rotate together to unroll the tarpaulin and stretch it over the insecure load in the skip.

One of the most common applications for flow dividers is for wheel motors in transmission circuits to give an element of 'Diff-lock'. The flow divider will ensure that there is always traction to both wheels even when one of them is over soft or slippery ground.

Figure 3 shows a typical circuit where the flow divider is switched in when needed. The flow divider works on pressure drop so is intrinsically inefficient, even though the pressure drop is low. In a transmission circuit this pressure drop would create excessive heat so it is necessary to have a system to select the diff-lock when required.

In a closed loop transmission circuit this can be achieved by using logic elements around each side of the flow divider. They are vented when the difflock is not required. This will work in both directions because the charge pump pressure is sufficient to keep the logic element vented allowing the flow to pass backwards through the valve.

The bypass can be achieved by using pilot operated spool valves or solenoid valves. By using one of these options there is no loss of flow through the vent line however there is a limit to the flow rate that these kinds of valves can handle.

Most of these machines are required to go round corners so one wheel has to go faster than the other. As previously described this would cause problems to the flow divider. It is also true that it is not necessary to maintain a perfect division under conditions requiring diff-lock. A limited slip differential is acceptable.

It has become common place to fit an orifice across the two legs of the flow divider to allow flow to pass from one side to the other when the vehicle is turning. As the differential pressures go up due to the extra load on the inner wheel a controlled flow can pass from one side to the other. The size of this orifice depends on the turning circle of the machine. In this way one wheel will be able to move faster than the other but pressure and therefore torque will remain on both of the wheels.

There are a number of different designs of flow divider on the market including valves that have a wider range of flow. This can be an advantage if the inlet flow varies from very low to the maximum rating of the valve but at low flows there is still an inaccuracy. Some designs can cope with very low flows but as with all flow dividers they are not 100% accurate and they can cause more problems than they solve. At low flows the standard spool type flow divider will function as a simple "T" connector and it is sometimes preferable to use this feature at low flow rates and the gain their benefits at higher rates.

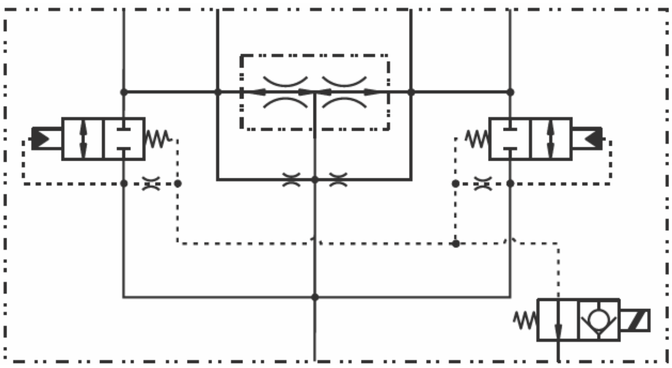

Figure 3. Difflock valve circuit

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Maintaining equilibrium

Article of Interest

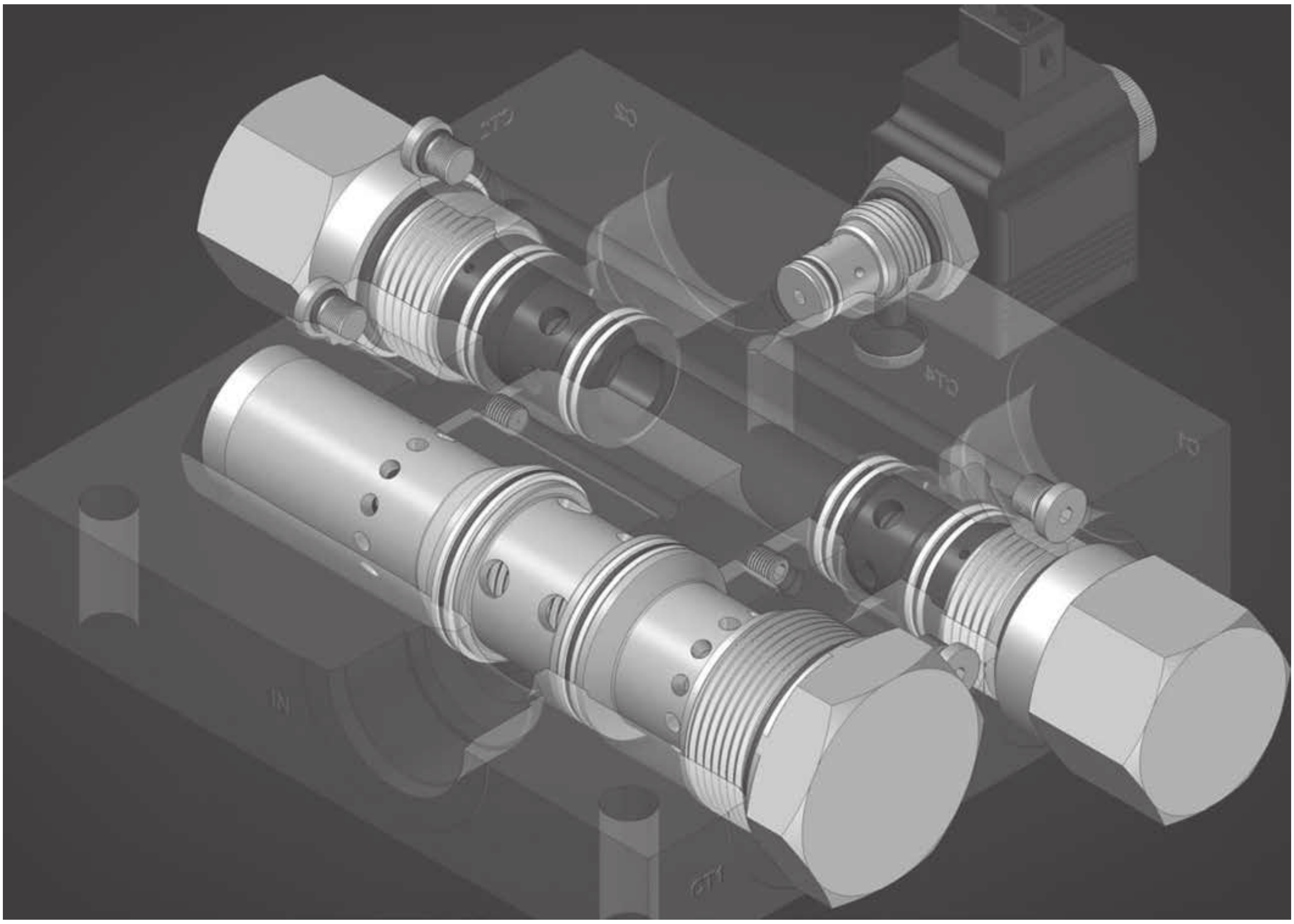

Figure 3. Diff-Lock valve including by-pass solenoid selector valves

Gear flow dividers are also available. These do not work in the same way, they are essentially two gear pumps/motors working in parallel on the same shaft. Each section will rotate at the same speed so the outlet flows remain constant from each leg. The accuracy of these depends on the leakage across the gears so is much better than the spool type. It is very important that relief valves are fitted to the outlet legs because if one leg meets a restriction the other legs will transmit their torque to the stalled line and intensify the pressure. If the gear flow divider has two sections then the pressure could double in the stalled line. If three sections are used then this pressure will triple. They do not suffer from the same pressure drop problem but are significantly more expensive.

Another way of maintaining equal movement on cylinders is the use of slave cylinders in which a number of double acting cylinders are securely and rigidly connected together at both rod and barrel and have a common inlet. Each outlet is connected individually to the inlet of the main working cylinders. (Obviously some provision for filling and maintaining the oil in the main cylinders has to be made.) As oil enters the common inlet of the slave cylinders the rods will extent together displacing fluid from each cylinder equally which operates the main working cylinders. This is a far more accurate way of controlling multiple cylinders but requires a more complicated control circuit and much more space.

A simple way of improving the accuracy and keeping the price down is to use restrictive style pressure compensated flow regulators mounted in parallel. It is important that there is always more flow available than the combined settings of the flow regulators. This solution requires the excess flow to go across a relief valve but can be used successfully with a pressure compensated pump. Each valve can be trimmed to give optimum performance.

The spool flow divider may appear to have a number of drawbacks but if we are to maintain cylinder or motor equilibrium in a cost effective manner then they are worth a look. Taking into account the point made previously proper application will give good results.

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