CHAPTER 8

Check Valves, Accumulators and Cylinders

what a check valve consists of

A check valve basically consists of a body with inlet and outlet ports and a movable member which is biased by spring force. The movable member can be a flapper, or plunger, but most often in hydraulic systems it is a ball or poppet.

how a check valve works

Fluid flow passes through a check valve in one direction only.

When system pressure at the check valve inlet is high enough to overcome the spring force biasing the poppet, the poppet is pushed off its seat. Flow passes through the valve. This is known as the check valve's free flow direction. When fluid flow enters through the outlet, the poppet is pushed on its seat. Flow through the valve is blocked.

check valves in a circuit

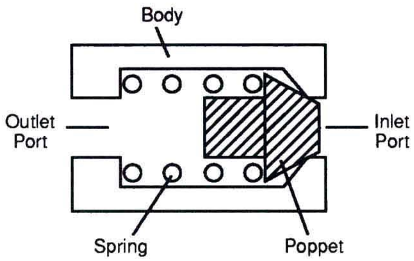
A check valve is a combination directional valve and pressure valve. It allows flow in only one direction and in that sense is a one-way directional valve.

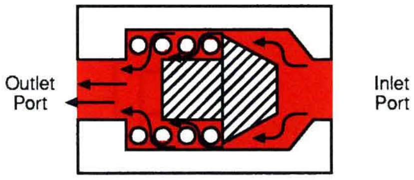
A check valve is often used in hydraulic systems as a bypass valve. It allows flow to get around components, like flow control valves which would restrict flow in a reverse direction.

A check valve is also used to isolate sections of a system or a system component, such as an accumulator. The check valve keeps an accumulator from dumping its flow over a relief valve or through the pump.

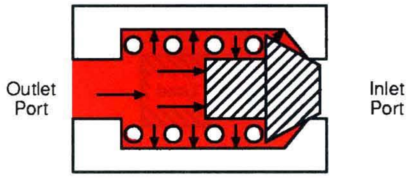
NOTE: Whenever a check valve is used in an accumulator circuit, surges in pressure and flow must be accounted for if the valve is to be selected and used properly. Selecting a valve type and size based only upon pump flow and relief valve pressure could result in severe component damage and malfunction. Component selection for such circuits is beyond the scope of this text. The circuit shown is simplified and used for illustration only.

The movable part in a check valve is usually biased by very light spring force. When a heavier

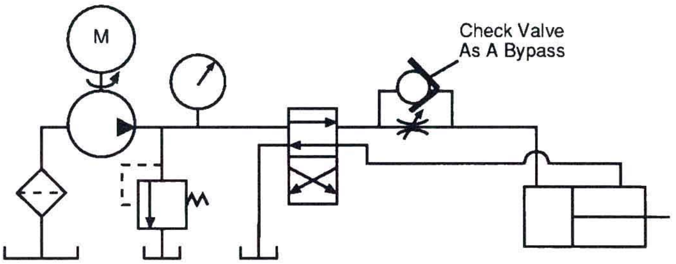


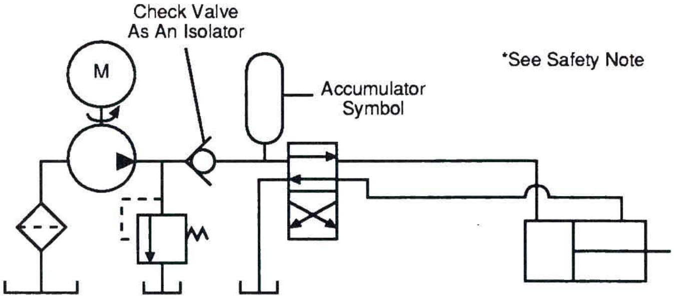


FREE FLOW

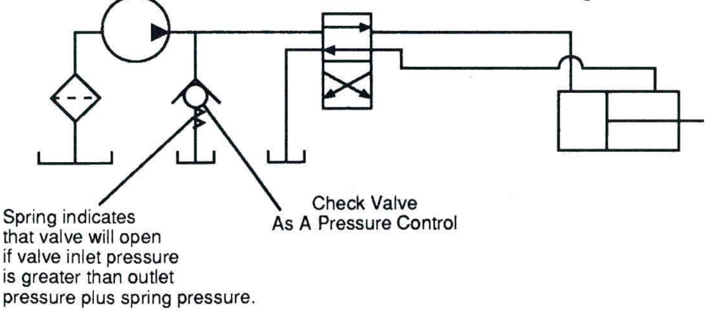


NO FLOW



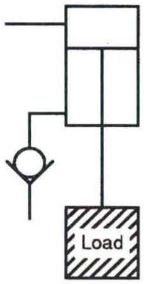


*Safety Note: In any accumulator circuit, a means could be available of automatically unloading the accumulator when the machine is shut down. See Page 8-6.



spring is used, a check valve can be used as a pressure control valve. (This is not commonly done).

suspending a load

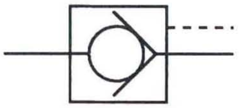


Hydraulic components that have a spool construction generally have a small bypass flow. This does not necessarily indicate that hydraulic components are poor quality since much of this bypass flow is actually designed into the component for lubrication reasons. Leakage does become a problem, however, when a load attached to a cylinder is required to be suspended indefinitely without drifting down. In this application, a sealing type check valve is used.

A check valve is generally, a low leakage device. As a matter of fact, check valves can be designed to be practically zero leakage devices.

A check valve may keep a load suspended almost indefinitely. But remember, a check valve is a one-way valve. When the time comes for the load to descend, the valve's movable member must be forced off its seat. This requires a special valve known as a pilot operated check valve.

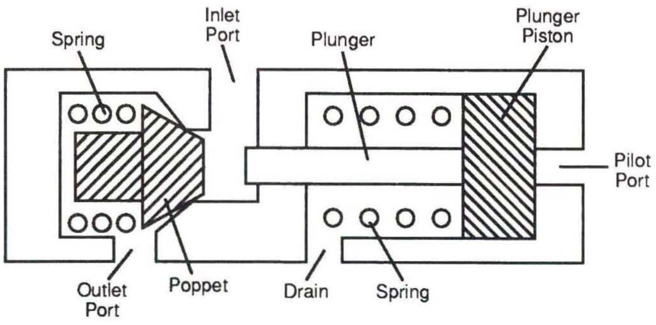
pilot operated check valve



Pilot Operated Check Valve Symbol

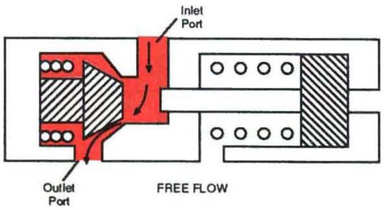
A pilot operated check valve allows free flow in one direction. In the opposite direction flow may pass when pilot pressure unseats the valve's movable member.

what a pilot operated check valve consists of

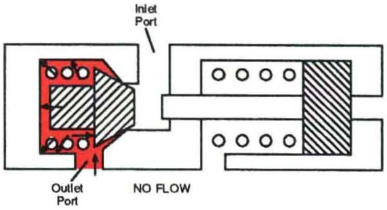


A pilot operated check valve consists of a valve body with inlet and outlet ports and a poppet biased by a spring, just as an ordinary check valve. Directly opposite the check valve poppet is a plunger and plunger piston which is biased by a light spring. Pilot pressure is sensed at the plunger piston through the pilot port. The plunger spring chamber has a drain.

how a pilot operated check valve works



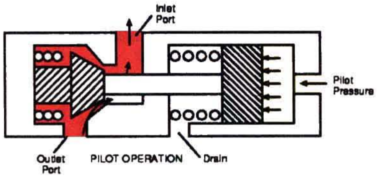
FREE FLOW



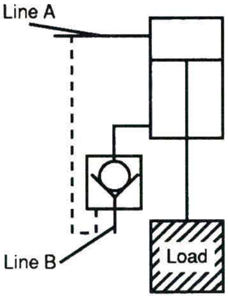
NO FLOW

A pilot operated check valve allows free flow from its inlet port to its outlet port just as an ordinary check valve.

Fluid flow attempting to pass through the valve from outlet to inlet port will force the poppet on its seat. Flow through the valve is blocked. When enough pilot pressure is sensed at the plunger piston, the plunger is moved and unseats the check valve poppet. Flow can pass through the valve from outlet to inlet as long as sufficient pilot pressure is acting on the plunger piston.



pilot operated check valves in a circuit



With a pilot operated check valve blocking flow out of cylinder line B, the load will stay suspended as long as the cylinder seals remain effective. When it is time to lower the load, system pressure is applied to the cylinder piston through line A.

Pilot pressure to operate the check valve is taken from this cylinder line. The check valve will remain open as long as enough pressure is available in line A.

To raise the load, fluid can easily pass through the valve since this is the valve's free flow direction.

In some applications, it is required that a load attached to a cylinder's piston rod be locked in place. To fill this requirement, a pilot operated check valve can be positioned in each cylinder line. The pilot operated check valve will block flow out of the cylinder. The load will be held as long as the cylinder seals remain effective and there are no leaks (line, cylinder, check valve, etc.).

NOTE: For absolute hold — special locking cylinders are required. They contain mechanical locks. Mechanically locking the load is the safest means of holding a load.

hydraulic accumulators


Accumulator Symbol

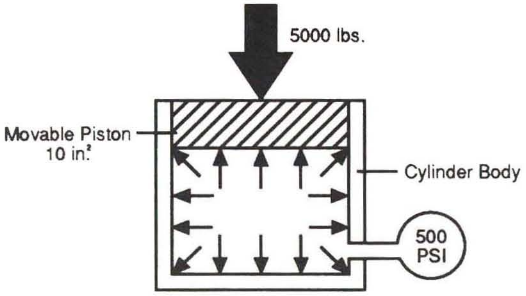
An accumulator stores hydraulic pressure. This hydraulic pressure is potential energy since it can change to working energy.

accumulator types

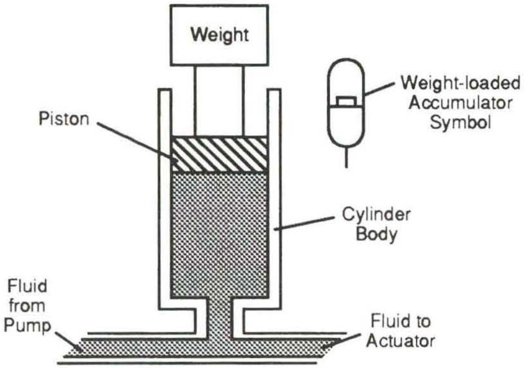
Hydraulic accumulators can be divided into weight-loaded, spring-loaded or hydro- pneumatic. Each classification identifies by what means an accumulator maintains a force on a liquid while it is stored.

weight-loaded accumulator

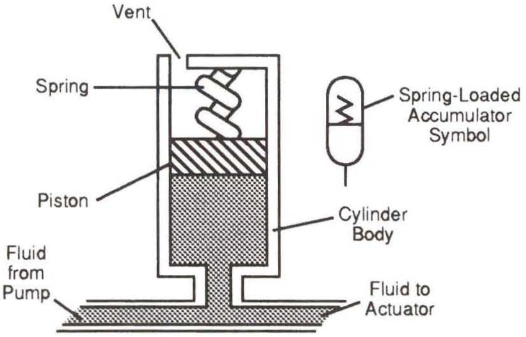
A weight-loaded accumulator maintains a force on the liquid it stores by means of heavy weights acting on a piston or ram. The weights can be made of any heavy material such as iron, concrete or even water.



Accumulator



Weight-Loaded Accumulator



Spring-Loaded Accumulator

Weight-loaded accumulators are generally quite large in some cases holding hundreds of gallons. They can service several hydraulic systems at one time and are most often used in mill and central hydraulic systems.

A desirable characteristic of a weight-loaded accumulator is that it stores fluid under a relatively constant pressure whether it is full or nearly empty; this will not be the case in other accumulator types. Because the weight applying the force to the liquid does not change, the same force is applied regardless of how much liquid is present in the accumulator.

An undesirable characteristic of a weight- loaded accumulator is shock generation. When a weight-loaded accumulator, discharging quickly, is suddenly stopped, the inertia of the weight could cause excessive pressure surges in a system. This can result in leaking fluid conductors and fittings, and early component failure due to metal fatigue.

spring-loaded accumulator

A spring-loaded accumulator applies a force to its stored liquid by means of a spring acting on a piston.

Spring-loaded accumulators are generally much smaller than weight-loaded accumulators with sizes holding up to several gallons. Spring-loaded accumulators usually serve individual hydraulic systems and generally operate at low pressure.

As liquid is pumped into a spring-loaded accumulator, the stored fluid pressure is determined by the compression rate of the spring. An accumulator of this type will have more stored pressure with the piston moved up and the spring compressed 10 inches than if it were only compressed 4 inches.

To avoid accumulation of leakage fluid, the spring chamber of a spring-loaded accumulator is vented. Leakage fluid will eventually discharge from the vent hole.

Spring-loaded accumulators are not externally drained back to tank because they can cause oil foaming. With an external drain terminating either above or below fluid level, leakage accumulated above the piston will tend to foam during accumulator operation. As the accumulator discharges rapidly, fluid above the piston will be unable to keep up with piston movement. A less-than-atmospheric pressure will be generated in the spring chamber resulting in dissolved air coming out of the liquid. When the accumulator is recharged, the piston moves up pushing the aerated oil to tank. Since air bubbles in a reservoir are undesirable, spring-loaded accumulators are not generally externally drained.

With spring chamber vented, spring-loaded accumulators demand immediate attention once their piston seal wears. If maintenance is not performed on a spring-loaded accumulator with a poor seal, a housekeeping problem could arise.

hydro-pneumatic accumulator

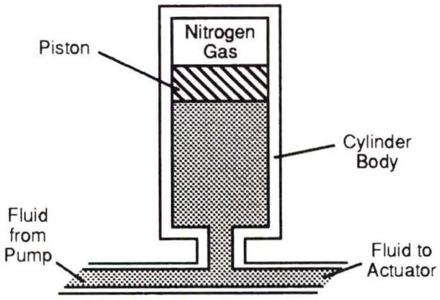
A hydro-pneumatic accumulator is the most commonly used accumulator in industrial hydraulic systems. This type accumulator applies a force to a liquid by using compressed gas.

Hydro-Pneumatic Accumulator Symbol

NOTE: In all cases of hydro-pneumatic accumulators applied to industrial systems, dry nitrogen is used. **COMPRESSED AIR SHOULD NEVER BE USED** because of the danger of exploding an air-oil vapor.

Hydro-pneumatic accumulators are divided into piston, diaphragm and bladder types. The name of each type indicates the device separating gas from liquid.

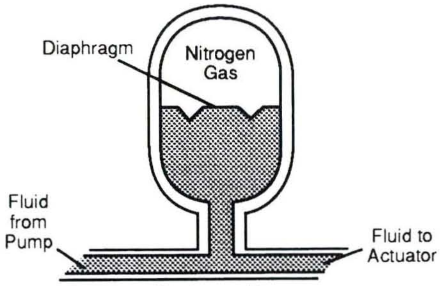
piston type accumulator



Piston Type Accumulator

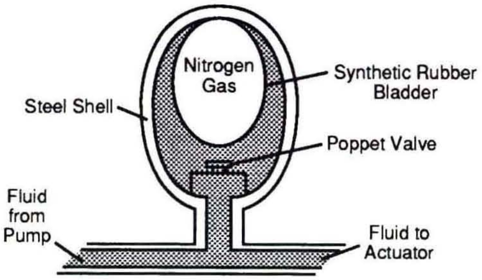
A piston type accumulator consists of a cylinder body and movable piston with resilient seals. Gas occupies the volume above the piston and is compressed as the cylinder body is charged with fluid. As fluid flows from the accumulator, gas pressure drops. When all liquid has been discharged, the piston has reached the end of its stroke and it covers the outlet keeping the gas within the accumulator.

diaphragm type accumulator



Diaphragm Type Accumulator

A diaphragm type accumulator consists of two metal hemispheres which are bolted together , but whose interior volume is separated by a synthetic rubber, gas occupies the space. As fluid enters the other chamber, gas is compressed. Once all liquid has been discharged, the diaphragm covers the outlet retaining the gas within the accumulator. The diaphragm will not be pushed through the thickness of the diaphragm.



Bladder Type Accumulator

bladder type accumulator

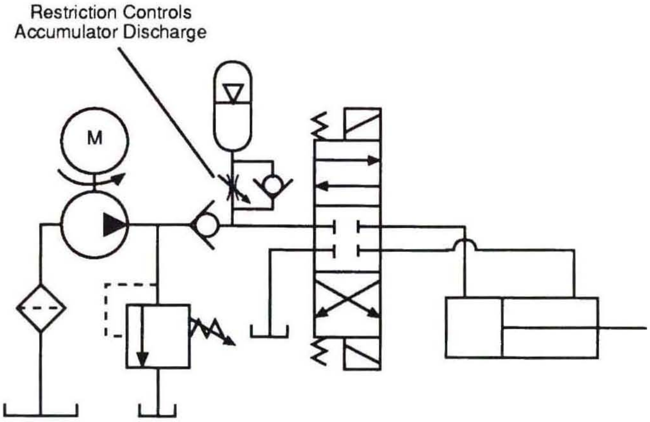
A bladder type accumulator consists of a synthetic rubber bladder inside a metal shell; the bladder contains the gas. As fluid enters the shell, gas in the bladder is compressed. Gas pressure decreases as fluid flows from the shell. When all liquid has been discharged, gas pressure attempts to push the bladder through the outlet. But, as the bladder contacts the poppet valve at the outlet, flow from the shell is automatically shut off.

We have seen the various types of accumulators. In the next section, we find that all three types can be used to develop system flow and maintain pressure. And, hydro-pneumatic accumulators can be used to absorb shock.

accumulators in a circuit

Accumulators can perform a variety of functions in a hydraulic system. Some of these are maintaining system pressure, supplementing pump flow, and absorbing system shock.

developing flow



Developing System Flow

SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

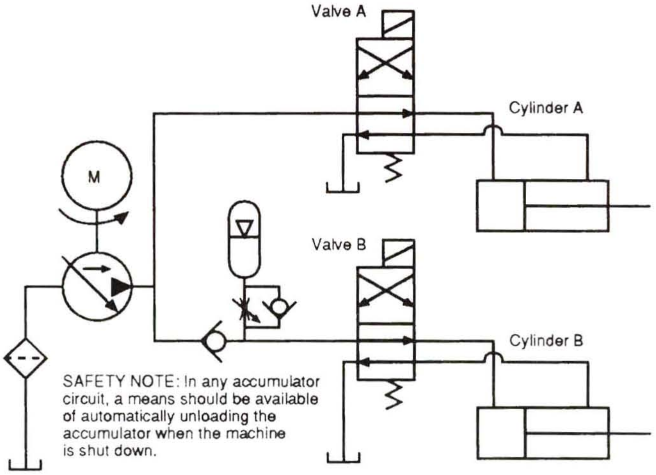
Developing liquid flow is one accumulator application. Since charged accumulators are a source of hydraulic potential energy, stored energy of an accumulator can be used to develop system flow when system demand is greater than pump delivery. For instance, if a machine is designed to cycle infrequently, a small displacement pump can be used to fill an accumulator over a period of time. When the moment arrives for the machine to operate, a directional valve is shifted downstream and the accumulator delivers the required pressurized flow to an actuator. Using an accumulator in combination with a small pump in this manner conserves peak horsepower. For instead of using a large pump/electric motor to generate a large horsepower all at once, the work can be evenly spread over a time period with a small pump/ electric motor.

maintaining pressure

Accumulators are used to maintain pressure. This can be required in one leg of a circuit while pump/ electric motor is delivering flow to another portion of the system.

In the circuit illustrated, two clamp cylinders are required to hold a part in place. As the directional valves are shifted, both cylinders extend and clamp at the pump's compensator setting. During this time, the accumulator is charged to the setting also.

System demands require that cylinder B maintain pressure while cylinder A retracts. As directional valve A is shifted, pressure at the pump as well as in line A drops quite low. Pressure at cylinder B is maintained because the accumulator has stored sufficient fluid under pressure to make up for any leakage in line B.

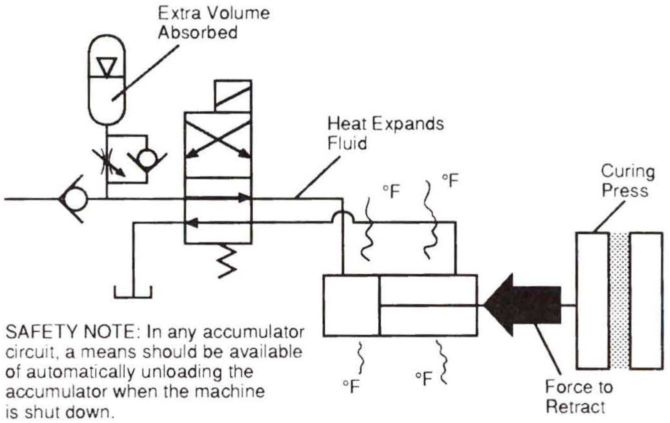


SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

Maintaining Pressure

Accumulators not only maintain pressure by compensating for pressure loss due to leakage, but they also compensate for pressure increase due to thermal fluid expansion or external mechanical forces acting on a cylinder.

In the illustrated circuit, assume that the cylinder is operating near a furnace where ambient temperatures are quite high. This causes the fluid to expand. With an accumulator in the circuit, the excess volume is taken up keeping the pressure relatively constant. Without an accumulator, pressure in the line would rise uncontrollably and may cause a component housing, fitting or conductor to crack.



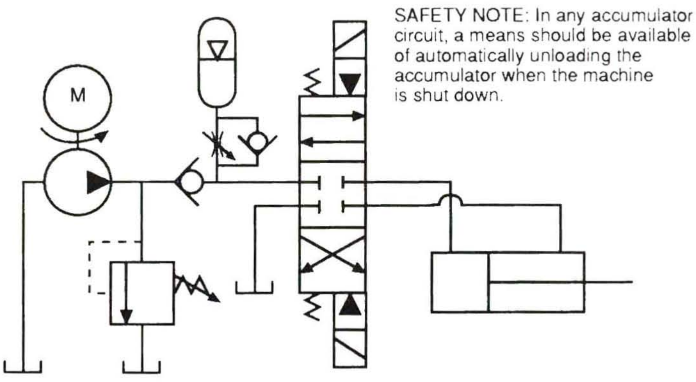
SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

The same situation can also occur if an external mechanical force acts to retract the cylinder. Assume now that the cylinder is clamping a curing press. As curing occurs, heat within the press causes it to expand resulting in a force acting to retract the piston rod. The accumulator once again absorbs the additional volume, maintaining the pressure at a relatively constant level.

absorbing shock

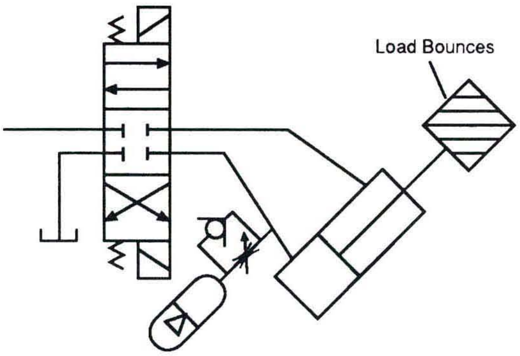
Hydro-pneumatic accumulators are sometimes used to absorb system shock even though in this application they are difficult to properly design into a system.

Shock in a hydraulic system may be developed from the inertia of a load attached to a cylinder or motor. Or, it may be caused by fluid inertia when system flow is suddenly blocked or changes direction as a directional valve is shifted quickly. An accumulator in the circuit will absorb some of the shock and not allow it to be transmitted fully throughout the system.



SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

In the circuit illustrated, a pump/electric motor is delivering 100 gpm (379 lpm) to a cylinder at the required working pressure. If the closed center directional valve upstream from the cylinder is centered while work is occurring, the pressurized flow of 100 gpm (379 lpm) will be stopped all at once resulting in hydraulic shock reverberating in the system. An accumulator positioned ahead of the directional valve absorbs and reduces the shock effects.



SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

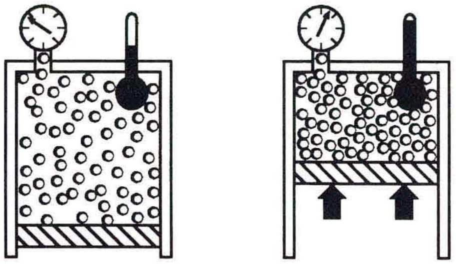
Shock may also occur in a hydraulic system due to external mechanical forces. In the circuit illustrated, the load attached to the cylinder has a tendency to bounce causing the rod to be pushed in and shock generated. An accumulator positioned in the cylinder line can help reduce the shock effects if properly precharged. If not, it could become overpressurized.

Since hydro-pneumatic accumulators are most common, we will for the remaining lesson deal with their operating characteristics. We will see what isothermal and adiabatic charging is and how precharge affects their operation.

isothermal and adiabatic charging

Since hydro-pneumatic accumulators use compressed gas to maintain pressure on a liquid, gas properties affect accumulator operation.

As a hydro-pneumatic accumulator is filled with liquid, gas is compressed, and that as a gas is compressed it heats up. With pressure remaining constant, a heated gas occupies more space than a gas at a lower temperature.

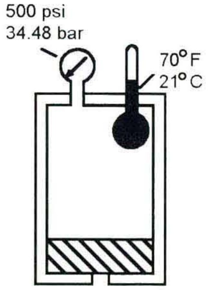


Gas Compression

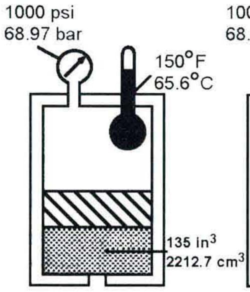
Isothermal describes the operation of an accumulator as the gas is maintained at a constant temperature. While an accumulator is being filled, isothermal operation indicates that the gas is being compressed slowly enough for the heat of compression to dissipate.

Adiabatic describes the operation of an accumulator as gas temperature changes. While an accumulator is being filled, adiabatic operation indicates that the gas is being compressed rapidly so that all heat of compression is retained.

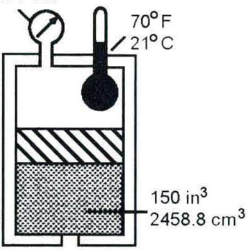
A hydro-pneumatic accumulator which is being charged with liquid up to a certain pressure, will hold more liquid if it is charged isothermally rather than adiabatically.



Gas at rest



Gas compressed adiabatically to a pressure



Gas compressed isothermally to same pressure

Illustrated is a piston accumulator void of liquid. A pressure gage at accumulator top indicates a gas pressure of 500 psi (34.48 bar); a thermometer indicates a temperature of 70°F (21°C). Assume that the accumulator is going to be filled with liquid until a pressure of 1000 psi (68.97 bar) is reached.

Fluid ceases to enter the accumulator at this point because this is the maximum pressure allowed to be developed by pump/electric motor.

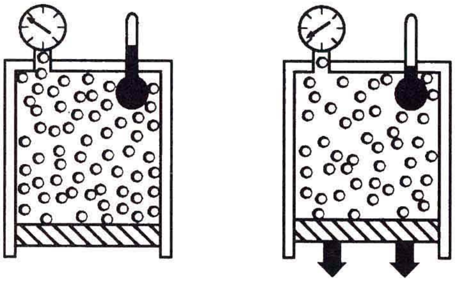
As the accumulator is charged adiabatically, pressure and temperature begin to climb. When pressure reaches 1000 psi (68.97 bar), fluid ceases to enter accumulator inlet. At that point, the temperature is 150°F (65.6°C) and the accumulator holds 135 in^3 (2212.65 cm^3) of fluid.

Assume now, that the accumulator is charged isothermally. Pressure begins to climb, but temperature remains the same. Charging takes place so slowly that heat of compression dissipates. When a pressure of 1000 psi (68.97 bar) is reached, fluid ceases to enter accumulator inlet. At that point, gas temperature is still 70°F (21°C) and the accumulator holds 150 in^3 (2458.5 cm^3) of fluid.

A hydro-pneumatic accumulator operated isothermally (slowly) will be charged with more liquid than if it were operated adiabatically (quickly).

isothermal and adiabatic discharging

As a hydro-pneumatic accumulator discharges liquid the gas expands and as a gas expands it cools. With pressure remaining constant, a cool gas occupies less space than a gas at an elevated temperature.

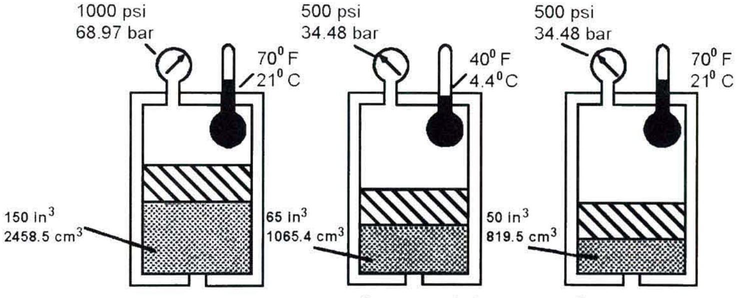


Gas Expansion

Isothermal and adiabatic describe the operation of an accumulator as it discharges fluid. An accumulator discharging fluid under isothermal conditions, indicates that discharge occurs slowly as gas expands; it is capable of acquiring heat from the ambient through accumulator walls or from the fluid. Adiabatic operation indicates that discharging occurs rapidly with no heat gain; as gas expands it cools.

A hydro-pneumatic accumulator which is discharging liquid until a lower pressure is reached, will discharge more liquid if it is discharged isothermally rather than adiabatically.

Illustrated is a piston accumulator which is charged with liquid to a pressure of 1000 psi (68.97 bar). A thermometer at accumulator top indicates a gas temperature of 70°F (21°C). Assume that the accumulator holds 150 in^3 (2458.5 cm^3) of fluid. Assume further that when a directional valve downstream is shifted, fluid will be discharged until accumulator pressure reaches 500 psi (34.48 bar). Fluid ceases to discharge from the accumulator at this time because 500 psi (34.48 bar) is the working



Gas compressed

Gas expanded adiabatically to a pressure

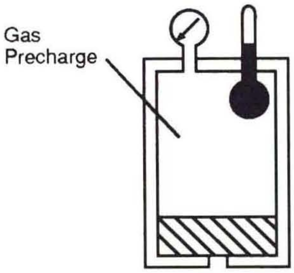
Gas expanded isothermally to same pressure

pressure of the system. This means a pressure differential no longer exists at this point to develop a discharge flow.

As the accumulator discharges adiabatically, gas pressure and temperature begin to drop. When gas pressure reaches 500 psi (34.5 bar), fluid ceases to exit the accumulator. At that point, temperature of the gas is 40°F (4.4°C) and the accumulator holds 65 in^3 (1065.35 cm^3) of fluid. 85 in^3 (1393 cm^3) of fluid has therefore been discharged.

Assume, now, that the accumulator is discharged isothermally. Pressure begins to drop, but gas temperature remains the same. Since discharging takes place slowly, the gas is able to acquire heat from the walls of the accumulator. When 500 psi (34.5 bar) is reached, fluid ceases to exit the accumulator. At that point, temperature is still 70°F (21°C) and the accumulator holds 50 in^3 (819.5 cm^3) of fluid. 100 in^3 (1639 cm^3) of fluid has therefore been discharged.

We have seen that more fluid enters and exits a hydro-pneumatic accumulator as it is operated isothermally. But, this is usually an ideal situation. Ordinarily, accumulators are charged and discharged adiabatically. In the following section, we find that the biggest concern is not how much the accumulator holds, but how much fluid is discharged before a lower pressure is reached. This is largely affected by gas precharge.



SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

precharge

The gas pressure present in a hydro-pneumatic accumulator when it is drained of hydraulic fluid is the accumulator precharge. This pressure significantly affects a hydro-pneumatic accumulator's usable volume and operation as a shock absorber.

precharge affects usable volume

A hydro-pneumatic accumulator which is used to develop system flow or maintain pressure operates between a maximum and minimum pressure. An accumulator is filled with fluid until a maximum pressure is attained. At an appropriate time, it discharges fluid until a lower pressure is reached. The accumulator is then recharged until the max pressure is reached. The liquid volume discharged between the two pressures is the accumulator's usable volume.

Gas precharge affects usable volume of an accumulator. This may be best shown by an example.

Assume that a 231 in³ (3786 cm³) hydro-pneumatic accumulator is used to develop flow in a particular system. The accumulator is charged with liquid from a pump until a system pressure of 2000 psi (137.9 bar) is reached. To develop a flow, it is allowed to discharge to 1500 psi (103.4 bar). The gas precharge selected will determine the amount of fluid which the accumulator pushes into the system.

From the illustrated chart, it can be seen that a 231 in³ (3786 cm³) accumulator with a gas precharge of 100 psi (6.89 bar) holds 210 in³ (3441.9 cm³) of hydraulic fluid at 2000 psi (137.9 bar) if charged isothermally. (The upper value numbers in each row indicate isothermic operation.) When discharged to 1500 psi (103.4 bar), the accumulator holds 202 in³ (3310.8 cm³). Between the two points, 8 in³ (131 cm³) have been discharged. With a 100 psi (6.89 bar) precharge, this accumulator holds much liquid, but discharges little.

With gas precharge increased to 1000 psi (68.96 bar), accumulator holds 93 in³ (1524.3 cm³) of oil at 2000 psi (137.9 bar) and 59.5 in³ (975 cm³) at 1500 psi (103.4 bar). 33.5 in³ (549.1 cm³) have been discharged.

In this instance, the accumulator does not hold as much fluid as before; however, more fluid is discharged. Increasing the precharge to 1400 psi (96.6 bar), we find that the accumulator holds the least amount of fluid, but the most is discharged. At 2000 psi (138 bar), 53.5 in³ (876.9 cm³) of oil are held by the accumulator. At 1500 psi (103.5 bar), 11.6 in³ (190.1 cm³) are contained. Usable volume in this case is 41.9 in³ (686.7 cm³).

As a hydro-pneumatic accumulator develops pressurized flow or maintains pressure, usable volume is an important consideration. While developing a flow, a certain amount of fluid must discharge between two pressures in order for system demands to be met. While maintaining pressure, an accumulator must have available within a certain pressure range sufficient fluid to compensate for leakage. Consequently, correct precharge is a very important element in hydro-pneumatic accumulator operation.

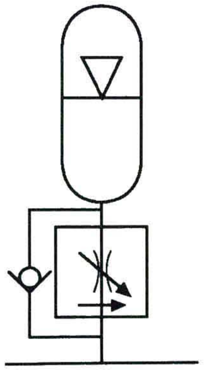
control of usable volume discharge

The usable volume of an accumulator should be discharged at a controlled rate. If an accumulator is required to maintain system pressure, this controlled rate is automatically achieved by the leakage fluid it has to replace. However, an accumu-

Adiabatic/*Isothermal* Accumulator Performance Chart (231 in.³ Accumulator)

	OPERATING PRESSURE — PSI (gage)																				
GAS PRECHARGE PRESSURE — PSI (gage)	100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100
100		86.6 *112*	113 *154*	144 *174*	158 *187*	168 *196*	175 *202*	182 *207*	186 *211*	190 *214*	192 *216*	196 *218*	198 *220*	200 *222*	202 *223*	204 *224*	206 *225*	207 *226*	209 *227*	210 *227*	211 *228*
200			57.4 *76.6*	39.7 *116*	112 *141*	126 *157*	138 *168*	147 *178*	155 *184*	161 *190*	166 *195*	170 *198*	174 *202*	178 *204*	181 *207*	184 *209*	186 *211*	188 *213*	190 *214*	192 *215*	194 *216*
300				43.4 *58.5*	71.4 *94.0*	91.1 *118*	105 *134*	118 *148*	127 *158*	136 *166*	143 *173*	148 *176*	153 *184*	157 *188*	162 *191*	165 *194*	169 *197*	172 *199*	174 *202*	177 *203*	179 *205*
400					34.2 *46.7*	58.8 *78.5*	77.3 *101*	92.0 *118*	103 *132*	114 *143*	121 *151*	128 *159*	135 *165*	141 *171*	145 *175*	149 *179*	153 *183*	157 *186*	160 *189*	163 *191*	165 *194*
500						28.5 *39.3*	50.2 *67.5*	67.0 *88.6*	80.5 *105*	91.8 *119*	102 *130*	110 *139*	117 *146*	123 *153*	128 *159*	134 *164*	138 *169*	142 *173*	146 *176*	149 *179*	152 *182*
600							24.6 *33.8*	43.6 *59.0*	58.8 *78.8*	72.1 *95.0*	83.2 *108*	92.4 *119*	101 *128*	108 *136*	114 *143*	120 *149*	126 *154*	130 *159*	132 *164*	136 *168*	140 *171*
700								21.7 *29.9*	38.6 *52.5*	53.0 *71.1*	65.1 *86.3*	75.5 *99.4*	84.6 *110*	92.6 *119*	99.5 *127*	106 *134*	112 *141*	117 *146*	121 *151*	125 *155*	129 *160*
800									19.1 *26.2*	35.0 *47.7*	48.0 *64.5*	59.3 *79.4*	69.4 *91.9*	78.1 *102*	85.8 *111*	92.5 *119*	99.8 *127*	105 *133*	110 *139*	114 *144*	119 *148*
900										17.4 *24.1*	31.6 *43.2*	43.6 *59.4*	54.7 *73.3*	63.9 *84.9*	72.5 *95.5*	80.0 *104*	86.8 *112*	92.8 *120*	98.5 *126*	104 *132*	108 *137*
1000											15.7 *21.5*	28.7 *39.5*	40.5 *55.0*	50.9 *68.2*	59.5 *79.6*	67.8 *89.7*	75.0 *98.4*	81.5 *106*	87.5 *113*	93.0 *120*	98.0 *125*
1100												14.2 *19.8*	26.8 *36.6*	37.4 *58.3*	47.2 *63.9*	55.9 *74.7*	63.4 *89.4*	70.4 *93.1*	76.9 *101*	82.6 *108*	88.0 *114*
1200													13.3 *18.6*	24.8 *34.2*	35.0 *47.7*	44.4 *60.0*	52.1 *70.2*	59.8 *79.8*	66.5 *88.2*	72.8 *95.7*	78.5 *103*
1300														12.3 *17.1*	23.1 *31.8*	32.5 *44.6*	41.0 *55.9*	49.6 *66.3*	56.4 *75.5*	63.1 *83.9*	69.1 *91.1*
1400															11.6 *15.9*	21.7 *29.9*	30.8 *42.2*	39.0 *53.0*	46.3 *62.7*	53.5 *71.9*	59.8 *80.0*
1500																10.6 *15.0*	20.2 *28.0*	28.9 *39.8*	36.9 *50.1*	44.4 *59.8*	51.9 *68.5*

Liquid Volume (in.³) Stored in Accumulator



SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

lator which is used to develop a pressurized flow can discharge its usable volume too rapidly as a downstream directional valve is shifted. For this reason, accumulators in this application are often equipped with a flow control and bypass check at their inlet-outlet port.

precharge affects shock absorber operation

Precharge of a hydro-pneumatic accumulator affects its operation as a shock absorber.

Shock generation in a hydraulic system is the result of fast pressure rises due to an external mechanical force acting on a cylinder or hydraulic motor, or the result of liquid crashing into a component as a valve is suddenly closed. An accumulator acts to reduce a shock effect by limiting pressure rise.

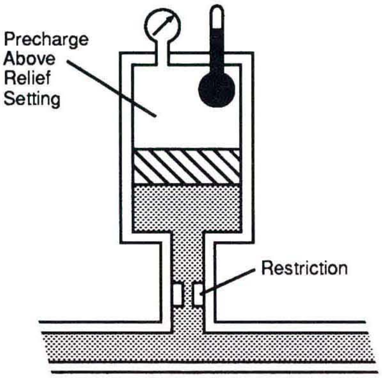
In a hydraulic system, as shock pressures develop, they attempt to displace or push the fluid to another part of the system. But, since liquid is relatively incompressible, it won't move or compress.

Without an accumulator in the line, shock pressures can climb to a high value because they have a relatively solid base on which to build. With an accumulator in the line, the base for shock pressure development becomes soft.

Above a certain system pressure as a shock pressure begins to build, an accumulator absorbs the volume of liquid the shock attempts to compress or displace. The line in which the accumulator is located becomes compressible above a certain point.

Gas precharge for a hydro-pneumatic accumulator used as a shock absorber is generally set slightly above the maximum working pressure of the line in which it is located. If the maximum pressure happens to be determined by the relief valve setting, gas precharge might be 100 psi (6.896 bar) above this.

For example, if a relief valve were set for a maximum pressure of 2000 psi (137.9 bar), accumulator precharge would be 2100 psi (144.8 bar). This means the accumulator would not function in the circuit until this pressure was reached. The accumulator later would be void of liquid until a pressure of 2100 psi (144.8 bar) was generated.



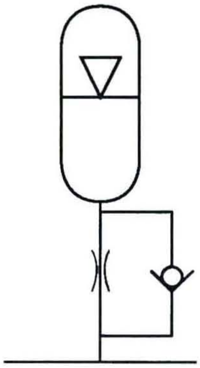
Adiabatic/Isothermal **Accumulator Performance Chart** (58 in³ Accumulator)

	OPERATING PRESSURE — PSI (gage)																				
GAS PRECHARGE PRESSURE — PSI (gage)	1000	1100	1200	1300	1400	1500	1600	1700	1800	1900	2000	2100	2200	2300	2400	2500	2600	2700	2800	2900	3000
1000		4 4 *6 1*	8 1 *11 2*	11 4 *15 5*	14 3 *19 2*	16 8 *22 4*	19 1 *25 3*	21 2 *27 8*	23 0 *29 9*	24 7 *32 0*	26 2 *33 7*	27 6 *35 4*	28 9 *36 8*	30 2 *38 2*	31 2 *39 4*	32 3 *40 5*	33 2 *41 6*	34 2 *42 6*	34 9 *43 4*	35 8 *44 3*	36 5 *45 1*
1100			4 0 *5 6*	7 6 *10 3*	10 5 *14 4*	13 3 *18 0*	15 8 *21 1*	17 9 *23 8*	19 9 *26 2*	21 7 *28 4*	23 3 *30 4*	24 9 *32 2*	26 2 *33 7*	27 5 *35 2*	28 6 *36 6*	29 8 *37 8*	30 8 *39 0*	31 8 *40 1*	32 8 *41 1*	33 7 *41 9*	34 4 *42 8*
1200				3 7 *5 2*	7 0 *9 7*	9 9 *13 5*	12 5 *16 9*	14 7 *19 8*	16 9 *22 5*	18 8 *24 9*	20 5 *27 0*	22 2 *29 0*	23 6 *30 7*	25 0 *32 4*	26 2 *33 7*	27 4 *35 1*	28 5 *36 4*	29 7 *37 7*	30 7 *38 8*	31 5 *39 7*	32 3 *40 6*
1300					3 5 *4 8*	6 5 *9 0*	9 2 *12 6*	11 6 *15 8*	13 9 *18 7*	15 9 *21 3*	17 8 *23 7*	19 5 *25 7*	21 0 *27 6*	22 5 *29 3*	23 8 *30 9*	25 1 *32 4*	26 2 *33 8*	27 3 *35 0*	28 3 *36 2*	29 2 *37 2*	30 2 *38 3*
1400						3 3 *4 5*	6 1 *8 4*	8 7 *11 9*	11 0 *14 9*	13 1 *17 7*	15 1 *20 3*	16 9 *22 6*	18 6 *24 6*	20 1 *26 4*	21 4 *28 1*	22 8 *29 7*	24 0 *31 2*	25 1 *32 5*	26 2 *33 8*	27 3 *34 9*	28 2 *36 0*
1500							3 0 *4 2*	5 7 *7 9*	8 2 *11 2*	10 4 *14 2*	12 5 *16 9*	14 4 *19 3*	16 1 *21 5*	17 7 *23 5*	19 2 *25.3*	20 5 *27 0*	21 9 *28 6*	23 0 *30 0*	24 2 *31 4*	25 2 *32 6*	26 2 *33 8*
1600								2 9 *4 0*	5 4 *7 5*	7 8 *10 7*	9 9 *13 5*	11 8 *16 0*	13 6 *18 4*	15 2 *20 5*	16 9 *22 5*	18 3 *24 3*	19 6 *26 0*	21 0 *27 6*	22 2 *28 9*	23 3 *30 3*	24 3 *31 7*
1700									2 7 *3 7*	5 1 *7 1*	7 3 *10 1*	9 3 *13 0*	11 3 *15 4*	13 0 *17 8*	14 7 *19 7*	16 2 *21 6*	17 5 *23 3*	18 9 *25 0*	20 2 *26 6*	21 4 *28 0*	22 4 *29 3*
1800										2 5 *3 5*	4 9 *6 8*	7 0 *9 7*	9 0 *12 2*	10 8 *14 8*	12 5 *16 9*	14 0 *18 9*	15 5 *20 8*	16 9 *22 6*	18 2 *24 2*	19 3 *25 6*	20 5 *27 1*
1900											2 4 *3 3*	4 7 *6 5*	6 7 *9 2*	8 6 *11 8*	10 3 *14 1*	11 9 *16 2*	13 5 *18 2*	14 8 *19 9*	16 2 *21 7*	17 5 *23 3*	18 6 *24 8*
2000												2 3 *3 2*	4 5 *6 2*	6 4 *8 8*	8 3 *11 4*	9 9 *13 5*	11 6 *15 6*	13 0 *17 5*	14 7 *19 2*	15 6 *20 9*	16 9 *22 5*
2100													2 2 *3 1*	4 2 *5 9*	6 1 *8 4*	7 8 *10 7*	9 5 *13 0*	11 0 *15 0*	12 5 *16 9*	13 8 *18 6*	15 1 *20 2*
2200														2 1 *3 0*	4 1 *5 6*	5 9 *8 2*	7 6 *10 4*	9 2 *12 5*	10 7 *14 5*	12 0 *16 3*	13 4 *18 1*
2300															2 0 *2 9*	3 9 *5 4*	5 6 *7 8*	7 3 *10 0*	8 7 *12 0*	10 3 *14 0*	11 6 *15 8*
2400																2 0 *2 7*	3 7 *5 2*	5 4 *7 5*	7 0 *9 7*	8 5 *11 7*	9 9 *13 5*
2500																	1 8 *2 6*	3 6 *5 0*	5 2 *7 3*	6 8 *9 4*	8 3 *11 4*
2600																		1 8 *2 5*	3 5 *4 8*	5 0 *6 9*	6 5 *9 0*
2700																			1 7 *2 4*	3 4 *4 7*	4 8 *6 7*

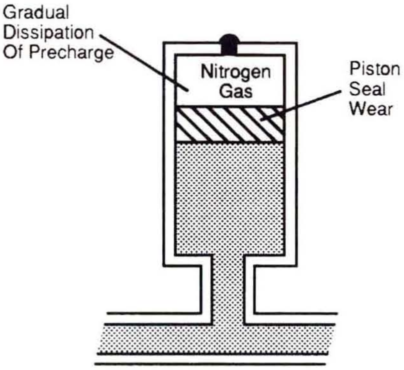
Liquid
Volume (in³)
Stored in
Accumulator

Assume that in a particular system, a shock pressure is defined as any pressure of 2100 psi (144.8 bar) and a 58 in³ (950.6 cm³) accumulator was required to absorb 4 in³ (65.6 cm³) of oil in order to dissipate the pressure. From this, it can be seen that a 58 in³ (950.6 cm³) accumulator operating adiabatically with a precharge of 2100 psi (144.8 bar) will allow pressure to climb to 2300 psi (158.6 bar) as it absorbs 4 in³ (65.6 cm³) of oil.

If the precharge were set too low, for instance 1500 psi (103.4 bar), the accumulator would hold 14.4 in³ (236 cm³) at 2100 psi (144.8 bar). In order to absorb 4 in³ (65.6 cm³) pressure would climb to over 2300 psi (158.6 bar). If the precharge were at a high value of 2500 psi (172.4 bar), pressure would climb to almost 2800 psi (193 bar) before 4 in³ (65.6 cm³) could be absorbed. Precharge of a hydro-pneumatic accumulator used as a shock absorber is quite important.



As an accumulator operates in a system as a shock absorber, it is generally required to get rid of the fluid it has accumulated in a controlled fashion. Commonly, accumulators in these applications are once again equipped with a restriction and bypass check valve. With this arrangement, an accumulator can accept its required fluid, yet any fluid accumulation can bleed off through the restriction.



losing gas precharge

Just because a hydro-pneumatic accumulator is charged once to the proper gas precharge, it does not mean it will remain charged to that pressure indefinitely. As accumulators operate, gas pressure can seep out through the gas valve. This can be due to a faulty or deteriorated seal in the valve, or an improperly seated poppet in the valve core.

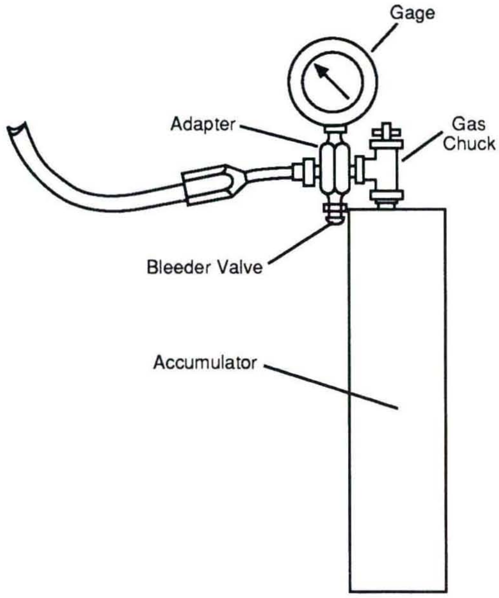
Hydro-pneumatic accumulators also lose gas precharge when discharging fluid. With bladder and diaphragm type accumulators, this usually occurs in a catastrophic manner as a result of a rupture in the synthetic rubber separator. When a piston type accumulator discharges, gas pressure can escape across the piston due to worn seals. Piston type accumulators give an indication of wear as gas precharge gradually dissipates.

checking gas precharge

Since proper gas precharge is an important consideration in hydro-pneumatic accumulator performance, precharge should be checked periodically.

A necessary piece of equipment for checking gas pressure is a precharging and gaging assembly. This assembly is primarily made up of a gas chuck, bleeder valve and pressure gage.

To check precharge, discharge the accumulator of fluid and remove the protective cap which frequently is found on the valve in accumulator top.

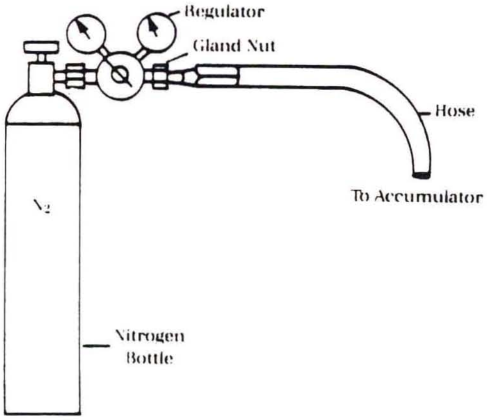


Screw the gas chuck handle of the assembly all the way out; check to see that the bleeder valve is closed. Attach the assembly to the accumulator gas valve at the gas chuck. Using a wrench, tighten gas chuck swivel nut securely onto gas valve. Turn gas chuck stem in; this depresses the core in the accumulator gas valve registering a gage pressure. This is the accumulator precharge.

If the accumulator is properly charged, back the gas chuck handle out and open the bleeder valve venting the assembly. Loosen the gas chuck swivel nut and remove the assembly. Replace the accumulator gas valve protective cover.

If it is found that the accumulator is overcharged, excess pressure can be bled off through the bleeder valve.

If it necessary to increase gas pressure, back the gas chuck handle out. Open the bleeder valve venting the assembly; then reclose the bleeder valve. At this point, the charging assembly will have to be connected to a nitrogen bottle.



With the nitrogen bottle gas valve off, connect a hose from the bottle gas valve to the charging assembly gas valve. Turn the gas chuck completely in depressing the accumulator core. Crack open the gas valve of the nitrogen bottle to slowly fill the accumulator. Shut off the valve when the gage indicates the desired pressure.

Once the gage indicates the appropriate pressure, shut off the bottle gas valve, turn out the gas chuck handle, open the gas bleeder valve. The charging hose and gaging assembly can then be removed.

In the following section, we will be concerned with unloading pump/electric motor once an accumulator is charged. We will see how this might be done with an unloading valve. But, we will find that venting a relief valve or using a differential unloading relief valve is a better means.

NOTE: Precautions must be taken to protect against accumulator overpressure during precharge and operation.

pump unloading in accumulator circuits

In a typical circuit, when an accumulator is charged and work is not required from any part of the system, pump/electric motor flow is unloaded to tank with the least possible pressure.

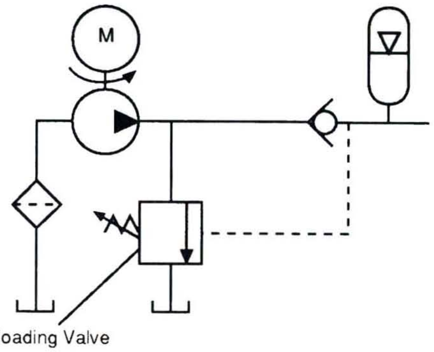
In the circuit illustrated, an unloading valve is used to dump flow back to tank once an accumulator is charged to the unloading valve setting. Frequently, unloading in this manner is only good for a few seconds at the most. With any leakage in the system downstream from the check valve, pressure in the accumulator will drop as fluid is discharged to compensate. This results in the valve gradually closing. The opening to tank through the valve progressively becomes more and more restrictive until accumulator pressure drops below the valve cracking pressure. Up to this point, pump/electric motor has to develop more and more power as the valve closes.

Once the valve closes, pump/electric motor must therefore generate power to recharge the accumulator to the unloading valve setting. This means the pump/electric motor works when it is supposed to be unloaded.

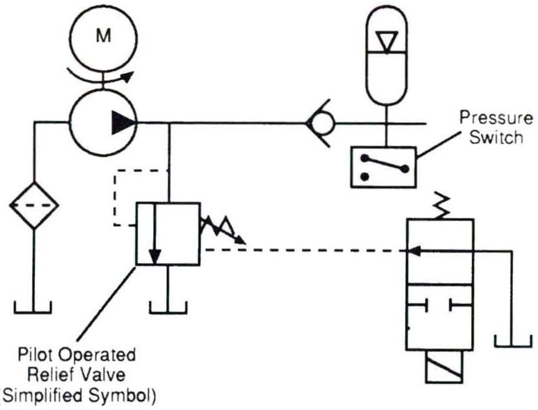
To keep a pump/electric motor fully unloaded until it is required to recharge an accumulator, an electric pressure switch can be used.

In the circuit illustrated, a pressure switch senses accumulator pressure sending and cutting-out electrical signals at various pressure levels. The electrical signals are transmitted to a normally non-passing, solenoid operated 2-way valve which vents a pilot operated relief valve. When the accumulator is charged to the pressure switch setting an electrical signal to the 2-way valve solenoid venting the relief valve and unloading pump/electric motor is sent. The setting of the pressure switch determines the pressure range within which a pump/electric motor works. Using a pressure switch to vent a relief valve, results in a pump/electric motor being fully unloaded when system conditions dictate.

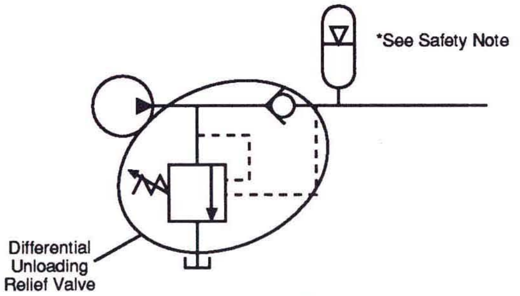
The wiring and additional piping required in venting a relief valve can be eliminated by using a differential unloading relief valve. In the following section, this valve is described. We find that its operation is dependent on a differential piston.



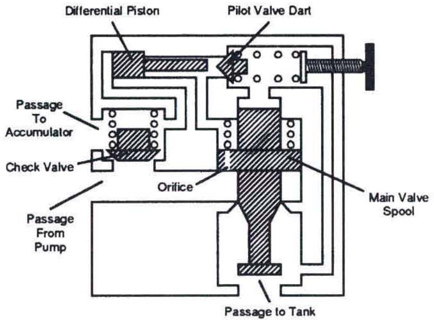
SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

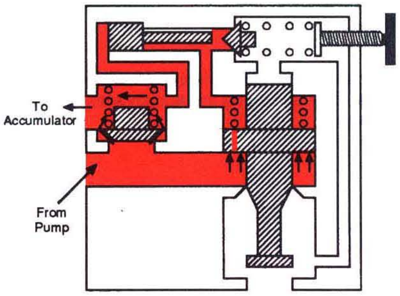


SAFETY NOTE: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down.

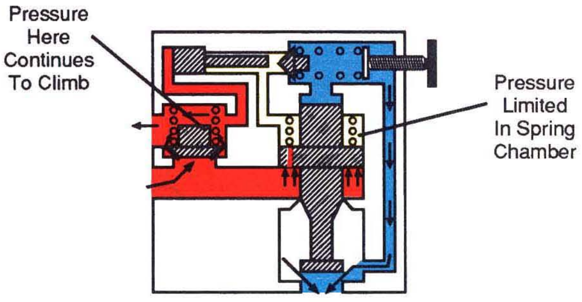


*Safety Note: In any accumulator circuit, a means should be available of automatically unloading the accumulator when the machine is shut down. See Page 8-6.

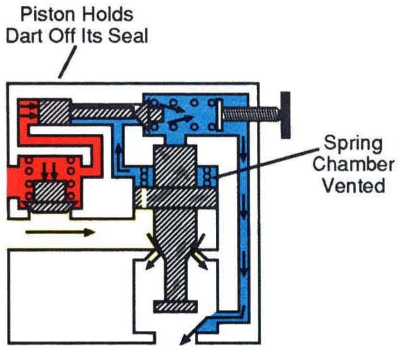




Accumulator Being Charged



Pilot Dart Unseated



Relief Valve Vented
By Differential Piston

differential unloading relief valve

Instead of using a pressure switch and solenoid valve to vent a relief valve while an accumulator is charged, one hydraulic component can be used a differential unloading relief valve. A differential unloading relief valve is specifically designed for use with accumulators. As its name implies, the valve unloads a pump/electric motor over a differential pressure range.

what a differential unloading relief valve consists of

A differential unloading relief valve consists of a pilot operated relief valve, check valve, and differential piston in one valve body. The valve body includes pump, tank and accumulator passages.

how a differential unloading relief valve works

In a differential unloading relief valve, check valve and pilot operated relief valve operate in their usual manner. The accumulator charges through the check valve.

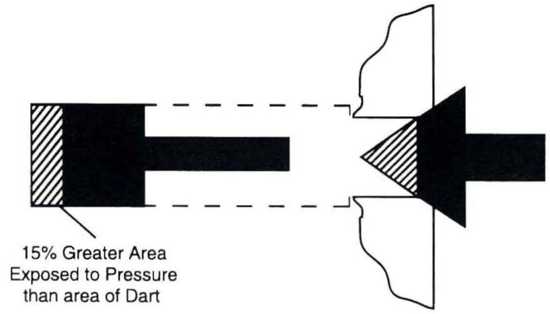
The differential piston is free to move within a bore opposite the pilot valve dart. Areas exposed to pressure at each end of the piston are equal. During the time the accumulator is being charged, pressure at each end of the piston is relatively equal. (Consider the pressure differential across the check valve to be negligible.) As a result, the piston does not move.

When a large enough pressure is present at the pilot valve dart, the dart is pushed off its seat. We have seen earlier that this action limits the pressure in the spring chamber of the main valve. With pressure limited in the spring chamber and also at one end of the differential piston, the piston is moved toward the pilot dart forcing the dart completely off its seat. This in effect releases the main spool spring chamber of pilot pressure, venting the relief valve and unloading the pump/electric motor. At the same time the check valve closes so that accumulator flow cannot discharge through the relief valve.

At this point, the accumulator's maximum pressure has been achieved and pump/electric motor is unloaded. The differential piston is the key to unloading the pump until the accumulator discharges to a lower pressure.

The differential piston has a 15% greater area exposed to pressure than the area of the pilot dart. Since force = pressure x area, the piston holds the pilot dart off its seat with a 15% greater force than the force which unseated the dart. This means that in order to reseat the pilot dart, the spring must acquire a 15% greater force from somewhere; or, it must wait until the pressure in the system falls off 15%. Of course, the dart reseats when system pressure falls off 15%.

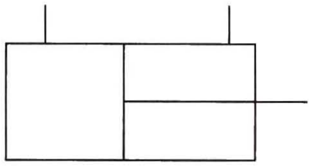
In this way, a differential unloading relief valve allows a pump/electric motor to unload when an accumulator is charged and to remain unloaded until recharging is required.



A limitation of a differential unloading relief valve is that the valve's secondary pressure is fixed because the difference in areas between piston and pilot dart is fixed. This is frequently 15% and in some cases 30% of the pilot valve setting. For example, a differential unloading relief valve with a 15% differential will unload between 1000 psi (69 bar) and 850 psi (59 bar) with its pilot valve adjusted for 1000 psi (69 bar). Or, with its pilot valve adjusted for 2000 psi (138 bar), it will unload between 2000 psi (138 bar) and 1700 psi (113 bar).

hydraulic cylinders

In all applications, hydraulic working energy must be converted to mechanical energy before any useful work can be done. Hydraulic cylinders convert hydraulic working energy into straight-line mechanical energy.

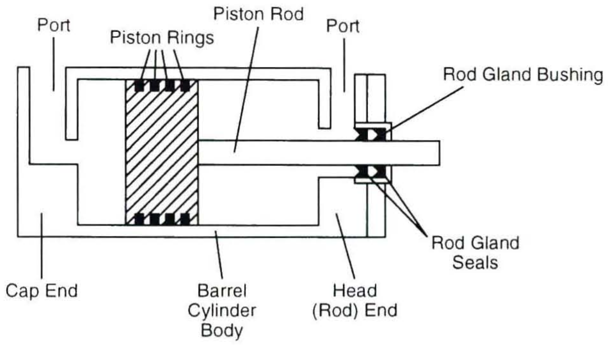


Cylinder Symbol

We have seen previously basic cylinder operation. Now we shall deal with cylinder construction.

what a cylinder consists of

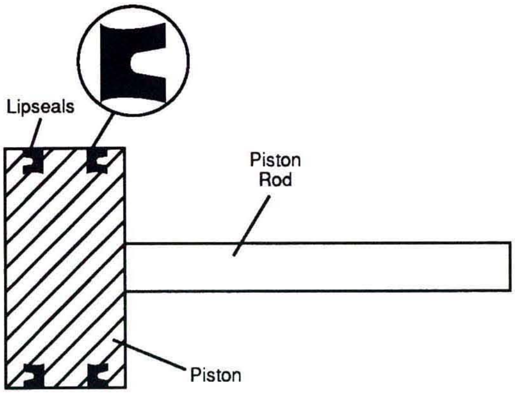
A hydraulic cylinder consists of a cylinder body, a movable piston and a piston rod attached to the piston. End caps are attached to the cylinder body barrel by threads, keeper rings, tie rods or a weld. (Industrial cylinders use tie rods.)



As the cylinder rod moves in and out, it is guided and supported by a removable bushing called a rod gland or rod bearing.

The end through which the rod protrudes is called the "head" or "rod head." The opposite side without the rod is termed the "cap." Inlet and outlet ports are located at the head and cap ends.

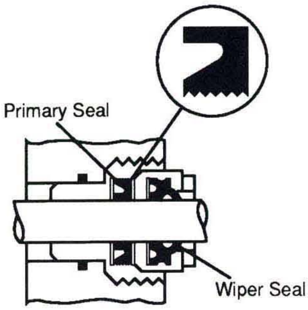
seals

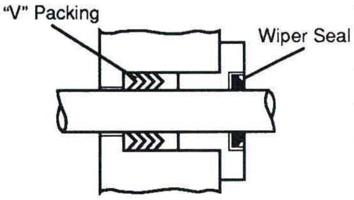


For proper operation, a positive seal must exist across a cylinder's piston as well as at the rod gland. Cylinder pistons are typically sealed by using lipseals, cast iron piston rings, or a single bidirectional sealing element.

Piston rings are durable, but may exhibit some leakages under normal conditions. Lipseals and bidirectional seals offer a more positive seal, but may be less durable in some applications.

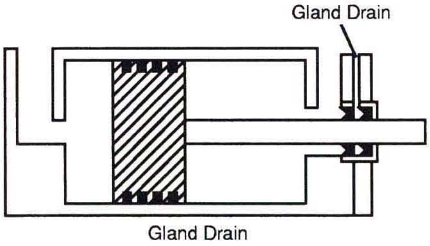
Rod gland seals come in several varieties. Some cylinders are equipped with a "V"-shaped or cup-shaped primary seal made of leather, polyurethane, nitrile or viton; and a wiper seal which prevents foreign materials from being drawn into the cylinder.





The seal material or compound should be verified to be compatible with the fluid media and operating conditions. One effective type of rod gland seal consists of a primary seal with serrated edges along its inside surface. The edges contact the rod continuously and scrape it clean of fluid. A second wiper seal, catches any fluid which may get by the primary seal and also wipes the rod of foreign material when the rod retracts.

gland drain



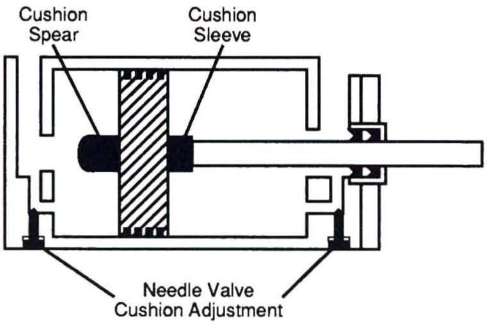
During the operation of the latter rod gland seal described above, any fluid which collects in the chamber between the primary and wiper seals is drawn back into the cylinder during the retraction stroke. But, on extremely long stroke cylinders (10 ft./3.05 M or more), there is a possibility that too much fluid may collect in the chamber causing a leaking rod gland. In these applications, or in any application where excessive amounts of fluid may collect in the gland, the rod gland should be externally drained.

hydraulic shock

When hydraulic working energy moving a cylinder's piston runs into a dead end (as at the end of a cylinder's stroke), the liquid inertia is changed into a concussion known as "hydraulic shock." If a substantial amount of working energy is stopped, the shock may damage the cylinder.

cushions

To protect against excessive shock a cylinder can be equipped with cushions. Cushions slow down a cylinder's piston movement just before reaching the end of its stroke. Cushions can be applied at either or both ends of a cylinder.



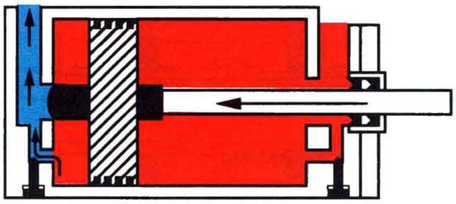
what a cushion consists of

A cushion consists of a needle valve flow control, a cushion spear attached to the cap end side of the piston and a cushion sleeve over the rod. These devices act like plugs at their own respective ends.

how a cushion works

As a cylinder piston approaches the end of its travel, the plug blocks the normal exit for a liquid and forces it to pass through the needle valve. At this point, some flow goes over the relief valve at the relief valve setting. The remaining liquid ahead of the cylinder piston is bled off through the needle valve and slows down the piston. The adjustment of the needle valve will determine the rate of deceleration.

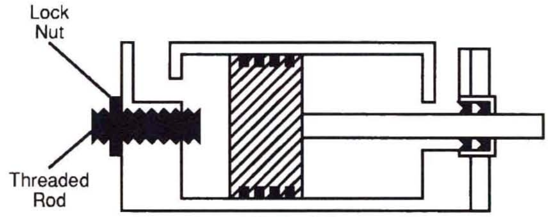
In the reverse direction, flow bypasses the needle valve by means of a check valve within the cylinder (Not shown).



Cushioning

stroke adjusters

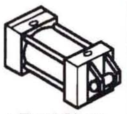
Sometimes the stroke length of a cylinder must be externally controlled. Periodic adjustment is accomplished with a threaded rod which can be screwed in or out of the cylinder cap. Each style of stroke adjuster must be examined for its effect on stopping, impact, shock, etc.



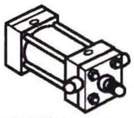
Stroke Adjuster

cylinder mounting styles

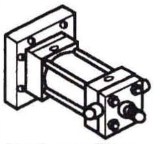
Cylinders can be mounted in a variety of ways, among which are flange, trunnion, side lug and side tapped, clevis, tie rod, and bolt mounting. Centerline mount would be a very good choice to minimize leakage due to cylinder movement.



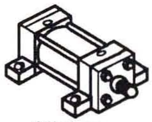
Cap Fixed Clevis



Head Trunnion



Cap Square Flange



Side Lugs

mechanical motions

Cylinders convert hydraulic working energy into straight line, or linear, mechanical motion. But, depending on the way in which they are attached to mechanical linkages, cylinders provide many different mechanical motions.

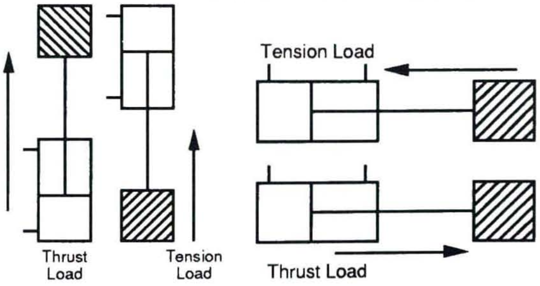
Straight Line Motion In Two Directions

Horizontal Parallel Motion

Practically Continous Rotary Motion

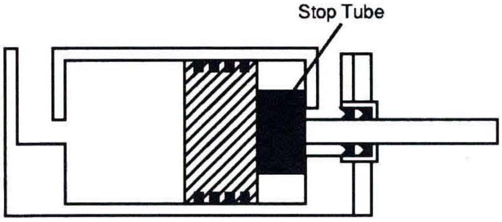
types of cylinder loads

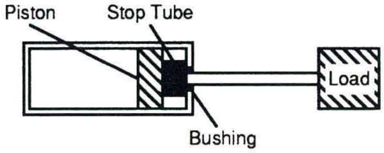
Cylinders can be used in an unlimited number of applications to move various types of loads. But, in general, a load which is pushed by a cylinder rod

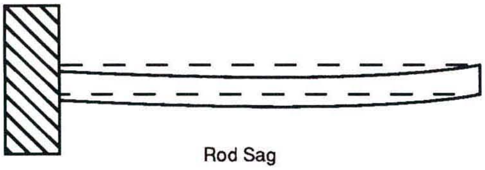


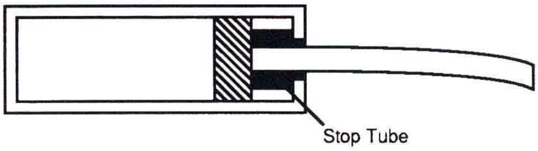
is termed a thrust load. A load which is pulled by a cylinder rod is called a tension load.

stop tube






Rod Sag



A stop tube is a solid, metal collar which fits over the piston rod. A stop tube keeps the piston and rod gland bushing separated when a long-stroke cylinder is fully extended.

Since it is a bearing, a rod gland bushing is designed to take some loading when supporting the rod as it moves in and out of the cylinder.

Along with being a bearing, a rod gland bushing is also a fulcrum for the piston rod. If the load attached to the piston rod of a long-stroke cylinder is not rigidly guided, then at full extension, the rod will tend to teeter-totter or jackknife at the bushing causing excessive loading. A stop tube in effect protects the rod gland bushing by reducing the bearing load at full extension between both piston and bushing.

Believe it or not, but the heavy, steel rods of long-stroke cylinders sag just because of their weight. A 5/8" (1.59 cm) diameter piston rod weighs 1 lb. per foot (.4536 kg per .3040 m) and will sag over 1 in. (2.54 cm) at the center of a 10 ft. (3.05 m) span.

On long-stroke, horizontally mounted cylinders, undesirable bearing loads are generated at the rod gland bushing because of the rod weight when the rod is fully extended. On these cylinders, a stop tube is used to separate bushing and piston when the rod is extended. This reduces the load on the rod gland bushing.

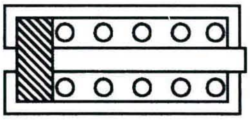
Most cylinders do not need stop tubes. To determine when a stop tube is required, or what the length of a stop tube should be, consult the cylinder manufacturer's catalog.

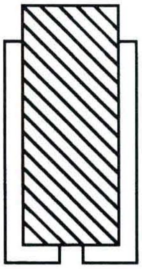
hydraulic cylinders
cylinder types

Hydraulic cylinders can be of various types. Some common cylinders are described below and in later lesson material are seen as they are applied in a circuit.

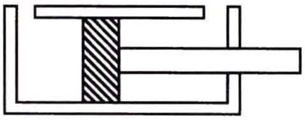
Single rod cylinder: cylinder with a piston rod extending from one end.

Double rod cylinder: cylinder with a single piston and a piston rod extending from both ends.


Single Acting
Spring Return Cylinder

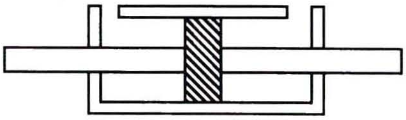

Ram

Double acting cylinder: cylinder in which fluid pressure is applied alternately to both sides of a cylinder piston to effect extension and retraction of a piston rod.

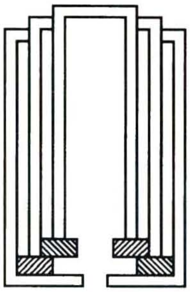


Double Acting Cylinder

Double Rod Cylinder

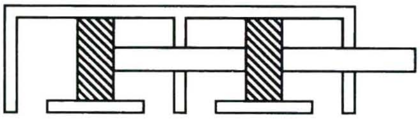


Telescoping cylinder: cylinder with nested multiple tubular rod segments which provide a long working stroke in a short retracted envelope.



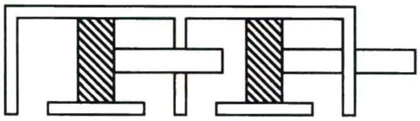
Telescoping Cylinder

Tandem cylinder: cylinder consisting of two or more cylinder bodies mounted in line with their piston rods connected to form a common piston rod; rod seals are installed between cylinder bodies to permit double acting operation of each.



Tandem Cylinder

Duplex cylinder: cylinder consisting of at least two cylinder bodies to permit double acting operation of each.

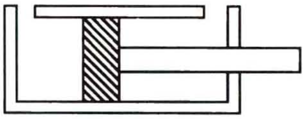


Duplex Cylinder

With single rod, double rod, double acting, tandem and duplex cylinders defined, let's now look at their operation in a system. We begin with a double acting, single rod cylinder.

double acting single rod cylinder operation

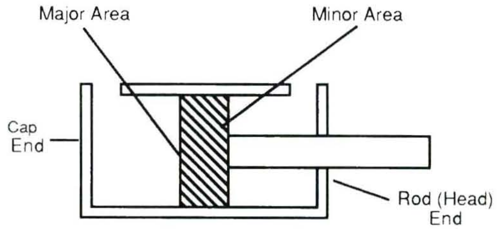
A very common cylinder in industrial hydraulic applications is a double acting, single rod cylinder. Just as other hydraulic cylinders, this type cylinder is concerned with accepting gpm and psi and converting it into mechanical force and piston rod motion.



Double Acting Cylinder

In the following material, it will be illustrated how rod speed of this type cylinder is determined by gpm, how mechanical force is affected by psi, and how both of these elements are affected by major and minor piston areas.

piston and effective piston area



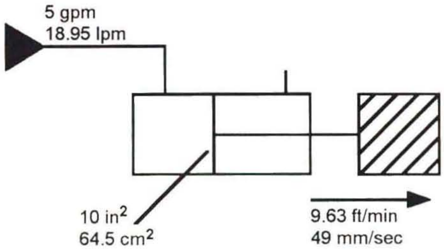
Piston and effective areas generally refer to a double acting, single rod cylinder. Piston major area indicates the piston area exposed to pressure at the cylinder cap side. Effective minor area (also called the annulus area) refers to the piston area exposed to pressure at the cylinder rod side. Since the rod covers a portion of the piston at this point, effective area is always less than piston area.

illustrating rod speed while extending

Rod speed of a cylinder is determined by how quickly the volume behind a piston can be filled with liquid. Rod speed is many times measured in of feet per minute or (ft/min) or meters per minute (m/min). This is described by the expression:

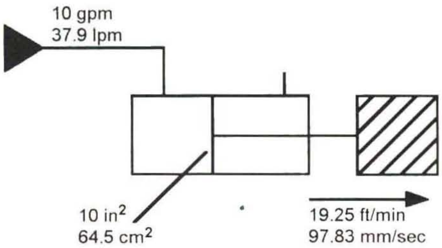
$$\underset{\text{(ft/min)}}{\text{Rod speed}} = \frac{\text{gpm x 19.25}}{\text{Piston area (in}^2\text{)}}$$

$$^*\text{(m/sec)} = \frac{\text{l/minx .167}}{\text{Piston area (cm}^2\text{)}}$$



Assume that a cylinder with a 10 in² (64.5 cm²) piston area receives a flow rate of 5 gpm (18.95 l/min). Using the above expression, we find that the rod speed is 9.63 ft/min (49 mm/sec).

$$\begin{array}{lccc} \begin{array}{l}\text{Rod speed}\\ \text{(ft/min)}\\ ^*\text{(m/sec)}\end{array} & = & \dfrac{\text{gpm x 19.25}}{\text{Piston area (in}^2\text{)}} & \dfrac{\text{l/minx .167}}{\text{(cm}^2\text{)}} \\ & = & \dfrac{5 \text{ x } 19.25}{10} & \dfrac{18.95 \text{ l/minx .167}}{64.5} \\ & = & \dfrac{96.25}{10} & \dfrac{3.165}{64.5} \\ & = & 9.63 \text{ ft/min} & 49\text{mm/sec} \end{array}$$



This means that 5 gpm (18.95 l/min) fills the volume behind the piston quickly enough to move piston and piston rod a distance of 9.63 ft. (49 mm) in one minute. If the same cylinder received twice the flow rate, (10 gpm/37.9 l/min), rod speed would be doubled (19.25 ft/min or 97.8 mm/sec).

*** If calculating m/sec and the solution is less than .1 m/sec, the solution should be expressed in mm/sec.**

The more flow a cylinder receives, the more quickly it will fill with liquid and the faster it will extend.

illustrating discharge flow while extending

Flow entering the cap end of a double-acting, single rod cylinder determines the rate at which a cylinder piston rod will extend. While this is occurring, of course, flow also discharges from the rod side. Discharge flow is an important concern in many systems and can be calculated by using the same basic formula.

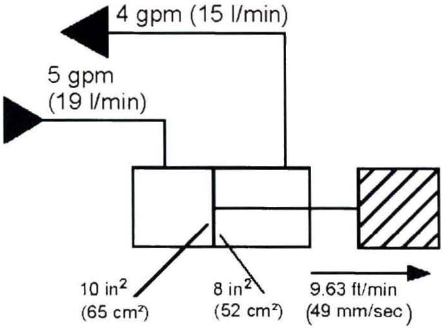
With the same cylinder as in the previous examples, assume that the rod has a cross sectional area of 2 in² (12.9 cm²) and that piston and piston rod extension speed is 9.63 ft/min (49 mm/sec) as calculated above. Using the same expression for rod speed, we find that rod speed and the minor area of the cylinder are known, but gpm (l/min) is not.

$$\text{Rod speed (ft/min)} = \frac{\text{gpm} \times 19.25}{\text{Minor piston area (in}^2\text{)}} \qquad \frac{\text{l/min} \times .167}{\text{cm}^2}$$

$$9.63 \text{ ft/min} = \frac{\text{gpm} \times 19.25}{8 \text{ in}^2}$$

$$49\text{mm/sec} = \frac{\text{l/min} \times .167}{52 \text{ cm}^2}$$

Rearranging the formula, we then solve for gpm (l/min) which is the discharge flow rate at the rod side.



$$\text{gpm (l/min)} = \frac{9.63 \text{ ft/min} \times 8 \text{ in}^2}{19.25} \qquad \frac{49\text{mm/sec} \times 52 \text{ cm}^2}{.167}$$

$$= \frac{77.04}{19.25} \qquad \frac{2528.4}{.167}$$

$$= 4.0 \text{ gal/min} \qquad 15 \text{ l/min}$$

This indicates as 5 gpm (19 l/min) enters the cap end of the cylinder, 4 gpm (15 l/min) discharges back to tank from the cylinder rod side.

While extending, discharge from a single rod cylinder is always less than the flow rate entering the cap end.

illustrating rod speed while retracting

During retraction, when full pump flow is directed to the rod side of a single rod cylinder, a piston rod will retract faster than it extended. This can be calculated by using the expression for rod speed. Except in this instance, instead of using the major piston area, the minor area is used.

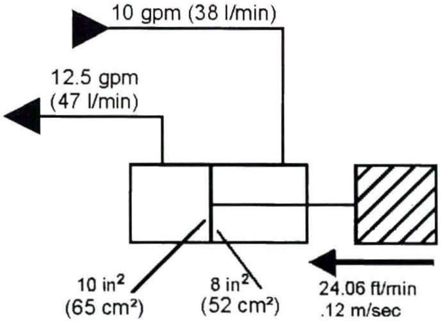
Assume that 10 gpm (38 l/min) enters the rod side of our cylinder during retraction:

$$\text{Retraction Rod speed (ft/min) (m/sec)} = \frac{\text{gpm} \times 19.25}{\text{Minor piston area (in}^2\text{)}} \qquad \frac{\text{l/min} \times .167}{\text{(cm}^2\text{)}}$$

$$= \frac{10 \text{ gpm} \times 19.25}{8 \text{ in}^2} \qquad \frac{38 \times .167}{52 \text{ cm}^2}$$

$$= \frac{192.5}{8} \qquad \frac{6.35}{52}$$

$$= 24.06 \text{ ft/min} \qquad .12 \text{ m/sec}$$



With flow to a cylinder remaining constant, double acting, single rod cylinders always retract faster than they extend.

illustrating discharge flow while retracting

During retraction, when full pump flow is directed to the rod side of a single rod cylinder, discharge flow will be greater than incoming flow. This can be calculated by using the same expression for gpm (l/min) used previously. In this instance, however, the major piston area is used.

Assume once again, that 10 gpm (38 l/min) enters the rod side of our cylinder during retraction.

$$\text{gpm} = \frac{\overset{\text{Rod}}{\text{speed (ft/min)}} \times \overset{\text{Major}}{\text{piston area (in}^2\text{)}}}{19.25}$$

$$= \frac{24.06 \text{ ft/min} \times 10 \text{ in}^2}{19.25}$$

$$= \frac{24.06}{19.25}$$

$$= 12.5 \text{ gpm discharge}$$

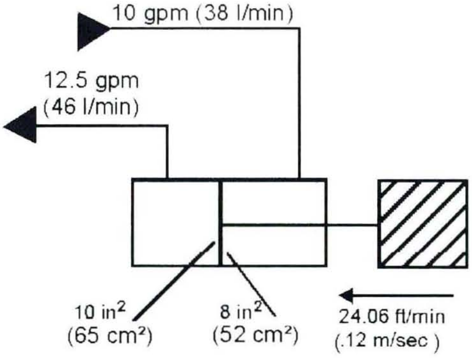
$$\text{l/min} = \frac{\overset{\text{Rod}}{\text{speed (m/sec)}} \times \overset{\text{Major}}{\text{piston area (cm}^2\text{)}}}{.167}$$

$$= \frac{.12 \text{ m/sec} \times 65 \text{ cm}^2}{.167}$$

$$= \frac{7.74}{.167}$$

$$= 46 \text{ l/min discharge}$$

Anytime a double acting, single rod cylinder is retracting, more flow is discharging from the cap end than entering the rod end. Therefore, pump flow is not necessarily the maximum flow rate in a system.



When the return side of a system is designed, consideration is given to the discharge flow from retracting single rod cylinders. This is one reason why piping, valving and filters at the return side are sized larger than their counterparts in the main system.

If an occasion arises where return side parts must be disassembled, ensure that a smaller sized component is not substituted for an original part upon reassembly.

cylinder force while extending

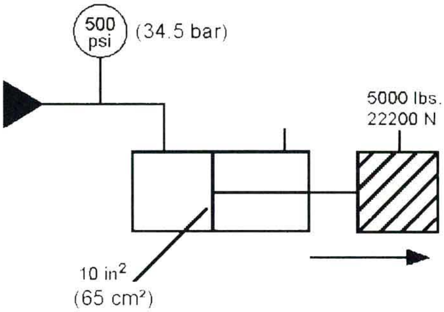
F = P x A

While extending, the mechanical force developed by a cylinder is the result of hydraulic pressure acting on the cap end of the cylinder piston. This is expressed by the formula:

$$\begin{aligned}\text{Force (lbs.)} &= \text{Pressure (psi)} \times \text{Area (in}^2)\\ \text{(Newtons)} &= \text{Pressure (bar)} \times \text{Area (m}^2)\end{aligned}$$

For example, if a load offers a resistance to move of 5000 lbs. and the area of the cylinder piston is 10 in², then a hydraulic pressure of 500 psi is required to equal the load as calculated by the formula:

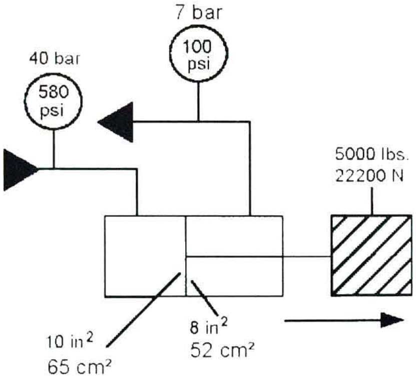
$$\begin{aligned}\text{Pressure (psi)}\ \text{(bar)} &= \frac{\text{Force (lbs.)}}{\text{Area (in}^2)} \quad \frac{\text{(Newtons)}}{(\text{cm}^2)}\\ &= \frac{5{,}000\ \text{lbs.}}{10\ \text{in}^2} \quad \frac{(22200\ \text{N})}{(65\text{cm}^2)}\\ &= 500\ \text{psi} \quad (34.5\ \text{bar})\end{aligned}$$



However, the assumption was made that pressure was zero on the other side of the piston. Even though the piston minor area is drained to tank while extending, tank line pressure, or back pressure, can be as high as 100 psi (7 bar) in some systems.

In our example, assume a back pressure of 100 psi (7 bar) during extension. This generates a force on the effective area of the piston to retract piston and piston rod. This force, together with the resistance offered by the load, must be overcome before the cylinder will extend at full speed. If the effective area of the piston were 8 in² (52 cm²) and the load offered 5000 lbs. (22200 N), then the force which must be overcome is 5800 lbs. (25752 N), which is made up of 5000 lbs. (22200 N) of the load and 800 lbs. (3552 N) (100 psi x 8 in²) generated by the back pressure. Pressure required at the cylinder piston is, therefore, 580 psi (40 bar).

$$\begin{aligned}\text{Pressure (psi)}\ \text{(bar)} &= \frac{\text{Force (lbs.)}}{\text{Area (in}^2)} \quad \frac{\text{(N)}}{(\text{cm}^2)}\\ &= \frac{5800\ \text{lbs.}}{10\ \text{in}^2} \quad \frac{(25752\text{N})}{(65\text{cm}^2)}\\ &= 580\ \text{psi} \quad (40\ \text{bar})\end{aligned}$$



Anytime a load is to be extended by a cylinder through a distance at a certain speed, pressure required at a cylinder piston is used to equal load resistances as well as liquid resistances flowing back to tank.

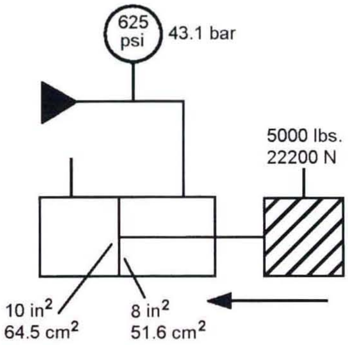
cylinder force while retracting

While retracting, the mechanical force developed by a cylinder is the result of hydraulic pressure acting on the effective area. This is also expressed by the formula:

$$\text{Force (lbs.)} = \text{Pressure (psi)} \times \text{Area (in}^2\text{)}$$
$$\text{Force (N)} = \text{Pressure (bar)} \times \text{Area (cm}^2\text{)}$$

In this instance, the effective area of the piston is used.

For example, if a load offers a resistance to move of 5000 lbs. (22200 N) and the effective area of the cylinder piston is 8 in^2 (51.60 cm^2), then a hydraulic pressure of 625 psi (43.1 bar) is required to equal the load as calculated by the formula:

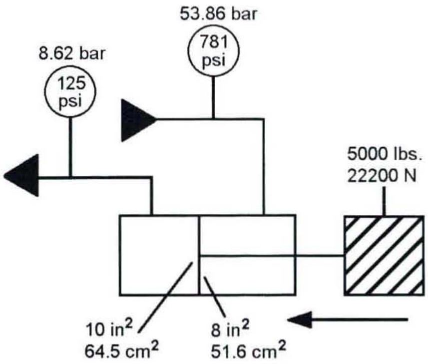


$$\begin{aligned} \begin{matrix}\text{Pressure (psi)}\\ \text{(bar)}\end{matrix} &= \frac{\text{Force (lbs.)}}{\text{Area (in}^2\text{)}} \quad \frac{\text{(N)}}{\text{(cm}^2\text{)}} \\ &= \frac{\text{5000 lbs.}}{\text{8 in}^2} \quad \frac{\text{(22200 N)}}{\text{(51.6 cm}^2\text{)}} \\ &= \text{625 psi} \quad \text{(43.1 bar)} \end{aligned}$$

This also assumes that backpressure is not present at the cap end. But, backpressure is even more pronounced while retracting than extending.

Assume once again that the load offers a resistance to move of 5000 lbs. (22200 N) and that backpressure is 125 psi (8.62 bar). This is a higher backpressure than seen during extension because discharge flow during retraction is usually more than during extension.

The total resistance which must be overcome is 5000 lbs. (22200 N) of the load and 1250 lbs. (5550 N) as a result of 125 psi (8.62 bar) acting on 10 in^2 (64.5 cm^2). This is a total of 6250 lbs. (27750 N) which is overcome by 781 psi (53.86 bar) acting on the 8 in^2 (51.6 cm^2) effective area as calculated by the formula:



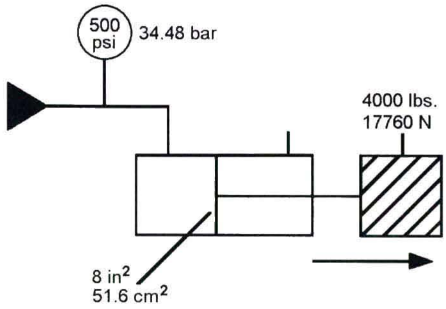
$$\begin{aligned} \begin{matrix}\text{Pressure (psi)}\\ \text{(bar)}\end{matrix} &= \frac{\text{Force (lbs.)}}{\text{Area (in}^2\text{)}} \quad \frac{\text{(N)}}{\text{(cm}^2\text{)}} \\ &= \frac{\text{6250 lbs.}}{\text{8 in}^2} \quad \begin{matrix}\text{(27750 N)}\\ \text{(51.6 cm}^2\text{)}\end{matrix} \\ &= \text{781 psi} \quad \text{(53.86 bar)} \end{aligned}$$

While retracting a single rod cylinder, back pressure at the cap end will usually be higher than while extending.

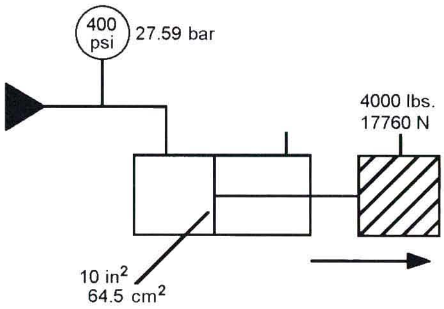
We have been shown how cylinder rod speed is affected by gpm (lpm) and piston area, and how cylinder force is affected by psi (bar) and piston area. In the next section, we see what is done when one or more of these elements is restricted.

affecting cylinder force

As illustrated, the force generated by a cylinder is a function of fluid pressure acting on the cylinder piston area. If a particular cylinder is required to generate more output force than it is currently developing, frequently fluid pressure is increased to the appropriate level.



In some cases, pressure generated by pump/electric motor is limited to a low value. In order to increase output force in this instance, cylinder size can be increased. This allows pump/electric motor to develop less pressure to achieve the same mechanical output force.

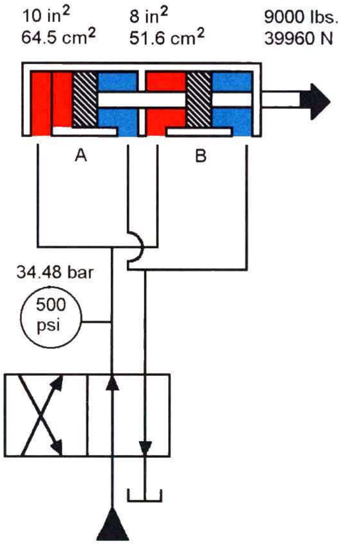


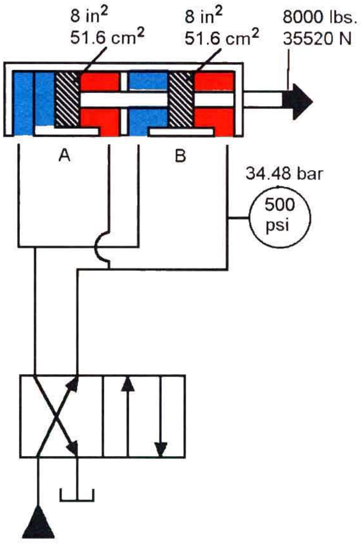
In some situations, both system pressure and cylinder bore size are not allowed to be increased. Machine dimensions are so fixed that a certain bore size cylinder can only be used. In these applications, a tandem cylinder can be employed.

tandem cylinder circuit

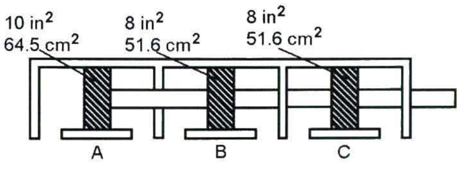
A tandem cylinder consists of two or more cylinder bodies mounted in line. Piston rods are connected to form a common piston rod, and rod seals are installed between cylinders to permit double acting operation of each. A tandem cylinder gives increased output force when cylinder bore size is limited, but not its overall length.

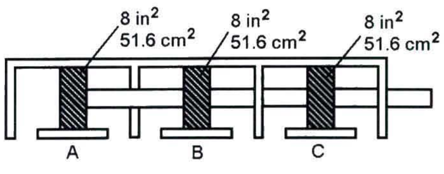
Assume that a cylinder with a piston area of 10 in^2 (64.5 cm^2) is the largest size which can be physically mounted on a machine. Yet, the maximum working pressure available to equal load resistances is only 500 psi (34.48 bar). This must move a load which offers a resistance of 9000 lbs. (39960 N).





In this application, a tandem cylinder is used which is made up of two cylinders with a major area of 10 in^2 (64.5 cm^2) and a minor area of 8 in^2 (51.6 cm^2). However, since one cylinder has a rod connected to both sides, 10 in^2 (64.5 cm^2) of piston A plus 8 in^2 (51.6 cm^2) of piston B are exposed to fluid





pressure during extension. During retraction, 16 in^2 (103.2 cm^2), the total minor areas of the pistons, are exposed to fluid pressure.

During extension with 500 psi (34.48 bar) acting on 10 in^2 (64.45 cm^2) of Piston A and 8 in^2 of Piston B, 9000 lbs. (39960 N) of force is developed. In the retraction stroke, 500 psi (34.48 bar) acts on the total 16 in^2 (103.2 cm^2) minor area resulting in a force of 8000 lbs. (35520 N). With 500 psi (34.48 bar), this cylinder can develop a maximum force of 9000 lbs. (39960 N) extending and 8000 lbs. (35520 N) retracting.

If another cylinder section were added to the group, the tandem set would have 26 in^2 (167.7 cm^2) of area exposed during extension and 24 in^2 (154.8 cm^2) during retraction. With a maximum working pressure capability of 500 psi (34.48 bar) in both directions, 13000 lbs. (57720 N) could be developed during extension and 12000 lbs. (53280 N) during retraction.

By using cylinder pistons in tandem, increased output force can be achieved when cylinder bore size and maximum working pressure are limited.

achieving mechanical positions

Besides transforming hydraulic power into linear mechanical power, cylinders are also used to achieve a mechanical position.

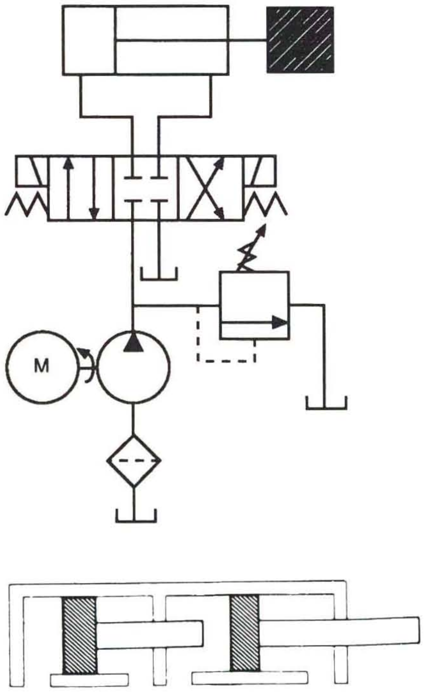
After a cylinder has moved a load through its stroke, a definite mechanical position has been achieved. As long as cylinder mounting and piston rod are not altered, a load can be depended on to reach this position time after time.

In some cases, it is desirable to have a load stop in an intermediate position. This can be accomplished by centering a closed center directional valve as the load reaches a definite point along the cylinder stroke.

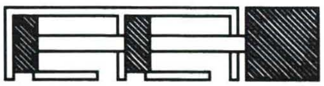
In some applications, a simple method of providing an accurate repeatable position is to use a duplex cylinder.

duplex cylinder circuit

Whereas a tandem cylinder consists of two or more cylinder bodies of equal stroke, a duplex cylinder is made up of at least two cylinder bodies with different strokes. Cylinder bodies are joined together, but piston rods are not connected. Rod seals are installed between cylinder bodies to permit double acting operation of each.



Duplex Cylinder



Duplex Cylinder Position 1 — Start

A duplex cylinder gives three possible mechanical positions for a load. One position is with both piston rods retracted. Another position is with the shorter rod extended its length. Extending the short piston rod in turn pushes the rod of the long stroke body and the load an equal distance. A third position is reached when the long piston rod is stroked the remainder of its length.

In the illustration, a duplex cylinder consists of 6 in. (15.2 cm) and 10 in. (25.4 cm) stroke cylinder bodies. Position 1 for the load is with both rods fully retracted. As a directional valve connected to the 6 in. (15.2 cm) stroke cylinder body is shifted, the piston rod extends its length moving the 10 in. (25.4 cm) stroke piston rod and the load 6 in. (15.2 cm) to position 2.

After the work operation has been performed at position 2, a directional valve connected to the 10 in. (25.4 cm) stroke cylinder body is shifted. This action extends the rod its remaining 4 in. (10.16 cm) achieving position 3 at 10 in. (25.4 cm).

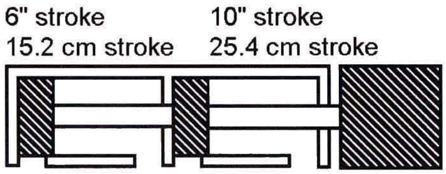
When the directional valves are shifted to retract, both rods return to the start position.

The three mechanical positions achieved with this particular duplex cylinder are start, 6 in. (15.2 cm) and 10 in. (25.4 cm).

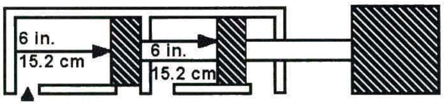
A duplex cylinder can also be arranged so that cylinder bodies are connected at their cap ends. In this case, one piston rod is attached to a machine member which remains stationary. With this set up, four positions can be achieved — three extending and one retracting.

A duplex cylinder with cylinder bodies connected at their cap ends is illustrated. One piston rod is mechanically attached to a machine member which does not move. One cylinder body has a stroke of 6 in. (15.2 cm); the other has an 8 in. (20.32 cm) stroke.

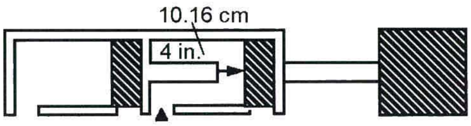
With both piston rods fully retracted, position 1 is achieved. As a directional valve connected to the 6 in. (15.2 cm) stroke body is shifted, fluid pressure acts on the major area of its piston; but, the piston cannot move. The same pressure acting on the piston also acts on the cap end of the cylinder body. Since the cylinder body is not attached to anything, both cylinder bodies, along with the load, move out a distance of 6 in. (15.2 cm). This is position 2.



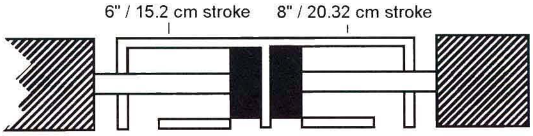
Duplex cylinder position 1 - Start



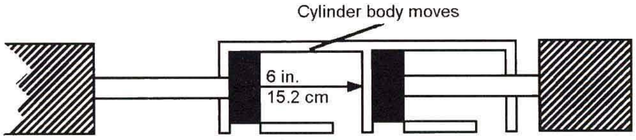
Duplex cylinder position 2 - 6 in. / 15.2 cm



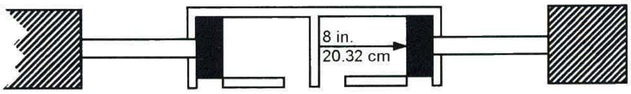
Duplex cylinder position 3 - 10 in. / 10.16 cm



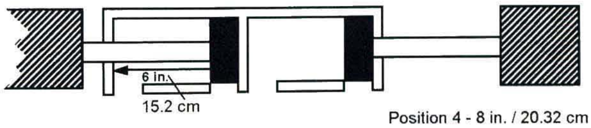
Position 1 - Start



Position 2 - 6 in. / 15.2 cm



Position 3 - 14 in. / 35.56 cm



Position 4 - 8 in. / 20.32 cm

At this point, the rod of the long stroke cylinder body is still retracted. As a directional valve connected to the cylinder body is shifted, the rod extends its length pushing the load out an additional 8 in. (20.32 cm). Position 3 is therefore 14 in. (35.56 cm).

This duplex cylinder has achieved three positions during its extension stroke. Position 4 is realized by retracting the 6 in. (15.2 cm) stroke cylinder first. As its directional valve is shifted, fluid pressure acts on the minor area of its piston. Since it is stationary, pressure acts on the rod end of the body pulling both bodies and load back 6 in. (15.2 cm) to position 4. This would be a distance of 8 in. (20.32 cm) from start. With this duplex arrangement, three positive mechanical positions are achieved during extension — start, 6 in. (15.2 cm), 14 in. (35.56 cm); and, one position is achieved during retraction — 8 in. (20.32 cm).

Applications of this type require the use of hoses as fluid conductors since cylinder bodies move.

affecting cylinder speed

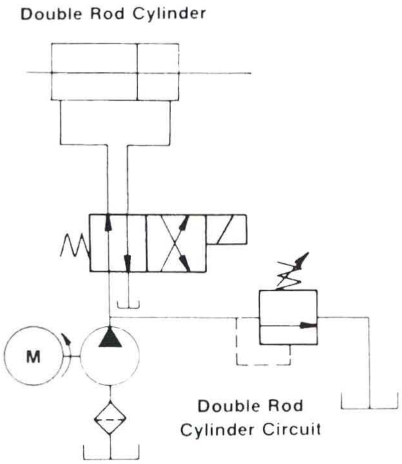
The speed at which a cylinder extends and retracts is frequently of great concern. As was illustrated, this is a function of how quickly the volume behind the cylinder piston is filled with liquid.

In the following section some typical attempts used to affect cylinder speed will be shown.

double rod cylinder circuit

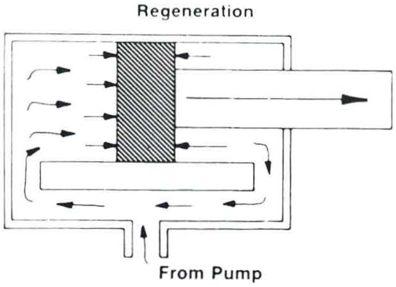
As shown previously, a double acting, single rod cylinder retracts faster than it extends. In some applications, it is required that a cylinder extend and retract at the same speed. One means of accomplishing this is with a double rod cylinder.

Since a double rod cylinder usually has the same diameter rod on both sides of the piston, piston areas exposed to system flow are equal. With the rate of flow to each side remaining constant, rod speed is the same whether extending or retracting.

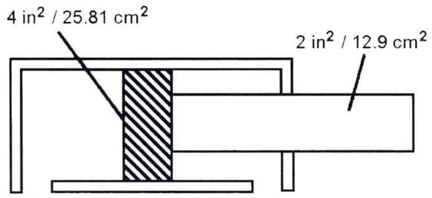


regeneration with 2:1 cylinder

In some systems, a cylinder's speed is increased by taking the discharge flow from the rod end of a cylinder and adding it to the flow to the cylinder's cap end. If a 2:1 cylinder is used in the system, the cylinder's speed will be the same whether extending or retracting.

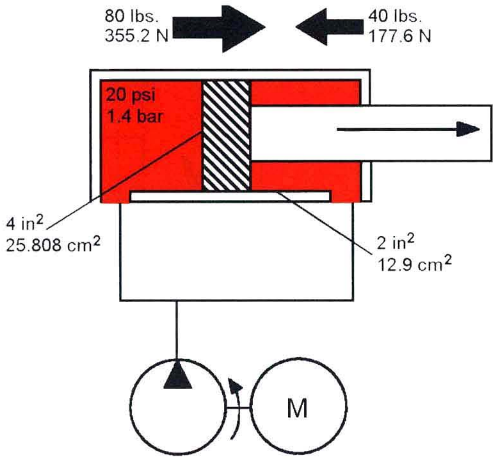


A 2:1 cylinder has a rod with a cross- sectional area equal to one half of the piston area. In other words, the rod side of the piston has one half the area exposed to pressure as the cap end side. (In actual practice the rod area is not precisely half the piston area. However, we will consider it so in order to facilitate our calculations.)



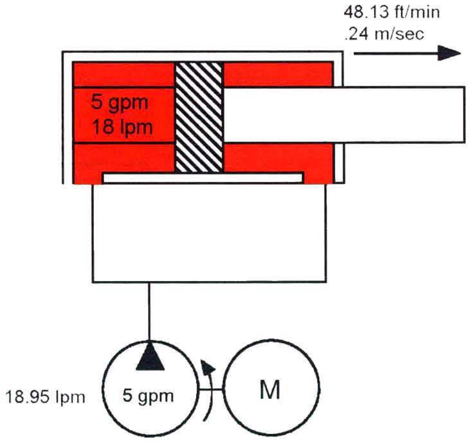
2:1 Cylinder

With the cylinder in a circuit, flow and pressure are directed to both sides of the piston at the same time. It may appear that the cylinder would be hydraulically locked. But, the difference in piston areas exposed to pressure results in a larger force being generated on the major piston area to extend the rod. With 20 psi (1.4 bar) on both sides of the piston, 80 lbs. (355.2 N) of force would be generated to extend the cylinder rod; and 40 lbs. (177.6 N) of force would be generated to retract the rod. This is a 40 lb. (177.6 N) differential in favor of extending the rod. The rod does extend. As the piston moves out, fluid which is displaced from the rod end switches position to the other side of the piston. This means pump flow is not required to fill the total volume behind the piston. Pump flow only has to fill behind an area equal to the cross sectional area of the rod.



With a 2:1 cylinder, this means the cylinder will extend twice as fast as normal with pump flow remaining the same. For example, if our 2:1 cylinder received 5 gpm (18.95 lpm) at its cap end, the following expression could be used to calculate rod speed:

$$\begin{aligned}
\underset{(\text{ft/min})\ (\text{m/sec})}{\text{Rod speed}} &= \frac{\text{gpm} \times 19.25}{\text{Rod area (in}^2)} \quad \frac{(\text{lpm} \times .167)}{(\text{cm}^2)} \\
&= \frac{5 \times 19.25}{2} \quad \frac{(18.95 \times .167)}{} \\
&= \frac{96.25}{2} \quad \frac{(3.165)}{(12.9)} \\
&= 48.13\ \text{ft/min} \quad (.24\ \text{m/sec})
\end{aligned}$$



Regeneration can only occur during rod extension. A characteristic of a 2:1 cylinder in a regenerative circuit is that rod speeds extending and retracting are basically the same.

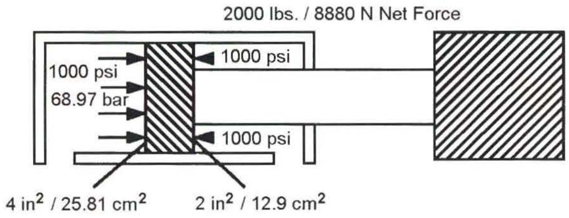
To retract the cylinder rod, the directional valve is shifted. The cap end of the cylinder is drained to tank. All pump flow and pressure is directed to the rod end side. Since the pump is filling the same volume as at the cap end side (half cap end volume), the rod retracts at the same speed.

cylinder force during regeneration

$$F = P \times A$$

A disadvantage of regeneration is that output force is reduced. Since fluid pressure is the same on both sides, the effective area on which the force is generated is the cross-sectional rod area.

In our example, as the cylinder contacted the work load, assume the pressure climbed to a relief valve setting of 1000 psi (68.97 bar). This results in 1000 psi (68.97 bar) on 4 in^2 (25.81 cm^2) equalling 4000 lbs (17760 N). on the major piston area. At the rod side, 1000 psi (68.97 bar) acting on the 2 in^2 (12.9 cm^2) minor area develops 2000 lbs. (8880 N) This is a net force of 2000 lbs. (8880 N) to extend the rod.



Net force in regenerative circuits is usually determined by using the cross-sectional area of the rod in calculations. Since areas on either side of the piston are balanced except for the rod area, net force is a result of pressure acting on a piston area equal to the rod cross-sectional area. In our example, rod area equals 2 in^2 (12.9 cm^2). 1000 psi (68.97 bar) acting on 2 in^2 (12.9 cm^2) results in a net force of 2000 lbs. (8880 N) to extend the rod as calculated previously.

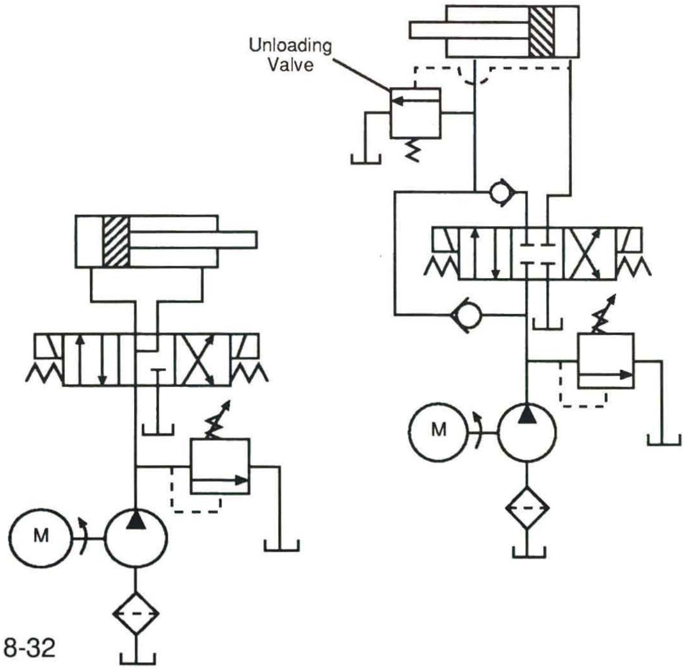
In calculations for speed and force during regeneration, the cross-sectional area of the rod is used, not the major piston area.

Connecting a cylinder in a regenerative circuit results in a faster rod speed while extending, but output force is reduced. Cylinder force is sacrificed for rod speed. And, with a 2:1 cylinder in a regenerative circuit, rod speeds extending and retracting are basically the same.

sample regenerative circuits

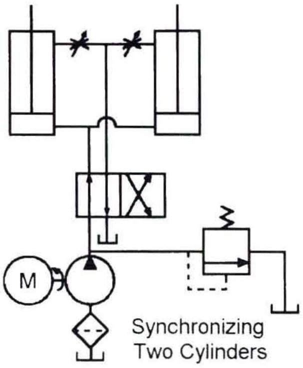
Since a force disadvantage exists while a cylinder regenerates, regeneration is frequently employed only to extend the rod to the work load. When work is to be done, the rod side of the cylinder is drained so that full force can be realized.

Illustrated are two common examples of regenerative circuits in which the rod side of the cylinder can be drained when necessary. In one circuit, this is accomplished through the center position of a directional valve. The other circuit employs an unloading valve.



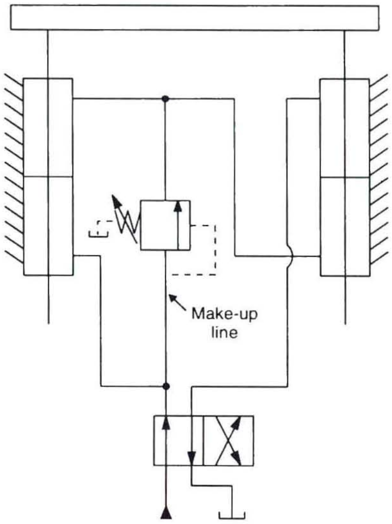
Now that we have seen how cylinders act in a circuit, in the next section we see how they are affected by wear.

synchronizing two cylinders


Synchronizing Two Cylinders

One of the most difficult, if not impossible, things to accomplish in hydraulic systems is synchronizing the movement of two cylinders, even when using the most sophisticated types of flow control valves. Typical values of synchronization range from 1/8" to 1/16" (.318 to .159 cm) depending upon cylinder stroke.

Even after the cylinders have been synchronized to within usable limits, in a relatively short period of time they will be out of synchronization because of the different wear characteristics of the cylinders and slightly different reactions of the flow control valves to the same set of conditions.

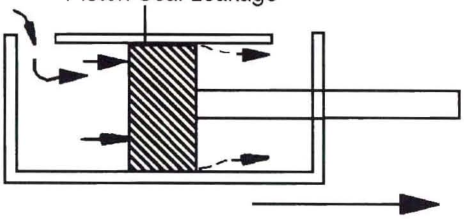


In order to achieve a more positive control of speed, some systems are designed so that the discharge flow from one cylinder is used as the input flow to another cylinder. Systems of this nature may still not be in perfect synchronization because of leakage. A characteristic of these circuits is that they are equipped with makeup and replenishing lines for the piping between cylinders.

NOTE: When two or more cylinders must stroke together, it is recommended that their piston rods be mechanically connected to one another. The connection must be rigid and could be a strong beam or the load itself.

piston seal leakage

Piston Seal Leakage



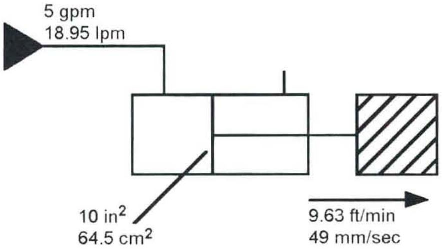
As a cylinder operates in a system, cylinder seals wear resulting in leakage at the piston rod and across the cylinder piston. Leakage at the rod seal results in a housekeeping problem and can readily be detected. Seal leakage across the cylinder piston is not as easily determined.

In the next section, we find how piston leakage causes rod speed to decrease and may even cause intensification in some cases. We then learn what checks can be made to determine if a piston seal is leaking excessively.

piston leakage affects rod speed

Piston seals are commonly cast iron rings or a resilient, synthetic compound. Leakage past piston rings or worn lip seals can reduce cylinder speed. However, a noticeable reduction in cylinder rod speed would require .5 gpm (1.89 l/min) or more leakage. At this point, the cylinder probably has extensive internal damage. In a clamping application, leakage past one piston is not too much of a problem. However, when there are many such cylinders on a machine, the clamping pressure may not be there because all the pump flow is leaking across the many cylinder pistons.

Rod speed of a cylinder is determined by how quickly pump flow can fill the volume behind a cylinder piston. This is the case whether the rod is extending or retracting. A cylinder piston with an excessively worn seal allows fluid to bypass. This fluid does not fill a volume behind a cylinder piston and therefore does not contribute to rod speed.



In the illustration, a cylinder with a major area of 10 in^2 (64.5 cm^2) receives 5 gpm (18.95 l/min) at its cap end. Rod speed extending is a function of 5 gpm (18.95 l/min) filling the volume behind 10 in^2 (64.5 cm^2) and is calculated by the following:

$$\text{Rod speed (ft/min) (*m/sec)} = \frac{\text{gpm x 19.25}}{\text{Piston area (in}^2\text{)}} \quad \frac{\text{(l/minx .167)}}{(\text{cm}^2)}$$

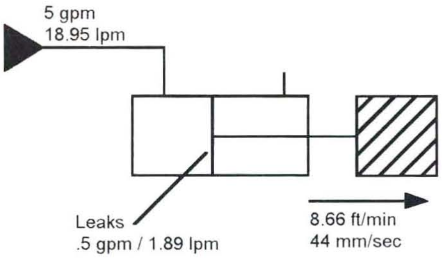
$$= \frac{5 \text{ x } 19.2}{10} \quad \frac{(18.95 \text{ x } .167)}{(64.5 \text{ cm}^2)}$$

$$= \frac{96.25}{10} \quad \frac{(3.165)}{(64.5 \text{ cm}^2)}$$

$$= 9.63 \text{ ft/min} \quad (49 \text{ mm/sec})$$

*** If calculating m/sec and the solution is less than .1 m/sec, the solution should be expressed in mm/sec.**

If a piston seal were worn excessively, bypass fluid would subtract from the 5 gpm (18.95 l/min) causing rod speed to decrease.



Assume that worn piston seals in the cylinder allow .5 gpm (1.89 l/min) to bypass during rod extension. Consequently, 4.5 gpm (17.05 l/min) fills the volume behind the piston even though 5 gpm (18.95 lpm) enters the cap end. 4.5 gpm (17.05 l/min) is used in the calculations resulting in a slower rod speed:

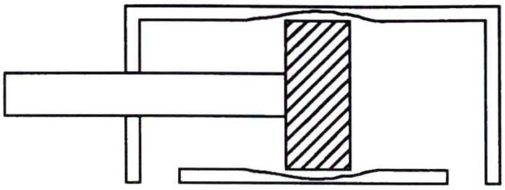
$$\text{Rod speed (ft/min) (*m/sec)} = \frac{\text{gpm x 19.25}}{\text{Piston area (in}^2\text{)}} \quad \frac{\text{(l/minx .167)}}{(\text{cm}^2)}$$

$$= \frac{4.5 \text{ x } 19.25}{10} \quad \frac{(17.05 \text{ x } .167)}{(64.5 \text{ cm}^2)}$$

$$= \frac{86.63}{10} \quad \frac{(2.847)}{(64.5 \text{ cm}^2)}$$

$$= 8.66 \text{ ft/min} \quad (44 \text{ *mm/sec})$$

Piston seal wear and the leakage which results causes cylinder rod speed to decrease even though full pump flow enters the cylinder. This means work will be done over a longer period of time. System operating temperature will increase because of the wasted hydraulic power flowing across the piston.



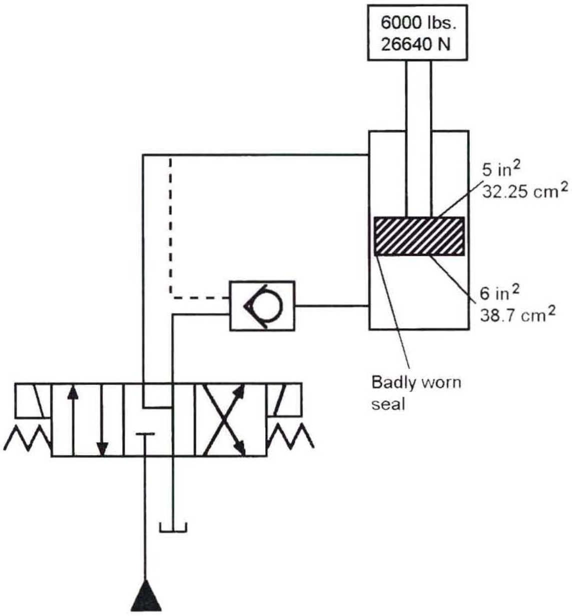
Cylinder Bore Worn in Mid-Stroke

Cylinders can leak excessively in only portions of their stroke. If a cylinder is primarily cycled in mid-stroke and excessive contamination is present in a fluid, piston seal and cylinder bore can become scored. This causes excessive leakage and reduced rod speed in mid-stroke only. During its cycle, once piston passes the worn portion of the bore, bypass fluid is reduced and rod speed increases.

intensification from piston leakage

In some cases, piston seal leakage can cause pressure intensification.

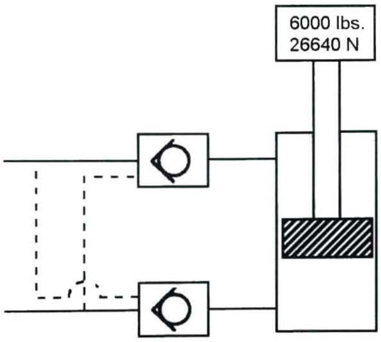
In the circuit illustrated, a cylinder in conjunction with a directional valve and pilot operated check valve is required to raise and hold a load in mid-stroke. The load is 6000 lbs. (26640 N) The cylinder piston has a major area of 6 in² (38.7 cm²) and a 5 in² (32.25 cm²) minor area.



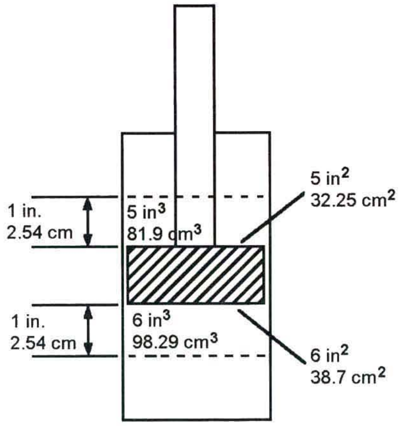
The cylinder in the circuit has an excessive leakage problem both at rod gland and piston seal. Since excessive leakage at a piston rod is usually quite obvious and a housekeeping problem, assume the rod gland seals are replaced. This still leaves the piston seal leakage.

Assume when the directional valve is shifted, flow at a pressure of 1000 psi (68.96 bar) enters the cap end of the cylinder raising the load. As the float center directional valve is centered, the load gradually falls because of leakage across the piston.

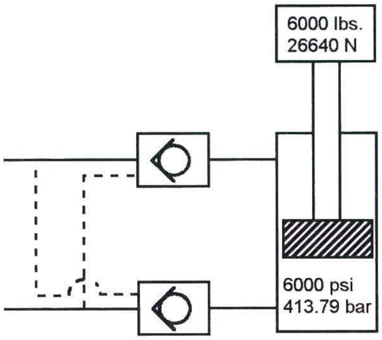
To remedy the problem, a pilot operated check valve is added to the rod-side cylinder line. Now when the directional valve is centered, fluid is not allowed to escape from the cylinder. Since a seal basically does not exist across the piston, liquid volumes on either side of the piston can communicate. Without a piston seal, piston and piston rod can be considered immersed in the liquid.



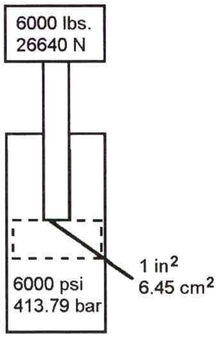
It may be felt that since a piston has no seals, the load, piston rod and piston will drift down even though fluid is trapped in the cylinder. This is not the case and may be explained by an example.



A piston and piston rod are immersed halfway in a cylinder filled with liquid. If the situation is analyzed, it can be seen that drift is impossible as long as liquid does not leak out of the cylinder. The piston has a piston area of 6 in^2 (38.7 cm^2) and a 5 in^2 (32.25 cm^2) minor area. If the piston moved down 1 in. (2.54 cm), 6 in^3 (98.29 cm^3) of oil would be displaced [6 in^2 (38.7 cm^2)] piston area x 1 in./ 2.54 cm stroke). At the other side of the piston, only a 5 in^3 (81.91 cm^3) space would be evacuated. Since 6 in^3 (98.29 cm^3) of oil does not fit into a 5 in^3 (81.91 cm^3) space, the piston cannot drift down unless some fluid leaks out of the cylinder.



Returning to our problem, we find that with piston seals worn excessively, pressures on either side of the piston are equal. Pressure acting on the piston area is offset by pressure acting on the effective area except for an area equal to the rod cross-sectional area. This is the area which supports the load. Since rod area is 1 in^2 (6.45 cm^2) and the load is 6000 lbs. (26640 N), pressure generated in the cylinder is 6000 psi (413.79 bar) (6000 lbs./1 in^2).

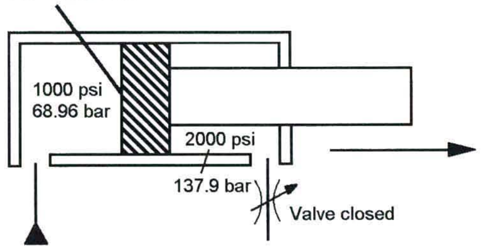


Another way of thinking of it would be to consider a rod immersed part way into a cylinder filled with liquid. If the rod had a cross-sectional area of 1 in^2 (6.45 cm^2) and supported 6000 lbs. (26640 N), 6000 psi (413.79 bar) would be generated in the cylinder. If a smaller-than-bore piston were attached to the rod, pressure would still be the same since any area of the piston not in contact with the rod would be cancelled out by equal pressures on both sides.

The generation of high pressure in this manner can be a source of ruptured seals and external leakage.

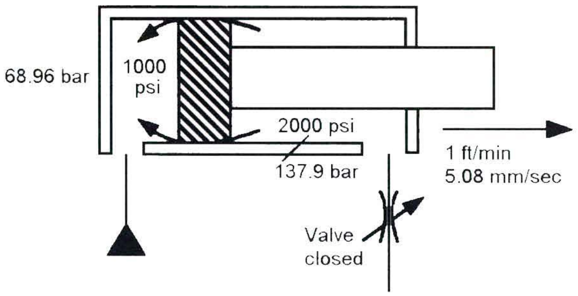
checking for piston seal leakage

Checking for piston seal leakage can be accomplished by seeing the effect of bypass flow on rod speed.



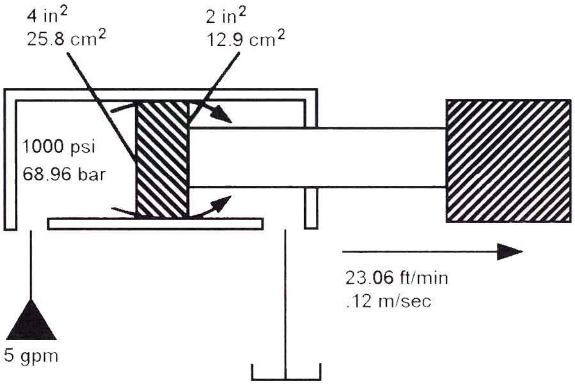
To check for piston leakage, a needle or shut off valve is piped into the rod side cylinder line. With the valve closed and the piston bottomed against the cap end, the cap end of the cylinder is subjected to full system pressure. The valve is then cracked open allowing the piston to move a short distance along its stroke. The valve is then closed. At this point, full system pressure will act on the major area of the piston resulting in an intensified pressure at the rod side.

In the illustration, a 2:1 cylinder has a piston area of 4 in^2 (25.8 cm^2) and an effective area of 2 in^2 (12.9 cm^2). With the relief valve set at 1000 psi (68.96 bar) and the rod end cylinder port blocked, 4000 lbs. (17760 N) (4 in^2 x 1000 psi) of force is generated on the piston area to extend the rod. This 4000 lbs. (17760 N) acts on the 2 in^2 effective area of the piston resulting in 2000 psi (137.9 bar) (4000 lbs./2 in^2) back pressure at the rod side.



With 1000 psi (68.96 bar) at the cap end and 2000 psi (137.9 bar) at the rod end, any fluid leakage will transfer from the rod side to the cap end side causing the piston rod to drift out. This check is performed at regular intervals along the cylinder stroke.

As a piston seal check is performed, the rate at which the rod drifts determines the reduction in rod speed as the cylinder operates in a system. In our illustrated 2:1 cylinder, assume that under test conditions the rod drifted out at a rate of 1 ft/min (5.08 mm/sec) as a 1000 psi (68.96 bar) differential existed across its piston. With the cylinder operating in a system at a differential of 1000 psi (68.96 bar), a reduction in speed of 1 ft/min (5.08 mm/sec) can be expected. With the cylinder receiving 5 gpm (18.95 lpm), the piston rod would extend at a rate of 23.06 ft/min (.12 m/sec). Whereas, if piston seals were in perfect condition, the rod would extend at 24.06 ft/min (.122 m/sec). If the reduced rod speed caused by piston seal leakage cannot be tolerated, the cylinder should be repaired or replaced.

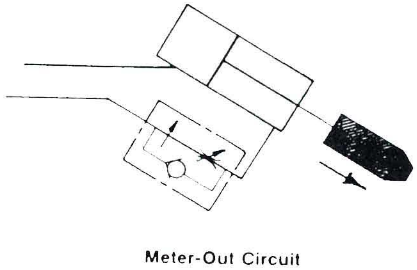


Keep in mind that cast iron piston rings can leak 1-3 in^3 (16.39-49.17 cm^3) of oil per minute at a pressure of 1000 psi (68.96 bar). They are designed to leak somewhat for purpose of lubrication. This flow should not be confused with leakage flow due to wear. Leakage flow (bypass) should be considered when studying or troubleshooting a hydraulic system.

In the next section, we find that an intensified pressure is present at the rod side of an unloaded, extending, single rod cylinder while flow is being metered out. This pressure can cause harm to a cylinder.

intensification at cylinder rod side

A flow control valve could be positioned at the rod side of a cylinder. This valve would then restrict flow from the cylinder and as a result the cylinder would not be allowed to runaway from pump flow.

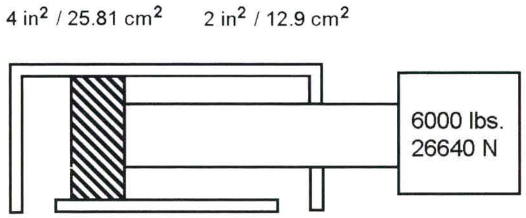


Meter-Out Circuit

This is known as a meter-out circuit.

The flow control valve kept the cylinder from running away by causing a back pressure to be generated on the minor area of piston. The resultant force kept the piston and piston rod under control.

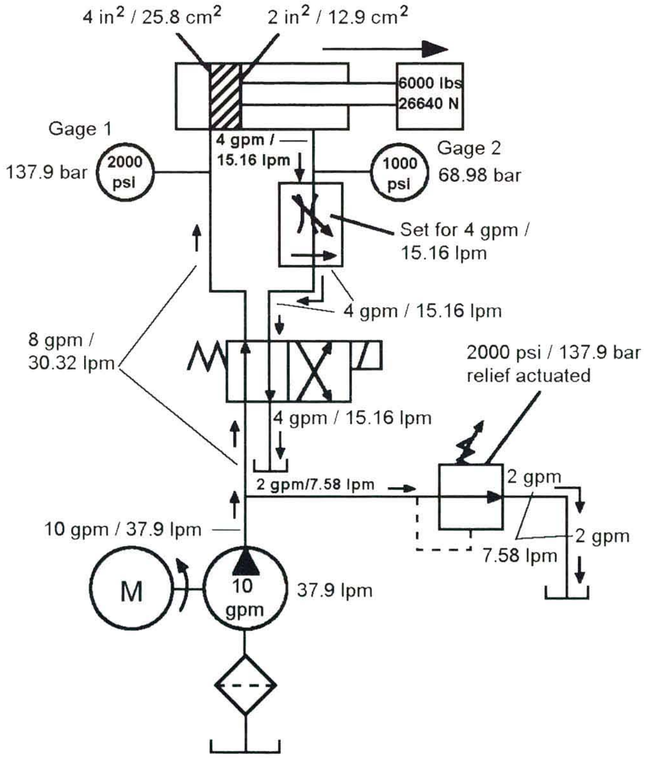
In any meter-out circuit in which a single rod cylinder is pushing out a load, whenever the pressure acting on the piston major area and its resultant force is more than is required to equal the load. The excess force develops a backpressure on the piston effective area.



In the illustration, a 2:1 cylinder is required to move a 6000 lb. (26640 N) load. The area of the piston is 4 in^2 (25.8 cm^2) and the effective area is 2 in^2. From the formula: pressure (psi) - force (lbs)/area (in^2), it can be calculated that 1500 psi (6000 lbs/4 in^2) must act on the piston area of the piston in order to equal the load.

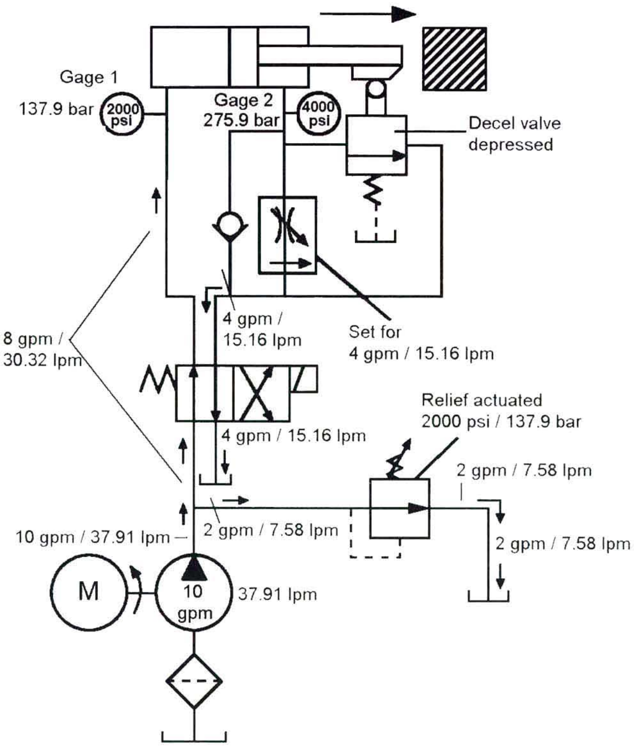
In the circuit illustration, assume that the pump flow is 10 gpm (37.9 lpm), relief valve setting is 2000 psi (137.9 bar), and the flow control valve is set to meter 4 gpm (15.16 lpm) out of the cylinder as the rod extends.

With a 2:1 cylinder, if 10 gpm (37.9 bar) enters the cap end, 5 gpm (18.95 lpm) discharges from the rod end. With the flow control set for 4 gpm (15.16 lpm), only 8 gpm (30.32 lpm) is allowed into the cylinder.



When pump/electric motor is turned on, pressure begins to build in the system. (This, of course, happens almost instantly.) When pressure reaches 1500 psi (103.4 bar), the load has been equalled. But, pump/ electric motor cannot put its flow of 10 gpm (37.9 lpm) into the system at 1500 psi (103.4 bar). More pressure is developed — 1600 (110.3 bar), 1700 (117.2 bar), 1800 (124.1 bar), 1900 psi (131 bar). When pressure reaches 2000 psi (137.9 bar), the relief valve cracks open enough to accept 2 gpm (7.58 lpm). 10 gpm (37.9 lpm) discharges from the pump at 2000 psi (137.9 bar). 8 gpm (30.32 lpm) at 2000 psi (137.9 bar) heads toward the cylinder; 2 gpm (7.58 lpm) at 2000 psi (137.9 bar) dumps over the relief valve.

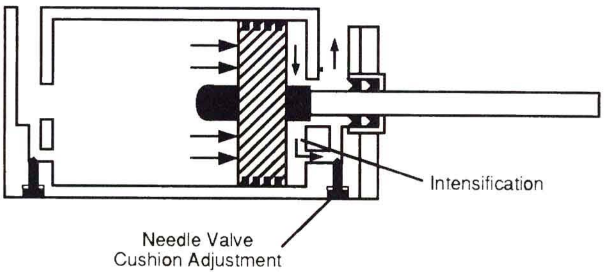
At this point in time, gage 1 at the cylinder cap end indicates 2000 psi (137.9 bar). 2000 psi (137.9 bar) on 4 in^2 (25.8 cm^2) results in 8000 lbs. (35520 N) extending the piston rod. 6000 lbs. (26640 N) is used to equal the load. The 2000 lbs. (8880 N) in excess is offset by a 1000 psi (68.96 bar) back pressure on the 2 in² (12.9 cm²) minor area. The back pressure is generated as a result of the flow control valve restriction. Gage 2 indicates 1000 psi (68.96 bar).



If the same situation occurred with no load, the restriction of the flow control valve would cause an extremely high pressure to be generated.

In the circuit illustrated, our 2:1 cylinder extends rapidly to the work load. At a point near the work, a deceleration valve is closed off forcing fluid to pass through a flow control valve as it exits cylinder rod side. The rod continues to extend for a short distance and then contacts the work load.

Between the time the deceleration valve closes and the load is contacted, 8000 lbs. (35520 N) of force is generated on the piston major area, but a load is not present to make use of it. This results in 8000 lbs. (35520 N) being absorbed by a backpressure acting on 2 in²/12.9 cm² (8000 lbs/ 2 in² = 4000 psi). Gage 2 indicates 4000 psi (275.9 bar) and continues to indicate 4000 psi (275.9 bar) until the load is contacted. Pressure intensification at the rod side of a cylinder can cause rod seals to leak or rupture.



Intensification can be expected to occur in the above manner any time an extending, single rod cylinder is being metered out without a load. Since cylinder cushions are also metered-out restrictions, pressure intensification will occur any time an extending, single rod cylinder goes into cushion. This does not affect the rod seal, but can cause leakage fluid to discharge from the needle valve cushion adjustment.

From previous illustrations, it has been shown that discharge flow from pump/electric motor is not necessarily the maximum flow rate in a system. The above example points out that a relief valve setting is not necessarily the maximum pressure in a system.

terms and idioms associated with check valves

BACK PRESSURE CHECK - A check valve used to cause the generation of a system pressure level required for the operation of other valves.

LOAD LOCK VALVE - Two pilot operated check valves in one valve body.

P O CHECK - pilot operated check valve.

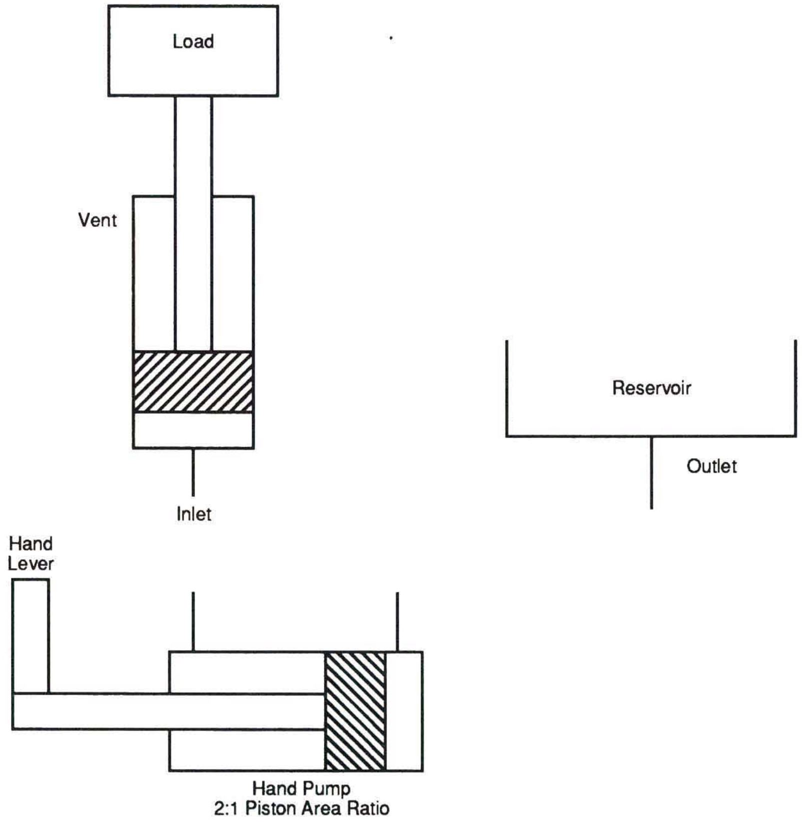
exercise
check valves, accumulators and cylinders
50 points

Instructions: Solve the problems.

1. **PROBLEM:** Connect the hand pump, cylinder and reservoir so that with each stroke of the hand lever (forward and backward), the load is raised the same amount. The load must not be allowed to drop when the hand pump is not pumping. To solve the problem, three check valves are needed. These are the only components which may be added to the circuit.

   Do not be concerned with lowering the load or refilling the reservoir.

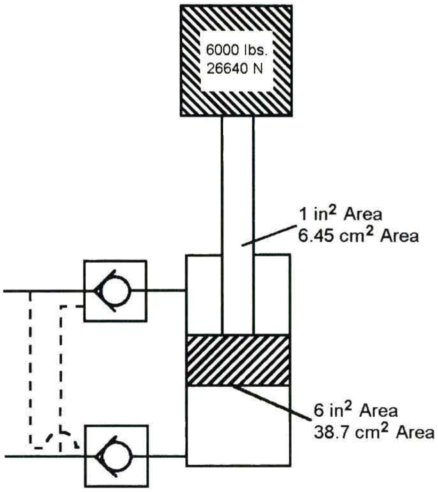
   In the hand pump, the rod side of the piston has half the area as the cap end side.



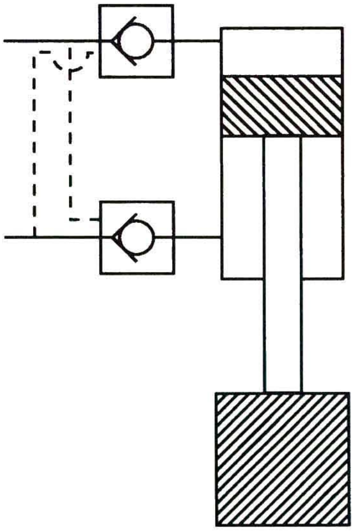
check valves, accumulators and cylinders (cont.)

2. **SITUATION:** The 6000 lb. (26640 N) load must be suspended, but excessive leakage across the piston causes the load to drift down. A pilot operated check valve was placed in each cylinder line to control the drift.

   PROBLEM: The cylinder's rod seal was blown out. Why?



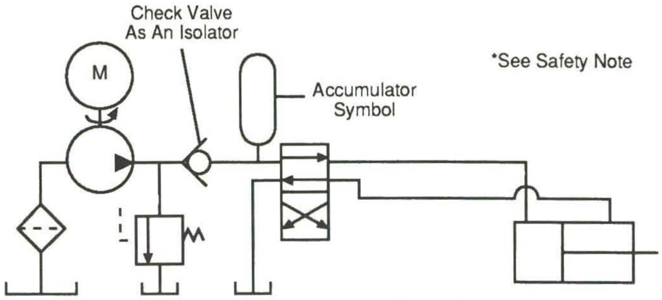
3. **PROBLEM:** Why can the load still drift down?
   (Assume pilot operated check valves seal perfectly.)



check valves, accumulators and cylinders (cont.)

4. **SITUATION:** For each machine cycle, a system requires a large volume flow for a short period of time. To conserve horsepower, pump flow should be allowed to return to tank at the least possible resistance when the accumulator is filled. This can be accomplished by connecting the pressure valve's pilot line to a certain section of the system.

   PROBLEM: Complete the design of the circuit by connecting the pressure valve's pilot line to the appropriate part of the system.



*Safety Note: In any accumulator circuit, a means could be available of automatically unloading the accumulator when the machine is shut down. See Page 8-6.

5. **PROBLEM:** In the regenerative circuit below, the components are connected so that the cylinder piston rod travels at the same speed extending and retracting. System pressure = 1000 psi (68.98 bar), gpm = 10 (lpm = 37.9).

   If the cylinder rod has a 2" (5.08 cm) diameter, what is the rod speed? ______________

   What is the maximum force which can be developed by this cylinder? ______________