Ministry of Man Power

Directorate General of Technological Education
Al Musanna College of Technology
Department of Engineering

Higher Diploma- Mechanical Engineering

COURSE NOTES ON

MIME 3140N Fluid Mechanics II

SEMESTER I - ACADEMIC YEAR 2017-18

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CHAPTER 1: FUNDAMENTALS OF FLUID POWER

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- Basic Hydraulic System components and Hydraulic Circuits
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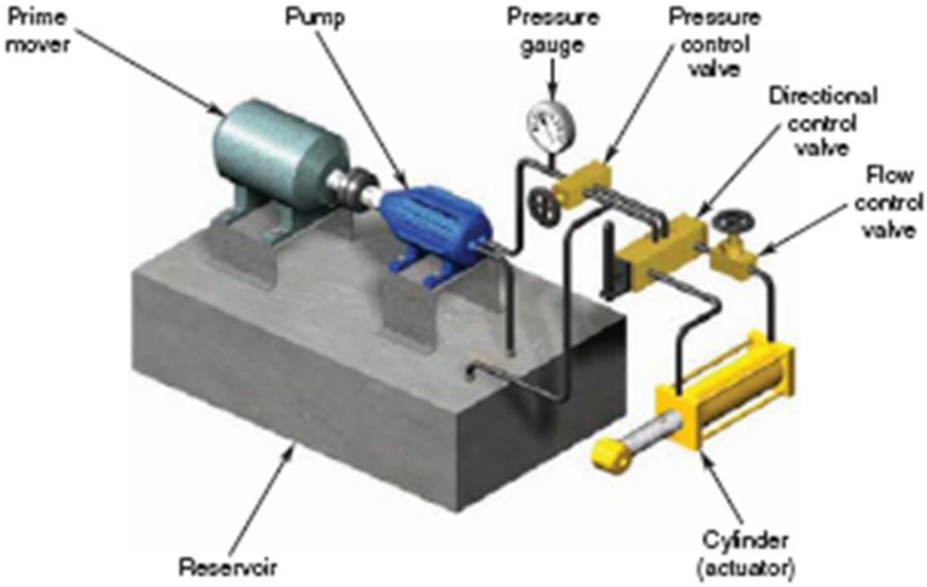
1.1 Introduction

Fluid power means using pressurized fluids in a confined system to accomplish work. Both liquids and gases are fluids. Fluid power is, thus, divided into two, Hydraulic Systems and Pneumatic Systems. Most hydraulic systems use petroleum oils, but often synthetic oils and water base fluids are used for safety reasons. Pneumatic systems use air which is exhausted to the atmosphere after doing the work.

A fluid power system accomplishes two main objectives. First, it provides substantial fluid force to move actuators in locations away from the power source where the two are connected by pipes, tubes, or hoses. A power source, for example a gasoline or diesel engine coupled to a hydraulic pump, can be housed in one area to power a cylinder or hydraulic motor one hundred meters or more away in another location. This is a decided advantage over systems using a mechanical drive train as the location of the output becomes less accessible. Second, fluid power systems accomplish highly accurate and precise movement of the actuator with relative ease.

Hydraulics and pneumatics have almost unlimited applications in the production of goods and services in nearly all sectors of any country. Several industries are dependent on the capabilities that fluid power affords. Among these are agriculture, aerospace and aviation, construction, defense, manufacturing and machine tool, marine, material handling, mining, transportation, undersea technology, and public utilities, including communications transmission systems.

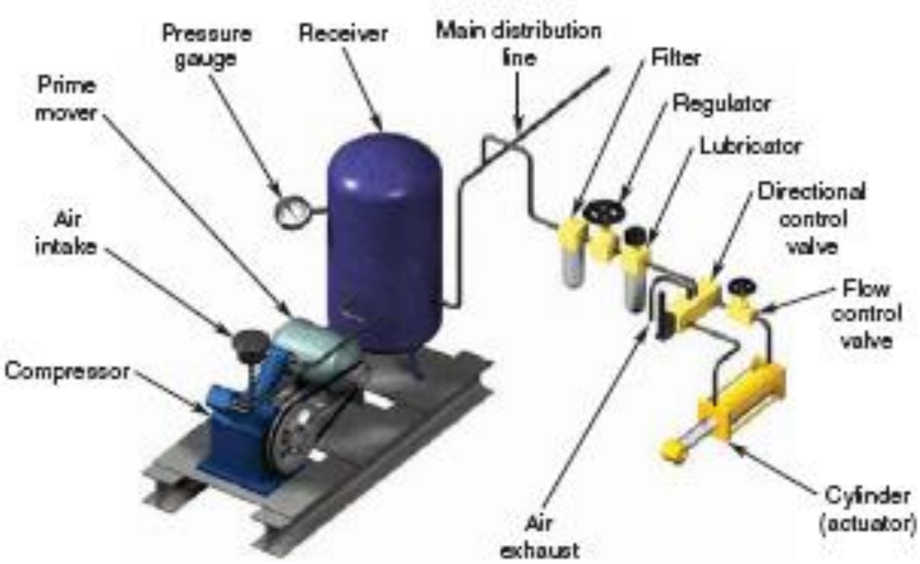


Hydraulics System

A suitable hydraulic system is shown in Figure. It consist a linear actuator with a movable piston. The system requires a liquid fluid to operate; expensive and messy and, consequently, the piping must act as a closed loop, with fluid transferred from a storage tank to one side of the piston, and returned from the other side of the piston to the tank. Fluid is drawn from the tank by a pump.

Cylinder movement is controlled by a three position changeover valve. To extend the cylinder, port A is connected to the pressure line and port B to the tank. To reverse the motion, port B is connected to the pressure line and port A to the tank. In its center position the valve locks the fluid into the cylinder (thereby holding it in position) and dead-ends the fluid lines (causing all the pump output fluid to return to the tank via the pressure regulator).

There are a few auxiliary points worthy of comment. First, speed control is easily achieved by regulating the volume flow rate to the cylinder. Precise control at low speeds is one of the main advantages of hydraulic systems. Second, travel limits are determined by the cylinder stroke and cylinders, generally, can be allowed to stall at the ends of travel so no over travel protection is required. Third, the pump needs to be turned by an external power source; almost certainly an AC induction motor which, in turn, requires a motor starter and overload protection. Fourth, hydraulic fluid needs to be very clean, hence a filter is needed to remove dirt particles before the fluid passes from the tank to the pump. One final point worth mentioning is that leaks of fluid from the system are unsightly, slippery (hence hazardous) and environmentally very undesirable.



Pneumatic system

Figure shows the components of a pneumatic system. The basic actuator is again a cylinder, with maximum force on the shaft being determined by air pressure and piston cross sectional area. Operating pressures in pneumatic systems are generally much lower than those in a hydraulic systems; 10 bar being typical which will lift 10 kg cm -2 of piston area, so a 16 cm diameter piston is required to lift the 2000 kg load specified in the previous section. Pneumatic systems therefore require larger actuators than hydraulic systems for the same load. The valve delivering air to the cylinder operates in a similar way to its hydraulic equivalent. One notable difference arises out of the simple fact that air is free; return air is simply vented to atmosphere Air is drawn from the atmosphere via an air filter and raised to required pressure by an air compressor (usually driven by an AC motor). Air also contains a significant amount of water vapor. Before the air can be used it must be cooled, and this results in the formation of condensation so, the air compressor must be followed by a cooler and air treatment unit. Compressibility of a gas makes it necessary to store a volume of pressurized gas in a reservoir, to be drawn on by the load. Without this reservoir, a slow exponential rise of pressure results in a similar slow cylinder movement when the valve is first opened. The air treatment unit is thus followed by an air reservoir. A pressure switch, fitted to the air reservoir, starts the compressor motor when pressure falls and stops it again when pressure reaches the required level.

COMPARISON OF DIFFERENT POWER SYSTEMS

System Property	Mechanical	Electrical	Pneumatic	Hydraulic
Input energy source	ICE and electric motor	ICE and hydraulic, air or steam turbines	ICE, electric motor, and pressure tank	ICE, electric motor, and air turbine
Energy transfer element	Mechanical parts, levers, shafts, gears	Electrical cables and magnetic field	Pipes and hoses	Pipes and hoses
Energy carrier	Rigid and elastic objects	Flow of electrons	Air	Hydraulic liquids
Power-to-weight ratio	Poor	Fair	Best	Best
Torque/inertia	Poor	Fair	Good	Best
Stiffness	Good	Poor	Fair	Best
Response speed	Fair	Best	Fair	Good
Dirt sensitivity	Best	Best	Fair	Fair
Relative cost	Best	Best	Good	Fair
Control	Fair	Best	Good	Good
Motion type	Mainly rotary	Mainly rotary	Linear or rotary	Linear or rotary

1.2 Pascal's Principle

Pascal's law or **the Principle of transmission of fluid-pressure** states that "pressure exerted anywhere in a confined incompressible fluid is transmitted equally in all directions throughout the fluid such that the pressure remains the same.

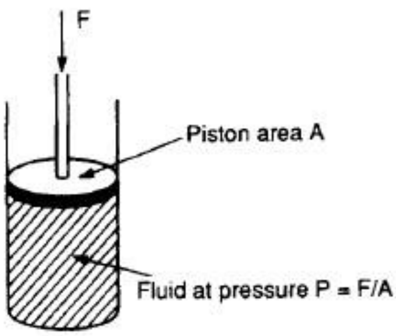
Pressure

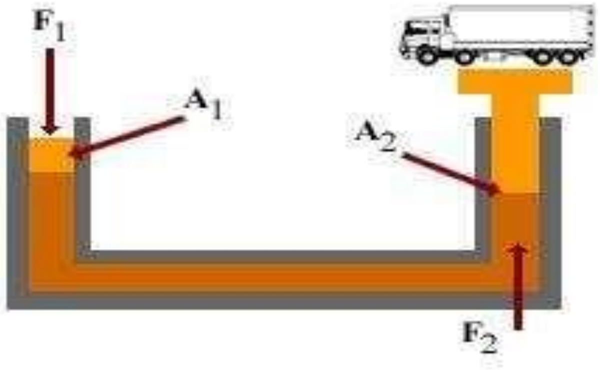
The SI system defines pressure as the force in Newton's per square Meter (N/m2). The SI unit of pressure is the Pascal (with 1 Pa = 1 N/m2). Since a pressure of 1 Pa is a relatively small pressure (1 Pa = 0.000145 psi), the term kilopascal (kPa) will be used in most practical hydraulics applications. 1 kPa = 1000 Pa = 0.145 psi.

$$1\ \mathrm{Pa} = 1\ \frac{\mathrm{N}}{\mathrm{m}^2} = 1\ \frac{\mathrm{kg}}{\mathrm{m}\cdot\mathrm{s}^2}$$

Pressure units

V·T·E	Pascal (Pa)	Bar (bar)	Technical atmosphere (at)	Standard atmosphere (atm)	Pounds per square inch (psi)
1 Pa	≡ 1 N/m^2	10^{-5}	1.0197×10^{-5}	9.8692×10^{-6}	$1.450\,377 \times 10^{-4}$
1 bar	10^5	≡ 10^6 dyn/cm^2	1.0197	0.986 92	14.503 77
1 at	$0.980\,665 \times 10^5$	0.980 665	≡ 1 kp/cm^2	0.967 8411	14.223 34
1 atm	$1.013\,25 \times 10^5$	1.013 25	1.0332	≡ p_0	14.695 95
1 Torr	133.3224	$1.333\,224 \times 10^{-3}$	$1.359\,551 \times 10^{-3}$	$1.315\,789 \times 10^{-3}$	$1.933\,678 \times 10^{-2}$
1 psi	6.8948×10^3	6.8948×10^{-2}	$7.030\,69 \times 10^{-2}$	6.8046×10^{-2}	≡ 1 lb$_F$ /in^2





It is often convenient to express pressure in terms of the height of a column of water, in meters or feet, instead of terms of psi or kPa. This is called **pressure head.**

According to Pascal"s Principle a multiplication of force can be achieved by the application of a fluid pressure in a hydraulic press. This implies for the two pistons.

$$p_1 = p_2$$

This allowes the lifting of a heavy load with a small force, as in a hydraulic lift, but of course there can be no multiplication of work. So in an ideal case with no friction:

$$p = \frac{F_1}{A_1} = \frac{F_2}{A_2}$$

Assume that the area A2 is ten times larger than A1, then a force F1 of 1 Newton applied on the first piston gets multiplied to a force F2 of 10 Newton by a factor of A2/A1 = 10 units.

$$F_2 = F_1 \frac{A_2}{A_1}$$

Derivation for Work done

Work is defined as force along (parallel) a distance:

$$W = Fd$$

Accordingly the first piston with a force F1 acting on it moves a distance of d1 and the second piston moves a distance of d2 with a force of F2.

The first piston accomplishes a work equivalent to

$$W_1 = F_1 d_1$$

and the second pistons accomplishes a work equivalent to

$$W_2 = F_2 d_2$$

Assuming no leakage between piston 1 and piston 2 and the lines connecting them, the volume of fluid displaced by the first piston is the same as the volume of fluid accommodated by the second cylinder.

$$V_1 = V_2$$

$$V_1 = A_1 d_1 \text{ and } V_2 = A_2 d_2$$

$$A_1 d_1 = A_2 d_2$$

Now the work done can be expressed as

$$W_1 = F_1 d_1 = F_1 \frac{A_2}{A_1} d_2 = F_2 d_2 = W_2$$

So the total work done remains the same. Energy cannot be created or destroyed it can only be transformed from one form to another. And Energy is the capacity of doing work.

Derivation for Power

Power is work per unit time:

$$P_1 = W_1/t_1$$

$$P_2 = W_2/t_2$$

Since, $t_1 = t_2$, No time lag experiences.

And, since $W_1 = W_2$.

Power is same.

Derivation for Velocity

Let the velocity of piston one is v_1 and the velocity of piston two be v_2.

$$V_1 = \frac{d_1}{t} \text{ and } V_2 = \frac{d_2}{t} \text{ since } t_1 = t_2 = t$$

Since d1 is greater than d2, piston one will move faster.

V_1 is greater than V_2

Flow rate

The flow rate is defined as the volume of fluid that is passing through a given cross sectional area per unit time. Or is defined as the volume divided by time, Q = V/ t

The SI unit is m^3/s.

Units in Hydraulic Systems

- Force in Newton [N]
- Pressure in [N/m2] or Pascal [Pa= N/m2] or bar [1 bar =100kPa]
- Work in joule [J] [1J = 1 Nm]
- Power Work/time [1 Watt = J/s]
- $P_{psi} = 0.145038 \times P_{kPa}$, $P_{kPa} = 6.89476 \times P_{psi}$

To **recapitulate,** in an ideal hydraulic **system:**

- Pressure remains the same
- Work remains the same
- Displaced fluid remains the same
- Force gets multiplied
- Distance traveled is different
- Velocity is different
- Power remains the same.

The conversion factors

Prefix	Symbol	Power	Prefix	Symbol	Power
mega-	M	10^6	centi-	c	10^{-2}
kilo-	k	10^3	milli-	m	10^{-3}
hecto-	h	10^2	micro-	μ	10^{-6}
deca-	D	10^1	nano-	n	10^{-9}
deci-	d	10^{-1}	pico-	p	10^{-12}

	cm	**METER**	km
1 centimeter =	1	10^{-2}	10^{-5}
1 **METER** =	100	1	10^{-3}
1 kilometer =	10^5	1000	1

	METER^3	cm^3	liter
1 **CUBIC METER** =	1	10^6	1000
1 cubic cm =	10^{-6}	1	1.000×10^{-3}
1 liter =	1.000×10^{-3}	1000	1

Equivalents

One Horsepower Equals: 746 Watts

One U.S. Gallon Equals: 3.785 Liters

One Liter Equals 0.2642 U.S. Gallons

1.3 Exercise examples:

1. The piston and cylinder with an inside diameter of 10 cm and are loaded with a force (F) of 500 N. What is the pressure (P) inside the cylinder?

2. For a piston with Ø40mm, the maximum pressure should not exceed 3500 KPa. Find the maximum force the system can withstand.

3. A piston and cylinder are required to support a force of 10 kN. Pressure should not exceed 70 bar. What is the required size of the cylinder?

4. An input cylinder with a diameter of 30 mm is connected to an output cylinder with a diameter of 80 mm. A force of 1000N is applied to the input cylinder. What is the output force? How far would we need to move the input cylinder to move the output cylinder 100 mm?

5. The power and load carrying capacity of a hydraulic cylinder (extension) are10 kW and 2000 N respectively. Find the piston velocity during extension. If the area of piston side and rod side is 2:1, find the retraction speed.

6. A hydraulic cylinder is used to compress a car body in 10 seconds. The operation requires a stroke length of 3 m and a force of 40000N.if a 7.5 N/mm^2 pump has been selected, find

(i) Required piston area and piston diameter.

(ii) The necessary pump flow rate

(iii) The mechanical power capacity in KW.

(iv) The hydraulic power.

7. A hydraulic cylinder is to compress a car body down to bale size is 8 sec. The operation requires a 3 m stroke and a 40000 N force. A pump with 10 MPa has been selected for the operation. Assuming 400N frictional force and leakage of 1.0 LPM in the cylinder, Find

1. The required piston area.
2. Actual pump flow rate.
3. The hydraulic power
4. The output power delivered by the cylinder to the load.
5. The efficiency of the cylinder with the given frictional force and leakage.

8. A Pump supplies 20 gallon/minute to a 50 mm diameter double acting hydraulic cylinder. The load acting during extending and retracting stroke is 5000 N and diameter of the piston rod is 25 mm. Find

a. The hydraulic pressure during extension stroke

b. The piston velocity in extension stroke

c. Cylinder capacity for extension stroke

d. Hydraulic pressure during return stroke

e. Find piston velocity during return stroke

f. Cylinder capacity for return stroke.

9. A cylinder with a bore diameter of 80 mm and a rod diameter of 25 mm is to be used in a system with a 60 LPM pump. What are the extension and retraction speeds?

Solved examples

A double acting hydraulic cylinder has a bore of 100 mm. The rod is 40 mm diameter and the stroke is 120 mm. It must produce a pushing force of 12 kN. The flow rate available in both directions is 12 dm^3/min.

Calculate:
i. The system pressure needed.
ii. The force with which it pulls given the same pressure.
iii. The speed on the outward stroke.
iv. The speed of retraction.
v. The power used on the outstroke.

Assume ideal conditions throughout.

SOLUTION

$A = \pi D2/4 = \pi \text{ x } 0.1^2/4 = 7.854 \text{ x } 10^{-3} \text{ m}^2$

$p = F/A = 12000/ 7.854 \text{ x } 10^{-3} = 1.528 \text{ x } 10^6 \text{ N/m}^2 = \text{or } 1.528 \text{ MPa}$

$a = \pi d^2/4 = \pi \text{ x } 0.04^2/4 = 1.257 \text{ x } 10^{-3} \text{ m}^2$

Pulling force $= p(A\text{-}a) = 1.528 \text{ x } 10^6 \text{ x } (7.854 \text{ x } 10^{-3} - 1.257 \text{ x } 10^{-3}) = 10008 \text{ N}$

Flow rate $Q = 0.012/60 = 20 \text{ x } 10^{-3} \text{ m}^3/\text{s}$

Speed on the outward stroke $= Q/A = 20 \text{ x } 10^{-3} / 7.854 \text{ x } 10^{-3} = 0.025$ m/s or 25 mm/s

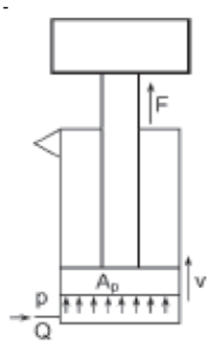
Speed of retraction $= Q/(A\text{-}a) = 20 \text{ x } 10^{-3}/(7.854 \text{ x } 10^{-3} - 5.027 \text{ x } 10^{-3}) = 0.03$ m/s or 30 mm/s

Power $= pQ = 1.528 \text{ x } 10^6 \text{ x } 20 \text{ x } 10^{-3} = 305.6$ Watts

1.4 Basic Hydraulic System Circuit and Components

Figure shows the circuit of a simple hydraulic system, drawn in both functional- sectional schemes and standard hydraulic symbols. The function of this system is summarized in the following:

- The prime mover supplies the system with the required mechanical power. The pump converts the input mechanical power to hydraulic power
- The energy-carrying liquid is transmitted through the hydraulic transmission lines: pipes and hoses. The hydraulic power is controlled by means of valves of different types. This circuit includes three different types of valves: a pressure control valve, a directional control valve, and a flow control (throttle check) valve.
- The controlled hydraulic power is communicated to the hydraulic cylinder, which converts it to the required mechanical power. Generally, the hydraulic power systems provide both rotary and linear motions.



Lifting a body vertically by a hydraulic cylinder

In the figure above a load is lifted by a hydraulic cylinder. This cylinder acts on the lifted body by a force F and drives it with a speed v. It is a single acting cylinder which extends by the pressure force and retracts by the body weight. The pressurized oil flows to the hydraulic cylinder at a flow rate Q (volumetric flow rate, m3/s) and its pressure is p. Neglecting the friction in the cylinder, the pressure force which drives the piston in the extension direction is given by

$$F = pA_p.$$

During the time period, ắt, the piston travels vertically a distance y.

The volume of oil that entered the cylinder during this period is

$$\boldsymbol{V = A_p y}$$

Then, the oil flow rate that entered the cylinder is

$$Q = \frac{V}{\Delta t} = \frac{\boldsymbol{A_p y}}{\Delta t} = A_p v$$

Assuming an ideal cylinder, then the hydraulic power inlet to the cylinder is

$$N = Fv = pA_p \, {}^{Q}\!/_{A_n} = Qp$$

$A_p = Piston\ area,\quad m^2$
$p = Pressure\ of\ inlet\ oil, Pa$
$Q = Flow\ rate, {}^{m^3}\!/_{s}$
$V = Piston\ swept\ volume, m^3$

The mechanical power delivered to the load equals the hydraulic power delivered to the cylinder. This equality is due to the assumption of zero internal leakages and zero frictional forces in the cylinder. The assumption of zero internal leakage is practical, for normal conditions. However, for aged seals, there may be non-negligible internal leakage. A part of the inlet flow leaks and the speed **v** becomes less than (**Q/Ap**). Also, a part of the pressure force overcomes the friction forces. Thus, the mechanical power output from the hydraulic cylinder is actually less than the input hydraulic power ($F_v < Q_p$).

1.5 Advantages and disadvantages of a hydraulic system

The main **advantage**s of the hydraulic power systems are the following:

1. High power-to-weight ratio.
2. Self-lubrication, Automatic lubricating provision to reduce to wear
3. Large load capacity with almost high accuracy and precision
4. High acceleration capability and a rapid response of the hydraulic motors.
5. High stiffness, which allows stopping loads at any intermediate position.
6. Possibility of energy storage in hydraulic accumulators.
7. Flexibility of transmission compared with mechanical systems.
8. Availability of both rotary and rectilinear motions.

Hydraulic power systems have the following **disadvantages:**

1. Hydraulic power is not readily available, unlike electrical. Hydraulic generators are required.
2. High cost of production due to the small clearances and high precision production process.
3. Limitation of the maximum and minimum operating temperature.
4. Fire hazard when using mineral oils.
5. Oil filtration problems. Special treatment is needed to protect from rust, corrosion, dirt etc.

CHAPTER 2: HYDRAULIC PUMPS AND ACTUATORS

In this chapter of Fluid Mechanics II you will learn:

- Principles of operation of a reciprocating pump
- Principles of operation of a rotary pump
- Definitions of *Heads and efficiencies* for a centrifugal pump
- Efficiencies for a pump system
- Efficiencies of a centrifugal pump
- Actuators.

2.1 Introduction

2.2 Hydraulic Machines (Pumps, Motors, Actuators)

Hydraulic machines are defined as those machines which convert either hydraulic energy (energy possessed by water) into mechanical energy (which may further be converted into electrical energy) or mechanical energy into hydraulic energy. The hydraulic machines, which convert the mechanical energy into hydraulic energy, are called pumps. The hydraulic machines, which convert the hydraulic energy into mechanical energy, are called actuators if the motion is linear or motor if the motion is rotary.

A pump is used to impart motion to a liquid. It provides the force required to transmit power and motion. These are mainly classified into two categories:

A. Non-positive displacement pumps

B. Positive displacement pumps.

A. Non-Positive Displacement Pumps

These pumps are also known as hydro-dynamic pumps. In these pumps the fluid is pressurized by the rotation of the propeller and the fluid pressure is proportional to the rotor speed. These pumps are generally used for **low-pressure and high-volume** flow applications. The fluid motion is generated due to rotating propeller. The fluid pressure and flow generated due to **inertia effect** of the fluid. These pumps provide a smooth and continuous flow but the flow output decreases with increase in system resistance (load). The flow output decreases because some of the **fluid slip back** at higher resistance. The fluid flow is completely stopped at very large system resistance and thus the volumetric efficiency will become zero. Therefore, the flow rate not only depends on the rotational speed but also on the resistance provided by the system.

The advantages of non-positive displacement pumps are:

1. Lower initial cost,
2. Less operating maintenance because of less moving parts,
3. Simplicity of operation,
4. Higher reliability and suitability with wide range of fluid.
5. These pumps provide a smooth and continuous flow.
6. Centrifugal pump is an example of non-positive displacement pumps.

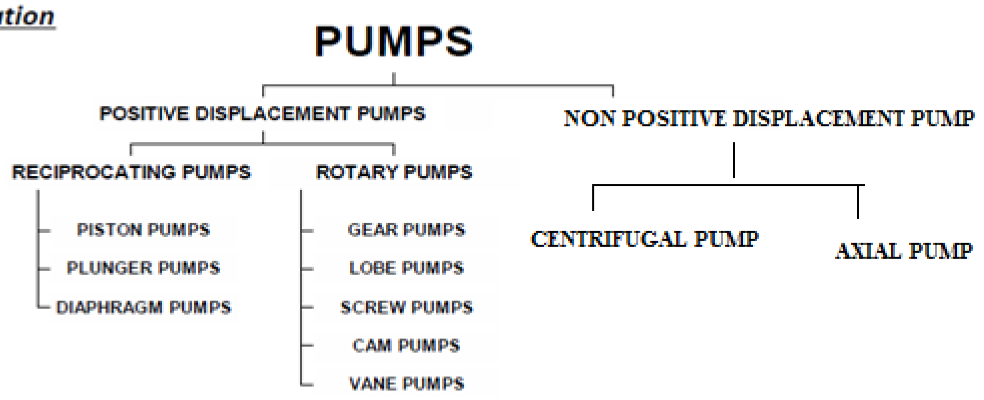
B. Positive displacement pump

The 'positive displacement' pumps are those pumps in which the liquid is sucked and then it is actually pushed or displaced due to the thrust exerted on it by a moving member, which results in lifting the liquid to the required height. Positive Displacement pumps do not use impellers, but rely on rotating or reciprocating parts to directly push the liquid in an enclosed cavity. These pumps deliver a constant volume of fluid in a cycle. The discharge quantity per revolution is fixed in these pumps and they produce fluid flow proportional to their displacement and rotor speed. The output fluid flow is constant and is independent of the system pressure (load). Displacement pumps fall into two major classes: reciprocating and rotary pumps.

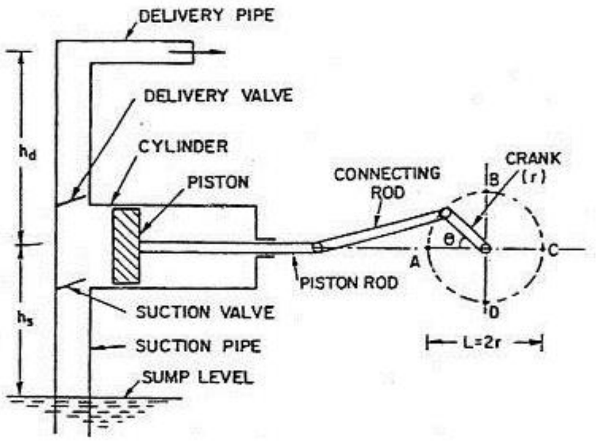
The advantages associated with these pumps are:

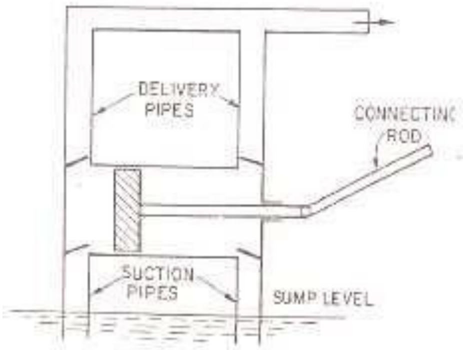
1. Since the high-pressure and low-pressure areas are separated there is no fluid back leak.
2. Capability to generate high pressures, up to 6800 Bar. And Q between 0.4 to 55,000 lpm
3. High volumetric efficiency,
4. High power to weight ratio,
5. Great flexibility of performance, can operate over a wide range of pressure requirements and speed ranges

Classification



2.3. Reciprocating Pumps





Piston pumps are meant for the high-pressure applications. These pumps convert the rotary motion of the input shaft to the reciprocating motion of the piston.

Working Principle reciprocating pump:

The suction and delivery valves are allow the water to flow in one direction only. Suction valve allows water from suction pipe to the cylinder which delivery valve allows water from cylinder to delivery pipe only. The movement of the piston is obtained by connecting the piston rod to crank by means of a connecting rod. The crank is rotated by means of an electric motor. When crank starts rotating, the piston moves to and fro in the cylinder. The movement of the piston towards right create partial vacuum in the cylinder. But on the surface of the liquid in the sump, atmospheric pressure is acting which is more than the pressure inside the cylinder. Thus the liquid is forced in the suction pipe from the sump. This liquid opens the suction valve and enters the cylinder.

When the movement of the piston is towards left, the pressure of the liquid inside the cylinder increases more than the atmospheric pressure. Hence suction valve closes and delivery valve opens. The liquid is forced into the delivery pipe and is raised to a required height. An **air vessel** is a closed chamber connected on the suction or delivery or both sides of the reciprocating pump to obtain a more uniform flow. It also helps to reduce the possibility of separation and cavitation, Allows pump to run at high speed and allow to save large amount of power.

A double acting reciprocating pump which does the pumping work on **both the sides of piston**; hence it is termed as Double acting Pump.

Advantages are: High efficiency, No priming needed, Can deliver water at high pressure, Can work in wide pressure range and Continuous rate of discharge.

Disadvantages are: High maintenance cost, No uniform torque, Low discharging capacity, Pulsating flow, Difficult to pump viscous fluid and high wear in parts.

2.4 Rotary Pumps

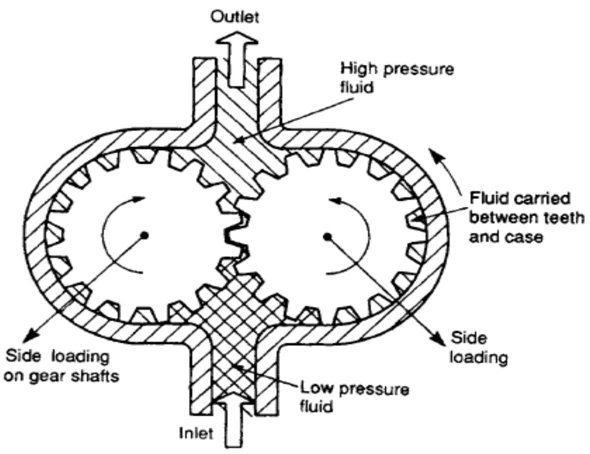
External gear pump

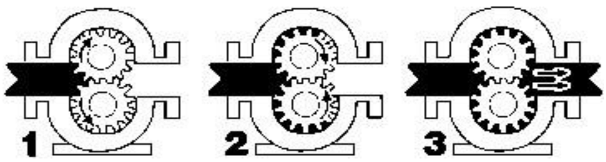
Working Principle:

Gear pump is a simple positive displacement pump. The external gear pump uses two identical gears rotating against each other - one gear is driven by a motor and it in turn drives the other gear. Each gear is supported by a shaft with bearings on both sides of the gear. It has two meshed gears revolving about their respective axes. The operation is based on the carrying of fluid between the teeth of the meshing gears and the pump housing. One of the gears is connected to the drive shaft and the second gear is driven by the meshing of the driven gear.

The operation of the pump:

As the teeth come out of mesh, vacuum is created at the inlet port and liquid flows into the pump. It is carried between the teeth and the casing to the discharge side of the pump. The teeth come back into mesh and the liquid is forced out the discharge port.





Advantages:

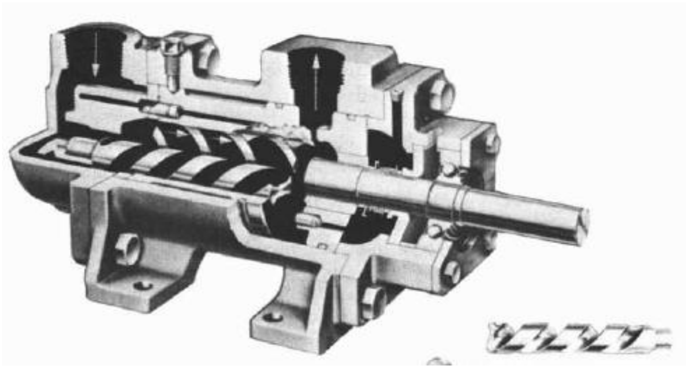
1. Close tolerances and shaft support on both sides of gears. So, they run at high pressures.
2. The rigid design allow for very high pressures range with high efficiency.
3. Have the ability to pump highly viscous fluids. So, it can handle high viscous fluids.
4. Subject to physical constraints the discharge pressure can be very high.

Applications with limitations

1. Small external gear pumps usually operate at 3450 rpm and larger at speeds up to 640 rpm.
2. Often used as lubrication pumps in machine tools.
3. They are not well suited to handling abrasive or extreme high temperature applications.
4. Gear pumps rely on precision clearances and have several rotating elements. So it is expensive.

2.5 Screw Pump:

The screw pump is an axial flow pump, which has two or three screws gears meshing in a closed housing. One of these gear is driven by the motor while the other gears are driven by this and in different direction, they tend to build up the suction pressure at bottom part, this makes upward and also centrifugal force is experienced. The screws are each mounted on shafts that run parallel to each other. The turning of the screws, and consequently the shafts to which they are mounted, draws the fluid through the pump. As with other forms of rotary pumps, the **clearance** between moving parts and the pump's casing is minimal. Side thrusts, often associated with gear and vane pumps, are not present in screw pumps.




Advantages of screw gears pumps:

1. Very low noise.
2. Very high speed reaches to 30000 rev/min.
3. Continuous flow (like centrifugal pump).
4. Used in plastic injection and in marine ships (military service).
5. Wide range of liquids and viscosities.
6. Wide range of flows and pressures

2.6 Vane Pump

The operation of the vane pump is based on eccentric mounting of the rotor and the casing. The rotor in a vane pump is connected to the prime mover through a shaft. The rotor contains radial slots of rectangular in shape and each slot contains a vane.

Working: As the rotor turns, centrifugal force causes the outer edge of each vane to slide along the surface of the housing cavity as the vanes slide in and out of the rotor slots. It provides a tight hydraulic seal to the fluid which is more at the higher rotation speed due to higher centrifugal force. This produces a suction cavity in the ring as the rotor rotates. It **creates vacuum** at the inlet and therefore, the fluid is pushed into the pump through the inlet. During one half of the rotation the oil enters between the vane and the housing then this area starts to decrease in the second half which permit the pressure to be produced, then the oil comes out through the output port as it is pressurized.

Key points:

Capacity depends upon eccentricity, expansion of vanes, and width of vanes and speed of rotor.

The fluid flow will not occur when the eccentricity is zero.

The operating range of these pumps varies from -32 °C to 260 °C.

Classifications of vane pumps:

External vane pumps can handle large solids.

Flexible vane pumps can handle only the small solids but create good vacuum.

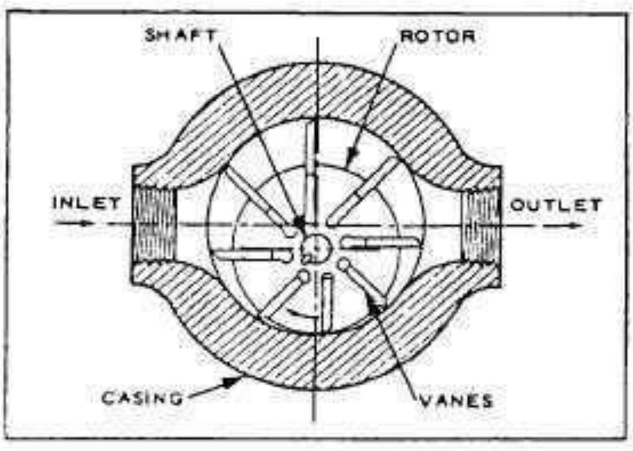
Sliding vane pumps can run dry for short periods of time and can handle small amounts of vapor.

The advantages of vane pumps are as follows:

1. They provide uniform discharge with negligible pulsations.
2. Their vanes are self-compensating for wear and vanes can be replaced easily.
3. They are light in weight and compact.
4. Volumetric and overall efficiencies are high.
5. Discharge is less sensitive to changes in viscosity and pressure variations.
6. Can handle thin liquids (low viscosity) at relatively higher pressure.

The disadvantages of vane pumps are as follows:

1. They are not suitable for abrasive liquids.
2. They require good seals.
3. They require good filtration systems and foreign particle can severely damage pump.
4. These pumps are not suitable for high speed applications.
5. The maintenance cost is also higher due to many moving parts.



Expression for the Theoretical Discharge of Vane Pumps

Let *D*C be the diameter of a cam ring in m, *D*R the diameter of rotor in m, *L* the width of rotor in m, *e* the eccentricity in m, *V*D- pump volume displacement in m3/rev and *e* max , maximum possible eccentricity in m.

From geometry the maximum possible eccentricity,

$$e_{\text{max}} = \frac{D_C - D_R}{2}$$

Themaximum value of eccentricity produces the maximum volumetric displacement

$$V_{D(\text{max})} = \frac{\pi}{4}(D_C^2 - D_R^2)L$$

this can be simplified as

$$V_{D(\text{max})} = \frac{\pi}{4}(D_c - D_R)(D_c + D_R)L$$

$$V_{D(\text{max})} = \frac{\pi}{4}(D_c + D_R)(2e_{\text{max}})L$$

The actual volumetric displacement occurs when $e_{\text{max}} = e$. Hence,

$$V_{D(\text{max})} = \frac{\pi}{2}(D_C + D_R)e\,L\ \text{m}^3/\text{rev}$$

When the pump rotates at N rev/min (RPM),the quality of discharge by the vane pump is given by

$$Q_T = v_D \times N$$

Theoretical discharge is

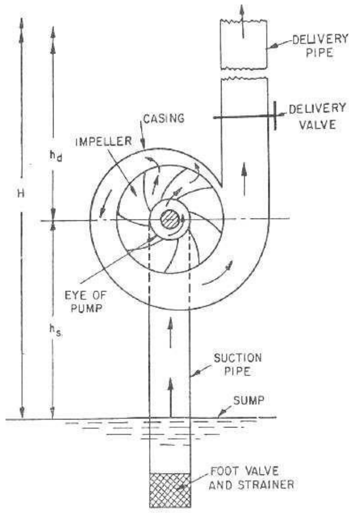
$$Q_T = \frac{\pi}{2}(D_C + D_R)e\,L\ \ \text{m}^3/\text{min}\ \ \mathbf{X\,N}$$

2.7 Centrifugal Pumps

This the most common type of pumping machinery used to move liquids through a piping system. Centrifugal pumps convert the mechanical energy into hydraulic energy by centrifugal force on the liquid. These pumps are suitable for low pressure, high volume flow applications. Since these pumps are not able to develop high pressure, these are not used in the fluid power industries. Low lift centrifugal pump are capable of working against heads up to 15 m. Medium lift centrifugal pump are basically used against the heads as high as 40 m and High lift centrifugal pumps are used to deliver liquids at heads above 40 m.

The main parts of a centrifugal pump are:

1. Impeller
2. Casing
3. Suction pipe
4. Delivery pipe



Impeller: The impeller is the rotating part of a centrifugal pump which consists of a series of backward curved vanes. The impeller is mounted on a shaft which is connected to the shaft of an electric motor.

Casing: The casing of the pump is an air tight passage surrounding the impeller and is designed in such a way that the kinetic energy of the water discharged at the outlet of the impeller is converted into pressure energy before the water leaves the casing and enters the delivery pipe.

Types of casings used.

Volute casing: Having a spiral shape in which area of flow increases gradually. The increase in area of flow decreases the velocity of flow. The efficiency is less due to the formation of eddies.

Vortex casing: In this type, a circular chamber is introduced between impeller and casing. The liquid from impellers enters to vortex casing and then to volute chamber.

Casing with guide blade arrangement, the impeller is surrounded by a series of guided blades and is designed in such a way that to have with minimum shock.

Suction pipe: Suction pipe is the pipe whose one end is connected to the inlet of the pump and the other end is dipped into water in a sump. **A foot valve** which is a non-return valve or one way type of valve is fitted at the lower end of the suction pipe. The foot valve open only in the upward direction. A strainer is also fitted at the lower end of the suction pipe.

Delivery pipe: Delivery pipe is a pipe whose one end is connected to the outlet of the pump and other end delivers the water at a required height.

Priming of a centrifugal pipe:

Priming of the pump is a process by which the suction pipe, casing of the pump and a portion of the delivery valve is completely filled up from outside source with the liquid to be raised by the pump before starting the pump. Thus the air from these parts of the pump is removed and filled with the liquid to be pumped.

2.8 Definitions of Heads and Efficiencies for a Centrifugal Pump

Suction Head (h_s). It is the vertical height of the centre line of the pump above the water surface in the tank from which water is lifted.

*Delivery Head (h_d).*The vertical distance between the centre line of the pump and the water surface in the tank to which water is delivered is known as delivery head.

Static Head (H_S) .The sum of suction head and delivery head is known as static head. This is represented by „H_s‟ and is written as $H_s=h_s+h_d$.

Total Head (or) Manometric Head (H_t). The total head is defined as the head against which a centrifugal pump has to work. It is given by the following expressions:

H_t = Total head at outlet of the pump – Total head at the inlet of the pump.

$$H_t = \left(\frac{p_o}{\rho g} + \frac{V_o^2}{2g} + Z_o\right) - \left(\frac{p_i}{\rho g} + \frac{V_i^2}{2g} + Z_i\right)$$

$$\boxed{H_t = h_s + h_d + h_{fs} + h_{fd} + \frac{V_d^2}{2g}}$$

$$H_t = \left(\frac{p_o}{\rho g} + \frac{V_o^2}{2g} + Z_o\right) - \left(\frac{p_i}{\rho g} + \frac{V_i^2}{2g} + Z_i\right)$$

$$\boxed{H_t = h_s + h_d + h_{fs} + h_{fd} + \frac{V_d^2}{2g}}$$

Pump Performance:

The performance of a pump is determined by the following efficiencies:

1. Volumetric efficiency: It is the ratio of actual flow rate of the pump to the theoretical flow rate of the pump. This is expressed as follows:

Volumetric Efficiency(η_{vol}) = $\frac{\text{Actual Discharge}}{\text{Theoretical Discharge}} \times 100$ $\qquad \eta_{vol} = \frac{Q_A}{Q_T} \times 100$

Theoretical discharge: Q_T **=Volume displaced /revolution X Speed in rpm**

$$Q_T\left(\frac{m^3}{min}\right) = V_D\left(\frac{m^3}{rev}\right) \times N(rpm)$$

Volumetric efficiency indicates the amount of leakage that takes place within the pump. This is due to manufacture tolerances and flexing of the pump casing under designed pressure operating conditions. For gear pumps, = 80%–90%, For vane pumps, = 90% - 92%. ,piston pumps, = 90–98%.

2. Mechanical efficiency: It is the ratio of the pump output power assuming no leakage to actual power delivered to the pump:

$$\text{Mechanical efficiency } (\eta_m) = \frac{\text{Pump output power assuming no leakages}}{\text{Actual power delivered to the pump}}$$

$$(\eta_{mech}) = \frac{P\left(\frac{N}{m^2}\right) \times Q_T\left(\frac{m^3}{s}\right)}{\frac{2\pi NT}{60}} \times 100$$

It indicates the amount of energy losses that occur for reasons other than leakage. This includes friction in bearings and between mating parts also energy losses due to fluid turbulence.

It can also be computed in terms of torque as follows:

$$\eta_m = \frac{\text{Theoretical torque required to operate the pump}}{\text{Actual torque delivered to the pump}} = \frac{T_T}{T_A}$$

Equation to find theoretical torque,

$$\textbf{Theoretical torque of a pump} = T_T = \frac{V_D(m^3) \times P(\frac{N}{m^2})}{2\pi}$$

3. Overall efficiency: It is defined as the ratio of actual power delivered by the pump to actual power delivered to the pump. Overall efficiency considers all energy losses.

$$\text{Overall efficiency } (\eta_o) = \frac{\text{Actual power delivered by the pump}}{\text{Actual power delivered to the pump}} = (\eta_o) = \frac{P\left(\frac{N}{m^2}\right) \times Q_A\left(\frac{m^3}{s}\right)}{\frac{2\pi NT}{60}} \times 100$$

It can be represented mathematically as = $\text{Overall efficiency } (\eta_o) = \eta_v \eta_m$, $\eta_o = \frac{Q_A}{Q_T} \times \frac{pQ_T}{T_A N}$

<u>Slip of the pump</u>

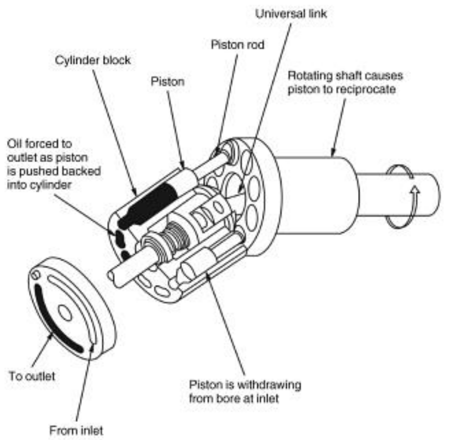
A major effect on positive displacement pump performance is the loss in flow due to slip. However, if slip occurs, the cavity will also be partly filled with fluid flowing back through the pump clearances from the outlet side. Usually slip increases with directly with pressure. Negative slip occurs when delivery pipe is short suction pipe is long and pump is running at high speed.

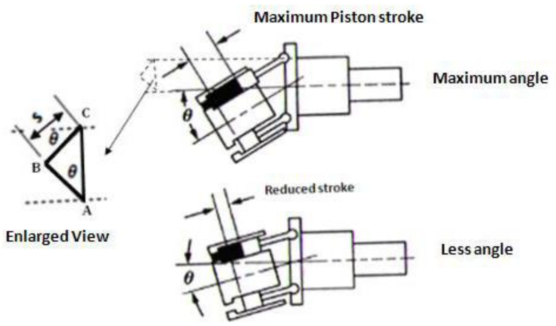
$$\text{Slip} = Q_{th} - Q_{act}$$

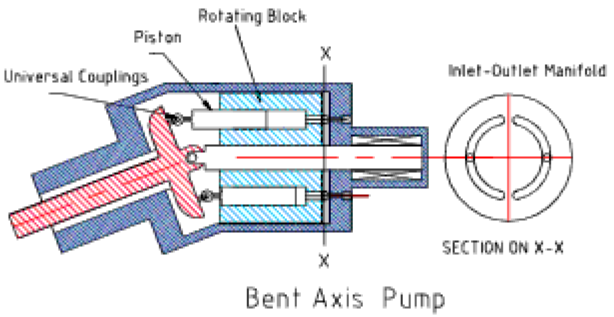
$$\text{Percentage Slip} = \frac{Q_{th} - Q_{act}}{Q_{th}} \times 100$$

Bent-Axis-Type Piston Pump

In this pump the reciprocating action is obtained by bending the axis of the cylinder block. It contains a cylinder block rotating with a drive shaft. However, the centerline of the cylinder block is set at an offset angle relative to the centerline of the drive shaft. A universal link connects the cylinder block to the drive shaft to provide alignment and positive drive. The cylinder block contains a number of pistons arranged along a circle. The piston rods are connected to the drive shaft flange by a ball and socket joints. The pistons are forced in and out of their bores as the distance between the drive shaft flange and cylinder block changes.

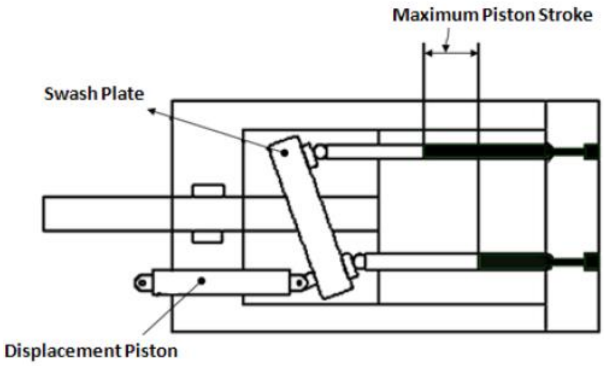


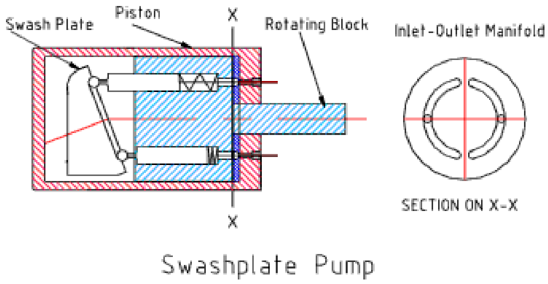




Swash-Plate-Type Piston Pump

In this type, the cylinder block and drive shaft are located on the same centerline. The pistons are connected to a shoe plate that bears against an angled swash plate. As the cylinder rotates, the pistons reciprocate because the piston shoes follow the angled surface of the swash plate. The outlet and inlet ports are located in the valve plate so that the pistons pass the inlet as they are being pulled out and pass the outlet as they are being forced back in. This type of pump can also be designed to have a variable displacement capability. The maximum swash plate angle is limited to 17.5° by construction.





Operation of a swash-plate-type piston pump

Volumetric Displacement and Theoretical Flow Rate of an Axial Piston Pump

Let θ be an offset angle, S the piston stroke in m, D the piston circle diameter, Y the number of pistons, A the piston area inm2, N the piston speed in RPM and Q the theoretical flow rate in m3/min.

From a right-angled triangle ABC Fig.

$$\tan\theta = \frac{\mathrm{BC}}{\mathrm{AB}} = \frac{S}{D}$$

$$\Rightarrow S = D \times \tan\theta \qquad (1.3)$$

The displacement volume of one piston $= AS\,\mathrm{m}^3$

Total displacement volume of Ynumber of pistons $= YAS\,\mathrm{m}^3$

$$V_{\mathrm{D}} = YAS \qquad (1.4)$$

From Eqs. (1.3) and (1.4), we have

$$V_{\mathrm{D}} = YAD\tan\theta\ \mathrm{m}^3/\mathrm{rev} \qquad (1.5)$$

Theoretical flow rate is

$$Q_{\mathrm{T}} = DANY\tan\theta\ \mathrm{m}^3/\mathrm{min}$$

	Pressure (Bar)	**Discharge(LPM)**	**MaximumSpeed (RPM)**	**Overall Efficiency**
Gear pump	20–175	7–570	1800–7000	75–90
Vane pump	20–175	2–950	2000–4000	75–90
Axial piston pump	70–350	2–1700	600–6000	85–95
Radial piston pump	50–250	20–700	600–1800	80–92

2.10 Exercises:

1. A pump of positive displacement type has a mechanical efficiency of 94% and a volumetric efficiency of 92%. What is its overall efficiency?

2. A pump having a 95% volumetric efficiency delivers 25 Lpm of oil at 1200 rpm. What is the volumetric displacement of the pump? (Ans: 2.193 x 10-5 m3)

3. A pump has a displacement of 80 cm3. It delivers 1.25 Lps at 1200 rpm and 75 bar. If the prime mover input torque is 110 N-m, find

i).The overall efficiency,

ii). Mechanical efficiency,

iii). Theoretical torque.

4. How much hydraulic power would a pump produce when operating at 125 bar and delivering 1.25 Lps of oil? What power rated electric motor would be selected to drive this pump if its overall efficiency is 88%. (Ans:15.625 kW & 17.756 kW).

5. A gear pump has a 75mm outside diameter, a 50mm inside diameter, and a 25mm width. If the actual pump flow at 1800rpm and discharge is 0.106 m3/min, what is the volumetric efficiency?

6. A vane pump has a rotor diameter of 50 mm, a cam ring diameter of 75mm, and a vane width of 50 mm. If the eccentricity is 8mm, determine the volumetric displacement.

7. A pump delivers 10 dm^3/min with a pressure rise of 80 bars. The shaft speed is 1420 rev/min and the nominal displacement is 8 cm3/rev. The Torque input is 11.4 Nm. (1 dm=10^{-1}m),(1 dm= 10 cm)

Calculate:

(1) The volumetric efficiency.

(2) The shaft power or mechanical power.

(3) The overall efficiency.

SOLVED EXAMPLES

8. A gear pump has a 75mm outside diameter, a 50mm inside diameter, and a 25mm width. If the volumetric efficiency is 90% at rated pressure, what is the corresponding actual flow-rate? The pump speed is 1000rpm.

$$\text{Volume} = \frac{\pi}{4}(D_o{}^2 - D_i{}^2)L$$

$$= \frac{\pi}{4} \times (0.075^2 - 0.05^2) \times 0.025 = 0.0000614 \text{ m3/rev}$$

$$V_D = 0.0614 \text{ L}$$

$$\text{Actual flow-rate}, Q_A = \eta_v \times Q_T$$

$$= 0.90 \times 0.0000614 \times 1000 = 0.0553 \text{ m3 /min}$$

$$Q_A = 55.3\ Lpm$$

9. A gear pump has an outside diameter of 82.6 mm, inside diameter of 57.2 mm and a width of 25.4 mm. If the actual pump flow is 1800 RPM and the flow rate is 0.00183 m3 /s. what is the volumetric efficiency? (86.11%)

Outside diameter $D_o = 82.6$ mm
Inside diameter $D_i = 57.2$ mm
Width $d = 25.4$ mm
Speed of pump $N = 1800$ RPM
Actual flow rate $= 0.00183\ m^3/s$
Theoretical flow rate

$$Q_T = \frac{\pi}{4} \times (D_o^2 - D_i^2) \times d \times \frac{N}{60}$$

$$= \frac{\pi}{4} \times (0.0826^2 - 0.0572^2) \times 0.0254 \times \frac{1800}{60}$$

$$= 2.125 \times 10^{-3}$$

Volumetric efficiency is

$$\eta_v = \frac{0.00183}{2.125 \times 10^{-3}} \times 100 = 86.11\%$$

10. A pump has a displacement volume of 120 cm^3. It delivers 0.0015 m^3/s at 1440 RPM and 60 bar. If the prime mover input torque is 130 Nm. Find,

i).Volumetric efficiency,

ii) Mechanical efficiency,

iii) Overall efficiency

iv).The theoretical torque required to operate the pump.

Solution: Given volumetric displacement, $V_D = 120\ \text{cm}^3$, $Q_A = 0.0015\ \text{m}^3/\text{s}$, $N = 1440$ rpm, $P = 60$ bar, input torque $T_A = 130$ N m.

Total number of working hours available $= 250 \times 12 = 3000$ h

Volumetric displacement in m^3/rev is

$$V_D = \frac{120\ \text{cm}^3}{\text{rev}} \times \left(\frac{1\ \text{m}}{100\ \text{cm}}\right)^3 = 0.000120\ \text{m}^3/\text{rev}$$

$$Q_T = V_D N = 0.000120 \times \frac{1440}{60}\ \text{rev/s} = 0.00288\ \text{m}^3/\text{s}$$

Now we can calculate the volumetric efficiency as

$$\eta_v = \frac{Q_A}{Q_T} = \frac{0.0015}{0.00288} = 52.08\%$$

Mechanical efficiency is given by

$$\eta_m = \frac{pQ_T}{T_A \omega} = \frac{60 \times 10^5 \times 0.00288}{130 \times 1440 \times \frac{2\pi}{60}} = \frac{17280}{19603} = 88.2\%$$

Note the product $T_A \omega$ gives power in units of Nm/s (W) where T_A has a unit of Nm and shaft speed has units of rad/s.

The overall efficiency is

$$\eta_o = \eta_m \times \eta_v = 0.882 \times 0.5208 = 0.459 = 45.9\%$$

Alternativelyoverall efficiency can also be calculated as

$$\eta_o = \frac{pQ_A}{T_A \omega} = \frac{60 \times 10^5 \times 0.0015}{130 \times 1440 \times \frac{2\pi}{60}} = \frac{9000}{19603} = 45.9\%$$

Now since the mechanical efficiency is known, we can calculate the theoretical torque as

$$T_T = T_A \times \eta_m = 130 \times .882 = 114.7\ \text{N m}$$

Thus, due to mechanical losses within the pump, 130 Nm of torque are required to drive the pump instead of 114.7 Nm.

11. A hydraulic motor has a displacement of 164 cm3 and operates with a pressure of 70 bar and a speed of 2000 rpm. If the actual flow rate consumed by the motor is 0.006 m3 /s and the actual torque delivered by the motor is 170 Nm, find

i).Volumetric efficiency,

ii) Mechanical efficiency,

iii) Overall efficiency

iv). actual power delivered by the motor?

Solution:

(a) We have

$$\eta_v = \frac{\text{Theoretical flow rate the motor should consume}}{\text{Actual flow rate consumed by the motor}} = \frac{Q_T}{Q_A}$$

Now $Q_A = 0.006\ \text{m}^3/\text{s}$. Theoretical flow rate is

$$Q_T = V_D \times N = 164 \times 10^{-6}\ (\text{m}^3/\text{rev}) \times \frac{2000}{60}\ (\text{rev/s}) = 0.0055\ \text{m}^3/\text{s}$$

So volumetric efficiency is

$$\eta_v = \frac{0.0055}{0.006} \times 100 = 91.67\%$$

(b) Mechanical efficiency is given by

$$\eta_m = \frac{\text{Actual torque delivered by the motor}}{\text{Theoretical torque motor should deliver}} = \frac{T_A}{T_T}$$

Theoretical torque,

$$T_T = \frac{p \times V_D}{2\pi} = \frac{70 \times 10^5 \times 164 \times 10^{-6}}{2\pi} = 182.71\ \text{N m}$$

So mechanical efficiency,

$$\eta_m = \frac{170}{182.71} = 93.04\%$$

(c) We have

$$\eta_o = \eta_m \times \eta_v = 0.9304 \times 0.9167 = 0.853 = 85.3\%$$

So overall efficiency is 85.3 %.

(d) Actual power is

$$T_A \omega = 170 \times \left(2000 \times \frac{2 \times \pi}{60}\right) = 35600\ \text{W} = 35.6\ \text{kW}$$

2.11 Efficiencies of a centrifugal pump.

In case of a centrifugal pump, the power is transmitted from the shaft of the electric motor to the shaft of the pump and then to the impeller. From the impeller, the power is given to the water. Thus power is decreasing from the shaft of the pump to the impeller and then to the water. The following are important efficiencies of a centrifugal pump.

(a) Mechanical efficiency (η_{mech})
(b) Hydraulic efficiency ($\eta_{hydraulic}$)
(c) Overall efficiency ($\eta_{overall}$)

Mechanical efficiency (η_{mech}) = $\frac{\text{Power At The Impeller}}{\text{Power At The Shaft}}$

Hydraulic efficiency ($\eta_{hydraulic}$) = $\frac{\text{Hydraulic Power Imparted To Fluid}}{\text{Power At The Impeller}}$

Overall efficiency ($\eta_{overall}$) = $\frac{\text{Hydraulic Power Imparted To Fluid}}{\text{Power At The Shaft}}$

Operating pressure and size ranges for hydraulic pump types

Pump Type	Maximum Pressure		Maximum Delivery (L/min)		Speed(RPM)		Min. Filtration(µm)	Pulsation	Noise Level(dB)	η (%)
	From	To	From	To	From	To				
External gear	40	300	0.25	760	500	3000	100	High	90	70–90
Internal gear	100	210	0.6	740	3000	4000	100	Low	85	75–90
Vane	50	140	6	360	500	3000	50	Low	80	65–80
Balanced vane	140	175	2	620	500	300	50	Low	85	70–90
Axial piston (swash plate)	200	350	1	1450	200	2000	25	High	90	80–90
Axial piston (bent-axis)	250	350	17	3500	200	2000	25	High	90	50–90
Radial piston	350	1720	0.3	1000	200	2000	50	High	90	80–90

2.12 Hydraulic Actuators

Fluid power actuators

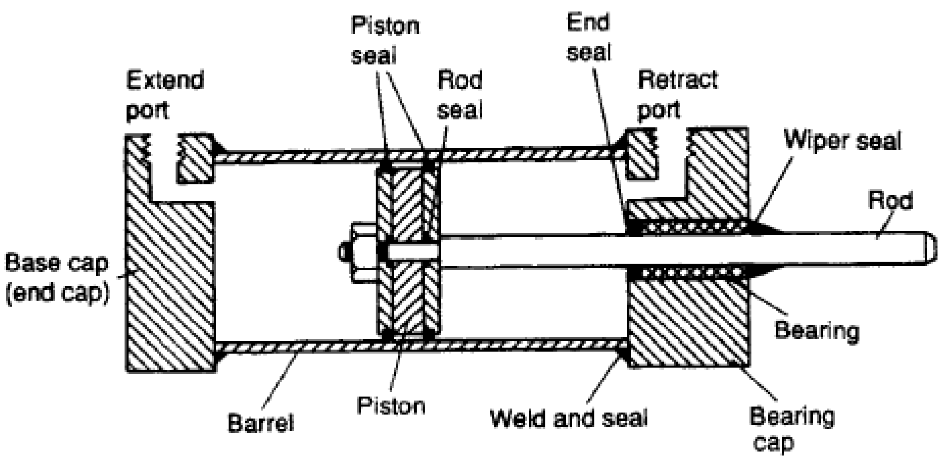
Fluid power actuators receive fluid from a pump driven by an electric motor. After the pressure, flow and direction of the fluid is controlled, the actuator converts its energy into linear or rotary motion to do some useful work. Cylinders account for more than 90% of the actuators used in fluid power systems for work output. The remaining 10% are rotary actuators to produce rotary output.

Construction

Pneumatic and hydraulic linear actuators are constructed in a similar manner, the major differences is in operating pressure (typically 100 bar for hydraulics and 10 bar for Pneumatics). There are five basic parts in a cylinder; two end caps (a base cap and a bearing cap) with port connections, a cylinder barrel, a piston and the rod itself. End caps can be secured to the barrel by welding, tie rods or by threaded connection. The inner surface of the barrel needs to be very smooth to prevent wear and leakage. Generally a **seamless drawn steel tube** is used which is machined (honed) to an accurate finish. In applications where the cylinder is used infrequently or may come into contact with corrosive materials, stainless steel, aluminium or brass tube may be used.

Pistons are usually made of cast iron or steel. The piston not only transmits force to the rod, but must also act as a sliding bearing in the barrel and provide a seal between high and low pressure sides. Piston seals are generally used between piston and barrel. Occasionally small leakage can be tolerated and seals are not used.

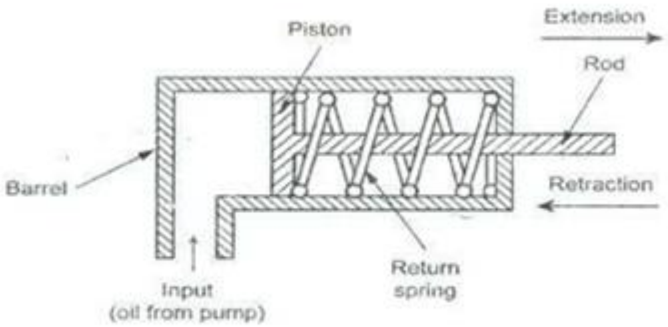
A bearing surface (such as bronze) is deposited on to the piston surface then honed to a finish similar to that of the barrel. The surface of the cylinder rod is exposed to the atmosphere when extended, and hence liable to suffer from the effects of dirt, moisture and corrosion. When retracted, these antisocial materials may be drawn back inside the barrel to cause problems inside the cylinder. **Heat treated chromium alloy steel** is generally used for strength and to reduce effects of corrosion. A wiper or **scraper seal** is fitted to the end cap to remove dust particles.

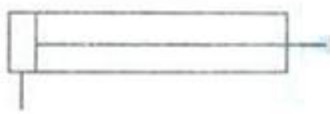


Types of Hydraulic actuators:

Single acting cylinder:

Single acting cylinder has only one port at one end of the cylinder barrel to allow the hydraulic fluid. The figure represents the diagram and symbol for the single acting cylinder. The extending movement of the cylinder is performed by the force applied to the piston by the hydraulic fluid. The piston is retracted (coming back) by the gravity or a compression spring and not by the action of hydraulic fluid. The rod attached to the piston extends out of the cylinder barrel to deliver the force. Single acting cylinders are usually available in short stroke lengths [maximum length up to 80 mm]. The advantage is very simple to operate and compact in size. The disadvantage is, as the cylinders are spring return cannot be used for large stroke lengths.





(b) Symbolic representation

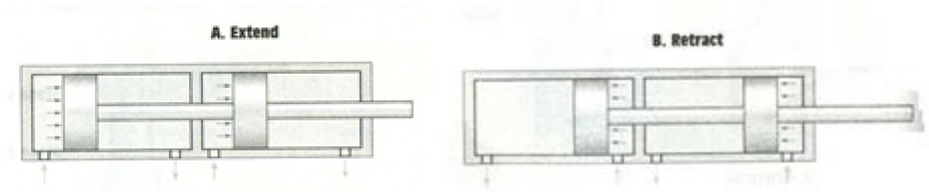
Double acting Cylinders:

In double acting cylinders, the liquid pressure can be applied to either side of the piston there by providing a hydraulic force in both directions. It is mostly used where larger stroke length is needed. During the extension stroke the pressurized fluid enters through the extend port. This fluid moves the piston towards the other end. During this stroke the fluid on the other end is pushed out through the retract port. During the retraction stroke the fluid enters through the retract port and .The fluid on the other end is pushed out of the cylinder through the extend port.

Double acting cylinders are available in diameters from few mm to around 300 mm and stroke lengths of few mm up to 2 meters. In this double acting cylinder the volumes on both the sides of the piston are not same. Volume on the extend port side is more, whereas the volume on the retract port side is less because of the presence of piston rod.Hence more force is transferred during the extending stroke.

Tandem Cylinders:

A tandem cylinder has two or more pistons assembled as a rigid unit with all pistons mounted on single rod. These cylinders are designed to provide a large working area and thus large forces for a given pressure. The tandem cylinder can produce almost twice the force from the same diameter. The two cylinders can be independently piped or drained to give extra force in one direction or in both directions.



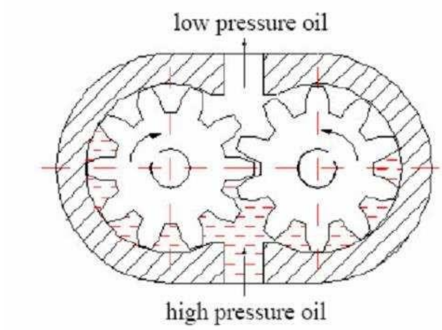
Telescopic Cylinders:

Telescopic cylinders are used where there is space constraint. A telescopic cylinder consists of a series of rams, which provide a long extension. The telescopic cylinder extends in stages, each stage consisting of a sleeve that fits inside the previous stage. They generally consist of a nest of tubes. The tubes are supported by bearing rings, the innermost (rear) set of which have grooves or channels to allow fluid flow. The front bearing assembly on each section includes seals and wiper rings. Stop rings limit the movement of each section, thus preventing separation.

Telescopic cylinders are available in both single-acting and double-acting models. They are more expensive than standard cylinders due to their more complex construction. These ram are retracted by gravity acting on the load or by pressurized fluid acting on the lip. Some of the applications are hydraulic cranes, dumps, trucks etc.

Rotary Actuators:

The function of a rotary actuator is to convert hydraulic energy into rotary mechanical energy. These are equivalent to electric motors; hence rotary actuators are called as hydraulic motors. They work on exactly the reverse principle to that of rotary pumps.



Gear motors are one of the continuous rotary hydraulic motor having fixed displacement. In this type both the gear wheels are driven and one of the wheels has an extended shaft to provide output torque. In the gear motor the rotary motion is produced by the hydraulic forces on the teeth.

2.13 Exercise Problems

1) A single acting reciprocating pump, running at 50 rpm, delivers 0.01 m^3/s of water. The diameter of the piston is 20 cm and stroke length 40 cm. Determine:

i) Theoretical discharge of pump

ii) Volumetric efficiency

ii) Slip and Percentage slip of pump

2) A double acting reciprocating pump, running at 40 rpm, is discharging 1.0 m3 of water per minute. The pump has a stroke of 40 cm. The diameter of the piston is 20 cm. The delivery and suction heads are 20 m and 5 m respectively. Find

i) Slip of the pump

ii) Volumetric efficiency and

iii) Power delivered by the pump

3. Determine the overall efficiency of a pump driven by a 10 HP prime mover if the pump delivers fluid at 40 LPM at a pressure of 10 MPa.

SOLVED EXAMPLES

5. An axial piston pump has nine pistons arranged on a piston of circle 125 mm diameters. The diameter of the piston is 15mm. The cylinder block is set to an off set angle of 10. If pump runs at 1000 RPM with an volumetric efficiency of 94 %. Find the flow rate in LPS.

Theoretical discharge of axial piston pump

$$Q_{T=} \text{DANY } \tan(\theta)$$

$$= 0.125 \times \frac{\pi}{4} \times 0.015^2 \times 9 \times \tan(10) \times 1000$$

$$= 0.03506 \frac{m^3}{\min} = \frac{0.03506}{60} = 5.8433x10^{-4} \frac{m^3}{\sec}$$

Actual discharge, $Q_A = Q_T \times \eta_{vol}$

$$= 5.8433x10^{-4} \text{x } 0.94 = 5.493x10^{-4} \frac{m^3}{\sec}$$

1L =1000cc = 1000 x$10^{-6} m^3 = 1000 m^3$

$5.493 m^3 = \frac{5.493x10^{-4}}{1000}$=0.5493 L

Thus actual discharge = $0.5493 \frac{L}{\sec} \cong 0.55$ LPS

6. A pump having a volumetric efficiency of 96% delivers 29 LPM of oil at 1000 RPM. What is the volumetric displacement of the pump?

Volumetric efficiency of the pump $\eta_v = 96\%$

Discharge of the pump = 29 LPM

Speed of pump $N = 1000$ rpm

Now

$$\eta_v = \frac{\text{Actual flow rate of the pump}}{\text{Theoritical flow rate of the pump}} = \frac{Q_A}{Q_T}$$

$$\Rightarrow 0.96 = \frac{29}{Q_T}$$

$$\Rightarrow Q_T = 30.208 \text{ LPM}$$

Volumetric displacement

$$V_D = \frac{Q_T}{N} = \frac{30.208 \times 10^{-3} \times 60}{60 \times 1000}$$

$$= 30.208 \times 10^{-6} \text{ m}^3 / \text{rev} = 0.0302 \text{ L} / \text{rev}$$

7. A pump has a displacement volume of 98.4 cm3. It delivers 0.00152 m3/s of oil at 1000 RPM and 70 bar. If the prime mover input torque is 124.3 Nm. What is the overall efficiency of pump? What is the theoretical torque required to operate the pump?

Volumetric discharge = 98.4 cm^3

Theoretical discharge is

$$Q_T = V_D \times \frac{N}{60} = 98.4 \times \frac{1000}{60} = 1.64 \times 10^{-3}\ \mathrm{m^3/s}$$

Volumetric efficiency is

$$\eta_v = \frac{1.52 \times 10^{-3}}{1.64 \times 10^{-3}} \times 100 = 92.68\ \%$$

Overall efficiency is

$$\eta_o = \frac{Q_A \times \text{pressure}}{T \times \omega} = \frac{1.52 \times 10^{-3} \times 70 \times 10^5 \times 60}{124.3 \times 2 \times 1000 \times \pi} \times 100 = 81.74\%$$

The mechanical efficiency is

$$\eta_{\text{mechanical}} = \frac{\eta_{\text{overall}}}{\eta_{\text{volumetric}}} = \frac{81.74}{92.78} = 88.2$$

Now

$$\text{Theoretical torque} = \text{Actual torque} \times \eta_{\text{mechanical}} = 124.3 \times 0.882 = 109.6\ \mathrm{Nm}$$

Note: Mechanical efficiency can also be calculated as

$$\eta_m = \frac{pQ_T}{T\omega}$$

$$= \frac{70 \times 10^5\ \mathrm{N/m^2} \times 0.00164\ \mathrm{m^3/s}}{124.3\ (\mathrm{N\,m}) \times \frac{1000}{60} \times 2\pi\ \mathrm{rad/s}}$$

$$= 0.882 = 88.2\%$$

CHAPTER 3: HYDRAULIC VALVES AND COMPONENTS

In this chapter of Fluid Mechanics II you will learn:

- Construction, operation and application of valve
- Symbols used to represent hydraulic system components including valve
- Construct simple hydraulic circuits
- Principle and working of Hydraulic elements

3.1 Valves and Circuits

Hydraulic and pneumatic systems require control valves to direct and regulate the flow of fluid from the pump or compressor to the various load devices like actuators. The operating principles are basically the same for both hydraulic and pneumatic system valves.

In Hydraulics the mechanical power delivered to the load are managed hydraulically by controlling the **pressure**, the **flow rate** or the **direction** of the flow. To accomplish these functions the following valves are used:

Valve according to function:

- Pressure Control Valves (PCVs)
- Directional Control Valves (DCVs direct- and pilot operated)
- Flow Control Valves (FCVs)
- Check valves.

Classification based on actuation mechanism:

Manual actuation

In this type, the spool is operated manually. To get the actuation, they use some mechanisms. Manual actuators are hand lever, push button and pedals etc.

Mechanical actuation

The DCV spool can be operated by using mechanical elements such as roller and cam, roller and plunger and rack and pinion etc. In these arrangements, the spool end is of roller or a pinion gear type. The plunger or cam or rack gear is attached to the actuator. These valves are subjected to wear.

Solenoid actuation

The solenoid actuation is also known as electrical actuation. The energized solenoid coil creates a magnetic force which pulls the armature into the coil. This movement of armature controls the spool position. The main advantage of solenoid actuation is its less switching time. However, electrical solenoids cannot generate large forces unless supplied with large amounts of electrical power. Heat generation poses a threat to extended use of these valves when energized over time

Hydraulic actuation

In this type of actuation, the hydraulic pressure is directly applied on the spool. The pilot port is located on one end of the valve. Fluid entering from pilot port operates against the piston and forces the spool to move forward. The needle valve is used to control the speed of the actuation.

Pneumatic actuation

DCV can also be operated by applying compressed air against a piston at either end of the valve spool. The construction of the system is similar to the hydraulic actuation. Actuation medium is the compressed air in pneumatic actuation system.

Classification based on construction and design:

Poppet valve:

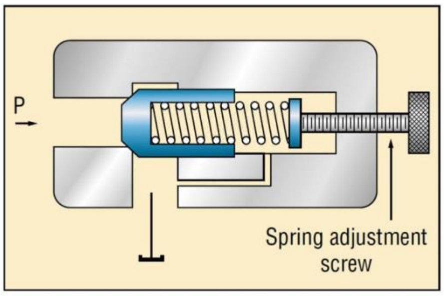
The poppet valve consists of a head, called poppet, a spring and a seat. The poppet may be spherical, conical or other shape. During operation, the valve either closed or opened. That is either poppet seated or unseated. The preset value for the pressure can be determined with the adjusting screw.

Advantages:

1. Low cost
2. Simple construction, easy repair and maintenance.
3. Negligible leakage
4. Less wear on internal seals contributes to a longer product life.
5. The larger internal surface area required by the poppet results in a higher flow rate than spool-style valves.

Disadvantage:

1. Limitation of number of ports
2. Poor controllability.
3. Poppet valves are unbalanced;
4. Back pressure can open the valve if supply pressure is removed. So, not a good choice for holding pressure downstream



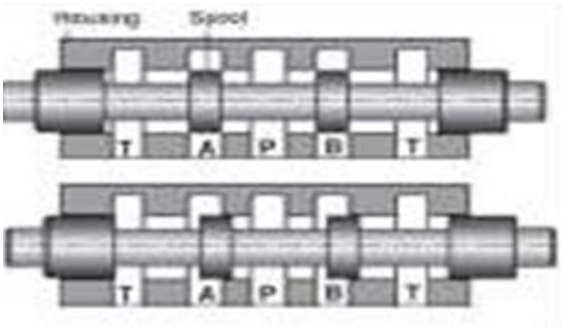
Spool valve (sliding spool):

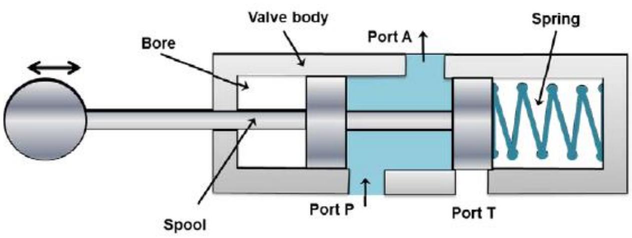
It consists of a shaft sliding in a bore which has large groove around the circumference. The spool mounted in a sleeve. The valve is usually symmetrical and its spool slides axially. The quality of seal or the amount of leakage depends on the amount of clearance, viscosity of fluid and the level of the pressure. Advantages are:

1. Increased number of ports
2. Greater controllability.
3. Spool valves are balanced.
4. Can be used to lock pressure downstream, no back pressure.
5. Not affected by pressure, therefore, less force is required to actuate the valve

Disadvantages:

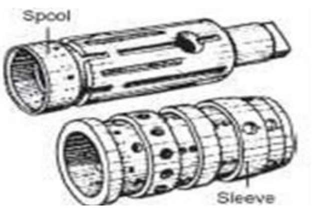
1. High initial cost and current cost
2. Increased leakage
3. Lower flow rate due to the smaller internal surface area required by the spool





Spool valve rotating type:

The valve consists of a spool mounted in a sleeve. As the spool is rotated within the stationary sleeve, the passages or recesses connect or block the ports in the sleeve. The spool rotates inside the sleeve. This class of valve is used in the steering system of some vehicles.

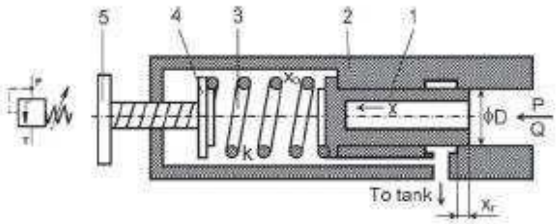


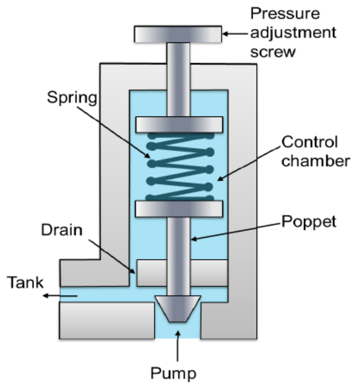
Types of valves based on control:

Although valves are used for many purposes, there are essentially two types of valves. An infinite position valve, which takes up any position between open and close can modulate or change flow rate or pressure. A relief valve or a throttle shutoff valve for example is one of an infinite type. Most control valves, however, are only used to allow or block flow of fluid. Such valves are called finite position valves, like the directional control valve.

3.2 Pressure Control Valves (PCVs)

Pressure Relief Valve



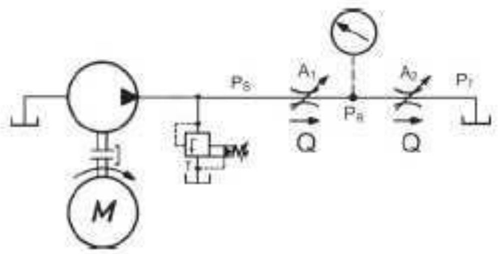


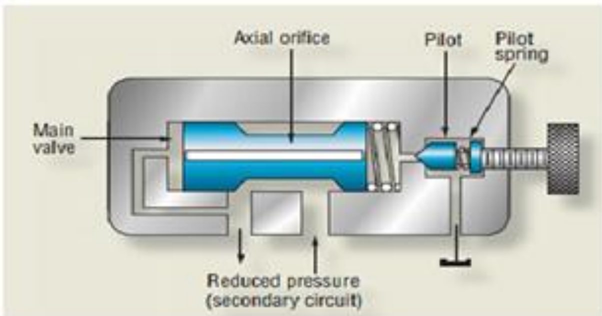
Relief valves are connected with high-pressure and return low pressure lines. They are used to limit the maximum **operating pressure in** the high pressure lines by releasing excess -oil to the tank. Schematic of direct pressure relief valve is shown in figure. This type of valves has two ports; one of which is connected to the pump and another is connected to the tank. It consists of a spring chamber where poppet is placed with a spring force. Generally, the spring is adjustable to set the maximum pressure limit of the system. The poppet is held in position by combined effect of spring force and dead weight of spool. As the pressure exceeds this combined force, the poppet raises and excess fluid bypassed to the reservoir (tank). The poppet again reseats as the pressure drops below the pre-set value. The pressure at which the valve opens is called the *cracking pressure.* The pressure at which the rated flow passes through the valve is termed *full flow pressure.*

A drain is also provided in the control chamber. It sends the fluid collected due to small leakage to the tank and thereby prevents the failure of the valve.

Pressure Reducing Valve

Pressure reducers are used when a subsystem operates at a pressure lower than that of the main system. Generally, the pressure reduction and control is carried out by means of throttling elements. The figure below illustrates the principle of operation of hydraulic pressure reducers. Two throttles are used to connect the reduced-pressure line to the high-pressure line and return (tank) lines. The reduced pressure, Pr, is increased by increasing the area, A1, or decreasing the area, A2, and vice versa.

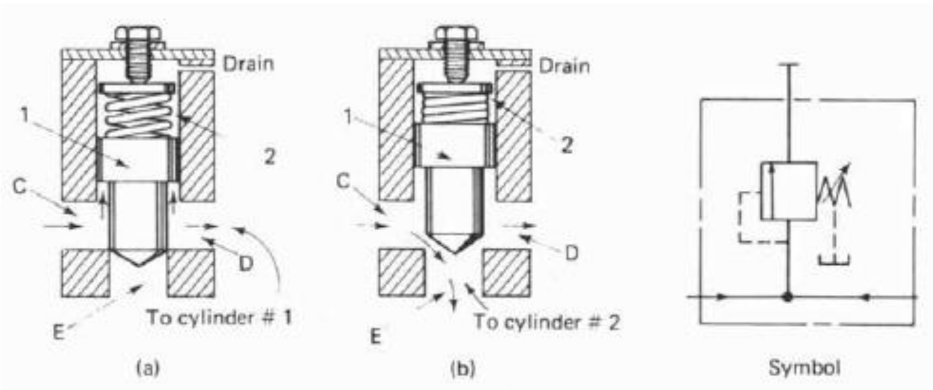




Sequence Valves

The sequence valves are used to create a certain sequence of operations. Sequence valves are also called multifunction valves and are used in various configurations to control sequencing, braking, unloading, load counter balancing, or other functions according to the pressure level in the system. The primary function of this type of valve is to divert flow in a predetermined sequence.

The operating principle of the sequence valve is seen in Fig. In the closed position *A*, fluid passes through the valve from C to *D* at low pressure. When the first step in the sequence has been completed and the clamping cylinder extends and stalls against the work piece, system pressure increases to act against the indicated area of the piston. Continued increase in pressure causes the piston to compress the spring and unseat the valve, thereby directing the flow of fluid at high pressure through port *E*. Fluid pressure is maintained in both branches of the circuit at high pressure so long as the sequence valve is open. Adjustment of the sequence valve is accomplished by compressing or extending the piston spring with the cap screw. If a return flow of fluid from port *E* to C is necessary to retract the second cylinder, a return check valve must be incorporated in the sequence valve.



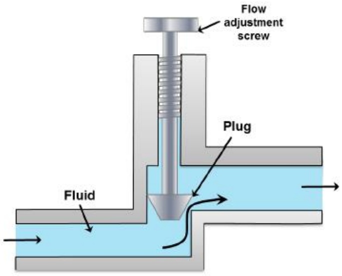
3.3 Flow Control Valves (FCVs)

The speed of actuator can be controlled by regulating the fluid flow. A flow control valve can regulate the flow or pressure of the fluid. The fluid flow is controlled by varying area of the valve opening through which fluid passes. The fluid flow can be decreased by reducing the area of the valve opening and it can be increased by increasing the area of the valve opening.

Types of Flow Control Valves

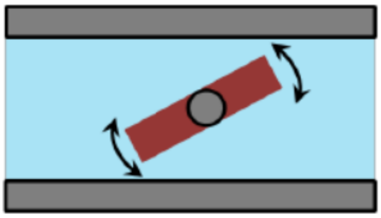
Plug or glove valve

The plug valve is quite commonly used valve. It is also termed as glove valve. This valve has a plug which can be adjusted in vertical direction by setting flow adjustment screw. The adjustment of plug alters the orifice size between plug and valve seat. The characteristics of these valves can be accurately predetermined by machining the taper of the plug. The valve body is made of glass or teflon. The plug can be made of plastic or glass.



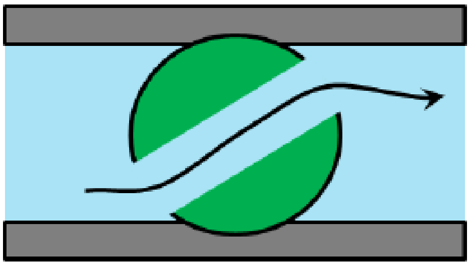
Butterfly valve

It consists of a disc which can rotate inside the pipe. The angle of disc determines the restriction. Butterfly valve can be made to any size and is widely used to control the flow of gas. These valves have many types which have for different pressure ranges and applications.



Ball Valve

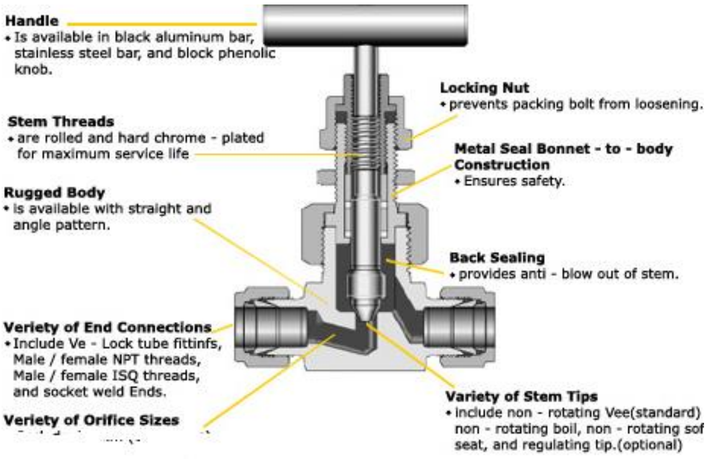
This type of flow control valve uses a ball rotated inside a machined seat. The ball has a through hole. It has very less leakage in its shut-off condition. These valves are durable and usually work perfectly for many years. They are excellent choice for shutoff applications. They do not offer fine control which may be necessary in throttling applications. These valves are widely used in industries because of their versatility, high supporting pressures (up to 1000 bar) and temperatures (up to 250°C). They are easy to repair and operate.



Needle Valve:

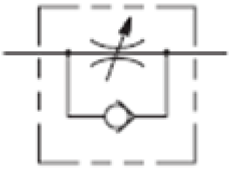
Provide excellent flow control and, depending on design, leak-tight shut-off. They consist of a long stem with highly engineered stem-tip geometry (vee- or needle-shaped) that fits precisely into a seat over the inlet. The stem is finely threaded, enabling precise flow control. Stem packing provides the seal to atmosphere.

Operation - The valve acts as a fixed orifice in a hydraulic circuit. The effective size of the orifice increases as the tapered needle is opened. Shutoff is provided when fully closed. The flow can be controlled and regulated with the use of a spindle. A needle valve has a relatively small orifice with a long, tapered seat, and a needle-shaped plunger on the end of a screw, which exactly fits the seat. Needle valves are usually used in flow-metering applications.



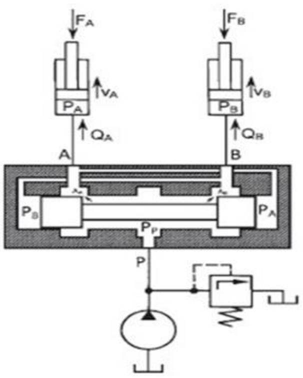
Throttle- check valves or Needle with a Reverse Check:

Throttle valves are used to restrict the fluid flow in both directions while the **throttle- check valves** restrict the flow in one direction only.



Flow divider:

The flow dividers are used to divide the fluid flow rate into two or more parts: either equal parts or by a certain division ratio. This system includes two symmetrical cylinders connected in parallel. The displacement of the cylinders should be synchronized during their extension. Therefore, a flow divider valve is installed. The flow divider acts to divide the pump flow equally between the two cylinders in the extension stroke. The oil flows into the mid-chamber of the spool valve, then through the spool valve restrictions to the cylinders lines (A and B). If the cylinders are equally loaded, then the right and left sides are symmetrical, having equal hydraulic resistance and the main spool is centered.

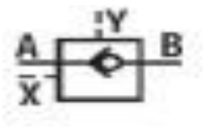


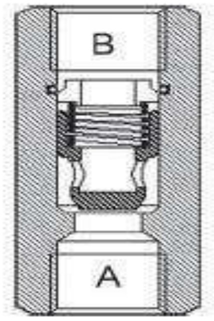
Check valves

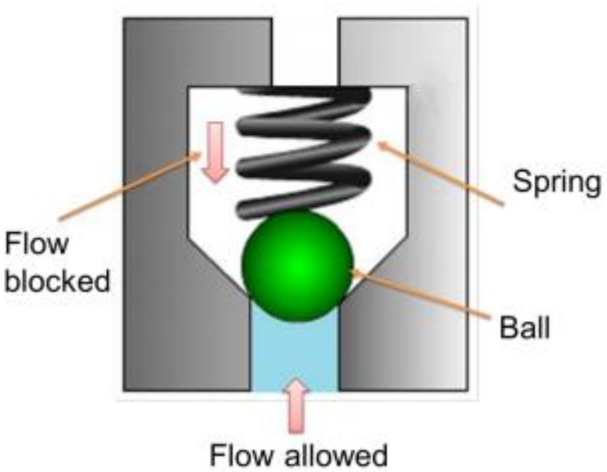
Check valves are generally used to allow for free flow in one direction, and prevent (obstruct) the fluid flow in the opposite direction. The direct-operated check valves consist of a simple poppet valve with a poppet loaded by a spring. The poppet rests against its seat, obstructing the direction from (B) to (A). It allows the fluid flow in the direction (A) to (B) if the pressure difference (PA – PB) is greater than the cracking pressure. Cracking pressure is defined as the pressure difference which produces a pressure force equal to the spring force.

When the fluid entering in the valve it will be pushed in opposite with a spring and poppet. The inline valve has holes around the angled seat face above the body seat to allow flow to pass. The right-angle design pushes the poppet out of the way and fluid flows by with little restriction. The cracking pressure is usually less than 10 bar for the check valves.

It can be in two types Direct operated and pilot operated. Pilot operated (Hydraulically or mechanically piloted). Note that **pilot valve** is a valve used to operate another valve or control. These valves can wear out or can generate the cracks after prolonged usage and therefore they are mostly made of plastics for easy repair and replacements.







3.4 Directional Control Valves (DCVs)

Directional control valves (DCVs) are used to start, stop, or change the direction of fluid flow.

DCV is mainly required for the following purposes:

- To start, stop, accelerate, decelerate and change the direction of fluid flow.
- To permit the free flow from the pump to the reservoir at low pressure when the pump's delivery is not needed into the system.
- To vent the relief valve by either electrical or mechanical control.
- To isolate certain branch of a circuit.

DCVs can be classified as follows:

- An internal valve mechanism like a poppet, a ball, a sliding spool, a rotary plug.
- Number of switching positions (usually 2 or 3).
- Number of connecting ports or ways.
- Method of valve actuation.

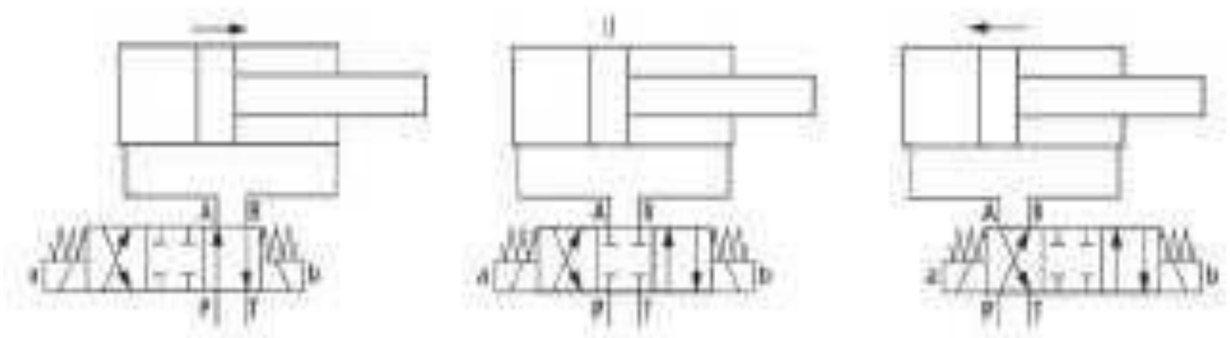
The DCV are classified in different categories.

2/2 dcv is having two ports and two positions.it also classified according to the position.

Normally closed DCV is does not allow the fluid from pump to actuator in its normal position and allows it when it is activated. Normally open DCV is does allows the fluid from pump to actuator in its normal position and does not allows it when it is activated.

3/2 DCV has 3 port and 2 positions. Generally this type solenoid valve are used in single acting cylinder. It can also be in both normally open and normally closed categories.

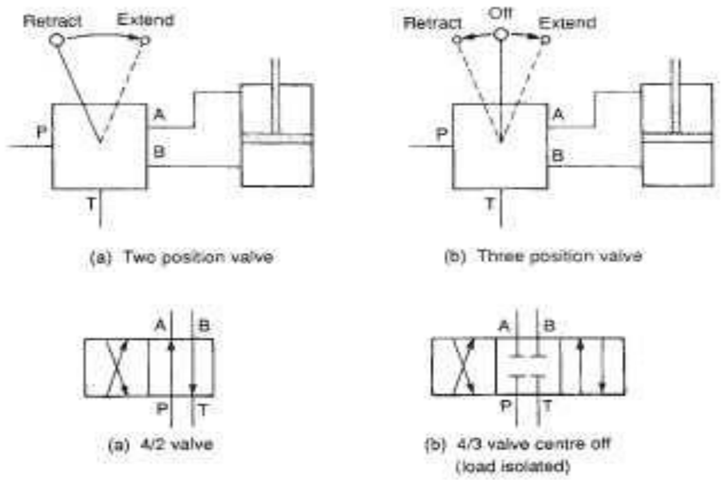
A 4/3 DCV has four ways and three positions. The application of a DCV in controlling the direction of motion of hydraulic cylinders is illustrated below. A 4/3 directional control valve is connected to the pressure line (P), return line (T), and cylinder lines (A and B). In its neutral position, the valve closes all of the four lines and the cylinder is stopped. By switching the valve to any of the other positions, the cylinder moves in the corresponding direction.



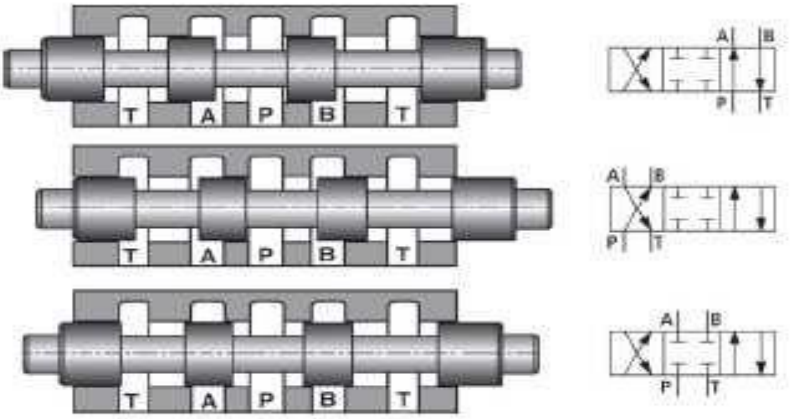
Schematically the above operation can be easily described as follows. Connections to valves are called "ports". A simple on/off valve has therefore two ports. Most control valves has however 4ports as shown below.

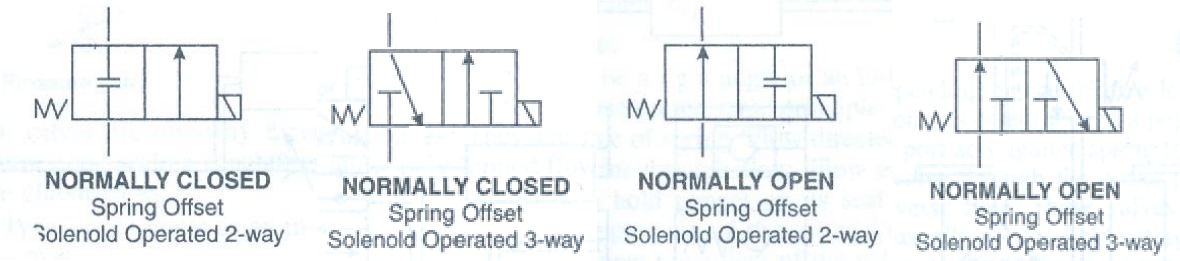
Another consideration is the number of control positions. The figure below shows two possible control schemes. Actuator "a" is controlled by a lever with two positions: extend or retract. This valve has two control positions, whereas actuator "b" has three positions: retract, off, and extend. The valve of actuator "a" is called 4/2 valve (4 port – 2 position valve) and that of "b" is called a 4/3 valve.

Possible position of a 4/3 valve:



(a) Two position valve (b) Three position valve
A B
P T
(a) 4/2 valve (b) 4/3 valve centre off (load isolated)





Hydraulic Components

3.5 Hydraulic Tanks

Also called reservoir. The hydraulic fluid reservoir holds:

1. Excess hydraulic fluid to accommodate volume changes from cylinder extension and contraction, temperature driven expansion and contraction, and leaks.
2. The reservoir is also designed to aid in separation of air from the fluid.
3. It work as a heat accumulator to cover losses in the system when peak power is used.
4. Reservoirs can also help separate dirt and other particulate from the oil, as the particulate will generally settle to the bottom of the tank.

3.6 Hydraulic Accumulator:

A hydraulic accumulator is an energy storage device. It is a pressure storage reservoir in which a non-compressible hydraulic fluid is held under pressure by an external source. That external source can be a spring, a raised weight, or a compressed gas.

Accumulators can provide several functions, such as:

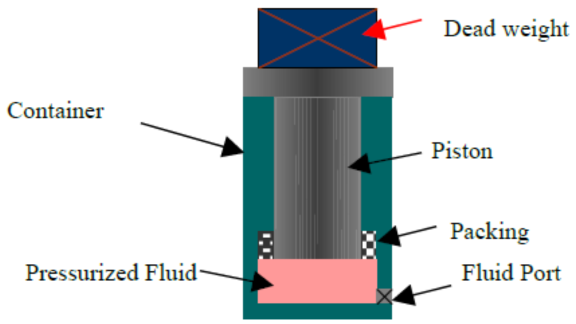
- Energy storage
- Compensation of leakage oil
- Compensation of temperature fluctuations
- Emergency operation
- Cushioning of pressure shocks which may occur at sudden switching of the valves
- Dampening vibrations

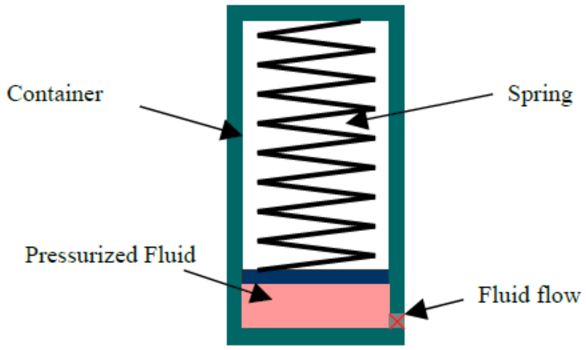
Advantages:

- Possibility of smaller pumps
- Lower installed power
- Less heat produced
- Simple maintenance and installation
- Increased service lifetime
- Immediate availability
- Unlimited storage life

Basic types of accumulator

Weight – Loaded Accumulator: This type consists of a vertical, heavy- wall steel cylinder, which incorporates a piston with packing to pressure leakage. A dead weight is attached to the top of the piston. The force of gravity of the dead weight provides the potential energy in the accumulator. This type of accumulator creates a constant fluid pressure throughout the full volume output of the unit regardless of the rate and quantity of output. The main disadvantage of this type of accumulator is extremely large size and heavy weight which makes it unsuitable for mobile equipment.

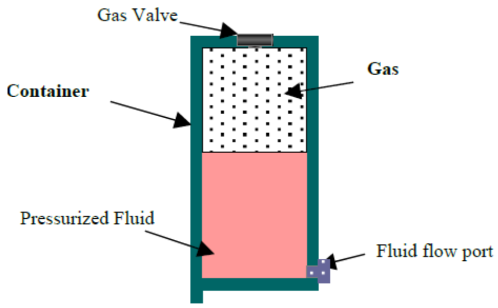




Spring – Loaded Accumulator: A spring loaded accumulator is similar to the weight – loaded type except that the piston is preloaded with a spring as shown in fig.The spring is the source of energy that acts against the piston, forcing the fluid into the hydraulic system. The pressure generated by this type of accumulator depends on the size and pre-loading of the spring.This type of accumulator should not be used for applications requiring high cycle rates because the spring will fatigue and lose its elasticity. The result is an inoperative accumulator.

Gas Loaded Accumulator-Non separator- Type Accumulator:-

The non-separator type of accumulator consists of a fully enclosed shell containing an oil port on the bottom and a gas charging valve on the top. The gas is confined in the top and the oil at the bottom of the shell. There is no physical separator between the gas and oil and thus the gas pushes directly on oil. The main advantage of this type is its ability to handle large volume of oil. The main disadvantage is absorption of gas in the oil due to the lack of a separator. Absorption of gas in the oil also makes the oil compressible, resulting in spongy operation of the hydraulic actuators. This type must be installed vertically to keep the gas confined at the top of the shell.

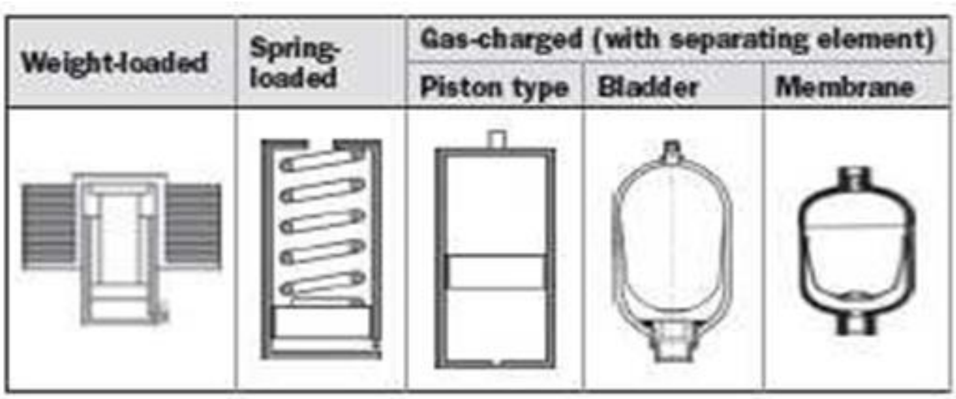


Separator – Type Accumulator: The commonly accepted design of gas loaded accumulators is the separator type. In this type there is a physical barrier between the gas and the oil. The three major type of separator accumulator are

i) **Piston type**: The piston type of accumulator consists of a cylinder containing a freely floating piston with proper seals. The piston serves as a barrier between the gas and oil. The main disadvantages of the piston types of accumulator are that they are expensive to manufacture and have practical size limitation. The principal advantage of the piston accumulator is its ability to handle very high or low temperature system fluids through the utilization to compatible O- ring seals.

ii) **Diaphragm Accumulator:** The diaphragm type accumulator consists of a diaphragm, secured in the shell, which serves as an elastic barrier between the oil and gas. A shutoff button, which is secured at the base of the diaphragm, covers the inlet of the line connection when the diaphragm is fully stretched. The primary advantage of this type of accumulator is its small weight to– volume ratio, which makes it suitable almost exclusively for mobile applications. The restriction is on the deflection of the diaphragm.

iii) **Bladder type Accumulator:** A bladder type- accumulator contains an elastic barrier (bladder) between the oil and gas. The bladder is fitted in the accumulator by means of a vulcanized gas- valve element and can be installed or removed through the shell opening at the poppet valve. Usually the bladder is filled with nitrogen and fitted in a welded or forged steel pressure vessel. The poppet valve closes the inlet when the accumulator bladder is fully expanded. This prevents the bladder from being pressed into the opening. The advantage of this type of accumulator is the positive sealing between the gas and oil chambers. And also they are fast acting. The disadvantage is that Nitrogen will permeate the foam bladder material over time and need to be periodically recharged.



3.7 Hydraulic Filters:

Filters are an important part of hydraulic systems. Metal particles are continually produced by mechanical components and need to be removed along with other contaminants. Common materials used in hydraulic filters are micro-fiberglass, phenolic-impregnated cellulose and polyester.

Filters may be positioned in many locations.

1. The filter may be located between the reservoir and the pump intake. Blockage of the filter will cause cavitation and possibly failure of the pump.
2. Sometimes the filter is located between the pump and the control valves. This arrangement is more expensive, since the filter housing is pressurized, but eliminates cavitation problems and protects the control valve from pump failures.
3. The third common filter location is just before the return line enters the reservoir. This location is relatively insensitive to blockage and does not require a pressurized housing, but contaminants that enter the reservoir from external sources are not filtered until passing through the system at least once.

3.8 Hydraulic Heat Exchangers

Excessive temperature is generated by activities of pumps, actuators, valves etc. Temperature causes the oil to become thin and reduces its lubricating characteristic (viscosity). In a hydraulic heat exchanger heat is dissipated by use of air or water cooler.

Heat exchangers are used in hydraulic systems to maintain the operating temperature of the fluid within specified limits. Nominal operating temperatures range between 120°F and 150°F. Excessive temperatures are caused by extreme environments, such as locations near hot metals in steel mills or foundries, excessive pumping of the hydraulic fluid and high cycle rates. High temperature promote thinning, oxidation, and breakdown of the hydraulic oil ; cause deterioration of seals, pickings ; and cause malfunction or inefficiency in components because required clearances between such parts as valve spools and bodies and pump and motor parts cannot be maintained.

3.9 Hydraulic Seals

Seals are used in hydraulic systems to prevent excessive internal and external leakage and to keep out contamination. Functions are

Prevent leakage, both internal and external, prevent the dust & other particles entering into the system, Maintain pressure, Enhance the service life & reliability of the hydraulic system.

Oil leakage, located anywhere in a hydraulic system, it reduces efficiency and increases power losses, temperature rise, environmental damage and safety hazards.

These leakages are

1. Internal leakage – component clearances between mating parts.
2. External leakage – loss of fluid from the system – improper assembly of pipe fittings

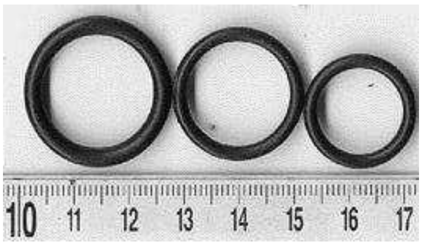
Seals are used in hydraulic systems to prevent excessive internal and external leakage and to keep out contamination. Seals are manufactured from a variety of materials, the choice being determined by the fluid, its operating pressure and the likely temperature range.

The earliest material was leather and, to a lesser extent, cork but these have been largely superseded by plastic and synthetic rubber materials. Leather cannot operate above 90, which is inadequate for many hydraulic systems. Natural rubber cannot be used in hydraulic systems as it tends to swell and perish in the presence of oil.

The earliest synthetic seal material was **neoprene,** but this has a limited temperature range (below 65^0C). It is unsuitable at high temperature because it has a tendency to vulcanize. The most common present-day material is **nitrile (Buna-N)** which has a wider temperature range (-50^0C to 100^0C) and is currently the cheapest seal material. **Silicon** has the highest temperature range (-100^0C to +250^0C) but is expensive and tends to tear. In pneumatic systems **Viton** (-20^0C to 190^0C) and **Teflon** (-80^0C to +200^0C) are the most common materials. These are more rigid and are often used as wiper or scraper seals on cylinders.

Types of seal configurations:

1. O-rings,
2. compression packaging,
3. piston cup packing,
4. piston rings and
5. wiper rings



Static seals and gaskets are used between mating surfaces where no relative movement occurs. The seal is usually compressed between the two adjacent parts by bolt s securing the two stationary parts together. Dynamic seals are used between the surfaces of part s where movement occurs and control both leakage and lubrication. Provision is made to keep dirt and other foreign matter from entering the system by the use of covering boots or rod wipers. The motion encountered by dynamic seal s is reciprocating or rotating, or a combination of both, oscillating.

CHAPTER 4: HYDRAULIC FLUIDS AND CIRCUITS

In this chapter of Fluid Mechanics II you will learn:

- Classification of hydraulic fluids
- Characteristic of hydraulic fluids
- Desirable properties of hydraulic fluids
- Different Symbols used to represent hydraulic system
- Construct simple hydraulic circuits

4.1 Introduction

Hydraulic fluids are used in hydrostatic power systems to transmit power. The power transmission is carried out by increasing, mainly, the pressure energy of the fluid. In addition to the *power transmission*, the hydraulic fluids serve to *lubricate* the contact surfaces, *cool* different elements, and *clean* the system. Water was the first fluid used for the transmission of fluid power. The main advantages of water as a hydraulic fluid are its availability, low cost, and fire resistance. On the other hand, water is of poor lubricity, has a narrow range of working temperature, and has a high rust-promoting tendency. These disadvantages limited its use to very special systems.

4.2 Desirable Properties for hydraulic fluids:

Compressibility:

Compressibility measures the relative change in volume of a fluid or solid as a response to a change in pressure. Hydraulic fluids need to have a very low compressibility. A low compressibility translates into a fast response time, resulting in a high-transmission velocity of pressure and low power loss.

The hydraulic oil compressibility has a direct impact on the transient behavior of the hydraulic system. Generally, the reduction of oil volume by 1% requires an increase of its pressure by 10 to 20 MPa.The bulk modulus of pure oil is nearly constant when operating at a certain temperature and pressure. However, when the oil includes bubbles of gases, air, or vapors, the bulk modulus of this mixture decreases due to the high compressibility of gases.

Viscosity:

Viscosity is the measure of resistance of fluid flow that is inverse measure of fluidity. It is important to keep the oil viscosity within a certain range during the system’s operation; otherwise, the operating conditions will change with temperature. The viscosity index (VI) of oil is a number used in industry to indicate the effect of temperature variation on the viscosity of the oil. A low VI signifies a large change of viscosity with temperature variation. On the other hand, a high VI means relatively little change in viscosity over a wide temperature range. The best one is that maintains constant viscosity throughout temperature changes.

The effect of oil pressure on the viscosity is much less than that of temperature. The viscosity of fluids increases as its pressure increases. However, the hydraulic fluid must not be so viscous at low temperature that it cannot be pumped. A fluid that is too thin also leads to rapid wear of moving parts or of parts that have heavy loads. The instruments used to measure the viscosity of a liquid are known as viscometers or viscosimeters.

Foaming

When foam is carried by a fluid, it degrades system performance and therefore should be eliminated. Foam usually can be prevented by eliminating air leaks within the system. However, two general types of foam still occur frequently: Surface foam, which usually collects on the fluid surface in a reservoir, and Entrained air. Surface foam is the easiest to eliminate, with defoaming additives or by proper sump design so that foam enters the sump and has time to dissipate. Entrained air can cause more serious problems because this foam is drawn into the system. In worst cases, it causes cavitation, a hammering action that can destroy parts. Entrained air is usually prevented by properly selecting the additive and base oils.

Fire Resistance

There are many hazardous applications where human safety requires the use of a fire-resistant fluid. Examples include coal mines, hot metal processing equipment, aircraft and marine fluid power systems. A fire-resisting fluid is one that can be ignited but does not support combustion when the ignition source is removed. Flammability is defined as the ease of ignition and ability to propagate the flame.

The following are the usual characteristics tested in order to determine the flammability of fluids:

1. **Flash point:** The temperature at which an oil surface gives off sufficient vapors to ignite when a flame is passed over the surface. A high flash point is desirable for hydraulic liquids because it indicates good resistance to combustion and a low degree of evaporation at normal temperatures.
2. **Fire point:** The temperature at which oil releases sufficient vapors to support combustion continuously for 5 s when a flame is passed over the surface.
3. **Autogenously ignition temperature:** The temperature at which ignition occurs spontaneously.

The density

The density is the mass per unit volume: $\rho = m/V$. The hydraulic oils are of low compressibility and volumetric thermal expansion. Therefore, under ordinary operating conditions, the oil density is practically constant. The density of mineral hydraulic oils ranges from 850 to 900 kg/m3. The oil density affects both the transient and steady state operations of the hydraulic systems.

Oxidation Stability

The oxidation stability is the ability of the fluid to resist chemical degradation by reaction with atmospheric oxygen. It is an extremely important particularly in high-temperature applications. The degradation of hydraulic fluids by oxidation can result in significant viscosity increases, development of corrosive organic acids, sludge, and varnish. Acids can attack system parts, particularly soft metals.

Thermal stability

Extended high-temperature operation and thermal cycling also encourage the formation of fluid decomposition products. The system should be designed to minimize these thermal problems, and the fluid should have additives that exhibit good thermal stability, inhibit oxidation, and neutralize acids as they form. The *pour point* of a fluid is the temperature 3° C above the temperature at which the fluid ceases to flow. As a general rule, the minimum temperature at which a fluid operates should be at least 10° C above the pour point.

4.3 Typically used hydraulic fluids

Mineral-based oils

These are the most widely used hydraulic fluids. They are chemically stable for reasonable operating temperatures. At higher temperatures, however, they suffer chemical breakdown. Premium grade mineral oils contain a package of additives to combat the effects of wear, oxidation, and foam formation, and to improve viscosity index and lubricity. The disadvantages are the flammability and the increase in viscosity at high pressures. The viscosity pressure characteristics limit use to pressures below 1000 bar.

Advantages:

1. They are relatively inexpensive and widely available.
2. Can be offered in suitable viscosity grades.
3. They are of good lubricity and noncorrosive
4. Compatible with most sealing materials with the exception of butyl rubber

Oil-in-Water Emulsion

This hydraulic fluid consists of tiny droplets of oil dispersed in a continuous water phase. The dilution is normally between 2% and 5% oil in water, and the characteristics of the fluid are more similar to water than oil. It is extremely fire-resistant, is highly incompressible, and has good cooling properties. Its main disadvantages are poor lubricity and low viscosity.

Water-in-Oil Emulsion

The water-in-oil emulsions are the most popular fire-resistant fluids. They have a continuous oil phase in which tiny droplets of water are dispersed. The usual dilution is 60% oil + 40% water. For optimum life, the operating temperatures should not exceed 25° C, but intermittent operation up to 50° C is permissible. At the higher temperature, water content is affected owing to evaporation, which decreases the emulsion's fire-resistance properties. When the system has been idle for long periods, there is a tendency for the oil and water to separate. However, during running, the pump will re-emulsify the fluid.

Water-Glycol Fluids

These fluids were developed primarily for use in aircraft because of their very low flammability characteristics. It consists of a solution of water, ethylene or diethylene glycol, polyglycol and an additive package. The additive package imparts corrosion résistance, oxidation resistance, and antiwear properties.

Synthetic Oils

Synthetic oils are the artificially made chemical compounds instead of crude oil and esters are formed by reacting oxoacid with hydroxil like the alcohol. Esters are derived from inorganic or organic acids. These are used in extreme temperatures because it has superior mechanical and chemical properties compared to mineral oil. They are used in industries such as plastic molding and die-casting, where unusually great fire risks occur. Their lubricating ability is similar to that of mineral oil.

Advantages over mineral-oil–based fluids:

1.Good thermal stability

2.Good oxidation stability

3.Good Viscosity-temperature properties (VI)

4.Low temperature fluidity

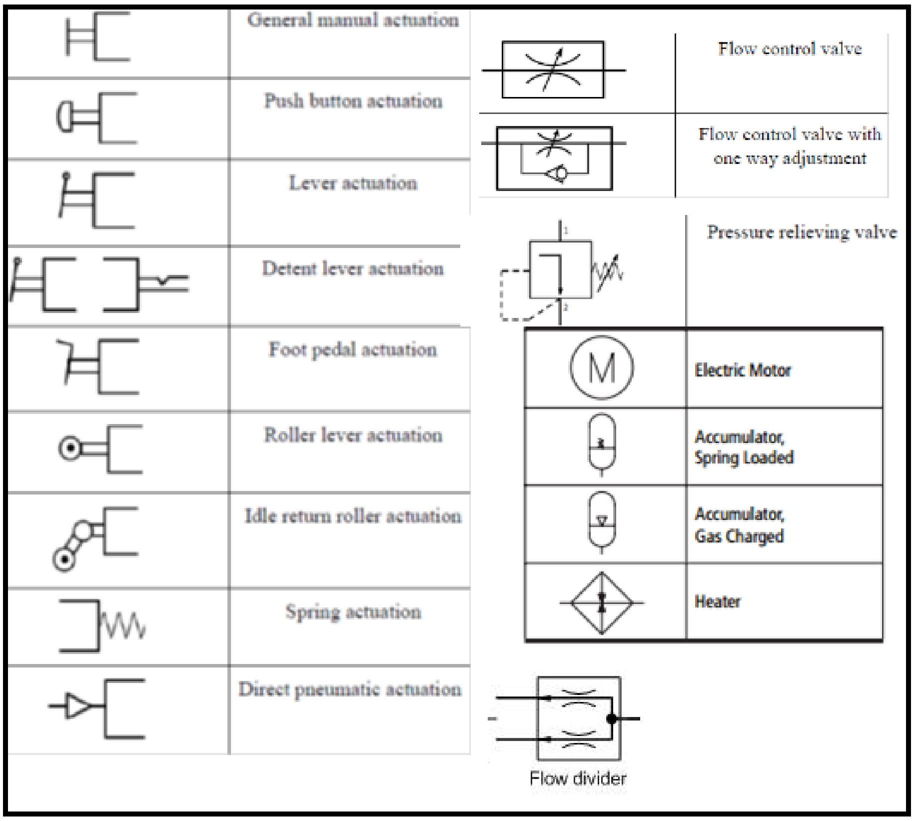
5.Operational temperature limits

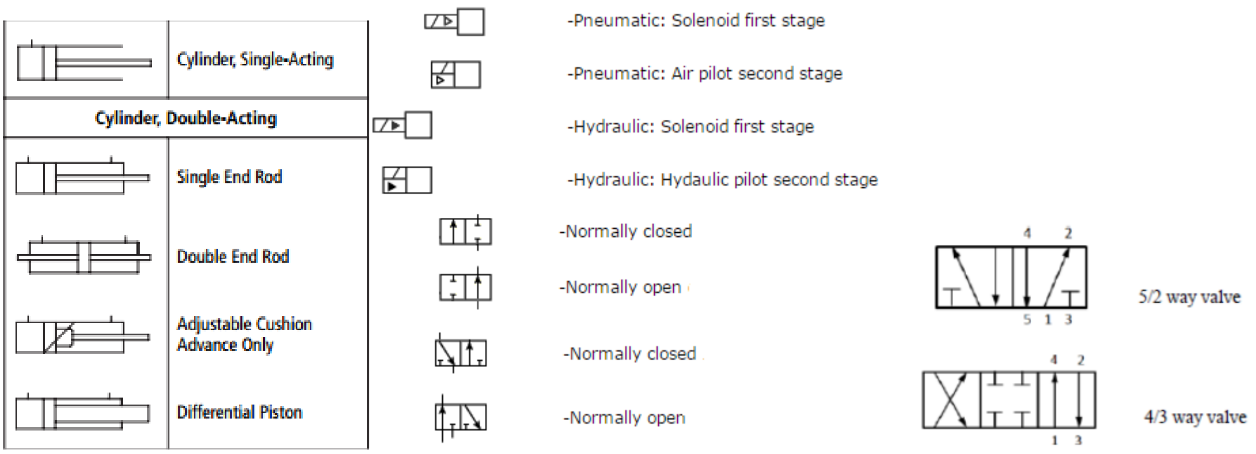
6.They have good fire resistance properties.

4.4 Hydraulic Circuits

Standard symbols allow fluid power schematic diagrams to be read and understood by persons in many different countries, even when they don't speak the same language. However, many companies today use the ISO or ANSI symbols as their standard for work with foreign suppliers and customers.

Basic Schematic Symbols Chart



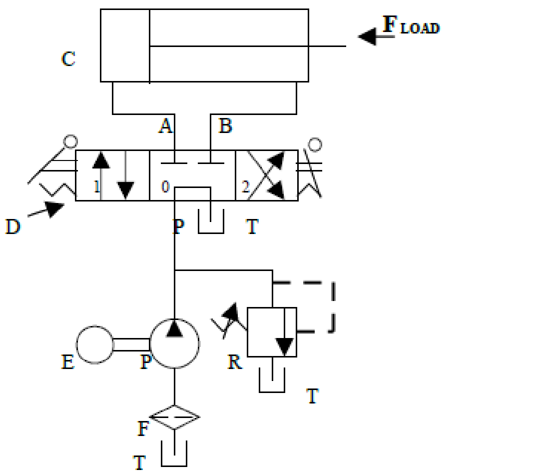


4.5 Double – acting hydraulic cylinder.

When the four ways valve is in centered configuration, the cylinder is hydraulically locked as the ports A and B is blocked. When the four way valve is actuated into the 1st position, the cylinder is extended against its load force F as oil flows from port P through port A. Also, oil in the rod end of the cylinder is free to flow back to the tank via the four way valve from port B through port T.

In the 2st position , the cylinder is retracts against as oil flows to the rod end of the cylinder from port P through port B. Oil in the blank end of the cylinder is returned to the tank from port A to T.

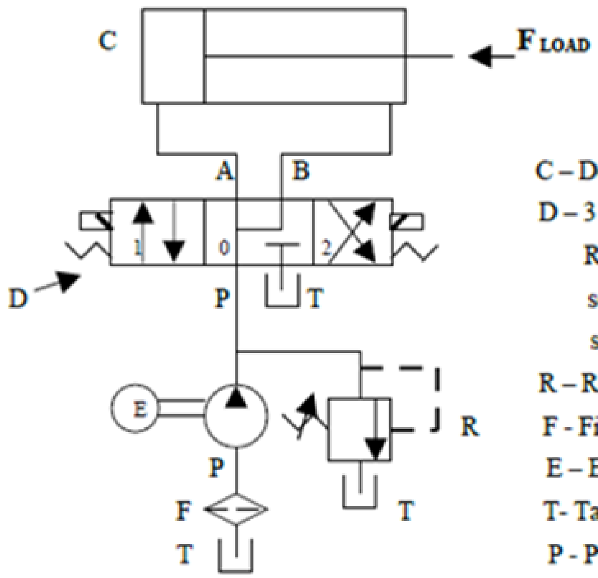
At the end of the stroke, there is no system demand for oil. Thus, the pump flow goes through the relief valve at its pressure- level setting unless the four- way valve is deactivated. In any event the system is protected from any cylinder overloads.



Control of Double -acting hydraulic cylinder.

C = Double acting cylinder
P = Pump
E = Electric Motor
T = Tank
F = Filter
R = Relief Valve
D =3-position, 4 way ,Tandem center, Manually operated and
Spring Centered DCV

4.6 Regenerative center in drilling machine.



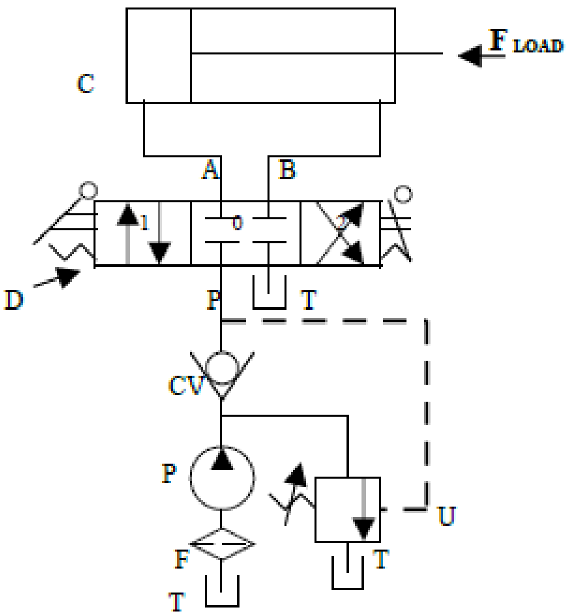
C – Double acting cylinder
D – 3 Position, 4 Way,
Regenerative center,
solenoid actuated,
spring centered DCV
R – Relief Valve
F - Filter
E – Electric Motor
T- Tank
P - Pump

Here a 3-position, 4-way, regenerative center directional control valve is used. When the DCV is in the spring-centered position, port P is connected to A and B and tank port T is blocked. In this position pump flow goes to A and flow from rod end of the cylinder also joins the pump flow to gives rapid spindle advance. (no work is done during this period).

The Oil from the rod end regenerates with the pump flow going to the blank end. This effectively increases pump flow to the blank end of the cylinder during the spring-centered mode of operation. The blank and rod ends are connected in parallel during the extending stroke of a regenerative center. When the DCV shifts to 1st position, P is connected to A and B to T gives slow feed (extension) when the drill starts to cut into the work piece. Similarly when the DCV shifts to 2nd position, P is connected to B and A is connected to T, since the ring area is less the cylinder will have fast return.

The speed of extension is greater than that for a regular double-acting cylinder because flow from the rod end (QR) regenerates with the pump flow (QP) to provide a total flow rate (QT), which is greater than the pump flow rate to the A side of the cylinder. (Area of blank end is more than rod end, thereby blank end provide least resistance)

4.7 Pump – unloading circuit:



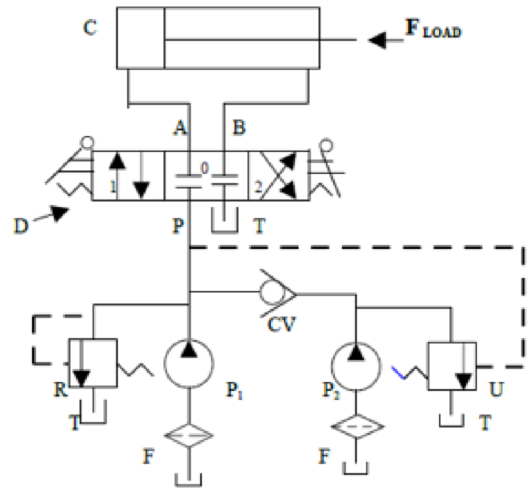
D =3-position, 4 way ,closed center, Manually operated and Spring Centered DCV

The unloading valve opens when the cylinder reaches the end of its extension stroke because the check valve keeps high-pressure oil in the pilot line of the unloading valve. When the DCV is shifted to retract the cylinder, the motion of the piston reduces the pressure in the pilot line of the unloading valve. This resets the unloading valve until the cylinder is fully retracted, at which point the unloading valve unloads the pump. Thus, the unloading valve unloads the pump at the ends of the extending and retraction strokes as well as in the spring-centered position of the DCV.

4.8 Double Pump Hydraulic system (High – Low circuit)

Figure shows a circuit that uses two pumps, one high-pressure, low-flow pump and the other low-pressure, high-flow pump. One can find application in a punch press in which the hydraulic ram must extend rapidly over a large distance with very low pressure but high flow requirements. However, during the short motion portion when the punching operation occurs, the pressure requirements are high due to the punching load. Since the cylinder travel is small during the punching operation, the flow-rate requirements are also low.

The circuit shown eliminates the necessity of having a very expensive high-pressure, high-flow pump. When the punching operation begins, the increased pressure opens the unloading valve to unload the low-pressure pump. The purpose of the relief valve is to protect the high-pressure pump from overpressure at the end of the cylinder stroke. The check valve protects the low-pressure pump from high pressure, which occurs during the punching operation, at the ends of the cylinder stroke, and when the DCV is in its spring-centered mode.

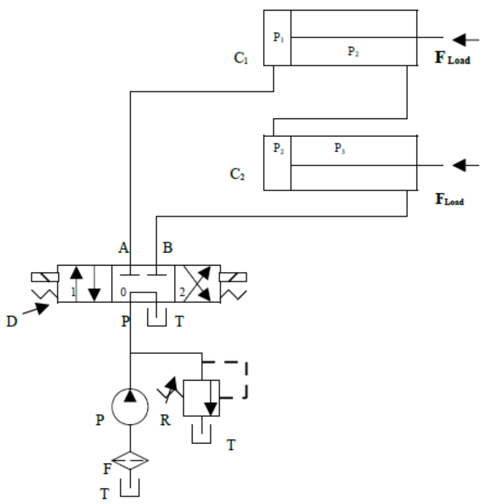


P_1 -- Low discharge, High pressure pump
P_2 -- High discharge, Low pressure pump
R -- Relief valve
U – Unloading valve
T - Tank
CV – Check valve
D – 3 Position, 4 Way, closed center, manual actuated DCV
C- Double acting cylinder
F - Filter

Double pump Hydraulic system

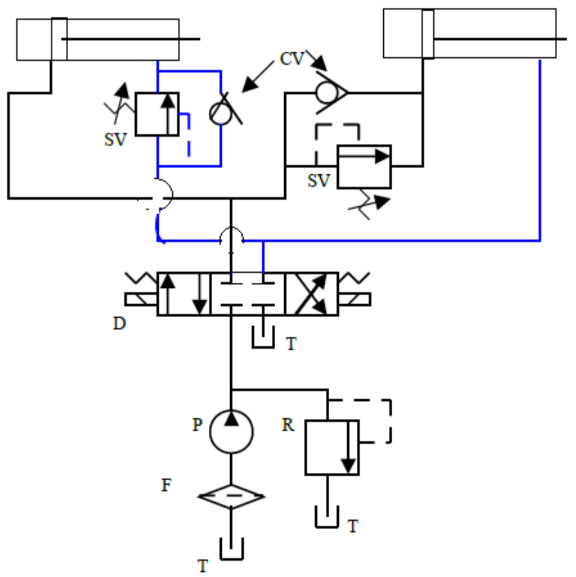
4.9 Cylinders connected in Series:

The circuit of Fig shows a simple way to synchronize two cylinders. Fluid from the pump is delivered to the blank end of cylinder 1, and fluid from the rod end of cylinder 1 is delivered to the blank end of cylinder 2. Fluid returns to the tank from the rod end of cylinder 2 via the DCV. Thus, the cylinders are hooked in series. For the two cylinders to be synchronized, the piston area of cylinder 2 must equal the difference between the areas of the piston and rod for cylinder 1. It should also be noted that the pump must be capable of delivering a pressure equal to that required for the piston of cylinder 1 by itself to overcome the loads acting on both cylinders. It should be noted that the pressure at the blank end of cylinder 1 and the rod end of cylinder 2 are equal as per Pascal's law.



4.10 Hydraulic Cylinder Sequencing circuit:

A sequence valve causes operations in a hydraulic circuit sequentially. Figure is an example where two sequence valves are used C1 and C2. When the DCV is shifted into its 1st position, the left cylinder extends completely, and only when the left cylinder pressure reaches the pressure setting of sequence valve, the valve opens and then the right cylinder extends. If the DCV is then shifted into its 2nd position, the right cylinder retracts fully, and then the left cylinder retracts. Hence this sequence of cylinder operation is controlled by the sequence valves. The spring centered position of the DCV locks both cylinders in place. One can find the application of this circuit in press circuit. For example, the left cylinder the clamping cylinder C1 could extend and clamp a work piece. Then the right cylinder C2, the punching cylinder extends to punch a hole in the work piece. The right cylinder then retracts the punch, and then the left cylinder retracts to declamp the work piece for removal. Obviously these machining operations must occur in the proper sequence as established by the sequence valves in the circuit.



Operation sequence is : $C_1+ C_2+ C_2- C_1-$
+ is Extension of piston
- is Retraction of piston

C_1 & C_2 = Double acting cylinder

P = Pump ; SV = Sequence Valve ;

R = Relief Valve

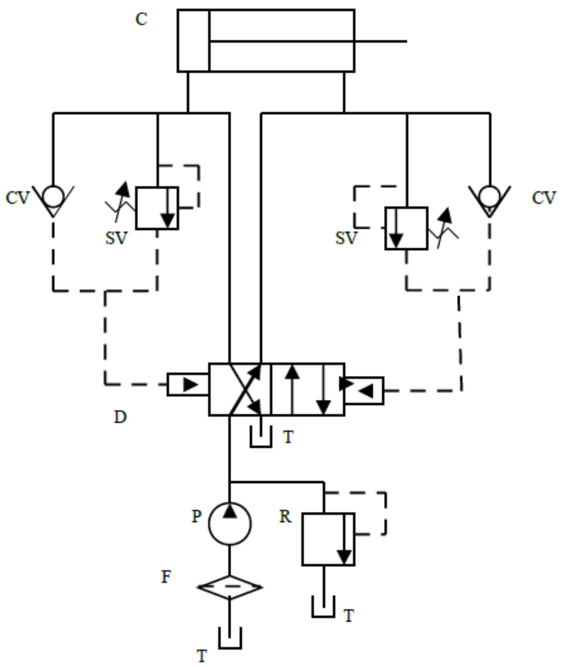
CV = Check Valve ;

T = Tank ; F = Filter

D =3-position, 4 way , Tandem center, Solenoid operated and Spring return DCV

4.11 Automatic Cylinder Reciprocating System:

Figure shows a circuit that produces continuous automatic reciprocation of a hydraulic cylinder. This is accomplished by using two sequence valves, each of which senses a stroke completion by the corresponding buildup of pressure. Each check valve and corresponding pilot line prevents shifting of the four-way valve until the particular stroke of the cylinder has been completed. The check valves are needed to allow pilot oil to leave either end of the DCV while pilot pressure is applied to the opposite end. This permits the spool of the DCV to shift as required.

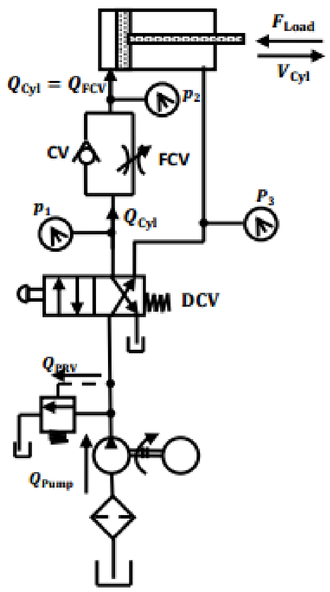


C = Double acting cylinder
P = Pump
SV = Sequence Valve
CV = Check Valve ; R = Relief Valve
T = Tank ; F = Filter
D =2-position, 4 way , pilot operated DCV

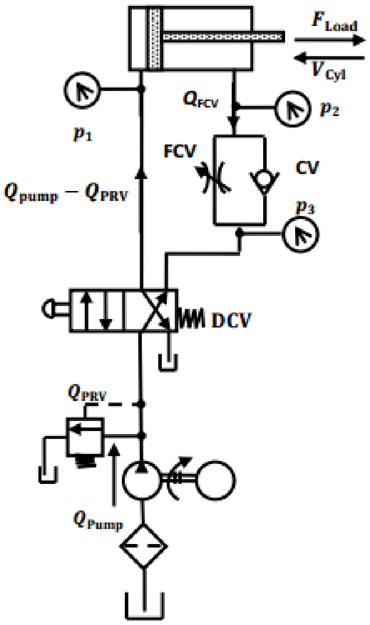
4.12 Speed control of Hydraulic Cylinder:

Speed control of a hydraulic cylinder is accomplished using a flow control valve. A flow control valve regulates the speed of the cylinder by controlling the flow rate to and of the actuator.

Meter – in Circuit: In this type of speed control, the flow control valve is placed between the pump and the actuator. There by, it controls the amount of fluid going into the actuator. The inlet flow into the cylinder is controlled using a flow-control valve. In the return stroke, however, the fluid can bypass the needle valve and flow through the check valve and hence the return speed is not controlled. This implies that the extending speed of the cylinder is controlled whereas the retracing speed is not.



<u>Meter – out Circuit</u>: In this type of speed control, the flow control valve is placed between the actuator and the tank. Thereby, it controls the amount of fluid going out of the actuator. When the cylinder extends, the flow coming from the pump into the cylinder is not controlled directly. However, the flow out of the cylinder is controlled using the flow-control valve (metering orifice). On the other hand, when the cylinder retracts, the flow passes through the check valve unopposed, bypassing the needle valve. Thus, only the speed during the extend stroke is controlled.



One drawback of a meter-out system is the possibility of excessive pressure buildup in the rod end of the cylinder while it is extending. This is due to the magnitude of back pressure that the flow control valve can create depending on its nearness to being fully closed as well as the size of the external load and the piston-to-rod area ratio of the cylinder. In addition an excessive pressure buildup in the rod end of the cylinder results in a large pressure drop across the flow control valve. This produce the undesirable effect of a high heat generation rate with a resulting increase in oil temperature.

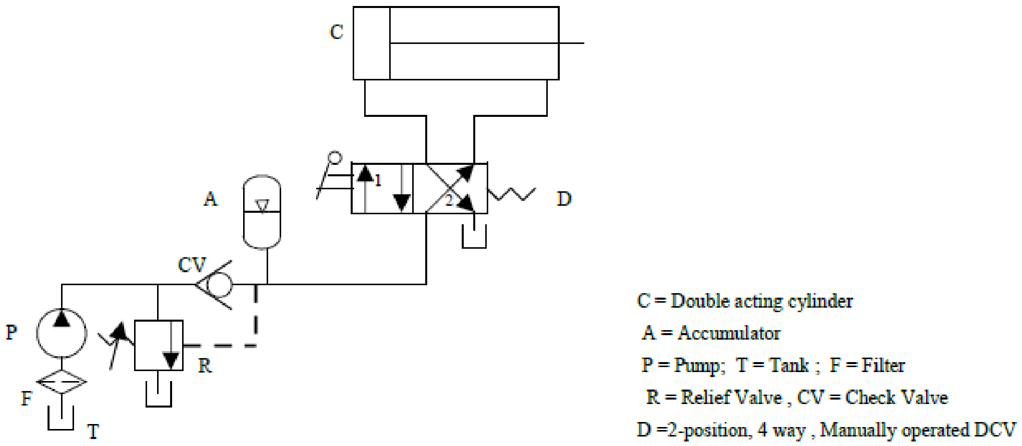
4.13 Accumulator Circuits:

<u>Accumulator as an auxiliary power source:</u>

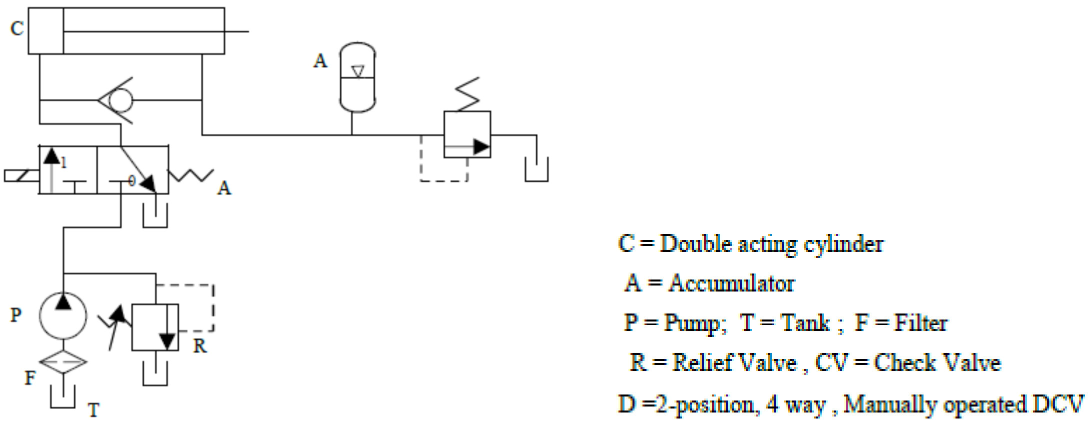
One of the most common applications of accumulator is as an auxiliary power source. The purpose of the accumulator in this application is to store oil delivered by the pump during a portion of the work cycle. The accumulator then releases this stored oil on demand to complete the cycle, thereby serving as a secondary power source to assist the pump. In such a system where intermittent operations are performed, the use of an accumulator results in being able to use a smaller size pump.

Figure shows a four way valve is used in conjunction with an accumulator. When the four way valve is manually actuated, oil flows from the accumulator to the blank end of the cylinder. This extends the piston until it reaches the end of its stroke. While the desired operation is occurring (the cylinder is fully extended position), the accumulator is being charged by the pump.

The four ways is then deactivated for the retraction of the cylinder. Oil from both the pump and accumulator is used to retract the cylinder rapidly. The check valve prevents the back flow of oil from the accumulator when the pump is not working. The control signal for the relief valve is obtained after.

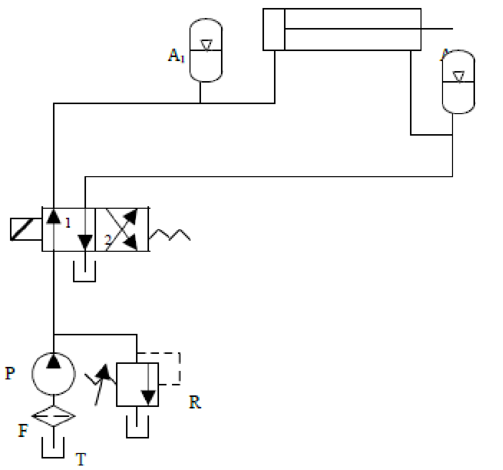


Accumulator as an Emergency power source.



In some hydraulic system, safety dictates that a cylinder be retracted even though the normal supply of oil pressure is lost due to a pump or electrical power failure. Such an application requires the use of an accumulator as an emergency power source. Figure shows such a application in which a solenoid actuated three way valve is used in conjunction with an accumulator. When the three way valve is energized, oil flows to the blank end of the cylinder and also through the check valve into the accumulator and rod end of the cylinder. The accumulator charges as the cylinder extends. If the pump fails due to an electrical failure, the solenoid will deenergize, shifting the valve to the spring -offset position. Then the oil stored under pressure is forced from the accumulator to the rod end of the cylinder. This retracts the cylinder to the starting position. In normal working, when the solenoid is deenergized, the valve shifts to the cylinder.

Accumulator as a hydraulic shock absorber :



C = Double acting cylinder
A_1, A_2 = Accumulators
P = Pump; T = Tank ; F = Filter
R = Relief Valve ,
D = 2/4 Solenoid actuated, spring return DCV

One of the most important industrial applications of accumulator is the elimination or reduction of high pressure pulsation or hydraulic shock Hydraulic shock is caused by the sudden stoppage, sudden impact load, or reversal with heavy loads. Hydraulic shock load may be reduced considerably if the deceleration time of the flowing fluid mass can be reduced. The accumulator should be installed as close to the shock source as possible. Here 2 accumulators are installed near the entry to the cylinder. The oil from the pump flow to the accumulator first and when accumulator is filled, the oil moves to the cylinder and piston starts moving.

Cylinder Extending Speed

The total flow rate entering the blank end (A) of the cylinder equals the pump flow rate plus the regenerative flow rate coming from the rod end of the cylinder:

$$Q_T = Q_P + Q_R$$
$$Q_P = Q_T - Q_R \quad \text{---(1)}$$

We know that the total flow rate equals the piston area multiplied by the extending speed of the piston (Vpext). Similarly, the regenerative flow rate equals the difference of the piston and rod areas (Ap - Ar) multiplied by the extending speed of the piston. Substituting these two relationships into the eq (1) yields

$$Q_P = A_p Vp_{ext} - (Ap - Ar)Vp_{ext}$$
$$\text{Therefore,} \quad Q_P = A_r Vp_{ext}$$

$$\text{Hence the extending speed of the piston, } V_{pext} = \frac{Q_P}{A_r} \quad \text{--- (2)}$$

Thus the extending speed equals the pump flow divided by the rod area. Thus, a small rod area (which produces a large regenerative flow) provides a large extending speed. In fact the extending speed can be greater than the retracting speed if the rod area is made small enough.

Ratio of Extending and Retracting Speeds

Let's determine under what condition the extending and retracting speeds are equal.

We know that the retracting speed (Vpret) equals the pump flow divided by the difference of the piston and rod areas

$$V_{pret} = \frac{Q_p}{A_p - A_r} \qquad \text{---(3)}$$

Dividing eq(1) with (4) we have $\dfrac{Vp_{ext}}{Vp_{ret}} = \dfrac{Qp / A_r}{Qp / (Ap - A_{r})} = \dfrac{A_p - A_r}{A_r}$

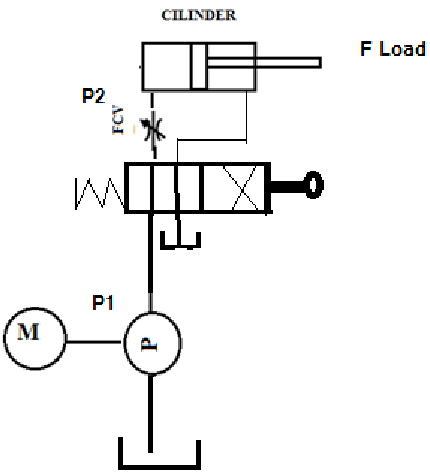
Simplifying we obtain the ratio of extension speed and retracting speed

$$\frac{Vp_{ext}}{Vp_{ret}} = \frac{Ap}{A_r} - 1 \qquad \text{--(4)}$$

We see that when the piston area equals two times the rod area, the extension and retraction speeds are equal. In general, the greater the ratio of piston area to rod area, the greater the ratio of extending speed to retracting speed.

Problem 1: An actuator forward speed is controlled by a meter-in circuit. The pressure setting of relief valve is 50 bar and the pump discharge = 30 litres /min. The cylinder has to carry a load of 3600 N during the forward motion. The area of piston is 15 cm2 and rod area = 8cm2. The flow control valve is set to allow only 10 litres/ min. Calculate the power input to pump, forward speed and return speed and efficiency of the circuit. Pump overall efficiency is 0.85.

Solution:



p2 = F_{load} / A_P = 3600 / 0.0015

= 2400000N/m2 = 24 bar.

Change in **Pressure =** 50-24 = 26 bar

Power input to the pump = p1*Q_P / η_o

=50 *105*0.0005/0.85

=2941watts= **2.94 kW Ans**

Forward speed, v_F = Q_{FCV} / AP

=0.00016 / 0.0015

= **0.16 m/s Ans**

Return speed, v_r = QP / (AP- Ar)

= 0.0005 / (0.0015 -0 .0008)

= **0.71m/s Ans**

Efficiency of circuit = Output / input = p2 QFCV / p1 QP/ η_o

= 24*10 5 *0.00016 / 2.94 kW

The efficiency is the ratio between output power to input power.

The output power can be get by finding mechanical power, P_m = F x V,

Here there is no friction loss so, hydraulic power is equal to mechanical power,

Pm = F x V=P2x Q_{FCV}

Problem 2. A Hydrostatic transmission operating at 70 bar pressure has the following characteristic for the pump and the motor:

Pump : Capacity of pump, $C_P = 82\ cm^3/rev$ (pump displacement)

Volumetric efficiency of pump, $\eta_{VP} = 82\ \%$

Mechanical efficiency of pump, $\eta_{MP} = 88\ \%$

Speed of pump , N = 500 rev / min

Motor: Capacity of motor, $C_M = ?$

Volumetric efficiency of motor, $\eta_{VM} = 92\ \%$

Mechanical efficiency of motor, $\eta_{MM} = 90\ \%$

Desired speed of motor , N = 400 rev / min

Actual Torque, $T_a = ?$

Solution:

Pump theoretical flow rate,

$$Q_P = \text{pump displacement} * \text{speed}$$
$$= 82 * 10^{-06} * 500 / 60$$
$$= 0.00068\ m^3 / sec$$

Actual flow rate to the motor,

$$Q_1 = Q_P * \eta_{VP}$$
$$= 0.00068 * 0.82$$
$$= 0.\ 00056\ m^3 / sec$$

Motor theoretical flow rate,

$$Q_M = Q_1 * \eta_{VM}$$
$$= 0.00056 * 0.92$$
$$= 0.\ 000515\ m^3 / sec$$

- Motor capacity, $C_M = Q_M /$ speed of motor

$$= 0.000515 / 400/60$$
$$= \underline{\mathbf{0.0000773\ m^3 / sec}}$$

Power delivered to motor, P_{hyd} = system pressure $* Q_1$

$$= 70 * 10^5 * 0.00056$$
$$= 3920 \text{ Watts} = 3.92 \text{ kW}$$

Mechanical power generated, $P_{Mech} = P_{hyd} * \eta_{VM} * \eta_{MM}$

$$= 3.92 * 0.92 * 0.90$$
$$= 3.246 \text{ kW}$$

- Actual Torque developed by motor, $T_a = P_{Mech} / 2\pi N$

$$= 3.246 * 1000 / \left(2\pi * \frac{400}{60}\right)$$
$$= \underline{\mathbf{77.49\ N\text{-}m}}$$

CHAPTER 5:PNEUMATICS SYSTEM

In this chapter of Fluid Mechanics II you will learn:

- Characteristics of pneumatics and pneumatic system elements
- Construction and operation of pneumatic system elements

5.1 Introduction

Pneumatic systems are power systems using compressed air as a working medium for the power transmission. Their principle of operation is similar to that of the hydraulic power systems. An air compressor converts the mechanical energy of the prime mover into, mainly, pressure energy of the compressed air. This transformation facilitates the transmission, storage, and control of energy. After compression, the compressed air should be prepared for use. The air preparation includes filtration, cooling, water separation, drying, and adding lubricating oil mist. The compressed air is stored in compressed air reservoirs and transmitted through transmission lines: pipes and hoses. The pneumatic power is controlled by means of a set of valves such as the pressure, flow, and directional control valves. Then, the pressure energy is converted to the required mechanical energy by means of the pneumatic cylinders and motors.

5.2 Characteristics of Pneumatic Systems

The static and dynamic characteristics of the pneumatic systems differ from those of the hydraulic systems due to the difference in the physical properties of the energy transmitting fluid, mainly the **high compressibility**, **low density**, and low viscosity of air.

Effects of Air Compressibility:

The fluid compressibility is the ability of fluid to change its volume due to pressure variation. It is evaluated by the bulk modulus, B, or the compressibility coefficient, β (β = 1/B). The bulk modulus is defined by the following relation.

$$B = \frac{-dp}{dV/V}$$

where
p = Applied pressure, Pa (abs)
V = Fluid volume, m3
B = is the Bulk modulus of air

The negative sign introduced since volume decreases as the pressure increases. At 10 MPa pressure, the air has a bulk modulus Ba= 1.4 × 107 Pa, for n = 1.4. This value is too small, compared with that of the hydraulic liquid (Boil = 1 to 2 GPa).

The Bulk modulus of air is very small compared to that of hydraulic liquid. So even if the air is subjected to high pressure it is much more compressible than liquid. This characteristic of compressibility of air allows us to store energy. The bulk modulus increases with the pressure increase and decreases with the temperature increase.

Characteristic of operations due to the high compressibility of air:

Time Delay of Response

The time delay is the time interval between the moment of opening the control valve and the beginning of motion of the working organ. This delay is caused by the gradual increase of pressure in the transmission lines and actuator chamber. The piston starts to move only when the pressure reaches the value needed to drive the load. The time delay depends upon the volume of the line and actuator chamber, the flow rate of the air, and the loading conditions, including the friction.

The Non-uniform Motion of a Pneumatic Cylinder Piston

The non-uniform motion of a pneumatic cylinder piston is caused by the variable friction in the cylinder and volumetric variation of inlet chamber due to piston displacement.

Pneumatic Systems Are Not Subject to Shocks

Hydraulic shocks result from the rapid change of liquid velocity in transmission lines. This change occurs due to the sudden closure or opening of the line by control valves as well as the sudden stopping of a piston at its end position. In the case of compressible fluids, the sudden closure of valves results in a gradual increase of fluid pressure. Consequently, the fluid speed decrease gradually. Taking into consideration the low air density and high compressibility, the pneumatic transmission lines are not subjected to these shocks.

Pneumatic Cylinders Need a Braking System for Position Locking

The variation of load affects the air pressure and volume in the actuating elements. Therefore, it is difficult to fix any intermediate position of the piston without using a mechanical locking element or by using an efficient electro-pneumatic servo system.

Limited Effect of Fluid Thermal Expansion on the Air Pressure

The pressure variation of a trapped volume of air due to temperature variation is too small compared with that of the hydraulic systems. This small variation of air pressure with temperature variation gives a good advantage to the pneumatic systems, especially in the case of systems subjected to a considerable variation in temperature

The Effect of Air Density (ρ)

Density defined in as the measure of the relative "heaviness" of objects with a constant volume. For gases the density may vary with the number of gas molecules in a constant volume. The air density changes with the pressure and temperature

For T = 288 K and p = 15 MPa, the compressed air density is ρ= 181 kg/m3, and at the atmospheric pressure, the air density is 1.21 kg/m3. Generally, the density of compressed air is much smaller than that of hydraulic liquids (for mineral hydraulic oils. (ρ = 800–900 kg/m3). This small density gives several advantages to the pneumatic systems such as

1. Protection against hydraulic shocks, due to small inertia forces and high compressibility.
2. Reduction of the total weight of the system
3. The air speed in transmission lines is greater than that of liquids for the same pressure difference. Therefore, small line diameters can be used, which lead to an additional reduction of the system weight.

The Effect of Air Viscosity

Viscosity describes the resistance to the laminar movement of two neighboring fluid layers against each other. Simply, viscosity is the resistance to flow. It results from the cohesion and interaction between molecules. Effects of viscosity are:

1. The friction losses in pneumatic transmission lines are very small, which allows reduction of the line diameter of the system.
2. The air is able to leak through the small clearance, mainly due to its small viscosity and density.
3. It is difficult to achieve full tightness of the pneumatic systems.

The dynamic viscosity of compressed air is very small compared with that of hydraulic liquids.at atmospheric temperature and pressure, typical mineral hydraulic fluids have a dynamic viscosity μ oil=$2x10^{-2}$ Pa s. under the same condition, the air viscosity is 2×10^{-5} pa s.

Other Characteristics of Pneumatic Systems

1. After its expansion, the air is expelled into the atmosphere. Therefore, only supply lines are used. There are no return lines.
2. Compressed air reservoirs are of considerable volume and weight.
3. The air is of poor lubricity. Therefore, friction surfaces need special lubrication.
4. The air contains a certain amount of water vapor. After compression and cooling, the vapor condenses. The condensed water should be removed to avoid filling the compressed air reservoir and rust formation. To do this, of air dryers are used.
5. Pneumatic systems are not fire hazards. However air reservoirs have the potential to explode.

5.3 Advantages and Disadvantages of Pneumatic Systems

Basic Advantages of Pneumatic Systems

a) Small weight of transmission lines due to the small diameter of lines.
b) Small weight of transmission lines due to low density of energy transmitting fluid; the air.
c) Small weight since, There are no return lines;
d) Availability of the energy transmission fluid, the air.
e) The system is fireproof.
f) Able to supply a great amount of energy during a short time period.

Basic Disadvantages of Pneumatic Systems

a) Difficult in system tightness.

b) Low working pressure due to the tightness problems and compressor design (within 10 bar for industrial systems and more than 200 bar for aerospace)

c) Difficulty of holding pneumatic actuators at intermediate positions.

d) Delay of response due to the time needed for filling the long lines with compressed air.

e) The variation of pressure in air reservoirs with temperature.

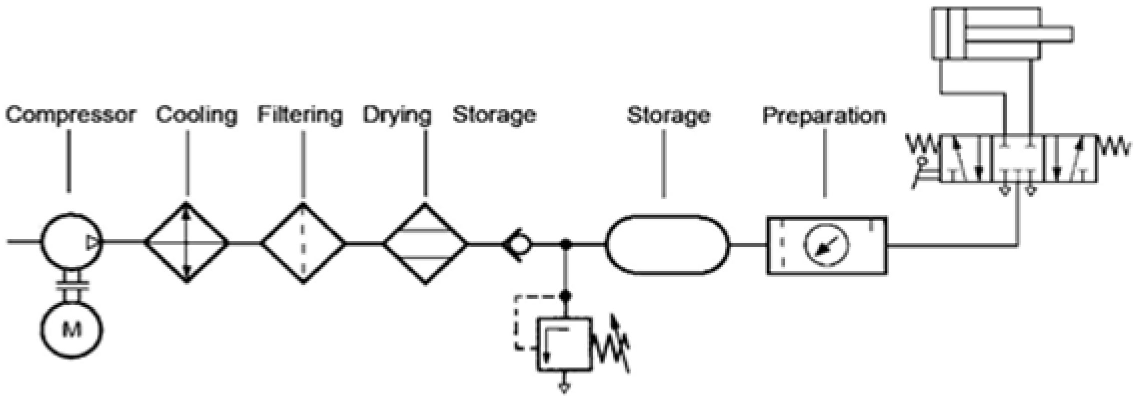
f) The possibility of the condensation of humidity and the freezing at low temperatures.

g) Special lubricators are needed due to the poor lubricity of air.

5.4 Basic Elements of Pneumatic Systems

Basic Pneumatic Circuits

Figure shows the basic circuit of a pneumatic system. The compressed air is prepared by means of the air preparation unit, including the compressor, filters, air drier, compressed air reservoir, cooler, and pressure control elements. The mechanical energy provided by the prime mover is converted by the compressor to, mainly, pressure energy. The compressed air is stored in an air reservoir of sufficient capacity. The maximum pressure at the compressor exit line is limited by a relief valve. The pressure in the air reservoir should be greater than that needed for system operation. Therefore, a pressure reducer is used to control the driving forces and save the compressed air.
Compact air preparation units are commercially available, and are comprised of a pressure reducer, an air filter, a lubricator, and a pressure gauge indicating its exit pressure. Finally, the cylinder is fed by the compressed air by means of a directional control valve.



5.5 Air Compressors

The function of a compressor is to compress gas and delivered it to the user through the piping. The main parameters are volumetric flow, intake pressure, and discharge pressure, rotating speed and compressor shaft power. The different classifications are displacement type and dynamic compressor. In displacement type the pressure increases because of change in volume of air trapped in a confined space. The dynamic type it is due to the acceleration of the moving gas and converting its energy.

Piston Compressors

In the piston class of compressors, the process phases of expansion, suction, compression, and discharge are accomplished by the reciprocating motion of a piston. A functional schematic of a piston type compressor and the associated theoretical P-V diagram are presented in the Figure.

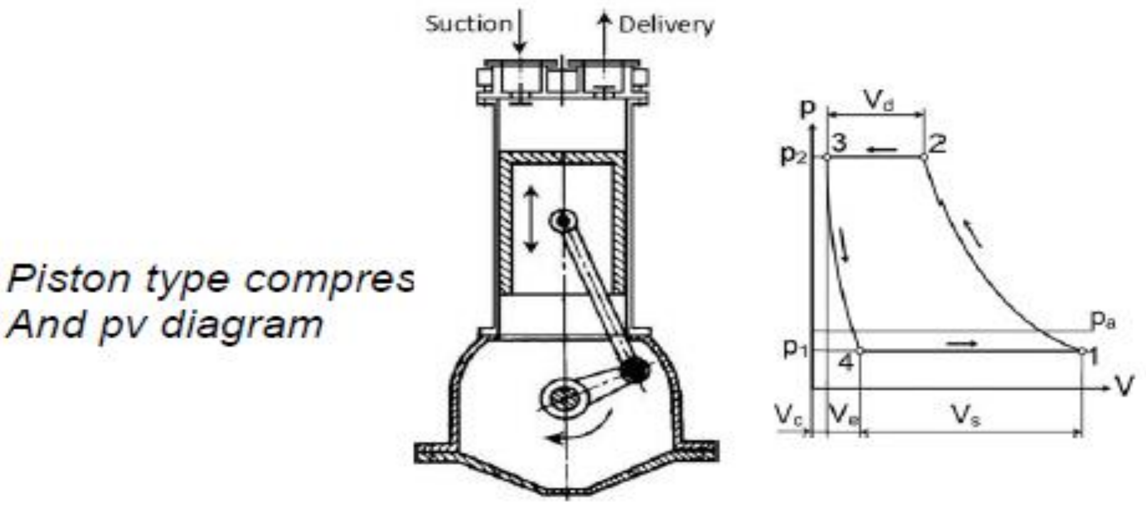
As the piston moves from the bottom dead point to its top dead point, it compresses the gas contained in the cylinder. The inlet valve is closed during the entire compression stroke. The discharge valve remains closed until the pressure in the cylinder overcomes both the load pressure and the exit valve cracking pressure. Then, the discharge valve opens and the piston displaces the gas into the discharge line. In the p-V diagram, the building pressure is represented by the line 1-2 and the gas discharge stroke by line 2-3. If p2 is the pressure in the cylinder during discharge, the volume of gas deliverd by the compressor at this pressure will be Vd.

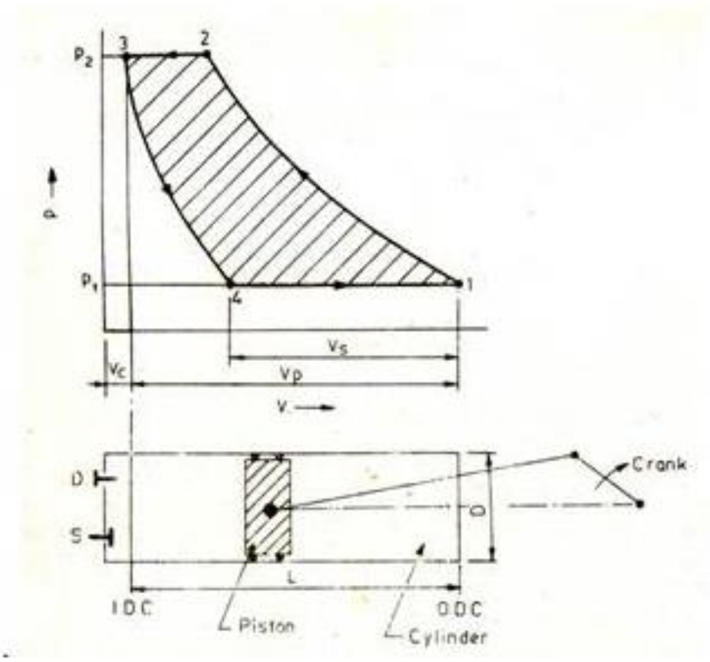
The compression line is a polytrope,given in the P-V diagram by the equation PV^{n}= constant.Theoretically,the discharge line 2-3 is an isobar,p2=constsnt.

The clearence Vc, of the cylinder is the volume of gas present in the cylinder minus the swept volume of the piston at its top dead center.At the beginning of the expansion stroke,the discharge valve will close and the clearence gas will expand along the line 3-4.By the end of the expansion process,this volume occupies the volume (Vc+Ve).

The expansion is polytropic.The gas expands untile pressure in the cylinder lowers to P1$\leq$ Pa,where Pa is the pressure in the compressor intake line.The intake valve will open,against the spring force,under the action of the force due to the pressure difference(Pa-P1).The piston will draw gas in to the cylinder during the expansion stroke.The pressure,P1,is always below Pa due to the fluid resistance in the intake system.

The suction is represented by the isobar 4-1.The resulting closed line1-2-3-4-1 is the theoretical indicator diagram.The actual indicator diagram differs mainly in suction and discharge lines.





Advantages of piston type compressor

1. Piston type compressors are available in wide range of capacity and pressure
2. Very high air pressure (250 bar) and air volume flow rate is possible with multi-staging.
3. Better mechanical balancing is possible by proper cylinder arrangement.
4. High overall efficiency compared to other compressor

Disadvantages of piston type compressor

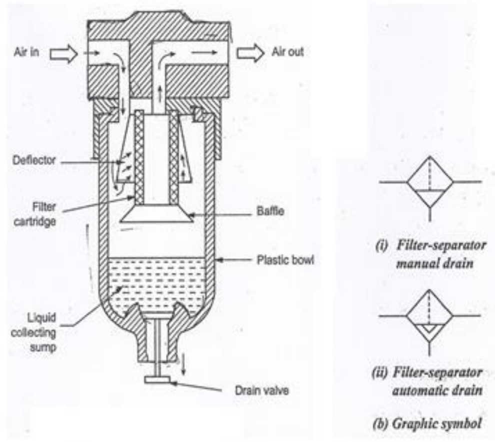
1. Reciprocating piston compressors generate inertia forces that shake the machine. Therefore , a rigid frame, fixed to solid foundation is often required
2. Reciprocating piston machines deliver a pulsating flow of air.
   Properly sized pulsation damping chambers or receiver tanks are required.
3. They are suited for small volumes of air at high pressures.

Pneumatic Reservoirs

Generally, the air compressor serves to charge the compressed air reservoir. The compressor operation can be controlled by a governor to keep the air reservoir pressure within certain limits. The pneumatic system is directly fed from the reservoir. However, in some special cases, the pneumatic system does not include an air compressor, such as in some aircrafts. They use pre-charged high-pressure compressed air bottles as a source of pneumatic energy.

Air Filters

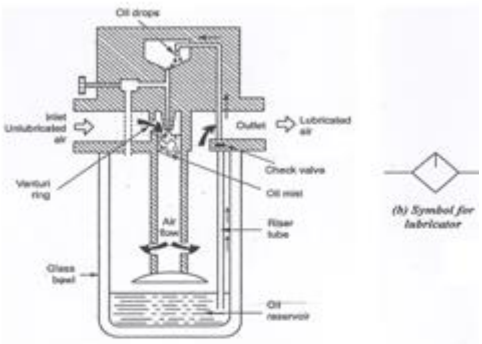
The solid particles and liquid droplets are removed from the compressed air by using air filters. The centrifugal force acting on the solids and water separates them from the air stream. They are then collected in the lower part of the filter. In the second stage, a fine filter is added to separate additional impurities. Condensed water is collected by means of drain valves. Certain air filters come with or without a water collector. Moreover, filters with water traps are drained automatically or manually.



Typical air-filter system

Air Lubricators

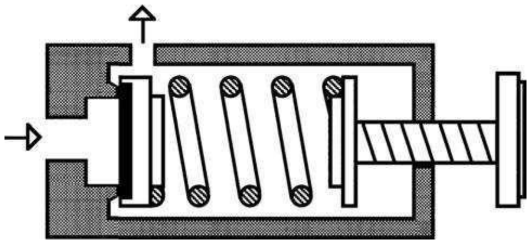
Oil fog lubricators are used to lubricate the compressed air by adding a fine fog of oil. Oil fog lubricators operate, mostly, according to the Venturi principle (see figure below). The reduction of area in the air path produces a vacuum. The oil is drawn up through a narrow pipe which reaches into the lubricating oil reservoir. The oil then drips into the flowing compressed air and forms a fine oil fog.



6.6 Pneumatic Control Valves

Relief Valves

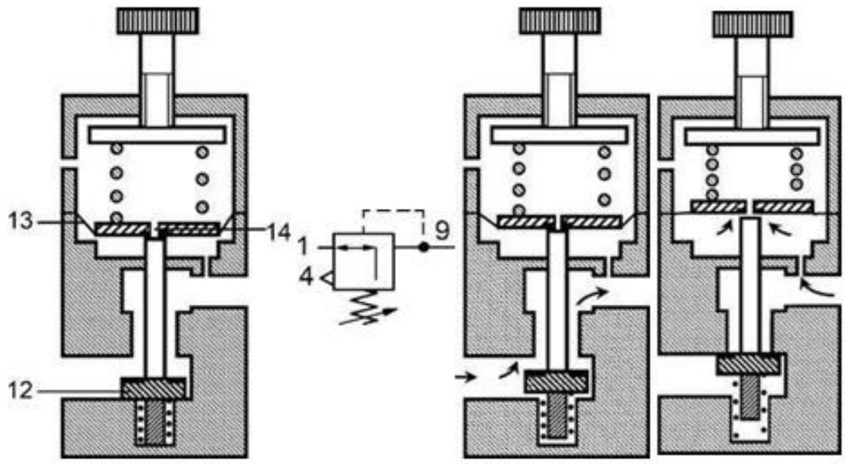
Usually, simple poppet-type relief valves are used to limit the maximum system pressure. The dimensions of pneumatic valves are much smaller compared with corresponding hydraulic valves because of the low-pressure losses. Therefore, direct-operated relief valves are usually used in pneumatic systems. The poppet, or its seat, is rubberized to reach the required tightness.



Relief Valve

Pressure Reducers

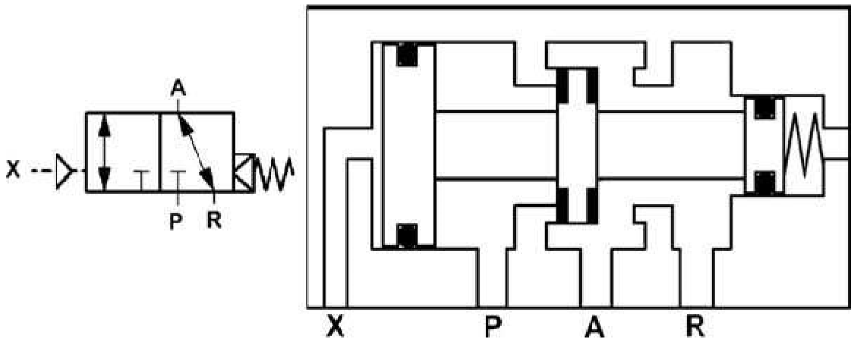
The pressure reducer is positioned downstream of the high-pressure compressed air reservoir. It is used to control the pressure of compressed air supplied to a subsystem. The ordinary pressure reducer has a rubberized poppet with a corresponding seat. The poppet controls the throttle area connecting the inlet (high pressure) with the outlet (reduced pressure) lines. The high- pressure air supplied to the valve is allowed to flow (expand) through the poppet valve. When the exit pressure rises, it acts on the piston through an internal pilot orifice. The piston and poppet move upward against the spring force, reducing the throttling area. When the exit pressure is increased to the required value, the poppet rests against its seat and the flow of air is stopped..



Construction and working of a venting - type pressure reducer

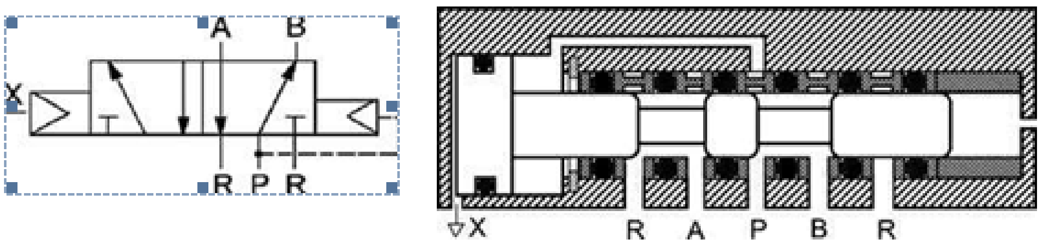
5.7 Directional Control Valves

Poppet-Type Directional Control Valves Figure below, shows a schematic of a 3/2 directional control valve (DCV) of the poppet type. The double face poppet is rubberized. The poppet is held to the left under the action of the spring and pressure forces. It closes the pressure line (P) and connects line (A) to exhaust line (R). When the pilot line (X) is pressurized, the poppet displaces to the right and rests against its right seat, connecting line (A) to the pressure line (P) and closing the venting line (R). The poppet-type DCV is of high resistance to leakage.



Directional Control Valve

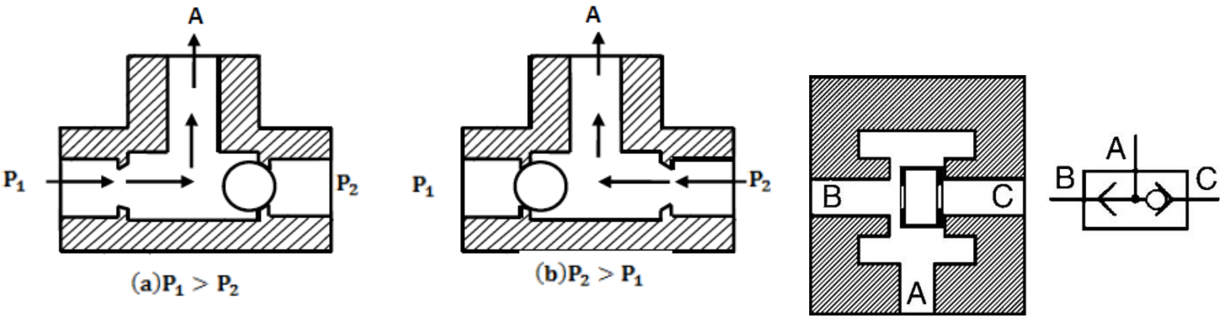
The spool-type DCV can be designed for a greater number of service ports, but the radial spool clearance is of very low resistance to air leakage. Therefore, this class of valves is equipped with sealing rings spaced by perforated metallic spacing rings (see Fig. below). The sealing rings introduce a considerable resistance force. Thus, this class of valves is usually pilot-operated when controlled electrically. This is due to the small force of the solenoid, which should be of limited volume.



A Spool-type Pneumatic Directional Control Valve

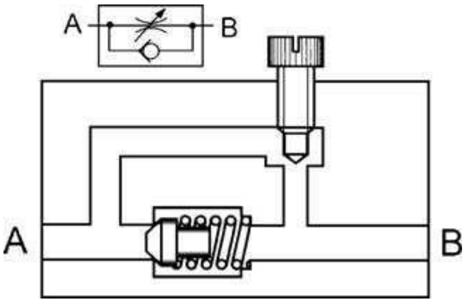
Shuttle Valves

A shuttle valve allows two alternate flow sources to be connected in a one-branch circuit. The valve has two inlets P1 and P2 and one outlet A. Outlet A receives flow from an inlet that is at a higher pressure. Figure shows the operation of a shuttle valve. If the pressure at P1 is greater than that at P2, the ball slides to the right and allows P1 to send flow to outlet A. If the pressure at P2 is greater than that at P1, the ball slides to the left and P2 supplies flow to outlet A. The construction of a shuttle valve is illustrated by below. Port (A) is pressurized if port (B) OR port (C) is pressurized.



5.8 Flow Control Valves

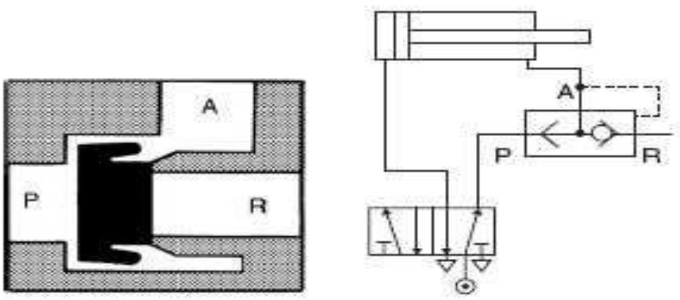
In general, the flow rate is controlled by using a simple throttling element, of fixed or variable area, with or without a parallel connected check valve as shown in Fig. below



Flow Control Valve

Quick Exhaust Valves

Quick exhaust valves are used wherever it is recommended to rapidly discharge the compressed air to maximize the cylinder speed, or for the quick release of brakes, for example. The air flows from port P to A with the exhaust line R closed (see Fig. below). When flowing from A, the line P is closed, and the air flows directly through exhaust port R.



CHAPTER 6: HYDRAULIC AND PNEUMATIC SYSTEMS TROUBLESHOOTING AND MAINTENANCE

In this chapter of Fluid Mechanics II you will learn:

- Troubleshooting to most common malfunction and poor performance of hydraulic system.
- Troubleshooting based on most probable cause of hydraulic system element malfunction.

Introduction

Hydraulic systems can be very simple, such as a hand pump pumping up a small hydraulic jack, or very complex, with several pumps, complex valving, accumulators, and many cylinders and actuators such as the hydraulic system of a commercial airliner. Yet, most of the problems encountered in all of these systems are often traced down to a few basic issues.

This chapter will help you to narrow down and solve most hydraulic system problems by focusing in on the most common causes of malfunction and poor performance. Many problems with hydraulic systems occur on startu p of a new system, after a repair or overhaul, after a new component is added, or when a component is adjusted or some other form of change occurs. In any machine, particularly in hydraulics, any change in one part of the system will affect all the other parts of the total system. This will result in problems that will require trouble shooting.

Primary Areas of Trouble Shooting Problems in Hydraulics

- Noisy Pump or Excessive Pump Vibration
- Low or Erratic Output Pressure
- No Pressure Output at all
- Hydraulic Cylinder not Moving
- Hydraulic Cylinder Slow response or Erratic
- Hydraulic Fluid Overheating

Trouble: Noisy Pump	
Cause	**Remedy**
Air leaking into System or air bubbles in intake oil	Be sure the oil reservoir is filled to the normal level and the oil intake is below the surface of the oil. Check pump packing, pipe and tubing connections, and all other points where air might leak into the system. If the oil level is low or the return line to the reservoir is installed above oil level, air bubbles will form in the oil reservoir. Check oil level and return-line
Cavitation	Check for a clogged or restricted intake line or a plugged air vent in the oil reservoir. Check all strainers in the intake line. Oil viscosity may be too high. Check the manufacturers' recommendations for the correct oil viscosity and type.
Loose or worn pump parts	Look for worn gaskets and packings; replace if necessary. There is usually no way to compensate for wear in a part; it is *always* better to replace it. Oil may be of improper grade or quality. Check the manufacturer's recommendations for the correct oil viscosity and type.
Stuck pump vanes, valves, pistons, etc.	Parts may be stuck by small particles such as; metallic chips, bits of lint, etc. If so, disassemble and clean them thoroughly. Avoid the use of files, emery cloth, steel hammers, etc., on machined surfaces. Products of oil deterioration such as gums, sludges, varnishes, and lacquers may be the cause of sticking. Use solvent to clean parts and dry thoroughly before reassembling. Be sure oil has sufficient resistance to deterioration and provides adequate protection against rusting and corrosion.
Filter or strainer too dirty. Filter too small	Filter and strainers must be kept clean enough to permit adequate flow. Check filter capacity. Be sure that original filter has not been replaced by one of smaller capacity. Use oil of quality high enough to prevent rapid sludge formation.
Pump running too fast	Determine recommended speed. Check pulley and gear sizes. Make sure that no one has installed a replacement motor with a faster speed than is recommended.

Trouble: Overheating	
Cause	**Remedy**
Oil viscosity too high	Check oil recommendation. If you‟re not sure of the oil viscosity in the system, it may be worth your while to drain the system and install oil of proper viscosity. Cold temperature conditions may cause oil of proper viscosity for “working temperature” to thicken too much on the way to the pump. In this case, use of oil with higher viscosity index may cure trouble.
Internal leakage too high	Check for wear and loose packings. Oil viscosity may be too low. Check the manufacturers oil viscosity and type recommendations. Under working conditions temperatures may increase to the point that the viscosity becomes too thin. Proceed with caution if you are tempted to try a higher viscosity oil.
Excessive discharge pressure	If oil viscosity is found to be OK, trouble may be caused by high setting of relief valve. If so, reset.
Poorly fitted pump parts	Poorly fitted parts may cause undue friction. Look for signs of excessive friction; be sure all parts are in alignment.viscosity and type.
Oil cooler clogged	On any machine equipped with an oil cooler, it is probable that high temperatures are expected. If temperatures run high normally, they‟ll go even higher if oil cooler passages are clogged. Clean cooler passages.
Low oil	If the oil supply is low, less oil will be available to adequately disperse heat. This will cause a rise in oil temperature, especially in machines without oil coolers. Be sure to fill to the proper oil level.

Trouble: Pump not pumping	
Cause	**Remedy**
Pump shaft turning in wrong direction	Shut down immediately. Some types of pumps can turn in either direction without causing damage; others are designed to turn in *one direction only* Check belts, pulleys, gears, and motor connections. Reversed leads on 3-phase motors are the most common cause of incorrect rotation.
Intake clogged or Low oil level	Check line from reservoir to pump. Be sure filter and strainers are not clogged. Be sure oil is up to recommended level in reservoir. Intake line must be below the oil level..
Air leak in intake	If any air at all is going through pump, it will probably be quite noisy. Pour oil over points suspected of leakage; if noise stops, you’ve found the leak.
Pump shaft speed too low	Some pumps will deliver oil over a wide range of speeds; others must turn at recommended speed to give appreciable flow. Find out first the speed recommended by the manufacturer; then, with a speed counter if possible, check the speed of the pump. If speed is too low, look for trouble in driving motor.
Oil too heavy	If oil is too heavy, some types of pumps cannot pick up prime. You can make a very rough check of viscosity by first getting some oil that is known to have the right viscosity. Then, with both oils at the same temperature, pour a quart of each oil through a small funnel. The heavier oil will take a noticeably longer time to run through. Oil that is too heavy can do great harm to hydraulic systems. Drain and refill with oil of the right viscosity.
Mechanical trouble (broken shaft, loose	Mechanical trouble is often accompanied by a noise that you can locate very easily. If disassembly is necessary, follow the manufacturer’s recommendations to the letter.

Trouble: Low Pressure in the System	
Cause	**Remedy**
Relief valve setting too low	If the relief valve setting is too low, oil may flow from the pump through the relief valve and back to the oil reservoir without reaching the point of use. To check the relief setting, block the discharge line *beyond the* relief valve and the check line pressure with pressure gauge.
Relief valve stuck open	Look for dirt or sludge in the relief valve. If the valve is dirty, disassemble and clean. A stuck valve may be an indication the system contains dirty or deteriorated oil. Be sure that the oil has high enough resistance to deterioration and varnishes.
Leak in system	Check the whole system for leaks. Serious leaks in the open are easy to detect, but leaks often occur in concealed piping. One routine in leak testing is to install a pressure gage in the discharge line near the pump and then block off the circuits progressively. When the gage pressure drops with the gage installed at a given point, the leak is between this point and the checkpoint just before it.
Broken, worn, or stuck pump parts	Install a pressure gage and block system just beyond the relief valve. If no significant pressure is developed and the relief valve is OK, look for mechanical trouble in the pump. Replace worn and broken parts.
Incorrect control valve setting; oil “short-circuited” to reservoir	If open-center directional control valves are unintentionally set in the neutral position, oil will return to the reservoir without meeting much resistance and very little pressure will be developed. Scored control-valve pistons and cylinders can cause this trouble. Replace worn parts.

Trouble: Erratic Action	
Cause	**Remedy**
Valves, pistons, etc., sticking or binding	First, check suspected parts for mechanical deficiencies such as misalignment of a shaft, worn bearings, etc. Then look for signs of dirt, oil sludge, varnishes and lacquers caused by oil deterioration. You can make up for mechanical deficiencies by replacing worn parts, but don"t forget that these deficiencies are often caused by the use of wrong oil..
Sluggishness when a machine is first started	Sluggishness is often caused by oil that is too thick at starting temperatures. If you can put up with this for a few minutes, oil may thin out enough to give satisfactory operation. But if oil does not thin out or if surrounding temperature remains relatively low, you may have to switch to oil

Cavitation.

During the working of positive displacement pump a vacuum is created at the inlet of the pump. This allows atmospheric pressure to push the fluid in. In some situations the vacuum may become excessive, and a phenomenon known as Cavitation occurs.

Name four popular methods to reduce cavitation

1) Keep the suction line velocities below 1.5 m/s

2) Keep the pump inlet as short as possible

3) Minimize the number of fittings in the inlet line

4) Mount the pump as close as possible to the reservoir.

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