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ΠΟΛΥΤΕΧΝΙΚΗ ΣΧΟΛΗ
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Hydraulic Systems

Analysis and Design

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We dedicate this thesis to our parents and in the memory of Aria.

K. Kaimenopoulos
D. Tsalagradas

Summary

Nowadays hydraulic systems are of high importance in the industrial as well as in the automotive, aeronautic and naval areas. The purpose of the present thesis is to introduce the reader to the function and analysis of hydraulic systems.

The thesis is based on a variety of bibliography sources aiming to provide a basic but complete spherical view of hydraulic systems. Thus, the structure is established by presenting the major designs of the different components that compose the hydraulic systems, introducing several efficient hydraulics subassemblies that correspond to different objectives and furthermore, analyzing and focusing the scientific areas of Fluid Mechanics, Transport Phenomena and Thermodynamics to better describe common met situations.

Hydraulic systems are unsurpassed in terms of speed and power in relation to other systems. The wide variety of components and their possible assemblies makes them very flexible. Furthermore, the fact that power is transferred through oil, minimizes and lubricates the moving parts providing high reliability and accuracy proving their strong position in today’s engineering.

Περίληψη

Σήμερα τα υδραυλικά συστήματα έχουν μεγάλη σημασία τόσο στις βιομηχανίες όσο στα αεροναυπηγικά, αυτοκινητοβιομηχανικά και ναυπηγικά πεδία. Σκοπός της παρούσας διπλωματικής εργασίας είναι να εισαγάγει τον αναγνώστη στη λειτουργία και την ανάλυση των υδραυλικών συστημάτων.

Η διπλωματική βασίζεται σε ποικίλες πηγές βιβλιογραφίας που αποσκοπούν στην παροχή μιας βασικής αλλά ολοκληρωμένης εικόνας των υδραυλικών συστημάτων. Έτσι, η δομή συντίθεται παρουσιάζοντας τις κύριες παραλλαγές των διαφόρων εξαρτημάτων που συνθέτουν τα υδραυλικά συστήματα, κατηγοριοποιώντας έναν αριθμό αποδοτικών υδραυλικών υποσυστημάτων που αντιστοιχούν σε διαφορετικούς στόχους και επιπλέον αναλύοντας και απλοποιώντας τις επιστημονικές περιοχές της Μηχανικής Ρευστών, των Φαινομένων Μεταφοράς και της Θερμοδυναμικής για την εξειδικευμένη περιγραφή κοινών καταστάσεων.

Τα υδραυλικά συστήματα είναι ανώτερα όσον αφορά την ταχύτητα και την ισχύ σε σχέση με άλλα συστήματα. Η μεγάλη ποικιλία εξαρτημάτων και πιθανών συνδυασμών τους, τα καθιστούν πολύ ευέλικτα. Επιπλέον, το γεγονός ότι η ισχύς μεταδίδεται μέσω λαδιού, ελαχιστοποιεί τα κινούμενα μέρη και τα λιπαίνει, παρέχοντας υψηλή αξιοπιστία και ακρίβεια αποδεικνύοντας την ισχυρή θέση τους στη σημερινή τεχνολογία.

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1. Introduction

1.1 Working Principle

The fast pace of technological evolution made possible increased production capabilities as well as the accomplishment of larger and more advanced constructions. However, along with those capabilities, an increased demand in terms of speed and power was raised, a problem that engineers had to face in the most efficient way.
The controlled movement of parts or a controlled application of force is a common requirement in the industries. These operations are often performed by using electrical machines or diesel, petrol and steam engines as a prime mover. These prime movers can provide various movements to the objects by using some mechanical attachments like screw jack, lever, rack and pinions etc. However, these are not the only prime movers. The enclosed fluids (liquids and gases) can also be used as prime movers to provide high magnitude-controlled motion (linear or rotary) and force to objects or substances. This kind of fluid-power based systems using pressurized incompressible liquids as transmission media are called Hydraulic Systems.
Compared to all other actuation technologies, including electric motors, fluid power is unsurpassed for force and power density and is capable of generating extremely high forces with relatively lightweight cylinder actuators which make it ideal for high speed, high force, high power applications. The hydraulic fluid transmits force applied at one point in the system to some other location and to produce any desired change in direction or magnitude of this force. To carry out this function in the most efficient manner, the hydraulic fluid must be relatively incompressible and must flow readily and because oil has a high bulk modulus, hydraulic systems can be finely controlled for precision motion applications.
The function of hydraulic systems is based on the incompressibility of fluids and the Pascal's Law which states that a pressure change occurring anywhere in a confined incompressible fluid is transmitted throughout the fluid such that the same change occurs everywhere. This way high output forces can be produced with relatively small inputs of pressure.

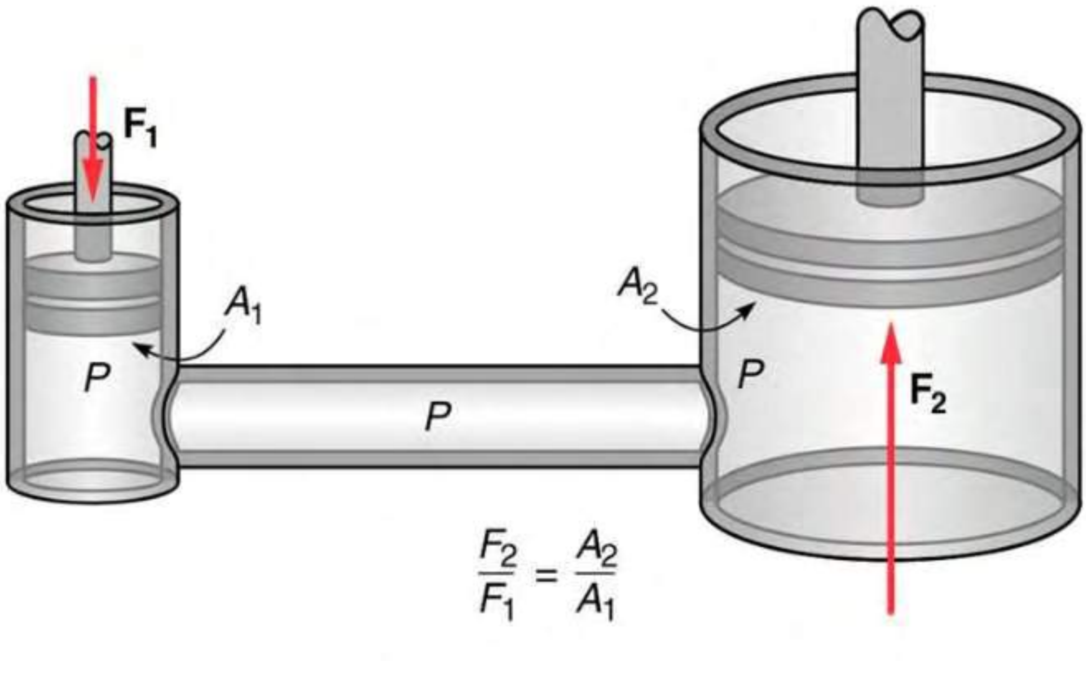


Figure 1-1 - Pascal's Law Illustration.
(me-mechanicalengineering.com)

Fluid power systems have a higher bandwidth than electric motors and can be used in applications that require fast starts, stops and reversals, or that require high frequency oscillations. Another major advantage of fluid power is compactness and flexibility. Hydraulic cylinders are relatively small and light for their weight and flexible hoses allows power to pass around corners, over joints and through tubes leading to compact packaging without sacrificing high force and high power.
However, fluid power is not as easy to generate as electric power and requires a heavy, noisy pump. Also, with the use of oil under pressure, leaking can occur at connections and seals and the oil itself can cavitate and retain air resulting in spongy performance and loss of precision. Furthermore, hydraulic and pneumatic systems become contaminated with particles and require careful filtering.
The physics of fluid power is more complex than that of electric motors which makes modeling and control more challenging. University and industry researchers are working hard not only to overcome these challenges but also to extend their use to new applications, for example tiny robots and wearable power-assist tools (Durfee, Sun, & Ven, 2015).

Figure 1-2 - Hydraulic Cylinder (www.hunger-hydraulics.com)

1.2 History of Hydraulic Systems

Hydraulics is a very ancient science. It traces back to the Egyptians and Babylonians, who constructed canals. Later the Roman and Egyptians, who –like the previous- had been more interested in the practical and constructional aspects of hydraulics than in theorizing. The first ones that tried to rationalize the nature of pressure and flow patterns were the Greeks, with the laws of hydrostatics and buoyancy. Although, development had been made it was very slow. This was the case until the Renaissance, when men such as Leonardo Da Vinci began to publish the results of their observations. Ideas which emerged then, respecting conservation of mass (continuity of flow), frictional resistance and the velocity of surface waves, are still in use, though sometimes in a more refined form.

In the 17th century, several brilliant men emerged. Descartes, Pascal, Newton, Boyle, Hooke and Leibnitz laid the foundations of modern mathematics and physics. This enabled researchers to perceive a logical pattern in the various aspects of mechanics. On this basis, four great pioneers -Bernoulli, Euler, Clairaut and D'Alembert- developed the academic discipline of hydrodynamics. The 19th century was a period of further advance. Hagen constructed experiments to investigate the effects of temperature on pipe flow. At almost the same time, Poiseuille developed equations for laminar flow in pipes. Further contributions were made by Weisbach, Bresse and Henri Darcy, who developed equations for frictional resistance in pipe and channel flows.
The rapid growth of industry in the 19th and 20th centuries was by now producing a demand for a better understanding of fluid flow phenomena. Navier, Stokes, Schwarz, Christoffel and other hydrodynamicists all contributed to the development of a formidable array of mathematical equations and methods. However, the real breakthrough came with the work of Prandtl. He proposed that flow was “divided into two interdependent parts. There is on the one hand the free fluid which can be treated as inviscid and on the other hand the transition layer at the fixed boundaries”. With this brilliant insight, Prandtl effectively fused together the two disparate schools of thought and laid the foundation for the development of the unified science of Fluid Mechanics.
The 20th century has, in consequence, seen tremendous advances in the understanding and application of fluid mechanics in almost every branch of engineering. Since 1945, the advent of the electronic computer, and advances in sensing and data logging equipment have revolutionized many aspects of hydraulics. Our understanding of the nature of turbulence, steady and unsteady flows in channels, sediment transport and maritime phenomena have developed rapidly. This has been matched by developments in software (Manring, 2005).
Figure 1-3 traces the development history of some typical water and oil hydraulics.

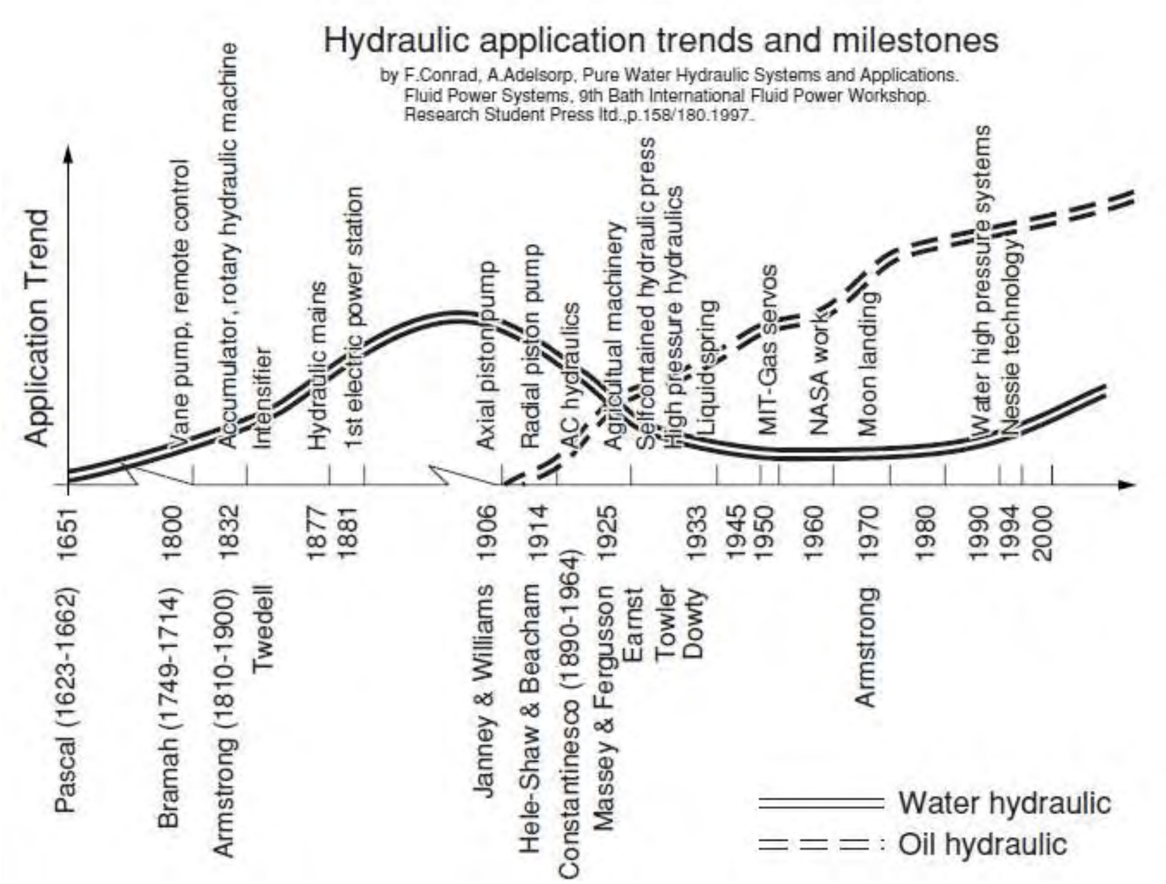


Figure 1-3 - Development History of Hydraulics
(Yuken Kogyo Co. - Basic Hydraulics and Components)

1.3 Applications

Hydraulics is applied in a wide range of industries: from construction machinery, automobiles, and airplanes (outdoor) to machine tools and press machines (indoor). Typical applications in each industrial field are listed below. Figure 1-4 shows photos of some of the applications.

- Construction machinery: earthmoving equipment (e.g. excavators, bulldozers, wheel loaders), cranes, tunnel boring equipment, rail equipment, building and construction machineries and drilling rigs
- Agricultural/forestry machinery: tractors, combines, rice planting machines, lawn mowers, and logging machines
- Industrial processing/forming machinery: steel mill, machine tools, and plastic processing, die casting, press, and sheet metal processing machines, automated production lines, loaders, textile machineries, R&D equipment and robotic systems
- Automobiles: power steering, transmissions, brake systems, shock absorbers and accessories for transport vehicles
- Industrial and special-purpose vehicles: fork lifts, platform vehicles, garbage trucks, concrete mixer trucks, concrete pump trucks, and accessories for transport vehicles (wing roofs and tail lifts)
- Ships/fishing machinery: steering, propulsion machinery, and deck cranes
- Aerospace machinery: steering, brake systems, and landing gear
- Testing machinery/simulator: vibration testers, flight simulators, and amusement machines

Special equipment: hydraulic lifts, vibration control systems for high-story buildings and trains, sluice gates, crushers, and compactors (Basic Hydraulics and Components, 2006).

Figure 1-4 - Top to bottom: Dump Truck. Excavator. Hydraulic Press (aviratgroup.wordpress.com, www.miromfg.com)

1.4 Components and Basic Operation Process

Basic components to be used in hydraulic systems are categorized as follows.

- Energy converters (hydraulic pumps, motors, and cylinders)
- Energy controllers (directional, pressure, and flow control valves)
- Accessories (reservoirs, filters, tubing, accumulators, sensors, etc.)

A power source (hydraulic package or unit) for practical systems consists of a hydraulic pump, a motor, and a reservoir. Depending on the required accuracy and operability, control valves are also incorporated in the systems. Recently, systems have become available that drive hydraulic pumps with servo motors and adjust the pump speed to control the flow and pressure. Figure 1-5 and Figure 1-6 show a circuit example of the most basic hydraulic systems (Basic Hydraulics and Components, 2006).

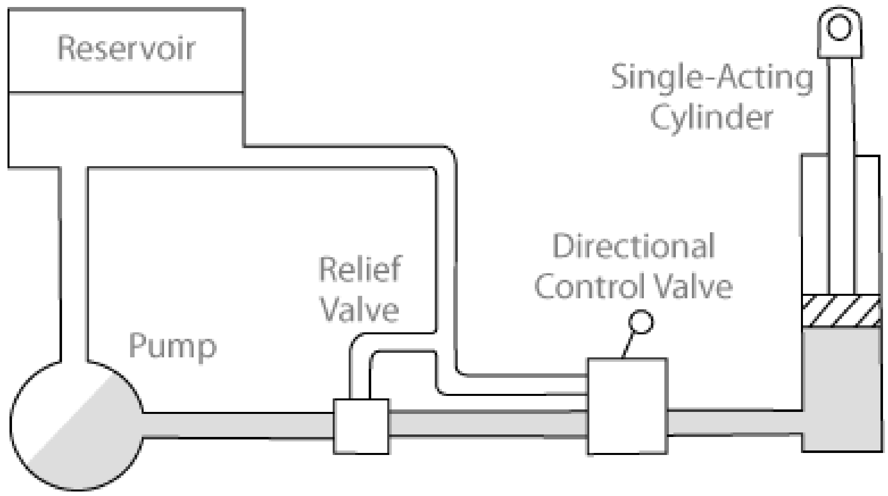


Figure 1-5 - Hydraulic Circuit Example.
(isccompanies.com)

Basic operation process:

1. The reservoir contains the appropriate quantity of oil in order for the system to operate properly.
2. The pump driven by an electric motor produces a flow of oil through the system.
3. The pressure regulator controls the maximum pressure in the system. If the pressure exceeds a set limit, the regulator directs the flow back to the tank and this way relieving the system from the pressure rise.
4. For normal operating pressures the regulator allows flow to the direction valve which can control whether the oil is going to flow from the upper to the lower or from the lower to the upper chamber of the hydraulic cylinder.
5. The hydraulic cylinder is the component through which the hydraulic power is transformed to work with the movement of the shaft.
6. The oil exiting the respective chamber of the cylinder returns to the tank after passing through a filter.

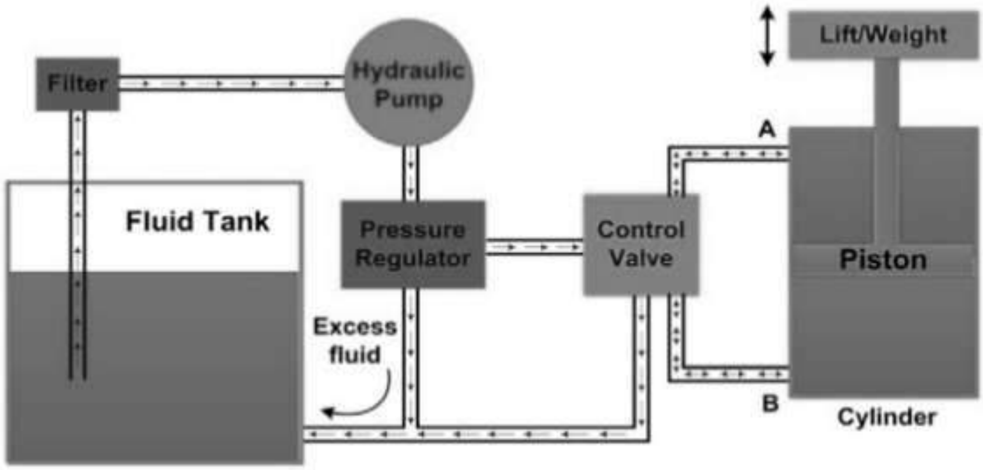


Figure 1-6 - Hydraulic Circuit Example
(Joshi S.N., Mechatronics and Manufacturing Automation)

1.5 Fundamentals

Analyzing the system dynamics of fluid power means using differential equations and simulations to examine the pressures and flows in components of a fluid power circuit, and the forces and motions of the mechanisms driven by the fluid power. Because fluid power systems change with time and because fluid power systems have energy storage elements, a dynamic system analysis approach must be taken which means the use of linear and nonlinear differential equations, linear and nonlinear simulations, time responses, transfer functions and frequency analysis.
Fluid power is one domain within the field of system dynamics, just as mechanical translational, mechanical rotational and electronic networks are system dynamic domains. Fluid power systems can be analyzed with the same mathematical tools used to describe spring-mass damper or inductor-capacitor-resistor systems. Like the other domains, fluid power has fundamental power variables and system elements connected in networks. Unlike other domains many its elements are nonlinear which makes closed-form analysis somewhat more challenging, but not difficult to simulate. Many concepts from transfer functions and basic closed loop control systems are used to analyze fluid power circuits. Like all system dynamics domains, fluid power is characterized by two power variables that when multiplied form power, and ideal lumped elements including two energy storing elements, one energy dissipating element, a flow source element and a pressure source element. Figure 1-7 shows the analogies between fluid power elements and elements in other domains. Lumping fluid power systems into elements is useful when analyzing complex circuits (Durfee, Sun, & Ven, 2015).

Domain	*Power Variable 1*	*Power Variable 2*	*Storage Element 1*	*Storage Element 2*	*Dissipative Element*
Translational	Force, F	Velocity, V	Mass, M	Spring, K	Damper, B
Rotational	Torque, T	Velocity, ω	Inertia, J	Spring, K	Damper, B
Electrical	Current, I	Voltage, V	Inductor, L	Capacitor, C	Resistor, R
Fluid Power	Flow, Q	Pressure, P	Inertance, I_f	Capacitor, C_f	Resistance, R_f

Figure 1-7 - Analogies
(W. Durfee, Z. Sun, J. V. Ven - Fluid Power System Dynamics)

1.6 Advantages and Disadvantages

Advantages

There are many unique features of hydraulic control compared to other types of control. These are fundamental and account for the wide use of hydraulic control. Some of the advantages are the following (Merritt, 1967):

- Heat generated by internal losses is a basic limitation of any machine. Lubricants deteriorate, mechanical parts seize, and insulation breaks down as temperature increases. Hydraulic components are superior to others in this respect since the fluid carries away the heat generated to a convenient heat exchanger. This feature permits smaller and lighter components.
- The hydraulic fluid also acts as a lubricant and makes possible long component life.
- There is no phenomenon in hydraulic components comparable to the saturation and losses in magnetic materials of electrical machines. The torque developed by an electric motor is proportional to current and is limited by magnetic saturation. The torque developed by hydraulic actuators (i.e., motors and pistons) is proportional to pressure difference and is limited only by safe stress levels. Therefore, hydraulic actuators develop relatively large torques for comparatively small devices.
- Electrical motors are basically a simple lag device from applied voltage to speed. Hydraulic actuators are basically a quadratic resonance from flow to speed with a high natural frequency. Therefore, hydraulic actuators have a higher speed of response with fast starts, stops, and speed reversals possible. Torque to inertia ratios are large with resulting high acceleration capability. overall, higher loop gains and bandwidths are possible with hydraulic actuators in servo loops.
- Hydraulic actuators may be operated under continuous, intermittent, reversing, and stalled conditions without damage. With relief valve protection, hydraulic actuators may be used for dynamic breaking. Larger speed ranges are possible with hydraulic actuators. Both linear and rotary actuators are available and add to the flexibility of hydraulic power elements.
- Hydraulic actuators have higher stiffness, that is, inverse of slope of speed-torque curves, compared to other drive devices since leakages are low. Hence there is little drop in speed as loads are applied. In closed loop systems this results in greater positional stiffness and less position error.
- Open and closed loop control of hydraulic actuators is relatively simple using valves and pumps.

Disadvantages

Although hydraulic controls offer many distinct advantages, several disadvantages tend to limit their use. Major disadvantages are the following (Merritt, 1967):

- Hydraulic power is not so readily available as that of electrical power This is not a serious threat to mobile and airborne applications but most certainly affects stationary applications.
- Small allowable tolerances results in high costs of hydraulic components.
- The hydraulic fluid imposes an upper temperature limit. Fire and explosion hazards exist if a hydraulic system is used near a source of ignition. However, these situations have improved with the availability of high temperature and fire-resistant fluids. Hydraulic systems are messy because it is difficult to maintain a system free from leaks, and there is always the possibility of complete loss of fluid if a break in the system occurs.
- It is impossible to maintain the fluid free of dirt and contamination. Contaminated oil can clog valves and actuators and, if the contaminant is abrasive, cause a

permanent loss in performance and/or failure. Contaminated oil is the chief source of hydraulic control failures. Clean oil and reliability are synonymous terms in hydraulic control.

- Basic design procedures are lacking and difficult to obtain because of the complexity of hydraulic control analysis. For example, the current flow through a resistor is described by a simple law—Ohm's law. In contrast, no single law exists which describes the hydraulic resistance of passages to flow. For this seemingly simple problem there are almost endless details of Reynolds number, laminar or turbulent flow, passage geometry, friction factors, and discharge coefficients to cope with. This factor limits the degree of sophistication of hydraulic control devices.
- Hydraulics are not so flexible, linear, accurate, and inexpensive as electronic and/or electromechanical devices in the manipulation of low power signals for purposes of mathematical computation, error detection, amplification, instrumentation, and compensation. Therefore, hydraulic devices are generally not desirable in the low power portions of control systems.

1.7 Thesis Overview

Hydraulic Components

Every hydraulic system consists of some integral parts, such as pumps, motors and valves. The purpose of this chapter is to introduce the most significant designs of each individual part and analyze their function. The included parts apart from the above mentioned are: reservoirs, filters, hoses, accumulators, heat exchangers and shock absorbers.

Fundamentals of Fluid Power

Three main objectives are analyzed in this chapter.

- The first refers to the properties of fluids. Except for the pneumatic systems which use air, the hydraulic systems use oil as a mean to transfer power, thus it is of high importance to understand the parameters that characterize a fluid like viscosity or bulk modulus.
- The second part refers to the mathematical scope of hydraulic systems. A combination of Transport Phenomena, Liquid Mechanics and Thermodynamics principles is used to describe such a system.
- Finally, the third object is to focus on specific situations found in hydraulic systems, where the general equations are specialized and simplified.

Hydraulic Systems

Having read about the components that assembly a hydraulic system and the mathematical approach that describes it, the main objective of the third chapter is to present different types of systems. Since there are many combinations of components that can lead to the same output, some efficient patterns have been established through the years depending on the type of task the hydraulic system is going to face. Furthermore, the schematics used for circuit design are presented as well as some examples of how to perform calculations on a hydraulic circuit.

2. Hydraulic Components

Mechanical components perform a basic function in a hydraulic power or control system and must satisfy numerous requirements to perform adequately in a given circuit. Furthermore, the hydraulic fluid influences the operation of the system components and they, in turn, affect the performance of the hydraulic fluid. The hydraulic fluid and most of the mechanical components that compose the hydraulic circuit are discussed in the current chapter (U.S. Army Material Command, 1971).

2.1 Pumps

Hydraulic pumps supply energy to the system, converting the torque and velocity of an input shaft to pressure and flow of the output fluid. Thereby convert mechanical energy to hydraulic energy. It provides the force required to transmit power. Pumps are rated in terms of flow and pressure. The flow rating (volumetric output) is the amount of liquid which can be delivered by the pump per time unit at a specified speed. A pump does not produce pressure. The pressure developed at the outlet depends on the resistance to flow in the circuit. Pumps are classified according to configuration or operating characteristics. They can also be classified as fixed or variable displacement devices (U.S. Army Material Command, 1971).

- ***Variable 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 cannot withstand high pressures and generally used for low-pressure and high-volume flow applications. The fluid motion is generated due to rotating propeller. These pumps provide a smooth and continuous flow but the flow output decreases with an increase in system resistance (load). Therefore, the flow rate not only depends on the rotational speed but also on the resistance provided by the system. The important advantages of non-positive displacement pumps are lower initial cost, less operating maintenance because of less moving parts, simplicity of operation, higher reliability and suitability with wide range of fluid etc. These pumps are primarily used for transporting fluids and find little use in the hydraulic or fluid power industries.

- ***Fixed displacement pump***

These pumps deliver a constant volume of fluid in a cycle. The discharge quantity per revolution is fixed and they produce fluid flow proportional to their displacement and rotor speed. They are used in most of the industrial fluid power applications. The output fluid flow is constant and is independent of the system pressure (load). The important advantage associated with these pumps is that the high-pressure and low-pressure areas (means input and output region) are separated and hence the fluid cannot leak back due to higher pressure at the outlets. These features make the positive displacement pump most suited and universally accepted for hydraulic systems. The important advantages of positive displacement pumps over non-positive displacement pumps include capability to generate high pressures, high volumetric efficiency, high power to weight ratio (Joshi, 2010).

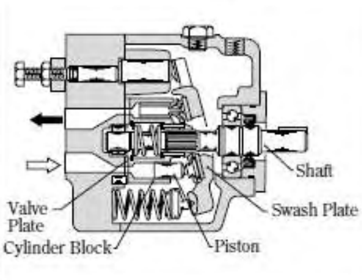
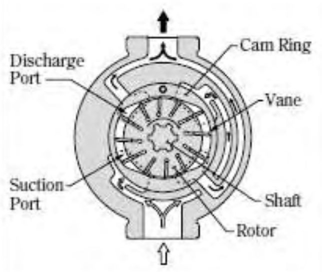
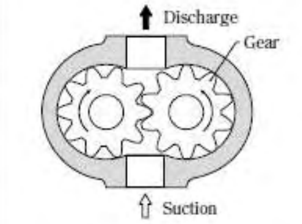
Type	Piston Pumps	Vane Pumps	Gear Pumps
Structure			
Operation Principle	Expansion and compression of a volume in a cylinder block with the piston stroke	Expansion and compression of volumes between the vanes and the cam ring	Movement of volumes between tooth spaces and the casing (the external gear pump is shown.)
Efficiency	· Generally the highest. · The valve plate is easily damaged and efficiency drops as the plate wears out.	· Generally low. · Can be compensated when the vane wears out.	· Generally low. · Drops as the gear wears out.
Contamination Resistance	Highly susceptible to foreign substances in oil.	Susceptible to foreign substances in oil, but less so than piston pumps.	Susceptible to foreign substances in oil, but hardly susceptible when the pumps are low pressure types.
Suction Ability	Low.	Middle.	High.
Variable Displacement Type	Easy to convert by changing the angle of the swash plate or bent axis.	Can be converted by changing the eccentricity of the cam ring for the unbalanced type.	Difficult.
Size and price	Generally large, heavy, and expensive.	Smallest and relatively inexpensive.	Small, light, and inexpensive.

Figure 2-1 - Characteristics of Pumps
(Yuken Kogyo Co. - Basic Hydraulics and Components)

Ideal pumps and motors are defined by the relations between fluid pressure P and flow Q and shaft torque T and velocity ω. For an ideal motor, input and output power is conserved. If ΔP is the pressure difference across the pump or motor, then the power balance for a pump or motor is (Durfee, Sun, & Ven, 2015):

$$Power = \Delta P\ Q = T\ \omega \quad (1.1)$$

If the volumetric displacement of the motor or pump is D_v, then the relations relating fluid to mechanical are:

$$T = D_v\ \Delta P \quad (1.2)$$

$$Q = D_v\ \omega \quad (1.3)$$

Real pumps and motors are not 100% efficient and, like cylinders, have an overall efficiency, which is made up of volumetric η_v and mechanical η_m efficiencies with:

$$\eta = \eta_v\ \eta_m \quad (1.4)$$

2.1.1 Gear Pumps

Gear pump is a sturdy and simple fixed displacement pump. It has two meshed gears revolving about their respective axes. These pumps operate with two gears engaged with each other and rotating to feed a hydraulic fluid from the suction area to the discharge area. These gears are the only moving parts in the pump. They are compact, relatively inexpensive and have few moving parts. The rigid design of the gears and houses allow for very high pressures and the ability to pump highly viscous fluids. The gear pumps are relatively resistant to working fluid contamination (Basic Hydraulics and Components, 2006), (Joshi, 2010).

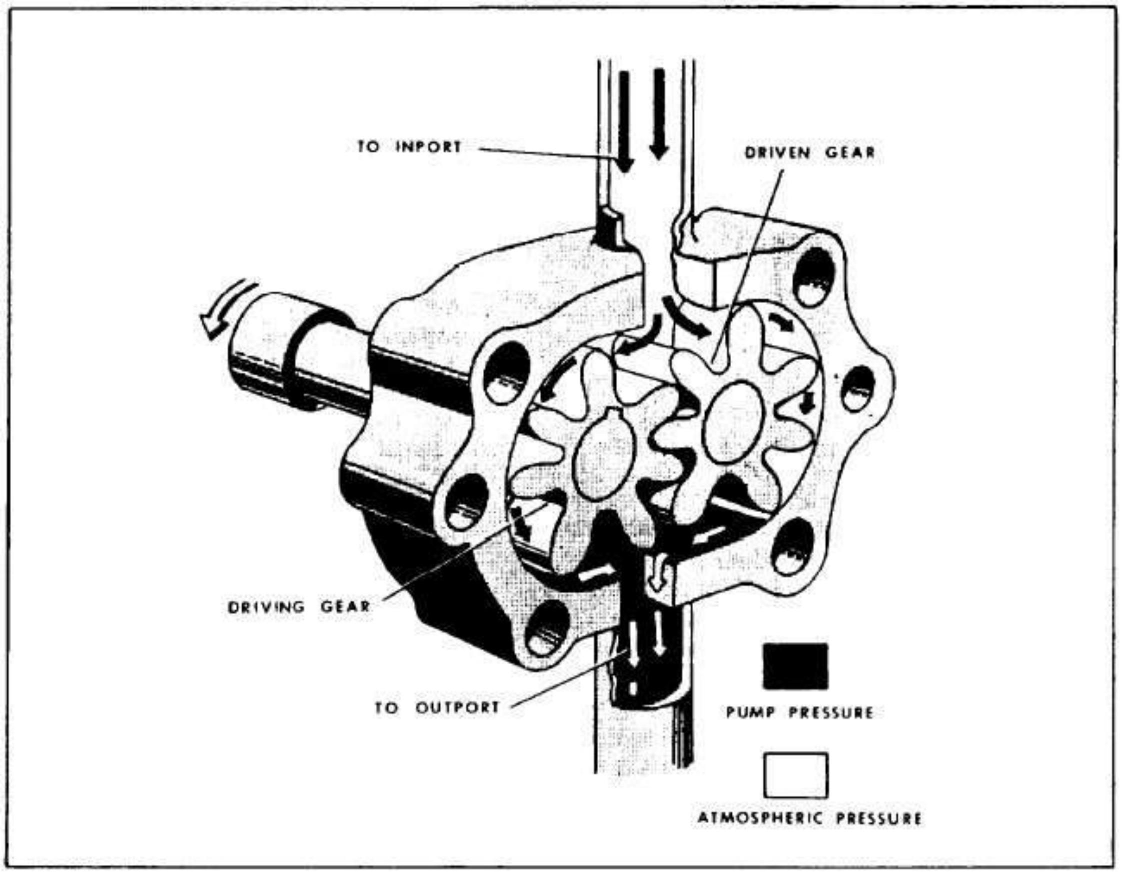


Figure 2-2 - Gear Pump
(MMC A. Beasley, Jr. - Fluid Power)

2.1.1.1 **External Gear Pumps**

The external gear pump consists of two externally meshed gears housed in a pump case as shown in Figure 2-3. One of the gears is coupled with a prime mover and is called as driving gear and another is called as driven gear. The rotating gear carries the fluid from the tank to the outlet pipe. When the gears rotate, volume of the chamber expands leading to pressure drop below atmospheric value. Therefore, vacuum is created and the fluid is pushed into the void due to atmospheric pressure. The fluid is trapped between housing and rotating teeth of the gears.

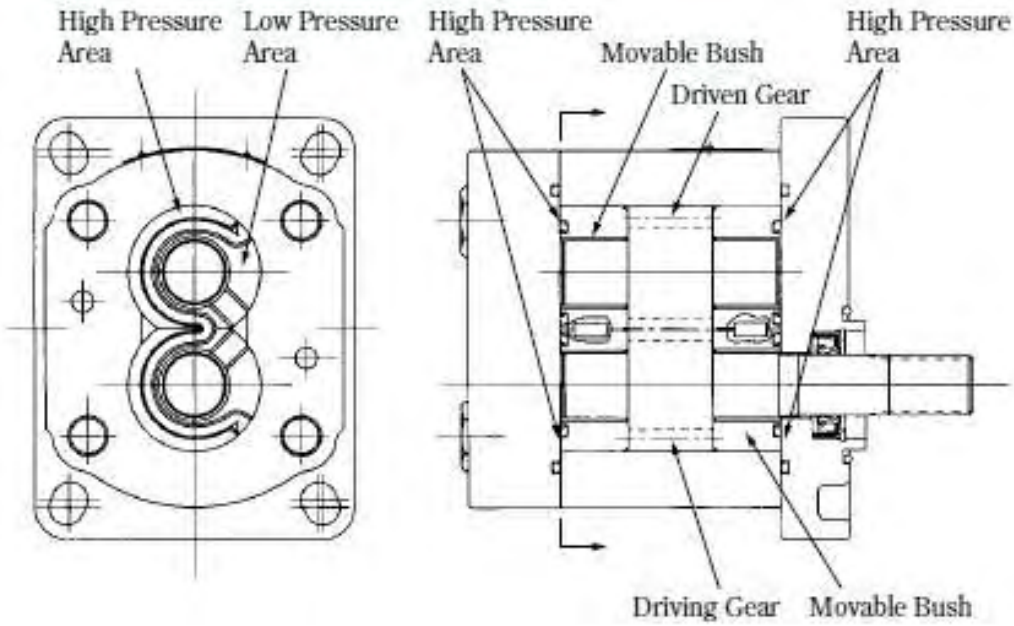


Figure 2-3 - External Gear Pump
(Department of the Army, Headquarters - Hydraulics)

The amount of fluid discharge is determined by the number of gear teeth, the volume of fluid between each pair of teeth and the speed of rotation. The important drawback of external gear pump is the unbalanced side load on its bearings. It is caused due to high pressure at the outlet and low pressure at the inlet which results in slower speeds and lower pressure ratings in addition to reducing the bearing life. Gear pumps are most commonly used for the hydraulic fluid power applications and are widely used in chemical installations to pump fluid with a certain viscosity (Joshi, 2010).

Types of external gear pumps (U.S. Army Material Command, 1971):

i. ***Spur gear pumps:*** A spur gear rotary hydraulic pump is illustrated in Figure 2-5. The two gears rotate in opposite directions and transfer liquid from the inlet to the outlet through the volume between the teeth and the housing. The output depends on tooth width and depth, and is largest for a minimum number of teeth. The spur gear pump is a fixed displacement pump.

ii. ***Helical gear pumps:*** A variation of the external spur gear pump is the helical gear pump. The fact that several teeth are engaged simultaneously allows the helical gear pump to carry larger loads at high speeds than can the spur gear pump. Operation is similar to that of the spur gear pump, but with less noise and usually smaller flow pulsations.

iii. ***Herringbone gear pumps:*** Another variation of the external gear pump incorporates herringbone gears. Like all gear pumps, the herringbone device is a constant displacement pump.

iv. ***Lobe pumps:*** Lobe pumps work on the similar principle of working as that of external gear pumps. However, the lobes do not make any contact like external gear pump (Figure 2-4). Similar to the external gear pump, the lobes rotate to create expanding volume at the inlet.

Figure 2-5 - Spur Gear Pump
(www.visionqci.com)

Figure 2-4 - Lobe Pump
(www.rodem.com)

2.1.1.2 **Internal Gear Pumps**

Internal gear pumps are exceptionally adaptable. It comprises of an internal gear, a regular spur gear, a crescent-shaped seal and an external housing. The schematic of internal gear pump is shown in Figure 2-6. Liquid enters the suction port between the rotor (large exterior gear) and idler (small interior gear) teeth. Liquid travels through the pump between the teeth and crescent. Crescent divides the liquid and acts as a seal between the suction and discharge ports. When the teeth mesh on the side opposite to the crescent seal, the fluid is forced out through the discharge port of the pump. This clearance between gears can be adjusted to accommodate high temperature, to handle high viscosity fluids and to accommodate the wear. However, they are not suitable for high speed and high-pressure applications (Bolton, 2003).

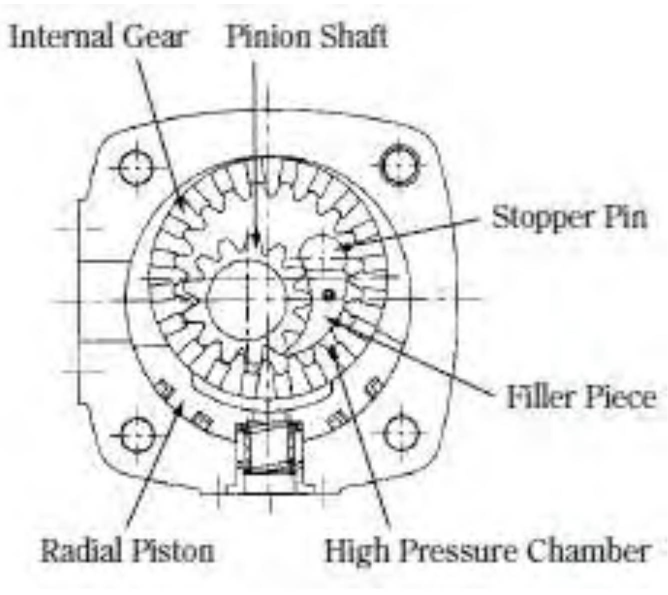


Figure 2-6 - Internal Gear Pump
(Department of the Army, Headquarters - Hydraulics)

Types of internal gear pumps (Department of the Army, 1997):

i. ***Crescent seal pumps:*** The crescent seal pump consists of an inner and outer gear separated by a crescent shaped seal (Figure 2-7). The gears rotate the same direction, with the inner gear rotating at a higher speed. The liquid is drawn into the pump at the point where the gear teeth begin to separate and is carried to the outlet in the space between the crescent and the teeth of both gears. The contact point of the gear teeth forms a seal, as does the small tip clearance at the crescent. This pump is generally used for low output applications.

ii. ***Gerotor pumps:*** Gerotor is a fixed displacement pump (Figure 2-8). The name Gerotor is derived from Generated Rotor. Gerotor pump is an internal gear pump without the crescent. It consists of two rotors, the inner and the outer rotor. The inner rotor has N teeth, and the outer rotor has $N + 1$ teeth. The inner rotor is located off-center and both rotors rotate. The geometry of the two rotors partitions the volume between them into N different dynamically changing volumes. During the rotation, volume of each partition changes continuously. Therefore, any given volume first increases, and then decreases. An increase in volume creates vacuum. This vacuum creates suction, and thus, this part of the cycle sucks the fluid. As the volume decreases, compression occurs. During this compression period, fluids can be pumped, or compressed (if they are gaseous fluids). The flow output is uniform and constant at the outlets.

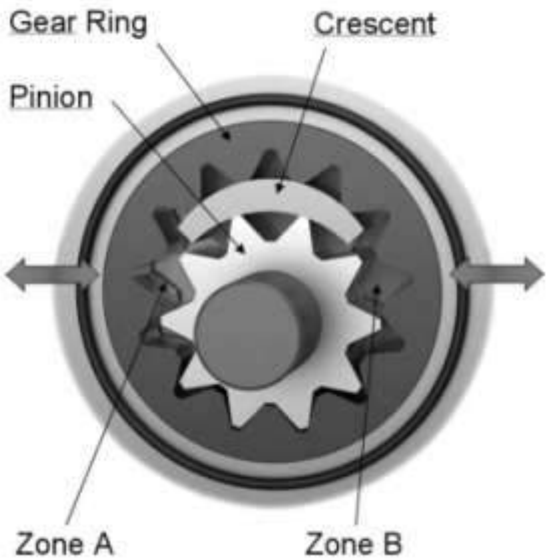


Figure 2-7 - Crescent Seal Pump (www.oemoffhighway.com)

Figure 2-8 - Gerotor Pump (www.appliedpumps.co.uk)

2.1.2 Vane Pumps

Vane pumps generate a pumping action by tracking of vanes along the casing wall. They generally consist of a rotor, vanes, ring and a port plate with inlet and outlet ports. The rotor in a vane pump is connected to the prime mover through a shaft. The vanes are located on the slotted rotor. The rotor is eccentrically placed inside a cam ring as shown in the Figure 2-9. When the prime mover rotates the rotor, the vanes are thrown outward due to centrifugal force. The vanes track along the ring. This produces a suction cavity in the ring as the rotor rotates and therefore, the fluid is pushed into the pump through the inlet. The fluid is carried around to the outlet by the vanes whose retraction causes the fluid to be expelled. The capacity of the pump depends upon the eccentricity, expansion of vanes, width of vanes and speed of the rotor.
Their simple construction results in a high degree of reliability and easy maintenance. They are relatively low in cost and exhibit long operating life. They have comparatively high volumetric and overall efficiencies and are available in a wide range of output ratings. These pumps are quieter because of their structure and are less susceptible to working fluid contamination than piston pumps. Therefore, they are conveniently used in a wide range of applications. Capacity and pressure ratings of a vane pump are generally

lower than the gear pumps, but reduced leakage gives an improved volumetric efficiency of around 95%.
They provide the following advantages: minimized discharge pressure pulsation, compactness and light weight for high output, less efficiency degradation due to vane wear, and reliability and ease of maintenance.
The schematic of vane pump working principle is shown in Figure 2-9. These pumps can handle thin liquids (low viscosity) at relatively higher pressure. However, they are not suitable for high speed applications and for the high viscosity fluids or fluids carrying some abrasive particles. The maintenance cost is also higher due to many moving parts (U.S. Army Material Command, 1971), (Basic Hydraulics and Components, 2006).

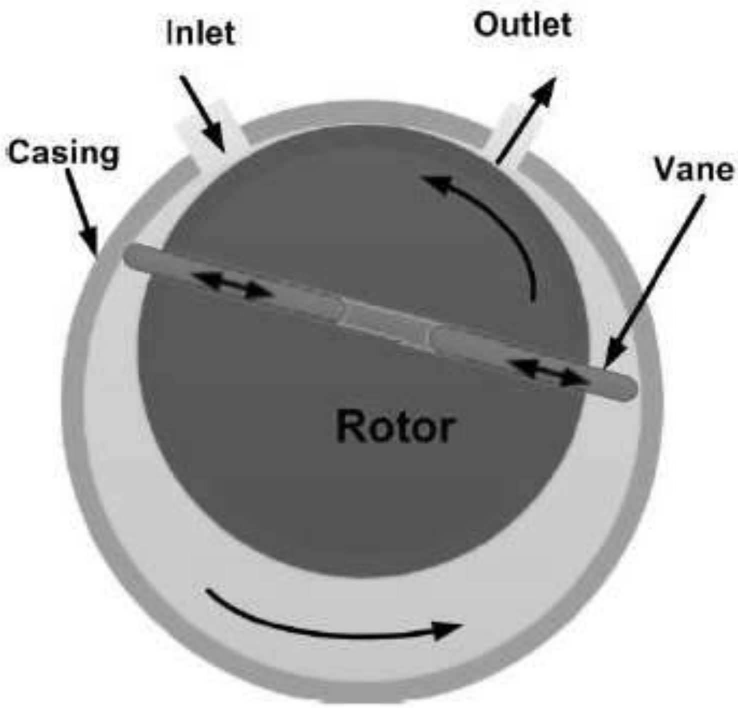


Figure 2-9 - Working Principle of Vane Pump
(S. N. Joshi - Mechatronics and Manufacturing Automation)

2.1.2.1 **Unbalanced Vane Pumps**

In the unbalanced vane pump, the rotor and cam housing are eccentric (Figure 2-10). The pump suction is generated in the region where the vanes begin to move outward. The liquid is carried around the rotor by the vanes, which form a seal with the housing and the end plates, and it is discharged as the vanes are forced back into the rotor slots by the eccentric housing. Unbalanced vane pumps can be either fixed or variable displacement pumps. In the fixed displacement pump the rotor-housing eccentricity is constant and, hence, the displacement volume is fixed. A constant volume of fluid is discharged during each revolution of the rotor. Variable displacement can be provided if the housing can be moved with respect to the rotor. This movement changes the eccentricity and, therefore, the displacement. In addition to sliding vanes, rolling vanes and swinging vanes are also available in unbalanced vane pumps.

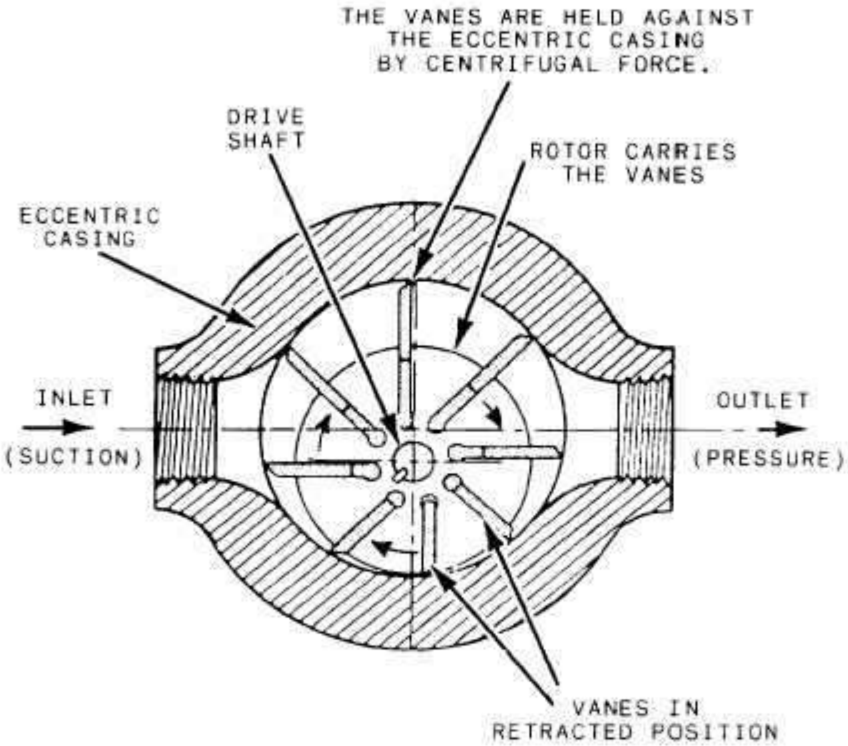


Figure 2-10 - Unbalanced Vane Pump (www.constructionmanuals.tpub.com)

Although the vane tips are held against the housing, still a small amount of leakage exists between rotor faces and body sides. Also, the vanes compensate to a large degree for wear at the vane tips or in the housing itself. The pressure difference between outlet and inlet ports creates a large amount of load on the vanes and a significant amount of side load on the rotor shaft which can lead to bearing failure (U.S. Army Material Command, 1993).

2.1.2.2 **Balanced Vane Pumps**

Hydraulic balance is achieved in the balanced vane pump in which the rotor is in an elliptic housing (Figure 2-11). This configuration creates two diametrically opposed displacement volumes. Pressure loading still occurs in the vanes, but the two identical pump halves create equal but opposite forces on the rotor. It leads to the zero-net force on the shaft and bearings. Thus, lives of pump and bearing increase significantly. Also, the sounds and vibrations decrease in the running mode of the pump. Balanced vane pumps are necessarily fixed displacement machines (U.S. Army Material Command, 1971), (Basic Hydraulics and Components, 2006).

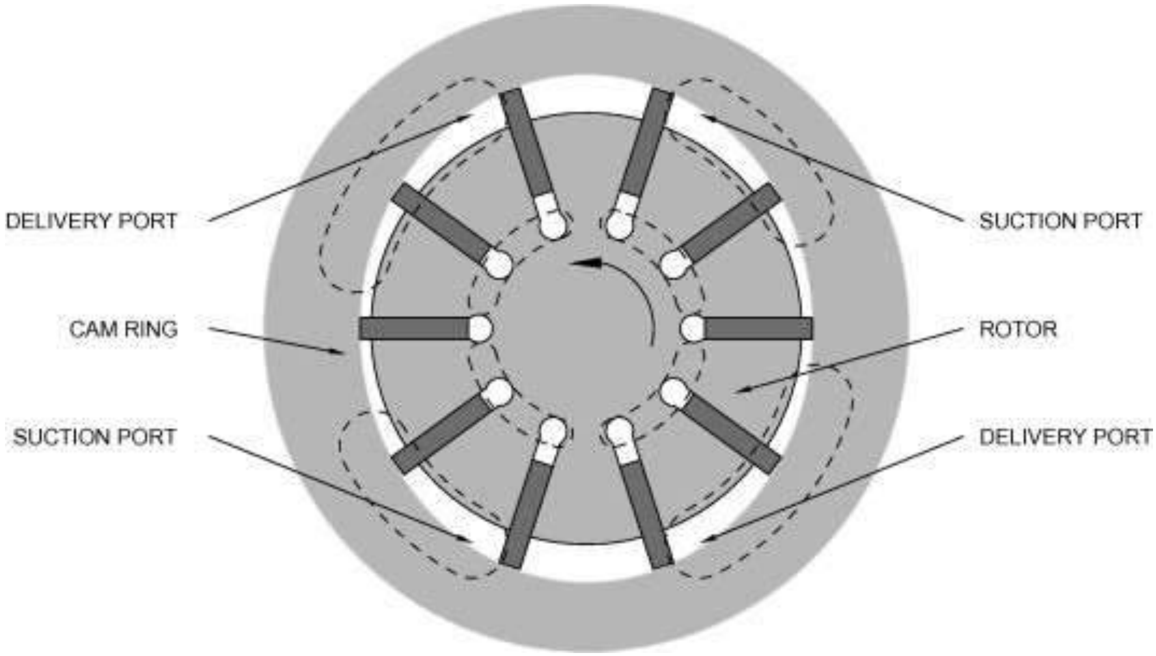


Figure 2-11 - Balanced Vane Pump (www.sciencedirect.com)

2.1.3 Piston Pumps

The applications for which the piston pump is well suited are determined by its two principal advantages high-pressure capability and high volumetric efficiency. They are easy to convert to the variable displacement type. Thus, they can operate with various control types. In addition, the piston pump can operate at speeds over 2,000 rpm, is available in a wide range of output ratings, and provides a compact, lightweight unit for high power applications, low noise level when flow path is linear, and better system economy in the higher power ranges. The piston pumps provide advantages including: high efficiency, ease of operation at high pressure, ease of conversion to the variable displacement type, and various applicable control types. Piston pumps are classified by the motion of the piston relative to the drive shaft. There are three categories-axial, radial, or rotating (U.S. Army Material Command, 1971), (Basic Hydraulics and Components, 2006).

Figure 2-12 - Piston Pump
(www.indiamart.com)

2.1.3.1 **Axial piston Pumps**

Axial piston pumps are fixed displacement pumps which converts rotary motion of the input shaft into an axial reciprocating motion of the pistons. These pumps have a number of pistons (usually an odd number) in a circular array within a housing which is commonly referred to as a cylinder block, rotor or barrel. Output can be controlled by manual, mechanical, or pressure-compensated controls. An axial-piston pump is shown in Figure 2-13. Rotary motion is converted to axial piston motion by means of the thrust cam, or wobble plate, mounted on the drive shaft. Variable displacement volume is provided by the internal valve arrangement (Department of the Army, 1997).

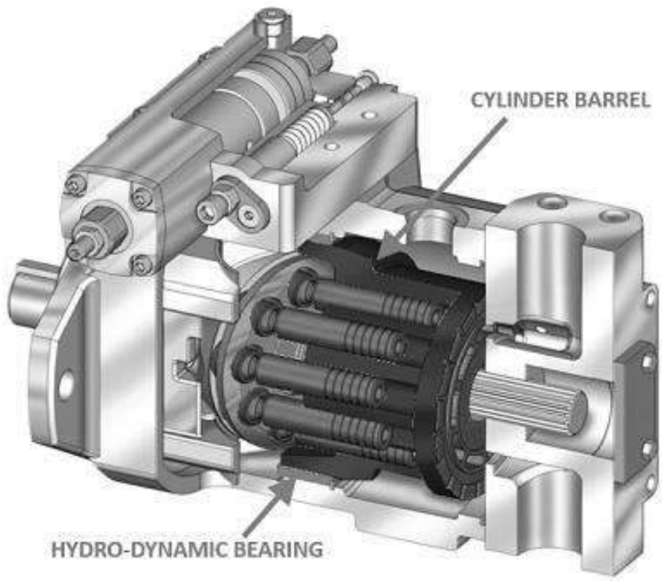


Figure 2-13 - Axial Piston Pump
(www.oilgear.com)

These pumps have sub-types as (U.S. Army Material Command, 1971):

a. Bent Axis Piston Pumps

Figure 2-14 shows the schematic of bent axis piston pump. In these pumps, the reciprocating action of the pistons is obtained by bending the axis of the cylinder block. The cylinder block rotates at an angle which is inclined to the drive shaft. Then it is turned by the drive shaft through a universal link. It is set at an offset angle with the drive shaft. Also, it contains a number of pistons along its perimeter. These pistons are forced in and out of their bores as the distance between the drive shaft flange and the cylinder block changes.

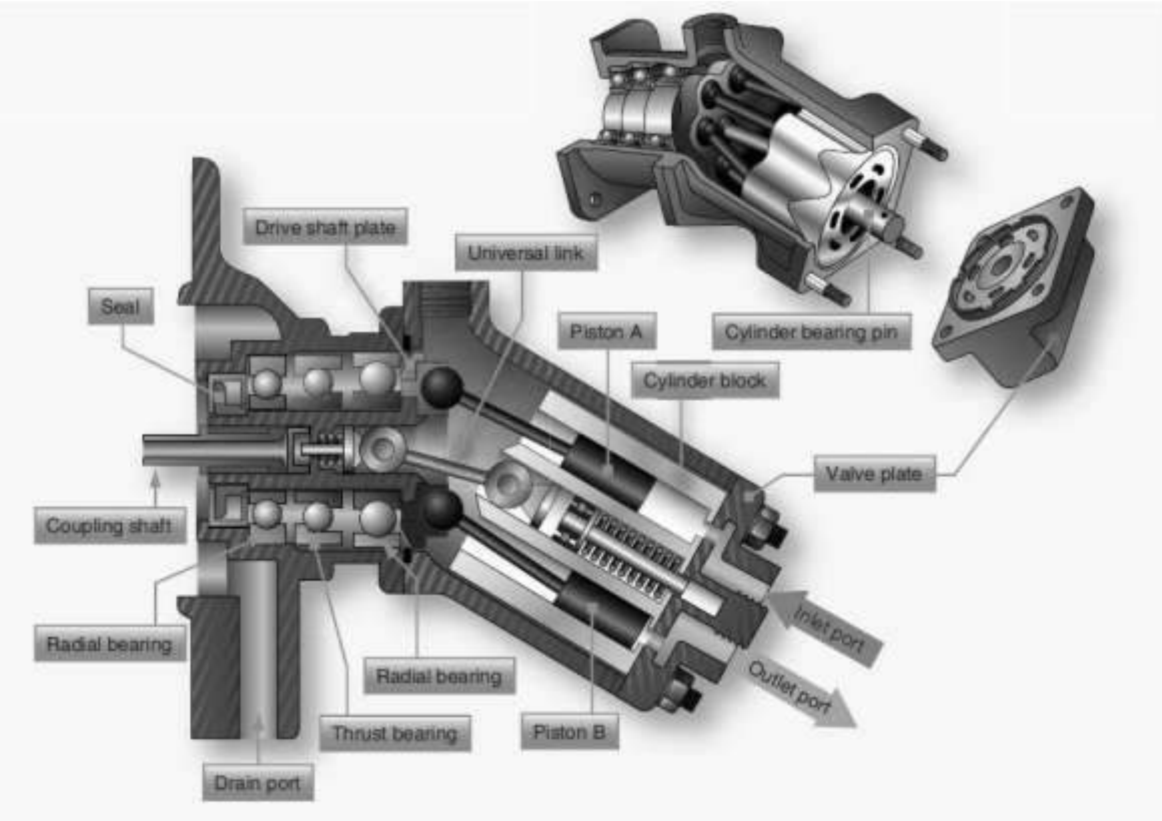


Figure 2-14 - Bent Axis Piston Pump
(www.flight-mechanic.com)

The volumetric displacement (discharge) of the pump is controlled by changing the offset angle. It makes the system simple and inexpensive. The discharge does not occur when the cylinder block is parallel to the drive shaft (offset angle is 0°). The offset angle can vary from 0° to 40°. The flow rate of the pump varies with the offset angle θ. The total fluid flow per stroke can be given as:

$$V_d = n\,A\,D\,\tan\theta \tag{1.5}$$

The flow rate of the pump can be given as:

$$V_d = n\,A\,D\,N\,\tan\theta \tag{1.6}$$

$$\tan\theta = \frac{S}{D} \tag{1.7}$$

where S is the piston stroke, D is piston diameter, n is the number of pistons, N is the speed of pump and A is the area of the piston.

b. Swash Plate Axial Piston Pump

A swash plate is a device that translates the rotary motion of a shaft into the reciprocating motion. It consists of a disk attached to a shaft as shown in Figure 2-15. If the disk is aligned perpendicular to the shaft, the disk will turn along with the rotating shaft without any reciprocating effect. Similarly, the edge of the inclined shaft will appear to oscillate along the shaft's length. This apparent linear motion increases with increase in the angle between disk and the shaft (offset angle).

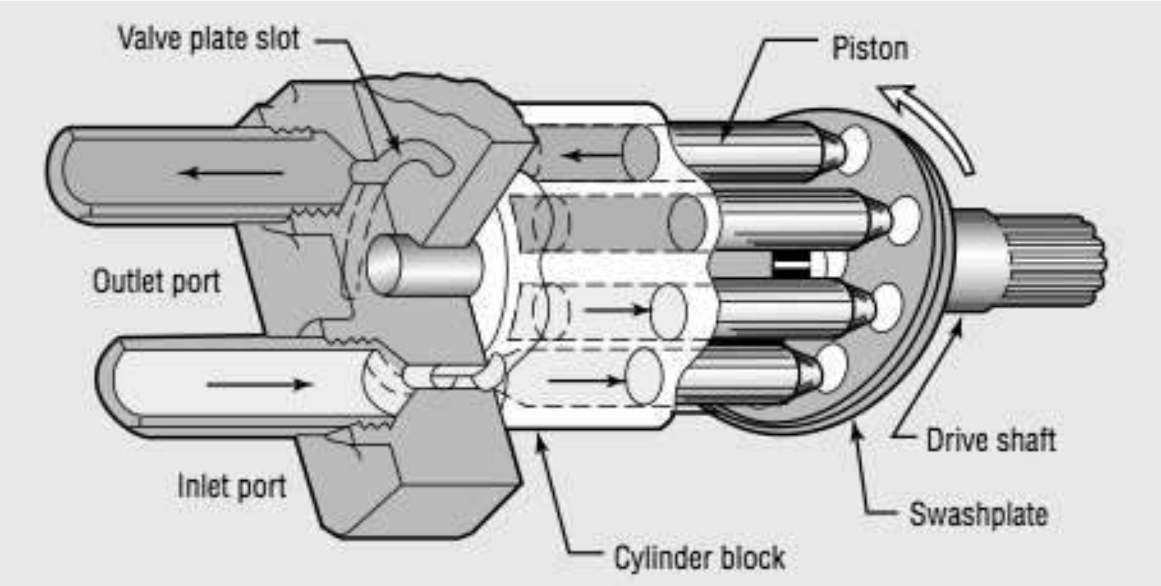


Figure 2-15 - Swash Plate Axial Piston Pump (www.hydraulicspneumatics.com)

In swash plate axial piston pump a series of pistons are aligned coaxially with a shaft through a swash plate to pump a fluid. The axial reciprocating motion of pistons is obtained by a swash plate that is either fixed or has variable degree of angle. As the piston barrel assembly rotates, the piston rotates around the shaft with the piston shoes in contact with the swash plate. The piston shoes follow the angled surface of the swash plate and the rotational motion of the shaft is converted into the reciprocating motion of the pistons. When the swash plate is perpendicular to the shaft, the reciprocating motion to the piston does not occur. As the swash plate angle increases, the piston follows the angle of the swash plate surface and hence it moves in and out of the barrel. The piston moves out of the cylinder barrel during one half of the cycle of rotation thereby generating an increasing volume, while during other half of the rotating cycle, the pistons move into the cylinder barrel generating a decreasing volume. This reciprocating motion of the piston results in the drawing in and pumping out of the fluid. Pump capacity can be controlled by varying the swash plate angle with the help of a separate hydraulic cylinder.

2.1.3.2 **Radial Piston Pumps**

The typical construction of radial piston pump is shown in Figure 2-16. The piston pump has pistons aligned radially in a cylindrical block. It consists of a shaft, a cylinder barrel with pistons and a rotor containing a reaction ring. The shaft directs the fluid in and out of the cylinder. Pistons are placed in radial bores around the rotor. The piston shoes ride on an eccentric ring which causes them to reciprocate as they rotate. The eccentricity determines the stroke of the pumping piston. Each piston is connected to inlet port when it starts extending while it is connected to the outlet port when start retracting. For initiating a pumping action, the reaction ring is moved eccentrically with respect to the shaft axis. As the cylinder barrel rotates, the pistons on one side travel outward. This draws the fluid in as the cylinder passes the suction port of the shaft. It is continued till the maximum eccentricity is reached. When the piston passes the maximum eccentricity, the shaft is forced inwards by the reaction ring. This forces the fluid to flow out of the cylinder and enter in the discharge (outlet) port of the pin (Department of the Army, 1997).

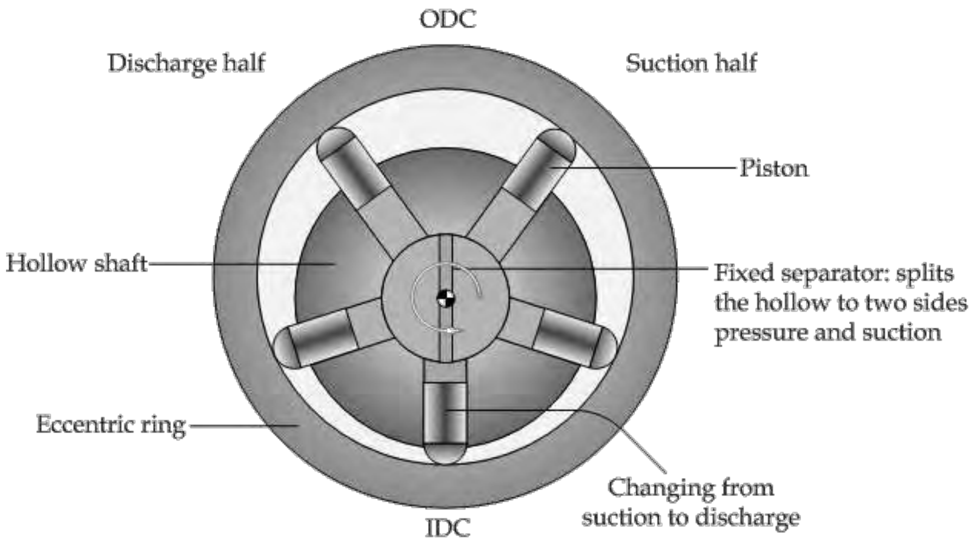


Figure 2-16 - Radial Piston Pump (www.mohsen.blogspot.com)

2.1.3.3 **Rotating Piston Pumps**

The rotating piston pump (sometimes called the rotary abutment pump) has three parallel synchronous shafts. Piston rotors are mounted on the outside shafts and seal dynamically against the cylindrical housing. The rotor mounted on the centre shaft forms an abutment valve. The rims of the piston rotors pass through a bucket cut in the centre rotor. Except when the rim is meshed with the abutment valve, a rolling contact seal is maintained between the rotors. Liquid is drawn into the right cylinder, pumped through to the left cylinder, and discharged by the left piston (Joshi, 2010).

2.1.4 **Screw Pumps**

A screw pump is an axial-flow gear pump and operates by rotating two or three screw shafts, which are aligned and engaged in parallel, to continuously convey a volume structured with screw leads. Figure 2-17 shows a two-rotor screw pump with helical gears. Liquid is introduced at the two ends and discharged at the center. The seal is formed by the contact of the two gears at the intersection of their addenda and by the small clearance between the gears and the pump housing. In pumps employing double helical gears, the thrust loads are balanced. This design is frequently employed in large pumps. Screw pumps are especially applicable where quiet operation is essential. In screw pumps, the gears must be in contact at the intersection of their addenda. This contact plus the minimum clearance at the outside diameter of the gears, provides a series of sealed chambers along the length of the screws. Screw pumps can also be arranged with three rotors. The center gear is the driver, and no timing gears are necessary. Because of the low noise level and reduced pulsation, they are used as hydraulic pressure sources for hydraulic lifts and submarines. Because gear pumps are less susceptible to working fluid contamination, they are also used for pumping cutting oils and lubricants (Basic Hydraulics and Components, 2006).

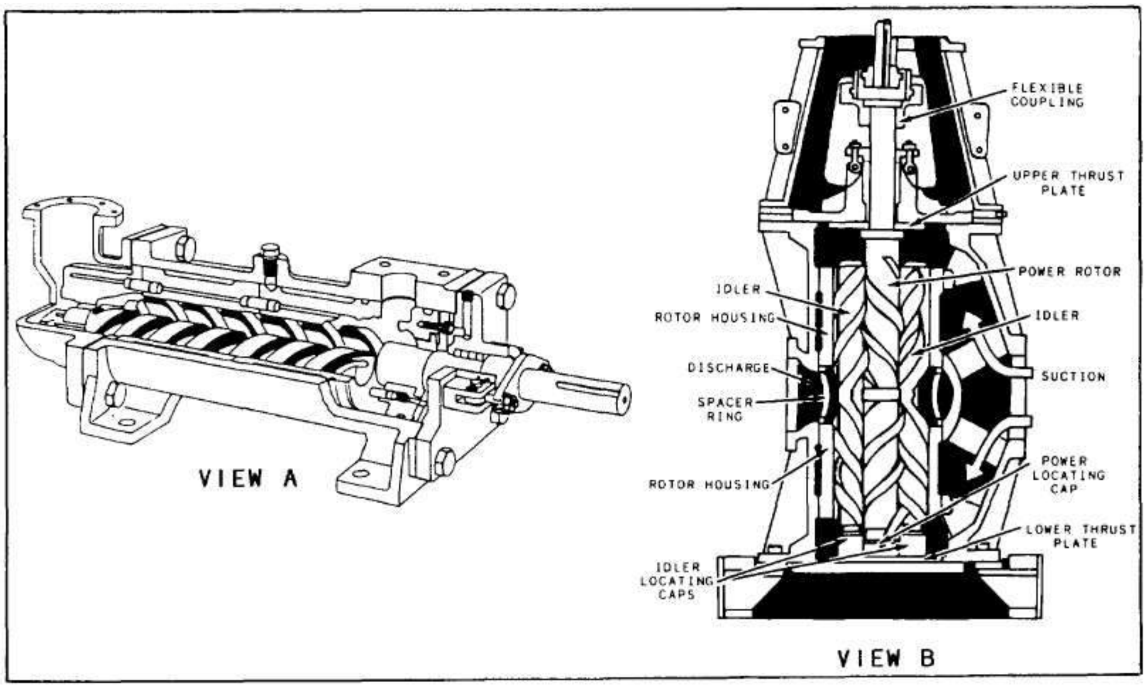


Figure 2-17 - Screw Pump
(MMC A. Beasley, Jr. - Fluid Power)

2.1.5 **Bellows Pumps**

The bellows is constructed of thin flexible metals or elastic non-metals. Compression of the bellows by the actuator provides the pumping action. Liquid flow is controlled by check valves at the inlet and outlet ports. Bellows pumps are not generally used in hydraulic systems and find only limited applications in other fields. One area of application has been the medical field where they have the advantage of simple construction and are easily fabricated of inexpensive disposable materials. Bellows pumps also are very effective pumps for liquids that are contaminated with solid particle material, and for abrasive slurries. There are no small clearance areas for the particles to damage or close (U.S. Army Material Command, 1971).

2.1.6 **Diaphragm Pumps**

The diaphragm is a flexible disk attached about its circumference to the pump housing. The actuator moves the diaphragm in a reciprocating motion which provides the pumping action. Diaphragms can be flexible sheets of metal or elastic non-metals. Liquid flow is controlled by check valves at the inlet and outlet ports. Diaphragm pumps are primarily low pressure, pulsating flow pumps and are not often used for hydraulic power sources. They are occasionally used as hydraulic power sources where a remote pump is required, such as in high temperature or corrosive environments. The valve head, containing the inlet and outlet valves, can be located in the hostile environment and the diaphragm unit located in a remote place. One of the most common uses of diaphragm pumps is the fuel pump on automobiles. Diaphragm pump advantages include simple construction, no lubrication problems, high volumetric efficiency, and insensitivity to contamination. Major disadvantages are the low pressures obtainable, pulsating flow, and low flow rates (MMC Beasley, 1990).

2.1.7 **Reciprocal Pumps**

These pumps have pistons installed at right angles against the pump rotating shaft in a plane including the shaft. The pistons reciprocate with alternately moving cranks or eccentric cams. Pumps suffer from discharge pulsation because they have a fewer pistons than the axial and radial types. These pumps are used for high pressure applications such as construction vehicles, cargo loading machines, and press machines. In addition, they can handle water or water-containing fluids as the working fluid because of the construction to be wider area applicable for lubrication (U.S. Army Material Command, 1971).

2.1.8 **Connection Between Pump and Drive Motor**

The physical connection between the hydraulic pump and its drive motor is not technically a hydraulic component. However, it is an important part of the hydraulic system, and in many cases, may be the weakest link in the power system. There are a number of methods for coupling the drive motor output shaft to the hydraulic pump input shaft. Some of the more common methods are keys and pins, flexible couplings, universal joints, clutches, and splines. The most frequently used connector in hydraulic systems is the spline. Splines offer the advantage of being able to transmit the maximum load with the smallest coupling diameter. In addition, they are self-centred, tend to equally distribute the load, and are simple to manufacture with standard gear cutting equipment. Their major disadvantage is the problem of wear. Even the best designed splines are subject to relative motion of the parts and are difficult to lubricate (U.S. Army Material Command, 1971).

2.2 Actuators/Motors

An actuator is a device for converting hydraulic energy to mechanical energy, and thus has a function opposite that of a pump. An actuator, or fluid motor, can be used to produce linear, rotary, or oscillatory motion.

Figure 2-18 - Hydraulic Cylinder
(W. Durfee, Z. Sun, J. V. Ven - Fluid Power System Dynamics)

2.2.1 **Linear Actuators/Cylinders**

A linear actuator or hydraulic cylinder converts fluid power pressure and flow to mechanical translational power force and velocity (Figure 2-18). Cylinders can push and pull, and when mounted around a joint, for example, as is done in an excavator, can actuate rotary motion. Cylinders come in single acting (push only), single acting with spring return and double-acting (push-pull) type. The rest of the section will focus on double-acting cylinders, which are most common in hydraulic applications. A cutaway illustration of a typical double-acting cylinder used for industrial applications is shown in Figure 2-19.

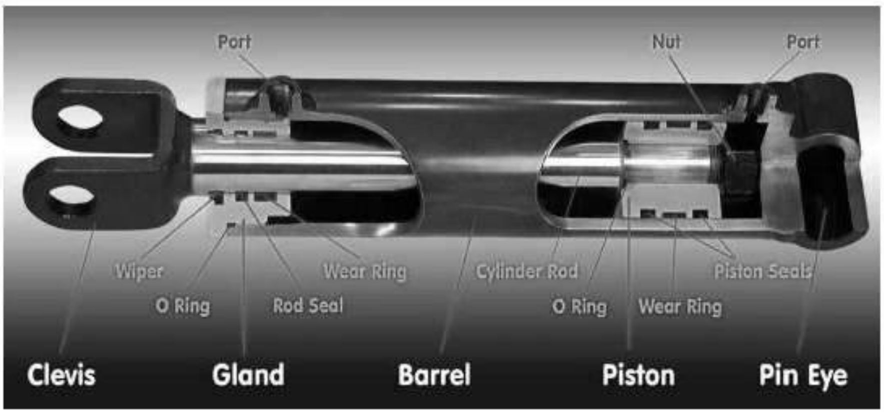


Figure 2-19 - Double Acting Cylinder
(Yuken Kogyo Co. - Basic Hydraulics and Components)

The end of the cylinder where the rod emerges is called the rod end and the other end is called the cap end. This distinction is important for modelling because the rod side of the piston within the cylinder has less surface area than the cap side of the piston. For the same pressure a double-acting cylinder can push with much greater force than it can pull. Cylinders have considerable friction, particularly around the piston because of its large circumference with wrap-around seals. The rod seal tends to be even tighter than the piston seals to prevent leaking of hydraulic oil, but because of the smaller circumference, rod seals play less of a role in cylinder friction. While the design of a cylinder is complex, the dynamic model used for most simulations simply captures the pressure-force transformation and sometimes includes the cylinder friction and leakage around the piston seal. The defining equations for an ideal, friction-free, leakless cylinder are (Durfee, Sun, & Ven, 2015):

$$F = P\ A \tag{1.8}$$

$$v = \frac{Q}{A} \tag{1.9}$$

The piston force depends on the difference in pressure across the piston, taking into account the area on each side. Piston force is:

$$F_P = P_1\ A_{cap} - P_2\ A_{rod} \quad (3.3) \tag{1.10}$$

where all pressures are gauge pressures with respect to atmospheric pressure. And the piston velocity is:

$$v = \frac{Q_1}{A_{cap}} = \frac{Q_2}{A_{rod}} \quad (3.4) \tag{1.11}$$

The rod area is the piston annulus around the rod and is:

$$A_{rod} = \frac{\pi}{4}(bore^2 - roddia^2) \tag{1.12}$$

Another consequence of the different areas on either side of the piston is that the oil flow in one port will not be equal to the flow out the other port. If the return line to the reservoir is long or the return line valve has small orifices, then the pressure builds up on the rod side of the cylinder when pushing can be significant and must be modelled. The overall efficiency of a cylinder is given by the ratio of the output mechanical power to the input fluid power:

$$\eta = \frac{F\ v}{P_i\ Q_i} \tag{1.13}$$

where P_i and Q_i refer to either the cap or rod side depending on whether the piston is pushing or pulling. Cylinder efficiency can be split into two parts, the force efficiency:

$$\eta_f = \frac{F}{P_i\ A_i} \tag{1.14}$$

and the volumetric efficiency:

$$\eta_v = \frac{A_i\ v}{Q_i} \tag{1.15}$$

with the overall efficiency being the product $\eta = \eta_v\ \eta_f$. Using these relations, Equations 1.10 and 1.11 can be modified for a non-ideal cylinder with friction and leakage:

$$F_P = \left(P_1\, A_{cap} - P_2\, A_{rod}\right) \eta_f \qquad (1.16)$$

$$v = \frac{Q_1}{A_{cap}}\, \eta_v = \frac{Q_2}{A_{rod}}\, \eta_v \qquad (1.17)$$

2.2.1.1 **Classification of Linear Actuators**

The many types of linear actuators which are available give rise to several criteria for classification (U.S. Army Material Command, 1971).

i. ***Rotating or non-rotating:*** In a rotating actuator the cylinder, rod, and piston can rotate. In many applications, such as on rotary machine tools, this feature is necessary to allow unrestricted motion of the piston rod. To permit stationary mounting of the fluid connections, a rotating seal is required. The non-rotating linear actuator, in which the cylinder is not free to rotate, is the most widely used fluid motor.

ii. ***Piston or plunger:*** The piston and rod assembly in a piston-type linear actuator serves to divide the cylinder volume into two separate chambers. The piston and attached sealing devices provide the seal between the two chambers. In a plunger-type there is no piston. The end of the reciprocating rod serves as the working face. The only seal provided is at the point where the plunger passes through the end of the cylinder. An external force is required to move the plunger into the cylinder. Both types provide a longer stroke and permit the use of the highest pressure.

iii. ***Rod classification:*** Linear actuators can also be classified as to rod type. A cylinder with one piston rod is termed a single-rod actuator. A double-rod actuator has piston rods extending from both ends of the cylinder. A telescoping rod consists of a series of nested rods which provide a long extension. Such rods are useful for applications requiring a long stroke but with only limited space available for the unextended rod. A positional rod is used where the stroke is split up into two or more portions. The cylinder can be actuated to any one of the positions.

iv. ***Cylinder action:*** The type of cylinder action is important in the specification of linear actuators. An actuator can be single-acting or double-acting. The single-acting type can move the piston rod in only one direction by the application of hydraulic pressure. A plunger-type actuator is a single-acting actuator. In the double-acting actuator, liquid pressure can be applied to either side of the piston, thereby providing a hydraulic force in both directions. Springs, external forces, or a combination of both can be used to assist return of the piston rod or plunger.

v. ***Single, tandem, and dual actuators:*** Yet another means of actuator classification is its assembly. Assemblies of actuators can be designed to obtain various types of cylinder operation. A tandem actuator is one in which two or more piston and rod combinations are assembled as a rigid unit with all pistons mounted on a single rod. Tandem pistons can also be designed to provide a large working area (and thus large forces for a given pressure) for a small cylinder diameter. The piston and rod assemblies of a dual actuator are not fastened together as in the tandem actuator. In most dual actuator designs, a given piston acts on another only in one direction. Tandem and dual actuators are frequently used in hydropneumatic systems where air is used as the power source and a hydraulic fluid is used for control.

vi. ***Cushioned or non-cushioned type:*** In non-cushioned actuators, no provision is made for controlled acceleration or deceleration of the piston assembly. Therefore, such units have speed and inertia limitations imposed at both ends of the stroke. Cushioned actuators are designed to enable the kinetic energy of the moving piston to be absorbed at the ends of the stroke and thereby reduce peak pressures and forces.

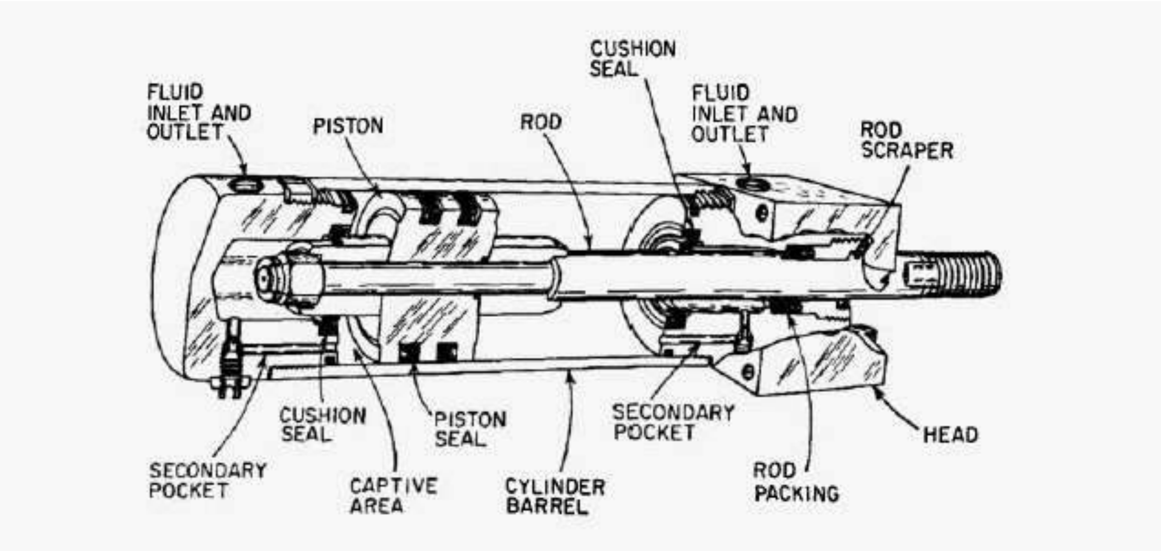


Figure 2-20 - Double Acting Cushioned Linear Actuator
(MMC A. Beasley, Jr. - Fluid Power)

2.2.2 Rotary Actuators or Motors

As in the case of a linear actuator, the function of a rotary actuator, or rotary fluid motor, is to convert hydraulic energy into mechanical energy. Rotary motors are usually rated in terms of the theoretical torque developed per 100 psi of inlet or differential pressure. The actual running torque and the stalled torque may be from 60% to 90% of the theoretical torque, depending on the type of motor. The running volumetric efficiency may vary from about 75% to 95%, again depending on the particular motor. The highest operating efficiency occurs near the rated torque and speed. The desirable features of the various types of rotary motors include:

- The ability to suddenly start, stop, and reverse without motor damage
- The ability to operate as a pump for braking
- A higher horsepower-to-weight ratio than any other conventional power source
- An infinitely-variable speed range
- The ability to operate through zero speed for overrunning loads
- The ability to accommodate contaminants in the fluid.

Rotary fluid motors are essentially rotary pumps operating in reverse. The mechanical characteristics of a particular rotary motor are nearly identical with those of the corresponding pump (Basic Hydraulics and Components, 2006).

Figure 2-21 - Motor
(www.rodem.com)

2.2.2.1 **Gear Motors**

Gear motors, like gear pumps, can be classified as external or internal gear units. Also, like gear pumps, they are fixed displacement devices. External gear motors include the gear-on-gear units such as the spur gear motor. Internal gear motors include the crescent seal types and the Gerotor unit (Department of the Army, 1997).

- ***Gear-on-gear motors:*** In the gear-on-gear motor, rotary motion is produced by the unbalanced hydraulic forces on the gear teeth which are exposed to the inlet pressure. An example is the spur gear motor which has the same mechanical features as the spur gear pump shown in Figure 2-5. Bearing loads generated by the hydraulic unbalance are high, as in the case with unbalanced gear pumps.

- ***Crescent seal motors:*** The crescent seal motor employs an inner and outer gear with a crescent-shaped seal separating the teeth during part of the revolution. Its operational features are the reverse of those of the crescent seal pump illustrated in Figure 2-7. Motor units of this type are suitable for high-speed, low-power operations at low-to-moderate pressure. Starting torque and running efficiencies are low.

- ***Gerotor motors:*** The Gerotor motor (see Figure 2-8 for corresponding pump) is suitable for high-speed operation and exhibits relatively high starting torque efficiency. Volumetric efficiency is relatively low and leakage rates are high at most speeds. The cost of Gerotor motors is relatively high in comparison with the other gear motors.

2.2.2.2 **Vane Motors**

Most vane motors are of the balanced type because hydraulic unbalance causes large radial bearing loads which limit the use of unbalanced vane motors to low pressure operation and applications where weight and space considerations do not preclude the use of large, heavy bearings. Therefore, most vane motors have a mechanical configuration similar to that of the balanced vane pump shown in Figure 2-11 and are thus fixed displacement units. To accommodate starting and low speed operation, it is usually necessary to provide a force -in addition to the centrifugal force- to move the vane radially outward. Springs are commonly used for this purpose. As with vane pumps, rolling and swinging vanes can also be used in vane motor design. The overall running efficiencies of vane motors are typically 80% to 85% (U.S. Army Material Command, 1971).

2.2.2.3 Limited-Rotation Motors

Limited-rotation motors, or rotary actuators, provide an oscillating power output. A variety of such units is available, all of which consist of one or more fluid chambers and a movable surface against which the fluid pressure is applied. Both vane-type and piston-type motors can be used to obtain an oscillatory output (U.S. Army Material Command, 1971).

- ***Vane type:*** There are two types of limited-rotation vane motors, the single-vane and the double-vane. The single-vane unit consists of a cylindrical housing, a shaft with a single vane, a barrier which limits the vane rotation, and end pieces which support the shaft . High-pressure liquid enters on one side of the vane, forcing the vane to rotate to the barrier. In the double-vane unit, the high-pressure fluid enters on one side of a vane and is ported through the shaft to the corresponding side of the other vane. In both the double and the single-vane units, seals are maintained between the rotor and the barriers and between the vanes and the housing.

- ***Piston type:*** Piston-driven actuators are available in several configurations designed to produce an oscillating output. The helix-spline unit employs a shaft with a helical screw which passes through the piston. A guide rod prevents rotation of the piston. The piston-rack unit consists of two or more pistons which provide the rack for a rack-and pinion system.

2.2.2.4 Piston Motors

Piston motors which generate a continuous rotary output motion (as opposed to linear actuators) can be classified in terms of the piston motion: axial, radial, or rotary. They can be fixed or variable displacement devices. They can operate at high pressures and have high volumetric efficiencies. The power-to-weight ratio of piston motors is not as favorable as that of gear and vane motors. Relative cost per horsepower is high (Merritt, 1967).

- ***Axial-piston type:*** The operation of an axial-piston motor is essentially the same as that of an axial piston pump except for the direction of flow (Figure 2-13). The high-pressure liquid introduced through the motor inlet forces the piston assembly against the thrust cam or wobble plate. The angular application of this force causes the plate to rotate and this rotation is transmitted by the shaft. The displacement can be varied by changing the angle of the thrust cam. Leakage is low under both running and stalled conditions.

- ***Radial-piston type:*** The radial-piston motor is also essentially its pump counterpart operating in reverse (Figure 2-16). Liquid enters the piston chamber through a central pin. The piston is forced radially outward against the thrust ring, thereby producing a force tangent to the piston chamber. The resulting torque causes the shaft to rotate. This motor type exhibits very high volumetric efficiencies and high torque, and is well-suited for low-speed application because of the small mass of the rotating parts.

- ***Rotary-piston type:*** The rotary-piston motor is the same as the rotary-piston pump except for the flow direction. Weight and space-to-power ratios are high, and cost per horsepower is usually high.

2.3 Valves

Control valves are essential and appear in all fluid power systems. Valves are sometimes categorized by function, which includes directional control valves for directing fluid flow to one or the other side of a cylinder or motor, pressure control valves for controlling the fluid pressure at a point and flow control valves for limiting the fluid flow rate in a line, which in turn limits the extension or retraction velocities of a piston. Valves are also characterized by the number of ports on the valve for connecting input and output lines and by the number of operating positions that the valve can assume. For example, a 3-way, 2-position valve commonly found in pneumatic systems has three ports for connecting supply line, exhaust or reservoir line and output line to the cylinder and two positions. In one position the supply line connects to the cylinder line extending the piston. In the other position the exhaust line connects to the cylinder retracting the piston, assuming the piston has a spring return. On/off valves can only be in the states defined by their positions while proportional valves are continuously variable and can take on any position in their working range. A servo valve is a proportional valve with an internal closed-loop feedback mechanism to maintain precise control over the valve behavior. Example valves are shown in Figure 2-22 (Durfee, Sun, & Ven, 2015).

Figure 2-22 - Types of Control Valves. Left to right: Hand-operated directional valve. On-off miniature, solenoid actuated valve. Precision proportional valves. High precision flapper-nozzle hydraulic servo valve
(W. Durfee, Z. Sun, J. V. Ven - Fluid Power System Dynamics)

Valve Types

Valves are classified according to their function in the hydraulic system. These basic types are pressure control valves, directional control valves, and flow control valves. Most valves can be regarded as some combination of these basic types.

2.3.1 Pressure Control Valves

A pressure control valve either limits the pressure in various circuit components or changes the direction of all or part of the flow when the pressure at a certain point reaches a specified level. Such controls are directly or indirectly actuated by some system-pressure level (U.S. Army Material Command, 1971).

2.3.1.1 Relief Valves

A relief valve limits the maximum pressure that can be applied to the part of the system to which it is connected. It acts as an orifice between the pressurized region and a secondary region at a lower pressure. In most applications, the relief valve is closed until the pressure attains a specified value. Then the flow through the valve increases as the system pressure rises until the entire system flow is vented to the low-pressure region. As the system pressure decreases, the valve closes (Basic Hydraulics and Components, 2006).

a. In the direct-acting pressure relief valve, the system pressure acts directly on the spring (Figure 2-25). These valves are small and have a simple structure for their capacity. However, they are likely to exhibit high-pressure override (a pressure characteristic observed when a fluid starts flowing from a valve and reaches the rated flow rate) and chattering. Therefore, they are used to control the pressure of relatively small flows and low-pressure systems or when relief valve conditions are expected only rarely. Valves of the size 1/8, in particular, are very popular for pilot pressure controls.

b. The differential relief valve can be constructed with a much lighter spring than the direct-acting type because the system pressure acts over only a differential area.

c. In the pilot operated relief valve, pressurized liquid is used to assist the spring (Figure 2-24). The liquid passes from the supply line through a restricted passage to a control chamber where it acts on a plunger to add to the spring force. The force is limited by a small capacity, direct-acting pilot relief valve. The pilot-operated relief valve is usually specified for systems which require frequent relieving. Placed in a vent circuit, they can perform remote control, unloading, or two-pressure control.

d. The solenoid controlled relief valve is a combination of a pilot operated relief valve and a small solenoid operated directional valve (Figure 2-23). Sending electrical signals to the directional valve can remotely unload pump pressure or conduct the two- or three-pressure control in hydraulic circuits.

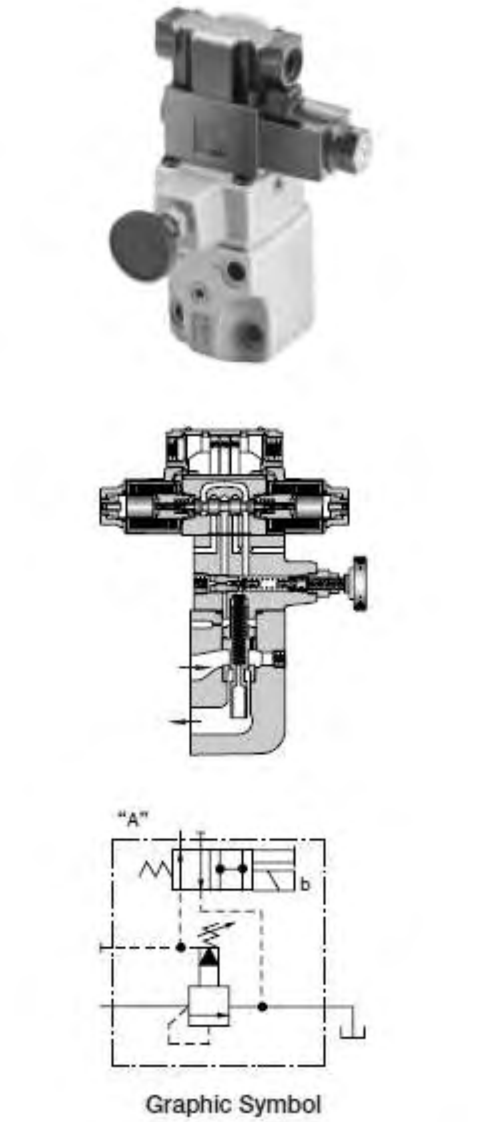


Figure 2-25 - Solenoid Controlled Relief Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

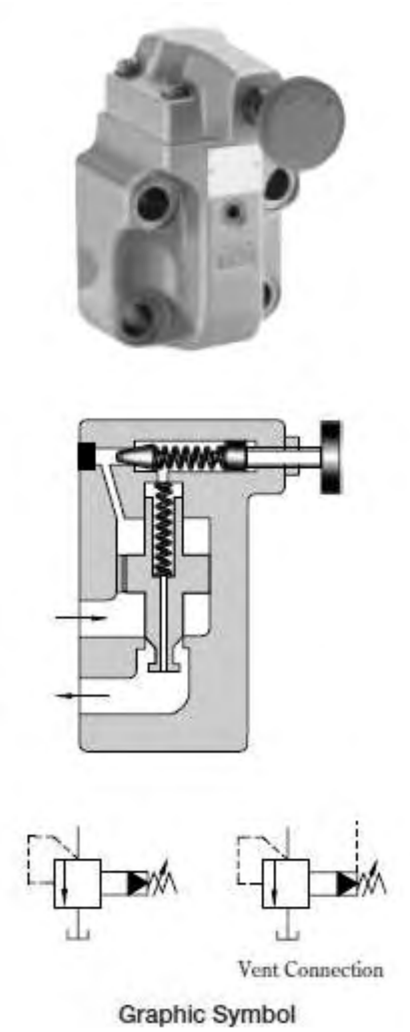


Figure 2-24 - Pilot Operated Relief Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

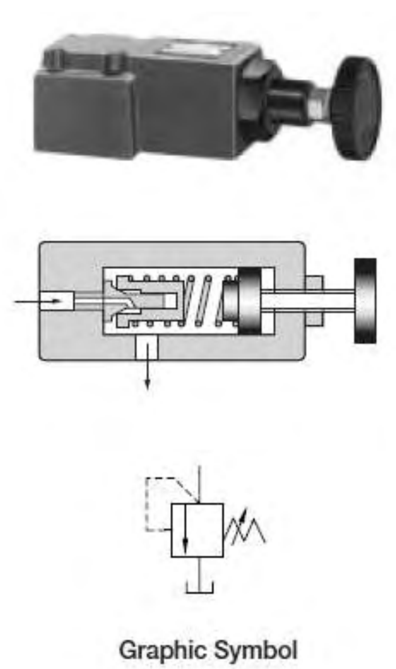


Figure 2-23 - Direct type Relief Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.1.2 **Unloading Valves**

An unloading valve provides a vent to a low pressure area when a specified pilot pressure is applied (Figure 2-26). They are used to operate pumps at the minimum load in an accumulator circuit or in a high-low pump circuit. A typical application is in a double pump system where a high volume, low pressure pump is completely loaded at maximum pressure, while a low volume, high pressure pump continues to develop higher pressure (Basic Hydraulics and Components, 2006).

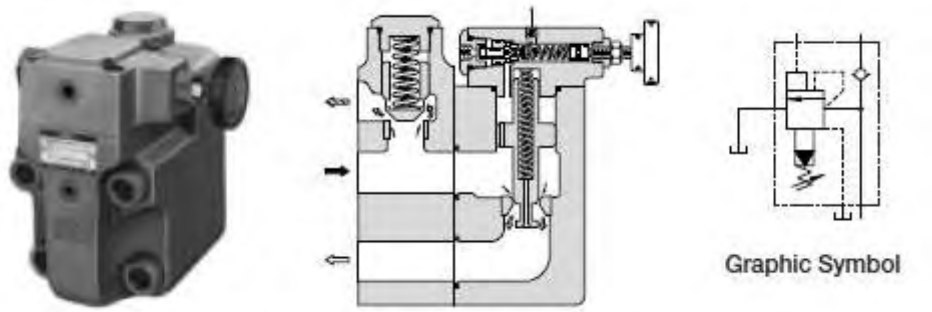


Figure 2-26 - Unloading Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.1.3 **Load Dividing Valves**

In a circuit with two pumps operating in series, the load can be equally divided between the pumps by a load dividing valve. The low pressure pump discharge is connected to a larger piston area. The low pressure flow tends to open the valve and relieve the pump discharge to the reservoir. The high pressure pump discharge is connected to the small area and, assisted by the spring, tends to close the valve. The ratio of the pressure produced at the low pressure pump to the discharge pressure of the high pressure pump is the same as the valve area ratio (U.S. Army Material Command, 1971).

2.3.1.4 **Brake Valves**

These valves smoothly stop actuators that have a large inertia force (Figure 2-27). When a directional control valve is closed, a relief valve at the outlet side releases the accumulating pressure while maintaining the circuit pressure at the preset level. At the inlet side, a check valve feeds the flow, supply of which has been blocked by the directional control valve, to reduce a risk of cavitation (Department of the Army, 1997).

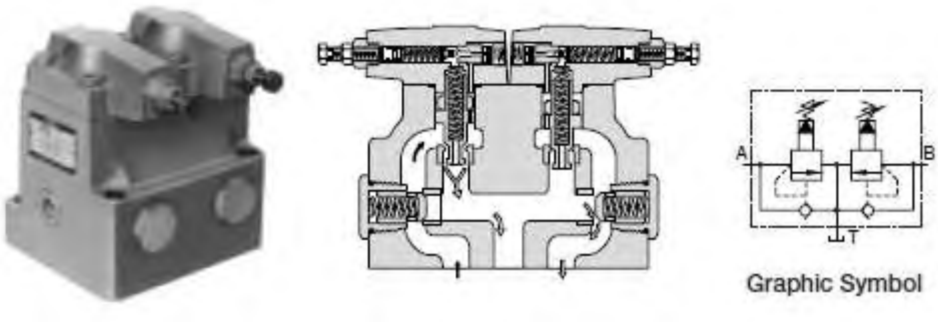


Figure 2-27 - Brake Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.1.5 **Sequence Valves**

The order of flow to different parts of a hydraulic system can be controlled by sequence valves. These valves control the sequential operation of two or more actuators. If the inlet pressure exceeds a preset level, they deliver effective pressure to the outlet side. This is accomplished by controlling minimum pressure. Either an internal or external pilot pressure can be applied. The valves can serve as pressure holding valves to maintain hydraulic pressure in a circuit. Figure 2-28 shows an externally piloted, externally drained sequence valve. The inlet pressure must reach a prescribed value before the flow is allowed to pass through the valve (Basic Hydraulics and Components, 2006).

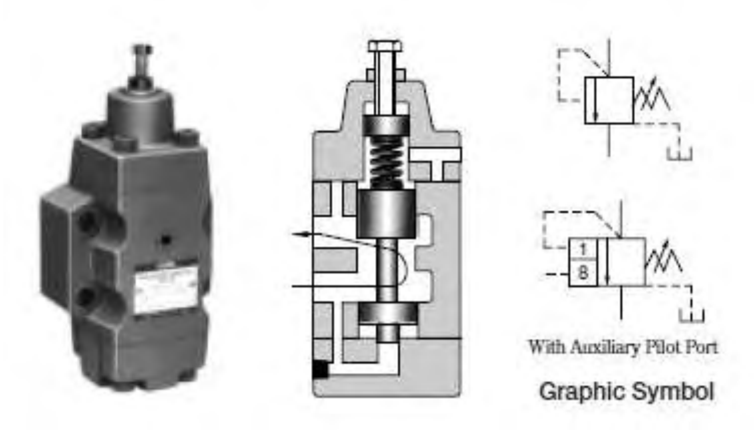


Figure 2-28 - Sequence Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.1.6 **Counterbalance Valves**

The counterbalance back pressure valve can be used to allow free flow in one direction but restricted flow in the opposite direction (Figure 2-29). These valves maintain hydraulic pressure in a hydraulic system or load backpressure on a cylinder. If the inlet pressure exceeds a preset level, flow is released to keep the pressure constant. They are accompanied with a check valve that allows the flow for lifting a cylinder up to freely pass. This valve can be used, for example, to prevent the weight of a vertically mounted piston from causing the piston to descend. The spring setting produces a back pressure on the piston which counterbalances the force of gravity (U.S. Army Material Command, 1971).

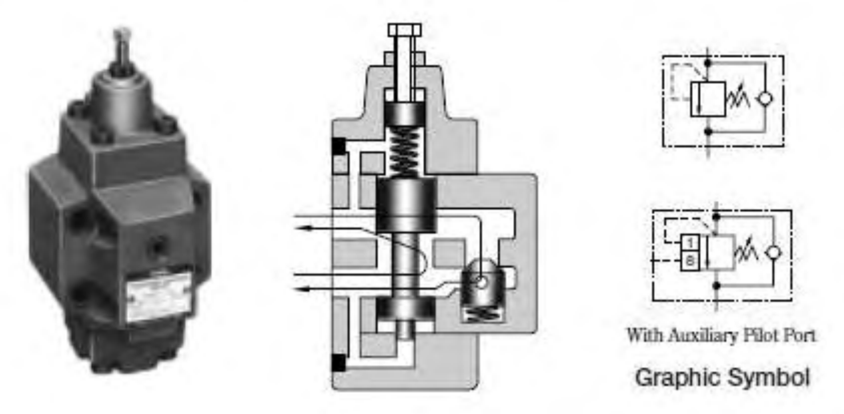


Figure 2-29 - Counterbalance Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.1.7 **Pressure Reducing Valves**

Pressure regulator or pressure reducing valves set hydraulic circuit pressure equal to or below a pressure in the main circuit. When the outlet pressure reaches a preset level, the valve opens, and the balanced piston moves to throttle a passage to keep the outlet pressure constant. The outlet pressure is maintained constant, regardless of the inlet pressure. In the type shown in Figure 2-30, the outlet pressure is balanced against a spring (Basic Hydraulics and Components, 2006).

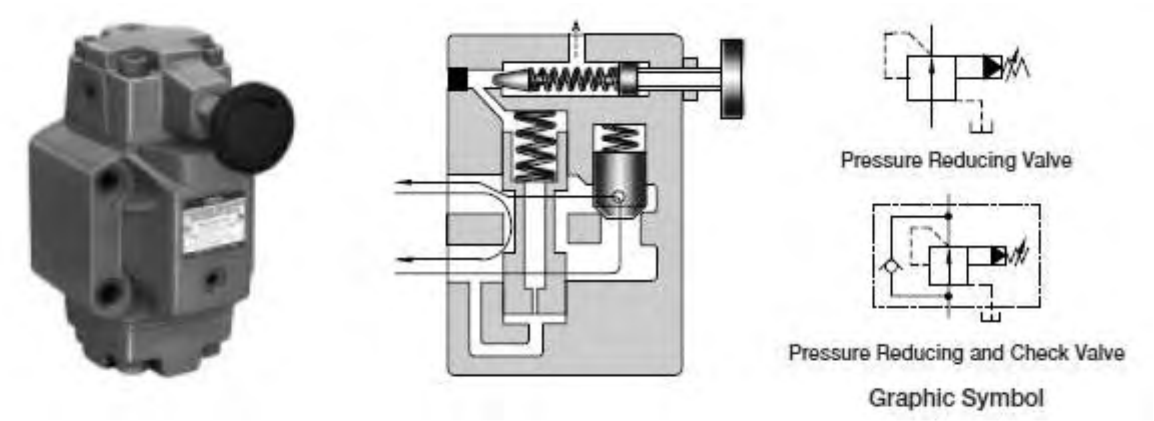


Figure 2-30 - Pressure Reducing and Check Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.1.8 **Pressure Switch**

Pressure switches are used in hydraulic systems to make or break an electrical circuit at a preset hydraulic pressure (Figure 2-31). The system pressure acts against an adjustable spring used to preset the switch. A sensing component made of semi-conducting materials detects the pressure. When the pressure reaches the specified value, the sensor is activated. The signal can be used to actuate a variety of control elements. Although the pressure switch is not a valve, it is a valuable control element in valve systems (Basic Hydraulics and Components, 2006).

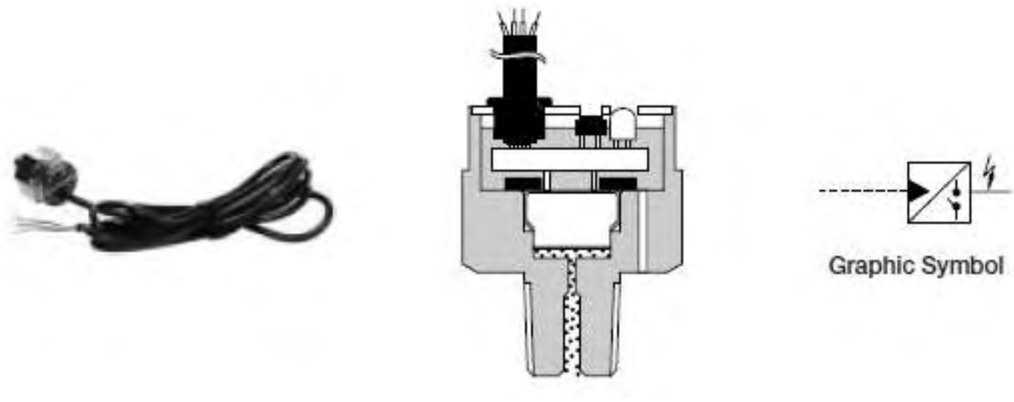


Figure 2-31 - Pressure Switch
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.1.9 **Balancing Valves**

These valves are combination valves that have pressure reducing and counterbalancing functions developed for applications such as a hydraulic balance circuit in a vertical machining center. When the pressure reducing function is employed, the outlet pressure is maintained at the preset level for pressure reduction, regardless of the inlet pressure. If the counterbalancing function is employed, the outlet pressure is maintained at the preset level for pressure relief, which is higher than the pressure reducing (U.S. Army Material Command, 1993).

2.3.1.10 **Hydraulic Fuses**

A hydraulic fuse employs a fragile diaphragm or similar device which fractures at a preset pressure. It can thus be used as a substitute for, or in conjunction with, a pressure control valve. Hydraulic fuses can be used with safety valves to prevent hydraulic fluid loss under normal operating conditions. They usually do not have automatic reset capabilities. It is necessary to manually replace the diaphragm if the hydraulic fuse is actuated (U.S. Army Material Command, 1971).

2.3.2 **Direction Control Valves**

Directional control valves control start/stop, directions, and acceleration/deceleration of hydraulic cylinders and motors. They can be used in a various applications, and a wide range of products is available. They are often used to control the operation of actuators. They can be categorized into three types: spool, poppet, and ball. The spool type can be either a sliding type or a rotary type. The former is the most popular for pressure balancing and high capacity. The poppet type offers excellent leak-tight capability (zero leak) for its poppet-seat contact. The ball type is an alternative for the poppet: a ball is used instead of a poppet (Basic Hydraulics and Components, 2006).

2.3.2.1 **Classification of Directional Control Valves**

- ***Classification by Port/Position Count***

The port count indicates the number of connectable lines, and the position count indicates the number of changeovers in the directional control valves. Figure 2-32 lists the classifications. The valves with four ports and three positions are very popular. The four ports include: pump port (P), tank port (T), and cylinder ports (A and B). The symbols are often appended with graphic symbols of the directional control valves (Basic Hydraulics and Components, 2006).

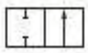
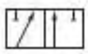
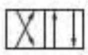
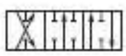
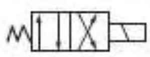
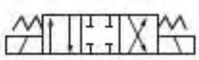
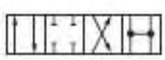
Classification		Graphic Symbol	Remarks
No. of Ports (Connections)	Two Ports		This valve has two ports to open/close a hydraulic line.
	Three Ports		This valve has three ports for changeover from the pump port to two ways only.
	Four Ports		This valve has four ports for a wide variety of purposes, including moving the actuator forward and backward or stopping it.
	Multiple Ports		This valve has five or more ports for special purposes.
No. of Positions	Two Positions		This valve has two positions.
	Three Positions		This valve has three positions.
	Multiple Positions		This valve has four or more positions for special purposes.

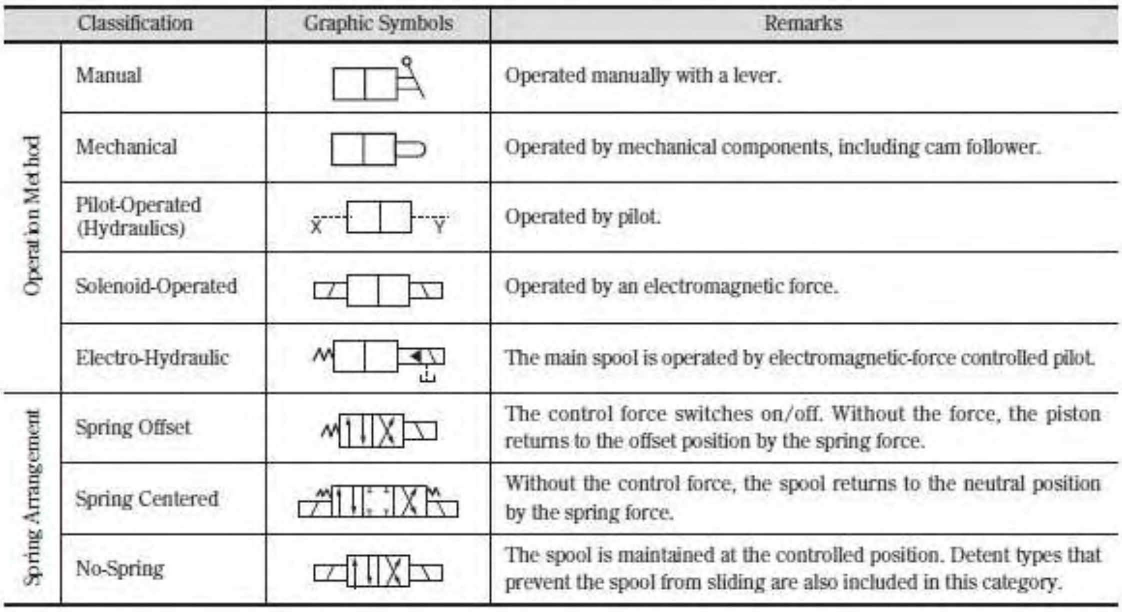
Figure 2-32 - Classification by Port/Position Count
(Yuken Kogyo Co. - Basic Hydraulics and Components)

- ***Classification by Spool Types***

The directional control valves in hydraulic systems must work such that when the spools are in a neutral position, the fluid flow patterns meet the purpose of the systems, in addition to causing reversible motion of the hydraulic cylinders and motors. For example, take the “three-position” valve in Figure 2-32. This closed-center valve (all ports are closed when the spool is in the neutral position) locks the cylinder at its position. The pressure in the pump line is maintained at a preset level for the relief valve or the variable pumps, and other systems can be operated as desired. On the other hand, the center-bypassed valve (ports P and T are open when the spool is in the neutral position) unloads the pump line while locking the cylinder, which is desirable for energy saving: lower heat generation and reduced pump load. However, the overall performance of the hydraulic system should be taken into consideration to choose the best valves for it (Basic Hydraulics and Components, 2006).

- ***Classification by Operation Method and Spring Arrangement***

These valves are classified according to the function, operation method, and spring arrangement. The operation method is classified into: manual, mechanical, hydraulic, solenoid-operated, electro-hydraulic, and pneumatic. The spring arrangement is classified into: spring offset for the two-position type, spring centred for the three-position type, and no spring for both two- and three-position types. The no-spring type includes the detent type, which holds the spool position at a certain position. Figure 2-33 shows classification of the valves according to the operation method and spring arrangement (Basic Hydraulics and Components, 2006).



Classification		Graphic Symbols	Remarks
Operation Method	Manual		Operated manually with a lever.
	Mechanical		Operated by mechanical components, including cam follower.
	Pilot-Operated (Hydraulics)	X Y	Operated by pilot.
	Solenoid-Operated		Operated by an electromagnetic force.
	Electro-Hydraulic		The main spool is operated by electromagnetic-force controlled pilot.
Spring Arrangement	Spring Offset		The control force switches on/off. Without the force, the piston returns to the offset position by the spring force.
	Spring Centered		Without the control force, the spool returns to the neutral position by the spring force.
	No-Spring		The spool is maintained at the controlled position. Detent types that prevent the spool from sliding are also included in this category.

Figure 2-33 - Classification by Operation Method/Spring Arrangement
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.2.2 Structure and Characteristics of Directional Control Valves

i. Solenoid Operated Directional Valves

These valves control the flow direction in hydraulic circuits, electrically operated with manual switches, limit switches, or pressure switches. They are the most popular for use in practical hydraulic systems. Three types of solenoids are available: for direct current (DC), for alternating current (AC), and with a rectifier. The solenoids can be grouped into wet and dry types in respect to the structure. The solenoid operated directional valves are commercially available in the many sizes, among them, the 1/8 size is most often selected for practical hydraulic systems. Figure 2-35 shows the 1/8 solenoid operated directional valve. Aside from the maximum working pressure, flow, tank-line back pressure, and changeover frequency, performance characteristics of the solenoid operated directional valves include power consumption.

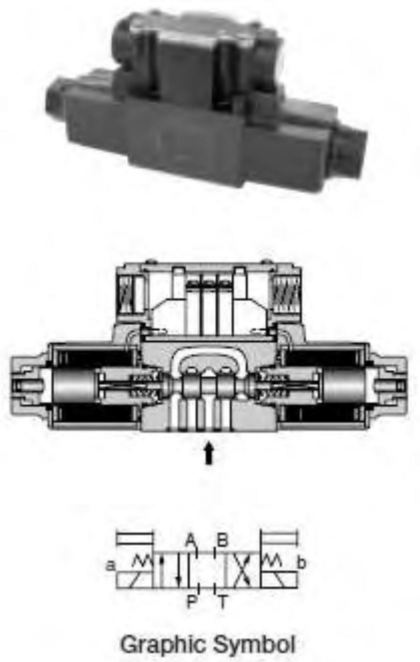


Figure 2-35 - Solenoid Operated Directional Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

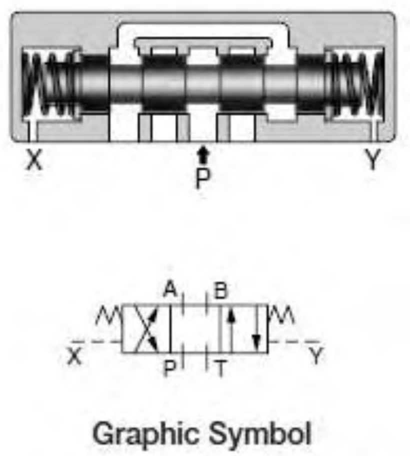


Figure 2-34 - Pilot Operated Directional Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

ii. Solenoid Controlled Pilot Operated Directional Valves

These valves are a combination of a small solenoid operated directional valve and a large pilot operated directional valve. The small four-way solenoid valve is used for directional control of the pilot line. The main valve (main spool) provides directional control of the main line. The pilot operated directional valve includes the spring offset, spring-centered and no-spring types. Figure 2-36 shows a solenoid controlled pilot operated directional valve of the 3/4 size.

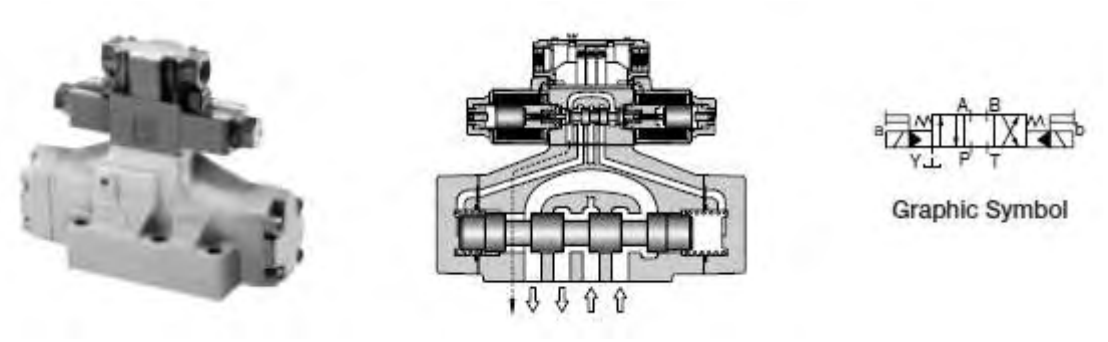


Figure 2-36 - Solenoid Controlled Pilot Operated Directional Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

iii. Poppet Type Solenoid Controlled Pilot Operated Directional Control Valves

These valves are solenoid controlled pilot operated directional valves, which are made multi-functional by granting individual poppet functions, such as directional control, flow control, and pressure control. They consist of a main valve with four poppets, a solenoid operated directional valve for the pilot line, and a pilot selector valve. These valves are used in large-scale hydraulic systems including press and compressing machines.

iv. Pilot Operated Directional Valves

These valves perform spool changeover by the hydraulic pilot. They are useful when the pilot directional control valve and the main directional control valve should be installed distant to each other. Figure 2-34 shows the pilot operated directional valve.

v. Manually Operated Directional Valves

These valves are manually operated to change the direction of hydraulic flow. They are available in the spool-operated and rotary types. These valves are structured in two types, detent and spring. The detent type maintains the spool position at the time the lever is operated, and the spool is returned to the center by the spring force. The spring type springs back the spool to either position of changeover, as is in the two-position valve. Figure 2-38 shows the spool-operated type and rotary type directional valves, respectively.

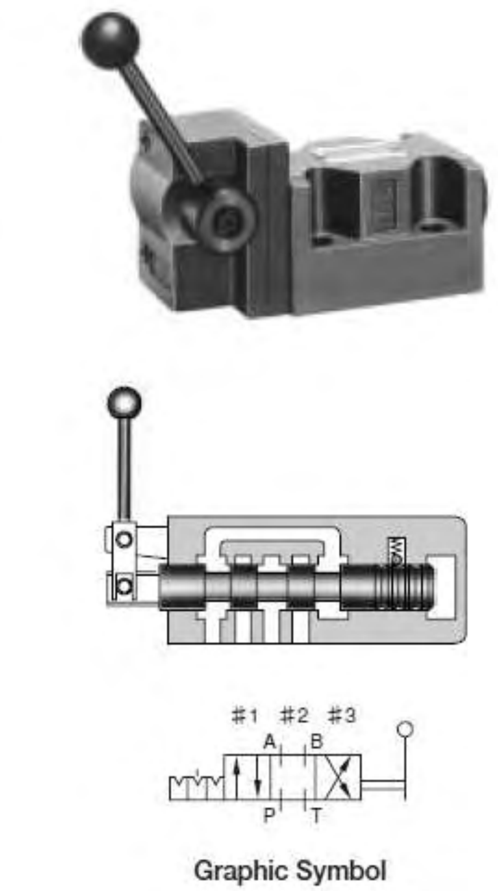


Figure 2-38 - Manually Operated Directional Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

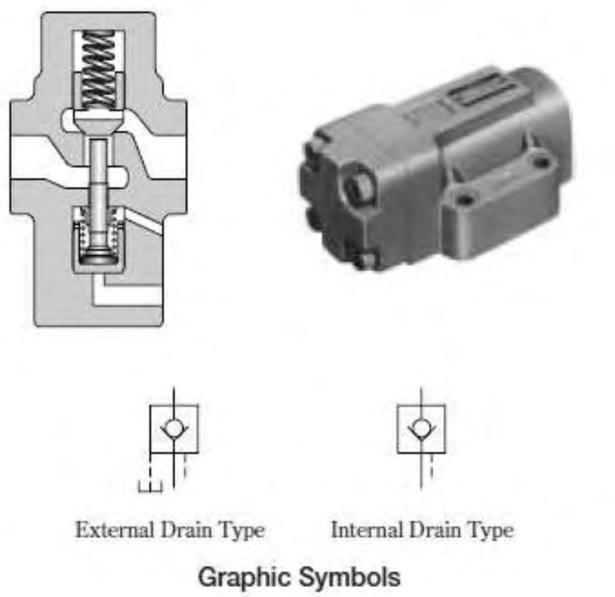


Figure 2-37 - Pilot Operated Check Valve (Yuken Kogyo Co. - Basic Hydraulics and Components)

vi. Check Valves

These valves allow free flow in one direction, while preventing flow in the reverse direction. When flow is in the normal direction, the liquid pressure acts against the spring tension to hold the poppet off the seat. When flow stops, the spring seats the poppet and liquid cannot pass in the reverse direction. Check valves are often incorporated in sequence valves or pressure reducing valves to let them open in one direction and bypass the free flow in the reverse direction.

vii. Pilot Operated Check Valves

These check valves have a pilot piston, which works with remote pressure to open the closed check valve, allowing reversed flow. Two types are available, standard and decompression. The decompression type has a main poppet valve combined with a decompression valve. When the pilot pressure increases to lift the pilot piston, the decompression poppet valve opens first, and then the main poppet after the pressure is reduced. These valves are used to moderate shock caused by a sudden pressure release, which often occurs during the return stroke of a press process. They maintain the actuator position and system pressure, opening/closing the seat, therefore, internal leakage can be kept at minimum. When they are used with a restrictor or a counterbalance valve that produces back pressure on the outlet side of reversed free flow, a counter force may work on the piston, which opens/closes the poppet continuously and vibrates the valves as a result. If the valves are in such a hydraulic circuit, the external drain type should be used. Figure 2-37 shows the pilot operated check valve.

viii. Multiple Control Valves

These valves have multiple functions, including direction, relief, and check control in one body and are mainly used for vehicles. Whether the functions are to be activated individually or simultaneously determines the circuit type: parallel, tandem, or series. When the directional control function is not working, the pump output flow goes through the valve into the tank. For this reason, power loss and heat generation are minimum. The mono block construction, which houses multiple spools in one body, and the sectional construction, which is modular by valve function, are available. For directional control, manual, solenoid, and proportional control valves are offered. (Basic Hydraulics and Components, 2006)

2.3.3 Flow Control Valves

Flow control valves are used to regulate the rate of liquid flow to different parts of a hydraulic system. Control of flow rate is a means by which the speed of hydraulic machine elements is controlled. The rate of flow to a particular system component is varied by throttling or by diverting the flow (U.S. Army Material Command, 1971).

2.3.3.1 Restrictors/One Way Restrictors

These valves regulate flow rates in hydraulic circuits (Figure 2-39). They have the advantages of plain structure, simple operation, and wide range of adjustment. On the other hand, they cannot accurately control the flow, even though the fixed restriction, the flow varies with the inlet-outlet differential pressure and the fluid viscosity. Therefore, these valves are placed where the pressure difference varies little, and high control accuracy is not required. The one-way restrictors regulate flow in one way, while allowing reversed flow to freely pass through (Basic Hydraulics and Components, 2006).

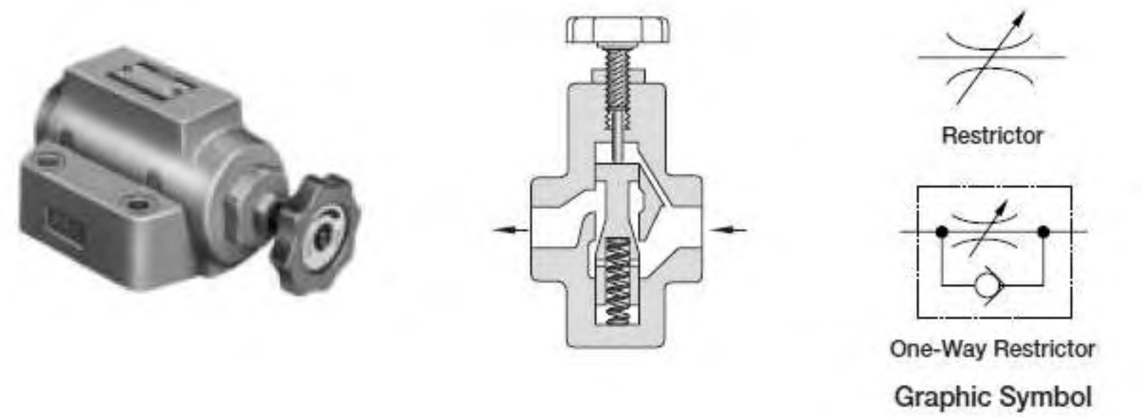


Figure 2-39 - Restrictor
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.3.2 Flow Control Valves/Flow Control and Check Valves

These valves consist of a pressure compensator (pressure reducing valve that keeps the pressure difference constant) and a restrictor (Figure 2-40). They maintain a constant flow rate, independent of the inlet-outlet differential pressure. Provided with a sharp-edge orifice, they can also work regardless of fluid temperature or viscosity. In a circuit where the flow rate is regulated to a low level, the control flow may be momentarily exceeded, leading to jumping of the actuator. This phenomenon is related to a time lag until the pressure compensating piston is properly positioned for flow control. To prevent the phenomenon, the piston stroke should be adjusted according to the inlet-outlet differential pressure. Flow control valves are basically used as follows (Basic Hydraulics and Components, 2006).

- ***Meter-In Control***

The control valve is connected in series with the cylinder inlet to directly control the input flow. Prior to the control valve, a relief valve is applied to excess flow, which escapes through a relief valve. In a circuit where load is applied in the direction of piston travel, the control valve may lose cylinder speed control.

- ***Meter-Out Control***

The control valve is connected in series with the cylinder outlet to directly control the output flow. Prior to the control valve, excess flow escapes through a relief valve to a tank. This circuit design is used for applications where the piston could move down faster than a control speed, as in the case of vertical drilling machines, or where there should always be a back pressure in the cylinder. Careful attention should be paid to the fact that the cylinder outlet pressure may rise above the relief pressure produced in the circuit.

- ***Bleed-Off Control***

The control valve is installed on a by-pass line to regulate flow to the tank and control the actuator speed. Compared to the other control circuits, this circuit works with small power consumption because discharge pressure of the pump is fully delivered to the load resistance. Given that the bleed flow is constant, the fluctuation of pump flow determines the actuator speed. In other words, the pump discharge flow directly influences the load and the pump's volumetric efficiency. This circuit does not allow for control of multiple actuators.

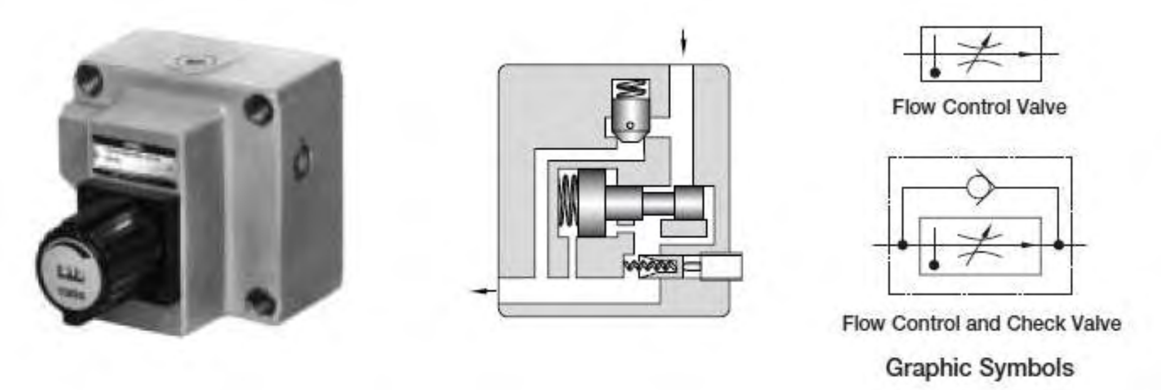


Figure 2-40 - Flow Control and Check Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.3.3 **Deceleration Valves/Deceleration and Check Valves**

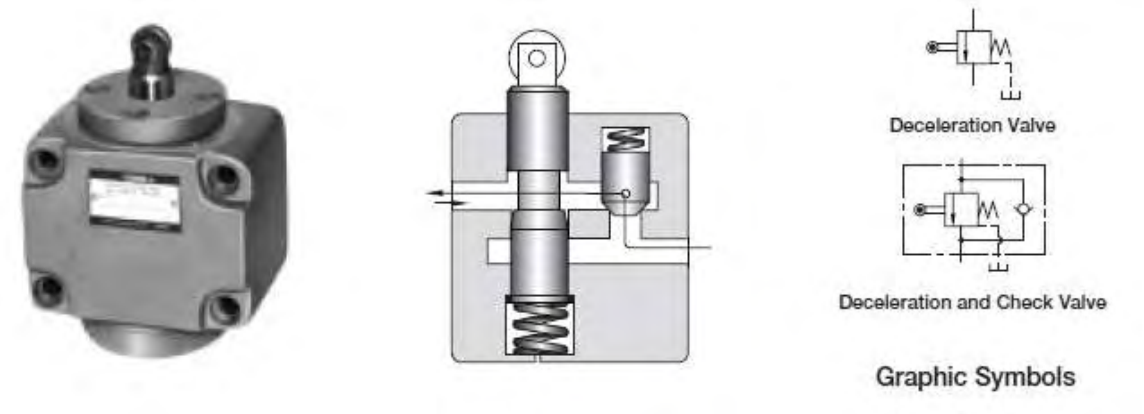


Figure 2-41 - Deceleration and Check Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

These valves continuously regulate flow rates, using a cam mechanism (Figure 2-41). Pushing the spool down decreases the flow rate for the normal open type and increases it for the normal close type. When the normal open type is installed to cushion the cylinder piston, accurate stroke end control is difficult. In this case, the restrictor and directional control valve should be adjusted so that the piston slowly returns to an intended position and then stops (Basic Hydraulics and Components, 2006).

2.3.3.4 **Feed Control Valves**

These valves are a combination of a flow control and check valve and a deceleration valve: they are used mainly for feed control of machine tools (Figure 2-42). Switching from rapid traverse to feed is made by a cam operation, and the feed speed is controlled with a flow control valve. Rapid return is free of cam actuation. Two-speed mode with two flow control valves is also available (Basic Hydraulics and Components, 2006).

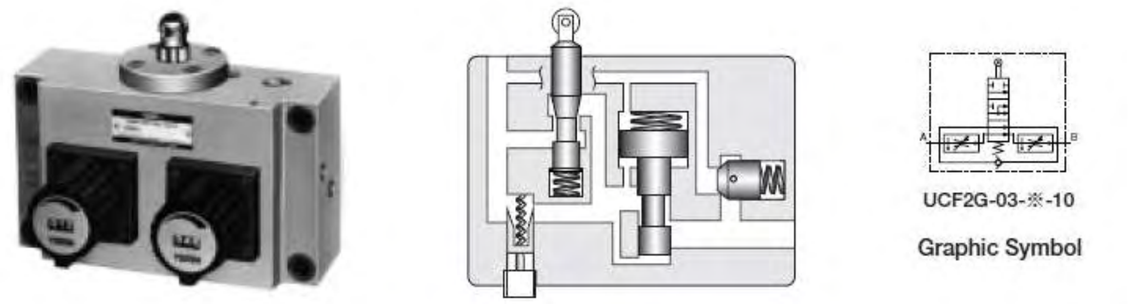


Figure 2-42 - Feed Control Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.3.5 **Flow divider valves**

Flow-divider valves utilize sliding elements to change the orifice area. They distribute the flow to multiple lines, each of which then has the same flow rate downstream of the valve. They are generally pressure-compensating valves and are frequently used to synchronize the motion of several linear actuators (U.S. Army Material Command, 1971).

2.3.4 **Modular Valves**

These valves, including the pressure control, flow control, and directional control types, have standardized mounting surfaces that conform to the ISO standards for solenoid operated directional valves. They are stacked on a base plate and referred to as sandwich valves or stack valves. They are available in nominal sizes of less than 1/8" and 1/8" to 1¼". These modular valves provide the following advantages.

- Compact stacking. They require very small mounting space.
- Easy circuit building. They eliminate the necessity for a large part of piping and assembly work, allowing easy and quick circuit building.
- Improved reliability. They are stacked together without piping: therefore, they are almost free from problems such as oil leakage, vibration, and noise.
- Easy maintenance due to integrated functions.

Note that the number of stacking layers is subject to the bolt strength. Also, due caution should be paid to the maximum flow and the pressure loss (Basic Hydraulics and Components, 2006).

2.3.5 **Logic Valves**

These valves consist of cartridge type elements and covers with pilot passages. Although they are two-port valves designed to simply open/close the poppets according to pressure signals from the pilot line, various types may be combined for direction, flow rate, and pressure control. Standard covers, which have several pressure signal ports and control valves, including pilot operated relief valves, are available for control purposes.
Logic valves provide fast-response, high-pressure, and high-flow control. They are typically applied to machines that involve high-speed actuator operation, such as die-cast machines, injection molding machines, and press machines. The logic valves have the following features (Basic Hydraulics and Components, 2006).

- Multifunction performance in terms of direction, flow, and pressure can be obtained by combining elements and covers.
- Various functions can be achieved, depending on the pilot line connection.
- Poppet-type elements virtually eliminate internal leakage and hydraulic locking. Because there are no overlaps, the response time is very short, permitting high-speed shifting.
- For high pressure, large capacity systems, optimum performance is achieved with low pressure losses.
- Since the logic valves are directly incorporated in cavities provided in blocks, the system faces fewer problems related to piping such as oil leakage, vibration, and noise, and higher reliability is achieved.
- Multi-function logic valves permit compact integrated hydraulic systems that reduce manifold dimensions and mass and achieve lower cost than that of the conventional types.

2.3.6 **Proportional Electro-Hydraulic Control Valves**

These valves and related devices work with electrical settings to provide continuous remote control of the pressure and flow in hydraulic circuits. For multi-stage pressure or flow control, various combinations of control valves have been used, however, proportional electro-hydraulic control valves and devices eliminate the necessity for those valves and greatly simplify the circuit architectures. These valves and devices, which permit remote control, allow hydraulic systems and their control rooms to be separately located. In other words, they are well suited for applications in large plants. Proportional electro-hydraulic control valves and devices, based on general-purpose hydraulic products, are easy to maintain and manage, highly resistant to contamination by fluids, and cost-effective for applications where very quick response and high accuracy are not required (Basic Hydraulics and Components, 2006).

2.3.7 **Servomechanism & Servo Valves**

A servomechanism is an automatic control system designed to operate in accordance with input control parameters. The mechanism continuously compares the input signal to the feedback signal to adjust the operating conditions for error correction. Commercially available servo systems vary according to their methods for error detection, amplification, communication, and output.

Hydraulic servo systems have been widely applied in general industrial areas, as well as in the airline, maritime, and military industries. Servo systems, capable of automatic position, speed, and force (load) control with high accuracy and quick response, are used for high-speed injection molding, die-casting, rolling mill, press machines, industrial robots, simulators, testing machinery, and table feeders.

A hydraulic servo system consists of an actuator (hydraulic motor/cylinder), servo valves, sensors, and a servo amplifier.

There are two types of electro-hydraulic servo valves: the pilot operated type, which drives a torque motor to amplify the hydraulic power with a nozzle flapper mechanism, and the direct type, which directly drives a spool with a linear motor and electrically provides feedbacks about the spool position. Mechanical servo valves are also available that have a stylus at one side of the spool to control the flow direction by the mechanical motion of the stylus (Basic Hydraulics and Components, 2006).

2.3.7.1 **Types**

i. Electro-Hydraulic Two-Stage Servo Valves

Nearly all types of servo valves are based on common principles. Electro-hydraulic two-stage servo valves generally operate with force feedback. Given that valve pressure drop is constant, the valves control the output flow in proportion to the input signal. Therefore, they can be used to drive a hydraulic cylinder or motor at a speed proportional to the input current. Figure 2-43 provides illustrations of an electro-hydraulic servo valve.

The valve contains identical torque motors in parallel, which serve as a nozzle flapper amplifier with movable coils and nozzles. Coil displacement always determines the spool position. To ensure reliable pilot operation, the valve is provided with a filter prior to the pilot line, as well as a high-performance line filter prior to the valve inlet.

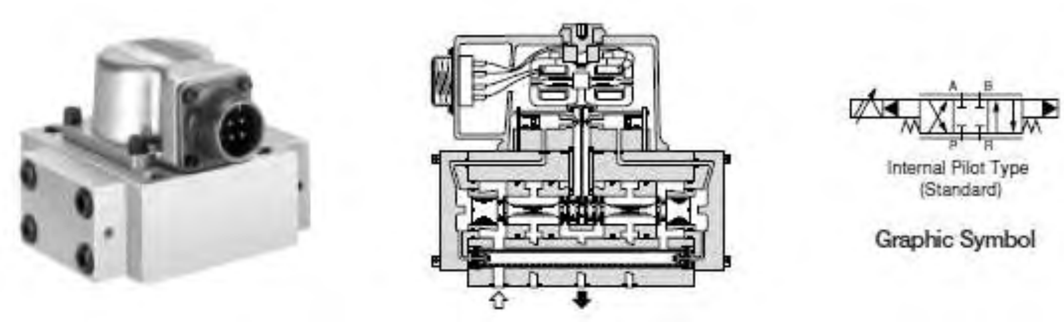


Figure 2-43 - Electro-Hydraulic Servo Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

ii. Direct Drive Servo Valves

The valves directly drive a spool with a small and high power linear motor. Direct drive servo valves electrically send the spool position data to the controller to provide quick response and high contamination resistance. These valves are available in two types:

direct spool control and pilot operation (a combination of small valves). Figure 2-44 shows illustrations of direct drive servo valves.

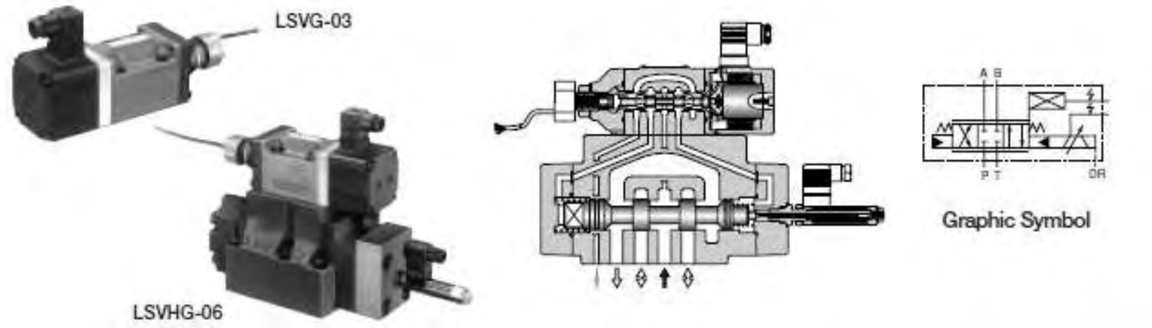


Figure 2-44 - Direct Drive Servo Valve
(Yuken Kogyo Co. - Basic Hydraulics and Components)

2.3.7.2 **Servo Supplementary Components**

a. Servo amplifiers drive servo valves, based on the same principles as the amplifiers for proportional electro-hydraulic control.
b. Position sensors, including potentiometers, synchronization generators, magnetic scales, and optical equipment (pulse encoder, digital position sensor, etc.), can accurately detect the component positions.
c. A hydraulic power source is required to provide constant hydraulic power for servo valve operation, and is incorporated in hydraulic circuits.

2.3.8 **Dynamic Models for Valves**

For dynamic modelling purposes, valves are fundamentally variable orifices where the area of the orifice depends on the valve position. For example, the core dynamic model of a solenoid proportional valve has the area of an orifice as a nonlinear function of the command signal to the solenoid. The basic equation for a valve is the orifice equation is (Durfee, Sun, & Ven, 2015):

$$Q = C_d\, A \sqrt{\frac{2\Delta P}{\rho}} \tag{1.18}$$

where C_d is valve coefficient, A is the area of the valve opening and P is the pressure drop across the valve. For valves with internal spools and rectangular orifice slots, the orifice opening area is proportional to the valve position, $A = \omega\, x$. To simplify analysis, the orifice equation can be linearized about a nominal operating point at the $x = 0$ valve position with leakage flow Q_0:

$$Q = Q_0 + K_q\, x + K_c\, P \tag{1.19}$$

The linearized valve is characterized by two parameters, the flow gain coefficient:

$$K_q = \frac{\partial Q}{\partial x} = C_d \sqrt{\frac{2P}{\rho}}\, \frac{\partial A}{\partial x} = C_d\, \omega \sqrt{\frac{2P}{\rho}} \tag{1.20}$$

and the flow pressure coefficient:

$$K_c = \frac{\partial Q}{\partial P} = \frac{A\, C_d}{\sqrt{2P\, \rho}} = \frac{\omega\, x\, C_d}{\sqrt{2P\, \rho}} \tag{1.21}$$

2.4 Accumulators

Hydraulic accumulators are used for temporarily storing hydraulic energy in the form of pressurized oil. The oil enters a chamber and acts against a piston or bladder to raise a weight, compress a spring or compress a gas (Figure 2-45). Accumulators are used to supply transient peak power, which reduces the flow rate requirement for the power supply and to act as shock absorbers for smoothing out pressure wave spikes. Accumulators are the equivalent to a capacitor in an electrical system and to a spring in a mechanical system. Accumulators can be used for pressure compensation, pulse damping, leakage compensation, emergency power, auxiliary pressure, and several other applications. They can also be used to apply pressure across a physical boundary between two liquids without contact or mixing of the liquids. This feature permits the pressurization of hazardous fluids, e.g., a volatile liquid, by means of a second liquid which can be safely pumped. Bladder type accumulators, pre-charged with nitrogen gas are the most common type for hydraulic systems. The capacity of a fluid capacitor is defined by its change in volume divided by its change in pressure (Durfee, Sun, & Ven, 2015):

$$C_f = \frac{\Delta V}{\Delta P} \tag{1.22}$$

Change in volume per time is flow rate and change in pressure per time is the derivative of pressure. This leads to the constitutive law for a linear fluid capacitor:

$$Q = C_f \, \dot{P} \tag{1.23}$$

For a gas-filled accumulator, the capacitance C_f will depend on the accumulator pre-charge. Another type of accumulator is a cylinder with the fluid pushing on one side of the piston against a stiff spring on the other side of piston. For these spring-loaded piston accumulators the capacitance is:

$$C_f = \frac{A^2}{K} \tag{1.24}$$

where A is the area of the piston and K is the spring constant.

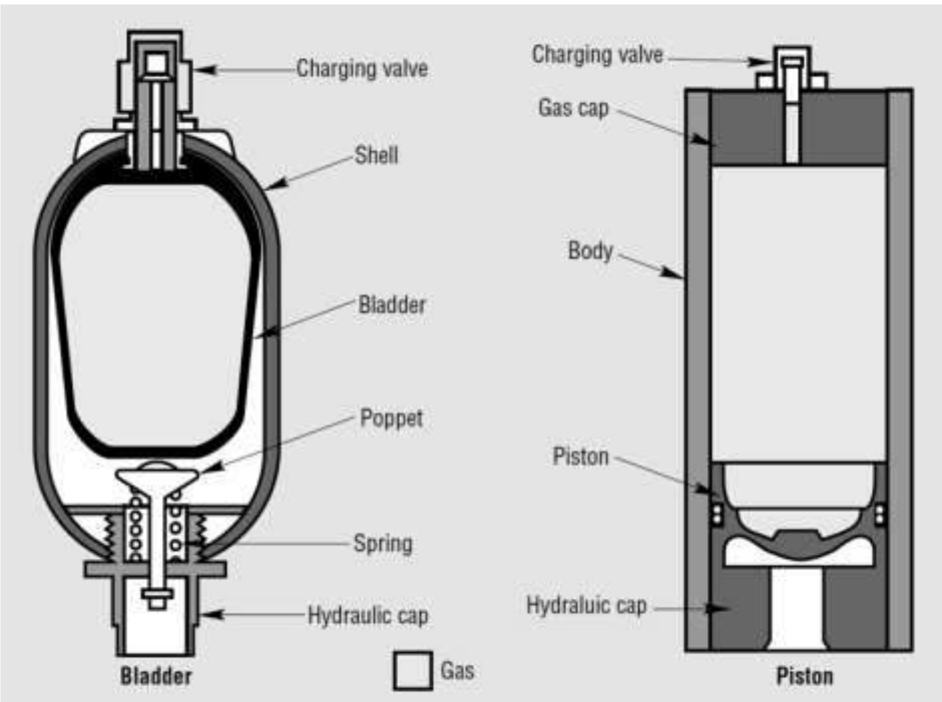


Figure 2-45 - Bladder and Piston Accumulators (www.hydraulicspneumatics.com)

2.4.1 **Accumulator Types**

Accumulators are classified in terms of the manner in which the load is applied. This is the major factor which influences design. Accumulators can be weight-loaded, spring-loaded, or pneumatic-loaded (Merritt, 1967).

i. Weight-loaded Accumulators

The weight-loaded accumulator consists of a piston mounted vertically in a cylinder. The piston rod or plunger is loaded with weights which provide potential energy to compress the fluid. This accumulator produces virtually constant pressure at all fluid levels. However, weight-loaded accumulators are very heavy and expensive. They also do not respond quickly to changes in the system demand. For these reasons, they are not often used in modern hydraulic systems.

ii. Spring-loaded Accumulators

The pressure varies with the amount of fluid in the accumulator since the spring force depends on displacement. Although such spring-loaded devices are easy to maintain, they are relatively bulky and costly. Most applications are for low-volume, low-pressure systems.

iii. Pneumatic-loaded Accumulators

There are two types of pneumatic-loaded accumulators. In one type, the gas which provides the load is in direct contact with the hydraulic fluid, whereas in the second type they are separated by a diaphragm, bladder, or piston.

- Non-separated type: Pressurization in a non-separated, pneumatic-loaded accumulator is achieved by introducing a pressurizing gas into a container above the liquid level. The pressurized storage vessel is a simple example of this type. Limit switches, which are actuated by liquid level, are usually used to limit pressure. This type can accommodate large liquid volumes, but aeration of the liquid often precludes their use in hydraulic systems.
- Separated type: Aeration in the pneumatic-loaded accumulator can be eliminated by providing a barrier between the pressurizing gas and the hydraulic fluid. Diaphragms, bladders, or pistons are used as barriers. The spherical vessel is separated into two compartments by a flexible diaphragm. One compartment is connected to the hydraulic system and the other to the high-pressure gas system. The bladder-type accumulator usually has a bladder inside a cylindrical shell with pressurized gas inside the bladder and the hydraulic fluid between the bladder and the housing.

2.5 Filters

During use, hydraulic oil picks up contaminating particles from wear of sliding metallic surface that add to residual contaminants from the oil manufacturing process, rust from metal and polymer particles from seal wear. These dirt particles are tiny grit that cause additional abrasive wear. Clumps of particles can clog tiny clearances in precision valves and cylinders and can lead to corrosion. All practical hydraulic systems require a filter in the circuit (Figure 2-46).

Filters are rated in terms of the degree of filtration. The ratings are usually expressed in microns ($1\mu m = 1 \times 10^{-6} m$). If a filter can remove 98% of the particles of a certain size or larger, then this particle size, expressed in microns, is termed the nominal filtration value. The absolute filtration value is the size of the smallest particle which the filter can completely remove from the flow. Filters are usually rated in terms of both nominal and absolute values. It is common practice to specify filters with an absolute filtration value equal to one-half of the smallest clearance or tolerance in the components which the filter must protect.

In-line filters have a fine mesh media formed from wire, paper or glass fibre, formed to create a large surface area for the fluid to pass through. The oil filter in a car is an example of a hydraulic filter. Sometimes the filter is included inside the reservoir or is part of an integrated power supply unit along with the motor, pump and reservoir. Selecting a filter is a trade-off between a media that traps fine contaminants and one that passes fluid with minimal resistance. The dynamic model for a filter is a nonlinear resistance $P = f(Q)$ that can be linearized about the nominal flow. If the pressure drop across the filter is small compared to other pressure drops in the system, the effects of the filter on the dynamic model can be ignored (Durfee, Sun, & Ven, 2015), (Merritt, 1967).

Figure 2-46 - Filters (www.hydraulicspneumatics.com)

2.5.1 **Classification of Filters**

Filters are classified according to the filter media, the configuration, or the filtering method. The filter media can be either the surface-type or the depth-type. The surface-type filtering media contain numerous orifices of relatively uniform size. Particles larger than the orifice size are trapped on the surface of the media. Depth-type media have long twisted paths through which the liquid must flow. Particles larger than the cross section of these flow paths are retained except perhaps for some particles which are larger in only one dimension. Wire mesh is an example of a surface filter medium. Depth media include sintered metal powders and fibrous materials such as paper, felt, glass, and cellulose. The T-type filter is the most widely used unit because it is compact and easy to clean or replace (Department of the Army, 1997).

2.5.2 **Filtering Methods/Types of filters**

There are three basic physical mechanisms by which filters can remove contaminants from a hydraulic fluid-mechanical, adsorbent, and absorbent. The filtering methods sometimes function in combination (U.S. Army Material Command, 1971).

2.5.2.1 **Mechanical Filters**

In a mechanical filter, particles are removed from the hydraulic fluid because of their inability to pass through the multitude of small holes or orifices in the filter. Metal or fabric screens are commonly used as the filter media. The size of particles which can be removed by this filter depends on the spacing between the disks. The filter can be cleaned while in service by revolving the central shaft. The stationary elements then act as wipers. Wire-screen mechanical filters can also be cleaned if care is taken not to force contaminant particles inside or through the elements.

2.5.2.2 **Adsorbent Filters**

Adsorption is the phenomenon by which particles of one material tend to adhere to solid or liquid surfaces. The filter medium in an adsorbent-type filter is finely divided to present maximum surface area to the flow. Materials used in the filter elements include activated clay, charcoal, fuller's earth, chemically treated paper, and bone black. The flow passages of the filter can also mechanically remove contaminants. One disadvantage of the adsorbent filter is the tendency to remove certain additives in the hydraulic fluid. Hence, it is not usually recommended for service with fluids which contain additives. Many adsorbent filter housings are designed to accommodate either an adsorbent filter element or a mechanical filter element.

2.5.2.3 **Absorbent Filters**

A porous, permeable medium is used as an element in an absorbent filter. Element materials include diatomaceous earth, wood, pulp, asbestos, paper, various textiles, and a variety of other substances. As the hydraulic fluid passes through the filter medium, contaminants are trapped by absorption. Water and water-soluble contaminants can be removed by some absorbent filters. The size of solid contaminant which can be filtered depends upon the permeability and porosity of the filter element.

2.6 Reservoirs

The main function of the reservoir is to provide a source of room temperature oil at atmospheric pressure (Figure 2-47). The reservoir is equivalent to the ground in an electrical system. Conceptually, a reservoir is nothing more than an oil storage tank connected to atmosphere through a breather and having pump and return lines to deliver and accept oil. In practice, a reservoir has additional functions including de-aerating, remove contaminants and acting as a heat exchanger. The dynamic model of a reservoir is to treat it as a ground, a source of zero pressure (Durfee, Sun, & Ven, 2015).

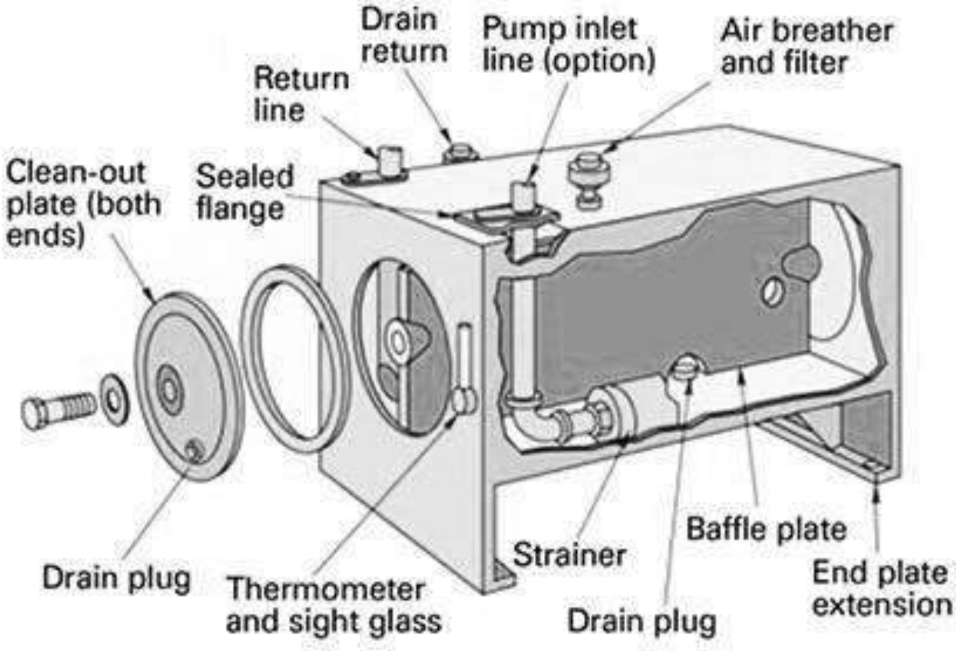


Figure 2-47 - Reservoir (www.fluidpowerjournal.com)

2.6.1 Capacity

Even before the conditioning functions of the reservoir are considered in design, the necessary capacity must be determined. The size and configuration depend on many factors. The minimum required capacity can vary from one to three times the volumetric rating of the pump in m^3 per minute. The reservoir should be sufficiently large to accommodate the liquid necessary to fill all system components if the liquid drains back to the reservoir. It should have sufficient capacity to maintain a liquid supply at the pump suction at all times. Sufficient liquid should be in the system to prevent the formation of vortices at the pump suction. Reservoir volume should be provided to allow time for solid contaminants and gases to separate from the liquid. This factor also depends on both the characteristics of the liquid and filtering system design. Adequate space above the liquid level should be provided to accommodate thermal expansion of the liquid. If the reservoir serves as the primary means of dissipating heat from the liquid, it should be large enough to accommodate the required cooling (U.S. Army Material Command, 1971).

2.6.2 Design

There are three basic reservoir arrangements: separate, integral, and dual-purpose. Separate reservoirs are commonly used in large stationary systems where space and weight are not important considerations. Integral reservoirs are spaces provided within the hydraulic system such as: piping, tubular structural members, or machine bases. Such a design minimizes space and weight requirements. However, the storage of a hot liquid within the system can sometimes cause thermal distortion of precision components.

If the liquid can serve as both a lubricant and a hydraulic fluid, individual reservoirs are not required. The reservoir in which such a liquid is stored is termed a dual-purpose reservoir. Baffles should be provided between the suction line and the fluid-return lines to prevent continual use of the same liquid. These also reduce the liquid velocity and thereby facilitate the settling of solid contaminants and de-aeration of the liquid. Lines which return liquid to the reservoir should be well below the liquid level to minimize aeration. Suction lines should also terminate below the minimum liquid level, but the inlet should be at least 1/2 pipe diameters above the tank floor. If the suction line strainer is not sufficiently beneath the liquid level at all times, a vortex could form and permit air to enter the suction line. Some reservoirs are pressurized to assist pump suction (U.S. Army Material Command, 1971).

2.7 **Hoses and Fittings**

The design of the liquid conductors is just as important as the design of other components of a hydraulic system. The hydraulic piping serves to contain and conduct the hydraulic fluid from one part of the system to another. Inadequate attention to piping design can lead to poor system operating characteristics and low efficiency. Hydraulic piping systems are usually constructed from three types of fluid conductors: pipe, tubing, and flexible hose. Hose is used to accommodate relative movement between components. Most of the stationary piping, however, is either tubing or pipe. The advantages of tubing include better appearance, greater flexibility, better reusability, fewer fittings, and less leakage. The principal advantage of pipe is its relatively low cost (Merritt, 1967).

Figure 2-48 - Hoses and Fittings (www.kehose.com)

2.7.1 **Hydraulic Line Size**

The sizing of liquid conductors for hydraulic piping depends on considerations of mechanical strength, volume flow rates, pressure drop, pressure surges, and compression time. When the components of the liquid distribution system have been converted to equivalent lengths of a common size, the pressure drop can be estimated by the following expression (U.S. Army Material Command, 1971):

$$\Delta P = 4f\frac{\rho v^2}{2}(\frac{L}{D}) \tag{1.25}$$

where, ΔP is the pressure drop, L the line length, D the inside line diameter, v the liquid's velocity and f the coefficient of friction of liquid in the fitting.

2.7.2 **Hose, Tubing and Pipe Settings**

Pipe and tubing fittings can be either threaded or permanent. Permanent methods include various forms of brazing, welding, swaging, and adhesive bonding. Such assembly methods are applied where low initial cost, reliability, and weight are important factors. Threaded pipe-fitting techniques include tapered pipe threads, flanges, SAE O-ring ports, and straight-thread ports with metal seals. Non-threaded tube fittings can be of three types: flare, self-flare, or flareless. Hose fittings are either permanent or reusable. Self-sealing couplings are sometimes used with hose in hydraulic piping systems (Merritt, 1967).

2.8 Heat Exchangers

Heat is generated in all hydraulic systems. The inherent mechanical and thermodynamic inefficiencies of pumps and motors result in heat generation. Much of this heat is transferred to the hydraulic fluid, causing a rise in fluid temperature. Since all hydraulic fluids exhibit a limited temperature range over which the viscosity and lubricating characteristics are optimum, the heat must be dissipated to assure satisfactory operation. Some heat is removed by dissipation to the environment. If this heat transfer is not sufficient to maintain the desired fluid temperature, it then becomes necessary to provide heat exchangers to supplement the natural dissipation. The analysis of temperature-control problems in a hydraulic system begins with an estimate of the total heat rejection from the system. This can be done in several ways. The heat rejection can be treated as the input power minus the actual mechanical work, based on a convenient time interval. It can also be estimated on the basis of pump output. The heat can be dissipated from the surfaces of the circuit components and, if necessary, removed by a heat exchanger (Figure 2-49), (Merritt, 1967).

Figure 2-49 - Heat Exchanger (www.indiamart.com)

2.8.1 Modes of Heat Transfer

Heat can be removed from the system by all three of the basic modes of heat transfer: conduction, convection, and radiation (Bontozoglou, 2003) (Merritt, 1967).

- ***Conduction***

Thermal conduction is the transfer of heat through a gas, liquid, or solid by means of collisions or intimate contact between the molecules. The amount of heat transferred by conduction is given by Fourier's Law. For simple one-dimensional flow this law reduces to:

$$Q = k\,A\,(\frac{dT}{dx}) \tag{1.26}$$

where, Q is the rate of heat flow, k is the thermal conductivity of material, A the area normal to direction of flow, dT the temperature difference between warmer and cooler surfaces of the material and dx the thickness of material. More general forms of the conduction equation must be used if the heat flow is other than one-dimensional.

- ***Convection***

Heat transfer by convection requires gross motion of liquid particles involving the transport of regions of the liquid at different temperatures. Free, or natural, convection occurs when the liquid particles move because of density gradients established by temperature gradients. If the liquid is circulated by external means, the process is called forced convection. Convection heat transfer rate Q is governed by a relation developed by Newton:

$$Q = h\,A\,\Delta T \tag{1.27}$$

where, Q is the rate of heat flow, h the convective film coefficient, A the area of surface exposed to the fluid and normal to heat flow direction and ΔT the temperature difference between the fluid and the surface.
This relation actually defines the film coefficient h. The film coefficient can be calculated only in the most ideal situations. Empirical relations are often employed to estimate a value of h.

- ***Radiation***

Thermal radiation involves the transport of thermal energy by means of electromagnetic radiation. The amount of heat transferred by radiation depends on the relative configuration of the areas which exchange heat, their temperatures, and the nature of their surfaces. The governing relation is:

$$Q = f\,A\,\sigma\,(T_2^4 - T_1^4) \tag{1.28}$$

where, Q is the rate of heat flow, f a dimensionless factor which accounts for the geometric orientation of the surfaces and their emittance, A the area of radiating surface, σ the Stefan-Boltzmann constant ($5.67 \times 10^{-8}\ W/(m^2K^4)$), T_2 the temperature of radiating surface and T_1 the temperature of sink or receiving surface.

2.8.2 **Overall Heat Transfer Coefficient**

The contributions of conduction, convection, and radiation to net heat transfer can be combined by making use of the concept of the overall heat transfer coefficient which is defined by the relation (Merritt, 1967):

$$Q = U\,A\,\Delta T_{total} \tag{1.29}$$

where, Q is the rate of heat flow, U the overall heat transfer coefficient, A the heat transfer area, ΔT_{total} the total temperature difference across which the heat is being transferred. The overall coefficient is a measure of the thermal conductance of the system:

$$\frac{1}{U} = \sum \frac{1}{h} + \sum R \tag{1.30}$$

Where, R is the thermal resistance of system component i and has units of m^2K/W. Additional factors such as dirt and scale deposits can contribute to the thermal resistance in heat exchangers. In such cases, appropriate terms are added to the system resistances in Equation 1.30.

2.8.3 **Types of Cooling Systems**

In general, heat transfer in a hydraulic system will involve some combination of the three modes of heat transfer. If it is found that the heat dissipated from the surfaces of the system is insufficient to maintain a satisfactory fluid temperature, then a heat exchanger is required. Most heat exchangers for hydraulic fluids are either air-cooled or water-cooled (U.S. Army Material Command, 1971).

- ***Air-cooled Heat Exchangers***

Air is often used as the coolant in heat exchangers for mobile hydraulic systems, or in stationary systems which generate moderate amounts of heat. A blower or fan is usually used to circulate the air across finned tubes through which the hydraulic fluid flows. Air-cooled units are limited to applications where the desired hydraulic fluid outlet temperature is at least 10°C above the dry bulb air temperature.

- ***Water-cooled Heat Exchangers***

The use of water as a coolant is common practice in stationary systems. The film coefficient on the water side is generally the same order of magnitude as on the hydraulic fluid side. Therefore, water-cooled heat exchangers are usually shell-and-tube type where the heat transfer area on the cold side is approximately the same as on the hot side. It consists of a tube bundle within a shell. The tubes are baffled so that the coolant flow in the shell is perpendicular to the tube axis. Most shell-and-tube heat exchangers used in hydraulic systems are either single- or double-pass units. In a single-pass unit, the two liquids generally flow in opposite directions. In a double-pass exchanger, the hydraulic fluid generally enters the same end at which the water enters and leaves. Water-cooled, shell-and-tube heat exchangers are compact and low in initial cost. They can be used to obtain lower hydraulic-fluid temperatures than the air-cooled exchangers. Limitations include the possibility of corrosion and the cost of a continuous water supply. Copper alloys are used in standard heat exchangers, and special corrosion-resistant alloys are available. Removable tube bundles facilitate cleaning.

2.9 Shock Absorbers

Fluid power is often used to cushion or absorb the impact caused when a moving mass must be stopped. If the energy of the moving mass is to be dissipated, a shock absorber is used (Figure 2-50). The working fluid in a shock absorber can be a liquid, a gas, or a combination of the two. Shock absorbers are available in a variety of different designs and configurations. Most shock absorbers, called non-regenerative shock absorbers, dissipate all of the energy of the moving mass. They rely on springs or other mechanisms to return the shock absorber to an equilibrium position. A common car shock absorber is an example. However, there is an important class of shock absorbers, called hydropneumatic mechanisms, which use pneumatic power to return the shock absorber to equilibrium (Department of the Army, 1997).

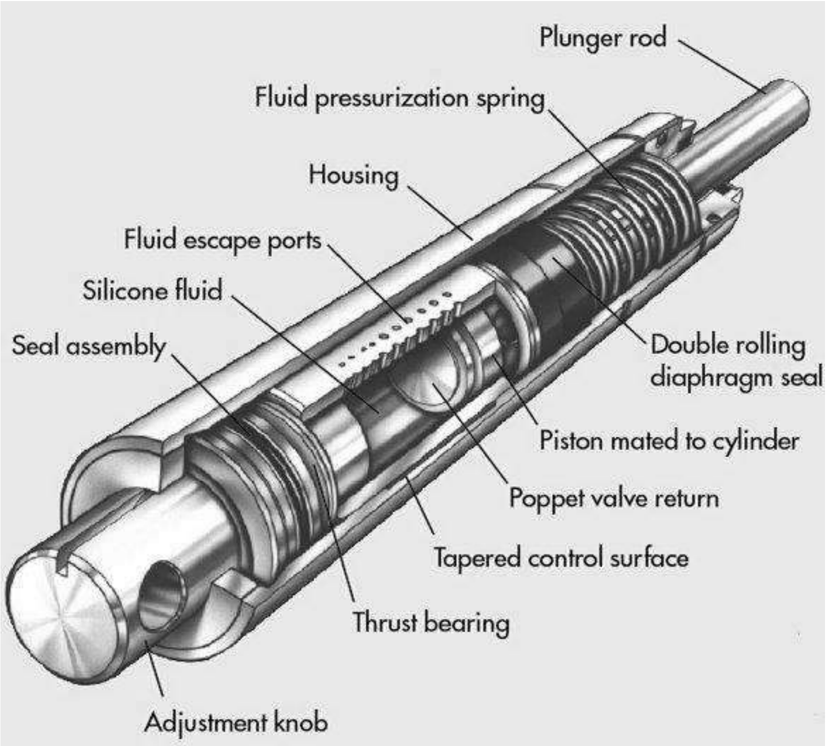


Figure 2-50 - Shock Absorber
(www.carnbikeexpert.com)

2.9.1 Hydraulic Heat Absorbers

A hydraulic shock absorber is normally used to completely stop a moving mass in a uniform manner. The shock absorber accomplishes the "smooth" deceleration by metering hydraulic fluid through orifices, converting work and kinetic energy into heat which is dissipated. The metering orifices may be fixed in size or adjustable so that deceleration rate may be varied.
When a moving mass strikes the bumper, the piston moves inward. The resisting pressure behind the piston closes the check valve. The piston then pushes the liquid through the metering orifices from the inner high-pressure chamber to the outer low-pressure chamber. The resistance to the liquid flow caused by the orifices acts against the piston to slow its motion. As the piston moves inward, it progressively blocks the orifices, increasing the resistance and uniformly decelerating the mass.

2.9.2 Hydropneumatic Shock Absorbers

A hydropneumatic shock absorber works on essentially the same principle as the hydraulic shock absorber except that both a liquid and a gas are used as the working fluids. As the mass to be stopped drives the piston inward, liquid metering through the orifices and compression of the gas brings the mass to a stop. The compressed gas then expands and returns the piston to the starting position. There are numerous variations of the hydropneumatic shock absorber. Many designs have a separate cylinder or container for the gas. Several methods of driving the piston into the gas cylinder are used.

2.9.3 Hydraulic Fluid Properties Perminet to Shock Absorbers

Properties of a hydraulic fluid that are most pertinent to its use in shock absorbers are:

- Bulk Modulus: If the shock absorber is to be fully non-regenerative, the hydraulic fluid should have a high bulk modulus so that a minimum of energy would be stored in the fluid, producing springback of the shock absorber. A liquid spring stores all of its energy in the working liquid and would therefore require a low bulk modulus.
- Density: Use of a higher density working fluid would produce a shorter stroke but higher working pressures.
- Viscosity: The working liquid viscosity is a factor in the rate at which the fluid will pass through metering orifices, lower viscosity would mean faster rates and faster strokes. Also, the viscosity should be shear stable and not exhibit excessive changes with large temperature changes.

2.9.4 Liquid Springs

A liquid spring is a regenerative shock absorber that stops a moving mass and stores the energy of the mass in the spring. A liquid spring depends on the compressibility, or bulk modulus, of the liquid for its action. A liquid with a high compressibility is confined in a cylinder. When a ram is pushed into the cylinder, the liquid is compressed and exerts a strong return force on the cylinder ram. Liquid springs provide high load absorbing capacities in small packages. A single liquid spring can provide as much load absorbing capacity as roughly 30 coil springs of the same length and diameter. They can be designed to reciprocate by providing little or no flow restriction or to act as a shock absorber or damper by restricting the flow with orifices in the piston. Disadvantages include high cost and sealing that results in high pressures. Loads must be high before a liquid spring becomes practical.

2.10 Hydraulic Fluid

2.10.1 Definition of a Fluid

All the components introduced above are designed in such a way that the hydraulic fluid which is the mean through which power is transmitted, can be controlled in the most efficient way.
A fluid may be defined broadly as a substance which deforms continuously when subjected to shear stress. This fluid can be made to flow if it is acted upon by a source of energy. This can be made clear by assuming the fluid being consisted of layers parallel to each other and letting a force act upon one of the layers in a direction parallel to its plane. This force divided by the area of the layer is called shear stress. As long as this shear stress is applied the layer will continue to move relative to its neighbouring layers. If the neighbouring layers offer no resistance to the movement of fluid, this fluid is said to be frictionless fluid or ideal fluid. The resistance the layers of a fluid offer to their neighbour layers is called viscosity of the fluid. (Practically speaking, ideal fluids do not exist in nature, but in many practical problems the resistance is either small or is not important, therefore can be ignored.) A fluid is always a continuous medium and there cannot be voids in it. The properties of a fluid, e.g., density, may, however, vary from place to place in the fluid.
In addition to shear force, fluid may also be subjected to compressive forces. These compressive forces tend to change the volume of the fluid and in turn its density. If the fluid yields to the effect of the compressive forces and changes its volume, it is compressible, otherwise it is incompressible. This property of the fluid is called bulk modulus (Manring, 2005).

2.10.2 Tasks of a Hydraulic Fluid

The hydraulic fluids used in hydraulic installations have to face multiple tasks:
- Pressure transfer
- Lubrication of the moving parts of devices
- Cooling (energy conversion produces heat)
- Cushioning of oscillations caused by pressure jerks
- Corrosion protection
- Scuff removal
- Signal transmission

2.10.3 Fluid Qualities

For hydraulic oils to be able to fulfil the requirements listed above, they must exhibit certain qualities under the relevant operating conditions such as:
- Lowest possible density.
- Minimal compressibility; (High bulk modulus)
- Viscosity not too low (lubricating film)
- Good viscosity-temperature characteristics
- Good viscosity-pressure characteristics
- Good ageing stability
- Low flammability
- Good material compatibility

In addition, hydraulic oils should fulfil the following requirements:
- Air release
- Non-frothing
- Resistance to cold
- Wear and Corrosion protection
- Water separable

2.10.4 Types of Hydraulic Fluids

Although the original fluid used with the traditional cast iron component hydraulic systems was water, it was soon found to have some major flaws. For one, as soon as the temperature dropped, it would freeze. If the climate conditions were too hot, it would then evaporate. Although water is still used in certain situations and applications, it will usually be emulsified with oil.

These days, the most typical hydraulic fluids are those made from refining mineral oil. In some cases, it's necessary to make them fire resistant and in these cases, they are likely to be manufactured from a variety of different materials blended together. The advantage of using mineral oil is that is can generally handle extreme temperatures. However, these fluids can also suffer from having a low flash point, sometimes between only 150° to 250°C. When there is a fire risk present, fire resistant fluids are typically used. Although water is a suitable addition where there is a risk of fire, it has some obvious issues. By adding 10% emulsified oil to water, it's possible to gain the required lubrication. Mixing 40% water with oil and special agents will produce a fluid that is fire resistant. A flash point as great as 600°C is possible from using synthetic fire-resistant fluids. However, these types of fluids can be very expensive.

To find out more about industrial liquid lubricants and their categorisation you can check the standards of ISO3448 and BS4231 (Lee, 2018).

Hydraulic Fluid is broken down into the following main categories:

- ***Mineral Oils***

Mineral oils are created as a result of refining crude oil and then improving their quality by adding certain substances. They may be labelled as HH which means that it's a refined mineral oil that is non-inhibited. HL has additives to make it anti corrosion and anti-rust. HM type has additives for anti-wear in addition to the additives of HL type.

- ***Fire Resistant Fluids***

There are 4 main types. HFAE is an oil in water emulsion. Type HFAB is a 40% water in oil emulsion. Type HRAS is a chemical solution in water and HFC is a water polymer solution containing water glycol. When a synthetic fluid is made from phosphate ester it's known as type HFDR. HFDS is a synthetic oil that is made of chlorinated hydrocarbons.

- ***Water / Oil Emulsions***

This is when the predominant substance (around 60%) is the oil. Chemicals are used to enable the water to mix into the oil (also known as emulsify). When the fluid touches a hot surface, the water will turn to vapour and prevent a fire from occurring. This mixture also offers good lubrication properties.

- ***Water Glycol***

Known as HFC it comprises of 40% water mixed with 60% glycol. The result is a solution. This mix has the benefit of being able to work at a lower temperature than an emulsion whilst being able to produce an improved temperature viscosity trait.

- ***Phosphate Esters***

Also known as HFDR, these fluids are resistant to fire and will not ignite unless they reach above the temperature of 550°C. The main downside with them is their tendency to be chemically active which leads to them stripping paint and destroying rubber. This means that it's necessary to use certain types of hoses, seals, etc that are able to withstand the chemical action. They can also melt the external insulation on electrical cables if they leak onto them. They are also known for being quite expensive.

When using hydraulic fluids, it’s critical that they are taken care of. Contamination accounts for up to 70% of faults in hydraulic system. It’s vital to avoid water, air and any solid matter from going into the fluid. This means that strict cleanliness is required when assembling units. Ideally it would take place in a dust free room that is designed to prevent contamination. After performing any work, a cleaning procedure should follow including the flushing of particles from pipes. Filtering systems should be used that can remove particles of between 3 microns to 10 microns (.001 mm = 1 micron). Finally, due to the high expense of oil, it’s imperative to maintain it to provide a maximum life. Its condition should be checked regularly with records taken for each machine. Contamination should be avoided and filters used.

3. Fundamentals of Hydraulics

There are two methods for studying the movement of flow (Nakayama, 1998). One is a method which follows any arbitrary particle with its kaleidoscopic changes in velocity and acceleration. This is called the Lagrangian method. The other is a method by which, rather than following any particular fluid particle, changes in velocity and pressure are studied at fixed positions in space x, y, z and at time t. This method is called the Eulerian method. Nowadays the latter method is more common and effective in most cases. Here we will explain the fundamental principles needed whenever fluid movements are studied.

3.1 The Logic Behind Hydraulics

3.1.1 Hydrostatic Pressure

Hydrostatic pressure is the pressure which rises above a certain level in a liquid owing to the weight of the liquid mass an example of which is the atmospheric pressure (Smits, 2017):

$$Ps = h\,\rho\,g \tag{3.1}$$

$Ps\,:\ hydrostatic\ pressure\ (gravitational\ pressure)\ in\ (psi)$
$h\,:\ level\ of\ the\ column\ of\ liquid\ in\ (in)$
$\rho\,:\ density\ of\ the\ liquid\ in\ (lb - sec^2/in^4)$
$g\,:\ acceleration\ due\ to\ gravity\ in\ (in/sec^2)$

The hydrostatic pressure, or simply "pressure" as it is known for short, does not depend on the type of vessel used. It is purely dependent on the height and density of the column of liquid as shown in Figure 3-1.

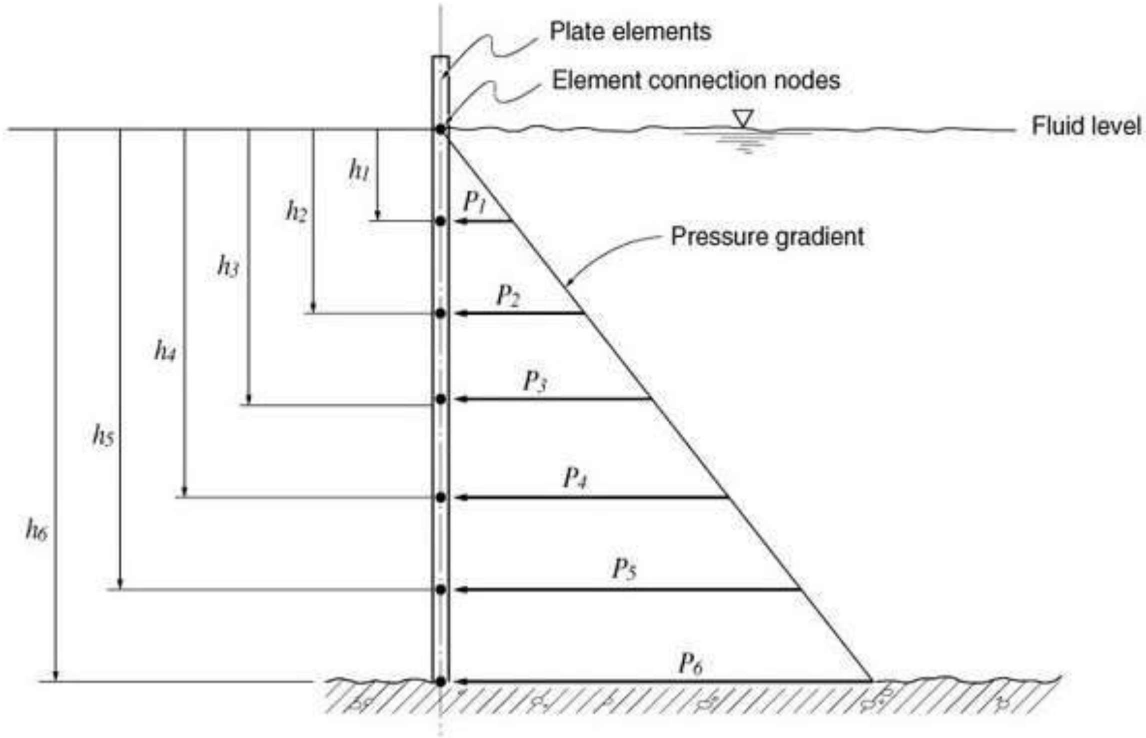


Figure 3-1 - Hydrostatic Pressure.
(manual.midasuser.com, article: Hydrostatic Pressure Loads)

3.1.2 **Pascal's Law – Pressure Transmission**

Pascal's law states that when there is an increase in pressure at any point in a confined fluid, there is an equal increase at every other point in the container (Fairman, 1996). For this reason, the shape of the container has no significance.

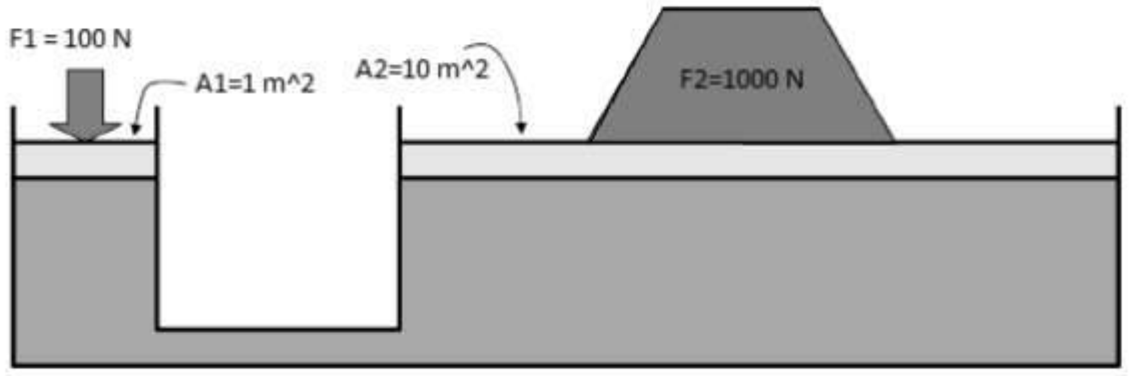


Figure 3-2 - Result of Pascal's Law. The increase in Pressure from Area 1 is equal with the increase in Pressure at Area 2. The Output Force F2 is 10 times F1.

Figure 3-2 shows two different pistons (A_1 and A_2). We apply a force of 100N at the piston with the small base (A_1) while on the larger piston (A_2) stands a mass of 100kg applying 1000N on its base due to gravity. The pressure rise from each action is calculated:

$$P_1 = \frac{F_1}{A_1} \quad and \quad P_2 = \frac{F_2}{A_2} \tag{3.2}$$

According to Pascal's Law in a closed system, $P_1 = P_2$. Thus,

$$F_2 = \frac{F_1 A_2}{A_1} \tag{3.3}$$

$Where,$
$P \ = \ Pressure\ (psi)$
$F \ = \ Force\ (lb)$
$A \ = \ Area\ (in^2)$

Small forces from the pressure piston can produce larger forces by enlarging the working piston surface. This is the fundamental principle which is applied in every hydraulic system from the jack to the lifting platform. The force F_1 must be sufficient for the fluid pressure to overcome the load resistance.

In the case where the pressure can be controlled like the double acting cylinder
The hydrostatic pressure P_1 exerts a force F_1 on the area A_1 which is transferred via the piston rod onto the small piston. Thus, the force $F1$ acts on the area A_2 and produces the hydrostatic pressure P_2. Since piston area A_2 is smaller than piston area A_1, the pressure P_2 is greater than the pressure P_1. Here too, the following law applies (Merkle, M.Thomes, & B.Schrader, 2003):

$$F_1 = F_2 \tag{3.4}$$

$$P_2 = \frac{P_1 A_1}{A_2} \tag{3.5}$$

3.1.3 Displacement Transmission

If a load F_2 is to be lifted a distance s_1 in line with the principle described above, the base A_1 must displace a specific quantity of liquid which lifts the base A_2 by a distance s_2. Assuming incompressible fluid, the volume must remain constant.

$$V_1 = V_2 \tag{3.6}$$

$$V_1 = s_1 \cdot A_1 \quad and \quad V_2 = s_2 \cdot A_2 \tag{3.7}$$

From this, it can be seen that the distance s_1 must be greater than the distance s_2 since the area A_1 is smaller than the area A_2.

$$s_1 = s_2 \cdot \frac{A_2}{A_1} \tag{3.8}$$

3.1.4 Flow Rate

Flow rate is the term used to describe the volume of liquid flowing through a pipe in a specific period of time. In hydraulics, the flow rate is designated as Q. The following equation applies:

$$Q = \frac{V}{t} = \frac{A\,s}{t} = A\,u \tag{3.9}$$

$Where,$
$u \ = Velocity\ of\ flowing\ particle\ (in/sec)$
$Q \ = \ Flow\ rate\ (in^2/sec)$
$V \ = \ Volume\ (in^3)$
$T \ = \ time\ (sec)$

The flow rate of a liquid in terms of volume per unit of time which flows through a pipe with several changes in cross-section is the same at all points in the pipe. This means that the liquid flows through small cross-sections faster than through large cross-sections.

$$Q_1 = Q_2 \tag{3.10}$$

$$u_2 = \frac{u_1 A_1}{A_2} \tag{3.11}$$

3.2 The Seven General Equations

Knowledge of the fundamental laws and equations which govern the flow of fluids is essential for the rational design of hydraulic control components and systems. This chapter will discuss the general equations of fluid motion, types of flow, and flow through conduits and orifices.

Fluids are made up of discrete particles - molecules. An accurate analysis would have to consider the motion of each particle, and this would be hopeless analytically. For example, the density at any geometrical point would depend on whether there exists a molecule at that point. Therefore, we must rely on "continuous" theory and consider the statistical properties of a fluid. This concept conflicts with molecular theory, but it is sufficiently accurate for engineering purposes.

Analytic description of general fluid flow requires that the motion of a small cube of fluid be defined. If such a cube can be sufficiently defined, it would be possible to proceed to more complex situations. An infinitesally small volume of fluid can be completely defined using eight parameters.

x coordinate	Temperature	Time
y coordinate	Pressure	
z coordinate	Density	
	Viscosity	

Therefore, seven independent equations are required in order that they may be solved simultaneously to obtain any of the parameters as a function of another or, as is more usually the case, to find any parameter as a function of time (Merritt, 1967).

3.2.1 Navier - Stokes Equations

The first three of these equations result when Newton's second law is applied to the three directions of motion.

$$\rho\left(\frac{\partial u}{\partial t}+u_x\frac{\partial u}{\partial x}+u_y\frac{\partial u}{\partial y}+u_z\frac{\partial u}{\partial z}\right)=\rho g_x-\frac{dP}{dx}+\mu\left(\frac{\partial u^2}{\partial^2 x}+\frac{\partial u^2}{\partial^2 y}+\frac{\partial u^2}{\partial^2 z}\right) \quad (3.12)$$

$$\rho\left(\frac{\partial u}{\partial t}+u_x\frac{\partial u}{\partial x}+u_y\frac{\partial u}{\partial y}+u_z\frac{\partial u}{\partial z}\right)=\rho g_y-\frac{dP}{dy}+\mu\left(\frac{\partial u^2}{\partial^2 x}+\frac{\partial u^2}{\partial^2 y}+\frac{\partial u^2}{\partial^2 z}\right) \quad (3.13)$$

$$\rho\left(\frac{\partial u}{\partial t}+u_x\frac{\partial u}{\partial x}+u_y\frac{\partial u}{\partial y}+u_z\frac{\partial u}{\partial z}\right)=\rho g_z-\frac{dP}{dz}+\mu\left(\frac{\partial u^2}{\partial^2 x}+\frac{\partial u^2}{\partial^2 y}+\frac{\partial u^2}{\partial^2 z}\right) \quad (3.14)$$

$Where,$
u_x, u_y, u_z $are\ velocity\ components\ in\ x, y, and\ z\ directions\ of\ Cartesian\ coordinates$
g_x, g_y, g_z $are\ body\ forces\ per\ unit\ volume\ in\ direction\ of\ coordinate\ axes$
$t = time$
$P = pressure\ per\ unit\ of\ area$
$\rho = mass\ density$
$\mu = absolute\ viscosity\ of\ the\ fluid$

These equations are a result of the law of conservation of momentum. The terms on the left side of these equations are a result of fluid inertia. The last three terms on the right-side result from viscous friction. If the inertia terms are neglected, the set of equations is called Stokes equations; if viscosity is neglected, the equations are called Euler's equations. The ratio of inertia force over viscous force is called Reynolds number and serves to weight the relative effects of viscosity and inertia terms of the Navier-Stokes equations. A large Reynolds number indicates that inertia terms are dominant, whereas a small number indicates the dominance of viscosity terms (Bontozoglou, 2003) (Merritt, 1967).

3.2.2 Continuity Equation - Conservation of Mass

Consider a control volume (Figure 3-3) in which there are weight flow rates W into and from the volume. Let the volume be Vo, and the accumulated or stored mass of fluid inside be m with a mass density of ρ. Since all fluid must be accounted for, as the medium is assumed continuous, the rate at which mass is stored must equal incoming mass flow rate minus outgoing mass flow rate (Merritt, 1967). Therefore,

$$\sum W_{in} - \sum W_{out} = g \cdot \frac{dm}{dt} = g \cdot \frac{d(\rho V_0)}{dt} \tag{3.15}$$

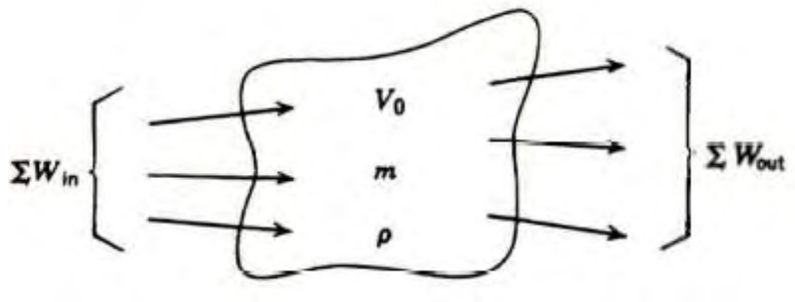


Figure 3-3 - Control volume where the continuity equation is applied. (Herbert Merritt - Hydraulic control systems)

3.2.3 Conservation of Energy – 1st Thermodynamic Law

3.2.3.1 Energy and Hydraulics

The first law of thermodynamics states that for any given system, the change in energy (ΔE) is equal to the difference between the heat transferred to the system (Q) and the work done by the system on its surroundings (W) during a given time interval.
The energy referred to in this principle represents the total energy of the system, which is the sum of the potential energy, kinetic energy, and internal (molecular) forms of energy such as electrical and chemical energy. Although internal energy may be significant for thermodynamic analyses, it is commonly neglected in hydraulic analyses because of its relatively small magnitude.

In hydraulic applications, energy values are often converted into units of energy per unit weight, resulting in units of length. Using these length equivalents gives engineers a better "feel" for the resulting behaviour of the system. When using these length equivalents, the engineer is expressing the energy of the system in terms of "head." The energy at any point within a hydraulic system is often expressed in three parts, as shown in Figure 3-4:

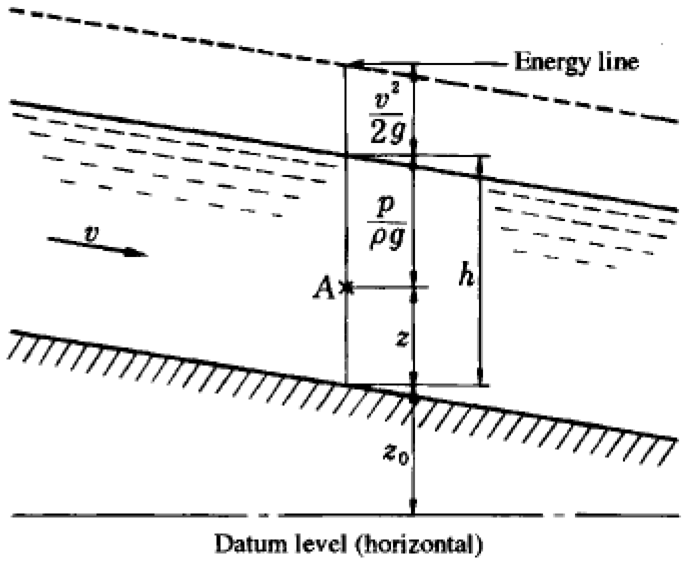


Figure 3-4 - Pressure, Elevation and Velocity Heads in an Open Channel. (Herbert Merritt - Hydraulic control systems)

Energy types

- Static
  - Potential (Elevation Head) (z)
  - Pressure Head ($P/\rho g$)
- Dynamic
  - Kinetic (Velocity Head) ($u^2/2g$)
  - Thermal (Friction)
  - Work

Hydraulic Grade

The hydraulic grade is the sum of the pressure head (p/γ) and elevation head (z). For open channel flow (in which the pressure head is zero), the hydraulic grade elevation is the same as the water surface elevation. For a pressure pipe, the hydraulic grade represents the height to which a water column would rise in a piezometer (a tube open to the atmosphere rising from the pipe). When the hydraulic grade is plotted as a profile along the length of the conveyance section, it is referred to as the hydraulic grade line, or HGL. Thus, if the depth of the channel is h, then

$$Hydraulic\ Grade = Pressure\ Head + Elevation\ Head = \frac{P}{\rho g} + z \tag{3.16}$$

Energy Grade

The energy grade is the sum of the hydraulic grade and the velocity head. This grade is the height to which a column of water would rise in a Pitot tube (an apparatus similar to a piezometer, but also accounting for fluid velocity). When plotted in profile, this parameter is often referred to as the energy grade line, or EGL. For a lake or reservoir in which the velocity is essentially zero, the EGL is equal to the HGL. Thus, If the pressure is P at a point A in the open channel in Figure 3-4, the Total Head or Energy Grade of the fluid at this point is:

$$Total\ Head = Velocity\ Head + Pressure\ Head + Elevation\ Head =>$$

$$Total\ Head = Velocity\ Head + Hydraulic\ Grade + z_0 =>$$

$$Total\ Head = \frac{u^2}{2g} + \frac{P}{\rho g} + z + z_0 \tag{3.17}$$

And by including the internal energy, the Total Energy is expressed:

$$Total\ Energy = U + \frac{u^2}{2g} + \frac{P}{\rho g} + z + z_0$$

or (3.18)

$$Total\ Energy = h + \frac{u^2}{2g} + z + z_0$$

$Where,$
$P\ =\ pressure\ (\ lbs/in^2)$
$\gamma\ =\ \rho g\ specific\ weight\ (lbs/in^3)$
$z\ =\ elevation\ (in)$
$u\ =\ velocity\ (in/sec)$
$U = internal\ energy\ per\ unit\ weight$
$h = U + pressure\ head, the\ enthalpy\ of\ the\ fluid\ (in - lb/lb)$

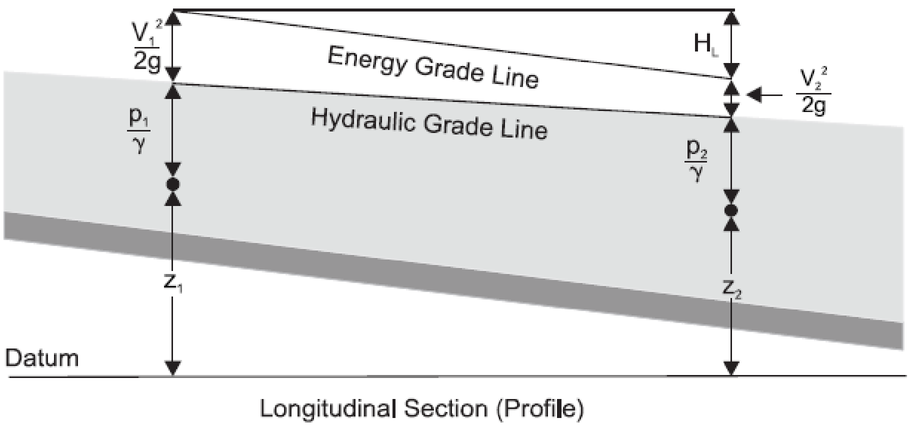


Figure 3-5 - The change of Energy Heads Downstream an Open Channel.
(Yuken Kogyo Co. Basic Hydraulics and Components)

Note that a point on the water surface of an open channel will have a pressure head of zero, but will have a positive elevation head higher than that of a point selected at the bottom of the channel for the same station.

In addition to pressure head, elevation head, and velocity head, energy may be added to a system by a pump (for example) and removed from the system by friction or other disturbances. These changes in energy are referred to as Head gains (H_G) and Head Losses (H_L), respectively. Because energy is conserved, the energy across any two points in the system must balance. This concept is demonstrated by the energy equation:

$$Total\ Head|_1 + Head\ Gain = Total\ Head|_2 + Head\ Loss$$

$$\frac{p_1}{(\rho g)_1} + z_1 + \frac{u_1^2}{2g} + H_G = \frac{p_2}{(\rho g)_2} + z_2 + \frac{u_2^2}{2g} + H_L \tag{3.19}$$

Where,
$p = pressure\ (lb/in^2)$
$\gamma = specific\ weight\ of\ the\ fluid\ (lb/in^3)$
$z = z_i + z_0\ elevation\ above\ a\ datum\ (in)$
$u = fluid\ velocity\ (in/sec)$
$g = gravitational\ acceleration\ (in/s^2)$
$H_G = head\ gain, such\ as\ from\ a\ pump\ (in)$
$H_L = combined\ head\ loss\ (in)$

(Merritt, 1967) (Manring, 2005) (Basic Hydraulic Principles, 2002)

Summary Table for Head Loss and Gain Terms

Energy equation for steady incompressible flow through control volume (inlet at 1, outlet at 2)

$$\left(\frac{p_1}{\rho g}+\alpha_1\frac{\bar{V}_1^2}{2g}+z_1\right)-\left(\frac{p_2}{\rho g}+\alpha_2\frac{\bar{V}_2^2}{2g}+z_2\right)=H_\ell+H_{\ell_m}-H_s$$

Head Term	Equation
Total head loss	$H_{\ell T}=\frac{(\breve{u}_2-\breve{u}_1)-\dot{Q}_{net}}{\dot{m}\,g}=H_\ell+H_{\ell_m}$
Major head loss (due to friction)	$H_\ell=f\left(\frac{L}{D}\right)\left(\frac{\bar{V}^2}{2g}\right)$
Minor head loss (in bends and fittings)	$H_{\ell_m}=K_L\left(\frac{\bar{V}^2}{2g}\right)$ or $H_{\ell_m}=f\left(\frac{\ell_{eq}}{D}\right)\left(\frac{\bar{V}^2}{2g}\right)$
Mechanical (or shaft) head	$H_m=\frac{\dot{W}_m}{\dot{m}\,g}$ or $H_s=\frac{\dot{W}_s}{\dot{m}\,g}$
Ideal pump head gain for swirl free inlet	$H_{i,SF}=\frac{\Delta p_{p,\max}}{\rho g}=\frac{U_2^2}{g}-\left(\frac{U_2\cot\beta_2}{2\pi r_2 b_2 g}\right)Q$, $U_2=\omega r_2$
Actual pump head gain for pump curve*	$H_p=H_m-H_{\ell p}=\frac{\Delta p_p}{\rho g}=\frac{\dot{W}_h}{\dot{m}\,g}$ $H_p=H_0-C_p Q^2$ where H_0 is *shutoff head*
Actual system head loss for system curve**	$H_{sys}=\frac{p_2-p_1}{\rho g}+z_0+C_{sys}Q^2$, $z_0=z_2-z_1$, $C_{sys}=\frac{1}{2g}\left(\frac{\alpha_2}{A_2^2}-\frac{\alpha_1}{A_1^2}\right)+\frac{(H_\ell+H_{\ell,m})_{sys}}{Q^2}$

Overall pump efficiency defined as: $\eta_p=\frac{\dot{W}_h}{\dot{W}_m}=\frac{H_p}{H_m}$

Best efficiency point (BEP) at maximum η: Q_{BEP} and $H_{p,BEP}$

Operating point at intersection of pump curve* and system curve** ($H_p=H_{sys}$):

$$Q_{OP}=\sqrt{\frac{H_0-z_0-(p_2-p_1)/(\rho g)}{C_p+C_{sys}}} \quad \text{and} \quad H_{OP}=H_0-\left(\frac{C_p}{C_p+C_{sys}}\right)\left(H_0-z_0-\frac{p_2-p_1}{\rho g}\right)$$

Figure 3-6 - Summary table of Head Gain and Head Loss terms.
(Ph.D., Kim A. Shollenberger. Energy Heads Table. Cal Poly State University, San Luis Obispo.)

Considering a volume (Figure 3-7) in which weight flow rates in are W_{in} and outflows are W_{out}. The fluid inside the volume is doing external work (expansion, shaft, and shear) of dW_x/dt *(in-lb/sec),* and heat is being transferred to the volume at a rate of dQ_h/dt *(in-lb/sec).* The statement of the first law is that the energy flow in minus the energy flow out must equal the rate at which energy is stored inside the volume.

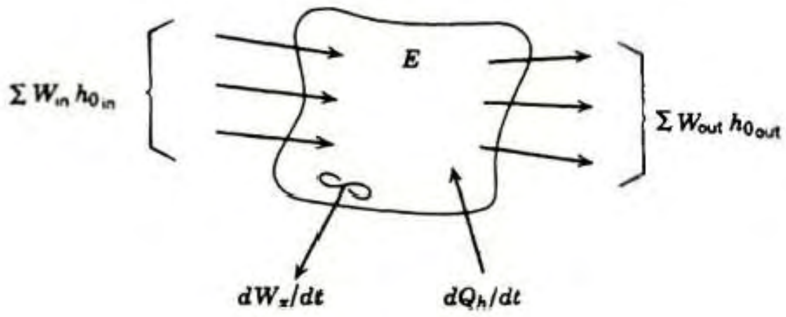


Figure 3-7 - Energies entering and leaving a Control Volume.
(Herbert Merritt - Hydraulic control systems)

Therefore,

$$\frac{dQ_h}{dt} - \frac{dW_x}{dt} + \sum W_{in}\, h_{0,in} - \sum W_{out} h_{0,out} = \frac{dE}{dt} \tag{3.20}$$

$Where,$
$ho = Total\ Energy, eq. 3.18\ (in - lb/lb)$
$E = \ total\ internal\ energy\ of\ fluid\ inside\ volume, (in - lb)$
$W = \ weight\ flow\ rate, (lb/sec)$
$dQ_h/dt = heat\ flow\ rate\ (in - lb\ /\ sec)$

This equation assumes the absence of capillary, electrical and magnetic forces, and that such a volume can be defined. For a liquid, the internal energy per pound is

$$U = 9339 C_p T \tag{3.21}$$

$Where,$
$T = liquid\ temperature, °F$
$Cp = specific\ heat, Btu/lb°F$
$9339 = the\ mechanical\ equivalent\ of\ heat, (in - lb/Btu.)$

Therefore, for steady flow of an incompressible liquid (i.e., no energy stored in the volume, $dE/dt = 0$, and $\gamma_1 = \gamma_2 = \gamma$) which enters and leaves a control volume at only one place with negligible changes in elevation (Figure 3-8), (3.20) becomes

$$\frac{dQ_h}{dt} - \frac{dW_x}{dt} + W_1(9339 C_p T_1 + \frac{P_1}{\rho g} + \frac{u_1^2}{2g}) - W_2(9339 C_p T_2 + \frac{P_2}{\rho g} + \frac{u_2^2}{2g}) = 0 \tag{3.22}$$

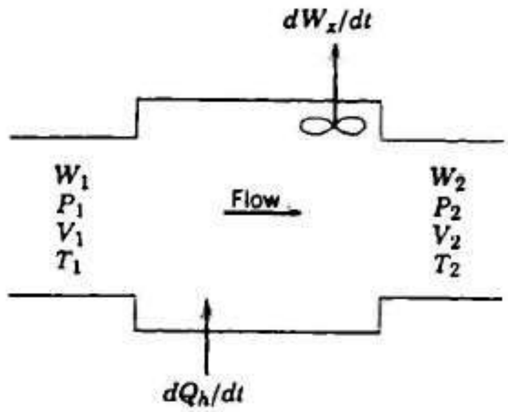


Figure 3-8 - Flow entering and leaving a Control Volume with heat added and work being done. (Herbert Merritt - Hydraulic control systems)

Considering the heat transferred to the volume two extreme cases are usually considered.

- Heat can be transferred at such a rate that the temperature remains constant. This condition is called isothermal and, since temperature is constant, the energy equation is not required.
- At the other extreme no heat is transferred, that is, $dQ/dt = 0$, and this condition is called adiabatic. In general, temperature changes have little effect on liquid flow because cubical expansion coefficients are small and cause negligible density change.

(Manring, 2005) (Merritt, 1967)

3.2.4 **Equation of State**

The density of a liquid is a function of both pressure and temperature. A function relating density, pressure, and temperature of a fluid is, by definition, the equation of state. Because changes in density as a function of pressure and temperature are small for a liquid, the first three terms of a Taylor's series for two variables may be used as an approximation. Therefore,

$$\rho = \rho_0 + \left(\frac{\partial \rho}{\partial P}\right)_T (P - P_0) + \left(\frac{\partial \rho}{\partial T}\right)_P (T - T_0) \qquad (3.23)$$

$where\ \rho, P, and\ T\ are\ the\ mass\ density, pressure, and\ temperature, respectively,$
$of\ the\ liquid\ about\ initial\ values\ of\ \rho_0, P_0, and\ T_0.$

A more convenient form for (3.23) is the linearized equation of state for a liquid.

$$\rho = \rho_0 \left[1 + \frac{1}{\beta}(P - P_0) - a(T - T_0)\right] \qquad (3.24)$$

Where

$$\beta = \rho_0 \left(\frac{\partial P}{\partial \rho}\right)_T \quad and \quad a = -\frac{1}{\rho_0}\left(\frac{\partial \rho}{\partial T}\right)_P$$

The mass density increases as pressure is increased and decreases with temperature increase (Merritt, 1967).

3.2.4.1 **Bulk Modulus**

In Equation 3.24, for V being the total volume and Vo the initial total volume of the liquid, yields:

$$\beta = -V_0 \left(\frac{\partial P}{\partial V}\right)_T \tag{3.25}$$

The quantity β is the change in pressure divided by the fractional change in volume at a constant temperature and is called the isothermal bulk modulus or simply bulk modulus of the liquid. The bulk modulus of a fluid characterizes the stiffness of the fluid acting as a spring and is always a positive quantity, for $(dP/dV)_T$ is always negative.
Interaction of the spring effect of a liquid and the mass of mechanical parts gives a resonance in nearly all hydraulic components. In most cases this resonance is the chief limitation to dynamic performance. For petroleum fluids a typical bulk modulus value is $220{,}000\ lb/in^2$. However, values this large are rarely achieved in practice because the bulk modulus decreases sharply with small amounts of air entrained in the liquid. In some cases, the elasticity of structural members, such as motor housings, can reduce the effective bulk modulus appreciably. Thus, it is important to calculate the total bulk modulus of a system combining the stiffnesses of fluid, air and mechanical components (Manring, 2005).

$$\frac{1}{\beta_{total}} = \frac{1}{\beta_{container}} + \frac{1}{\beta_{liquid}} + \frac{1}{\beta_{gas}} \tag{3.26}$$

3.2.4.2 **Cubical Expansion Coefficient.**

Respectively, in Equation 3.24, for V being the total volume and Vo the initial total volume of the liquid, yields:

$$\alpha = -\frac{1}{V_o} \left(\frac{\partial V}{\partial T}\right)_P \tag{3.27}$$

which is the fractional change in volume due to a change in temperature.
The cubical expansion coefficient for petroleum base fluids is about $a = 0.5\ x\ 10^{-3}\ (°F^{-1})$, that is, there is about a 5% increase in volume for each $100°F$ of temperature increase. (Manring, 2005)

3.2.5 **Fluid Viscosity**

The viscosity of a fluid is the measure of its resistance to gradual deformation by shear stress or tensile stress. Viscosity is the property of a fluid which opposes the relative motion between two surfaces of the fluid that are moving at different velocities. In simple terms, viscosity means friction between the molecules of fluid. For example, honey has a higher viscosity than water. When the fluid is forced through a tube, the particles which compose the fluid generally move more quickly near the tube's axis and more slowly near its walls; therefore, some stress (such as a pressure difference between the two ends of the tube) is needed to overcome the friction between particle layers to keep the fluid moving. For a given velocity pattern, the stress required is proportional to the fluid's viscosity.
A fluid that has no resistance to shear stress is known as an ideal or inviscid fluid. Zero viscosity is observed only at very low temperatures in super-fluids. Otherwise, all fluids have positive viscosity and are technically said to be viscous or viscid. A fluid with a relatively high viscosity, such as pitch, may appear to be a solid.

Viscosity is an important property of any fluid. It is necessary for hydrodynamic lubrication, and a suitable value is required for many other purposes. Close-fitting surfaces in relative motion occur in most hydraulic components. If the viscosity of the fluid is too low, leakage flows increase; if the viscosity is too large, component efficiencies decrease because of additional power loss in fluid friction. Viscosity is of such significance that it is common practice to designate the fluid by its viscosity at a certain temperature.

Isaac Newton was the first to give a quantitative definition of viscosity. Referring to the piston and cylinder of Figure 3-9, in which the radial clearance Cr is filled with a fluid.

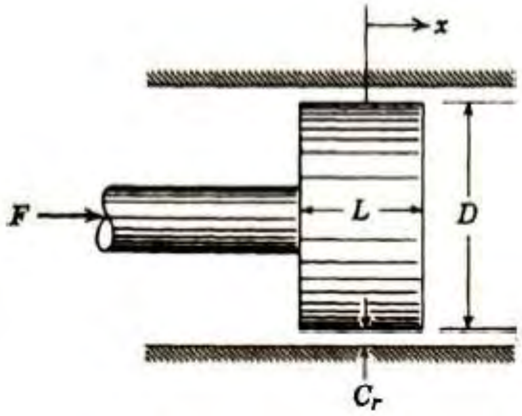


Figure 3-9 - Piston Concentric in Cylinder.
(Herbert Merritt - Hydraulic control systems)

Newton observed that a force was necessary to cause relative motion. This force is a measure of the internal friction of the fluid or its resistance to shear and is proportional to the area in contact and to the velocity and is inversely proportional to the film thickness.

$$F = \mu \frac{A}{C_r}\frac{dx}{dt} = \mu \frac{\pi DL}{C_r}\frac{dx}{dt} \tag{3.28}$$

The ratio of absolute viscosity to mass density occurs in many equations (Navier-Stokes, Reynolds number, etc.) and is easily measured by many viscometers. This ratio is, by definition, the kinematic viscosity v of the fluid, that is (Merritt, 1967),

$$v = \frac{\mu}{\rho} \tag{3.29}$$

3.2.5.1 **Viscosity as a Function of Pressure and Temperature**

The viscosity of liquids decreases markedly with temperature increase and increases, but to a much lesser degree, with increased pressure. The viscosity variation with temperature is the more important and may be approximated by an equation of the form:

$$\mu = \mu_0 e^{-\lambda(T-T_0)} \tag{3.30}$$

$where$
$\mu\ =\ absolute\ viscosity\ at\ temperature\ T, (Ib - sec/in^2)$
$\mu_0\ =\ viscosity\ at\ a\ reference\ temperature\ Tg, (Ib - sec/in^2)$
$\lambda\ =\ a\ constant\ which\ depends\ on\ the\ liquid, (1/°F)$
$T\ =\ temperature, (°F)$

The equations which describe fluid flow are nonlinear partial differential equations with complex boundary conditions. Needless to say, no general solutions of these equations have been found. There is therefore no general theoretical treatment of fluid motion. The general equations do serve to define the scope of any problem involving fluids. In many instances certain approximations can be made which reduce the complexity of these equations and permit solutions accurate enough for most purposes (Merritt, 1967).

3.3 Types of Fluid Flow in Closed Conduits

Flow in closed conduits is of particular interest and includes flow in pipes, sudden enlargements and contractions in pipe sections, flow-through fittings, and flow through restrictions in pipes such as orifices. Some general comments on fluid flow can be made. The forces which affect fluid flow are:

- Body forces such as gravity and buoyancy
- Forces due to fluid inertia
- Forces arising from internal fluid friction (viscosity)
- Forces due to surface tension
- Electric and magnetic fields.

3.3.1 Reynold's Number

In most cases, only those forces arising from fluid inertia and viscosity are significant. Experience shows that flows in nature are generally dominated either by viscosity or inertia of the fluid. Therefore, it is useful to define a quantity which describes the relative significance of these two forces in a given flow situation.
The dimensionless ratio of inertia force to viscous force is called Reynolds number and defined by

$$Re = \frac{\rho\, u_{avg}\, a}{\mu} \tag{3.31}$$

Where,
$\rho = fluid\ mass\ density$
$\mu = absolute\ viscosity$
$u = average\ velocity\ of\ flow$
$a = a\ characteristic\ dimension\ of\ the\ particular\ flow\ situation.$

The characteristic length used for Reynolds number is inside pipe diameter D, and the average flow velocity is volumetric flow rate divided by pipe area, that is,

$$u_{avg} = \frac{Q}{A} = \frac{4Q}{\pi D^2} \tag{3.32}$$

And the Reynolds number is given by

$$Re = \frac{\rho\, u_{avg}\, D}{\mu} = \frac{4\rho Q}{\pi \mu D} \tag{3.33}$$

The Reynolds number, based on hydraulic diameter, should be computed to obtain a rough idea of the type of flow. Referring to Figure 3-11, we note that flow through an annulus increases as the shaft becomes more eccentric. In the extreme case in which the shaft touches the cylinder wall, the eccentricity equals the radial clearance and the flow is 2.5 times that obtained with shaft and cylinder concentric. In practice the eccentricity is not known and an average flow between the two extremes might be used. This relation is useful in establishing the radial clearance in a seal so that leakage flow requirements are not exceeded.
Turbulent flow in closed conduits of noncircular cross section may be approximately computed from formulas given if the diameter is considered to be the hydraulic diameter. The hydraulic diameter is defined by:

$$Dh = \frac{4A}{S} \tag{3.34}$$

$where\ A\ is\ the\ flow\ section\ area\ and\ S\ is\ the\ flow\ section\ perimeter.$

For a circular section the hydraulic diameter becomes the inside pipe diameter. The concept of hydraulic diameter cannot be used for laminar flows because such flows are highly dependent on passage geometry. Transition Reynolds number may be approximately determined based on the hydraulic radius. Transition from laminar to turbulent flow has been experimentally observed to occur in the range $2000 < \mathrm{R} < 4000$. Below $\mathrm{R} = 2000$, the flow is always laminar; above $\mathrm{R} = 4000$ the flow is usually, but not always, turbulent. It is possible to have laminar flow at Reynolds number considerably above 4000 if extreme care is taken to avoid disturbances which would lead to turbulence. However, these instances are exceptional, and the high limit of 4000 is a good rule (Merritt, 1967).

3.3.1.1 **Viscosity Dominated Flow**

- ***Laminar Flow (Viscous Flow)***

It is dominated by viscosity forces and characterized by an orderly, smooth, parallel line motion of the fluid.
(Merritt, 1967)

3.3.1.2 **Inertia Dominated Flow**

- ***Turbulent Flow***

It is generally inertia dominated flow and characterized by irregular, erratic, eddy-like paths of the fluid particles (Merritt, 1967).

- ***Potential Flow***

In some cases of inertia dominated flow, viscosity is important only in a layer, called the boundary layer, next to a solid boundary while the main body of flow outside of the boundary layer is inertia dominated and behaves in an orderly fashion similar to that of laminar flow. If the boundary layer forces can be neglected, the resulting flow is called potential or streamline flow, an example of which is flow through an orifice. Potential flow is non-turbulent, streamline, and frictionless, so that the Reynolds number is infinite. However, the term turbulent is generally used to designate flows at high Reynolds numbers. Assuming one-dimensional, steady, incompressible, frictionless ($\mu = 0$) flow with no body forces, the Navier-Stokes equations reduce to

$$u\frac{\partial u}{\partial x} = -\frac{1}{\rho}\frac{\partial P}{\partial x} \tag{3.35}$$

which may be integrated to yield Bernoulli’s equation with negligible gravity forces and is applicable to streamline of potential flow

$$\frac{P}{\rho g} + \frac{u^2}{2g} = constant \tag{3.36}$$

Note that if the velocity u increases, the pressure must decrease and vice versa, that is, the Total Head at any section is a constant. Generally, laminar flows can be solved from the Navier-Stokes equations if the geometry of the flow is simple. Potential flows can be described by Bernoulli's equation.(Nakayama, 1998)

3.3.2 **Laminar Flow in Pipes**

3.3.2.1 **Transition Length**

Let us first consider steady laminar flow in pipes. Such pipes are often termed capillary tubes because the small diameters usually result in laminar flow. However, it should be recognized that low velocities or large viscosities can also result in laminar flow in pipes of larger diameter. As fluid enters a pipe (Figure 3-10) the velocity profile is constant at a value u if there is rounding of the inlet. The fluid velocity at the pipe wall is zero, and this layer of fluid exerts considerable shear forces on the inner layers whose velocities must exceed u to satisfy the law of continuity. The boundary layer thus formed increases in thickness until the center of the pipe is reached. The velocity profile then becomes parabolic and remains parabolic throughout the length of pipe. Let u_0 denote the peak velocity of the entrance velocity profiles. For a parabolic profile, the peak velocity is $2u$. The ratio u_0/u then varies from unity at the entrance to two at the length of pipe where a parabolic velocity profile is established. The inlet length where the peak velocity is within 1% of the final peak velocity of $2u$ $(i.e., u_0/u = 1.98)$ is called the transition length. The transition length for laminar flow is

$$L_t = 0.0575DRe \tag{3.37}$$

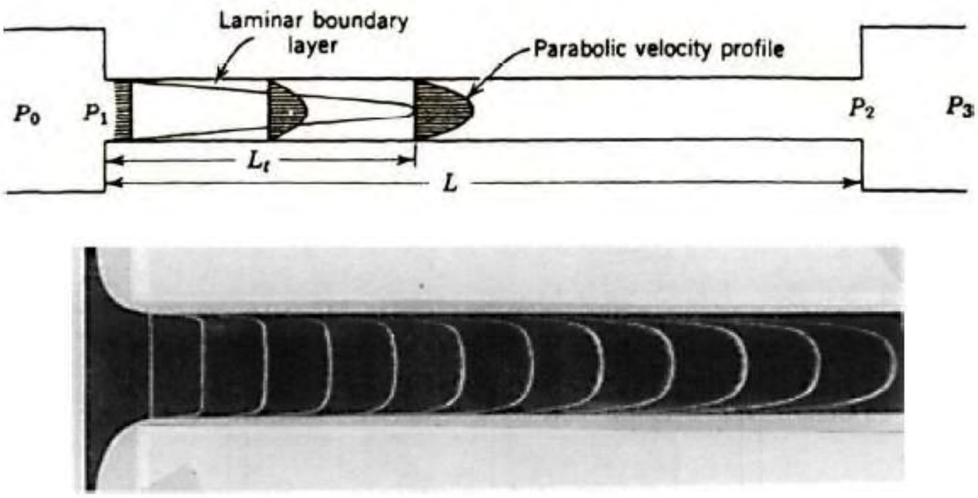


Figure 3-10 - Laminar Flow in pipe.
(Herbert Merritt - Hydraulic control systems)

Laminar flow is dominated by viscous forces when it achieves a parabolic velocity profile. However, both inertia and viscous forces affect the pressure drop in the transition length. (Manring, 2005)

3.3.2.2 **Pressure Drop After the Transition Length**

i. Hagen-Poiseuille Law: $(L/DRe > 0.001)$

$$\frac{P_0 - P_2}{\frac{1}{2}\rho u_{avg}^2} = 64\frac{L}{D}\left(\frac{1}{Re}\right) + 2.28 \tag{3.38}$$

Combining 3.32, 3.33 and 3.38, yields

$$P_0 - P_2 = \frac{128\mu LQ}{\pi D^4}\left(1 + \frac{2.28}{64}\frac{DRe}{L}\right) \quad (3.39)$$

The first term in (3.39) is the well-known Hagen-Poiseuille law for fully developed laminar flow in pipes. The second term accounts for losses due to fluid inertia because the inner layers are being accelerated in the transition length.
In order to include entrances and exits for example, an abrupt square-edged entrance from a large reservoir has a loss coefficient of 0.5. An abrupt exit into a large reservoir has a loss coefficient of 1, which means that all the kinetic energy of the issuing fluid is lost in the turbulent mixing with fluid in the reservoir; hence, $P_2 \approx P_3$. These losses can be factored into (3.39) by replacing the value 2.28 by 2.78. Thus, the pressure drop in a capillary with sharp-edged entrance and exit is

$$P_0 - P_3 = \frac{128\mu LQ}{\pi D^4}\left(1 + \frac{2.78}{64}\frac{DRe}{L}\right) \quad (3.40)$$

ii. Shapiro, Siegel and Kline: $(L/DRe < 0.001)$

$$\frac{P_0 - P_2}{\frac{1}{2}\rho u_{avg}^2} = 1 + 13.74\sqrt{\frac{L}{DRe}} \quad (3.41)$$

which applies for a short tube with well-rounded entry.
If we include a loss coefficient of 0.5 for a square-edged entry, assuming a square edged exit such as $P_2 \approx P_3$ and because $u_{avg} = Q/A$, we obtain (Merritt, 1967),

$$P_0 - P_3 = \left(1.5 + 13.74\sqrt{\frac{L}{DRe}}\right)\frac{1}{2}\,\rho\left(\frac{Q}{A}\right)^2 \quad (3.42)$$

3.3.2.3 **Friction Factor for Laminar Flow**

Equating (3-46 seen later) to the Hagen-Poiseuille law, the friction factor for laminar flow

$$f = \frac{64}{Re} \quad (3.43)$$

3.3.2.4 **Pressure Drop for Steady Laminar Flow with Friction**

However, a much more convenient representation for laminar flow can be obtained as follows. The Hagen-Poiseuille law with friction factor, applicable to fully developed laminar flow, can be written as

$$\frac{P_1 - P_2}{L} = \frac{128\mu}{\pi D^4}Q \quad (3.44)$$

(*a*) Laminar flow through an elliptical tube

$$Q = \frac{\pi a^3 b^3}{4\mu L(a^2 + b^2)}(P_1 - P_2)$$

If $a = b = r$, then Hagen-Poiseuille law for a circular tube is obtained:

$$Q = \frac{\pi r^4}{8\mu L}(P_1 - P_2)$$

(*b*) Laminar flow through rectangular passages ($w \geq h$, i.e., w is selected to be the larger dimension)

$$Q = \frac{wh^3}{12\mu L}\left[1 - \frac{192h}{\pi^5 w}\tanh\frac{\pi w}{2h}\right](P_1 - P_2)$$

If $w = h$ (square cross section)

$$Q = \frac{w^4}{28.4\mu L}(P_1 - P_2)$$

If $w \gg h$, then

$$Q = \frac{wh^3}{12\mu L}(P_1 - P_2)$$

(*c*) Laminar flow through triangular passages
Equilateral triangle cross section

$$Q = \frac{s^4}{185\mu L}(P_1 - P_2)$$

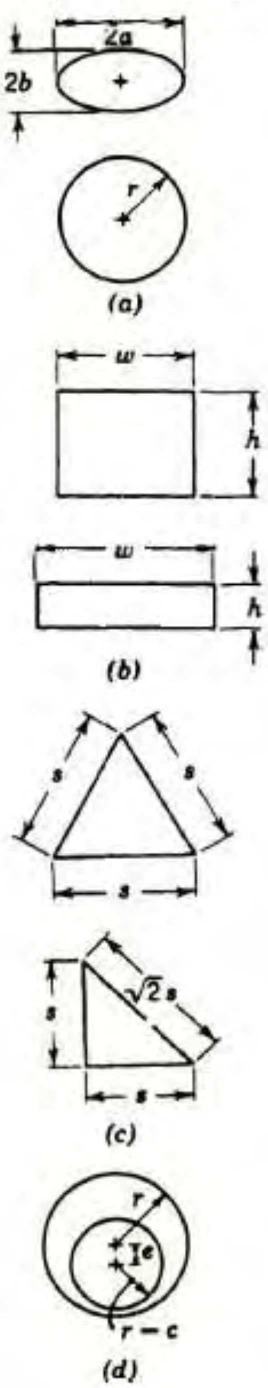
Right angle triangle cross section

$$Q = \frac{s^4}{155.5\mu L}(P_1 - P_2)$$

(*d*) Laminar flow in annulus between circular shaft and cylinder ($c \ll r$)

$$Q = \frac{\pi r c^3}{6\mu L}\left[1 + \frac{3}{2}\left(\frac{e}{c}\right)^2\right](P_1 - P_2)$$

Passage cross sections



Nomenclature:

Q = volumetric flow rate, in.^{3}/sec
$P_1 - P_2$ = pressure drop in direction of flow, psi
L = passage length, in.
μ = fluid viscosity, lb-sec/in.^{2}
w = passage width, in.
h = passage height, in.
r = tube radius, in.
c = radial clearance, in.
e = eccentricity of shaft, in.
a, b = axes of ellipse, in.
s = side of a triangle, in.

Figure 3-11 - Laminar Flow through various passages with cross sections illustrated. (Noah Manring, Hydraulic Control Systems. Wiley, 2005.)

3.3.2.5 **Pressure Drop in the Transition Length**

If the length of pipe is less than the transition length (Manring, 2005), then the pressure drop is determined from the curve in Figure 3-12. A trial and error solution is required if the pressure drop is known and flow is desired. Again, the entrance and exit losses must be added to the result.

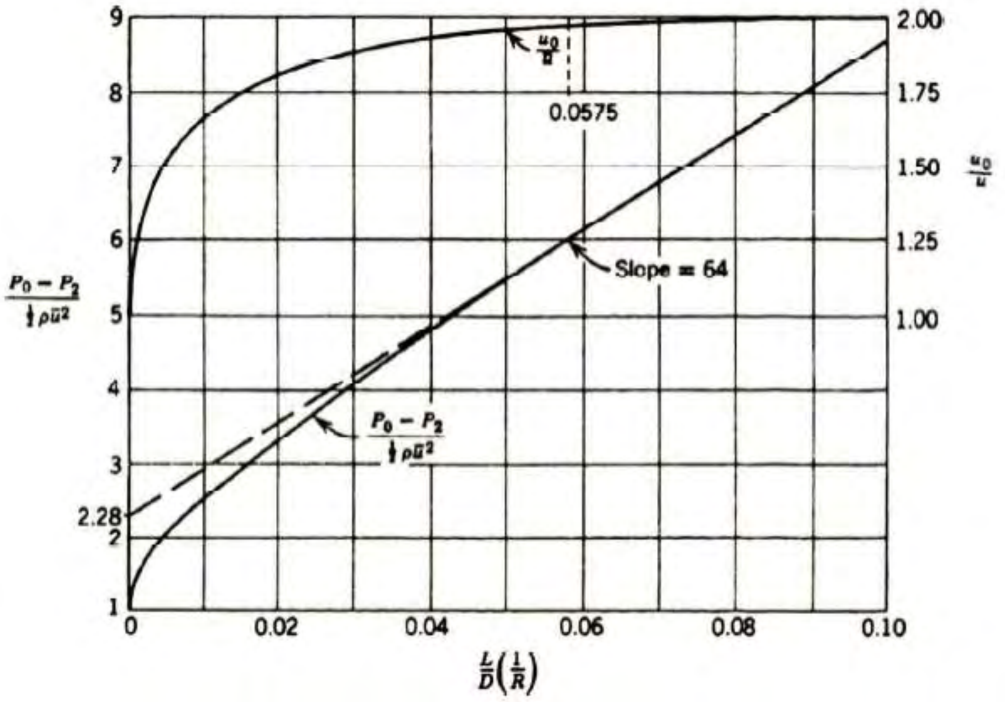


Figure 3-12 - Pressure Drop and peak velocity of Steady Laminar Flow in a pipe. (Herbert Merritt - Hydraulic control systems)

3.3.2.6 **Tube Length and Laminar Flow**

A concern to the designer is the length of tube required for laminar flow to be dominant. Taking the extreme limit of laminar flow, $R = 2000$ and selecting a length to diameter ratio of 800, we find that the parenthesis in (3.40) has a value of 1.11. Therefore, ratios of $L/D > 800$ give no more than 11 % error in pressure drop when computed from the Hagen-Poiseuille law. If $R < 2000$, correspondingly shorter lengths can be used. Using 10% error in pressure computation as a criterion and referring to (3.40), we have design ratios given by

$$\frac{L}{D} \geq 0.434Re \tag{3.45}$$

This relation insures that the capillary is the dominant resistance when pressures are measured in the end reservoirs.

3.3.3 **Turbulent Flow in Pipes**

Flow patterns and equations for turbulent flow in pipes (Merritt, 1967) are based largely on experimental observations. As flow enters the pipe (Figure 3-13), the initial boundary layer is laminar but becomes turbulent (except for a very thin laminar sublayer) after a very short distance. This turbulent boundary layer increases in thickness to the center of the pipe in a transition length of about 25 to 40 pipe diameters. A rather blunt velocity profile, with a peak velocity of about $1.2u$, is then established and remains throughout the pipe length.

Figure 3-13 - Turbulent Flow in a pipe.
(Herbert Merritt - Hydraulic control systems)

3.3.3.1 **Pressure Drop for Steady Turbulent Flow**

The empirical Darcy-Weisbach equation giving the pressure drop for fully developed turbulent flow is

$$P_1 - P_2 = f\frac{L}{D}\frac{\rho u_{avg}^2}{2} \tag{3.46}$$

Where f is the friction factor which depends on Reynolds number and pipe roughness. The additional pressure drop due to the transition length is about $0.09\ \rho u_{avg}^2/2$ and is negligible in most computations. Pressure drops due to entrance and exit losses are also usually negligible.

3.3.3.2 **Friction Factor of Turbulent Flow in Smooth Pipes**

In the turbulent flow range Blasius experimentally determined the friction factor to be

$$f = \frac{0.3164}{Re^{0.25}} \tag{3.47}$$

Prandtl's universal law of friction for smooth pipes (3.48) is applicable

$$\frac{1}{\sqrt{f}} = 2log_{10}\left(Re\sqrt{f}\right) - 0.8 \tag{3.48}$$

for arbitrarily large Reynolds numbers but is somewhat difficult to manipulate mathematically. However, it covers most cases in hydraulic control because the pipes are smooth and the flow velocities are normally kept below $15\ ft/sec$ to avoid large pressure surges with sudden valve closures (see water-hammer), and this results in Reynolds numbers being less than 10^5.
If the flow is known and the pressure drop is required, then Reynolds number can be directly computed and the friction factor selected from Figure 3-14. The pressure drop is then obtained from (3-46). (Durfee, Sun, & Ven, 2015) (Merritt, 1967)

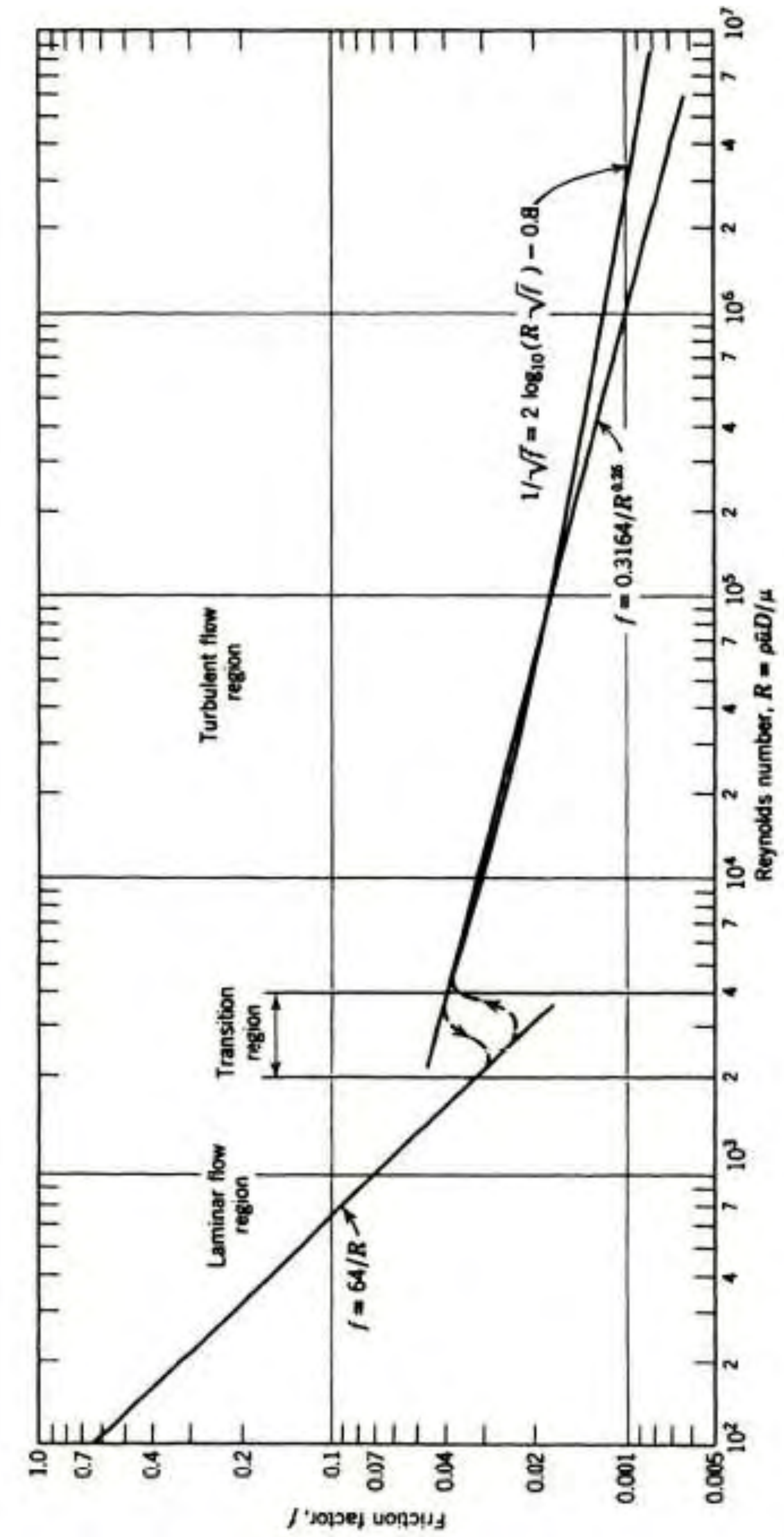


Figure 3-14 - Friction Factor for smooth pipes (Herbert Merritt - Hydraulic control systems)

If needed, the turbulent friction factor data for rough piles in the Moody diagram (Figure 3-15).

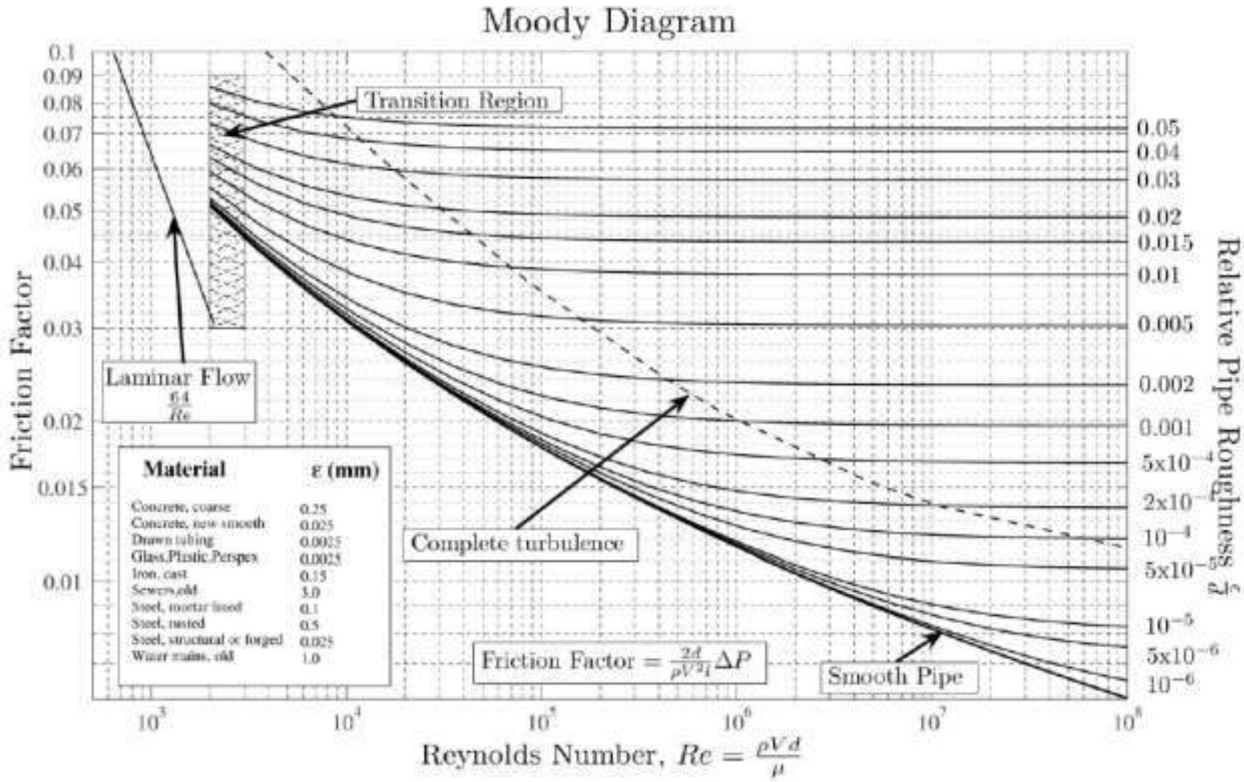


Figure 3-15 - The Moody Diagram where the friction factor for rough pipes in the turbulent region can be found. (Wikipedia)

For fully developed turbulent flow for smooth pipes (with friction factor), equations (3.33), (3-46), and (3-47) may be combined to yield

$$\frac{P_1 - P_2}{L} = 0.242\frac{\mu^{0.25}\rho^{0.75}}{D^{4.75}}Q^{1.75} \qquad (3.49)$$

$where$
$P_1 - P_2 = pressure\ drop, (psi)$
$Q = volumetric\ flow\ rate, (in.^3/sec)$
$L = pipe\ length, (in.)$
$D = pipe\ inside\ diameter, (in.)$
$\rho = fluid\ mass\ density, (lb - sec^2/in^4)$
$\mu = fluid\ absolute\ viscosity, (lb - sec/in^2)$

For a given fluid and selected inside diameters, the Hagen-Poiseuille law (3.44) is plotted for $R < 2000$, and the Darcy-Weisbach equation (3.46) is plotted for $R > 4000$ as illustrated in Figure 3-16. Such plots are very useful in design since flow, pressure drop, and pipe size arc read directly without explicit computation of Reynolds number. Note the increase in the rate of pressure gradient along the pipe with flow in the turbulent region. For this reason, laminar flow is desirable; however, the resulting pipe is usually unnecessarily large. Usually the flow is determined from load velocity requirements, and the pipe size is selected so that the pressure drop is moderate. Pipe selection criteria of $15\ ft/sec$ maximum flow velocity and $1\ psi/ft$ pressure drop are common.

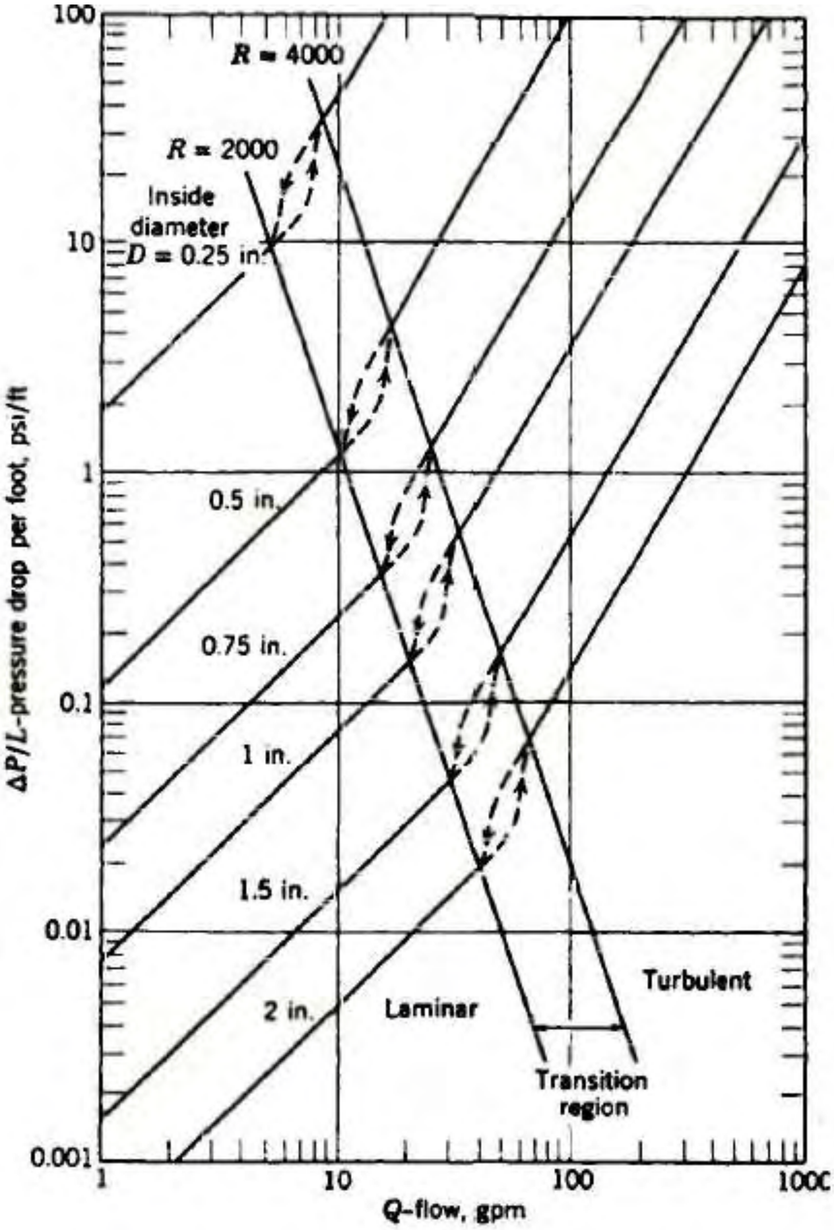


Figure 3-16 - Pressure Drop per foot for smooth pipe. Fluid density is $0.0307\ lb/in^3$ and viscosity is $4\ x\ 10^{-6}\ lb - sec/in^2$.
(Herbert Merritt - Hydraulic control systems)

3.4 Flow through Orifices

Orifices are a basic means for the control of fluid power (Manring, 2005) (Merritt, 1967). Flow characteristics of orifices play a major role in the design of many hydraulic control devices. An orifice is a sudden restriction of short length (ideally zero length for a sharp-edged orifice) in a flow passage and may have a fixed or variable area. Two types of flow regime exist (Figure 3-17), depending on whether inertia or viscous forces dominate. The flow velocity through an orifice must increase above that in the upstream region to satisfy the law of continuity. At high Reynolds numbers, the pressure drop across the orifice is caused by the acceleration of the fluid particles from the upstream velocity to the higher jet velocity. At low Reynolds numbers, the pressure drop is caused by the internal shear forces resulting from fluid viscosity.

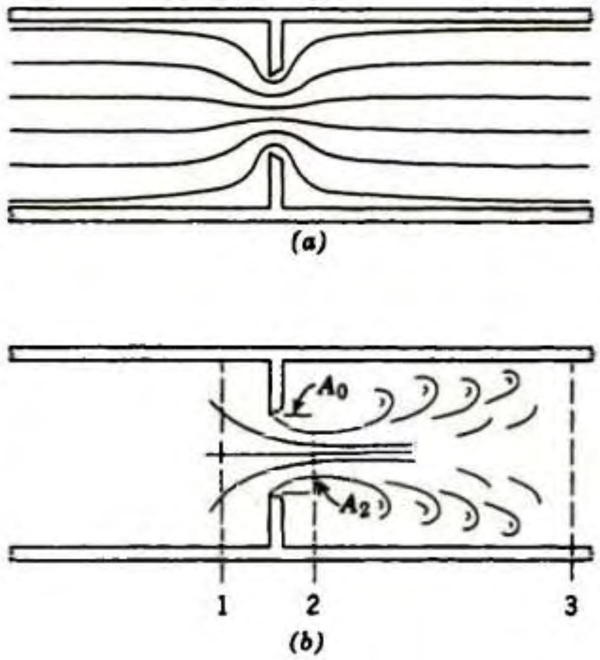


Figure 3-17 - Flow through an orifice. (a) Laminar, (b) Turbulent Flow. (Herbert Merritt - Hydraulic control systems)

Orifices and the orifice equations have the following applications:

- Regulating the flow out of detention ponds
- Regulating the flow through channels in the form of radial and sluice gates
- Approximating the interception capacity of submerged drainage inlets in sag
- Approximating the flow allowed through a submerged culvert operating under inlet control
- Measuring flow

3.4.1 Turbulent Orifice Flow

Since most orifice flows occur at high Reynolds numbers, this region is of greater importance. Such flows are often referred to as “turbulent”, but the term does not have quite the same meaning as in pipe flow. Referring to Figure 3-17b, the fluid particles are accelerated to the jet velocity between points 1 and 2. The flow between these points is streamline or potential flow and experience justifies the use of Bernoulli’s equation in this region. The area of the issuing jet is smaller than the orifice area because the fluid particles have inertia and are moving in a curved path at the orifice opening. The point along the jet where the jet area becomes a minimum is called the *vena contracta*. The ratio of stream area at the vena contracta A_2 to the orifice area A_0 is called the contraction coefficient C_c.

$$A_2 = C_c A_0 \tag{3.50}$$

For round orifices, the vena contracta occurs at approximately half an orifice diameter downstream and point 1 is about the same distance upstream (for a slit type orifice, these same distances are about $b/2$). Thus, the fluid is accelerated in a total distance of about one orifice diameter. Between points 2 and 3 of Figure 3-17b, there is turbulence and violent mixing of the issuing jet with the fluid in the downstream region. The kinetic energy of the jet is converted into an increase in internal energy (temperature) of the fluid by the turbulence. Since the kinetic energy of the jet is not recovered, pressures P_2 and P_3 are approximately equal. This may be shown by analysing the section between 2 and 3 as a sudden expansion.
The pressure difference required to accelerate the fluid particles from the lower upstream velocity to the higher jet velocity is found by applying Bernoulli's equation between points 1 and 2. Therefore

$$u_2^2 - u_1^2 = \frac{2}{\rho}(P_1 - P_2) \tag{3.51}$$

And for incompressible flow

$$A_1 u_1 = A_2 u_2 = A_3 u_3 \tag{3.52}$$

Combining 3.51 and 3.52,

$$u_2 = \frac{1}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \sqrt{\frac{2}{\rho}(P_1 - P_2)} \tag{3.53}$$

Because of viscous 'friction, the jet velocity is slightly less than that given by (3.53), and an empirical factor called the velocity coefficient C_v, is introduced to account for this discrepancy. C_v is usually around 0.98 and is approximated by unity in most computations. Since $Q = A_2 u_2$, the volumetric flow rate at the vena contracta then becomes

$$Q = \frac{C_v A_2}{\sqrt{1 - \left(\frac{A_2}{A_1}\right)^2}} \sqrt{\frac{2}{\rho}(P_1 - P_2)} \tag{3.54}$$

Because it is more convenient to use orifice area rather than vena contracta area, (3.54) and (3.50) can be combined to yield

$$Q = C_d A_0 \sqrt{\frac{2}{\rho}(P_1 - P_2)} \tag{3.55}$$

where C_d called the discharge coefficient, is given by

$$C_d = \frac{C_v C_c}{\sqrt{1 - C_c^2 \left(\frac{A_0}{A_1}\right)^2}} \tag{3.56}$$

Since $C_v \approx 1$ and A_0 is usually much less than A_1, the discharge coefficient is approximately equal to the contraction coefficient. The contraction coefficient is difficult to compute but solutions have been made for round and slit-type sharp-edged orifices and are plotted in Figure 3-18. Experience shows that the theoretical value of $C_c = \pi/(\pi + 2) = 0.611$, can be used for all sharp-edged orifices, regardless of the particular geometry, if the flow is turbulent and $A_0 \ll A_1$. For this reason, a discharge coefficient of $\mathrm{C_d} \approx 0.60$ is often assumed for all orifices and, since $C_d\sqrt{2/\rho} \approx 100\ (in^2/\sqrt{Ib} - sec)$, the orifice equation takes the familiar form

$$Q = 100 A_0 \sqrt{P_1 - P_2} \tag{3.57}$$

$Where,$
$P_1, P_2 = pressures\ (psi).$
$A_0 = orifice\ area\ (in^2).$
$Q = volumetric\ flow\ rate\ (in^3/sec).$

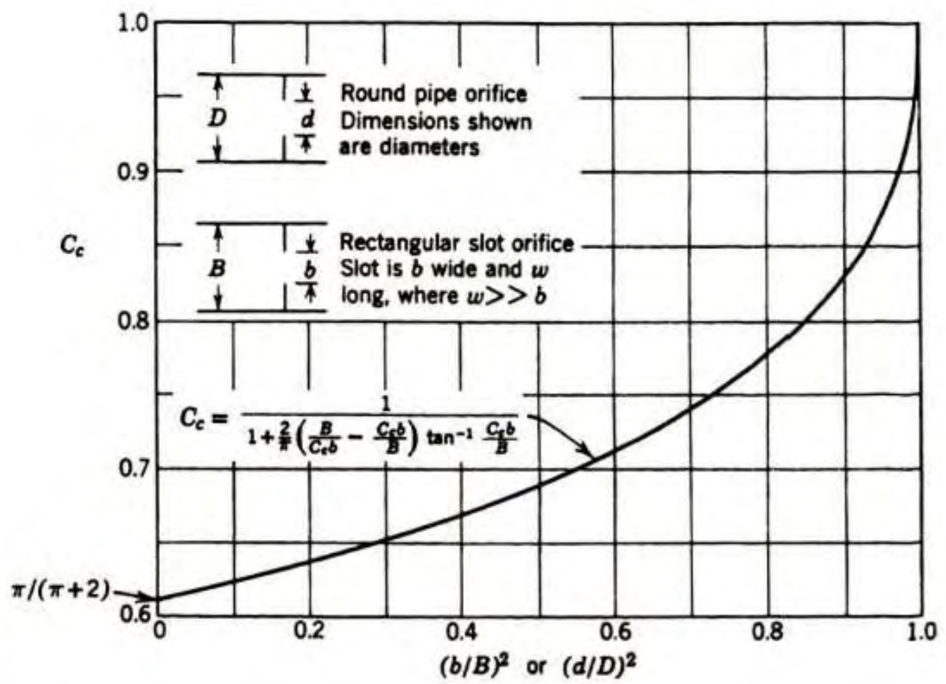


Figure 3-18 - Contraction Coefficients for round and slot type orifices. (Herbert Merritt - Hydraulic control systems)

Sharp-edged orifices are desirable for their predictable characteristics and insensitivity to temperature changes. However, cost frequently prohibits their use, especially as fixed restrictors, and orifices with length are often employed. An average discharge coefficient for such short tube orifices can be obtained as follows: Comparing (3.55) with (3.38) and (3.42), respectively, the discharge coefficient plotted in Figure 3-19 can be identified as

$$C_d = \left(1.5 + 13.74\sqrt{\frac{L}{DRe}}\right)^{-\frac{1}{2}} \quad for \quad \frac{DRe}{L} > 50$$

(3.58)

$$C_d = \left(2.28 + 64\frac{L}{DRe}\right)^{-1/2} \quad for \quad \frac{DRe}{L} < 50$$

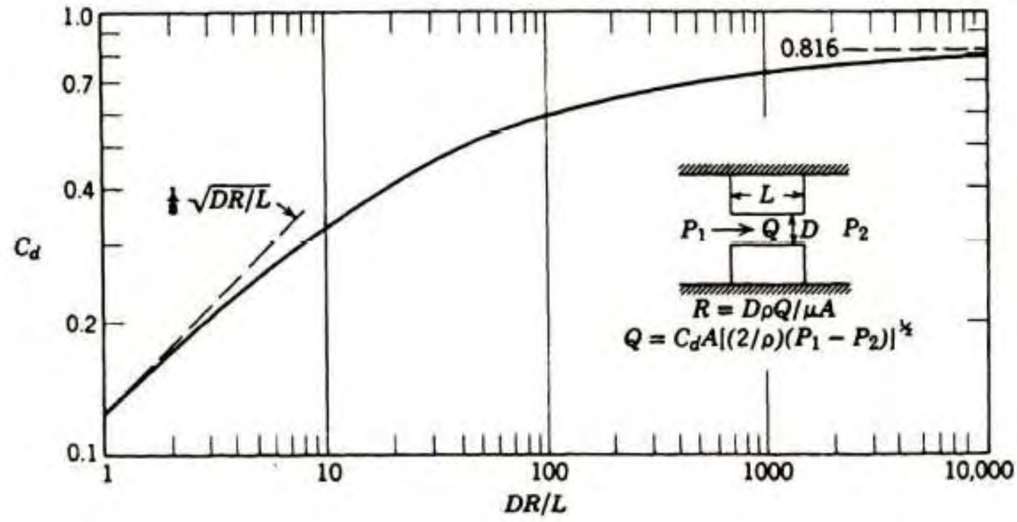


Figure 3-19 - Discharge Coefficient for short tube orifice. (Noah Manring, Hydraulic Control Systems. Wiley, 2005.)

3.4.2 Laminar Orifice Flow

At low temperatures (Merritt, 1967), low orifice pressure drops, and/or small orifice openings, the Reynolds number may become sufficiently low to permit laminar flow. Reynolds number for an orifice is defined by

$$Re = \frac{\rho\left(\frac{Q}{A_0}\right)D_h}{\mu} \tag{3.59}$$

$where,$
$Q/Ao\ is\ the\ jet\ velocity\ at\ the\ orifice\ opening\ and$
$Dh\ is\ the\ hydraulic\ diameter\ of\ the\ opening.$

For a circular orifice of diameter d the hydraulic diameter is $Dh = d$. For a rectangular slit orifice of width w and height b where $w >> b$, the hydraulic diameter, defined by (3.34), becomes

$$D_h = \frac{4bw}{2(b+w)} \approx 2b \tag{3.60}$$

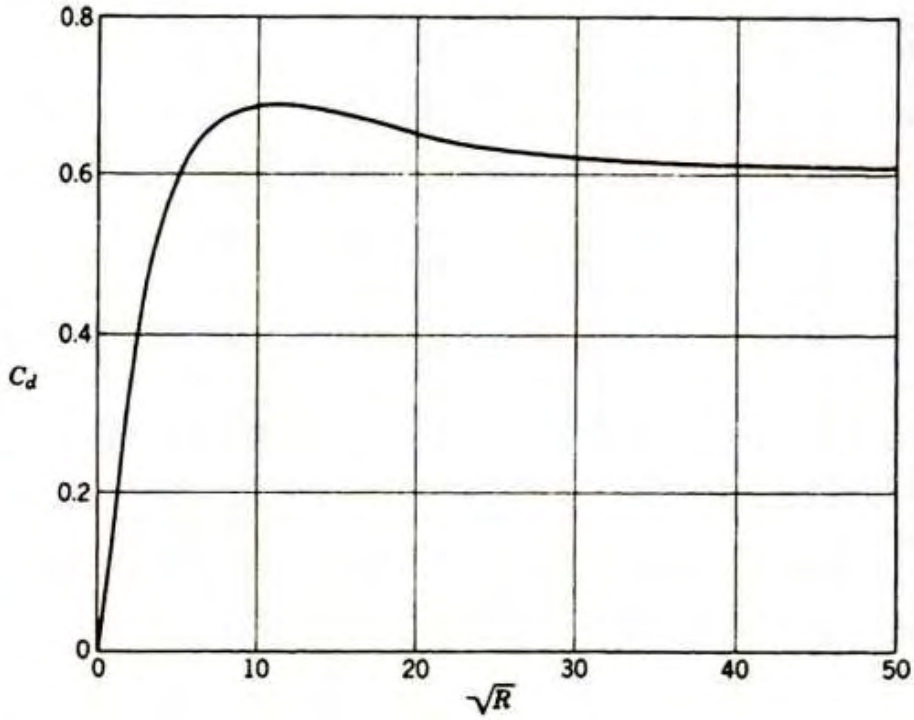


Figure 3-20 - Typical plot of Discharge Coefficient versus Reynolds Number for an orifice. (Noah Manring, Hydraulic Control Systems. Wiley, 2005.)

Although the analysis leading to (3.55) is not valid at low Reynolds numbers, attempts have been made to extend this equation to the laminar region by plotting discharge coefficient as a function of Reynolds number. A typical plot of such data is shown in Figure 3-20.

For $Re < 10$ many investigators have found the discharge coefficient to be directly proportional to the square root of Reynolds number; that is

$$C_d = \delta\sqrt{Re} \tag{3.61}$$

The quantity δ depends on geometry and is called the laminar flow coefficient. Substituting (3.59) and (3.61) into (3.55) yields

$$Q = \frac{2\delta^2 D_h A_0}{\mu}(P_1 - P_2) \tag{3.62}$$

for low Reynolds numbers. Note that flow is directly related to pressure difference and, since mass density is absent, dominated by fluid viscosity.

Wuest has theoretically determined expressions for laminar flow through sharp-edged orifices. For a circular orifice in an infinite plane (i.e., $d << D$ in Figure 3-18), the result is

$$Q = \frac{\pi d^3}{50.4\mu}(P_1 - P_2) \tag{3.63}$$

For a rectangular slit of height b and width w in an infinite plane (i.e., $b \ll B$ in Figure 3-18) with $w \gg b,$ the result is

$$Q = \frac{\pi b^2 w}{32\mu}(P_1 - P_2) \tag{3.64}$$

Equating (3.62) to (3.63) and to (3.64) gives $\delta = 0.2$ for a sharp-edged round orifice and $\delta = 0.157$ for a sharp-edged slit orifice.

Viersma represents the discharge coefficient by asymptotes defined by (3.61) in the laminar and $Cd = 0.611$ in the turbulent regions as shown in Figure 3-21. The transition Reynolds number Re_t is defined by the intersection point of the two asymptotes, that is

$$Re_t = \left(\frac{0.611}{\delta}\right)^2 \tag{3.65}$$

For $\delta = 0.2$, the transition Reynolds number is $Re_t = 9.3$ and increases as δ is decreased.

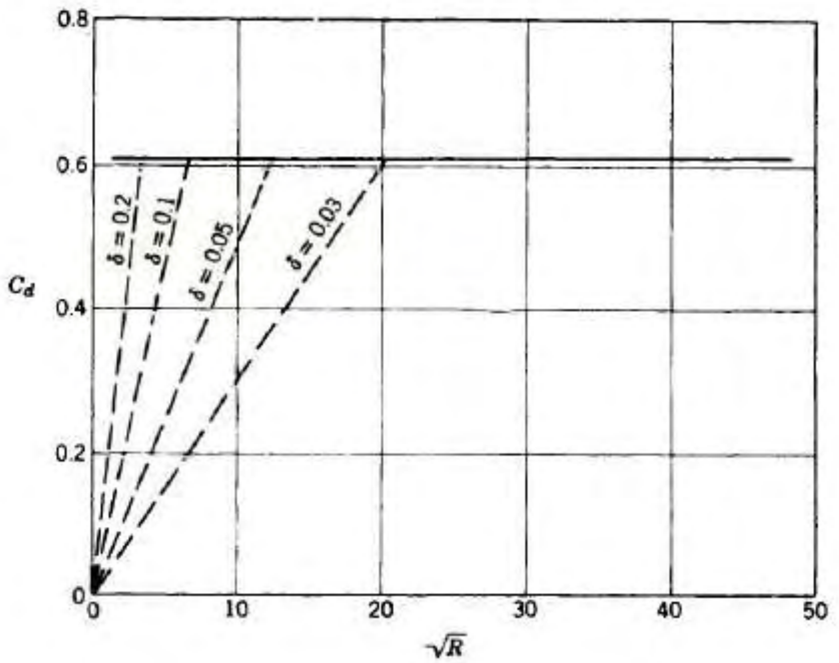


Figure 3-21 - Asymptotic approximation of Discharge Coefficient. (Herbert Merritt - Hydraulic control systems)

In many instances it is desirable to evaluate the flow coefficient, $\Delta Q/\Delta(P_1 - P_2)$ of an orifice with no initial pressure drop. If turbulent flow is assumed, this coefficient is infinite as seen by differentiation of (3.55). However, the flow becomes laminar when the pressure drop is small and the flow coefficient has a finite value which can be estimated from (3.63) and (3.64).

In summary, orifice flow is laminar for $R < Re_t$ with flow rates directly related to pressure drop as given by (3.62). Near Re_t,both inertia and viscosity are important.

For $R > Re_t$ the flow can be treated as turbulent and described by the orifice equation (3.55). The orifice equation is commonly used for all situations with a total disregard for the types of flow that can be encountered. This is justified in most cases but can lead to gross errors in certain instances.

3.5 Minor Losses

The term minor losses (Merritt, 1967) (Shollenberger) (Taborek, 1959) refers to those energy loses caused by bends fittings and sudden changes in flow cross section. These losses are empirically described by

$$H_L = K\frac{u^2}{2g} = \frac{K}{2g}\left(\frac{Q}{A}\right)^2 \tag{3.66}$$

$Where,$
$u\ is\ the\ fluid\ velocity$
$Q\ is\ the\ volumetric\ flow\ rate$
$A\ is\ the\ passage\ area\ (if\ two\ areas, the\ one\ of\ the\ smaller\ cross\ section\ is\ used)$
$K\ is\ called\ the\ resistance\ or\ loss\ coefficient$

Values for K are given in Figure 3-22 for many cases of interest.

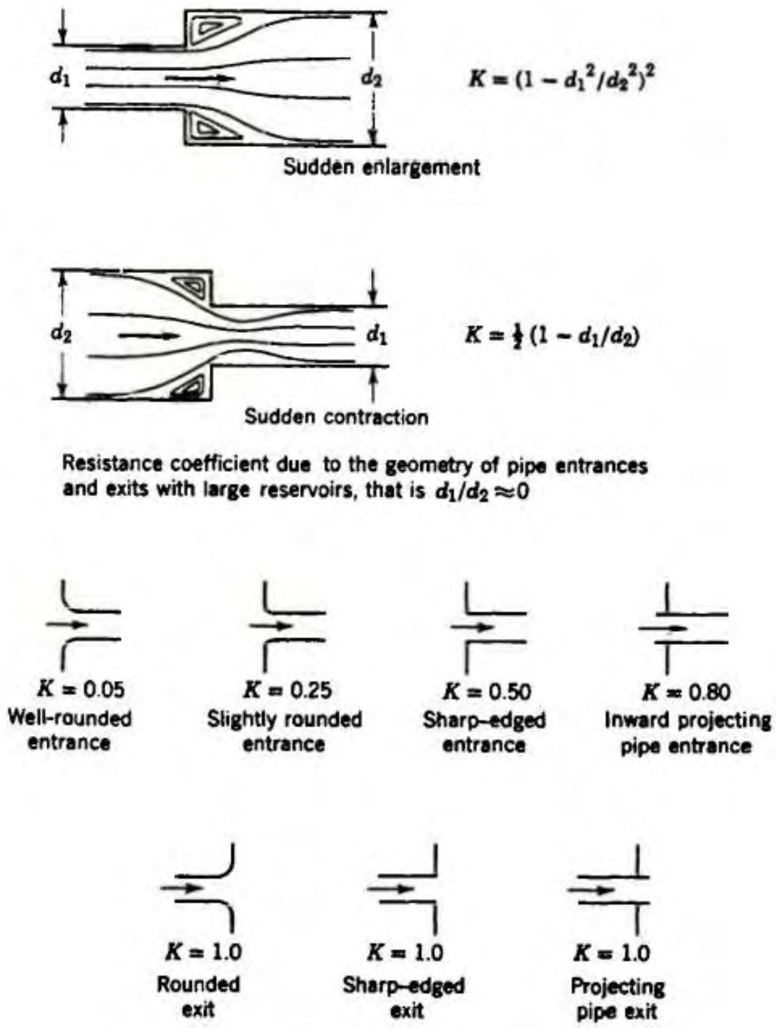


Figure 3-22 - Resistance Coefficients due to abrupt changes in pipe cross section and due to the geometry of pipe entrances and exits.
(Noah Manring, Hydraulic Control Systems. Wiley, 2005.)

Bernoulli's equation applied between points 2 and 3 of the abrupt exit in Figure 3-22 would predict a pressure increase. However, there is a loss in energy which must be considered. Because $K = 1$ for all reasonably abrupt exits into a large reservoir (i.e., $d_2/d_1 >> 1$ the head loss is $u^2/2g$. This means that the entire kinetic energy of the fluid entering the reservoir is converted into heat energy by the turbulent mixing which takes place. Hence, there is little or no recovery in pressure during the expansion so that $P_2 \approx P_3$. The exit must be smooth and diverge gradually to achieve significant pressure recovery. For most cases in hydraulic control it is sufficient to assume that $P_2 = P_3$ for an exit into a large chamber.

When fluid flow encounters a sudden contraction, a vena contracta is formed between approximately one half to one pipe diameters downstream. Because the conversion of pressure energy into kinetic energy at the inlet is very efficient, most of the energy loss occurs due to the expansion of the fluid stream from the vena contracta. Application of Bernoulli's equation to points 0 and 2 of the sudden contraction in Figure 3-22 yields

$$\frac{P_0}{\gamma} + \frac{u_0^2}{2g} = \frac{P_2}{\gamma} + \frac{u_2^2}{2g} + K\frac{u_2^2}{2g} \tag{3.67}$$

where K is given in Figure 3-22 for different inlet geometries. Because $u_0 \ll u_2$, u_0 can be neglected and we obtain

$$P_0 - P_2 = (1 + K)\frac{\rho}{2}u_2^2 \tag{3.68}$$

Although point 2 is downstream of the vena contracta, the distinction between points 1 and 2 is often overlooked and $(P_0 - P_2)$ is referred to as the inlet pressure drop.

Centrifugal forces and secondary flow patterns result in a pressure drop in pipe bends. There is much variation in test data for resistance coefficients of bends and those values given in Figure 3-23 should be considered approximate. Because the fluid velocities at points 1 and 2 in Figure 3-23 are the same, application of Bernoulli's equation yields

$$P_1 - P_2 = K\frac{\rho}{2}\left(\frac{Q}{A}\right)^2 \tag{3.69}$$

as the pressure drop due to the bend. The pressure drop for the length of the bend is determined from (3.46) and added to that due to the bend. Pressure drops in fittings and valves vary widely and data should be obtained from component manufacturers or by direct measurement.

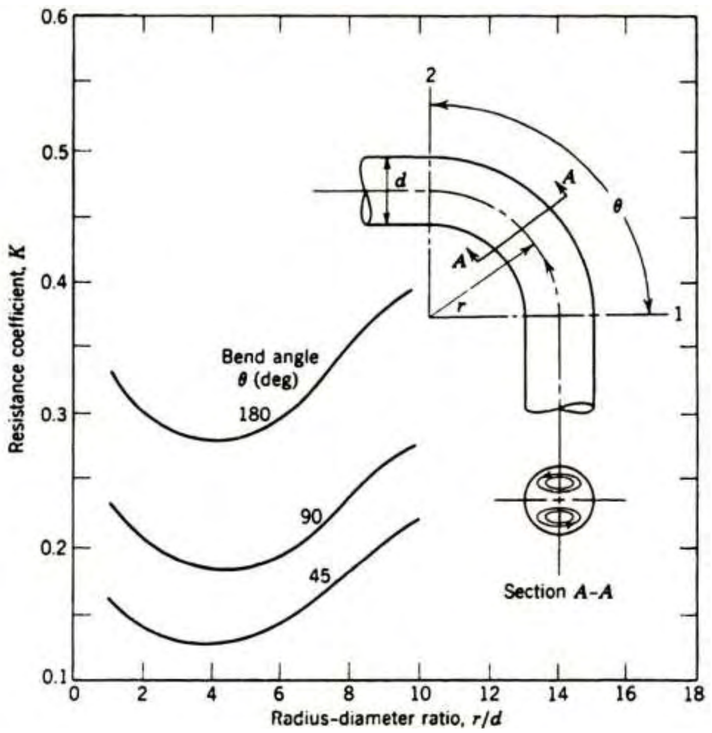


Figure 3-23 - Resistance Coefficients of pipe bends.
(J. J. Taborek, "Fundamentals of Line Flow" 1959)

3.6 Power Loss and Temperature Rise

Hydraulic horsepower is the product of pressure drop and flow ΔPQ, and in many instances this power is consumed by fluid friction and increases the internal energy of the fluid (Merritt, 1967). The power used by all hydraulic resistances such as orifices, valves, pipes, capillaries, and minor losses, is converted into a temperature increase of the fluid. Most of the horsepower produced by hydraulic motors is used as shaft work, but power used by internal and cross leakages is converted into heat.
The power converted into heat energy by a hydraulic resistance is $S_{heat} = 9339\gamma C_p \Delta TQ$. Equating this to the supplied hydraulic horsepower of $S = \Delta P \cdot Q$ yields a temperature rise in the direction of flow of

$$\Delta T = \frac{\Delta P}{9339 C_p \rho g} \tag{3.70}$$

Thus, there is about a $1°F$ rise in temperature across any hydraulic resistance (orifice, pipe, etc.) for each $140\ psi$ drop if the fluid has a petroleum
base for which $\gamma\ =\ 0.03\ lb/in^3$. .
Equation 3.70 may be derived in a more rigorous manner from (3.22). If there is no heat added or work done and the flow is steady (i.e., $W_1 = W_2$) then (3.22) becomes

$$9339 C_p T_1 + \frac{P_1}{\rho g} + \frac{u_1^2}{2g} = 9339 C_p T_2 + \frac{P_2}{\rho g} + \frac{u_2^2}{2g} \tag{3.71}$$

In most cases the inlet and outlet is about the same size so that $u_1 = u_2$ and (3.71) reduces to (3.70).
The temperature rise given by (3.70) is useful in determining the heat generated in hydraulic systems.

Because hydraulic systems generate heat, they must operate above ambient temperature to dissipate the heat to the environment or to reject it to the coolant of a heat exchanger. Operation at excessive temperatures can break down the oil, causing sludges, varnishes, etc., which can clog orifices. Decrease in viscosity and lubricity at elevated temperatures may drastically shorten service life of components such as pumps. Seals, packings, hoses, filters, etc. have a definite temperature range for satisfactory operation. Thermal distortion in hydraulic components and adjacent structures may be undesirable. Therefore, it is quite important that the hydraulic system be designed so that a heat balance is achieved at a satisfactory operating temperature.

There are several sources of heat generation in hydraulic systems.

- The major heat generators are the orifices and valves in the system used to throttle and control the flow. The hydraulic power consumed by these devices is dissipated in heating the fluid and, to a much lesser extent, local heating of a valve itself. It is important to realize that valves are inherently heat generators, but this is the price that must be paid for the ability to control. Relief valves and servo-valves are good examples of heat generators.
- Another source of heat generation is the resistive pressure drops in hydraulic lines, fittings, filters, and passageways in components such as valves, motors, and heat exchangers. Undersized and/or dirty passages should be avoided to minimize heat losses.

- Leakage flow losses in pumps, motors, and valves add to the heat generated by a system.
- Seal friction, mechanical friction, windage losses, and viscous drag between surfaces in pumps and motors also generate heat.
- The compression of oil and, especially, entrained air to higher pressures in pumps during the pumping portion of the stroke causes heat generation. Rapidly cycling of gas-charged accumulators can cause gas temperatures higher than that of the oil which results in heat flow to the oil.

Each of the latter four sources of heat are usually much smaller than that produced by the metering orifices of control valves but collectively represent a significant contribution. The hydraulic system can absorb heat from external sources such as prime movers which also must be taken into account.
The operating temperature of the oil in a system should be determined to see whether it is satisfactory. This requires a computation of the heat generated and the natural heat dissipation capability of the system. One method of determining the heat generated is to add up the losses of each component. This technique is tedious and requires estimation of many quantities, such as leakage rates, efficiencies, and pressure drops. Because the hydraulic horsepower initially generated must be used in mechanical power at the output or converted into heat, a far better technique is to compute the total hydraulic horsepower generated by the pump (or pumps) and subtract the mechanical power delivered to the load by actuation devices (pistons or motors) to yield the heat that must be dissipated.
The mechanical horsepower and heat power equivalents of hydraulic pressure and flow will prove useful. Manipulation of appropriate conversion constants yields the horsepower developed as

$$hp = \frac{\Delta P\ Q}{1714.3} \tag{3.72}$$

and the equivalent heat power as

$$q = 2540\ hp = 1.485\ \Delta P\ Q \tag{3.73}$$

where
$\Delta P\ =\ pressure\ difference\ across\ device, (psi)$
$Q\ =\ flow\ through\ device, (gal/min)$
$q\ =\ heat\ power, (Btu/hr)$
$hp\ =\ horsepower\ (1\ hp\ =\ 550\ ft-lb/sec\ =\ 42.4\ Btu/min)$

The hydraulic horsepower initially generated depends on the type of power supply. For a constant pressure supply with a bypass type (i.e., relief valve) regulator, full power is generated at all times and the heat power is

$$q|_{PF} = 1.485 P_s Q_p\ \ \frac{Btu}{hr} \tag{3.74}$$

$where$
$P_s\ =\ constant\ supply\ pressure, (psi)$
$Q_p\ =\ (60/231) Dp \cdot N\ =\ ideal\ pump\ flow, (gal/\mathrm{min})$

Because the total power generated is desired, the ideal rather than actual pump flow (actual is the ideal flow minus leakage) is used because the difference represents a heat loss.

If the constant pressure supply used a stroke regulated variable delivery pump, the heat power will be less and depends on the load flow, Q_L. Therefore

$$q|_{PV} = 1.485 P_s Q_p \left(\frac{Q_L}{Q_p}\right) \tag{3.75}$$

where now denotes the pump flow at maximum stroke and the ratio Q_L/Q_p is between zero and one. An analysis of the load duty cycle is necessary to establish Q_L. If the duty cycle is not repetitive, then Q_L/Q_p might be estimated at say 0.5 or the most conservative value of unity used. It should be clear that the two equations given do not represent the heat generated by the pump but rather the heat power equivalent of the generated hydraulic power.

The load duty cycle must now be analysed to determine the average mechanical power delivered to the load. If the output actuator is holding a given position during most of the cycle, no power is consumed. This is usually the case in servo-controlled systems where nearly all the generated power is eventually dissipated as heat. Systems using a variable delivery pump have much less heat dissipation when the actuator is holding position because the pump flow is reduced to only that necessary to supply leakage losses. However, the heat losses would be comparable to that of a bypass supply if the actuator moved at high velocities and required low pressure differences, but such loads are exceptional.

The heat power to be dissipated is the equivalent heat power generated at the pump minus the mechanical power at the actuator. Heat is dissipated in hydraulic systems by the three basic methods of conduction, radiation, and convection. Heat conduction to adjacent structures is the principal cooling means when the reservoir is built-in. The heat conducted is given (Bontozoglou, 2003) by

$$q = kA\frac{dT}{dx} \quad \frac{Btu}{hr} \tag{3.76}$$

$where$
$k\ is\ the\ thermal\ conductivity\ of\ the\ structure\ material, (Btu/in - hr - °F)$
$dT/dx\ is\ the\ thermal\ gradient\ in\ the\ direction\ of\ heat\ flow, (°F/in)$
$A\ is\ the\ area\ normal\ to\ heat\ flow\ path, (in^2)$

For complex and irregular structures, as they all are, it is obviously difficult to identify the thermal gradients, and specific tests are required to determine the oil temperature.

Separate reservoirs are self-cooled by radiation and, to a lesser degree, by free convection of heat from the heat sink formed by the mass of the oil, reservoir, housings, and tubing. The heat transferred from separate reservoirs is usually written

$$q = U\ A\ \Delta T \tag{3.77}$$

where
$U\ =\ overall\ heat\ transfer\ coefficient, (Btu/in^2 - hr - °F)$
$\Delta T\ =\ temperature\ difference\ between\ oil\ and\ ambient, (°F)$
$q\ =\ heat\ dissipation\ rate, (Btu/hr)$
$A =\ surface\ area\ of\ reservoir, (in^2)$
$(usually\ taken\ as\ the\ area\ of\ the\ sides\ and\ bottom, if\ above\ the\ floor, but\ not\ the\ top)$

3.7 Pressure Transients in Hydraulic Conduits

3.7.1 **Waterhammer**

When fluid flowing in a conduit is suddenly stopped due to a rapid valve closure at the end of the conduit, a very large pressure transient may result (Manring, 2005) (Merritt, 1967). This phenomenon is called waterhammer because it is usually accompanied by considerable noise. The fluid adjacent to the valve is stopped initially and a pressure wave, which heads the increasing amount of fluid being brought to a standstill, travels back to the fluid source at velocity c given by

$$c = \left(\frac{\beta_e}{\rho}\right)^{\frac{1}{2}} \tag{3.78}$$

$where$
$c\ =\ velocity\ of\ sound\ in\ the\ fluid, (in./sec)$
$\beta e =\ effective\ bulk\ modulus\ (incl. fluid\ and\ mechanical\ compliance), (lb/in^2)$
$\rho\ =\ mass\ density\ of\ fluid, (lb - sec^2/in^4)$
$Values\ of\ c\ in\ the\ range\ 35{,}000\ to\ 50{,}000\ in./sec\ are\ common.$

When the pressure wave arrives at the source end of the conduit (in L/c seconds where L is the conduit length), then the kinetic energy of the moving mass of fluid has been completely stored as potential energy in the elasticity of conduit and fluid and the pressure of the compressed fluid, P_{IC}, is a maximum. At this time a decompression wave forms and travels back to the valve. These waves continue to travel back and forth with the associated interchanges of kinetic and potential energies until friction expends the energy involved.

At the instant of valve closure the kinetic energy of the moving fluid is

$$KE = \frac{1}{2}M_f u_0^2 = \frac{1}{2}\rho L A u_0^2 \tag{3.79}$$

where A is the conduit area and u_0 is the initial velocity of the fluid.

The potential energy stored in the compressed fluid is

$$PE = \frac{1}{2}\frac{LA}{\beta_e}P_{IC}^2 \tag{3.80}$$

where P_{IC} is the pressure rise due to the instant valve closure.

Equating these energies yields an expression for pressure rise.

$$P_{IC} = \rho c u_0 \tag{3.81}$$

It is apparent from (3.81) that the most effective and only way, since ρ and c are fixed, to reduce this pressure surge is to design pipe systems to have low original fluid velocities by keeping pipe areas large. If fluid velocities are limited to a maximum of $15\ ft/sec$, then the instant closure pressure rise (above the steady state level) is about $750\ psi$, which is generally considered a safe design value and is a criterion for conduit selection. It is interesting to note that P_{IC} is independent of line length.
Equation 3.81 is valid and the closure is considered instantaneous if the valve closure time T is less than that required for one round trip of the pressure wave, that is,

$$T \leq T_c = \frac{2L}{c} \tag{3.82}$$

Where Tc is commonly called the critical closure time. For short lines this inequality is generally not satisfied. In this event the pressure rise will depend on line length, steady state pressure level, and valve closure time in addition to P_{IC}. The mathematical equations which describe this situation are unwieldy but solutions can be made and expressed in graphical form (Figure 3-24).
(Manring, 2005)

3.7.2 **Quick's Chart**

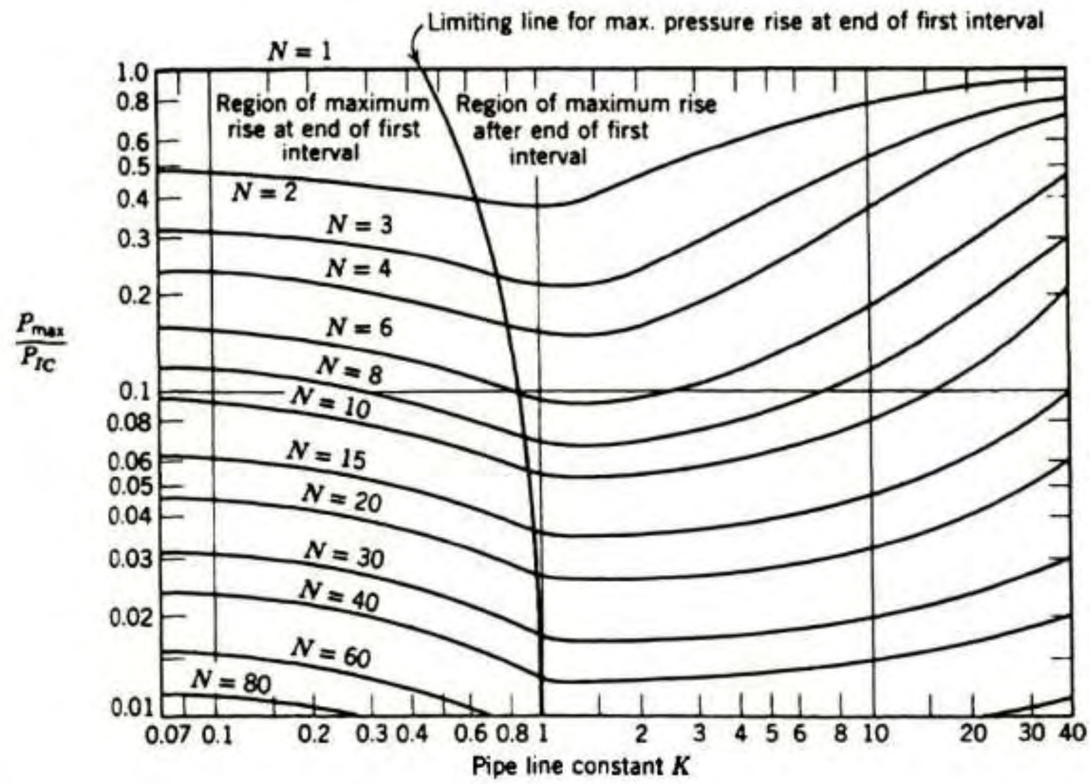


Figure 3-24 - Quick's Chart showing maximum pressure rise with uniform valve closure. (Herbert Merritt - Hydraulic control systems)

Quick's chart assumes uniform valve closure and is the most convenient graphical technique (Figure 3-24).

To use Quick's chart first compute the quantities K and N in which

$K = \dfrac{P_{IC}}{2P_0} = pipe\ line\ constant, dimensionless$
$N = T/T_C = number\ of\ critical\ closure\ time\ intervals, dimensionless$
$P_0 = steady\ state\ pressure\ in\ conduit, (psi)$
$T_C = critical\ closure\ time, (sec)$
$T\ =\ time\ required\ for\ uniform\ valve\ closure, (sec)$
$P_{max}\ =\ pressure\ rise\ in\ conduit\ above\ P_0\ due\ to\ uniform\ valve\ closure$
$(for\ instant\ closure, P_{max}\ =\ P_{IC}), (Psi)$

Now, from the chart obtain the value of P_{max}/P_{IC} from which, since P_{IC} is computed from (3.81), P_{max} is computed. Note that P_{max} will always be less than P_{IC}

In summary, the most severe pressure surge in a single pipe is that caused by an instantaneous valve closure (i.e., $T \leq T_C$ and is given by (3.81). If the valve closure is not instantaneous (i.e., $T > T_C$ then uniform valve closure should be assumed and P_{max} obtained from Quick's chart. This latter case is typical for short lines (less than $10\ ft$ if $T > 0.005\ sec$). However, because P_{IC} is the maximum surge pressure, instant valve closure is usually assumed for design purposes.
(Merritt, 1967)

3.8 Summary

At the outset it was determined that seven equations are required to define a situation involving fluids. These seven equations reduce considerably if the fluid is a liquid. The first three equations, the Navier- Stokes equations, are reduced to the application of certain formulas which were discussed in Sections 3.4, 3.5 and 3.6.

As a general rule, only those equations that describe intentionally inserted hydraulic resistances are used in a dynamic analysis because these are usually the dominant restrictors. Resistances of flow passages such as pipes, bends, and fittings, are often neglected. Therefore, the formulas most often used are the orifice equation, (3.55) and those given in Figure 3-11.

Because cubical expansion coefficients are small for liquids, the direct effect of temperature on fluid density and, consequently, on fluid flow is often negligible. This is not to say that thermal gradients never exist. It is simply that these gradients have little influence on flow conditions. Therefore, it is usually sufficient to include temperature by evaluating fluid properties at the operating temperature.

It is generally assumed that isothermal conditions exist in liquid flow. The assumption of constant temperature eliminates the need for the energy equation and reduces the equation of state to the simple form

$$\rho = \rho_i + \frac{\rho_i}{\beta} P \tag{3.83}$$

where ρ_i and β are the mass density and bulk modulus at zero pressure. The continuity equation (3.15) can be written

$$\sum W_{in} - \sum W_{out} = g \cdot \frac{d(\rho V_0)}{dt} = g\rho \cdot \frac{d(V_0)}{dt} + gV \cdot \frac{d(\rho)}{dt} \tag{3.84}$$

Noting that weight flow rate can be written $W = g\rho Q$, we can combine (3.83) and (3.84) to yield

$$\sum Q_{in} - \sum Q_{out} = \frac{dV_0}{dt} + \frac{V_0}{\beta}\frac{dP}{dt} \tag{3.85}$$

Thus, the continuity equation and the equation of state are combined into the more useful form given by (3.85). The first term on the right side is the flow consumed by expansion of the control volume; if the volume is fixed, this term is zero. The second term is the compressibility flow and describes the flow resulting from pressure changes.

Need for the seventh equation is eliminated by assuming that viscosity is constant. Therefore, all seven of the initial equations have been accounted for. In pneumatic systems the temperature may vary and a slightly different reduction of the initial equations is required.

4. Hydraulic Circuit Analysis

Hydraulic circuits are composed of pumps, pressure control valves, directional valves, flow control valves, actuators and accessories, and change their compositions according to objectives and specifications. Many compositions are possible for a single objective. The most efficient circuit is (Merkle, M.Thomes, & B.Schrader, 2003):

i. Safe and completely optimal to meet objective.
ii. Capable of smooth movement.
iii. Energy efficient.
iv. Effective for initial and running costs.
v. Easy to maintain.

4.1 Graphic and Circuit Symbols

Simple graphic and circuit symbols are used for individual components to enable clear representation of hydraulic systems in diagrams. A symbol identifies a component and its function, but it does not provide any information about its design. The symbols to be used are laid down in DIN ISO 1219. The most important symbols are dealt with in this chapter (Merkle, M.Thomes, & B.Schrader, 2003).

An arrow drawn at an angle through the symbol indicates that setting possibilities exist.

4.1.1 Pumps and Motors

Hydraulic pumps and motors are represented by means of a circle which shows where the drive or output shaft is located. Triangles within the circle give information about the direction of flow. These triangles are filled in, since hydraulic fluids are used for hydraulics. If a gaseous pressure medium were being used, as is the case in pneumatics, the triangles would not be filled in. The symbols for hydraulic motors and hydraulic pumps can only be distinguished from one another by the fact that the arrows indicating the direction of flow are drawn pointing one way for the pumps and the other for the motors.

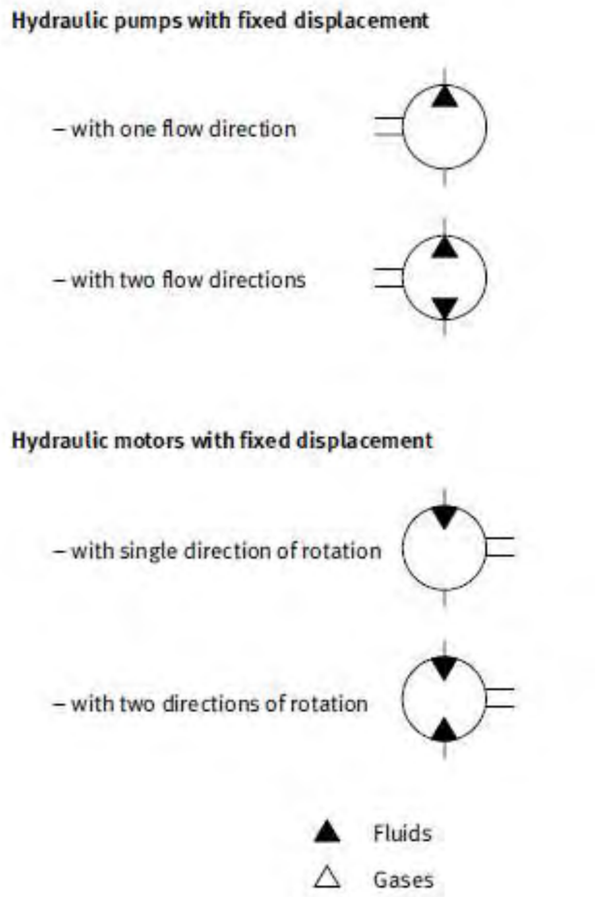


Figure 4-1 - Schematics of Hydraulic Pumps and Motors with fixed displacement.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.1.2 **Directional Control Valves**

Directional control valves are shown by means of several connected squares.

• The number of squares indicates the number of switching positions possible for a valve.
• Arrows within the squares indicate the flow direction.
• Lines indicate how the ports are interconnected in the various switching positions.

There are two possible methods of port designation. One method is to use the letters P, T, A, B and L, the other is to label ports alphabetically A, B, C, D, etc. The former method is generally preferred. Ports should always be labelled with the valve in the rest position. Where there is no rest position, they are allocated to the switching position assumed by the valve when the system is in its initial position.

The rest position is defined as the position automatically assumed by the valve on removal of the actuating force.

When labelling directional control valves, it is first necessary to specify the number of ports followed by the number of switching positions. Directional control valves have at least two switching positions and at least two ports. In such an instance, the valve would be designated a 2/2-way valve. The following diagrams show other directional control valves and their circuit symbols.

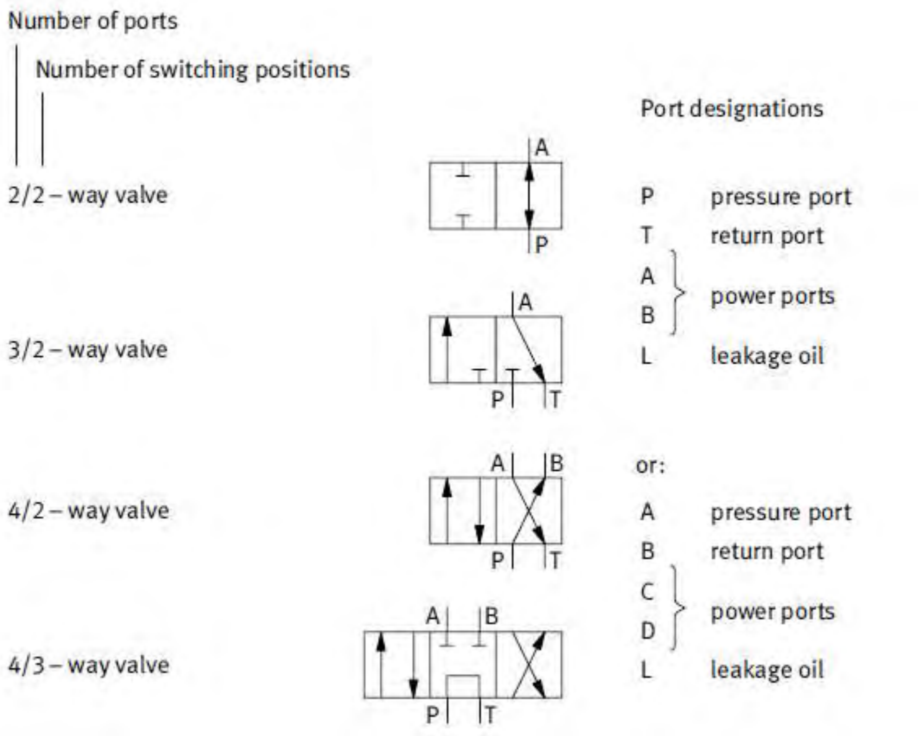


Figure 4-2 - Schematics of Directional Control Valves.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.1.3 **Types of Actuation**

The switching position of a directional control valve can be changed by various actuation methods. The symbol for the valve is elaborated by the addition of the symbol indicating the actuation method. In the case of some of the actuation methods shown, such as push button, pedal, lever with detent, a spring is always necessary for resetting. Resetting may also be achieved by switching the valve a second time, e.g. in the case of a valve with hand lever and detent setting.

Listed below are the symbols for the most important actuation methods. Refer to DIN ISO 1219 for other methods of actuation.

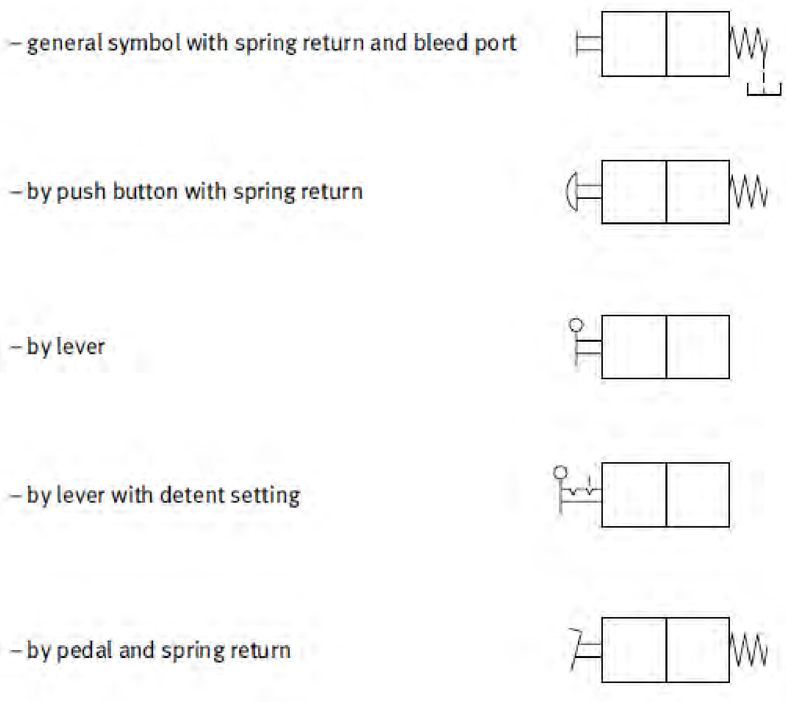


Figure 4-3 - Schematics of different types of Mechanical Actuations. (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.1.4 **Pressure Valves**

Pressure valves are represented using squares. The flow direction is indicated by an arrow. The valve ports can be labelled as P (pressure port) and T (tank connection) or A and B.

The position of the valve within the square indicates whether the valve is normally open or normally closed.

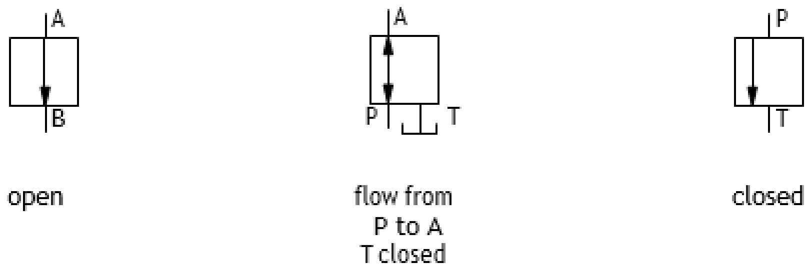


Figure 4-4 - Schematics of open and closed Valve Position. (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

A further distinction is made between set and adjustable pressure valves. The latter are indicated by a diagonal arrow through the spring.

Figure 4-5 - Schematic of set and adjustable Pressure Valves.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

Pressure valves are divided into pressure relief valves and pressure regulators:

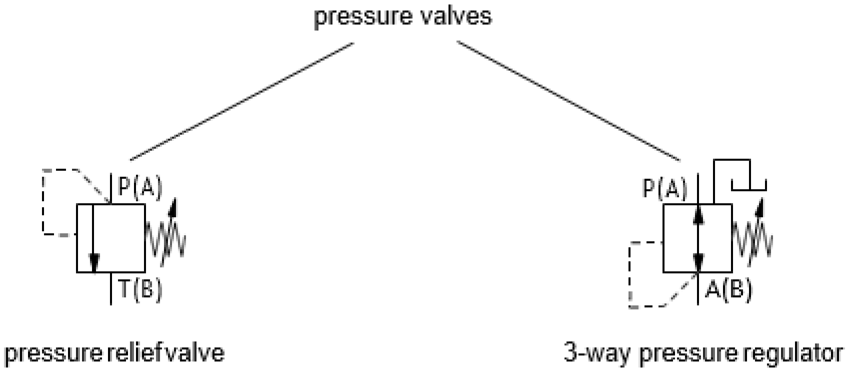


Figure 4-6 - Schematic of Pressure Relief Valve and Pressure Regulator Valve.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.1.5 Flow Control Valves

In the case of flow control valves, a distinction is made between those affected by viscosity and those unaffected. Flow control valves unaffected by viscosity are termed orifices. Throttles constitute resistances in a hydraulic system. These valves are depicted as a rectangle into which are drawn the symbol for the variable throttle and an arrow to represent the pressure balance. The diagonal arrow running through the rectangle indicates that the valve is adjustable. There is a special symbol to represent the 2-way flow control valve.

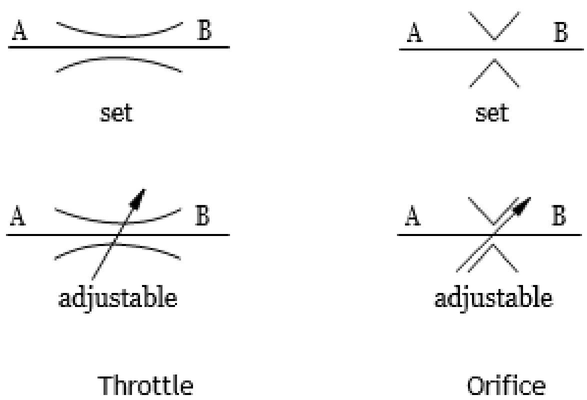


Figure 4-7 - Throttle and Orifice schematics
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

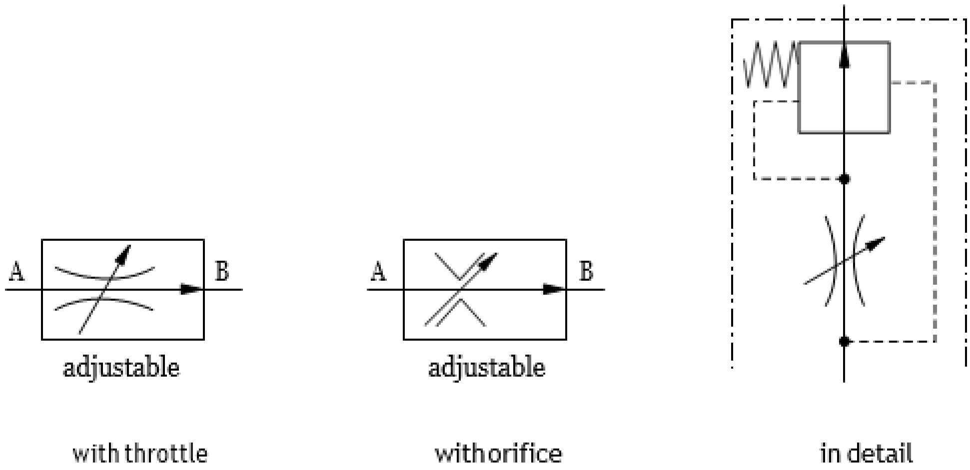


Figure 4-8 - Two way Flow Control Valve schematics
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.1.6 **Non-Return Valves**

The symbol for non-return valves is a ball which is pressed against a sealing seat. This seat is drawn as an open triangle in which the ball rests. The point of the triangle indicates the blocked direction and not the flow direction. Pilot controlled non-return valves are shown as a square into which the symbol for the non-return valve is drawn. The pilot control for the valve is indicated by a control connection shown in the form of a broken line. The pilot port is labelled with the letter X.

Shut-off valves are shown in circuit diagrams as two triangles facing one another. They are used to depressurise the systems manually or to relieve accumulators. In principle, wherever lines have to be opened or closed manually.

Figure 4-9 - Non-Return Valves schematics
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

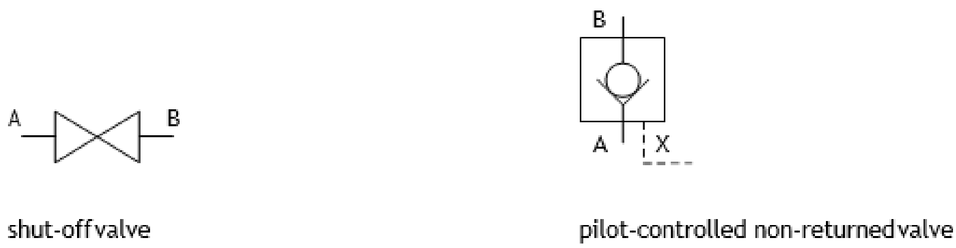


Figure 4-10 - Non-Return Valves
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.1.7 **Cylinders**

Cylinders are classified as either single-acting or double-acting.

- ***Single Acting Cylinder***

Single acting cylinders just have one port, i.e. only the full piston surface can be pressurised with hydraulic fluid. These cylinders are returned either by the effect of external forces – indicated by the symbol with the open bearing cap – or by a spring. The spring is then also drawn into the symbol.

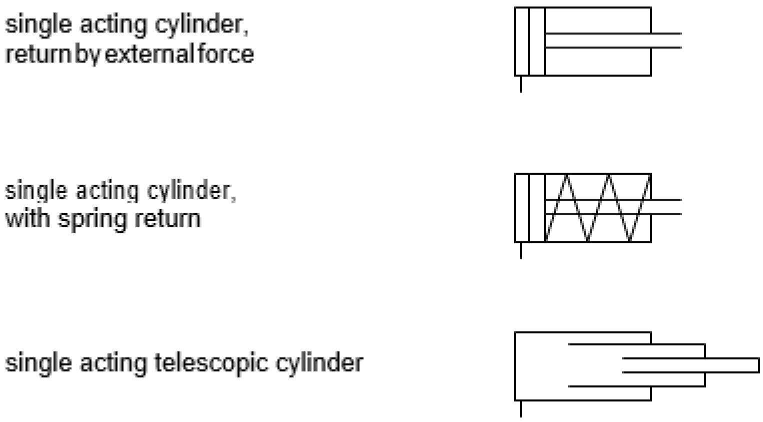


Figure 4-11 - Single Acting Cylinders schematics
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

- ***Double Acting Cylinder***

Double acting cylinders have two ports for supplying either side of the piston with hydraulic fluid. It can be seen from the symbol for a double acting cylinder with single piston rod that the piston area is greater than the annular piston surface.

Conversely, the symbol for the cylinder with a through piston rod shows that these areas are of the same size (synchronous cylinder). The symbol for the differential cylinder can be distinguished from that for the double-acting cylinder by the two lines added to the end of the piston rod. The area ratio is 2:1. Like single-acting telescopic cylinders, double-acting ones are symbolized by pistons located one inside the other. In the case of the double-acting cylinder with end position cushioning, the cushioning piston is indicated in the symbol by a rectangle.

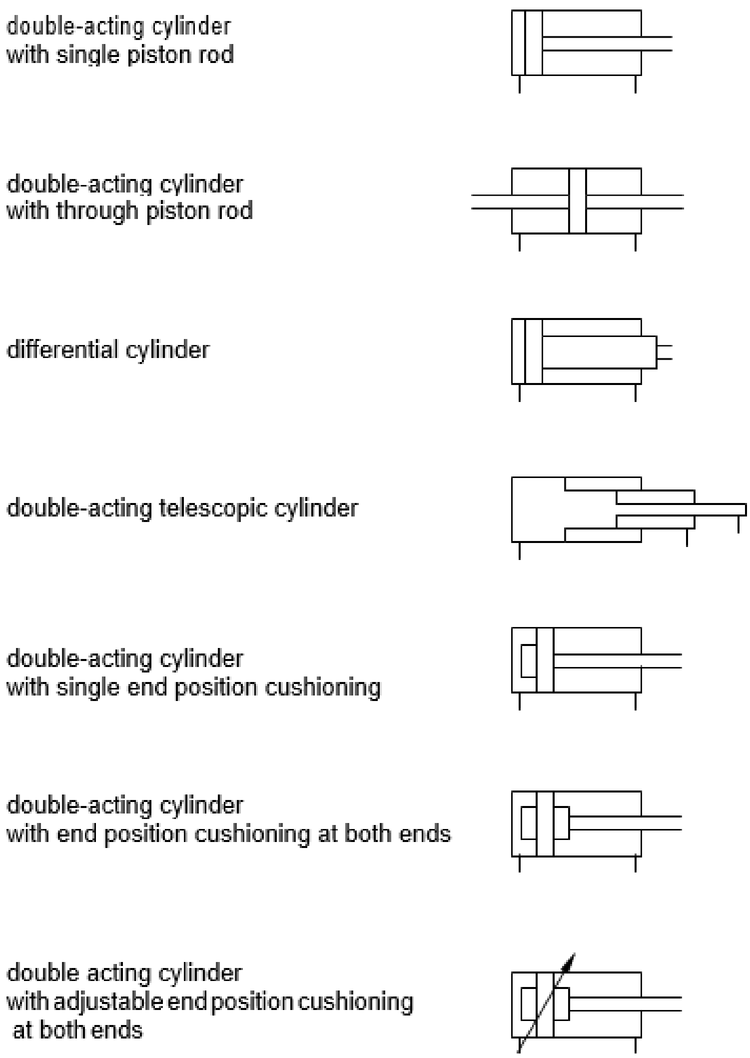


Figure 4-12 - Double Acting Cylinders schematics (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.1.8 Transfer of Energy and Conditioning of the Pressure Medium

The following symbols are used in circuit diagrams for energy transfer and conditioning of the pressure medium.

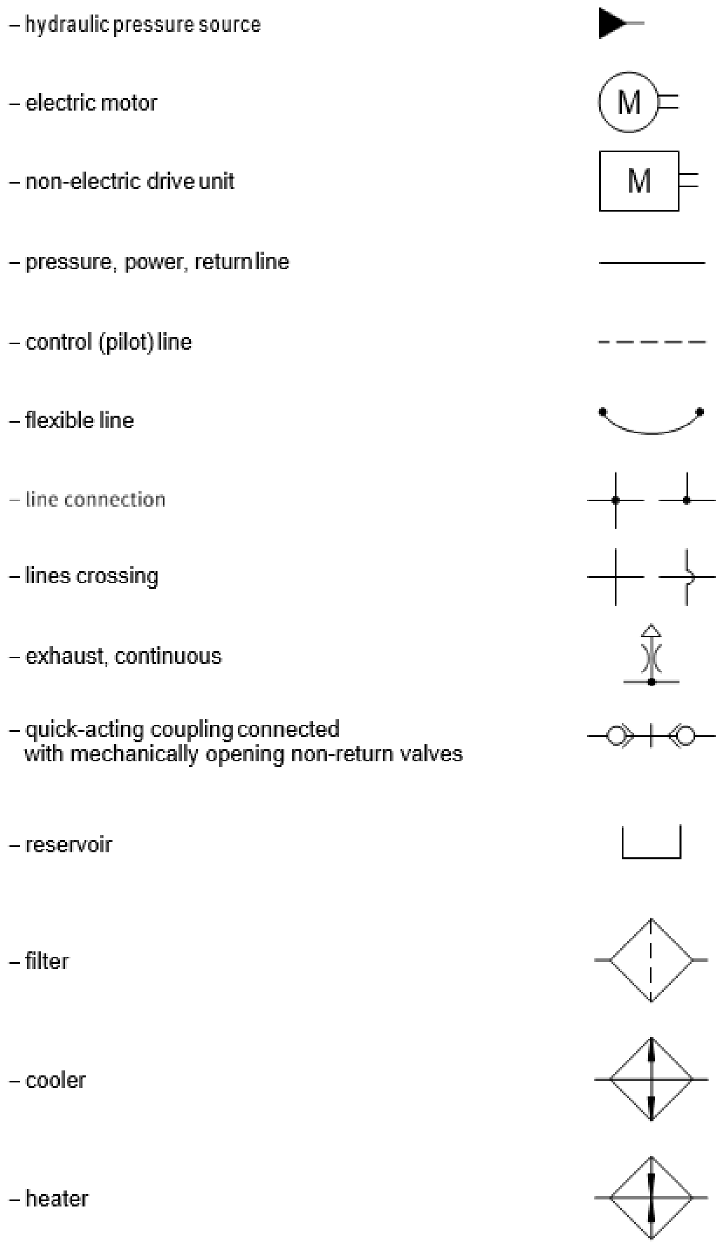


Figure 4-13 - Symbols for Energy Transferring and Conditioning of the Pressure medium.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

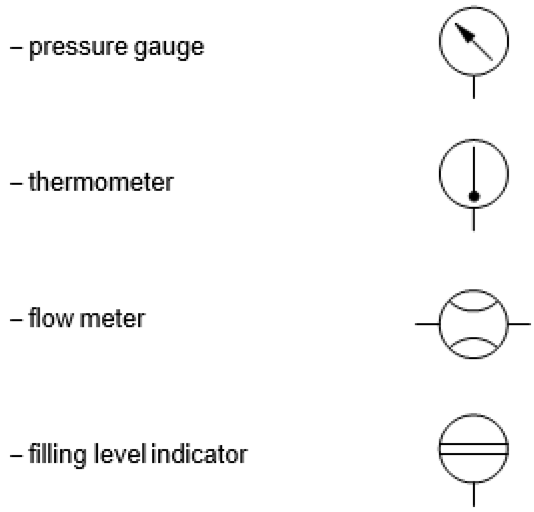


Figure 4-14 - Measuring devices schematics (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

If several devices are brought together in a single housing, the symbols for the individual devices are placed into a box made up of broken lines from which the connections are led away.

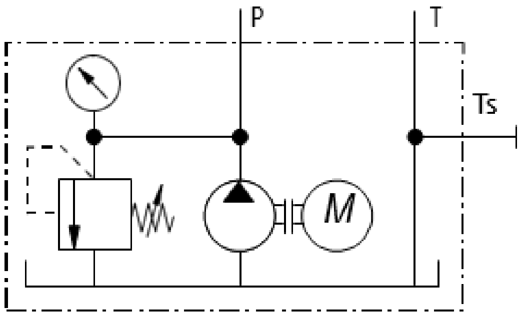


Figure 4-16 - Hydraulic Power Pack (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

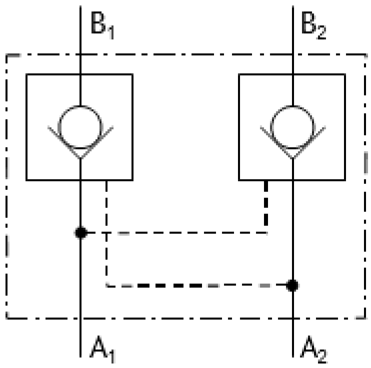


Figure 4-15 - Pilot Operated Double Non-Return Valve (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.2 Design and Representation of a Hydraulic System

A hydraulic system can be divided into the following sections (Merkle, M.Thomes, & B.Schrader, 2003) (Festo):

- ***The signal control section***
- ***The power section***

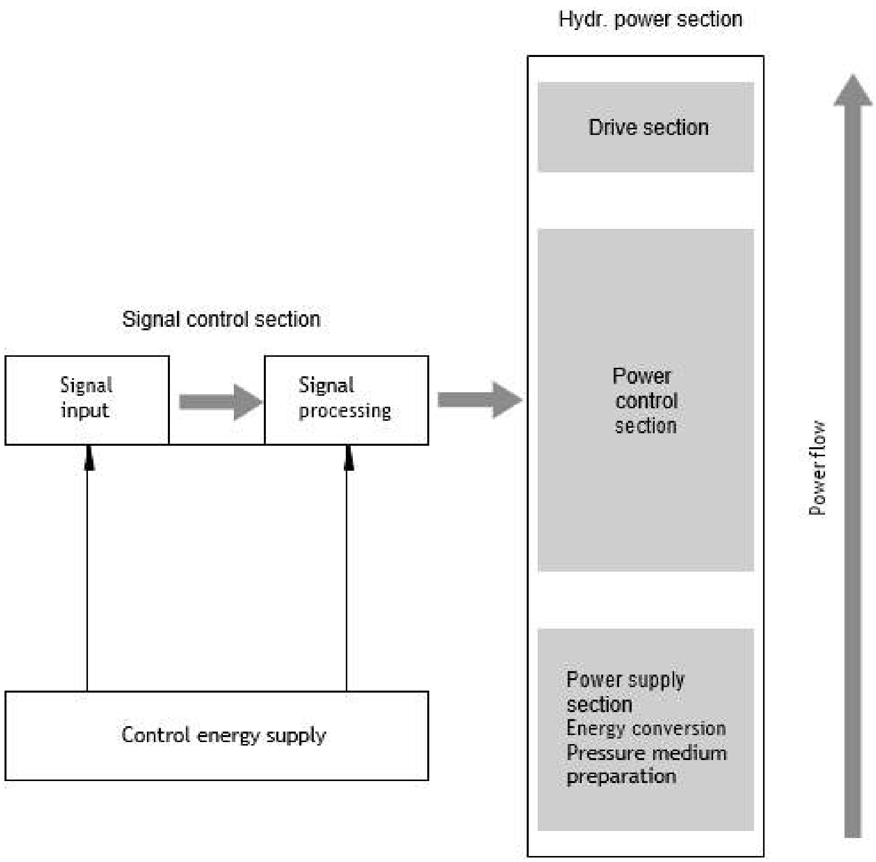


Figure 4-17 - Structure of a Hydraulic System
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.2.1 **Signal Control Section**

The signal control section is divided into signal input (sensing) and signal processing (processing).

Signal input may be carried out:

- Manually
- Mechanically
- Contactlessly

Signals can be processed by the following means:

- by the operator
- by electricity
- by electronics
- by pneumatics
- by mechanics
- by hydraulics

4.2.2 **Hydraulic Power Section**

The hydraulic power can be divided up into the power supply section, the power control section and the drive section (working section). The power supply section is made up of the energy conversion part and the pressure medium conditioning part. In this part of the hydraulic system, the hydraulic power is generated and the pressure medium conditioned.

The following components are used for energy conversion – converting electrical energy into mechanical and then into hydraulic energy:

- Electric motor
- Internal combustion engine
- Coupling
- Pump
- Pressure indicator
- Protective circuitry

The following components are used to condition the hydraulic fluid:

- Filter
- Cooler
- Heater
- Thermometer
- Pressure gauge
- Hydraulic fluid
- Reservoir
- Filling level indicator

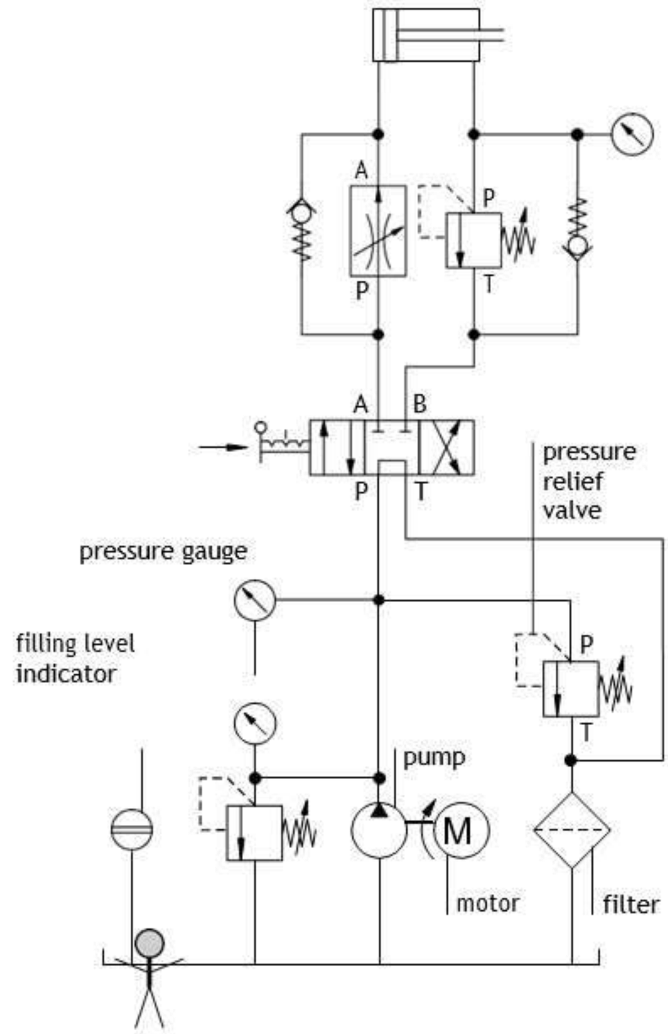


Figure 4-18 - Simple Hydraulic Circuit.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

The power is supplied to the drive section by the power control section in accordance with the control problem. The following components perform this task:

- directional control valves
- flow control valves
- pressure valves
- non-return valves.

The drive section of a hydraulic system is the part of the system which executes the various working movements of a machine or manufacturing system. The energy contained in the hydraulic fluid is used for the execution of movements or the generation of forces (e. g. clamping processes). This is achieved using the following components:

- cylinders
- motors

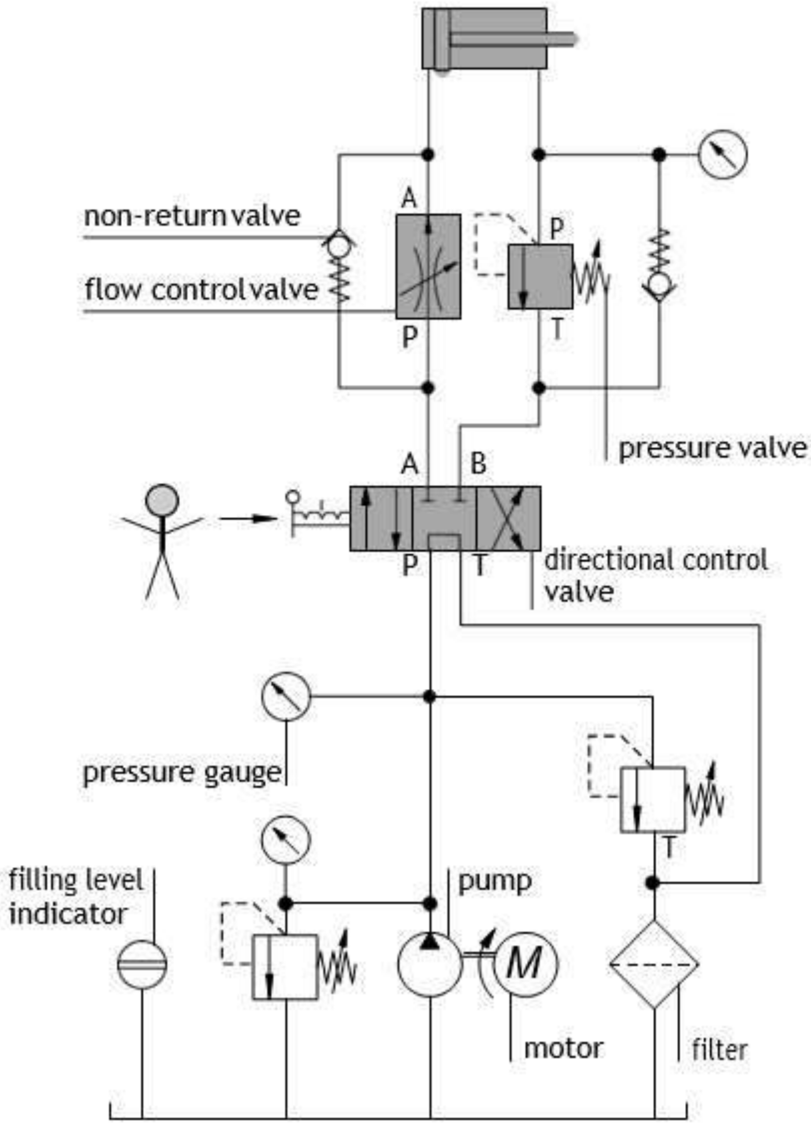


Figure 4-19 - Simple Hydraulic System. The drive section is highlighted. (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

4.2.3 Types of Hydraulic Circuits Representations

A suitable type of representation is required in order to reproduce movement sequences and operating statuses of working elements and control elements clearly.

The following types of representation are of importance (Merkle, M.Thomes, & B.Schrader, 2003):

- ***Positional Sketch***

The positional sketch is a drawing or schematic diagram of a production installation or machine etc. It should be easily understandable and should include only the most important information. It shows the spatial arrangement of the components.

The positional sketch in the Figure shows the position of cylinder Z1 and its function: Z1 is intended to lift the hood of the tempering furnace.

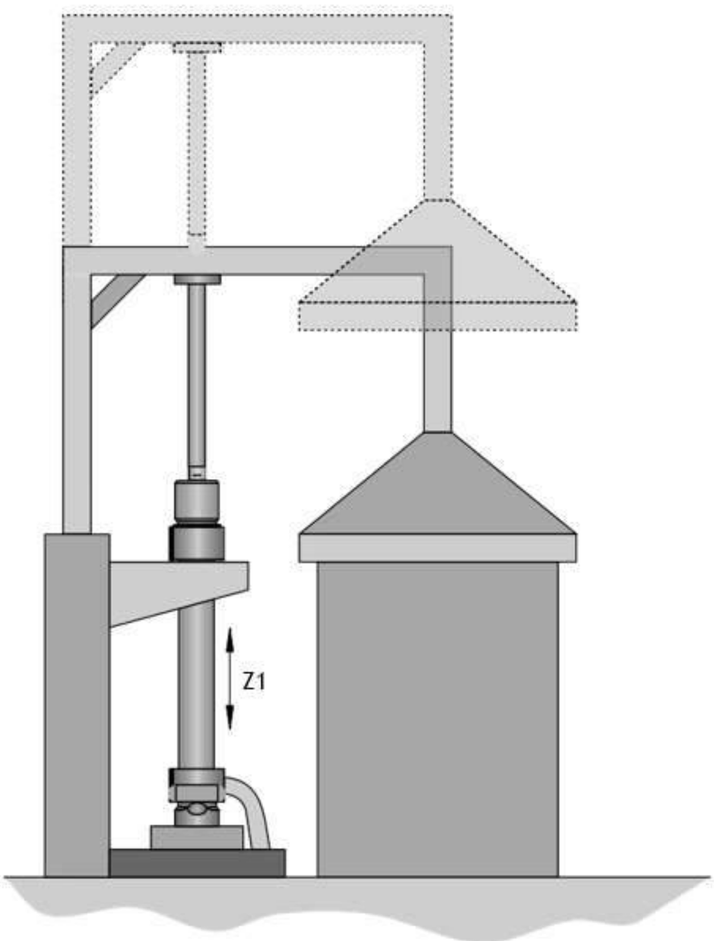


Figure 4-20 - Positional sketch example
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

- ***Circuit Diagram***

The circuit diagram describes the functional structure of the hydraulic system.

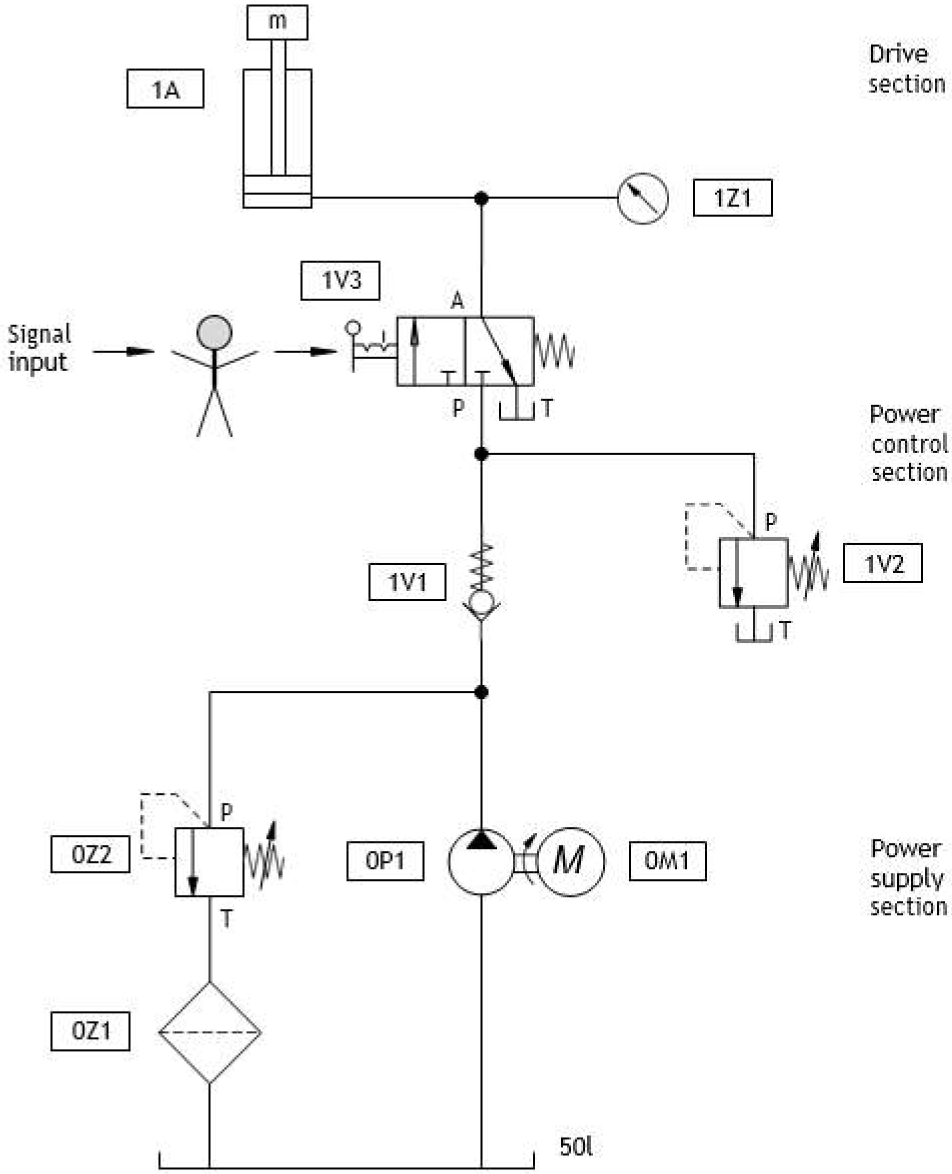


Figure 4-21 - Circuit diagram with labels next to each component.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

The power supply section of the system with filter (0Z1), pressure-relief valve (0Z2), pump (0P1) and electric motor (0M1) is depicted in the lower part of the circuit diagram shown for the hydraulic device of the tempering furnace.

The power control section with the non-return valve (1V1), the 3/2-way valve (1V3) and the pressure-relief valve (1V2) is located at the centre of the circuit diagram. The 3/2-way valve (1V3) with the hand lever for signal input forms the “system-person” interface.

Like the drive section, the power control section is assigned to the power section. In this hydraulic device, the drive section consists of the single-acting cylinder 1A. In the circuit diagram, the technical data are often additionally specified with the devices in accordance with DIN 24347.

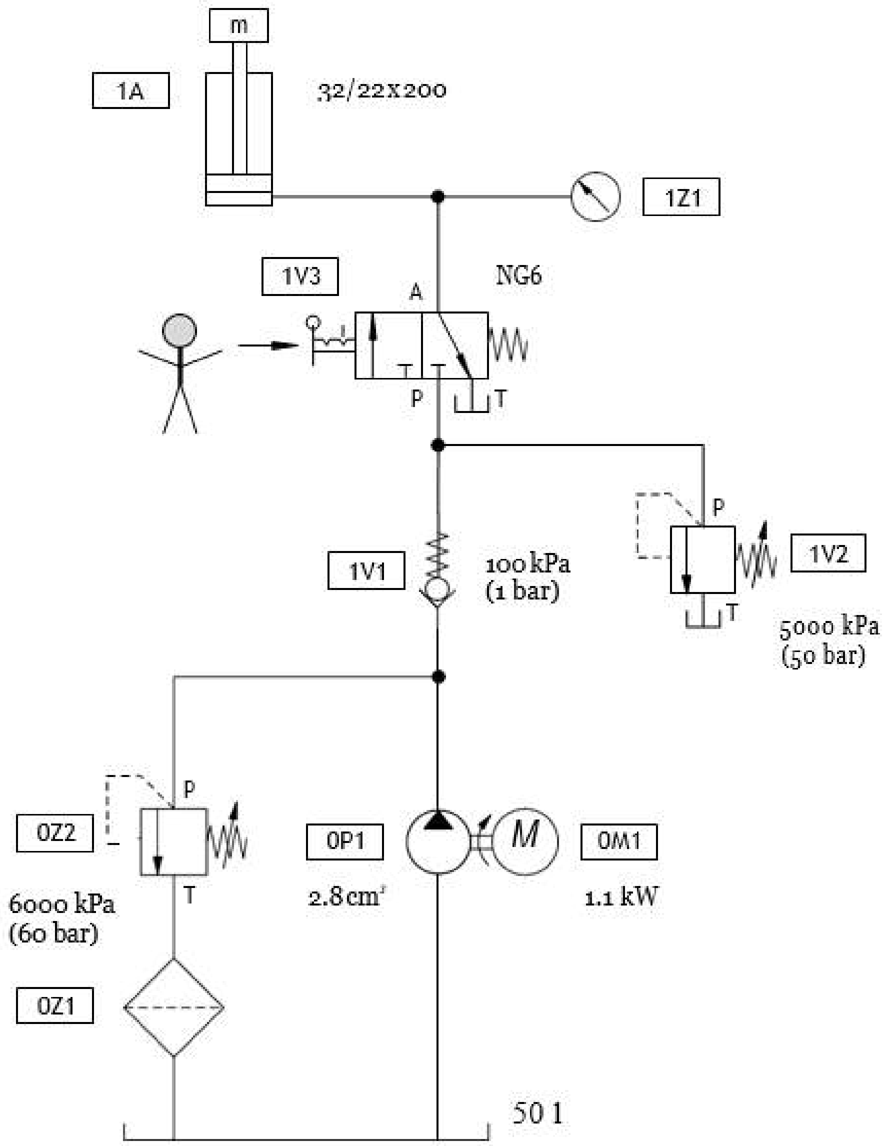


Figure 4-22 - Circuit diagram with labels and parameters mentioned next to each component. (D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

Or in a table form:

Equipment	Specifications	Example values
Reservoirs	Volume in litres to the highest permissible oil level	Max. 50 l
	Type of hydraulic fluid	ISO VG 22 type Al or HLP
Electric motors	Rated capacity in kW	1.1 kW
	Rated speed in rpm	1420 rpm
Fixed displacement pumps and variable-displacement pumps	Geometric delivery rate in cm^3	Gear pump 2.8 cm^3/revolution
Pressure valves	Set pressure in bar or permissible pressure range for the system	Operating pressure 50 bar
Non-return valve	Opening pressure	1 bar
Cylinder	Cylinder inner diameter/piston rod diameter stroke in mm. The function (e. g. clamping, lifting, flat turning etc.) must be entered above every cylinder	32/22·200 1A lifting
Filter	Nominal flow rate in l/min ß...at Δp...bar	
Flexible hose	Nominal diameter (inner diameter) in mm	6 mm
Hydraulic motor	Capacity in cm^3 Speed in rpm	v = 12.9 cm^3 n = 1162.8 rpm at Q = 15 cm^3/min M = 1 Nm
Directional control valve	Nominal size	NG 6

Figure 4-23 - Circuit diagram in Table form.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

- ***Function Diagram***

Function diagrams of working machines and production installations can be represented graphically in the form of diagrams. These diagrams are called function diagrams. They represent statuses and changes in status of individual components of a working machine or production installation in an easily understood and clear manner.

The following example shows a lifting device controlled by electromagnetic directional control valves.

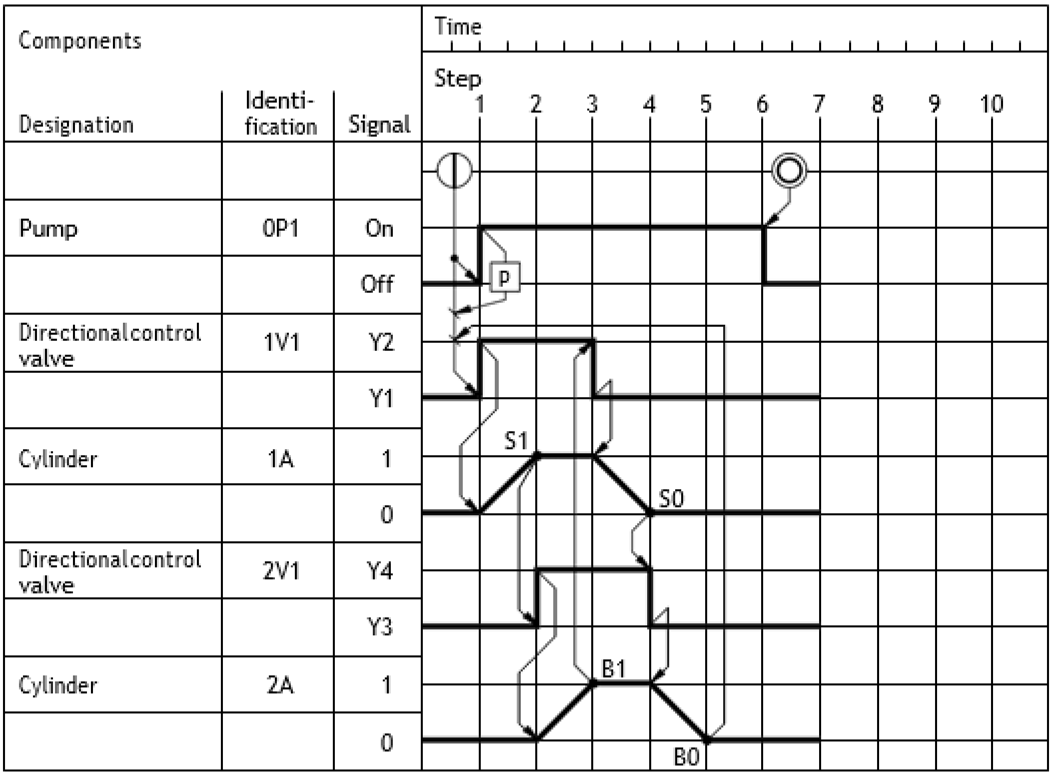


Figure 4-24 - Function diagram example.
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

- ***Function Chart.***

A function chart is a flow chart in which the control sequence is strictly divided into steps. Each step is executed only after the previous step has been completed and all step enabling conditions have been fulfilled.

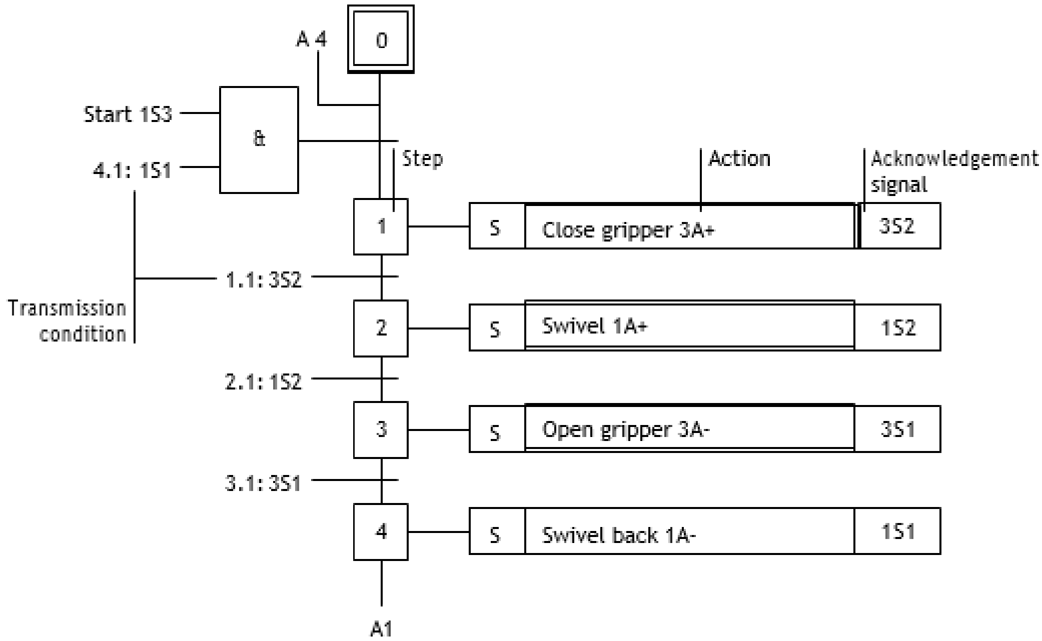


Figure 4-25 - Function chart example
(D.Merkle, M.Thomes and B.Schrader. Hydraulics Basic Level Textbook)

- ***Displacement-step and time diagram***

The displacement-step diagram and the displacement-time diagram are used for motion sequences (Festo). The displacement-step diagram represents the operating sequence of the actuators; the displacement is recorded in relation to the sequence step.

If a control system incorporates a number of actuators, they are shown in the same way and are drawn one below the other. Their interrelation can be seen by comparing the steps.

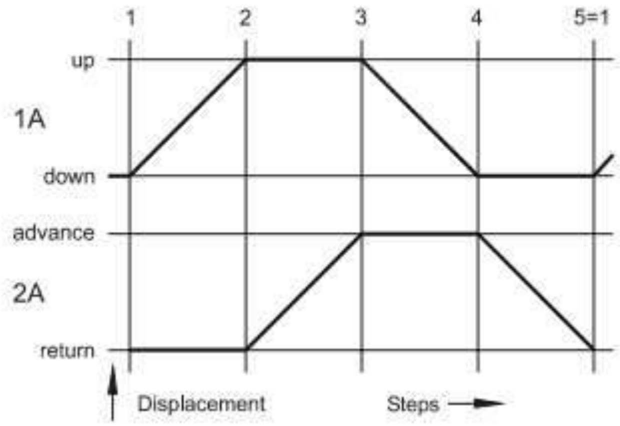


Figure 4-26 - Displacement step diagram. (Festo Didactic)

In this case there are two cylinders 1A and 2A. In step 1, cylinder 1A extends and then cylinder 2A extends in step 2. In step 3, cylinder 1A retracts and in step 4, cylinder 2A retracts. Step number 5 is equivalent to step 1.

In the case of a displacement-time diagram, the displacement is plotted in relation to the time.

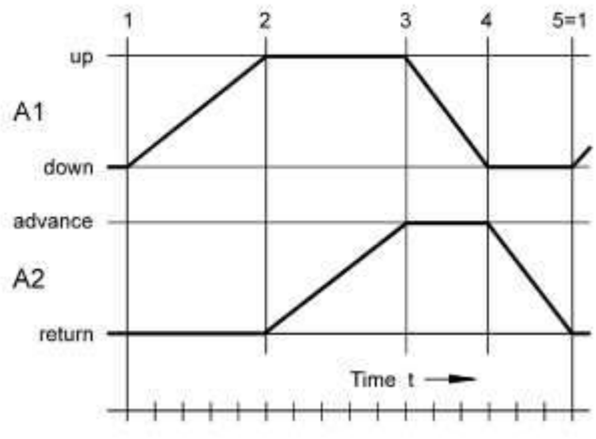


Figure 4-27 - Displacement time diagram (Festo Didactic)

4.3 Types of Hydraulic Systems

Depending on the objective, a proper hydraulic system can be assembled in different ways through different combinations of hydraulic components. For this reason, some categories of hydraulic systems have been developed for the most common groups of similar objectives and each one of which, has different specifications in order to apply best to each group.

The followings are general circuit types of hydraulic systems (Yuken Kogyo Co., 2006)(8 types, 29 examples).

4.3.1 **Unload Circuits**

These circuits (Yuken Kogyo Co., 2006) enhance the product life and efficiency, at the same time, keeping the power consumption and heat generation at a low level when the system is in waiting mode. Hydraulic power is expressed in the following equation.

$$L(kW) = \frac{PQ}{60} \tag{4.1}$$

$P\ :\ Pressure\ (MPa)$
$Q\ :\ Flow\ Rate\ (L/min)$

As shown, hydraulic power is proportional to the product of pressure and flow rate. Reducing either exponent then leads to lessened hydraulic power.

4.3.1.1 **Open Center Circuits**

An open center system (U.S. Army Material Command, 1971) is one having fluid flow, but no pressure in the system when the actuating mechanisms are idle. The pump circulates the fluid from the reservoir, through the selector valves, and back to the reservoir. The open center system may employ any number of subsystems, with a selector valve for each subsystem. Unlike the closed center system, the selector valves of the open center system are always connected in series with each other. In this arrangement, the system pressure line goes through each selector valve. Fluid is always allowed free passage through each selector valve and back to the reservoir until one of the selector valves is positioned to operate a mechanism. When one of the selector valves is positioned to operate an actuating device, fluid is directed from the pump through one of the working lines to the actuator. With the selector valve in this position, the flow of fluid through the valve to the reservoir is blocked. The pressure builds up in the system to overcome the resistance and moves the piston of the actuating cylinder; fluid from the opposite end of the actuator returns to the selector valve and flows back to the reservoir.

Operation of the system following actuation of the component depends on the type of selector valve being used. Several types of selector valves are used in conjunction with the open center system. One type is both manually engaged and manually disengaged. First, the valve is manually moved to an operating position. Then, the actuating mechanism reaches the end of its operating cycle, and the pump output continues until the system relief valve relieves the pressure. The relief valve unseats and allows the fluid to flow back to the reservoir. The system pressure remains at the relief valve set pressure until the selector valve is manually returned to the neutral position. This action reopens the open center flow and allows the system pressure to drop to line resistance pressure. The manually engaged and pressure disengaged type of selector valve is similar to the valve previously discussed. When the actuating mechanism reaches the end of its cycle, the pressure continues to rise to a predetermined pressure. The valve automatically returns to the neutral position and to open center flow.

Figure 4-29 shows a circuit in which output flow from the pump is by-passed to the reservoir with the spool of the PT connection valve at the center position. In the case of solenoid pilot operated directional valves, as shown in Figure 4-28 the check valve is required to maintain the minimum pilot pressure of the valve.

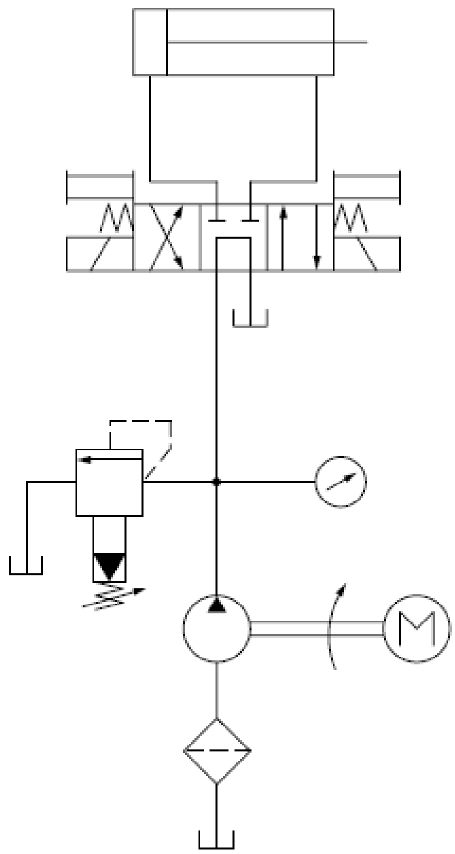


Figure 4-29 - Open center circuit where the output flow from the pump is bypassed to the reservoir through the Valve. (Yuken Kogyo, Basic Hydraulics and Components)

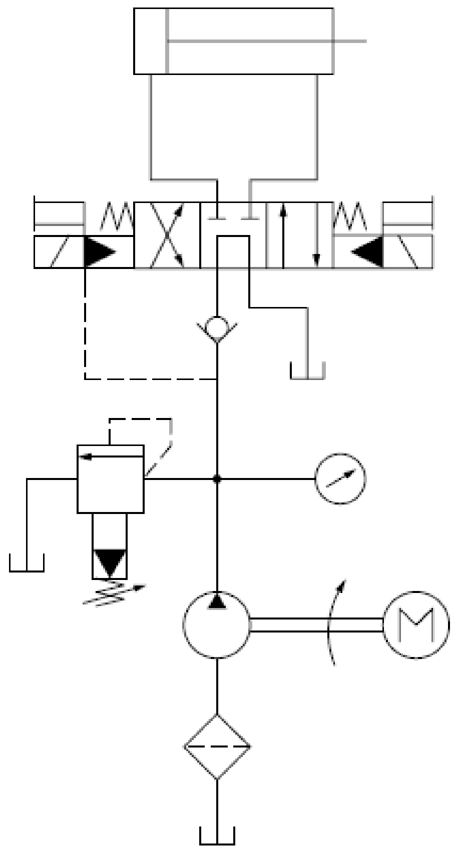


Figure 4-28 - Open center circuit where the need to maintain the pilot pressure of the valve requires the use of a Check Valve. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.1.2 **Circuit by Pressure Compensated Pump**

This circuit reduces the amount of output flow rate by using the variable displacement pump with pressure compensating function. The pressure of the system is raised to a set level even when it is in idle.

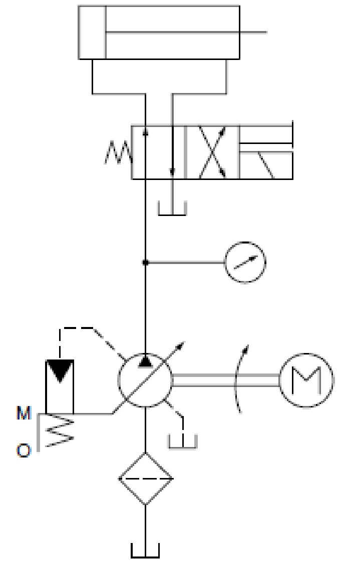


Figure 4-30 - Pressure Compensated Pump. Higher pressure leads to lower flow rate.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.1.3 **Circuit with Accumulator**

Pressure switch PS is correlated with the solenoid valve; when the circuit pressure equals the pressure set in the pressure switch PS, the solenoid valve is turned off, and output flow is by-passed to the reservoir, but the accumulator keeps the circuit pressure constant.

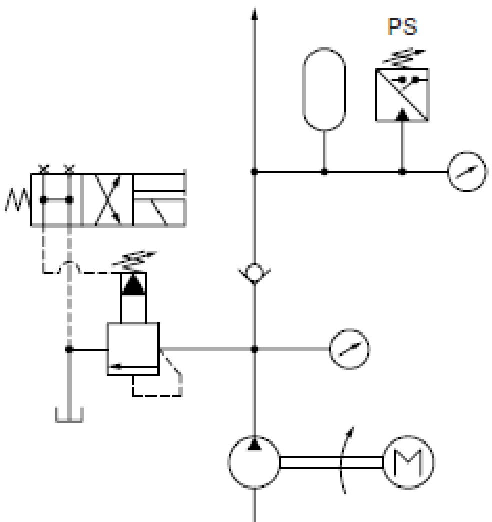


Figure 4-31 - Circuit with Accumulator (PS). If pressure exceeds a limit the flow is directed to the reservoir but the accumulator maintains the pressure to the rest of the system.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.1.4 **Pump-Relief Unload Circuit**

The circuit shown Figure 4-33 contains a low pressure and large volume pump (1) and a high pressure and small volume pump (2). The circuit raises its efficiency by using (1) and (2). In a case where the pressure in the circuit is lower than the pressure set at the unload valve, the output flow rate from (1 and (2) are gathered and provided to the circuit. In contrast, in a case where pressure in the circuit is higher than the pressure set at the unload valve, the output flow rate from (1) is by-passed to the reservoir by the unload valve, and the pressure is unloaded. In this case, pressure within the circuit is kept at a certain level by the output flow rate from (2) only. This circuit provides flow rate

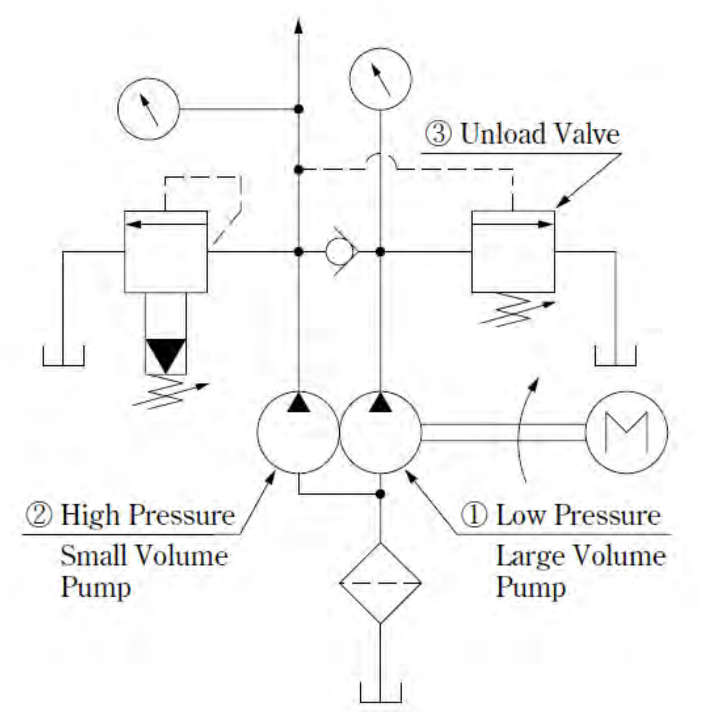


Figure 4-32 - Pump relief -unload valve circuit. The relief valve activates or deactivates the high pressure pump.
(Yuken Kogyo, Basic Hydraulics and Components)

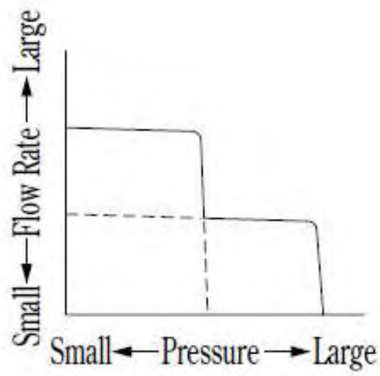


Figure 4-33 - Flow vs Pressure diagram of a circuit with one high pressure pump, one low pressure pump and a relief valve.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.1.5 **Circuit with Two-Pressure Two-Control Type**

When pump pressure is lower than PL, the angle of the swash plate becomes maximum, and the output flow reaches to the maximum (the flow rate goes to QH). In contrast, a pump pressure higher than PL causes a smaller swash plate angle and smaller output flow (QL). When the circuit pressure equals pump pressure, the swash plate angle falls close to zero, and output flow reduces to the amount of internal leakage. In this circuit, the power of the electric motor is kept small.

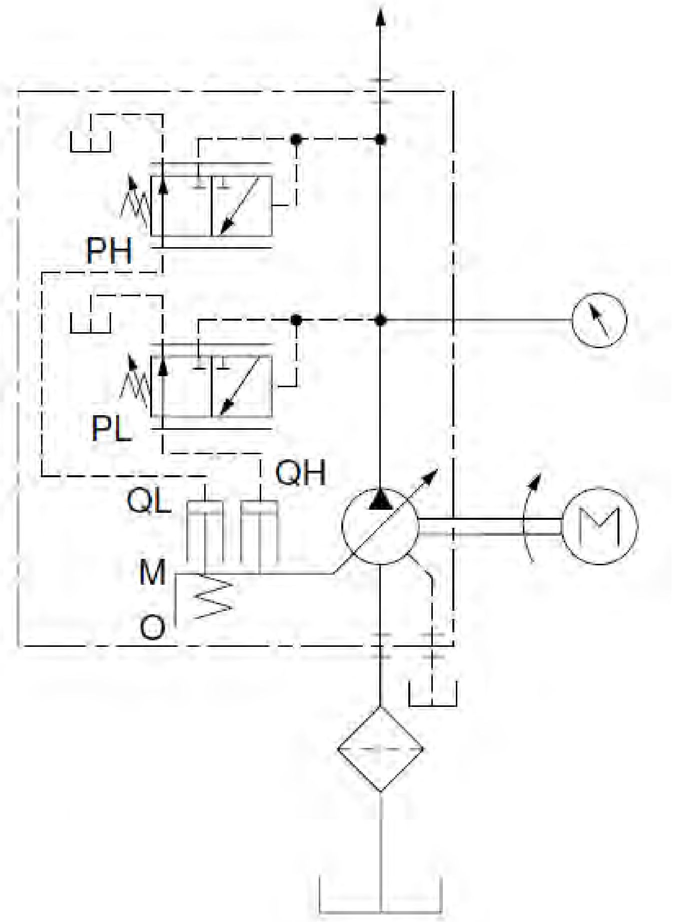
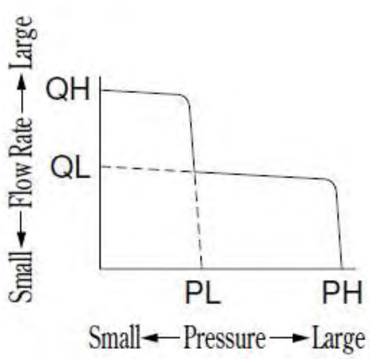


Figure 4-34 - Two pressure two control type circuit.
(Yuken Kogyo, Basic Hydraulics and Components)



4.3.2 **Pressure Control Circuits**

(Yuken Kogyo Co., 2006)

4.3.2.1 **Two-Pressure Circuit with Decompression and Check Valves**

In the cylinder-forwarding process (solenoid OFF), circuit pressure is kept at 10 MPa (1450 psi). But, in the cylinder- returning process (solenoid ON), circuit pressure is controlled at 7 MPa (1015 psi) by the decompression valve.

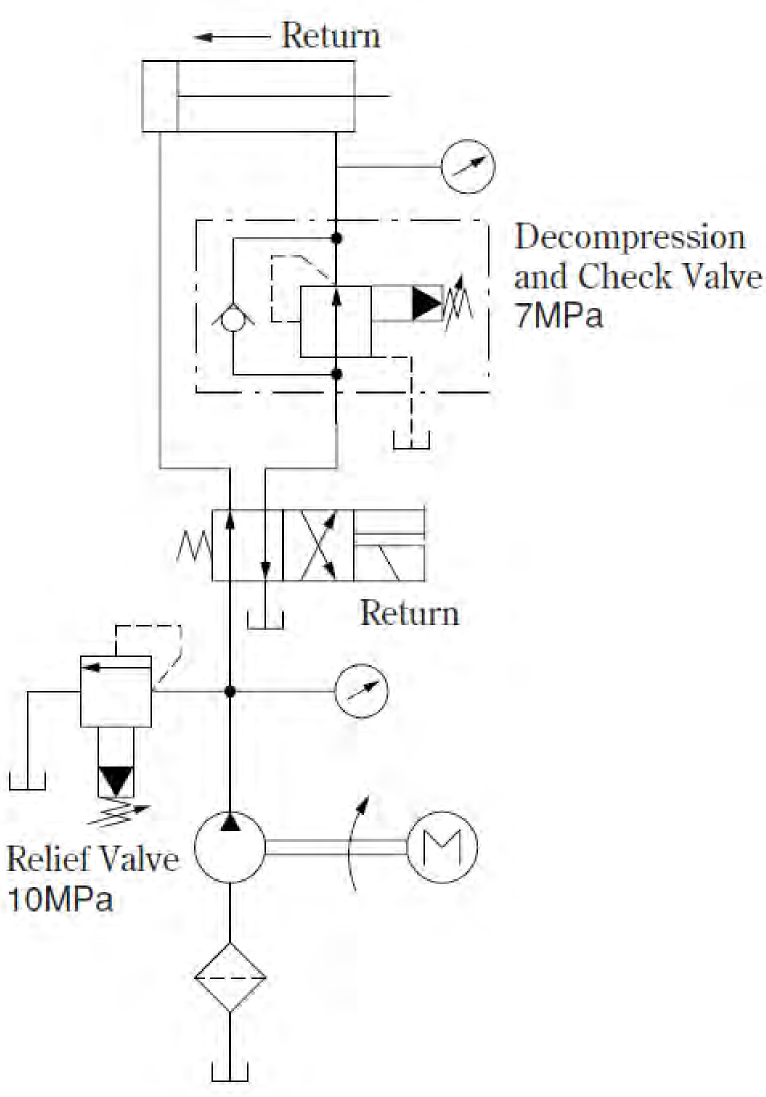


Figure 4-35 - Two-Pressure Circuit with Decompression and Check Valves. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.2.2 **Decompression Circuit I**

Releasing compressed working fluid instantly back to a reservoir generates shock waves. Compressed working fluid must be released gradually. In Figure 4-36, compressed working fluid in the cylinder cap is released gradually through the flow control valve. This reduces circuit pressure slowly, thus preventing shock waves.

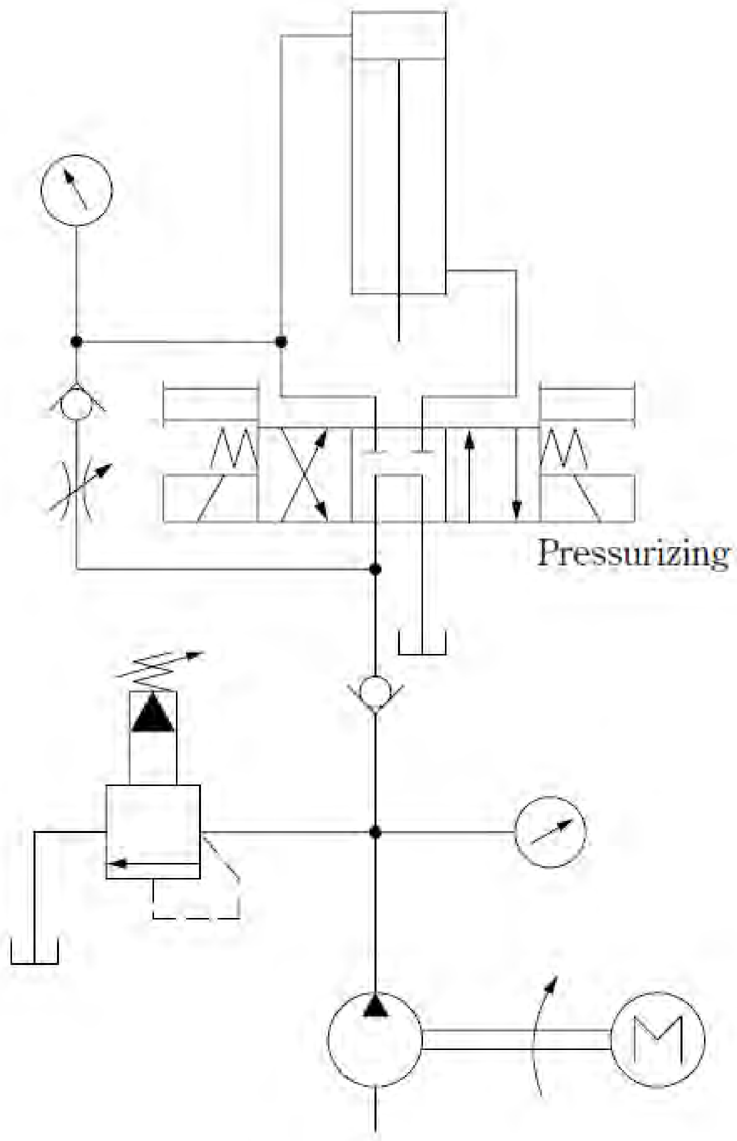


Figure 4-36 - Decompression circuit. The pressure drops gradually by passing through a flow control valve in order to avoid shockwaves from instant relief.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.2.3 Decompression Circuit II

After completion of pressure release, this circuit achieves automatic lift of the cylinder by electric signals from the pressure switch. After compressing working fluid in the cylinder cap, the solenoid of the decompression valve is turned ON. Then, the compressed working fluid is released back to the reservoir through the flow control valve. Pressure in the circuit drops until it hits a certain point set in the switch PS. Then, the switch transmits signals out, and turns the solenoid valve ON to lift the cylinder. By this sequence, a smooth lift-up-and-down motion is achieved.

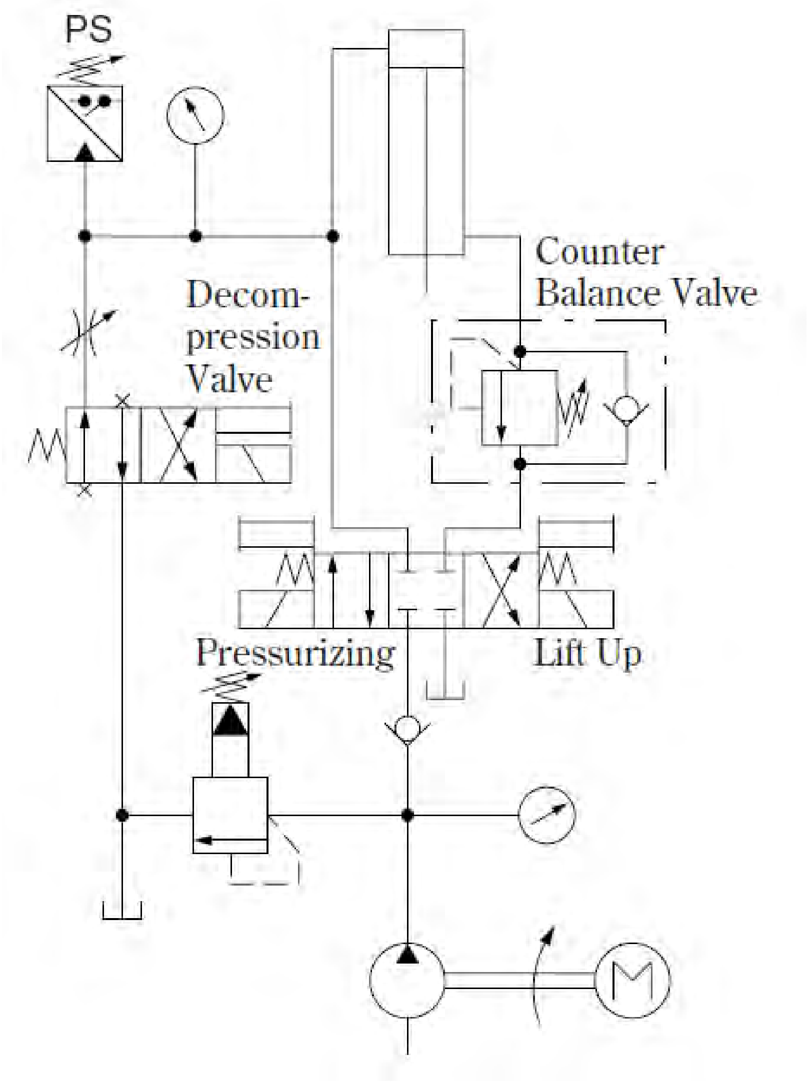


Figure 4-37 - Decompression circuit. A flow control valve is used to transmit the power gradually at the piston achieving smoother motions of the shaft. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.2.4 **Weight Balancing Circuit**

This circuit balances (holds up) the weight by using the balancing valve. The balancing valve cannot hold up the weight if the pump does not work and does not generate enough energy or pressure to hold up the weight. Therefore, a pilot operated check valve is commonly used in the circuit.

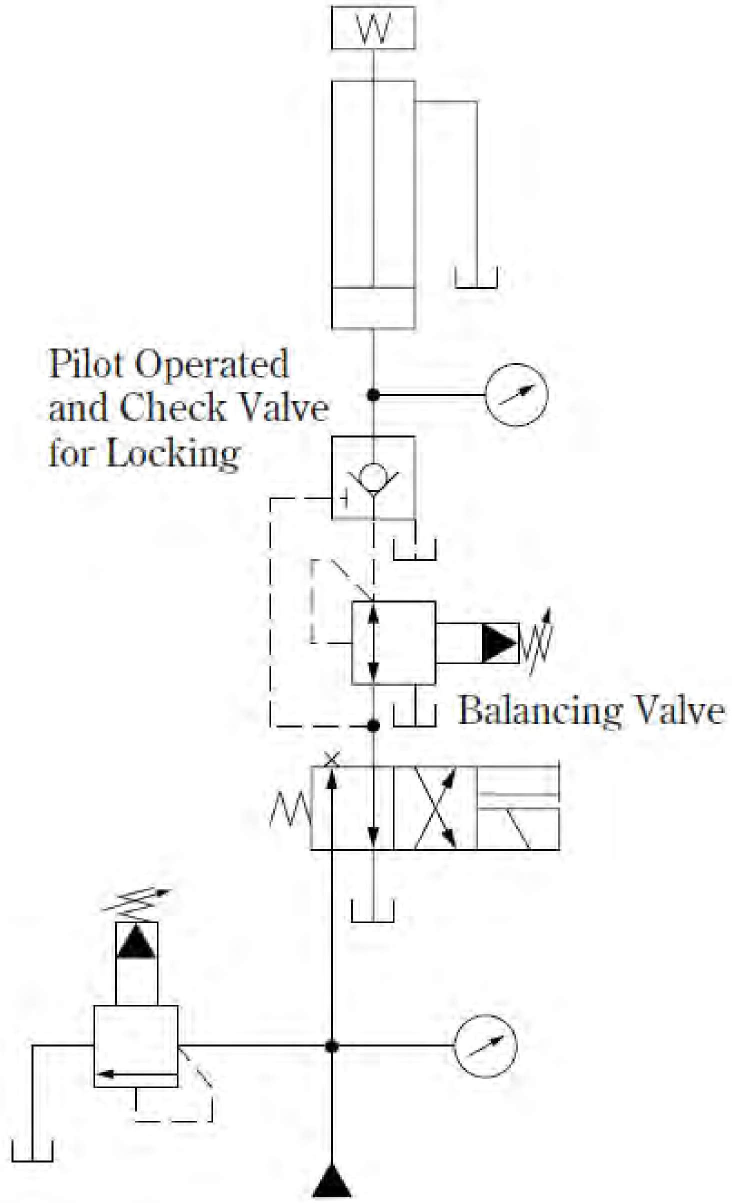


Figure 4-38 - Weight balancing circuit. With the use of a pilot operated check valve, blocking the flow, the weight itself builds up the pressure that holds it with out the need of oil flow.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.3 **Speed Control Circuits**

(Yuken Kogyo Co., 2006) (U.S. Army Material Command, 1971)

4.3.3.1 **Speed Change Circuit I**

This circuit changes the speed of the cylinder motion by employing two flow control valves. Shifting to high speed, change over the solenoid operated directional valve for low speed first then change over the solenoid valve for high speed so that shock is kept small

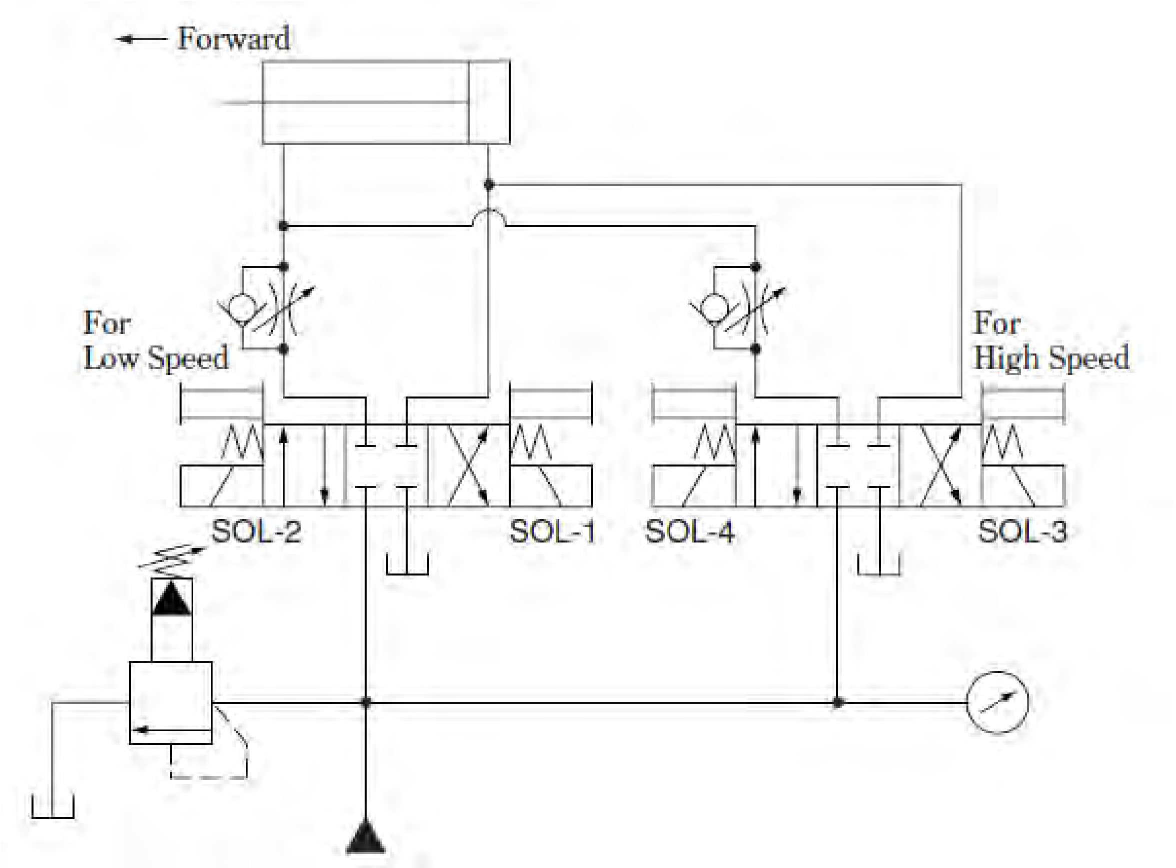


Figure 4-39 - Speed change circuit using one high speed flow control valve and a low speed one.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.3.2 **Speed Change Circuit II**

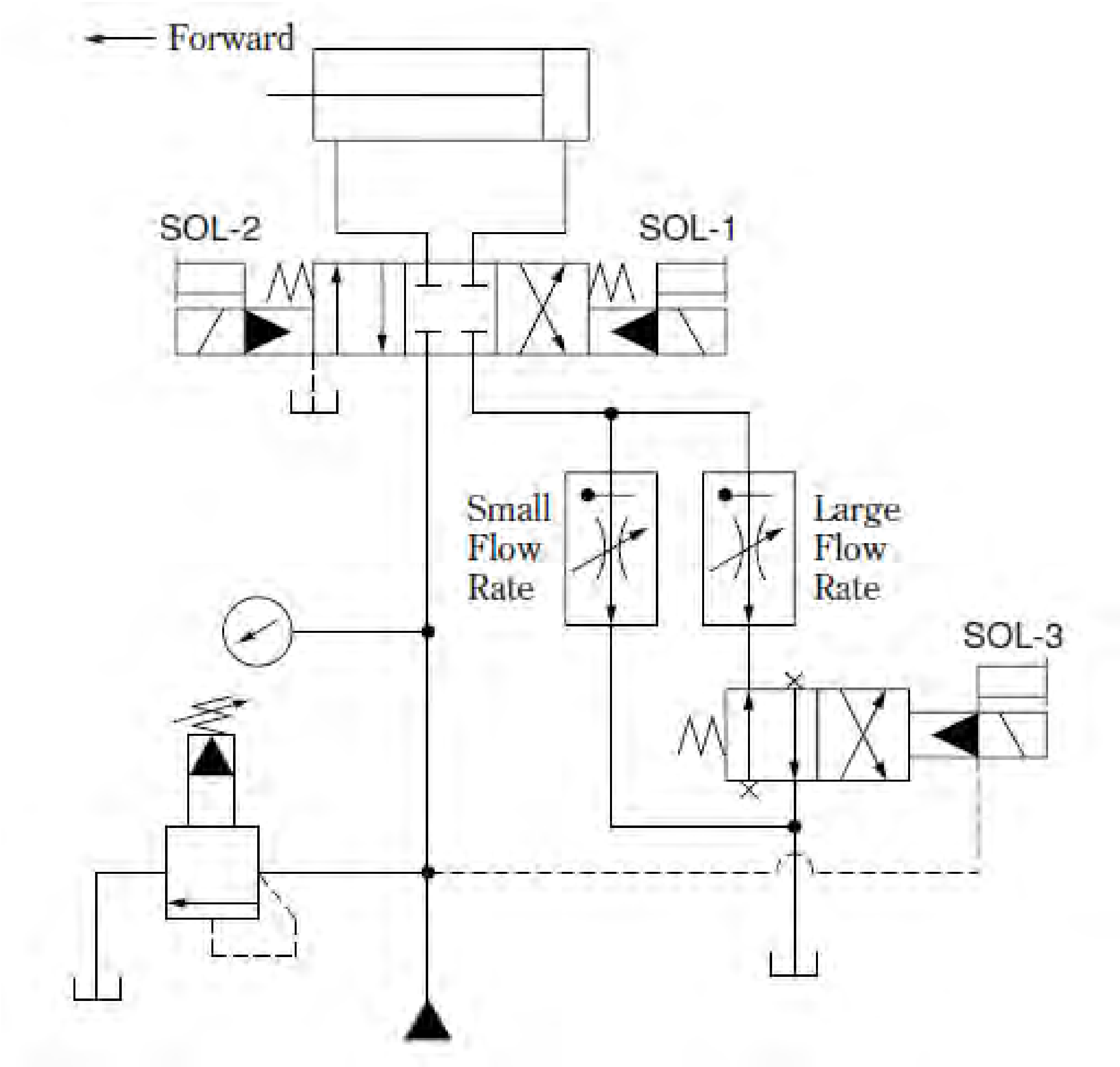


Figure 4-40 - Change speed circuit.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.3.3 Circuit with Proportional Electro-Hydraulic Directional and Flow Control Valve

Optimal flow rate (speed) is achieved by controlling the amount of the spool shifted in the proportional electro- hydraulic valve; the spool is shifted proportional to the amount of electric signal received. The actuator is controlled smoothly with this valve, and the hydraulic circuit is simplified, as shown in Figure 4-41.

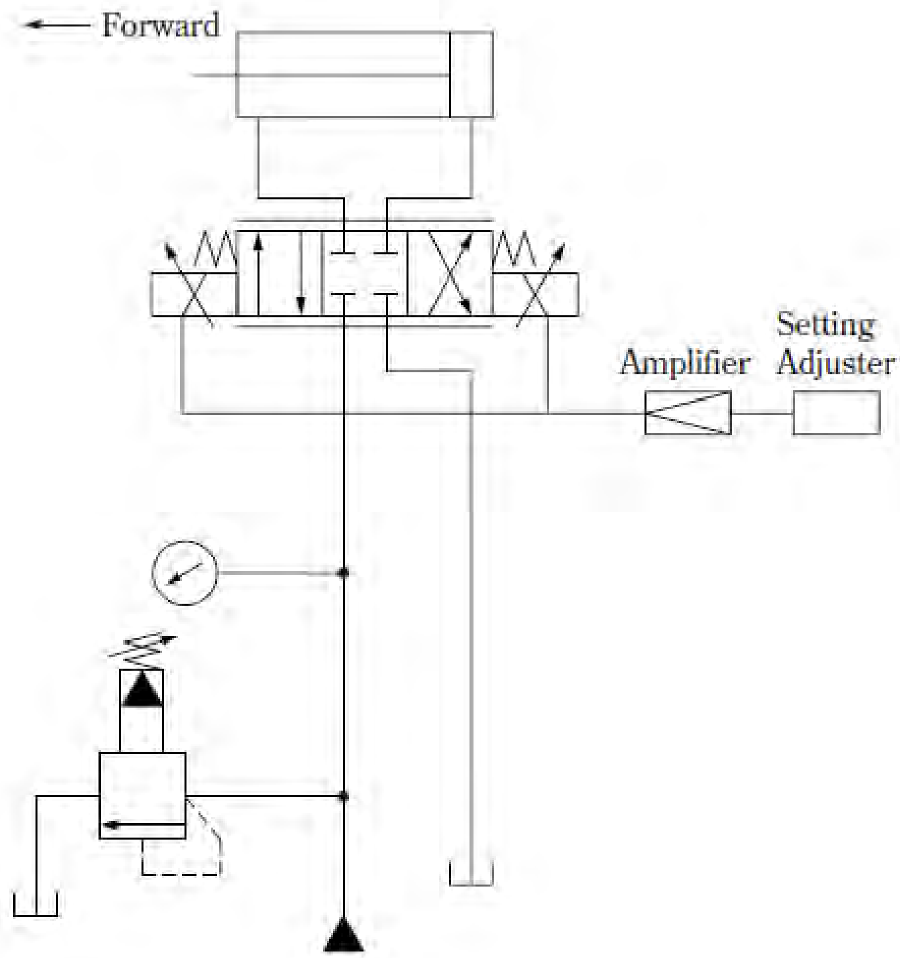


Figure 4-41 - Circuit with Proportional Electro-Hydraulic Directional and Flow Control Valve (Yuken Kogyo, Basic Hydraulics and Components)

4.3.3.4 **Differential Circuit**

When the cylinder is pushed forward, working fluid discharged from the cylinder head is added back to the cylinder cap because of the difference in surface area between the cap and the head of the cylinder. This achieves a faster-cylinder-forward motion, compared to a circuit with only one channel of incoming flow.

$$V = \frac{Pump\ Output\ Flow\ Rate}{Rod\ Area} \quad (4.2)$$

$$F = Pressure\ Supplied\ x\ Rod\ Area \quad (4.3)$$

The relationship between forward speed V and the rod area, and between output force F and the rod area, are obtained as follows. V and F are functions of the rod area. The size of load pressure and pressure loss require due attention.

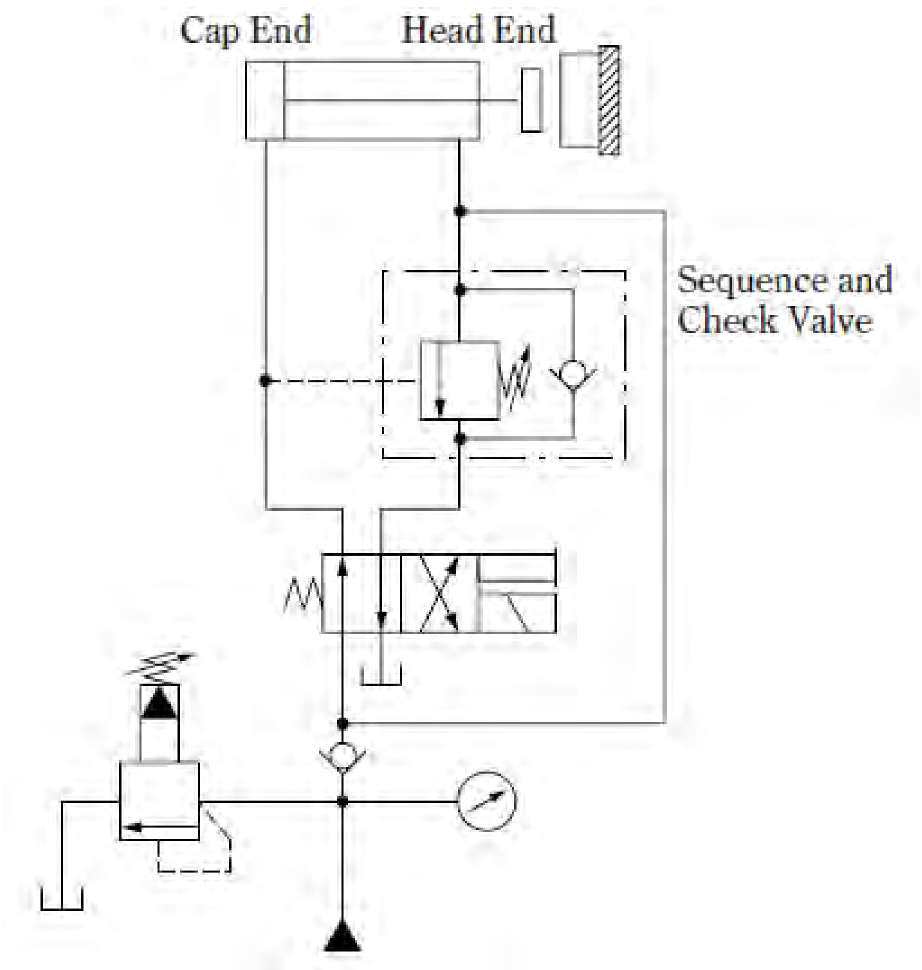


Figure 4-42 - Differential circuit where discharged oil from the cylinder is used to add up pressure to the other end in order to equalize or generally control the forward vs the reverse velocities of the cylinder's shaft.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.3.5 Pre-Fill Valve Circuit

In this circuit, the subsidiary cylinders and the pre-fill valve help the main cylinder achieve pumping function. This circuit drastically reduces the pump-output volume required for the high-speed up-and-down cylinder motion of the press machine.

As the subsidiary cylinders move downward, the main cylinder is pulled down with them, sucking fluid from the reservoir through the pre-fill valve. At the end of the downward motion, the sequence valve is opened, and working fluid is directed to the main cylinder, which then generates a great pressure force on the press.

After the pressure is released, the solenoid valve is set for the high-speed upward motion. The main cylinder is pulled up again with the subsidiary cylinders, pushing

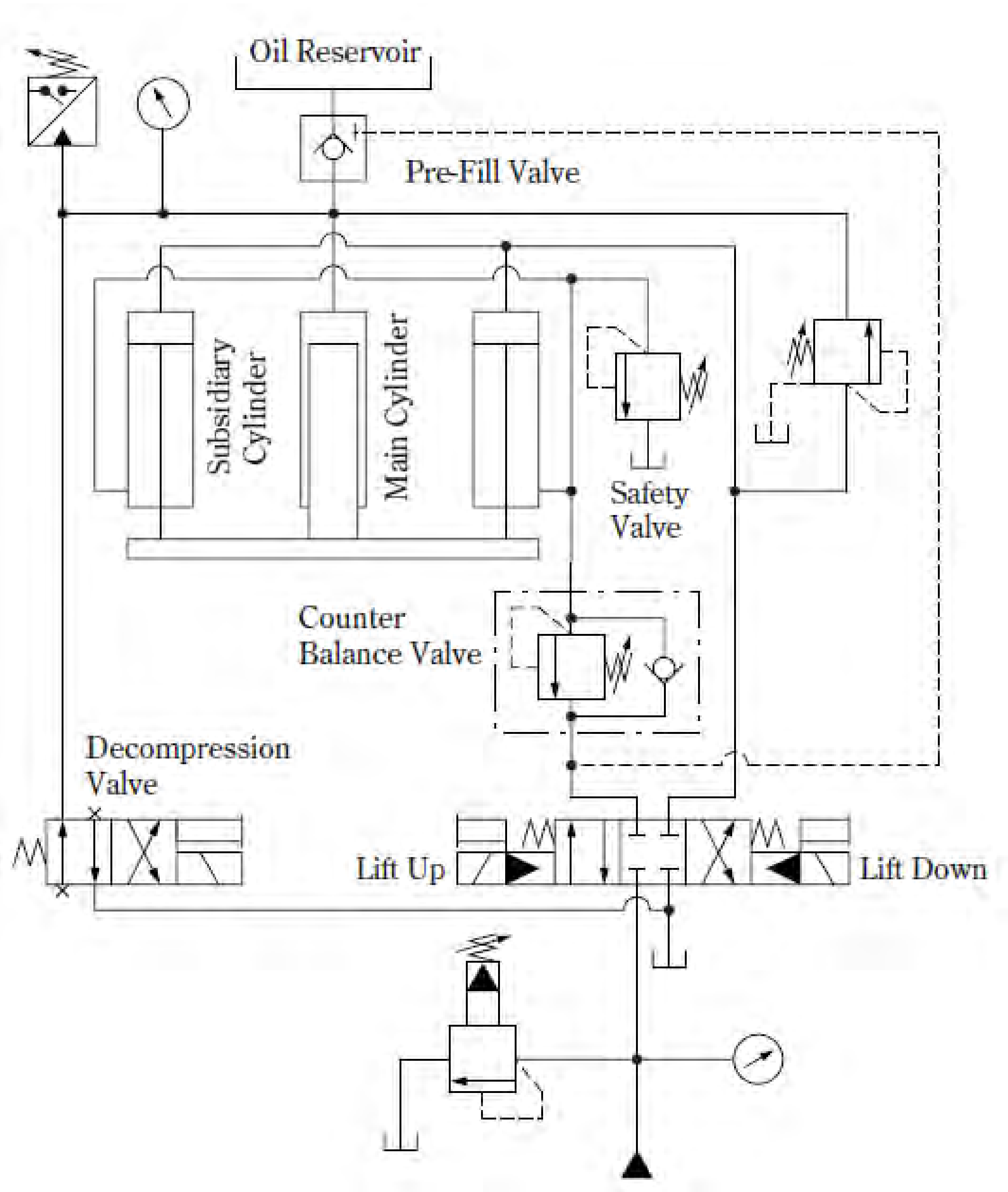


Figure 4-43 - Pre-Fill valve circuit.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.4 **Filter Circuits**

Objectives of filtering and filtration rating change depending on where a filter is set in a circuit. One of the objectives, other than filtering contaminants is to let fluid flow at a certain rate. Another objective is to protect filter and provide a by-pass valve. Examples are shown in Figure 4-44 to Figure 4-47.

Figure 4-48 and Figure 4-49 show circuits in which a clogged filter does not have an adverse impact. Also, filtration rating in these circuits can reach high class performance, (class 8 in the NAS Cleanliness Requirements with 1 pass quality,), when the ideal flow rate is applied to the filters.

4.3.4.1 **Pump Filter Circuit**

This circuit's objective is to protect the hydraulic pump.

This circuit has two types: in one type, a filter is set inside a reservoir on a pump suction port (sometimes it is called a strainer), and in the other, a filter is set outside a reservoir to aid in maintenance. The filtration rates of these circuit types are limited to around 100 μm because of the influence on pump suction resistance.

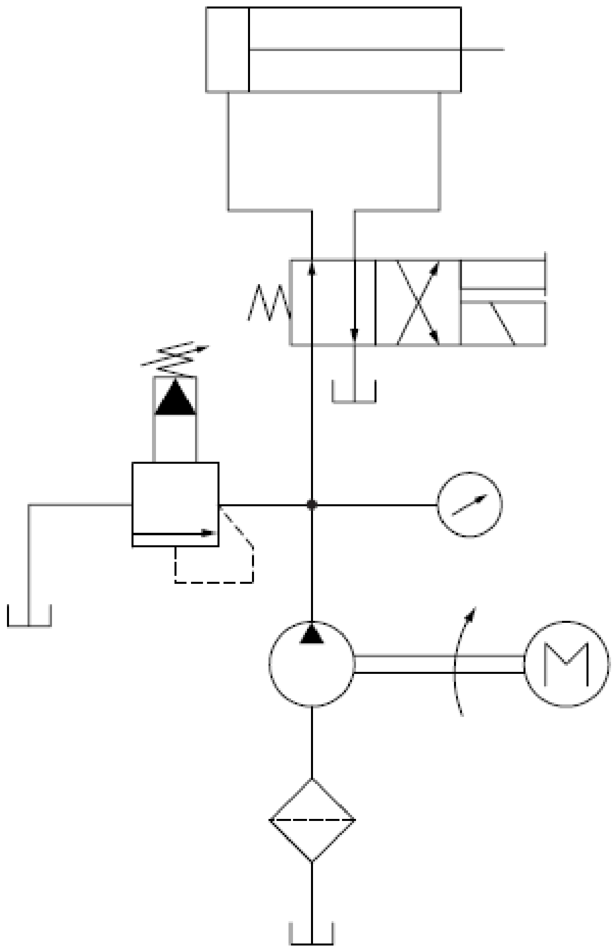


Figure 4-44 - Pump filter circuit.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.4.2 **Pressure Line Filter Circuit I**

This circuit protects the directional and other control valves in the line. Its filtration rate ranges from about 10 μm (solenoid/proportional valve) to 3 μm (servo valve, etc.).

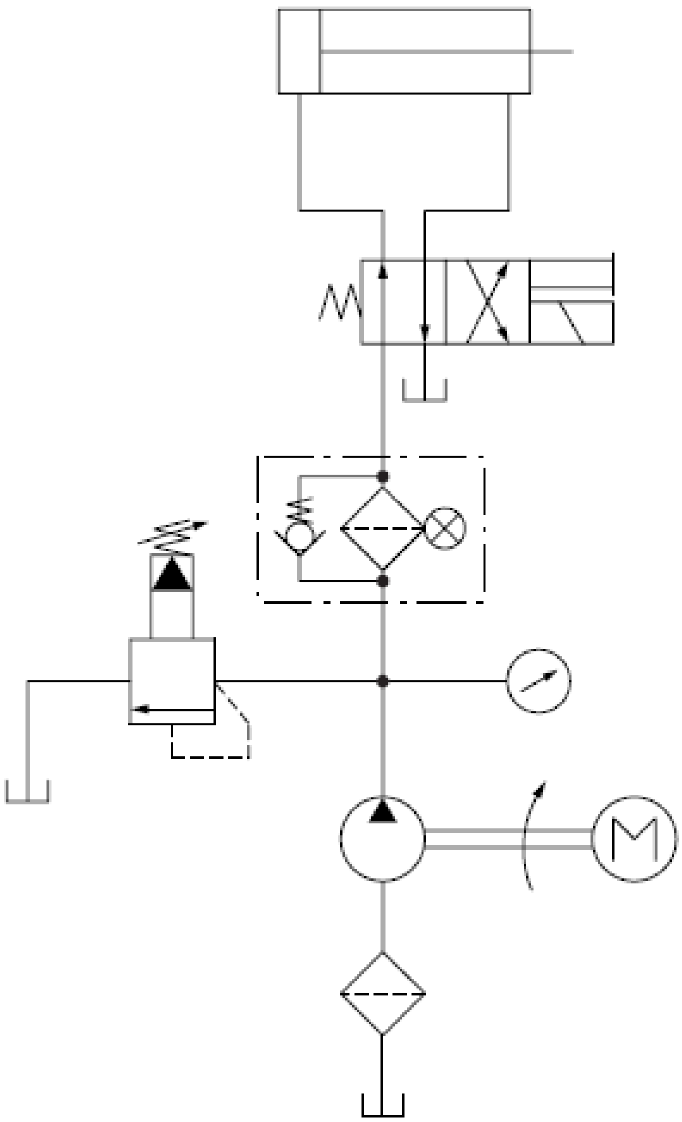


Figure 4-45 - Pressure line filter circuit
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.4.3 **Pressure Line Filter Circuit II**

This circuit has an objective similar to that of Figure 4-45, but it filters out contaminants from the cylinder. It is better to filter out contaminants from only one direction with the anti-reverse-flow valve.

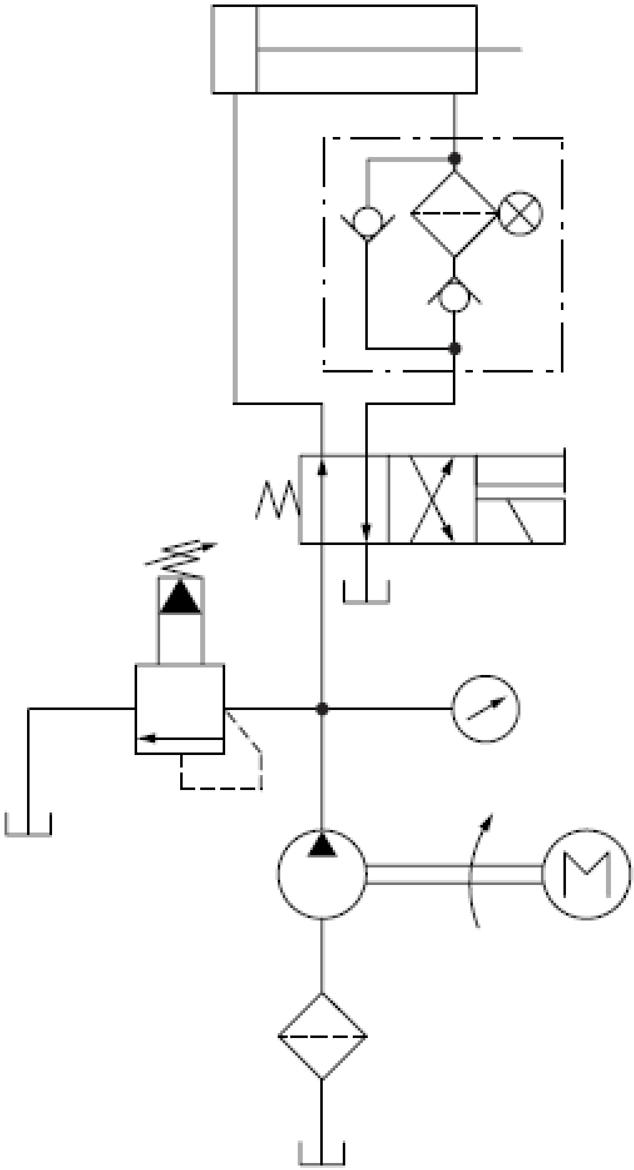


Figure 4-46 - Pressure line filter circuit Filters contaminants from and to the cylinder. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.4.4 **Return Line Filter Circuit**

This circuit filters working fluid going back to the reservoir. The filtration rate is approximately 10 μm to 20 μm.

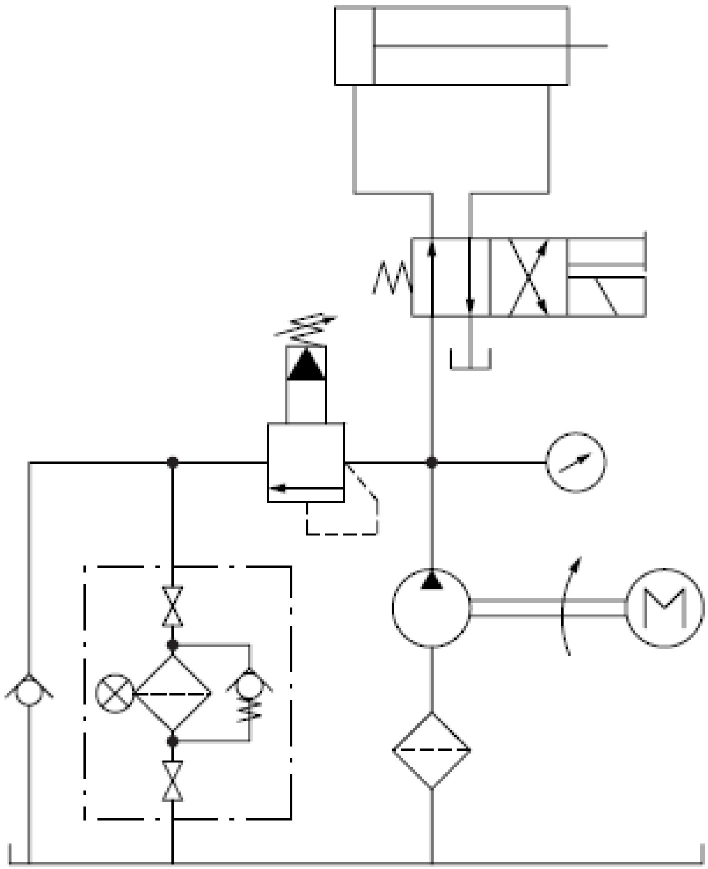


Figure 4-47 - Return line filter circuit.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.4.5 **Pressure Line Bleed-Off Filter Circuit**

This circuit filters a small amount of the by-passed outlet flow from the pump ($about\ 1\ to\ 2\ L/min$ ($0.26 \sim 0.53\ U.S.GPM$)).

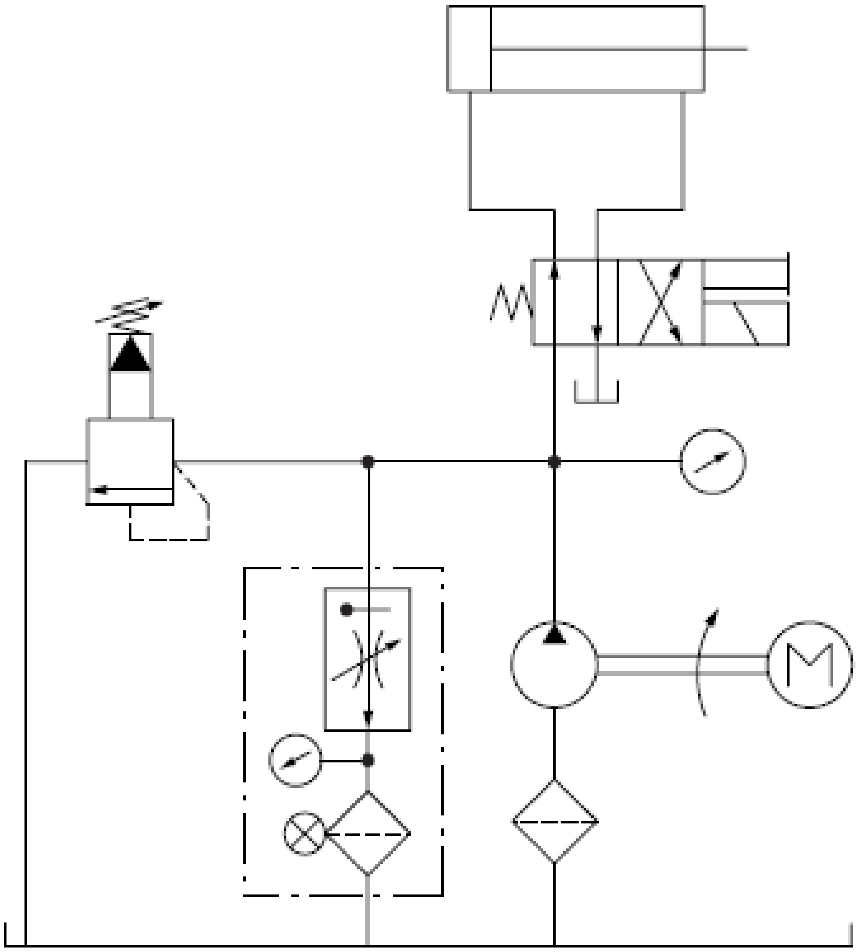


Figure 4-48 - Pressure line bleed-off filter circuit
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.4.6 **Off-Line Filter Circuit**

By using a pump and an electric motor specialized for filtration, this circuit filters contaminants even when the main hydraulic pump is not turned on. This circuit has the best filtration of all the filtration circuits.

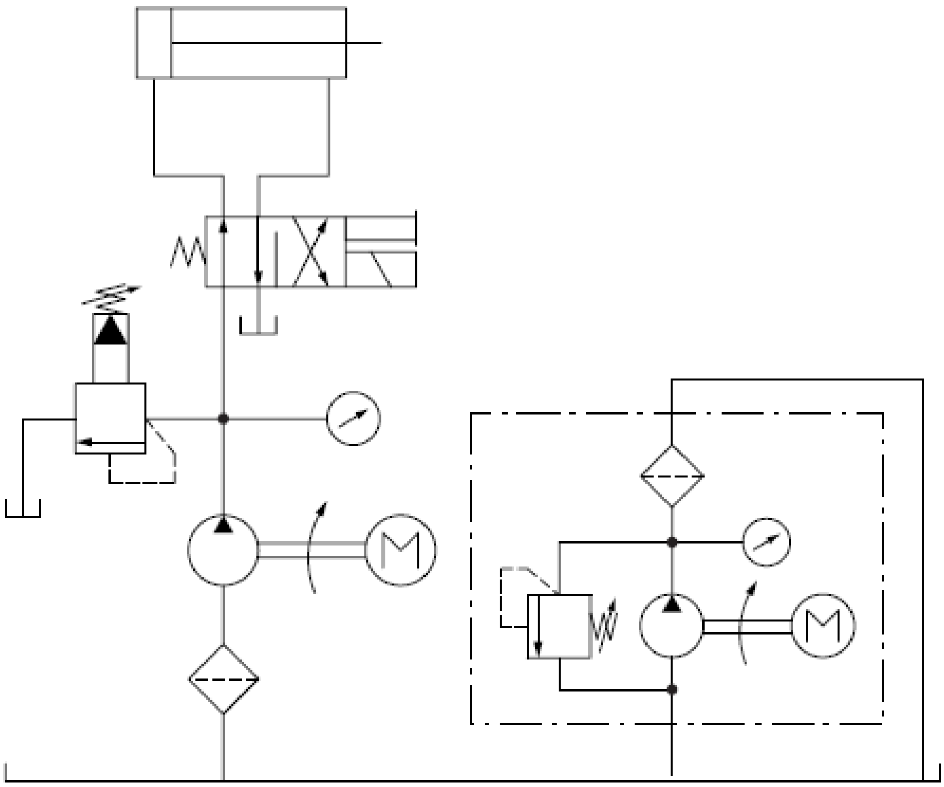


Figure 4-49 - Off-line filter circuit. It's a separate system dedicated to filtering and does not affect the main hydraulic system. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.5 Synchronizing Circuits

This circuit synchronizes movements of multiple actuators. Minimal error should never be overlooked, however, to avoid the accumulation of synchronization errors, it is always recommended that the circuit be set in such a way that the error is corrected at the end of an operation with one, full cylinder stroke, instead of the repeated, half-way cylinder motion. (Yuken Kogyo Co., 2006)

4.3.5.1 Synchronizing Circuit with Mechanical Combination

This circuit realizes a synchronized motion by mechanically combined cylinder rods. In the following figure, the relationship between the two main cylinders and two other auxiliary cylinders is also a mechanical combination. This circuit does not necessitate a control valve for synchronization. Synchronization errors would be happened by production accuracy and rigid of mechanism.

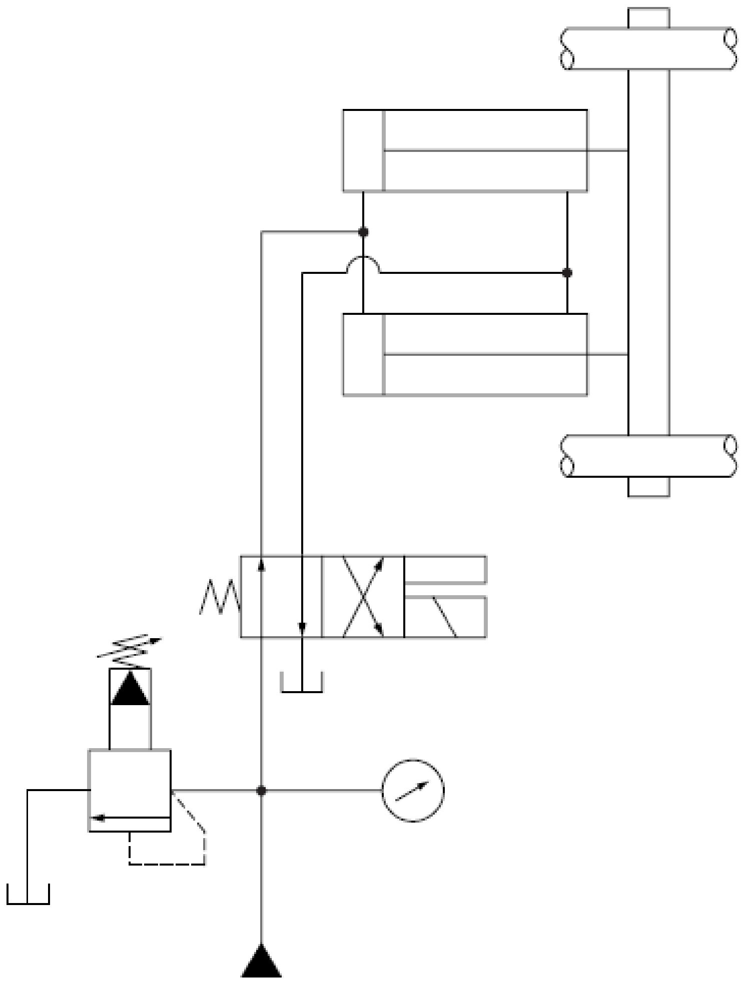


Figure 4-50 - Synchronized system with mechanical combination. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.5.2 **Synchronizing Circuit with Flow Control Valves**

In this circuit, the flow control valve controls fluid flowing in and out of the cylinders. Generally, a high accuracy valve is employed.

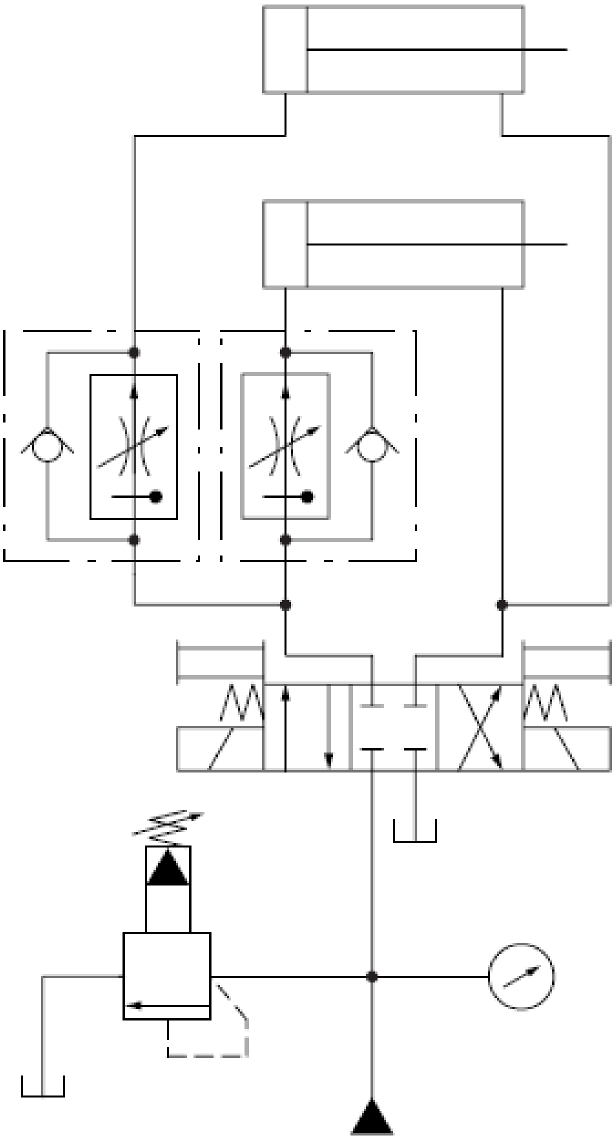


Figure 4-51 - Synchronized system with the use of a high accuracy flow control valve.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.5.3 **Circuit with Flow Divider**

This circuit utilizes the flow divider specialized for synchronization.

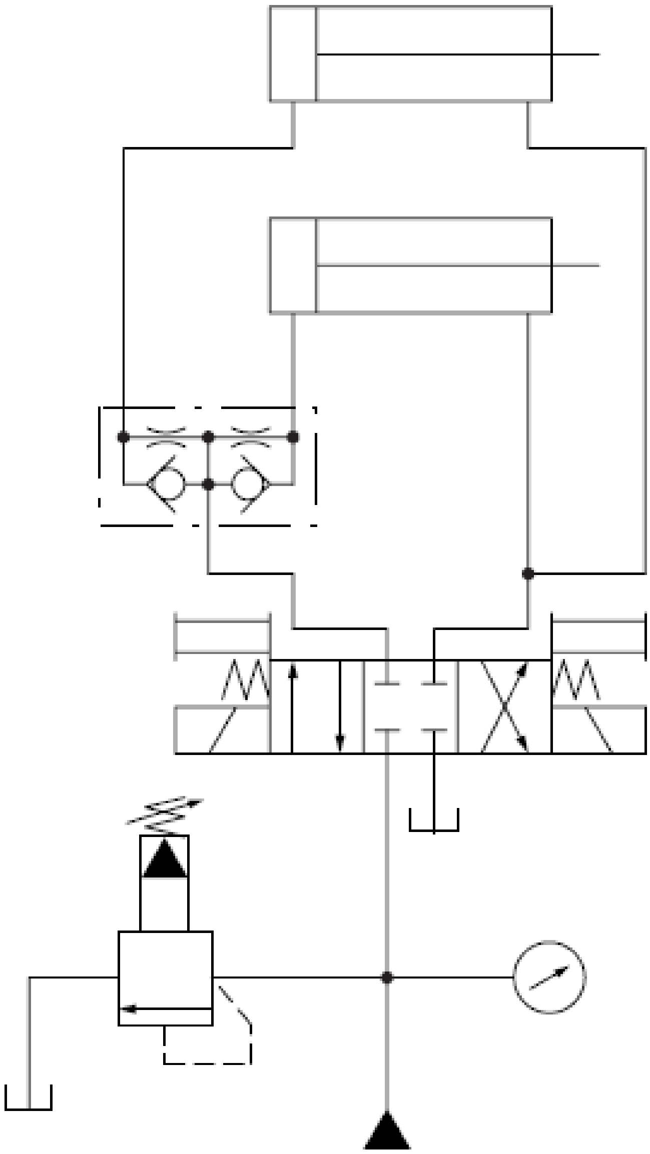


Figure 4-52 - Synchronized system using a flow divider.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.5.4 **Circuit with Synchronized Hydraulic Motors**

With the shafts combined, these motors can displace the same amount of working fluid to and from each cylinder. The accuracy of the amount of displacement controls the accuracy of synchronization. Therefore, if the volumetric efficiency is the same, setting the circuit with high speed motors reduces synchronization errors.

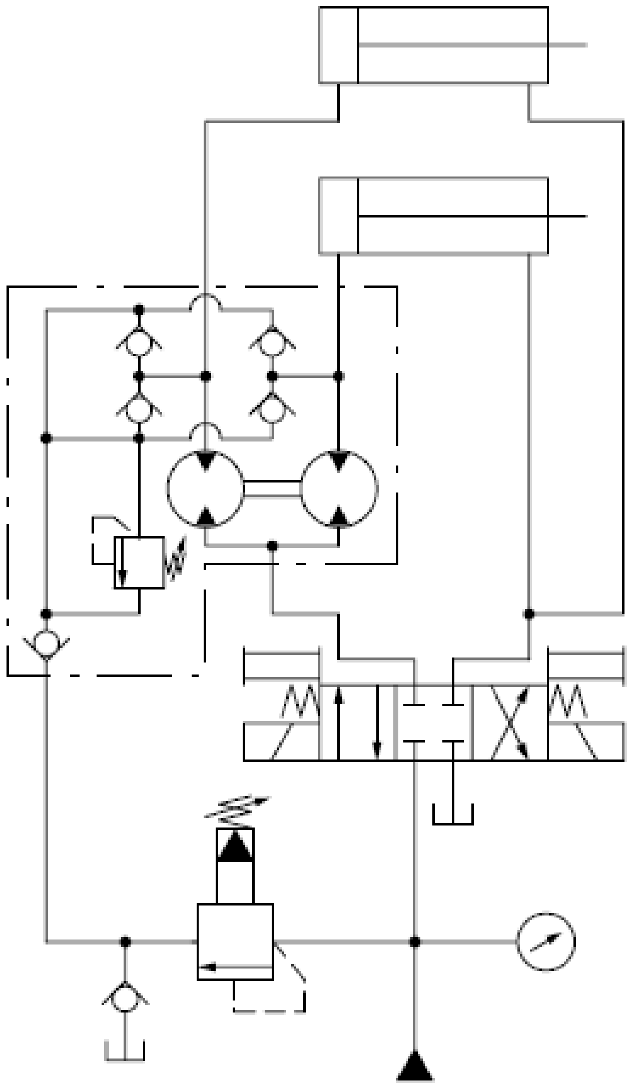


Figure 4-53 - Synchronized system using hydraulic motors. The synchronization error depends on the deviation of displacement between the motors. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.5.5 **Circuit with Synchronized Cylinders**

This circuit realizes a very accurate synchronized motion via combined synchronizing cylinders. But, sometimes, spacing becomes an issue because it requires the volume of all the cylinders to be the same.

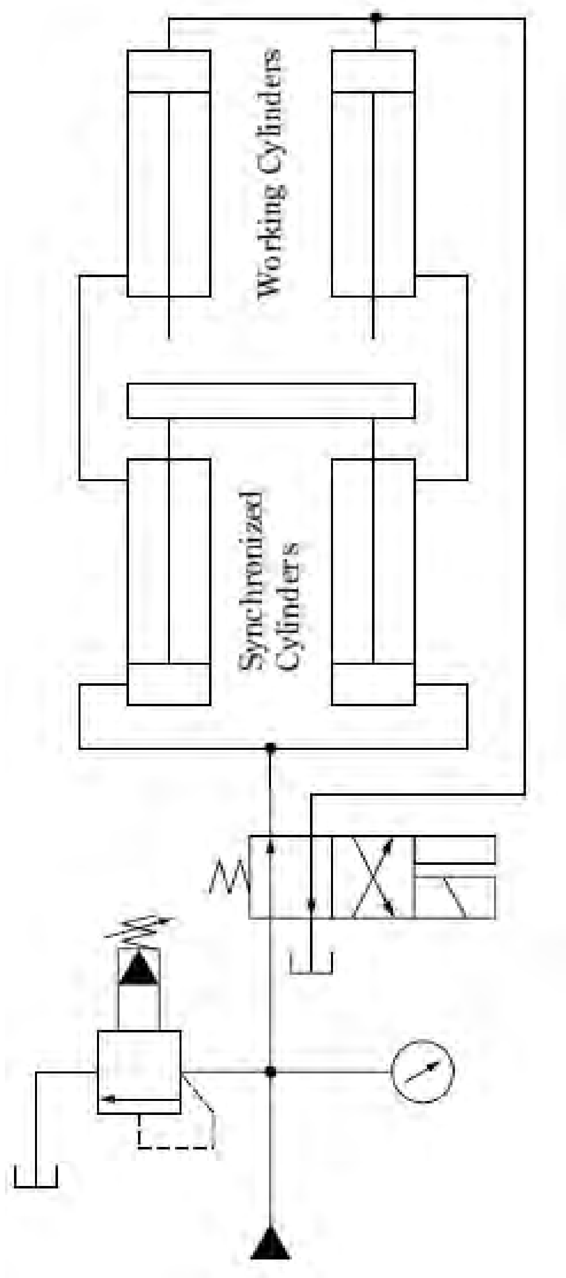


Figure 4-54 - System with synchronized cylinders. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.5.6 **Circuit with Servo Valve**

This circuit detects the position of two cylinders and uses two servo valves to control the amount of working fluid required to adjust synchronization errors. The following figure is an example of such feedback-synchronization control. Rather than detecting the position of one cylinder and giving the feedback to the other cylinder for synchronization, it is better and more accurate with less time lag if each cylinder works separately and their positions are controlled by different servo valves.

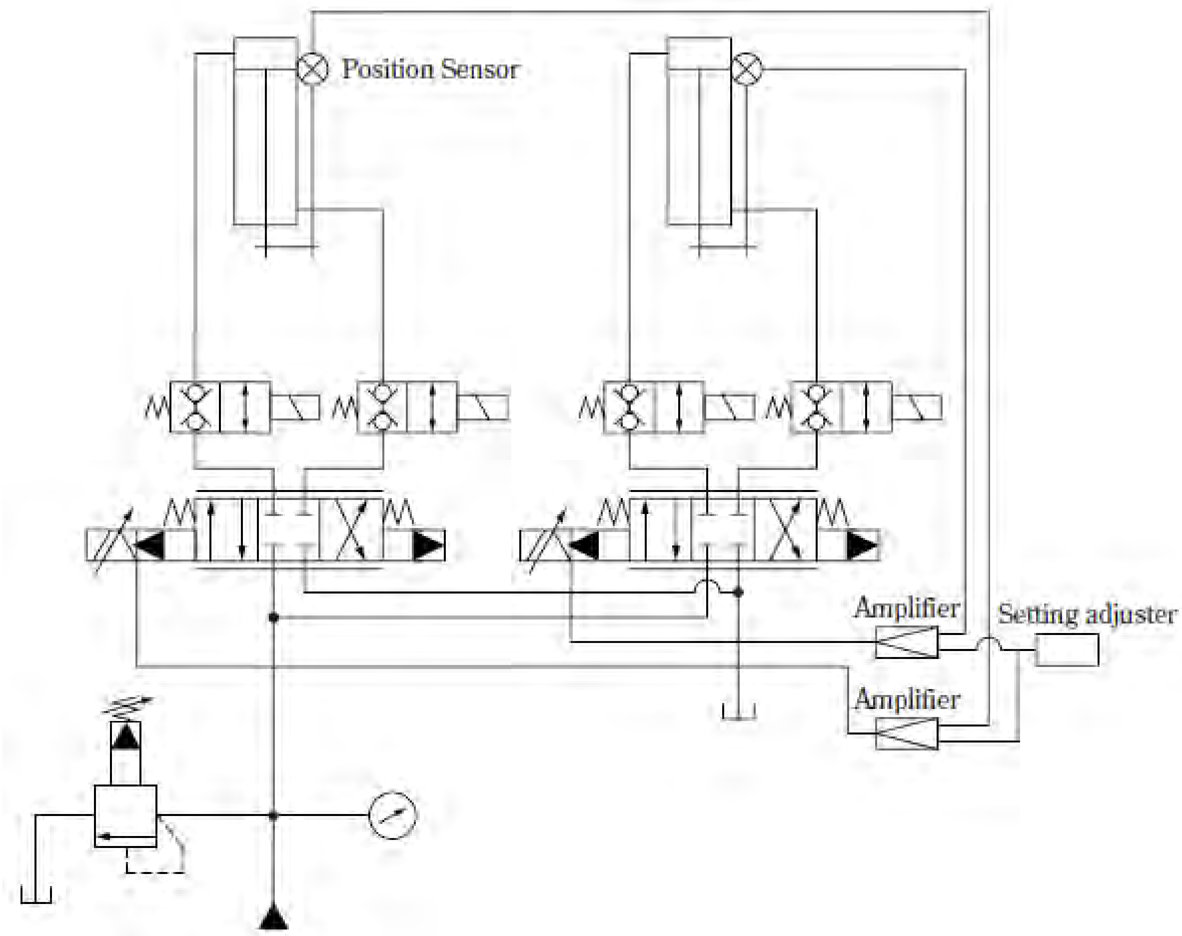


Figure 4-55 - Circuit with servovalve.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.6 **Intensifying Circuits**

4.3.6.1 **Circuit with a Cylinder as Intensifier**

This circuit intensifies pressure by using the difference between cap and head area in cylinders. In the following figure, the solenoid valve for adding pressure is turned ON. Working fluid channelled through the sequence valve (1) pushes the working cylinder head forward until it hits an object. The contact between the cylinder head and the object eventually increases the pressure inside the line. Then, the circuit delivers the pressurized working fluid to the intensifying cylinder in which the fluid is pressurized yet further. The highly pressurized working fluid in the intensifying cylinder is then supplied back to the working cylinder. The decompression valve on the primary side of the intensifying cylinder adjusts the output power. Also, in the process of returning the cylinders, it is important to note that the intensifying cylinder is returned by the sequence valve (2) (using the counter balance valve as the sequence valve).

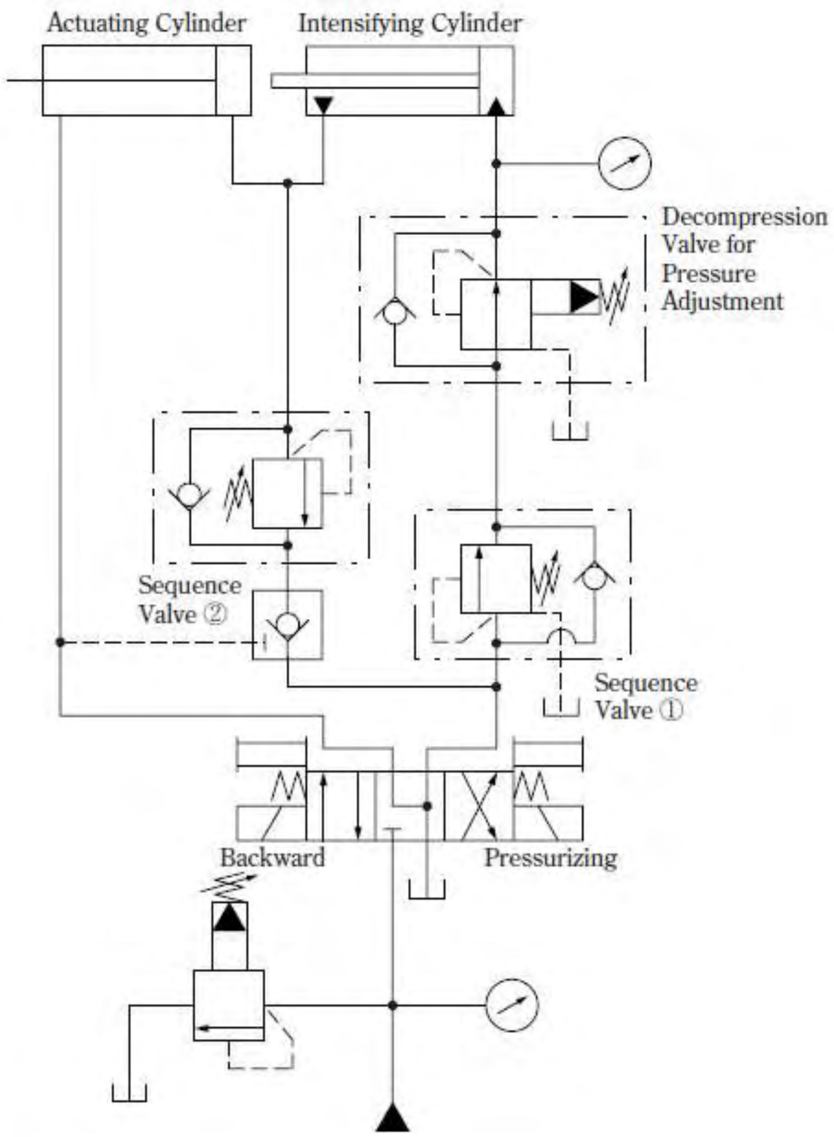


Figure 4-56 - Circuit with Intensifying Cylinders.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.7 **Brake Circuits**

(Yuken Kogyo Co., 2006) (U.S. Army Material Command, 1971)

4.3.7.1 **Brake Circuit with Hydraulic Motor (Fig. 15.28)**

This figure is an example of a motor that turns both directions. With the solenoid valve in position (1), the hydraulic motor turns right. After that, the solenoid valve is in the middle position, but the hydraulic motor keeps working as a pump because of inertia. Discharged working fluid runs through the check valve (4) and returns to the reservoir with back pressure given by the relief valve. The primary side of the motor becomes low pressure, thus working fluid through the check valve (3) is supplied into the line. In the case of a left turn, the check valve (2) and (5) are used. (Yuken Kogyo Co., 2006)

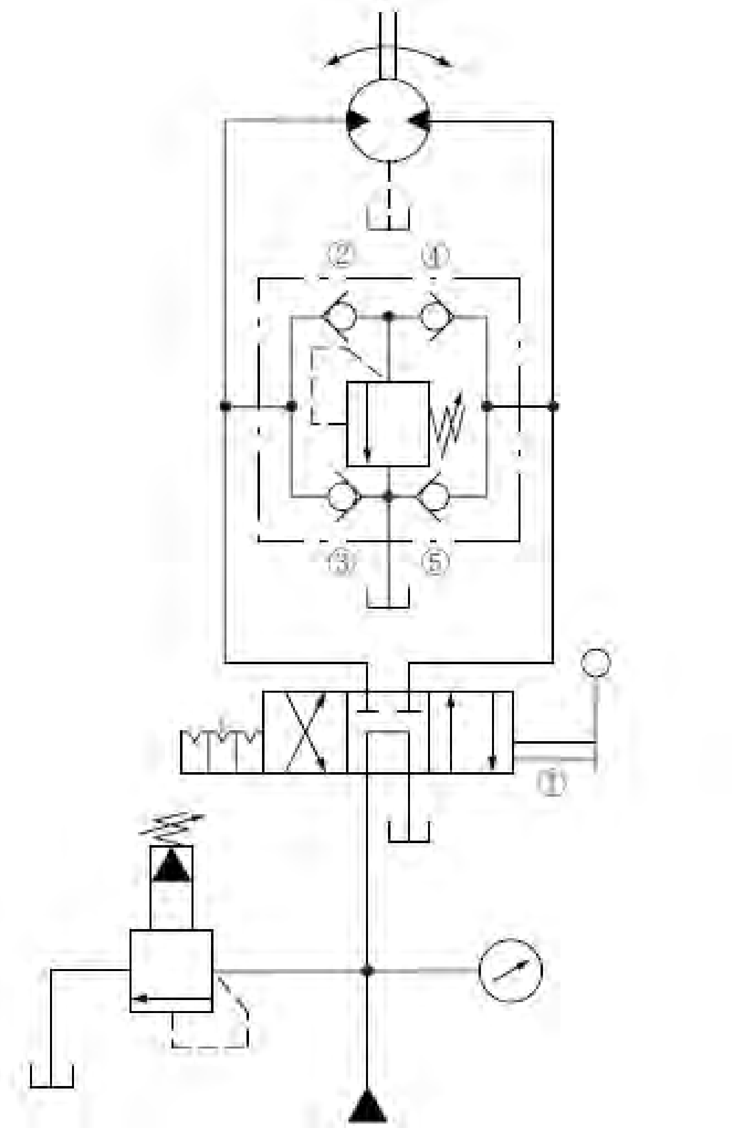


Figure 4-57 - Brake circuit with hydraulic motor.
(Yuken Kogyo, Basic Hydraulics and Components)

4.3.8 **Closed Circuits**

4.3.8.1 **Closed-Center Circuit**

In the closed-center system (U.S. Army Material Command, 1971), the fluid is under pressure whenever the power pump is operating. The three actuators are arranged in parallel and actuating units B and C are operating at the same time, while actuating unit A is not operating. This system differs from the open-center system in that the selector or directional control valves are arranged in parallel and not in series. The means of controlling pump pressure varies in the closed-center system. If a constant delivery pump is used, the system pressure is regulated by a pressure regulator. A relief valve acts as a backup safety device in case the regulator fails. If a variable displacement pump is used, system pressure is controlled by the pump's integral pressure mechanism compensator. The compensator automatically varies the volume output. When pressure approaches normal system pressure, the compensator begins to reduce the flow output of the pump. The pump is fully compensated (near zero flow) when normal system pressure is attained. When the pump is in this fully compensated condition, its internal bypass mechanism provides fluid circulation through the pump for cooling and lubrication. A relief valve is installed in the system as a safety backup. (Figure 4-58). An advantage of the open-center system over the closed-center system is that the continuous pressurization of the system is eliminated. Since the pressure is built up gradually after the selector valve is moved to an operating position, there is very little shock from pressure surges. This action provides a smoother operation of the actuating mechanisms. The operation is slower than the closed-center system, in which the pressure is available the moment the selector valve is positioned. Since most aircraft applications require instantaneous operation, closed-center systems are the most widely used.

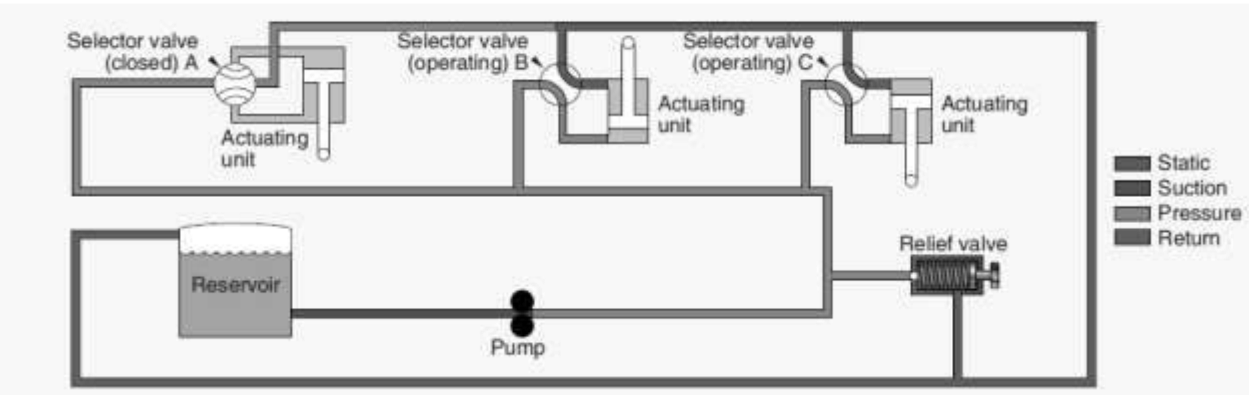


Figure 4-58 - Closed Circuit for Vehicle
(Fluid Power Aviation)

Closed circuits (Yuken Kogyo Co., 2006) are widely employed in vehicles performing running, circling or HST (Hydro Static Transmission: no shift change for speed change) functions. One of the characteristics of this circuit is to use a pump as a hydraulic motor to absorb the power: this is a reverse use of the pumping function of a motor found in the previous section on brake circuits. In addition, pressure inside the line is low because the hydraulic pump controls the speed of the vehicle. This system is more efficient and achieves less heat generation when compared to valve control systems. The feed pump fills and replaces working fluid internally, and it supplies clean fluid through a filter. The circuit must be made in such a way that safety valve (1) has a higher pressure than that of safety valve (2), and that working fluid from the feed pump is discharged to the reservoir via the flushing valve.

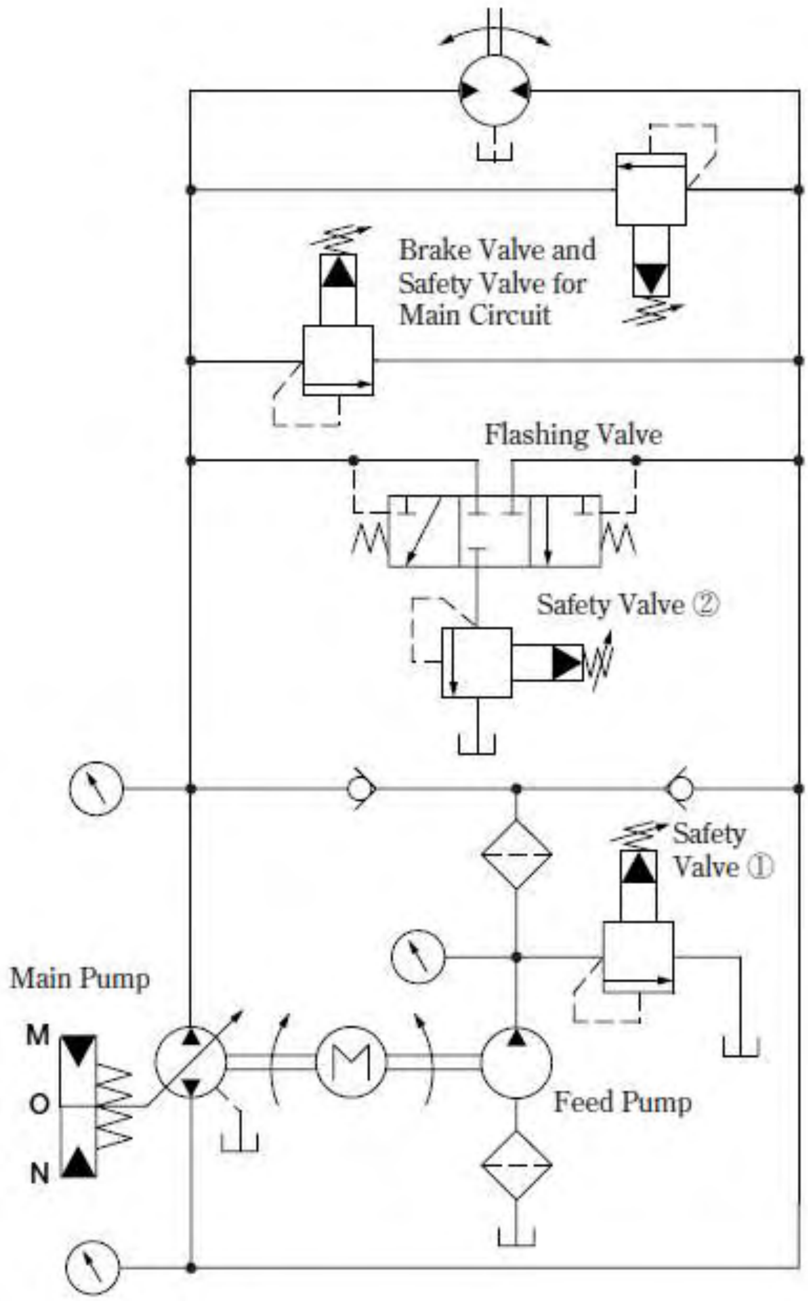


Figure 4-59 - Closed circuit. Except for the main pump, another one is used as a hydraulic motor. (Yuken Kogyo, Basic Hydraulics and Components)

4.3.9 **Hydraulic Power Pack System**

A hydraulic power pack (U.S. Army Material Command, 1971) is a small unit that consists of an electric pump, filters, reservoir, valves, and pressure relief valve. [Figure 4-60]

Figure 4-60 - Hydraulic Power Pack (Fluid Power Aviation)

The advantage of the power pack is that there is no need for a centralized hydraulic power supply system and long stretches of hydraulic lines, which reduces weight. Power packs could be driven by either an engine gearbox or electric motor. Integration of essential valves, filters, sensors, and transducers reduces system weight, virtually eliminates any opportunity for external leakage, and simplifies troubleshooting. Some power pack systems have an integrated actuator. These systems are used to control the stabilizer trim, landing gear, or flight control surfaces directly, thus eliminating the need for a centralized hydraulic system.

5. Conclusions

The following deductions were made through the elaboration of this thesis:

- Taking advantage of the fluid properties, hydraulic systems achieve great superiority in terms of speed and power compared to other systems.
- Because of their flexibility and high performance, hydraulics are applied in a wide range of industries: from construction machinery, automobiles, and airplanes (outdoor) to machine tools and press machines (indoor).
- Hydraulic systems consist of the following components: pumps, motors, valves, accumulators, filters, reservoirs, hoses and fittings, heat exchangers and shock absorbers. Each one of the above may be found in different designs depending on the system requirements.
- The analysis of fluid power systems brings together the scientific areas of Fluid Mechanics, Transport Phenomena and Thermodynamics. Complex Mechanical components are used for the establishment of such systems; Electronics and Automations are often seen in the control section of the systems.
- Hydraulic components provide a wide variety of systems that can be assembled for a specific goal. For this reason, assembly patterns have been established in favor of efficiency depending on the objective that has to be faced by the system.
- Except for transmitting power, the hydraulic oil also lubricates the system, protecting the components from wear and corrosion and making hydraulic systems reliable and resistant in time.

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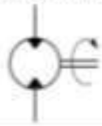
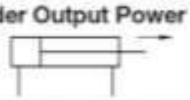
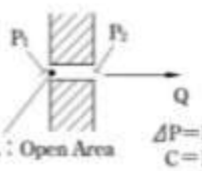
7. Appendix

	SI Unit	US Unit
Hydraulic Pumps: ● Hydraulic Power (Pump Output Power)	$L_o=\frac{P\cdot Q}{60}$ L_o : Hydraulic Power kW P : Pressure MPa Q : Flow Rate L/min ※1 kW=1 kN·m/s =60 kN·m/min	$L_o=\frac{P\cdot Q}{1714}$ L_o : Hydraulic Power HP P : Pressure psi Q : Flow Rate U.S.GPM
Hydraulic Pumps: ● Input Power	$L_i=\frac{2\pi TN}{60\,000}$ L_i : Input Power kW T : Shaft Torque N·m N : Rotation Speed r/min	$L_i=\frac{T\cdot N}{5\,252}$ L_i : Input Power HP T : Shaft Torque lbf·ft N : Rotation Speed r/min
Hydraulic Pumps: ● Volumetric Efficiency	$\eta_V=\frac{Q_P}{Q_O}\times 100$ η_V : Overall Efficiency % Q_P : Output Flow Rate at Pressure P L/min (U.S.GPM) Q_O : Output Flow Rate at No pressure L/min (U.S.GPM) ※ Q_O-Q_P= Total Leakage Amount at Pump Inside	
Hydraulic Pumps: ● Overall Efficiency	$\eta=\frac{L_o}{L_i}\times 100 =\frac{P\cdot Q}{60L_i}\times 100$ η : Overall Efficiency % L_o : Hydraulic Power kW L_i : Input Power kW P : Output Pressure MPa Q : Output Flow Rate L/min	$\eta=\frac{L_o}{L_i}\times 100 =\frac{P\cdot Q}{1714L_i}\times 100$ η : Overall Efficiency % L_o : Hydraulic Power HP L_i : Input Power HP P : Output Pressure psi Q : Output Flow Rate U.S.GPM
● Output Power of Hydraulic Motor 	$L=\frac{2\pi T\cdot N}{60\,000}$ L : Output Power kW T : Torque Nm N : Rotation Speed r/min	$L=\frac{T\cdot N}{5\,252}$ L : Output Power HP T : Torque lbf·ft N : Rotation Speed r/min
● Cylinder Output Power 	$L=\frac{F\cdot V}{60}$ L : Output Power kW F : Thrust kN V : Speed m/min	$L=\frac{F\cdot V}{33\,000}$ L : Output Power HP F : Thrust lbf V : Speed ft/min
● Power Loss of a Valve Flow Rate : Q Pressure : P_1 [Valve] Pressure : P_2 Pressure Loss : $\Delta P=P_1-P_2$ Power Loss at Valve Inlet/Outlet : L	$L=\frac{\Delta P\cdot Q}{60}$ L : kW ΔP : MPa Q : L/min	$L=\frac{\Delta P\cdot Q}{1714}$ L : HP ΔP : psi Q : U.S.GPM
● Viscosity (Absolute Viscosity) and Kinematic Viscosity	$\mu=\rho\cdot\nu_1=\rho\cdot\nu_2\times 10^{-6}$ μ : Viscosity (Absolute Viscosity) Pa·s(=N·s/m²) ρ : Density kg/m³ ν_1 : Kinematic Viscosity m²/s ν_2 : Kinematic Viscosity mm²/s	
● Reynolds Number  R : Reynolds Number ν : Kinematic Viscosity	$R=\frac{V\cdot d}{\nu_1}=\frac{4\,000\,Q}{60\pi d\cdot\nu_1}=\frac{2120\,Q}{d\cdot\nu_2}$ R : Dimensionless V : cm/s d : cm ν_1 : cm²/s ν_2 : mm²/s{cSt} Q : L/min	※R<2 300 ········· Laminar Flow R>2 300 ········· Turbulent Flow
● Flow of Orifice  A : Open Area $\Delta P=P_1-P_2$ C = Flow Rate Index γ = Specific Weight ρ = Density	$Q=C\cdot A\sqrt{\frac{2\Delta P}{\rho}\times 10^6}\times 6$ Q : L/min C : Dimensionless A : cm² ρ : kg/m³ ΔP : MPa	$Q=C\cdot A\sqrt{\frac{2g}{\gamma}\cdot\Delta P}$ Q : in³/s C : Dimensionless A : in² g : 386.4 in/s² γ : lbf/in³ ΔP : psi
	Note) Flow rate index is affected by forms of flow lines and Reynolds number, and is usually about 0.6 to 09.	

Useful Equations (Yuken Kogyo - Basic Hydraulics and Components)