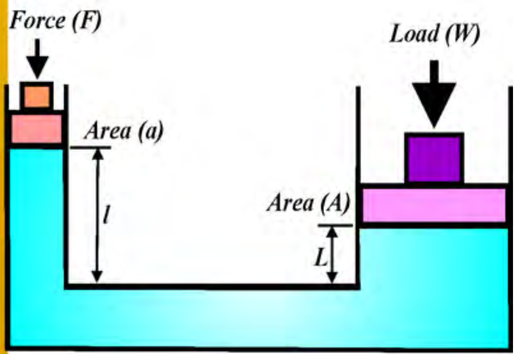
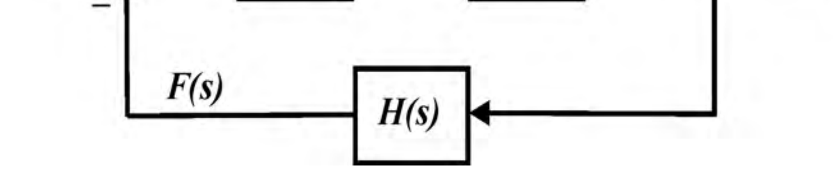


Ahmed Abu Hanieh

FLUID POWER CONTROL

Second Edition



FLUID POWER CONTROL

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Ahmed Abu Hanieh

Jerusalem

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To my beloved homeland

Palestine

إني رأيتُ أنه ما كتب أحدُهم في يومِه كتابًا إلا قال في غدِه،
لو غُيِّرَ هذا لكان أحسن، ولو زِيْدَ ذاك لكان يُستَحسن، ولو قُدِّمَ
هذا لكان أفضل، ولو تُرِكَ ذاك لكان أجمل، وهذا من أعظم العِبر،
وهو دليلٌ على استيلاءِ النقصِ على جملةِ البشر

العماد الأصفهاني

Once an author writes a book, next day he will say:
Changing this will make it better, adding that will make it stronger,
preceding this will make it beautiful, removing that will make it wonderful.
This is a great lesson and an evidence of human imperfection.

Al Imad Al Asfahani

About the author

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Professor of Mechanical Engineering, works in the department of Mechanical and Mechatronics Engineering at Birzeit University - Palestine since 2004. Participated in more than seventeen research and capacity building (CBHE) projects like Tempus and Erasmus+ projects. Participated in Erasmus Mundus and Erasmus+ International Credit Mobility (ICM) projects with different EU universities and Quality Improvement Fund (QIF) projects funded by the International Bank. Established and directed two Master programs (Sustainable Engineering in Production and Renewable Energy Management). Acts as a Palestinian Higher Education Reforms Expert (HERE) and as a board member of the Higher Council of Innovation and Excellence (HCIE). Supervised Doctorate students in European Universities. Supervised several Bachelor graduation projects and Master theses. Author of three books, more than 20 journal articles and more than 30 conference papers.

Preface

The idea of developing a second edition for this book came out after fifteen years of teaching the fluid power control course at Birzeit University besides to the long experience of the author in the real hydraulic and pneumatic systems which is reflected on the different applications of the discussed circuits. Most of the existing books in this field discuss hydraulic and pneumatic systems in concentrating on the design and components of the system without going deep enough into the problem of dynamic modelling and control of these systems. This book attempts to compromise between theoretical modelling and practical understanding of fluid power systems by using modern control theory based on implementing Newton's second law in second order differential equations transformed into direct relationships between inputs and outputs via transfer functions or state space approach. More examples have been added to the book and more problems are added to each chapter. The arrangement of chapters has been changed and an additional chapter about hydraulic oils and piping systems has been added too.

Chapter one tackles a brief overview on some hydraulic and pneumatic applications working on hydraulic and pneumatic power. The second chapter starts with fluid mechanics background that discusses the basic principles of Pascal and Bernoulli principles necessary for the fluid power calculations taking into account the pressure losses due to friction effect in pipes and fittings besides to other theories like perfect gas laws and Newton's second law. Chapter three handles the different techniques of modelling used to represent linear and rotary actuators, the control valves and the hydrostatic transmission systems. First and second order models are taken into account including the effect of leakage and compressibility on the general system behaviour. Non-linear modelling is discussed briefly at the end of the chapter. Chapter four discusses some control techniques using servo and proportional valves. Pump operated and valve operated servo control systems are discussed in this chapter besides to block diagrams. Electro-mechanical controls are depicted in chapter five discussing relays, solenoids and voice coil actuators with their applications in proportional control systems, fluid power symbols are discussed at the end of this chapter. Chapter six presents the analysis of selected basic hydraulic circuits with corresponding applications while the pneumatic circuits are discussed and analysed in chapter seven. Chapters six and seven help in understanding the best design techniques for fluid power circuits. The basic principles and designs of different hydraulic and pneumatic components are shown in chapter eight. This chapter uses schematic drawings and real photos to represent the different designs of pumps, compressors, valves and actuators. A new chapter (chapter nine) has been added to this edition of the book to discuss the characteristics and types of hydraulics oils, pipes, hoses and fittings. Finally, general maintenance aspects of hydraulic and pneumatic systems are discussed in chapter ten showing the different problems with their possible causes and remedies.

Ahmed Abu Hanieh

Jerusalem

August 2021

Contents

Chapter 1

Fluid Power Industrial Applications

1.1 Introduction

Fluid power solutions have the greatest contribution in the modern industrial revolution. It is worth mentioning that there is a wide variety of applications for fluid power systems. Most of heavy duty equipment working in industrial sectors depend mainly on either hydraulic or pneumatic principle where the main power source in these systems come from oil in hydraulic systems and from air in pneumatic solutions. The early principles of control depended on using mechanical techniques as the main source of power to introduce force and motion into the required application. Nowadays, most of the fluid power systems are controlled by electrical and digital control techniques which facilitates their use and implementation and gives higher performance to the power systems. The next sections discuss some hydraulic and pneumatic instruments used in industrial applications.

1.2 Hydraulic jack

The simplest hydraulic system is expressed in the hydraulic jack shown in Figure 1.1. This lifting jack is used to lift trucks and vehicles during maintenance and tyre changing process. It consists of a single acting linear actuator supplied by a manual pump and non-return valve. The manual pump is a reciprocating piston pump operated by a manual handle to deliver the oil into the lifting piston. The non-return valve (check valve) is used to prevent oil from returning back to the pump due to the load.

The jack is allowed to retract by turning the adjusting screw that opens the check valve allowing the oil to return back after finishing the required job. The manual pump is sometimes replaced by pneumatic compressed air that serves as power source to exert force on the oil causing air over oil effect.

Figure 1.1: *Hydraulic jack - 20 tons, (Courtesy of Torin)*

1.3 Hydraulic loader

Figure 1.2 shows a picture of a hydraulic loader excavator vehicle used in dig and fill operations of construction. The whole system is driven by an internal combustion engine that provides the main pump with the required rotational speed and torque. The main hydraulic pump is a positive displacement pump (piston, vane or gear pump). The pump's oil delivery is accumulated in the system leading to increase the pressure. To actuate the different degrees of freedom of the loader, the compressed oil is transferred to the linear and rotary actuators through directional and flow control valves.
The loader is moved in two main degrees of freedom; where two hydraulic pistons in parallel are used to lift it up while other two parallel pistons

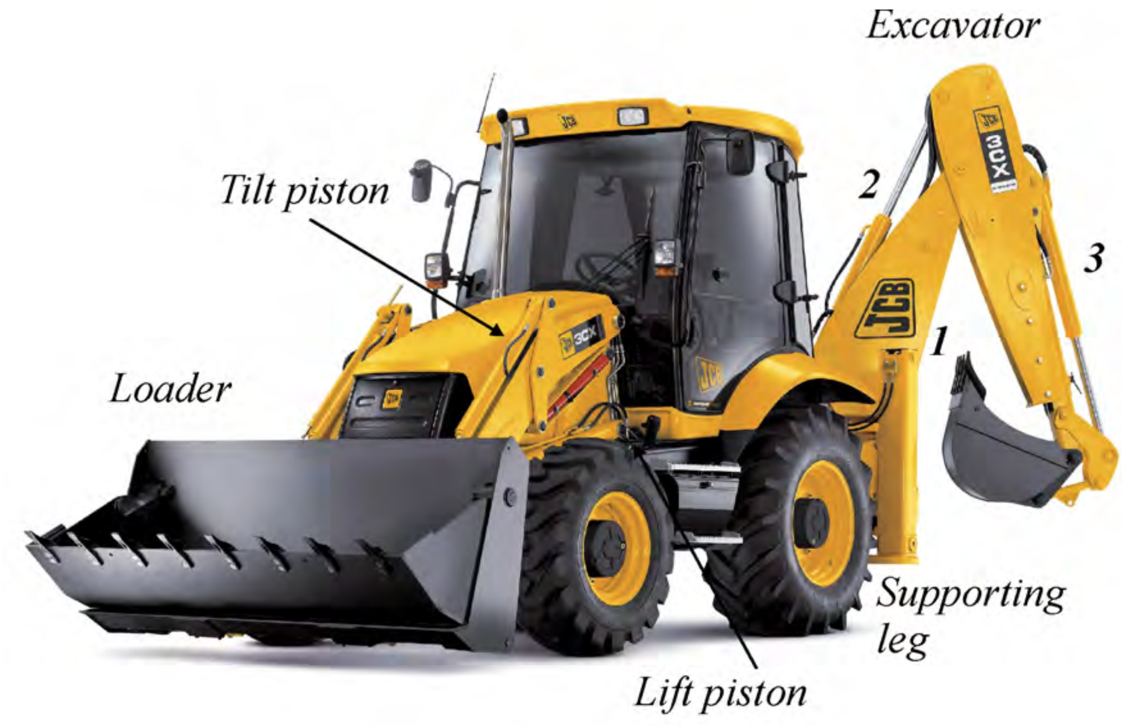

Figure 1.2: *Loader excavator machine (Courtesy of JCB)*

are used to rotate the bucket in a tilting motion. Every couple of pistons are identical and influenced by the same load because they are attached to the same rigid body. Thus, the two pistons can be connected in parallel to double the force in keeping a synchronized motion. The force and displacement of each piston is calculated by estimating the ratio between the action arm (distance between the piston's head and the centre of rotation) and the reaction arm (distance between the bucket and the centre of rotation). To obtain the best synchronization, the diameters of the two pistons should be identical and the forces acting on them are equal in magnitude and direction.

1.4 Backhoe excavator

The backhoe excavator is used in construction works for digging and excavating soil and aggregate. Some vehicles include the backhoe together with the loader like the system shown previously in Figure 1.2, other designs have only excavators driven by the Internal Combustion (IC) engine where the pump is used to supply and move the different motions of the backhoe shown in Figure 1.3.

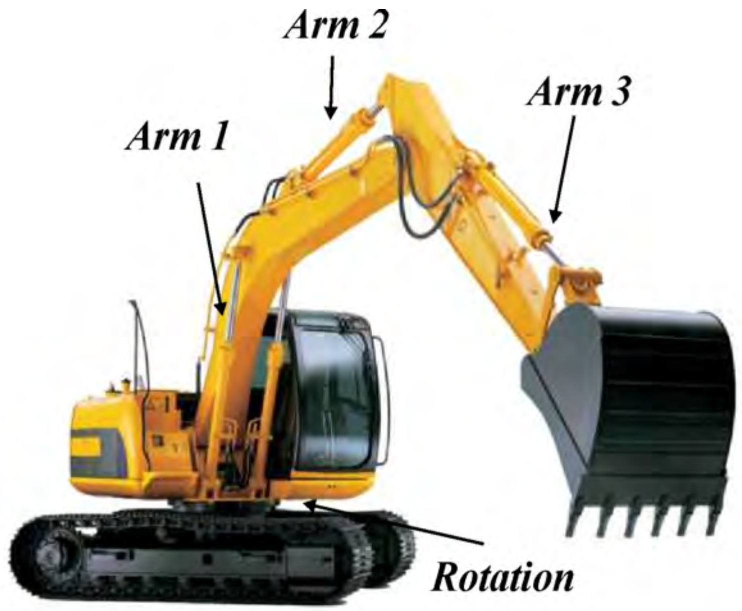

Figure 1.3: *Backhoe design and operation (JCB)*

An excavator has four degrees of freedom where two linear actuators are used to rotate the two arms of the boom, a third piston is used to rotate the end bucket in a tilting manner to fill and empty. The fourth motion is for the rotational steering motion necessary to rotate the whole system in both senses of direction clockwise and counter clockwise. The steering motion can be done either by using a rotary hydraulic motor or by linear actuators connected to rack and pinion gears connected to the main column of the boom.

In most cases, this backhoe is used to operate a jack hammer used to break or crumble hard rocks, asphalt and concrete. The jack hammer is a device that operates by using hydraulic oil and Nitrogen power cell that behaves like an accumulator causing high impulsive force on the head of the tool which can be returned back by means of a strong spring at each impact stroke. Jack hammer is shown in Figure 1.4.

Figure 1.4: *Jack hammer used to break rocks (Courtesy of Beilite)*

1.5 Concrete mixer

Concrete mixer shown in Figure 1.5 consists of a specially designed mixing drum able to rotate around its centre. The rotation is achieved by rotating the main shaft centred in the front side of the drum while the back side has a circular ring simply supported by two freely rotating steel rollers. Spiral blades are welded on the internal surface of the drum to mix the concrete when rotated clockwise and to deliver it out from the back hole when rotated counter clockwise.

Figure 1.5 shows a concrete mixer, the main rotational power is provided by a hydrostatic power transmission system consisting of a swash plate axial piston reciprocating pump and a hydraulic motor. The pump can be coupled directly to the flywheel of an IC engine or by taking power from the vehicle's gearbox through a power take-off instrument (PTO). Most of the hydraulic power is transformed into torque here because the rotational speed of the mixer does not exceed 20 rpm. The rotational speed of the mixer is controlled by both; increasing the rotation speed of the IC engine and changing the angle of the swash plate of the pump.

Figure 1.5: *Concrete mixer (Courtesy of Liebherr)*

1.6 Concrete pump

Concrete pump is considered one of the most important instruments used in heavy duty construction works. The design of this machine depends mainly on the application requirements where it can be mounted on a truck as shown in Figure 1.6 or it can be fixed on the construction site in the case of huge and high buildings. The operating system of this pump can be divided into three parts:

- Safety hydraulic supporting legs.
- Multi-arm extension boom.
- Reciprocating pumping system.

To avoid exerting load on the body and chassis of the truck, four safety hydraulic supporting legs are connected to the chassis of the pump. When these legs are extended, they make the shape of X letter which enables them to react against the moment coming from the boom's weight. Each leg contains two hydraulic pistons; one for extending the leg aside and the other is for lifting the whole system up. It is advised to extend the legs to the maximum distance away from the truck and to adjust their height till reaching a horizontally balanced situation for the pump in all directions. The ready mixed concrete is pumped through pipes attached to the boom arms and articulated by a set of elbows at each joint, to reach to the construction site. These pipes are installed and fixed to the boom that can reach from 18 to 100 meters according to the required distance. The boom consists of three to six articulated arms depending on the design. Each arm of the boom is hinged at the end of the previous one and rotated by a hydraulic linear actuator supported on the previous one causing a relative rotational motion between each two arms. The rotational motion or steering degree of freedom of the complete boom system is operated either by using a rotary hydraulic motor or by a linear actuator with rack and pinion mechanism.

The main system used to pump concrete into pipes is the pumping system shown in Figure 1.7. Pumping system consists of two concrete pistons operated by two hydraulic linear actuators and an S-valve rock connector rotated by reversing pistons. The two pumping pistons are connected in reciprocating manner where the extension stroke of one of them begins at the end of the retraction stroke of the other. The S-valve connector that has the shape of S letter or in other designs it takes the elephant trunk shape or ball valve. It is used to transport the concrete from the pumping piston to the transmission pipe. This valve is rotated by the reversing pistons to be aligned with the pumping pistons in a reciprocating manner. This rotational motion is synchronized with the stroke of the pumping pistons. Limiting the strokes of the pumping actuators and synchronizing them with the reversing pistons are synchronized with the motion of the pumping pistons and can be controlled in one of two ways:

- Mechanical control using pressure relief valves that open and close according to the pressure in the system.

- Electrical control using proximity sensors, relays and solenoids.

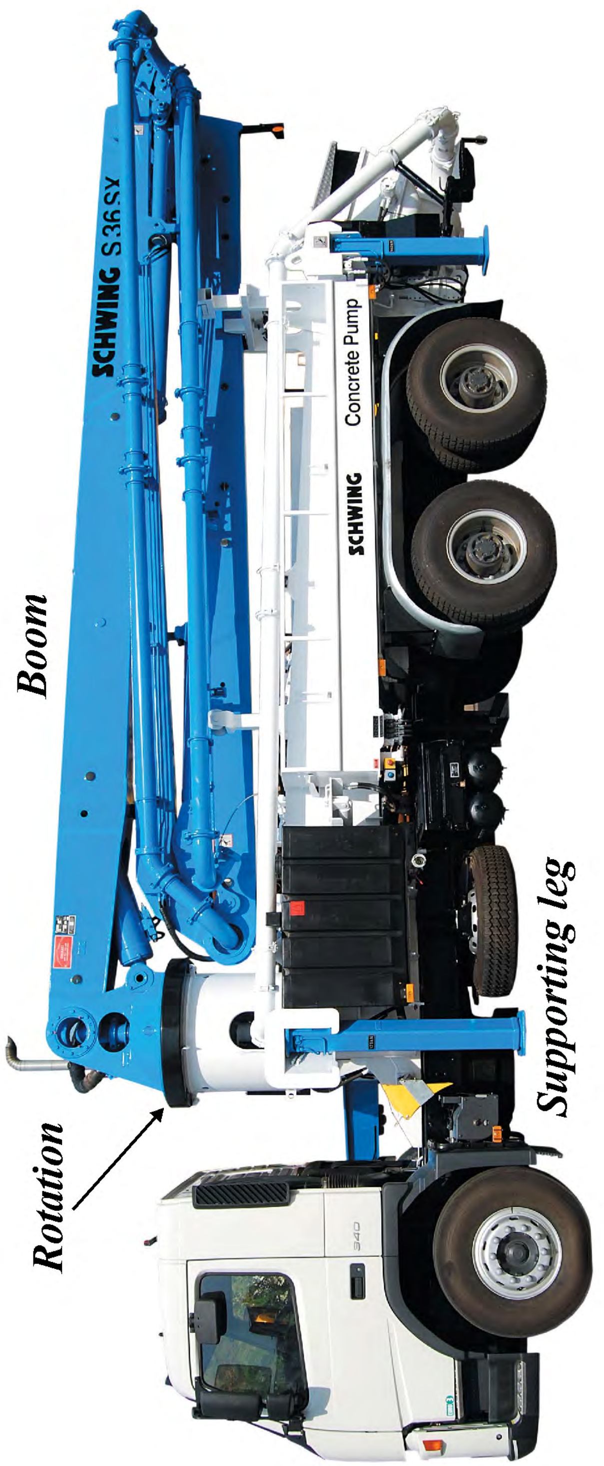

Figure 1.6: *Truck mounted concrete pump (Courtesy of Schwing)*

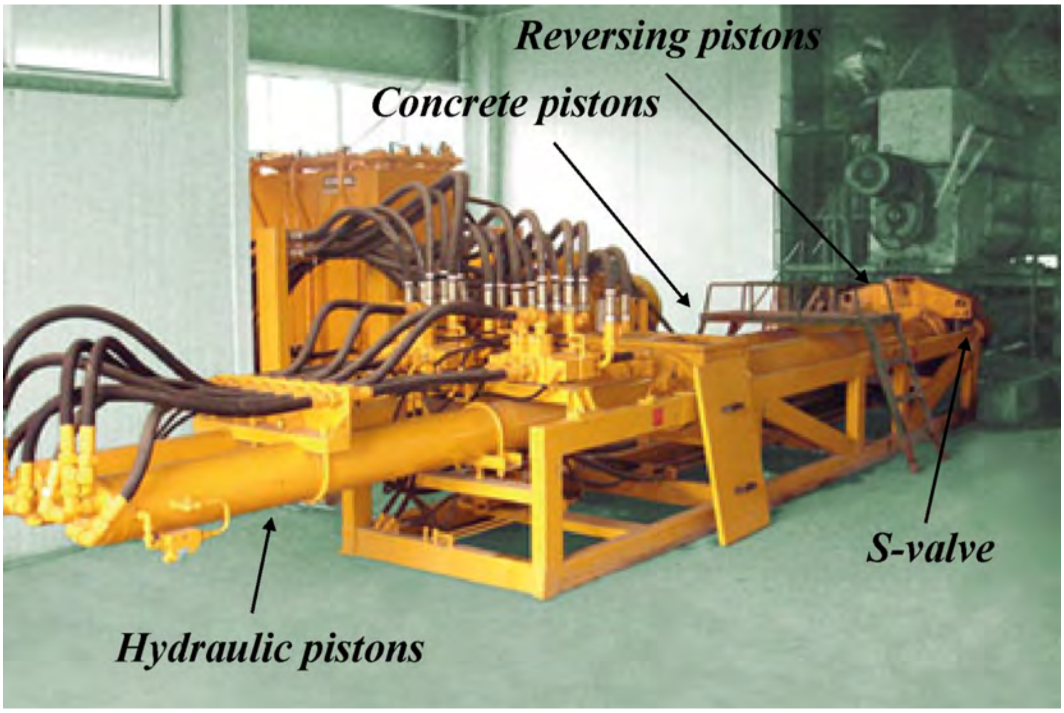

Figure 1.7: *Concrete pumping system (Schwing)*

1.7 Fork lift

Fork lift is a device used to lift loads and containers to transport them from point to point. Figure 1.8 shows a typical fork lift. It functions in two degrees of freedom to complete the lifting operation; lifting up and tilting back. These two motions are operated by two hydraulic pistons. The lifting piston must have a long stroke to enable raising the load to a high place while the tilting piston needs a very short stroke to tilt the load backwards by a maximum angle of 10 degrees to avoid dropping it during motion. Sprockets and chains are sometimes used to extend the stroke of the lifting piston. The driving motion of the whole lift truck is operated by a hydraulic rotary motor connected to the wheels.

1.8 Hydraulic crane

Figure 1.9 shows a telescopic hydraulic crane used to lift and position loads on top of buildings and high places. Cranes are designed according to the

Figure 1.8: *Hydraulic fork lift (Courtesy of Justdial)*

required applications specifying the needed length and lifted weight. Apart from the extension telescopic motion, a crane has two main motions; lifting and rotation. Lifting motion can be operated by a linear hydraulic piston and the rotational motion can be done either by a hydraulic rotary motor or a linear actuator with rack and pinion mechanism. The extension of crane's length is done by a telescopic hydraulic actuator. Cranes need safety hydraulic supporting legs to react against the exerting moments of the boom and load.

1.9 Metal scrap press

Metal scrap collected for recycling processes and old cars take a very large volume and space in storage. To reduce the required storage space, presses like the one shown in Figure 1.10 are used to press the scrap volume to reduced size cubes. Scrap press consists of a U shaped chamber of cast iron with a front covering gate. The upper side (cover) is functioned by a

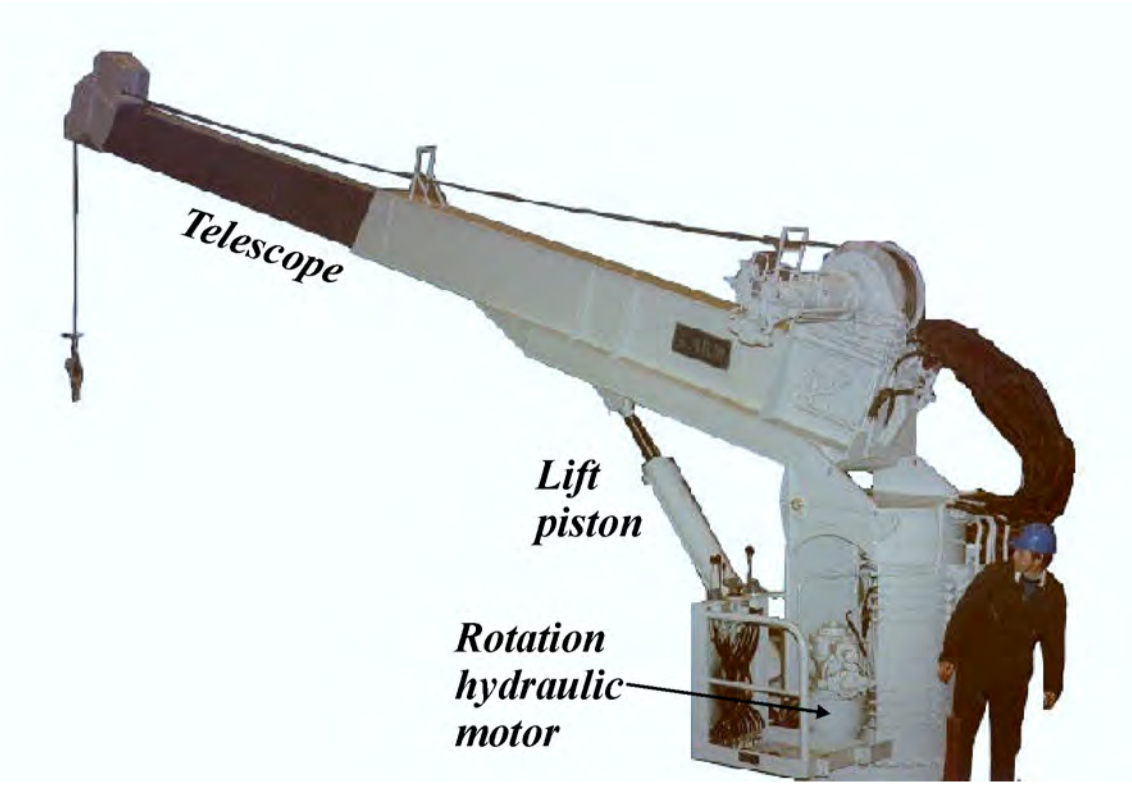

Figure 1.9: *Telescopic hydraulic crane (Courtesy of Karmoy Winches)*

hydraulic piston to press the scrap from the upper side and the back side is functioned by another hydraulic piston to make the final pressing motion converting the scrap into a small cube, the later piston is used expel the cube out of the press. Eventually, the front gate is opened and the back press piston is extended to slide the cube out of the press.

1.10 Leakage testing machine

Production of water plastic bottles by blowing techniques is accompanied by leakage problems according to defects and errors done either by workers or by the machine itself. This leads to the need for a leakage testing machine like the one shown in Figure 1.11. A leakage tester consists of a testing head, a conveyor belt and an ejector. The testing head contains a pipe to blow compressed air into the bottle and another pipe connected to a pressure gauge for measurement. The head is pushed down to cover the bottle by means of a pneumatic piston.

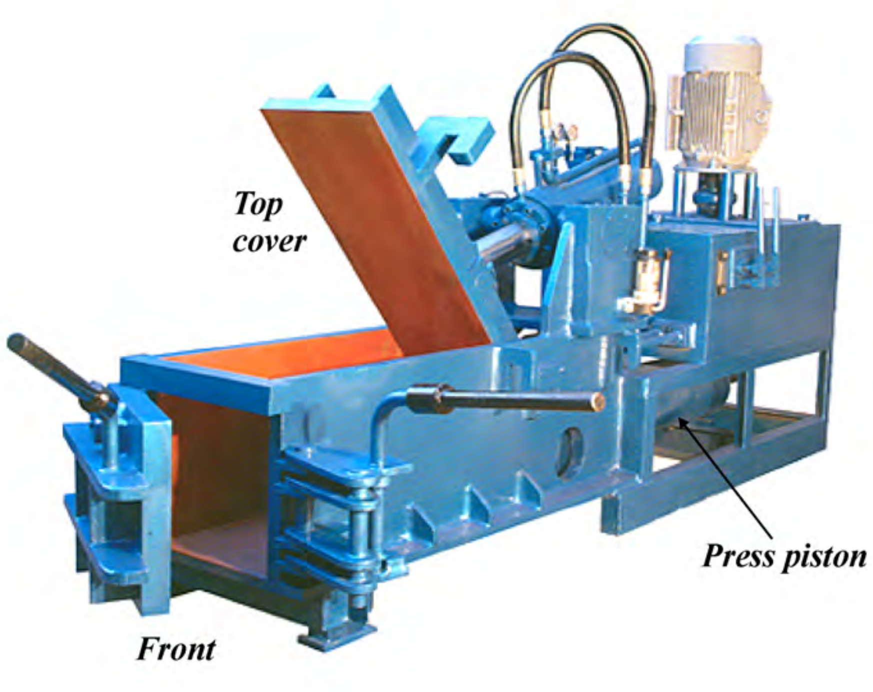

Figure 1.10: *Hydraulic metal scrap press (Courtesy of Advanced Hydrau-Tech PVT)*

Figure 1.11: *Bottle leakage testing machine (Courtesy of Piotech)*

During the testing operation, the blowing pipe blows compressed air inside the bottle till reaching a specific pressure and the pressure gauge measures the pressure difference inside the bottle for a few seconds. If the pressure stays at the same level, the bottle succeeds and passes. Otherwise, if the pressure descends down, this means having a leakage and the bottle fails the test. Failure of the bottle sends a signal to the ejector spring loaded, single acting pneumatic piston to extend and eject the defected bottle away from the line. The conveyor belt moves to approach the next bottle and positions it under the testing head.

1.11 Tyre changing machine

Flat or worn tyres are changed or fixed using tyre changing machine as shown in Figure 1.12. Tyre changing machine contains three main systems:

- Detachment mechanism.
- Clamping table.
- Rolling out head.

The detachment mechanism is a blade operated by a pneumatic piston and used to detach the rubber tyre from the metallic rim on both sides. The clamping table contains four movable jaws actuated by two pneumatic pistons. The table is rotated by an electric or pneumatic motor for the purpose of tyre rolling out. The tyre is rolled out during the rotation of the table using the rolling out head that can be moved manually up and down.

1.12 Pneumatic industrial tools

Rotating hand-held tools can be operated using pneumatic power like the ones shown in Figure 1.13. The cutting tools in these machines and devices are connected to a main rotor that acts as a turbine or a vane in the path of the compressed air. Different speeds and actions can be achieved using air pressure in these pneumatic cutting and industrial tools.

Figure 1.12: *Tyre changing machine (Courtesy of Corghi)*

Figure 1.13: *Different pneumatic tools (courtesy of Industrial Hardware)*

1.13 Problems

1. What is the main function of the check valve in the hydraulic jack?
2. What is the total number of degrees of freedom in a backhoe-excavator-loader machine?
3. What are the sources of vibration disturbance in a truck concrete mixer?
4. Consider a 36 meter long concrete pump, what is the volume of the concrete hopper knowing that the nominal diameter of the concrete pipes is 5.5 inch?
5. Suggest five hydraulic applications other than those mentioned in this chapter.
6. What is the best technique to eject the defective bottle in the plastic bottle leakage tester?
7. What is the type of machine used to extract the tyre from the rim in the heavy duty truck wheels?
8. Explain the principle of intermittent motion in the pneumatic air ratchet device, and what is it used for?
9. Suggest five pneumatic applications other than those mentioned in the chapter.
10. Find an application where hydraulic and pneumatic systems are used together and explain its function.

Chapter 2

Fundamentals of Fluid Power

2.1 Introduction

"*All people are partners in sharing three sources; Water, Food and Energy (Fire)",* said prophet Mohammed more than 1400 years ago [1]. This fact conforms with the basic principle of conducting recent international research about "Water Energy Food Nexus". Man started thinking about different sources of power and energy since the early ages of creation. Fire was the first discovered source of power and energy, then people started using liquids (namely water) that has been replaced later by oil to avoid oxidation and corrosion. It has been found that gases can have similar behaviour under pressure as well. Liquid and gas phases of material have been given the term *Fluid*. Flowing fluid is considered as a power transmission medium because the continuous flow is capable of changing the existing energy from one shape to another. Fluid power is a general term used in mechanical engineering for Hydraulics and Pneumatics . This source of energy proved having higher power than any other power handling method which led engineers to consider it as the best solution for heavy duty works. The overall power in fluids depends mainly on the flow and pressure values. The higher the pressure, the smaller the flow to produce the required power for a specific task. These pressure and flow levels can be obtained by using pumps in the hydraulic system and compressors in the pneumatic system. The most common known pump is the human heart while the human lungs can be considered as the most famous compressor .

Various industrial applications depend more and more on fluid power, it is even logic to say that fluid power touches every part in engineering applications because of its contribution in automation which is becoming essential to increase productivity of different manufacturing and handling processes. Ease of control can be considered as one of the main advantages of fluid power systems where mechanical, electrical and manual controls are applied easily to the various hydraulic and pneumatic systems. These systems have the ability to amplify the force in a high ratio which makes them useful for very high loads and torques. Besides to the fact that they can introduce constant steady forces and torques leading to high stability and performance. Despite simplicity, safety and high power to weight ratio, fluid power systems suffer from some disadvantages like high noise, liquid leakage and exposure to high pressures that can be hazardous to operations and operators [2].

Figure 2.1: *Old manual pump (Courtesy of King Pumps)*

The first invented manual pump was used to pump water out of wells for the purposes of drinking and irrigation. The basic principle of this pump, shown in Figure 2.1, is based on using a reciprocating positive displacement method where a piston slides inside a cylinder changing the volume occupied by the fluid in every stroke back and forth. A non return valve is used to prevent water from returning back when motion is reversed.

2.2 Power transmission methods

Power transmission systems depend mainly on the source of energy used to produce this power, they are divided into mechanical , electromechanical , pneumatic and hydraulic . The influence of these power transmission and handling systems is described as follows:

- To obtain high torque to load inertia or power to weight ratio, hydraulic system is the best solution followed by pneumatics while mechanical and electromechanical systems give worse results.
- The steady state stiffness of hydraulics is higher than that of the mechanical system while in pneumatics and electromechanical systems stiffness is extremely weak.
- The friction level in electromechanical system is much better than all other systems but it is more sensitive to external noise.

The previous discussion shows how much fluid power systems (specially hydraulics) are superior to other power transmission methods but it is worth having a look at a comparison between hydraulics and pneumatics. Table 2.1 shows this comparison [3].

A basic hydraulic system consists mainly of a tank or reservoir for the fluid (oil), a pump , control valves , an actuator (linear or rotary) and a set of pipes as shown in Figure 6.1. The tank is normally open to atmosphere leading to zero gauge pressure of oil at this point, Properties of the tank for best design are:

Table 2.1: *Comparison between hydraulic and pneumatic systems*

Pneumatics	**Hydraulics**
Fluid is compressible (Air)	Fluid is incompressible (Oil)
Relatively low fluid pressure	Very high fluid pressure
Limited dynamic response	Good dynamic response
Delay time of pistons is big	Very smaller delay time
Higher friction due to dryness	Lower friction due to viscous lubrication
No cavitation effect	Exposed to cavitation
Ability of operation at high temperatures	Temperature is limited to oil characteristics

- The tank should have a wide surface area exposed to atmosphere to enable better heat exchange with the surrounding air and reduce the lowering speed of the oil surface during operation.
- The volume of the tank should be at least twice the volume of the whole hydraulic system to compensate leakage and avoid cavitation. This can be done by estimating the total volume of all hydraulic components that include oil (pipes, actuators, valves, ...) and building the tank to contain a quantity of oil double that volume.
- A permanent magnet is usually fixed at the bottom of the tank to collect all metallic chips and prevent them from swimming into the system irritating the moving parts.

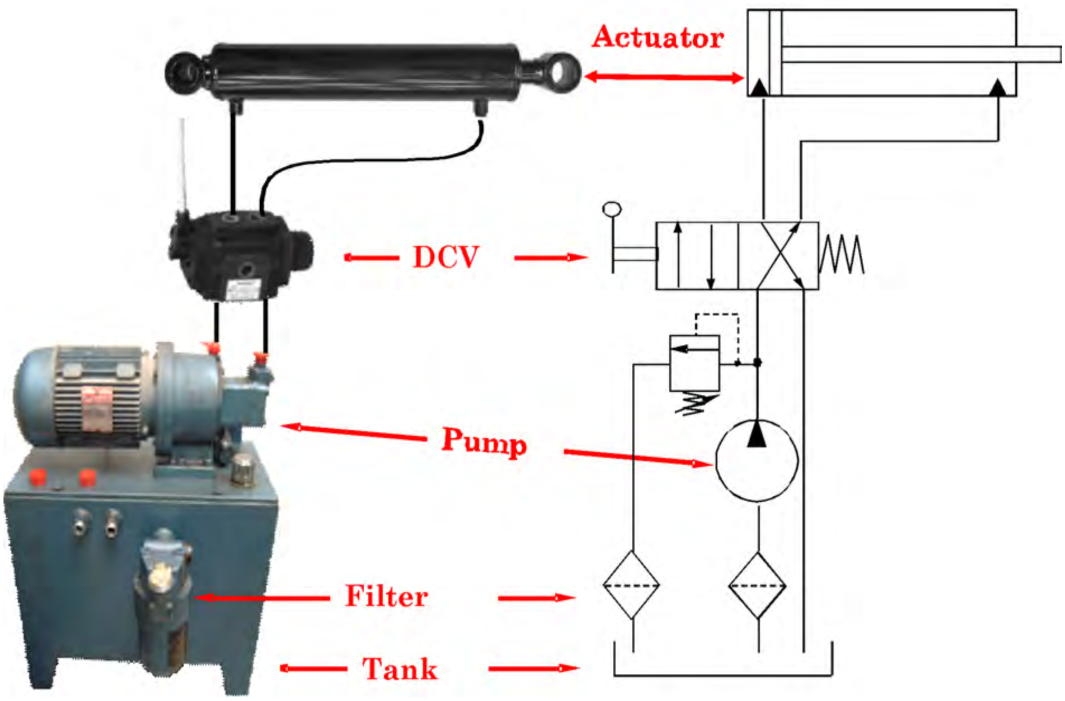

Figure 2.2: *Basic hydraulic system*

The tank feeds oil to the pump through a filter or strainer. The pump delivers fluid continuously at a specific flow rate according to its capacity, the accumulation of this flow inside the pipes leads to increase the pressure in the system when exposed to an external load. A control valve is used here to decide the required operation and direction of fluid. The force produced by the pressurised fluid is used to extend or retract the piston. A basic pneumatic system consists of a compressor , a storage tank for the fluid (air), a Filter Regulator Lubricator (FRL), control valves, an actuator (linear or rotary) and a set of pipes as shown in Figure 7.1.

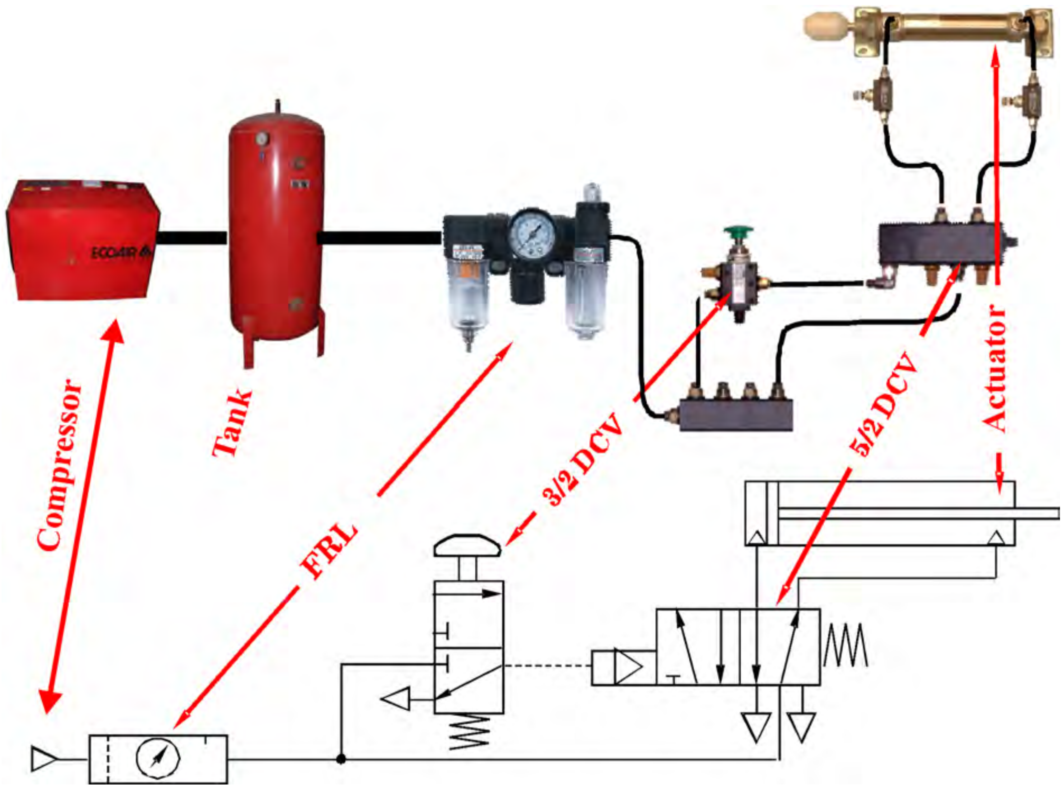

Figure 2.3: *Basic pneumatic system*

The compressor can be single stage or multi-stages, it extracts air from the atmosphere through a filter and compresses it into the storage tank. The pressure inside the tank increases till reaching a specific point where the compressor turns off automatically. The pressurised air flowing in pipes is controlled and directed by means of a control valve that determines the direction and operation of this air that leads, eventually, to extend or retract the actuation piston. The tank should have a draining outlet used to drain the condensed water in the tank that comes in the shape of vapour from atmosphere and condenses due to high pressure. The filter in the

FRL is used to filter the air exiting the tank before reaching to sensitive components, the regulator is a globe valve used to control the compressed air pressure to the required value corresponding to the application and the lubricator is used to spray a small quantity of oil in the compressed air to avoid high dryness and reduce friction in the system.

2.3 Basic theories

Science is becoming multi-disciplinary specially in design problems, where it is impossible to obtain a good design without taking into account all related components. When a system engineer determines to design a fluid power system, he has to take into account the following disciplines:

- *Applied mechanics :* Static, kinematic and dynamic calculations are totally involved here.
- *Vibration analysis:* High pressure and temperature accompanied with harmonic disturbances cause a high level of vibrations that can be catastrophic to the mechanical system. Vibration theory and dynamic modelling helps in obtaining adequate dynamic models for the different fluid power systems.
- *Fluid mechanics :* This is considered the main field to be tackled here as far as the systems depend mainly on the motion of fluids.
- *Thermodynamics :* Heat can have a great influence on these systems specially in hydraulics where high temperature reduces, significantly, the efficiency of hydraulic system by changing oil characteristics.

The fundamental calculations are based on the following basic laws:

- Perfect gas law.
- Newton's laws.
- Torricelli's theorem.
- Pascal's law.
- Bernoulli's equation of energy.

2.3.1 Perfect gas law

In the pneumatic case, the flowing fluid is a gas where there is a high influence of temperature and pressure that causes a change in density and can be governed by the Perfect Gas Law [8]:

$$PV = mRT \tag{2.1}$$

where P is the absolute pressure , T is the absolute temperature in Kelvin and R is the gas constant that can be calculated from the universal gas constant R_u divided by the molecular weight MW of gas:

$$R = \frac{R_u}{MW} \tag{2.2}$$

Knowing that the universal gas constant $R_u = 8315 J/kg.K$ and the molecular weight of air $MW = 28.97$, results in a gas constant value for air equals $R = 287$. Note here that the absolute pressure is calculated by adding the measured gauge pressure to the atmospheric pressure that equals at sea level to 101.3 kPa. and the temperature in Kelvin is calculated by adding the Celsius temperature to 273.15.

As special cases of the perfect gas law, one can express the following:

- If a given constant mass of a gas is compressed or expanded at a constant temperature, the absolute pressure is inversely proportional to the volume which leads to Boyle's law:

$$P_1 V_1 = P_2 V_2 \tag{2.3}$$

- If a given constant mass of a gas changes its temperature under constant pressure, the volume is directly proportional to the temperature leading to Charle's Law :

$$\frac{V_1}{T_1} = \frac{V_2}{T_2} \tag{2.4}$$

- If a given constant mass of a gas changes its temperature under constant volume, the pressure is directly proportional to the temperature leading to Gay Laussac's Law :

$$\frac{P_1}{T_1} = \frac{P_2}{T_2} \tag{2.5}$$

- Combining the three previous laws in equations (2.3), (2.4) and (2.5), results in the most commonly accepted form called the combined gas law :

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} = Constant \tag{2.6}$$

When a system is expanded or compressed at a constant temperature it is said to be $Isothermal$. This is possible only if the gas is allowed to expand freely and slowly without doing external work. In the case of accumulator, rapid expansion or compression is done with no heat losses. This process is called *Adiabatic* and can be determined as follows

$$PV^{\gamma} = K = constant \tag{2.7}$$

where

$$\gamma = \frac{C_p}{C_v}$$

knowing that C_p is the specific heat of fluid at constant pressure and C_v is the specific heat of fluid at constant volume. This leads to the following calculations:

$$\left(\frac{T_1}{T_2}\right) = \left(\frac{V_2}{V_1}\right)^{\gamma-1} = \left(\frac{P_1}{P_2}\right)^{\frac{\gamma-1}{\gamma}} \tag{2.8}$$

or

$$\left(\frac{P_1}{P_2}\right) = \left(\frac{V_2}{V_1}\right)^{\gamma} = \left(\frac{T_1}{T_2}\right)^{\frac{\gamma-1}{\gamma}} \tag{2.9}$$

or

$$\left(\frac{V_1}{V_2}\right) = \left(\frac{P_2}{P_1}\right)^{\frac{1}{\gamma}} = \left(\frac{T_2}{T_1}\right)^{\frac{1}{\gamma-1}} \tag{2.10}$$

Example 2.1

A pneumatic air reservoir with a capacity of 150 liter is filled with a compressed air at a gauge pressure of 900 kPa at a temperature of 45^oC. The air is cooled to a temperature of 20^oC. Determine the final pressure in the reservoir.

Solution

To calculate the absolute pressure and temperature
$P_1 = 900 + 101.3 = 1001.3$ kPa
$T_1 = 45 + 273.15 = 318.15$ K
$T_2 = 20 + 273.15 = 293.15$ K

The combined gas law in equation (2.6) can be applied here, but the volume is constant, then equation (2.5) can also be applied:

$$P_2 = P_1\frac{T_2}{T_1} = 1001.3\frac{293.15}{318.15} = 922.6 \text{ kPa (absolute)}$$

2.3.2 Newton's laws of motion

Newton's second law of motion states that the induced force F is directly proportional to the acceleration a of a moving material. The proportionality constant is defined as the mass m of the matter or the mass moment of inertia when the motion is a rotational one [4], [5]:

$$F = ma = m\frac{dv}{dt} \tag{2.11}$$

where v is the velocity of the matter. It is known that the mass is defined by the multiplication between the density ρ and the volume V where, $m = \rho V$. This leads to the equation:

$$F = \rho V \frac{dv}{dt} \quad or \quad F = \rho \frac{dV}{dt} v \tag{2.12}$$

The rate of change of the volume is equal to the fluid flow rate Q which leads to the equation

$$F = \rho Q v \tag{2.13}$$

The foregoing discussion is applied directly in the case of flowing liquid from a liquid jet or nozzle exerting a force on a body.

Example 2.2

A 100 litre per minute water jet is used to raise a box with 25 kg mass. Determine the diameter of the nozzle.

Solution

The flow rate of the jet

$$Q = 100 \frac{l}{min} \times \frac{m^3}{1000l} \times \frac{min}{60sec} = 0.00167 m^3/s$$

The force exerted by the 25 kg box equals

$$F = 25 \times 9.81 = 245.25N$$

The velocity of the water is

$$v = \frac{F}{\rho Q} = \frac{245.25}{1000 \times 0.00167} = 146.8 m/s$$

The nozzle's area is

$$A = \frac{Q}{v} = \frac{0.00167}{146.8} = 1.13 \times 10^{-5}$$

The diameter is

$$D = \sqrt{\frac{4 \times A}{\pi}} = 0.0038m = 3.8mm$$

2.3.3 Torricelli's theorem

Evangelista Torricelli who lived from 1608 to 1647 put forward a theorem stating that; the velocity of a free jet of a fluid is proportional to the square root of the head producing the jet [6]. This theorem is demonstrated in Figure 2.4. Having the same atmospheric pressure, energy conservation can be applied between points A and B.

$$\frac{1}{2}mv^2 = mgh \tag{2.14}$$

$$v = \sqrt{2gh} \tag{2.15}$$

This leads to the direct relationship between the velocity v and the head h, where the head is related to the liquid pressure P by the relation $h = P/\rho g$.

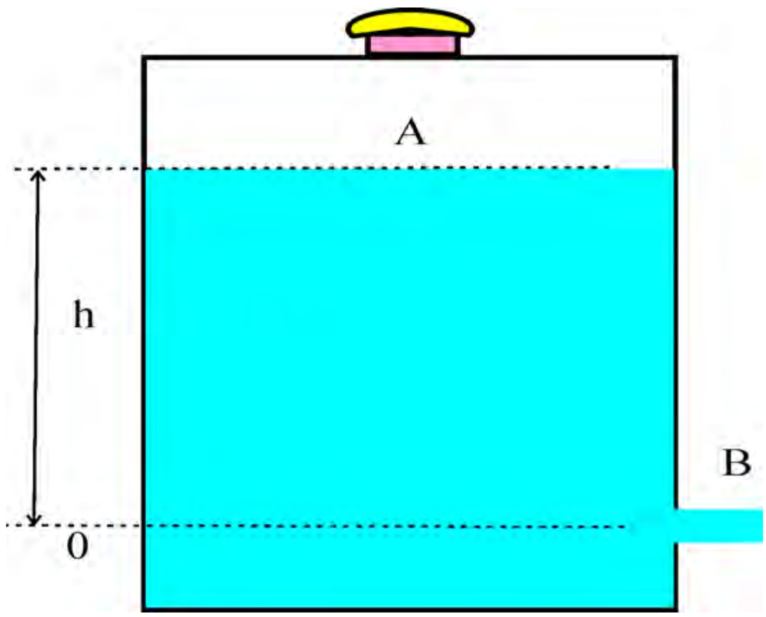

Figure 2.4: *Torricelli's theorem*

Example 2.3

A 100 litre tank with a 50 cm diameter circular cross section provides the hydraulic pump with a hydraulic oil. Determine the speed and flow rate of the oil flowing into the pump's inlet.

Solution
The height of the hydraulic tank is

$$h = \frac{volume}{Area} = \frac{100 \times 10^{-3}}{\pi(0.25)^2} = 0.51m$$

The velocity of oil exiting the tank and entering the pump

$$v = \sqrt{2gh} = \sqrt{2 \times 9.81 \times 0.51} = 10m/s$$

2.3.4 Pascal's law

The principle of power transmitted by fluid has been put forward by Blaise Pascal, [7]. Power transmission here is similar to power transmission in a human arm muscle or a lever arm where the multiplication of the action force by its arm should overcome the multiplication of the reaction force by its arm to cause motion.

Pascal's theorem states that:

- Fluid pressure has the same value throughout an enclosed fluid in a vessel.
- Pressure acts equally in all directions at the same time.
- Pressure acts at right angle to any surface in contact with the fluid.

The main idea is shown in Figure 2.5. The system consists of a force cylinder with an area a and an acting force F that causes the piston to move a distance l called the pumping arm, it is connected to a load cylinder with an area A and a supported load W that causes this piston to move a distance L called the load ram. The pressure of fluid is calculated as follows:

$$P = \frac{F}{a} = \frac{W}{A} \tag{2.16}$$

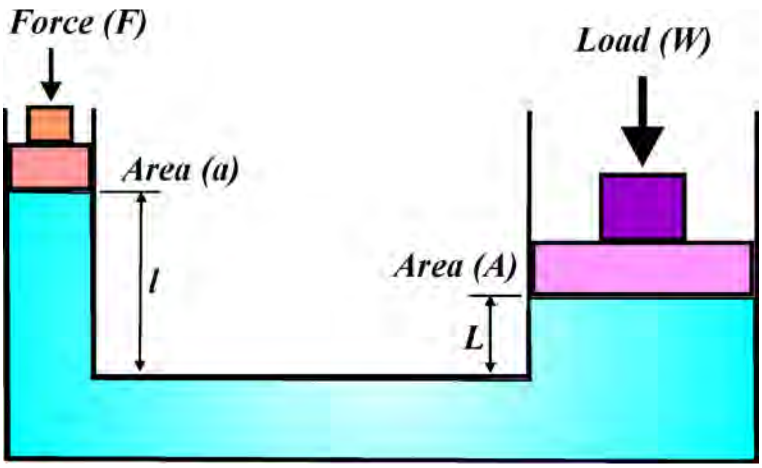

Figure 2.5: *Pascal's Theorem*

or taking the force ratio as

$$\frac{W}{F} = \frac{A}{a} \tag{2.17}$$

Example 2.4

Consider the hydraulic piston shown in Figure 2.6 used to push a load of 1000 N. The piston is actuated by a manual pump that compresses a fluid to the piston through a pipe with inside diameter of 10 mm. The diameter of the cylinder is 100 mm and the diameter of the rod is 40 mm. Determine the human force needed to act on the piston
a- in Extension stroke.
b- in Retraction stroke.

Solution
The pressure is assumed to have the same value in pipe and cylinder. The area of the pipe is

$$A_{pipe} = \frac{\pi}{4}(0.01)^2 = 0.0000785m^2$$

a- For the extension stroke, the area of the piston side A_p is

$$A_p = \frac{\pi}{4}(0.1)^2 = 0.00785m^2$$

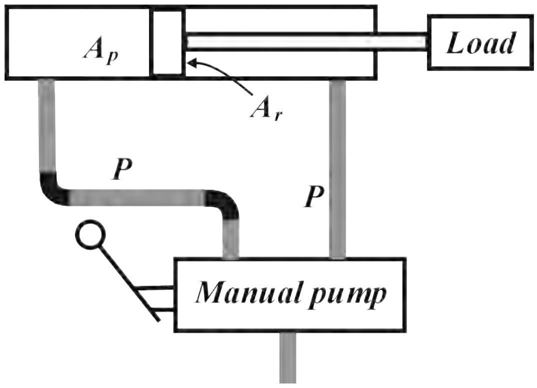

Figure 2.6: *A hydraulic piston, for example 2.4*

The hand force F_{hand} needed to act on the manual handle in this case is

$$F_{hand} = \frac{A_{pipe}}{A_p}(Load) = \frac{0.0000785}{0.00785}(1000) = 10N$$

b- For the retraction stroke, the area of the rod side A_r is

$$A_r = \frac{\pi}{4}[(0.1)^2 - (0.04)^2] = 0.006594m^2$$

The hand force F_{hand} needed to act on the manual handle in this case is

$$F_{hand} = \frac{A_{pipe}}{A_r}(Load) = \frac{0.0000785}{0.006594}(1000) = 11.9N$$

Assuming incompressible fluid, the displaced volume V is calculated from $V = AL = al$ and the total work is calculated from

$$Work = PV = PAL \tag{2.18}$$

The SI units for the Work are
$Work = P(N/m^2).V(m^3) = N.m$

The mechanical Work is estimated by the multiplication of the force and the displacement resulting in an acting force on the left hand side $F = PA$ and a resulting force on the load $W = PA$. It is clear that the magnification of the force depends on the area ratio A/a.
On the other hand, the hydraulic power is equal to $Power = PQ$ where Q is the volume flow per unit time $Q = V/time$. The SI units for the

hydraulic power are
$Power = P(N/m^2).Q(m^3/s) = N.m/s = Watt$

In practical application, this theorem is applied to the fluid power cylinder (linear actuator) where the commercially used units are based on expressing the flow rate in litre per minutes (l/min) and pressure in (bar)

$$Q(l/min) = \frac{Q}{60}(l/s) = \frac{Q}{60 \times 10^3}(m^3/s)$$

$$P(bar) = P \times 10^5(N/m^2)$$

The hydraulic power

$$Power = Q(l/min)(\frac{1}{60 \times 10^3})(m^3/s) \times P(bar)(1 \times 10^5)(N/m^2)$$

$$= \frac{QP}{600}10^3(N.m/s) = \frac{QP}{600}10^3(Watts)$$

$$= \frac{Q(l/min) \times P(bar)}{600} = Power(KW)$$

Example 2.5

A hydraulic pump has a flow rate of 60 l/min and capable of operating a pressure of 20 bar as a maximum load. What is the power of this pump.

Solution
The pumps flow rate in standard units

$$Q = 60\frac{l}{min}\frac{m^3}{1000l}\frac{min}{60sec} = 0.001m^3/sec$$

The pressure in standard units

$$P = 20 \times 1.013 \times 10^5 = 2.026 \times 10^6 Pa$$

The Power is

$$Power = P \times Q = 0.001 \times 2.026 \times 10^6 = 2.026 \times 10^3 = 2.026kW$$

Or simply

$$Power = \frac{Q(l/min) \times P(bar)}{600} = \frac{60 \times 20}{600} = 2kW$$

2.3.5 Bernoulli's equation of energy

Daniel Bernoulli who lived between 1700 to 1782 developed the concept of energy conservation for fluids flowing in a pipe with changing cross-sectional area and connecting between two different levels as shown in Figure 2.7. Conservation of energy concept means that energy can neither be created nor destroyed. When a pipe is installed between different elevations, the total energy of fluid (kinetic , potential , hydraulic and losses) is conserved throughout the pipe [6].

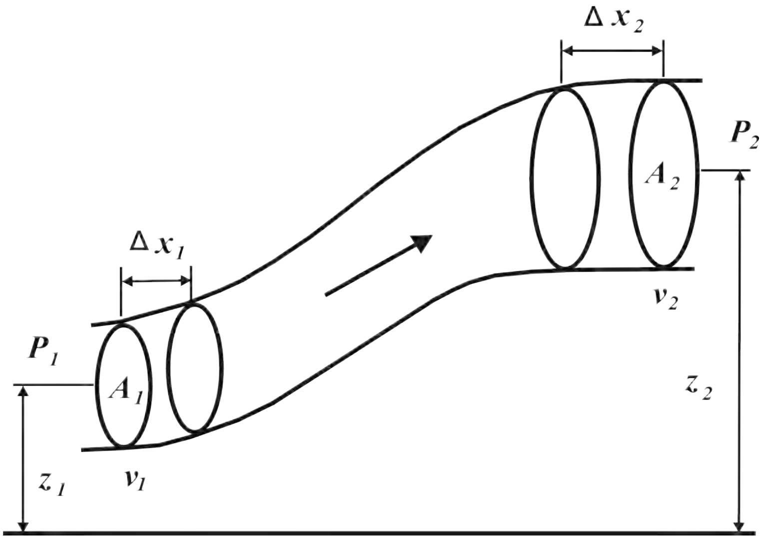

Figure 2.7: *Bernoulli's Theorem*

Consider an incompressible, nonviscous, pressurized fluid flowing in a pipe from point (1) at an elevation z_1 with a cross-sectional area A_1 and a pressure P_1 to point (2) at an elevation z_2 with a cross-sectional area A_2 and a pressure P_2. The force at the input of the pipe is the multiplication of P_1 and A_1 and the volume of fluid pushed to the pipe a displacement Δx_1 is the multiplication of A_1 and Δx_1 [7]:

$$\Delta V_1 = A_1 \Delta x_1$$

Similarly, the volume of fluid reaching at the exit of the pipe is

$$\Delta V_2 = A_2 \Delta x_2$$

The incompressibility of the fluid means that the volume remains the same throughout,

$$\Delta V_1 = \Delta V_2 = \Delta V$$

Hence, calculating the work done at the inlet and outlet of the pipe

$$Work_{in} = P_1 A_1 \Delta x_1 = P_1 \Delta V$$

$$Work_{out} = P_2 A_2 \Delta x_2 = P_2 \Delta V$$

The change in kinetic energy between the inlet and outlet is given by

$$\Delta KE = \frac{1}{2}\Delta m_2 v_2^2 - \frac{1}{2}\Delta m_1 v_1^2 \tag{2.19}$$

where the mass at the inlet is

$$\Delta m_1 = \rho A_1 \Delta x_1$$

Similarly, the mass of fluid reaching at the exit of the pipe is

$$\Delta m_2 = \rho A_2 \Delta x_2$$

The conservation of mass (no leakage) leads to the fact that the mass remains the same throughout the pipe,

$$\Delta m_1 = \Delta m_2 = \Delta m$$

The change of potential energy due to weight as a function of the height measured from an inertial reference reads

$$\Delta PE = \Delta m_2 g z_2 - \Delta m_1 g z_1 = \Delta m g(z_2 - z_1) \tag{2.20}$$

Applying the rule of conservation of energy

$$\text{Work in} = \text{Kinetic Energy} + \text{Potential Energy} + \text{Work out}$$

$$P_1 \Delta V = \Delta KE + \Delta PE + P_2 \Delta V$$

$$P_1 \Delta V = \frac{1}{2}\Delta m(v_2^2 - v_1^2) + \Delta m g(z_2 - z_1) + P_2 \Delta V \tag{2.21}$$

But it is known that $\Delta m/\Delta V = \rho$, thus, dividing equation (2.21) by ΔV results in

$$P_1 = P_2 + \frac{1}{2}\rho v_2^2 - \frac{1}{2}\rho v_1^2 + \rho g z_2 - \rho g z_1 \tag{2.22}$$

On the other hand, dividing equation (2.21) by Δt, knowing that $\Delta V/\Delta t = Q$ and adding the effect of friction losses leads to the equation

$$P_1 Q = \frac{1}{2}\rho Q(v_2^2 - v_1^2) + \rho g Q(z_2 - z_1) + P_2 Q + \textit{friction losses} \tag{2.23}$$

Eventually, equation (2.22) can be divided by the quantity ρg that defines the specific weight of the fluid γ. This leads to the general form of Bernoulli's equation. For hydraulic applications, the pump head H_p, motor head H_m and friction losses head H_L are added to the equation to form the general modified form of Bernoulli's equation

$$z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g} \tag{2.24}$$

Example 2.6

Consider the hydraulic system shown in Figure 2.8, the system consists of a hydraulic pump fed by an oil tank through a filter or strainer. The pump delivers a flow of 120 l/min adding a power of 3.7 KW. The pipe has a 25 mm inside diameter and transports oil with specific gravity 0.9. The oil moves a hydraulic motor and returns back to the tank. considering the portion of the system between point (1) on the oil surface in the tank and point (2) at a height of 6 m on the inlet of the hydraulic motor, dertermine the pressure at point (2) knowing that the head loss H_L between points 1 and 2 is 9 m of oil.

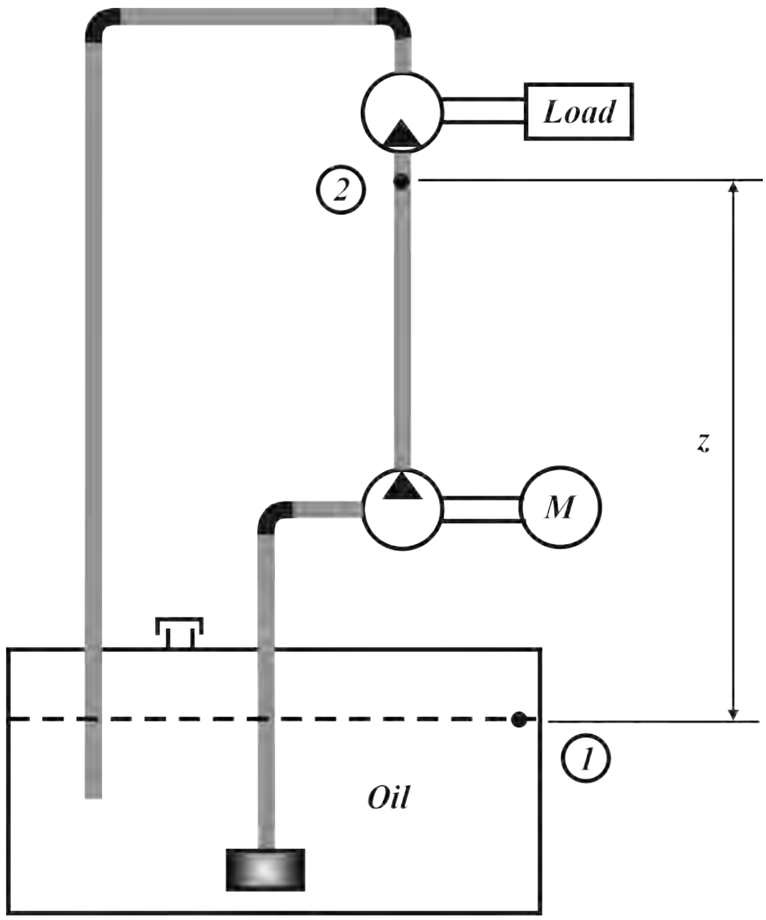

Figure 2.8: *Hydraulic system for example 2.6*

Solution

Applying Bernoulli's equation of conservation of energy

$$z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

Looking at the different terms of the equation

- The hydraulic motor is not included in the calculations, so $H_m = 0$.
- The oil surface is considered large enough and its motion is negligible, so $v_1 = 0$.
- The oil tank is open to the atmosphere, thus, gauge pressure $P_1 = 0$.
- The reference of height is taken at point 1, so the head of point 1 $z_1 = 0$.
- The head of point 2 is given as $z_2 = 6m$.
- The head of friction losses in pipes is given by $H_L = 9m$.

Substituting the previous values in Bernoulli's equation

$$0 + 0 + 0 + H_p - 0 - 9 = 6 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

This leads to

$$\frac{P_2}{\gamma} = H_p - \frac{v_2^2}{2g} - 15$$

Calculating γ

$$\gamma = S.G\gamma_{water}$$

$$\gamma = (0.9)(9800) = 8820 \ kg/m^2s^2$$

The flow rate

$$Q = 120(l/min)(min/60s)(m^3/1000l) = 0.002 \ m^3/s$$

The head of the pump

$$H_p = \frac{Power}{\gamma Q}$$

$$H_p = \frac{3.7 \times 10^3}{(8820)(0.002)} = 209.7\ m$$

The inside area of the pipe

$$A = \frac{\pi}{4}D^2 = \frac{\pi}{4}(0.025)^2 = 0.00049\ m^2$$

The velocity at point 2

$$v_2 = \frac{Q}{A} = \frac{0.002}{0.00049} = 4.1\ m/s$$

The velocity effect on Bernoulli's equation

$$\frac{v_2^2}{2g} = \frac{(4.1)^2}{2 \times 9.8} = 0.85\ m$$

Substituting again in the equation of energy

$$\frac{P_2}{\gamma} = 209.7 - 0.85 - 15 = 193.85\ m$$

Finally, solving for the pressure P_2

$$P_2 = (193.85)(8820) = 1.7 \times 10^6\ Pa = 1.7\ MPa$$

2.4 Friction losses

2.4.1 Friction losses in hydraulic systems

Figure 2.9 shows a general hydraulic system consists mainly of a hydraulic double acting hydraulic piston driving a load. The pump is driven by an electric motor and the pump's suction line takes the oil from the reservoir after being filtered by a filter or strainer. A check (non return) valve is installed between the pump and the tank to prevent oil from returning

back to the tank in stall conditions. The direction and speed of fluid flow is controlled by the directional control valve. This operation involves a mechanical energy coming in to the system from the electric motor, a mechanical energy going out of the system by the hydraulic piston and a heat energy dissipated from the system due to friction losses. As reducing heat losses needs expensive changes in different parts, designers try to find a compromise between loss reduction and high costs [2].
The dissipated heat caused by friction between fluid and pipe depends on

- The roughness of the path: the more tortous the path, the greater the losses.
- The pipe dimensions: the smaller the pipe diameter and the longer the pipe, the greater the losses.
- The viscosity of the fluid: the higher the fluid viscosity, the greater the losses

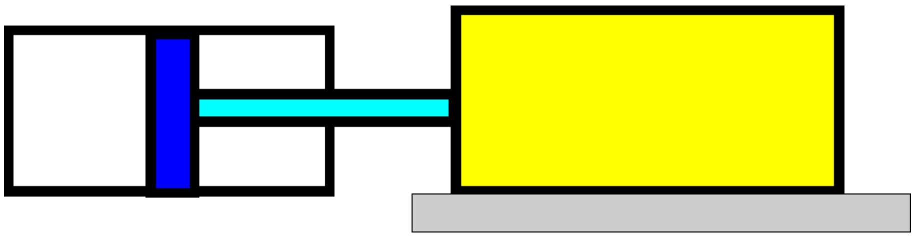

Figure 2.9: *General hydraulic system with linear actuator*

Reynolds number

When a fluid passes through a pipe, the velocity of fluid layer near the walls of the cylinder reaches zero while the maximum velocity occurs at the greatest distance from the wall which is at the centreline of the pipe. There are three shapes of fluid flow; laminar, transient and turbulent. The determination of the type of flow depends mainly on the fluid velocity and viscosity from one side and the size and shape of the pipe from the other side. The most common indicator used to determine the type of flow is *Reynolds number* (Re). Osborn Reynolds in 1833 carried out a set of experiments in which he found that the nature of flow depends on a dimensionless number that he called *Reynolds number* (Re). The value of Reynolds number varies as follows:

$$2000 \leq Re \leq 4000 \textit{ for smooth surface (new pipe)}$$

$$1200 \leq Re \leq 2500 \textit{ for corrugated surface (old pipe)}$$

Reynolds number can be calculated from equation 2.25:

$$Re = \frac{vD}{\nu} = \frac{vD\rho}{\mu} \tag{2.25}$$

where v is the fluid velocity, ρ is the fluid density, D is the hydraulic diameter of the pipe, ν is the kinematic viscosity and μ is the dynamic viscosity.
The pipe's cross sectional shape is not always circular, it can have any shape. Thus, the hydraulic diameter is calculated by

$$D = \frac{4 \times \textit{flow section area}}{\textit{flow section perimeter}} \tag{2.26}$$

The viscosity of a fluid is the measure of its resistance to flow. High viscosity of fluid leads to energy dissipation during fluid flow. Dynamic viscosity μ is the ratio between the shear stress and the shear rate. Kinematic viscosity ν can be calculated from the dynamic viscosiy

$$\nu = \frac{\mu}{\rho} \tag{2.27}$$

The dimension of ν is given in Stoke St or m^2/s, where $1St = cm^2/s = 10^{-4}m^2/s$. In smaller units, $1cSt = 10^{-6}m^2/s$. Similarly, dimension of μ is given by $Poise$, where $1kg/m.s = 10Poise$. In smaller units, $1Poise = 1g/cm.s$.

Example 2.7

Consider an oil flowing at a rate of 1.6 l/s in an equilateral triangular pipe with a side $y = 15mm$. If the kinematic viscosity of the oil is 35 cSt, find Reynolds number and discuss the results.

Solution

$$Cross\ sectional\ area = \frac{y\sqrt{y^2 - (y/2)^2}}{2} = 9.743 \times 10^{-5} m^2$$

$$Perimeter = 3y = 0.045m$$

$$D = \frac{4 \times 9.743 \times 10^{-5}}{0.045} = 0.00866m$$

$$v = \frac{Q}{A} = \frac{1.6 \times 10^{-3}}{9.743 \times 10^{-5}} = 164.2m/s$$

$$Re = \frac{164.2 \times 0.00866}{35 \times 10^{-6}} = 4063.3$$

The value of Reynolds numbers shows that the flow is turbulent which implies the necessity to increase the pipe diameter in order to have a laminar flow to avoid pressure drop and high noise and to reduce energy dissipation.

Darcy's equation

The main cause of head loss in fluid power systems is friction. The friction losses are dissipated to the environment in a shape of heat energy due to direct contact between fluid and pipes or fittings . these losses are transformed into head H_L that can be calculated in pipes, for both laminar and turbulent flow, from Darcy's equation

$$H_L = f(\frac{L}{D})(\frac{v^2}{2g}) \tag{2.28}$$

where
f = friction factor (dimensionless)
L = length of pipe (m, ft)
D = inside diameter of pipe (m, ft)
v = fluid velocity (m/s, ft/s)
g = gravitational acceleration (m/s^2, ft/s^2)

Friction factor f for laminar flow can be calculated by

$$f = \frac{64}{Re} \tag{2.29}$$

This results in calculating the head loss for laminar flow as follows

$$H_L = \frac{64}{Re}(\frac{L}{D})(\frac{v^2}{2g}) \tag{2.30}$$

friction factor f for turbulent flow can be taken from Moody diagram shown in Figure 2.10.

Moody diagram shows that the friction factor f can be picked out from one of the curves depending on Reynold's number from one side and the relative roughness from another side. Relative roughness can be estimated by

$$Relative\ Roughness = \frac{\epsilon}{D} \tag{2.31}$$

where D is the inside diameter of the pipe and ϵ is the height of the insdie corrugation in the pipe. ϵ values for different materials are given in the table included in Figure 2.10. Moody diagram has the following charcteristics:

- Moody diagram is plotted on a log-log scale because of high differences in values.

- The transient flow where Reynold's number is between 2000 and 4000 has no clear values because flow is not possibly predicted.

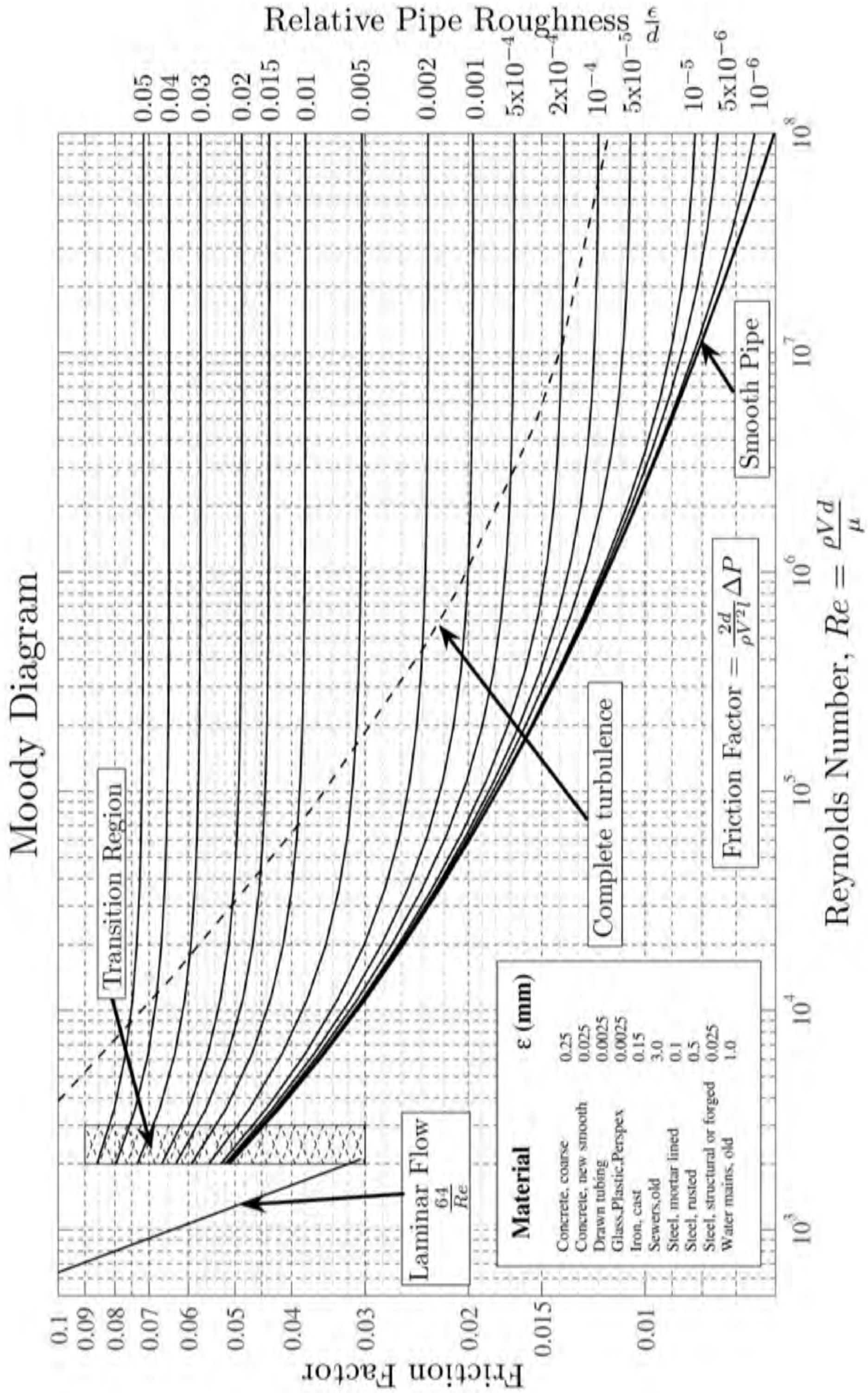

Figure 2.10: *Moody Diagram*

- At low Reynold's number values (less than 2000), flow is laminar, thus, friction coefficient is a constant value $f = 64/Re$.
- At high Reynold's number values (more than 4000), flow is turbulent, thus, friction coefficient is picked out of the curve where the values of Re and ϵ/D intersect. Interpolation is needed for further accuracy.

Valves and Fittings

It is almost impossible to have a fluid power circuit without valves or fittings. Elbows , tees and return bends are the most common fittings needed to branch or direct a pipeline. On the other hand, globe valves, gate valves, check valves and directional control valves have high influence on the head losses in the system. To include the effect of these valves and fittings in the head loss of the system, equation (2.28) has been truncated into the following equation

$$H_L = K\frac{v^2}{2g} \tag{2.32}$$

where K is the constant of proportionality or K factor. It is clear from the relationship between equation (2.32) and equation (2.28) that the ratio L/D has been set to a unit and the main factor influencing here is K. The proportionality constant K has been determined experimentally and is given in Table 2.2 for the different fittings shown in Figure 2.11.

In the case of specific job valves like directional control valves and different types of spool valves, empirical curves are plotted for each valve by the manufacturer depending on the design and characteristics of the valve itself. Figure 2.12 shows a cutaway of a spool directional control valve. Figure 2.13 shows a set of curves representing pressure losses in spool valve. These curves give the relationship between the flow rate passing through the valve and the pressure drop across the valve due to friction and flow resistance.

Table 2.2: *K factor for different fittings and valves*

Fitting	**K factor**
Globe valve	
Wide open	10.0
Half open	12.5
Gate valve	
Wide open	0.19
3/4 open	0.9
1/2 open	4.5
1/4 open	24
Return bend	2.2
Standard tee	1.8
Standard elbow	0.9
45^o elbow	0.42
90^o elbow	0.75
Check valve	4.0

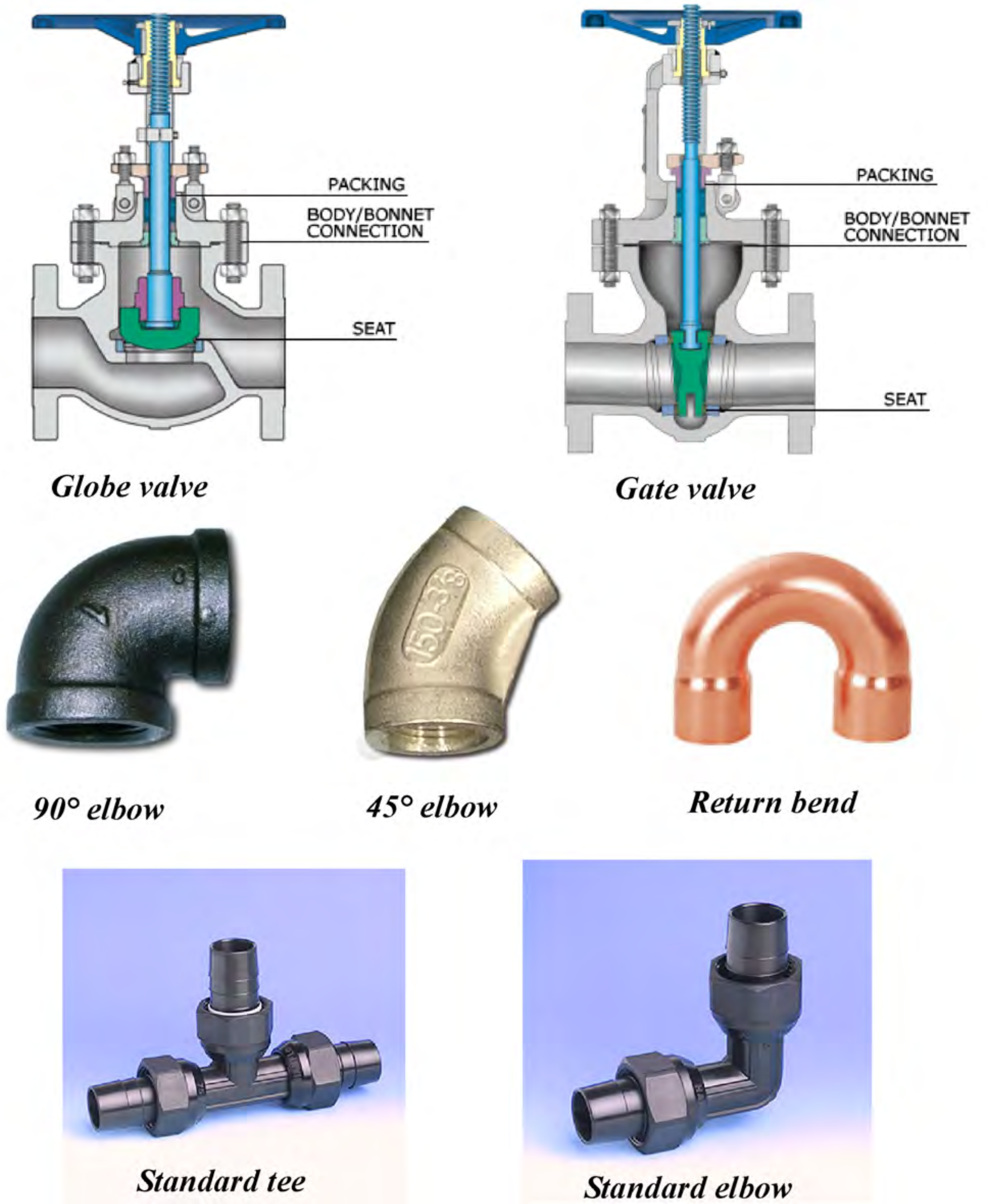

Figure 2.11: *Fittings and valves (Courtesy of PAR and Y.T Metals)*

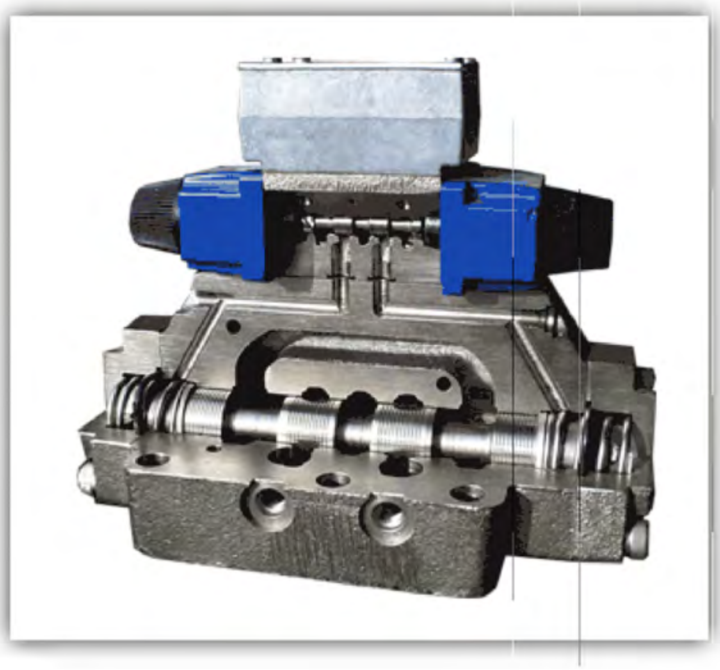

Figure 2.12: *Cutaway of directional control valve (Courtesy of Hoyen Inc.)*

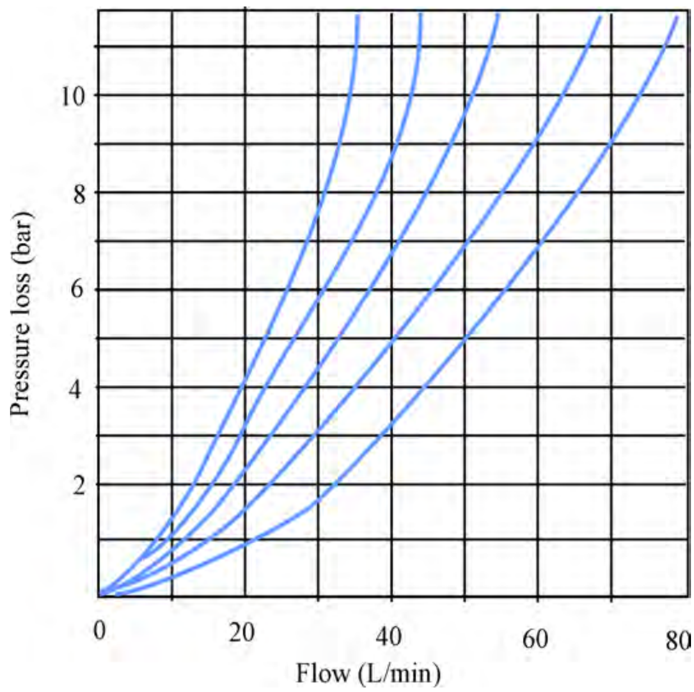

Figure 2.13: *Pressure loss versus flow rate in a directional control valve*

Equivalent length

Valves and fittings are usually installed in a fluid power circuit and connected to pipes with specific diameter D. An easy way to estimate the head loss of valves and fittings is to assume replacing them with a portion of pipe having the same diameter D and an equivalent length L_e. This can be done by assuming the same losses in fittings and equivalent pipe

$$H_{L(fitting)} = H_{L(pipe)} \tag{2.33}$$

Substituting the corresponding values for both sides from equation (2.28) and equation (2.32) gives

$$K(\frac{v^2}{2g}) = f(\frac{L_e}{D})(\frac{v^2}{2g}) \tag{2.34}$$

Eliminating velocities from both side, since they have the same value results in

$$L_e = \frac{KD}{f} \tag{2.35}$$

The value of equivalent length in equation (2.35) can be used then in equation (2.28) to calculate the head loss in valve or fitting or, one can add the value of L_e to the total length of the pipe before applying equation (2.28).

Example 2.8

Consider the hydraulic system in Figure 2.14. The pump adds a power of 3.73 KW to a fluid of specific gravity equals to 0.9 and a kinematic viscosity of 100 cSt. The fluid flow rate is 0.00190 m^3/s. The pipe has a 25.4 mm inside diameter and dimensions of $L_1 = 1.22m$, $L_2 = 0.3m$ and $L_3 = 4.88m$. Knowing that the pressure ar point (1) is atmospheric pressure and taking into account the friction losses, determine the pressure at point (2).

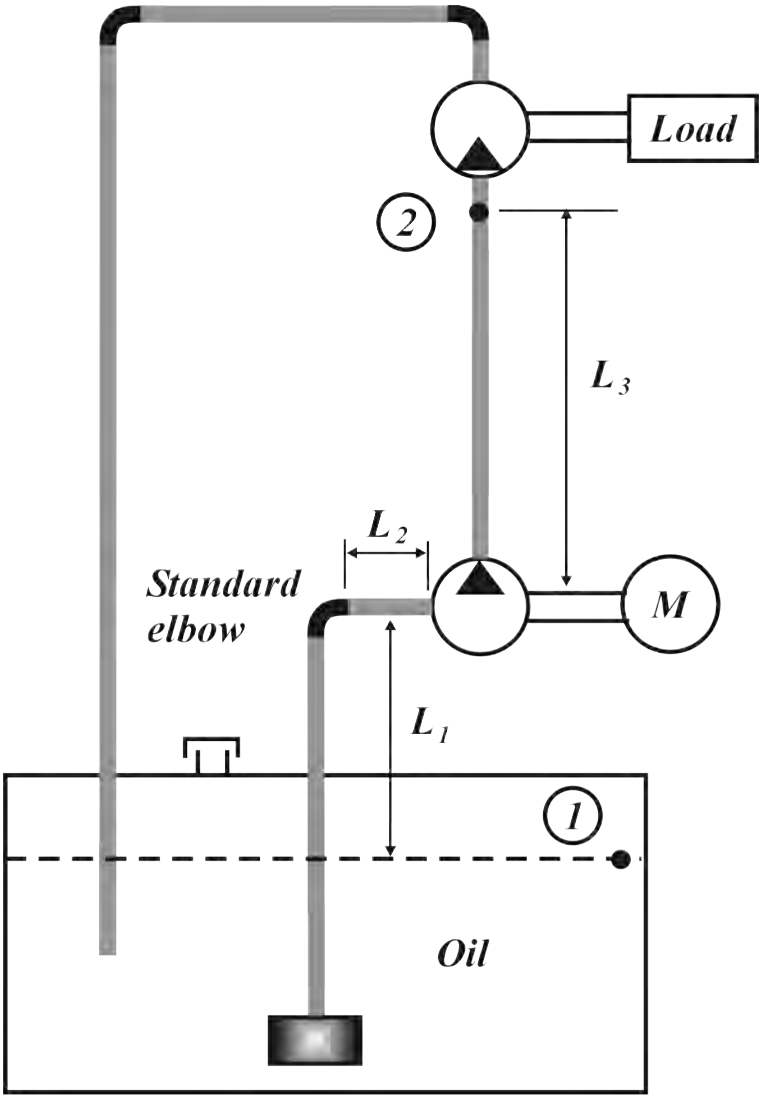

Figure 2.14: *Hydraulic system with friction effect, for example 2.8*

Solution

Applying Bernoulli's equation of motion for the conservation of energy

$$z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

Finding the different parameters of Bernoulli's equation
$z_1 = 0$ (*Height reference*)
$H_m = 0$ (*Hydraulic motor is not included*)
$v_1 = 0$ (*Large oil reservoir*)

$P_1 = 0$ (*Oil tank is open to atmosphere*)

$$z_2 = L_1 + L_3 = 1.22 + 4.88 = 6.1m$$

$$v_2 = \frac{Q}{A} = \frac{0.0019}{(\pi/4)(0.0254)} = 3.74m/s$$

The contribution of velocity to the head at point (2)

$$\frac{v_2^2}{2g} = \frac{3.74}{2(9.8)} = 0.714m$$

The head of the pump is calculated by

$$H_p = \frac{Pump\ hydraulic\ power}{\gamma \times Q}$$

$$H_p = \frac{3730}{(0.9)(9800) \times (0.0019)} = 223.1m$$

To determine the head loss due to friction, Reynold's number must be calculated at point (2)

$$Re = \frac{vD}{\nu} = \frac{(3.74)(0.0254)}{100 \times 10^{-6}} = 944$$

The flow at point (2) is laminar and the friction coefficient f is given by

$$f = \frac{64}{Re} = \frac{64}{944} = 0.0678$$

Calculating the equivalent length for the standard elbow (K=0.9)

$$L_e = \frac{KD}{f} = \frac{(0.9)(0.0254)}{0.0678} = 0.337m$$

The total length is

$$L_T = L_1 + L_2 + L_3 + L_e = 1.22 + 0.3 + 4.88 + .337 = 6.737m$$

The head loss is

$$H_L = f(\frac{L_T}{D})(\frac{v^2}{2g}) = (0.0678)(\frac{6.737}{0.0254})(0.714) = 12.9m$$

Substituting in Bernoulli's equation

$$0 + 0 + 0 + 223.1 - 0 - 12.9 = 6.1 + \frac{P_2}{\gamma} + 0.714$$

$$\frac{P_2}{\gamma} = 203.4m$$

Solving for P_2

$$P_2 = \gamma \times 203.4 = (0.9)(9800)(203.4) = 1790000Pa = 1.79MPa$$

Example 2.9

Consider the hydraulic system in Figure 2.15. Oil flows at 12 gpm from a tank through 1 in internal diameter pipes and elbows with (k = 0.75 each), and through a pump and motor as shown. The strain at the inlet has a pressure drop of 2 psi. The pump adds 3 hp and the motor extracts 1 hp. the oil has a specific gravity S.G = 0.9 and a kinematic viscosity $\nu = 105$ cSt. What is the pressure at point 2.

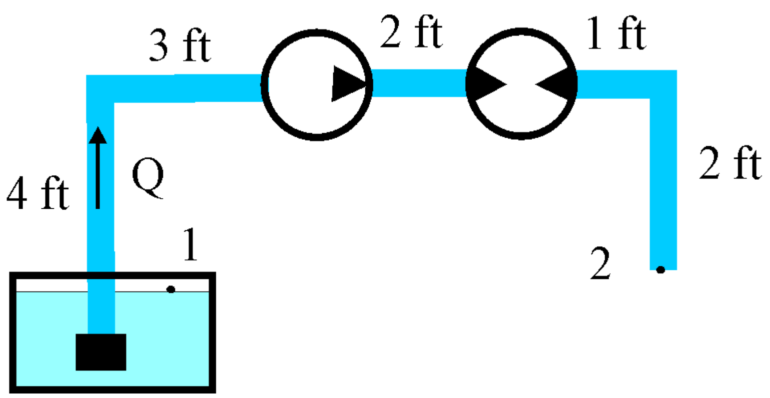

Figure 2.15: *Hydraulic system with friction effect, for example 2.9*

Solution

Apply Bernoulli's equation of energy as follows:

$$z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

Finding the different parameters of Bernoulli's equation
$z_1 = 0$ (*Height reference*)
$v_1 = 0$ (*Large oil reservoir*)
$P_1 = 0$ (*Oil tank is open to atmosphere*)
$z_2 = 4 - 2 = 2ft$ (*height difference between point 1 and point 2*)

The velocity

$$v_2 = \frac{Q}{A}$$

$$v_2 = (\frac{12.gal}{min})(\frac{4}{\pi(1.in)^2})(\frac{231.in^3}{gal})(\frac{min}{60.s})(\frac{ft}{12.in}) = 4.902ft/s$$

$$\frac{v_2^2}{2g} = \frac{4.902^2}{2 \times 32.2} = 0.373$$

Finding Reynold's number knowing that
$1ft.in/s = 7740cSt$

$$Re = \frac{7740 v_2 D}{\nu} = (\frac{7740.cSt.s}{ft.in})(\frac{1}{105.cSt})(\frac{4.902.ft}{s})(\frac{1.in}{1}) = 361$$

$Re < 2000$ this means having laminar flow, then the friction coefficient is

$$f = \frac{64}{Re} = \frac{64}{361} = 0.177$$

The total length of the pipes equals to the pipe length plus the equivalent length of the two elbows

$$L = L_{pipe} + 2\frac{K.D}{f}$$

$$L = 4.ft + 3.ft + 2.ft + 1.ft + 2.ft + 2(\frac{(0.75)(1)}{0.177})(\frac{ft}{12.in}) = 12.71ft$$

The head losses due to pipe and fittings friction is calculated by

$$H_L = f(\frac{L}{D})(\frac{v^2}{2g})$$

$$H_{Lf} = (0.177)(\frac{12.71ft}{1.in})(\frac{12.in}{ft})(\frac{(4.902ft/s)^2}{2(32.2ft/s^2)}) = 10.07ft$$

The strainer adds another loss due to pressure drop of 2 psi

$$H_{LP} = \frac{\Delta P}{\gamma} = (\frac{2.lb}{in^2})(\frac{ft^3}{(0.9)(62.4lb)})(\frac{(12in)^2}{ft^2}) = 5.13ft$$

the total head loss due to friction and strainer is
$H_L = H_{Lf} + H_{LP} = 10.07 + 5.13 = 15.20ft$
The head of the pump can be calculated from the equation

$$H_P = (\frac{3950gpm.ft}{hp})(\frac{P_{pump}}{Q.SG})$$

in numbers

$$H_P = (\frac{3950.gpm.ft}{hp})(\frac{3.hp}{(12.gpm)(0.9)}) = 1097ft$$

Similarly, the head of the hydraulic motor can be calculated from the equation

$$H_m = (\frac{3950gpm.ft}{hp})(\frac{P_{motor}}{Q.SG})$$

in numbers

$$H_m = (\frac{3950.gpm.ft}{hp})(\frac{1.hp}{(12.gpm)(0.9)}) = 366ft$$

Now implementing Bernoulli's equation for this problem

$$z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

in numbers

$$0 + 0 + 0 + 1097 - 366 - 15.2 = 2 + \frac{P_2}{\gamma} + 0.373$$

$$P_2 = (-2 + 1097 - 366 - 15.2 - 0.373)ft \times (\frac{0.9 \times 62.4lb}{ft^3})(\frac{ft^2}{(12in)^2}) = 278psi$$

2.4.2 Friction losses in pneumatic systems

Pressure losses in pneumatic systems can be calculated from Harris formula. Harris formula is an empirical formula used to calculate pressure loss directly in British units as follows:

$$\Delta P = \frac{CLQ^2}{3600(CR)D^5} \tag{2.36}$$

where
ΔP = Pressure loss (psi).
L = Length of pipe (ft).
Q = flow rate (ft^3/min).
D = Inside diameter of pipe (in).
CR = Compression ratio,

$$CR = \frac{Pressure\ in\ pipe}{Atmospheric\ pressure}$$

C = Experimentally determined coefficient, which is for schedule 40 commercial pipes

$$C = \frac{0.1025}{D^{0.31}}$$

Substituting this value in Harris formula, equation (2.36) results in

$$\Delta P = \frac{(0.1025)LQ^2}{3600(CR)D^{5.31}} \tag{2.37}$$

This formula can be used directly to calculate the pressure loss in pneumatic pipes when the system is known in Imperial or British units. A similar empirical formula is also used for SI metric units and it reads

$$\Delta P = \frac{7.57 \times 10^4 LQ^{1.85}}{pD^5} \tag{2.38}$$

where
ΔP = pressure drop (kg/cm^2)
Q = air volume flow rate at atmospheric conditions (m^3/min)

L = length of pipe (m)
D = inside pipe diameter (mm)
p = initial gauge pressure (kg/cm^2)
$1kg/cm^2 = 98068Pa = 0.98bar = 14.2psi$

For practical design it is possible to use the nomogram calculator scale shown in Figure 2.16. The Nomogram contains three scales one for the flow rate, the second for the pipe diameter and the third scale is for pressure loss values. The designer can define the air flow rate and the inside diameter of the pipe in the graph, then with a simple ruler he can draw a line between the two points and extend it to cross the third scale that contains the values of pressure losses. The point of intersection with this scale indicates the value of pressure losses in the pipes. This chart contains values in both unit systems SI and Imperial systems and calculated at a working pressure of 7 bar (100 psi).

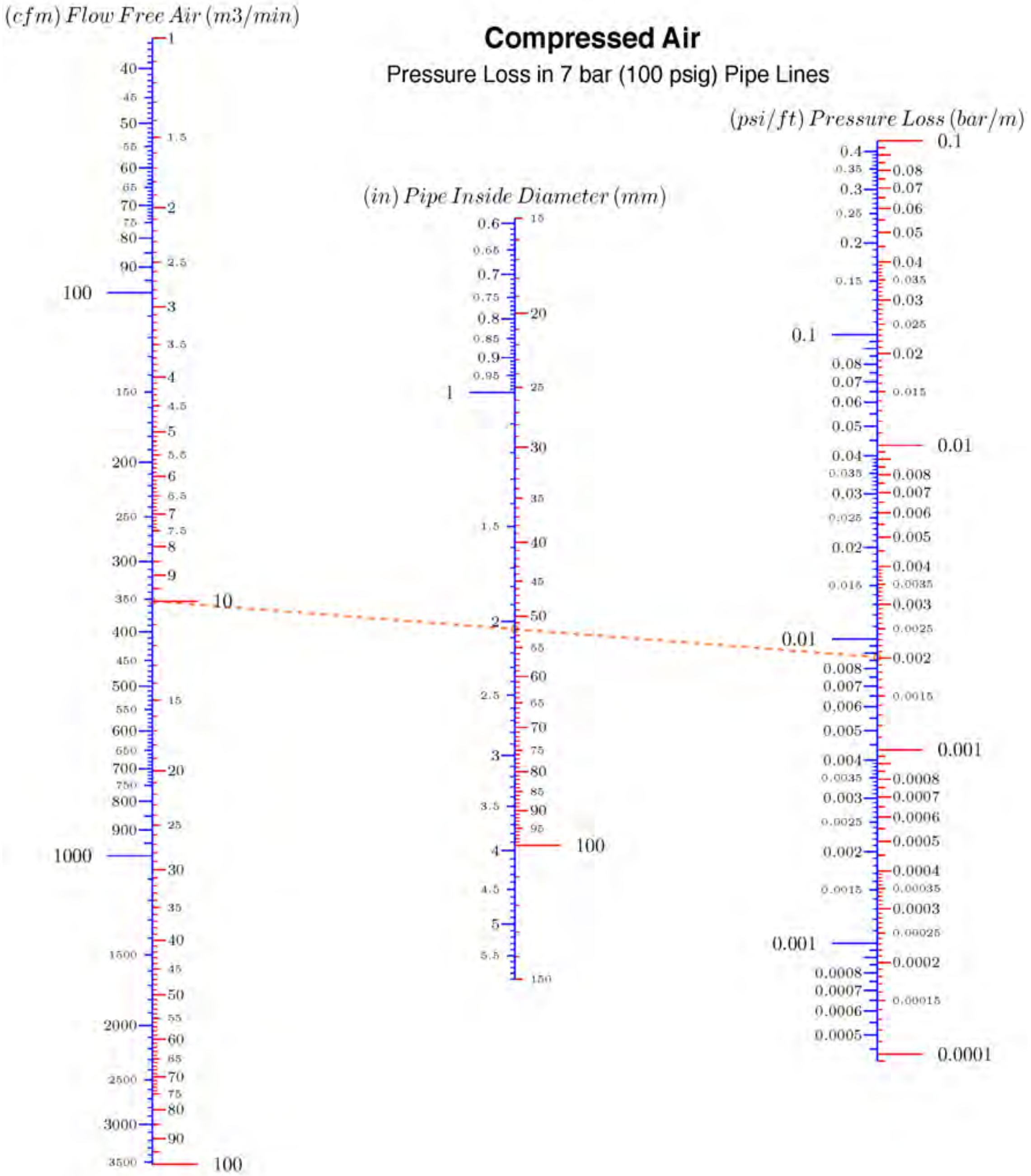

Figure 2.16: *Nomogram scale calculator to estimate the pressure losses in pneumatic pipes*

Example 2.10

Consider a compressed air pneumatic system where a 25 mm pipe transfers the compressed air from the tank to an actuator at a distance of 12 m. The initial gauge pressure in the pipe is 7 bar and it flows at a flow rate of 100 l/s. Calculate the total pressure drop in the pipe.

Solution

$Q = 100 l/s = 0.1 m^3/s = 6 m^3/min$
$L = 12m$
$D = 25mm$
$p = 7bar = (7/0.98) = 7.14 kg/cm^2$

Applying Harris formula for SI units

$$\Delta P = \frac{7.57 \times 10^4 L Q^{1.85}}{p D^5}$$

$$\Delta P = \frac{7.57 \times 10^4 (12)(6)^{1.85}}{(7.14)(25)^5}$$

$$\Delta P = 0.3585 kg/cm^2 = 0.351 bar$$

Or one can look at the Nomogram shown in Figure 2.16, taking the flow rate on the left column as $Q = 6m^3/min$ and the diameter at the middle column as $D = 25mm$, then with a straight ruler you can draw a line connecting the two previous values and extend it to cross the right column containing the values of pressure loss. The point where the line crosses the right column is $0.029 bar/m$ which represents the pressure loss per meter. To estimate the total pressure loss in the pipe just multiply by the length of the pipe (12 m) which results 0.35 bar.

2.5 Problems

1. Determine and explain the power transmission system in the following systems:

- Passenger cars
- Wheel loaders
- Chain excavators
- Voice coil actuators
- Concrete patching plant

2. Consider a fluid power actuator wanted to be extended from 0 mm to 20 mm stroke. Draw a rough plot for the time response of the piston showing the time delay and the dynamic response once for hydraulic piston and another time for a pneumatic actuator. suggest a time delay and settling time for each actuator.

3. Nominate all basic components of a hydraulic system and of a pneumatic system.

4. What should be the characteristics of the hydraulic tank (reservoir)?

5. What should be the characteristics of the pneumatic tank?

6. What is the function of the FRL in pneumatic system?

7. Indicate three differences between the hydraulic system and the pneumatic system from functionality point of view.

8. A pump supplies oil at 0.002 m^3/s to a 50mm diameter double acting cylinder and a rod diameter is 20mm. If the load is 6000N both in extending and retracting, find:

    - a. Piston velocity during the extension stroke and retraction stroke
    - b. Pressure during the extension stroke and retraction stroke
    - c. Power during the extension stroke and retraction stroke

9. A hydraulic cylinder has to move a table of mass 1.3 kg. Speed of the cylinder is to be accelerated up to a velocity of 13 cm/s in 0.5 seconds and brought to stop within a distance of 2 cm. Assume coefficient of sliding friction as 0.15 and cylinder bore diameter as 50 mm. Calculate the required pressure.

10. A cylinder has a bore of 125mm diameter and a rod of 70mm diameter. It drives a load of 2000 kg vertically up and down at a maximum velocity of 3 m/s. The load is slowed down to rest in the cushion length of 50mm. If the relief valve is set at 140 bar, determine the average pressure in the cushions while extending and retracting.

11. For the crane system given in Figure 2.17, find the pressure required to lift a load of 9000 N if the cylinder's bore is 100 mm.

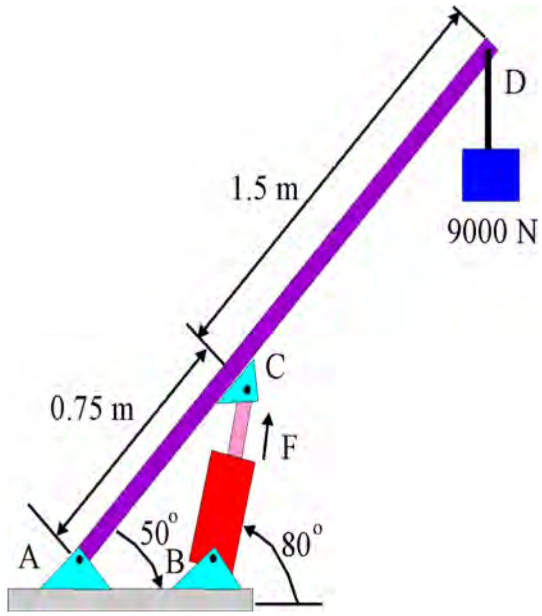

Figure 2.17: *Crane with loading*

12. A mass of 2000kg is to be accelerated up to a velocity of 1 m/s from rest over a distance of 50 mm. The coefficient of friction between the load and the guides is 0.15. Select the bore of the cylinder required to accelerate this load if the maximum allowable pressure at the full bore end is 100 bar. Take seal friction to be equivalent to a pressure drop of 5 bar.

13. A hydraulic cylinder is to accelerate a load of 50 Tonnes horizontally form rest with a velocity of 10 m/min to 50 mm as shown in Figure 2.18. Take coefficient of friction between the load and the guide as 0.1. Assume zero back pressure. Determine

a) a suitable size of standard metric cylinder if the maximum allowable pressure at the cylinder is 180 bar
b) the fluid flow rate required to drive the piston forward at 3 m/min

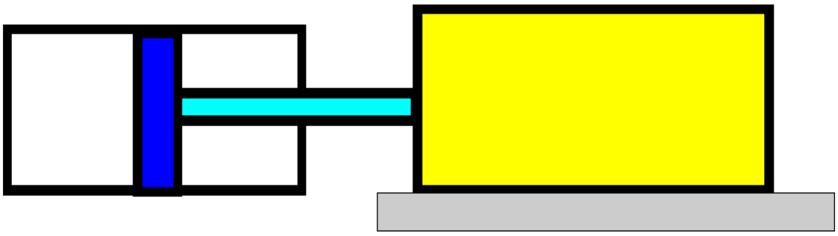

Figure 2.18: *Hydraulic cylinder with 50 tonnes load and friction*

14. Oil flows at a rate of 3 gpm through a horizontal 0.75 inch ID pipe for 10 feet, passes through a 90 standard elbow into a vertical pipe which drops for 12 feet, passes through a second 90 elbow into a horizontal pipe for another 14 feet as shown in Figure 2.19. The oil has a specific weight γ=54 lb./ft3 and a kinematic viscosity ν=75cSt . If the initial pressure is 90 psi, what is the final pressure?

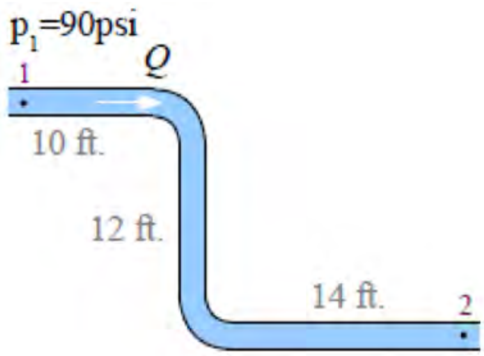

Figure 2.19: *Hydraulic fluid flow in a pipe*

15. A horizontal nozzle discharges water into the atmosphere. The inlet has a bore area of 600 mm^2 and the exit has a bore are of 200 mm^2. Calculate the flow rate when the inlet pressure equals 400 Pa. Assume no energy losses.

16. The system shown in the Figure 2.20 contains a pump delivering high-pressure oil to a hydraulic motor, which drives an external load via rotating shaft. The following data are given:
    Pump:
    Flow rate (Q_p = 154 l/min), efficiency (p = 85%), inlet pressure (P_{in_p} = -27.5 KPa).
    Hydraulic motor:
    Flow rate (Q_m = 138.5 l/min), efficiency (η_m = 83%), speed (ω_m = 1057 rpm), inlet pressure ($P_i n_m$ = 3447 KPa), outlet pressure (P_{out_m} = 34.47 KPa).
    Pipe between pump and motor:
    Length (L = 15.2 m), Diameter (D = 25.4 mm). level difference ($Z_2 - Z_1$ = 6 m), elbows 90^o (2 elbows with k = 0.75 each), one check valve (k = 4.0).
    Oil: Viscosity (ν = 125 cSt), specific gravity (S.G = 0.9), γ_{water} = 9800, Stoke = cm^2/s.
    Find the following:
    a- Pump discharge pressure P1.
    b- Input Horse power required to drive the pump.
    c- Motor output Horse power.
    d- Motor output torque.
    e- Overall efficiency of the system

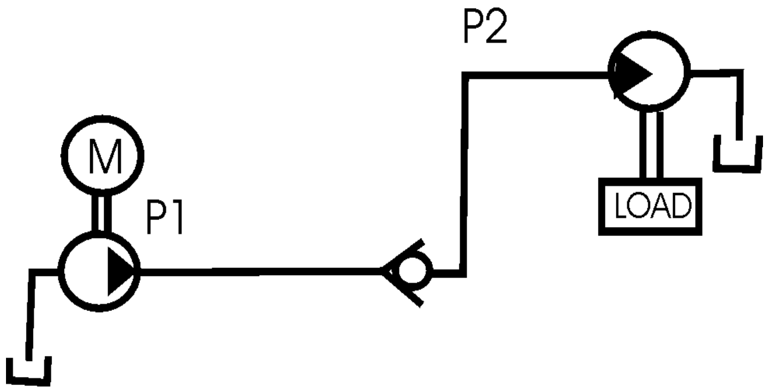

Figure 2.20: *Hydraulic pump and hydraulic motor*

17. A hydraulic cylinder having a bore of 125 mm, a rod of 80 mm diameter and a stroke of 350 mm, is to fully extend in 15 seconds. The cylinder exerts a force of 20 tons in extension and 10 tons in retraction, determine: a) The theoretical system pressure when the cylinder is extending. b) The theoretical system pressure when the cylinder is retracting. c) The required theoretical pump delivery. d) The actual pump displacement if the volumetric efficiency is 90% and the pump is driven at 1440 rev/min.

18. Find the pressure P_2 in the circuit shown in Figure 2.21 including friction losses, knowing that:
    Pressure $P1 = 7bar$
    Flow rate $Q = 0.002m3/s$
    Inner diameter $D = 38mm$
    Oil specific gravity $SG = 0.9$
    Viscosity $= 0.0001m2/s$
    For 90^o elbow $Kfactor = 0.75$

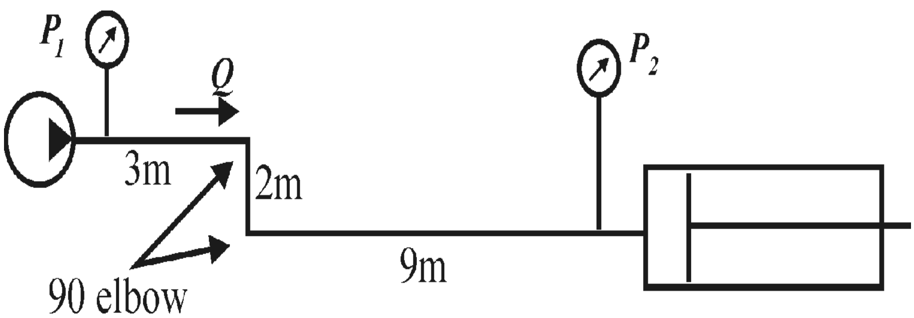

Figure 2.21: *Hydraulic pump and hydraulic actuator*

19. A compressed air is enclosed in a reservoir at a pressure of 100 psi. The air is transported through a 30 ft long with 0.5 inch diameter pipe at a flow rate of 25 ft^3/min to a pneumatic piston. Determine the bore diameter of the piston required to lift a mass of 10 lb mass. (1 atm = 2116 lb/ft^3, g = 32.2 ft/s^2)

20. A cylinder has a bore of 125mm diameter and a rod of 70mm diameter

(Figure 2.22). It drives a load of 2000 kg vertically up and down at a maximum velocity of 3 m/s. The load is slowed down to rest in the cushion length of 50mm. If the relief valve is set at 140 bar, determine the average pressure in the cushions while extending and retracting.

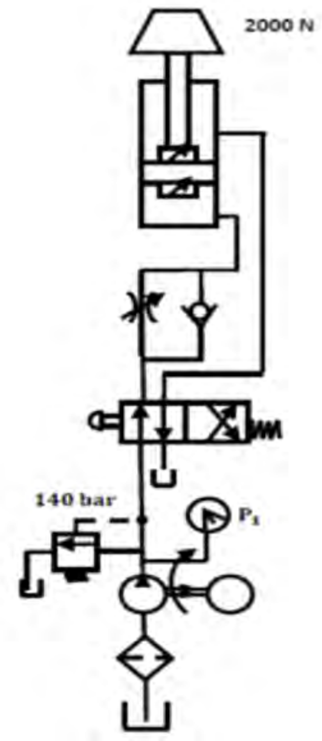

Figure 2.22: *Vertical loading hydraulic system*

21. Consider the following series hydraulic cylinder arrangement in Figure 2.23. Knowing that the rod area have to be 0.5 the cylinder area in both cylinders. The force F_1=1000N and the force F_2=2000N and the pressure out from the pump P_1=300 kPa required to move the two cylinders in synchronization the distance of 100 cm in one second.
   1- Find the bore areas and strokes of the two cylinders.
   2- What is the required flow rate from the pump.

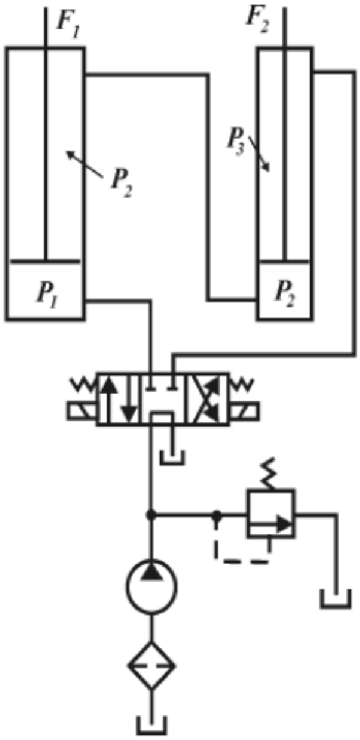

Figure 2.23: *Two hydraulic actuators in series*

22. A 3-meters height hydraulic elevator consists of:
    1- Electric motor on the ground
    2- Oil tank and 4 meters long, 25 mm diameter cast iron pipes on the ground
    3- Oil (hydraulic stiffness = 1.5×10^7 N/m, kinetic viscosity = 100 cSt, S.G = 0.9, damping coefficient = 1000 N.s/m)
    4- hydraulic pump (efficiency = 85%) on the ground near the tank
    5- Hydraulic linear actuator (150 mm bore and 100 mm rod diameter) on the ground
    6- Cabin (500 kg mass and 0.1 m/s required speed)
    Determine:
    a) The volume of the tank (best design)
    b) Pressure and flow rate in the actuator
    c) Pressure and flow rate of the pump (include friction losses)
    d) Power of the electric motor

2.6 References

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Chapter 3

Modelling of Fluid Power Systems

3.1 Introduction

To study the behaviour of any mechanical system, it is important to represent this system using mathematical equations. The most modern way of representation is expressed in building a second order differential equation of motion for the system which is called $Modelling$ [1] . This equation of motion can be built using one of the following ways:

- Newton's second law of motion which states that the induced force is directly proportional to the acceleration of the body and the summation of external forces acting on the body is equal to the inertial force of the body itself [2].
- Conservation of energy : where the energy of the body is conserved but it changes from one shape to another as the body changes its position or state [2].
- Finite element modelling is the most precise numerical way of modelling. It is handled by dividing the body into small pieces to be analysed and studied. this can represent the virtual real behaviour of the system [3].
- Modal analysis is performed by installing measurement sensors on different parts of the system. The signals retrieved from these sensors are inserted into a mathematical calculation to simulate the real behaviour of the system [4].

This book will focus on using Newton's law of motion to establish equations of motion .

3.2 Block diagrams

Solving the equation of motion leads to build relations between the different parameters of the systems. These relations are called the $TransferFunctions$. Building an equation of motion and deriving transfer function will be discussed later in this chapter. Transfer function expresses the relation between an input and an output of the system taking into account the influence of all other parameters. Transfer function is usually denoted by the letter G or other letters. It is time dependent relationship and can be expressed in Laplace transform to be used in frequency domain calculations $G(s)$. The relationships and operations between transfer functions can be studied by using block diagrams [5]. Figure 3.1 shows a block diagram of three transfer functions in series where they can be represented by the multiplication of the functions. Transfer functions in parallel are represented by the summation of these functions as shown in Figure 3.2. In the case of parallel connection, the point where the inputs branch before entering to the blocks is called *Take-off* point and the point of summing the outputs is called *Sum* point.

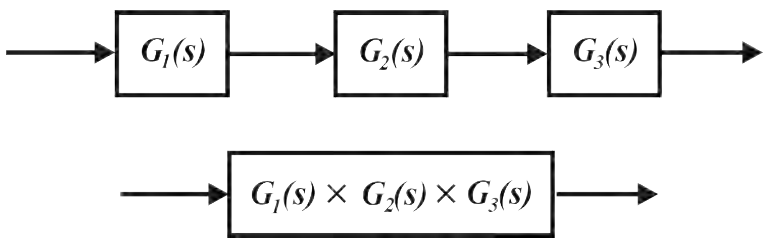

Figure 3.1: *Blocks in series*

The aim of the control system is to minimize the error by seeking for zero value between the input and the signal fed back from the output. Such a system is called *Feedback system* and is shown in Figure 3.3. The feedback control system shown in Figure 3.3 consists of two transfer functions $G_1(s)$

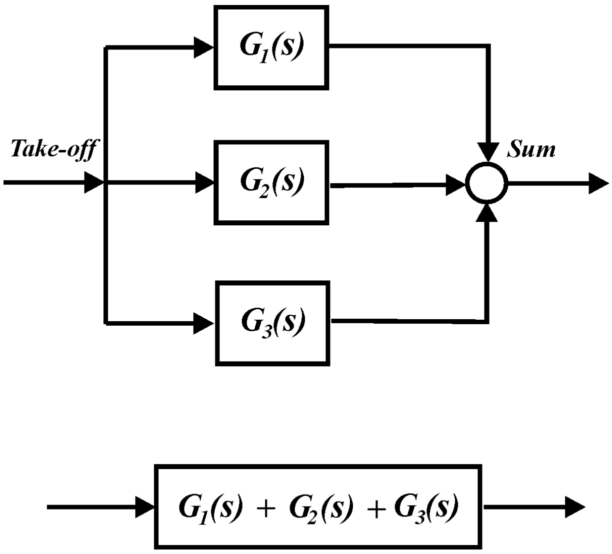

Figure 3.2: *Blocks in parallel*

and $G_2(s)$ connected in series. The input reference signal coming into the system is $I(s)$ and the output signal is $O(s)$. The error signal $E(s)$ is minimized by feeding the output signal back and subtracting it from the input after multiplying by the compensator transfer function $H(s)$ yielding the feedback signal $F(s)$. The transfer function of the system is the ratio between the input $I(s)$ and the output $O(s)$ can be found by manipulating the blocks to find $(O(s)/I(s))$.

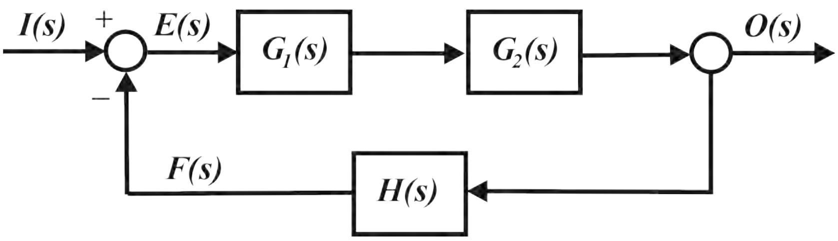

Figure 3.3: *Feedback control system*

Looking at the main feedforward path,

$$O(s) = G_1(s)G_2(s)E(s)$$

Looking at the feedback path,

$$F(s) = H(s)O(s)$$

The error signal at the summation junction,

$$E(s) = I(s) - F(s)$$

Hence,

$$E(s) = I(s) - H(s)O(s)$$

and

$$O(s) = G_1(s)G_2(s)[I(s) - H(s)O(s)]$$

This leads to the ratio between $I(s)$ and $O(s)$

$$\frac{O(s)}{I(s)} = G_1(s)G_2(s) - G_1(s)G_2(s)\frac{H(s)O(s)}{I(s)}$$

Or,

$$\frac{O(s)}{I(s)}[1 + G_1(s)G_2(s)] = G_1(s)G_2(s)$$

Finally,

$$\frac{O(s)}{I(s)} = \frac{G_1(s)G_2(s)}{1 + G_1(s)G_2(s)H(s)} \tag{3.1}$$

This transfer function is considered the general closed-loop transfer function for the feedback control system [6].

Example 3.1

Find the transfer function between u as an input and y as an output in the block diagram shown in Figure 3.4.

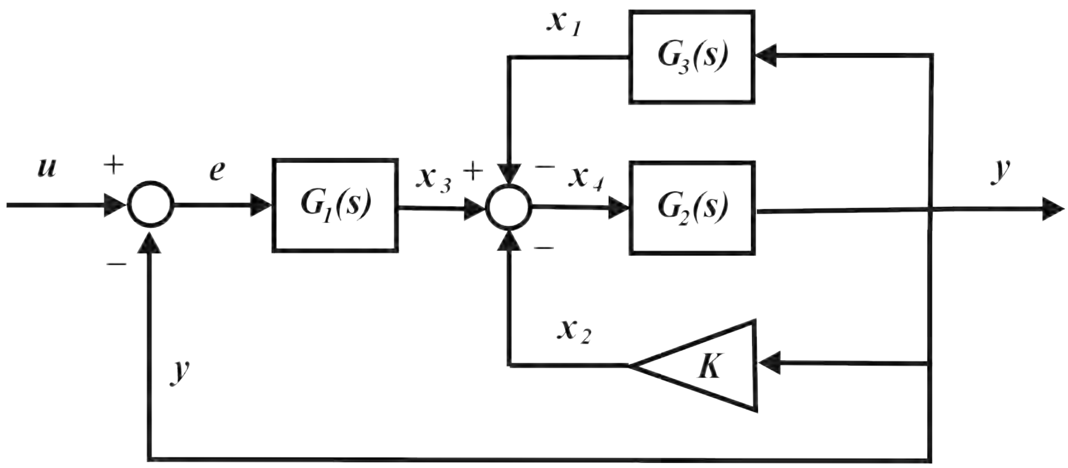

Figure 3.4: *Block diagram for example 3.1*

Solution

Starting at the summation junction

$$x_4 = x_3 - x_2 - x_1$$

$$x_4 = x_3 - G_3(s)y - Ky$$

But,

$$y = G_2(s)x_4 = G_2(s)[x_3 - G_3y - Ky]$$

Manipulation gives

$$y = \frac{G_2(s)}{1 + G_2(s)G_3(s) + K}$$

Looking at the other summation junction

$$e = u - y$$

and

$$x_3 = G_1(s)e$$

then,

$$y = \left(\frac{G_1(s)G_2(s)}{1 + G_2(s)(G_3(s) + K)} \right) e$$

Or,

$$y = \left(\frac{G_1(s)G_2(s)}{1 + G_2(s)(G_3(s) + K)} \right) (u - y)$$

Thus,

$$y = u \left(\frac{G_1(s)G_2(s)}{1 + G_2(s)(G_3(s) + K)} \right) / \left(1 + \frac{G_1(s)G_2(s)}{1 + G_2(s)(G_3(s) + K)} \right)$$

Final mnipulation gives the transfer function

$$\frac{y}{u} = \frac{G_1(s)G_2(s)}{1 + G_2(s)[G_3(s) + G_1(s) + K]}$$

Example 3.2

Consider the two parallel pistons driven by a hydraulic pump through a directional control valve shown in Figure 3.5. Find the transfer function between pump displacement X as an input and the actuator displacement Y as an output in the block diagram shown in Figure 3.5.

Solution

The system can be represented by the block diagram shown in Figure 3.6, where:

X = Pump controller displacement
Q = Pump flow rate
$GP(s)$ = Pump transfer function between controller displacement and flow rate
F = feedback signal from controller
E = Error signal
$GV(s)$ = Valve transfer function

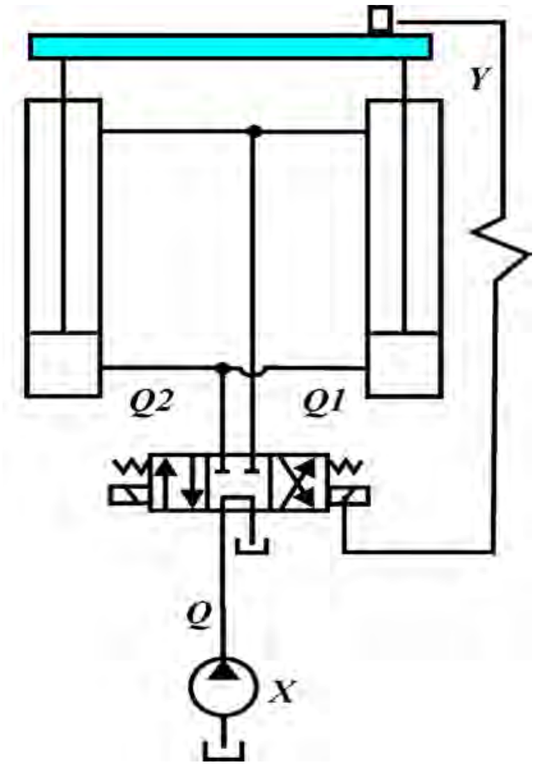

Figure 3.5: *Two pistons driven by a pump*

$G1(s)$ = Actuator 1 transfer function
$G2(s)$ = Actuator 2 transfer function
$H(s)$ = Controller (compensator) transfer function
Y = Displacement of the actuators

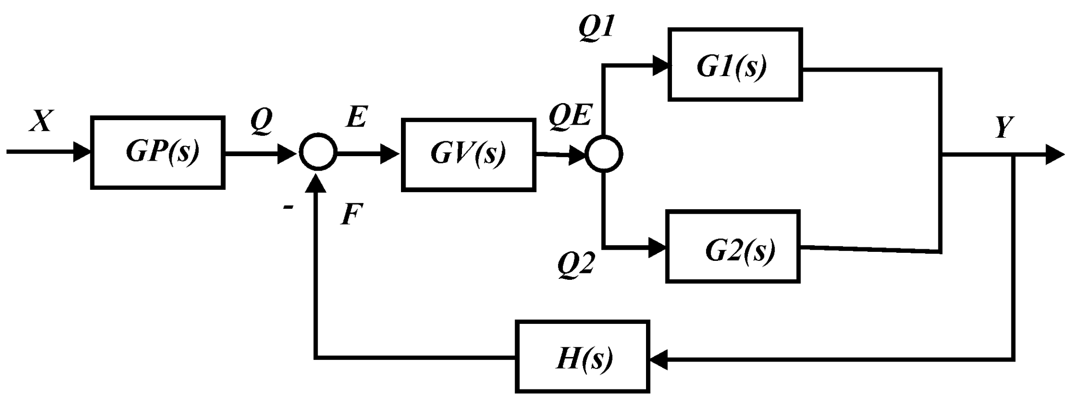

Figure 3.6: *Block diagram for the two pistons example*

$E = Q - F$
$Q = X.GP(s)$
$F = Y.H(s)$
So
$E = X.GP(s) - Y.H(s)$
$QE = E.GV(s)$
$QE = GV(s)(X.GP(s) - Y.H(s))$
$Y = QE(G1(s) + G2(s))$
$Y = (GV(s).X.GP(s) - GV(s).Y.H(s))(G1(s) + G2(s))$
or
$Y = GV(s).X.GP(s).G1(s)+GV(s).X.GP(s).G2(s)-GV(s).Y.H(s).G1(s)-GV(s).Y.H(s).G2(s)$
separating variables
$Y(1+GV(s).H(s).G1(s)+GV(s).H(s).G2(s)) = X(GV(s).GP(s).G1(s)+GV(s).GP(s).G2(s))$

The transfer function reads

$$\frac{Y}{X} = \frac{GV(s).GP(s)(G1(s) + G2(s))}{1 + GV(s).H(s)(G1(s) + G2(s))}$$

3.3 Conceptual modelling of a transfer function

As mentioned at the beginning of this chapter, Newton's law will be used to build the transfer function. Assuming a lumped parameter model, mass, spring and damper are the basic elements of this model [7]. The Inertia force is given by

$$F = m\frac{d^2x}{dt^2} = m\frac{dv}{dt} \tag{3.2}$$

where, F is the force, m is the mass and x is the displacement, reciprocating the equation gives

$$v = \frac{1}{m}\int F dt \tag{3.3}$$

where v is the velocity. In the case of rotation, the mass is replaced by the Inertia I , the force is replaced by the torque T and the displacement is replaced by the angle θ

$$T = I\frac{d^2\theta}{dt^2} = I\frac{d\omega}{dt} \tag{3.4}$$

Where ω is the angular velocity. The mass or inertia element is a kinetic energy storage element. The spring is considered as a potential energy storage element, the spring force reads

$$F = k(x_1 - x_2) \tag{3.5}$$

or,

$$F = k\int(v_1 - v_2)dt \tag{3.6}$$

In the rotation case

$$T = k(\theta_1 - \theta_2) \tag{3.7}$$

When a damper is added to the system, it bahaves as an energy dissipation element in the form of heat, the damping force is given by

$$F = C(v_1 - v_2) \tag{3.8}$$

In the rotation case

$$T = C(\omega_1 - \omega_2) \tag{3.9}$$

3.4 Effort and flow

The modelling of inputs and outputs of any system can be represented by effort e and flow f as shown in Figure 3.7. The overall power is the multiplication of the effort and the flow

$$Power = effort \times flow = e \times f$$

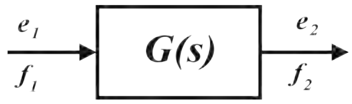

Figure 3.7: *Effort and flow representation*

and the efficiency is

$$Efficiency = \frac{Power_{out}}{Power_{in}}$$

The transfer function is called the impedance where

$$Input\ Impedance = \frac{e_1}{f_1}$$

$$Output\ Impedance = \frac{e_2}{f_2}$$

If the flow is constant $f_1 = f_2 = f$, then

$$Transfer\ Function\ \frac{e_2}{e_1} = \frac{e_2}{f} \times \frac{f}{e_1}$$

According to causality theorem, an effort F can be a cause that leads to effect v (Figure 3.8) and a flow v can be a cause leading to effect F (Figure 3.9).

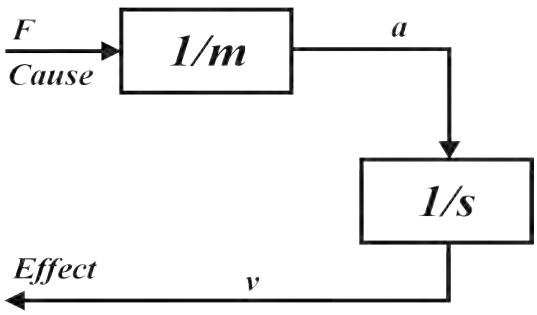

Figure 3.8: *Effort causes flow*

Table 3.1 shows a comparison between the different general systems indicating the effort and flow for each system

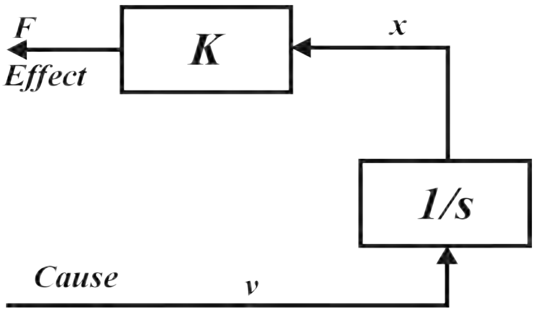

Figure 3.9: *Flow causes effort*

Table 3.1: *Comparison between general systems*

General system	**Effort** (e)	**Flow** (f)	$\int (f)dt$	$\int (e)dt$
Mechanical	Force (F)	Velocity (v)	Displacement (x)	Momentum (L)
Mechanical	Torque (T)	Angular velocity (ω)	Angle (θ)	Angular momentum (H)
Electrical	Voltage (V)	Current (i)	Charge (q)	Flux (ϕ)
Fluid	Pressure (P)	Flow rate (Q)	Volume (V)	Pressure momentum (Φ)

Example 3.3

Consider an electric motor with a voltage V and current I, produces a torque T and angular speed ω. The motor drives a hydraulic pump that produces a flow rate Q and capable of operating a system at a pressure P. Find the impedance values at the inputs and outputs for both; the motor and the pump and find the overall transfer functions.

Solution

For the electric motor:
Input impedance = $e_1/f_1 = V/I$
Output impedance = $e_2/f_2 = T/\omega$
Transfer functions:
$e_2/e_1 = T/V$

$e_2/f_1 = T/I$
$f_2/e_1 = \omega/V$
$f_2/f_1 = \omega/I$

For the hydraulic pump:
Input impedance $= e_2/f_2 = T/\omega$
Output impedance $= e_4/f_4 = P/Q$
Transfer functions:
$e_4/e_2 = P/T$
$e_4/f_2 = P/\omega$
$f_4/e_2 = Q/T$
$f_4/f_2 = Q/\omega$

Looking at the whole system, one can build four transfer functions:
$e_4/e_1 = (e_4/e_2)(e_2/e_1) = (P/T)(T/V) = P/V$
$e_4/f_1 = (e_4/e_2)(e_2/f_1) = (P/T)(T/I) = P/I$
$f_4/e_1 = (f_4/f_2)(f_2/e_1) = (Q/\omega)(\omega/V) = Q/V$
$f_4/f_1 = (f_4/f_2)(f_2/f_1) = (Q/\omega)(\omega/I) = Q/I$

3.5 Linear modelling of a piston

3.5.1 First order transfer function

A fluid power piston without mass effect can be modelled as shown in Figure 3.10 where, k is the stiffness of fluid, C is the viscous damping of moving piston, x is the input displacement motion of fluid and y is the output displacement of the piston. The fluid is under a pressure P and acts at a piston area A. This system can be represented by the block diagram shown in Figure 3.11.

The resultant of the forces acting on the spring damper system shown in Figure 3.11 is equal to zero

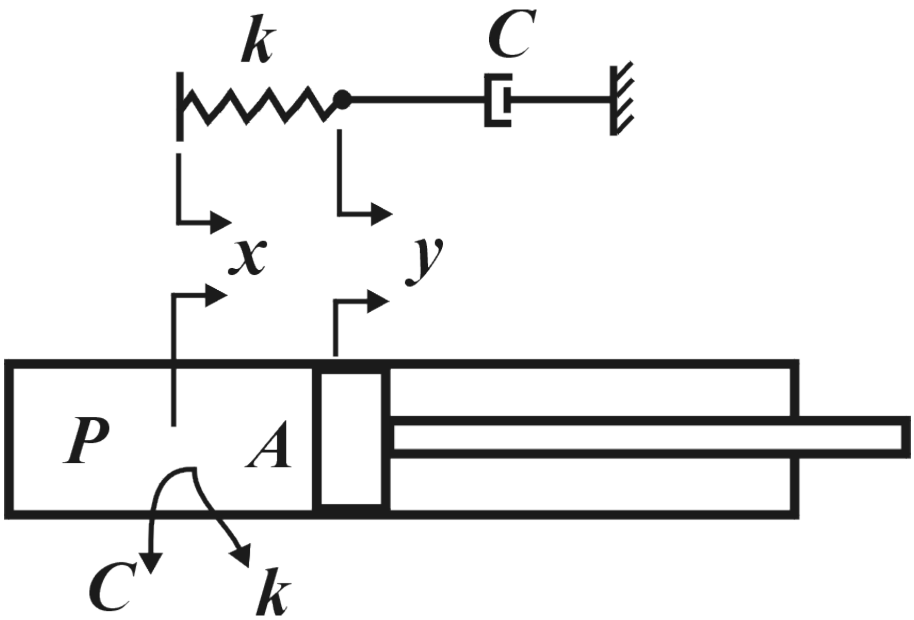

Figure 3.10: *First order dynamic model of a piston*

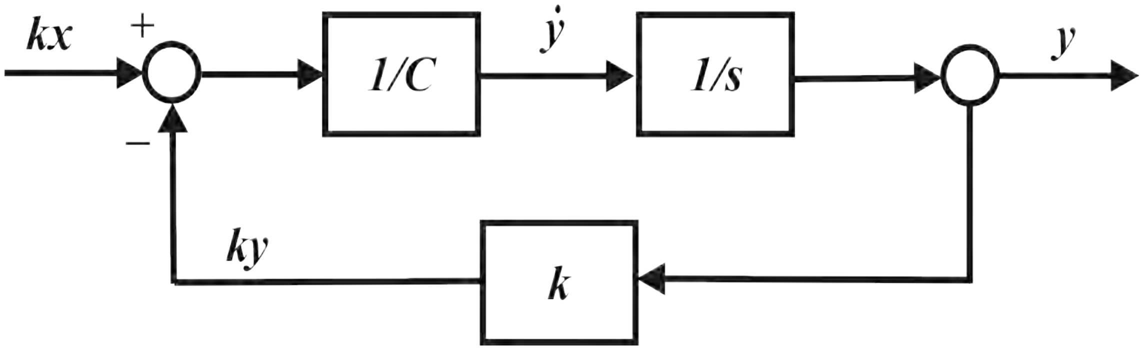

Figure 3.11: *First order block diagram of a piston*

$$kx - ky - C\dot{y} = 0 \qquad (3.10)$$

or

$$kx - ky = C\frac{dy}{dt}$$

Assume having a unit input $x = 1$, then

$$C\frac{dy}{dt} = k(1 - y)$$

leading to the first order differential equation

$$dy = \frac{k}{C}(1-y)dt$$

or

$$\frac{1}{1-y}dy = \frac{k}{C}dt$$

Integrating both sides of the equation

$$\int \frac{1}{1-y}dy = \int \frac{k}{C}dt$$

$$ln(1-y) = \frac{k}{C}t$$

Solving the equation for the output y

$$y = 1 - e^{-\frac{k}{C}t}$$

Thus, finding the output y in time domaine

$$y = 1 - e^{-t/\frac{C}{k}}$$

or

$$y = 1 - e^{-t/\tau}$$

where, $\tau = C/k$ is the time constant.
If the input is a sinusoidal harmonic motion at a frequency ω and an amplitude X, the output will be a harmonic motion at a new amplitude Y and a phase shift ϕ.

Input $x(t) = Xsin(\omega t)$
Output $y(t) = Ysin(\omega t + \phi)$

Referring back to Figure 3.11, the summation of forces reads

$$x = \frac{C}{k}\dot{y} + y$$

In Laplace transform,

$$x = \frac{C}{k}sy + y$$

where s is the Laplace variable and can be related to the frequancy in the Real-Imaginary plane by $s = j\omega$. j here is the imaginary number $j = \sqrt{-1}$.

$$x = \frac{C}{k}j\omega y + y$$

or

$$x = y(\frac{C}{k}j\omega + 1)$$

The transfer function between x as an input and y as an output is then given by

$$\frac{y}{x} = \frac{1}{1 + (C/k)j\omega}$$

It has been shown that $C/k = \tau$ and $j\omega = s$. Back to Laplace transform

$$\frac{y(s)}{x(s)} = \frac{1}{1 + \tau s}$$

This transfer function represents a first order low-pass filter with a pole at $s = -1/\tau$. The Bode plot for this transfer function is shown in Figure 3.12. The magnitude is calculated in decibles (dB) where

$$dB = 20 \times log_{10}|\frac{y}{x}|$$

and the phase angle ϕ is in degrees where

$$\phi = tan^{-1}(-\tau\omega)$$

It is noticed that the roll-off of the transfer function at high frequency is equal to $-20dB/decade$ which corresponds to the first order frequency response function.

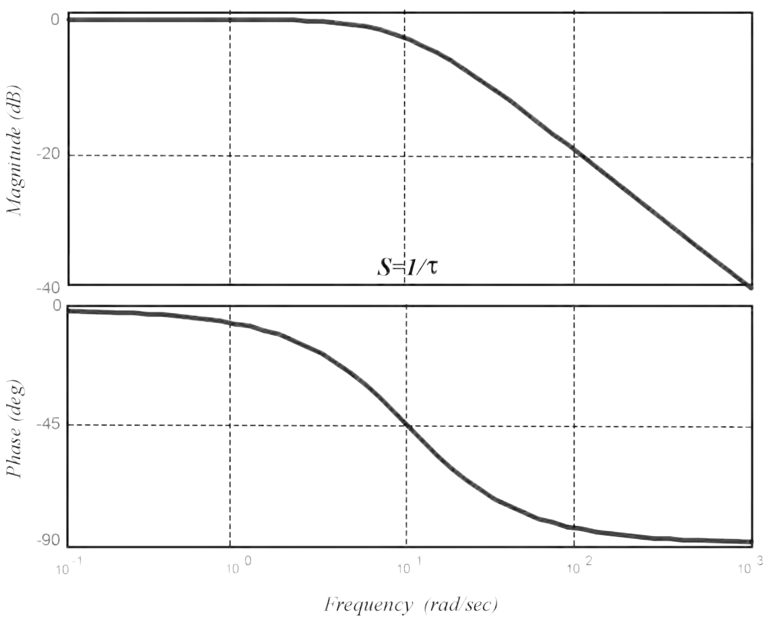

Figure 3.12: *Frequency response function of first order low-pass filter*

3.5.2 Second order transfer function

To build a more convenient model, a fluid power piston with mass effect can be modelled as shown in Figure 3.13, where m is the mass of the piston rod and suspended load, k is the stiffness of fluid, C is the viscous damping of moving piston, x is the input displacement motion of fluid and y is the output displacement of the piston. The fluid is under a pressure P and acts at a piston area A. This system can be represented by the block diagram shown in Figure 3.14.

Looking at this system, one can determine the forces acting on the system:

Inertia force

$$F_m = m\ddot{y} = ms^2 y$$

Damping force

$$F_C = C\dot{y} = -Csy$$

Spring force

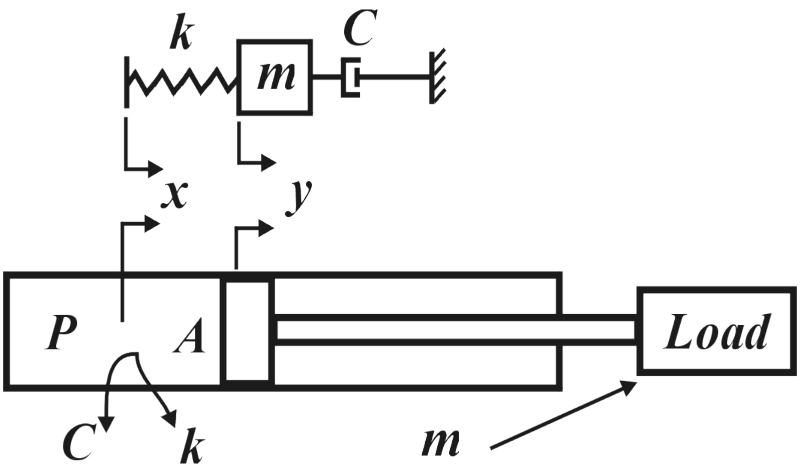

Figure 3.13: *Second order dynamic model of a piston*

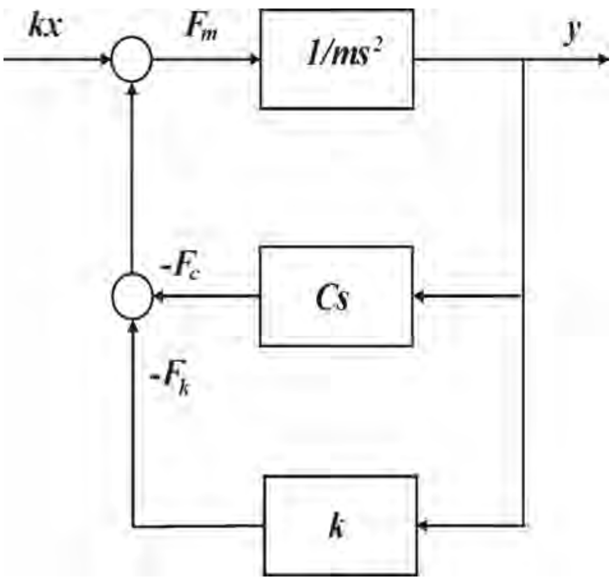

Figure 3.14: *Second order block diagram of a piston*

$$F_k = k(x - y)$$

Applying Newton's law for the summation of forces results in

$$kx - F_C - F_k = F_m$$

Substituting the vlues of the forces givs

$$ms^2y + Csy + ky = kx \tag{3.11}$$

or

$$(ms^2 + Cs + k)y = kx$$

The transfer function between x and y is

$$\frac{y}{x} = \frac{k}{ms^2 + Cs + k} \tag{3.12}$$

or

$$\frac{y}{x} = \frac{k/m}{s^2 + (C/m)s + (k/m)}$$

but it known that the natural frequency $\omega_n = \sqrt{k/m}$ and the damping ratio $\xi = C/2m\omega_n$, then

$$\frac{y}{x} = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2} \tag{3.13}$$

This transfer function represents a second order low-pass filter with a natural frequency ω_n and a damping ratio ξ. Figure 3.15 shows a Bode plot of the transfer function where the overshoot on the natural frequency is determined by the damping ratio ξ . The roll-off at high frequency is $-40dB/decade$ which gives a higher vibration isolation performance [8].

Example 3.4

Assume the linear hydraulic actuator shown in 3.16 where the input is the force caused by the fluid displacement x and the output is the piston rod displacement y. The load mass is $1000kg$, the hydraulic stiffness of the oil is $k = 1 \times 10^5 N.s^2/m$ and the viscous damping in the piston is $C = 10N.s/m$
a- Draw a block diagram representing the whole system
b- Find the second order linear dynamic differential equation
c- Find the transfer function between x and y as a function of the natural frequency and damping ratio.

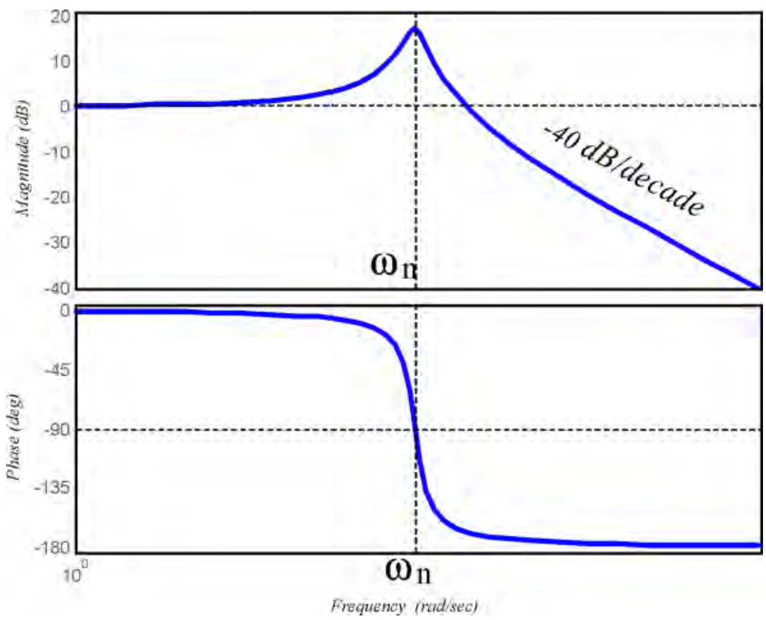

Figure 3.15: *Frequency response function of second order Low-pass filter*

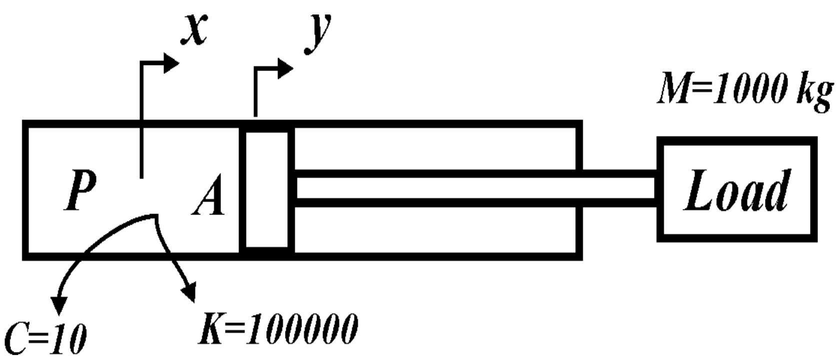

Figure 3.16: *Model for a double acting linear actuator*

Solution

The block diagram of the actuator is shown in Figure 3.17

The linear actuator can be represented by the free body diagram shown in Figure 3.18 where both plots on the right and on the left are the same and can be represented by the equation of motion:

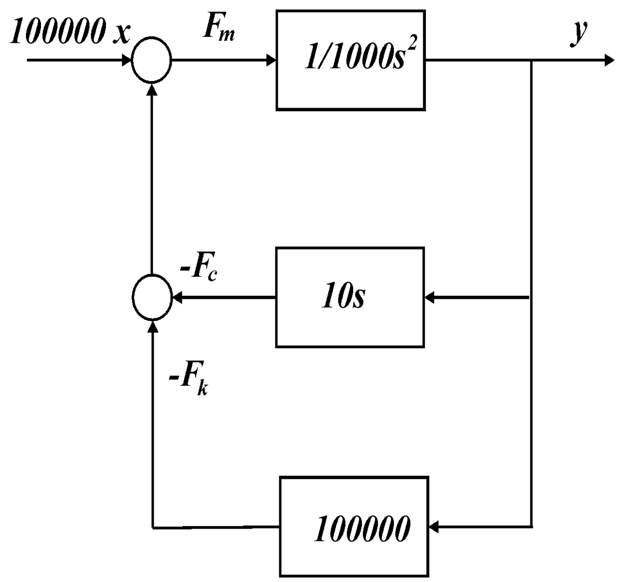

Figure 3.17: *Second order block diagram of a piston*

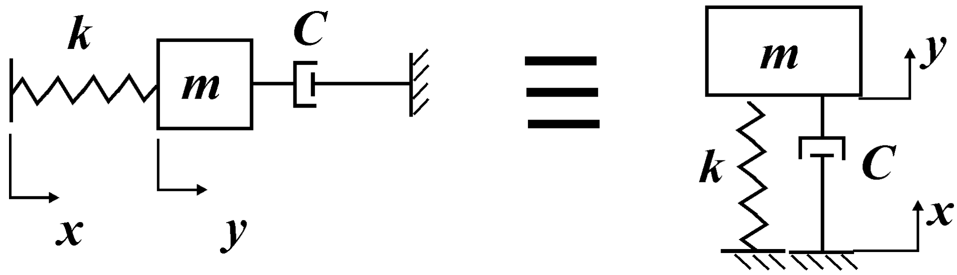

Figure 3.18: *Free body diagram for a double acting linear actuator*

$$ms^2y + Csy + ky = kx$$

or

$$(ms^2 + Cs + k)y = kx$$

The transfer function between x and y is

$$\frac{y}{x} = \frac{k}{ms^2 + Cs + k}$$

or

$$\frac{y}{x} = \frac{k/m}{s^2 + (C/m)s + (k/m)}$$

The transfer function represented in numerical form reads

$$\frac{y}{x} = \frac{100000/1000}{s^2 + (10/1000)s + (100000/1000)}$$

in more simplified from

$$\frac{y}{x} = \frac{100}{s^2 + (0.01)s + (100)}$$

but it known that the natural frequency $\omega_n = \sqrt{k/m} = \sqrt{100000/1000} = \sqrt{100} = 10rad/s$ and the damping ratio $\xi = C/2m\omega_n = 10/(2 \times 1000 \times 10) = 0.0005$.

Thus, the transfer function can be represented in terms of the natural frequency and damping ratio as follows:

$$\frac{y}{x} = \frac{\omega_n^2}{s^2 + 2\xi\omega_n s + \omega_n^2}$$

or in numerical form

$$\frac{y}{x} = \frac{10^2}{s^2 + 2(0.0005)(10)s + 10^2}$$

or simplified as

$$\frac{y}{x} = \frac{100}{s^2 + (0.01)s + 100}$$

This transfer function can be represented in the Bode plot (frequency response function) shown in Figure 3.19 where the overshoot appears on the natural frequency at 10 rad/sec. the magnitude at the overshoot can be calculated from the transfer function where the natural frequency coincides with the perturbation frequency
$s = j\omega_n = \sqrt{-1}\omega_n = \sqrt{-1} \times 10$
at natural frequency $s^2 = (-1)(10)^2$ the magnitude of transfer function is

$$\frac{y}{x} = \frac{(100)}{(-100) + (0.01)(10) + (100)} = 1000 = 60dB$$

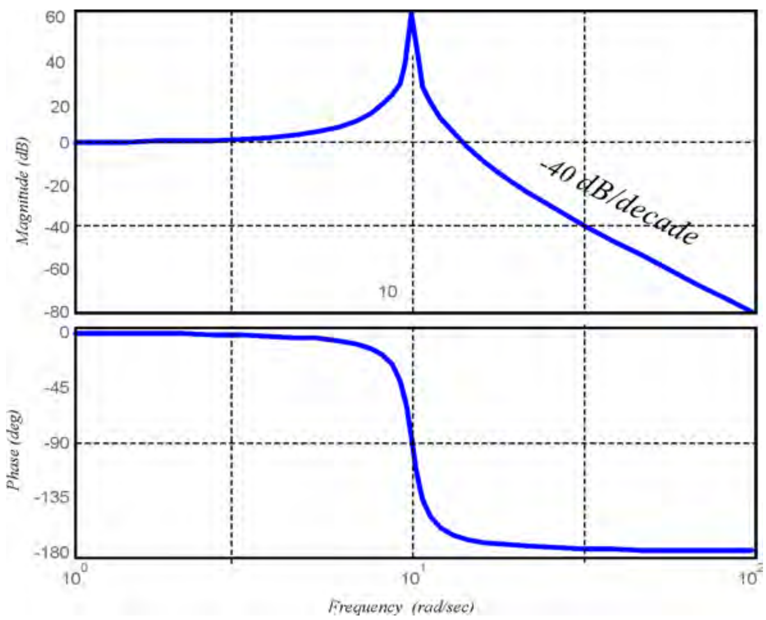

Figure 3.19: *Frequency response function of a linear actuator*

3.6 State space approach

The block diagram in Figure 3.20 depicts a state space modelling of a general dynamic system. A system can be represented by a set of first order linear differential equations [6]:

$$\dot{x} = Ax + Bu \tag{3.14}$$

$$y = Cx + Du \tag{3.15}$$

where

u = input vector.
y = output vector.
x = state vector.
A = system matrix.
B = input matrix.
C = output matrix.
D = feedthrough matrix.

The state vector is not unique and can be selected from the variables that can give information about the system. The simple oscillator shown in Figure 3.20 consists of a mass m, a spring k and a damping coefficient C.

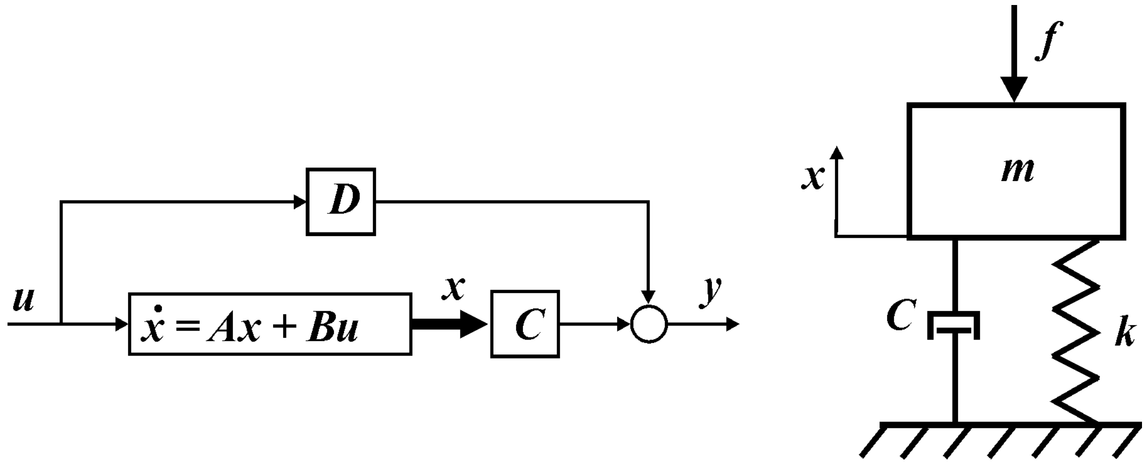

Figure 3.20: *State space representation of a simple oscillator*

The second order equation of motion of the simple oscillator reads

$$\ddot{x} + 2\xi\omega_n\dot{x} + \omega_n^2 x = \frac{1}{m}f \tag{3.16}$$

The state variables can be selected as the displacement x and the velocity $\dot{x}$ as follows

$$x_1 = x$$

$$x_2 = \dot{x}$$

then the derivatives of the state variables are

$$\dot{x}_1 = \dot{x} = x_2$$

$$\dot{x}_2 = \ddot{x} = -2\xi\omega_n\dot{x} - \omega_n^2 x + \frac{1}{m}f$$

In matrix form

$$\begin{pmatrix} \dot{x}_1 \\ \dot{x}_2 \end{pmatrix} = \begin{pmatrix} \dot{x} \\ \ddot{x} \end{pmatrix} = \overbrace{\begin{pmatrix} 0 & 1 \\ -\omega_n^2 & -2\xi\omega_n \end{pmatrix}}^{A} \begin{pmatrix} x \\ \dot{x} \end{pmatrix} + \overbrace{\begin{pmatrix} 0 \\ \frac{1}{m} \end{pmatrix}}^{B} f \tag{3.17}$$

The system matrix here reads

$$A = \begin{pmatrix} 0 & 1 \\ -\omega_n^2 & -2\xi\omega_n \end{pmatrix}$$

and the input matrix reads

$$B = \begin{pmatrix} 0 \\ \frac{1}{m} \end{pmatrix}$$

If the measured output is the displacement x, $y = x_1 = x$, then the output matrix reads

$$C = \begin{pmatrix} 1 & 0 \end{pmatrix}$$

If the measured output is the velocity $\dot{x}$, $y = x_2 = \dot{x}$, then the output matrix reads

$$C = \begin{pmatrix} 0 & 1 \end{pmatrix}$$

In these two cases, the feedthrough matrix D is a zero matrix $D = (0)$ which means that there is no direct influence of the input on the output. If the measured output is the acceleration, $y = \dot{x}_2 = \ddot{x}$, then the feedthrough matrix $D = 1/m$ and the output equation reads

$$y = \underbrace{\begin{pmatrix} -\omega_n^2 & -2\xi\omega_n \end{pmatrix}}_{C} \begin{pmatrix} x \\ \dot{x} \end{pmatrix} + \underbrace{\frac{1}{m}}_{D} f$$

This shows that the output matrix is

$$C = \begin{pmatrix} -\omega_n^2 & -2\xi\omega_n \end{pmatrix}$$

and the feedthrough matrix is

$$D = \frac{1}{m}$$

Once getting the state space four matrices (A, B, C and D), one can calculate the set of transfer functions relating the inputs u and the outputs y using the following equation:

$$y(s) = [C(sI - A)^{-1}B + D]u(s) \tag{3.18}$$

Where $u(s)$ and $y(s)$ are the input and output respectively as functions of Laplace variable in Laplace transform, s is the Laplace variable of the disturbance frequency on the imaginary axis $s = j\omega$ and I is the identity matrix (a matrix with ones on its diagonal and zeros for the other elements). This results in a number of transfer functions equals to the multiplication of the number of inputs by the number of outputs. If the number of inputs is u and the number of outputs y then the number of transfer functions TF is:

$$TF = u \times y$$

Considering the input vector $u = \{u_1, u_2, u_3, ...\}$ and the output vector $y = \{y_1, y_2, y_3, ...\}$, The set of transfer functions become

$$TF = \{(\frac{y_1}{u_1}), (\frac{y_1}{u_2}), (\frac{y_1}{u_3}), ..., (\frac{y_2}{u_1}), (\frac{y_2}{u_2}), (\frac{y_2}{u_3}), ..., (\frac{y_3}{u_1}), (\frac{y_3}{u_2}), (\frac{y_3}{u_3}), ...\}$$

So far it has been discussed how to obtain the different transfer functions from state space matrices in a manual way. When the system has more than three degrees of freedom, this calculation by hand becomes almost impossible, thus, there is a need to use a numerical technique. MATLAB software is one of the best solutions for this calculation where its control toolbox is programmed to calculate these transfer functions. First step to calculate the transfer function by Matlab is to separate the required functional matrices by selecting the inputs and outputs, a new set of matrices $[A1, B1, C1, D1]$ can be defined by selecting them from the original four matrices defining the input and output using the function ($ssselect$):

$$[A1, B1, C1, D1] = ssselect(A, B, C, D, u_1, y_3)$$

In this function $[A1, B1, C1, D1]$ is a set of state space matrices that define the transfer function between the first input u_1 and the third output y_3.

This set of state space matrices can be transferred into the rational shape of transfer function (n/d) = (numerator/denominator) as follows

$$[n, d] = ss2tf(A, B, C, D, 1, 3)$$

In order to print this transfer function on the command editor as a function of Laplace variable s, one can use the function $printsys(n, d)$. To present this transfer function in frequency domain using Bode plot, one can use the function $bode(A1, B1, C1, D1)$ or $bode(n, d)$; both give the same plot.

Example 3.5

Consider the two piston system shown in Figure 3.21. The inputs to the system are the two hydraulic forces F_1 and F_2 and the outputs are the two displacements x_1 and x_2. m_1 and m_2 are the two masses of the loads with the connected rods. k_1 and C_1 are the hydraulic stiffness and the viscous damping of piston (1) respectively and k_2 and C_2 are the hydraulic stiffness and the viscous damping of piston (2) respectively. Determine the state space matrices A, B, C and D in symbolic form.
Assume the following numbers for the masses, the stiffnesses and the damping coefficients:
$m_1 = 1$, $k_1 = 1000$, $C_1 = 1.5$, $m_2 = 1$, $k_2 = 10000$, $C_2 = 1$

draw the Bode plots for the transfer functions x_1/F_1 and x_2/F_2.

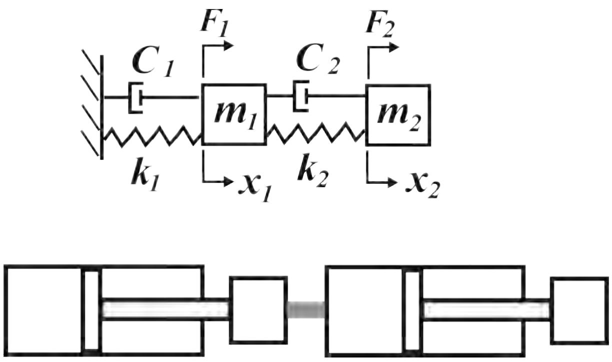

Figure 3.21: *Two piston system for example 3.5*

Solution

This is a two degrees of freedom system that can be represented by two second order differential equations

$$m_1\ddot{x}_1 = F_1 - k_1x_1 - C_1\dot{x}_1 + k_2(x_2 - x_1) + C_2(\dot{x}_2 - \dot{x}_1)$$

$$m_2\ddot{x}_2 = F_2 - k_2(x_2 - x_1) - C_2(\dot{x}_2 - \dot{x}_1)$$

Solving for the accelerations $\ddot{x}_1$ and $\ddot{x}_2$

$$\ddot{x}_1 = \frac{1}{m_1}F_1 + x_1\left(\frac{-k_1 - k_2}{m_1}\right) + \dot{x}_1\left(\frac{-C_1 - C_2}{m_1}\right) + x_2\left(\frac{k_2}{m_1}\right) + \dot{x}_2\left(\frac{C_2}{m_1}\right)$$

$$\ddot{x}_2 = \frac{1}{m_2}F_2 + x_1\left(\frac{k_2}{m_2}\right) + \dot{x}_1\left(\frac{C_2}{m_2}\right) + x_2\left(\frac{-k_2}{m_2}\right) + \dot{x}_2\left(\frac{-C_2}{m_2}\right)$$

Now building the state space equations from the equations of motion

$$\begin{pmatrix} \dot{x}_1 \\ \ddot{x}_1 \\ \dot{x}_2 \\ \ddot{x}_2 \end{pmatrix} = \begin{pmatrix} 0 & 1 & 0 & 0 \\ \frac{-k_1-k_2}{m_1} & \frac{-C_1-C_2}{m_1} & \frac{k_2}{m_1} & \frac{C_2}{m_1} \\ 0 & 0 & 0 & 1 \\ \frac{k_2}{m_2} & \frac{C_2}{m_2} & \frac{-k_2}{m_1} & \frac{-C_2}{m_1} \end{pmatrix} \begin{pmatrix} x_1 \\ \dot{x}_1 \\ x_2 \\ \dot{x}_2 \end{pmatrix} + \begin{pmatrix} 0 & 0 \\ \frac{1}{m_1} & 0 \\ 0 & 0 \\ 0 & \frac{1}{m_2} \end{pmatrix} \begin{pmatrix} F_1 \\ F_2 \end{pmatrix}$$

The outputs are the displacements x_1 and x_2

$$\begin{pmatrix} x_1 \\ x_2 \end{pmatrix} = \begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \end{pmatrix} \begin{pmatrix} x_1 \\ \dot{x}_1 \\ x_2 \\ \dot{x}_2 \end{pmatrix} + \begin{pmatrix} 0 & 0 \\ 0 & 0 \end{pmatrix} \begin{pmatrix} F_1 \\ F_2 \end{pmatrix}$$

These four matrices can be entered into MATLAB to find the Bode plots by introducing values for the different variables:
$m_1 = 1$, $k_1 = 1000$, $C_1 = 1.5$, $m_2 = 1$, $k_2 = 10000$, $C_2 = 1$.
The two bode plots representing x_1/F_1 and x_2/F_2 are shown in Figure 3.22 and Figure 3.23, respectively

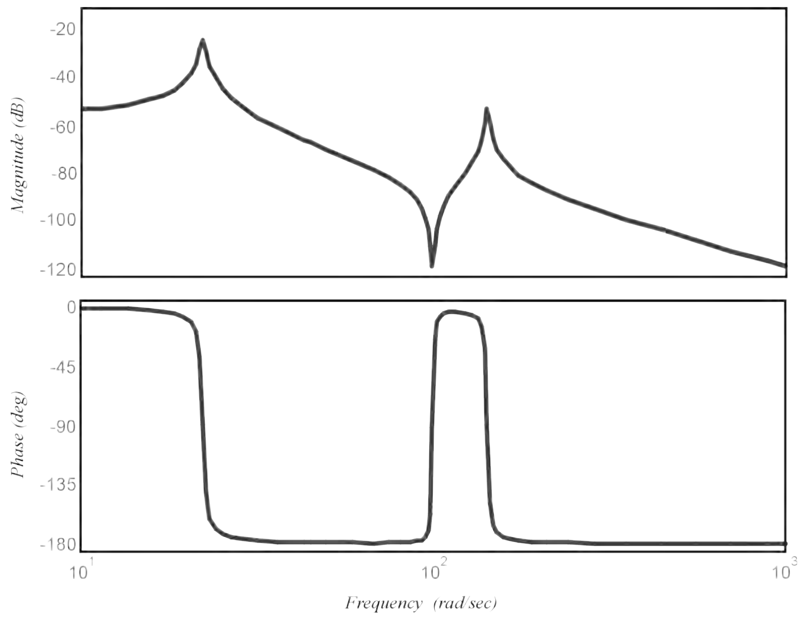

Figure 3.22: *Bode plot for the transfer function x_1/F_1*

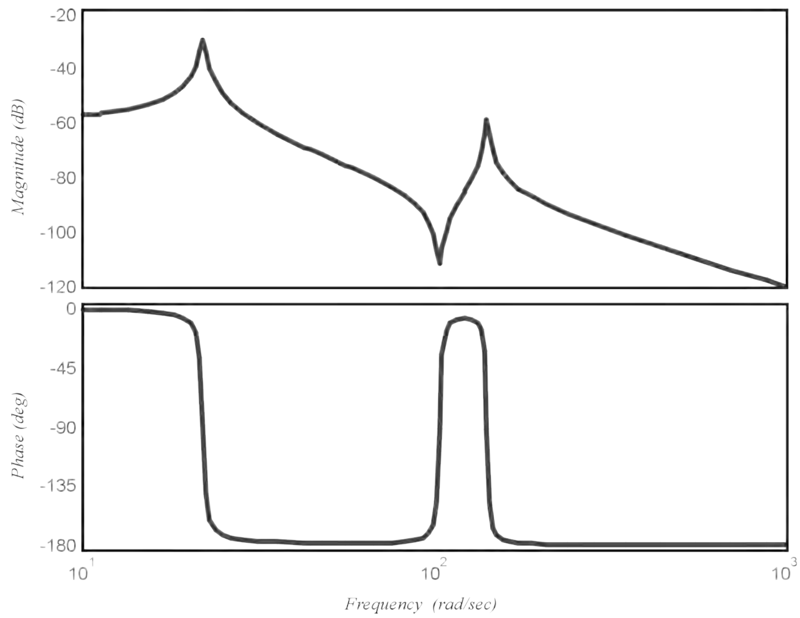

Figure 3.23: *Bode plot for the transfer function x_2/F_2*

3.7 Linear modelling of rotating elements

For angular motion, rotary actuators are needed, there are several types of rotary actuators; gear type, piston type and vane type. When the actuator is a rotary element, the equation of motion reads

$$I\ddot{\theta} + C_\theta\dot{\theta} + k_\theta\theta = T \tag{3.19}$$

or

$$I\alpha + C_\theta\omega + k_\theta\theta = T \tag{3.20}$$

where

I = Moment of inertia.
T = Torque
k_θ = Angular stiffness
C_θ = Angular damping
θ = Angle of rotation
$\omega = \dot{\theta}$ = Angular velocity
$\alpha = \ddot{\theta}$ = Angular acceleration

This model can be treated as the previous one, either by finding transfer functions or state space approach. The torque T can be calculated from

$$T = PAR$$

where P is the pressure in the system, A is the area on which the pressure P acts and R is the distance from the centre of pressure to the centre of rotation.

Example 3.6

Consider the two degrees of freedom industrial robot shown in Figure 3.24. The inputs to the system are the two torques of the hydraulic motors T_1 and T_2 and the outputs are the two angular displacements (angles) θ_1 and θ_2 and their derivatives that represent the velocity and acceleration at each joint. I_1 and I_2 are the two moments of inertia of the robot arms. k_1 and C_1 are the hydraulic stiffness and the viscous damping of hydraulic motor (1) respectively and k_2 and C_2 are the hydraulic stiffness and the viscous damping of the hydraulic motor (2) respectively. Determine the state space matrices A, B, C and D in symbolic form.

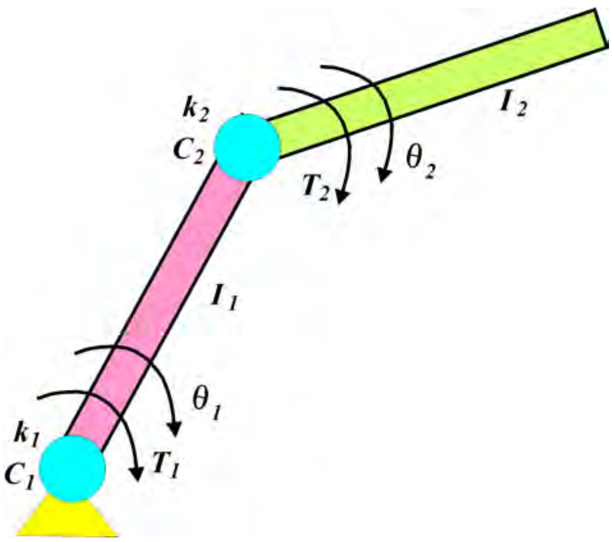

Figure 3.24: *Two arm robot for example 6.2*

Solution

This is a two degrees of freedom system that can be represented by two second order differential equations

$$I_1\ddot{\theta}_1 = T_1 - k_1\theta_1 - C_1\dot{\theta}_1 + k_2(\theta_2 - \theta_1) + C_2(\dot{\theta}_2 - \dot{\theta}_1)$$

$$I_2\ddot{\theta}_2 = T_2 - k_2(\theta_2 - \theta_1) - C_2(\dot{\theta}_2 - \dot{\theta}_1)$$

Solving for the accelerations $\ddot{\theta}_1$ and $\ddot{\theta}_2$

$$\ddot{\theta}_1 = \frac{1}{I_1}T_1 + \theta_1\left(\frac{-k_1-k_2}{I_1}\right) + \dot{\theta}_1\left(\frac{-C_1-C_2}{I_1}\right) + \theta_2\left(\frac{k_2}{I_1}\right) + \dot{\theta}_2\left(\frac{C_2}{I_1}\right)$$

$$\ddot{\theta}_2 = \frac{1}{I_2}T_2 + \theta_1\left(\frac{k_2}{I_2}\right) + \dot{\theta}_1\left(\frac{C_2}{I_2}\right) + \theta_2\left(\frac{-k_2}{I_2}\right) + \dot{\theta}_2\left(\frac{-C_2}{I_2}\right)$$

Now building the state space equations from the equations of motion

$$\begin{pmatrix} \dot{\theta}_1 \\ \ddot{\theta}_1 \\ \dot{\theta}_2 \\ \ddot{\theta}_2 \end{pmatrix} = \begin{pmatrix} 0 & 1 & 0 & 0 \\ \frac{-k_1-k_2}{I_1} & \frac{-C_1-C_2}{I_1} & \frac{k_2}{I_1} & \frac{C_2}{I_1} \\ 0 & 0 & 0 & 1 \\ \frac{k_2}{I_2} & \frac{C_2}{I_2} & \frac{-k_2}{I_1} & \frac{-C_2}{I_1} \end{pmatrix} \begin{pmatrix} \theta_1 \\ \dot{\theta}_1 \\ \theta_2 \\ \dot{\theta}_2 \end{pmatrix} + \begin{pmatrix} 0 & 0 \\ \frac{1}{I_1} & 0 \\ 0 & 0 \\ 0 & \frac{1}{I_2} \end{pmatrix} \begin{pmatrix} T_1 \\ T_2 \end{pmatrix}$$

The outputs are the angular displacements, velocities and accelerations θ_1, $\dot{\theta}_1$, $\ddot{\theta}_1$, θ_2, $\dot{\theta}_2$ and $\ddot{\theta}_2$

$$\begin{pmatrix} \theta_1 \\ \dot{\theta}_1 \\ \ddot{\theta}_1 \\ \theta_2 \\ \dot{\theta}_2 \\ \ddot{\theta}_2 \end{pmatrix} = \begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ \frac{-k_1-k_2}{I_1} & \frac{-C_1-C_2}{I_1} & \frac{k_2}{I_1} & \frac{C_2}{I_1} \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ \frac{k_2}{I_2} & \frac{C_2}{I_2} & \frac{-k_2}{I_1} & \frac{-C_2}{I_1} \end{pmatrix} \begin{pmatrix} \theta_1 \\ \dot{\theta}_1 \\ \theta_2 \\ \dot{\theta}_2 \end{pmatrix} + \begin{pmatrix} 0 & 0 \\ 0 & 0 \\ \frac{1}{I_1} & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & \frac{1}{I_2} \end{pmatrix} \begin{pmatrix} T_1 \\ T_2 \end{pmatrix}$$

3.8 Modelling of control valves

A control valve is the main element in fluid power system. The most common control valve is the spool valve shown in Figure 3.25. It consists of a spool with circular cross section slides inside a cylindrical casing. The spool is made of lands connected together by central rods. The general equation of motion of this type of valves can be represented by the following equation of motion, [9], [10]

$$P_1A_1 - P_2A_2 = m\ddot{x} + C\dot{x} + kx \tag{3.21}$$

where

P_1 = Pilot pressure acting on the left land of the spool.
P_2 = Pilot pressure acting on the right land of the spool.
A_1 = Cross sectional area of the left land of the spool.
A_2 = Cross sectional area of the right land of the spool.
m = The mass of the sliding spool.
C = The viscous damping coefficient in the valve.
k = The stiffness factor of the fluid.

The pressure values included in equation (3.21) are the low pilot pressure values of the pilot fluid used to move the spool and not the high system pressure used to move the load or the piston. Looking at the port from which fluid passes in or out of the valve with high pressure, the relationship between the flow rate and the pressure which represents the steady state response can be determined as follows

$$Q = C_d A \sqrt{\frac{2\Delta P}{\rho}} \tag{3.22}$$

where

Q = The fluid flow rate through the port.
ΔP = The difference of high pressure passing through the valve.
C_d = The discharge coefficient of the orifice (port) which is typically 0.6 to 0.65 for laminar flow and 0.7 to 0.8 for turbulent flow [13].
A = Cross sectional area of the port.
ρ = The fluid density .

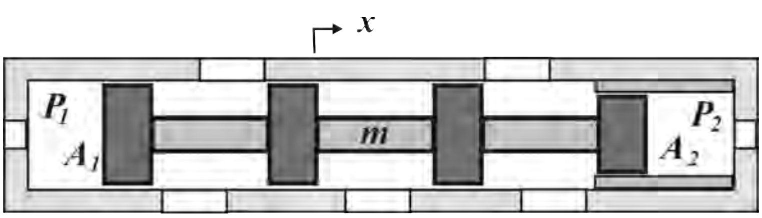

Figure 3.25: *General design of spool valve*

Example 3.7

Consider a two-way spool valve in hydraulic system where the port is a square shape with 5 mm dimensions. The pressure P_1 at the inlet is equal to 1.7×10^6 MPa and at the outlet equals to 1.5×10^6 MPa. The flow is laminar and the discharge coefficient $C_d = 0.6$. The spool slides through the valve to open and close the port depending on the demand of the flow rate. Draw a plot showing the relationship between the spool displacement and the flow rate through the valve. The oil density $\rho = 800 kg/m^3$

Solution

The area of the port A is given by the width $x = 5mm$ and the height ranges $y = 0 - 5mm$ due to the spool displacement.

$$A = x \times y$$

The flow rate is calculated by the equation:

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

or by numbers

$$Q = (0.6)(0.005)y \sqrt{\frac{2(1.7 \times 10^6 - 1.5 \times 10^6)}{800}}$$

or

$$Q = 0.0671y = K_v y$$

Where K_v is denoted the valve coefficient. Figure 3.26 shows this relationship between the flow rate and the spool displacement

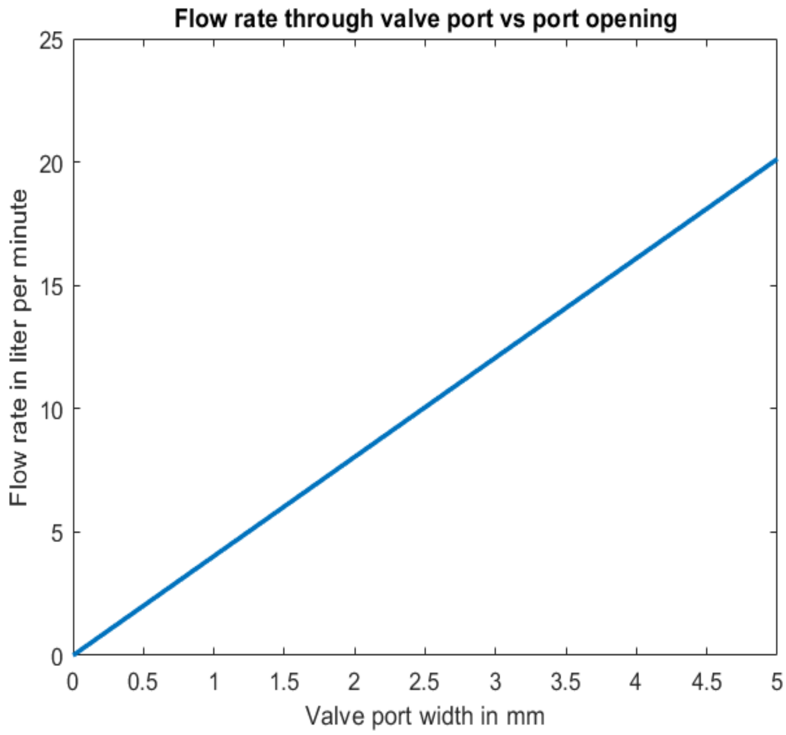

Figure 3.26: *Flow rate vs spool displacement in spool valve for example 3.7*

3.9 Non-linear modelling

So far, all the discussed different models have been considered as linear models. The rapid change of a non-linear load accompanied with the need for a variable displacement pump that leads to variable pressure can be a strong reason for the non-linearity of fluid power systems. Furthermore, non-linearity is added to the system due to dry friction between the sliding pistons and the casing cylinders. This non-linearity led some of the researchers to use non-linear control techniques like H-infinity, Neural Networks and Fuzzy logic control to increase the control performance and robustness like those in [11], [12].

Considering the double acting linear hydraulic actuator shown in Figure 3.27 the input flow rate to the actuator Q_1 is equal to the output flow rate Q_2 and the piston area on the left A_1 is equal to the area at the right A_2; according to the continuity equation, thus:

$$Q = Q_1 = Q_2$$

$$A = A_1 = A_2$$

The nominal flow rate to and from the piston should be equal to $A\dot{y}$, the compressibility influence is $\dot{P}/K$ and the leakage effect on the flow rate is $C\Delta P$. this leads to the fact that the input and output flow rates are

$$Q_1 = A\dot{y} + \frac{\dot{P}_1}{K} + (P_1 - P_2)C$$

$$Q_2 = A\dot{y} + \frac{\dot{P}_2}{K} + (P_1 - P_2)C$$

Knowing that $Q_1 = Q_2 = Q$; results in

$$Q = A\dot{y} + \frac{(\dot{P}_1 - \dot{P}_2)}{2K} + (P_1 - P_2)C \tag{3.23}$$

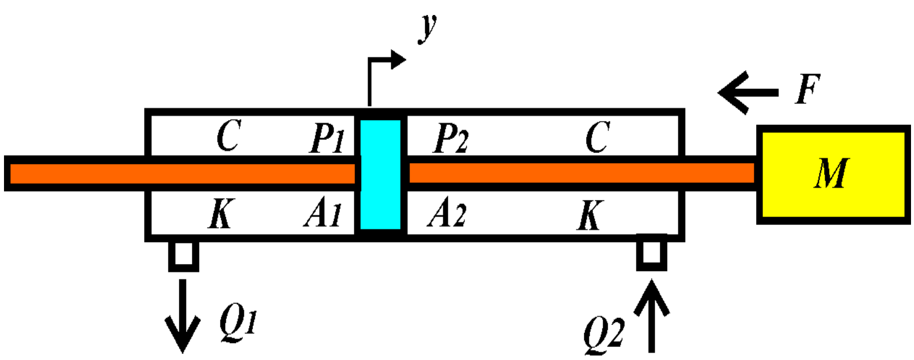

Figure 3.27: *Hydraulic actuator*

To determine the differential equation of motion consider the hydraulic actuator in Figure 3.27. The main problem causing non-linearity here is the damping coefficient that comes from friction damping which depends on the path and increases its order during the travel stroke of the actuator. Assuming that $P = P_1 - P_2$, the equation of motion in this case reads

$$M\frac{d^2y}{dt^2} = PA - (K + a_1)y - (C + a_2)\frac{dy}{dt} - a_3\frac{d^2y}{dt^2} - a_4\frac{d^3y}{dt^3} - F \qquad (3.24)$$

Where the coefficients (a_1, a_2, a_3, a_4) can be determined experimentally or analytically using Taylor expansion [14]. More details about this non-linear effect is necessary for researchers and can be found in deeper discussions tackled in specific books.

3.10 Problems

1. Consider a linear hydraulic actuator (piston) with a cross-sectional area A supporting a load with a mass m (including the mass of the rod). The hydraulic stiffness of the fluid is k and the viscous damping coefficient is C. The volume flow rate of the fluid at the inlet Q causes a velocity $\dot{y}$ of the pistons rod.
   a- Find the transfer function between Q as an input and $\dot{y}$ as an output in Laplace transform in terms of the natural frequency ω, the damping ratio ζ and the area A.
   b- Draw an open-loop block diagram for the system showing details of all acting forces and components.

2. Consider the hydraulic system shown in Figure 3.28, where;

$$k = 10$$
$$H_1(s) = 4s$$
$$H_2(s) = \frac{2}{1+4s}$$
$$G(s) = \frac{8}{s(s+2)(s+3)}$$

   a- Find the relationship (transfer function) between x and y
   b- What is the value of transfer function at very low frequency and

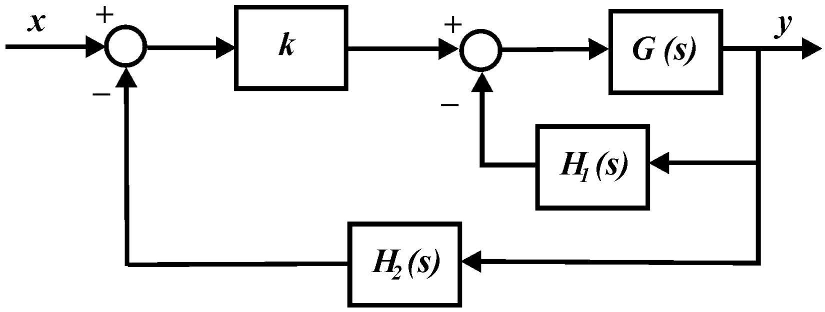

Figure 3.28: *Hydraulic system*

very high frequency

3. A hydraulic cylinder is used to support a mass M as shown in Figure 3.29. The natural frequency of the system is f_n (Hz), the damping ration is ζ, and the piston force is denoted by F. Using the force F as an input and the displacement x as an output and using Laplace transform:
a- Find the equation of motion governing the system.
b- Find the transfer function between F and x.
c- Find the state space matrices A, B, C, D. (take the displacement and the velocity of the ram as state variables).
d- Using a feedback control technique with integration compensator draw a detailed block diagram describing the closed-loop system.

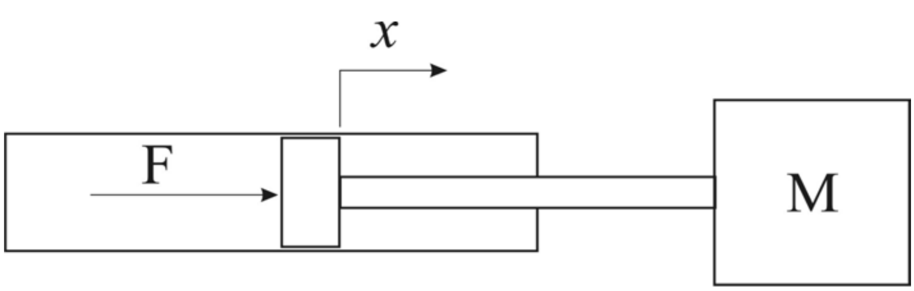

Figure 3.29: *Hydraulic actuator with load*

4. Consider a rotary pneumatic vane type actuator with an inlet air pressure P acting on a vane area A at a distance R from the centre of rotation. The actuator holds a load with Inertia J and needs a torque T related linearly to the input pressure. The pneumatic air stiffness is H N/m and the pneumatic damping is coefficient D $N.s/m$.

   a- Find the equation of motion of the system.
   b- Find the transfer function between the input pressure P and the output angular rotational displacement of the motor θ.
   c- Find the state space matrices taking the angular displacement and angular velocity as state variables using the input P and the output as the angular acceleration of the motor $\ddot{\theta}$.

5. In the quarter-car model shown in Figure 3.30, $M_1 = 100kg$ is the mass of the wheel and chassis, $K_1 = 1 \times 10^5 N/m$ and $C_1 = 5N.s/m$ are the pneumatic stiffness and damping of the wheel. $M_2 = 250kg$ is the mass of the quarter-car, $K_2 = 1.5 \times 10^6$ and $C_2 = 10N.s/m$ are the spring stiffness and viscous damping of the suspension system. F_1 is the ground corrugation force causing wheel displacement Y_1 and F_2 is the engine harmonic force causing body displacement Y_2. Write the differential equations of motion and find the state space matrices taking Y_1 and Y_2 and their derivatives as state variables.

6. Consider the spool valve shown in Figure 3.31 with a spool mass m and the absolute spool displacement x. $A_1 = A_2 = 8 \times 10^{-5}$ and the pressure difference $P_1 = 65$ bar and $P_2 = 52$ bar. Assume $C = 7N.s/m$ and $k = 1.2 \times 10^5 N/m$ as the damping and stiffness of oil respectively.
   a- Design the block diagram for this system.
   b- Find the transfer function between the pressure difference P as an input and x as an output.
   c- Draw the Bode plot for the previous transfer function.

7. A 3-meters height hydraulic elevator consists of electric motor on the

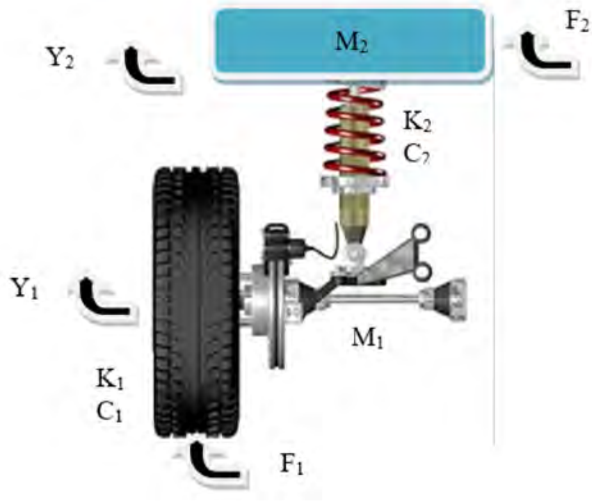

Figure 3.30: *Quarter car suspension*

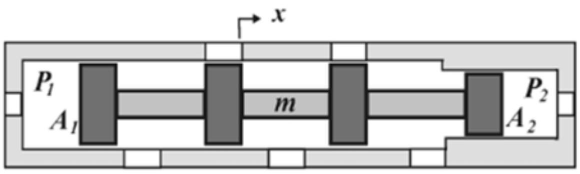

Figure 3.31: *Spool valve*

ground, oil tank and 4 meters long, 25 mm diameter cast iron pipes on the ground, oil (hydraulic stiffness = 1.5×10^7 N/m, kinetic viscosity = 100 cSt, specific gravity S.G = 0.9, damping coefficient = 1000 N.s/m), hydraulic pump (efficiency = 85%) on the ground near the tank, hydraulic linear actuator (150 mm bore and 100 mm rod diameter) on the ground and cabin (500 kg mass with 0.1 m/s required speed), determine:

a- The volume of the tank (best design)
b- Pressure and flow rate in the actuator
c- Pressure and flow rate of the pump (include friction losses)
d- Power of the electric motor
e- Draw block diagram to represent the whole system with pump displacement as the main input and the actuators displacement as the output
f- Find the transfer function between the pumps displacement and

actuators displacement
g- Draw Bode plot (by hand) for the transfer function in the previous point and find the natural frequency and damping ratio
h- Draw the hydraulic circuit with all necessary components in symbolic form

8. If a system is represented by the equation of motion

$$450\ddot{x}(t) + 105\dot{x}(t) + 10x(t) = f(t)$$

Find the expression of the time response if the system is subject to a unit input force. Take zero boundary conditions.

9. The 3-arm hydraulic robot shown in Figure 3.32 consists of three arms actuated by three hydraulic actuators:

a- Determine the three equations of motion of the system
b- Determine the four matrices (A, B, C, D) using the state space approach taking the three angular displacements and three angular velocities as the state variables
c- Calculate the nine transfer functions between the three forces of the actuators as inputs and the three angular accelerations of the arms using the analytical form
d- Calculate the time response functions for the three angular displacements

10. Considering the 3-arm hydraulic robot shown in Figure 3.32:

a) Draw the nine transfer functions calculated in frequency domain using MATLAB software
b) Plot the time domain responses of the three angular displacements on MATLAB
c) Change the values of hydraulic stiffness and damping of the actuators and report on the MATLAB results

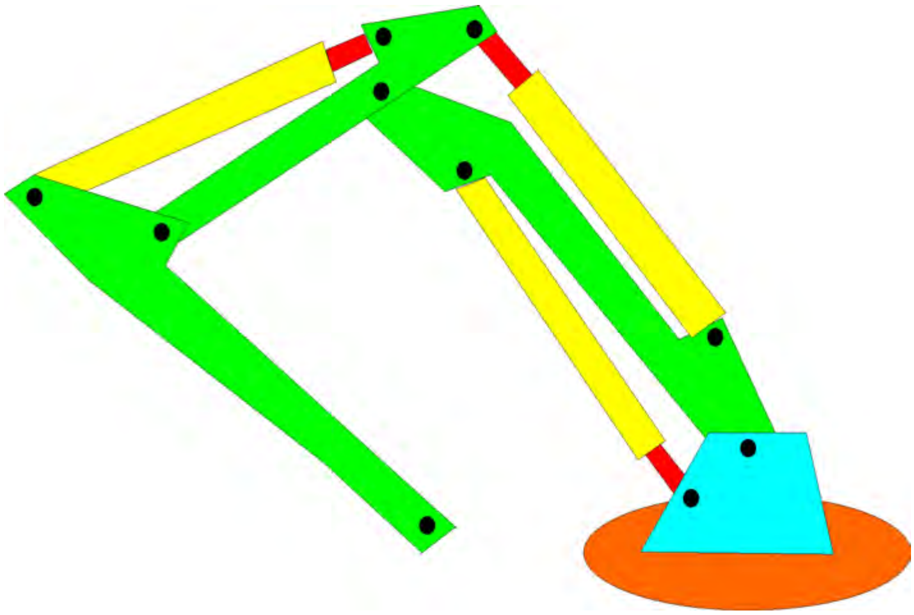

Figure 3.32: *3-DOF hydraulic robot*

11. Write a MATLAB program to simulate the behaviour of chain bulldozer knowing that the driving chain is driven by a hydraulic rotating motor and the loader is driven by linear hydraulic actuators. Look for a specific type of chain loader and insert it dimensions and characteristics in the problem.

12. Derive the non-linear equation of motion of the hydraulic actuator shown in Equation 3.24.

3.11 References

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Chapter 4

Control of Fluid Power Systems

Fluid power systems stay useless unless they are organized and controlled to be directed to specific mechanical and industrial applications. Control techniques depend strongly on the deep understanding of the system and the type of fluid used [1]. Control system can be one the following:

- **Mechanical control:** by utilizing the fluid itself at low pressure to actuate the control valves that in turn control the high pressure fluid and pass it to move the actuators to move a specific application.
- **Electrical control:** that can be analog or digital. This way of control utilizes the signals measured by sensors near the load or any signal fed directly to actuate linear or rotary electric motors whose motion is used to actuate the control valve. Here, the control valve is also used to move the end effecting actuators.

4.1 Servo Control systems

Usually, the control systems include both electrical and mechanical components. The most common control system is the *Servo* control system. A control system is said to be servo if the input signal is amplified (*Conditioning*) and if there is a signal fed back from the output to the input in a closed-loop manner (*Feedback*). The closed-loop feedback system shown in Figure 4.1 is usually called $follow - up$ system that aims at keeping the output at a given value by minimizing the error.

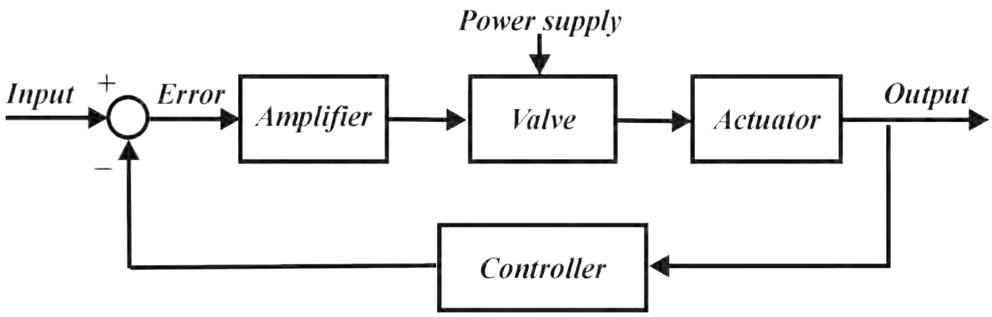


Figure 4.1: *Block diagram of closed-loop feedback servo system*

A follow-up hydraulic closed-loop feedback servo system undergoes two types of systems:

1. Valve operated servo control system: used for lower power applications.

2. Pump operated servo control system: used for high power applications.

4.1.1 Valve operated servo control

A valve operated servo control system is usually used for lower power applications. It operates with a constant displacement pump and the valve acts as an orifice which increases the resistance in the system increasing the temperature and decreasing the performance [2]. Generally, a valve operated servo control system has the following characteristics:

- The circuit is easily designed and constructed with simple components.
- It has a rapid dynamic response because of having lower inertia.
- One single pump can be enough to give power for the whole system whereas valves are distributed amongst the different actuators and applications.

One of the most important concepts in valves is the valve lap where the spool valve can have zero lap, underlap or overlap as shown in Figure 4.2. The influence of lap types on pressure and flow rate across the valve is shown in Figure 4.3.

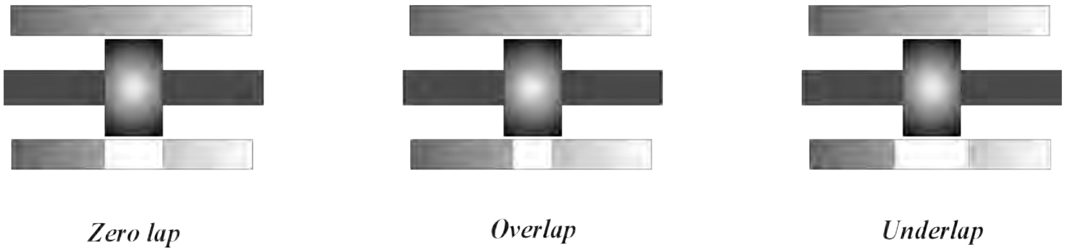


Figure 4.2: *Valve lap (design of valve port and spool land)*

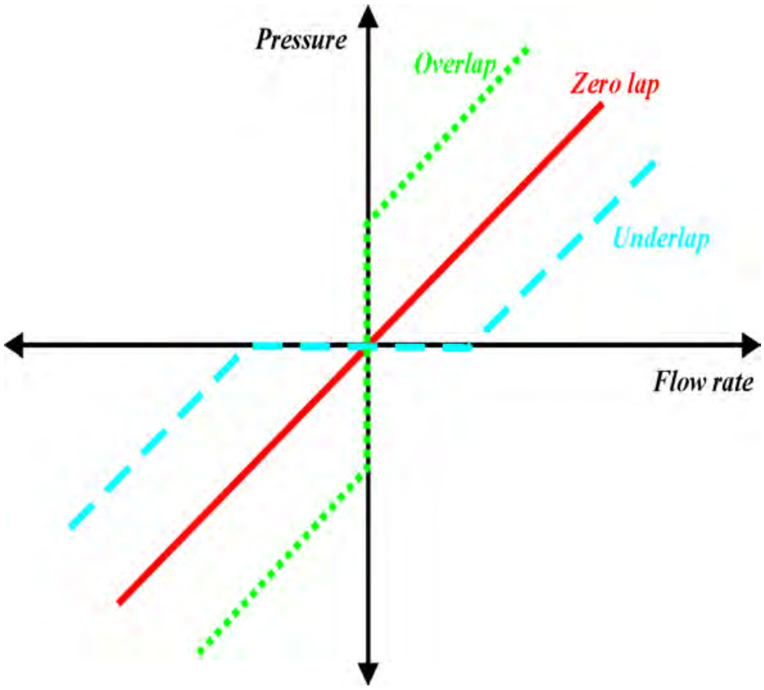


Figure 4.3: *Pressure and flow rate influenced by valve lap*

- *Zero lap:* the width of the spool land is exactly the same as the width of the port which leads to a proportional relation between the displacement of the spool and the quantity of fluid flowing through the valve. This is practically difficult to manufacture and not easy to obtain.

- *Overlap:* the width of the land is bigger than that of the port which leads to having a dead zone where there is a movement of the spool without having any flow in this region.

- *Underlap:* the width of the land is smaller than that of the port leading to a continuation in flow even when the spool land is on the mid point of the port (null point).

Since it is not easy to manufacture a zero lap spool because of the need to high accuracy, this design is usually replaced by an overlap with notches on the edge of the land. The underlap design is usefull in achieving a high response at null point and is used to compensate for the loss in fluid when there is a leakage in the system.
The discharge equation in the valve reads

$$Q = C_d A \sqrt{\frac{2\Delta P}{\rho}} \tag{4.1}$$

where

$Q =$ The fluid flow rate through the port.
$\Delta P =$ The difference of high pressure passing through the valve.
$C_d =$ The discharge coefficient of the orifice (port).
$A =$ Cross sectional area of the port.
$\rho =$ The fluid density.

Example 4.1

Consider the valve servo copy machine shown in Figure 4.4. Determine the transfer function between the stylus displacement u and the tool holder displacement y.

Solution

The system consists of a spool valve and a double acting piston. Both, the spool of the valve and the rod of the piston are connected to the feedback lever arm. The end of the lever arm is influenced by the template input displacement u leading to the tool holder machining displacement y passing through the intermediate valve spool displacement x.

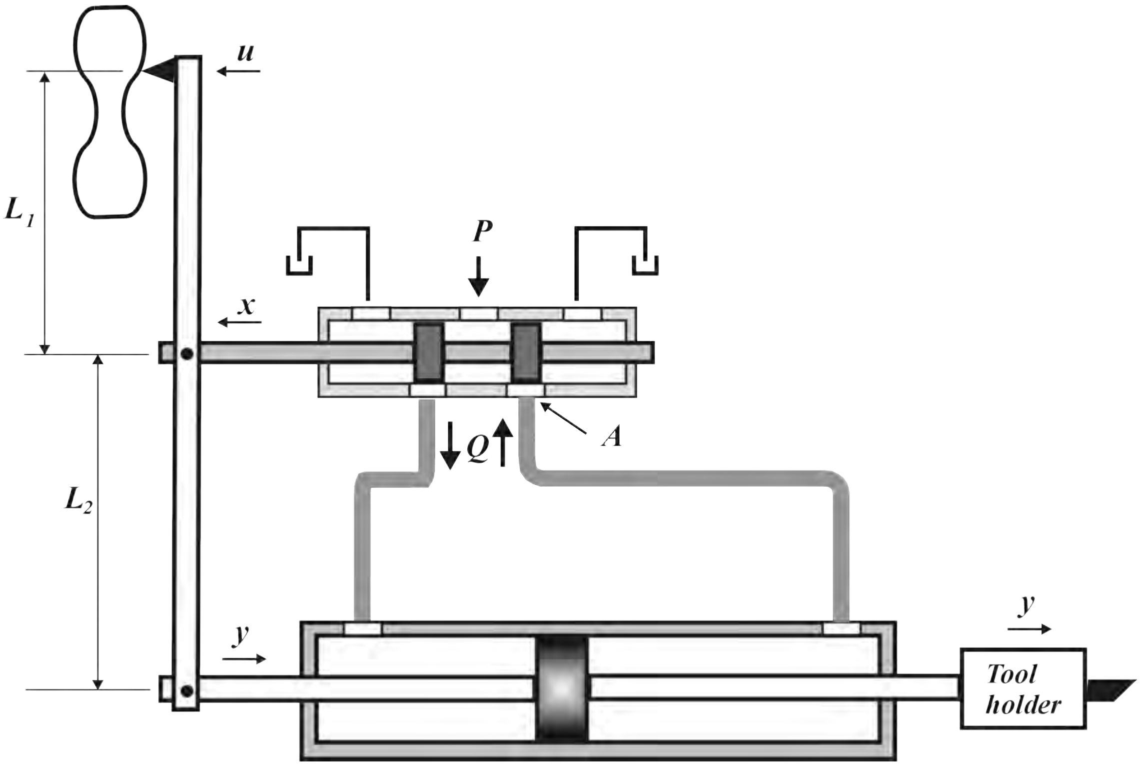


Figure 4.4: *Valve servo Copy mechanism for example 4.1*

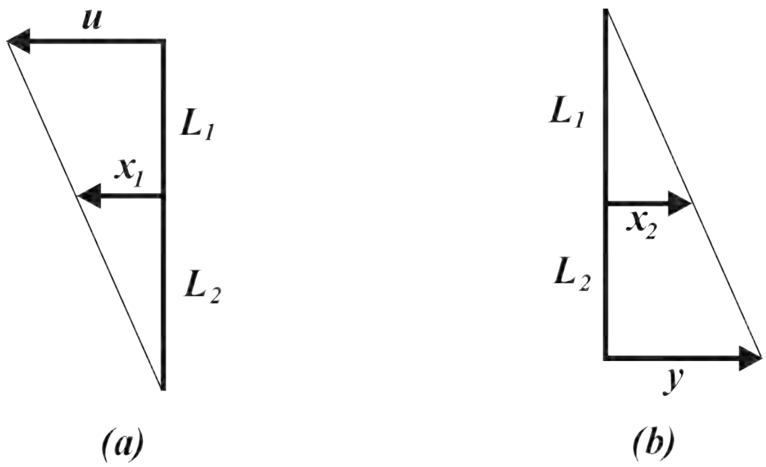


Figure 4.5: *Displacemenet triangles for example 4.1*

Looking at Figure 4.5(a), from similar triangles

$$\frac{x_1}{L_2} = \frac{u}{L_1 + L_2}$$

solving for x_1

$$x_1 = \frac{uL_2}{L_1 + L_2}$$

Looking at Figure 4.5(b), from similar triangles

$$\frac{x_2}{L_1} = \frac{y}{L_1 + L_2}$$

solving for x_2

$$x_2 = \frac{yL_1}{L_1 + L_2}$$

The spool displacement x

$$x = x_1 - x_2 = \frac{uL_2}{L_1 + L_2} - \frac{yL_1}{L_1 + L_2}$$

for a special case when $L_1 = L_2$

$$x = \frac{u - y}{2}$$

If there is no leakage in the flow considering a zero lap servo valve, the discharge equation reads

$$Q = C_d x d \sqrt{\frac{2\Delta P}{\rho}}$$

where d is the average width of the port (area of the port is $A = xd$). For incompressible hydraulic fluid with a constant pressure input to the valve

$$Q = K_v x$$

where $K_v = C_d d\sqrt{2\Delta P/\rho}$ is the valve constant at specific constant pressure and specific density. On the other hand, the flow rate from the valve to the piston is calculated from

$$Q = A\frac{dy}{dt}$$

Equating the two equations of flow,

$$K_v x = A\frac{dy}{dt}$$

or

$$K_v \frac{u-y}{2} = A\frac{dy}{dt}$$

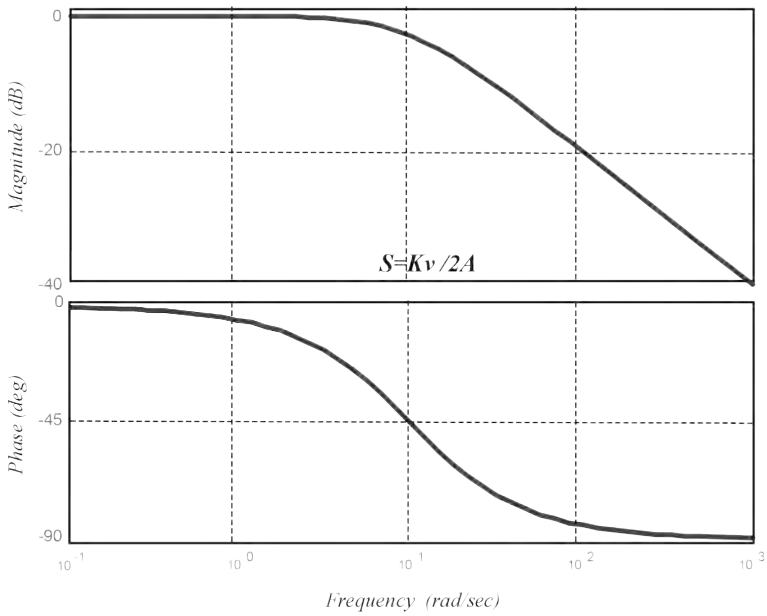


Figure 4.6: *Bode plot for the first order valve servo copy machine*

Applying Laplace transform with zero initial conditions

$$K_v(u(s) - y(s)) = 2Asy(s)$$

using separation of variables to solve the first order equation

$$K_v u(s) = y(s)(K_v + 2As)$$

Solving for the transfer function y/u

$$\frac{y(s)}{u(s)} = \frac{K_v}{(K_v + 2As)}$$

or

$$\frac{y(s)}{u(s)} = \frac{1}{(1 + \frac{2A}{K_v}s)}$$

$$\frac{y(s)}{u(s)} = \frac{1}{(1 + \tau s)}$$

This transfer function is a first order frequency response function with a time constant $\tau = 2A/K_v$. The Bode plot for this transfer function is shown in Figure 4.6.

Example 4.2

Consider the system shown in Figure 4.7. The system consists of a linear hydraulic actuator with a piston side area ($A_P = 8 \times 10^{-3}m^2$) and a rod side area ($A_R = 0.5A_P$), the actuator is required to drive a load of $1000kg$ through a variable distance y. The actuator is operated by a four port, three position, solenoid controlled, directional control spool valve where the width of the port $d = 10mm$ and a discharge coefficient $C_d = 0.8$, the pressure drop across the port $\Delta P = 1kPa$ and the oil density $\rho = 800kg/m^2$. The solenoid moves the spool a variable distance x to control the actuator corrected by a feedback signal comes from a displacement sensor on the load to the current of the solenoid ($I = y - x$).
1- Find the transfer function between I as an input and y as an output in extension and retraction strokes.
2- Knowing that the oil hydraulic stiffness is $k = 1 \times 10^6 N/m$ and the damping coefficient $C = 10N.s/m$, find the transfer function between the pressure input to the actuator and the displacement y of the load as an output in extension and retraction strokes.

Solution

1- transfer function y/I:
The flow rate coming out of the spool valve is

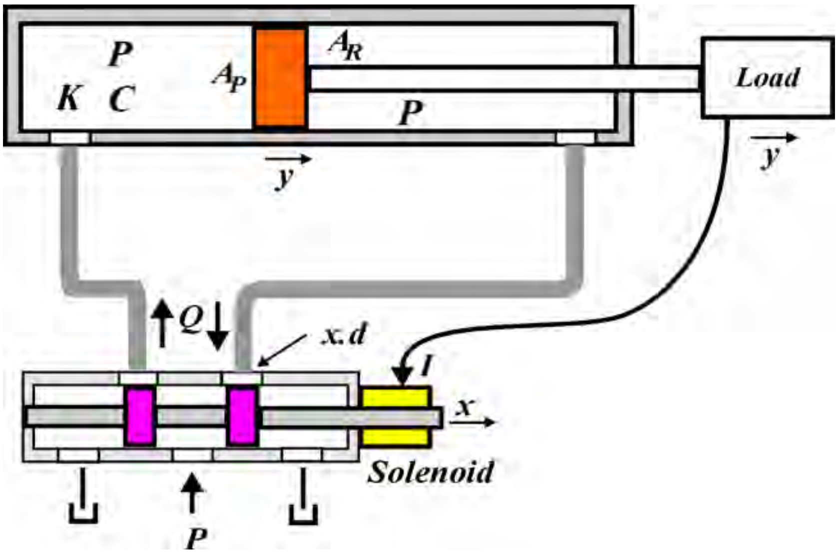


Figure 4.7: *spool valve operating a linear actuator*

$$Q = C_d x d \sqrt{\frac{2\Delta P}{\rho}} = K_v x$$

where

$$K_v = C_d d \sqrt{\frac{2\Delta P}{\rho}}$$

$$K_v = (0.8)(0.01)\sqrt{\frac{2(1000)}{800}}$$

$$K_v = 0.012$$

$$Q = (0.012)x = (0.012)(I - y)$$

The flow rate coming into the actuator is

$$Q = A_P \dot{y} = K_v(I - y)$$

in Laplace transform

$$A_P s y = K_v(I - y)$$

$$A_P s y = K_v I - K_v y$$

$$y(A_P s + K_v) = K_v I$$

So the transfer function

$$\frac{y}{I} = \frac{K_v}{A_P s + K_v}$$

or

$$\frac{y}{I} = \frac{1}{1 + \frac{A_P}{K_v} s}$$

where the time constant

$$\tau = \frac{A_P}{K_v}$$

In numerical form for extension

$$\frac{y}{I} = \frac{1}{1 + \frac{8 \times 10^{-3}}{0.012} s}$$

or

$$\frac{y}{I} = \frac{1}{1 + 0.67s}$$

where the time constant

$$\tau = 0.67 seconds$$

or the corner frequency

$$\omega = 1/0.67 = 1.5 Hz$$

In numerical form for retraction

$$\frac{y}{I} = \frac{1}{1 + \frac{4 \times 10^{-3}}{0.012} s}$$

or

$$\frac{y}{I} = \frac{1}{1 + 0.33s}$$

where the time constant

$$\tau = 0.33 seconds$$

or the corner frequency

$$\omega = 1/0.33 = 3 Hz$$

Or as shown in Figure 4.9

2- Transfer function y/P

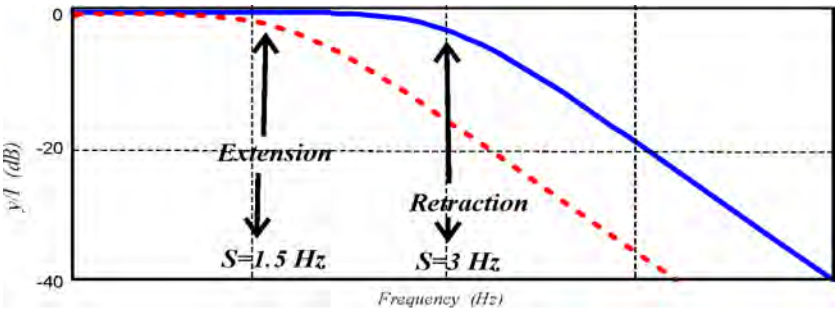


Figure 4.8: *Transfer function between the current I and the actuator displacement y in Example 4.2*

$$m\ddot{y} + C\dot{y} + ky = PA_P$$

in Laplace transform

$$ms^2y + Csy + ky = PA_P$$

$$y(ms^2 + Cs + k) = PA_P$$

The transfer function y/P

$$\frac{y}{P} = \frac{A_P}{ms^2 + Cs + k}$$

$$\frac{y}{P} = \frac{A_P/m}{(s^2 + \frac{C}{m}s + \frac{k}{m})}$$

In numerical form for extension

$$\frac{y}{P} = \frac{8 \times 10^{-3}/1000}{s^2 + \frac{10}{1000}s + \frac{1 \times 10^6}{1000}}$$

$$\frac{y}{P} = \frac{8 \times 10^{-6}}{s^2 + (0.01)s + 1 \times 10^3}$$

In numerical form for retraction

$$\frac{y}{P} = \frac{4 \times 10^{-6}}{s^2 + (0.01)s + 1 \times 10^3}$$

Or as shown in Figure 4.9

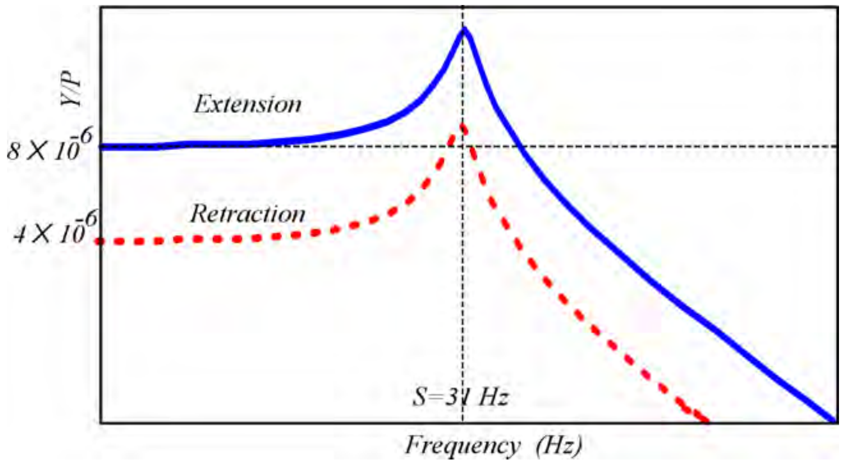


Figure 4.9: *Transfer function between the pressure P and the actuator displacement y in Example 4.2*

4.1.2 Pump operated servo control

A pump operated servo control system is usually used for large power applications especially hydraulic rotating motors. It operates with a variable displacement pump where the system is controlled by changing the displacement of the pump [3]. A pump operated servo control system has the following general characteristics:

- The circuit is compact with more complicated components.
- It gives much higher power to drive high inertia loads.
- Friction losses are minimized in this system which increases the efficiency of the system.
- Used in general to drive hydraulic motors (Hydrostatic transmission system).

The pump operated servo control system is usually used for hydrostatic power transmission systems where a variable displacement pump with a displacement y driven by an electric motor or internal combustion engine at a speed Ω_p, delivers hydraulic oil with a flow rate of q to rotate a hydraulic motor at a speed of Ω_m.

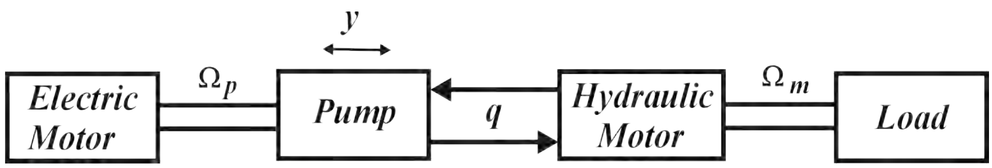


Figure 4.10: *Block diagram of a hydrostatic transmission system*

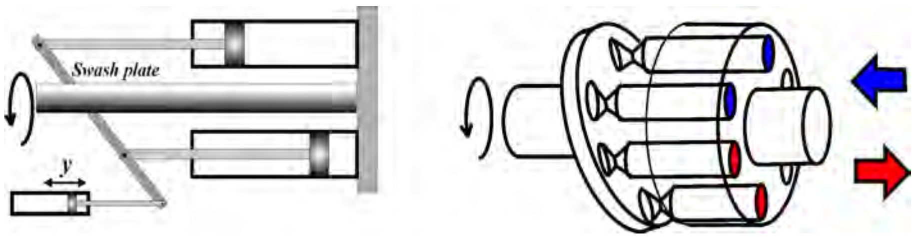


Figure 4.11: *Section and assembly of a reciprocating variable displacement swash plate pump*

Consider the hydrostatic transmission system shown in Figure 4.10 with a reciprocating positive variable displacement pump shown in Figure 4.11. Assuming there is neither leakage nor compressibility in the system

$$\text{Flow from the pump} = \text{Flow to the motor}$$

$$q_p = q_m = q$$

$$\Omega_p d_p = \Omega_m d_m$$

where $\Omega_p = \textit{constant}$ is the constant angular speed of the pumps rotor and d_m is the displacement of the motor per radian. If the control piston in the pump moves a displacement y as shown in Figure 4.11, the flow rate of the pump is proportional to that displacement

$$q_p = K_p y$$

where K_p is the pump flow constant at a constant speed, this leads to

$$q_p = q_m = K_p y = \Omega_m d_m$$

Hence, finding the transfer function between the pump's control displacement and the speed of the motor

$$\frac{\Omega_m}{y} = \frac{K_p}{d_m} \tag{4.2}$$

This gives a constant relationship between the input and the output without being influenced by the dynamics of the system which is too much ideal because the maximum speed cannot be achieved instantaneously at zero time.

Leakage effect

In real hydraulic systems, leakage occurs in pumps and hydraulic motors which includes a significant influence on the flow rate in the system. The leakage coefficients of the pump and motor are denoted by λ_p and λ_m, respectively. The leakage coefficients are proportional to the pressure value in the system [4].

Pump leakage $= \lambda_p P_p$.
Actual flow in pump $= K_p y - \lambda_p P_p = q$.
Motor leakage $= \lambda_m P_m$.
Actual flow to motor $= q - \lambda_m P_m = \Omega_m d_m$

so

$$\Omega_m d_m = K_p y - \lambda_p P_p - \lambda_m P_m$$

If the combined leakage coefficient λ for both pump and motor is

$$\lambda = \lambda_p + \lambda_m$$

and the pressure drop is negligible, such that

$$P = P_p = P_m$$

then

$$\Omega_m d_m = K_p y - \lambda P \tag{4.3}$$

Equating the output mechanical power to the input hydraulic power in the motor reads

$$Power = T_m\Omega_m = P_m q_m$$

where T_m is the torque output of the motor

$$T_m\Omega_m = P_m\Omega_m d_m$$

therefore,

$$T_m = P_m d_m$$

but the torque can be related to the inertia I and the angular acceleration α by

$$T_m = \alpha I = \frac{d\Omega_m}{dt} I$$

thus

$$P_m d_m = \frac{d\Omega_m}{dt} I$$

Manipulating Equation (4.3) gives

$$\Omega_m d_m = K_p y - \lambda P$$

$$P = \frac{K_p y - \Omega_m d_m}{\lambda}$$

multiplying both sides by d_m gives

$$P d_m = \frac{d_m K_p y - \Omega_m d_m^2}{\lambda}$$

therefore,

$$T_m = P d_m$$

or

$$I\frac{d\Omega_m}{dt} = \frac{d_m}{\lambda}(K_p y - \Omega_m d_m)$$

Using Laplace transform with zero initial conditions

$$Is\Omega_m(s) = \frac{d_m}{\lambda}(K_p y(s) - \Omega_m(s)d_m)$$

$$\Omega_m(s)\left(Is + \frac{d_m^2}{\lambda}\right) = \frac{d_m}{\lambda}K_p y(s)$$

Solving for the transfer function between $y(s)$ and $\Omega_m(s)$

$$\frac{\Omega_m(s)}{y(s)} = \frac{d_m K_p/\lambda}{Is + d_m^2/\lambda}$$

multiplying numerator and denominator by λ/d_m^2 gives

$$\frac{\Omega_m(s)}{y(s)} = \frac{K_p}{d_m}\left(\frac{1}{1 + (\lambda I/d_m^2)s}\right) \tag{4.4}$$

This represents a first order low-pass filter

$$\frac{\Omega_m(s)}{y(s)} = \frac{K_p}{d_m}\left(\frac{1}{1 + \tau s}\right)$$

with a time constant $\tau = \lambda I/d_m^2$. The bode plot of this transfer function is shown in Figure 4.12.

Compressibility effect

Hydraulic oils are considered, ideally, incompressible for simplicity of calculations. However, there is a small amount of compressibility in these fluids that introduces some flexibility causing dynamic influence on the system.

In solid mechanics, the modulus of elasticity is considered as the measure of flexibility of solid materials. Similarly, flexibility of fluids is measured by a constant called *Bulk modulus B* and can be defined by

$$B = \frac{Volumetric\ stress}{Volumetric\ strain}$$

or

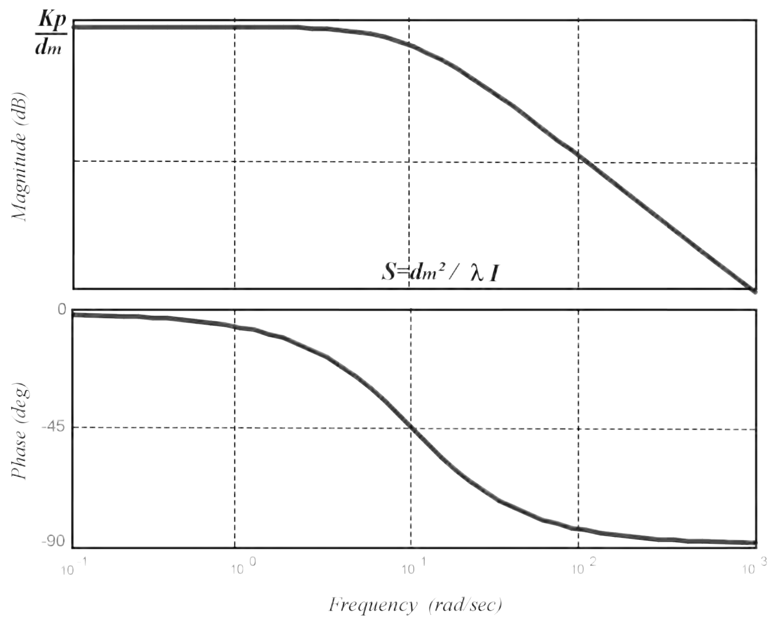


Figure 4.12: *Frequency response function* $\Omega_m(s)/y(s)$ *with leakage effect*

$$B = \frac{P}{\Delta V/V}$$

where P is the pressure and V is the original volume of fluid between the pump and the motor. Compressibility has a direct influence on the pressure. Although, it causes a loss in the flow q_c, where

$$q_c = \frac{d}{dt}\Delta V = \frac{d}{dt}\left(\frac{VP}{B}\right) = \left(\frac{V}{B}\right)\frac{d}{dt}P$$

The actual flow reaches to the motor is

$$\Omega_m d_m = \underbrace{K_p y}_{Pump\ delivery} - \underbrace{\lambda P}_{Leakage\ loss} - \underbrace{\left(\frac{V}{B}\right)\frac{d}{dt}P}_{Compressibility\ loss} \tag{4.5}$$

From previous discussion, it is known that

$$Pd_m = T_m = I\frac{d}{dt}\Omega_m$$

or

$$P = \frac{I}{d_m}\frac{d\Omega_m}{dt}$$

Substituting back in equation (4.5) results in

$$\Omega_m d_m = K_p y - \lambda\left(\frac{I}{d_m}\frac{d\Omega_m}{dt}\right) - \frac{V}{B}\frac{d}{dt}\left(\frac{I}{d_m}\frac{d\Omega_m}{dt}\right)$$

or

$$\Omega_m d_m = K_p y - \lambda\left(\frac{I}{d_m}\frac{d\Omega_m}{dt}\right) - \frac{VI}{Bd_m}\frac{d^2\Omega_m}{dt^2}$$

Using Laplace transform

$$\Omega_m(s)d_m = (K_p y(s)) - \left(\frac{\lambda I}{d_m}s\Omega_m(s)\right) - \left(\frac{VI}{Bd_m}s^2\Omega_m(s)\right)$$

Manipulating

$$\Omega_m(s)\left(d_m + \left(\frac{\lambda I}{d_m}\right)s + \left(\frac{VI}{Bd_m}\right)s^2\right) = K_p y(s)$$

Solving for the transfer function Ω_m/y gives

$$\frac{\Omega_m(s)}{y(s)} = \frac{K_p}{d_m + (\lambda I/d_m)\,s + (VI/Bd_m)\,s^2}$$

or

$$\frac{\Omega_m(s)}{y(s)} = \frac{K_p}{d_m}\left(\frac{Bd_m^2/VI}{s^2 + (\lambda B/V)s + (Bd_m^2/VI)}\right) \tag{4.6}$$

Which represents a second order low-pass filter. But the general characteristic transfer function reads

$$\frac{\Omega_m(s)}{y(s)} = \frac{K_p}{d_m}\left(\frac{\omega^2}{s^2 + 2\xi\omega s + \omega^2}\right) \tag{4.7}$$

Matching equations (4.6) and (4.7) gives the following parameters of the system

$$\omega^2 = \frac{Bd_m^2}{VI}$$

and

$$2\xi\omega = \frac{\lambda B}{V}$$

where ω is the undamped natural frequency and ξ is the damping ratio in the system. From the previous comparison, the natural frequency reads

$$\omega = \sqrt{\frac{Bd_m^2}{VI}}$$

It is important for any hydraulic system to increase the stiffness and push the natural frequency to the highest possible value. The following steps can be taken into account to increase the natural frequency:

- Increasing the motor displacement d_m by increasing the pump flow rate.
- Decreasing the original volume V of the fluid between the pump and the motor. This is possible by installing the motor close enough to the pump.
- Minimizing the load inertia.

Knowing the natural frequency of the system, the hydraulic stiffness can be determined as follows: for a revolution of 1 radian, the volume difference is equal to the motor displacement $\Delta V = d_m$, therefore

$$P_m = \frac{Bd_m}{V}$$

and the torque of the motor is

$$T_m = d_m P_m = d_m \left(\frac{Bd_m}{V}\right) = \frac{Bd_m^2}{V}$$

and can be calculated from

$$T_m = K_H \times \theta$$

where K_H is the hydraulic stiffness and θ is the angle of rotation. But $\theta = 1$ radian, then $K_H = T_m$

$$K_H = \frac{Bd_m^2}{V} \tag{4.8}$$

Solving for the natural frequency

$$\omega = \sqrt{\frac{K_H}{I}} = \sqrt{\frac{Bd_m^2}{VI}}$$

The damping ratio can be calculated from the previous parameters of the system as follows

$$2\xi\omega = \frac{\lambda B}{V}$$

substituting the value of the natural frequency gives

$$\xi = \sqrt{\frac{\lambda^2 BI}{4Vd_m^2}} \tag{4.9}$$

It is clear from the previous discussion that the damping ratio depends strongly on the leakage in the system. Knowing the different parameters of the system, the transfer function in equation (4.6) can be presented using the Bode plot shown in Figure 4.13.

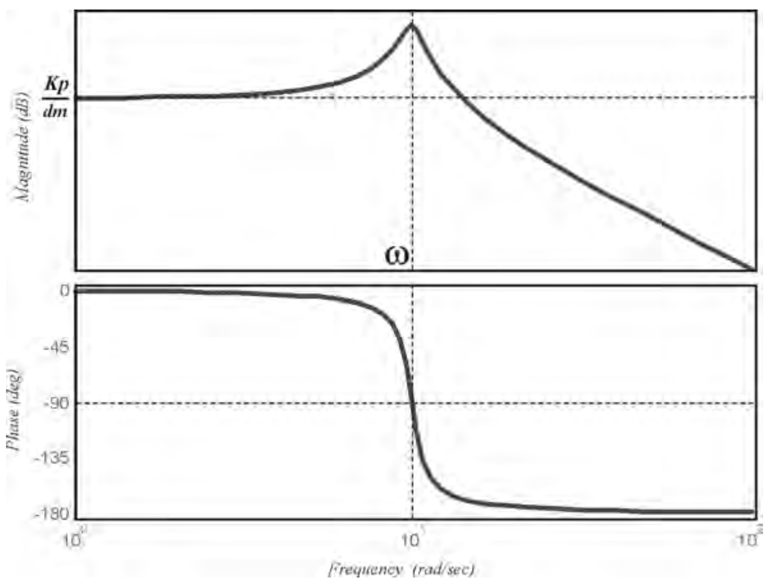


Figure 4.13: *Frequency response function $\Omega_m(s)/y(s)$ with leakage and compressibilty effects*

Example 4.3

Consider a reversible hydrostatic transmission system with a variable displacement pump and a constant displacement hydraulic motor with the following characteristics:

Leakage coefficient of pump and motor $\lambda = 0.01$ l/min/bar.
Load inertia $I = 300\ N.m.s^2$.
The motor displacement $d_m = 25$ ml/radian.
The maximum motor speed $\Omega_m = 200$ rpm.
Motor acceleration $\alpha = 1.05\ rad/s^2$.
Pump speed $\Omega_p = 1400$ rpm.
Overall effeiciecy $\eta = 85\%$.
Pump control stroke $y = 0.1$ m.

Calculate:

1. System pressure (pipe friction losses are negligible).
2. The actual pump capacity.

3. The power of the electric motor needed to drive the pump.

4. The time constant τ and the frequency response function Ω_m/y.

Solution

1. The torque at the load

$$
\begin{aligned}
T_m &= I\alpha \\
T_m &= 300(Nms^2) \times 1.05\ (rad/s^2) \\
T_m &= 315\ N.m
\end{aligned}
$$

But the hydraulic motor torque is

$$
\begin{aligned}
T_m &= P_m d_m \\
P_m &= T_m/d_m = 315(Nm)/25 \times 10^{-6}\ (m^3) \\
P_m &= 125.7 \times 10^5 Pa = 125.7\ bar
\end{aligned}
$$

2. The fluid flow rate reaching to the motor is

$$
\begin{aligned}
q_m &= \Omega_m d_m \times 2\pi \\
q_m &= 200(rev/min) \times 25 \times 10^{-6}(m^3) \times 2\pi \\
q_m &= 31.4 \times 10^{-3} m^3/min = 31.4\ l/min
\end{aligned}
$$

The total leakage from the pump and the motor is

$$
\begin{aligned}
q_l &= \lambda P_m \\
q_l &= 0.01(l/min/bar) \times 125.7(bar) \\
q_l &= 1.26\ l/min
\end{aligned}
$$

The total actual pump capacity is

$q_p = q_m + q_l$
$q_p = 31.4 + 1.26 = 32.66 \; l/min$

3. Electric motor power in (KW) is

$$Electric\ motor\ power = \frac{flow(l/min) \times Prsssure\ (bar)}{600 \times Overall\ effeciency}$$

$$Electric\ motor\ power = \frac{32.66(l/min) \times 125.7(bar)}{600 \times 0.85} = 8.04\ KW$$

4. The time constant

$$\tau = \frac{\lambda I}{d_m^2}$$

$$\lambda = 0.01 \frac{l}{min.bar} \times \frac{1min}{60s} \times \frac{1bar}{10^5(N/m^2)} = \frac{1 \times 10^{-10}}{60}(m^5/N/s)$$

$$\tau = \frac{(10^{-10}/60)(300)}{(25 \times 10^{-6})^2} = 0.8\ s$$

The transfer function

$$\frac{\Omega_m}{y} = \frac{K_p}{d_m}\left(\frac{1}{1+\tau s}\right)$$

The pump coefficient

$$K_p = \frac{q_p}{y} = \frac{32.66}{60 \times 10^3} \times \frac{1}{0.1} = 5.44 \times 10^{-3}\ m^2/s$$

Therefore

$$\frac{\Omega_m}{y} = \frac{5.44 \times 10^{-3}}{25 \times 10^{-6}}\left(\frac{1}{1+0.8s}\right)$$

$$\frac{\Omega_m}{y} = 217.6\left(\frac{1}{1+0.8s}\right)$$

Example 4.4

Consider a concrete mixer drum driven by reversible hydrostatic transmission system with a variable displacement pump and a constant displacement hydraulic motor with the following characteristics:

Leakage coefficient of pump and motor $\lambda = 0.02$ l/min/bar.
Load inertia $I = 500\ N.m.s^2$.
The motor displacement $d_m = 45$ ml/radian.
The maximum motor speed $\Omega_m = 150$ rpm.
Motor acceleration $\alpha = 1.5\ rad/s^2$.
Pump speed $\Omega_p = 1400$ rpm.
Overall volume of the system $V = 2litre$.
Oil used is VG 68 with a Bulk modulus of elasticity $B = 1.2GPa$.
Maximum pump delivery is $Q = 60l/min$
Maximum displacement for swash plate $y = 10cm$
Calculate:

1. The pressure in the system.
2. Transfer function between swash plate pump displacement y as an input motor's angular speed Ω_m as an output.
3. The natural frequency, damping ratio and hydraulic stiffness of the system.
4. Show how to improve the bandwidth of the system.

Solution

1. To calculate the pressure

$$T_m = I\alpha = P_m d_m$$

$$(500)(1.5) = P_m(45 \times 10^{-6})$$

$$P_m = 16.66 \times 10^6 Pa = 166.6bar$$

2. To calculate the transfer function

$$\frac{\Omega_m(s)}{y(s)} = \frac{K_p}{d_m}\left(\frac{Bd_m^2/VI}{s^2 + (\lambda B/V)s + (Bd_m^2/VI)}\right)$$

To calculate the pump coefficient K_p

$$K_p = \frac{Q}{y} = \left(\frac{60(\frac{l}{min})(\frac{min}{60s})(\frac{m^3}{1000l})}{0.1}\right) = 0.01$$

To estimate the leakage coefficient λ in standard units

$$\lambda = 0.02\frac{l}{min.bar}\times\frac{1min}{60s}\times\frac{1bar}{10^5(N/m^2)} = \frac{2\times 10^{-10}}{60}(m^5/N/s) = 3.3\times 10^{-9}$$

$$\frac{\Omega_m(s)}{y(s)} = \frac{0.01}{45\times 10^{-6}}\left(\frac{\frac{(1.2\times 10^9)(45\times 10^{-6})^2}{(2\times 10^{-3})(500)}}{s^2 + \frac{(3.3\times 10^{-9})(1.2\times 10^9)}{(2\times 10^{-3})}s + \frac{(1.2\times 10^9)(45\times 10^{-6})^2}{(2\times 10^{-3})(500)}}\right)$$

$$\frac{\Omega_m(s)}{y(s)} = 285.7\left(\frac{2.43}{s^2 + 1.98\times 10^{-3}s + 2.43}\right)$$

3. The natural frequency

$$\omega_n = \sqrt{2.43} = 1.55rad/sec$$

The damping ratio

$$\xi = \frac{1.98\times 10^{-3}}{2\times 1.55} = 6.38\times 10^{-4}$$

The hydraulic stiffness

$$K_H = \frac{B(d_m)^2}{V} = \frac{(1.2\times 10^9)(45\times 10^{-6})^2}{2\times 10^{-3}} = 1.2\times 10^3 N/m$$

4. The natural frequency is small and the hydraulic stiffness is extremely law, this makes the bandwidth very narrow. The bandwidth can be enlarged by minimizing the volume of the system where the pump and hydraulic motor can be mounted close to each others.

4.2 Problems

1. Consider the the spring loaded hydraulic piston shown in Figure 4.14 that moves a distance y by means of pilot pressure coming from pressure line P_2 and divided between the restrictor R and the piston. Find the transfer function between the flow rate Q and the displacement y and draw the response in frequency domain.

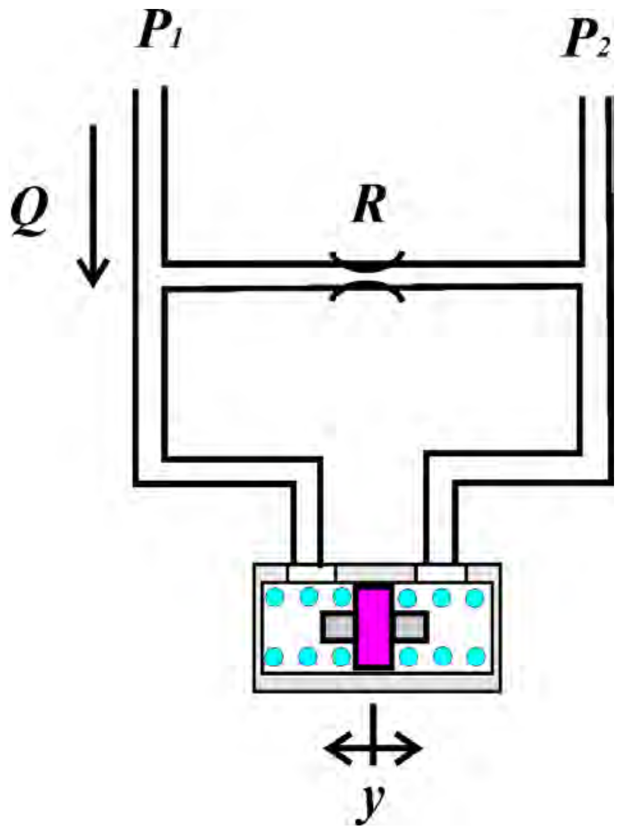


Figure 4.14: *A spring loaded hydraulic piston with a pressure control*

2. Consider the system shown in Figure 4.15, find the transfer function y/x knowing that

$$K = 10$$

$$H_1(s) = 4s$$

$$G_1 = \frac{8}{s(s+2)(s+3)}$$

$$H_2(s) = \frac{2}{1+4s}$$

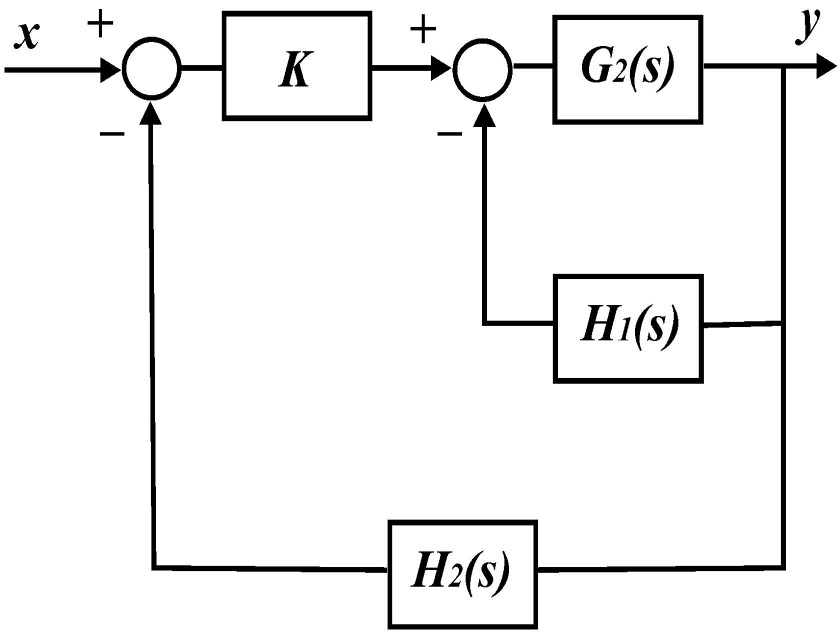


Figure 4.15: *A block diagram with two feedback loops*

3. Consider the hydraulic process shown in 4.16. Knowing that the capacitance of each tank is the ratio between the change in quantity of liquid to the change of height of the liquid and the capacity of the tank is the actual volume. Find the transfer function between Q_3 and h_1. R_1 and R_2 are the resistance to flow in each tank.

4. Consider the system represented by the equation of motion

$$450\frac{d^2x}{dt^2} + 105\frac{dx}{dt} + 10x = f(t)$$

Find the time response $x(t)$ for a step unit input taking zero initial displacement and velocity.

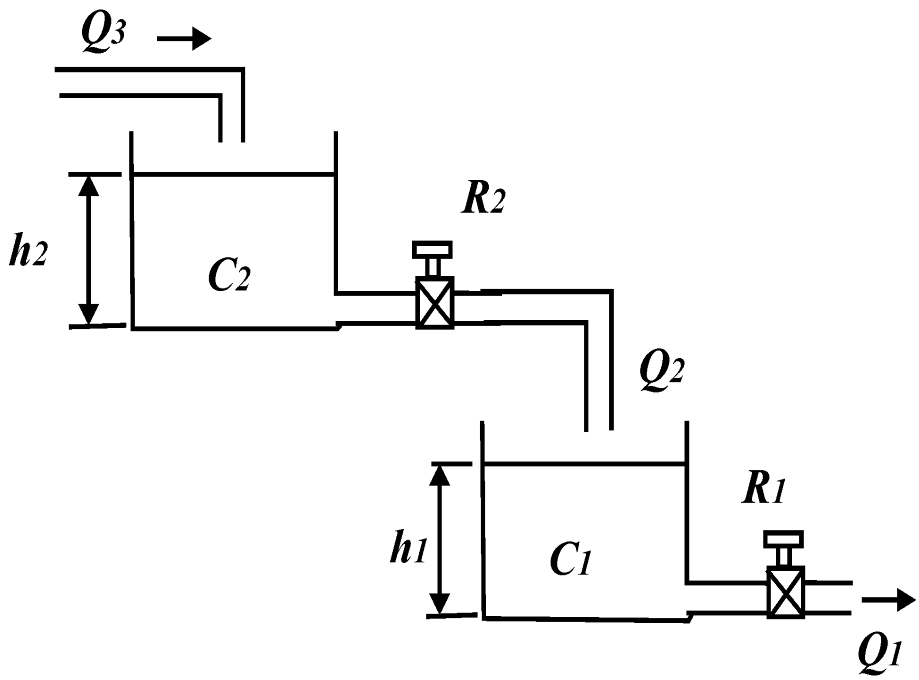


Figure 4.16: *A system of two hydraulic tanks*

5. A hydraulic motor produces a torque of $27N.m/rad$ and its shaft drives with inertia of $1.4kg.m^2$ and a damping coefficient $4N.m.s/rad$. calculate the transfer function and draw the bode plot between shaft output angle and the swash plate angle. Calculate the damping ratio.

6. Knowing that the transfer function of a hydrostatic transmission system is defined by the following second order low-pass filter

$$\frac{\Omega_m}{y} = \frac{K_p}{d_m} \frac{Bd_m^2}{VIs^2 + \lambda BIs + Bd_m^2}$$

   Find the following:
   a- The natural frequency of the system (ω).
   b- The damping ratio (ξ).
   c- The hydraulic stiffness (H_s).

7. Assume a simple closed-loop hydrostatic transmission system that consists of a hydraulic variable displacement pump and a hydraulic

motor. Taking into account the leakage and compressibility effects, prove that:

$$\frac{\Omega_m}{y} = \frac{K_p}{d_m} \frac{Bd_m^2/VI}{s^2 + \lambda(B/V)s + Bd_m^2/VI}$$

8. Consider a rotary pneumatic vane type actuator with an inlet air pressure P acting on a vane area A at a distance R from the center of rotation. The actuator holds a load with Inertia J and torque T related linearly to the input pressure. The pneumatic stiffness is H and the pneumatic damping is D.

   a- Find the equation of motion of the system
   b- Find the transfer function between the input pressure P and the output angular rotation of the motor θ.
   c- Find the state space matrices taking the angular displacement and angular velocity as state variables using the input P and output as the angular acceleration of the motor.

9. Consider a reversible hydrostatic transmission system with a variable displacement pump and a constant displacement hydraulic motor with the following characteristics:

   Leakage coefficient of pump and motor $\lambda = 0.01 l/min/bar$.
   Load inertia $I = 300 N.m.s^2$.
   The motor displacement$d_m = 25 ml/radian$.
   Pump delivery $q_p = 32.6 l/m$.
   Pump control stroke $y = 0.1 m$.

   Calculate:
   a. The time constant τ.
   b. The frequency response function $_m/y$ in Laplace transform.
   c. Draw a Bode plot for the transfer function $_m/y$.

10. Consider a reversible hydrostatic transmission system with a variable displacement pump and a constant displacement hydraulic motor with the following characteristics:

    Leakage coefficient of pump and motor = $0.01 l/min/bar$.
    Bulk modulus of rigidity of oil = $1500 MPa$
    System Volume = $0.0001 m^3$
    Load inertia = $300 N.m.s^2$
    The motor displacement = $25 ml/radian$.
    The maximum motor speed = $200 rpm$.
    Motor acceleration = $1.05 rad/s^2$.
    Pump speed = $1400 rpm$.
    Pump control stroke $y = 0.1m$.

    Considering leakage and compressibility, Calculate:

    1. System pressure (pipe friction losses are negligible).
    2. The actual pump flow rate.
    3. The frequency response function between hydraulic motor speed and pump stroke y.
    4. The natural frequency and the hydraulic stiffness
    5. Draw the bode plot

11. Consider the robot shown in Figure 4.17:

    The robot consists of:
    1- Hydraulic oil with bulk modulus=$1.5 \times 10^9 Pa$, kinematic viscosity= $200 cSt$, Specific gravity= 0.9, in $13mm$ diameter rusted steel pipes with 5 standard 90^o elbows at the pressure line and other 5 at the return line.
    2- Hydraulic pump at M_1 with volume $V_p = 0.01 m^3$, leakage coefficient= $5 \times 10^{-11} m^5/N/s$, pump speed = $1400 rpm$, $y = 0.1$, $K_p = 4 \times 10^{-3}$, Pump power=$1 KW$

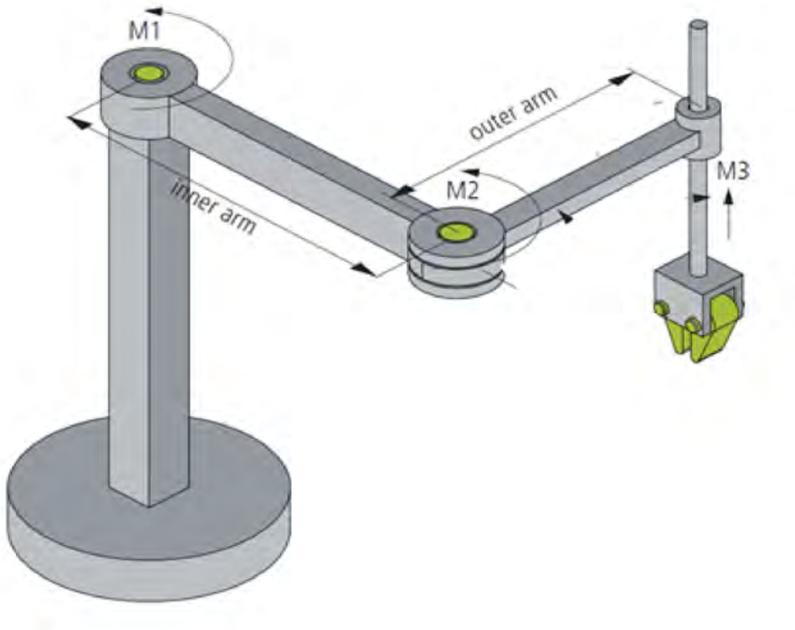


Figure 4.17: *A three DOFs robot*

3- Reversible hydraulic motor at M_1 with volume $V_{M1} = 0.01m^3$, head=1.5m, displacement= $2l/radian$, max. motor speed = $200rpm$ and motor acceleration = $1.2rad/s^2$
4- Reversible hydraulic motor at M_2 with volume $V_{M2} = 0.01m^3$, head=1.5m, displacement=$2l/radian$, max. motor speed = $200rpm$ and motor acceleration = $1.2rad/s^2$
5- Double acting hydraulic linear piston at M_3 with piston diameter $D_{M3} = 5cm$, rod diameter $R_{M3} = 2cm$, stroke=30cm
6- Inner arm with length $L_1 = 0.6m$ and mass $m_1 = 3kg$
7- Outer arm with length $L_2 = 0.5m$ and mass $m_2 = 2kg$
8- Load at the end effector equals to $100kg$

Determine:

1- The volume of the hydraulic reservoir (Tank).
2- The pressure in the piston needed to lift the load when the pump is off, in the case when the piston rod is in up position and when the rod is down position.

3- The pressure in the piston when the pump is on.
4- Build a block diagram that contains the pump, the hydraulic motors and the hydraulic piston with the input as the displacement of the pump and the output as the displacement of the load.
5- Consider that the hydraulic motor M_1 forms a hydrostatic transmission system with the pump and find the transfer function between the angular speed of motor M_1 and pump displacement (y). find the natural frequency, hydraulic stiffness and damping ratio, draw the bode plot for this transfer function. (Take leakage and compressibility into account).
6- Build a three degrees of freedom dynamic model taking the two hydraulic motors at M_1 and M_2 and the piston at M_3. (Neglect the pump). The inputs are the toque T_1 for motor M_1 and torque T_2 for motor M_2 and pressure P for piston M_3. The outputs are angular displacements M_1 and M_2 and linear displacement M_3 and velocity for M_3 and acceleration for M_3. Use state space approach selecting the displacement and velocity at M_1, displacement and velocity at M_2 and displacement and velocity at M_3 as state variables. Use the oil hydraulic stiffness and damping calculated in (5). Write the equations of motion and derive the four matrices of state space (A, B, C, D).
7- Draw the bode plots between inputs and outputs.

4.3 References

[1] F. Yeaple, *Fluid Power Design Handbook*, Third Ed., Marcel Dekker,1996.

[2] J. Pippenger and T. Hicks, *Industrial Hydraulics*, McGraw-Hill, 1979.

[3] H. Stewart, *Hydraulic and Pneumatic Power for Production*, Industrial Press Inc., 1977.

[4] M. Pinches and J. Ashby, *Power Hydraulics*, Prentice Hall, 1989.

Chapter 5

Mechatronics of Fluid Power Systems

5.1 Electromagnetic background

When a bobbin φ is approached to a permanent magnet with the shown polarity $[S-N]$ (South-North), an electromagnetic force is induced in the bobbin and electric current is induced in the shown direction if the circuit is closed (see Figure 5.1).

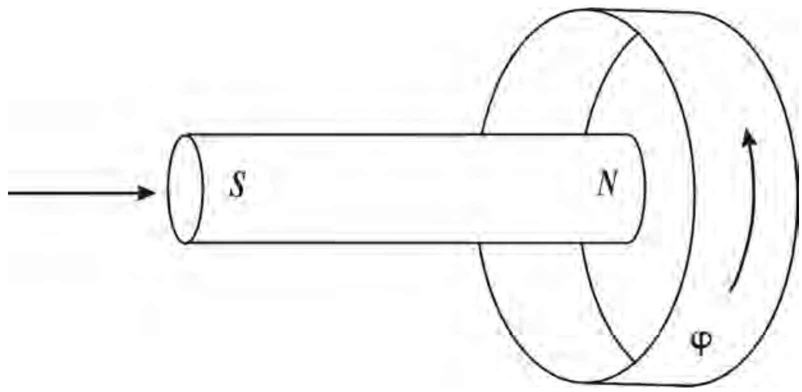


Figure 5.1: *Moving magnet in a coil*

The magnitude of the induced voltage is determined by Faraday's law. Assume a magnet with a cross-sectional area S and is bounded by the closed contour φ. If the magnetic flux ϕ linking S varies with time, a voltage V is induced around φ. This voltage is given by Faraday's law [1]:

$$V = -\frac{d\phi}{dt} = -L\frac{di}{dt} = -\frac{d}{dt}\int_S BdS \tag{5.1}$$

In circuit theory L is called the self-inductance of the element φ and V is called the voltage of the self-inductance. B is the magnetic field density [2]. The sense of direction of the induced voltage is determined by Lenz's law [3] which states that: *The voltage induced by a changing flux has a polarity such that the current established in a closed path gives rise to a flux which opposes the change in flux.* In other words, one can say that; in the induced current there is an electromagnetic inertia that opposes the variation of flux, like the reaction of mass inertia against the velocity variation. The voltage sense can be determined easily by applying the left hand rule, where, the pounce finger is the field and the index is the direction of movement, then the middle finger indicates the current. This polarity prediction assumes that the conductor experiences a magnetic force which opposes its motion. When a closed conducting loop is in motion with a relative velocity U with respect to a stationary magnet that generates a constant field B, this includes change in the shape. In this case Equation (5.1) can be applied:

$$V = -\frac{d}{dt}\int_S BdS = \oint (U \times B)dl \tag{5.2}$$

If the velocity U and the field B are at right angles and the conductor is normal to both, then a conductor with a length l will have a voltage:

$$V = BlU \tag{5.3}$$

According to the previous assumptions, B and l are constants. This leads to express the linear relation between the voltage and the velocity as follows:

$$V = GU \tag{5.4}$$

Where G is the *transduction constant* (in [V/m/s] or [N/A]) which can be called also the $Lorentz\ force\ constant$. From the previous discussion, one can see that the relation between the magnet velocity and the induced voltage depends strongly on the magnetic field density B and the length of the conductor l. In the case of coaxial cylindrical magnet and cylindrical coil, the length of the conductor is equal to the length of one turn of the coil multiplied by the number of turns.

5.1.1 Forces in the magnetic field

If a charged particle Q is moving in an external magnetic field B with a velocity V,as shown in Figure 5.2, the force F exerted on this particle is given by

$$F = Q(V \times B) \tag{5.5}$$

The direction of this force is defined by the right hand rule. In Figure 5.2, if the magnetic field B goes into the page and the particle moves to the right, then the force will be upwards.

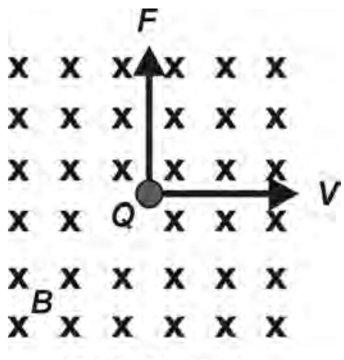


Figure 5.2: *Force on a charged particle in a magnetic field*

Since the rate of change of the charge with respect to time is the current,

$$I = \frac{dQ}{dt} \tag{5.6}$$

Replacing the charged particle by a current carrying conductor with a length l, the differential force on the conductor becomes

$$dF = dQ(V \times B) = (Idt)(V \times B) = I(dl \times B)$$

Where $dl = Vdt$. If the conductor is straight and the field is constant along it, both sides of the differential equation can be integrated to get

$$F = I(L \times B) = ILBSin\theta \tag{5.7}$$

Again, the direction of the exerted force is defined by the right hand rule. One can imagine, as in Figure 5.3, the field lines as stretched rubber bands that push the conductor exerting this magnetic force (the imagination of Michael Faraday). In the case when the conductor is a circular bobbin with a radius R, and N number of turns, equation (5.7) takes the form

$$F = ILBN \tag{5.8}$$

Such that $L = 2\pi R$. Here, $Sin\theta = 1$ because the magnetic field is always normal to the current.

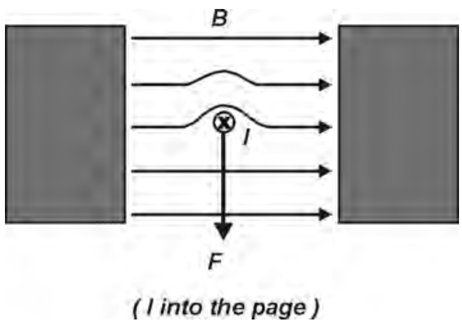


Figure 5.3: *Force on a conductor due to magnetic field (Faraday's imagination)*

5.2 Solenoids and relays

Solenoid is an electromechanical device consists of a cylindrical coil wrapped around a coaxial armature (see Figure 5.4). The magnetic armature is free to move axially inside the coil. The solenoid is an actuation device used to convert electromagnetic energy into linear motion. When an electric current passes in the coil, the induced electromagnetic force in the magnetic field forces the magnetic armature to move axially back or forth according to the current sense of direction applying the right hand rule. Solenoids are used to move the spool of the spool type valve in both, hydraulic and pneumatic valves forming the so called *Solenoid Valves.* Spools are pushed by the solenoid armatures to open or close ports and change position of valve. Figure 5.5 shows a real solenoid valve used for pneumatic applications.

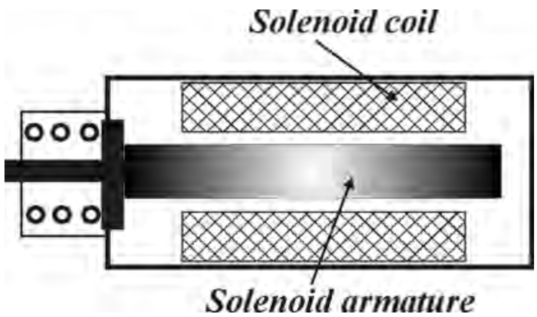


Figure 5.4: *Electromechanical solenoid design*

Figure 5.5: *Pneumatic solenoid valve (Courtesy of Fenghua Xingyu)*

Another main application on using electromagnetic power to control fluid power systems is *Relays*. Relay is an electromagnetic switch that consists of a contacting arm pulled and pushed by a solenoid (see Figure 5.6). When passing a low voltage electric current in the coil of the solenoid the induced electromagnetic force moves the armature back or forth causing the contacting arm to move and close the high voltage circuit to operate the required function. De-energizing the coil of the relay returns the arm to its centre point opening the high voltage circuit and stopping the operation. Since relays are switches, a relay will switch one or more poles contacting when energizing the coil in one of three ways:

- Normally open **(NO):** The circuit is normally disconnected when the relay is de-energized, it contacts when the coil is energized.
- Normally closed **(NC):** The circuit is normally connected when the relay is de-energized, it disconnects when the coil is energized.
- Change over **(CO):** Contacts control two circuits: one normally-open contact and one normally-closed contact with a common terminal.

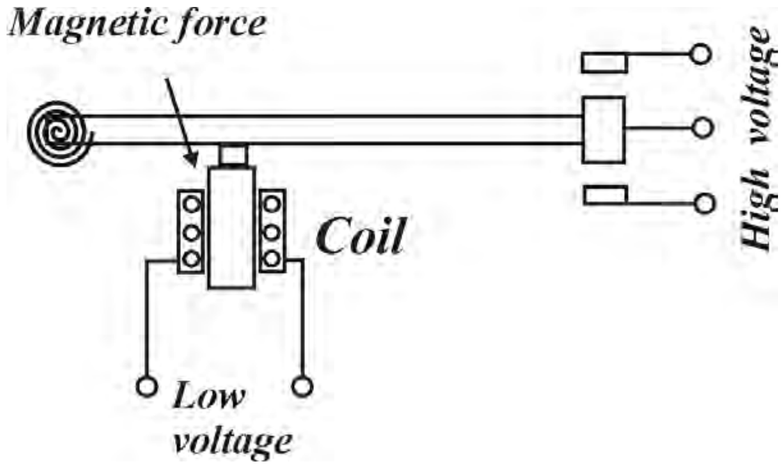


Figure 5.6: *Electromagnetic Relay*

5.3 Voice coil linear actuator

Another type of linear actuator is the so called *Voice Coil Actuator.* Voice coil actuator depends mainly on using magnetic field to induce an electromagnetic force. On the contrary of the solenoid, the moving part in the voice coil is the coil itself. Another advantage of the voice coil is that it can be used for precise displacements but for short strokes.

5.3.1 Basic architecture of voice coil

Figure 5.7 shows a cross-section of the internal basic design of a voice coil actuator. This type of actuators is made up of two components; a moving part and a fixed part. The upper part in the figure is the moving part and

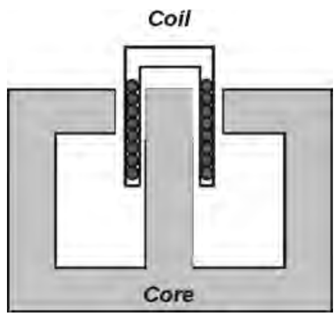


Figure 5.7: *Basic schematic architecture for a voice coil actuator*

consists of a group of wires wound in a tubular form around the coil holder. The holder can be manufactured out of plastic or fibre or any other non-conductive material depending on the load carried by this part to reduce the effect of eddy currents on the motion. The coil itself is formed out of a conducting wire like copper. The diameter of the wire and the dimensions of the holder are determined according to the application. The stationary member is made up of a permanent magnet. The polarity of the magnet can be radial or axial according to the design, and this is one of the basic concepts and differences between the three designs that will be explained later. This member can be not only a permanent magnet but it can also contain a ferromagnetic material like the iron or some steel alloys to pass the magnetic flux generated by the permanent magnet. In the voice coil, the magnetic cycle should be closed to have a continuous flux. Although there is an air gap that allows the coil to pass through, but this air gap should be as small as possible to reduce the fringe in the flux at that point [4].

As mentioned in the previous section, when a current carrying conductor crosses the stream lines of a magnetic field; a force is exerted on the conductor. This force is called $Lorentz\ force$ and is a function of the magnetic field density B, the current applied in the conductor I, the length of the conductor and the angle between the magnetic field and the current. The force is maximum when the field is normal to the current. It creates a linear motion in the moving part because the force is linearly proportional to the current applied in the coil, and this leads to achieve a good control. The displacement (stroke) moved by the voice coil can vary from microns up to a few millimetres. From the force point of view, the voice coil is considered a weak force producer relative to its size, in other words, it is not as weight-power-efficient as other devices.

5.3.2 Modelling of voice coil

The relative linear motion between the two parts of the voice coil is mainly caused by the current passing across the magnetic field. The output force of the actuator is linearly proportional to the input current. The ratio between the force and the current is called the actuator constant, this ratio is constant for each actuator and depends on the size, design and shape of the actuator itself.

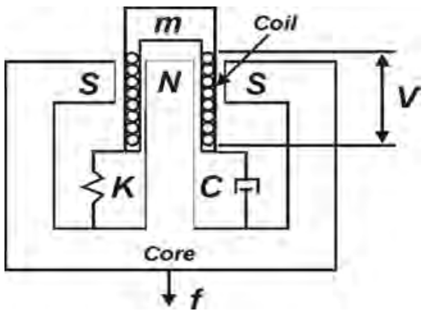


Figure 5.8: *installation of a voice coil actuator against a fixed reference*

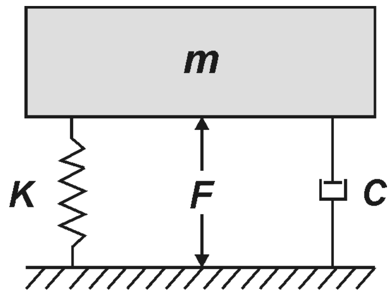


Figure 5.9: *Mass, spring and dash-pot representation of the voice coil*

To study the behaviour of a voice coil, it can be installed against a fixed reference. In this case, the mass m will act as a charged load, (Figure 5.8). The actuator is mounted on the top of a force sensor to measure the output force. This installation can be represented by a simple mass with a spring and a dash-pot, as shown in Figure 5.9, Which leads to a single degree of freedom model with the following equation of motion:

$$m\ddot{x} + C\dot{x} + Kx = F \tag{5.9}$$

But $F = GI$, where G is the *Lorentz force constant*. Applying the Laplace transform on Equation (5.9) we get:

$$s^2x + 2\xi\omega_n sx + \omega_n^2 x = \frac{G}{m}I$$

$$\frac{x}{I} = \frac{\frac{G}{m}}{s^2 + 2\xi\omega_n s + \omega_n^2} \tag{5.10}$$

$$\frac{F}{I} = \frac{-Gs^2}{s^2 + 2\xi\omega_n s + \omega_n^2} \tag{5.11}$$

Where the natural frequency $\omega_n^2 = \frac{K}{m}$ and the damping part of the equation is $2\xi\omega_n = \frac{C}{m}$, including contributions from the structure and from the back electro-motive force ($e.m.f$), because here the $e.m.f$ appears as an additional damping in the system, [5]. The transfer function expressed in Equation (5.11) is plotted in Figure 5.10. This Bode plot enables us to predict the characteristics of the system.

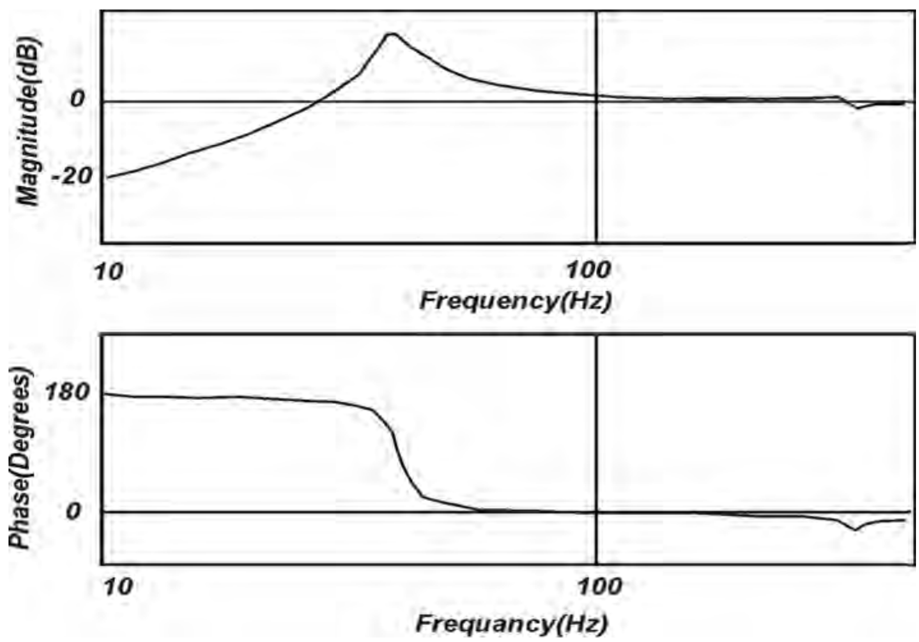


Figure 5.10: *Transfer function of the voice coil with the current as an input and the force as an output*

To determine the force constant G of the actuator which leads to define the force, a blocked transfer function is measured (see Figure 5.12). To obtain this plot one can block both sides of the voice coil actuator to prevent it from moving and apply a current into the coil as shown in Figure 5.11.

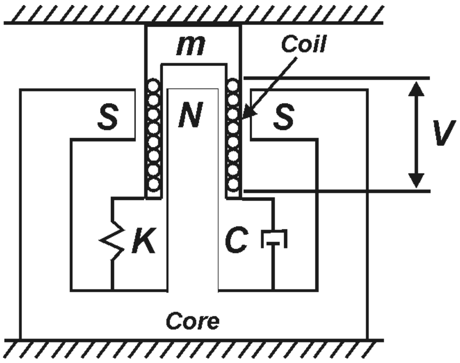


Figure 5.11: *Blocked installation of a voice coil*

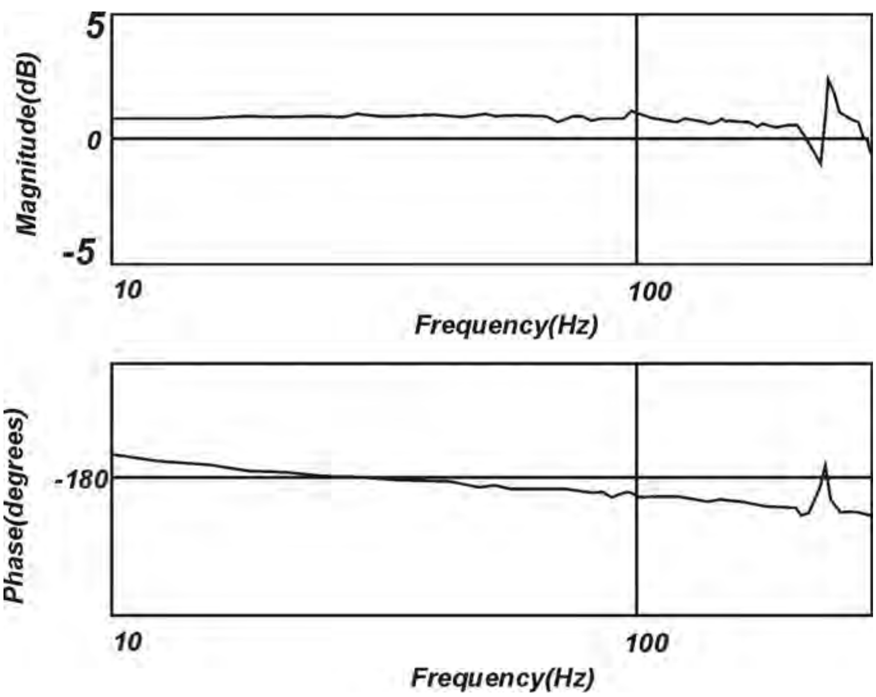


Figure 5.12: *Blocked transfer function of the voice coil with the current as an input and the force as an output*

5.3.3 Configurations of voice coil actuators

Several designs can be proposed to build an actuator based on the voice coil principle. Here, three configurations have been selected considering that they are the most common used designs.

Radial toroid voice coil actuator

The first design depends on employing a toroidal permanent magnet with radial polarity. The magnet is included coaxially in a cylindrical ferromagnetic material. A core from the same material exists along the central axis of the cylinder (Figure 5.13). Thanks to the large axial dimension of the permanent magnet, relatively large number of turns in the coil can be wound, which enables to obtain higher force. One of the restrictions that may face the designer here is the difficulty to obtain a large axial length for one magnet, but still there is a possibility to stack several magnets on top of each others. Another difficulty is that it is more expensive to produce a permanent magnet with radial polarity which increases the cost of the actuator. In order to minimize the volume and weight of the actuator, a special steel alloy with high magnetic permeability is used. Special alloys are used to avoid reaching the saturation limit in the ferromagnetic part. All the coil turns are accounted here in the force calculation. Because the flux generating area of the magnet is larger than the area of the bobbin coil [4].

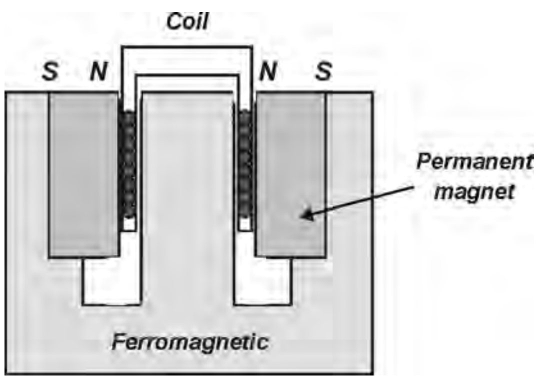


Figure 5.13: *Radial toroid voice coil actuator*

Axial toroid voice coil actuator

Another way of the design is shown in Figure 5.14. In this type, a toroidal shaped magnet is used here too, but this time the polarity is selected to be axial. To guide the magnetic flux generated by the magnet, a hollow disk of ferromagnetic material is mounted on the top of the magnet. This piece of metal aims at concentrating the magnetic field at the air gap where it crosses the coil current. On the other side of the magnet, another solid disk is mounted. In the centre of this disk and along the central axis, a core of the ferromagnetic material is installed. This type of configuration is easier to manufacture and is more economic than the previous one. To concentrate the flux in the air gap, the axial area near the gap should be minimized. This leads the designer to try to increase the number of turns of the coil to keep the force level, which, unfortunately, needs to increase the air gap. To do so, an optimization technique should be held to compromise all the dimensions including the air gap. In this configuration the axial distance of the coil is larger than the thickness of the hollow disk to enable the magnetic flux to cross the same number of turns during the stroke of the motion.

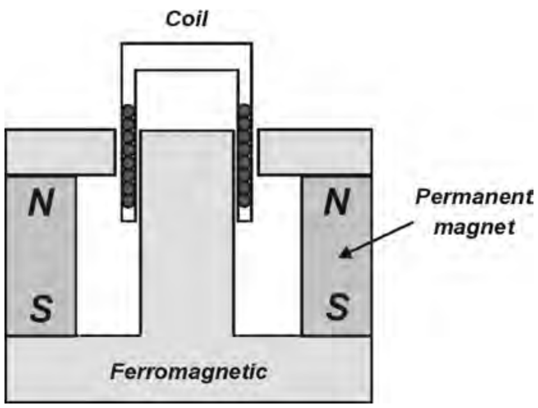


Figure 5.14: *Axial toroid voice coil actuator*

Axial disk voice coil actuator

This configuration is different from the previous two by the shape of the magnet and the situation where it is installed. Here the magnet is a circular disk put just under the core along the central axis of the cylinder (see

Figure 5.15). The magnetic flux, flowing axially out of the permanent magnet, passes through the ferromagnetic material to be concentrated at the air gap where the current carrying coil crosses the field lines generating a force. The central core should be well centred over the magnet and inside the outer cup to keep a uniform air gap and avoid flux leakage. The axial distance of the coil here also should be larger than the thickness of the ferromagnetic material at the air gap to cover the same number of turns during the motion of the bobbin [6].

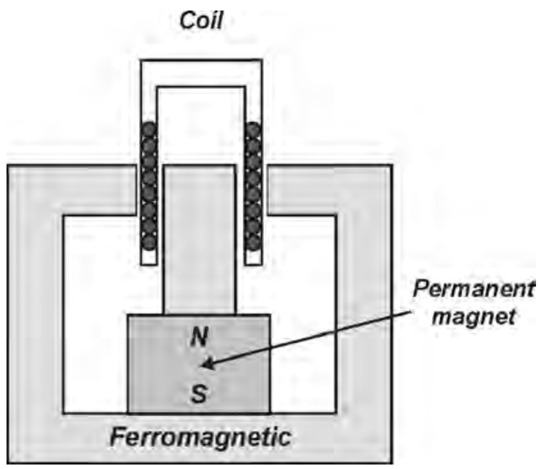


Figure 5.15: *Axial disk voice coil actuator*

5.4 Servo valves

Servo valve is a directional control valve that has an infinite possibilities of spool positions. This means that in servo valve, it is possible to locate the spool in any position leading to have any accurate area of the port having the required accurate flow rate for the system, this can be done by several ways as will be seen later. Servo valve is always accompanied with feedback loop that helps in obtaining the required position by eliminating the error in that position; feedback process can either be electrical or mechanical depending on the design of the system. Single stage spool valve (shown in Figure 5.16) is the basic design of the servo valve where the spool is driven by the torque motor via a mechanical mechanism to determine the final position. The main problem here is the static friction between the spool lands and the casing. This problem is solved by adding a dither signal to the main control signal. The dither signal is a signal with a very small amplitude and high frequency that reaches to 100 Hz. This signal oscillates the spool around its rest position with very small strokes to overcome the static friction force and keep the spool in continuous dynamic motion.

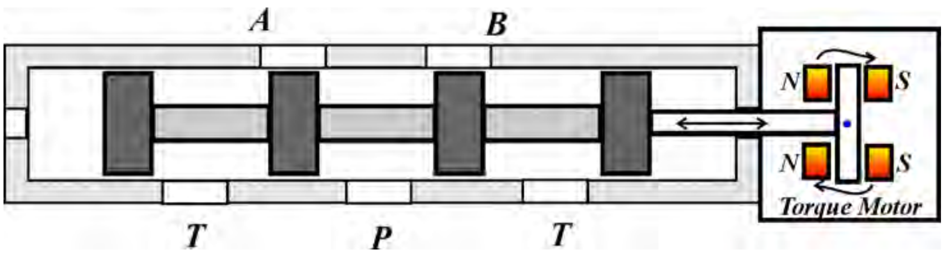


Figure 5.16: *Single stage spool valve directly operated by a torque motor*

Another type of servo valves is the flapper type shown in Figure 5.17. The principle of this type is based on moving the flapper plate to the right or the left by the magnetic force induced by the electromagnetic coils. When the flapper moves to the left it closes the outlet of the left nozzle increasing the pressure in that side and causing the spool to move to the right. The value of the produced force here is controlled by the restrictor mounted on that pipe.

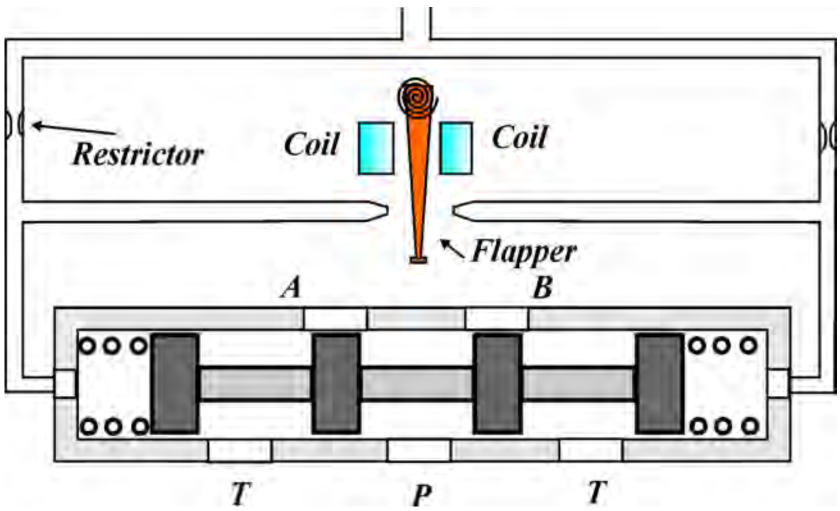


Figure 5.17: *Flapper servo valve*

The third type of servo valves is the jet servo valve shown in Figure 5.18. The main jet pipe is pushed right or left here by means of the electromagnetic force induced by the coils. Pushing the jet to the right causes the fluid to flow from the jet to the right side pipe forcing the spool to move from the right to the left.

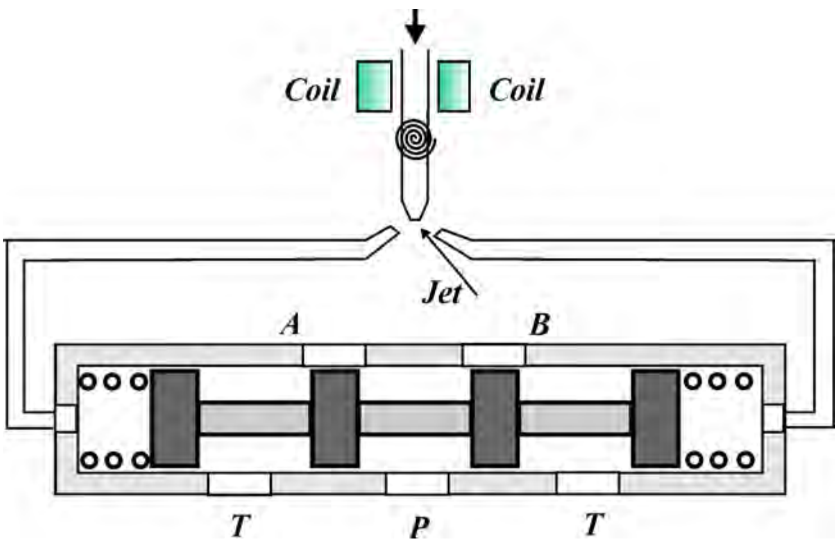


Figure 5.18: *Jet servo valve*

5.5 Proportional valves

In servo valves, the armature moves in a full stroke either open or close (Bing-bang operation). This can be applied for small size applications but in most of the heavy duty applications there is a need to open or close the valve gradually to avoid shocks and rapid transition response from one side and to control flow and pressure according to the load from the other side. This gradual movement can be achieved by using proportional solenoids or voice coil actuators. The force exerted by the armature of the solenoid here is proportional to the current input to the coil of the solenoid. While the relation between the armature force and the displacement of the armature is constant as far as the armature is immersed in the magnetic field of the coil where the force begins decaying after that. Voltage is not adequate for solenoid control because the coil resistance is temperature dependent. A current amplifier is needed here to provide the circuit with an amplified and conditioned current output to control the solenoid as shown in Figure 5.19. The current feedback loop is necessary to reduce the hysteresis caused by the change of the current direction.

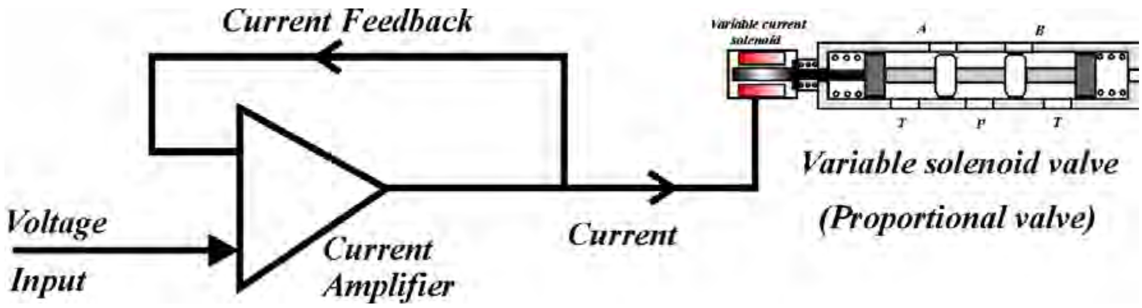


Figure 5.19: *Current control of a proportional solenoid valve*

The block diagram and time response of the current amplifier required to control the proportional valves is shown in Figure 5.20. The amplifier consists of a ramp up step generator that determines the rate of acceleration of the spool to reach the constant speed of motion at the maximum current and a ramp down signal generator that determines the deceleration at which the spool moves before reaching the final position. Another component in the amplification system is the dither. A dither signal is a low level signal superimposed to the input; it is an AC oscillating signal at a rate of about 100 Hz. This signal keeps the spool oscillating around its rest position to overcome static friction and increase the spool's response. The feedback signal fed to the amplifier can come from the following sources:

- *Hysteresis control:* the output current of the amplifier is measured and fed back to the amplifier again to reduce the hysteresis effect which is considered very much higher in proportional valves than servo valves.

- *Spool position control:* a displacement transducer can be attached to the spool of the valve measuring its position and feeding it back to the current amplifier to be corrected according to the required position.

- *Load speed or position control:* the position or the speed of the output load can be measured and fed back to the amplifier to be determined as desired by the operator.

Although there is a feedback loop in the hysteresis control and spool position control but still the circuit is considered an open-loop unless there is a feedback signal coming from the load to control the output.

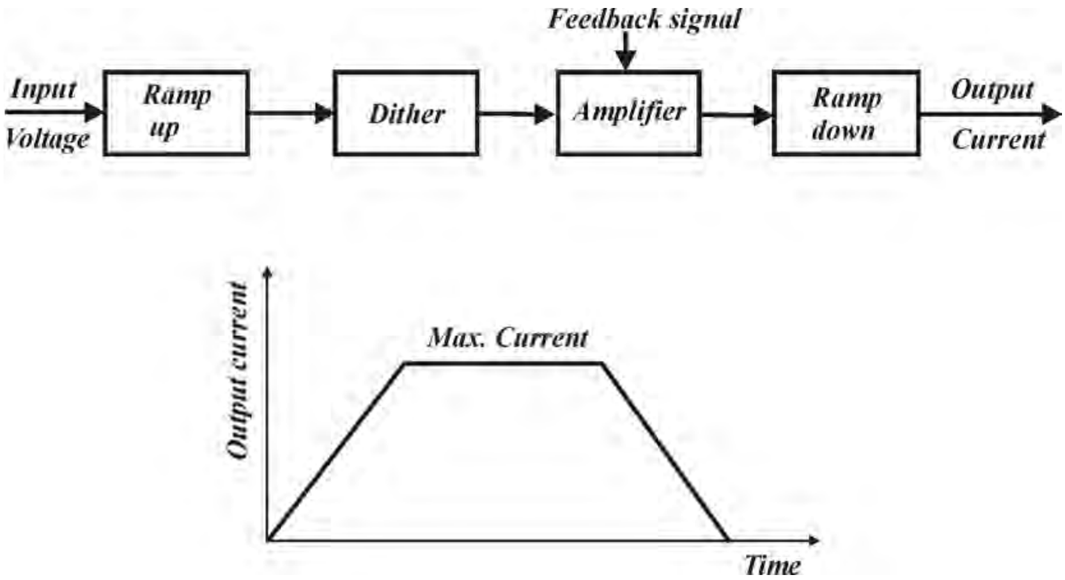


Figure 5.20: *Block diagram and time response of the amplifier used for proportional valves*

Although servo valves are more accurate and convenient than proportional valves for their better response and hysteresis effect, but they are much more expensive to manufacture. Thus, proportional valves are considered an economic and achievable solution when high accuracy and fast response are not essential. Table 5.1 shows a comparison between servo and proportional valves.

5.6 Proportional force control

Variable current is used to control the force of the proportional valves. Using voltage control is not precise enough because the increase in tem-

Table 5.1: *Fluid power symbols (Miscellaneous)*

Characteristic	**Proportional valve**	**Servo valve**
Valve lap	Overlap with dead zone	Zero or underlap without dead zone
Response time	40 - 60 ms	5 - 10 ms
Operating frequency	10 Hz	100 Hz
Hysteresis	1% - 5%	0.1%

perature changes the resistance of the coil. Therefore, the output force is proportional to the input current. A control spring can be added to the valve to help in spool stability and to have more accurate control although this means the need for a higher force to overcome the spring stiffness. As mentioned before, it is difficult and expensive to manufacture zero-lap lands spool valves. Thus, over-lap spool valves are used in proportional valves which implies having a dead zone in which there is a current value (200 mA) without having any fluid flow from the ports as shown in Figure 5.21. To avoid the problem of the dead zone and improve the valve response, notches are added to the lands of the spool as shown in Figure 5.22.

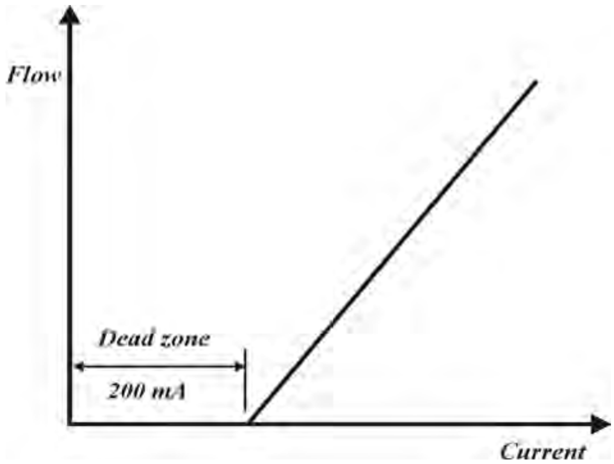


Figure 5.21: *Flow current relationship in proportional valves*

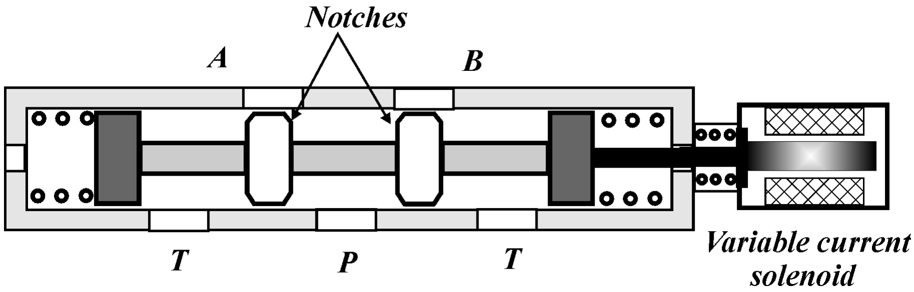


Figure 5.22: *Proportional directional control valve with notched spool*

5.7 Proportional spool position control

The flow rate through the spool valve can be controlled precisely by controlling the position of the spool that allows to open or close gradually the operation ports of the valve. Current feedback control can be done for the spool position control as shown in Figure 5.23. A Linear Variable Differential Transformer (LVDT) or a potentiometer can be used as a transducer fixed to the spool of the valve. The output voltage of the transducer is proportional to the displacement of the spool, this position is fed back to the current amplifier to be compared with the required input position previously determined by the operator. According to this comparison, the current fed into the solenoid is adjusted to fit the required position. Figure 5.23 shows an example of spool position feedback control for a two directional hydraulic motor, where a transducer is used to control the position of the spool but this is not enough to control the actual flow reaching to the load. Therefore, a closed-loop speed feedback control is applied by measuring the speed of the hydraulic motor (using a tachogenerator) and feeding it back to the a current amplifier to be compared to the desired input speed. The influence of the dead zone in the proportional valve should be taken into account in spool position control.

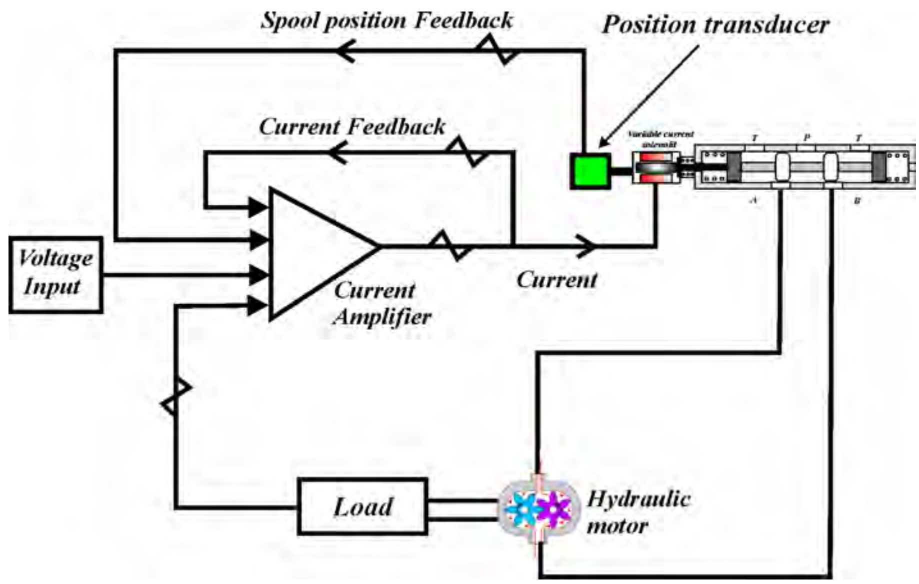


Figure 5.23: *Closed-loop speed control with spool position control for a hydraulic motor*

5.8 Proportional pressure control

In single stage proportional relief valve shown in Figure 5.24, the solenoid acts on the poppet of the valve with a force proportional to the applied current. When the valve is Normally Open (NO) by means of the spring inside the solenoid, the solenoid is actuated to close the valve, the fluid pressure needs to overcome the difference between solenoid force and the spring force to open the valve again. In the other case, the valve is Normally Closed (NC) and the solenoid is used to change the stiffness of the spring changing the force needed to open it and thus changing the fluid pressure required to open the port and allow the fluid to pass.

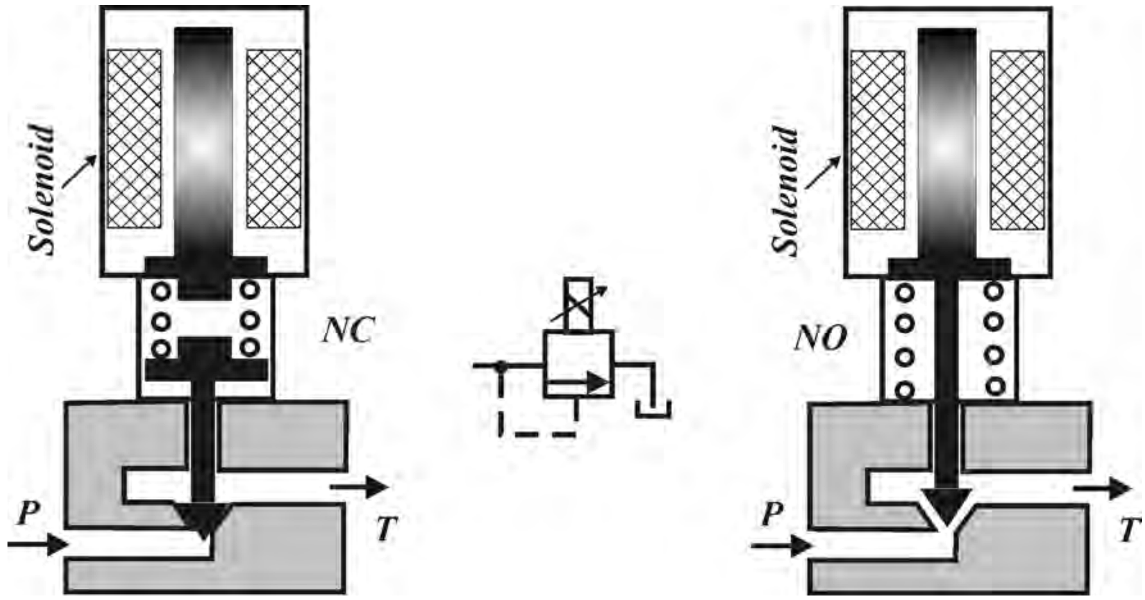


Figure 5.24: *Pressure relief valve with solenoid control; Normally Open (NO) and Normally Closed (NC)*

Increasing the control current flowing into the solenoid increases the pressure needed to overcome the solenoid force and to open the nozzle of the valve having a maximum flow through it. This relation is shown in Figure 5.25 where it shows that the relation is not really linear but there is some non-linearity in the shown curves.

Another main example of using solenoids to control the fluid pressure is shown in Figure 5.26. This system shows a conventional pressure regulating valve. When the solenoid is energized, the fluid flows through the spool of the valve with full pressure value. When the solenoid is de-energized, the spring returns the spool back to its rest position where part of the flowing fluid is fed back to the tank. Intermediate positions of pressure reduction by returning part of the fluid back to the tank can be obtained by changing the control current flowing to the solenoid. Eventually, the pressure of the output fluid is proportional to the control current of the solenoid.

Single stage proportional valves are limited to low flow capacities, less than $5l/min$. To obtain higher flow rates, two stage proportional directional

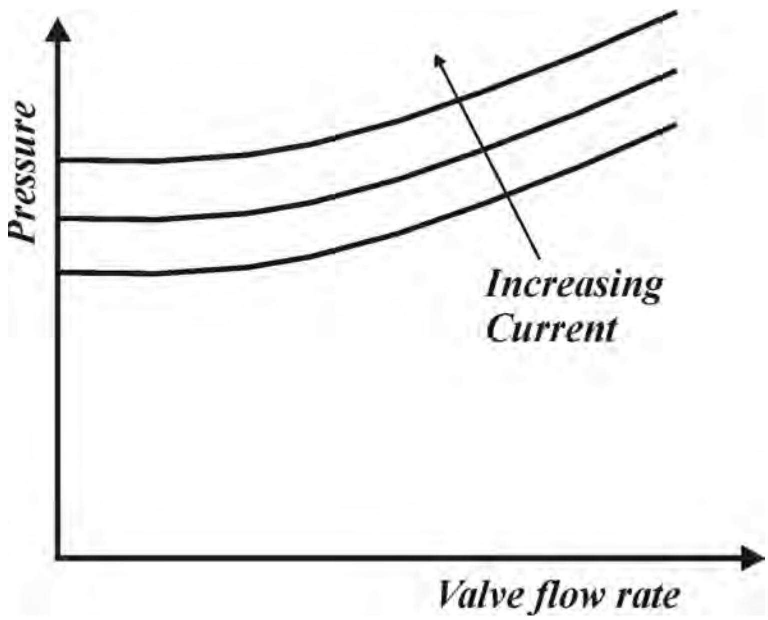


Figure 5.25: *Relationship between pressure and flow rate of fluid through the valve with increasing the control current of the solenoid*

control valves are required like the one shown in Figure 5.27. When the left solenoid is energized, it opens the pressure relief valve allowing the pilot fluid to flow to the left side chamber of the directional control valve. Pressure accumulates behind the spool till it overcomes the spring force acting on the spool. Hence, the spool moves to the right opening the way to the operational pressure to flow through port B. This spool is overlapped with notches and thus it opens gradually having a dead zone of 40-60 ms. Flow through the valve increases non-linearly with the increase of the control current but this changes as the pressure drop across the valve increases as shown in Figure 5.28.

5.9 Proportional flow control

Flow in proportional valves is usually controlled by adjusting the spool position with changing the current of the solenoid but this flow is limited to the size of the port. If this flow is not enough, two ports are connected

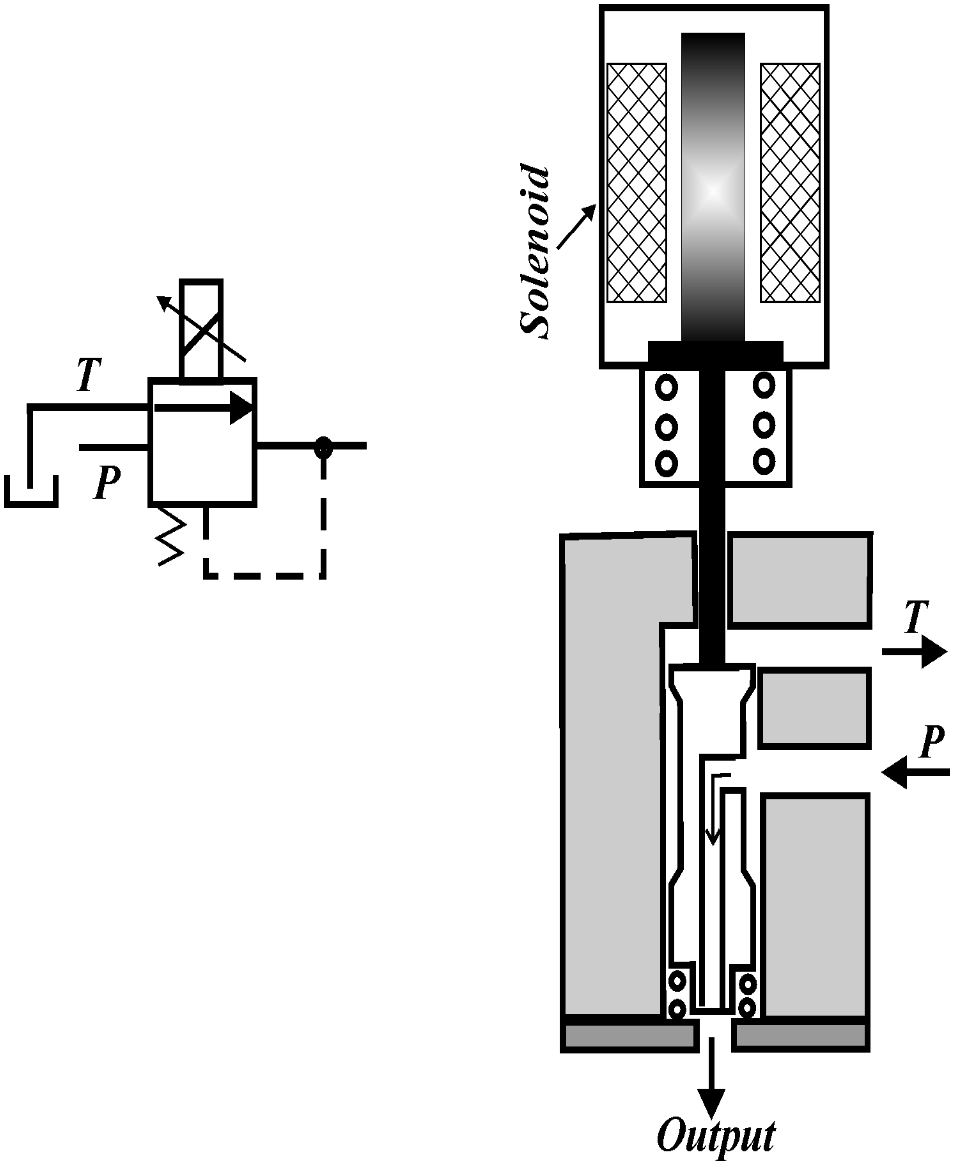


Figure 5.26: *Pressure reduction valve*

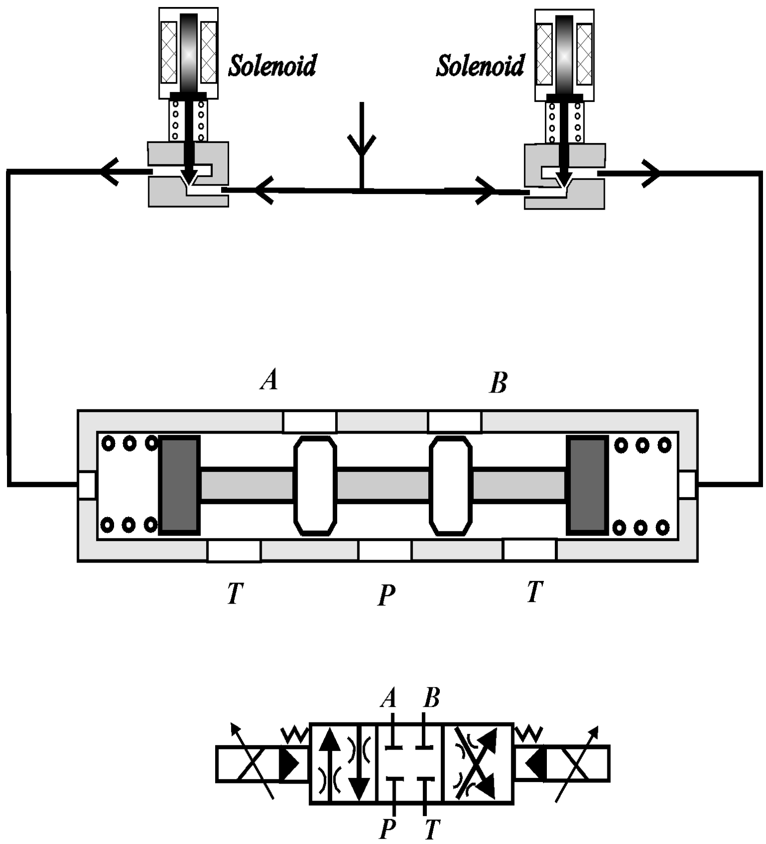


Figure 5.27: *Two stage proportional directional control valve*

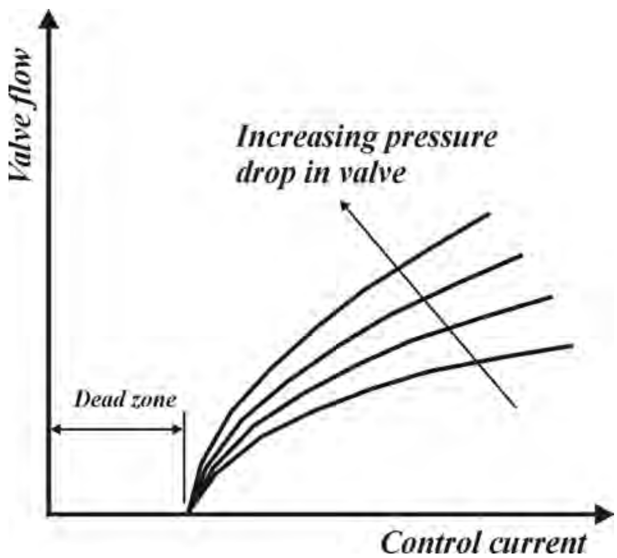


Figure 5.28: *Flow-current relationship with increasing pressure drop in two-stage proportional valve*

together to double the flow as shown in Figure 5.29. In this case too, the flow is proportional to the current passing through the solenoid but the relationship here is not linear and can be adjusted by adding notches to the lands of the spool.

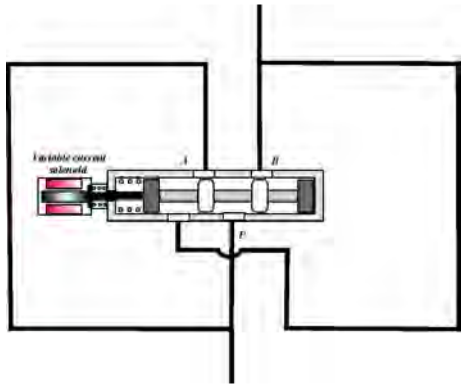


Figure 5.29: *Double flow of four port directional control valve*

Combining the ports to get a double flow can be accompanied by a pressure drop downstream or upstream. Doubling the flow can be obtained without being influenced by the pressure drop by adding a pressure compensation setup that can compensate for the pressure loss by feeding a pilot line of the pressure back to the pressure relief valve and varying the flow orifice to maintain a constant pressure drop as shown in Figure 5.30.

5.10 Speed control of actuators

The aim for controlling actuators is to regulate the speed of the ram to fit for specific applications. The technique used to control the piston speed is by using a cam connected to the piston rod as shown in Figure 5.31. The cam is designed in a way to be able to push the spool of the two position control valve leading the fluid to pass through a flow control that in turn regulates the speed of the piston.

The same techniques used to control the speed of linear actuators can be

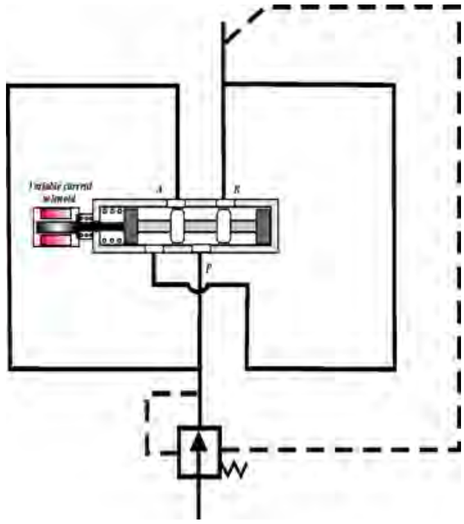


Figure 5.30: *Pressure compensation in double flow of four port directional control valve*

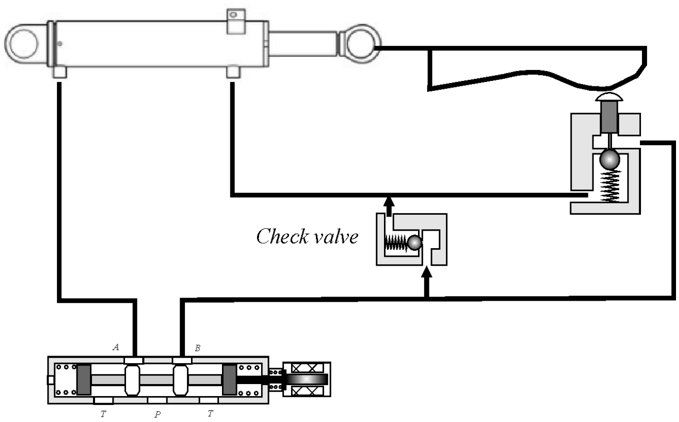


Figure 5.31: *Cam operated speed control of the actuator*

used to control the speed of the rotary hydraulic motors. The disadvantage of these techniques is the slow response due to using proportional valves. In specific applications where rapid response is a need, servo controls are used. Variable displacement pump servo control technique is considered the best way to control hydraulic motors.
The position of the linear actuators (pistons) can be controlled by adding position transducers to the spool of the directional control valve besides to using position sensors on the load to overcome the dead zone difference and feed a signal back to the solenoid through the current amplifier as discussed previously.

5.11 Programmable Logic Controllers (PLC)

5.11.1 PLC layout

Programmable Logic Controller (PLC) shown in Figure 5.32 is a special microprocessor controller that uses programmable memory to store instructions. These instructions are used to function logic orders to control machines and make industrial operations. PLC looks like a computer but it is directed towards industrial applications. Logic operations like (IF) statements and (FOR) loops can be implemented in PLC easily relating inputs to outputs. Figure 5.33 depicts a general layout of PLC controllers where the inputs of the system are processed by the input interface before being fed into the PLC processor (Central Processing Unit - CPU) to be processed according to the programmed instructions, all required instructions are inserted into the processor using an external PC or laptop. A human-machine communication interface is used to allow operators to control processes and operations. The final commands and instructions are transferred to the valve controllers of fluid power system (solenoids, relays, voice coils, ...) through the output interface.

The processor, the input interface and output interface need power supply to convert the input voltage to 5V in the processor and to execute

Figure 5.32: *Programmable Logic Controller - PLC (Courtesy of EATON)*

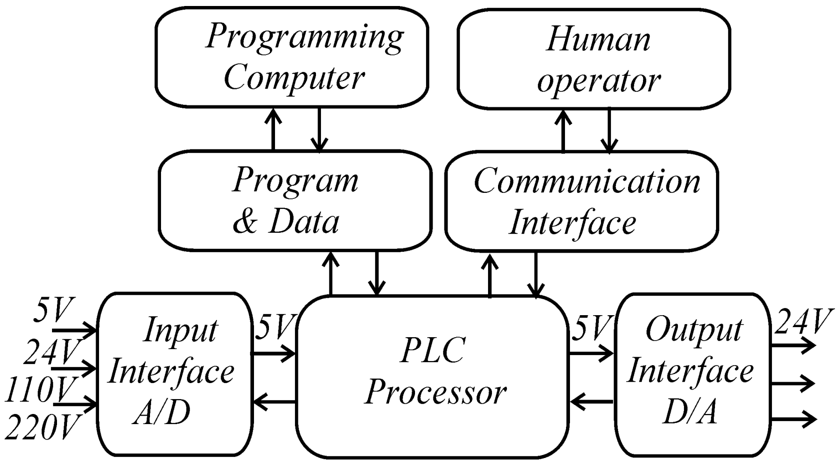


Figure 5.33: *Block diagram for the layout of PLC controller*

the different processing and amplification operations in the CPU. The processed instructions are controlled by the commands programmed in the programming computer and stored in the memory to be used when required. Communication interface is used by the operator to enter any interaction commands during operation and to connect to other PLCs if necessary. The inputs to the PLC can be switches, sensors or external commands and the output signals can be fed into solenoids, relays or voice coils to function pneumatic or hydraulic valves or actuators; or they can be used to function electric or electromagnetic motors. Input interfaces can have 4, 8, 16, 32 module with AC or DC voltage that can be (5V, 24V, 110V or 220V) while all are converted to 5V.

5.11.2 PLC programming

To memorize the instructions and store them in the PLC processor it is necessary to use a programming computer, this computer uses different ways of software to do the job:

1. Ladder Diagrams (LD)
2. Instruction Lists (IL)
3. Functional Block Diagrams (FBD)
4. Sequential Function Charts (SFC)

Ladder Diagram is considered the basic way of programming where other ways depend on it; this is why this section will concentrate mainly on this this technique. Figure 5.34 shows the main structure of Ladder Diagram where the vertical lines are the power lines and the horizontal lines are the rungs containing the program commands. The program starts from left of the first rung moving to the right and at the end of each rung it returns to the left start of the next one. Figure 5.35 depicts the different symbols used in LD programming showing the symbols representing the push-bottom, contact and load and how they are placed in the rungs.

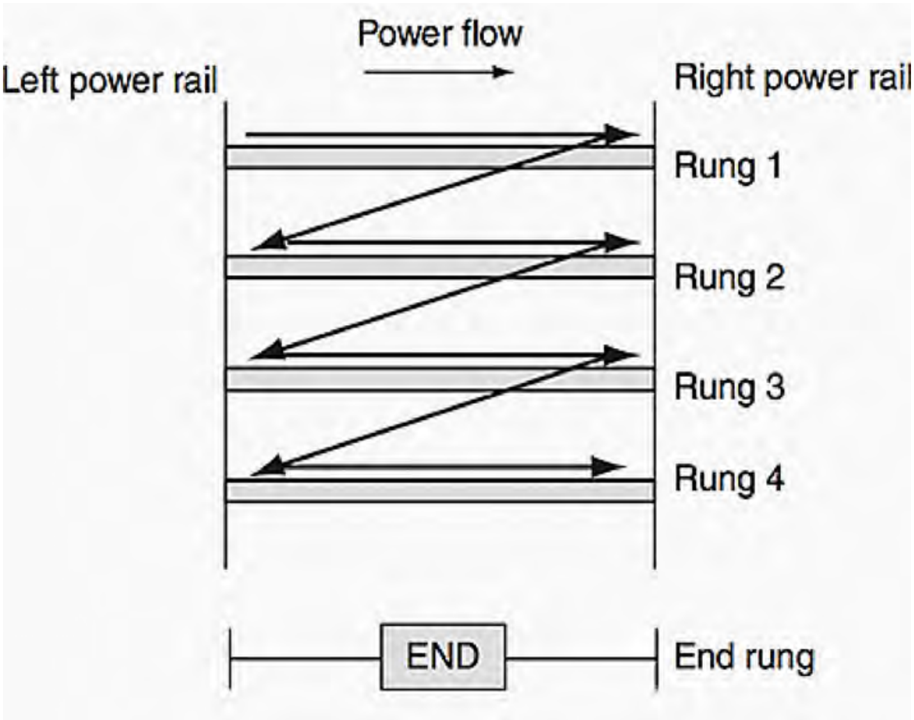


Figure 5.34: *Power rails and rung sequence in Ladder Diagram*

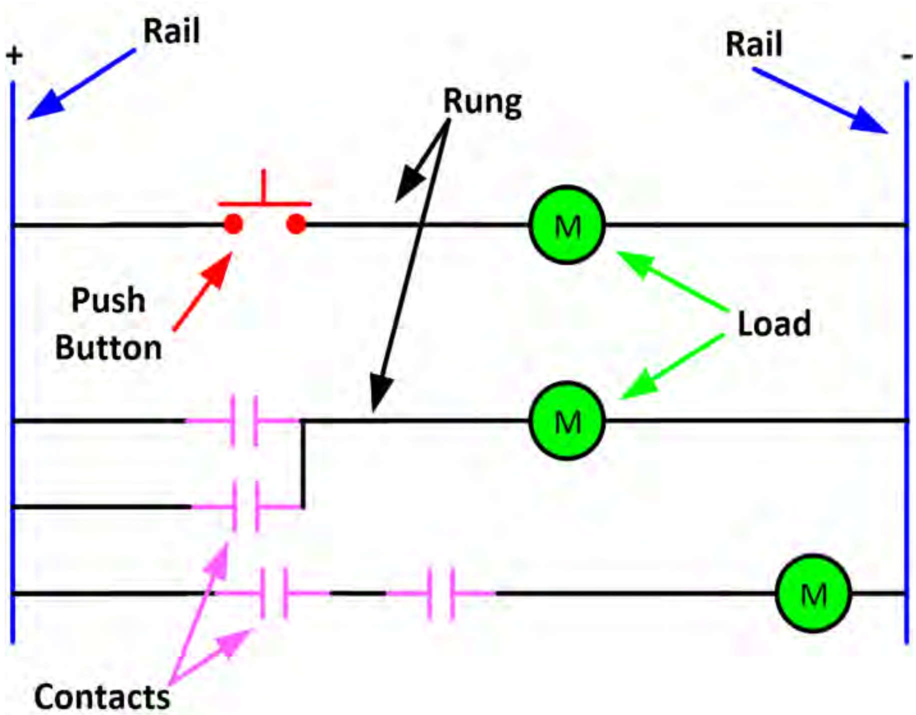


Figure 5.35: *Nomenclature of symbols in Ladder Diagram*

Looking at Figure 5.36 one can see how AND/OR logic statement can be set in Ladder Diagram. In the first rung, to function the output $Y0$, the program functions the contact $X0$ first and then it is necessary to function either the contact $X1$ or the contact $X2$. In the second rung, to function the output $Y1$, it is necessary to function either $X1$ or $X2$ then the contact $X0$ is functioned.

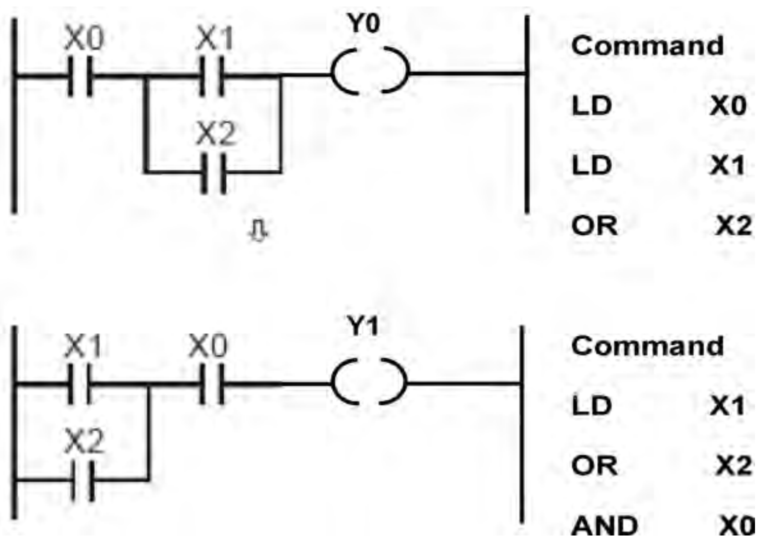


Figure 5.36: *Ladder logic AND/OR statement*

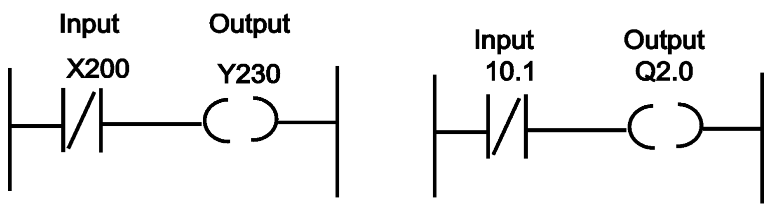


Figure 5.37: *Ladder logic input and output representation depending on the type of PLC manufacturer*

Figure 5.37 shows two types of representation of the inputs and outputs and how they written in LD. On the left the variable $X200$ is used to represent the input and the variable $Y230$ to represent the output, while on the right the input is just a number 10.1 while the output is $Q2.0$. Note that the contact here is normally closed. The foregoing discussion took into account very basic commands and explanation of Ladder Diagram programming for PLC; other commands and statements can be found for further implementation in programming like timer, counter and other advanced statements.

Example 5.1

Write a ladder diagram program to control a double acting hydraulic actuator for both extension and retraction strokes using limit switches for feedback signal of end strokes.

Solution

Consider Figure 5.38 for the schematic circuit and Ladder Diagram of this example:

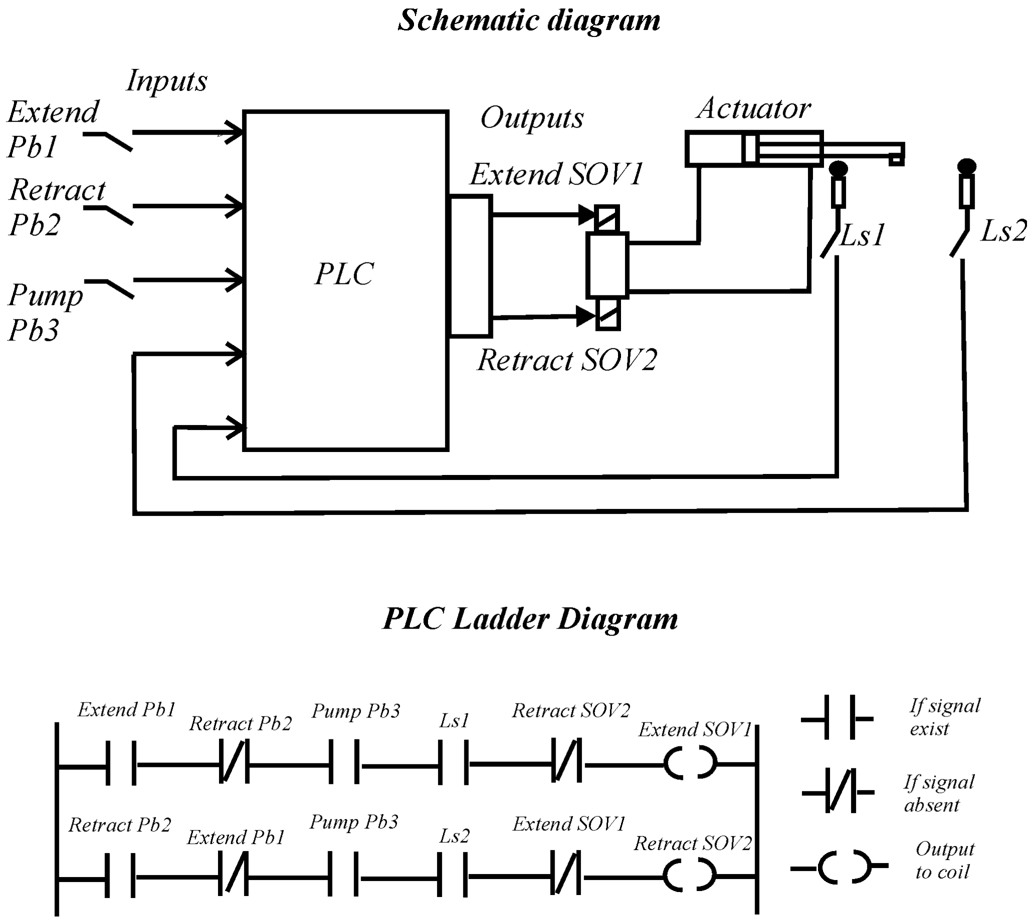


Figure 5.38: *Ladder logic input and output representation depending on the type of PLC manufacturer*

5.12 Fluid power symbols

It is not easy to represent fluid power circuits by technical drawing every time it is required to explain a circuit. Therefore, it is important to use a simplified way of representation. Special symbols are used to represent every part of the circuit. These symbols are considered as a language code and can be understood by all the system engineers working in this field.

5.12.1 Symbols of lines

Table 5.2 shows the symbols used to draw lines and basic variation symbols. In this table, the solid straight line is used to represent the main flow solid pipe line while the solid curved line represents a flexible hose main flow line. The dotted line is used for the pilot low pressure line that brings the control signal and the dash-dotted line is used to enclose specific parts of the circuit for the purpose of grouping. Note here that using inclined arrow means having variable parameters and using perpendicular arrow means having pressure compensation. The solid arrow head is usually used for hydraulic circuits while empty arrow head is used for pneumatic circuits.

5.12.2 Symbols of pumps, compressors and prime movers

Table 5.3 depicts a definition for the different power sources in fluid power systems. Single and reversible direction hydraulic pumps with fixed and variable flow control is shown besides to the symbols of the different prime movers (electric motors and internal combustion engines).

5.12.3 Symbols of actuators

Some symbols used for the actuators are shown in Table 5.4. Symbols of rotary actuators or hydraulic motors are shown for single and reversible direction with fixed and variable displacement control. In linear actuators, it is possible to have a double lined symbol or a single lined symbols (simplified form). It is possible also to show if the piston is with single end or double end rod. Besides to the possibility to show the cushioned design either from one side or from both sides.

Table 5.2: *Fluid power symbols (Lines)*

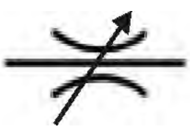
Symbol	**Definition**
	Main line
	Pilot line
	Enclosure outline
	Hydraulic flow direction
	Pneumatic flow direction
	Flexible pipe line
	Constant flow restriction
	Variable flow restriction
	Pressure Compenation (small perpendicular arrow)
	Temperature effect
	Vented reservoir (hydraulic)

Table 5.3: *Fluid power symbols (pumps, compressors and prime movers)*

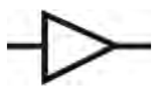
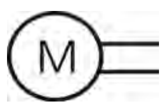
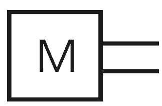
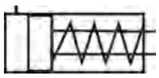
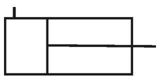
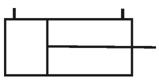
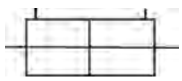
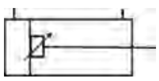
Symbol	**Definition**
	Single direction, fixed displacement pump
	Reversible, fixed displacement pump
	Reversible, variable displacement pump
	Air compressor
	Electric motor
	Internal combustion engine

Table 5.4: *Fluid power symbols (linear and rotary actuators)*

Symbol	**Definition**
	Single direction, fixed displacement hydraulic motor
	Reversible, fixed displacement hydraulic motor
	Reversible, variable displacement hydraulic motor
	Single acting spring loaded actuator
	Simplified symbol of single acting actuator
	Double acting, single end rod actuator
	Double acting, double end rod actuator
	Double acting actuator with adjustable cushion

5.12.4 Symbols of Valves

Table 5.5 shows the different symbolic representations of valves. Different ways of control operations are shown with the corresponding definitions. The table explores how to represent the number of positions and number of ports (ways) for each control valve using boxes and arrows. As an example of how to define a control valve, look at Figure 5.39. The definition for this valve is:

Three position, four port, spring centered, solenoid controlled directional control valve.

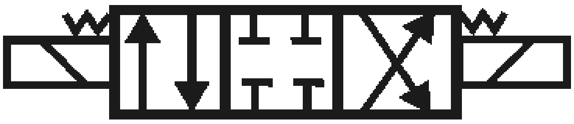


Figure 5.39: *Symbolic representation of a control valve*

5.12.5 Miscellaneous symbols

Table 5.6 includes other miscellaneous symbols of gas and spring loaded accumulators, pressure and temperature indicators, besides to the symbols of heaters, coolers and filters. The FRL (Filter Regulator Lubricator) is shown in the table too. FRL is usually used in pneumatic systems.

5.12.6 Comparison between logic and fluid power symbols

Most of fluid power systems are controlled by electric or electronic components. These electronic components are represented by logic symbols. Pneumatic controls are used instead of electric ones in specific cases. Tables 5.7 and 5.8 show a comparison between logic symbols and fluid power symbols used for the same purpose.

Table 5.5: *Fluid power symbols (valves)*

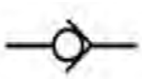
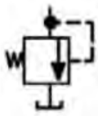
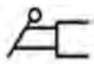
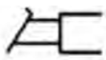
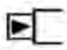
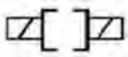
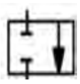
Symbol	**Definition**
	Butterfly manual ON-OFF valve
	Non return (check) valve
	Pressure relief valve
	Manual hand control
	Pedal foot control
	Pressure pilot control
	electric solenoid control
	One position four port valve
	Two position four port valve
	Three position four port valve

Table 5.6: *Fluid power symbols (Miscellaneous)*

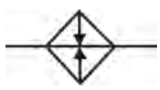
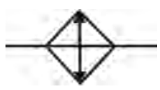
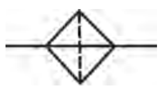
Symbol	**Definition**
	Spring loaded accumulator
	Gas loaded accumulator
	Heater
	Cooler
	Filter or strainer
	Pressure indicator
	Temperature indicator
	Filter Regulator Lubricator (FRL)

Table 5.7: *Comparison between logic symbols and fluid power symbols (basic symbols)*

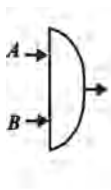
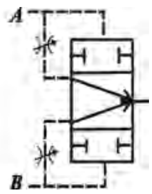
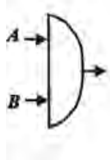
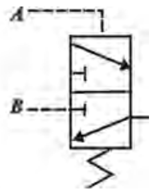
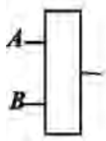
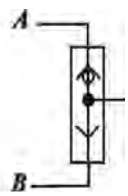
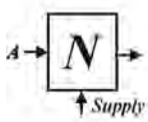
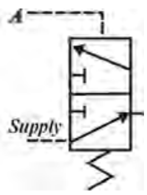
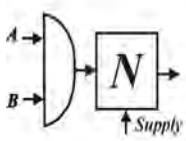
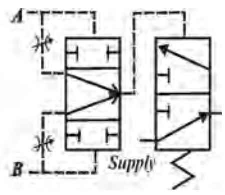
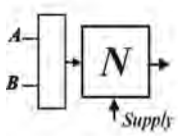
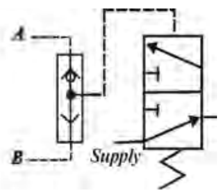
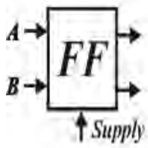
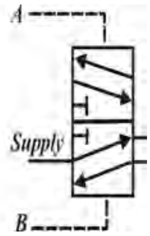
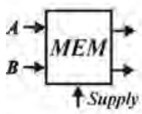
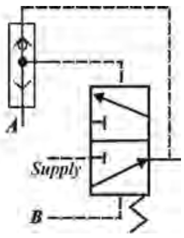
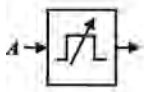
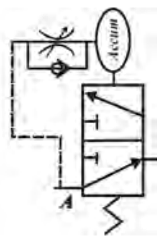
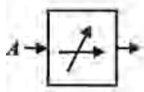
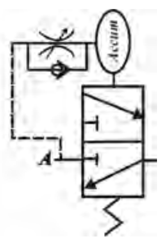
Control element	**Logic symbol**	**Fluid power symbol**
And		
Yes		
Or		
Not		
Nand		
Nor		

Table 5.8: *Comparison between logic symbols and fluid power symbols (miscellaneous symbols)*

Control element	**Logic symbol**	**Fluid power symbol**
Flip flop		
Memory		
Step shot		
Delay timer		

5.13 Problems

1. Derive the magnetic force equation ($F = ILBN$) and show where it is used
2. Draw in schematic form the design of the solenoid and show the principle of working
3. What is the difference between the different designs of voice coil actuators and what does it depend on.
4. What is the main difference between the solenoid and the voice actuator and how can you exploit these characteristics in controlling proportional control valves.
5. A voice coil actuator has a Lorentz Force Constant (G) equals to 1.5, a natural frequency of 5 Hz and damping ratio of 0.03. Find the transfer function between the current input to the actuator and the force output from it and draw a bode plot for this transfer function using MATLAB.
6. Explain how manufacturers can decide the G constant of the voice coil actuator experimentally.
7. Explain using drawing and text the difference between jet and flapper servo valves.
8. What is the purpose of using current feedback in current amplifiers.
9. What is the function of the dither used in the control circuit of proportional control valve.
10. Make a comparison between servo valve and proportional valve.
11. What if the influence of the valve lap and how does it influence the valve dynamic response and dead zone.
12. Consider a hydraulic motor controlled by a Directional Control Valve DCV with spool position feedback and motor speed feedback. Draw a block diagram and derive the transfer function between hydraulic motor speed and the input current to the solenoid of DCV.

13. What is the difference between NO and NC pressure relief valves.

14. Explain the double flow control circuit and show how pressure difference is compensated.

15. Consider the chemical substances mixing process shown in Figure 5.40, where pump 1 is used to deliver chemical 1 and is denoted by Y0, pump 2 is used to deliver chemical 2 and is denoted by Y1, the mixer motor is denoted by Y2, the drain valve by Y3 and the output delivery pump by Y4. It is required to function pump 1 for 5 seconds then stops and pump 2 is functioned for 3 seconds then stops. the mixer motor is required to function for 60 seconds then stops. the main delivery valve after that is opened for 8 seconds during which pump 3 delivers the mixture. Write a logic ladder diagram program to operate the complete process.

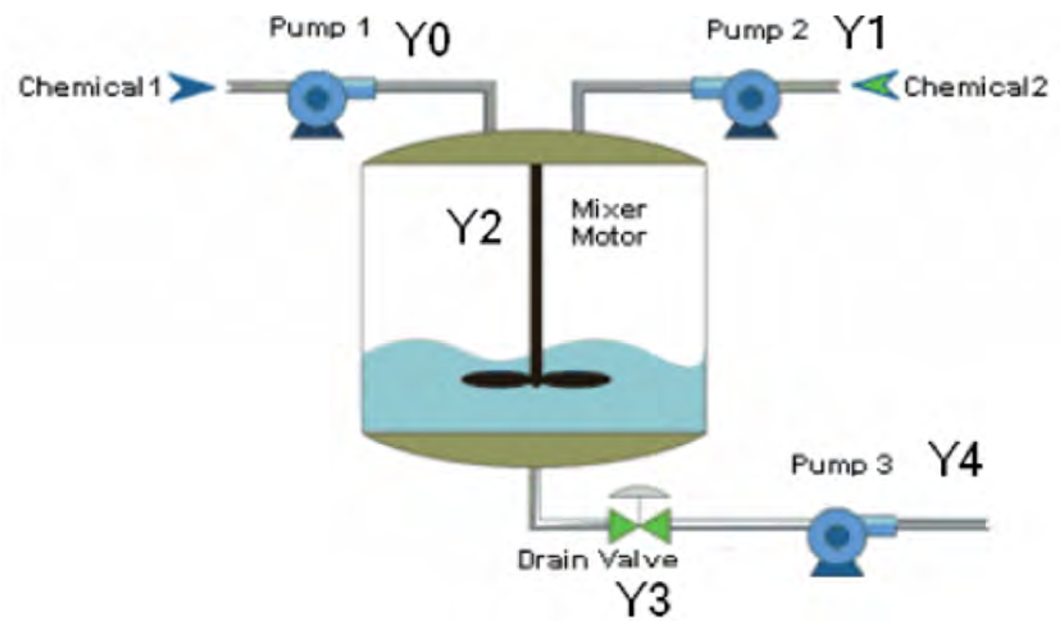


Figure 5.40: *Chemical substances mixing process*

16. Write a Ladder Diagram program to control two reciprocating pneumatic actuators where actuator 2 starts extension when actuator 1 ends retraction and vice versa. Limit switches are placed at the start and end of strokes of each actuator.

5.14 References

[1] J. D. Kraus, *Electromagnetics*, 4th ed., McGraw-Hill international editions, electrical engineering series, 1992.

[2] J.A Edminister, *Schaum's theory and problems, "Electromagnetics"*. Schaum's outline series, McGraw-Hill, Inc., 1993.

[3] A. Fouille, *Electrotechnique a l'usage des ingenieurs.* Dunod, Paris, 1961.

[4] Kimco magnetic division "BEI", *Voice coil actuators and applications guide.* BEI technologies Inc., 1998.

[5] A. Preumont, *Vibration control of active structures*, 1st. ed., Press Universitaire de Bruxelles, pp33-34, 1996-1997.

[6] Brüel & Kjaer, *Brüel & Kjaer, catalog of shakers,* Brüel & Kjaer ,1992.

Chapter 6

Hydraulic Circuits Design and Analysis

6.1 Introduction

This chapter will focus on the design and analysis of different hydraulic circuits and their applications. The circuits discussed here are the systems controlled by hydro-mechanical and electro-mechanical controls. Any mechanically controlled hydraulic circuit like the general one shown in Figure 6.1 consists mainly of:

- Hydraulic tank (reservoir filled with hydraulic liquid). The tank should have a capacity equals or more than twice the system's volume, should have a large surface area to allow better heat transfer and sometimes contain a magnet at its bottom to collect the metallic wastes.

- Hydraulic pump (reciprocating, gear, vane or any other type). Characteristics and selection of pumps will be discussed in next chapters.

- Hydraulic actuator (linear piston or rotary motor). This is the main part of the circuit that provides motion and energy to loads.

- Hydraulic valve (pressure relief, flow control or directional control). Used to control the actuator's motion and regulate its speed and acceleration.

- Prime mover (electric or internal combustion motor). This is the main source of energy used to drive the pumps.

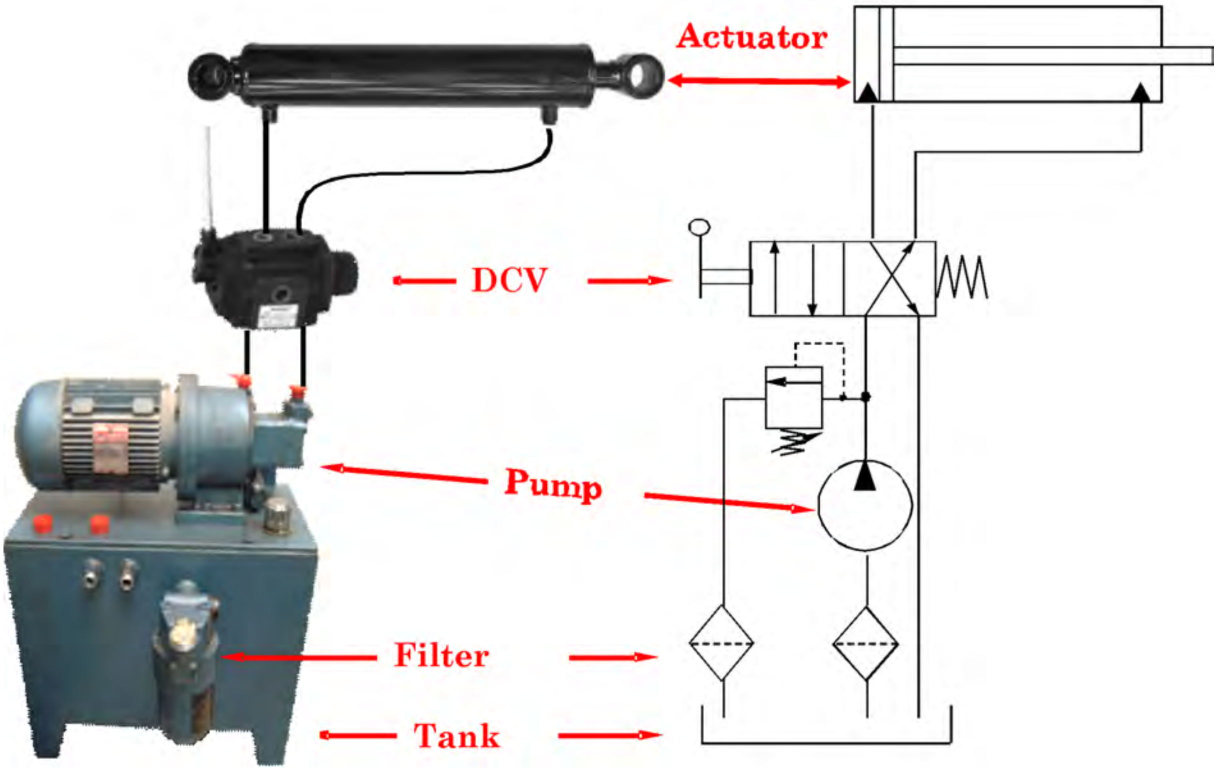


Figure 6.1: *Basic hydraulic system*

Several considerations must be taken into account when designing a hydraulic system:

- Safety of operator and operation (red line): Safety should not be violated for any reason, any item in the system contradicts with safety should be declined. Safety is considered a dead-line or red line that must be met even if it is on the account of cost or efficiency.

- Efficiency and performance of the whole system: After meeting the safety requirements efficiency is considered the second item to maintain in order to get the highest output power using the lowest input one.

- Cost requirements: Hydraulic systems are economic and industrial devices, thus, it is important to minimize the cost of these devices as much as you can after reaching the safety and efficiency requirements.

- Simplicity and easiness: Hydraulic systems are usually complicated because of the need to function more than one operation at the same time. This means that the designer should take into account to simplify the design as much as he can to reduce the cost of manufacturing and maintenance.

The following sections will discuss different basic hydraulic circuits using fluid power symbols listed in the previous chapter that conform with the American National Standards Institute (ANSI).

6.2 Single-acting hydraulic cylinder

6.2.1 Application

The simplest system is the single-acting spring-loaded cylinder shown in Figure 6.2. It is used when the needed action is one sense of direction like pushers and ejectors in production lines.

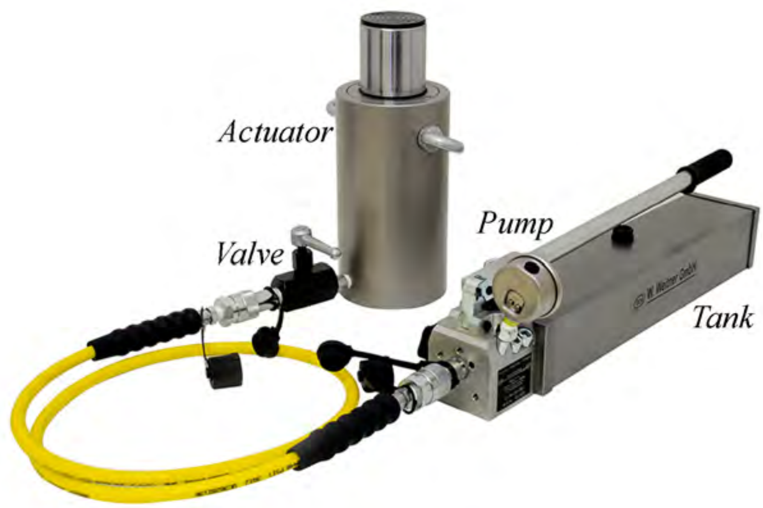


Figure 6.2: *Hydraulic single-acting cylinder with manual pump (Courtesy of SURKON)*

6.2.2 Analysis

Figure 6.3 shows a schematic circuit for a single-acting hydraulic actuator driven by a fixed displacement and single direction hydraulic manual pump and controlled by a two-position three-port manually controlled directional control valve.

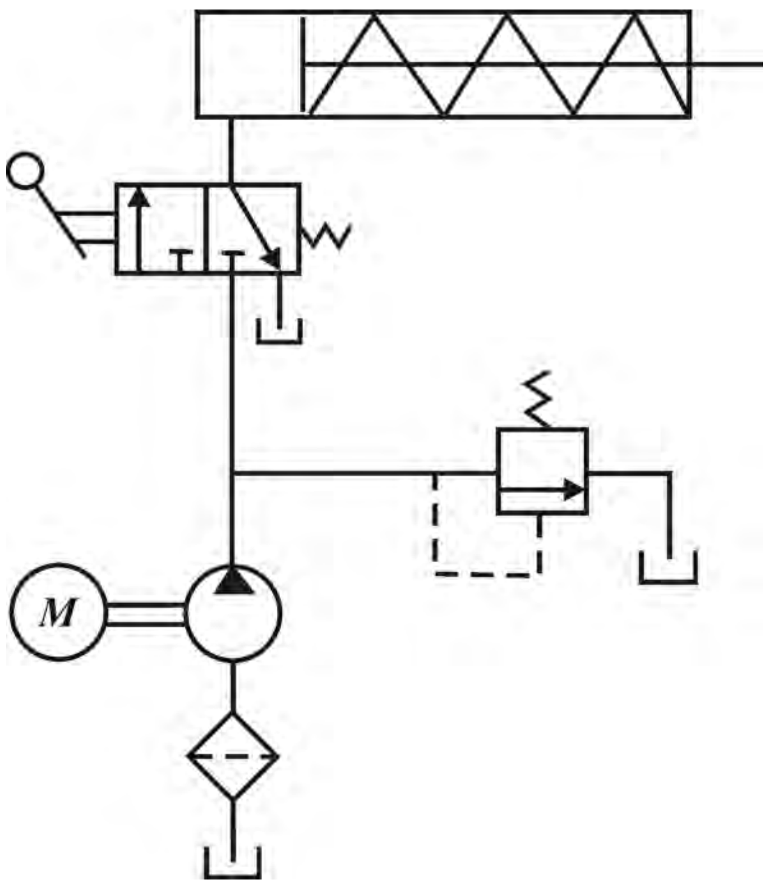


Figure 6.3: *Hydraulic single-acting cylinder*

1. The whole circuit is supplied by a single directional constant displacement pump that pumps the fluid from a tank through a filter or strainer. Pump is driven by an electric motor.

2. The pump is followed by a pressure relief valve calibrated to allow the fluid to run away from the circuit back to the tank when the pressure inside the circuit reaches the calibrated value pressure.

3. This circuit is controlled easily by the two-position three-port manually-controlled directional valve shown.

4. The valve is forced normally by a spring to the right position that draws the fluid from the piston back to the tank and the spring inside the piston forces the rod to return back (retraction stroke).

5. When the manual handle of the directional valve is operated, the

spool moves to the left position passing the fluid to push and extend the piston against its spring (extension stroke).

6.3 Double-acting hydraulic cylinder

6.3.1 Application

Figure 6.4 includes and double-acting hydraulic actuator driven by a manual pump and manually controlled directional control valve. This type of actuator is used when there is a need for external power to extend and retract the actuator.

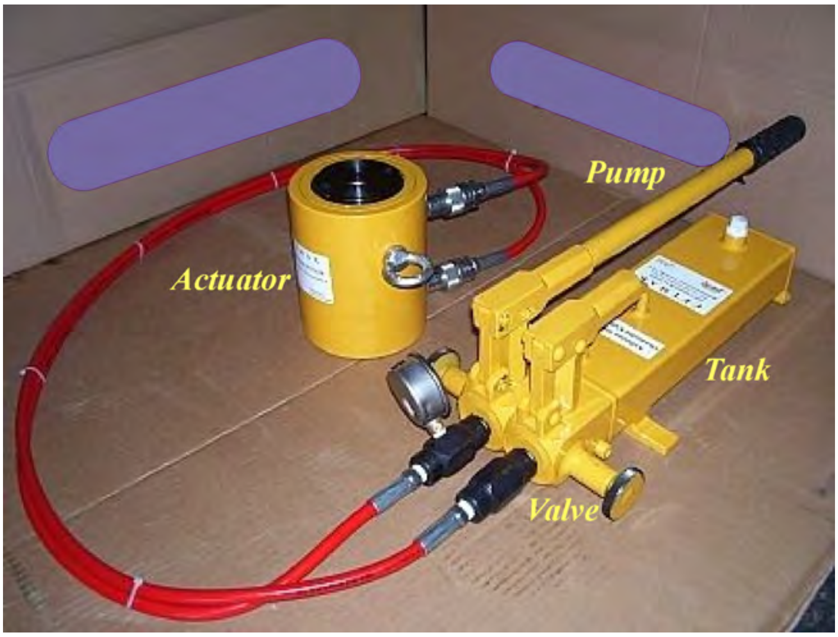


Figure 6.4: *Hydraulic double-acting cylinder with manual pump (Courtesy of RITM industry)*

6.3.2 Analysis

A double-acting hydraulic cylinder shown in Figure 6.5 is used in most of the hydraulic applications where a linear actuation is required in two directions.

1. The pump driven by an electric motor extracts the oil from the oil reservoir via a filter or strainer.

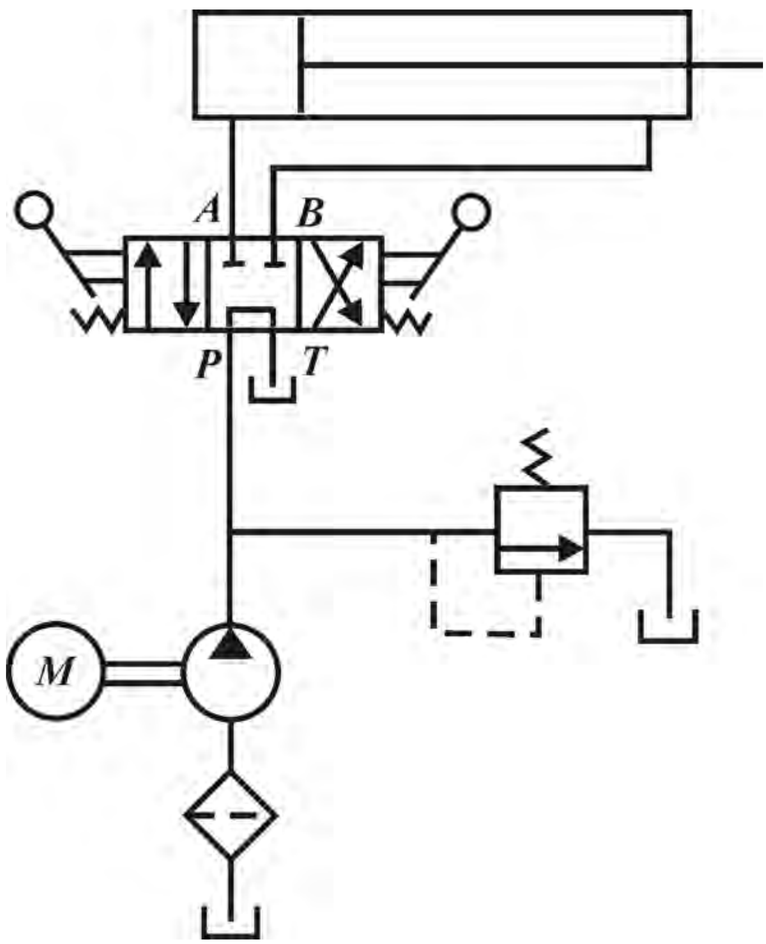


Figure 6.5: *Hydraulic double-acting cylinder*

2. The pressure relief valve (PRV) allows the oil pressure in the system to reach a specific value after which the PRV opens passing some fluid back to the tank to regulate the pressure.

3. The circuit is controlled by the three-position four-port spring-centred manually-operated directional control valve (DCV).

4. When the DCV is on its centre position, there is no fluid flow through it which keeps the piston hydraulically locked in its position.

5. Moving the manual handle of the DCV to the left moves the spool of the valve to the left position, which allows the oil to flow from the pressure line P to port A exerting force on the blank side of the piston extending it to the right. This motion allows the oil to flow from port

B coming from the rod side to the tank T.

6. Moving the manual handle of the DCV to the right moves the spool of the valve to the right position, which allows the oil to flow from the pressure line P to port B exerting force on the rod side and retracting the piston to the left. This motion allows the oil to flow from port A coming from the blank side to the tank T.

7. When the piston reaches the dead end on either side, the pressure increases in the pressure line and this can be regulated by the PRV.

6.4 Regenerative hydraulic cylinder

6.4.1 Application

The regenerative circuit shown in Figure 6.6 is usually used to increase the speed of the piston in extension stroke keeping the retraction speed the same as the normal double-acting cylinder. This circuit can be used in shapers to increase the speed of the return stroke (instead of quick return mechanism) and in drilling machines as will be discussed later.

6.4.2 Analysis

1. The pump extracts oil from the oil tank via a filter or strainer and the PRV aims at keeping specific pressure in the system when the piston reaches dead ends.

2. One of the output ports of the 3-position 4-way DCV is blocked to allow summing the flow in one port.

3. When the DCV is moved to the right position, the piston retracts normally like a normal double-acting cylinder.

4. When the DCV is set to its left position, the fluid flows to extend the piston to the right sense of direction but with a speed higher than the normal double-acting cylinder because the flow from the rod side Q_R regenerates and sums with the flow of the pump Q_P resulting in

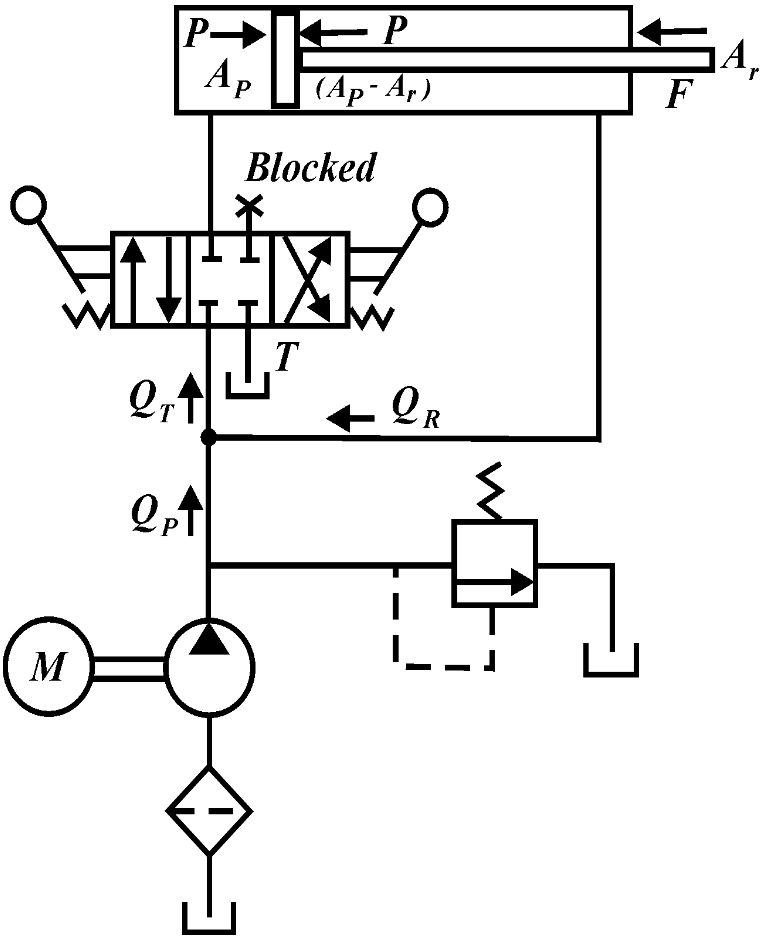


Figure 6.6: *Regenerative hydraulic cylinder circuit*

a total flow rate Q_T. Increasing the total flow increases the speed of the fluid increasing the piston speed.

Estimating the flow rate of the pump in the extension stroke of the piston:

$$Q_P = Q_T - Q_R$$

Denoting the extension speed of the piston as v, the area of the piston side as A_p and the area of the rod side as $A_p - A_r$, the pump flow rate becomes:

$$Q_P = A_p v - (A_p - A_r)v$$

Solving for the extension speed of the piston gives:

$$v = \frac{Q_P}{A_r}$$

But the force exerted by the regenerative circuit is calculated by:

$$F = PA_r$$

Which is considered lower than the force obtained by normal double-acting cylinder because the rod area is less than the piston area. This leads to the fact that the regenerative circuit increases the speed but decreases the obtained power to carry the load.

One of the main applications on regenerative circuit is the hydraulic drilling machine circuit shown in Figure 6.7.

A hydraulic drilling machine is controlled by a three-position four-way DCV where one of the output ports is regenerated with the other output port. The left position of the DCV causes a normal (slow) extension to operate the drilling feed. The right position causes a normal retraction of the piston after the drilling operation. The spring centred position causes a rapid advance in the extension stroke to approach the drill to the workpiece before beginning the drilling operation to save time in work.

Example 6.1

Consider a hydraulic drilling machine with a regenerative circuit functioning a double acting actuator with a bore of $100mm$ and a rod of $40mm$. The exerted force on the actuator is $10kN$. Knowing that the oil flow rate reaching to the piston equals $0.004m^3/s$, calculate the required flow rate delivered from the pump and the pressure in the system.

Solution

The area of the piston and the rod

$$A_P = \pi \times (0.05)^2 = 0.0079m^2$$

$$A_r = \pi \times (0.02)^2 = 0.0013m^2$$

The velocity of the piston

$$v = \frac{Q_P}{A_r} = \frac{Q_T}{A_P}$$

$$v = \frac{0.004}{0.0079} = 0.5m/s$$

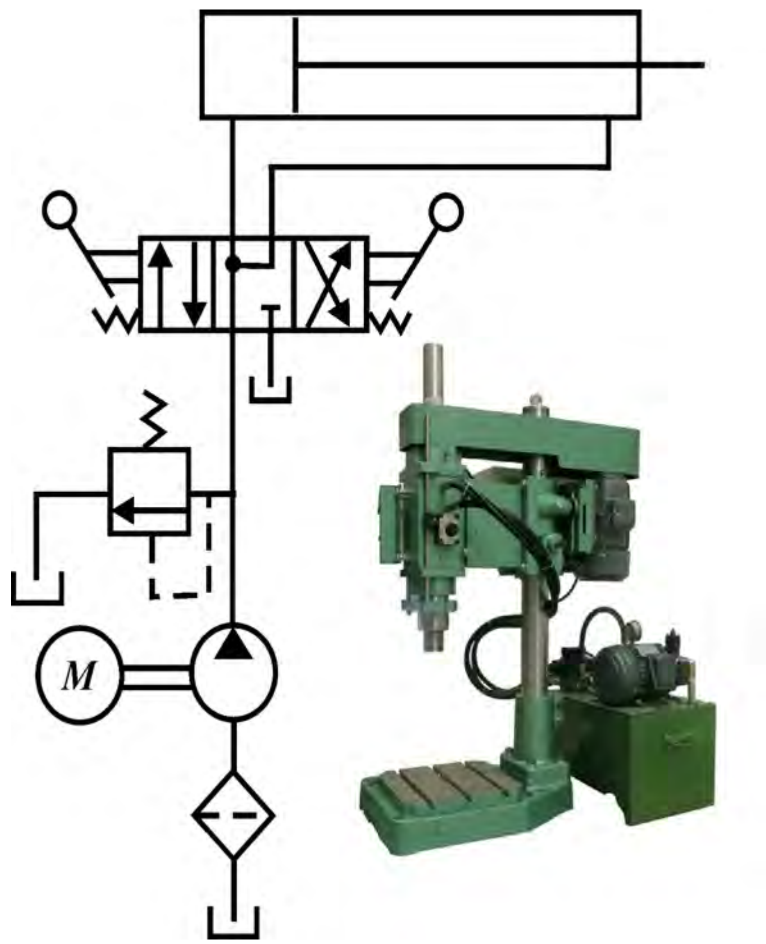


Figure 6.7: *Drilling machine circuit (Courtesy of MEC)*

The flow rate delivered by the pump

$$Q_P = A_p v - (A_p - A_r)v$$

in numbers

$$Q_P = (0.0079 \times 0.5) - (0.0079 - 0.0013) \times 0.5 = 6.5 \times 10^{-4} m^3/s$$

The pressure in the system

$$P = \frac{F}{A_P} = \frac{10 \times 10^3}{0.0079} = 1.26 \times 10^6 Pa = 12.6 bar$$

6.5 Double-pump hydraulic system

6.5.1 Application

The circuit shown in Figure 6.8 is based on using a high-pressure low-flow pump and another low-pressure high-flow pump to feed the system with oil. This circuit is used in general to design a hydraulic punch press for shearing and forming of sheet metals. The main purpose of this circuit is to introduce a rapid extension stroke of the piston under low pressure for the free of load stroke and a slow retraction stroke with high pressure to exert high force when the punch approaches the workpiece to execute the cutting or forming operation.

6.5.2 Analysis

1. The high-flow low-pressure pump delivers oil to the circuit when there is no punching load on the piston in the extension stroke.
2. The three-position four-port DCV is used to control the direction of operation.
3. When the punch approaches the workpiece, the high punching load increases the pressure in the system which opens the unloading PRV by the pilot line eliminating the influence of the high-flow pump and operating the low-flow high-pressure pump only to exert higher force on the load executing the punching operation.
4. As soon as the punching operation is finished, the pressure in the system reduces leading the pilot line to close the unloading PRV and to return the effect of the high-flow pump and increasing the speed of the punch in the retraction stroke.

6.6 Locked cylinder hydraulic system

6.6.1 Application

In most of the applications where the hydraulic system is in contact with with human beings, there is a need to lock the cylinder in its position to

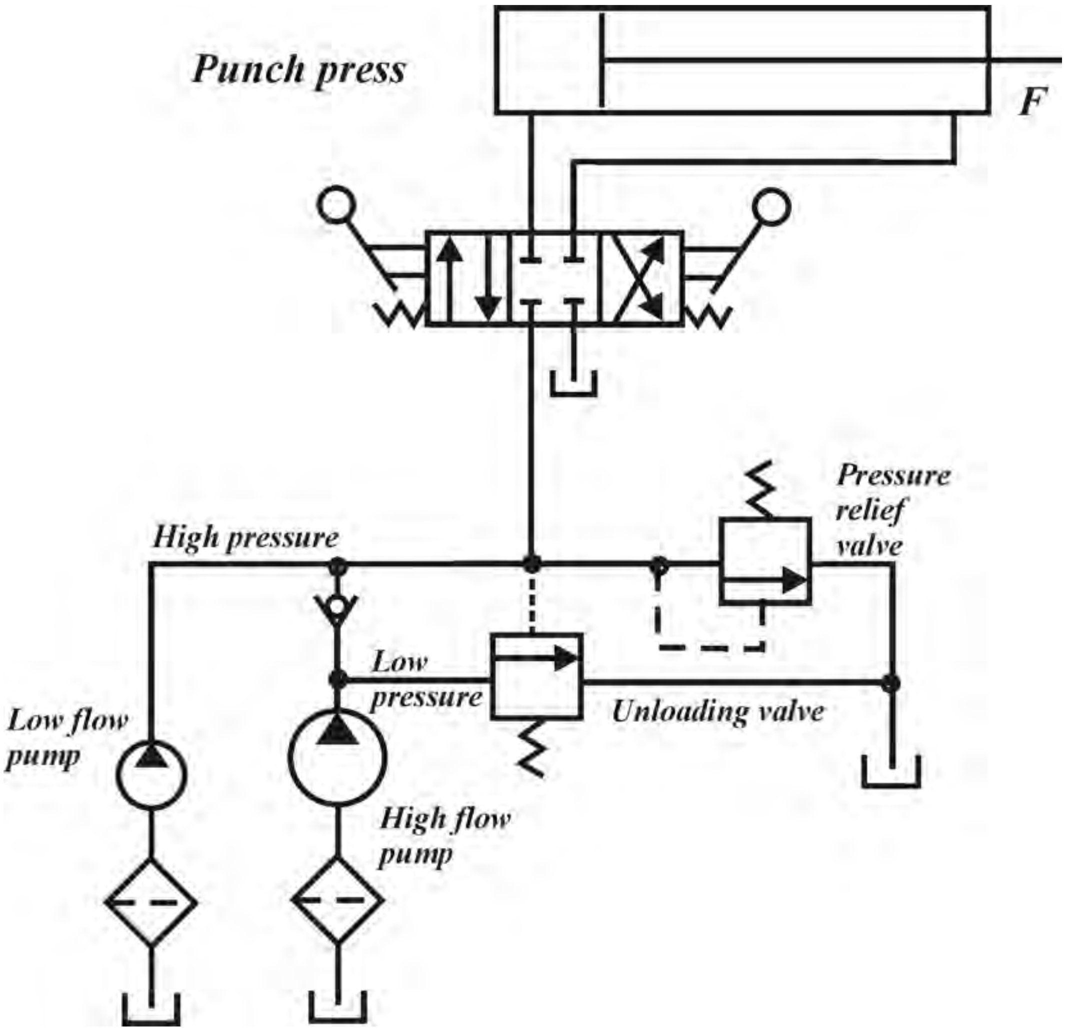


Figure 6.8: *Double pump hydraulic system (Punch press)*

avoid causing harm to people. This can be done by using pilot check valves as shown in Figure 6.9. Examples of these applications are lifts, cranes, concrete pumps, fork lifts and loaders.

6.6.2 Analysis

1. Oil is pressurized into the system by a pump fed by an oil reservoir via an oil filter. A PRV is used here to regulate and avoid the overload pressure in the circuit.

2. Setting the three-position four-way DCV to the left position causes an increase in the pressure line to extend the piston.

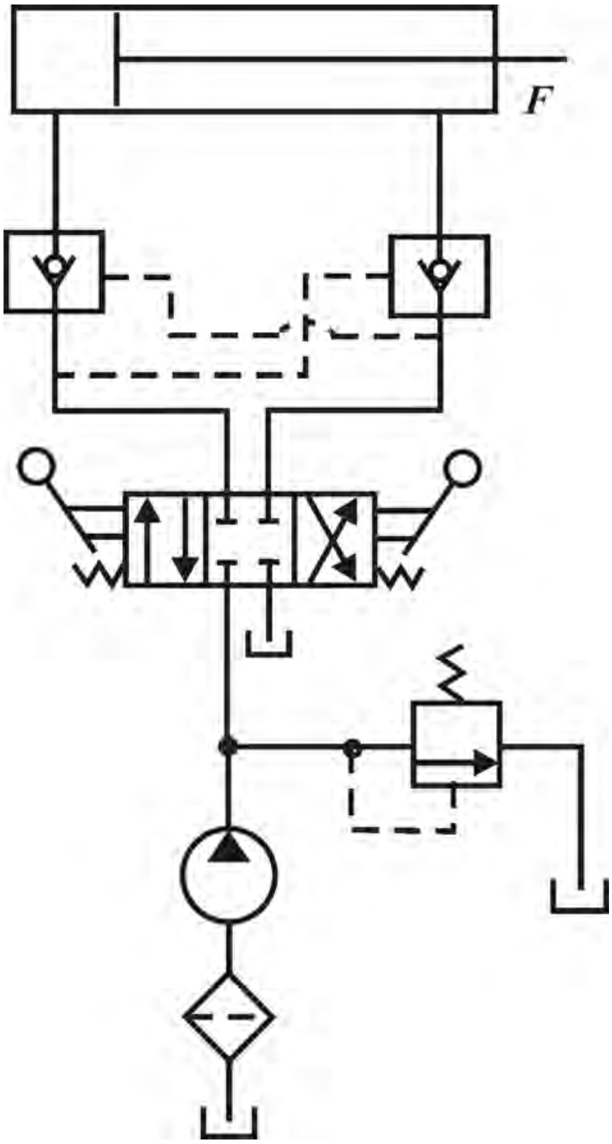


Figure 6.9: *Locked cylinder hydraulic system using check valves*

3. Oil can pass freely from the left hand check valve but the right hand check valve resists this flow.

4. When pressure increases in the left line, the pilot line opens the right check valve permitting the fluid to flow and allowing the piston to extend.

5. changing the DCV to the right position, reverses the operation causing the left check valve to open and the piston to retract.

6. when the DCV is on its middle spring-centered position, the piston is locked in its place without being able to extend or retract under any external load.

6.7 Counterbalance hydraulic system

6.7.1 Application

A counterbalance installation (shown in Figure 6.10) is used to protect a vertically mounted hydraulic cylinder from moving under vertical loads when the pump is idling. This system is used when a load is hanged in an upward position to a hydraulic cylinder.

6.7.2 Analysis

1. Oil is pressurized into the system by the pump fed by an oil reservoir via an oil filter. A PRV is used here to regulate and avoid the overload pressure in the circuit.

2. The system is controlled by a three-position four-way spring-centred solenoid-controlled DCV.

3. Moving the spool to the left position allows the oil to flow through the check valve to the rod side of the piston causing retraction operation upwards. The required pressure here is to overcome the load.

4. Changing the DCV to the right position passes the flow to the blank side of the piston to extend it. The oil flow out of the piston is

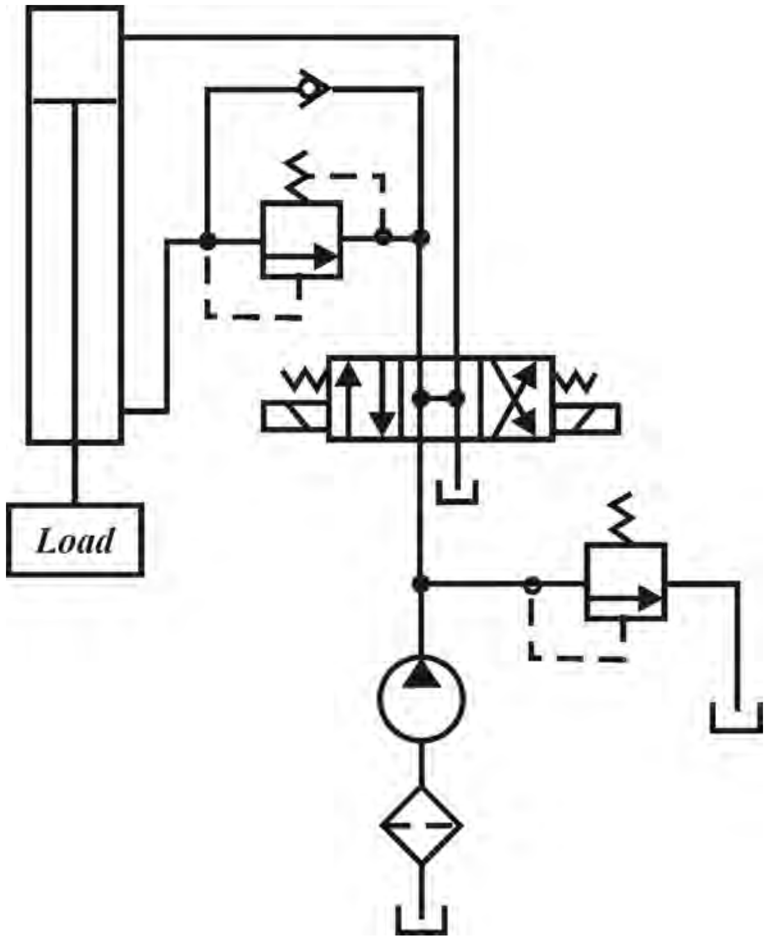


Figure 6.10: *Counterbalance hydraulic system*

restricted by the pilot PRV. A specific value of pressure is needed to do this operation. Thus, the PRV is set to a pressure value that can overcome the load with a small overhead.

5. When the DCV is on centre position, all lines are open to the tank including the pressure line coming from the pump which causes the pump to circulate oil to the tank. In this position, the load is kept hanged by the pressure of the PRV that behaves like a counterbalance.

6.8 Sequence cylinder hydraulic system

6.8.1 Application

The cylinder sequencing circuit shown in Figure 6.11 is used for executing operations in sequence like folding the sides of a metal scrap press or packaging and arranging boxes in warehouses.

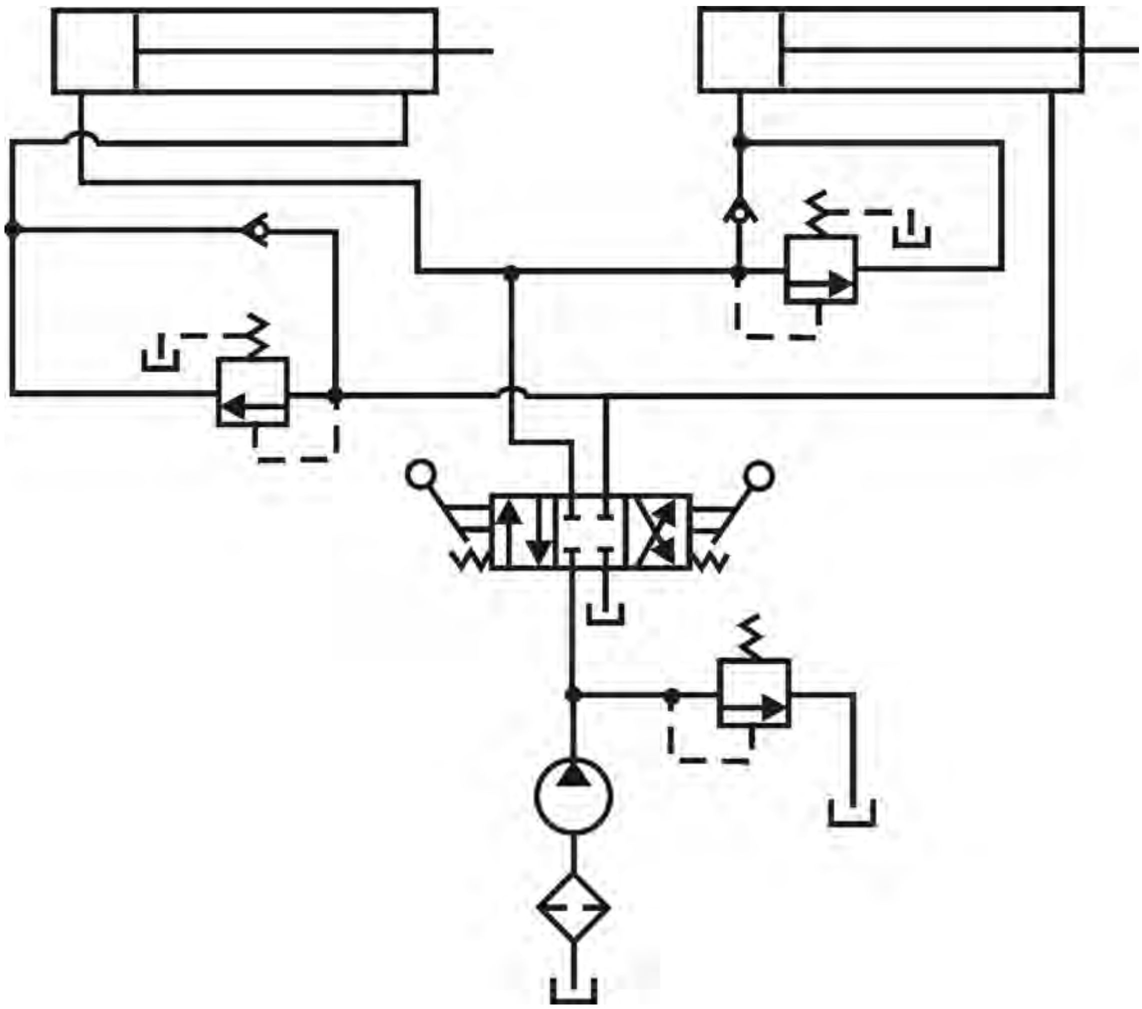


Figure 6.11: *Sequence cylinder hydraulic system*

6.8.2 Analysis

1. Oil is pressurized into the system by the pump fed by an oil reservoir via an oil filter. A PRV is used here to regulate and avoid the overload

pressure in the circuit.

2. When the DCV is moved to the left position, the oil selects to pass to the left cylinder which is the easiest way leading to full extension of this cylinder.
3. Reaching the dead end (full stroke) of the left cylinder increases the oil pressure in this line which opens the PRV at the inlet of the right cylinder and extends the right cylinder.
4. From the previous two steps 2 and 3, one can see that the operation is executed by extending the left piston first and then the second piston in sequence.
5. The retraction operation occurs when the spool of the DCV is pushed to the right position which causes retracting the right cylinder first followed by the retraction of the left cylinder in sequence.
6. In each operation, the oil returns back to the tank through the check valve installed on the by-pass line passing around the PRV.
7. The spring centered mid position of the DCV blocks the two cylinders in their positions.

6.9 Automatic reciprocating hydraulic system

6.9.1 Application

The automatic reciprocating system shown in Figure 6.12 can be used to function a double-acting hydraulic cylinder back and forth automatically. This system is used to operate the pumping reverser in some concrete pumps and in the automatic reciprocating motion of the table of a surface finishing machine.

6.9.2 Analysis

1. Oil is pressurized into the system by the pump fed by an oil reservoir via an oil filter. A PRV is used here to regulate and avoid the overload pressure in the circuit.

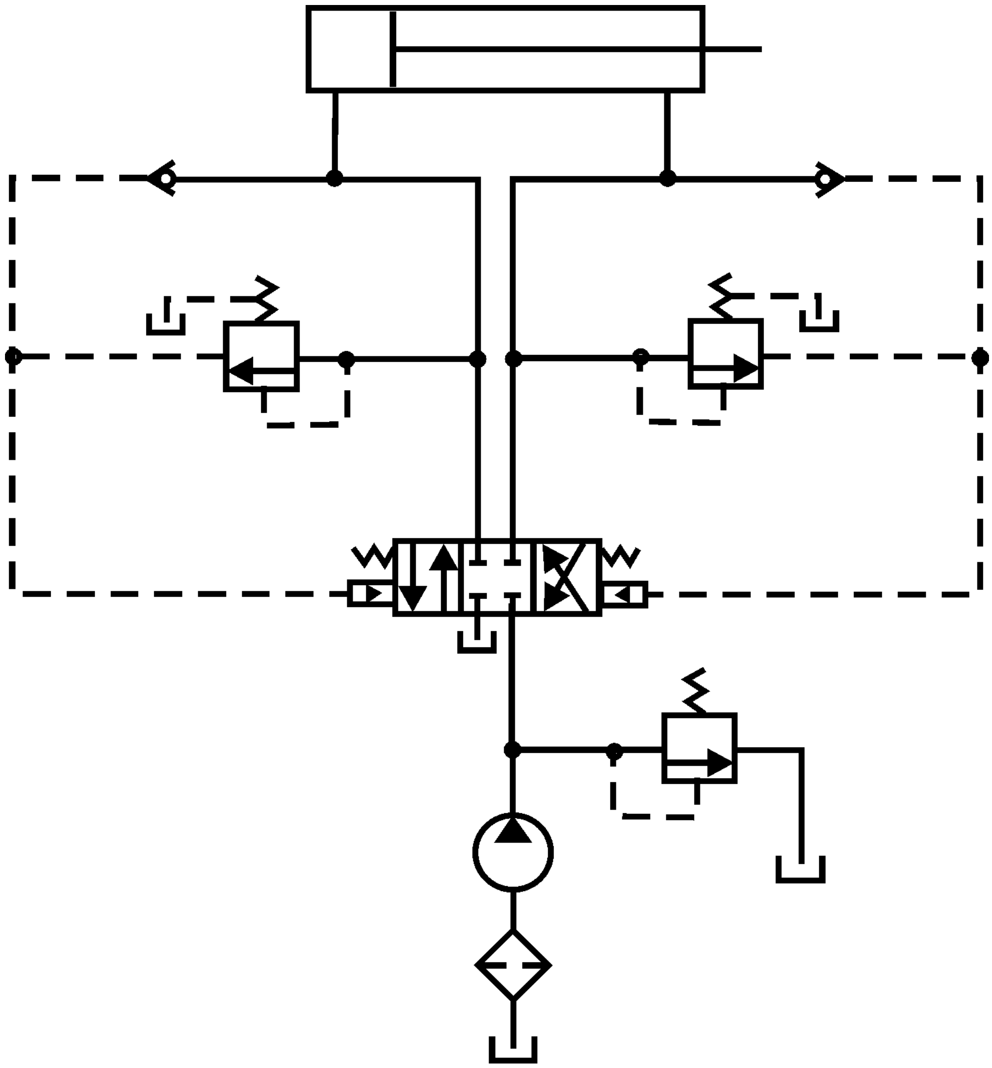


Figure 6.12: *Automatic reciprocating hydraulic system*

2. A three-position, four-way spring-centred and pilot-operated DCV is used to control this circuit.

3. Leaving the DCV on the centre position keeps the cylinder blocked in its position.

4. The reciprocating operation begins by moving the DCV to any of the right or left positions.

5. Beginning from the right position of the DCV; the pressurized oil flows in the left line and extends the cylinder till reaching full stroke.

6. At full extension stroke, the pressure builds up in the left line and opens the left PRV sending oil via the pilot line to the left pilot control in the DCV moving the spool to the left position.

7. The spool now is on the left position of the DCV; the pressurized oil flows in the right line and retracts the cylinder till reaching zero stroke.

8. At complete retraction, the pressure builds up in the right line and opens the right PRV sending oil via the pilot line to the right pilot control in the DCV moving the spool to the right position.

9. And so on and so forth ...

6.10 Parallel connected hydraulic cylinders

6.10.1 Application

Having an identical motion of two cylinders in synchronization can be obtained by connecting the two cylinders in parallel as shown in Figure 6.13. Two conditions must be satisfied to obtain this synchronization:

- The two cylinders have to be completely identical.
- The two loads acting on the cylinders have to be equal.

It is worth mentioning that it is impossible to have two identical cylinders because of the differences in friction and accuracy in manufacturing. On the other hand, it is not easy to get two equal loads on the cylinders. Nevertheless, cylinders in parallel are used specially in loader bucket and heavy booms that need two pistons to act at the same point.

6.10.2 Analysis

1. The pump, tank, filter and PRV have the same function as in previous circuits.

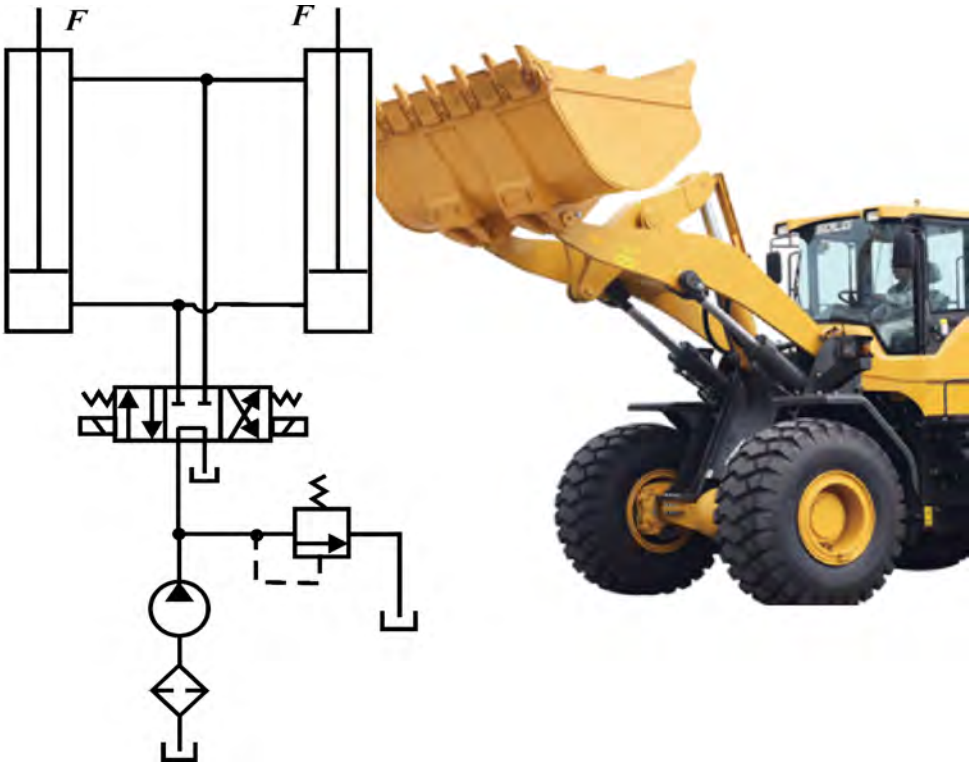


Figure 6.13: *Hydraulic cylinders connected in parallel (Courtesy of SDLG)*

2. leaving the three-position four-way solenoid-controlled DCV on the centre position by the influence of the springs causes the pump to circulate the oil to the tank without influencing the system.
3. Moving the DCV to the left position pressurizes the oil to extend the two pistons simultaneously if they are identical and if the two loads are equal.
4. Moving the DCV to the right position pumps the oil to retract the two pistons under the same conditions.

6.11 Series connected hydraulic cylinders

6.11.1 Application

The best way to obtain complete synchronization for two cylinders regardless of the acting loads is connecting them in series as shown in Figure 6.14. The condition for this connection is to have the blank area of the second cylinder equals to the difference between the blank area and the rod area of the first cylinder.

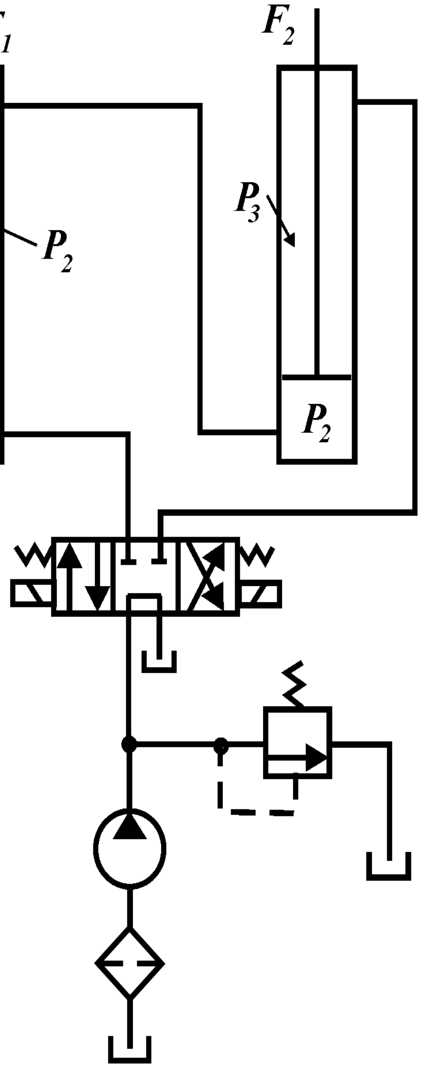


Figure 6.14: *Hydraulic cylinders connected in series*

6.11.2 Analysis

1. The pump, tank, filter and PRV have the same function as in previous circuits.

2. leaving the three-position four-way solenoid-controlled DCV on the centre position by the influence of the springs causes the pump to circulate the oil to the tank without influencing the system.

3. Setting the DCV to the left position pushes the fluid to the blank end of piston 1 to extend this piston with a pressure P_1. The oil in the rod side of piston 1 is drained with a pressure P_2 to become the inlet to piston 2 with the same pressure. Finally, the fluid in the rod side of piston 2 is drained back to the tank at a new pressure P_3.

Applying the continuity equation and denoting Q for the flow rate, A for the area and v for the velocity with the subscripts 1 and 2 for cylinders 1 and 2 respectively.

$$(Q_{out})_1 = (Q_{in})_2$$

Since $Q = Av$,

$$(Av)_1 = (Av)_2$$

or,

$$((A_P)_1 - (A_R)_1)v_1 = (A_P)_2 v_2$$

Assuming that the fluid is incompressible, the synchronization means that $v_1 = v_2$, then

$$(A_P)_1 - (A_R)_1 = (A_P)_2$$

Note that the pressure P_1 is the total pressure needed to carry the two loads and the output pressure P_3 is zero gauge unless there is a PRV at the return line to the tank. Newtons second law can be applied for the summation of the force on cylinder 1:

$$P_1(A_P)_1 - P_2((A_P)_1 - (A_R)_1) = F_1 \tag{6.1}$$

Summing the forces on cylinder 2 results

$$P_2(A_P)_2 - P_3((A_P)_2 - (A_R)_2) = F_2 \tag{6.2}$$

But $P_3 = 0$ and $(A_P)_2 = (A_P)_1 - (A_R)_1$, then

$$P_2((A_P)_1 - (A_R)_1) = F_2 \tag{6.3}$$

Summing equations (6.1) and (6.3) gives

$$P_1(A_P)_1 = F_1 + F_2 \tag{6.4}$$

Example 6.2

It is required to lift two different masses $m_1 = 100kg$ and $m_2 = 200kg$ in synchronization at similar speeds with two different hydraulic pistons. If we have a piston with a bore of $120mm$ and a rod diameter of $70mm$. Find the pressure in the line out of the pump and the dimensions of the second actuator required here.

Solution

The solution is to connect the existing piston with another piston in series and mount the two masses on top of each one of them.
The piston side area of the first piston

$$(A_P)_1 = \pi \times (0.06)^2 = 0.0113m^3$$

The rod side area of the first piston

$$(A_R)_1 = \pi \times (0.035)^2 = 0.0038m^3$$

The pump is connected directly to the piston side of the first piston that raises the two loads together, thus the pressure on the pump line is calculated by the equation:

$$P_1 = \frac{F_1 + F_2}{(A_P)_1}$$

The total pressure is:

$$P_1 = \frac{m_1 \times g + m_2 \times g}{(A_P)_1} = \frac{100 \times 9.81 + 200 \times 9.81}{0.0113} = 2.6 \times 10^5 = 2.6bar$$

The piston side area of the second piston is:

$$(A_P)_2 = (A_P)_1 - (A_R)_1$$

$$(A_P)_2 = 0.0113 - 0.0038 = 0.0075m^3$$

The bore diameter of the second required piston is:

$$(D_P)_2 = \sqrt{\frac{4 \times 0.0075}{\pi}} = 0.0975m = 97.5mm$$

6.12 Fail-safe circuit

6.12.1 Application

The fail-safe circuit drawn in Figure 6.15 is used to protect the machine from accidentally falling on the operator in machines working in contact directly with people. It is used to avoid any damage or overload from hurting operators.

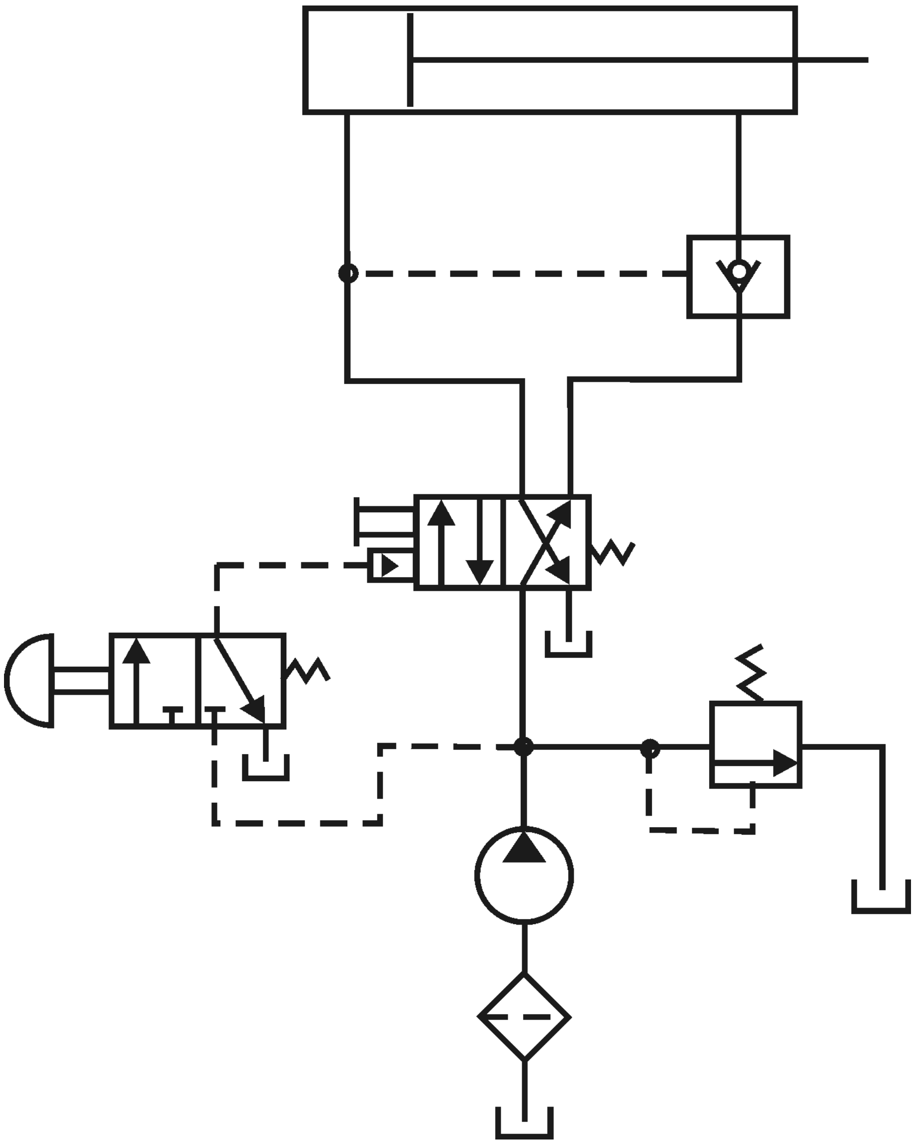


Figure 6.15: *Hydraulic fail-safe circuit*

6.12.2 Analysis

1. The pump, filter, tank and PRV are used as usual for all the previous circuits.

2. A two-position four-way, pilot-controlled DCV is used here.

3. When the DCV is set to the right position by means of the spring, the piston retracts to its end position.

4. The piston is prevented from extending by the effect of the check valve installed on the return line unless the push-button valve is pressed.

5. when the push-button valve is pushed, it operates the DCV to extend the piston after opening the check valve through the pilot line.

Figure 6.16 shows a two-handed safety circuit. For further protection, the operator needs to push the two push-buttons to pilot operate the DCV to extend the piston. Otherwise, the piston will stay retracted no matter if the operator pushes one button or nothing.

6.13 Meter-in cylinder speed control

6.13.1 Application

Meter-in speed control (Figure 6.17) is used to regulate the speed of the hydraulic pistons by controlling the flow rate of the oil entering to the cylinder when the speed is significant in specific applications.

6.13.2 Analysis

1. The pump, filter, tank and PRV are used as usual for all the previous circuits.

2. This circuit works exactly like the operation of a double-acting hydraulic cylinder with the difference of installing flow control valves (FCV) on the hydraulic lines by-passed by check valves.

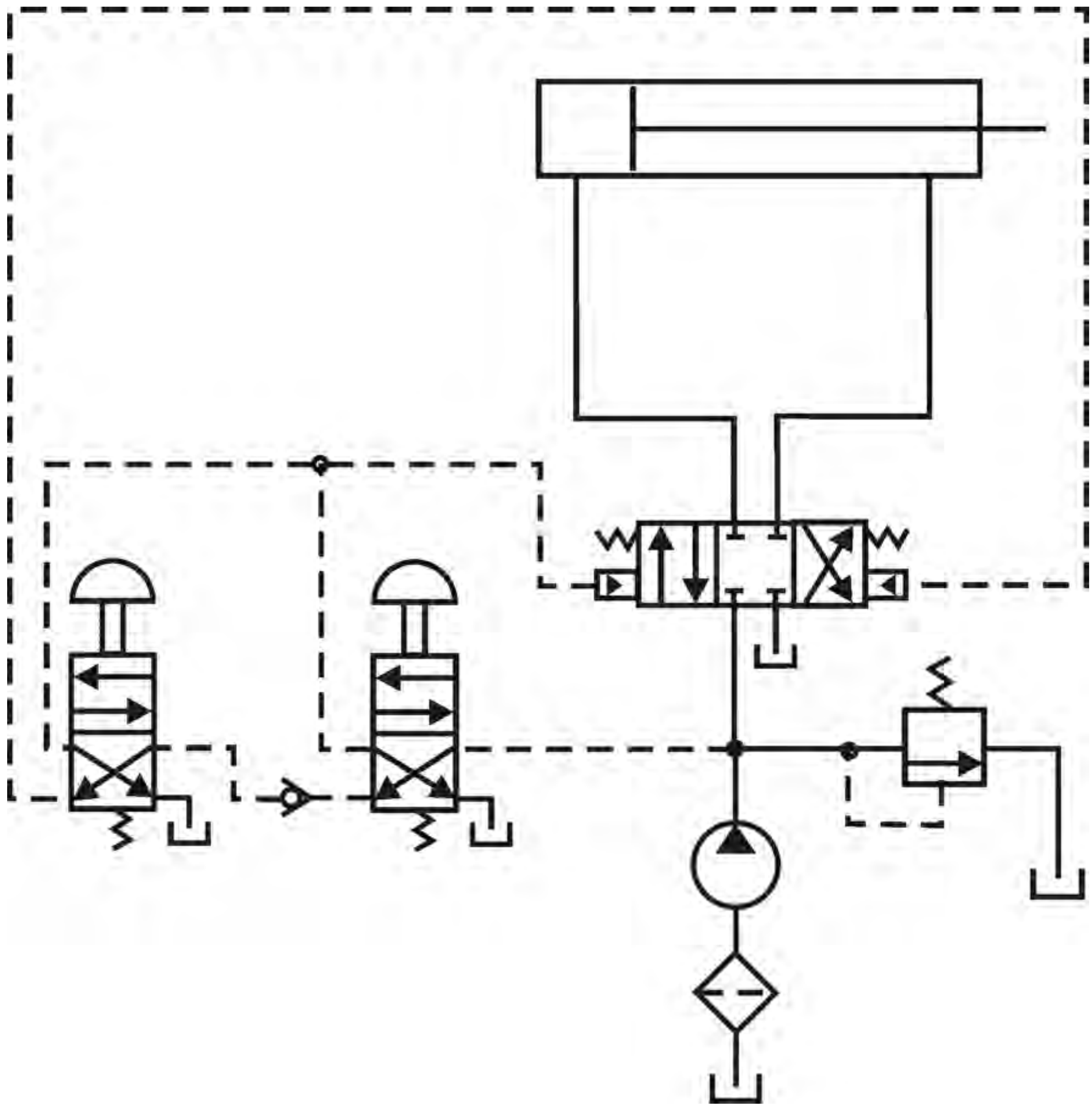


Figure 6.16: *Two-handed safety circuit*

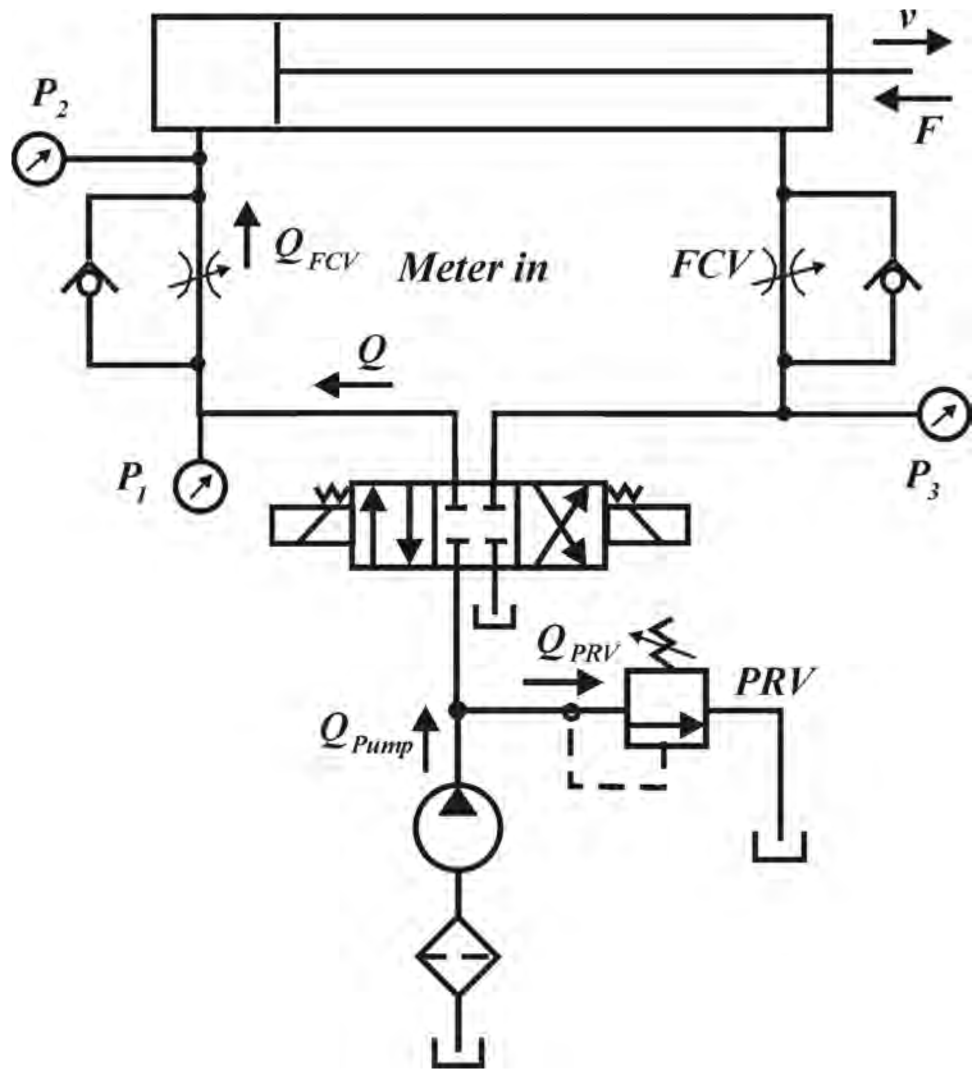


Figure 6.17: *Meter-in speed control circuit*

3. The inlet oil is forced to pass through the FCV to control the speed while at the outlet, it can pass through the check valve without restrictions.

4. Controlling the flow rate at the piston inlet controls the speed of the piston.

6.14 Meter-out cylinder speed control

6.14.1 Application

Meter-out speed control (Figure 6.18) is also used to regulate the speed of the hydraulic pistons by controlling the flow rate of the oil exiting the cylinder when the speed in significant in specific applications.

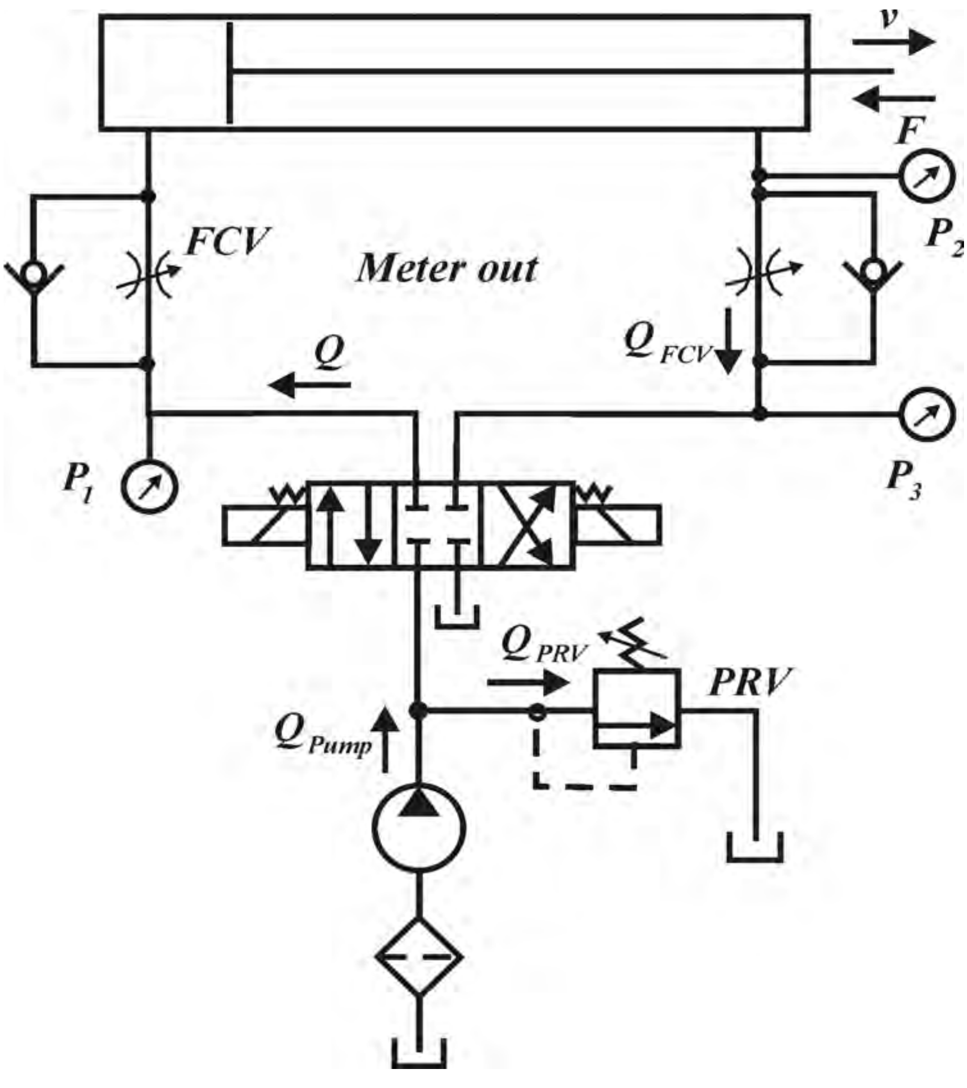


Figure 6.18: *Meter-out speed control circuit*

6.14.2 Analysis

1. The pump, filter, tank and PRV are used as usual for all the previous circuits.

2. This circuit also works exactly like the double-acting hydraulic cylinder with the difference of installing flow control valves (FCV) on the hydraulic lines by-passed by check valves too.

3. The outlet oil is forced to pass through the FCV to control the speed while at the inlet it can pass through the check valve without restrictions.

4. Controlling the flow rate at the piston outlet controls the speed of the piston.

5. The disadvantage of meter-out control is that the pressure builds up excessively in the rod side of the piston which increases the temperature reducing the efficiency.

6.15 Hydraulic motor speed control

6.15.1 Application

Figure 6.19 depicts an open-loop speed control circuit of a hydraulic motor accomplished by a pressure-compensated flow control valve FCV.

6.15.2 Analysis

1. The pump, filter, tank and PRV are used as explained for all the previous circuits.

2. When the three-position four-way DCV is spring-centred, the pump circulates the oil in an idle manner and the hydraulic motor is hydraulically blocked.

3. Pushing the DCV to the left position passes the oil to rotate the hydraulic motor in one direction.

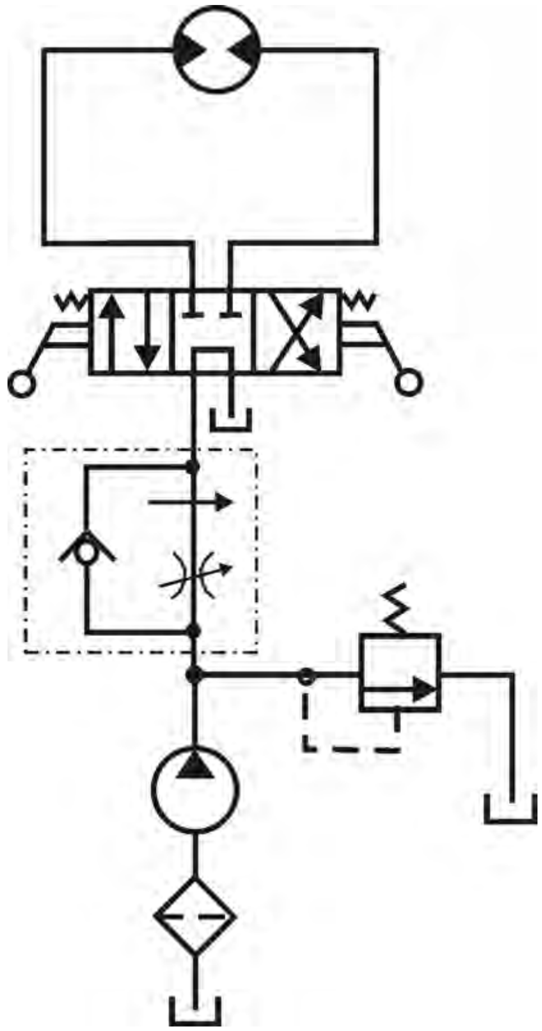


Figure 6.19: *Pressure compensated speed control for a hydraulic motor*

4. The speed of the motor is controlled by changing the throttle of the FCV and if the there is an excess pressure according to the load it can be drained by the PRV.

5. Pushing the DCV to the right position passes the oil to rotate the hydraulic motor in the other direction.

6. The by-pass check valve aims at compensating and equalizing the pressure on the two sides of the FCV.

6.16 Hydraulic motor braking system

6.16.1 Application

The circuit in Figure 6.20 shows an open-loop installation of a two-directional hydraulic motor with braking system. This is used when the motor operates a high inertia load applications.

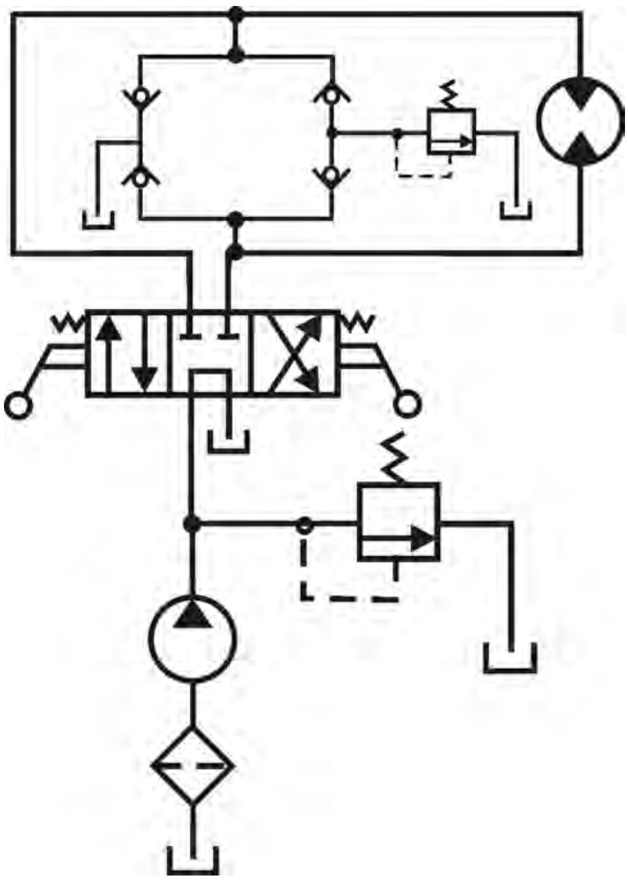


Figure 6.20: *Braking system for a hydraulic motor*

6.16.2 Analysis

1. The oil is pressurized after being drawn from the tank by the pump and the DCV changes the direction of rotation of the hydraulic motor by changing position manually.
2. When the DCV is spring centred to mid position the hydraulic motor is blocked. If the motor is driving a high inertia load, this can cause

flywheel effect on it which forces it to move slightly pushing fluid to one side. This fluid will be drained back to the tank via the PRV installed between the two check valves on the right.

3. Leaking fluid caused by high inertia reduces the amount of fluid in the circuit. To avoid pulling air, the other line with two check valves on the left, pumps oil from the tank into the circuit.

6.17 Hydrostatic transmission system

6.17.1 Application

The closed-loop circuit of a reversible variable displacement pump and two-directional hydraulic motor is shown in Figure 6.21. This is usually called "Hydrostatic Transmission System". One of the most common applications for this circuit is the concrete mixer.

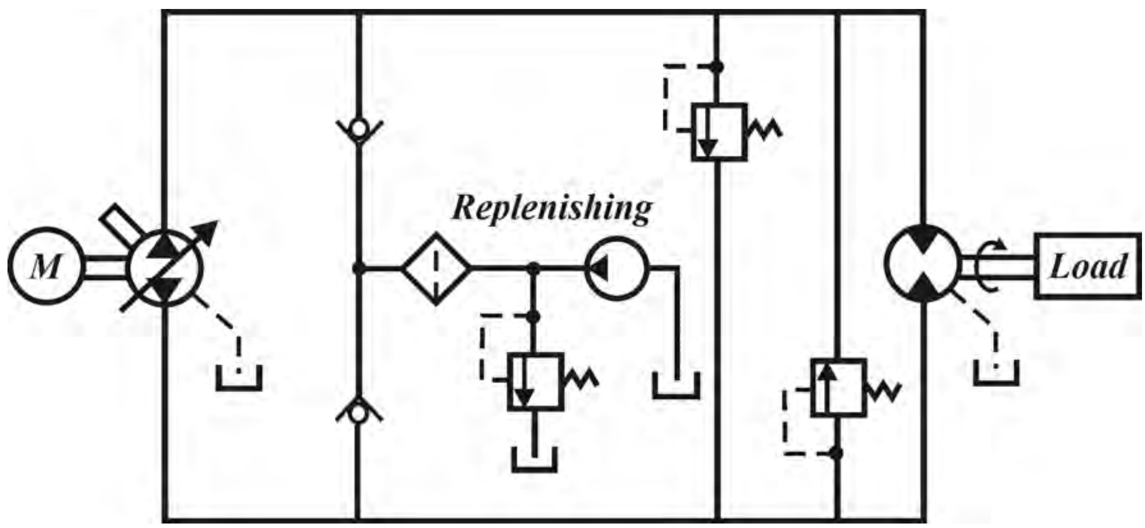


Figure 6.21: *Hydrostatic transmission system*

6.17.2 Analysis

1. The system is driven by a reversible variable displacement hydraulic pump driven by an electric motor.

2. Changing the direction of the pump changes the direction of rotation of the hydraulic motor.
3. The motor speed is controlled infinitely by varying the flow rate (displacement) of the pump.
4. When the motor is overloaded, the oil is circulated through one of the overload PRVs depending on the direction of rotation without damaging the system.
5. When the motor is influenced by the high inertia of the load, part of the fluid is leaked to the tank and is replenished from the tank to the system by one of the replenishing check valves according to the direction of rotation.

6.18 Air over oil system

6.18.1 Application

Figure 6.22 depicts an air over oil circuit where the air is used instead of the hydraulic pump. This kind of circuits is used in vehicle lifting jacks.

6.18.2 Analysis

1. The compressed air flows coming from an air storage (surge) tank fed by a compressor.
2. The air passes through a Filter Regulator Lubricator (FRL) to a three-position, three-way DCV.
3. The circuit is supplied with a PRV to exhaust the air out of the circuit to the atmosphere in the case of built-up pressure.
4. The compressed air enters into a vessel in which it exerts pressure on an oil separated by a diaphragm.
5. The oil, in turn, passes to extend the hydraulic piston to lift the load.
6. When the piston retracts under the load, the oil passes through a FCV to control its return speed to avoid impact.

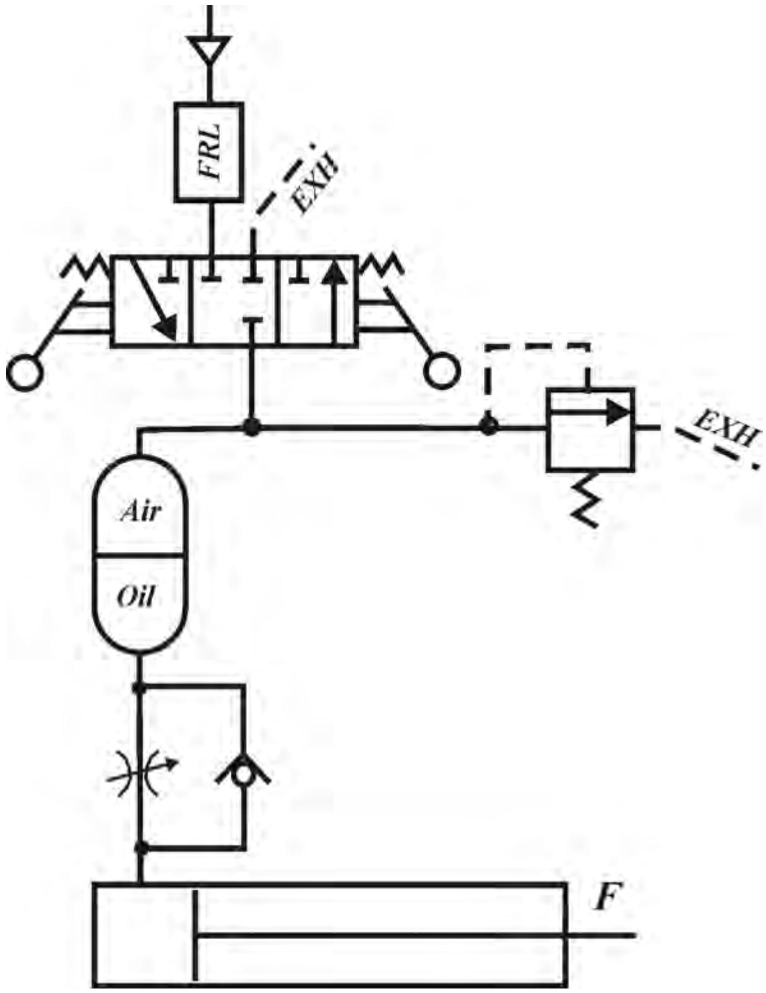


Figure 6.22: *Air over oil system*

6.19 Gas loaded accumulator

6.19.1 Application

Figure 6.23 shows how to use a gas loaded accumulator to give high impact to the motion of the piston. An example of this circuit is used in the pumping system of a concrete pump when there is a blockage in the concrete line the operator functions the accumulator to exert an impact force on the blocked concrete to move it forward and then the pumping process proceeds normally.

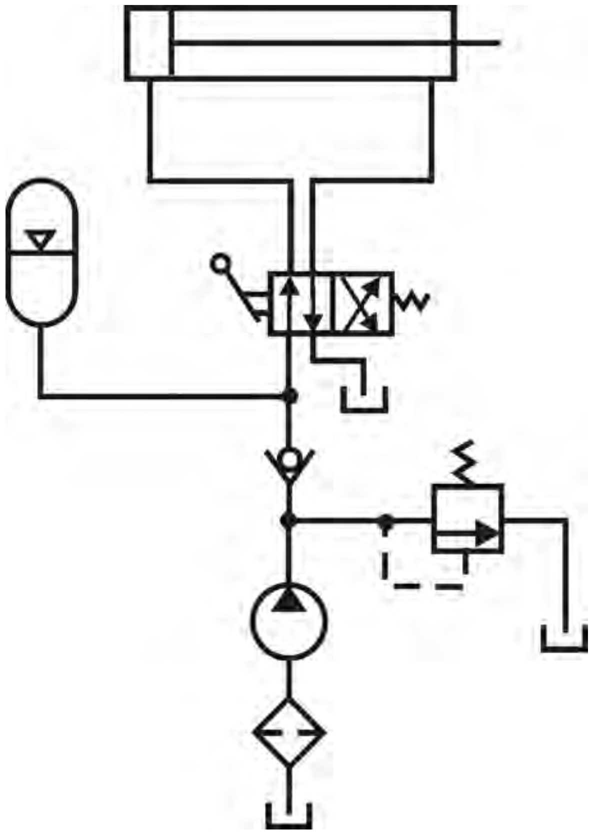


Figure 6.23: *Gas loaded accumulator*

6.19.2 Analysis

When the piston reaches a dead end, the accumulator is filled with oil compressed by the loaded gas. Changing the direction of the stroke begins the action with a high impact caused by the accumulator to overcome any blockage or high load effect. The rest of the circuit is a normal double acting cylinder circuit.

6.20 Control of pumps

Pressure and flow are two variables required to be controlled in pumps. The simplest way used to control the pressure and have a constant pressure value in the system is using a proportional pressure relief valve (PRV) that can be adjusted to the required pressure. The pressure relief valve will open only when the pressure in the system exceeds the required value where it allow the fluid to pass returning back to the tank. Figure 6.24 shows a control circuit used to control the pressure and the flow of a variable displacement pump.

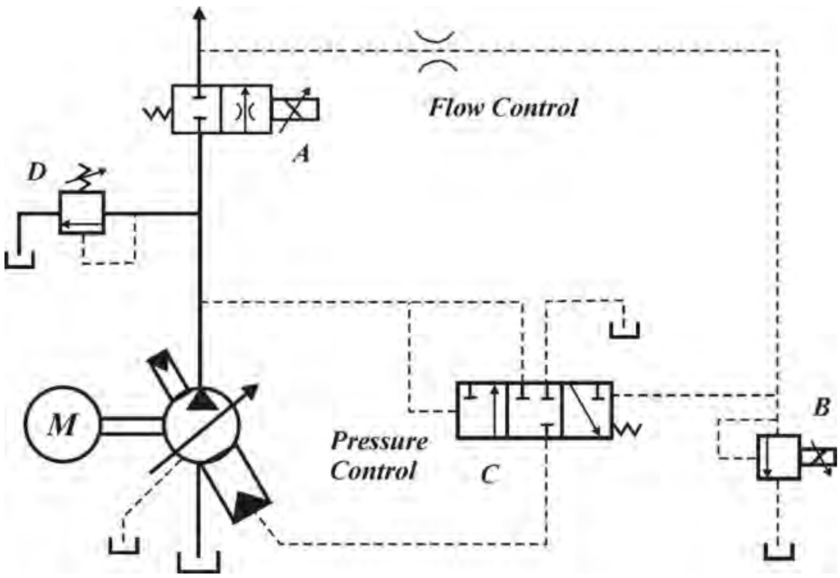


Figure 6.24: *Pump proportional control, pressure and flow*

The flow of the pump returns back after a specific pressure to the tank via the pressure relief valve D until applying current to the solenoid of the directional valve A. Moving the spool of valve A to the right allows to control the flow by the throttle in valve A. Leaving the pressure relief valve B closed increases the pressure in the pilot line and moves the spool of the three position three port directional control valve C to the right increasing the displacement of the pump. If the system pressure is larger than the pressure value set to valve B, this valve opens passing the fluid to the tank and the spool of valve C moves to the left opening the large pump piston to the tank.

6.21 Problems

1. State five applications for the single acting piston with drawing.

2. What is the main advantage of the double acting piston over the single

acting piston.

3. Design a double acting actuator for a fork lift to load and unload a one ton weight and calculate the pressure in the system in loading and unloading processes.

4. Consider a $2.5 \times 6 \times 1.5m^3$ truck dumper used for sand transportation (sand density $= 1400kg/m^3$). Design a hydraulic single acting actuator place at the free end of the dumper to tilt it around hinges mounted on the other end, what is the size of the required hydraulic pump.

5. Consider the general hydraulic system shown in 6.25. It is required to raise a load of $10kN$ at a speed of $100mm/s$. The system is horizontal and the $100mm$ bore actuator is connected to pump by a $10mm$ diameter pipe (neglect fittings). Suggest the hydraulic stiffness and damping coefficient:

   1- Draw the circuit in hydraulic symbolic form.
   2- Calculate the pressure and velocity of oil at the pump outlet (no friction losses).
   3- Calculate the electric current required for the single phase ($220V$) electric motor required to drive the pump (efficiency 90%).
   4- Build a numerical state space model for the actuator (input: pressure and output: speed).
   5- Find the frequency response function between the pressure input to the actuator and the load speed and draw it roughly by hand in frequency domain.

6. Consider the clamping system shown in Figure 6.26:
   1- What is the function of the 0.5 liter accumulator
   2- If the pressure of the pressure relief valve is set to 150 bar, what is the gas charge pressure of the accumulator
   3- Suggest an application for the circuit

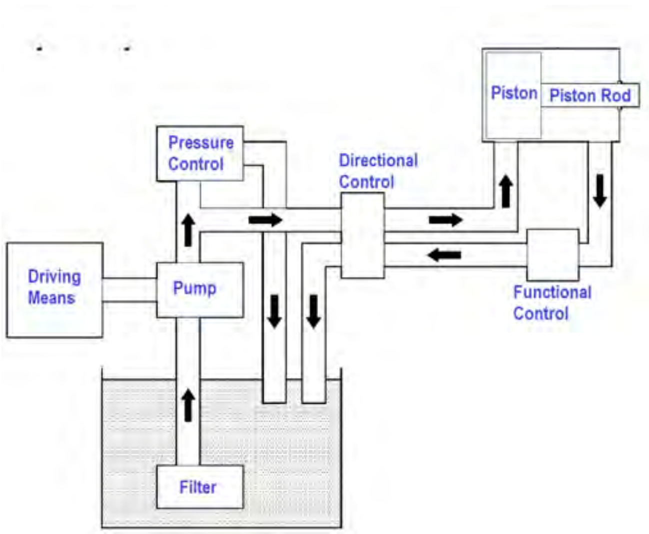


Figure 6.25: *General hydraulic system*

4- What is the function of the pilot line connected to the check valve

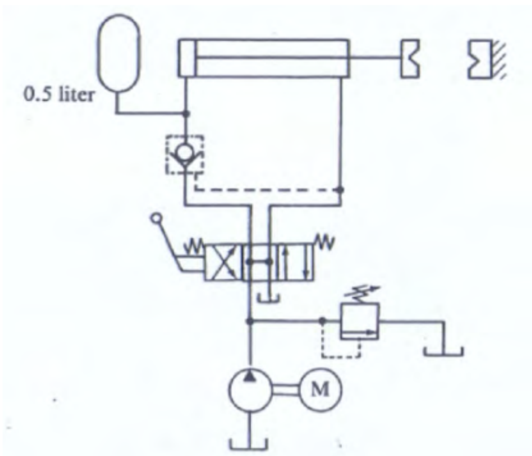


Figure 6.26: *Hydraulic clamping system with accumulator*

7. Consider the clamping system shown in Figure 6.27:
   1- What is the function of the pressure relief valve A
   2- Analyze the circuit
   3- Suggest an application for the circuit
   4- nominate all components of the circuit

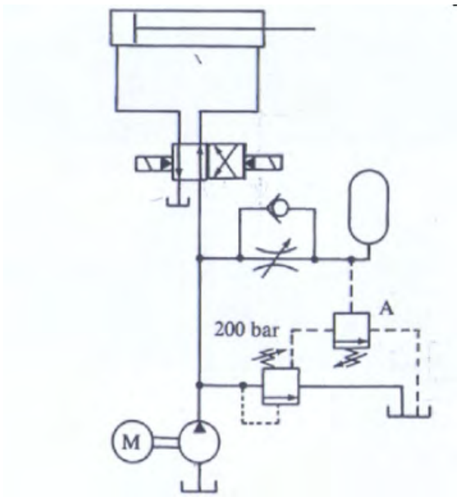


Figure 6.27: *Pumping system with accumulator*

8. Consider the clamping system shown in Figure 6.28:
   1- What is the function of the accumulators A and B
   2- nominate all components of the circuit
   3- Analyze the circuit
   4- Suggest an application for the circuit
   5- What is the purpose of the electrical supplies

9. Design a hydraulic circuit that can operate two hydraulic rotary actuators in sequence mode using hydraulic valves for the control. Find two applications for this circuit.

10. Draw the following hydraulic components in schematic and in symbolic forms:

    a- Swash plate piston pump
    b- Variable cushion double end double acting linear actuator
    c- 4/2 spring centered pilot operated spool type DCV

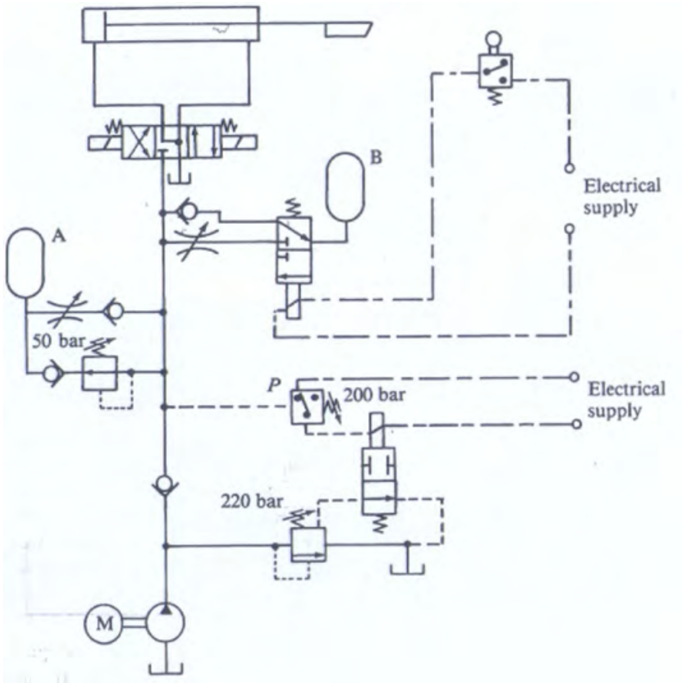


Figure 6.28: *Hydraulic system with two accumulators*

11. Design a hydraulic circuit to function and control a metal sheet bending machine where two workers work on this machine at the same time. (Take into account workers safety and machines safety).

12. Draw the following circuits and explain their processes:
    a- A reversible hydrostatic transmission system
    b- Meter-in speed control circuit for hydraulic cylinder

13. Draw a two-stage controlled (solenoid + Pilot), 3/4 , spring centered hydraulic directional control valve; a) in schematic form, b) in symbolic form.

14. A mass of 2000 kg is to be accelerated up to a velocity of 1 m/s from rest over a distance of 50 mm. The coefficient of friction between the load and the guides is 0.15. Select the bore of the cylinder required to accelerate this load if the maximum allowable pressure at the full bore end is 100 bar. Take seal friction to be equivalent to a pressure drop of 5 bar.

15. Consider the clamping system shown in Figure 6.29:
    1. Define all components on the same figure
    2. What is the application of the circuit?
    3. Analyse the following circuit in 3 steps (extension, retraction and mid position of the main directional control valve).

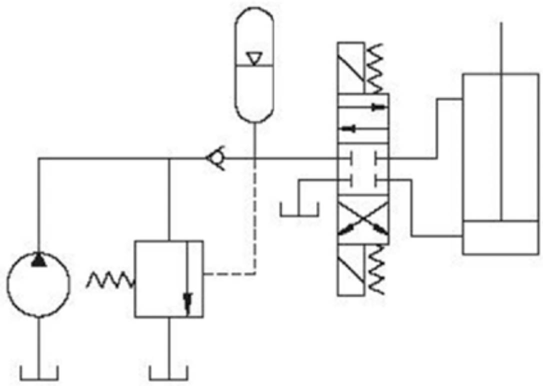


Figure 6.29: *Hydraulic system with one accumulators*

16. Draw a symbolic circuit that represents the following hydraulic system shown in Figure 6.30:

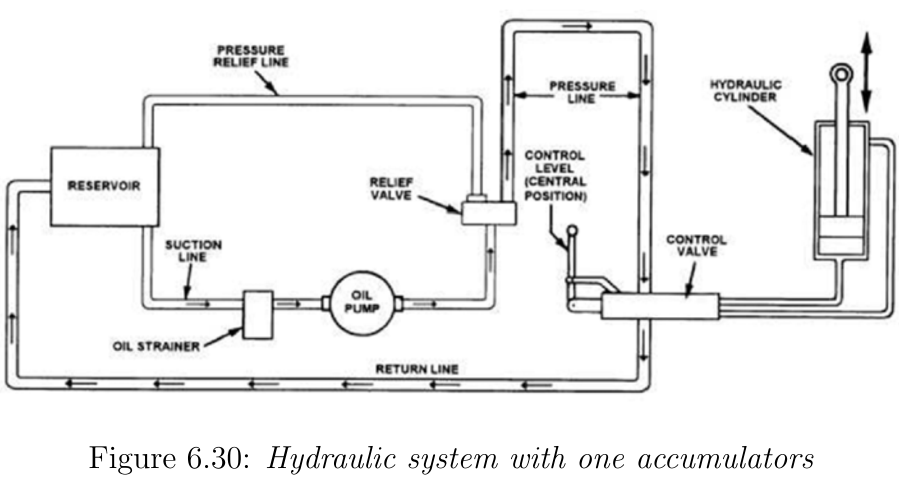


Figure 6.30: *Hydraulic system with one accumulators*

17. Draw a circuit with two hydraulic pistons in series and show analytically how these two pistons can move in complete synchronization even if the loads on the pistons are different.

18. For the meter-in and meter-out control techniques in hydraulic circuits:
a- Write two main differences between the two techniques.
b- What are these techniques used for and suggest an application?
c- Use symbolic drawing with hydraulic symbols to draw the two circuits.

19. Design and draw a hydraulic circuit to move two different loads in a synchronized manner (same speed) for a distance of $50cm$. If the loads are $200kg$ and $100kg$. Knowing that the system pressure is equal to $3bar$. Find the dimensions of each actuator knowing that the rod diameter must be equal to 40% of the cylinder diameter.

20. For a hydrostatic transmission system, the pump coefficient is $0.1m^2/s$, the motor displacement is $1m^3/radian$, the leakage coefficient is $2 \times 10^{-4}\ m^3/sec/Pa$, the load inertia is $100N.m.s^2$, the bulk modulus of the oil is $500N/m^2$ and the total volume of the system is $0.05m^3$

Find the following:
a- Find the transfer function between the pump displacement as an input and the motor speed as an output.
b- The natural frequency of the system.
c- The damping ratio.
d- The hydraulic stiffness.
e- Draw Bode plot for the transfer function.

21. Consider the series hydraulic cylinders arrangement shown in Figure 6.31. Knowing that the rod area have to be 0.5 the cylinder area in both cylinders. The force $F_1 = 1000N$ and the force $F_2 = 2000N$ and the pressure out from the pump $P_1 = 300kPa$ required to move the

two cylinders in synchronization the distance of $100cm$ in one second.
1- Find the bore areas and strokes of the two cylinders.
2- What is the required flow rate from the pump.

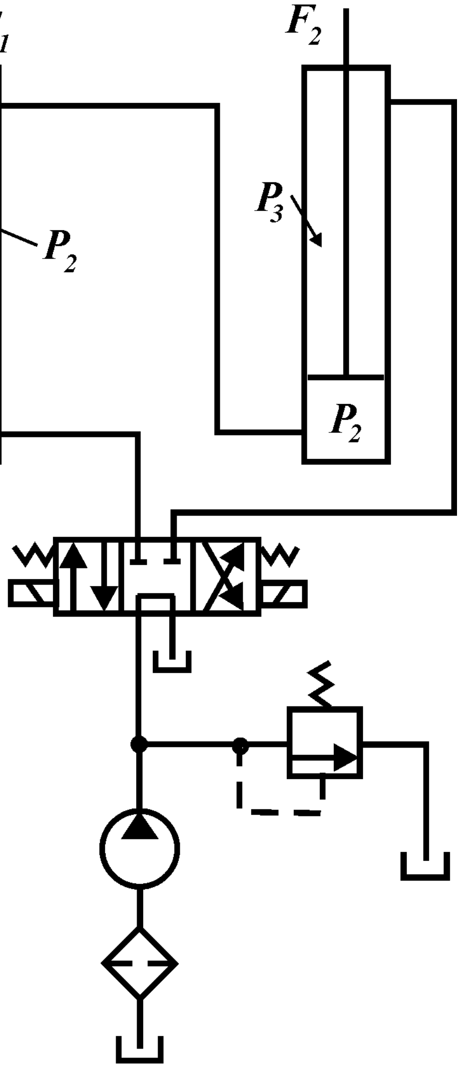


Figure 6.31: *Hydraulic system with two actuators in series*

22. The Scara robot in Figure 6.32 consists of:
    1- Hydraulic oil with bulk modulus=$1.5 \times 10^9 Pa$, kinematic viscosity= $200cSt$, Specific gravity= 0.9, in $13mm$ diameter rusted steel pipes with 5 standard 90^o elbows at the pressure line and other 5 at

the return line.
2- Hydraulic pump at $M1$ with volume $V_p = 0.01m^3$, leakage coefficient=$5\times 10^{-11}m5/N/s$, pump speed = $1400rpm$, $y = 0.1$, $K_p = 4x10^{-3}$, Pump power=$1kW$
3- Reversible hydraulic motor at $M1$ with volume $V_{M1} = 0.01m^3$, head=$1.5m$, displacement=$2l/radian$, max. motor speed = $200rpm$ and motor acceleration = $1.2rad/s^2$
4- Reversible hydraulic motor at $M2$ with volume $V_{M2} = 0.01m^3$, head=$1.5m$, displacement=$2l/radian$, max. motor speed = $200rpm$ and motor acceleration = $1.2rad/s^2$
5- Double acting hydraulic linear piston at $M3$ with piston diameter $D_{M3} = 5cm$, rod diameter $R_{M3} = 2cm$, stroke=$30cm$
6- Inner arm with length $L_1 = 0.6m$ and mass $m_1 = 3kg$
7- Outer arm with length $L_2 = 0.5m$ and mass $m_2 = 2kg$
8- Load at the end effector equals to $100kg$

Determine:

1- The volume of the hydraulic reservoir (Tank).
2- The pressure in the piston needed to lift the load when the pump is off, in the case when the piston rod is in up position and when the rod is down position.
3- The pressure in the piston when the pump is on.
4- Build a block diagram that contains the pump, the hydraulic motors and the hydraulic piston with the input as the displacement of the pump and the output as the displacement of the load.
5- Consider that the hydraulic motor $M1$ forms a hydrostatic transmission system with the pump and find the transfer function between the angular speed of motor $M1$ and pump displacement (y). find the natural frequency, hydraulic stiffness and damping ratio, draw the bode plot for this transfer function. (Take leakage and compressibility into account).
6- Build a three degrees of freedom dynamic model taking the two hydraulic motors at $M1$ and $M2$ and the piston at $M3$. (Neglect the

pump). The inputs are the toque $T1$ for motor $M1$ and torque $T2$ for motor $M2$ and pressure P for piston $M3$. The outputs are angular displacements $M1$ and $M2$ and linear displacement $M3$ and velocity for $M3$ and acceleration for $M3$. Use state space approach selecting the displacement and velocity at $M1$, displacement and velocity at $M2$ and displacement and velocity at $M3$ as state variables. Use the oil hydraulic stiffness and damping calculated in (5). Write the equations of motion and derive the four matrices of state space (A, B, C, D).

7- Draw the bode plots between inputs and outputs.

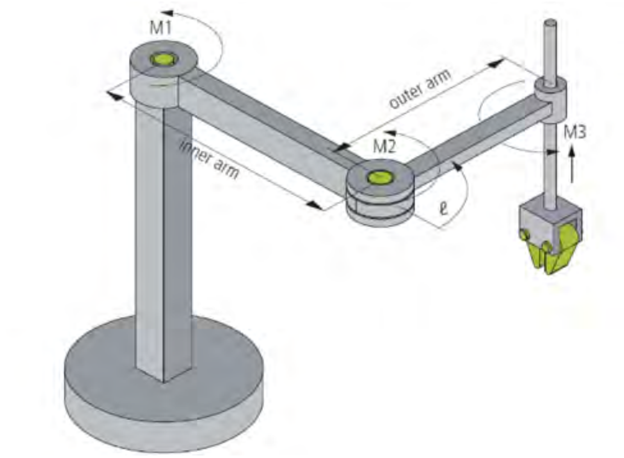


Figure 6.32: *Scara robot*

6.22 References

[1] A. Esposito, *Fluid Power with Applications*, Prentice Hall, 2003.

[2] A. Preumont, *Vibration control of active structures*, 1st. ed., Press Universitaire de Bruxelles, pp33-34, 1996-1997.

[3] M. Pinches and J. Ashby, *Power Hydraulics*, Prentice Hall, 1989.

Chapter 7

Pneumatic Circuits Design and Analysis

7.1 Introduction

Pneumatic solution is considered the most common relatively high power and low cost solution to supply systems with power and motion. The circuits discussed in this chapter concentrate on the systems controlled by mechanical and electro-mechanical valves and controls. A general pneumatic circuit like the one shown in Figure 7.1 consists mainly of:

- Air compressor (piston, screw or rotary type).
- Air reservoir (tank to be filled with compressed air).
- Filter-Regulator-Lubricator (FRL).
- Pneumatic actuator (linear piston or rotary motor).
- Pneumatic valve (pressure relief, flow control or directional control).

The same considerations taken in hydraulic circuits must be taken into account when designing a pneumatic system, these considerations are:

- Safety of operator and operation.
- Efficiency and performance of the whole system.
- Cost requirements.

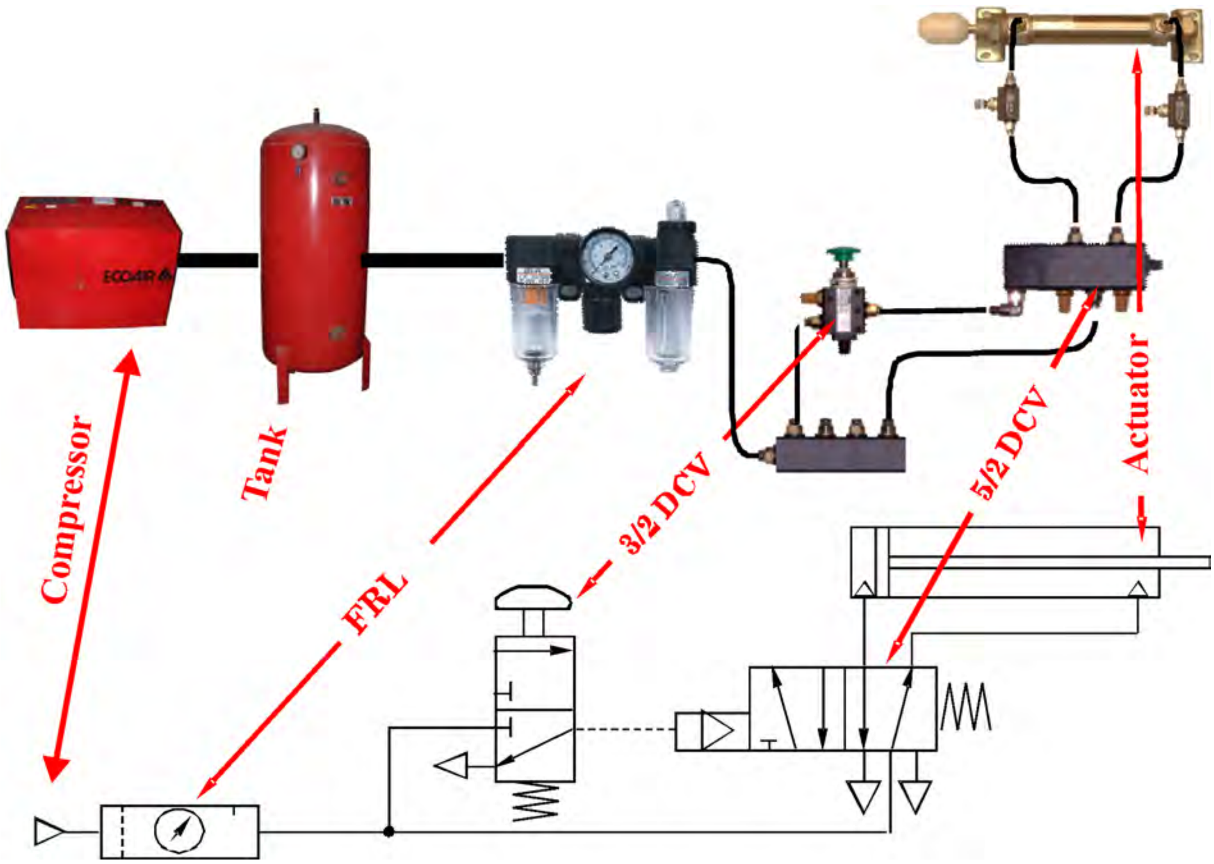


Figure 7.1: *Basic pneumatic system*

- Simplicity and easiness.

Different pneumatic circuits will be discussed in the following sections using the fluid power symbols listed before and that conform with the American National Standards Institute (ANSI).

7.2 Single-acting pneumatic cylinder

7.2.1 Application

The single-acting cylinder shown in Figure 7.2 is used when there is a need to have an action in a single sense of direction while the return stroke of the cylinder is actuated by means of a spring. An example of application is the pushers and rejectors used to reject manufactured plastic vessels that fail the leakage test in a production line. Another application is the can crusher shown in Figure 7.2.

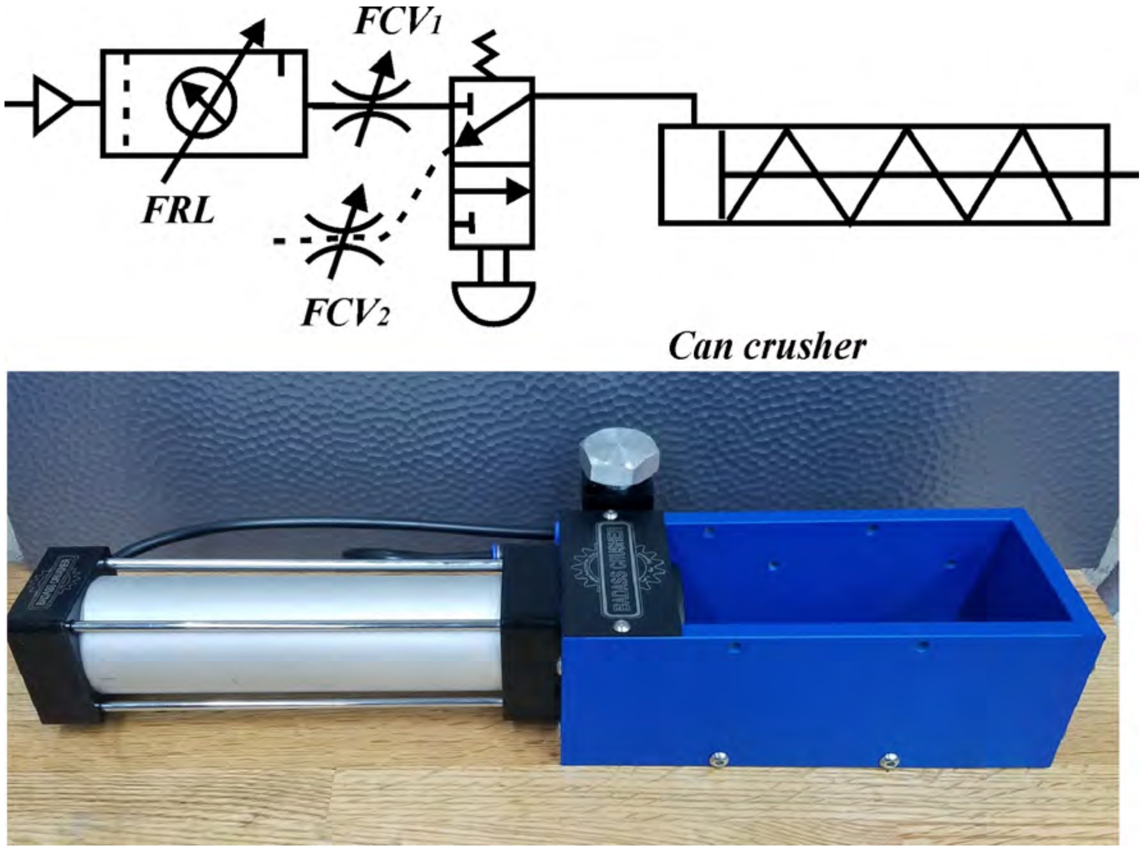


Figure 7.2: *Pneumatic single-acting cylinder and can crusher (courtesy of Badass crushers)*

7.2.2 Analysis

1. The inlet to the system begins from compressed air coming from the air storage tank after being pressurized by a compressor.

2. The compressed air passes through a Filter Regulator Lubricator (FRL) in which it is filtered and lubricated by adding some oil drops. The FRL serves also to regulate the air pressure before being fed to the application.

3. If the button of push-button directional valve is pressed the air flow is allowed to pass through the valve to extend the piston. The speed

of the extension stroke of the piston is controlled by the Flow Control Valve (FCV1).

4. Releasing the push-button valve returns the valve to the spring loaded position and opens the port of the piston to the exhaust port.

5. The piston is returned back by the spring force and the air is exhausted through the Flow Control Valve (FCV2) that controls the speed of the retraction stroke.

7.3 Double-acting pneumatic cylinder

7.3.1 Application

Figure 7.3 depicts a double-acting piston which is used in general for most of the applications that need actions in the two senses of direction; extension and retraction. This figure shows a pneumatic sliding gate (knife gate) as an application for the double-acting piston.

7.3.2 Analysis

1. The inlet to the system begins from compressed air coming from the air storage tank after being pressurized by a compressor and passed through an FRL.

2. A two-position five-port push-button directional control valve is used to change the direction of motion of the piston.

3. Pushing the button of the directional valve passes the compressed air to extend the piston and the excess air is vented through the exhaust port of the valve.

4. Releasing the button returns the control valve to the spring loaded position that allows the compressed air to retract the piston back exhausting the air from the exhaust port of the valve.

5. The speeds of the extension and retraction strokes are controlled by the variable flow control valves installed on the different ports.

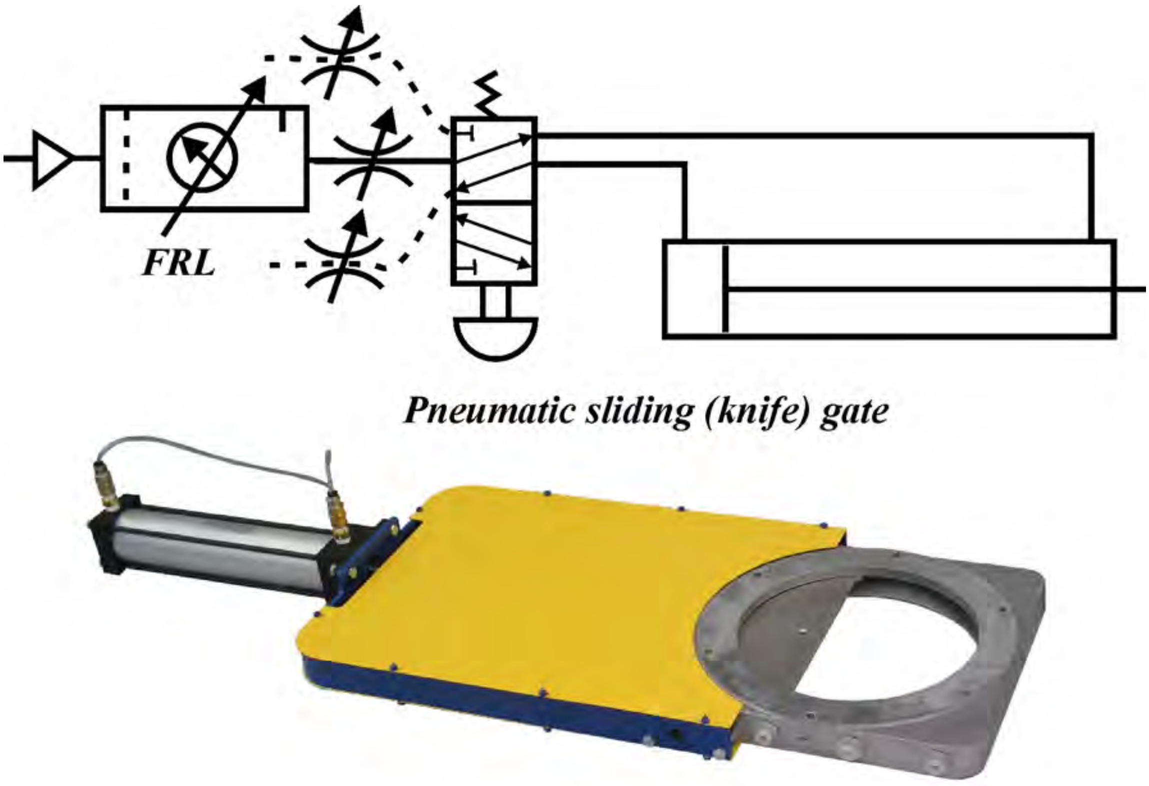


Figure 7.3: *Pneumatic double-acting cylinder and knife gate (courtesy of BISCO)*

7.4 Air pilot control of pneumatic cylinder

7.4.1 Application

The air pilot control can be used to change the direction of actuation of the pneumatic piston as shown in Figure 7.4. This circuit is used to limit the stroke of the piston at both sides like the one used in the reciprocating table of the grinding machine.

7.4.2 Analysis

1. The inlet to the system begins from compressed air coming from the air storage tank after being pressurized by a compressor and passed through an FRL.

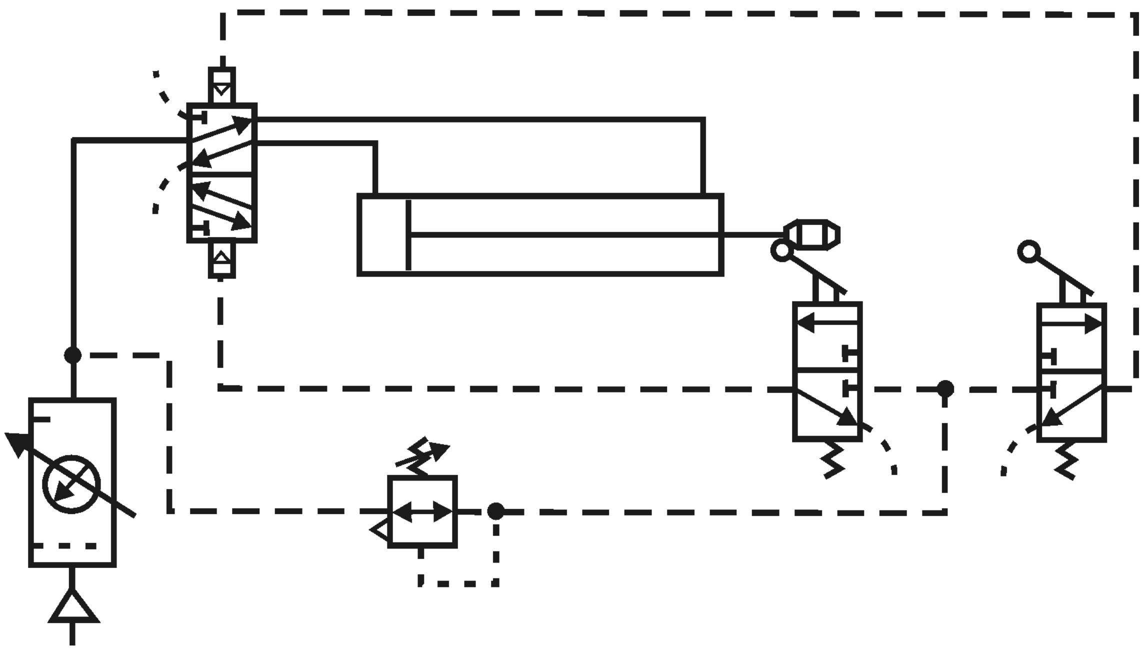


Figure 7.4: *Air pilot control of a pneumatic double-acting cylinder*

2. A two-position five-way pilot-actuated DCV is used to change the direction of actuation of the piston by passing the high pressure air to the piston everytime when the position is changed.

3. An air pressure relief valve is used to reduce the pressure (sometimes to 10% of the actuation pressure) to be used for the pilot operation that controls the position of the DCV.

4. When the piston reaches the end of the extension stroke, the tip of the piston pushes the handle of the push-button valve changing its position and passing the pilot air to the second position of the DCV. This changes the position of the DCV and passes the pressurized air to retract the piston.

5. When the piston retracts to the end of the stroke it pushes the handle of the second push-button valve repeating the same operation and extending the piston again, and so on and so forth.

6. This circuit leads to a reciprocating motion of the cylinder.

7.5 Cycle timing of pneumatic cylinder

7.5.1 Application

The circuit shown in Figure 7.5 is used for the timing of the cylinder cycle where a limited stroke is needed after which the piston is retracted again.

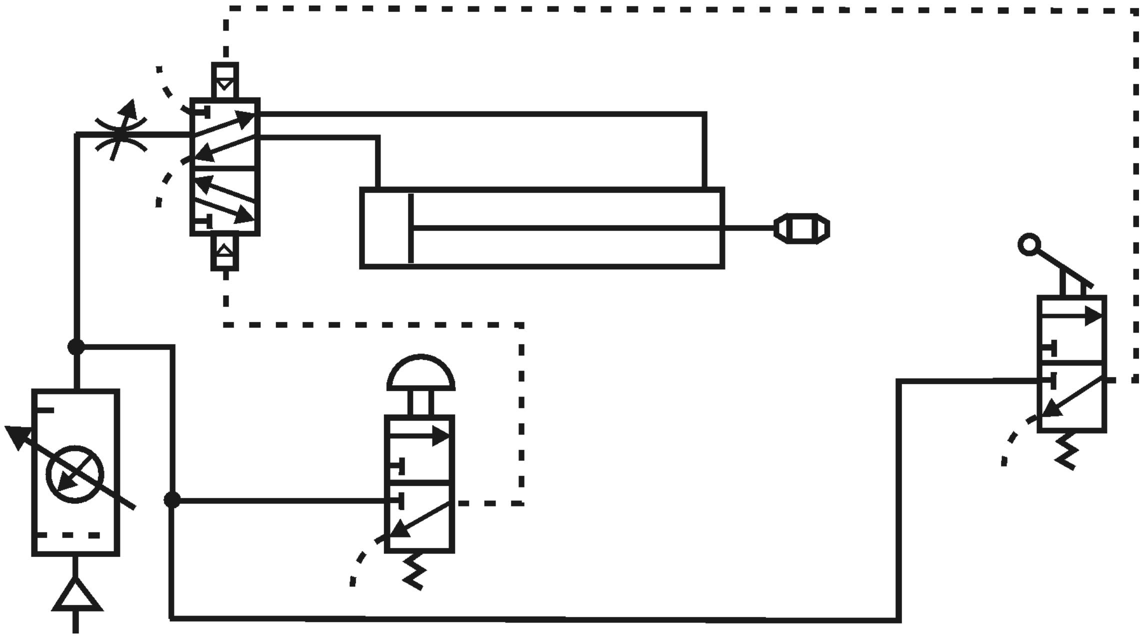


Figure 7.5: *Cycle timing of pneumatic cylinder*

7.5.2 Analysis

1. The inlet to the system begins from compressed air coming from the air storage tank after being pressurized by a compressor and passed through an FRL.

2. When the push-button valve is pressed manually, the position of the two-position five-way valve is set to extend the stroke of the piston.

3. The tip of the piston presses the handle of the limit valve to return the piston back in a retraction stroke.

4. To make another cycle of extension and retraction, the push-button valve must be pressed manually again.

5. The FCV is used to control the extension and retraction speed of the piston.

7.6 Two-speed pneumatic cylinder

7.6.1 Application

Figure 7.6 shows a circuit serves to introduce two different speeds for a piston during the same stroke. This circuit is used in cutting and forming press when there is a need to accelerate the piston before reaching the load and decelerating it during the forming operation to increase the force.

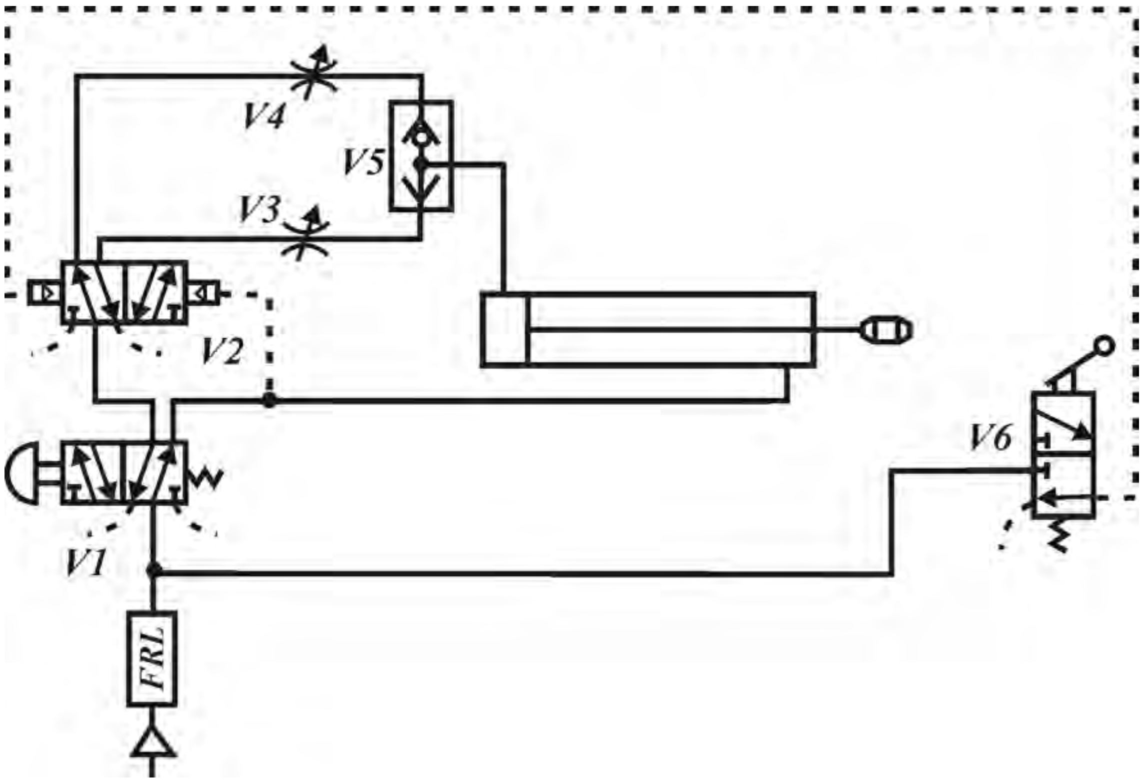


Figure 7.6: *Two-speed pneumatic cylinder*

7.6.2 Analysis

1. The system is fed by a compressed air via an FRL.

2. Pressing the button of the push-button valve V1 passes the compressed air through the DCV V2 and the FCV V4 to the shuttle valve V5 to begin extending the piston in a high speed depending on the flow controlled by the FCV V4.

3. When the tip of the piston presses the handle of valve V6, the pilot air flows to change the position of the DCV V2. This operation passes the compressed air through the FCV V3 via the shuttle valve to continue extending the piston but with a different lower speed depending on the flow controlled by the FCV V3.

4. Releasing the push-button valve V1 retracts the piston to the original position.

7.7 Two-handed safety circuit for pneumatic cylinder

7.7.1 Application

Figure 7.7 shows a safety circuit used to hold the piston in the retraction position unless the two push-button valves are pressed. This circuit is necessary in the applications close to human beings to avoid injuries caused by unconscious operating of the cylinder using one single button.

7.7.2 Analysis

1. The system is fed by a compressed air via an FRL.

2. Pushing the two palm-button valves passes the compressed air to right side envelope of the three-position five-way pilot-actuated DCV causing the piston to extend.

3. Note that pushing one valve will exhaust the air out and will not actuate the piston.

4. Releasing any of the pushed valves will return the DCV to the left position causing the piston to retract.

5. safety here comes from the necessity to push both palm-button valves to extend the piston.

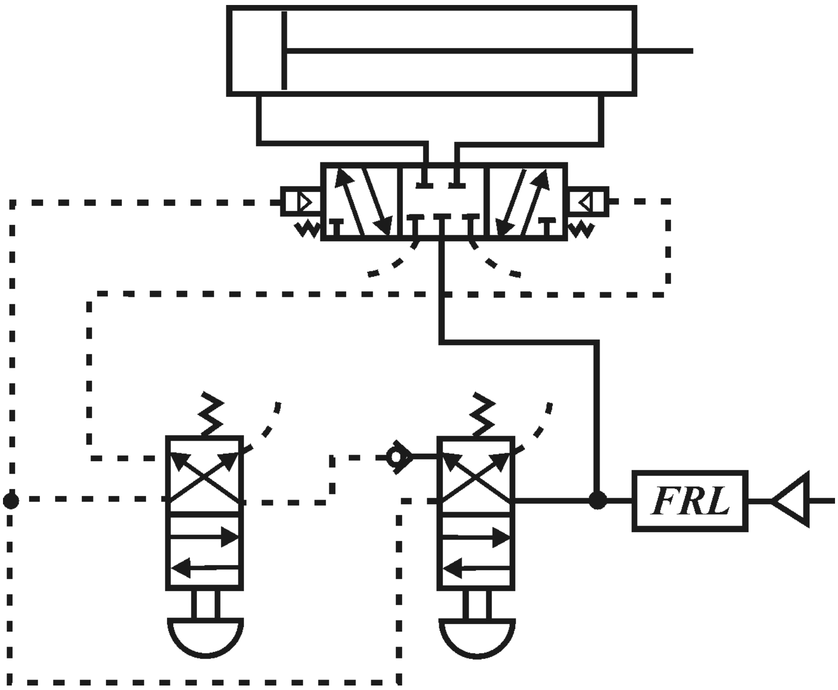


Figure 7.7: *Two-handed safety circuit for pneumatic cylinder*

7.8 Control of air motor

7.8.1 Application

The circuit shown in Figure 7.8 represents an operation procedure for an air motor where one push-button is used to actuate the motor and the other push-button is used to stop it.

7.8.2 Analysis

1. The system is fed by a compressed air via an FRL.
2. Pushing the left hand push-button will pass the compressed air to actuate the air motor by changing the position of the two-position five-way pilot-operated DCV.
3. The rotational speed of the air motor is controlled by variable flow control valve.
4. Pushing the right hand push-button will change the position of the DCV leading to vent the air out and to stop the air motor.

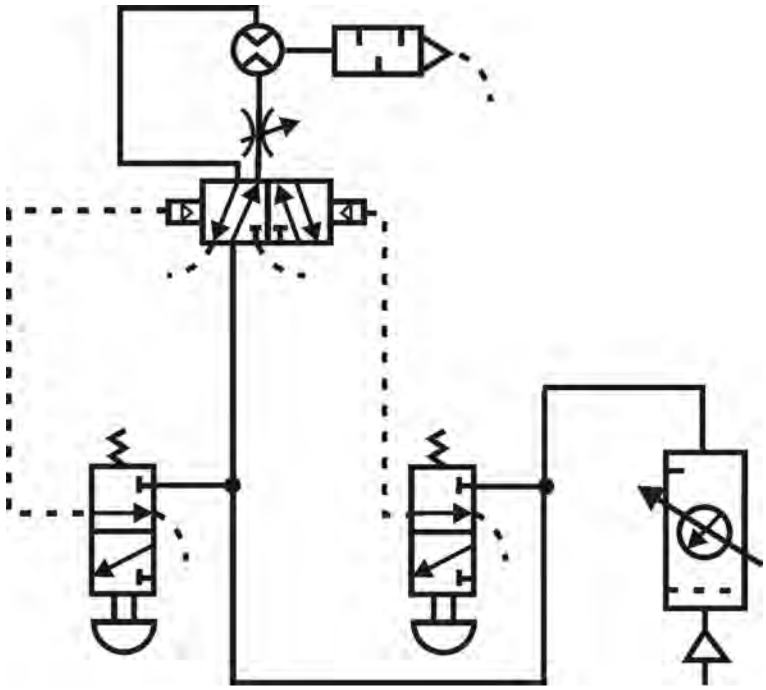


Figure 7.8: *Control of air motor*

7.9 Deceleration of a pneumatic cylinder

7.9.1 Application

Figure 7.9 shows a circuit used to decelerate the speed of the air piston when it is exposed to high loads to avoid shocks at the ends of the strokes.

7.9.2 Analysis

1. The system is fed by a compressed air via an FRL.

2. When there is a large weight acting on the piston in the retraction stroke, the pilot air passes through FCV V3 and changes the position of DCV V5 to force the exhausting air to be restricted by FCV V7 and decelerating the cylinder.

3. The same process is repeated in the extension stroke through valves V6, V2 and V4.

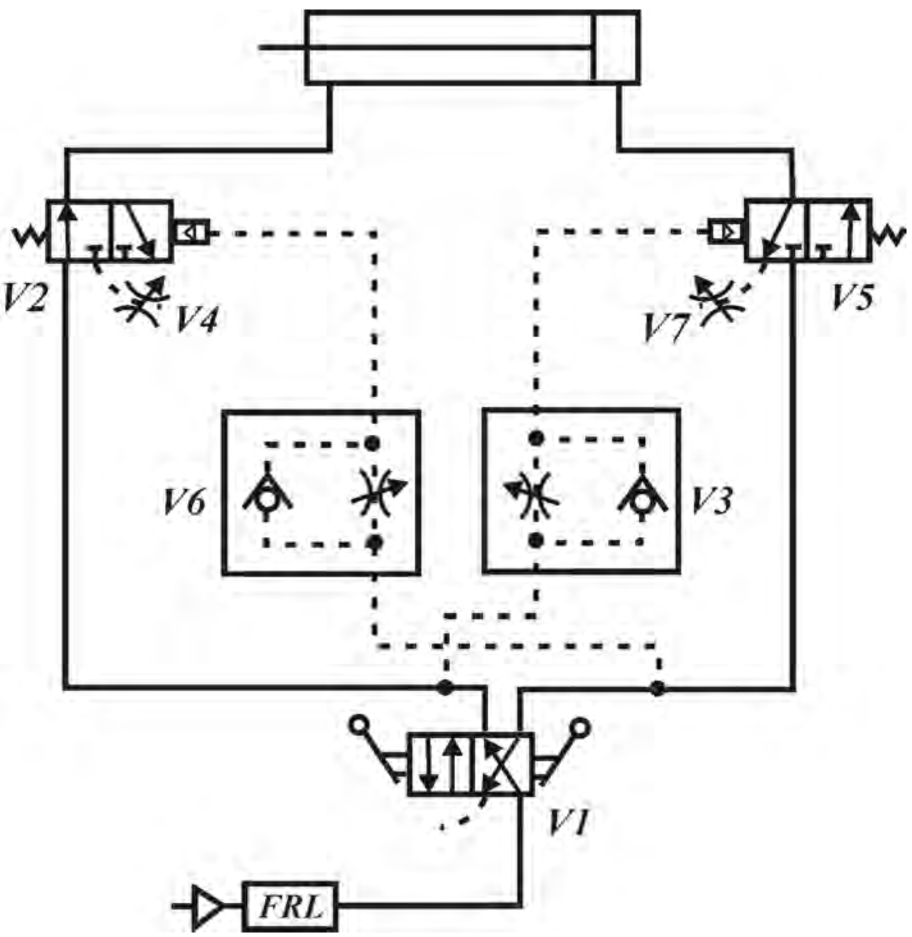


Figure 7.9: *Deceleration of a pneumatic cylinder*

7.10 Problems

1. What is the principle of leakage testing machine used to test the air leakage in plastic vessels and bottles.
2. Show how the failing plastic bottle is rejected in the leakage testing machine.
3. Draw a pneumatic circuit for pneumatic actuators and valves used to open and close the gates of the aggregate containers in a concrete batching plant.
4. Consider the pneumatic circuit in Figure 7.10, Analyse the circuit and nominate all components. What is the purpose of the shuttle valve in this circuit.

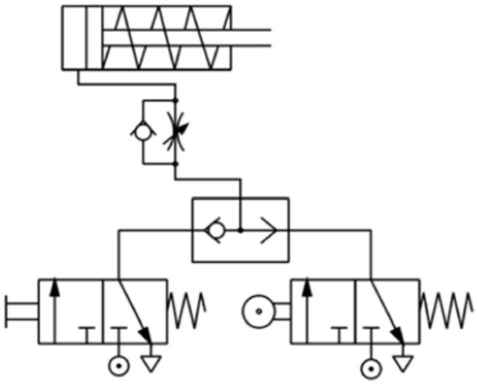


Figure 7.10: *pneumatic actuator with two inputs*

5. Consider the pneumatic circuit in Figure 7.11, Analyse the circuit and nominate all components. What are the flow control valves used for? and suggest an application for the circuit.

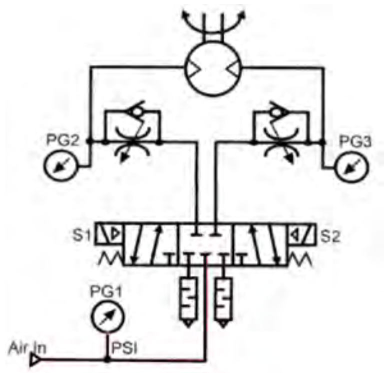


Figure 7.11: *pneumatic motor control circuit*

6. If the circuit in Figure 7.11 is controlled by a PLC, construct a ladder diagram program to control the circuit.

7.11 References

[1] A. Esposito, *Fluid Power with Applications*, Prentice Hall, 2003.

Chapter 8

Fluid Power Components

8.1 Introduction

Any fluid power system consists mainly of the following components:

- Electrical or thermal energy sources: Prime movers (Electrical or Internal Combustion motor)
- Hydraulic or pneumatic energy sources (pumps for hydraulics and compressors for pneumatics).
- Energy consumers: Actuators (linear or rotary).
- Control parts: Valves (depending on the required operation).
- Other parts: Tanks, filters, fittings, pipes and hoses.

The following section discusses the pumps used in hydraulic systems taking into account the procedure to select a pump and a few types of pumps used in different applications. Compressors of pneumatic systems are discussed in the following section. Linear and rotary types of actuators are analysed in the next section while different types of valves will be handled in the last section.

8.2 Pumps

Pump is the main source of energy in a hydraulic system. Pumps are used to convert mechanical energy to hydraulic energy. If there is no load on the pump, the main output of the pump is its flow rate of incompressible fluid. The mechanical energy given to the fluid by the pump can be transformed to hydraulic energy if a force is exerted by the load. In other words, the cumulation of the fluid flow leads to increasing the pressure of fluid in the system. Positive displacement pumps are used in hydraulic power for their capability of reaching high pressure that can reach more than 1000 bar.

8.2.1 Pump selection

Many parameters should be taken into account before selecting the type of pump:

- Maximum pressure in the system.
- Maximum required flow rate.
- Speed of the prime mover.
- Control technique.
- Fluid type and contamination level.
- Noise tolerance.
- Size and weight.
- Efficiency.
- Cost.
- Availability of the pump.

Taking into account the previous parameters, the following procedure can be followed to select a pump and hydraulic system successfully:

1. Select the type and size of the required actuator depending on the force exerted by the existing load.
2. Calculate the delivery flow rate needed from the pump depending on the required speed of the actuator moving the load (using continuity equation).

3. Calculate the maximum pressure in the system depending on the force exerted by the load and the area of the pipes and actuators (using Pascal theorem).

4. Determine the speed of the pump and prime mover. This is related to the pump's delivery (displacement) (by converting the hydraulic power to mechanical power over a specific efficiency).

5. Determine the type of the pump that can give the previously calculated parameters (pump calculations will be discussed later).

6. Select the size of the tank and piping needed for the system. This includes the calculation of the pressure and head losses due to friction and head difference.

7. Estimate the cost of the system.

8.2.2 Pump performance

The best pump selection can be done by taking into account obtaining the best performance of its operation. The best performance can be decided by looking at the performance curve shown in Figure 8.1.
The pump performance curve is drawn by operating the pump at variable flow rate and reading the accumulated pressure in the system at the pump's outlet. The system curve is obtained by calculating the constant static head in the system and summing it to the pressure loss coming from friction losses in the system, this gives the total pressure value that can be drawn as a function of flow rate variation. The best operating point for the pump is the point of intersection between the two curves (pump performance curve and system curve). The best operating point is not necessarily the best efficiency point because the best efficiency point of the pump is where the pump performance curve intersects the pump efficiency curve taken from the pump's data sheet. The best efficiency point (BEP) may correspond to higher pressure but less flow rate which may not meet the requirements of the system.

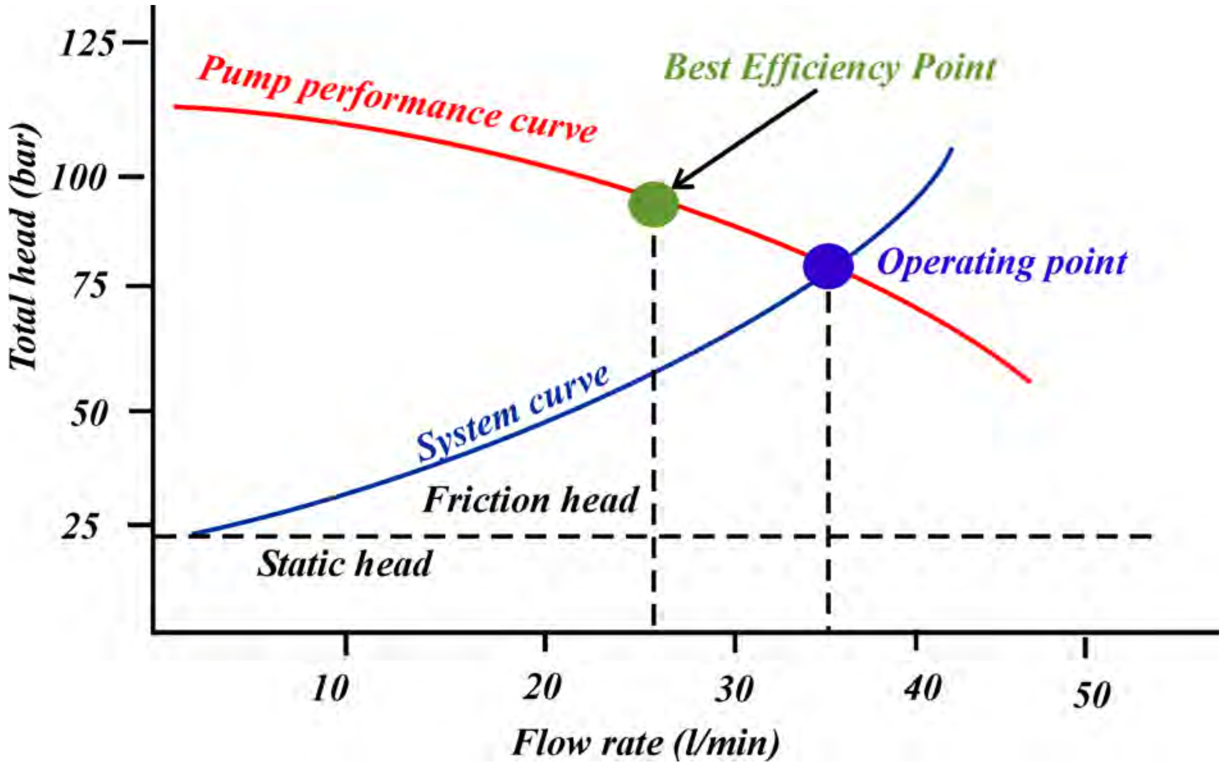


Figure 8.1: *Pump performance curve*

8.2.3 Pump control

Start/stop control
Start/stop control or known as On/Off control is done by switching the pump on and off depending on the system requirements. this can be done by connecting the pump a storage tank (accumulator) that serves to store the pressurized fluid, the pump shuts off at a calibrated pressure. The fluid flows from the accumulator tank at a steady flow rate and when the pressure reaches a specific pressure the pump is turned on again. This control technique minimizes the energy consumed by the pump and the run times of the pump can be synchronized at the low-rate periods.

Variable speed control
In some hydraulic systems, flow rate output from the pumps can be controlled by changing the rotational speed of the pump.When the pump speed is increased the number of cycles per minute is increased which increases the amount of fluid transmitted by the pump either it is vane, reciprocating

or any type of pump; this in turn will increase the flow rate at the pump's outlet. Pump speed can be varied simply by the following ways:

- Changing the speed of the prime mover motor specially when the prime mover is internal combustion engine. The speed of three phase electric motors can be varied by varying the electricity frequency.
- Using variable speed control through torque converter and controlled gearbox to regulate the speed of the pump's spindle which regulates the flow rate delivered from the pump.

Swash plate control
Swash plate is used usually in multi-cylinder reciprocating pumps. changing the angle of inclination of the swash plate changes the volume of cylinders at the intake and the outlet which changes the amount of flow rate delivered by the pump.

Example 8.1

A pump has a displacement volume of $100cm^3$. It delivers $0.0015m^3/s$ at 1000 rpm and 70 bars. If the electric motor torque is 120 N.m. What is the overall efficiency of the pump.
Solution
The volumetric displacement d_p:

$$d_p = 100\frac{cm^3}{rev} \times \frac{1m^3}{100cm^3} = 1 \times 10^{-4} m^3/rev$$

The total flow rate is:

$$Q_T = d_p \times \Omega_p = 1 \times 10^{-4}\frac{m^3}{rev} \times \frac{1000rev}{60s} = 0.00167m^3/s$$

The volumetric efficiency is the (delivered flow rate/Total flow rate):

$$\eta_v = \frac{0.0015}{0.00167} \times 100 = 89.8\%$$

The mechanical efficiency is:

$$\eta_m = \frac{PQ_T}{T\Omega_p} \times 100 = \frac{(70 \times 10^5 N/m^2)(0.00167 m^3/s)}{(120 N.m)(1000(\frac{2\pi}{60}) rad/s)} \times 100 = 93.0\%$$

The overall efficiency is

$$\eta_o = \eta_v \times \eta_m = \frac{89.8 \times 93}{100} = 83.5\%$$

The following sections will handle and discuss some types of pumps used in hydraulic systems

8.2.4 Gear pump

Gear pump is one of the simplest designs of pumps. It consists of two gears in mesh where one of them is connected to the prime mover and rotates the second one by teeth contact. The gear mesh causes a high noise level in the pump. The pumping occurs by enclosing a small amount of fluid between the teeth and the case of the pump as shown in Figure 8.2. The torque on the gear shaft transfers the fluid from the low pressure part at the inlet to the high pressure side at the outlet of the pump causing a continuous flow that can reach up to 700 l/min. The pressure at the outlet depends on the force exerted by the load and can reach up to 150 bar. Gear pumps are very sensitive to particle contamination of the fluid where it is necessary to filter the fluid before the pump to avoid any erosion of the teeth and casing caused by friction due to dirt and particles in the fluid. Any wear in the gear or housing leads to leakage of fluid and reduces the efficiency of the pump. Another type of gear pumps uses an internal gear in mesh with an external gear with smaller diameter. In this later case, a crescent is enclosed between the two gears to contain the fluid and transfer it from the inlet to the outlet. The delivery of this type of pumps can reach up to 750 l/min and the operation pressure ranges from 170 to 300 bar.

8.2.5 Lobe pump

Another type of gear pumps is the lobe pump shown in Figure 8.3. Lobe pump consists of two gears in mesh where each gear is made of three teeth

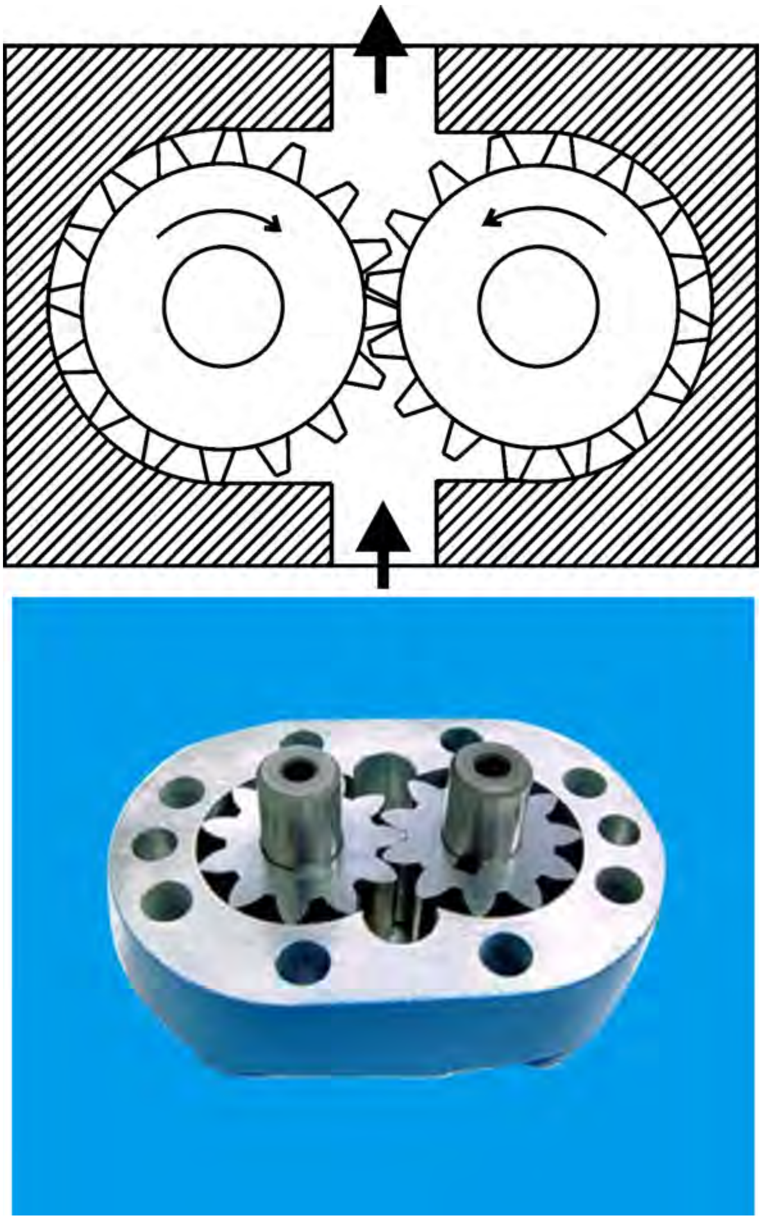


Figure 8.2: *Gear Pump (courtesy of Dynamic Pump Services)*

only. This type of pumps leads to a higher delivery due to the higher volume enclosed between the teeth and the casing but the reduction in the number of pulsations causes less continuity of flow compared to the normal gear pumps. This pump is less noisy than the previous one because each lobe is driven separately which reduces the contact between them reducing the noise level.

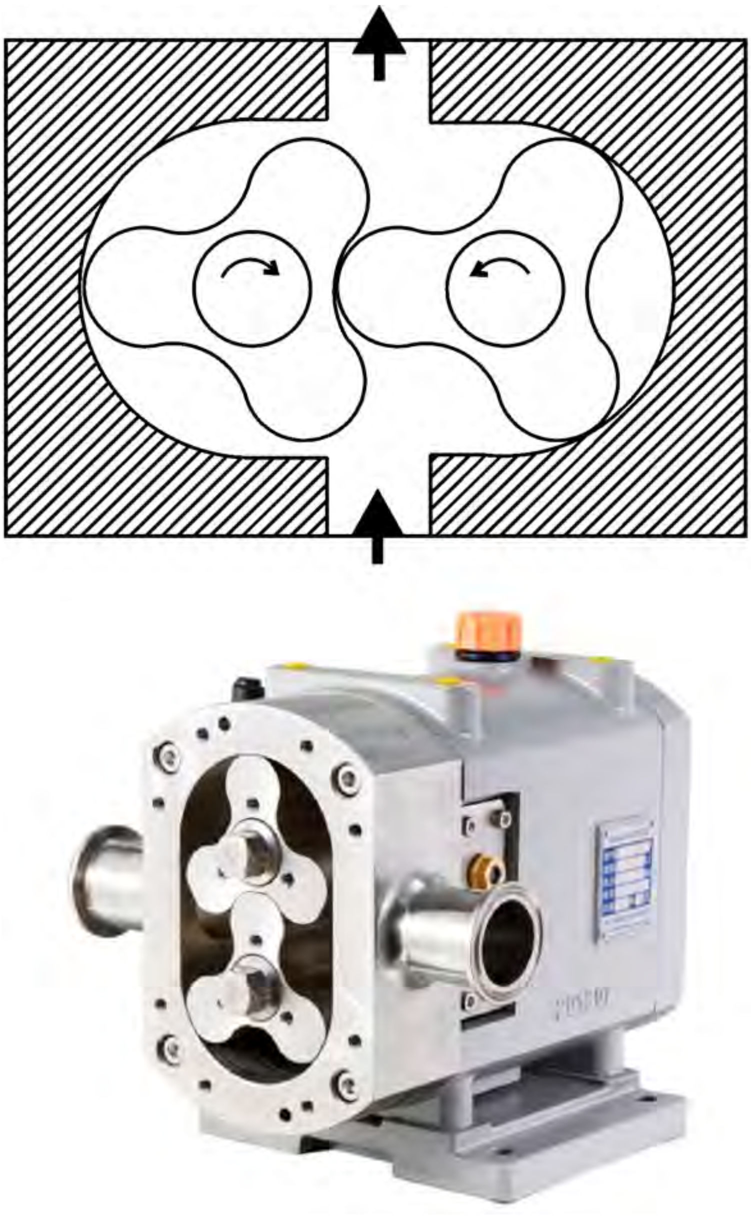


Figure 8.3: *lobe Pump (Courtesy of Zhejiang Ligao Pump Technology)*

8.2.6 Gerotor pump

The gerotor pump shown in Figure 8.4 is made out of an internal gear in mesh with an external one that has one tooth less than the outer gear. The displacement of this pump depends on the volume enclosed between the two gears due to the difference of one tooth. All types of gear pumps have the problem of high leakage because of the necessity to have a clearance between the gears.

8.2.7 Vane pump

Vane pump shown in Figure 8.5 consists of a rotor that contains radial slots and rotates inside a cam ring. Vanes are plates sliding radially inside the slots due to either the centrifugal force or pre-loaded springs under the vanes. Unlike the gear pumps, the problem of high leakage is solved in vane pumps because the vanes are pushed against the cam ring but this causes a high wear for the vanes. Eccentricity of the rotation in vane pump causes side force on the shaft. This problem is solved by using an elliptical cam ring with the rotor in the middle which is called balanced vane pump. The delivery of this type of pumps can reach up to 600 l/min and the operation pressure ranges from 100 to 170 bar.

8.2.8 Radial piston pump

The radial pump shown in Figure 8.6 consists of a ring housing and a concentric barrel containing the cylinders radially. A cam shaft rotates eccentrically inside the barrel to push the pistons radially outwards inside the cylinders. The pistons stay in contact with the cam by the effect of springs. The cam shape of the shaft's rotation causes the pistons to extend with different strokes depending on their position relative to the cam. This motion causes the cylinders to extract the fluid at the minor radius of the cam and deliver it at the major radius. The displacement of the pump depends on the cross sectional area of the pistons and the difference between the minor and major diameters of the cam. The delivery of this type of pumps can reach up to 1000 l/min and the operation pressure can reach up to 1500 bar.

Figure 8.4: *Gerotor Pump (Courtesy of Cascon Pumps)*

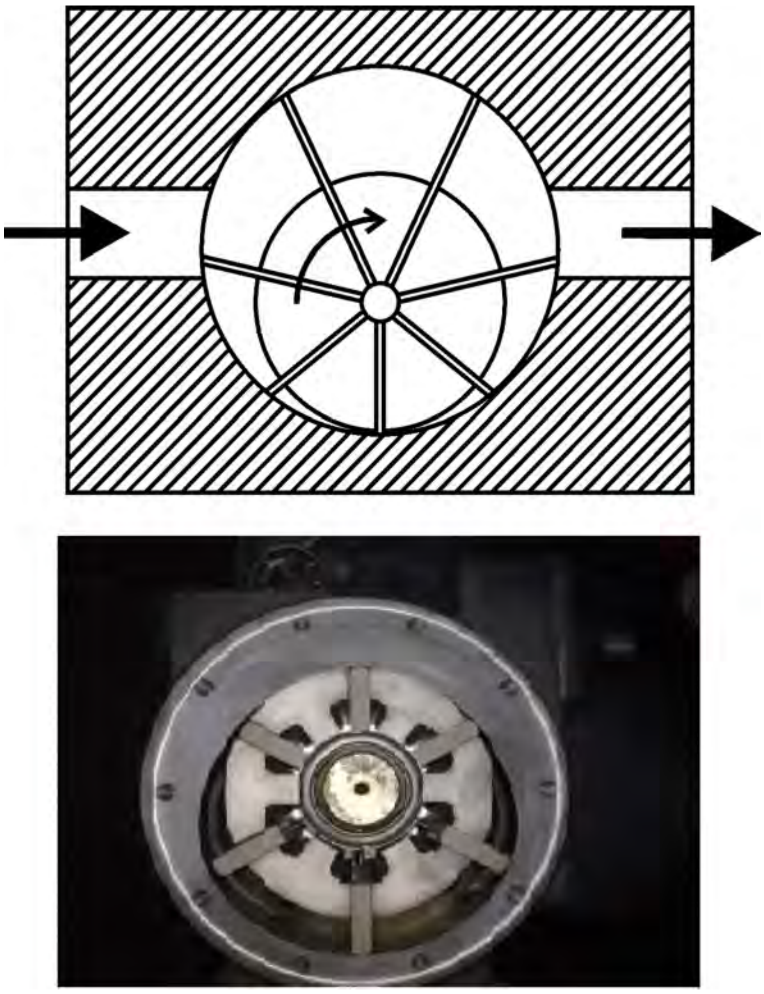


Figure 8.5: *Vane Pump (Courtesy of North Ridge Pumps)*

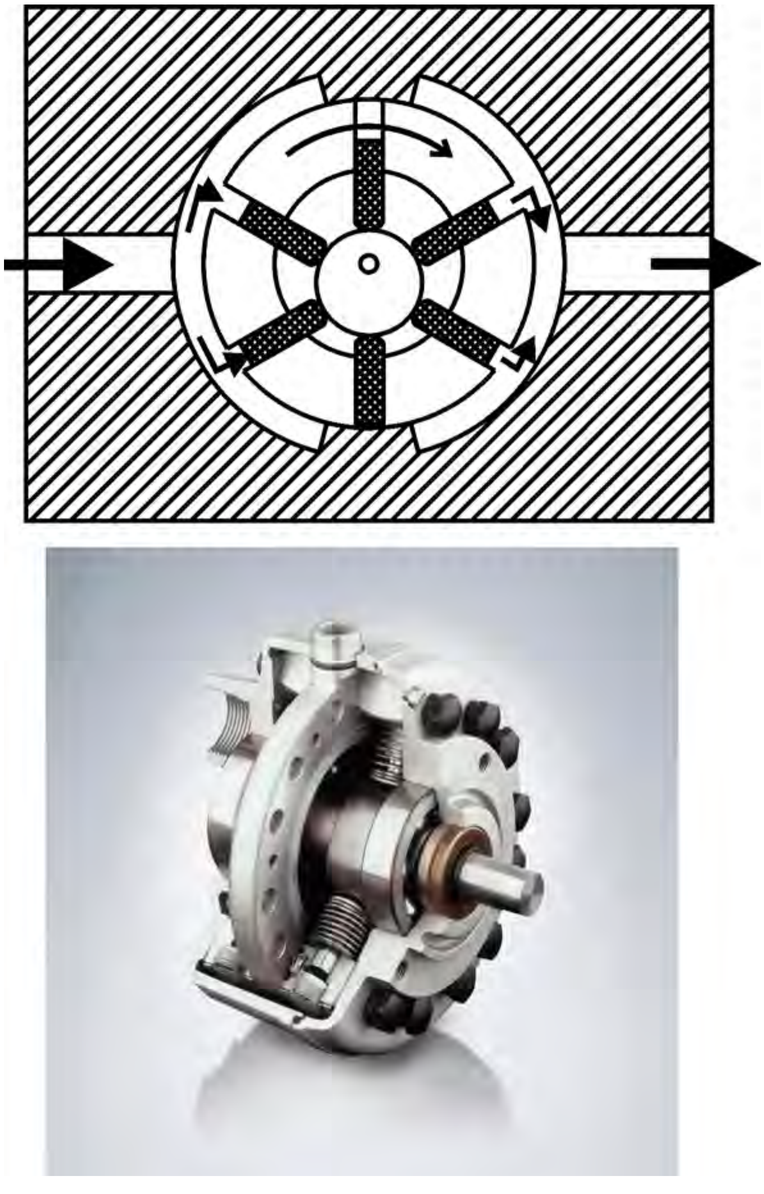
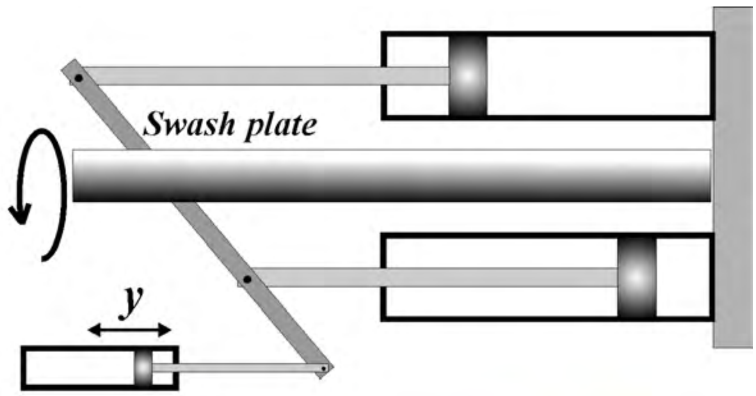


Figure 8.6: *Radial piston Pump (Courtesy of Rexroth company)*

8.2.9 Axial piston pump

Figure 8.7 shows a swash plate axial pump. The pistons are included axially in a rotating barrel, the rods of the pistons are connected to a swash plate inclined with a specific angle and connected (sliding contact) to the piston rods sliding inside the rotating barrel. The angle of inclination of the swash plate determines the displacement of the pump by changing the stroke of the different pistons. This type of pumps is very common in variable displacement pumps and very widely used in a wide spread range of applications. The delivery of this type of pumps can reach up to 3500 l/min and the operation pressure ranges from 200 to 350 bar.



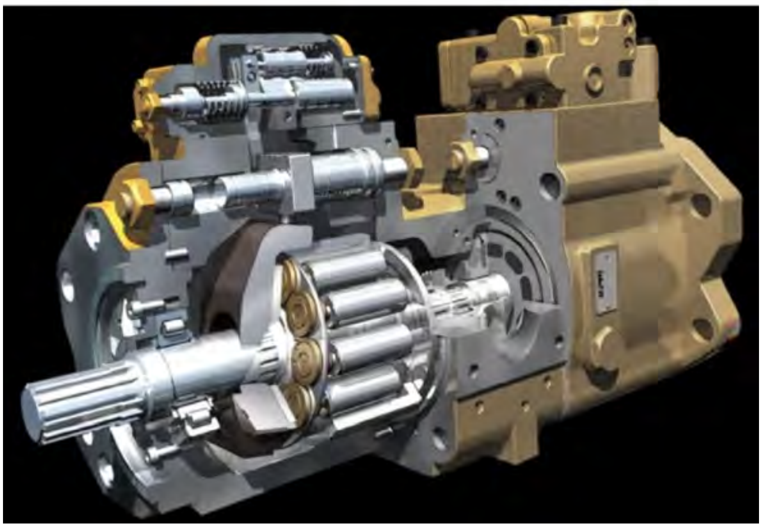


Figure 8.7: *Swash plate axial piston Pump (Courtesy of Cross company)*

Figure 8.8 shows a bent-axis axial pump. In this design the complete assembly of the barrel, pistons and back plate are connected to the rotating axis by a universal joint to overcome the angle difference between the rotation axes.

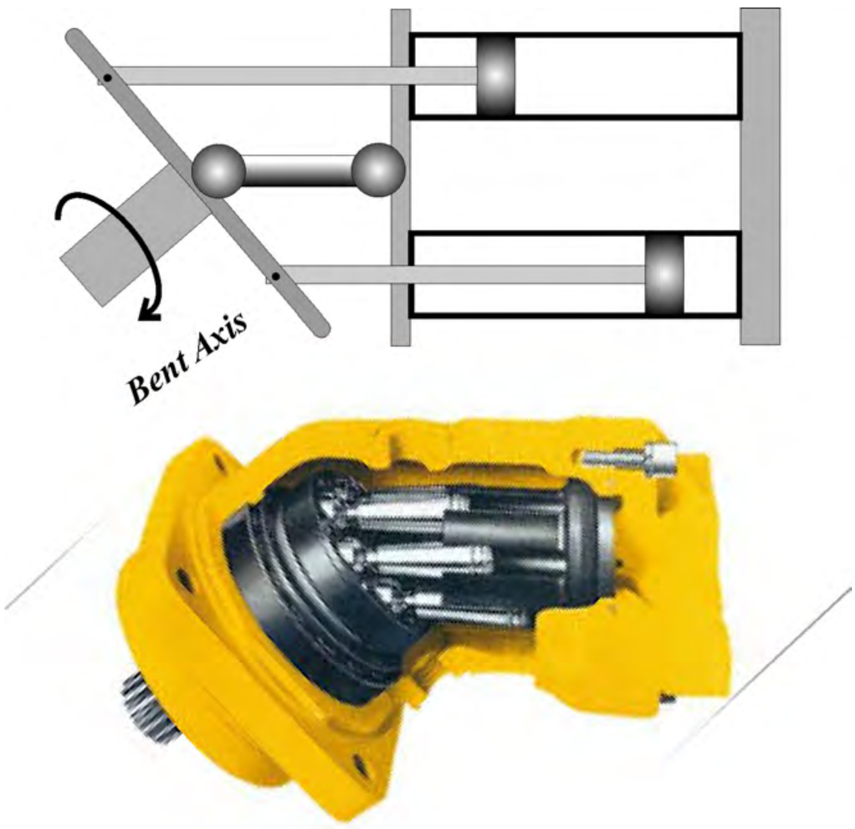


Figure 8.8: *Bent-axis Axial piston Pump (Courtesy of Insane Hydraulics)*

8.3 Compressors

The main component in pneumatic system is the compressor. Compressors are usually driven by electric motors to produce pressurized air and store it in a closed reservoir to be used in exerting force and pressure on loads. Unlike hydraulic systems, pneumatic systems are open to the atmosphere where compressed air is exhausted out after being used. Compressing air leads to increase its temperature which explores the need to use after-coolers to reduce its temperature. In the case of two-stage compressors, intercoolers are used between the stages. Filters and water traps are used to get rid of the water entrapped in the compressed air. This water comes from the condensation of vapour existing as a humidity in the atmospheric air. Positive displacement compressors are mostly used in pneumatic systems because they have the ability to produce higher pressure.

8.3.1 Reciprocating Piston compressors

The most commonly used compressor is the piston type shown in Figure 8.9. The basic principle of piston compressors is based on turning rotational motion into linear motion through a slider-crank mechanism where the rotational speed of the crank is transformed to a linear motion of the piston inside the cylinder. When the piston is pulled down, the inlet check valve opens and allows air to enter to the cylinder. Pushing the piston upwards in a return stroke, closes the inlet check valve and opens the outlet check valve forcing the air in the cylinder to flow out. A single-stage piston compressor can compress the air up to 15 bar.

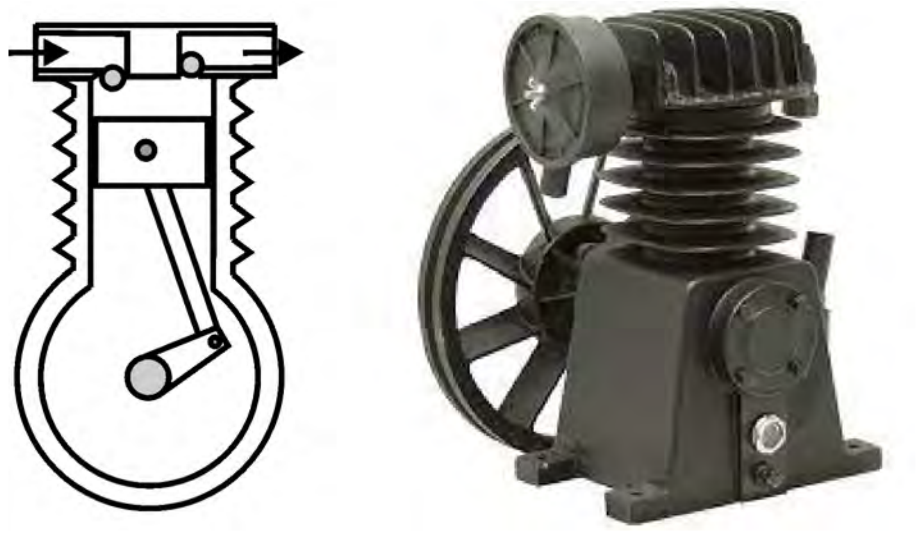


Figure 8.9: *Piston Compressor (Courtesy of Taizhou Bori Machinery Co. Ltd.)*

The operation of any reciprocating compressor can be represented by the PV diagram shown in Figure 8.10. The compression stage from point 1 to point 2 is done by opening the intake valve and shutting the exhaust valve while moving the piston inside the cylinder towards reducing the compressed volume which increases the pressure to the maximum value P_2 according to the gas laws. The delivery stage from point 2 to point 3 occurs by opening the exhaust valve and shutting the intake valve while the piston is still moving to shrink the chamber to the minimum volume V_1 allowing the compressed air to flow out at constant pressure P_2. The expansion stage from point 3 to point 4 is done by pulling the piston back to increase the volume which decreases the pressure to the minimum value P_1. The last intake stage from point 4 to point 1 is done by pulling the piston to the end stroke expanding the volume to the maximum value V_2 and returning the system to the initial point again.

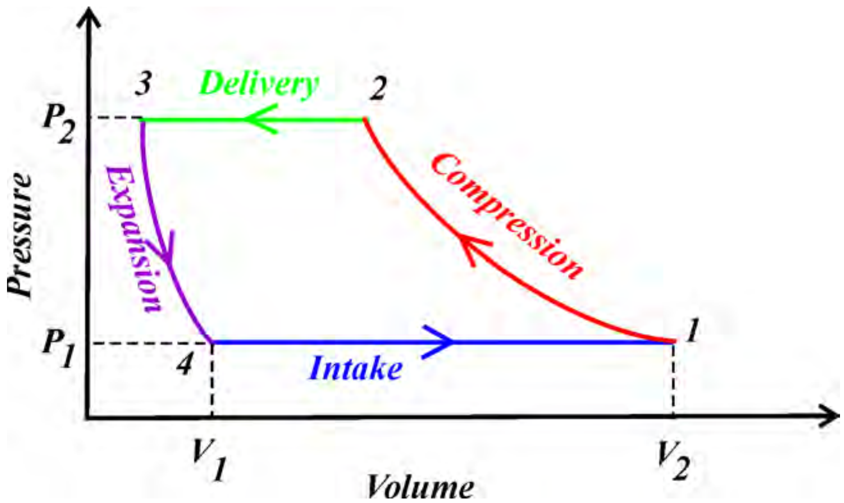


Figure 8.10: *PV diagram for reciprocating air compressor*

To get higher pressures that may reach up to 50 bar, it is necessary to use a multi-stage compressor. The two-stage compressor shown in Figure 8.11 is built of two piston compressors, the large one is used for low pressure compression while the smaller one compresses the same air to a higher pressure. An intercooler is used between them to reduce the air temperature before being compressed to the maximum pressure. Both stages are usually driven by the same motor.

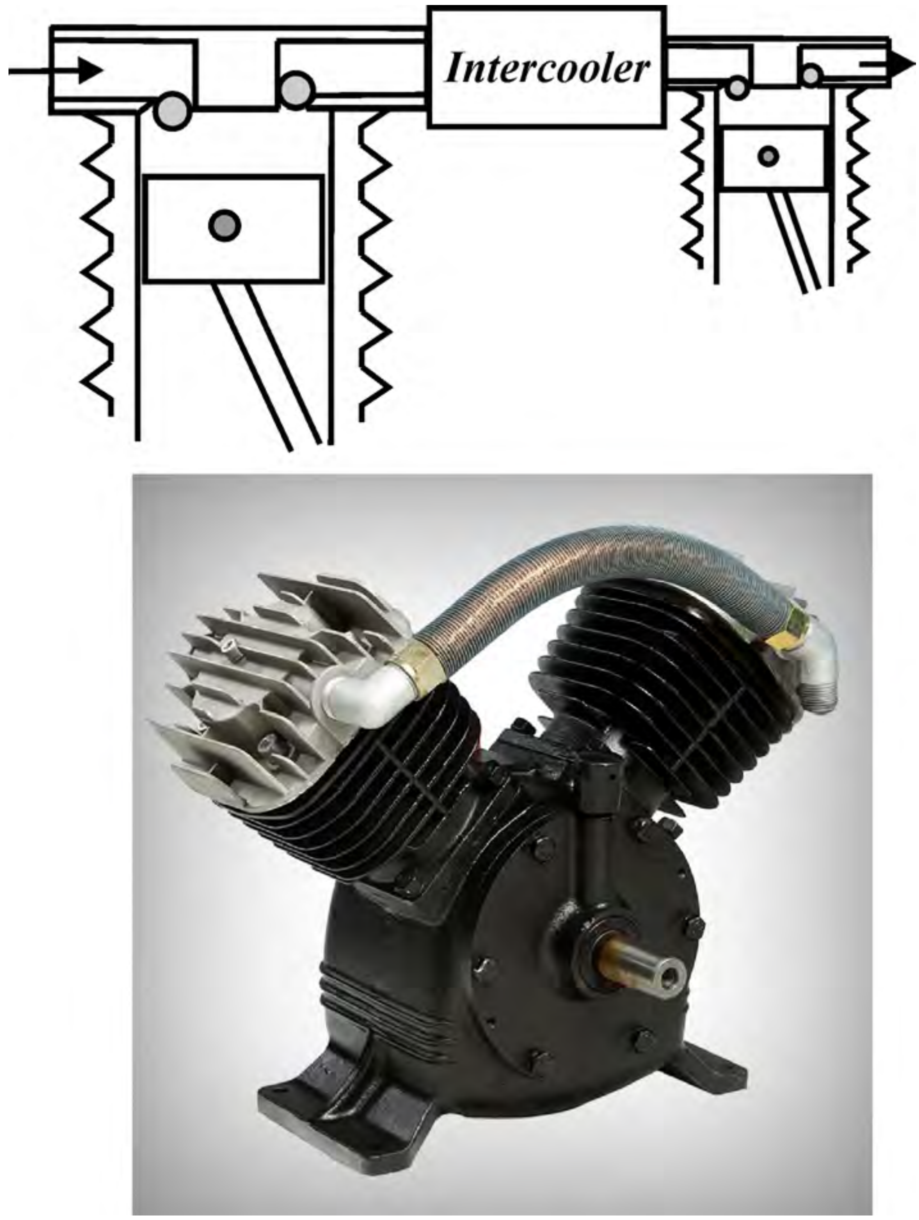


Figure 8.11: *Two-stage piston Compressor (Courtesy of Detroit Air)*

8.3.2 Diaphragm compressors

Piston compressors can cause a contamination to the outlet which makes a problem in food and medical industries. This problem is solved by separating air from moving parts by a flexible diaphragm as shown in the diaphragm compressor in Figure 8.12. Diaphragm compressors are often used in food, chemical and medical industries to avoid contamination from reaching to products.

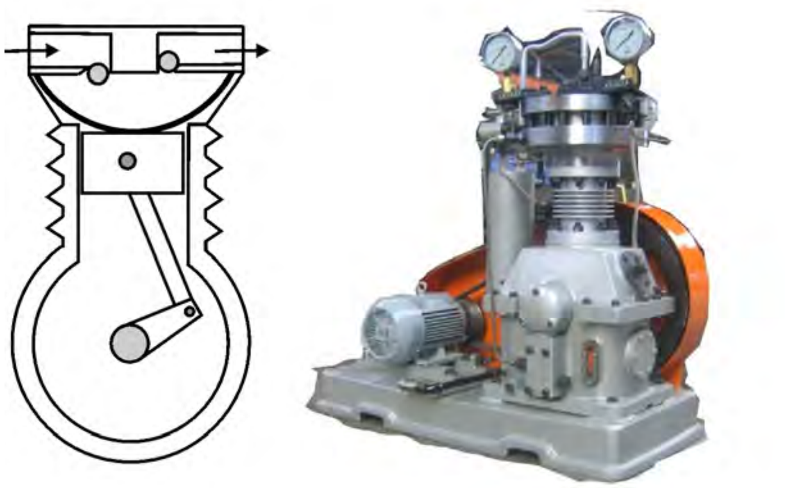


Figure 8.12: *Diaphragm Compressor (Courtesy of Shandong China Coal Group)*

8.3.3 Vane compressors

Vane compressors (Figure 8.13) have a similar design to vane pumps where the vanes can be loaded by springs or centrifugal force. This type of compressors cannot reach more than 3 bar pressure unless it is designed in two-stage where it can reach up to 10 bar.

8.3.4 Lobe compressors

Another type of rotary compressors is the lobe compressor shown in Figure 8.14. Lobe compressors are used to produce high volume flow rates but with lower pressure 1 - 2 bar. Rotating the lobes leads to enclose the air between the lobes and the casing and transfer it towards the outlet.

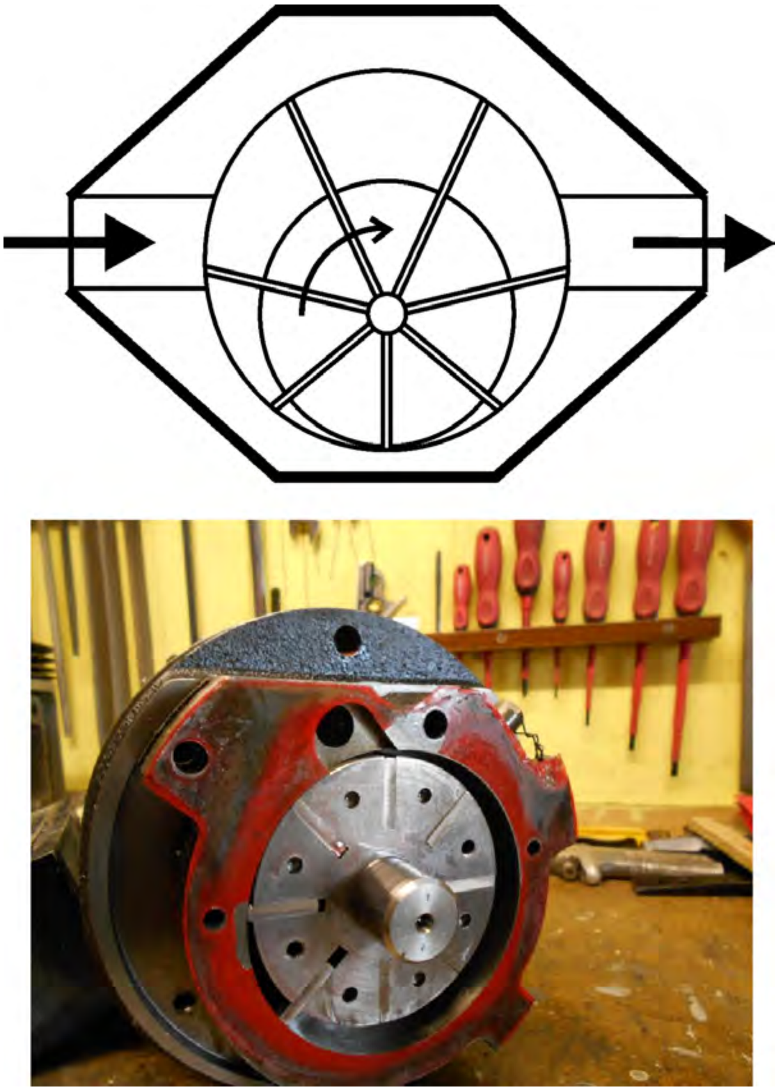


Figure 8.13: *Vane Compressor (Courtesy of Linquip)*

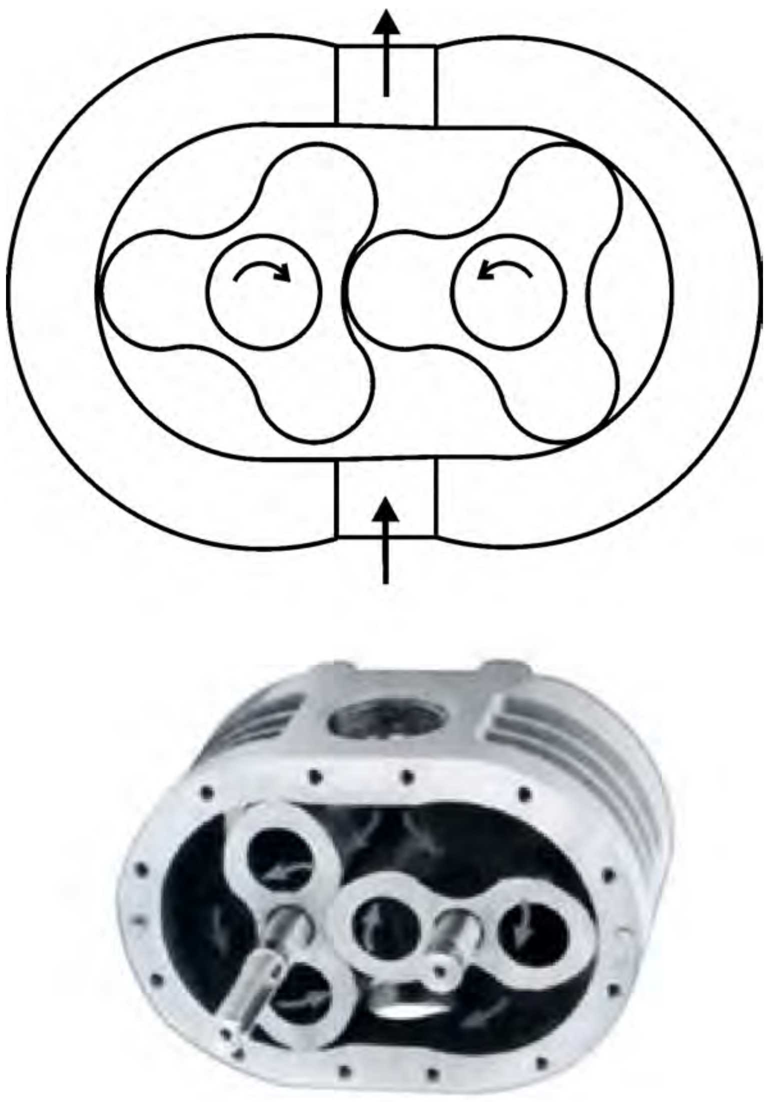


Figure 8.14: *Lobe Compressor (Courtesy of Delta Hybrid)*

8.3.5 Screw compressors

Screw compressor shown in Figure 8.15 is usually used in the applications where a high pressure (more than 20 bar) with a very low flow rate is required. High accuracy screws with very small clearance are used to enclose the air between the screws and the casing pushing it towards the outlet.

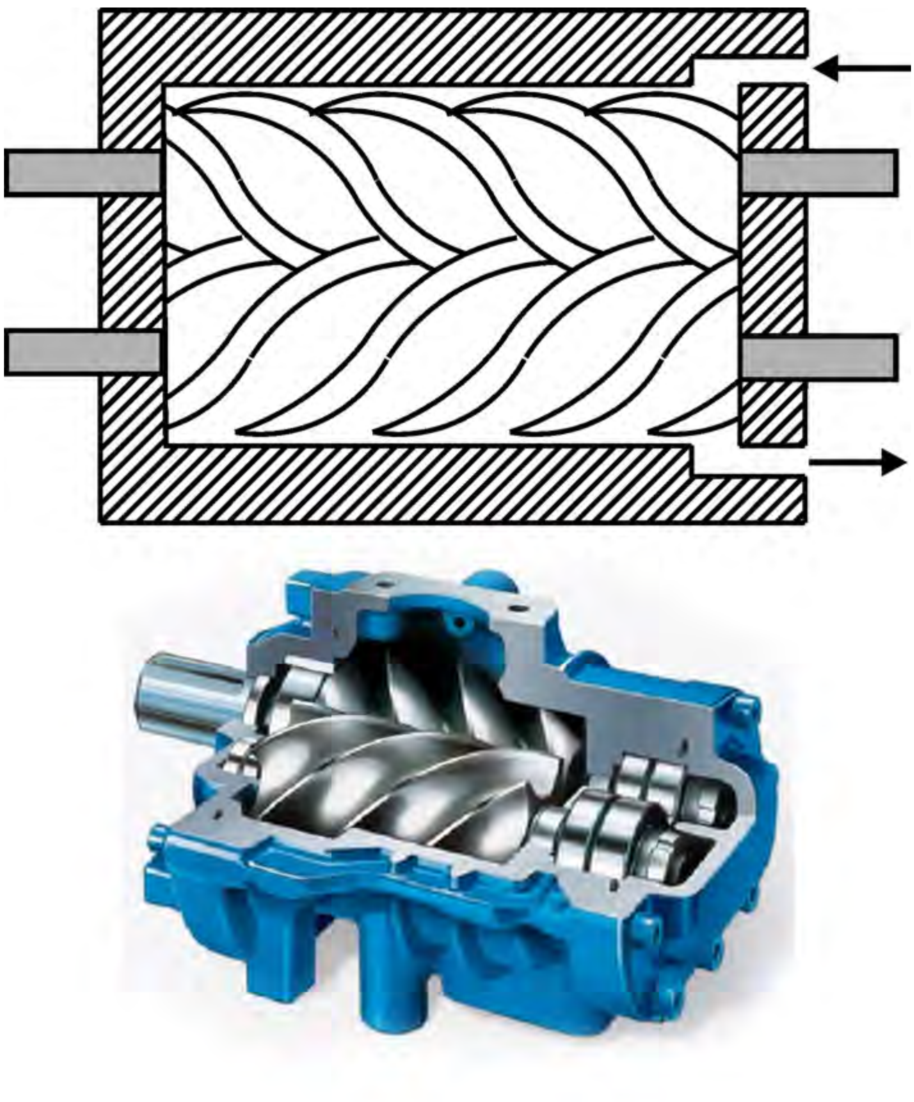


Figure 8.15: *Screw Compressor (Courtesy of BOGE)*

8.4 Actuators

Fluid power remains useless unless it is transferred to act on a specific load. This action can be done by an actuator. Actuators can be either linear or rotary depending on the required type of motion. A linear actuator consists of a cylindrical piston sliding axially inside a hollow cylinder by the influence of the force exerted on it by the fluid pressure. The resulting force is transmitted to the load by a rod connected to the piston. This force reads

$$F = PA \tag{8.1}$$

Where F is the induced force, P is the pressure of the compressed fluid and A is the inside cross sectional area of the piston. Rotary actuators have different shapes and designs like vane, gear and lobe types. The rotation torque is transmitted to the load by means of a rotating shaft connected to the rotating surface exposed to the fluid's pressure. This torque can be calculated by

$$T = PAR \tag{8.2}$$

Where T is the induced torque, P is the pressure of the compressed fluid, A is the area of surface exposed to the pressure in the rotating element and R is the distance between the center of the action of the pressure on the area A and the center of rotation of the rotating shaft. The following sections discuss different types of actuators.

Example 8.2

It is required to move a load of $10kN$ at a speed of $10cm/s$ using a hydraulic actuator and a pump. what is the pressure and flow rate of the pump knowing that the bore of the piston is $100mm$ and the its rod diameter is $40mm$. Calculate in extension and retraction strokes.

Solution
The piston side area is

$$A_p = (\frac{\pi}{4})(100 \times 10^{-3})^2 = 0.00785m^2$$

The rod area is

$$A_r = (\frac{\pi}{4})(40 \times 10^{-3})^2 = 0.00125m^2$$

Extension stroke:
The pressure in the system is:

$$P = \frac{F}{A_p} = \frac{10000}{0.00785} = 1.27 \times 10^6 Pa = 12.7bar$$

The flow rate is:

$$Q = vA_p = (0.1)(0.00785) = 0.000785m^3/s$$

$$Q = (0.000785m^3/s)(1000l/m^3)(60s/min) = 47.1l/min$$

Retraction stroke:
The pressure in the system is:

$$P = \frac{F}{A_p - A_r} = \frac{10000}{0.00785 - 0.00125} = 1.51 \times 10^6 Pa = 15.1bar$$

The flow rate is:

$$Q = v(A_p - A_r) = (0.1)(0.00785 - 0.00125) = 0.00066m^3/s$$

$$Q = (0.00066m^3/s)(1000l/m^3)(60s/min) = 39.6l/min$$

It is clear that in the retraction stroke the system needs higher pressure to move the same load but at the same time it needs less flow rate to move the same load at the same speed.

8.4.1 Single-acting actuator

The simplest design of an actuator is the single-acting actuator shown in Figure 8.16. This actuator is used to exert force and displacement in the extension stroke while the retraction stroke is actuated by the return spring installed internally or externally on the actuator.

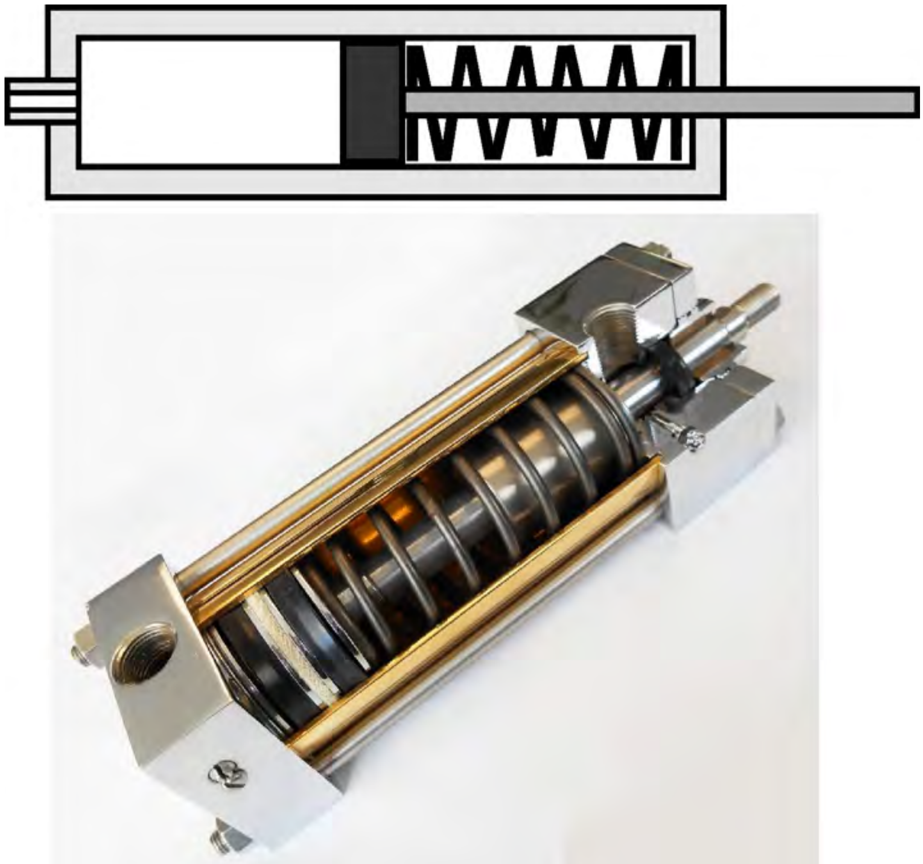


Figure 8.16: *Single-acting actuator (Courtesy of Lehigh Fluid Power)*

8.4.2 Double-acting actuator

To obtain double sided controlled motion of the linear actuator, the double acting piston in Figure 8.17 is used. A double-acting actuator consists mainly of a piston, rod, cylinder and two caps. The internal surface of the cylinder barrel should have a smooth surface finish to reduce the friction effect caused by the contact between the piston and the cylinder. Piston seals are wound around the piston as rubber gaskets to avoid leakage of fluid through the clearance between the piston and the cylinder. End seals and wiper seals are other rubber gaskets installed between the rod and the cylinder to avoid leaking the fluid out of the actuator. Compressing the fluid into the actuator from the extend port leads to extend the piston with the desired displacement while the exerted force depends on the operational pressure. On the other hand, retraction is obtained by pushing the fluid into the retract port.

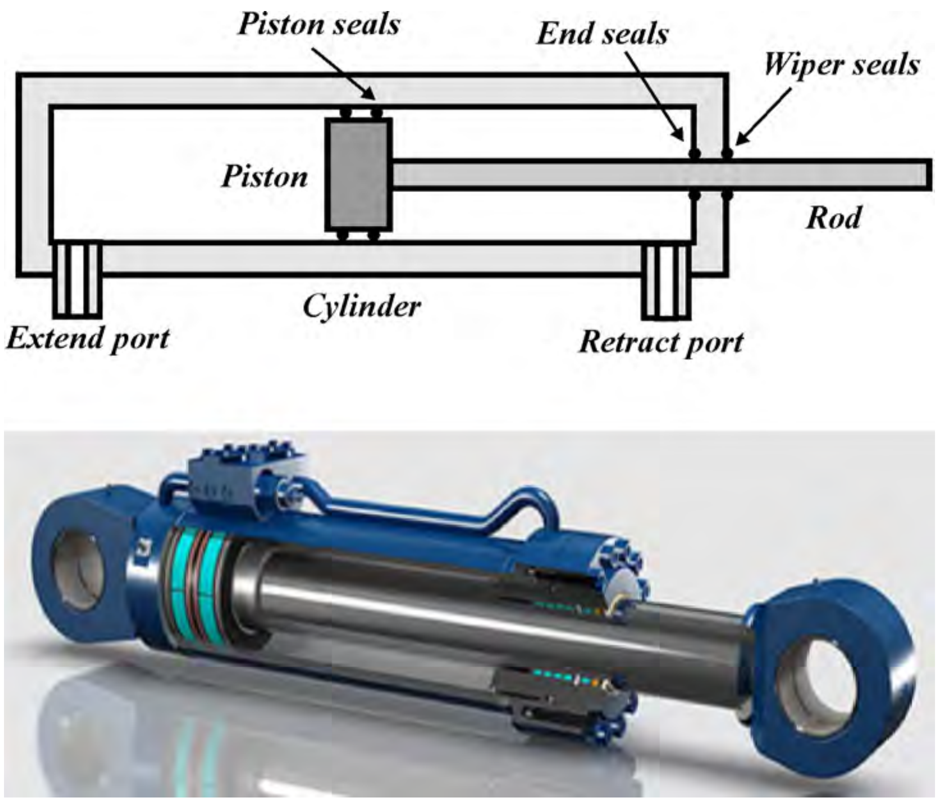


Figure 8.17: *Double-acting actuator (Courtesy of DALLAST)*

8.4.3 Telescopic actuator

Figure 8.18 depicts a two-stage telescopic piston used to have a longer actuation with a longer distance of the end rod. When oil is applied to the blank side of piston A pressure is applied to both sides of the piston exerting different forces on both sides due to different areas. The difference of the force causes piston A to move to the right till reaching the full stroke then piston B begins to extend. To retract the piston, oil is applied to the port on the rod side of piston B retracting piston B first followed by piston A.

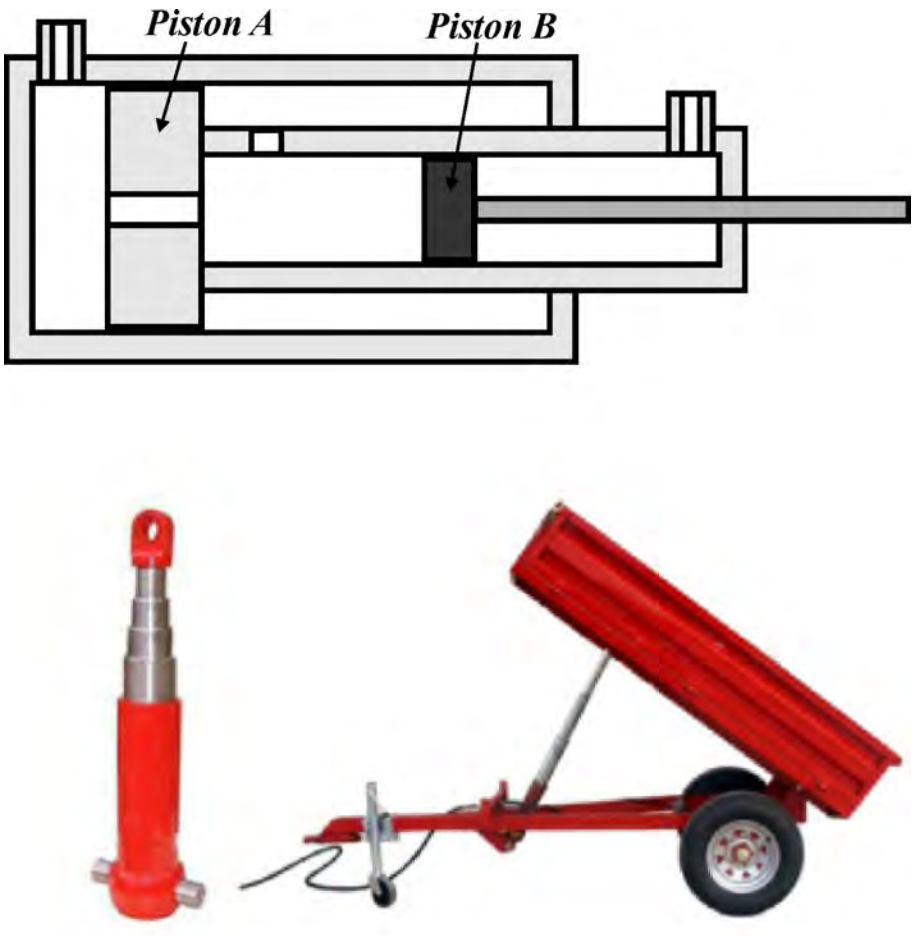


Figure 8.18: *Telescopic actuator (Courtesy of DALLAST)*

8.4.4 Cushions of actuator

Cushioning is used to decelerate the speed of extension or retraction of the piston near the end points of the stroke. Figure 8.19 shows a deceleration cushion added at the end of the retraction stroke of a piston. A plunger is installed at the blank side of the piston. This plunger enters inside a cylinder installed at the end cap of the cylinder reducing the flow of the exhausting fluid from the cylinder which is forced to exit through the needle valve slowing down the piston speed.

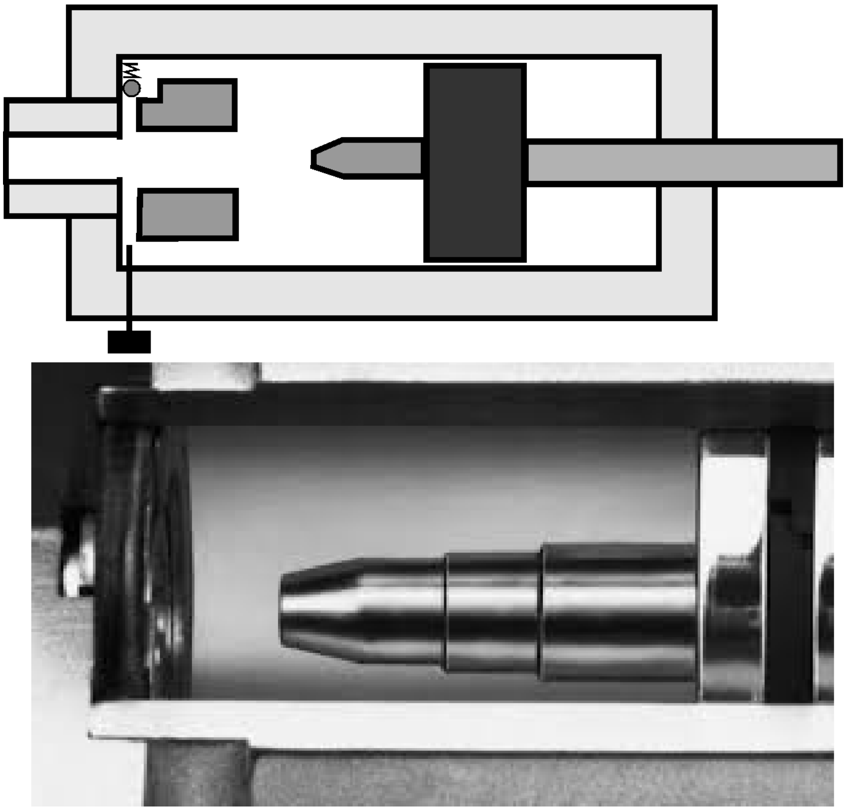


Figure 8.19: *Cushioning of an actuator (Courtesy of Parker Actuator Products)*

8.4.5 Gear motor

Rotary actuators can have one of three designs; gear motors, vane motors or piston motors. Gear motor (shown in Figure 8.20) consists of two gears in mesh enclosed in a casing. The inlet port enters the fluid at high pressure to rotate the gears passing between the teeth and casing to be exhausted from the low pressure port. The direction of rotation is as shown in Figure 8.20. Gear motor suffers from leakage between the teeth and casing because of the need to have enough clearance.

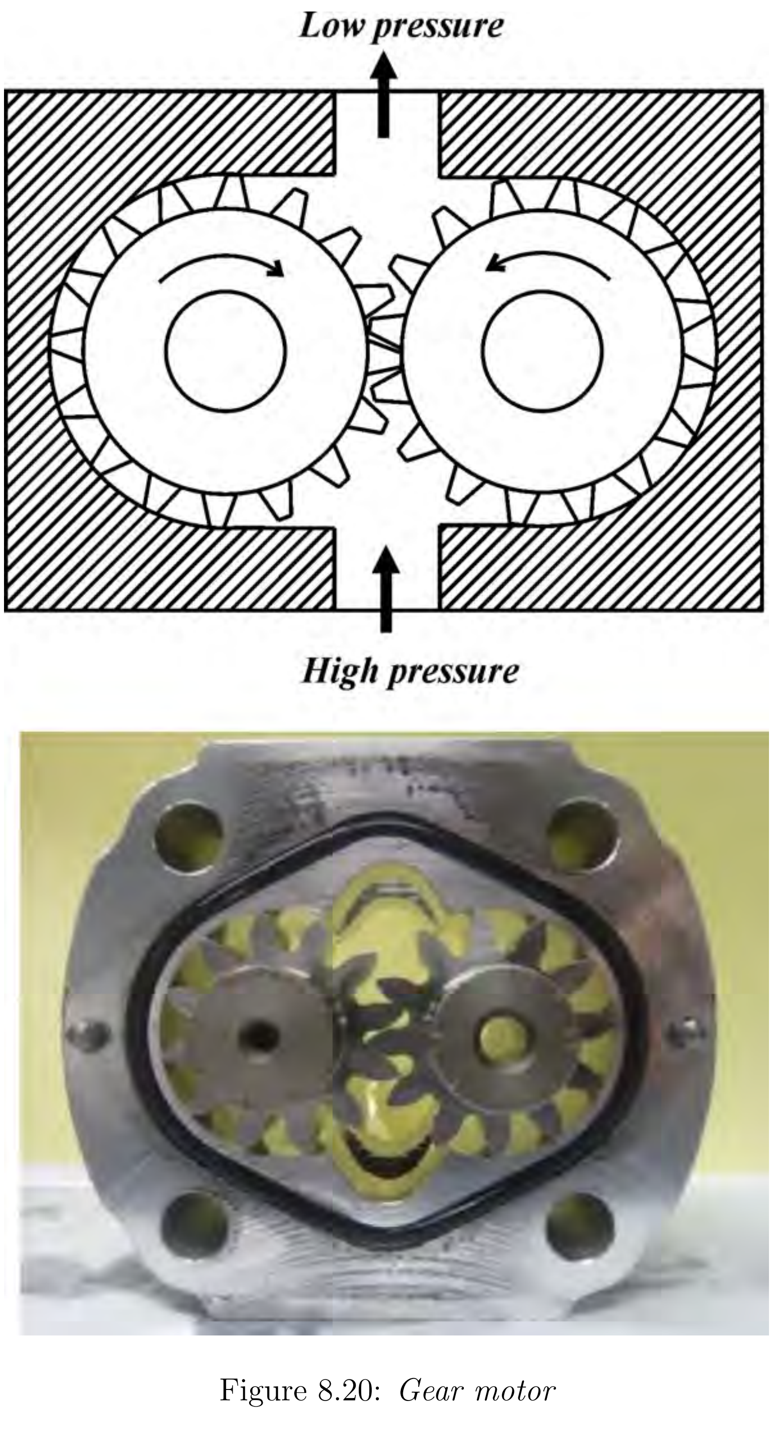


Figure 8.20: *Gear motor*

8.4.6 Vane motor

Vane motor shown in Figure 8.21 is similar to the design of the vane pump discussed earlier. Vane motor consists of several vanes sliding around a rotor inside a cam casing. Vane motors have less leakage effect than gear motors but the contact between the vanes and the casing is exposed to high friction effect. This type of actuators is usually used in low speed applications. Force unbalance can be solved by using dual design.

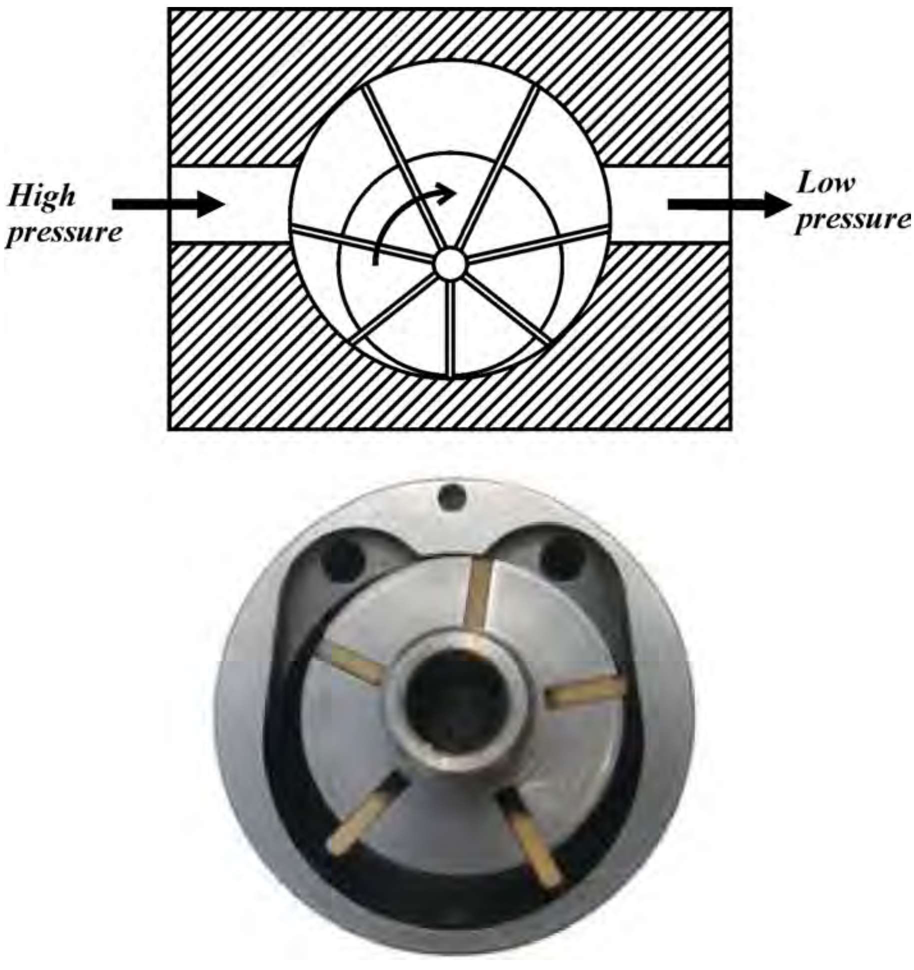


Figure 8.21: *Vane motor (Courtesy of FAUJI AUTO ELECTRIC PVT LTD)*

8.4.7 Piston motor

Higher torques and speeds can be achieved by using piston design actuators. Figure 8.22 shows a piston rotary actuator where compressed fluid extends the axial pistons leading to rotate the swash plate. Speeds are adjusted by changing the inclination angle of the swash plate just like the piston pump.

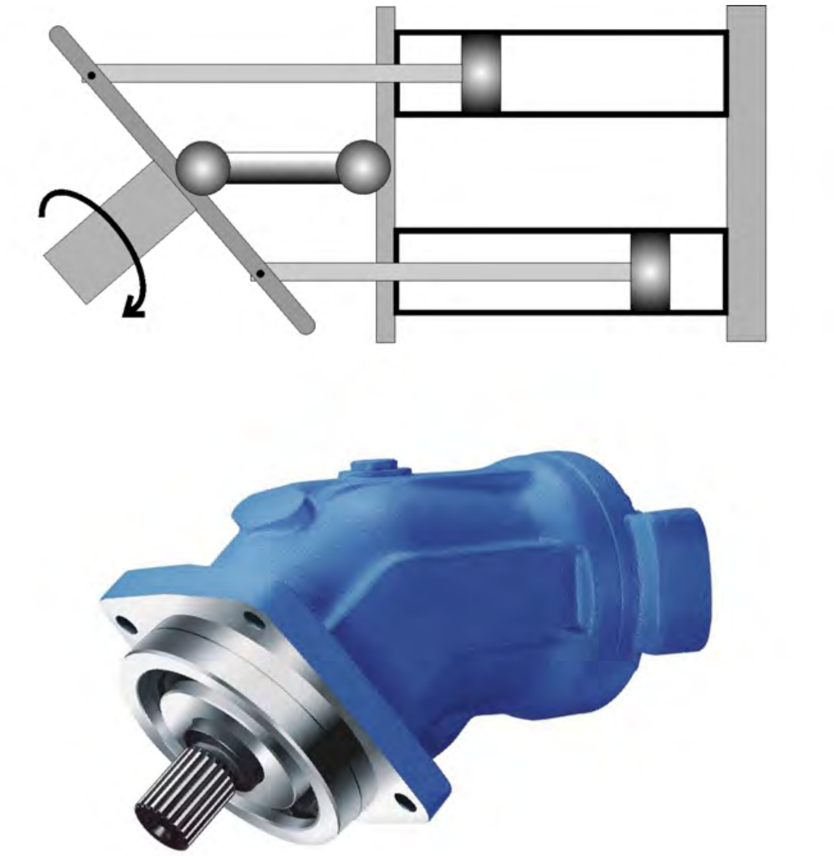


Figure 8.22: *Piston motor (Courtesy of Rexroth)*

8.5 Valves

Any hydraulic or pneumatic system needs different types of valves to direct and regulate the flowing fluid either oil or air before reaching to the application. These valves can be divided into two main types:

- Infinite position valves: this type of valves can take any required position to close and open gradually determining the area of the port (orifice) in order to control the flow rate that in turn regulates the speed of the actuator. Besides to the fact that it can stop at any position.
- Finite position valves: these valves are also called ON/OFF valves that serve to either fully open or fully close to pass or to stop the fluid flow.

The conceptual design of the different types of valves is almost the same for hydraulic and pneumatic applications. The only difference is the need for stronger materials and seals in the case of hydraulics because they need to support higher pressures. The following sections discuss different general designs of valve types.

8.5.1 Check valve

Check valves, or called non-return valves are used to keep the direction of flow in one sense of direction preventing the fluid from flowing back in the opposite direction. Figure 8.23 shows a simple design of the check valve where a ball mounted on a prestressed spring is used to close the orifice in one direction. When the force induced on the ball's surface by the fluid pressure exceeds the spring force, the ball is pushed down allowing the fluid to flow through the valve. The arrows on the drawing determine the direction of flow that cannot be reversed unless the ball is pushed against the spring by another mean like a mechanical handle, pliot pressure or a solenoid.

8.5.2 Poppet valve

Poppet valve shown in Figure 8.24 is similar to the the check valve discussed in the previous section except that the poppet valve is operated by a manual handle that serves to push the ball against the spring force to open the orifice and allow the fluid to pass freely in the direction shown in the figure. In the opposite direction, the valves behaves like a normal check valve.

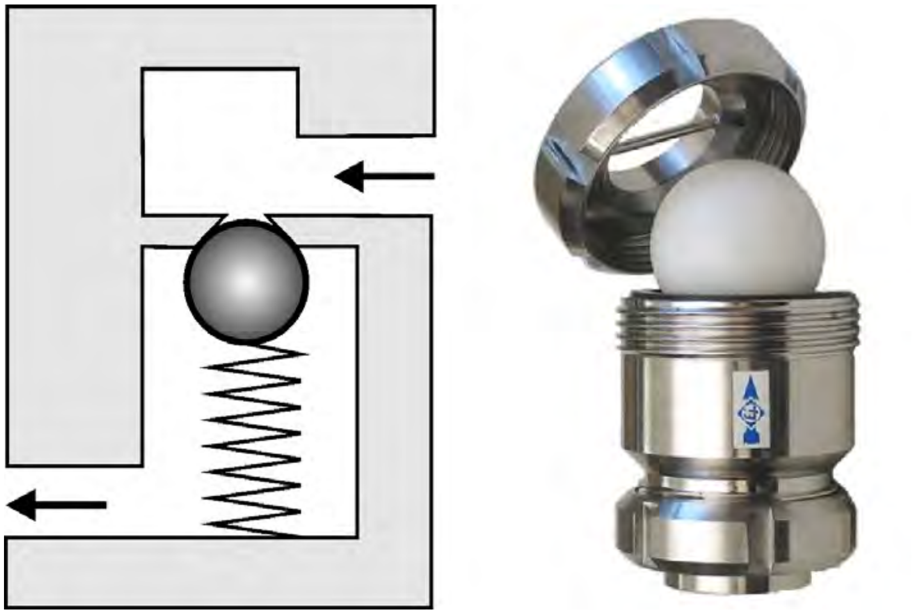


Figure 8.23: *Check valve (Courtesy of Equip Inox)*

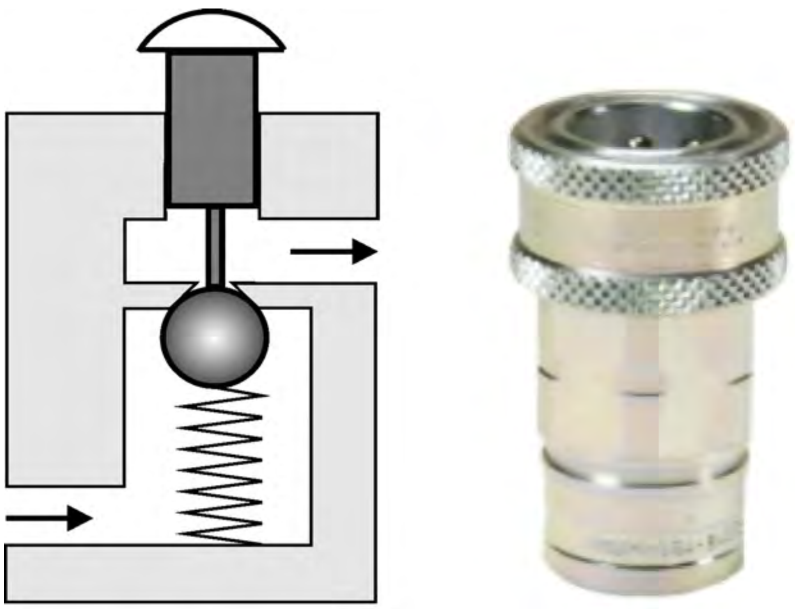


Figure 8.24: *Poppet valve (Courtesy of Dixon FTP)*

8.5.3 Spool valve

Figure 8.25 shows a directional control spool valve. Spool valves are the most commonly used valves to control the direction of flow. Spool valve consists mainly of a spool rod with cylindrical lands allowed to slide inside an envelope vented to the main inlets and outlets of the valve. When the spool is pushed to the right, pressurised fluid coming from the pump flows through port B to the application while port A is open to the tank in this case to drain the fluid existing in the piston back to the tank. Pushing the spool to the left reverses the operation. Spool can be operated either manually or automatically using a pilot pressure or a solenoid. A third action can be achieved by keeping the spool in a center position by two springs mounted on both sides of the valve.

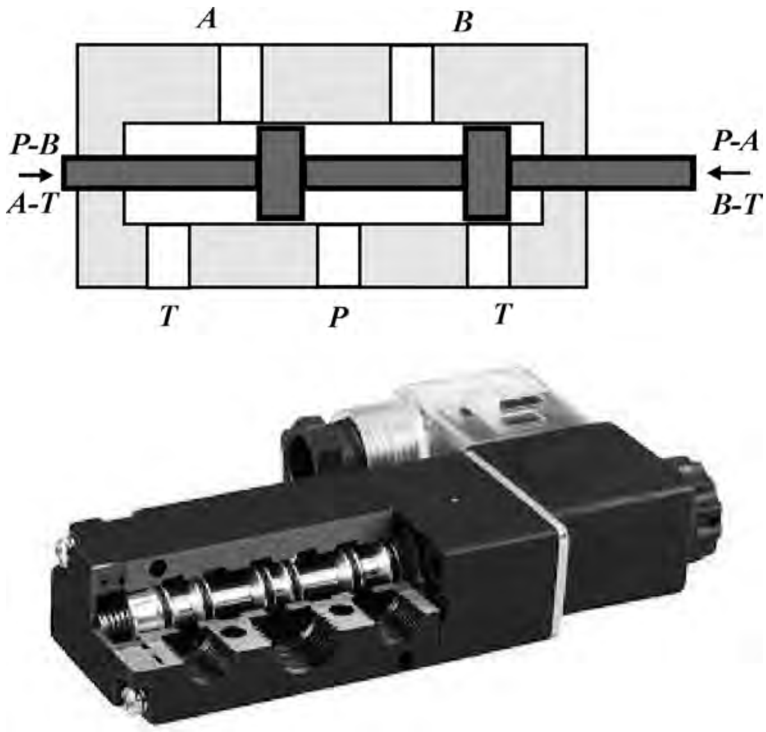


Figure 8.25: *Directional Control spool valve (Courtesy of Tameson)*

8.5.4 Rotary valve

The direction of fluid flow can be controlled by rotary valves as the design shown in Figure 8.26. The outer cylindrical casing containing the inlet and outlet ports is fixed. The inner rotating rotor consists of two perpendicular channels used to connect specific ports together when rotated. In the shown position, the pressure port is connected to port B and port A is connected to the tank. Other positions can be obtained by rotating the internal rotor either clockwise or counter-clockwise.

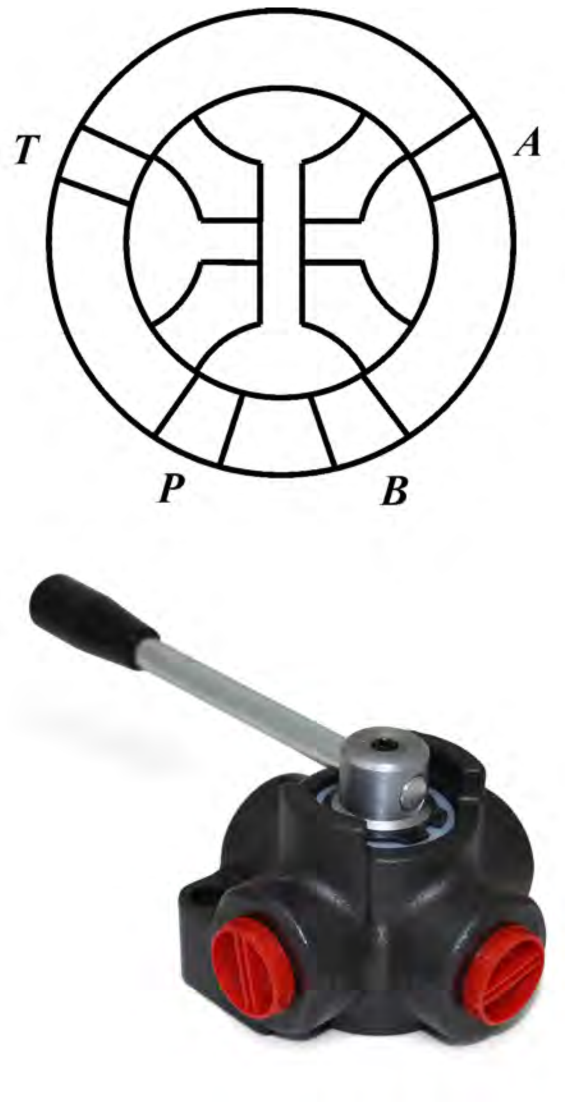


Figure 8.26: *Rotary valve (Courtesy of Summit Hydraulics)*

8.5.5 Shuttle valve

Shuttle valve or double check valve shown in Figure 8.27 is used to keep a constant pressure in the system when the flow comes from either of two inlets X or Y. The valve consists of a ball enclosed in an envelope. When the fluid enters from inlet X, the ball is pushed to close inlet Y allowing the fluid to flow through the valve to port A. The same action occurs when the flow comes from inlet Y closing X and allowing fluid to flow from Y to A. Shuttle valves are rarely found in hydraulic systems, rather they are commonly used in pneumatic circuits. In some designs, ball of the shuttle valve can be loaded by a spring to one side and can be pushed to the other side by the influence of fluid pressure.

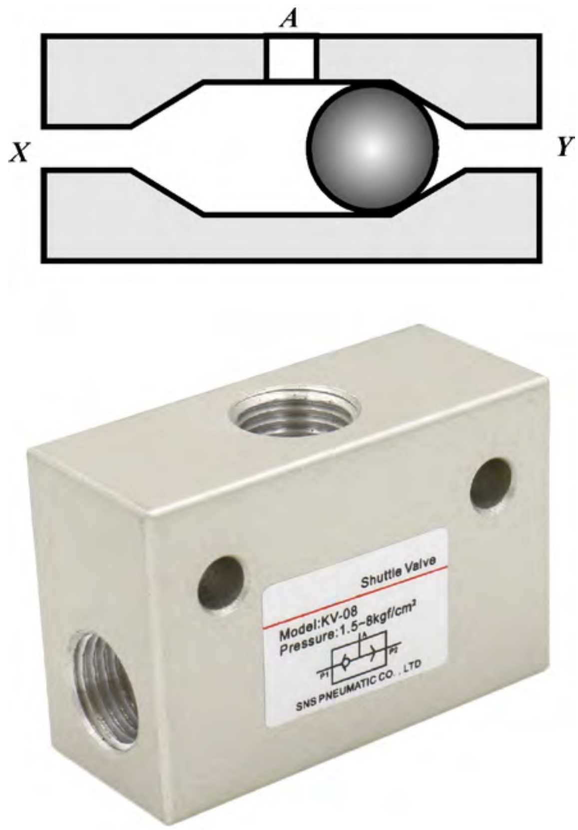


Figure 8.27: *Shuttle valve (Courtesy of Heschen)*

8.5.6 Sequence valve

Sequence valve is a combination of a directional control valve and a pressure relief valve. Figure 8.28 shows a sequence valve that consists of a directional control spool valve connected to two pressure relief valves. When the spool moves to allow the fluid to extend piston 1 the pressure relief valve remains closed till piston 1 reaches its end stroke and thus the pressure increases in that line, at this moment, the relief valve opens allowing the fluid to flow to extend piston 2 in a sequence form. The same operation occurs in the retraction stroke.

8.5.7 Pilot operated valve

Figure 8.29 shows a pilot operated check valve. The pilot fluid flows to push the small piston connected to a pushing rod that serves to push the ball against the spring opening the main port to allow the operation fluid to pass to the application. The main advantage of the pilot operation of the check valve is that it enables the valve to allow the reversed flow of fluid when actuated. Normally, the pressure of the pilot line does not exceed ten percent of the operation pressure.

8.5.8 Time delay valve

Figure 8.30 shows a time delay valve that can be used to delay the actuation of the valve delaying the operation with a time depending on the adjustment of the valve. Time delay valve contains a reservoir in the way of the incoming pilot fluid besides to a needle valve at the inlet. Delay of the operation is achieved by changing the orifice of the needle valve where filling the reservoir takes a time before reaching the required pressure to push the ball of the check valve.

8.5.9 Modular valve

Modular valve is a way to connect valves to the inlets and outlets. This is done by having a base containing internal pipes and channels to connect between the different ports of the valves while several valves are connected

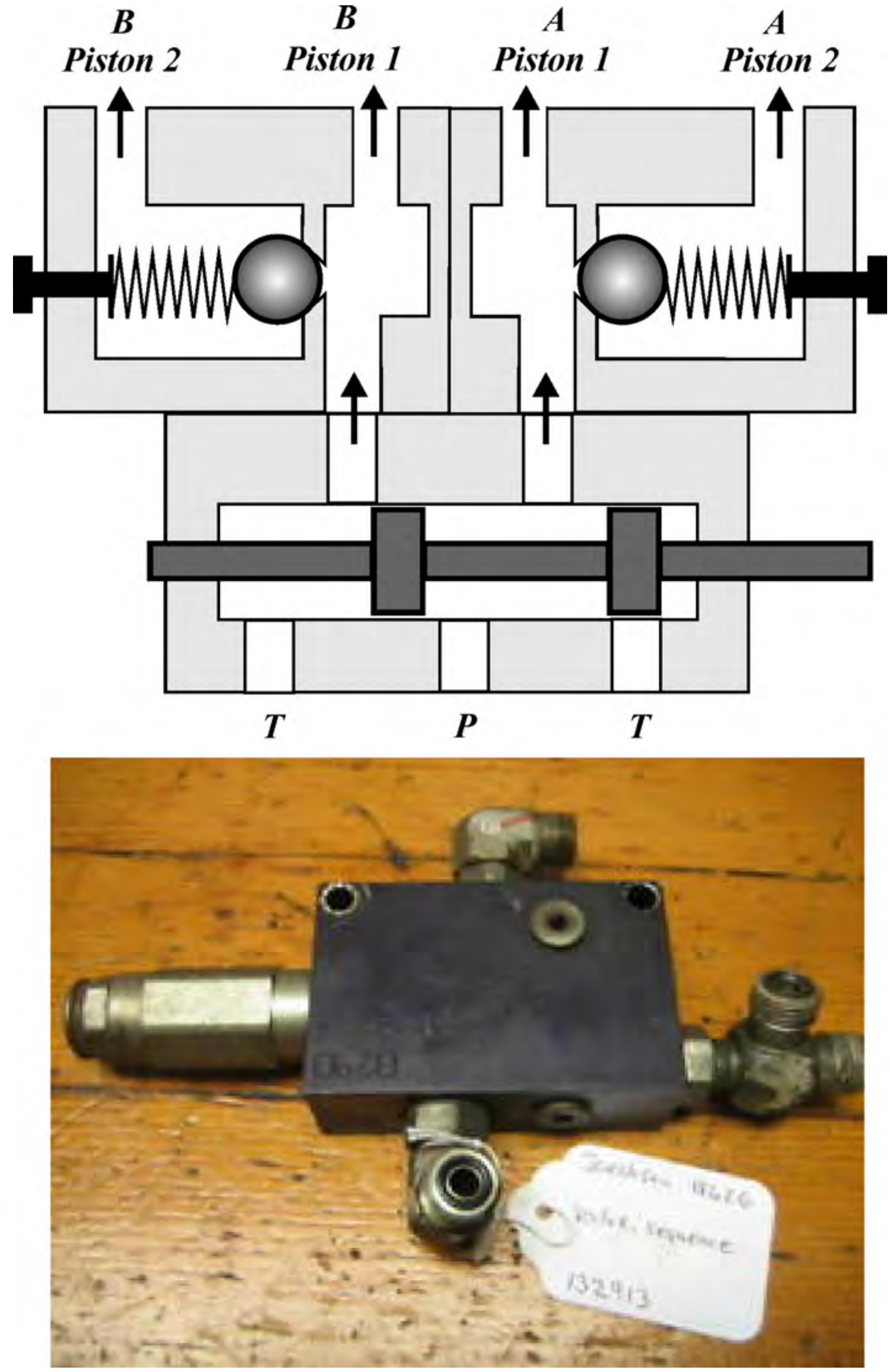


Figure 8.28: *Sequence valve (Courtesy of JACOBSEN)*

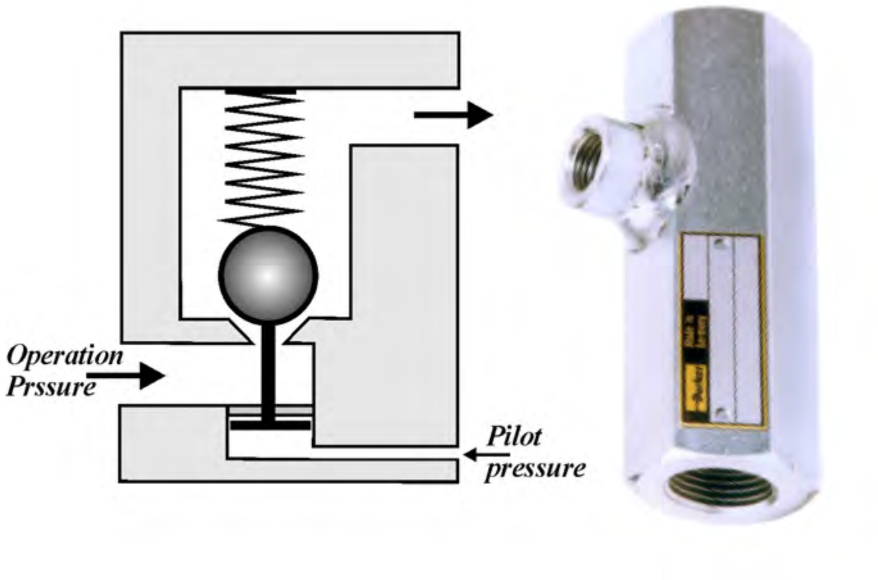


Figure 8.29: *Pilot operated check valve (Courtesy of Parker Hannifin)*

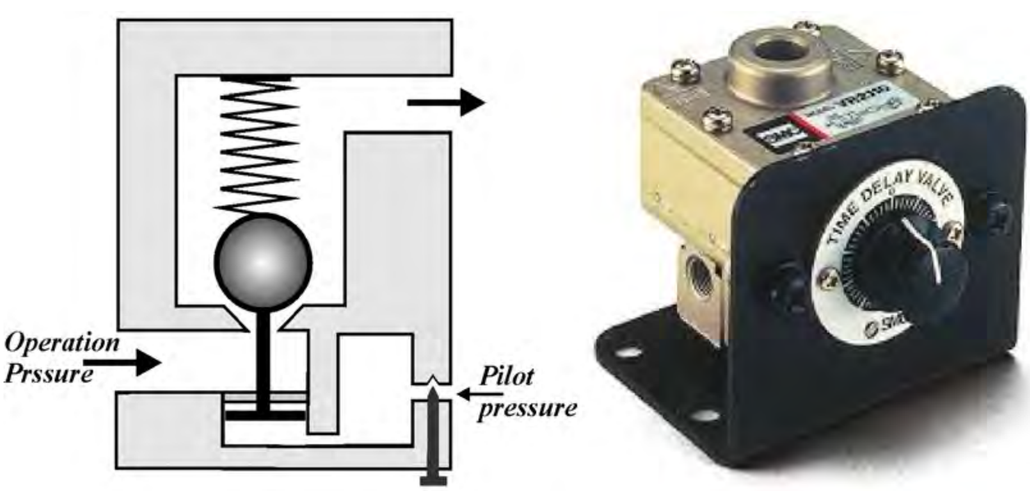


Figure 8.30: *Time delay check valve (Courtesy of SMC)*

to this base. Normally, the inlet to the modular valves is common while the outlets are different to reach to the different applications. Figure 8.31 shows an example of modular valves.

Figure 8.31: *Modular valves (Courtesy of Hydrastar)*

8.6 Problems

1. What does the best efficiency point in pump selection depend on.

2. How to calculate the static head and friction head in the pump performance curve.

3. If the operating point of the pump is on $30l/min$ and $200bars$, what is the maximum load that can be carried by a 150mm bore actuator and is its maximum speed.

4. It is required to rotate a load at a $3N.m$ torque and angular speed of $20rpm$, select a pump for a volumetric efficiency of 85% and select the prime mover to function the pump assuming a mechanical efficiency of 80%.

5. What are the main differences between the gear pump and lobe pump, suggest applications for both of them.

6. What are the differences between the axial piston swash plate pump and bent axis pump.

7. What is the main reason for using intercooler between the two stages of the two-stage compressor.

8. What are the advantages and applications of the diaphragm compressor.

9. Oil is pumped in a $25mm$ diameter pipe at a speed of $4m/s$, what is the speed of connected $200mm$ bore actuator.

10. Air pressure in a single acting $80mm$ pneumatic actuator is $8bars$ and the actuator contains a return spring with $1000N/m$ stiffness coefficient, what is the maximum load it can move a distance of $100mm$.

11. What is the range of pressure used to operate the pilot operated check valve.

12. Build a second order differential model for spool directional control valve and suggest parameters.

13. Explain how to install four DCV valves in a modular way to control a four boom concrete pump.

8.7 References

[1] A. Parr, *Hydraulics and Pneumatics, A technician's and Engineer's Guide*, Newnes, 1991.

[2] A. Esposito, *Fluid Power with Applications*, Prentice Hall, 2003.

Chapter 9

Oils and Piping of Hydraulic Systems

9.1 Introduction

Fluids are used in fluid power systems as power transmission means where they transmit force and displacement from one point to another. Some of these fluids are compressible gases and others are incompressible liquids. The first used liquid for hydraulic systems was water because it is cheap and available but the main problem encounters here is the oxidation reactions causing rust to the used metallic materials. In the beginning of the 20th century industrial people started to think about using mineral oils as a mean in hydraulic systems instead of water and this solved the problem partially where it helped in the lubrication of the moving parts and reduced the corrosion but not completely because it is highly flammable and exposed to fire when operated at high temperatures and still exposed to rust and oxidation. This created the need to develop the synthetic industrial fluids where chemicals and additives are added to the fluid ending with inflammable and anti-corrosion fluids.

9.2 Hydraulic oil Properties

9.2.1 Viscosity

Viscosity is defined as (the resistance to flow), it represents the resistance of a specific fluid to overcome the velocity of flow in a vessel. Dynamic viscosity is defined as the ratio between the shear stress and the velocity gradient (du/dy) [1], for Newtonian fluids the dynamic viscosity is independent of velocity gradient but it is influenced by the change in temperature and pressure, coefficient of dynamic viscosity is shown in Newton's law in equation 9.1:

$$\mu = \frac{\tau}{du/dy} \tag{9.1}$$

where

μ = Dynamic viscosity ($1 Poise = 0.1 N.s/m^2$)
τ = Shear stress (N/m^2)
u = Fluid velocity (m/s)
y = Displacement perpendicular to velocity vector (m)

Kinematic viscosity is the ratio of the dynamic viscosity to the density as shown in equation 9.2

$$\nu = \frac{\mu}{\rho} \tag{9.2}$$

where
ν = Kinematic viscosity ($1 St = 10^{-4} m^2/s$)
ρ = Oil density (kg/m^3)

Viscosity Grade (VG) is an indicator used to show the viscosity of fluids where the number following the letters (VG) indicates the kinematic viscosity in centi-Stoke (cSt) at a temperature 40^oC. For example the oil having a grade (VG 68) means that its viscosity equals 68 cSt at a temperature 40^oC. The most commonly used hydraulic oils are the grades (VG 10, VG 22, VG 32, VG 46, VG 68 and VG 100). Viscosity Index (VI) is another indicator calculated for the viscosity of oils at 40^oC and at 100^oC, VI indicates the speed of the oil's viscosity variation due to temperature change. The effect of temperature change on viscosity of oils is shown in Figure 9.1 where the viscosity of low grade oils decreases at lower temperatures. This means that low grade oils (VG 22 and VG 32) are used for cold places like

North Europe and long pipelines while high grade oils (VG 46 and VG 68) are used in warm and hot countries like Africa and Middle East. VG 100 is a special oil used for extremely high temperatures like internal combustion engines. The sharp decrease in oil viscosity leads to high increase in Reynold's number turning the flow to turbulent flow developing bubbles in the fluid that causes cavitation that leads to pump corrosion. On the other hand, reducing viscosity causes excessive oil leakage either internally or externally that increases the non-linearity in the system and affects the control possibilities due to losses of flow rate and pressure in the system [1].

The effect of pressure on viscosity is much less than temperature as shown in Figure 9.2. The effect of pressure on viscosity at low pressures from 0 to 300 bars is negligible while at higher pressures it depends mainly on the temperature where at 40^oC viscosity increases quadratically as a function of the pressure and at 100^oC the increase is very slow.

9.2.2 Hydraulic oil density

Density is defined as the mass per unit volume. Hydraulic mineral oils are considered to have low compressibility and low volumetric thermal expansion factor under operating conditions. Therefore, their density is considered constant under these operation conditions [2]. Density of mineral hydraulic oils vary from 850 - 900 kg/m^3. Oil density has a high influence on velocities and pressure on oil passing through sharp-edge orifice. consider that the velocity and pressure before orifice are v_1 and P_1 respectively, and the velocity and pressure after orifice are v_2 and P_2 respectively. Applying Bernoulli's equation of energy before and after leads to the relationship in equation 9.3:

$$\rho = \frac{2(P_1 - P_2)}{(v_2^2 - v_1^2)} \tag{9.3}$$

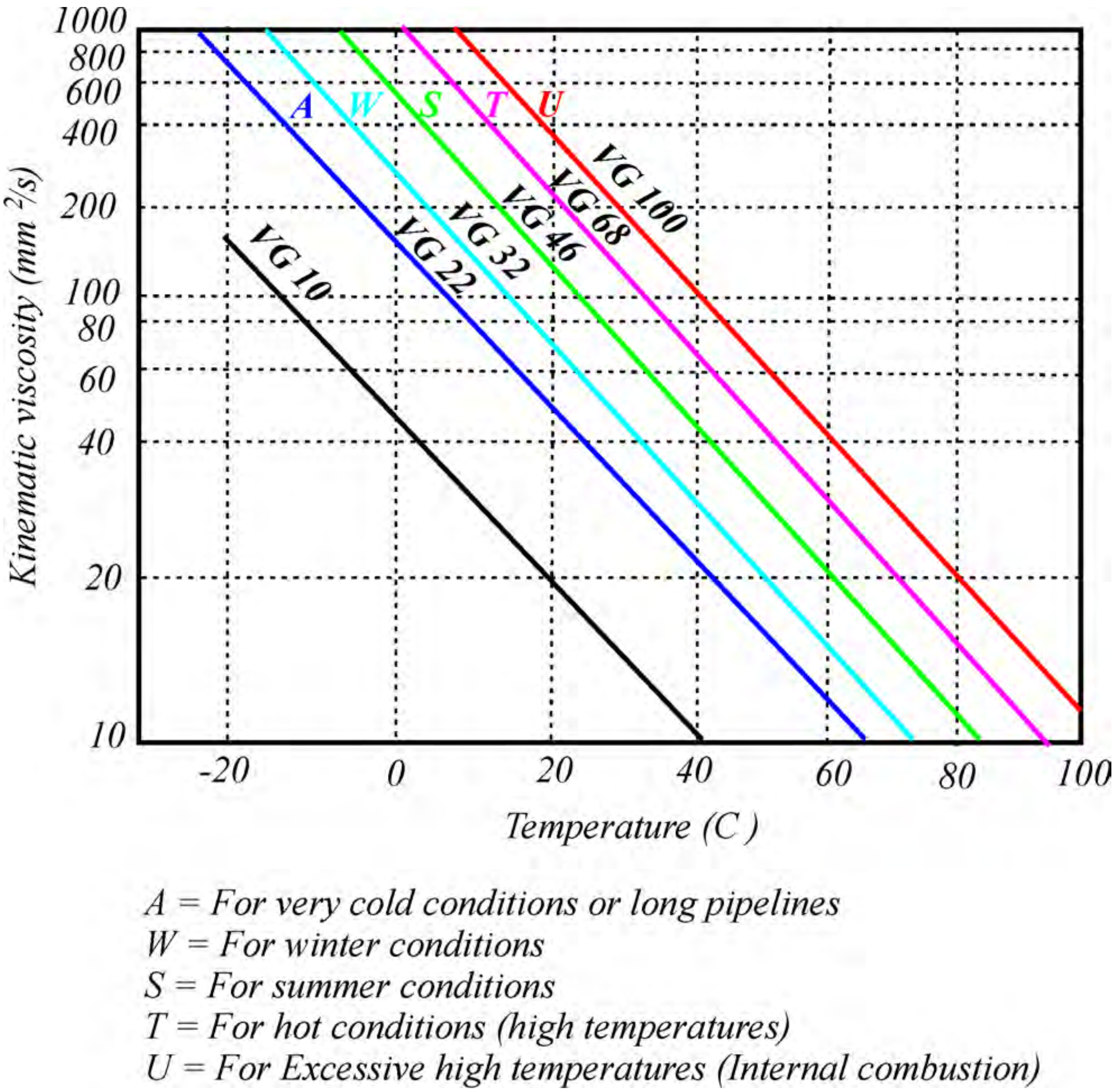


Figure 9.1: *Viscosity variation of different hydraulic oils due to temperature change*

9.2.3 Hydraulic oil compressibility

Compressibility is defined as the ability of fluid to change its volume under applied pressure. Gases are compressible fluids because they have high ability to change their volumes under pressure while liquids are considered incompressible, but this is an approximation because liquids and oils are also compressible but they have a very low compressibility [3]. Compress-

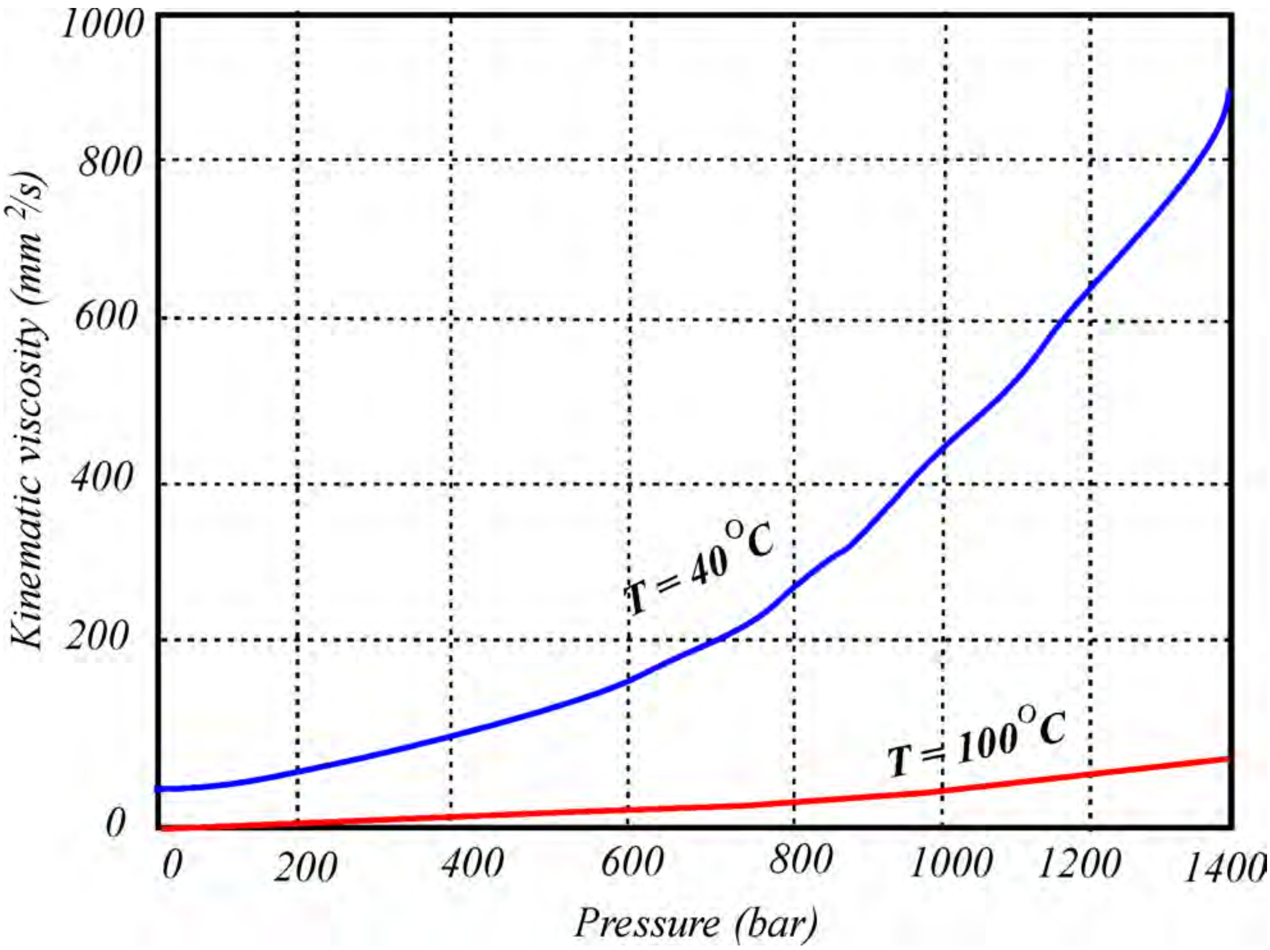


Figure 9.2: *Viscosity variation of a specific hydraulic oils due to pressure change*

ibility is measured by the Bulk modulus of liquids B which is defined as the ratio of pressure variation to volume variation as shown in equation 9.4:

$$B = \frac{(P_1 - P_2)}{(\Delta V/V)} \tag{9.4}$$

Where

$(P_1 - P_2)$ = Pressure variation (Pa).
ΔV = Change of volume (m^3).
V = Initial volume (m^3).
B = Bulk modulus of liquids, typically B = 1-2 GPa for mineral fluids.

Bulk modulus is influenced by pressure and temperature, it increases by increasing pressure and and decreases by increasing temperature. Compressibility has a remarkable influence on linearity of the system where it increases the dynamics of the hydraulic system causing higher order of its model leading to more complications in controlling the fluid by restricting the bandwidth of operation due to the appearance of dynamic resonance created by compressibility [4].

9.2.4 Thermal expansion of hydraulic oils

Dynamic motion and operation of hydraulic liquids causes high temperature [5]. This temperature causes thermal expansion to these fluids governed by equation 9.5

$$\Delta V = \alpha V \Delta T \tag{9.5}$$

where
ΔV = Change of volume (m^3)
α = Thermal expansion coefficient, for mineral oils $\alpha = 0.0007K^{-1}$
ΔT = Temperature variation ^oC
V = Initial volume (m^3)

9.2.5 Other properties

Besides to the previously mentioned properties, there are other properties for the mineral hydraulic oils:

- Vapour pressure at which liquid will boil causing bubbles that lead to cavitation.
- Lubrication and anti-wear where the oil is required to form a film between the moving parts to avoid friction and wear.
- Chemical stability where the oils are required to resist oxidation and deterioration to last the longest period possible.
- Foaming due to dissolved air in the liquid which makes bubbles causing cavitation.

- Cleanliness from air contamination and other types of contaminants that may cause wear to moving and contacting parts.
- Water contamination causes rust and wear to part surfaces.
- Acidity and toxicity.

9.3 Hydraulic transmission pipelines

9.3.1 Hydraulic tubes

Hydraulic tubes are usually manufactured of carbon steel and are used to transmit high pressurized oil through fixed lines. Selection of hydraulic tubes depends on:

- Material
- Size
- Length
- Fluid type
- Temperature
- Pressure
- Design

Tube diameter is usually calculated using the continuity equation knowing the flow rate and velocity of fluid. Typical fluid velocities in the pressure line vary from 2 to 6 m/s while in the suction or return line from 0.6 to 1.6 m/s. When designing hydraulic tubes designer must take into account to bend tubes as much as he can to avoid using fittings and he must use symmetrical installation to have a neat shape and easier installation. Table 9.1 show some examples of tube sizes, their operation pressure and maximum burst pressure at which it explodes.

Table 9.1: *Sizes, operation pressures and burst pressures of hydraulic tubes*

Tube size (mm)	**Wall thickness (mm)**	**Working pressure (bar)**	**Burst pressure (bar)**
6	1	250	1300
6	2	730	2600
10	1	150	789
10	2	400	1500
15	1.5	170	790
15	2	250	1000
20	2	180	780
20	4	440	1600
25	2	150	600
25	3	240	940
30	2	120	520
30	4	280	1000
35	2	100	450
35	3	170	670
42	2	84	370
42	4	190	740
50	3	120	420
60	3	100	340
75	3	76	310
90	3.5	75	300
115	4	67	270
140	4.5	63	250
165	5	60	230
220	6	54	210
273	6	44	180

9.3.2 Connections and fittings

To connect tubes together, different types of fittings and connectors are required depending on the application, installation and design. Some of these fittings are shown in Figure 9.3.

Figure 9.3: *Fittings and connectors for hydraulic tubes (Courtesy of Global Industrial)*

9.3.3 Hoses

Hoses are flexible pipes used to connect tubes at moving or vibrating points to allow for flexibility of connection during operation. Flexible hoses are made of multiple materials like fluoropolymers and silicone, elastomers, metal, and thermoplastics. Composite or laminated materials are also common. Rubber and elastomeric hydraulic hose is a strong choice for better flexibility. Hoses are constructed as shown in Figure 9.4 from the following layers:

1. Inner synthetic rubber tube, it also may have an internal dressing layer of Viton.
2. High-abrasion-resistant metallic screen layers.
3. Closely braided high tensile steel wire layers separated by anti-friction rubber layer.
4. Abrasion resistant synthetic rubber cover.
5. Textile layer.

Figure 9.4: *Construction of hydraulic flexible hose (Courtesy of Maha hydraulics)*

Hoses are connected to tubes and to each others using different types of end fitting like the ones shown in Figure 9.5. The following procedure is done to install an end fitting to a hose:

1. The outer textile layer of the hose is pealed.

2. The hose is inserted inside the outer ferrule of the fitting.
3. The barbed size of the inner main fitting is inserted into the hose.
4. the end fitting is inserted into a hydraulic press machine to reduce the diameter of the outer ferrule.
5. fitting is ready.

During installation of hoses one must take care not stretch the hose too much or give it excessive length. sharp or perpendicular curves should be avoided and hoses should not be loose to touch each other or any part to avoid wear caused by vibrations.

Figure 9.5: *End fittings for hydraulic flexible hoses (Courtesy of Global hydraulics)*

9.4 Problems

1. An oil with a viscosity grade VG 68 passing through a 25 mm diameter pipe at a flow rate of 30 l/min. Calculate the Reynold's number and decide whether the flow is laminar or turbulent.

2. What is the dynamic viscosity of the oil if you know that the kinematic viscosity is 46 cSt.

3. What is the dynamic viscosity of the VG 100 hydraulic oil.

4. What is the best hydraulic oil grade to be used in Palestine?

5. What is the percentage of reduction in oil viscosity if the temperature of the system is increases from 40 to 100 ^{o}C knowing that the system working pressure is 700 bars.

6. What is the change of volume if a 20 litre hydraulic system is heated from 40 to 100 ^{o}C.

7. Make a comparison between foaming and vapour pressure.

8. If the working pressure is 200 bars and the velocity of oil is 3 m/s, select a range of pipes adequate for this system.

9. Draw the best configurations for flexible hose installation in the different positions.

9.5 References

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Chapter 10

Maintenance of Fluid Power Systems

10.1 Introduction

Prevention is better than cure. This is the best sentence to begin with before talking about maintenance of mechanical systems in general especially fluid power systems. Most of working companies spend a lot of money training their maintenance personnel to troubleshoot a hydraulic system. Although, they should focus on preventing system failure to spend less time and money on troubleshooting a hydraulic or pneumatic system. In the following sections, a detailed discussion about maintenance and troubleshooting of both hydraulic and pneumatic systems will be presented.

10.2 Maintenance of hydraulic systems

10.2.1 Maintenance categories

Maintenance of hydraulic systems can be focused in the following categories:

1. *High temperature of oil:* High temperature influences deeply the properties of oil, like viscosity which reduces the efficiency of the system. To prevent that, a fluid temperature alarm should be installed in the system and all high temperature indications must be investigated and rectified immediately.

   Excess heat of oil can be caused by the following:

- Wear of the pump.
- Oil contamination.
- The air flow through the oil cooler (heat exchanger) is too low .
- The oil level is low.

2. *Cavitation:* Occurs when the volume of fluid demanded by any part of a hydraulic circuit exceeds the volume of fluid being supplied.

3. *Damage of seals and rings:* Cylinders have seals and rings that can be damaged by excess pressure and contaminants in the fluid. to avoid that make sure that the hydraulic fluid is clean. A cylinder is designed to take loads along its axis only. Side loads can decrease cylinder's life by causing excess wear on seals and rod.

4. *Contamination:* Contamination have two types:

   - Particulates: Such as dirt, sand, metal or rubber wear particles that come from internal wear during maintenance, attachment changes and machine operation.
   - Chemicals: They come from oil, water and air.

   Fluid contamination damages hydraulic system in two ways: the first is that it reduces system efficiency. Efficiency losses usually occur slowly and can reach (20 %) before the operator detects a loss in performance. These invisible efficiency losses also can increase fuel consumption. Contamination also accelerates component wear, where (75-85 %) of hydraulic pump, motor, cylinder and valve failures can be traced to contamination. Contamination can be avoided by the following ways:

   - Oil storage and transfer by using transfer filter cart.
   - Change filters carefully.
   - Carefully hose assembly and storage.
   - Avoiding mistakes in changing filters.

10.2.2 Recommended Maintenance technique

The following steps can be followed to apply the best maintenance technique:

(a) Perform daily inspections.

(b) Writing maintenance reports.

(c) Inspection of valves and other parts.

(d) Watch temperature and pressure gauges.

(e) Keep hydraulic tank filled.

To ensure a long life and reliability of the hydraulic systems, the most important maintenance recommendations include:

1. Keep hydraulic fluids cool. The bulk oil temperature at the exterior of the reservoir should not exceed 60^oC and the exterior of all components must be kept clean to ensure that no hot spots develop as a result of accumulated dust and dirt.

2. Keep hydraulic fluids dry. Water content generally should never exceed 1000 ppm (0.1%) in hydraulic systems using mineral base or synthetic fluids.

3. Repair fluid leaks immediately. If oil can escape, dirt, dust and air can re-enter the system. Keep in mind that an external leak of one drop of oil per second is equal to 1600 litres in a 12 month period.

4. Keep hydraulic fluid clean. It is known that 75 to 80% of hydraulic component failures are caused by fluid contamination with dirt, water, wear particles and other foreign material.

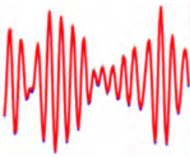
5. Establish an effective oil analysis program. The fluid used in a hydraulic system is a critical component of that system and its condition should be monitored as part of an effective maintenance and reliability program.

10.2.3 Problems, possible causes and remedies

Problem 1: Pump not pumping

Table 10.1 shows the different causes and remedies of the problem when the pump is not pumping any fluid. The first reason to check is the pump's direction of rotation as it may rotate in wrong direction. If the pump is connected through a gearbox it is required to reverse the direction of gear rotation but if it is connected to a 3-phase electric motor, then just exchanging the connection of two phase wires will reverse the direction of rotation. Another reason for not pumping is having a clogged intake for the pump. This can be caused by the cumulation of dirt in the intake pipe and in this case it possible to clean or change the pipe. There could be a non-return (check) valve at the intake pipe, any blockage or damage to this valve will cause closure to the suction line, thus, replace or remove this valve. Low level oil in the tank due to leakage can be one of the non-pumping reasons. Tank should include at least twice the volume of the system's oil quantity, always check oil level and add if needed. Another problem that causes the pump to stop pumping is the leakage of air into the suction pipe. This is a serious problem and can cause severe damage to the pump due to cavitation. The indicator of having this problem is hearing high noise for the pump during operation. It is necessary here to stop the pump directly and check the intake pipe to solve leakage. Otherwise, this will cause damage to the pump's parts that costs a huge amount of money to be repaired. Low speed of the pump can be one of reasons for not pumping. In this case, the speed of the prime mover should be checked and increases, if the prime mover is an internal combustion engine it will be enough to increase the IC engine speed but in the case of electric motors it could be required to replace the motor with a higher speed one. Sometimes the heaviness or the high viscosity of the used hydraulic oil can be a reason for not pumping. When making the hydraulic design, one should select the proper oil for the proper machine and the proper area. For example oils with a viscosity grade of VG10, VG22 and VG32 can be selected for cold and medium conditions countries while the oils VG46 and VG68 are selected for warm and hot countries, VG100 is usually used for very high temperature condition like internal combustion engines.

Table 10.1: *Problem: Pump not pumping*

Possible cause	**Remedy**
Pump rotates in wrong direction In case of 3 phase prime mover	Reverse direction of rotation Check 3 phase connections
Clogged intake 	Check pipe from tank to pump 
Low oil level Best volume	Fill to adequate level Twice the system volume
Air leak intake 	Check noise of pump 
Pump speed too low	Check driving motor speed
Oil too heavy	Check viscosity and replace oil VG22, VG32, VG46, VG68, VG100

Problem 2: Noisy pump

Table 10.2 lists the reasons and solutions for the problem of noise induced in the pump during working. If air leaks into the suction pipe it causes bubbles that reach to the pump inlet parts causing pitting and corrosion on these parts. In this situation the maintenance team should check the oil level in the reservoir and refill it if low besides that they should also check the intake pipe if contains holes or loose connections that allow air to leak into the pipe. Pump need to be checked periodically to fasten its screws and tighten its loose parts. gaskets of the pump are subject to dry or tear during repeated operation, thus, they need to be checked and replaced from time to time. Bad mounts can cause noisy pump.

Wear and erosion happens to moving metallic parts in the hydraulic system produces small metallic chips that contaminate the oil. These metallic chips swim through the pipes till reaching the pump and then they can cause serious damage or blockage to the pump. If the pump is vane pump, these chips can clog between the vanes and the casing sticking the vanes or scratching the casing. If the pump is a reciprocating piston pump, these chips can enter between the moving pistons and their cylinders blocking them from motion or scratching the cylinders liners. In both cases the pump will produce high noise and there will be a need to maintain the pump and repair the spoiled parts.

Dirty of small size filter or strainer and the inlet of the pump can prevent oil from flowing with enough quantity to the pump, this makes the delivery flow rate of the pump more than the intake flow rate causing negative pressure at the inlet leading to cavitation and high noise for the pump. The speed of the prime mover should be adequate to the pump running possibilities, otherwise, the pump will run at a higher speed than it is designed for and will cause high noise. The motor speed should be reduced to a proper speed in this case. The intake pipe should be installed higher than the pump while the return line should pour oil at the oil level and not above to avoid producing bubbles and turbulence.

Table 10.2: *Problem: Noisy pump*

Possible cause	**Remedy**
Air leaking into system	Fill oil reservoir and check intake pipe
Air bubbles in intake oil	Check level of intake pipe
Pump cavitation	Look at cavitation problem next section
Loose pump parts	Tighten parts and check gaskets
Stuck pump vanes and pistons	Check for metallic contaminants
Dirty filter or strainer	Clean or replace
Small size filter	Check filter size
Pump running too fast	Determine appropriate speed
Return line above fluid level	Extend return line

Problem 3: Pump cavitation

Table 10.3 depicts the cavitation problem which is considered one of the most dangerous problems occur in hydraulic systems. Cavitation is a phenomenon in which the static pressure of a liquid reduces to below the liquid's vapour pressure, leading to the formation of small vapour-filled cavities in the liquid. When the filter or strainer is small or clogged for some reason it causes cavitation. The solution here is to replace the filter with a proper one. The suction line is another main reason where diameter of the intake should be larger than that of the delivery line. The suction pipe should checked for bends and its layout be modified to be higher than the pump's level. If the fluid is too cold or unsuitable for the operation of the region of operation then its viscosity will prevent it from flowing properly into the suction line causing cavitation. Each tank or reservoir contains an air breather to equate atmospheric pressure on the oil surface. If this breather is clogged or closed for any reason it should be repaired or replaced. If there is a restrictor or flow control valve on the suction line it should be checked and opened to allow fluid to flow freely to the pump. The speed of pump is very important that if it runs faster to produce more flow rate than the incoming one this will cause cavitation. Speed of the pump should be reduced in this situation. Pump should be mounted lower than the oil level to allow fluid to flow freely by gravity to the pump.

Problem 4: System overheat

Table 10.4 discusses the problem of overheat occurs in hydraulic systems. High viscosity grade oils can cause over heat to the system because it induces high resistance to motion and needs higher pressure to be pushed into piping system. In this case oil should be replaced. Loose parts can cause internal leakage of fluid and this leads to heating oils and components. All parts should be checked and repaired. The pressure relief valve is used to set the system operating pressure. If the relief valve is set to high pressure this will increase pressure in the system causing higher temperature of oil and overheat of components. Higher friction between moving parts and corrugated pipelines can cause overheat to the system. The oil cooler should be checked from time to time to avoid blockage and keep cooling continuously. Oil level in the tank is another reason for overheat, always check oil level. a proper cooling system can save the hydraulic system

Table 10.3: *Problem: Pump cavitation*

Possible cause	**Remedy**
Clogged or too small strainer	Clean or renew
Bore of suction line too small	Fit larger bore pipes
Too many bends in suction line	Modify pipe layout
Suction line too long	Reduce length or fit larger bore pipes
Fluid too cold	Heat fluid to recommended temperature
Unsuitable fluid	Replace with correct fluid
Air breather blocked	Clean or replace element
Restriction in suction line	Open or modify valves
Pump running too fast	Reduce to recommended speed
Pump mounted above oil level	Modify pump installation

from damage. Sometimes designers use air cooling by fan and radiator to reduce oil temperature and other times they use water cooling by passing cold water stream to exchange heat with oil lines.

Table 10.4: *Problem: System overheat*

Possible cause	**Remedy**
Oil viscosity too high	Replace with lower viscosity
Internal leakage	Check for wear and loose parts
High discharge pressure	Proper setting of relief valve
High friction	Check loose pump parts
Clogged oil cooler	Clean cooler
Low oil level	Fill to proper oil level

Problem 5: Low or no pressure in the system
(Table 10.5 handles the causes and remedies of the lack of pressure in the hydraulic system. The first option to check for low pressure in the system is the pressure relief valve and to make sure that it is set to the proper pressure and make sure it is not stuck or blocked. Leakage in the system can cause pressure reduction, so repair leakage as soon as it is noticed. If the pump is broken or has damaged parts it will lose its ability to afford

pressure. The regulation of the control valves also has great influence on the pressure in the system.

Table 10.5: *Problem: Low or no pressure in the system*

Possible cause	**Remedy**
Low setting of relief valve	Reset relief valve
Relief valve stuck open	Clean or replace
Leak in system	Check pressure drop
Broken pump parts	Replace broken parts
Incorrect control valve	Replace parts

Problem 6: Cylinder slow or not moving

Table 10.6 deals with the problems of cylinder motion. If the cylinder does not move this can be caused by a failure in the directional control valve, one should check if electrical current reaches to the solenoid or not. Sometimes the actuator stops or moves in a slow motion due to lack of pressure in the system. In this case mechanic should return back to lack of pressure reasons and check accordingly. This problem arises in special cases when there is defect either in the hydraulic pipe or in the actuator body itself. Calculations may show that the operating actuator does not fit to the exposed load leading to incapability of the piton to move. air leakage into the system causes compressibility in the fluid that absorbs motion given to the actuator without allowing the actuator to move.

Table 10.6: *Problem: Hydraulic cylinder slow or not moving*

Possible cause	**Remedy**
Directional valve failure	Check power input to solenoids
Insufficient pressure supplied	Check system pressure
Hydraulic line problem	Check for dented or crushed hoses
Defective actuator	Check piston rod bent or dented cylinder
Load exceeds capacity of actuator	Check system pressure and size of piston
Hydraulic circuit error	Check valve backwards
Air in system	Bleed air from system
Defective or worn pump	repair or replace

10.3 Maintenance of pneumatic systems

10.3.1 Maintenance parts of pneumatic systems

The main parts used to maintain pneumatic systems can be listed as follows:

1. **Coolers**
   As temperature of air increases during compression, removing heat during compression reduces the work required to raise the pressure of the air. Heat can be removed from the air compressor to the surrounding air or to water. Air-cooled compressors (Figure 10.1) pass the hot lubricating oil from the compressor and compressed air through finned-tube heat exchangers and force ambient air across the heat exchangers using a cooling air fan.

Figure 10.1: *Pneumatic compressor aftercooler (Courtesy of Xinxiang Zhenhua Radiator Co., Ltd.)*

   Cooling fan horsepower is typically about 5% of the power of the compressor motor. Water-cooled compressors use water-to-air heat exchangers to remove heat from the lubricating oil and compressed air. In many applications, this heat is eventually rejected to the atmosphere by a cooling tower. Increasing cooling by decreasing the temperature of the cooling air or water improves compressor efficiency and output capacity.

2. **Dryers**

   The two most common types of dryers for removing moisture from compressed air lines are refrigerated dryers and desiccant dryers. Dryers (Figure 10.2) are typically sized to handle the peak air compressor air flow. As compressed air cools, water vapour can condense out of the air and should be removed from the compressed air system through drains.

Figure 10.2: *Pneumatic 2-stage air filter dryer (Courtesy of Rockwood)*

3. **Filter, Regulator and Lubricator** (FRL)

   The filter removes particulates entrained in the compressed air and may have a trap or drain at the bottom. Regulator reduces downstream air pressure. Regulators have pressure gauges and valves to adjust the downstream pressure. Lubricator looks like the filter, but have a clear bubble or screw assembly on top for adding oil.

Figure 10.3: *Pneumatic Filter Regulator Lubricator (Courtesy of Pneumatic Plus)*

4. **Condensate drains**

   Condensate drains (Figure 10.4) should be located after the cooler, underneath the receiver tank, at low points in the system, after filters, regulators and other devices that result in a large pressure drop.

Figure 10.4: *Pneumatic Condensate drain (Courtesy of SMC)*

10.3.2 Air treatment

Figure 10.5 shows a typical drawing of air filter indicating the different components required to exist in any air filter.

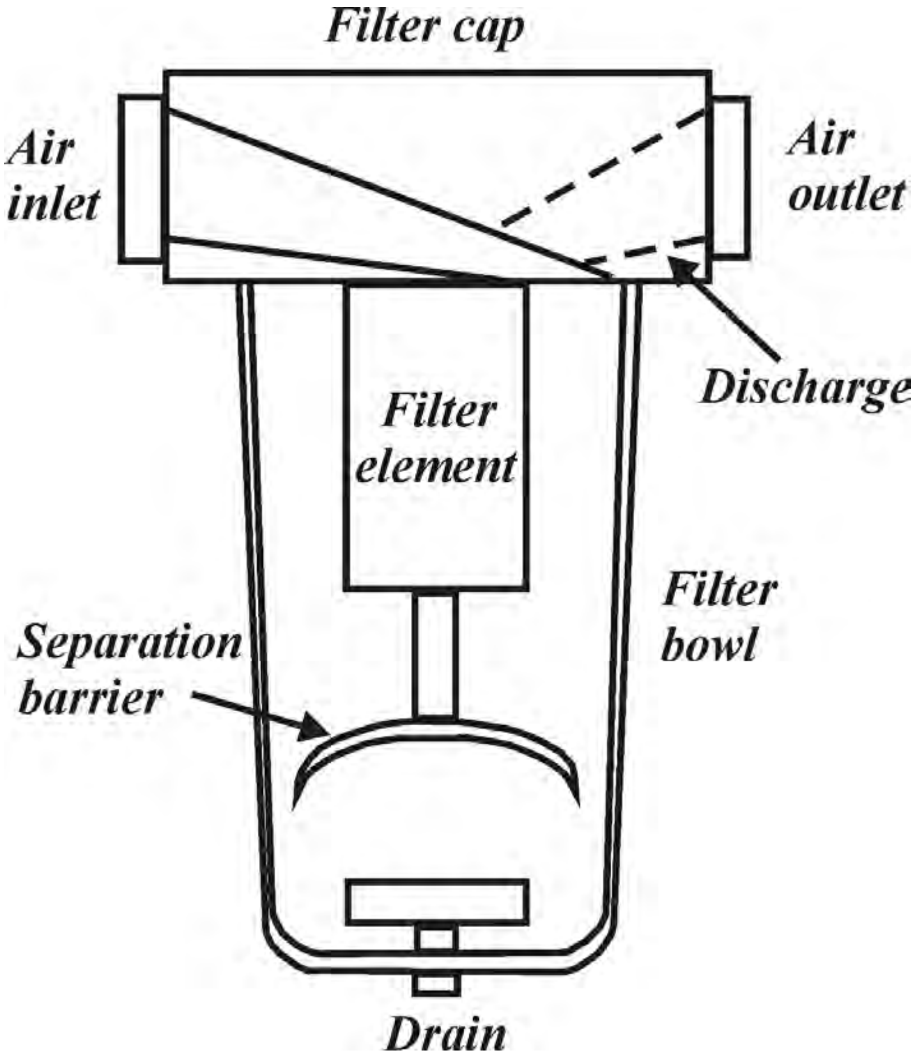


Figure 10.5: *Typical drawing of air filter*

Air filters can be classified according to the filter element as follows:

- **Fibreglass filters**
  This throwaway air filter is the most common type. Layered fibreglass fibres are laid over each other to form the filter media and typically are reinforced with a metal grating that supports the fibreglass to prevent failure and collapse

- **Polyester filters**
  These filters are similar to fibreglass filters but typically have a higher resistance to airflow and a superior dust-stopping ability

10.4 Problems

1. What are the main reasons for overheat in the hydraulic system?
2. What is cavitation and how doe it occur?
3. specify one way to solve solid metal contamination in hydraulic oil?
4. Discuss the consequences of water existing in hydraulic oil.
5. Indicate the steps of inspection that a mechanic should follow when the flow rate of the pump is very weak.
6. What does the noise of the pump depend on?
7. Make a dynamic model for a vane pump and show how to calculate the resonance frequency of its noise.
8. How does cavitation influence pump end of life and how is it solved?
9. What are the main parts changed when the piston pump is repaired?
10. Discuss the influence of overheat on the hydraulic oil.
11. Make a design for water cooling system to overcome hydraulic oil overheat.
12. How water cooling for hydraulic oil is done.
13. List the reasons and remedies for low pressure in the hydraulic system.
14. When the hydraulic actuator moves slowly, what are the possible reasons and solutions.
15. What are the reasons of overheat in air compressors and what are the type of coolers.
16. What is the design of a dryer and what is it used for?

17. Draw a filter regulator lubricator and explain its function.
18. What are the types of and designs of air treatment filters.

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