ME 5451 – Hydraulics and Pneumatics

Lecture -1

Date: 25-02-2021 Time slot: 10:30-12:00 noon.

Contents

1. Introduction to course - need
2. Organization of syllabus
3. Course plan
4. Resources for study

Course Instructor: Dr. A. Siddharthan

Prerequisite knowledge

Fluid Mechanics – Bernoulli's Theorem, Pascal’s law, friction Losses, Pumps, efficiencies

Gas Laws-

Hydraulics- Fluid Power Application

Look around and recollect components where oil/ air used for force manipulation!

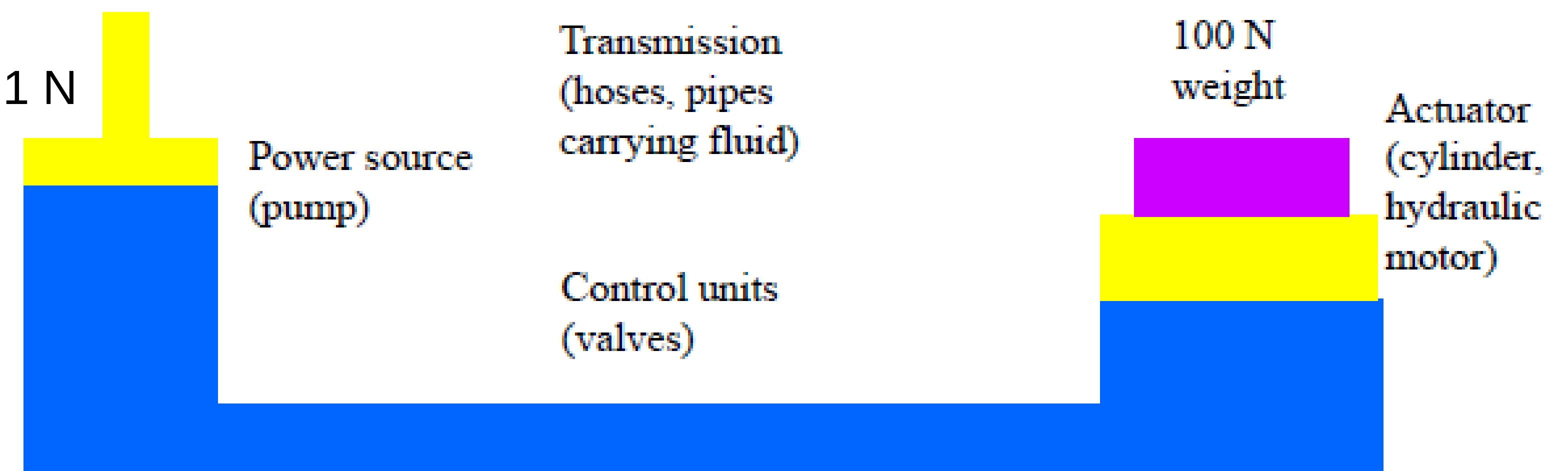
- Hydraulic lift in automobile service station and lorries to unload.
- Shock absorbers of automobiles
- Door closer in office and in bus/train
- Brakes, ABS Braking
- Power Steering
- Hydraulic Press, Injection moulding etc
- Concrete Breakers ....

Difference between hydraulic machine in Fluid Machinery and Fluid Power based on oil and gas? Work done is Not based on kinetic energy, rather pressure !

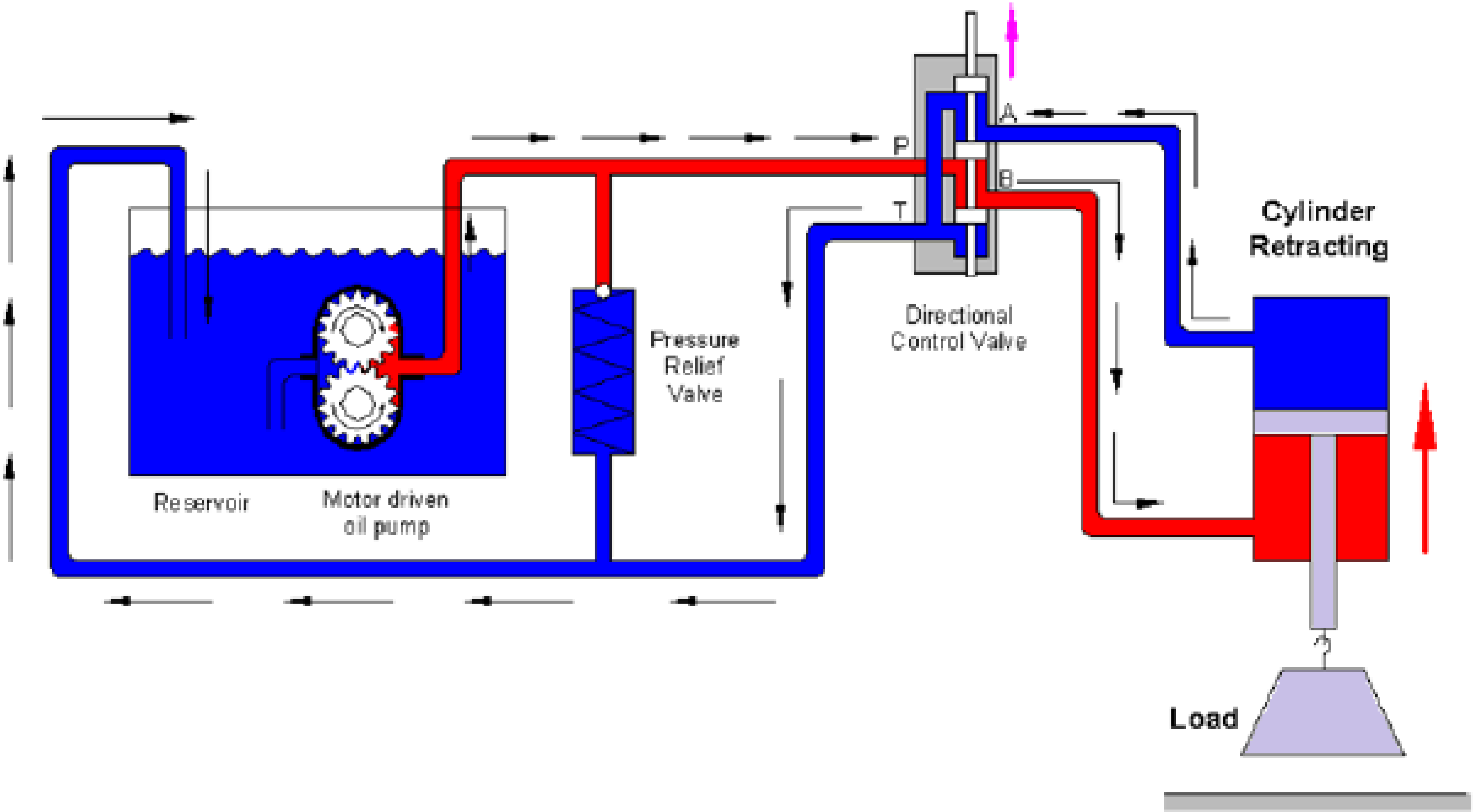
Pascal: “Pressure applied on a confined fluid is transmitted in all directions with equal force on equal areas”.

Multiplication of Force

40,000 Ton Forging Press



Fork Lifts



P
A
B
T
Cylinder Retracting
Directional Control Valve
Pressure Relief Valve
Reservoir
Motor driven oil pump
Load

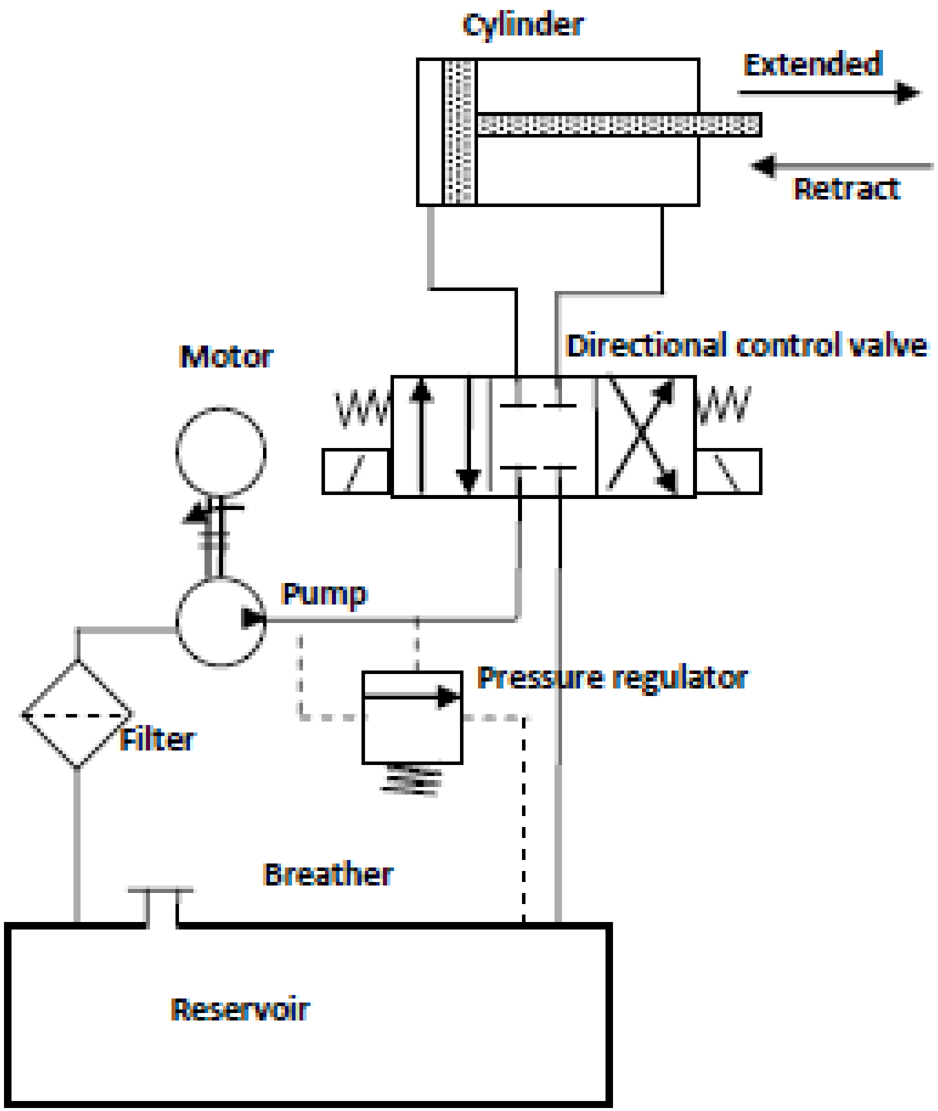
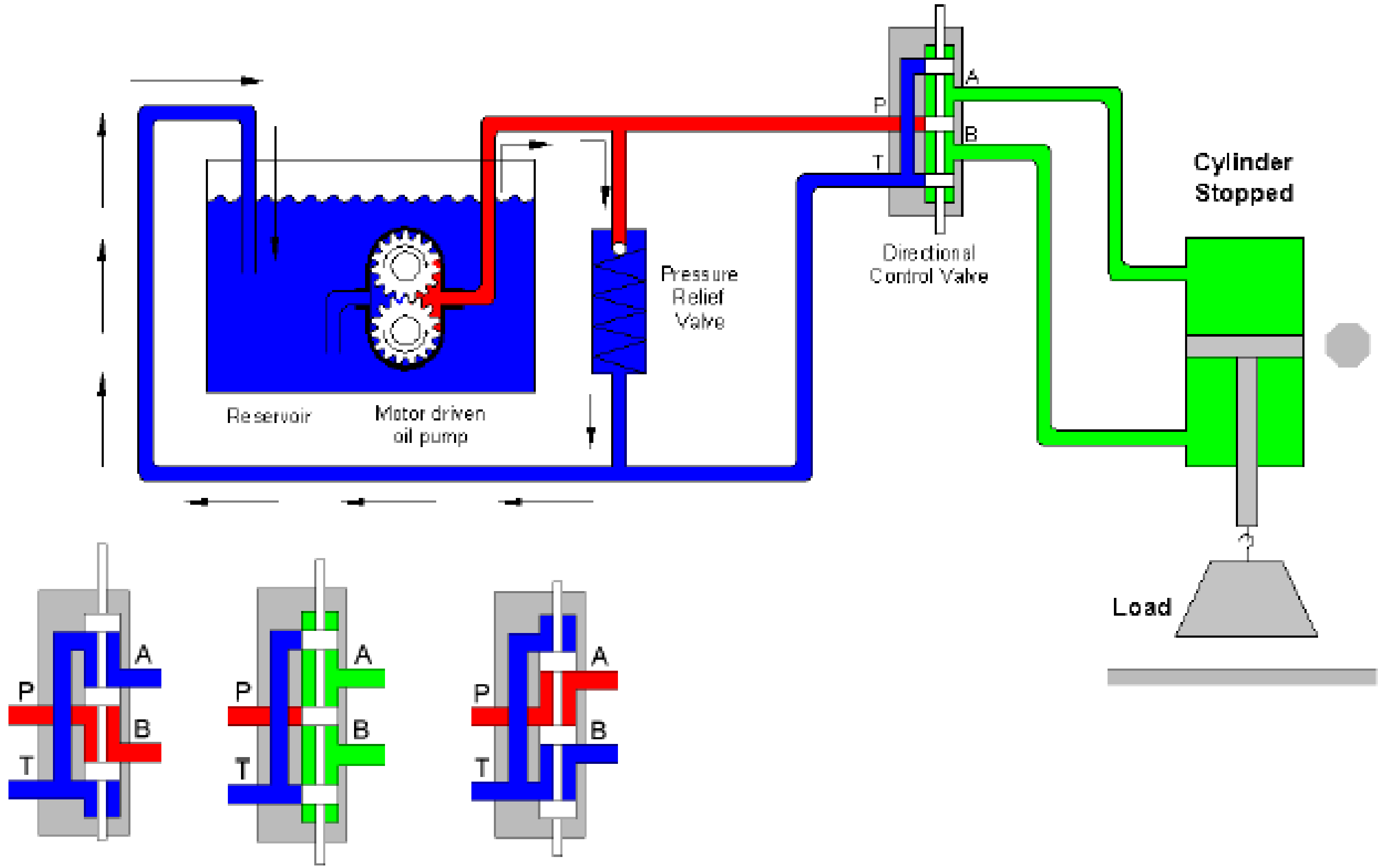
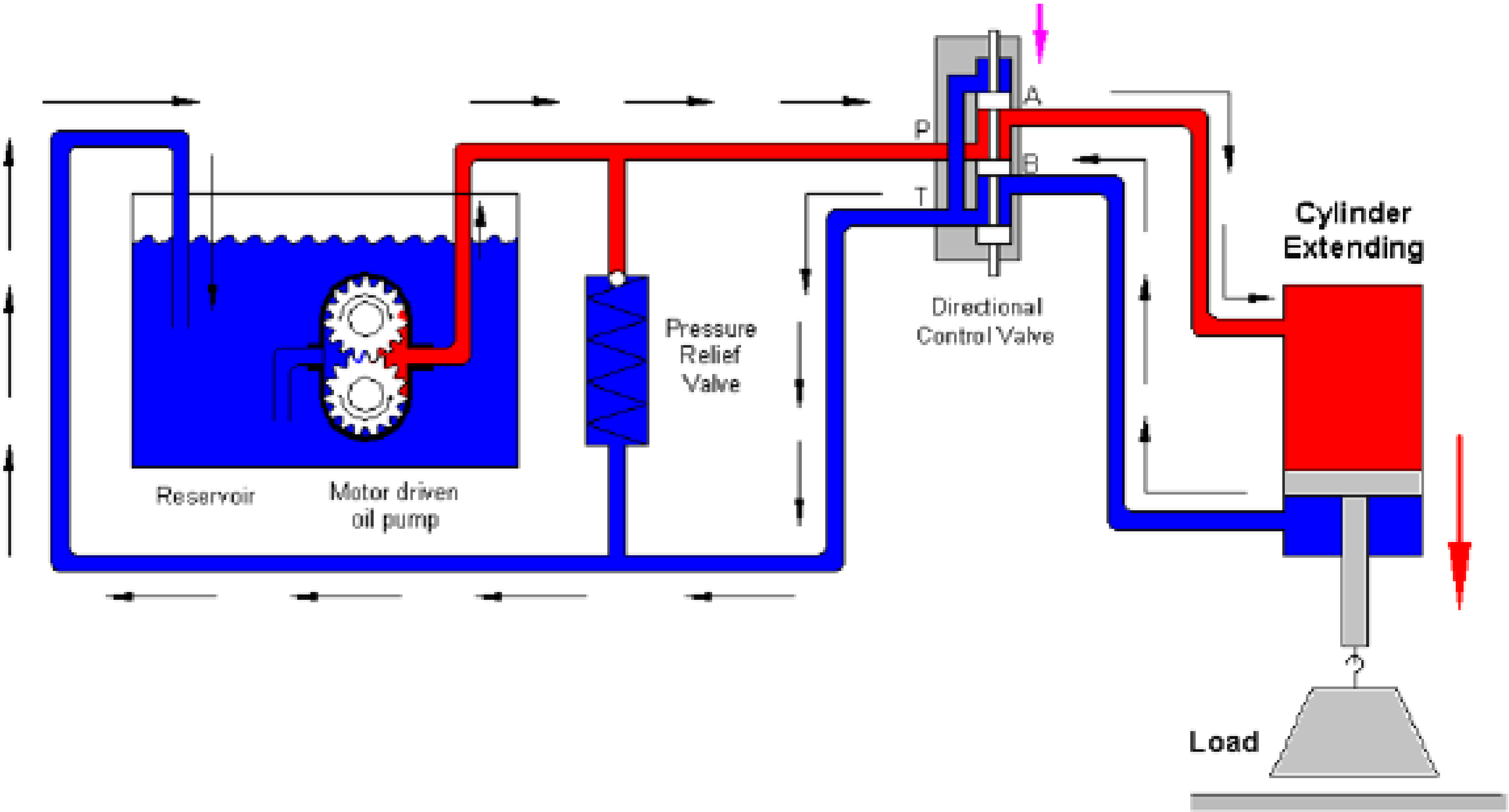


Figure 1.2 Components of a hydraulic system (shown using symbols).



A
P
B
T
Directional Control Valve
Cylinder Stopped
Pressure Relief Valve
Reservoir
Motor driven oil pump
Load
A
P
B
T
A
P
B
T
A
P
B
T



P
A
B
T
Cylinder Extending
Directional Control Valve
Pressure Relief Valve
Reservoir
Motor driven oil pump
Load

History of development of Fluid Power

1800	Branch	Power transmission by water	Hydraulic Press Leather cup Seal technology
1900			
1910	Constaninesco		First ideas of Alternating Flow Hydraulics (AFS)
1920	Hele Shaw Beachan		
1930	Thomas Dowty		High Pressure Vane pumps Synthetic rubber seals
1940			Electro hydraulic servos & rubber sac accumulators
1950	MIT Labs	Hydraulics & Pneumatics	Hot gas servos
1960	Growing research effort throughout Industry, universities & Technical colleges. Power Station.		Air fluidics Digital hydraulics, oil fluidics
1980			oil hydraulics Micro process and computer controlled
2000	Rapid growth in Electronics and Manufacturing		Hybrid Systems Energy Saving System
2010	Technology		Wide application of proportional valves.
	Nano Technology		Micro-hydraulic system Nano Fluidics.

Advantages and disadvantages of Fluid Power Transmission over Mechanical and Electrical Power Transmissions are briefly as follows.

Advantage

1. Material medium - heat dissipation,
2. Inertia to turque ratio

3. Mechanically Stiff
   ↑ (Oil hydraulic)
4. High attainable speed response.
5. The same medium can be used for both drive as well as control.

Disadvantages of fluid power

- Leaks!
- Fire hazard (flash point approx. 220 F) → **Water hydraulics**
- Difficulty in precise control
  - inherently nonlinear
  - fluid compressibility
  - control accuracy and efficiency

  → **• Electrohydraulics**
  • Advanced control
- Prone to contamination (reliability)
- Efficiency (compared to purely mechanical)
  - Poor efficiency using current control methods → **New control principles**
- Noise → noise cancellation, better hose design

Table 1.3 Comparison of different power systems</table>

Property	Mechanical	Electrical	Pneumatic	Hydraulic
Input energy source	I C engines Electric motor	I C engines Water/gas turbines	I C engines Pressure tank	I C engines Electric motor Air turbine
Energy transfer element	Levers, gears, shafts	Electrical cables and magnetic field	Pipes and hoses	Pipes and hoses
Energy carrier	Rigid and elastic objects	Flow of electrons	Air	Hydraulic liquids
Power-to-weight ratio	Poor	Fair	Best	Best
Torque/inertia	Poor	Fair	Good	Best
Stiffness	Good	Poor	Fair	Best
Response speed	Fair	Best	Fair	Good
Dirt sensitivity	Best	Best	Fair	Fair
Relative cost	Best	Best	Good	Fair
Control	Fair	Best	Good	Good
Motion type	Mainly rotary	Mainly rotary	Linear or rotary	Linear or rotary

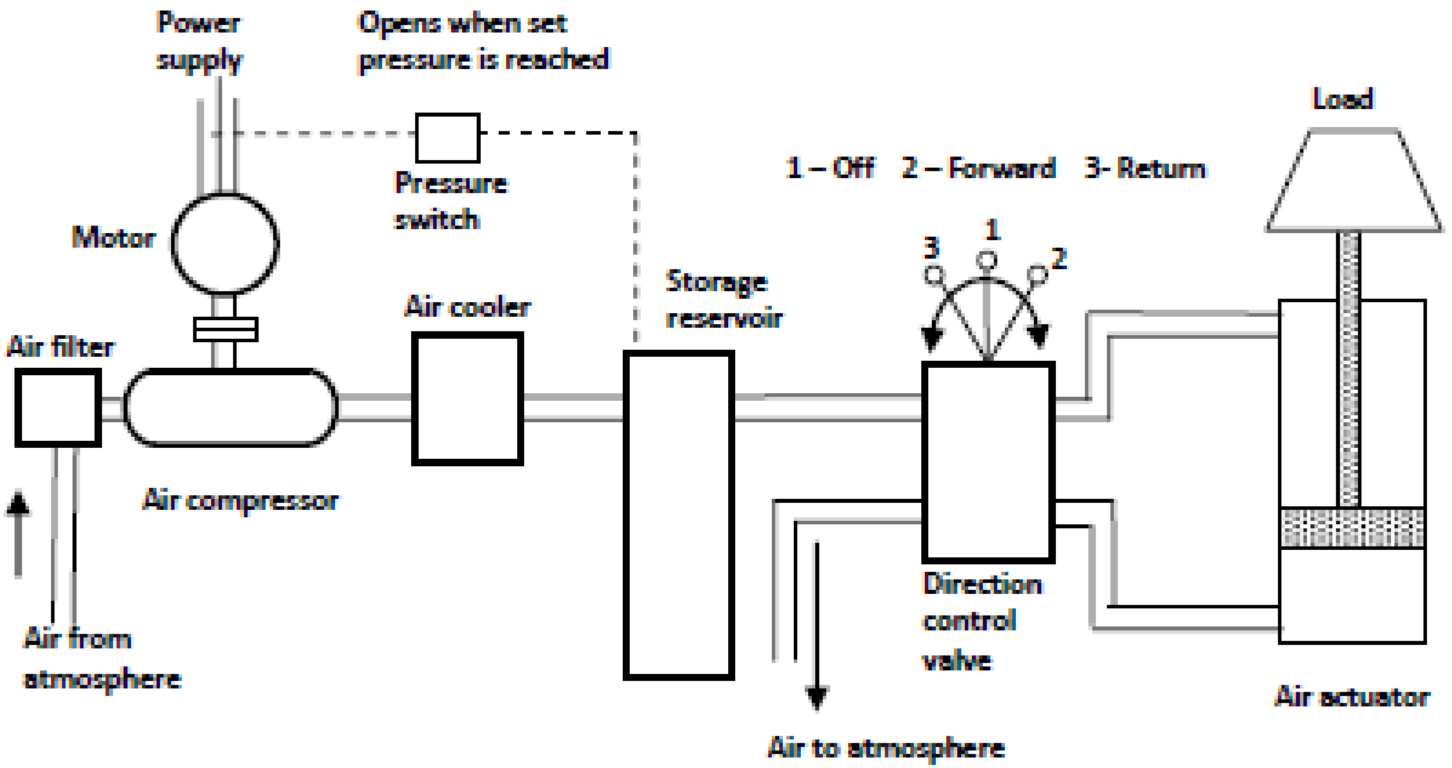


Figure 1.3 Components of a pneumatic system.

Table 1.2 Comparison between a hydraulic and a pneumatic system</table>

S. No.	Hydraulic System	Pneumatic System
1.	It employs a pressurized liquid as a fluid	It employs a compressed gas, usually air, as a fluid
2.	An oil hydraulic system operates at pressures up to 700 bar	A pneumatic system usually operates at 5–10 bar
3.	Generally designed as closed system	Usually designed as open system
4.	The system slows down when leakage occurs	Leakage does not affect the system much
5.	Valve operations are difficult	Valve operations are easy
6.	Heavier in weight	Lighter in weight
7.	Pumps are used to provide pressurized liquids	Compressors are used to provide compressed gases
8.	The system is unsafe to fire hazards	The system is free from fire hazards
9.	Automatic lubrication is provided	Special arrangements for lubrication are needed

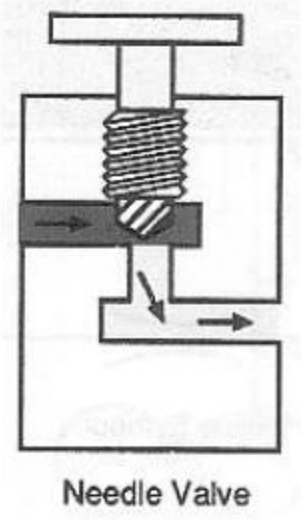
Advantages of a Fluid Power System

- Fluid power systems are simple, easy to operate and can be controlled accurately
- Multiplication and variation of forces
- Multifunction control
- Low-speed torque
- Constant force or torque
- Economical
- Low weight to power ratio
- Infinite Speed control
- Instant reversal of direction of rotation
- System can be halted

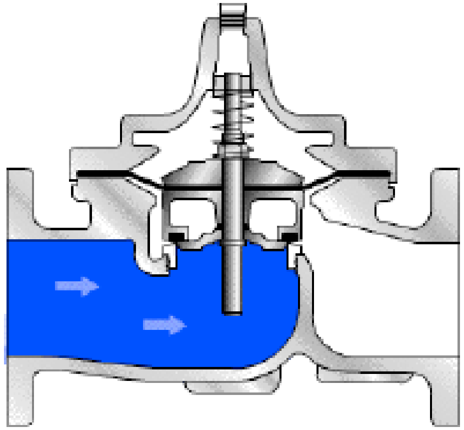
Fluid Power Fundamental

- Pressure → force or torque
- Flow → velocity

Needle Valve

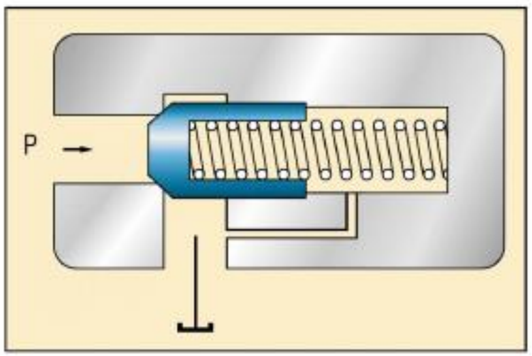


$$Q = C_d A_0 \sqrt{\frac{2}{\rho}(P_1 - P_2)}$$

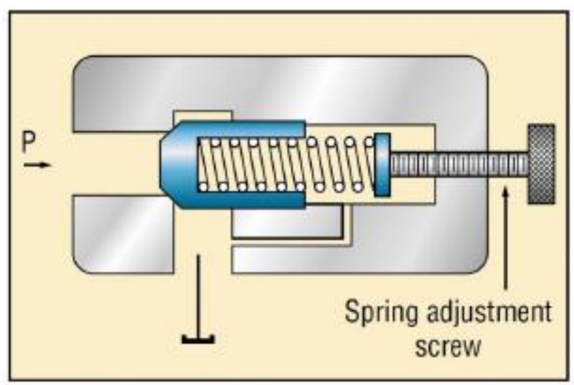


Fluid Power Fundamental

- Pressure → force or torque

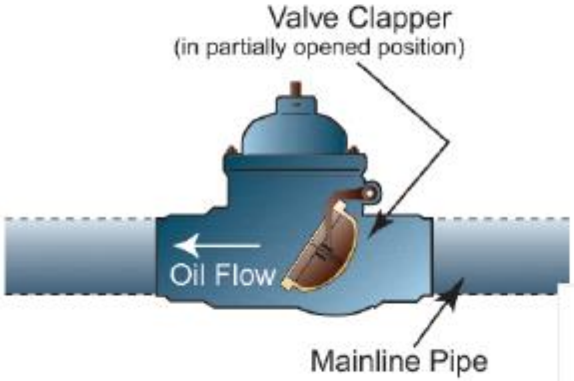


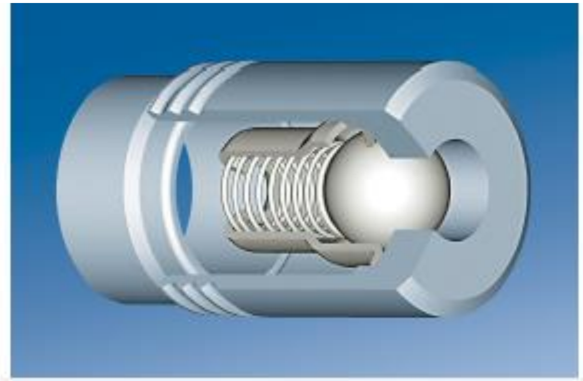
A pressure control valve is used to reduce the amount of pressure in a tank or system of pipes.

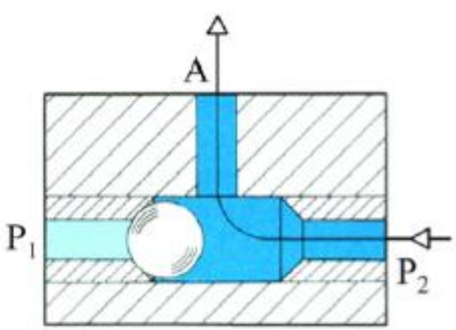


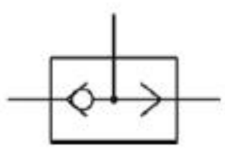
Directional control valves

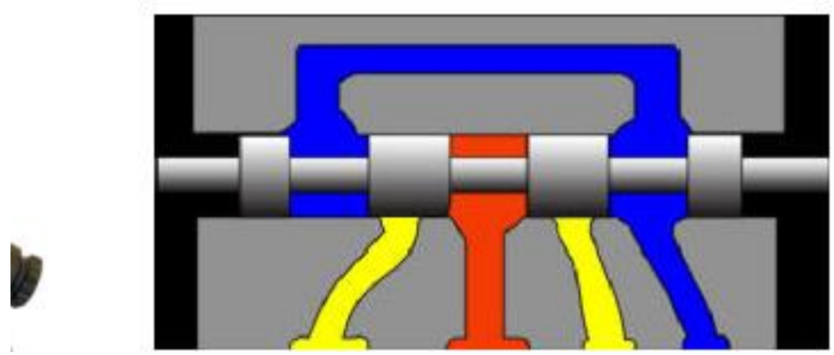
- Check Valves

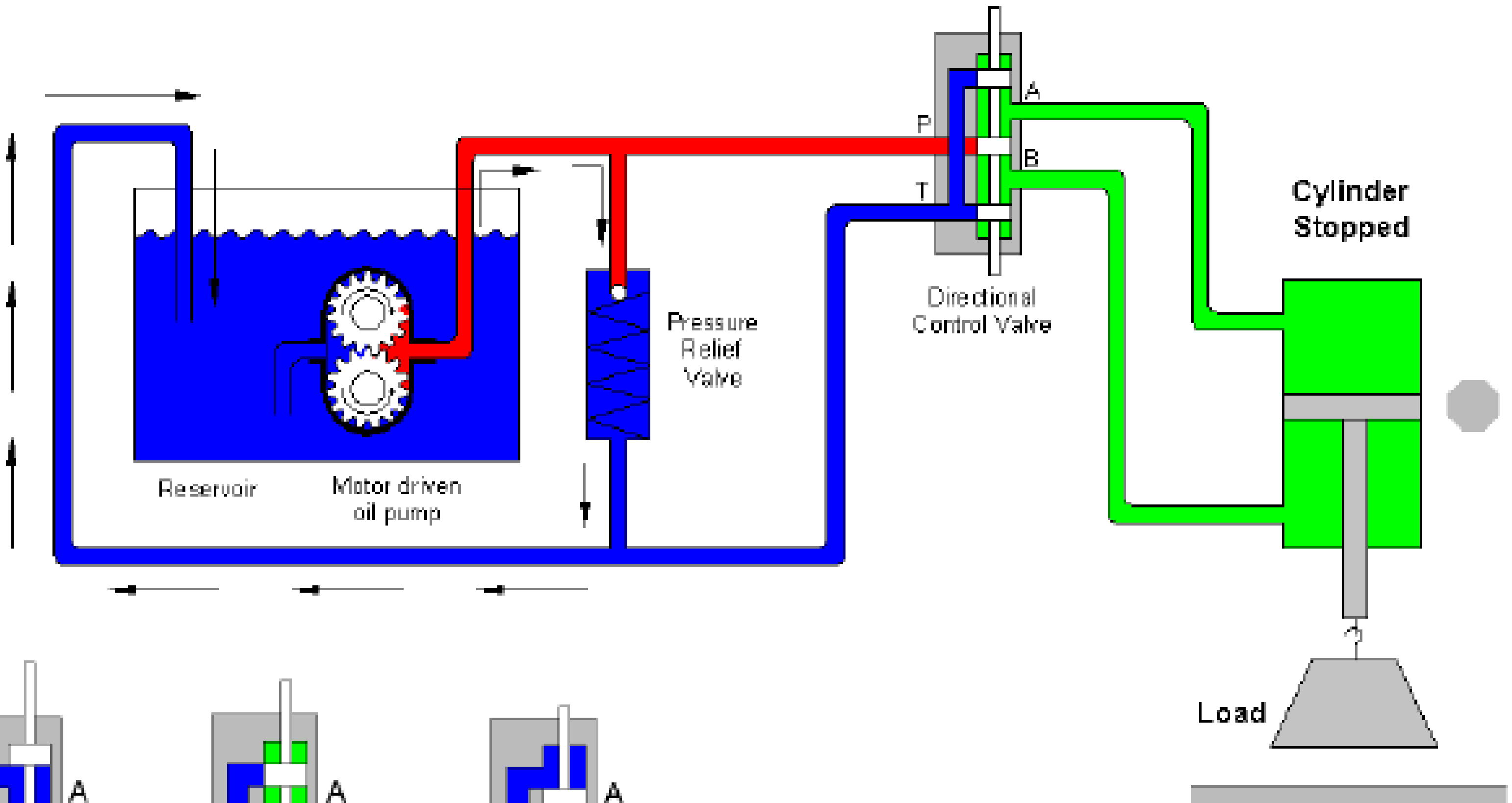


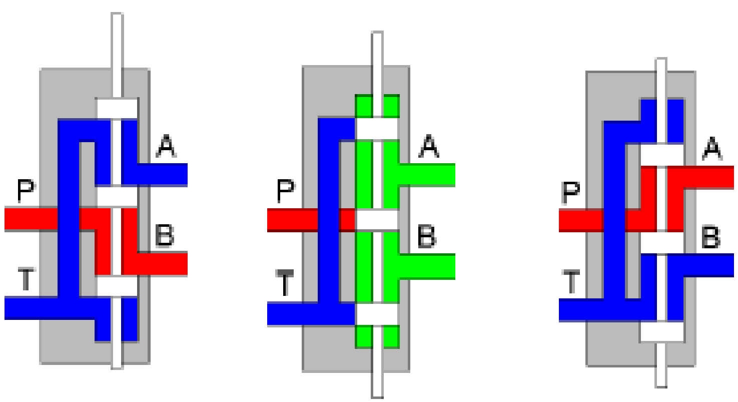












The valve shown has 4 ports and 3 positions so it is designated as a 4/3 directional control valve.

COURSE OUTCOMES

Upon completion of this course, the students will be able to:

- Identify hydraulic and pneumatic components and its symbol and usage.
- Ability to design hydraulic and pneumatic circuits.

RESOURCES

TEXT BOOKS:

1. **Anthony Esposito, “Fluid Power with Applications”, Prentice Hall, 2009.**
2. James A. Sullivan, “Fluid Power Theory and Applications”, Fourth Edition, Prentice Hall, 1997.

NPTEL course Material

1. Fluid Power Control
   https://nptel.ac.in/courses/112/106/112106175/#
2. Fundamentals to Industrial Oil hydraulics and Pneumatics
   https://nptel.ac.in/courses/112/105/112105047/#

REFERENCES

1. Shanmugasundaram.K, “Hydraulic and Pneumatic Controls”. Chand & Co, 2006.
2. Majumdar, S.R., “Oil Hydraulics Systems – Principles and Maintenance”, Tata McGRaw Hill,2001.
3. Majumdar, S.R., “Pneumatic Systems – Principles and Maintenance”, Tata McGRaw Hill,2007.
4. Dudley, A. Pease and John J Pippenger, “Basic Fluid Power”, Prentice Hall, 1987
5. **Srinivasan.R, “Hydraulic and Pneumatic Controls”, Vijay Nicole Imprints, 2008**
6. Joshi.P, “Pneumatic Control”, Wiley India, 2008.
7. **Jagadeesha T, “Pneumatics Concepts, Design and Applications”, Universities Press, 2015**

The piston overlap determines the oil leakage rate. Overlapping is significant for all types of valve. The most favorable overlap is selected in accordance with the application.

1. **Positive switching overlap:** During the reversing procedure, all parts are briefly closed against one another. Hence, switching imparts "pressure peaks" and make hard advance.
2. **Negative switching overlap:** During the reversing procedure, all ports are briefly interconnected. Pressure collapses briefly (load drops down).
3. **Zero overlap:** Edges meet. Important for fast switching, short switching paths.
4. **Pressure advanced opening:** The pump is first of all connected to the power componentand then the power component is discharged into the reservoir.

5. Outlet advanced opening: The outlet of the power component is first discharged to the reservoir before the inlet is connected to the pump.

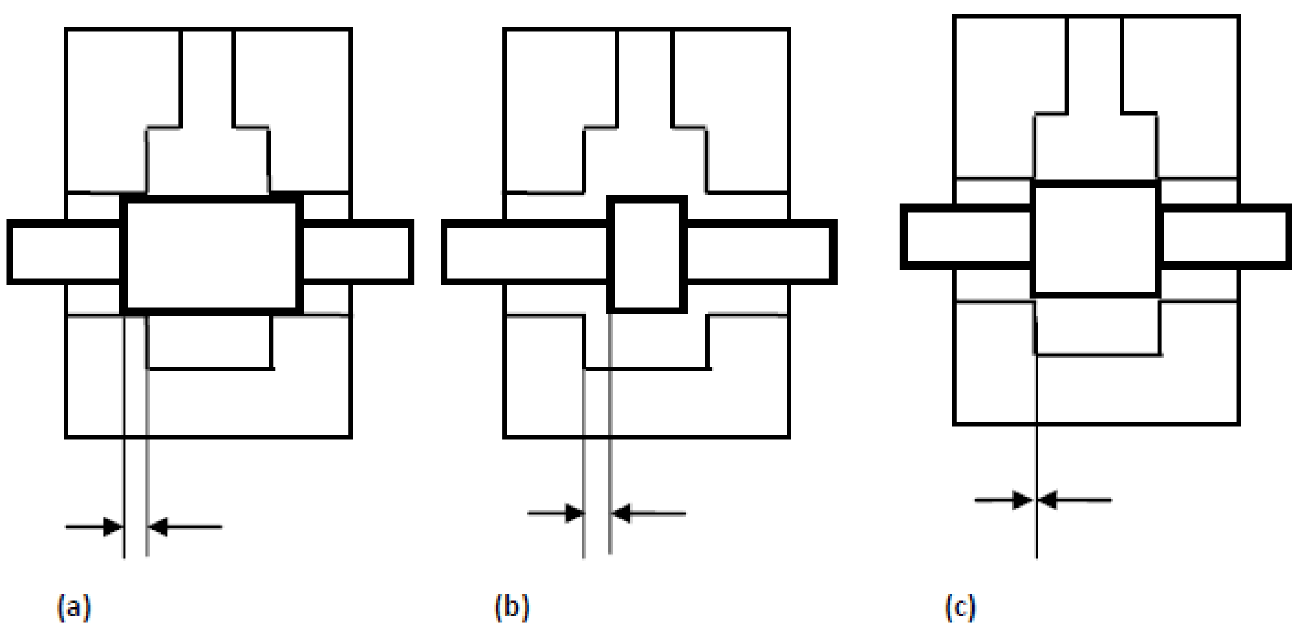


Figure 1.27 Valve overlap: (a) Positive overlap; (b) negative overlap; (c) zero overlap.

1.8 Cartridge Valves

Cartridge valves consist of a valve shell that can be mounted in a standard recess in a valve block or manifold. The machine manufacturer does not have to worry about tolerances of moving spools and poppets because these are taken care by the hydraulic valve manufacturer. This is very advantageous for

batch production and modularized packages or integrated circuits. Cartridge valves eliminate expensive and potentially leaking pipework and connectors.Cartridge valves can be used as follows:

1. Leak-proof direction control valve.
2. Check valve to obtain unidirectional flow.
3. Throttle valve to control and limit the rate of flow.

1.8.1 Poppet-Type Cartridge Valves

In some designs, a poppet fits into cavity and is held in position by a cover or a top plate that contains all pilot connections. Others are designed to fit the standard cavities used by some conventional cartridge valves. Logic elements that have a balanced poppet or spools can be modulated and are largely used as pressure controls. Those with unbalanced poppets are primarily used for switching functions such as directional controls or where the poppet movement can be limited as flow controls.

The principal advantages of poppet-type valves are as follows:

1. A very high flow rate for a relatively small physical size.
2. A positive seal can be obtained.
3. May be extremely rapid acting but can also be easily adopted for soft switching.
4. The shape of the poppet or spool together with its seat can be varied to give different operating characteristics to the valve assembly.

Figure 1.23 shows how a large double-acting cylinder can be controlled using cartridge valves. A small double-solenoid-operated directioncontrol valve feeds pilot pressure signals to four cartridge valves that are coupled in pairs to each end of the double-acting cylinder. One cartridge valve from each pair is permanently connected to the tank drain line and the other to the pump pressure line. In the position drawn, all the four valves are held closed by the pilot pressure signals and the cylinder position is locked. When solenoid A is energized, pilot pressure is maintained on valves 1 and 3 which remain closed. But valves 2 and 4 are released and open under the influence of fluid pressure in the main system. The fluid under pressure, therefore, flows from the pump through the cartridge valve to the piston side of the cylinder and the cylinder extends. The fluid from the rod end of the cylinder flows through valve 4 back to the tank. When solenoid B is energized, cartridge valves 2 and 4 are closed under pilot pressure and valves 1 and 2 are released, causing the cylinder to retract.

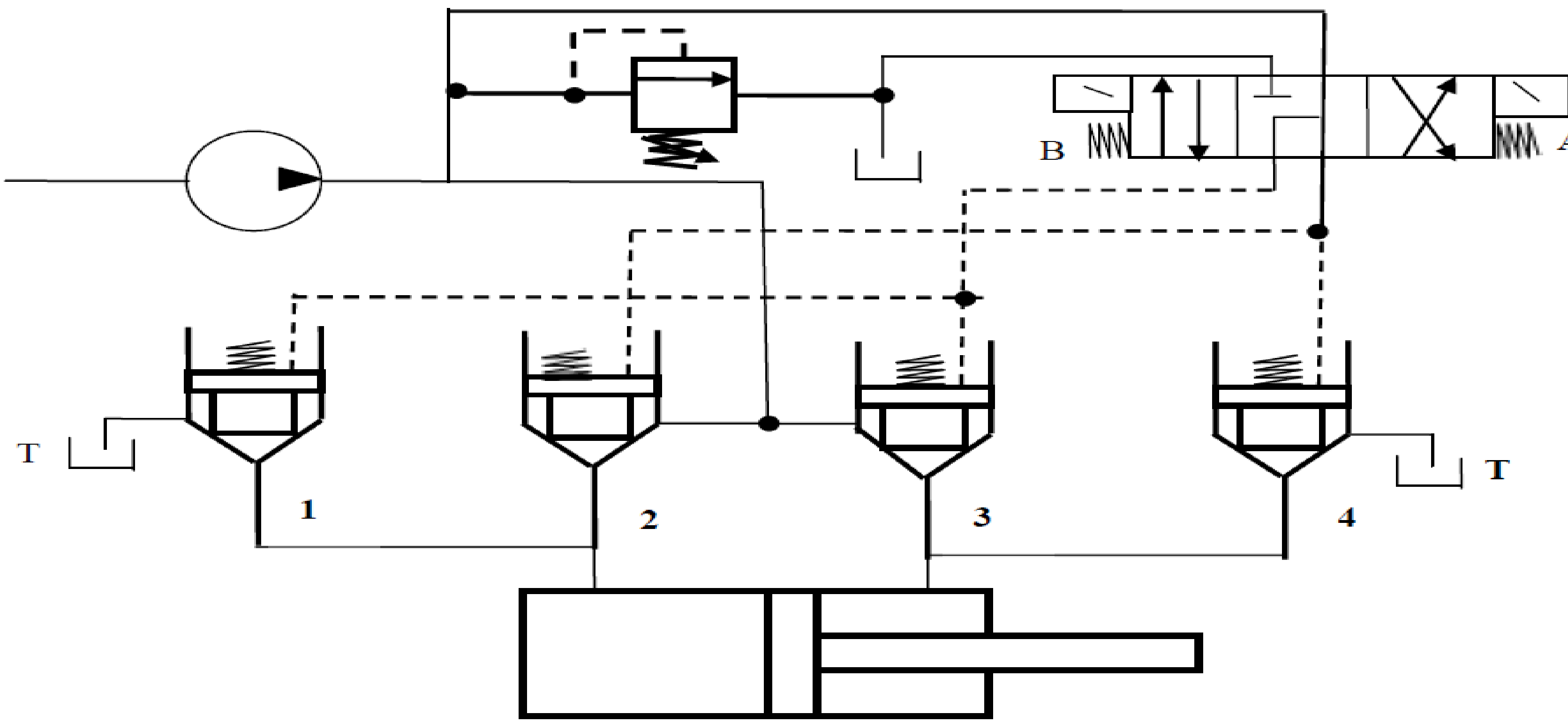


Figure 1.23 Control of a double-acting cylinder using cartridge logic valves.

1.4.1 Force Position Control

The electrical control to the proportional valve normally uses a variable current rather than a variable voltage. If a voltage control system is adopted, any variation in coil resistance caused by temperature change will result in a change in current. This problem is eliminated by using a current control system. It is possible to control a force electrically. By applying the force to a compression spring, its deflection can be controlled. If the spool in a valve (as in Fig. 1.8) is acted on by a spring at one end and a proportional solenoid on the other, the orifice size can be varied along with the control current.

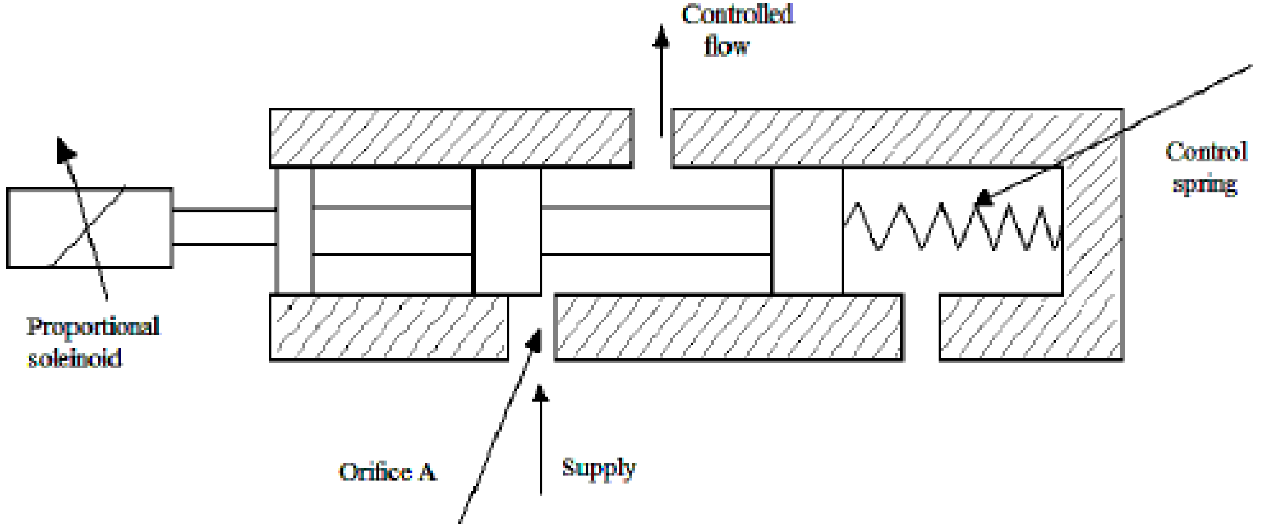
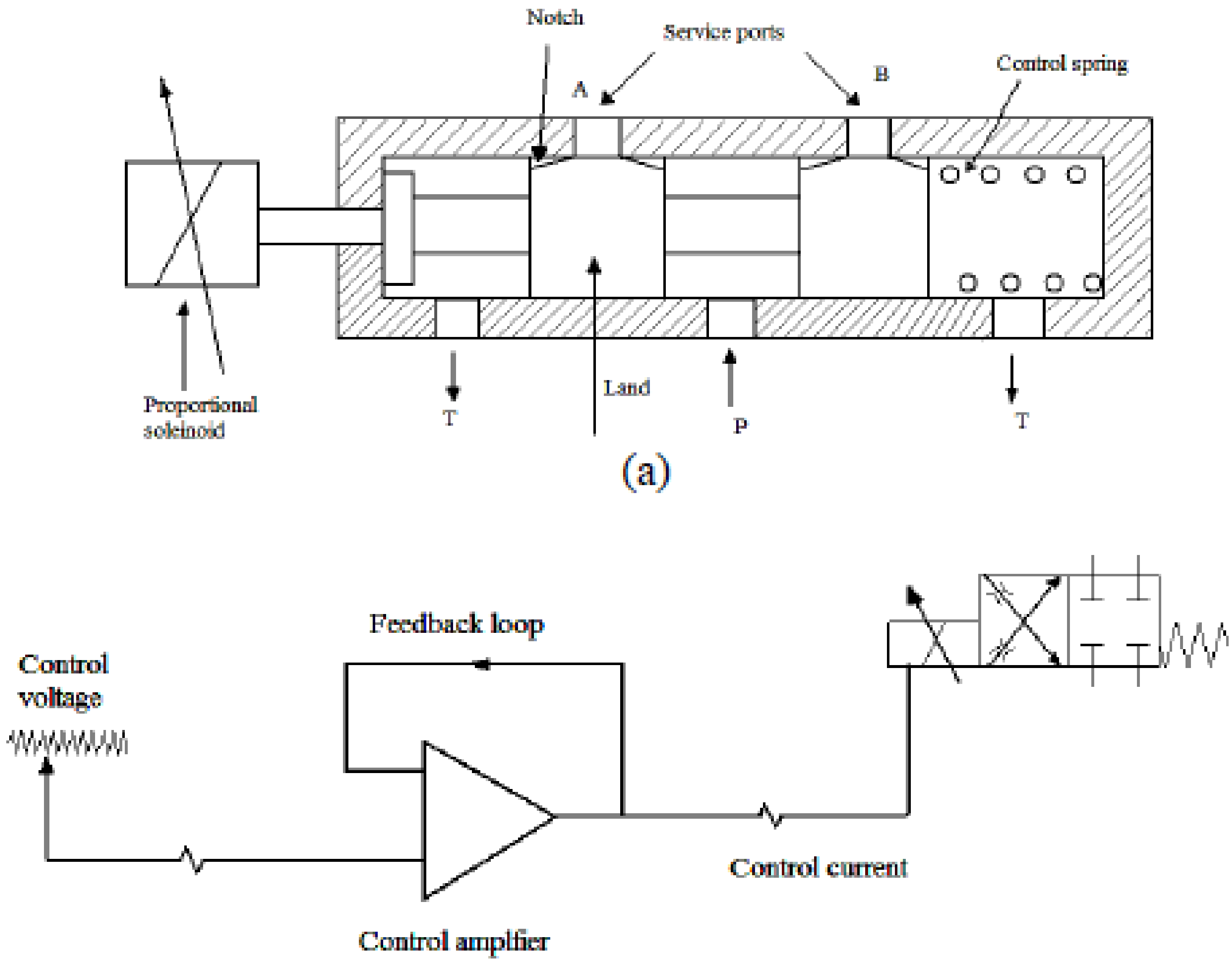


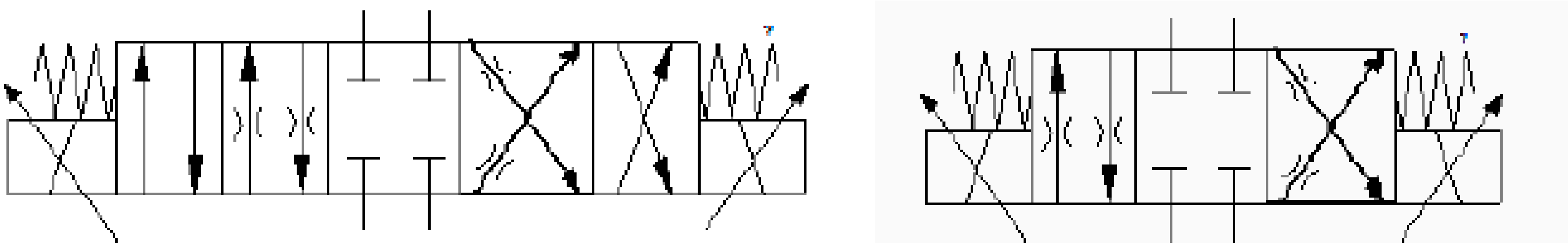
Figure 1.8 Diagrammatic section of a proportional control valve.



(a)

(b)

Figure 1.10 Notched spool proportional valve.(a) Valve construction;(b) electrical control diagram.



(a) (b)

Figure 1.11 The symbols of a proportional directional control valve: (a) Five position;(b) three position .

1.4.2 Spool Positional Control

In order to increase the accuracy and extend the range of applications of proportional control valves, a linear transducer may be fitted to measure the spool position. The output from the transducer is a voltage that is proportional to the spool displacement and it continuously varies through the total spool movement. The actual position of the spool is fed back via the transducer to the electrical control system and then compared with the required position, the control current being adjusted accordingly. Such a system is shown in Fig. 1.12.

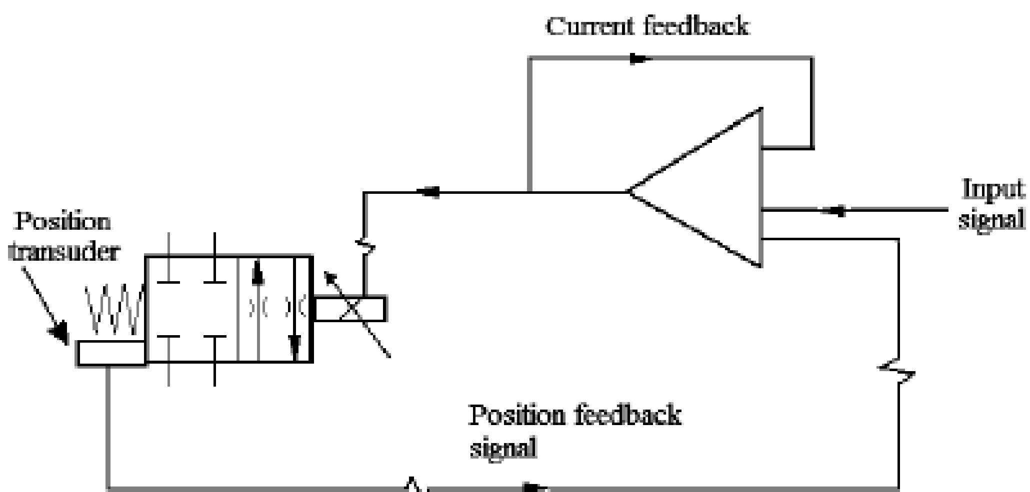


Figure 1.12 The symbols of a proportional directional control valve.

1.4.3 Proportional Pressure Control

In a conventional pressure control valve, a spring is used to control the pressure at which the valve operates. The spring is replaced by a DC solenoid in the case of proportional valves; the force set up by the solenoid is controlled by being dependent on the current flowing through it.

1.4.3.1 Single-Stage ProportionalRelief Valves

Direct-acting proportional relief valves are shown in Fig. 1.14. The proportional solenoid exerts a force on the poppet keeping the valve closed, until the hydraulic pressure at port P overcomes this force and opens the valve. In the design of the relief valve, the proportional solenoid acts directly on the valve poppet.

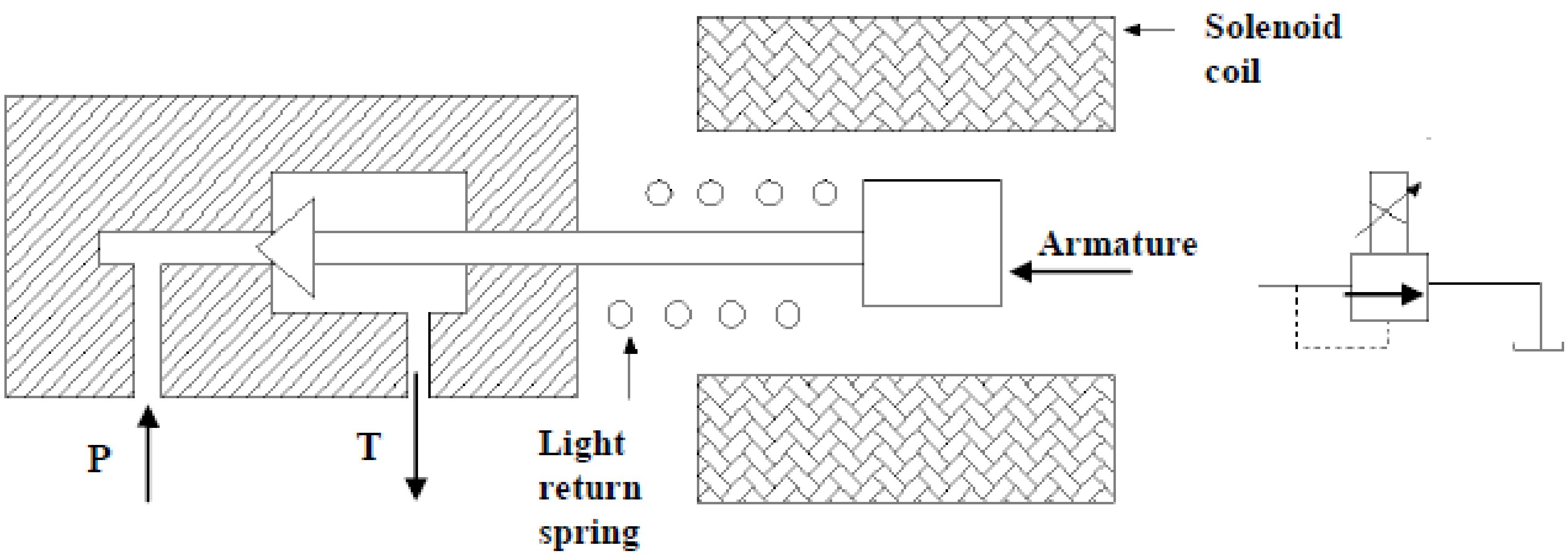


Figure 1.14 Direct-acting proportional relief valve.

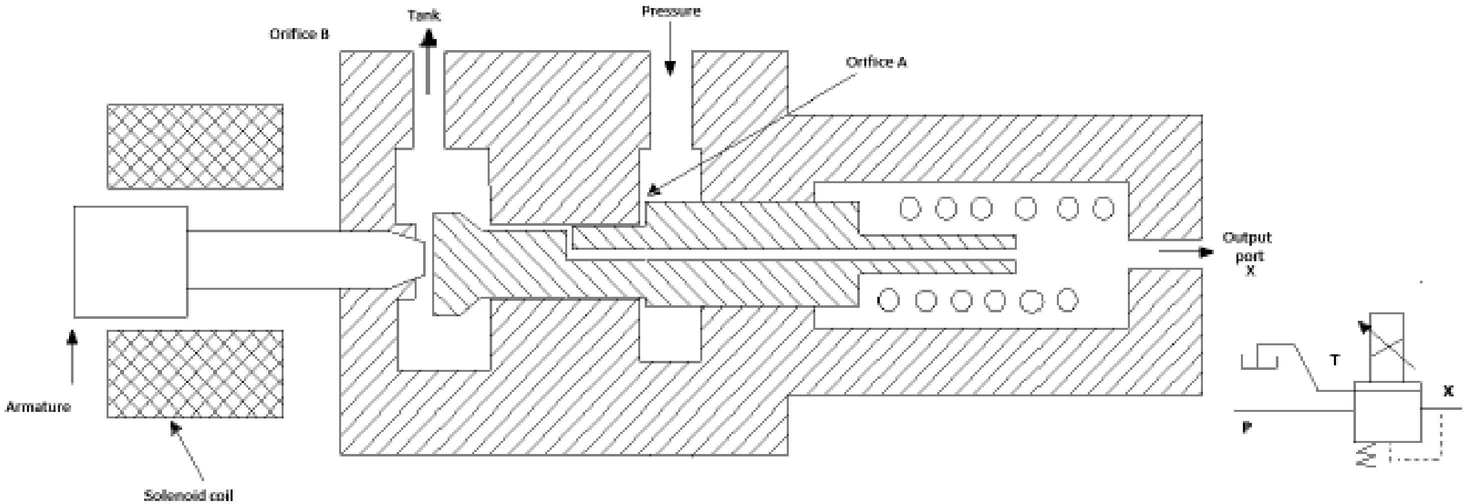


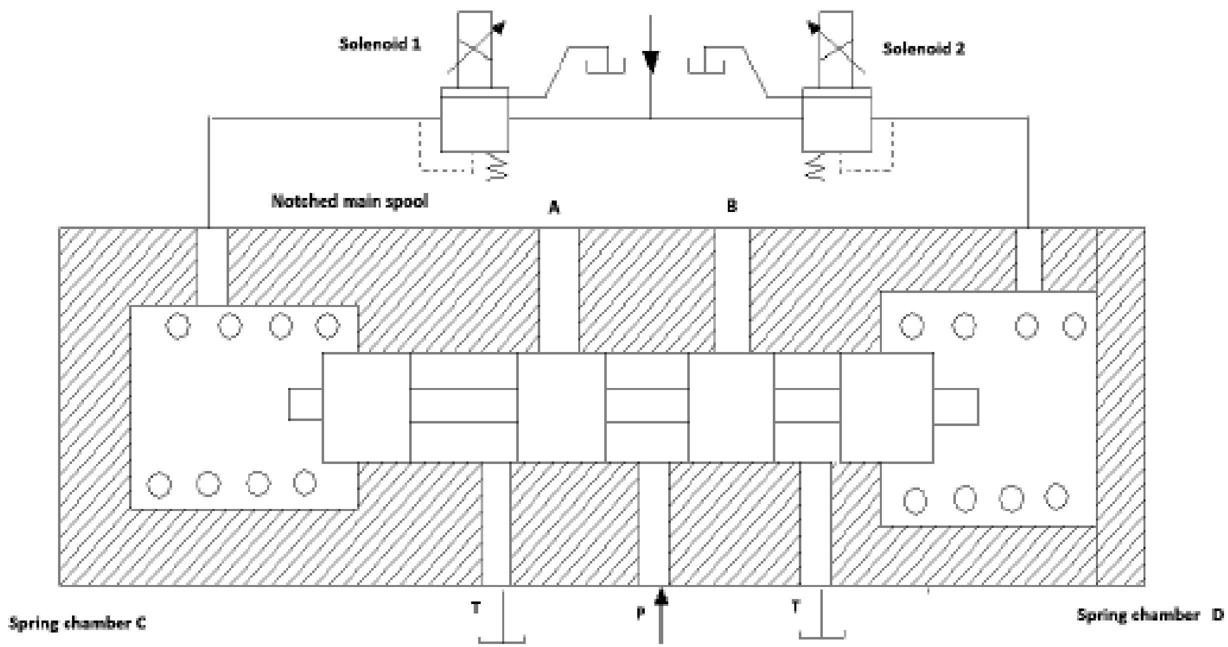
Figure 1.15 Proportional pressure-reducing valve.

1.4.4 Two-Stage Proportional Valves

The valves already discussed have a maximum flow capacity of 5 LPM; to obtain highe rates in valves, two-stage versions are available. A single-stage proportional pressure valve is used to pilot the main valve. These operate in a manner similar to conventional tw valves.

1.4.5 Two-Stage Proportional Directional Control Valves

The pressure output from a proportional pressure-reducing valve is directed to move the sp the main valve against a control spring. Energizing solenoid 1 causes pressure to be app pilot port X and hence to current in solenoid 1. As the main spool lands are notched, a mov to the right progressively opens the flow paths from P to B and A to T. De-energizing sole de-pressurizes spring chamber C and the control spring centralizes the spool.



(a)

SERVO VALVES

A servomechanism is defined as an automatic device for controlling a large amount of power by means of a very small amount of power and automatically correcting the performance of a mechanism. The automatic and continuous correction requires return of information from the mechanism– feedback, in other words. Therefore, a servo valve is operated without feedback and it is not a true servomechanism. In Chapter 17 we have studied about proportional valves and Table 1.1 gives comparison between servo valve and electrohydraulic proportional control valves (EHPV).

Table 1.1 Comparison of servo valves and electrohydraulic proportional valves

Feature	**Servo Valve**	**EHPV**
Electrical operator	Torque motor	Proportional solenoid
Manufacturing precision	Extremely high	Moderately high
Feedback circuitry	Main system as well as valve	Valve (depending on type), main system (seldom)
Cost (compared with a solenoid valve)	Very expensive	Moderately expensive

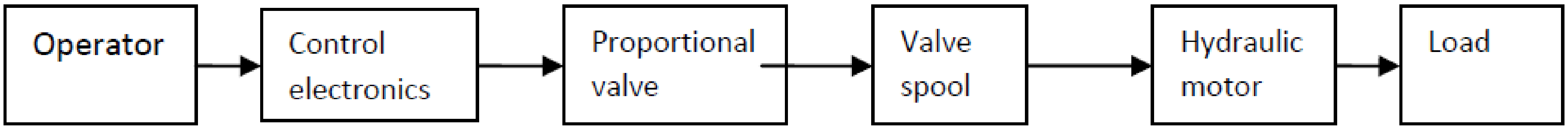


Figure 1.2 Proportional valve block diagram.

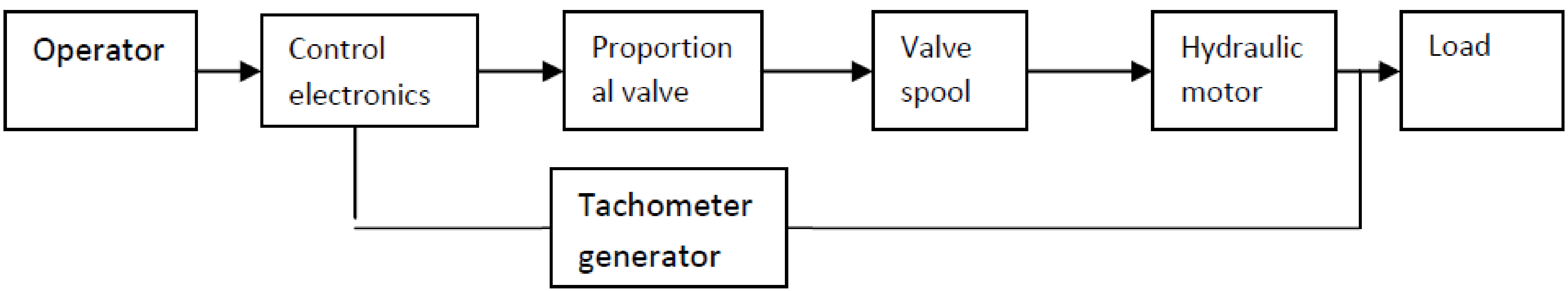


Figure 1.3 Servo control provides automatic and continuous corrections for any changes in motor rpm.

1.4 Servo Valves

Servo valves can be used in virtually any aspect of fluid power system operations, including the following:

1. Positioning of cylinders and rotary actuators.
2. Speed of cylinders and motors.
3. Cylinder force and motor torque.
4. Acceleration and deceleration.
5. System pressure.
6. Flow rate.

1.4.1 Torque Motor

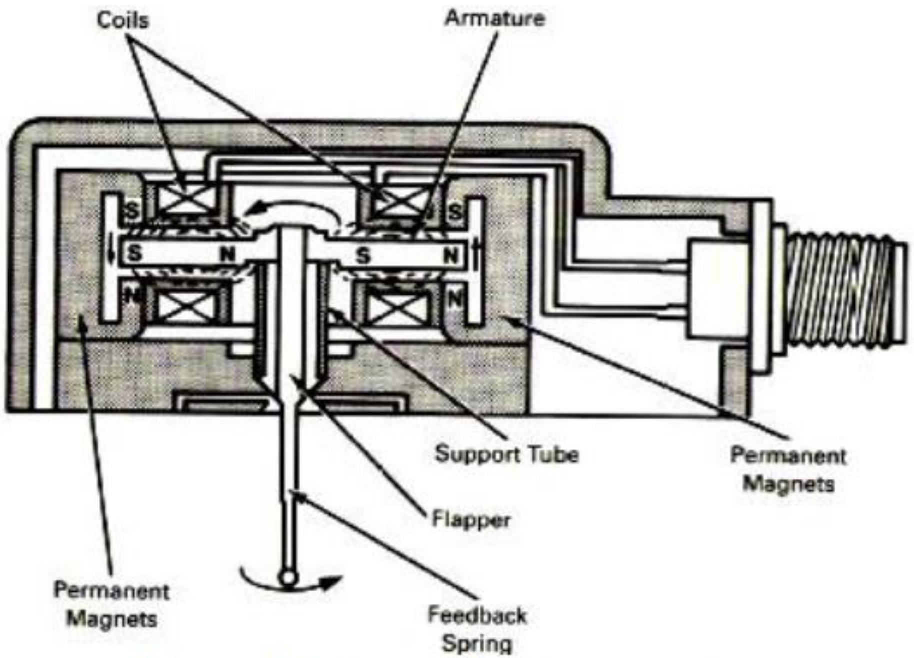


Figure 1.4 Servo valve torque motor.

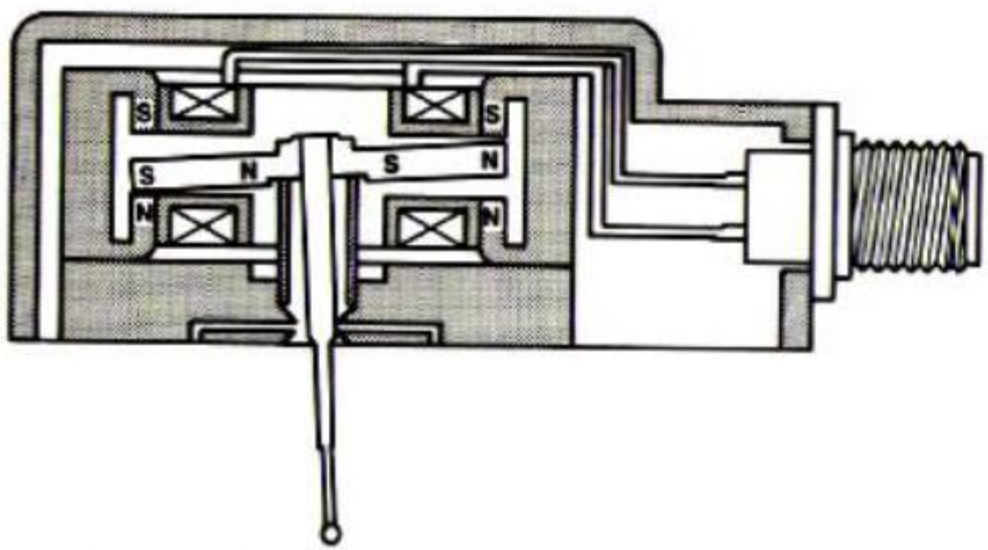


Figure 1.5 Servo torque motor operation.

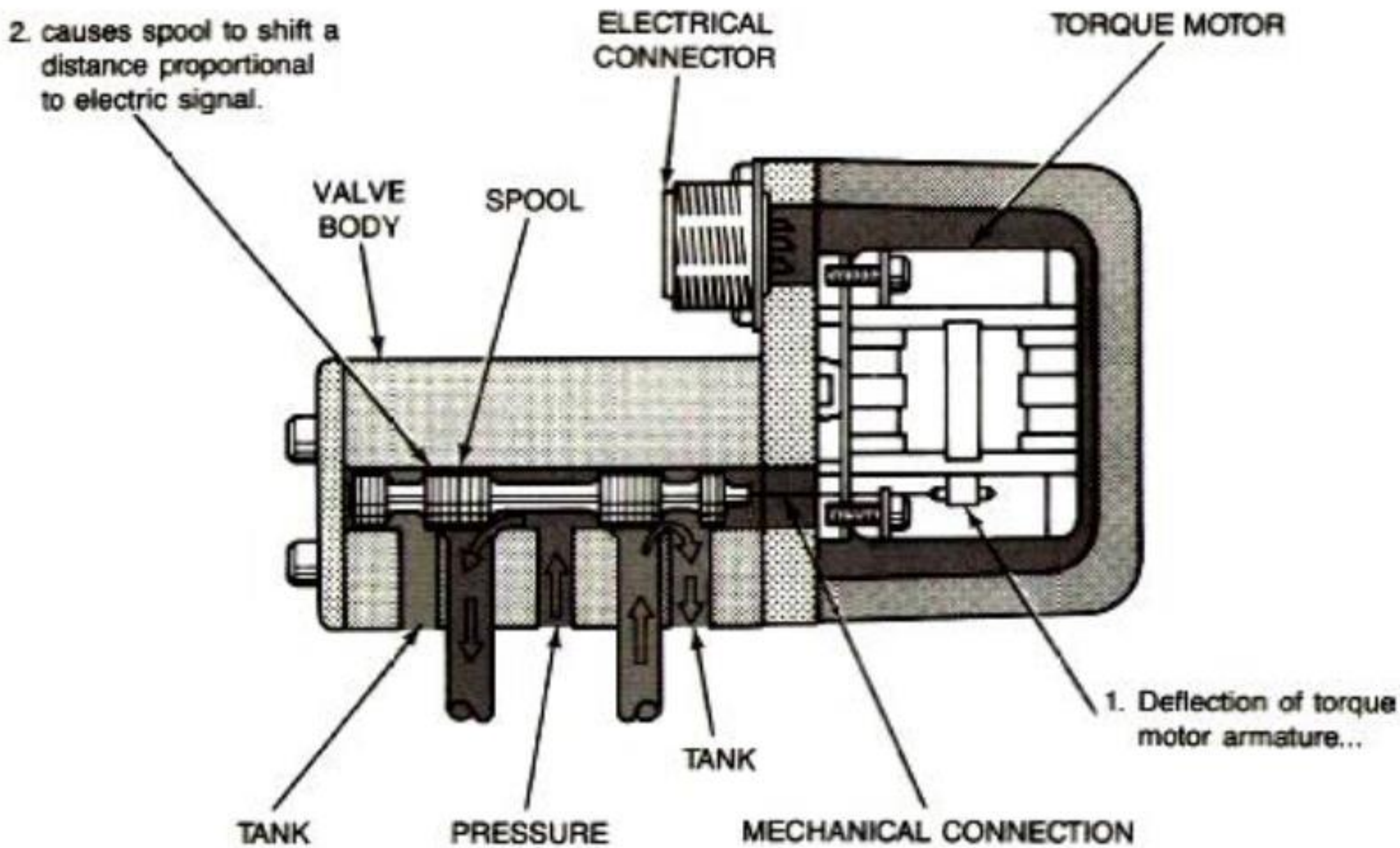


Figure 1.8 Single-stage spool-type servo valve.

1.5.5Two-Stage Servo Valve

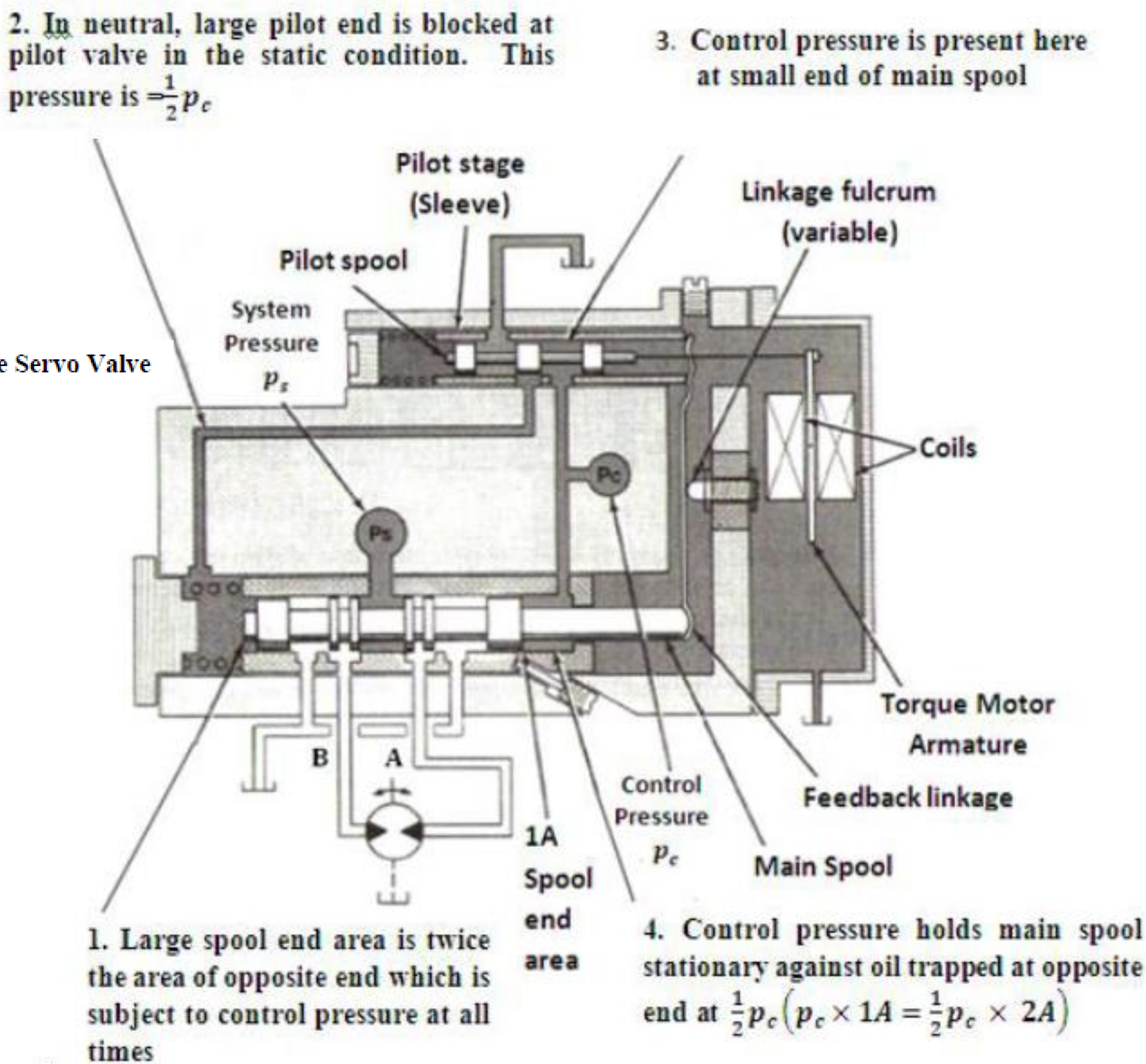


Figure 1.9 Two-stage spool-type servo valve.

1.5.6Double-Flapper Nozzle Pilot Stage

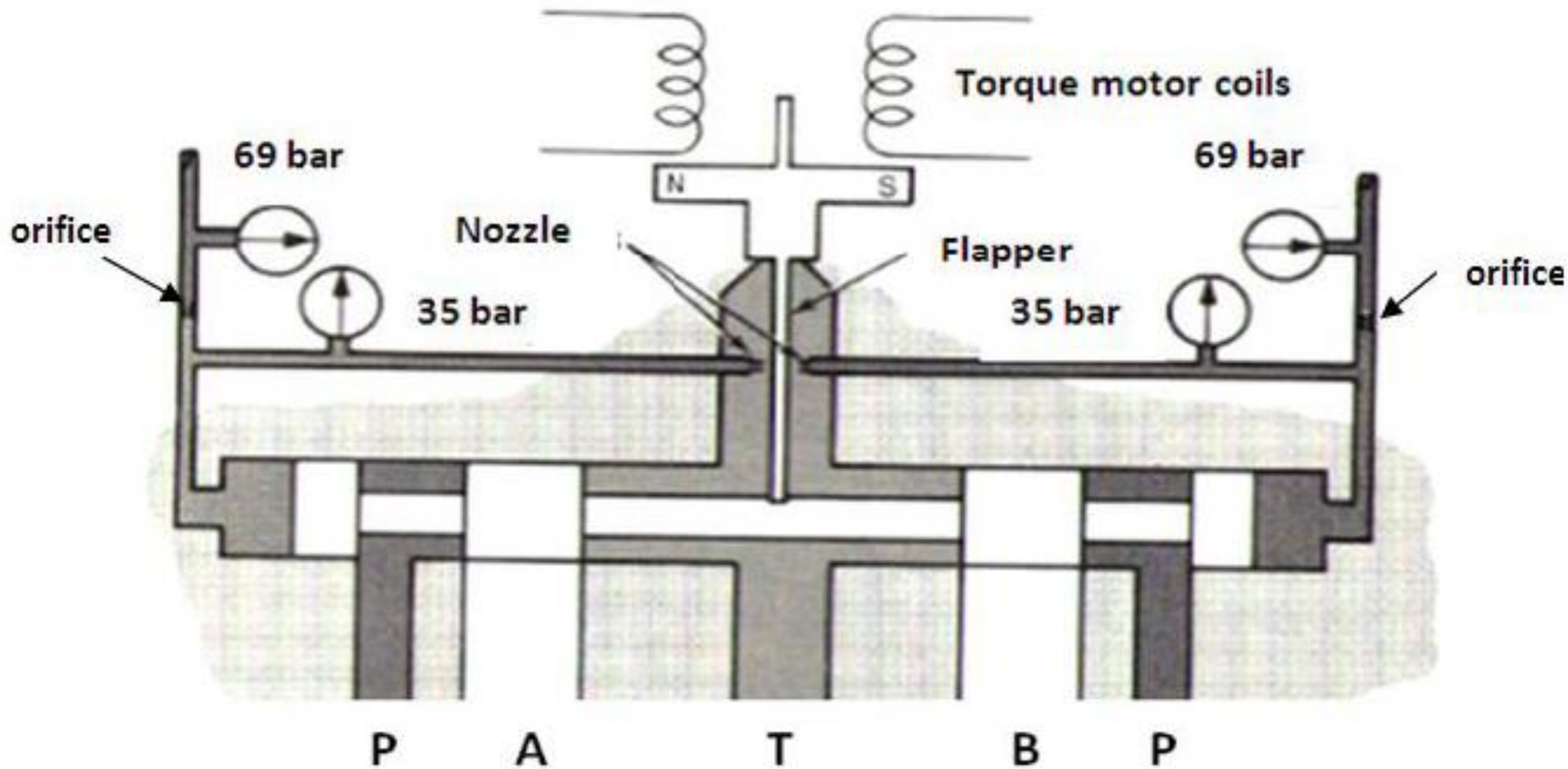


Figure 1.10 Two-stage spool-type servo valve.

1.5.7Jet Pipe Servo Valve

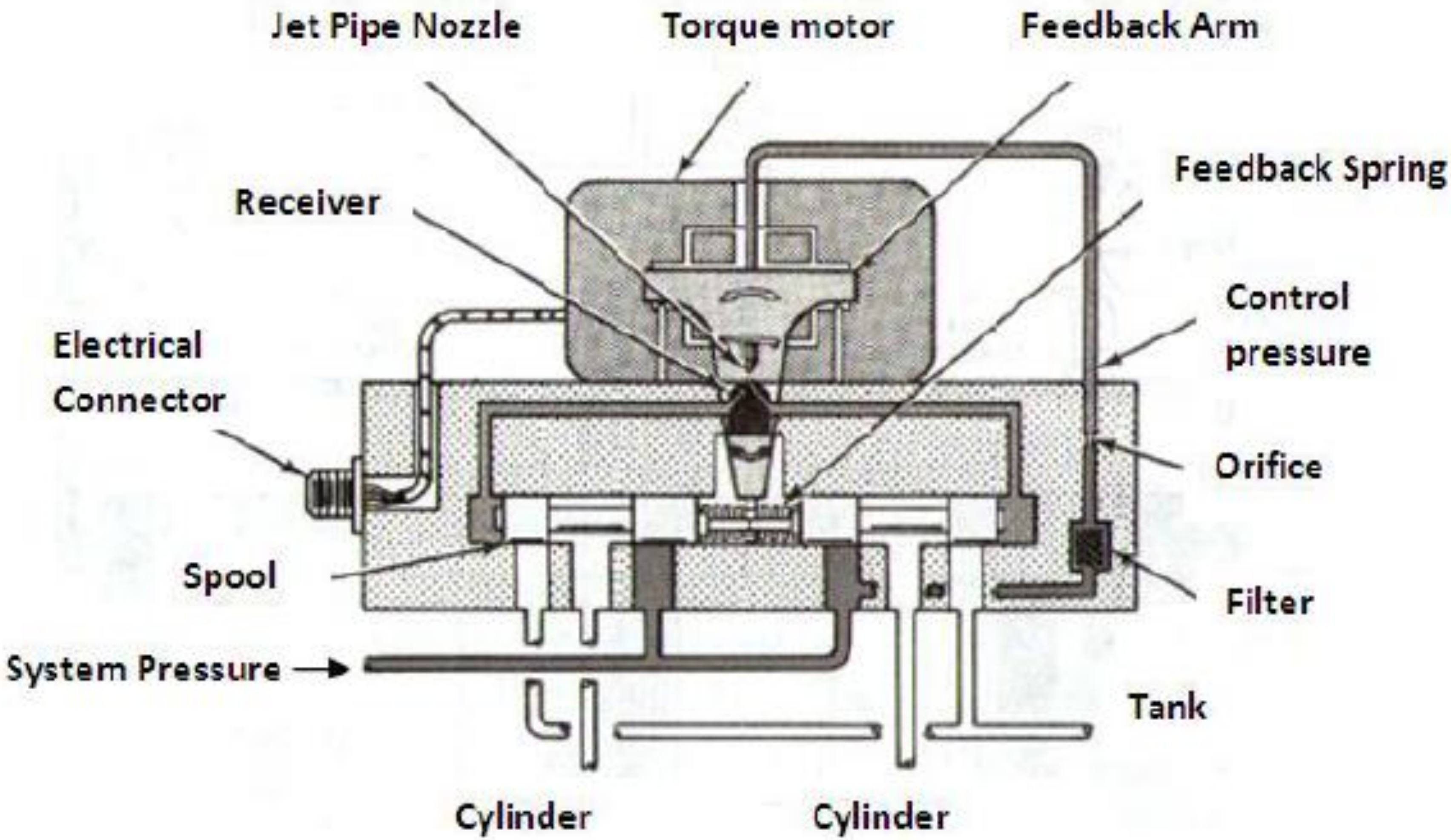
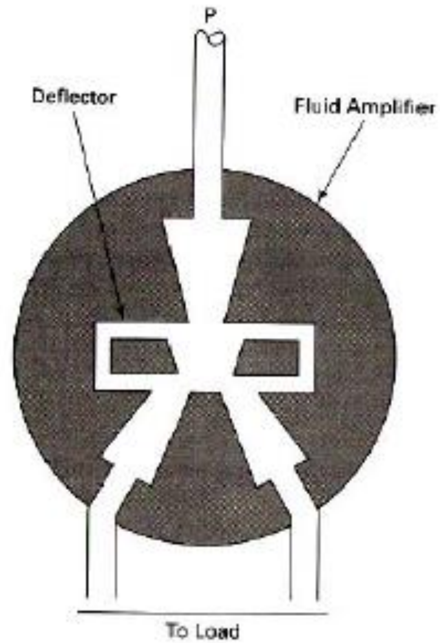


Figure 1.11 Two-stage spool-type servo valve.



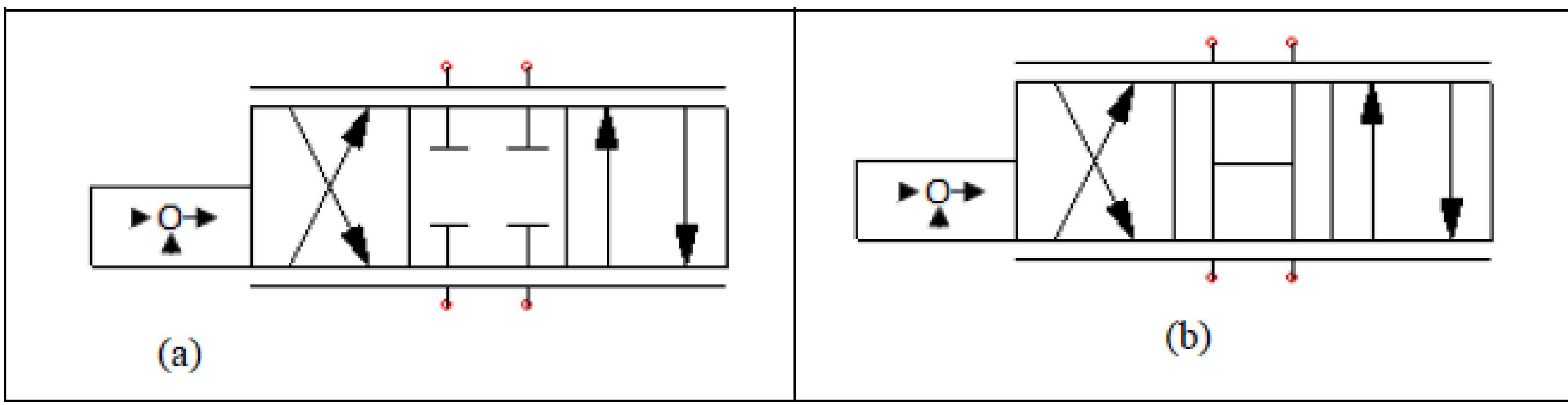


Figure 1.13 Servo symbols: (a) Line-to-line and overlapped spool; (b) underlapped spool.

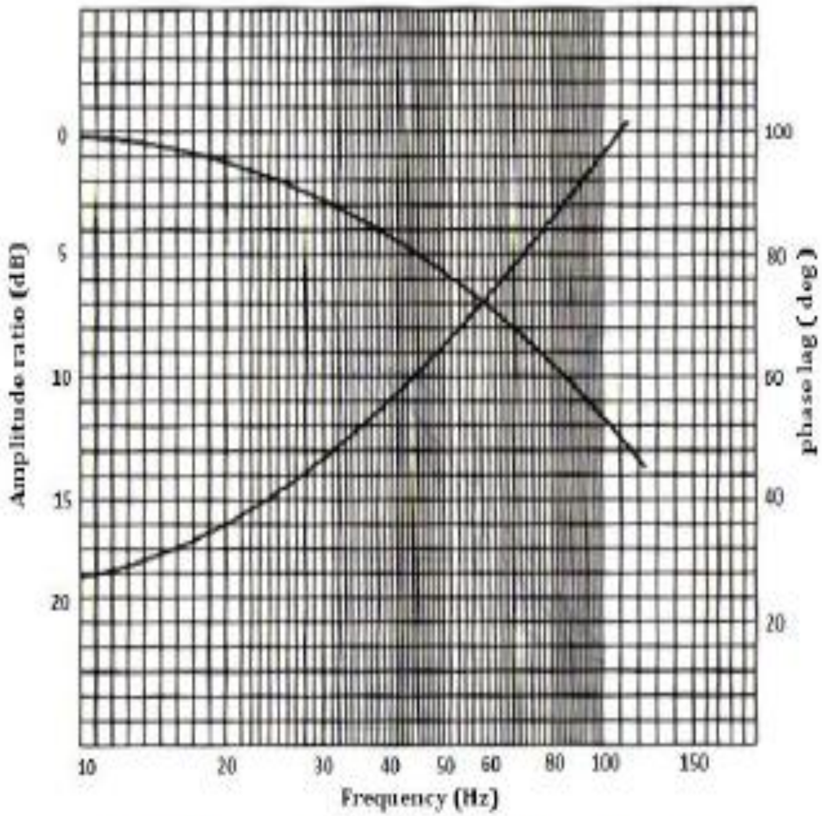


Figure 1.14 Valve performance.

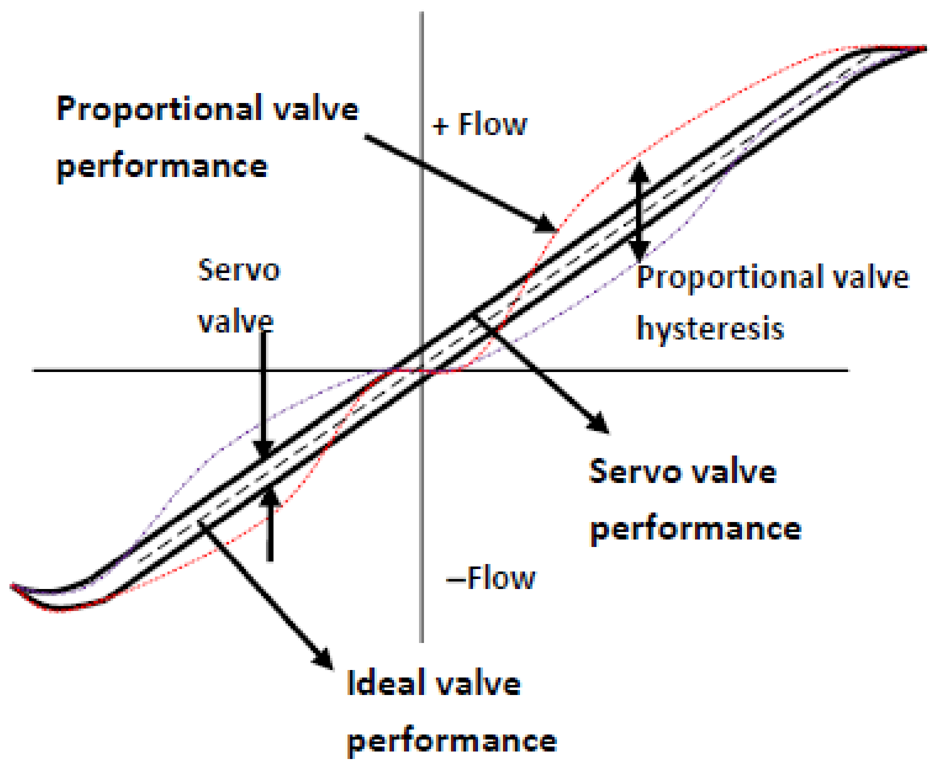


Figure 1.15 Hysteresis for a servo valve and a typical proportional valve

A second important valve characteristic is the valve dead band. Dead band occurs only at the null position, as shown in Fig. 1.16. It is defined as the current required to move the spool from the exact centered position to the position where the first flow output is seen. It is usually expressed in milliamps or percent-rated current. Dead band is the result of the spool inertia, overlap, static friction and any other forces that might impede the initial motion.

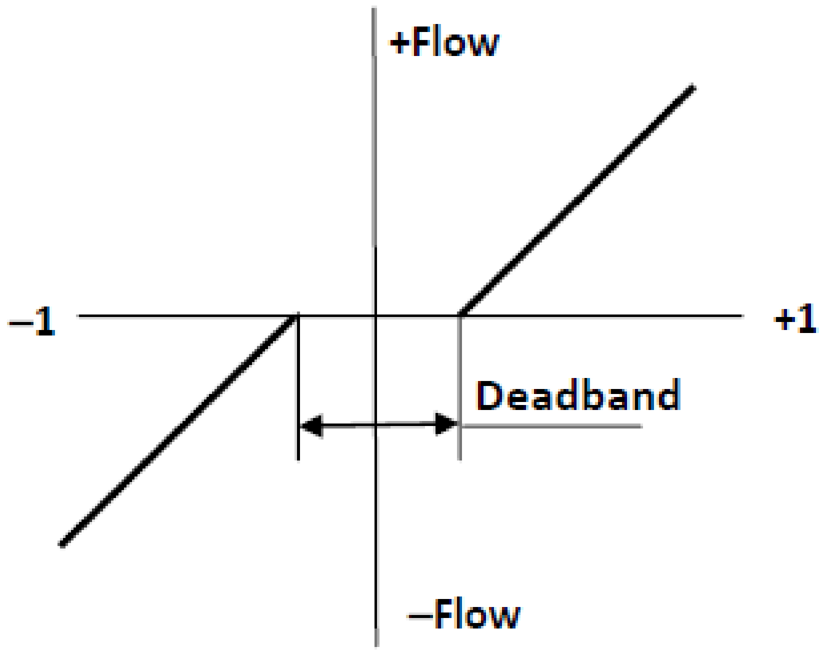
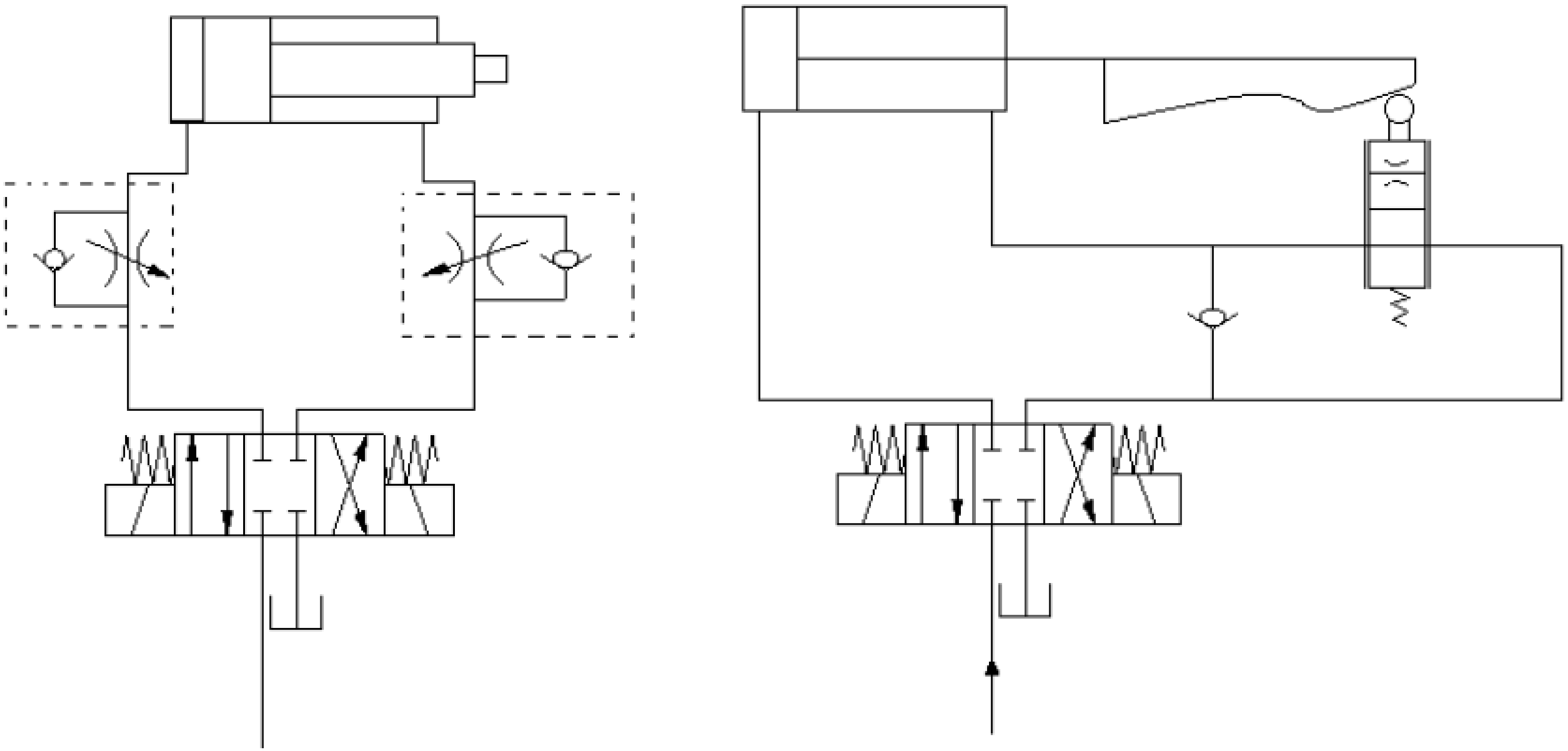


Figure 1.16 Dead band of a valve.

A similar phenomenon is threshold. Threshold current is the smallest input current required to overcome spool inertia and other impending forces to cause the spool to move. The primary difference between threshold and dead band is that threshold occurs throughout the spool stroke, whereas dead band occurs only at the null position. Threshold contributes to the dead band rating.

Control of Actuators

Speed Control of Cylinders



(a) (b)

Figure 1.21 (a) Meter-out speed control. (b) Cam-operated speed control.

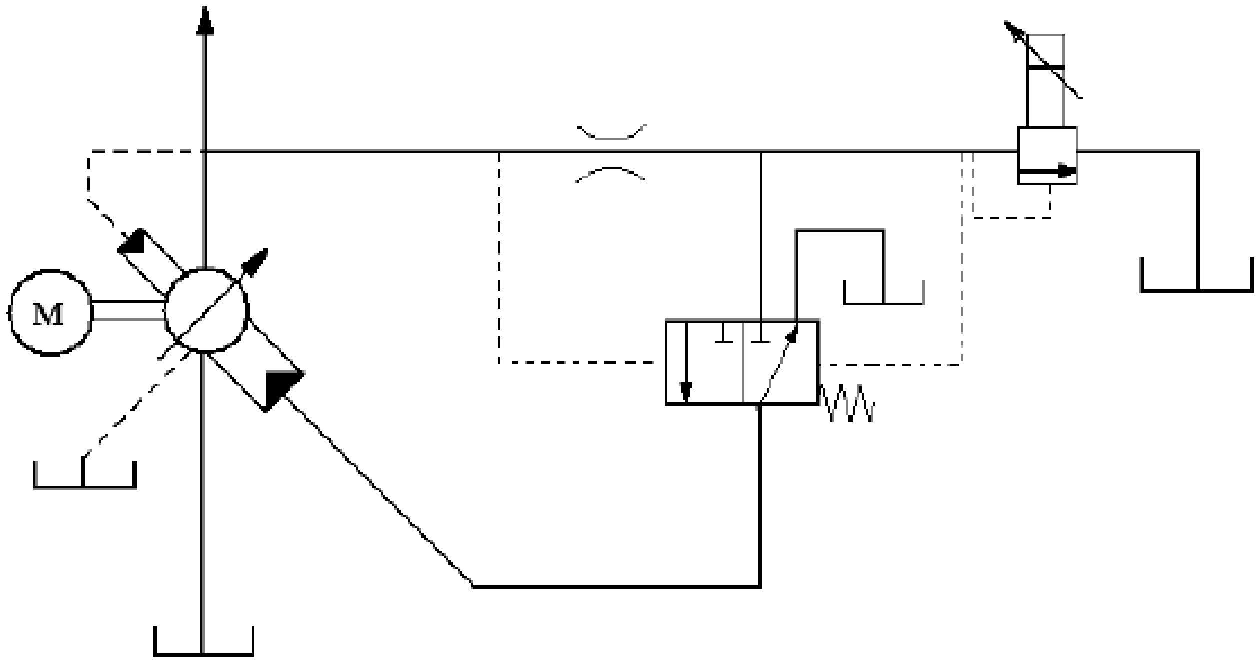


Figure 1.22 Pressure-compensated pump with proportional pressure control.

1. Safety of machine and personnel in the event of power failures.
2. Performance of given operation with minimum losses.
3. Cost of the component used in the circuit.

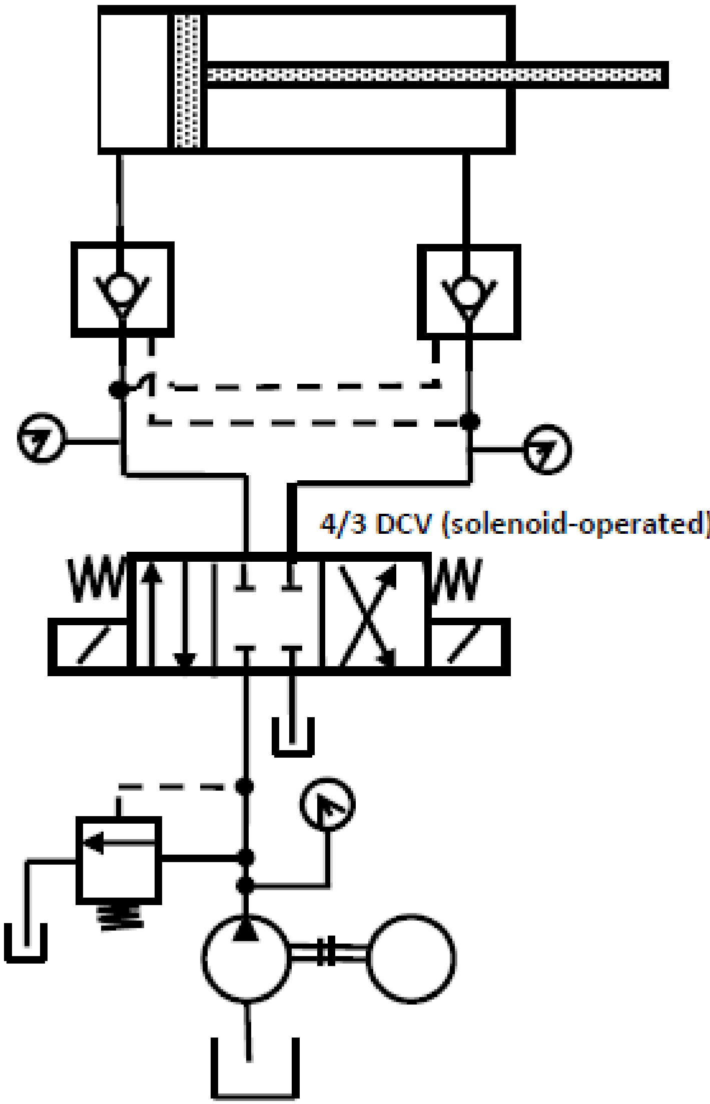


Figure 1.9 Locked cylinders with pilot check valves.

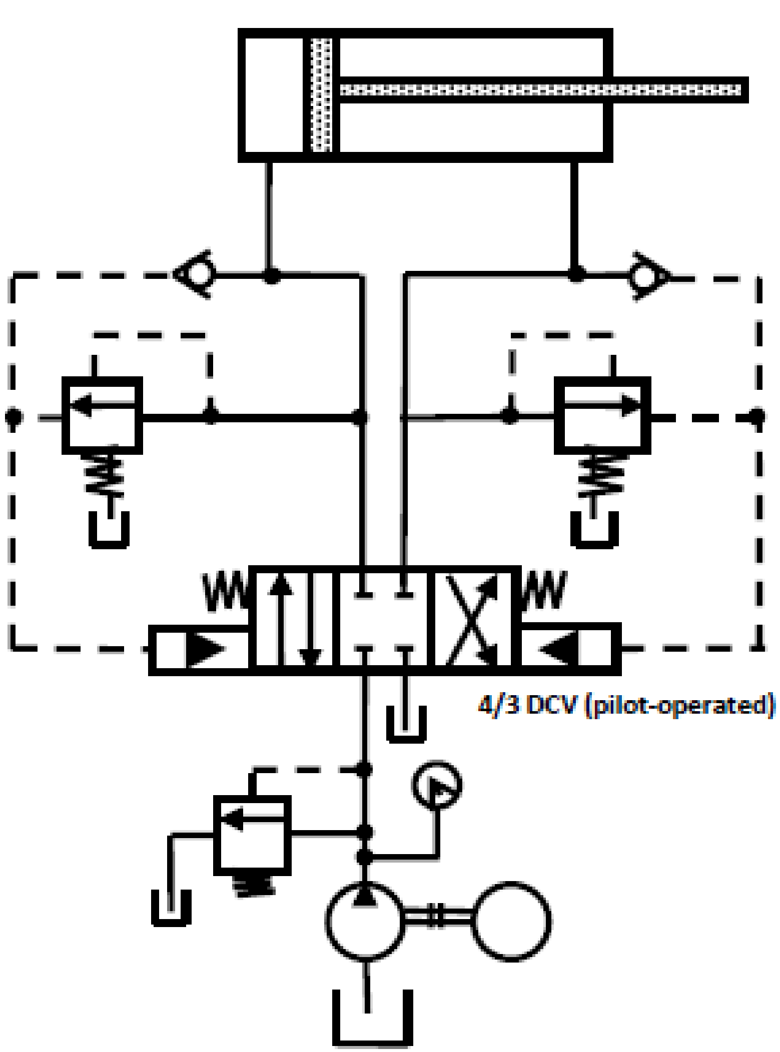


Figure 1.8 Sequencing circuit.

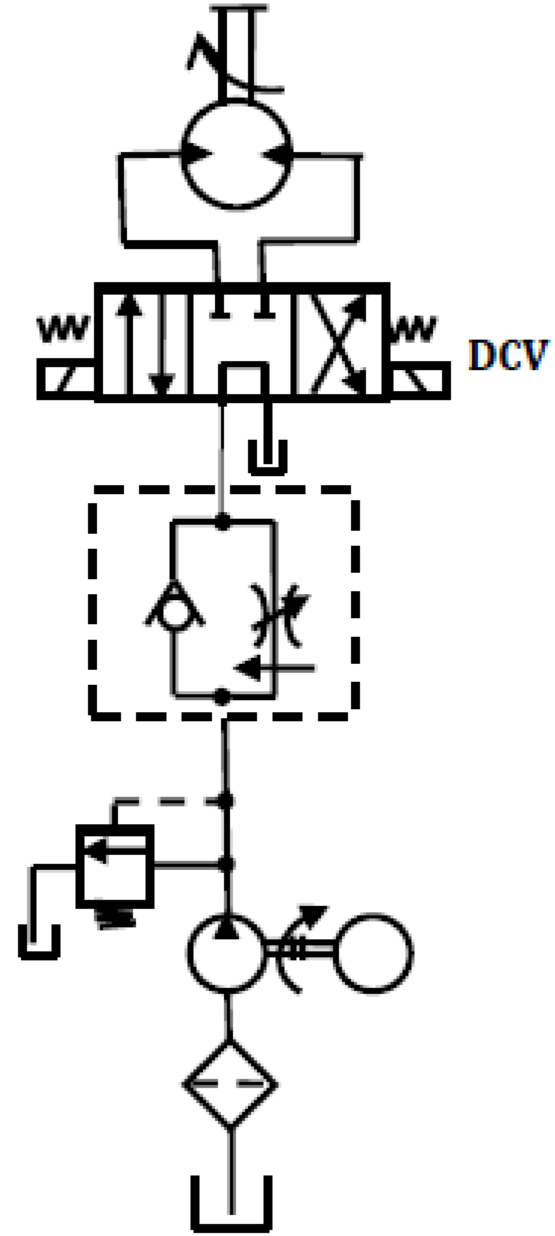


Figure 1.12 Speed control of a motor.

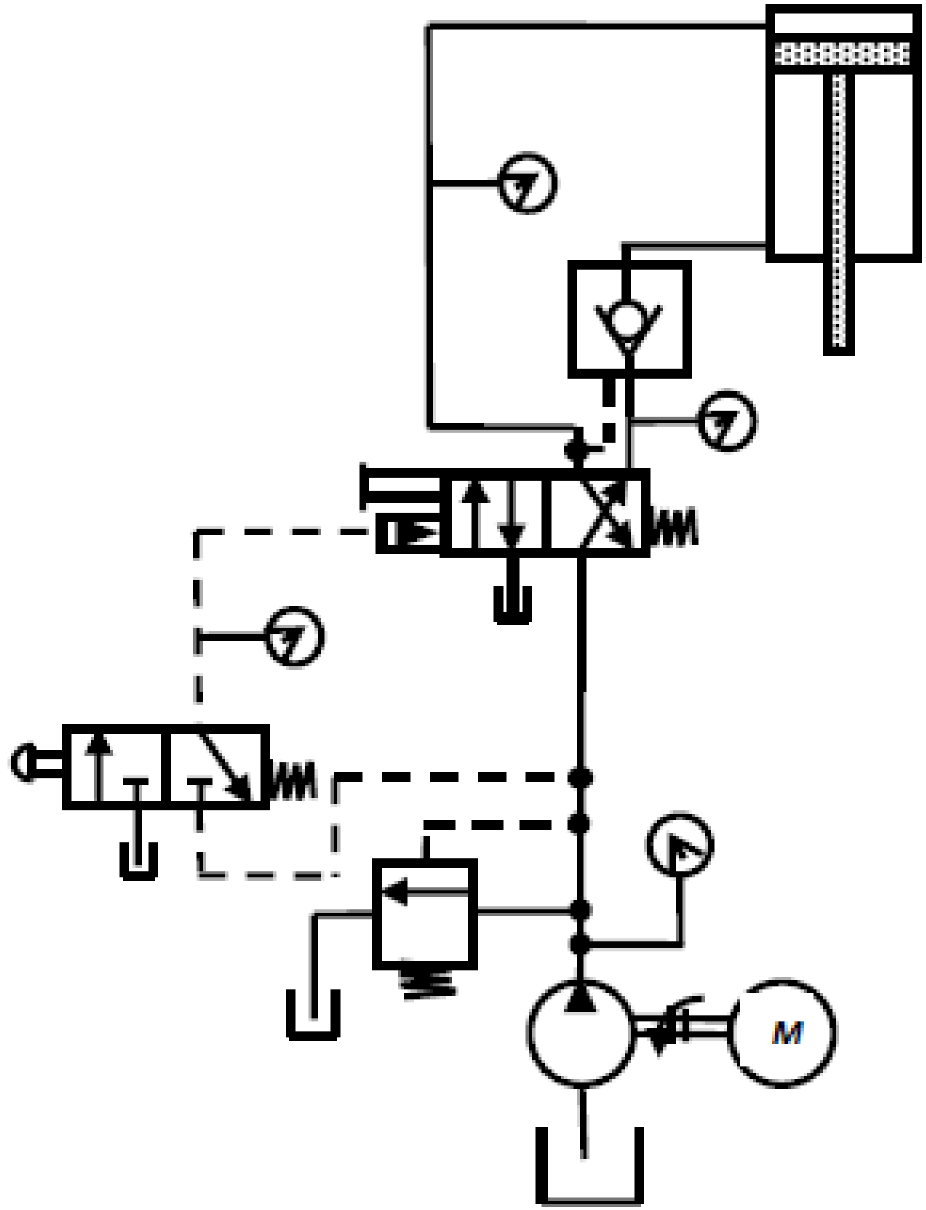


Figure 1.13 Fail-safe circuits – inadvertent cylinder extension.

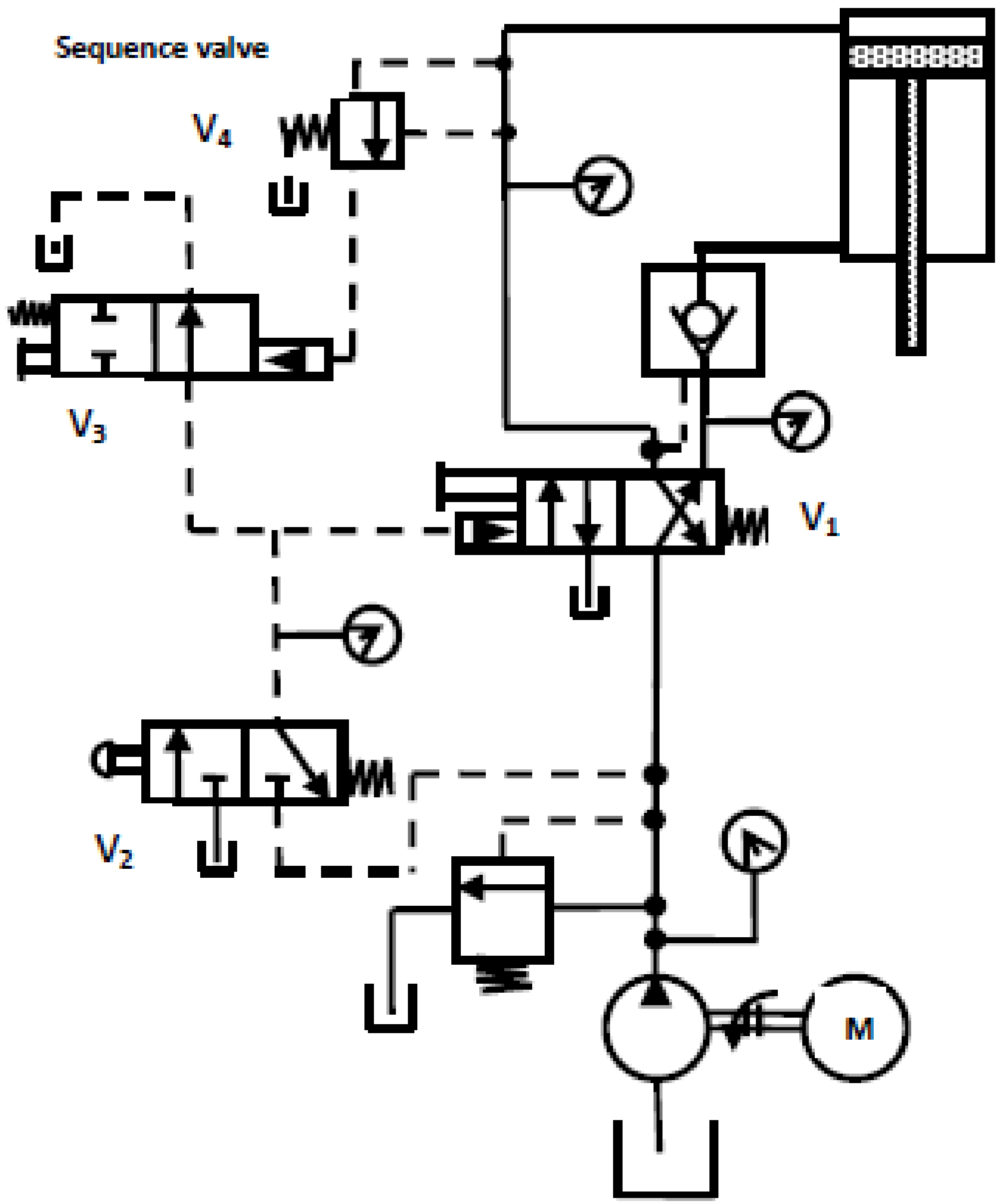


Figure 1.14 Fail-safe circuits –overload protection.

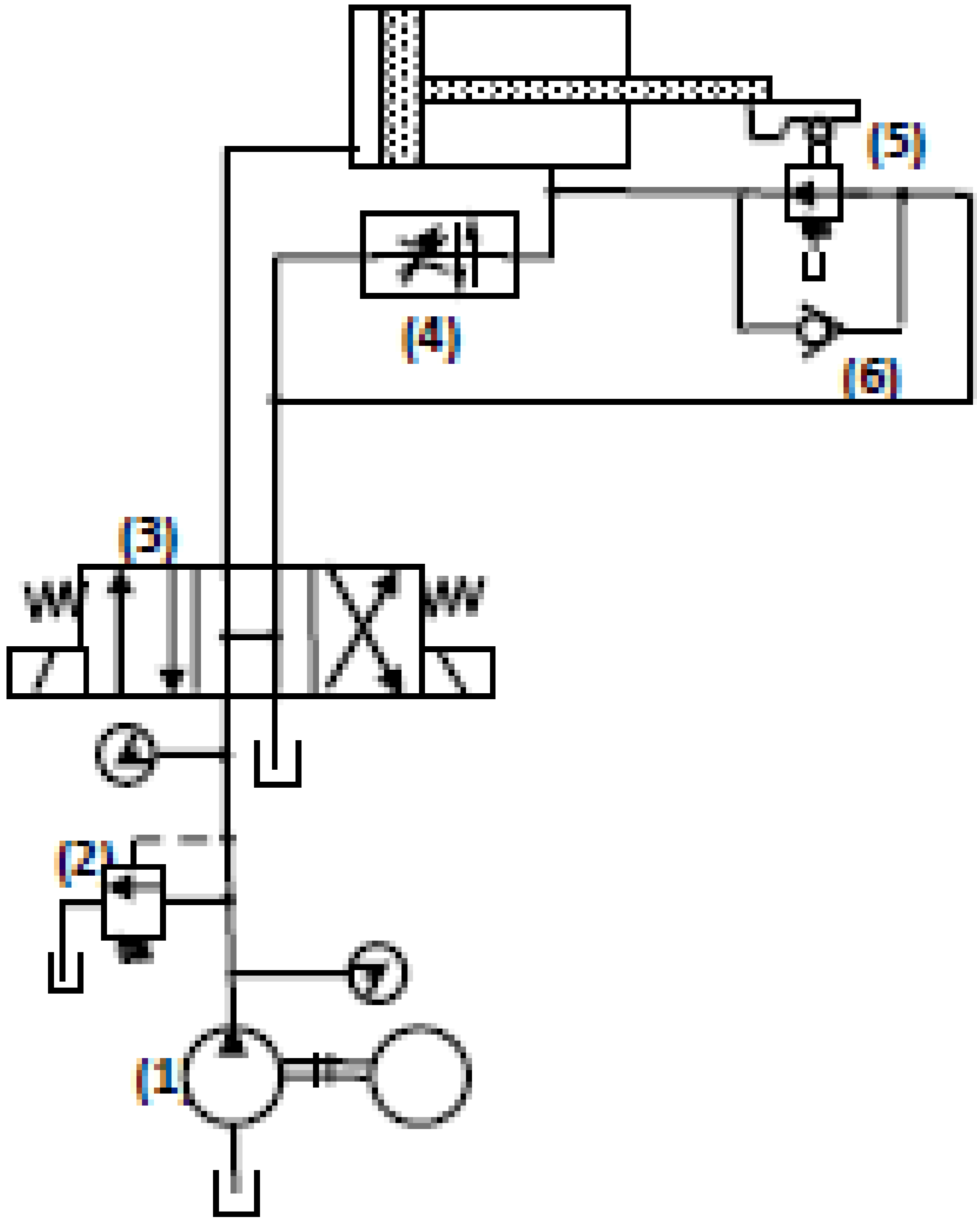


Figure 1.16 Rapid traverse and feed circuit – alternate circuit.

For the circuit of Fig. 1.19, give the sequence of operation of cylinders 1 and 2 when the pump is turned ON. Assume that both cylinders are initially fully retracted.

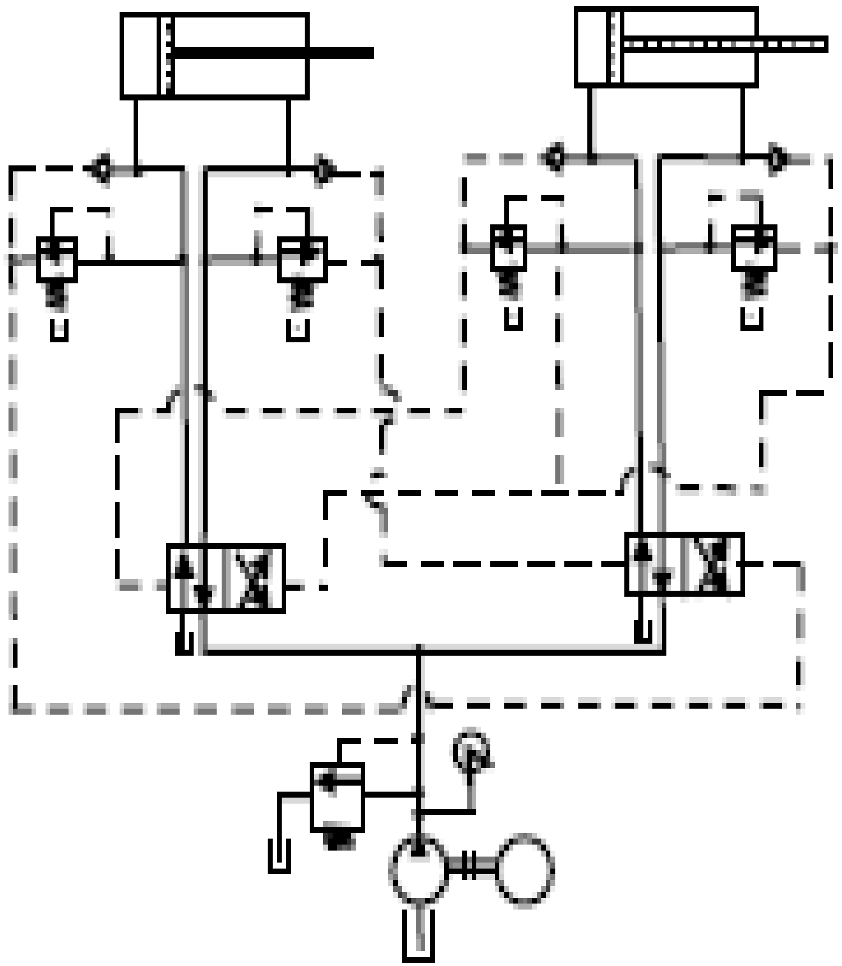
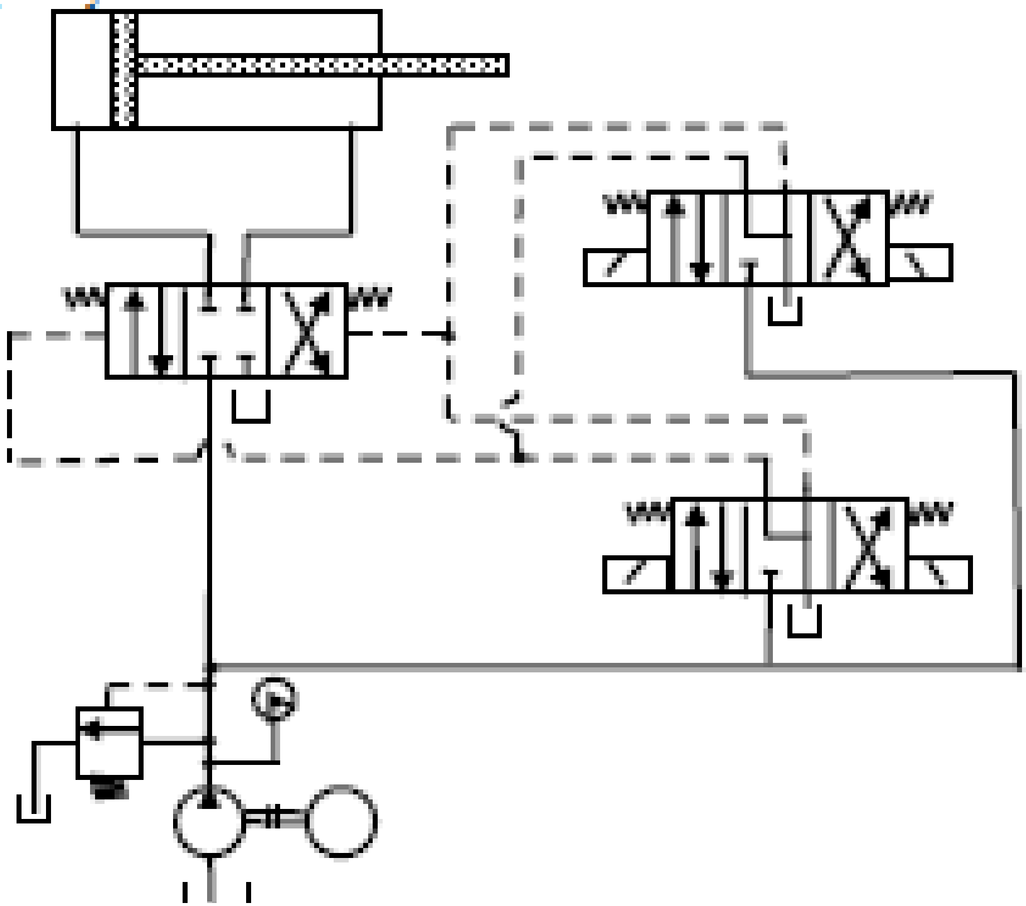


Figure 1.19

Solution: Cylinder 1 extends, cylinder 2 extends. Cylinder 1 retracts, cylinder 2 retracts.

What safety features does Fig. 1.20possess in addition to a pressure-relief valve. If the load on cylinder 1 is greater than the load on cylinder 2, how will the cylinder move when DCV is shifted into the extending or retracting mode? Explain your answer.



Solution: Both solenoid-actuated DCVs must be actuated in order to extend or retract the hydraulic cylinder.

Cylinder 2 will extend through its complete stroke receiving full pump flow while cylinder 1 will not move. The moment cylinder 2 will extend through its complete stroke, cylinder 1 will receive full pump flow and extend through its complete stroke. This is because the system pressure builds up until load resistance is overcome to move cylinder 2 with the smaller load. Then the pressure continues to increase until the load on cylinder 1 is overcome. This then causes cylinder 1 to extend. In retraction mode, the cylinders move in the same sequence.

ME 5451 – Hydraulics and Pneumatics

Lecture -3

Date: 04-03-2021 Time slot: 10:30-12:10 p.m.

Contents

1. Refreshing lec-2
2. Fluid power – energy- efficiency
3. Sources of Hydraulic power : Pumping Theory – Pump Classification
4. Gear Pump

Course Instructor: Dr. A. Siddharthan

$$\text{Hydraulic power } (W) = \text{Pressure} \times \text{Flow}$$
$$= p\ (\mathrm{N/m^2}) \times Q\ (\mathrm{m^3/s})$$
$$= p \times Q\ (\mathrm{N\,m/s}) = p \times Q\ (\mathrm{W})$$

It is usual to express flow rate in liters/minute (LPM) and pressure in bars. To calculate hydraulic power using these units, a conversion has to be made. Thus,

$$Q\ (\mathrm{L/min}) = Q/60\ (\mathrm{L/s})$$
$$\Rightarrow Q\left(\frac{\mathrm{L}}{\mathrm{min}}\right) = \frac{Q}{60}\left(\frac{\mathrm{L}}{\mathrm{s}}\right) = \frac{Q}{60\times 10^3}\left(\frac{\mathrm{m^3}}{\mathrm{s}}\right)$$
$$\Rightarrow p\,(\mathrm{bar}) = p \times 10^5\,\frac{\mathrm{N}}{\mathrm{m^2}}$$

Hydraulic power is

$$Q\left(\frac{1}{\mathrm{min}}\right) \times p\,(\mathrm{bar}) \times \frac{1\times 10^5}{60\times 10^3}\left(\frac{\mathrm{m^3}}{\mathrm{s}} \times \frac{\mathrm{N}}{\mathrm{m^2}}\right)$$
$$\Rightarrow Q \times p\,(\mathrm{bar}) \times \frac{1\times 10^3}{600}\ (\mathrm{W}) = \frac{Q(\mathrm{LPM}) \times p\,(\mathrm{bar})}{600}\ (\mathrm{kW})$$

Thus, hydraulic power (kW) is

$$\frac{\text{Flow (LPM)} \times \text{Pressure (bar)}}{600}$$

In the SI metric system, all forms of power are expressed in watt. The pump head Hp in units of meters can be related to pump power in units of watt by using $p = \gamma h$. So

Example 1.15

A cylinder with a piston of diameter 8 cm and a rod of diameter 3 cm receives fluid at 30 LPM. If the cylinder has a stroke of 35 cm, what is the maximum cycle rate that can be accomplished?

Solution: We know that

Volume of oil displaced per minute (m^3/min) = Area × Stroke length × No. of cycles per minute

So

$$Q = \frac{\pi(0.08^2)\,\mathrm{m}^2}{4} \times \left\{\frac{35}{100}\right\}(\mathrm{m}) \times N(\mathrm{cycles/min}) + \frac{\pi(0.08^2 - 0.03^2)\,\mathrm{m}^2}{4} \times \left\{\frac{35}{100}\right\}(\mathrm{m}) \times N(\mathrm{cycles/min})$$

$$= 0.03\,\mathrm{m}^3/\mathrm{min}$$

$$\Rightarrow 0.030 = 0.00176 + 0.0015 \times N$$

$$\Rightarrow N = 9.2 \text{ cycles/min}$$

Example 1.16

A hydraulic pump delivers a fluid at 50 LPM and 10000 kPa. How much hydraulic power does the pump produce?

Solution: We have

$$Q = 50 \text{ LPM} = \frac{50}{60 \times 10^3} = 0.833 \times 10^{-3} \text{m}^3/\text{s}$$

Now

$$\mathbf{1\ L = 1000\ cc = 1000} \times 10^{-6} \text{ m}^3 = 10^{-3} \text{m}^3$$

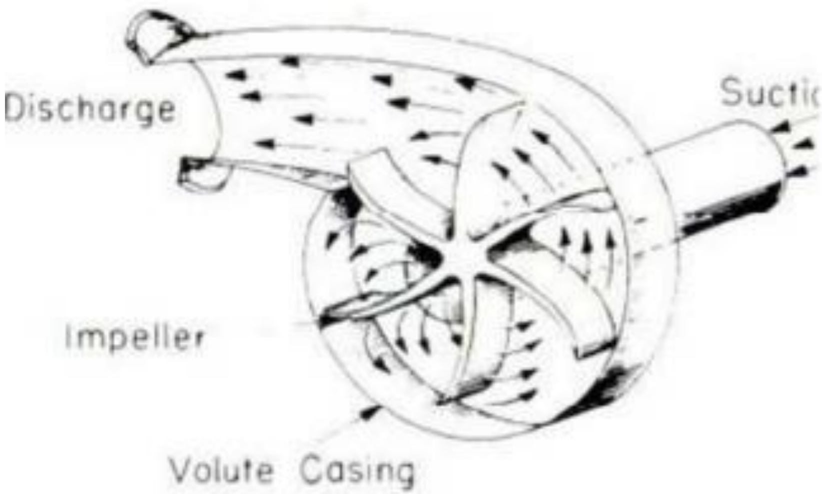
So

$$\text{Power (kW)} = p \text{ (kPa)} \times Q \text{ (m}^3/\text{s)}$$

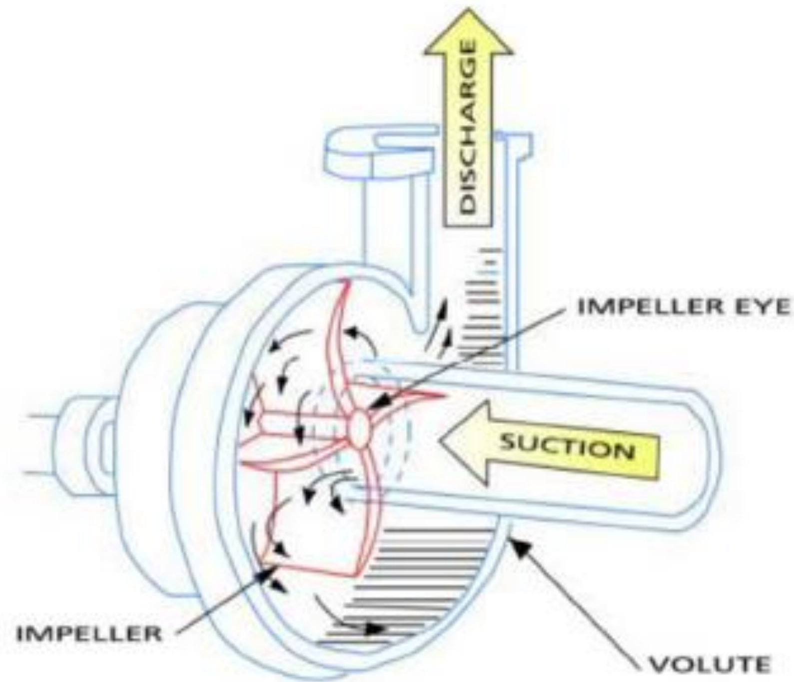
$$= 10000 \times 0.833 \times 10^{-3}$$

$$= 8.33 \text{ kW} = 8330 \text{ W}$$

Non positive displacement pumps



Centrifugal Pump



Impeller Pump

Non-positive displacement pump

Flow does not depend on kinematics only - pressure important

- Also called hydro-dynamic pump (pressure dependent)

Smooth flow

Examples: centrifugal (impeller) pump, axial (propeller) pump

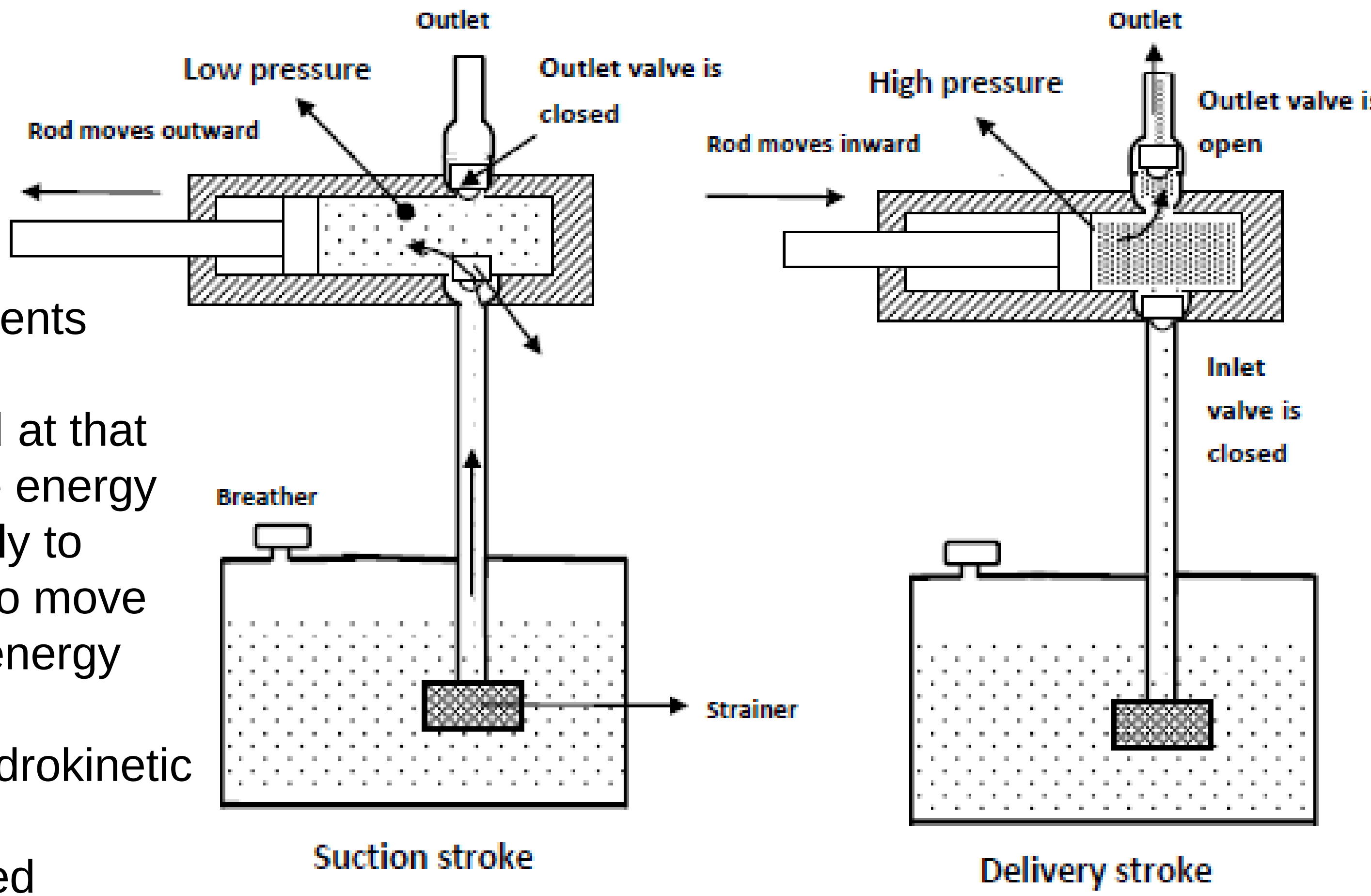
Does not have positive internal seal against leakage

If outlet blocks, Q = 0 while shaft can still turn

- Volumetric efficiency = actual flow / flow estimated from shaft speed
  = 0%

1.3Pumping Theory

A positive displacement hydraulic pump is a device used for converting mechanical energy into hydraulic energy.It is driven by a prime mover such as an electric motor.It basically performs twofunctions.First, it creates a partial vacuum at thepump inlet port.This vacuum enables atmospheric pressure to force the fluid from the reservoir into the pump.Second,the mechanical action of the pump traps this fluid within the pumping cavities,transports it through the pump and forces it into the hydraulic system. It is important to note that pumps create flow not pressure. Pressure is created by the resistance to flow.



Only all the components must be
 (i) able to withstand at that pressure and (ii) the energy source must be apply to
 supply the energy to move the piston. In such energy transfer there is no hydrodynamic or hydrokinetic effect. That is why
such a pump is called *hydrostatic pump.*

Figure 1.2Illustration of pumping theory

Positive vs. non-positive displacement pumps

- Positive displacement pumps
  - most hydraulic pumps are positive displacement
  - high pressure 700 bar
  - high volumetric efficiency (leakage is small)
  - large ranges of pressure and speed available
  - can be stalled !

- Non-positive displacement pumps
  - many pneumatic pumps are non-positive displacement
  - used for transporting fluid rather than transmitting power
  - low pressure 20 bar, high volume flow
  - blood pump (less mechanical damage to cells)

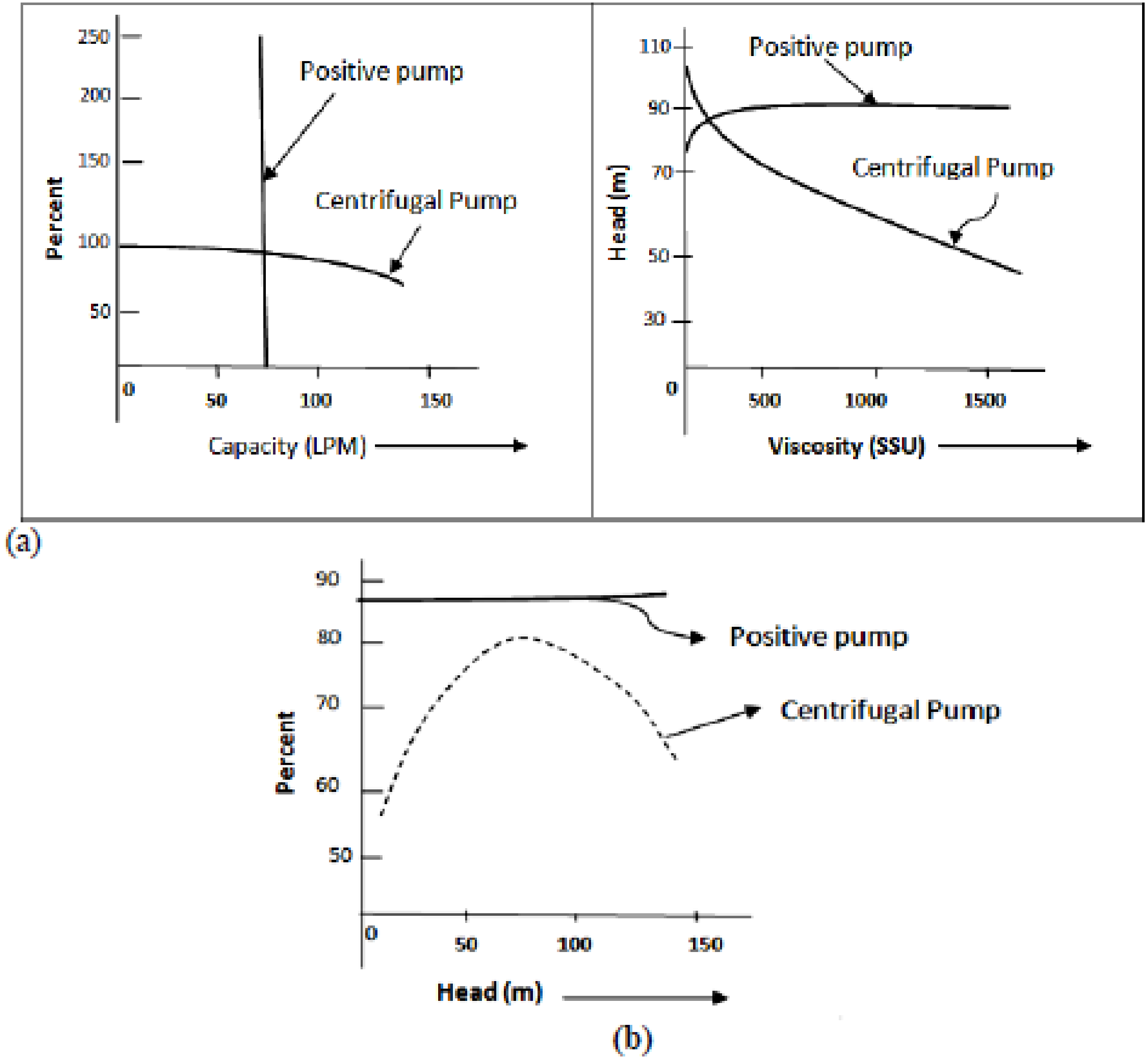


Figure 1.1 Performance curves for positive and non-positive displacement pumps

The function of a pump is to convert mechanical energy into hydraulic energy. It is the heart of any hydraulic system because it generates the force necessary to move the load. Mechanical energy is delivered to the pump using a prime mover such as an electric motor.Partial vacuum is created at the inlet due to the mechanical rotation of pump shaft. Vacuum permits atmospheric pressure to force the fluid through the inlet line and into the pump. The pump then pushes the fluid mechanically into the fluid power actuated devices such as a motor or a cylinder.

Pumps are classified into three different ways and must be considered in any discussion of fluid power equipment.

1. **Classification based on displacement:**

- Non-positive displacement pumps (hydrodynamic pumps).
- Positive displacement pumps (hydrostatic pumps).

2. **Classification based on delivery:**

- Constant delivery pumps.
- Variable delivery pumps.

3. **Classification based on motion:**

- Rotary pump.
- Reciprocating pump.

Types of positive displacement pumps

- Gear pump (fixed displacement)
  - internal gear (gerotor)
  - external gear

- Vane pump
  - fixed or variable displacement
  - pressure compensated

- Piston pump
  - axial design
  - radial design

External gear pump

- Driving gear and driven gear
- Inlet fluid flow is trapped between the rotating gear teeth and the housing
- The fluid is carried around the outside of the gears to the outlet side of the pump
- As the fluid can not seep back along the path it came nor between the engaged gear teeth (they create a seal,) it must exit the outlet port.

Advantages and disadvantages of gear pumps

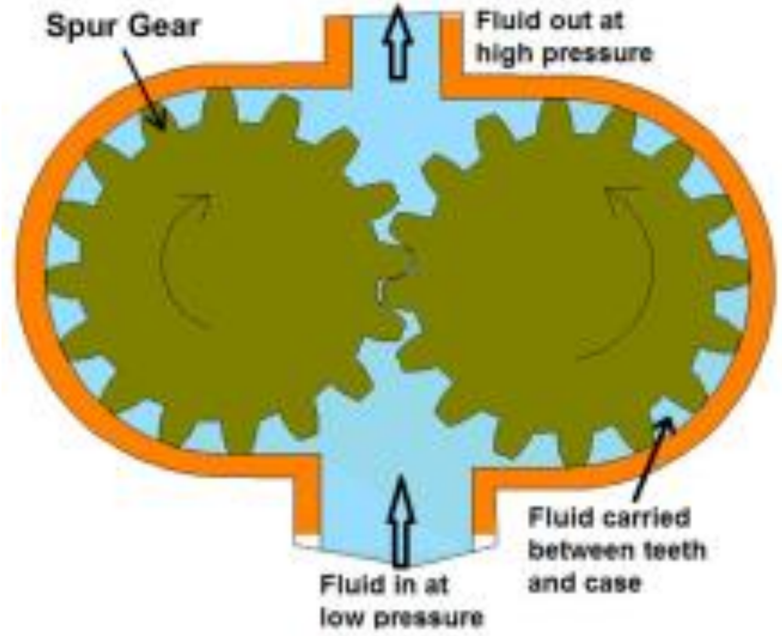


Figure 5.2.1 Gear pump

The advantages are as follows:

1. They are self-priming.
2. They give constant delivery for a given speed.
3. They are compact and light in weight.
4. Volumetric efficiency is high.

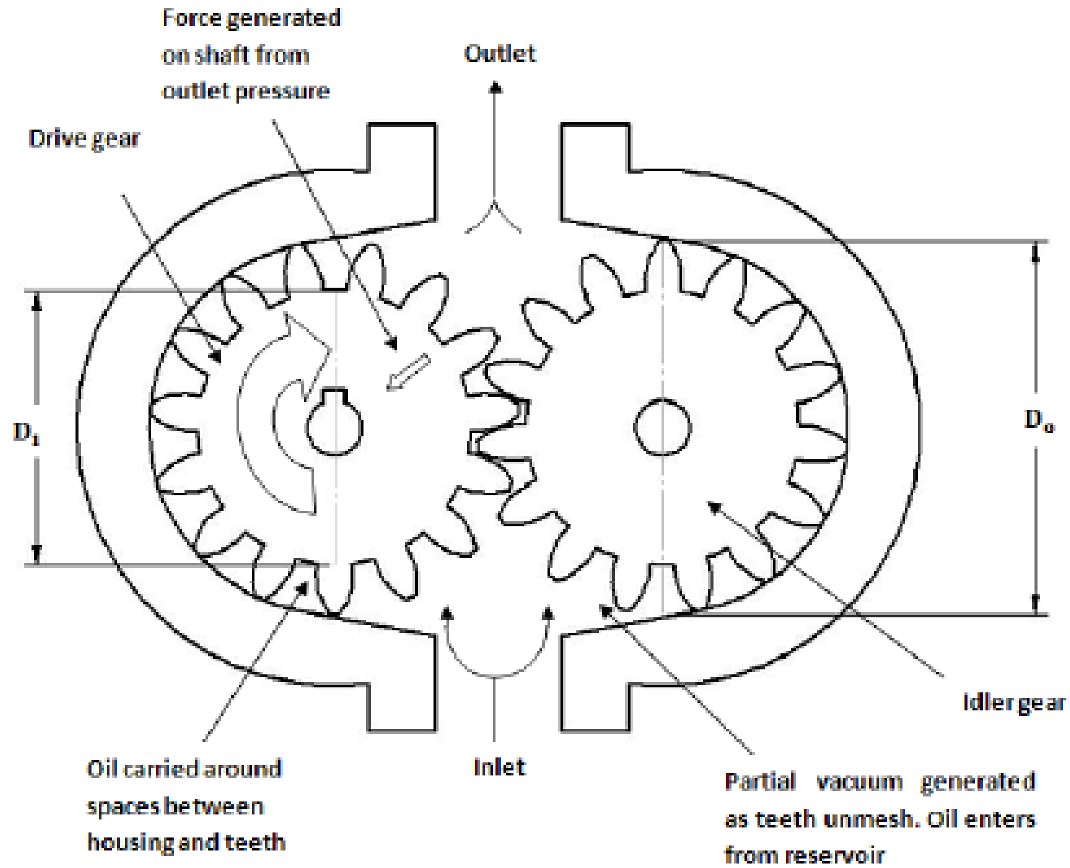
The disadvantages are as follows:

1. The liquid to be pumped must be clean, otherwis
2. Variable speed drives are required to change the delivery.
3. If they run dry, parts can be damaged because the fluid to be pumped is used as lubricant.

Generally gear pumps are used to pump:

- Petrochemicals: Pure or filled bitumen, pitch, diesel oil, crude oil, lube oil etc.
- Chemicals: Sodium silicate, acids, plastics, mixed chemicals, isocyanates etc.
- Paint and ink
- Resins and adhesives
- Pulp and paper: acid, soap, lye, black liquor, kaolin, lime, latex, sludge etc.
- Food: Chocolate, cacao butter, fillers, sugar, vegetable fats and oils, molasses, animal food etc.

D_o =the outside diameter of gear teeth
D_i= the inside diameter of gear teeth
L =the width of gear teeth
N=the speed of pump in RPM
V_D=the displacement of pump in m/rev
M= module of gear
z=number of gear teeth
α= pressure angle



Volume displacement is

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

$$D_i = D_o - 2(\text{Addendum} + \text{Dendendum})$$

Theoretical discharge is

$$Q_T\ (\text{m}^3/\text{min}) = V_D\ (\text{m}^3/\text{rev}) \times N\ (\text{rev/min})$$

If the gear is specified by its module and number of teeth, then the theoretical discharge can be found by

$$Q_T = 2\pi L m^2 N\left[z + \left(1 + \frac{\pi^2 \cos^2 20}{12}\right)\right]\ \text{m}^3/\text{min}$$

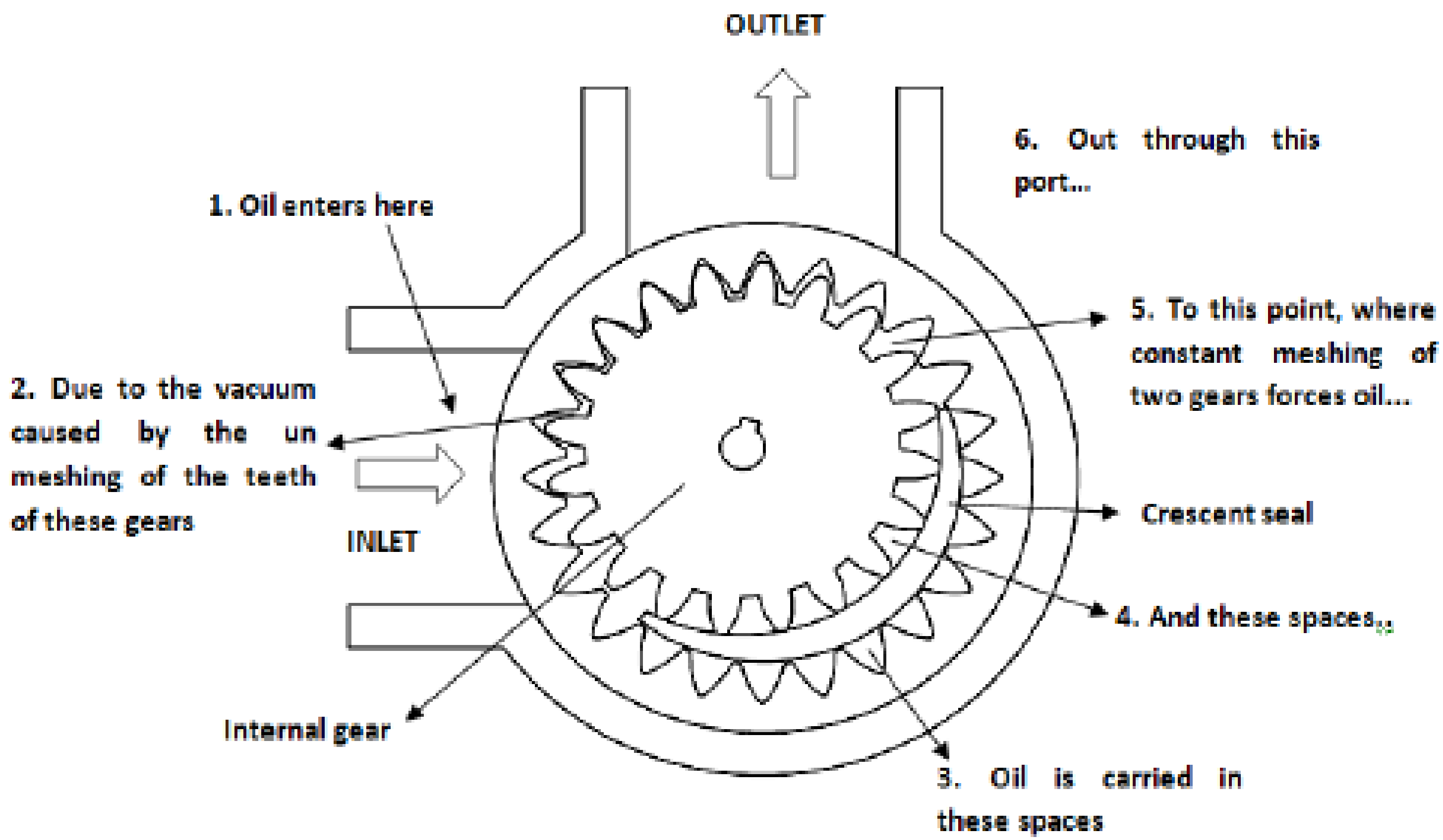


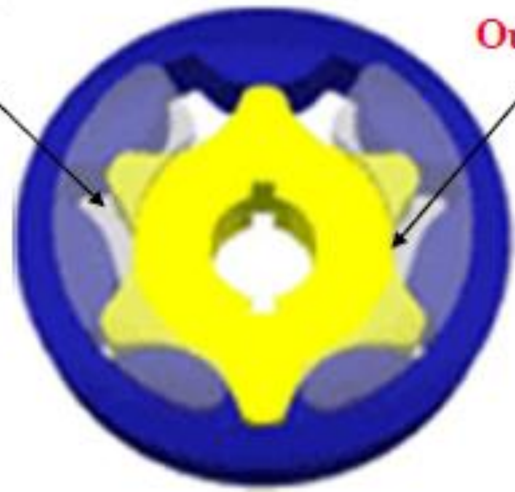
Figure 1.5Operation of an internal gear pump

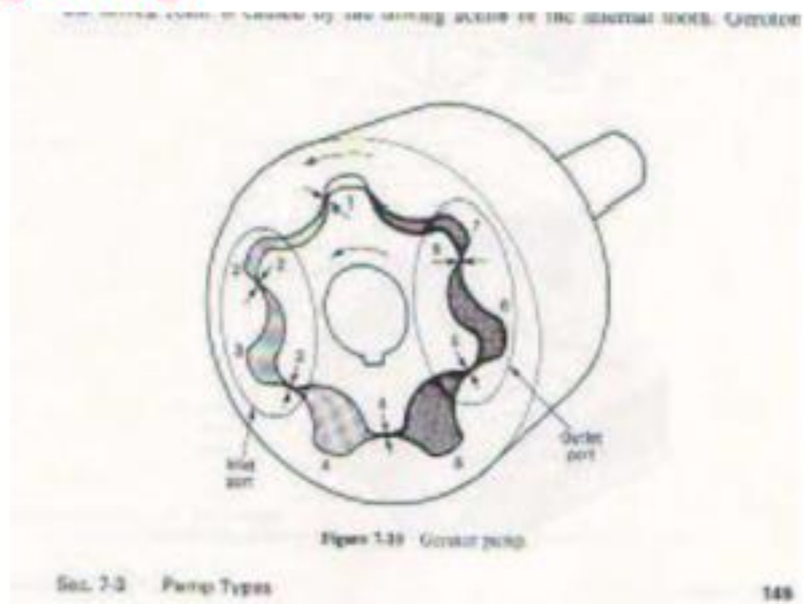
Applications

Some common internal gear pump applications are:

- All varieties of fuel oil and lube oil
- Resins and Polymers
- Alcohols and solvents
- Asphalt, Bitumen, and Tar
- Polyurethane foam (Isocyanate and polyol)
- Food products such as corn syrup, chocolate, and peanut butter
- Paint, inks, and pigments
- Soaps and surfactants
- Glycol

Gerotor pump



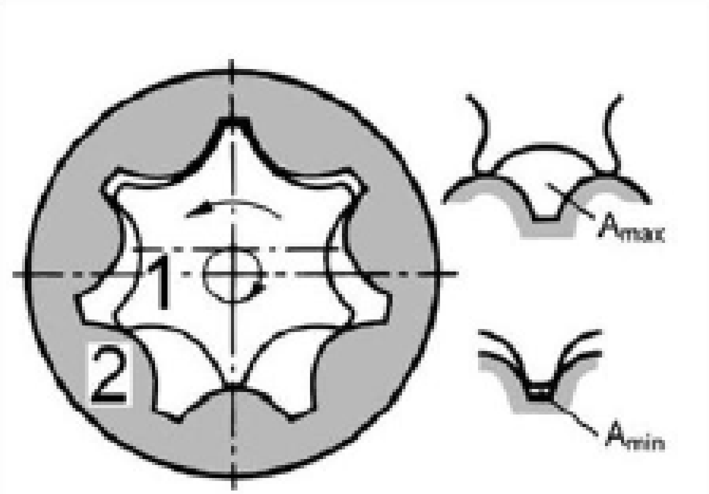


- Inner gerotor is slightly offset from external gear
- Gerotor has 1 fewer teeth than outer gear
  - Gerotor rotates slightly faster than outer gear
- Displacement = (roughly) volume of missing tooth
  - Pockets increase and decrease in volume corresponding to filling and pumping
- Lower pressure application: <138 bar
- Displacements (determined by length): 0.002 to 0.2 L

center. During the second half of the revolution, the spaces collapse, displacing the fluid to the outlet port formed at the side plate. The geometric volume of the gerotor pump is given as

$$V_D = b\,Z\,(A_{max} - A_{min})$$

where b is the tooth height, Z is the number of rotor teeth, A_{max} is the maximum area between male and female gears (unmeshed – occurs at inlet) and A_{min} is the minimum area between male and female gears (meshed – occurs at outlet).

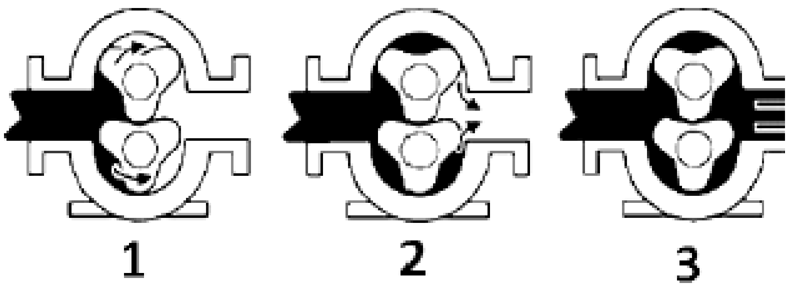


Applications

Gerotors are widely used in industries and are produced in variety of shapes and sizes by a number of different methods. These pumps are primarily suitable for low pressure applications such as lubrication systems or hot oil filtration systems, but can also be found in low to moderate pressure hydraulic applications. However common applications are as follows:

- Light fuel oils
- Lube oil
- Cooking oils
- Hydraulic fluid

Stages of operation of Lobe pump



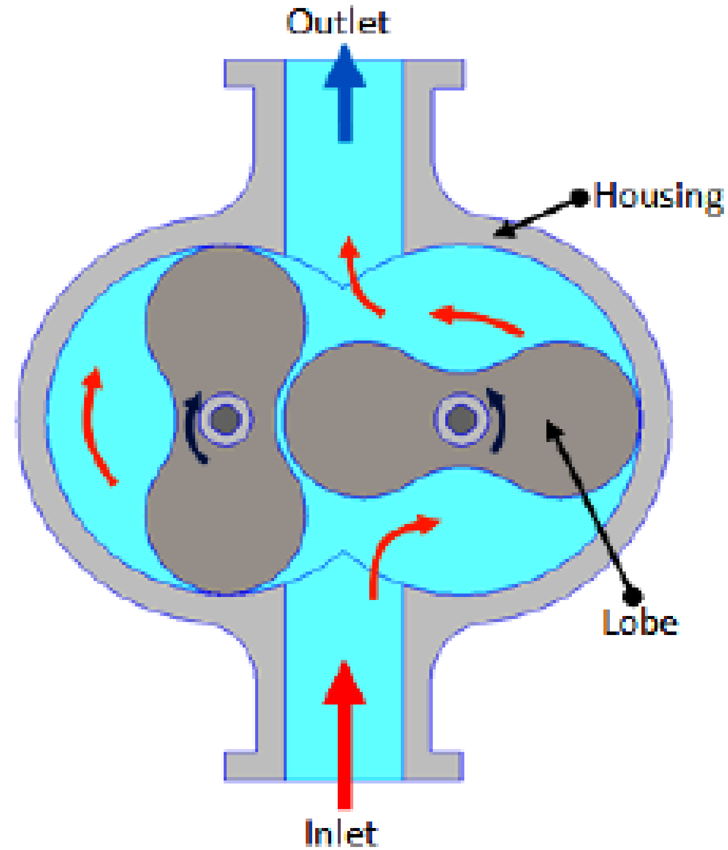


Figure 5.2.3 Lobe pump

Because of superb sanitary qualities, high efficiency, reliability, corrosion resistance and good clean-in-place and steam-in-place (CIP/SIP) characteristics, Lobe pumps are widely used in industries such as pulp and paper, chemical, food, beverage, pharmaceutical and biotechnology etc. These pumps can handle solids (e.g., cherries and olives), slurries, pastes, and a variety of liquids. A gentle pumping action minimizes product degradation. They also offer continuous and intermittent reversible flows. Flow is relatively independent of changes in process pressure and therefore, the output is constant and continuous.

1.5.1Advantages

The advantages of lobe pumps are as follows:

1. Lobe pumps can handle solids, slurries, pastes and many liquid.
2. No metal-to-metal contact.
3. Superior CIP(Cleaning in Place) /SIP(Sterilization in Place) capabilities.
4. Long-term dry run (with lubrication to seals).
5. Non-pulsating discharge.

1.5.2Disadvantages

The disadvantages of lobe pumps are as follows:

1. Require timing gears.
2. Require two seals.
3. Reduced lift with thin liquids.

1.5.3Applications

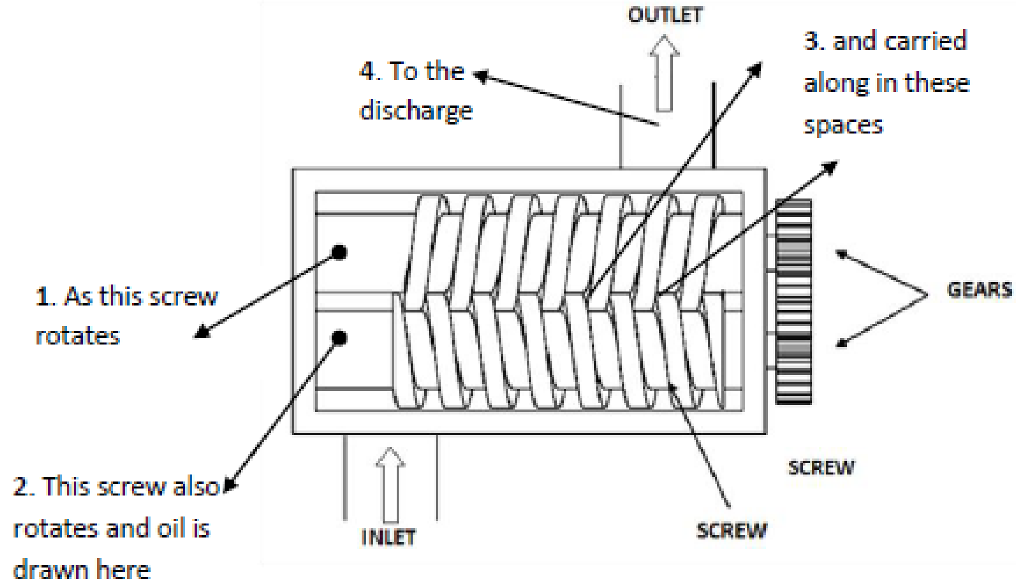
Common rotary lobe pump applications include, but are not limited to, the following:

1. Polymers.
2. Paper coatings.
3. Soaps and surfactants.
4. Paints and dyes.
5. Rubber and adhesives.
6. Pharmaceuticals.
7. Food applications.

SCREW PUMP

These pumps have two or more gear-driven helical meshing screws in a close fitting case to develop the desired pressure. These screws mesh to form a fluid-type seal between the screws and casing

When the screws turn, the space between the threads is divided into compartments. As the screws rotate, the inlet side of the pump is flooded with hydraulic fluid because of partial vacuum.



Ina screw pump, a chamber is formed between thread and housing as shown in Fig.1.11. The following expression gives the volumetric displacement

$$V_D = \frac{\pi}{4}(D^2 - d^2)s - D^2\left\{\frac{\alpha}{2} - \frac{\sin 2\alpha}{2}\right\}s$$

Heres is the stroke length and $\cos(\alpha) = \dfrac{D+d}{2D}$

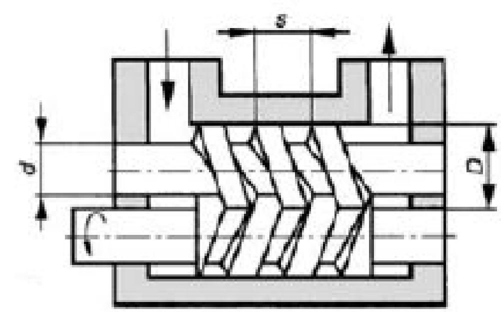


Figure 1.11 Volumetric displacement of a screw pump

Here the fluid does not rotate but moves linearly as a nut on threads. Thus, there are no pulsations at a higher speed; it is a very quiet operating

Advantages and disadvantages of screw pump

Theadvantages are as follows:

1. They are self-priming and more reliable.
2. They are quiet due to rolling action of screw spindles.
3. They can handle liquids containing gases and vapor.
4. They have long service life.

The disadvantages are as follows:

1. They are bulky and heavy.
2. They are sensitive to viscosity changes of the fluid.
3. They have low volumetric and mechanical efficiencies.
4. Manufacturing cost of precision screw is high.

ME 5451 – Hydraulics and Pneumatics

Lecture -4

Date: 04-03-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Refreshing lec-3
2. Gerotor, Lobe and Screw Pump
3. Vane Pump- Fixed and Variable displacement
4. Piston Pump- Fixed and Variable displacement
5. Efficiency –Comparison of pumps
6. Performance curves
7. Problems

Course Instructor: Dr. A. Siddharthan

Vane Pumps

1. **Unbalanced vane pump:** Unbalanced vane pumps are of two varieties:

- Unbalanced vane pump with fixed delivery.
- Unbalanced vane pump with pressure-compensated variable delivery.

2. **Balanced vane pump.**

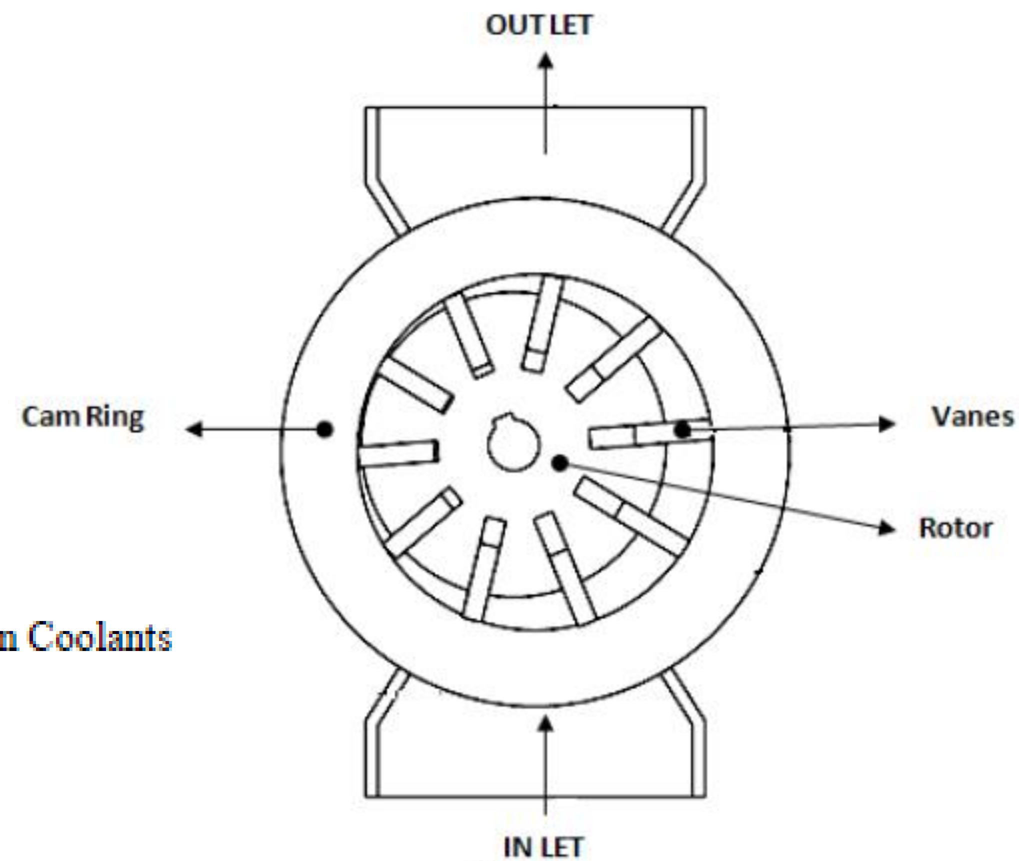
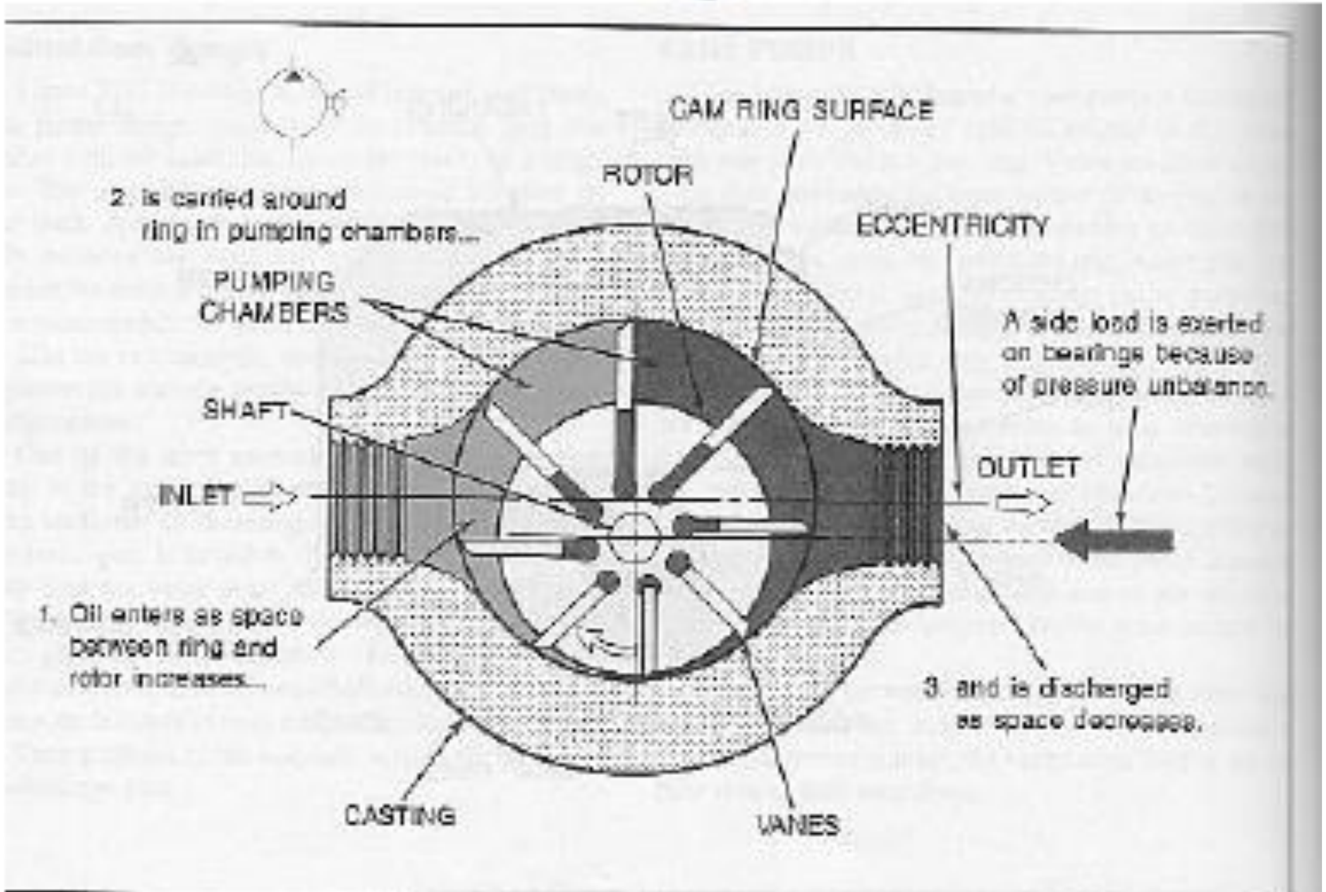


Figure 1.12 Simple vane pump

applications for the pumping of following fluids:

- Aerosol and Propellants
- Aviation Service - Fuel Transfer, Deicing
- Auto Industry - Fuels, Lubes, Refrigeration Coolants
- Bulk Transfer of LPG and NH3
- LPG Cylinder Filling
- Alcohols
- Refrigeration - Freons, Ammonia
- Solvents
- Aqueous solutions



CAM RING SURFACE
ROTOR
2. is carried around
ring in pumping chambers...
ECCENTRICITY
PUMPING
CHAMBERS
A side load is exerted
on bearings because
of pressure unbalance.
SHAFT
OUTLET
INLET
1. Oil enters as space
between ring and
rotor increases...
3. and is discharged
as space decreases.
CASTING
VANES

1.7.4 Expression for the Theoretical Discharge of Vane Pumps

Let D_C be the diameter ofa cam ring in m, $D_R$the diameter of rotor in m, L the width of rotor in m, e the eccentricity in m, V_D the pump volume displacement in m^3/rev and e_{max} the maximum possible eccentricity in m.

From geometry (Fig.1.13) the maximum possible eccentricity,

$$e_{\max} = \frac{D_C - D_R}{2} \qquad (1.1)$$

Themaximum value of eccentricity produces the maximum volumetric displacement

$$V_{D(\max)} = \frac{\pi}{4}(D_C^2 - D_R^2)L \qquad (1.2)$$

Using Equation (1.1), Equation (1.2) can be simplified as

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

<

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

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

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

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

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

Theoretical discharge

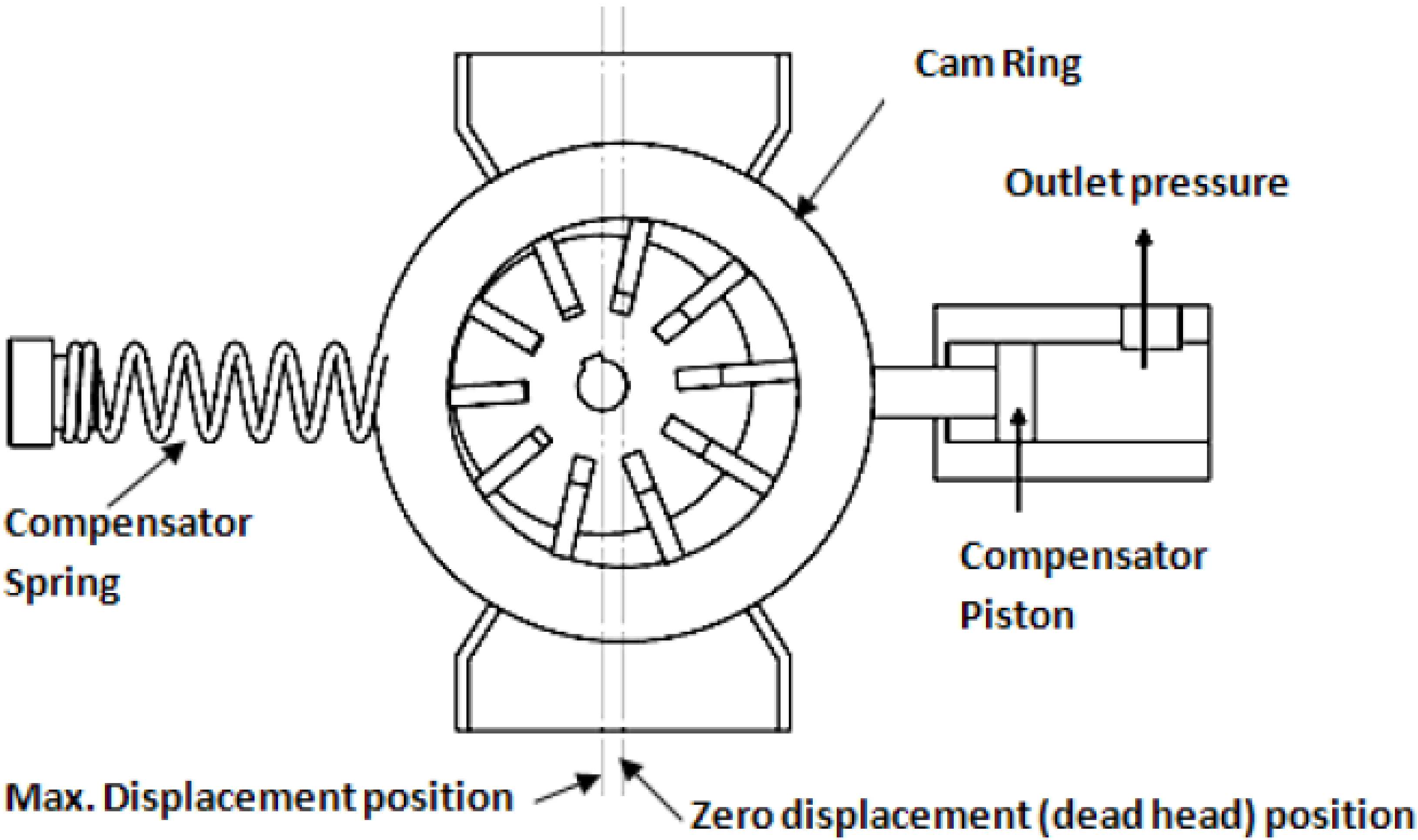


Figure 1.14 Operation of a variable displacement vane pump

Pressure Compensated Vane Pump

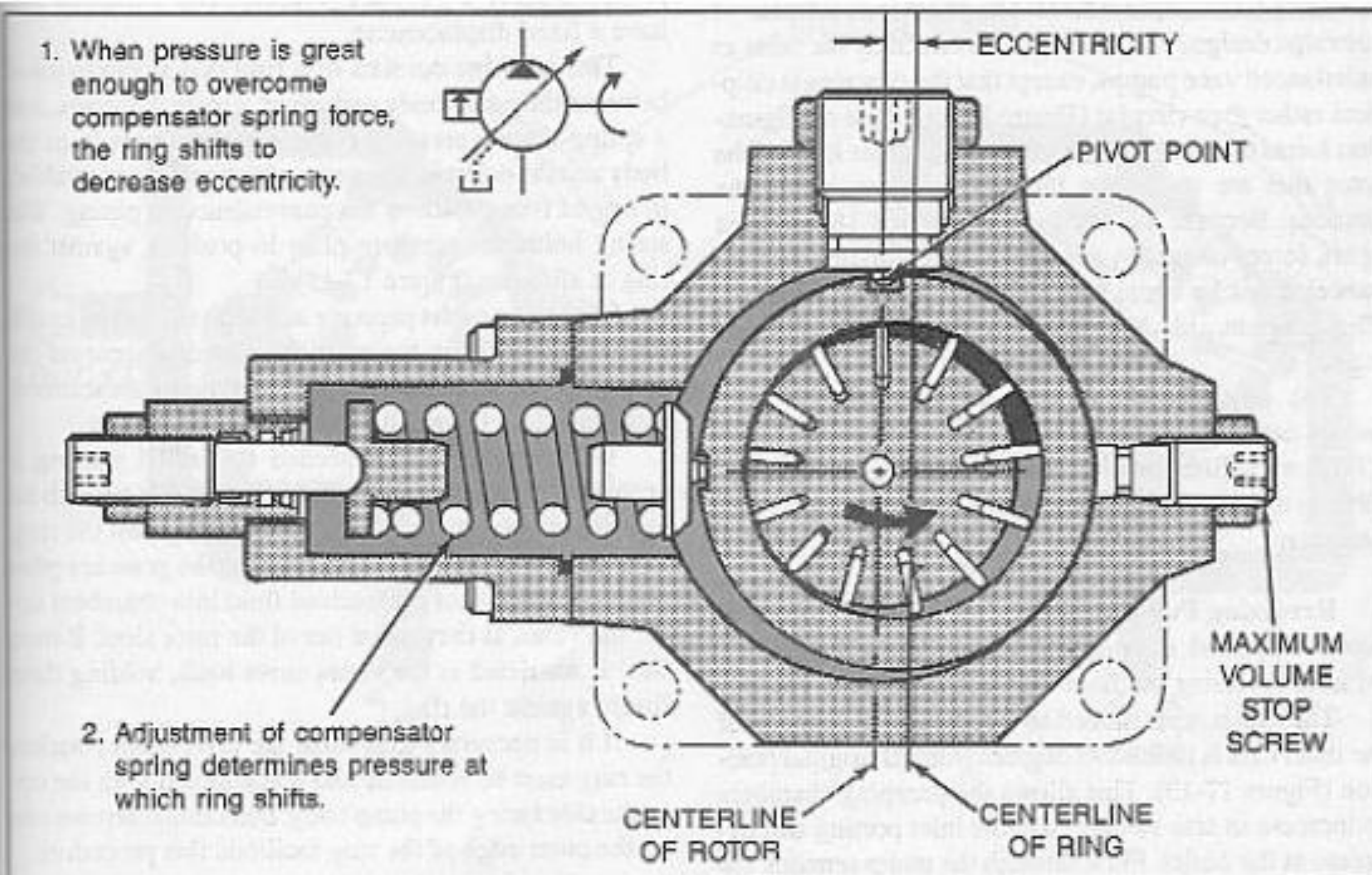


Figure 17-10. Unbalanced variable displacement vane pump (pressure compensated).

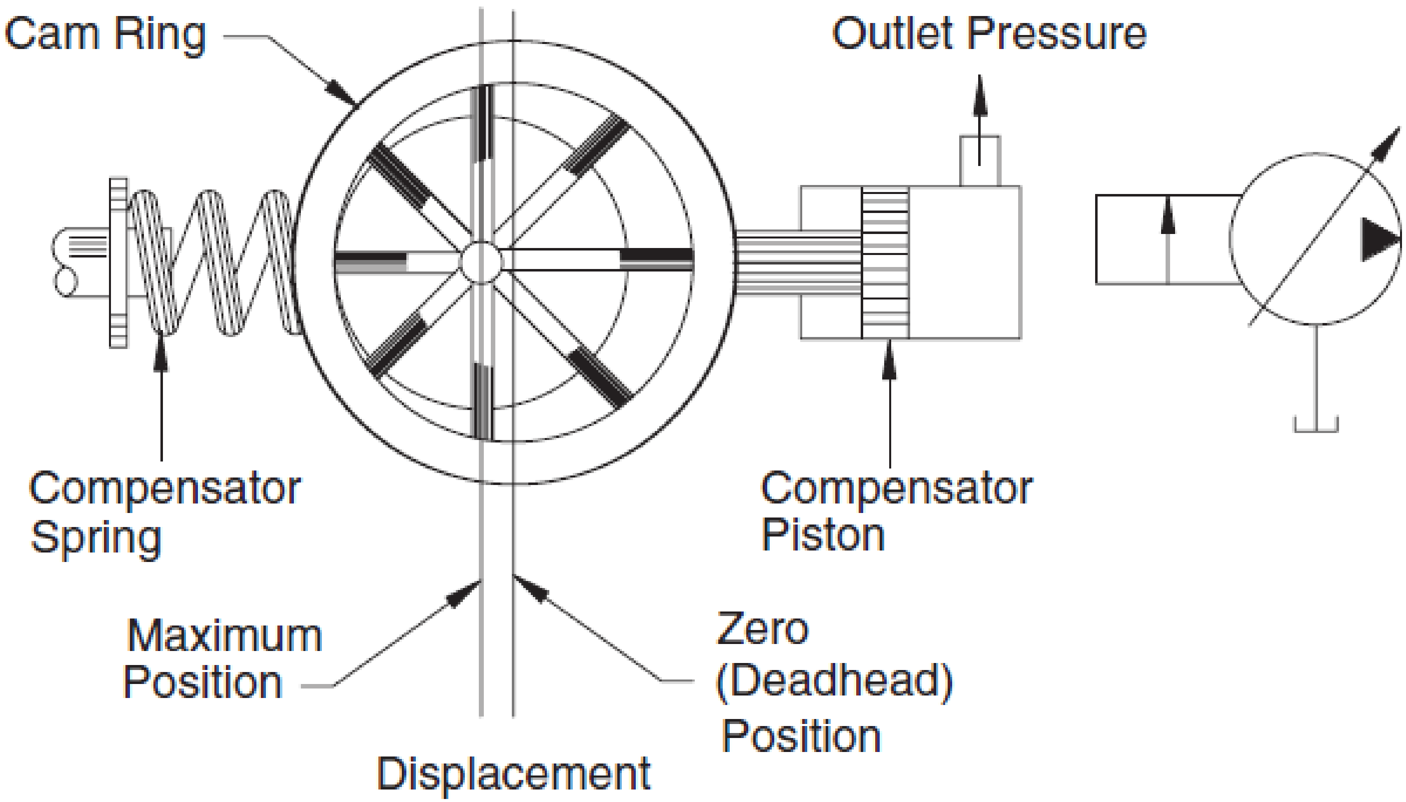


FIGURE 4.9
Pressure-compensated vane pump.

The advantages of vane pumps are as follows:

1. Vane pumps are self-priming, robust and supply constant delivery at a given speed.
2. They provide uniform discharge with negligible pulsations.
3. Their vanes are self-compensating for wear and vanes can be replaced easily.
4. These pumps do not require check valves.
5. They are light in weight and compact.
6. They can handle liquids containing vapors and gases.
7. Volumetric and overall efficiencies are high.
8. Discharge is less sensitive to changes in viscosity and pressure variations.

The disadvantages of vane pumps are as follows:

1. Relief valves are required to protect the pump in case of sudden closure of delivery.
2. They are not suitable for abrasive liquids.
3. They require good seals.
4. They require good filtration systems and foreign particle can severely damage pump.

2. Balanced vane pump.

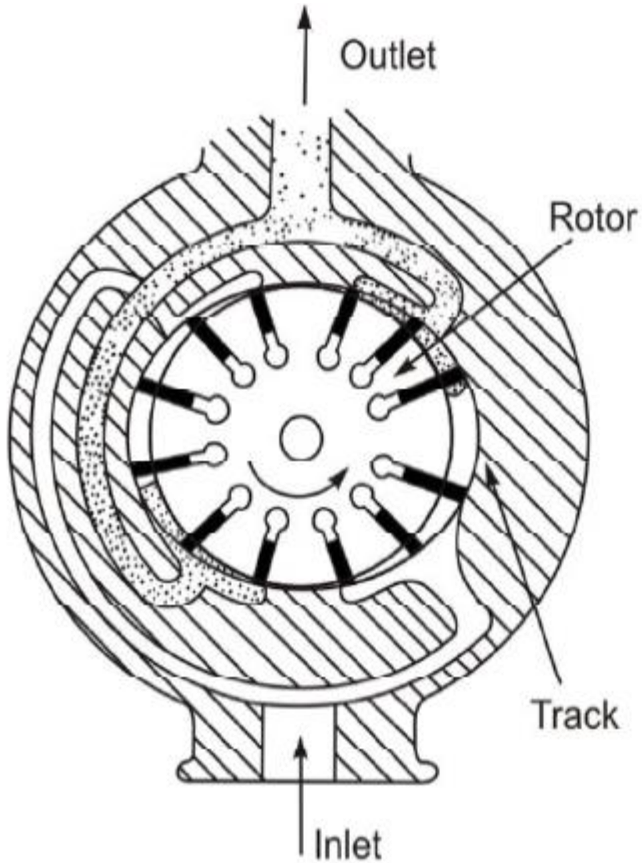


Fig. 5.18-2 : Balanced Vane type pump or motor
With Double Eccentric Rotor.

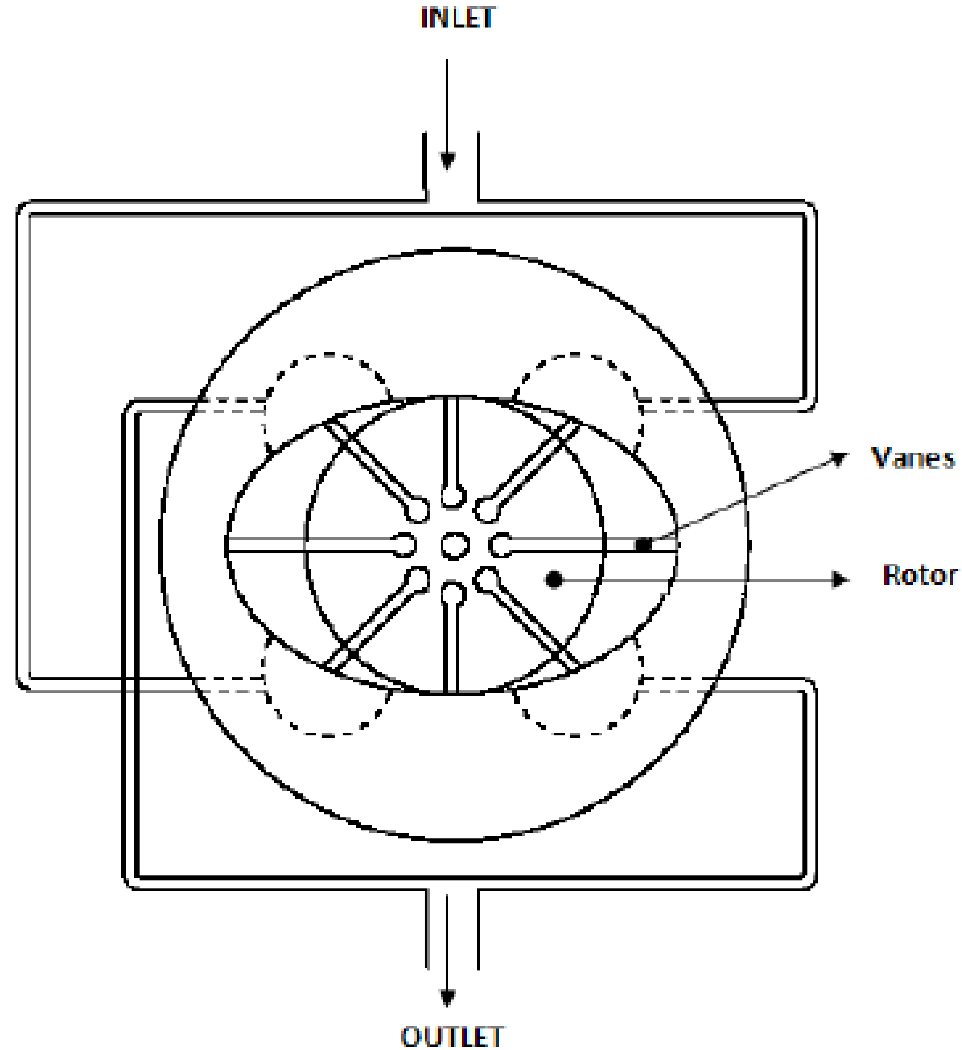


Figure 1.15 Operation of a balanced vane pump

Advantages and disadvantagesofbalancedvane pumps

The advantages of balanced vane pumps are as follows:

1. The balanced pump eliminates the bearing side loads and therefore high operating pressure can be used.
2. The service life is high compared to unbalanced type due to less wear and tear.

The disadvantages of balanced vane pumps are as follows:

1. They are fixed displacement pumps.
2. Design is more complicated.
3. Manufacturing cost is high compared to unbalanced type.

1.8Piston Pumps

Piston pumps are of the following two types:

1. **Axial piston pump:**These pumps are of two designs:

    - Bent-axis-type piston pump.
    - Swash-plate-type piston pump.

2. **Radial piston pump.**

Axial Piston Pumps :

Fig. 5.18-4 shows schematic view of an axial piston hydrostatic unit. In pump version the ba
with pistons, equispaced on a pitch circle, rotates on a central shaft housed in a casing, while
piston ends slides on a stationary inclined plate, called as *swash plate*. With a mechanism the
ends are always kept in touch with the swash plate. As a result the pistons reciprocate generat
suction and compression of volumes. The oil in and oil out are executed through the kidney p
plate, fixed to the housing with a force contact to the barrel in opposite side of the swash plat
unit can be made variable displacement by varying the swash plate angle α . As the radial rea
load acts on piston at piston-barrel interface α is usually kept within 25 degree.

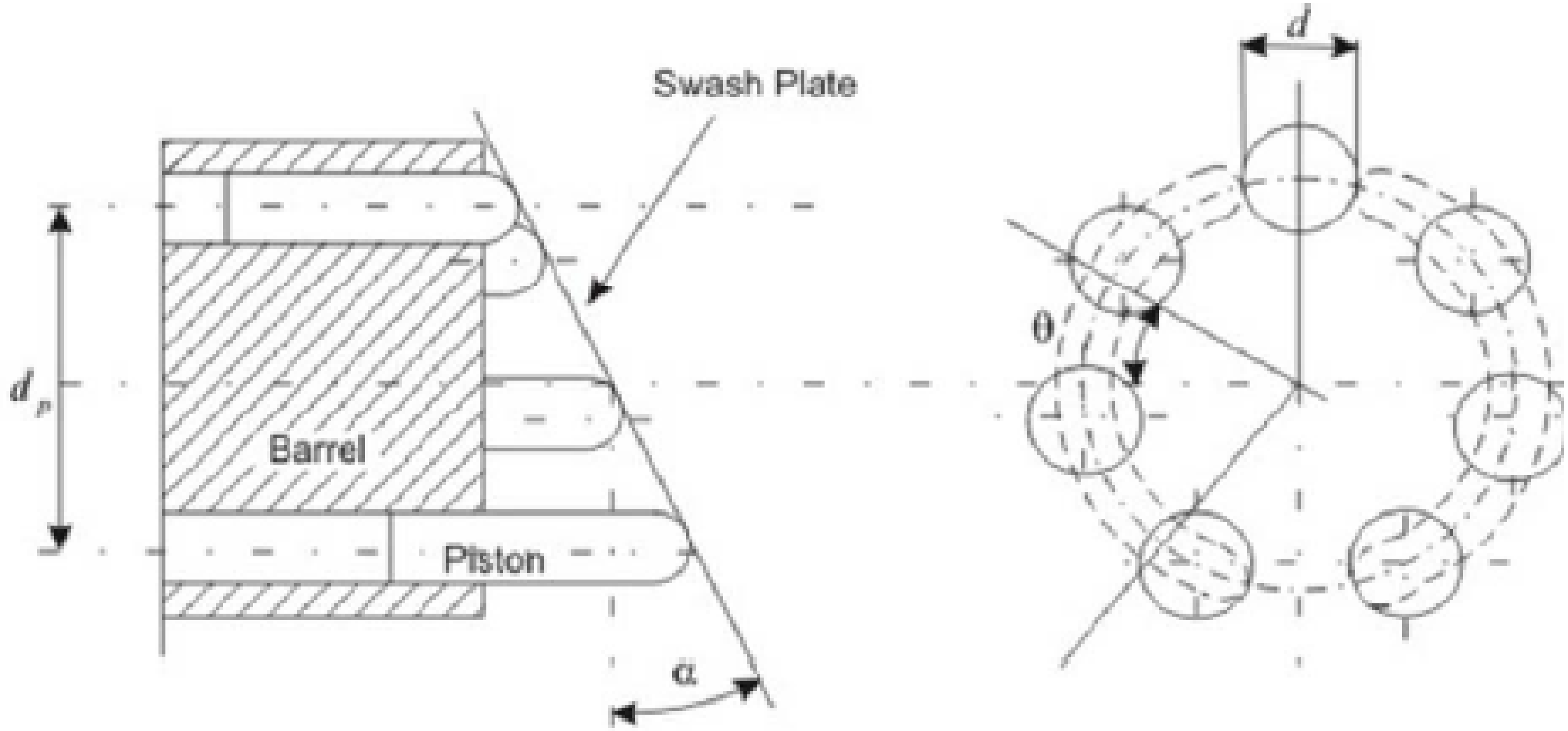
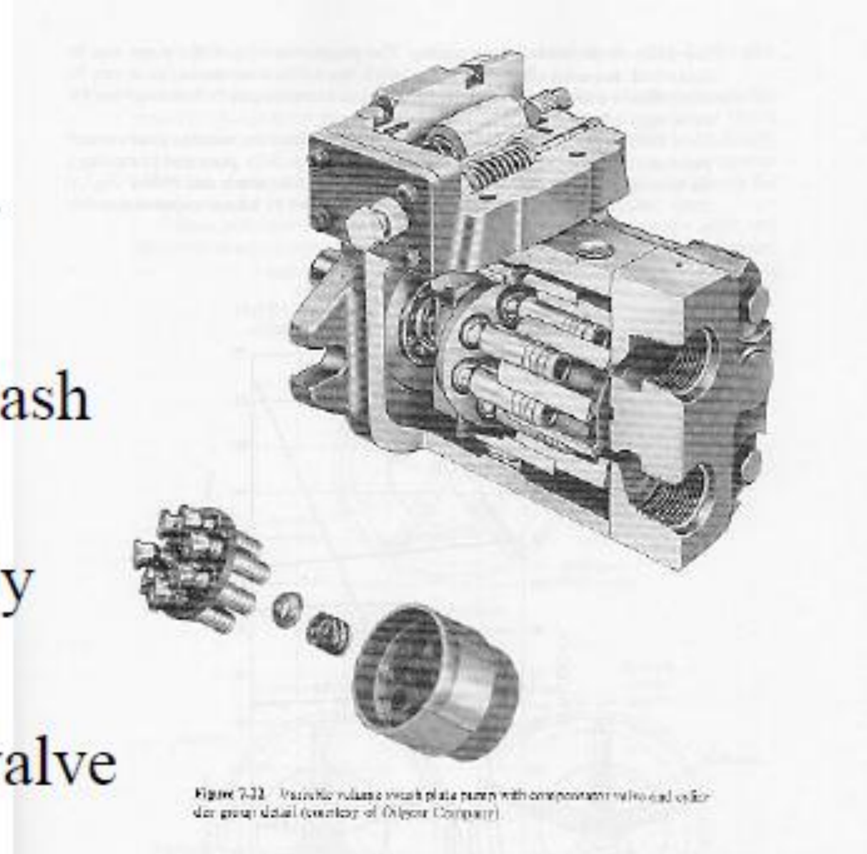


Fig. 5.18-4 : Axial Piston Pump/Motor (Schematic View).

Axial Piston Pump

- Pistons rotate with cylinder block
- Pistons translate against swash plate
- Displacement determined by swash plate angle
- Fluid enters/exits through valve plate



The swept volume D_p (Geometric Displacement per revolution) can be expressed as:

$$D_p = n\frac{\pi d^2}{4} d_p \tan\alpha \qquad \ldots 5.18\text{-}4$$

where d_p the pitch circle diameter on which the piston are laid

d = diameter of the piston.

α = swash plate tilt angle

n = number of piston

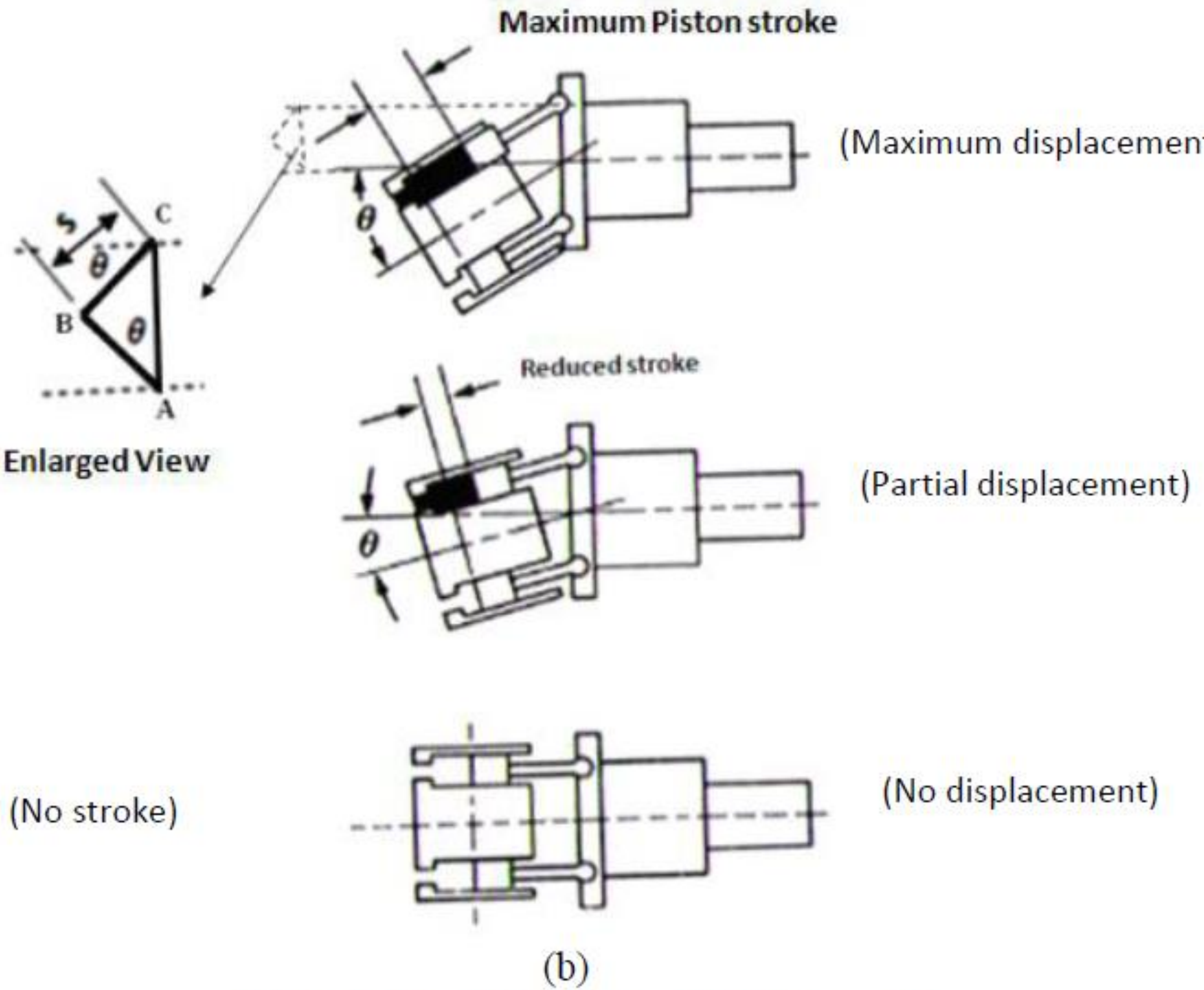


Figure 1.19 Stroke changes with offset angle

From a right-angled triangle ABC [Fig. 1.19(b)]

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

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

The displacement volume of one piston = AS m^3

Total displacement volume of Y number of pistons = YAS m^3

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

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

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

Theoretical flow rate is

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

Bend Axis Pumps

In bend axis units (pump or motor) the axis of the (rotating) barrel can be bent from the rotating shaft axis as shown in Fig. 5.18-8. Contrary to the swash plate type the angle α can be made as high as 45 degree, as no radial load acts on piston at piston barrel interface. Therefore, much more swept volume is available in comparison to axial swash plate type pump. However, valve plate and barrel bending arrangement needs more engineering complicacy, particularly to make this variety as a variable displacement unit.

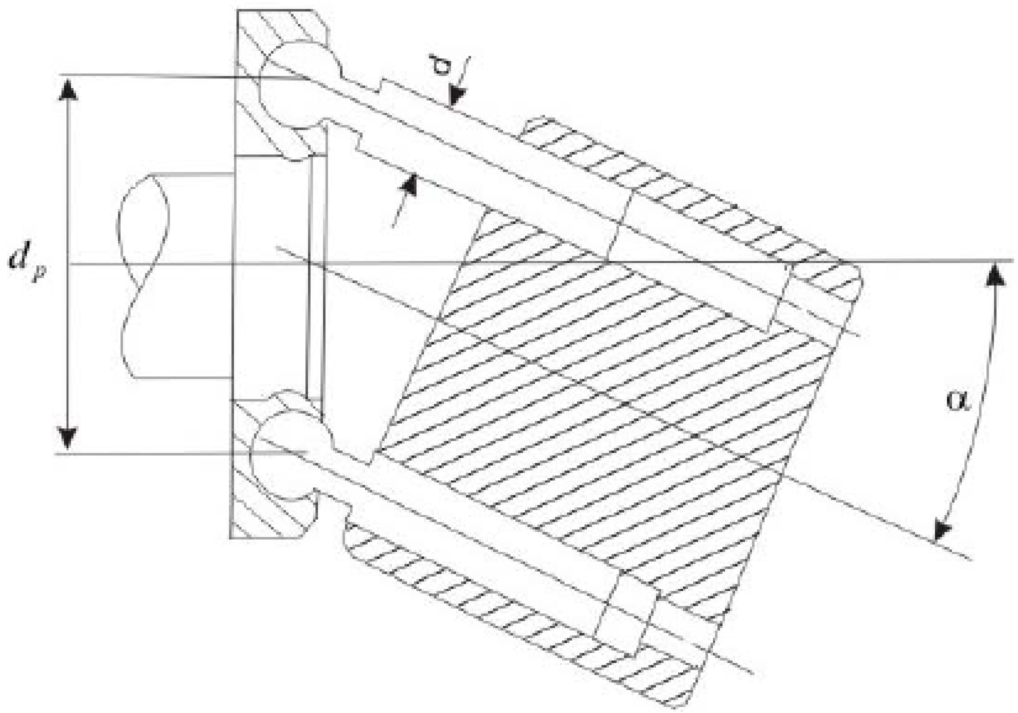
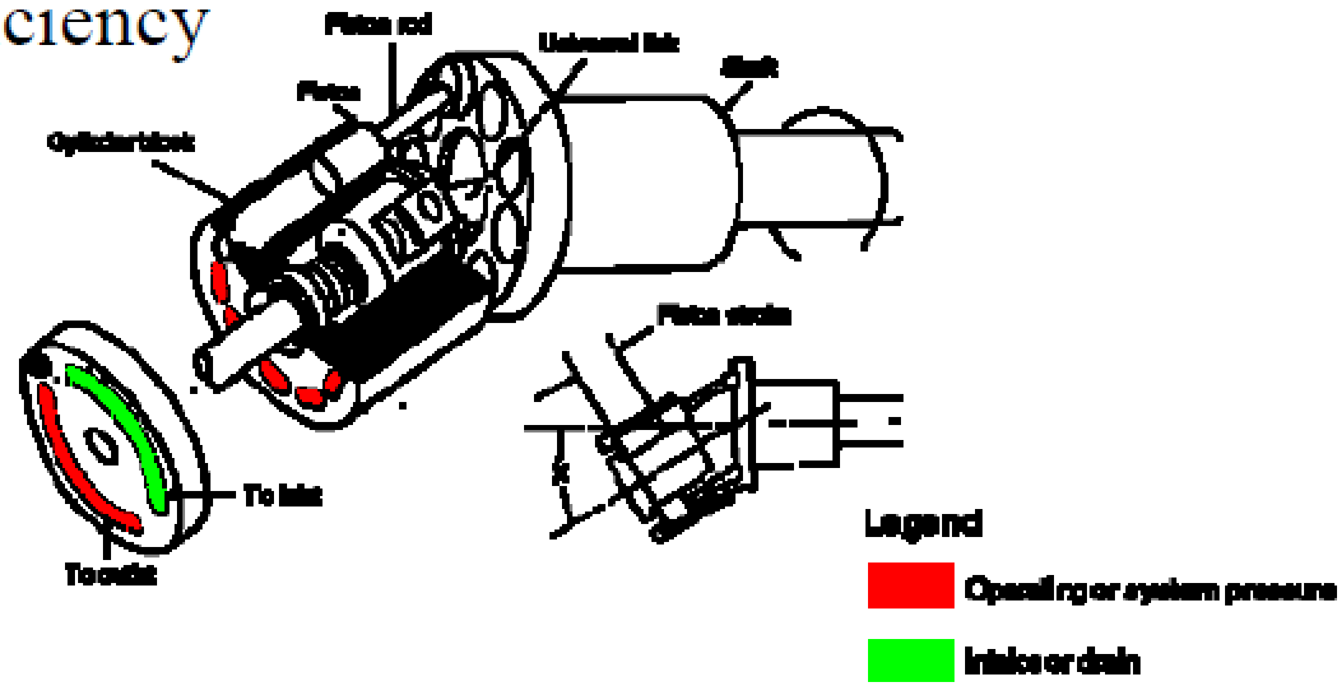


Fig. 5.18-8 : Bend Axis Pump (Schematic View).

Bent Axis Pump

- Drive shaft coupled to cylinder block
- Stationary valve plate
- Low piston side load
- High efficiency



Radial Piston Pump/Motor

Referring to the Fig. 5.18-9, in radial piston unit a cylindrical block having equispaced cylindrical pistons in radial direction is placed in a circular housing eccentrically. While the shaft (placed centrally to the housing and eccentric to the cylindrical block) rotates the piston reciprocate as the ends always touch the casing due to centrifugal force, pressure force and or the spring force. In the cylindrical block valve is placed close to the shaft. Usually such a unit is used as a motor. The piston diameter can be made relatively large in comparison the inline axial piston unit, and the eccentricity is made small. The motor unit can discharge high torques at low speeds. Combining an axial piston pump and a radial piston motor an output of low speed high torque (LSHT) is available for the HST system with high speed low torque input.

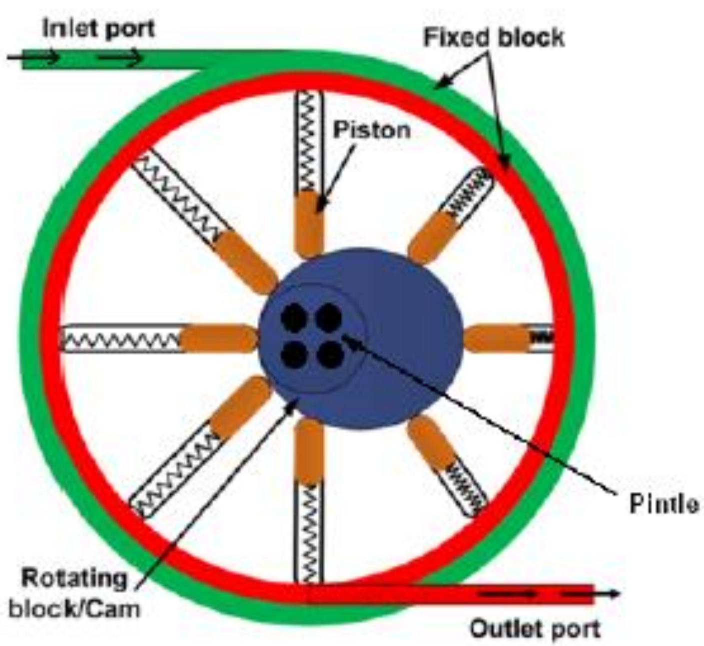


Figure 5.3.7 Radial piston pump

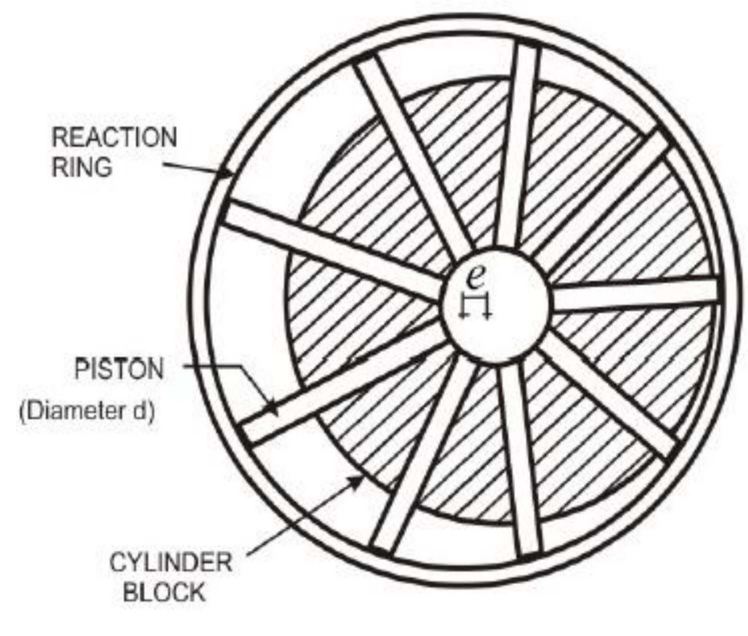
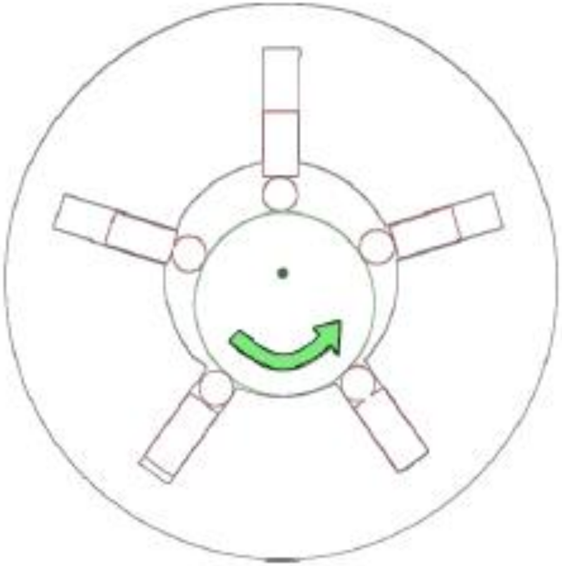


Fig. 5.18-9 : Radial Piston Pump/Motor

Radial Piston Pump

- Cam moves pistons radially
- Displacement determined by cam profile
- Displacement variation can be achieved by moving the cam (not common)
- High pressure capable, and efficient
- Pancake profile



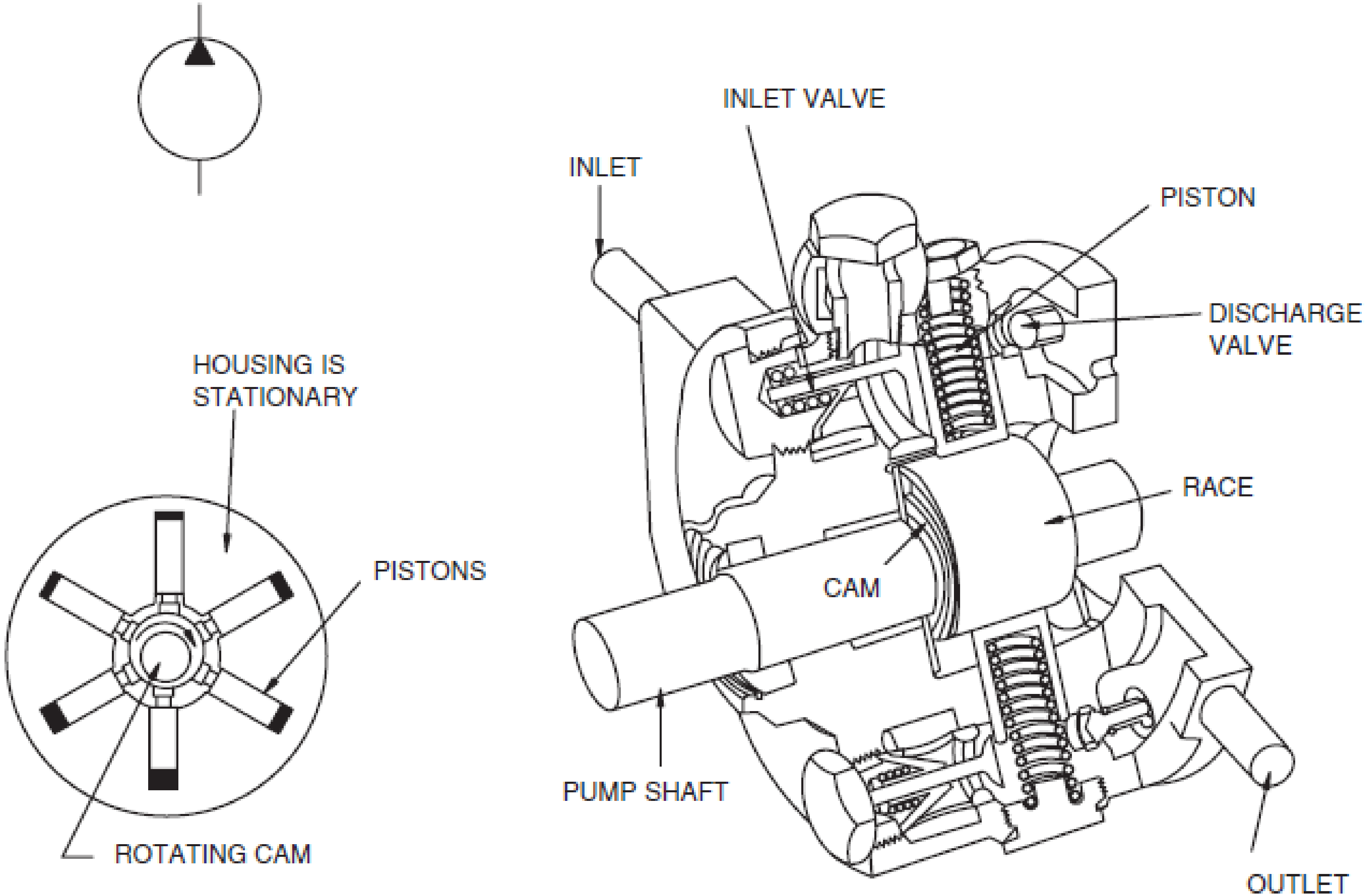


FIGURE 4.16
Schematic of radial piston pump.

Ball Piston type Hydrostatic unit

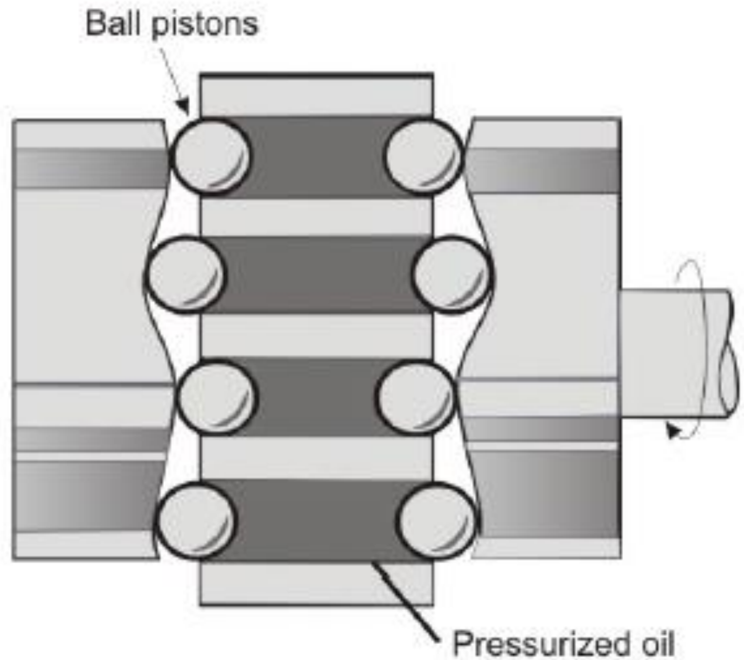


Fig. 5.18-10 : Axial Ball Piston Motor

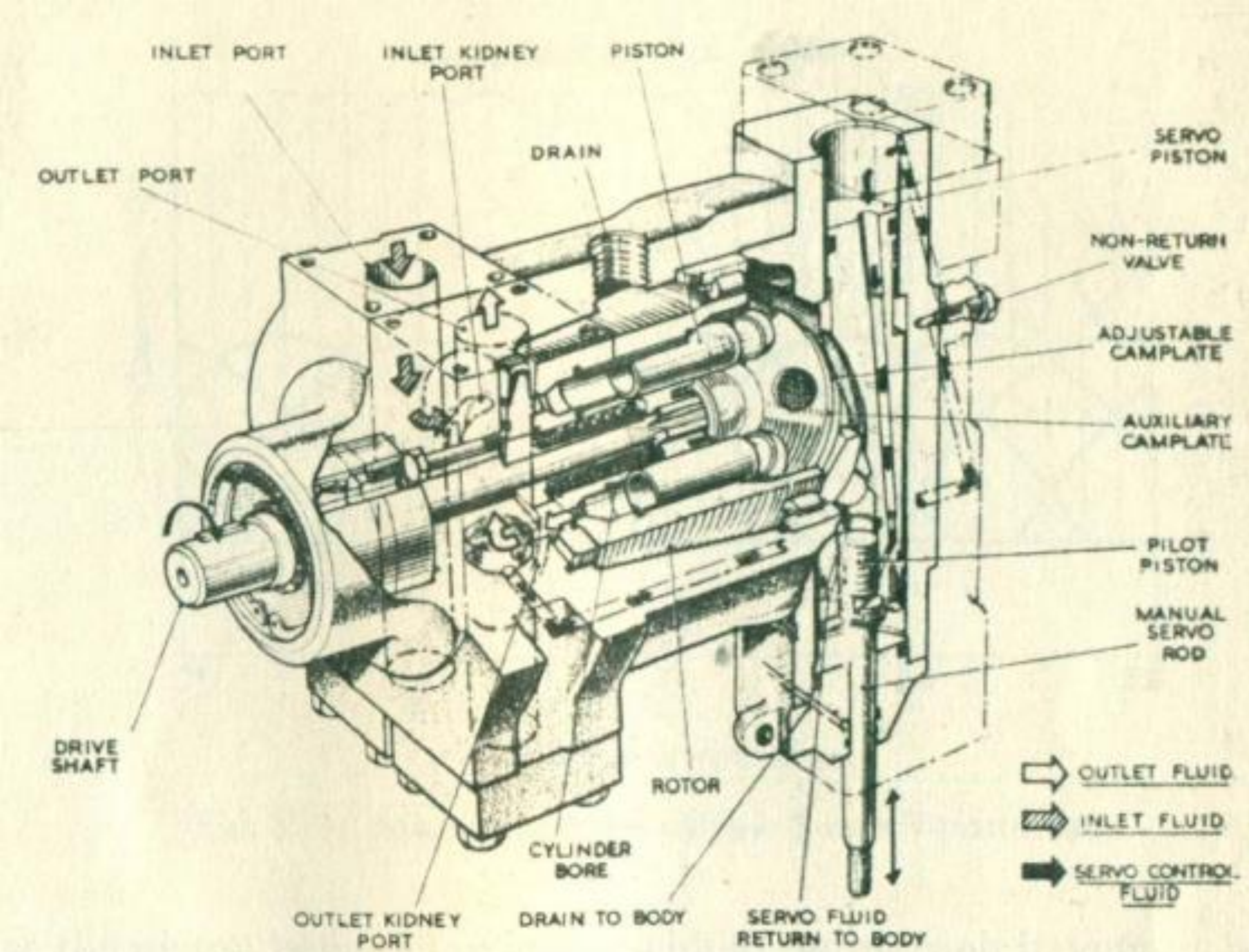


Fig. 5.18-7 : Lucas axial piston pump

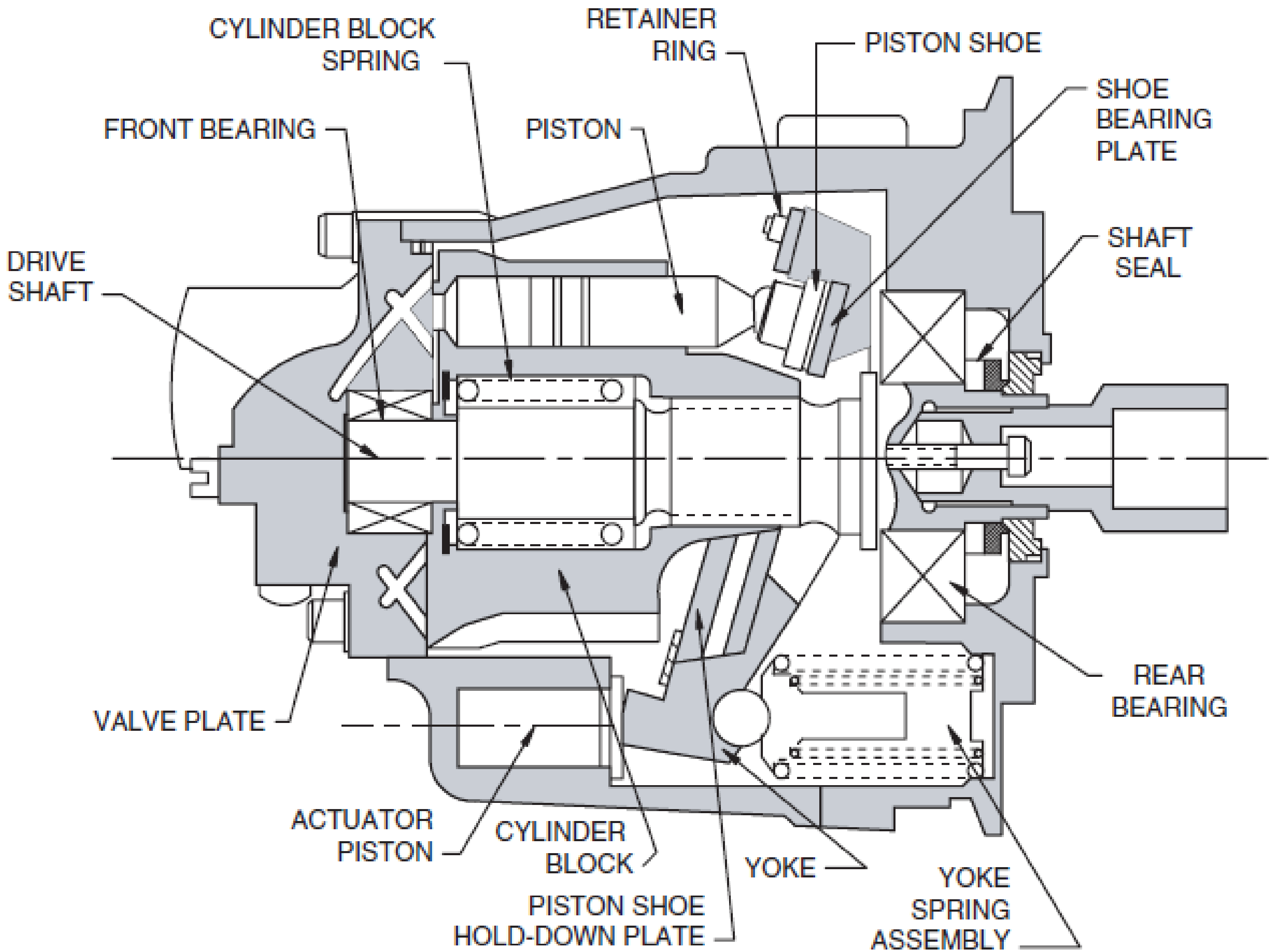


FIGURE 4.14
Implementation of axial piston pump design. (Reprinted with permission from Eaton Hydrau-

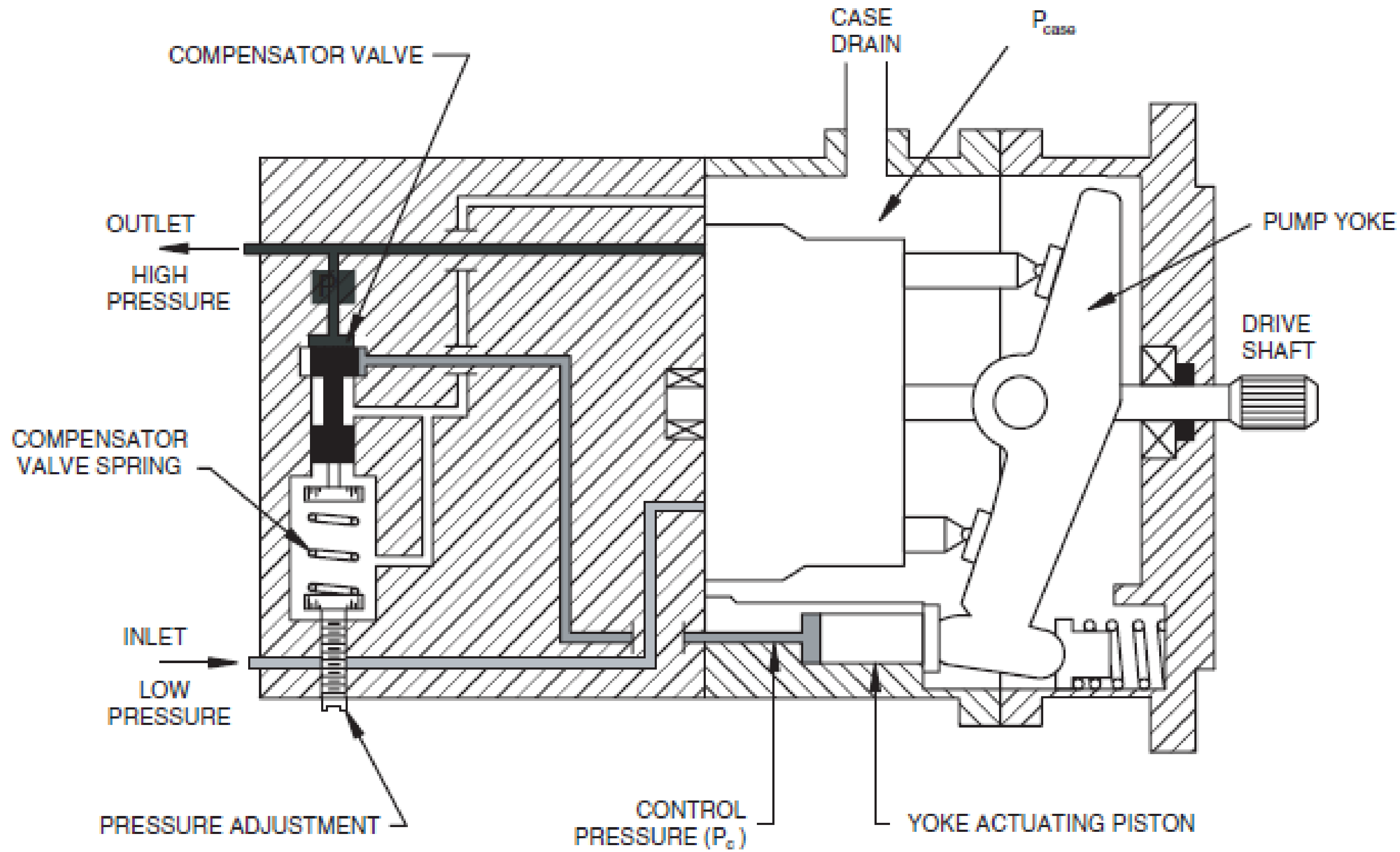


FIGURE 4.15
Schematic showing axial piston pump with pressure compensation.

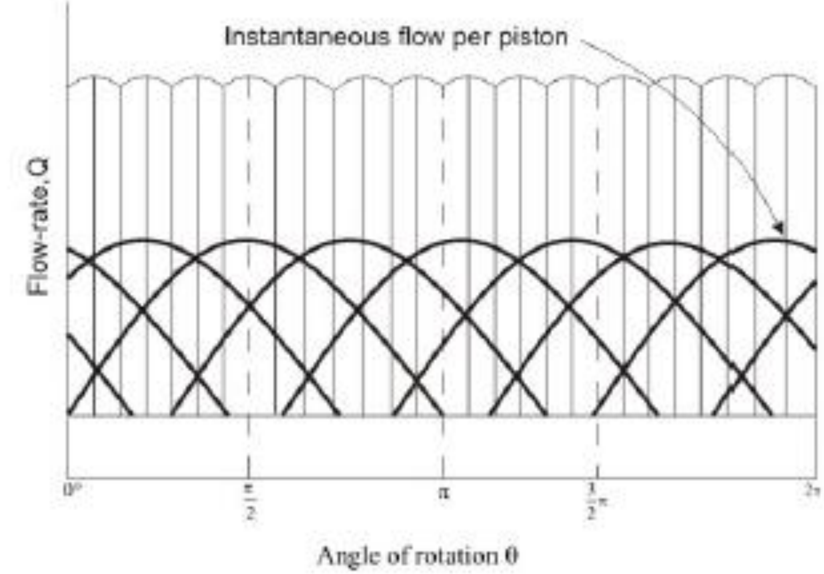


Fig. 5.18-5 : Plot of flow rate of 7 piston pump

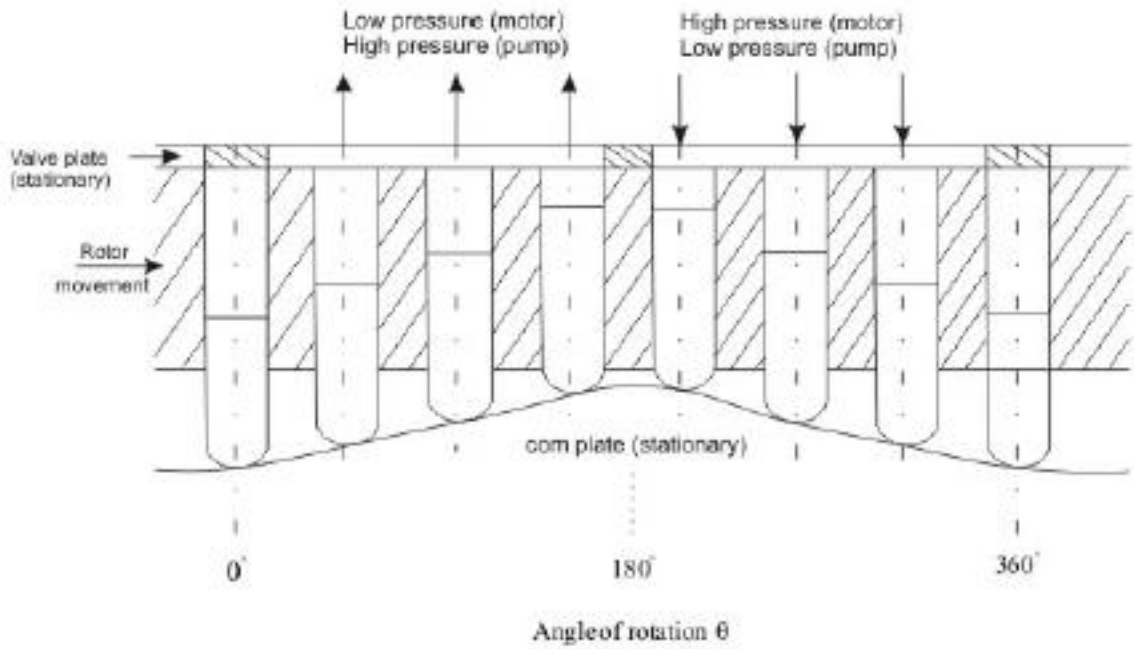


Fig. 5.18-6 : Development of the pitch circle along with all pistons showing the functions of valve plate and swash plate cam.

Comparison of Hydraulic Pumps

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

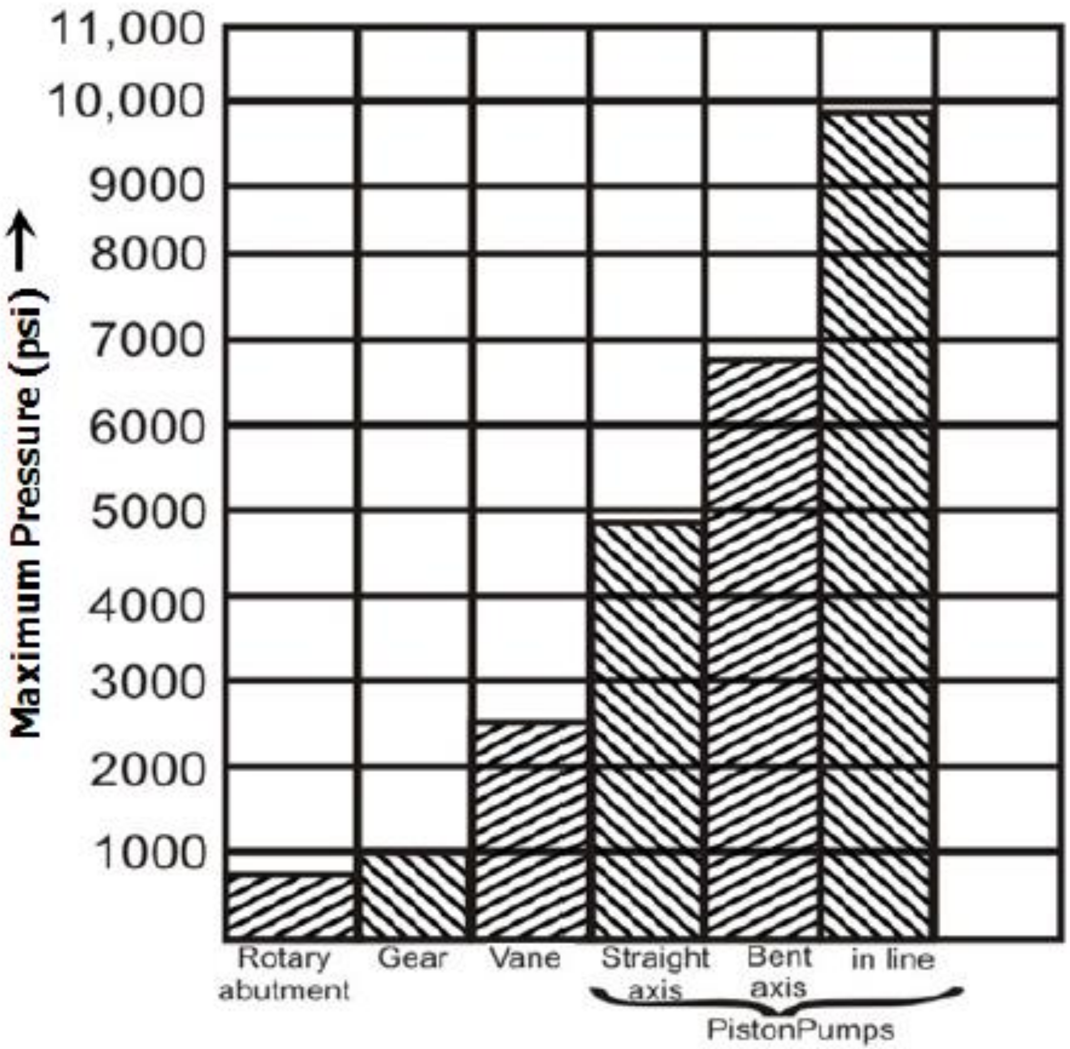


Fig. 5.17-1 : Practical pressure range in different model of pump & motor

Pump Performance

$$\text{Volumetric efficiency } (\eta_v) = \frac{\text{Actual flow rate of the pump}}{\text{Theoretical flow rate of the pump}}$$

$$= \frac{Q_A}{Q_T}$$

Volumetric efficiency (η_v) indicates the amount of leakage that takes place within the pump.This is due to manufacture tolerances and flexing of the pump casing under designed pressure operating conditions.

For gear pumps, $\eta_v = 80\%–90\%$.

For vane pumps, $\eta_v = 92\%$.

For piston pumps, $\eta_v = 90\%–98\%$.

2. Mechanical efficiency (η_m): Itis the ratio of the pump output power assuming no leakage to actual power delivered to the pump:

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

Mechanicalefficiency(η_m) indicates the amount of energy losses that occur for reasons other than leakage. This includes friction in bearings and between mating parts. This includes the energy losses due to fluid turbulence. Mechanical efficiencies are about 90%–95%. We also have the relation

$$\eta_m = \frac{p\,Q_T}{T_A N}$$

where$p$ is the pump discharge pressure in Pa or N/m^2, Q_T is the theoretical flow rate of the pump in m^3/s, T_A is the actual torque delivered to the pump in Nm and N is the speed of the pump in rad/s.

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

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

$$= \frac{T_T}{T_A}$$

The theoretical torque (T_T) required to operate the pump is the torque that would be required if there were no leakage.

The theoretical torque (T_T) is determined as follows

$$T_T(\text{N m}) = \frac{V_{D_N}}{2\pi}\left(\text{m}^3 \times \frac{N}{\text{m}^2}\right) = \text{N m}$$

The actual torque (T_A) is determined as follows

$$\text{Actual torque } T_A \text{ (N m)} = \frac{P}{\omega}\left(\frac{\text{N m/s}}{\text{rad/s}}\right) = \text{N m}$$

where$\omega = 2\pi N/60$. Here$N$is the speed in RPM.

3. Overall efficiency (η_o)**:**It is defined as the ratio of actual power delivered by the pump to actual power delivered to the pump

$$\text{Overall efficiency }(\eta_o) = \frac{\text{Actual power delivered by the pump}}{\text{Actual power delivered to the pump}}$$

Overall efficiency (η_o) considers all energy losses and can be represented mathematically as follows:

$$\text{Overall efficiency }(\eta_o) = \eta_v \eta_m$$

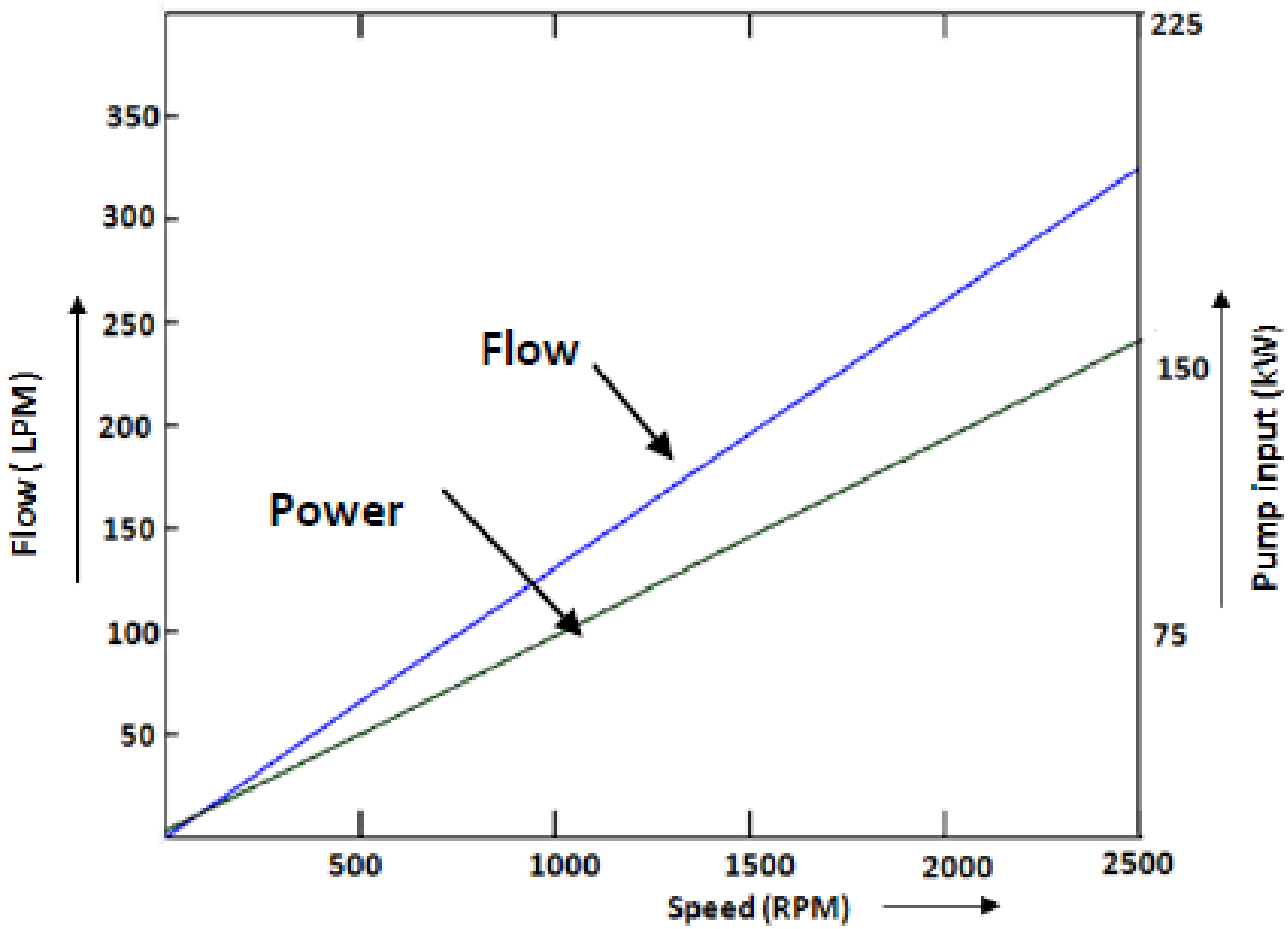
$$\Rightarrow \eta_o = \frac{Q_A}{Q_T} \times \frac{pQ_T}{T_A N}$$

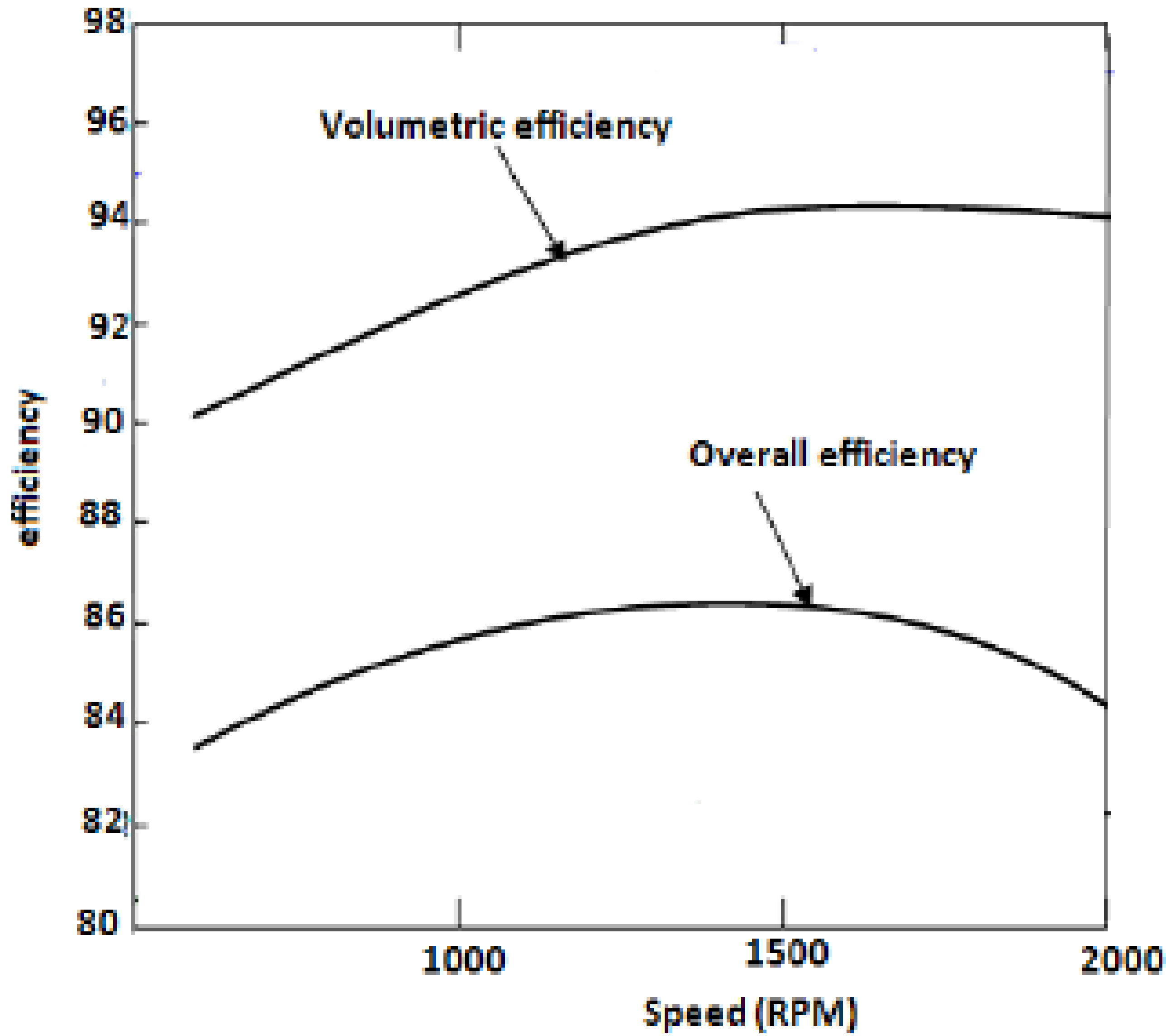
Comparison of Power Losses in Three Pump Designs

Pump Design	Power Loss as a Percentage of Input Power[a]		
	Due to Friction	Due to Leakage	Total
Gerotor	–	15.0	15.0
Vane	11.0	7.6	18.6
Piston	5.0	1.7	6.7

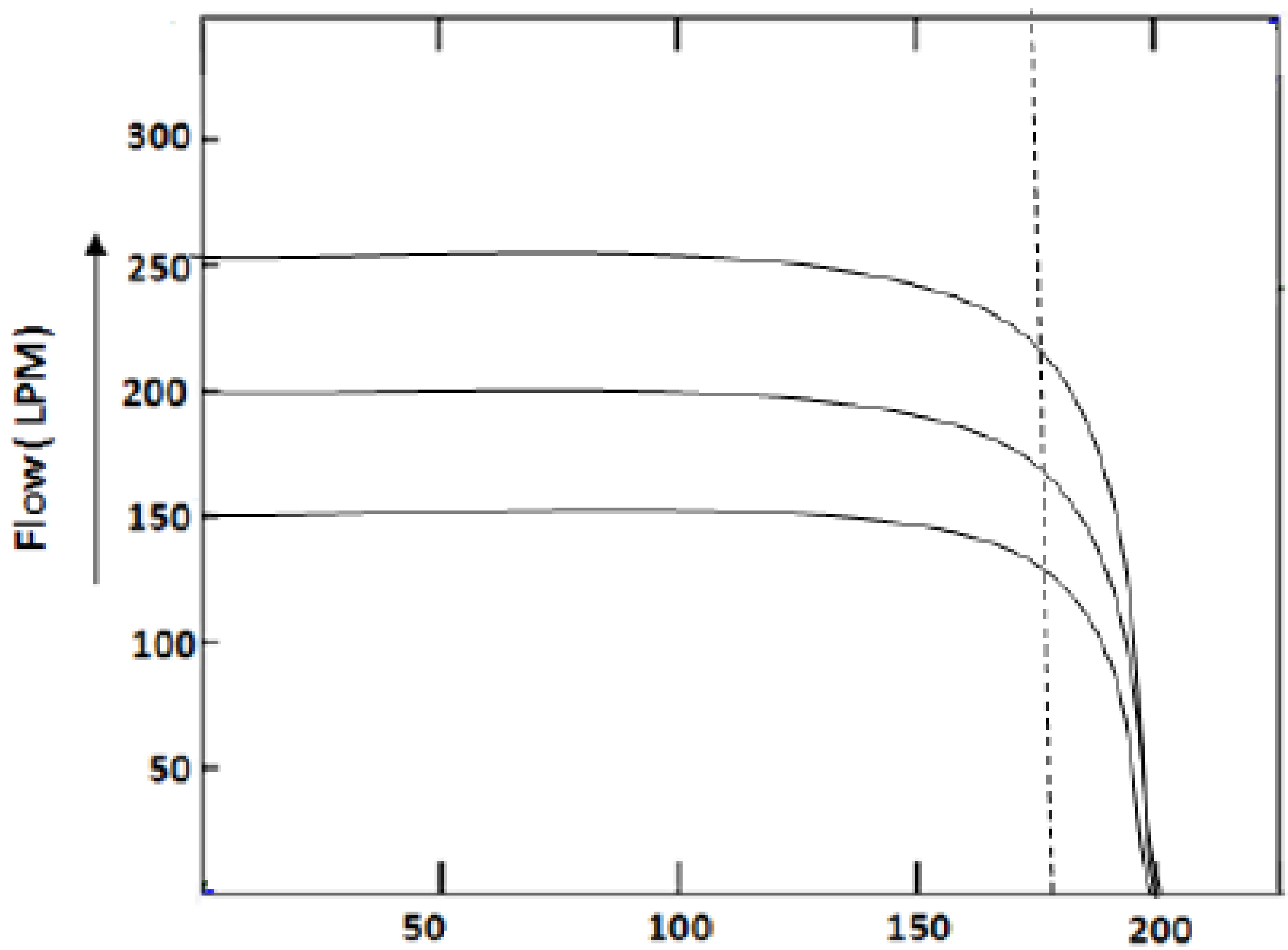
a. Comparison made at 1200 rpm operating speed and 1500 psi pressures

PERFORMANCE CURVES

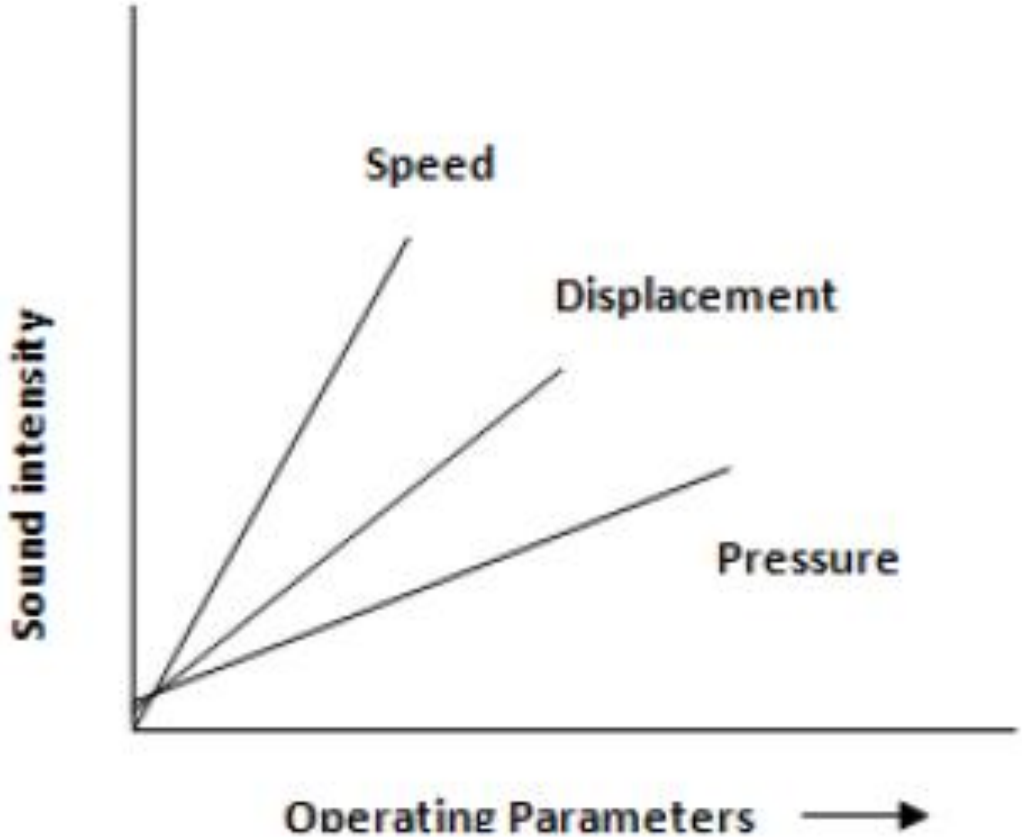




Volumetric efficiency
Overall efficiency
efficiency
98
96
94
92
90
88
86
84
82
80
1000
1500
2000
Speed (RPM)



Flow (LPM)
300
250
200
150
100
50
50
100
150
200


Speed
Displacement
Pressure
Sound intensity
Operating Parameters

1.13.1 Factors Causing Cavitation

Cavitation is caused by the following factors:

1. Undersized plumbing.
2. Clogged lines or suction filters.
3. High fluid viscosity.
4. Too much elevation head between the reservoir and the pump inlet.

1.13.2 Rules to Eliminate (Control) Cavitation

Following are the rules to control cavitation:

1. Keep suction line velocities below 1.2 m/s.
2. Keep the pump inlet lines as short as possible.
3. Minimize the number of fittings in the inlet line.
4. Mount the pump as close as possible to the reservoir.
5. Use low-pressure drop inlet filters.
6. Use proper oil as recommended by the pump manufacturer.

1.14Pump Selection

The main parameters affecting the selection of a particular type of pump are as follows:

1. Maximum operating pressure.
2. Maximum delivery.
3. Type of control.
4. Pump drive speed.
5. Type of fluid.
6. Pump contamination tolerance.
7. Pump noise.
8. Size and weight of a pump.
9. Pump efficiency.
10. Cost.
11. Availability and interchangeability.
12. Maintenance and spares.

Table 1.4 Operating pressure and size ranges for hydraulic pump types

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

Example 1.6

A vane pump has a rotor diameter of 63.5 mm, a cam ring diameter of 88.9 mm and a vane width of 50.8 mm. What must be eccentricity for it to have a volumetric displacement of 115cm^3?

Solution: Volumetric displacement is

$$V_D = \pi\left(\frac{D_C + D_R}{2}\right)Le$$

where D_c is the diameter of the cam ring, D_R is the diameter of the rotor, e is the eccentricity and L is the width of the vane pump. So we have

$$115\times10^{-6} = \pi\times\frac{0.0889+0.0635}{2}\times e\times0.0508$$

Therefore eccentricity

$$e = 9.456\times10^{-3}\text{ m} = 9.456\text{ mm}$$

Example 1.9

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

Solution:

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

Discharge of the pump = 29 LPM

Speed of pump $N = 1000$ rpm

Now

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

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

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

Volumetric displacement

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

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

Example 1.13

A pump has a displacement volume of 98.4 cm^3. It delivers 0.0152 m^3/s of oil at 1000 RPM and 70 bar. If the prime mover input torque is 124.3 Nm. What is the overall efficiency of pump? What is the theoretical torque required to operate the pump?

Solution:

Volumetric discharge $= 98.4 \text{ cm}^3$

Theoretical discharge is

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

Volumetric efficiency is

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

Overall efficiency is

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

The mechanical efficiency is

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

Now

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

Note: Mechanical efficiency can also be calculated as

$$\eta_{\mathrm{m}} = \frac{pQ_{\mathrm{T}}}{T\omega}$$

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

$$= 0.882 = 88.2\%$$

ME 5451 – Hydraulics and Pneumatics

Lecture -5

Date: 05-03-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Refreshing lec-4
2. Efficiency –Comparison of pumps
3. Performance curves
4. Problems

Course Instructor: Dr. A. Siddharthan

- maximum discharge pressure for continuous operation
- maximum intermittent operating pressure;
- maximum peak pressure for short peaks only (fig. 55).

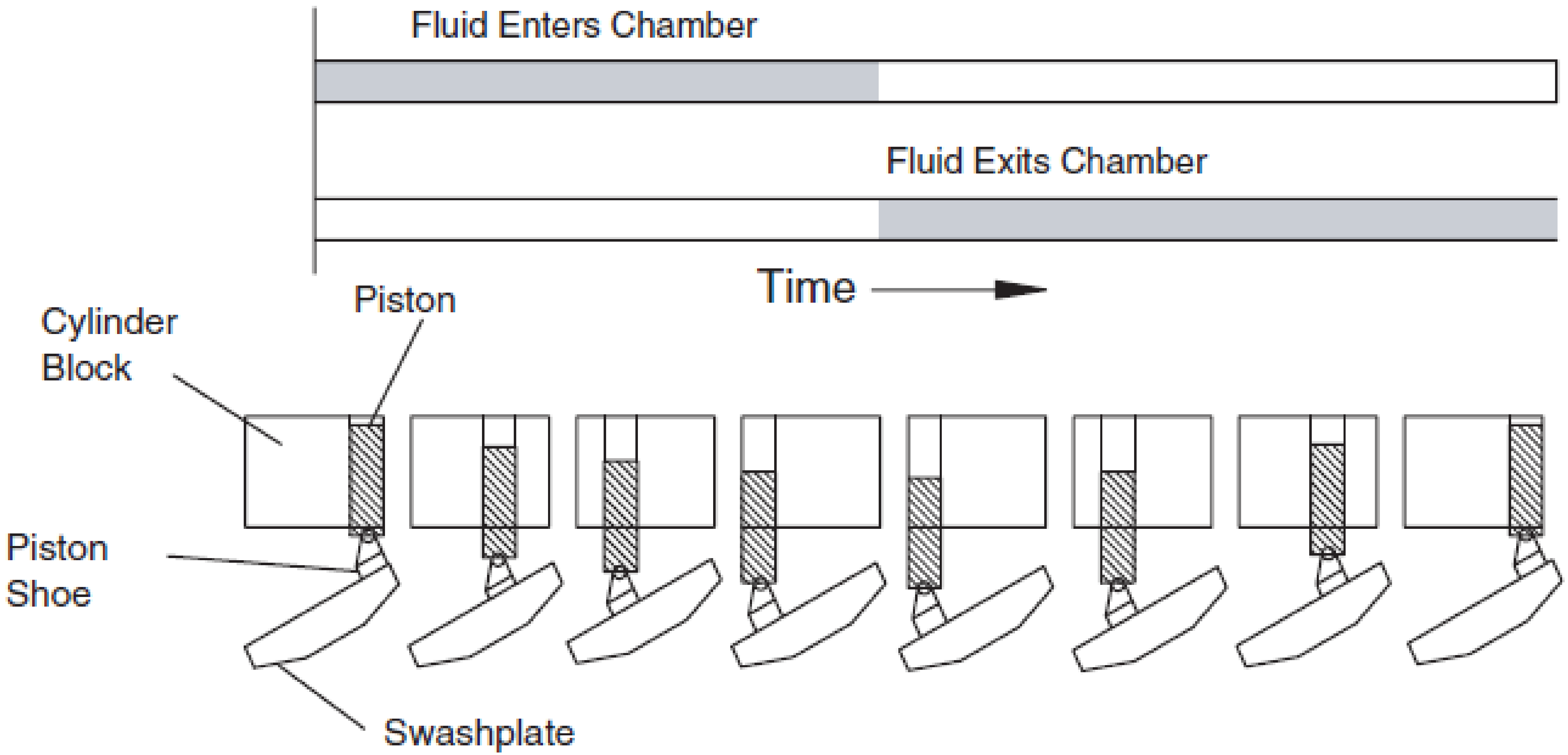


FIGURE 4.13
Schematic illustrating the motion of one piston during a single rotation of the cylinder block. Axis of rotation is in the plane of paper.

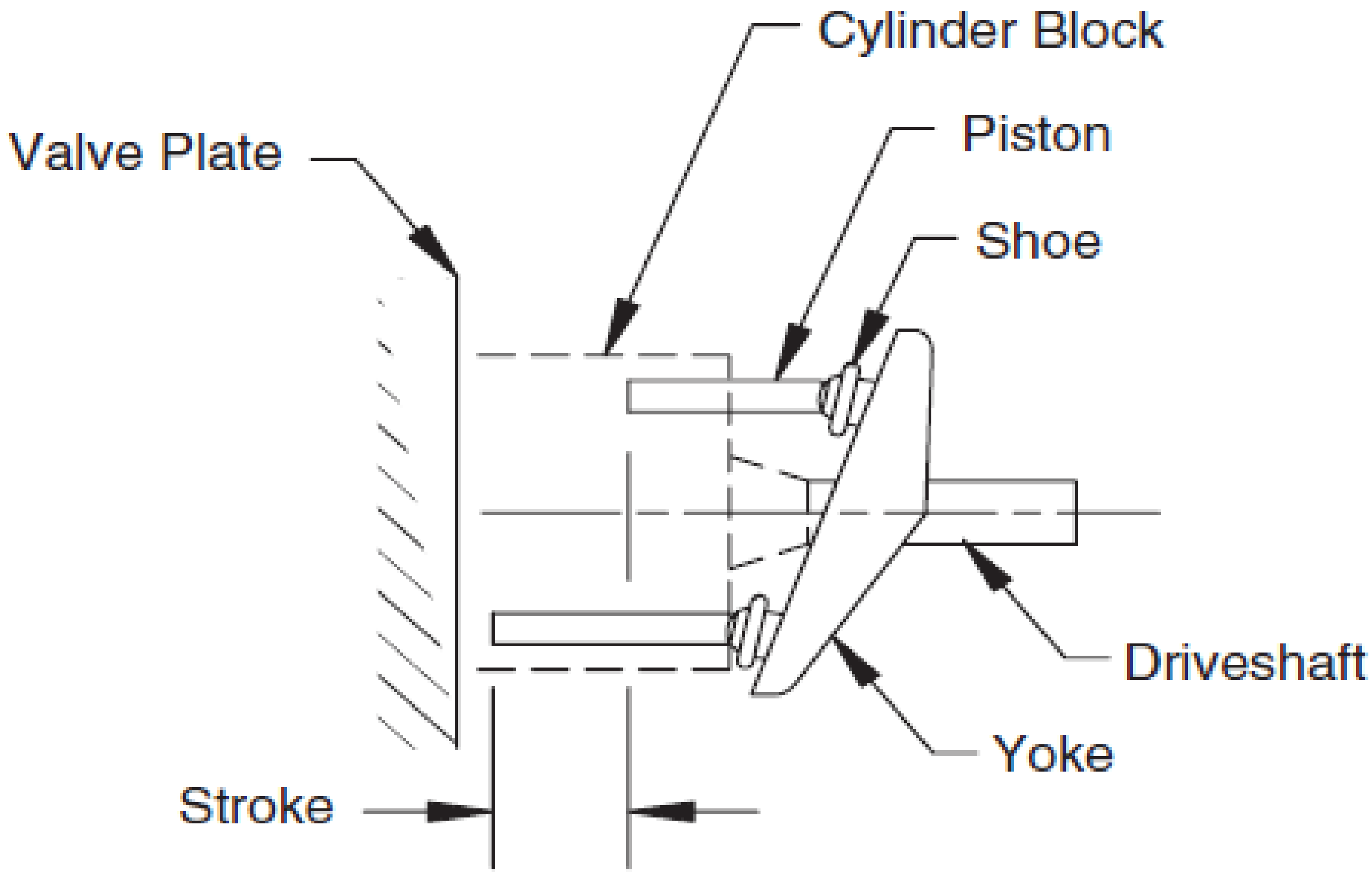


FIGURE 4.12
Schematic of axial piston pump.

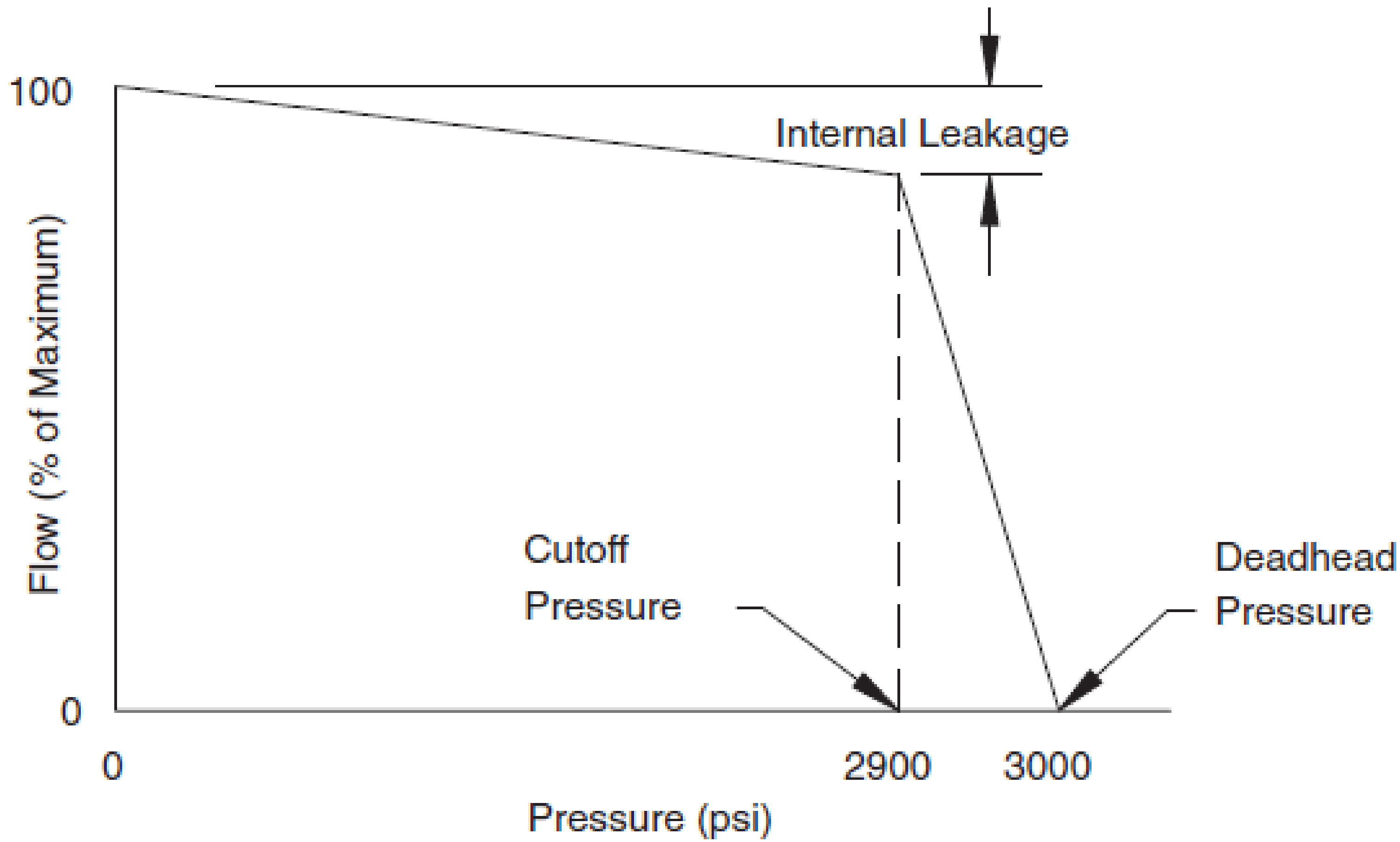
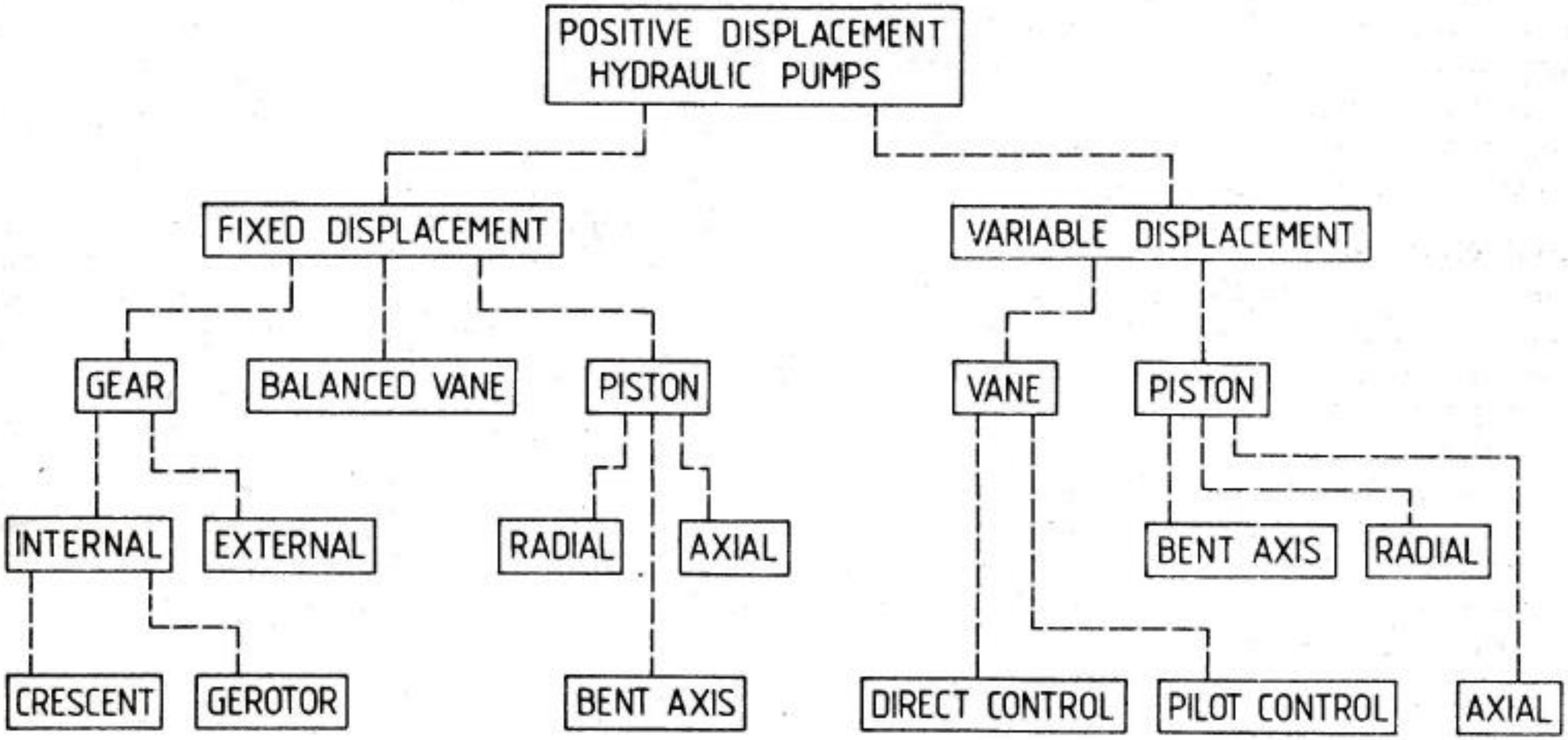


FIGURE 4.10
Typical flow vs. pressure performance curve for a pressure-compensated variable displacement vane pump.

Pump Principle	Pressure (bar) p max		Speed (RPM)		Q_{max} (l/min)	Pressure Fluctuations	Noise Level dBa	Efficiency tot	Filtration min
	from	to	n min	n max					
Gear	40	100	500	3000	300	pulsating	90	50–80	100
Gear, hydrost. bal.	100	200	500	6000	200	pulsating	90	80–90	50
Internal Gear (gerotor)	50	70	500	2000	100	low pulsation	85	60–80	100
Internal Gear, (crescent)	150	300	500	2000	50	low pulsation	65	70–90	50
Screw	50	140	500	3000	100	free of pulsation	75	60–80	50
Vane	50	100	500	3000	100	low pulsation	80	65–80	50
Vane, hydrost. bal.	140	175	500	3000	300	low pulsation	85	70–90	50
Variable Vane	40	100	1000	2000	200	low pulsation	80	70–80	50
Fixed Vane	100	140	500	2000	100	low pulsation	80	70–85	50
Cam	30	50	—	—	200	low pulsation	—	—	—
Axial Piston, swashplate	200	250	200	2000	3000	pulsating	90	80–90	25
Axial Piston, bent axis	250	350	200	2000	500	pulsating	90	80–90	25
Radial Piston	350	650	200	2000	100	pulsating	90	80–90	50
In-Line-Piston	350	500	50	1000	300	pulsating	—	—	50


POSITIVE DISPLACEMENT HYDRAULIC PUMPS
FIXED DISPLACEMENT
VARIABLE DISPLACEMENT
GEAR
BALANCED VANE
PISTON
VANE
PISTON
INTERNAL
EXTERNAL
RADIAL
AXIAL
BENT AXIS
RADIAL
CRESCENT
GEROTOR
BENT AXIS
DIRECT CONTROL
PILOT CONTROL
AXIAL

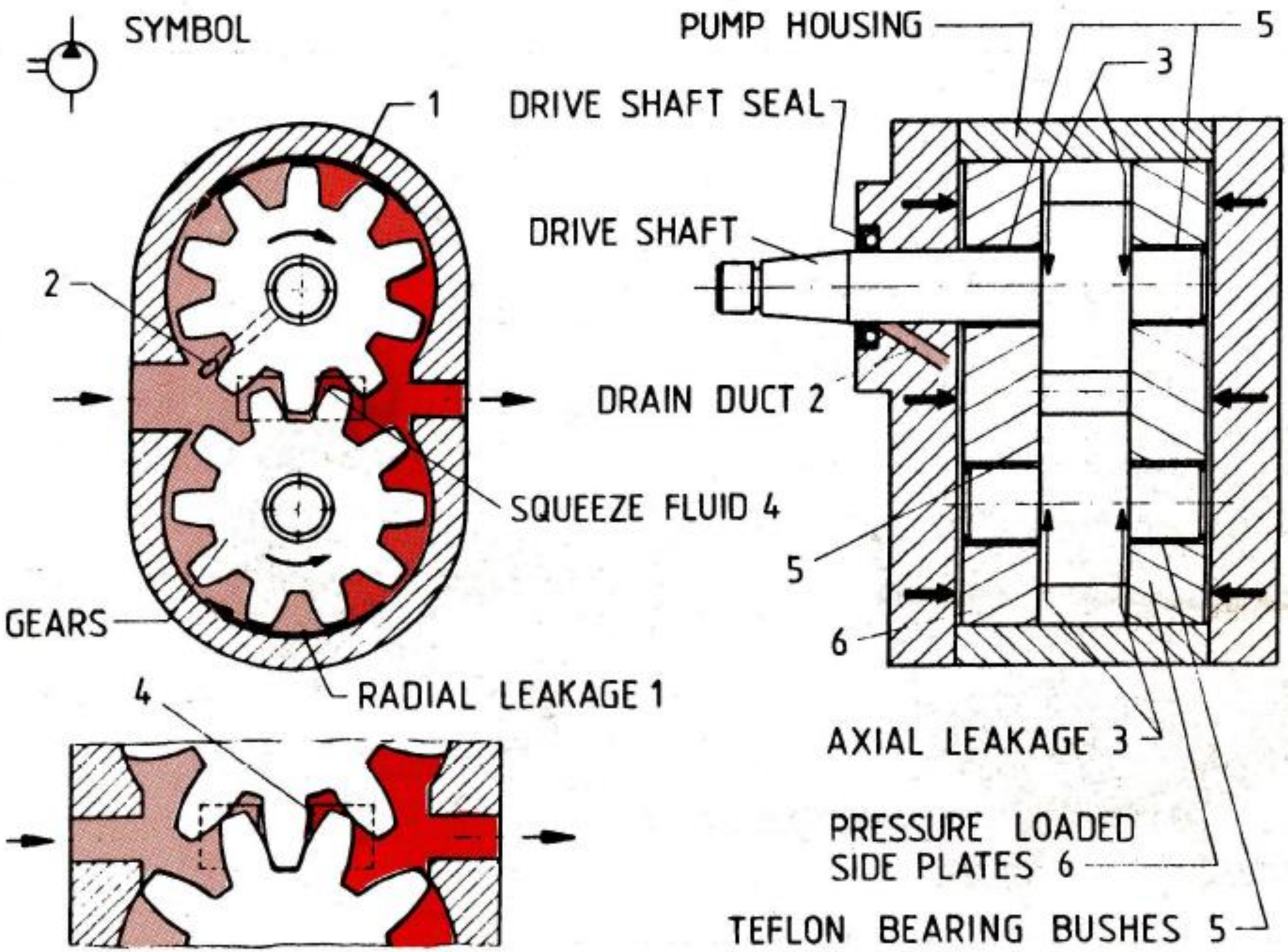


Fig. 59 External gear pump.

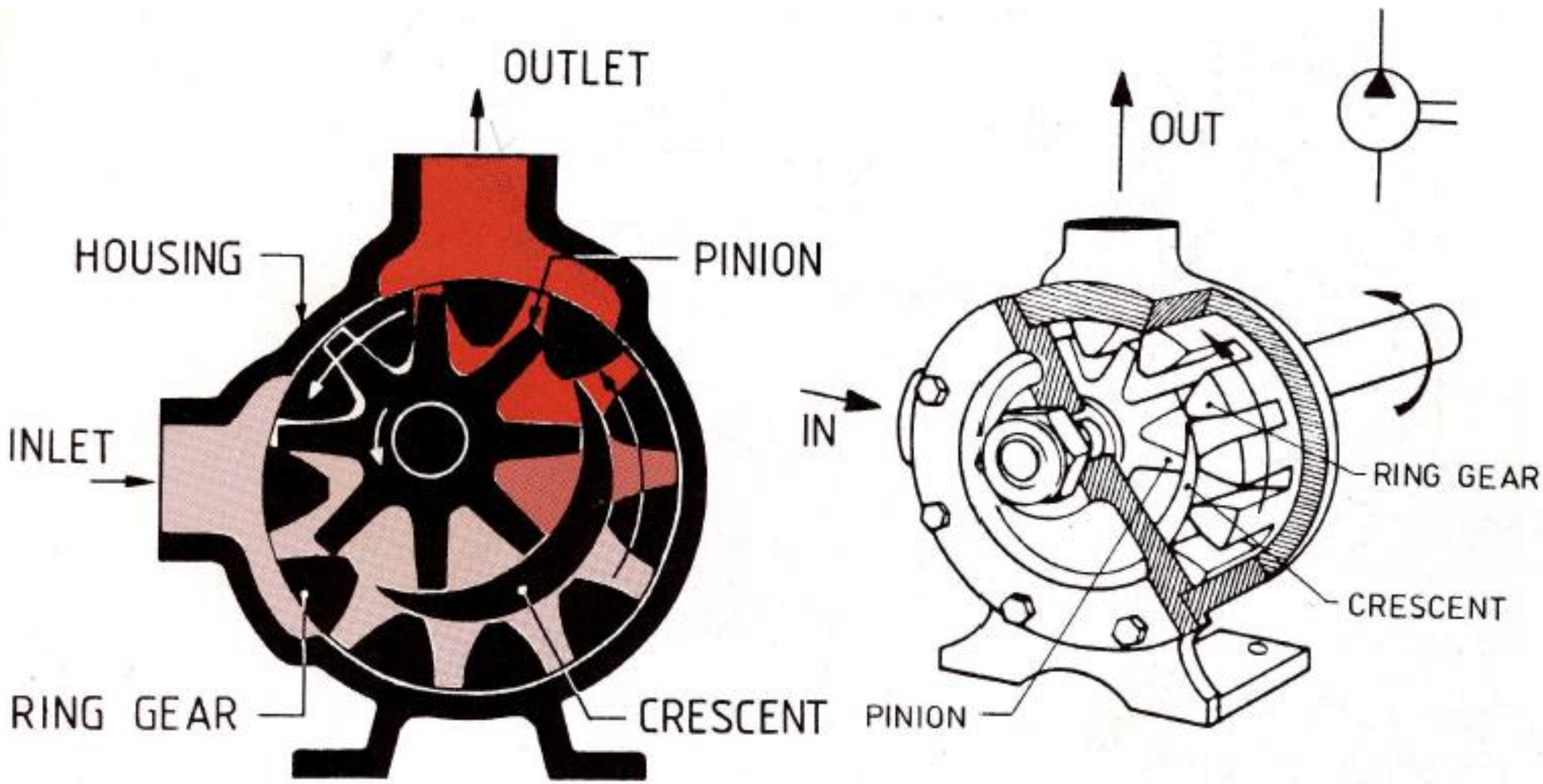


Fig. 60 Crescent type internal gear pump.

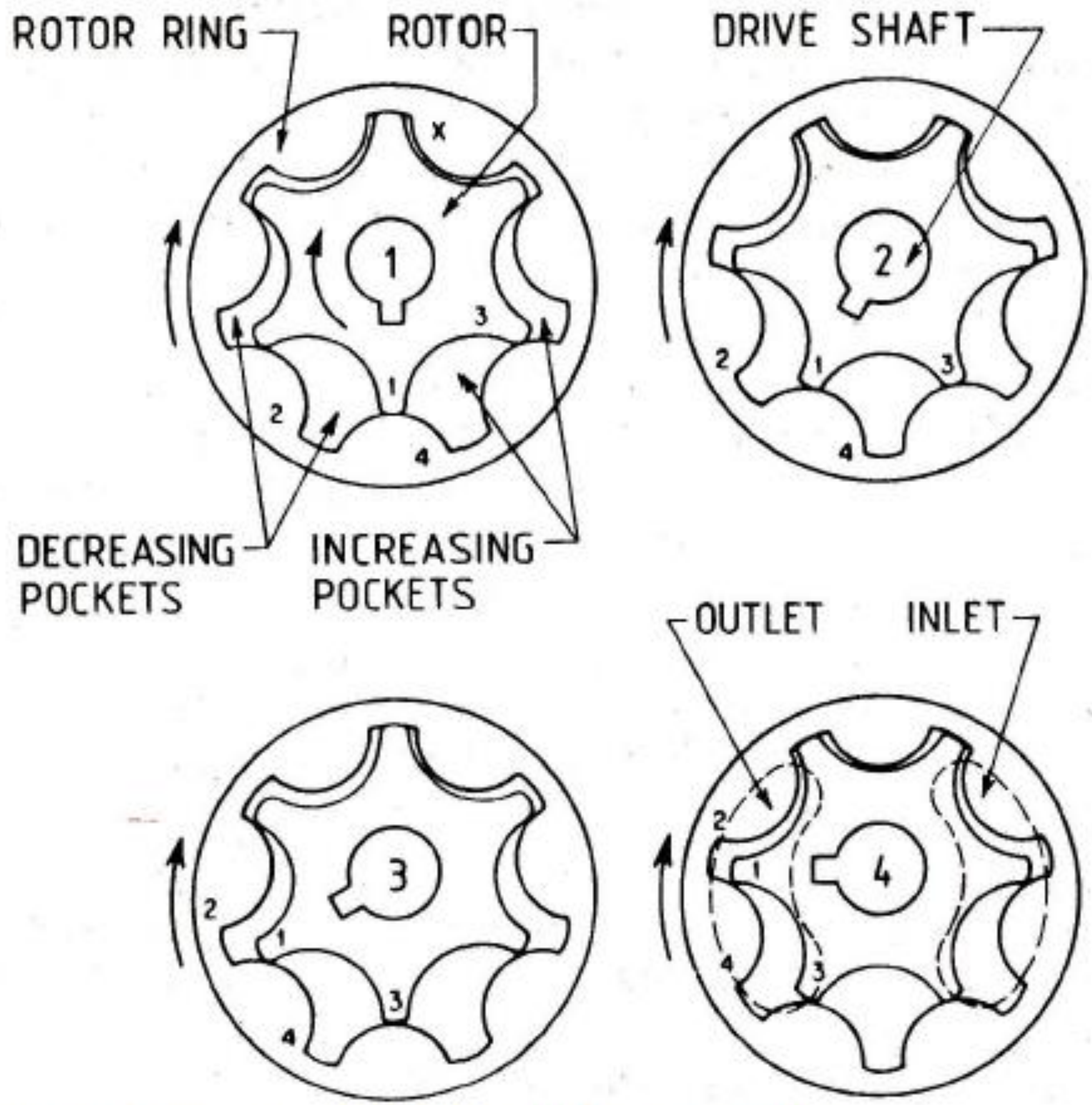


Fig. 61 Gerotor type internal gear pump

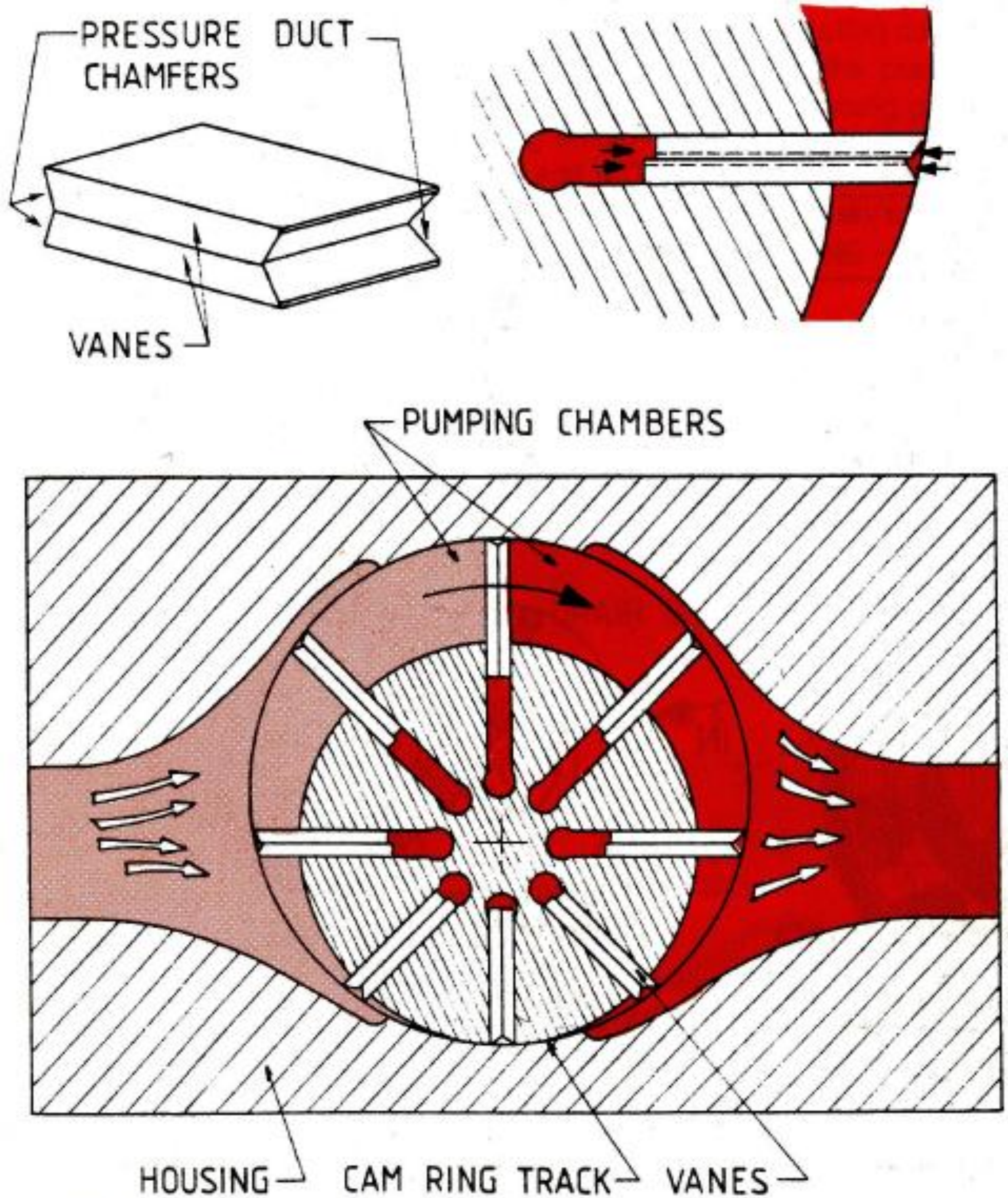


Fig 62 Function principle of vane pumps

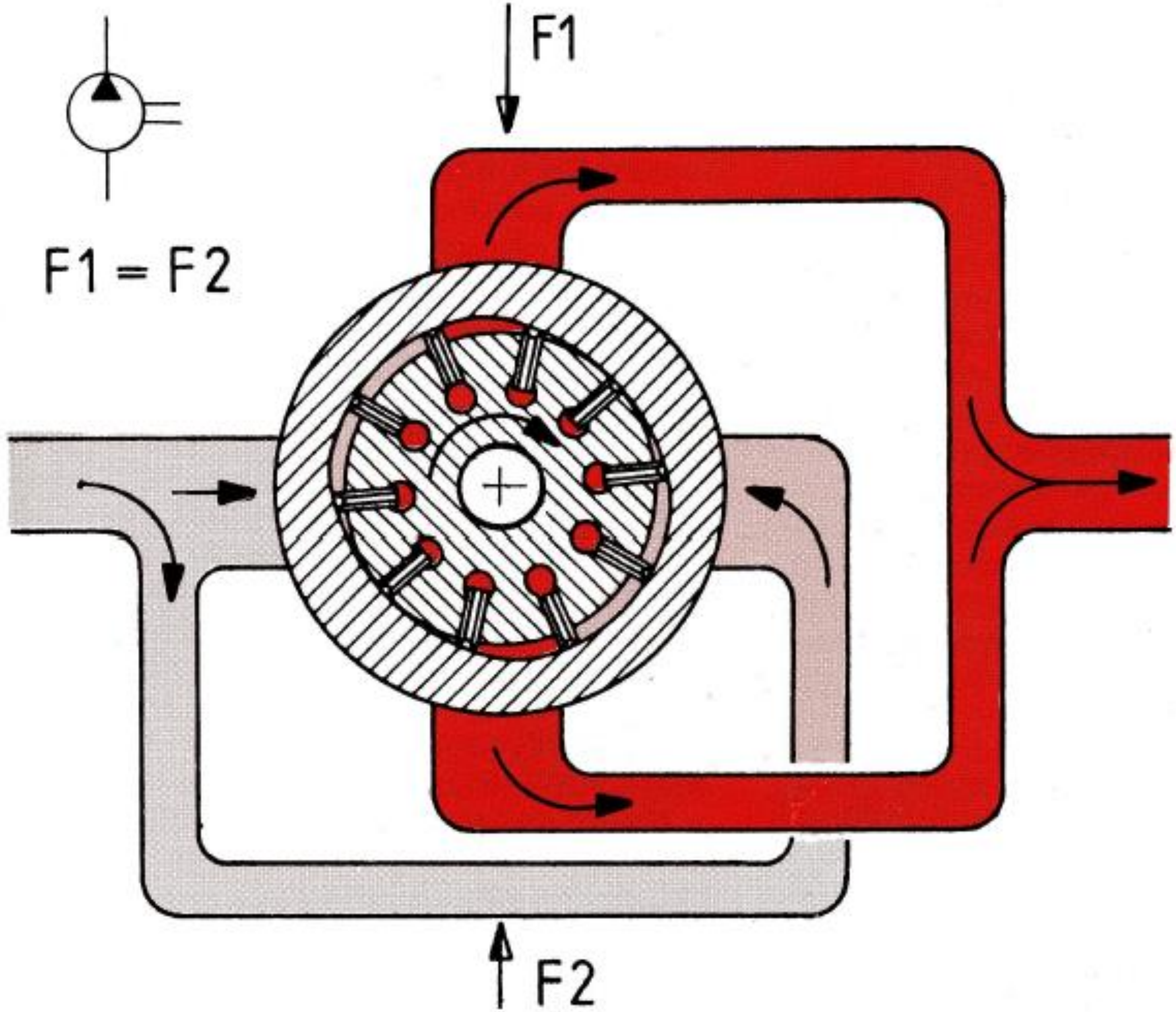


Fig. 63 Function principle of a balanced vane pump.

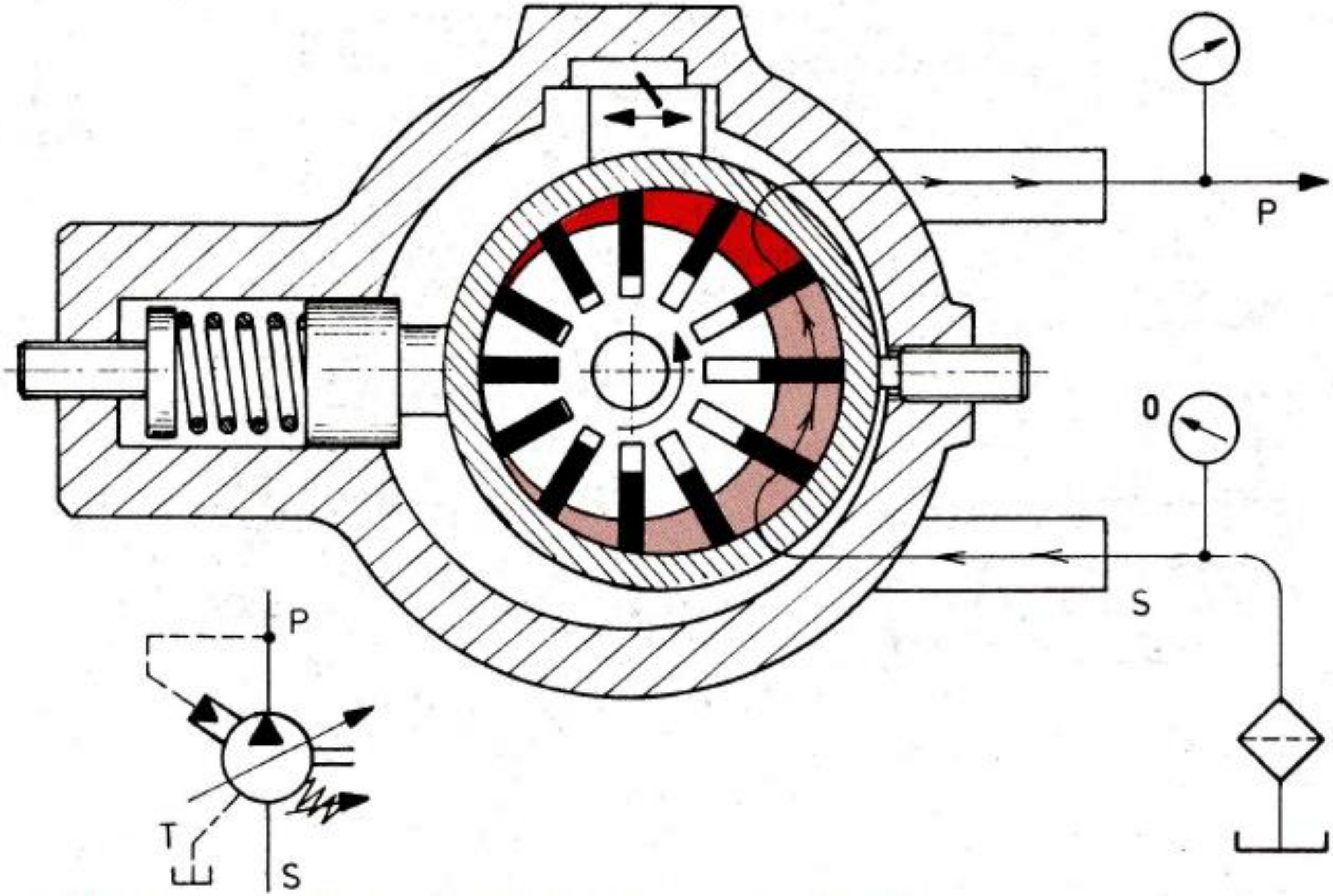
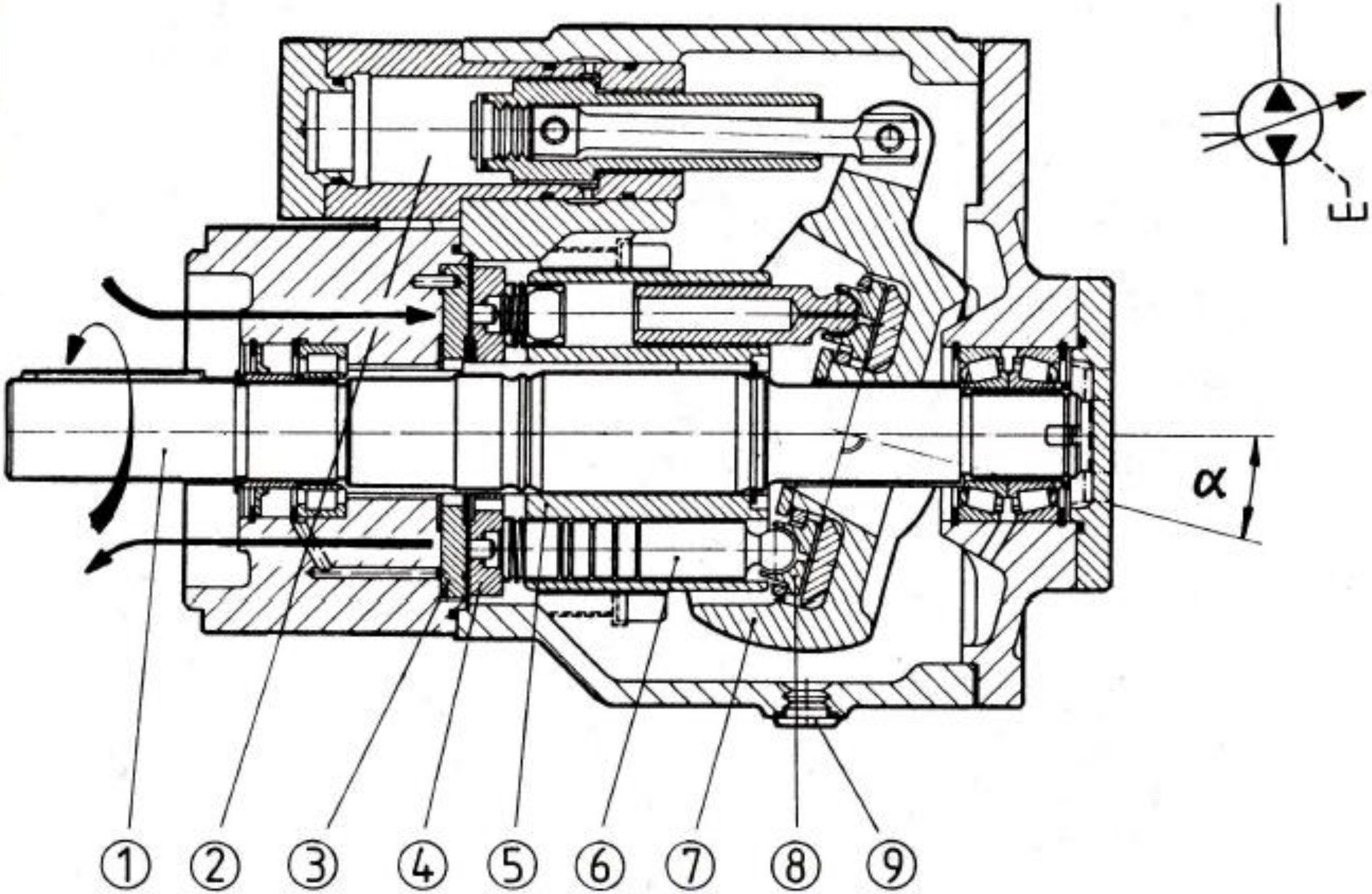


Fig. 64 Function principle of a variable displacement vane pump.



① Drive shaft
② Pressure controller
③ Valve plate (fixed)
④ Valve plate (rotating)
⑤ Cylinder block
⑥ Pistons
⑦ Swash plate (yoke)
⑧ Slipper pads
⑨ Case drain port

Fig 65 Swash plate, variable displacement piston pump

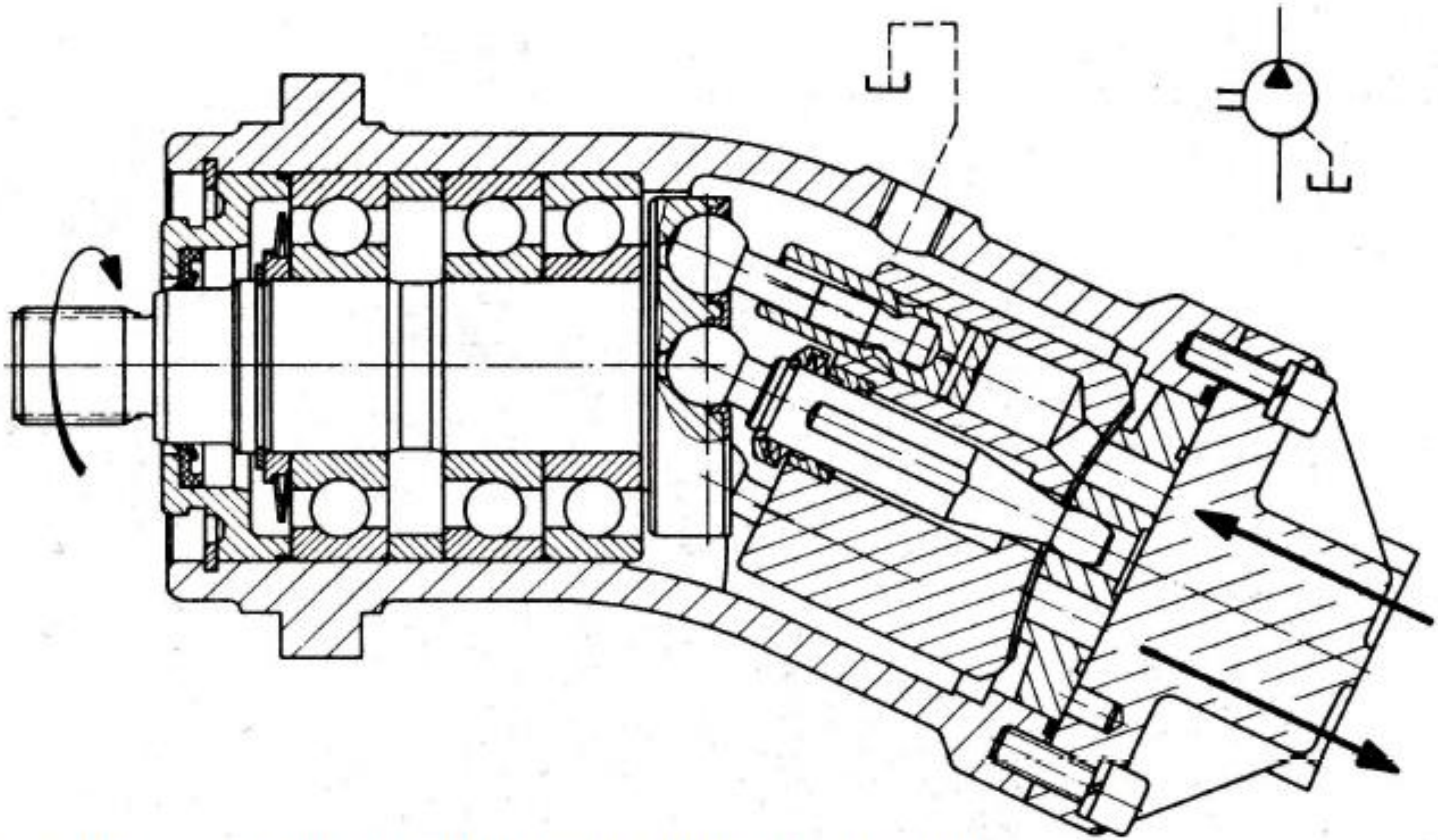


Fig. 67 Bent axis, fixed displacement piston pump.

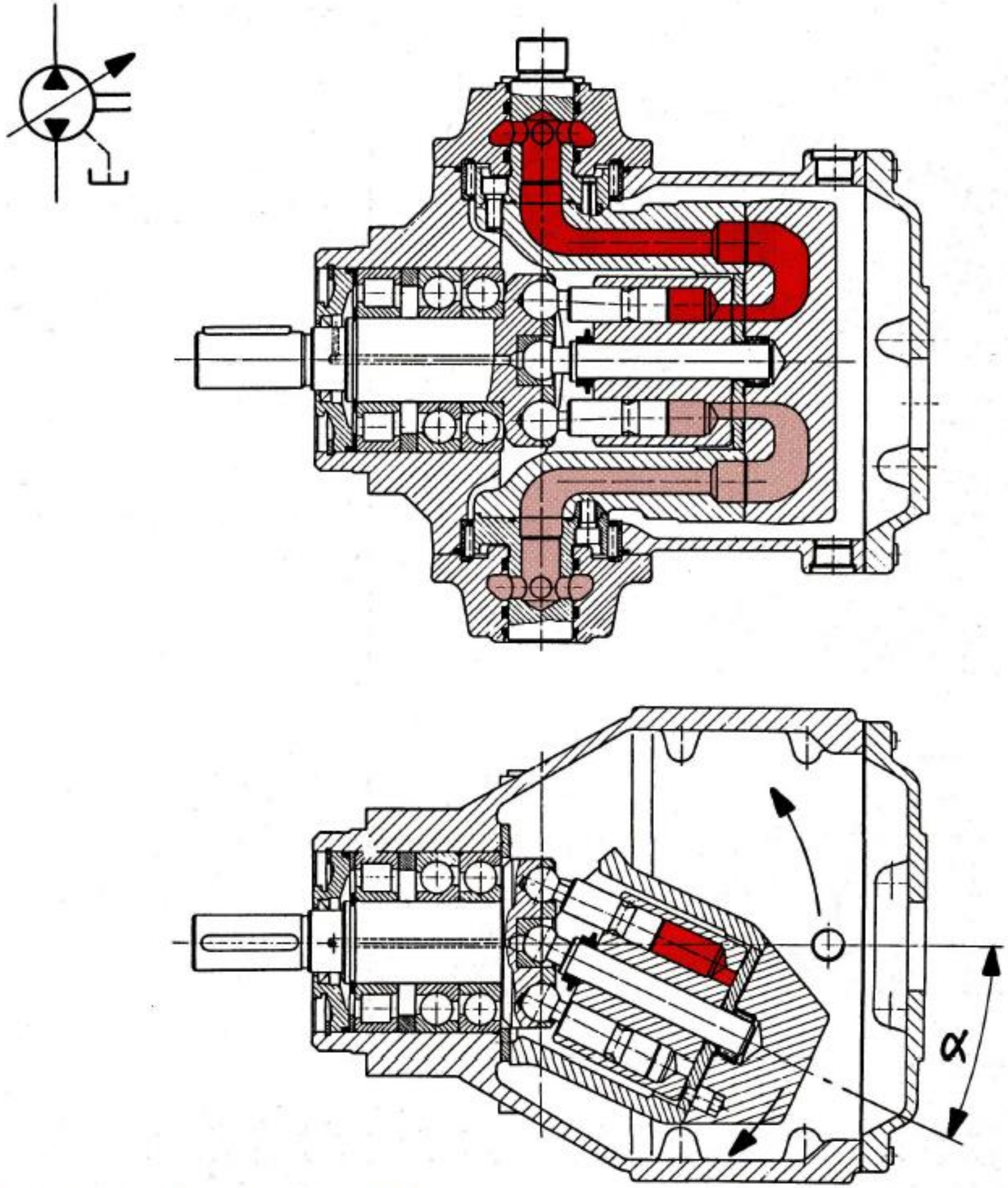


Fig 68 Bent Axis, variable displacement piston pump.

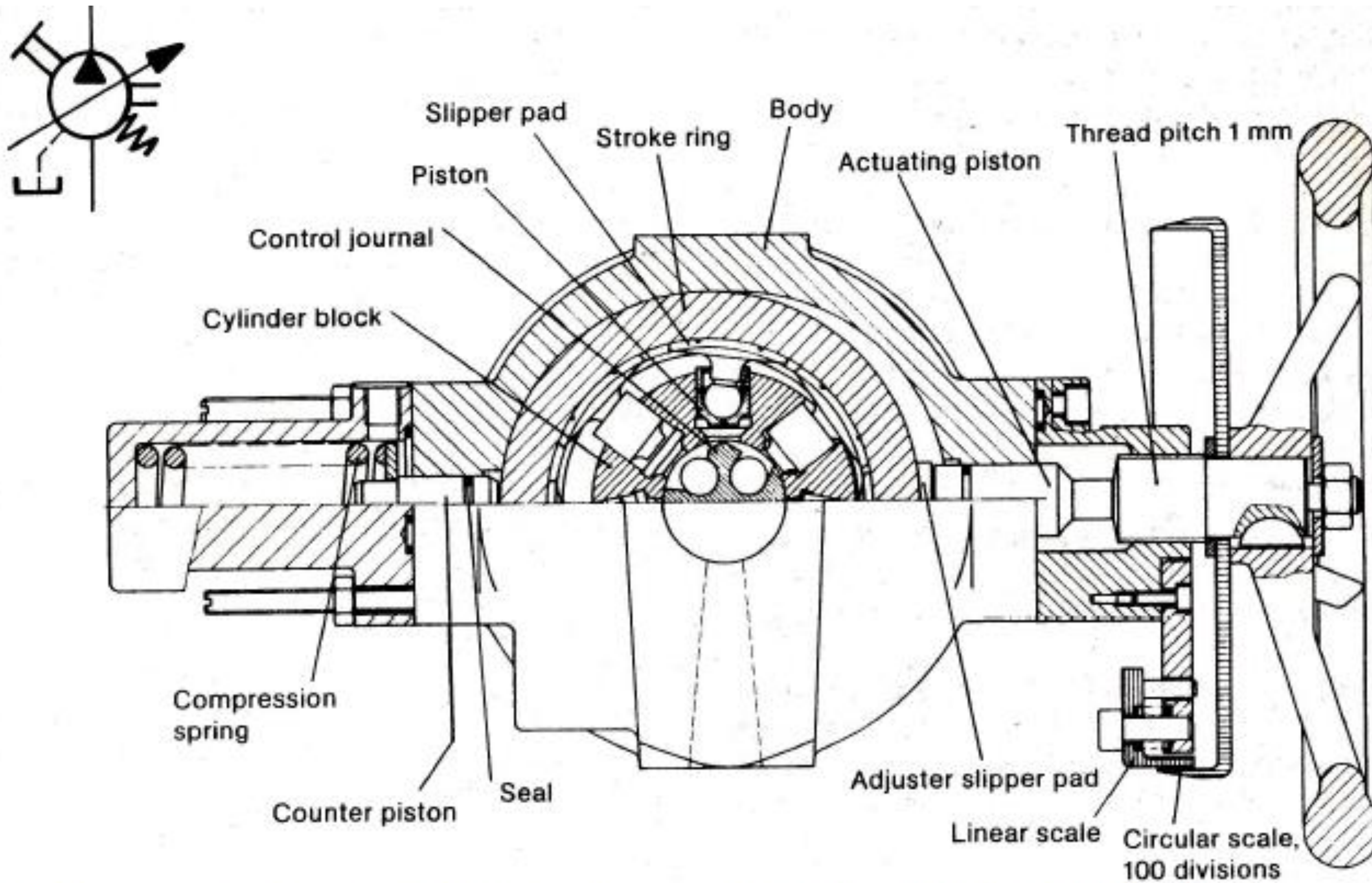


Fig 71 Radial piston pump, variable displacement with hand wheel adjustment

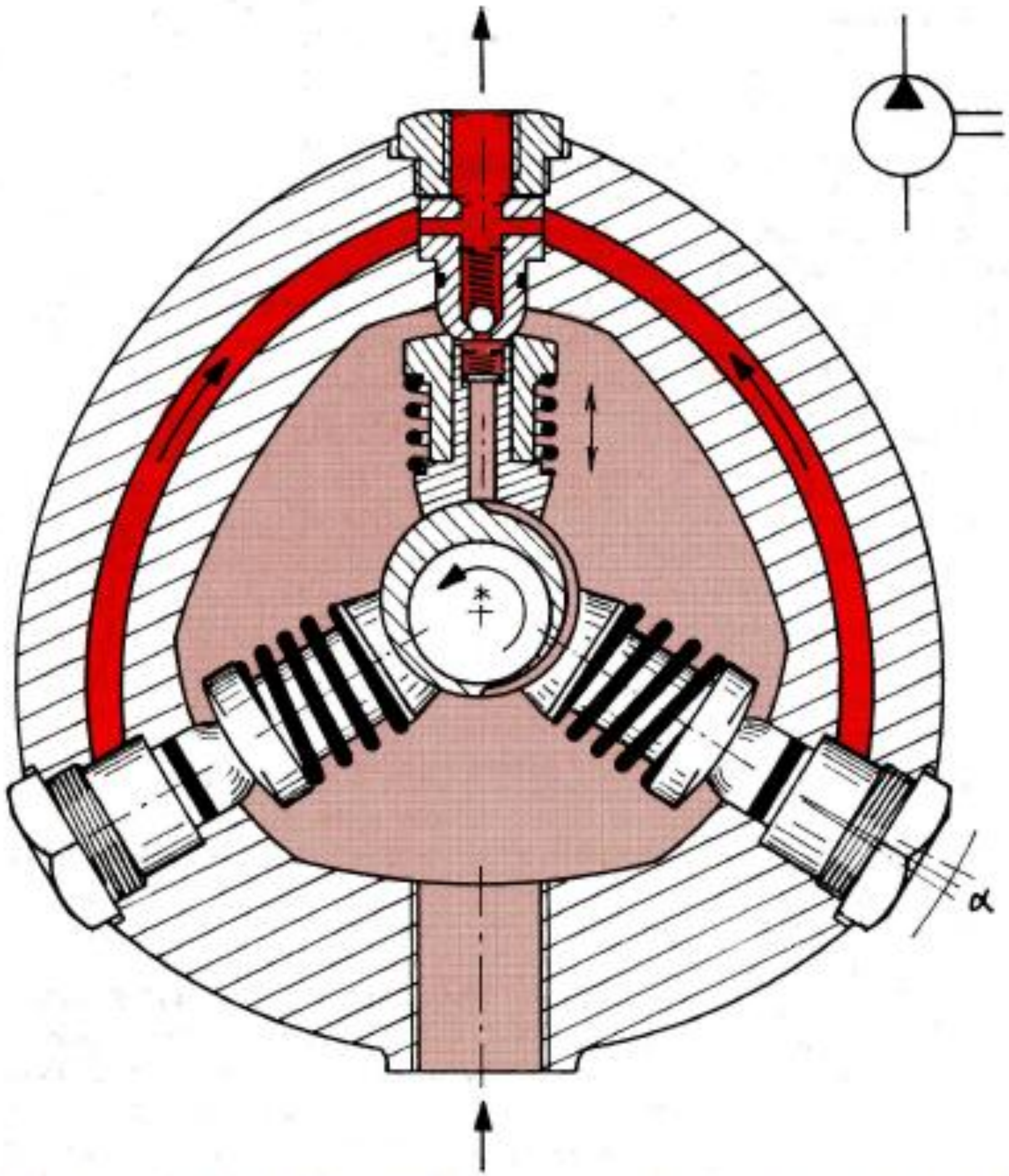


Fig 73 Radial piston pump (cam actuated type with self priming)

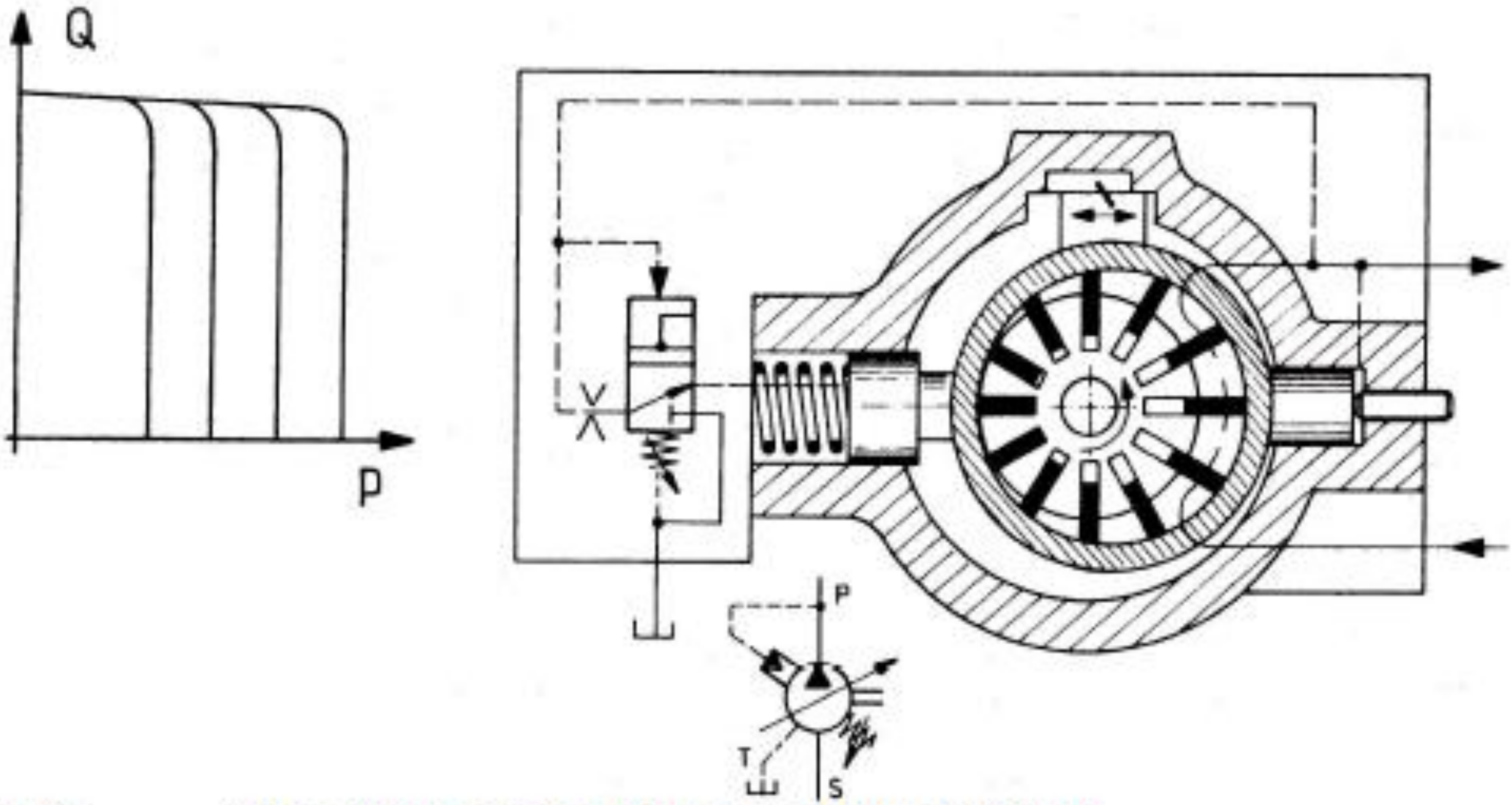


Fig 78 Vane pump with step type pressure controller

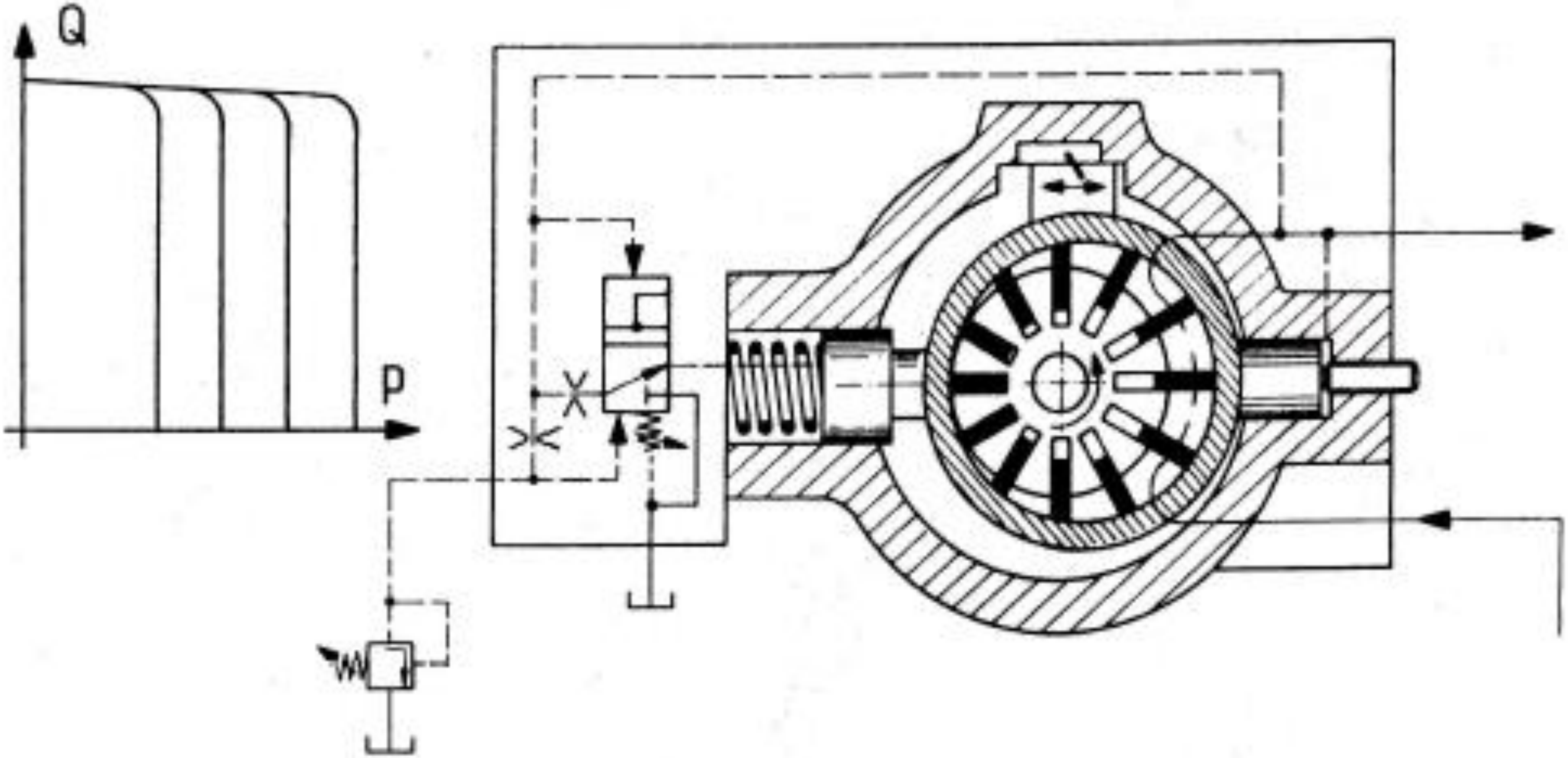
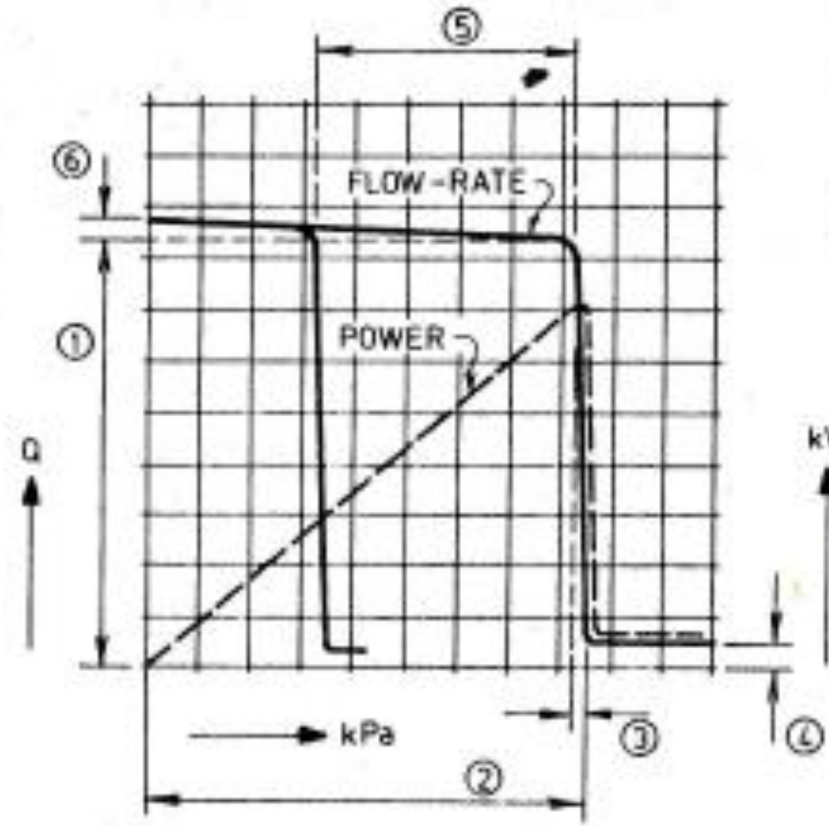


Fig 81 Vane pump with step type pressure controller and remote maximum pressure adjustment

"STEP" CUT-OFF PUMP CONTROL



1 Maximum flowrate prior to cut-off

2 Maximum adjusted system pressure

3 Spring constant

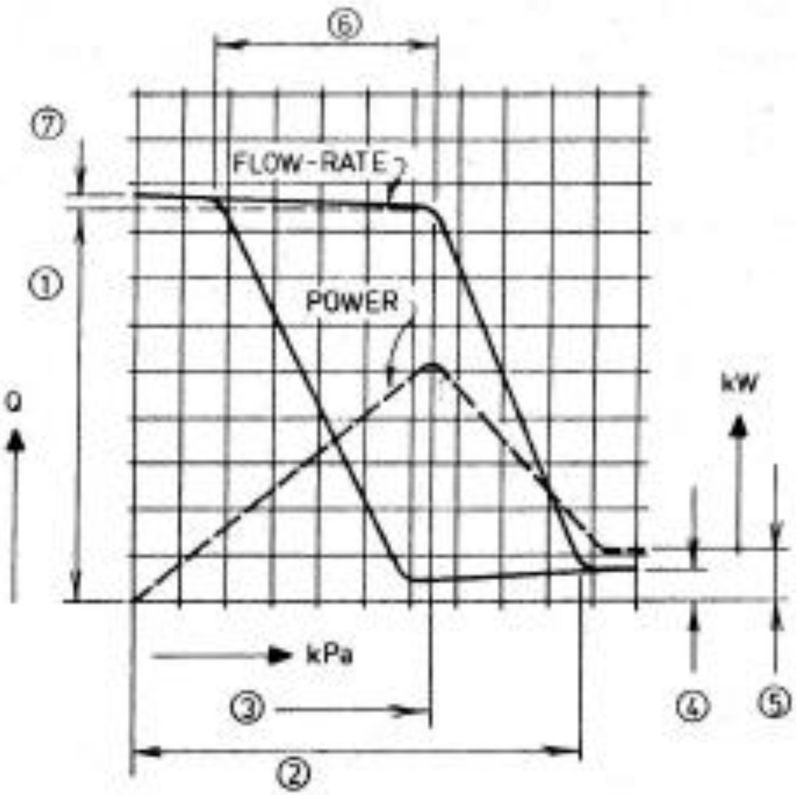
4 Minimum required flowrate to hold system pressure

5 Adjustment range

6 Decline in volumetric efficiency with increasing pressure

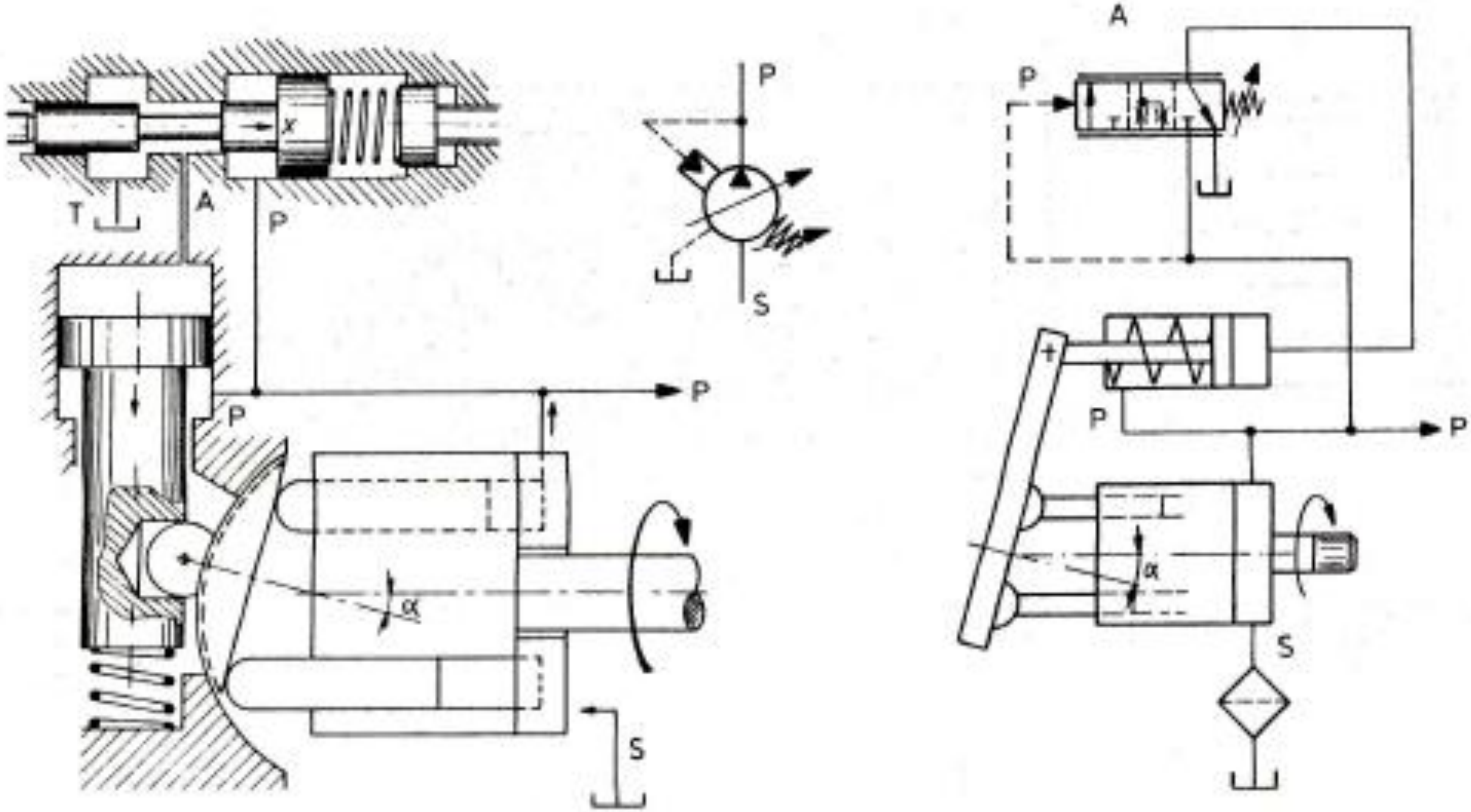
Fig 77 Pressure flow diagram for step type pump control.

"RAMP" CUT-OFF PUMP CONTROL



1 Maximum flowrate prior to cut-off
2 Maximum adjusted system pressure
3 Cut-off point
4 Minimum required flowrate to hold system pressure despite internal system leakage
5 Power requirement at minimum flow
6 Adjustment range
7 Decline in volumetric efficiency with increasing pressure

Fig 76 Pressure flow diagram for ramp type pump control.



PRESSURE CONTROLLER

Fig 80 Step type pressure controller for swash plate piston pump

ME 7553 – Hydraulics and Pneumatics

Lecture -6

Date: 12-03-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Refreshing lec-5
2. Rotary Actuators
3. Performance curves

Course Instructor: Dr. A. Siddharthan

HYDRAULIC MOTORS

Motors convert fluid energy back into mechanical energy and thus are the mirror image of pumps. It is not surprising that the same mechanisms are used for both. The typical motor designs are gear, vane, and piston.

Motor torque is divided into three separate groups:

1. **Starting torque:** The starting torque is the turning force the motor exerts from a dead stop.
2. **Running torque:** Running torque is exerted when the motor is running and changes whenever there is a change in fluid pressure.
3. **Stalling torque:** Stalling torque is the torque necessary to stop the motor.

Electric Motor	Hydraulic Motor
Electric motors cannot be stopped instantly. Their direction of rotation cannot be reversed instantly. This is because of air gap between the rotor and stator and the weak magnetic field.	Hydraulic motors can be stalled for any length of time. Their direction of rotation can be instantly reversed and their rotational speed can be infinitely varied without affecting their torque. They can be braked instantly and have immense torque capacities.
Electric motors are heavy and bulky.	Hydraulic motors are very compact compared to electric motors. For the same power, they occupy about 25% of the space required by electric motors and weigh about 10% of electric motors.
Moment of inertia-to-torque ratio is nearly 100.	Moment of inertia-to-torque ratio is nearly 1.

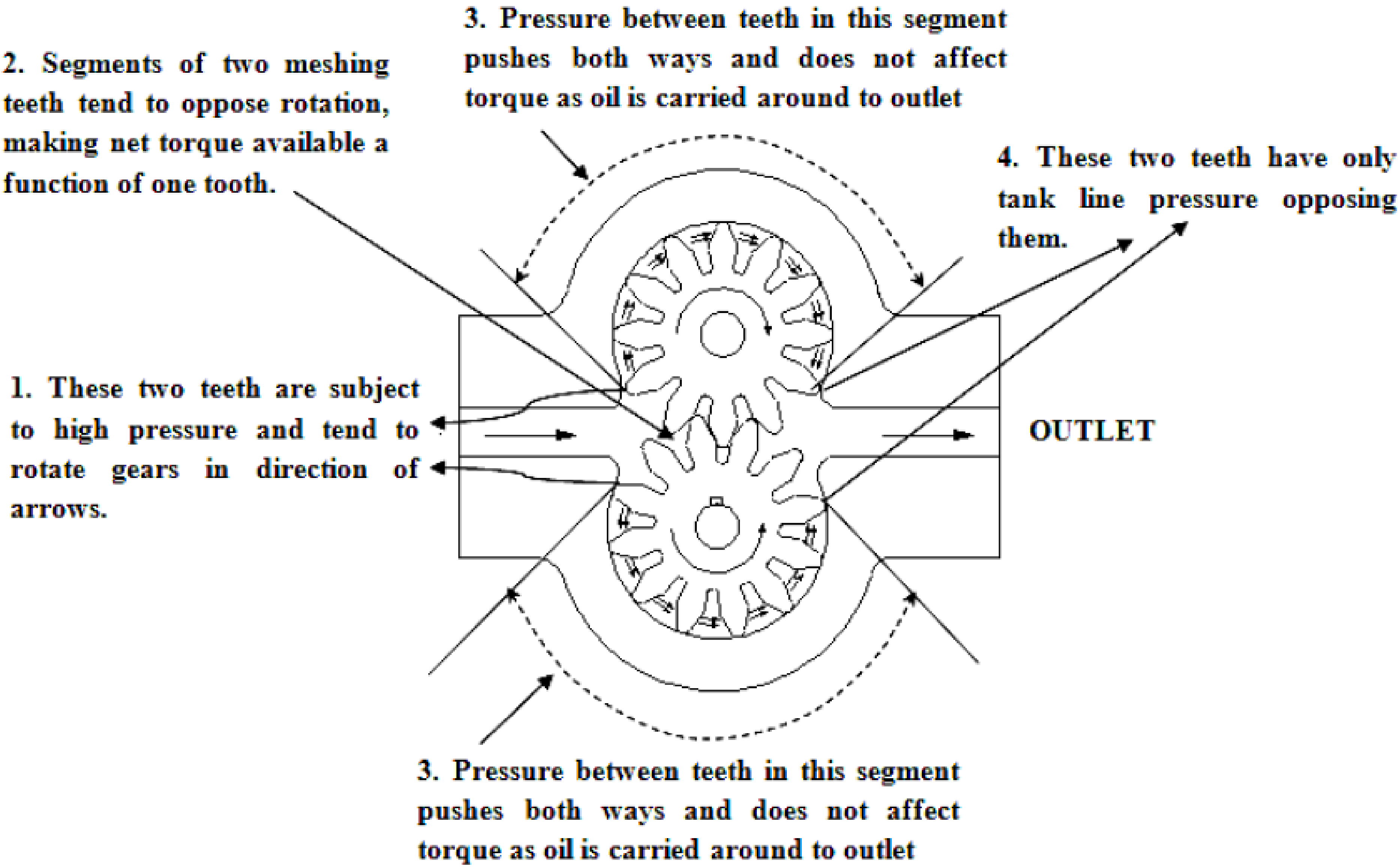


Figure 1.1 Gear motor

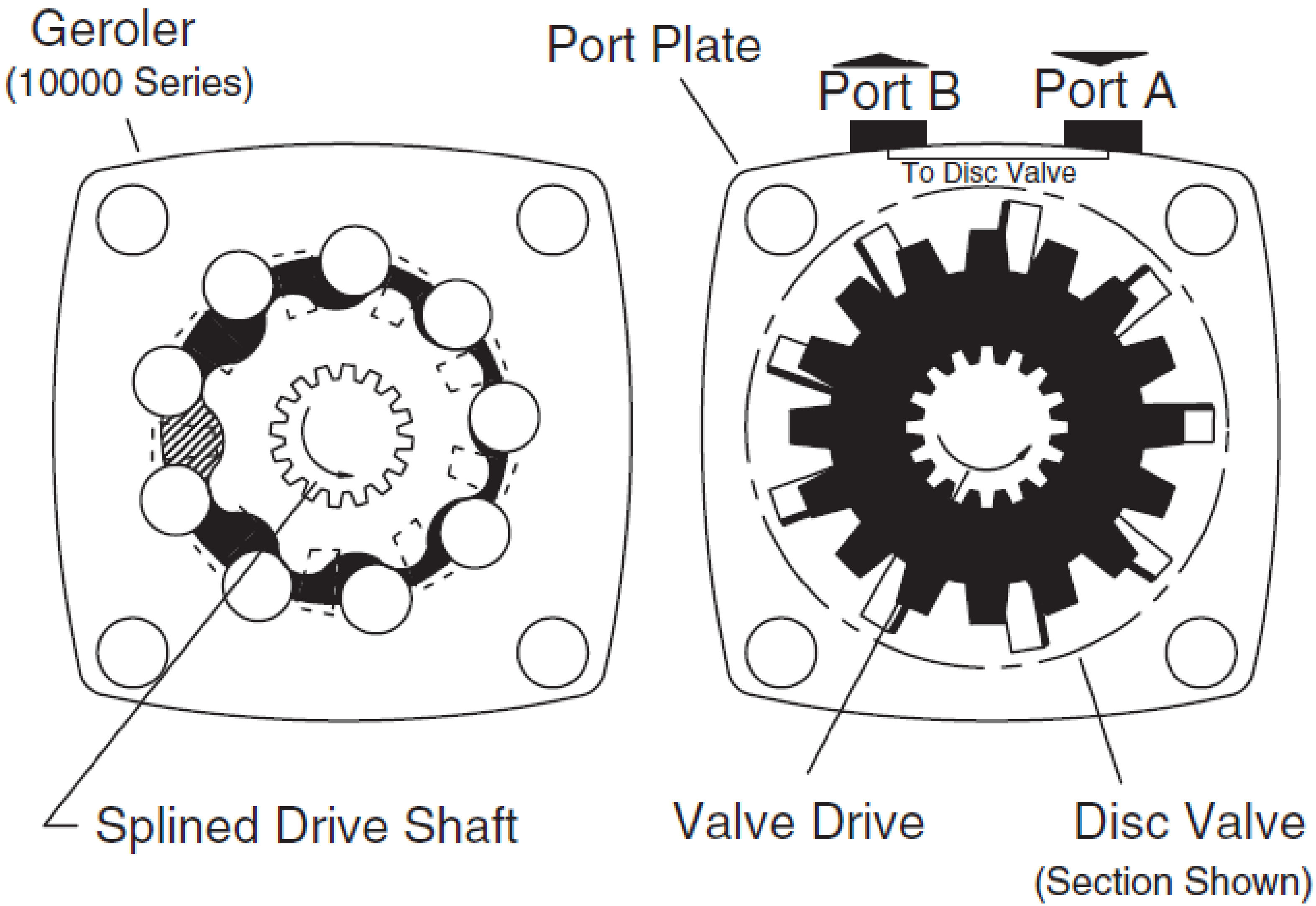


FIGURE 5.7
Geroler motor design. (*Source:* courtesy of Eaton Corp., Hydraulic Div.)

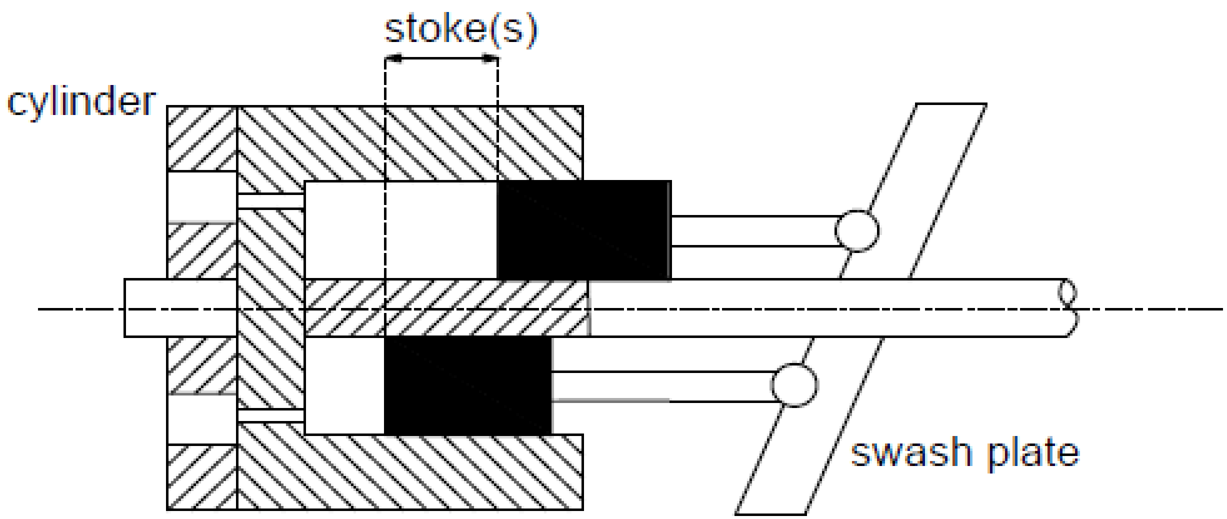


Figure1.5 Swash-plate piston motor

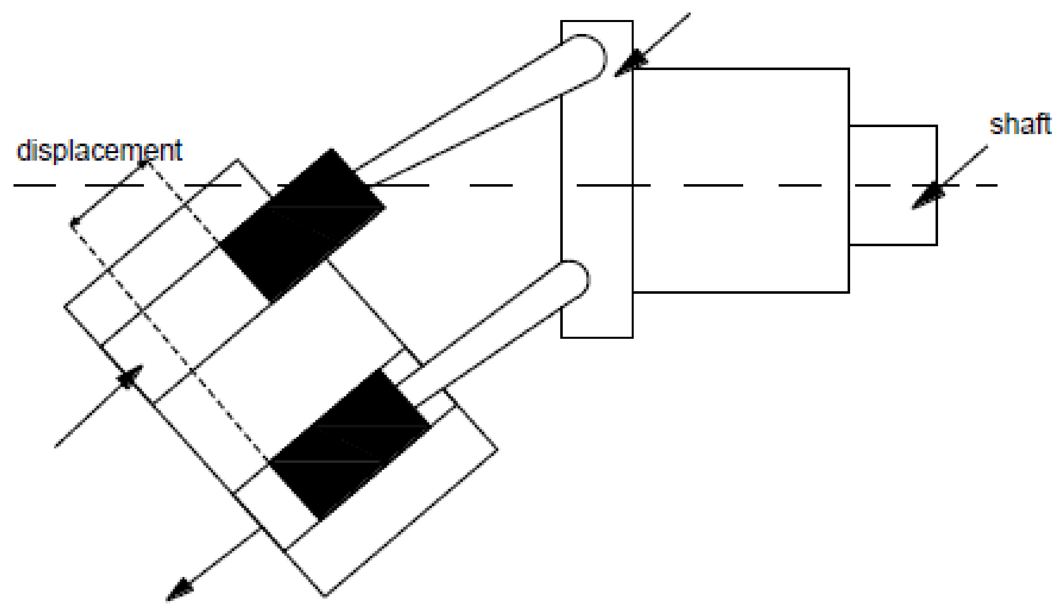


Figure1.6 Inline piston motor

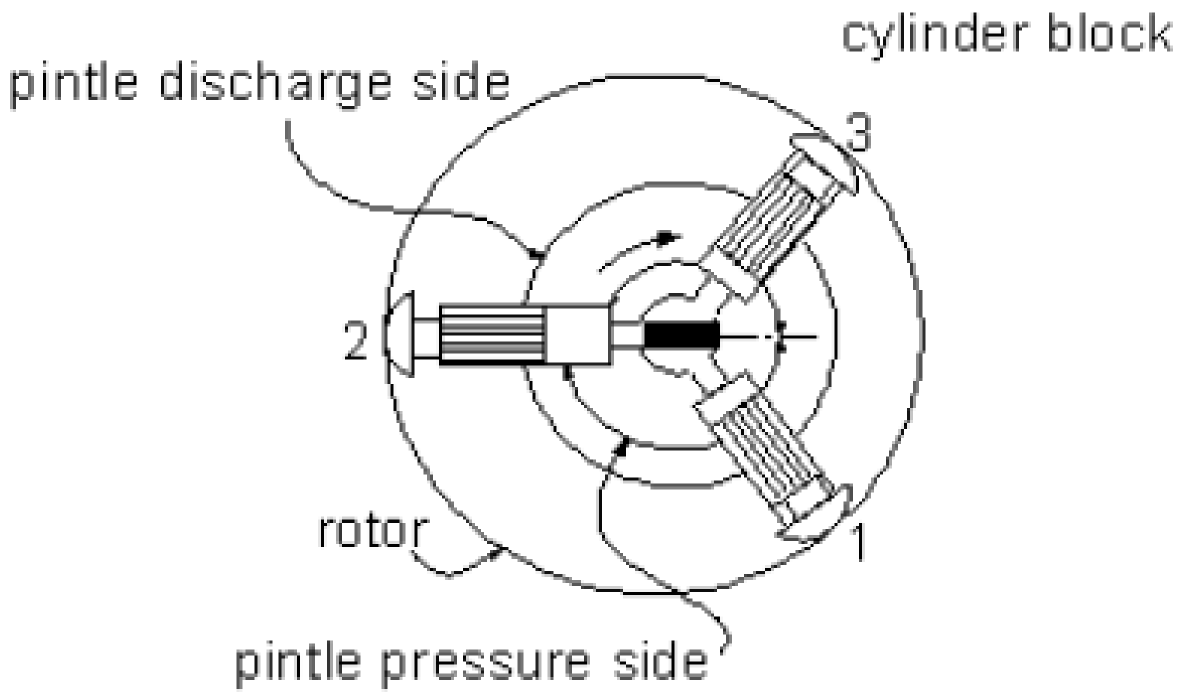


Figure1.8 Radial piston motor

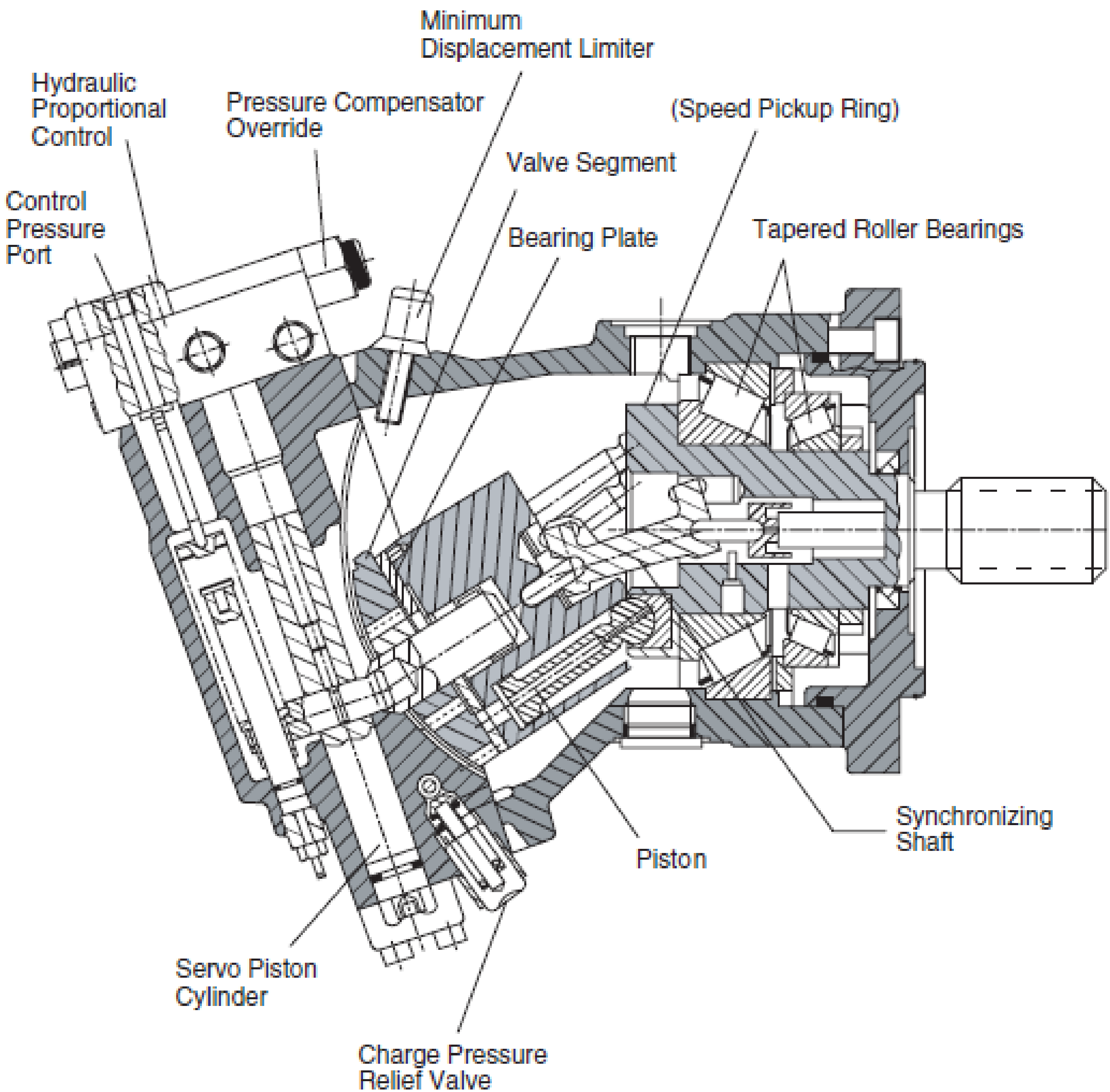
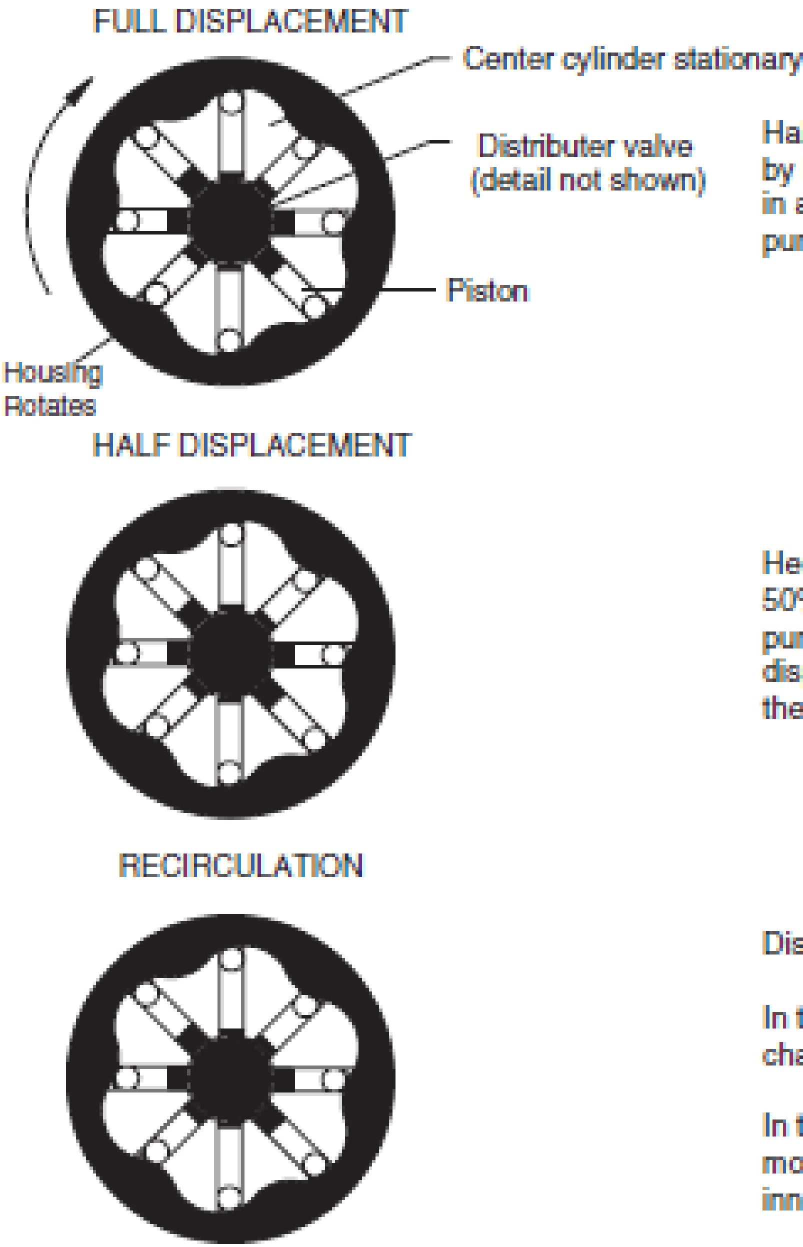


FIGURE 5.17
Diagram of a variable displacement, bent axis motor. (Reprinted with permission from Sauer-



Half the number of the pistons are always in a working stroke, fed by high pressure from the pump. The remaining group of pistons is in a retracting stroke, feeding oil out from the motor and back to the pump.

Here, 25% of the pistons are working, forcing the cam ring to rotate, and 50% of the pistons are on the return stroke, feeding oil back to the pump. Hence, the motor is operating with half its normal rated displacement and thus performing at twice the normal speed and half the normal torque compared to full displacement operation.

Disengagement can be achieved in two different ways:

In the RECIRCULATION mode, all motor ports are connected to charge pressure. The motor can rotate freely within the rated speed.

In the FREEWHEELING mode, all the main connections of the motor are connected to the drain-line. The pistons are forced to their inner position by the case pressure and the motor casing rotates freely.

FIGURE 5.18
Radial piston motor. (*Source:* courtesy of Reidville Hydraulics & Manufacturing.)

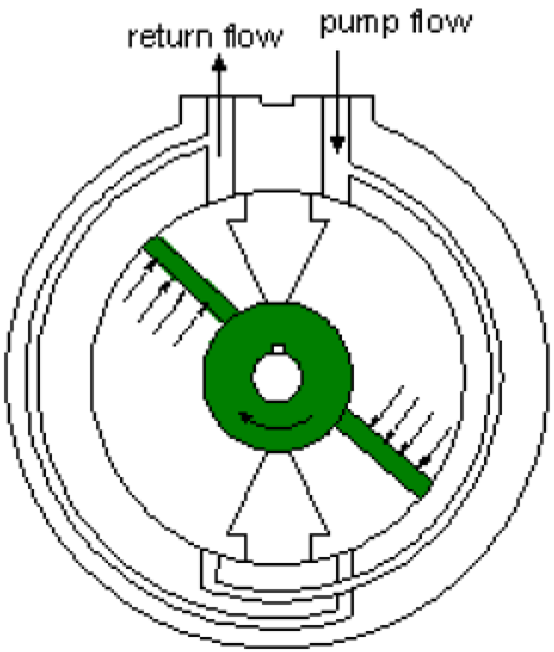


Figure1.10 Two-vane type

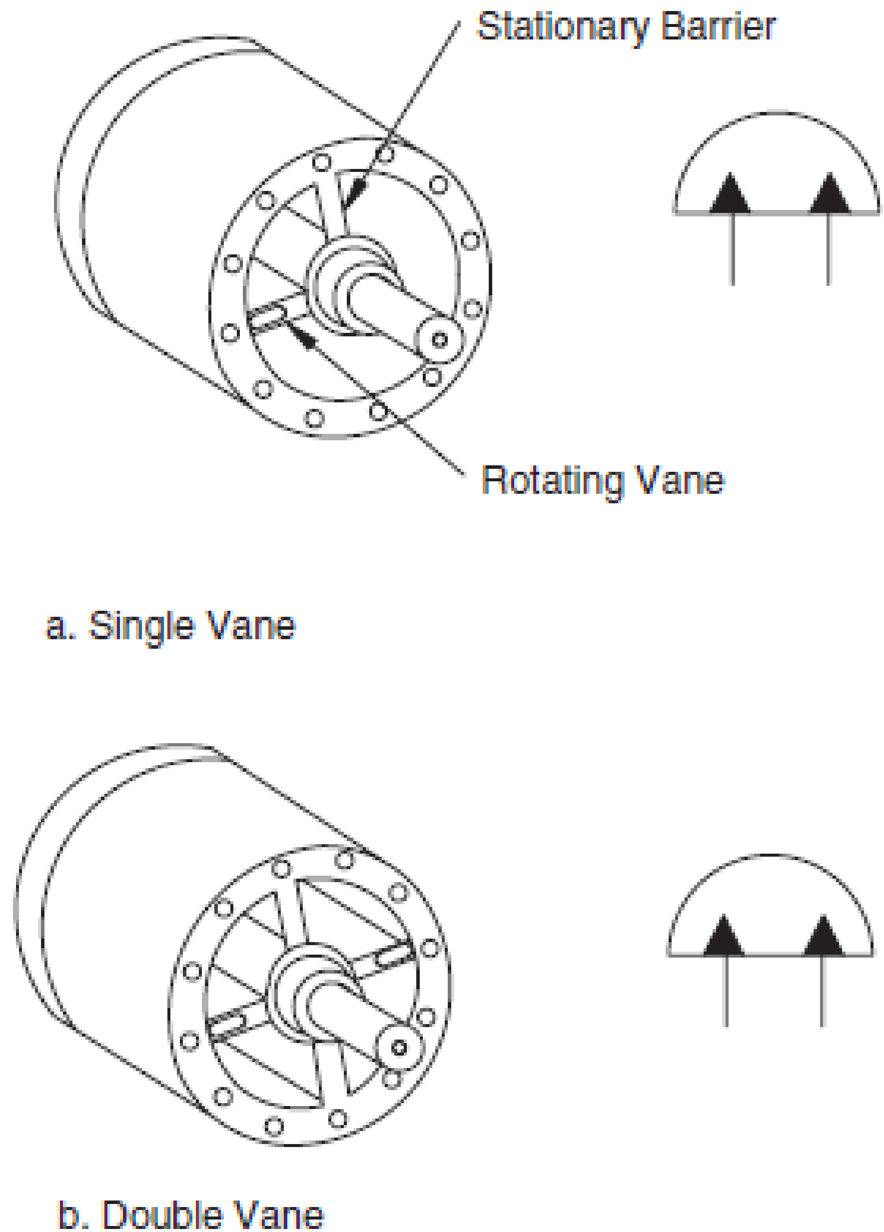


FIGURE 5.21
Single vane motor used for limited-rotation applications.

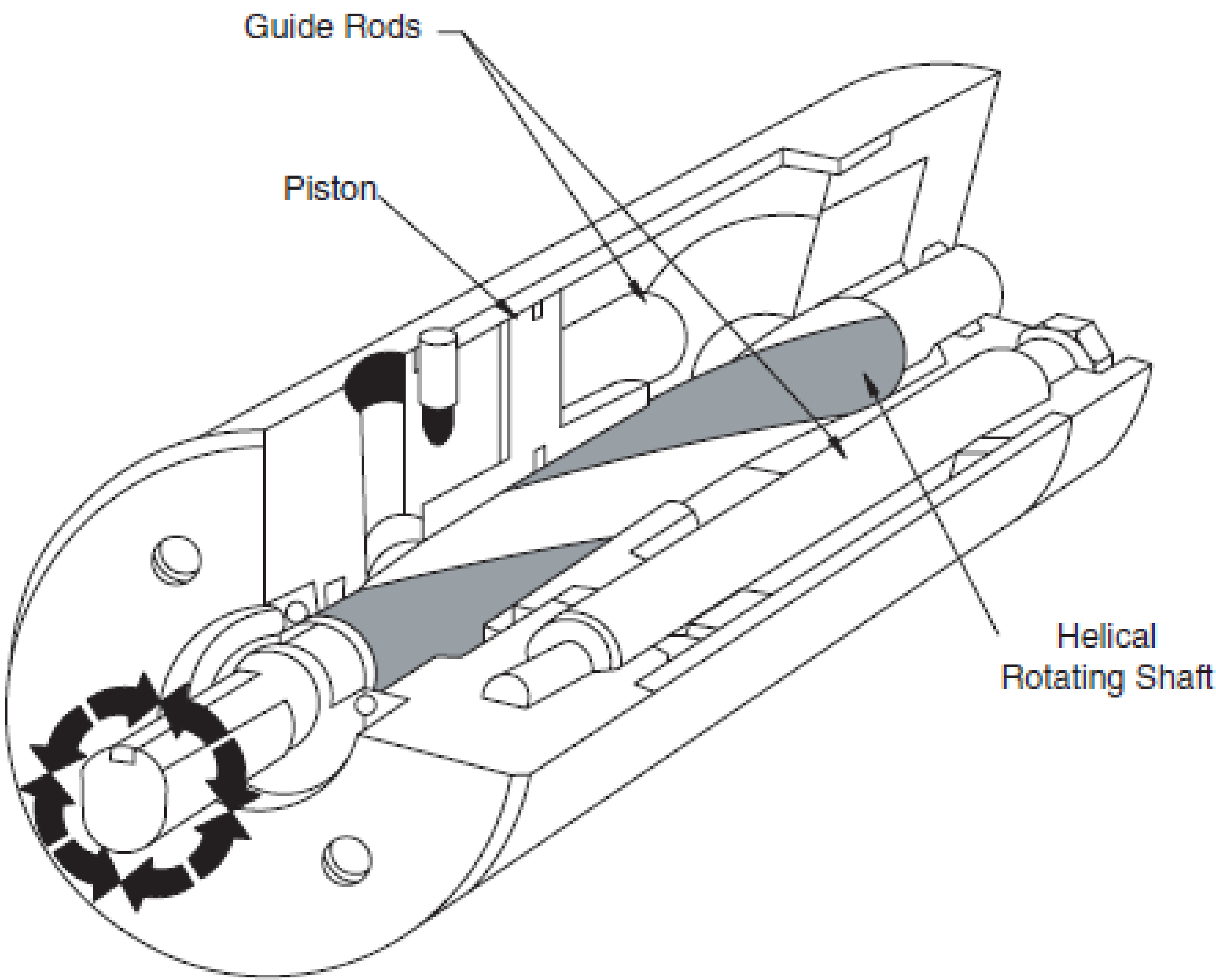


FIGURE 5.22
Limited-rotation actuator with helical shaft.

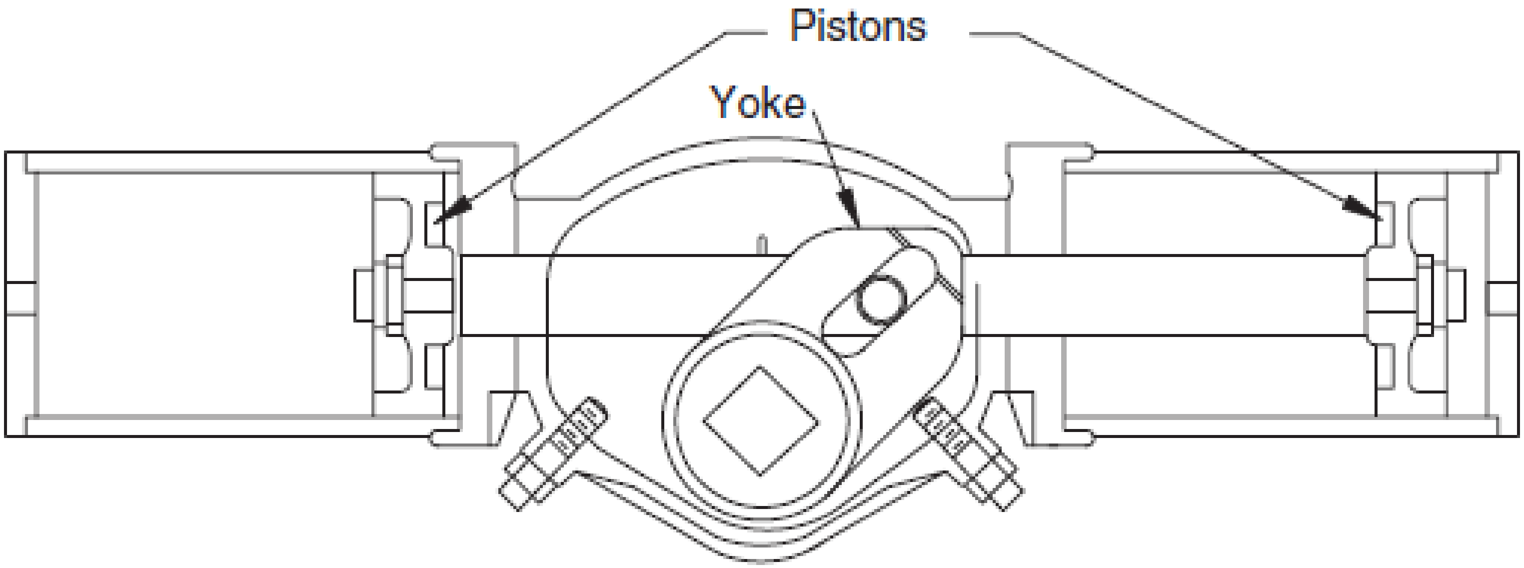


FIGURE 5.23
Skotch yoke rotary actuator.

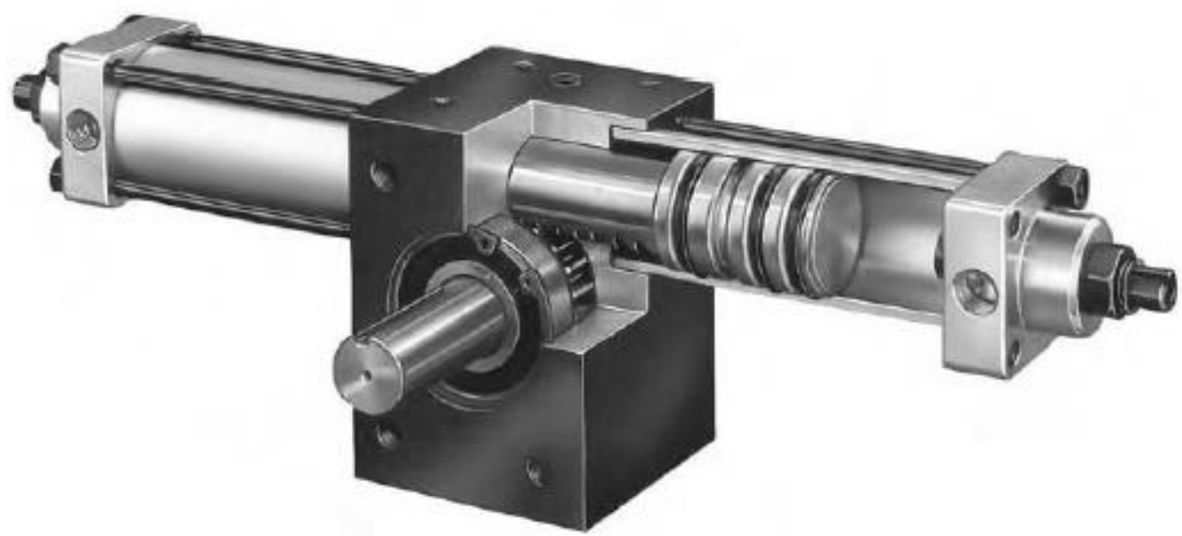


FIGURE 5.24
Rack-and-pinion rotary actuator. (*Source*: courtesy of PHD, Inc., Ft. Wayne, IN.)

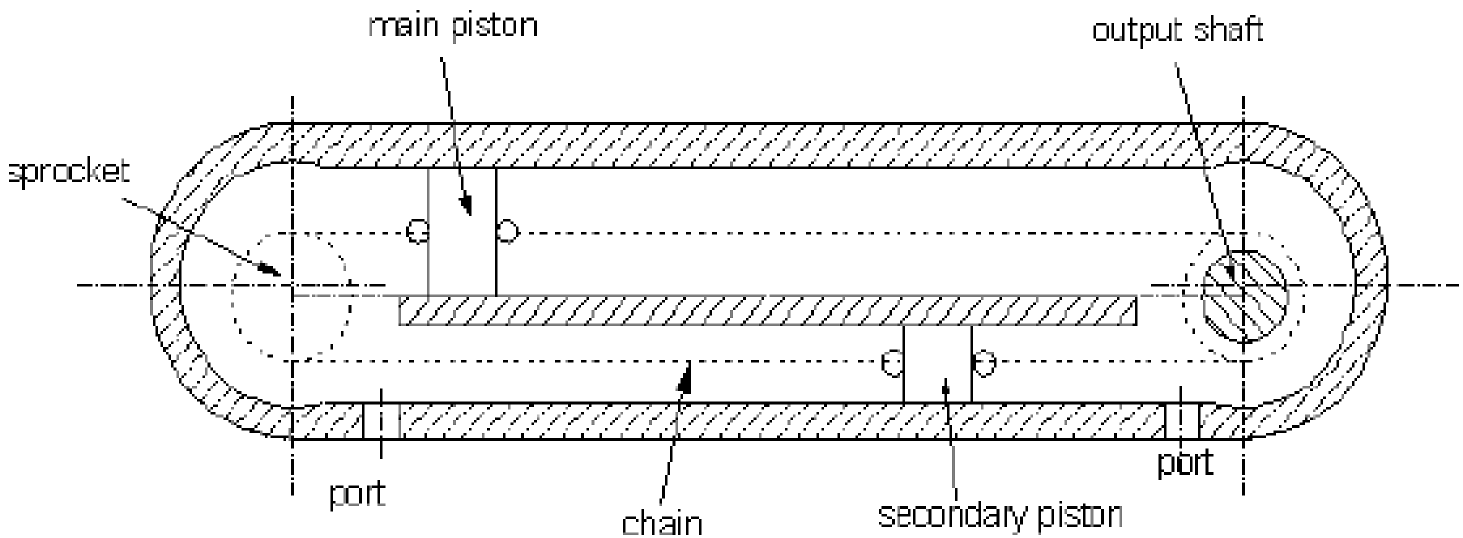


Figure1.11 Chain and sprocket

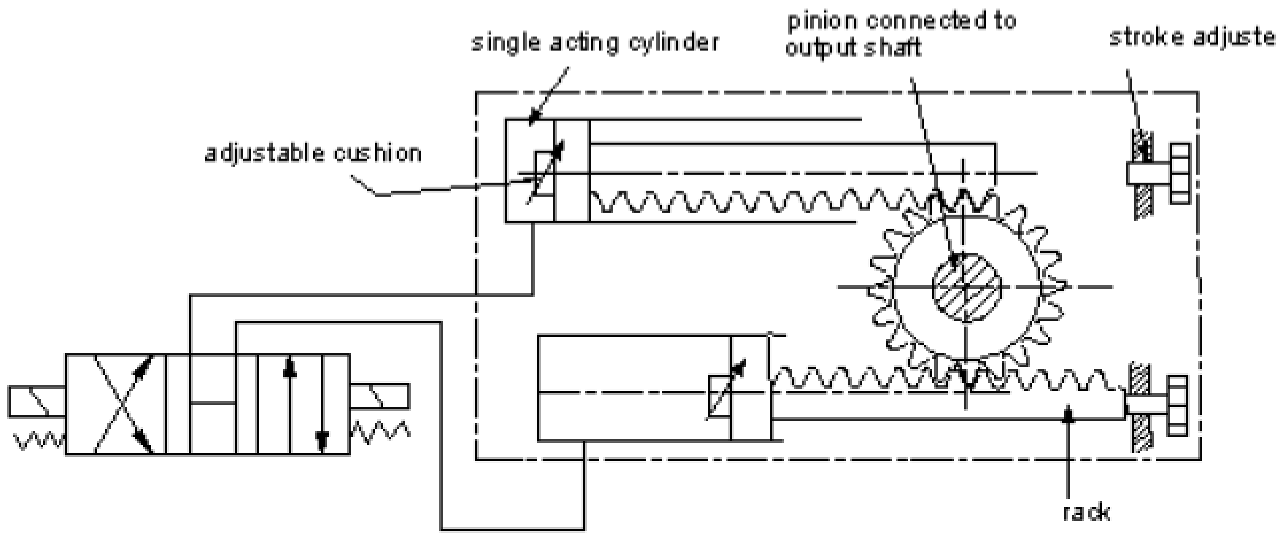


Figure1.12 Rack and rotary actuator

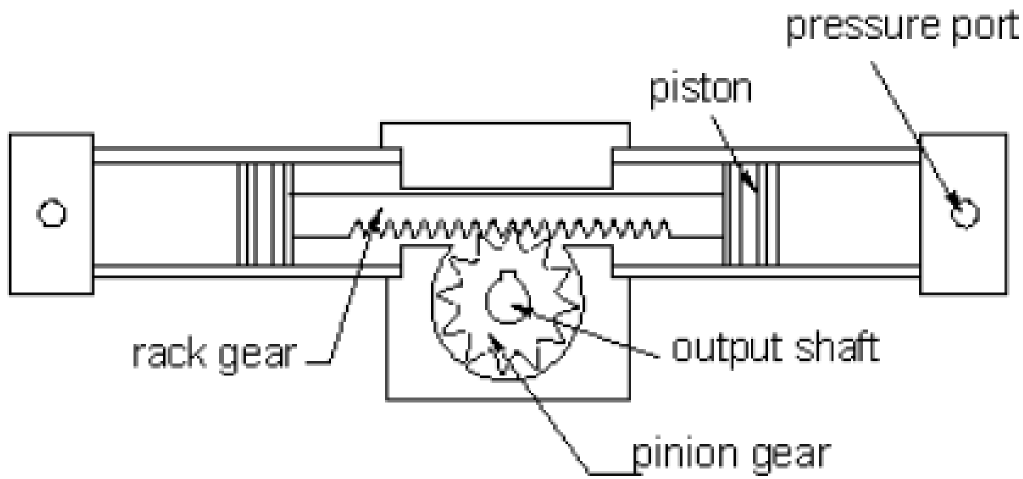


Figure 1.13 Rack and rotary actuator

1.11Hydraulic Motor:Theoretical Torque, Power and Flow Rate

The torque generated by africtionless hydraulic motor is known as a theoretical torque.Theoretical torque can be calculated by the following formula:

$$T_{\mathrm{T}} = \frac{p \times V_{\mathrm{D}}}{2\pi}$$

where V_{D} is the volumetric displacement in m^3/rev and p is the pressure in N/m^2.The power developed by a frictionless motor is known as theoretical power.It can be calculated by the following formula:

$$P_{\mathrm{T}} = T_{\mathrm{T}} \times \omega$$

$$(W) = (\mathrm{Nm})\ \mathrm{rad/s} = W$$

where T_{T} is the theoretical torque in Nm, ω is the speed of the motor in rad/s and $\omega = 2\pi N / 60$, where N is the speed of the motor in rev/min.The flow ratea hydraulic motor would consume if there were no leakage is known as the theoretical flow rate Q_{T}.Mathematically, theoretical flow rate is given by

$$Q_{\mathrm{T}} = V_{\mathrm{D}} n$$

where V_{D} is the volumetric discharge in m^3/rev, n is the speed of motor in rev/s = N/60 and N is the speed of motor in rpm.

1. Volumetric efficiency: The volumetric efficiency of a hydraulic motor is the ratio of theoretical flow rate to actual flow rate required to achieve a particular speed. The motor uses more flow than the theoretical due to leakage:

$$\eta_{\mathrm{v}} = \frac{\text{Theoretical flow rate the motor should be supplied with}}{\text{Actual flow rate supplied to the motor}} = \frac{Q_{\mathrm{T}}}{Q_{\mathrm{A}}}$$

2. Mechanical efficiency: The mechanical efficiency of a hydraulic motor is the ratio of actual work done to the theoretical work done per revolution. The output torque of a hydraulic motor is less than theoretical torque due to mechanical friction between the mating parts:

$$\eta_{\mathrm{m}} = \frac{\text{Actual torque delivered by the motor}}{\text{Torque the motor should theoretically deliver}} = \frac{T_{\mathrm{A}}}{T_{\mathrm{T}}}$$

Here,theoretical torque and actual torque are given by

$$T_{\mathrm{T}} = \frac{V_{\mathrm{D}} \times p}{2\pi}$$

$$T_{\mathrm{A}} = \frac{\text{Actual wattage delivered by the motor}}{N}$$

3. Overall efficiency: The overall efficiency of a motor is the ratio of output power to input power of the motor. Output power is mechanical power output at the shaft and input power is fluid energy supplied to the inlet of the hydraulic motor:

$$\eta_o = \frac{\text{Actual power delivered by the motor (mechanical)}}{\text{Actual power delivered to the motor (hydraulic)}}$$

$$\eta_o = \frac{T_A \times N}{p \times Q_A}$$

$$= \frac{T_A \times T_T \times N}{T_T \times p \times Q_A}$$

$$= \frac{T_A \times V_D \times p \times N}{T_T \times p \times Q_A \times 2\pi}$$

$$= \frac{T_A \times Q_T}{T_T \times Q_A}$$

$$\Rightarrow \eta_o = \eta_V \eta_m$$

Overall efficiency = Volumetric efficiency × Mechanical efficiency

Stall torque is of critical importance for mobile applications. Because of inertia, a high torque is required to start a stationary vehicle. Sometimes hydraulic motors must be sized on the basis of *stall torque* rather than *operating torque* characteristics.

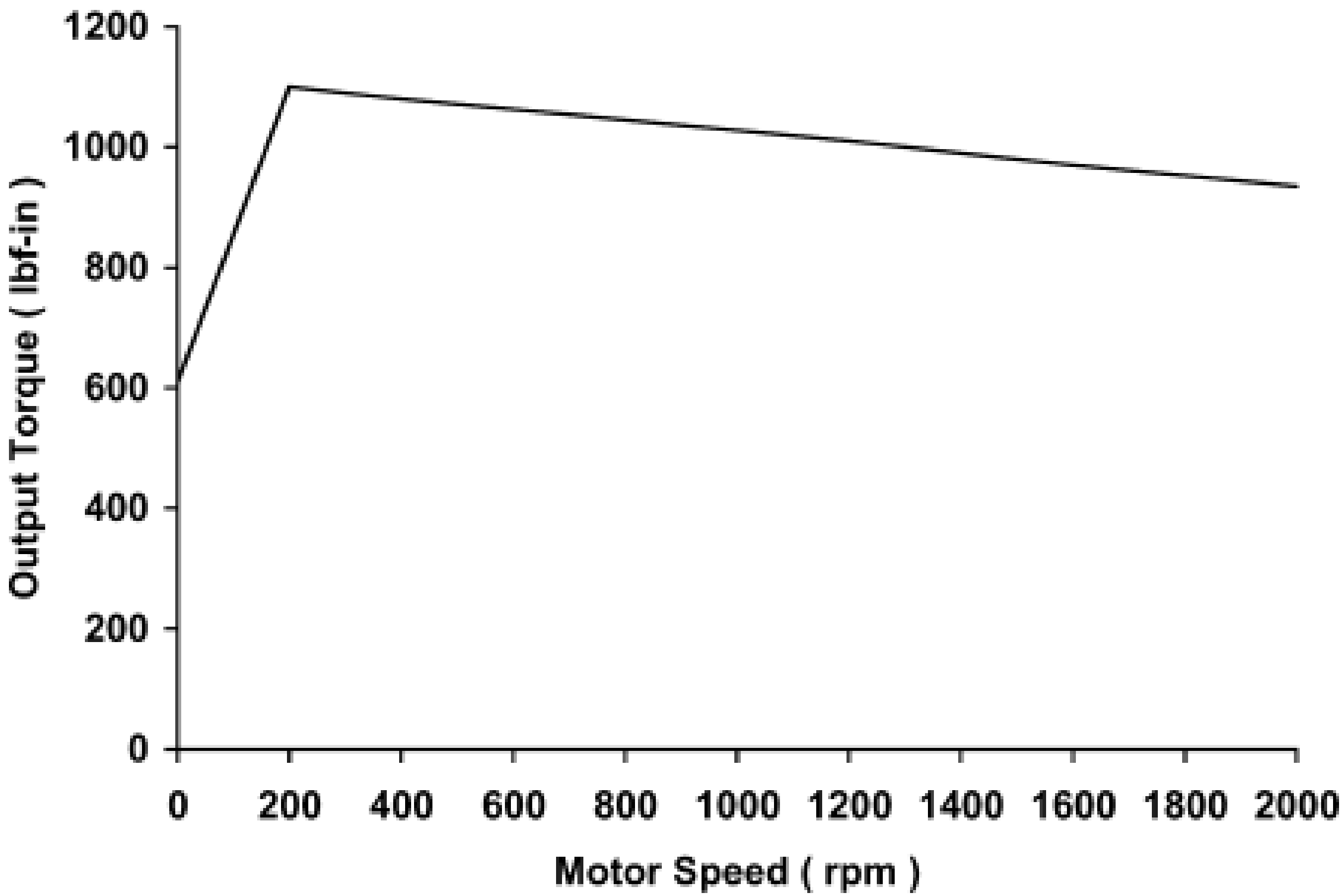


FIGURE 5.1
Torque output vs. motor output speed for typical high-speed hydraulic motor.

Performance Data for Gerotor-Type Motor Supplied with 10 GPM Flow

	Efficiency (%)		
Pressure drop across motor (psi)	**Volumetric, e_{vm}**	**Torque, e_{tm}**	**Overall, e_{om}**
1000	99.7	86.5	86.0
2000	90.5	87.0	78.7
3000	73.5	85.5	62.8

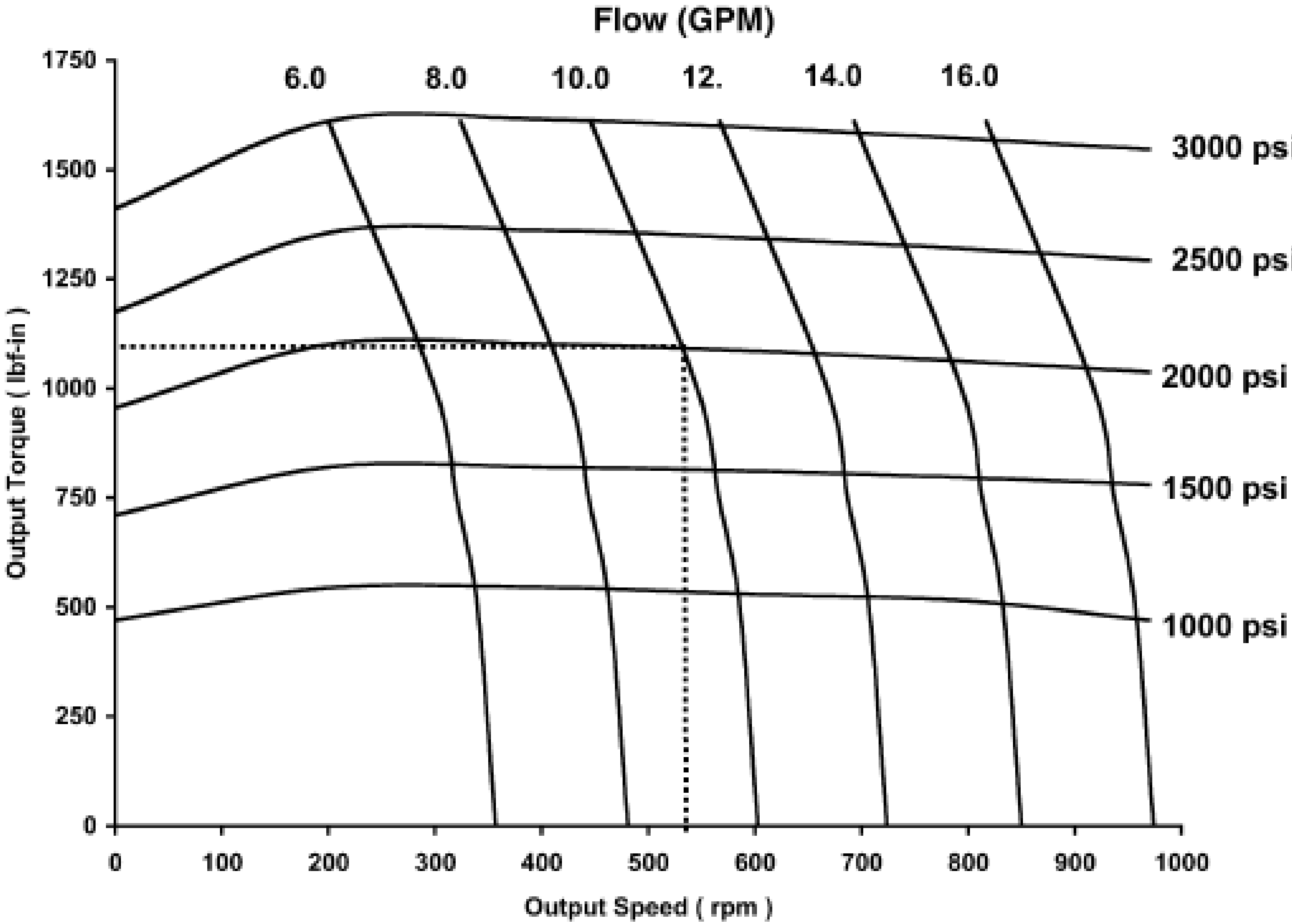


FIGURE 5.2
Performance data for gerotor-type motor (Vickers Model CR-04).

Torque loss is defined by

$$T_\ell = T_{mth} - T$$

where T_{mth} = theoretical torque (lb_f-in)
T = measured torque (lb_f-in)

TABLE 5.2

Torque Loss Data for Gerotor-Type Motor Supplied with 10 GPM Flow

Pressure drop across motor (psi)	Torque loss T_l (lb_f-in)
1000	83.8
2000	161.4
3000	270.0

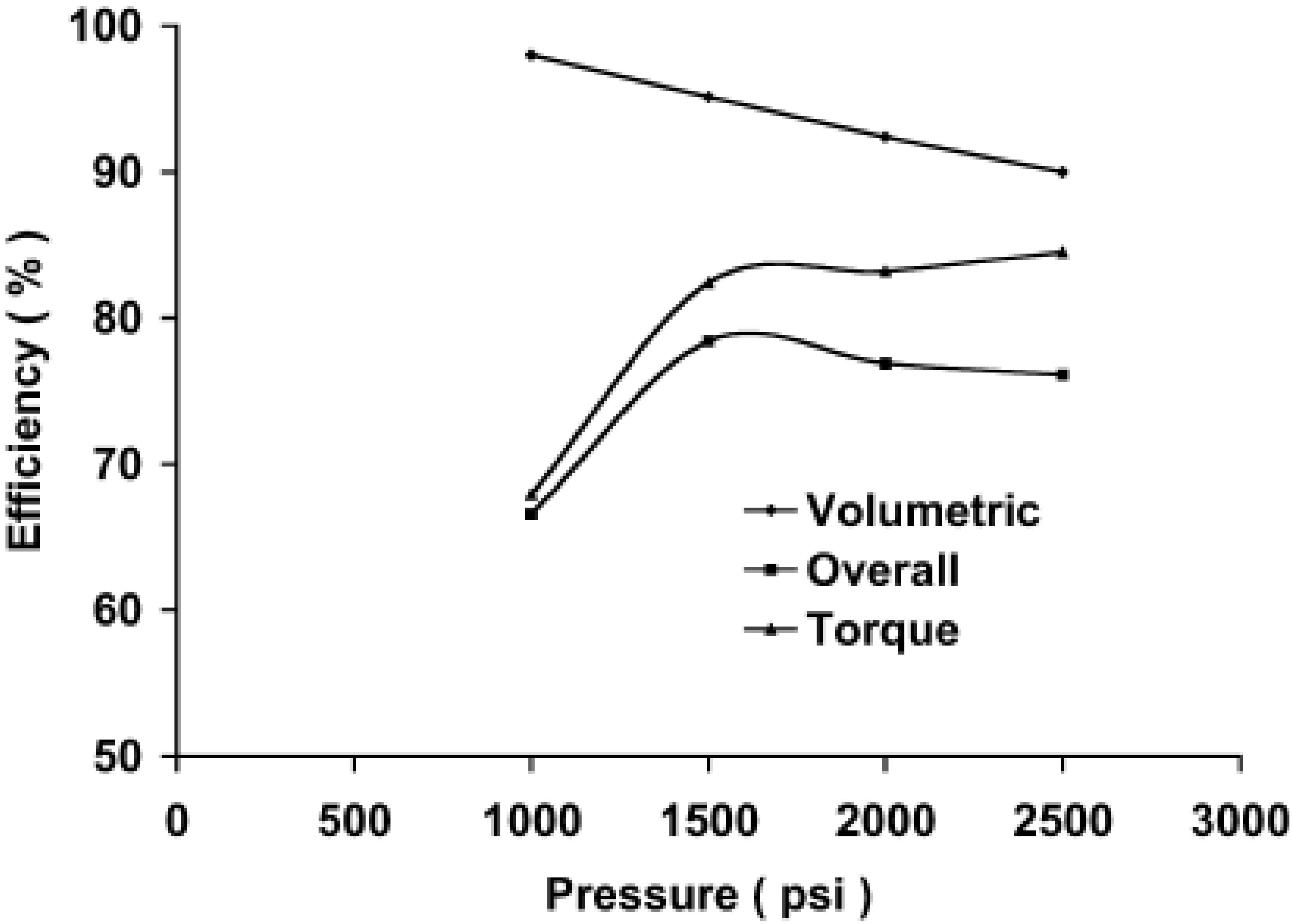


FIGURE 5.3
Efficiencies for gear motor with 36 GPM input flow.

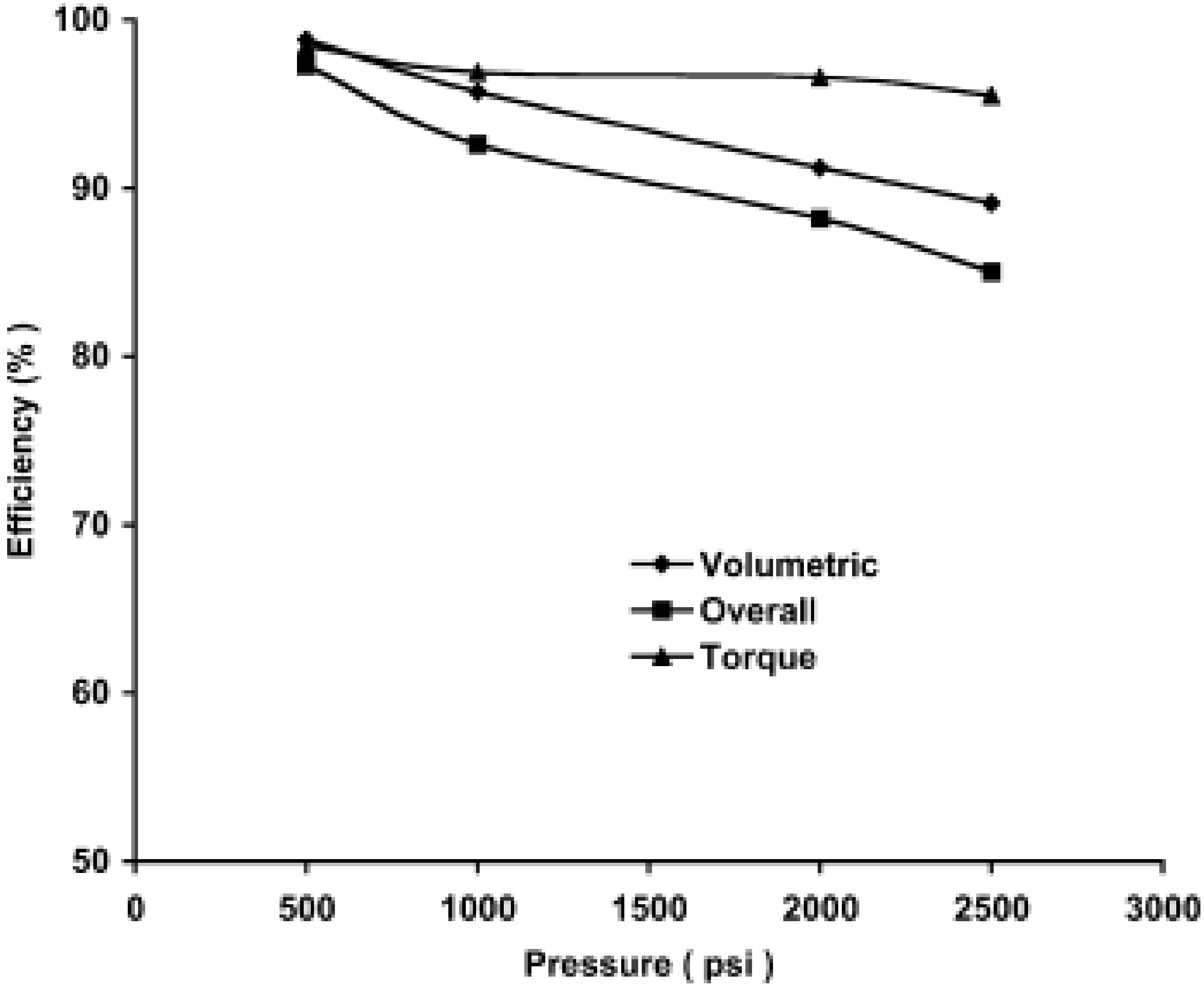


FIGURE 5.4
Efficiencies for vane motor with 35 GPM input flow.

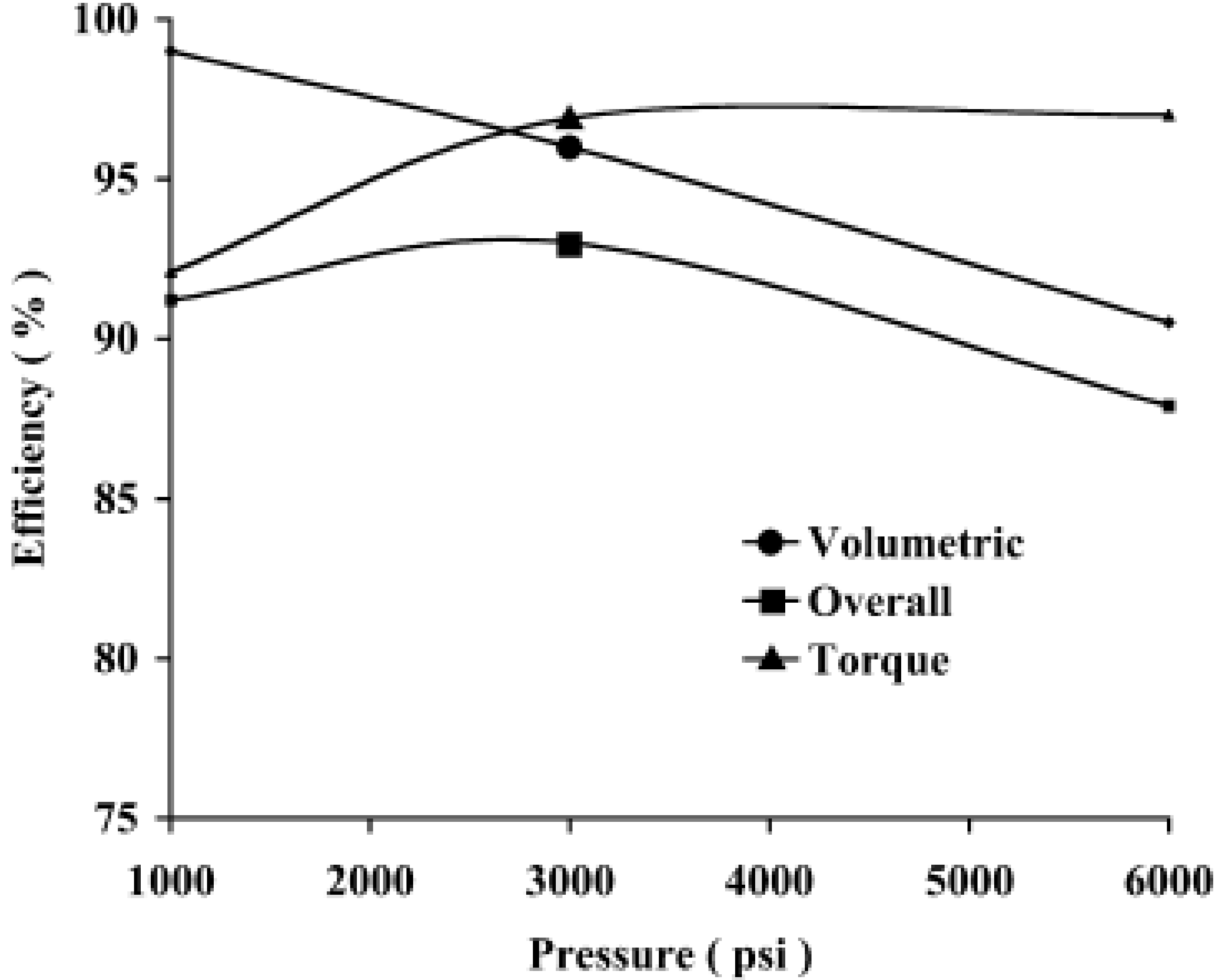


FIGURE 5.5
Efficiencies for piston motor with 36 GPM input flow.

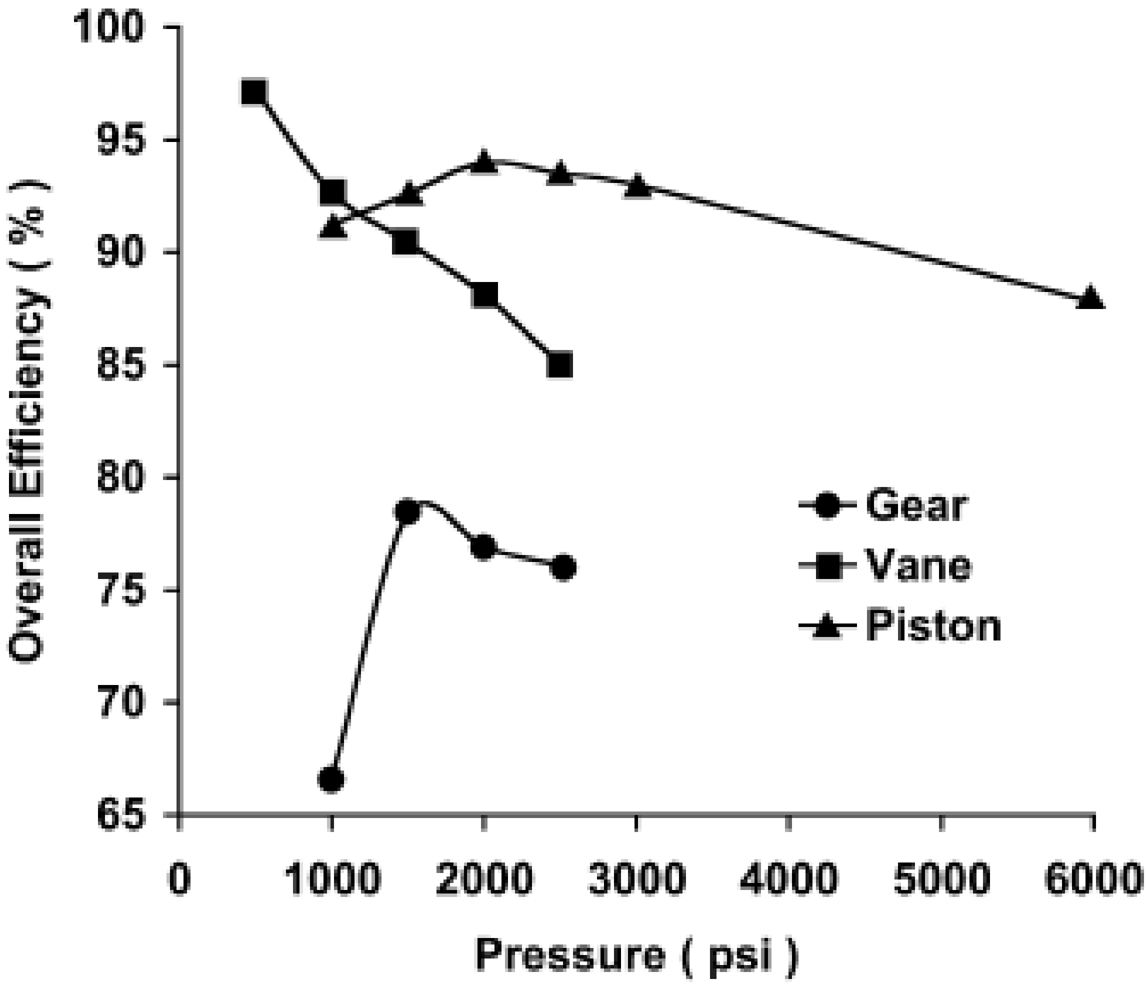


FIGURE 5.6
Comparison of overall efficiencies for gear, vane, and piston high-speed motor at a constant input flow of 35 GPM (nominal).

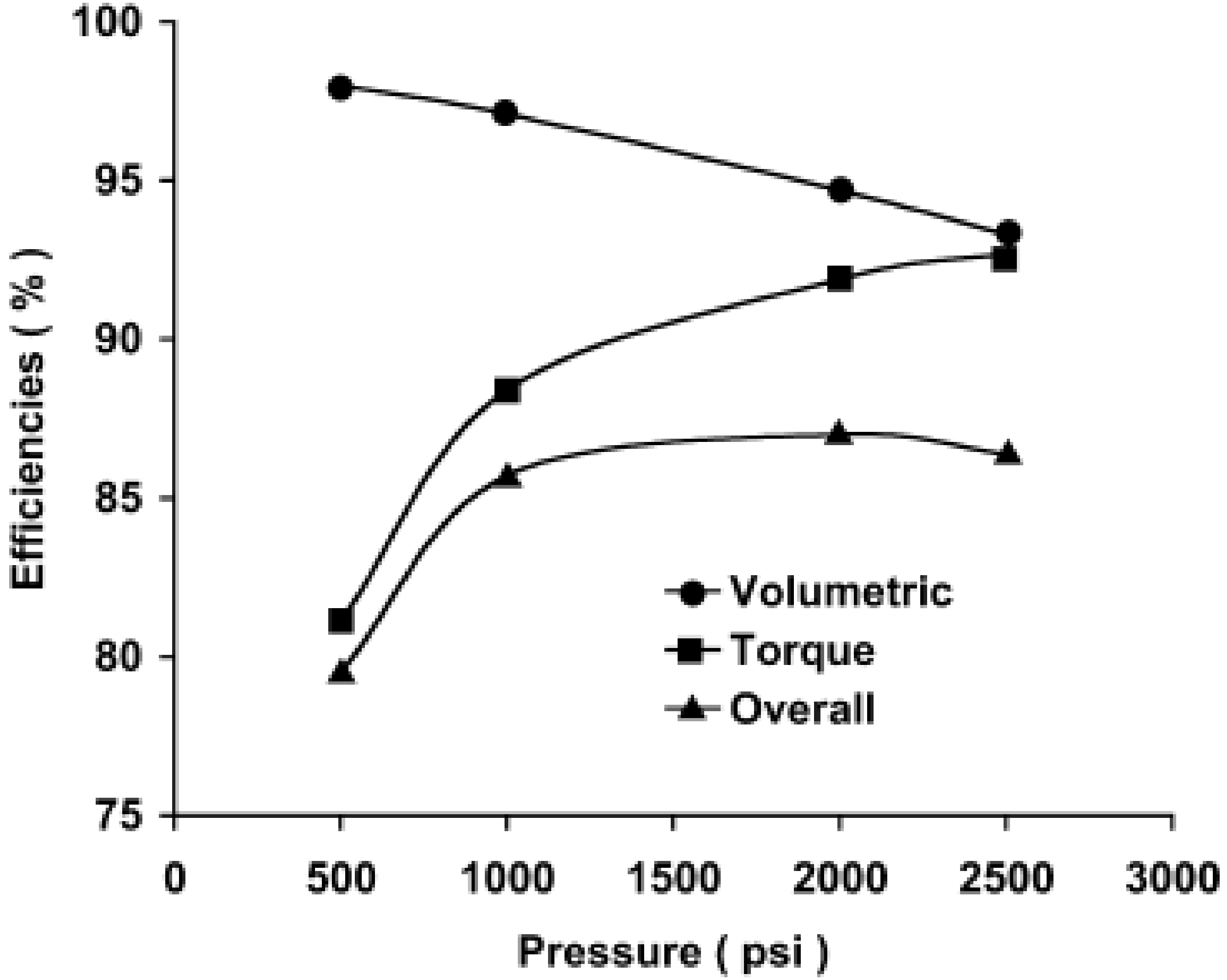


FIGURE 5.8
Efficiency for geroler motor (low-speed, high-torque) with 36 GPM input flow.

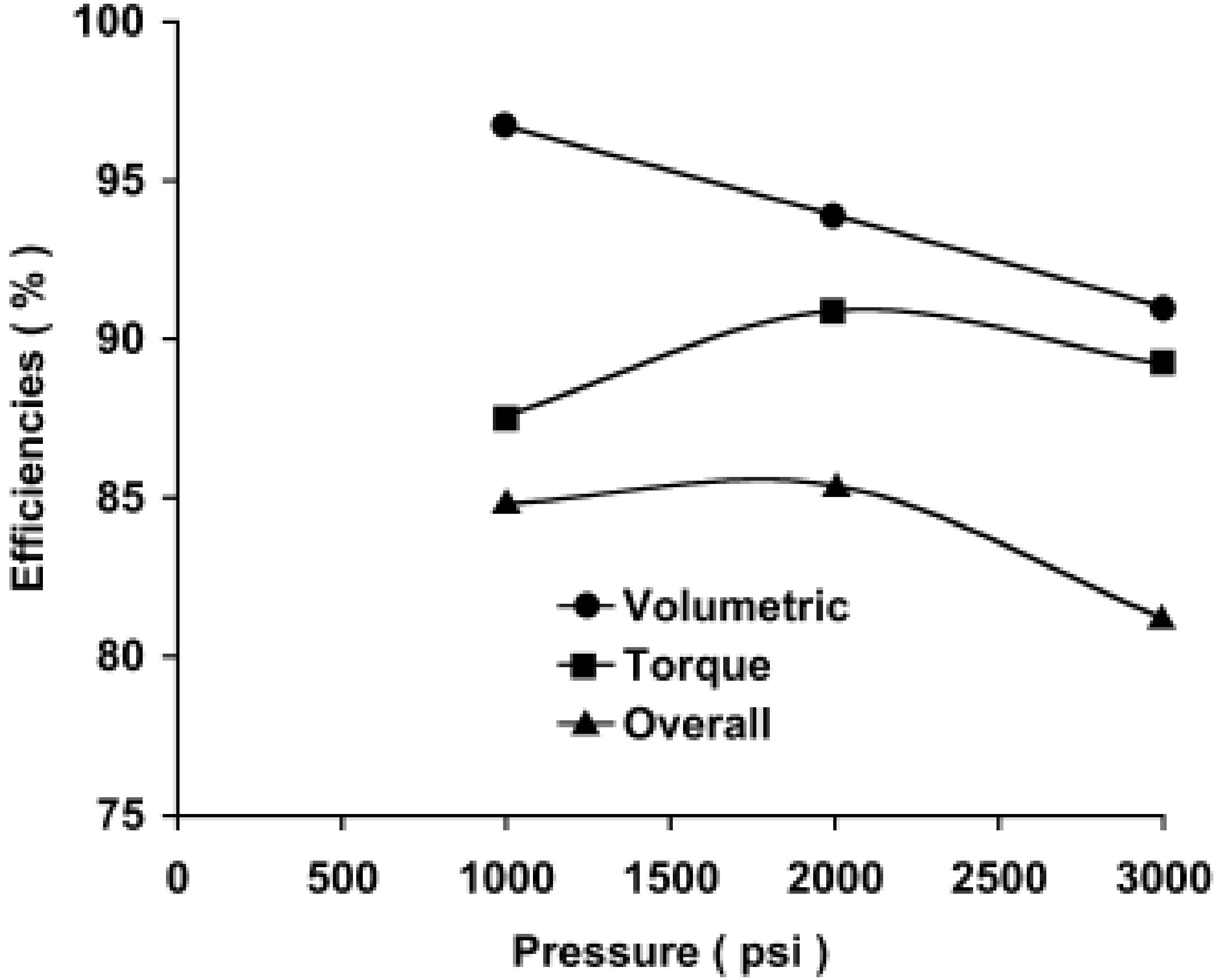


FIGURE 5.9
Efficiency for vane motor (low-speed, high-torque) with 35 GPM input flow.

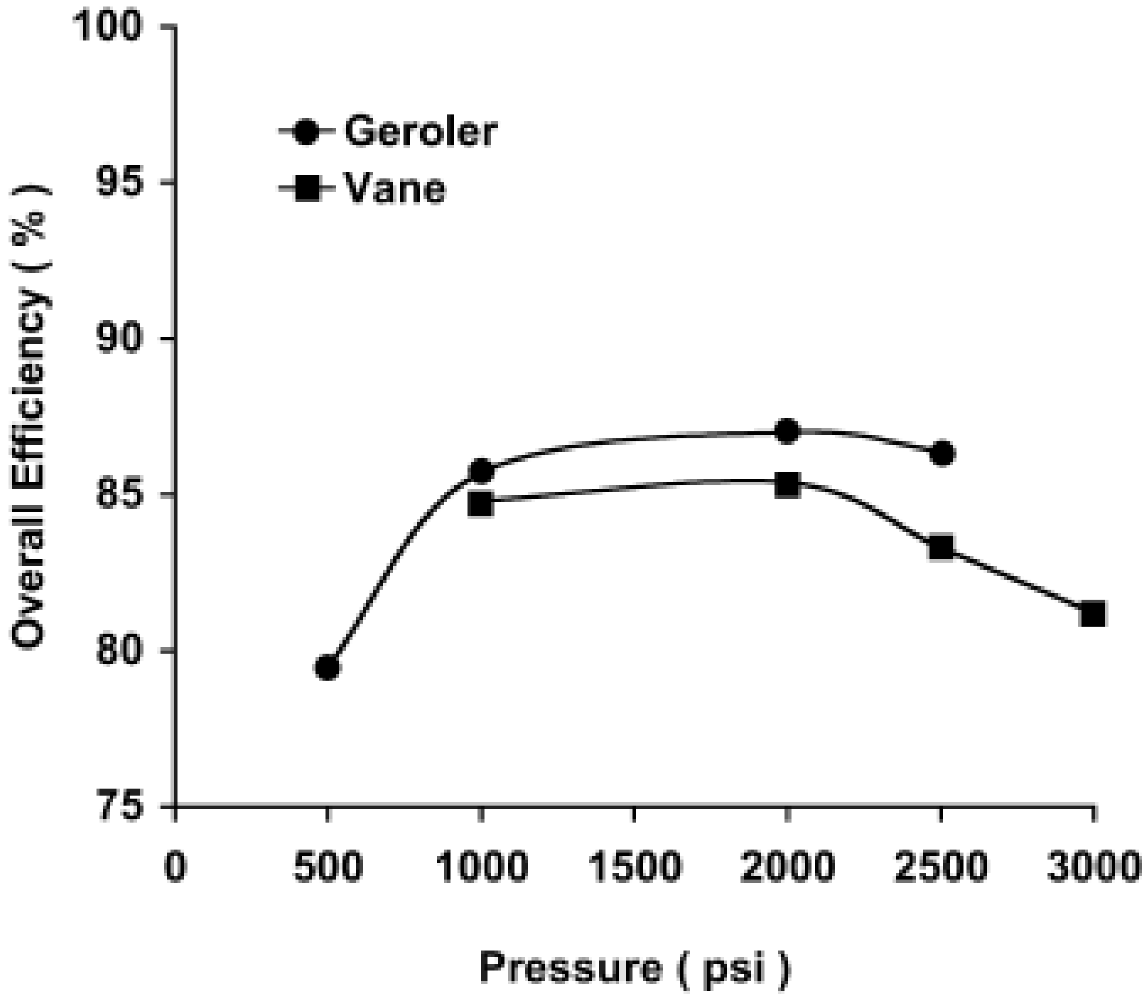


FIGURE 5.10
Comparison of overall efficiency for geroler and vane motor designs (low-speed, high-torque) at a constant flow of 35 GPM (nominal).

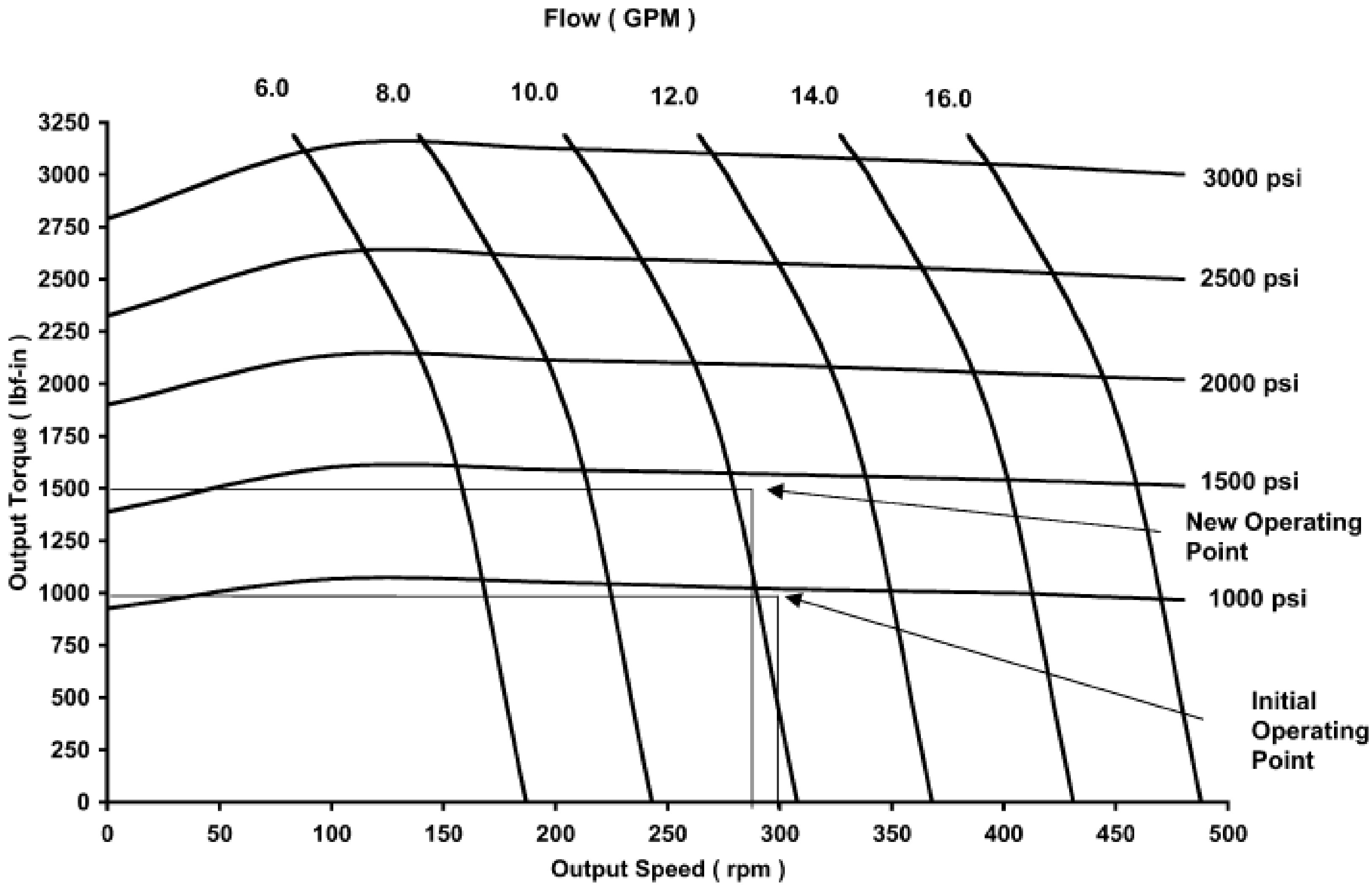


FIGURE 5.11
Manufacturer's test data for low-speed, high-torque motor (Vickers Model CR-08).

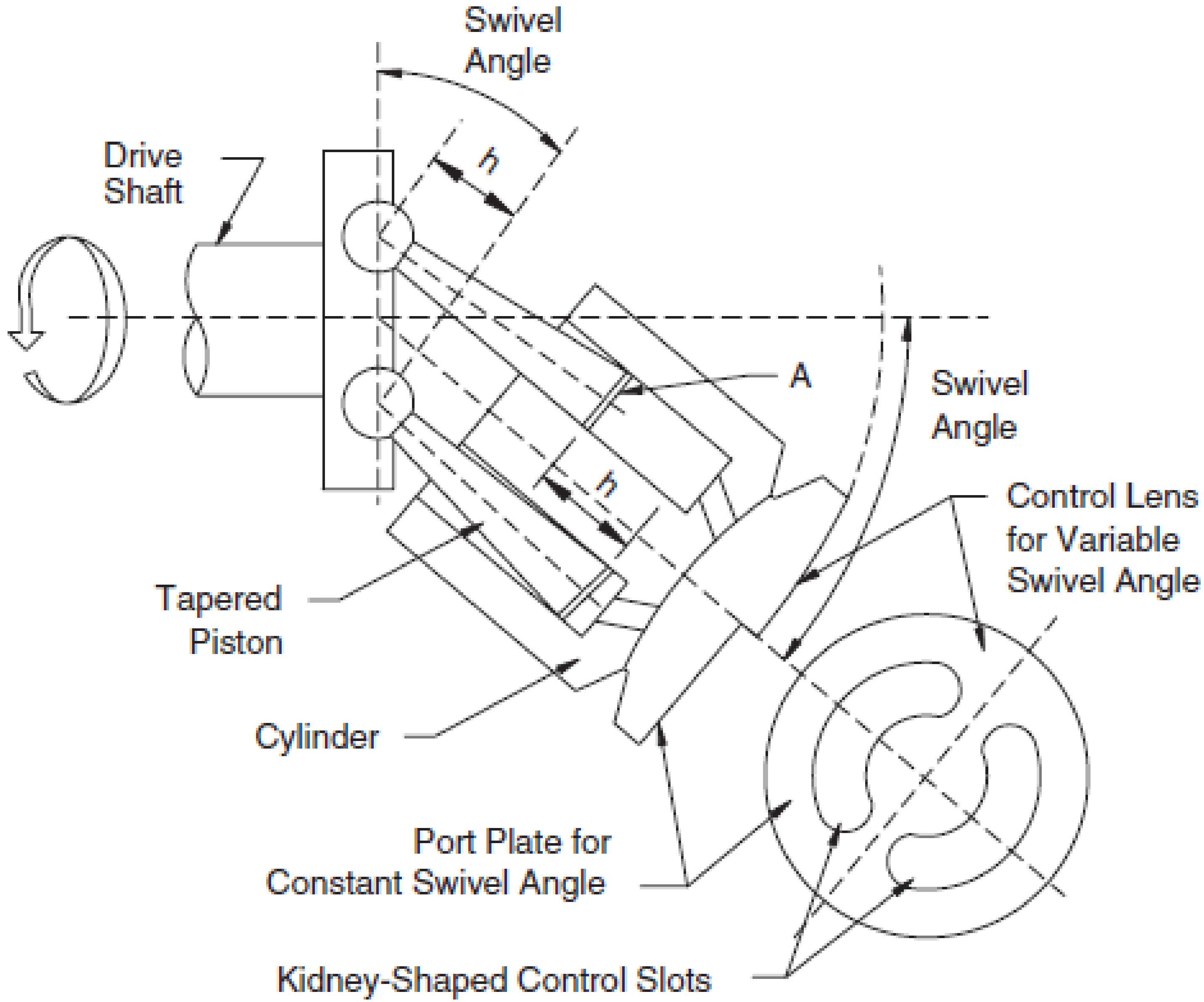


FIGURE 5.15
Diagram of bent axis motor (Dimension A is the area of the individual piston, and h is the stroke.)

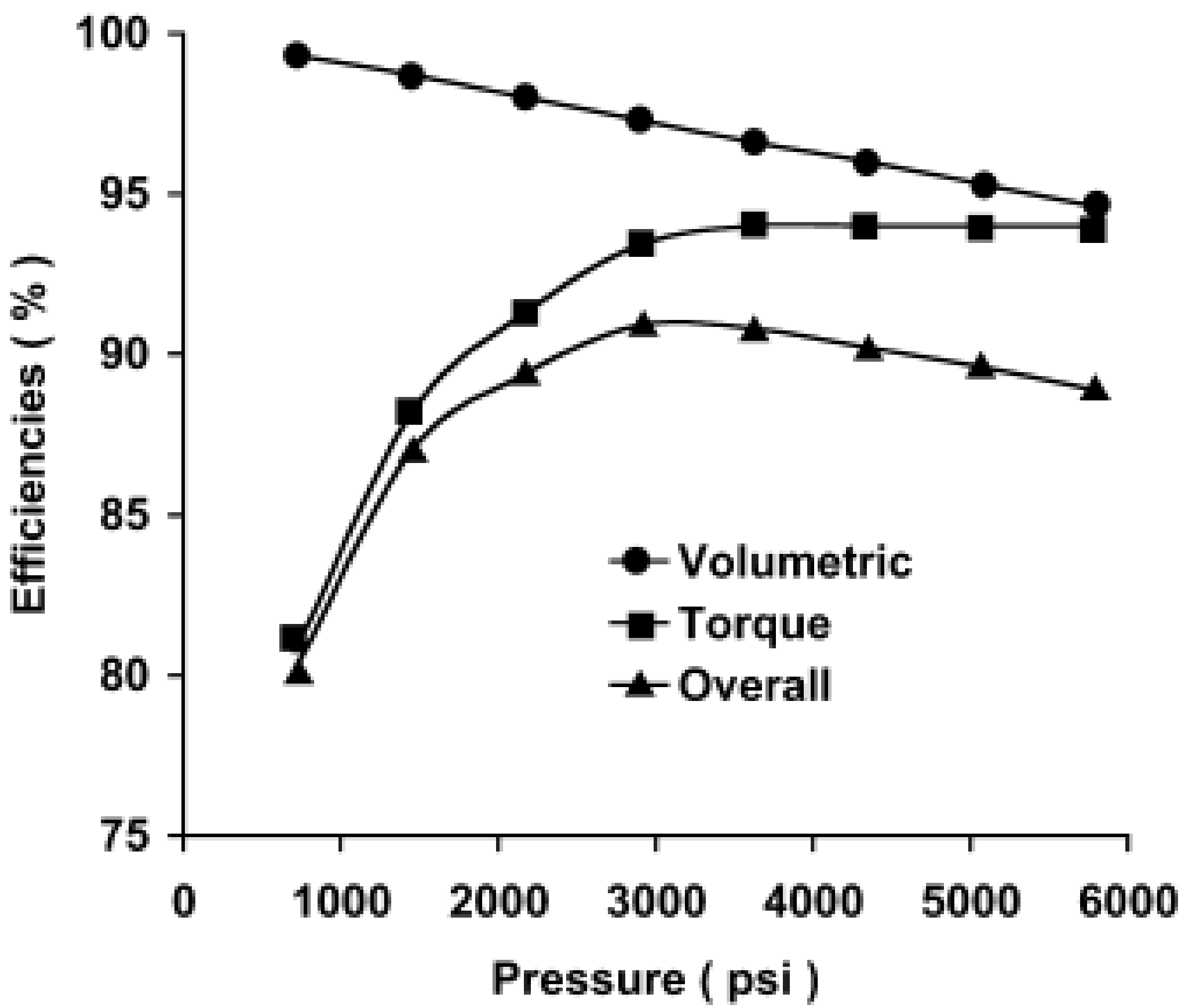


FIGURE 5.16
Efficiencies for bent axis motor operated at 1800 rpm.

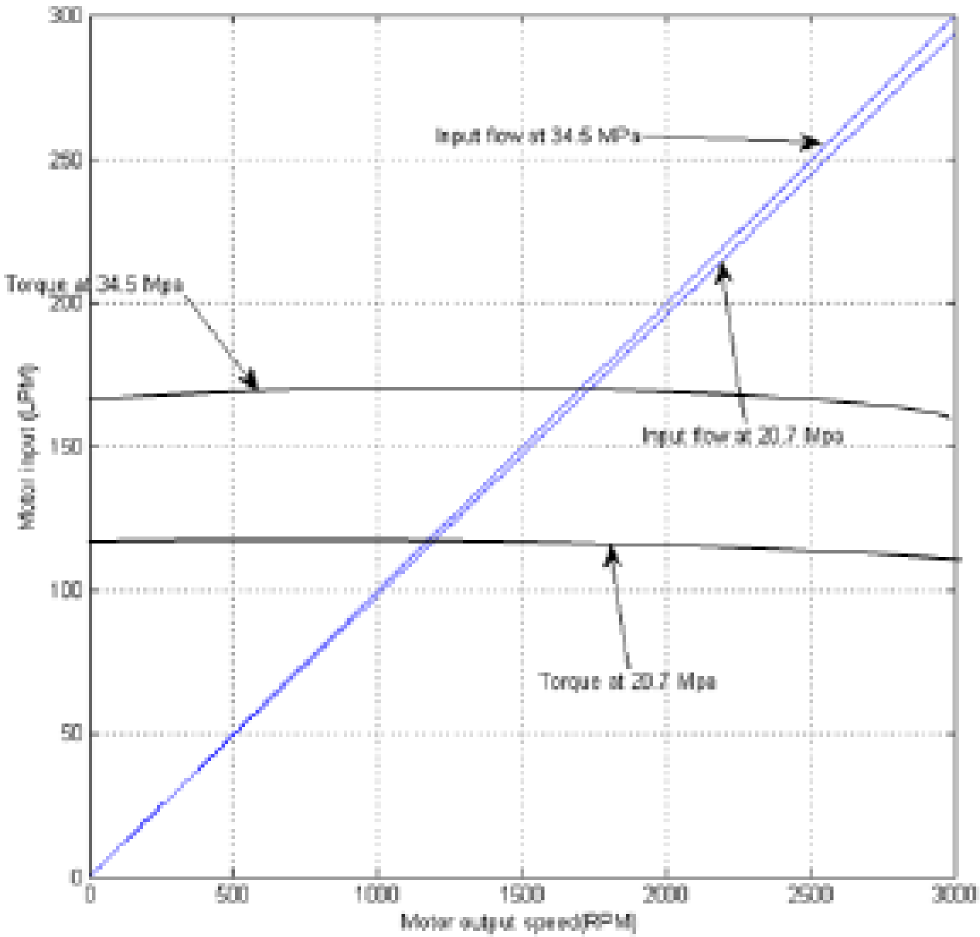


Figure 1.16 Motor input flow versus motor output torque

Figure 1.17gives the curves of overall and volumetric efficiencies as a function of motor speed(RPM) for pressure levels of 34.5 and 20.7 MPa.

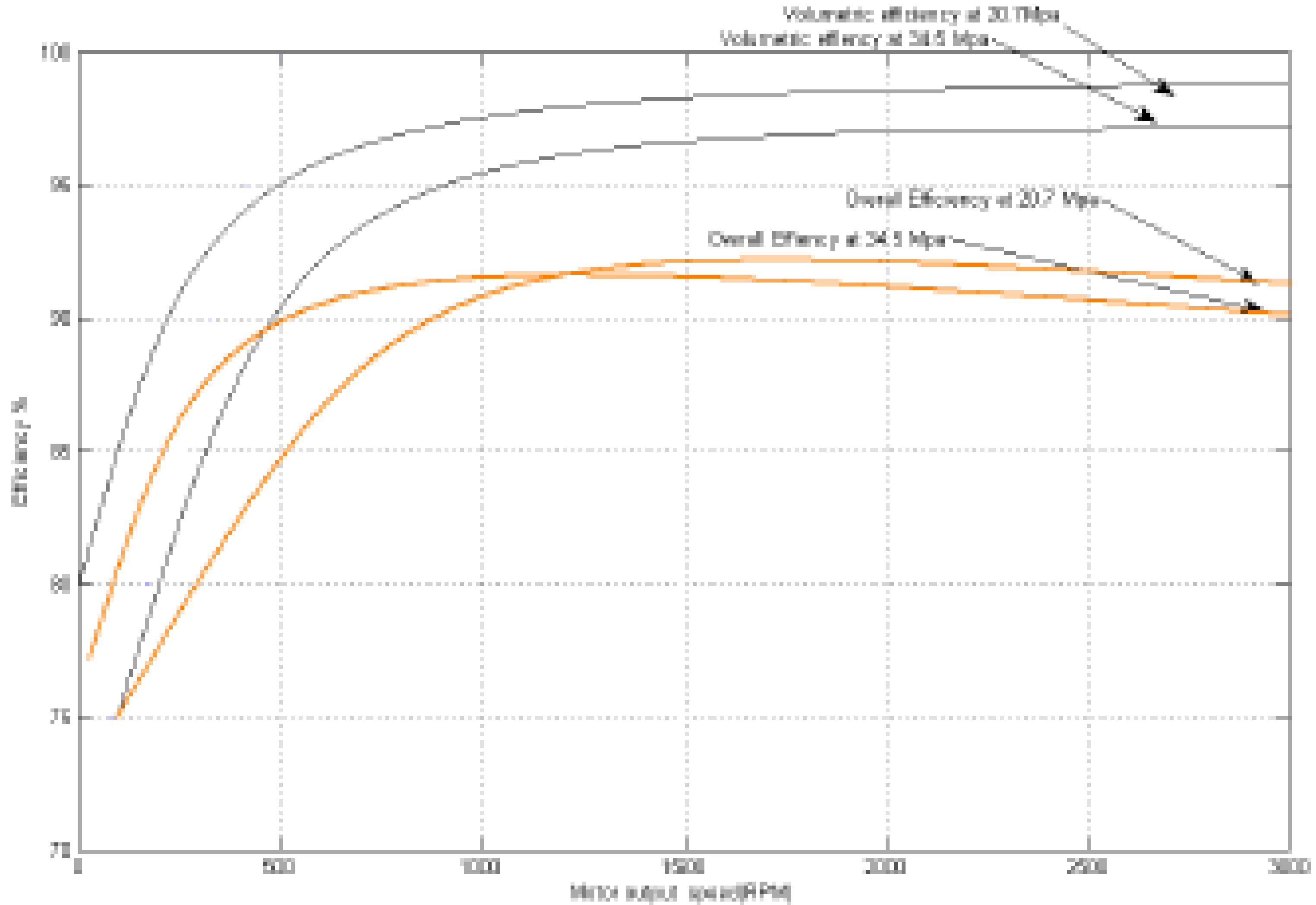


Figure1.17Performance curves for a 100 cm³ variable displacement motor

Example1.3

A hydraulic motor receives a flow rate of 72 LPM at a pressure of 12000 kPa. If the motor speed is 800 RPM, determine the actual torque delivered by the motor assuming the efficiency 100%?

Solution

Method I

Actual flow rate

$$Q_A = 72 \text{ LPM} = \frac{72\times10^{-3}}{60} = 1.2\times10^{-3} \text{ m}^3/\text{s}$$

Speed of motor N = 800 RPM. So

$$\omega = 800\times2\pi/60 = 83.78 \text{ rad/s}$$

Pressure = 12000×10^3 Pa.

Overall efficiency can be calculated using

$$\eta_o = \frac{T_A \times N}{P\times Q_A}$$

Substituting the values we get

$$1 = \frac{T_A \times 83.78}{12000\times10^3 \times1.2\times10^{-3}}$$

$$\Rightarrow T_A = 171.88 \text{ N m}$$

So the actual torque $T_A = 171.88$ N m.

Method II

$$\text{Hydraulic power} = pQ = 12000 \times \frac{72}{60} \times 10^{-3} = 14.4 \text{ kW}$$

$$T\,(\text{Nm}) \times \omega(\text{rad/s}) = 14400 \text{ W}$$

So

$$T = \frac{14400}{800 \times \frac{2\pi}{60}} = 172 \text{ N m}$$

Example 1.4

A hydraulic motor has a 100 cm^3 volumetric displacement. If it has a pressure rating of 140 bar and receives oil from a 0.001 m^3/s theoretical flow rate pump, find the motor (a) speed, (b) theoretical torque, (c) theoretical kW power.

Solution:

(a) Speed: We have the theoretical flow rate given by

$$Q_\text{T} = V_\text{D} \times n$$

$$\Rightarrow 0.001 = 100 \times 10^{6} \times n$$

$$\Rightarrow n = 10 \text{ RPS(revolutions per second)}$$

and

$$N = 600 \text{ RPM}$$

(b) Theoretical torque

$$T_\text{T} = \frac{p \times V_\text{D}}{2\pi} = \frac{140 \times 10^5 \times 100 \times 10^{-6}}{2\pi} = 222.82 \text{ N m}$$

(c) Theoretical kW power

$$P = Q_\text{T} \times p = 0.001 \text{ m}^3/\text{s} \times 140 \times 10^5 \text{ N/m}^2 = 14000 \text{ W} = 14 \text{ kW}$$

Alternately,

$$\text{Power} = T_\text{T}\omega = 222.82 \times 10 \times 2\pi = 14000 \text{ W} = 14 \text{ kW}$$

Example 1.10

A hydraulic motor receives a flow rate of 72 LPM at a pressure of 12000 kPa. If the motor speed is 800 RPM and ifthe motor has a power loss of 3 kW, find the motor actual output torque and overall efficiency.

Solution: We have

$$72\text{ LPM} = 0.0012\ \text{m}^3/\text{s}$$

Now we calculate the hydraulic power given to motor using

Hydraulic power $= pQ = 0.0012\ \text{m}^3/\text{s} \times 12000 = 14400\text{ W} = 14.4\text{ kW}$

Actual power is obtained by subtracting the losses,

$$\text{Actual power} = T\omega = 14.4 - 3 = 11.4\text{ kW}$$

$$\Rightarrow T = \frac{11400}{800 \times \dfrac{2\pi}{60}} = 136\text{ N m}$$

The overall efficiency is

$$\text{Overall efficiency} = \frac{11.4}{14.4} = 0.792 = 79.2\ \%$$

Example 1.9

A hydraulic motor has a displacement of 164 cm^3 and operates with a pressure of 70 bar and a speed of 2000 rpm. If the actual flow rate consumed by the motor is 0.006 m^3/s and the actual torque delivered by the motor is 170 Nm, find (a) η_v, (b) η_m, (c) η_o and (d) actual power delivered by the motor?

Solution:

(a) We have

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

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

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

So volumetric efficiency is

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

(b) Mechanical efficiency is given by

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

Theoretical torque,

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

So mechanical efficiency,

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

(c) We have

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

So overall efficiency is 85.3 %.

(d) Actual power is

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

Example 1.12

A hydrostatic transmission operating at 105 bar pressure has the following characteristics:

Pump	Motor
$V_d = 100\ \text{cm}^3$	$V_d = ?$
$\eta_v = 85\%$	$\eta_v = 94\%$
$\eta_m = 90\%$	$\eta_m = 92\%$
$N = 1000\text{rpm}$	$N = 600\ \text{rpm}$

Find the (a) displacement of motor and (b) motor output torque.

Solution:

(a) Pump theoretical flow rate

$$Q_{\text{T-pump}} = V_d \times N = \frac{100 \times 10^{-6} \times 1000}{60} = 1.667 \times 10^{-3}\ \text{m}^3/\text{s}$$

Actual flow rate

$$Q_{\text{A-pump}} = \eta_V \times Q_T = 1.667 \times 10^{-3} \times 0.85 = 1.42 \times 10^{-3}\ \text{m}^3/\text{s}$$

Actual flow from the pump is the actual flow to the motor. So for the motor

$$Q_{\text{A-motor}} = 1.42 \times 10^{-3}\ \text{m}^3/\text{s}$$

$$Q_{\text{T-motor}} = \eta_V \times Q_A = 1.42 \times 10^{-3} \times 0.94 = 1.332 \times 10^{-3}$$

So the theoretical flow rate, $Q_{\text{T-motor}} = 1.332 \times 10^{-3}\ \text{m}^3/\text{s}$. Now

$$Q_{\text{T-motor}} = V_{\text{D-motor}} \times N$$

$$\Rightarrow V_{\text{D-motor}} = \frac{Q_{\text{T-motor}}}{N_{\text{motor}}} = \frac{1.332 \times 10^{-3}}{600/60} = 1.332 \times 10^{-4} = 133\ \text{cm}^3/\text{rev}$$

So for the motor, the displacement is 133 cm^3/rev.

(b) Torque delivered by the motor

To calculate torque delivered by the motor, let us first calculate the actual power to motor

$$\text{Power}_{\text{actual to motor}} = p\,Q = 105\times10^5 \times 0.00142 = 14900 \text{ W}$$

Now

$$\text{Power}_{\text{actual by motor}} = \text{Power}_{\text{actual to motor}} \times \text{Mechanical efficiency} \times \text{volumetricefficiency}$$

$$\text{Power}_{\text{actual by motor}} = 14900\times0.94\times0.92 = 12900 \text{ W}$$

$$\text{Torque}_{\text{actual by motor}} = \frac{12900}{\dfrac{600\times2\pi}{60}} = 205 \text{ Nm}$$

ME 7553 – Hydraulics and Pneumatics

Lecture -8

Date: 03-04-2021 Time slot: 08:30-10:10 a.m.

Contents

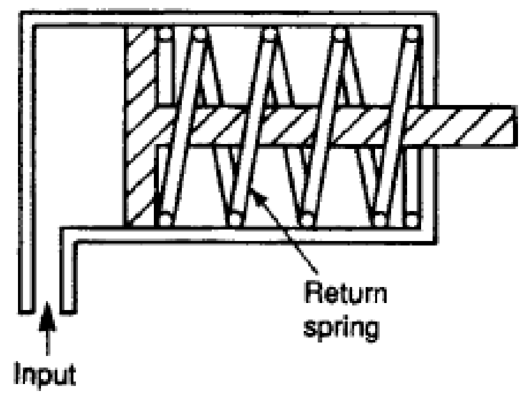
1. Linear Actuators
2. Applications of linear actuators
3. Hydraulic cushioning
4. Problems
5. Directional control valve

Course Instructor: Dr. A. Siddharthan

1.2 Types of Hydraulic Cylinders

Hydraulic cylinders are of the following types:

- Single-acting cylinders.
- Double-acting cylinders.
- Telescopic cylinders.
- Tandem cylinders.



Single-acting cylinder

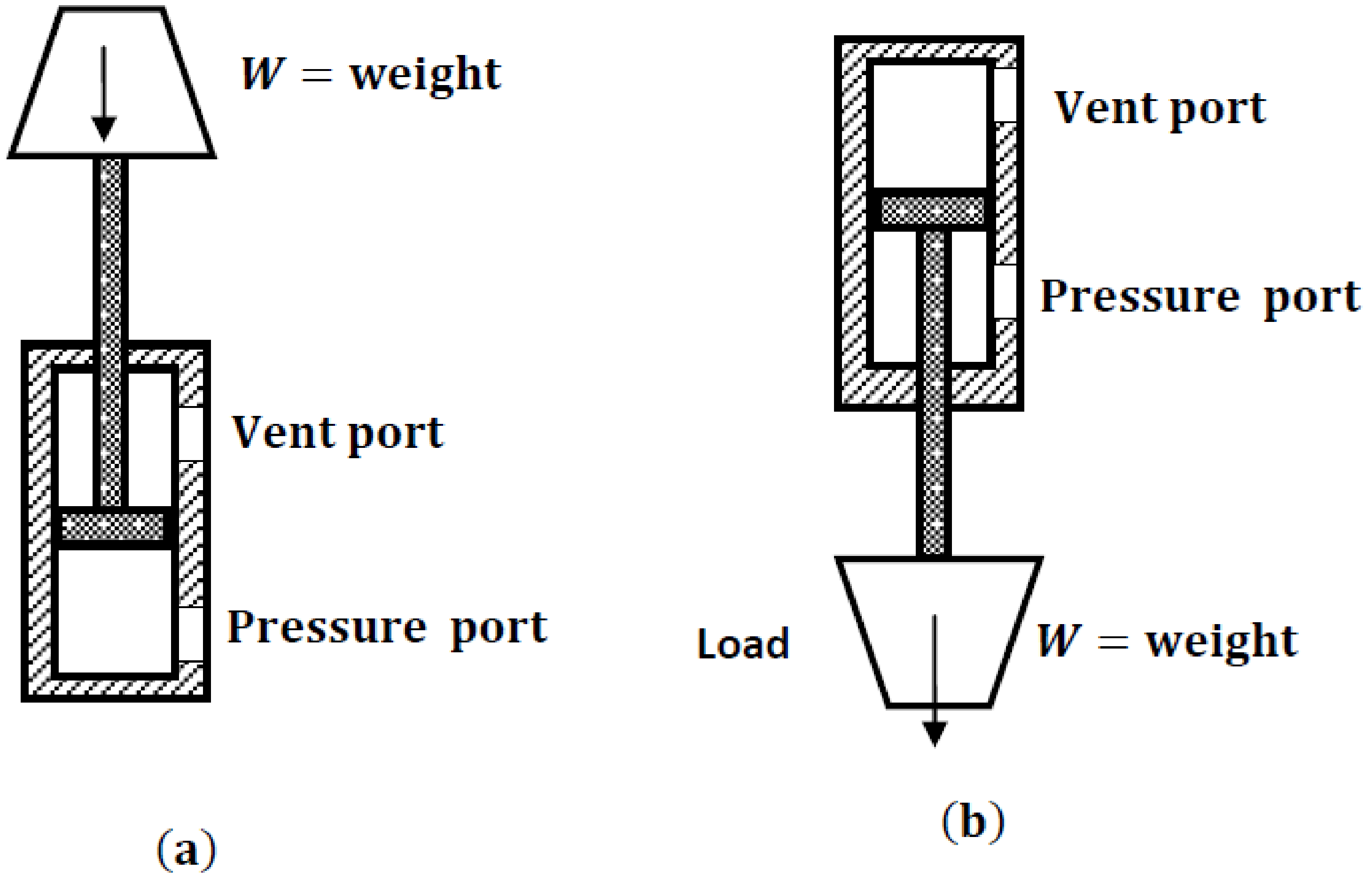
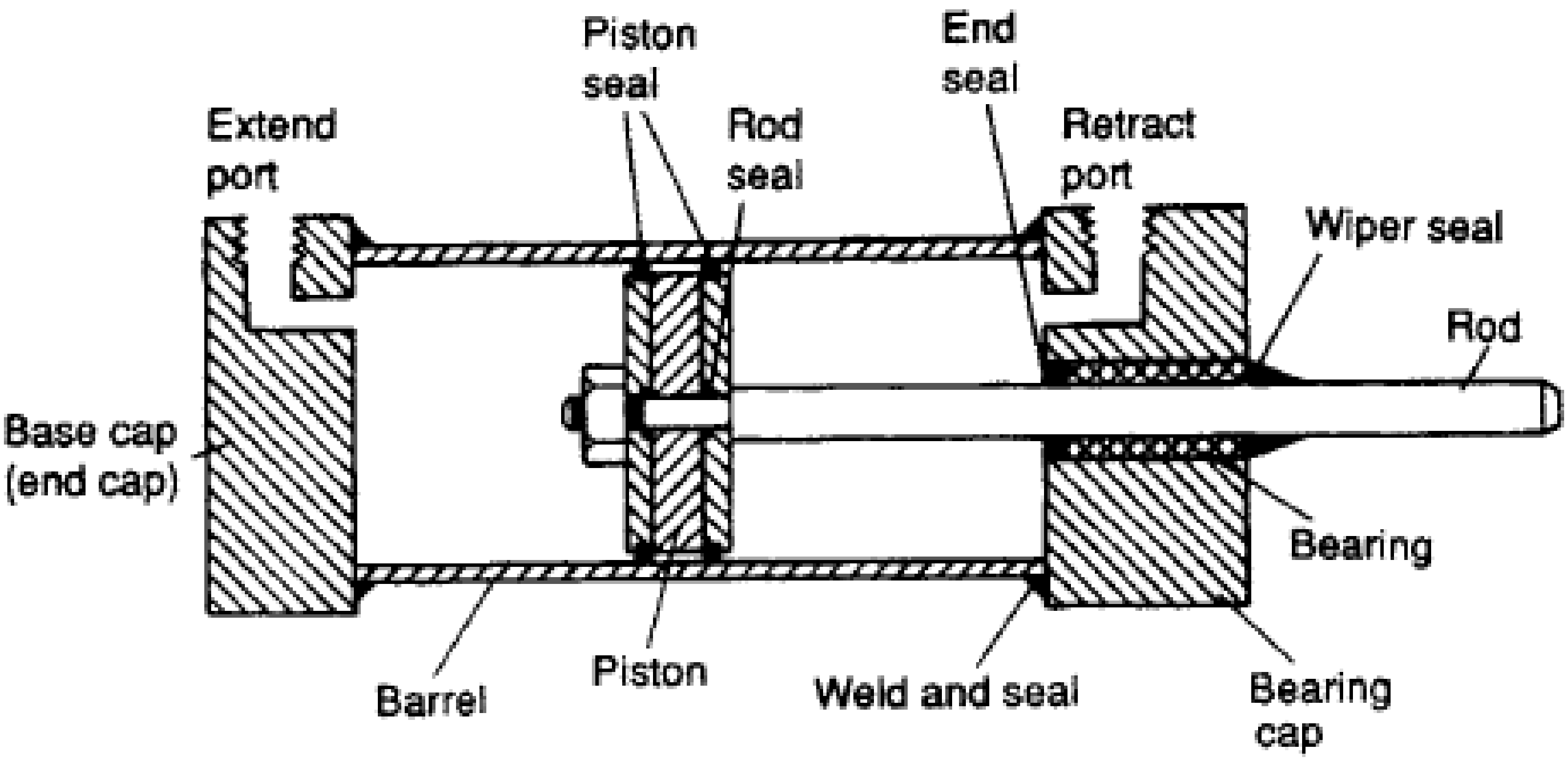
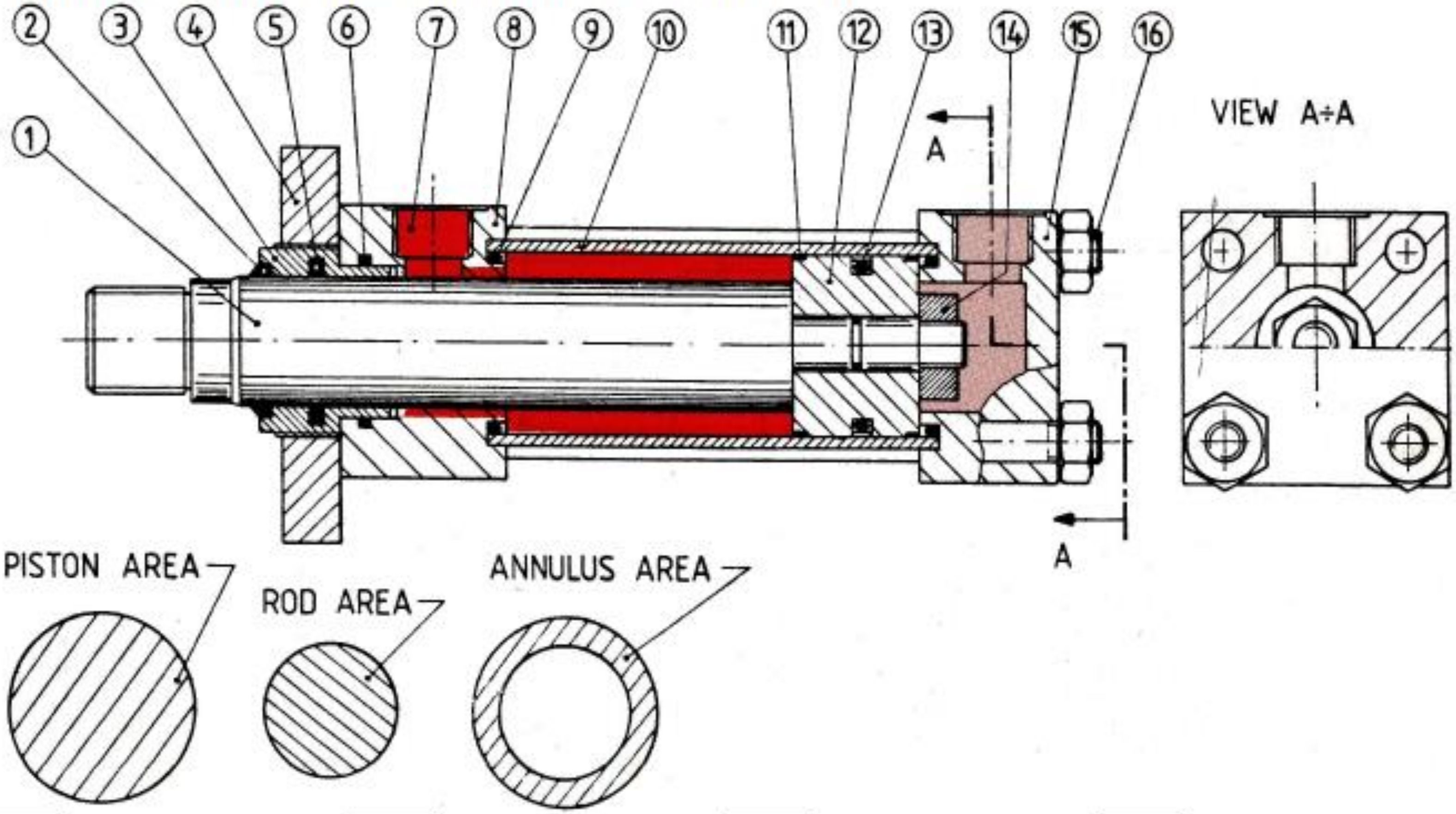


Figure 1.2 Gravity-return single-acting cylinder: (a) Push type; (b) pull type


Piston seal
End seal
Extend port
Rod seal
Retract port
Wiper seal
Rod
Base cap (end cap)
Bearing
Barrel
Piston
Weld and seal
Bearing cap

Actuator types

Single-acting actuators permit the application of hydraulic force in one direction only. These actuators are normally mounted in vertical direction, thus permitting the load to return the piston to its initial position Where the actuator must be mounted horizontally, an built in spring is used to cause retraction (fig. 85).



1	Piston rod	2	Rod wiper	3	Guide bush	4	Mounting flange
5	Rod seal	6	Bush seal	7	Oil port	8	Cylinder head
9	Tube seal	10	Tube (barrel)	11	Guide ring	12	Piston
13	Piston seal	14	Lock nut	15	Cylinder gap	16	Tie rod

Fig. 84 Typical tie-rod type actuator (cylinder).

Seals in linear actuators

Seals are grouped into static and dynamic applications. Static seals are fitted between two rigidly connected components. The seals between the piston rod and the piston in fig. 87A and C are static seals, and so are the seals that prevent external leakage past the joint where the tube is mounted onto the cap in fig. 8713, C, and D.

Dynamic seals prevent leakage between components which move relative to each other. Therefore, dynamic seals are subject to wear, whereas static seals are essentially wear free. The seals between the moving piston and the stationary actuator tube (fig. 87A and C) are typically dynamic seals, and so are the seals that stop hydraulic fluid from escaping past the clearance gap between the moving piston rod and the end cap (head)

(fig. 87B and C). The rod-wiper ring may also be regarded as a dynamic seal. It prevents contaminants from being drawn into the actuator when the rod retracts (fig. 87B and D).

0-ring type seals are generally used as static seals whereas dynamic seals may range from simple Chevron type compression packings to complex moulded cup or lip seals. Chevron seals (fig. 87A and B) are less suitable for smooth piston movement at low pressures and low speed, but are an excellent seal for high system pressure and high stroking speeds. Lip-ring seals (fig. 87C and D) provide good sealing even under stationary piston conditions. Lip-ring and glide-ring seals are used for applications with exceptionally low friction requirements. For glide-ring seal applications, the piston is fitted with guide rings (Fig 86)

Actuator Construction

The actuator tube is usually of cold-drawn seamless steel with a microfinish honed bore of a surface quality of R, < 1.3,um. Piston rods may be made from highgrade heat-treated steel, surface hardened and hard chrome-plated for some applications. Surface quality of the rod is about 0.2,um. Induction hardening of the surface provides greater protection against mechanical damage and results in a longer life of the dynamic seals. Where the actuator is to be used in an aggressive environment, high tensile stainless may be specified for the piston rod, and sometimes also hard chrome-plating to provide a wear-resistant surface. Tube and heads are either screwed or welded to each other or tie-rods or wire retainers may be used (fig. 87).

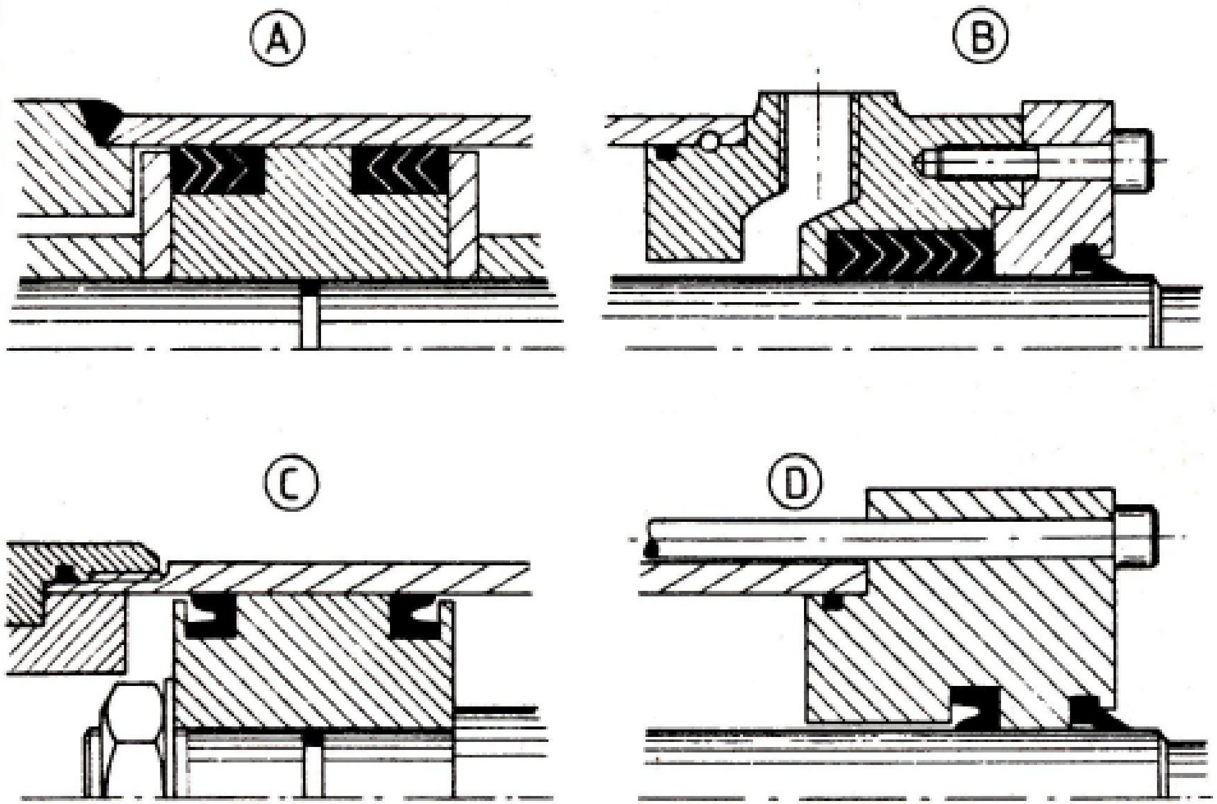
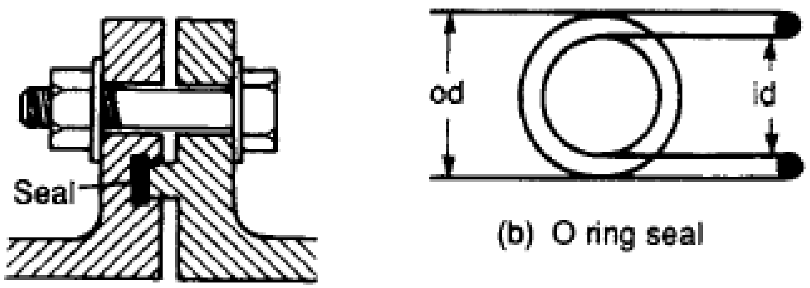


Fig 87 Various types of seals and tube to head connecting methods

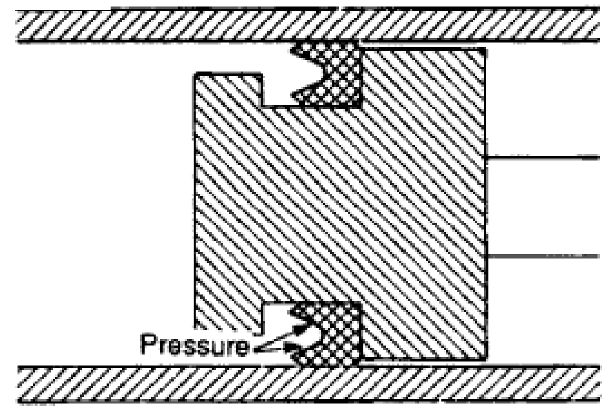
The earliest synthetic seal material was neoprene, but this has a limited temperature range (below 65°C). The most common present-day material is nitrile (buna-N) which has a wider temperature range (–50°C to 100°C) and is currently the cheapest seal material. Silicon has the highest temperature range (–100°C to +250°C) but is expensive and tends to tear.



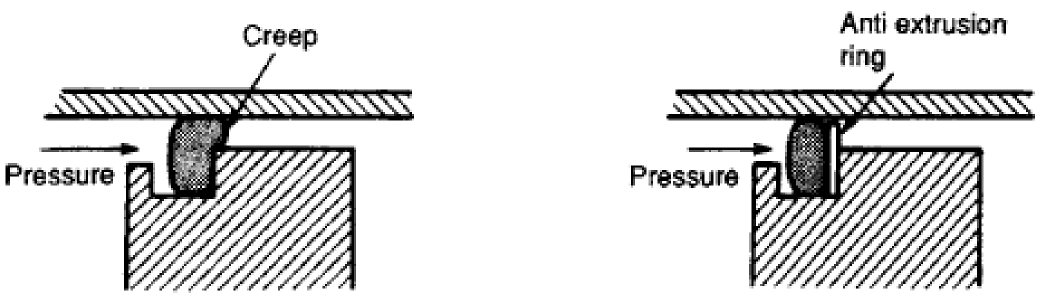
(b) O ring seal

(a) Simple static seal

7 *Static seals*



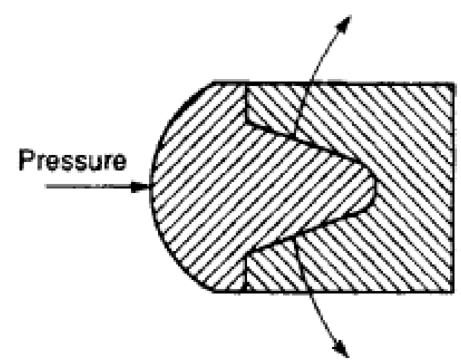
The U ring seal



(a) Unsecured seal creeps into gap

(b) Backup ring prevents creep

Figure 5.20 *Anti-extrusion ring*



The composite seal

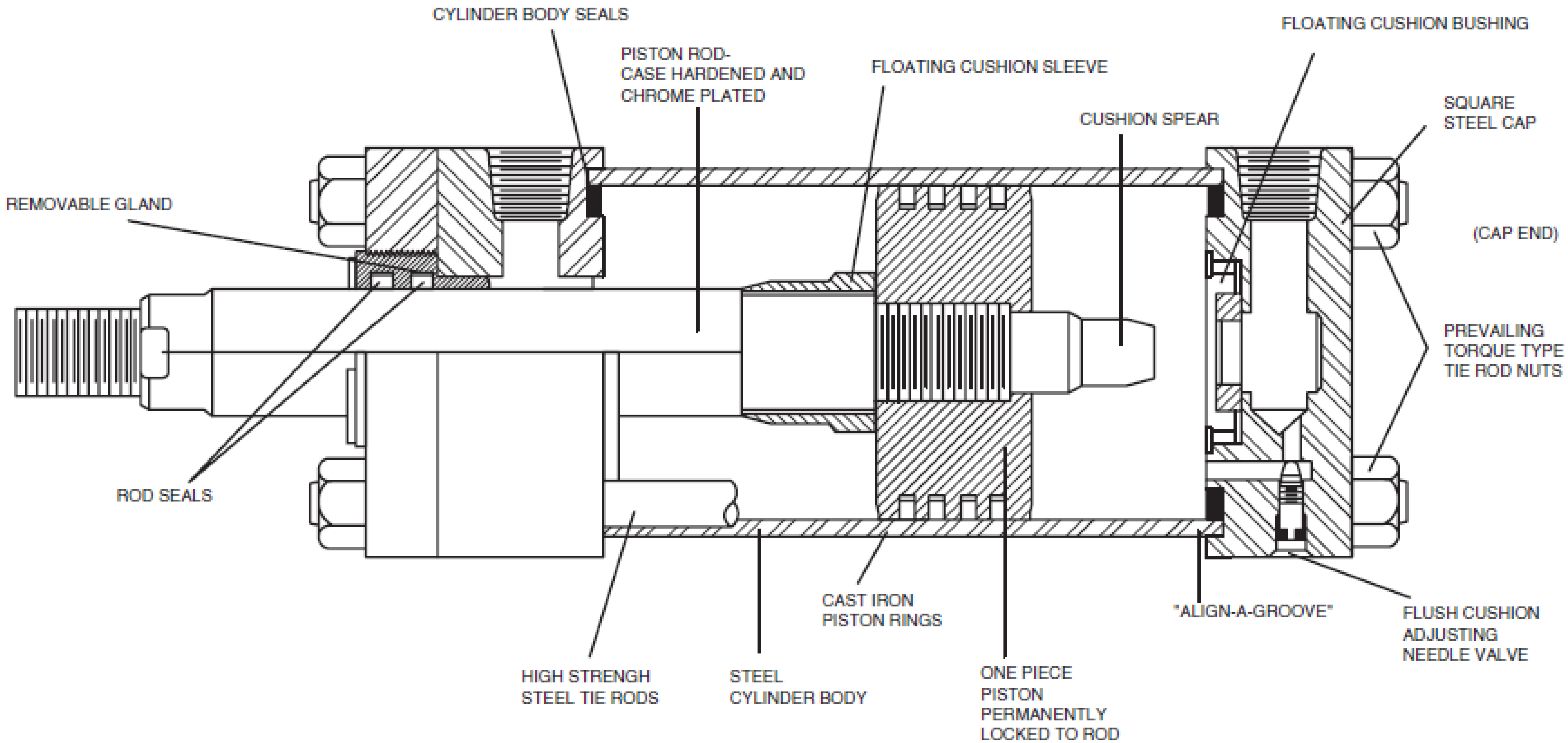


FIGURE 7.18
Typical cylinder construction for cylinders used for industrial applications. (Reprinted with permission from Parker Hannifin Corp.)

1.4Cylinder Force, Velocity and Power

The output force (F) and piston velocity (v) of double-acting cylinders are not the same for extension and retraction strokes.

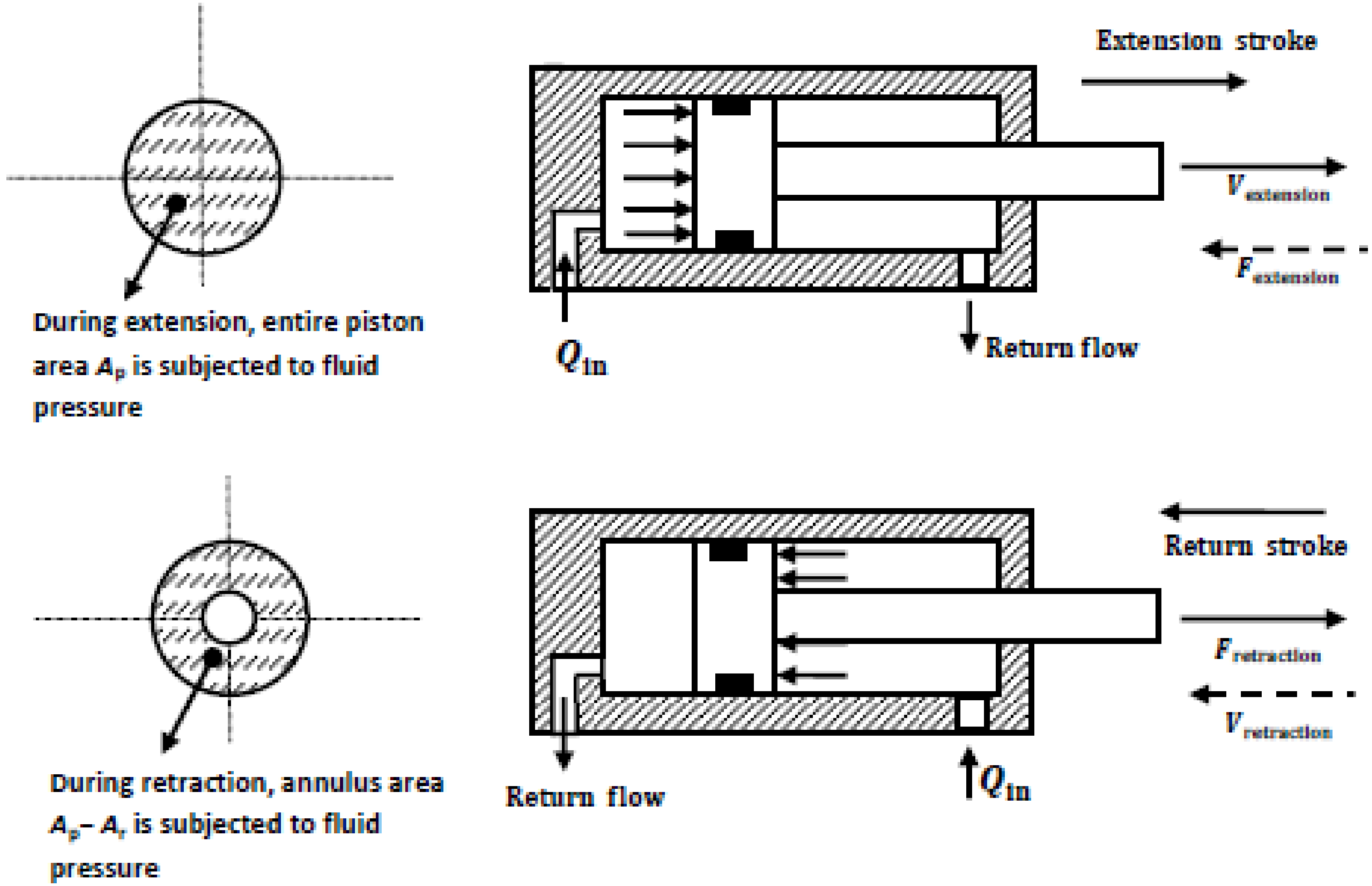


Figure 1.11 Effective area during (a) extension strokes and (b)retraction strokes

During the extension stroke, the fluid pressure acts on the entire piston area (A_r), while during the retraction stroke, the fluid pressure acts on the annular area ($A_p - A_r$). This difference in area accounts for the difference in output forces during extension and retraction strokes. Because A_r is greater than $A_p - A_r$), the extension force is greater than the retraction force for the same operating pressure.

Force and velocity during extension stroke

Velocity

$$v_{ext} = \frac{Q_{in}}{A_p}$$

Force

$$F_{ext} = p \times A_p$$

Force and velocity during retraction stroke

Velocity

$$v_{ext} = \frac{Q_{in}}{A_p - A_r}$$

Force,

$$F_{ext} = p \times (A_p - A_r)$$

Power developed by a hydraulic cylinder (both in extension and retraction) is

$$\text{Power} = \text{Force} \times \text{Velocity} = F \times V$$

In metric units, the kW power developed for either extension or retraction stroke is

$$\text{Power (kW)} = v_p\,(\text{m/s}) \times F\,(\text{kN})$$
$$= Q_{in}\,(\text{m}^3/\text{s}) \times p\,(\text{kPa})$$

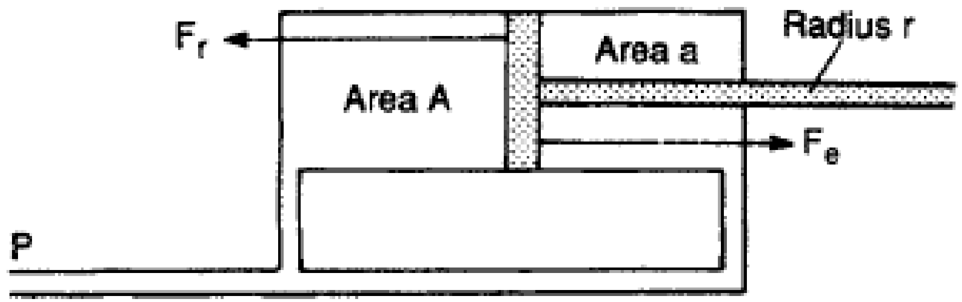
Power during extension is

$$P_{ext} = F_{ext} \times v_{ext} = p \times A_p \times \frac{Q_{in}}{A_p} = p \times Q_{in} \tag{1.1}$$

Power during retraction is

$$P_{ret} = F_{ret} \times v_{ret}$$
$$= p \times (A_p - A_r) \times \frac{Q_{in}}{A_p - A_r}$$
$$= p \times Q_{in} \tag{1.2}$$

Comparing Equation. (1.1) and (1.2), we can conclude that the powers during extension and retraction strokes are the same.



Pressure applied to both sides of piston

1.2.2.2 **Double-Acting Cylinder with a Piston Rod on Both Sides**

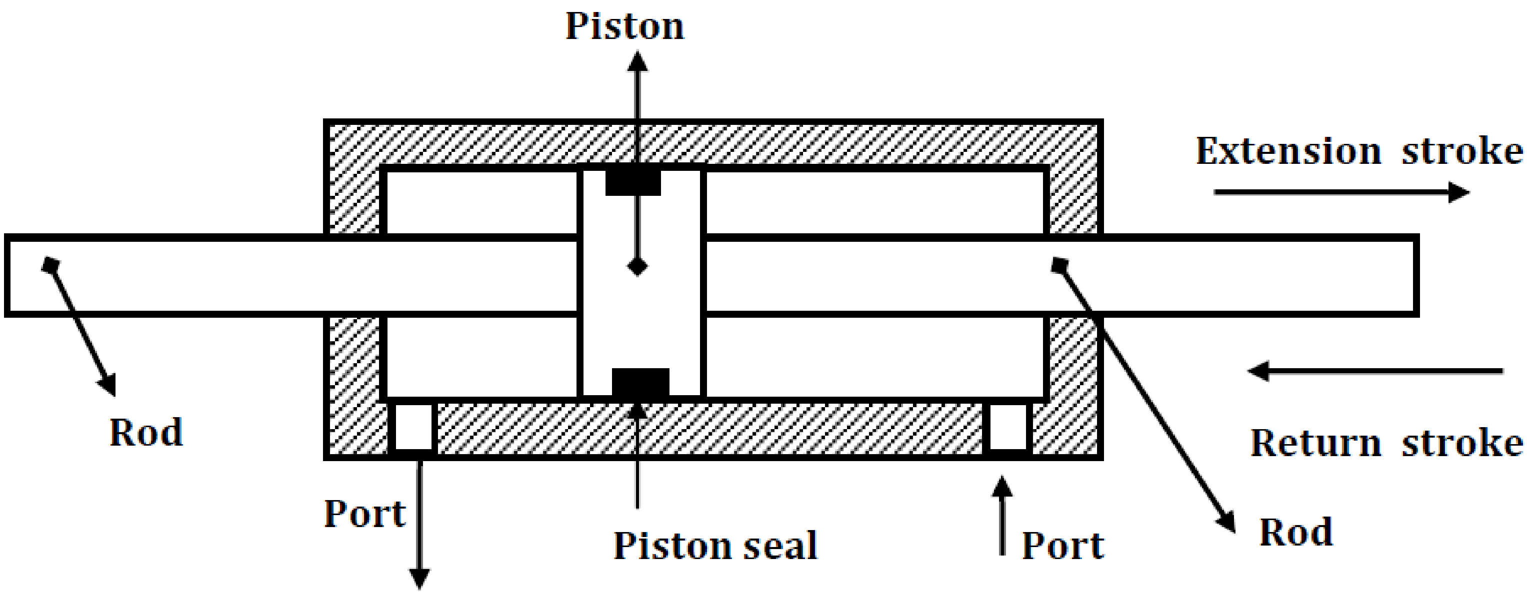
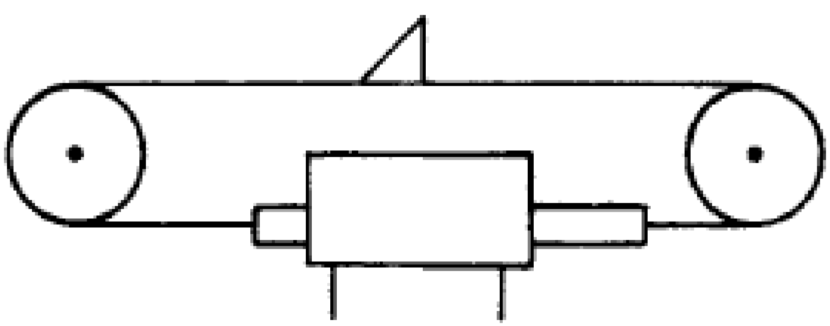


Figure 1.5 Double-acting cylinder with a piston rod on one side



(b) Typical application

Double rod cylinder (with equal extend/retract force)

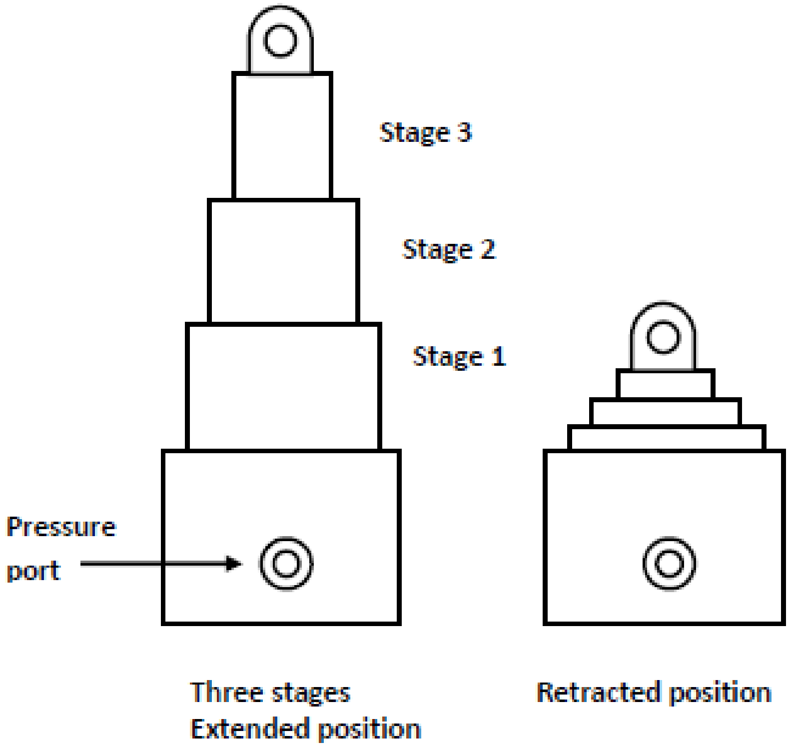


Figure 1.6 Telescopic cylinder

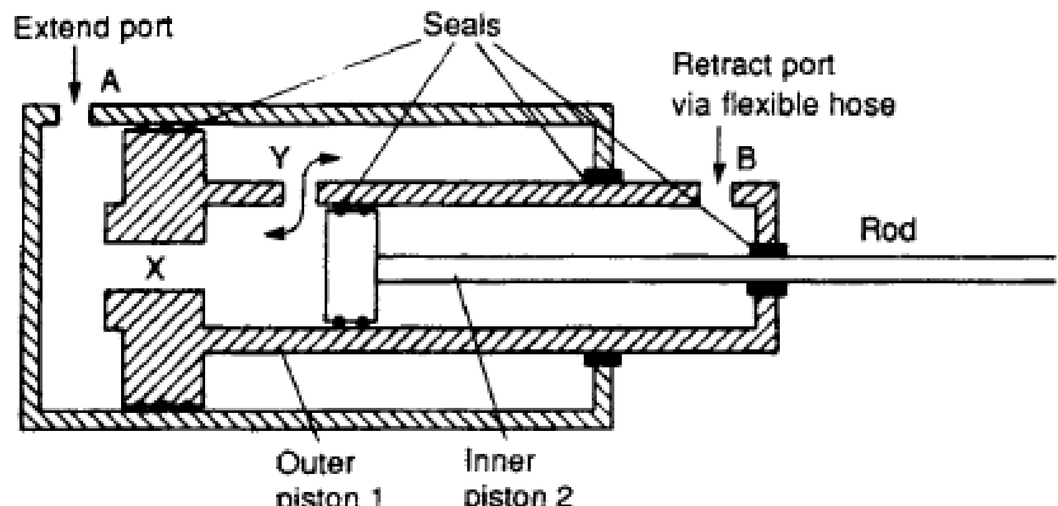


Figure 5.12 *Two-stage telescopic piston*

1.2.4 Tandem Cylinder

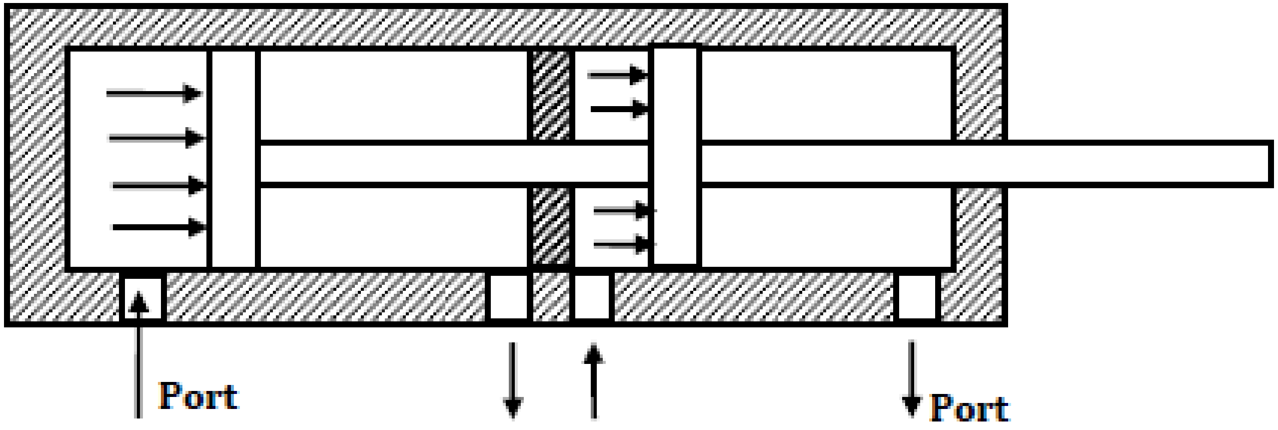


Figure 1.7Tandem cylinder

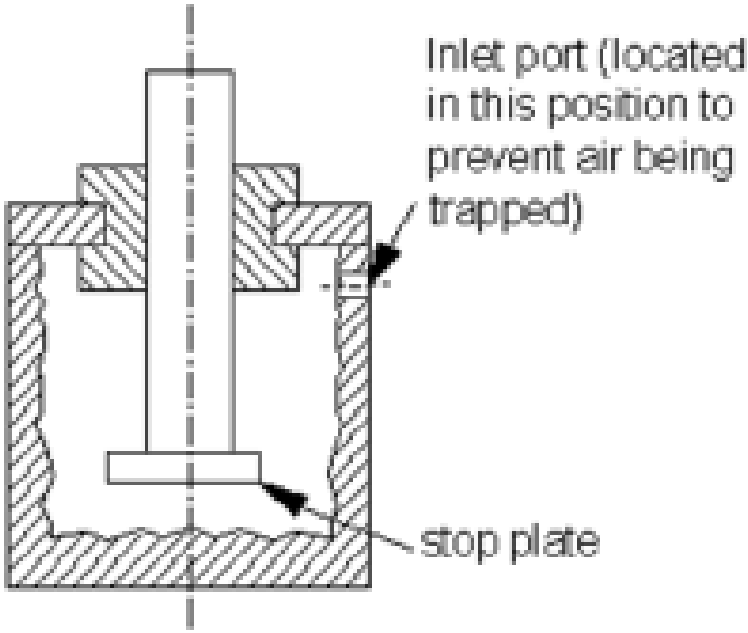


Figure 1.8Displacement cylinders

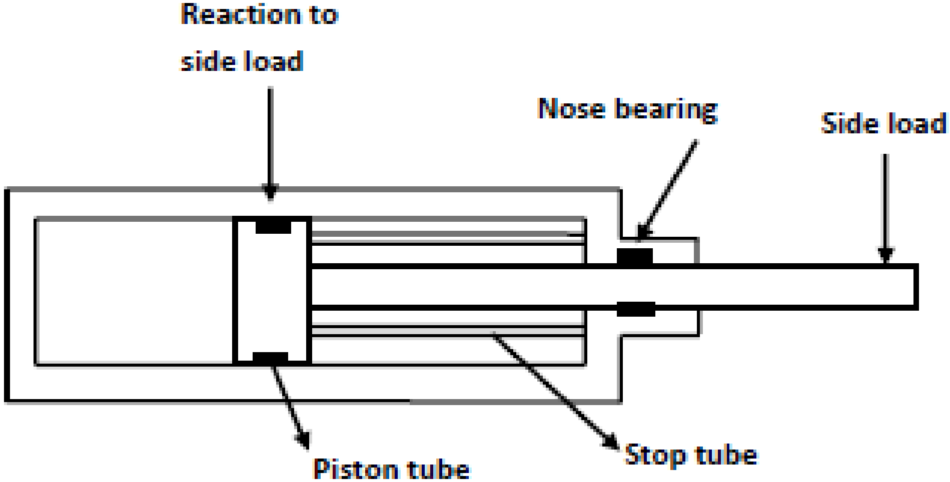
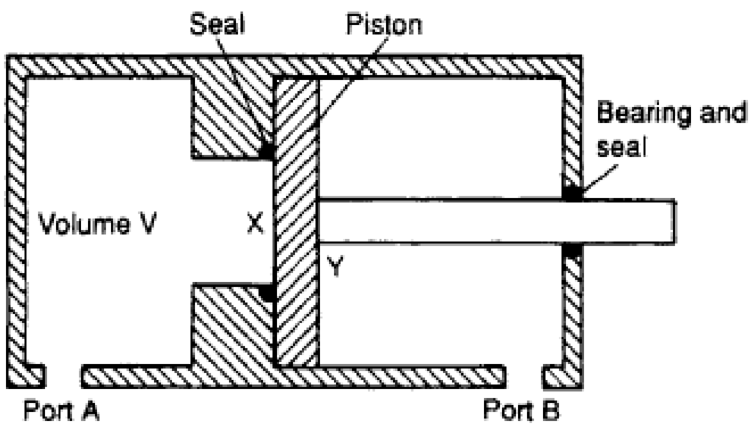


Figure 1. 33Use of a stop tube to minimize side loading



An impact cylinder

End-position cushioning

Cushioning, or end-position cushioning, refers to braking and deceleration of the final stroke portion until standstill occurs. Cushioning becomes essential above a certain stroke speed. The kinetic energy released on impact at the stroke end must be absorbed by the stroke limit-stops, which are built into the end caps. Their capacity to absorb this energy depends on the elasticity of their material.

A hydraulic braking function (end-position cushioning) must therefore be applied where piston speeds (v) exceed 0.1 m/sec. Figure 82 shows a cross-section of the end-position cushioning mechanism of the end cap. The cushioning of the rod end cap is similar

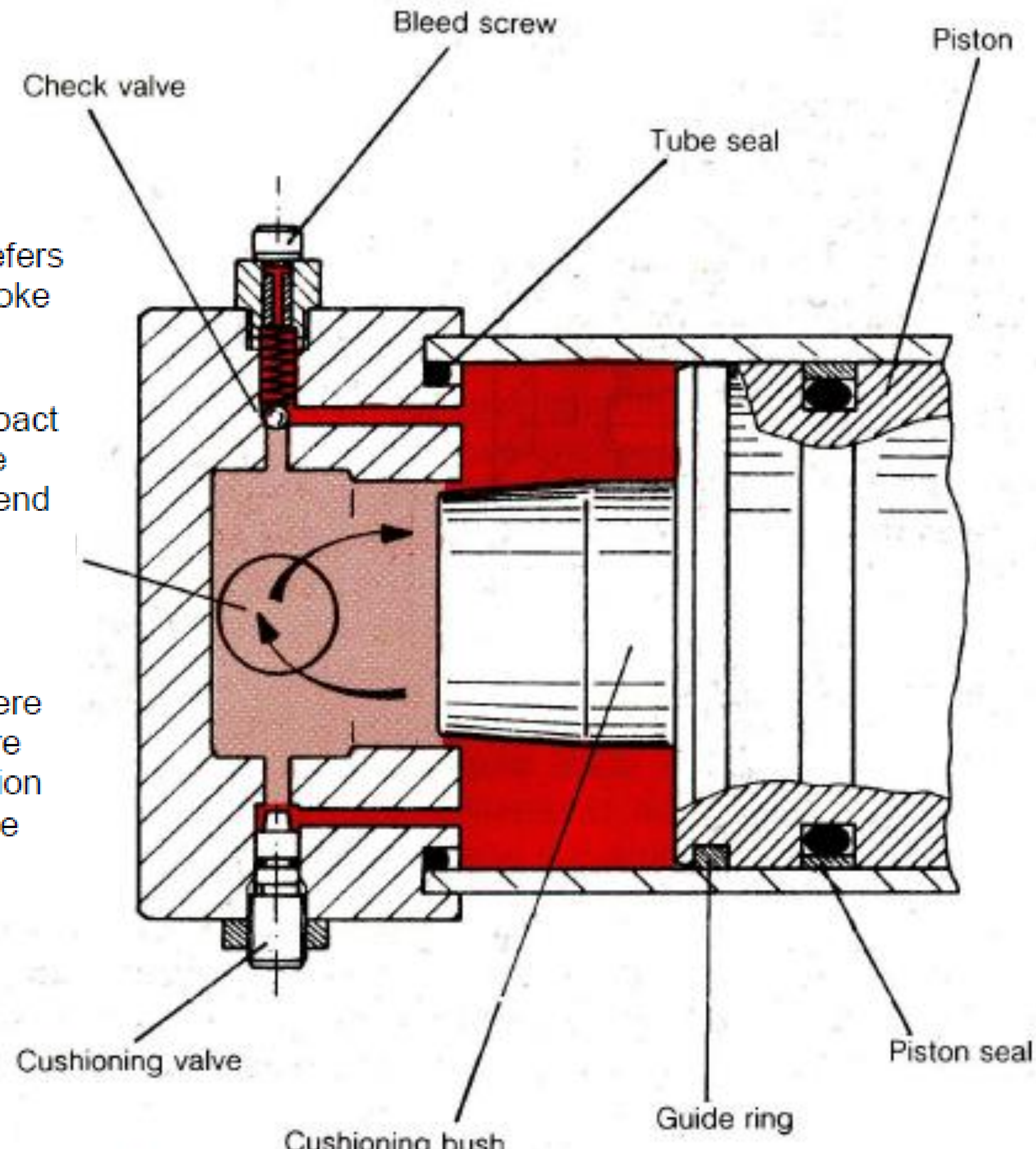


Fig 86 Typical end position cushioning position.

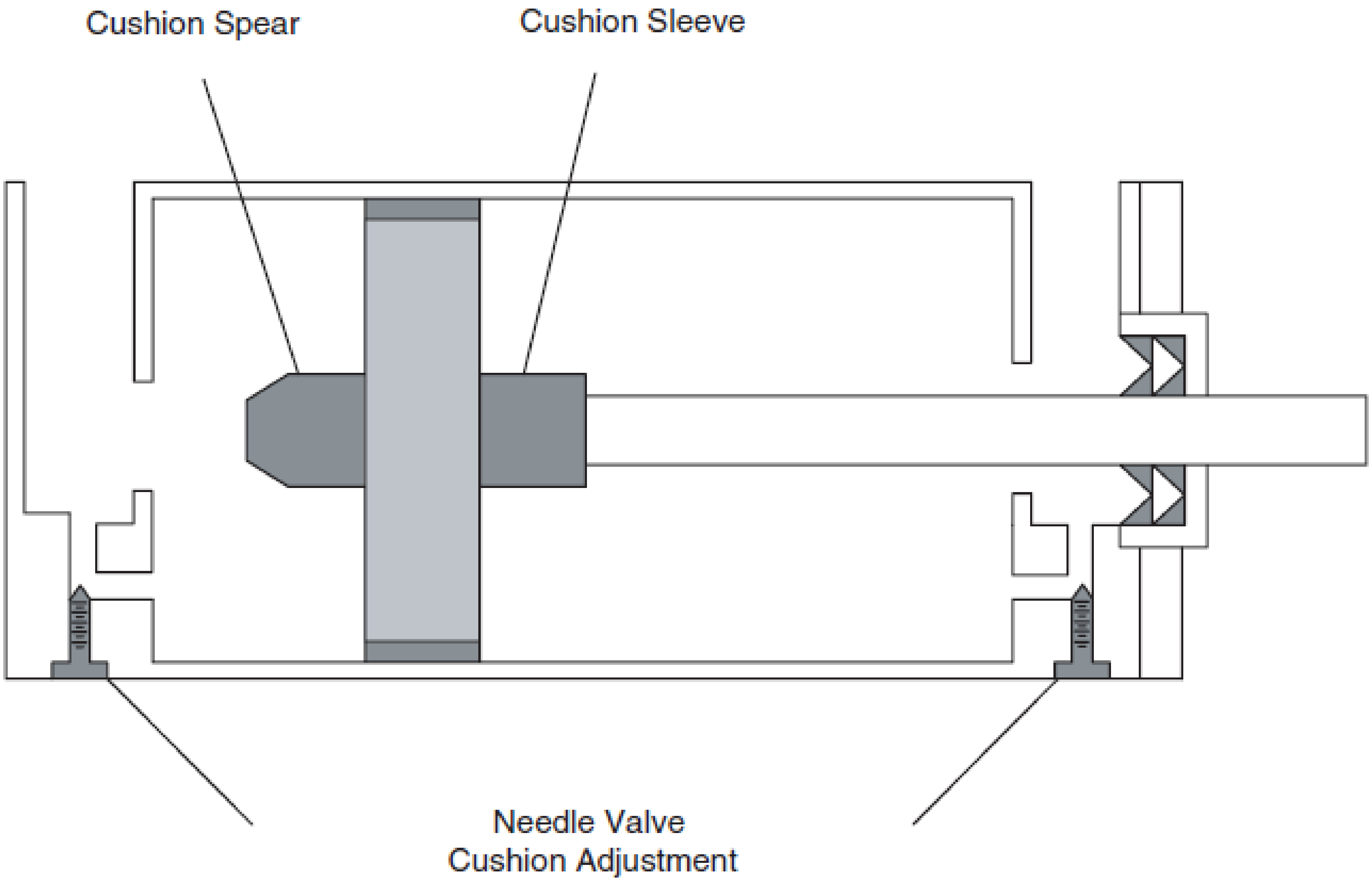


FIGURE 7.6
Schematic showing a technique for cushioning a hydraulic cylinder.

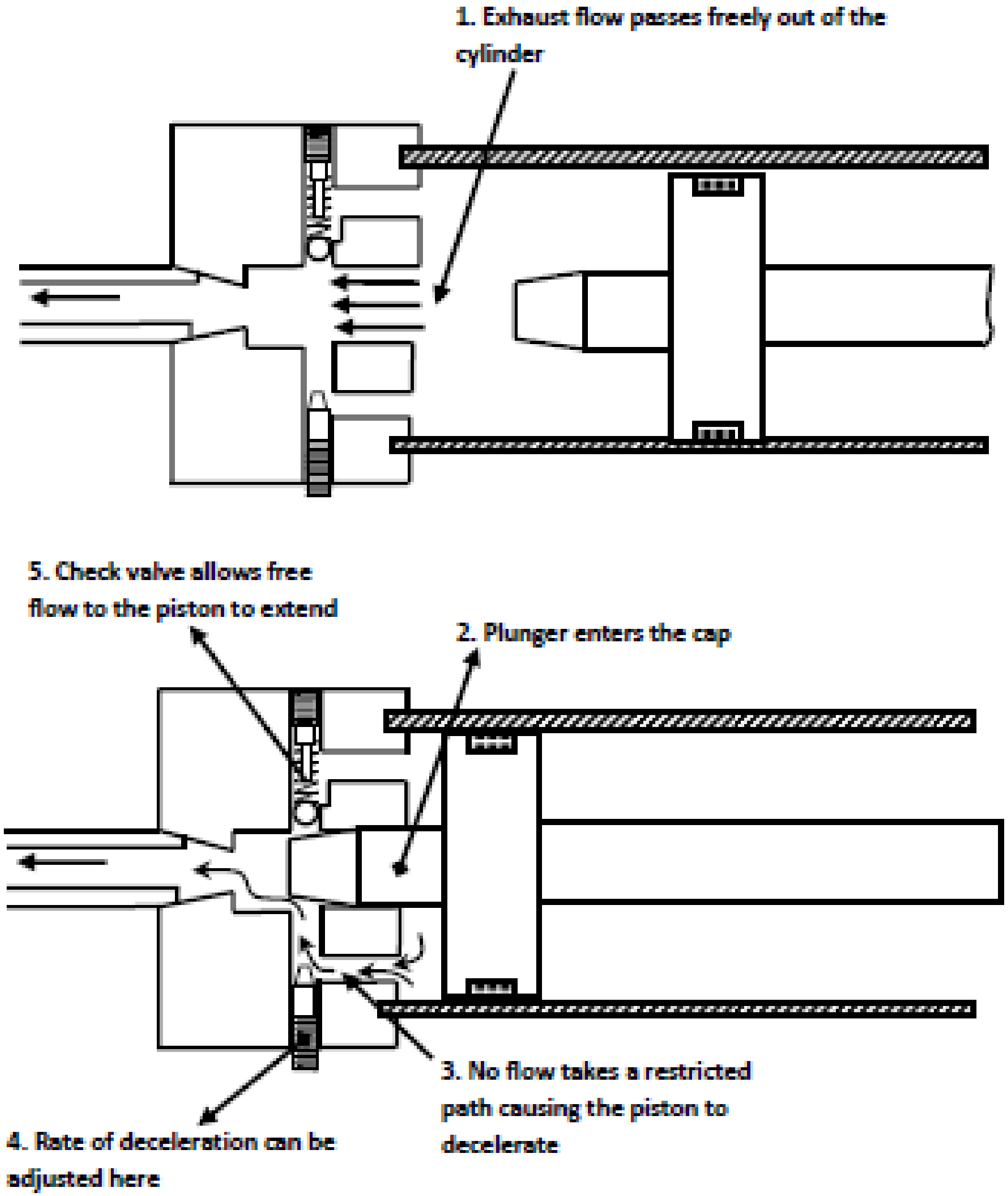


Figure 1. 28 Operation of cylinder cushions

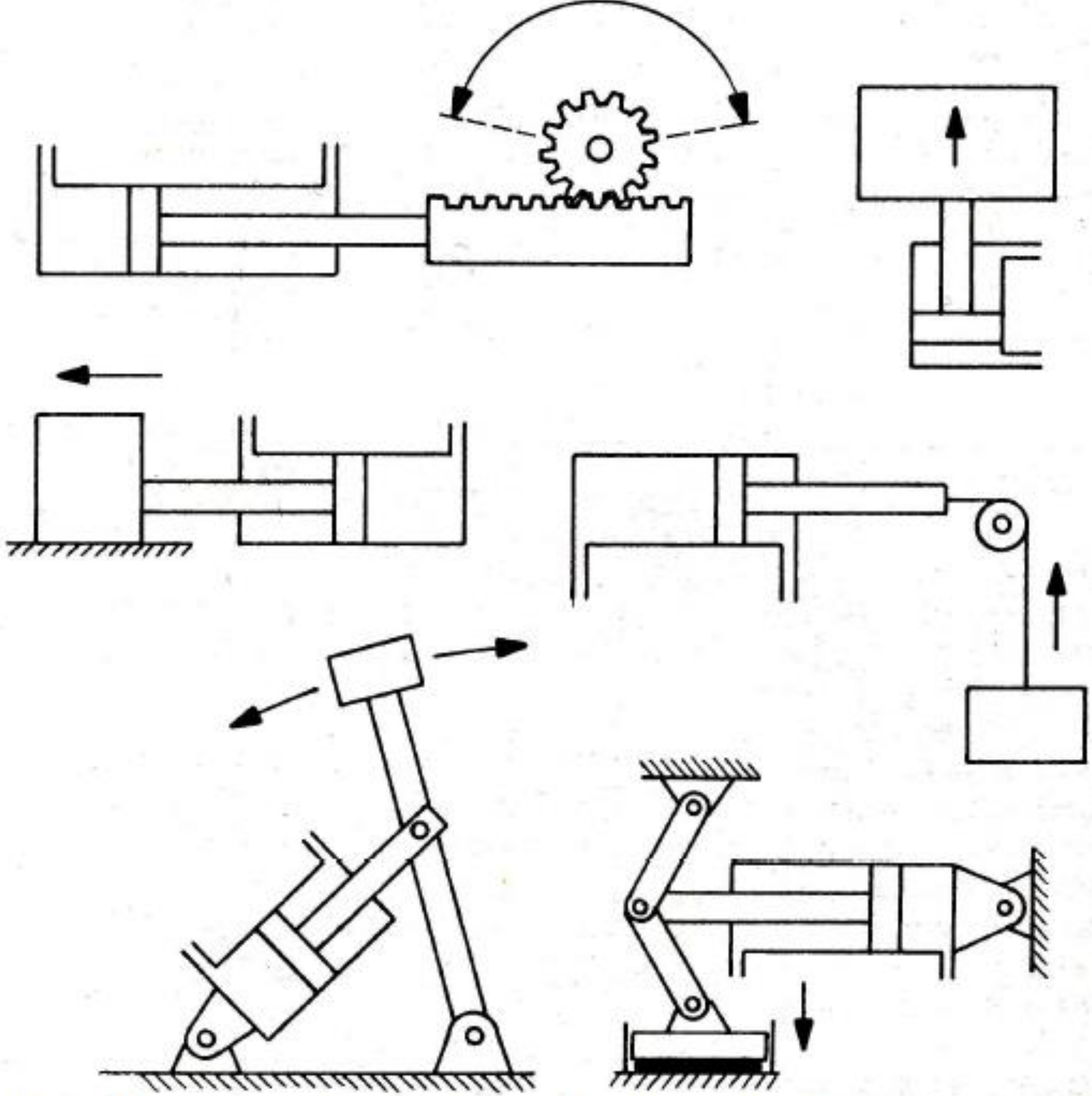


Fig 83 Linkages and loads attached to linear actuators

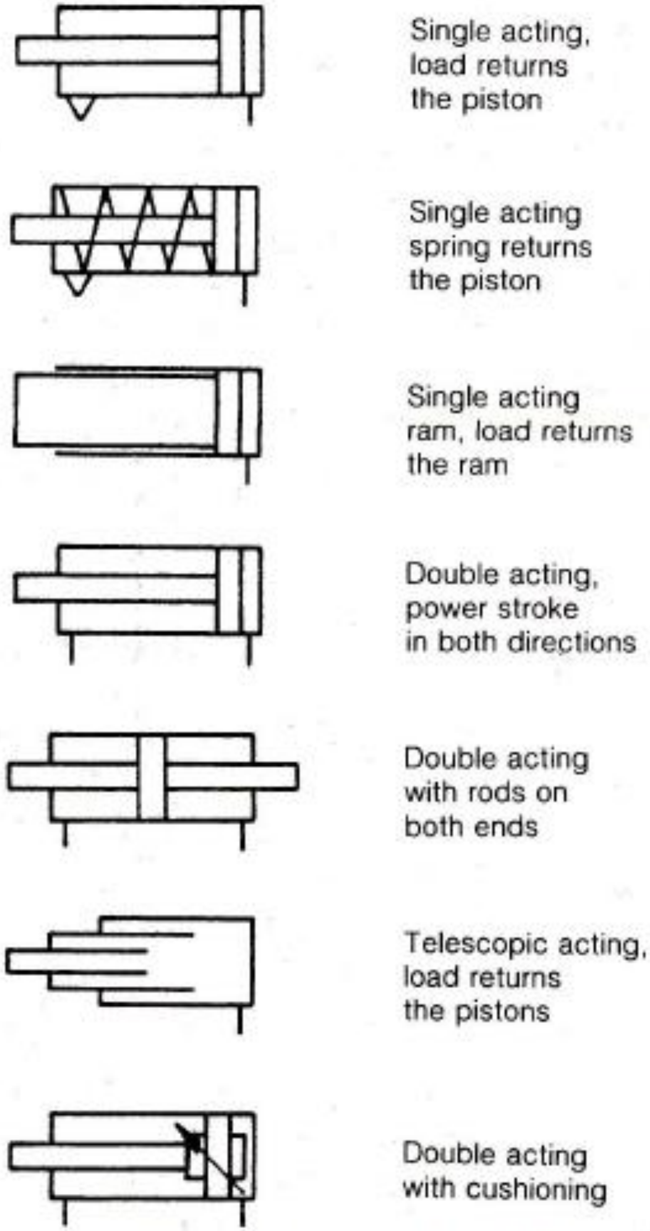


Fig. 85 Graphic symbols for the more commonly used linear actuators.

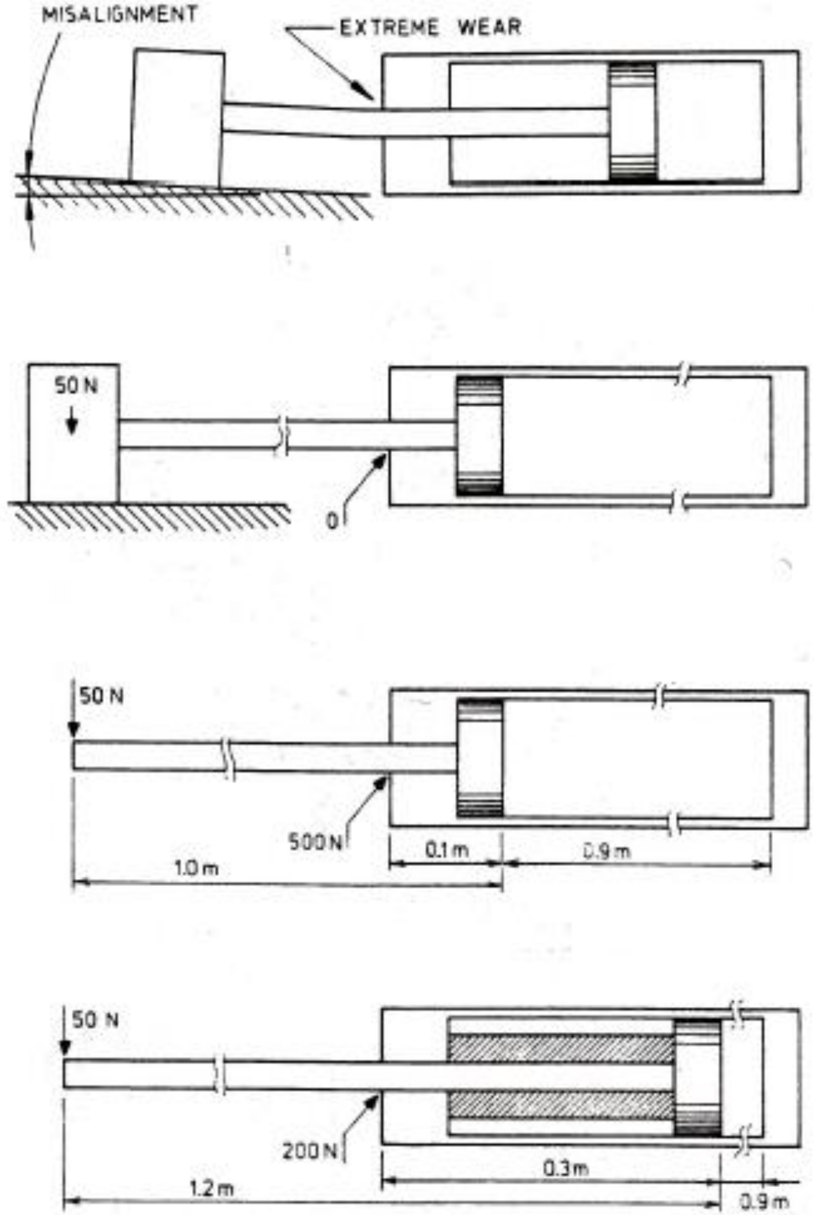
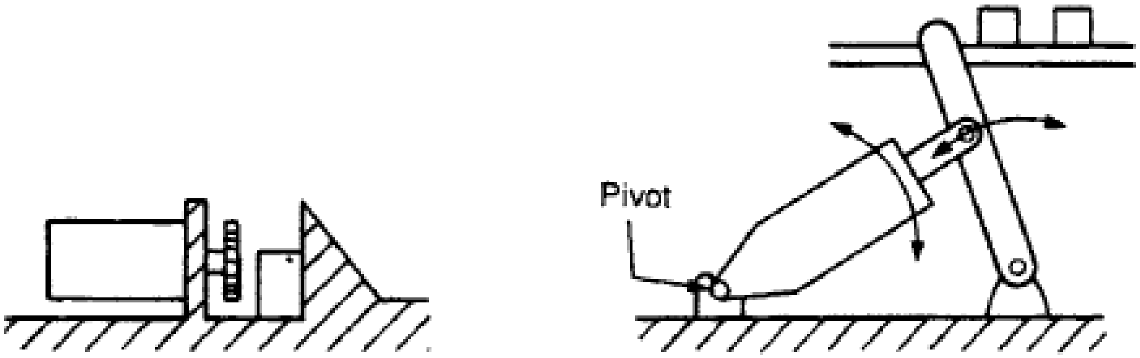

MISALIGNMENT
EXTREME WEAR
50 N
0
50 N
500 N
0.1 m
0.9 m
1.0 m
50 N
200 N
0.3 m
1.2 m
0.9 m

Fig 88



(a) Clamp using front flange mounting

(b) Pusher using rear trunnion mounting

Figure 5.14 *Basic mounting types*

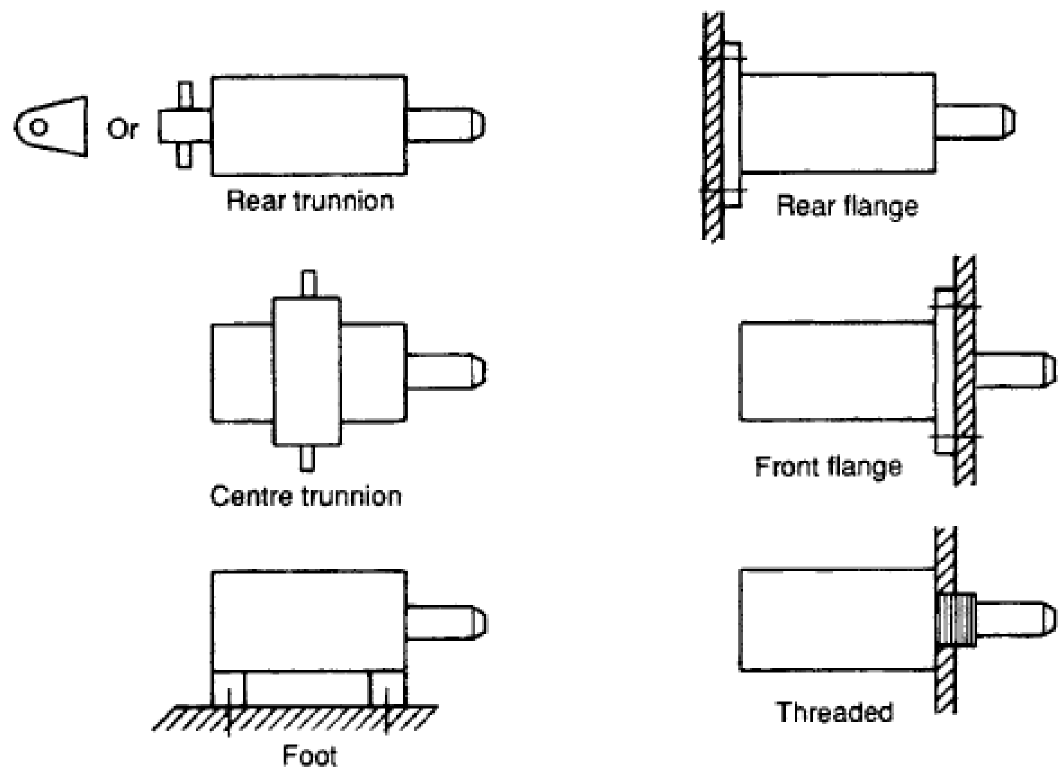


Figure 5.15 *Methods of cylinder mounting*

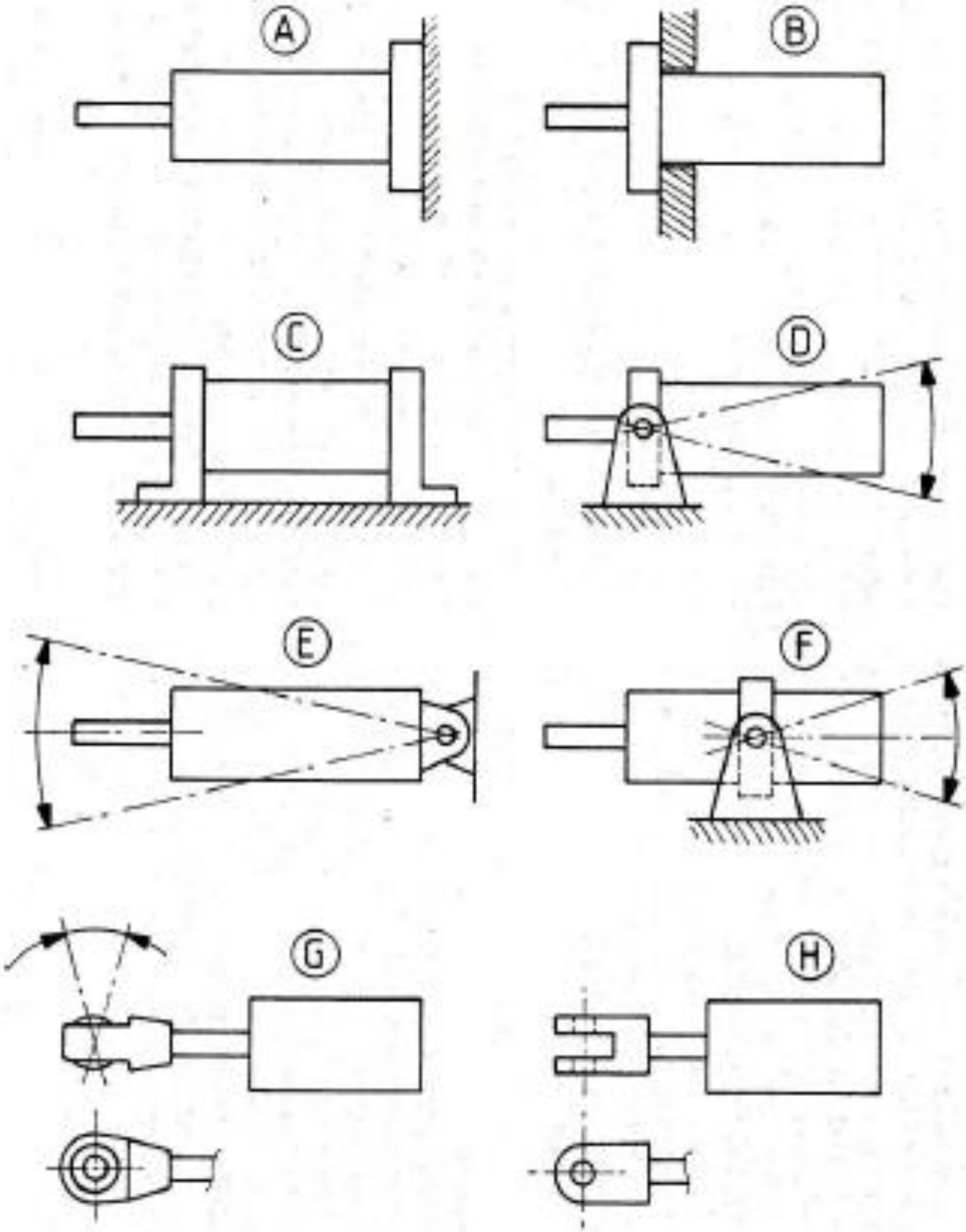


Fig 89

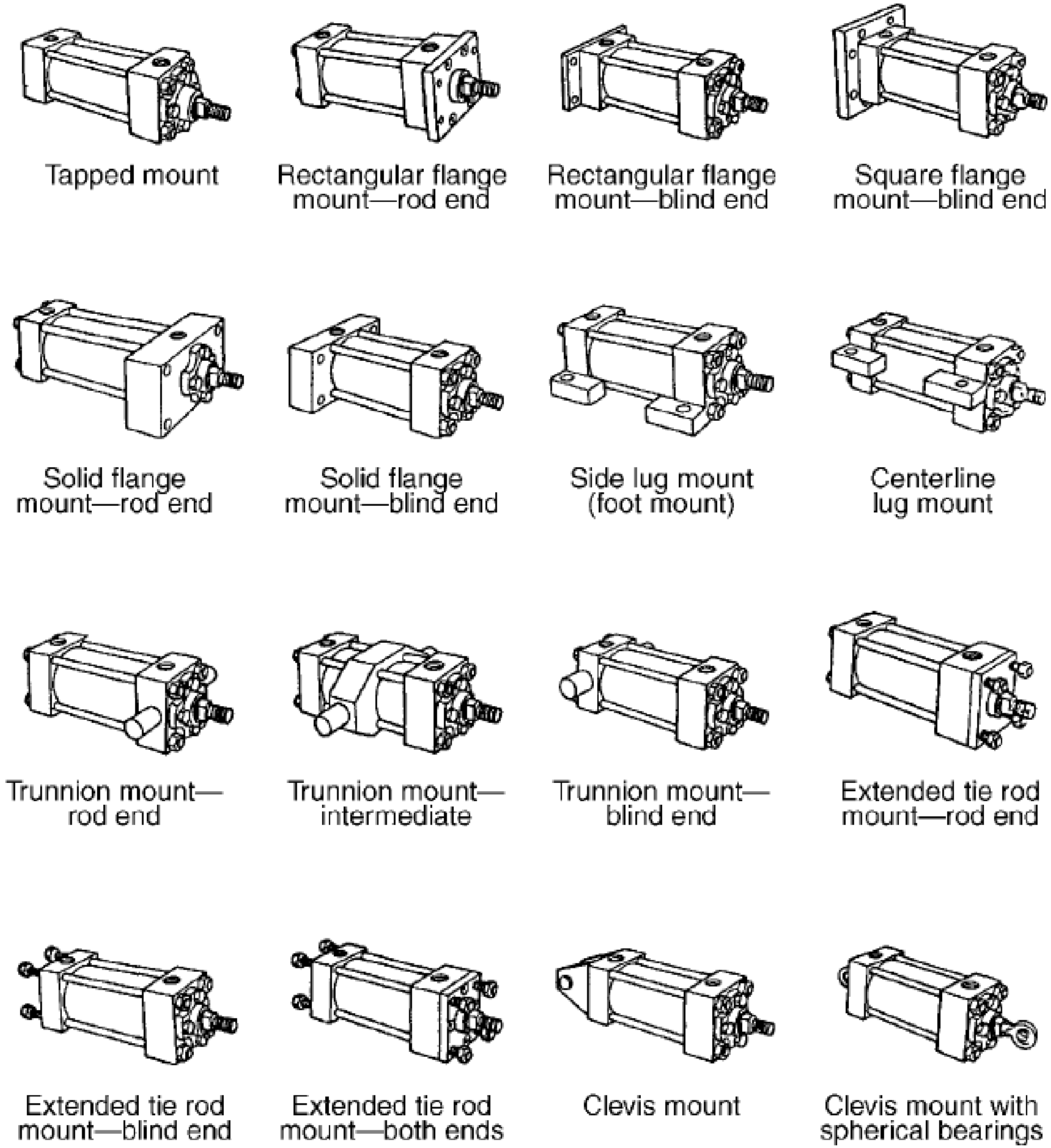
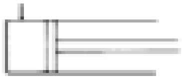
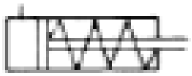
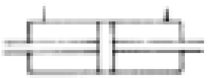
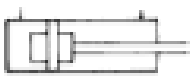
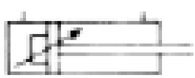
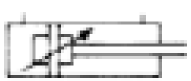
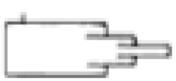
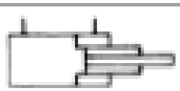


FIGURE 7.15
Partial listing of cylinder mounting methods.

Graphic symbol	Item	Description
	Single-acting cylinder	Return stroke by external force
		Return stroke through a spring
	Double-acting cylinder	Single rod
		Double rod
	Cylinder with fixed stroke end cushioning	Cushioning on one side
		Cushioning on both sides
	Cylinder with adjustable stroke end cushioning	Cushioning on one side
		Cushioning on both sides
	Telescopic cylinder	Single-acting
		Double-acting

Actuator sizing

The main criteria on which the size of the actuator is based are:

- force output for extension and retraction;
- piston speed for extension and retraction;
- mechanical stability of the actuator.

Piston rod buckling

Column failure, or buckling of the piston rod, will occur if the actuator stroke in relation to the piston rod diameter is, at a required force output, out of (safe) proportion. Piston rod buckling is calculated according to the "Euler formula", where the piston rod is regarded as the buckling member (fig. 93).

$$\text{Euler formula: } K = \frac{\pi^2 \text{ x } E \text{ x } I}{S_K^{\,2}}$$

Note: Under this condition the piston rod buckles! The maximum safe operating load (in Newtons) is:

$$F = \frac{K}{S}$$

K = critical load (N)
S_K = free buckling length (m) (fig. 93)
S = safety factor (usually 2.5-3.5)
E = elasticity modulus (Pa) - for steel: 2.1 x 10^{10} x 9.80665
F = force
D = rod diameter (m)

I = moment of inertia (m^4) - $\dfrac{\pi \text{ x } d^4}{64}$

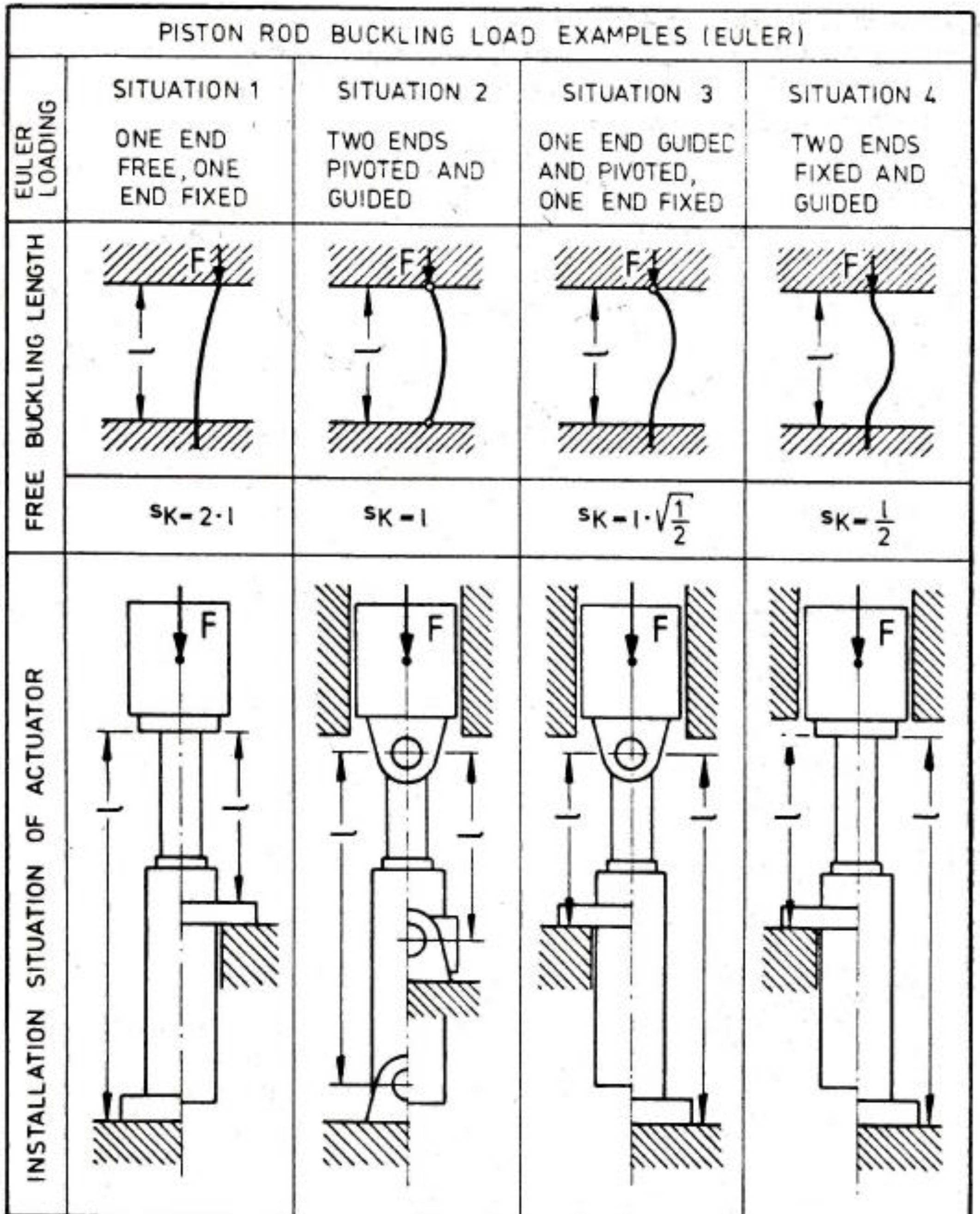
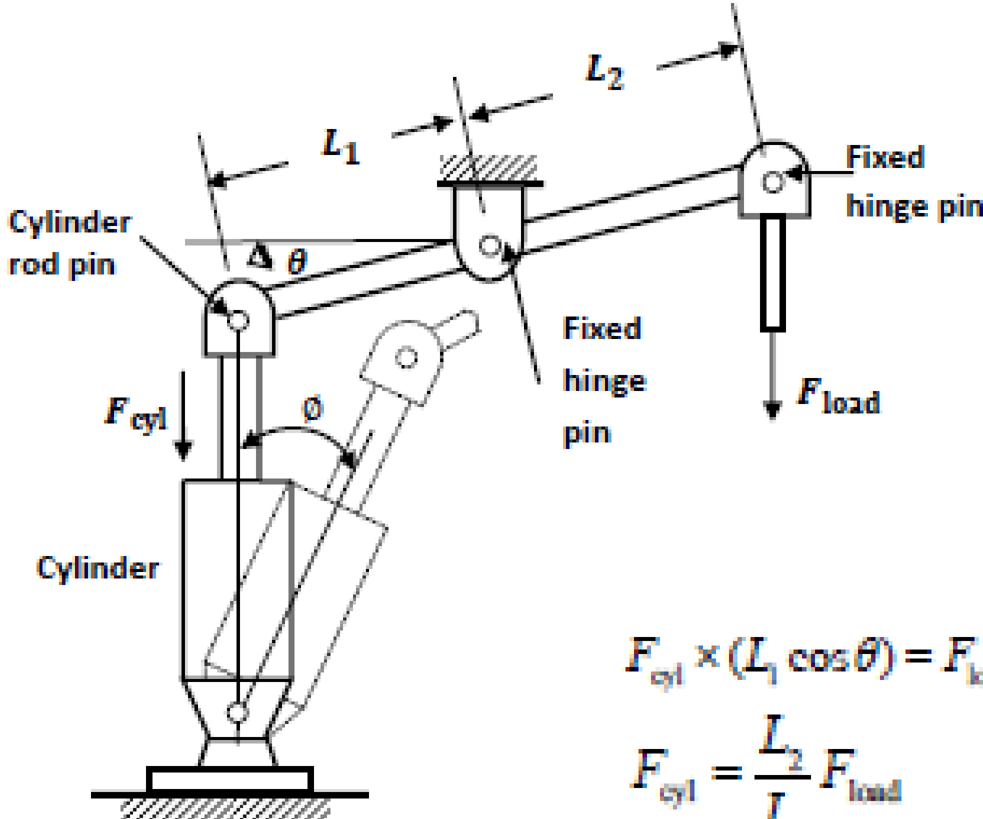


Fig.93 Piston rod buckling load examples (Euler).

1. 7.1 First-Class Lever System



$$F_{cyl} \times (L_1 \cos\theta) = F_{load} \times L_2 \cos\theta$$

$$F_{cyl} = \frac{L_2}{L_1} F_{load} \qquad (1.\ 3)$$

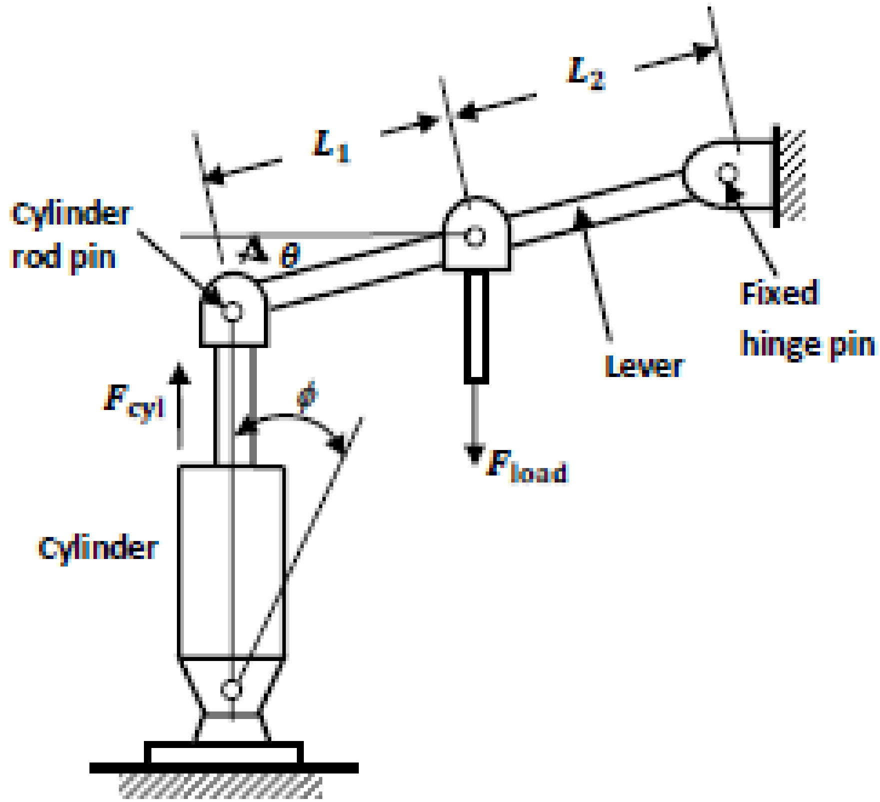
Suppose the centerline of the hydraulic cylinder tilts by an offset angle ϕ from the vertical; the relationship becomes

$$F_{cyl} \cos\phi \times (L_1)(\cos\theta) = F_{load} \times L_2 \cos\theta$$

$$F_{cyl} = \frac{L_2}{(L_1)\cos\phi} F_{load} \qquad (1.\ 4)$$

1. 7.2 Second-Class Lever System

In this lever system, the loading point is in between the cylinder and the hinge point as shown in Fig.1. 22.



$$F_{cyl}\cos\phi \times (L_1 + L_2)(\cos\theta) = F_{load} \times L_2 \cos\theta$$

$$F_{cyl} = \frac{L_2}{(L_1 + L_2)\cos\phi} F_{load} \quad (1.\ 5)$$

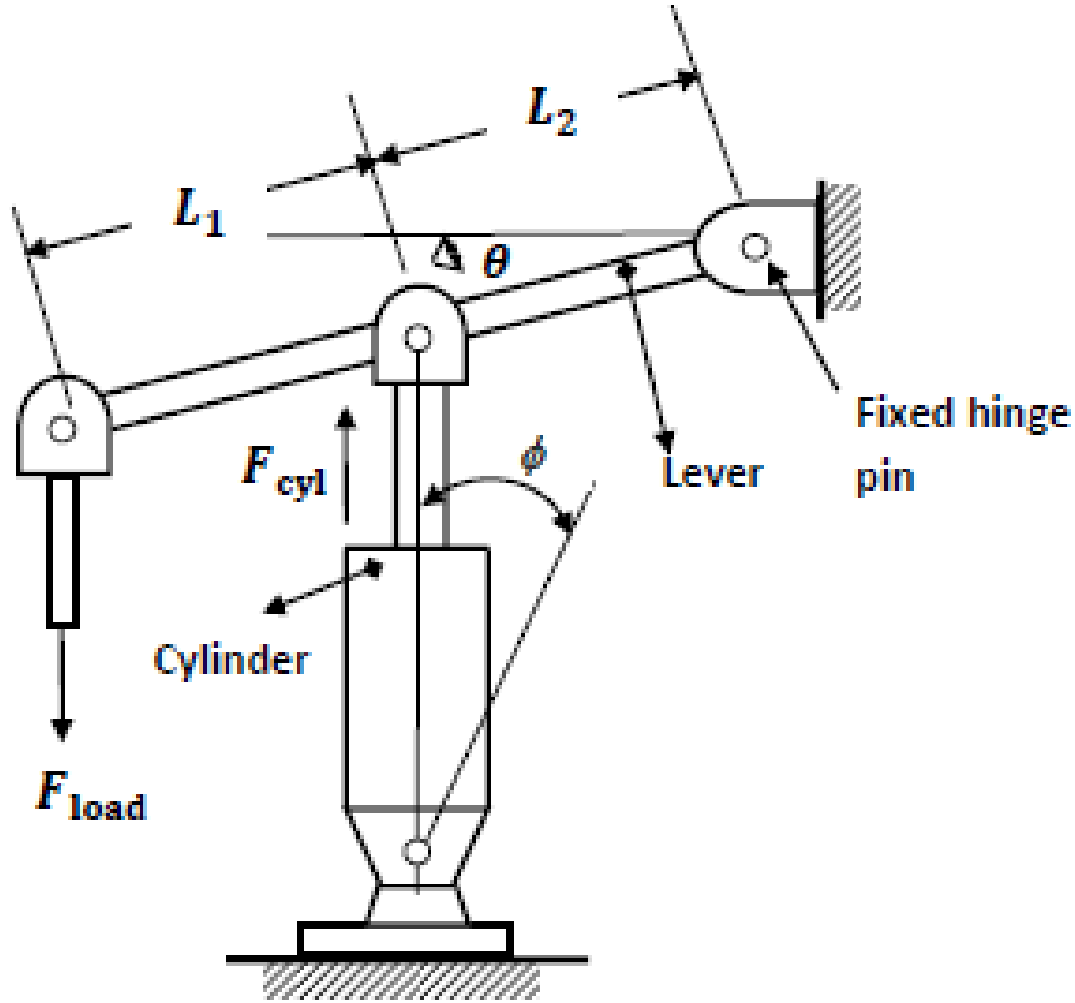


Figure 1. 23 Third-class lever system

Equating moments about the hinge point, we can write

$$F_{cyl} \cos\phi \times (L_2 \cos\theta) = F_{load} \times (L_1 + L_2) \cos\theta$$

$$F_{cyl} = \frac{L_1 + L_2}{L_2 \cos\phi} F_{load}$$

Example 1.5

An 8 cm diameter hydraulic cylinder has a 4 cm diameter rod. If the cylinder receives flow at 100 LPM and 12 MPa, find the (a) extension and retraction speeds and (b) extension and retraction load carrying capacities.

Solution:

Let us first convert the flow in LPM to m^3/s before we calculate forward velocity Q_{in}=100 LPM = 100/(1000 × 60) =1/600 m^3/s

Now

D_C = Diameter of cylinder = 8 cm = 8×10^{-2} m

d_r = Diameter of piston rod = 4 cm = 4×10^{-2} m

p = 12 MPa = 12×10^6 N/m^2 or Pa

(a) Forward velocity is given by

$$v_{ext} = \frac{Q_{in}}{A_p} = \frac{1/600}{\pi d^2/4} = 0.3315 \text{ m/s}$$

Return velocity is given by

$$v_{ret} = \frac{Q_{in}}{(A_p - A_r)} = \frac{\frac{1}{600}}{\frac{\pi(d_C^2 - d_r^2)}{4}} = 0.442 \text{ m/s}$$

(b) Force during extension is given by

$$F_{ext} = p \times a_p = 12 \times 10^6 \frac{\pi(8\times 10^{-2})^2}{4} = 60318.57 \text{ N}$$

$$F_{ret} = p \times (A_p - A_r)$$

$$= 12 \times \frac{10^6 \times \pi[(8\times 10^{-2})^2 - (4\times 10^{-2})^2]}{4}$$

$$= 42238.9 \text{ N} = 45.24 \text{ kN}$$

Example 1.12

A 10000 N weight is to be lowered by a vertical cylinder as shown in Fig. 1.18. The cylinder has a 75 mm diameter piston and 50 mm diameter rod. The weight is to decelerate from 100 m/min to a stop in 0.5 s. Determine the required pressure in the rod end of the cylinder during the deceleration motion.

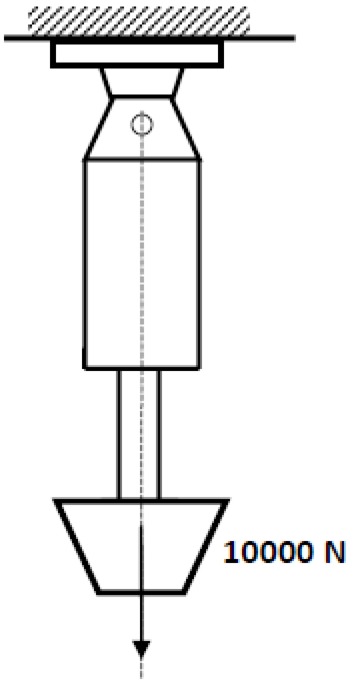


Figure 1.18

Solution: As per Newton's law of motion, we have

$$\sum F = m\ a$$

where

$$a = \frac{100\dfrac{\text{m}}{\text{min}}\times\dfrac{1\,\text{min}}{60\,\text{s}}}{0.5\,\text{s}} = \frac{1.67\,\text{m/s}}{0.5\,\text{s}} = 3.34\,\text{m/s}^2$$

Summing forces on the 10000 N weight, we have

$$p(A_\text{p} - A_\text{r}) - 10000\,\text{N} = \frac{10000\,\text{N}}{9.81\,\text{m/s}^2}\times 3.34\ \text{m/s}^2$$

$$\Rightarrow p\ (\text{N/m}^2)\times\frac{\pi}{4}(0.075^2 - 0.050^2)\ \text{m}^2 - 10000\,\text{N} = 3408\,\text{N}$$

Solving we get

$$p = 5450000\ (\text{N/m}^2) = 5450000\,\text{Pa} = 5450\,\text{kPa}$$

Example 1.13

A 27000 N weight is being pushed up on an inclined surface at a constant speed by a cylinder, as shown in Fig. 1.19. The coefficient of friction between the weight and the inclined surface equals 0.15.

(a) Determine the required cylinder piston diameter for the pressure of 6894 kPa,

(b) Determine the required cylinder piston diameter, if the weight is to accelerate from a 0 mm/s to a 1524 mm/s in 0.5 s.

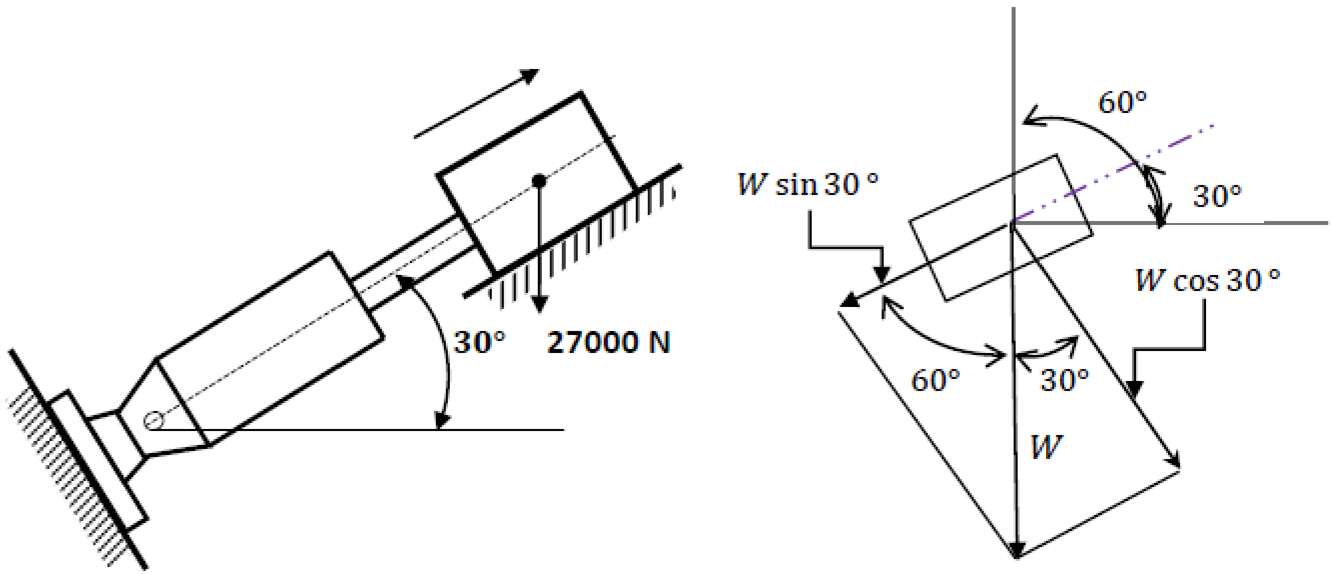


Figure 1.19

Therefore, the frictional force f acting along the axis of the cylinder is

$$\begin{aligned} f &= \mu \times W \cos 30° \\ &= 0.15 \times 27000 \times \cos 30° \\ &= 3507\,\mathrm{N} \end{aligned}$$

Total force on the cylinder is frictional force and vertical component:

$$\begin{aligned} F_{\text{cylinder}} &= f + W \sin 30° \\ &= 3507 + 27000 \times \sin 30° \\ &= 17007\ \mathrm{N} \end{aligned}$$

$$F_{\text{cylinder}} = pA_{\mathrm{p}} = 17007\ \mathrm{N}$$

Diameter to resist 17007 N is given by

$$F_{\text{cylinder}} = 6894000 \times \frac{\pi}{4} D^2 = 17007\ \mathrm{N}$$

$$\Rightarrow D = 0.05605\ \mathrm{m} = 56\ \mathrm{mm}$$

(b) Cylinder piston diameter if the weight is to accelerate from a 0 mm/s to a 1524 mm/s in 0.5 s

As per Newton's law of motion, we have calculate the acceleration

$$\sum F = m\,a$$

where

$$a = \frac{1524\,\text{mm/s}}{0.5\,\text{s}} = 3048\ \text{mm/s}^2 = 3.048\ \text{m/s}^2$$

Summing forces on the 27000 N weight using values determined in (a), we have

$$pA_p - 17007 = \frac{27000 \times 3.048}{9.81}$$

$$\Rightarrow \qquad 6894000 \times \frac{\pi}{4} D^2 - 17007 = \frac{27000 \times 3.048}{9.81}$$

$$\Rightarrow \qquad 5414535\,D^2 - 17007 = 8389$$

$$\Rightarrow D = 0.0685\ \text{m} = 68.5\ \text{mm}$$

Example 1.15

A mass of 2000 kg is to be accelerated horizontally up to a velocity of 1 m/s from the rest over a distance of 50 mm (Fig. 1.20). The coefficient of friction between the load and guide is 0.15. Calculate 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). Assume that the back pressure at the annulus end of the cylinder is zero.

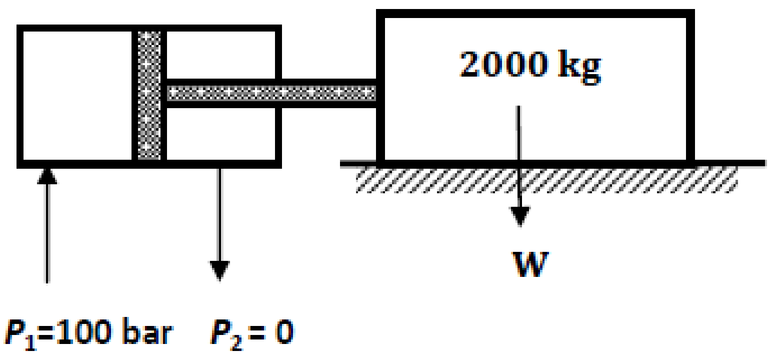


Figure 1.20

Solution: In this case, $u = 0$, $v = 1$ m/s, $s = 0.05$ m and a is unknown. Using the equation

$$v^2 = u^2 + 2as$$

We have,

$$1^2 = 0^2 + 2a \times 0.05 \Rightarrow a = 10 \text{ m/s}^2$$

The force to accelerate the load is given by

$$F = \left(\frac{w}{g}\right) a = \left(\frac{2000 \times 9.81}{9.81}\right) 10 = 20000 \text{N}$$

The force P to overcome the load friction is given by

$$P = \mu W = 0.15 \times 2000 \times 9.81 = 2943 \text{ N}$$

The total force to accelerate the load and overcome friction is

$$F + P = 20000 + 2943 = 22943 \text{ N}$$

The cylinder area required for a given thrust is calculated from

$$\text{Thrust} = \text{Force} \times \text{Area}$$

The pressure available is pressure at the full bore end of the cylinder minus the equivalent seal break-out pressure.

$$\text{Pressure available} = 100 - 5 = 95 \text{ bar} = 95 \times 10^5 \text{ bar}$$

Area is given by

$$\text{Area} = \frac{22943}{195 \times 10^5}$$
$$= 0.002415\,\text{m}^2 = 2415\,\text{mm}^2$$

Now area is also given by

$$\text{Area} = \frac{\pi D^2}{4}$$

where D is the diameter. Comparing the two equations , we get $D = 55.4$ mm.
The cylinder diameter is thus 55.4 mm. This neglects the effect of any back pressure. The nearest standard cylinder above has a 63 mm diameter bore.

Example 1.2

A three-stage displacement-type telescopic cylinder is used to tilt the body of a lorry (Fig. 1.10). When the lorry is fully laden, the cylinder has to exert a force equivalent to 4000 kg at all points in its stroke. The outside diameters of the tubes forming the three stages are 60, 80 and 100 mm. If the pump powering the cylinder delivers 10 LPM, calculate the extend speed and pressure required for each stage of the cylinder when tilting a fully laden lorry.

$$\text{First-stage pressure} = \frac{\text{Load}}{\text{Area}}$$

$$= \frac{4000 \times 9.81}{(\pi/4) \times (0.1)^2}\ \text{N/m}^2 = 5 \times 10^6\ (\text{N/m}^2) = 50\ \text{bar}$$

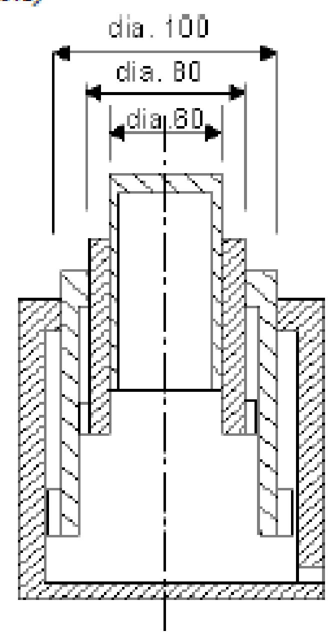


Figure 1.10

Solution:

First-stage

First-stage diameter = 100 mm

$$\text{First-stage speed} = \frac{\text{Quantity flowing}}{\text{Area}}$$

$$= \frac{10 \times 10^{-3}}{(\pi/4) \times (0.1)^2} \left(\frac{\text{m}^3}{\text{min} \times \text{m}^2} \right) = \frac{4}{\pi} = 1.27\ \text{m/min}$$

Second-stage:

Second-stage diameter = 80 mm

$$\text{Second-stage speed} = \frac{\text{Quantity flowing}}{\text{Area}}$$

$$= \frac{10\times10^{-3}}{(\pi/4)\times(0.08)^2}\left(\frac{\text{m}^3}{\text{min}\times\text{m}^2}\right) = 1.99 \ \text{m/min}$$

$$\text{Second-stage pressure} = \frac{\text{Load}}{\text{Area}}$$

$$= \frac{4000\times9.81}{(\pi/4)\times(0.08)^2}\text{N/m}^2 = 7.81\times10^6(\text{N/m}^2) = 78.1\text{bar}$$

Third-stage:

Third-stage diameter = 60 mm

$$\text{Third-stage speed} = \frac{\text{Quantity flowing}}{\text{Area}}$$

$$= \frac{10\times10^{-3}}{(\pi/4)\times(0.06)^2}\left(\frac{\text{m}^3}{\text{min}\times\text{m}^2}\right)$$

$$= 3.54 \ \text{m/min}$$

$$\text{Third-stage pressure} = \frac{\text{Load}}{\text{Area}}$$

$$= \frac{4000\times9.81}{(\pi/4)\times(0.06)^2}\text{N/m}^2 = 13.9\times10^6 = 139 \text{ bar}$$

Telescopic cylinders are made in a standard range for vehicle applications. Although non-standard cylinders can be obtained, they tend to be very expensive if ordered as a single piece.

Example 1. 19

Figure 1. 27 shows a toggle mechanism. Find the output load force for a hydraulic cylinder force of 1 N.

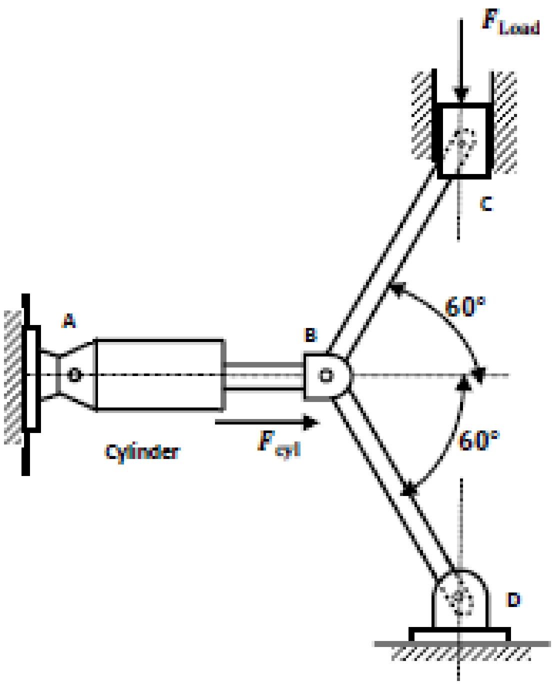


Figure 1. 27

Solution: Setting the sum of the forces on pin C equal to zero (from Newton's law of motion), force (F) $= ma = 0$ because $a = 0$ for constant velocity motion, yields the following for the x- and y-axes:

y-axis: $F_{BC}\sin 60° - F_{BD}\sin 60° = 0 \Rightarrow F_{BC} = F_{BD}$

x-axis: $F_{cyl} - F_{BC}\cos 60° - F_{BD}\cos 60° = 0$

$$F_{cyl} - 2F_{BC}\cos 60° = 0$$

$$\Rightarrow F_{BC} = \frac{F_{cyl}}{2\cos 60°}$$

Similarly, setting the sum of forces on pin C equal to zero for the y-axis direction yields

$$F_{BC}\sin 60° - F_{load} = 0$$

Therefore, we have

$$F_{load} = F_{BC}\sin 60°$$

$$= \frac{\sin 60°}{2\cos 60°} \times F_{cyl}$$

$$= \frac{\tan 60°}{2} \times 1000 = 866\text{N}$$

Example 1. 20

A cylinder has a bore of 125 mm diameter and a rod of 70 mm diameter. It drives a load of 2000 kg vertically up and down at a maximum velocity of 3 m/s. The lift speed is set by adjusting the pump displacement and the retract speed by a flow control valve. The load is slowed down to rest in the cushion length of 50 mm. If the relief valve is set at 140 bar, determine the average pressure in the cushions on extend and retract. (Neglect pressure drops in pipe work and valves.)

Solution: Kinetic energy of load

$$\text{Kinetic energy} = (1/2)\ \text{Mass} \times \text{Velocity}^2$$

$$= (1/2)\ (2000) \times 3^2 = 9000\ \text{N m}$$

Average force to retard load over 50 mm is

$$\frac{\text{Kinetic energy}}{\text{Distance}} = \frac{9000 \times 10^3}{50} = 180\ \text{kN}$$

The force acting on the load is

$$\text{Load} = 2000\ \text{kg} = 2000 \times 9.81 = 19.6\ \text{kN}$$

$$\text{Annulus area} = \frac{\pi}{4}(0.125^2 - 0.07^2) = 0.0084\ \text{m}^2$$

$$\text{Full bore area} = \left(\frac{\pi}{4}\right) \times (0.125^2) = 0.0123\ \text{m}^2$$

The kinetic energy of the load is opposed by the cushion force and the action of gravity on the load. Cushion pressure to absorb the kinetic energy of load when extending is

$$\frac{(180 \times 10^3) - (19.6 \times 10^3)}{(8.4 \times 10^{-3})}\ (\text{N/m}^2) = 19.1 \times 10^6 = 191\ \text{bar}$$

When the piston enters the cushion, the pressure on the full bore side of the piston rises to relief valve pressure. This pressure on the full bore side drives the piston into the cushions, and so increases the cushion pressure needed to retard the load. The cushion pressure to overcome the hydraulic pressure on the full bore end is

$$\text{Pressure} \times \frac{\text{Full bore area}}{\text{Annulus area}} = 140 \times \frac{12.3 \times 10^{-3}}{8.4 \times 10^{-3}} = 205\ \text{bar}$$

Thus, the average pressure in the cushion on the extend stroke is (190 + 205) = 395 bar.
During cushioning, the effective annular area is reduced as the cushion sleeve enters the cushion. This has been neglected in the calculation, and in practice, the cushion pressure is even greater.
When the load is retracted, forces act on the load. The back pressure owing to the flow control valve in the circuit is minimal once the piston enters the cushion and is neglected in this calculation.
The force in the cushion has to overcome the kinetic energy of the load, the weight of the load and the force due to the hydraulic pressure. The force owing to the hydraulic pressure is

$$\text{Force} = \text{Pressure} \times \text{Annulus area}$$
$$= (140 \times 10^5) \times (8.4 \times 10^{-3})\ \text{N}$$
$$= 117.6\ \text{kN}$$

Also

$$\text{Cushion force} = 180 + 19.6 + 117.6 = 317.2\ \text{kN}$$

After knowing the force, we can find cushion pressure =

$$\text{Cushion pressure} = \frac{\text{Force}}{\text{Area}} = \frac{317.2}{0.0123}\ (\text{kN/m}^2) = 25800 = 258\ \text{bar}$$

The average pressure in the cushion retracting is 258 bar. Again this value is somewhat higher as the cushion spike reduces the effective cushion area below that used.

Example 1. 21

A pump delivers oil at a rate of 1.15 LPS into the blank end of the 76.2 mm diameter hydraulic cylinder shown in Fig. 1. 30. The pistons decelerate over a distance of 19.05 mm at the end of its extension stroke. The cylinder drives a 6672 N weight which slides on a flat horizontal surface having a coefficient of friction (CF) equal to 0.12. The pressure relief valve setting equals 51.7125 bar. Therefore, the maximum pressure (p_1) at the blank end of the cylinder equals 51.7125 bar while the cushion decelerates the piston. Find the maximum pressure (p_2) developed by the cushion.

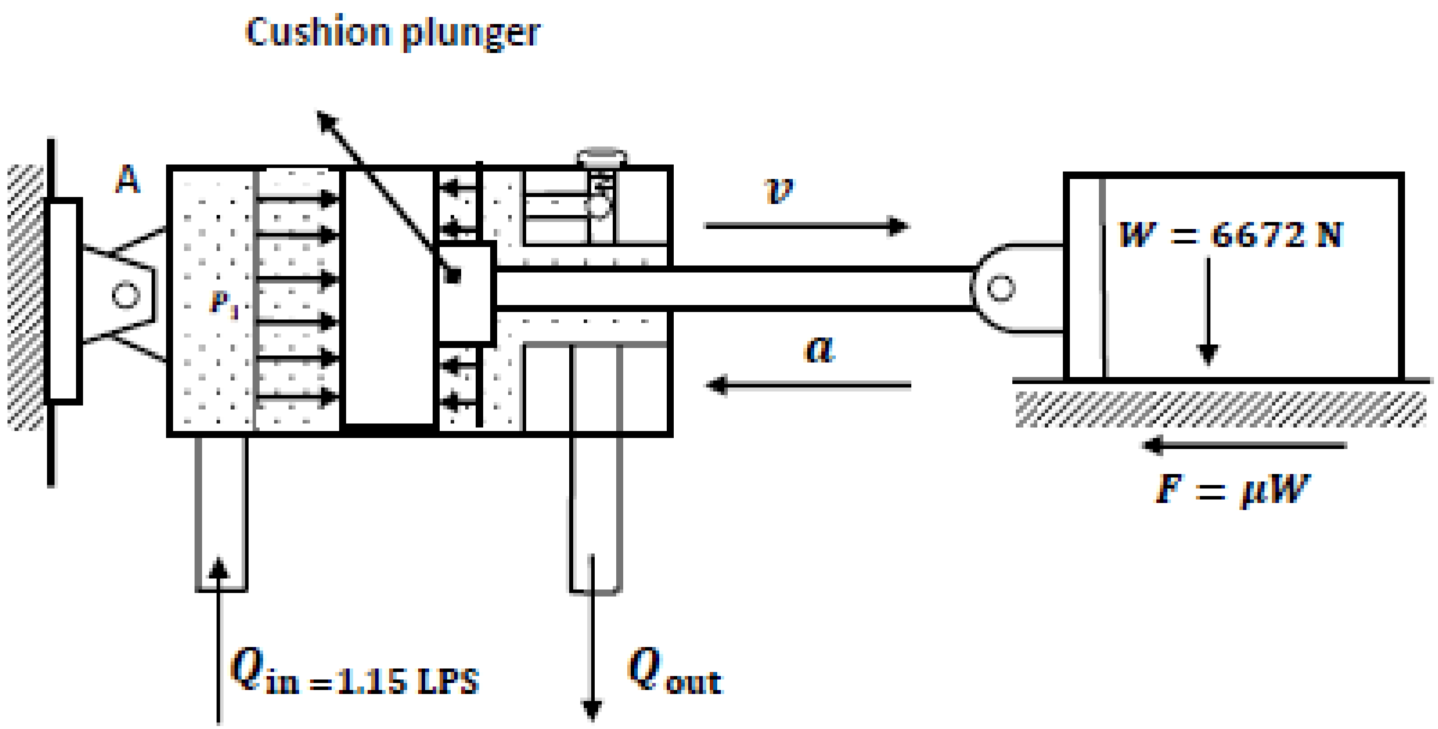


Figure 1. 30

Solution:

Step 1: Calculate the steady piston velocity v prior to deceleration:

$$v=\frac{Q_{\text{pump}}}{A_{\text{piston}}}=\frac{\frac{1.15}{1000}}{\frac{\frac{\pi}{4}(0.0762^2)}{1}}=\frac{1.15}{4.5}=0.255\text{ m/s}$$

Step 2: Calculate the deceleration a of the piston during the 19.05 mm displacement S using the constant acceleration or deceleration equation:

$$v^2=2as$$

Substituting the values and solving for deceleration we get

$$a=\frac{v^2}{2s}=\frac{0.255^2}{2(19.05\times 10^{-3})}=17.06\text{ m/s}^2$$

Step 3: Using Newton's laws of motion, the net force acting on the system is equated as

$$\sum F=ma$$

Consider the forces that tend to slow down the system as positive forces as we are solving for deceleration. The mass under consideration m is equal to the sum of all the masses of moving members (piston, road and load). Because the weight of the piston and road is small compared to the weight of the load, the weight of the piston and rod is ignored. The frictional forces acting between the weight W and the horizontal support surface equal *coefficient of friction* (CF) times W. This frictional force is the external force acting on the cylinder while it moves the weight.

Substituting into Newton's equations yields

$$p_2(A_{\text{piston}}-A_{\text{cushion}})+\text{CF}\times W-p_1(A_{\text{piston}})=\frac{W}{g}$$

$$p_2 = \frac{\left[(6672)\left(\frac{17.06}{9.81}\right)\right] + 51.7125\left(\frac{\pi}{4}\right)(76.2\times10^{-3})^2 - (0.12)(6672)}{\left[\frac{\pi}{4}(76.2\times10^{-3})^2 - \frac{\pi}{4}\left(25.4\times10^{-3}\right)^2\right]} = 59.0212\,\text{bar}$$

Thus, the hydraulic cylinder must be designed to withstand an operating pressure of $59.0212\,\text{bar}$ rather than the pressure relief setting of $51.7125\,\text{bar}$.

Example

Calculate the force output for a linear actuator and the required pump flow rate for the following specifications:

Piston diameter = 50 mm

Rod diameter =25 mm

Stroke = 600 mm

Piston speed = l2 mm/s

Efficiency (η_{hm}) = 95%

Pressure = 4000 kPa

$$\text{Force}_{(\text{Ext})} = \frac{4000 \text{ x } 10^3 \text{ x } 0.05^2 \text{ x } 0.7854 \text{ x } 95}{10^3 \text{ x } 100}$$

$$= 7.46 \text{ kN}$$

$$\text{Force}_{(\text{Ret})} = \frac{4000 \text{ x } 10^3 \text{ x } (0.05^2 - 0.025^2) \text{ x } 0.7854 \text{ x } 95}{10^3 \text{ x } 100}$$

$$= 5.6 \text{ kN}$$

$$\text{Flow rate}_{(\text{Ext})} = \frac{0.012 \text{ x } 0.05^2 \text{ x } 0.7854}{\text{s}}$$

$$= 0.000024 \text{m}^3/\text{s}$$

$$= 0.24 \text{ L/s}$$

$$\text{Flow rate}_{(\text{Ret})} = \frac{0.012 \text{ x } (0.05^2 - 0.025^2) \text{ x } 0.7854}{\text{s}}$$

$$= 0.0000177 \text{ m}^3/\text{s}$$

$$= 0.0177 \text{ L/s}$$

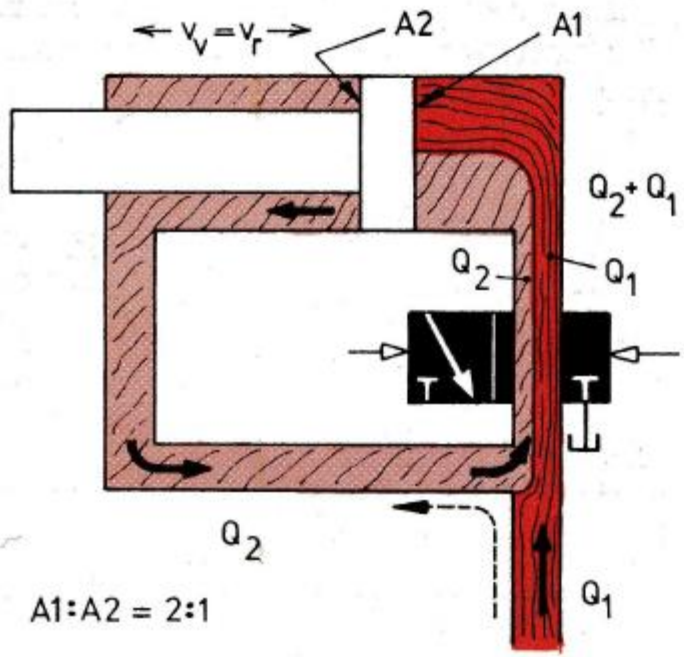
Regenerative actuator control

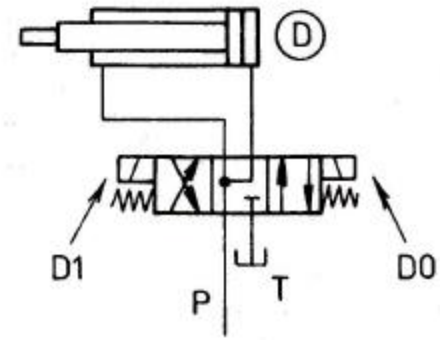
Regeneration is achieved by using suitable valving which connects the exhaust flow from the extending piston rod with the in-rushing fluid on the piston end. In this way, the exhaust fluid, which would normally return to tank, joins the fluid flow from the pump, thus causing the piston rod to extend with increased speed.

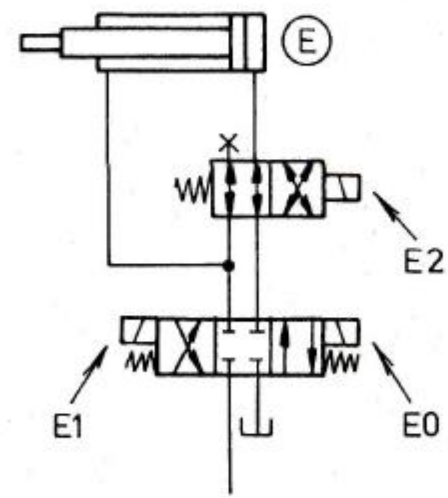
If the full piston area (A1) has a ratio of 2:1 to the annulus area (A2), then the piston rod will extend and retract with equal speed (fig. 94). During regeneration, equal pressure is acting onto both sides of the piston. Therefore, the net thrust for extension can be calculated as:

$Force_{(Ext)} = p \times (A1 - A2)$, where A1 - A2-rod area.

Thus, the extension force is the product of the piston rod area and the system pressure. A similar calculation can be made for the retraction stroke. Since the annulus area and the rod area are equal (with a 2:1 ratio), the retraction force can be calculated as:







E1	E0	E2	
0	0	0	Actuator blocked
0	0	1	———
0	1	0	Retraction
0	1	1	Fast extension
1	0	0	Normal extension
1	0	1	———
1	1	0	———
1	1	1	———

Fig. 94 Function of regenerative actuator with alternative control methods.

$Force_{(RET)} = p \times$ Annulus Area (rod area = annulus area).

Thus, the forces for extension and retraction are equal. With a 2:1 ratio, the exhaust volume for the pushed-out fluid during extension is one half the volume formed by the full piston area (A1). Hence, the pump must fill the other half. Therefore, the regenerated volume and the pumped volume are equal ($V_R = V_P$), and the required volume for extension is:

$V_{ext} = V_R + V_P$, or $V_{Ext} = V \times 2$.

Furthermore, the required volume for retraction is V_R or $V \times 1$ (remember, V_R and V_P are equal).

Piston rod speed is calculated with the formula:

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

where Q stands for flow rate,
A for effective piston area, and
v for velocity or speed.

As previously shown, the extension volume is twice the retraction volume ($V_{ext} = V \times 2$; $V_{RET} = V \times 1$; volume ratio 2:1). The flow rate for retraction is equal to the flow rate of the pump. Using the piston rod speed formula one can calculate ratio 2:1). The flow rate for retraction is equal to the flow rate of the pump. Using the piston rod speed formula one can calculate:

where
$Q \times 1$ = pump flow rate
$Q \times 2$ = pump flow rate + exhaust flow rate
$A \times 1$ =annulus area
$A \times 2$ = piston area

Therefore, regenerative control with a piston-to-rod area ratio of 2:1 provides equal speed and equal force for extension and retraction

ME 7553 – Hydraulics and Pneumatics

Lecture -9

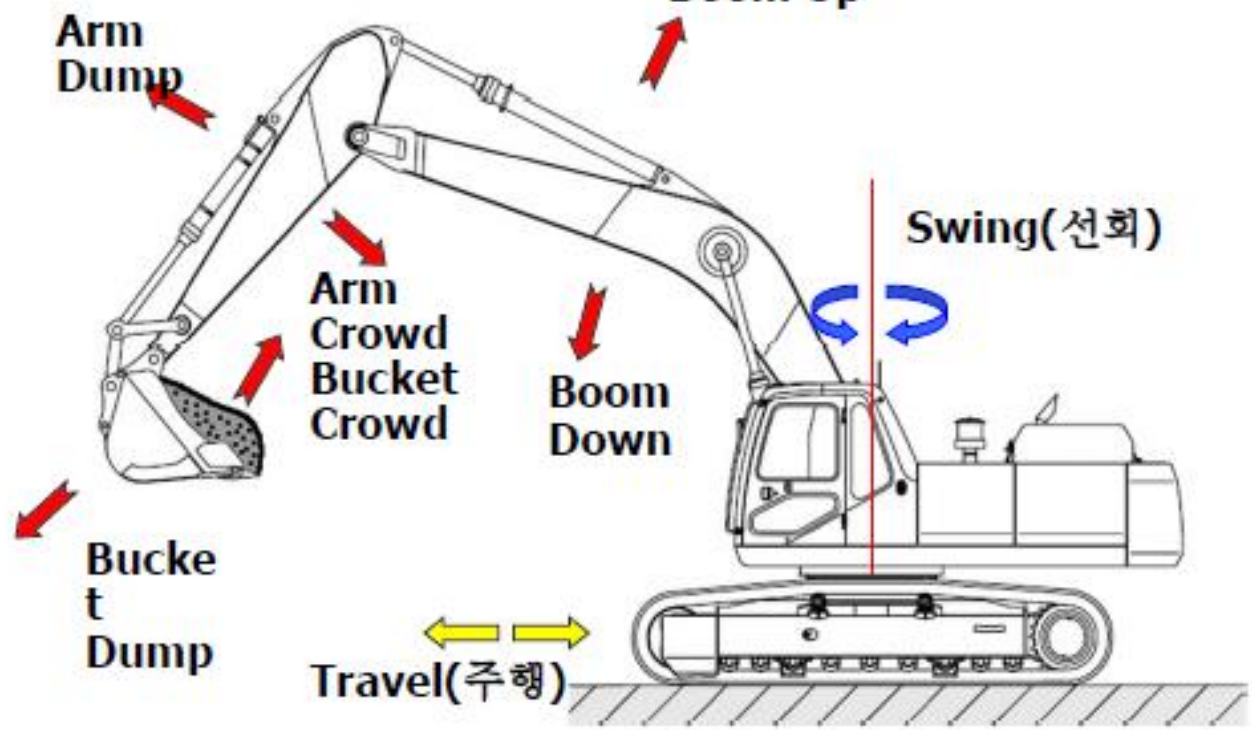
Date: 08-04-2021 Time slot: 10:30-12:10 p.m.

Contents

1. Review of Lecture 8
2. Problems
3. Directional control valve

Course Instructor: Dr. A. Siddharthan

Hydraulic Excavator



6. A hydraulic cylinder has to move a table of weight 13kN. Speed of the cylinder is to be accelerated upto a velocity of 0.13m/s in 0.5 seconds and brought to a stop within a distance of 0.02m. Assume coefficient of sliding friction as 0.15 and cylinder bore diameter as 50mm. Calculate the surge pressure.

Solution

Initial velocity $u = 0\text{m/s}$

Final velocity $v = 0.13\text{m/s}$

Acceleration $a = \dfrac{v-u}{t} = \dfrac{0.13-0}{0.5}$

$= 0.26\text{m/s}^2$

Force required to move the piston = Dynamic force + frictional force

$$= \left[\frac{w}{g} \times a\right] + \mu.w$$

$$= \left[\frac{13000}{9.81} \times 0.26\right] + (0.15 \times 13000)$$

$$= 2294.5\text{N}$$

To overcome this force, the pressure required in the hydraulic cylinder is

$$P_1 = \frac{2294.5}{\pi/4 \times (0.05)^2}$$

$$= 11.69 \times 10^5 \text{ N/m}^2$$

$$= 11.69 \text{ bar}$$

From the equation for velocity, acceleration and distance $v^2 - u^2 = 2as$

$$a = \frac{v^2 - u^2}{2s}$$

$$= \frac{0 - 0.13^2}{2 \times 0.02}$$

$= -0.4225 \text{ m/s}^2$ (The –ve sign indicates that it is deceleration)

The total force required to stop the motion of a cylinder

$$= \frac{13000}{9.81} \times 0.4225 + 13000 \times 0.15$$

$$= 2510 \text{N}$$

Then pressure created by this opposing force is

$$P_2 = \frac{2510}{\pi/4 \times (0.05)^2}$$

$$= 12.78 \times 10^5 \text{ N/m}^2$$

$$= 12.78 \text{bar}$$

Thus surge pressure

$$P_s = P_1 + P_2$$

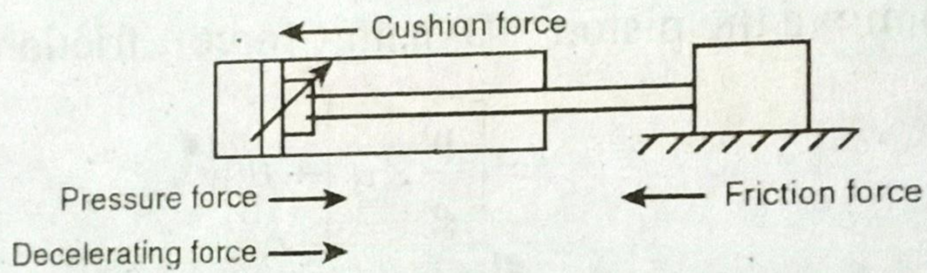
$$= 11.69 + 12.78$$

$$= 24.47 \text{ bar}$$

$= 24.47$ bar

7. A cylinder has a bore of 80mm diameter and a rod of 45mm diameter. It drives a load of 6700N, travelling at a velocity of 15m/min. The load slides on a flat horizontal surface having a coefficient of friction of 0.12. The load is to be decelerated to rest within a cushion length of 20mm. If the relief valve is set at 50 bar, compute the fluid pressure developed in the cushion.

Solution



Cushion length $s = 20\text{mm} = 0.02\text{m}$

Velocity $u = 15\text{m/min} = 0.25 \text{ m/s}$

From the eqution of motion.

$v^2 = u^2 + 2as$ (final velocity is zero)

$\therefore a = \dfrac{-u^2}{2s}$ (Negative sign indicates deceleration)

Decelerating force to retard load $= \dfrac{w}{g} \times a$

$$= \frac{w}{g} \times \frac{u^2}{2s}$$

$$= \frac{6700}{9.81} \times \frac{0.25^2}{2 \times 0.02}$$

$$= 1067\text{N}$$

Pressure force on blank end $= P \times A$

$$= 50 \times 10^5 \times \pi/4 \times 0.08^2$$

$$= 25133\text{N}$$

Friction force

$$= \mu w$$

$$= 0.12 \times 6700$$

$$= 804\text{ N}$$

$\therefore$ Cushion force = (Pressure force + Decelerating force) – Friction force

$$= 25133 + 1067 - 804$$

$$= 25396 \text{N}$$

Fluid pressure developed at the cushion $= \dfrac{F}{(A_P - A_R)}$

$$= \frac{25396}{\pi / 4(0.08^2 - 0.045^2)}$$

$$= 74 \text{bar}$$

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 3m/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.

Solution

From the equations of motion,

$v^2 = u^2 + 2as$ (final velocity is zero)

$$a = \frac{-u^2}{2s}$$

$\therefore$ Decelerating force to retend $= ma$

$$= \frac{2000 \times 3^2}{2 \times 0.05}$$

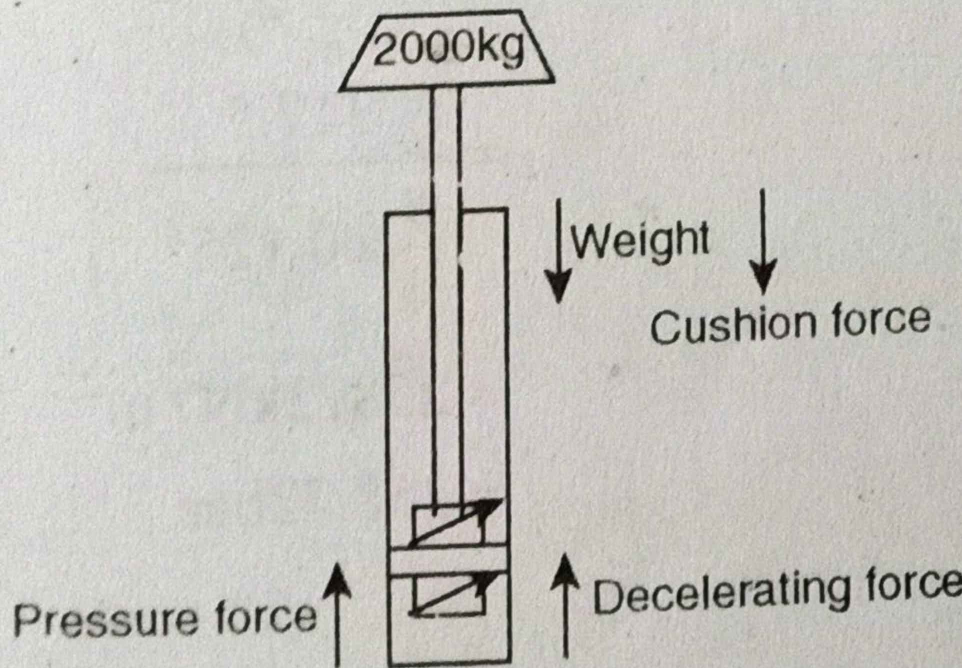
$$= 180\,\text{kN}$$

Weight of the load $= mg$

$$= 2000 \times 9.81$$

$$= 19.6\,\text{kN}$$

a. During extension



$$\text{Pressure force} = P \times A_p$$
$$= 140 \times 10^5 \times \pi/4 \times 0.125^2$$
$$= 171.8\,\text{kN}$$

Cushion force = (Pressure force + Decelerating force – weight)

$= 171.8 + 180 - 19.6$

$= 332.2\ \text{kN}$

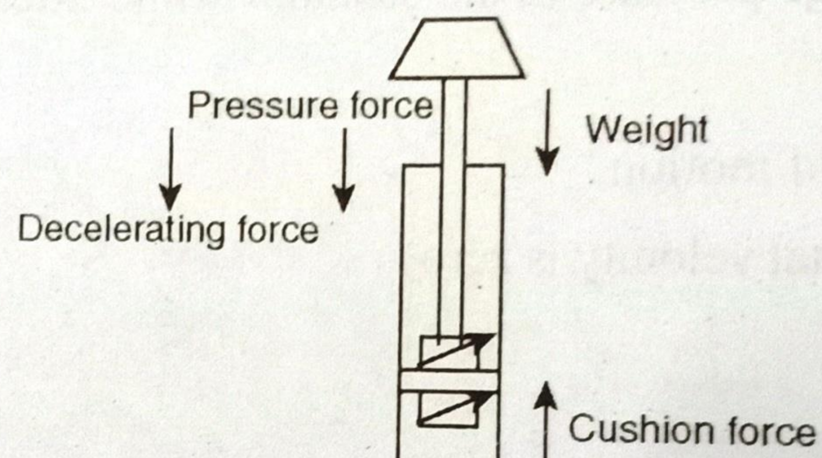
$$\text{Cushion pressure} = \frac{\text{Cushion force}}{(A_P - A_R)}$$

$$= \frac{332.2}{\frac{\pi}{4}(0.125^2 - 0.07^2)}$$

$$= 39438\text{kN}/\text{m}^2$$

$$= 394.38\text{bar}$$

b. During retraction



$$\text{Pressure force} = P \times (A_P - A_R)$$

$$= 140 \times 10^5 \times \frac{\pi}{4}(0.125^2 - 0.07^2)$$

$$= 117.9\,\text{kN}$$

$$\text{Cushion force} = (\text{Pressure force} + \text{Decelerating force} + \text{Weight})$$

$$= 117.9 + 180 + 19.6$$

$$= 317.5\,\text{kN}$$

$$\text{Cushion pressure} = \frac{\text{Cushion force}}{A_P}$$

$$= \frac{317.5}{\frac{\pi}{4} \times 0.125^2}$$

$$= 25872\,\text{kN/m}^2$$

$$= 258.72\,\text{bar}$$

1.2Directional Control Valves

A valve is a device that receives an external signal (mechanical, fluid pilot signal, electrical or electronics) to release, stop or redirect the fluid that flows through it. The function of a DCV is to control the direction of fluid flow in any hydraulic system. A DCV does this by changing the position of internal movable parts. To be more specific, a DCV is mainly required for the following purposes:

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

Any valve contains ports that are external openings through which a fluid can enter and exit via connecting pipelines. The number of ports on a DCV is identified using the term "way." Thus, a valve with four ports is a four-way valve A DCV consists of a valve body or valve housing and a valve mechanism usually mounted on a sub-plate. The ports of a sub-plate are threaded to hold the tube fittings which connect the valve to the fluid conductor lines. The valve mechanism directs the fluid to selected output ports or stops the fluid from passing through the valve. DCVs can be classified based on fluid path, design characteristics, control methods and construction.

1.2.1 Classification of DCVs based Fluid Path

Based on fluid path, DCVs can be classified as follows:

- Check valves.
- Shuttle valves.
- Two-way valves.
- Three-way valves.
- Four-way valves.

1.2.3 Classification of DCVs based on the Control Method

Based on the control method, DCVs can be classified as follows:

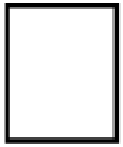
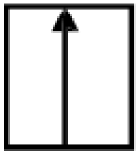
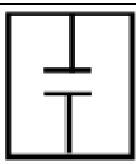
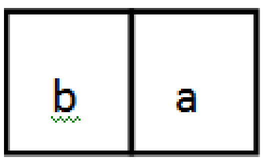
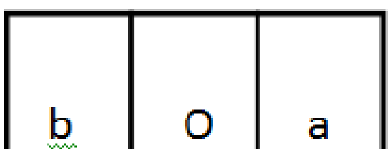
- **Direct controlled DCV:** A valve is actuated directly on the valve spool. This is suitable for small-sized valves.
- **Indirect controlled DCV:** A valve is actuated by a pilot line or using a solenoid or by the combination of electrohydraulic and electro-pneumatic means. The use of solenoid reduces the size of the valve. This is suitable for large-sized valves.

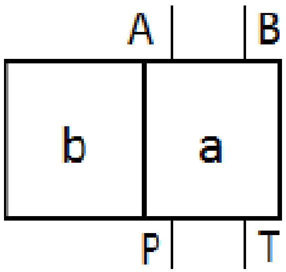
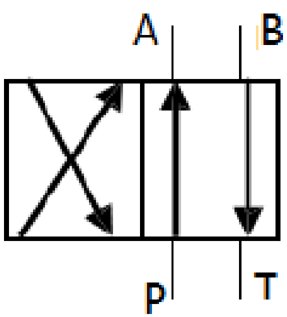
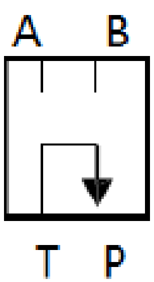
1.2.4 Classification of DCVs based on the Construction of Internal Moving Parts

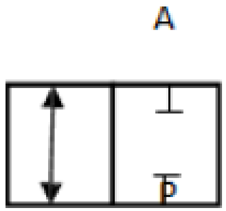
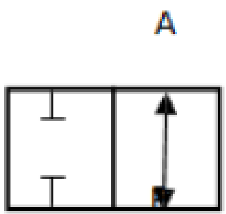
Based on the construction of internal moving parts, DCVs can be classified as follows:

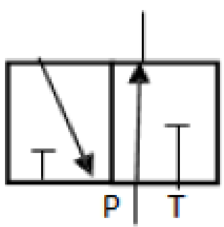
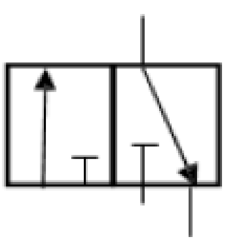
- **Rotary spool type:** In this type, the spool is rotated to change the direction of fluid. It has longitudinal grooves. The rotary spools are usually manually operated.
- **Sliding spool type:** This consists of a specially shaped spool and a means of positioning the spool. The spool is fitted with precision into the body bore through the longitudinal axis of the valve body. The lands of the spool divide this bore into a series of separate chambers. The ports of the valve body lead into these chambers and the position of the spool determines the nature of inter-connection between the ports.

Table 1.1

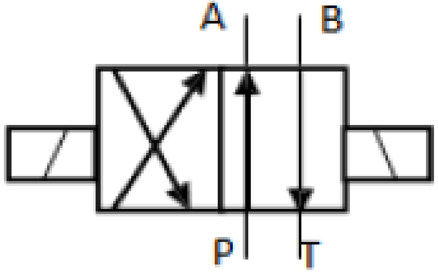
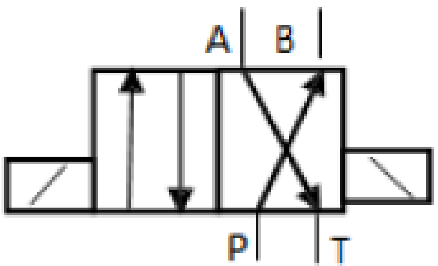
	Each individual switching portion is shown in a square
	Flow path is indicated by means of arrow within a square
	Closed position
	Two-position valve
	Three-position valve

	Ports added to the two-position valve
	Two flow paths
	Two ports are connected, two ports are closed

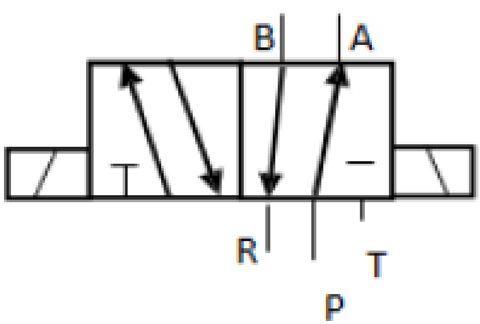
2/2-way valve: 2-ports and 2-position DCV	
	Normally closed position: P is not connected to A. When the valve is not actuated, the way is closed.
	Normally open position: P is connected to A. When the valve is not actuated, the way is open.

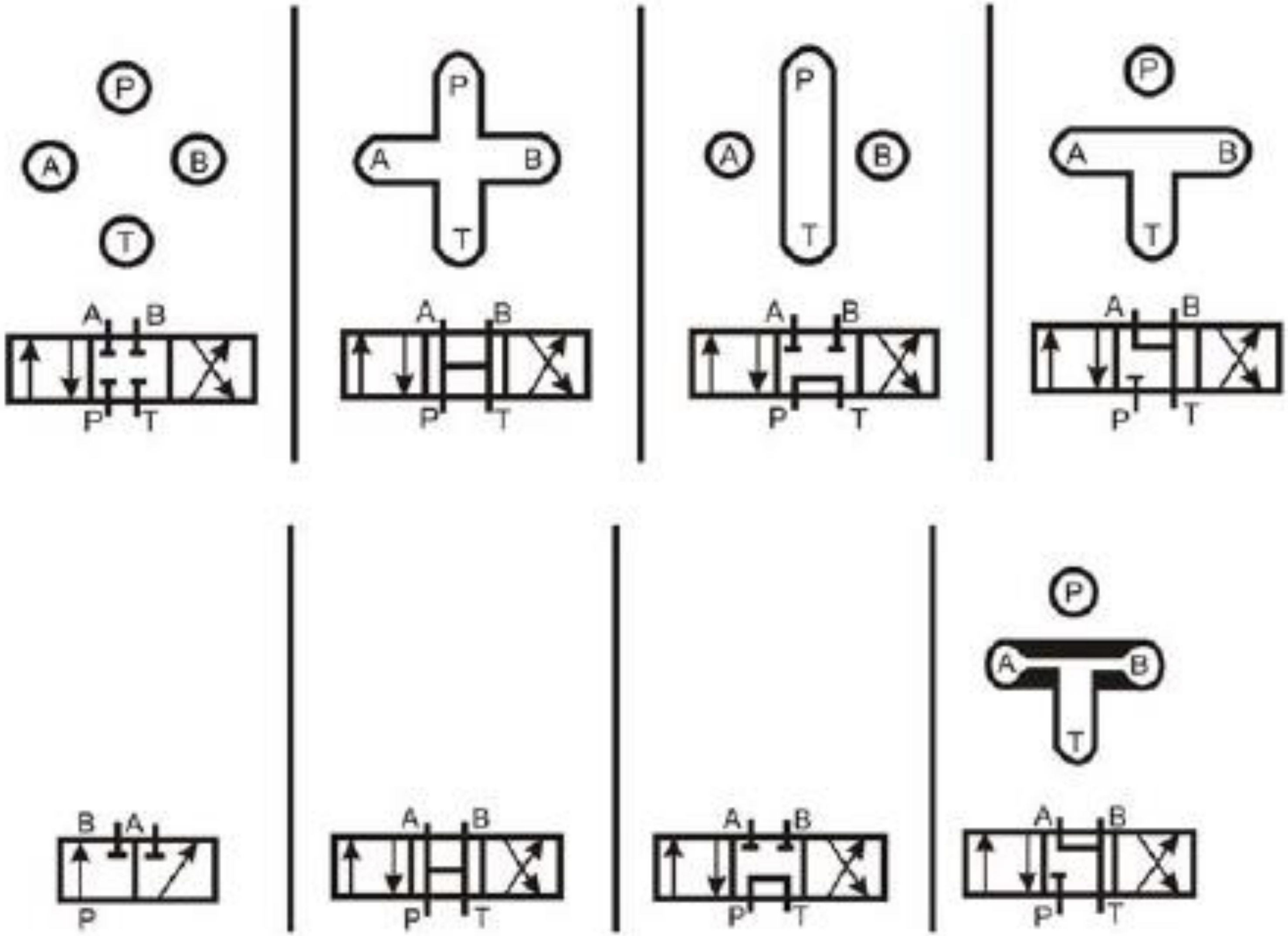
3/2 way valve : 3ports and 2 position DCV	
	Normally open position: P is connected to A. When the valve is not actuated, the way is open.
	Normally open position: P is connected to A. When the valve is actuated, the way is closed

4/2-way valve – 4-port and 2-position DCV

	P is connected to A B is connected to T
	Position 2: P is connected to B A is connected to T

5/2-way valve – 5-port and 2-position DCV

	Normal position: P is connected to B A is connected to R


P
A
B
T
A
B
P
T
P
A
B
T
A
B
P
T
P
A
B
T
A
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P
T

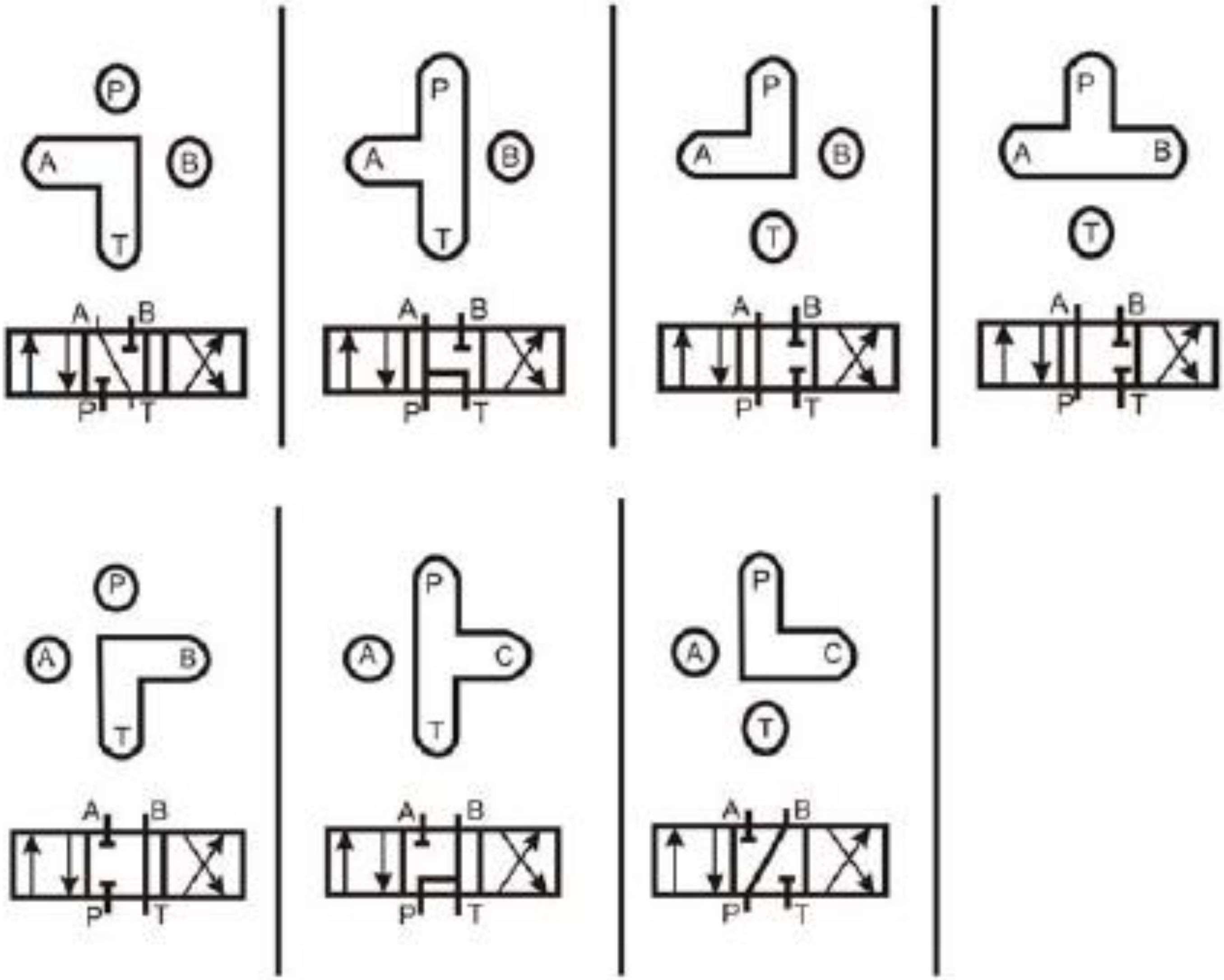


Fig.- 3.8-7 : Spool Positions, Port Connections and Symbols for DC Valves

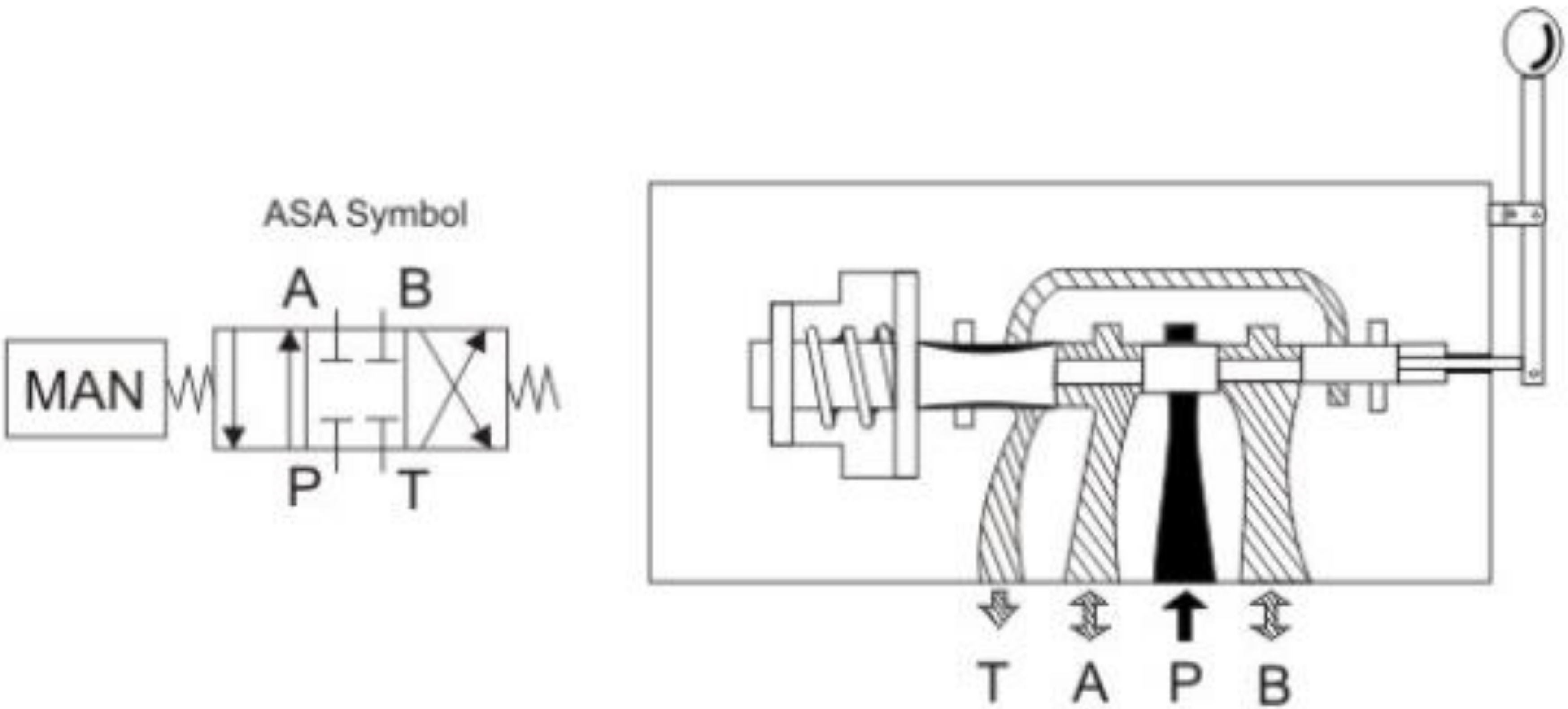
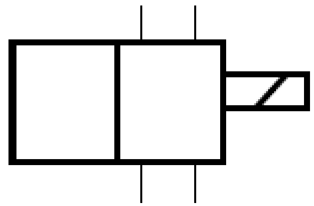
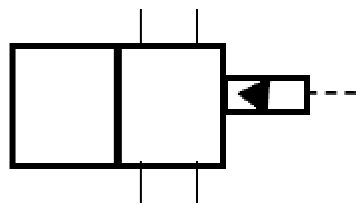
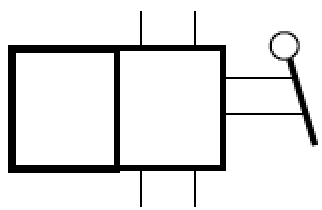
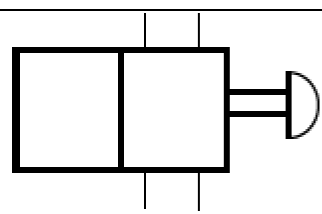
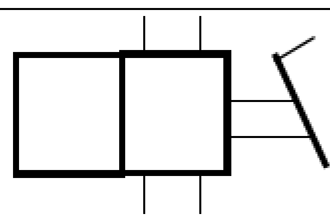


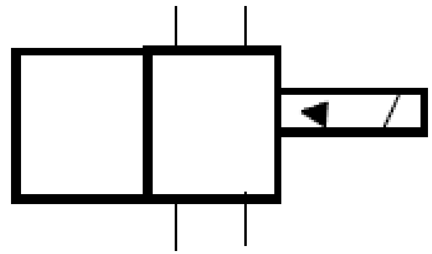
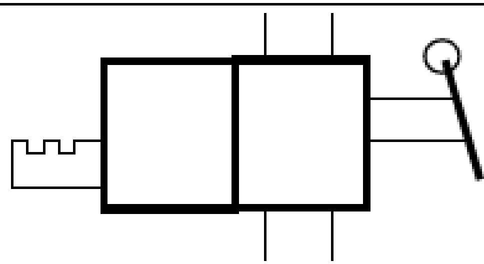
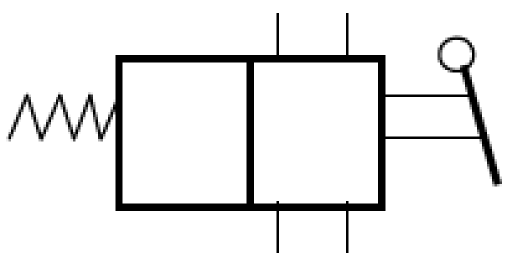
Fig.- 3.6-8 : Close Centre Direction Control Valve

1.3Actuating Devices

Direction control valves may be actuated by a variety of methods. Actuation is the method of moving the valve element from one position to another. There are four basic methods of actuation: Manual, mechanical, solenoid-operated and pilot-operated. Several combinations of actuation are possible using these four basic methods. Graphical symbols of such combinations are given in Table 1.3.

- **Manually operated:** In manually operated DCVs, the spool is shifted manually by moving a handle pushing a button or stepping on a foot pedal. When the handle is not operated, the spool returns to its original position by means of a spring.
- **Mechanically operated:** The spool is shifted by mechanical linkages such as cam and rollers.
- **Solenoid operated:** When an electric coil or a solenoid is energized, it creates a magnetic force that pulls the armature into the coil. This causes the armature to push the spool of the valve.
- **Pilot operated:** A DCV can also be shifted by applying a pilot signal (either hydraulic or pneumatic) against a piston at either end of the valve spool. When pilot pressure is introduced, it pushes the piston to shift the spool.

	Solenoid operated
	Pilot operated
	Manual operated
	Push button
	Foot operated

	Pilot-operated solenoid
	Two-position detent
	Spring return

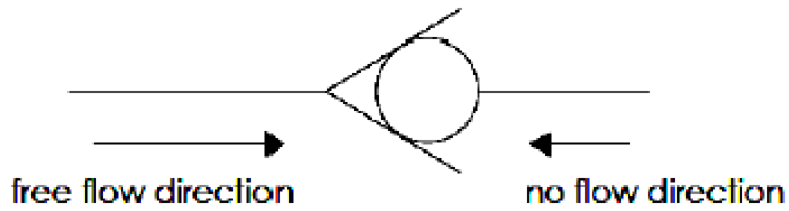
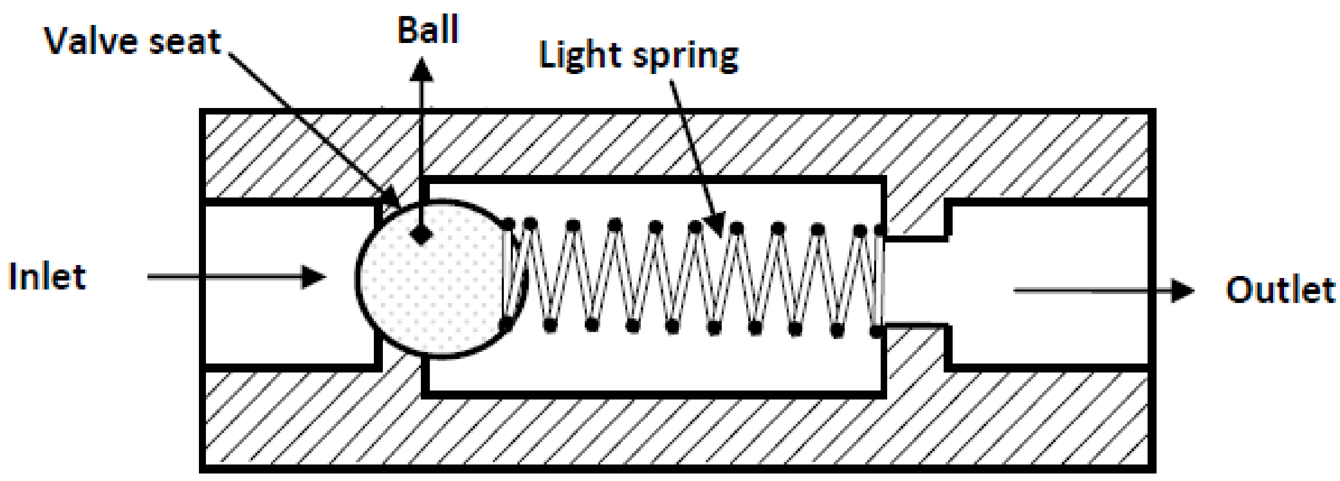


Figure 1.1 Graphical symbol of a check valve.

In Fig. 1.2, a light spring holds the ball against the valve seat. Flow coming into the inlet pushes the ball off the seat against the light force of the spring and continues to the outlet. A very low pressure is required to hold the valve open in this direction. If the flow tries to enter from the opposite direction, the pressure pushes the ball against the seat and the flow cannot pass through.



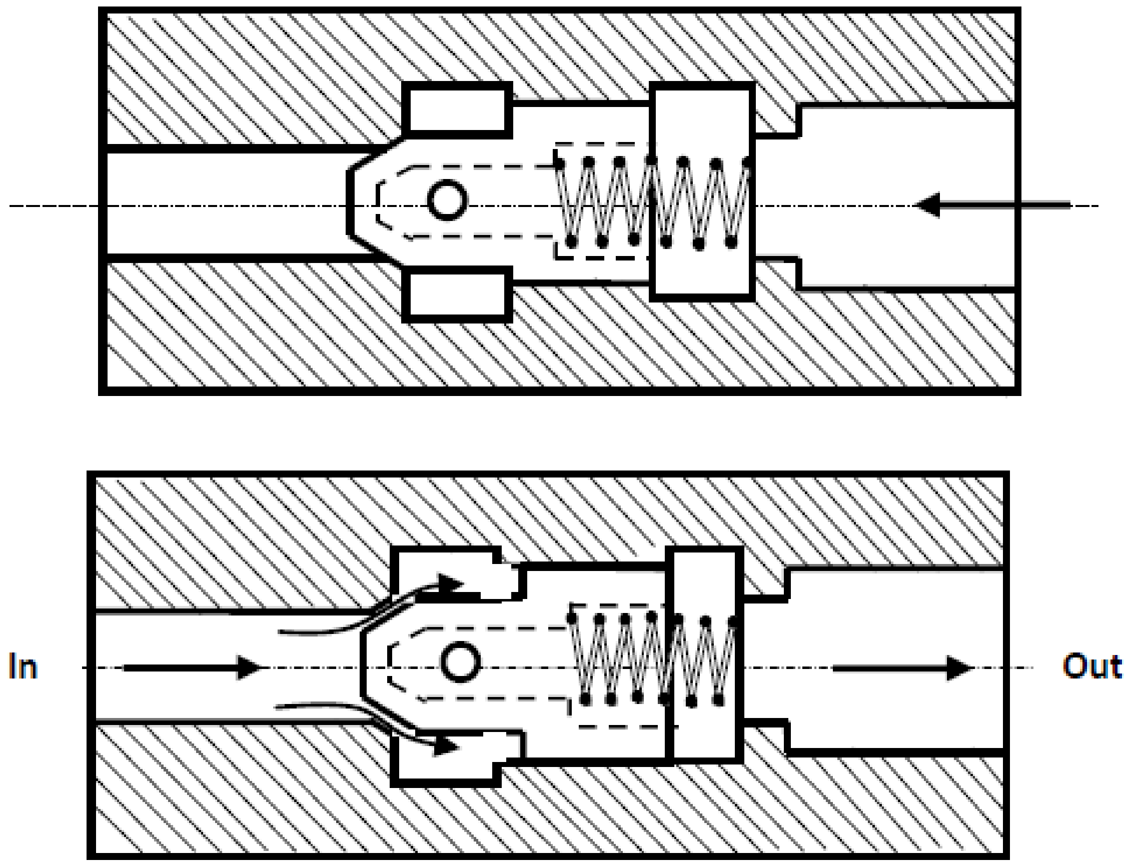


Figure1.3 Poppet check valve: (a) Open and (b) closed position

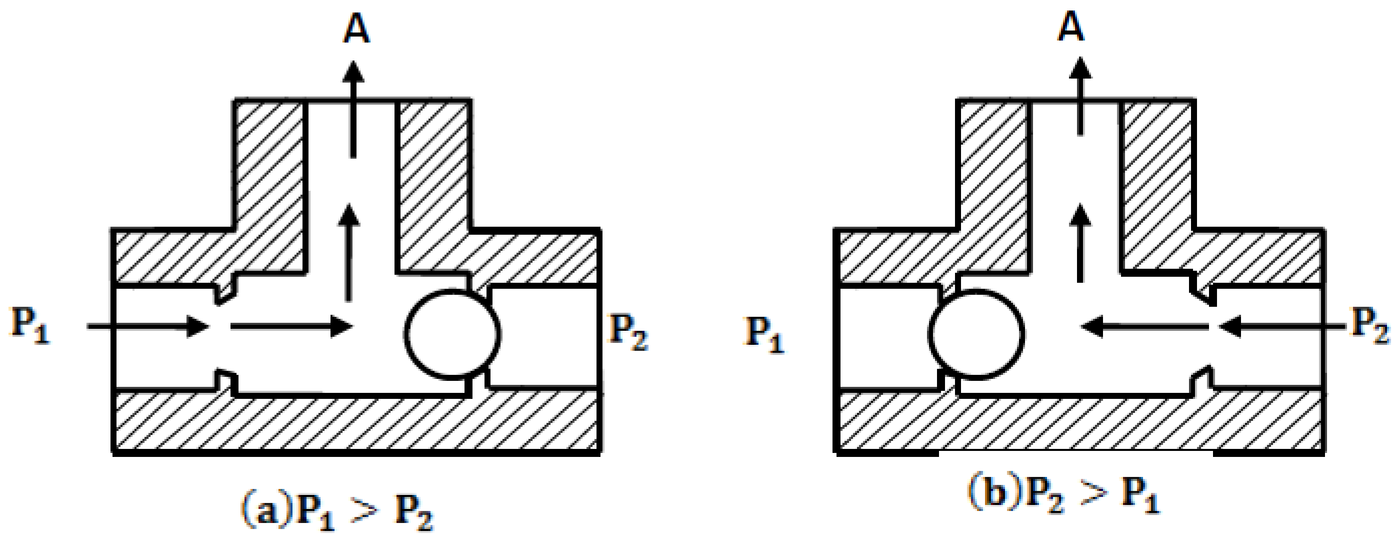


Figure 1.5 Shuttle valve: (a) Flow from left to outlet and (b) flow from right to outlet in Fig. 1.5.

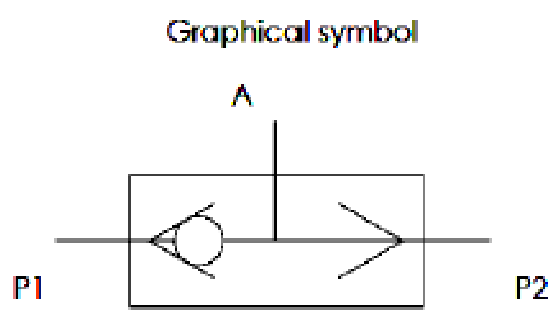


Fig 1.6 symbol of Shuttle valve

ME 7553 – Hydraulics and Pneumatics

Lecture -10

Date: 29-04-2021 Time slot: 08:30-10:10 a.m.

Contents

1. Lecture 9
2. Directional control valve

Course Instructor: Dr. A. Siddharthan

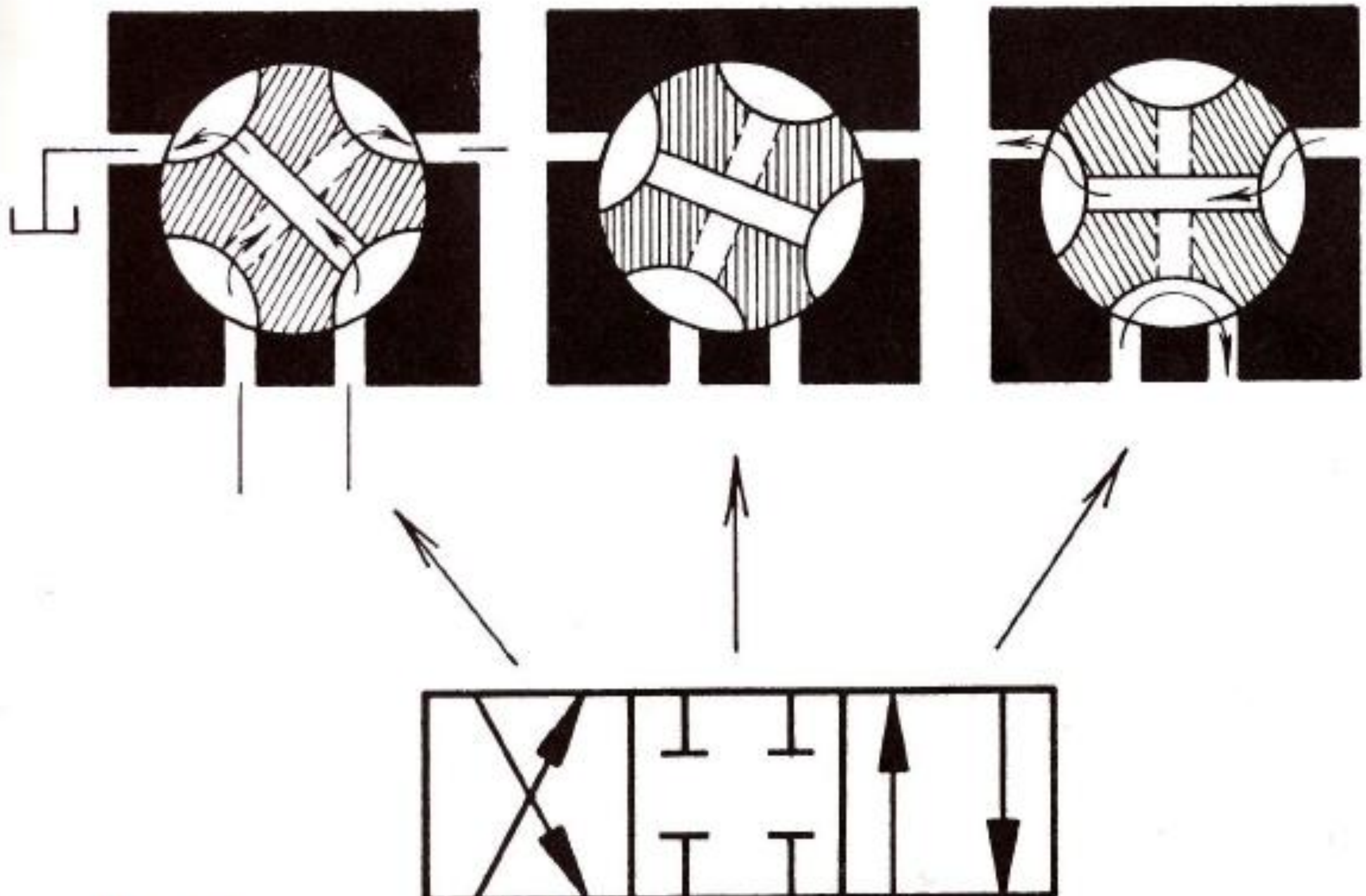


Fig 21 Rotary plug valve mechanism

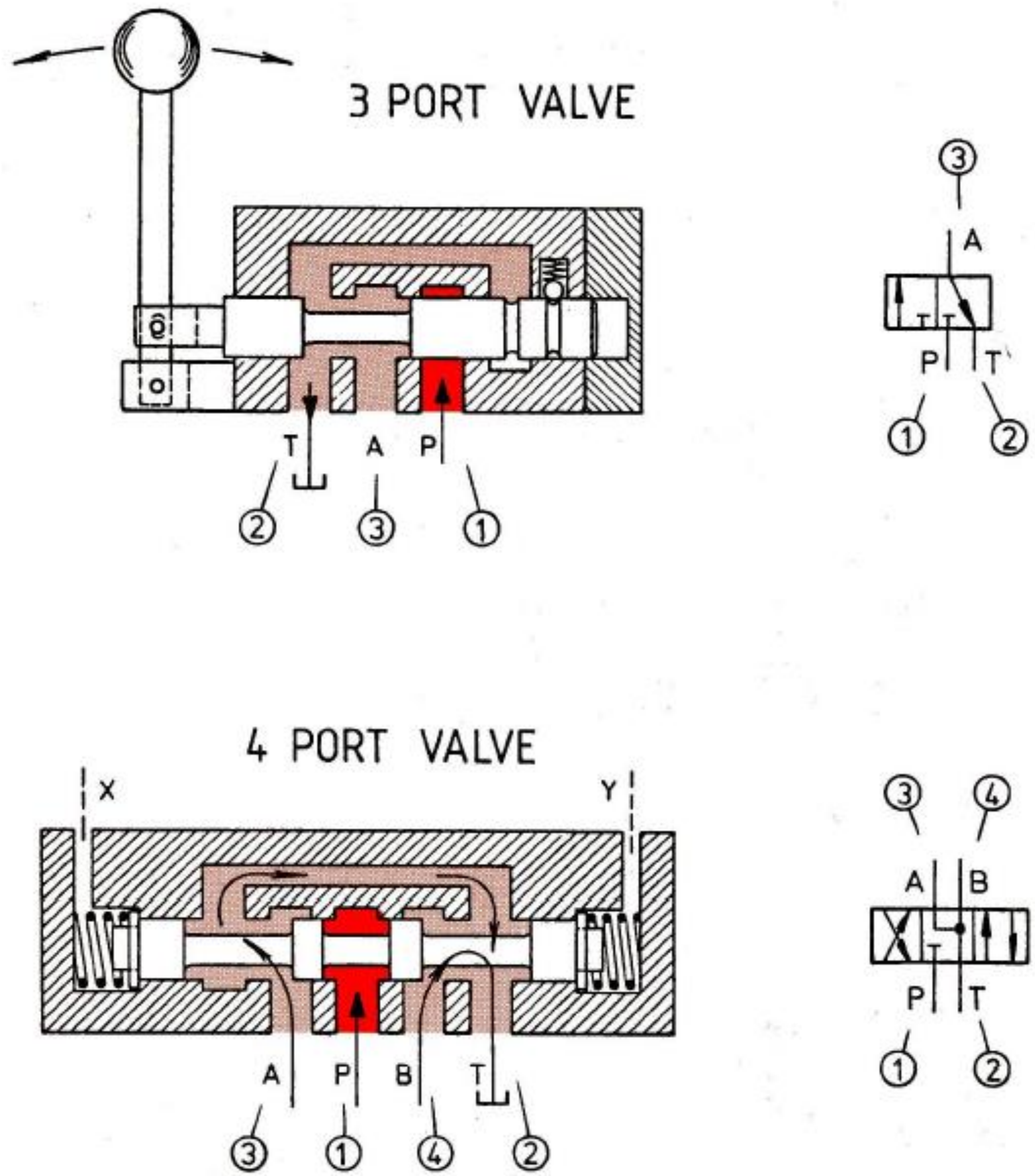
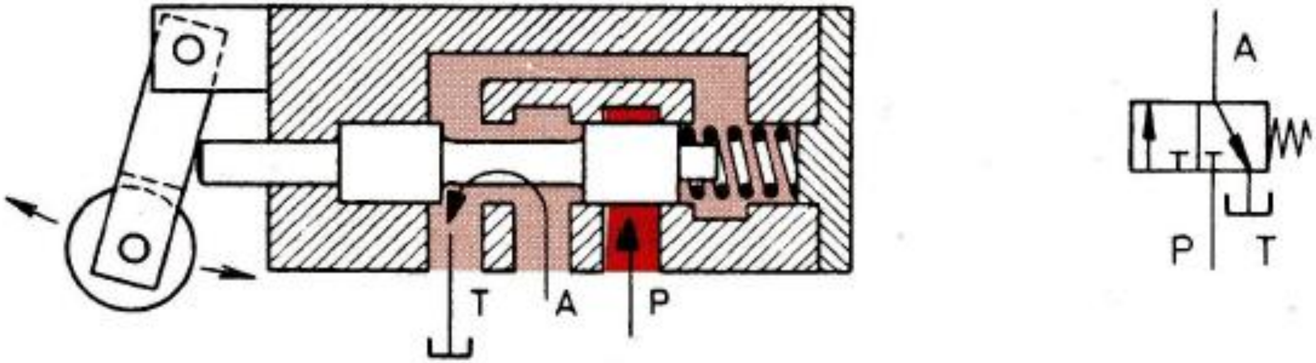


Fig. 20 Valve ports.

NORMALLY CLOSED VALVE



NORMALLY OPEN VALVE

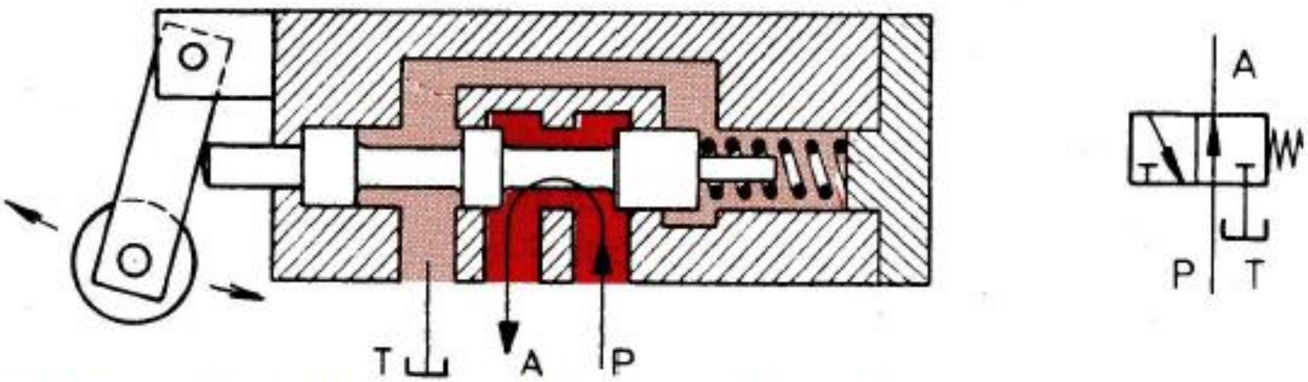
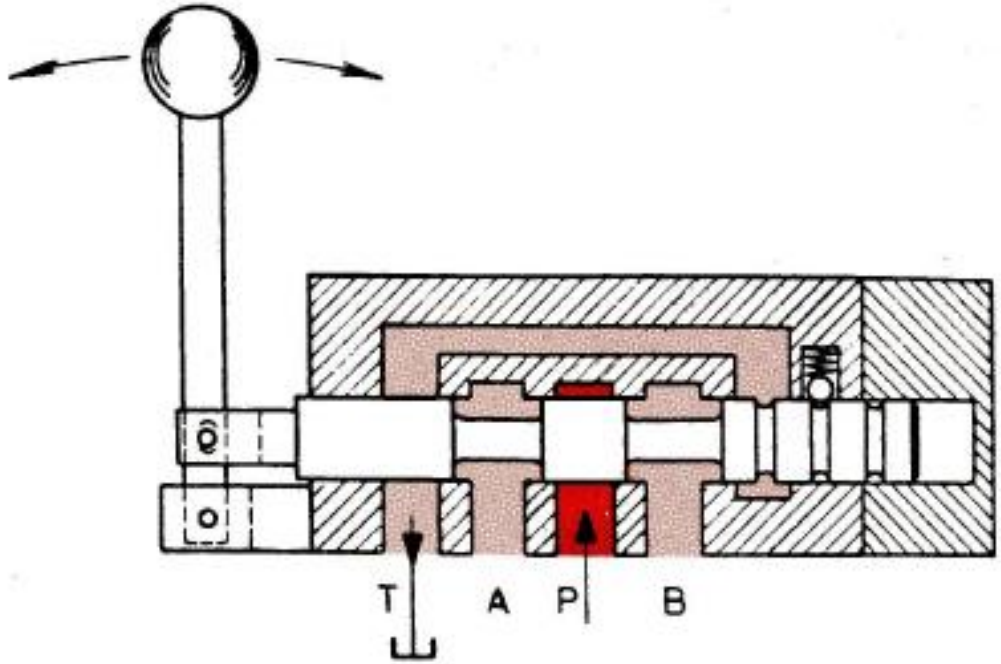
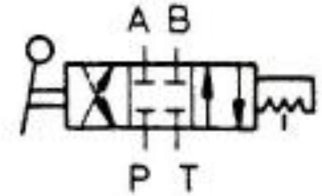


Fig 22 Valve with spring bias (normal position)

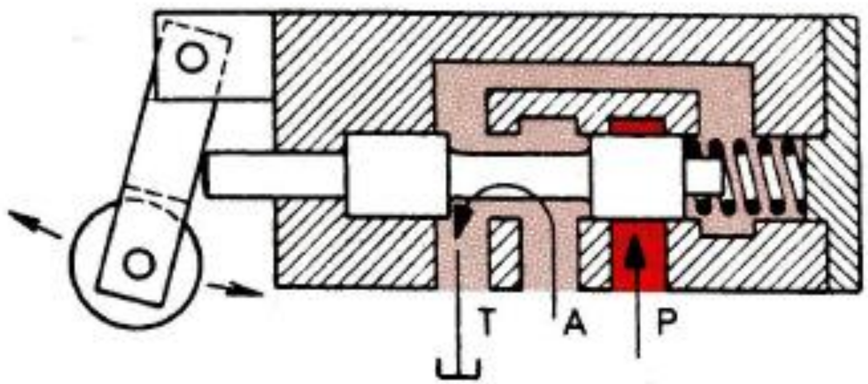
A

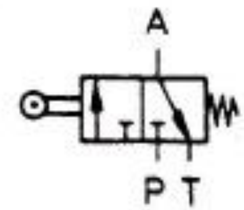




MANUAL ACTUATION

B





MECHANICAL ACTUATION

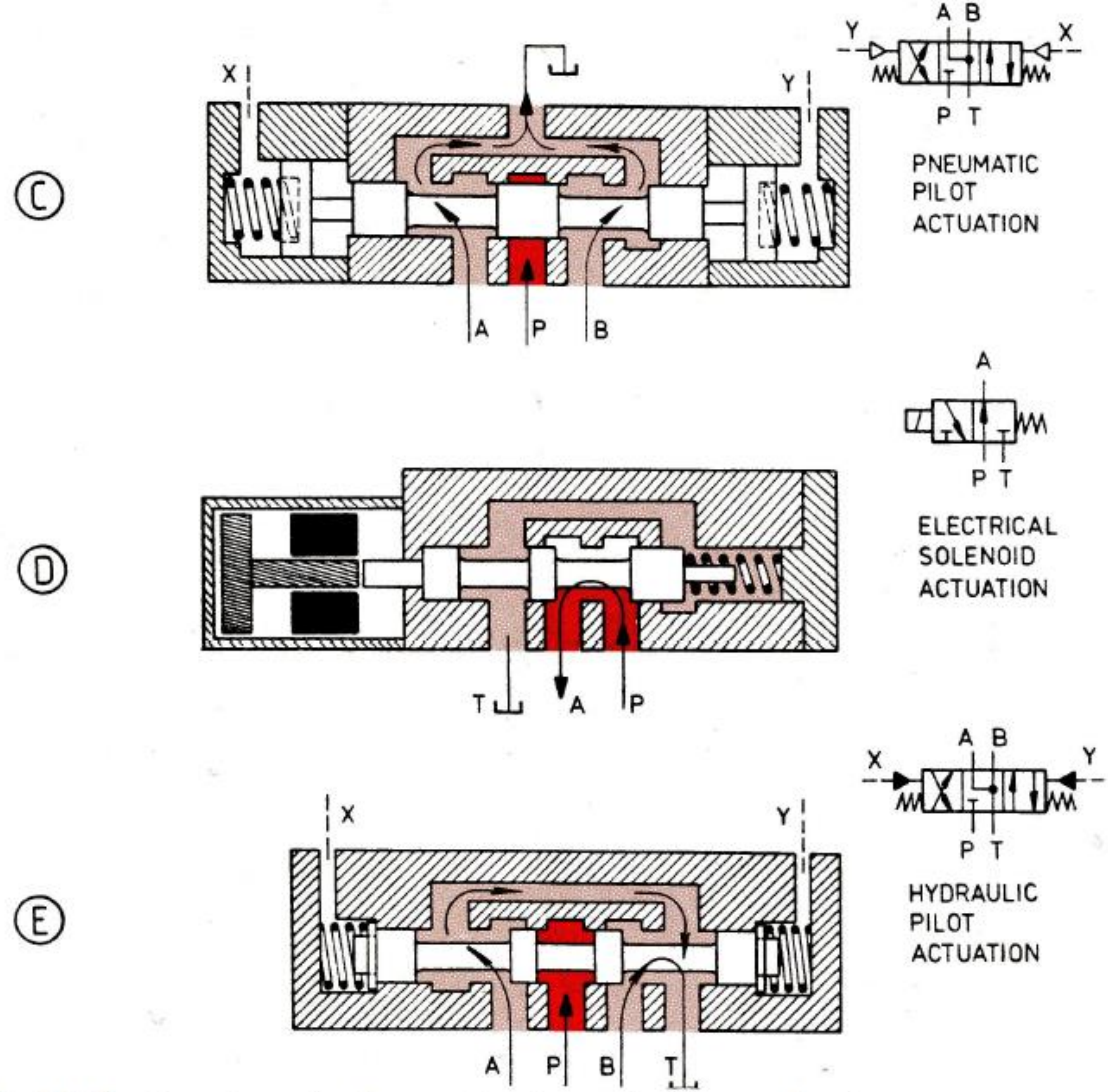


Fig 25 Basic valve actuation methods applied to spool valves

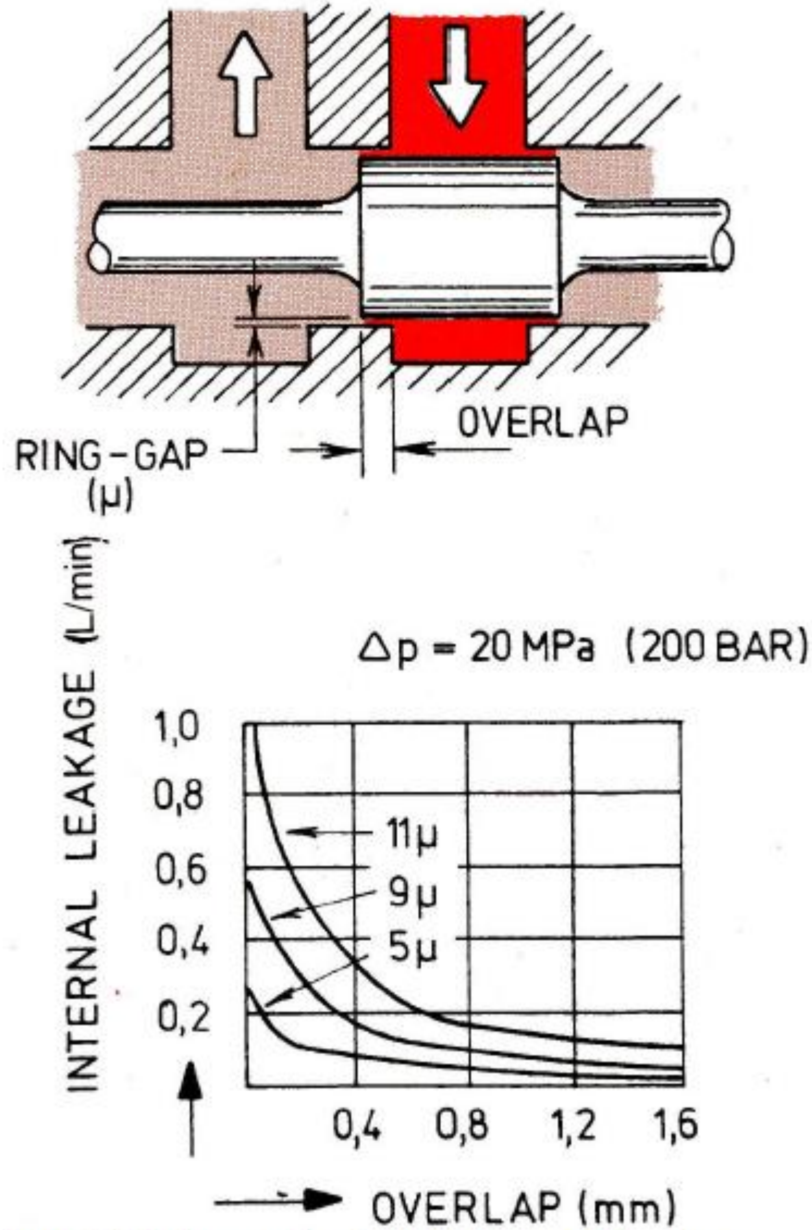


Fig. 26 Internal leakage on spool valves.

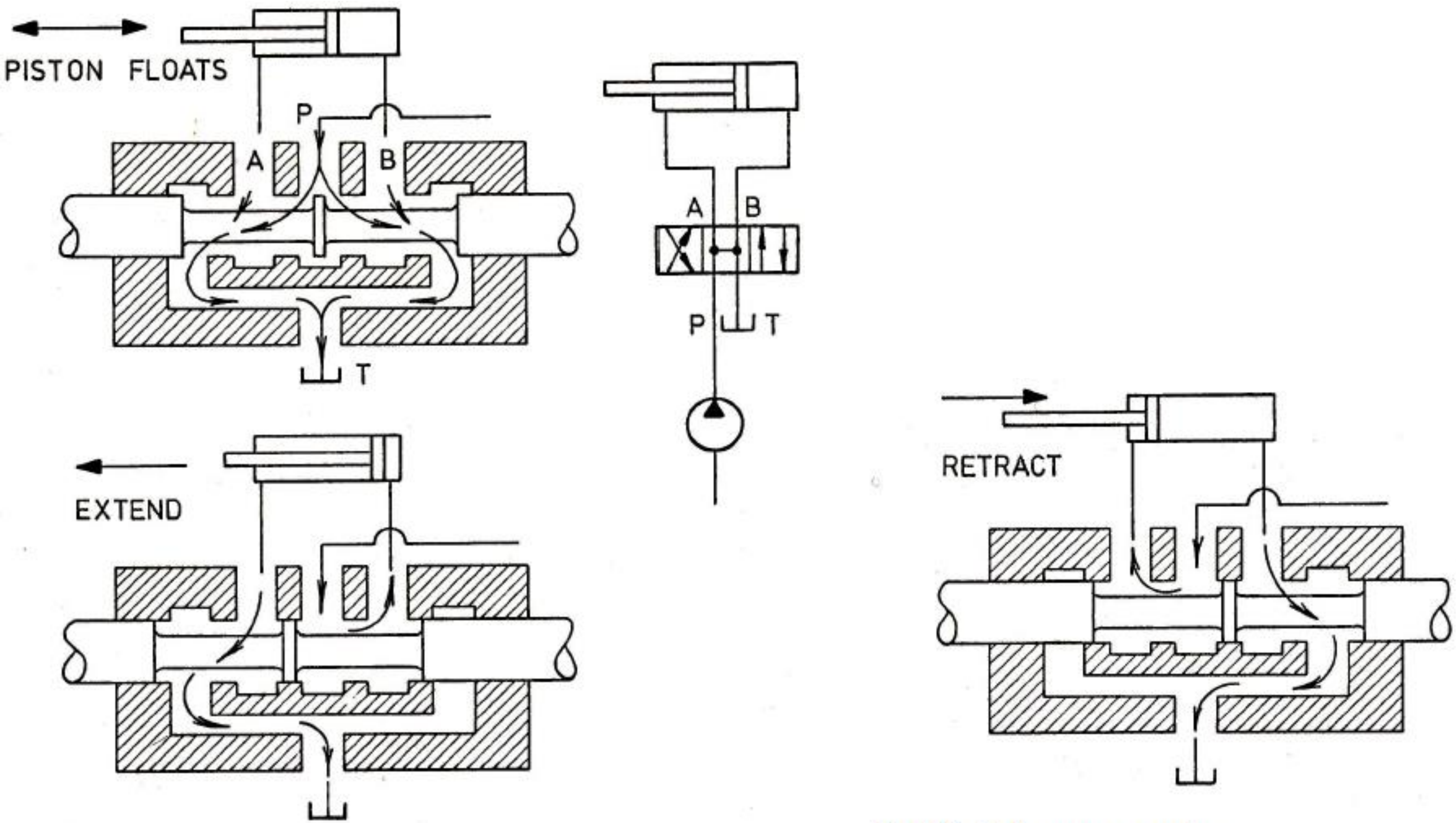
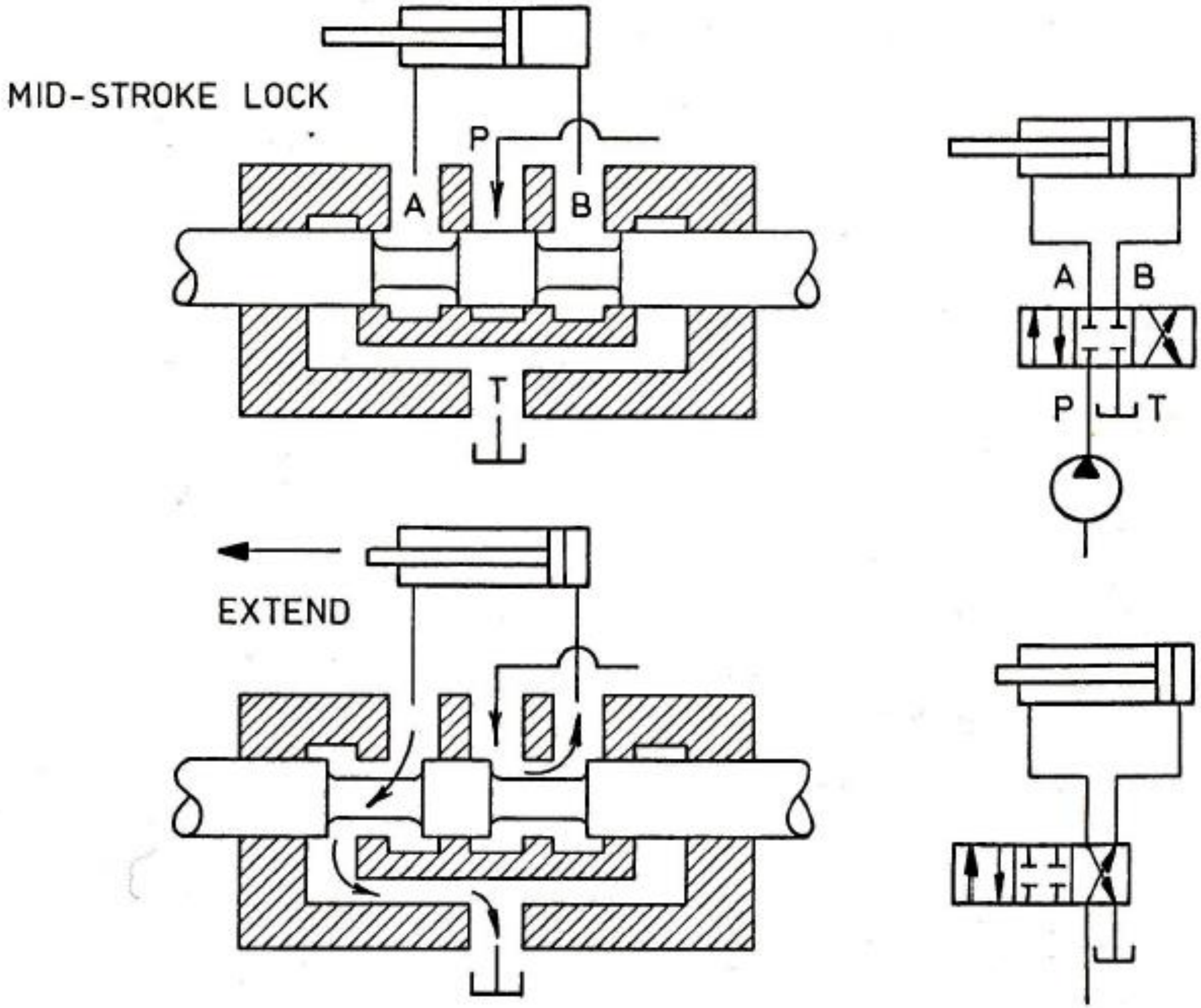


Fig 27 Fully open centre



MID-STROKE LOCK
P
A
B
T
A
B
P
T
EXTEND

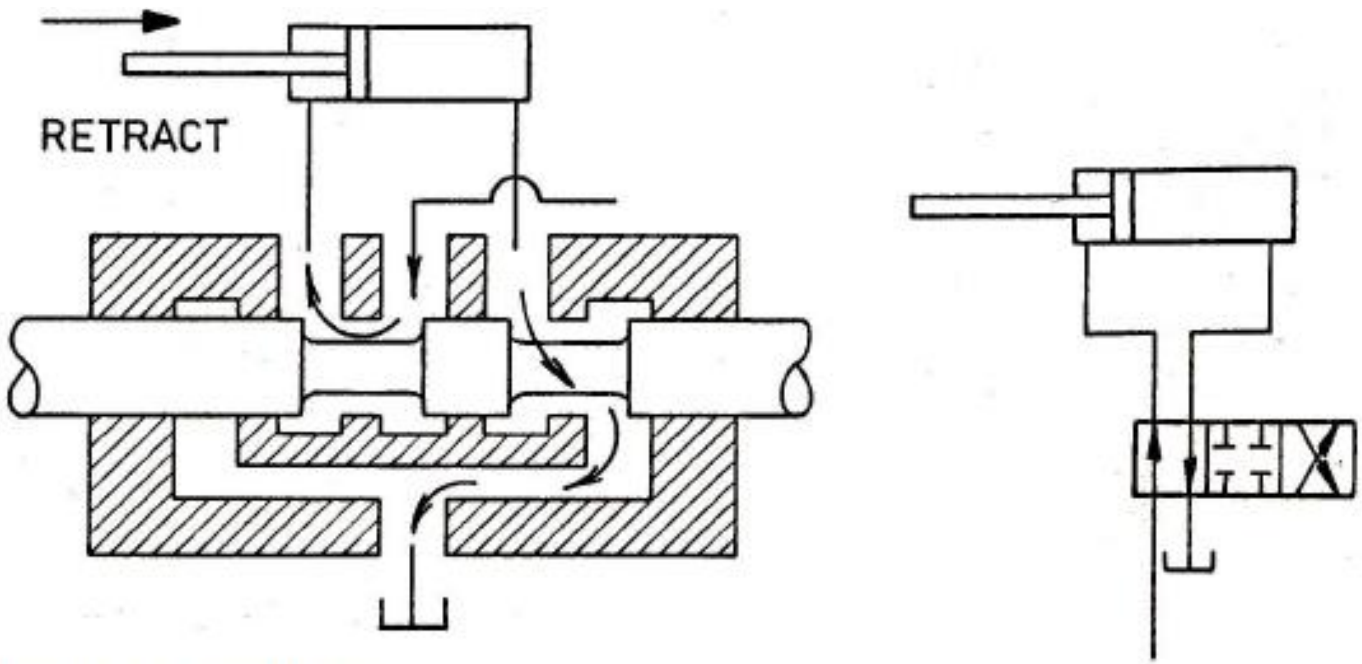
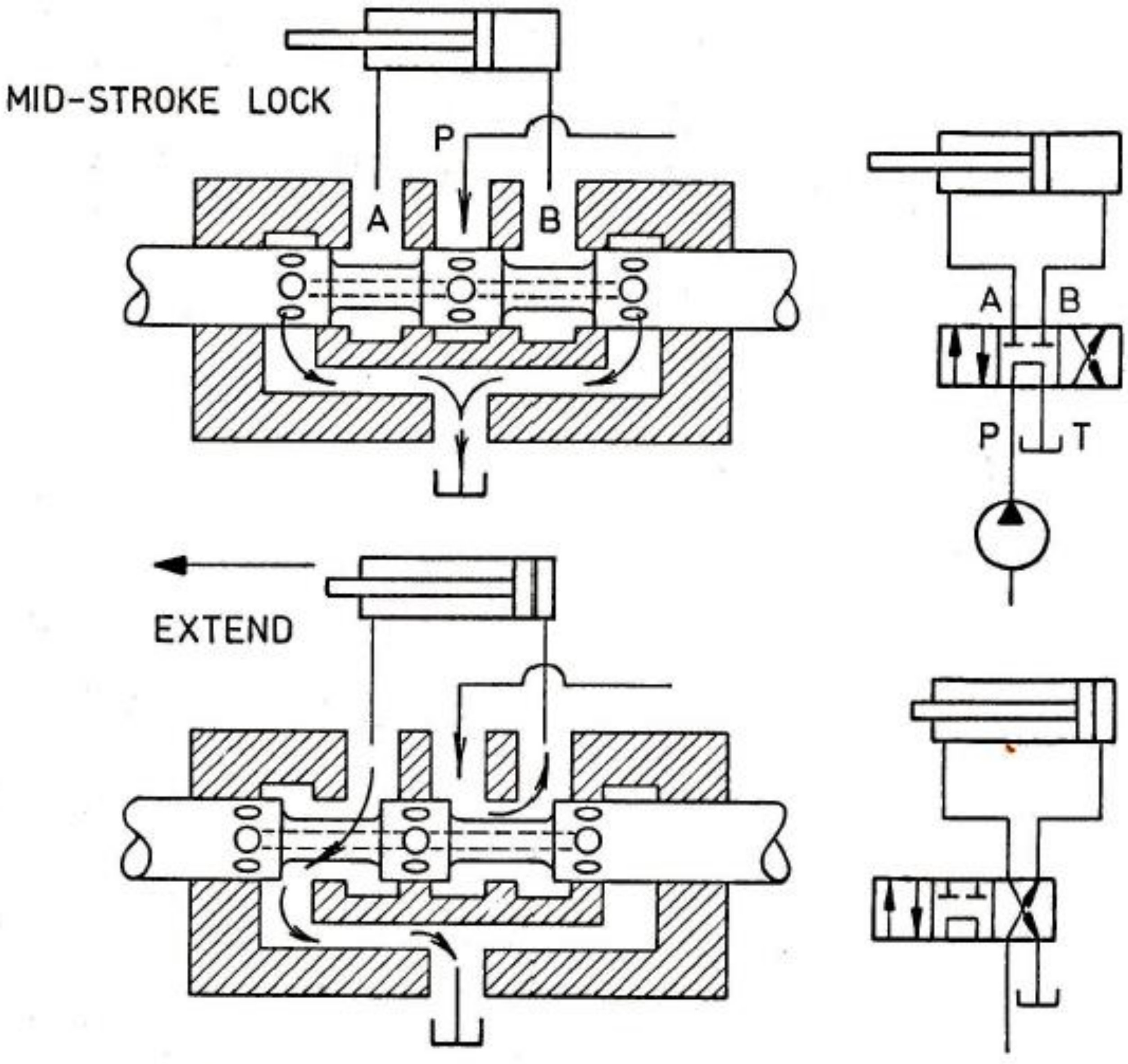


Fig 28 Fully closed centre



MID-STROKE LOCK
P
A
B
A
B
P
T
EXTEND

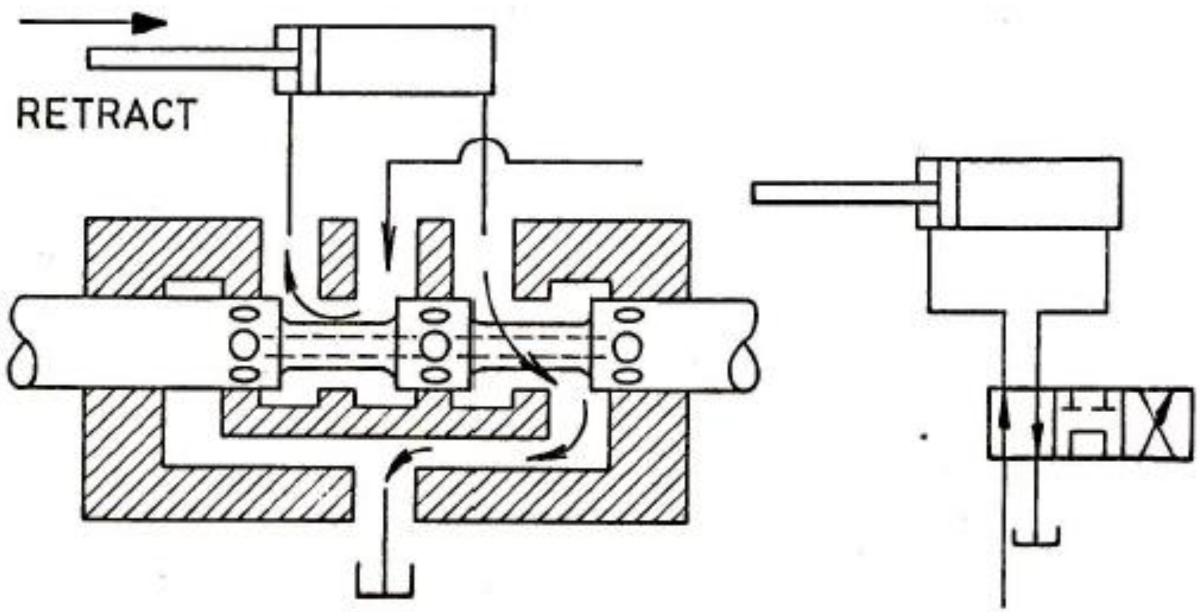


Fig 29 Tandem centre

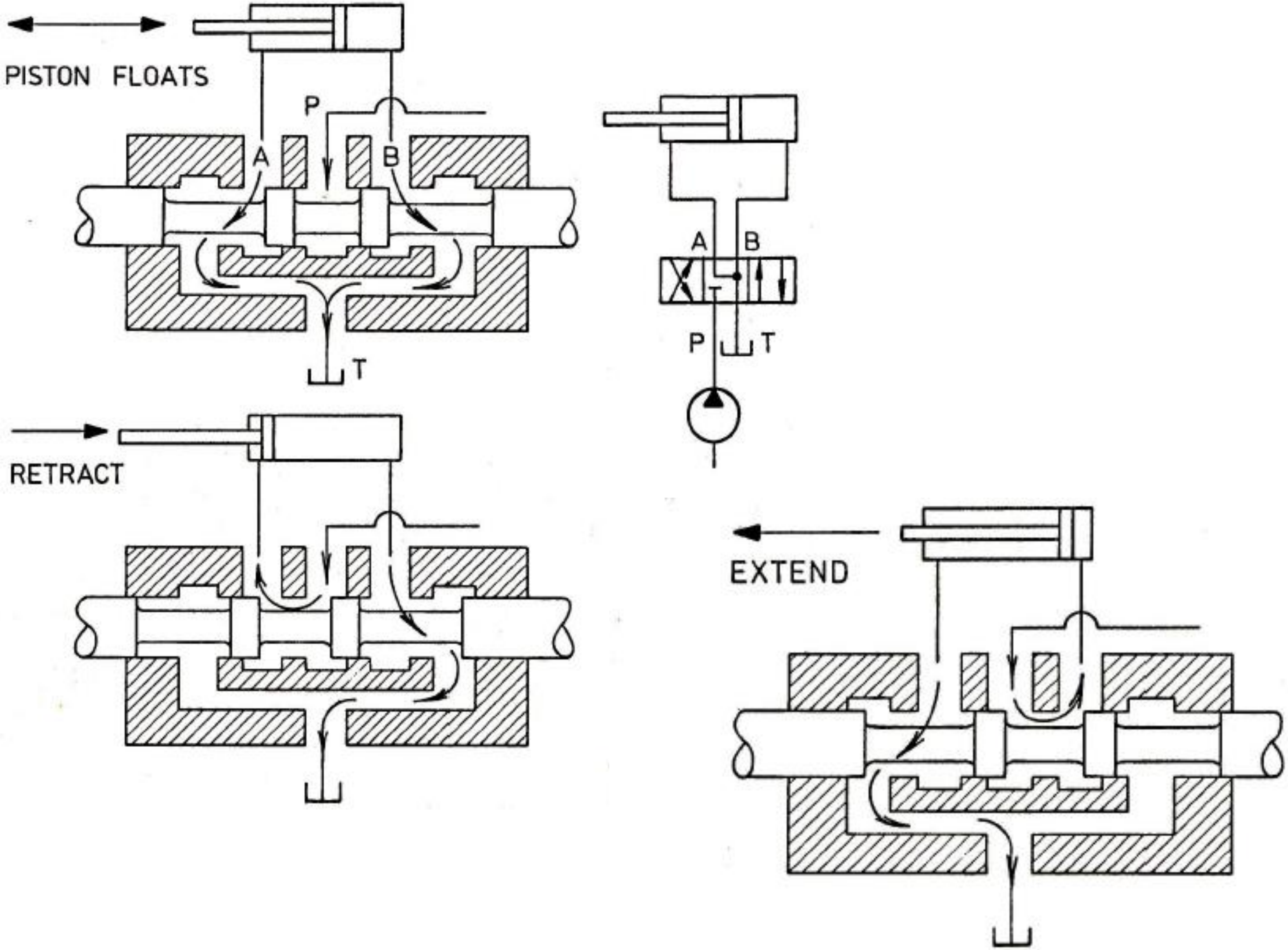


Fig 30 Float centre

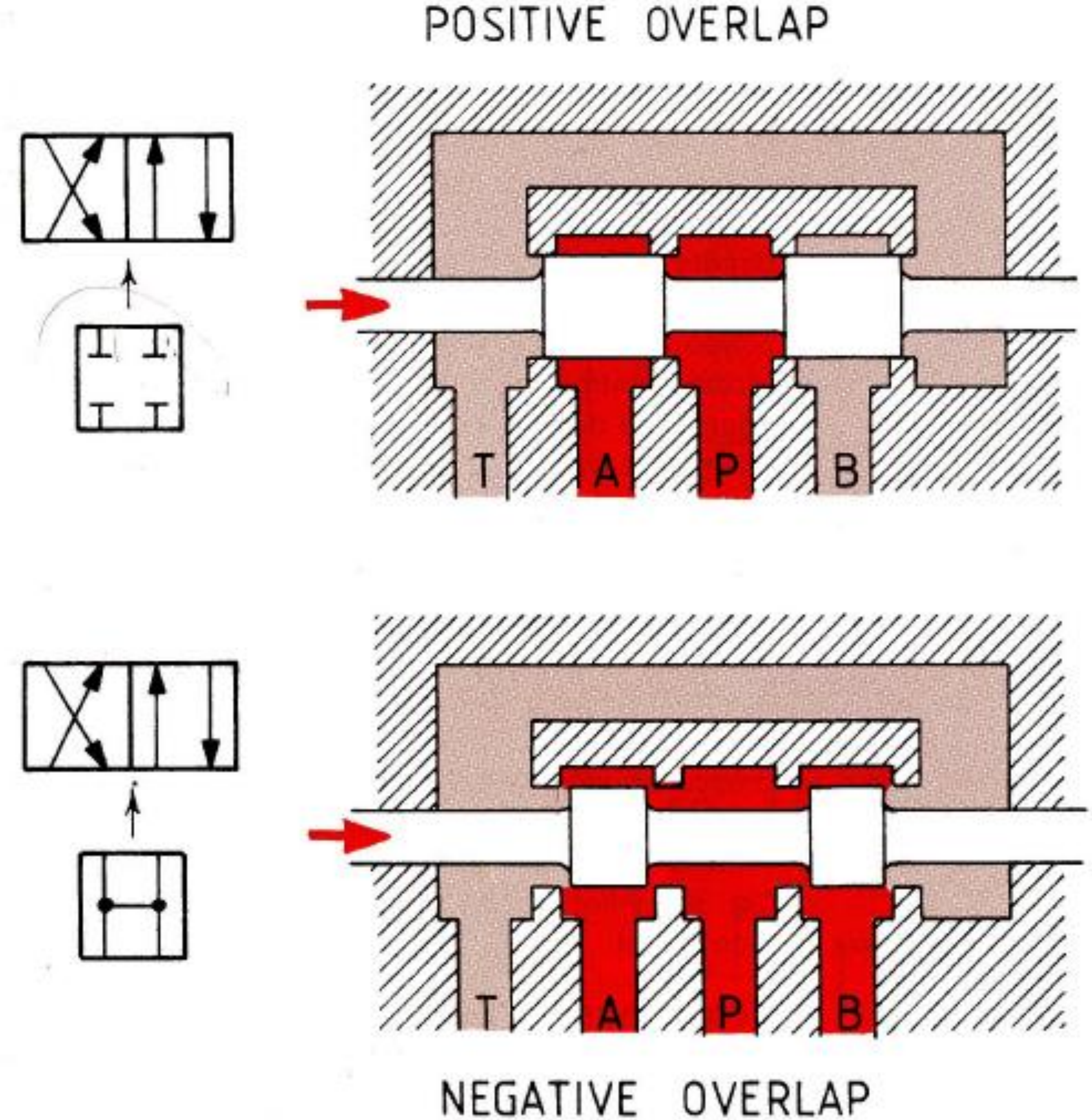


Fig. 31 Different spools for positive or negative overlap.

The fully closed centre crossover is achieved with a positive spool overlap (fig. 31). For a short period all valve ports are sealed against each other. Thus system pressure on the actuators is prevented from collapsing during the crossover. The fully closed centre crossover may however cause undesirable pressure peaks in the system which vary with the amount of fluid flow and switching time.

The fully open centre crossover is achieved with negative spool overlap (also called underlap) (fig. 31). For a short period all valve ports are connected to each other. This results in smooth, pressure-peak free switching during the crossover. However, undesirable actuator movements may occur with certain load conditions.

The flow gain K_q is the slope of the approximate line in the figure, which can double its valve near null with negative lap. The magnitude of K_q is the most important parameter of a valve and often also of any system incorporating the valve.

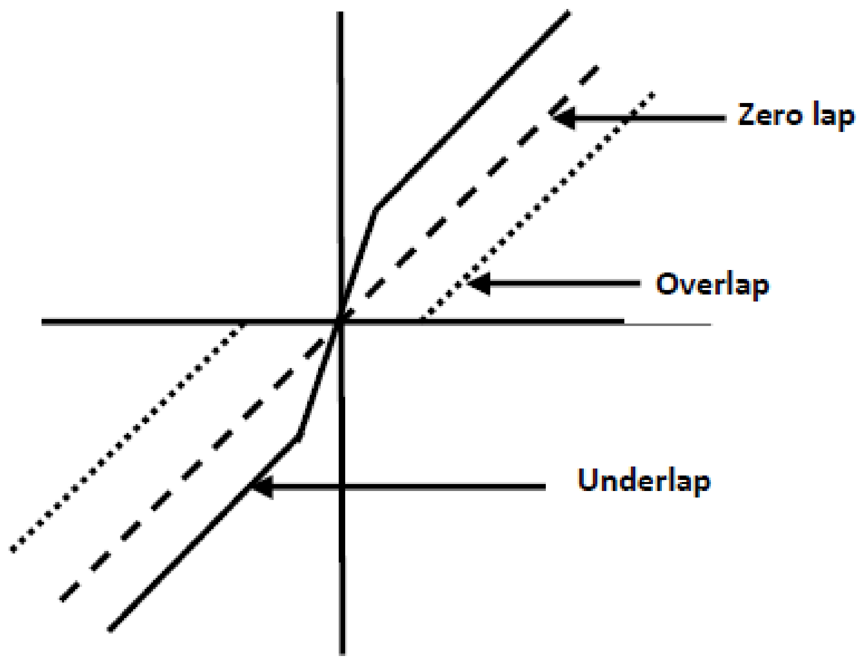


Figure 1.2 Flow rates versus valve displacement for constant pressure drop.

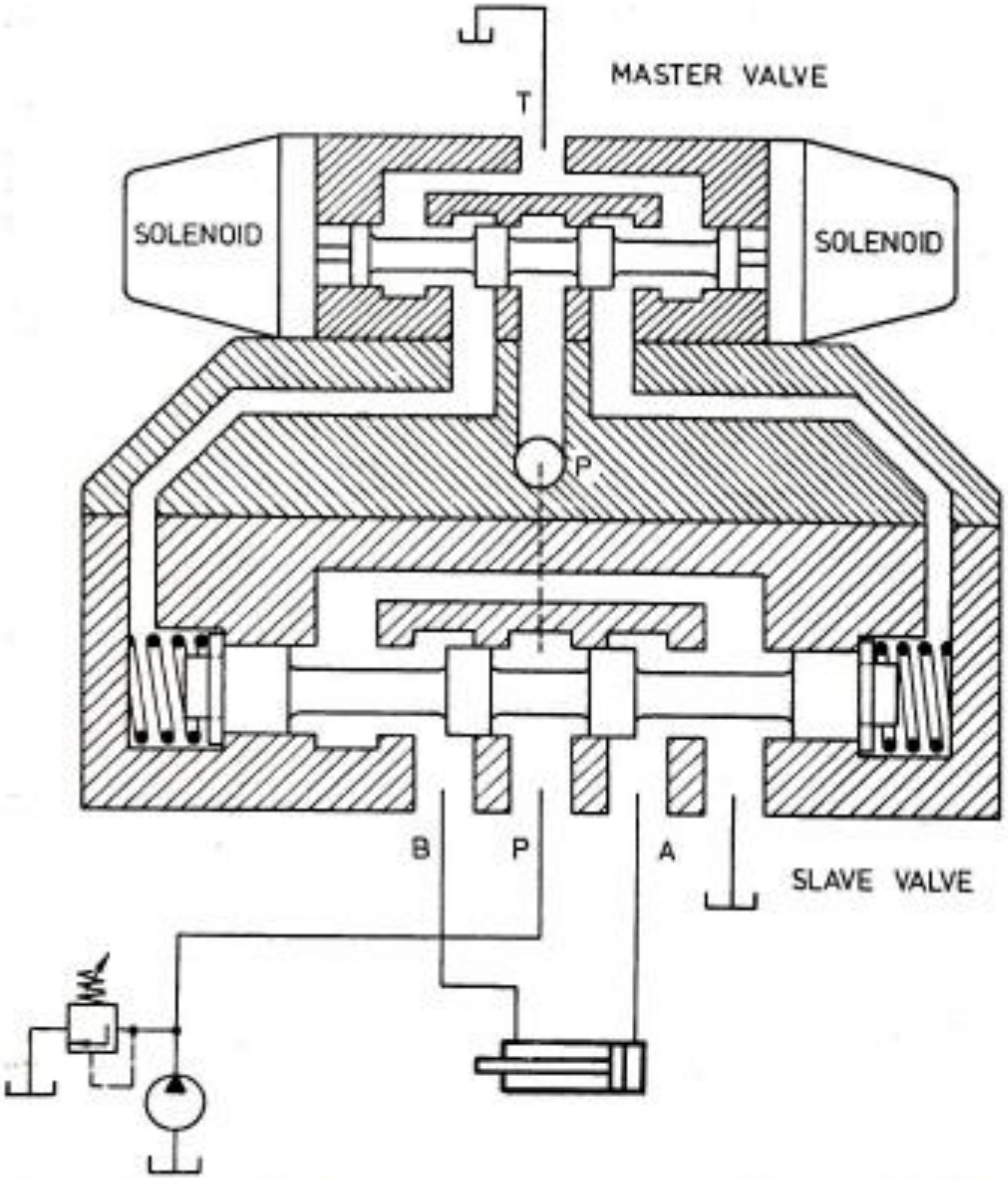


Fig 34 Pilot operated valve: the master valve sits on top of the slave valve.

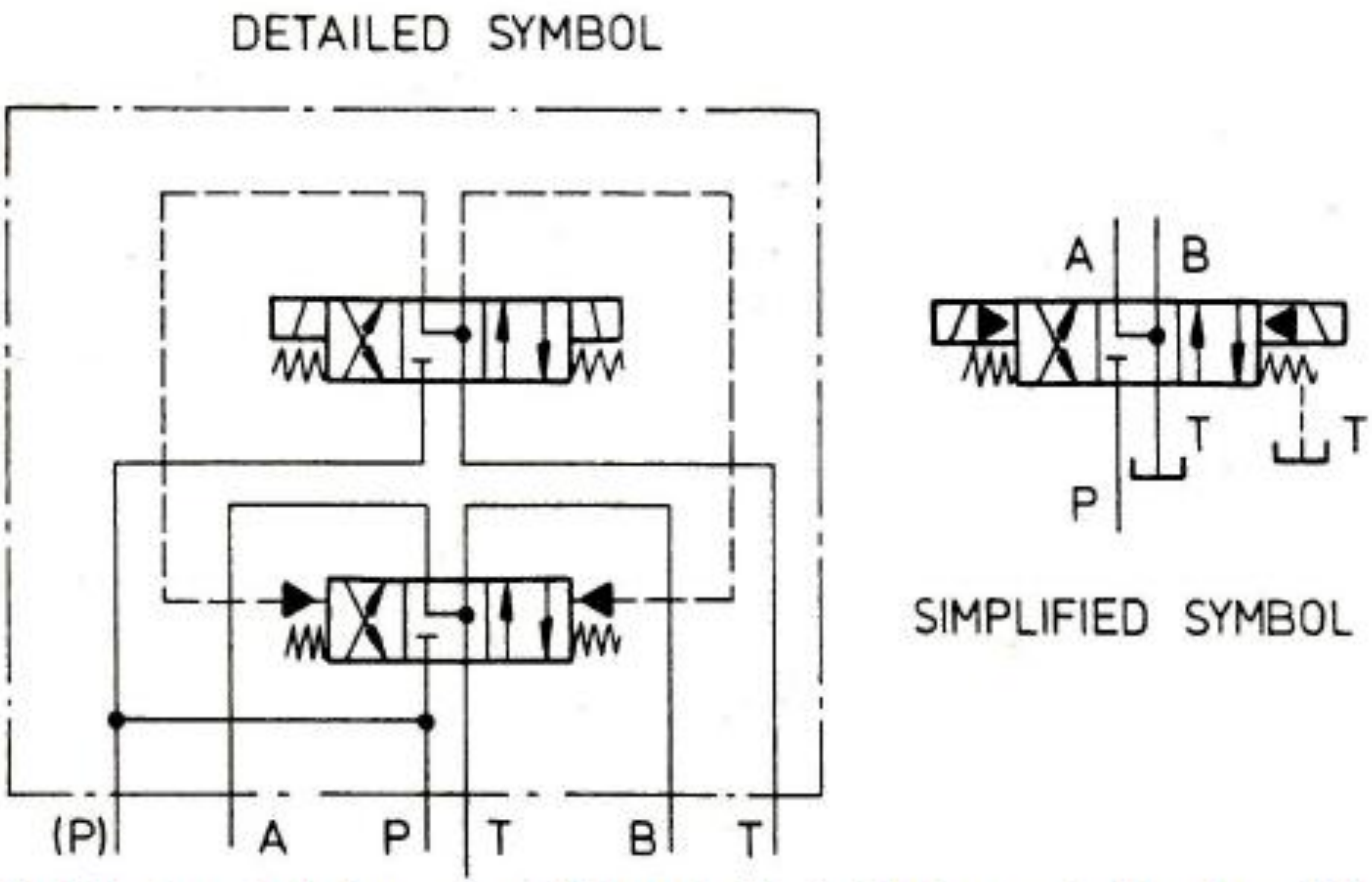


Fig 35 Symbols for solenoid controlled pilot operated valve. (Also available with pneumatically pilot operated master valve.

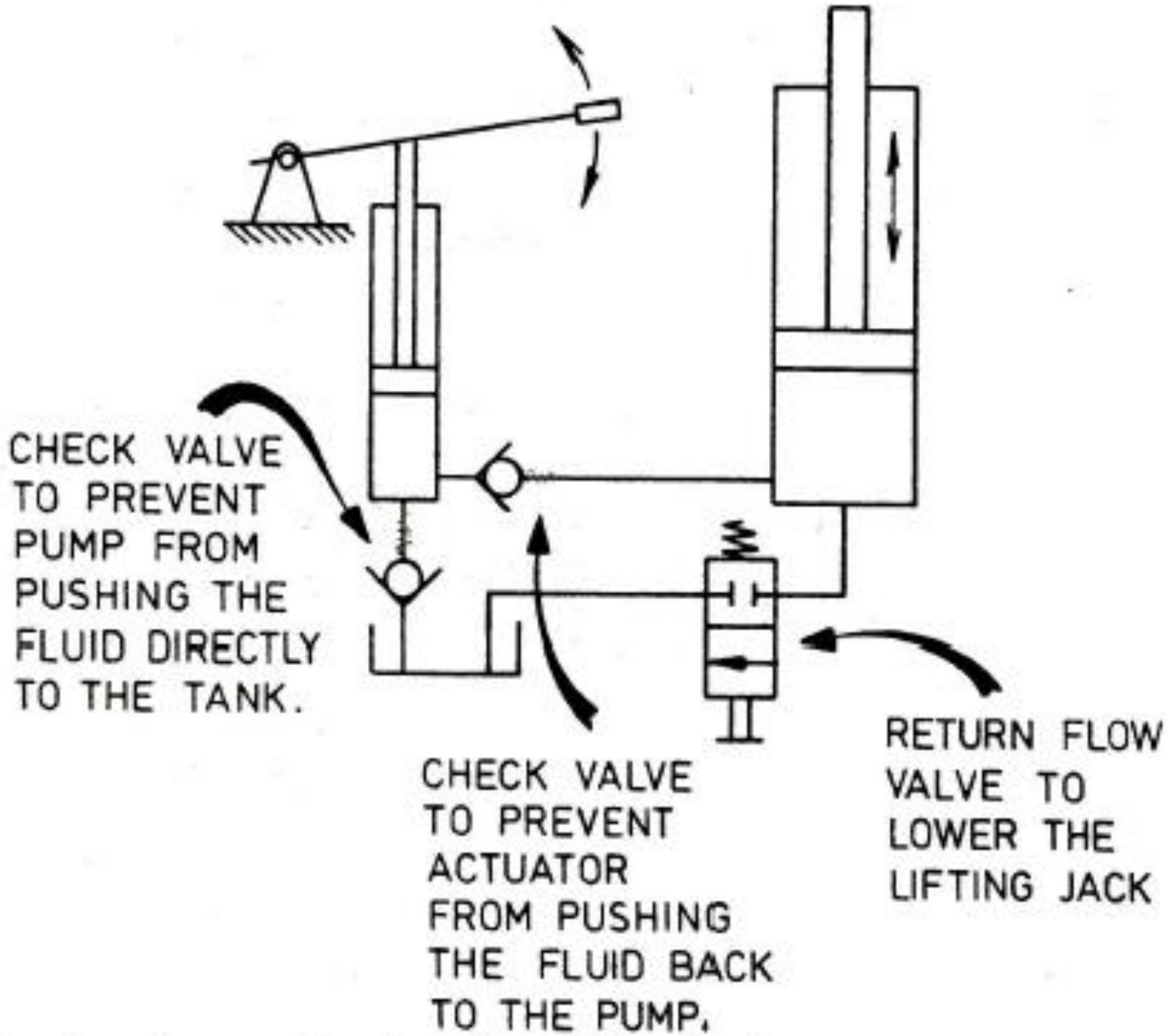


Fig. 41 Typical check valve application in a lifting jack.

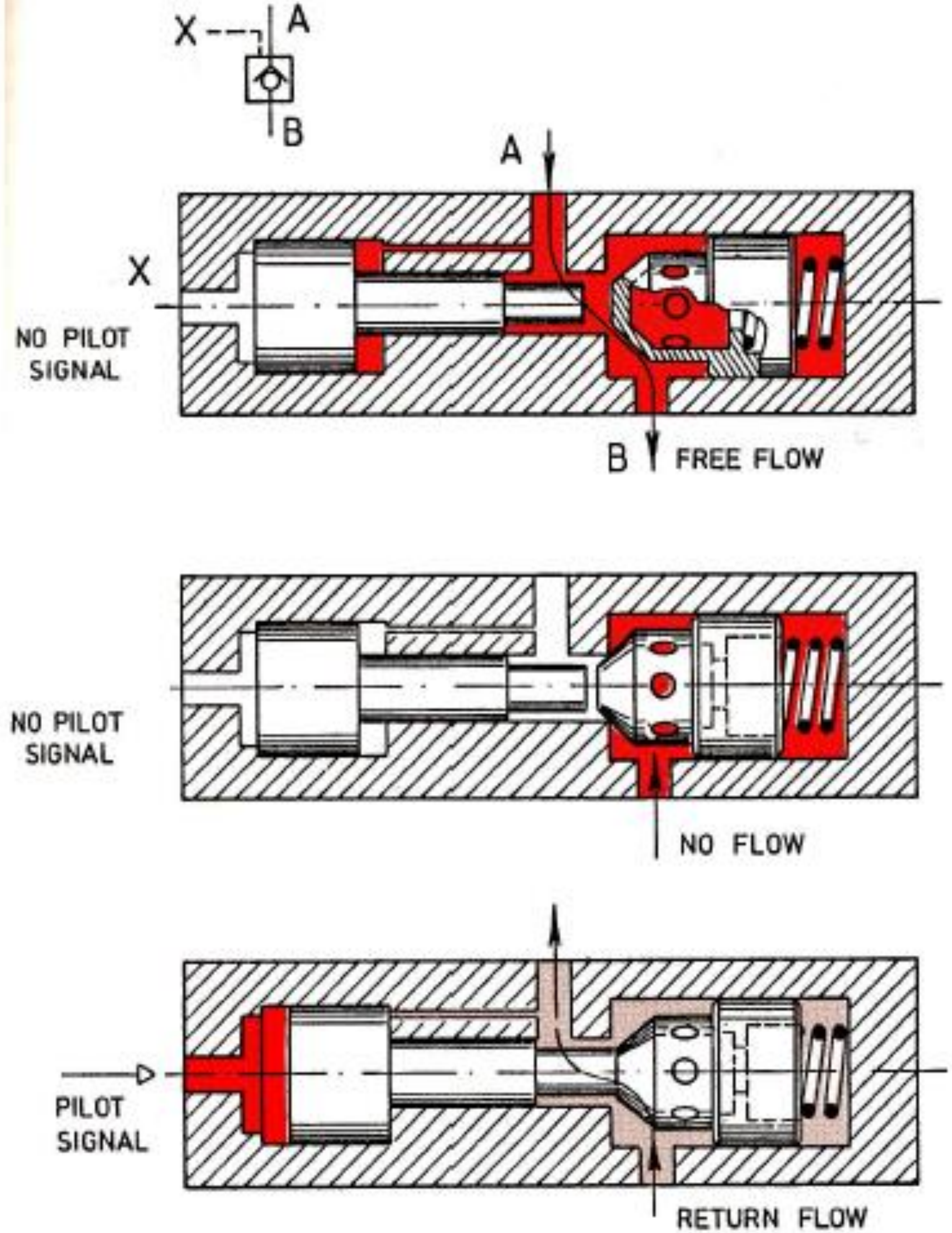


Fig 42 Pilot operated check valve

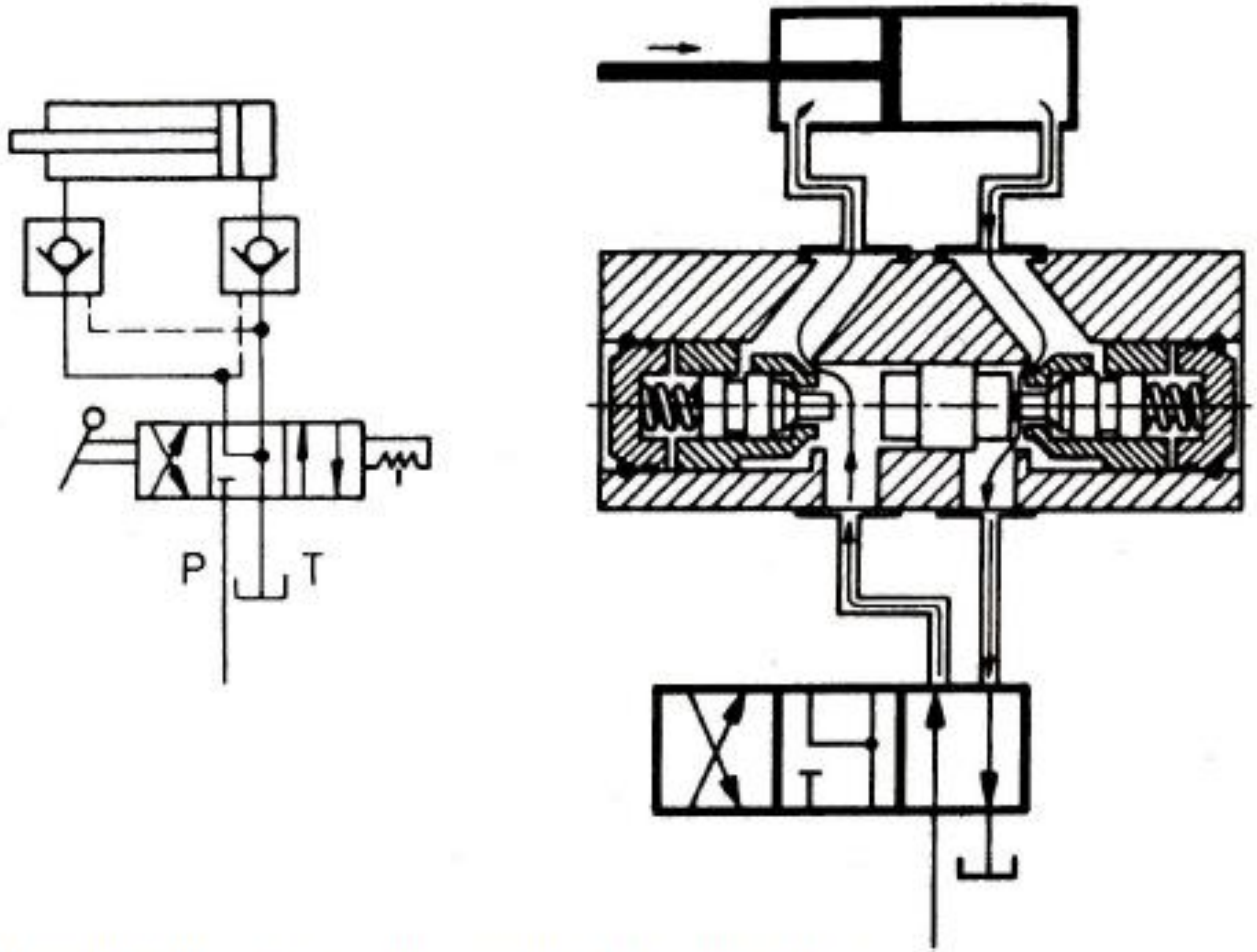


Fig 43 Typical application for pilot operated check valve

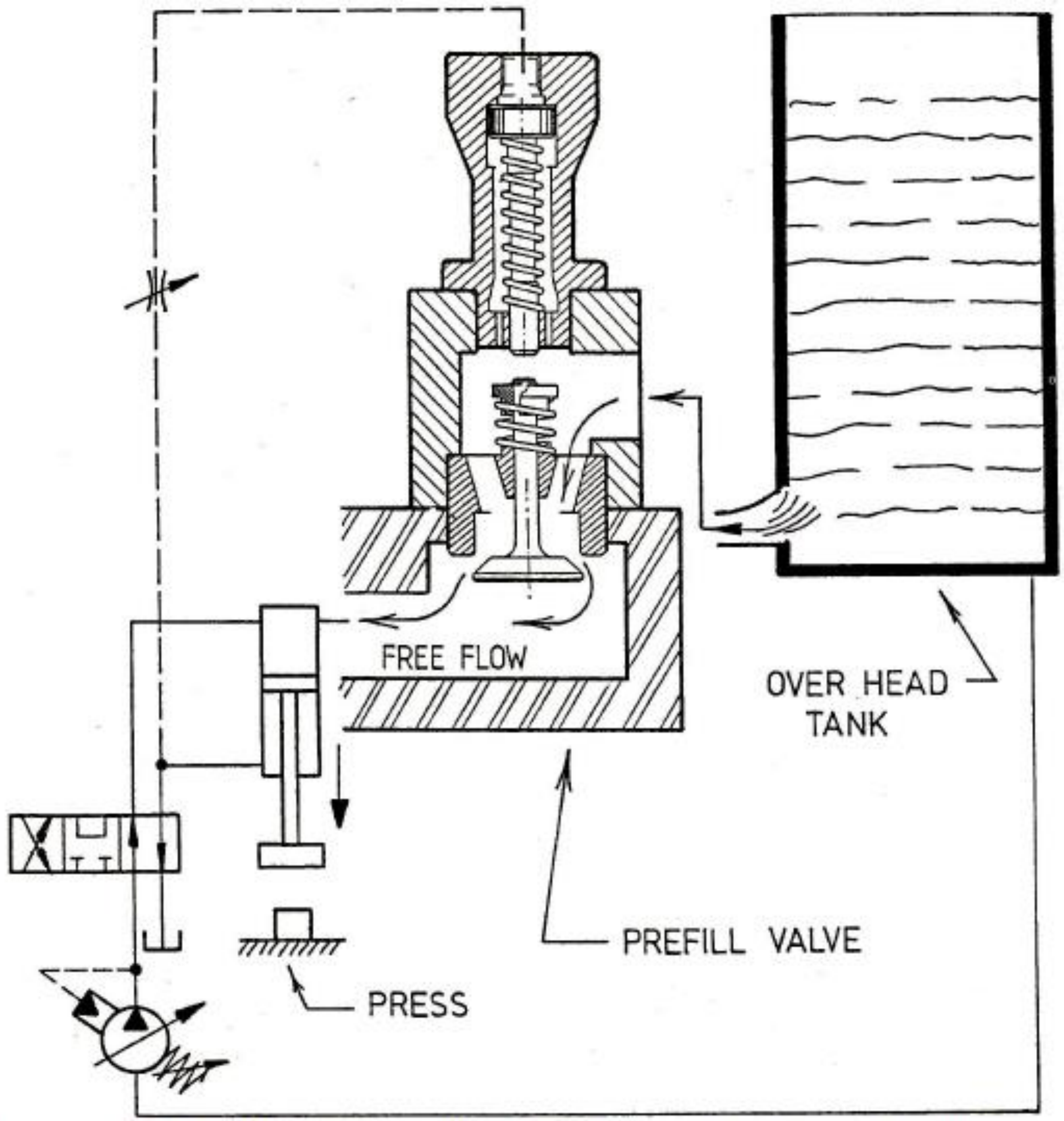


Fig 44 Pilot operated check valve used as pre-fill valve in a press circuit

ME 7553 – Hydraulics and Pneumatics

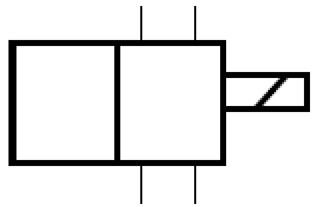
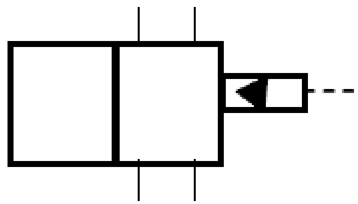
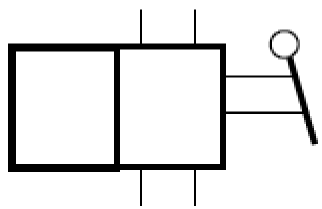
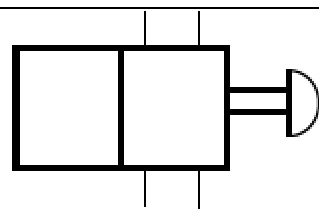
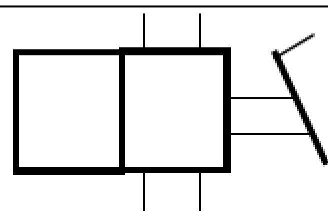
Lecture -11

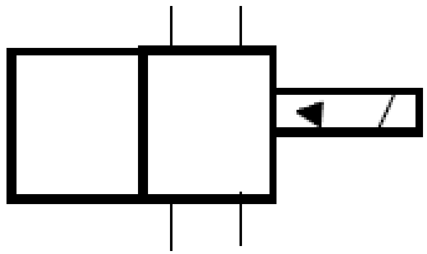
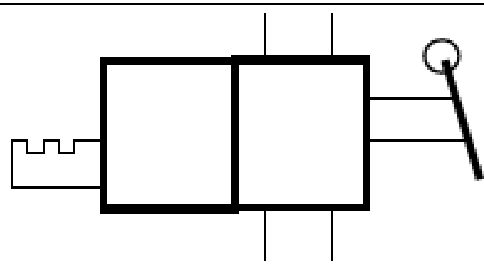
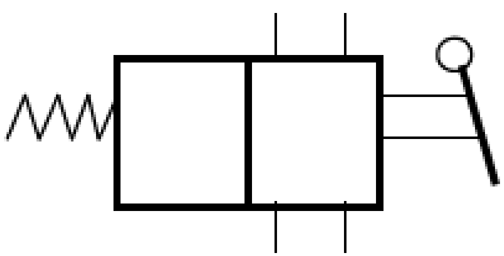
Date: 29-04-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Review of Lecture 10
2. Pressure Control Valve- simple Pressure Relief Valve(PRV) and Compound PRV
3. Unloading Valve

Course Instructor: Dr. A. Siddharthan

	Solenoid operated
	Pilot operated
	Manual operated
	Push button
	Foot operated

	Pilot-operated solenoid
	Two-position detent
	Spring return

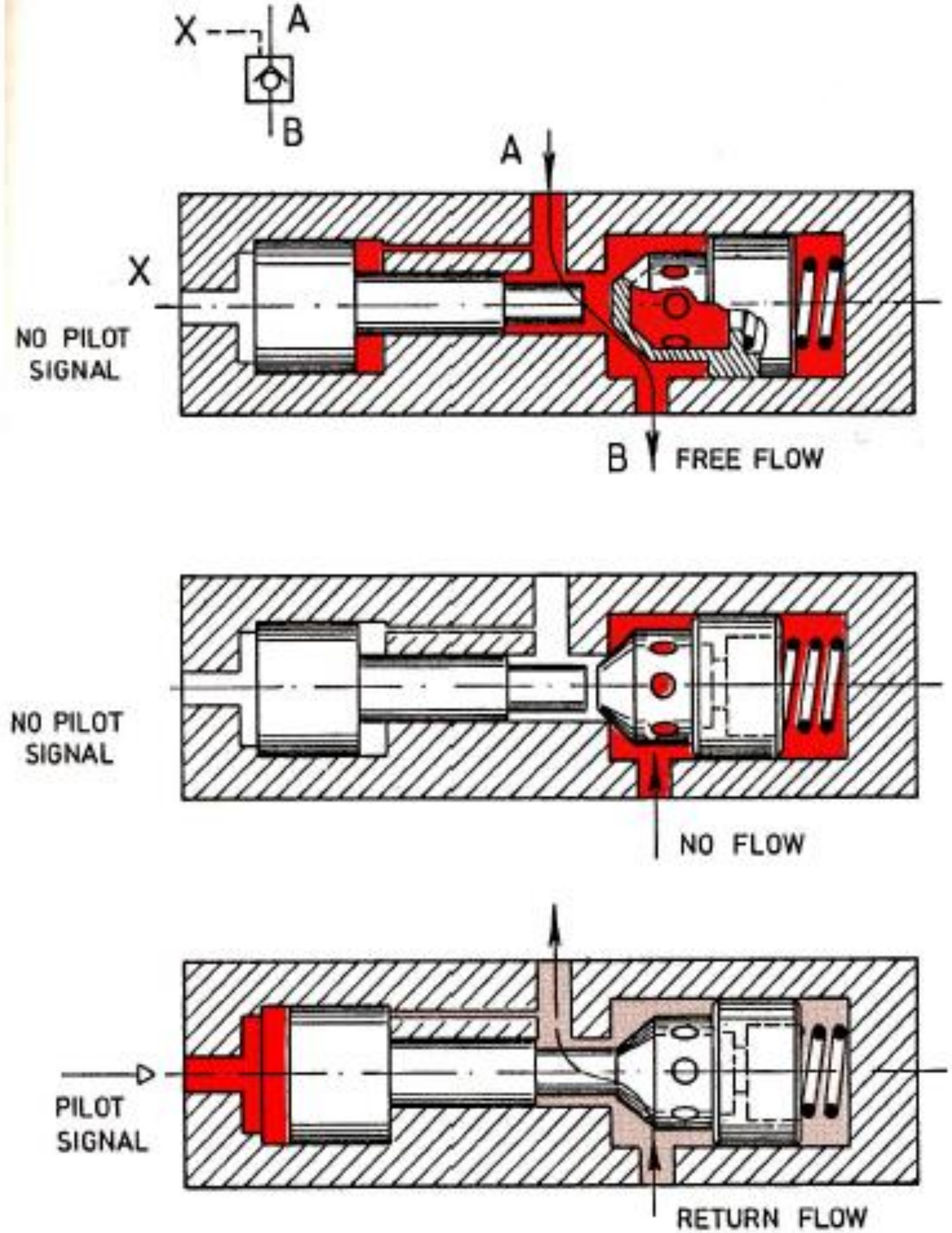


Fig 42 Pilot operated check valve

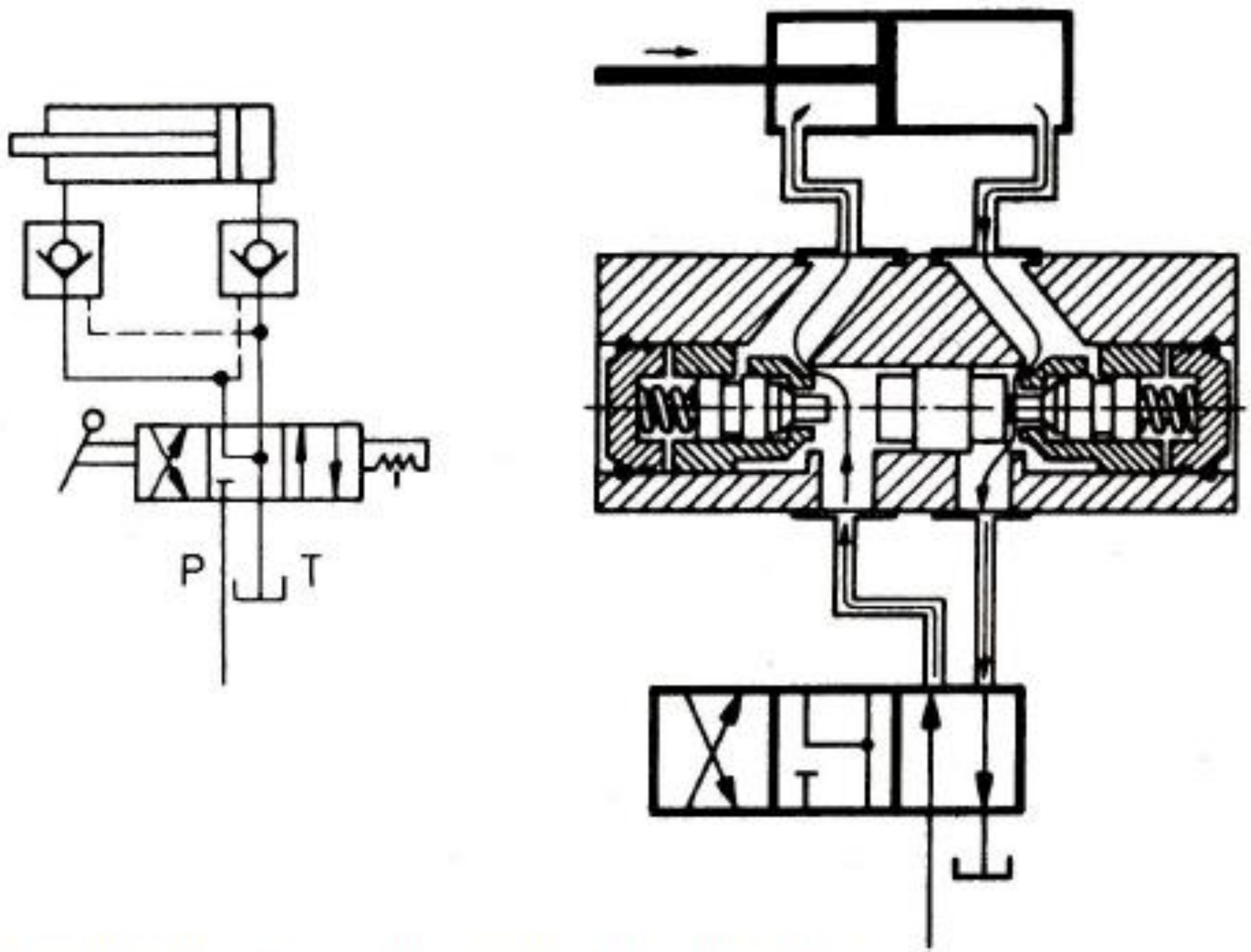


Fig 43 Typical application for pilot operated check valve

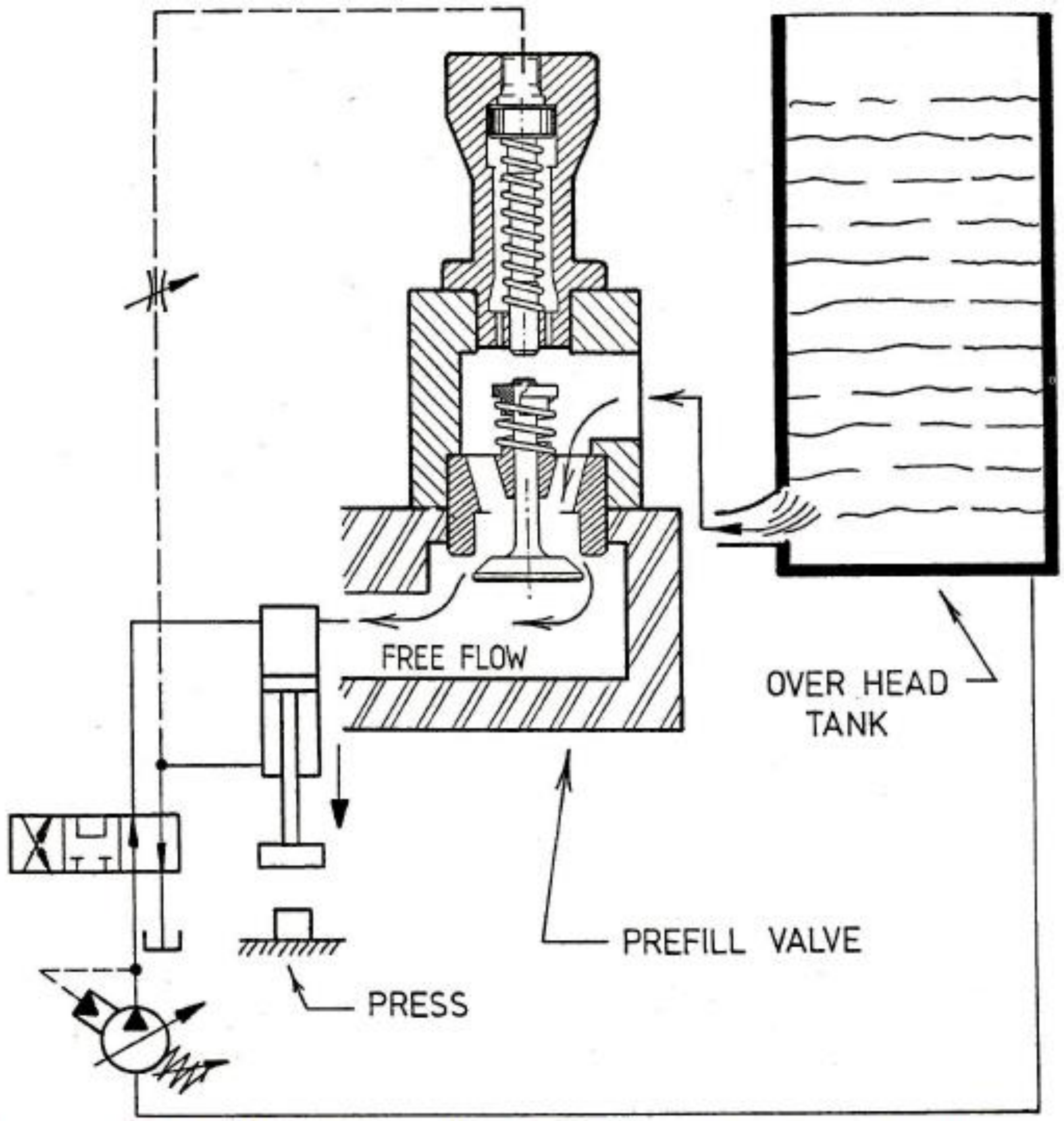


Fig 44 Pilot operated check valve used as pre-fill valve in a press circuit

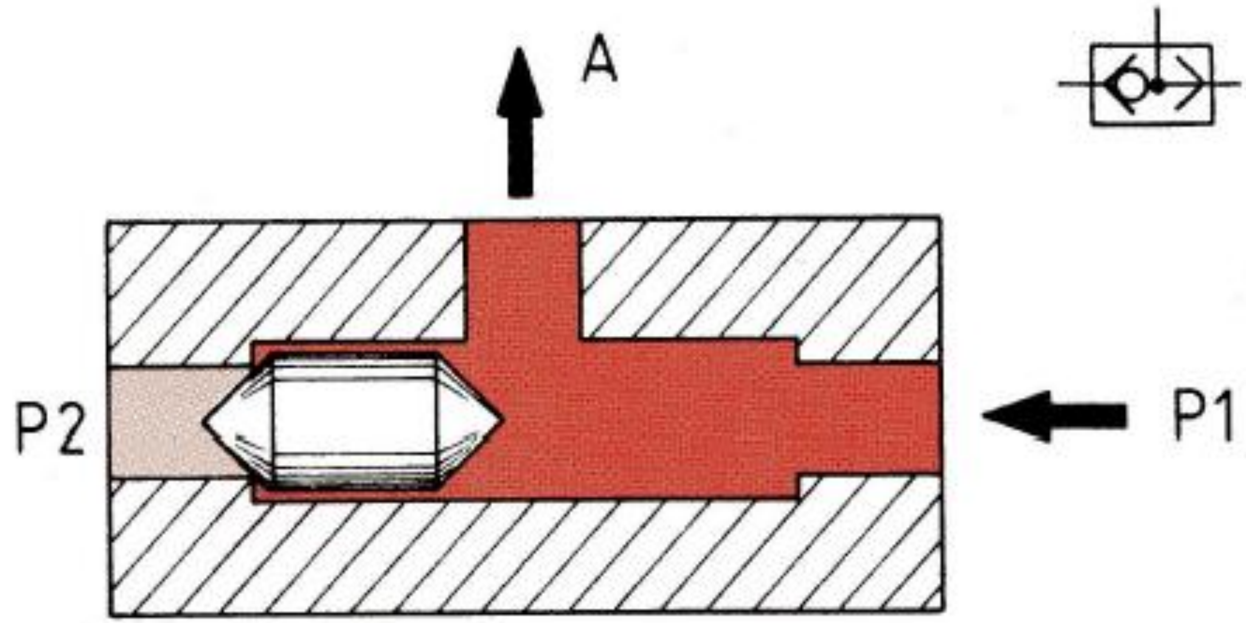


Fig 45 "OR" function valve (shuttle valve)

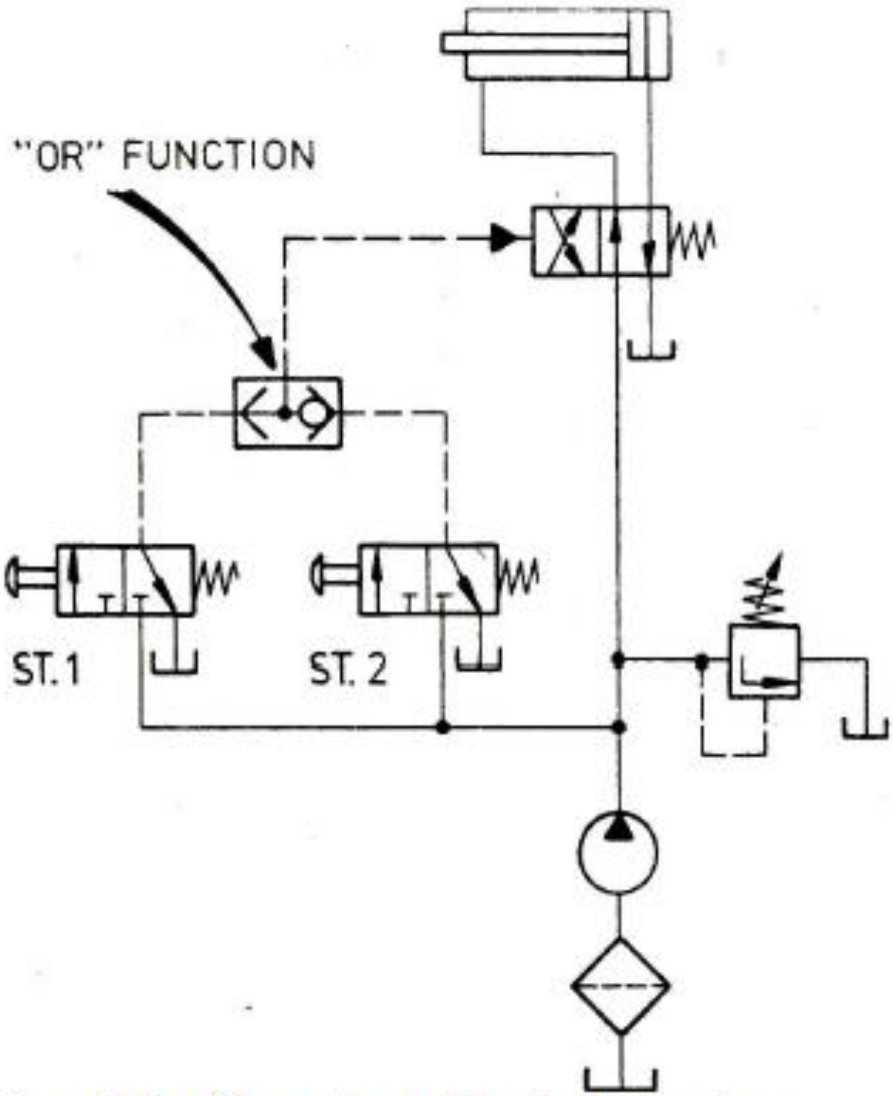


Fig. 46 "or" function with shuttle valve.

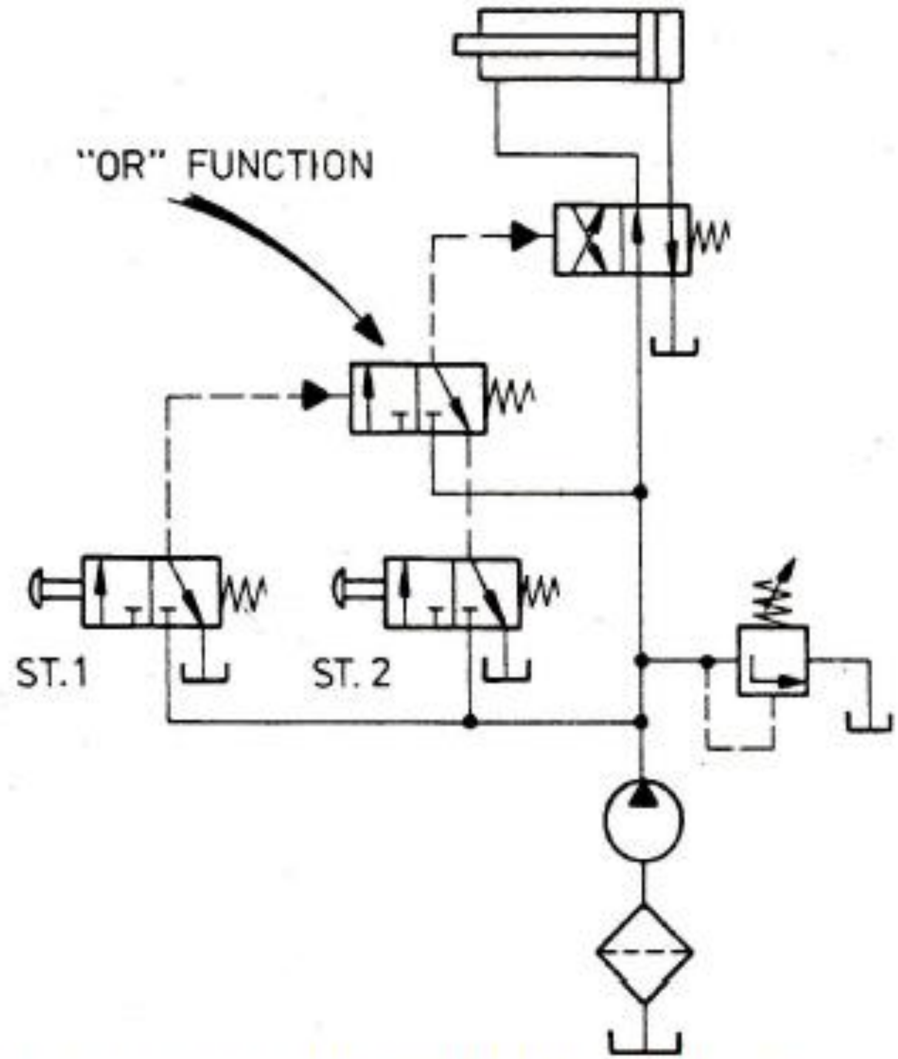


Fig. 47 "or" function with two position, three port directional control valve.

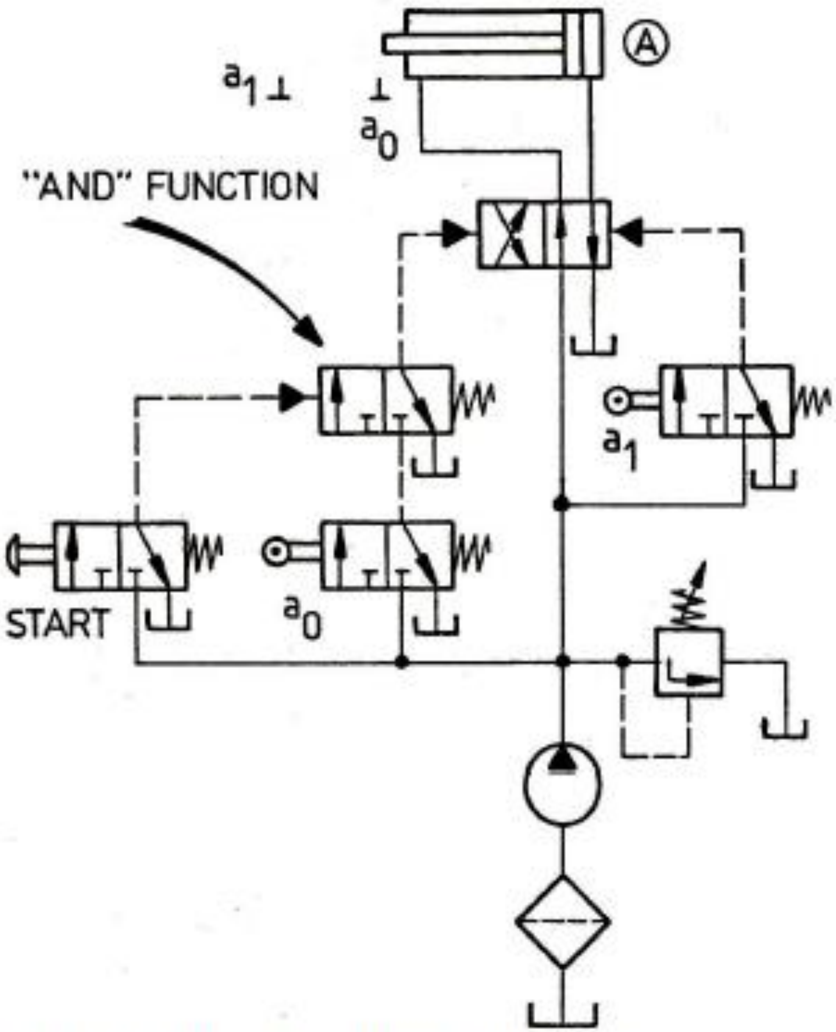


Fig 48 "AND" function with directional control valve

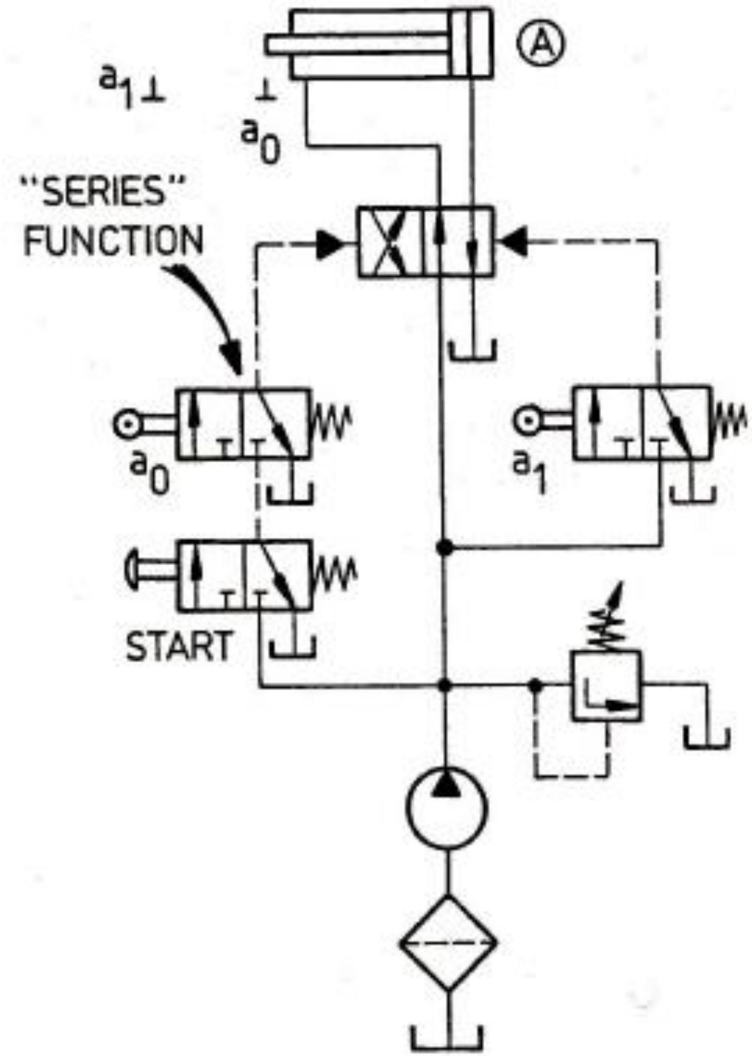


Fig 49 "AND" function by series arrangement

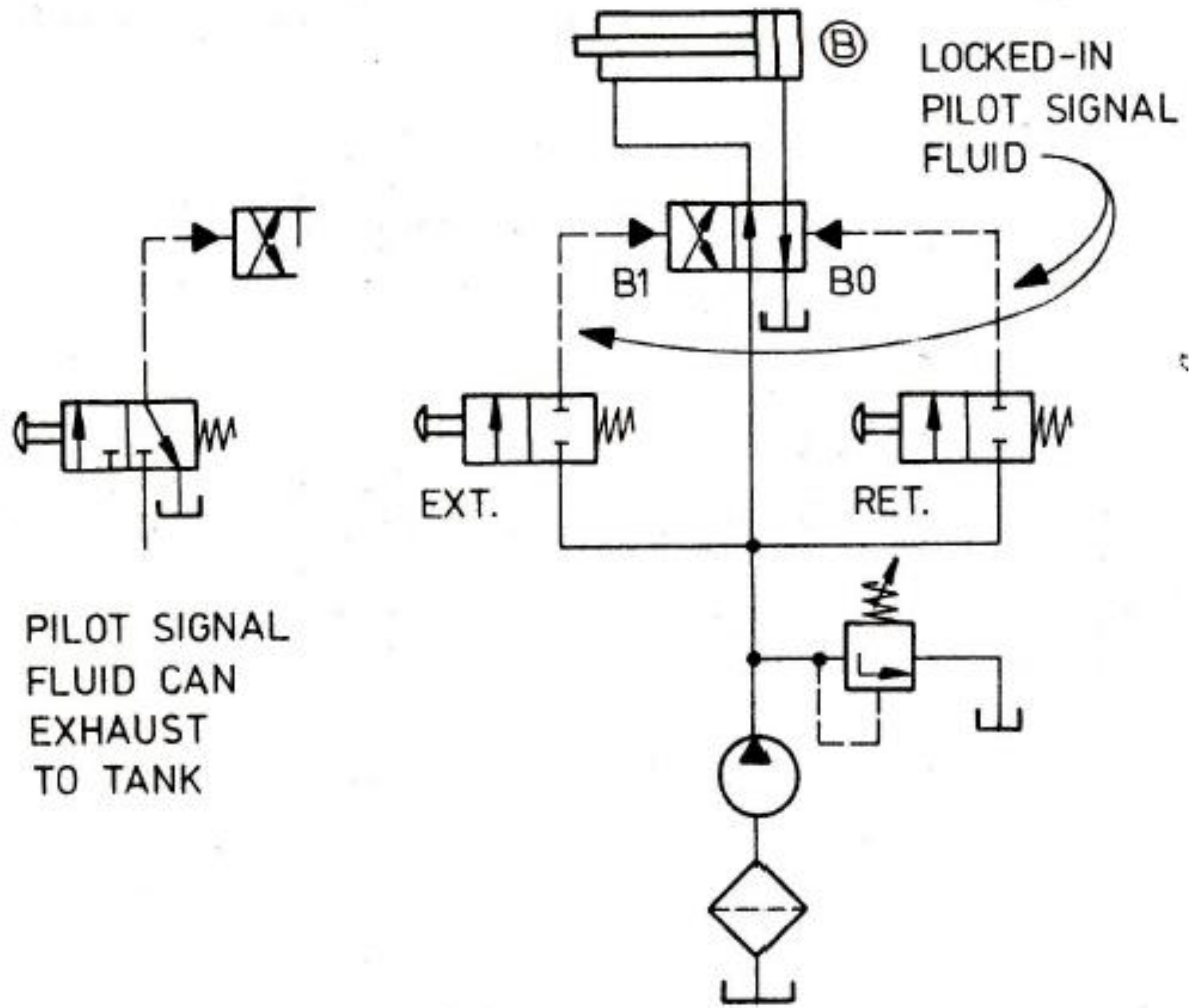


Fig 50 Prevention of locked in pilot signal

Pressure Controls

Pressure control valves are used in hydraulic systems to control actuator force (force = pressure x area), and to determine and (pre) select pressure levels at which certain machine operations must occur. Pressure controls are in the main used to perform the following system functions:

1. To limit maximum system pressure in a hydraulic circuit or sub-circuit, and thus provide overload protection.
2. To provide re-direction of pump flow to tank, while system pressure must be maintained (system unloading).
3. To provide re-direction of pump flow to tank while system pressure is not maintained (system offloading).
4. To offer resistance to fluid flow at selectable pressure levels (counterbalance force).
5. To provide an alternative flow path for the fluid at selected pressure levels (pressure sequencing).
6. To reduce (or step down) pressure levels from the main circuit to a lower pressure in a sub-circuit.

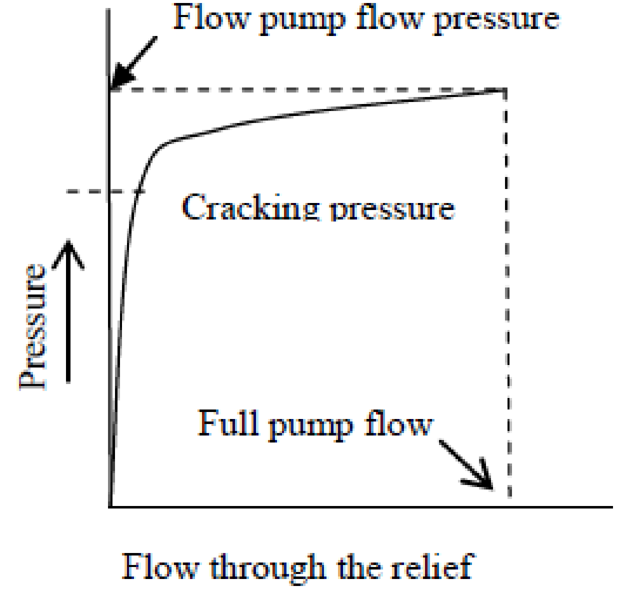


Figure 1.3 Characteristics of a relief valve.

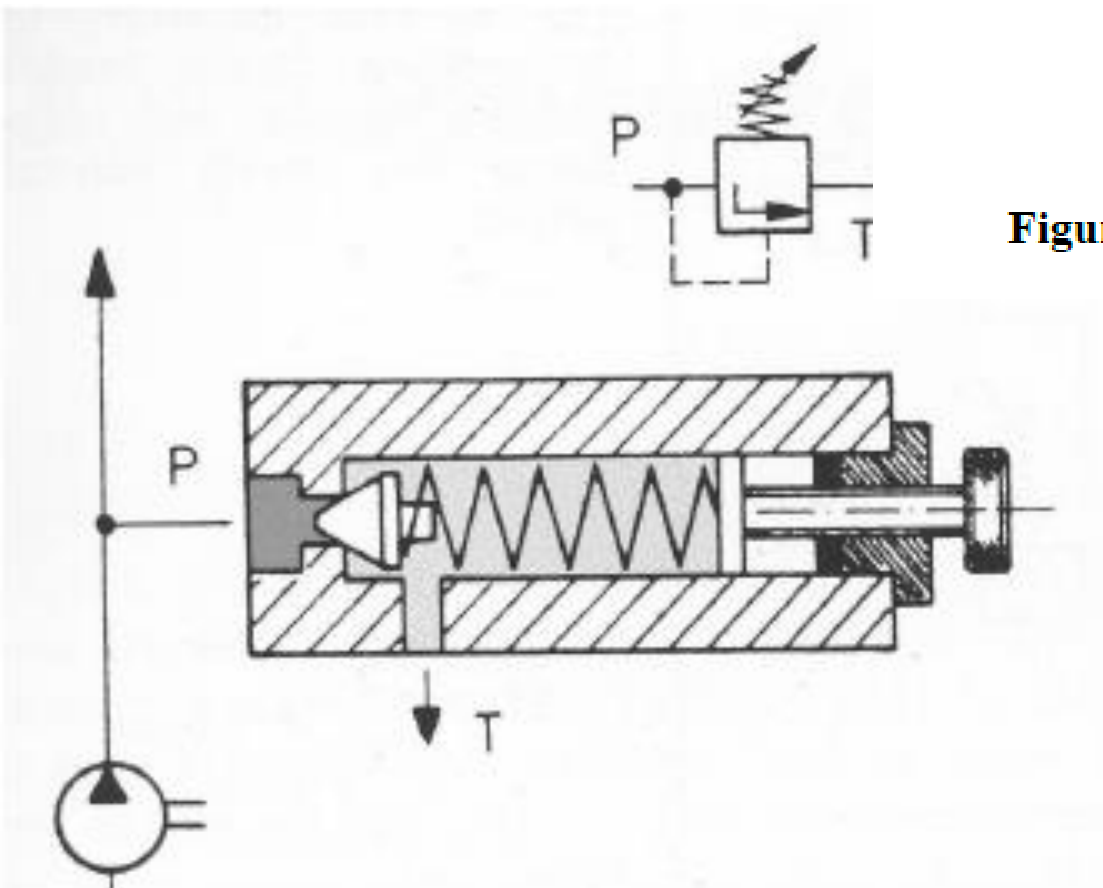
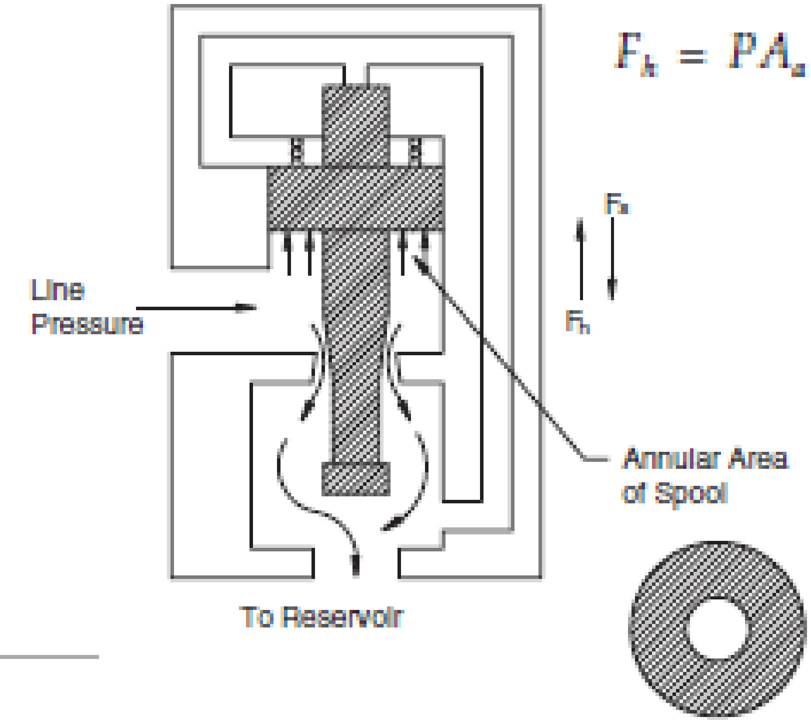
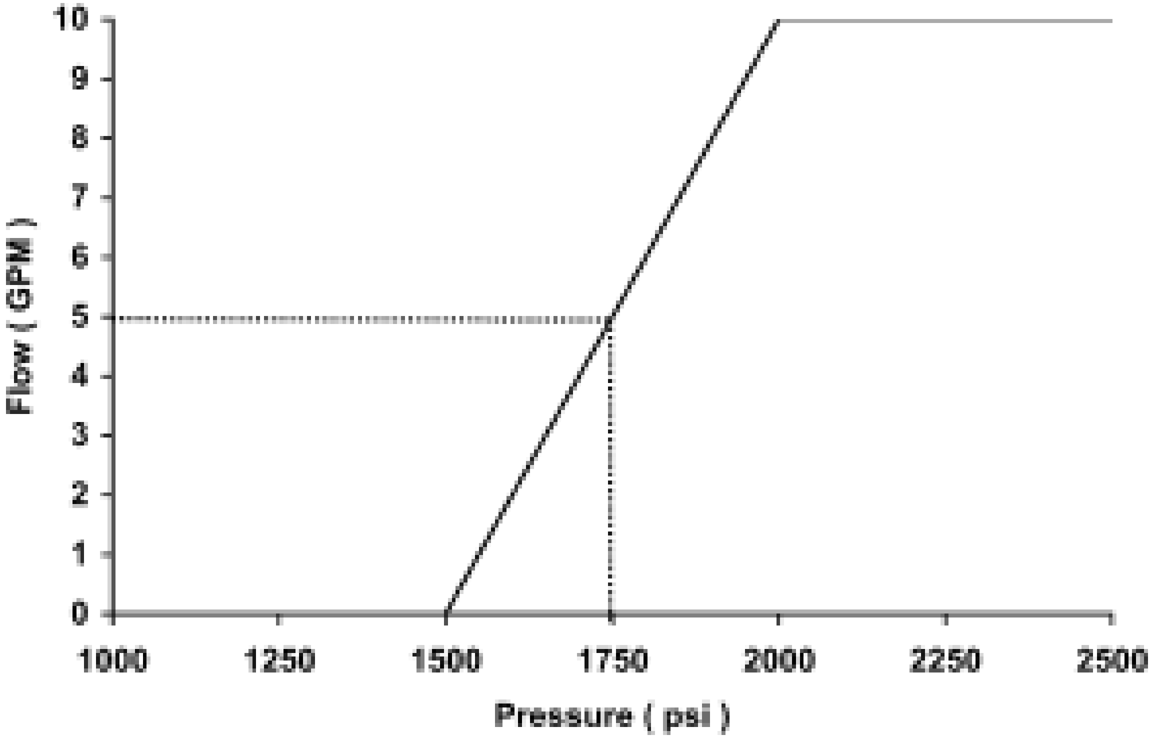


Fig. 97 Direct acting relief valve (simple relief valve).

$$F_h = PA_a$$



Fs
Fh
Line Pressure
Annular Area of Spool
To Reservoir



10
9
8
7
6
5
4
3
2
1
0
Flow (GPM)
1000
1250
1500
1750
2000
2250
2500
Pressure (psi)

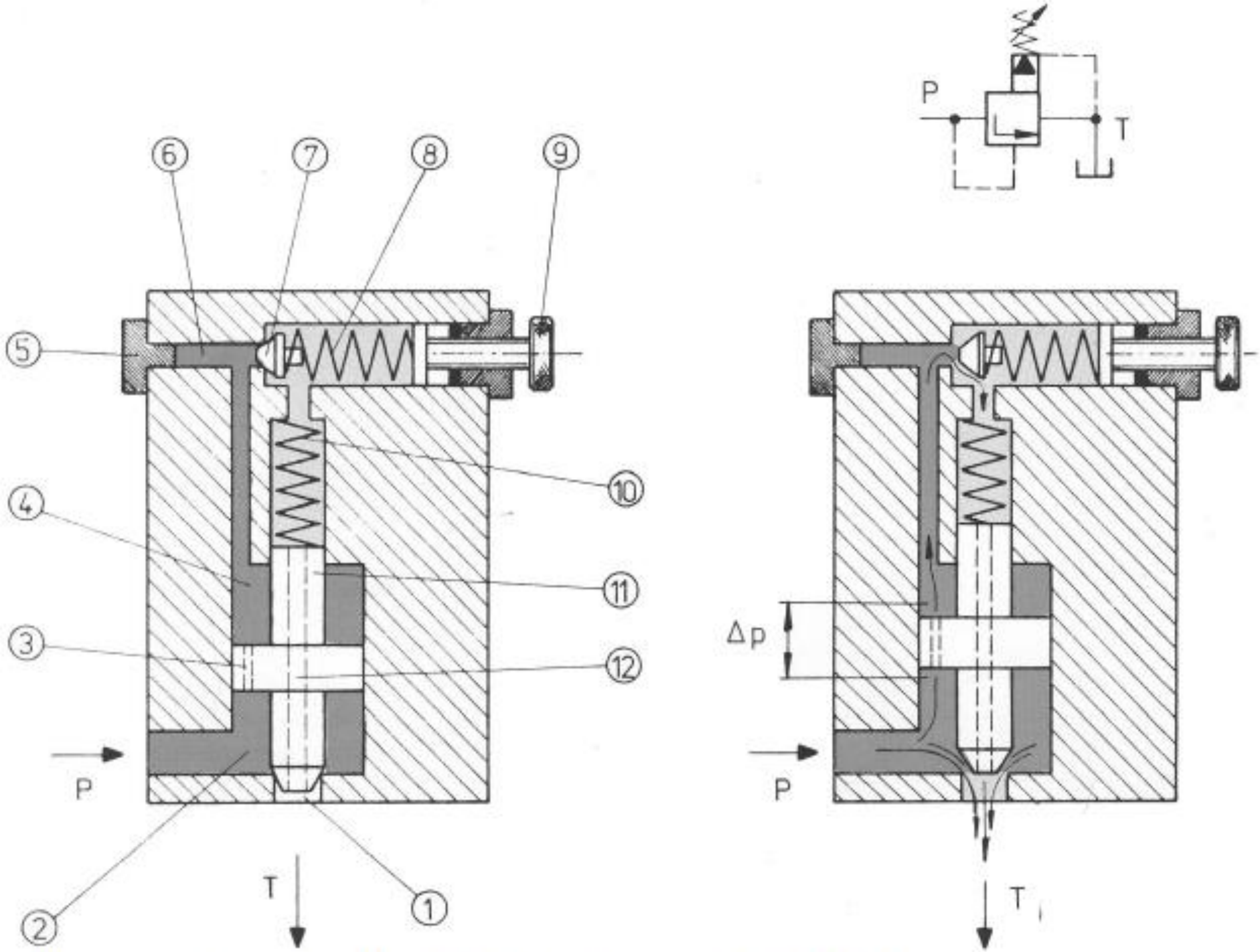


Fig 99 Compound relief valve (pilot operated relief valve)

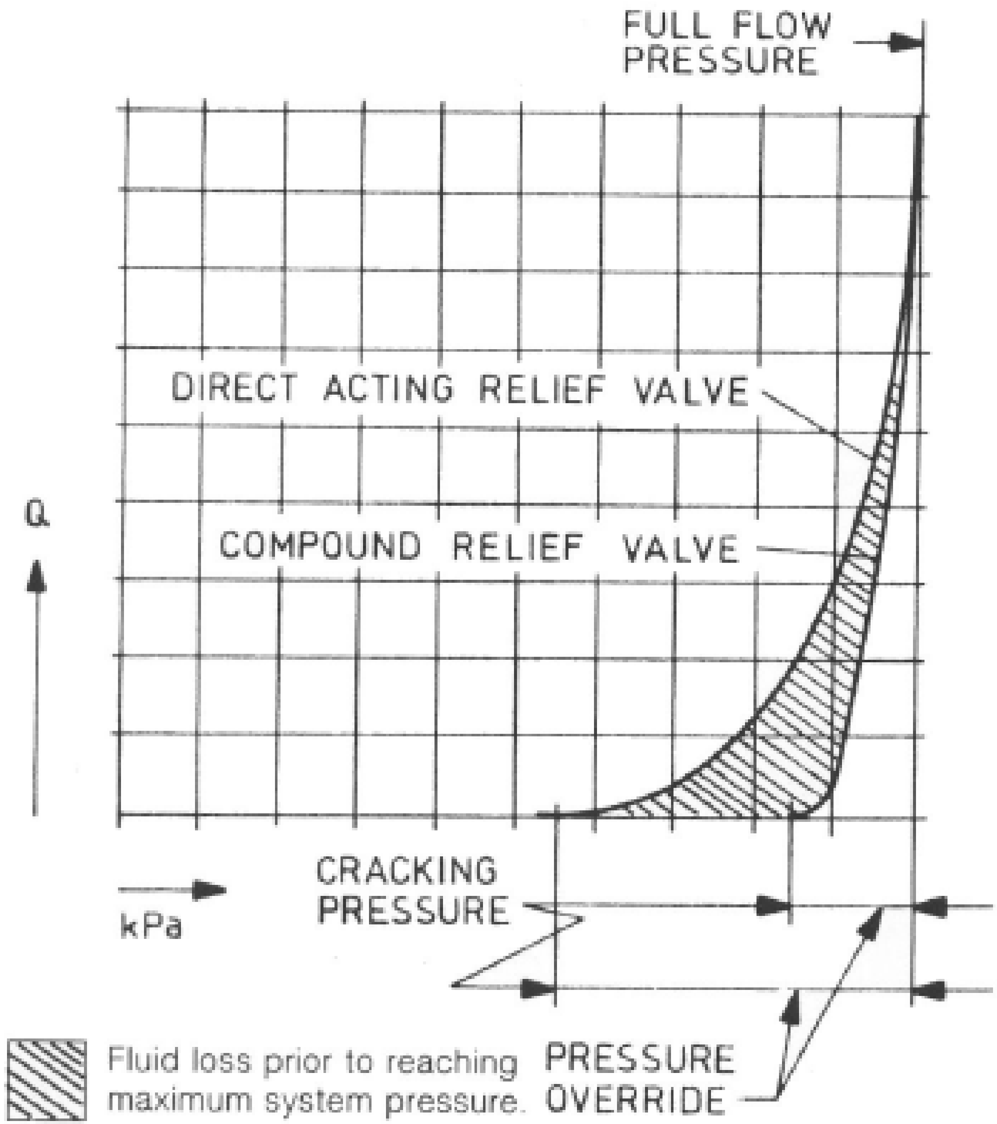
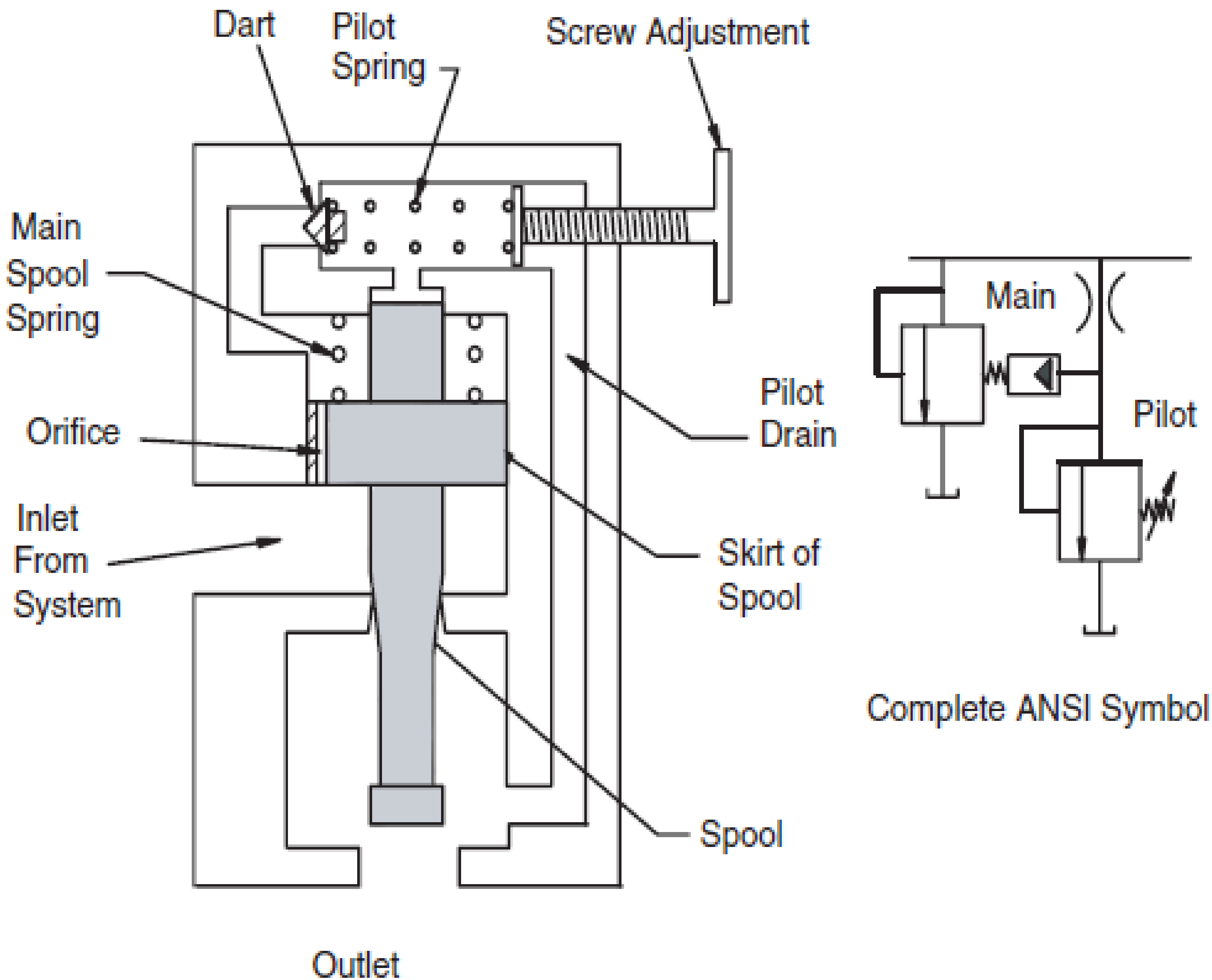
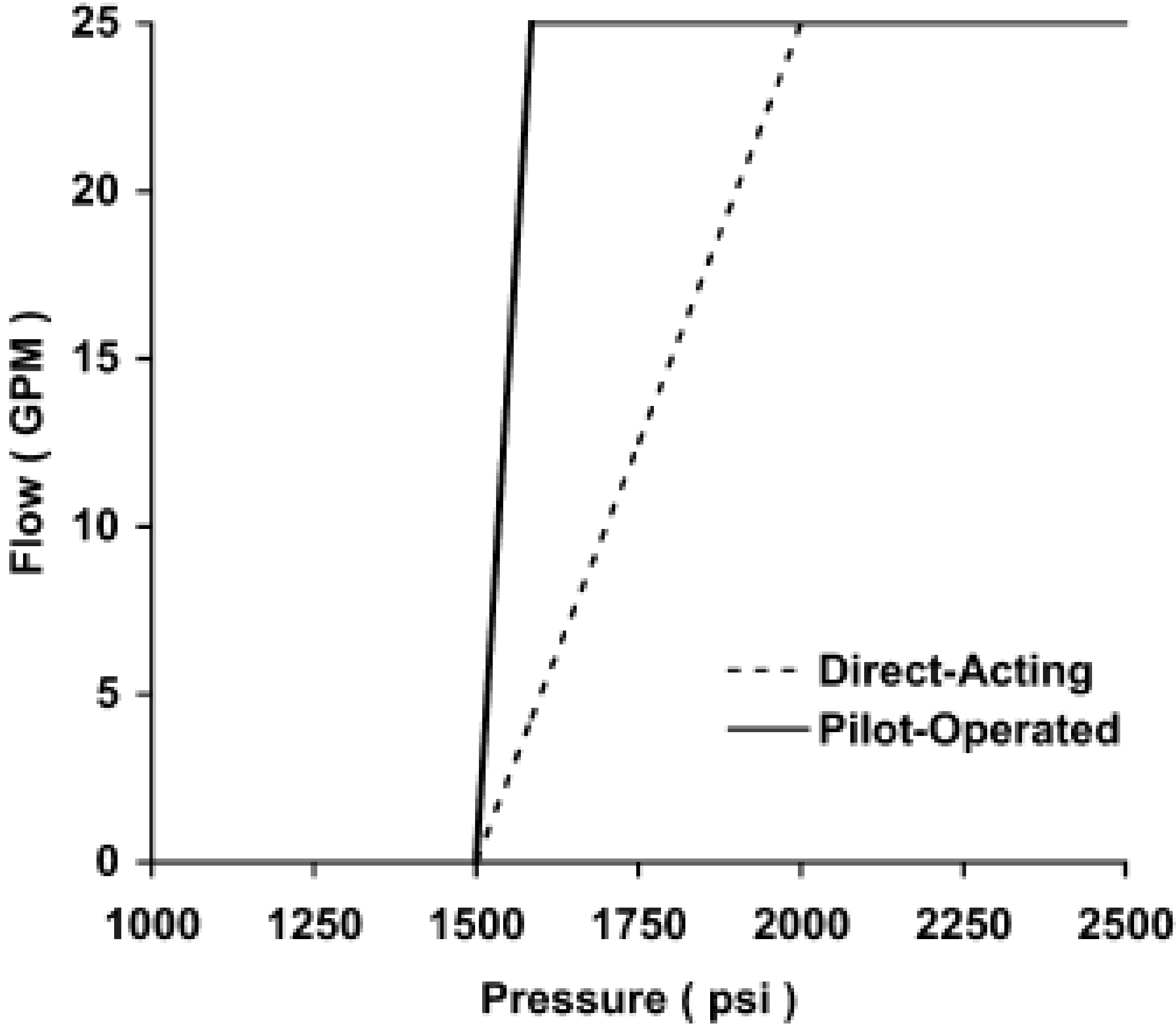


Fig. 98 Q-P graph for pressure-flow relief behaviour, comparing compound and simple relief valves.


Dart
Pilot
Spring
Screw Adjustment
Main
Spool
Spring
Orifice
Inlet
From
System
Pilot
Drain
Skirt of
Spool
Spool
Outlet
Main
Pilot
Complete ANSI Symbol



25
20
15
10
5
0
Flow (GPM)
1000
1250
1500
1750
2000
2250
2500
Pressure (psi)
Direct-Acting
Pilot-Operated

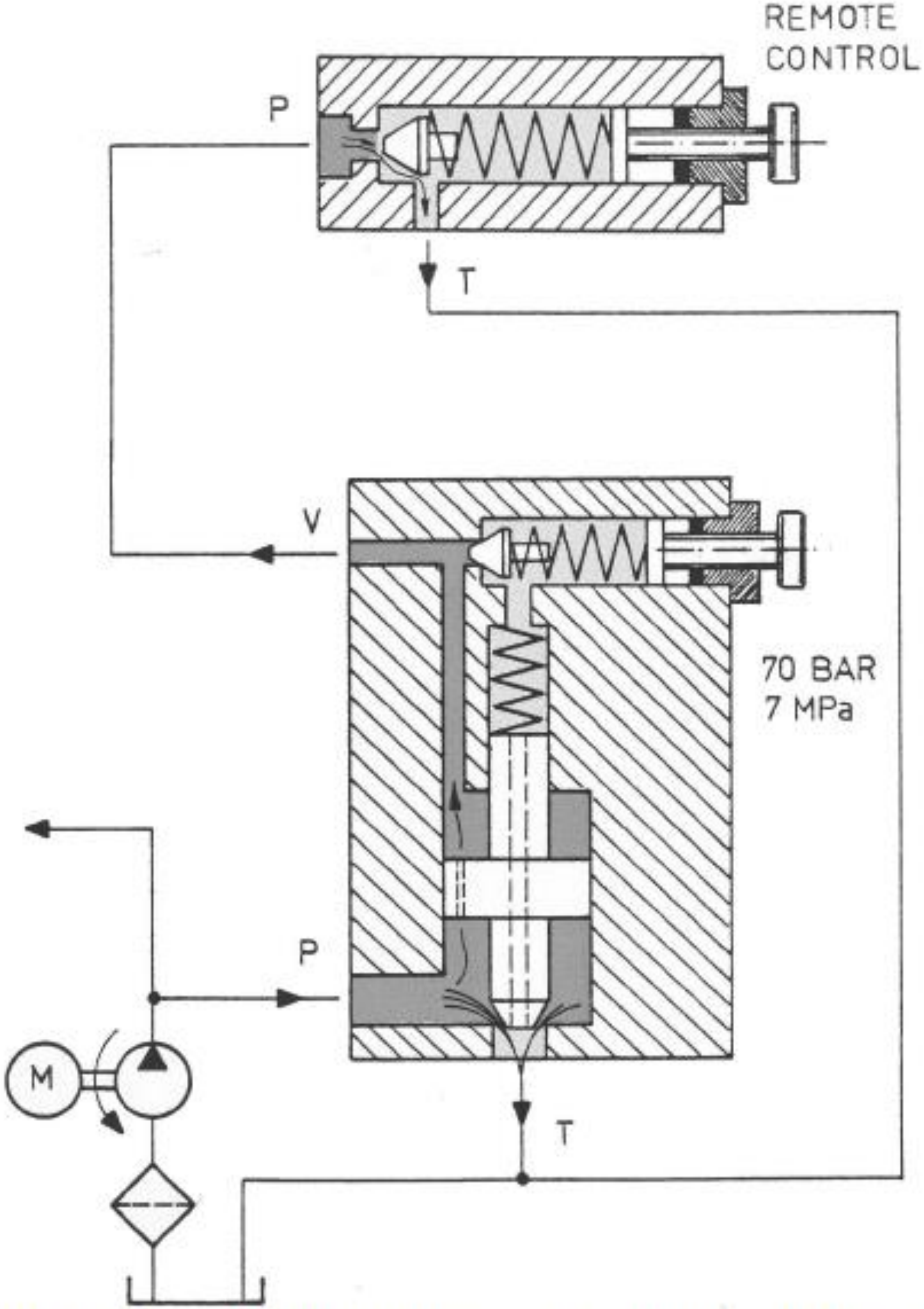


Fig 101 Remote pressure control: the direct acting relief valve could also be replaced by a compound type relief valve.

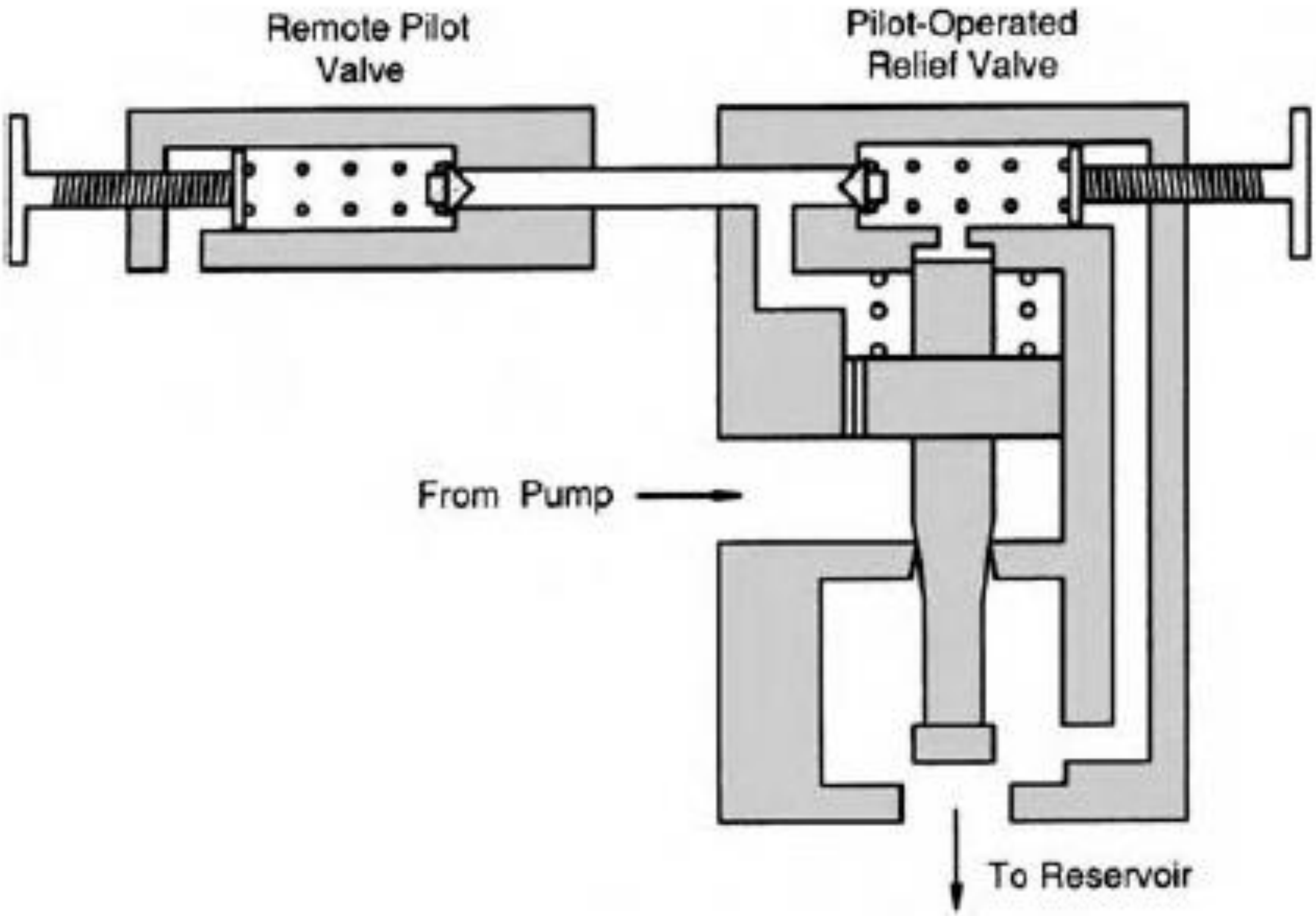
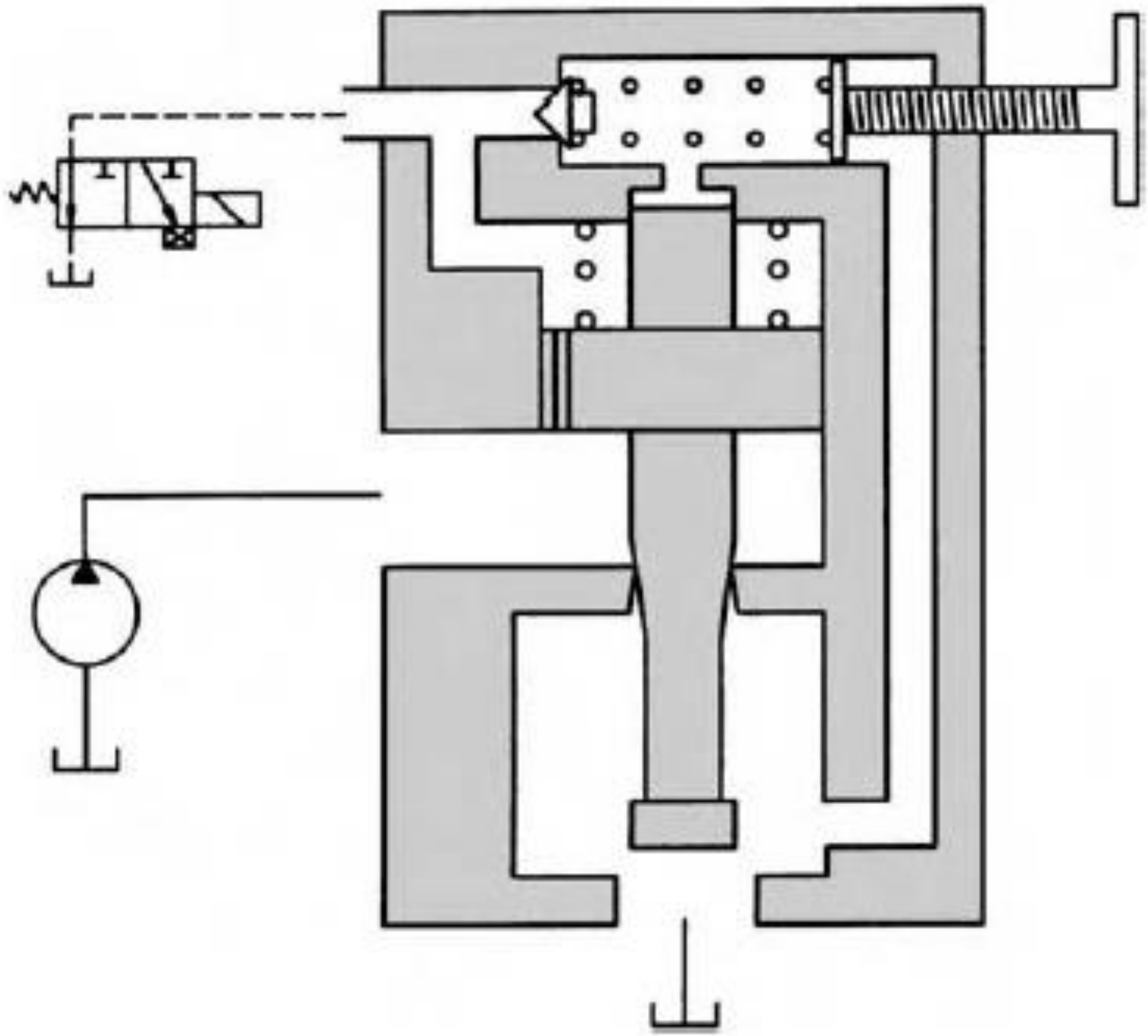


FIGURE 3.9
Pilot-operated relief valve with remote pilot. (Reprinted with permission from Parker Hannifin

Pilot Operated
Relief Valve



A pressure-relief valve has a pressure setting of 140 bar. Compute the kW loss across this valve if it returns all the flow back to the tank from a 0.0016 m^3/s pump.

Solution: We have

$$\begin{aligned} \text{kW power} &= pQ \\ &= (140\times10^5)\times(0.0016\times10^{-3}) \\ &= 22.4\ \text{kW} \end{aligned}$$

Example 1.2

A pressure-relief valve contains a poppet with an area of 4.2 cm^2 on which the system pressure acts. During assembly, a spring with a spring constant of 3300 N/cm is installed in the valve to hold the poppet against its seat. The adjustment mechanism is then set so that the spring is initially compressed to 0.5 cm from its free-length condition. In order to pass full pump flow through the valve at the pressure-relief valve pressure setting, the poppet must move 0.30 cm from its fully closed position.

(a) Determine the cracking pressure.

(b) Determine the full pump flow pressure (pressure-relief valve pressure setting).

(c) What should be the initial compression of the spring in pressure-relief valve if the full pump flow is to be 40% greater than the cracking pressure?

Solution:

(a) Cracking pressure:

Force required to fully close is the product of initial displacement and spring constant

$$F_{\text{valve closed}} = K\,S_{\text{initial}} = 3200\ \text{N/cm} \times 0.50\ \text{cm} = 1600\ \text{N}$$

Now we can calculate the cracking pressure knowing the cracking force

$$p_{\text{cracking}} A_{\text{poppet}} = 1600\ \text{N}$$

$$\Rightarrow p_{\text{cracking}} (4.20 \times 10^{-4}\ \text{m}^2) = 1600\ \text{N}$$

$$\Rightarrow p_{\text{cracking}} = 381 \times 10^4\ \text{N/m}^2 = 3.81\ \text{MPa}$$

(b) Full pump flow pressure (pressure-relief valve pressure setting):

Force required to fully open is the product of final displacement and spring constant

$$F_{\text{fully open}} = K\,S_{\text{fully open}} = 3200\ \text{N/cm} \times 0.8\ \text{cm} = 2560\ \text{N}$$

Now this force must be equal to product of full pump pressure and area of poppet.

$$p_{\text{full pump flow}} A_{\text{poppet}} = 2650\ \text{N}$$

$$\Rightarrow p_{\text{full pump flow}} (4.20 \times 10^{-4}\ \text{m}^2) = 2650\ \text{N}$$

$$\Rightarrow p_{\text{full pump flow}} = 610 \times 10^4\ \text{N/m}^2 = 6.10\ \text{MPa}$$

(c) Initial compression of spring:

$$F_{\text{valve closed}} = K\,l = 3200l = p_{\text{cracking}} A_{\text{poppet}}$$

Now cracking pressure can be calculated as follows

$$p_{\text{cracking}} = \frac{\text{Force}}{\text{Area}} = \frac{3200\,l}{4.20\times10^{-4}} = 762\times10^{4}\,l$$

Also we know that force required to fully open is given by product of full pump flow and area of poppet.

$$\begin{aligned} F_{\text{fully open}} &= K\ (l+0.3) \\ &= 3200(l+0.3) \\ &= 3200l + 960 \\ &= p_{\text{full pump flow}} A_{\text{poppet}} \end{aligned}$$

Now

$$p_{\text{full pump flow}} = \frac{3200l + 960}{4.20\times10^{-4}} = (762l + 229)10^{4}$$

We can now calculate the ratio of pump full flow pressure to cracking pressure as

$$\frac{p_{\text{full pump flow}}}{p_{\text{cracking}}} = \frac{(762l + 229)10^{4}}{(762l)10^{4}} = 1.40$$

Solving we get $l = 0.75$ cm.

A pressure-relief valve contains a poppet with a 3.87 cm^2 area on which the system pressure acts. The poppet must move 0.381 cm from its fully closed position in order to pass pump flow at the pressure-relief valve setting (full pump flow pressure). The pressure required to overcome the external load is 68.95 bar. Assume that the pressure-relief valve setting is 50% higher than the pressure required to overcome the external load. If the valve-cracking pressure is 10% higher than the pressure required to overcome the external load, find the following:

(a) The required spring constant of the compression in the valve.

(b) The required initial compression of the spring from its free length condition as set by the spring adjustment mechanism of the pressure-relief valve.

Solution:

a) At full pump flow pressure, spring force equals hydraulic force on the poppet: Total spring compression (S) = Initial compression (L) + Full poppet stroke

$$\Rightarrow k(L + 0.00381) = 4002.5 \text{ N}$$

$$\Rightarrow kL + 0.00381\,k = 4002.5 \text{ N}$$

Also at cracking pressure, spring force equals hydraulic force on the poppet. Thus, we have

Spring force= Cracking force

$$kL = 0.00381\,k$$

$$= 1.1 \times 68.95 \times 10^5 \text{ N/m}^2 \times 3.87 \times 10^{-4} \text{ m}^2$$

$$= 2935.8 \text{ N}$$

Substituting values of k in $kL + 0.00381\,k = 4002.5$ N, we get

$$2935.8 + 0.00381\,k = 4002.5 \text{ N}$$
$$\Rightarrow k = 279986.22 \text{ N/m}$$

(b) From part (a), we have

$$k = 279986.22 \text{ N/m}$$
$$kL = 2935.8 \text{ N}$$

This implies

$$279986.22 \times L = 2935.8 \text{ N}$$

$$\Rightarrow \quad L = \frac{2935.8}{279986.22} \text{ m} = 0.0104 \text{ m} = 1.04 \text{ cm} = 10.4 \text{ mm}$$

Unloading valve (accumulator charging valve)

The unloading valve (also called accumulator charging valve or differential unloading valve) is in its design closely related to the compound-relief valve (compare figs. 99 and 104). This valve is used to accomplish the following switching and pressure control functions:

- limit maximum system pressure;
- charge the accumulator to maximum system pressure and maintain a working volume and pressure in the accumulator;
- unload the pump when the desired accumulator pressure is reached (e.g. maintain system pressure to actuators).

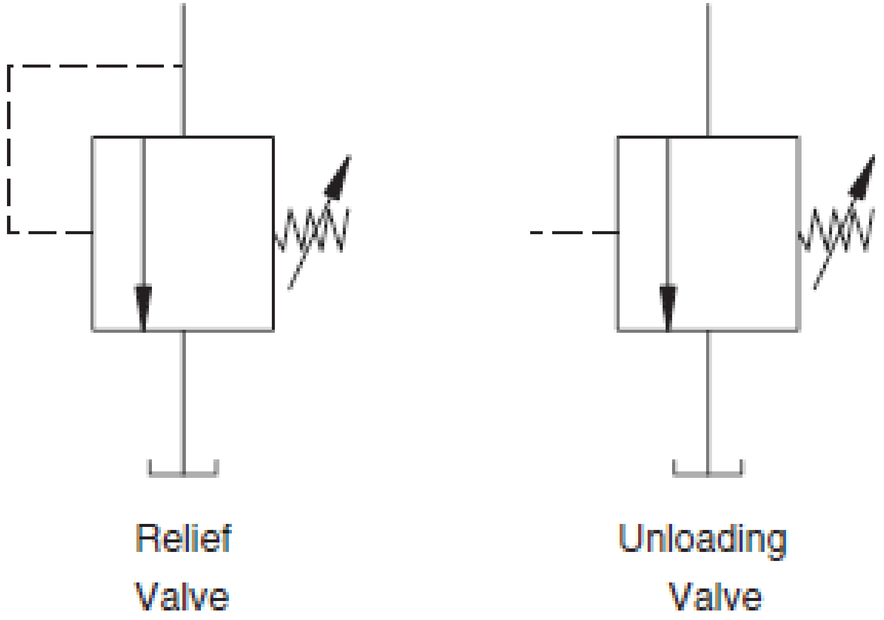
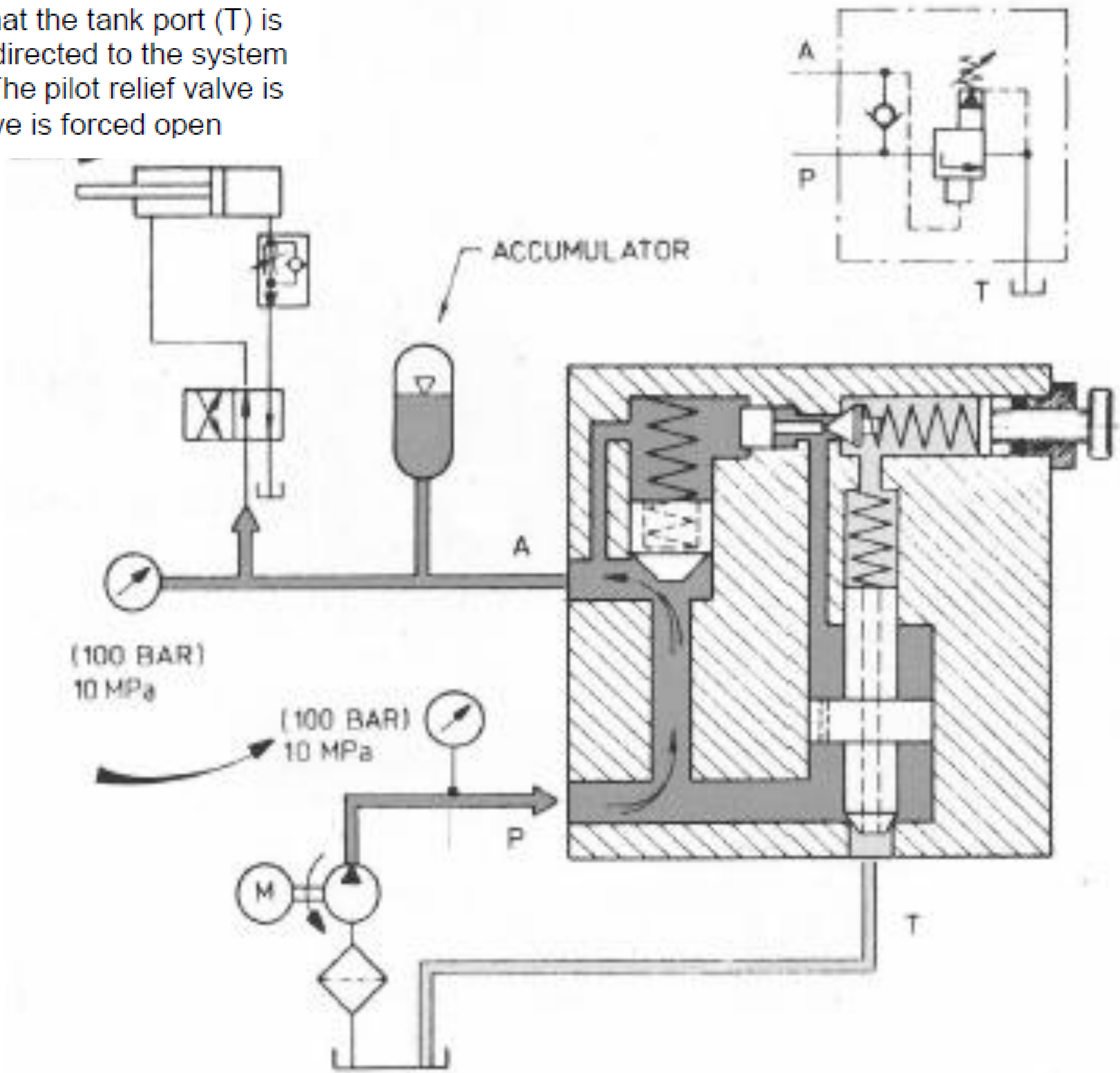


FIGURE 3.14
Comparison of symbols for relief valve and unloading valve.

Fig. 104 Unloading valve in accumulator charging position. Note that the tank port (T) is closed and pump flow is directed to the system and to the accumulator. The pilot relief valve is closed and the check valve is forced open



Pump unloading with accumulator and electric control

The switching and pressure control functions for the accumulator charging circui discussed here are identical to the functions achieved by the unloading valve circuit. A normally-open directional control valve actuated by an electric pressure switch is used to vent or de-vent the compound-relief valve (fig. 106).

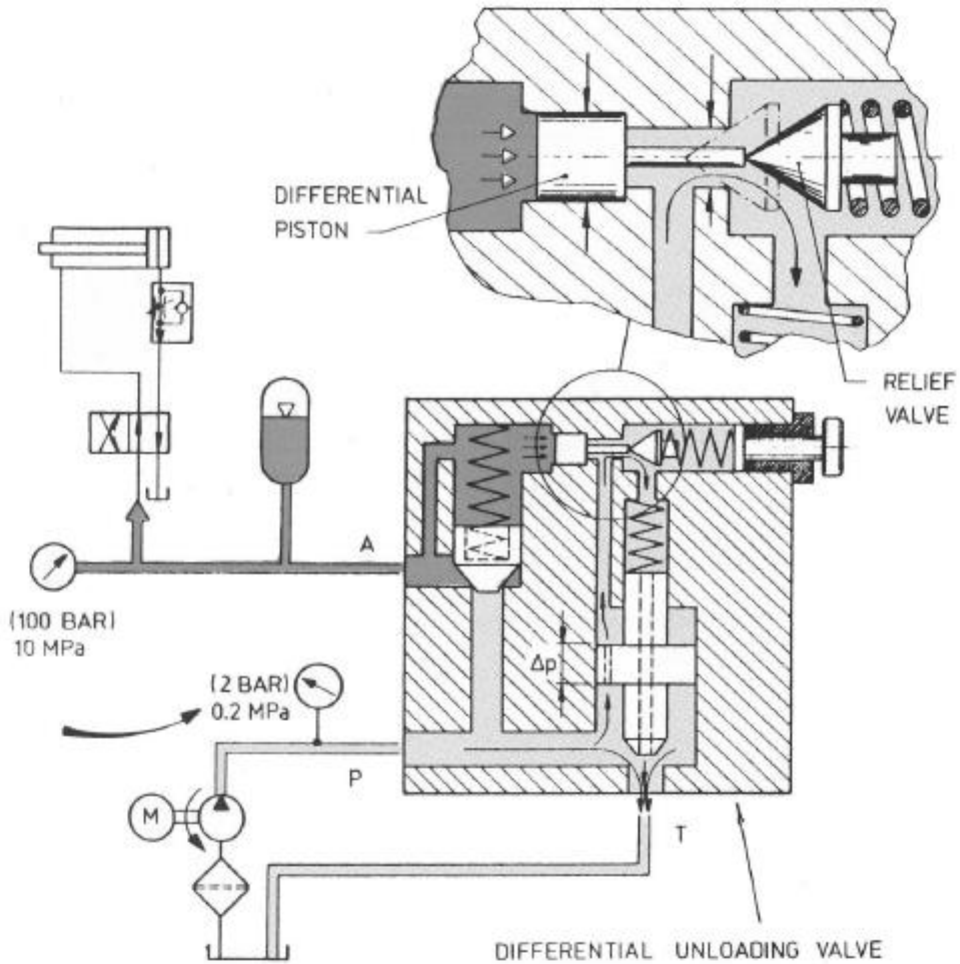


Fig. 105 Unloading valve in pump unloading position. Note that the thank port (T) is open and the pilot relief valve poppet is forced open by the system pressure acting onto the differential piston. The check valve is closed (see also enlarged section and compare the pressure gauge readings with the previous illustration).

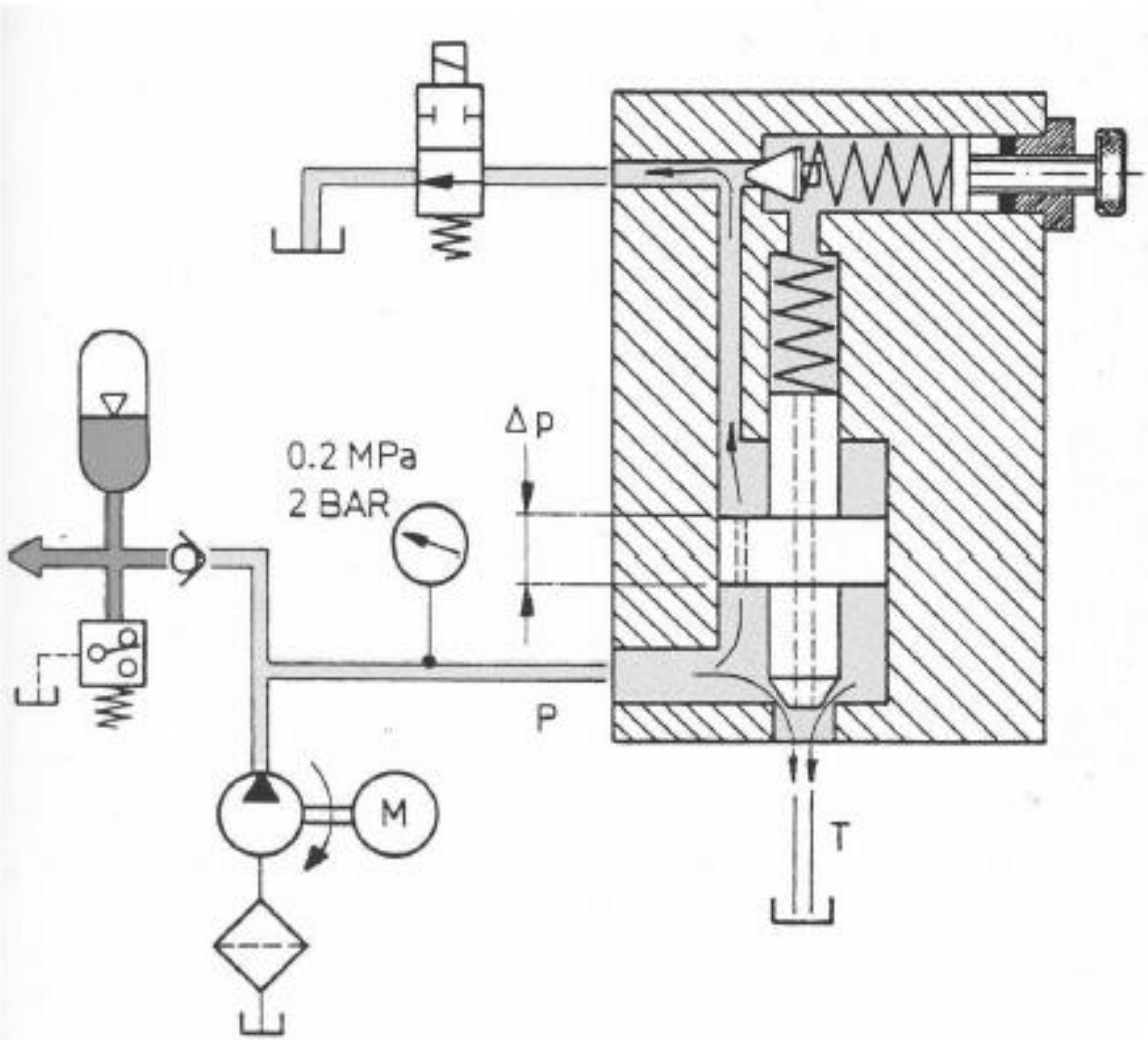


Fig. 106 Pump unloading circuit with accumulator and electric pressure switch, depicting the charging cycle.

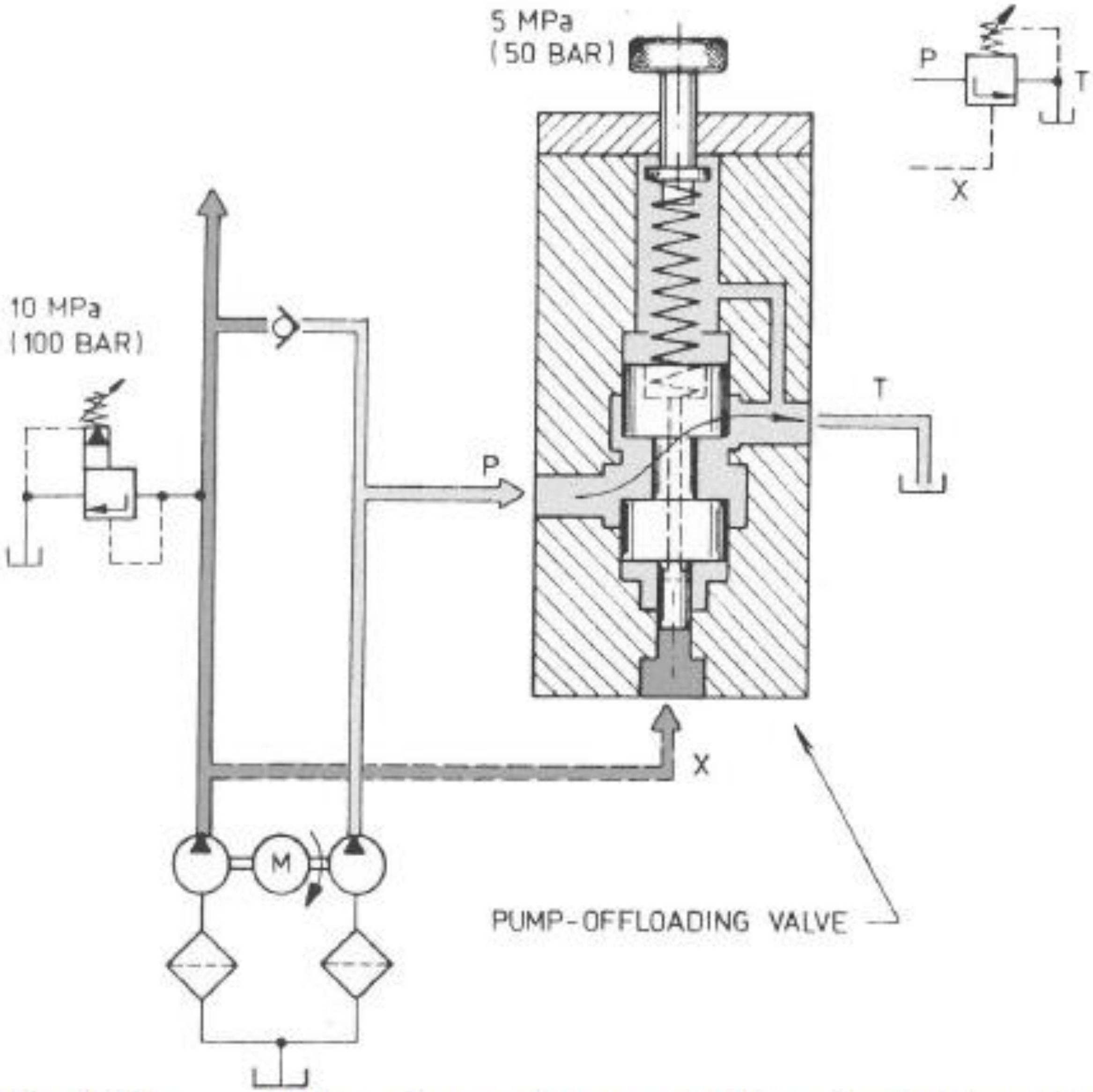


Fig. 107 Offloading valve controlling the high volume, low pressure pump in a "high-low" circuit.

ME 7553 – Hydraulics and Pneumatics

Lecture -12

Date: 30-04-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Review of Lecture 11
2. Counter Balance Valve
3. Break Valve
4. Pressure Reducing Valve
5. Sequence Valve
6. Flow Control Valve

Course Instructor: Dr. A. Siddharthan

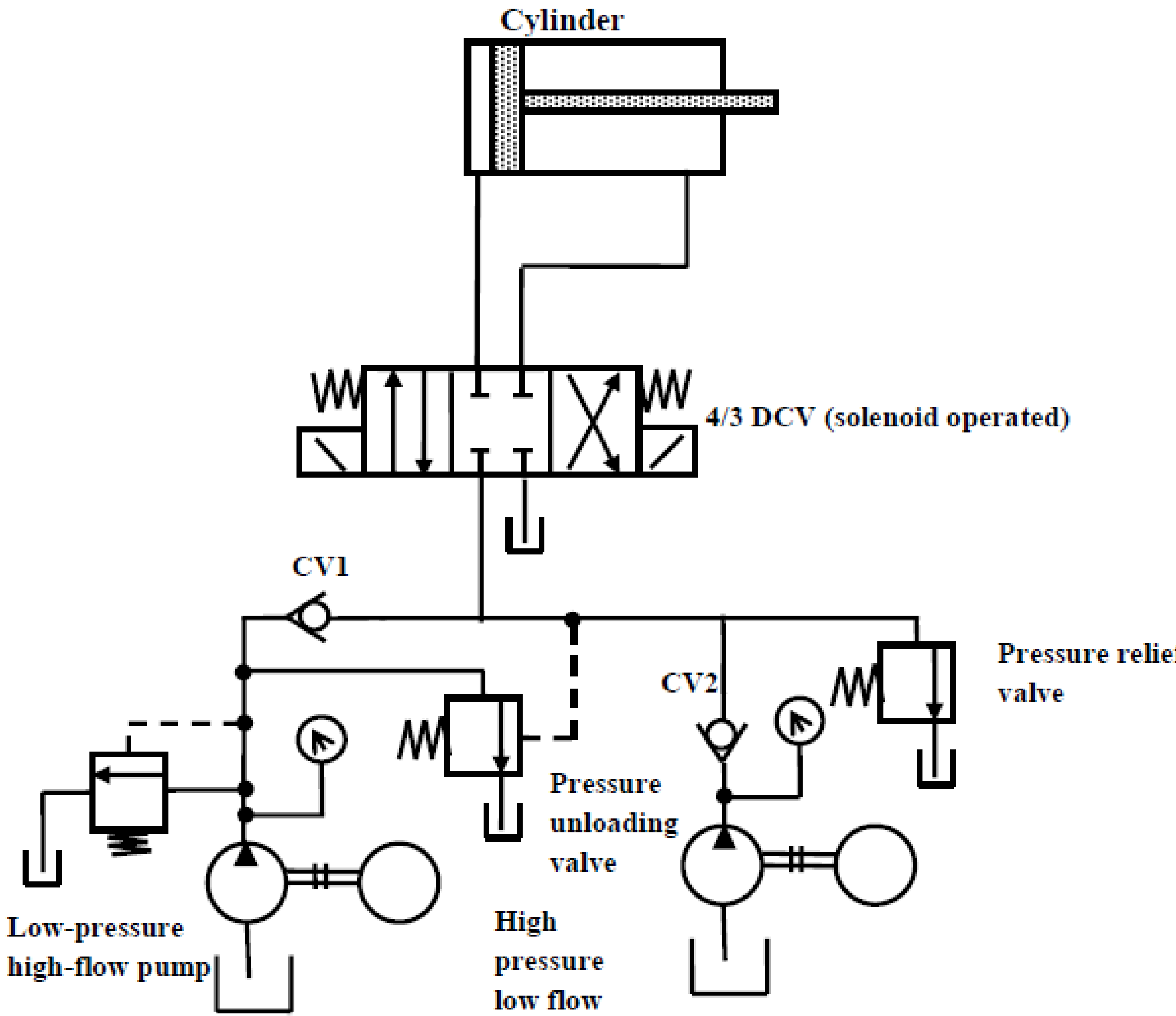
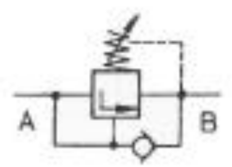
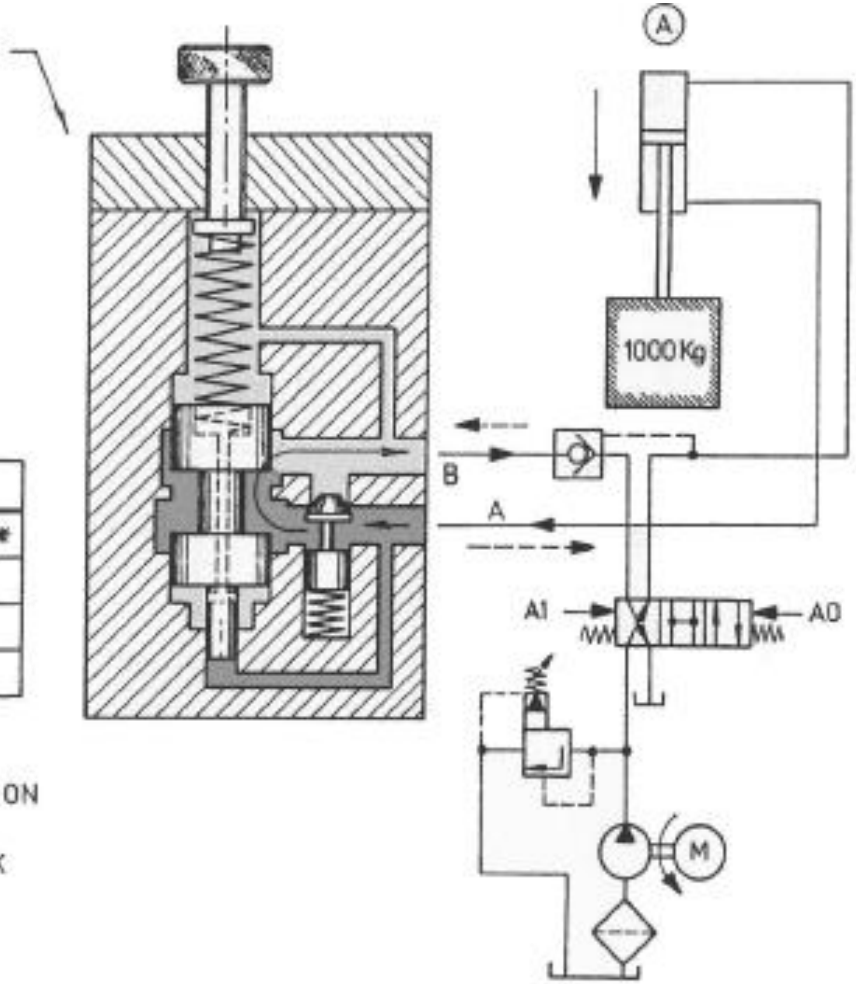


Figure1.11 Application of unloading valve in a punching press (high–low circuit).

Counterbalance valve (back pressure valve)

The counterbalance valve is applied to create a back pressure or cushioning pressure on the underside of a vertically-moving piston, to prevent the suspended load from "free falling" because of gravity whilst it is being lowered (fig. 108). This counteracting or counterbalancing function has given the valve its name.





A0	A1	
0	0	LOAD SUSPENDED *
0	1	LOAD LOWERED
1	0	LOAD LIFTED
1	1	———

* THE LOAD MAY BE STOPPED IN ANY POSITION AND PUMP FLOW IS RECIRCULATED TO TANK

Fig 108 Counterbalance vale circuit. The function table shows the required DCV pilot signals for lifting lowering, and load suspension.

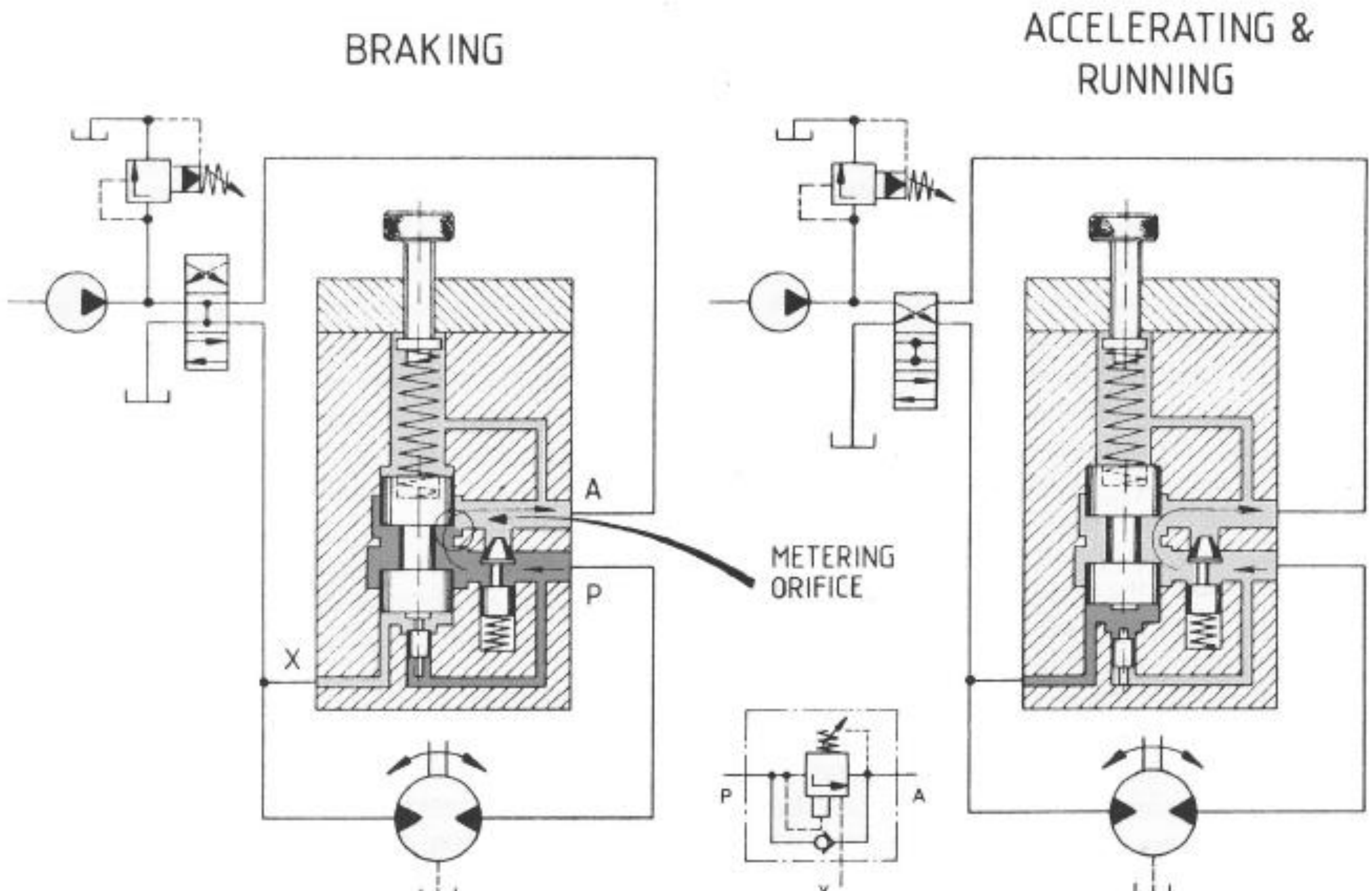
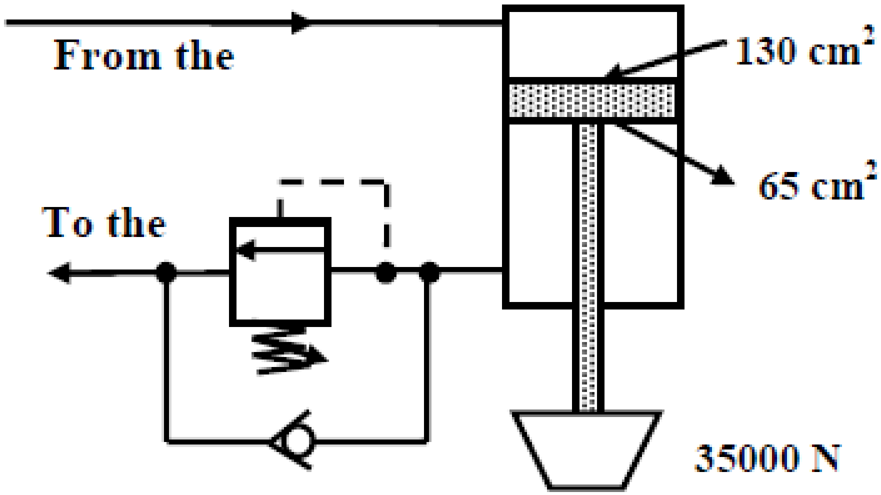


Fig 109

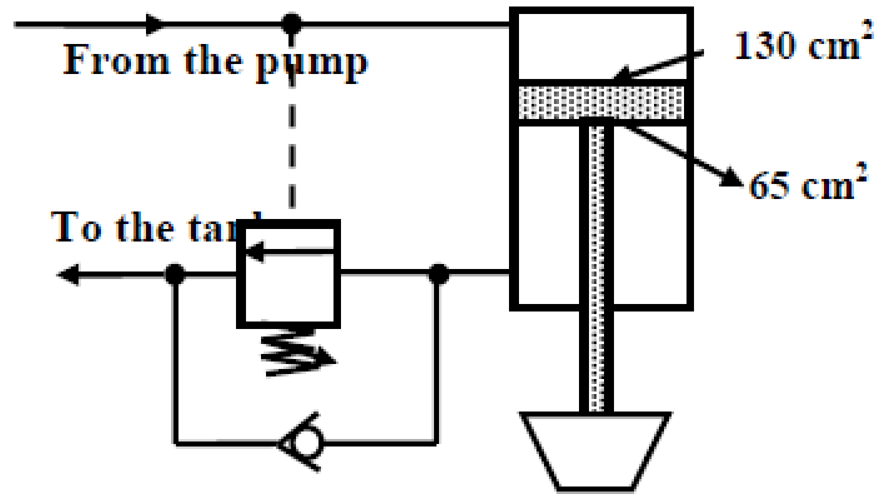
Brake valve

The brake valve is closely related to the counterbalance valve, and serves - when mounted in the exhaust line of a hydraulic motor - the following functions:

- It prevents the hydraulic motor from over-speeding when an over-spinning load is applied to the motor shaft.
- It prevents an excessive pressure build-up during deceleration, and controls the rate of deceleration.



From the
130 cm²
65 cm²
To the
35000 N
55 + bar



From the pump
130 cm²
65 cm²
To the tank

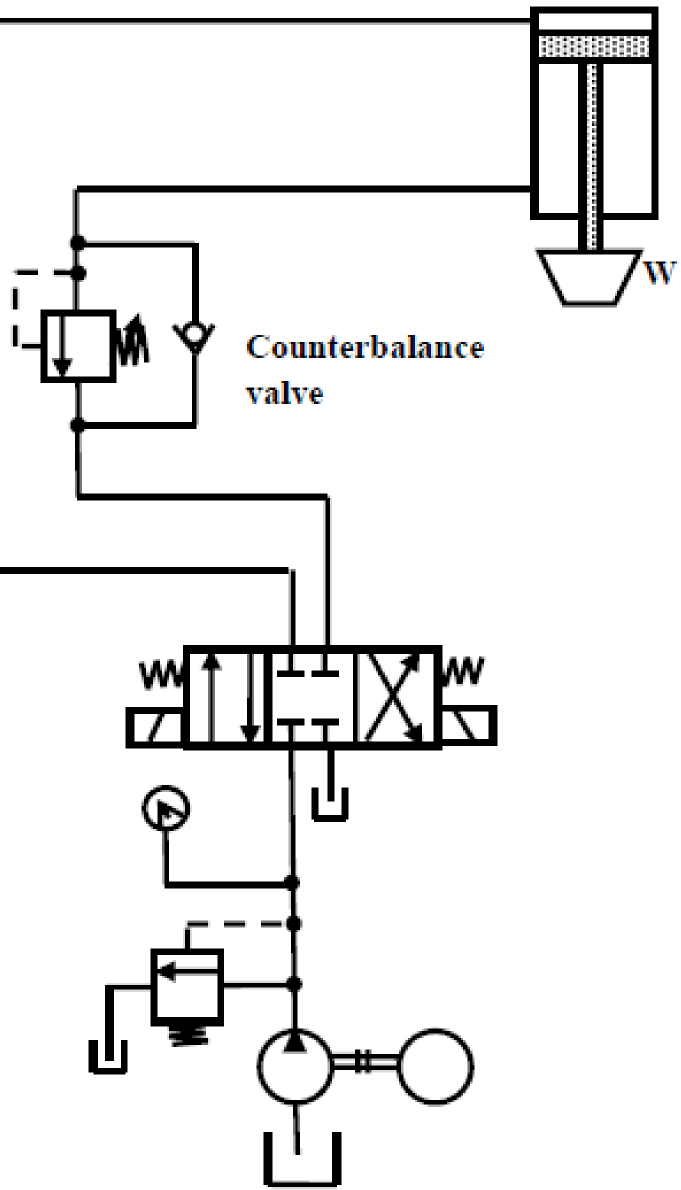


Figure 1.16 Application of a counterbalance valve.

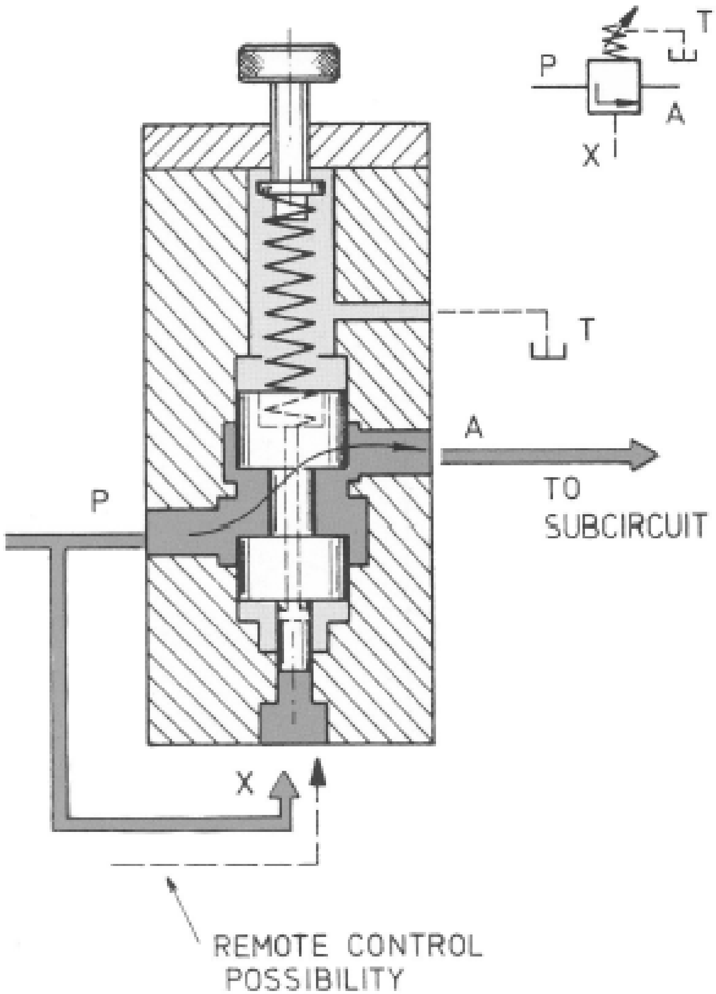


Fig. 110 Direct acting sequence valve

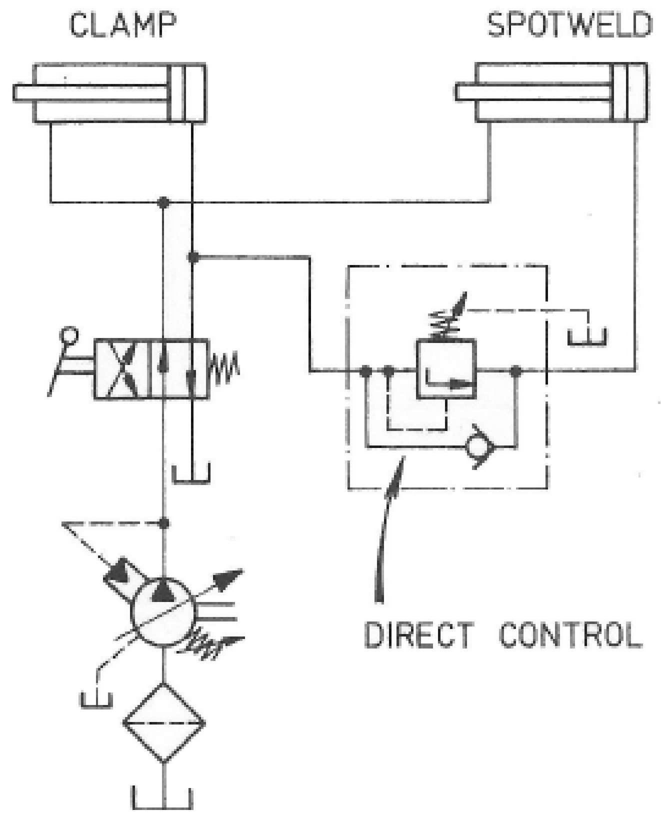


Fig. 111 Clamp and spot weld circuit. The sequence valve is directly controlled. The circuit provides pressure sequencing for the second sequential step (spotweld) as soon as the pre-set clamping pressure is reached

Sequence valve (single stage)

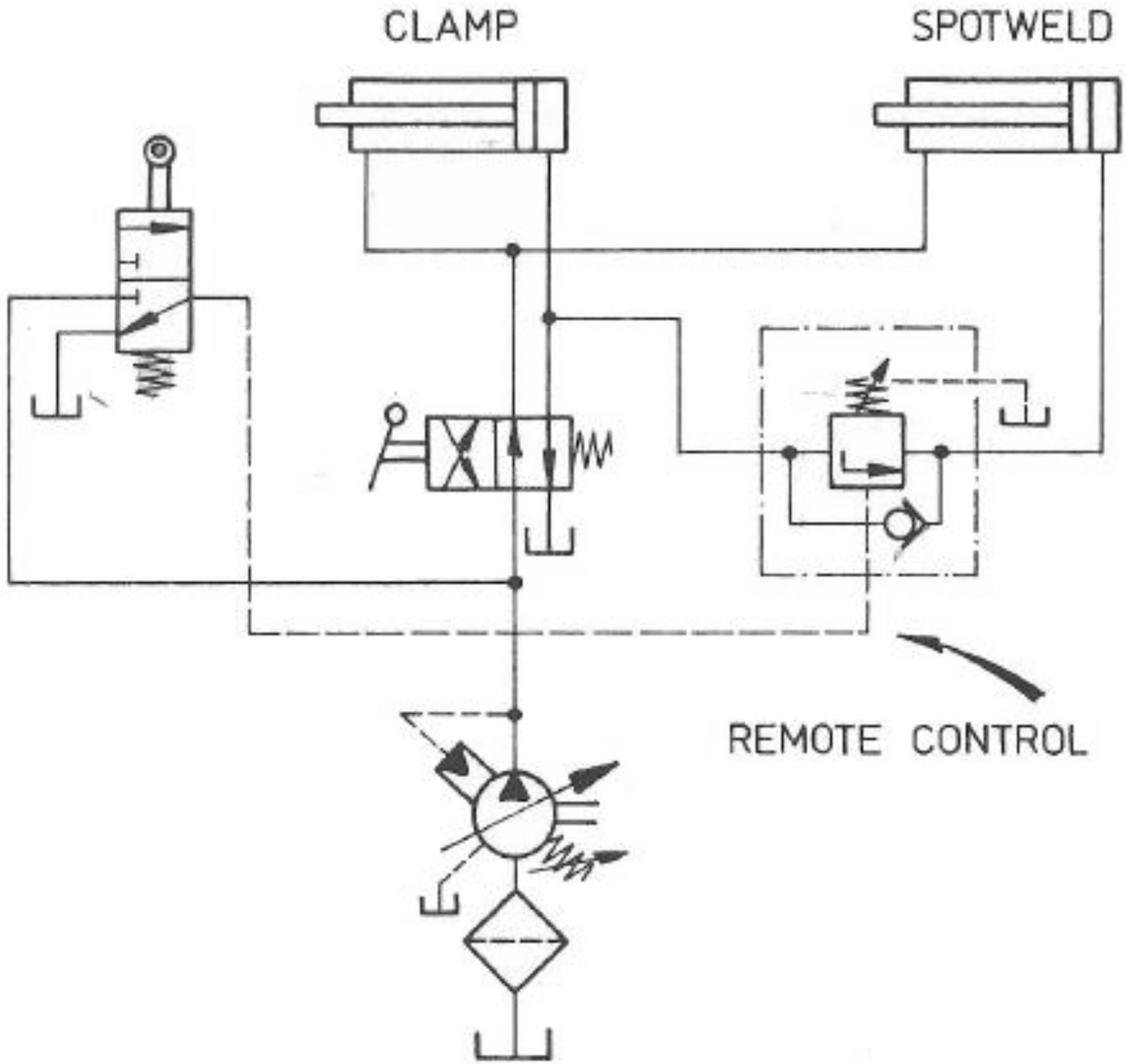


Fig. 112 Clamp and spot weld circuit. The sequence valve is remotely controlled. The circuit thus provides pressure as well as position sequencing for the second sequential step ("and" function).

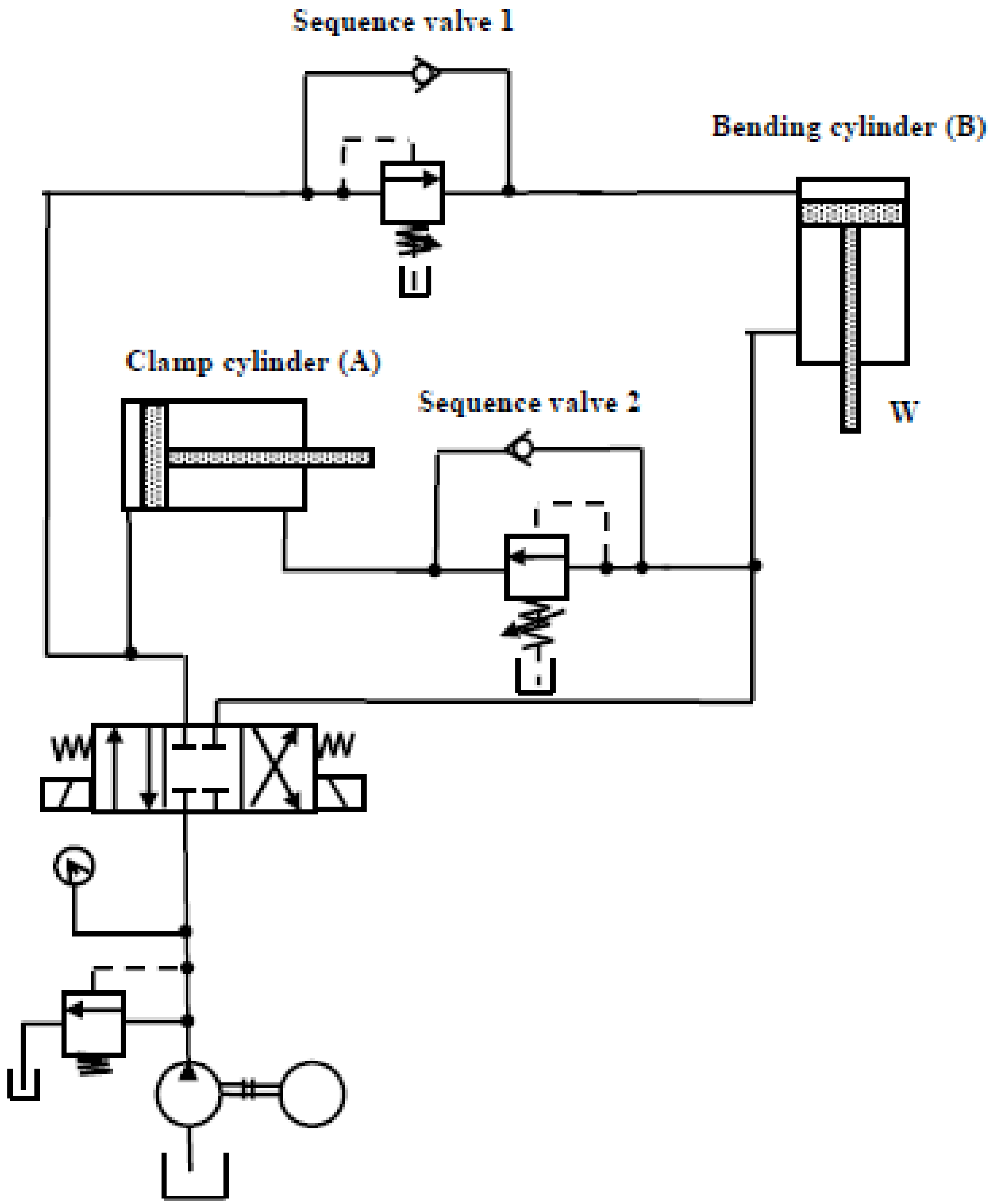


Figure 1.20Application of a sequence valve.

Pressure reducing valve (pilot-operated)

In some fluid power systems it is desirable (and often necessary), to operate a subcircuit at a lower pressure than the main system. Pressure reducing valves are used for this purpose. In contrast to the "normally closed" pressure control valves discussed so far, the pressure reducing valve is "normally open".
The main function of this valve is to limit and maintain a constant downstream pressure (subcircuit pressure), regardless of pressure fluctuations in the main circuit upstream (fig. 114).

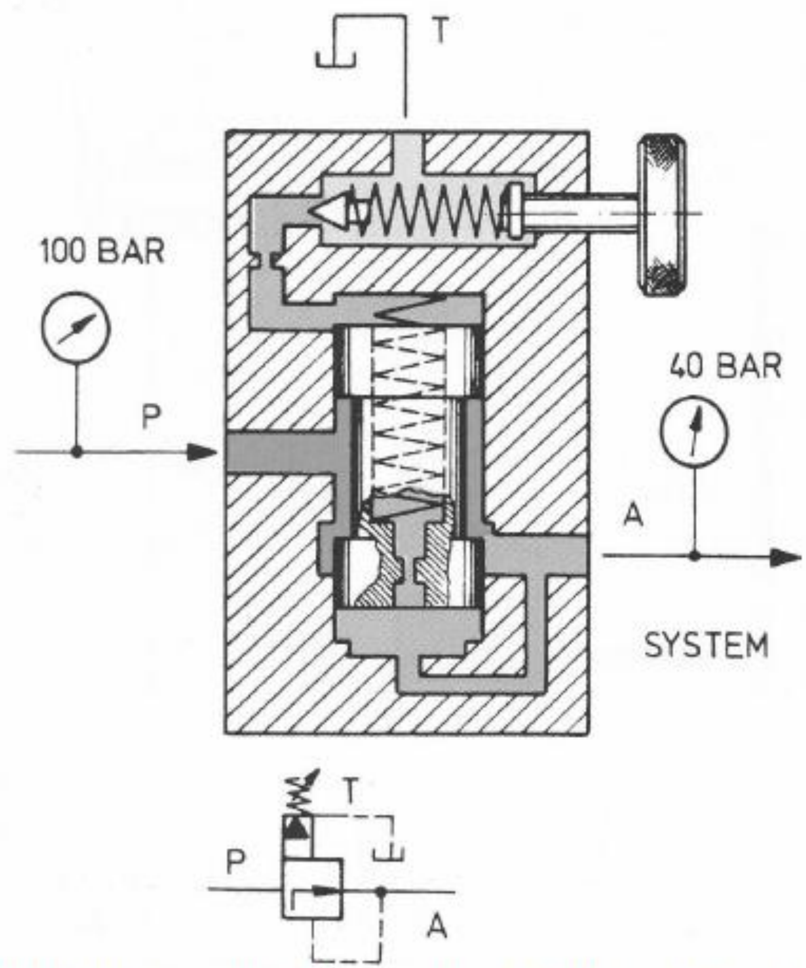


Fig. 114 Pilot operated pressure reducing valve.

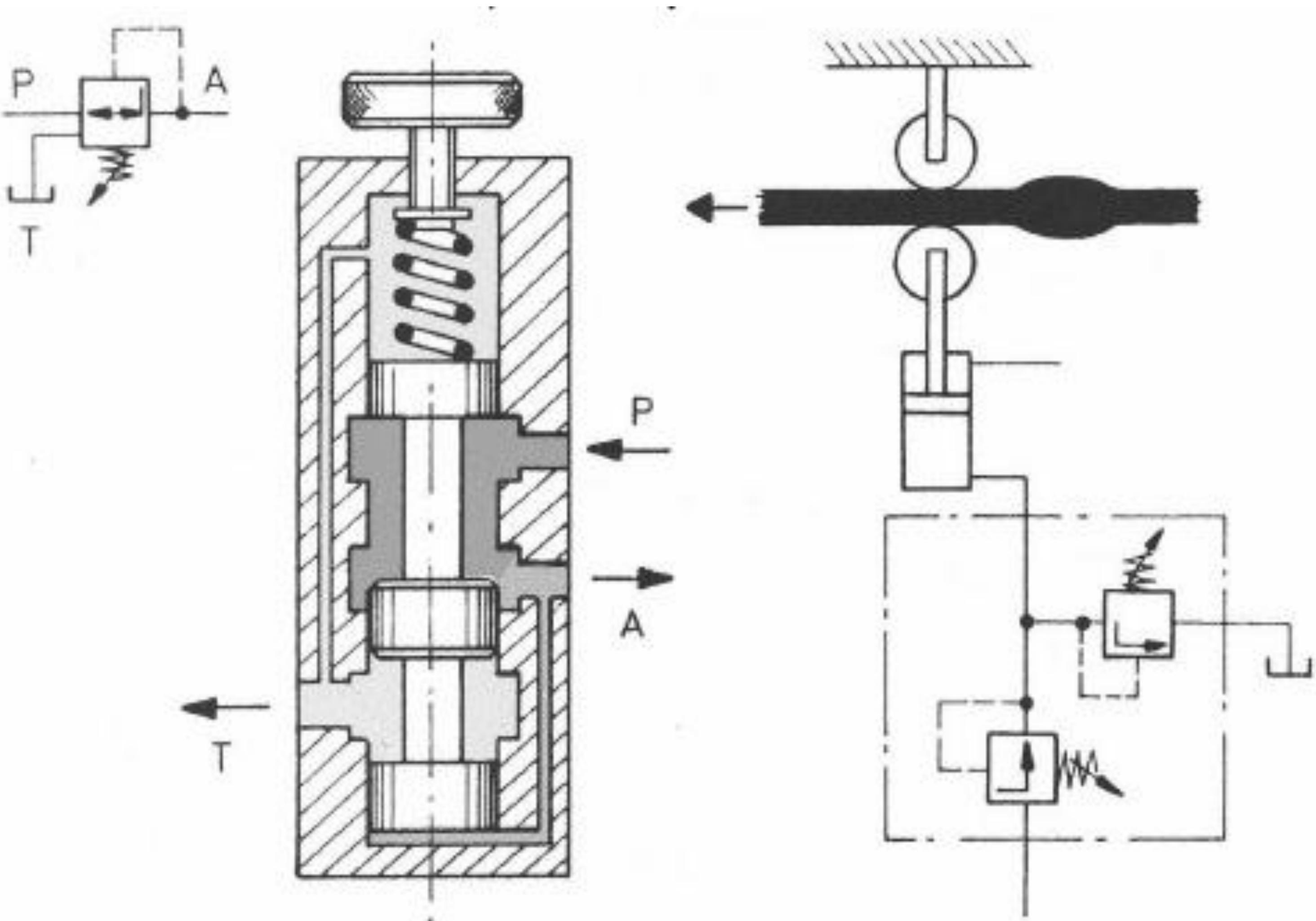


Fig. 115 Direct acting pressure reducing valve with secondary system relief function.

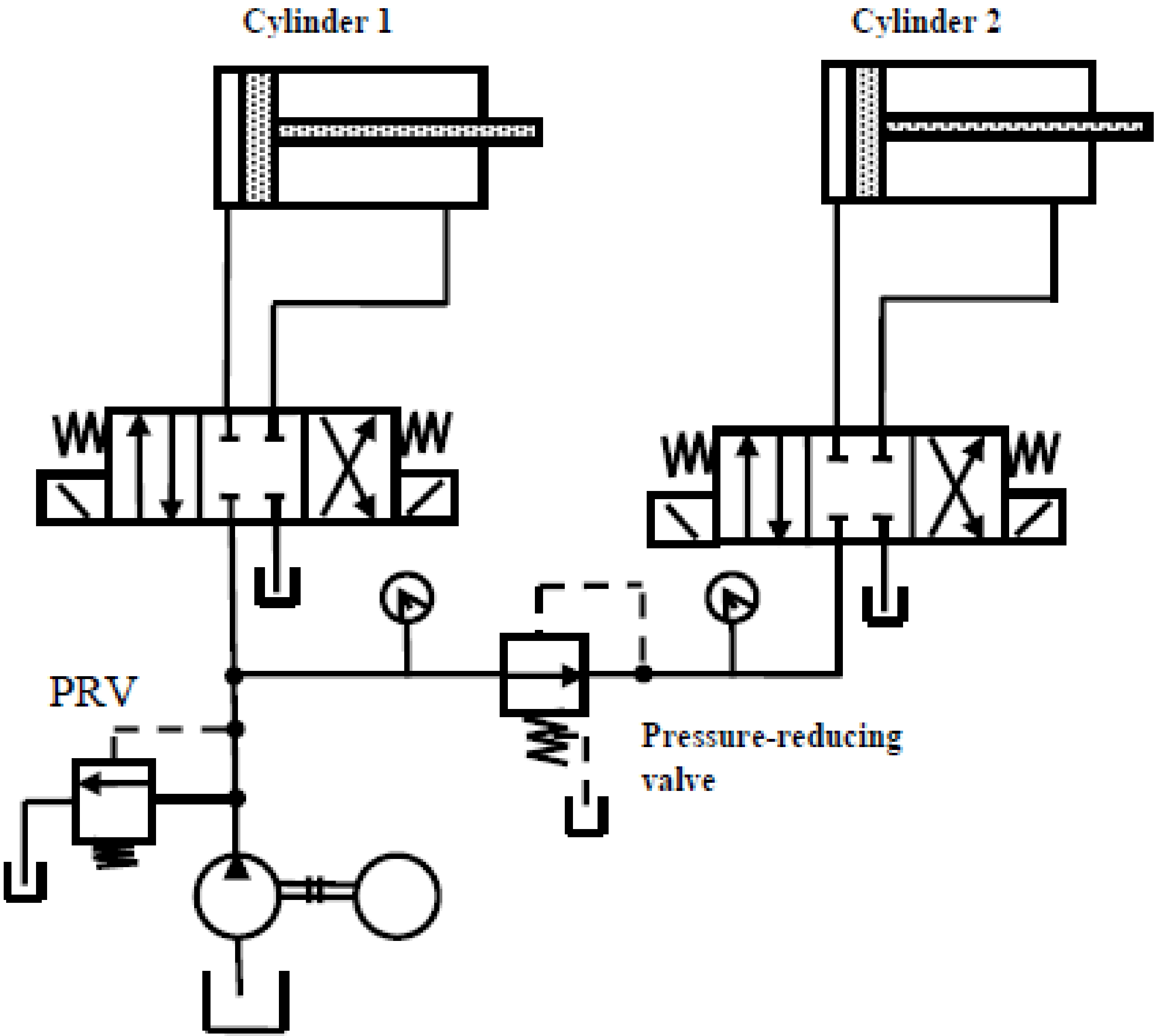


Figure 1.9 Application of a pressure-reducing valve.

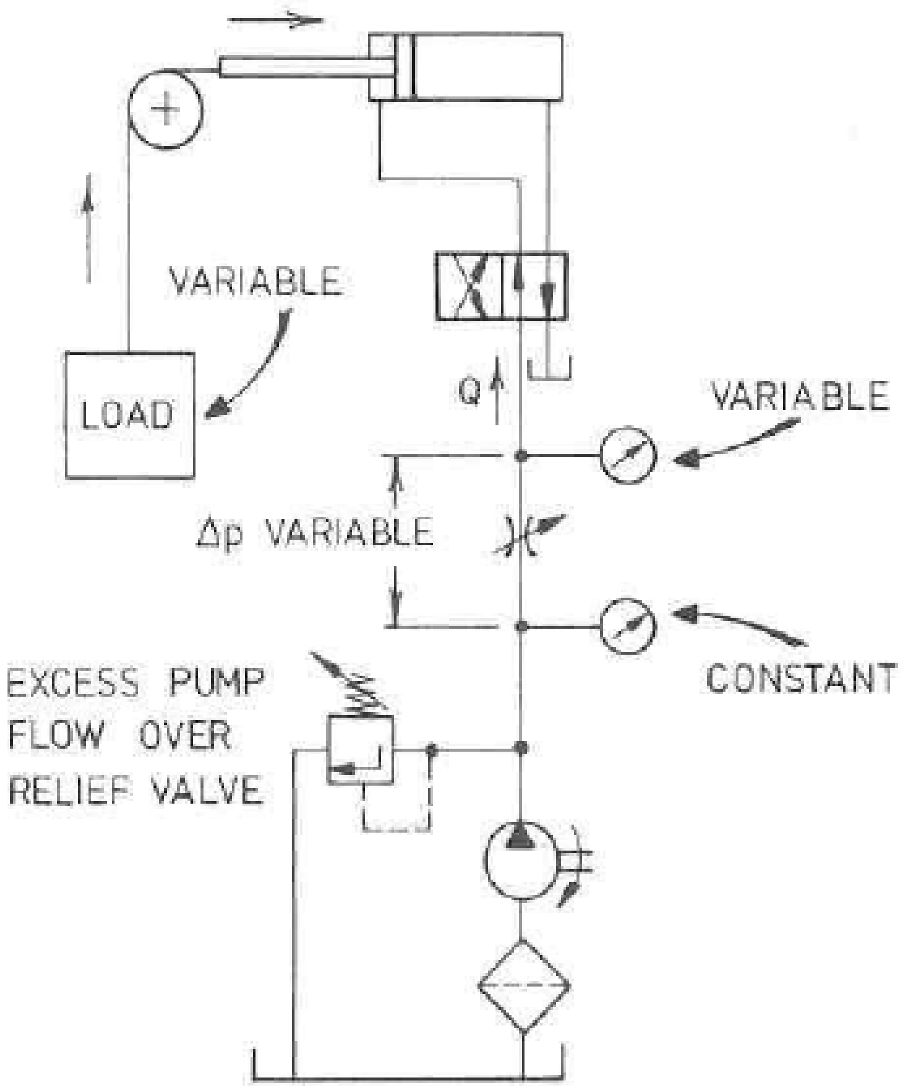
Flow Controls

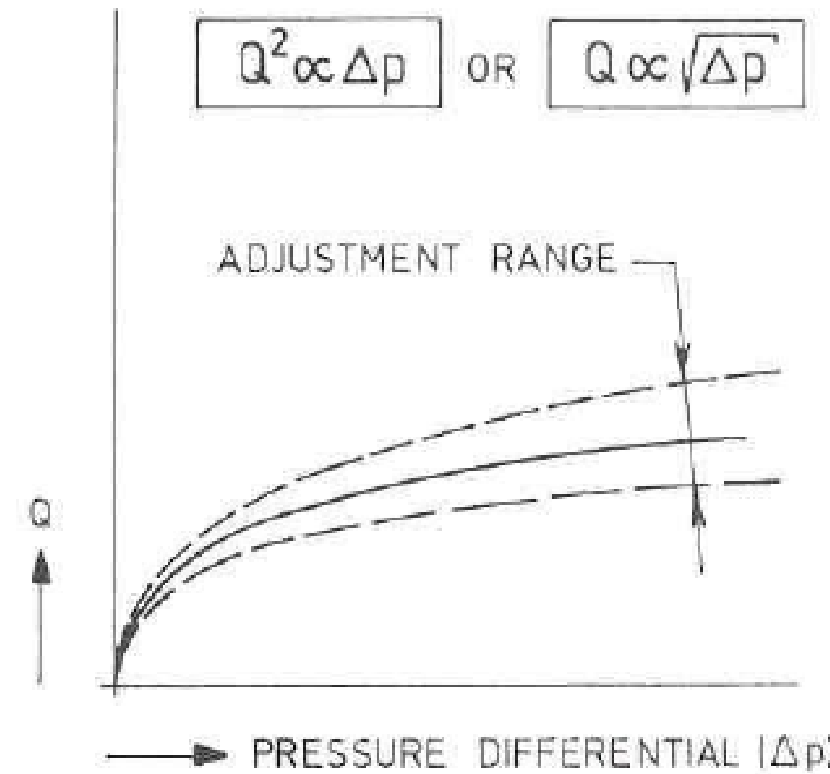
Flow-control valves are used in hydraulic systems to control the rate of flow from one part of the system to another. Flow-control devices accomplish one or more of the following control functions:

- limit the maximum speed of linear actuators and hydraulic motors

  $$\left(\frac{\text{Flowrate}}{\text{Piston Area}} = \text{Piston Speed} \right)$$

- limit the maximum power available to subcircuits by controlling the flow to them (power = flowrate x pressure);
- proportionally divide or regulate the pump flow to various branches of the circuit.


VARIABLE
LOAD
Q
VARIABLE
Δp VARIABLE
CONSTANT
EXCESS PUMP
FLOW OVER
RELIEF VALVE


$Q^2 \propto \Delta p$
OR
$Q \propto \sqrt{\Delta p}$
ADJUSTMENT RANGE
Q
PRESSURE DIFFERENTIAL (Δp)

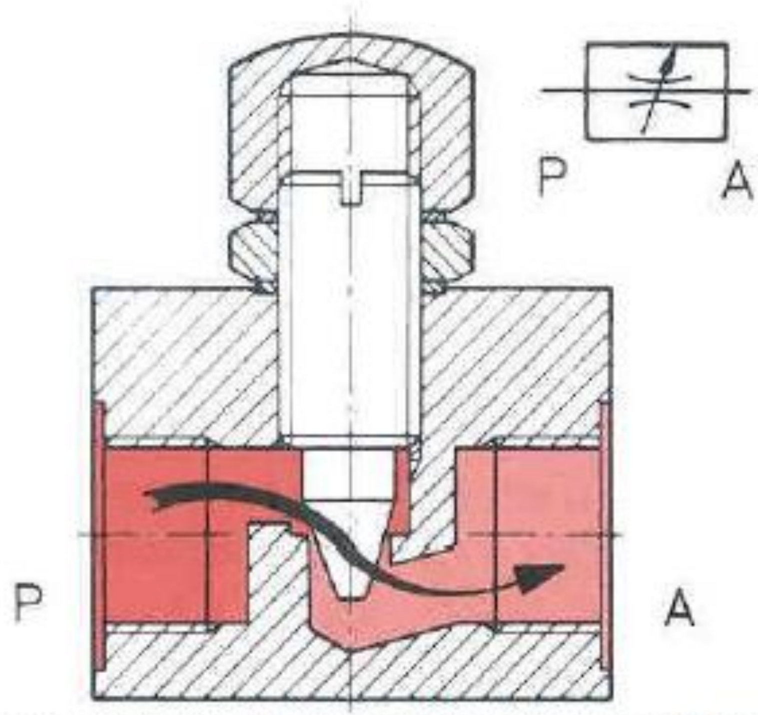


Fig. 120 Simple restrictor flow control valve.

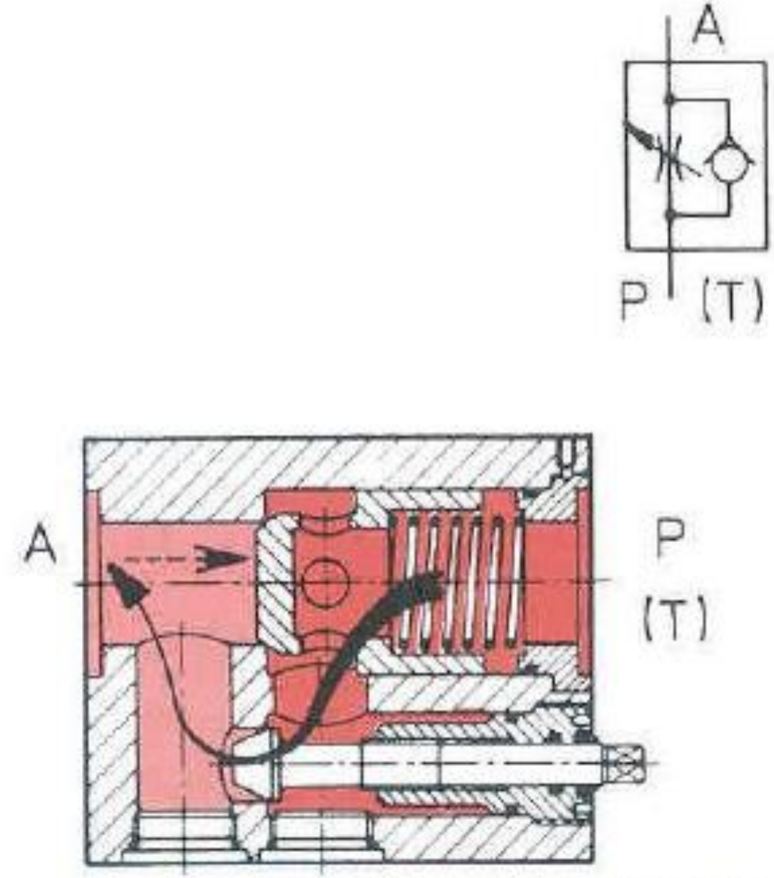
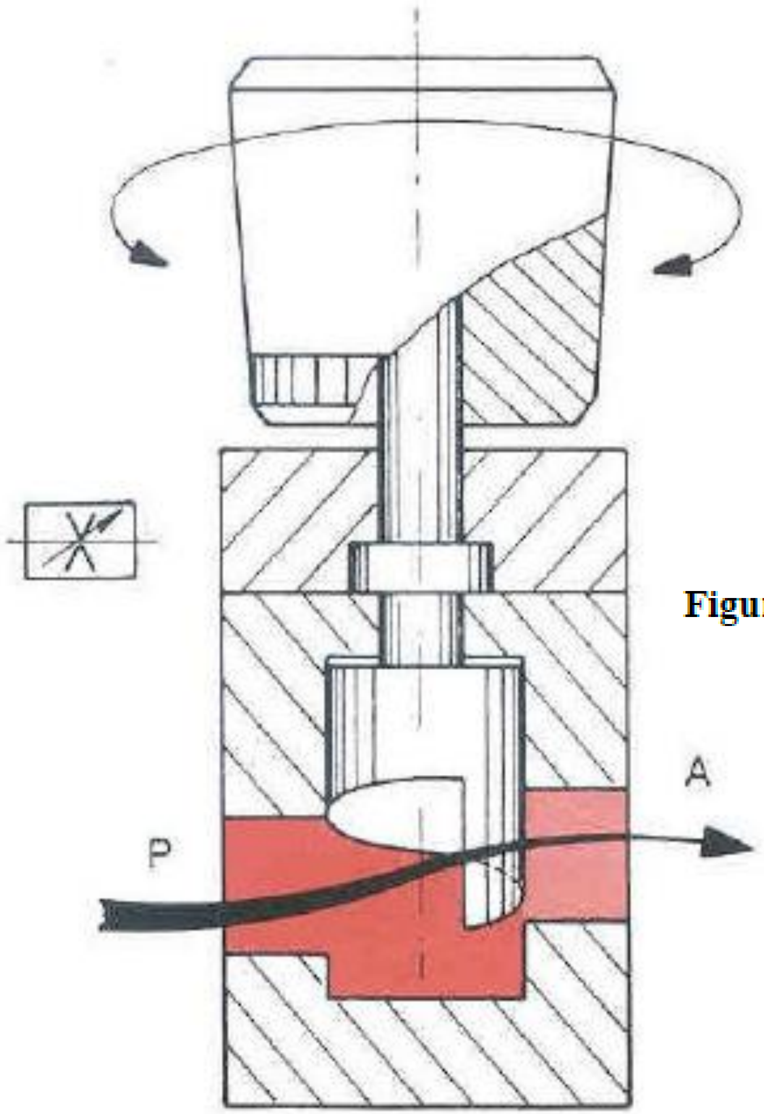


Fig. 121 Simple restrictor with reverse free-flow check



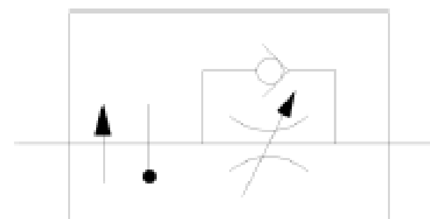


Figure 1.7 Pressure- and temperature-compensated flow-control valve.

Fig. 123 Temperature compensated flow control valve.

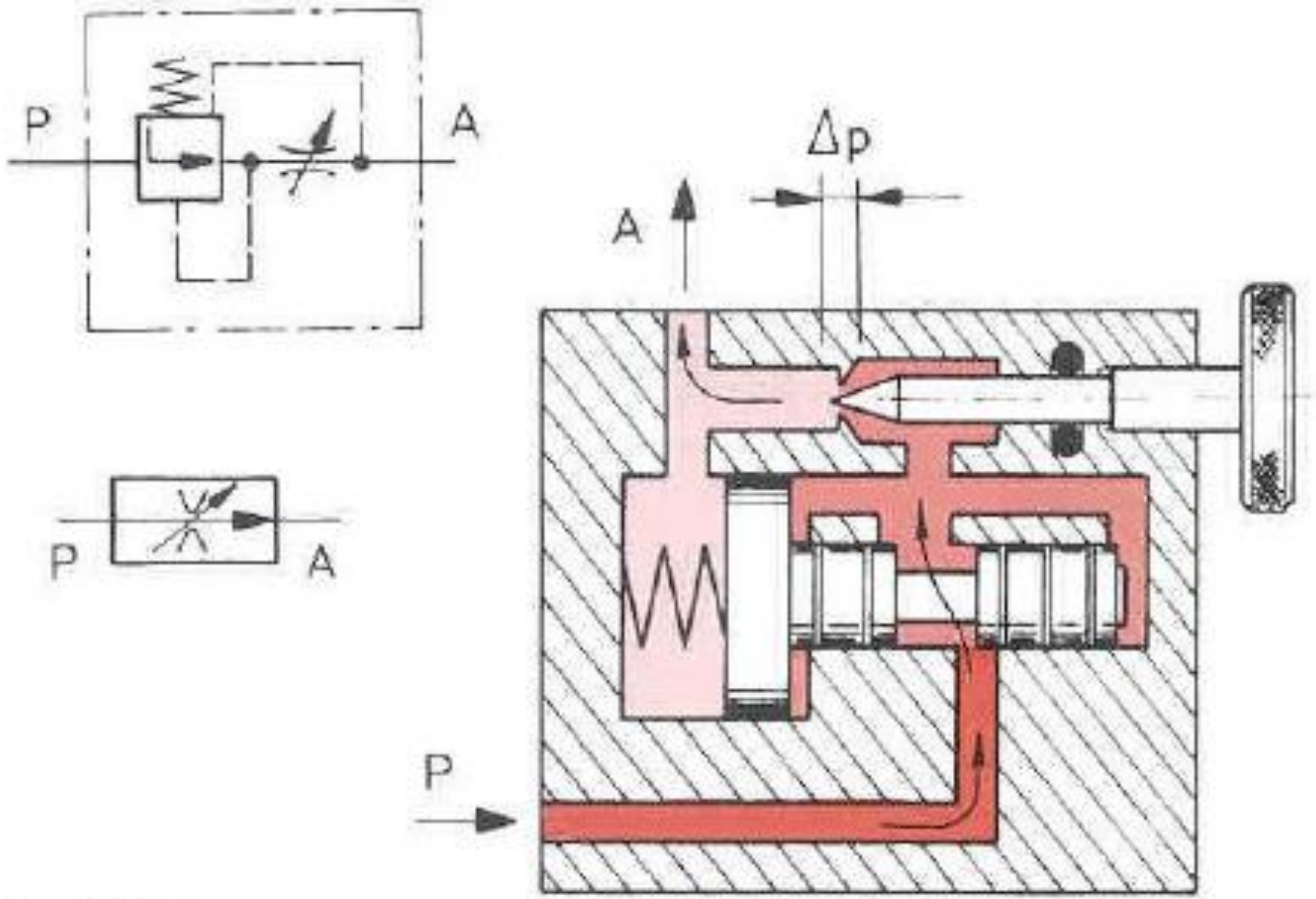


Fig. 124 Pressure compensated flow control valve.

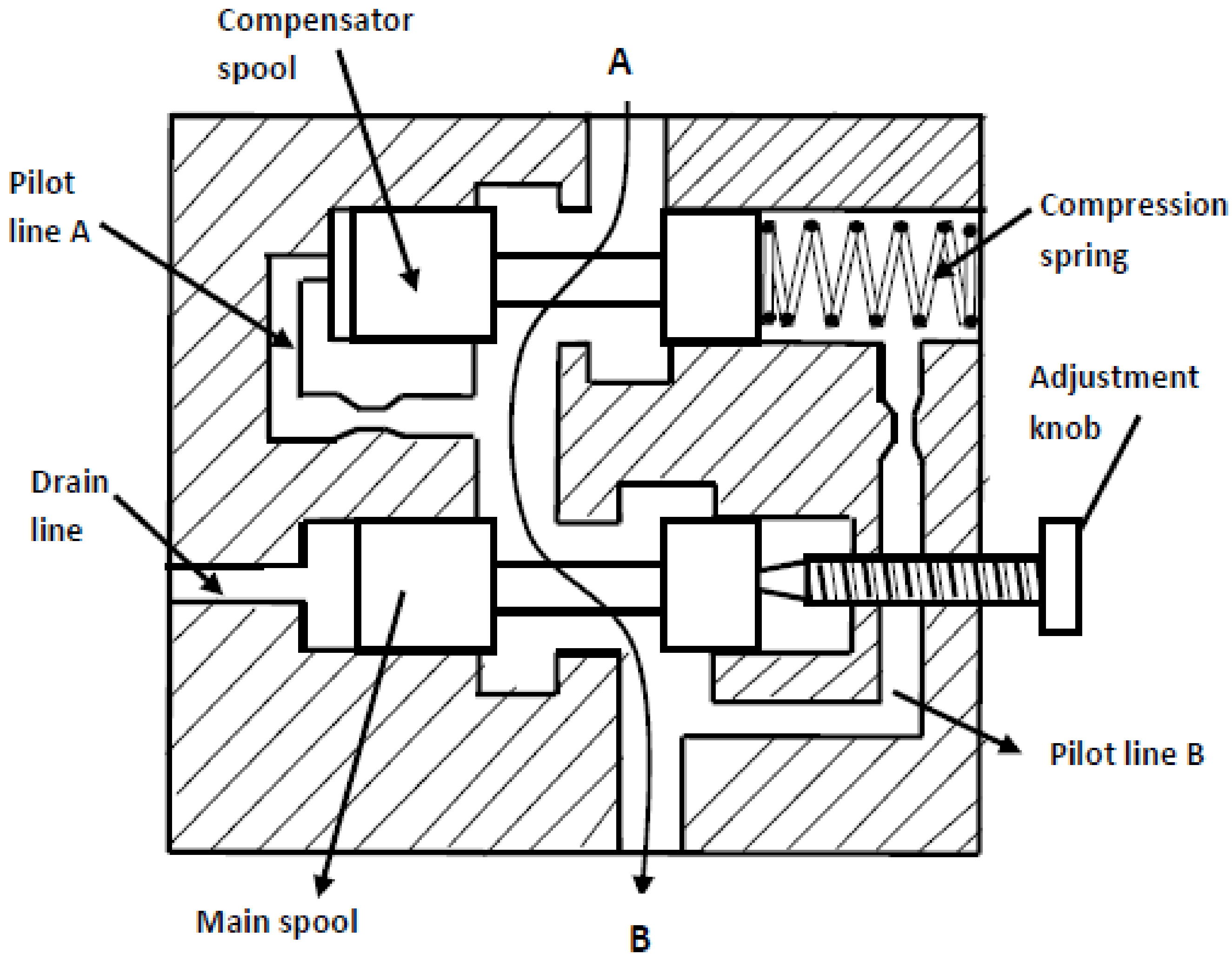


Figure 1.5 Sectional view of a pressure-compensated flow-control valve.

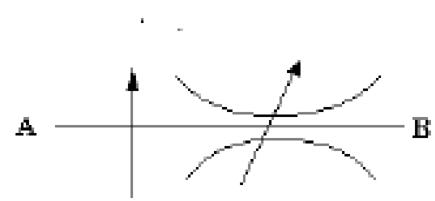


Figure 1.6 Graphic symbol of a pressure-compensated flow-control valve.

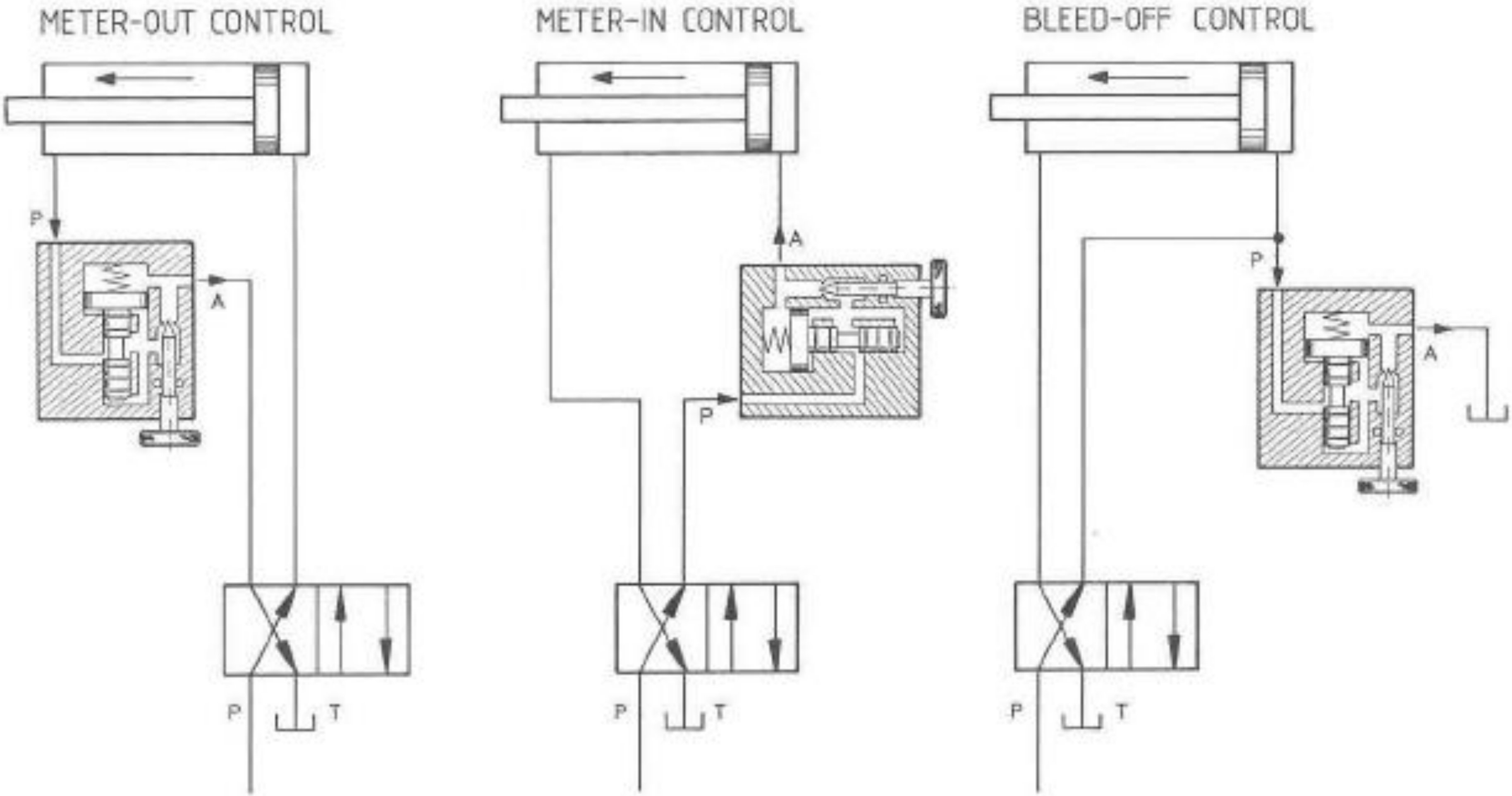


Fig 125 Flow control methods for restrictor type valves

Meter-out flow control
This speed control method is highly accurate, and used wherever a free-falling load or overhauling load tends to get out of control ("runaway" condition).

The flow control valve is located between the actuator and the directional-control valve, and controls the exhaust fluid from the actuator. If both actuator strokes are to be controlled, the valve can be installed in the tank line of the directional-control valve. But one must beware of the pressure intensification exceeding the T port rating of the directional-control valve if actuators with oversize piston rods are used. If only one stroke is to be speed controlled, a reverse free-flow check valve would be required for rapid retraction.

As a disadvantage it must be mentioned, that excess pump flow, which cannot pass through the flow control valve, is pushed over the system relief valve.
Meter-in flow control
This speed control method is also highly accurate, and is used where the load on the actuator resists the stroke at all times (no "runaway" condition).

The flow control valve is located in the feed-line on the actuator and where only one stroke is to be speed con- trolled, a reverse free-flow check valve would be required to provide rapid retraction. If both actuator strokes are to be speed controlled, the flow control valve may be installed between the pump and the directional-control valve. However, stroke speeds could then not be regulated individually.

Here too, excess pump flow is pushed over the system relief valve.

Bleed-off flow control

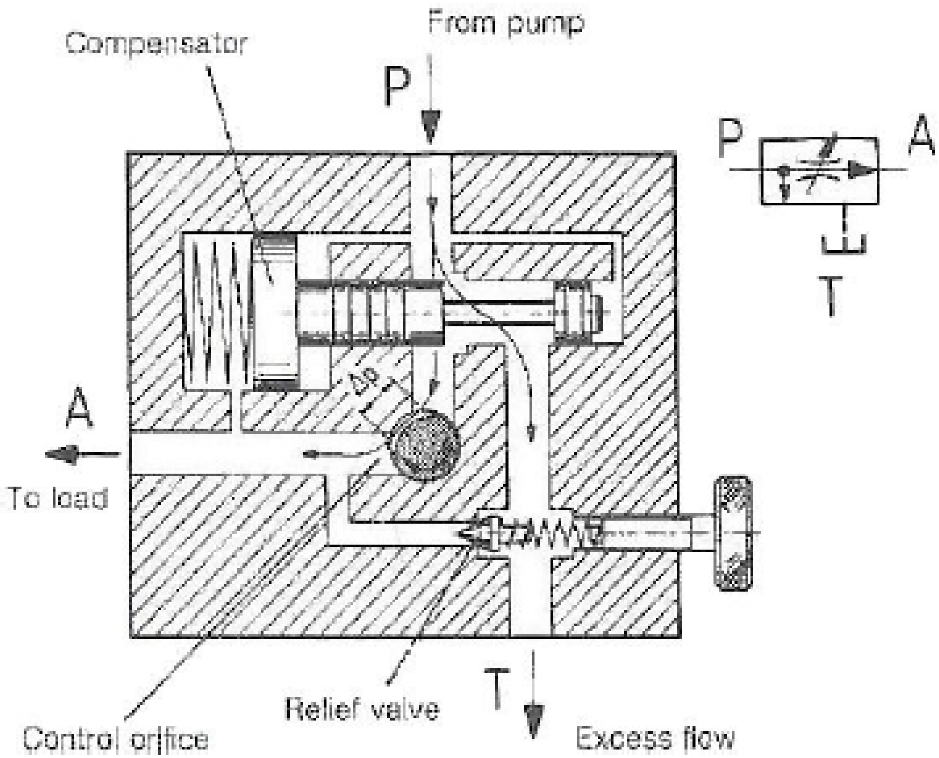
This speed control method has a power-saving advantage, as the pump operates always at the pressure required by the workload, and the excess pump flow returns via the flow control valve to tank, without being pushed over the relief valve.

The method is not as accurate as meter-in, since the measured flow goes to tank and the remaining flow into the actuator. This makes the actuator speed subject to varying pump-delivery.

Bleed-off flow control does not require a reverse free- flow check valve. It must be noted, that bleed-off control is not suitable for "runaway" load conditions.

By-pass flow control valve

This valve controls flow to an actuator and diverts any excess (surplus) flow to tank (figs. 126 and 127). The in- built pressure relief valve is an additional feature of the by-pass valve and provides overload protection from excessive workload pressure build-up.



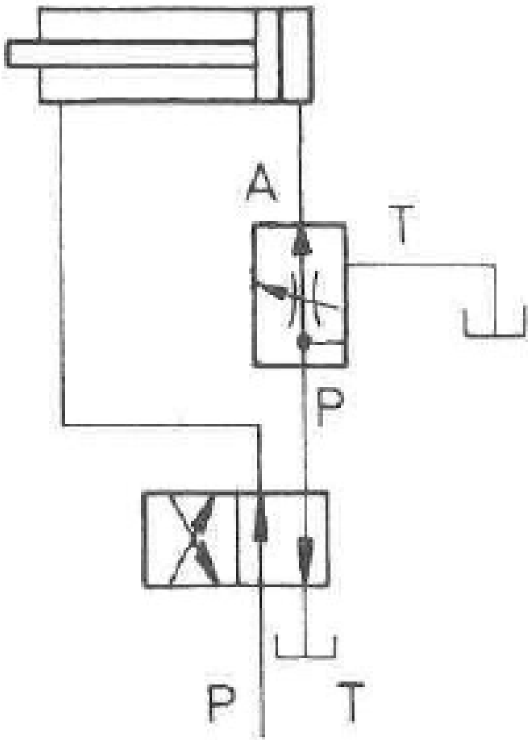


Fig. 126 left: By-pass flow control valve.

Fig. 127 above: By-pass flow control valve in a meter-in control application

Example 1.1

A 55-mm diameter sharp-edged orifice is placed in a pipeline to measure the flow rate. If the measured pressure drop is 300 kPa and the fluid specific gravity is 0.90, find the flow rate in units of m^3/s.

Solution: For a sharp-edged orifice, we can write

$$Q = 0.0851\, A\, C_V \sqrt{\frac{\Delta p}{SG}}$$

where Q is the volume flow rate in LPM, C_V is the capacity coefficient = 0.80 for the sharp-edge orifice, c = 0.6 for a square-edged orifice, A is the area of orifice opening in mm^2, Δp is the pressure drop across the orifice (kPa) and SG is the specific gravity of the flowing fluid = 0.9. Now,

$$A_{\text{orifice}} = \frac{\pi}{4}(D_{\text{orifice}}{}^2) = \frac{\pi}{4}(55^2) = 2376\ \text{mm}^2$$

Using the orifice equation we can find the flow rate as

$$Q\ (\text{LPM}) = 0.0851 \times 2376 \times 0.80 \sqrt{\frac{300}{0.9}}$$

$$= 2953.3\ \text{LPM} = 0.0492\ \ \text{m}^3/\text{s}$$

Example 1.4

The system shown in Fig. 1.12 has a hydraulic cylinder with a suspended load W. The cylinder piston and rod diameters are 50.8 and 25.4 mm, respectively. The pressure-relief valve setting is 5150 kPa. Determine the pressure p_2 for a constant cylinder speed:

(a) $W = 8890$ N
(b) $W = 0$ (load is removed)
(c) Determine the cylinder speeds for parts (a) and (b) if the flow-control valve has a capacity coefficient of $0.72\,\text{LPM}/\sqrt{\text{kPa}}$. The fluid is hydraulic oil with a specific gravity of 0.90.

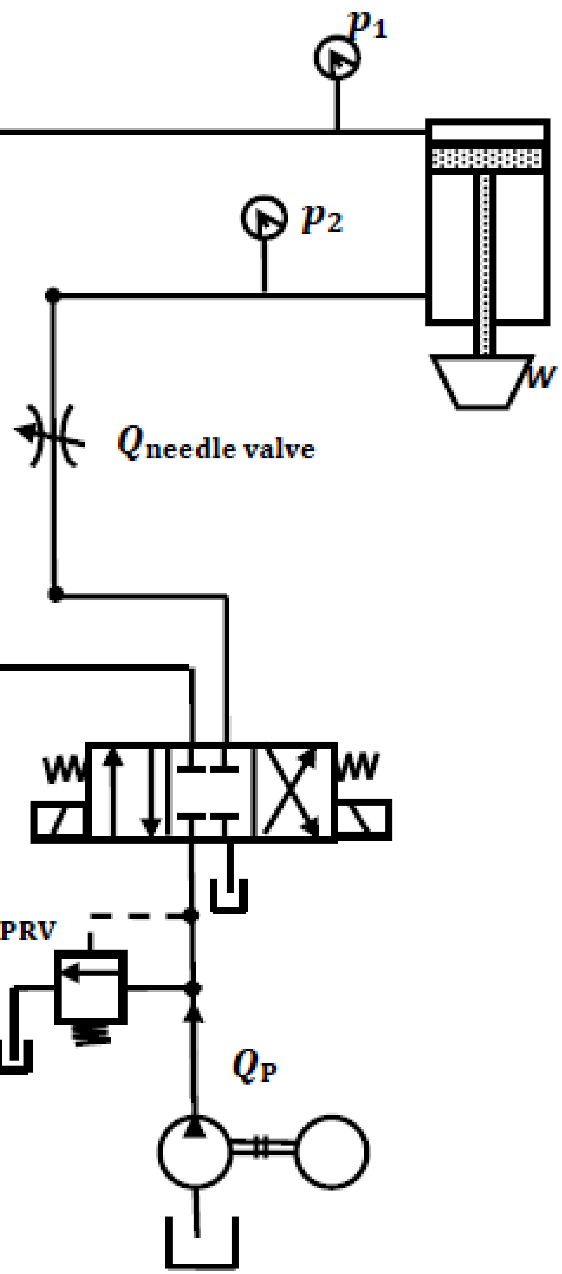


Figure 1.12 Hydraulic cylinder with a suspended weight.

$$A_p = \frac{\pi}{4}(D_p^2) = \frac{\pi}{4}(0.0508^2) = 0.00203\,\text{m}^2$$

$$A_r = \frac{\pi}{4}(D_R^2) = \frac{\pi}{4}(0.0254^2) = 0.000506\,\text{m}^2$$

So

$$A_p - A_r = 0.00152\ \text{m}^2$$

Case 1: If $W = 8890$ N.

$$-W - p_1 A_p + p_2(A_p - A_r) = 0$$

$$\Rightarrow -8890 - 5150\times 10^3\,\text{N/m}^2 \times 2.03\times 10^{-3}\ \text{m}^2 + p_2(0.00152\ \text{m}^2) = 0$$

$$\Rightarrow -8890 - 10450\,\text{m}^2 + p_2(0.00152) = 0$$

$$\Rightarrow p_2 = 12700\,\text{kPa}$$

Case 2: If $W = 0$.

$$0 - 5150\times 10^3\ \text{N/m}^2 \times 2.03\times 10^{-3}\ \text{m}^2 + p_2(0.00152\,\text{m}^2) = 0$$

$$\Rightarrow p_2 = 6880\,\text{kPa}$$

Case 3: Cylinder speed for case 1: For a sharp-edged orifice, we can write

$$Q = C_V\sqrt{\frac{\Delta p}{SG}} = 0.72\sqrt{\frac{12700}{0.9}} = 85.5\,\text{LPM}$$

where $\Delta p = p_2$ because the flow-control valve discharges directly to the oil tank. This is the flow rate through the flow-control valve and thus the flow rate of the fluid leaving the hydraulic cylinder. Thus, we have

$$v_p(A_p - A_r) = Q$$

$$\Rightarrow v_p\ (\text{m/s})(0.00152)\ \text{m}^2 = 85.5\ \text{L/min} \times \frac{1\ \text{m}^3}{10^3\ \text{L}} \times \frac{1\ \text{min}}{60\text{s}}$$

$$\Rightarrow v_p = 0.938\ \text{m/s}$$

Case 4: Cylinder speed for case 2. We have

$$Q = C_V\sqrt{\frac{\Delta p}{\text{SG}}} = 0.72\sqrt{\frac{6880}{0.9}} = 63\text{LPM}{=}63\,\text{L/min} \times \frac{1\ \text{m}^3}{10^3\ \text{L}} \times \frac{1\ \text{min}}{60\text{s}} = 0.00105\frac{\text{m}^3}{s}$$

Also we can write

$$\begin{aligned} Q &= \text{Velocity}\times\text{Area} \\ &= v_p \times A \\ &= 0.00105\frac{\text{m}^3}{s} \end{aligned}$$

$$\Rightarrow v_p\ (\text{m/s})(0.00152)\ \text{m}^2 = 0.00105$$

$$\Rightarrow v_p = 0.691\ \text{m/s}$$

Deceleration valve

(cam-operated flow control)

Deceleration valves are used to control piston speed during some parts of the actuator stroke. The valve is controlled by means of a cam, and depending on the cam shape, it can either increase or decrease (step-up and step-down) the actuator speed. A normally-open valve reduces flow when the plunger is depressed by the cam, whereas for the normally closed valve, a depression of the plunger increases the flow through the valve (figs. 128A & B).

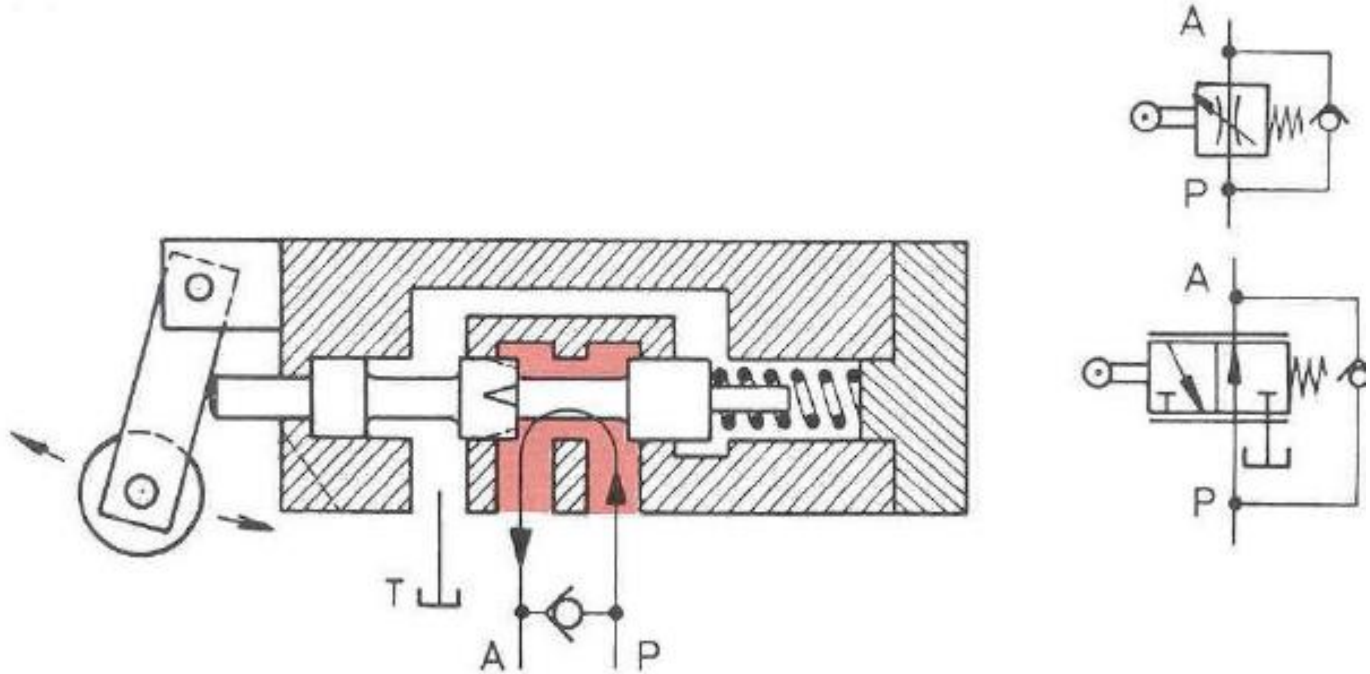


Fig 128A Deceleration valve (normally open)

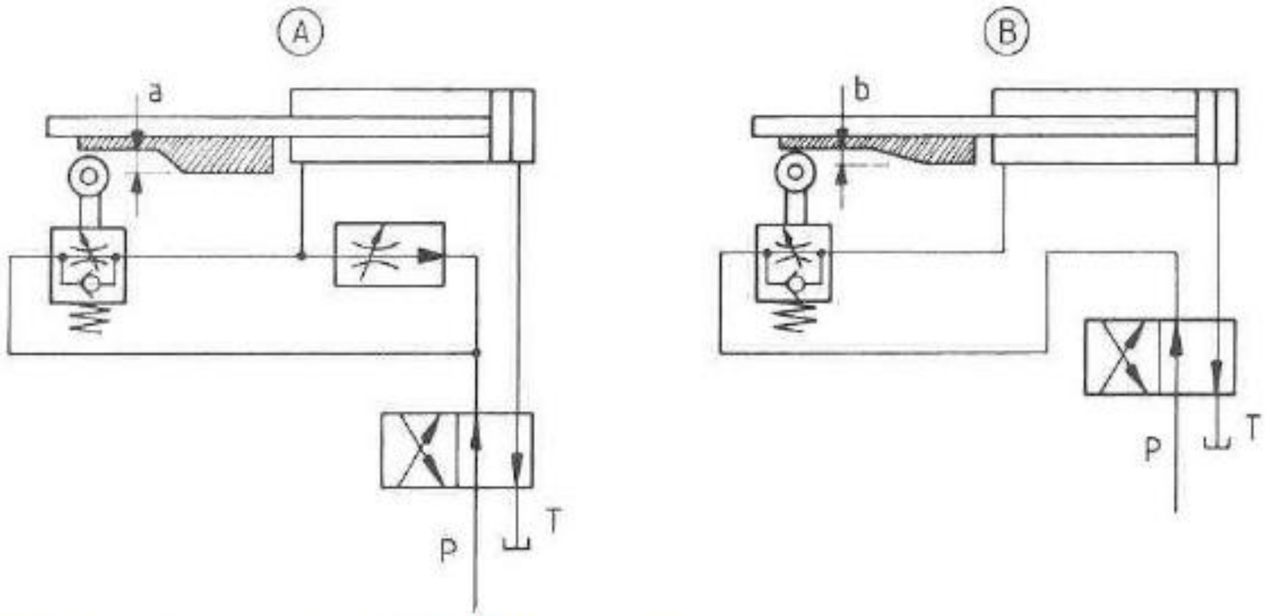


Fig 128B Applications for deceleration valves

Flow divider valves

Flow divider valves are used to split (divide) the flow from a pump into two equal or, where applicable, non- equal parts. The flow division is kept constant, regardless of load fluctuations on the flow outlets.

Priority flow divider valve (Priority flow control valve)

The priority flow divider is used on power steering or power braking on mobile hydraulic equipment. The variable input flow, for example, from a pump being driven off an idling diesel engine, is split into a constant (priority) flow and an excess flow which may be utilised for other functions. The priority flow leads to the power steering, the excess flow to other services on the machine.

Some pump manufacturers integrate the priority flow divider into the pump housing.

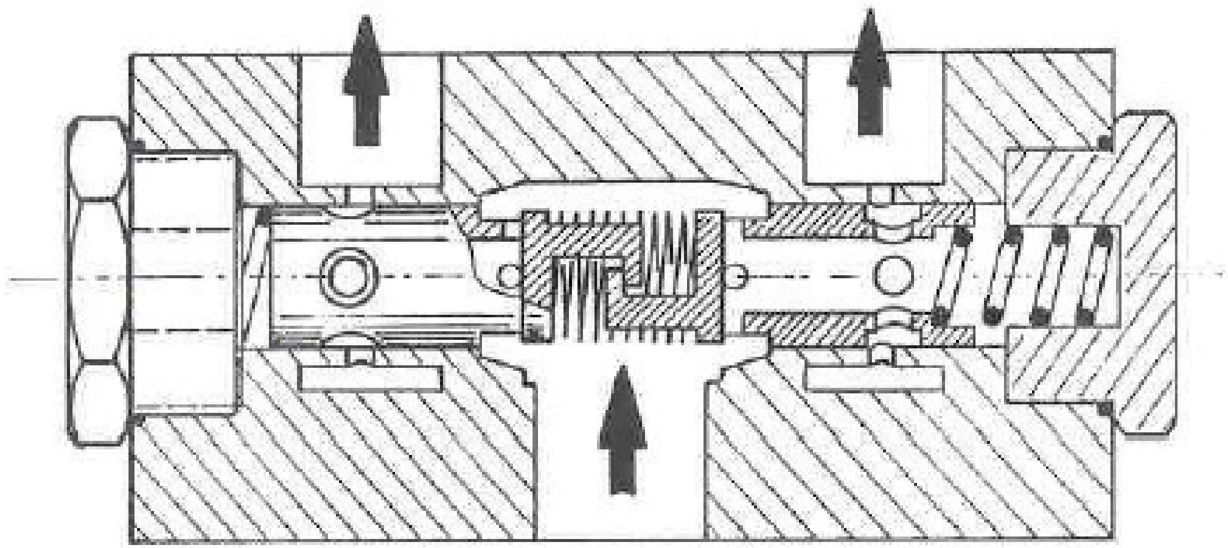


Fig 129 Flow divider valve (flow combiner)

ME 5451 – Hydraulics and Pneumatics

Lecture -13

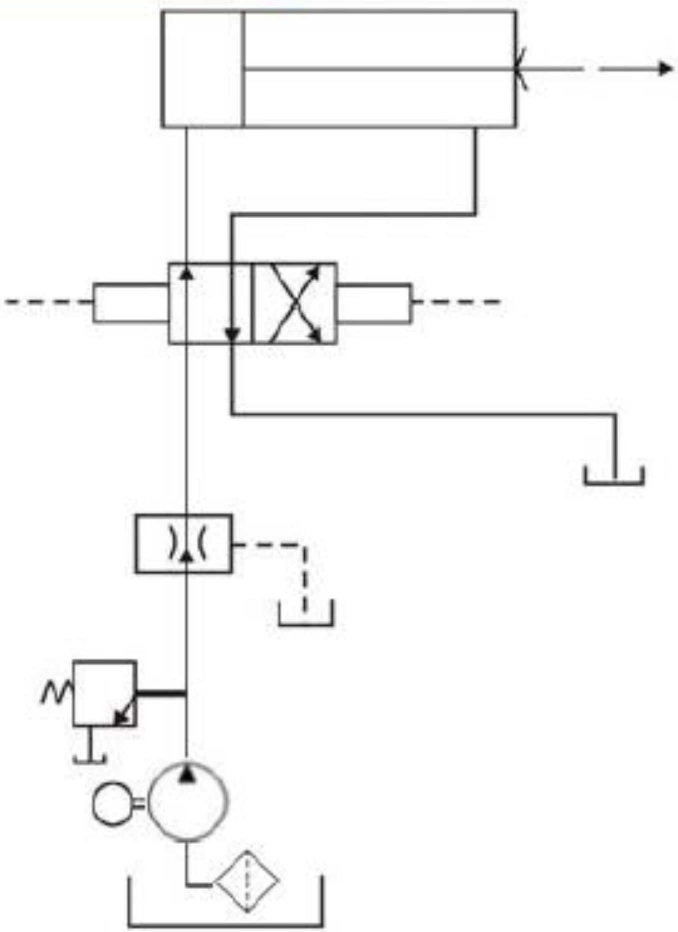
Date: 06-05-2021   Time slot: 08:30-10:10 a.m.

Contents

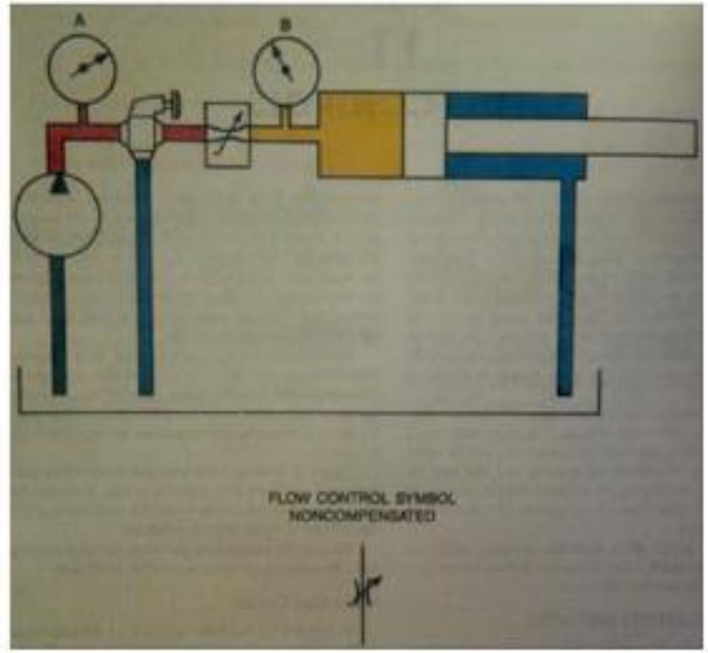
1. Review of Lecture 12
2. Flow Control Valve –Meter in, Meter out, Bleed off
3. Problems on FCV
4. Flow Divider

Course Instructor: Dr. A. Siddharthan

Meter-in-Control



Meter-in-Control
In both directions

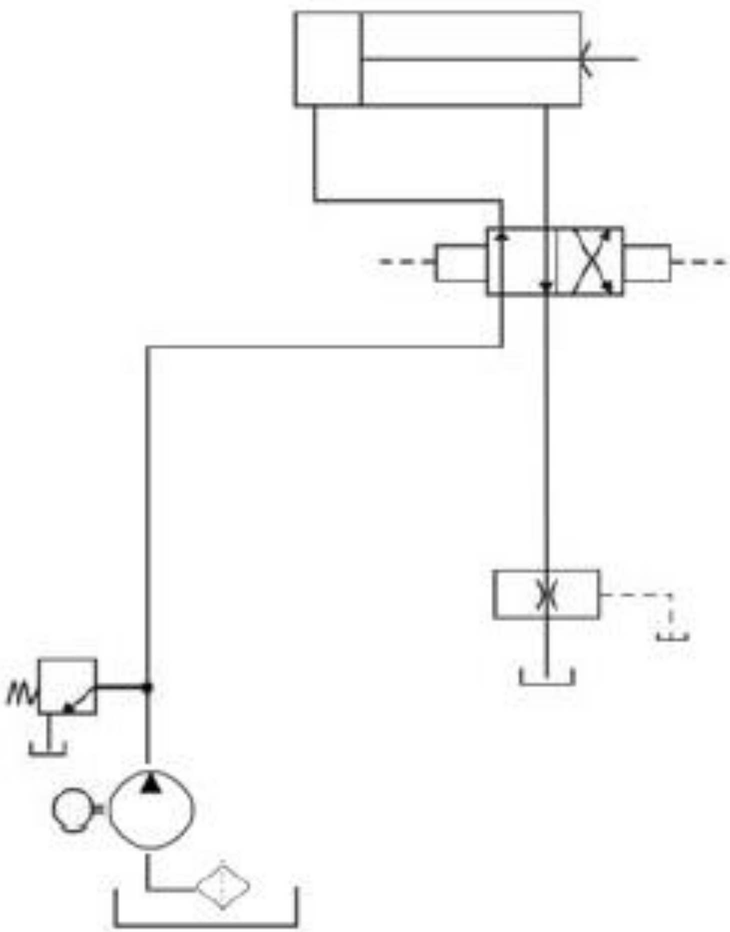


The flow of the fluid can be controlled as it enters the cylinder or motor making it a meter - in –application.

Example- operation of a grinding table or a welding table.

Fig.- 3.9-5 : Meter-in-Control (In both directions).

Meter-out-Control



Meter-out-Control
In both directions

Control of the fluid as it leaves the cylinder is known as a meter-out control .

It is widely used in machine tools often as a quick return mechanism.

Also, it is commonly used on drills.

When the drill point is about to break through it tends to drag, but metering the fluid out of the cylinder prevents the drill from a fast break through of the material being drilled.

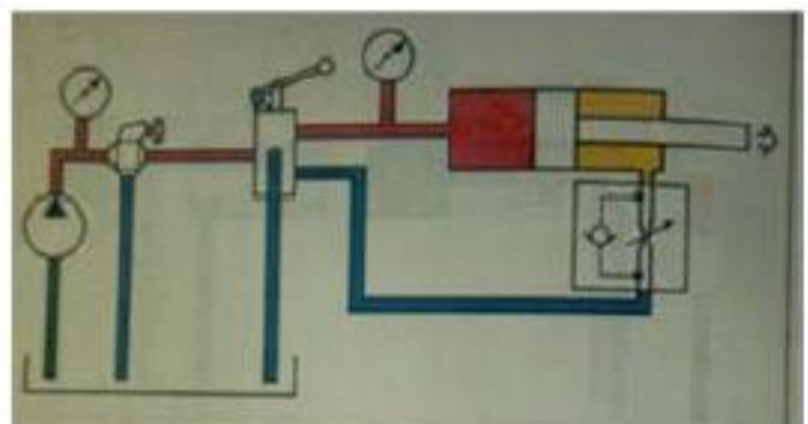
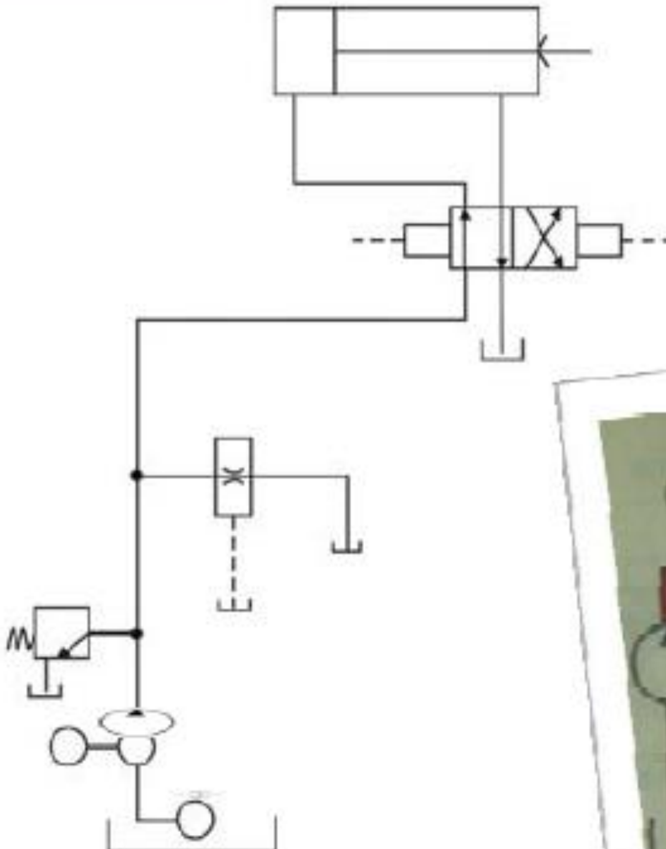


Fig.- 3.9-6 : Meter-out-Control (In both directions).

Bleed-Off-Control

Diverting the flow of fluid to the tank is known as the bleed-off function. Used in planer table or broaching. However, applications are limited.



Bleed-off-Control
In both directions

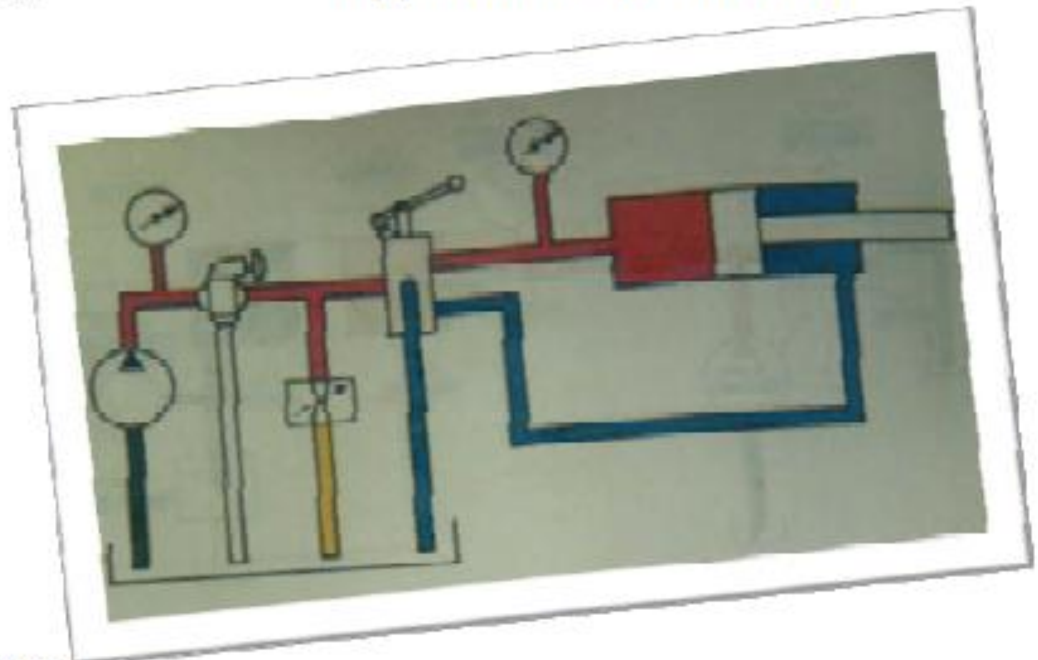
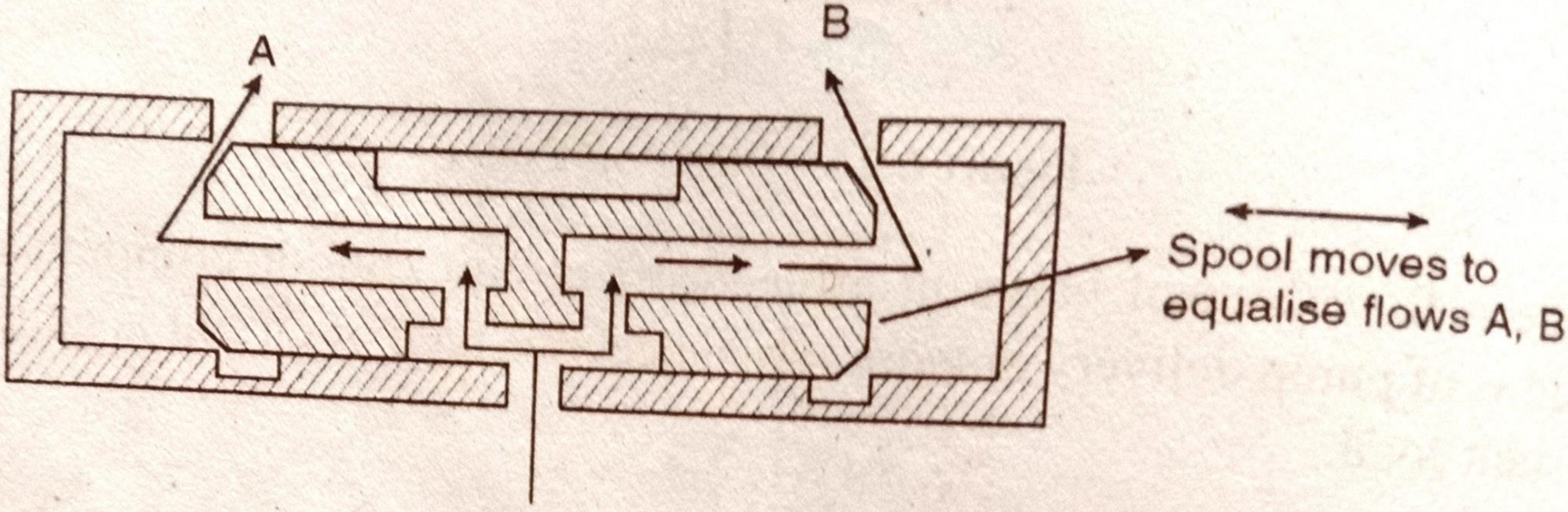
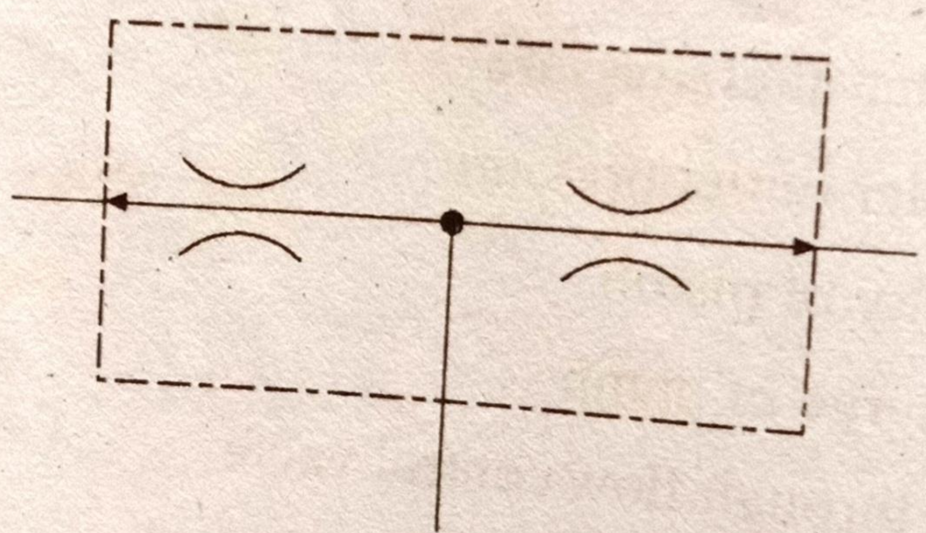


Fig.- 3.9-7: Bleed-off-Control (In both directions).



(a)



(b)

Figure 6.25 (a) Valve Type Flow Divider (b) Sym

Motor type flow divider is used for dividing one input flow into proportional, multiple branch output flows. It consists of several hydraulic motors connected together mechanically by a common shaft. One input fluid stream is split into as many output streams as there are motor sections in the flow divider. Because all motor sections rotate at the same speed, output flow rates are proportional and depend on the displacements of motor sections. Motor flow dividers can usually handle larger flows than valve type. Greater accuracy can be achieved by coupling piston motors together but this may be expensive.

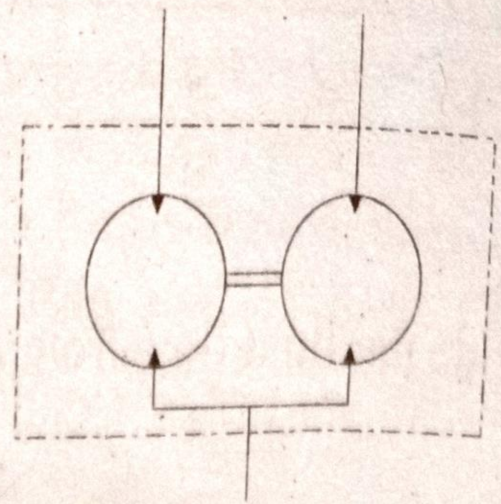


Figure 6.26 Motortype Flow Divider.

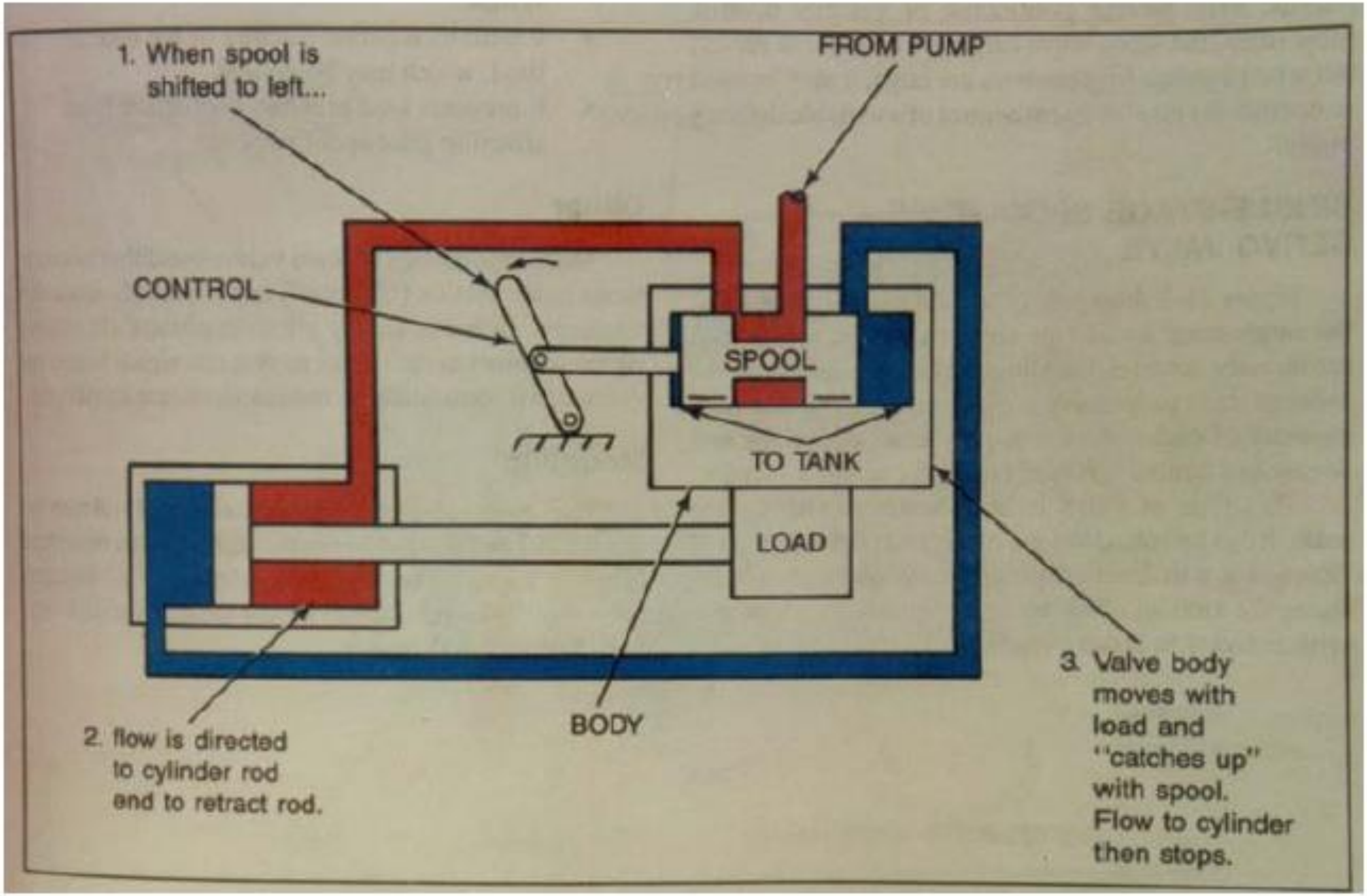
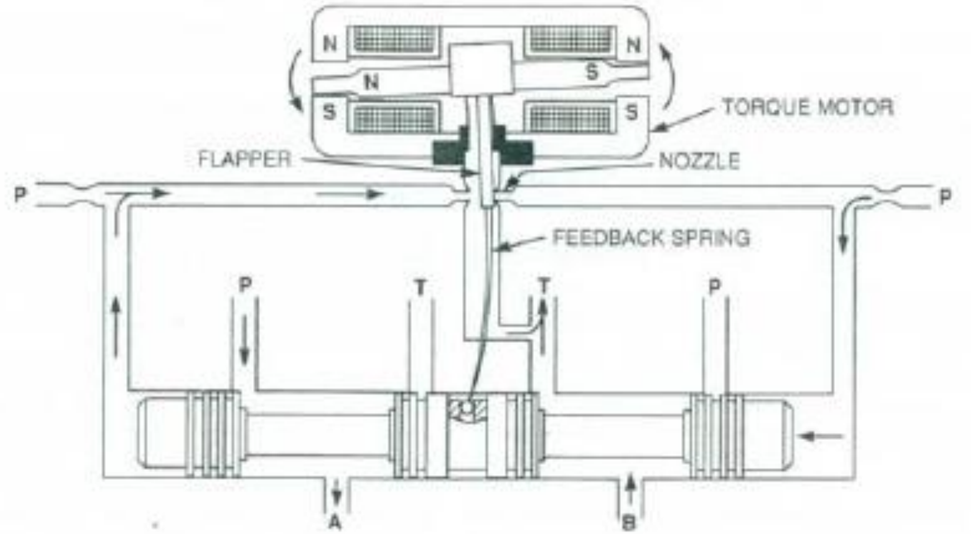


Fig.- 1: Schematic view of Mechanical Servo Valve



(a) Single stage Servo Valves (Flapper nozzle type)

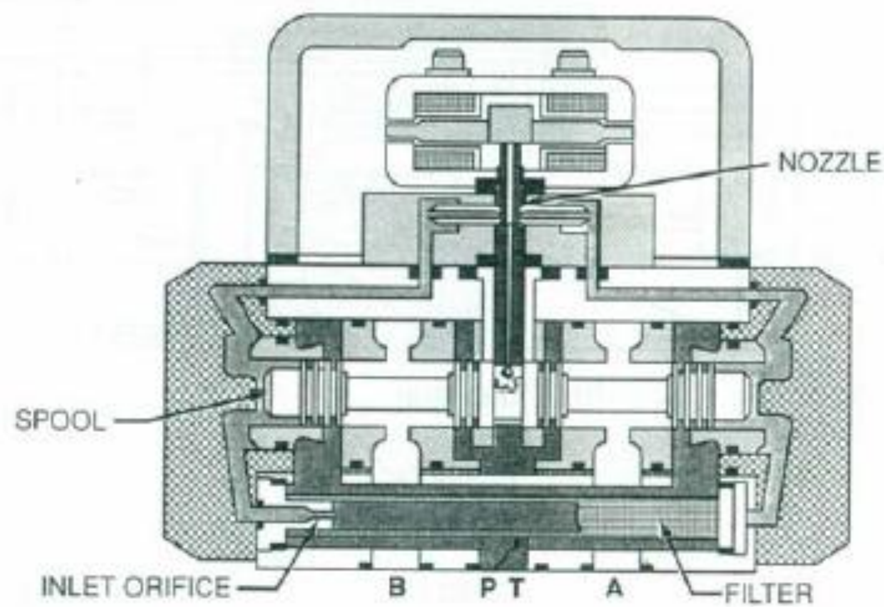


Fig.- 2: Servo Valves

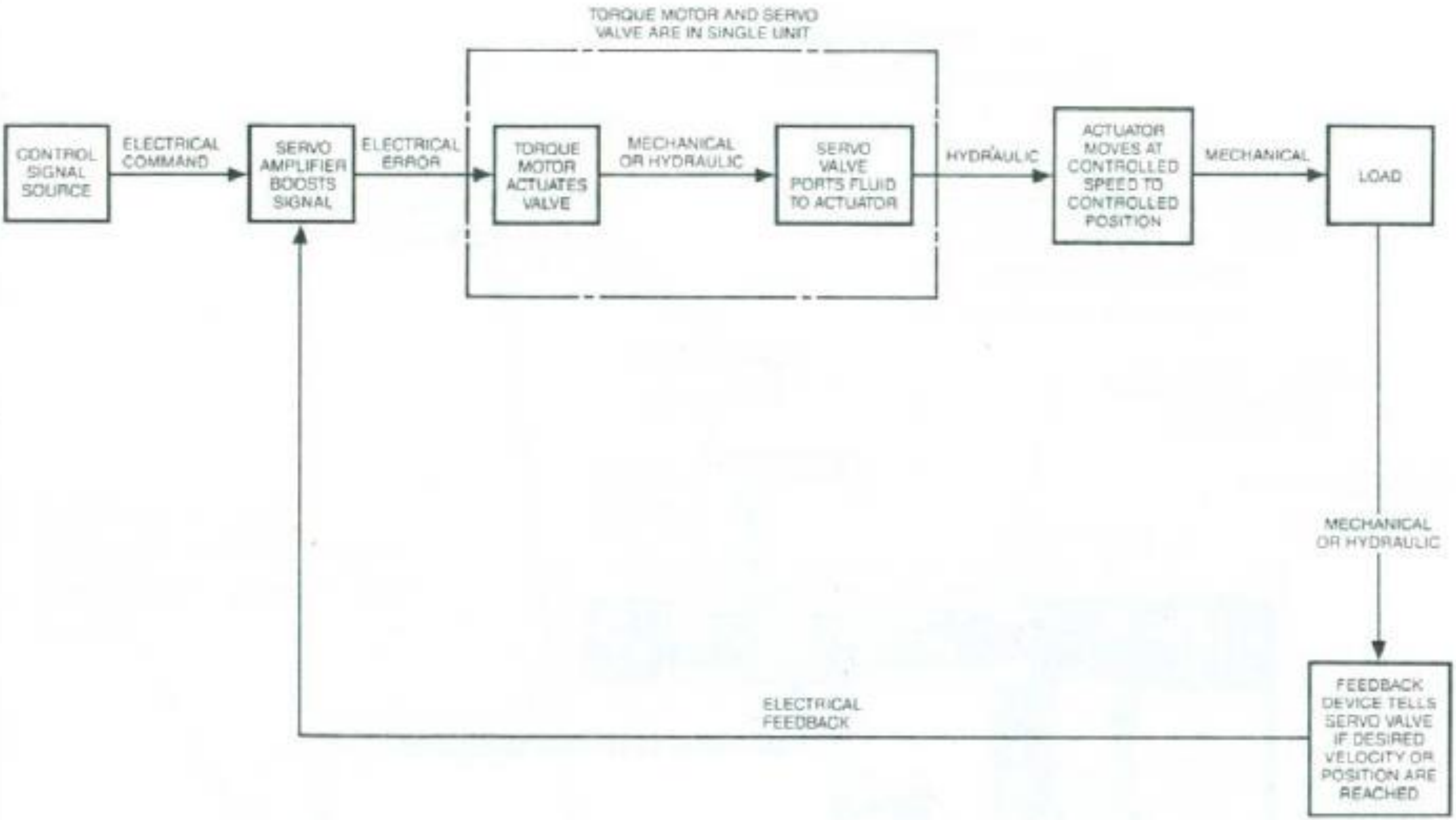
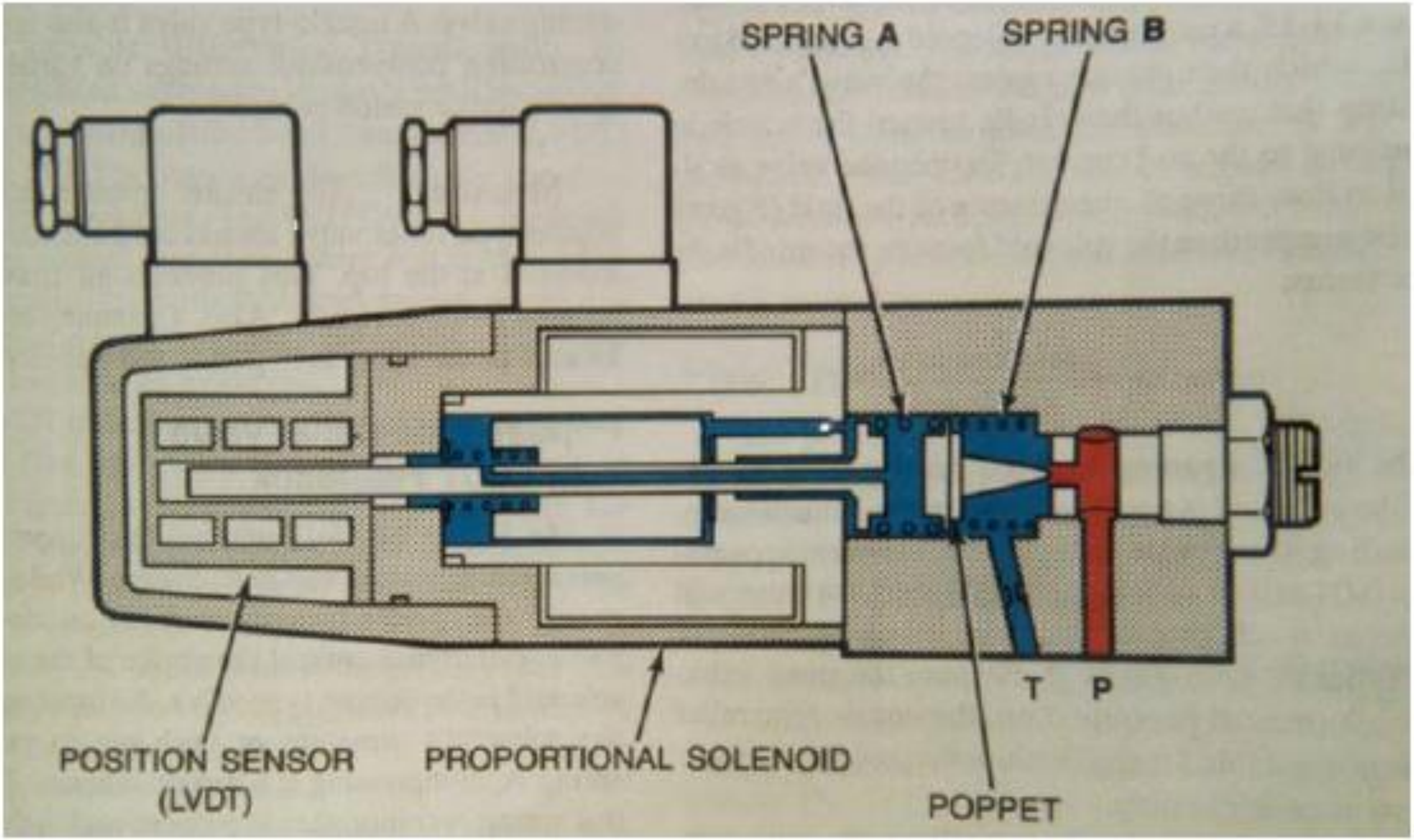
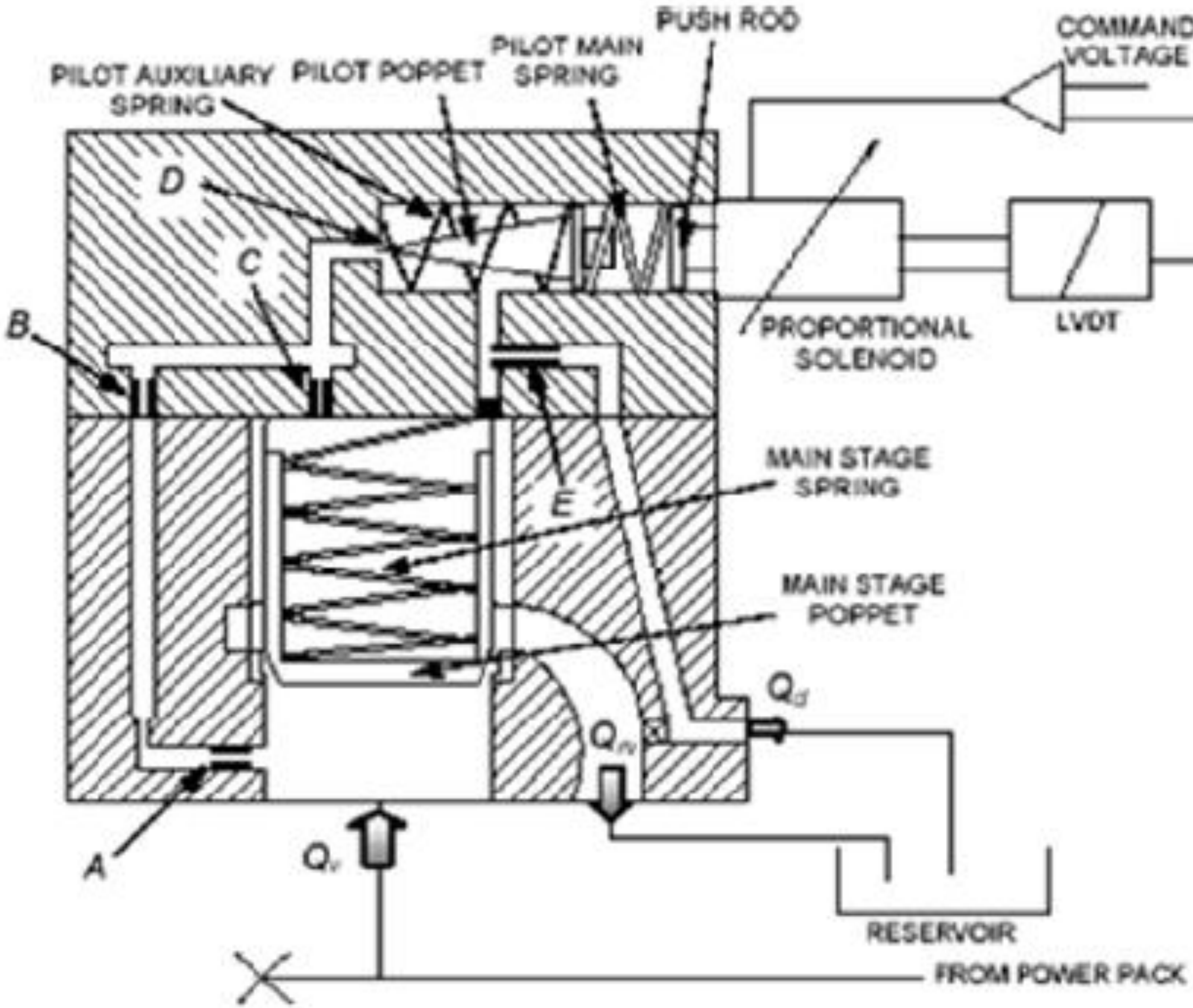


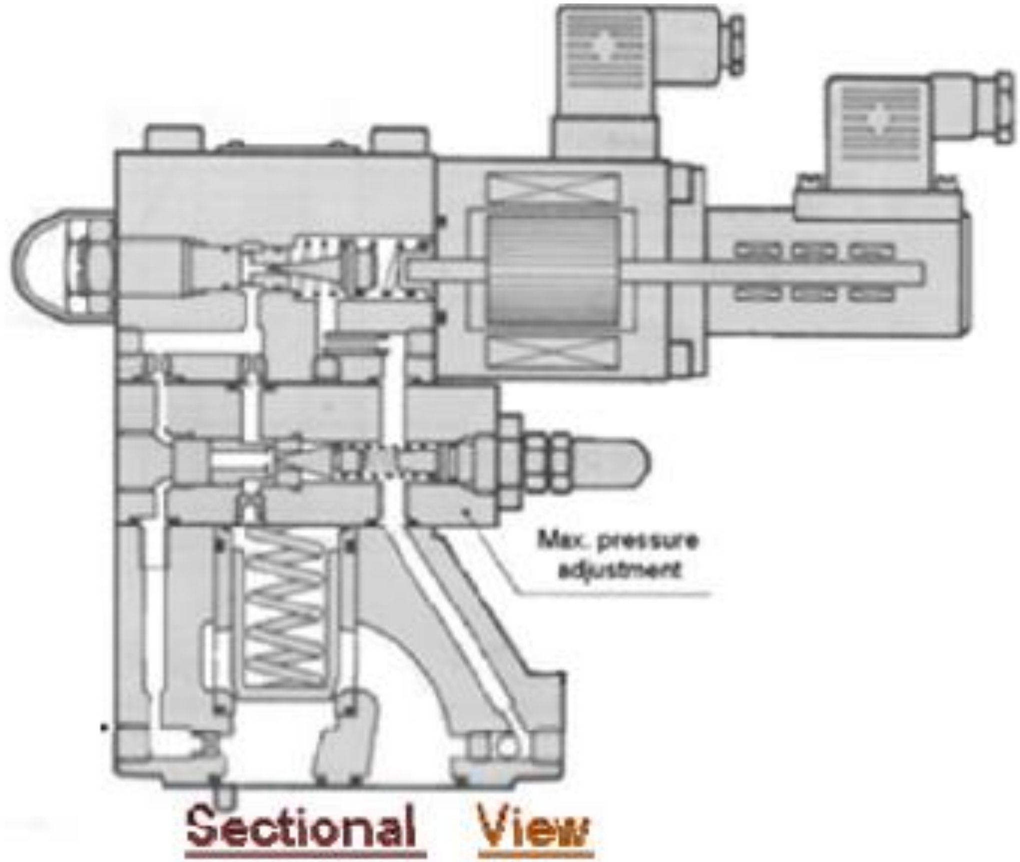
Fig.- 3: Block diagram of a Servo valve system.



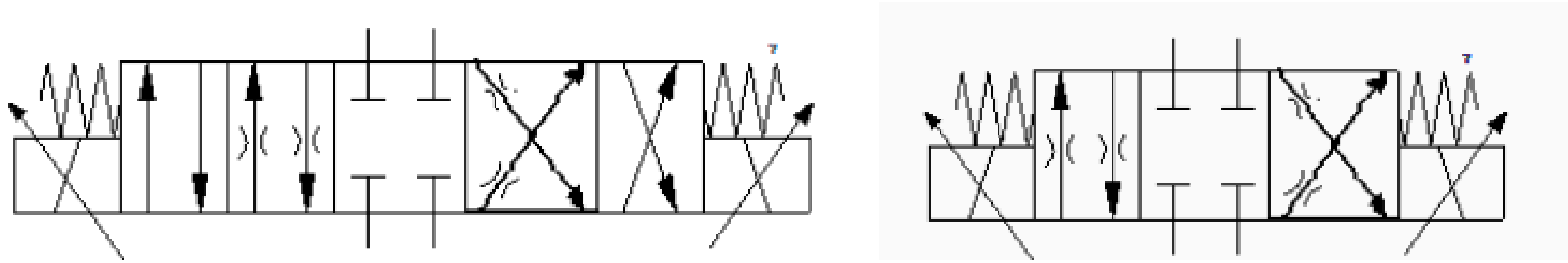
(a) Single stage Proportional Solenoid Pressure Relief Valve with LVDT feedback.



Schematic Diagram



(b) Two stage Proportional Solenoid Pressure Relief Valve with LVDT feedback
And Maximum Relief setting.



(a) (b)

Figure 1.11 The symbols of a proportional directional control valve: (a) Five position;(b) three position .

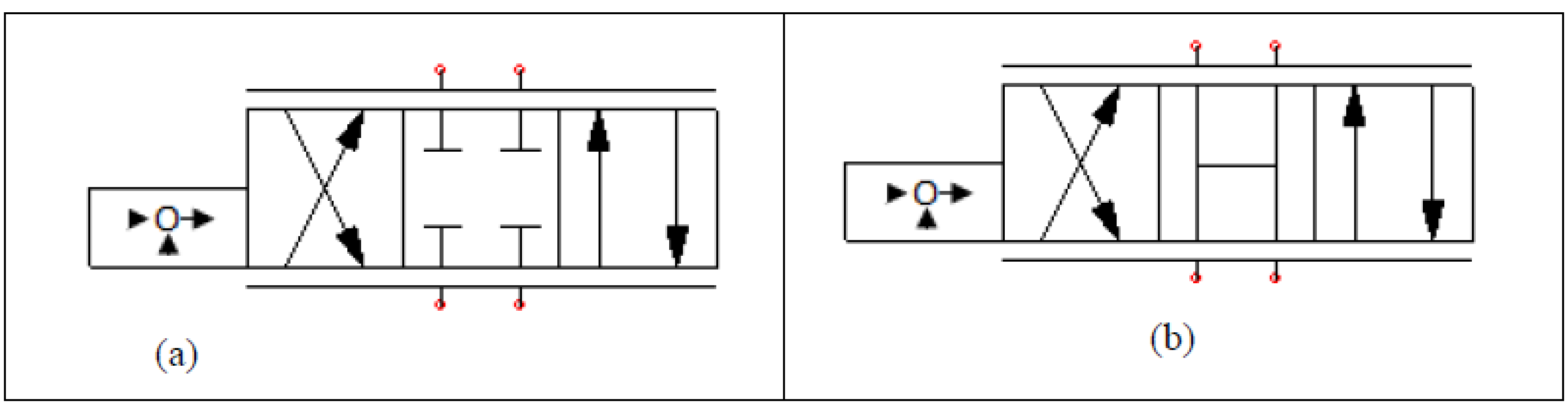


Figure 1.13 Servo symbols: (a) Line-to-line and overlapped spool; (b) underlapped spool.

ME 5451 – Hydraulics and Pneumatics

Lecture -14

Date: 07-05-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Problems on FCV
2. Review of Lecture 13
3. Servo Control Valve
4. Proportional control valve
5. Accessories : Reservoirs, Pressure Switches

Course Instructor: Dr. A. Siddharthan

A cylinder has to exert a forward thrust of 100 kN and a reverse thrust of 10 kN. The effects of using various methods of regulating the extend speed will be considered. In all the cases, the retract speed should be approximately 5 m/min utilizing full pump flow. Assume that the maximum pump pressure is 160 bar and the pressure drops over the following components and their associated pipe work (where they are used):

Filter = 3 bar

Directional control valve (DCV) = 2 bar

Flow-control valve (controlled flow) = 10 bar

Flow-control valve (check valve) = 3 bar

Determine the following:

(a) The cylinder size (assume the piston-to-rod area ratio to be 2:1).
(b) Pump size.
(c) Circuit efficiency when using the following:

Case 1: No flow controls (calculate the extend speed).

Case 2: Meter-in flow control for extend speed 0.5 m/min.

Case 3: Meter-out flow control for extend speed 0.05 m/min.

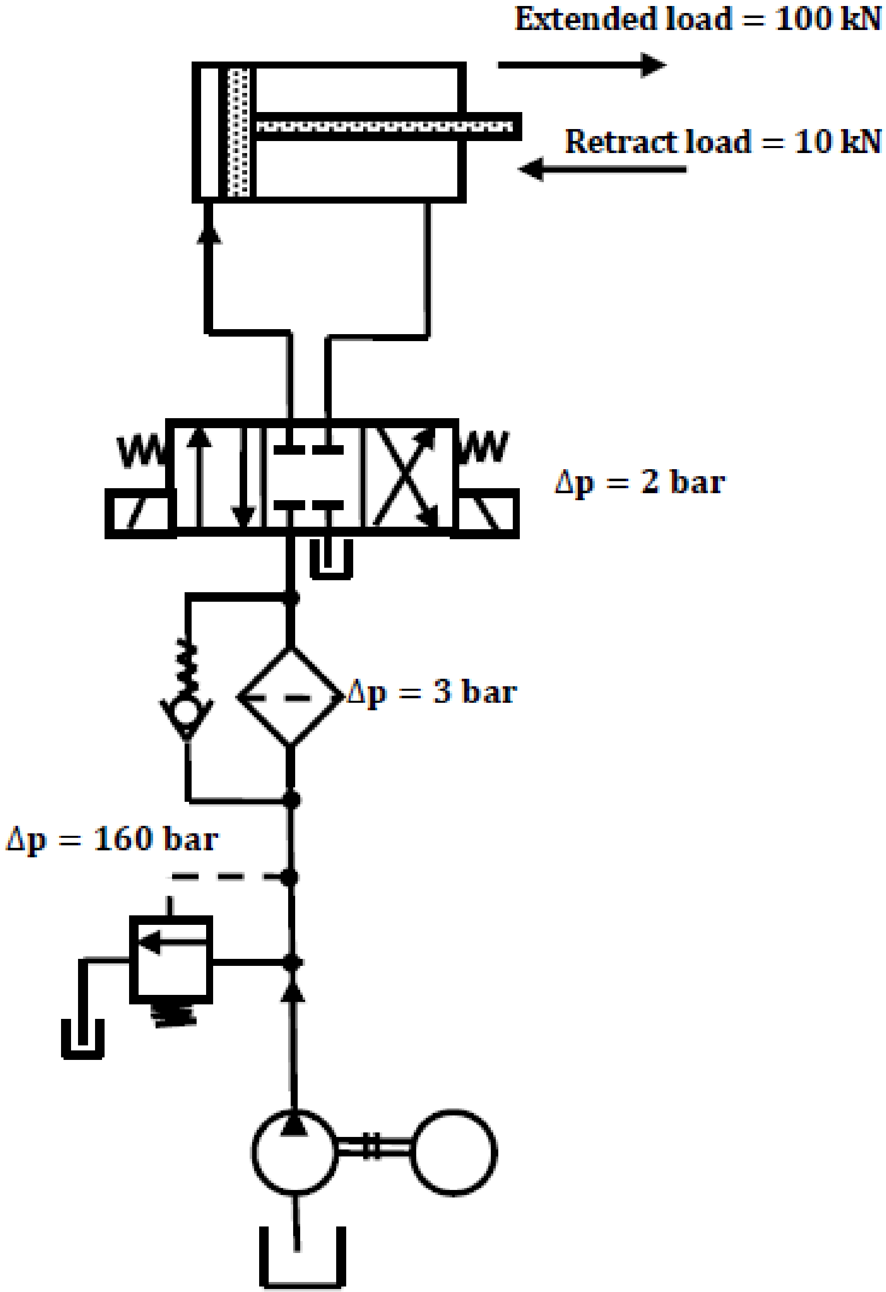


Figure 1.13 Hydraulic cylinder with no control.

Case 1: No flow controls (Fig. 1.13)

Part (a) No flow controls

Maximum available pressure at the full bore end of cylinder = 160 − 3 − 2 = 155 bar

Back pressure at the annulus side of cylinder = 2 bar.

This is equivalent to 1 bar at the full bore end because of the 2:1 area ratio. Therefore, the maximum pressure available to overcome load at the full bore end is 155 − 1 = 154 bar

Full bore area = Load/Pressure = $\dfrac{100 \times 103}{154 \times 105}$ =0.00649 m^2

Piston diameter = $\left(\dfrac{4 \times 0.00649}{\pi}\right)^{1/2}$

Select a standard cylinder, say with 100-mm bore and 70-mm rod diameter. Then

Full bore area = 7.85×10^{-3} m^2

Annulus area = 4.00×10^{-3} m^2

This is approximately a 2:1 ratio.

Part (b)No flow controls

Flow rate for a return speed of 5 m/min is given by

$$\text{Area} \times \text{Velocity} = 4.00 \times 10^{-3} \times 5 \text{ m}^3/\text{min} = 20 \text{ LPM}$$

Extend speed = $\dfrac{20 \times 10^{-3}}{7.85 \times 10^{-3}}$ = 2.55 m/min

Pressure to overcome load on extend = $\dfrac{100 \times 10^{3}}{7.85 \times 10^{-3}}$ = 12.7 MPa = 127 bar

Pressure to overcome load on retract = $\dfrac{10 \times 10^{3}}{4.00 \times 10^{-3}}$ = 2.5 MPa = 25 bar

(i) Pressure at pump on extend (working back from the DCV tank port)

Pressure drop over DCV B to T	$2 \times (1/2)$	1
Load-induced pressure		127
Pressure drop over DCV P to A		2
Pressure drop over filter		3

Therefore, pressure drop required at the pump during extend stroke = 133 bar

Relief-valve setting = 133 + 10% = 146 bar

(ii) Pressure required at the pump on retract (working from the DCV port as before) is

$$(2 \times 2) + 25 + 2 + 3 = 34 \text{ bar}$$

Note: The relief valve will not be working other than at the extremities of the cylinder stroke. Also, when movement is not required, pump flow can be discharged to the tank at low pressure through the center condition of the DCV.

Part (c) No flow controls

System efficiency:

$$\frac{\text{Energy required to overcome load on the cylinder}}{\text{Total energy into fluid}} = \frac{\text{Flow to the cylinder} \times \text{Pressure owing to load}}{\text{Flow from the pump} \times \text{Pressure at the pump}}$$

$$\text{Efficiency on extend stroke} = \frac{20 \times 127}{20 \times 133} \times 100 = 95.5\ \%$$

$$\text{Efficiency on retract stroke} = \frac{20 \times 25}{20 \times 34} \times 100 = 73.5\ \%$$

Case 2: Meter-in flow control for the extend speed of 0.5 m/min(Fig. 1.14)

Part (a) meter in controls

From case 1,

Select a standard cylinder, say with 100-mm bore and 70-mm rod diameter.

Cylinder 100-mm bore diameter × 70-mm rod diameter

Full bore area 7.85×10^{-3} m^2

Annulus area = 4.00×10^{-3} m^2

Load-induced pressure on extend = 127 bar

Load-induced pressure on retract = 25 bar

Pump flow rate = 20 L/min

Part (b) meter in controls

Flow rate required for the extend speed of 0.5 m/min is

$$7.85 \times 10^{-3} \times 0.5 = 3.93 \times 10^{-3} \text{ m}^3/\text{min} = 3.93 \text{ L/min}$$

Working back from the DCV tank port:

Pressure required at the pump on retract is

$$(2 \times 2) + (2 \times 3) + 25 + 2 + 3 = 40 \text{ bar}$$

Pressure required on the pump at extend is

$$2 \times (1/2) + 127 + 10 + 2 + 3 = 143 \text{ bar}$$

Relief-valve setting = 143 + 10% = 157 bar

This is close to the maximum working pressure of the pump (160 bar). In practice, it would be advisable to select either a pump with a higher working pressure (210 bar) or use the next standard

Part (c) meter in controls

Now that a flow-control valve has been introduced when the cylinder is on the extend stroke, the excess fluid will be discharged over the relief valve.

$$\text{System efficiency on extend} = \frac{3.93 \times 127}{20 \times 157} \times 100 = 15.9\%$$

$$\text{System efficiency on retract} = \frac{20 \times 25}{20 \times 40} \times 100 = 62.5\%$$

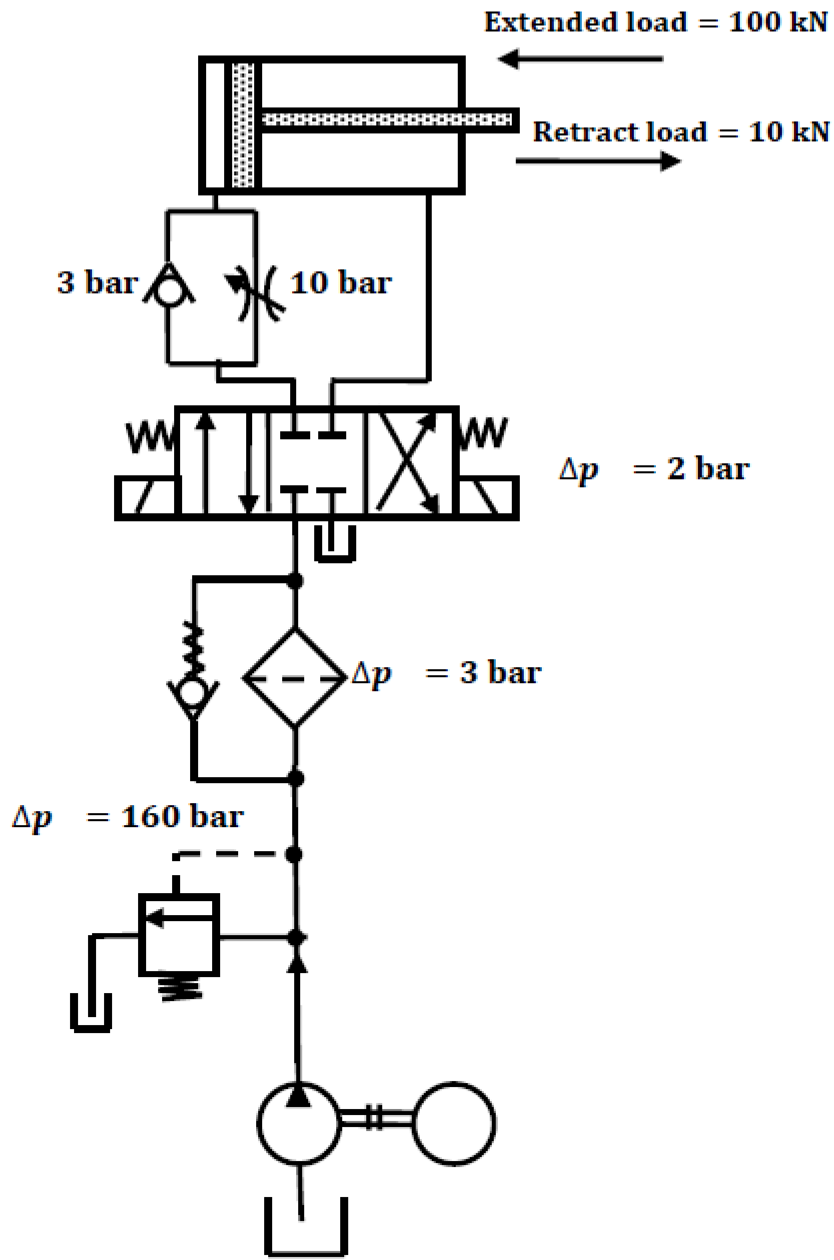


Figure 1.14 Hydraulic cylinder with meter-in control

Case 3: Meter-out flow control for the extend speed of 0.5 m/min(Fig. 1.15)
Cylinder, load, flow rate and pump details are as before (partsa and b of meter in control).

Part (c) meter out controls
Working back from the DCV tank port:

Pressure required at the pump on retract is

$$(2 \times 2) + 25 + 3 + 2 + 3 = 37 \text{ bar}$$

Pressure required at the pump on extend is

$$[2 \times (1/2)] + [10 \times (1/2)] + 127 + 2 + 3 = 138 \text{ bar}$$

Relief-valve setting = 38 + 10 % = 152 bar

$$\text{System efficiency on extend} = \frac{3.93 \times 127}{20 \times 152} \times 100 = 16.4\%$$

$$\text{System efficiency on retract} = \frac{20 \times 25}{20 \times 37} \times 100 = 67.6\%$$

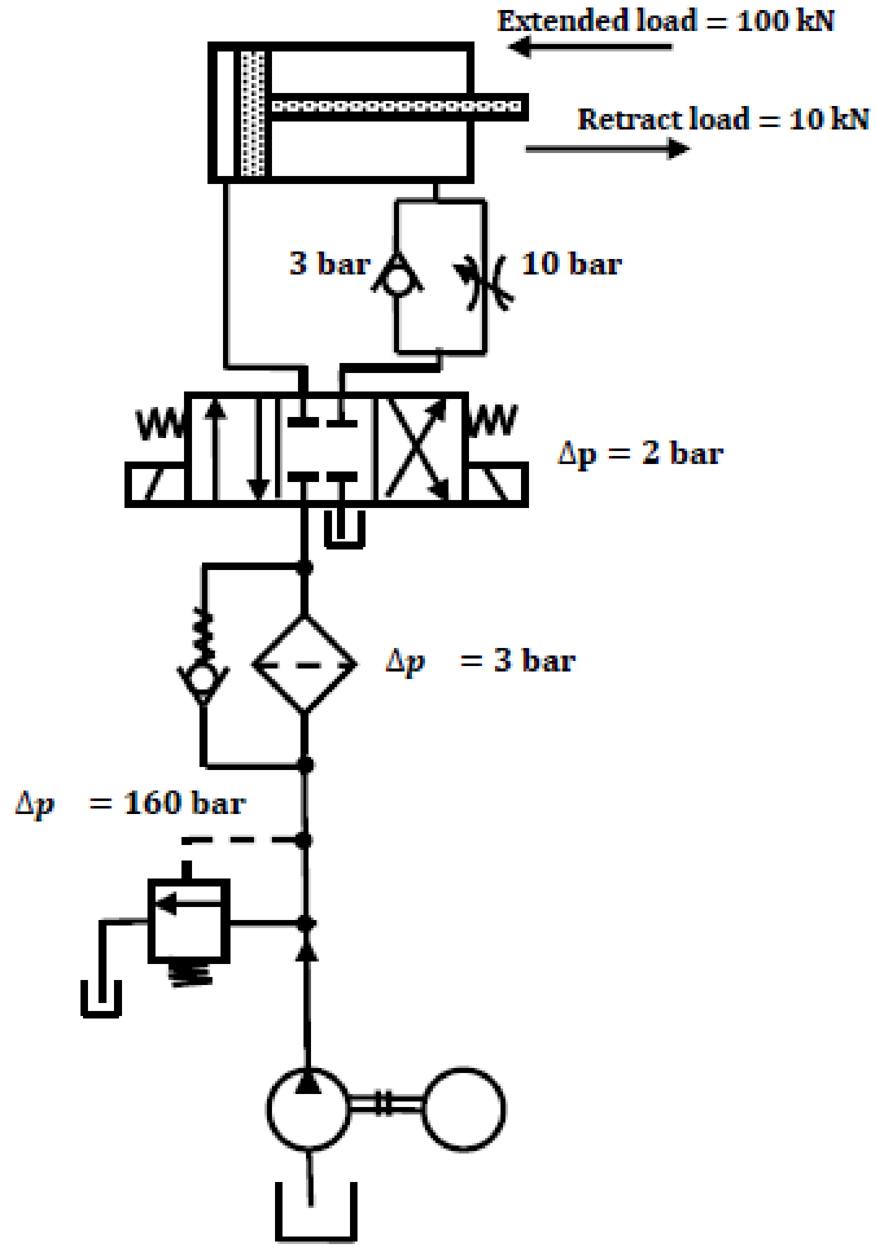


Figure 1.15 Hydrauliccylinder with meter-out control.

Discussion of results of all three cases: No control, meter-in and meter-out.
As can be seen, meter-out is marginally more efficient than meter-in owing to the ratio of piston to piston rod area. Both systems are equally efficient when used with through-rod cylinders or hydraulic motors. It must be remembered that meter-out should prevent any tendency of the load to run away.

In both cases, if the system runs light, that is, extends against a low load, excessive heat is generated over the flow controls in addition to the heat generated over the relief valve. Consequently, there is further reduction in the efficiency. Also, in these circumstances, with meter-out flow control, very high pressure intensification can occur on the annulus side of the cylinder and within the pipe work between the cylinder and the flow-control valve. Take a situation where meter-out circuit is just considered. The load on extend is reduced to 5 kN without any corresponding reduction in the relief-valve settings.

Flow into the full bore end is 3.91 L/min.
Therefore, excess flow from the pump is

$$20 - 3.93 = 16.07 \text{ L/min}$$

that passes over the relief valve at 152 bar.
The pressure at the full bore end of the cylinder is = 152 – 3 – 2 = 147 bar
This exerts a force that is resisted by the load and the reactive back pressure on the annulus side:

$$147 - \left(\frac{5 \times 10^3}{7.85 \times 10^{-3} \times 10^5} \right) = (2 + 10 + p) \times \frac{4.00}{7.85}$$

where$p$ is the pressure within the annulus side of the cylinder and between the cylinder ant the flow-control valve. So

$$p = [(147 - 6.4) \times 7.85/4.00] - 12 = 264 \text{ bar}$$

The system efficiency on extend is

$$\frac{3.93 \times 6.4}{20 \times 152} \times 100 = 0.83\%$$

Almost all of the input power is wasted and dissipates as heat into the fluid, mainly across the relief and flow-control valves.

Figure 1.16 shows a hydraulic circuit where the actuator speed is controlled by a meter-in system employing a series pressure-compensated valve. Determine the power input to the pump under a steady-state condition. If the series compensation is replaced by parallel compensation, and the load and speed of the actuator remain unchanged, determine the change of overall efficiency of the circuit.

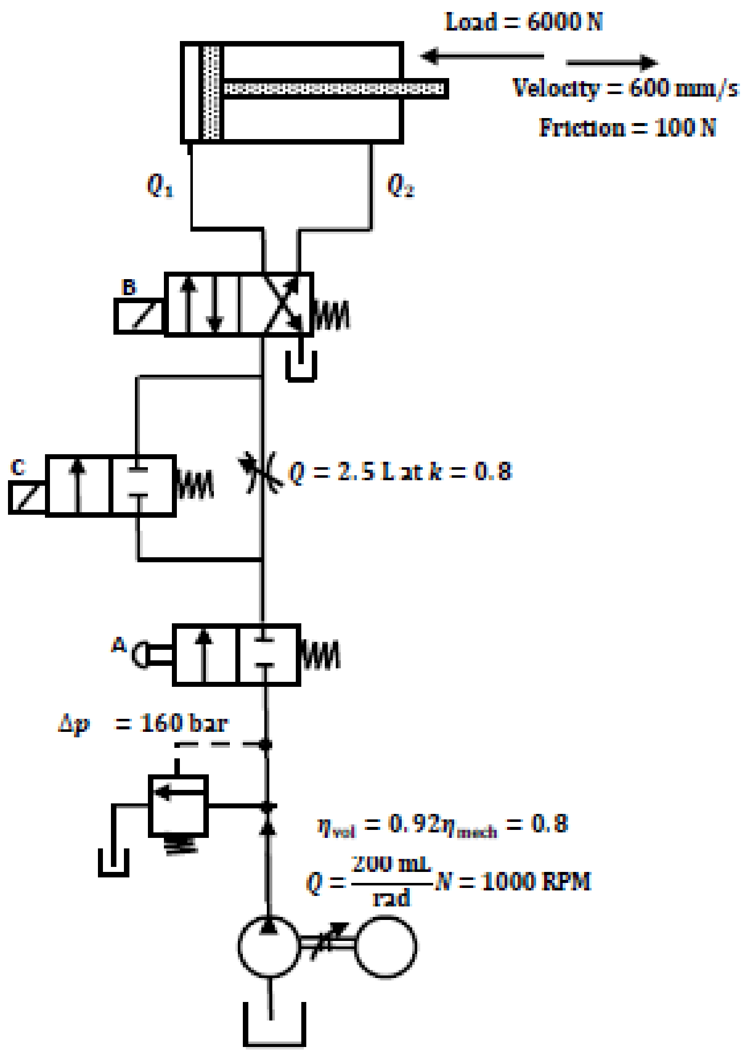


Figure 1.16 Hydraulic cylinder with a pressure-compensated valve.

Solution:

For valve A we have $Q = 0.5\ \sqrt{\Delta p}$

For valve B we have $Q = 0.4\ \sqrt{\Delta p}$

For valve C we have $Q = 0.3\ \sqrt{\Delta p}$

Now referring to Fig. 1.16, let us calculate the flow to piston side of the cylinder:

$$Q_1 = \frac{\pi}{4} \times 60^2 \times 600 = 1.7 \text{ L/s}$$

Flow from the return side of the cylinder is

$$Q_2 = 1.4 \text{ L/s}$$

Pump flow is given by

$$Q_p = 2\pi \times 1000/60 \times 20 = 2.1 \text{ L/s}$$

$$Q_3 = \eta_{vol} \times Q_p = 1.93 \text{ L/s}$$

Power input to the pump = $150 \times 10^5 \times 2.1 \times 10^{-3} \times 1/0.8$ W = 39.4 kW

Power output to the actuator = $6100/1000 \times 600/1000 = 3.6$ kW

Therefore, system efficiency = $3.66/39.4 \times 100 = 9\%$

Pressure loss at valve B due to $Q_2 = 1 \times (1.4/0.4)^2 = 12.2$ bar

Pressure at the head end of actuator

$$p \times \frac{\pi}{4} \times 60^2 \times 10^{-6} = 6100 + 12.2 \times 10^5 \times \frac{\pi}{4} \times (602 - 252) \times 10^{-6}$$

$$\Rightarrow p = 29 \text{ bar}$$

Pressure losses at B due to $Q_1 = (1.7/0.4)^2 = 18$ bar
Pressure losses at valve A $= (1.7/0.5)^2 = 11.6$ bar
Therefore, the total pressure, excluding that lost in the pressure-compensated valve if it is of series type, is

$$29 + 18 + 11.6 + 4 = 62.6 \text{ bar}$$

Hence, 150 – 62.6 = 87.4 bar is dropped in the pressure-compensated valve if it is of series type.
For a parallel pressure-compensated valve, the excess oil $Q - Q_1$ would bypass at

$$62.6 \text{ bar} - 11.6 \text{ bar} = 51 \text{ bar}$$

The pump delivery would be at 62.6 bar and hence the total power consumption is

$$62.6 \times 10^5 \times 2.1 \times 10^{-3} \times 1/0.8 \text{ W} = 16.5 \text{ kW}$$

System efficiency = 22.2%

1.3.1 Proportional Solenoids

All standard solenoids have no intermediate positions; rather they are always at one end or the other of the solenoid stroke. The magnetic flux attempts to drive the plunger to its fully closed position when the coil is energized. The force developed by the solenoid is a function of square of the solenoid current and inverse function of square of the air gap. The result is that the force increases as the air gap closes as well as when the current increases. A typical force–displacement

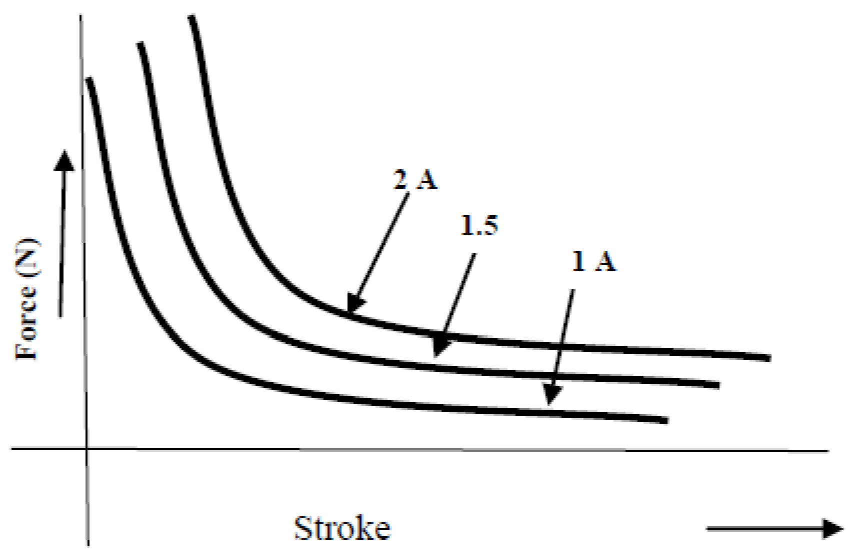


Figure 1.2 Solenoid force versus stoke curves with increasing current.

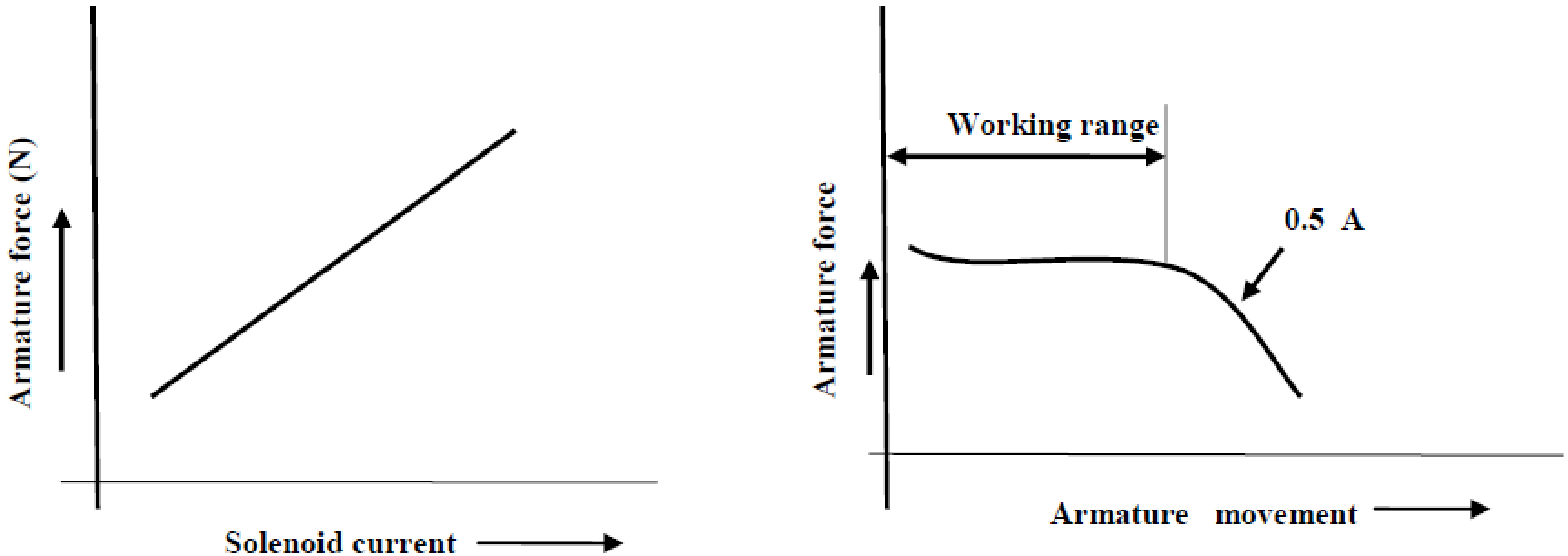


Figure 1.7 Proportional solenoid characteristics.

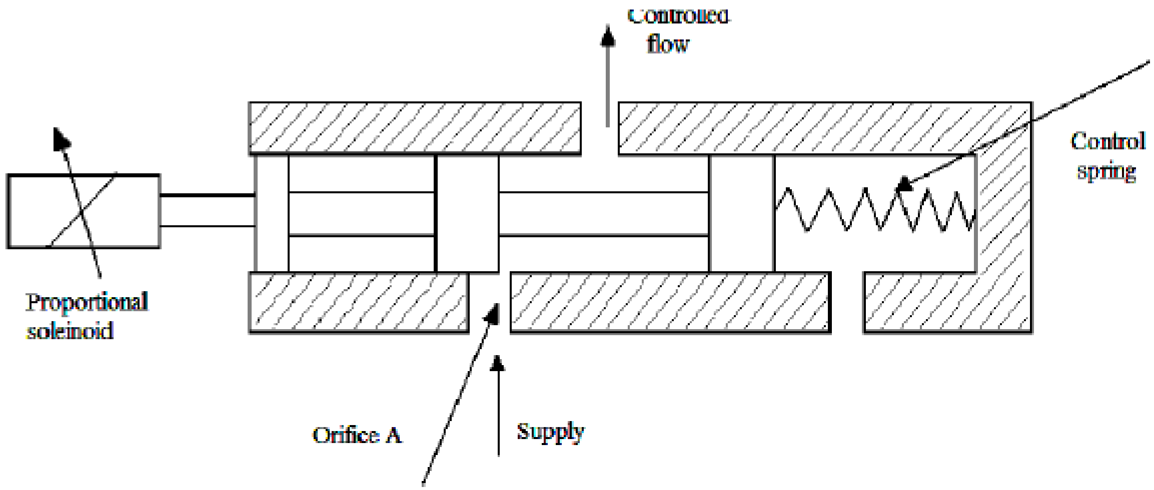


Figure 1.8 Diagrammatic section of a proportional control valve.

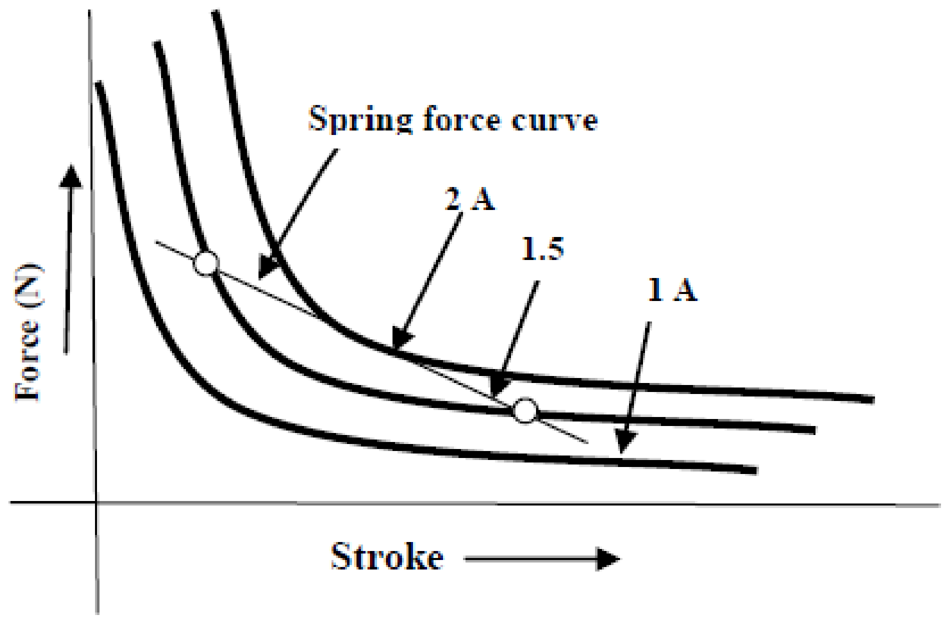


Figure 1.3 Solenoid force versus stroke with spring force.

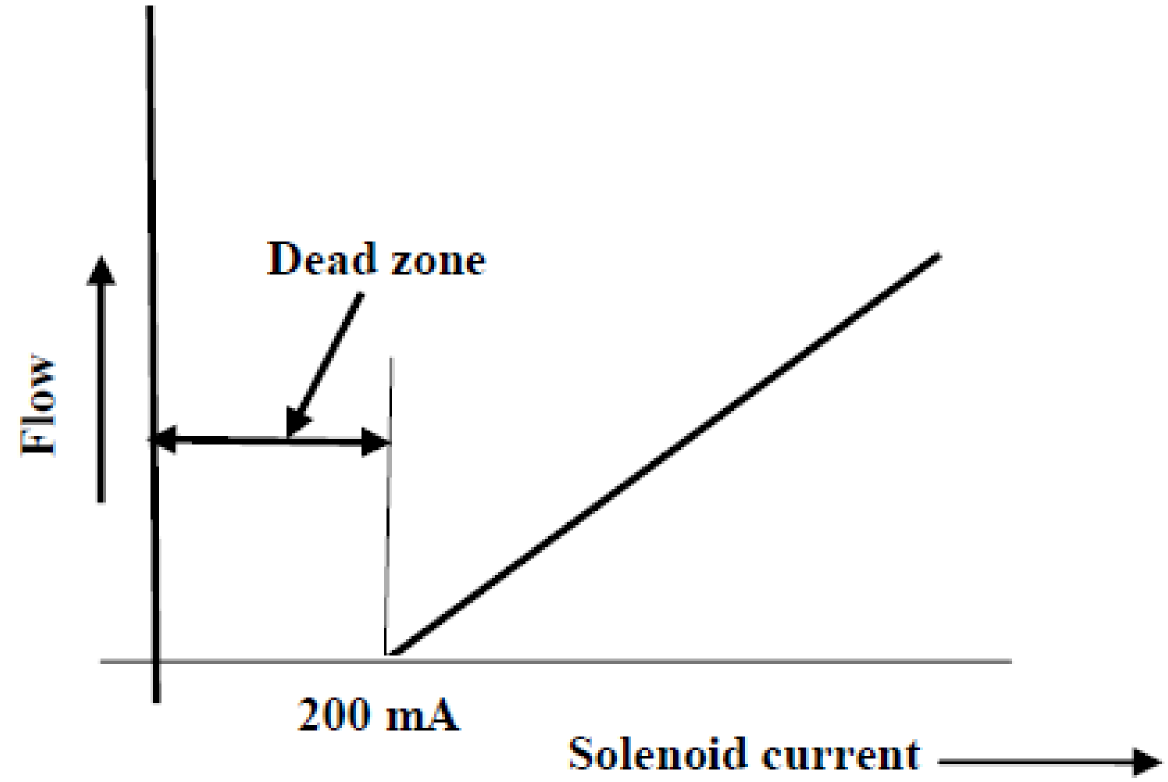
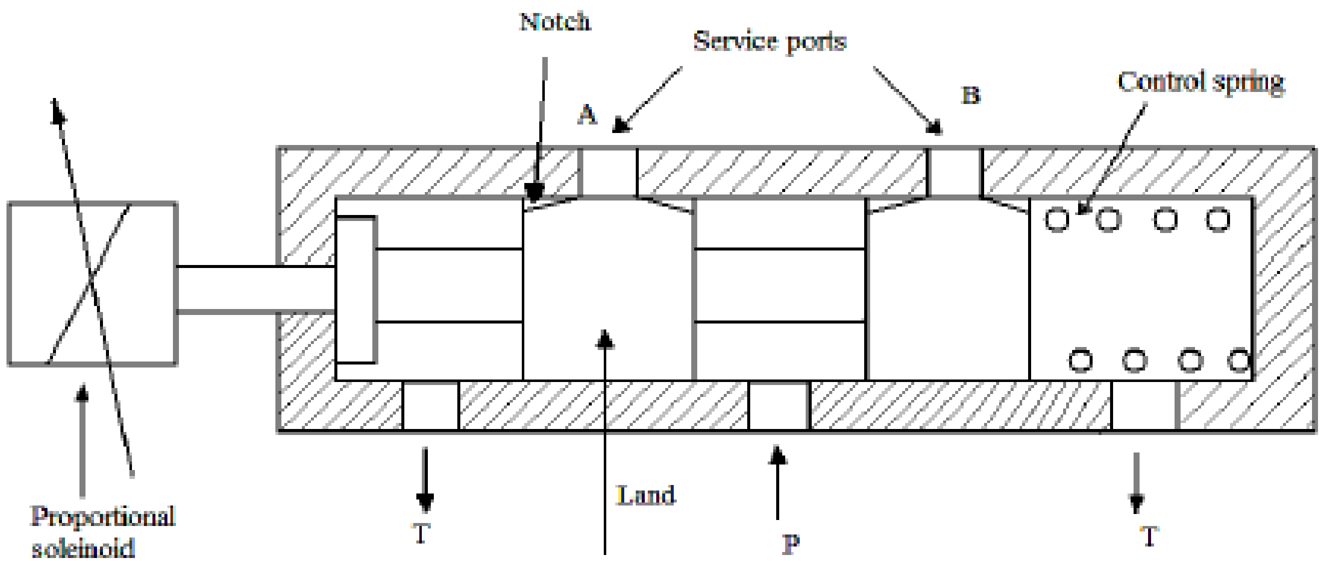
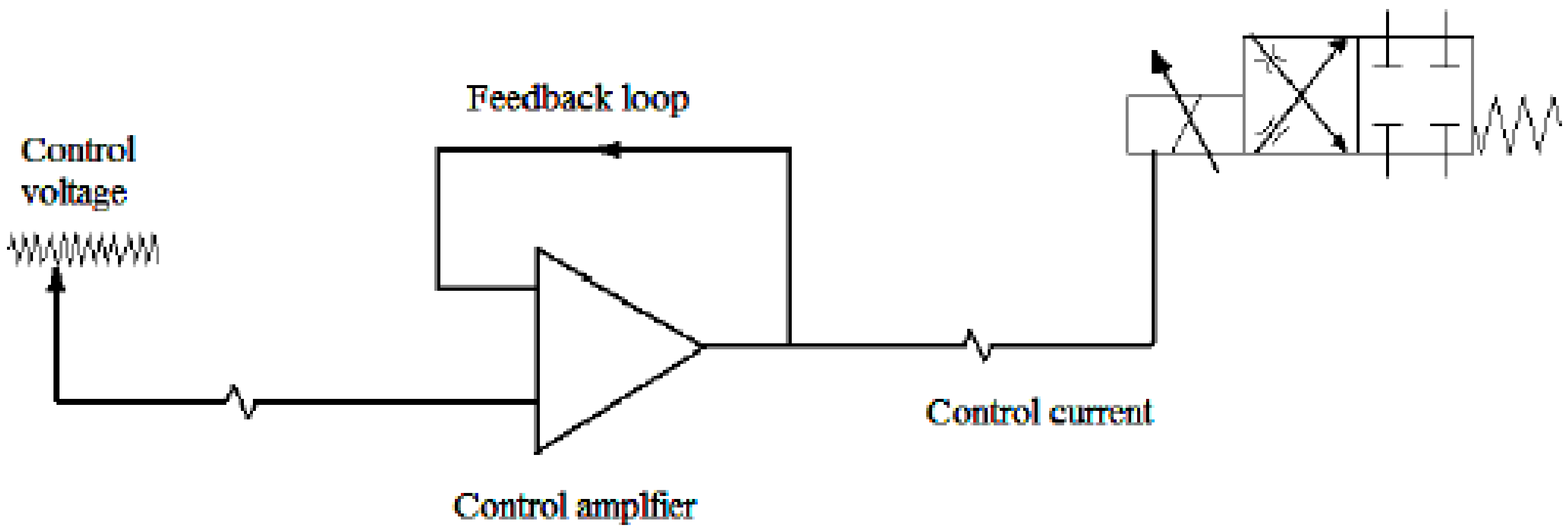


Figure 1.9 Flow current characteristics of a proportional control valve.


Notch
Service ports
A
B
Control spring
Proportional soleinoid
T
Land
P
T

(a)


Feedback loop
Control voltage
Control current
Control amplfier

(b)

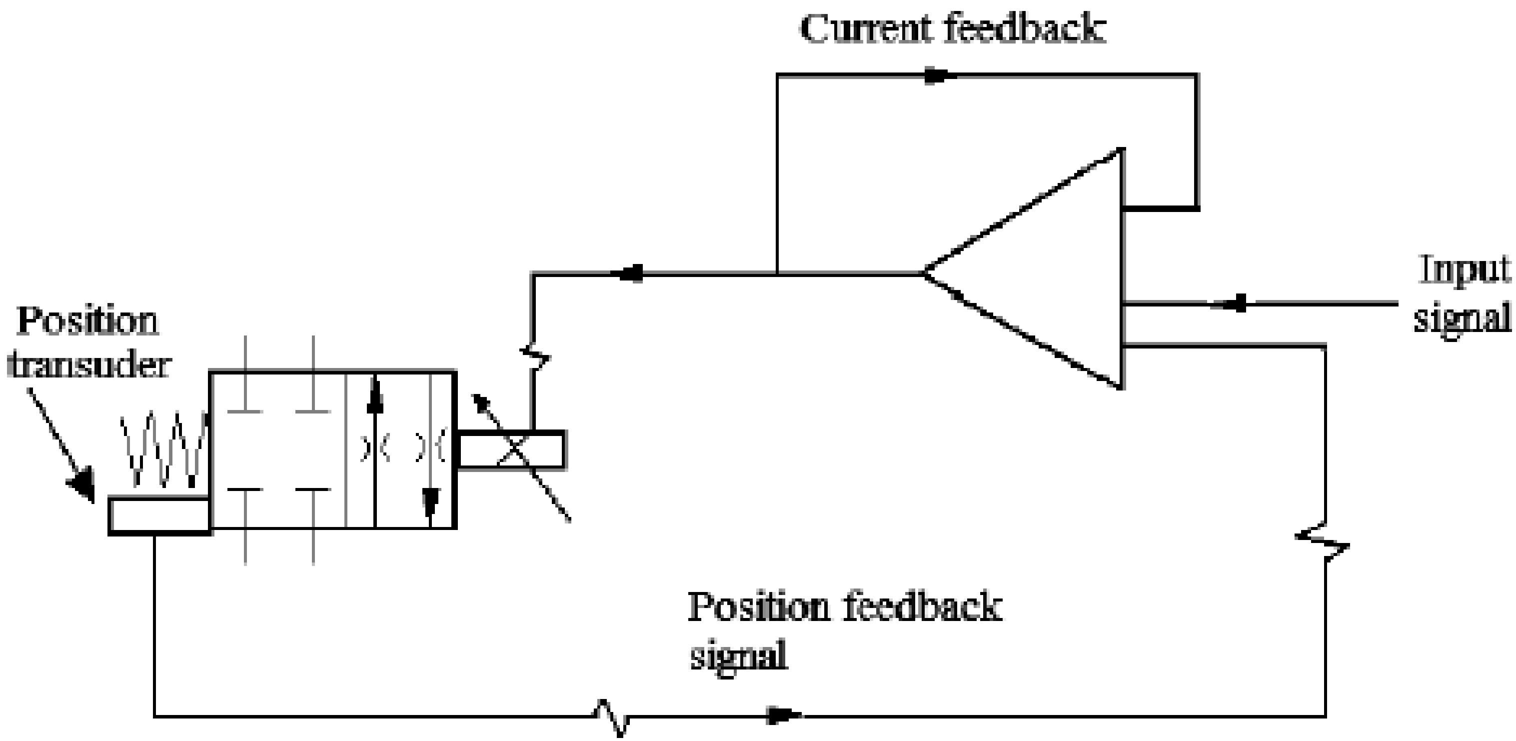


Figure 1.12 The symbols of a proportional directional control valve.

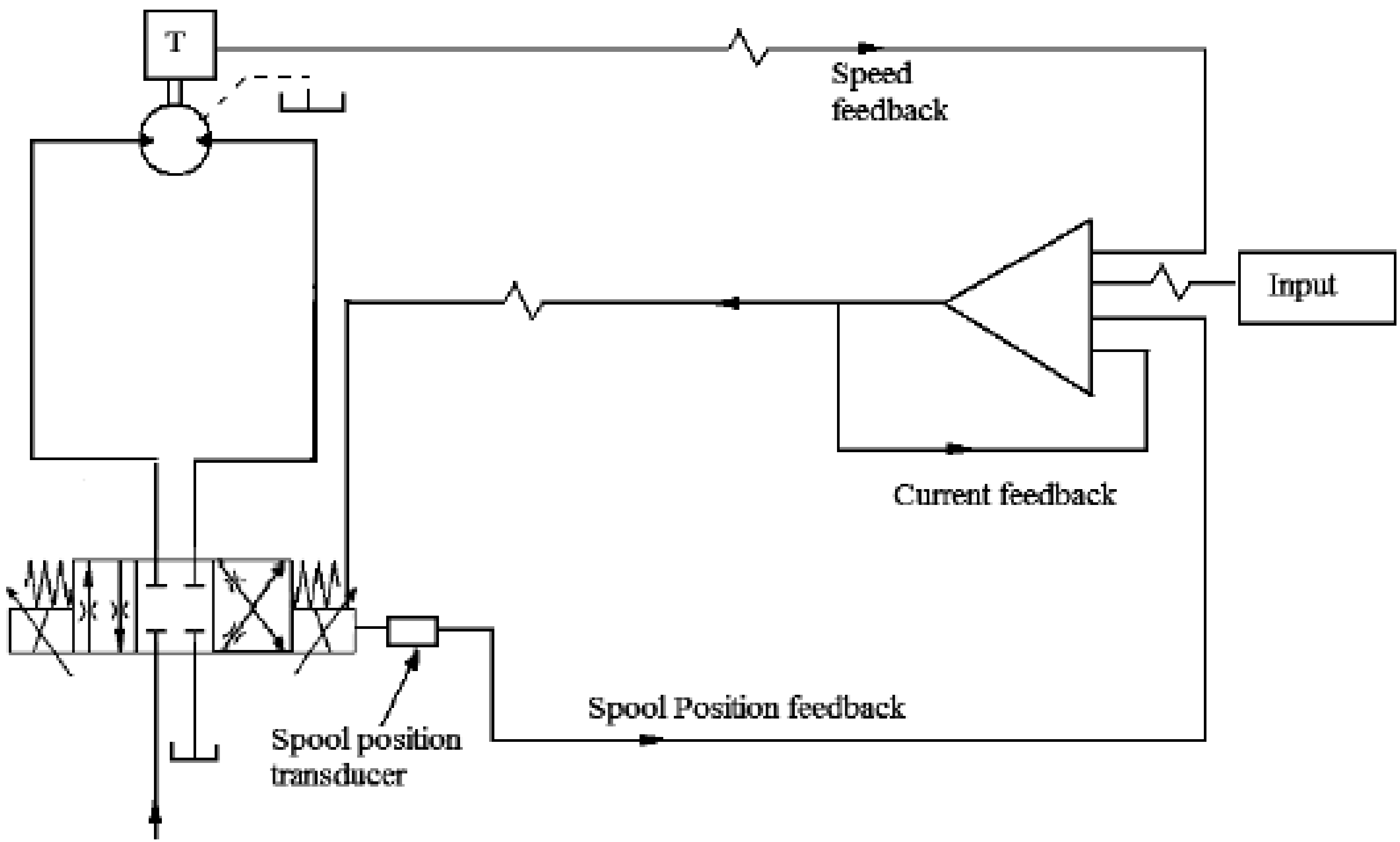


Figure 1.13 Closed-loop speed control with a proportional valve.

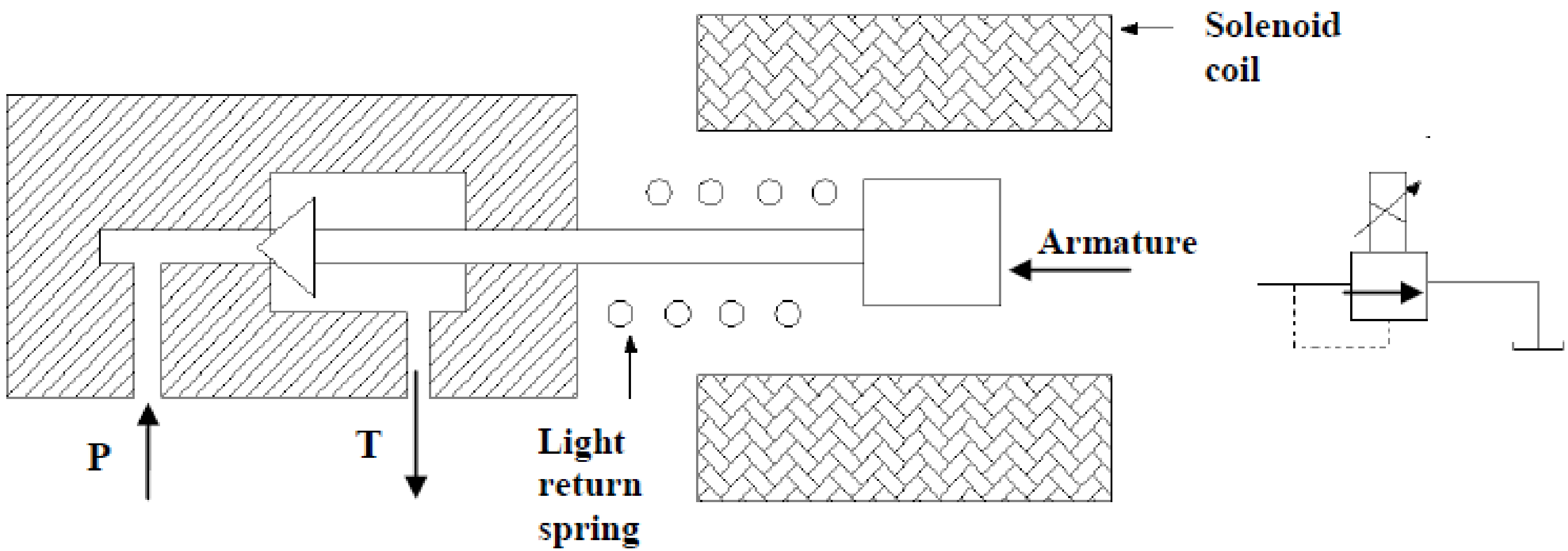


Figure 1.14 Direct-acting proportional relief valve.

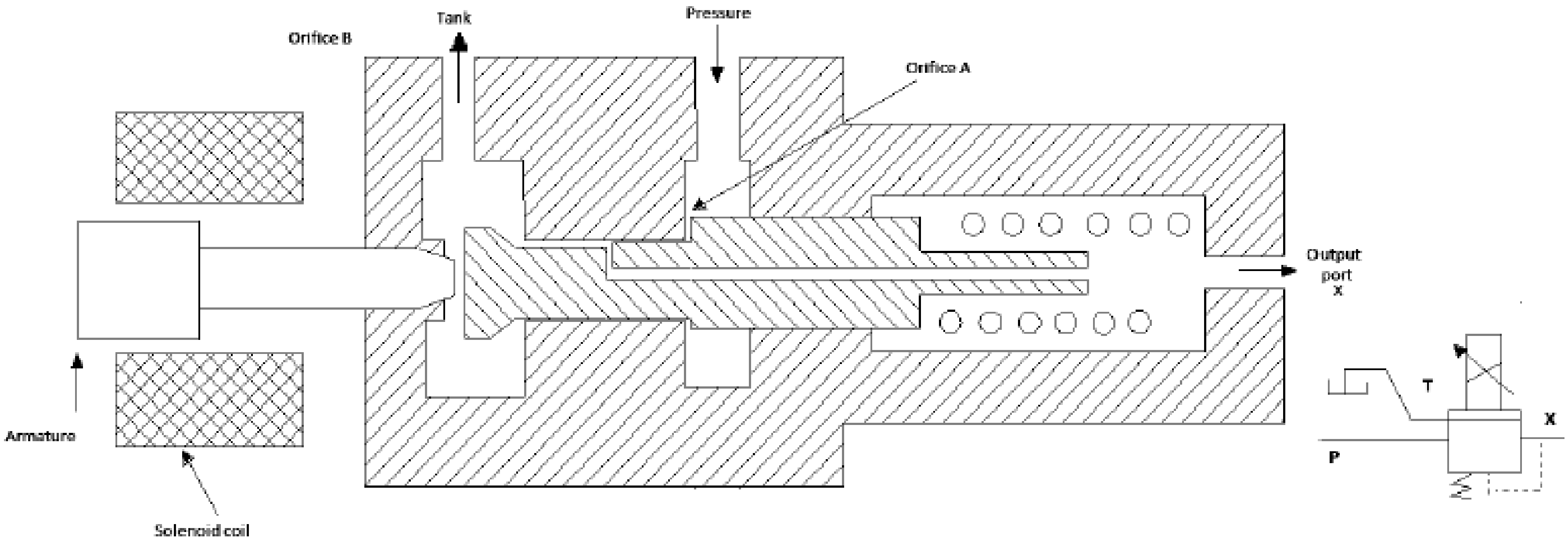


Figure 1.15 Proportional pressure-reducing valve.

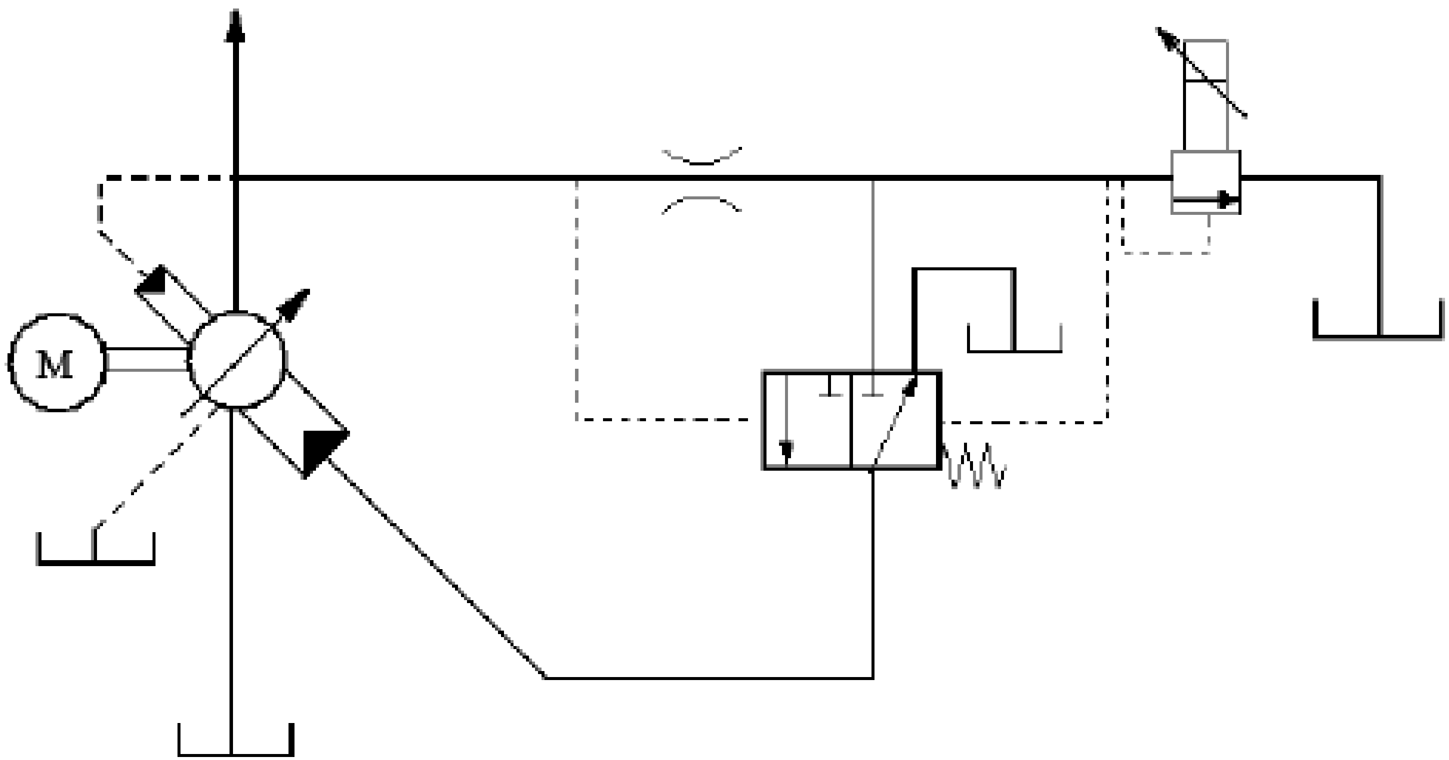


Figure 1.22 Pressure-compensated pump with proportional pressure control.

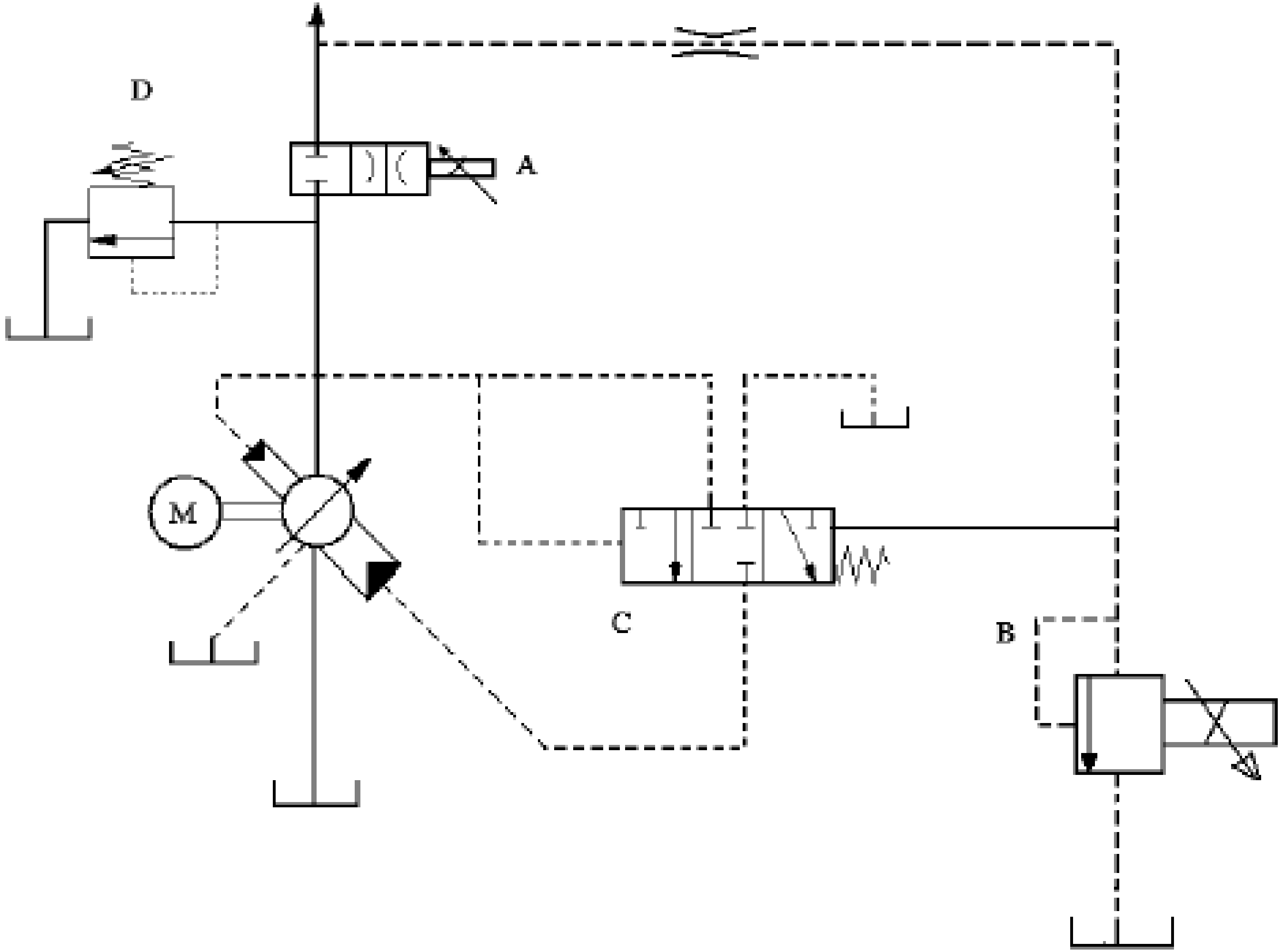


Figure 1.23 Pump with proportional pressure and flow control.

Parameter	**Proportional Hydraulic Valve**	**Electrohydraulic Servo Valve**
Valve lap	Overlap spool, causing a "dead zone" on either side of the null position.	Zero or underlap valve spool. No dead zone.
Response time for the valve spool to move fully over	40–60 ms	5–10 ms
Maximum operating frequency	Approx. 10 Hz	Approx. 100 Hz
Hysteresis	Without armature feedback approx. 5% With armature position feedback approx. 1%	Approx. 0.1%

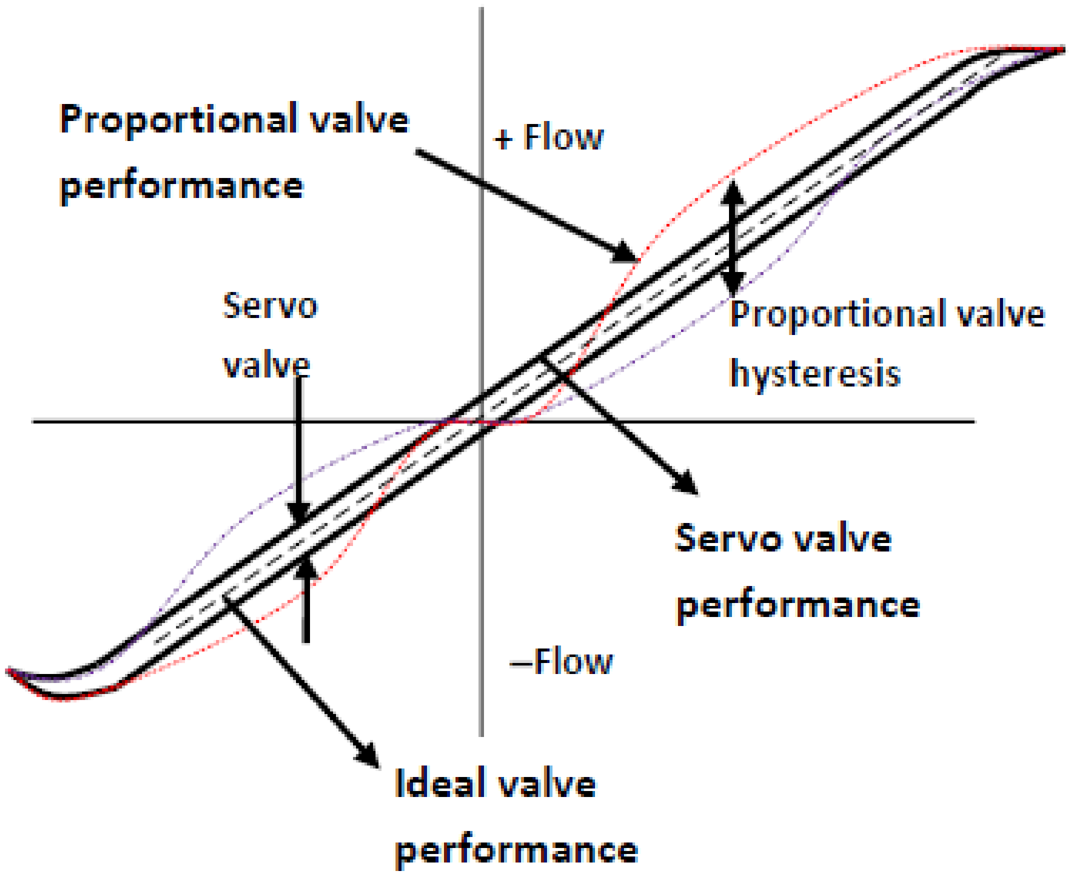


Figure 1.15 Hysteresis for a servo valve and a typical proportional valve.

Table 1.1 Comparison of servo valves and electrohydraulic proportional valves

Feature	**Servo Valve**	**EHPV**
Electrical operator	Torque motor	Proportional solenoid
Manufacturing precision	Extremely high	Moderately high
Feedback circuitry	Main system as well as valve	Valve (depending on type), main system (seldom)
Cost (compared with a solenoid valve)	Very expensive	Moderately expensive

1.4.1 Torque Motor

A torque motor is illustrated in Fig. 1.4. It is a simple electromagnetic device consisting of one or two permanent magnets, two pole pieces, a ferromagnetic armature and two coils. The permanent magnet polarizes the upper and lower pole pieces, so that they present equal and opposite magnetic fields. Torque motors are very low-power devices operated on low-voltage DC power.

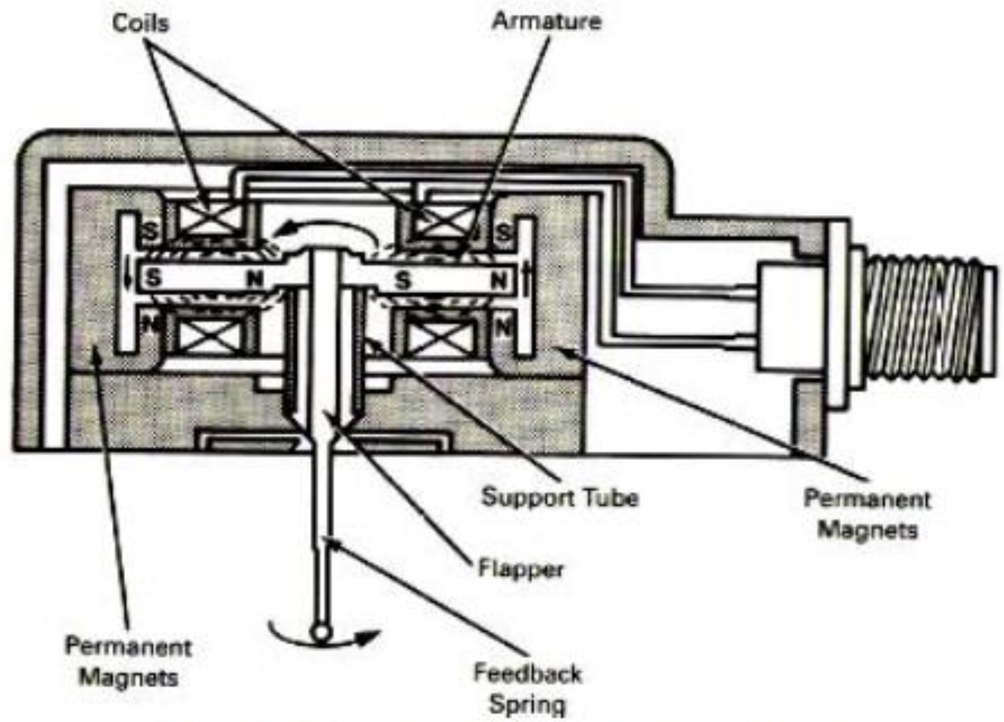


Figure 1.4 Servo valve torque motor.

magnetic field is generated. The polarity of the field depends on the direction of the current flow. In Fig. 1.5, the current flow causes the left end to become the South Pole and right end to become the North Pole, resulting in counter-clockwise rotation of the armature.

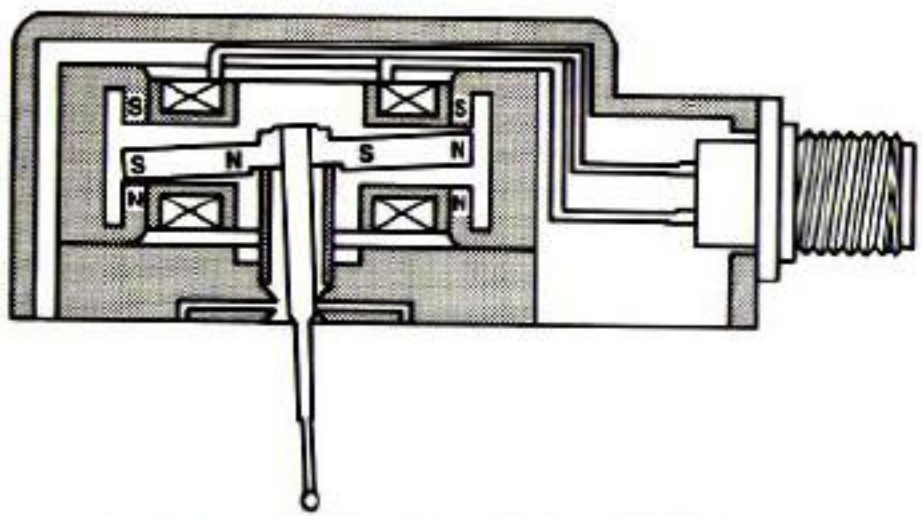


Figure 1.5 Servo torque motor operation.

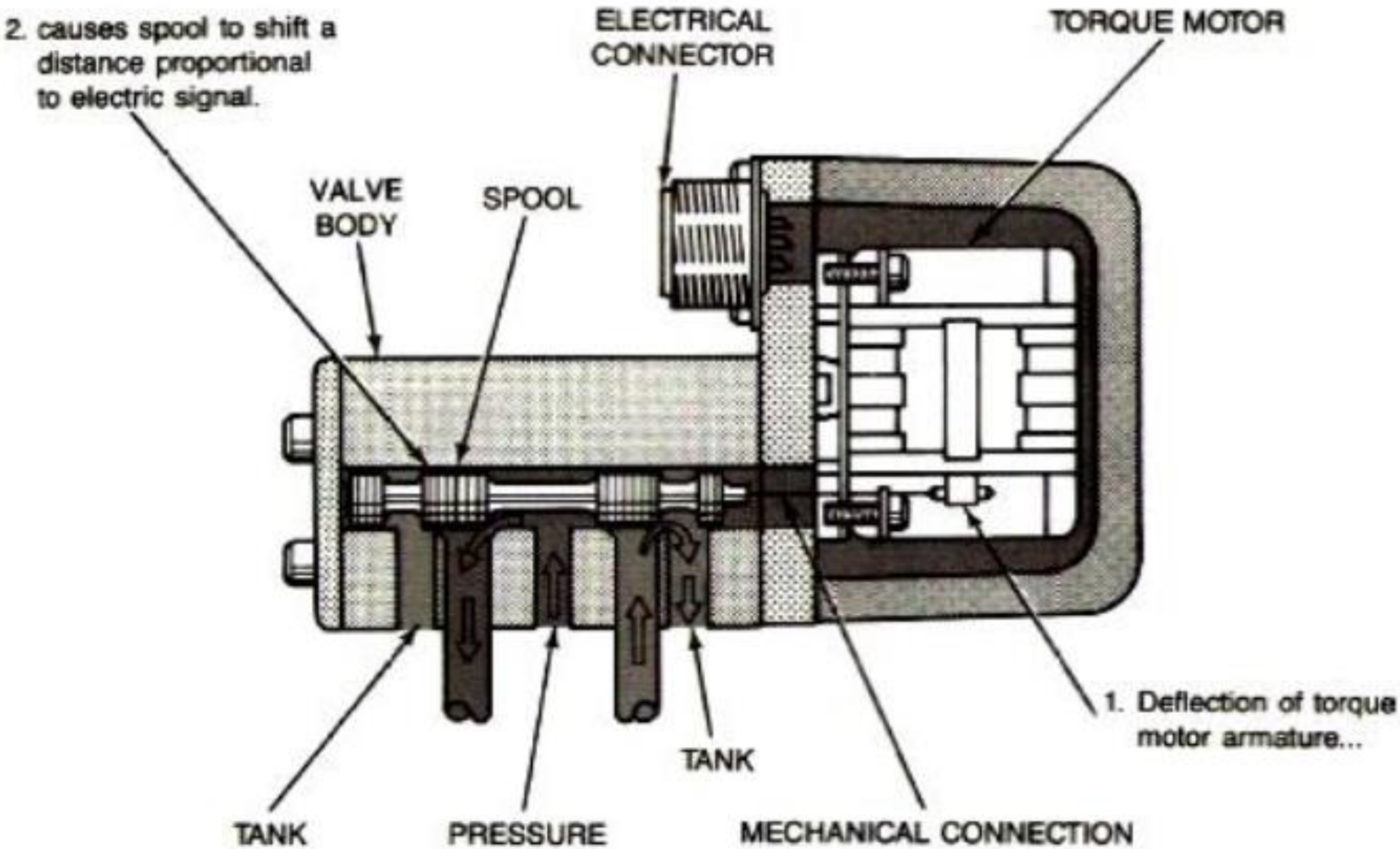


Figure 1.8 Single-stage spool-type servo valve.

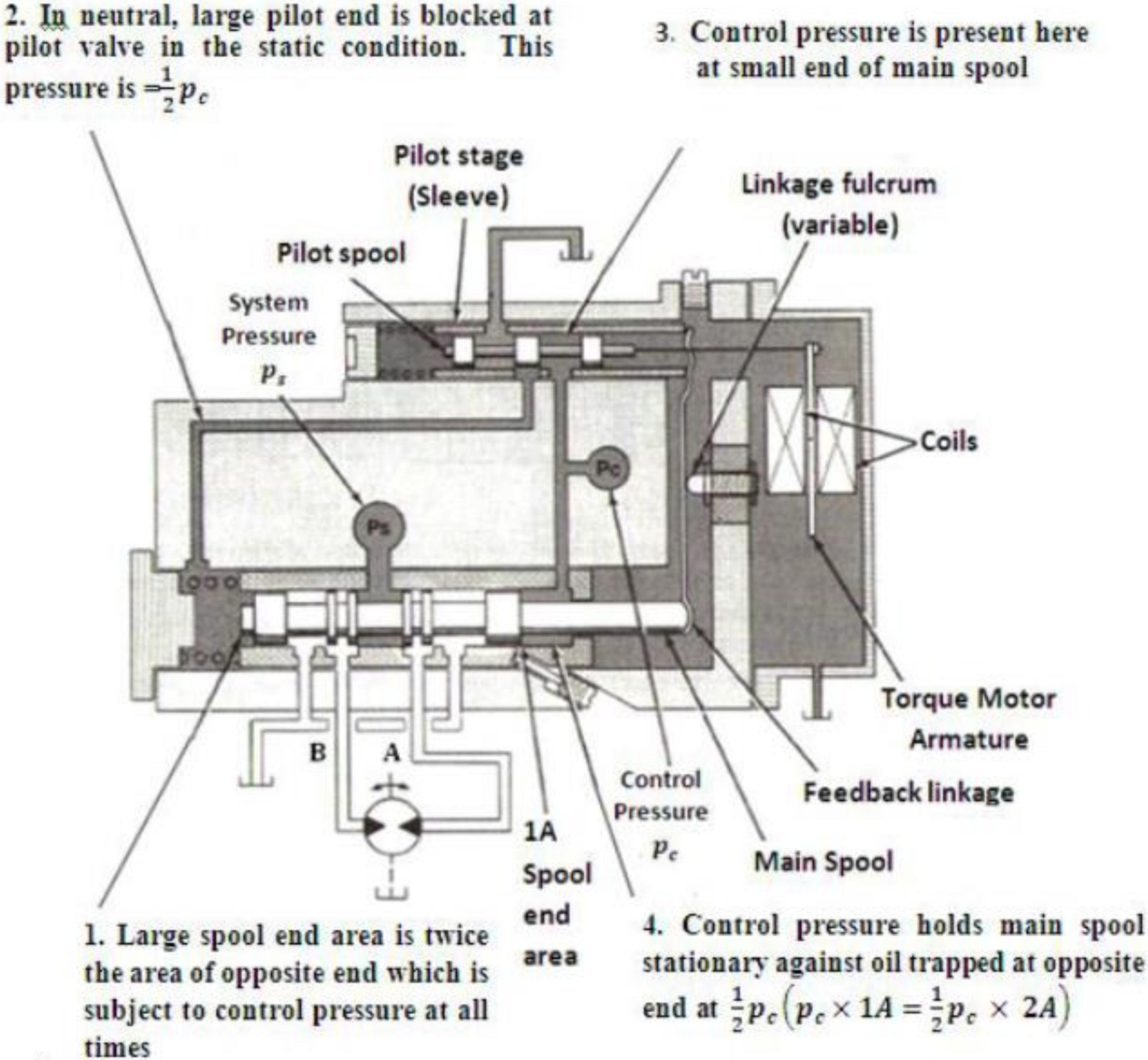


Figure 1.9Two-stage spool-type servo valve.

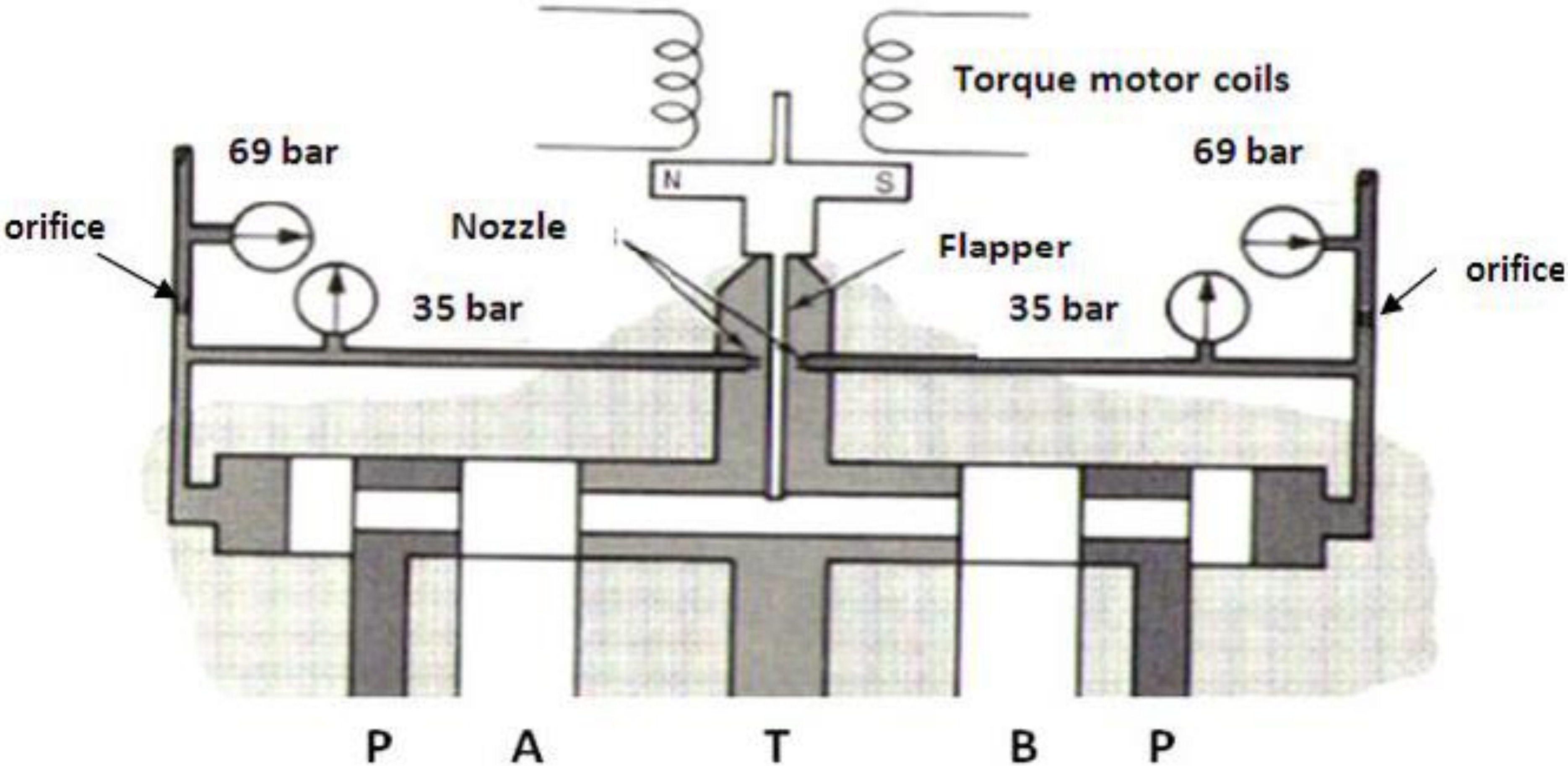


Figure 1.10 Two-stage spool-type servo valve.

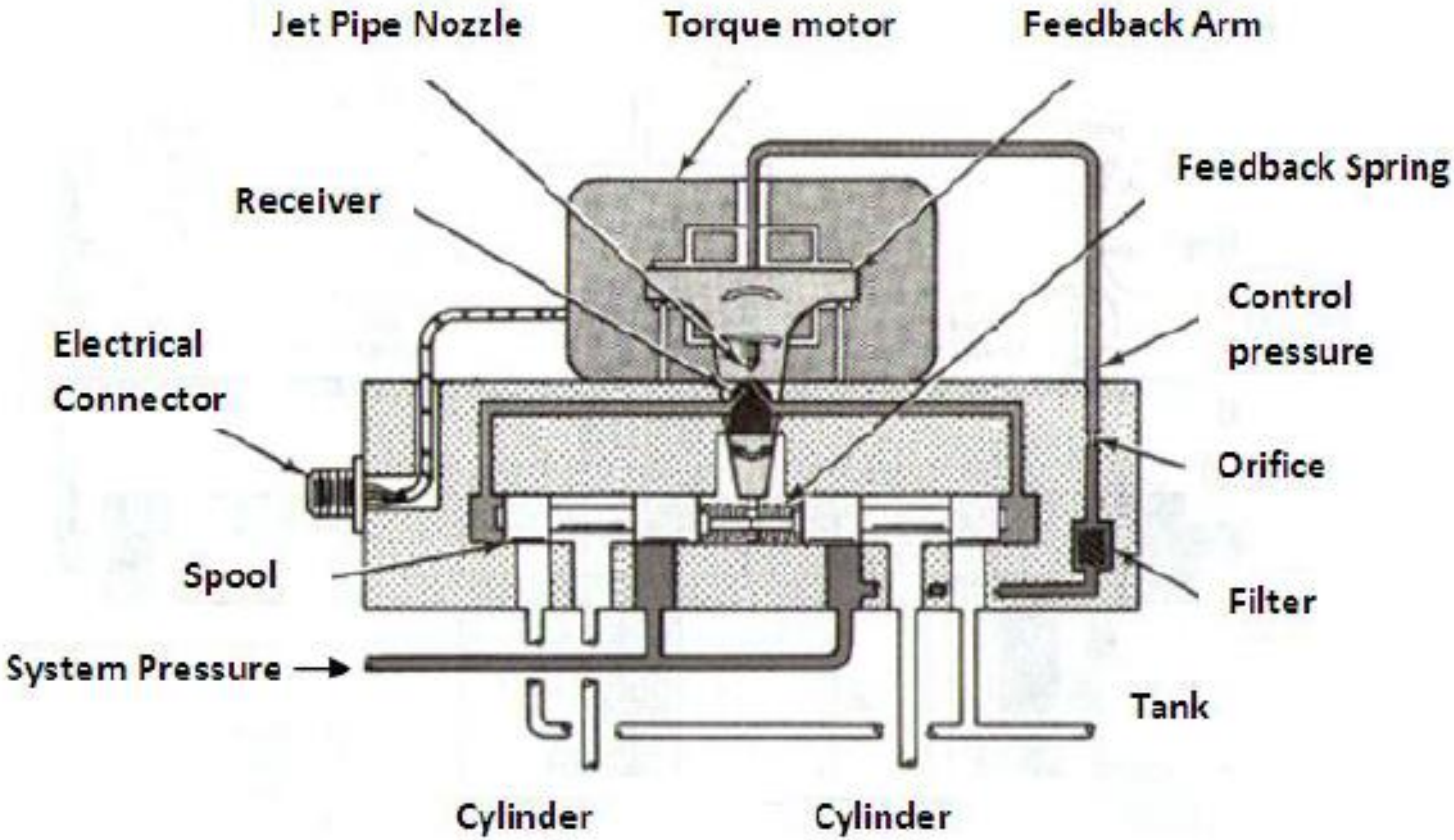


Figure 1.11 Two-stage spool-type servo valve.

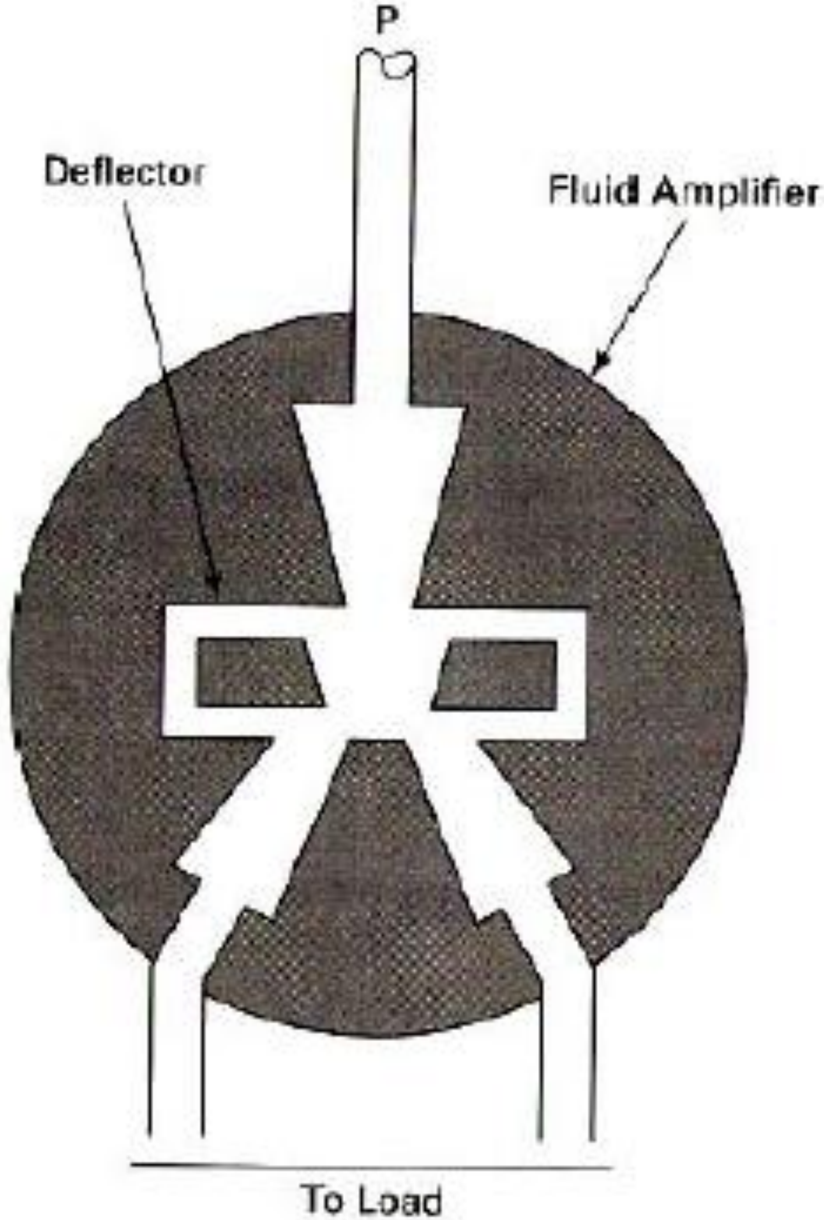


Figure 1.12 Deflector jet valve.

ACCESSORIES USED IN FLUID POWER SYSTEMS

1. Internal leakage: This occurs in hydraulic components built with operating clearances. Moving parts need to be lubricated and leakage path may be designed solely for this purpose. Internal leakage does not cause loss of fluid because the fluid returns to the reservoir. This leakage increases the clearances between mating parts due to wear. If the entire system leakage becomes large enough, the actuators do not operate properly.

2. External leakage: External leakage represents loss of fluid from the system. It also represents a safety hazard. Improper assembly of pipe fittings is the most common cause of external leakage. Over-tightened fittings may become damages or vibrations can cause properly tightened fittings to become loose. Failure to connect drain lines, excessive operating pressure and contamination might cause the fluid to externally leak.

1.2 Functions of Seals

Seals are used in hydraulic systems to prevent excessive internal and external leakage and to keep out contamination.Various functions of seals include the following:

1. They prevent leakage – both internal and external.
2. They prevent dust and other particles from entering into the system.
3. They maintain pressure.
4. They enhance the service life and reliability of the hydraulic system.

1.2.1 Classification of Hydraulic Seals

Hydraulic seals can be classified as follows:

1. **According to the method of sealing:**

   - *Positive sealing:* A positive seal prevents even a minute amount of oil from getting past. A positive seal does not allow any leakage whatsoever (external or internal).
   - *Non-positive sealing:* A non-positive seal allows a small amount of internal leakage, such as the clearance of the piston to provide a lubrication film.

2. **According to the relative motion existing between the seals and other parts:**

   - **Static seals:** These are used between mating parts that do not move relative toone another. Typical examples are flange gaskets and seals, o-rings, etc. These are relatively simple. They are essentially non-wearing and usually trouble-free if assembled properly.
   - **Dynamic seals:** These are assembled between mating parts that move relative to each other. Hence, dynamic seals are subject to wear because one of the mating parts rubs against the seal.

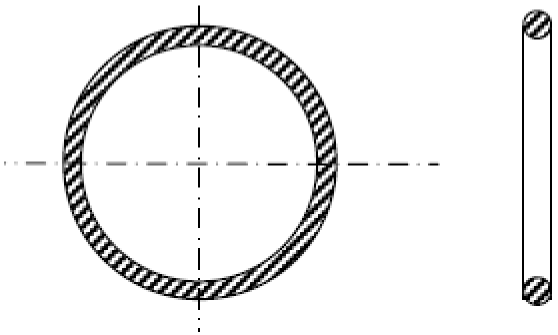
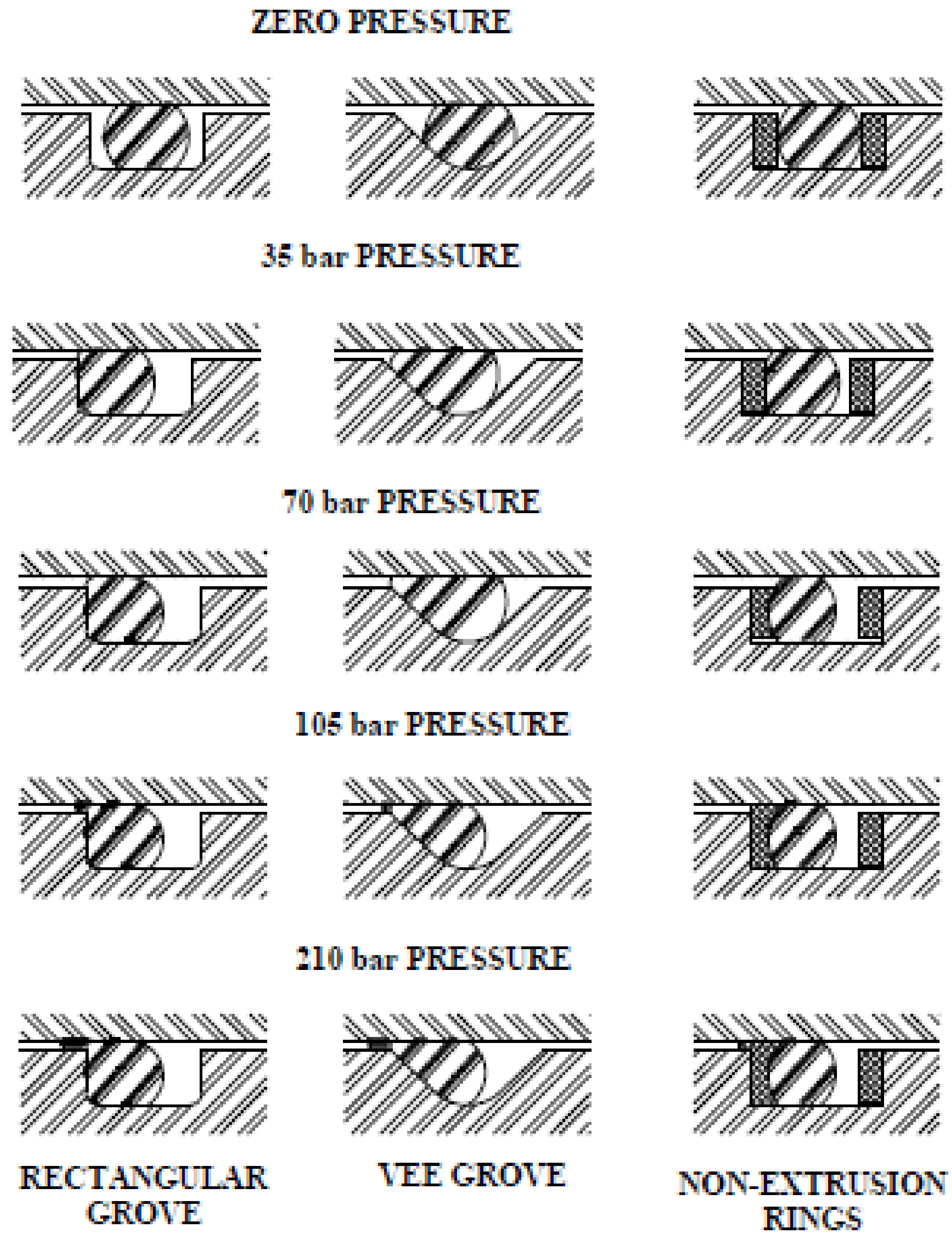
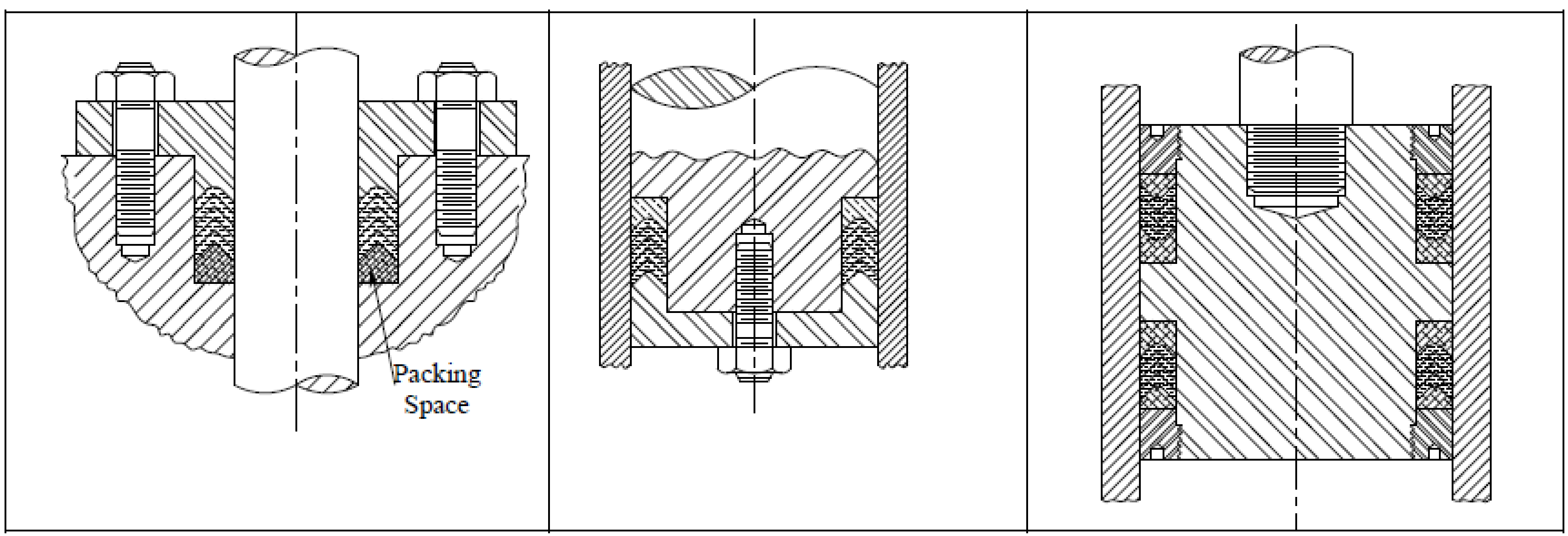


Figure 1.1 O-ring.



(a) (b)

Figure 1.3(a)V-ring seal and (b) U-ring seal.



(a) (b) (c)

Figure 1.4 V packings:(a) An outside packed V-ring installation;(b) installation of V packing on the end of the ram;(c) V packing for double-acting cylinder.

- **Leather:** This material is rugged and inexpensive. However, it tends to squeezewhen dry and cannot operate above 90°C which is inadequate for many hydraulic systems. Leather does not operate well at cold temperatures to about −50°C .
- **Buna-N:** This material is rugged and inexpensive and wears well. It has a rather wide operating temperature range (−45–110°C) during which it maintains its good sealing characteristics.
- **Silicone:** This elastomer has an extremely wide temperature range (−65–232°C). Hence, it is widely used for rotating shaft seals and static seals where a wide operating temperature is expected. Silicone is not used for reciprocating seal applications because it has a low tear resistance.

- **Neoprene:** This material has a temperature range of 50–120°C. It is unsuitable above 120°C because it has a tendency to vulcanize.
- **Tetrafluoroethylene:** This material is the most widely used plastic for seals of hydraulic systems. It is a tough, chemically inert, waxy solid, which can be processed only by compacting and sintering. It has an excellent resistance to chemical breakdown up to temperatures of 370°C. It also has an extremely low coefficient of friction. One major drawback is its tendency to flow under pressure, forming thin, feathery films. This tendency to flow can be greatly reduced by the use of filler materials such as graphite, metal wires, glass fibers and asbestos.
- **Viton:** This material contains about 65% fluorine. It has become almost a standard material for elastomer-type seals for use at elevated temperatures up to 240°C. Its minimum operating temperature is 28°C.

1.3 Durometer Hardness Tester

The physical properties frequently used to describe the behavior of elastomers are as follows: hardness, coefficient of friction, volume change, compression set, tensile strength, elongation modulus, tear strength, squeeze stretch, coefficient of thermal expansion and permeability. Among these physical properties, hardness is the most important because it has a direct relationship to service performance.

A durometer is an instrument used to measure the indentation hardness of rubber and rubber-like materials. As shown, the hardness scale has a range from 0 to 100. The durometer measures 100 when pressed firmly on flat glass. High durometer readings indicate a great resistance to denting and thus a hard material. A durometer hardness of 70 is the most common value.

A hardness of 80 is usually specified for rotating motion to eliminate the tendency toward side motion and bunching in the groove. The values between 50 and 60 are used for static seals on rough surfaces. Hard seal materials (values between 80 and 90) have less breakaway friction than softer materials, which have a greater tendency to deform and flow into surface irregularities. As a result, harder materials are used for dynamic seals.

1.4 Reservoirs

The functions of a fluid reservoir in a power hydraulic system are as follows:

1. To provide a chamber in which any change in the volume of fluid in a hydraulic circuit can be accommodated. When the cylinder extends, there is an increased volume of fluid in the circuit and consequently there is a decrease in the reservoir level.
2. To provide a filling point for the system.
3. To serve as a storage space for the hydraulic fluid used in the system.
4. It is used as the location where the fluid is conditioned.
5. To provide a volume of fluid which is relatively stationery to allow entrained air to separate out and heavy contaminants to settle. The reservoir is where sludge, water and metal slips settle.
6. It is a place where the entrained air picked up by the oil is allowed to escape.
7. To accomplish the dissipation of heat by its proper design and to provide a radiating and convective surface to allow the fluid to cool.

A baffle plate extends lengthwise across the center of the tank. The purpose of the baffle plate is to separate the pump inlet line from the return line to prevent the same fluid from recirculating continuously within the tank. The functions of a baffle plate are as follows:

1. To permit foreign substances to settle to bottom.
2. To allow entrained air to escape from oil.
3. To prevent localized turbulence in the reservoir.
4. To promote heat dissipation through reservoir walls.

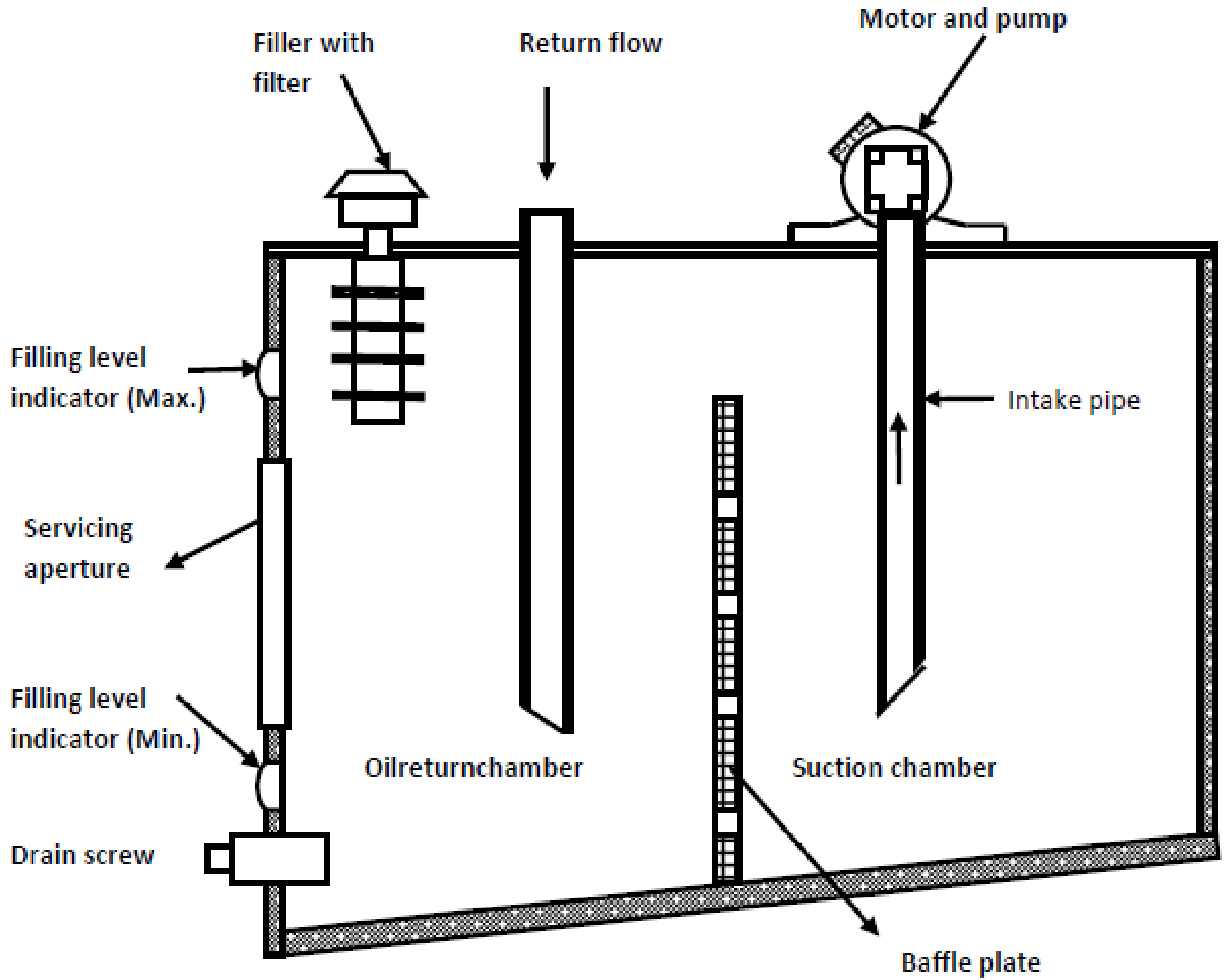


Figure 1.11 Hydraulic reservoir.

1.4.3 Sizing of the Reservoir

The reservoir capacity should be adequate to cater for changes in fluid volume within the system, and with sufficient surface area to provide system cooling. An oversize reservoir can present some disadvantages such as increased cost, size and longer warming-up periods when starting from cold. There are many empirical rules for sizing reservoirs.

The sizing of a reservoir is based on the following criteria:

1. The minimum reservoir capacity should be twice the pump delivery per minute. This must be regarded as an absolute minimum and may not be sufficient to allow for the volume changes in the system.
2. The reservoir capacity should be three to four times the pump delivery per minute. This may well be too high a volume for mobile application.
3. The reservoir capacity should be 2–15 L per installed horse power. This may result in very large reservoirs when high-pressure systems are used.
4. It must make allowance for dirt and chips to settle and for air to escape.
5. It must be able to hold all the oil.
6. It must maintain the oil level high enough to prevent the whirlpool effect.
7. It should have a large surface area to dissipate heat generated in the system.
8. It should have an adequate air space to allow for the thermal expansion of oil.

Table 1.1 Operating temperatures for various types of fluids

Temperature	**Mineral Oil (°C)**	**Water in Oil (60/40) (°C)**	**Water Glycol (°C)**	**Phosphate Ester (°C)**
Maximum local temperature	100	65	65	150
Maximum temperature for continuous operation	65	40	40	95
Maximum temperature for optimum fluid life	40	25	25	65

1. Water tube coolers: Water tube oil coolers consist of a series of interconnected copper tubes through which the cooling water passes surrounded by a jacket through which the hydraulic fluid passes. It is quiet in operation and can be arranged so that the oil pressure is higher than the water pressure; consequently, any leakage is more likely to be of oil into the water that is less serious than the contamination of the hydraulic fluid. When a water tube cooler is used, there is considerable flow of water involved and a separate water cooling tower and a circulating water supply may be necessary. Usually, the water supply is thermostatically controlled so that it is only switched ON when required.

2. Air blast coolers: An air blast cooler is similar in construction to a vehicle radiator with a powered air fan. It should be situated in a cool area so that cold air is blown over the radiator. An air blast cooler tends to be noisy but on small installations is preferable to water coolers owing to running and installation costs. Air blast coolers are now available to fit between the pump and the electric motor as part of the bell housing and coupling. Such coolers do not need a separate electric motor drive, but it is necessary to take into account the extra power needed when sizing the pump drive.

1. Filters: They are devices whose primary function is the retention, by some fine porous medium, of insoluble contaminants from fluid. Filters are used to pick up smaller contaminant particles because they are able to accumulate them better than a strainer. Generally, a filter consists of fabricated steel housing with an inlet and an outlet. The filter elements are held in position by springs or other retaining devices. Because the filter element is not capable of being cleaned, that is, when the filter becomes dirty, it is discarded and replaced by a new one. Particle sizes removed by filters are measured in microns. The smallest sized particle that can be removed is as small as 1 μm. A strainer is a device whose function is to remove large particles from a fluid using a wire screen. The smallest sized particle that can be removed by a strainer is as small as 0.15 mm or 150 μm.

2. Hydraulic strainers: A strainer is a coarse filter. Fluid flows more or less straight through it. A strainer is constructed of a fine wire mesh screen or of screening consisting of a specially processed wire of varying thickness wrapped around metal frames. It does not provide as fine a screening action as filters do, but offers less resistance to flow and is used in pump suction lines where pressure drop must be kept to a minimum. A strainer should be as large as possible or wherever this is not practical, two or more may be used in parallel.

1. According to the filtering methods:

- **Mechanical filters:** This type normally contains a metal or cloth screen or a series of metal disks separated by thin spacers. Mechanical filters are capable of removing only relatively coarse particles from the fluid.
- **Absorption filters:** These filters are porous and permeable materials such as paper, wood pulp, diatomaceous earth, cloth, cellulose and asbestos. Paper filters are impregnated with a resin to provide added strength. In this type of filters, the particles are actually absorbed as the fluid permeates the material. Hence, these filters are used for extremely small particle filtration.
- **Adsorbent filters:** Adsorption is a surface phenomenon and refers to the tendency of particles to cling to the surface of the filters. Thus, the capacity of such a filter depends on the amount of surface area available. Adsorbent materials used include activated clay and chemically treated paper.

2. According to the size of pores in the material:

- **Surface filters:** These are nothing but simple screens used to clean oil passing through their pores. The screen thickness is very thin and dirty unwanted particles are collected at the top surface of the screen when the oil passes, for example, strainer.
- **Depth filters:** These contain a thick-walled filter medium through which the oil is made to flow and the undesirable foreign particles are retained. Much finer particles are arrested and the capacity is much higher than surface filters.

Advantages of suction filters:

(a) A suction filter protects the pump from dirt in the reservoir. Because the suction filter is outside the reservoir, an indicator telling when the filter element is dirty can be used.
(b) The filter element can be serviced without dismantling the suction line or reservoir (easy to maintain).

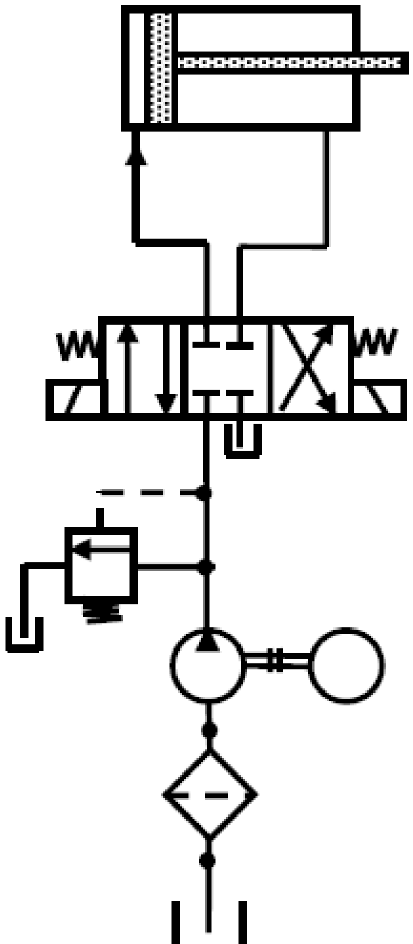


Figure 1.15 Suction filter.

Disadvantages of suction filters:

(a) A suction filter may starve the pump if not sized properly.

Advantages of a pressure line filter:

(a) A pressure filter can filter very fine contaminants because the system pressure is available to push the fluid through the element.
(b) A pressure filter can protect a specific component from the harm of deteriorating particles generated from an upstream component.

Disadvantages of a pressure line filter:

(a) The housing of a pressure filter must be designed for high pressure because it operates at full system pressure. This makes the filter expensive.
(b) If pressure differential and fluid velocity are high enough, dirt can be pushed through the element or the element may tear or collapse.

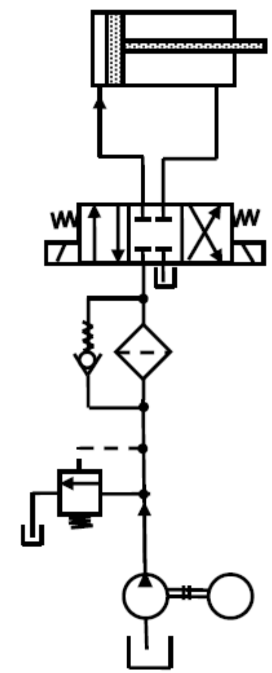


Figure 1.16 Pressure filter.

Advantages of a return line filter:

(a)A return line filter catches the dirt in the system before it enters the reservoir.
(b)The filter housing does not operate under full system pressure and is therefore less expensive than a pressure filter.

Disadvantages of a return line filter:

(a) There is no direct protection for circuit components.
(b) In return line full flow filters, flow surges from discharging cylinders, actuators and accumulators must be considered when sizing.

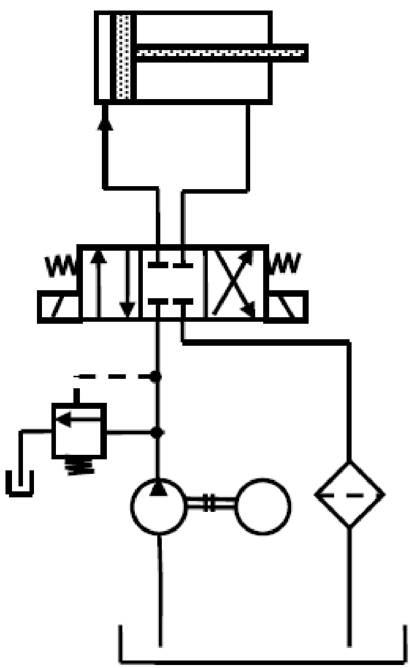


Figure 1.17 Return line filter.

1.6.3 Beta Ratio of Filters

Filters are rated according to the smallest size of particles they can trap. Filter ratings are identified by nominal and absolute values in micrometers. A filter with a nominal rating of 10 µm is supposed to trap up to 95% of the entering particles greater than 10 µm in size. The absolute rating represents the size of the largest pore or opening in the filter and thus indicates the largest size particle that could go through. Hence, absolute rating of a 10 µm nominal size filter would be greater than 10 µm.

A better parameter for establishing how well a filter traps particles is called the beta ratio or beta rating. The beta ratio is determined during laboratory testing of a filter receiving a steady-state flow containing a fine dust of selected particle size. The test begins with a clean filter and ends when pressure drop across the filter reaches a specified value indicating that the filter has reached the saturation point. This occurs when contaminant capacity has been reached.

By mathematical definition, the beta ratio equals the number of upstream particles of size greater than Nµm divided by the number of downstream particles having size greater than Nµm where N is the selected particle size for the given filter. The ratio is represented by the following equation:

$$\text{Beta ratio} = \frac{\text{No. of upstream particles of size} > N\ \mu\text{m}}{\text{No. of downstream particles of size} > N\ \mu\text{m}} \qquad (1.4)$$

A beta ratio of 1 would mean that no particle above specified N are trapped by the filter. A beta ratio of 50 means that 50 particles are trapped for every one that gets through. Most filters have a beta ratio greater than 75:

$$\text{Beta efficiency} = \frac{\text{No. of upstream particles} - \text{No. of downstre am particles}}{\text{No. of upstream particles}}$$

Thus,

$$\text{Beta efficiency} = 1 - \frac{1}{\text{Beta ratio}}$$

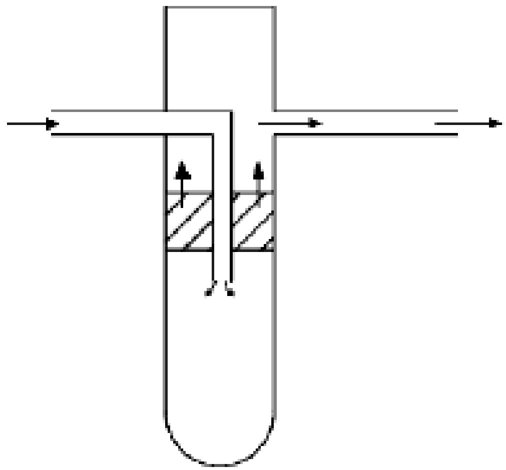


Figure 1.18 Full flow filter.

Proportional filters (bypass filters): In some hydraulic system applications, only a portion of oil is passed through the filter instead of entire volume and the main flow is directly passed without filtration through a restricted passage (Fig. 1.19).

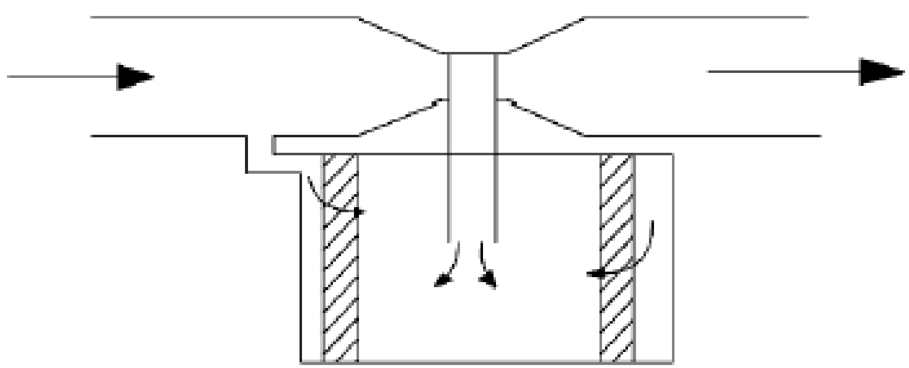


Figure 1.19 Proportional filter.

1.7 Heat Exchangers

The heating up of hydraulic oil beyond tolerable limits in an otherwise well-designed hydraulic systemis usually a phenomenon associated with high-pressure, high-flow systems cycling at high frequencies. The input power in such systems is usually more than 40 kW.

Heat is generated in hydraulic systems because no component can operate at 100% efficiency. Significant sources of heat include the pump, pressure-relief valves and flow control valves. Heat can cause the hydraulic fluid temperature to exceed its normal operating range of 35–70°C. Excessive temperature hastens the oxidation of the hydraulic oil and causes it to become too thin. This promotes deterioration of seals and packing and accelerates wear between closely fitting parts of hydraulic components of valves, pumps and actuators.

The sum of the energy less the energy dissipated through the reservoirs and flow lines by convection and radiation constitutes the net energy that the heat exchanger must dissipate or the rate of heat transfer. The maximum oil temperature at the inlet point to the heat exchanger can now be determined by using the equation

$$H = mC_{\mathrm{p}}(T_1 - T_0) \tag{1.6}$$

Here H is the heat transfer rate in kJ/s; C_p is the specific heat at constant pressure that for hydraulic oil = 1.97 kJ/kg°K; T_1 is the oil temperature at the inlet to the heat exchanger, which typically lies between 55 and 65°C; T_0 is the oil temperature at the outlet, which is the desired oil temperature that the heat exchanger must maintain. A suitable value may be assumed for T_0 which is consistent with the system requirement. Also m is the mass flow rate determined by the equation

$$m = \rho \times q \quad (\mathrm{kg/s}) \tag{1.7}$$

where ρ is the density of hydraulic oil in kg/m^3 and q is the oil flow rate in m^3/s. If the difference between T_1 and T_0 is large, then it justifies the need for a heat exchanger. If the difference is only marginal, the heat

$$Q = \frac{(860)(P_{\mathrm{w}})}{T_1 - T_{\mathrm{w}}} \tag{1.8}$$

where P_w is the wasted energy in kW, Q is the water flow rate through the heat exchanger in L/min, T_1 is the temperature of the oil and T_w is the water temperature. When the flow capacity as determined from the equation is much larger than the capacity of available heat exchangers, then two smaller heat exchangers of equal sizes can be used in parallel. Standard heat exchangers would have the oil-to-water ratio of either 1:1 or 2:1. This means that for every liter of oil circulated, one or half a liter of water must be circulated to

Example 1.2

Determine the beta ratio of a filter when, during test operation, 30000 particles greater than 20 μm enter the filter and 1050 of these particles pass through the filter. What is the beta efficiency?

Solution: We have

$$\text{Beta ratio} = \frac{\text{No. of upstream particles of size} > N\,\mu\text{m}}{\text{No. of downstream paricles of size} > N\,\mu\text{m}} = \frac{30000}{1050} = 28.6$$

$$\text{Beta efficiency} = \frac{\text{No. of upstream particles} - \text{No. of downstream particles}}{\text{No. of upstream particles}} = \frac{30000 - 1050}{30000} = 96.5\%$$

Beta efficiency could also be calculated as

$$\text{Beta efficiency} = 1 - \frac{1}{\text{Beta ratio}} = 1 - \frac{1}{28.6} = 96.5\%$$

Example 1.4

Oil at 49°C and 69 bar is flowing through a pressure-relief valve at 38 LPM. What is the downstream oil temperature?

Solution: First we calculate the power lost/wasted:

$$\text{Power lost} = p\ (\text{bar}) \times Q\ (\text{LPM})$$

$$= (69 \times 10^5\ \text{N/m}^2)\ (38 \times 10^{-3}\ \text{m}^3/\text{min}) \times 1/60\ (\text{s})$$

$$= 4370\ \text{W} = 4.37\ \text{kW}$$

Next we calculate the oil flow rate in units of kg/s and the temperature increase.

$$\text{Oil flow rate (kg/s)} = 895 \times \text{Oil flow rate (m}^3\text{/s)}$$

$$= 895 \times 38 \times \frac{10^{-3}}{60} = 0.6\ \text{kg/s}$$

Now

$$\text{Temperature}\ (^\circ\text{C}) = \frac{\text{Heat generation rate (kW)}}{\text{Oil specific heat}\left(\frac{\text{kJ}}{\text{kg}}\,^\circ\text{C}\right) \times \text{Oil flow rate (kg / s)}}$$

$$= \frac{4.37}{1.8 \times 0.6} = 4^\circ\text{C}$$

Downstream oil temperature = 49 + 4 = 53 °C

Example 1.9

What would be an adequate size of a reservoir for a hydraulic system using 0.0005 m^3/s pump?

Solution: Size of the reservoir is three to four times the capacity of the pump, that is

$$\text{Size of reservoir} = 4 \times \text{capacity of pump (LPM)}$$

$$= 4 \times 0.0005 = 4 \times 30 (\text{LPM}) = 120 \text{ L tank is required}$$

Example 1.10

A pump delivers oil to a hydraulic motor at 20 LPM at a pressure of 15 MPa. If the motor delivers 4 kW and 80% of the power loss is due to internal leakage, which heats the oil, calculate the heat-generation rate in kJ/min.

Solution: We have

$$\text{Pump power} = \frac{0.02}{60} \times 150000 = 5 \text{ kW}$$

Motor delivers 4 kW; therefore loss is 1 kW. Now

$$\text{Power loss due to leakage} = 0.8 \times 1 = 0.8 \text{ kW}$$

$$\text{Power loss due to leakage} = 0.8 \times 60 = 48 \text{ kJ / min}$$

Example 1.8

A hydraulic pump operates at 140 bar and delivers oil at 0.001 m^3/s to a hydraulic actuator. Oil discharg through a pressure relief valve during 60% of the cycle time. The pump has an overall efficiency of 82 and 15% of power is lost due to frictional pressure losses in the hydraulic lines. What heat exchang rating is required to dissipate all the generated heat?

Solution: We have

$$\text{Pump power loss} = \text{Pump power input} - \text{Pump power output}$$

$$\text{Pump power loss} = \frac{\text{Power output}}{\text{Overall efficiency}} - \text{Power output}$$

$$\text{Pump power loss} = \left\{\frac{1}{\eta_0} - 1\right\}\text{pump power output}$$

$$= \left(\frac{1}{82} - 1\right) \times \left(\frac{140 \times 10^5 \times 0.001}{1000}\right) = 3.073 \text{ kW}$$

$$= (0.60) \times \left(\frac{140 \times 10^5 \times 0.001}{1000}\right) = 8.4 \text{ kW}$$

Also

$$\text{Line average loss} = \{0.60\} \times 0.15 \times \left(\frac{140 \times 10^5 \times 0.001}{1000}\right) = 1.26 \text{ kW}$$

Therefore

$$\text{Total loss} = 3.073 + 8.4 + 1.26 = 12.77 \text{ kW}$$

Select heat exchanger rating of 12.77 kW.

ME 5451 – Hydraulics and Pneumatics

Lecture -15

Date: 13-05-2021   Time slot: 08:30-10:10 a.m.

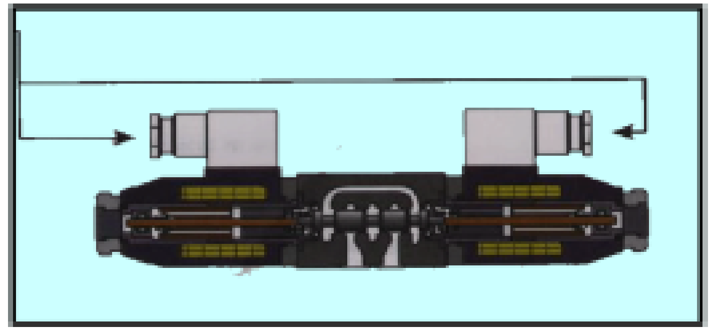
Contents

1. Review of Lecture 14
2. Seals
3. Reservoirs,
4. Heat Exchangers
5. switches

Course Instructor: Dr. A. Siddharthan

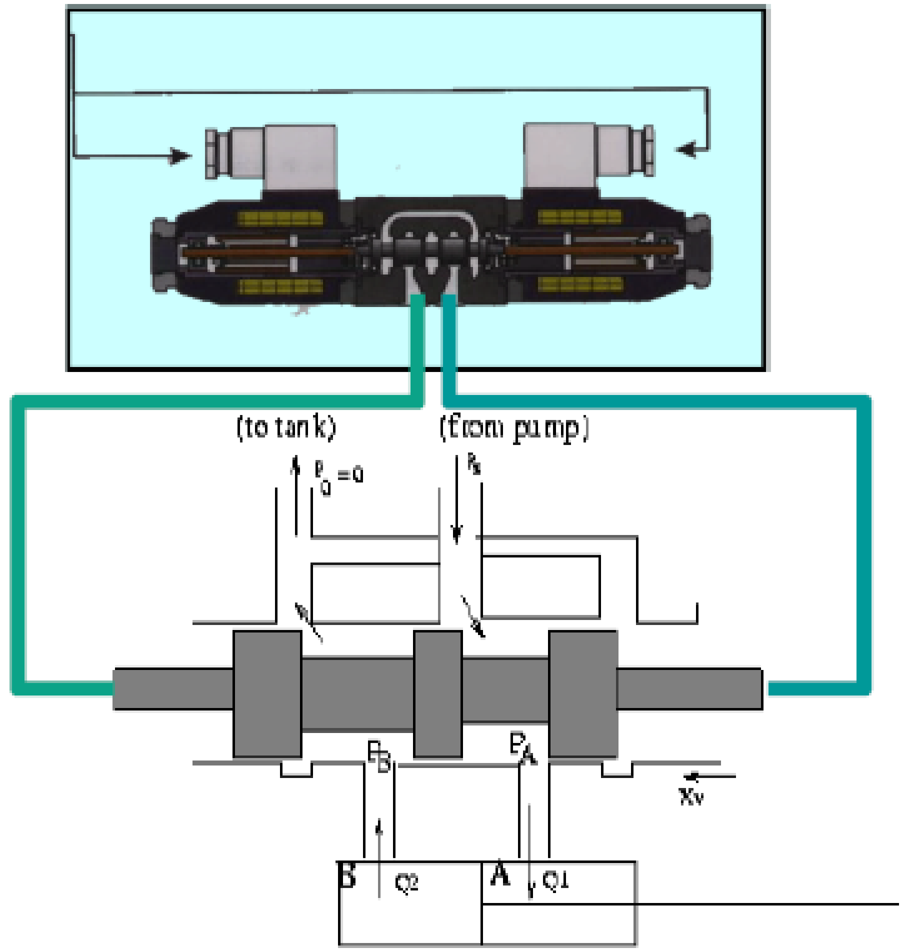
Single Stage Proportional Valves

- **Advantages:**
  - Simple design
  - Reliable
  - Cost effective



- **Disadvantages**:
  - Poor dynamic performance (bandwidth)
  - At high flow rates and bandwidths, large stroking force is needed
  - Large (and expensive) solenoids / torque motors needed.
  - Low end market …..

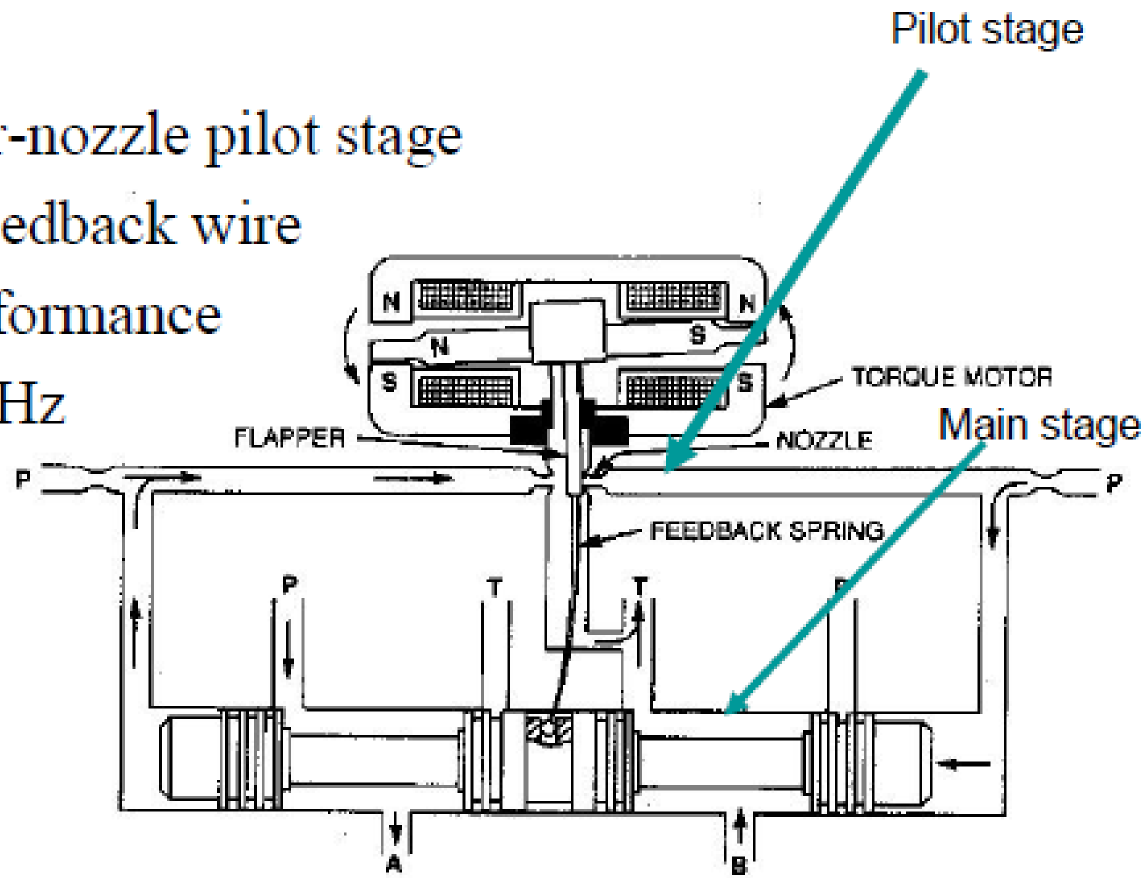
Multi-stage valves



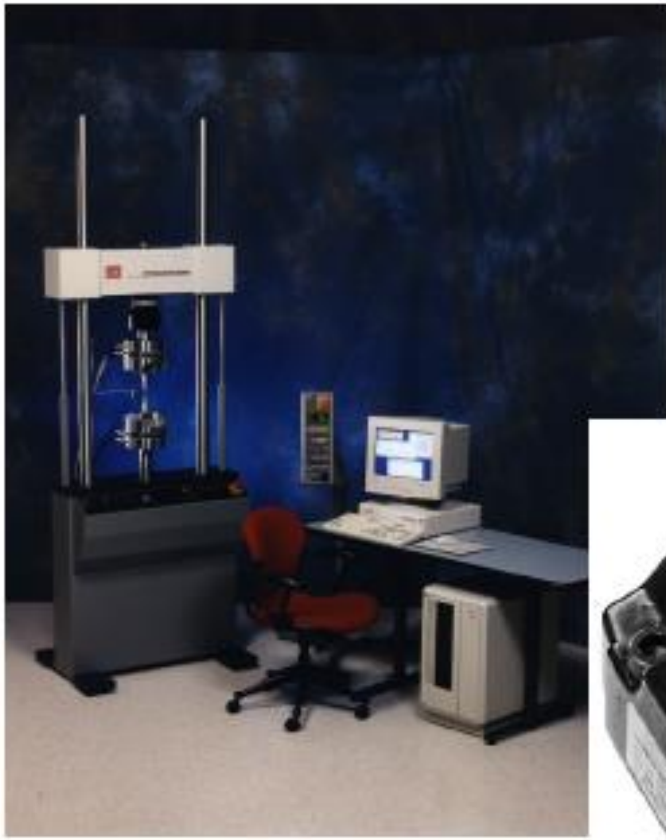
- Use hydraulic force to drive the spool …..

Electrohydraulic servo-valve

- Multi-stage valve
- Typically uses a flapper-nozzle pilot stage
- Built-in feedback via feedback wire
- Very high dynamic performance
- Bandwidth = 100-200+Hz

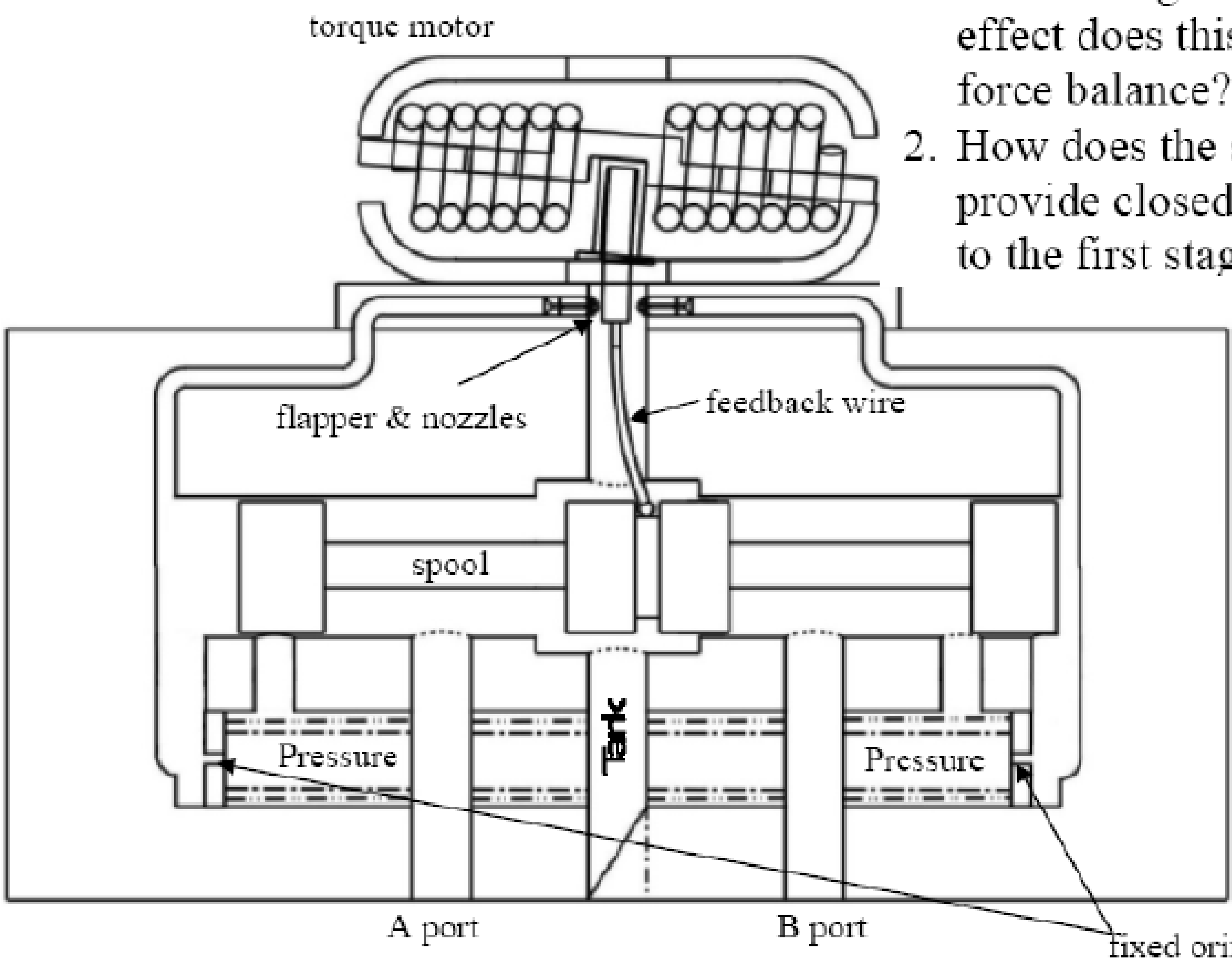


Servovalves



Servo-Valve Diagrams from:

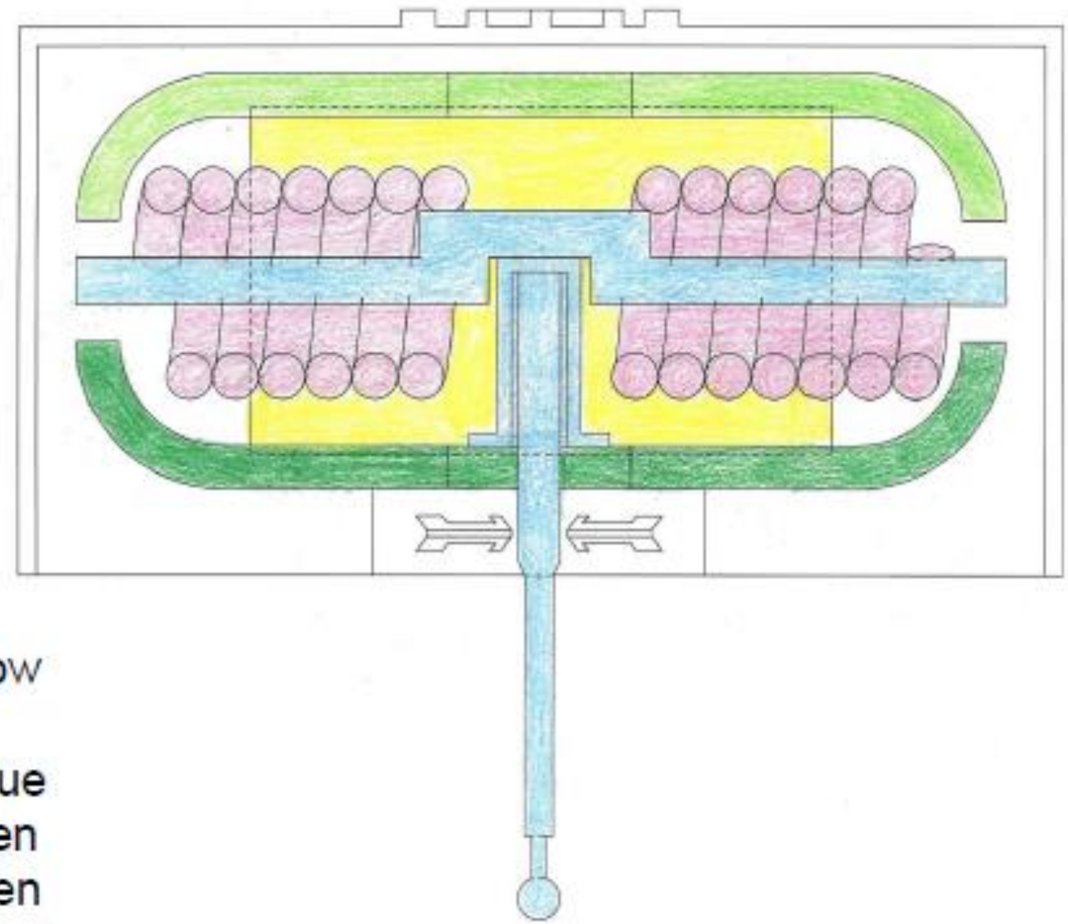
Servo Valve Function



1. Rotation of the torque motor restricts one jet and allow free flow through the other. What effect does this have on spool force balance?
2. How does the second stage provide closed-loop feedback to the first stage?

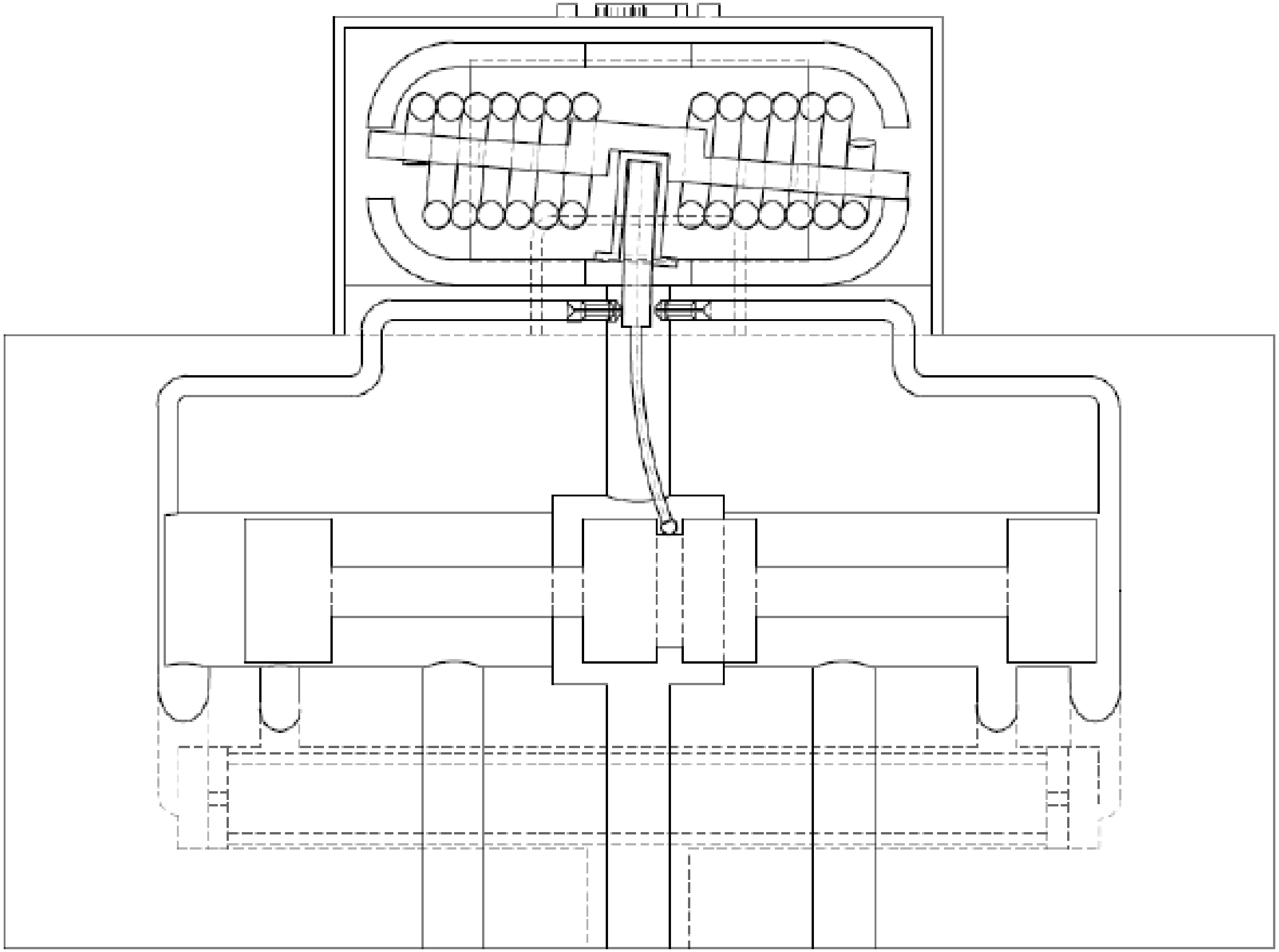
First Stage – Torque-motor-armature

Transforms electrical current into torque, which changes relative nozzle pressures

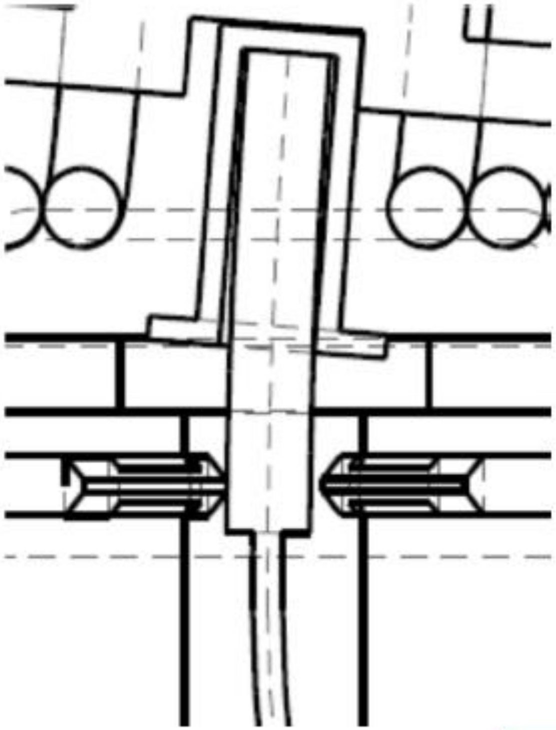


A. Permanent Magnet (2) - yellow
B. Wire Coil - pink
C. Armature Assembly – light blue
D. Upper Pole Piece – light green
E. Lower Pole Piece – dark green

First Stage - Armature rotation

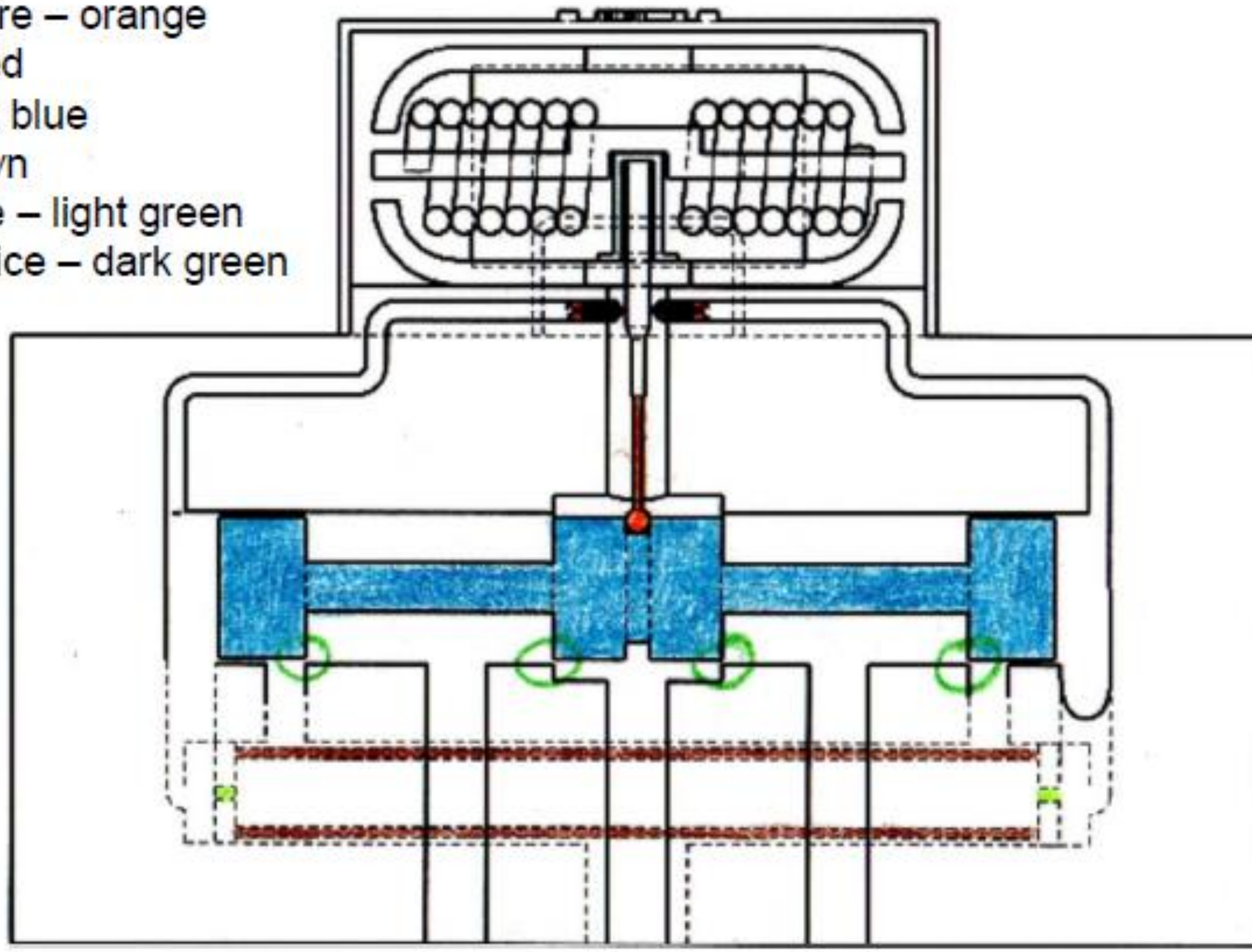


First Stage – Nozzle and Flapper



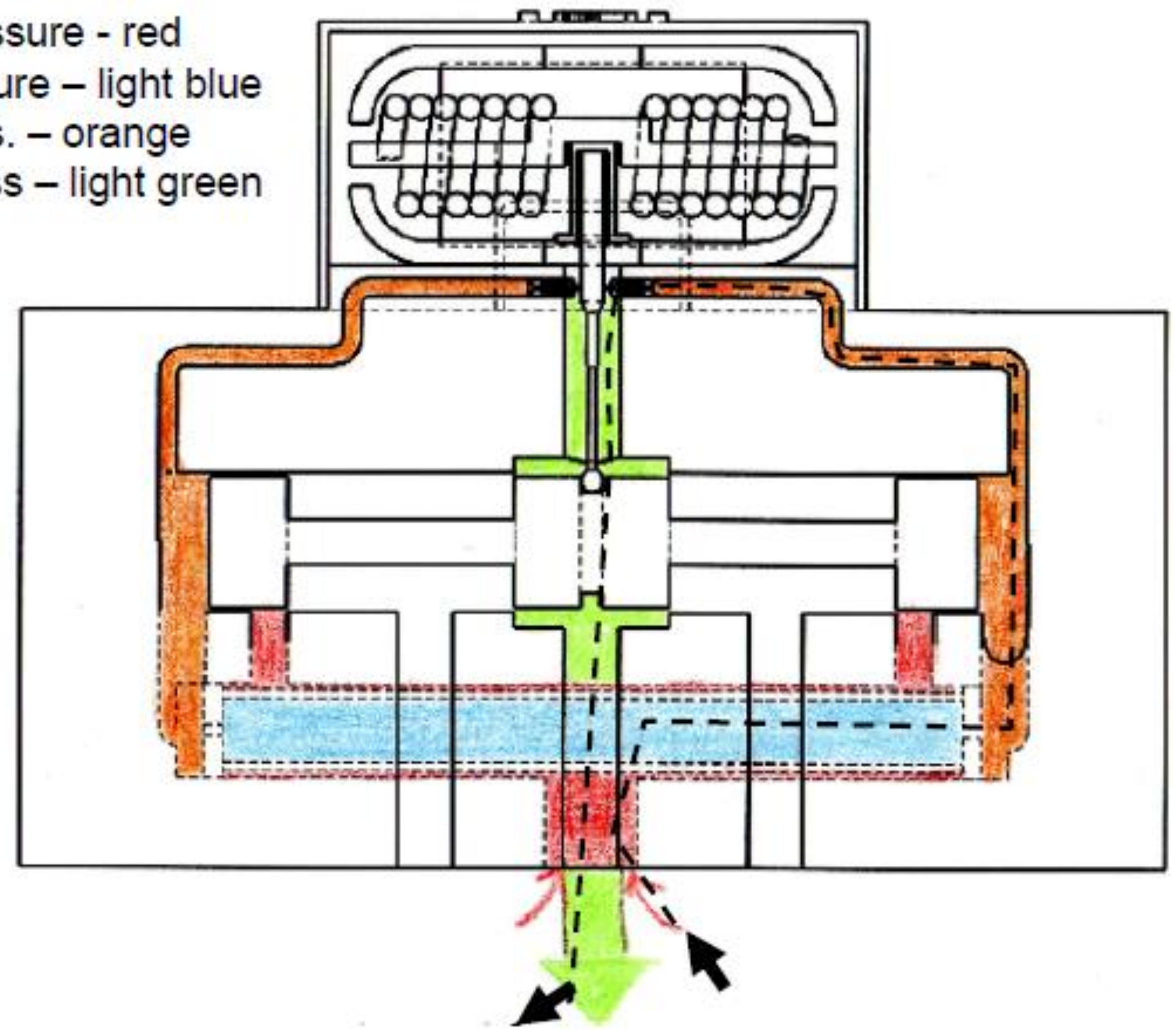
Second Stage

J. Feedback wire – orange
K. Nozzles – red
L. Spool – dark blue
M. Filter – brown
N. Fixed Orifice – light green
O. Variable orifice – dark green

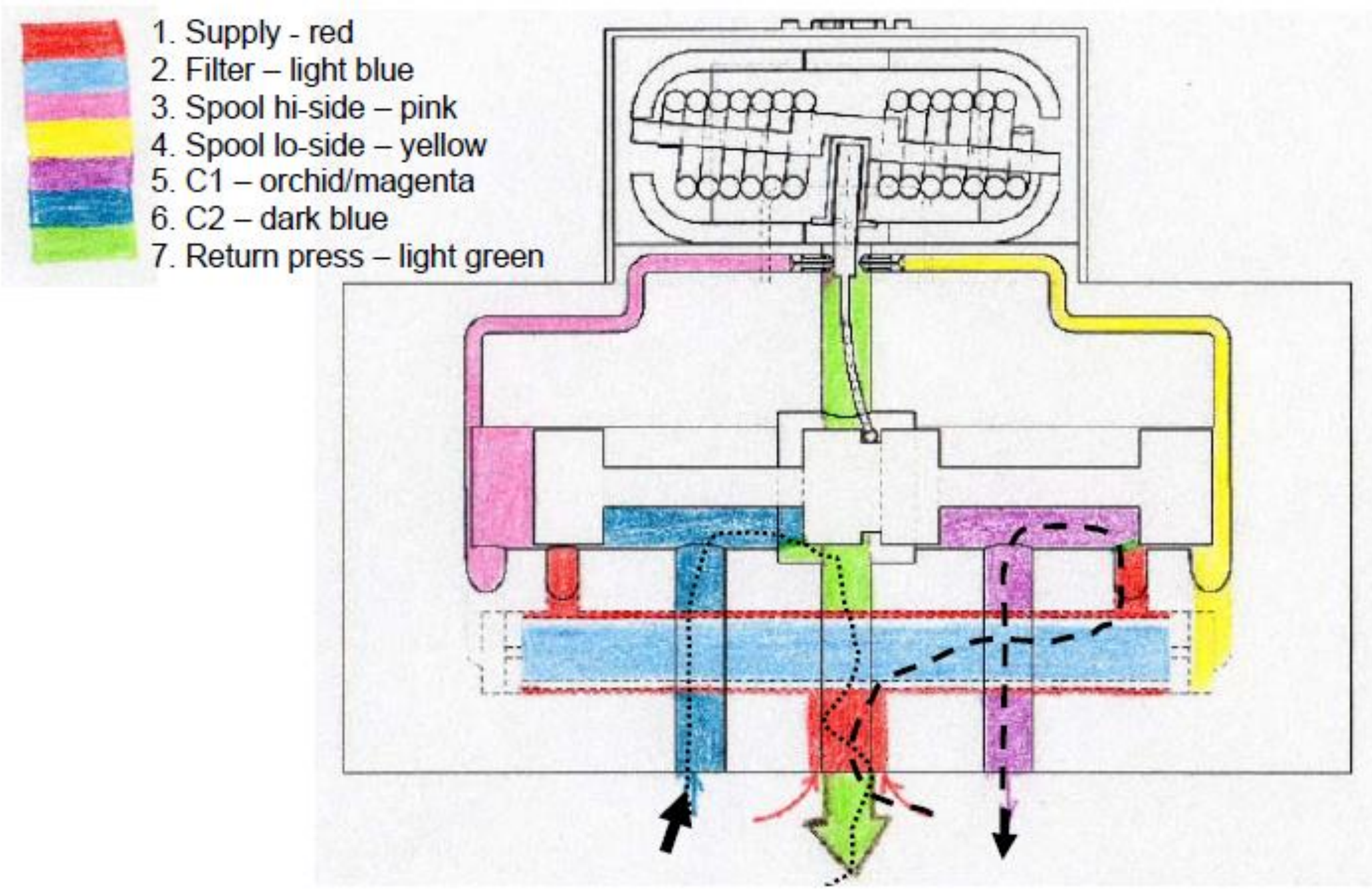


Null Position – Oil Flow path

1. Supply pressure - red
2. Filter pressure – light blue
3. Spool press. – orange
4. Return press – light green



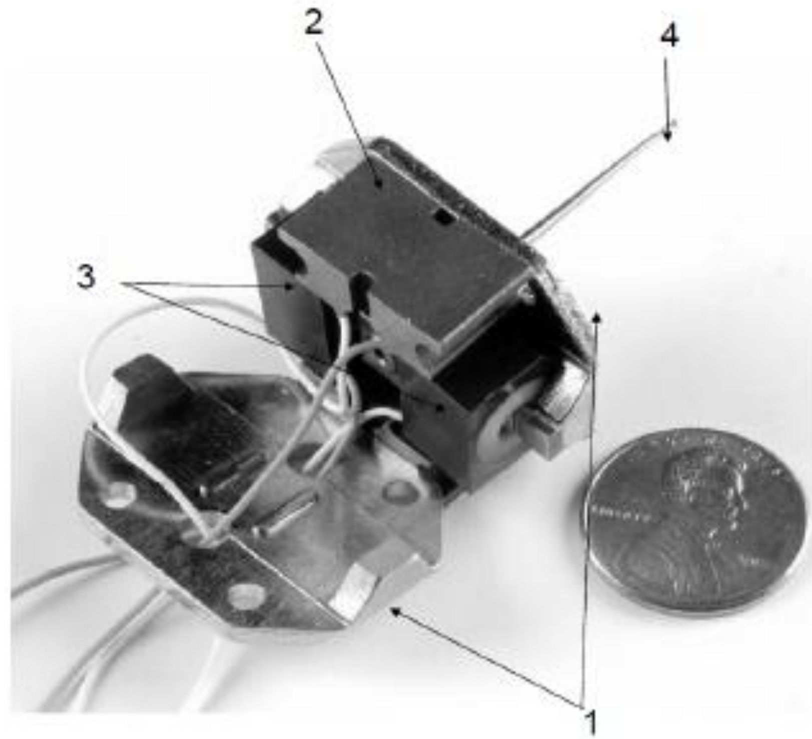
Activated Position – Oil Flow Path



Servo Valve Parts

1. Spool
2. Nozzle
3. Fixed Orifice
4. Tubular Filter
5. Disc Filter

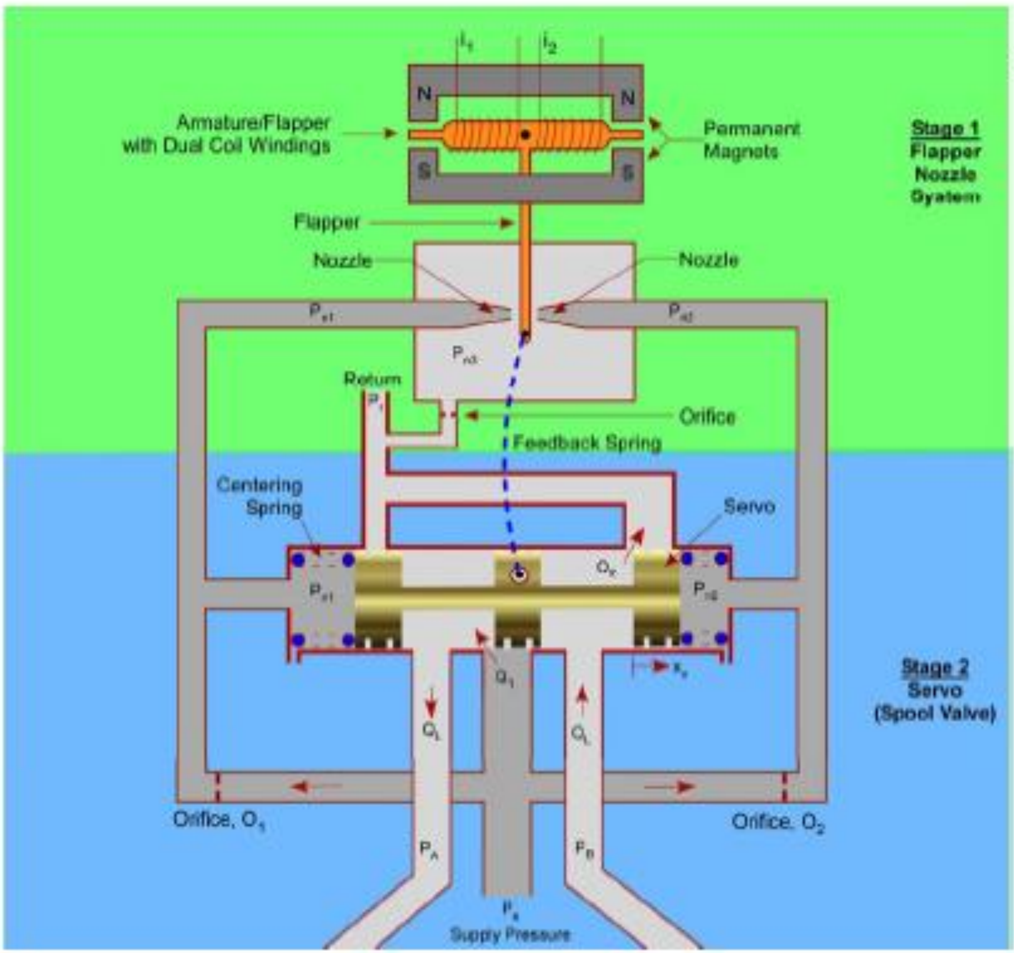
Torque Motor Parts

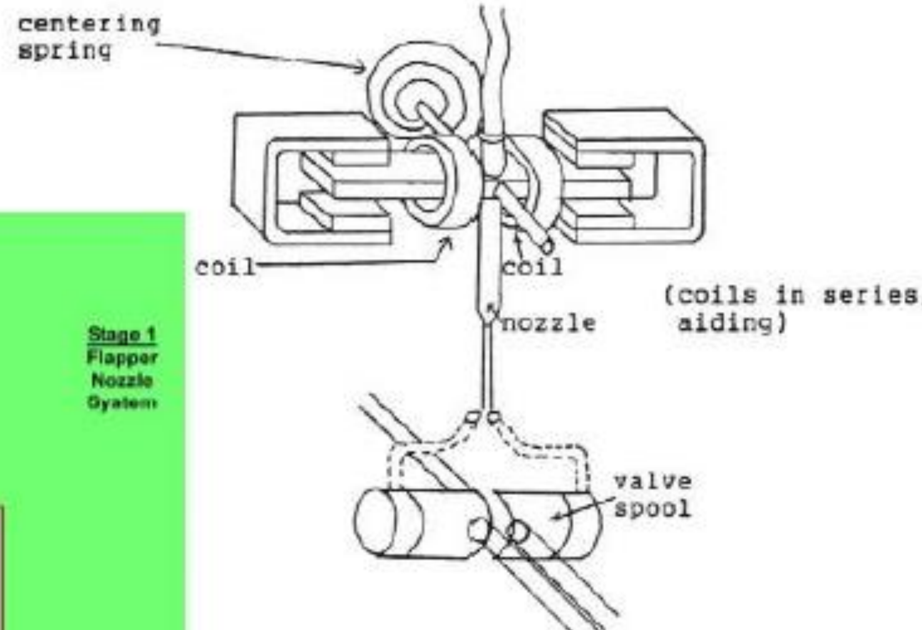


1. Torque motor armature assembly with signal connector
2. Permanent magnet
3. Electromagnet coil
4. Armature/flexure tube/flapper/feedback wire assembly

1. Upper and lower pole pieces
2. Permanent magnet
3. Electromagnet coil
4. Feedback wire

Nozzle & Flapper vs. Jet Tube





Servo Valve Benefits / Characteristics

- Feedback from 2nd Stage to 1st
- Low Mass Torque Motor
- Large ΔP in 1st Stage
- Frictionless / Isolated 1st Stage
- Mechanically / Hydraulically Symmetric
- 300-1000 Hz Natural Frequency
- Good Linearity

Negatives

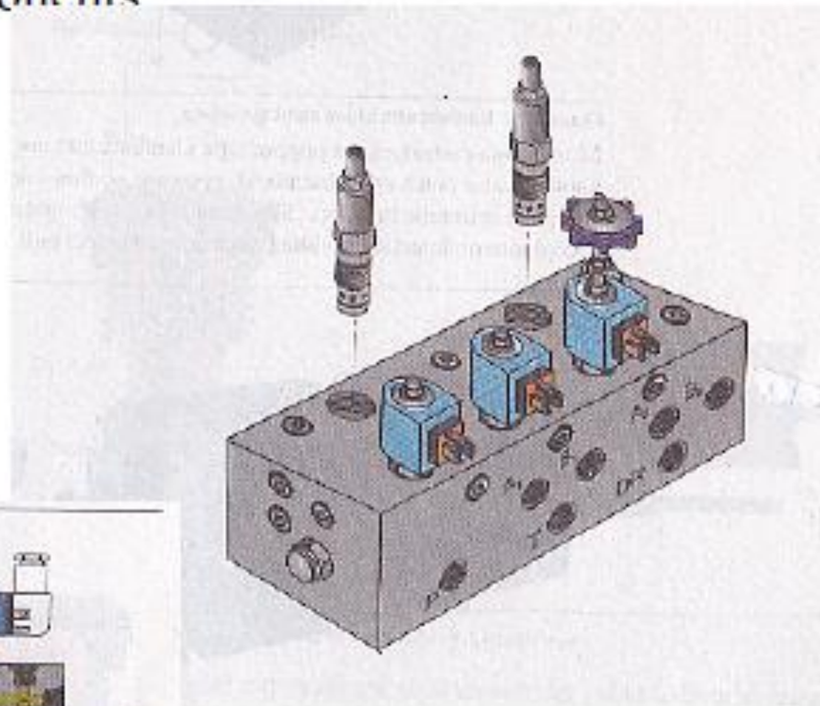
- Contamination Susceptible
  - < 3 micron filtration
- High Cost
  - Torque motor
  - Critically Lapped

Cartridge Valves

- Integrate circuits with many components using a manifold

- Manifold = aluminum block with internal passages

- Standard configurations



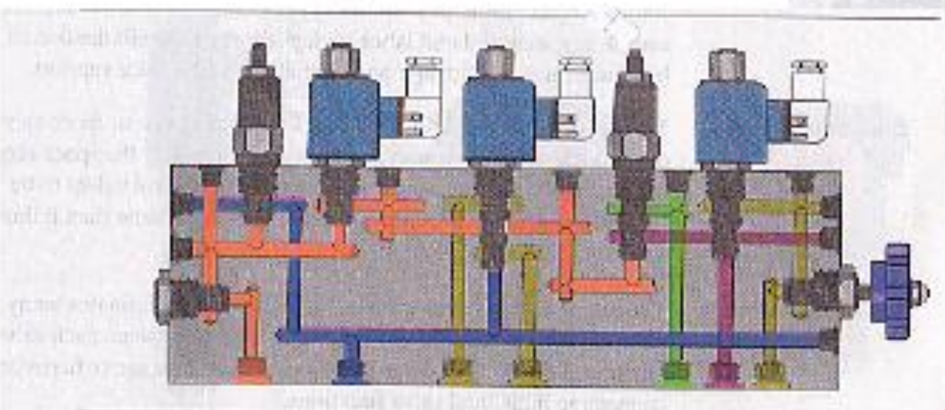


Figure 12.8 Internal plumbing of a screw-in cartridge valve manifold.

Screw-in Cartridge Valve

- Single piece implements the function
- Plugs into standard manifold porting
  - manifold itself being custom built

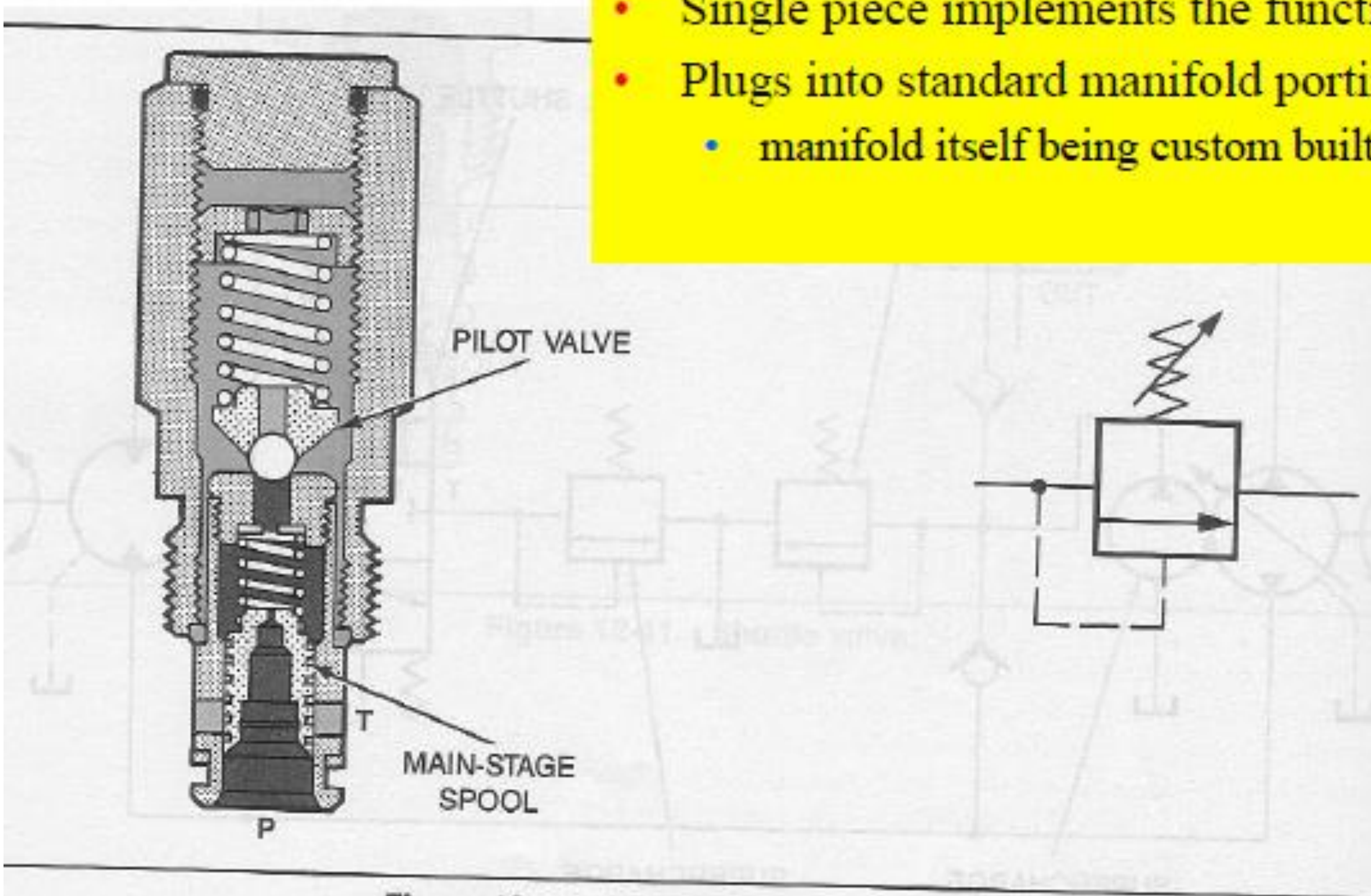


Figure 12-44. Pilot-operated relief valve.

ME 5451 – Hydraulics and Pneumatics

Lecture -16

Date: 13-05-2021 Time slot: 8:30-10:10 a.m.

Contents

1. Problems on heat and power losses
2. Accumulators
3. Intensifiers
4. Industrial hydraulic circuits

Course Instructor: Dr. A. Siddharthan

ACCUMULATORS

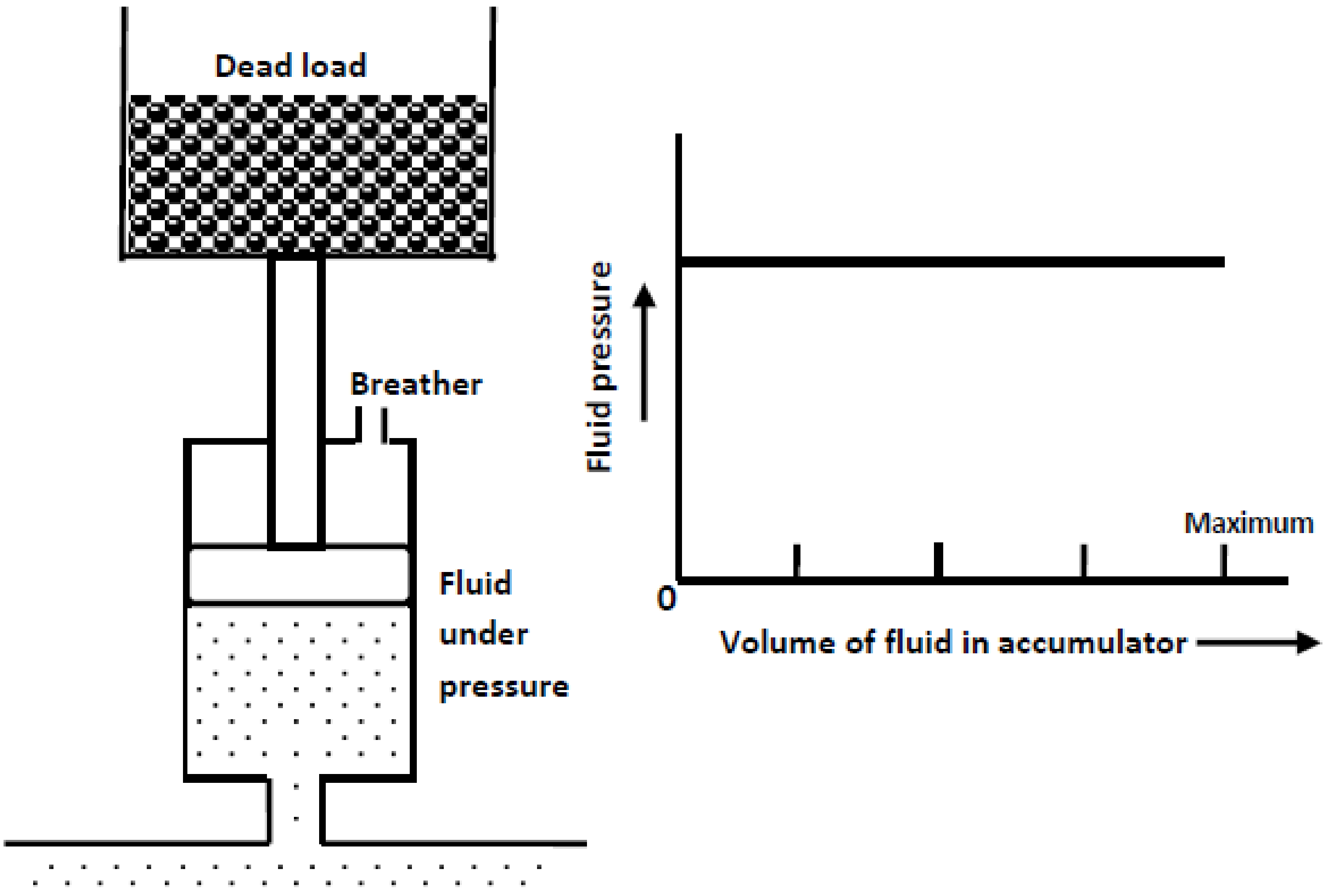


Figure 1.1 Dead weight accumulator.

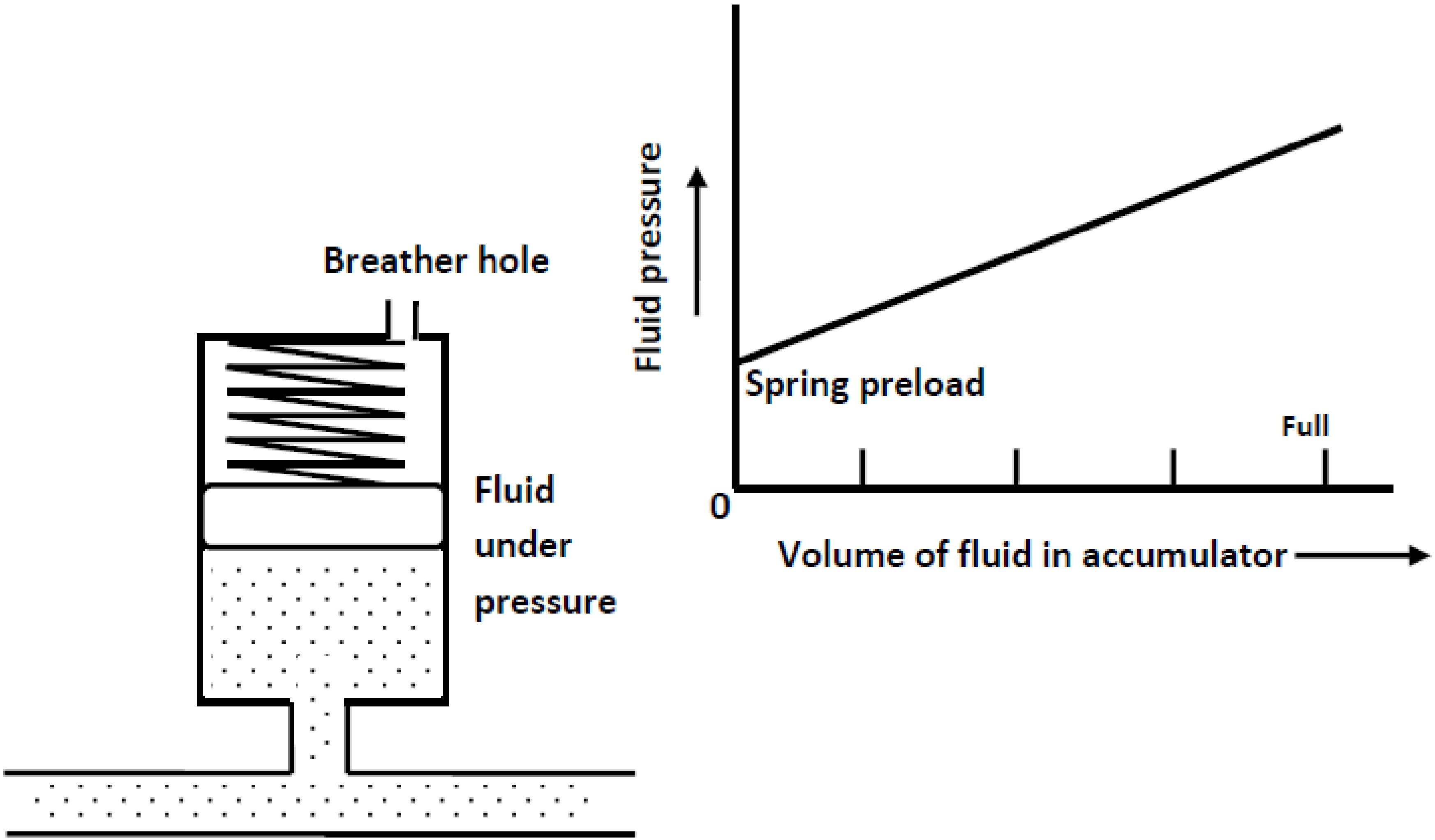


Figure 1.2 Spring-loaded accumulator.

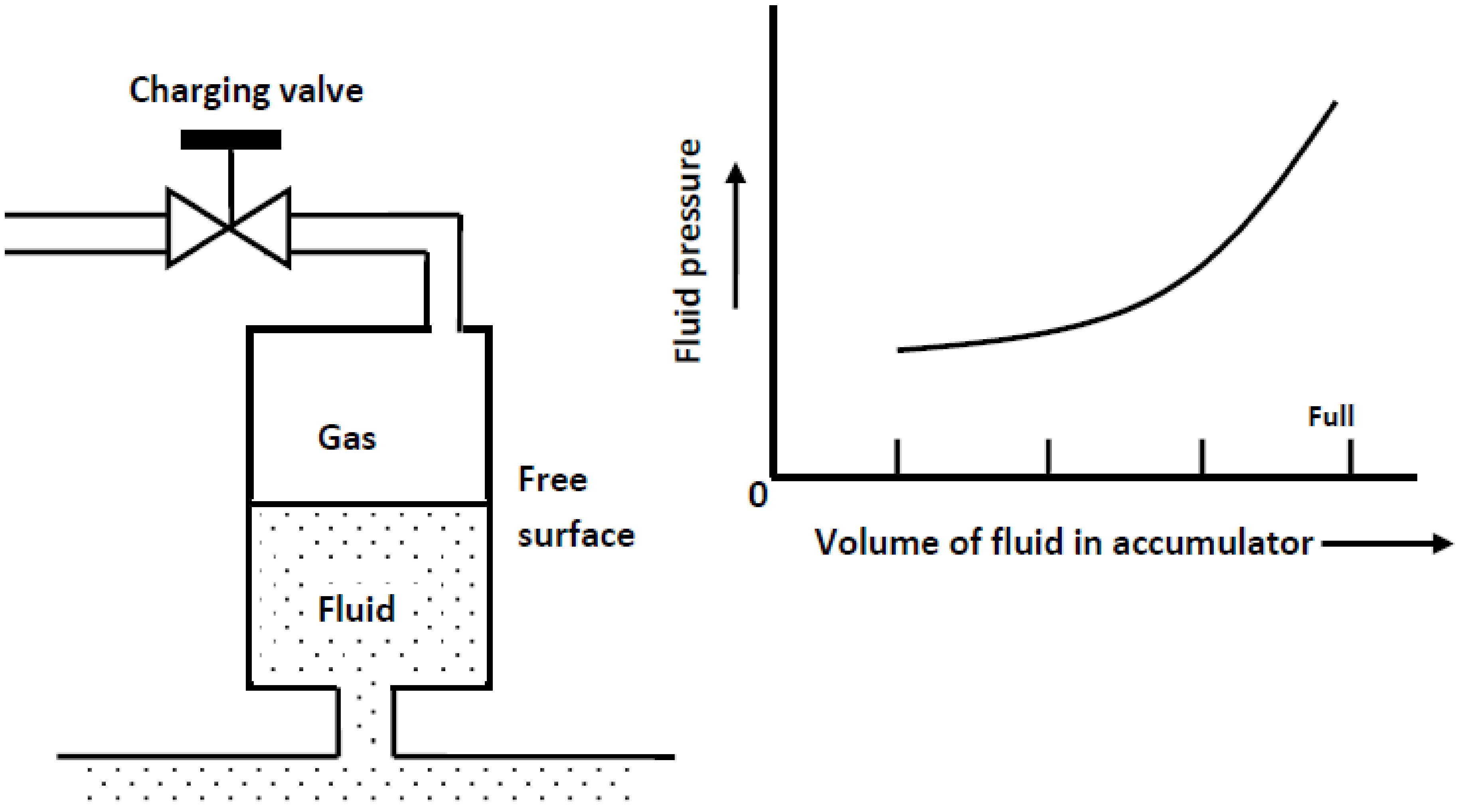


Figure 1.3 Gas-loaded accumulator.

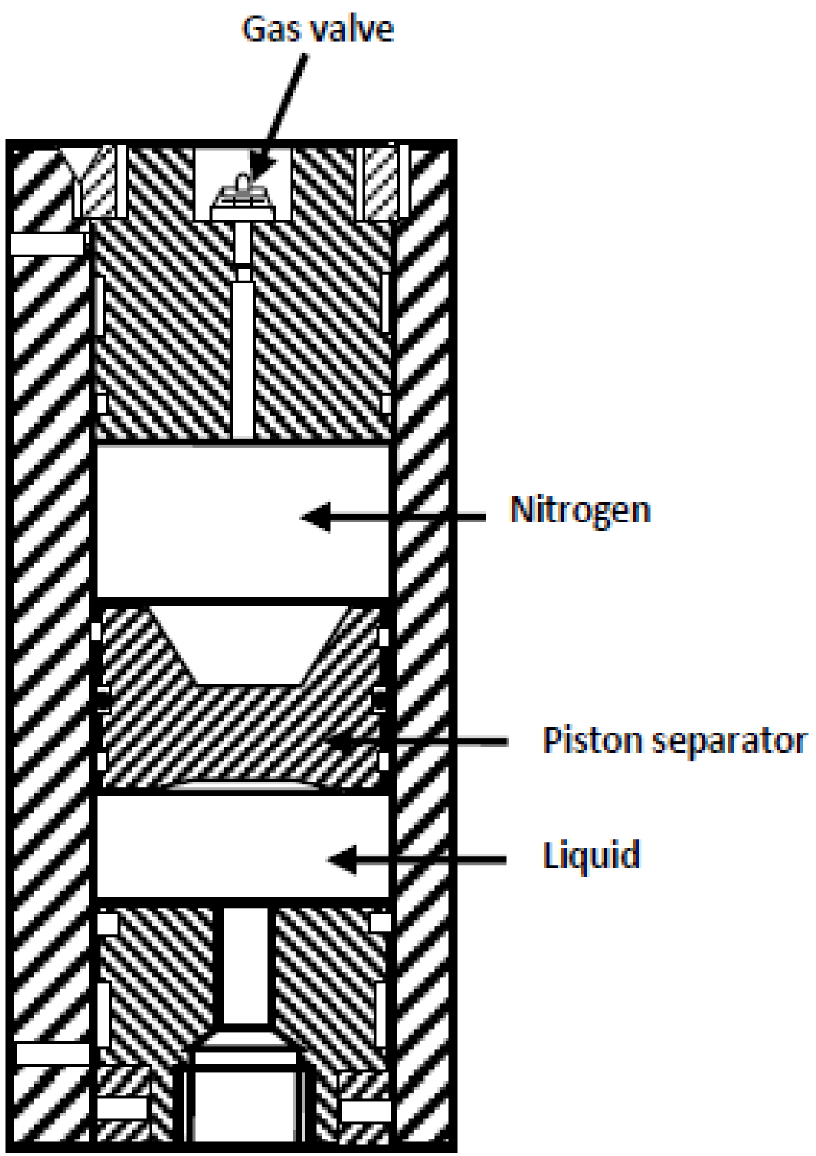


Figure 1.4 Piston-type accumulator.

*Diaphragm accumulator:*In this type, the hydraulic fluid and nitrogen gas are separated by a synthetic rubber diaphragm. Schematic diagram of diaphragm accumulator is shown in Fig. 1.5. The advantage of a diaphragm accumulator over a piston accumulator is that it has no sliding surface that requires lubrication and can therefore be used with fluids having poor lubricating qualities. It is less sensitive to contamination due to lack of any close-fitting components.

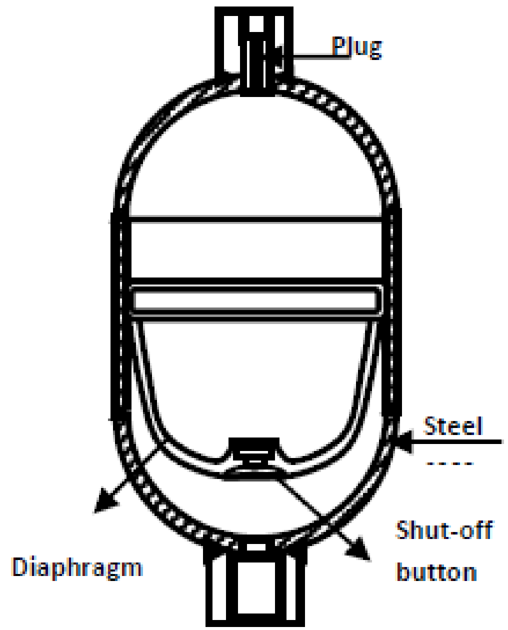


Figure 1.5 Diaphragm-type accumulator.

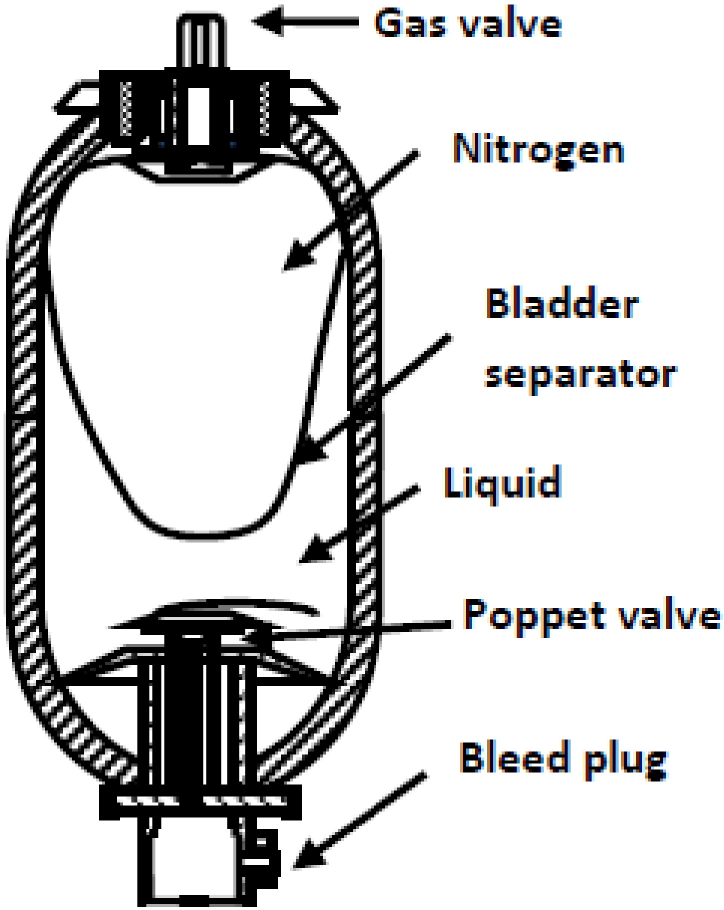


Figure 1.6 Bladder-type accumulator.

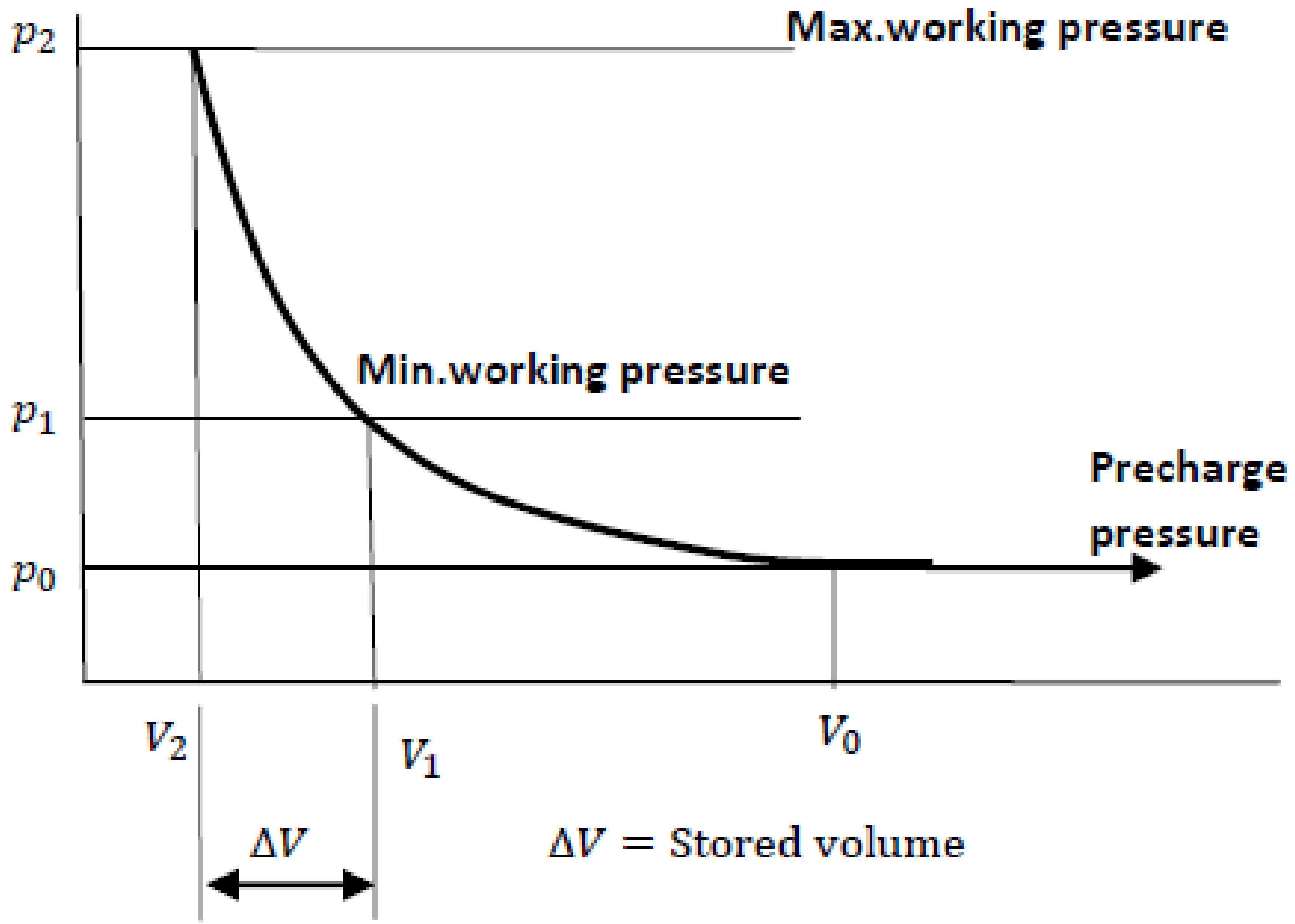


Figure 1.7 Pressure–volume diagram.

1.2.2 Sizing Accumulators for Adiabatic Condition

Starting from the basic formula $p_0V_0^n = p_1V_1^n = p_2V_2^n$, it can be shown that for adiabatic conditions, the values of maximum nitrogen volume V_0 at the pre-charge pressure p_0 and the stored volume of oil are given by the following equations:

$$\Delta V = V_0\left[\left(\frac{p_0}{p_1}\right)^{0.7143} - \left(\frac{p_0}{p_2}\right)^{0.7143}\right]$$

$$\Rightarrow V_0 = \frac{\Delta V}{[(p_0 / p_1)^{0.7143} - (p_0 / p_2)^{0.7143}]} \qquad (1.9)$$

Here, the value of the polytropic exponent n is taken equal to 1.4 and therefore $1/n$ becomes 0.7143. Here again intermediate values can be used for more accurate results.

1.2.3 Sizing Accumulators for Emergency Reserve

This is a typical application where both isothermal and adiabatic conditions prevail due to slow storage and quick discharge. For this condition, the accumulator volume is given by

$$V_0 = \frac{\Delta V(p_2 / p_0)}{[(p_2 / p_1)^{0.7143} - 1]}$$

$$\Rightarrow \Delta V = V_0 p_0 \frac{\left[\left(\frac{p_2}{p_1}\right)^{0.7143} - 1\right]}{p_2} \qquad (1.10)$$

1.2.4 Sizing Accumulators for Pulsation Damping

Pulsation damping is typically an adiabatic condition because both storage and discharge have to be accomplished in a very short time. Because pressure pulsation is a phenomenon associated with piston pumps, the stored volume ΔV is a product of the pump displacement q in liters and a constant $\boldsymbol{k}$ that depends on whether the pump is single-acting or double-acting and the number of pistons involved. Pressure pulsation is highest in a single-acting single-piston pump delivering large flows at high pressures. Here the k factor is about 0.69. A single-piston double-acting pump has the same k factor as that of a double-piston single-acting pump whose k factor is equal to 0.29. A three- or four-piston single-acting pump has a k factor of 0.12. For any other pump configuration, an average k factor of 0.05 can be taken with reasonable accuracy.

The stored volume is given by

$\Delta V = kq$ where q is the pump flow rate in LPM/RPM × number of piston. Now

$$V_0 = \frac{\Delta V}{[(p_0 / p_1)^{0.7143} - (p_0 / p_2)^{0.7143}]} \tag{1.11}$$

where$p_1 = (p - x)$ and $p_2 = (p + x)$. Here, p is the average working pressure in bar and x is given by $(a \times p)/100$ bar , where a is percentage of pulsation.

1. **Accumulator as an auxiliary power source:** The purpose of accumulator in this application is to store the oil delivered by the pump during a portion of the work cycle. The accumulator then releases the stored oil on demand to complete the cycle, thereby serving as a secondary power source.

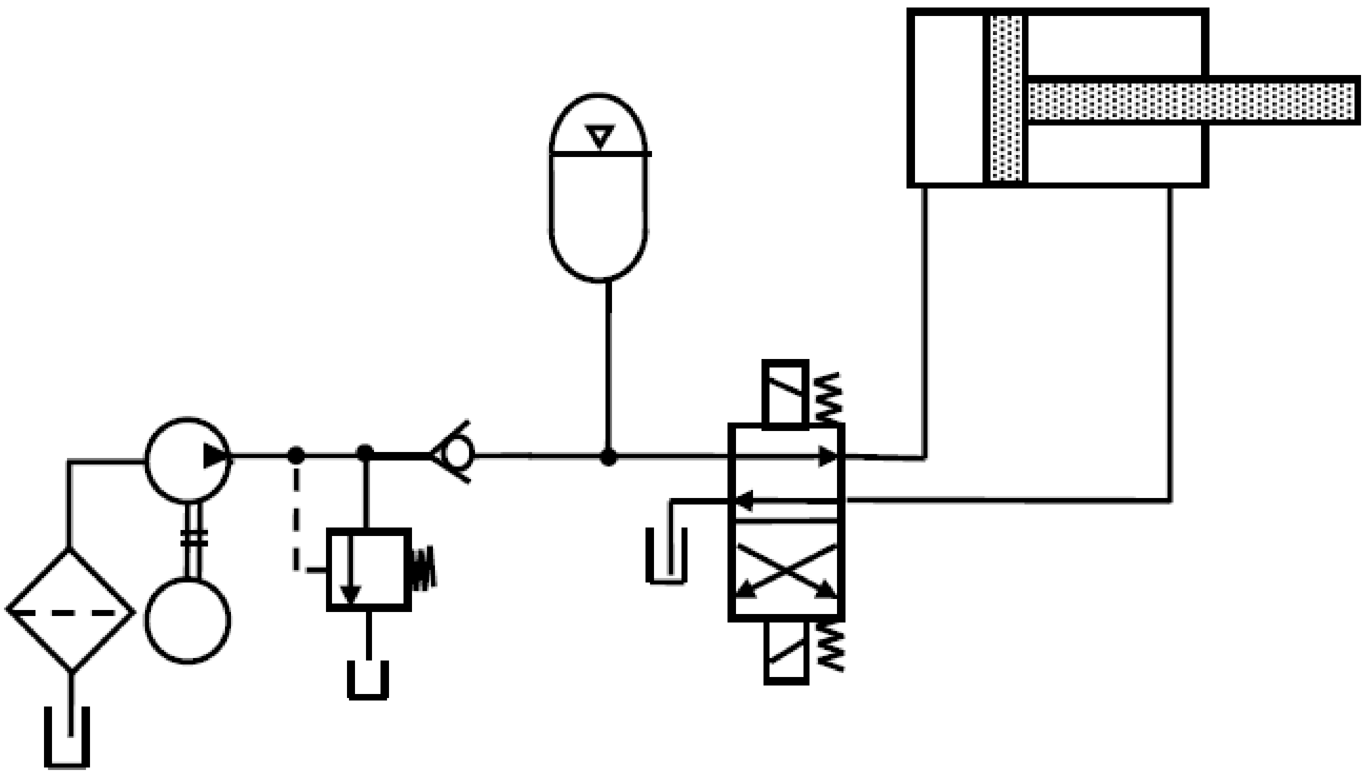


Figure 1.8 Accumulator as an auxiliary power source.

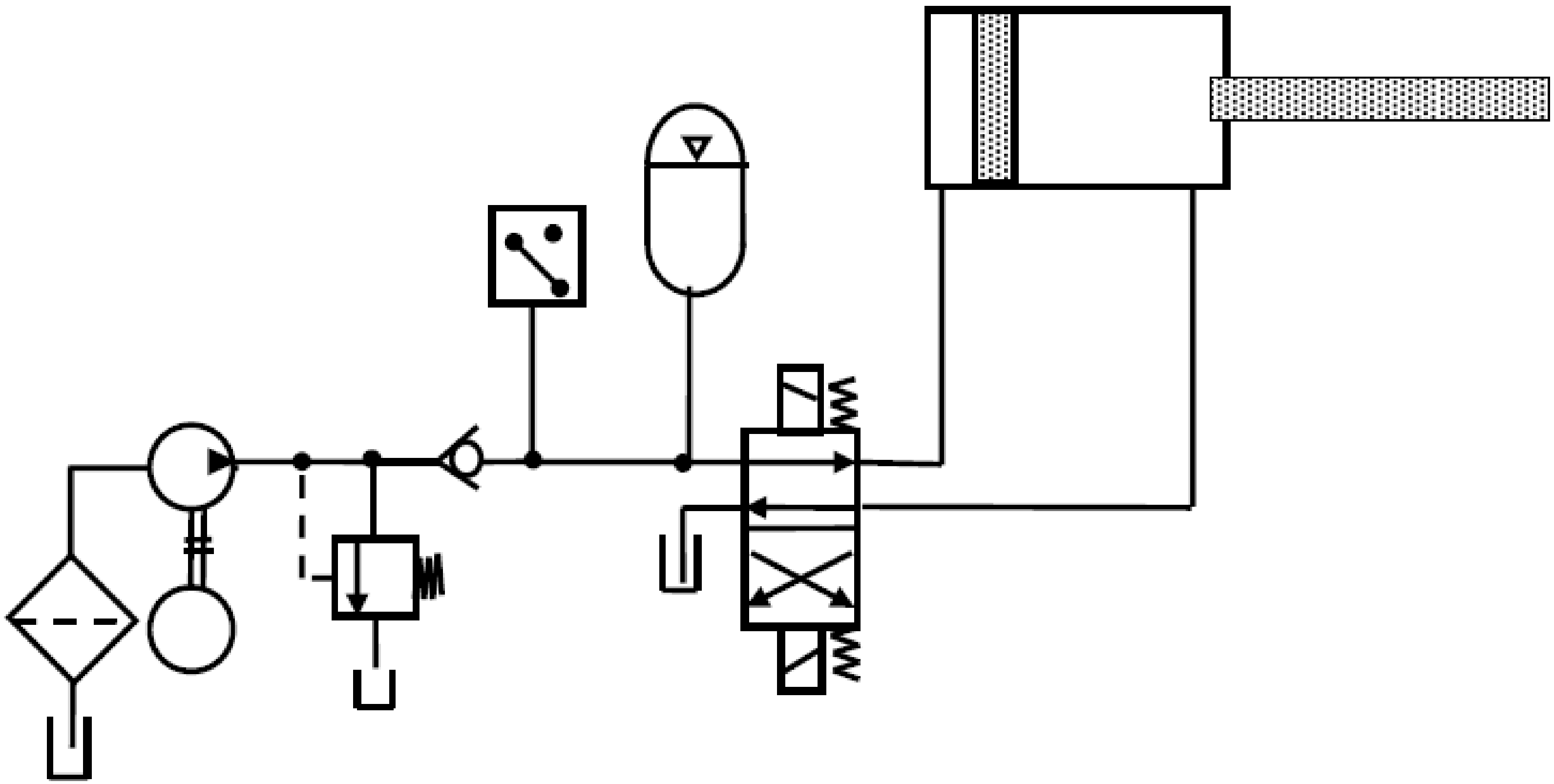


Figure 1.9 Accumulator as a leakage compensator.

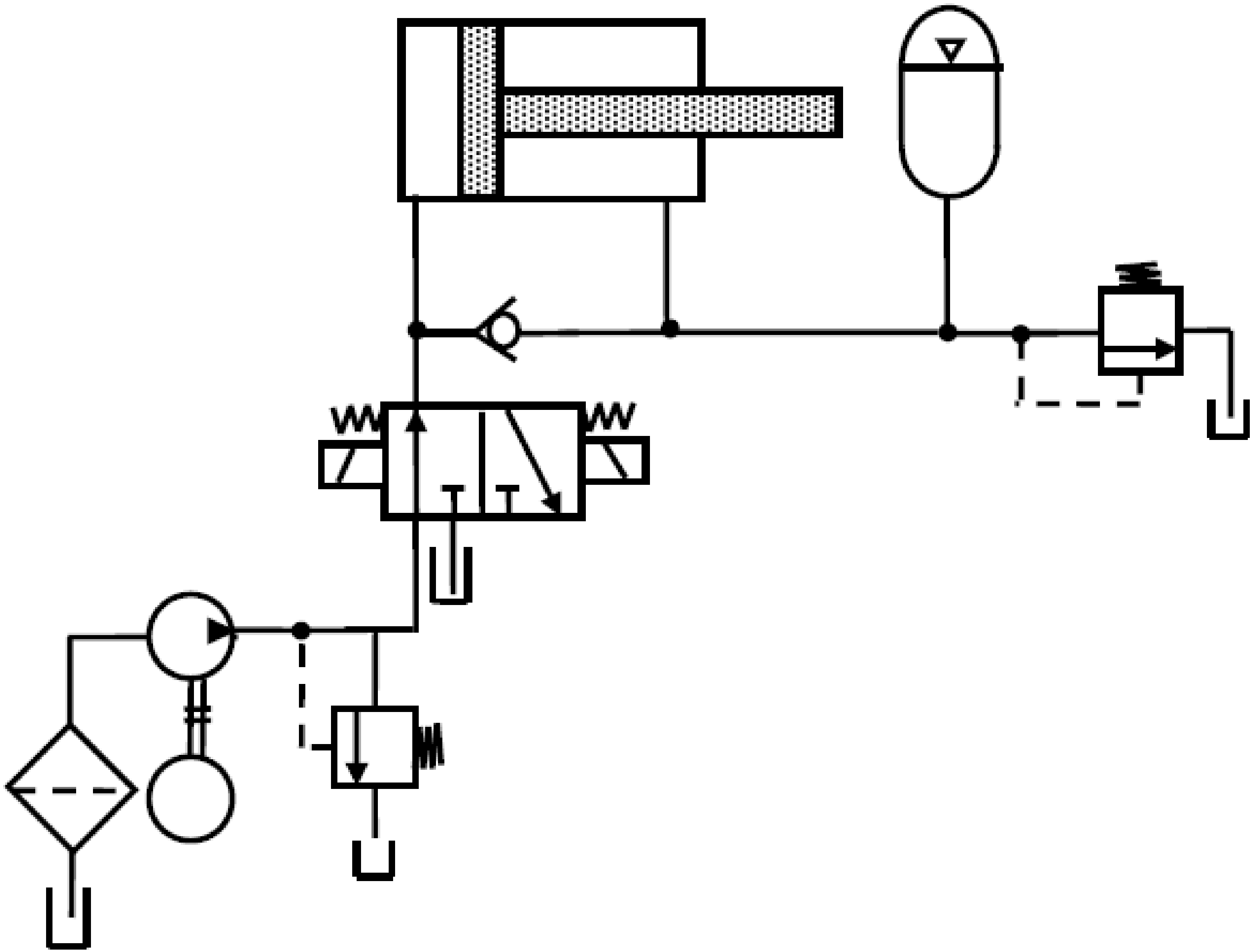


Figure 1.10 Accumulator as an emergency power source.

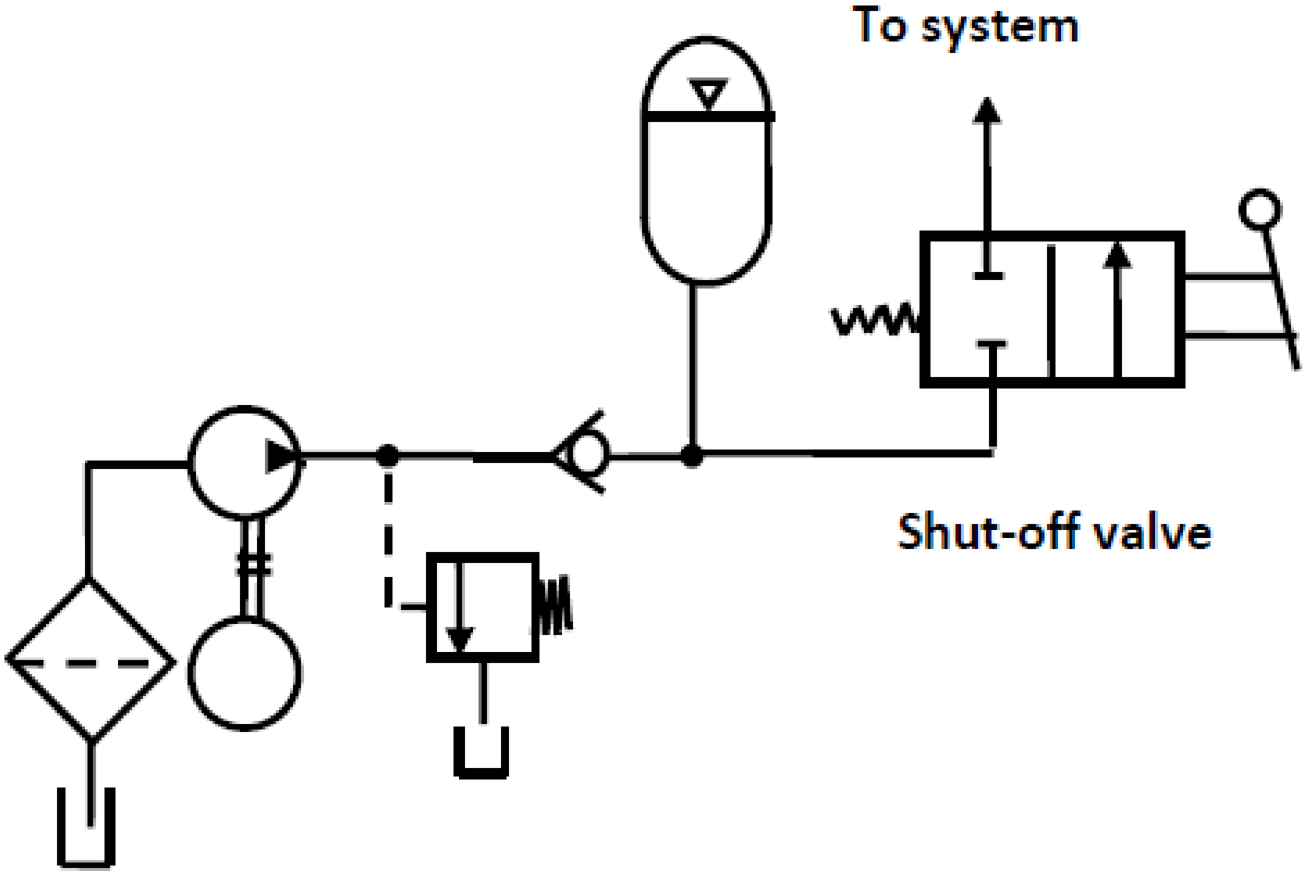


Figure 1.11 Accumulator as a hydraulic shock absorber.

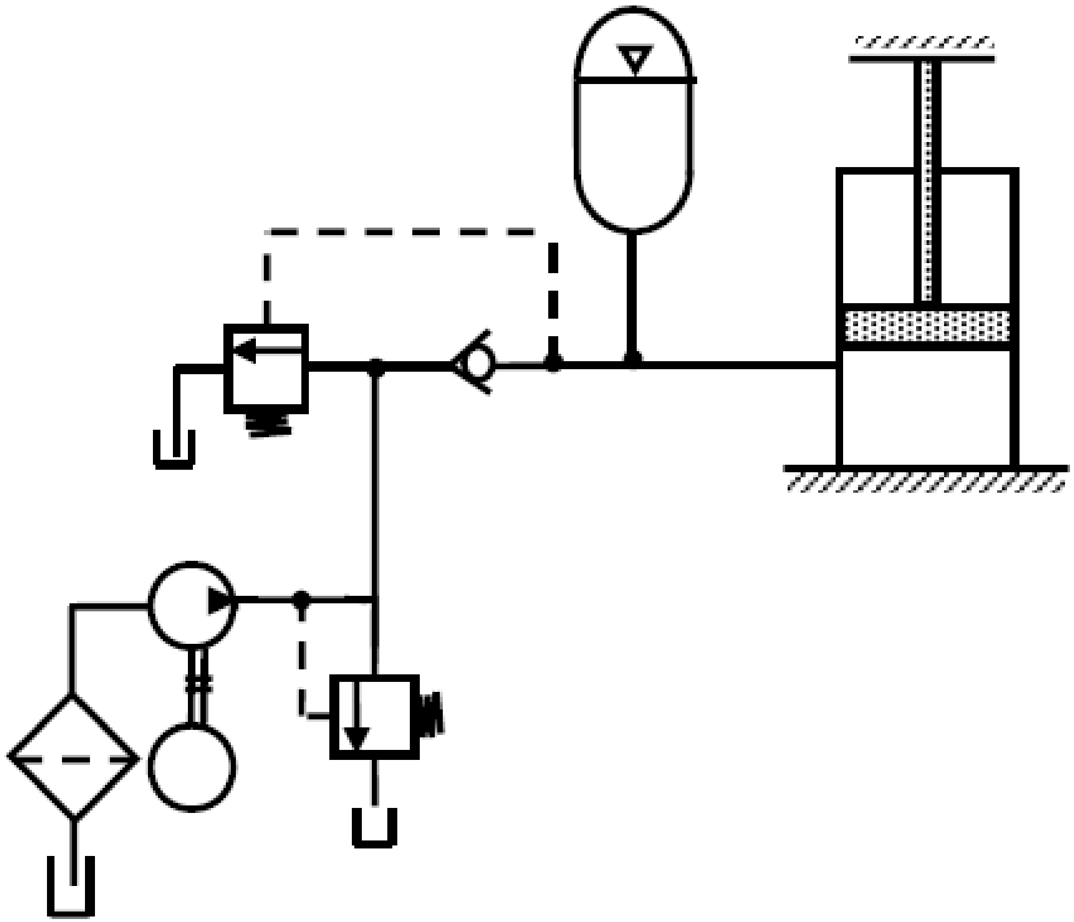


Figure 1.12 Accumulator as a thermal expansion compensator.

Isothermal conditions can be considered to exist if the accumulator is used as a volume compensator, leakage compensator and pressure compensator or as a lubrication compensator. In all other cases, such as energy accumulation, pulsation damping, emergency power source, dynamic pressure compensator, shock absorber, hydraulic spring, etc., expansion and compression process may be considered to take place under "adiabatic" conditions. Generally, the adiabatic condition is considered to exist if the compression or expansion period is less than 3 min.

1.2 Accumulator Selection

After ascertaining the type of accumulator that is appropriate for the purpose envisaged, what remains is determining the volume of the form of "high pressure fluid." Accumulators are manufactured to a variety of pressure ratings and the one chosen should be rated for a pressure more than the maximum system working pressure p_2.

However, the values of the following basic parameters should be established before proceeding further: Working pressures p_1 and p_2. The value of p_2 is found from the ratio $p_2 / p_0 \leq 4$. The maximum gas pre-charge pressure is found from the relationship $p_0 \leq 0.9 p_1$ or $p_0 \geq 0.25 p_1$. The gas pre-charge pressure must be as close as possible to the minimum working pressure p_1 to obtain maximum storage. Special values for p_0 are used in pulsation damping and shock absorber applications ($p_0 = 0.8 p_1$). Other parameters to be determined are the volume of fluid ΔV that needs to be stored ($\Delta V = 0.75 V_0$), the maximum required flow rate and the operating temperature.

Example 1.1

A hydraulic cylinder has to move a certain load through a certain distance in 1 s at a pressure of 140 bar. An accumulator is integrated into the circuit to provide peak power. The accumulator is charged for the first 20 s and discharged in 2 s. The delivery expected from the accumulator is 0.6 L in 2 s as the pressure falls from 250 to 140 bar. Calculate the accumulator volume. The operating temperature is +25–70°C. Also calculate the reduction in input power due to the accumulator.

Solution: This is a case of isothermal compression and adiabatic expansion. The equation considered here is

$$V_0 = \frac{\Delta V(p_2 / p_0)}{[(p_2 / p_1)^{0.7143} - 1]}$$

Because the maximum system pressure is above 200 bar, we consider a higher value of 1.6 for the adiabatic index n. Therefore, 1/1.6 = 0.625. Inserting this value into the above equation, we have $p_0 = 0.8 p_1 = 0.9 \times 140 = 126$ bar (gauge) =127 bar (absolute). So

$$V_0 = \frac{\Delta V(p_2 / p_0)}{[(p_2 / p_1)^{0.7143} - 1]}$$

$$= \frac{0.6(251/127)}{[(251/141)^{0.7143} - 1]}$$

$$= 2.74 \text{ L}$$

If we apply the correction for temperature change because the pre-charge pressure p_0 was based on the maximum temperature indicated, we have the new corrected volume

$$V_{0T} = 2.74\ (343/298) = 3.15 \text{ L}$$

A 3.5 L accumulator would effectively serve the purpose. The delivery from the accumulator is 0.6 L in 2 s, 0.3 L/s or 18 L/min.

In the absence of accumulator, the pump has to supply all of this delivery at a pressure of 140 bar. The HP required would be

$$(18 \times 140)/600 = 4.2 \text{ kW}$$

Here the efficiency of the pump is not considered. If the accumulator is included in the circuit, the pump has to deliver 0.6/20 = 0.03 L/s or 1.8 L/min sufficient to charge the accumulator to a pressure of 250 kgf/cm^2 within the time interval of 20 s. Here again the flow required to retract the cylinder is not considered.

The power requirement in this case would be $(1.8 \times 250)/600 = 0.75$ kW.

The power saved is $(4.2 - 0.75) = 3.45$ kW.

Example 1.2

A hydraulic molding press is kept closed at a maximum system pressure of 200 kgf/cm^2 for a duration of 60 min during the curing period. The maximum leakage permitted during this period is 2 cm^3/minute and minimum fall in pressure permitted is 198 kgf/cm^2. Calculate the accumulator volume.

Solution: This is an application where the accumulator is used as a leakage compensator under isothermal conditions. Therefore, the equation used for this is

$$V_0 = \frac{\Delta V}{[p_0 / p_1 - p_0 / p_2]}$$

Here $\Delta V = (2\times60)/1000 = 0.12$ L, $p_0 = (0.9\times198) = 178$ kgf/cm^2.Inserting these values in the above equation, we have

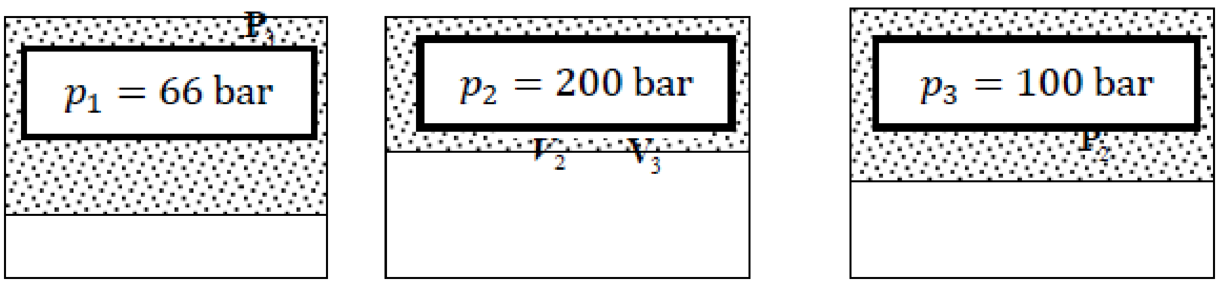
$$V_0 = \frac{0.12}{[179 / 199 - 179 / 201]} = 13.3\text{L}$$

A standard 15 L accumulator would meet the requirement.

Example 1.7

What size of accumulator is necessary to supply 500 cc of fluid in a hydraulic system of maximum pressure of 200 bar that drops to 100 bar minimum? Assuming N_2 gas pre-charged of 66 bar, find adiabatic and isothermal solution.

Solution: Stages of pre-cl , charging and delivery are shown in Fig. 1.13.



(a) Isothermal condition:

$$V_3 - V_2 = \text{volume of oil that can be delivered} = 0.005\text{m}^3$$

$$V_3 = V_2 + 0.005$$

Using $p_2V_2 = \quad p_3V_3$ we get

$$V_2 = \frac{p_3}{p_2} \times V_3 = \left[\frac{100+1.013}{200+1.013}\right] \times [V_2 + 0.005]$$

$$= 0.5025\ V_2 + 0.002512$$

$$= 0.00505\ \text{m}^3$$

Using $p_1V_1 = p_2V_2$ we get

$$V_1 = \frac{p_2}{p_1} \times V_2 = \left[\frac{200+1.013}{60+1.013}\right] \times 0.00505 = 0.01663\ \text{m}^3$$

(b) Adiabatic condition:

$$p_2V_2^{1.4} = p_3V_3^{1.4}$$

$$\Rightarrow V_2 = V_3 \times \left(\frac{p_1}{p_2}\right)^{1/1.4}$$

$$= (V_2 + 0.005)\left(\frac{101.013}{201.013}\right)^{0.714}$$

$$= 0.6118V_2 + 0.003059$$

$$= 0.00788 \text{ m}^3$$

Using $p_1V_1^{1.4} = p_2V_2^{1.4}$ we get

$$V_1 = 0.00788\left(\frac{201.013}{67.013}\right)^{0.714} = 0.01726 \text{ m}^3$$

Example 1.12

A circuit has been designed to crush a car body into bale using a 152 mm diameter hydraulic cylinder. The hydraulic is to extend 2.54 m during a period of 10 s. The time between crushing strokes is 5 min. The following accumulator gas absolute pressures are given: p_1(gas pre-charge pressure) = 84 bar (abs), p_2(gas charge pressure when the pump is turned ON) = 210 bar (abs) = pressure relief value setting, p_3(minimum pressure required to actuate load) = 126 bar (abs).

(a) Calculate the required size of the accumulator.

(b) What are the pump hydraulic kW power and flow requirements with and without accumulator?

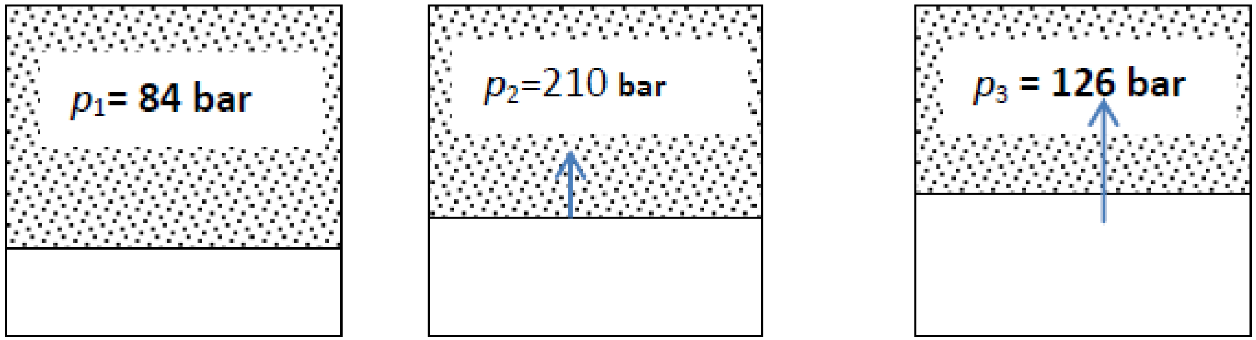


Figure 1.17

Let the pre-charging pressure be p_1 (85 bar). Gas is compressed by incoming oil from pressure 84 to 210 bar and accumulator is discharged till the pressure reaches 126 bar.

(a) Without accumulator: Let the compression and expansion of gas follow isothermal law:

$$p_1V_1 = p_2V_2 = p_3V_3$$

Here V_c is the volume of hydraulic cylinder. It can accommodate $(V_3 - V_2)$ amount of oil

$$V_c = (V_3 - V_2)$$

$$\Rightarrow p_3V_3 = p_2V_2$$

$$\Rightarrow V_3 = \frac{p_2V_2}{p_2} = \frac{210 \times V_2}{126} = 1.67V_2 \quad (1.13)$$

$$\Rightarrow V_c = \frac{\pi}{4}d^2I = \frac{\pi}{4}(0.152)^2 \times 2.54 = 0.0461\,\text{m}^3 = (V_3 - V_2) \quad (1.14)$$

Using Eq. (1.13) in Eq. (1.14) and solving, we get

$$V_2 = 0.0688\ \text{m}^3$$

$$V_3 = 0.155\ \text{m}^3$$

$$V_1 = \frac{p_2V_2}{p_1} = \frac{210 \times 0.0688}{840} = 0.172\,\text{m}^3 = 172\,\text{L}$$

(b) With accumulator: The pump charges accumulator in every 2.5 min. In other words, two times in five minutes.

Flow supplied by the pump

$$Q_p = \frac{2(V_3 - V_2)}{30} = \frac{2(46.1)}{300} = 0.307\,\text{LPS}$$

Neglecting all losses, power supplied to the pumpis

$$p_{\text{pump}} = p_2 \times Q_{\text{pump}}$$

$$= \frac{(210\times10^5)(0.307\times10^{-3})}{1000} = 6.45\text{ kW}$$

Without accumulator: The pump extends cylinder in 10 s. Flow supplied by the pump is

$$Q_p = \frac{46.1}{10} = 0.461\text{ LPS}$$

Neglecting all losses, power supplied to the pump is

$$p_{\text{pump}} = p_2 \times Q_{\text{pump}}$$

$$= \frac{(126\times10^5)(461\times10^{-5})}{1000} = 58.1\text{ kW}$$

It can be seen that flow and power requirement by the pump is more without accumulator.

ME 5451 – Hydraulics and Pneumatics

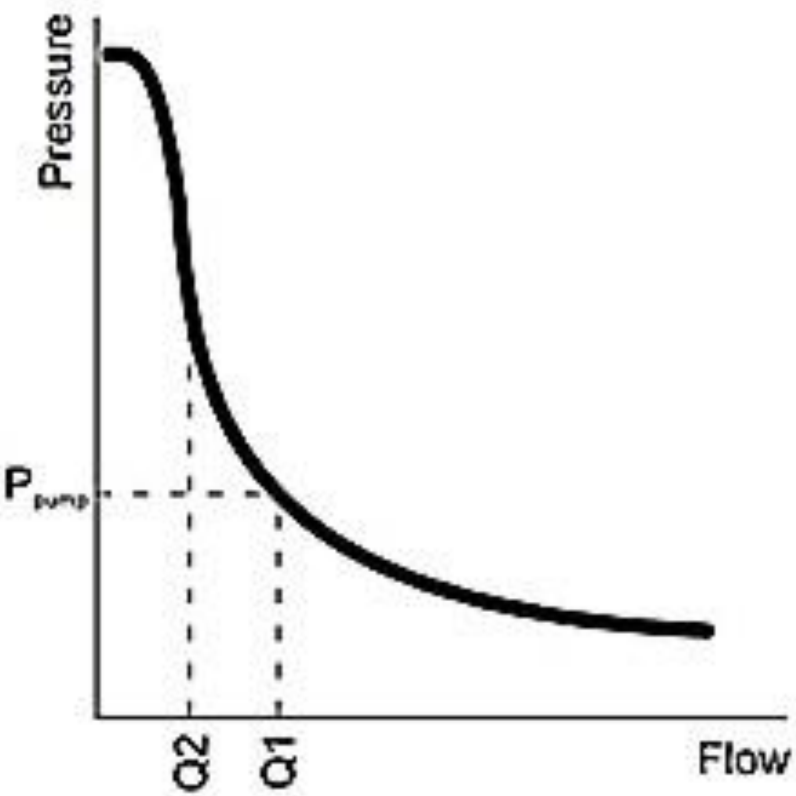
Lecture -17

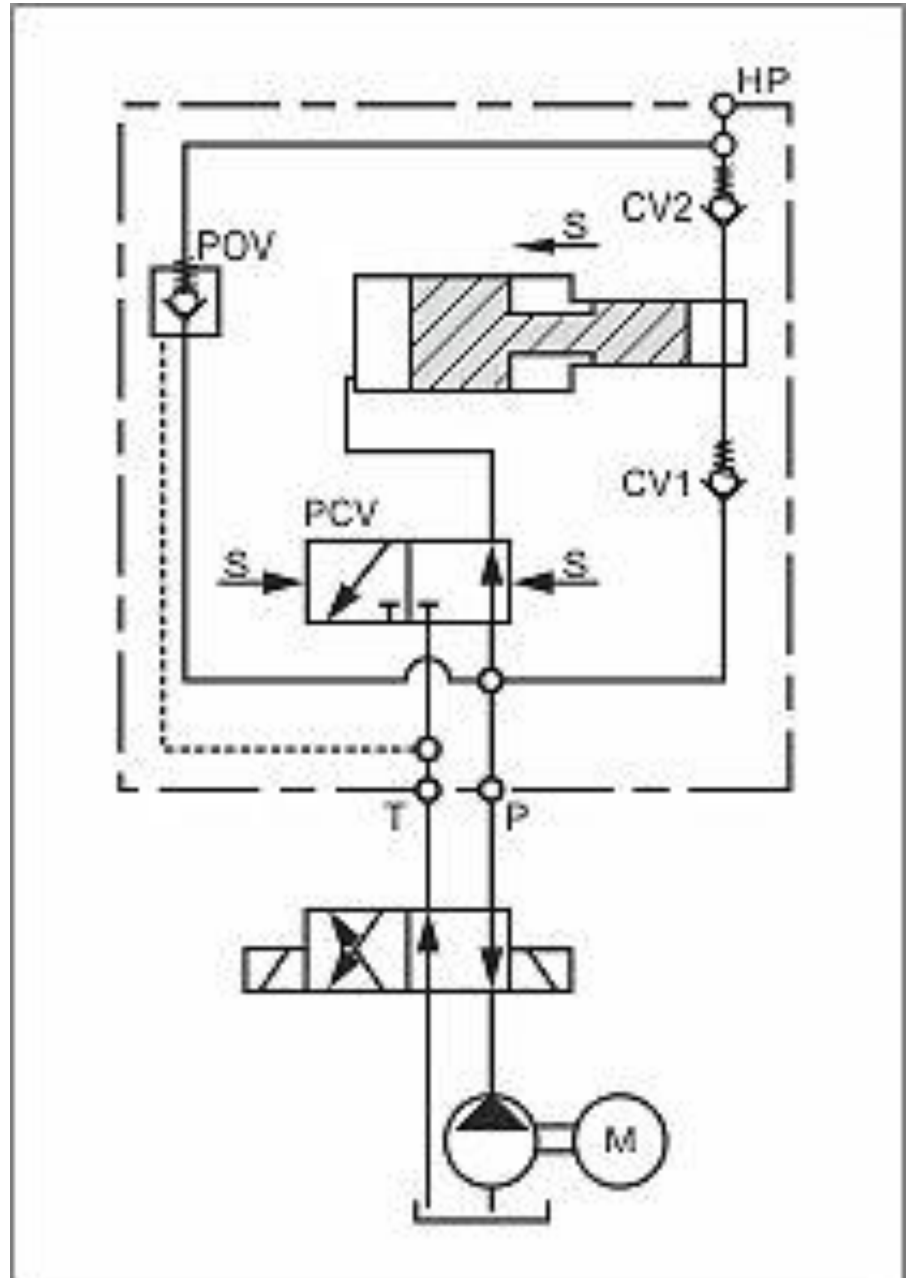
Date: 14-05-2021 Time slot: 08:30-10:10 a.m.

Contents

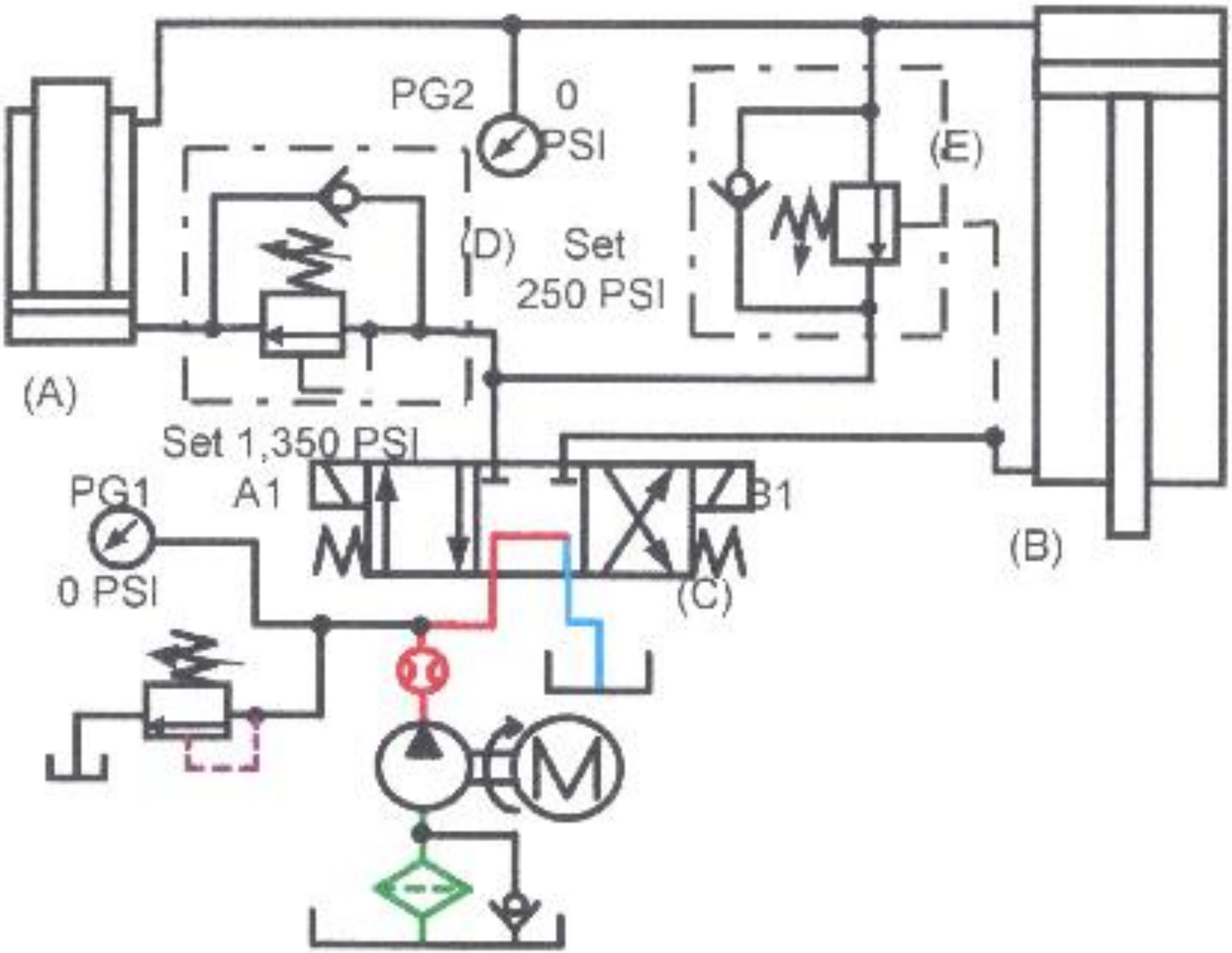
1. Intensifiers
2. Industrial hydraulic circuits

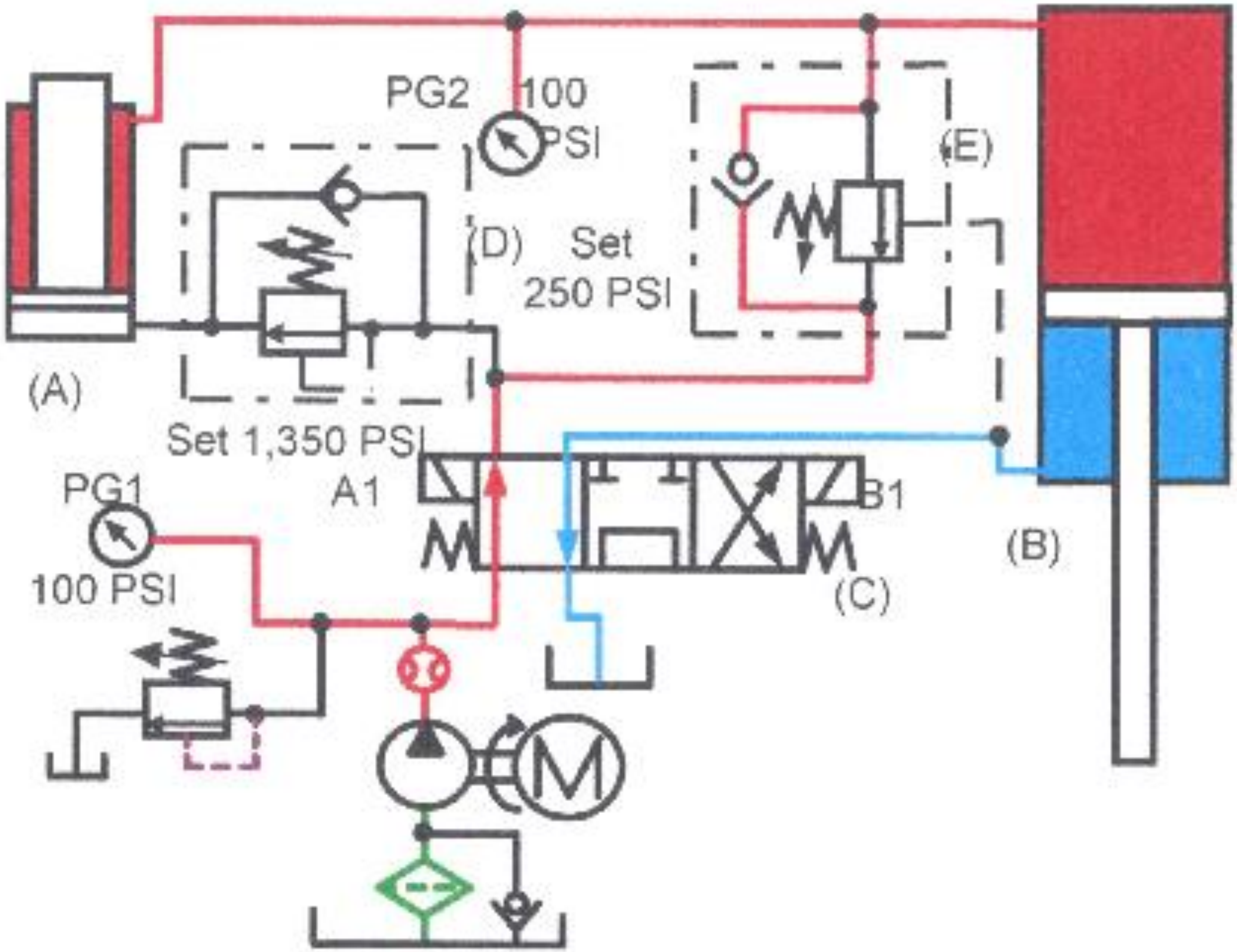
Course Instructor: Dr. A. Siddharthan

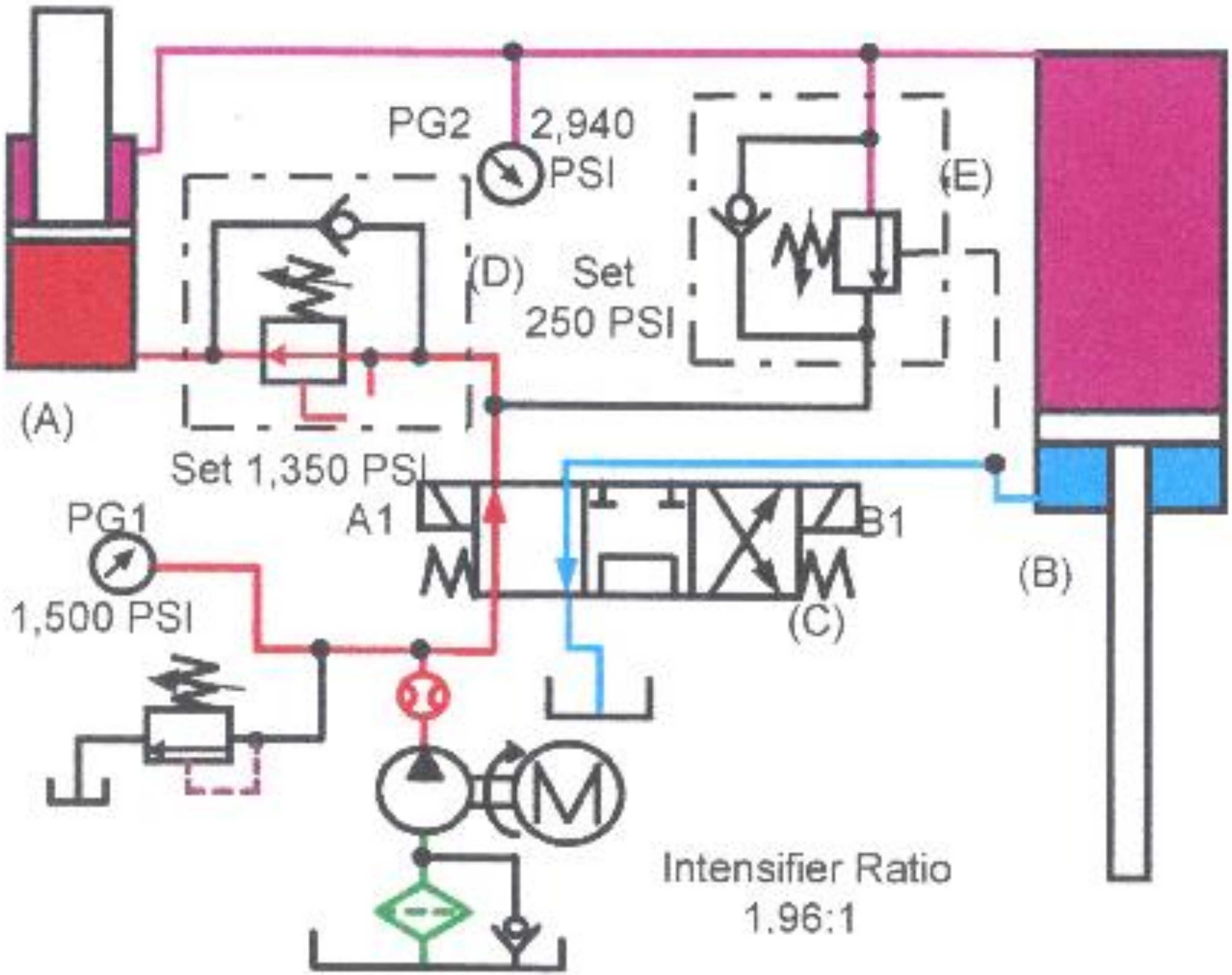

Pressure
P_{pump}
Q2
Q1
Flow


HP
CV2
S
POV
CV1
PCV
S
S
T
P
M

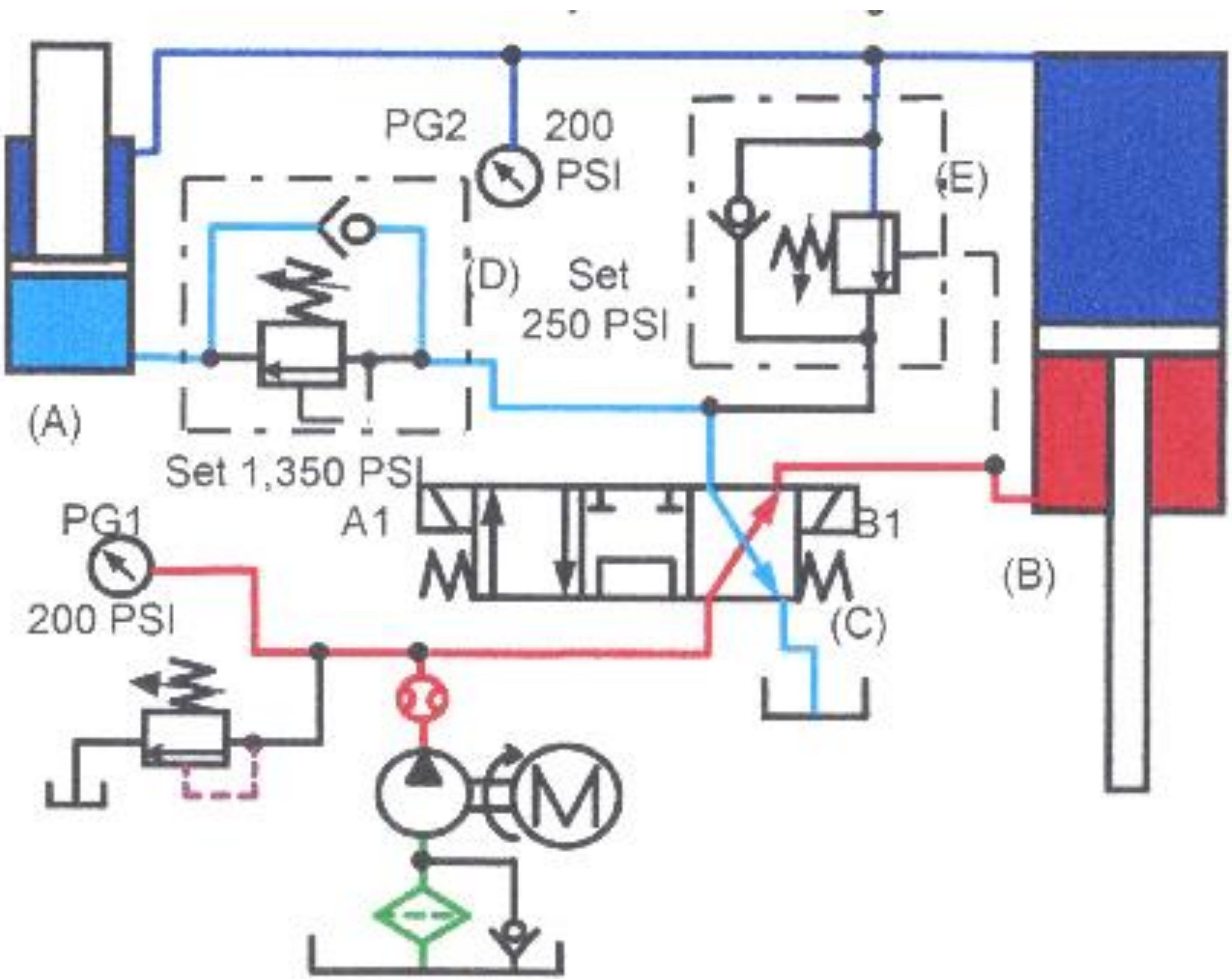
Ordering Code*	Ratio (i)	Inlet Flow (max.) (LPM)	Outlet Flow Q1 (LPM)	Outlet Flow Q2 (LPM)	Inlet Pressure (bar)	Outlet Pres (bar)
MP-T-1.5	1.5	8.0	0.8	0.3	200	300
MP-T-2.0	2.0	8.0	0.8	0.2	200	400
MP-T-3.4	3.4	15.0	2.2	0.5	200	680
MP-T-4.0	4.0	14.0	1.8	0.4	200	800
MP-T-5.0	5.0	14.0	1.4	0.3	160	800
MP-T-7.0	7.0	13.0	1.1	0.2	114	800
MP-T-9.0	9.0	13.0	0.7	0.1	89	800

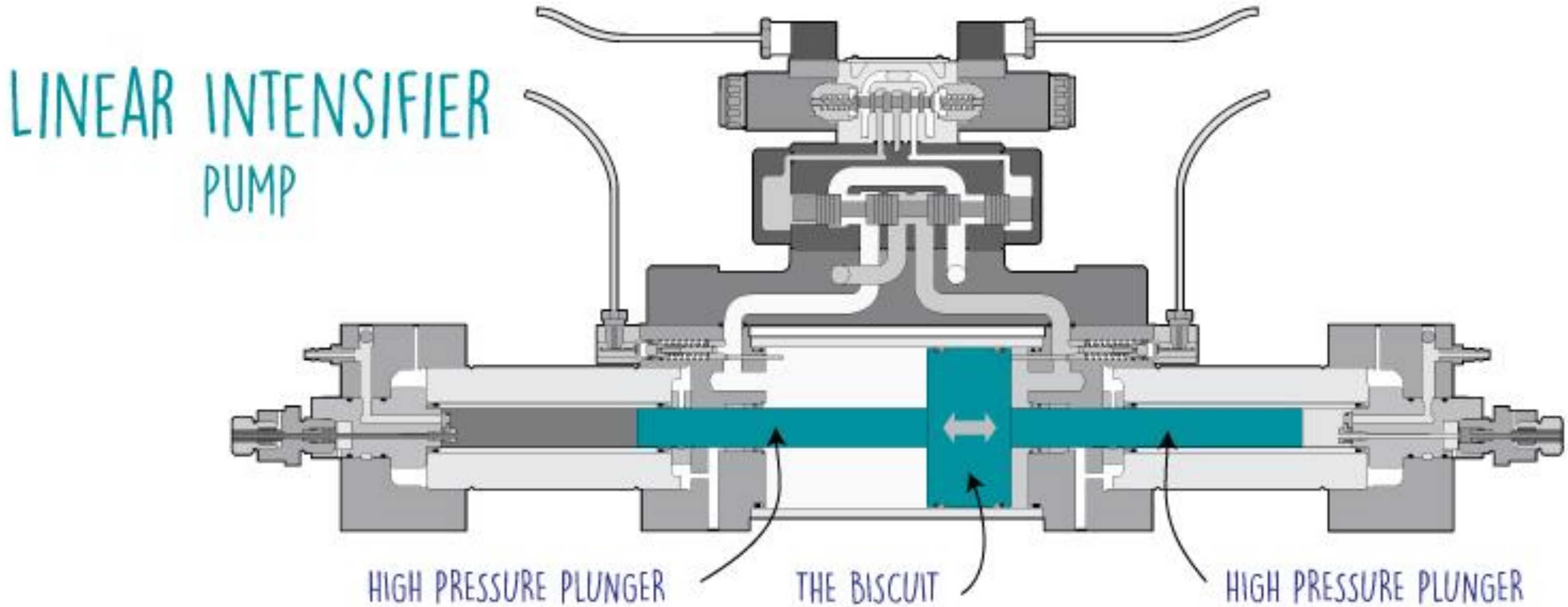

PG2
0
PSI
(E)
(D)
Set
250 PSI
(A)
Set 1,350 PSI
PG1
A1
B1
0 PSI
(C)
(B)
M

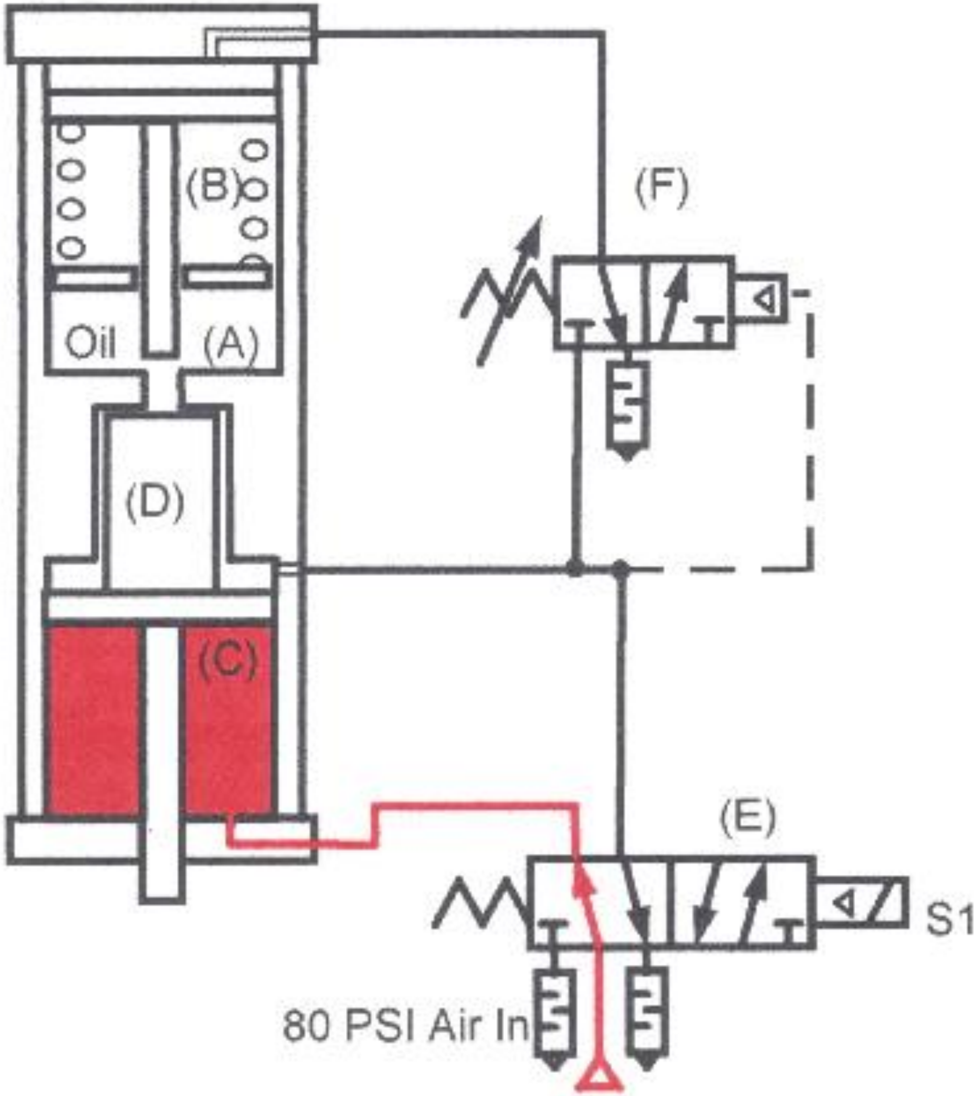

PG2
100
PSI
(E)
(D)
Set
250 PSI
(A)
Set 1,350 PSI
A1
B1
PG1
100 PSI
(C)
(B)
M



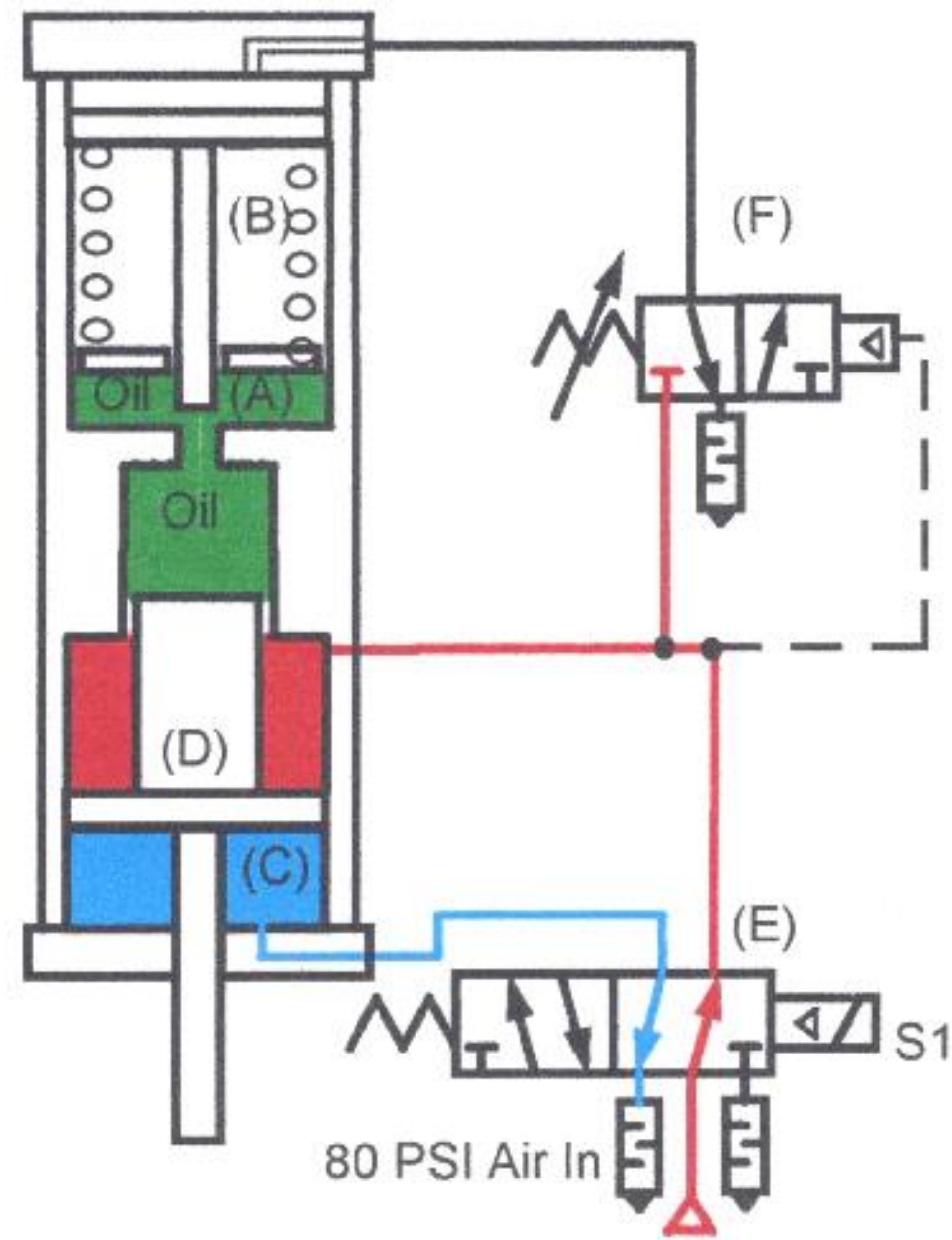
PG2
2,940
PSI
(E)
(D)
Set
250 PSI
(A)
Set 1,350 PSI
PG1
A1
B1
1,500 PSI
(B)
(C)
Intensifier Ratio
1.96:1


PG2
200
PSI
(E)
(D)
Set
250 PSI
(A)
Set 1,350 PSI
A1
B1
PG1
200 PSI
(B)
(C)
M

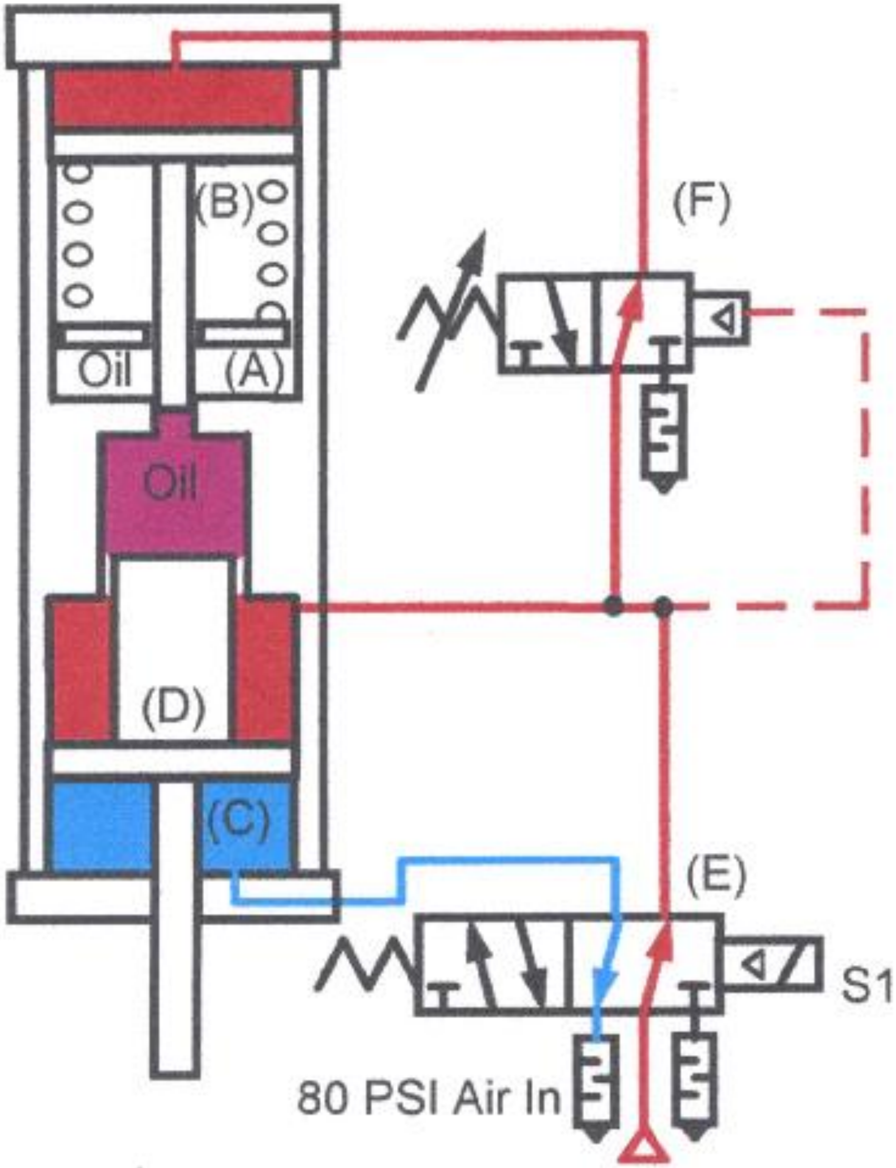

LINEAR INTENSIFIER
PUMP
HIGH PRESSURE PLUNGER
THE BISCUIT
HIGH PRESSURE PLUNGER



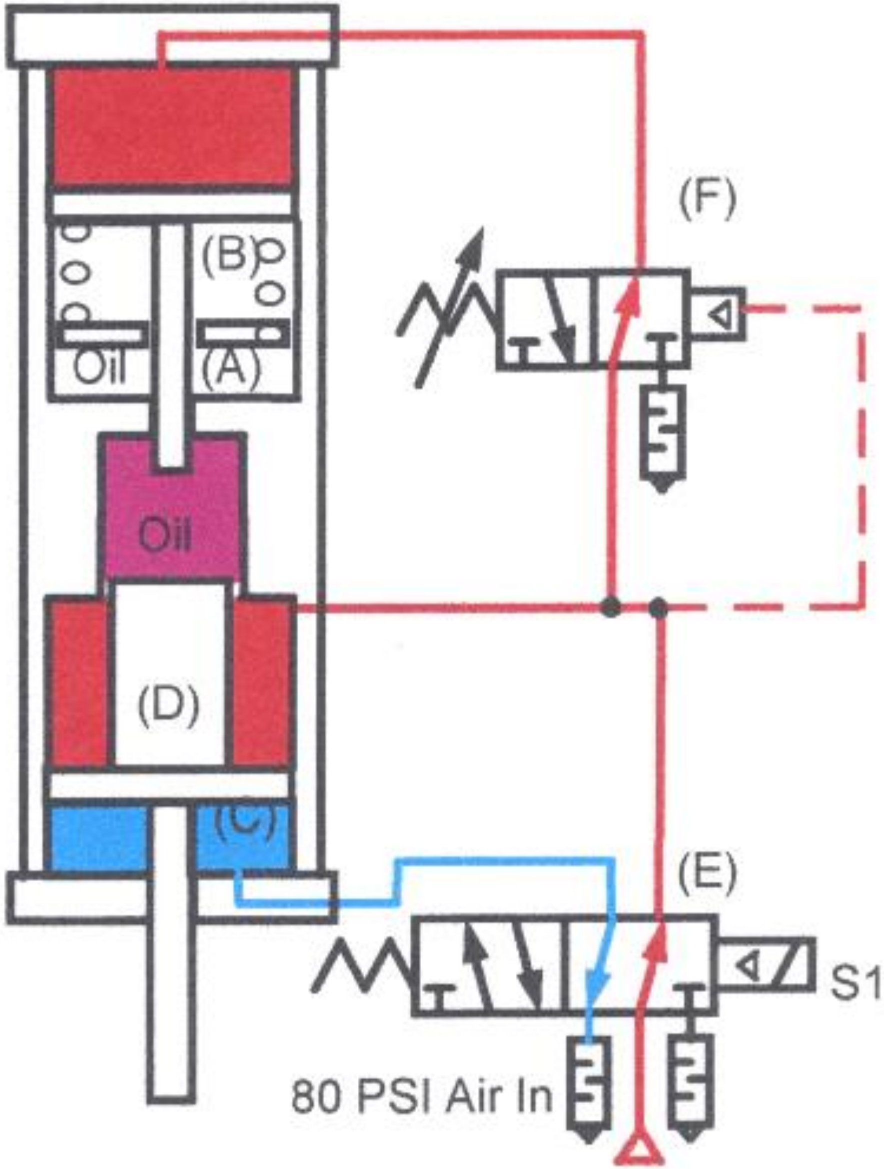
Special air-oil intensifier cylinder. System on and ready.



Special air-oil intensifier cylinder. Fast advance at low force.



Special air-oil intensifier cylinder. Starting high-pressure cycle.



Special air-oil intensifier cylinder extending at low speed with high force.

Regenerative Cylinder Circuit

The total flow rate Q_T entering the blank end of the cylinder is giv

$$Q_T = Q_p + Q_r$$

where Q_p is the pump flow rate and Q_r is the regenerative flow or flow from the rod end.

$$\text{Pump flow rate } (Q_p) = Q_T - Q_r$$

But the total flow rate acting on the blank rod end is given by

$$(Q_T) = A_p v_{ext}$$

Similarly, theflow rate from the rod end is given by

$$(Q_r) = (A_p - A_r) v_{ext}$$

So pump flow rate is

$$Q_p = A_p v_{ext} - (A_p - A_r) v_{ext}$$

$$\Rightarrow Q_p = A_r v_{ext}$$

The extending speed of the piston is given as

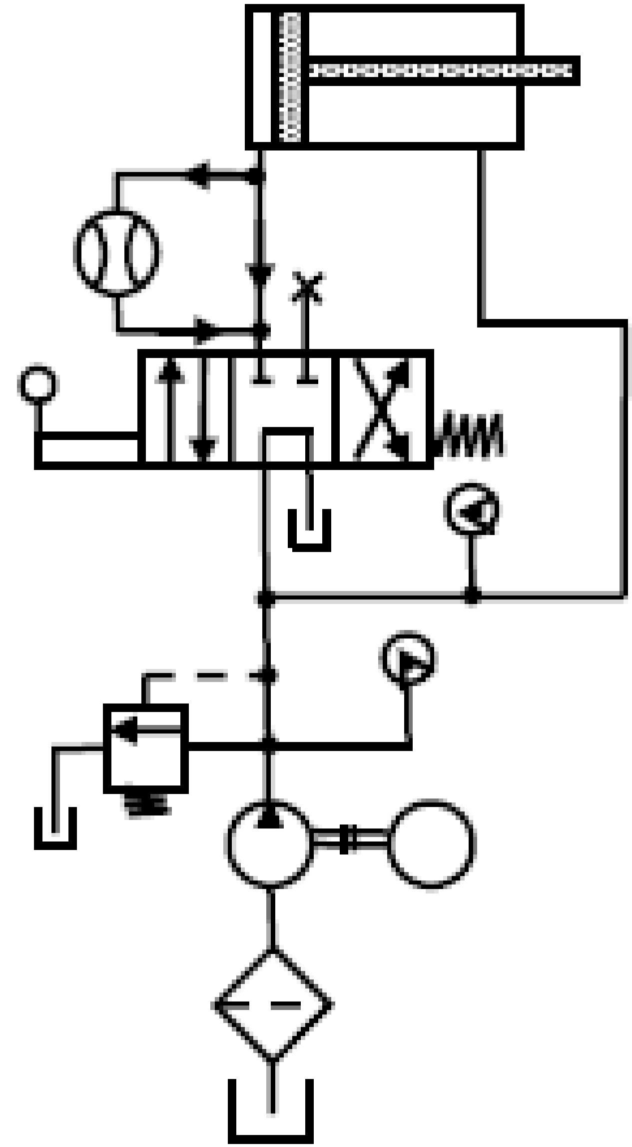
$$v_{ext} = \frac{Q_p}{A_r}$$

Thus, a small area provides a large extending speed. The extending speed can be greater than the retracting speed if the rod area is made smaller. The retraction speed is given by

$$v_{ret} = \frac{Q_p}{A_p - A_r}$$

The ratio of extending and retracting speed is given as

$$\frac{v_{ext}}{v_{ret}} = \frac{Q_p / A_r}{Q_p / (A_p - A_r)} = \frac{A_p - A_r}{A_r} = \frac{A_p}{A_r} - 1$$



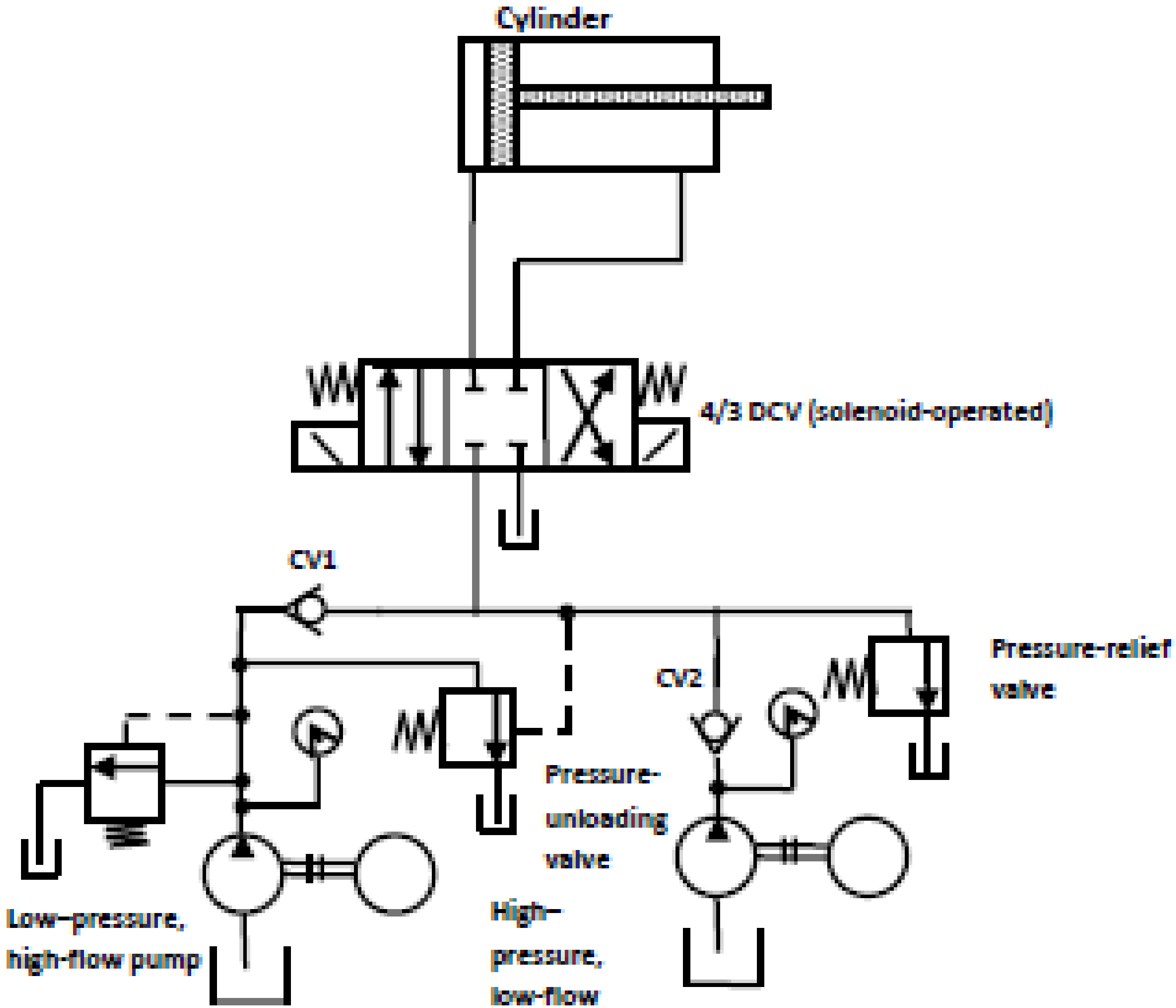


Figure 1.5 Double-pump circuit.

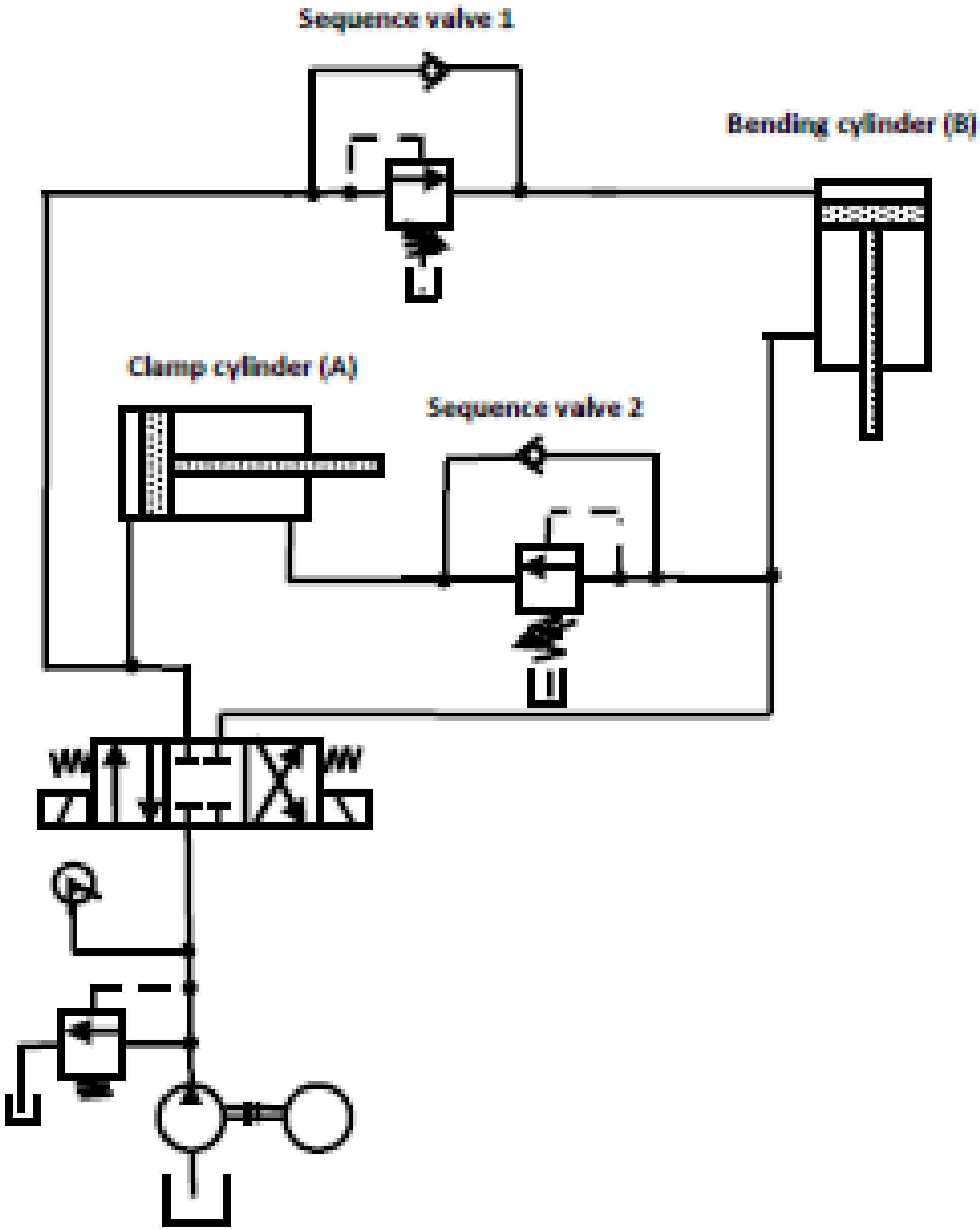


Figure 1.7 Sequencing circuit.

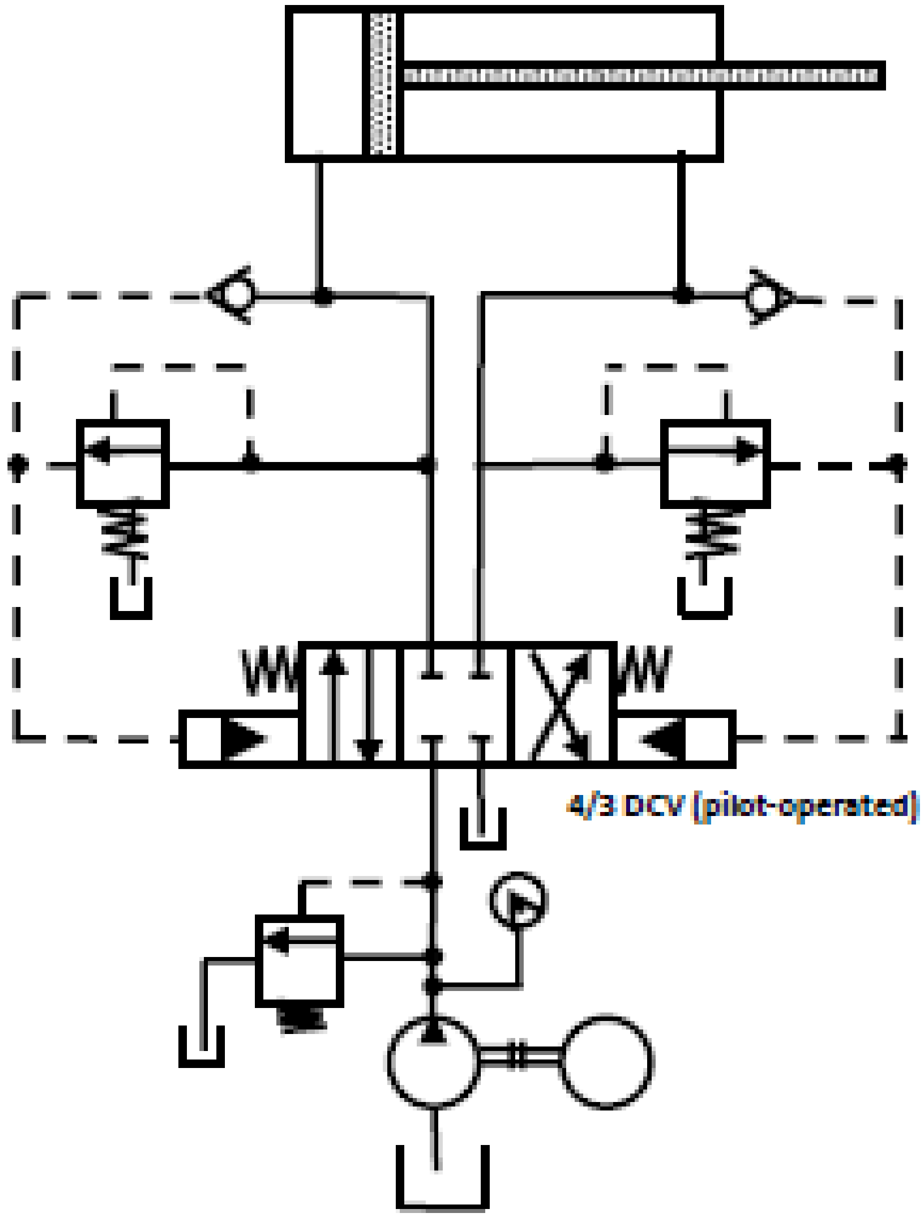
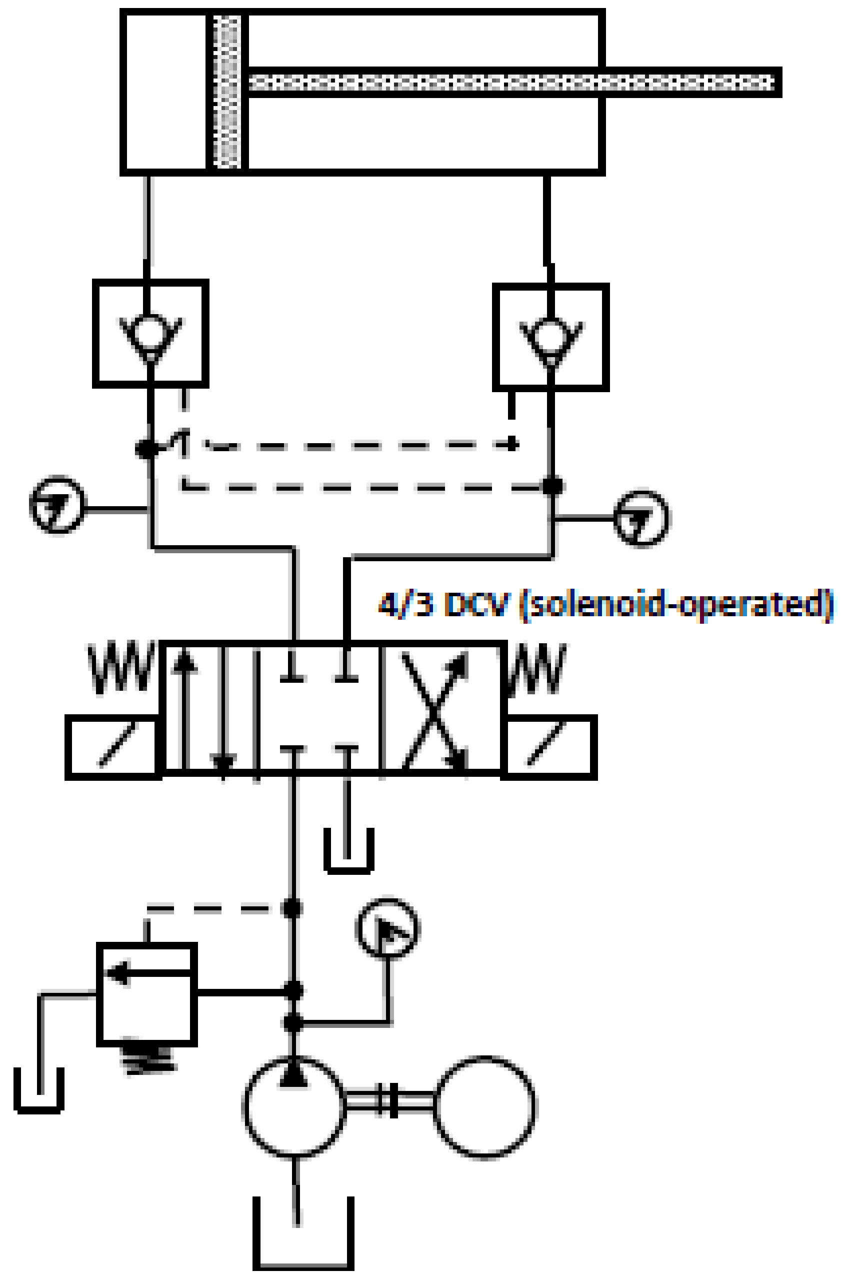


Figure 1.8Sequencing circuit.

Locked cylinders with pilot check valves.



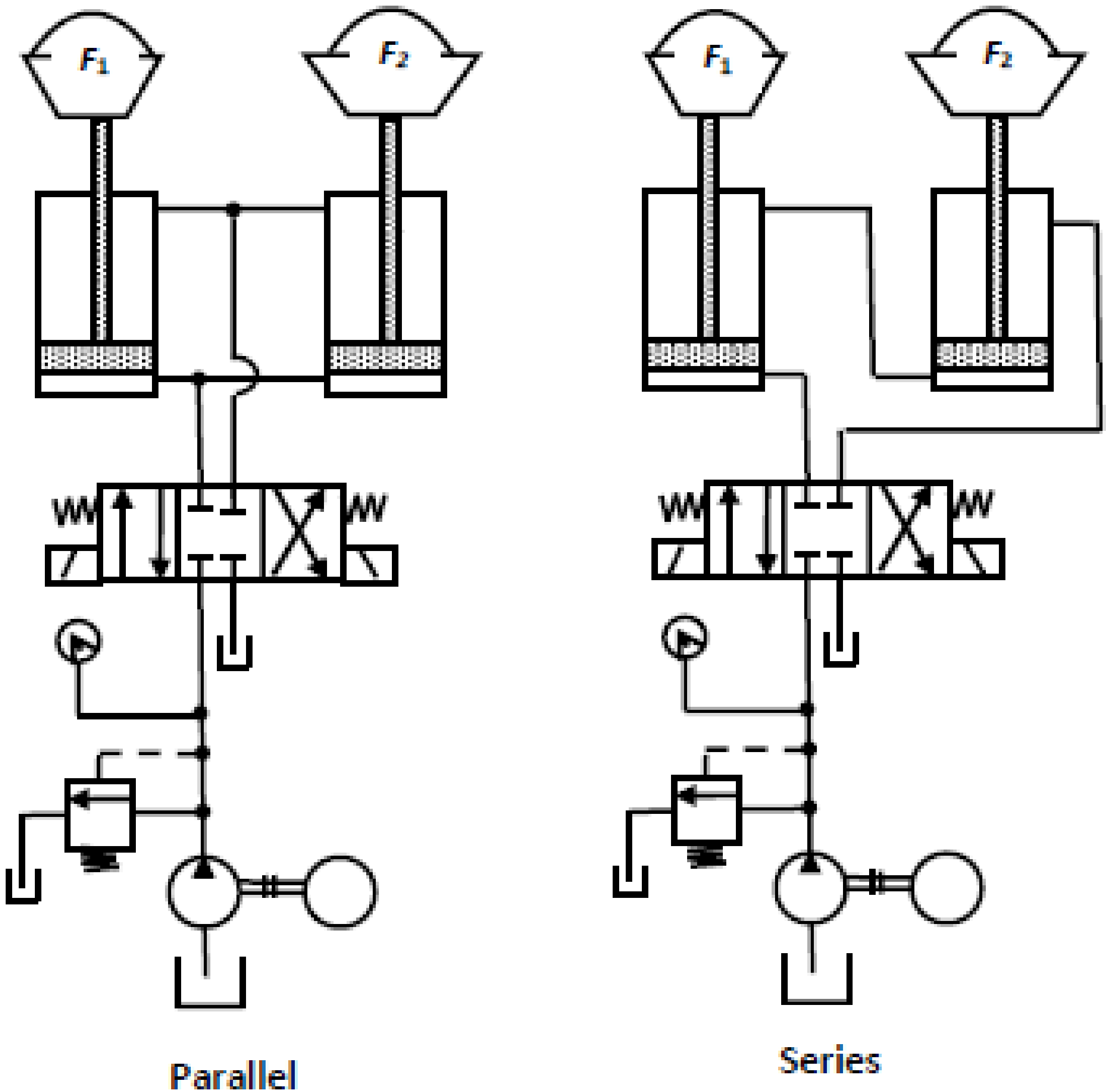


Figure 1.10Cylinders in parallel and series.

$$Q_{\text{out (cylinder1)}} = Q_{\text{in (cylinder 2)}}$$

we get

$$(A_{p1} - A_{r1})v_1 = A_{p2}v_2$$

For synchronization, $v_1 = v_2$. Therefore,

$$(A_{p1} - A_{r1}) = A_{p2} \tag{1.1}$$

The pump must deliver a pressure equal to that required for the piston of cylinder 1 by itself to overcome loads acting on both extending cylinders. We know that the pressure acting at the blank end of cylinder 2 is equal to the pressure acting at the rod end of cylinder 1.

Forces acting on cylinder 1 give

$$p_1A_{p1} - p_2(A_{p1} - A_{r1}) = F_1$$

Forces acting on cylinder 2 give

$$p_2A_{p2} - p_3(A_{p2} - A_{r2}) = F_2$$

UsingEq. (1.1)and noting that $p_3 = 0$ (it is connected to the tank), we have

$$p_1A_{p1} - p_2(A_{p2}) = F_1 \tag{1.2}$$

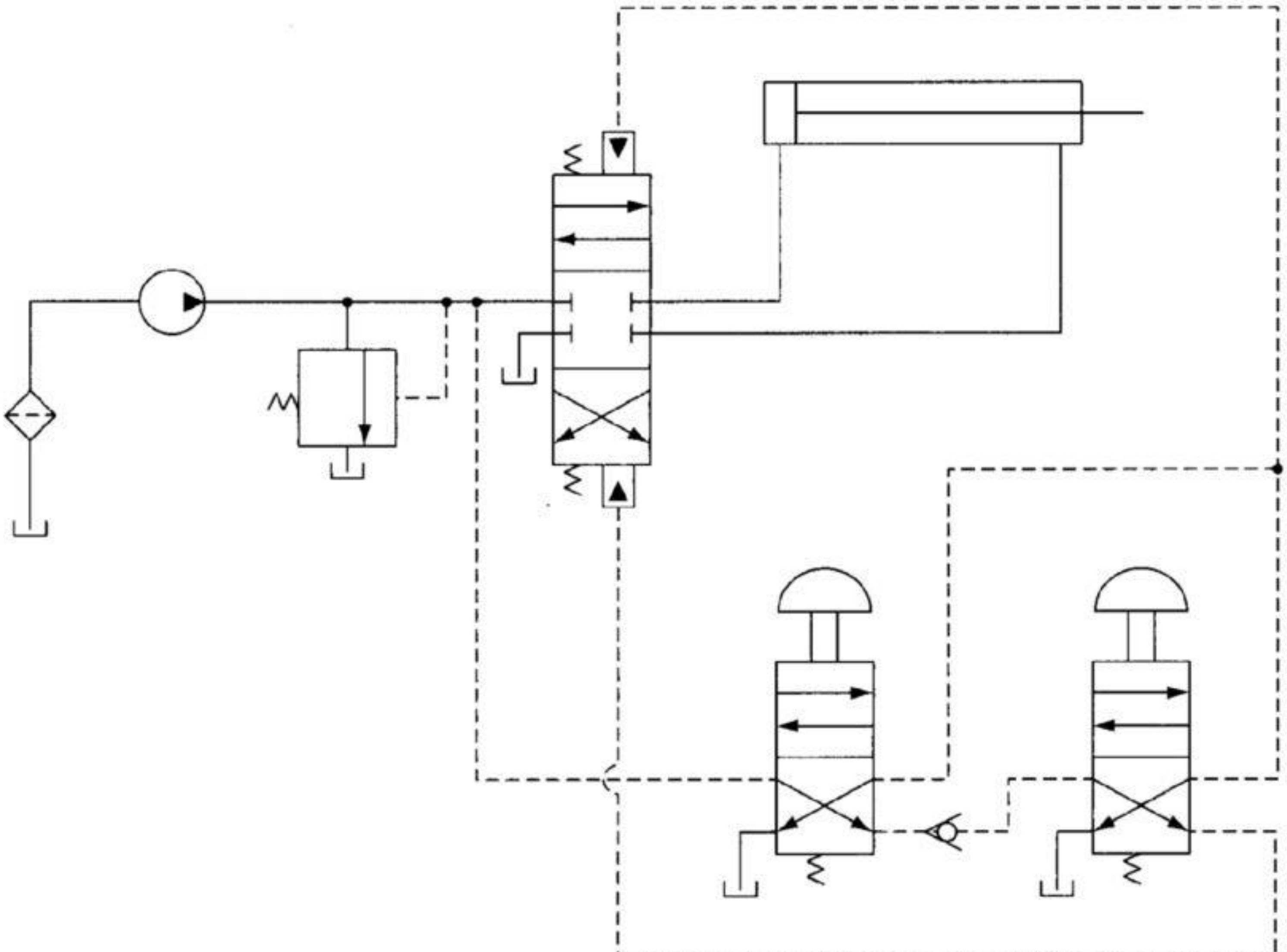
$$p_2(A_{p2}) - 0 = F_2 \tag{1.3}$$

Now, Eq. (1.2) + Eq. (1.3) gives

$$p_1A_{p1} = F_1 + F_2 \tag{1.4}$$

If Eqs. (1.1) and (1.4) are met in a hydraulic circuit, the cylinders hooked in series operate in synchronization.

Two-Handed Safety System



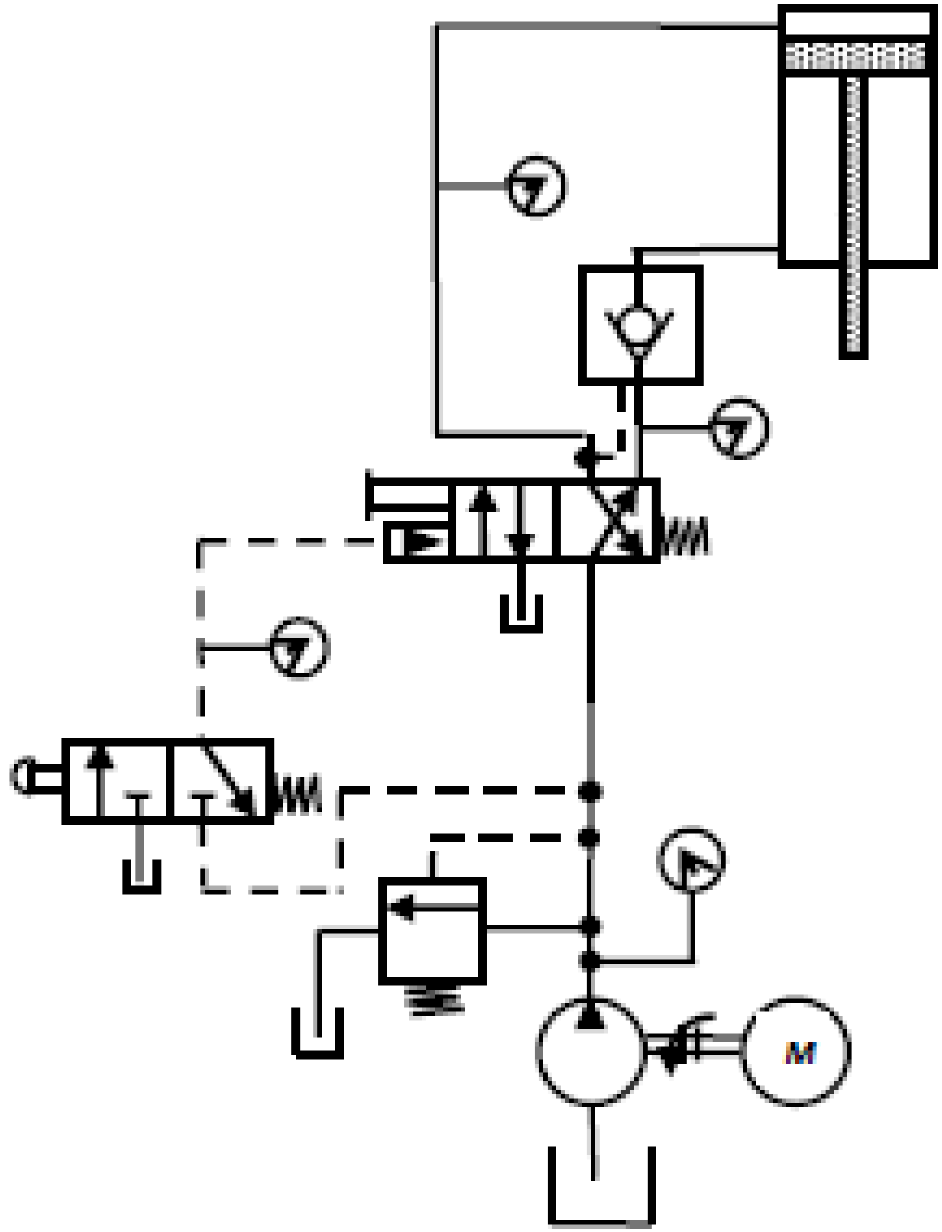


Figure 1.13 Fail-safe circuits – inadvertent cylinder extension.

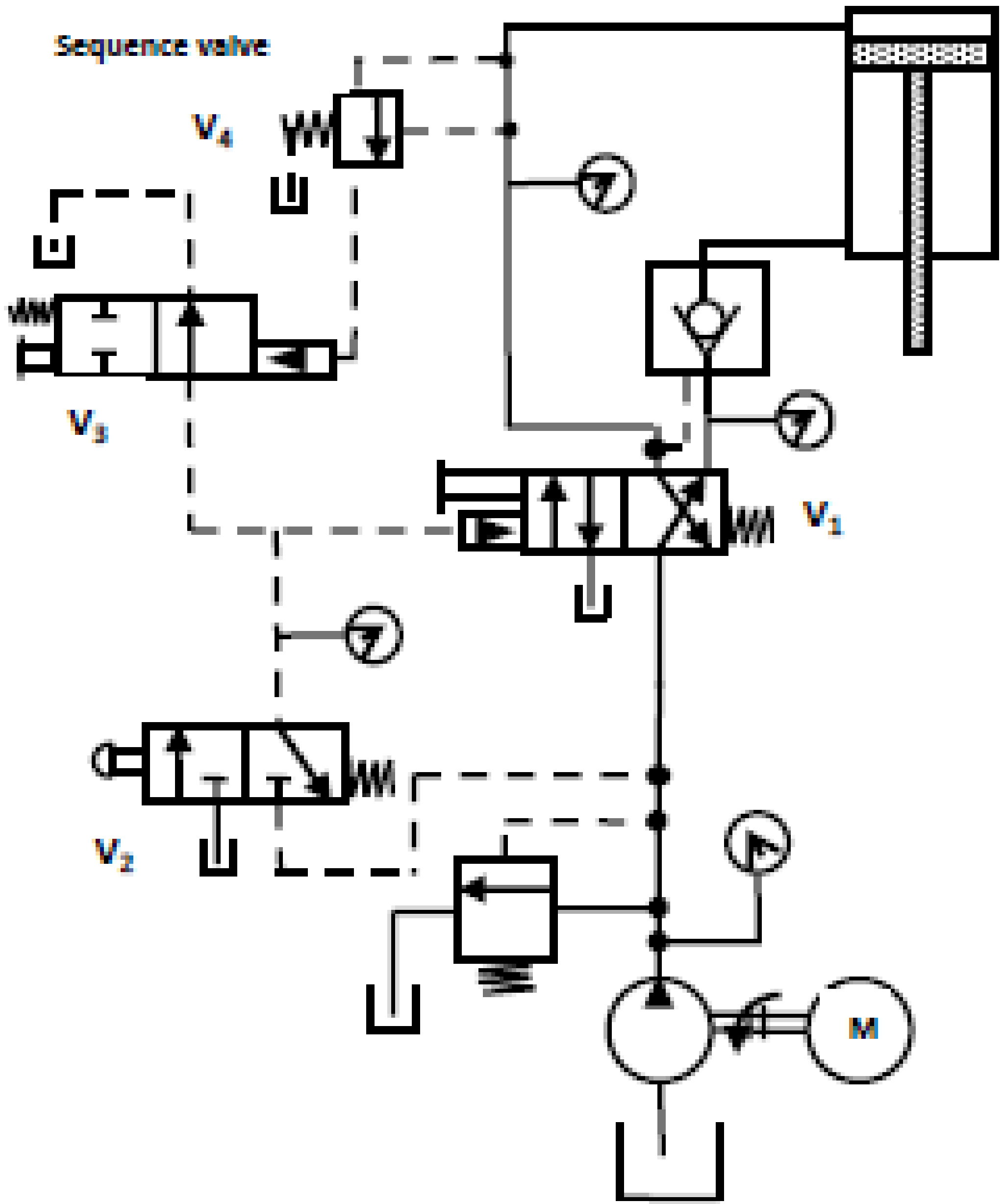


Figure 1.14 Fail-safe circuits –overload protection.

ME 5451 – Hydraulics and Pneumatics

Lecture -18

Date: 20-5-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Review of lecture 17
2. Industrial hydraulic circuits

Course Instructor: Dr. A. Siddharthan

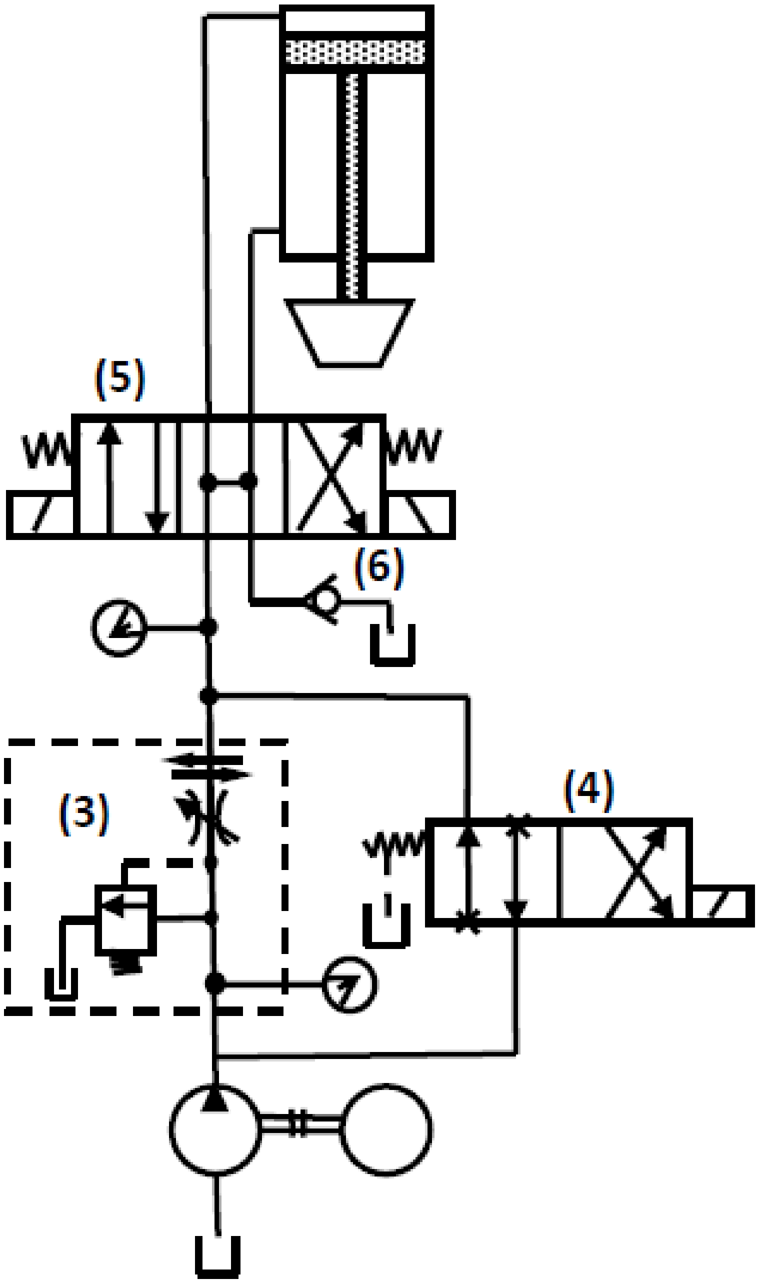


Figure 1.15 Rapid traverse and feed circuit.

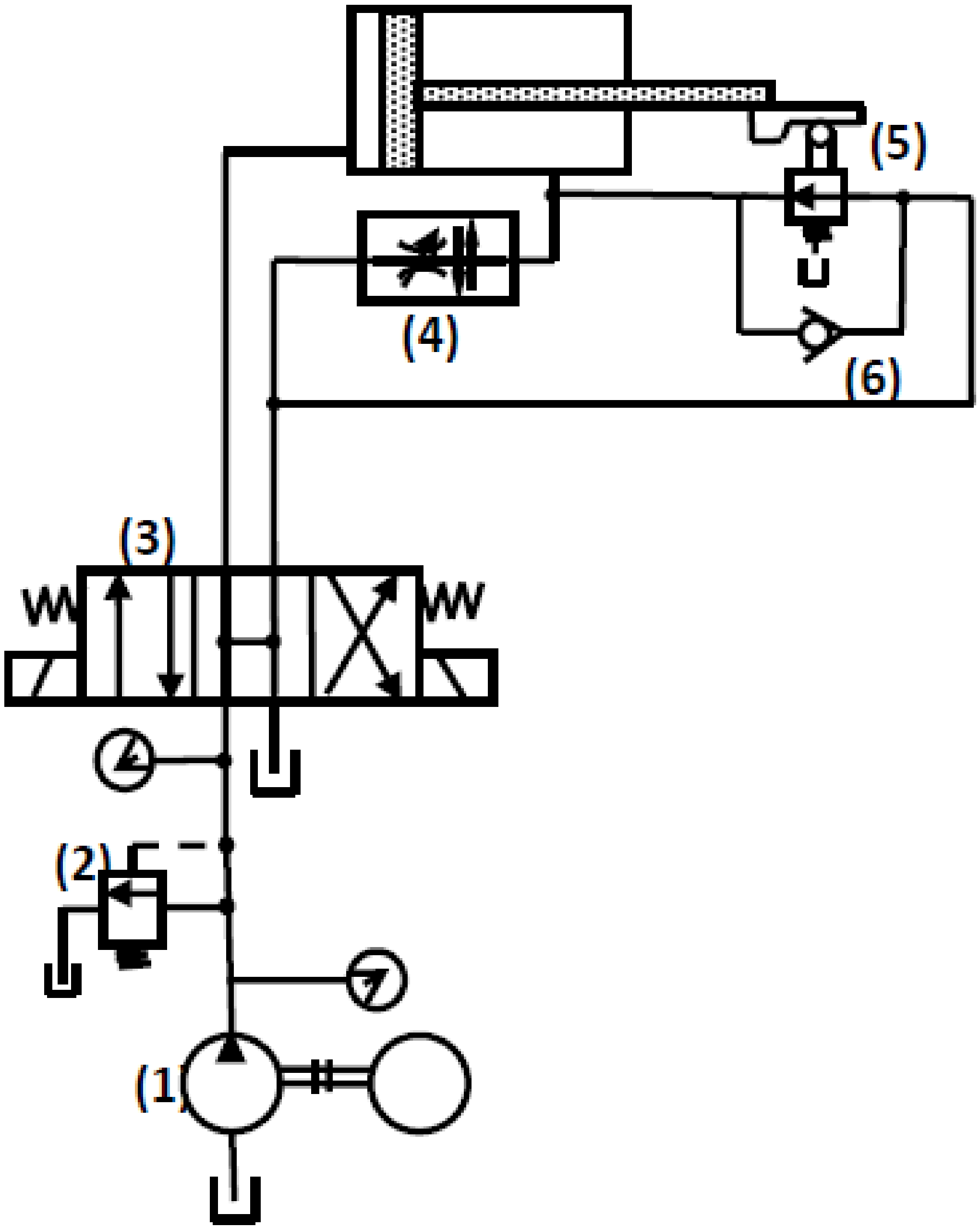


Figure 1.16 Rapid traverse and feed circuit – alternate circuit.

Example 1.7

For the hydraulic system is shown in Fig. 1.22

(a) What is the pump pressure for forward stroke if the cylinder loads are 22000 N each and cylinder 1 has the piston area of 65 cm^2 and zero back pressure?

(b) What is pump pressure for retraction stroke (loads pull to right), if the piston and rod areas of cylinder 2 equal to 50 cm^2 and 15 cm^2, respectively, and zero back pressure?

(c) Solve using a back pressure p_3 of 300 kPa instead of zero, the piston area and rod area of cylinder 2 equal 50 and 15 cm^2, respectively.

(a)Pressure acting during forward stroke is

$$p_1 = \frac{F_1 + F_2}{A_{p1}} = \frac{22000 + 22000}{65 \times 10^{-4}} = 6.77 \text{ MPa}$$

(b)For cylinder 2 we can write

$$p_3(A_{p2} - A_{r2}) - p_2 A_{p2} = F_2$$

For cylinder 1, force balance gives

$$p_2(A_{p1} - A_{r1}) = F_1$$

But $A_{p2} = A_{p1} - A_{r1}$. So we can write

$$p_2 A_{p2} = F_1$$

and rod side pressure of second cylinder is given by

$$p_3 = \frac{F_1 + F_2}{A_{p2} - A_{r2}} = \frac{22000 + 22000}{35 \times 10^{-4}} = 12570000 \text{ Pa} = 12.57 \text{ MPa}$$

(c)For cylinder 1, we have

$$p_1 A_{p1} - p_2 (A_{p1} - A_{r1}) = F_1$$

Similarly for cylinder 2, we have

$$p_2 A_{p2} - p_3 (A_{p2} - A_{r2}) = F_2$$

Adding both equations and noting that $A_{p2} = A_{p1} - A_{r1}$ yield

$$p_1 A_{p1} - p_3 (A_{p2} - A_{r2}) = F_1 + F_2$$

$$\Rightarrow p_1 = \left\{ \frac{F_1 + F_2 + p_3 (A_{p2} - A_{r2})}{A_{p1}} \right\}$$

$$\Rightarrow p_1 = \left\{ \frac{22000\ \text{N} + 22000\ \text{N} + 300000\ \text{N}/\text{m}^2 (50 - 15)\ \text{cm}^2 \times 10^{-4}\ \text{m}^2}{65\ \text{cm}^2 \times 10^{-4}\ \text{m}^2} \right\}$$

$$\Rightarrow p_1 = 6.93\ \text{MPa}$$

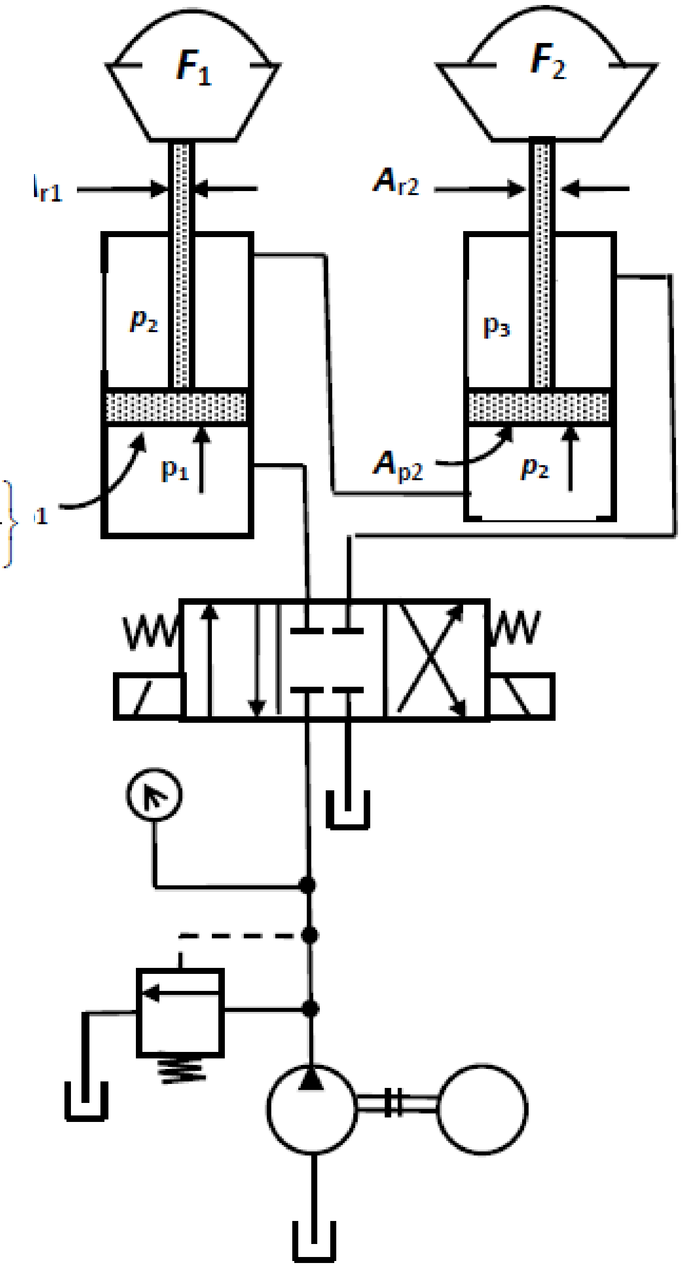
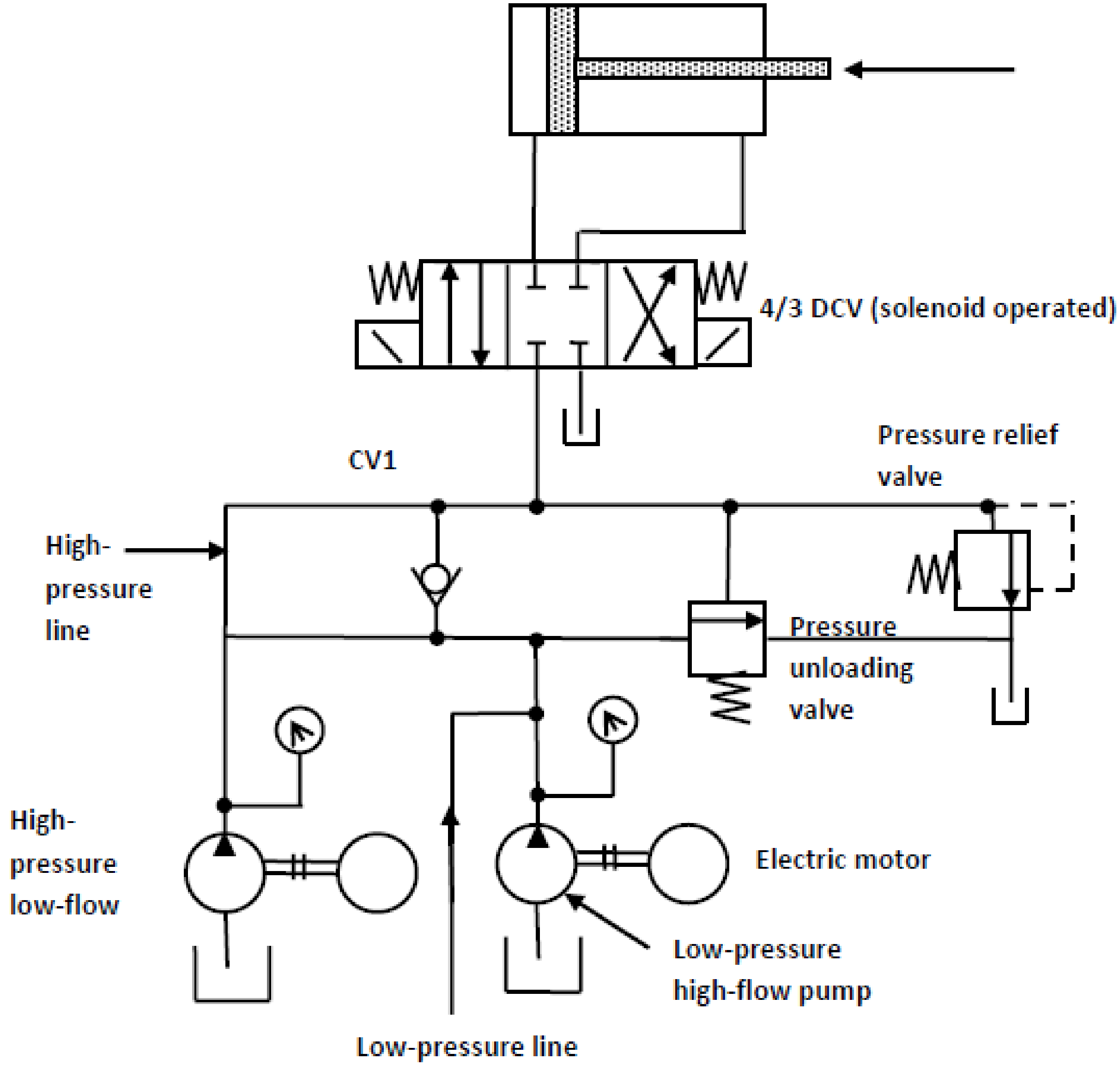


Figure 1.22

Example 1.8

For the double-pump system in Fig. 1.23, what should be pressure setting of the unloading valve and pressure-relief valve under the following conditions:

(a) Sheet metal punching operation requires a force of 8000 N.
(b) A hydraulic cylinder has a 3.75 cm diameter piston and a 1.25 cm diameter rod.
(c) During the rapid extension of the cylinder, a frictional pressure loss of 675 kPa occurs in the line from the high-flow pump to the blank end of the cylinder. During the same time, a 350 kPa pressure loss occurs in the return line from the rod end of the cylinder to the oil tank. Frictional pressure losses in these lines are negligibly small during the punching operation.
(d) Assume that the unloading valve and relief-valve pressure setting (for their full pump flow requirements) should be 50% higher than the pressure required to overcome frictional pressure losses and the cylinder punching load, respectively.



4/3 DCV (solenoid operated)
Pressure relief valve
CV1
High-pressure line
Pressure unloading valve
High-pressure low-flow
Electric motor
Low-pressure high-flow pump
Low-pressure line

Unloading valve: Back pressure force on the cylinder equals pressure loss in the return line times the effective area of the cylinder $(A_p - A_r)$:

$$F_{\text{back pressure}} = 350000\frac{\text{N}}{\text{m}^2}\times\frac{\pi}{4}(0.0375^2 - 0.0125^2)\ \text{m}^2 = 344\ \text{N}$$

Pressure at the blank end of the cylinder required to overcome back pressure force equals the back pressure force divided by the area of the cylinder piston:

$$p_{\text{cyl blank end}} = \frac{344\ \text{N}}{\frac{\pi}{4}(0.0375^2)\text{m}^2} = 311\ \text{kPa}$$

Thus,the pressure setting of unloading valve equals

$$1.50(675+311)\ \text{kPa} = 1480\ \text{kPa}$$

Pressure relief valve: Pressure required to overcome the punching operation equals the punching load divided by the area of the cylinder piston:

$$p_{\text{punching}} = \frac{8000\ \text{N}}{\frac{\pi}{4}(0.0375^2)\text{m}^2} = 7240\ \text{kPa}$$

Thus, the pressure setting of pressure-relief valve equals

$$1.50\times 7240\ \text{kPa} = 10860\ \text{kPa}$$

ME 5451 – Hydraulics and Pneumatics

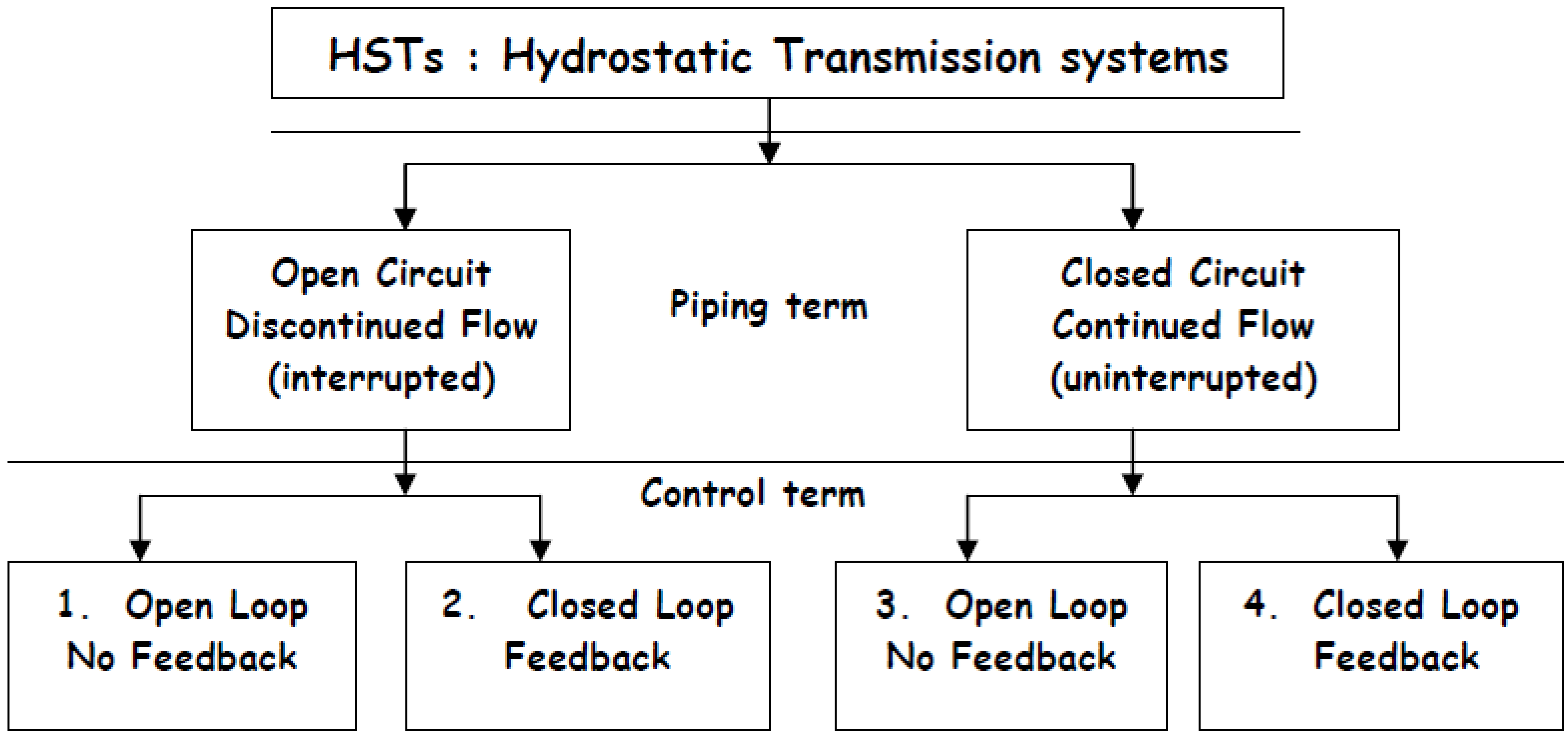
Lecture -19

Date: 27-05-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Review of lecture 18
2. Industrial hydraulic circuits

Course Instructor: Dr. A. Siddharthan


HSTs : Hydrostatic Transmission systems
Open Circuit
Discontinued Flow
(interrupted)
Piping term
Closed Circuit
Continued Flow
(uninterrupted)
Control term
1. Open Loop
No Feedback
2. Closed Loop
Feedback
3. Open Loop
No Feedback
4. Closed Loop
Feedback

(a)

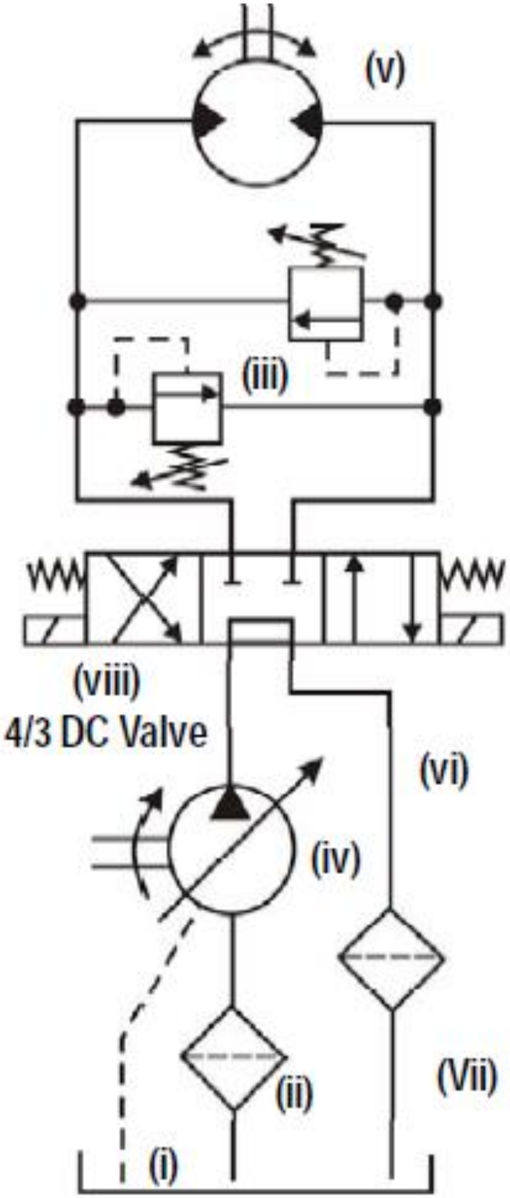


Fig. 6.23-2 : Open Circuit HST

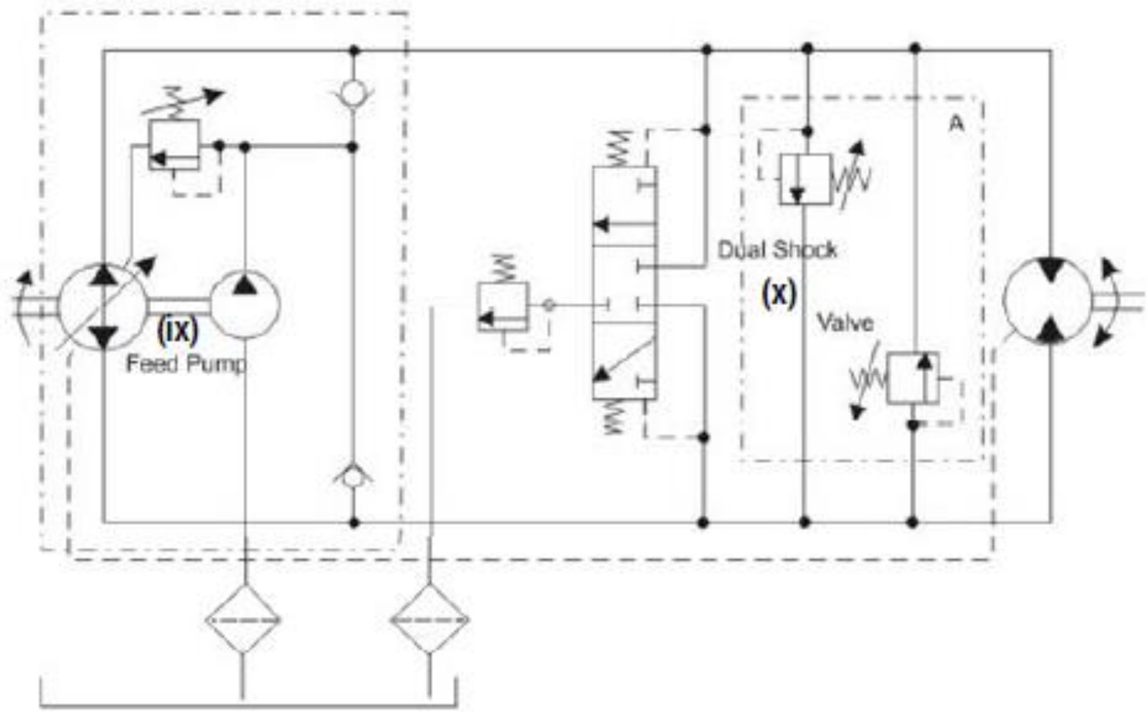
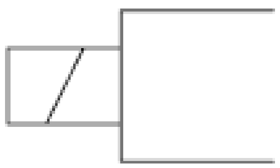


Fig. 6.23-3 : Closed Circuit HST

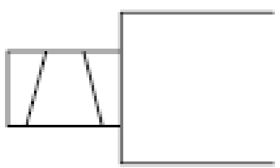
Displacement		Transmission		Output Speed	Commonly Known as
PUMP	MOTOR	POWER	TORQUE		
Fixed	Fixed	Constant	Constant	Constant	--
Variable	Fixed	Variable	Constant	Variable	Constant Torque System
Fixed	Variable	Constant	Variable	Variable	Constant Power System
Variable	Variable	Variable	Variable	Variable	--

Actuation modes of directional control valves in electro-hydraulics

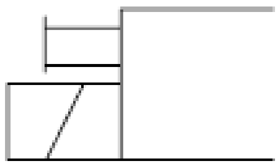
Solenoid with one winding



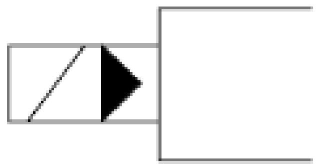
Solenoid with two opposing windings



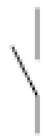
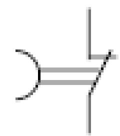
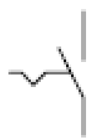
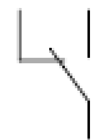
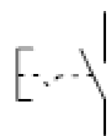
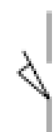
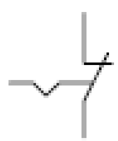
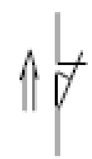
Solenoid with manual override



Two-stage (pilot-actuated) valve;
the piloted directional control valve is
electromagnetically actuated

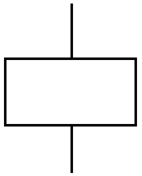


Switching elements

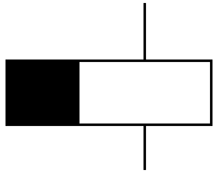
	normally open contact		normally closed contact delays when dropping off
	normally open contact, latched		changeover contact
	normally open contact, closes in delayed mode		control switch with normally open contact
	normally closed contact		limit switch
	normally closed contact, latched		limit switch (actuated normally open contact)

Electromechanical switching elements

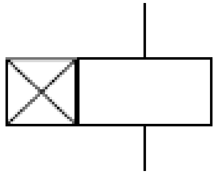
relay, contactor



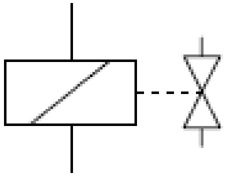
relay with switch-off delay



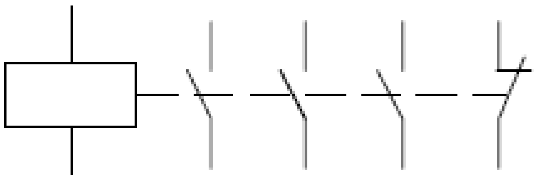
relay with switch-on delay



shut-off valve,
electromechanically actuated

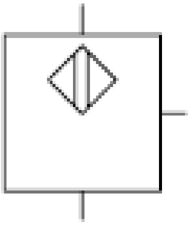


relay with three normally
open contacts and
one normally closed contact

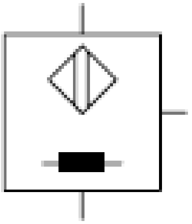


Block symbols for proximity sensors

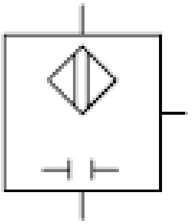
proximity sensor, general



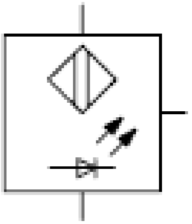
proximity sensor, inductive



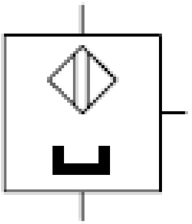
proximity sensor, capacitive



proximity sensor, optical



proximity sensor, magnetic



ME 5451 – Hydraulics and Pneumatics

Lecture -20

Date: 27-5-2021   Time slot: 08:30-10:10 a.m.

Contents

1. Review of lecture 19
2. Electrohydraulic circuits
3. Pneumatics – Basics
4. Compressors
5. Filter Regulator Lubricator

Course Instructor: Dr. A. Siddharthan

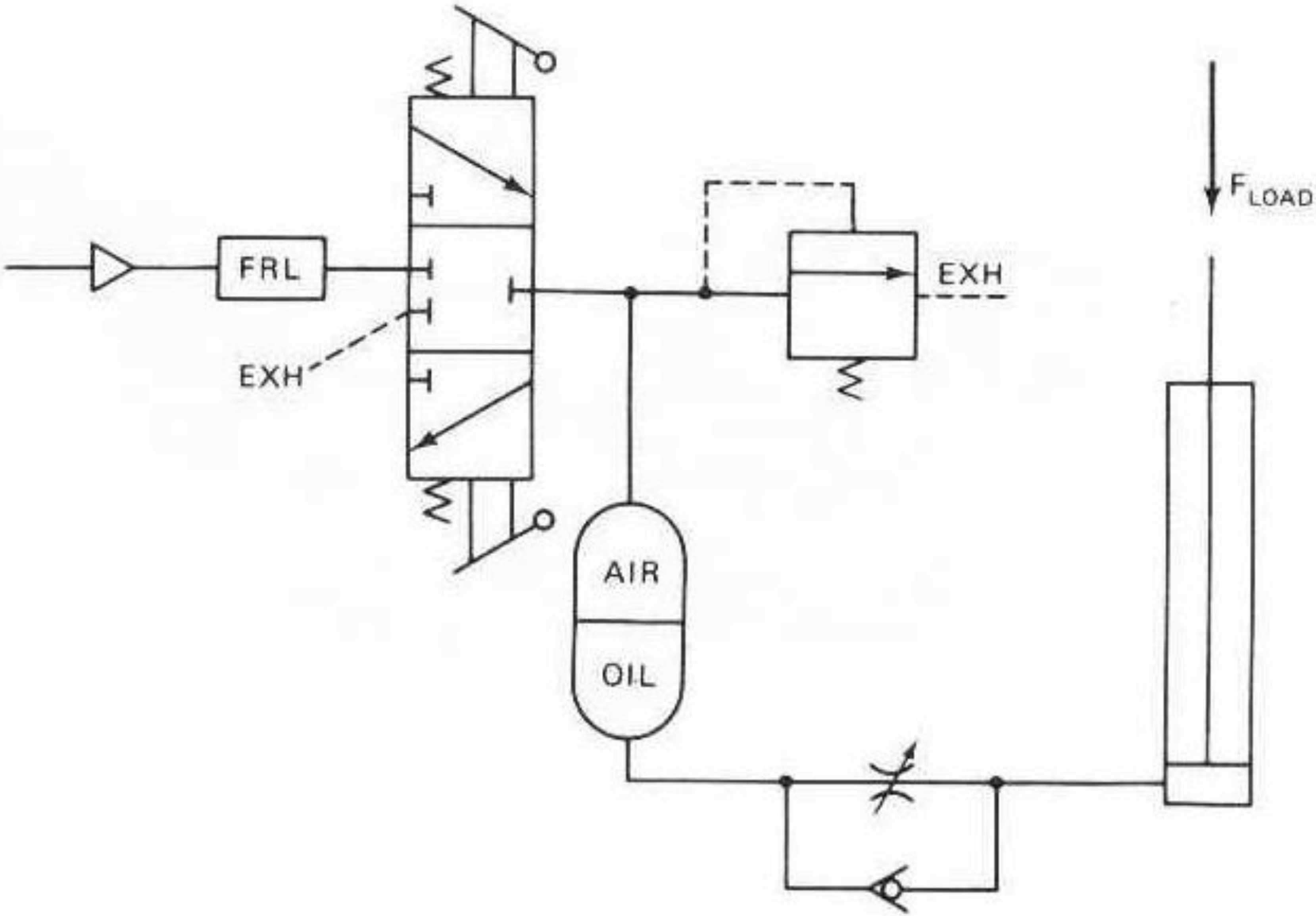


Figure 9-24. Air-over-oil circuit.

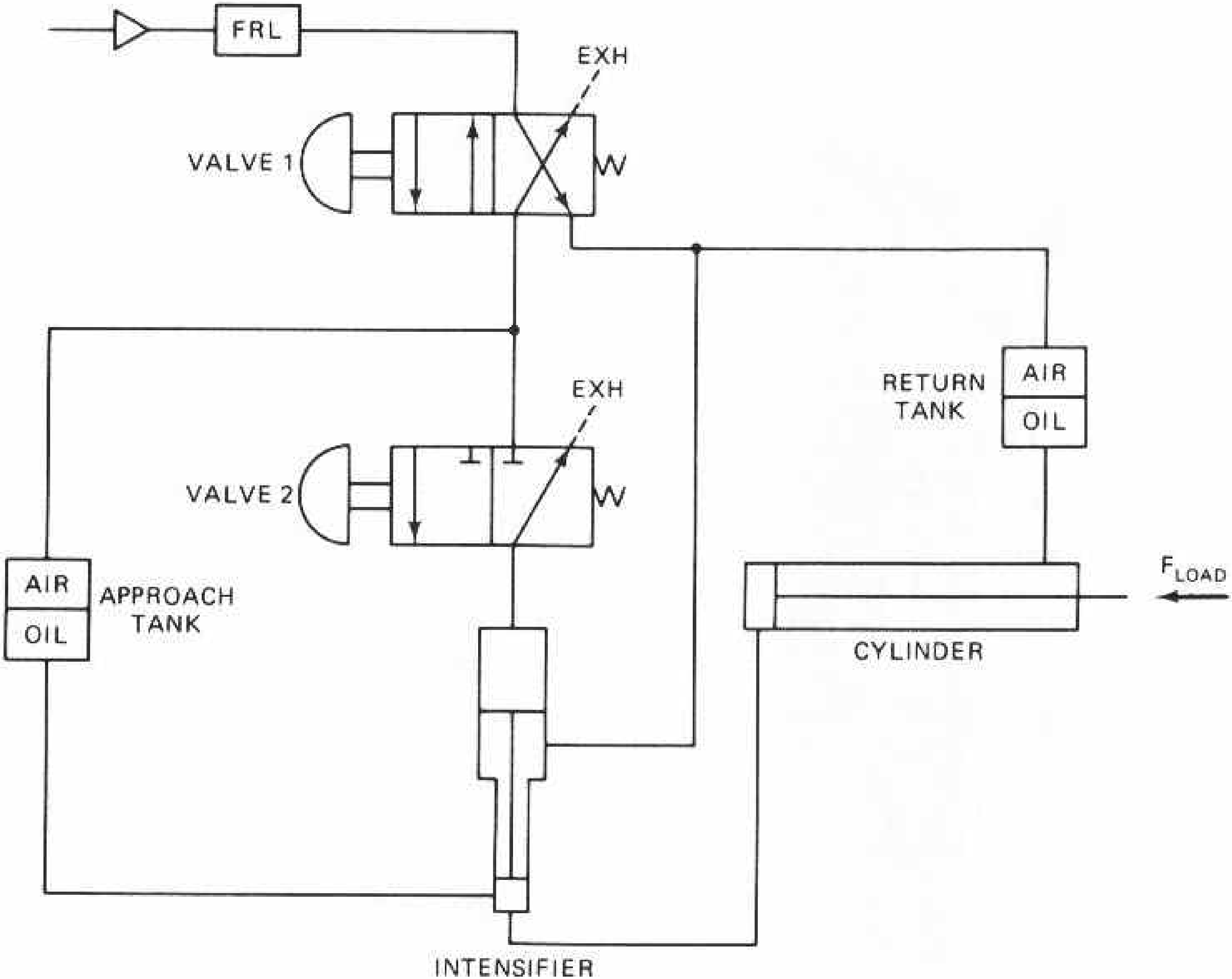
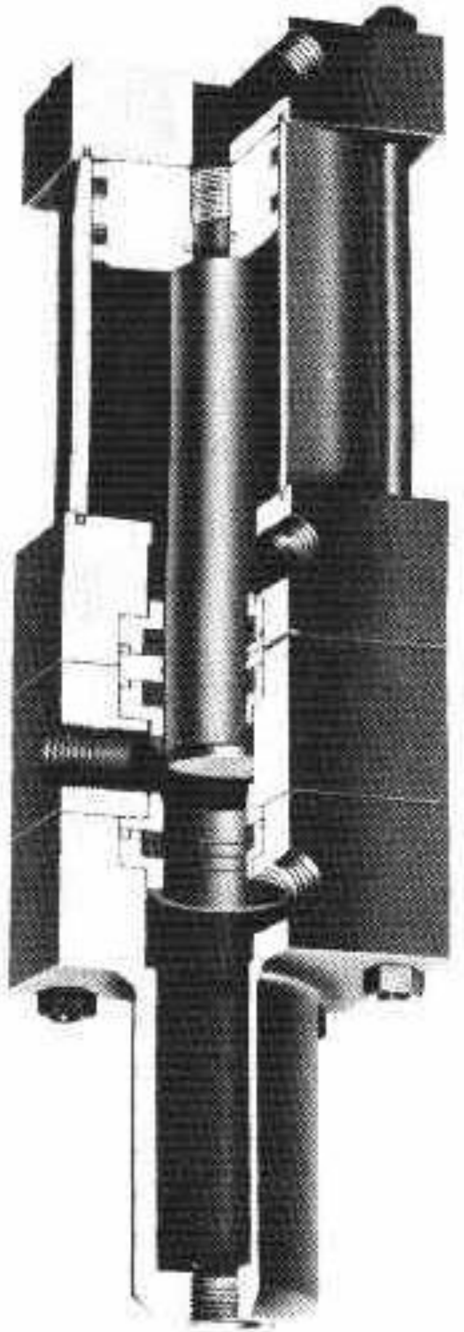
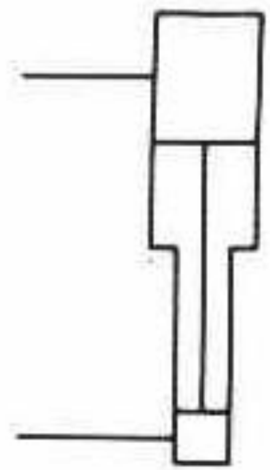


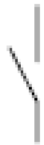
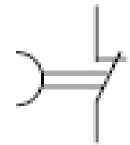
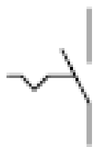
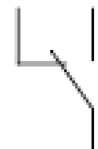
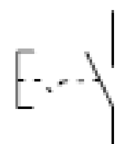
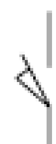
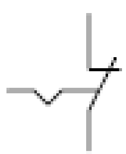
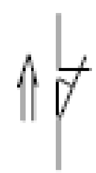
Figure 9-25. Air-over-oil intensifier circuit.





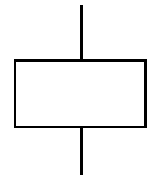
ANSI SYMBOL

Switching elements

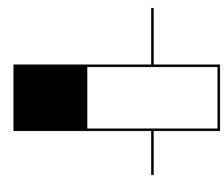
	normally open contact		normally closed contact delays when dropping off
	normally open contact, latched		changeover contact
	normally open contact, closes in delayed mode		control switch with normally open contact
	normally closed contact		limit switch
	normally closed contact, latched		limit switch (actuated normally open contact)

Electromechanical switching elements

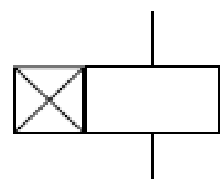
relay, contactor



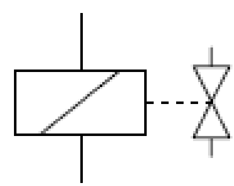
relay with switch-off delay



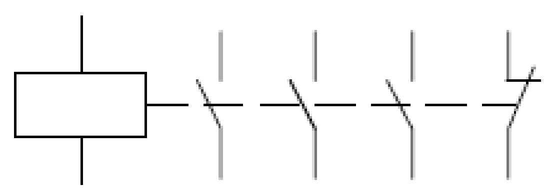
relay with switch-on delay



shut-off valve,
electromechanically actuated

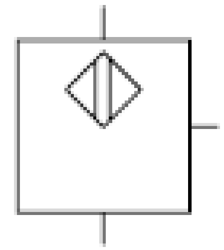


relay with three normally
open contacts and
one normally closed contact

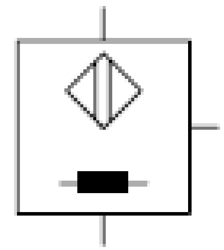


Block symbols for proximity sensors

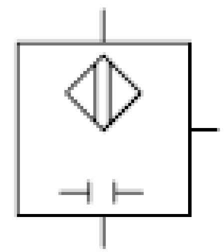
proximity sensor, general



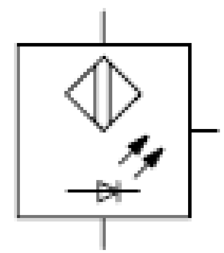
proximity sensor, inductive



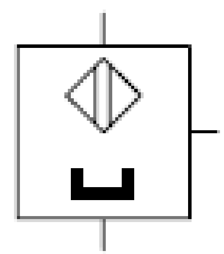
proximity sensor, capacitive



proximity sensor, optical



proximity sensor, magnetic



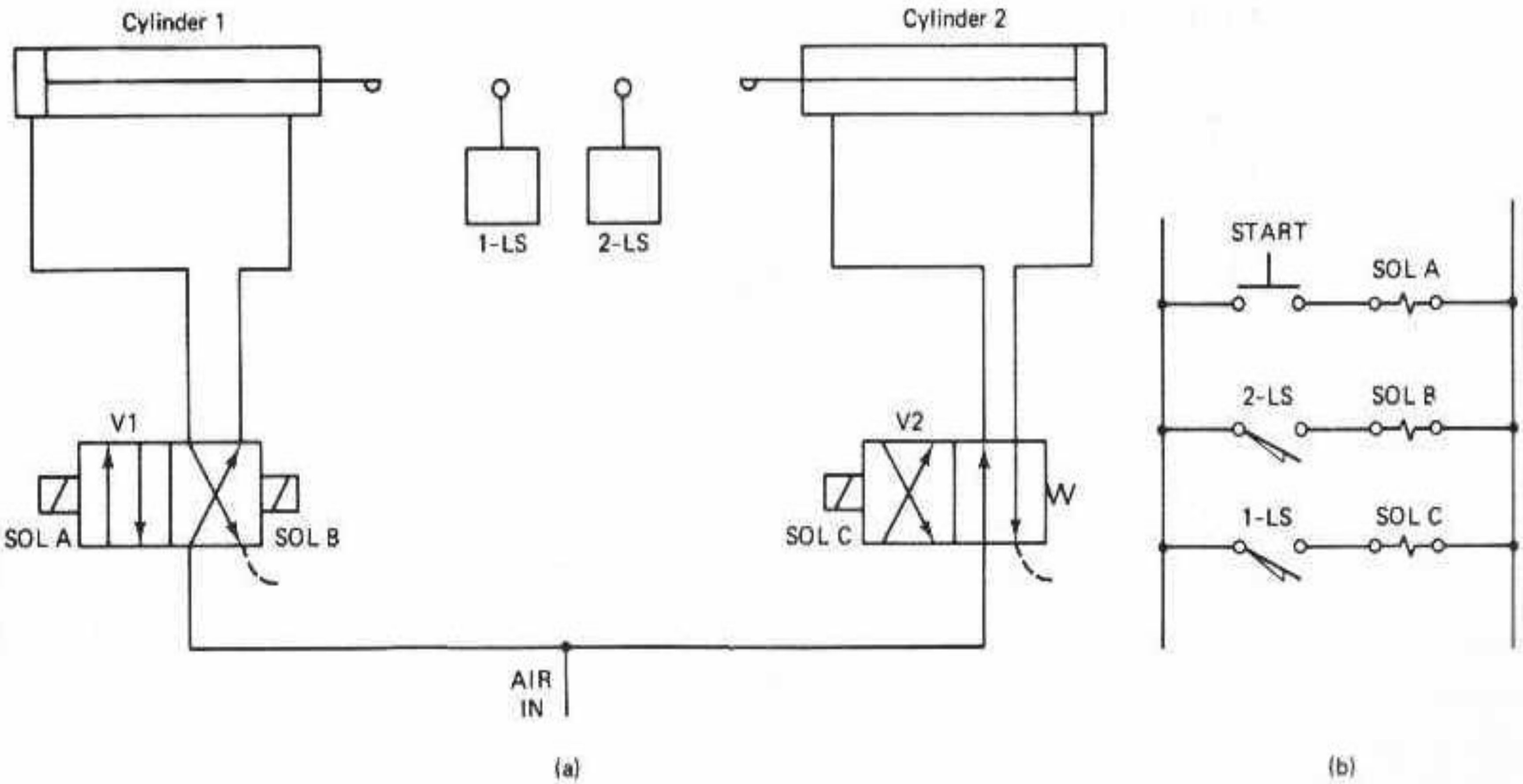
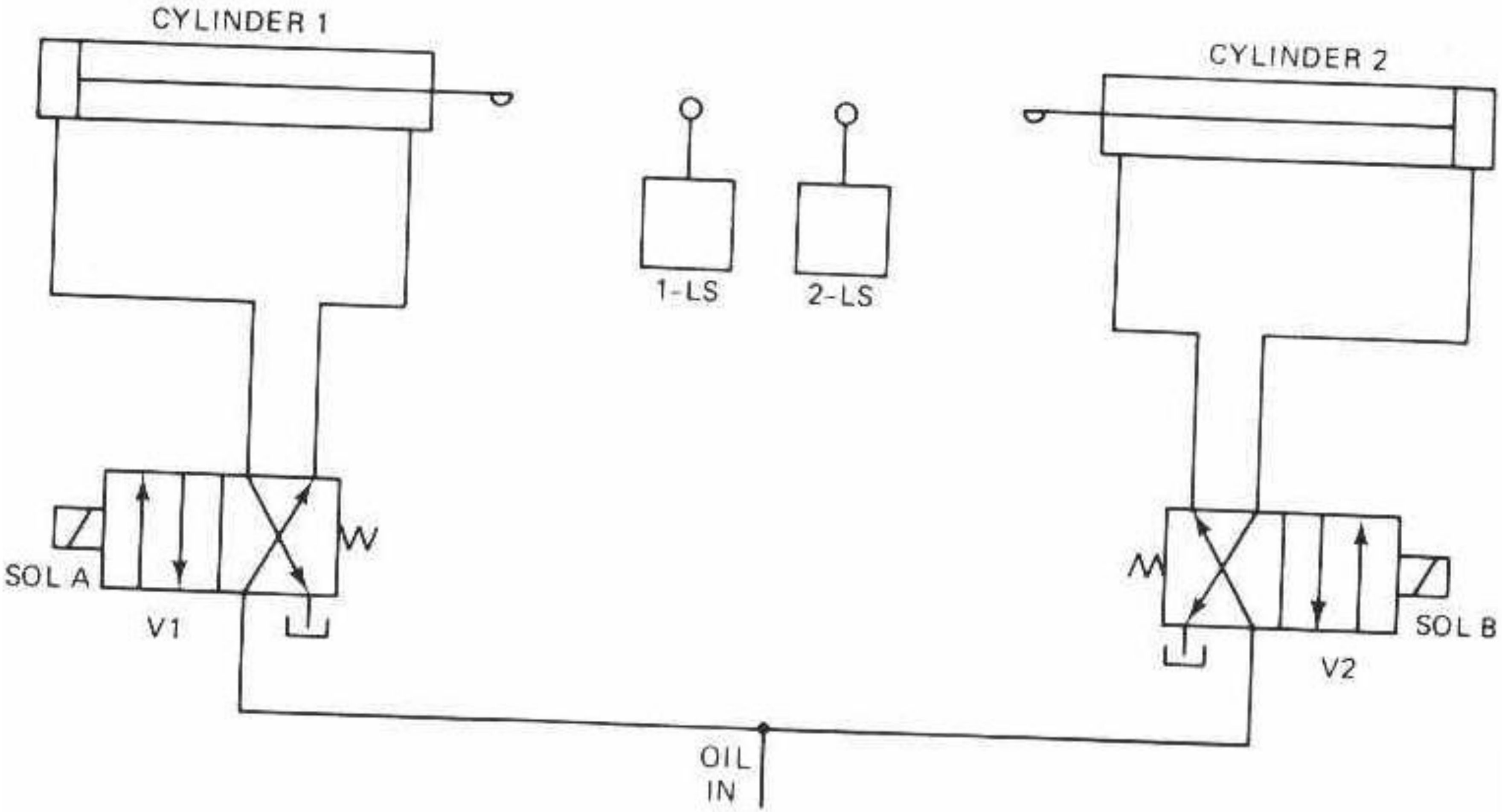
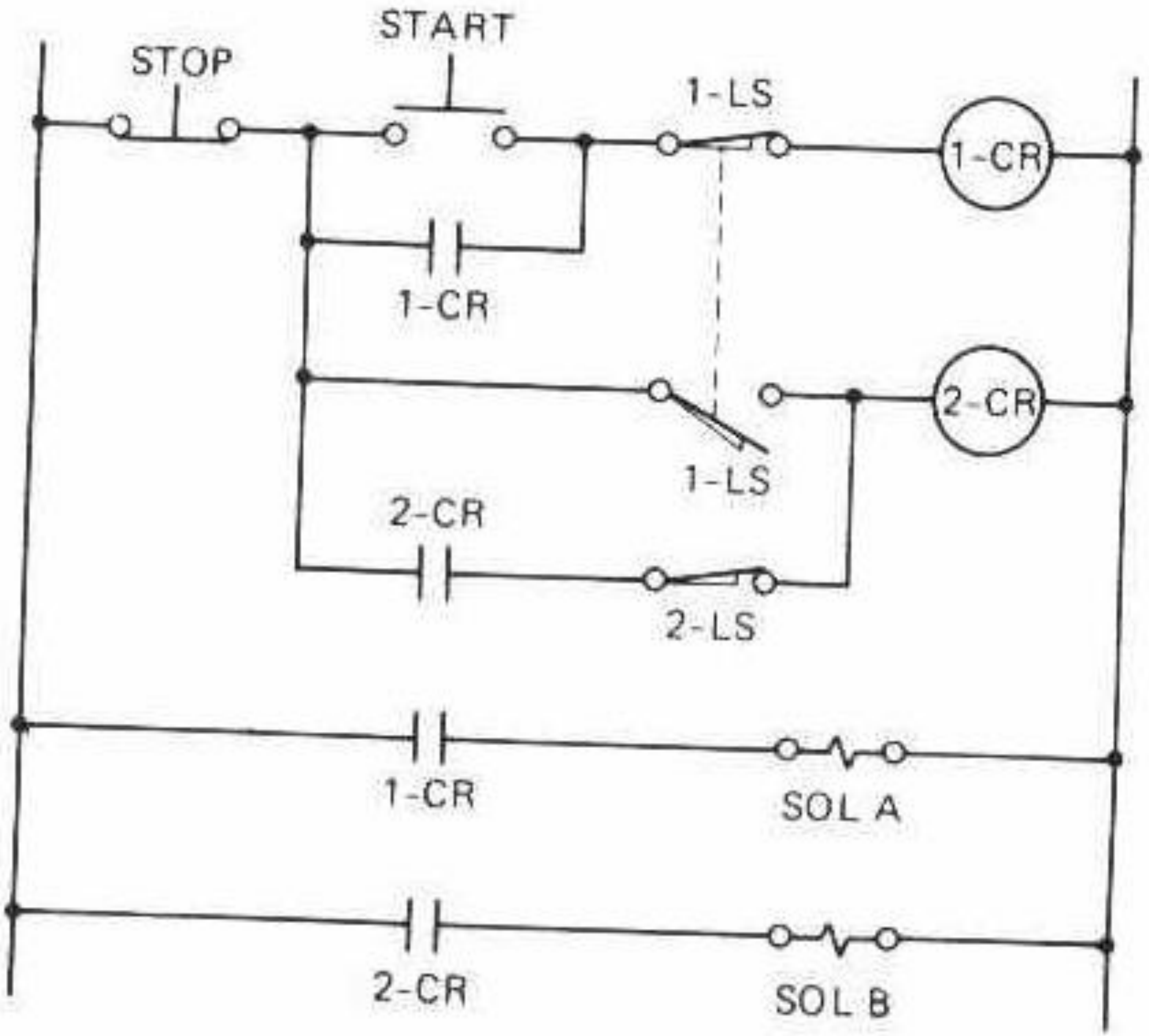


Figure 13-14. Dual cylinder sequencing circuit.

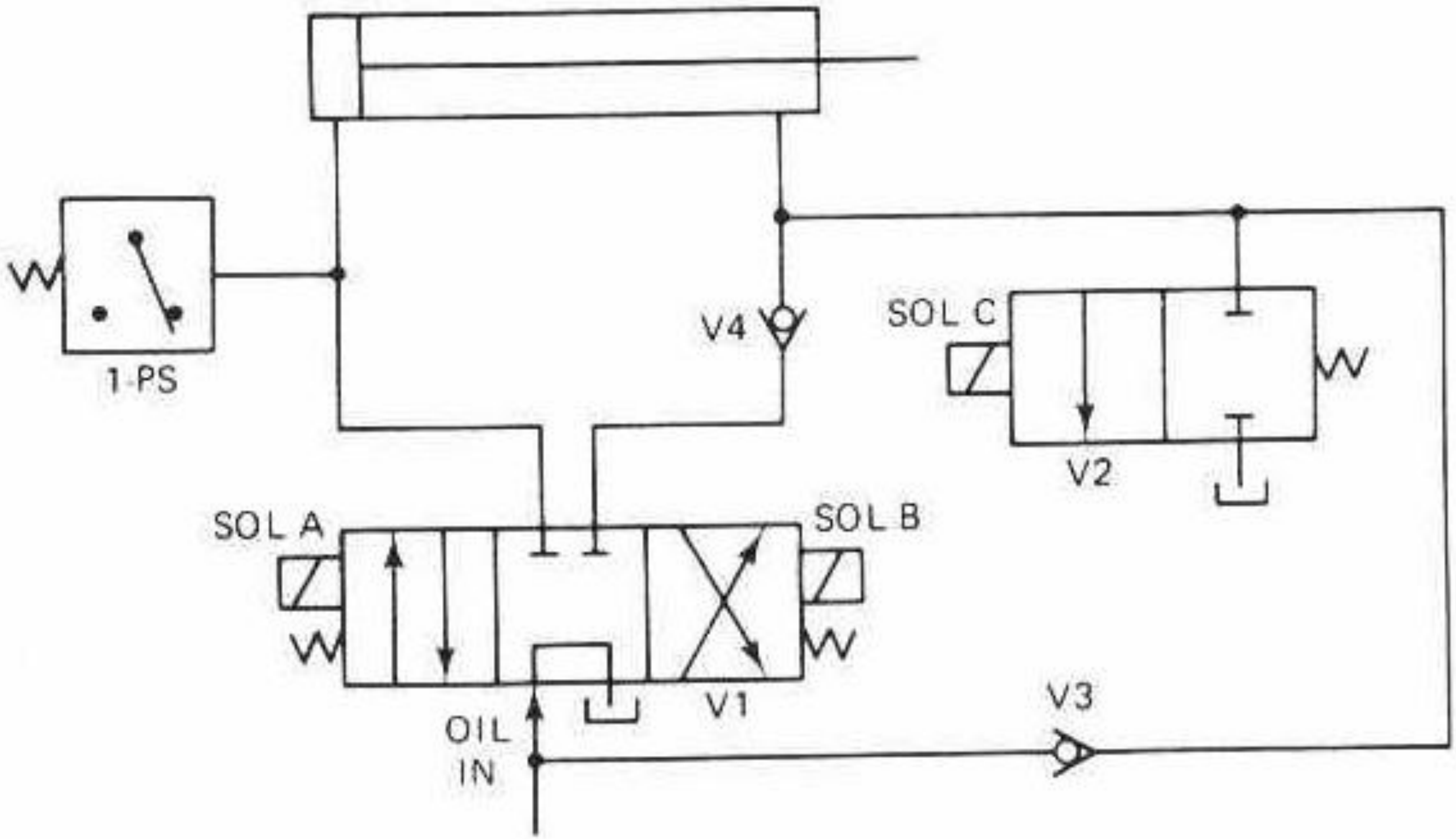
1. Cylinder 1 extends.
2. Cylinder 2 extends.
3. Both cylinders retract.
4. Cycle is ended.



CYLINDER 1
CYLINDER 2
1-LS
2-LS
SOL A
V1
V2
SOL B
OIL
IN



1. Cylinder 1 extends.
2. Cylinder 2 extends while cylinder 1 retracts.
3. Cylinder 2 retracts.
4. Cycle is ended.



(a)

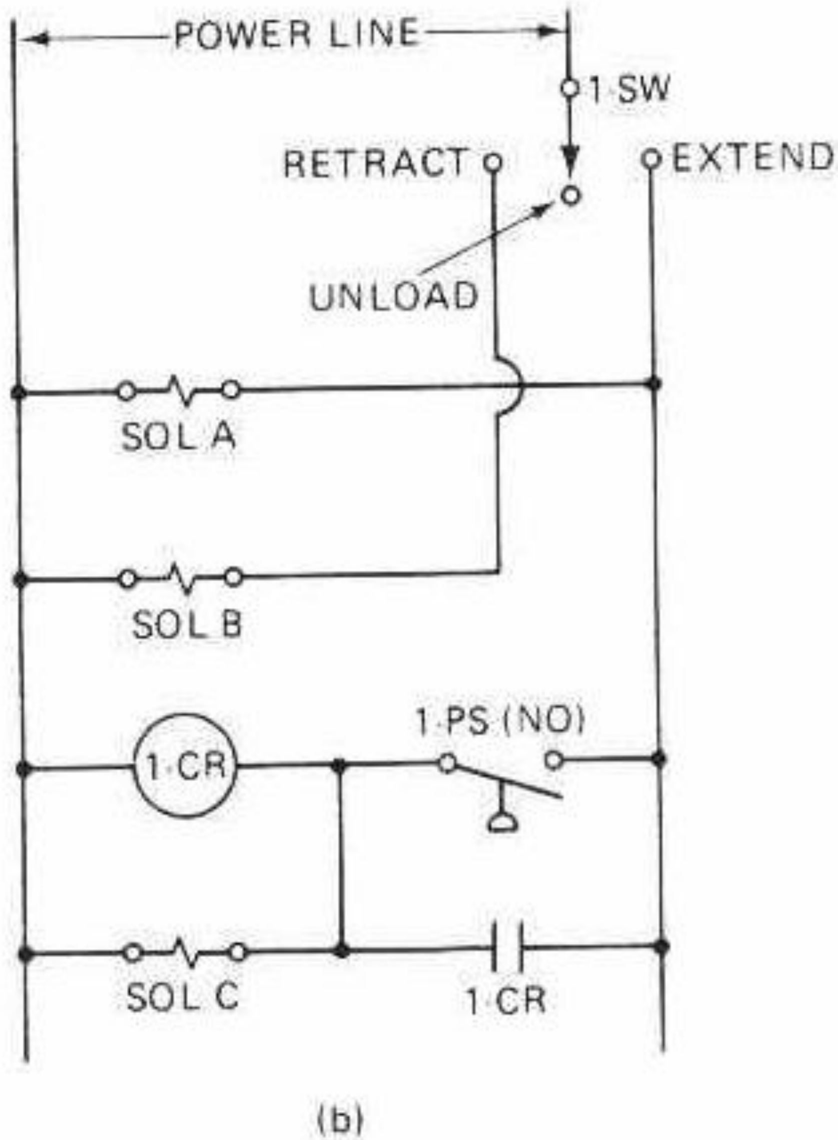
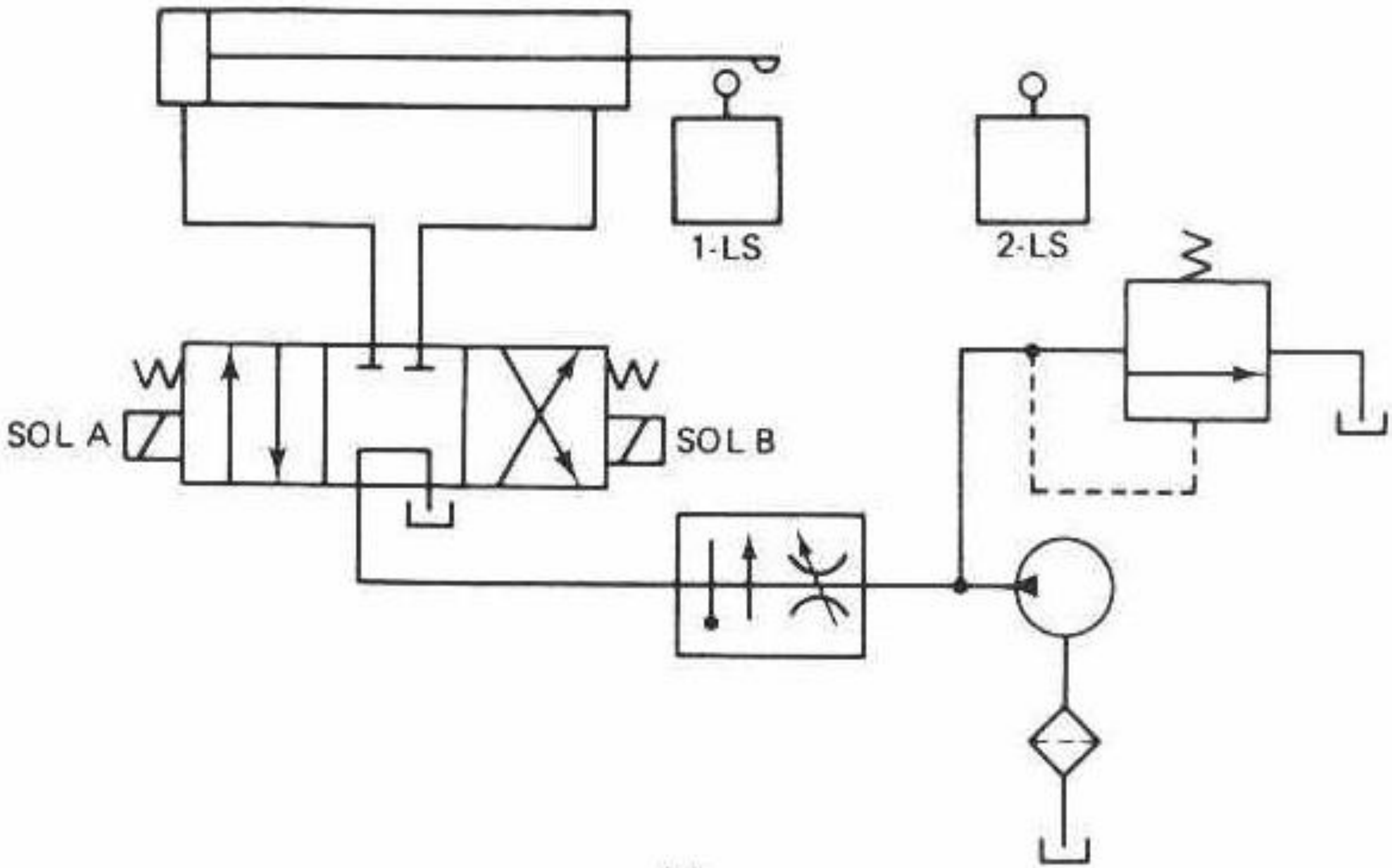
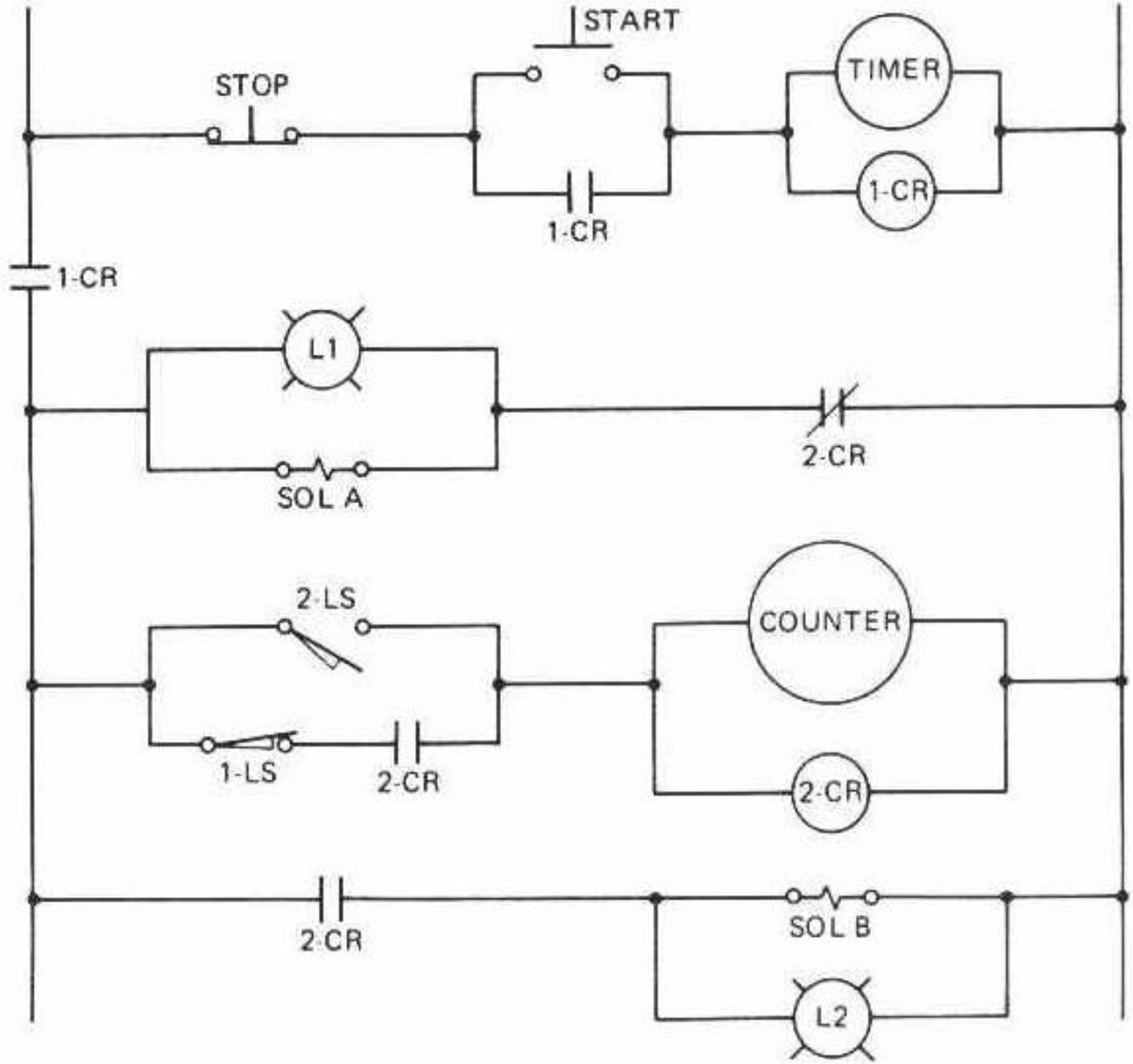


Figure 13-17. Electrical control of regenerative circuit.



(a)



START
STOP
TIMER
1-CR
1-CR
1-CR
L1
SOL A
2-CR
2-LS
COUNTER
1-LS
2-CR
2-CR
2-CR
SOL B
L2

$$\text{Relative humidity of air} = \frac{\text{Absoulte humidity}}{\text{humidity at saturatrion}} \times 100$$

Gas Laws

To understand Pneumatic systems, we must first understand the behavior of gases. Their behavior is described by the perfect **gas laws.**

Ideal gas Equation : $p.V = m.R.T$ $\Rightarrow$ $p.V = \text{cons. } T$

Boyle's Law : $p_1 V_1 = p_2 V_2$ (Temp. is constant) $p \; \alpha \; 1/V$

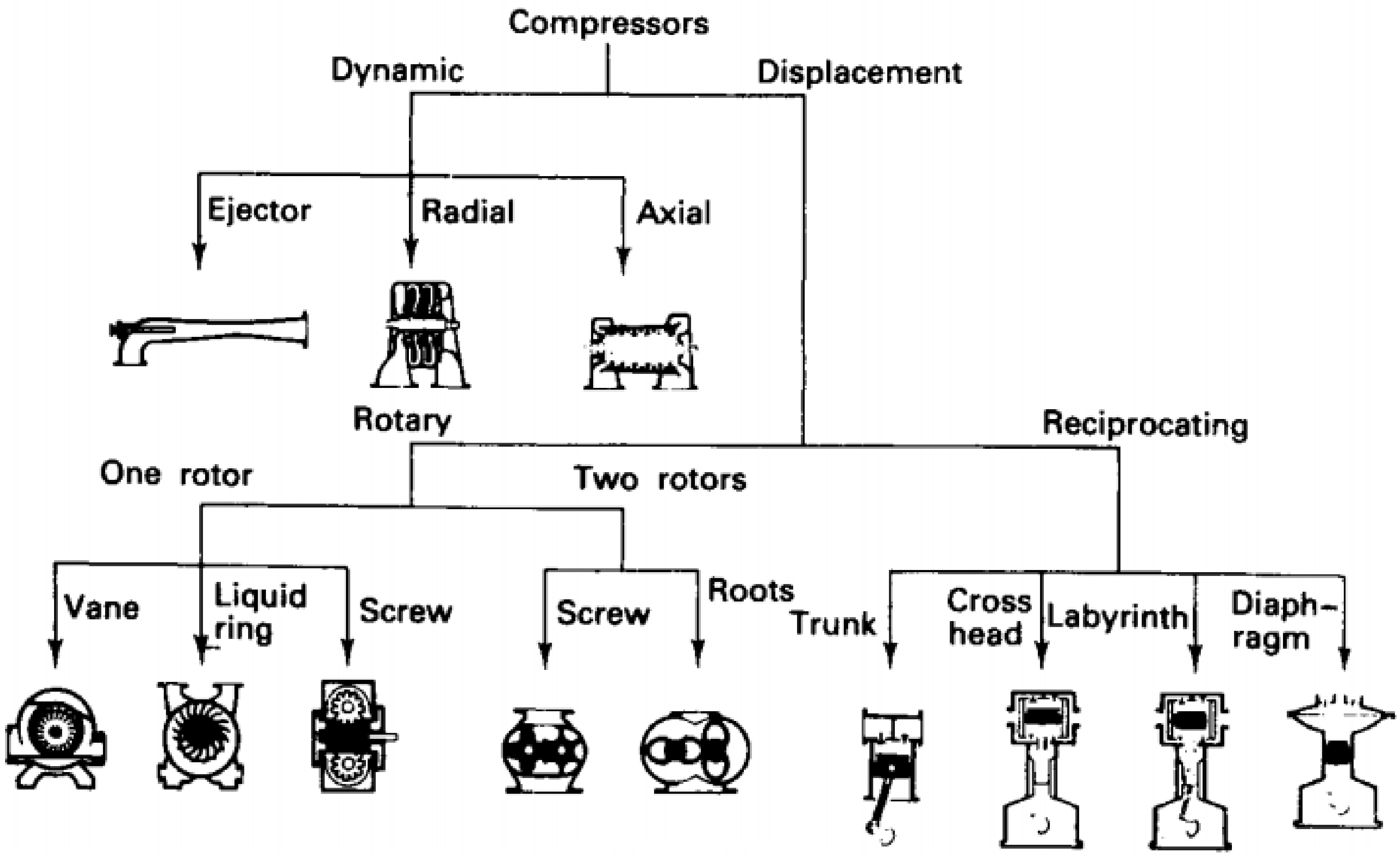
Charle's law : $V_1/T_1 = V_2/T_2$ (Pressure is constant) $V \; \alpha \; T$

Gay-Lussac's Law $p_1/T_1 = p_2/T_2$ (Volume is constant) $p \; \alpha \; T$

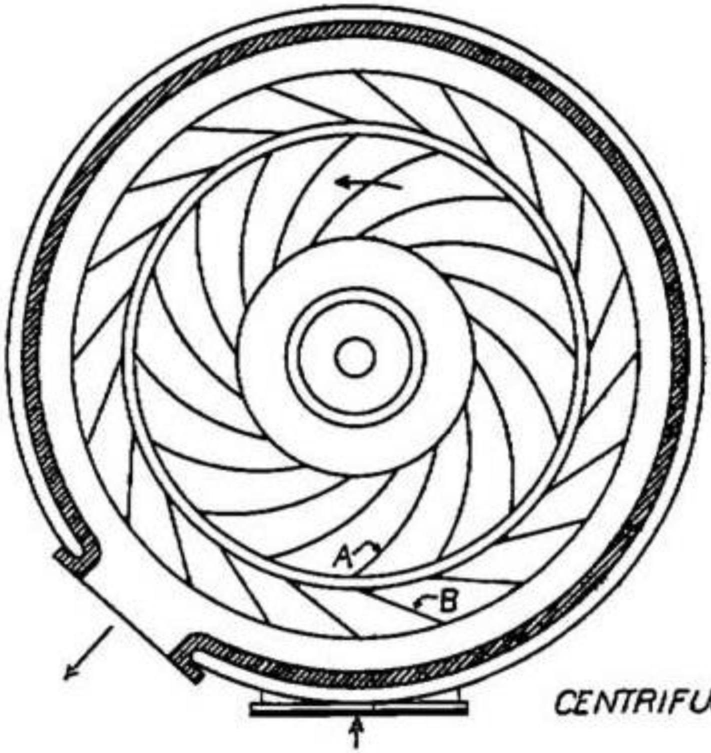
Table 1.4 Comparison between Hydraulic and Pneumatic systems

Hydraulic system	Pneumatic system
It employs a pressurized liquid as fluid	it employs a compressed gas usually air as a fluid
Oil hydraulics system operates at pressures upto 700 bar.	Pneumatics systems usually operate at 5 to 10 bar.
Generally designed for closed systems	Pneumatic systems are usually designed as open system
System get slow down of leakage occurs	Leakage does not affect the system much more
Valve operations are difficult	Easy to operate the valves
Heavier in weight	Light in weight
Pumps are used to provide pressurized liquids	Compressors are used to provide compressed gas
System is unsafe to fire hazards	System is free from fire hazards
Automatic lubrication is provided	Special arrangements for lubrication needed.

Compressor Classification

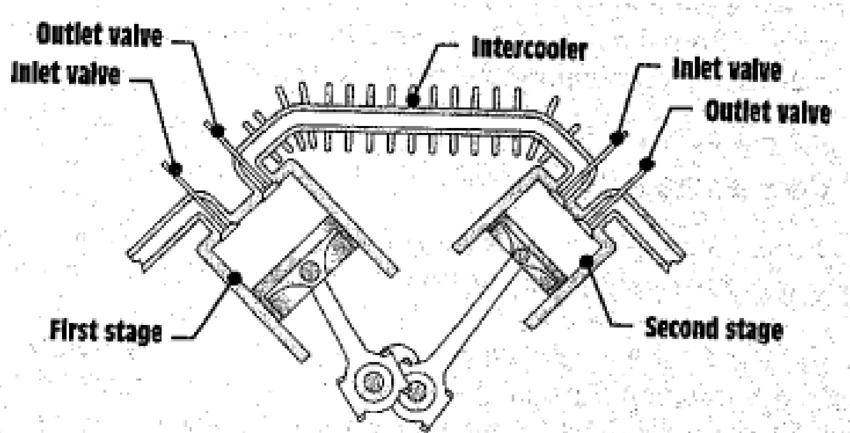


Non-positive Compressor (Centrifugal type)

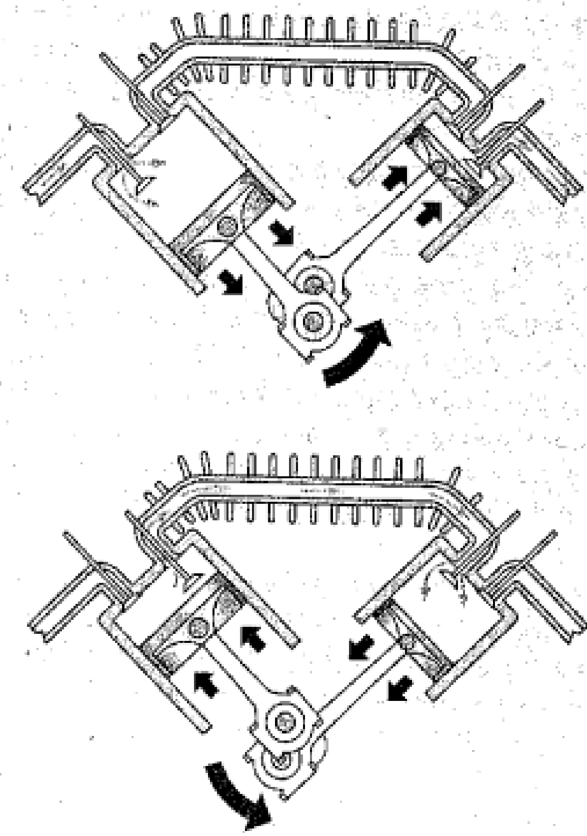


Cycling of Compressor

- Cut-in Pressure
- Cut-out Pressure



Two stage compressor



Cycling of the air pump is controlled by Pressure switch

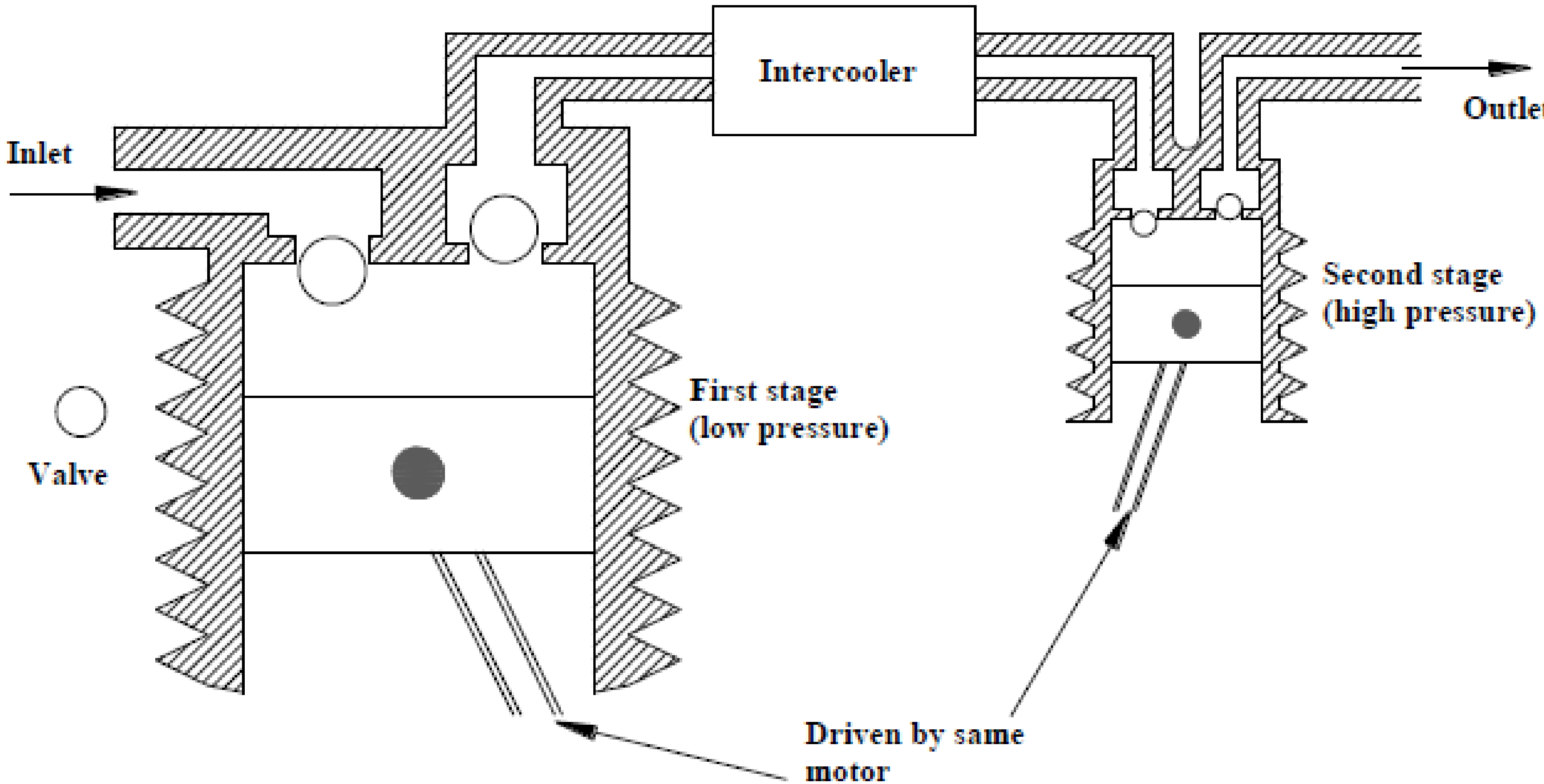


Figure 1.11 Multi stage piston air compressor with intercooler

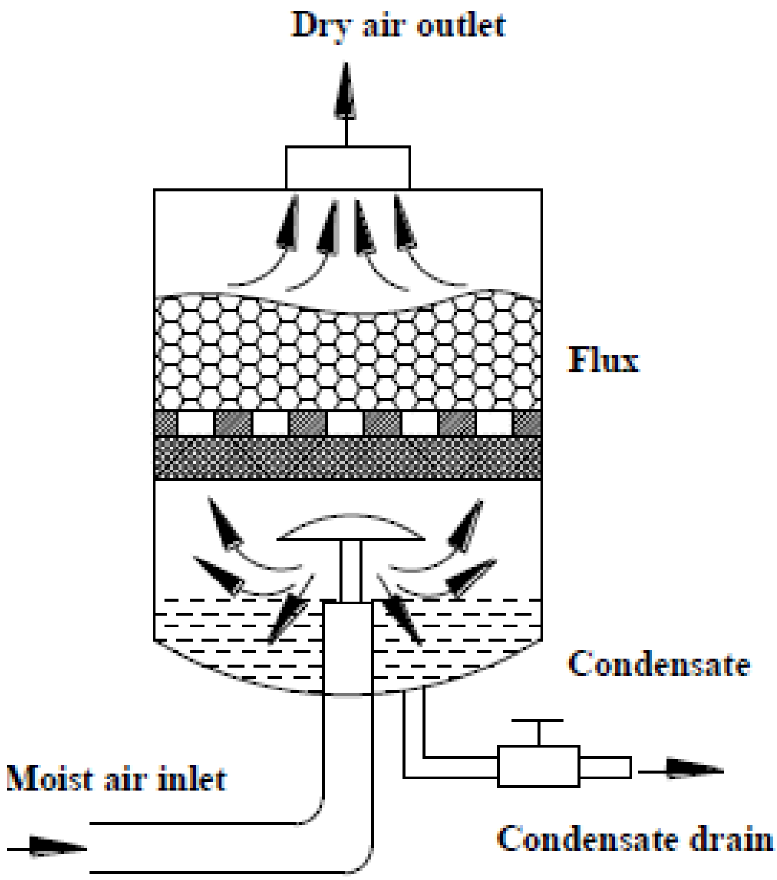


Figure 1.1 Absorption dryer

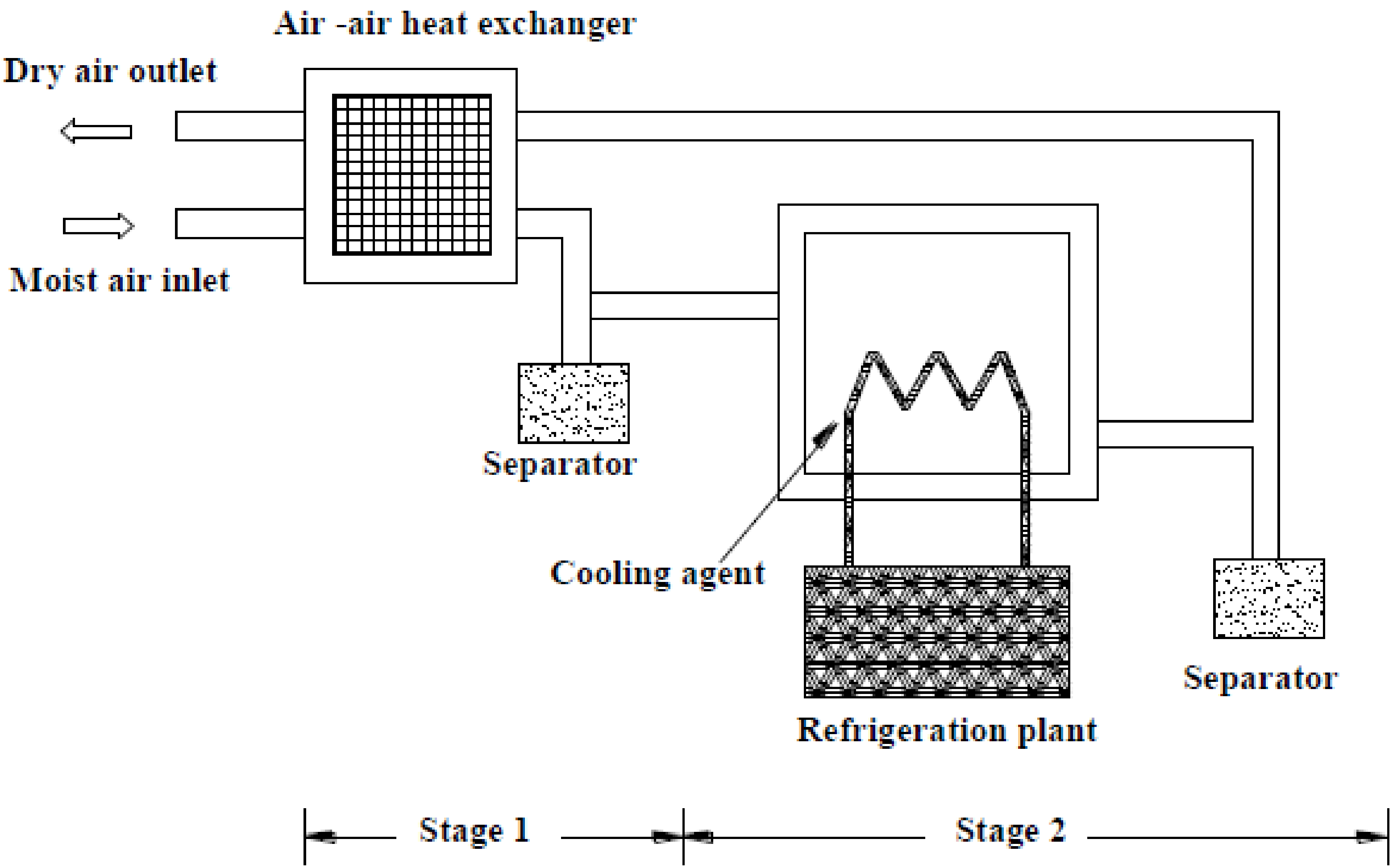


Figure 1.3 Refrigerated dryer

Type	Advantages	Disadvantages
Absorption	Pressure dew point + 16 °C	Inlet temperature must not exceed 30 °C
	Low capital cost	Drying agents are consumables and therefore must be regularly replenished. Highly corrosive chemical are used. They are not environmental friendly
Refrigeration	+ 3 °C pressure dew point	Output dew point will vary with approach temperature at the inlet and cleanliness of heat exchanger
	Input temperature can be as high as 16 °C	
Adsorption	Achievable pressure dew point of -40 °C	High capital cost
		High operating cost
		Use of micro filters adds cost to prevent the residue from chemicals.

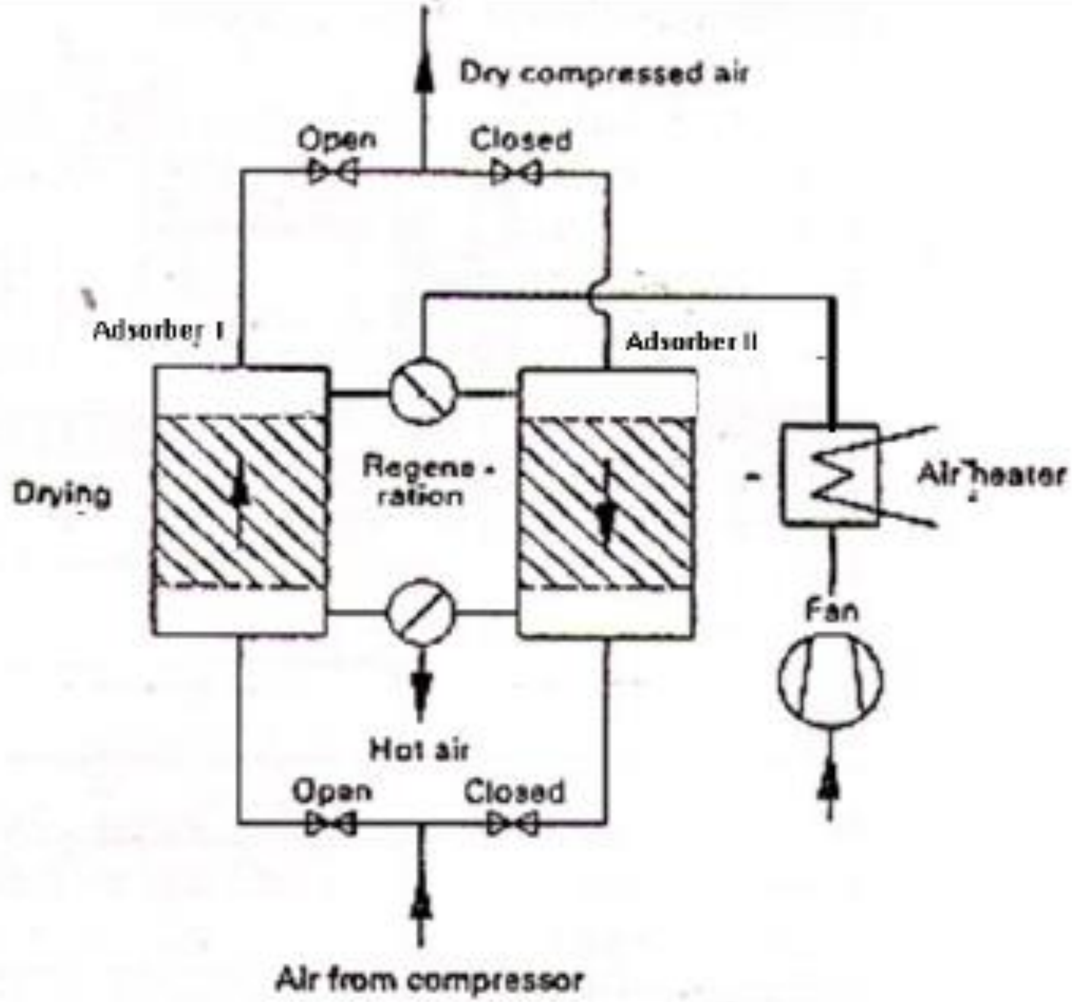


Figure 1.2 Adsorption type dryer

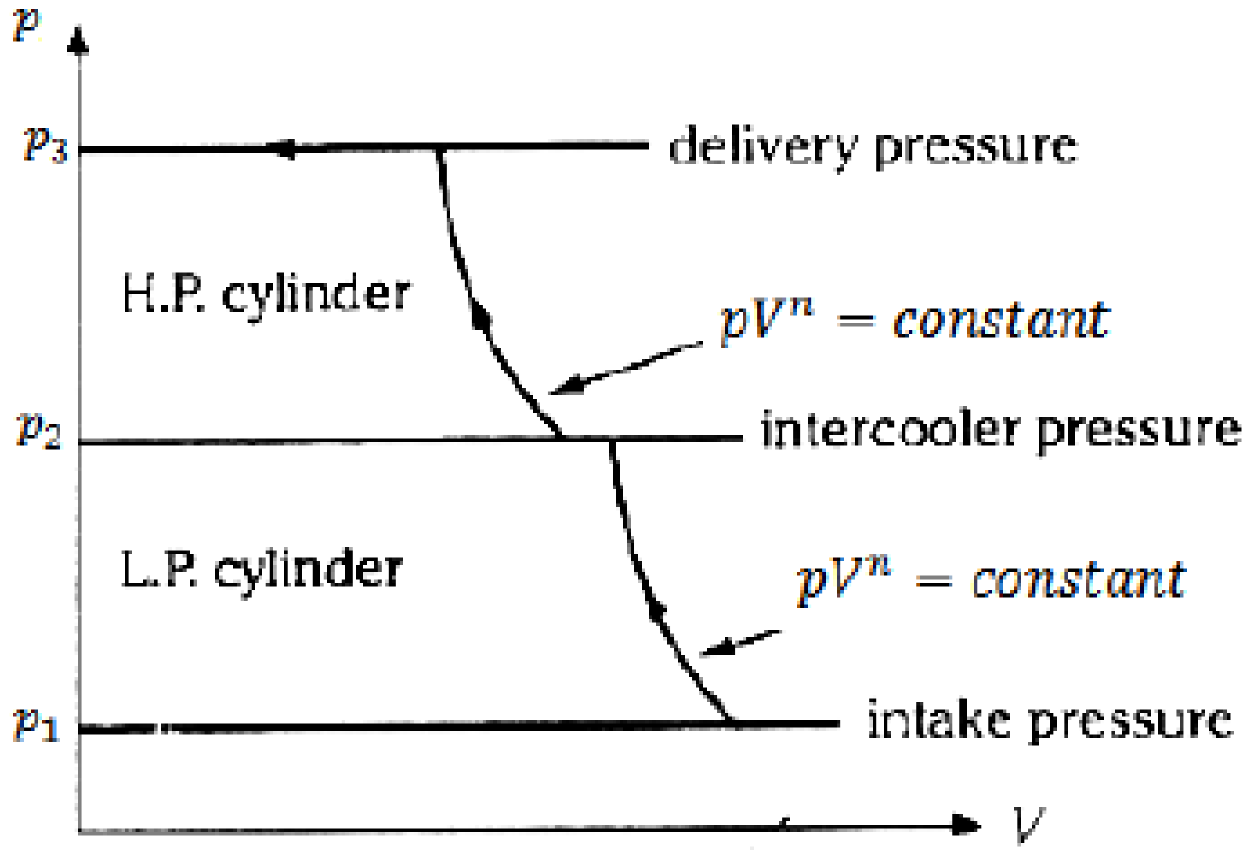


Figure 1.15 PV Diagram for two stage air compressor

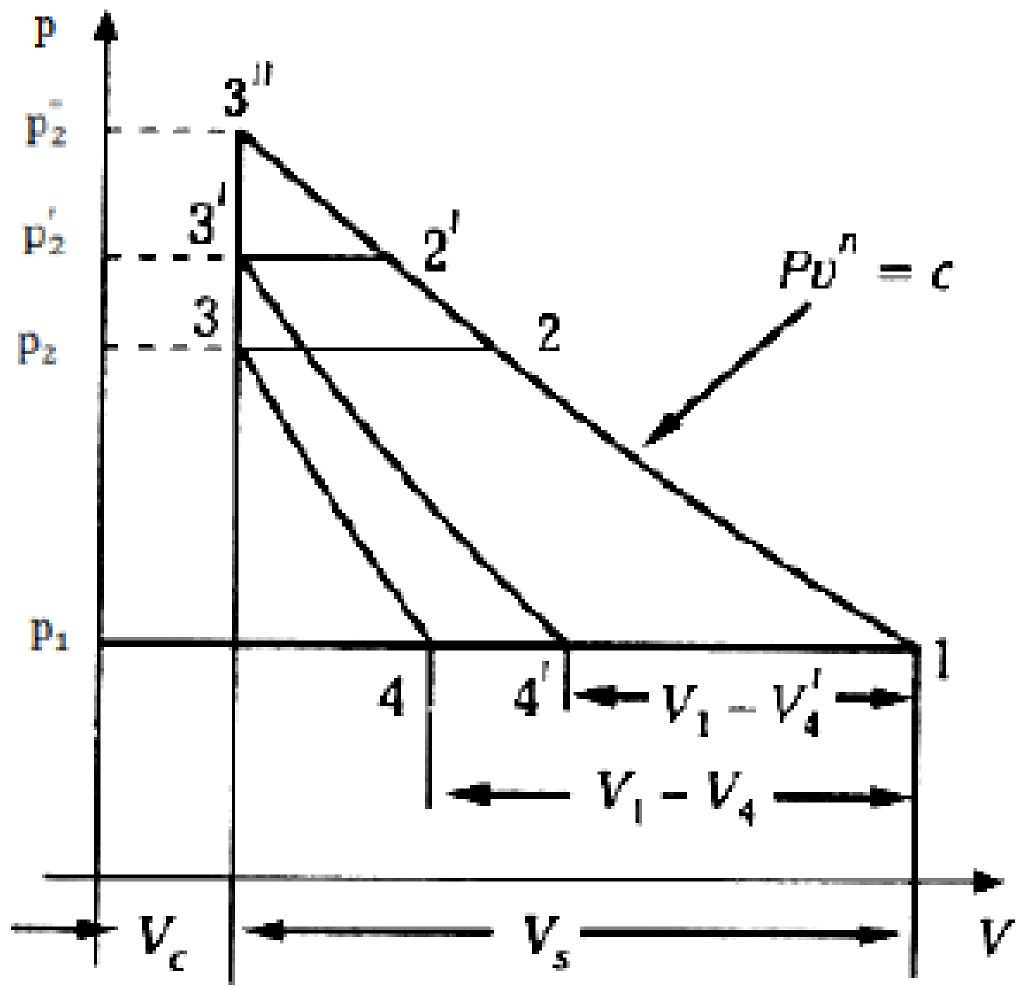
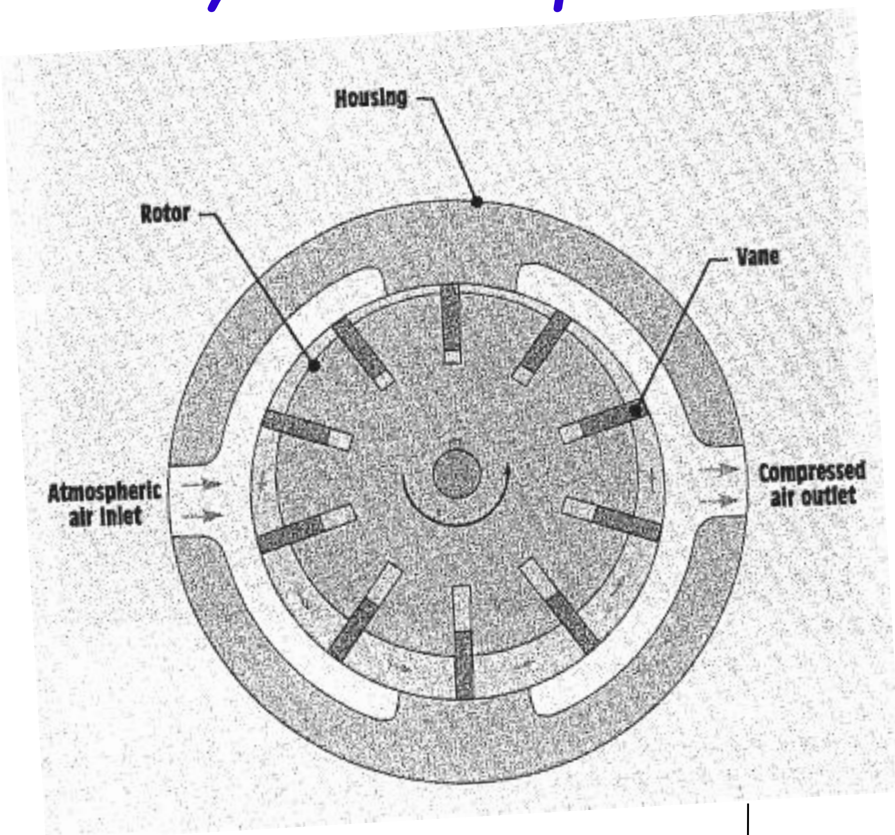
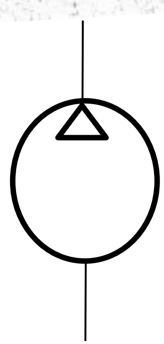


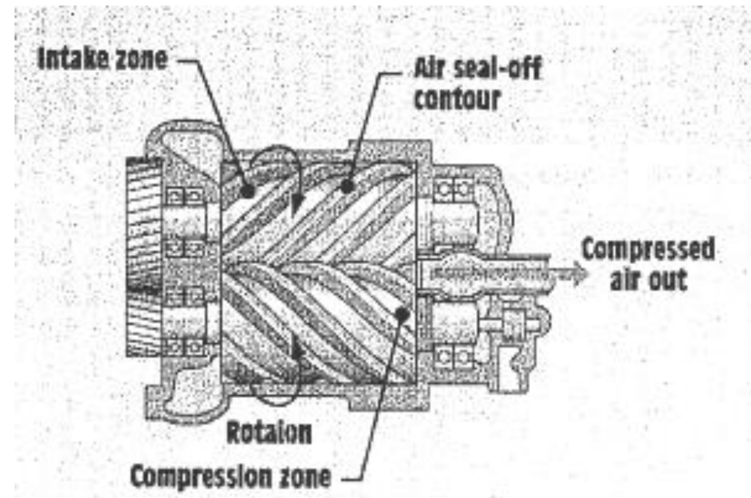
Figure 1.13 Multi stage compressions

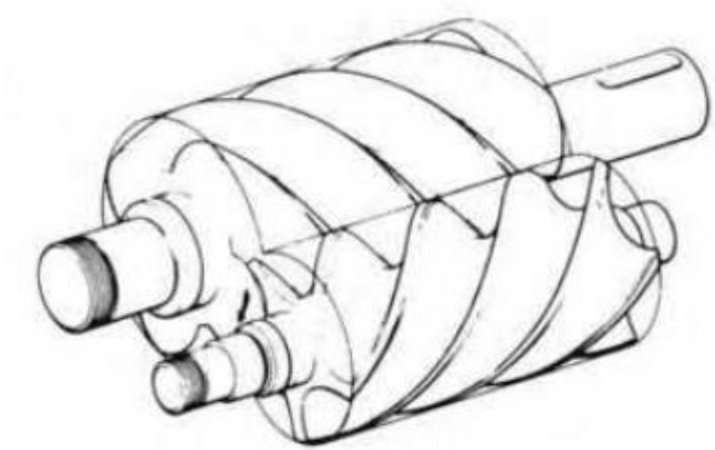
Rotary vane Compressor





Rotary Screw Compressor





Item	Reciprocating	Rotary vane	Rotary screw	Centrifugal
Efficiency at full load	High	Medium-high	High	High
Efficiency at part load	High due to staging	Poor: below 60% of full load	Poor: below 60% of full load	Poor: below 60% of full load
Efficiency at no load(power as % of full load)	High (10%-25%)	Medium (30%-40%)	High Poor (255-60%)	High-medium (20%-30%)
Noise level	Noisy	Quiet	Quiet it enclosed	Quiet
Size	Large	Compact	Compact	Compact
Oil carry over	Moderate	Low-medium	Low	Low
Vibration	High	Less	Less	Less
Maintenance	Many wearing parts	Few wearing parts	Very few wearing parts	Sensitive to dust in air
Capacity	Low-high	Low-medium	Low-high	Medium-high
Pressure	Medium- very high	Low-medium	Medium-high	Medium-high

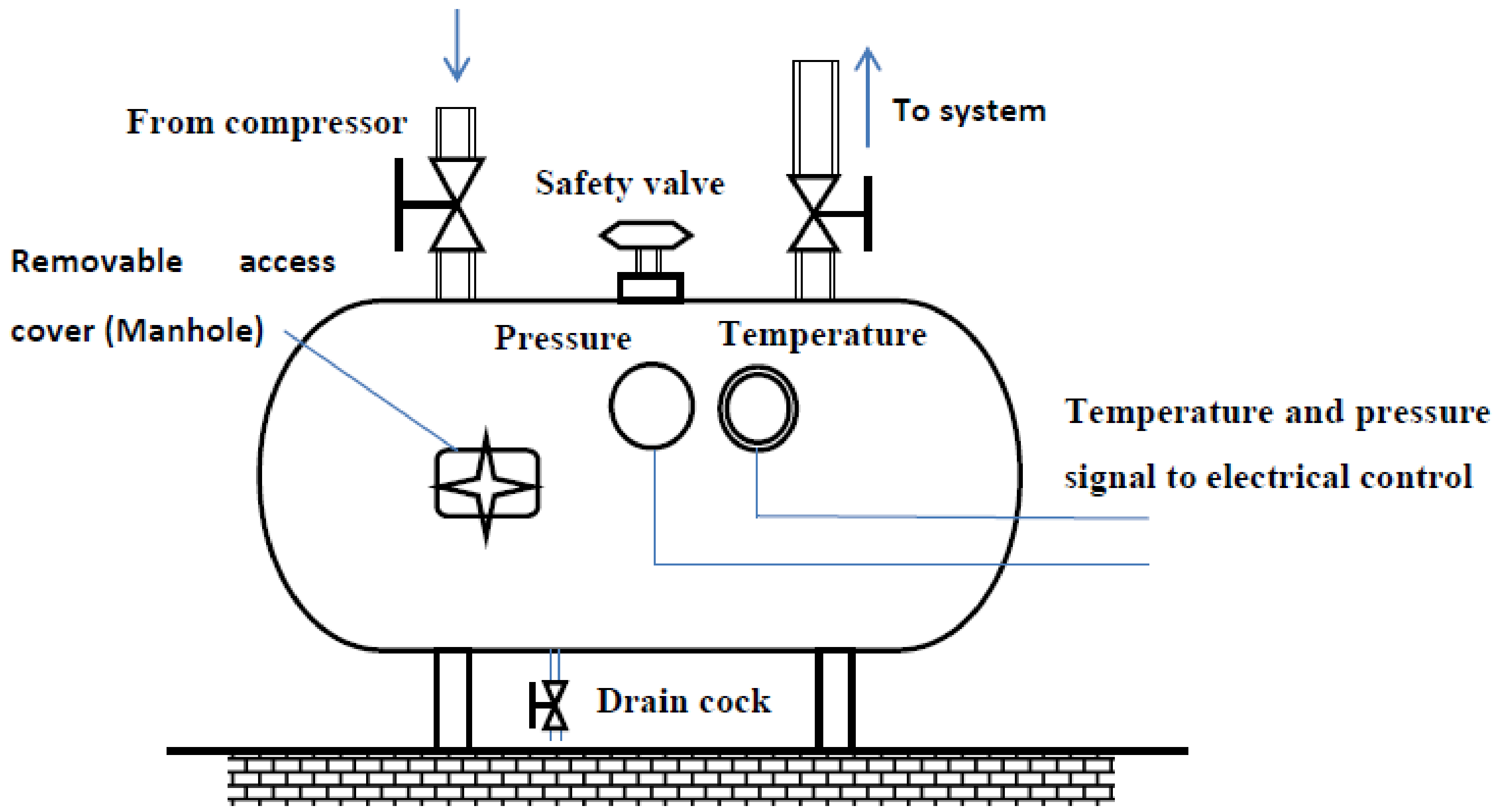


Figure 1.8 Components of Air receiver

$Q_1 and\ Q_2 = volume\ flow\ rate\ of\ air\ at\ the\ compress\ inlet\ and\ out\ let\ \ (m^3/min)$

$p_1 and\ p_2 = absoulte\ pressure\ of\ air\ at\ the\ compressor\ inlet\ and\ outlest\ \ (kPa(abs)$

$T_1 and\ T_2 = Absolute\ temperature\ of\ air\ at\ the\ compressor\ \ at\ inlet\ and\ out\ let\ \ (k)$

Using general gas law

$$Q_1 = Q_2 \left(\frac{p_2}{p_1}\right)\left(\frac{T_1}{T_2}\right)$$

Case 2 : Air is supplied to the receiver during the time interval in which air is being draw off

Then, The air receiver size can be determined by using the following empirical equation

$$V_r = \frac{101\,t\,[Q_r - Q_c]}{[p_{max} - p_{min}]}$$

V_r = receiver size (m^3)

t = time that receiver can supply required amount of air , (min)

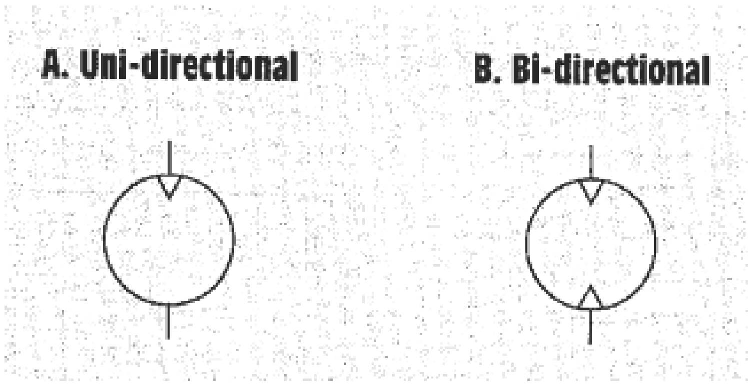
Q_r = consumption rate of pneumatic system (standard m^3/min)

Q_c = outflow rate of pneumatic system (standard m^3/min)

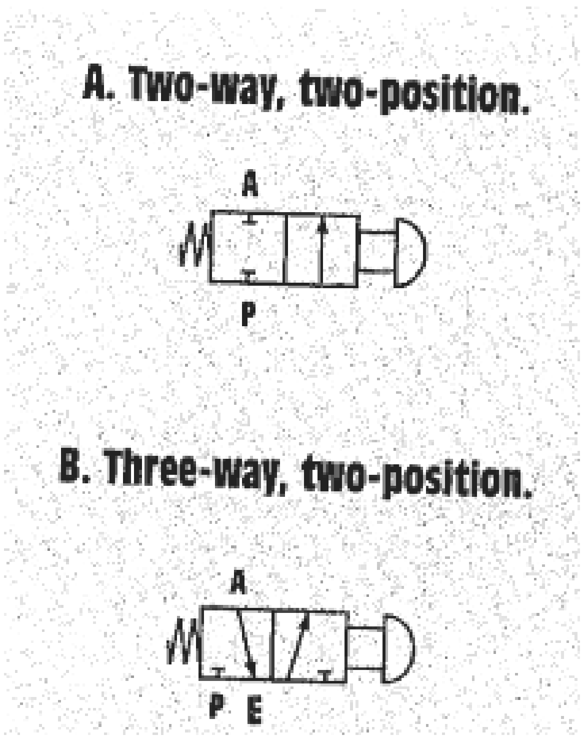
p_{max} = maximum pressure level in receiver (kPa)

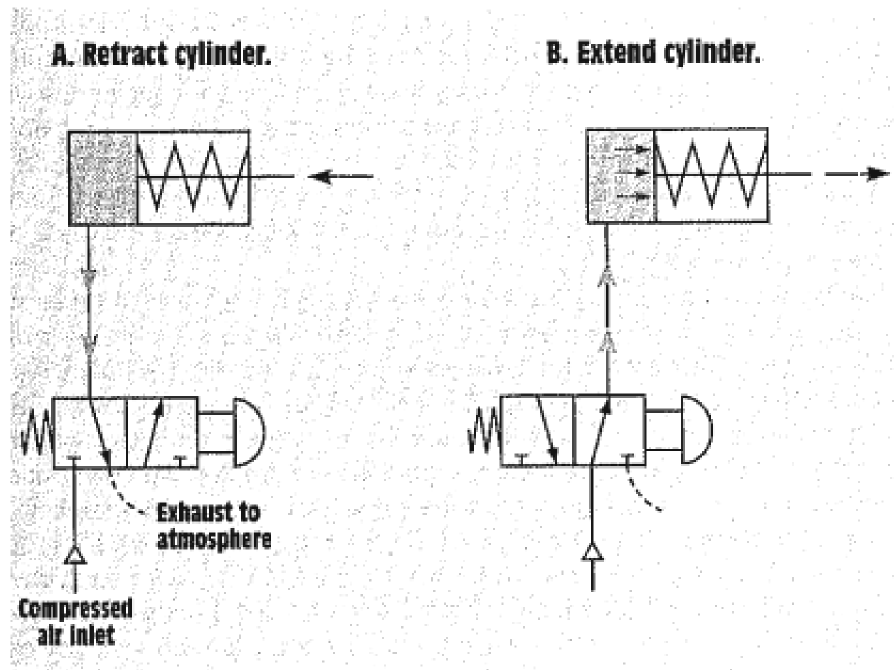
p_{min} = maximum pressure level in receiver (kPa)

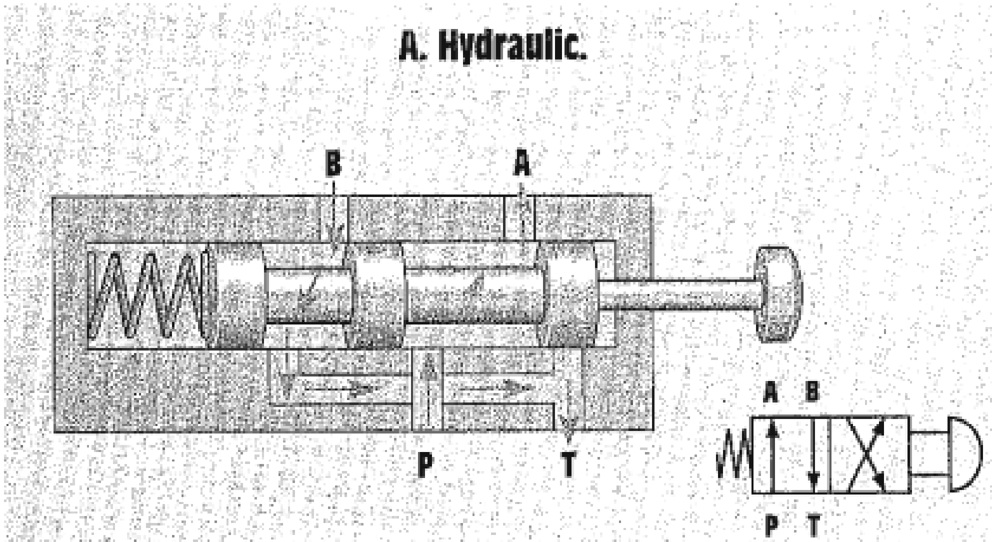
Pneumatic motor Graphic symbol



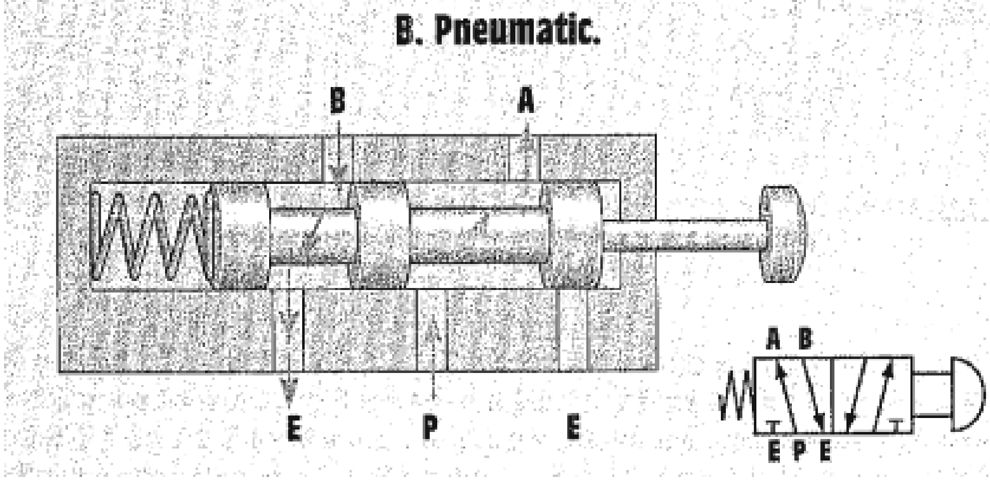
Pneumatic DCV Graphic symbol

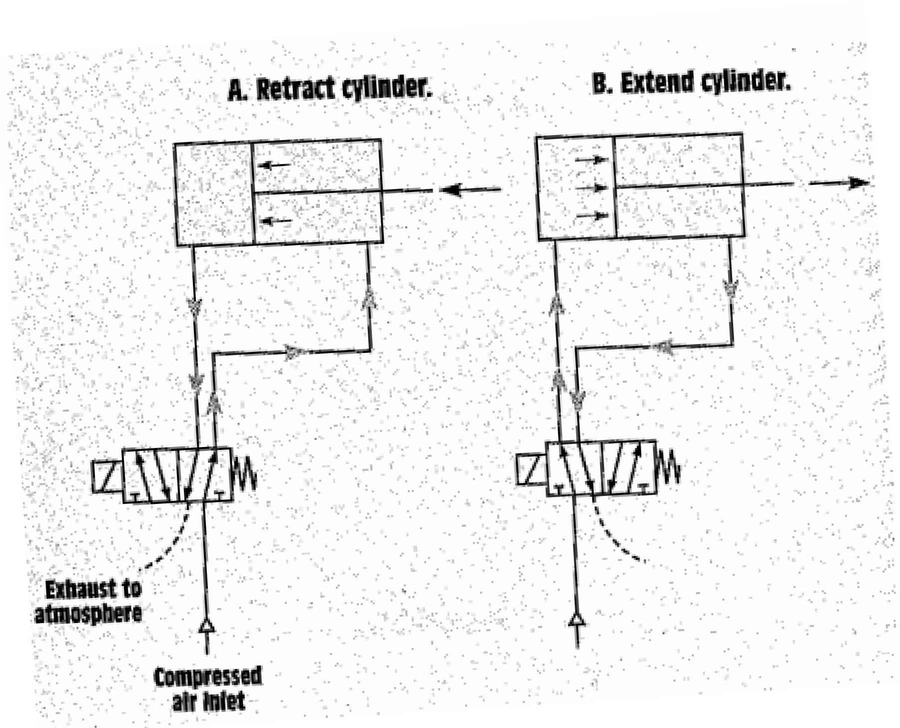




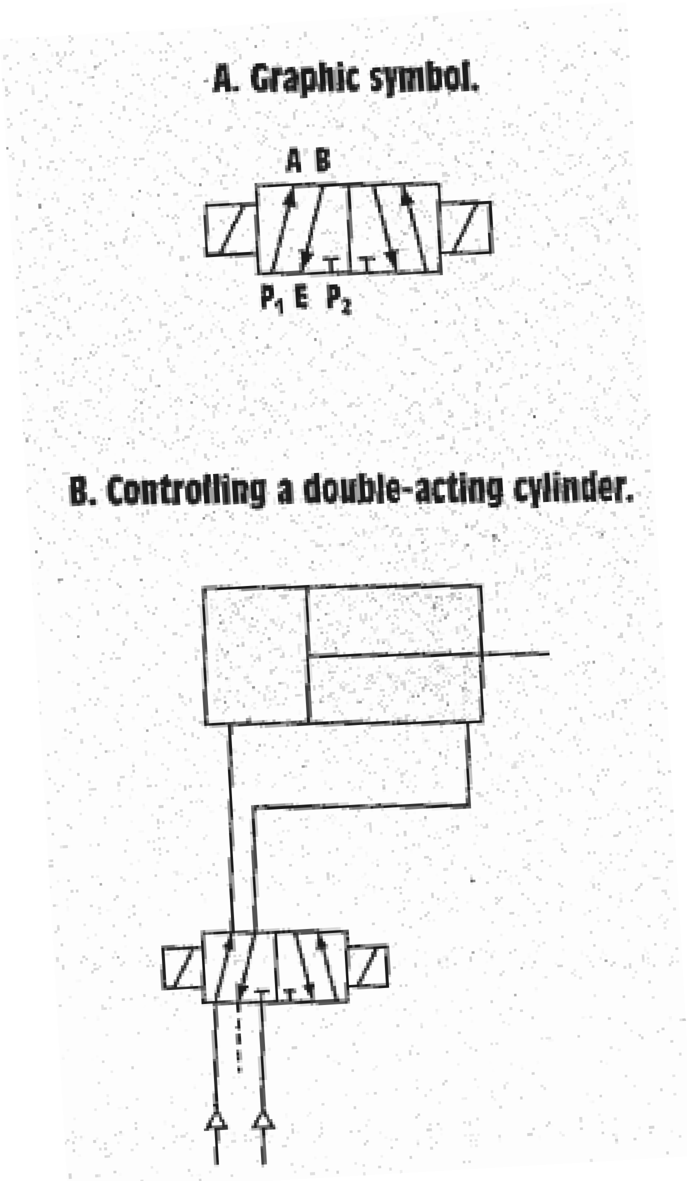


Pneumatic DCV Graphic symbol

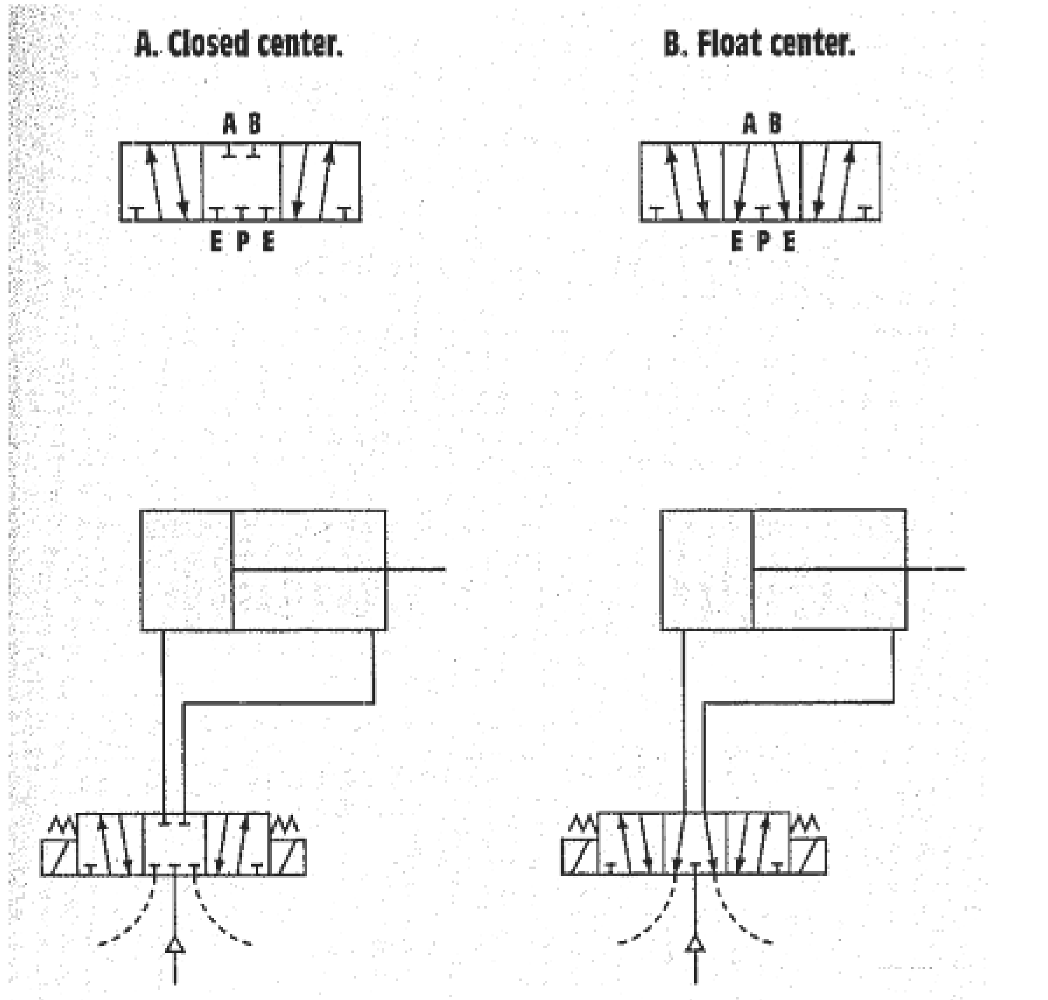




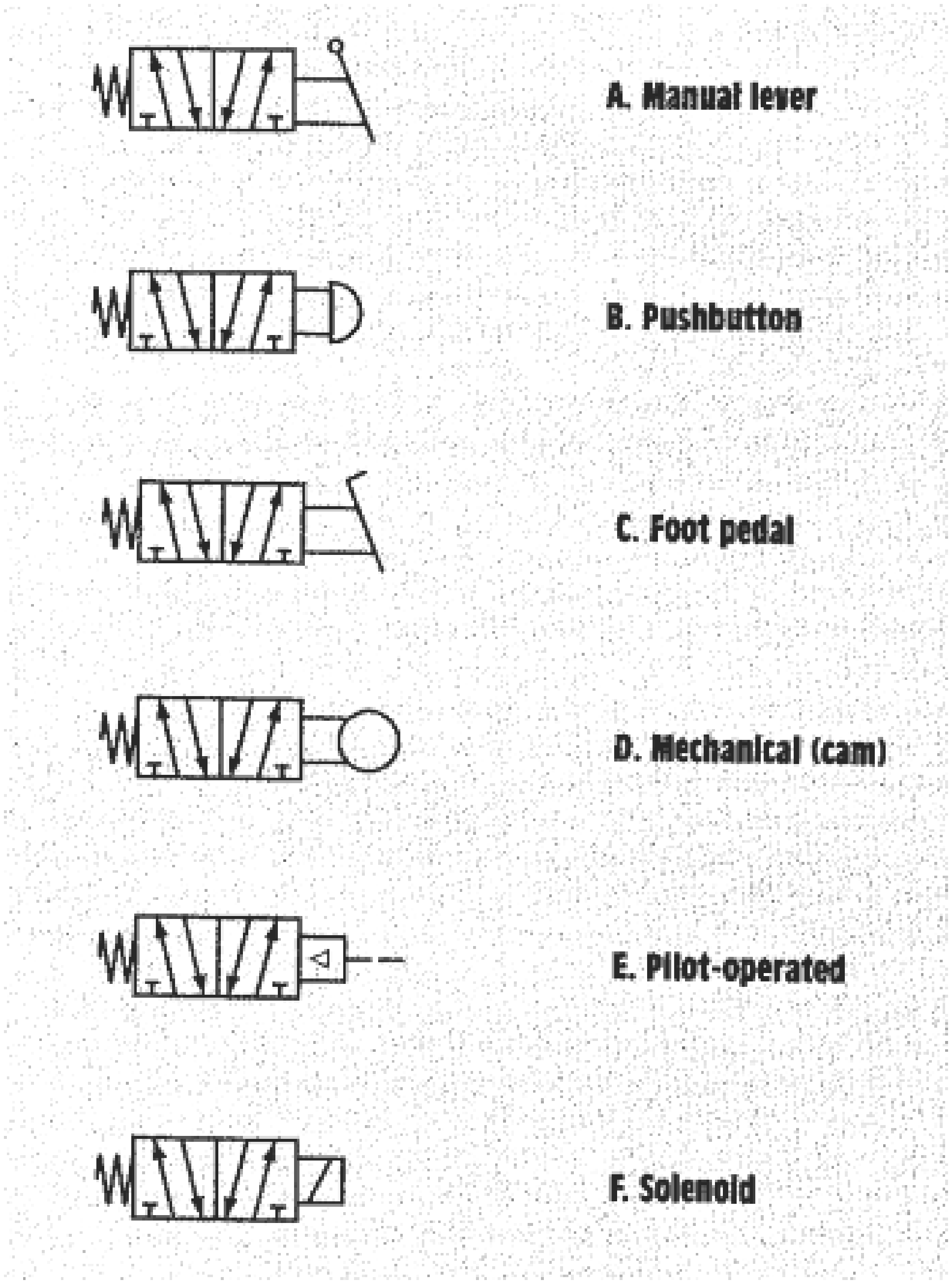
5/3 DCV



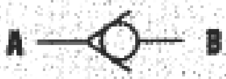
Standard DCV neutral position



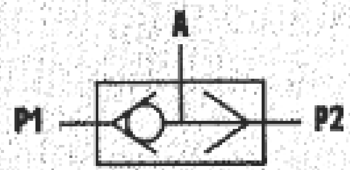
Actuation types



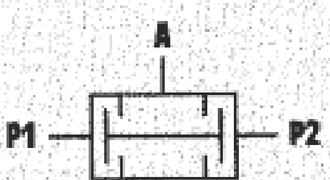
DCV's



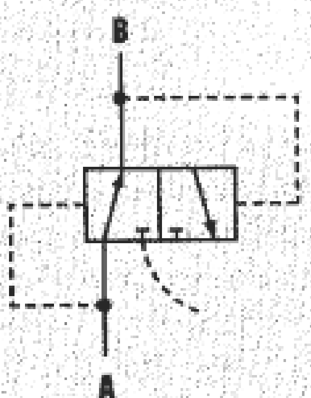
A. Check valve.



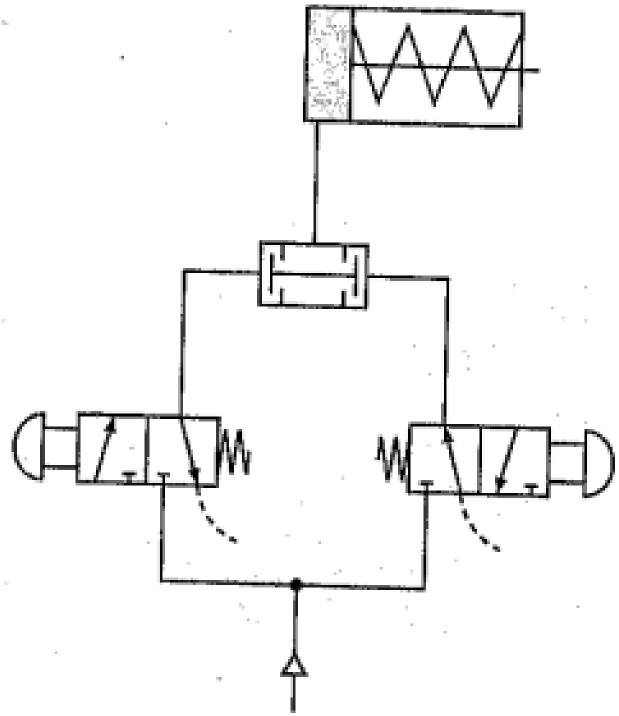
B. Shuttle (OR) valve.



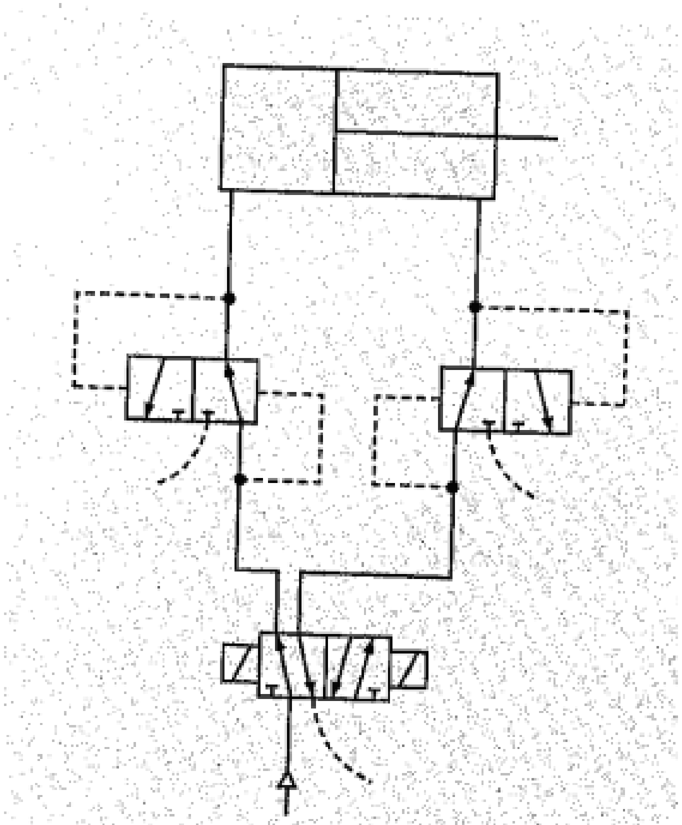
C. AND valve.



D. Quick exhaust valve.

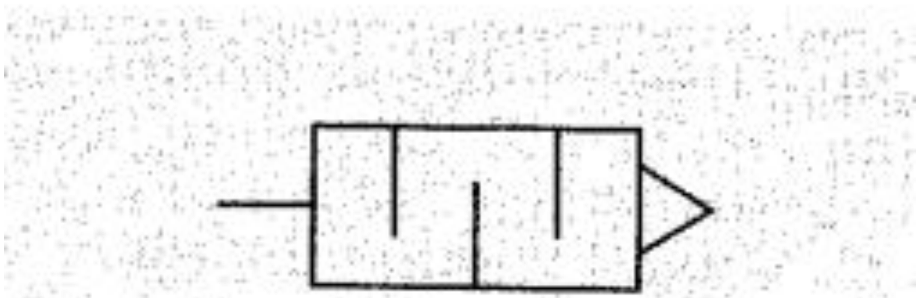


AND Valve

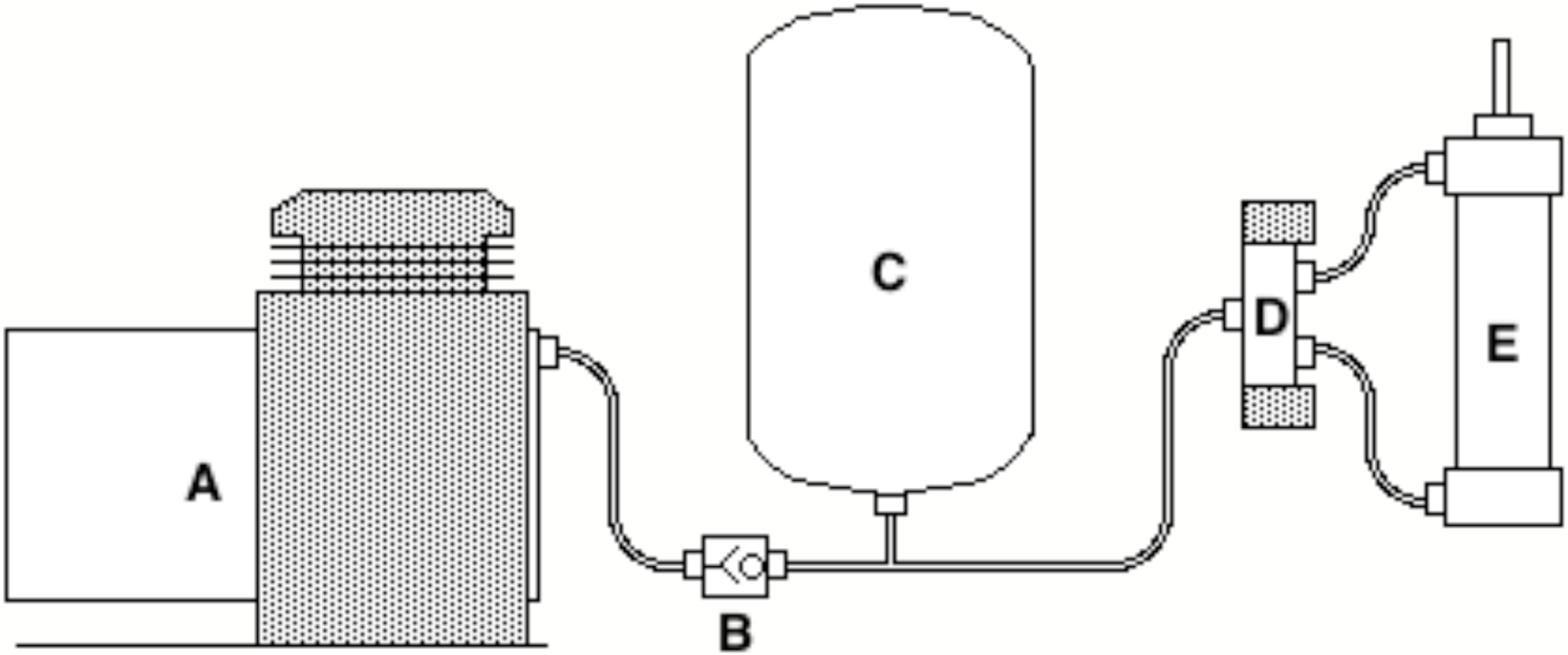


Quick Exhaust Valve

Pneumatic Silencer



Simple Pneumatic system

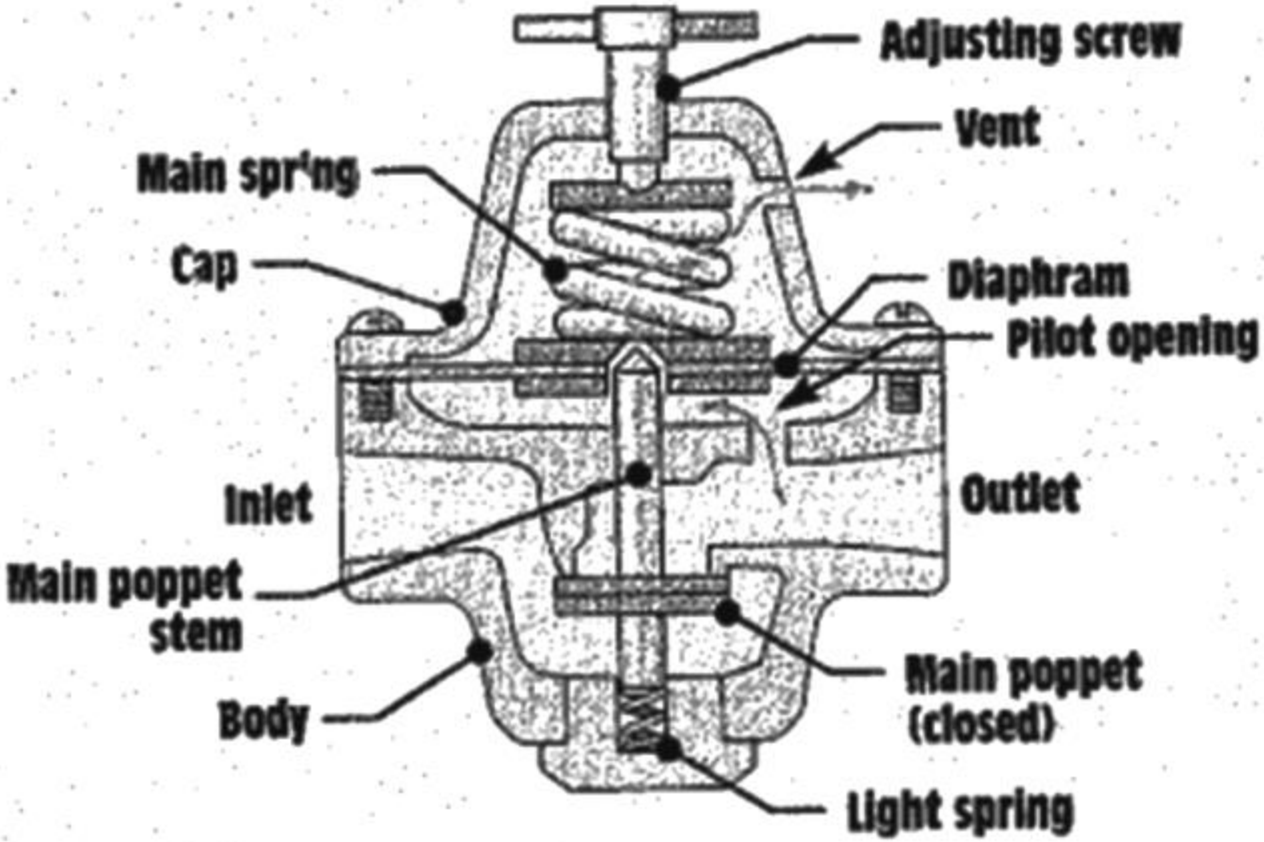


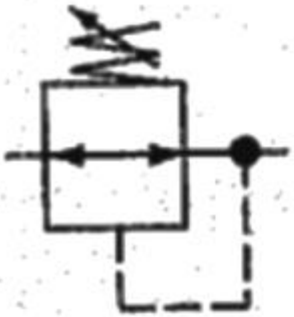
A – Compressor B – Check valve C – Accumulator

D – Directional valve E – Actuator

Air Preparation

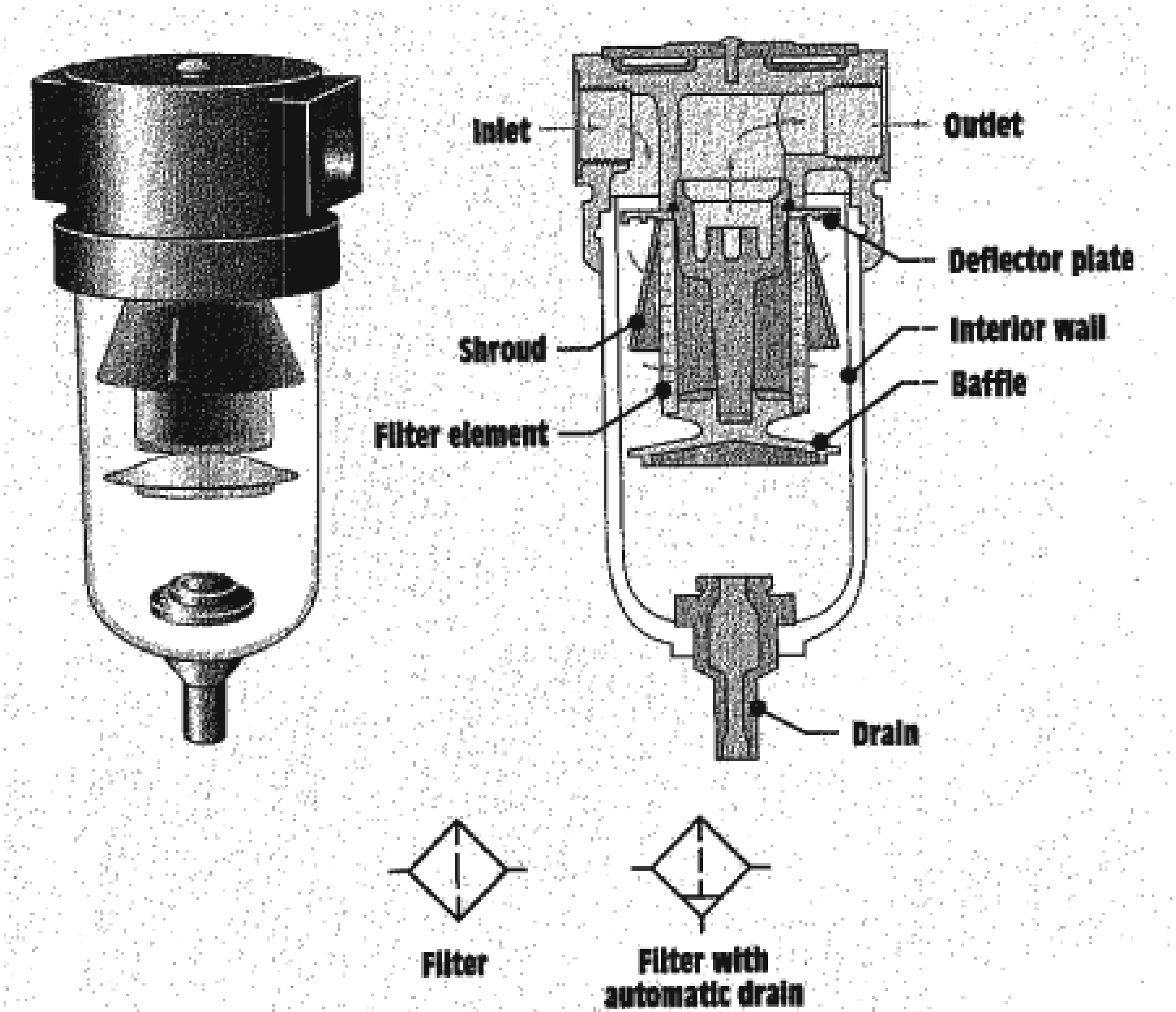
(i) Pressure Regulator





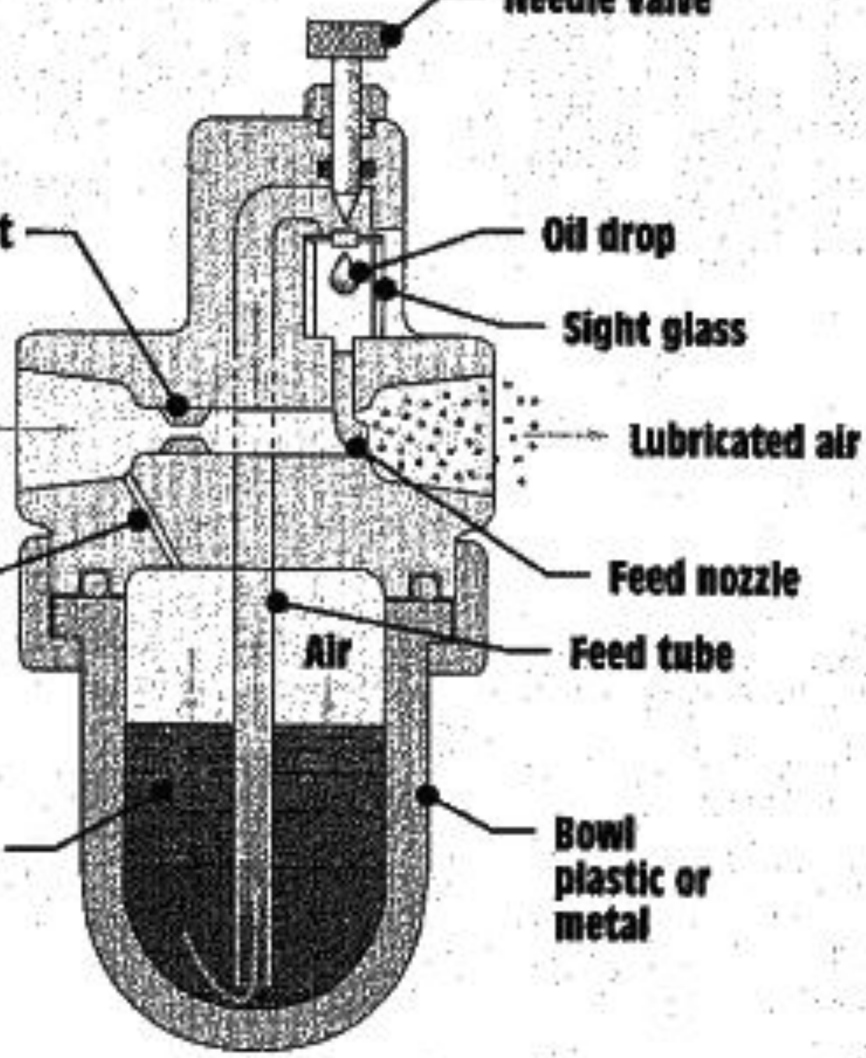
Air Preparation

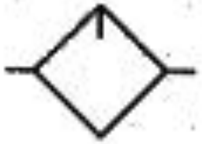
(ii) Filters



Air Preparation

(iii) Lubricator

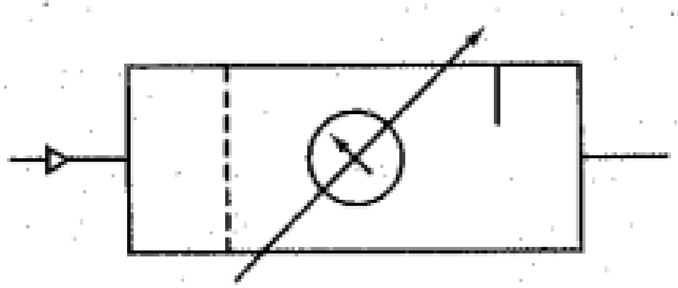




Symbol

Air Preparation

Complete FRL unit



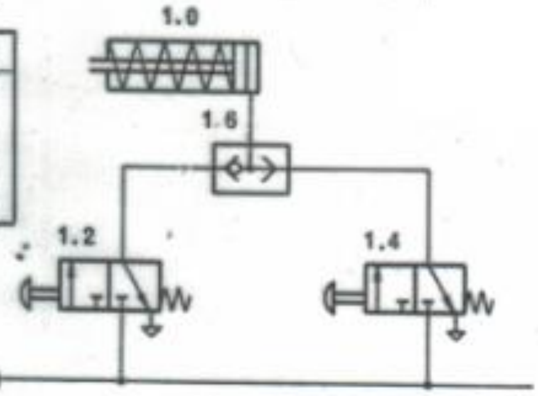
Pneumatic Circuit Design Consideration

When analyzing or designing a pneumatic circuit, the following four important considerations must be taken into account:

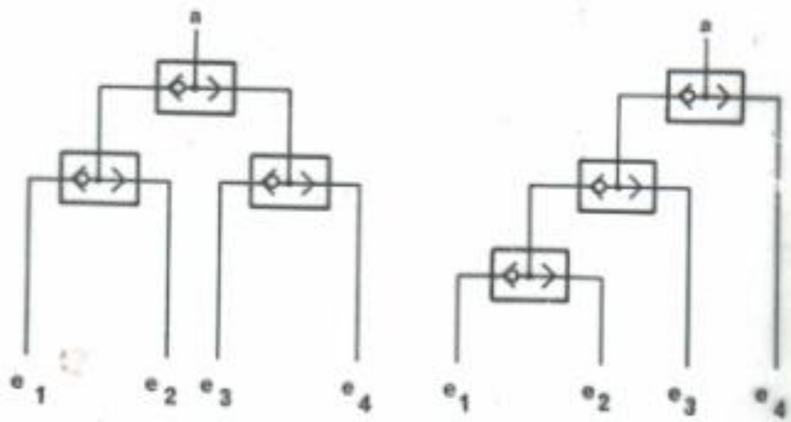
1. Safety of operation
2. Performance of desired function
3. Efficiency of operation
4. Costs

The truth table for two input OR gate is

A	B	Y
0	0	0
1	0	1
0	1	1
1	1	1

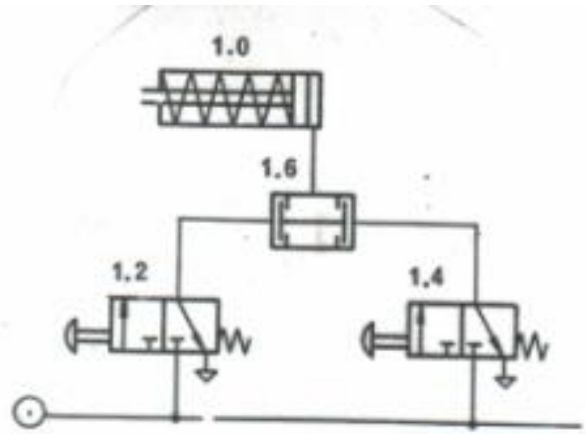


Shuttle Valve in Series



Required no. of valves (n_v) for one outlet a $= e_n - 1$

where e_n - no. of Signals to be grouped

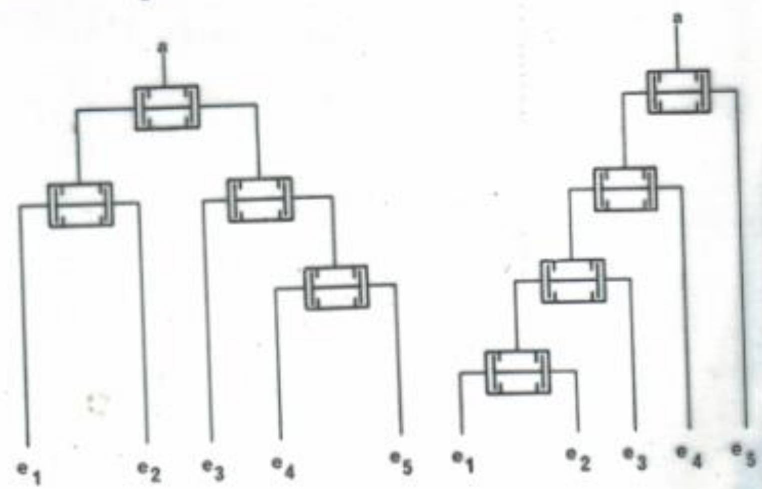


The truth table for a two input AND gate is given below

Truth table		
A	B	Y
0	0	0
1	0	0
0	1	0
1	1	1

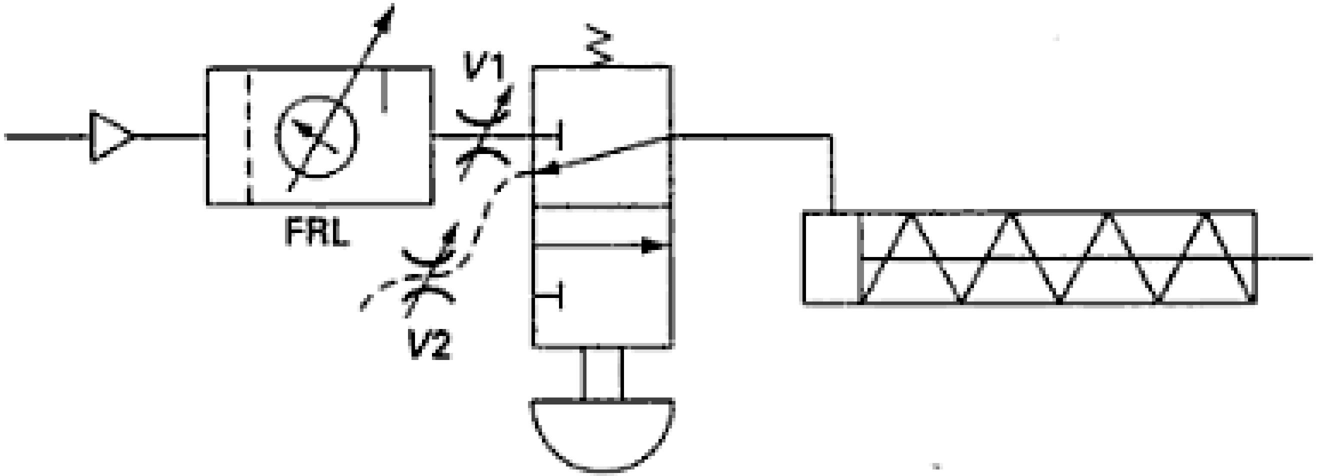
* Signal is given at the outlet only if both inlet signals are present

Possible arrangements of the AND valve.

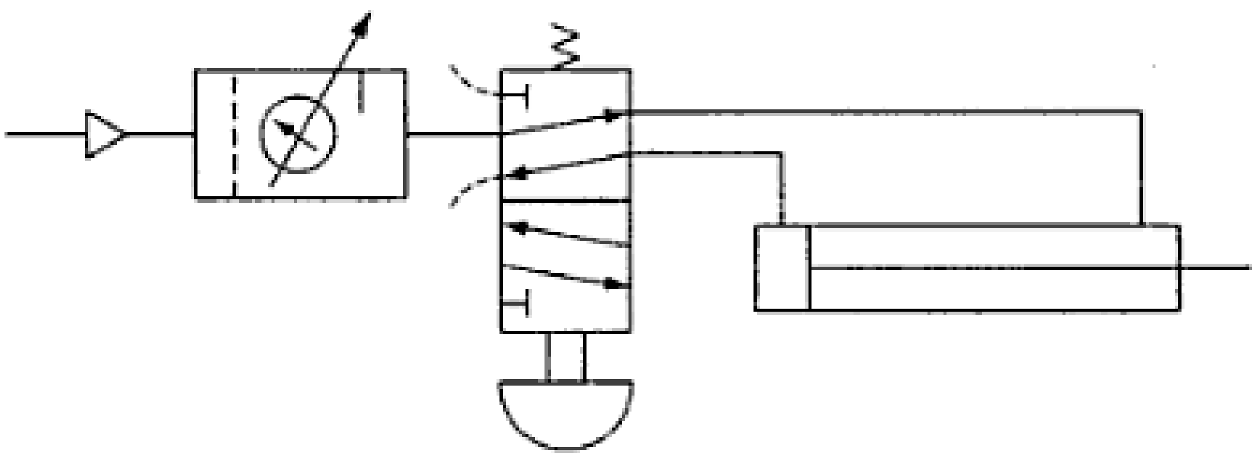


* $n_v = e_n - 1$

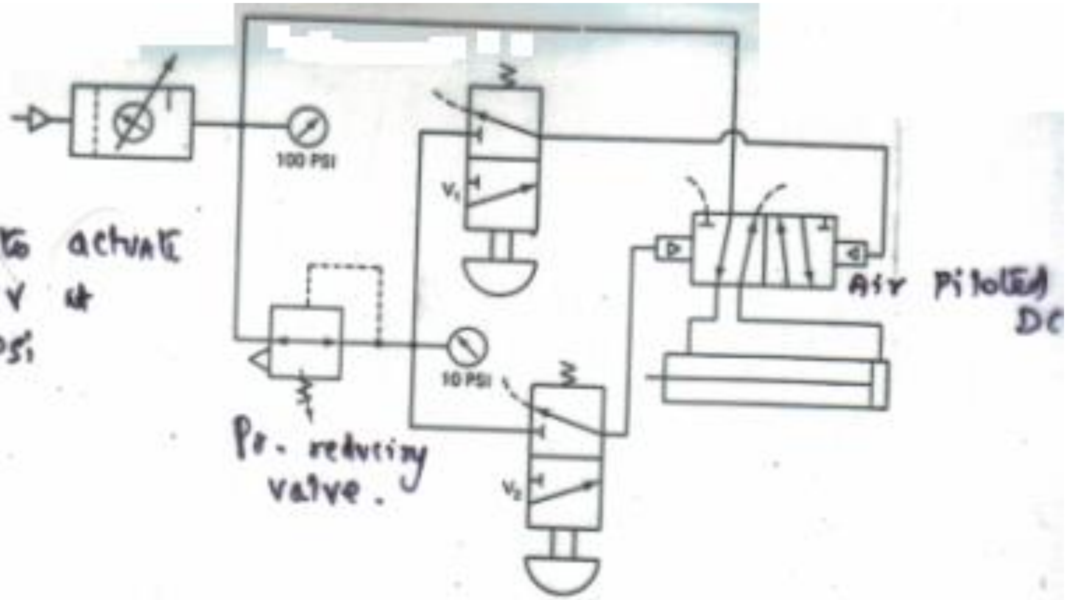
Single acting cylinder



Double acting cylinder

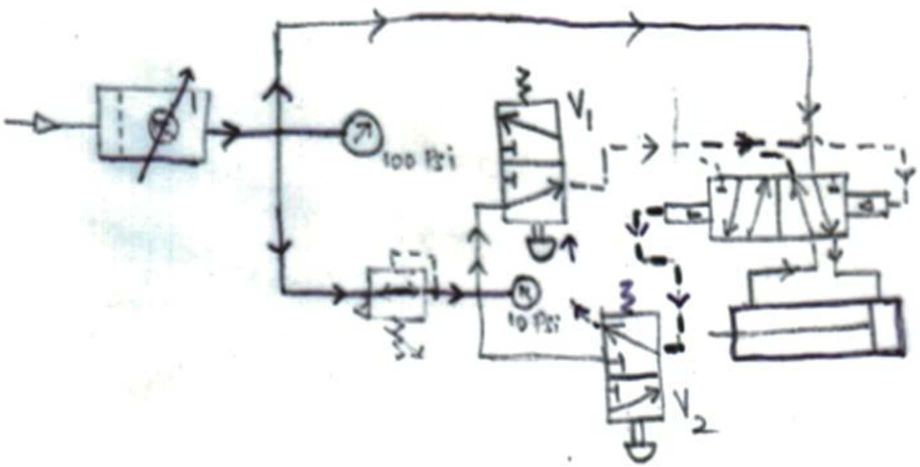


* V_1, V_2 used to actuate the Air-piloted DCV at low pr eg. 10 PSi

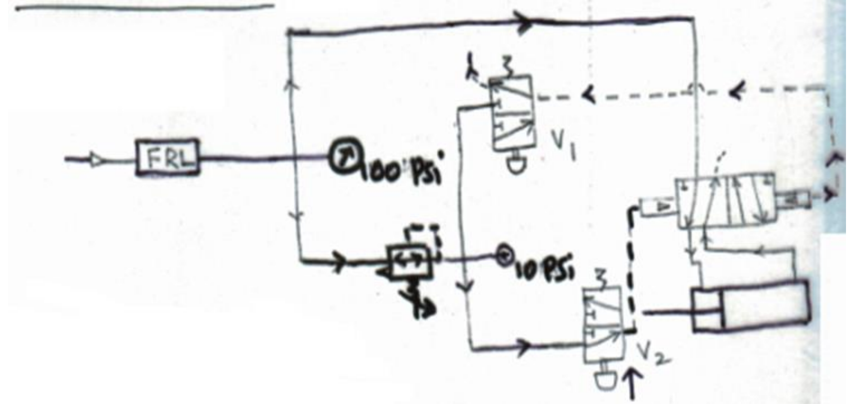


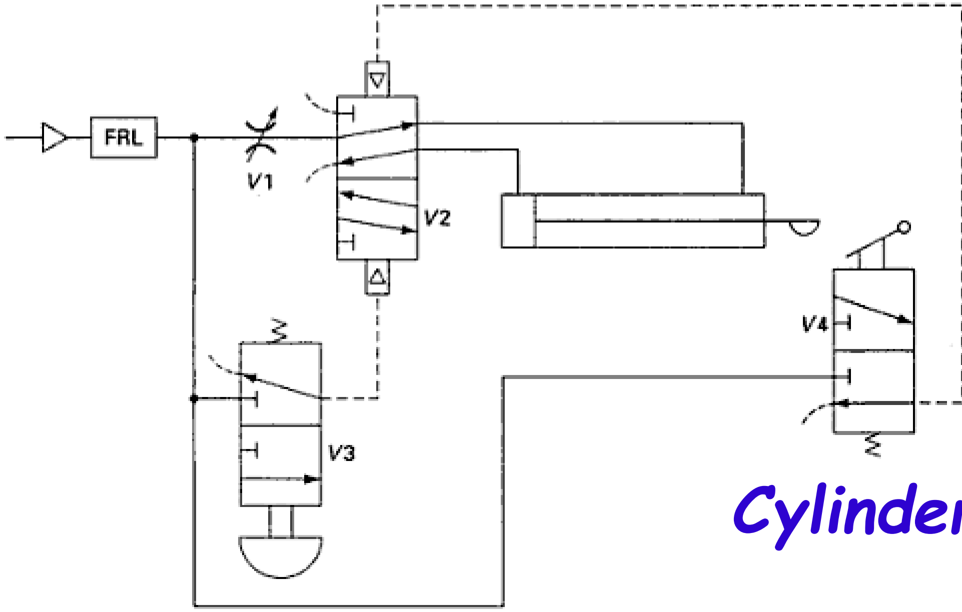
Air pilot control of double acting cylinder

V_1 actuates (Extend)



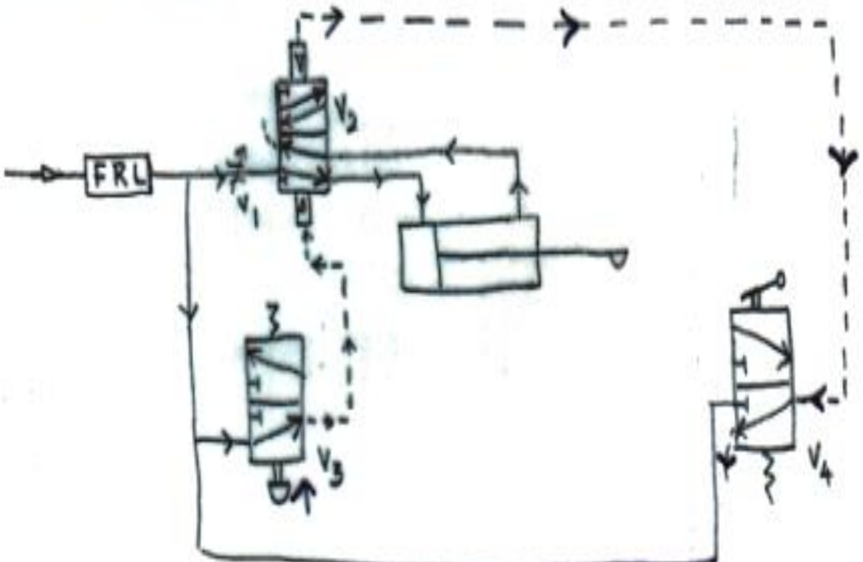
V_2 actuates (Retract)



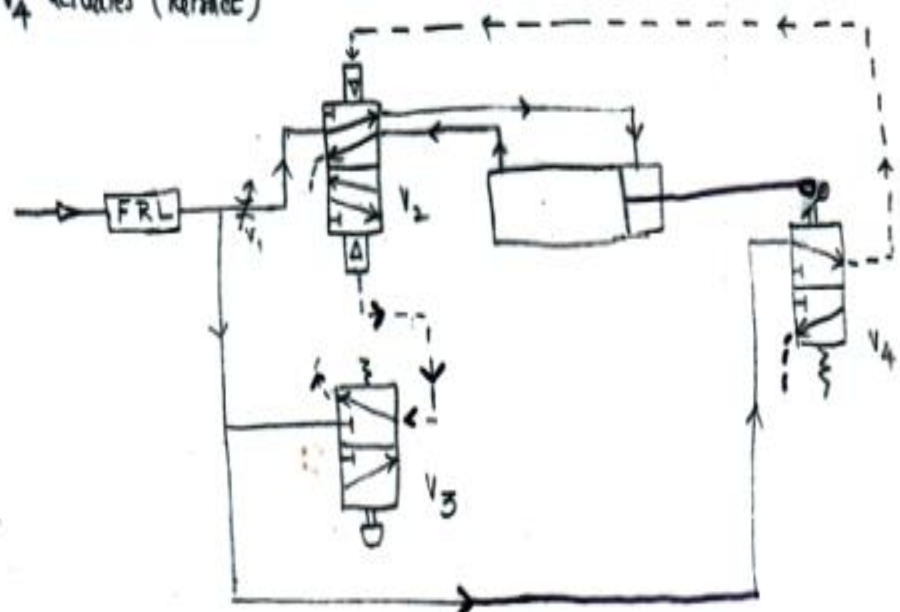


Cylinder Cycle timing system

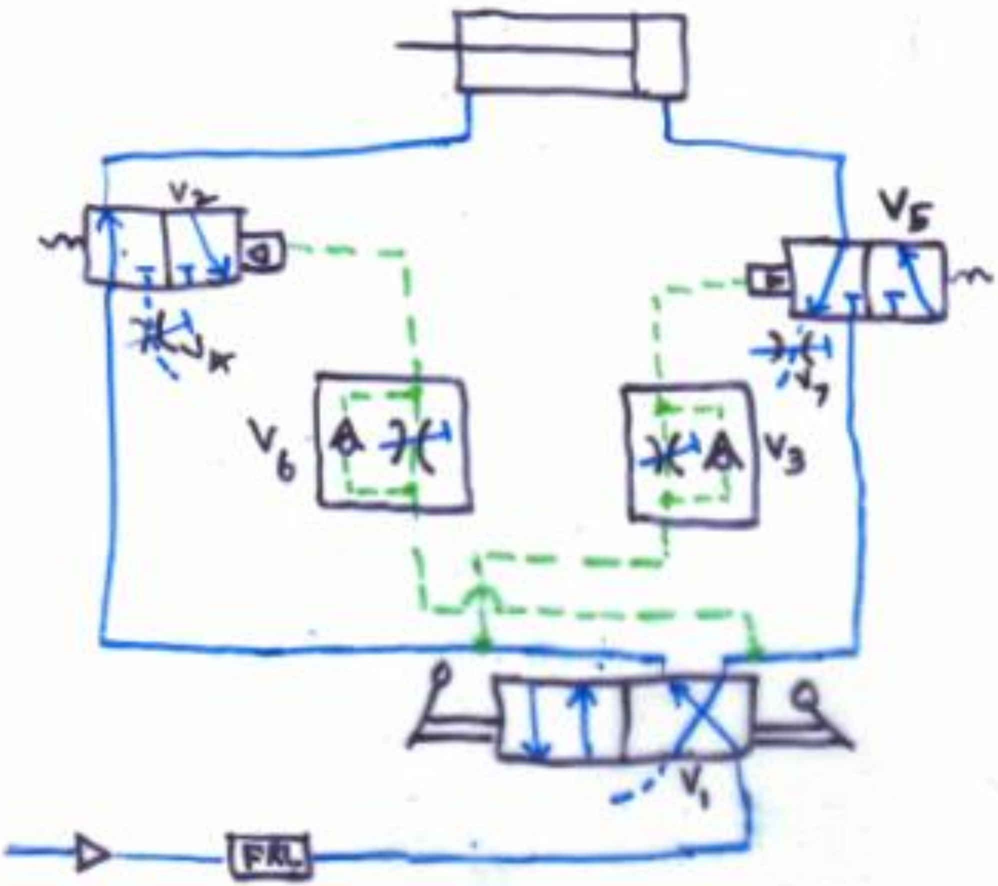
V_3 actuates (Extend)



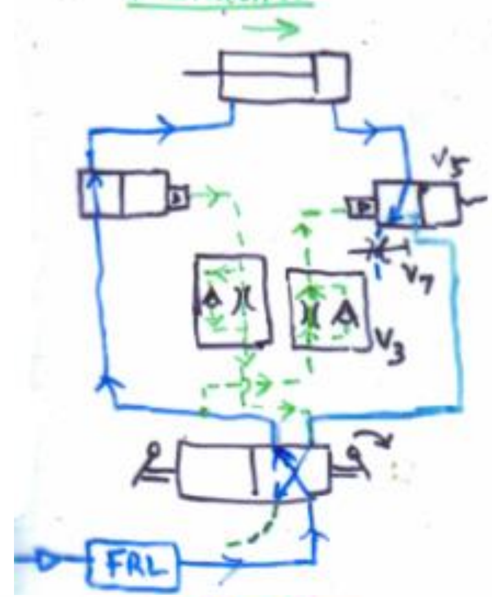
V_4 actuates (Retract)



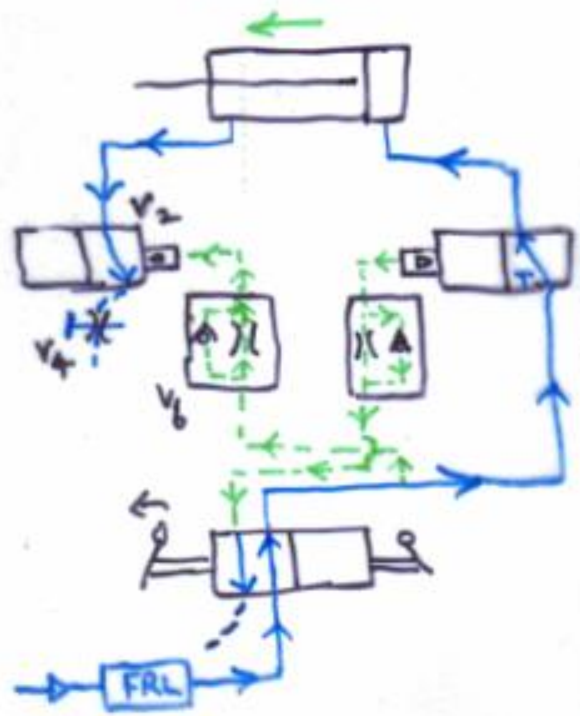
Decelleration air cushion
of a pneumatic circuit



Retraction

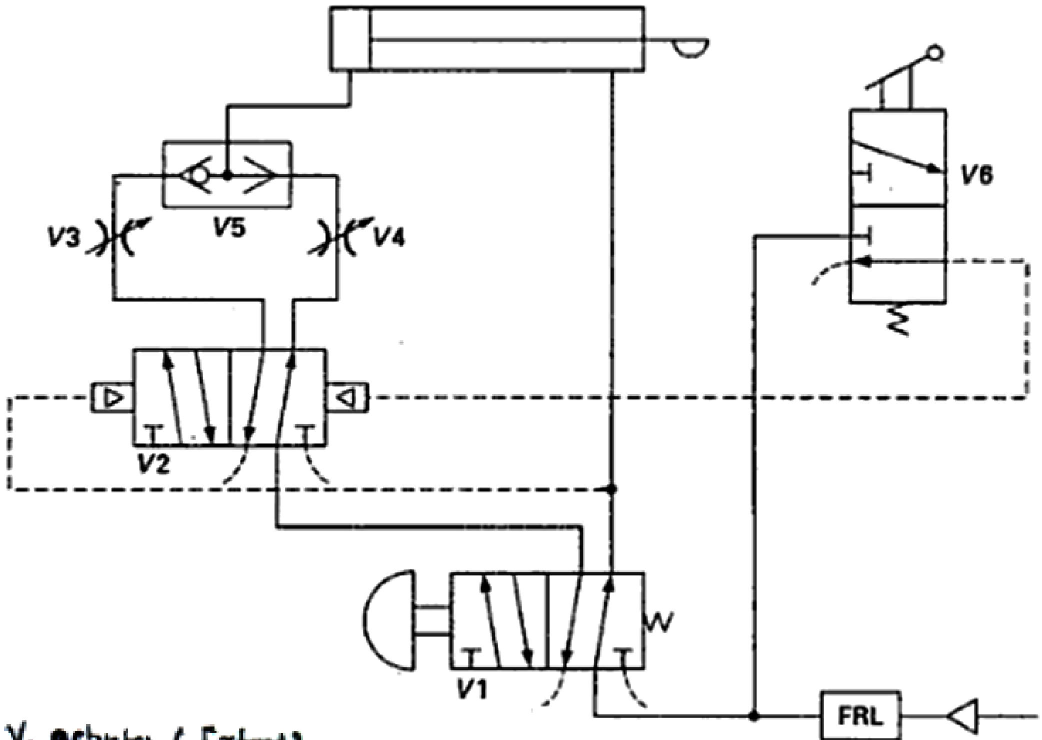


Extension

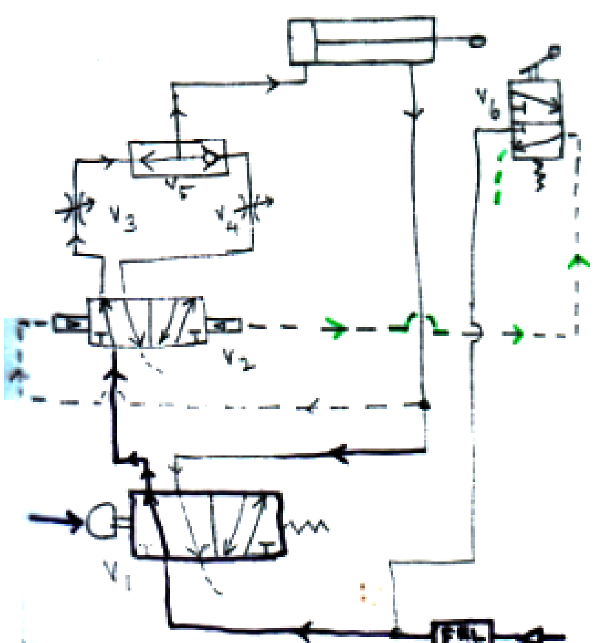


2-step speed control

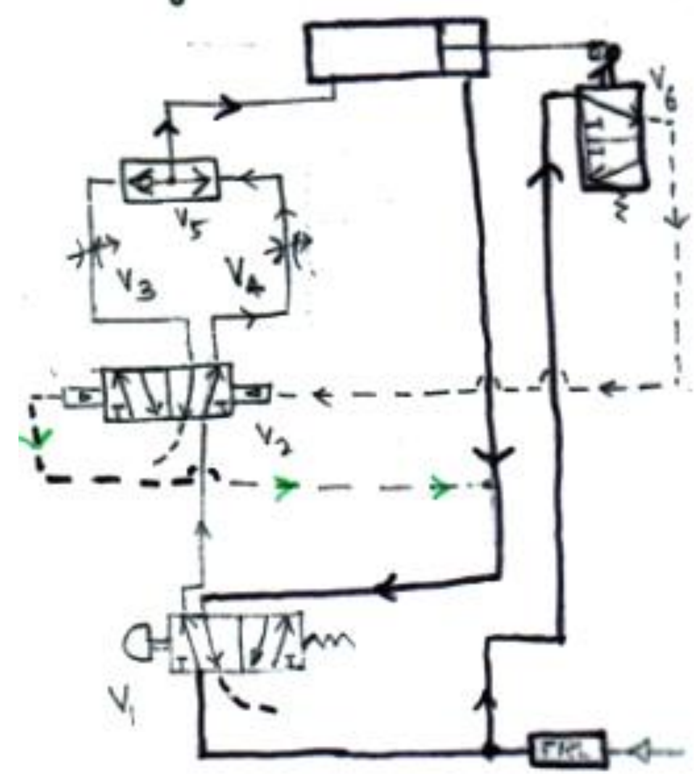
Assume V3 allows more flow rate than V4



V3 actuates (Extend)



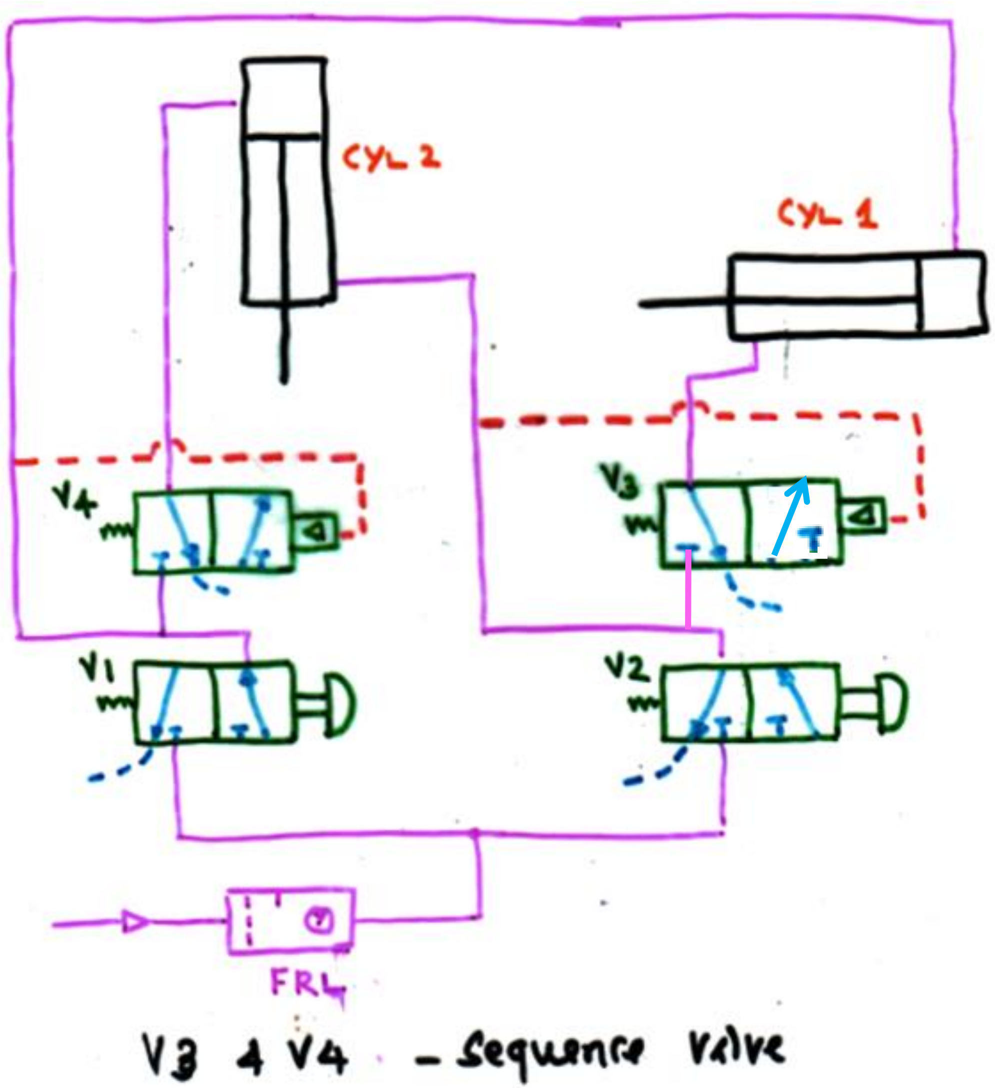
V4 actuates (End of Extension stroke)
Via V6



What happens to the cylinders in each case?

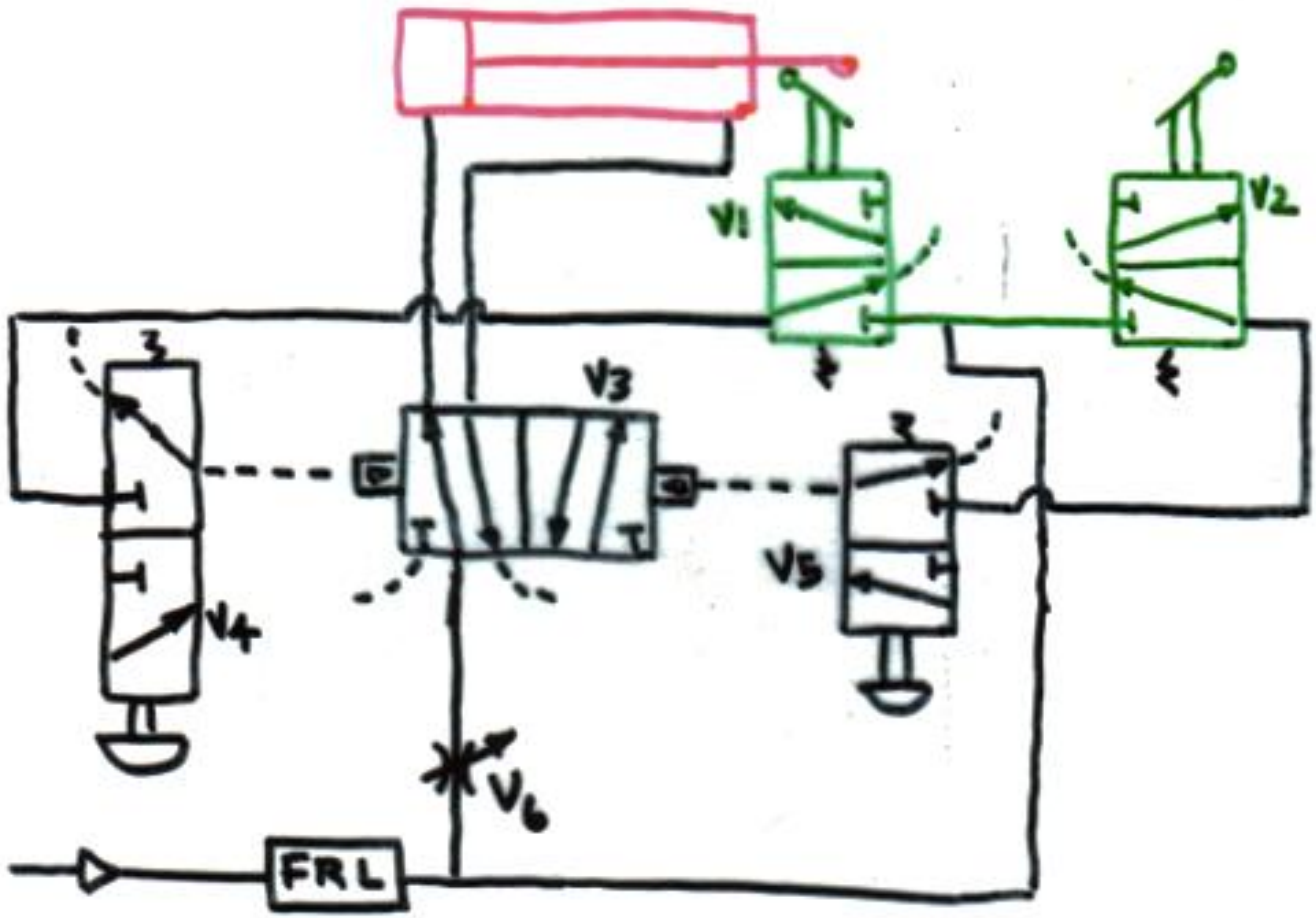
a. **V1 is actuated and held**

b. **V1 is released and V2 is actuated and held**



Consider the circuit

a. What happens to the cylinder when V4 is used?
b. What happens to the cylinder when V5 is used?



PNEUMATIC LOGIC CONTROL SYSTEMS

These systems use logic devices that switch a fluid, from one outlet of the device to another outlet.

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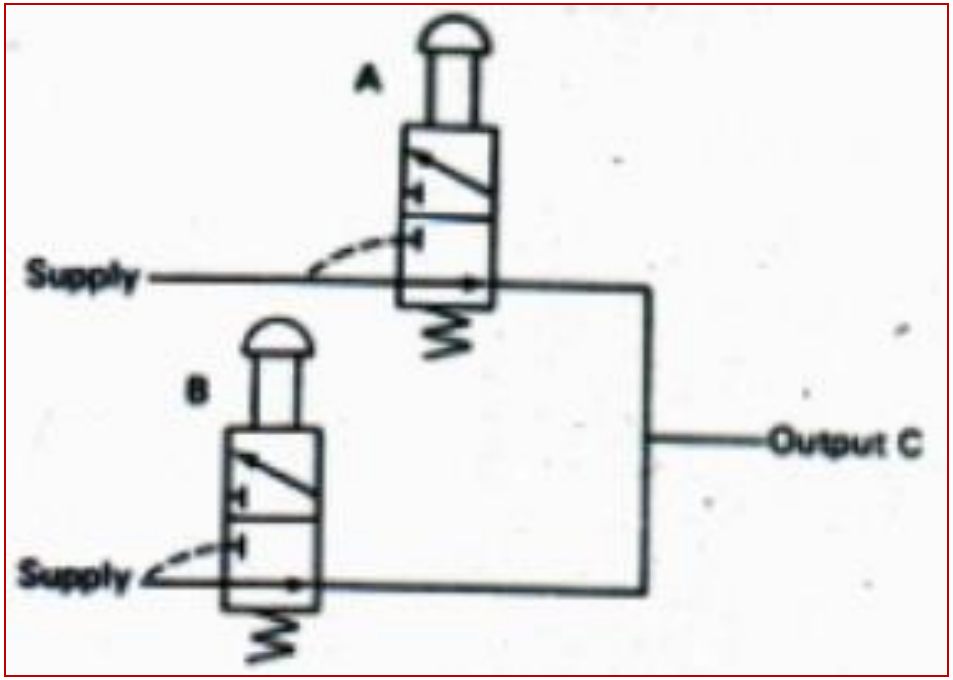
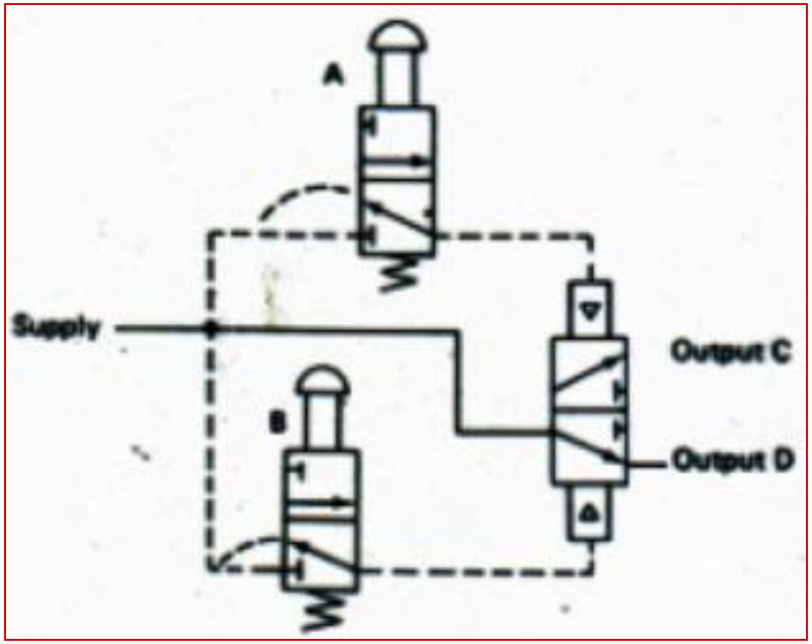
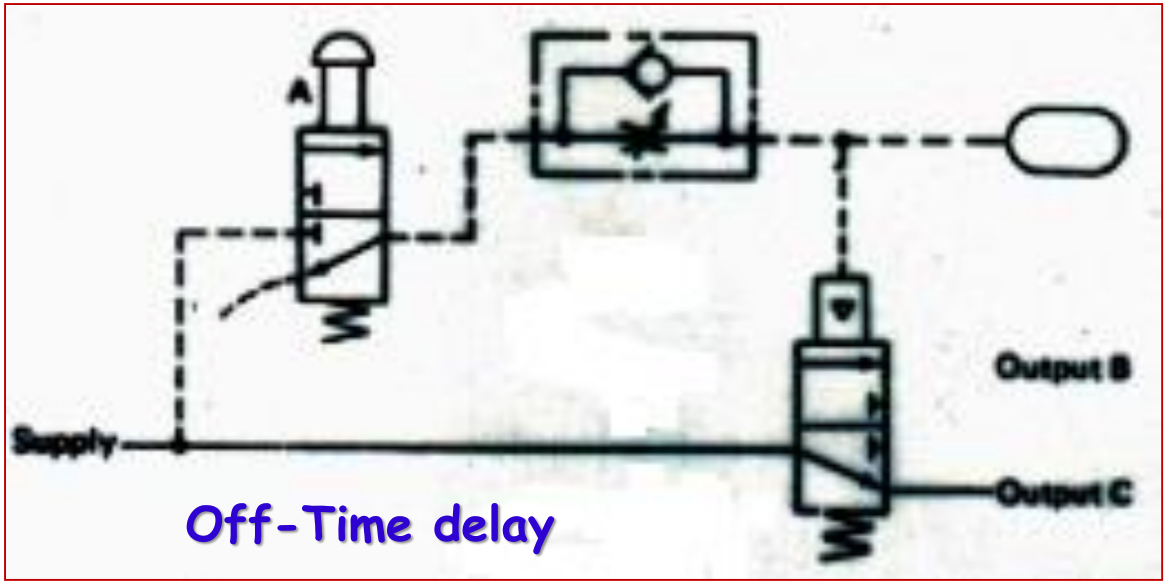


Memory Circuit

Memory is the ability of the circuit to retain a bias

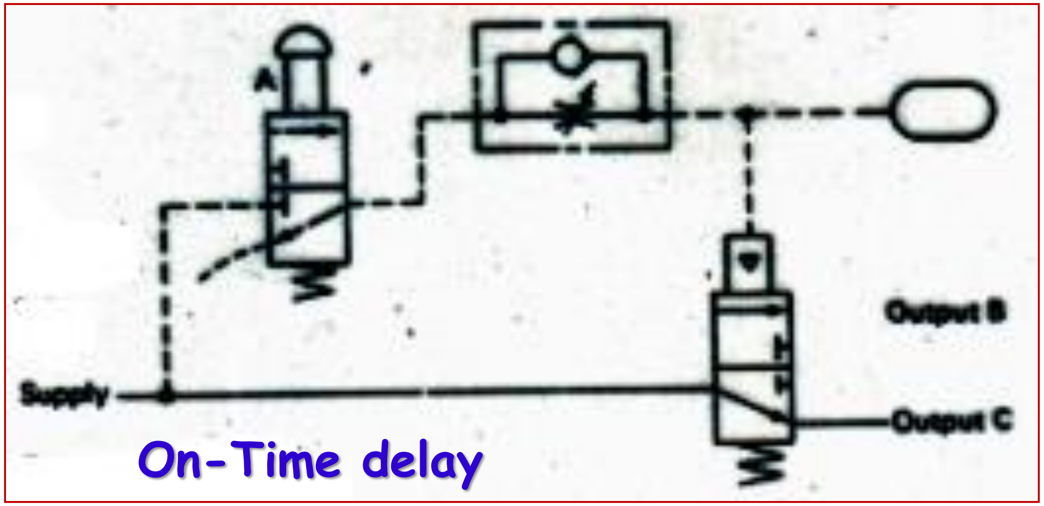


Timing-out Circuit or Limited Memory Circuit



Time functions are introduced to delay operation of the circuit, usually by placing a resistance in the circuit.

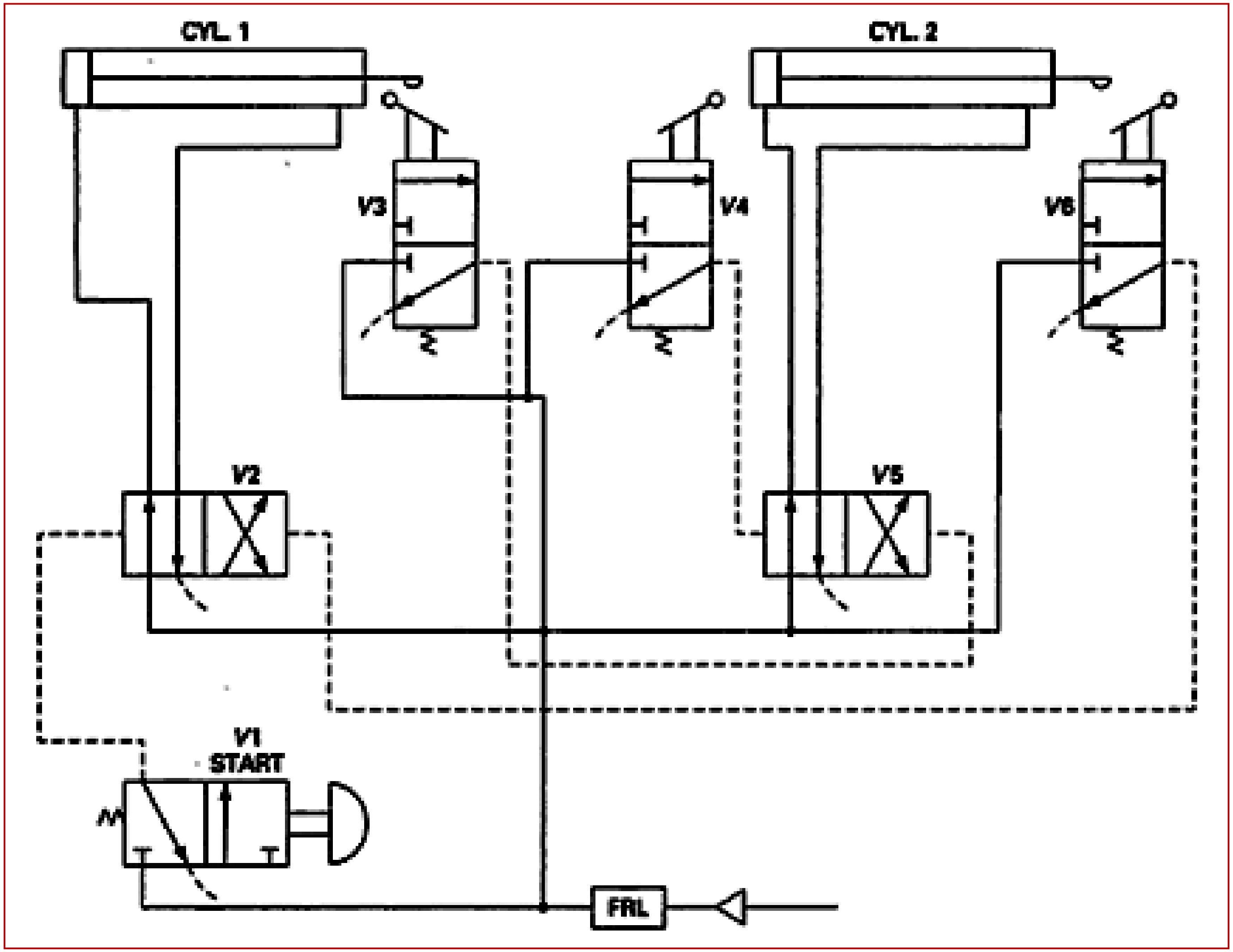
Timing-in Circuit



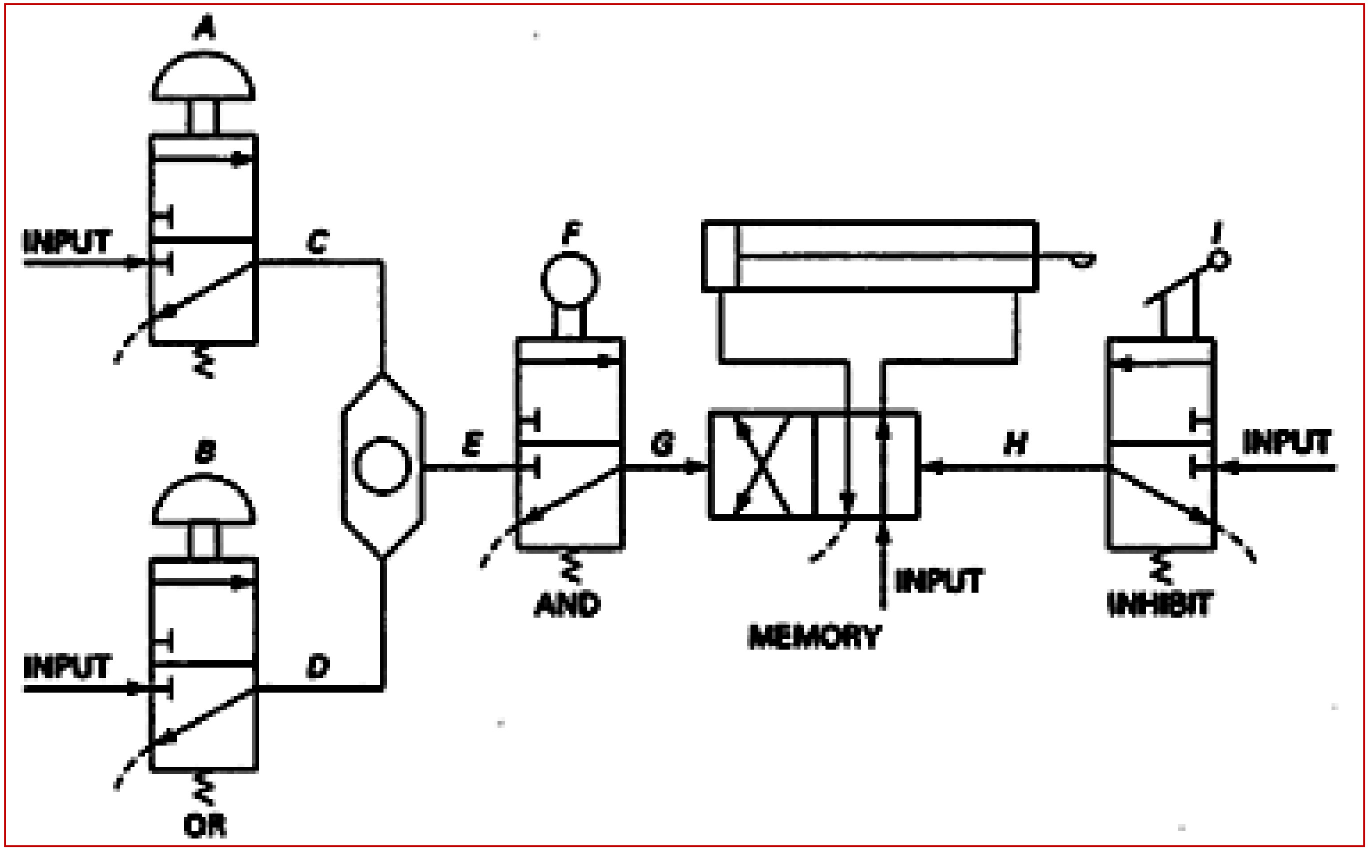
<u>Unlimited MEMORY:</u>
Command signal reverts by manually

<u>Limited MEMORY:</u>
Command signal reverts by automatically

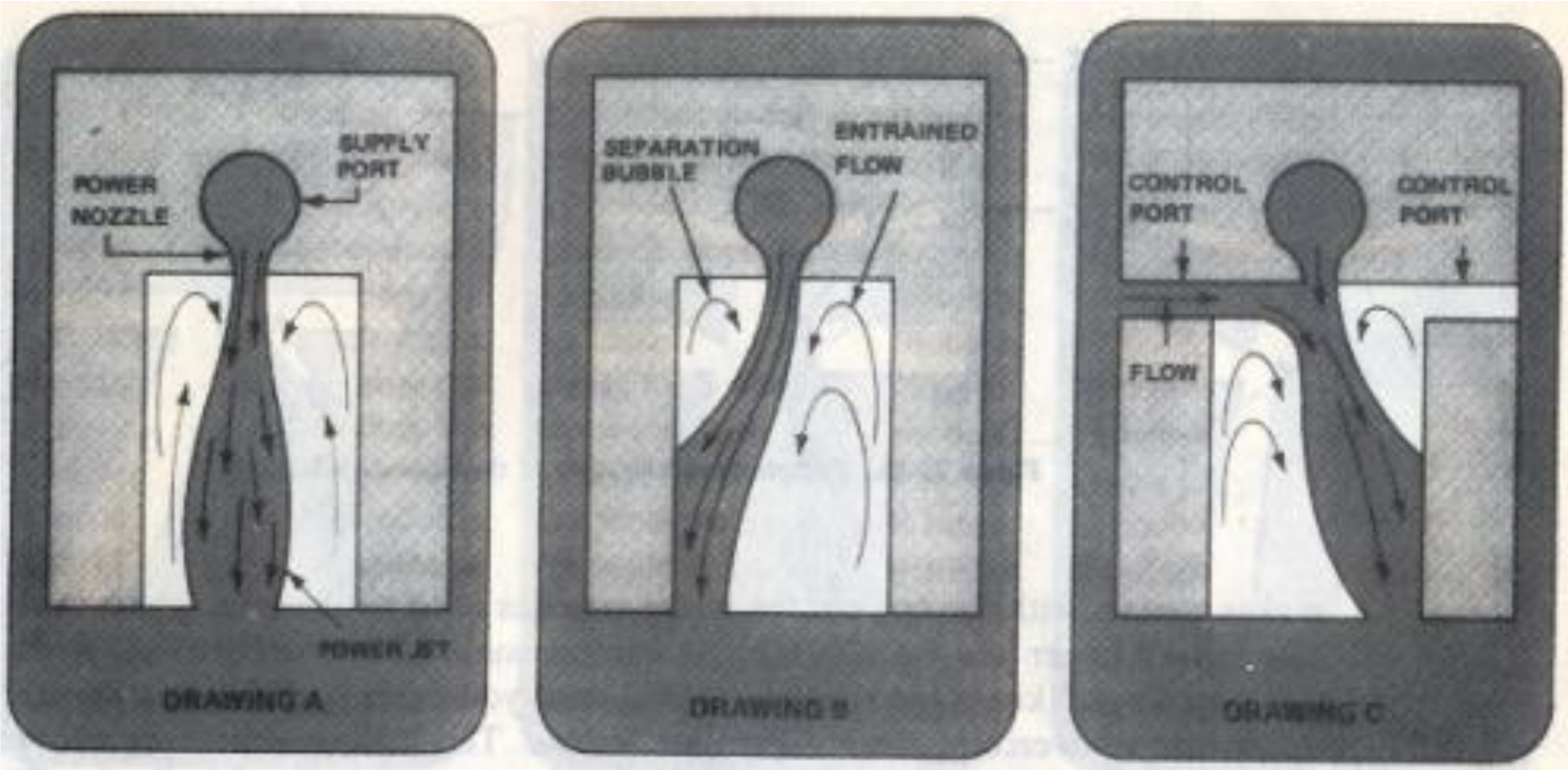
MPL Cylinder sequencing circuit

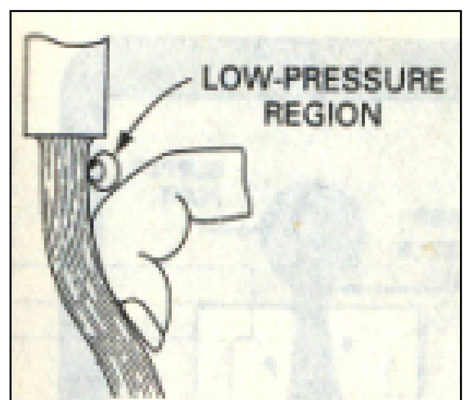


MPL control of a Double acting cylinder

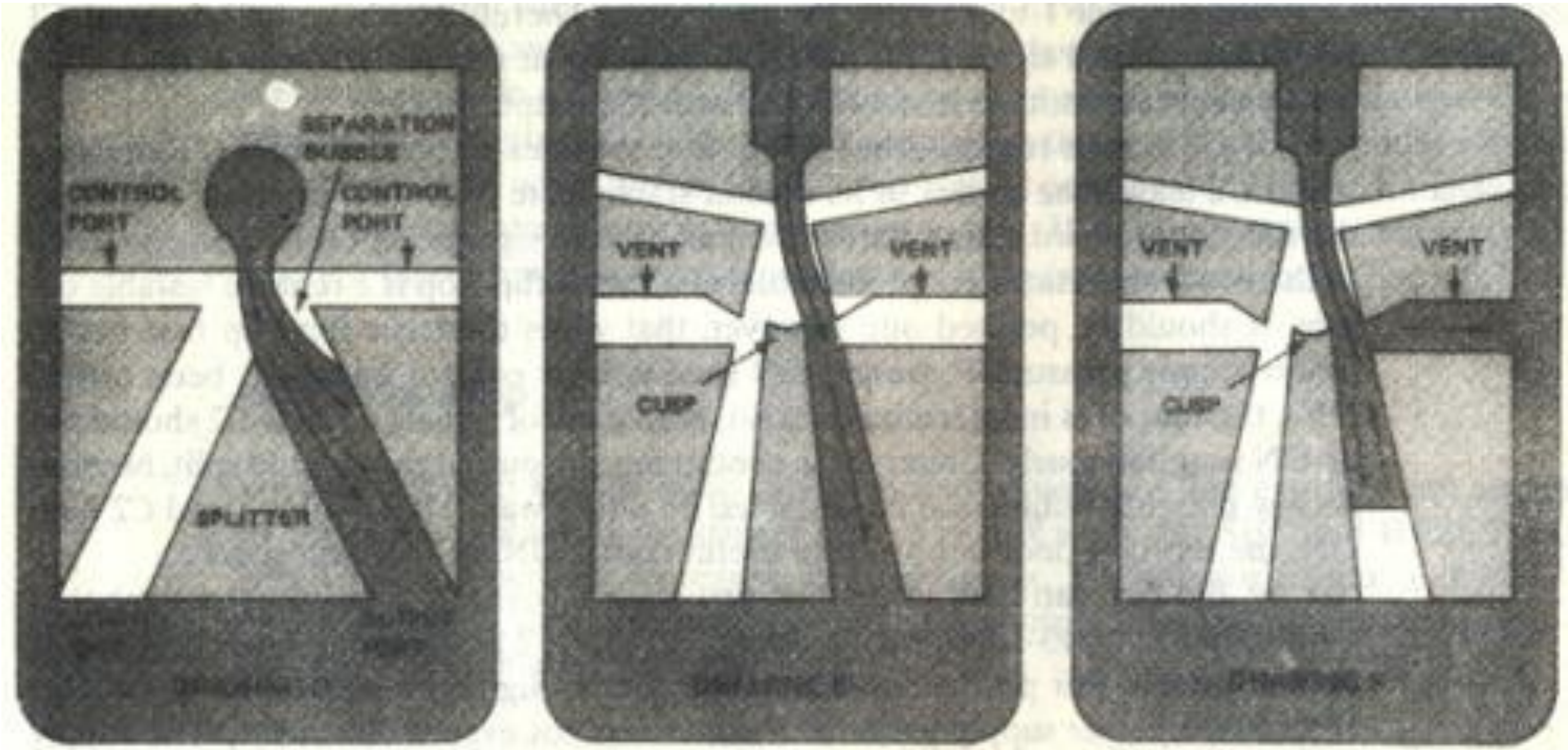


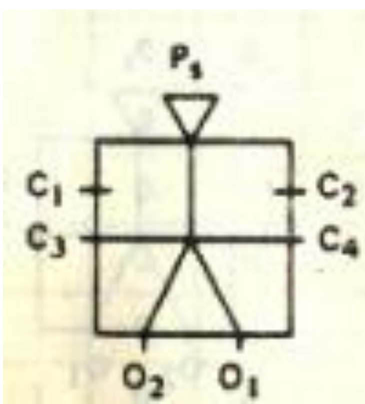
Coanda effect devive





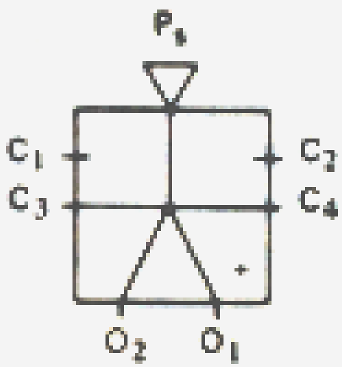
Basic Bistable flip-flop





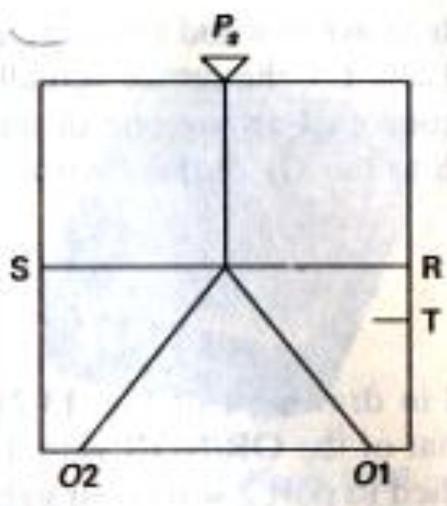
C1	C2	01	02
1	0	1	0
0	0	1	0
0	1	0	1
0	0	0	1

Preferenced flip flop



C1	C2	O1	O2
0	0	1	0
0	1	0	1
0	0	0	1
1	0	1	0
0	0	1	0

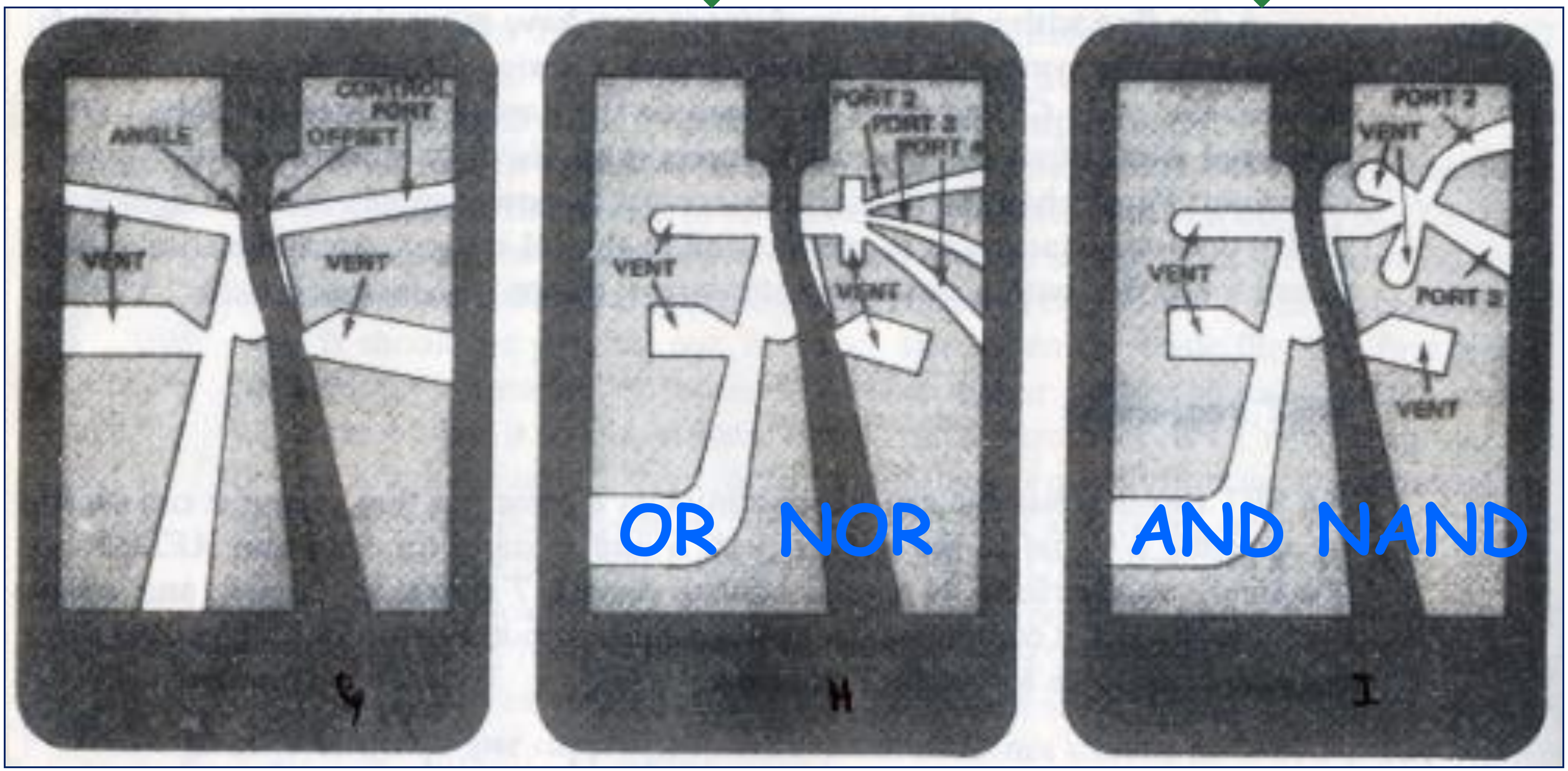
SRT flip flop



T	S	R	O1	O2
0	1	0	1	0
0	0	0	1	0
0	0	1	0	1
0	0	0	0	1
1	0	0	1	0
0	0	0	1	0
1	0	0	0	1
0	0	0	0	1

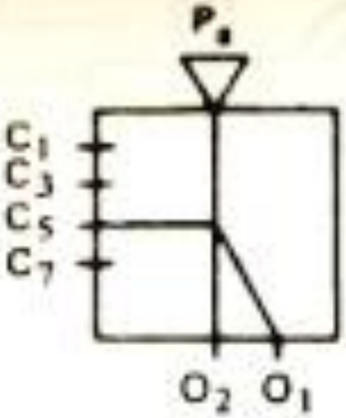
Monostable flip flop

OR/NOR AND/NAND



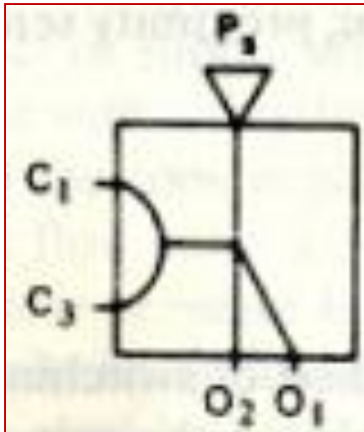
Monostable device is slightly asymmetrical. As a result, this device favors one leg.

OR/NOR



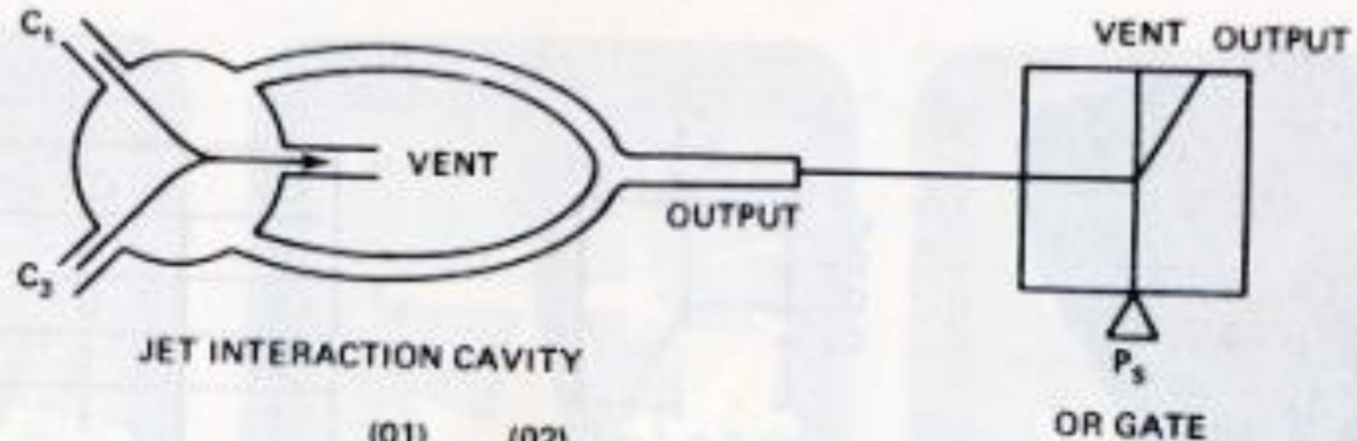
C1	C3	01	02
0	0	0	1
1	0	1	0
0	1	1	0
1	1	1	0

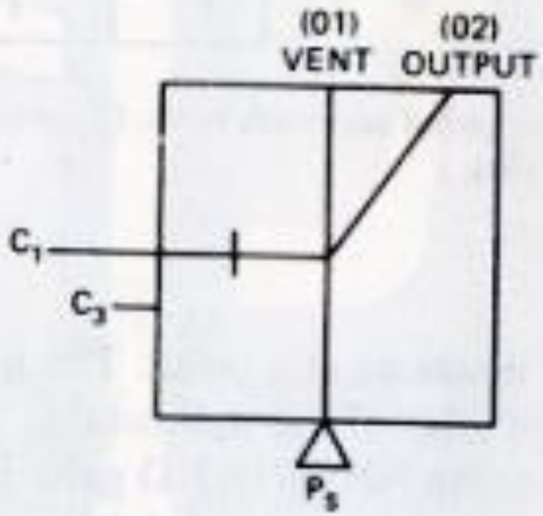
AND/NAND



C1	C3	01	02
0	0	0	1
1	0	0	1
0	1	0	1
1	1	1	0

Exclusive OR





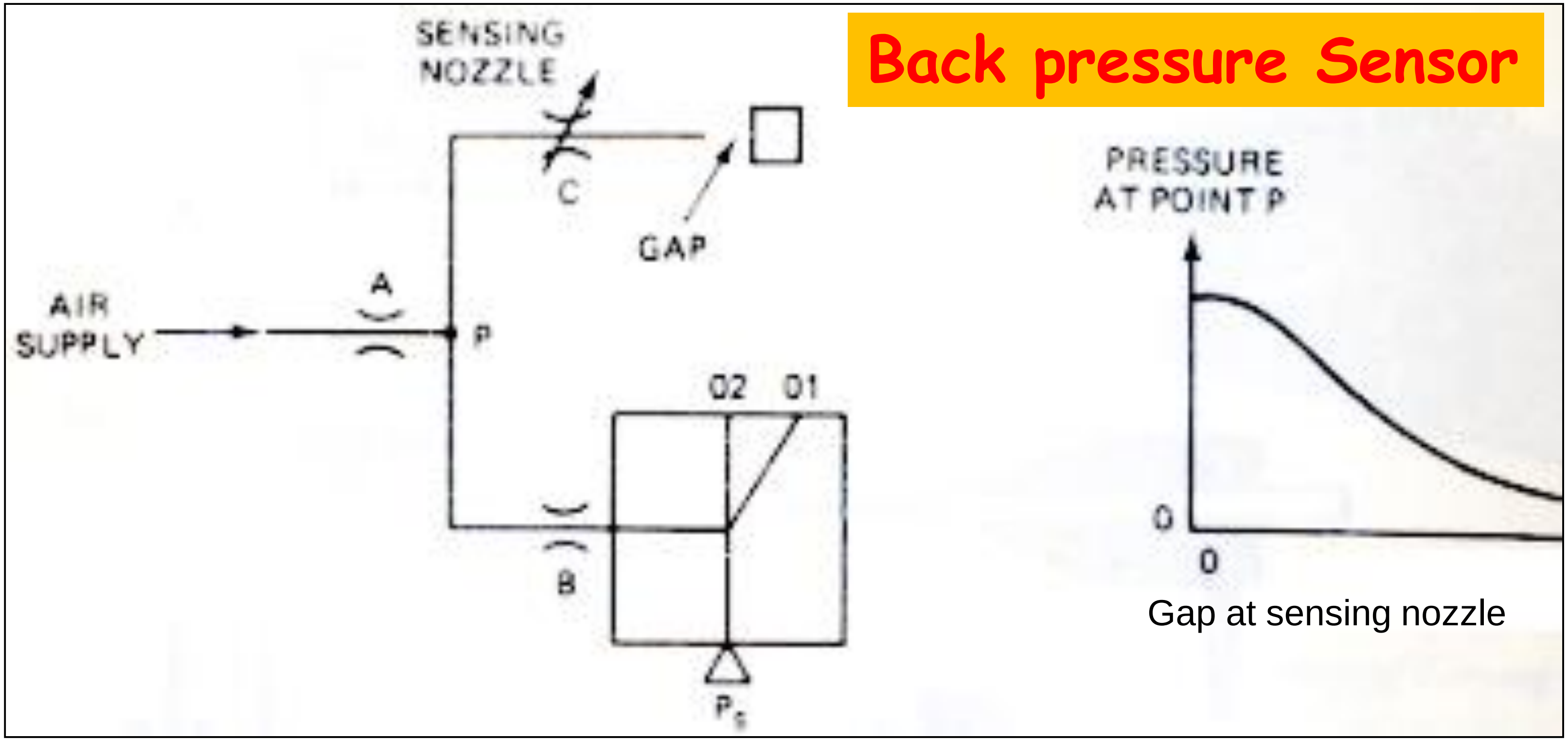
C1	C2	O1	O2
0	0	1	0
1	0	0	1
0	1	0	1
1	1	1	0

Fluid Sensor

It's a device that senses a change in some parameter and as a result causes a related change in another parameter that can be recognized and inierpreted.

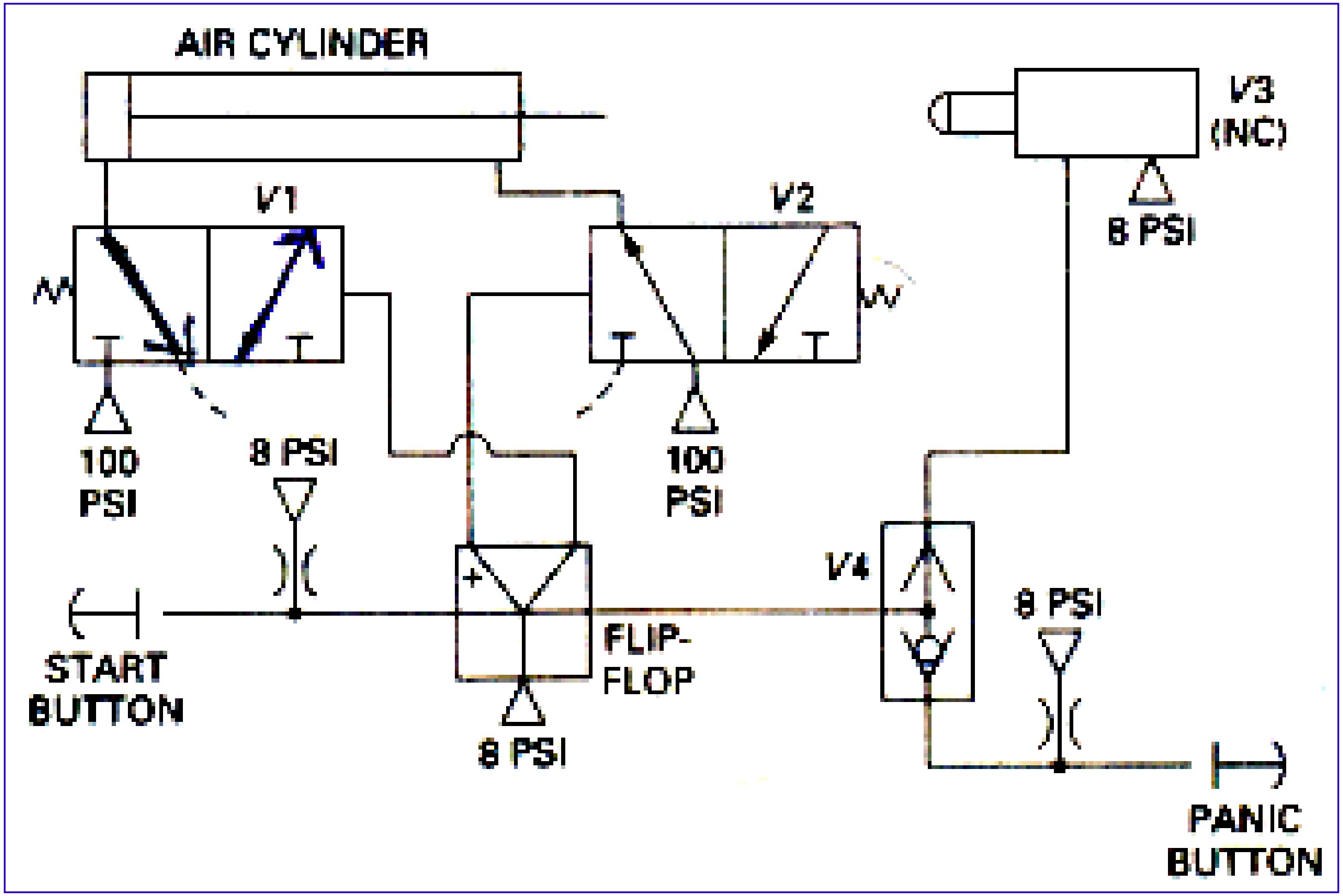
- Back-pressure sensor
- Proximity sensor
- Interruptible sensor
- Contact sensor

Back pressure Sensor

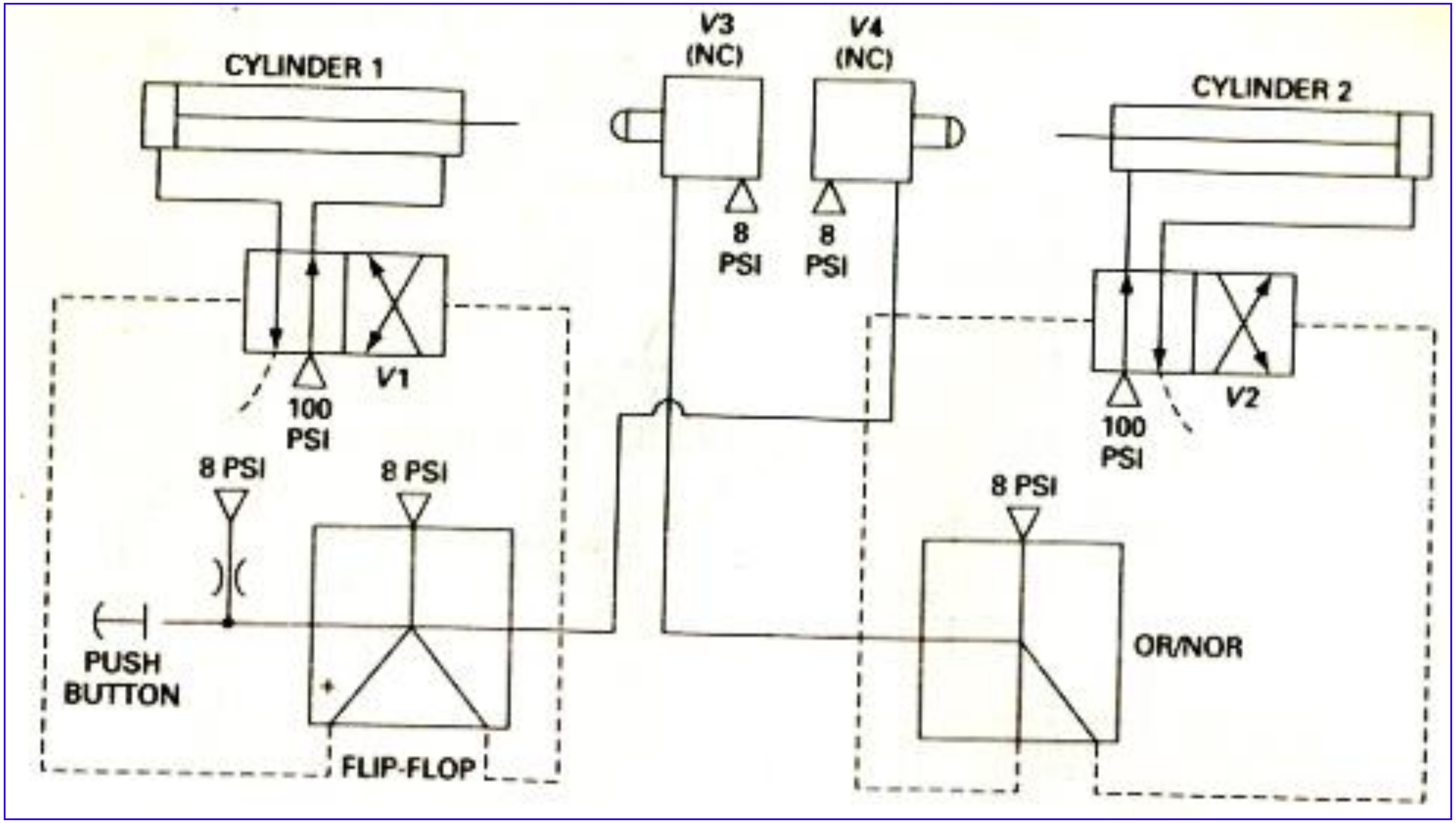


Gap at sensing nozzle

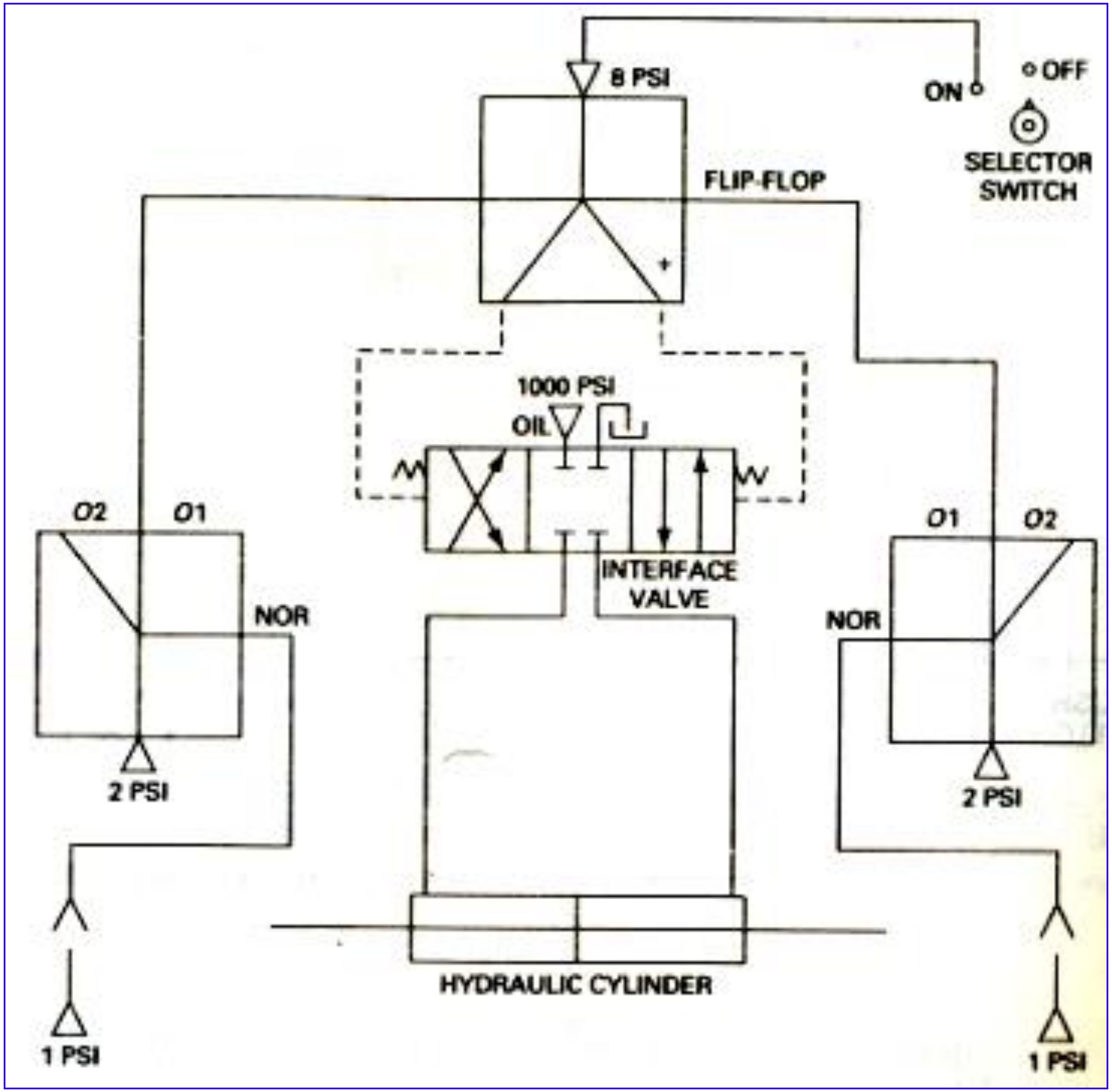
Control of Air cylinder using preferenced Flip flop



Sequencing circuit using preferenced Flip flop OR/NOR Monostable device



Continuous reciprocation of Hydraulic cylinder



ME 7553 – Hydraulics and Pneumatics

Lecture -21

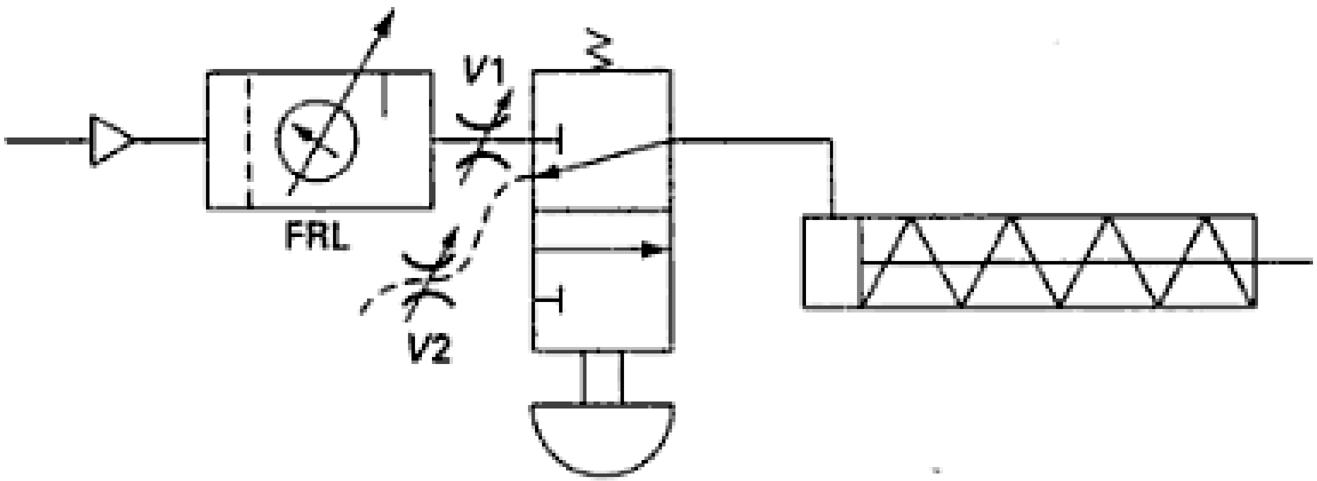
Date: 13-10-2020 Time slot: 08:30-09:20 a.m.

Contents

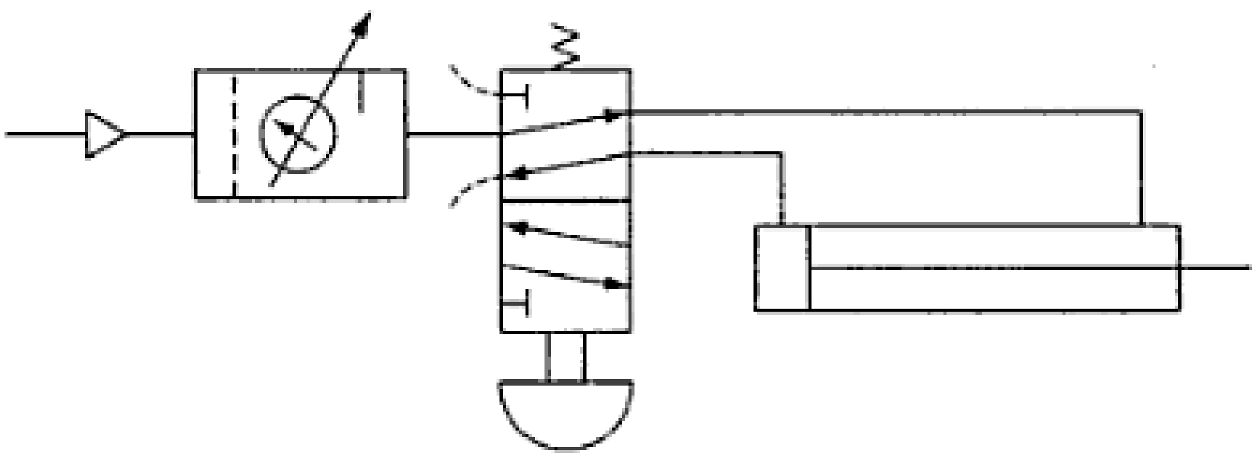
1. Review of lecture 20
2. Valves
3. Pneumatics circuits

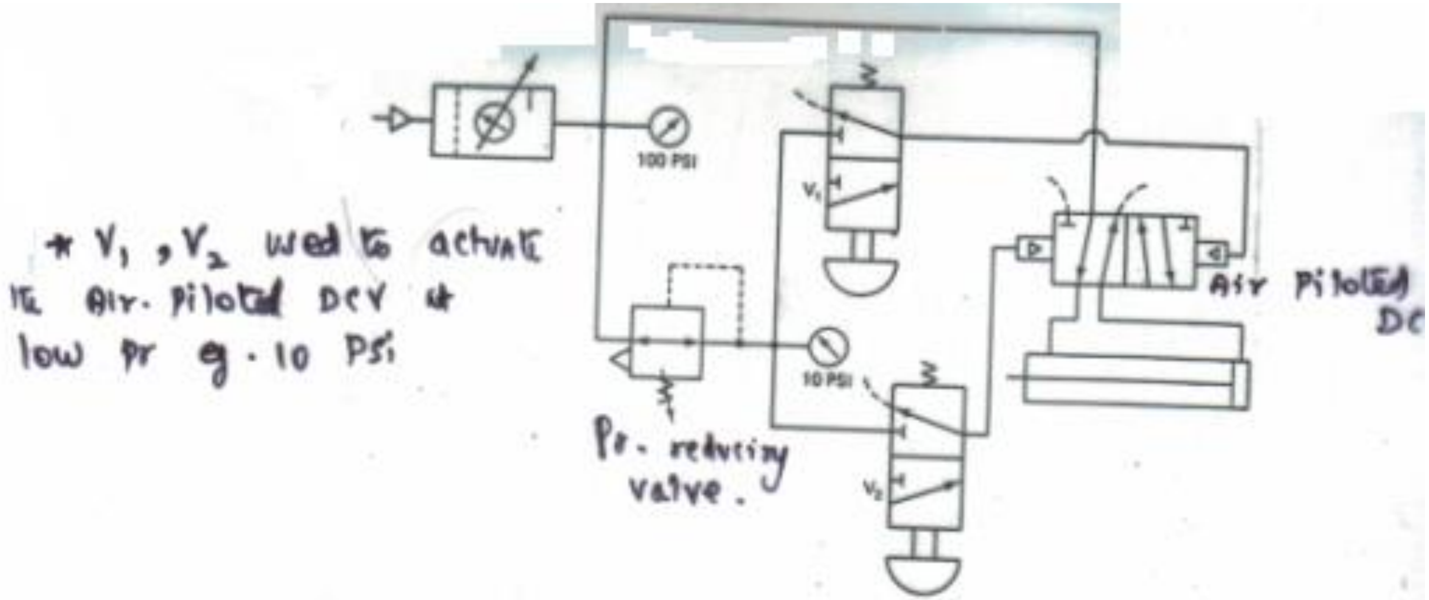
Course Instructor: Dr. A. Siddharthan

Single acting cylinder

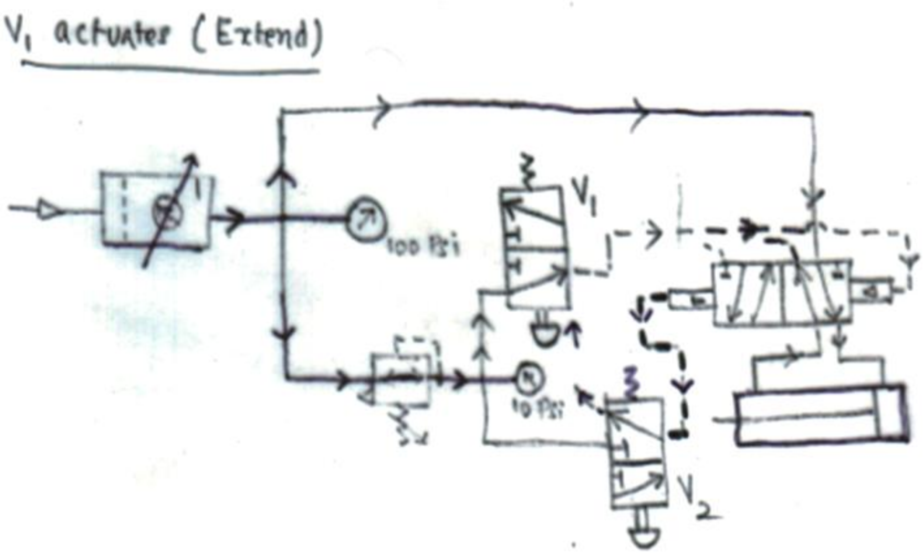


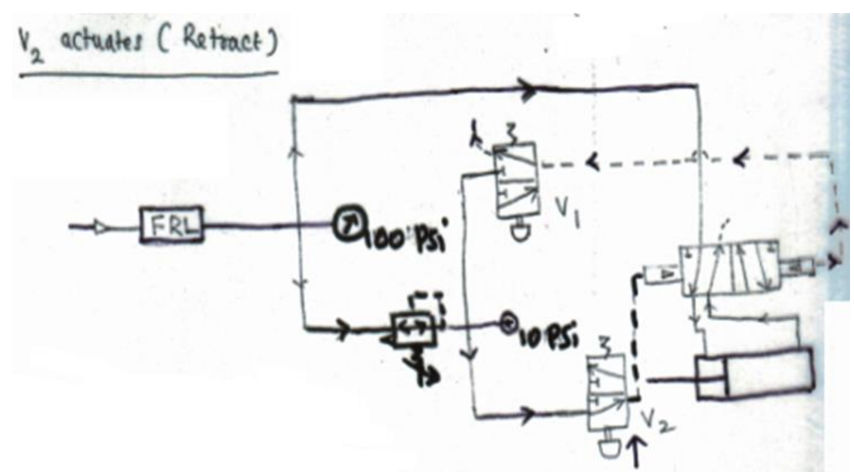
Double acting cylinder

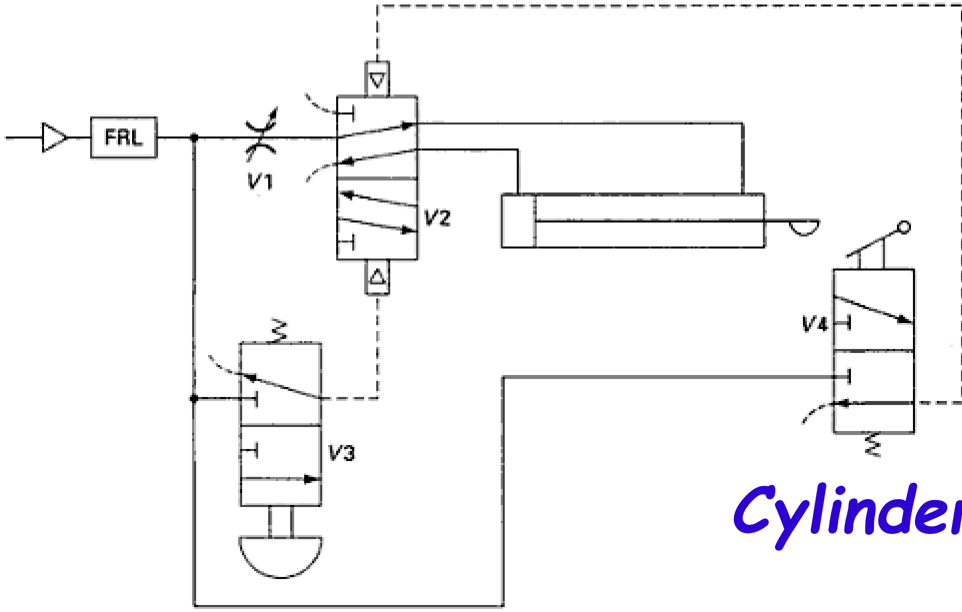




Air pilot control of double acting cylinder

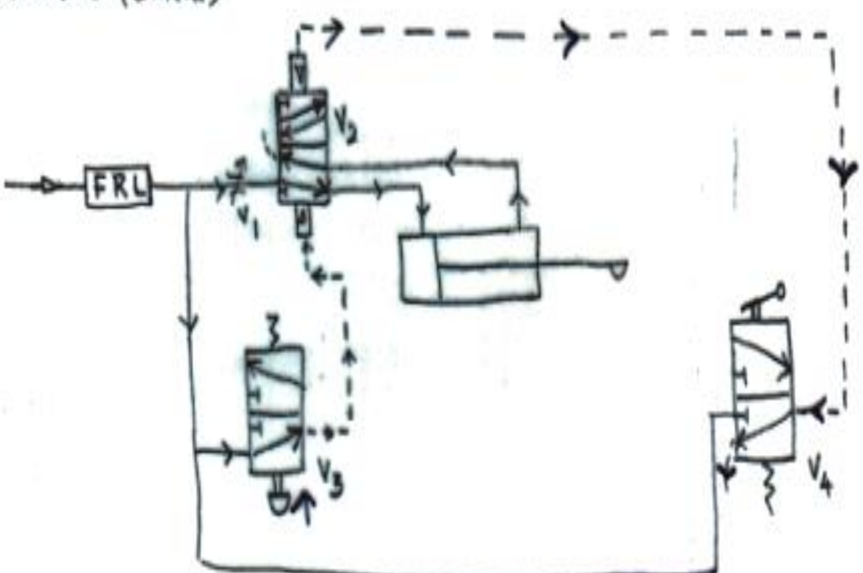




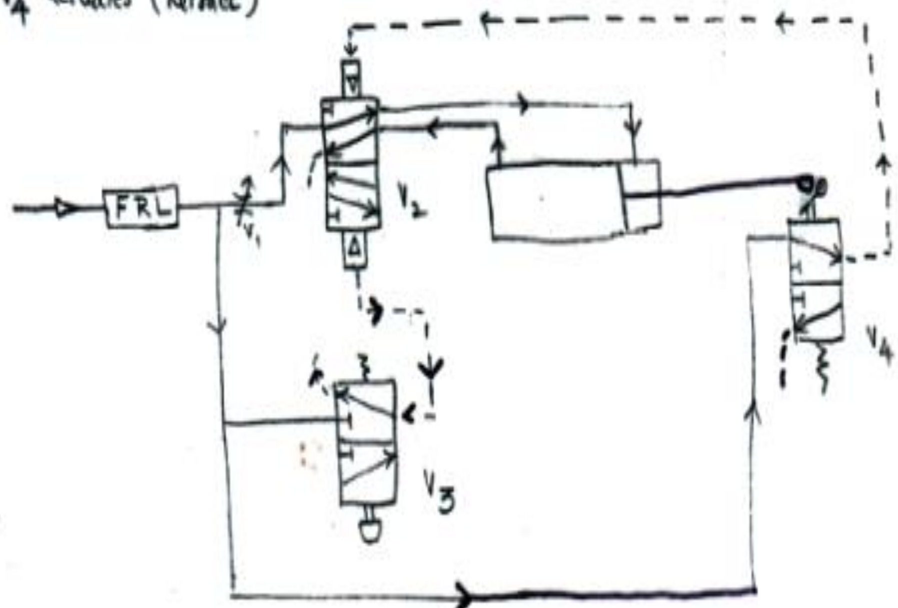


Cylinder Cycle timing system

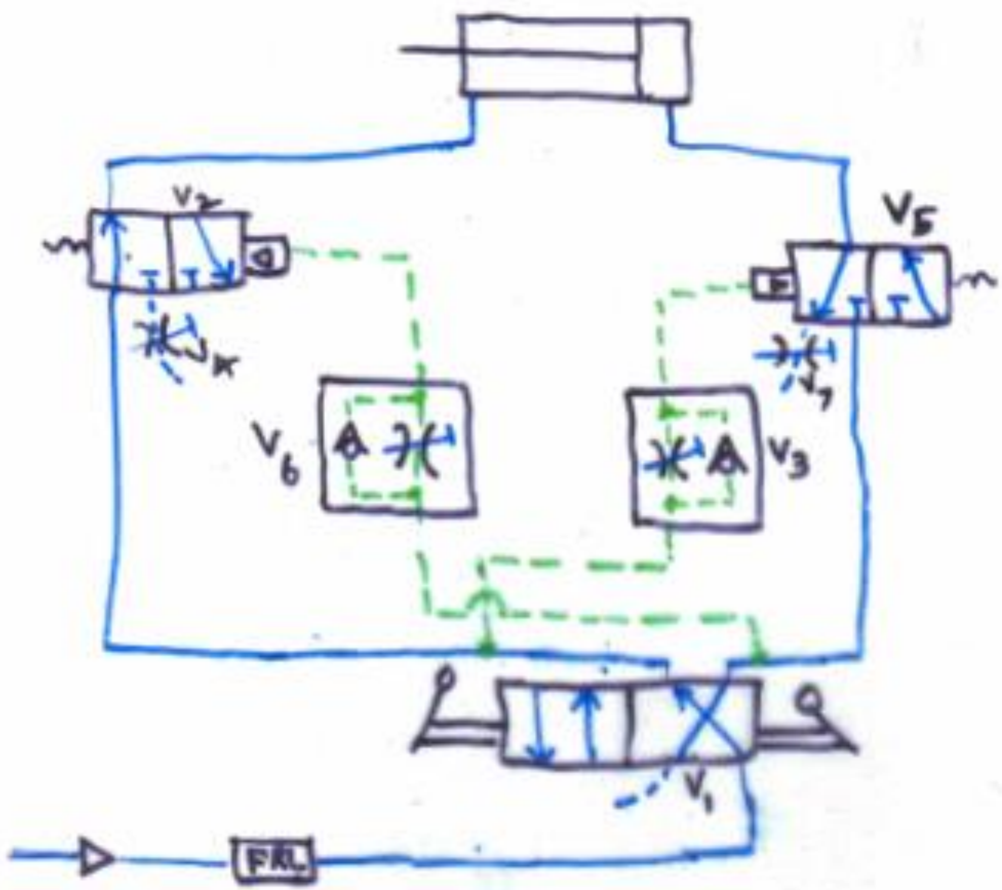
V_3 actuates (Extend)

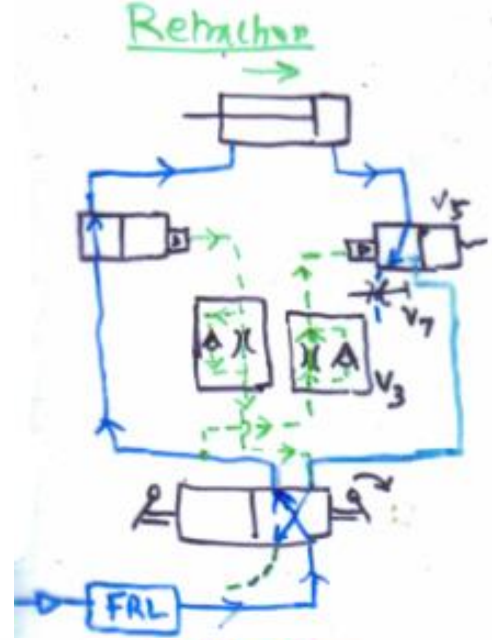


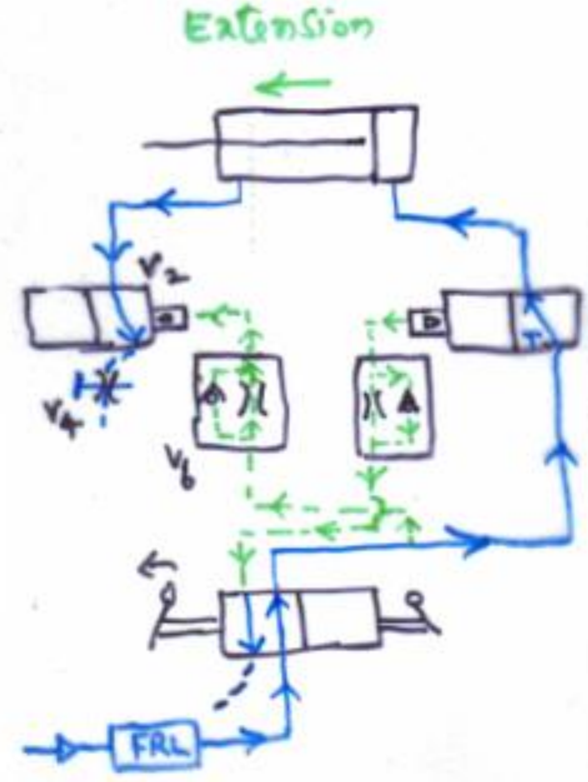
V_4 actuates (Retract)

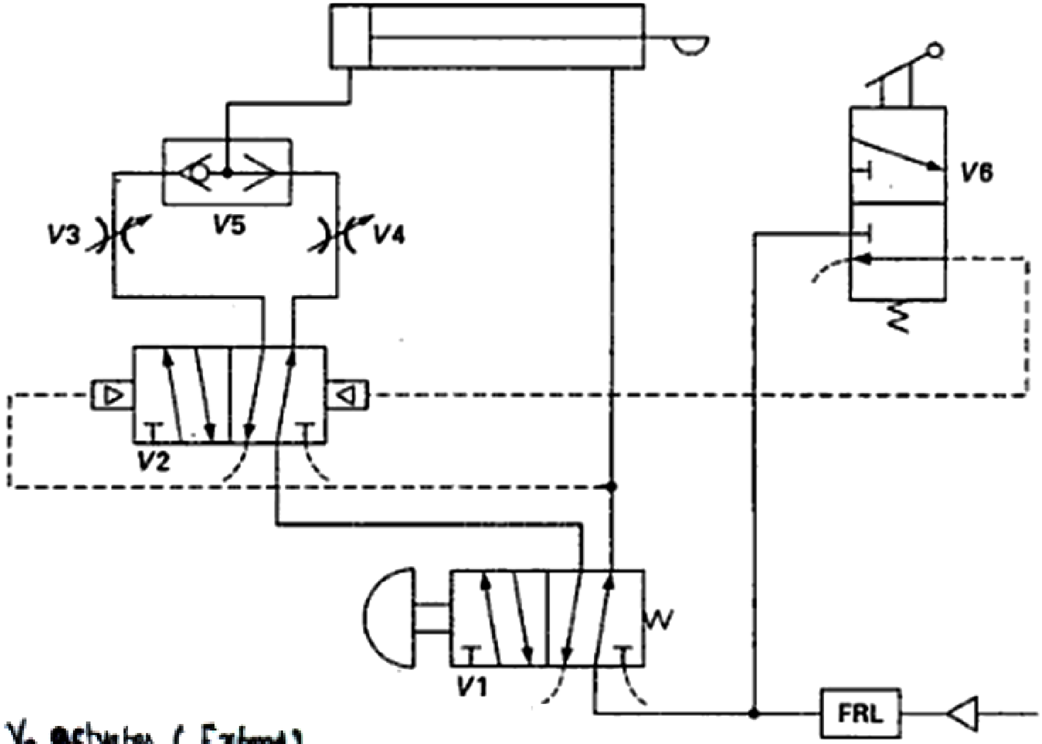


Decelleration air cushion
of a pneumatic circuit





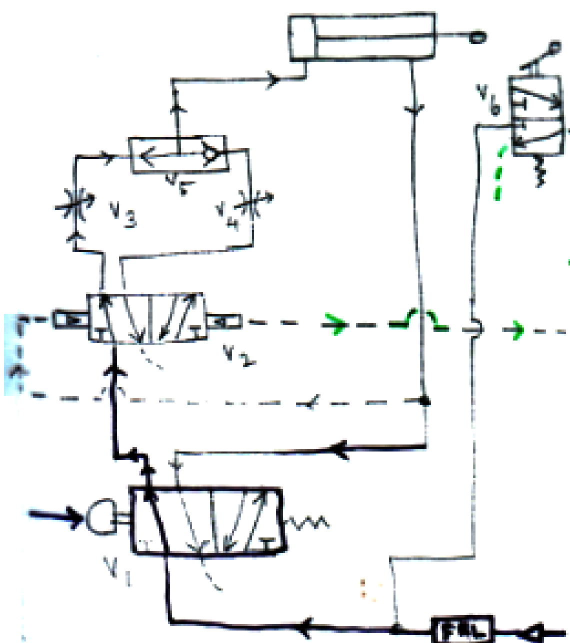




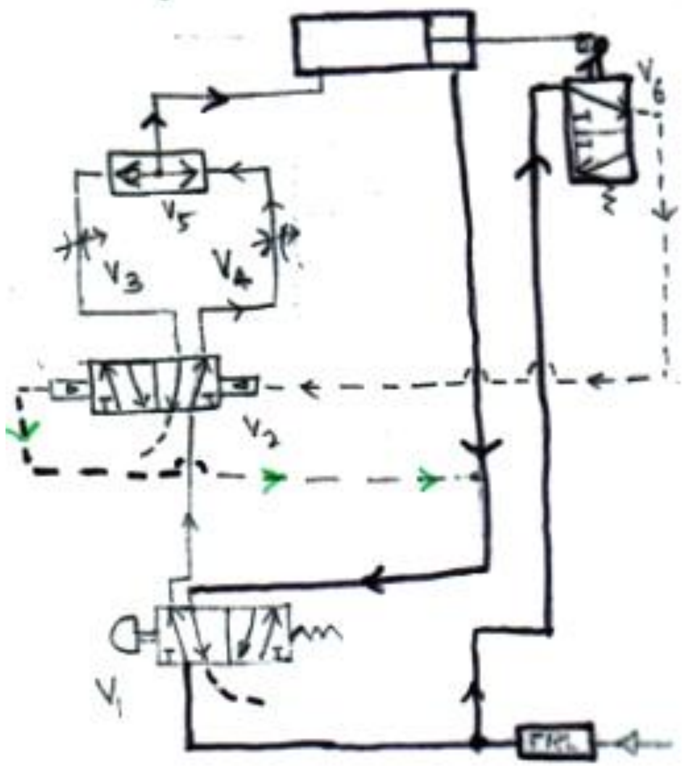
2-step speed control

Assume V3 allows more flow rate than V4

V_3 actuates (Extend)



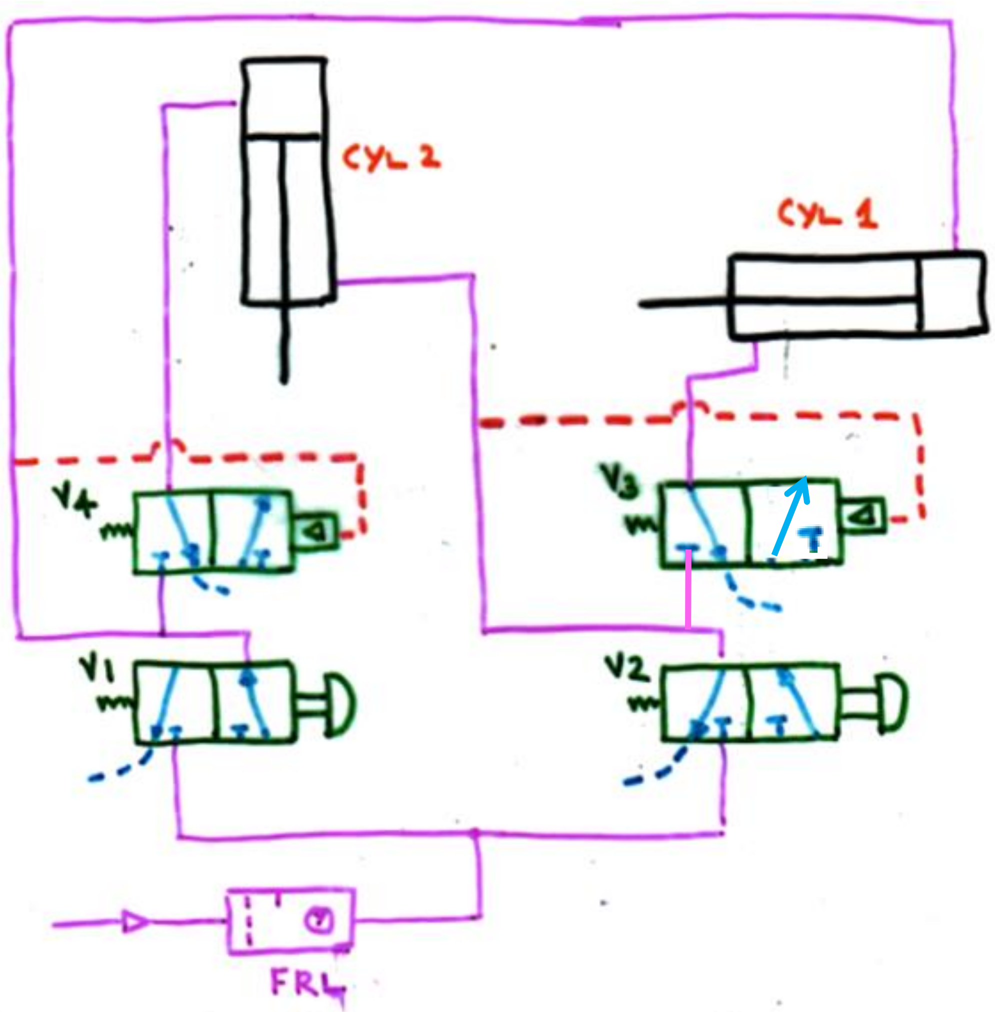
V_4 actuates (End of Extension stroke) via V_6



What happens to the cylinders in each case?

a. **V1 is actuated and held**

b. **V1 is released and V2 is actuated and held**

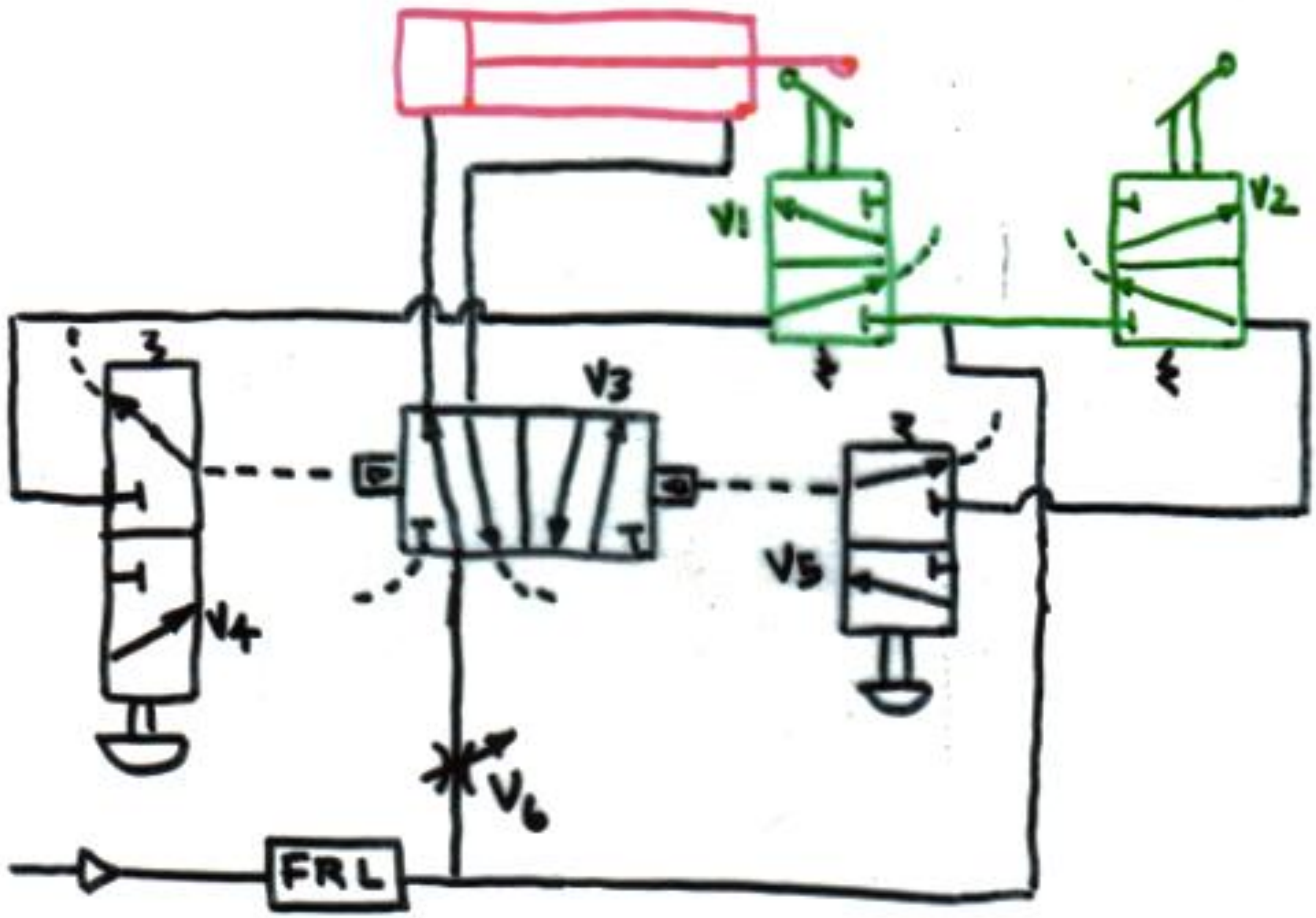


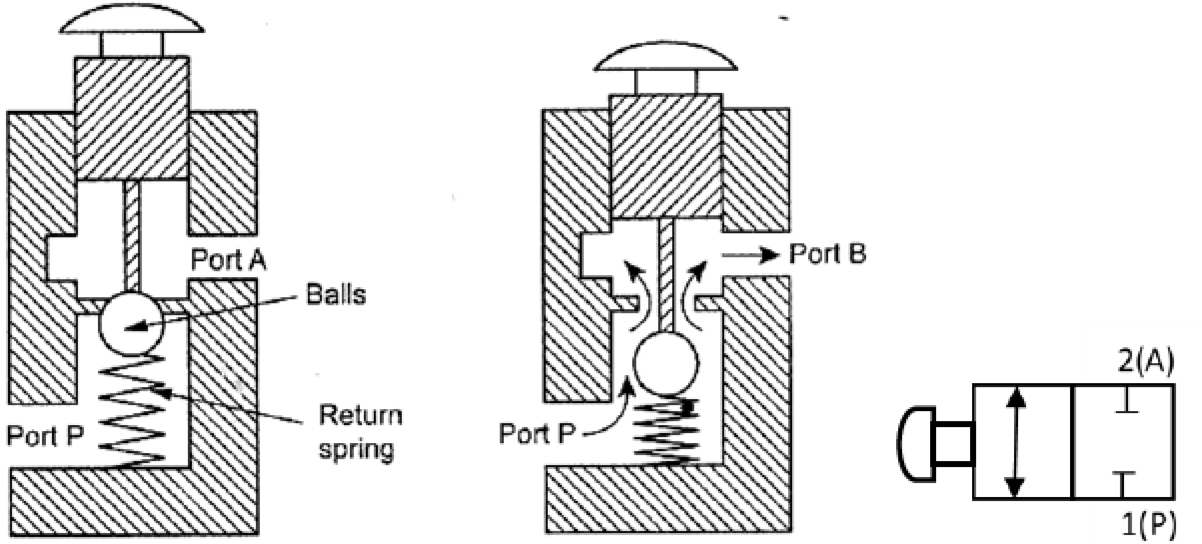
V3 & V4 - Sequence valve

Consider the circuit

a. What happens to the cylinder when V4 is used?

b. What happens to the cylinder when V5 is used?





(a) Poppet position 1 *(b) Poppet position 2* (c) Symbol

Figure 1.1 Two/Two Ball seat Pop

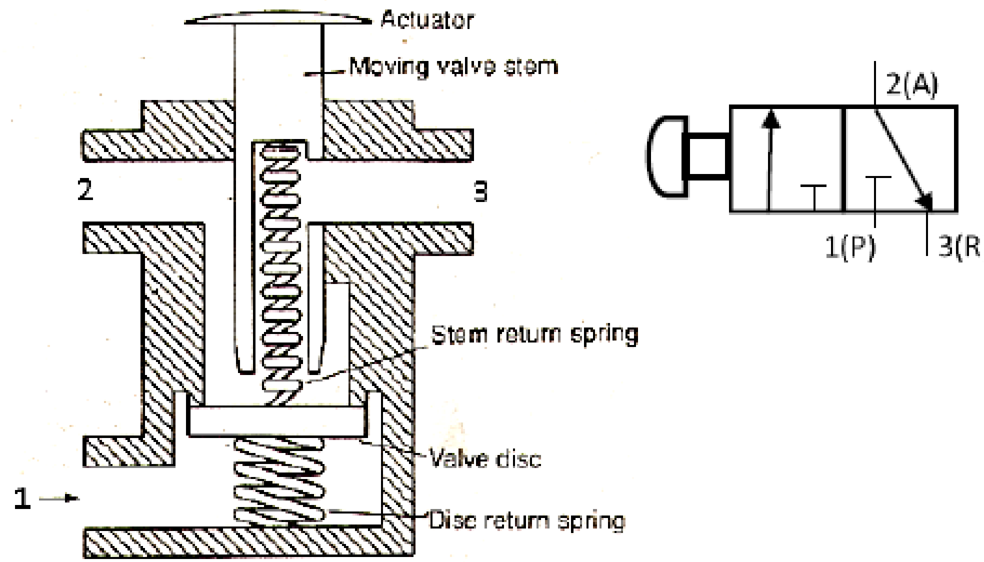


Figure 1.2 Disc seat poppet valve

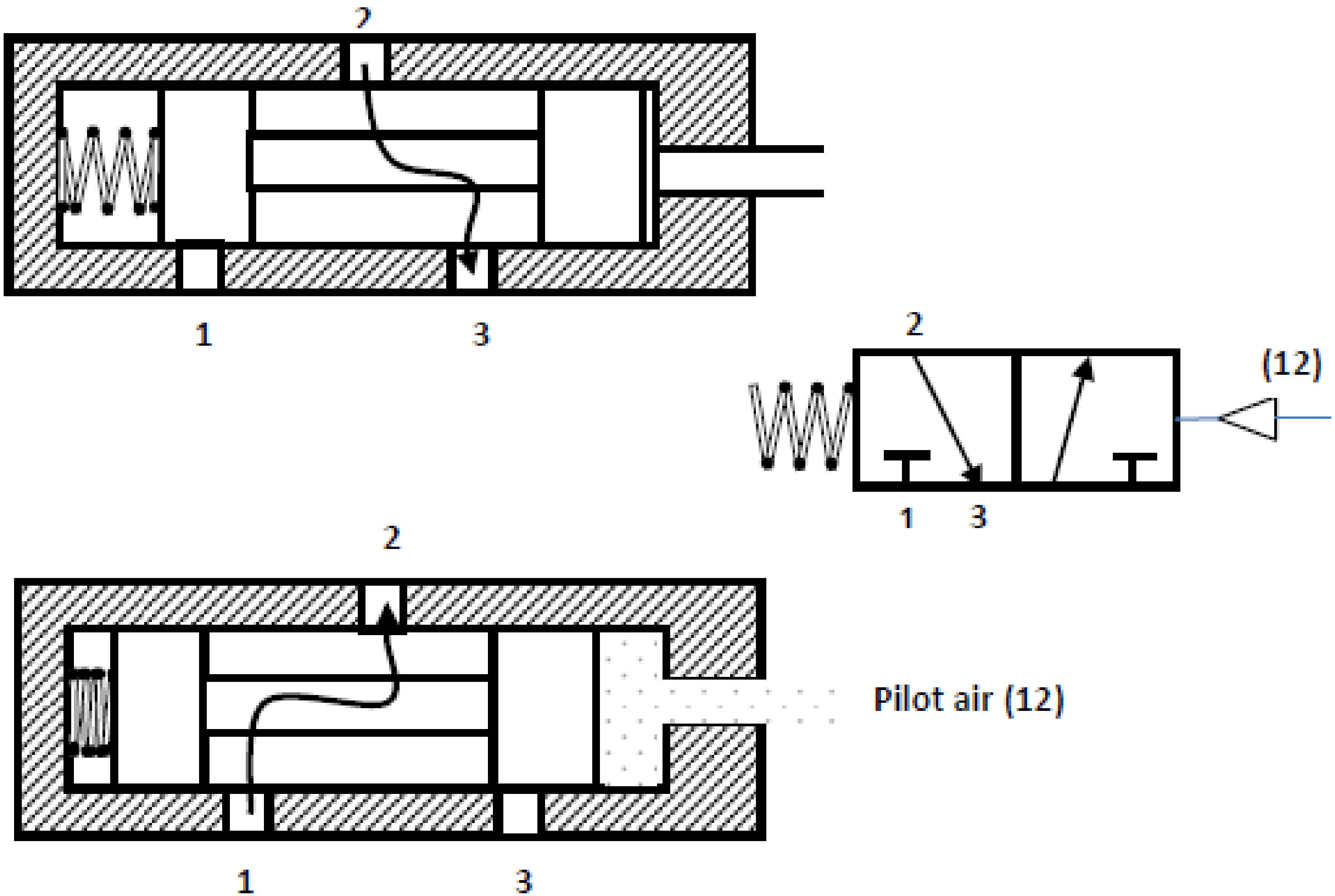


Figure 1.7 3/2 Directional control valve (pneumatically operated)

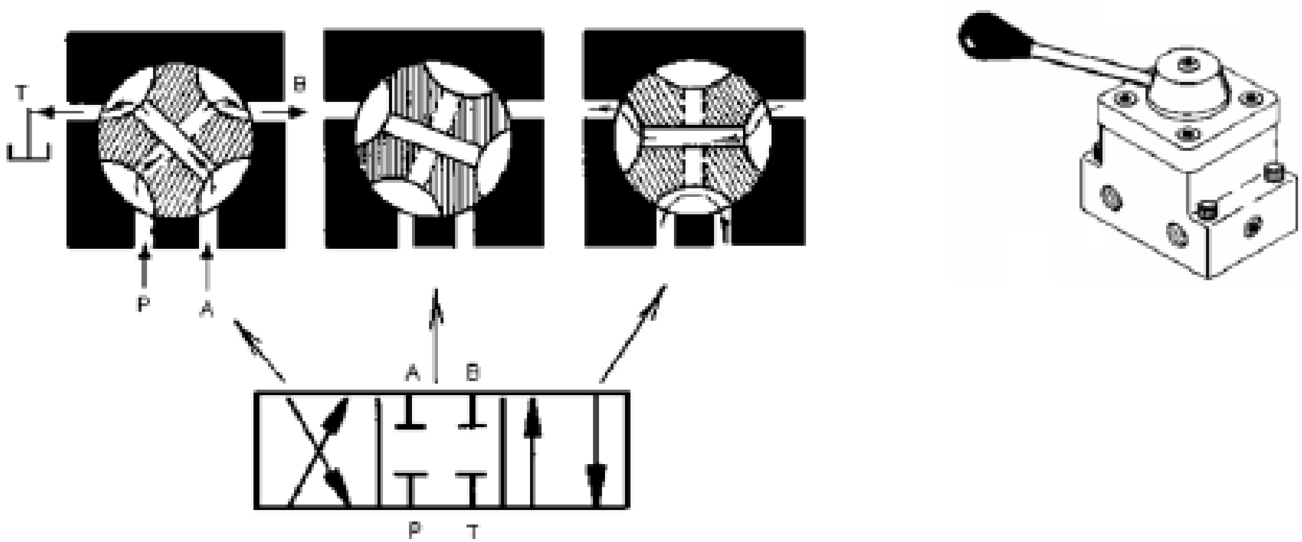


Figure 1.15 Three different positions of 4/3 way rotary spool directional control valve.

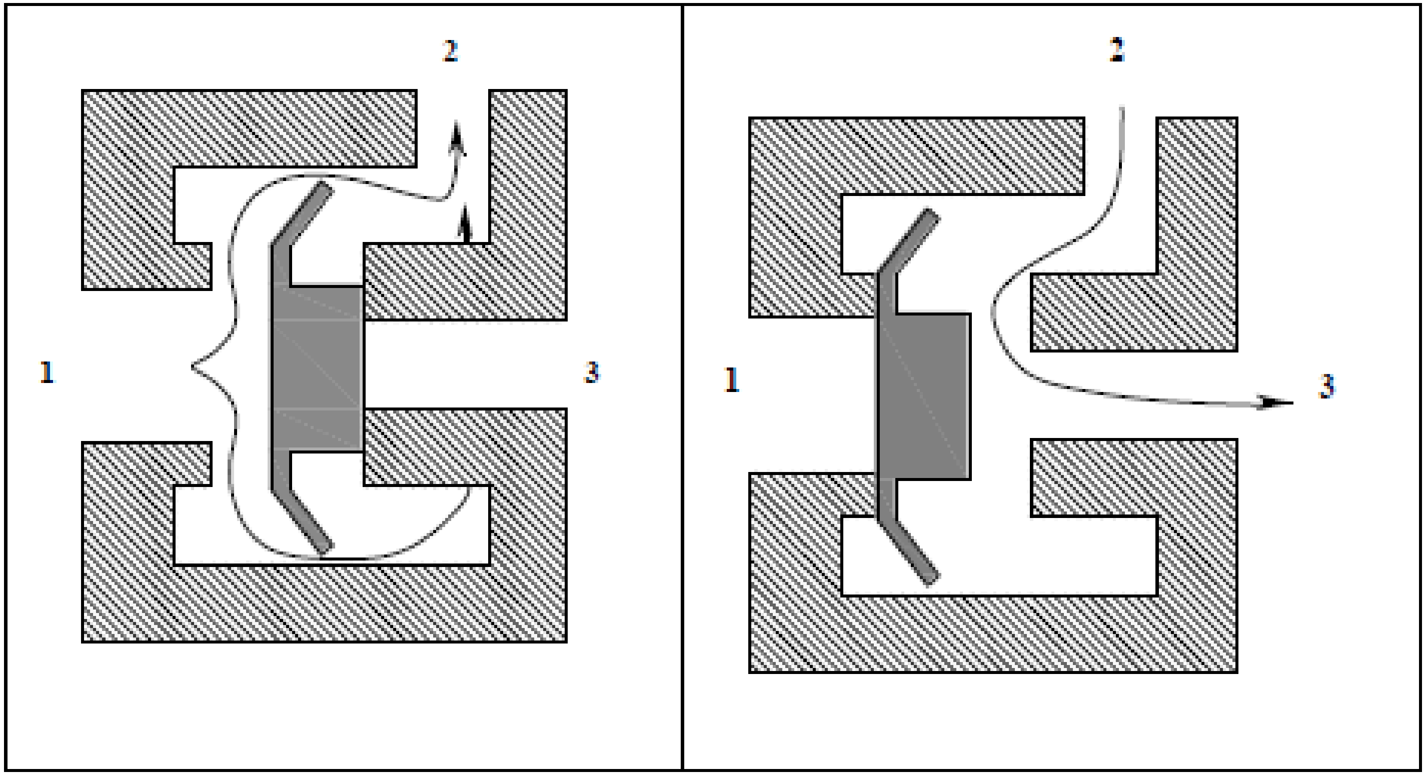


Figure 1.38 Functional diagram of quick exhaust valve.

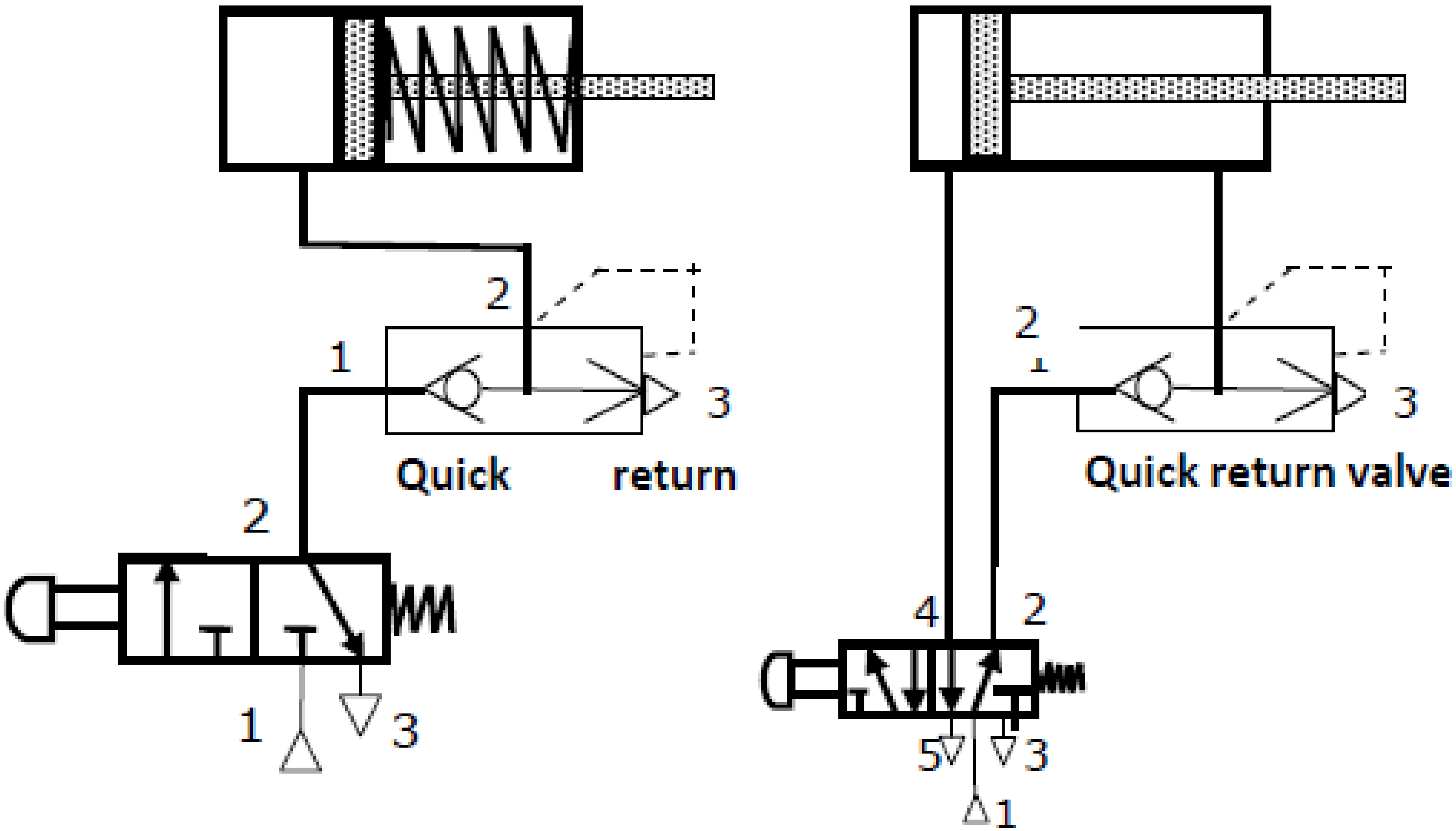


Figure 1.39 Application of quick exhaust valve.

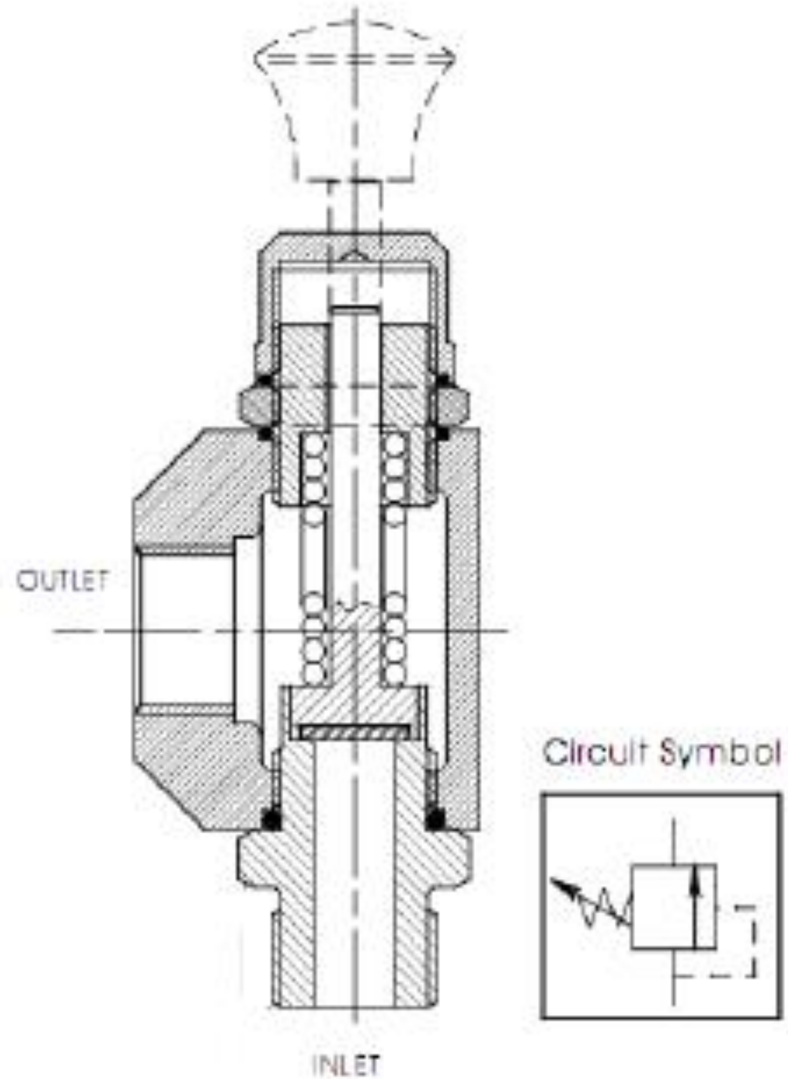


Figure 1.43 Pressure limiting valve

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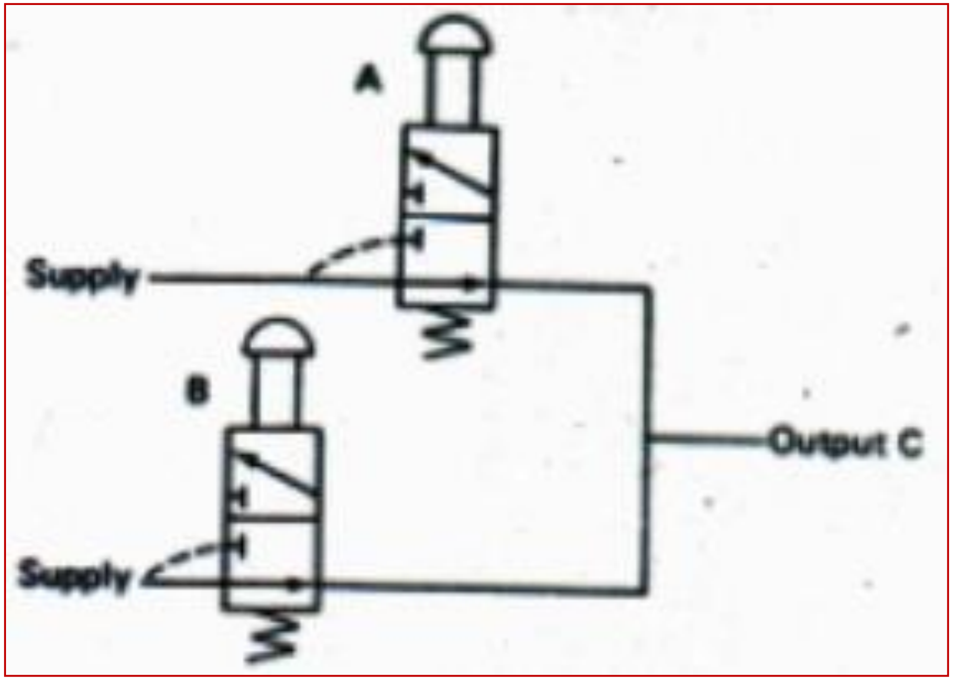

Memory Circuit

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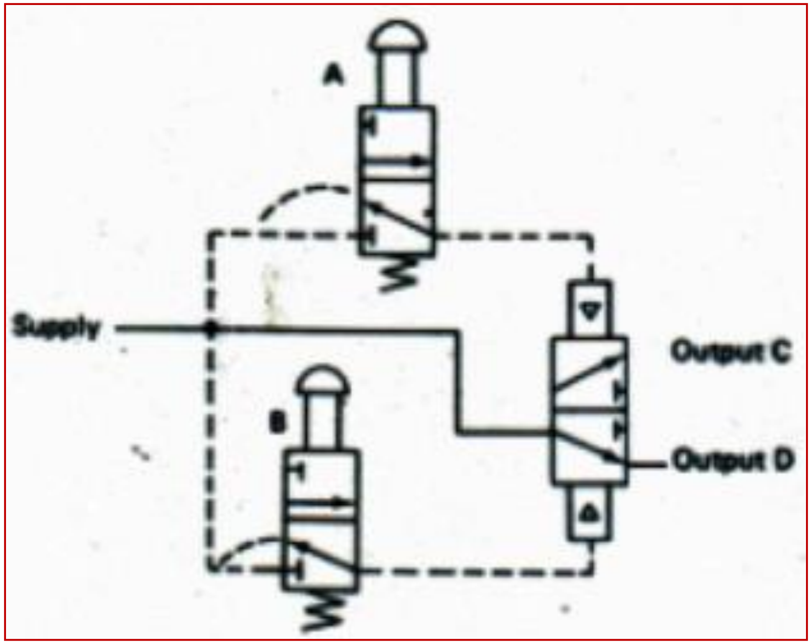
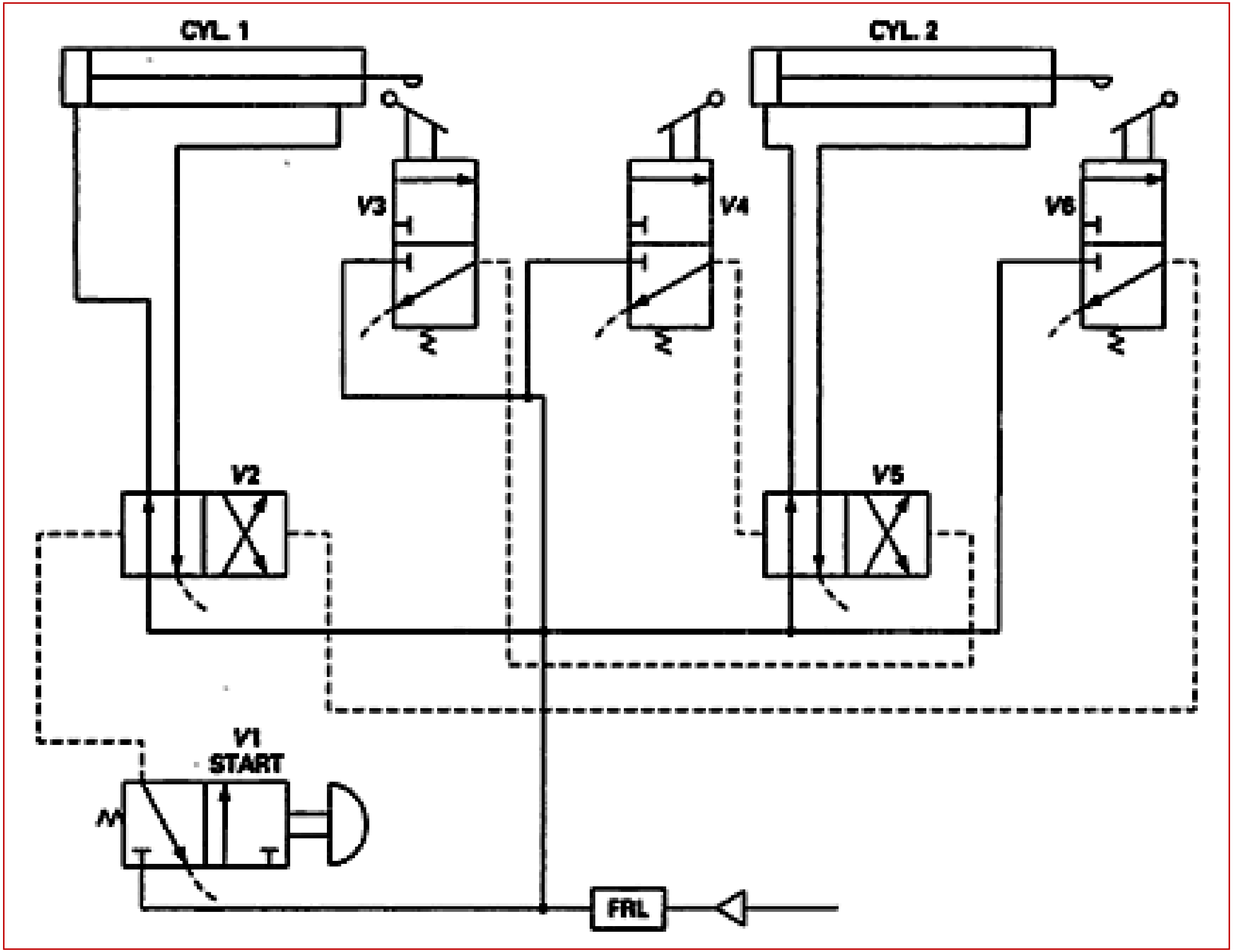
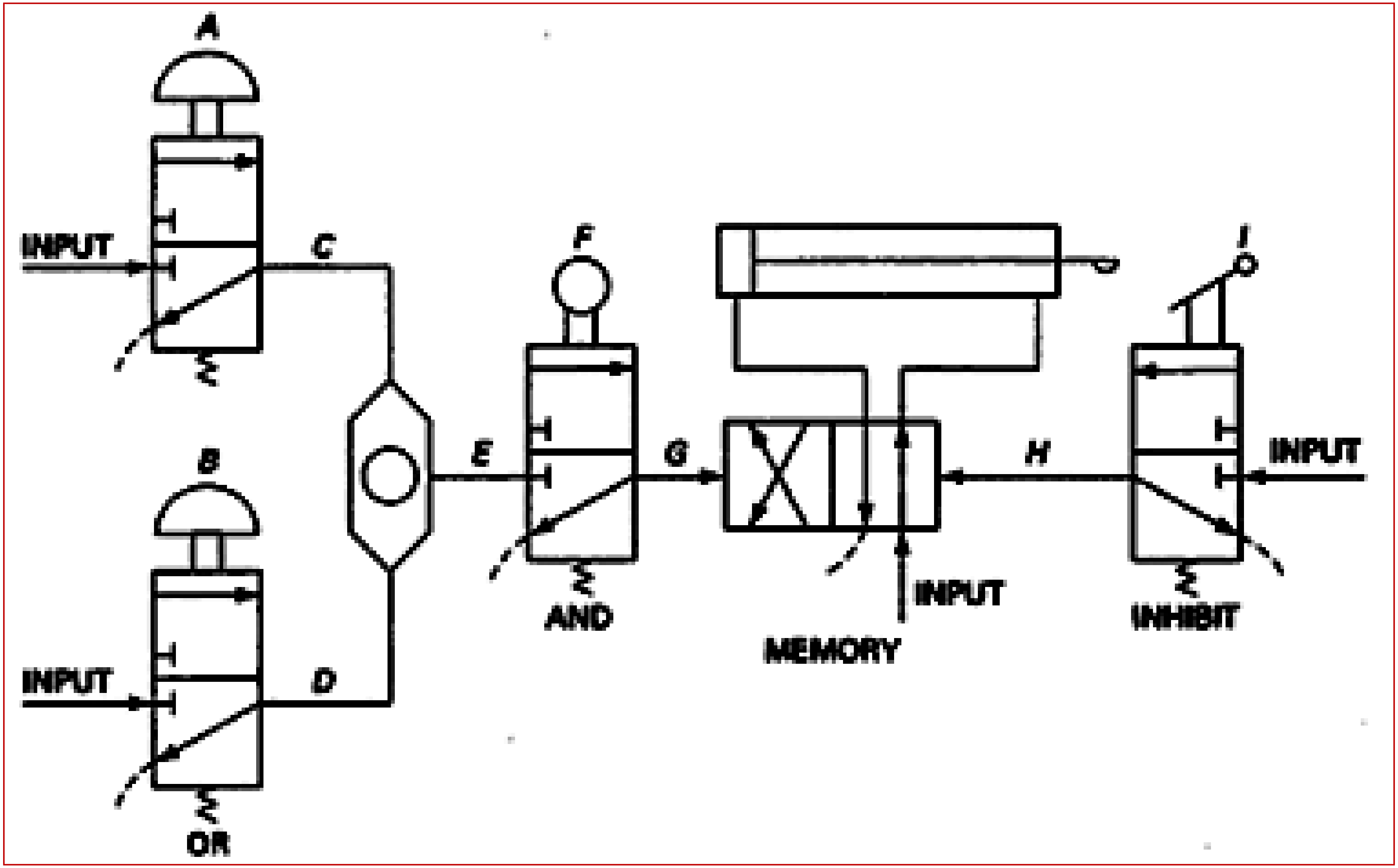


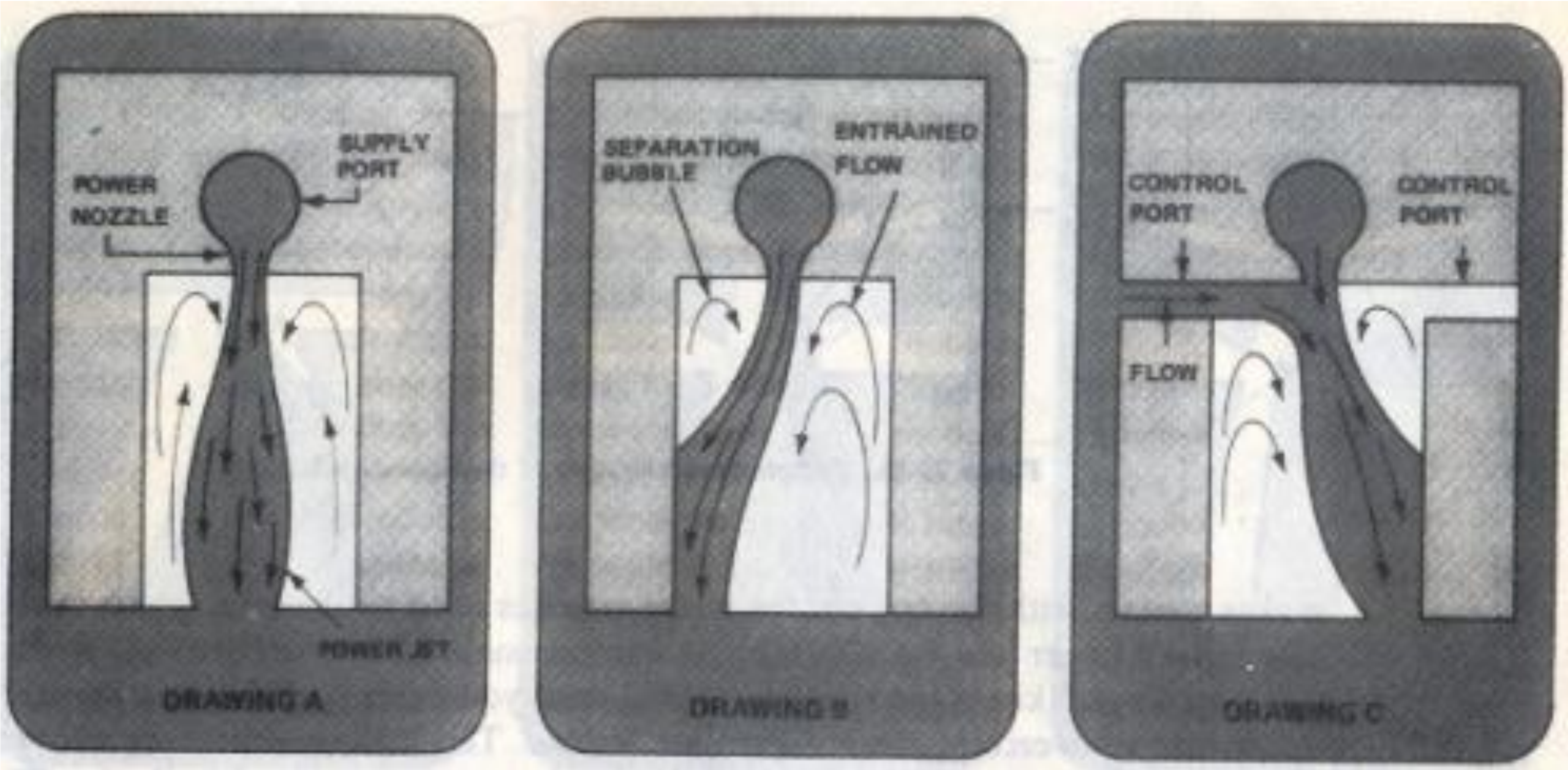
MPL Cylinder sequencing circuit

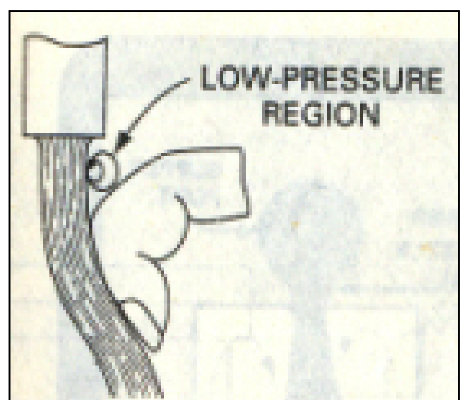


MPL control of a Double acting cylinder

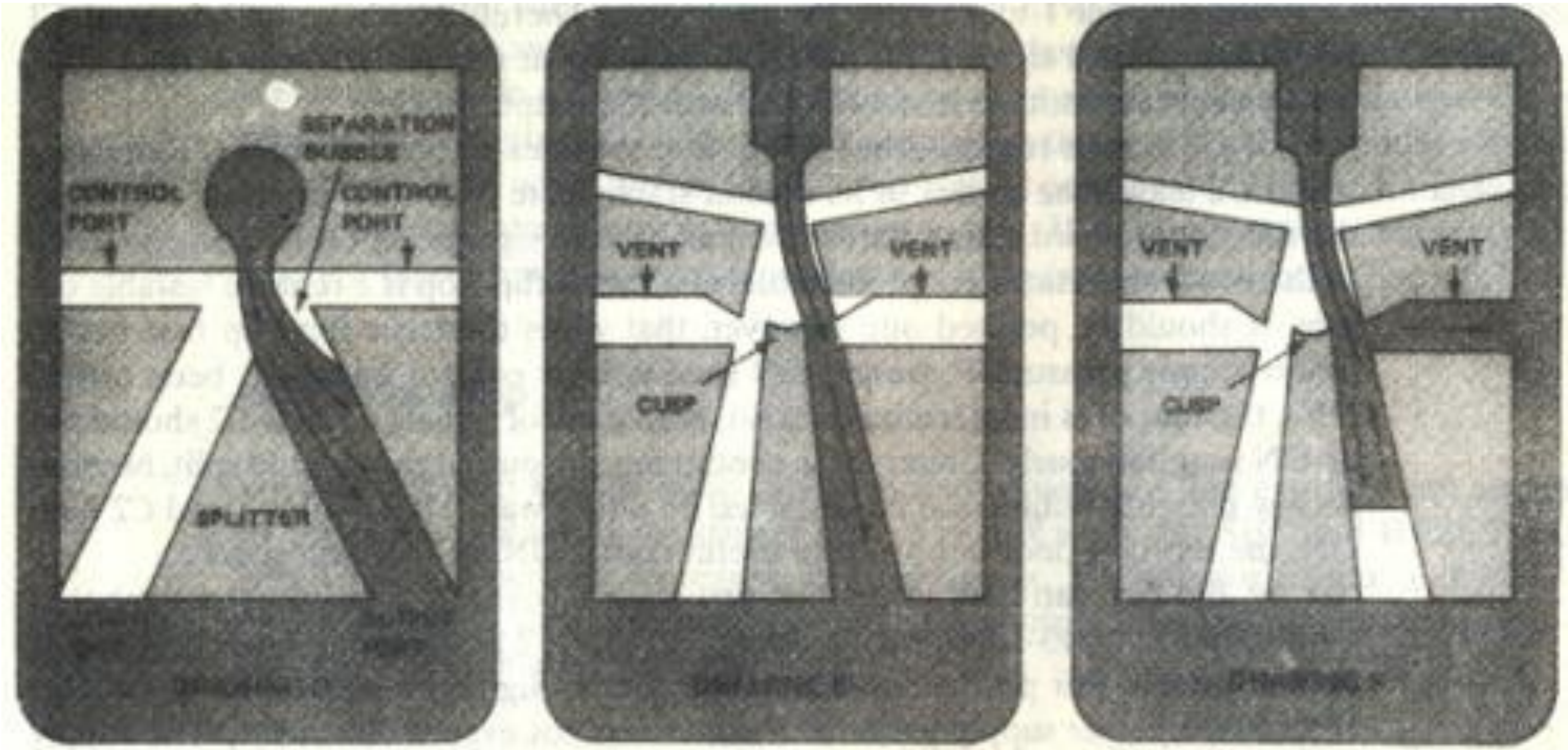


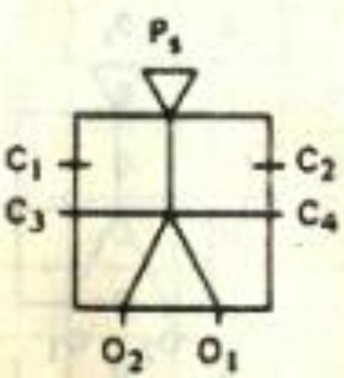
Coanda effect devive





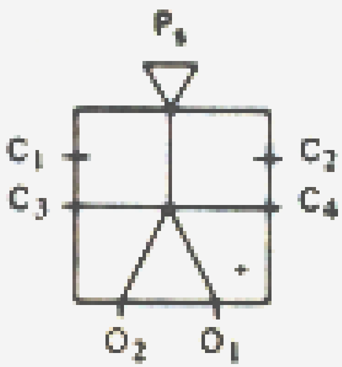
Basic Bistable flip-flop





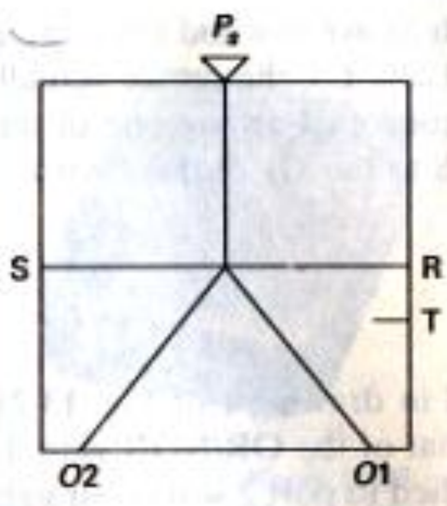
C1	C2	01	02
1	0	1	0
0	0	1	0
0	1	0	1
0	0	0	1

Preferenced flip flop



C1	C2	O1	O2
0	0	1	0
0	1	0	1
0	0	0	1
1	0	1	0
0	0	1	0

SRT flip flop



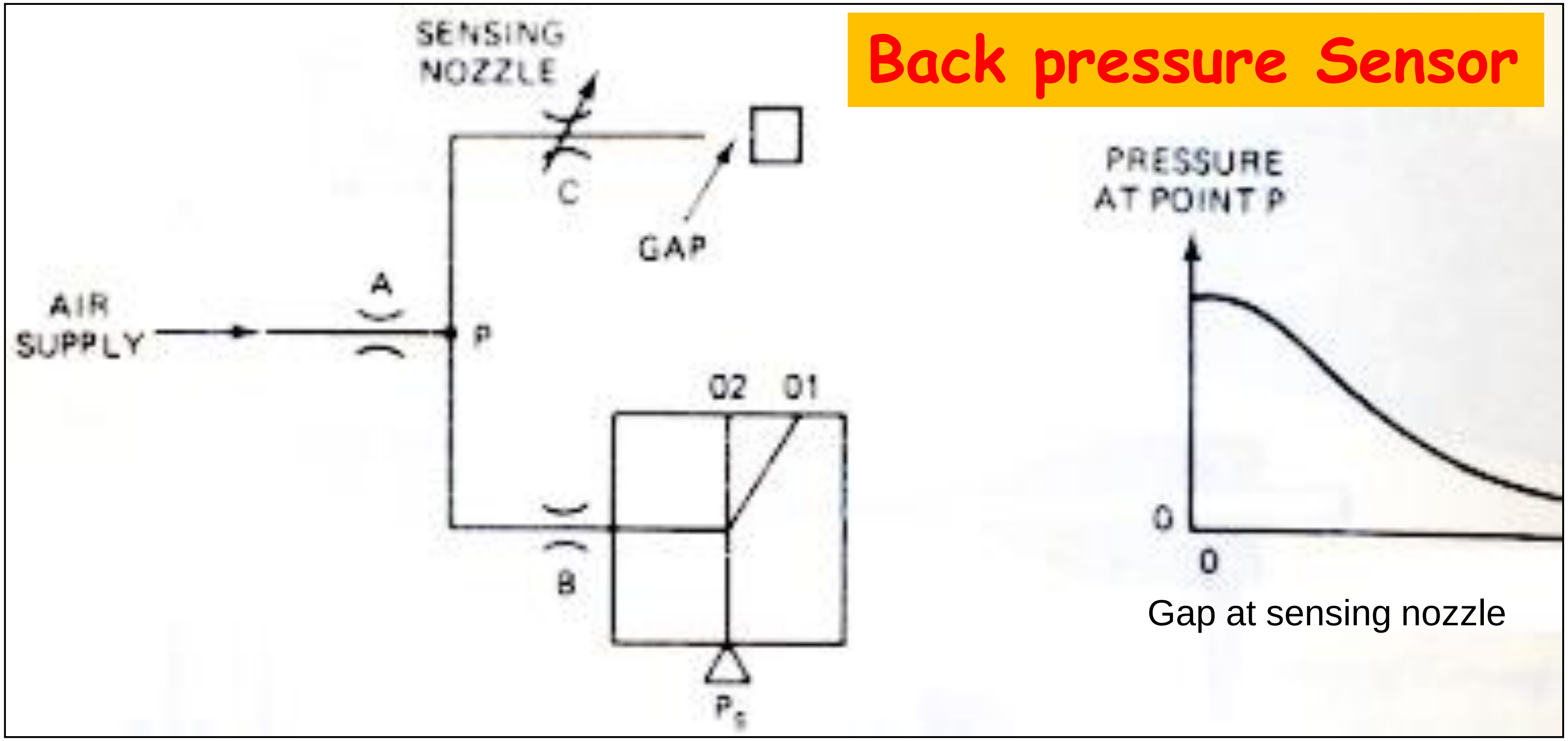
T	S	R	O1	O2
0	1	0	1	0
0	0	0	1	0
0	0	1	0	1
0	0	0	0	1
1	0	0	1	0
0	0	0	1	0
1	0	0	0	1
0	0	0	0	1

Fluid Sensor

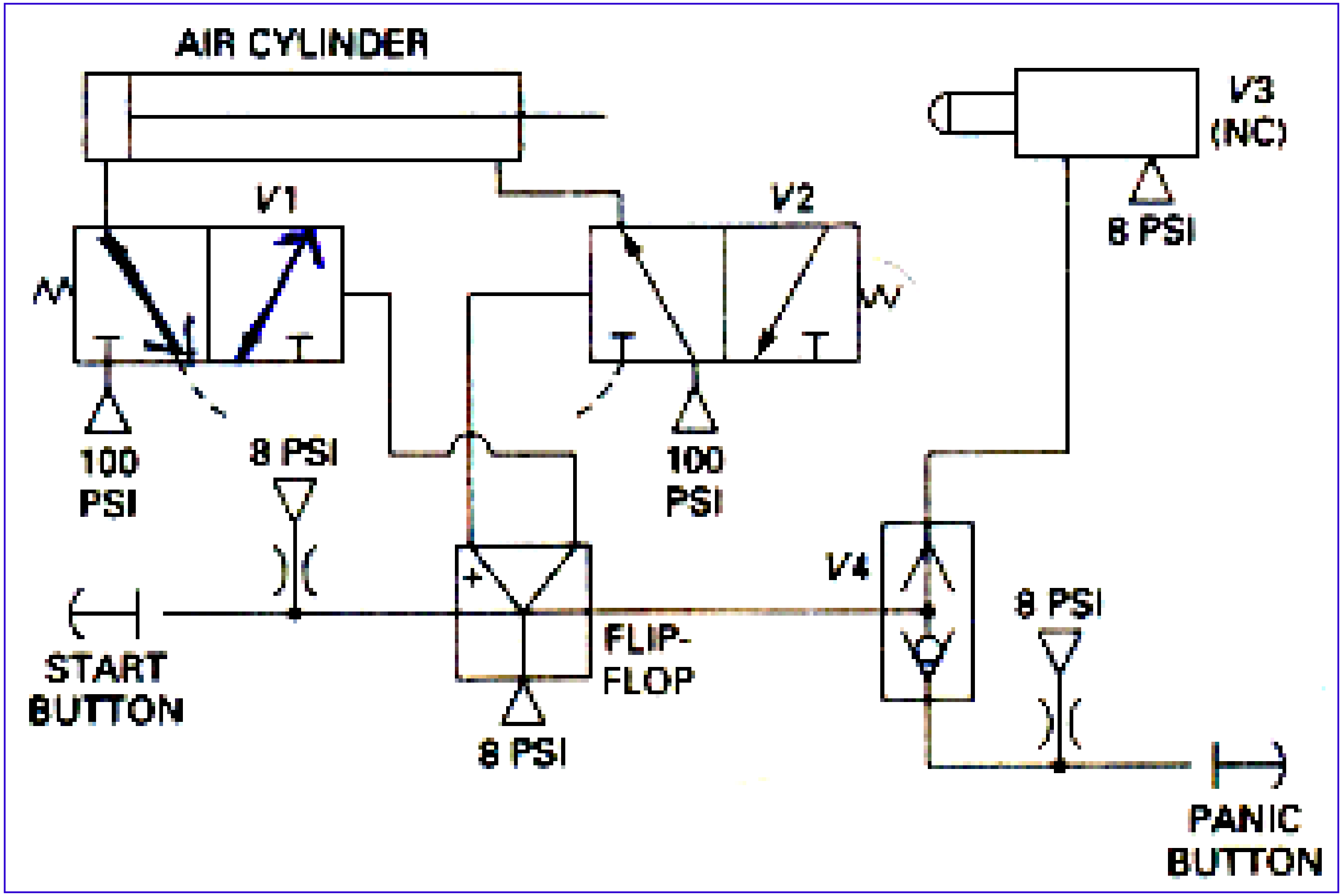
It's a device that senses a change in some parameter and as a result causes a related change in another parameter that can be recognized and inierpreted.

- Back-pressure sensor
- Proximity sensor
- Interruptible sensor
- Contact sensor

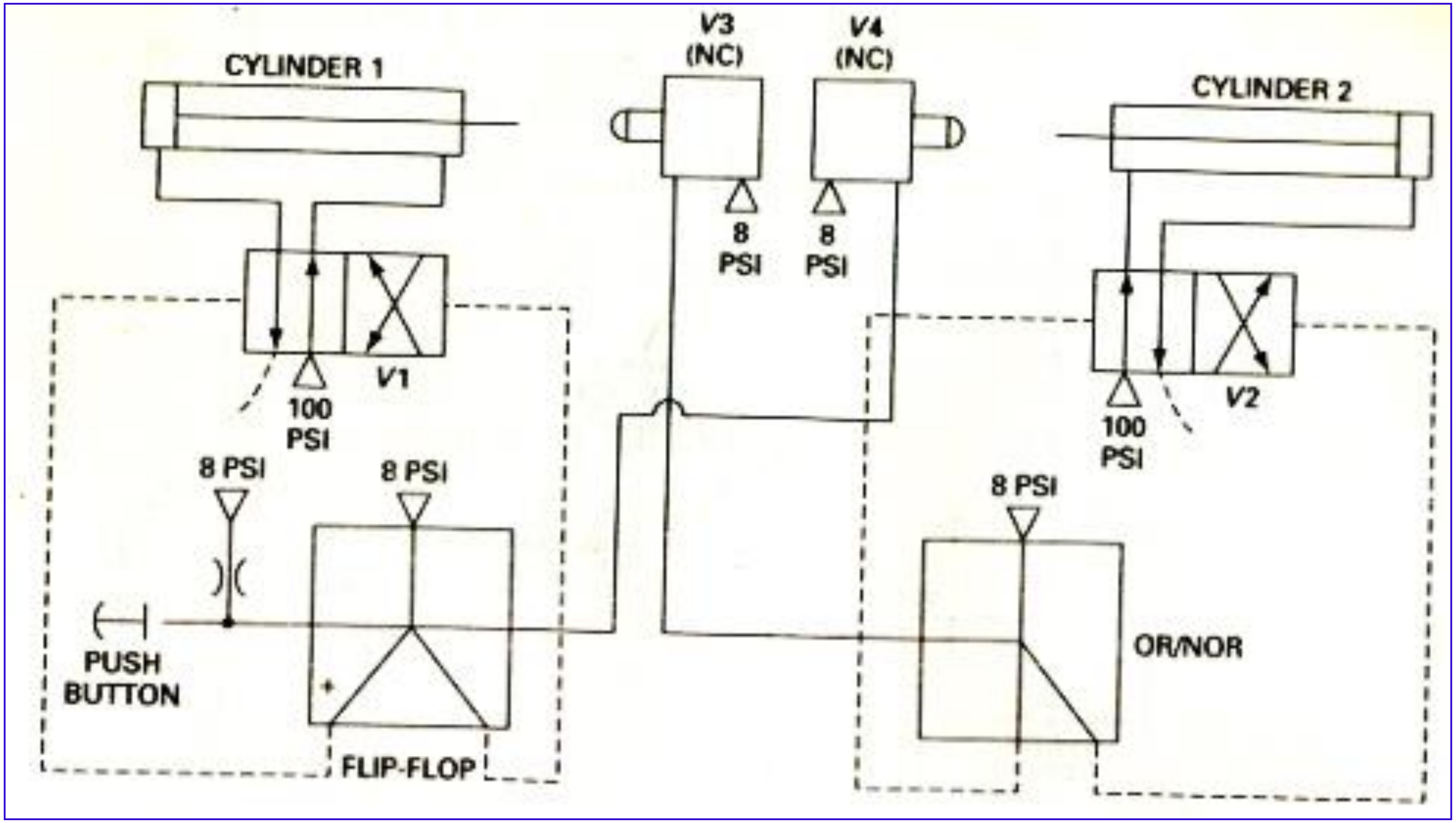
Back pressure Sensor



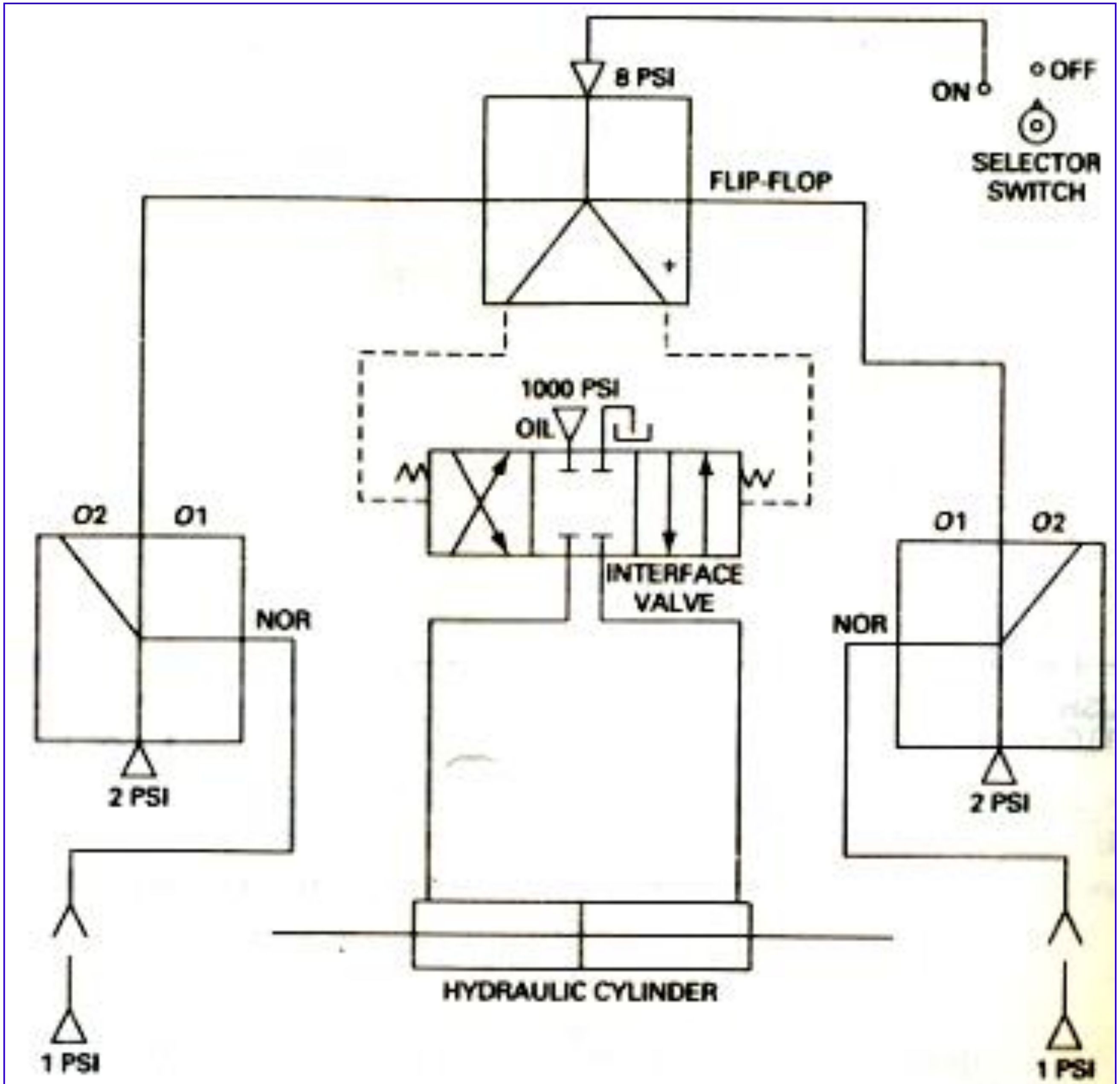
Control of Air cylinder using preferenced Flip flop



Sequencing circuit using preferenced Flip flop OR/NOR Monostable device



Continuous reciprocation of Hydraulic cylinder



Example 2 : Two cylinders are used to transfer parts from a magazine onto a chute (Figure 1.12). When a push button is pressed, the first cylinder extends. Pushing the part from the magazine and positions it in preparation for transfer by the second cylinder onto the out feed chute. Once the part is transferred, the first cylinder retracts, followed by the second. Confirmation of all extended and retracted positions are required.

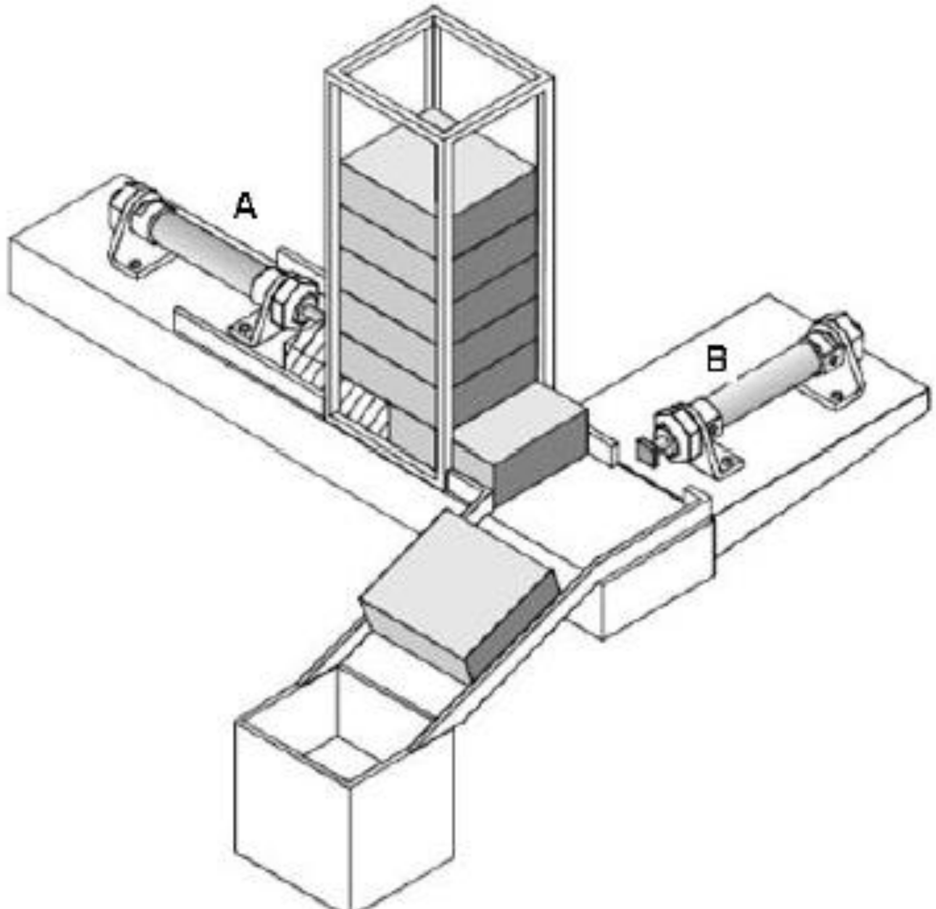


Figure 1.12 Positional diagram

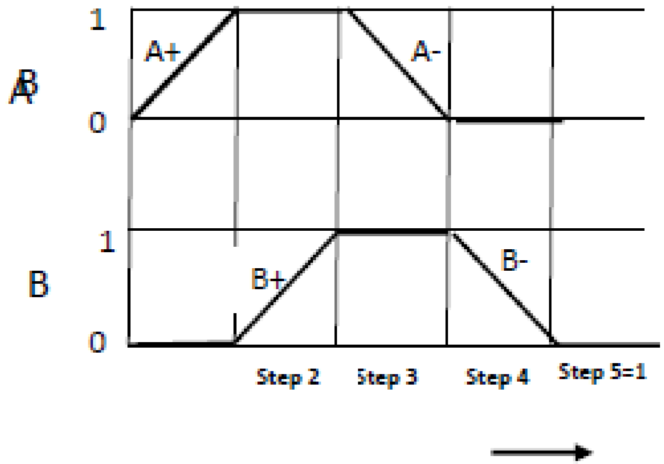


Figure 1.14 Displacement step diagram

Step 5 Draw the Displacement –time diagram (Figure 1.15)

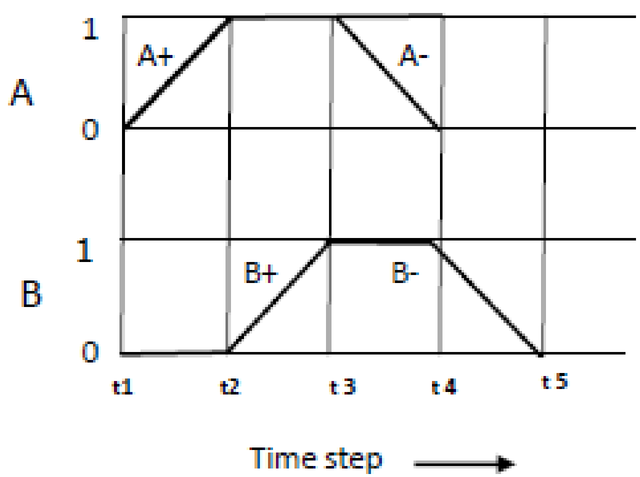


Figure 1.15 Displacement time diagram

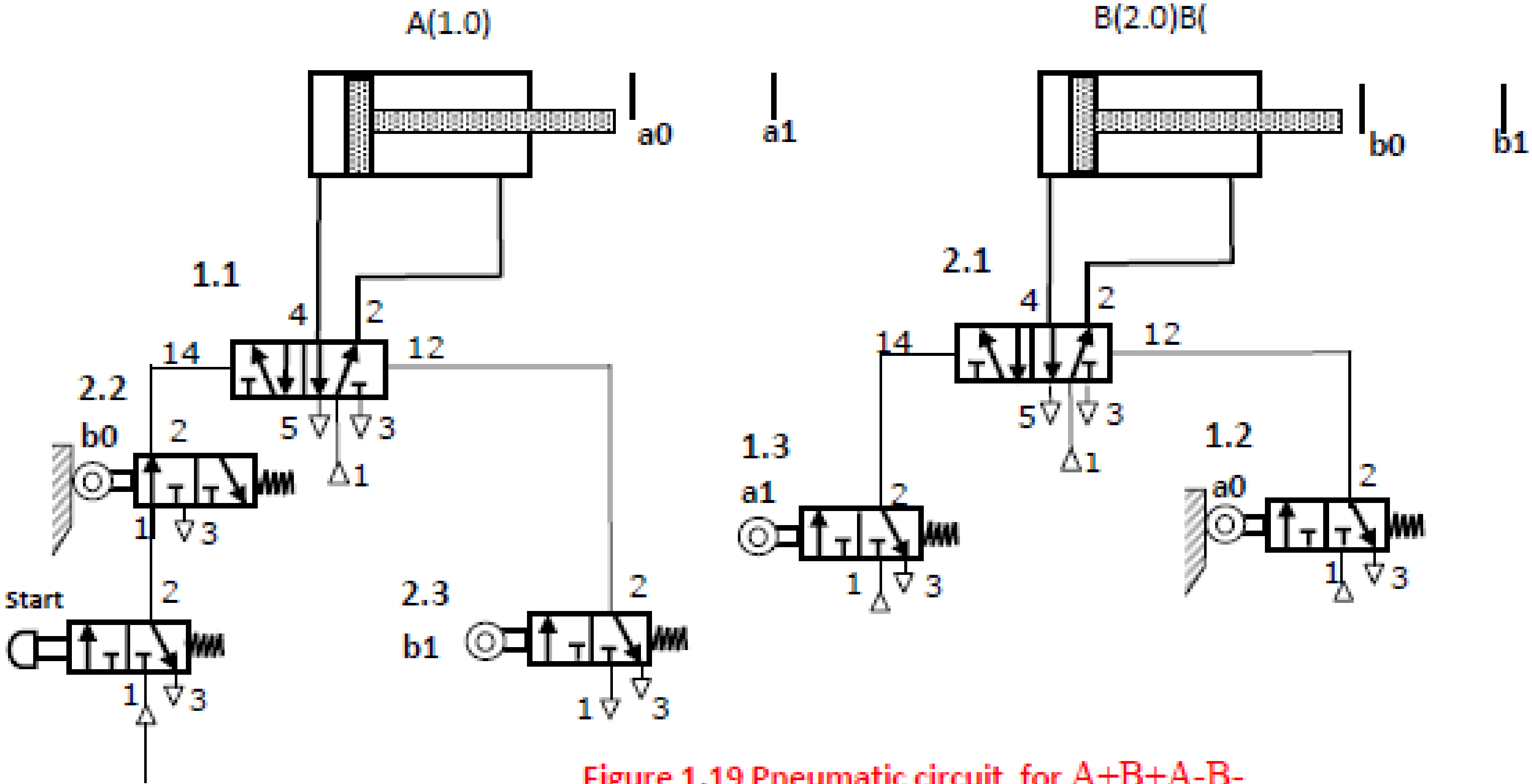


Figure 1.19 Pneumatic circuit for A+B+A-B-

1.2.3 Elimination of Signal Conflict

Various methods are used to solve problem of signal conflicts in multi cylinder circuits.

a) Idle return roller

b) Reversing valves (memory valves)

c) Modules as combination of valves

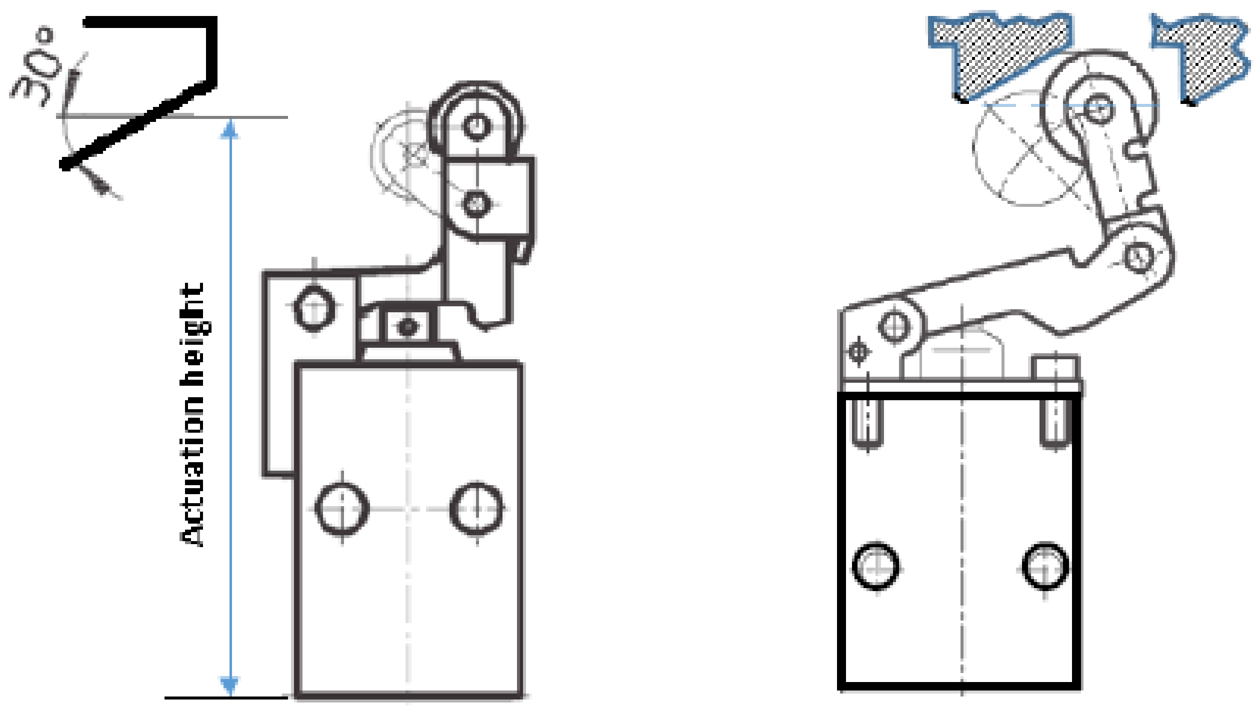


Figure 1.25 Two designs of Idle return rollers

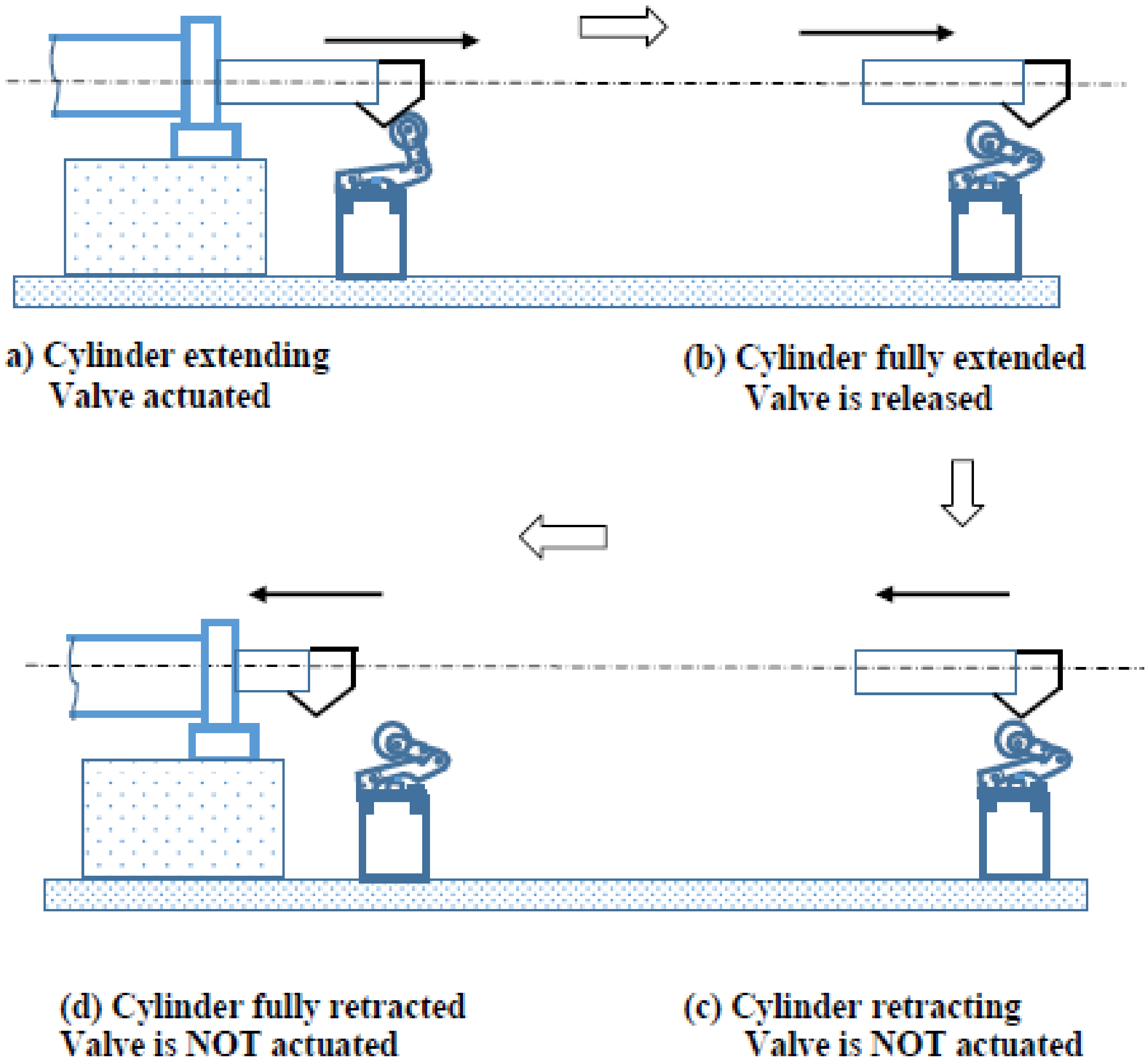
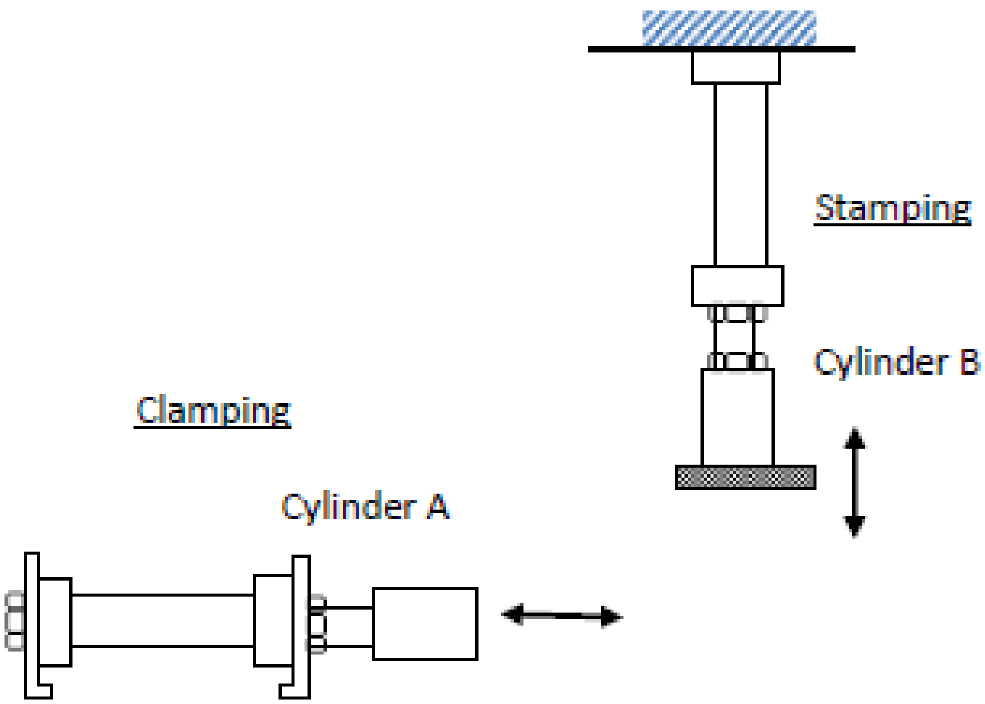


Figure 1.26 Actions of Idle return rollers

First cylinder A extends and brings under stamping station where cylinder B is located. Cylinder B then extends and stamps the job. Cylinder A can return back only cylinder B has retracted fully.

Step 2: Draw the positional layout. (Figure 1.28)



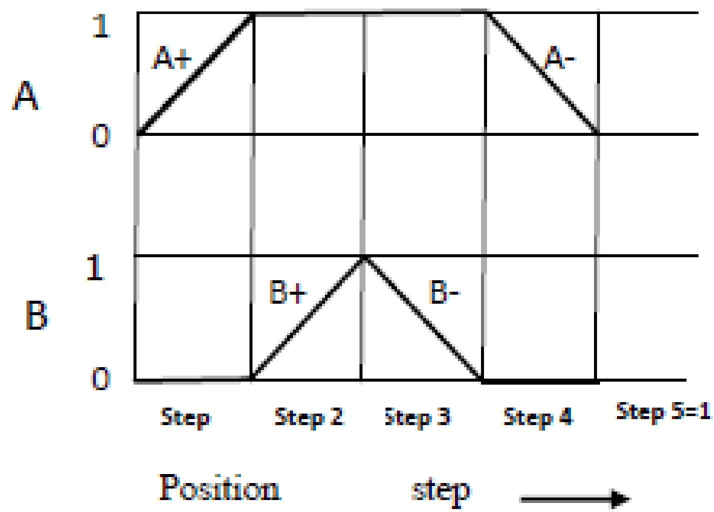
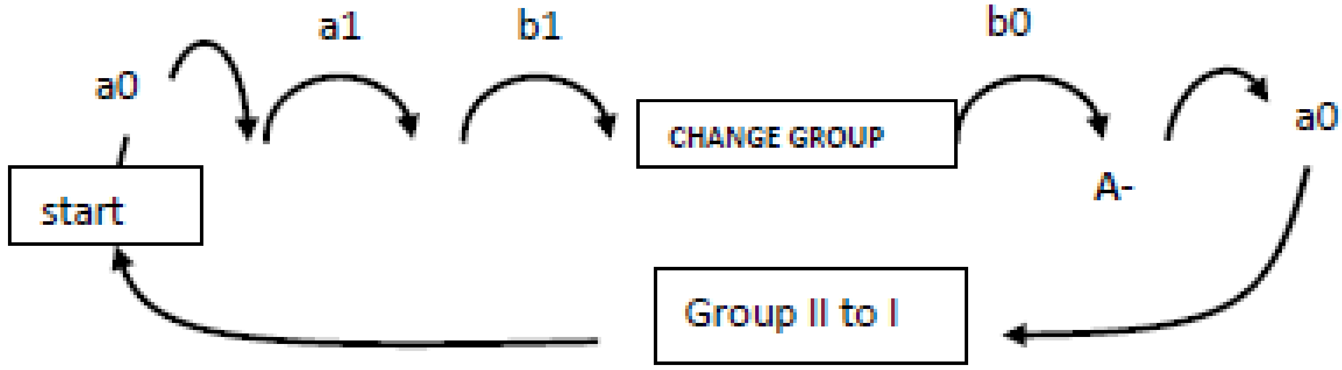
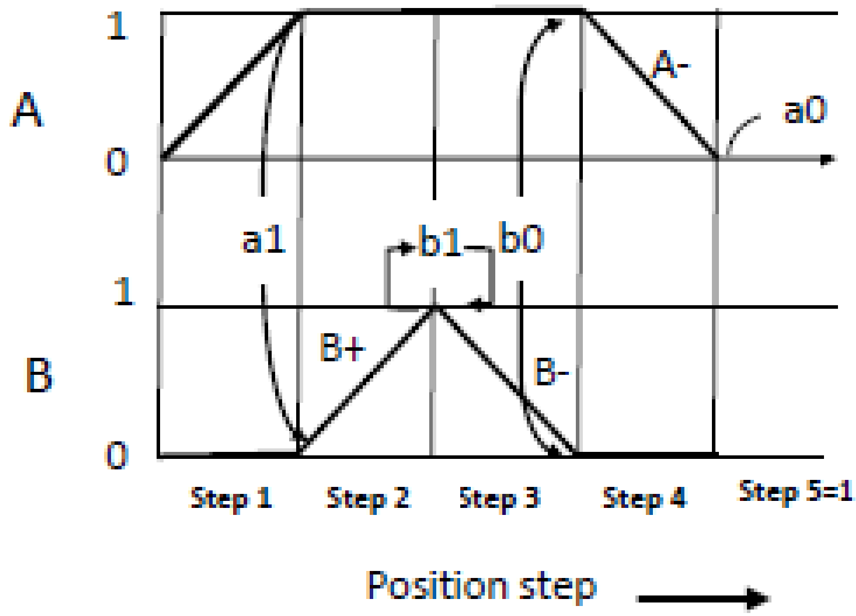


Figure 1.29 Displacement step diagram

Figure 1.28 Positional diagram





In our example of A+ B+ B- A- we can group as

A+ B+	B- A-
Group 1	Group 2

Figure 1.31 Displacement time diagram

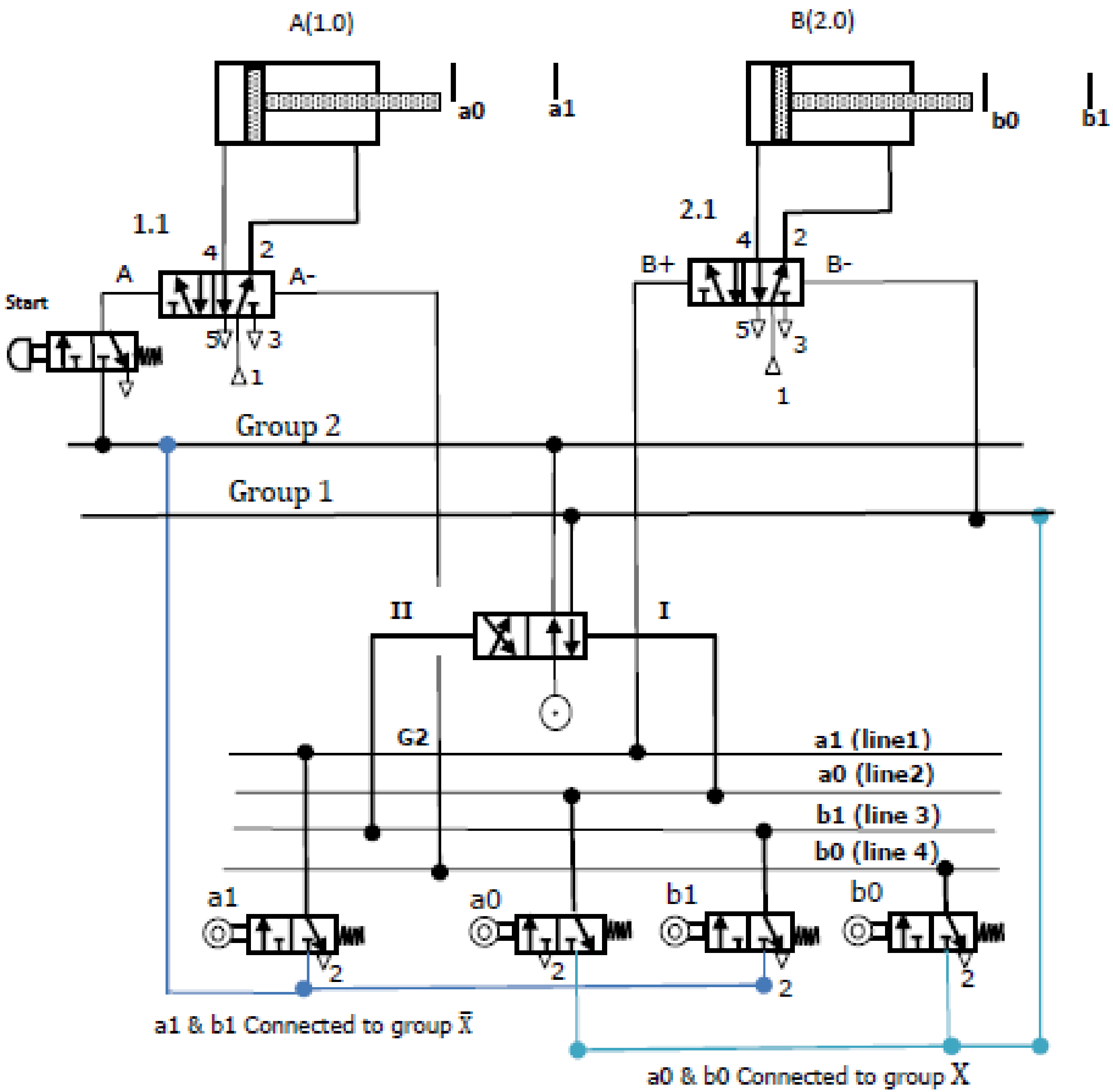


Figure 1.35 Pneumatic circuits for A+ B+ B- A-

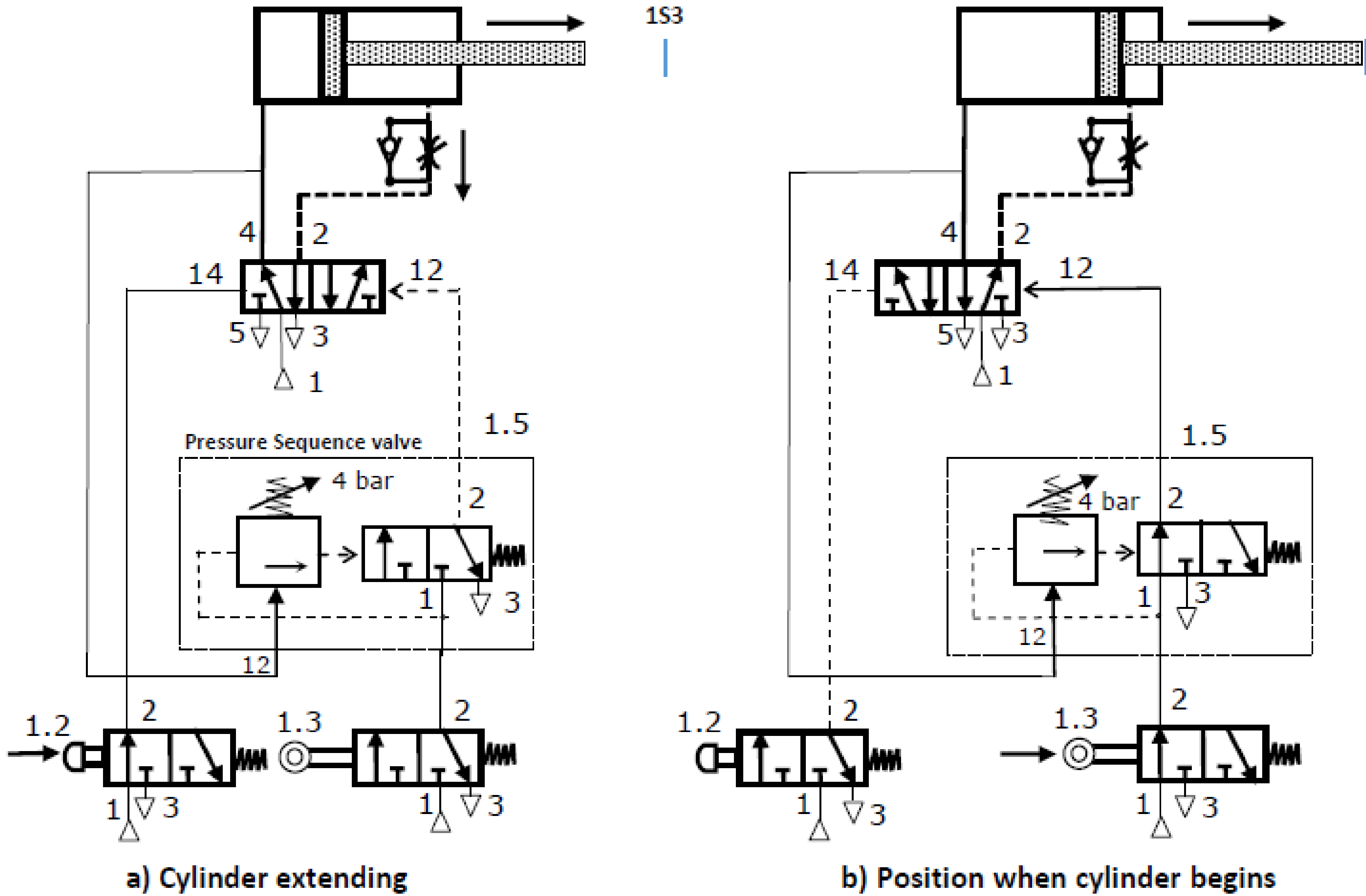
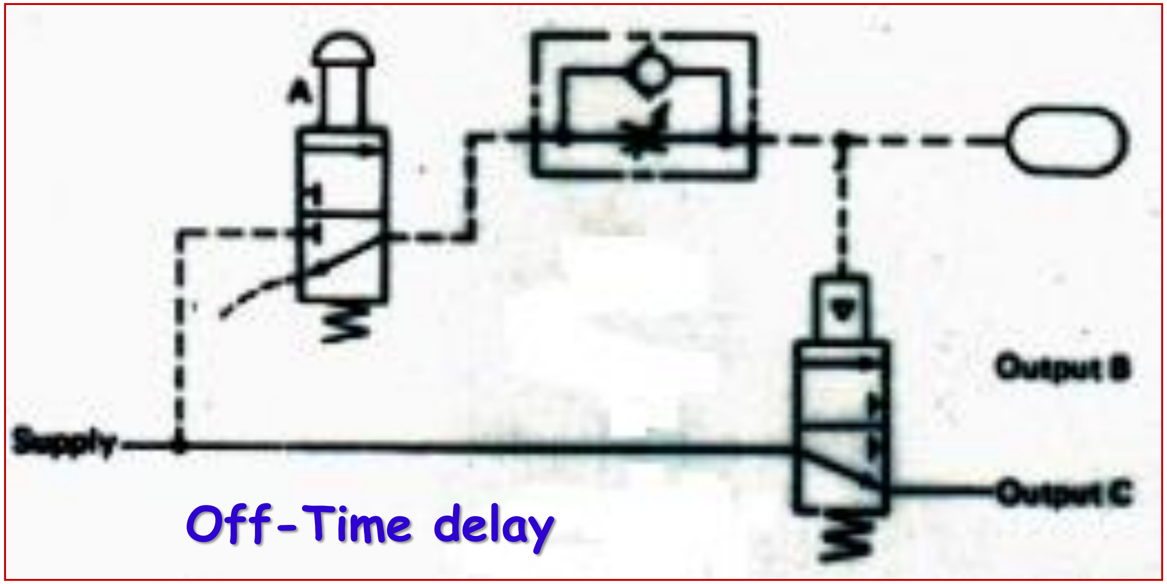


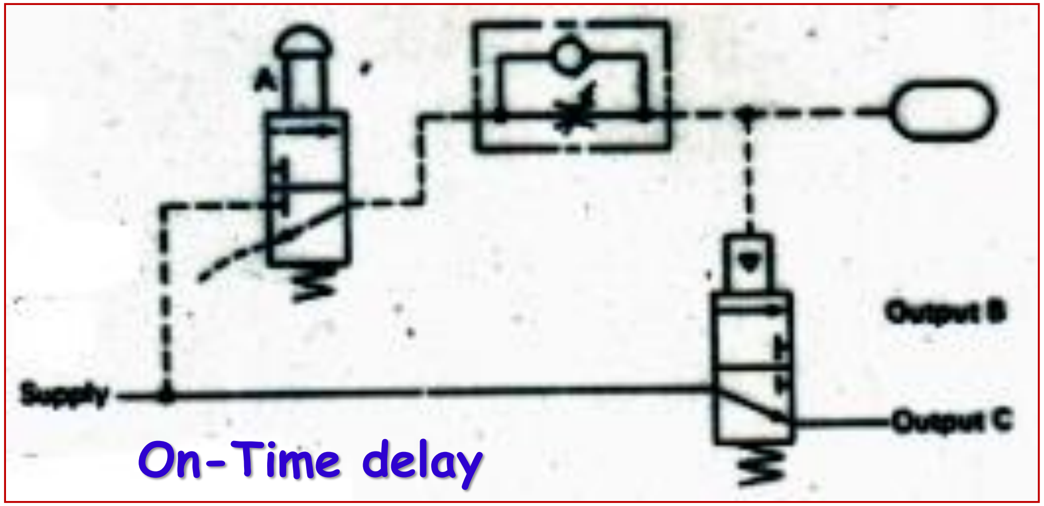
Figure 1.18 Pressure dependant control valve

Timing-out Circuit or Limited Memory Circuit



Time functions are introduced to delay operation of the circuit, usually by placing a resistance in the circuit.

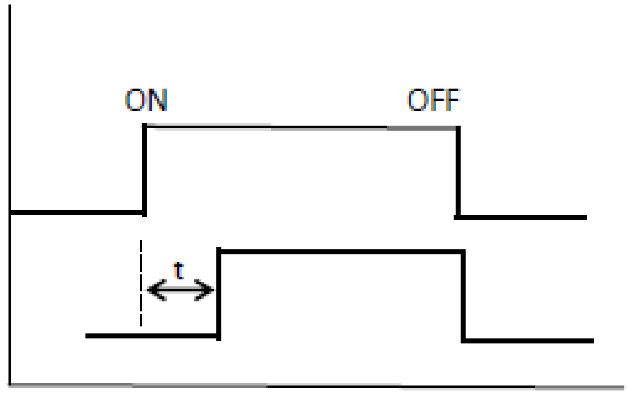
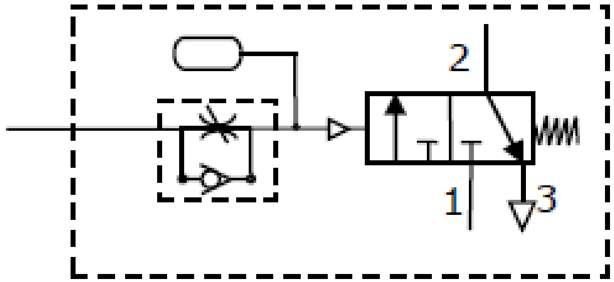
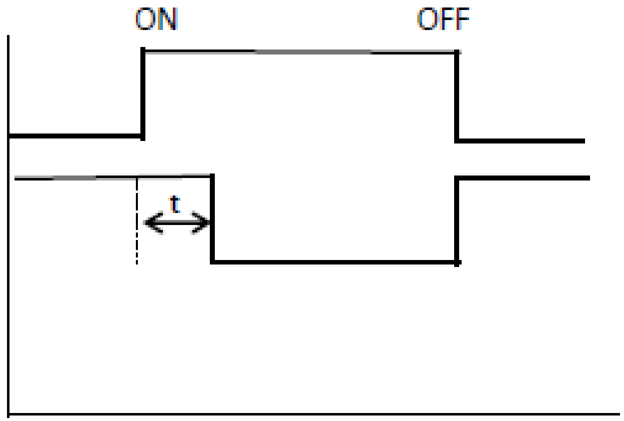
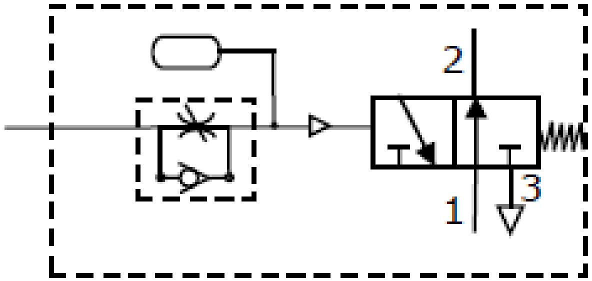
Timing-in Circuit

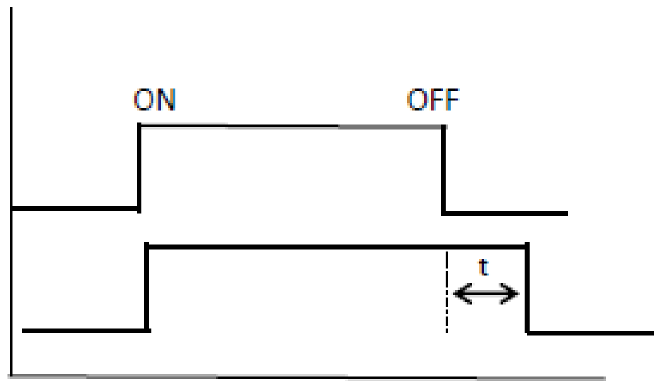


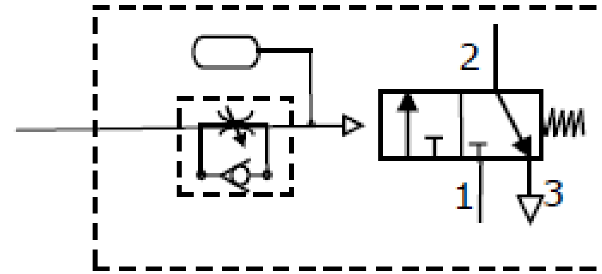
<u>Unlimited MEMORY:</u>
Command signal reverts by manually

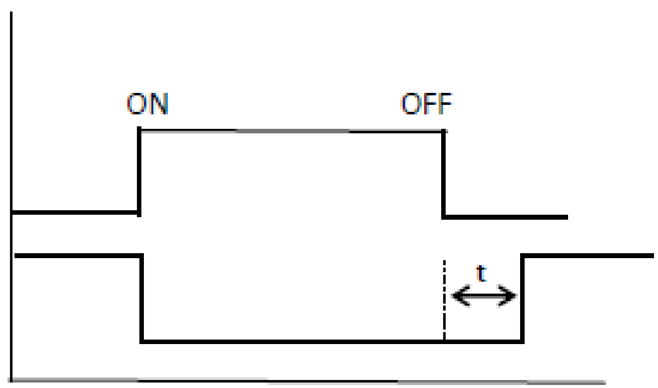
<u>Limited MEMORY:</u>
Command signal reverts by automatically

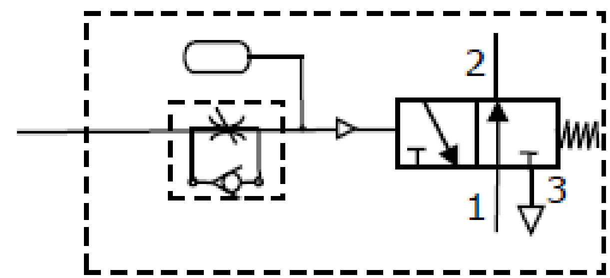
Table 1.1 Timing diagrams for all four type of Pneumatic delay valve

Timing diagram	Symbol
	
	

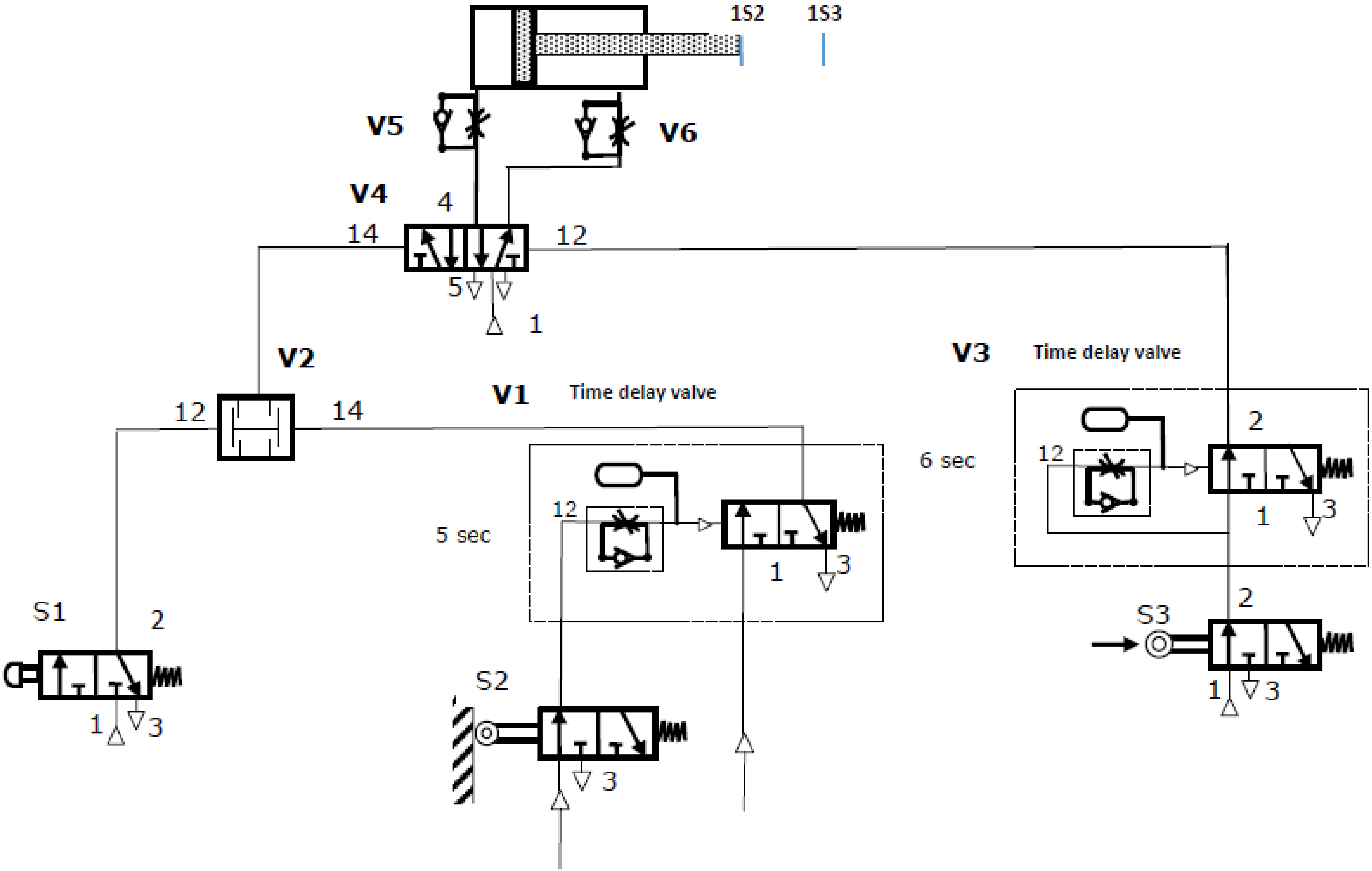

ON
OFF
t


2
1
3


ON
OFF
t


2
1
3

Example 12: A double acting cylinder is used to press together glued components . Upon operation of a press button, the clamping cylinder slowly advances. Once the fully extended position is reached , the cylinder is to remain for a time of t = 6 seconds and then immediately retract to the initial position. A new start cycle is only possible after the cylinder has fully retracted and after a delay of 5 seconds. During this delay the finished part is manually removed and replaced with new parts for gluing. The retracting speed should be fast, but adjustable.



1S2
1S3
V5
V6
V4
4
14
12
5
1
V2
12
14
V1
Time delay valve
5 sec
12
1
3
V3
Time delay valve
6 sec
12
2
1
3
S1
2
1
3
S2
3
S3
2
1
3

ME 5451 – Hydraulics and Pneumatics

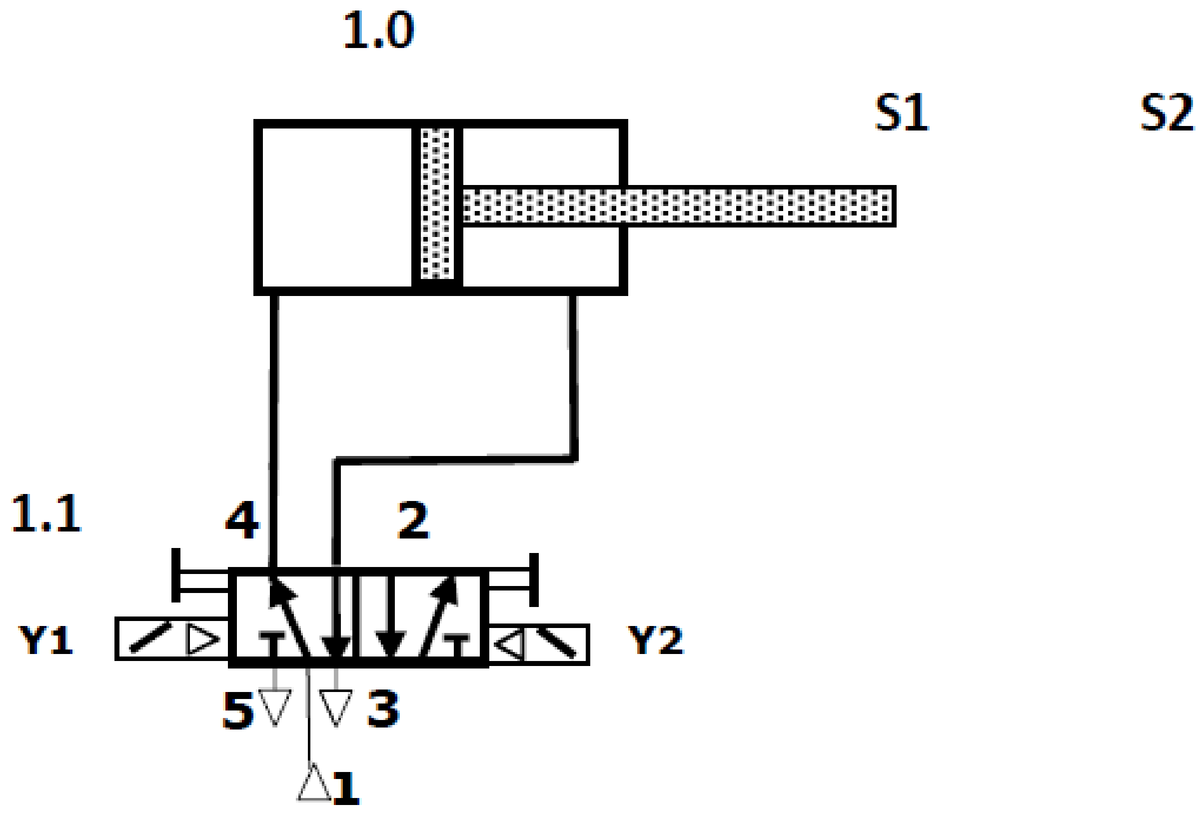
Lecture -22

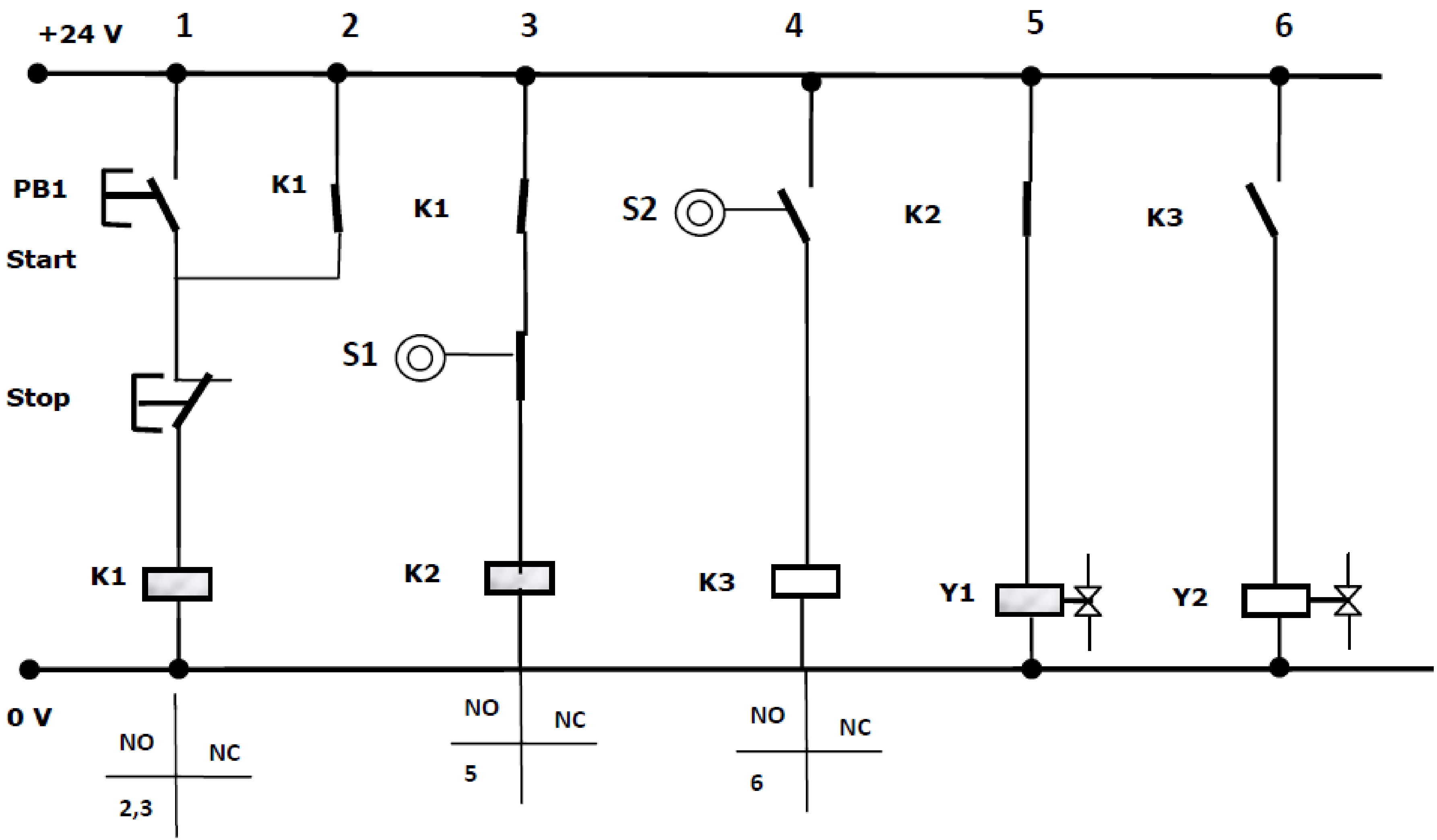
Date: 28-05-2021 Time slot: 08:30-10:10 a.m.

Contents

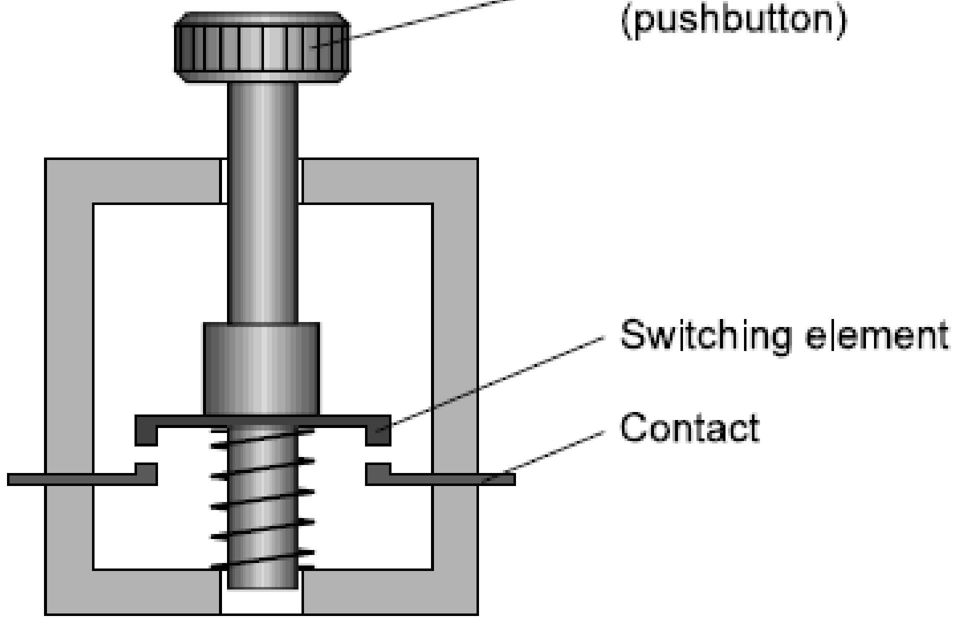
1. Review of lecture 21
2. Electro Pneumatics circuits

Course Instructor: Dr. A. Siddharthan


1.0
S1
S2
1.1
4
2
Y1
Y2
5
3
1



+24 V
1
2
3
4
5
6
PB1
Start
K1
K1
S2
K2
K3
S1
Stop
K1
K2
K3
Y1
Y2
0 V
NO
NC
2,3
NO
NC
5
NO
NC
6



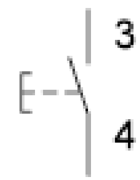
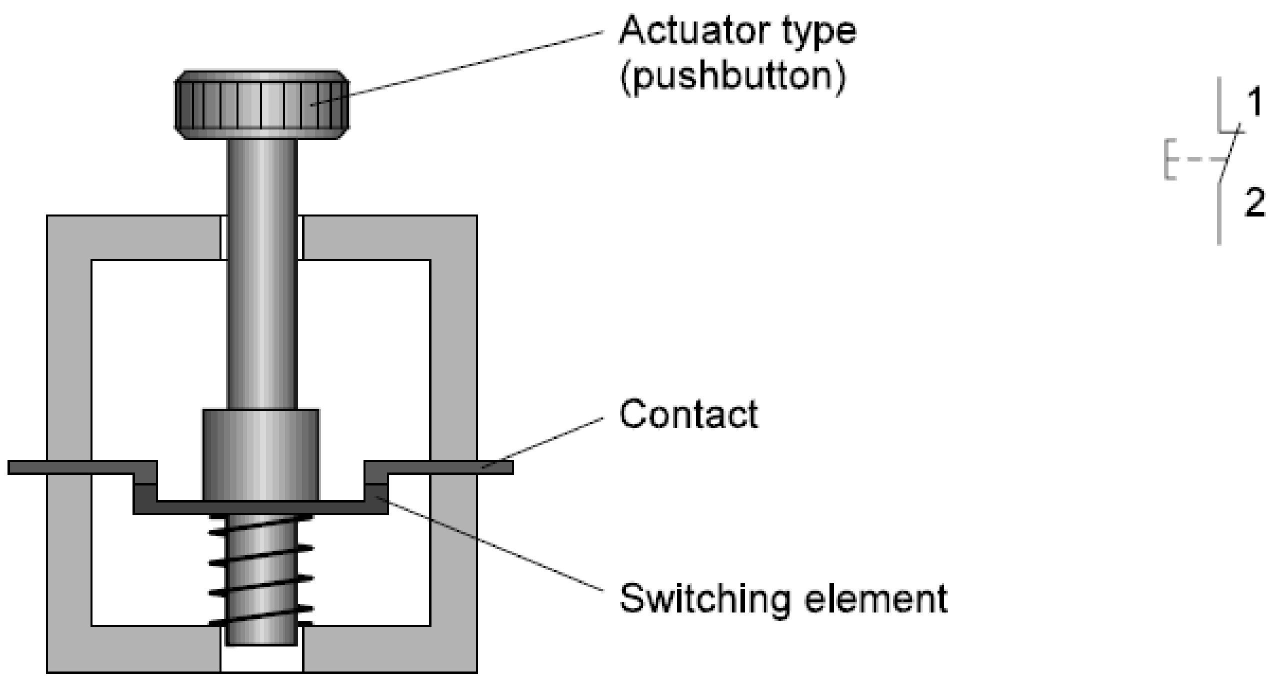


Fig. 3.2:
Normally open contact (make)
– section and symbol

Fig. 3.3:
Normally open contact (break) – section and symbol



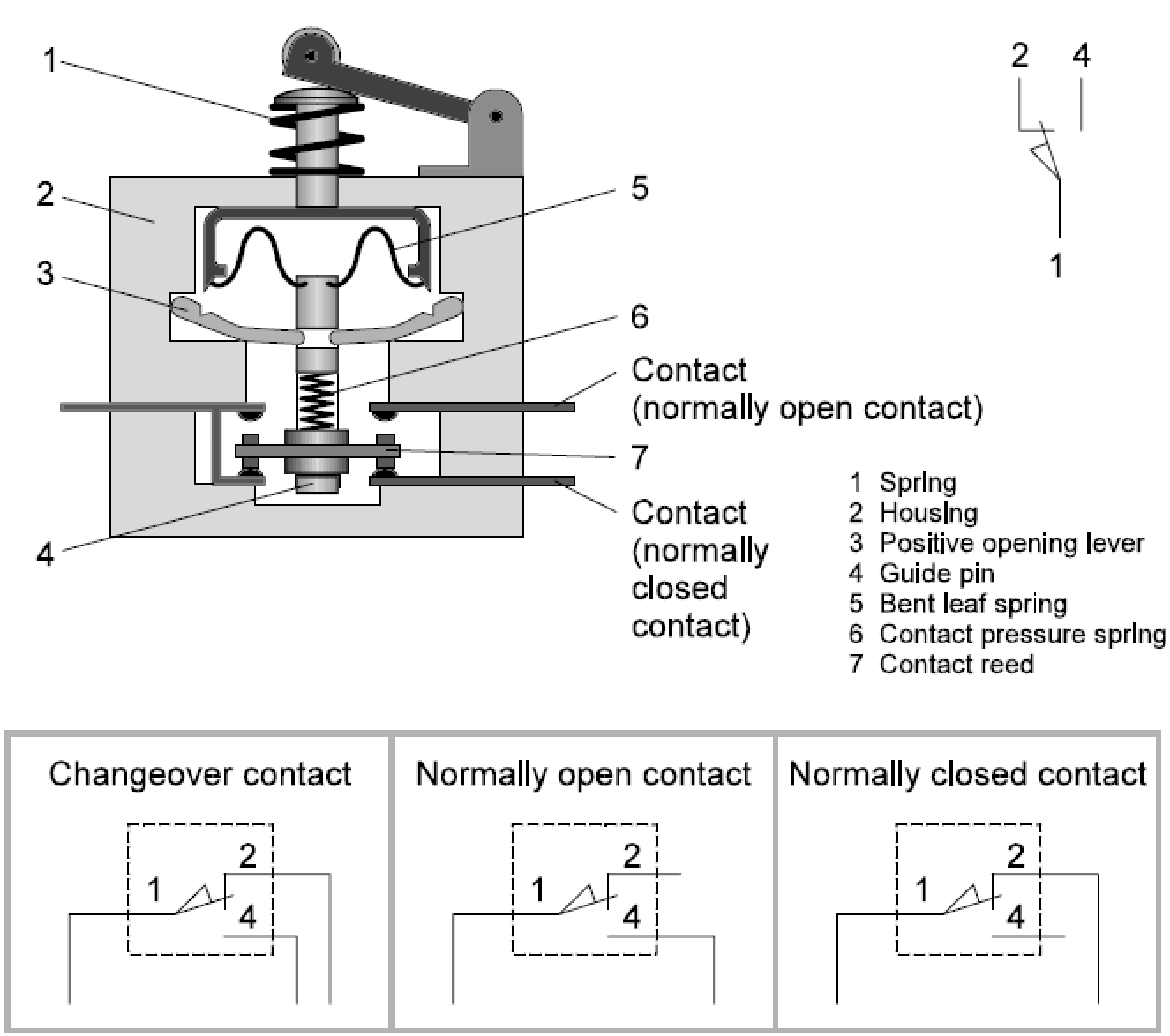


Fig. 3.5: Mechanical limit switch: construction and connection possibilities

Fig. 3.6:
Reed switch
(normally open contact)

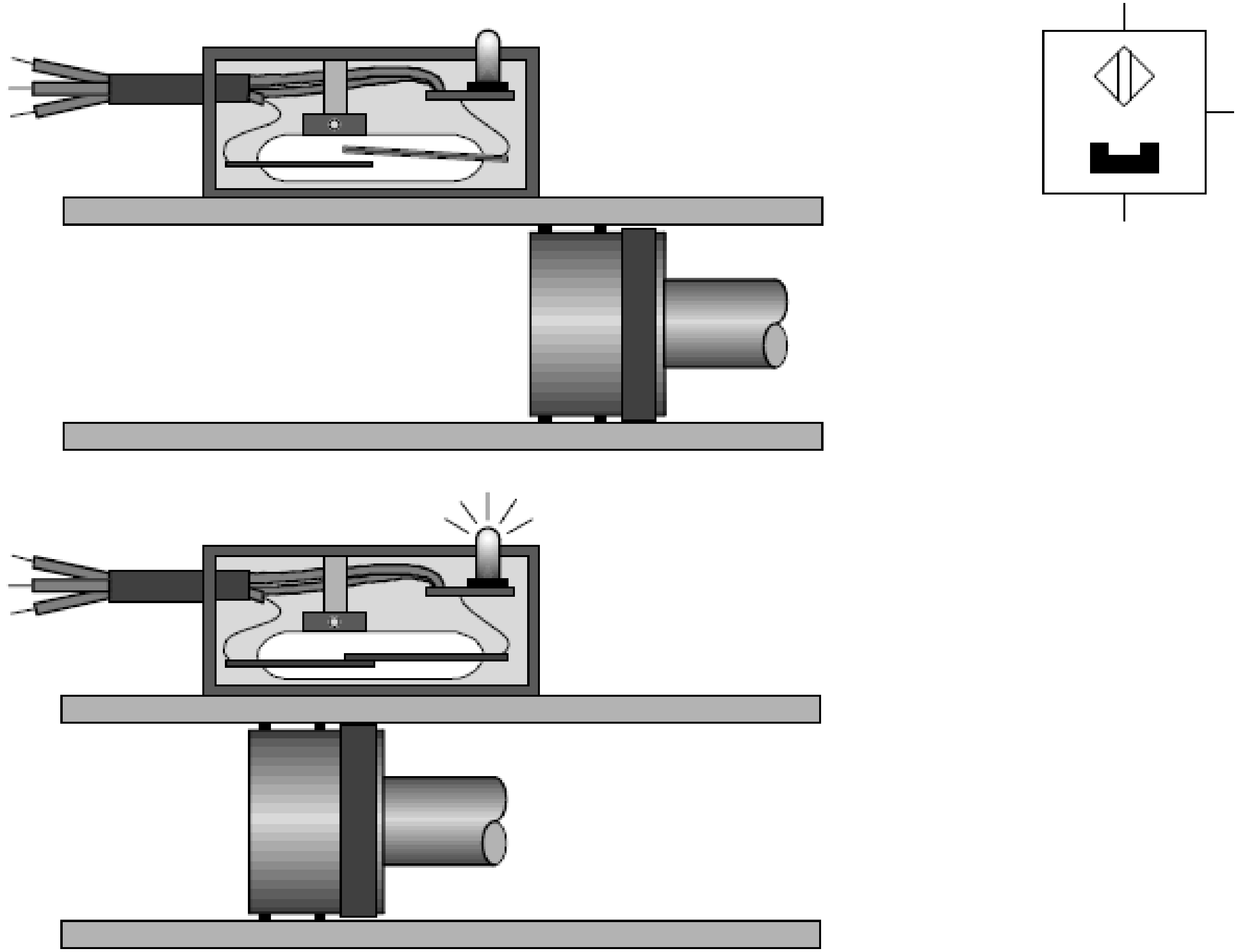
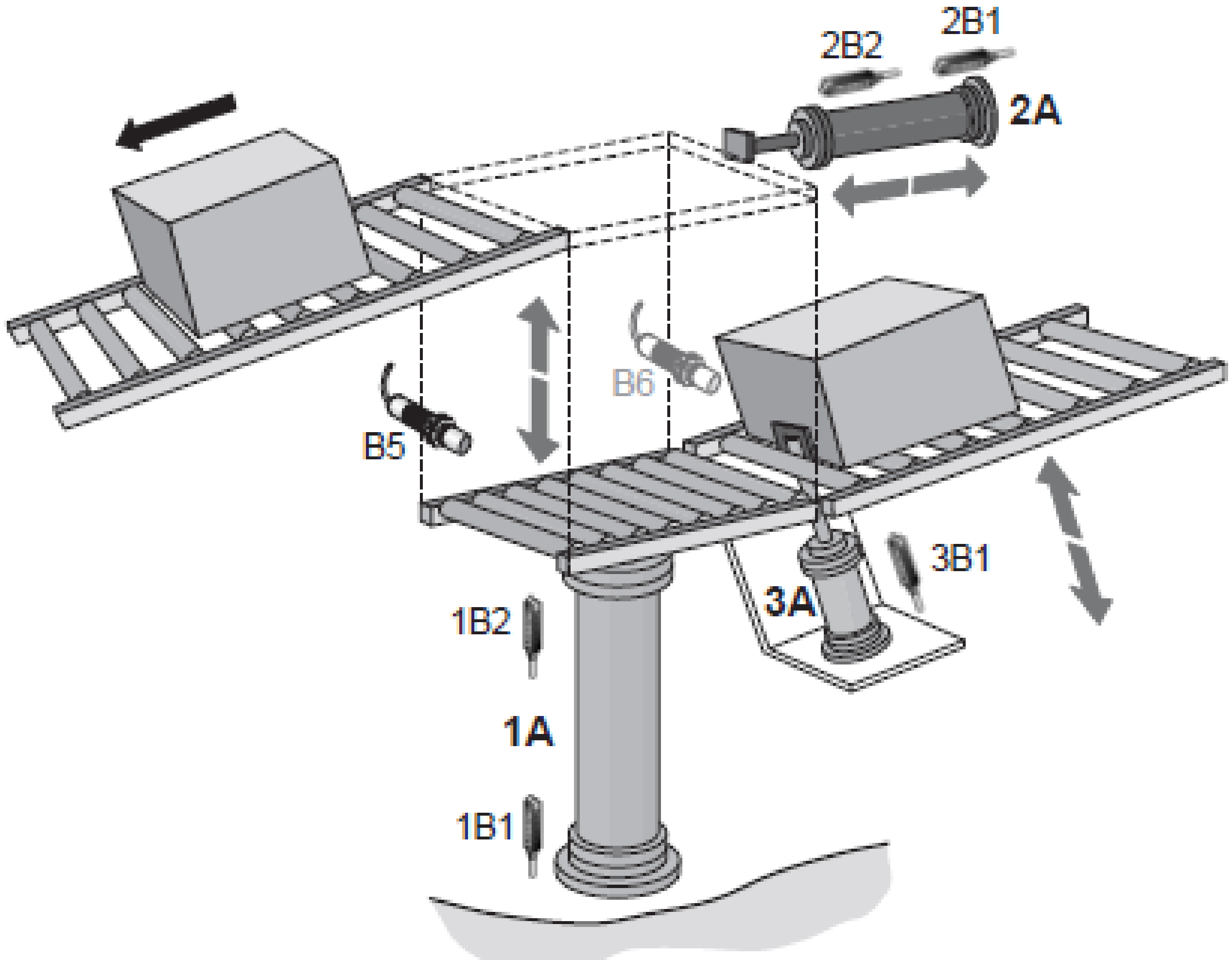


Fig. 5.2:
Positional sketch of the lifting device



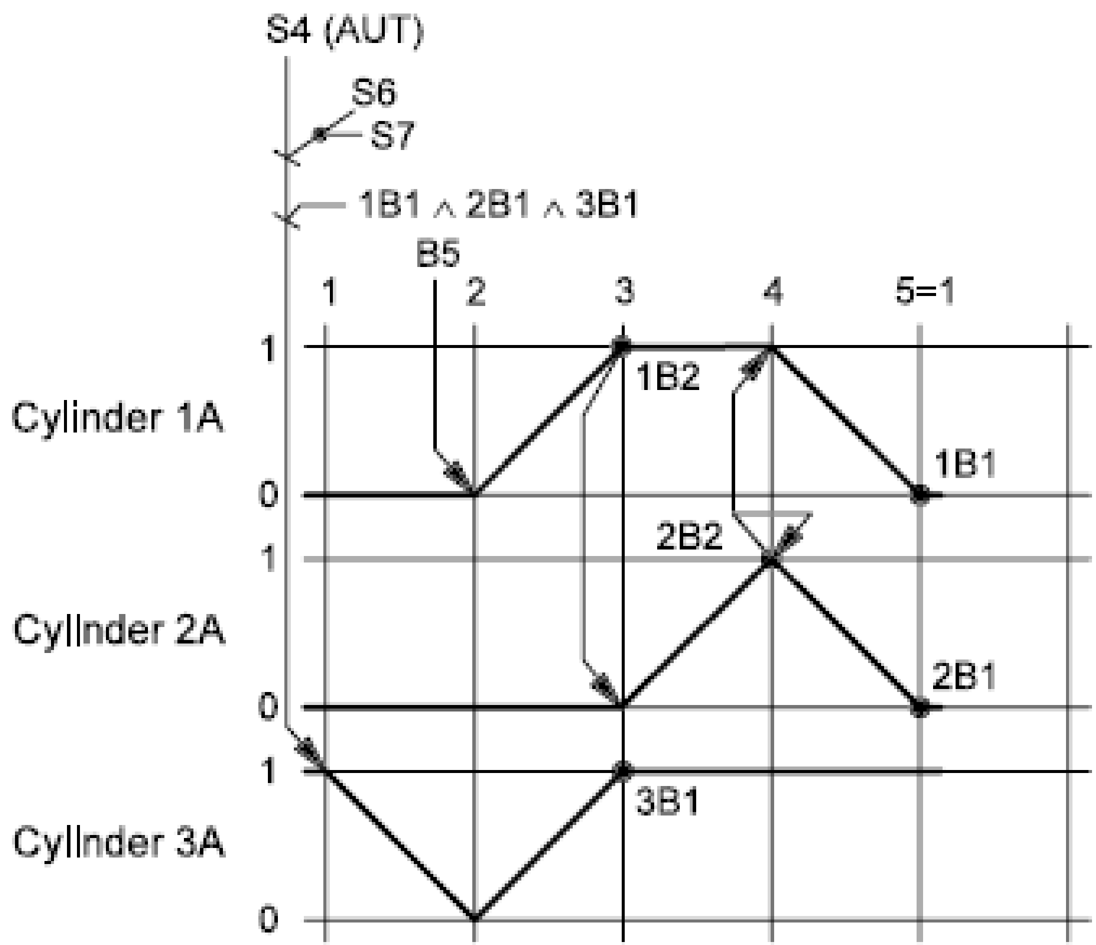
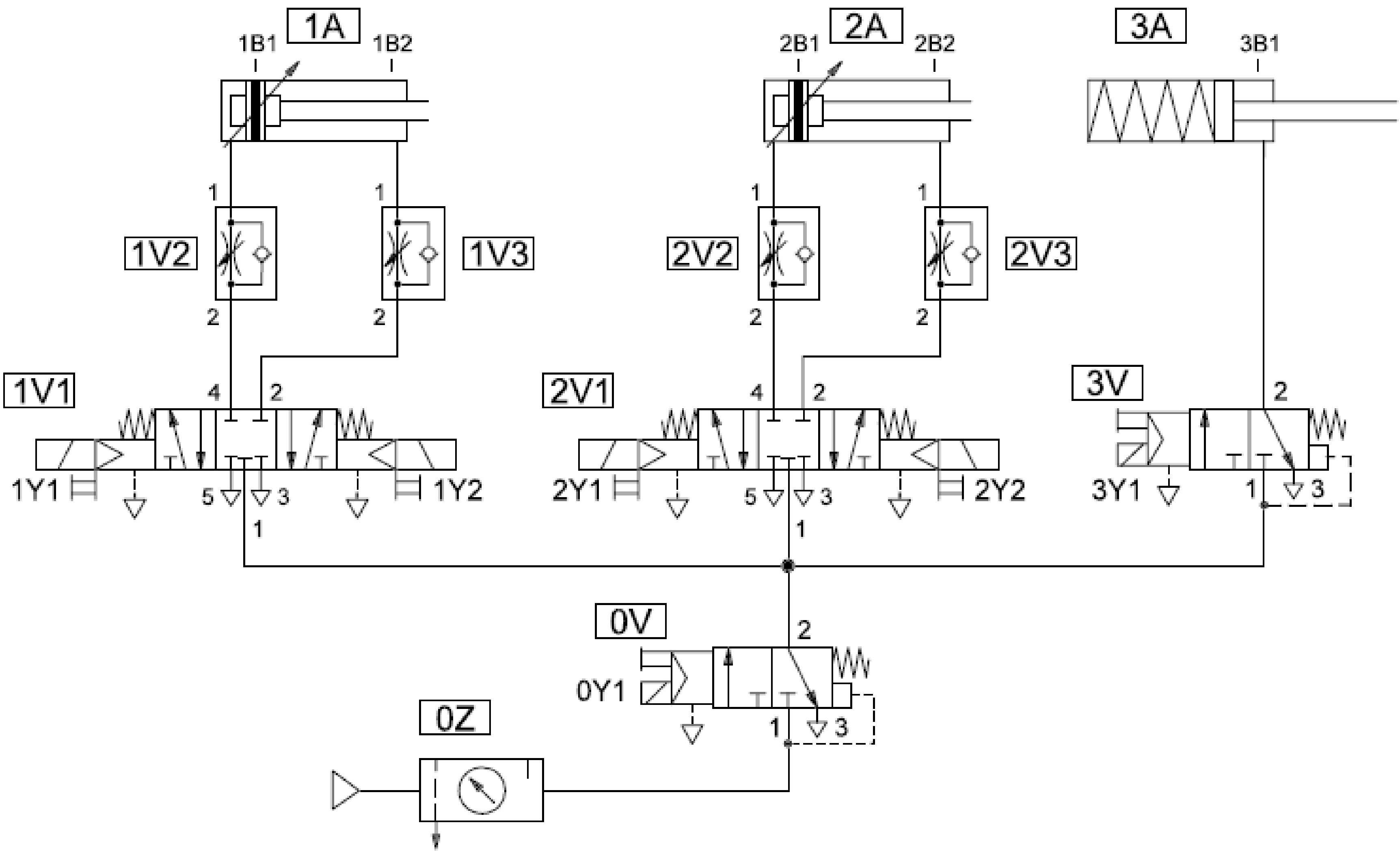
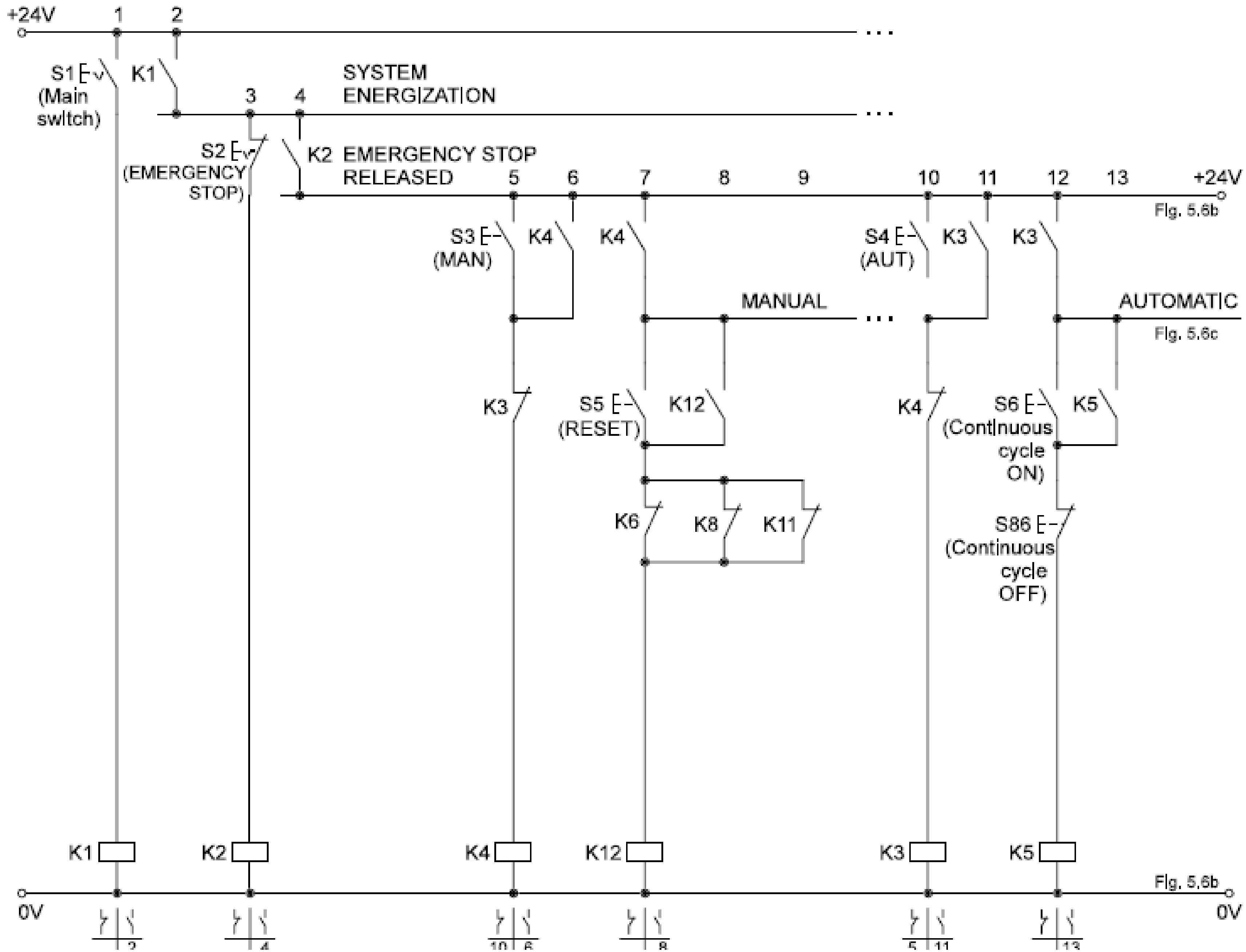


Fig. 5.4:
Displacement-step diagram for the lifting device

Fig. 5.5:
Pneumatic circuit diagram
of the lifting device

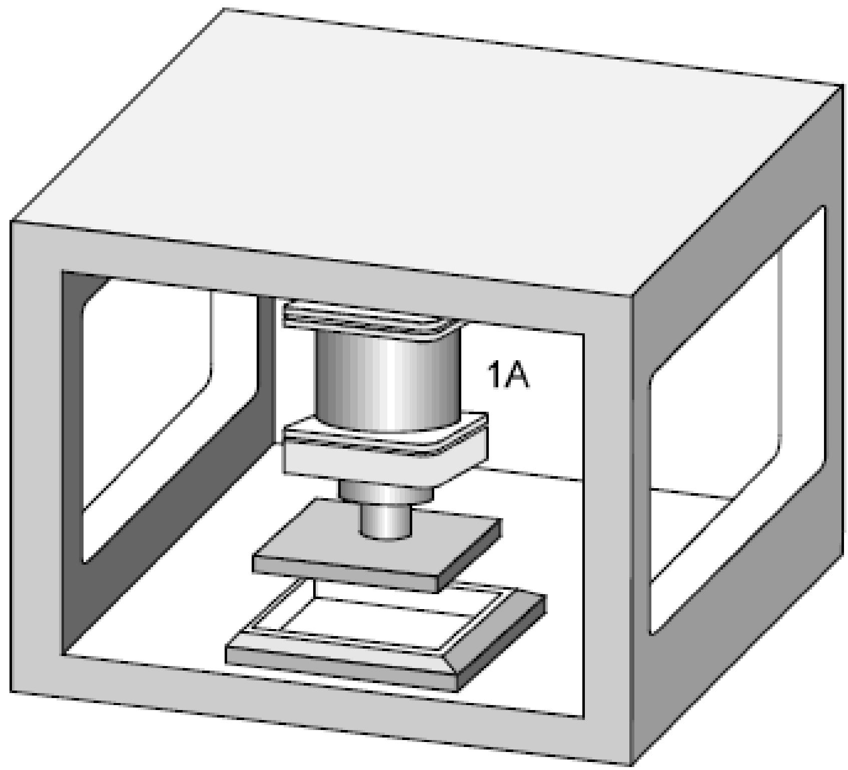


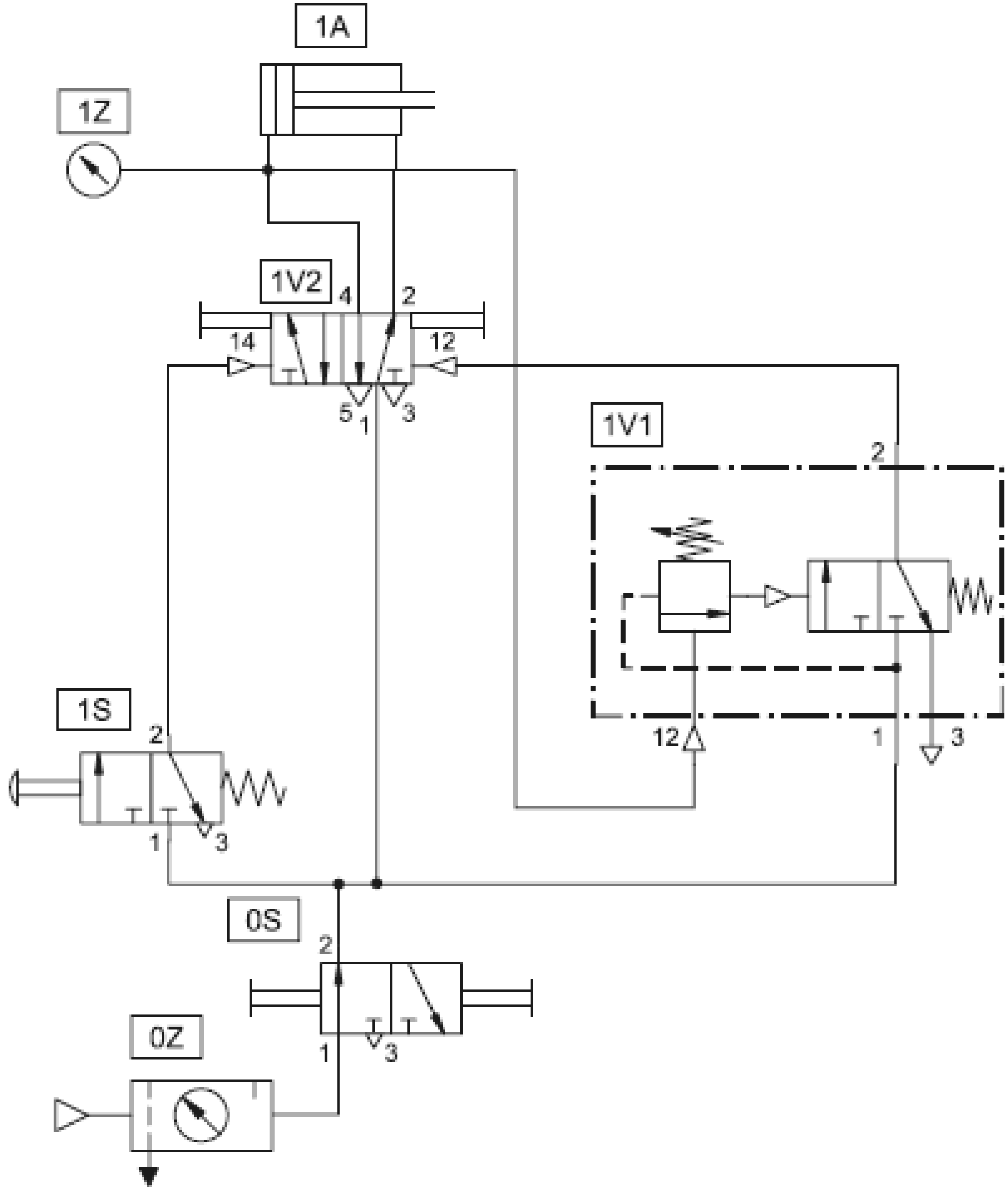

+24V
1
2
S1
(Main switch)
K1
3
4
SYSTEM ENERGIZATION
S2
(EMERGENCY STOP)
K2
EMERGENCY STOP RELEASED
5
6
7
8
9
10
11
12
13
+24V
Fig. 5.6b
S3
(MAN)
K4
K4
S4
(AUT)
K3
K3
MANUAL
AUTOMATIC
Fig. 5.6c
K3
S5
(RESET)
K12
K4
S6
(Continuous cycle ON)
K5
K6
K8
K11
S86
(Continuous cycle OFF)
K1
K2
K4
K12
K3
K5
0V
Fig. 5.6b
0V
2
4
10
6
8
5
11
13

The Problem

A plastic component is embossed using a die driven by a double-acting cylinder. The die is to advance and emboss the plastic when a push button is operated. The return of the die is to be effected when a preset pressure is reached. The embossing pressure is to be adjustable.

Fig. 5.24
Positional sketch





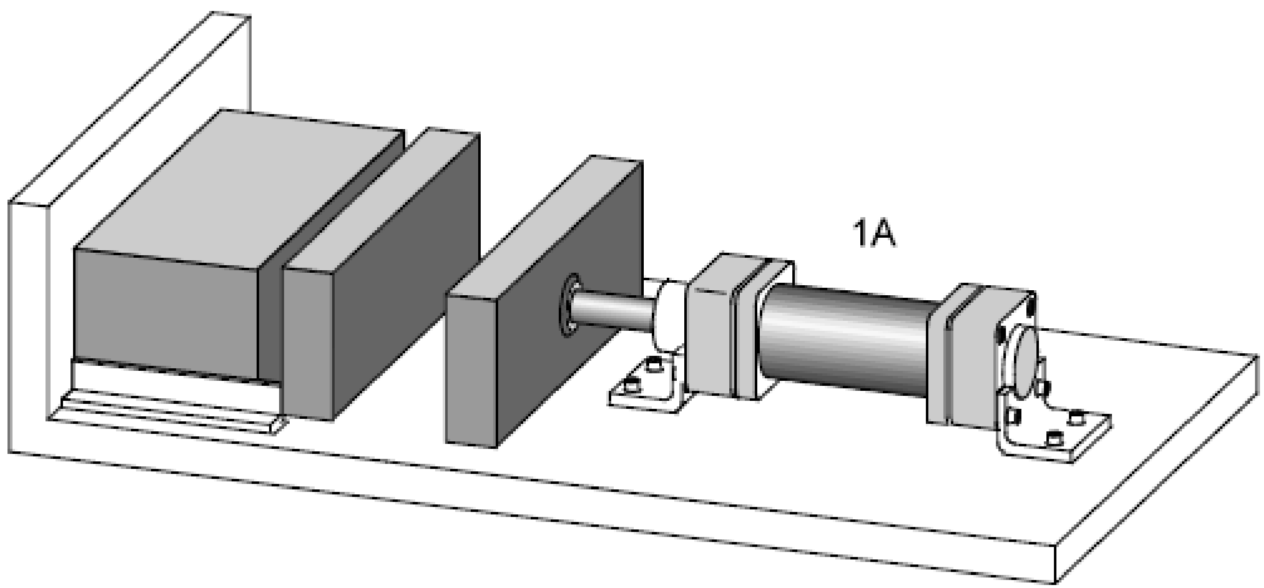
1A
1Z
1V2
4
2
14
12
5
1
3
1V1
2
12
1
3
1S
2
1
3
0S
2
1
3
0Z

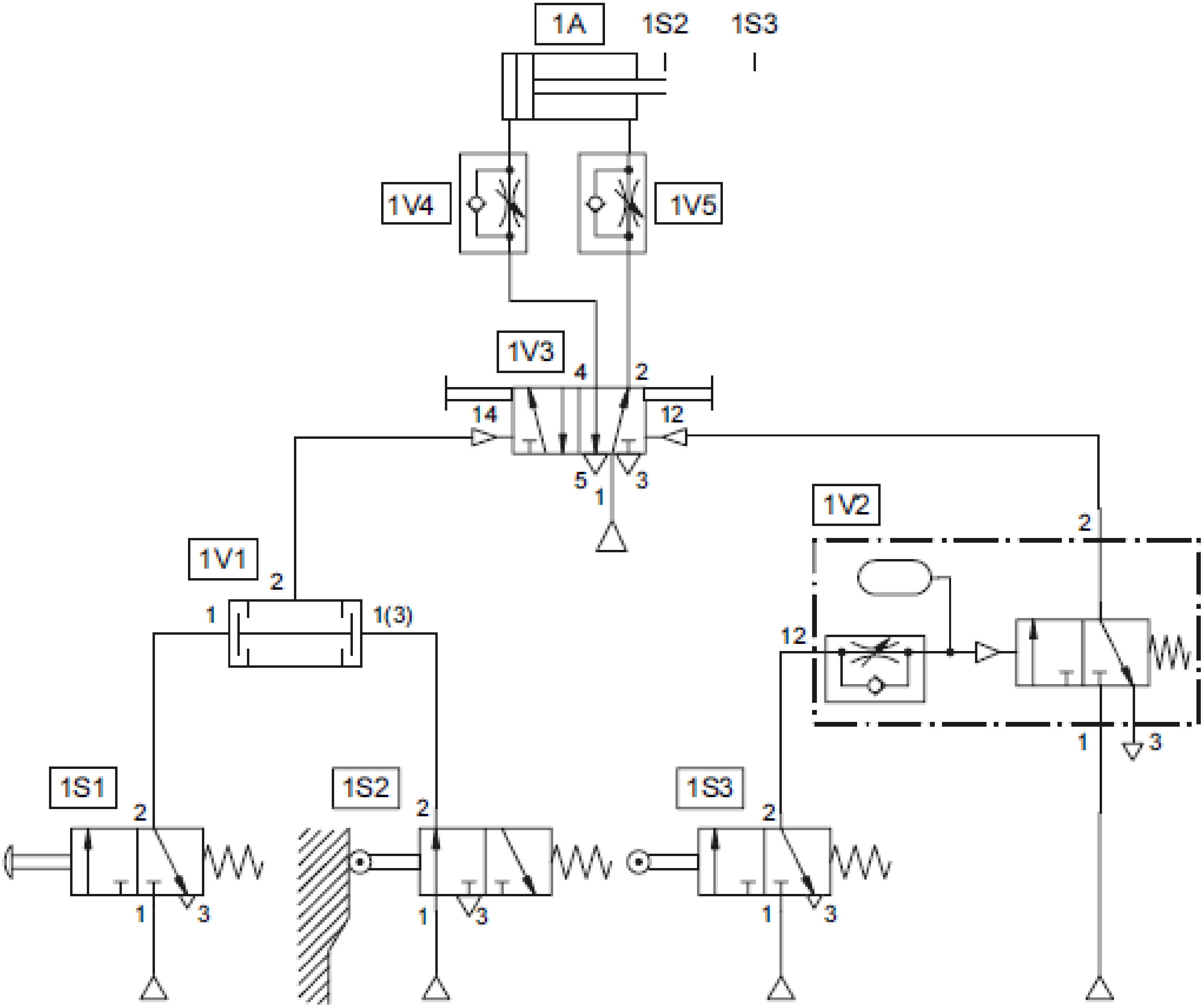
The Problem

A double-acting cylinder is used to press together glued components. Upon operation of a push button, the clamping cylinder extends.

Once the fully advanced position is reached, the cylinder is to remain for a time of T= 6 seconds and then immediately retract to the initial position. The cylinder retraction is to be adjustable. A new start cycle is only possible after the cylinder has fully retracted.

Fig. 5.28
Positional sketch





1A
1S2
1S3
1V4
1V5
1V3
4
2
14
12
5
1
3
1V2
2
1V1
2
1
1(3)
12
1
3
1S1
2
1
3
1S2
2
1
3
1S3
2
1
3