Fluid Power Control

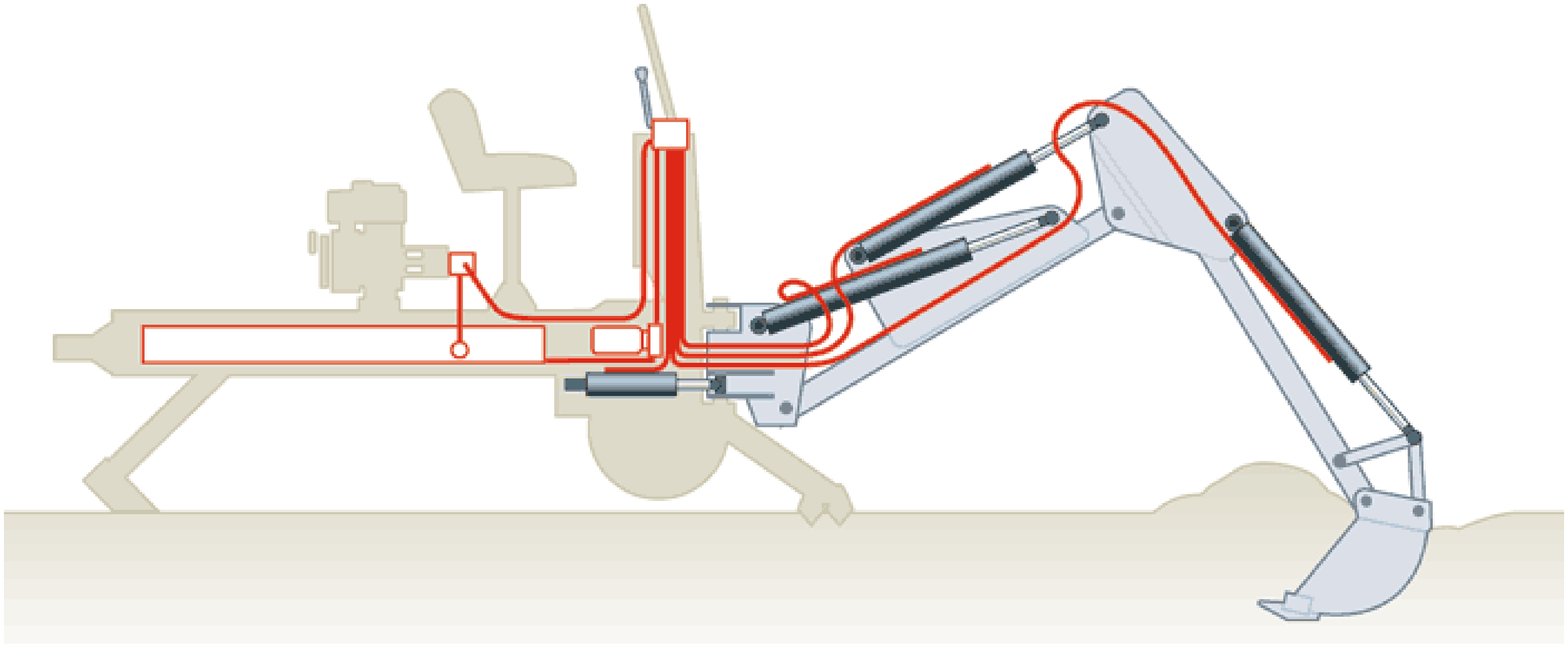
Ahmed Abu Hanieh

Introduction

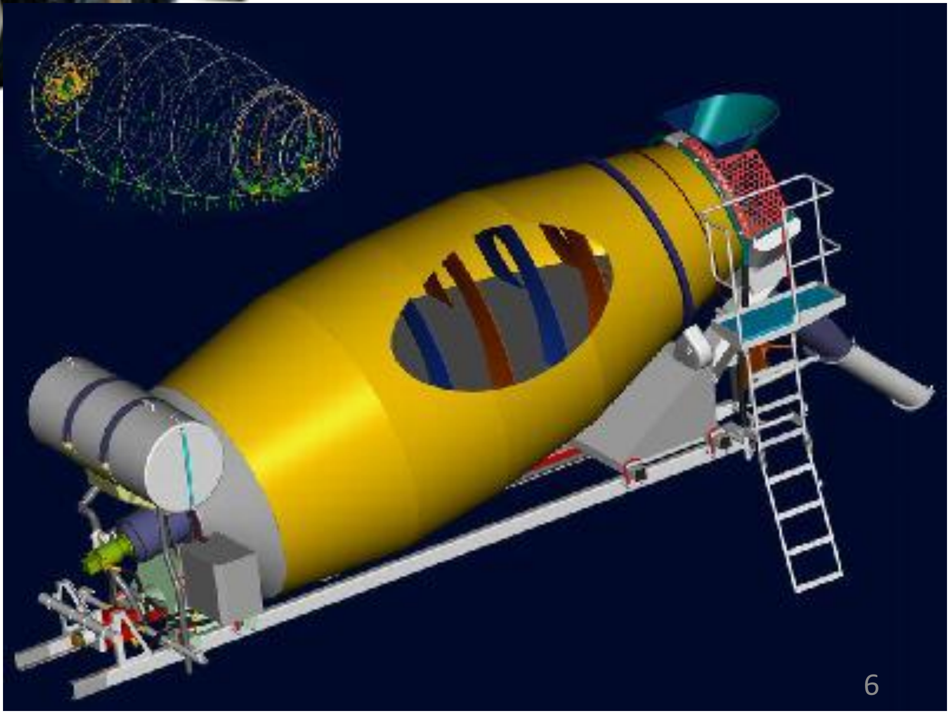
Hydraulic jack

Loader

Excavator

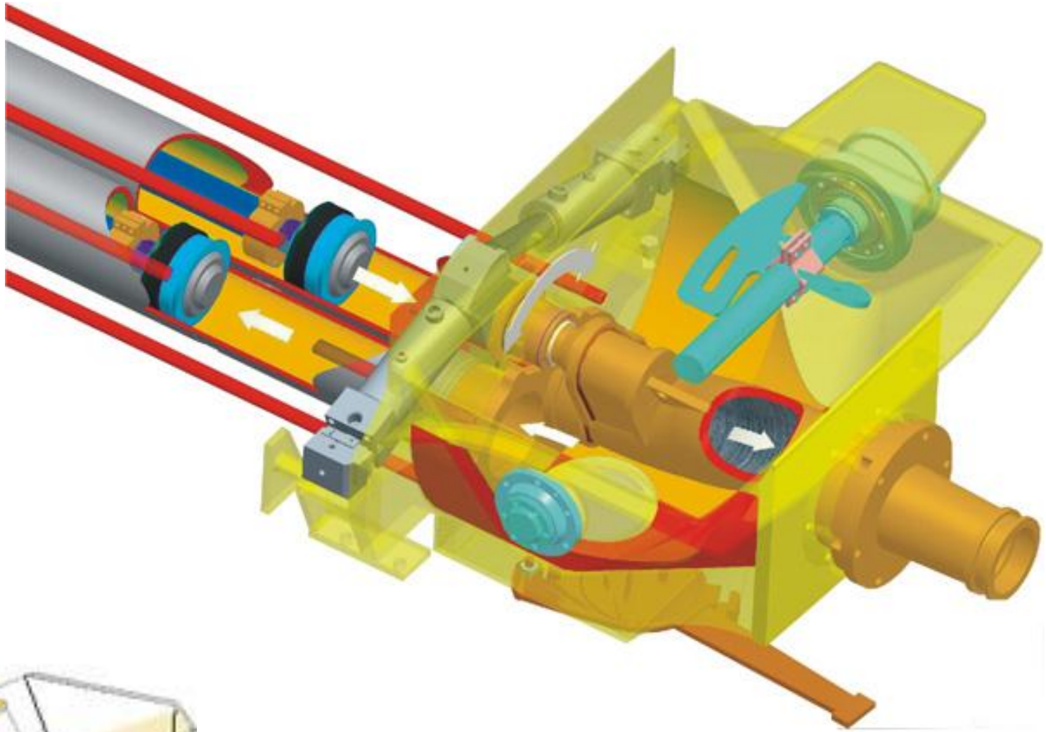


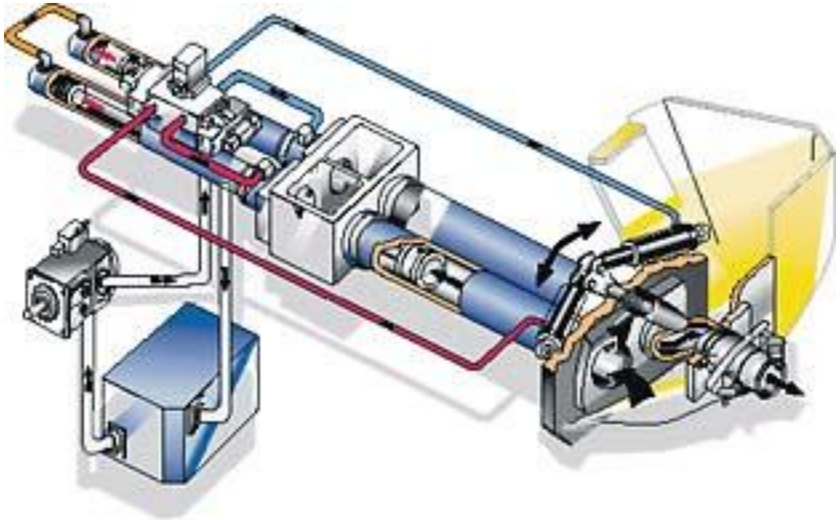
Concrete Mixer



Concrete Pump

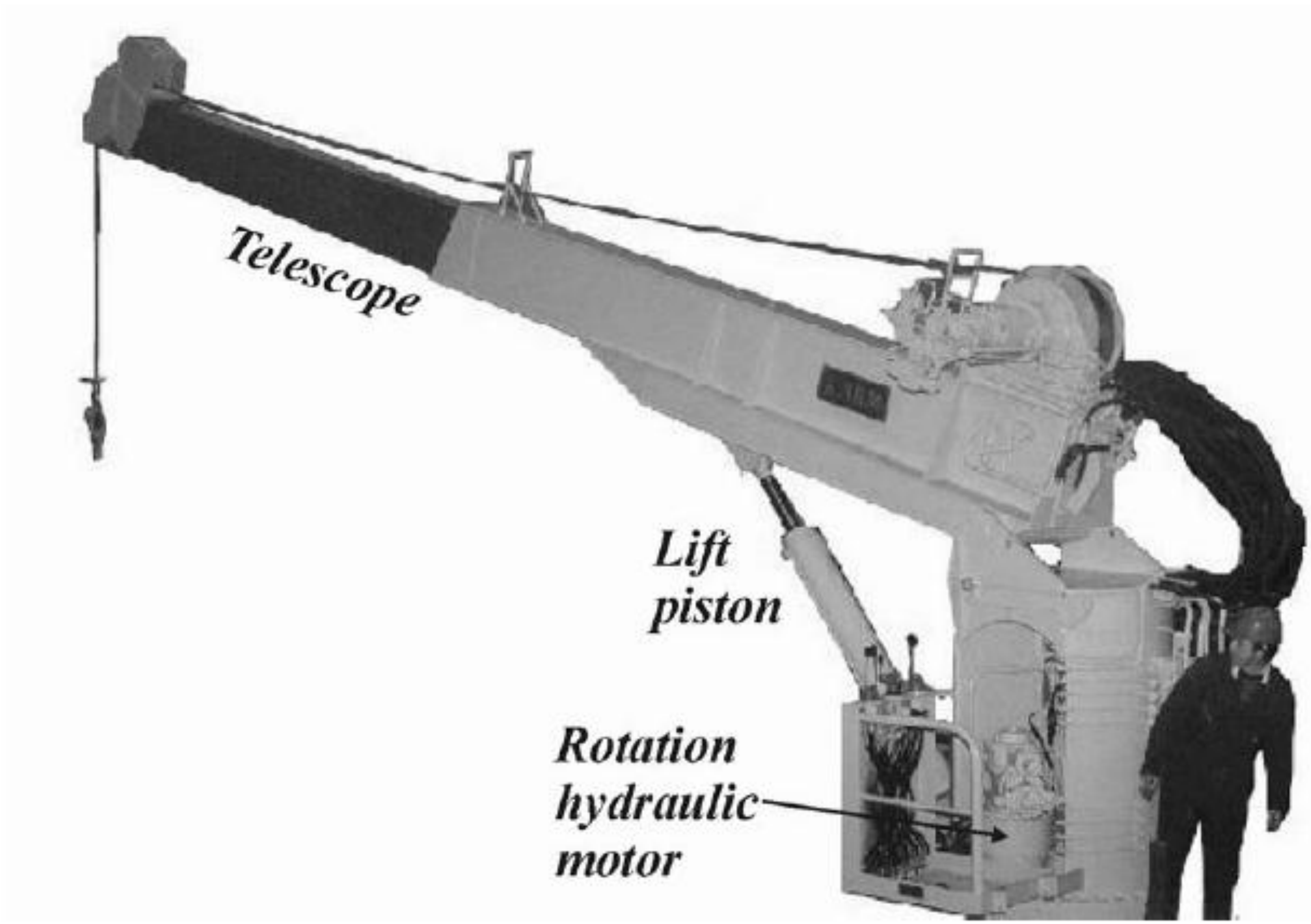
Pumping system



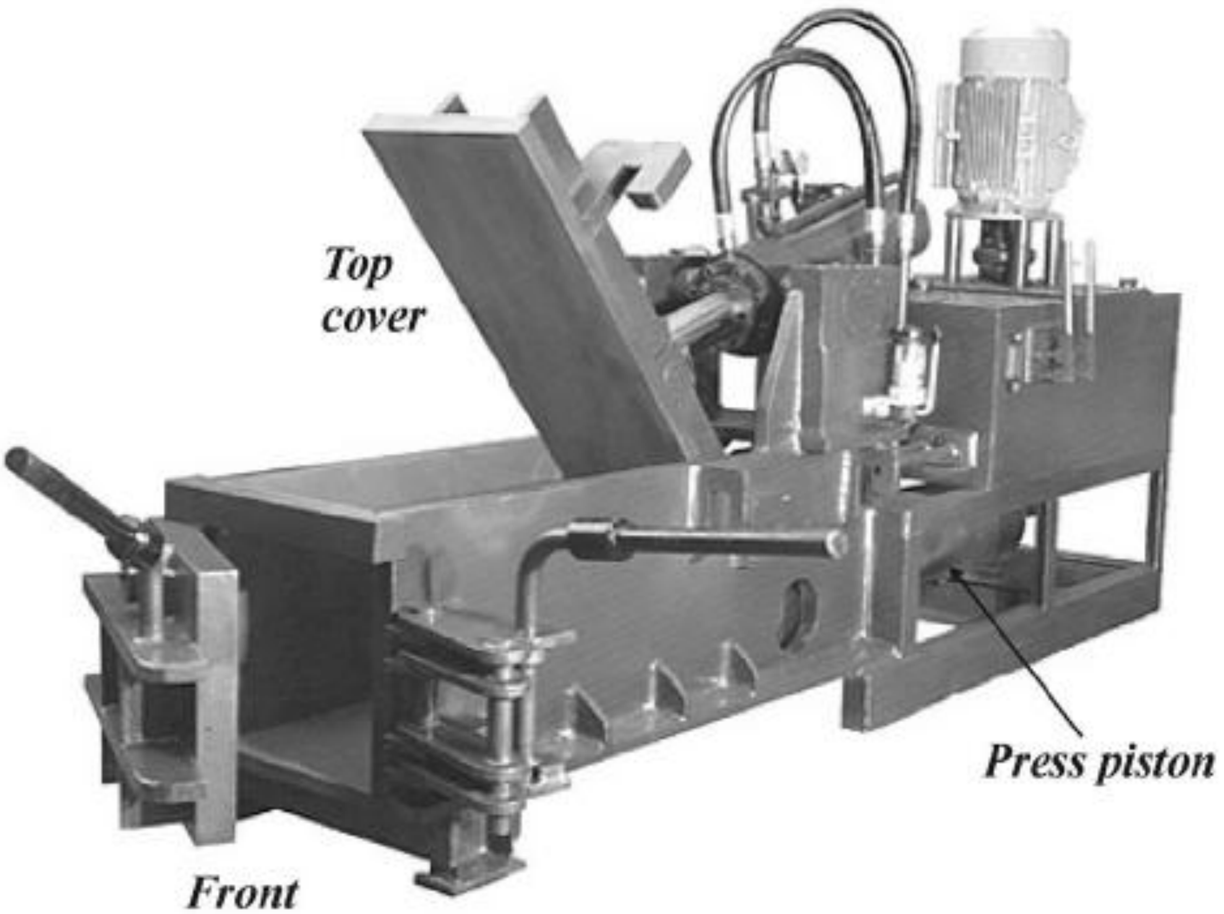


Fork lift

Telescopic crane



Scrap press



Leakage tester

Tyre changing machine

Pneumatic tools

Fundamentals of Fluid Power

Introduction

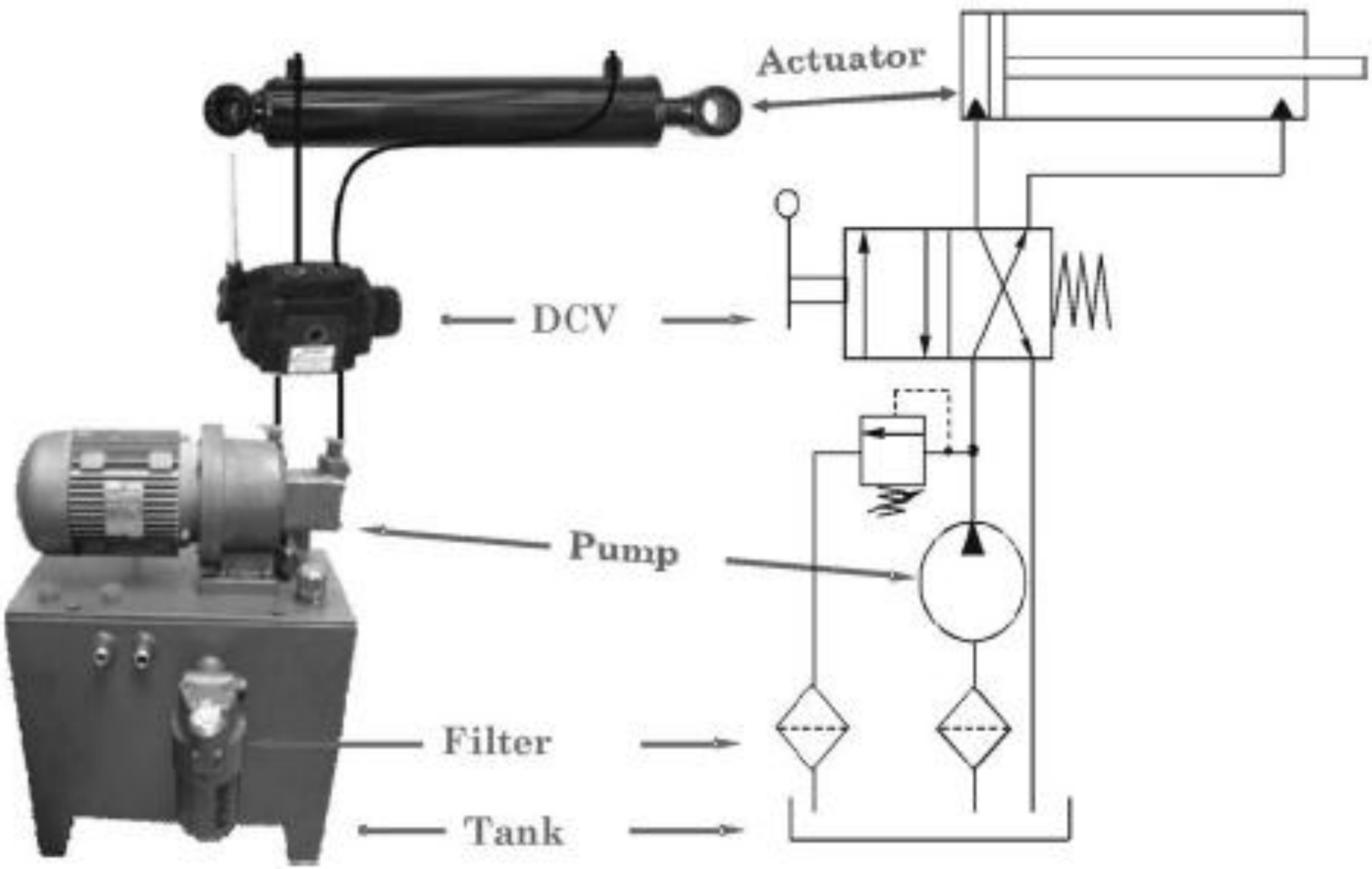
Power transmission

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

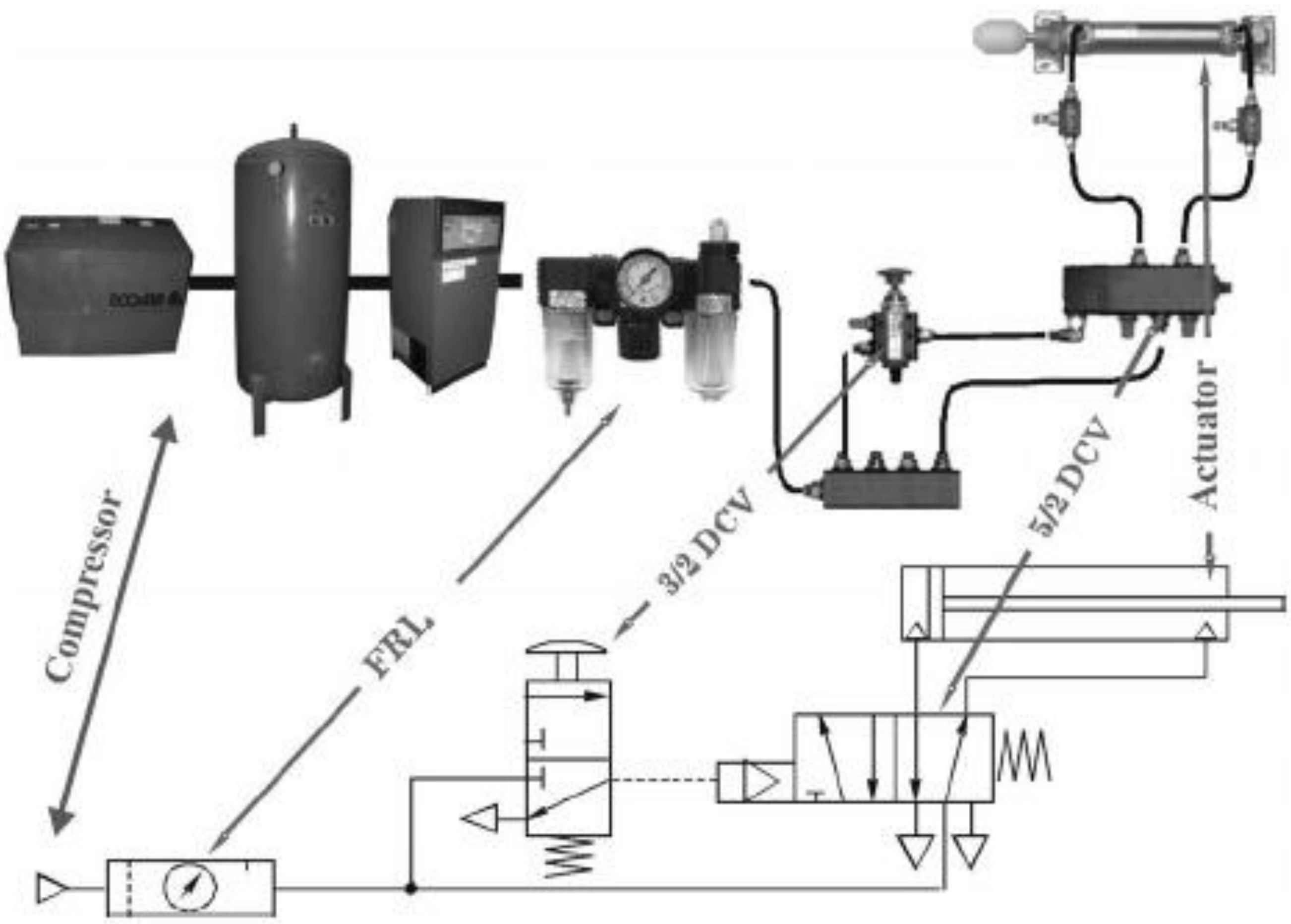
Comparison

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

Hydraulic system



Pneumatic system



Basic theories

- Applied Mechanics
- Vibration Analysis
- Fluid Mechanics
- Thermodynamics

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

Newton's Law

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

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

$$F = \rho Q v$$

Perfect gas law

$$PV = mRT$$

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

$$R_u = 8315 J/kg.K$$

$$\text{air } MW = 28.97$$

$$P_1 V_1 = P_2 V_2$$

Boyl's law

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

Charle's law

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

Gay Laussac's law

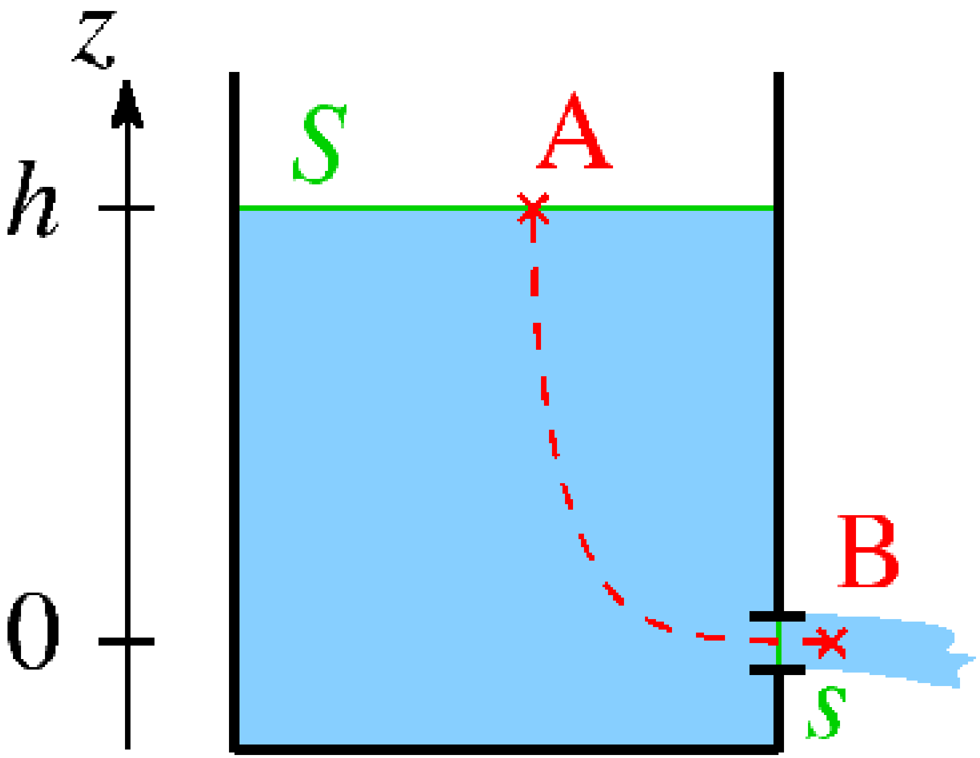
$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} = Constant$$

Combined gas law

Example 1.1

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

Toricelli's theorem

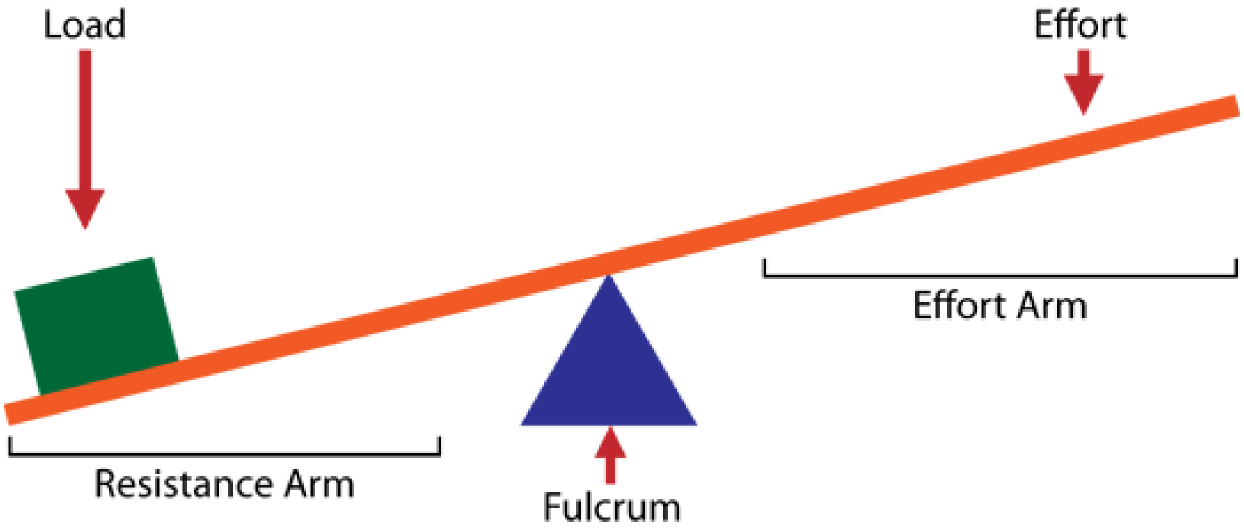


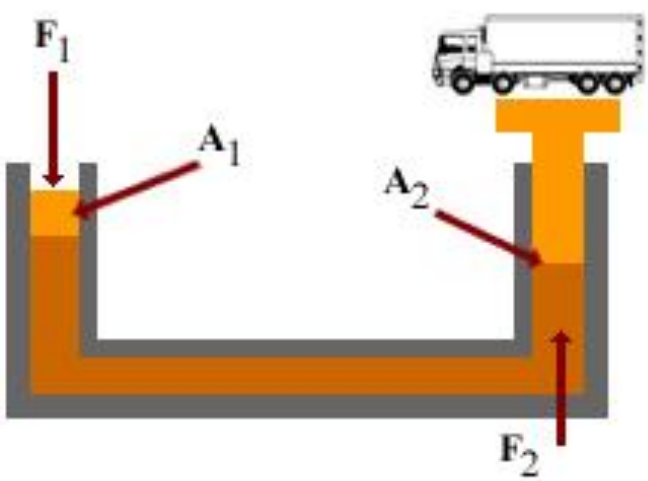
$$\frac{1}{2}mv^2 = mgh$$

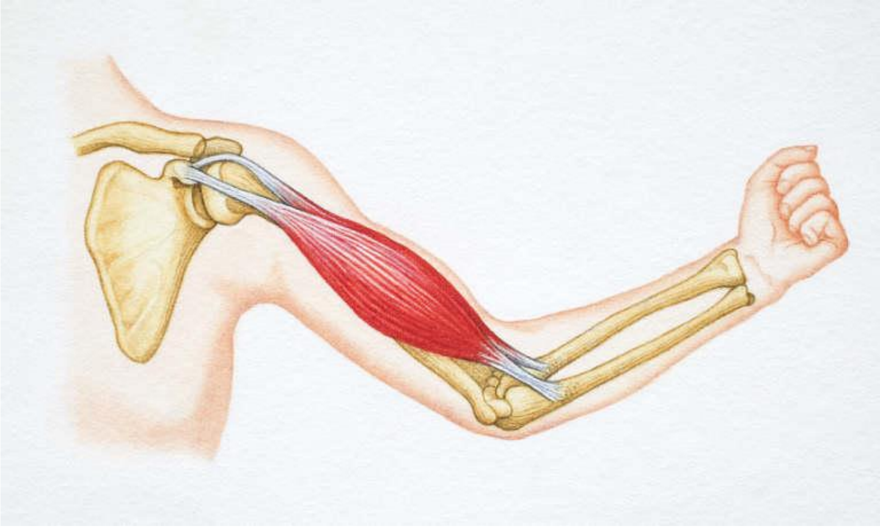
$$h = P/\rho g$$

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

Pascal's theorem



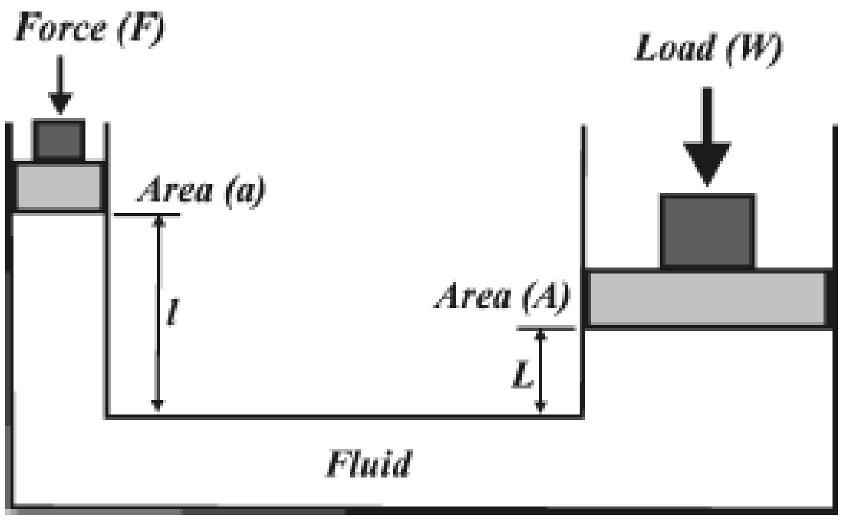




Pascal's theorem

Pascal's theorem states that:

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



Pascal's theorem

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

$$V = AL \bar{=} al$$

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

$$Work = PV = PAL$$

$$W = PA$$

The SI units for the work are

$$Work = P(N/m^2).V(m^3) = N.m$$

$$F = PA$$

$$Power = PQ$$

$$Q = V/time.$$

$$\check{P}ower = \dot{P}(N/m^2).Q(m^3/s) = N.m/s = Watt$$

Pascal's theorem

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

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

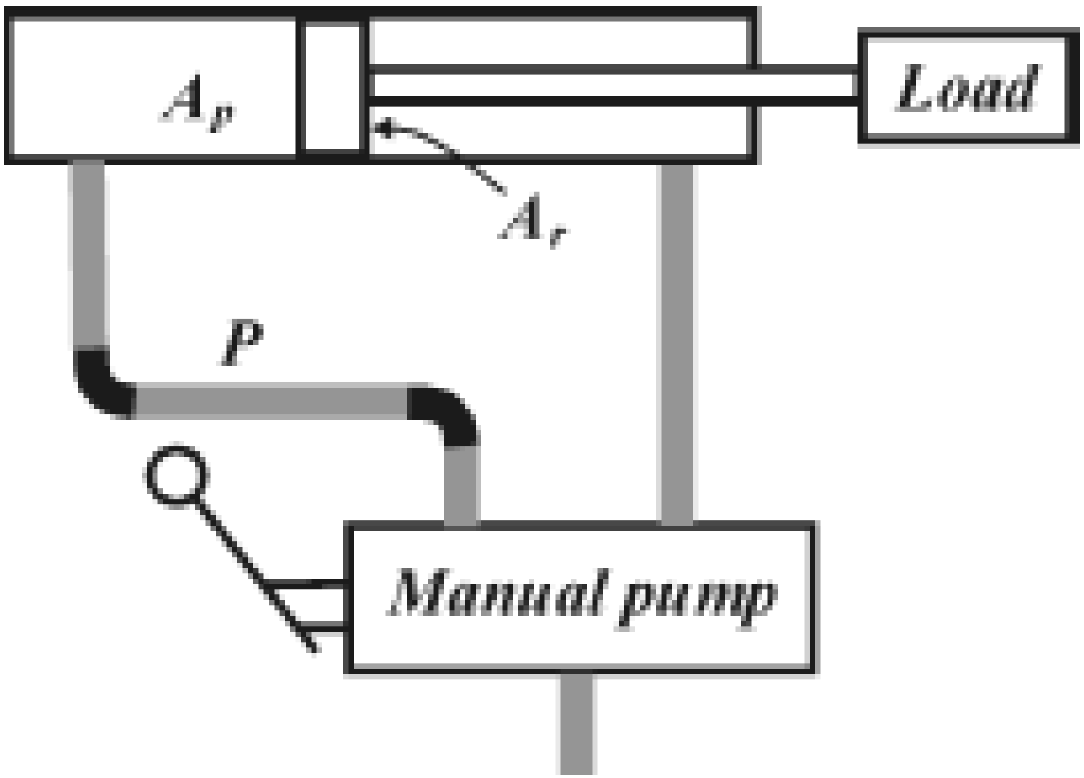
The hydraulic power

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

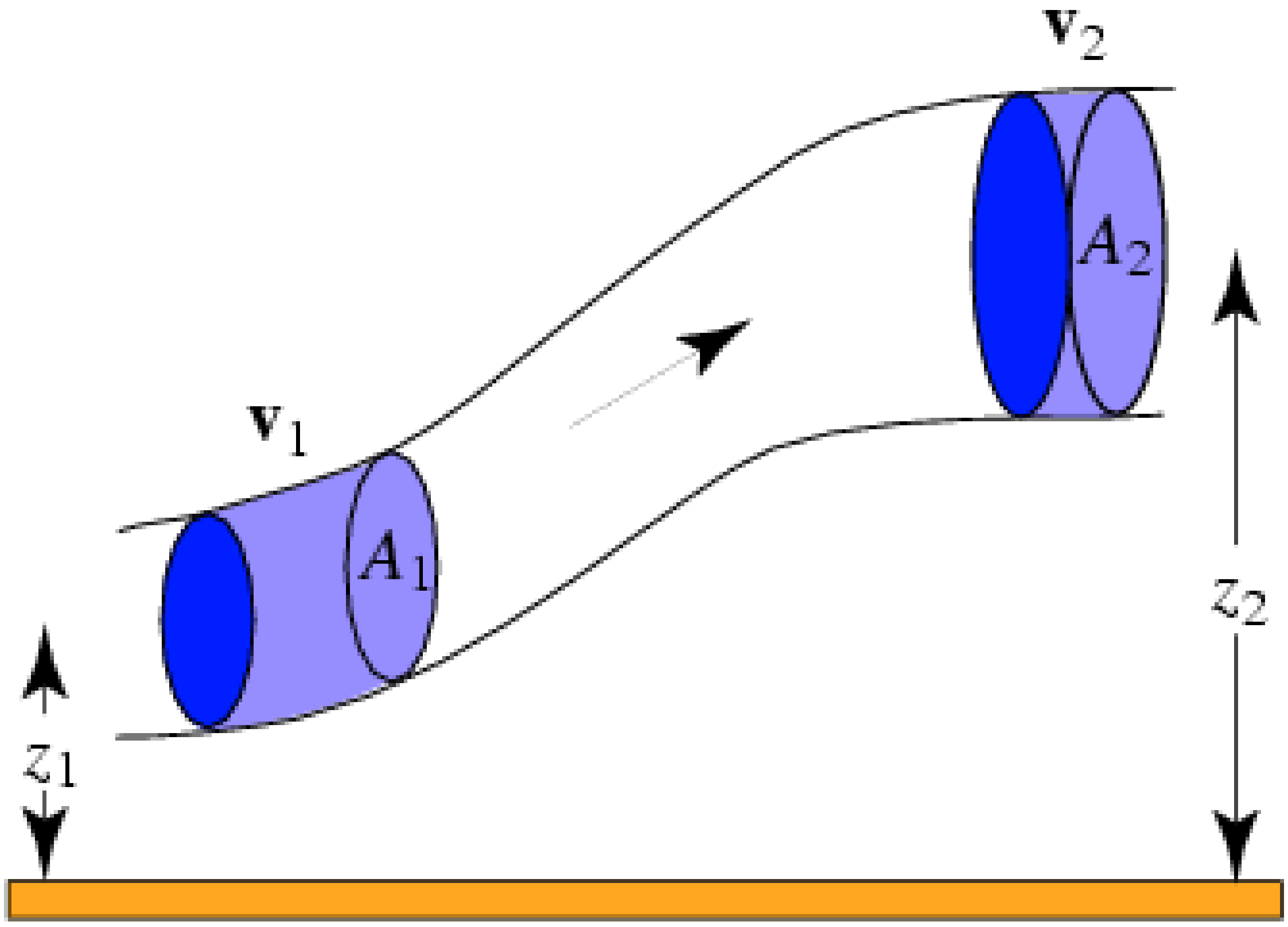
$$= \frac{QP}{600} 10^3 (N.m/s) = \frac{QP}{600} 10^3 (Watts)$$

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

Example 1.2



Bernoulli's Equation



Bernoulli's Equation

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

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

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

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

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

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

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

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

Bernoulli's Equation

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

$$\Delta KE = \frac{1}{2}\Delta m_2 v_2^2 - \frac{1}{2}\Delta m_1 v_1^2 \tag{1.19}$$

where the mass at the inlet is

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

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

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

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

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

$$\Delta PE = \Delta m_2 g z_2 - \Delta m_1 g z_1 = \Delta m g(z_2 - z_1) \tag{1.20}$$

Bernoulli's Equation

Applying the rule of conservation of energy

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

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

$$P_1 \Delta V = \frac{1}{2} \Delta m (v_2^2 - v_1^2) + \Delta m g (z_2 - z_1) + P_2 \Delta V \qquad (1.21)$$

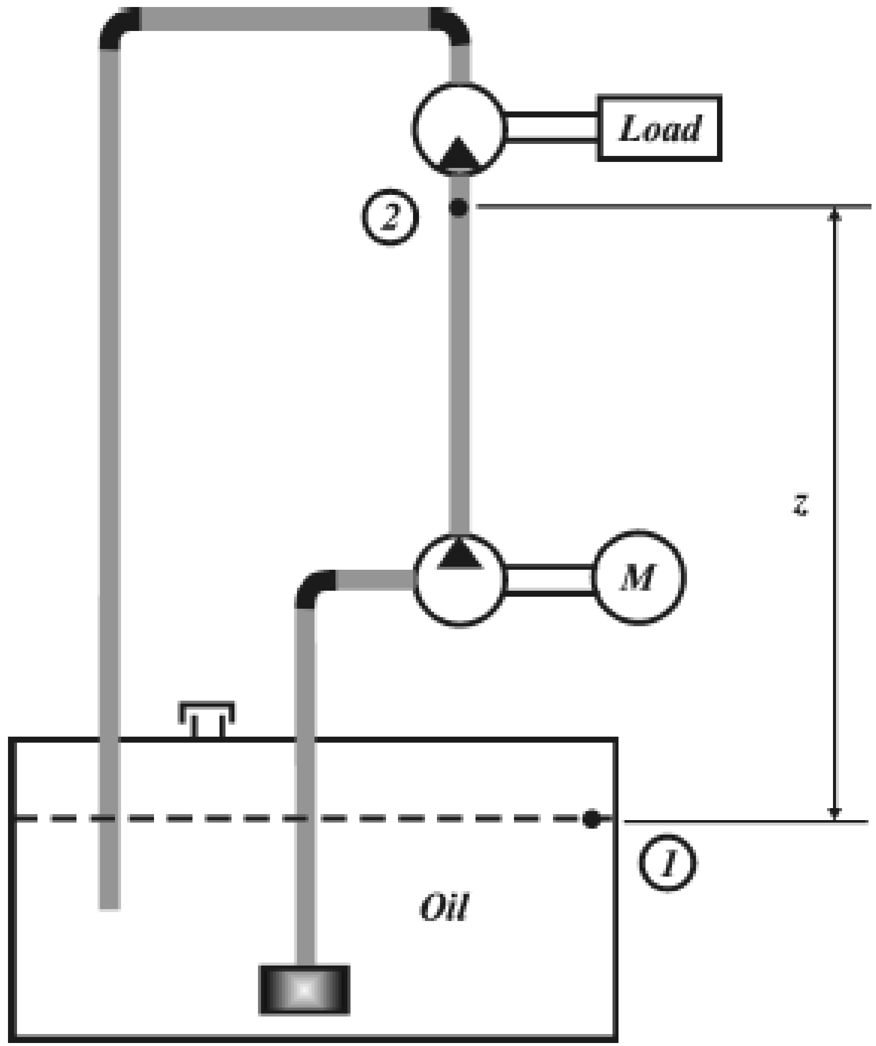
But it is known that $\Delta m / \Delta V = \rho$, thus, dividing equation (1.21) by ΔV results in

$$P_1 = P_2 + \frac{1}{2} \rho v_2^2 + \rho g (z_2 - \frac{1}{2} v_1^2) - \rho g z_1 \qquad (1.22)$$

$$P_1 Q = \frac{1}{2} \rho Q (v_2^2 - v_1^2) + \rho g Q (z_2 - z_1) + P_2 Q + \textit{friction losses} \qquad (1.23)$$

$$z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g} \qquad (1.24)$$

Example 1.3

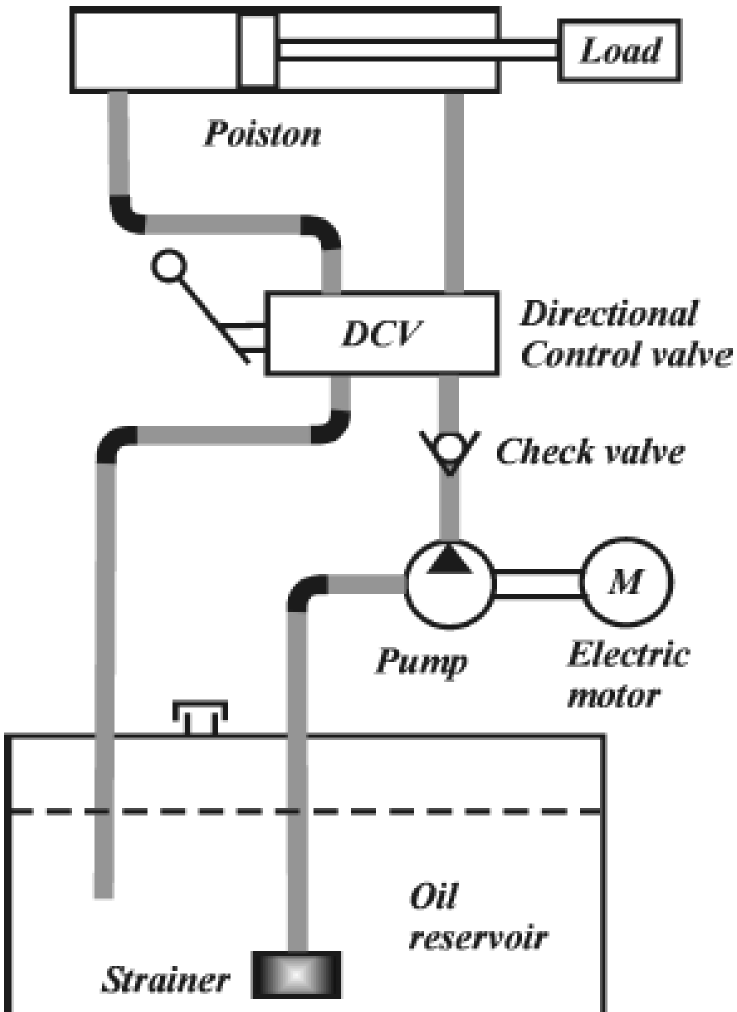


Friction losses

Dissipated heat depends on:

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

Friction losses



Reynold's Number

$$2000 \leq Re \leq 4000 \; for\ smooth\ surface\ (new\ pipe)$$

$$1200 \leq Re \leq 2500 \; for\ corrugated\ surface\ (old\ pipe)$$

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

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

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

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

Example 1.4

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

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

$$Perimeter = 3y = 0.045m$$

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

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

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

Darcy's Equation

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

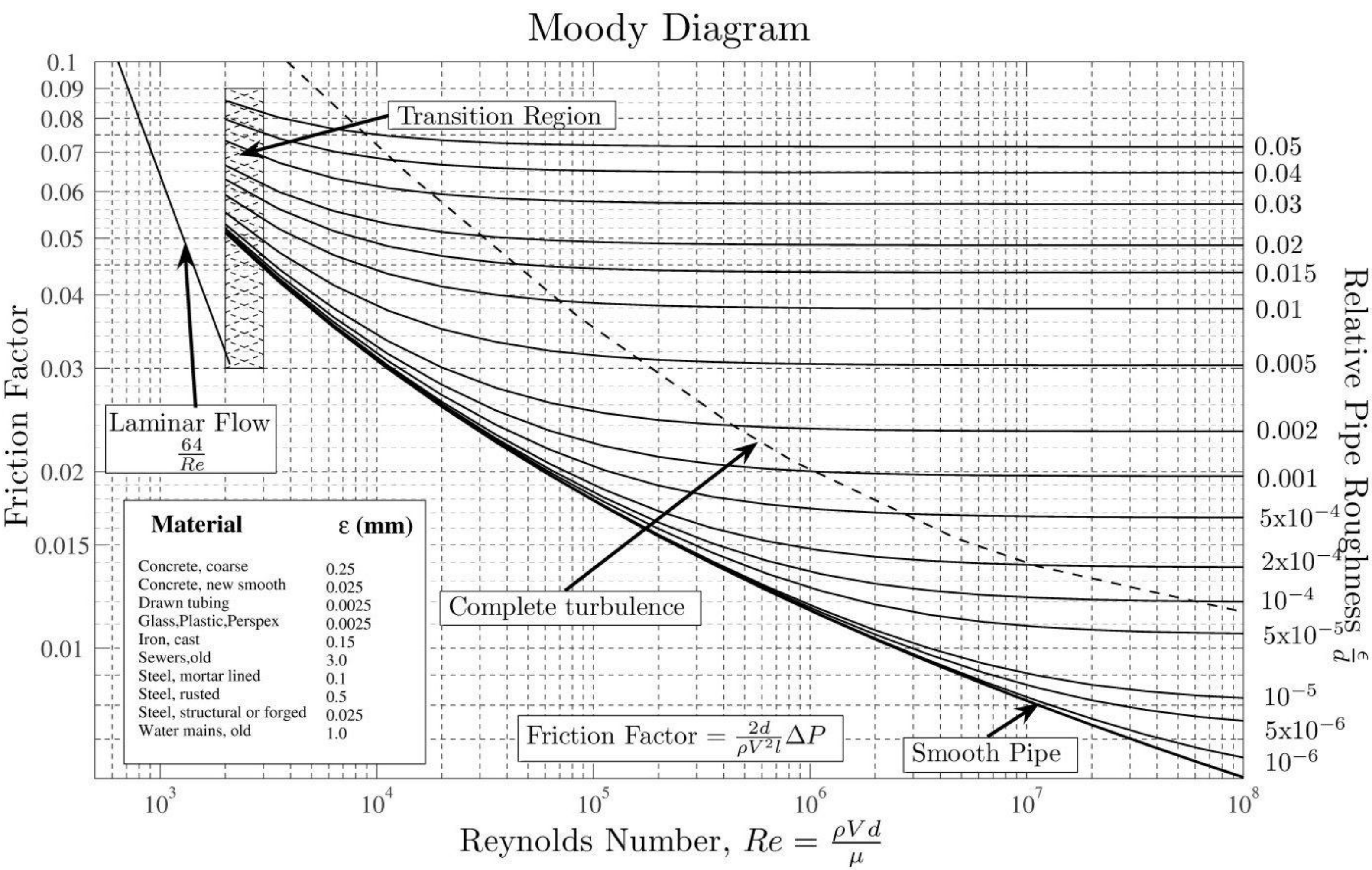
where
f = friction factor (dimensionless)
L = length of pipe (m, ft)
D = inside diameter of pipe (m, ft)
v = fluid velocity (m/s, ft/s)
g = gravitational acceleration $(m/s^2, ft/s^2)$

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

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

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

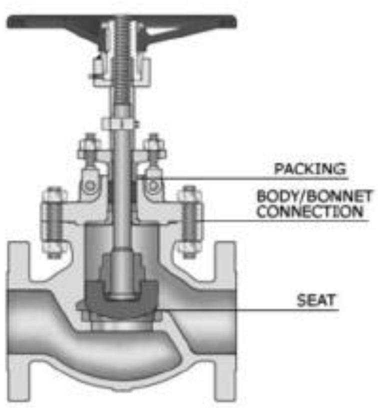
Moody diagram



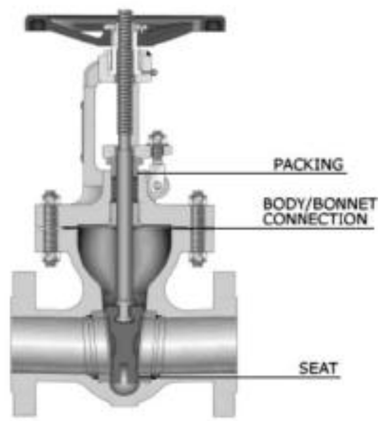
Moody diagram

- Moody diagram is plotted on a log-log scale because of high differences in values.
- The transient flow where Reynold's number is between 2000 and 4000 has no clear values because flow is not possibly predicted.
- At low Reynold's number values (less than 2000), flow is laminar, thus, friction coefficient is a constant value $f = 64/Re$.
- At high Reynold's number values (more than 4000), flow is turbulent, thus, friction coefficient is picked out of the curve where the values of Re and ϵ/D intersect. Interpolation is needed for further accuracy.

Valves and fittings



Globe valve



Gate valve

90° elbow

45° elbow

Return bend

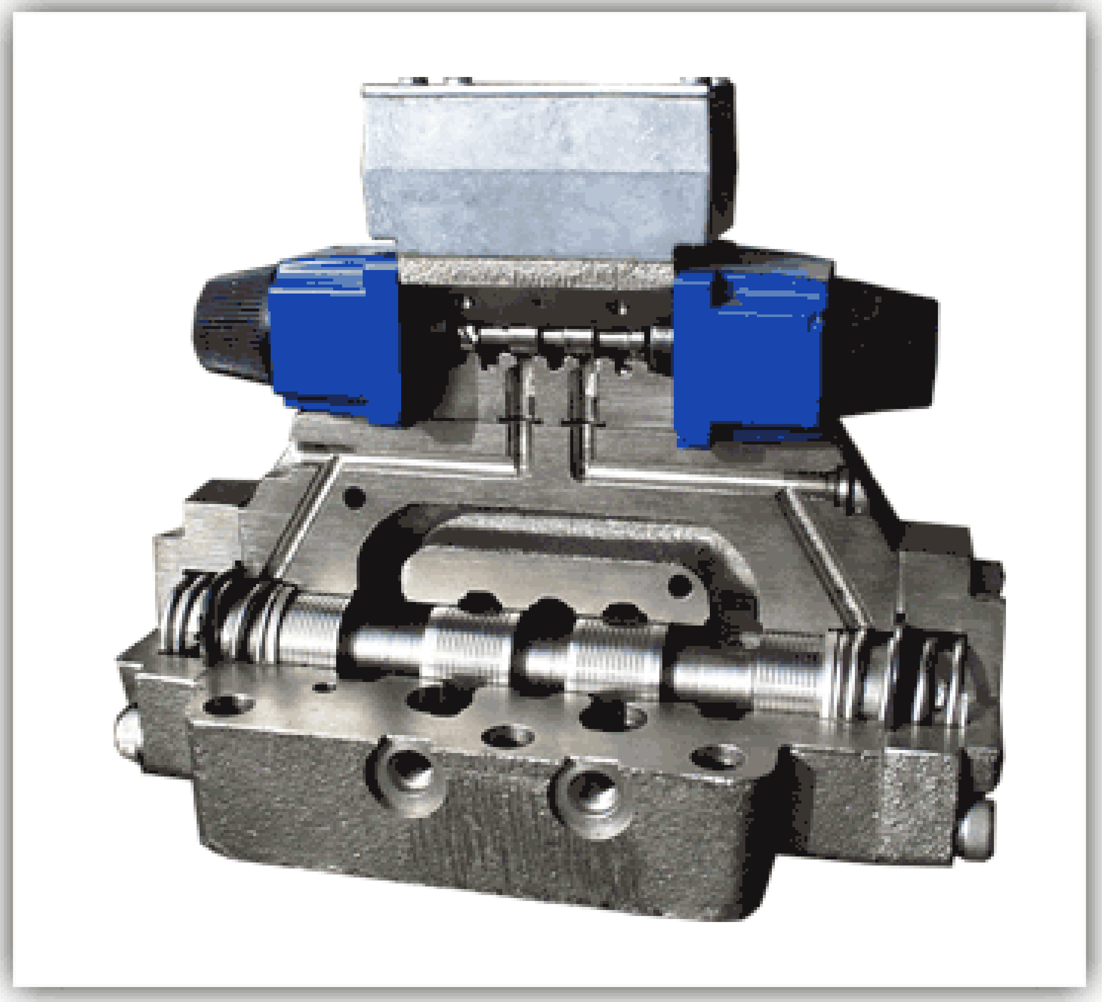
Standard tee

Standard elbow

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

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

Valves



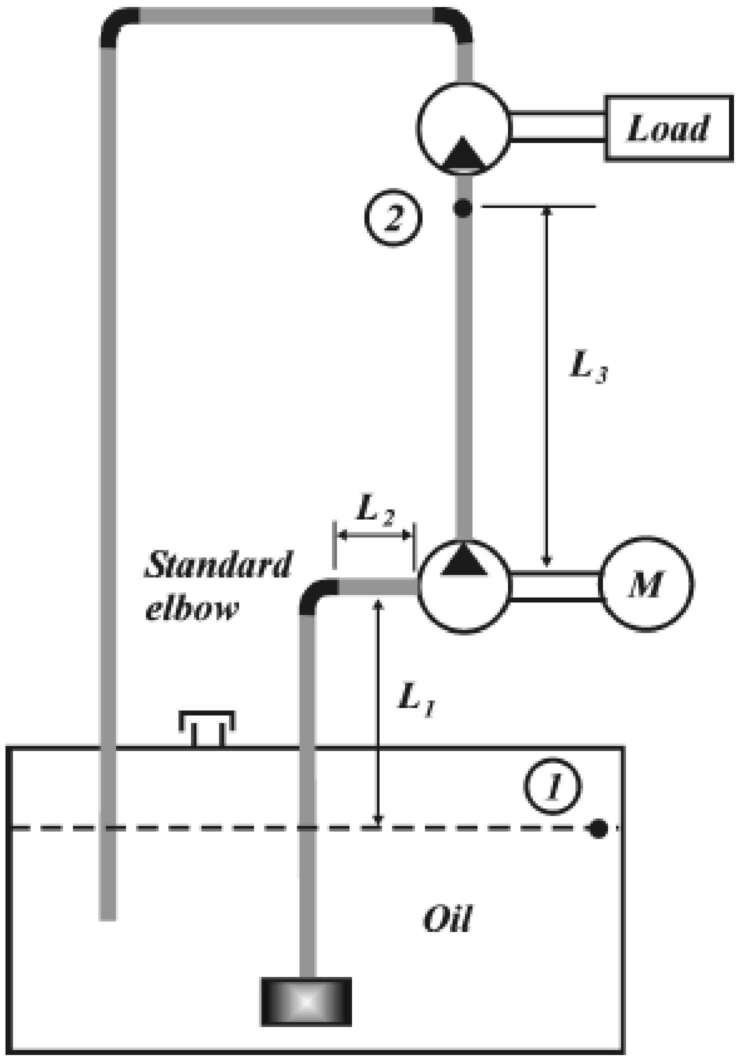
Equivalent length

$$H_{L(fitting)} = H_{L(pipe)}$$

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

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

Example 1.5



Pneumatic friction

$$P_f = \frac{CLQ^2}{3600(CR)D^5}$$

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

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

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

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

$$P_f = \frac{(0.1025)LQ^2}{3600(CR)D^{5.31}}$$

Harris formula

Fluid Power Control

Ahmed Abu Hanieh

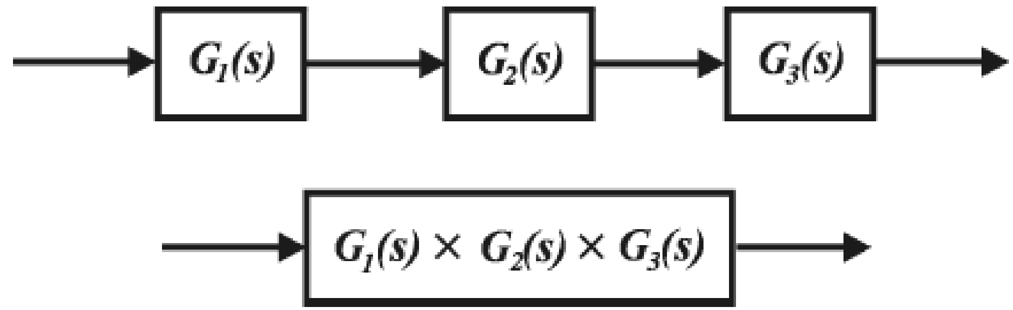
Modelling

Modelling techniques

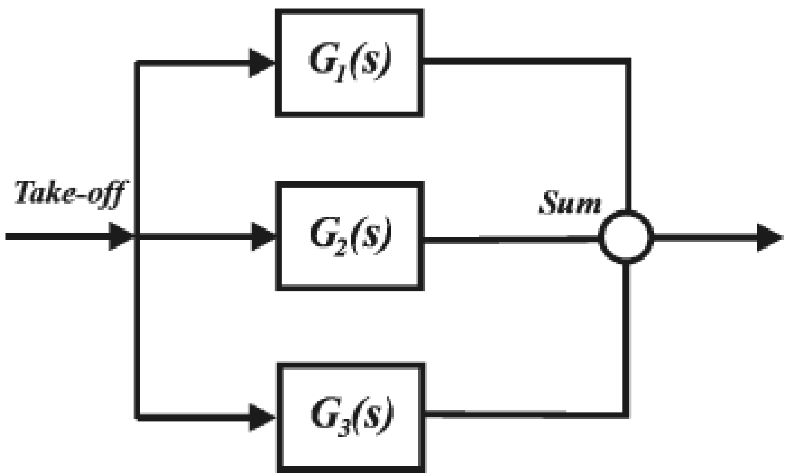
A second order differential equation

- Newton laws
- Conservation of Energy method
- Finite element method
- Modal analysis

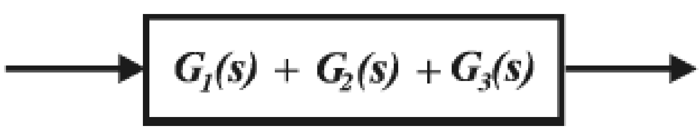
Block diagrams



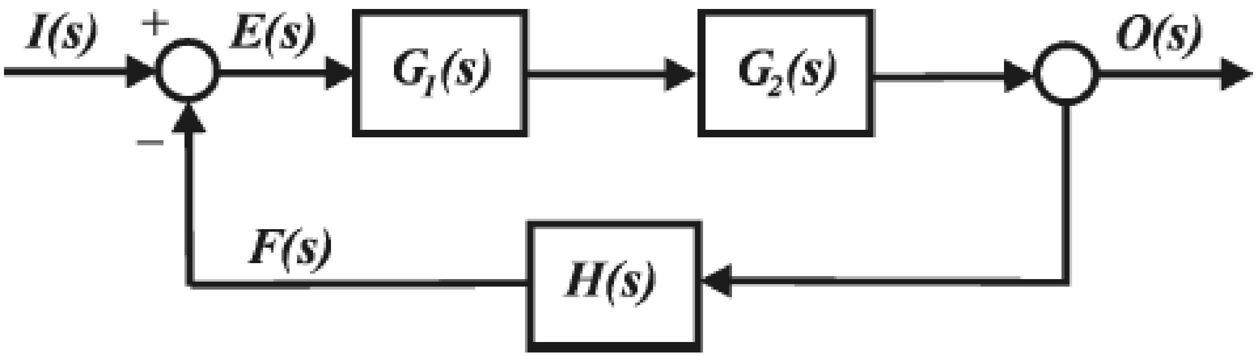
Series



Parallel



Feedback Control loop



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

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

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

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

Feedback Control loop

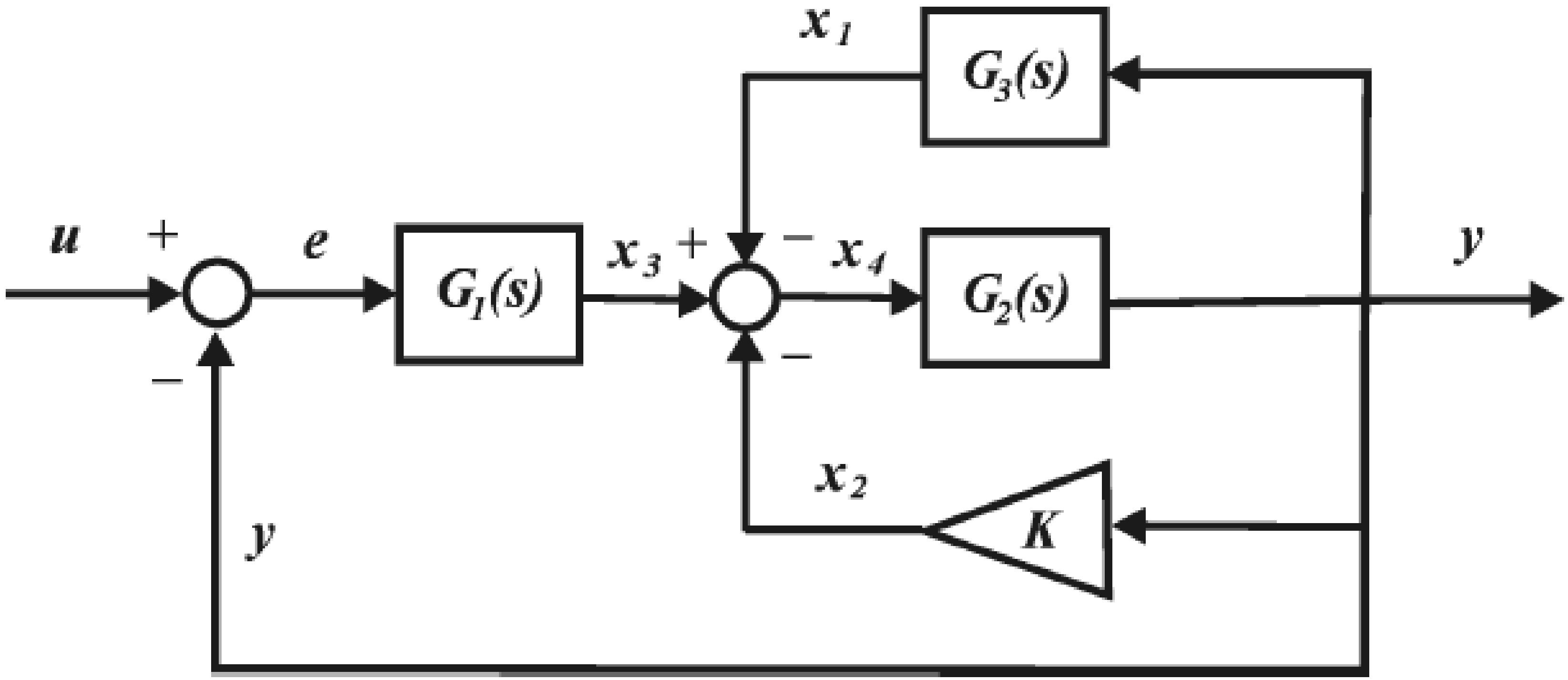
$$O(s) = G_1(s)G_2(s)[I(s) - H(s)O(s)]$$

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

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

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

Example 2.1



Conceptual modelling

$$\left.\begin{array}{c} F = m\dfrac{d^2x}{dt^2} = m\dfrac{dv}{dt} \\ \\ v = \dfrac{1}{m}\displaystyle\int F dt \\ \\ T = I\dfrac{d^2\theta}{dt^2} = I\dfrac{d\omega}{dt} \end{array}\right\} \text{Mass}$$

$$\left.\begin{array}{c} F = k(x_1 - x_2) \\ \\ F = k\displaystyle\int (v_1 - v_2) dt \\ \\ T = k(\theta_1 - \theta_2) \end{array}\right\} \text{Spring}$$

$$\left.\begin{array}{c} F = C(v_1 - v_2) \\ \\ T = C(\omega_1 - \omega_2) \end{array}\right\} \text{Damper}$$

Effort and Flow

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

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

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

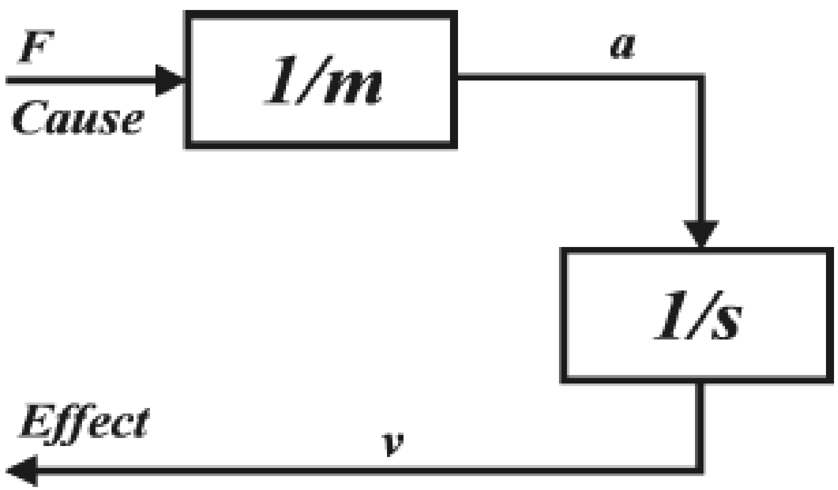
$$Output\ Impedance = \frac{e_2}{f_2}$$

$$f_1 = f_2 = f$$

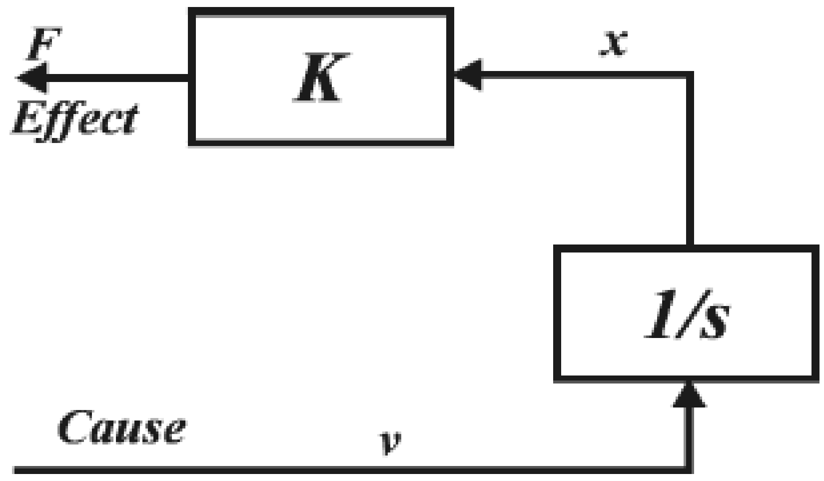
$$Transfer\ Function\ \frac{e_2}{e_1} = \frac{e_2}{f} \times \frac{f}{e_1}$$

Effort and Flow

Effort causes flow



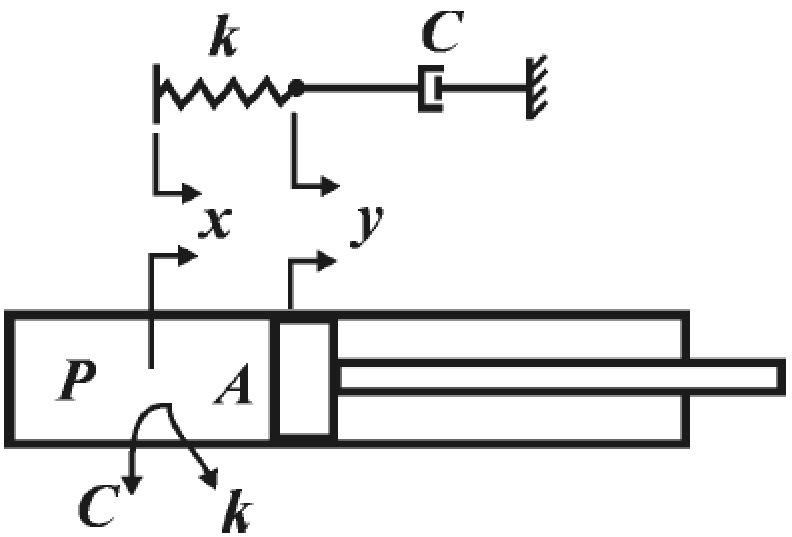
Flow causes effort

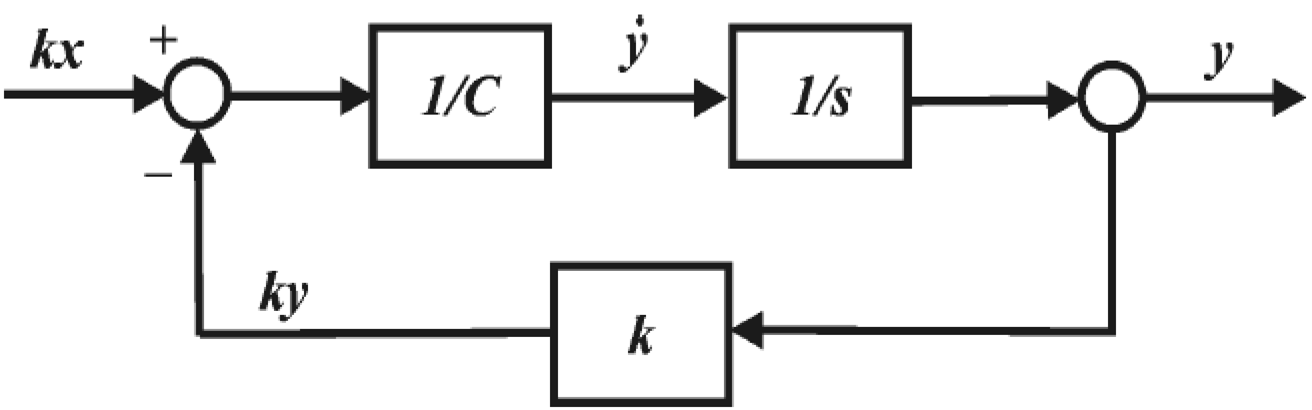


Comparison

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

First order modelling of a piston





First order

$$kx - ky - C\dot{y} = 0$$

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

g a unit input $x = 1$, then

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

first order differential equation

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

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

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

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

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

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

$\tau = C/k$ is the time constant

input is a sinusoidal harmonic motion at a frequency ω

output will be a harmonic motion at a new amplitude Y and a phase shift ϕ.

Input $x(t) = X sin(\omega t)$
Output $y(t) = Y sin(\omega t + \phi)$

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

$$s = j\omega$$

$$j = \sqrt{-1}.$$

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

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

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

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

$$C/k = \tau \text{ and } j\omega = s.$$

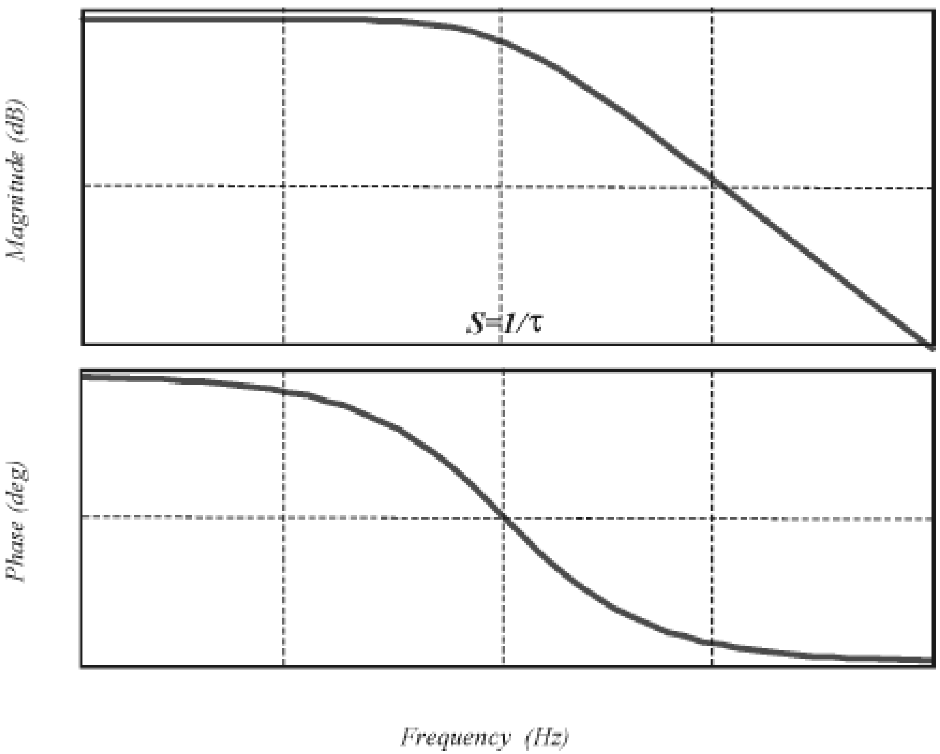
$$\frac{y(s)}{x(s)} = \frac{1}{1 + \tau s}$$

$$s = -1/\tau$$

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

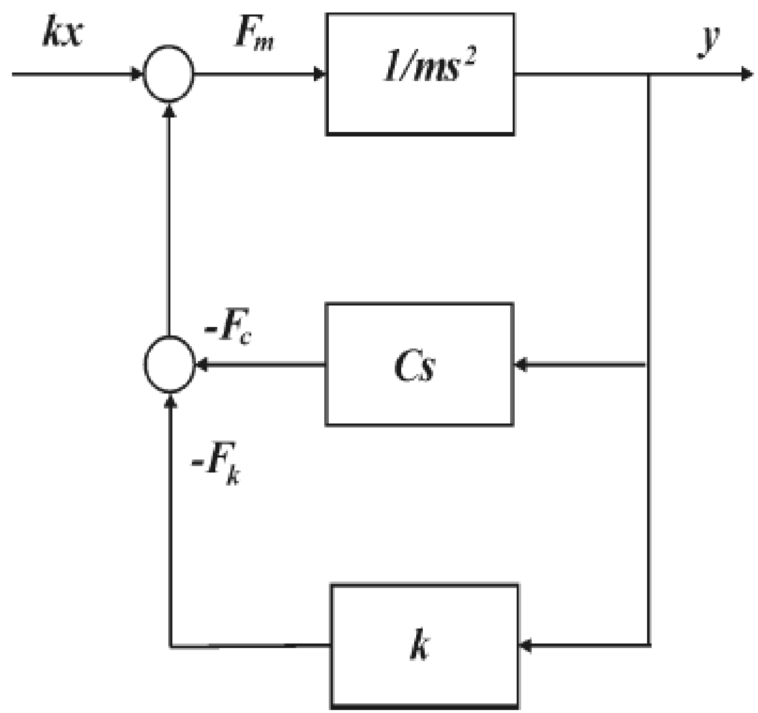
$$\phi = tan^{-1}(-\tau\omega)$$

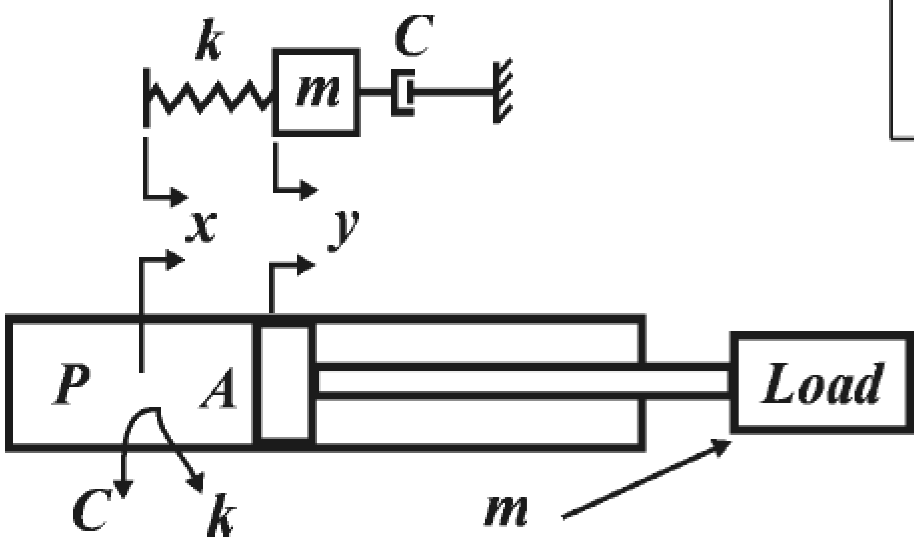
$$-20dB/decade$$



Roll-off

Second order TF





Second order

Inertia force

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

Damping force

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

Spring force

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

$$kx - F_C - F_k = F_m$$

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

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

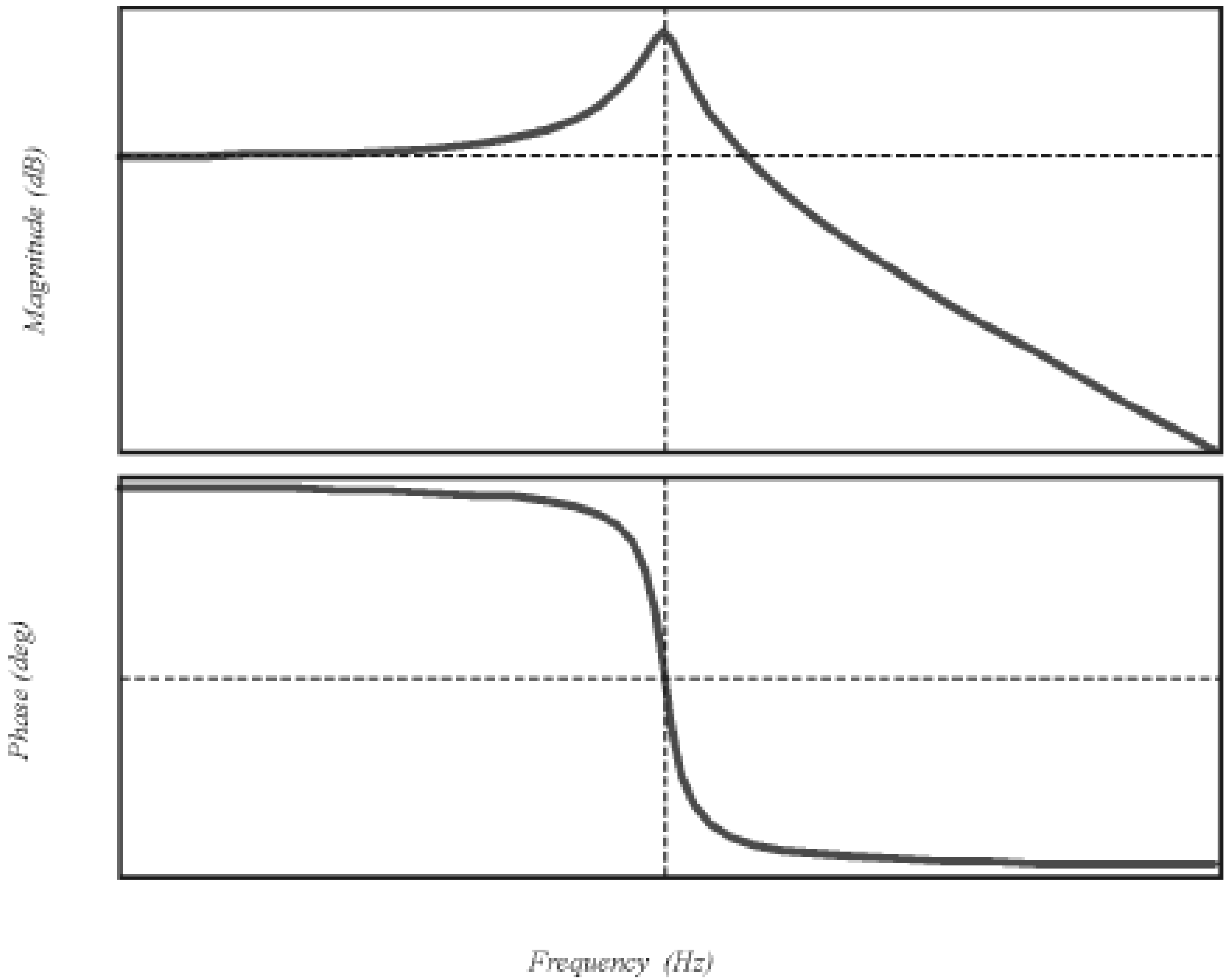
$$\frac{y}{x} = \frac{k}{ms^2 + Cs + k}$$

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

$$\omega_n = \sqrt{k/m}$$

$$\xi = C/2m\omega_n$$

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

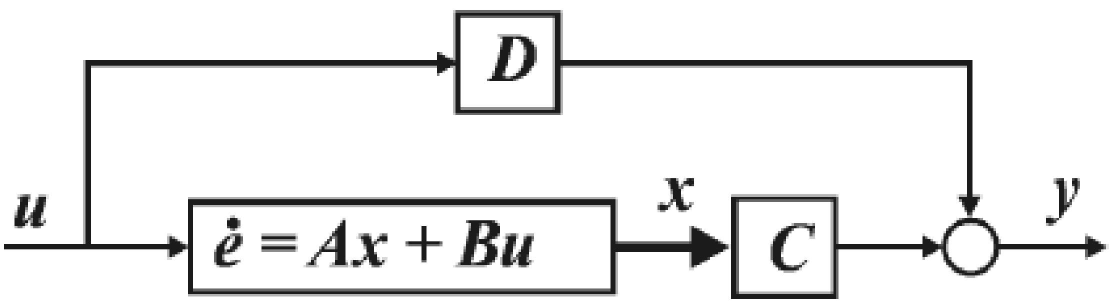

Magnitude (dB)
Phase (deg)
Frequency (Hz)

State space approach

$$\dot{e} = Ax + Bu$$

$$y = Cx + Du$$

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



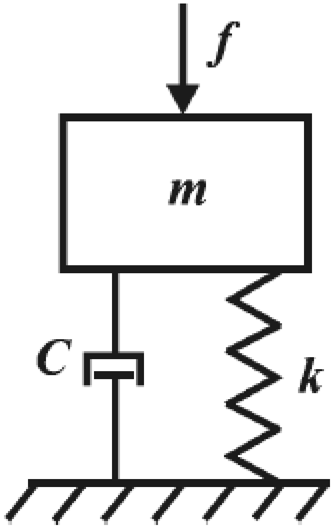
Simple oscillator

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

$$e_1 = x$$

$$e_2 = \dot{x}$$

$$\dot{e}_1 = \dot{x} = e_2$$



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

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

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

output is the displacement x $\quad y = e_1 = x$

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

output is the velocity $\dot{x}$, $\quad y = e_2 = \dot{x}$

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

$$D = (0)$$

output is the acceleration, $y = \dot{e}_2 = \ddot{x}$

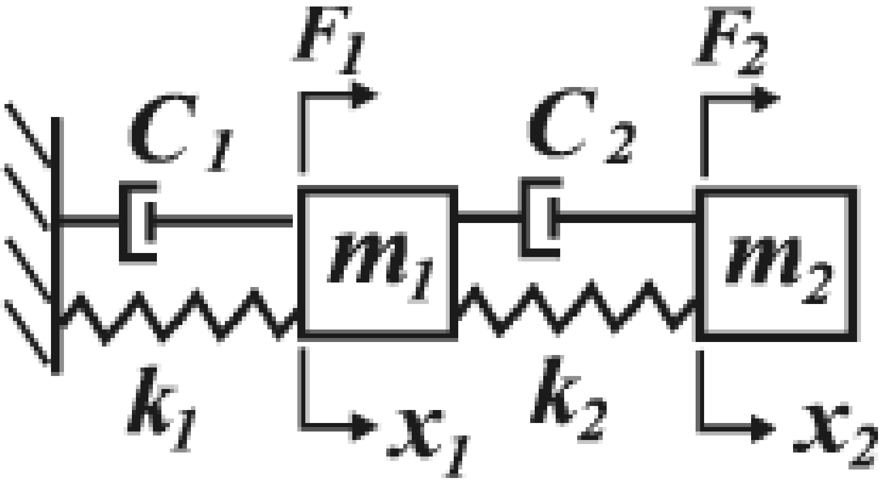
$$D = 1/m$$

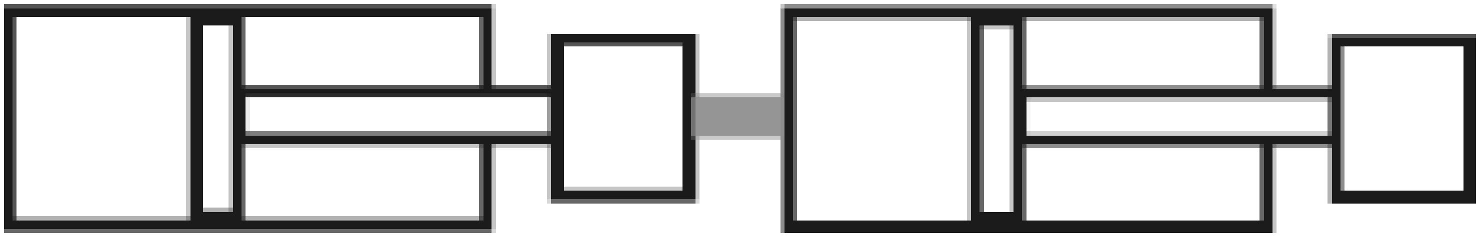
$$y = \underbrace{\begin{pmatrix} -\omega_n^2 & -2\xi\omega_n \end{pmatrix}}_{C} \begin{pmatrix} x \\ \dot{x} \end{pmatrix} + \underbrace{\frac{1}{m}}_{D} f$$

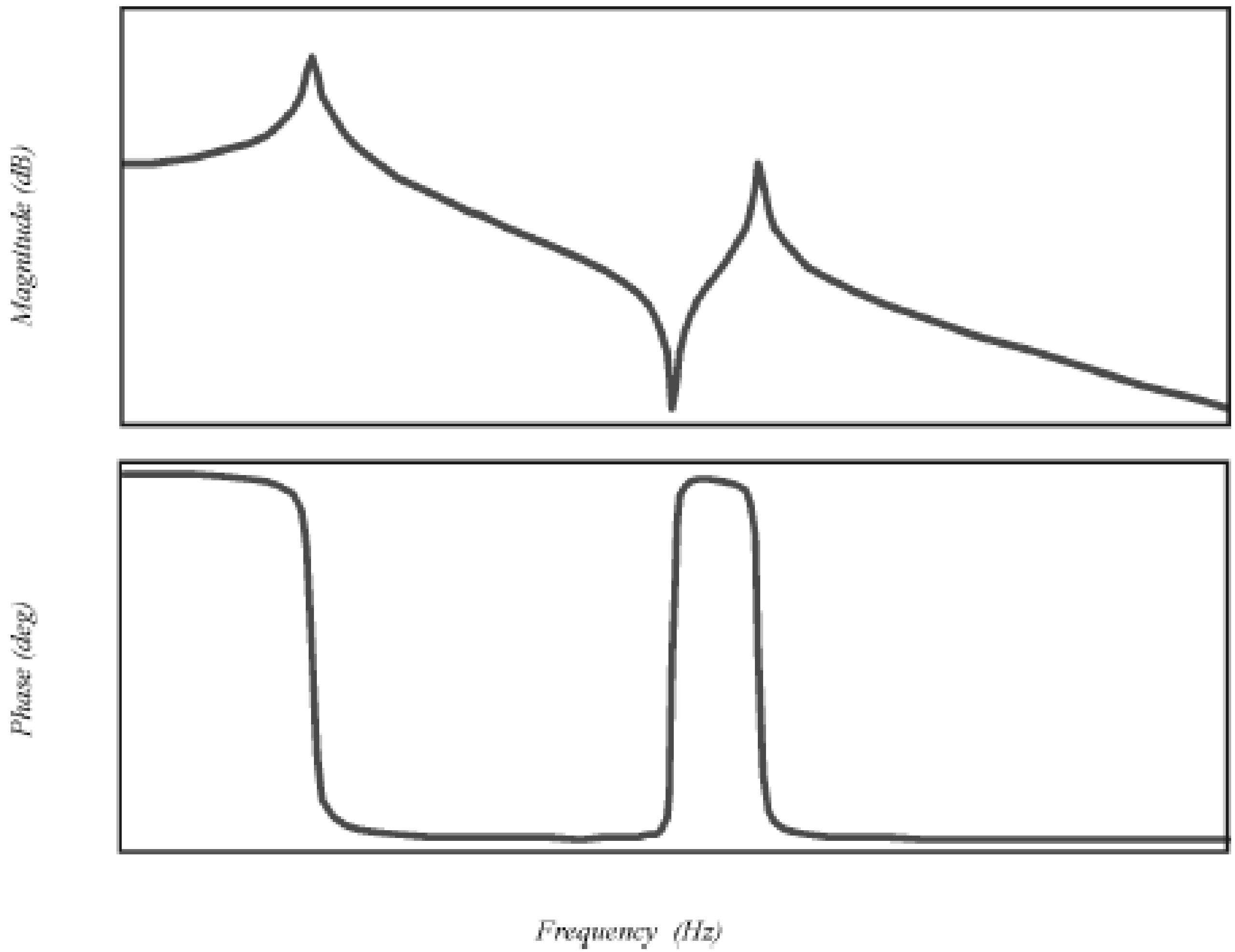
$$C = \begin{pmatrix} -\omega_n^2 & -2\xi\omega_n \end{pmatrix}$$

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

Example 2.2






Magnitude (dB)
Phase (deg)
Frequency (Hz)

Rotating element

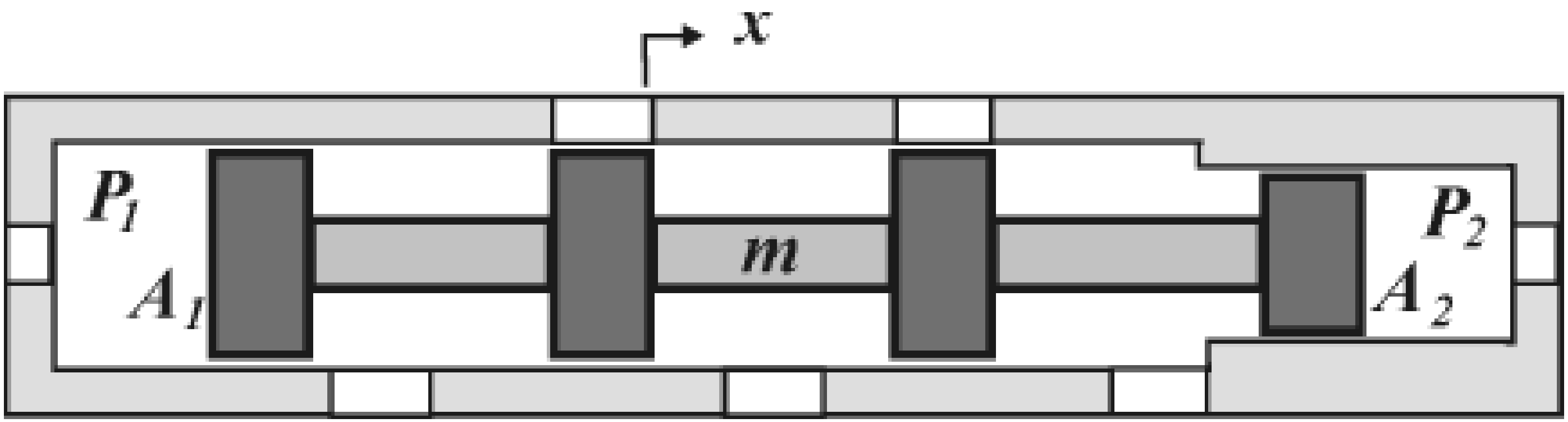
$$I\ddot{\theta} + C_\theta\dot{\theta} + k_\theta\theta = T$$

$$I\alpha + C_\theta\omega + k_\theta\theta = T$$

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

$$T = PAR$$

Control valve



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

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

Control valve

$$P_1 A_1 - P_2 A_2 = m\ddot{x} + C\dot{x} + kx$$

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

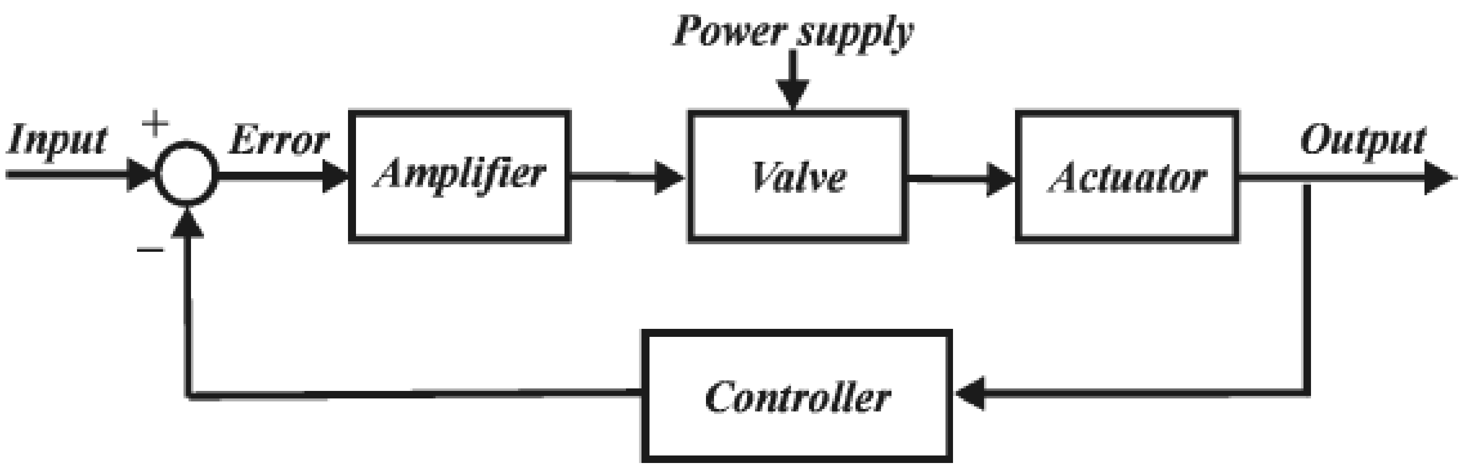
Control systems

Control techniques

Mechanical

Electrical

Servo control



Conditions for servo control system:

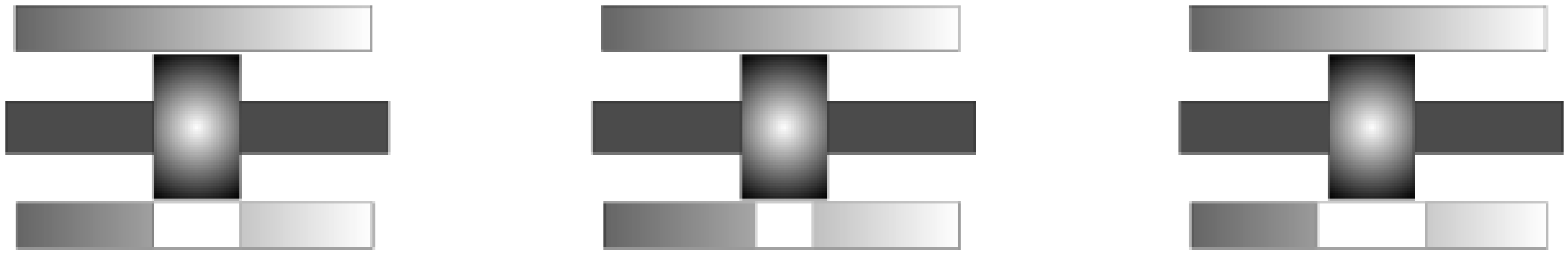
1- Amplification (Conditioning)
2- Feedback

Follow-up control system undergoes:

1- Valve operated servo control system
2- Pump operated servo control system

Valve operated servo control system

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



Zero lap *Overlap* *Underlap*

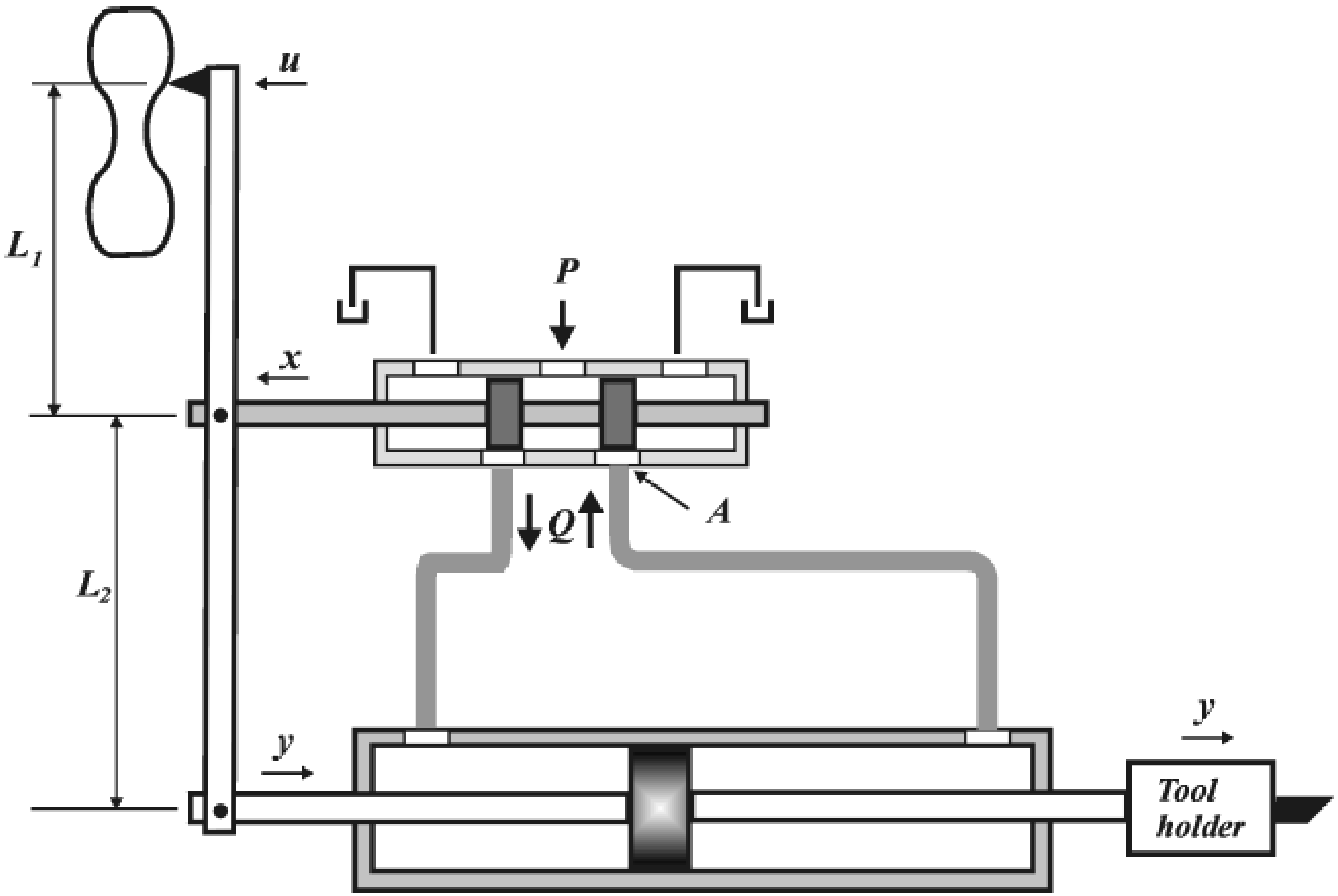
Analysis

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

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

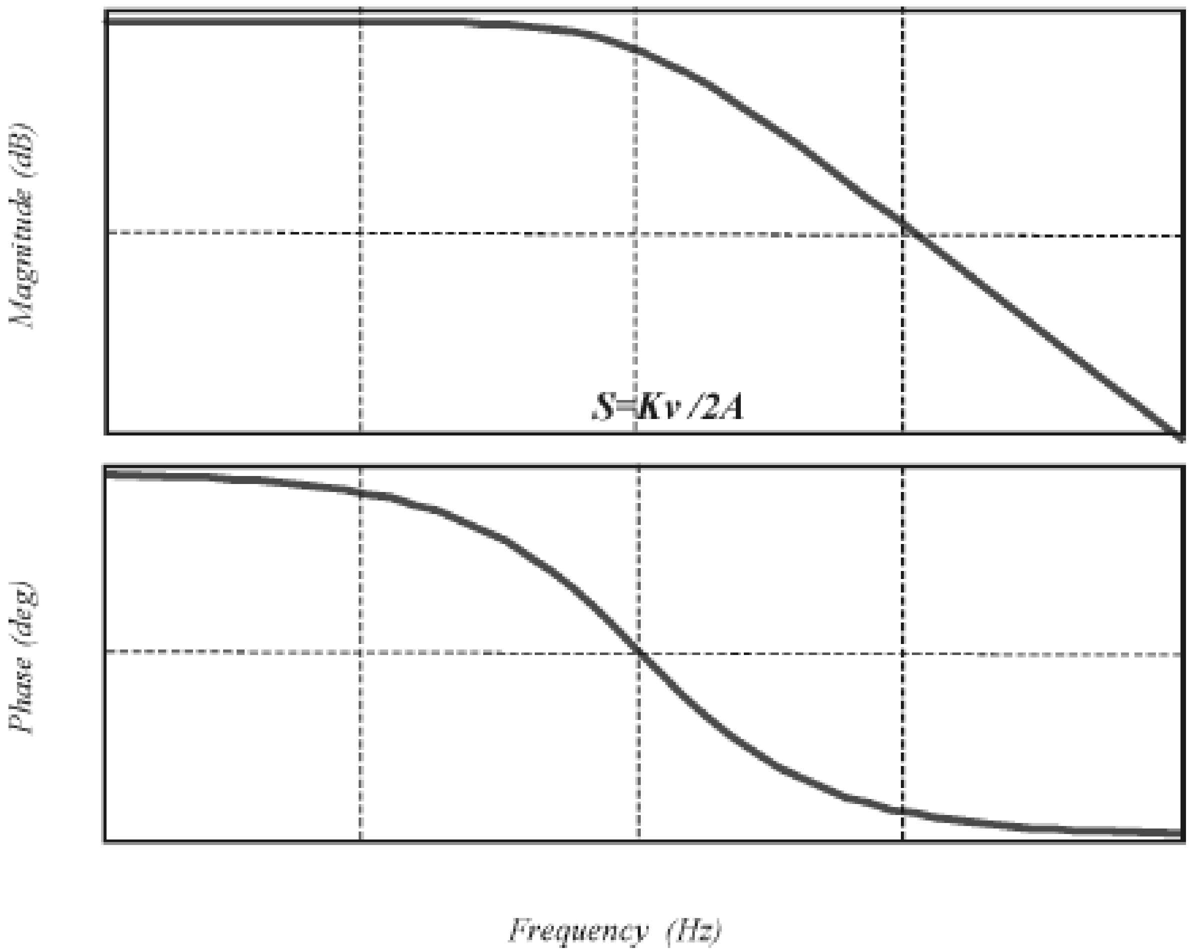
$A =$ Cross sectional area of the port.
$\rho =$ The fluid density.

Example 3.1



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

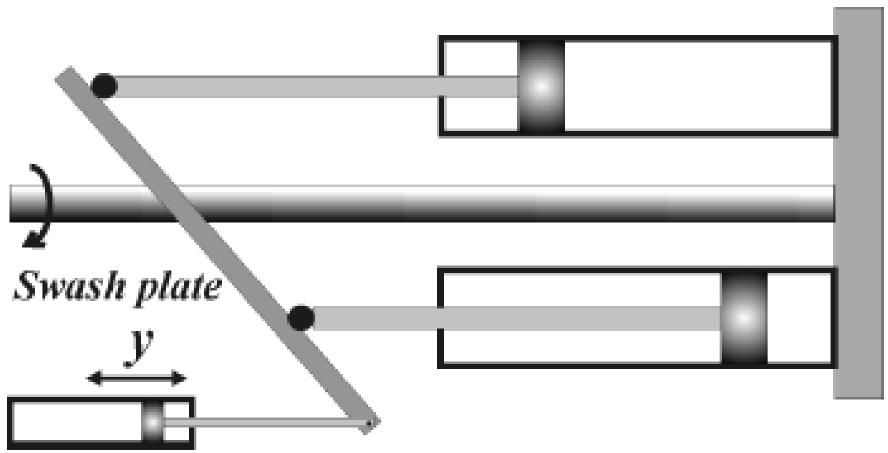
$$\tau = 2A/K_v$$

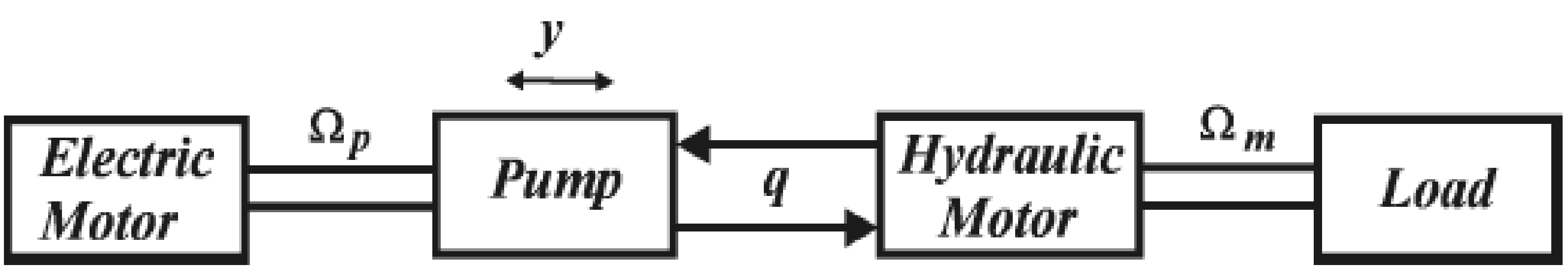


Pump operated servo control systems

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

Hydrostatic transmission system





Analysis

Flow from the pump = Flow to the motor

$$q_p = q_m = q$$

$$\Omega_p d_p = \Omega_m d_m \qquad \Omega_p = constant$$

d_m is the displacement of the motor per radian

$$q_p = K_p y$$

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

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

Leakage effect

Pump leakage $= \lambda_p P_p$.
Actual flow in pump $= K_p y - \lambda_p P_p = q$.

Motor leakage $= \lambda_m P_m$.
Actual flow to motor $= q - \lambda_m P_m = \Omega_m d_m$

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

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

$$P = P_p = P_m$$

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

Leakage effect

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

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

$$T_m = P_m d_m$$

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

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

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

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

multiplying both sides by d_m gives

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

therefore,

$$T_m = P d_m$$

or

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

Using Laplace transform with zero initial conditions

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

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

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

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

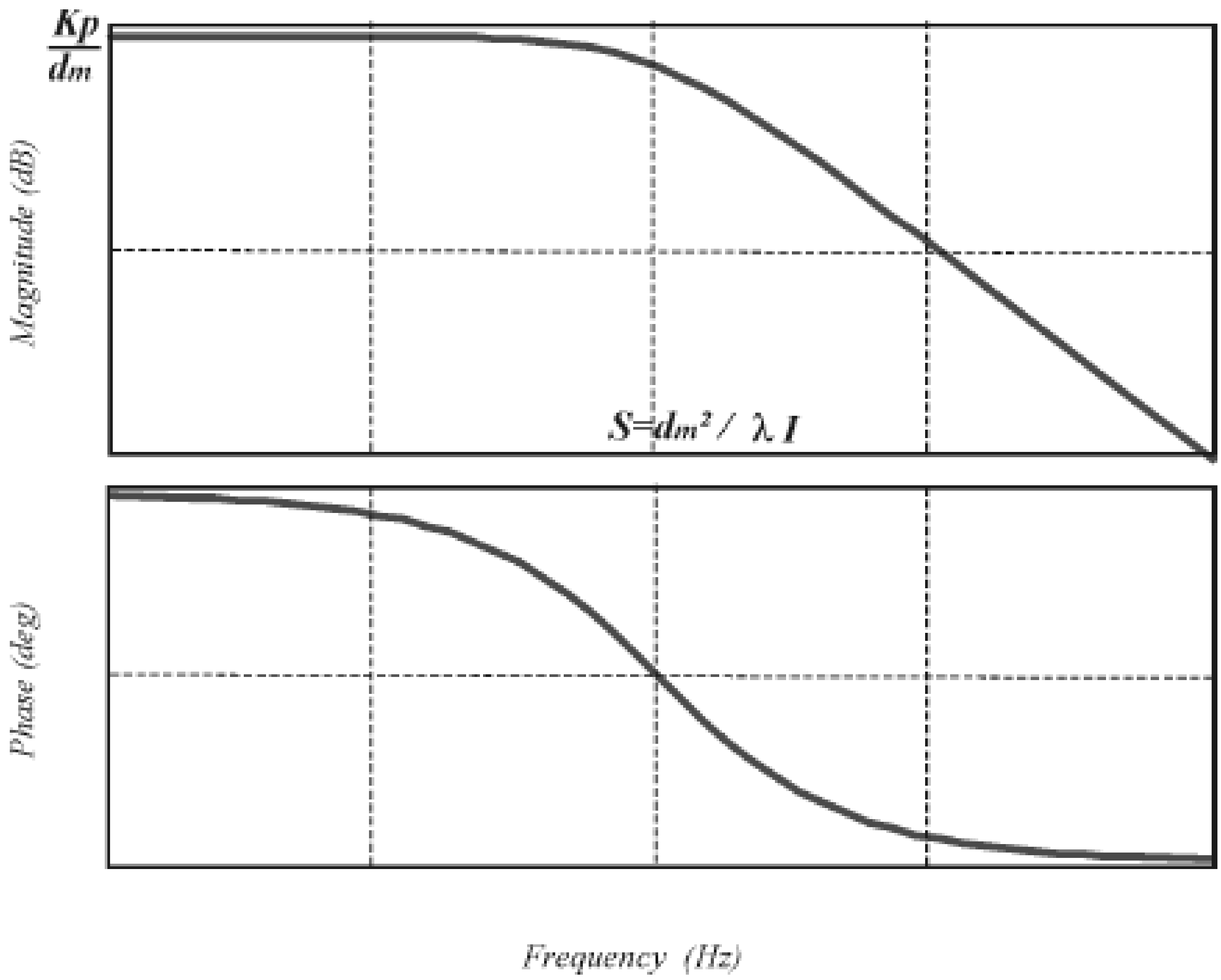
multiplying numerator and denominator by λ/d_m^2 gives

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

This represents a first order low-pass filter

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

time constant $\tau = \lambda I/d_m^2$


$\frac{Kp}{dm}$
Magnitude (dB)
$S = dm^2 / \lambda I$
Phase (deg)
Frequency (Hz)

Compressibility effect

$$B = \frac{Volumetric\ stress}{Volumetric\ strain} \qquad Bulk\ modulus$$

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

$$q_c = \frac{d}{dt}\Delta V = \frac{d}{dt}\left(\frac{VP}{B}\right) = \left(\frac{V}{B}\right)\frac{d}{dt}P \qquad \text{loss in the flow } q_c$$

The actual flow reches to the motor is

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

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

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

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

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

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

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

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

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

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

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

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

Natural frequency

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

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

$$\text{motor displacement } \Delta V = d_m$$

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

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

$$T_m = K_H \times \theta$$

where K_H is the hydraulic stiffness and θ is the angle of rotation

$$\theta = 1 \text{ radian, then } K_H = T_m$$

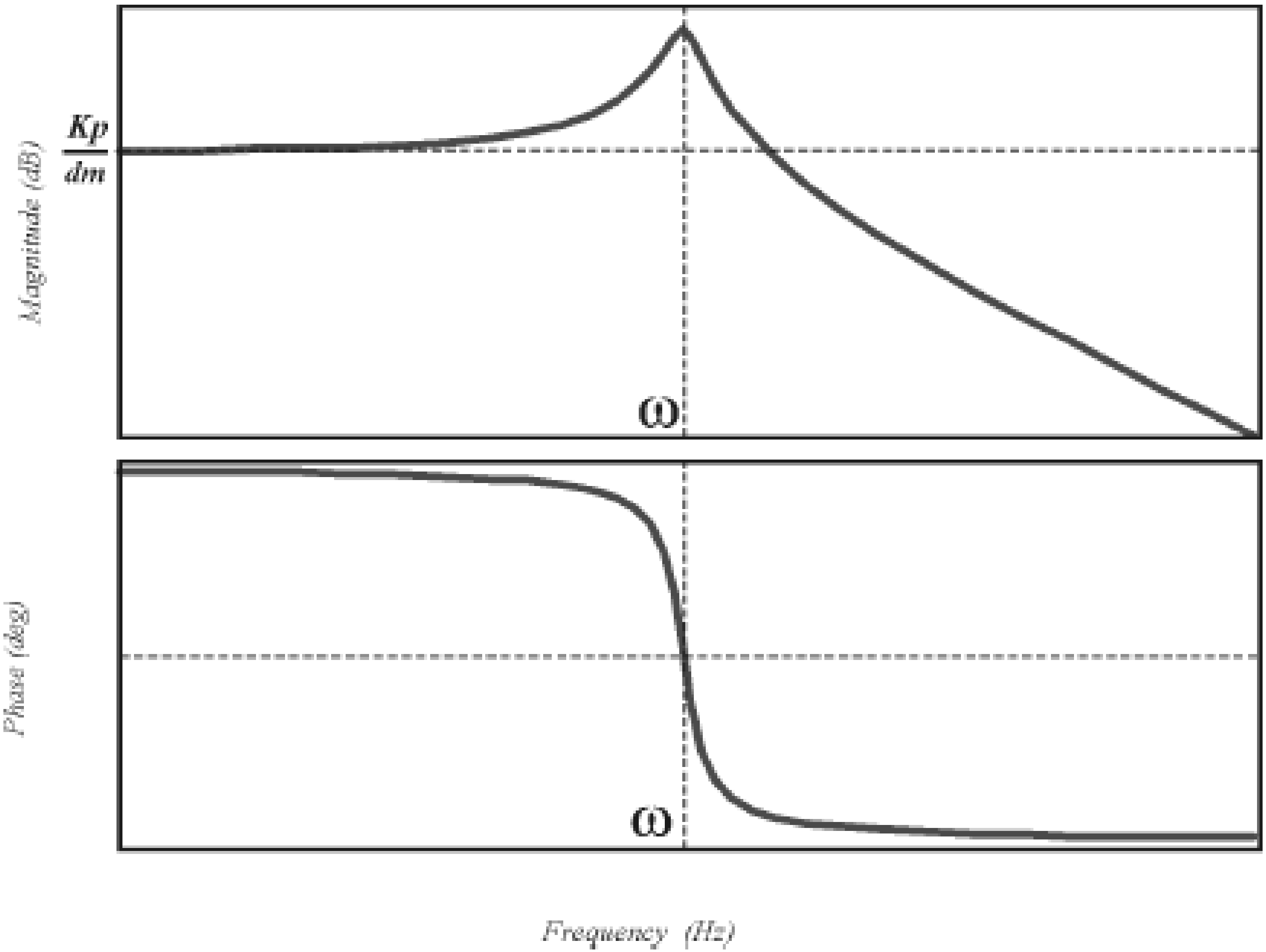
$$K_H = \frac{Bd_m^2}{V}$$

Solving for the natural frequency

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

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

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



Kp
dm
Magnitude (dB)
ω
Phase (deg)
ω
Frequency (Hz)

Example 3.2

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

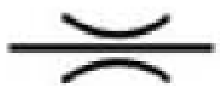
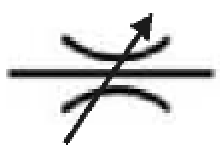
Leakage coefficient of pump and motor $\lambda = 0.01$ l/min/bar.
Load inertia $I = 300\ N.m.s^2$.
The motor displacement $d_m = 25$ ml/radian.
The maximum motor speed $\Omega_m = 200$ rpm.
Motor acceleration $\alpha = 1.05\ rad/s^2$.
Pump speed $\Omega_p = 1400$ rpm.
Overall effeiciecy $\eta = 85\%$.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

Example 3.2

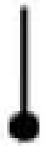
Neglecting the compressibility effect, calculate:

1. System pressure (pipe friction losses are negligible).
2. The actual pump capacity.
3. The power of the electric motor needed to drive the pump.
4. The time constant τ and the frequency response function Ω_m/y.

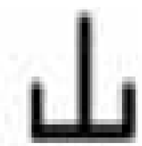
Fluid Power symbols

	Main line
	Pilot line
	Enclosure outline
	Hydraulic flow direction
	Pneumatic flow direction
	Flexible pipe line
	Constant flow restriction
	Variable flow restriction

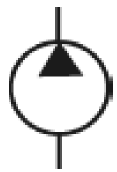
Pressure Compenation (small perpendicular arrow)



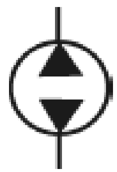
Temperature effect



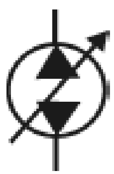
Vented reservoir (hydraulic)



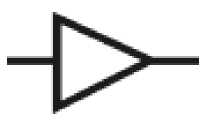
Single direction, fixed displacement pump



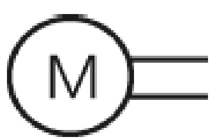
Reversible, fixed displacement pump



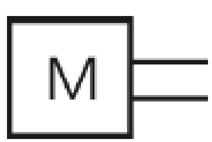
Reversible, variable displacement pump



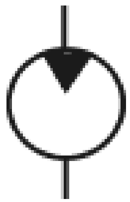
Air compressor



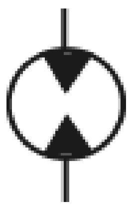
Electric motor



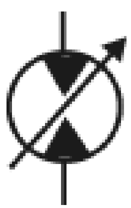
Internal combustion engine



Single direction, fixed displacement hydraulic motor

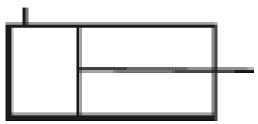


Reversible, fixed displacement hydraulic motor

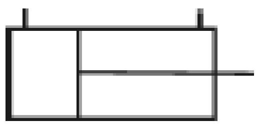


Reversible, variable displacement hydraulic motor

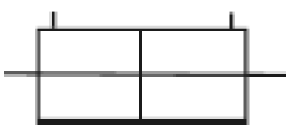
Single acting spring loaded actuator



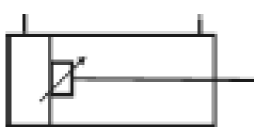
Simplified symbol of single acting actuator



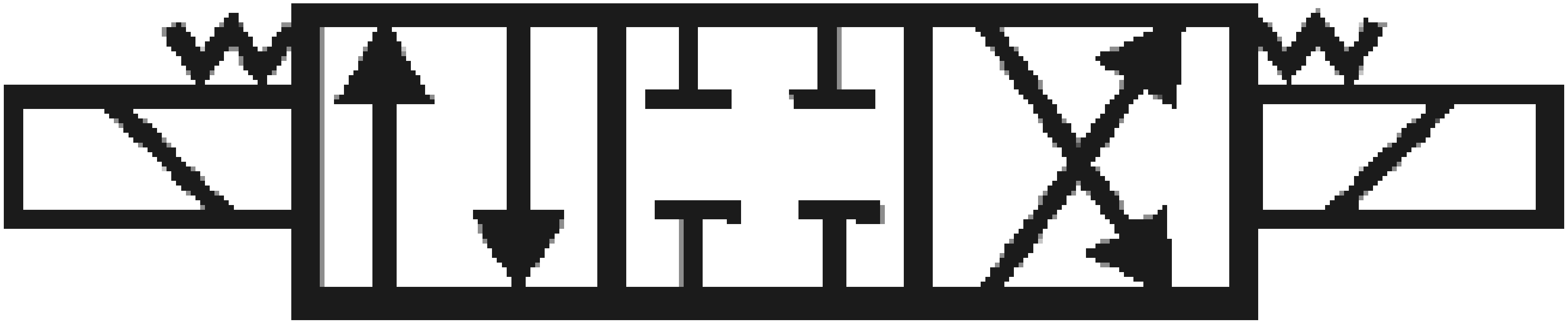
Double acting, single end rod actuator

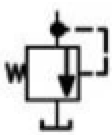
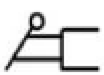
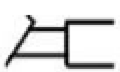


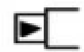
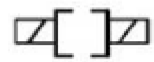
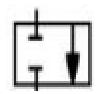
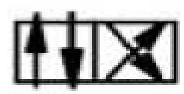
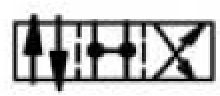
Double acting, double end rod actuator

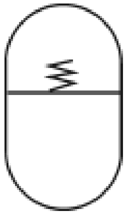


Double acting actuator with adjastable cushion

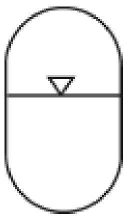


	Butterfly manual ON-OFF valve
	Non return (check) valve
	Pressure relief valve
	Manual hand control
	Pedal foot control

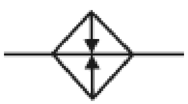
	Pressure pilot control
	electric solenoid control
	One position of a control valve
	Two position four port valve
	Three position four port valve



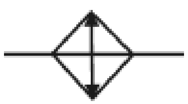
Spring loaded accumulator



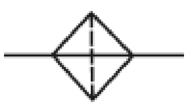
Gas loaded accumulator



Heater

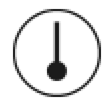


Cooler



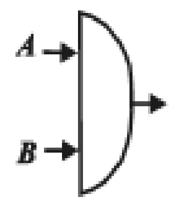
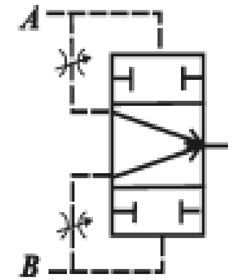
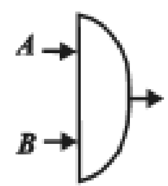
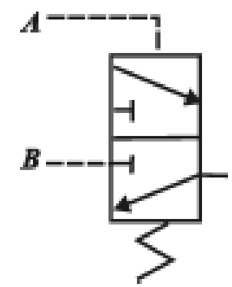
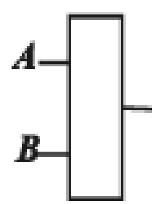
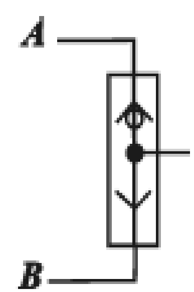
Filter or strainer

Pressure indicator

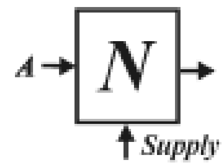


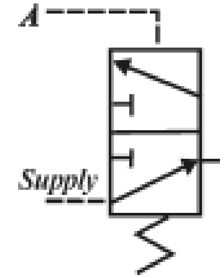
Temperature indicator

Filter Regulator Lubricator (FRL)

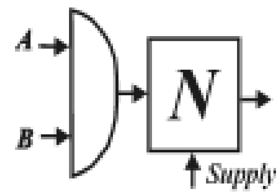
Control element	Logic symbol	Fluid power symbol
And		
Yes		
Or		

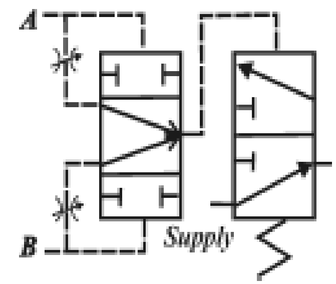
Not



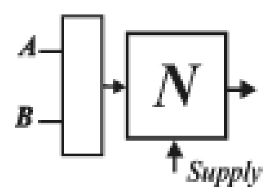


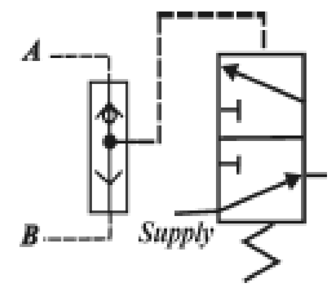
Nand



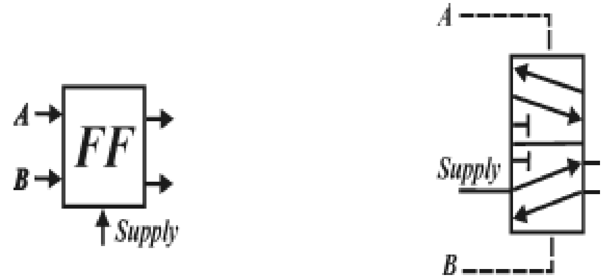


Nor

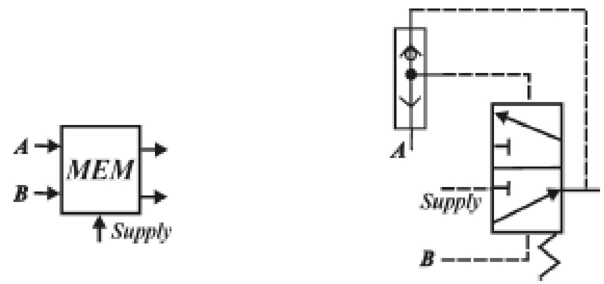




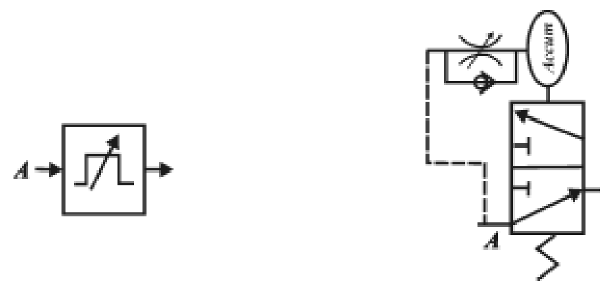
Flip flop



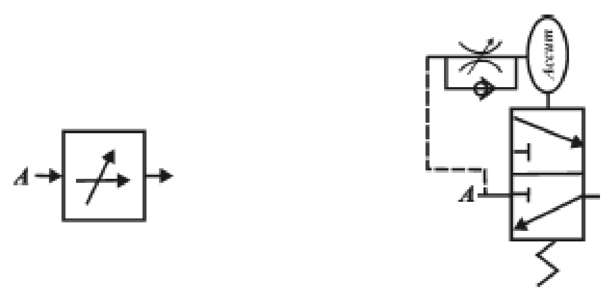
Memory



Step shot



Delay timer



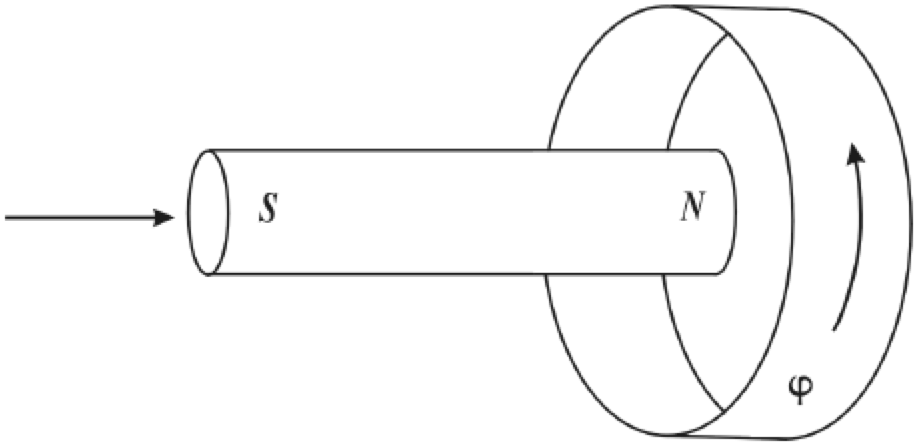
Electro-mechanical control

Electromagnetism

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

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

$$V = BlU$$



$$V = GU$$

G is the *transduction constant* (in [V/m/s] or [N/A])

Lorentz force constant

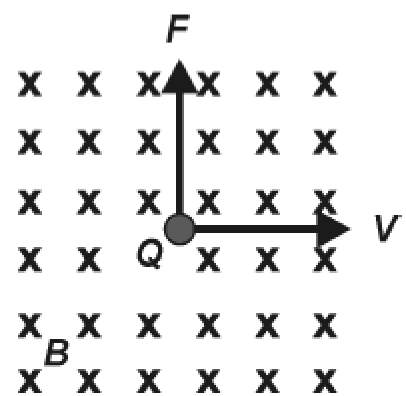
Force in magnetic field

$$F = Q(V \times B)$$

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

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

$$dl = Vdt$$

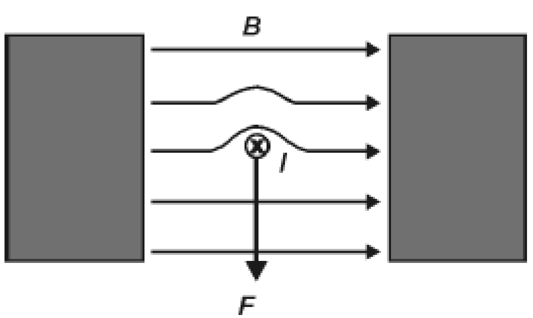


(B into the page)

$$F = I(L \times B) = ILBSin\theta$$

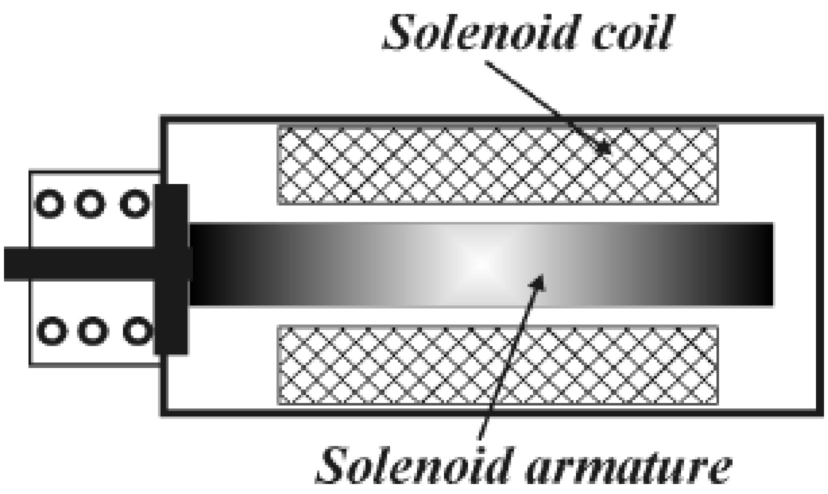
$$F = ILBN$$

$$L = 2\pi R.\ \text{Here},\ Sin\theta = 1$$

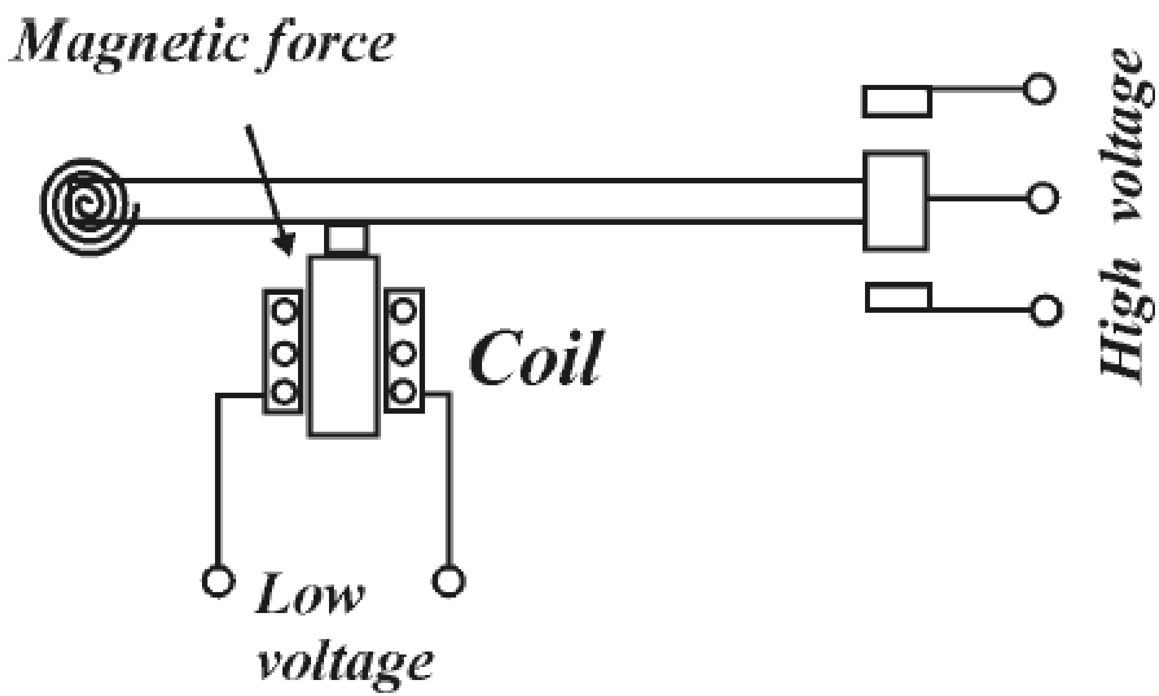


(I into the page)

Solenoid

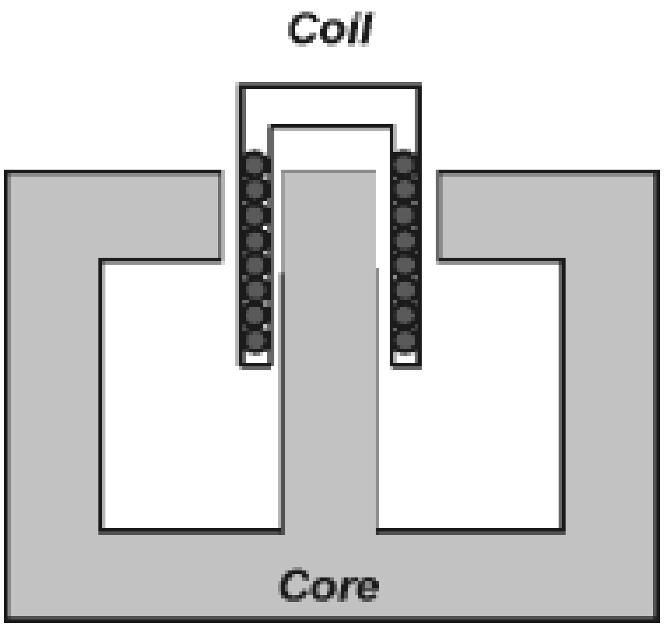


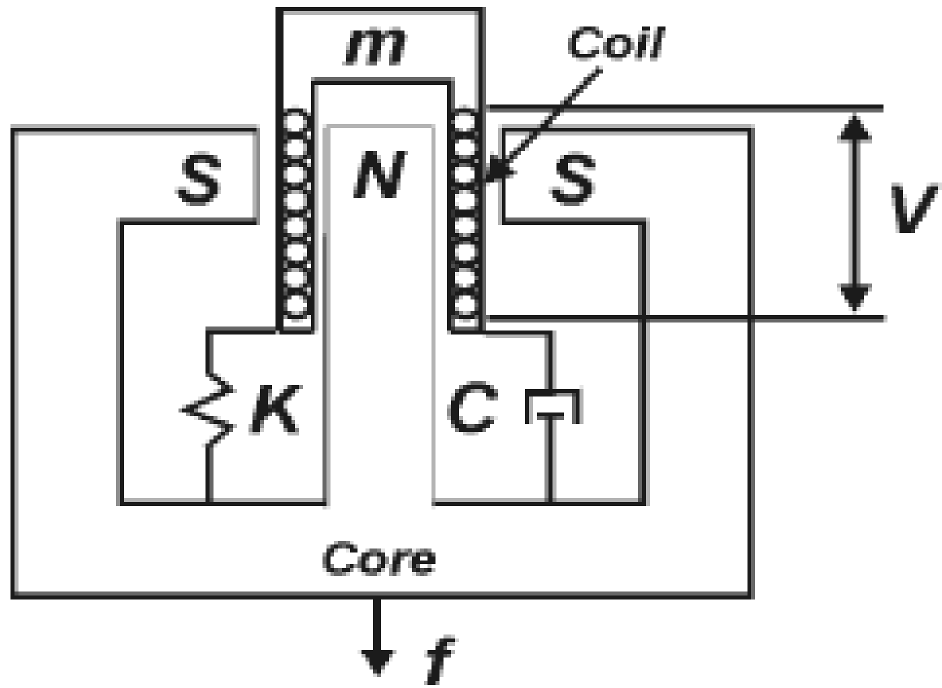
Relay



Normally open (NO)
Normally closed (NC)
Change Over (CO)

Voice coil





Voice coil modelling

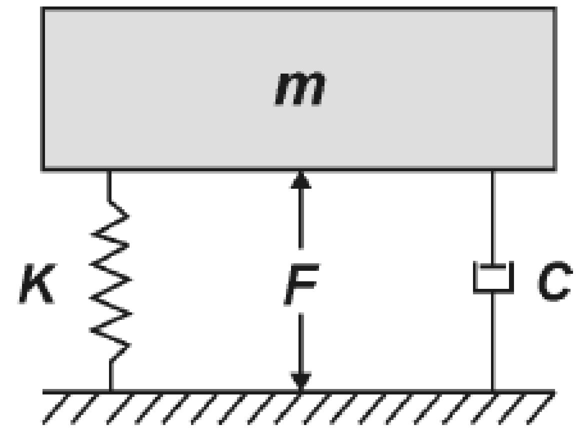
$$m\ddot{x} + C\dot{x} + Kx = F$$

$$F \;=\; GI, \text{ where } G \text{ is the } \mathit{Lorentz\ \ force\ \ constant}$$

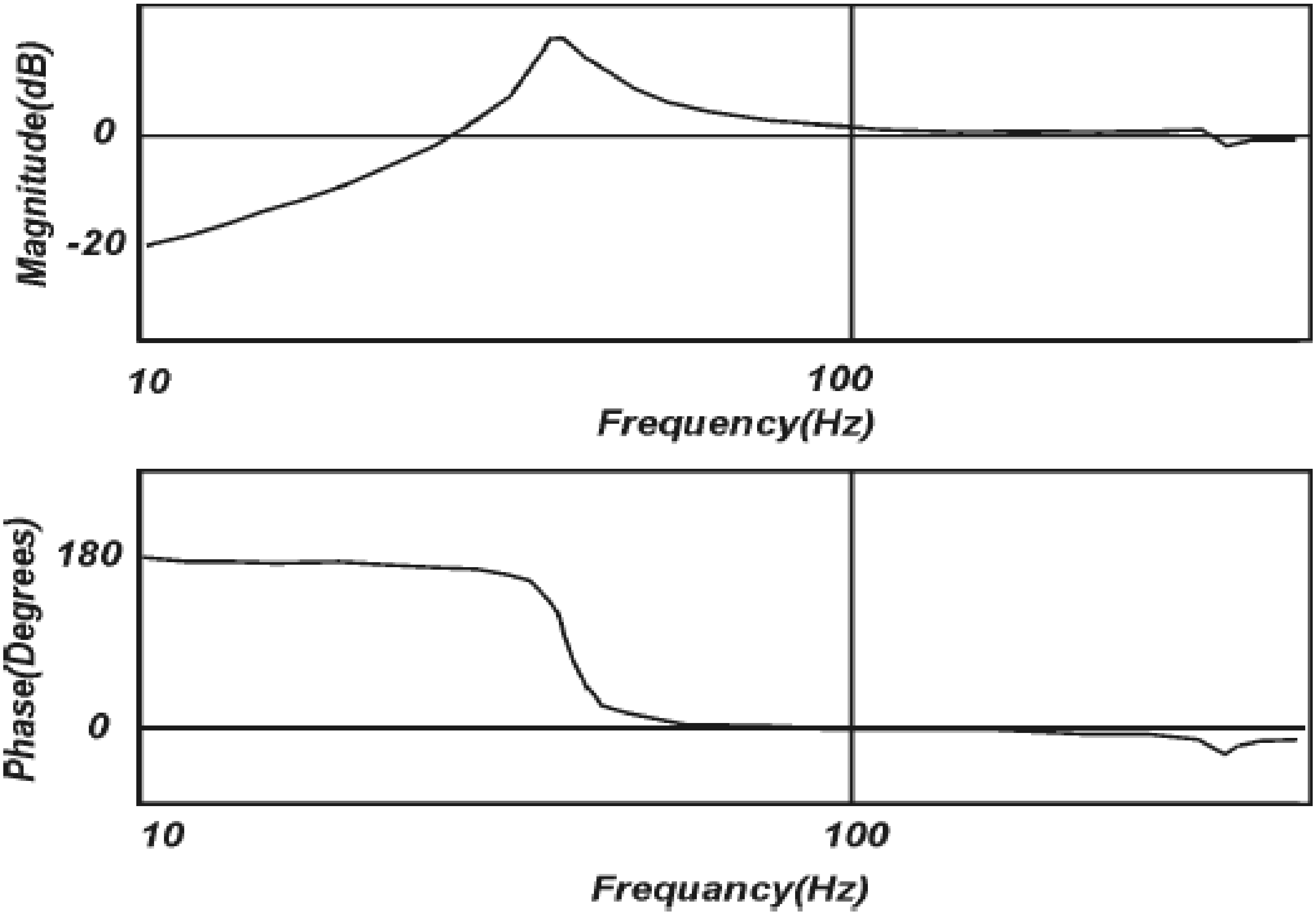
$$s^2x + 2\xi\omega_n sx + \omega_n^2 x = \frac{G}{m}I$$

$$\frac{x}{I} = \frac{\frac{G}{m}}{s^2 + 2\xi\omega_n s + \omega_n^2}$$

$$\frac{F}{I} = \frac{-Gs^2}{s^2 + 2\xi\omega_n s + \omega_n^2}$$

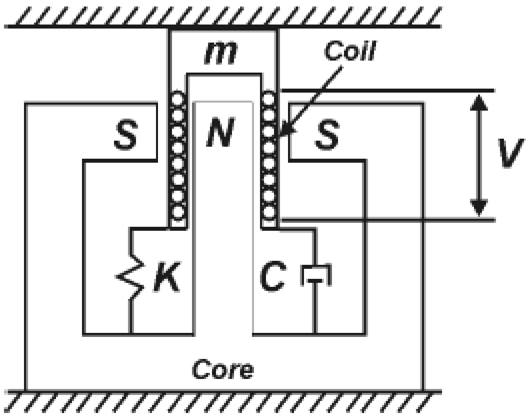


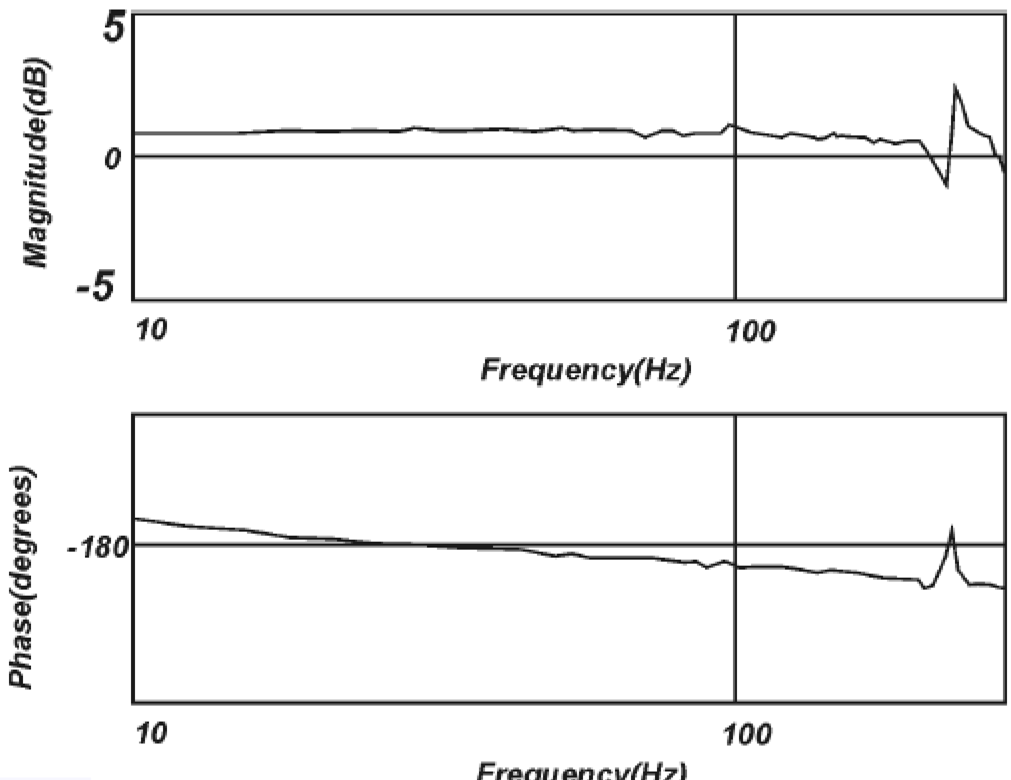
$$\omega_n^2 = \frac{K}{m} \qquad\qquad 2\xi\omega_n = \frac{C}{m}$$



Magnitude(dB)
0
-20
10
100
Frequency(Hz)
Phase(Degrees)
180
0
10
100
Frequancy(Hz)

Finding G


m
Coil
S
N
S
V
K
C
Core


5
0
-5
Magnitude(dB)
10
100
Frequency(Hz)
-180
Phase(degrees)
10
100
Frequency(Hz)

Configurations

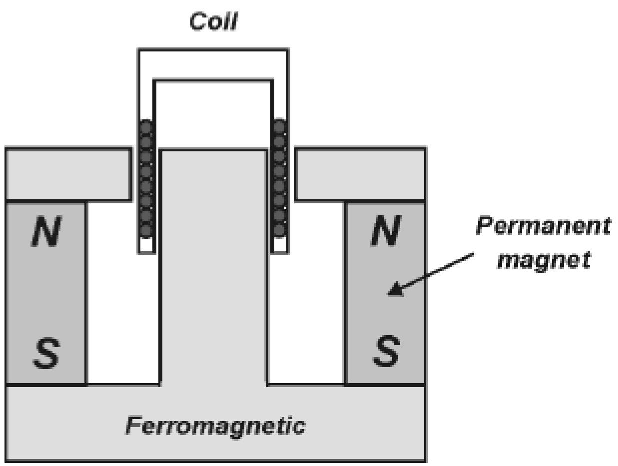


Figure 4.14: *Axial toroid voice coil actuator*

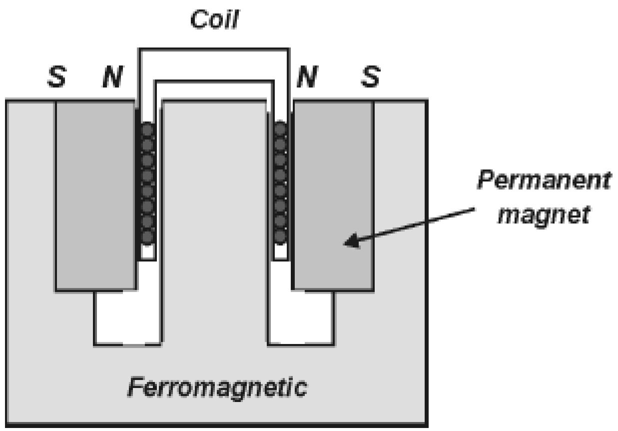


Figure 4.13: *Radial toroid voice coil actuator*

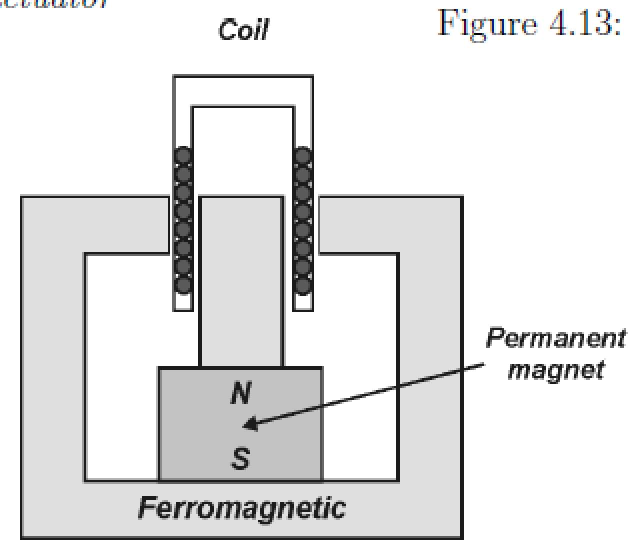


Figure 4.15: *Axial disk voice coil actuator*

Servo valves

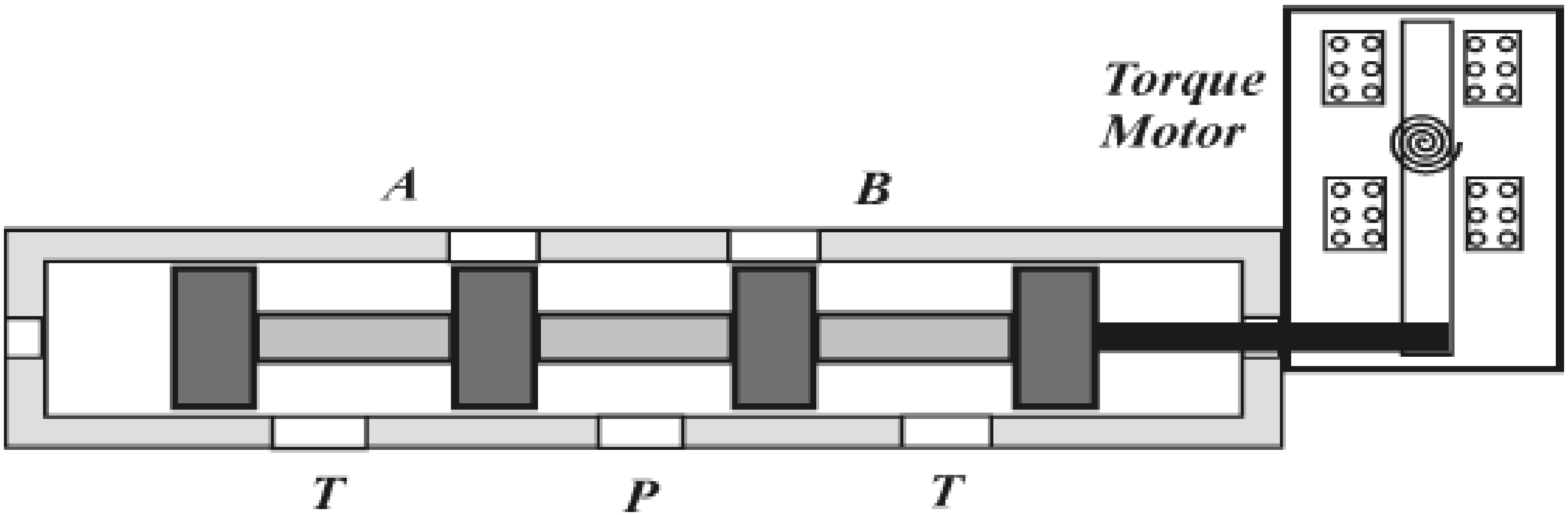


Figure 4.16: *Single stage spool valve directly operated by a torque motor*

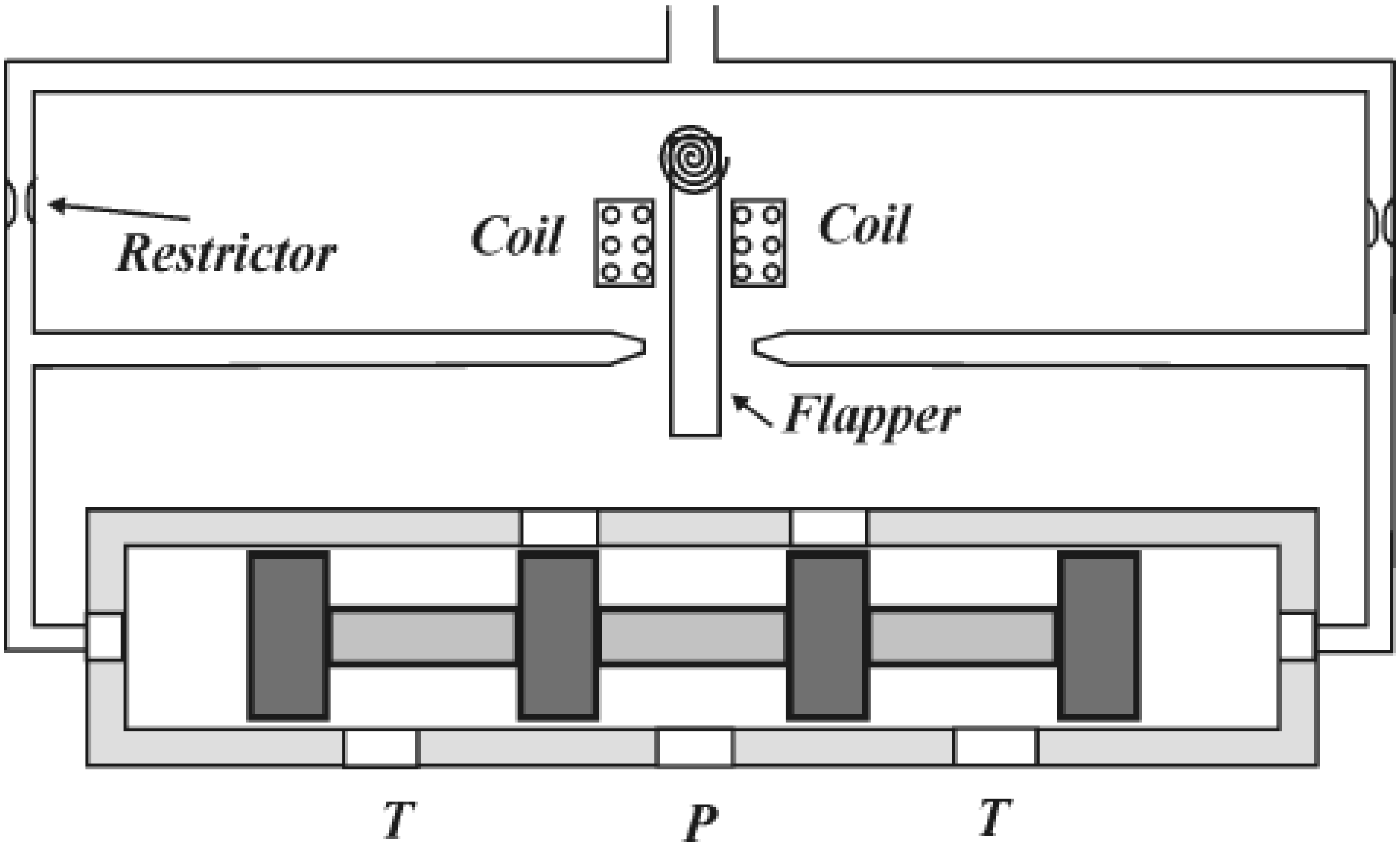


Figure 4.17: *Flapper servo valve*

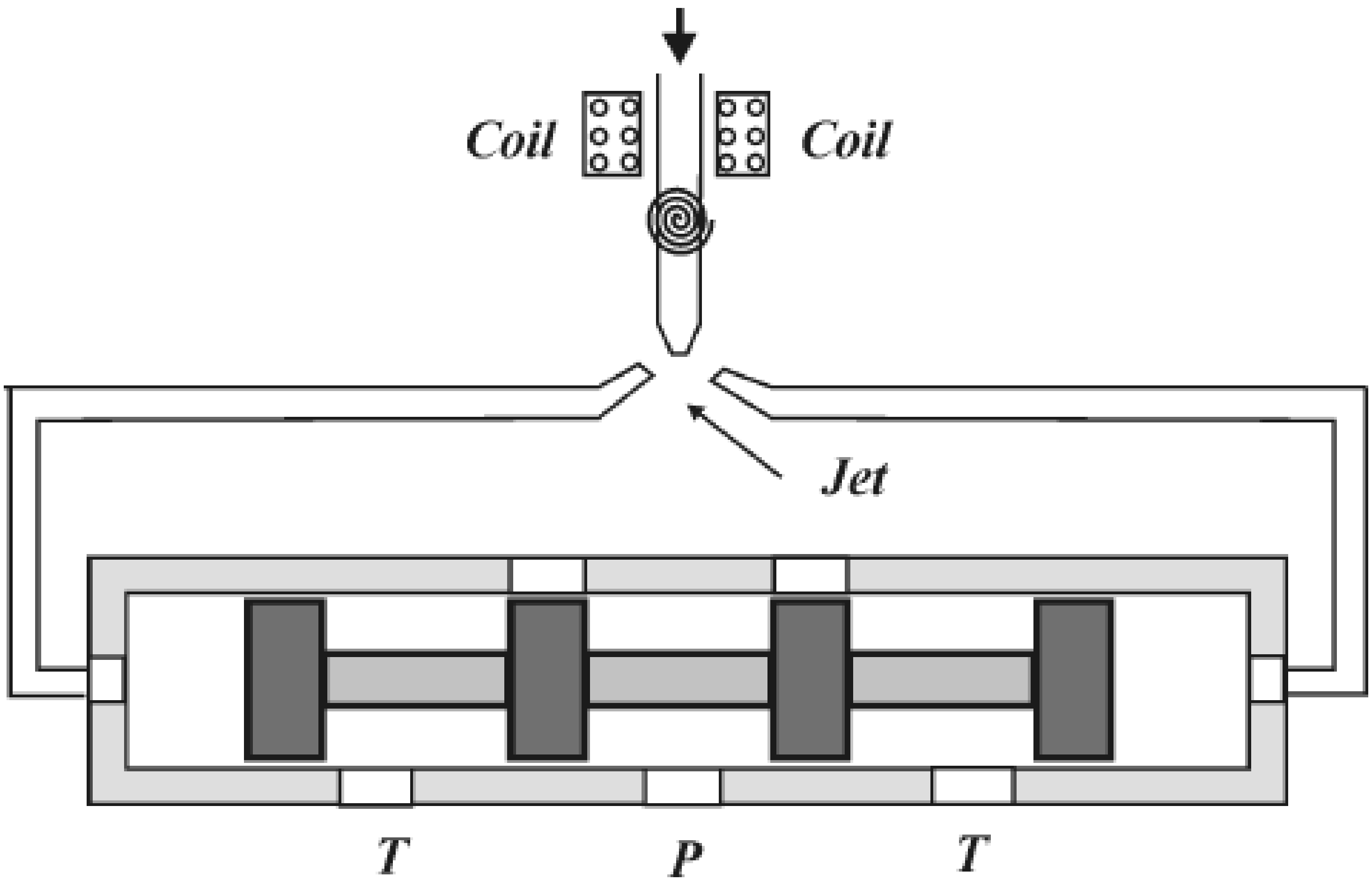


Figure 4.18: *Jet servo valve*

Proportional valves

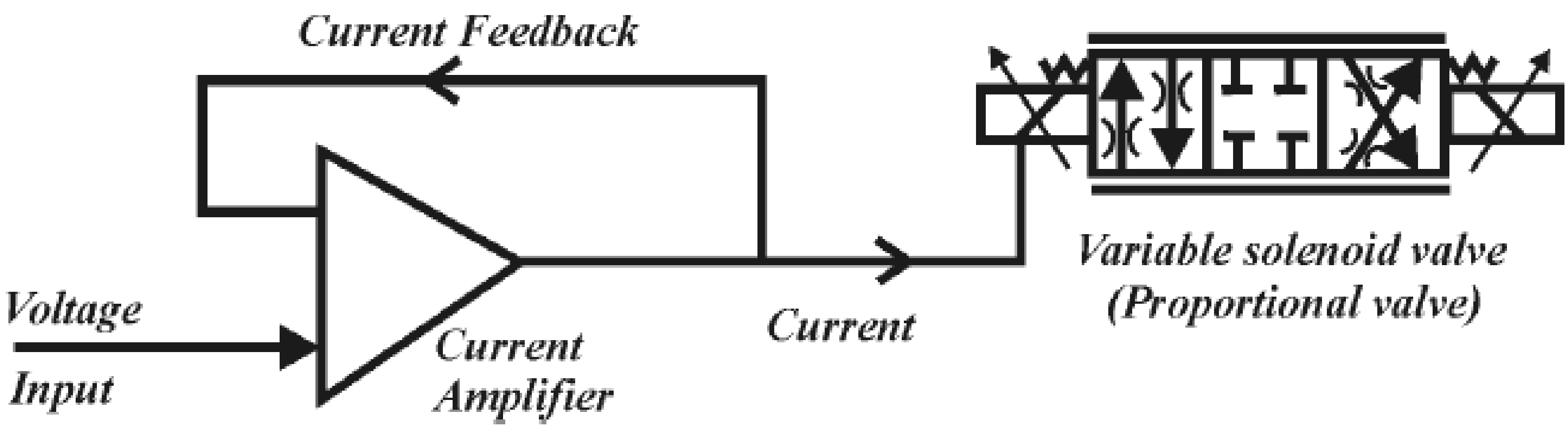


Figure 4.19: *Current control of a proportional solenoid valve*

Feedback sources

- *Hysteresis control:* the output current of the amplifier is measured and fed back to the amplifier again to reduce the hysteresis effect which is considered very much higher in proportional valves than servo valves.

- *Spool position control:* a displacement transducer can be attached to the spool of the valve measuring its position and feeding it back to the current amplifier to be corrected according to the required position.

- *Load speed or position control:* the position or the speed of the output load can be measured and fed back to the amplifier to be determined as desired by the operator.

Amplifier

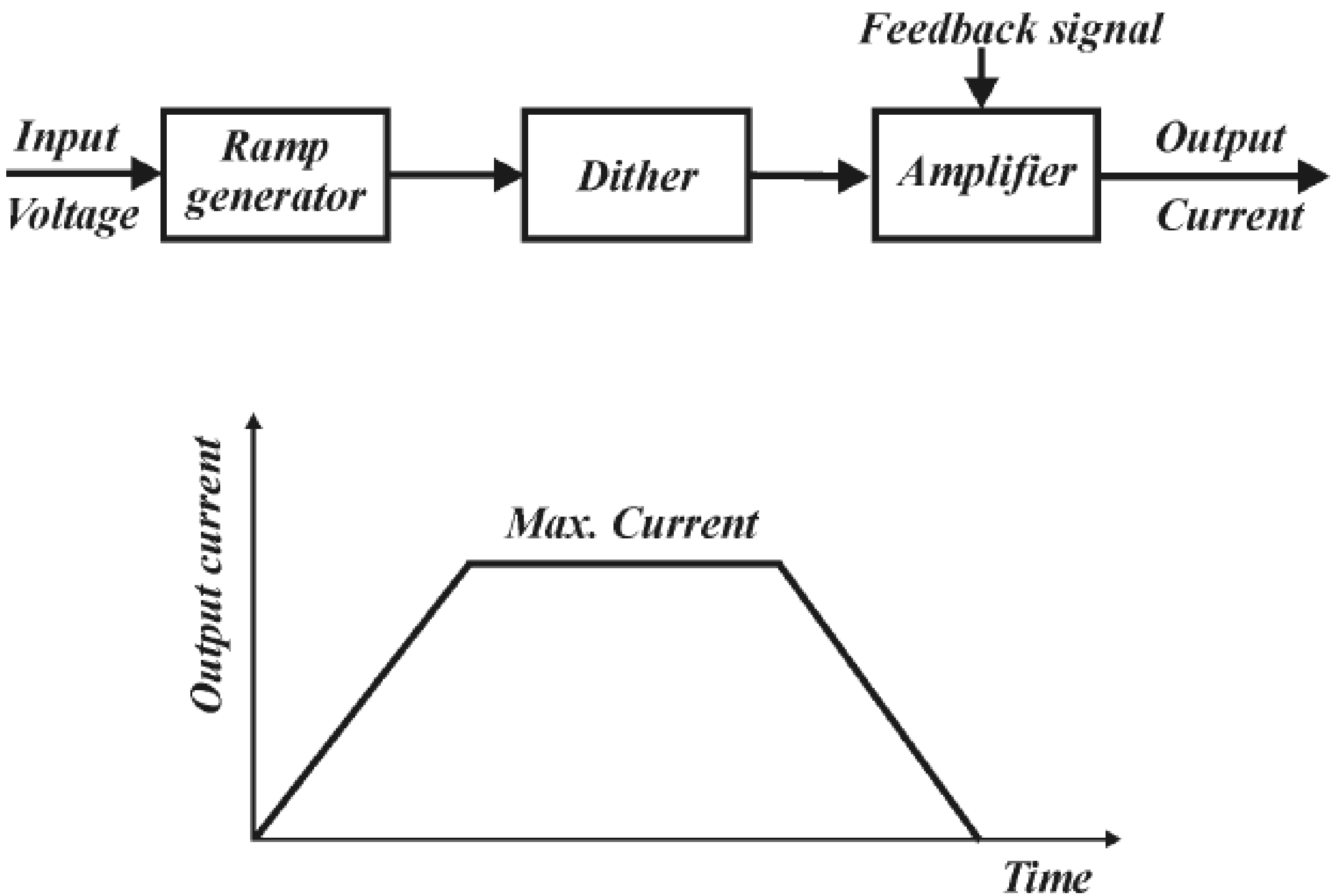


Figure 4.20: *Block diagram and time response of the amplifier used for proportional valves*

Comparison

Characteristic	Proportional valve	Servo valve
Valve lap	Overlap with dead zone	Zero or underlap without dead zone
Response time	40 - 60 ms	5 - 10 ms
Operating frequency	10 Hz	100 Hz
Hysteresis	1% - 5%	0.1%

Dead zone

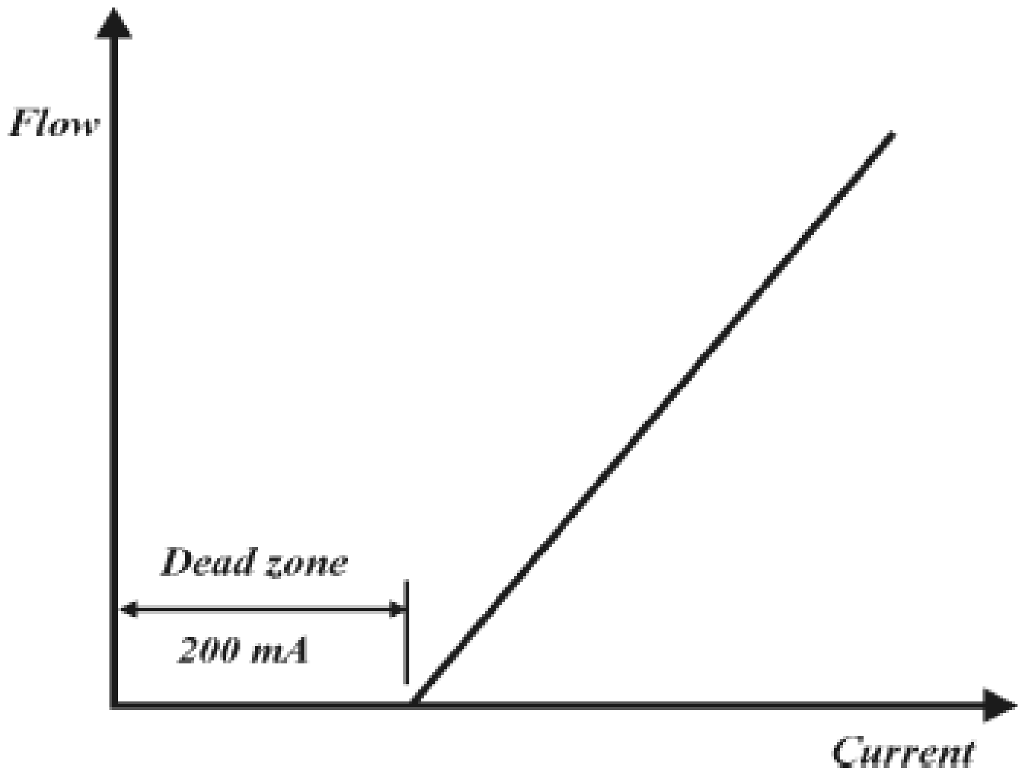


Figure 4.21: *Flow current relationship in proportional valves*

Notches

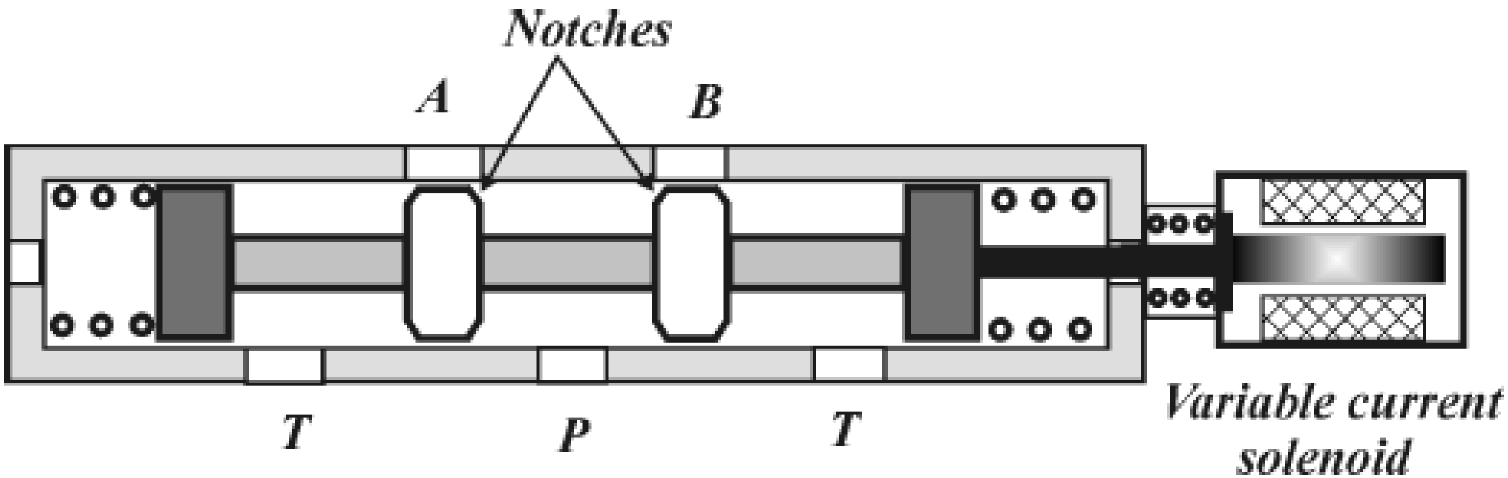


Figure 4.22: *Proportional directional control valve with notched spool*

Spool position control

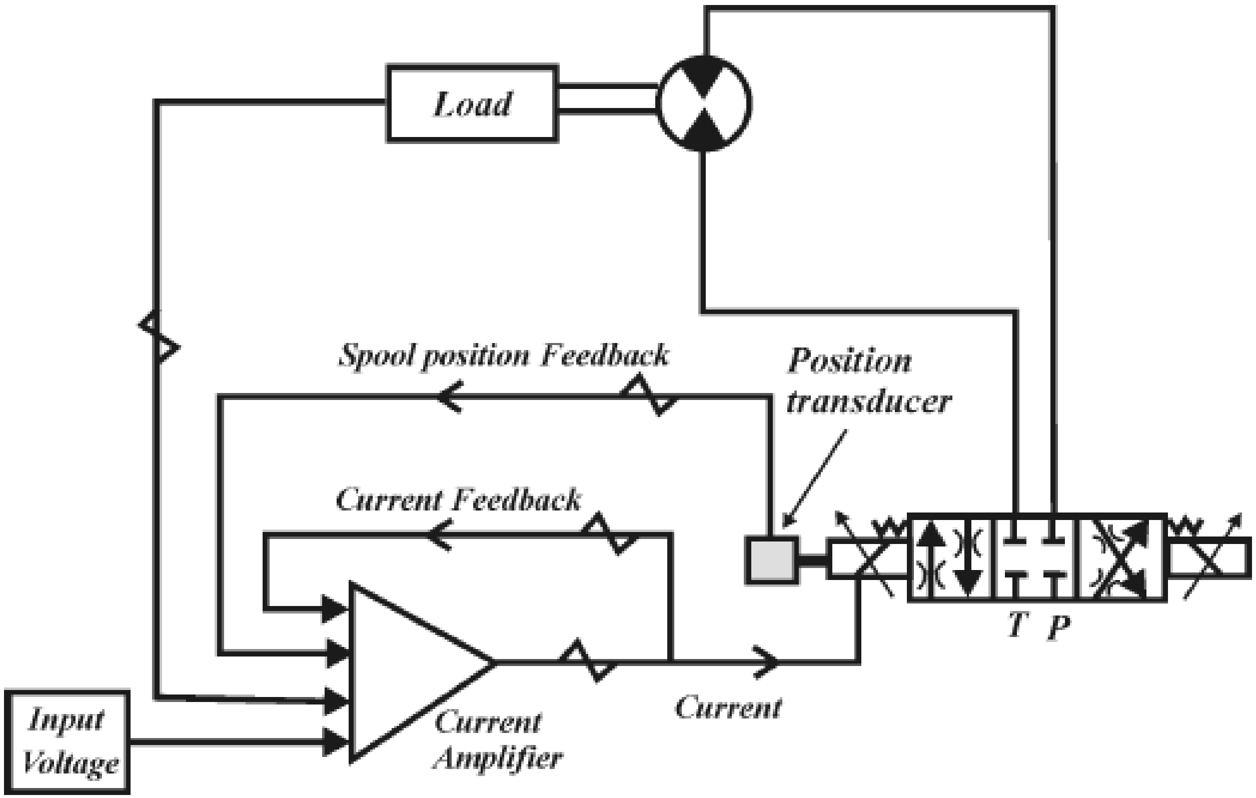


Figure 4.23: *Closed-loop speed control with spool position control for a hydraulic motor*

Pressure control

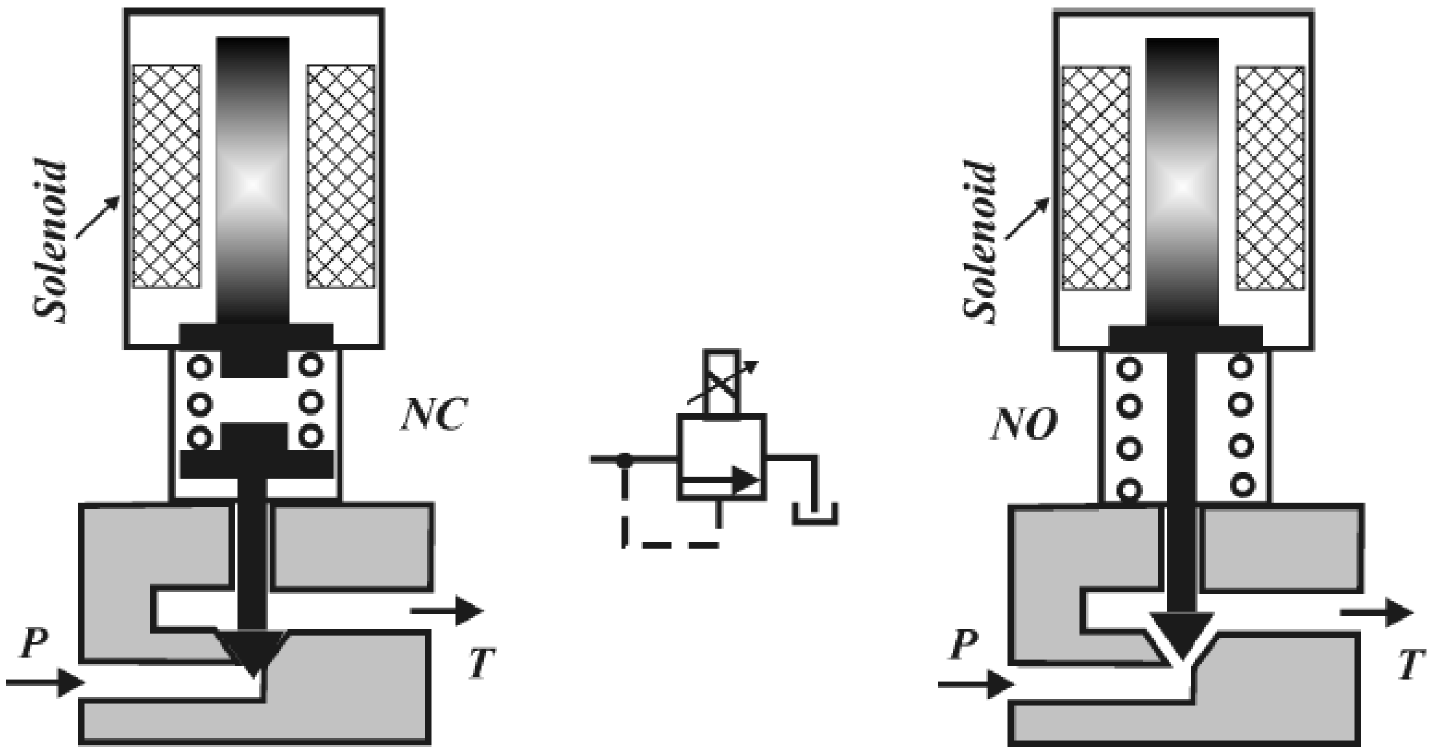


Figure 4.24: *Pressure relief valve with solenoid control; Normally Open (NO) and Normally Closed (NC)*

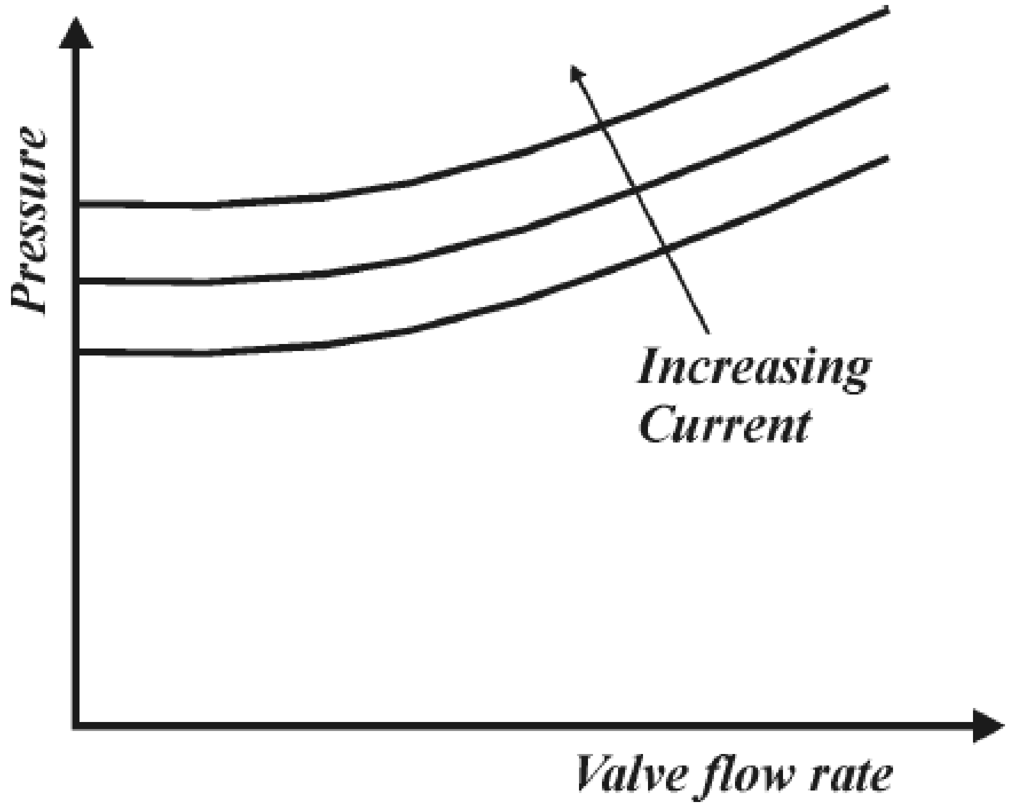
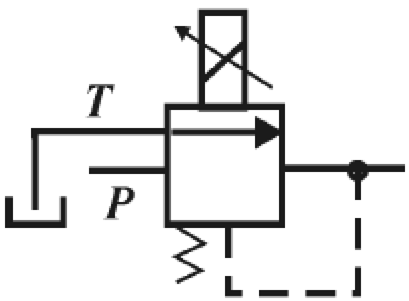


Figure 4.25: *Relationship between pressure and flow rate of fluid through the valve with increasing the control current of the solenoid for NC valve*

Pressure reduction



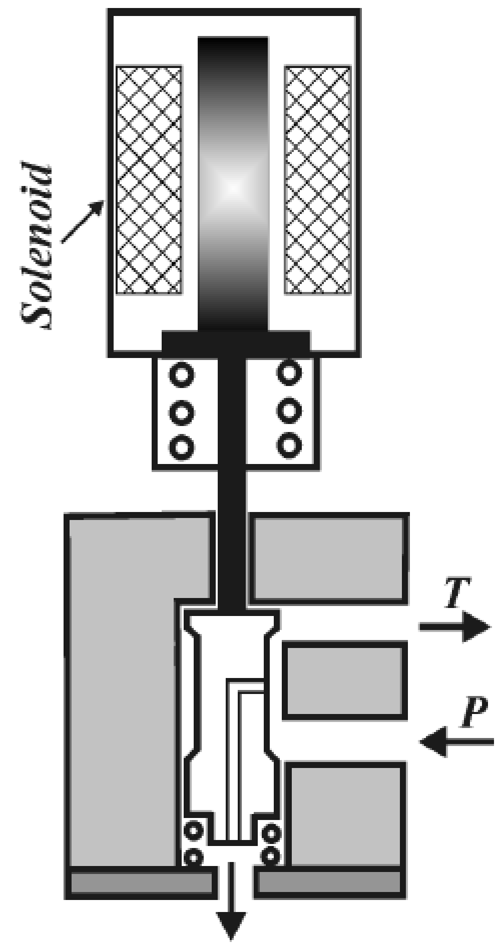
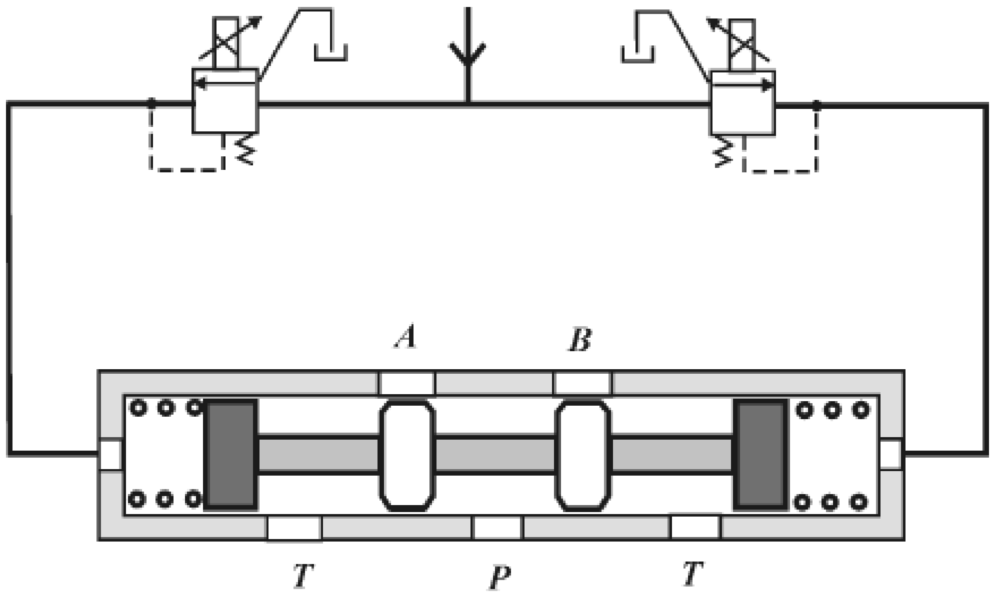


Figure 4.26: *Pressure reduction valve*

Two stage control



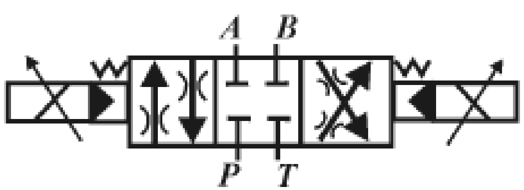


Figure 4.27: *Two stage proportional directional control valve*

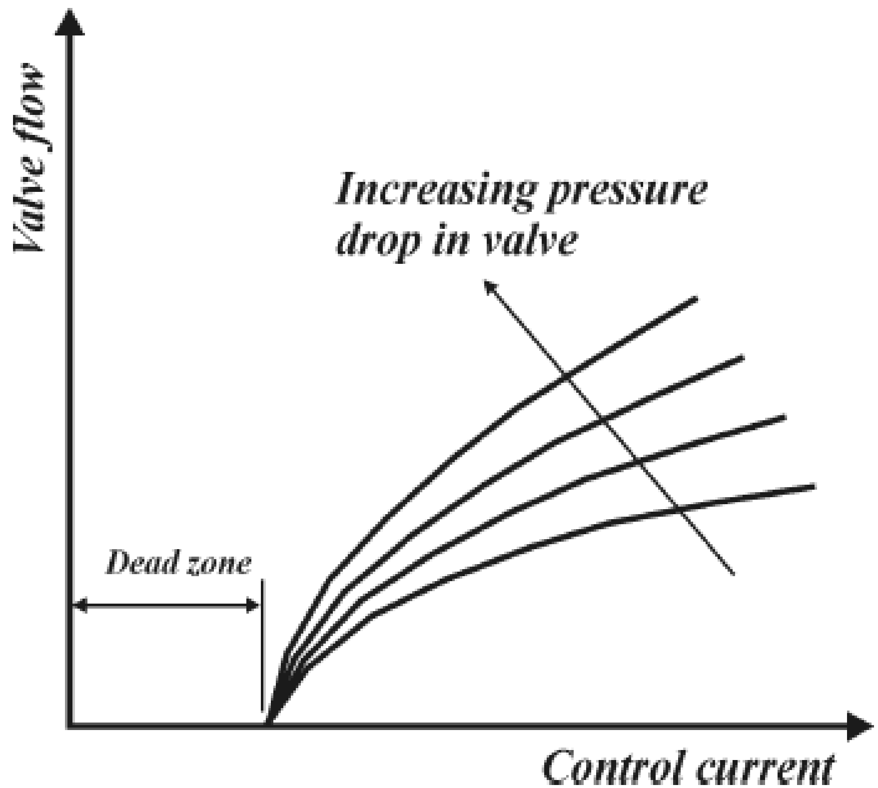


Figure 4.28: *Flow-current relationship with increasing pressure drop in two-stage proportional valve*

Double flow control

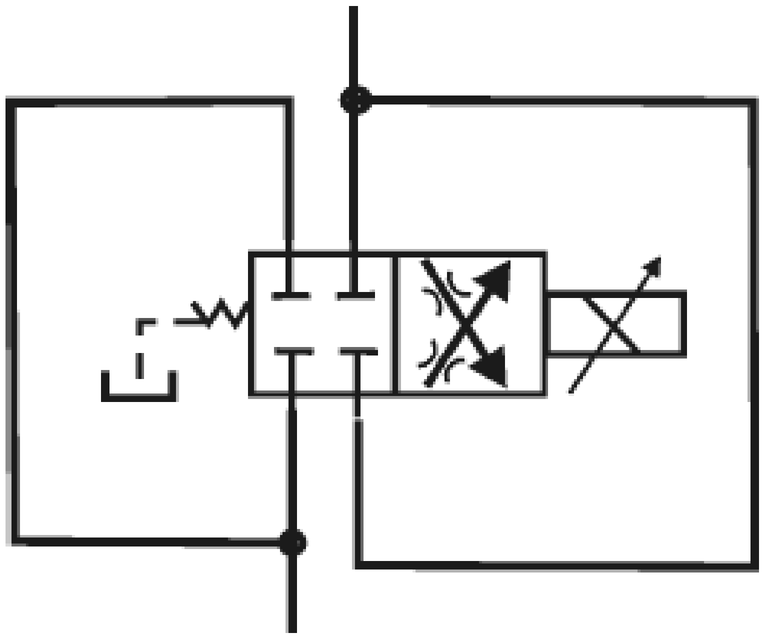


Figure 4.29: *Double flow of four port directional control valve*

Pressure compensation

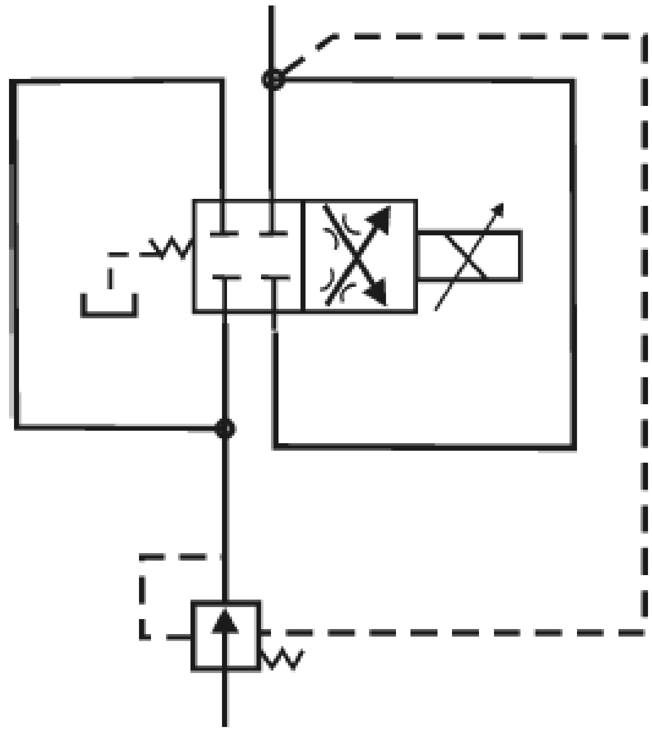


Figure 4.30: *Pressure compensation in double flow of four port directional control valve*

Actuator control

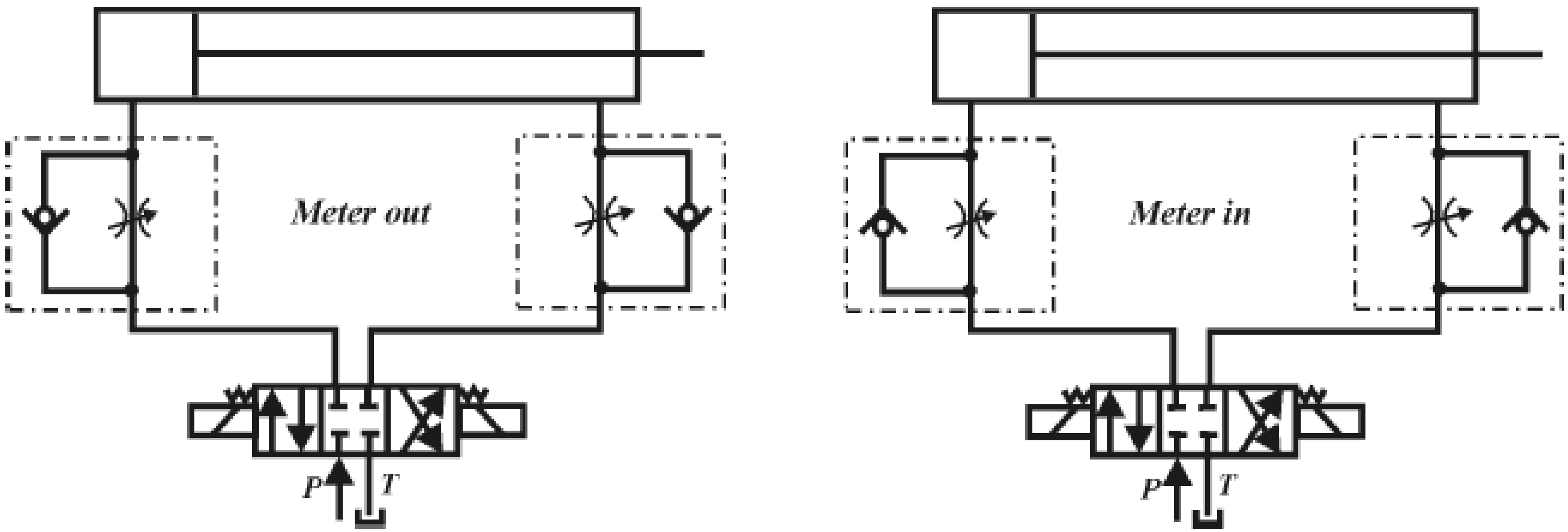


Figure 4.31: *Meter-in and meter-out actuator speed control*

Actuator control

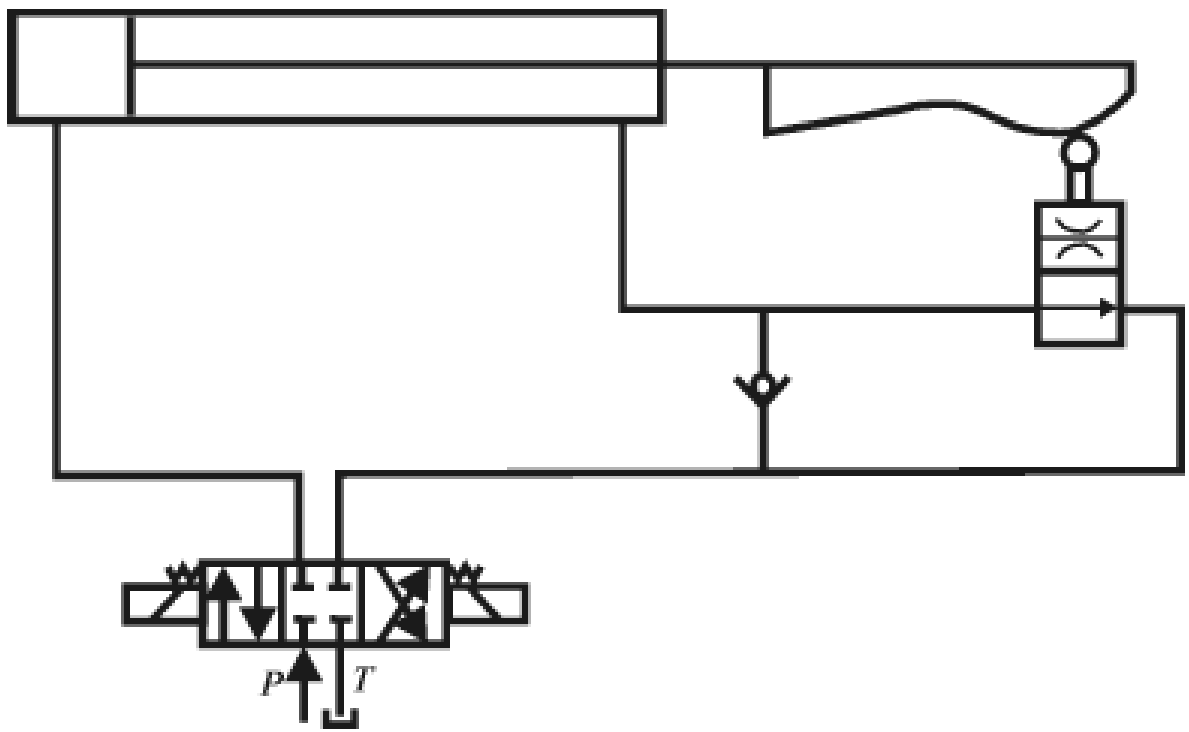


Figure 4.32: *Cam operated speed control of the actuator*

Pump control

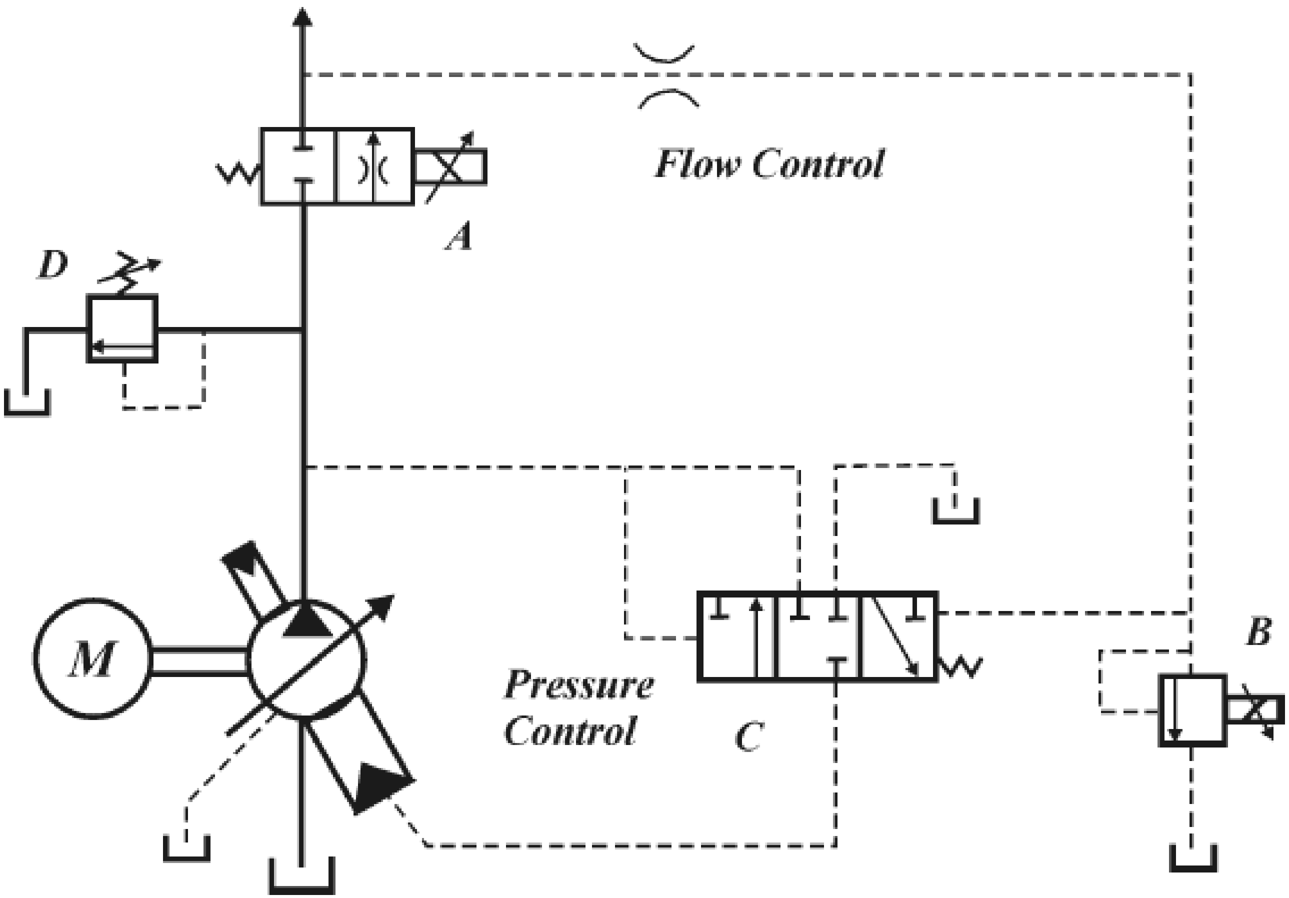


Figure 4.33: *Pump proportional control, pressure and flow*

Fluid Power Control

Ahmed Abu Hanieh

Hydraulic circuits

Hydraulic circuit components

- Hydraulic tank (reservoir filled with hydraulic liquid).
- Hydraulic pump (reciprocating, gear, vane or any other type).
- Hydraulic actuator (linear piston or rotary motor).
- Hydraulic valve (pressure relief, flow control or directional control).
- Prime mover (electric or internal combustion motor)

Hydraulic circuit design

- Saftey of operator and operation.
- Efficiency and performance of the whole system.
- Cost requirements.
- Simplicity and easiness.

Amercian National Standards Institute (ANSI)

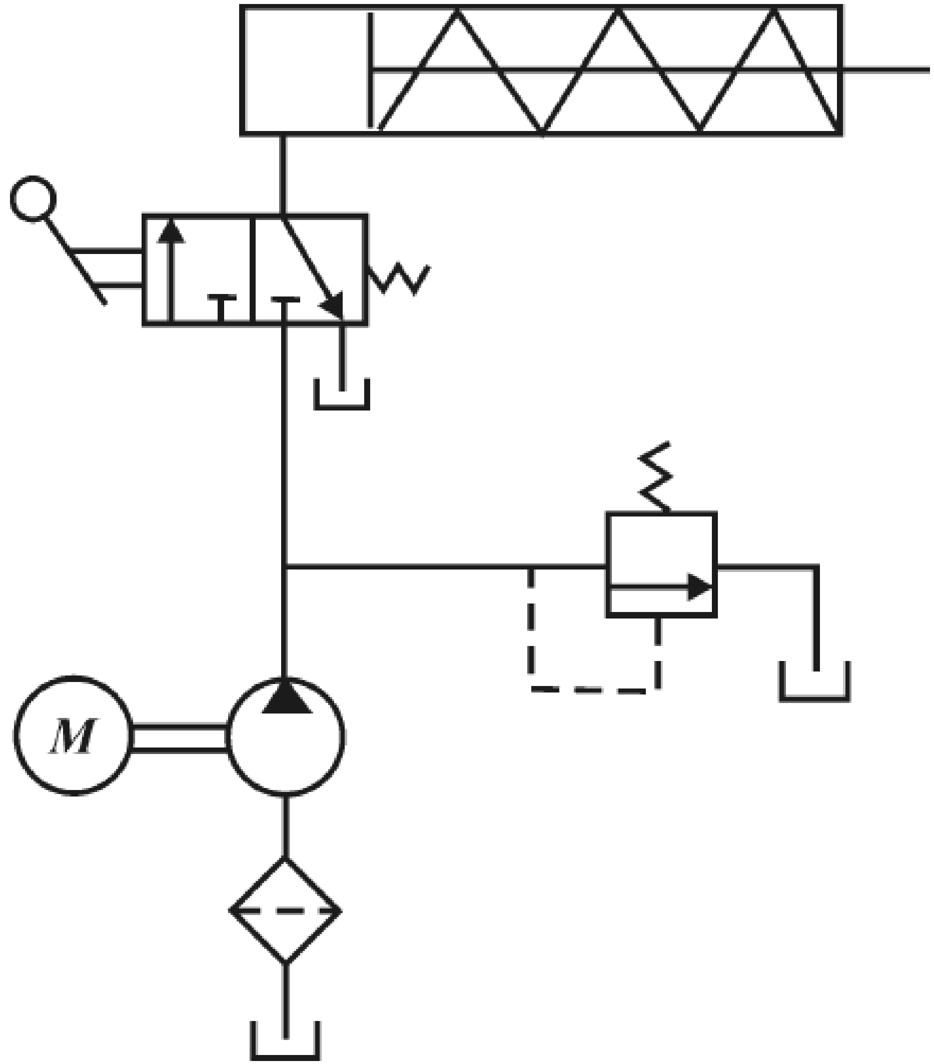


Figure 5.1: *Hydraulic single-acting cylinder*

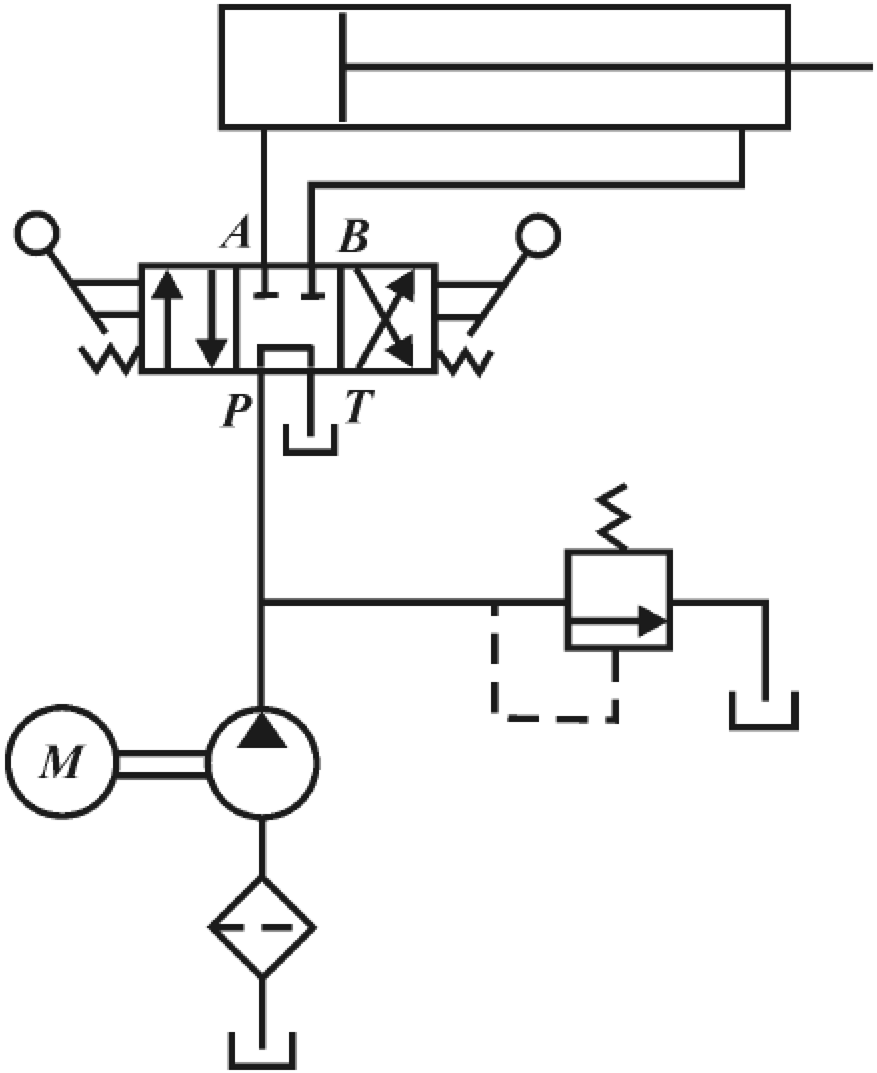


Figure 5.2: *Hydraulic double-acting cylinder*

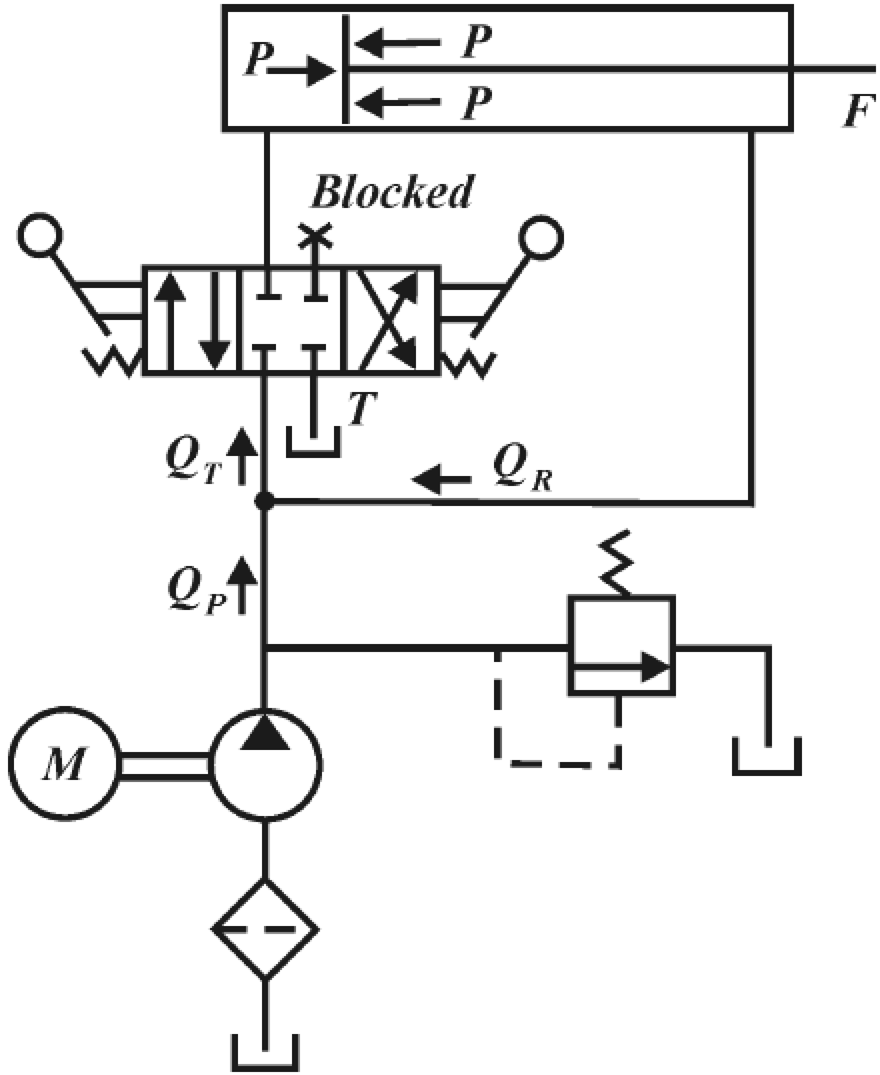


Figure 5.3: *Regenerative hydraulic cylinder circuit*

Estimating the flow rate of the pump in the extension stroke of the piston:

$$Q_P = Q_T - Q_R$$

Denoting the extension speed of the piston as v, the area of the piston side as A_p and the area of the rod side as $A_p - A_r$, the pump flow rate becomes:

$$Q_P = A_p v - (A_p - A_r)v$$

Solving for the extension speed of the piston gives:

$$v = \frac{Q_P}{A_r}$$

But the force exerted by the regenrative circuit is calculated by:

$$F = PA_r$$

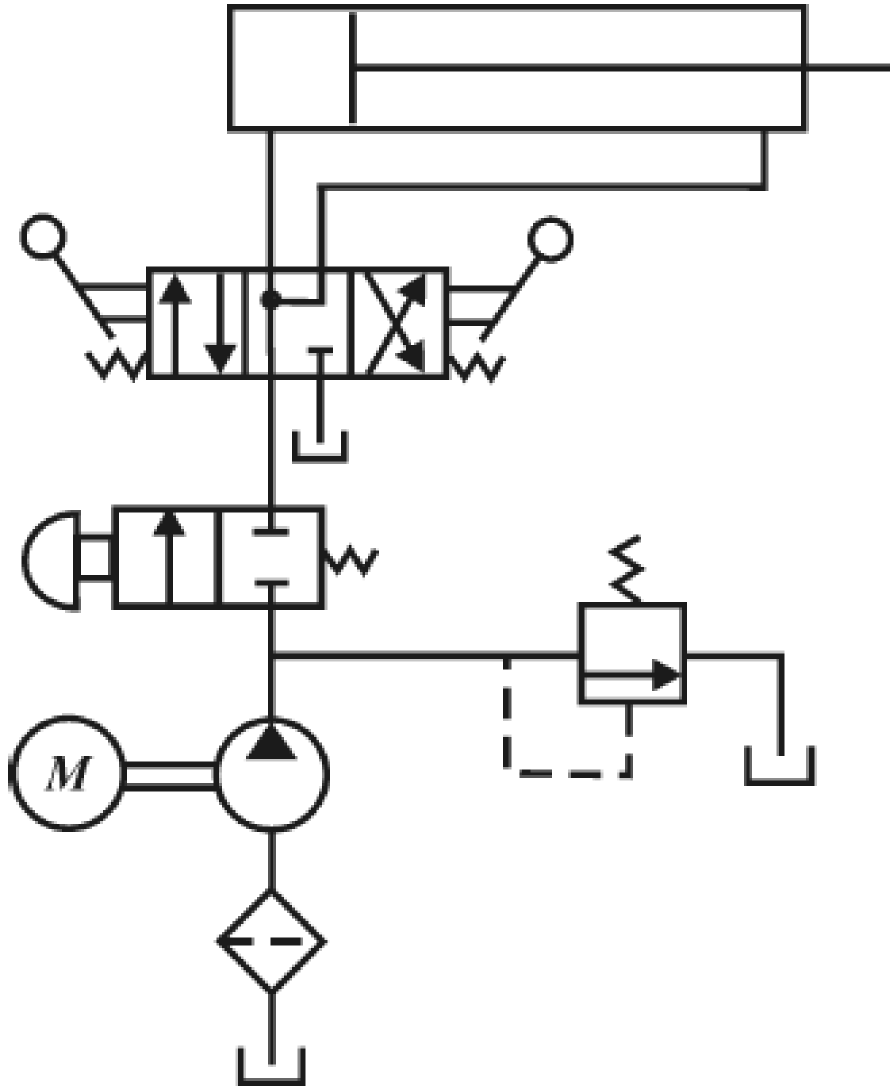


Figure 5.4: *Drilling machine circuit*

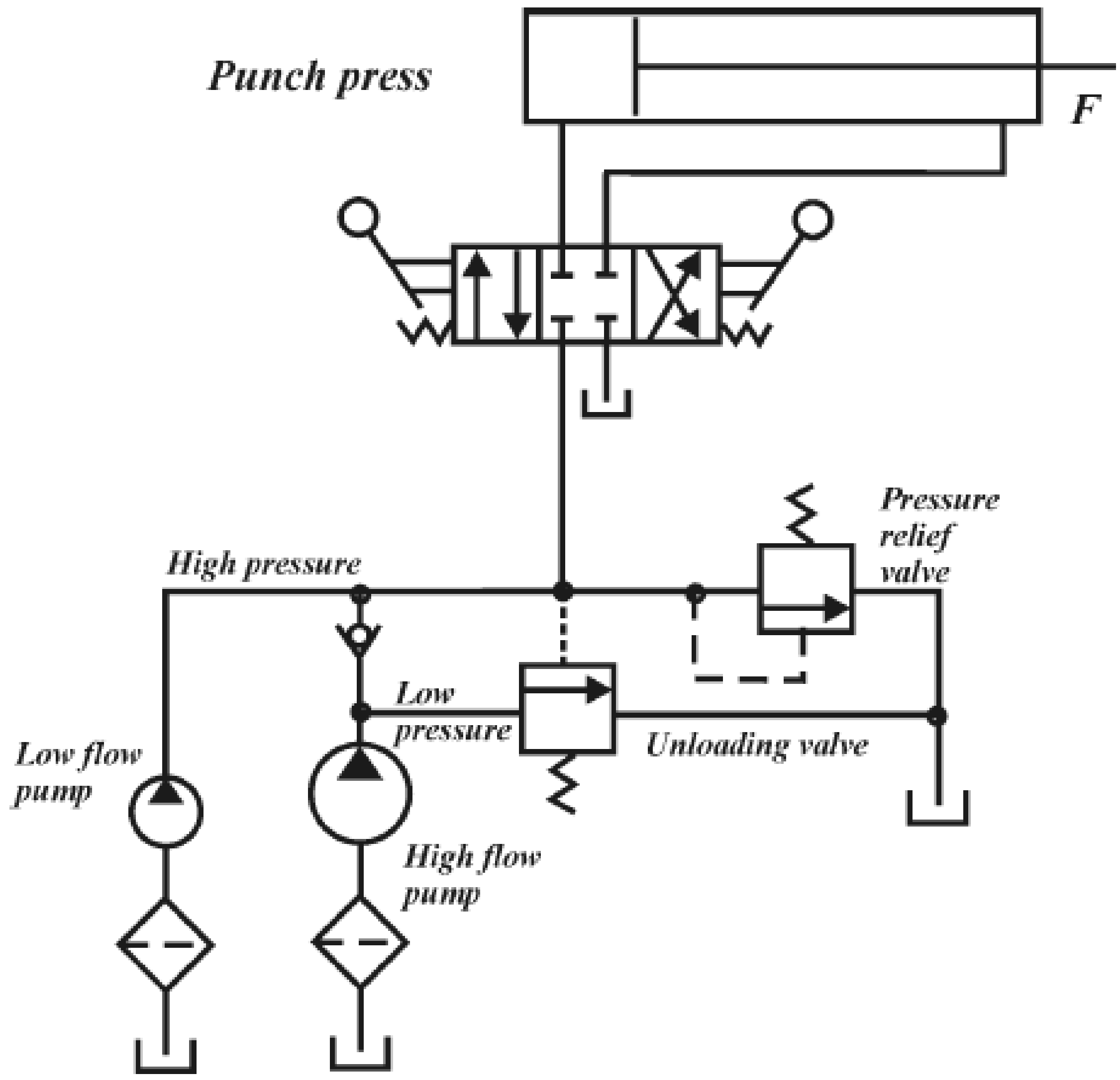


Figure 5.5: *Double pump hydraulic system (Punch press)*

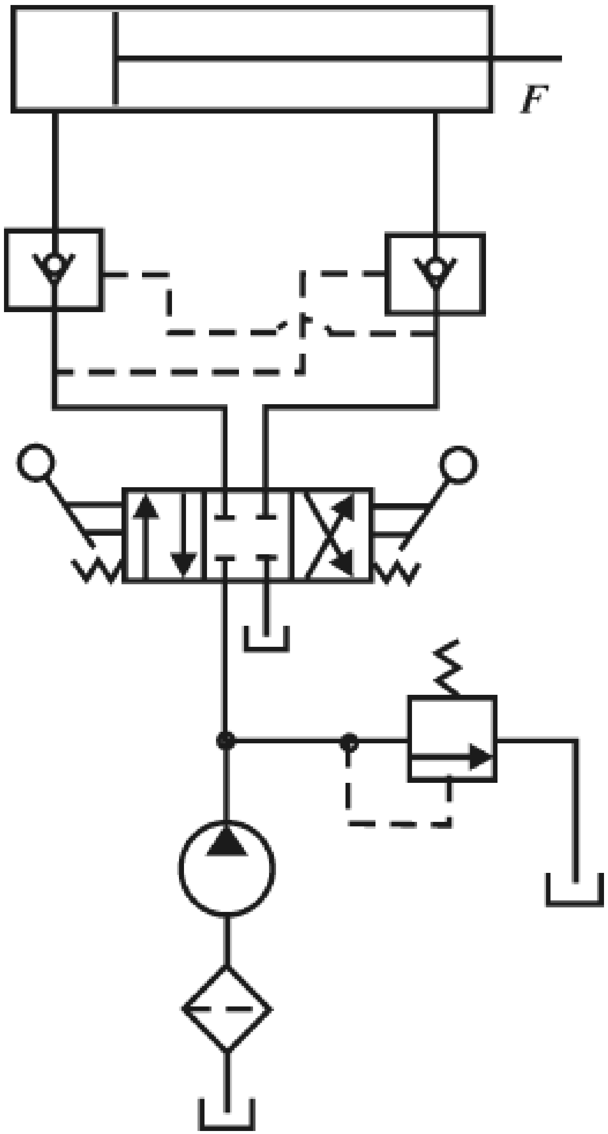


Figure 5.6: *Locked cylinder hydraulic system using check valves*

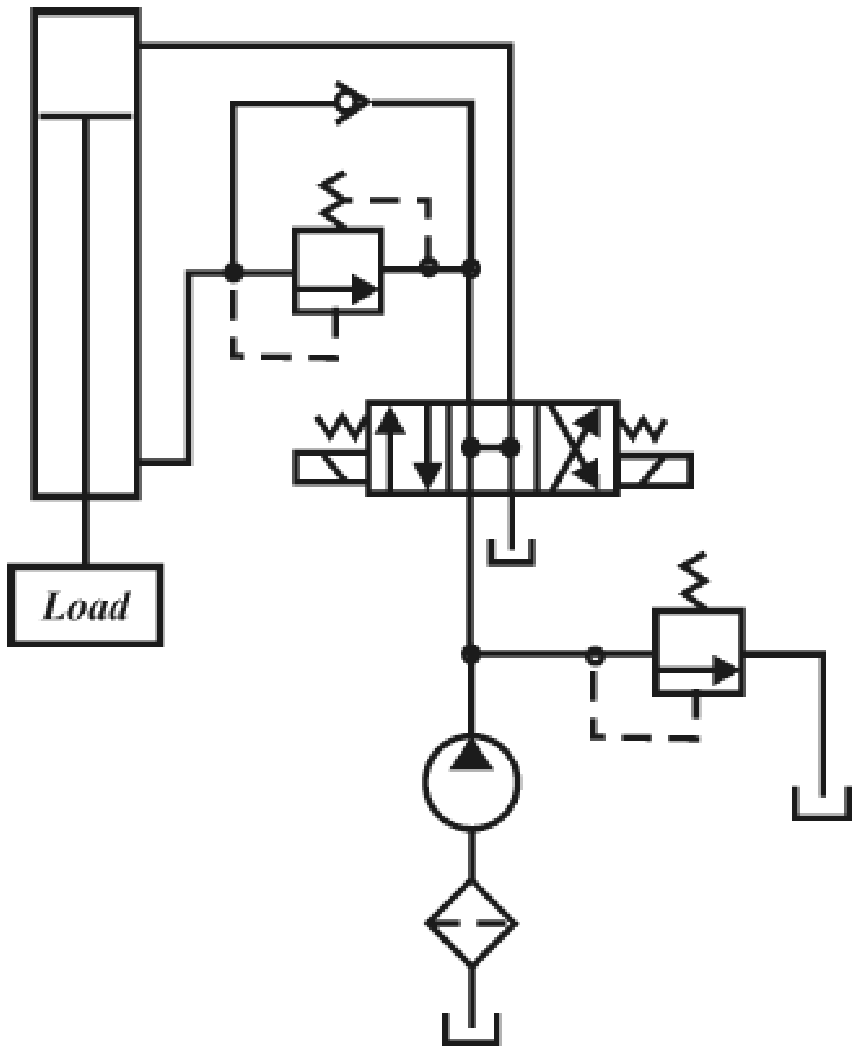


Figure 5.7: *Counterbalance hydraulic system*

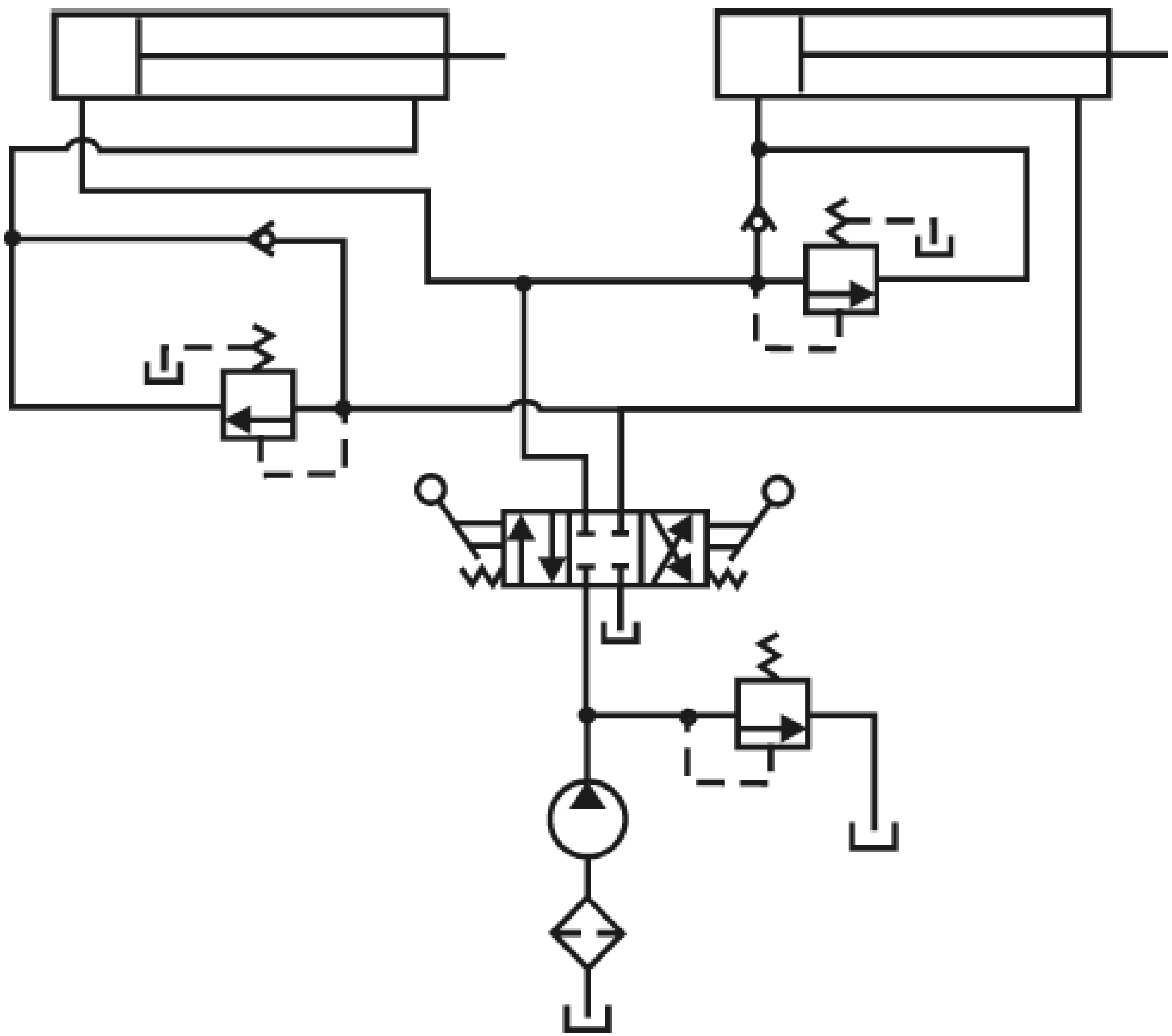


Figure 5.8: *Sequence cylinder hydraulic system*

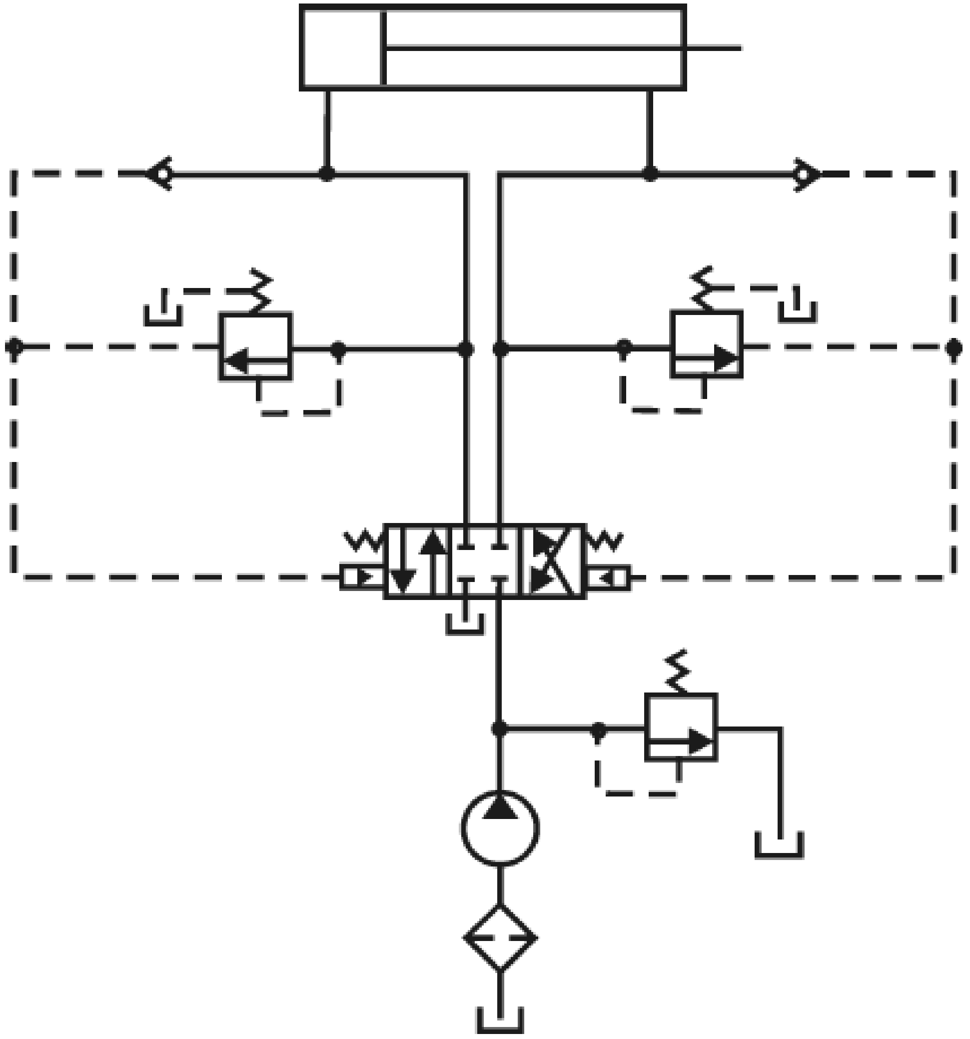


Figure 5.9: *Automatic reciprocating hydraulic system*

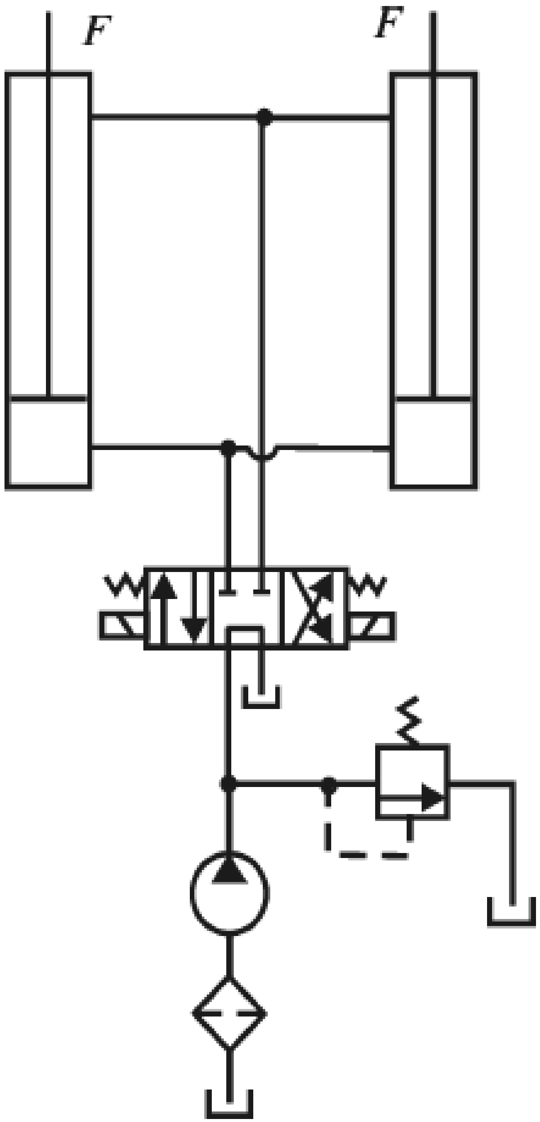


Figure 5.10: *Hydraulic cylinders connected in parallel*

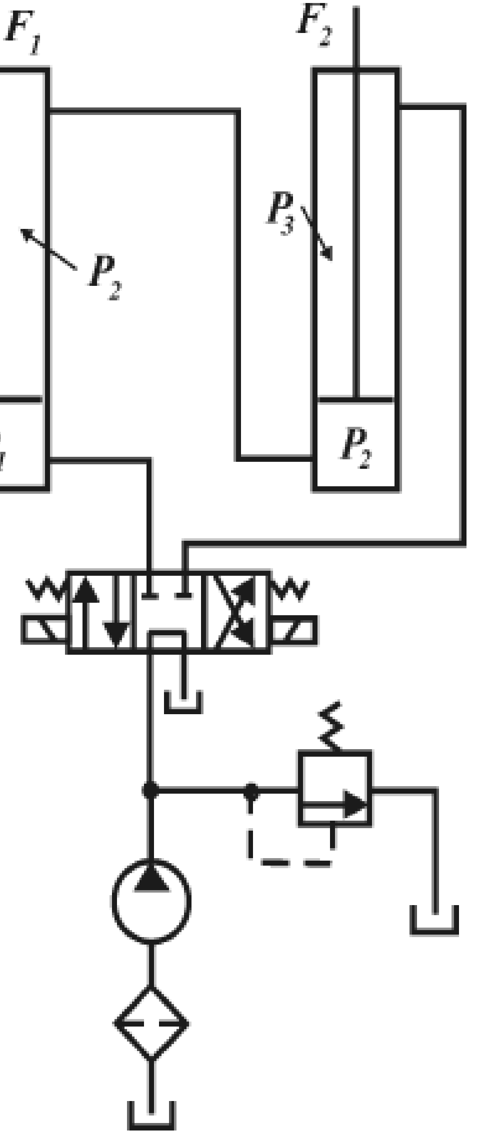


Figure 5.11: *Hydraulic cylinders connected in series*

$$(Q_{out})_1 = (Q_{in})_2$$

Since $Q = Av$,

$$(Av)_1 = (Av)_2$$

or,

$$((A_P)_1 - (A_R)_1)v_1 = (A_P)_2 v_2$$

Assuming that the fluid is incompressible, the synchronization means that $v_1 = v_2$, then

$$(A_P)_1 - (A_R)_1 = (A_P)_2$$

$$P_1(A_P)_1 - P_2((A_P)_1 - (A_R)_1) = F_1$$

Summing the forces on cylinder 2 results

$$P_2(A_P)_2 - P_3((A_P)_2 - (A_R)_2) = F_2$$

But $P_3 = 0$ and $(A_P)_2 = (A_P)_1 - (A_R)_1$, then

$$P_2((A_P)_1 - (A_R)_1) = F_2$$

Summing equations (5.1) and (5.3) gives

$$P_1(A_P)_1 = F_1 + F_2$$

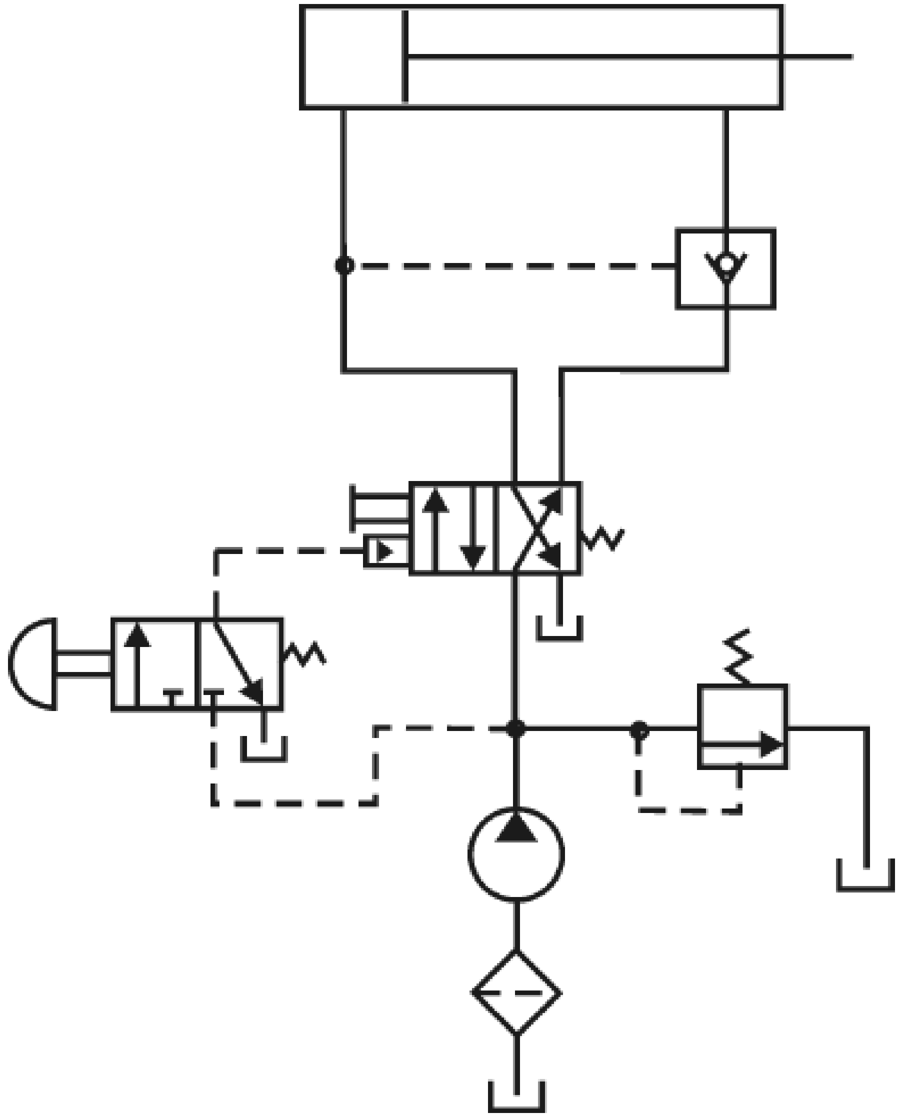


Figure 5.12: *Hydraulic fail-safe circuit*

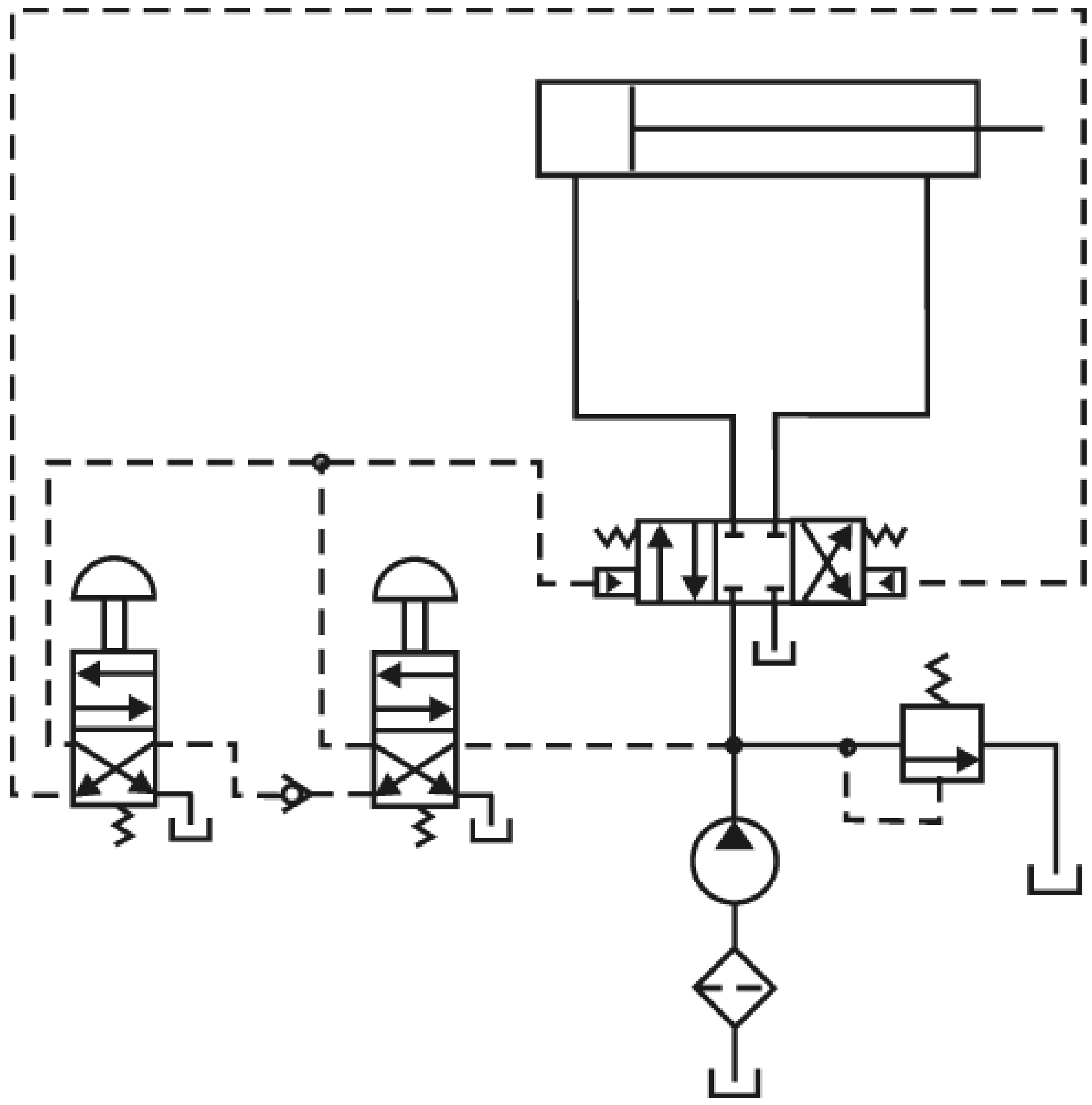


Figure 5.13: *Two-handed safety circuit*

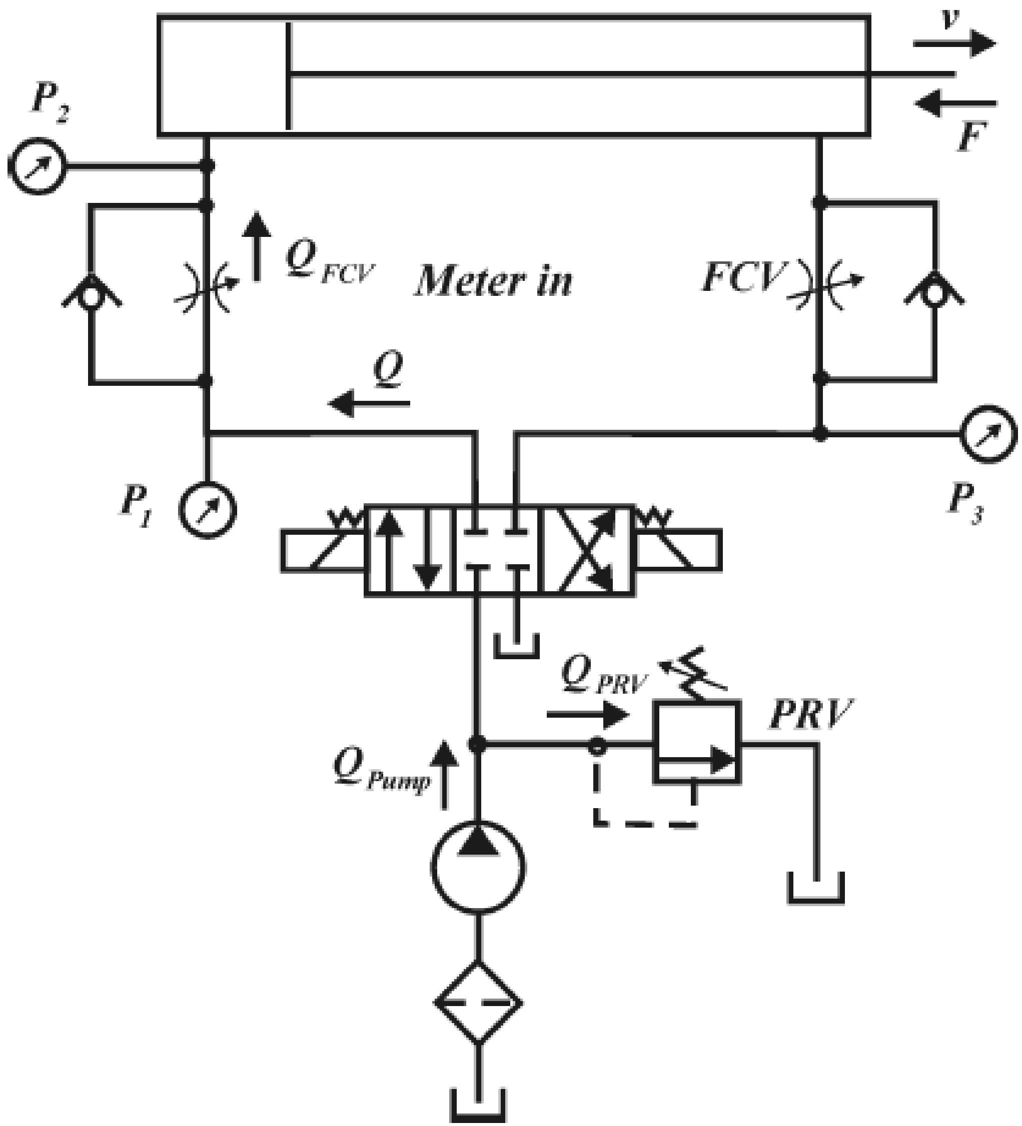


Figure 5.14: *Meter-in speed control circuit*

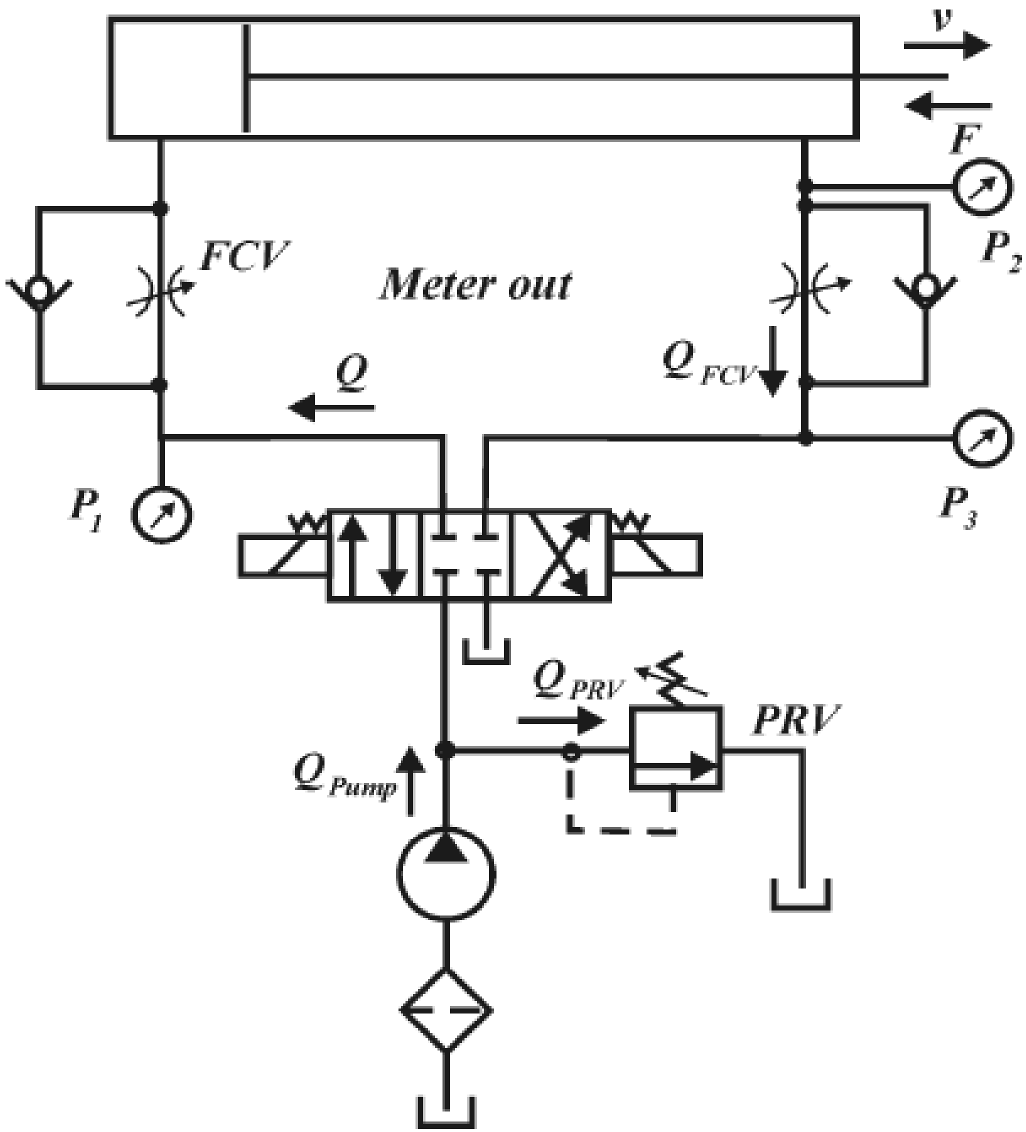


Figure 5.15: *Meter-out speed control circuit*

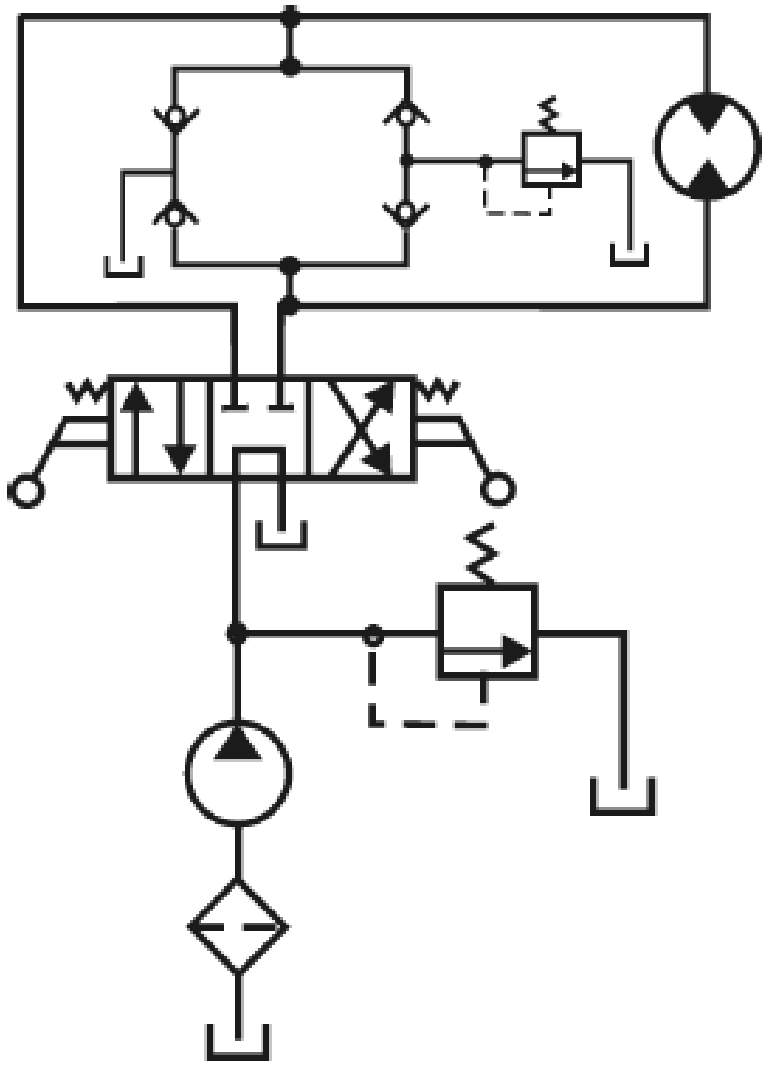


Figure 5.17: *Braking system for a hydraulic motor*

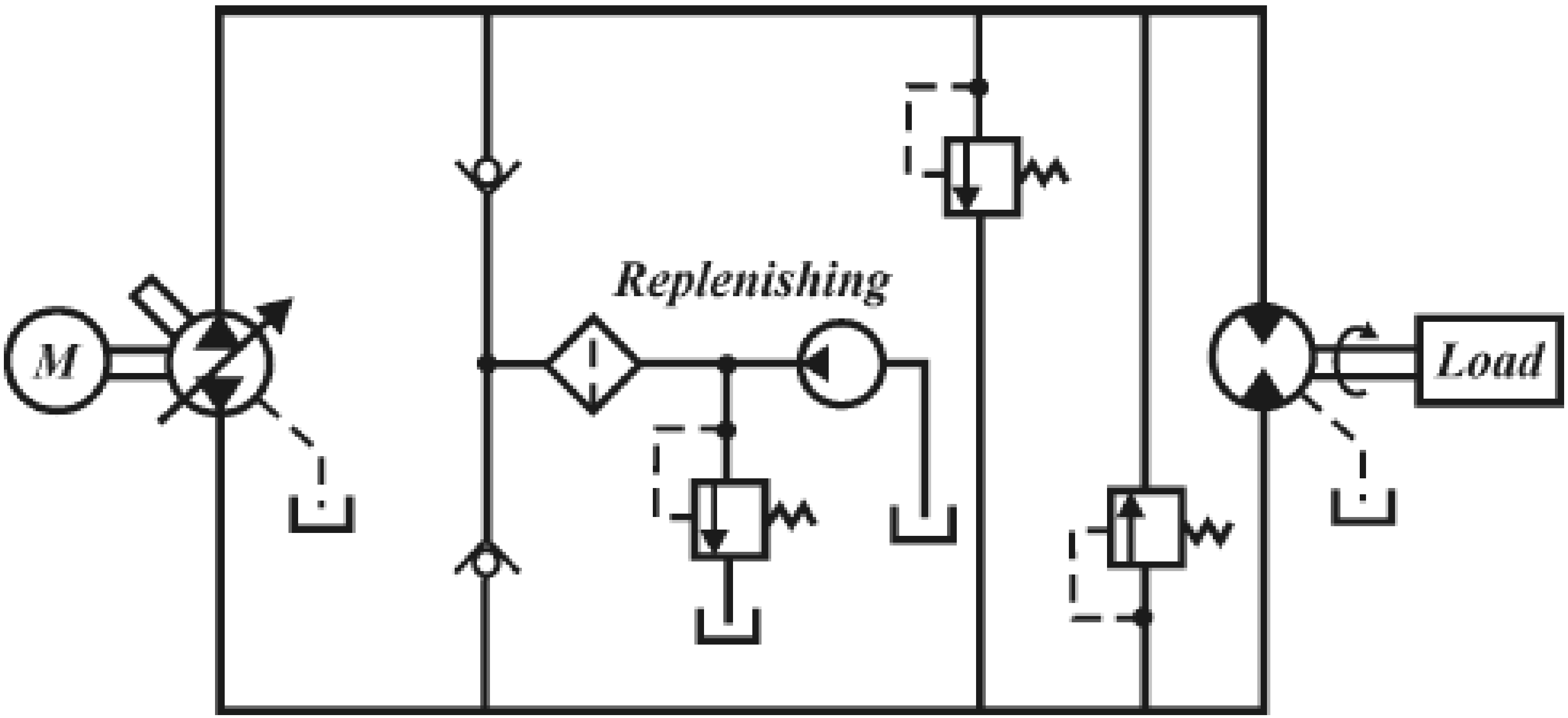


Figure 5.18: *Hydrostatic transmission system*

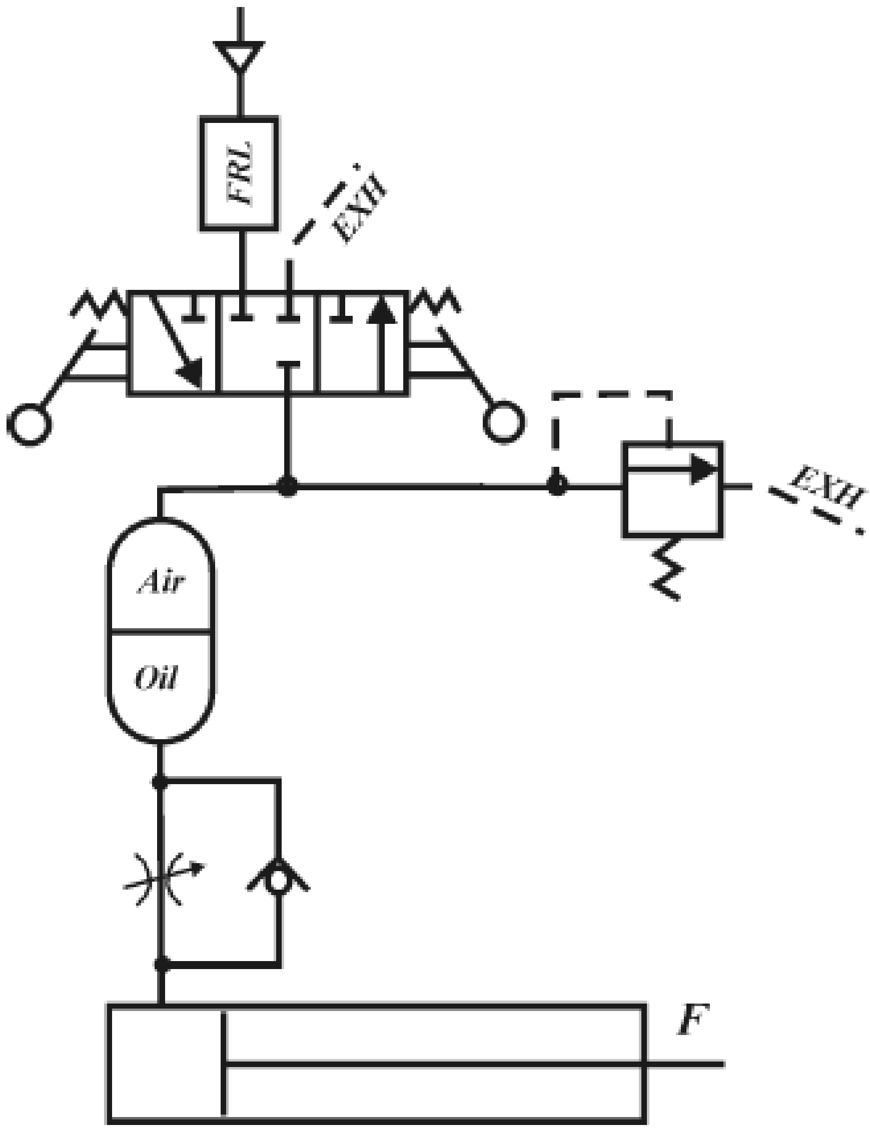


Figure 5.19: *Air over oil system*

Pneumatic circuits

Pneumatic circuit components

- Air compressor (piston, screw or rotary type).
- Air reservoir (tank to be filled with compressed air).
- Filter-Regulator-Lubricator (FRL).
- Pneumatic actuator (linear piston or rotary motor).
- Pneumatic valve (pressure relief, flow control or directional control).

Pneumatic circuit design

- Saftey of operator and operation.
- Efficiency and performance of the whole system.
- Cost requirements.
- Simplicity and easiness.

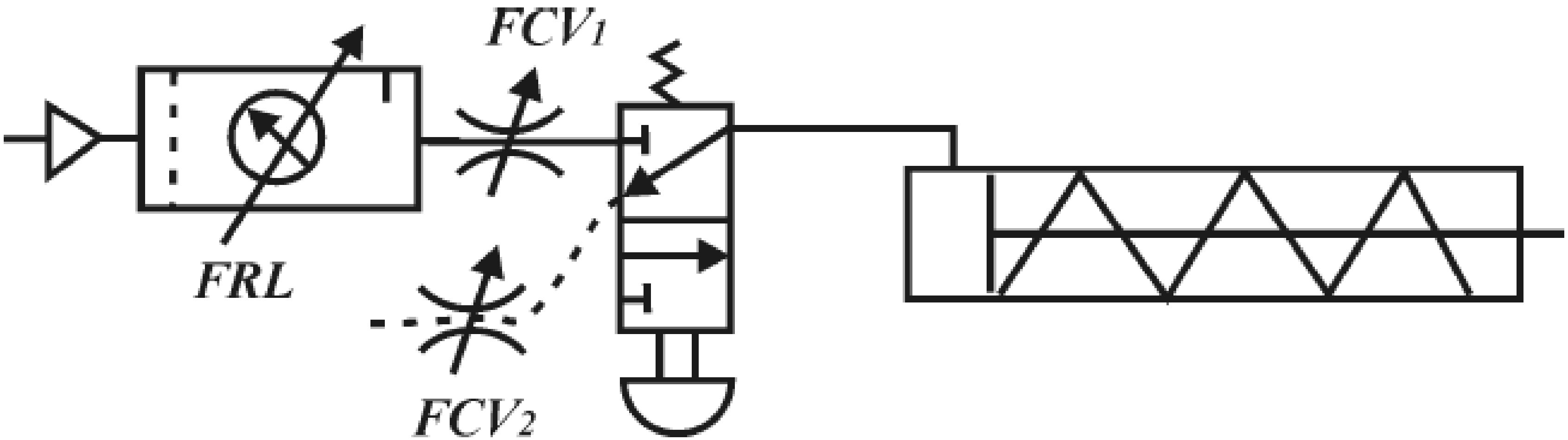


Figure 6.1: *Pneumatic single-acting cylinder*

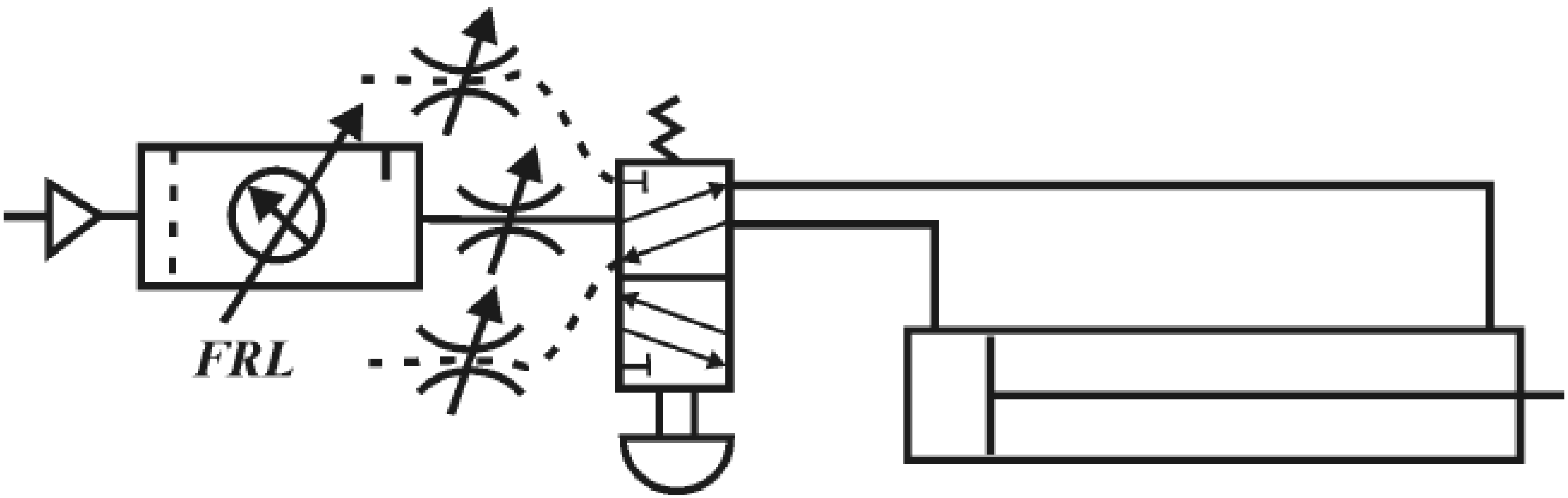


Figure 6.2: *Pneumatic double-acting cylinder*

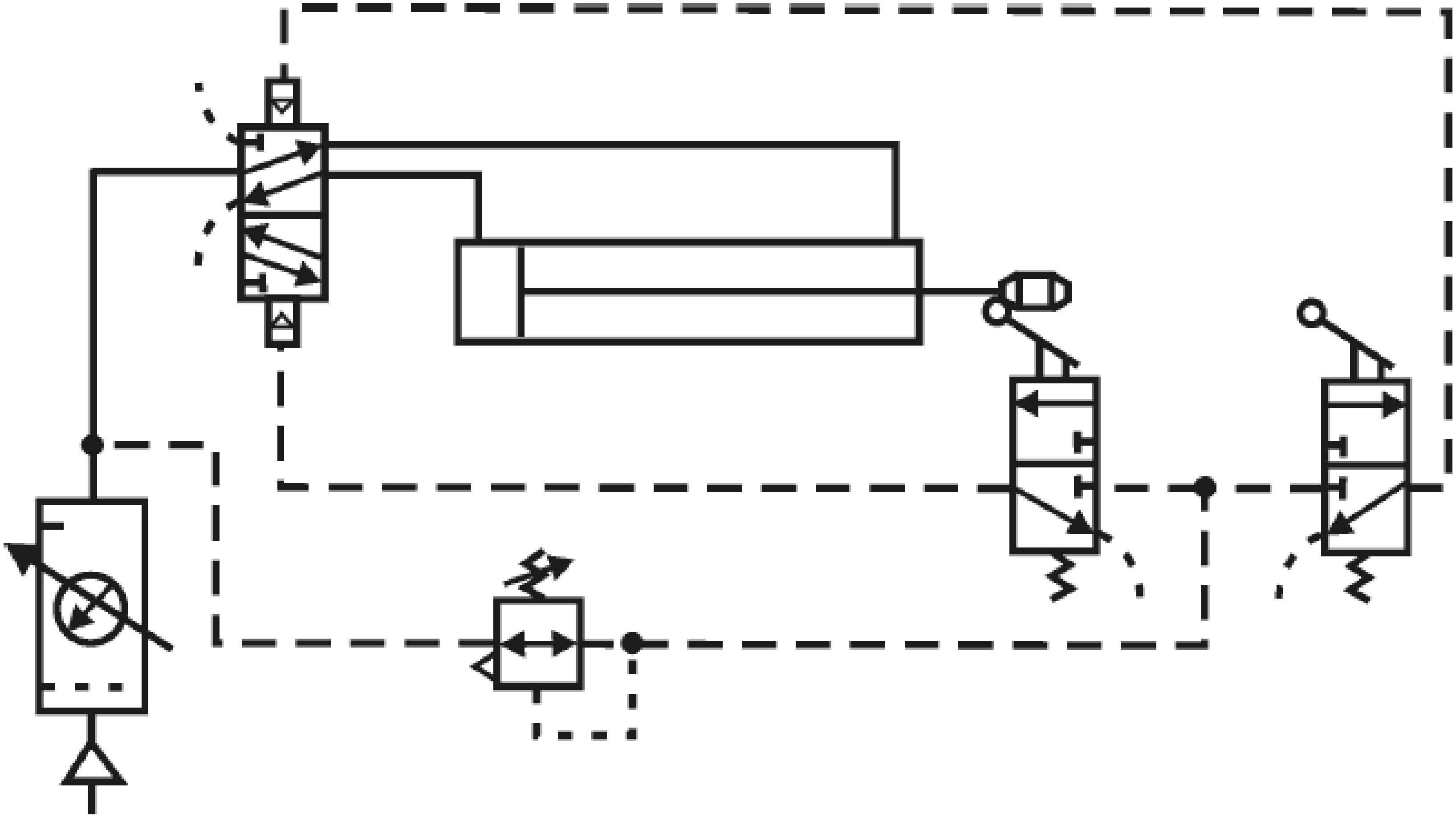


Figure 6.3: *Air pilot control of a pneumatic double-acting cylinder*

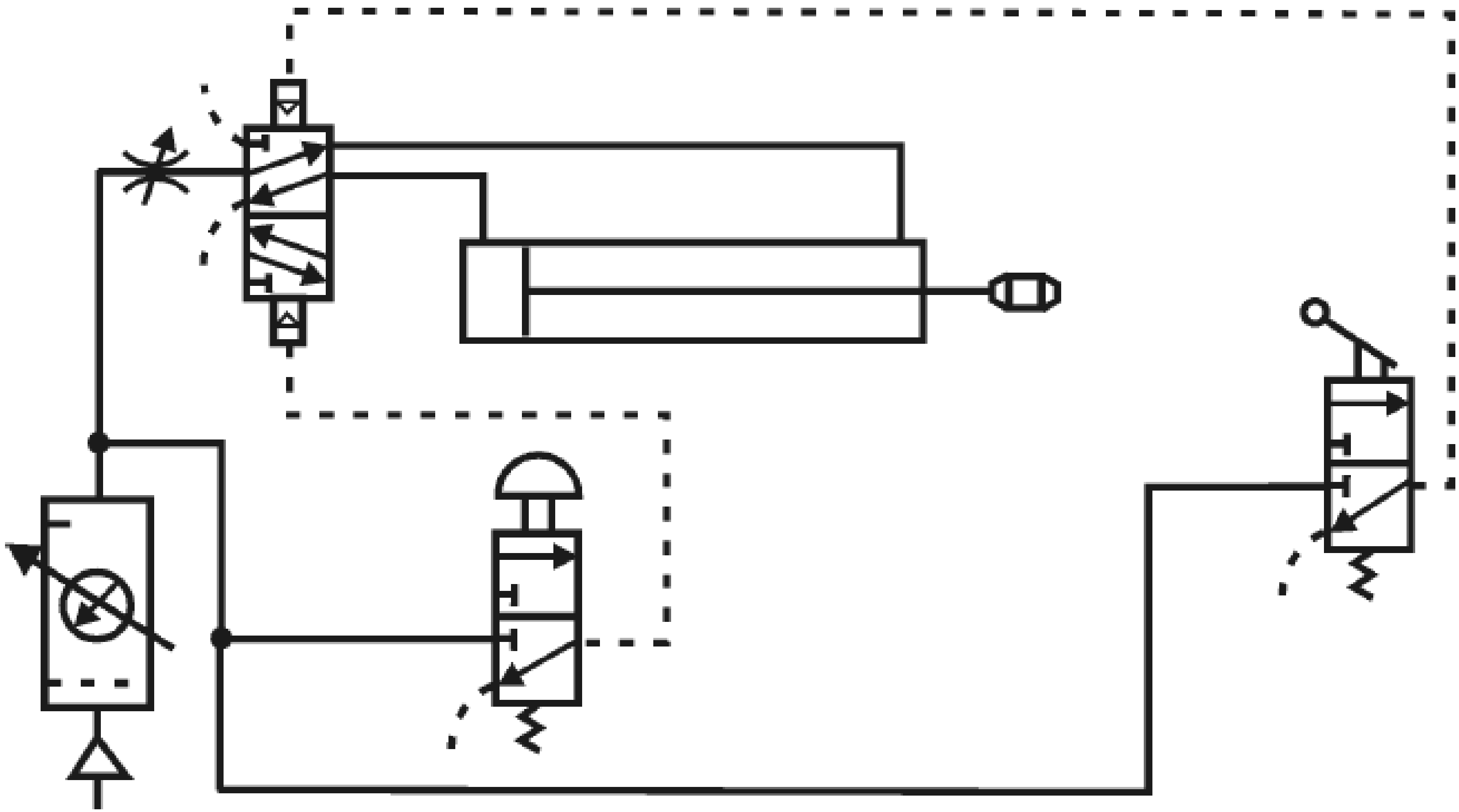


Figure 6.4: *Cycle timing of pneumatic cylinder*

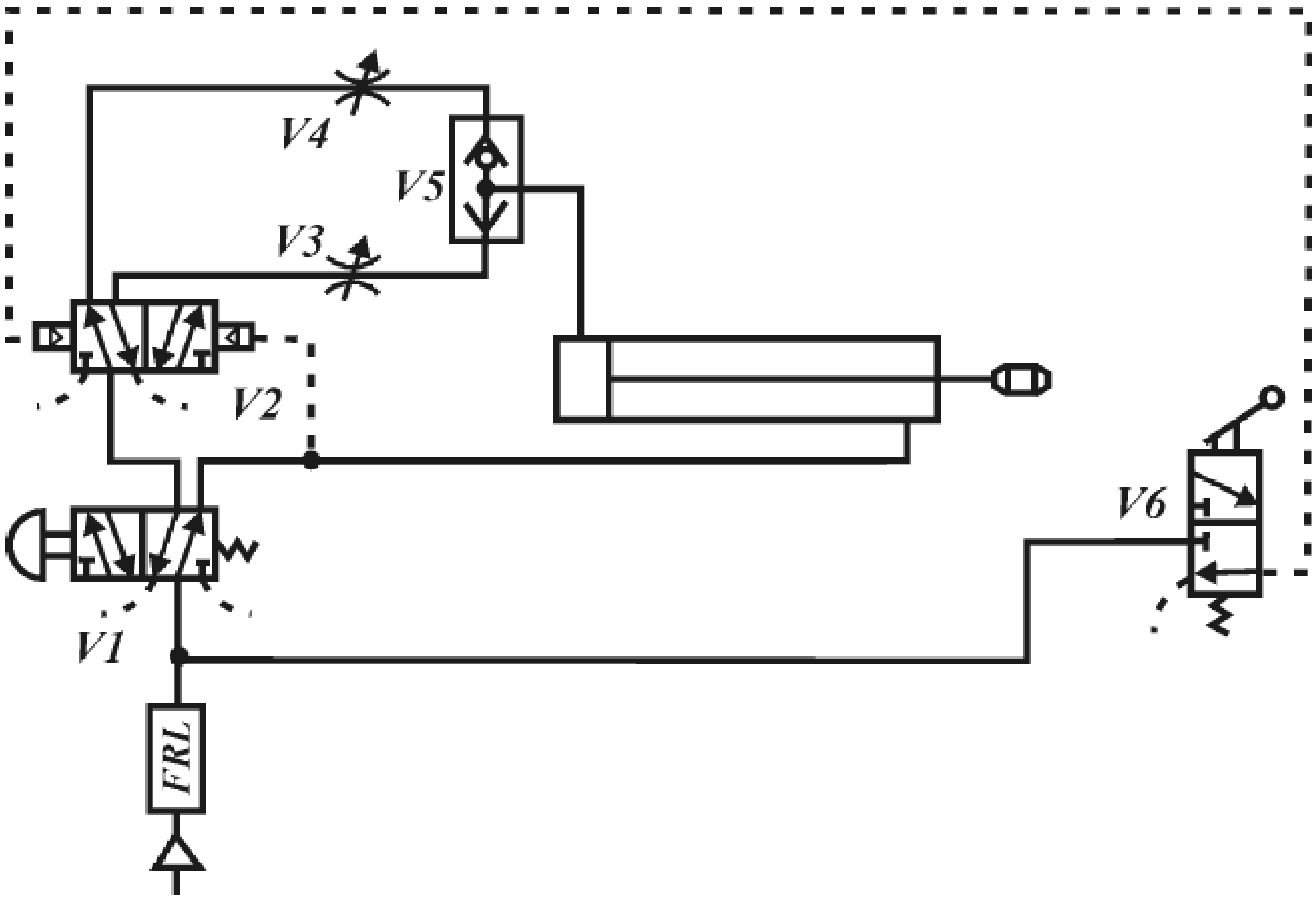


Figure 6.5: *Two-speed pneumatic cylinder*

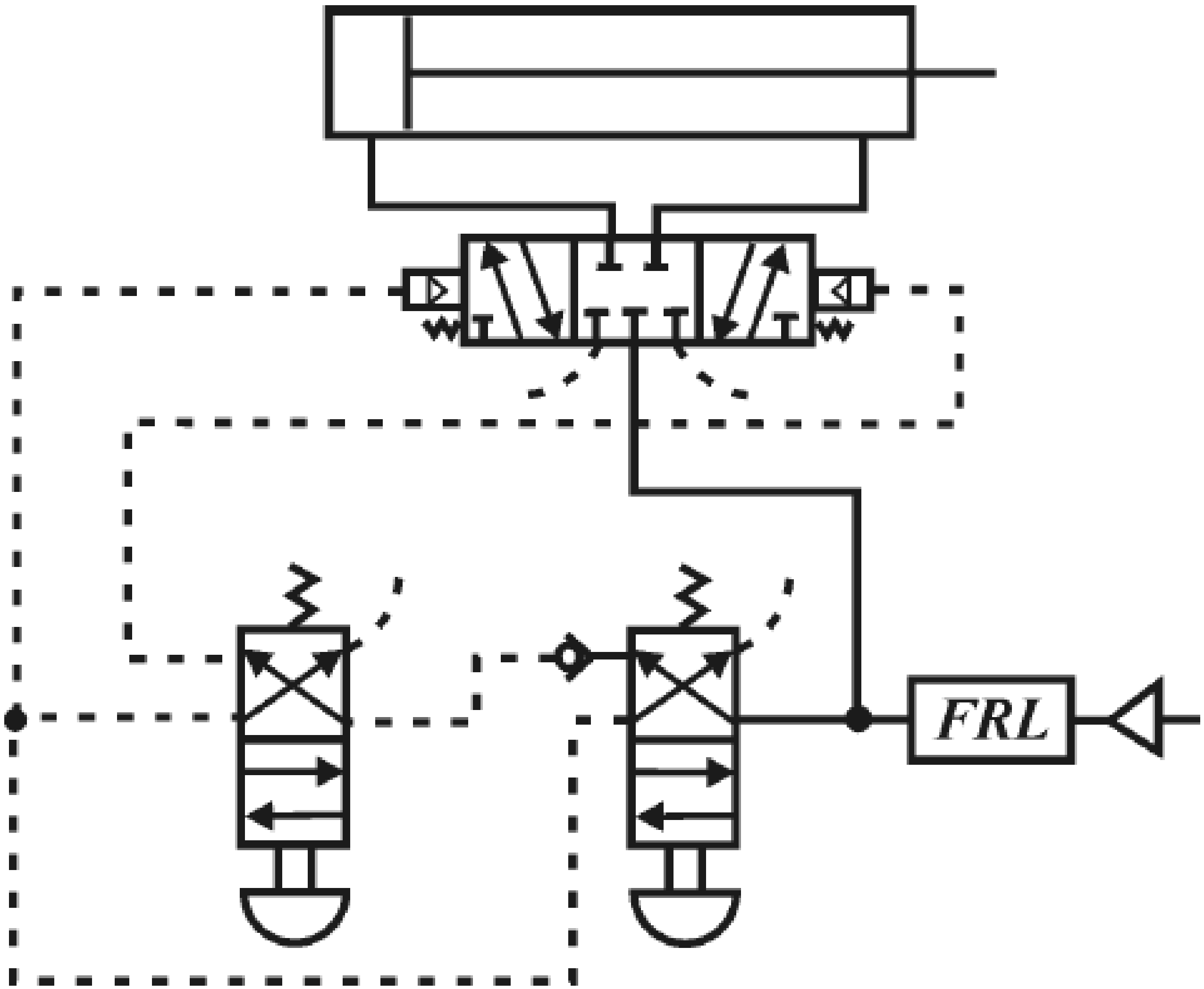


Figure 6.6: *Two-handed safety circuit for pneumatic cylinder*

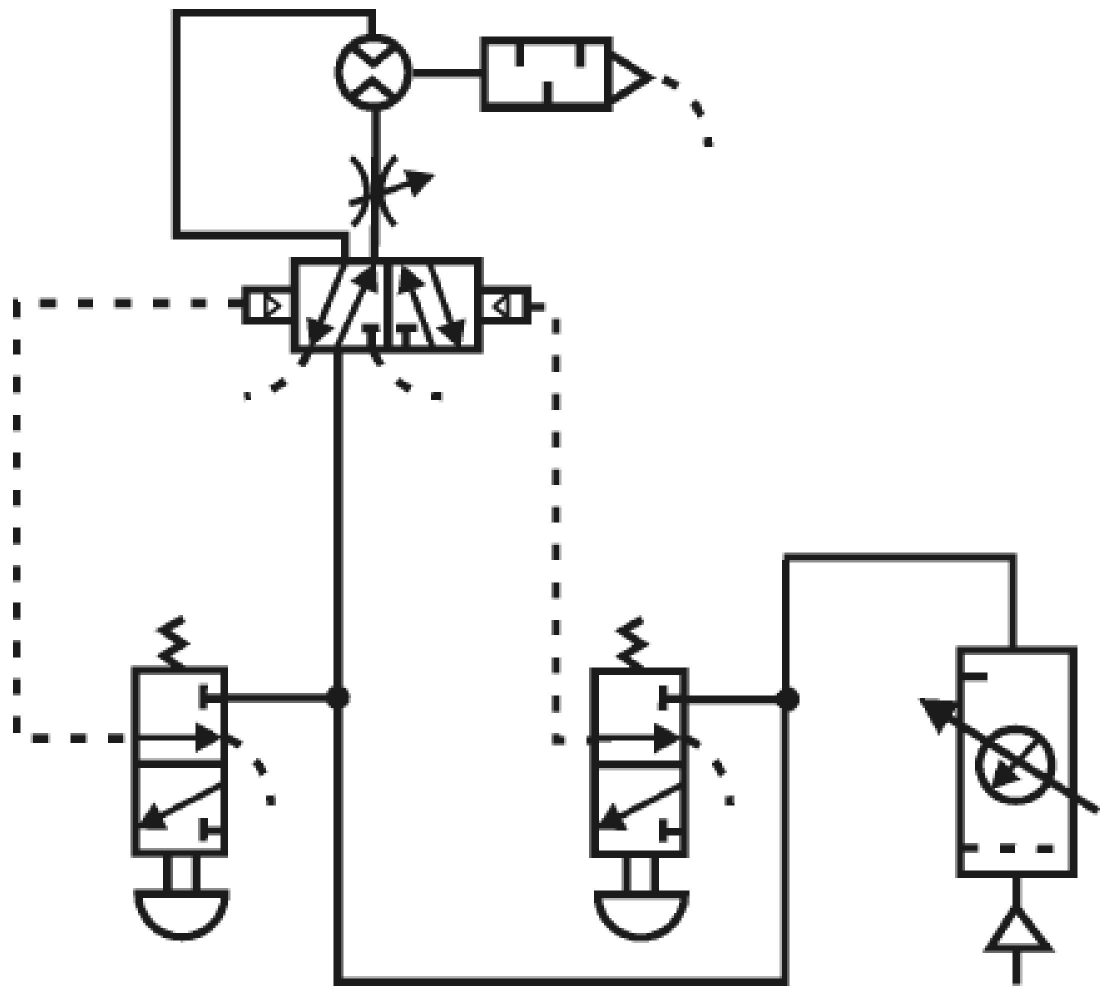


Figure 6.7: *Control of air motor*

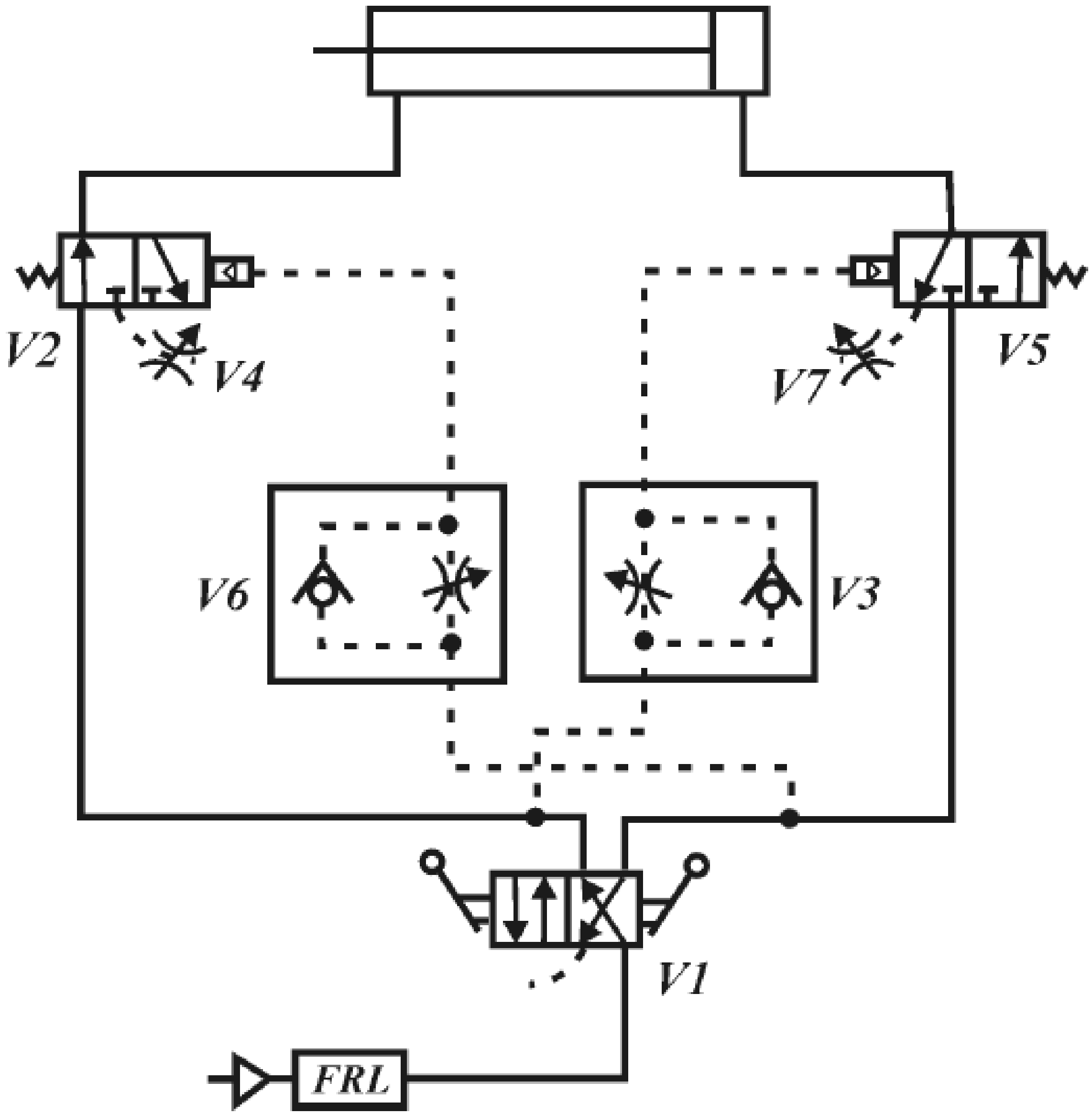


Figure 6.8: *Deceleration of a pneumatic cylinder*

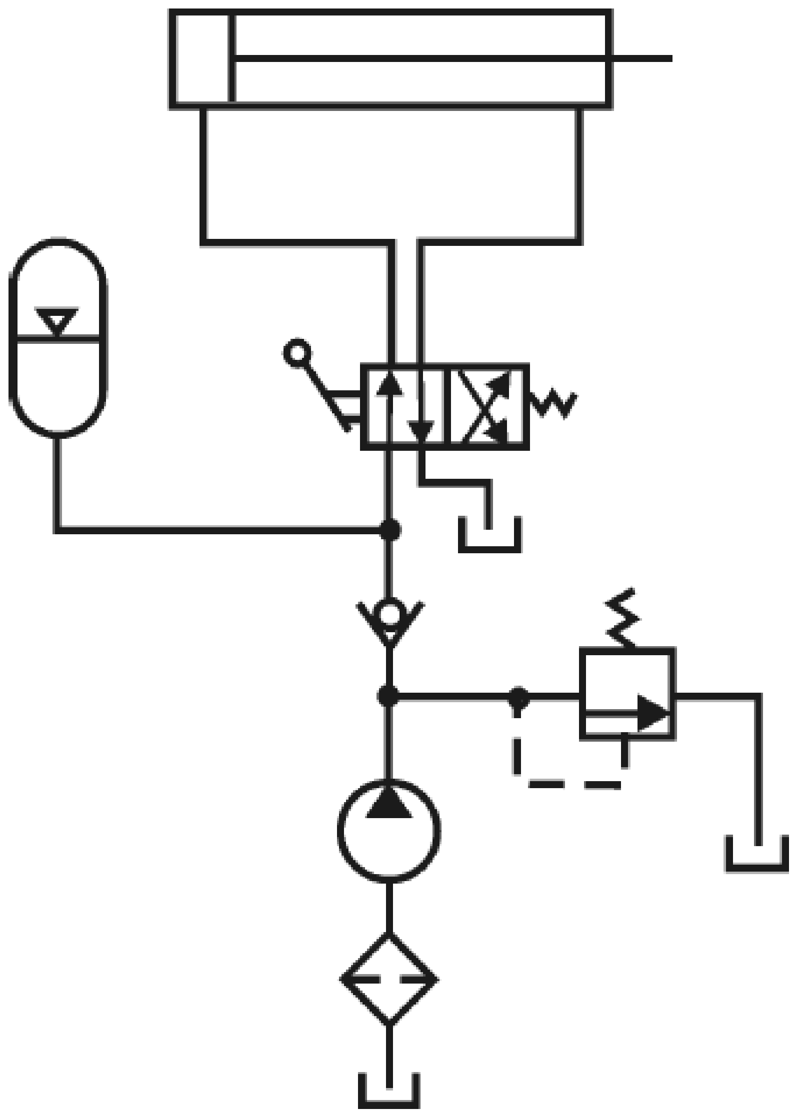


Figure 6.9: *Gas loaded accumulator*

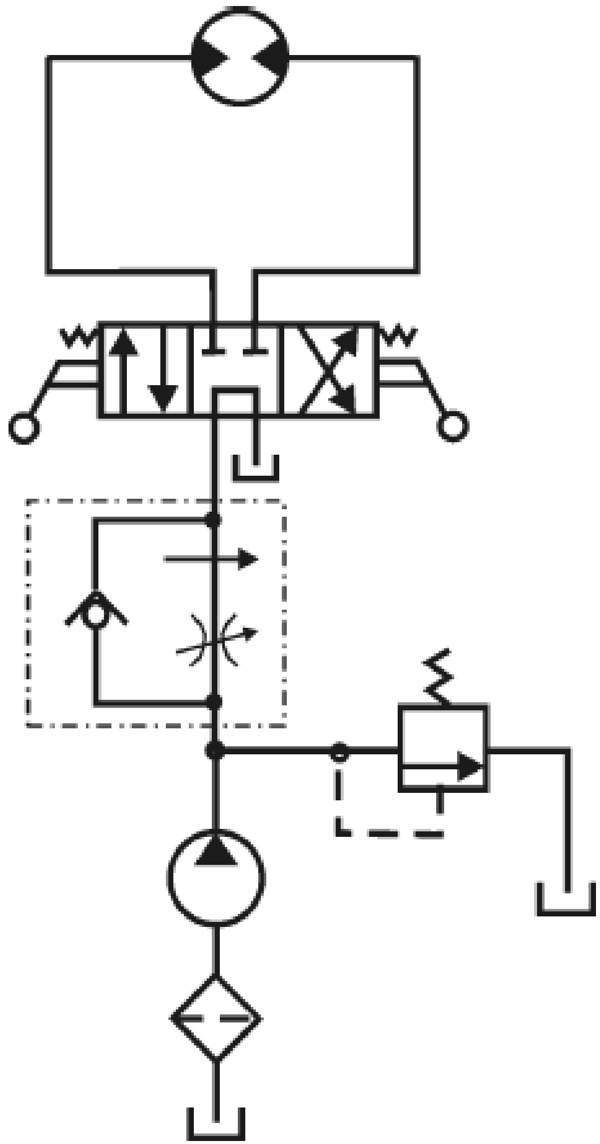


Figure 5.16: *Pressure compensated speed control for a hydraulic motor*