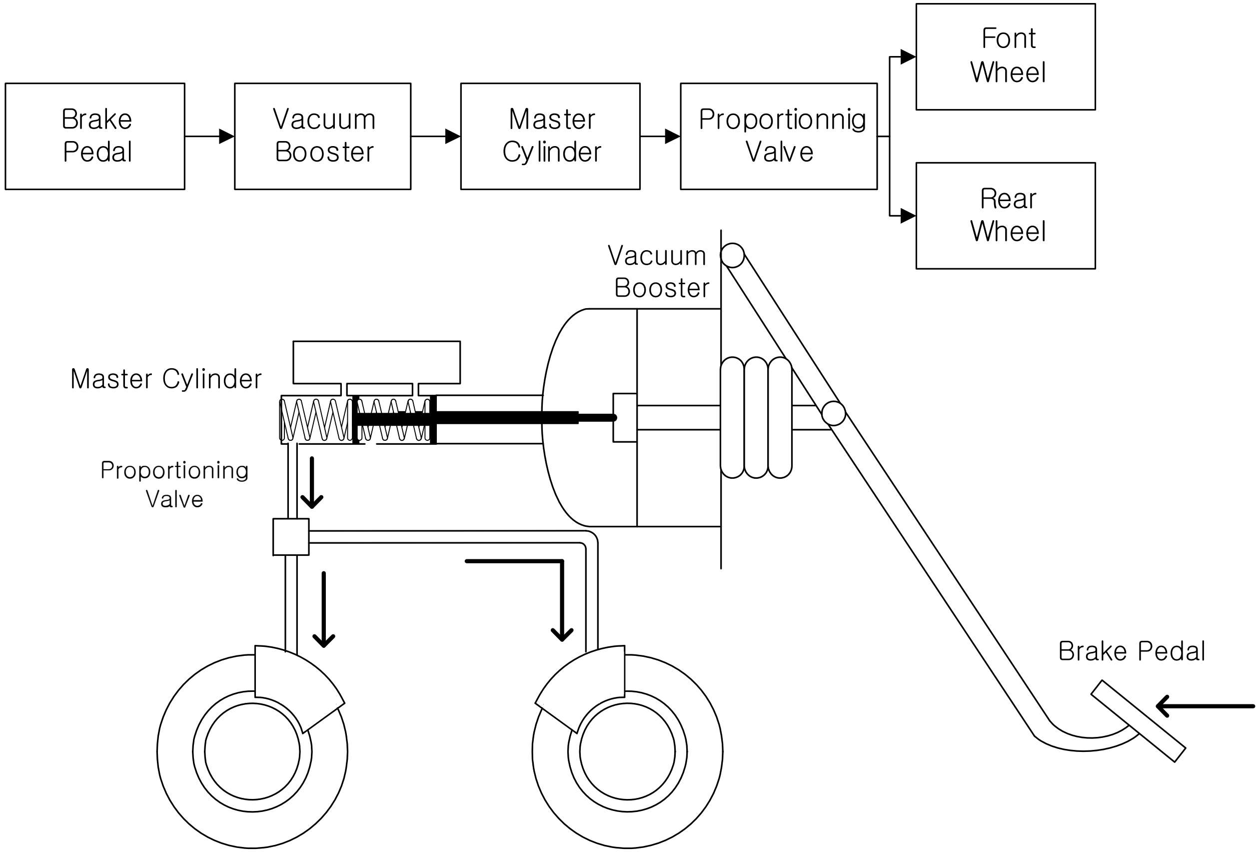
Lecture Note 8-1
Hydraulic Systems

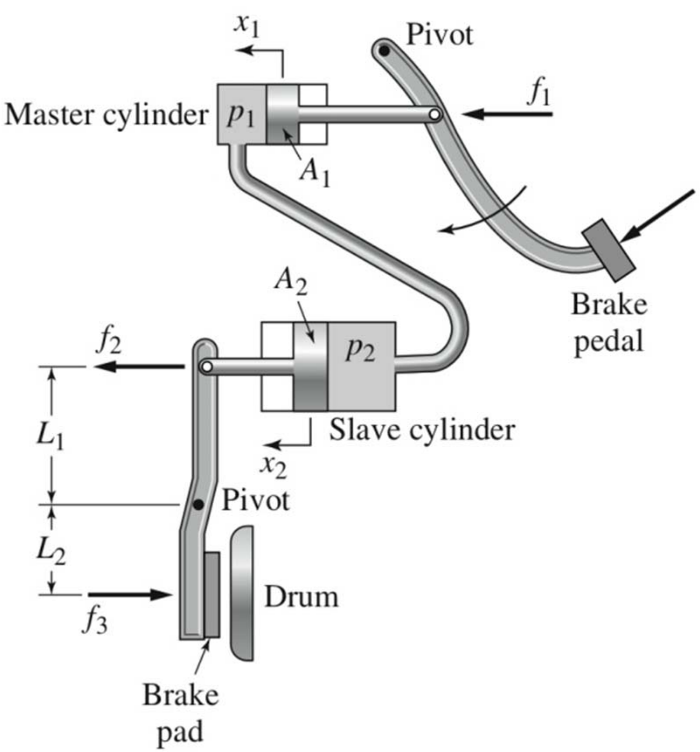
Brake Model



Fundamental structure of a hydraulic brake

Conservation of Mass, Force and Pressure

$$\dot{m} = q_{mi} - q_{mo}$$



$$i)\ Volume \quad A_1 dx_1 = A_2 dx_2$$

$$\rightarrow dx_2 = \frac{A_1}{A_2} dx_1$$

$$ii)\ \text{Pr}essue \quad p_2 = p_1 + \rho g h$$

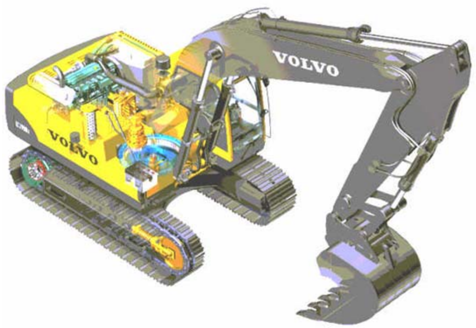
$$iii)\ \textbf{Force} \quad f_1 = p_1 A_1$$

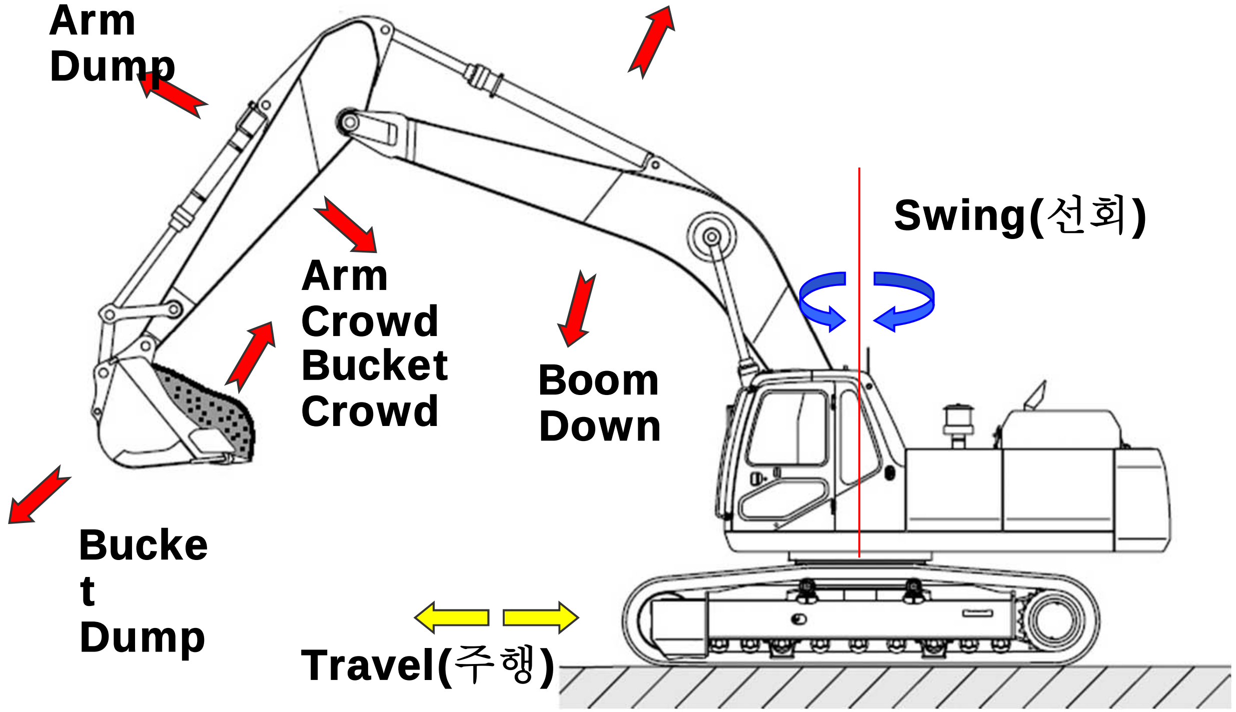
$$f_2 = p_2 A_2$$

$$f_2 = \frac{A_2}{A_1} f_1 \qquad (P_1 \approx P_2)$$

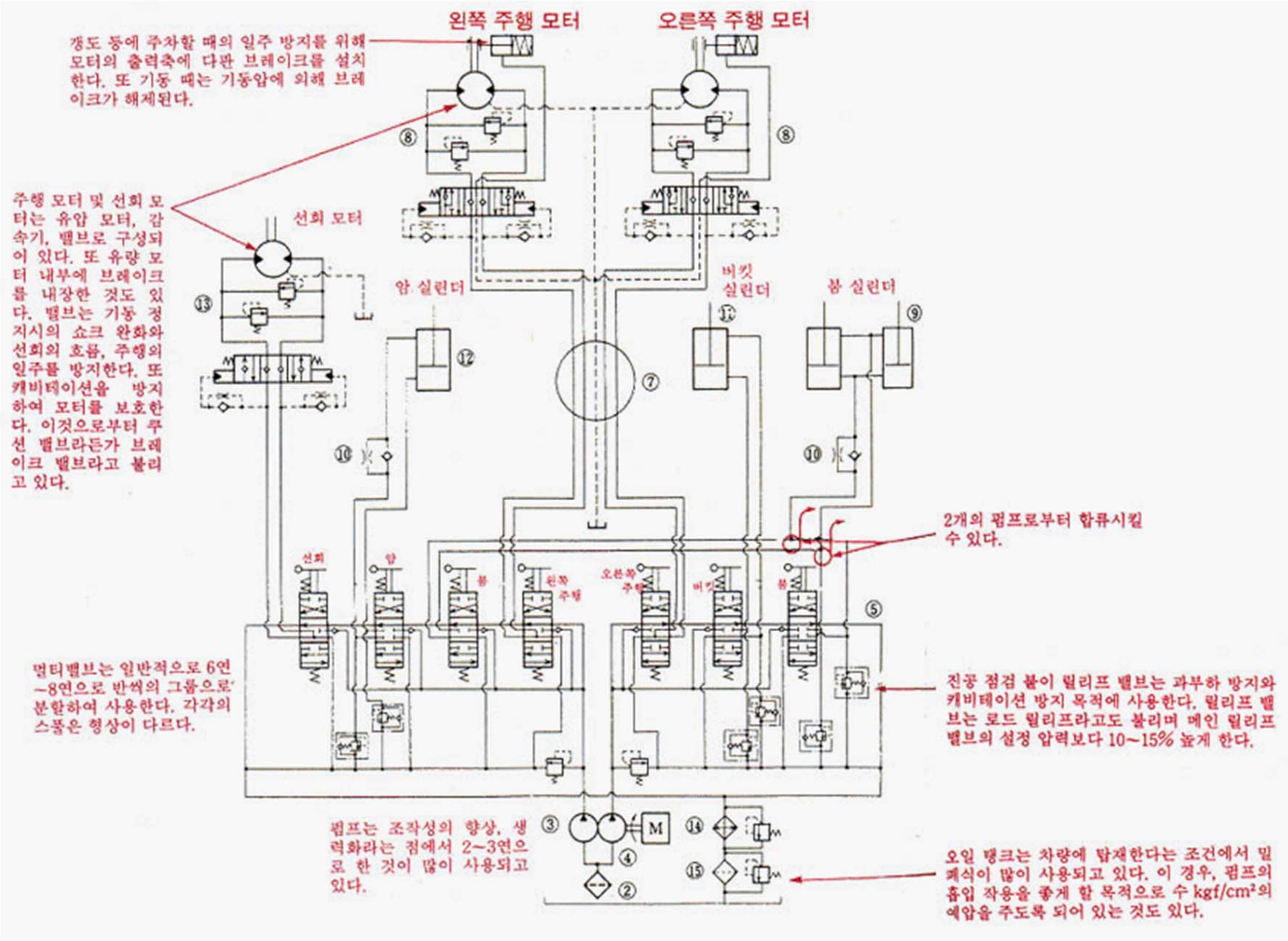
$$f_3 = \frac{L_1}{L_2} f_2 = \frac{L_1}{L_2} \frac{A_2}{A_1} f_1$$

Hydraulic Excavator



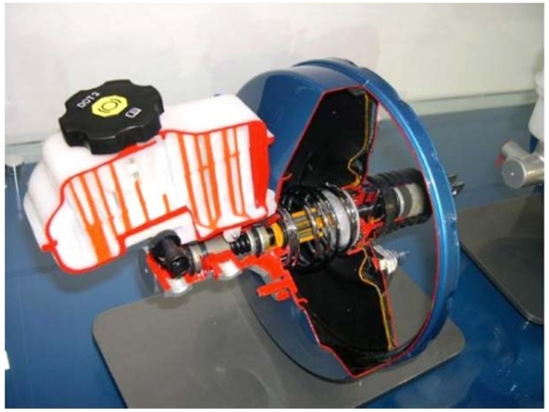


유압굴삭기 회로도

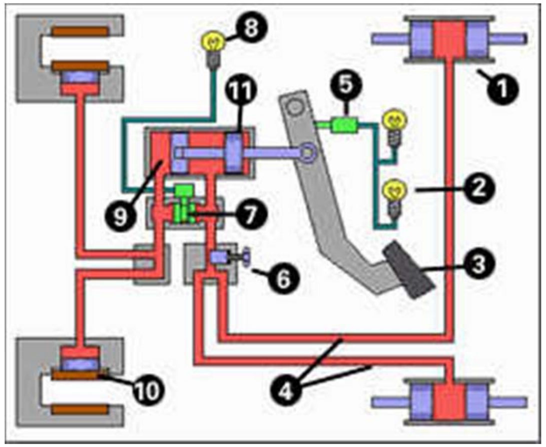


왼쪽 주행 모터
오른쪽 주행 모터
경도 등에 주차할 때의 일주 방지를 위해 모터의 출력축에 다판 브레이크를 설치 한다. 또 기동 때는 기동압에 의해 브레이크가 해제된다.
주행 모터 및 선회 모터는 유압 모터, 감속기, 밸브로 구성되어 있다. 또 유량 모터 내부에 브레이크를 내장한 것도 있다. 밸브는 기동 정지시의 쇼크 완화와 선회의 흐름, 주행의 일주를 방지한다. 또 캐비테이션을 방지하여 모터를 보호한다. 이것으로부터 쿠션 밸브라든가 브레이크 밸브라고 불리고 있다.
선회 모터
암 실린더
버킷 실린더
붐 실린더
2개의 펌프로부터 합류시킬 수 있다.
선회
암
붐
왼쪽 주행
오른쪽 주행
버킷
멀티밸브는 일반적으로 6연~8연으로 반씩의 그룹으로 분할하여 사용한다. 각각의 스풀은 형상이 다르다.
진공 점검 붙이 릴리프 밸브는 과부하 방지와 캐비테이션 방지 목적에 사용한다. 릴리프 밸브는 로드 릴리프라고도 불리며 메인 릴리프 밸브의 설정 압력보다 10~15% 높게 한다.
펌프는 조작성의 향상, 생력화라는 점에서 2~3연으로 한 것이 많이 사용되고 있다.
M
오일 탱크는 차량에 탑재한다는 조건에서 밀폐식이 많이 사용되고 있다. 이 경우, 펌프의 흡입 작용을 좋게 할 목적으로 수 kgf/cm²의 예압을 주도록 되어 있는 것도 있다.

Hydraulic Brake Systems



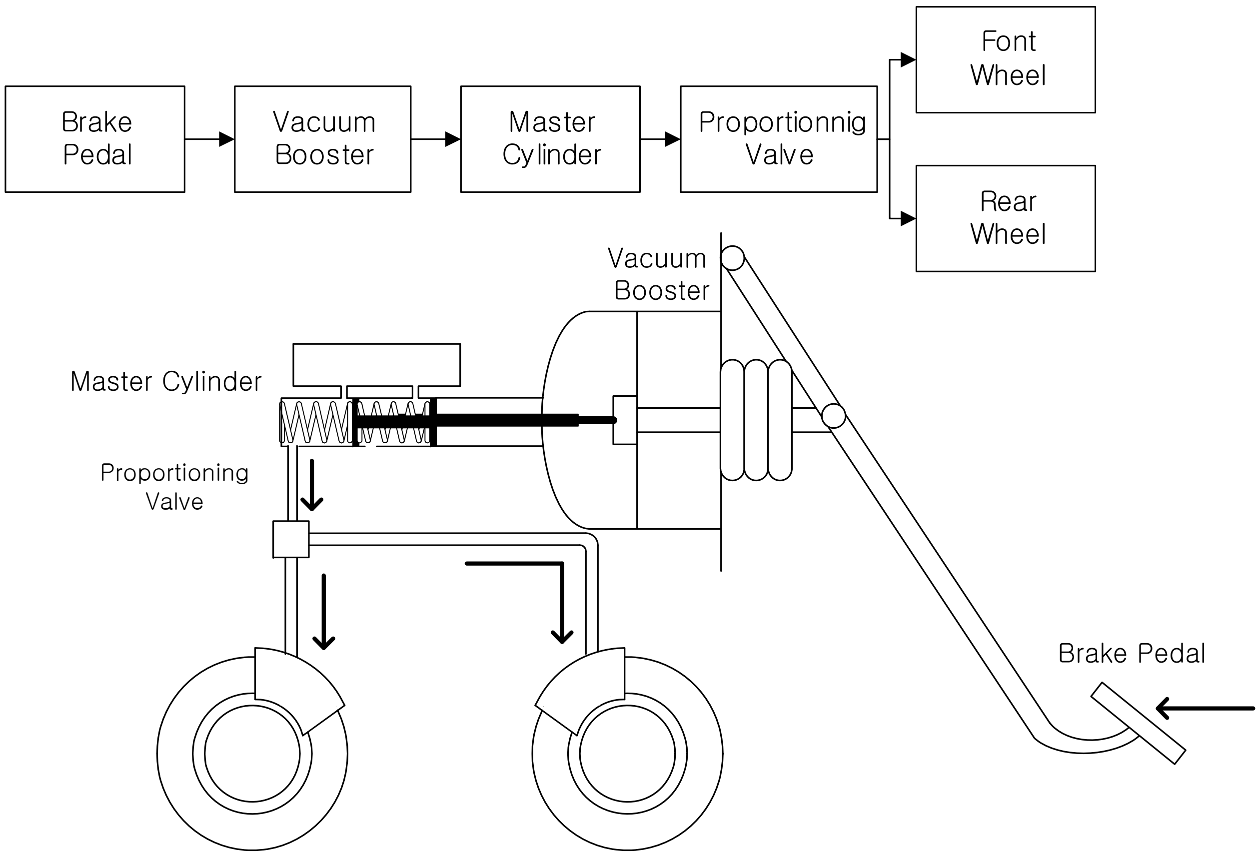
Brake System



1) Wheel Cylinder
2) Brake Light
3) Brake Pedal
4) Rear Brake Lines
5) Stop Light Switch (Mechanical)
6) Front/Rear Balance Valve
7) Pressure Differentiavl Valve
8) Brake Warning Lamp
9) Brake Fluid
10) Brake Pad
11) Master Cylinder

Vehicle Model - Brake Model

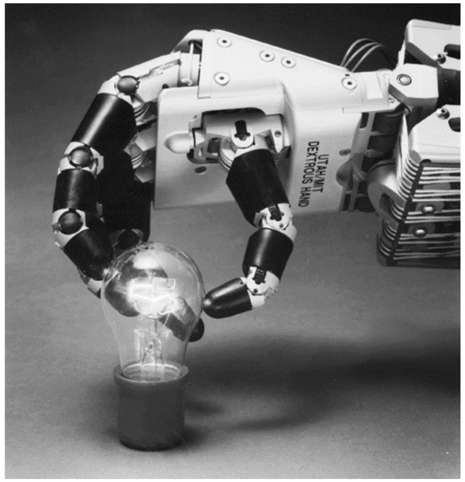
Brake Model



Fundamental structure of a hydraulic brake

Applications of Fluid Power

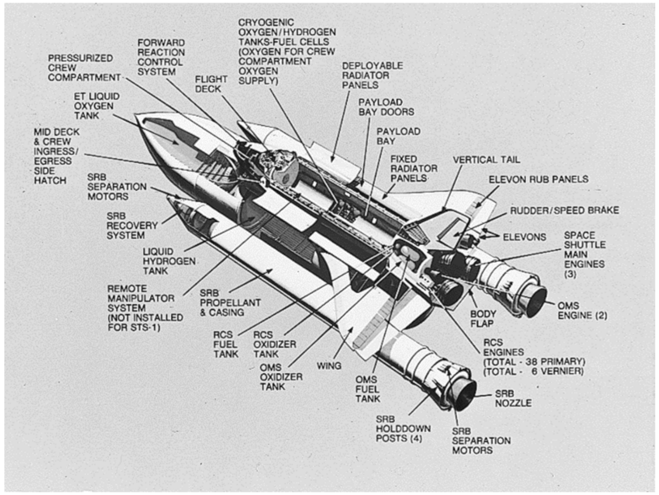
- Pneumatically controlled dexterous hand



- Hydraulically powered dexterous

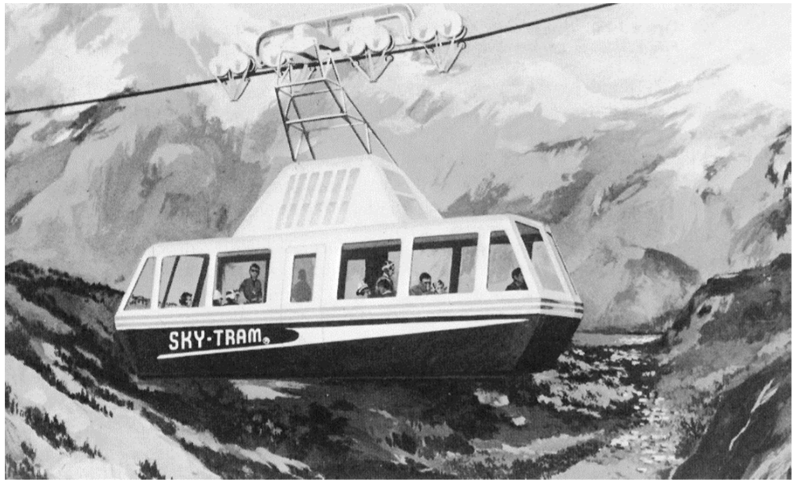
- Space shuttle Columbia

- Space shuttle vehicle

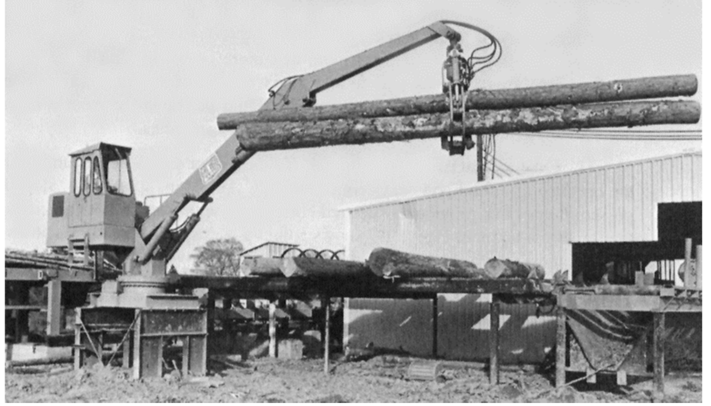


Applications of Fluid Power

- Hydraulically powered Sky-tram

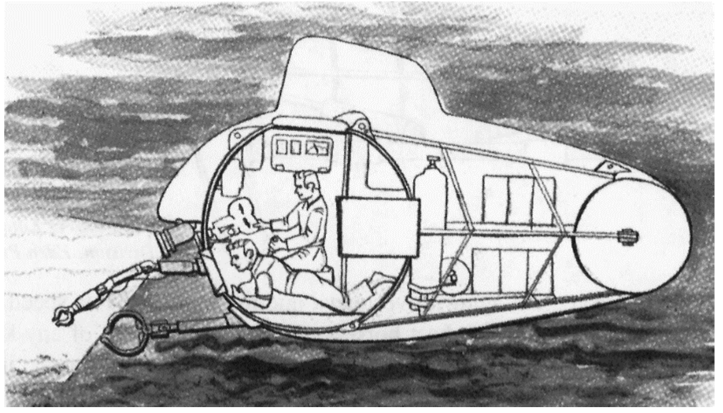


- Hydraulically driven turntable

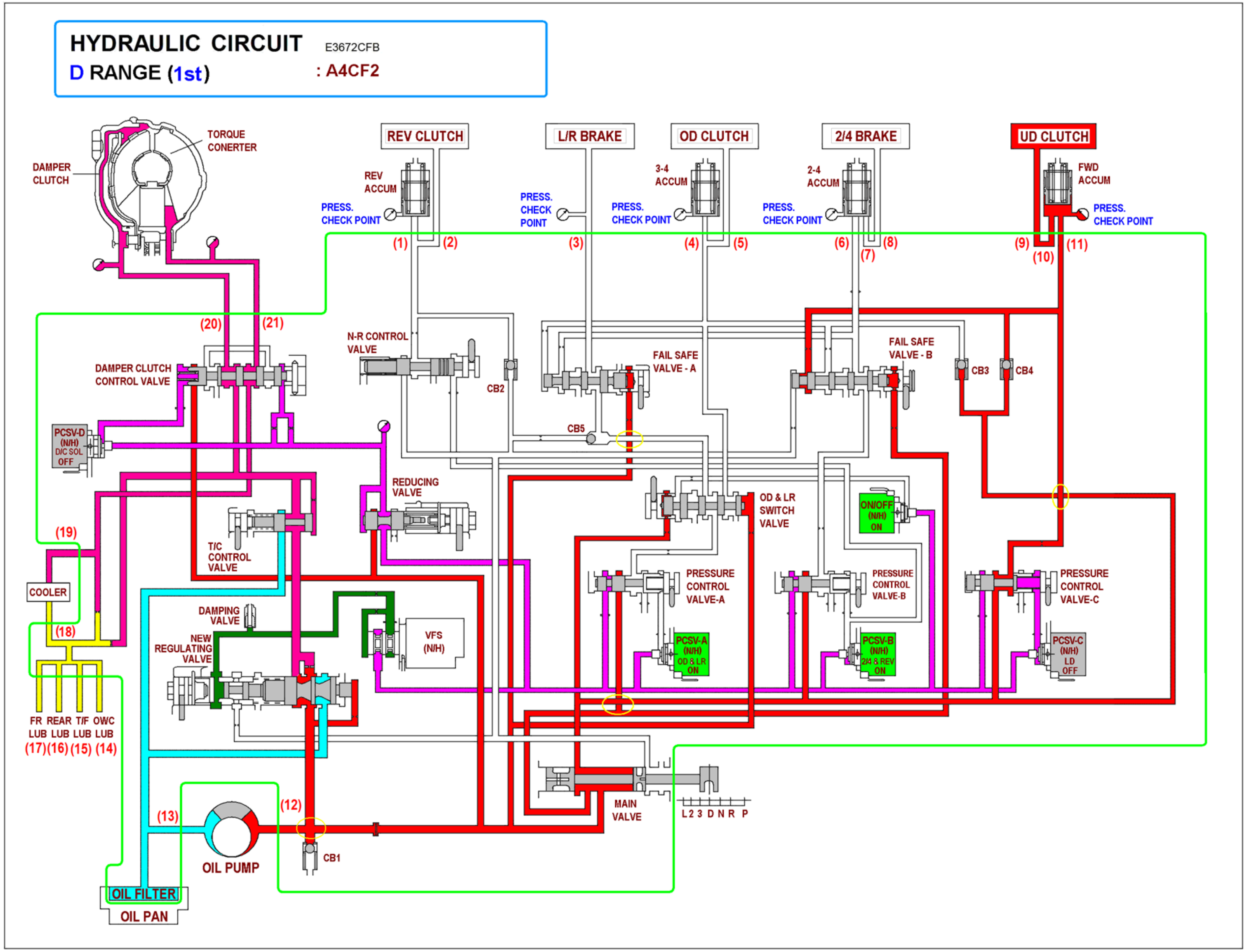


- Hydraulic power brush drive

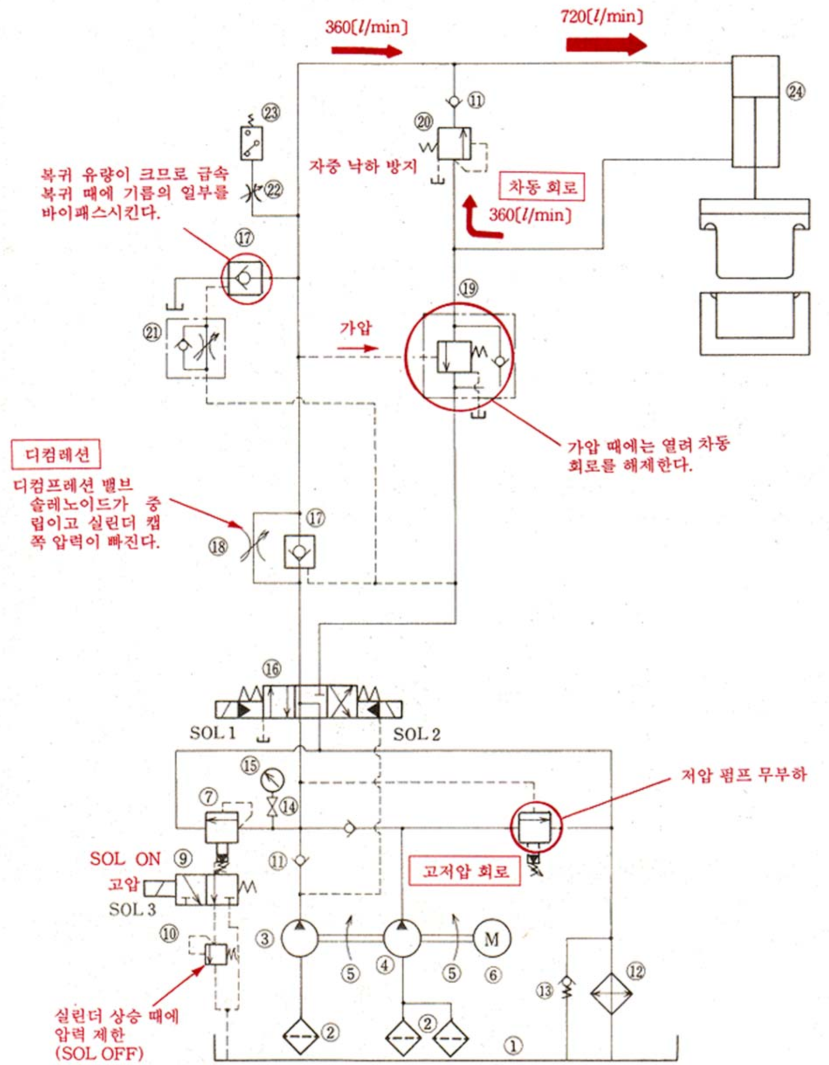
- Oceanography



Automatic Transmission



Press

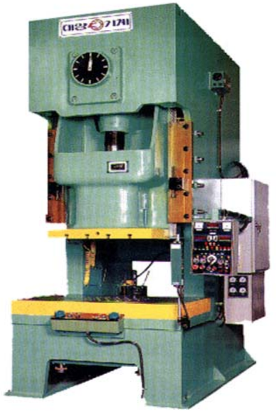


360[l/min]
720[l/min]
복귀 유량이 크므로 급속
복귀 때에 기름의 일부를
바이패스시킨다.
자중 낙하 방지
차동 회로
360[l/min]
가압
가압 때에는 열려 차동
회로를 해제한다.
디컴레션
디컴프레션 밸브
솔레노이드가 중
립이고 실린더 캡
쪽 압력이 빠진다.
SOL 1
SOL 2
저압 펌프 무부하
SOL ON
고압
SOL 3
고저압 회로
실린더 상승 때에
압력 제한
(SOL OFF)
M

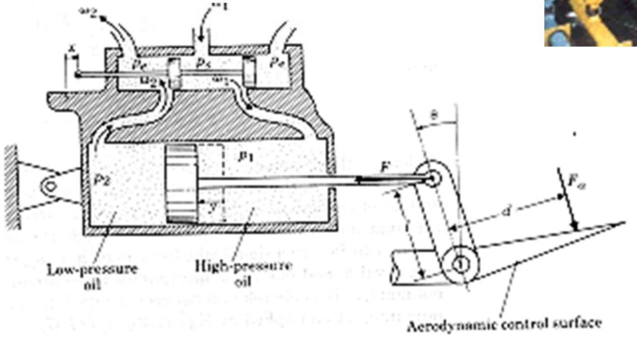
Linear Actuators

Motion
Simulator

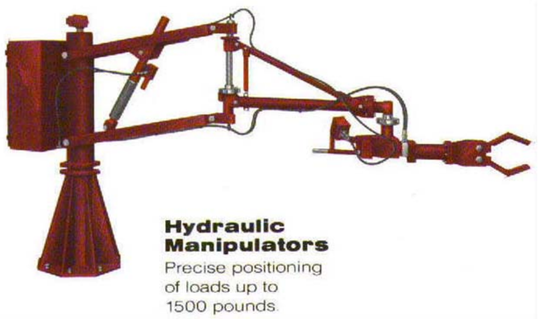
Press



Airplane

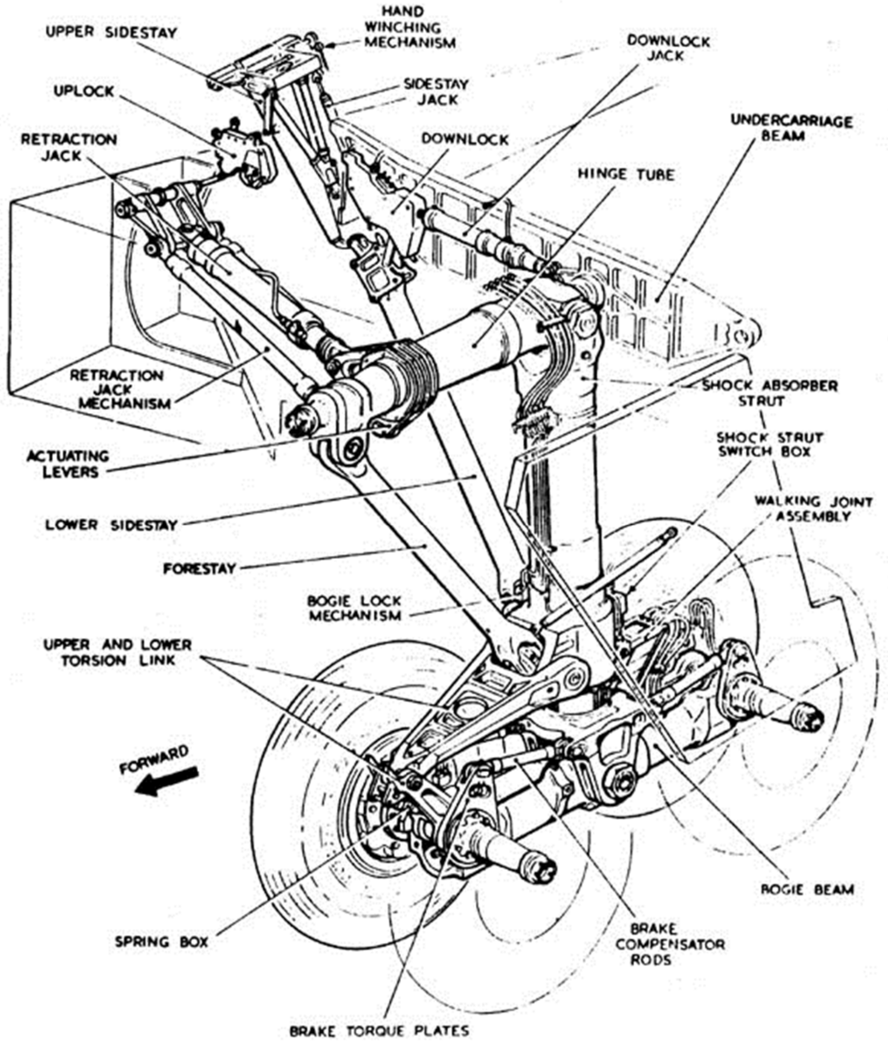


Robot

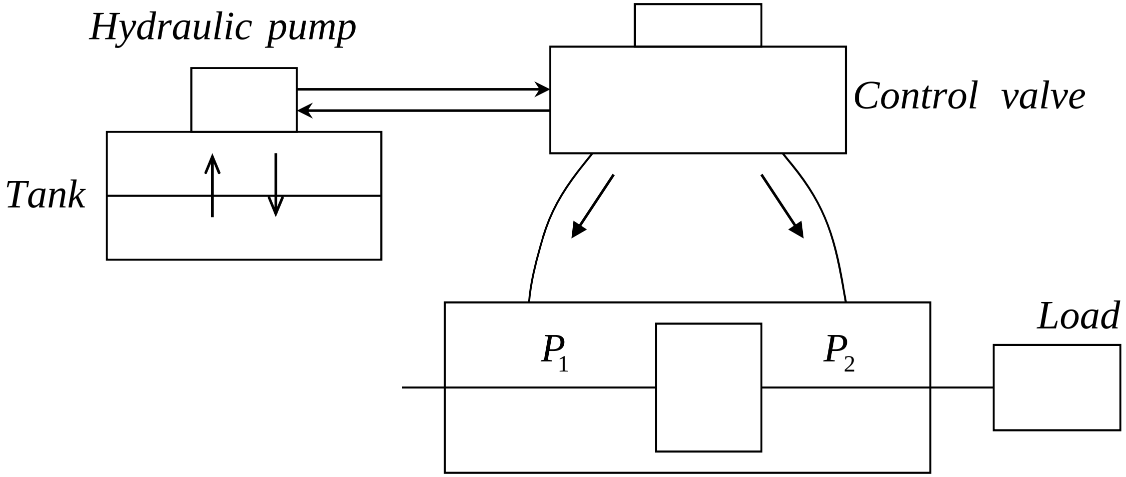


Hydraulic Systems : Landing Gear System

Landing gear system of AIRBUS A330



Hydraulic Systems



Hydraulic actuator $(70 \sim 210 bar)$

$$F = A_P \cdot (P_1 - P_2)$$

Why hydraulic ?

Internal combustion Engine

Turbine

Electric motor

Hydraulic actuator

……

Why hydraulic ?

1. smaller and lighter

   horsepower to weight ratio > 2 hp/lb

2. heat/lubrication – long component life
3. no saturation and losses

   - saturation and losses in magnetic materials of electrical machine
   - torque limit only by safe stress levels

4. high natural frequency/high speed of response/high loop gains

   - electrical motors, a simple lag device from applied voltage to speed

5. dynamic breaking with relief valve without damage

Disadvantages

1. not so readily available

2. small allowable tolerances result in high costs

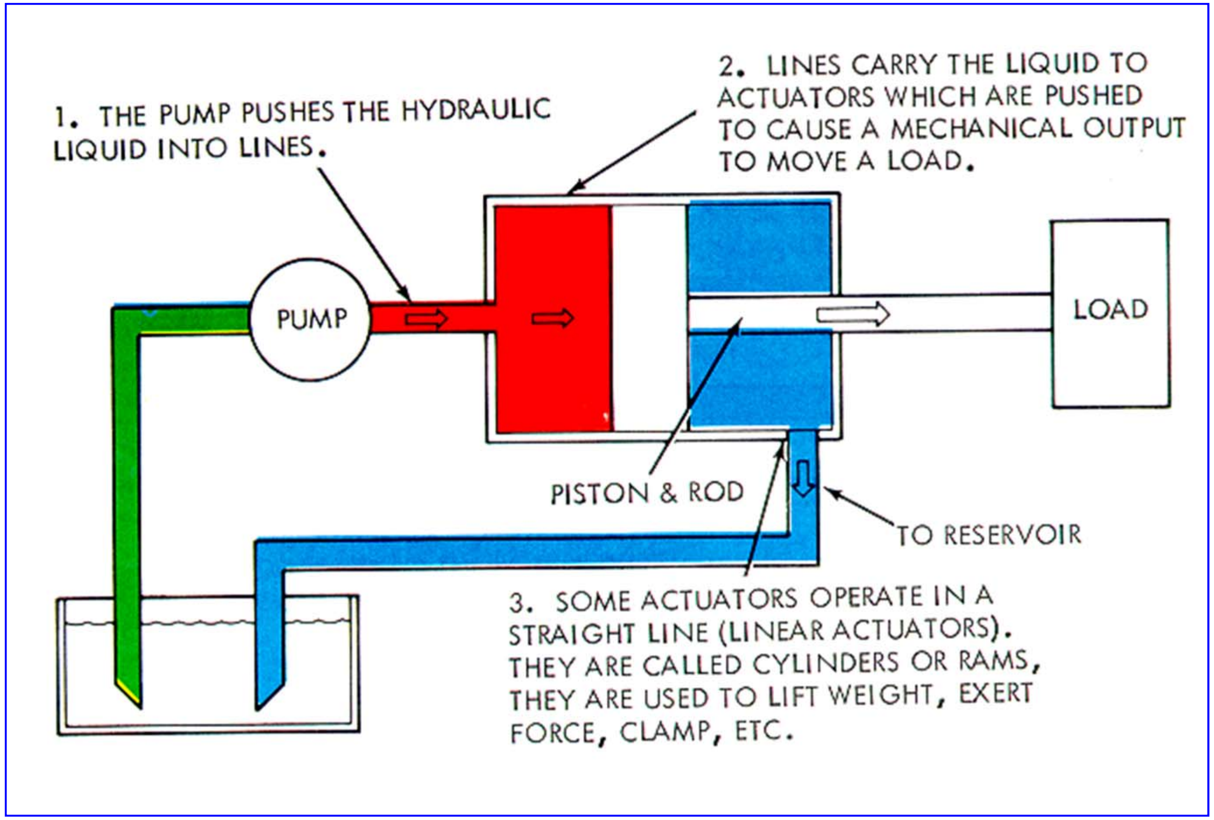
3. hydraulic fluids imposes upper temperature limit.

4. fluid contamination: dirt and contamination

5. basic design procedures are lacking and difficult, complexity of hydraulic control analysis

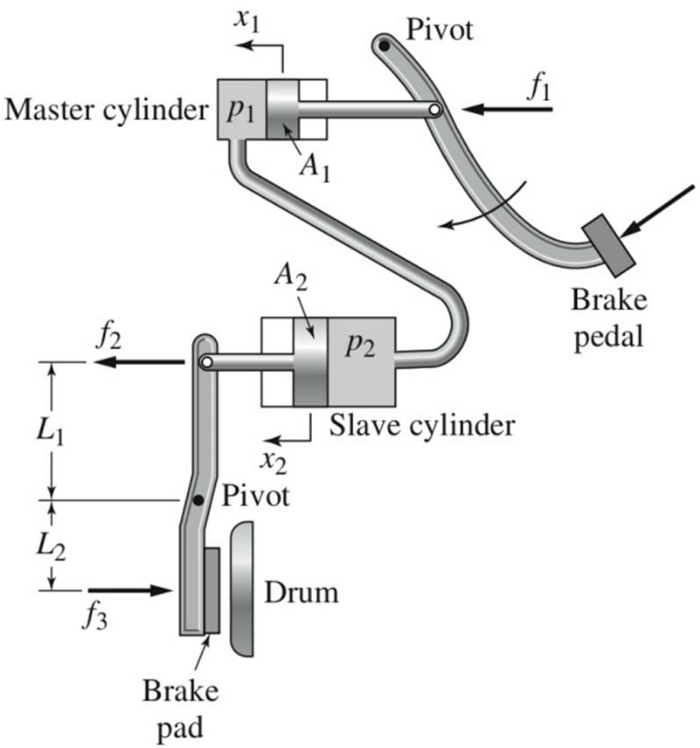
6. not so flexible, linear, accurate, and inexpensive as electronic and/or electromechanical devices

Primary Functions of a Hydraulic Fluid



Conservation of Mass, Force and Pressure

$$\dot{m} = q_{mi} - q_{mo}$$



$i)\ Volume \quad A_1 dx_1 = A_2 dx_2$

$$\rightarrow dx_2 = \frac{A_1}{A_2} dx_1$$

$ii)\ \text{Pr}\,essue \quad p_2 = p_1 + \rho g h$

$iii)$ **Force** $\quad f_1 = p_1 A_1$

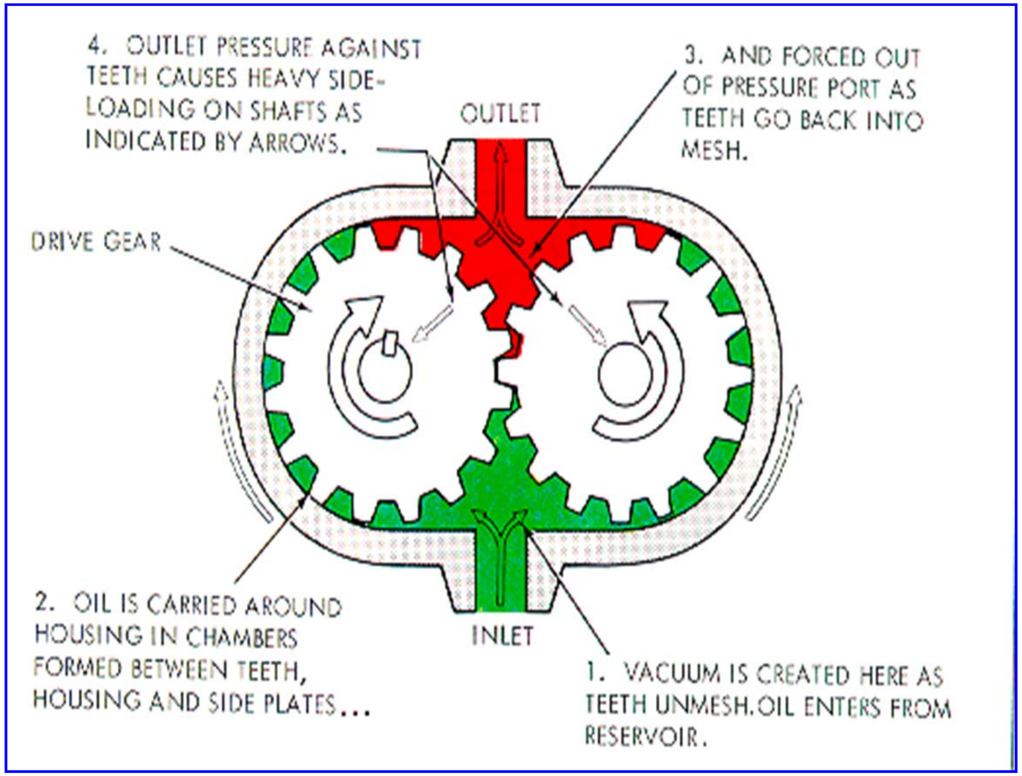
$$f_2 = p_2 A_2$$

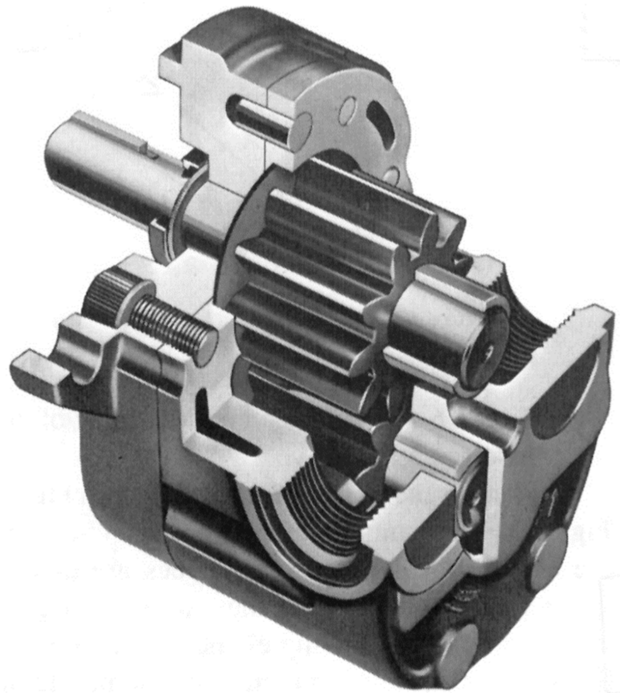
$$f_2 = \frac{A_2}{A_1} f_1 \qquad (P_1 \approx P_2)$$

$$f_3 = \frac{L_1}{L_2} f_2 = \frac{L_1}{L_2} \frac{A_2}{A_1} f_1$$

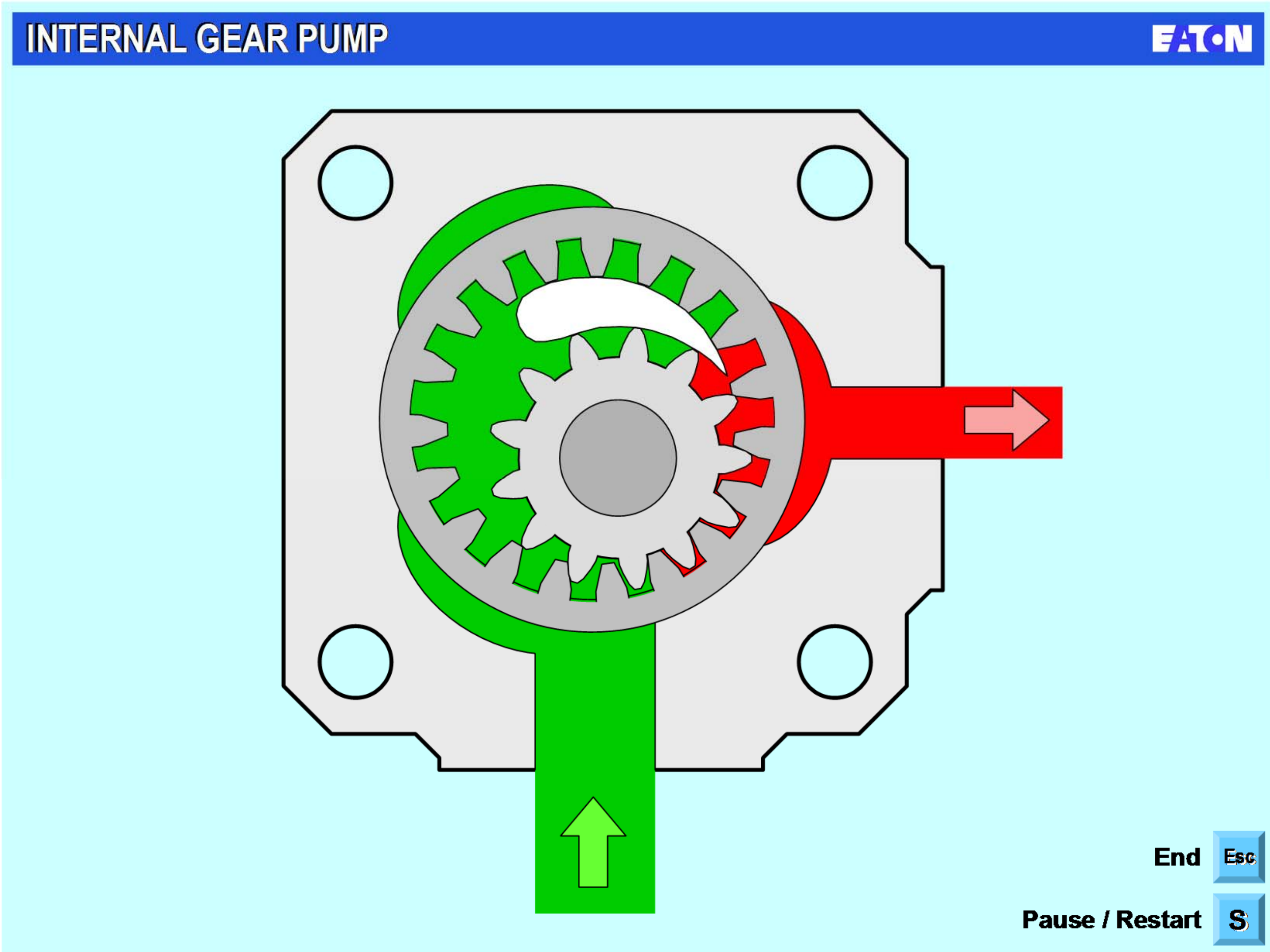
Gear Pumps (External)

- fixed displacement pump
- uses spur gear (teeth are parallel to the axis of the gear)
- noisy at relatively high speeds



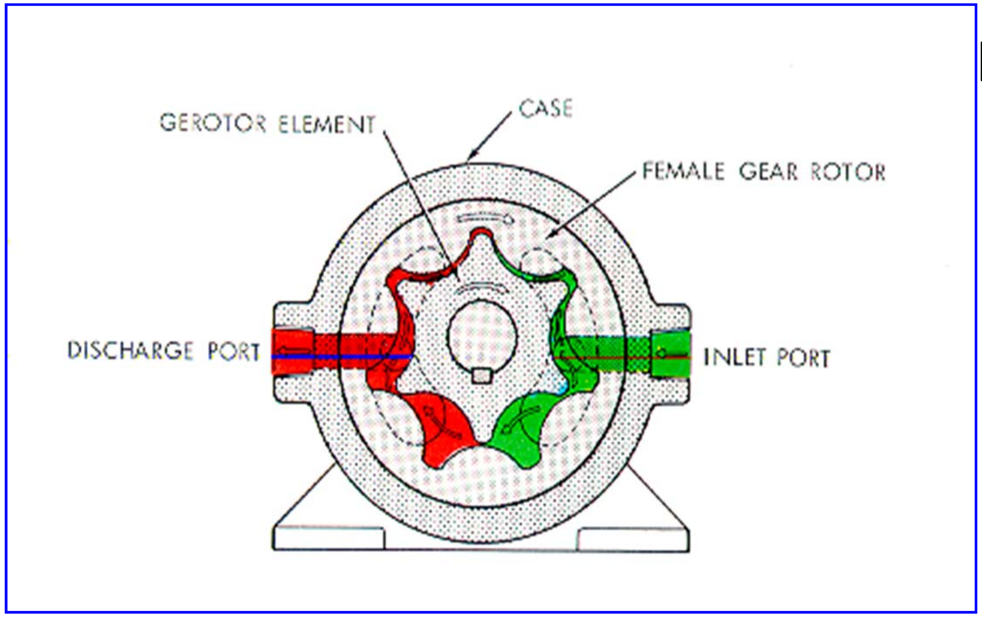


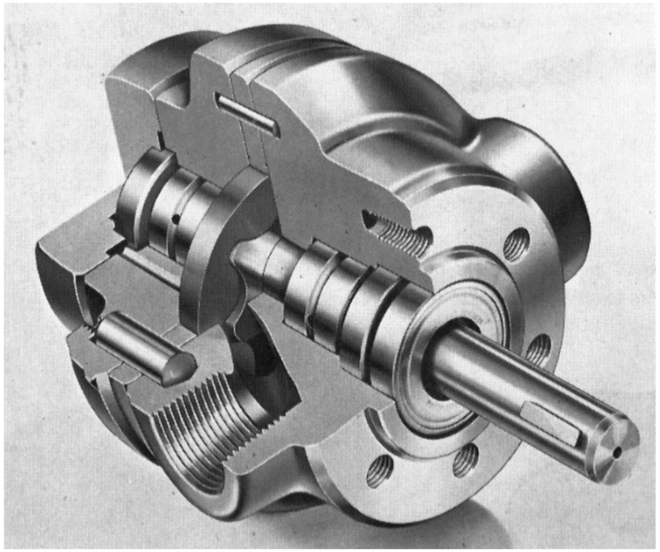
Internal Gear Pump



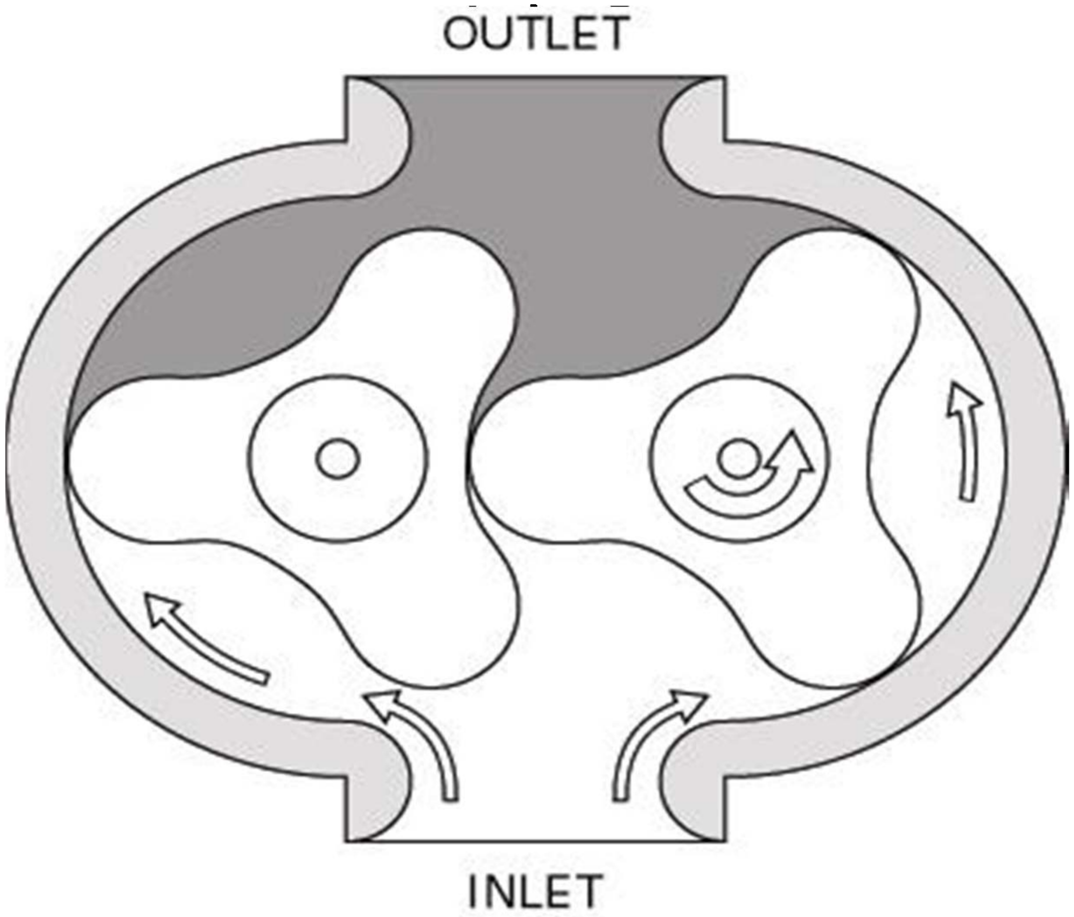
Internal Gear Pump

p

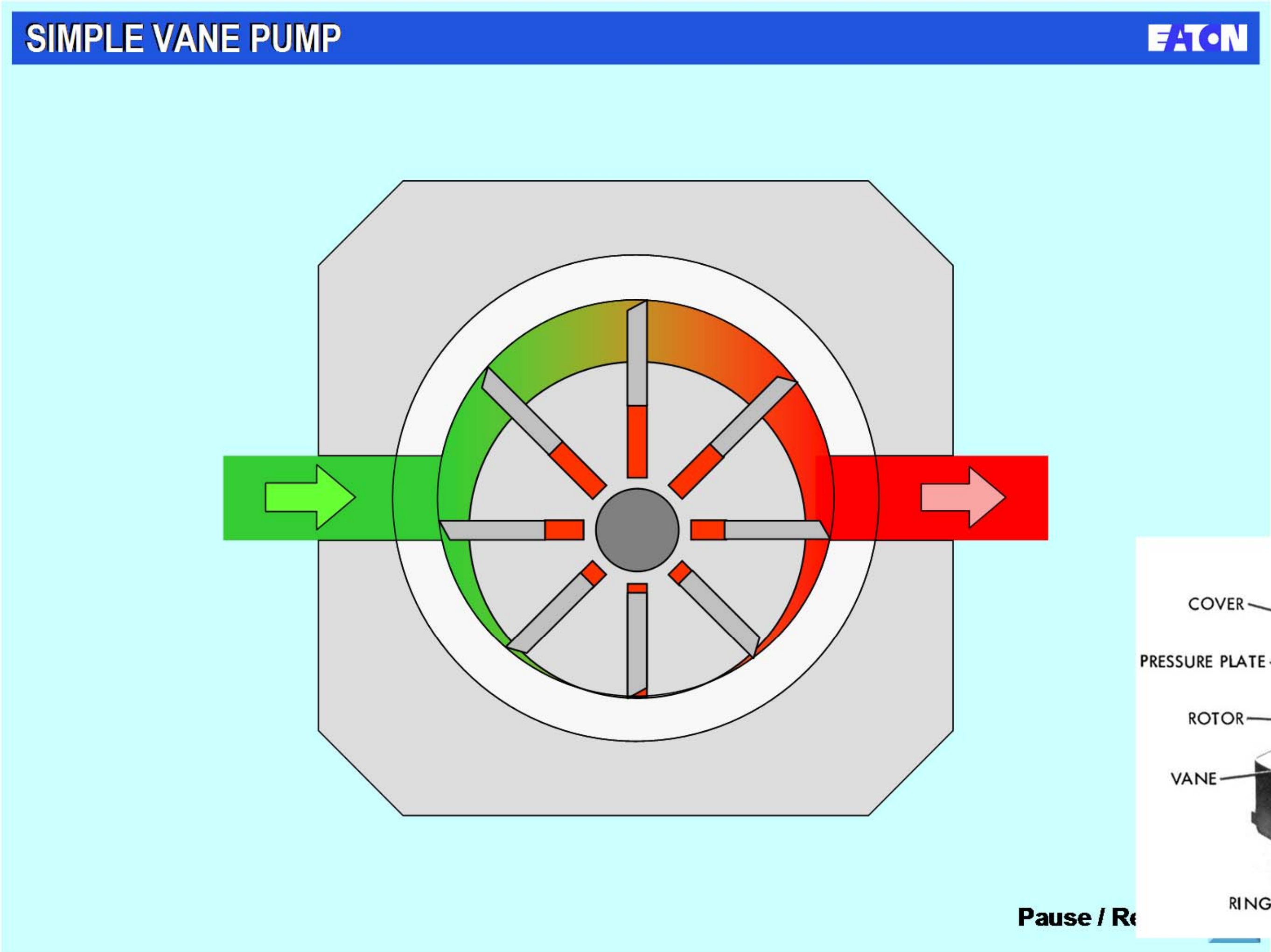


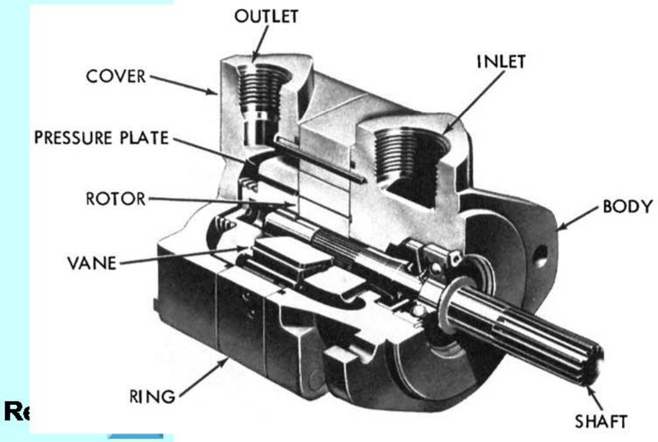


Internal Gear Pump

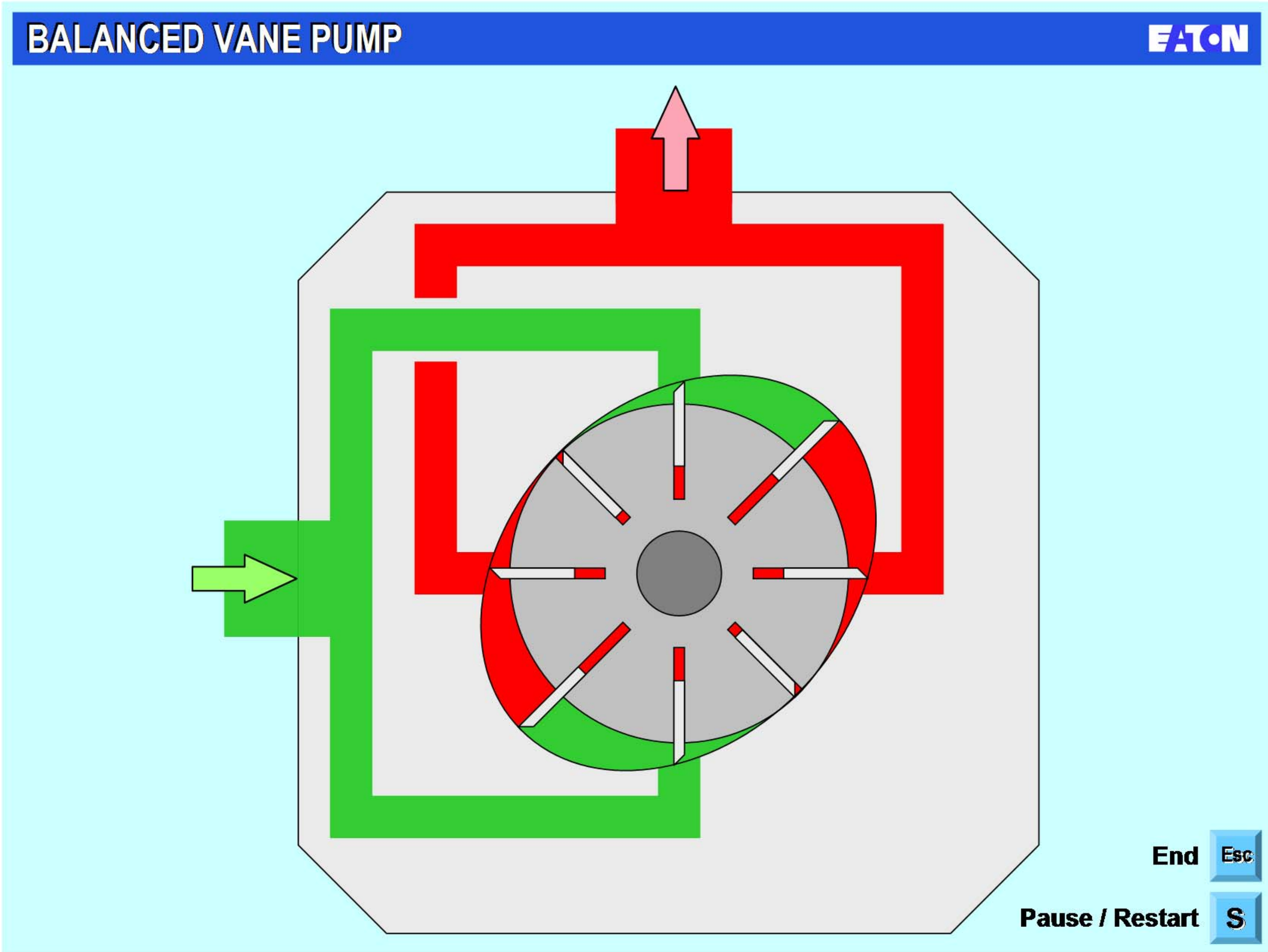


Simple Vane Pump

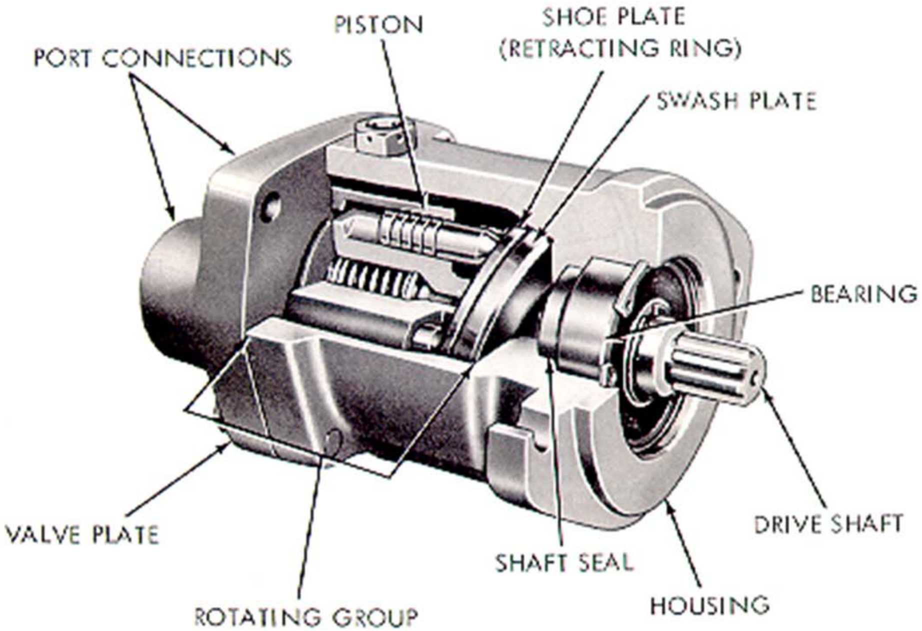


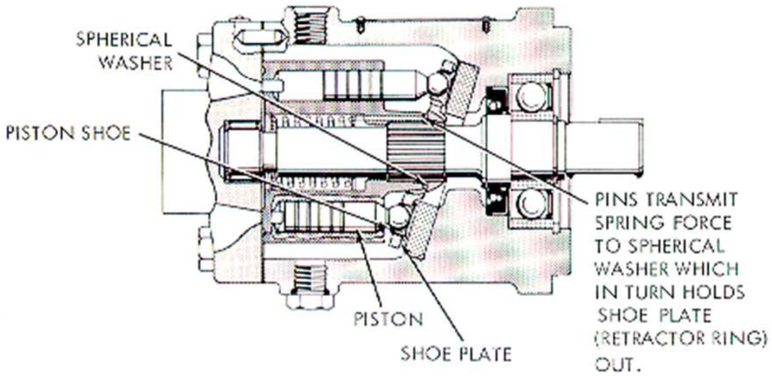


Balanced Vane Pump

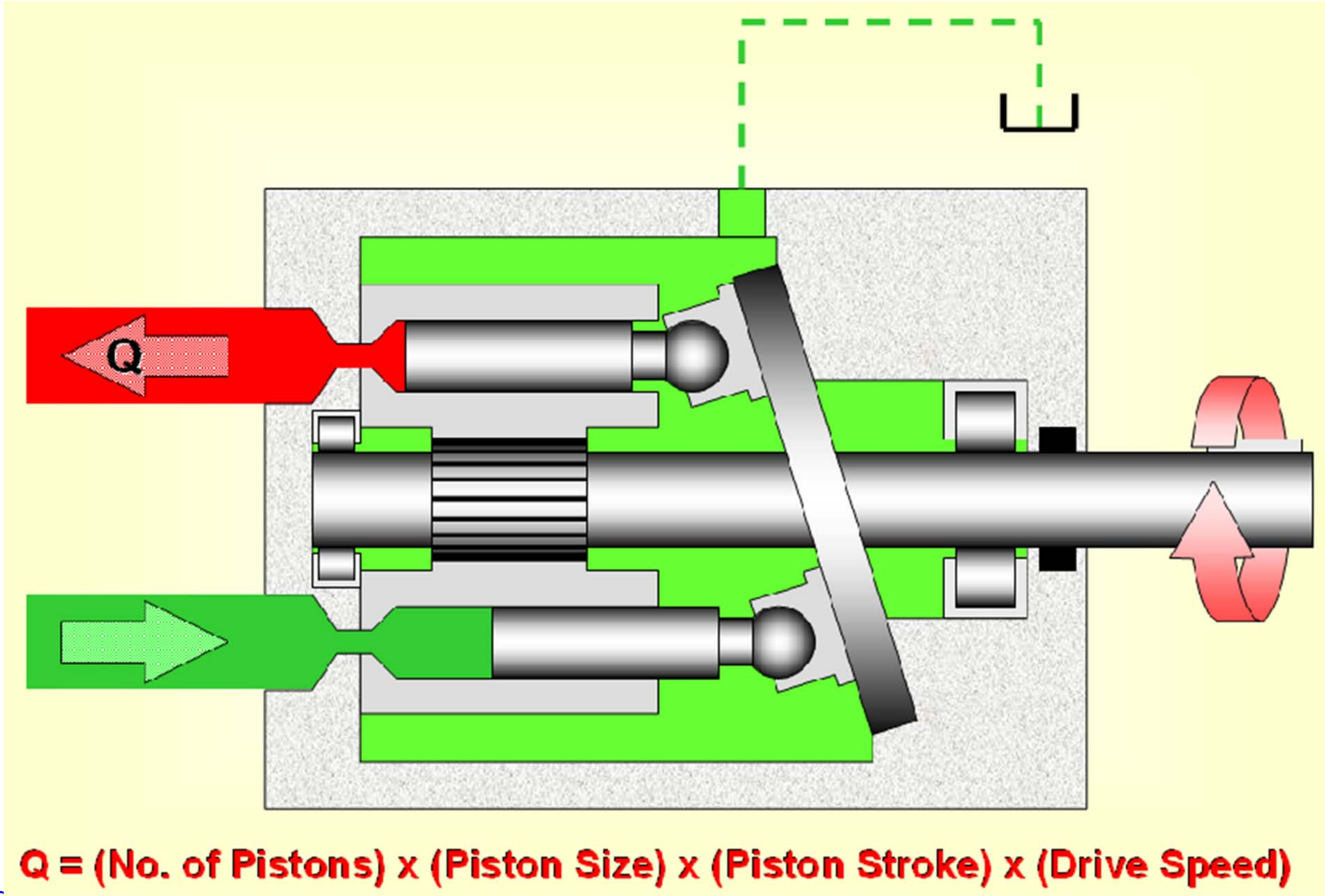


Piston Pump (Swash Plate Type)



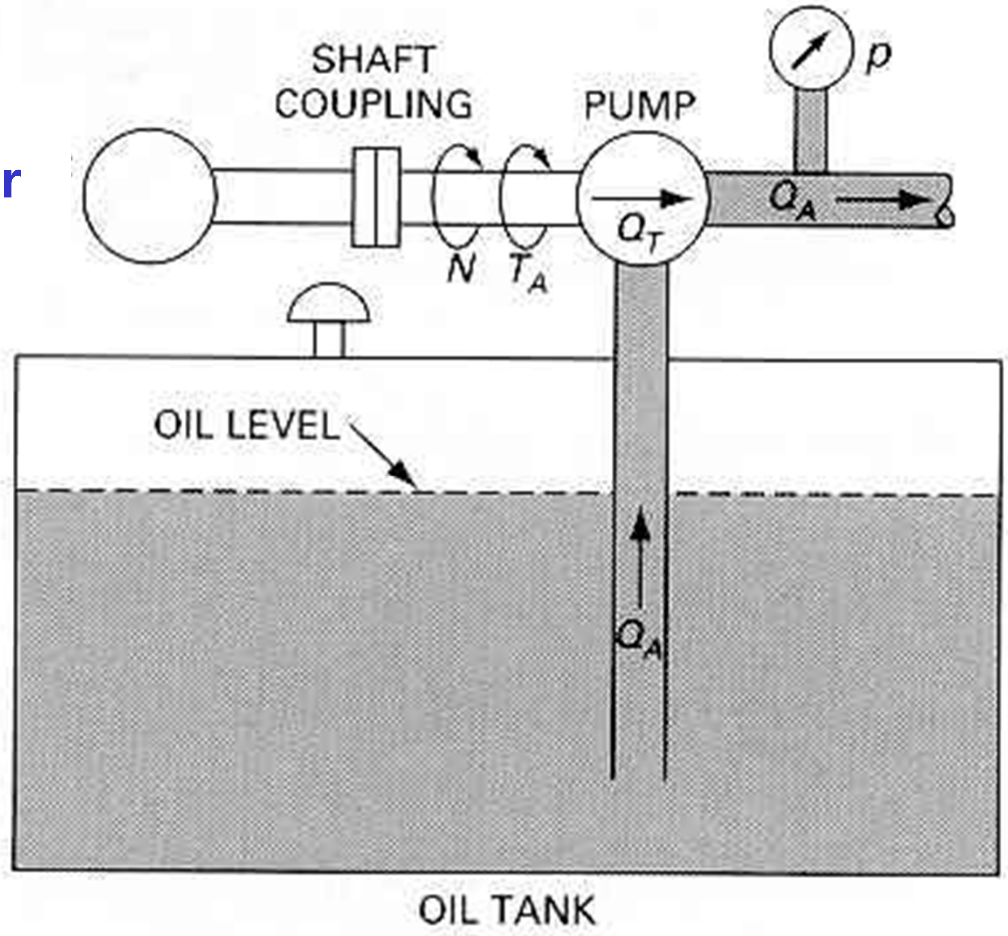


Piston Pump



Pump

Electric
Motor or
Engine



Hydraulic Pump

- **the heart of a hydraulic system**
- **converts mechanical energy into hydraulic energy**
- **hydrostatic pump**
- **Displacement**
  - the amount of fluid ejected per revolution
  - unit: cm^3/rev, cc/rev, cm^3/rad, cc/rad

Pump Torque

- **Mechanical power supplied to pump**

$$H_m = T\omega$$

- **Hydraulic power delivered by pump**

$$H_p = PQ$$

$$P: pressure\ rise\ across\ the\ pump$$

$$Q: delivery\ rate$$

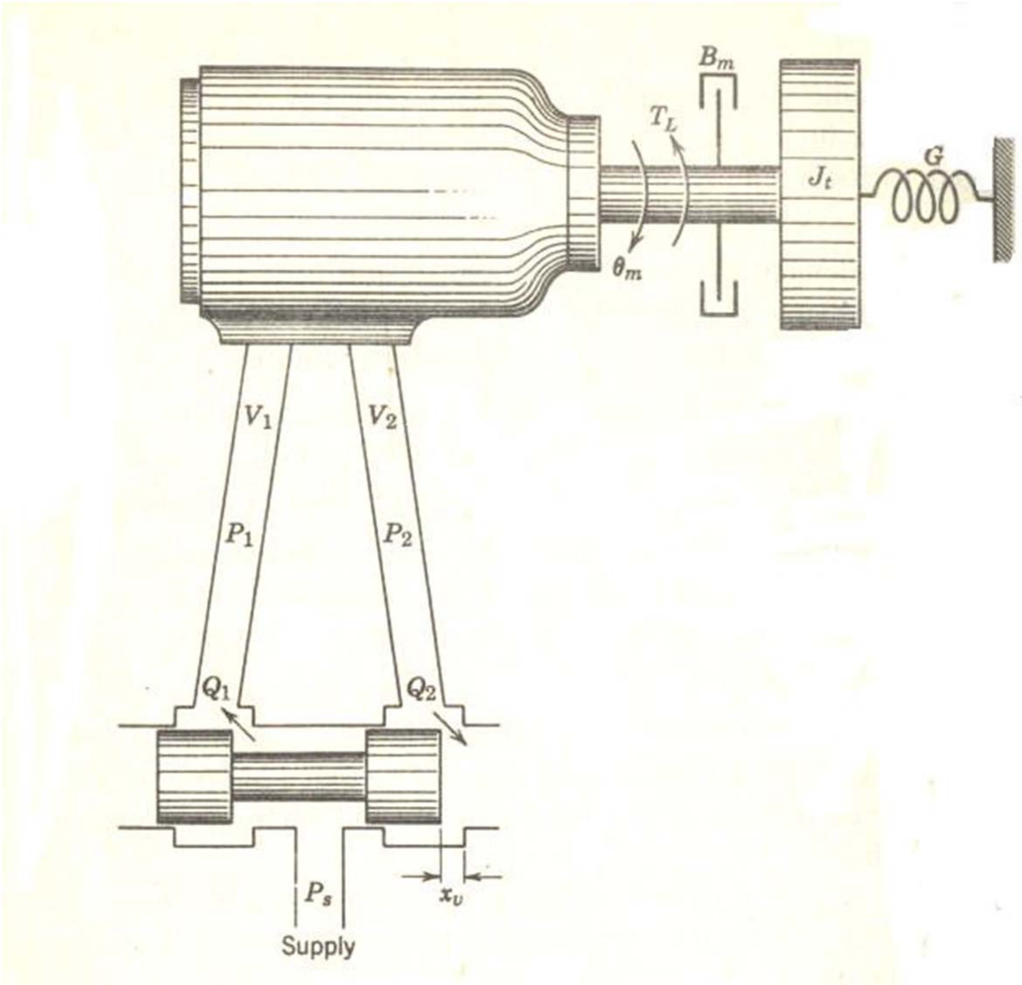
- **Therefore**

$$T_{th}\omega = PQ_{th} = P\omega D_p$$

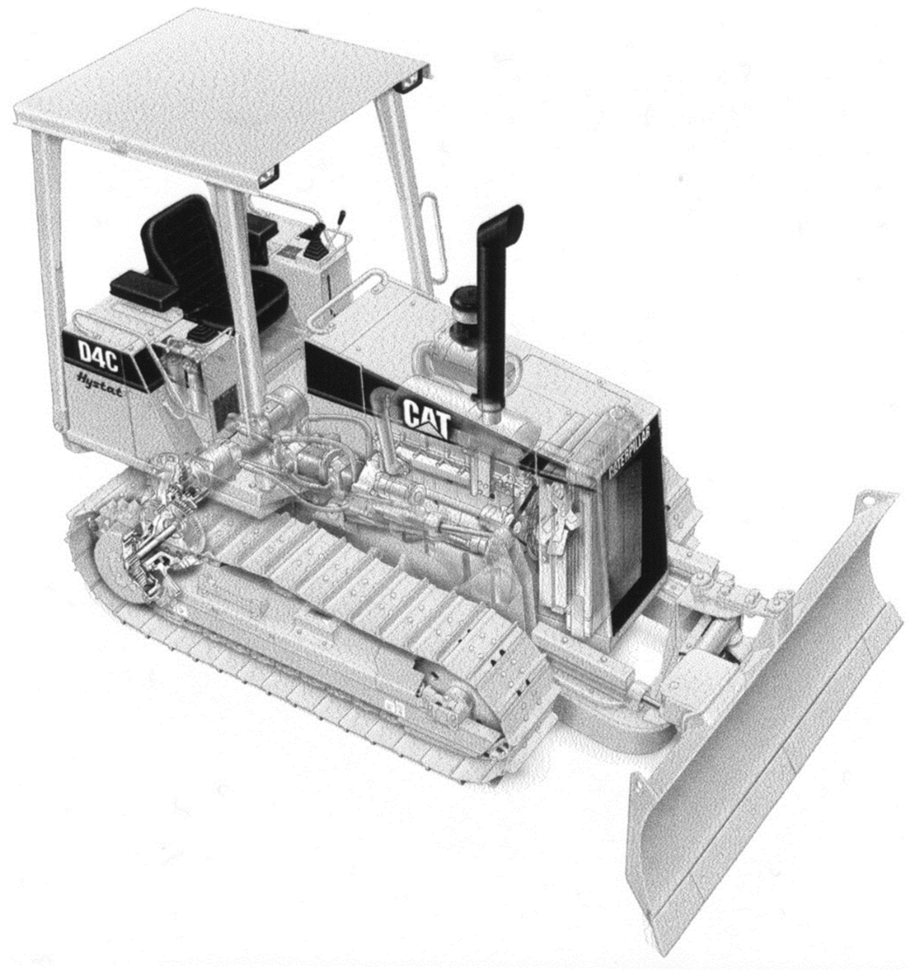
$$T_{th} = PD_p \qquad \text{여기서} D_p : \text{펌프배제용적}[m3/rad]$$

Hydraulic Motors and Actuators

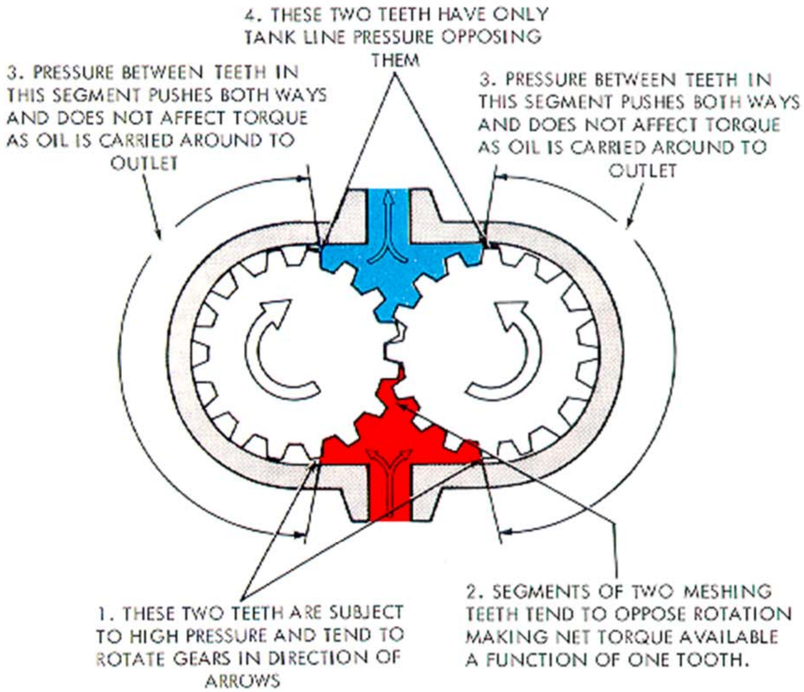
Hydraulic Systems : Valve-motor Combination

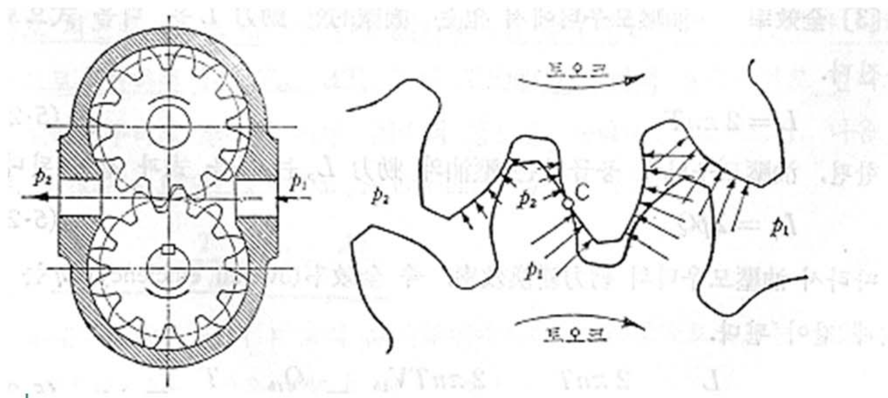


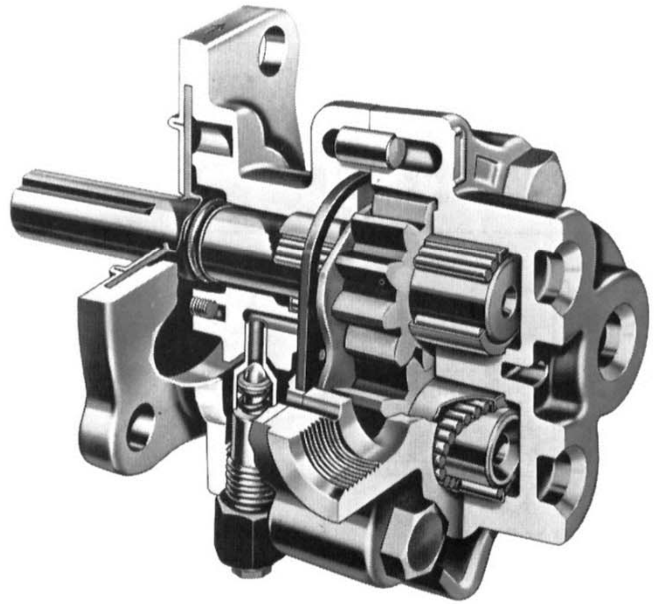
Application of Hydraulic Motors



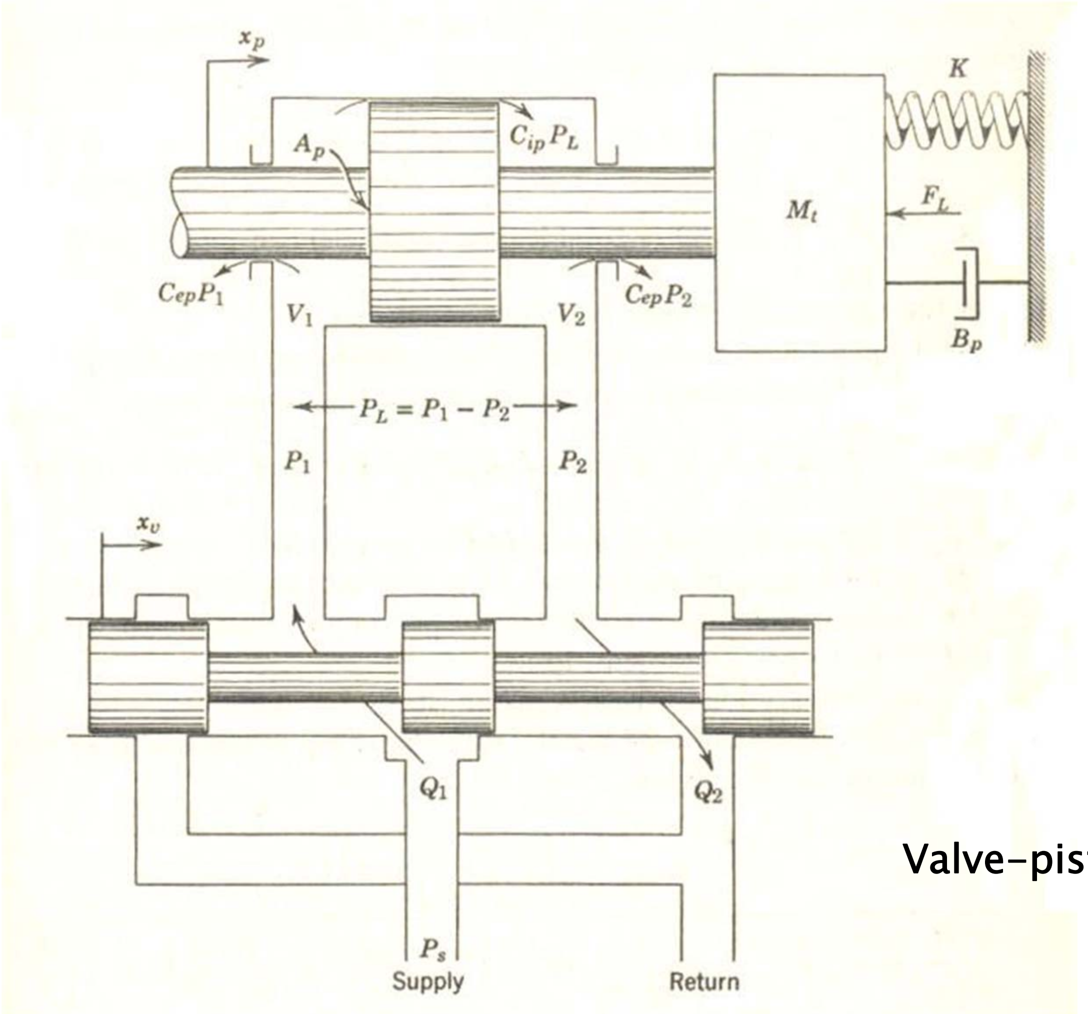
Gear Motors


4. THESE TWO TEETH HAVE ONLY TANK LINE PRESSURE OPPOSING THEM
3. PRESSURE BETWEEN TEETH IN THIS SEGMENT PUSHES BOTH WAYS AND DOES NOT AFFECT TORQUE AS OIL IS CARRIED AROUND TO OUTLET
3. PRESSURE BETWEEN TEETH IN THIS SEGMENT PUSHES BOTH WAYS AND DOES NOT AFFECT TORQUE AS OIL IS CARRIED AROUND TO OUTLET
1. THESE TWO TEETH ARE SUBJECT TO HIGH PRESSURE AND TEND TO ROTATE GEARS IN DIRECTION OF ARROWS
2. SEGMENTS OF TWO MESHING TEETH TEND TO OPPOSE ROTATION MAKING NET TORQUE AVAILABLE A FUNCTION OF ONE TOOTH.


토오크
p_2
p_1
p_2
p_2
C
p_1
p_1
토오크

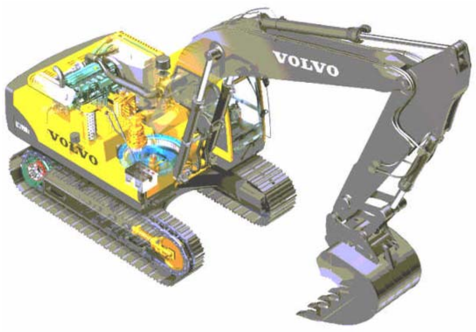


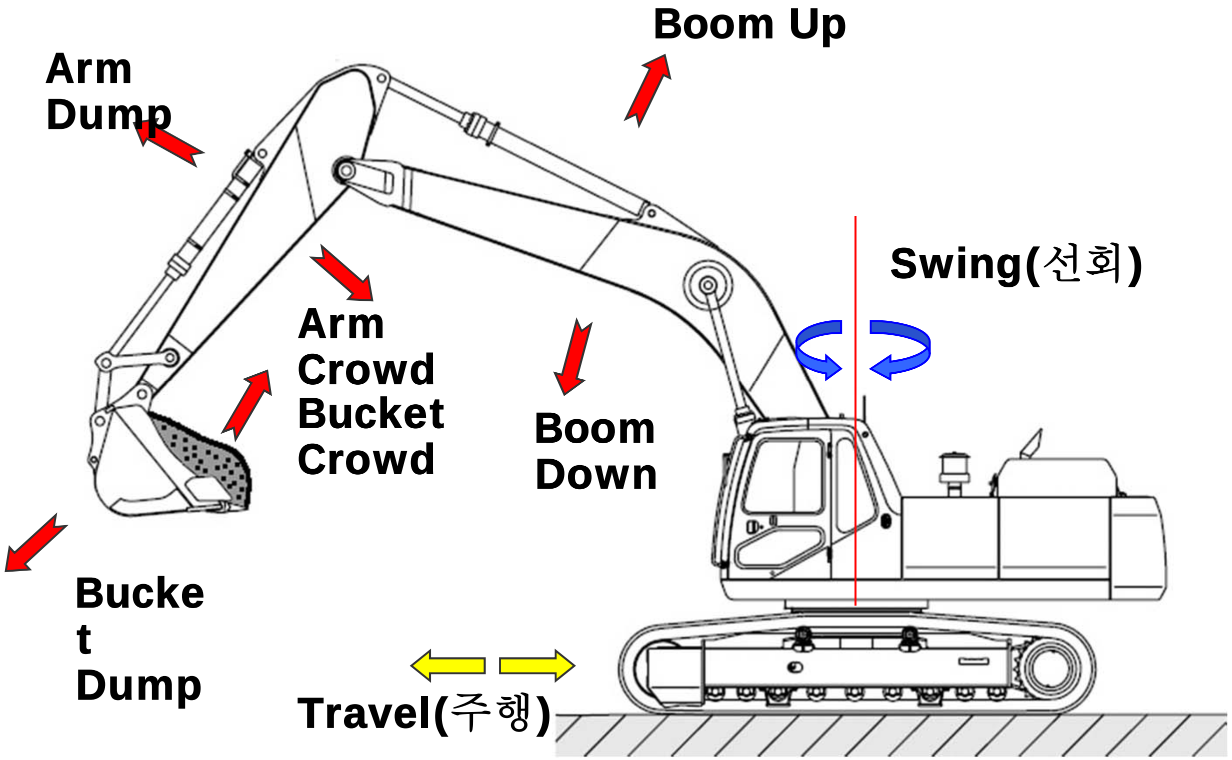
Hydraulic Systems : Valve-piston Combination



Valve–piston combination

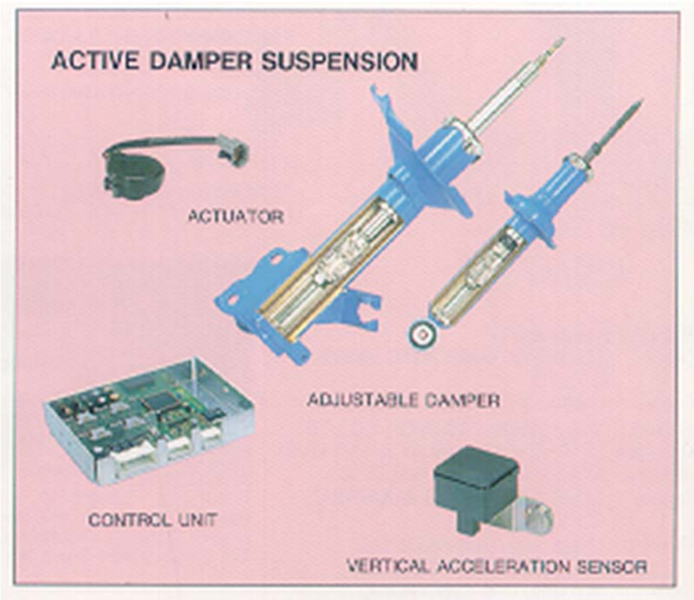
Hydraulic Excavator

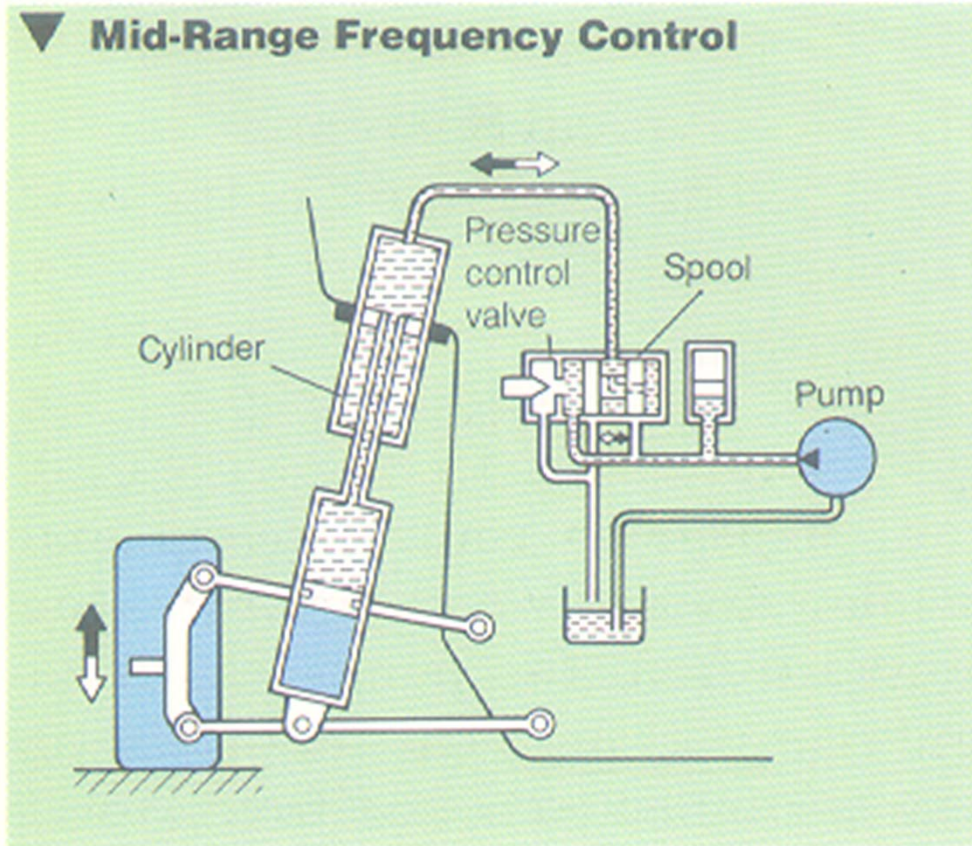




Hydraulic Excavator

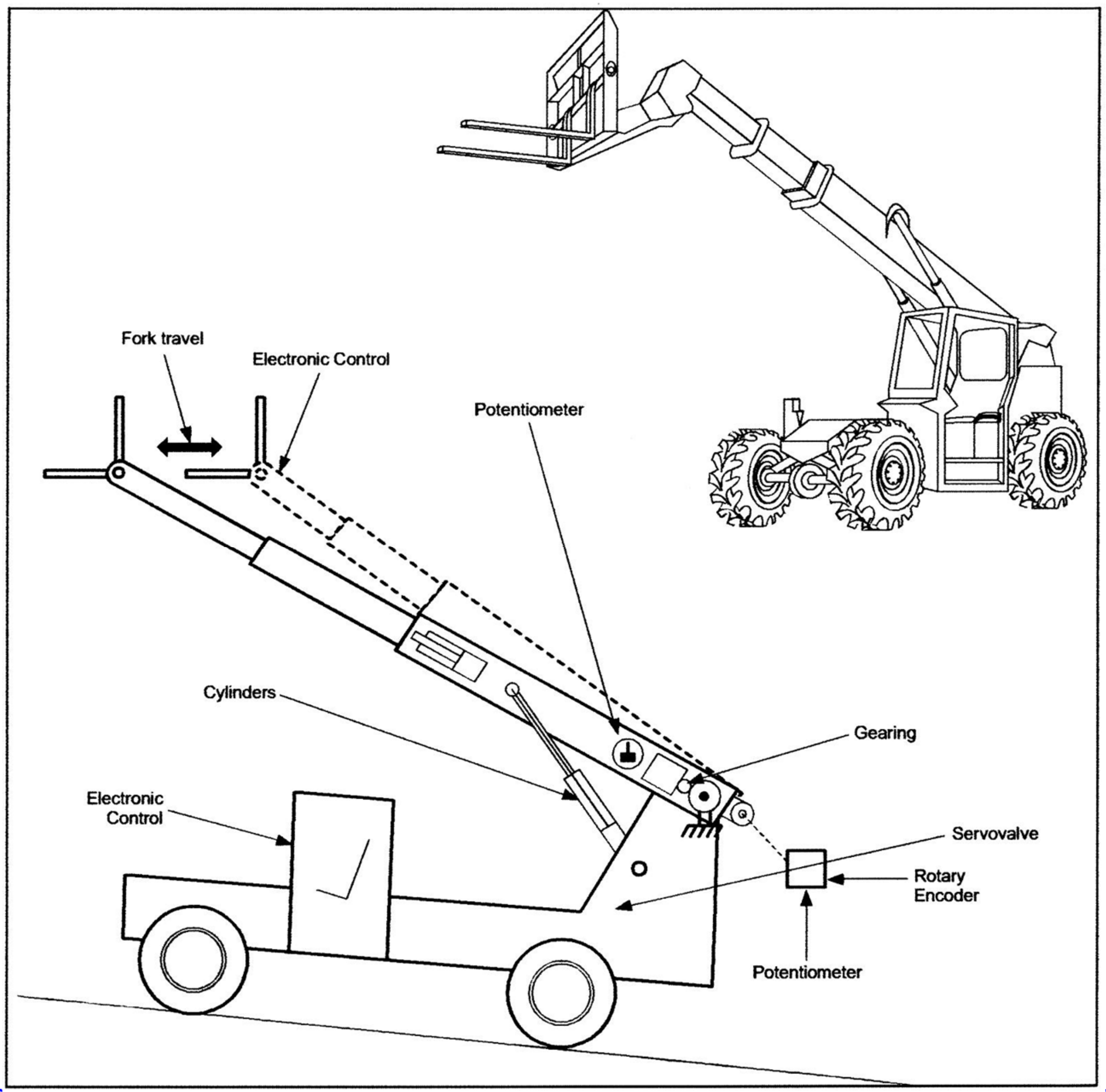
Automotive Application



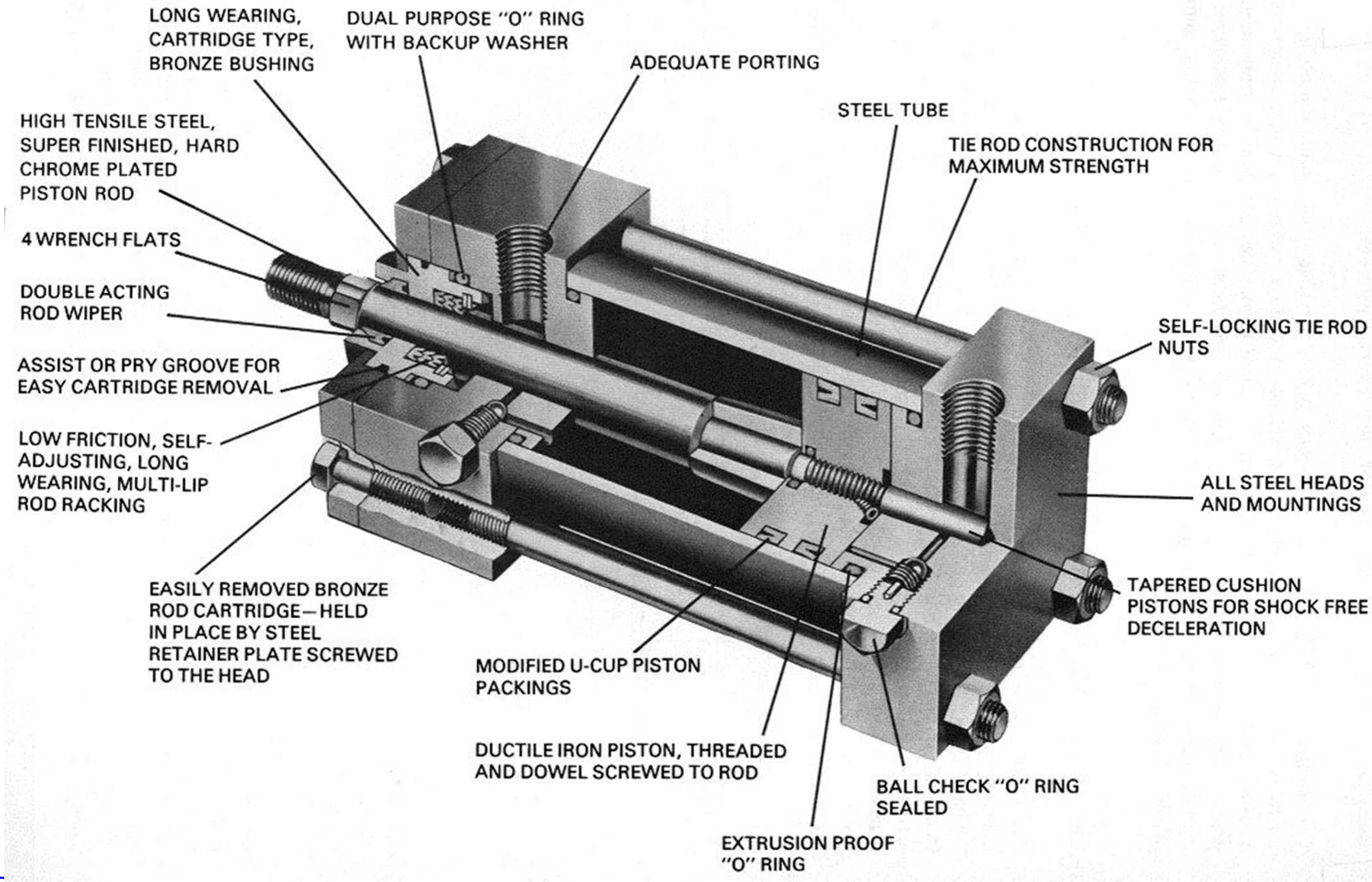


Active Suspension

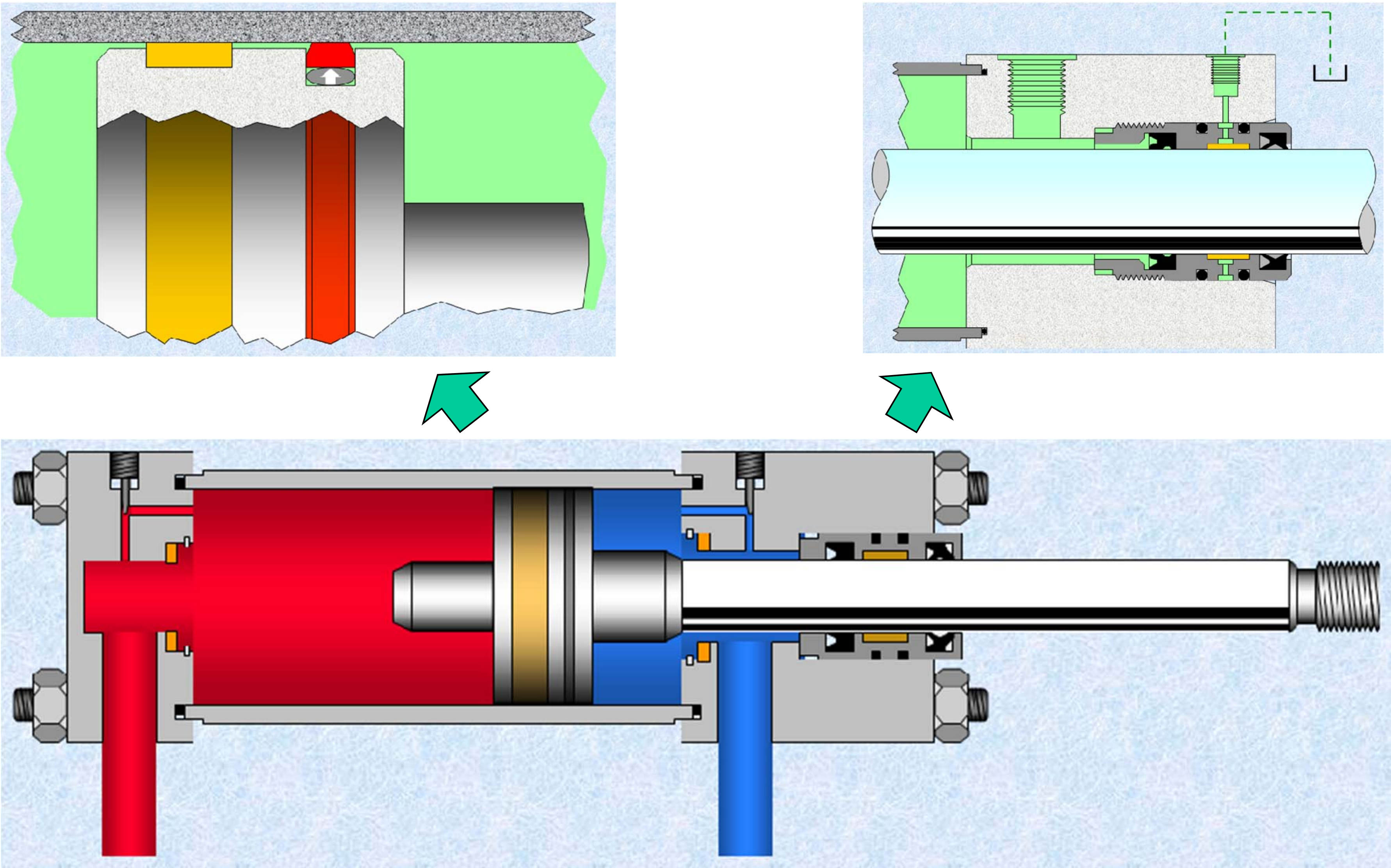
Rough terrain forklift driven by hydraulic cylinders



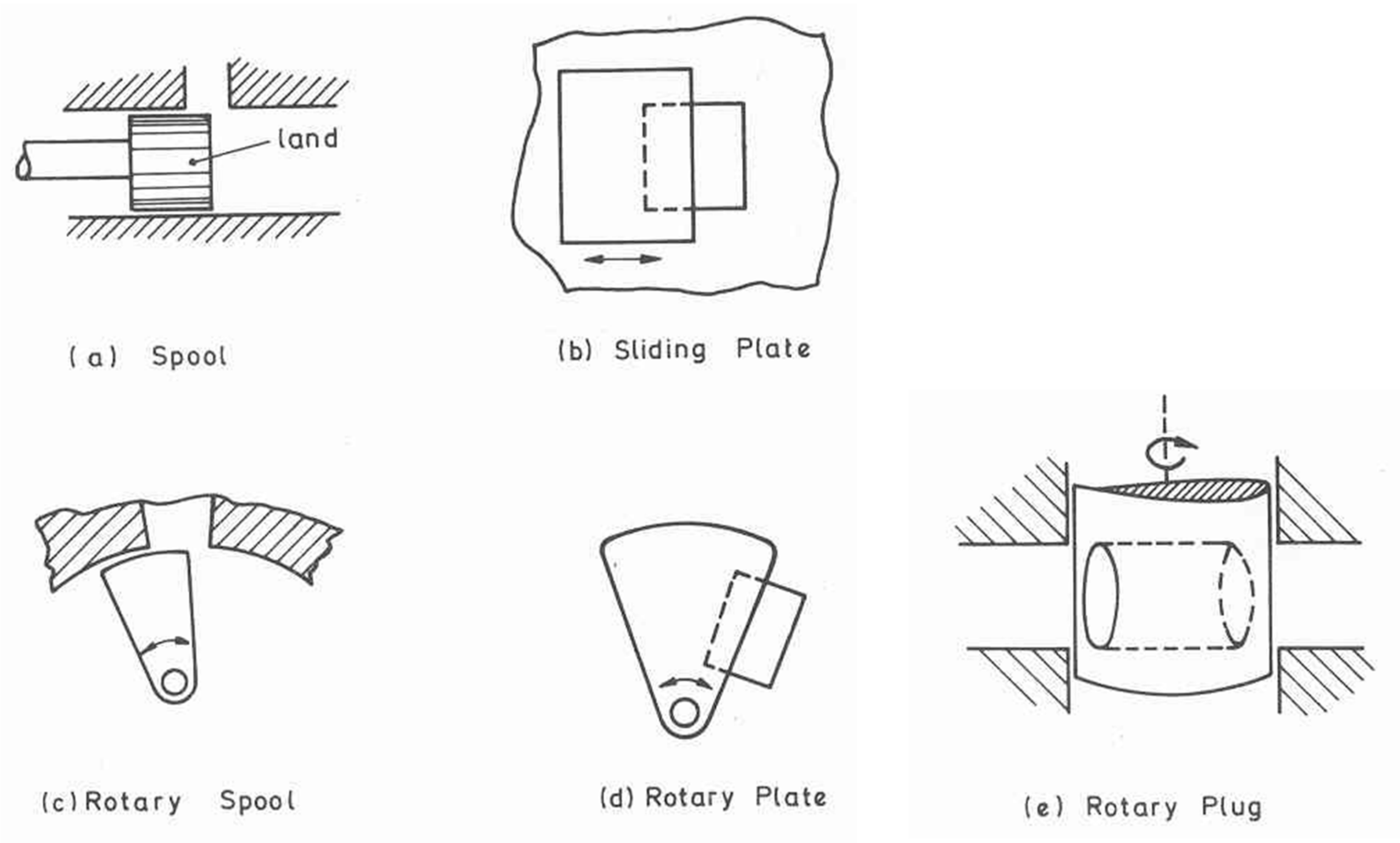
Double-acting Cylinder Design



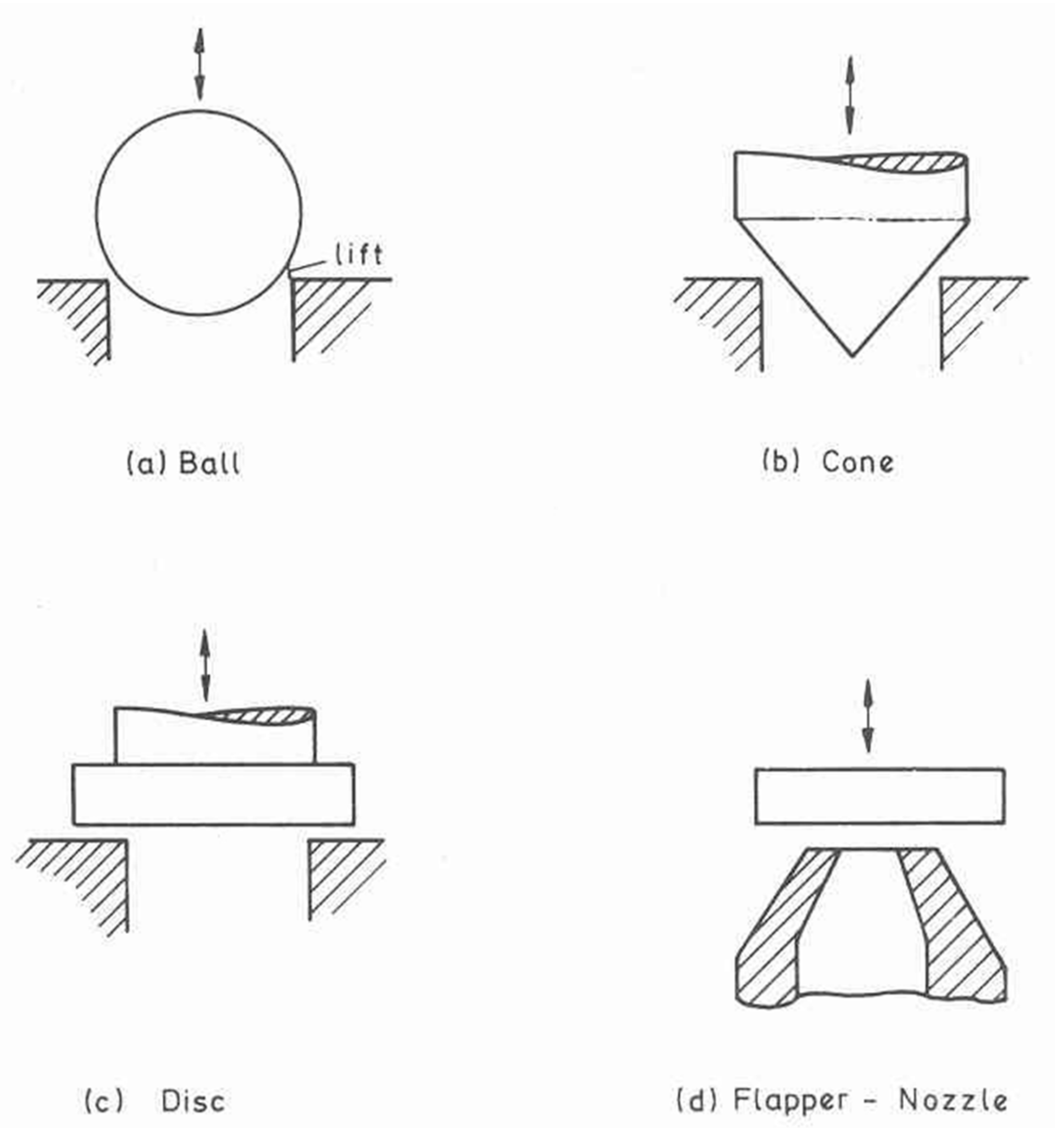
Cylinder Construction



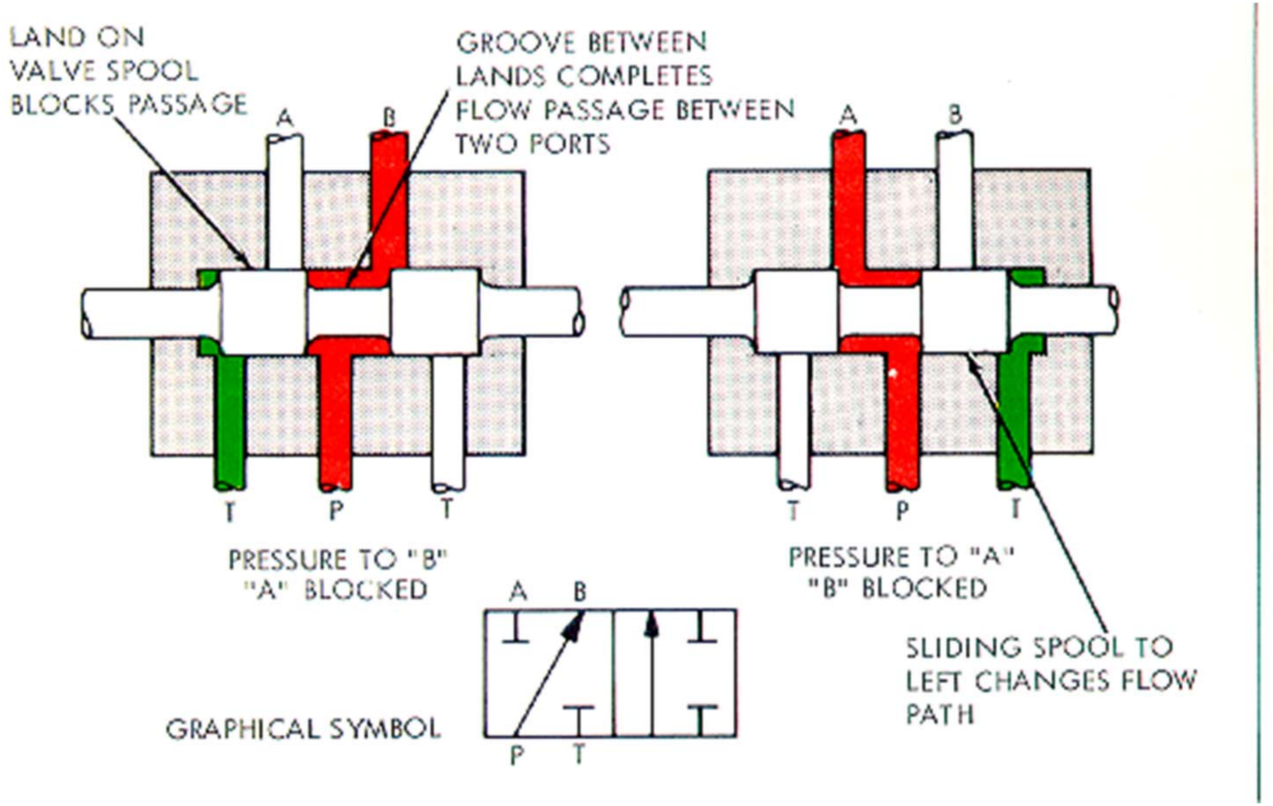
Types of Valves: shearing elements



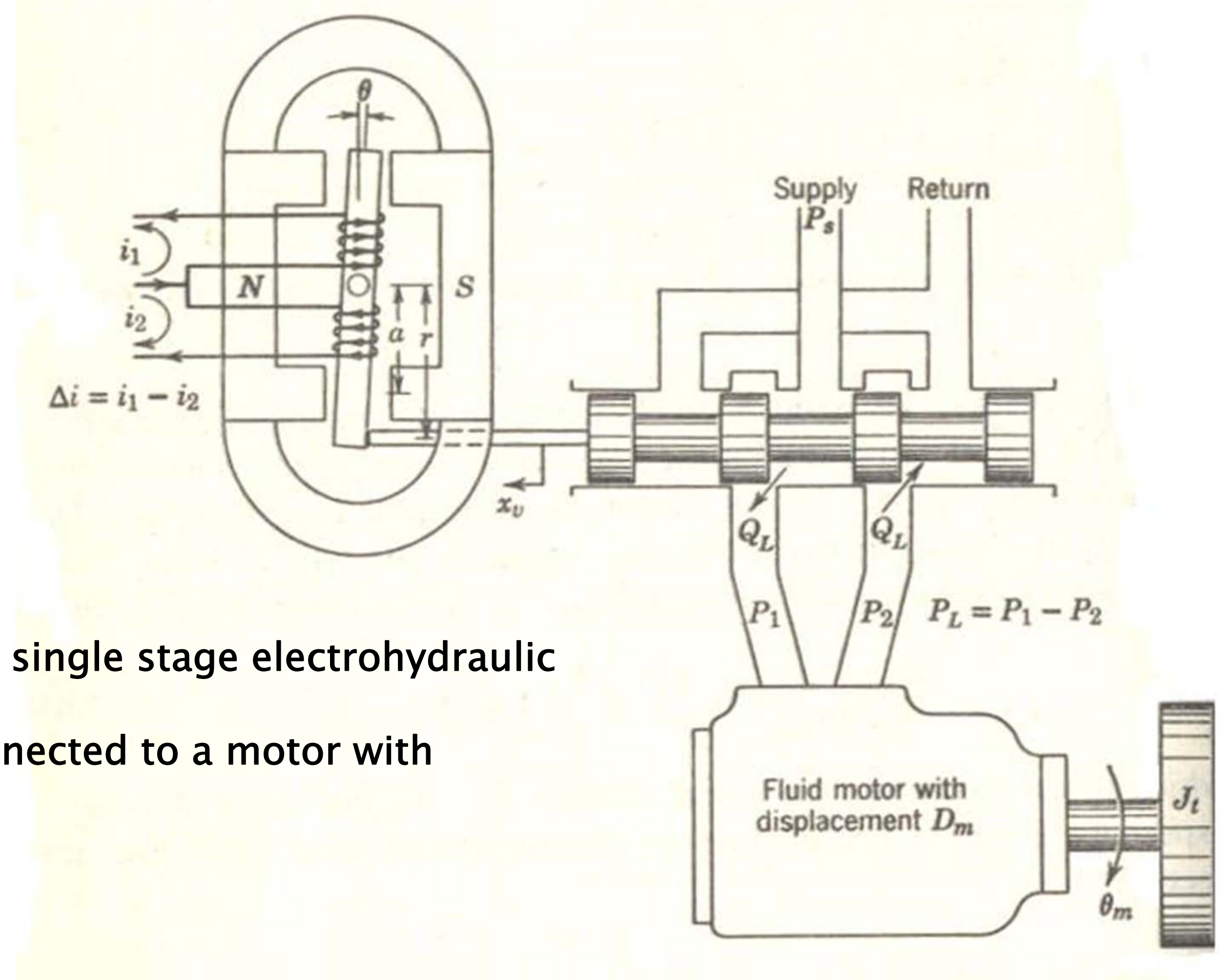
Types of Valves: seating elements



Directional Control Valve

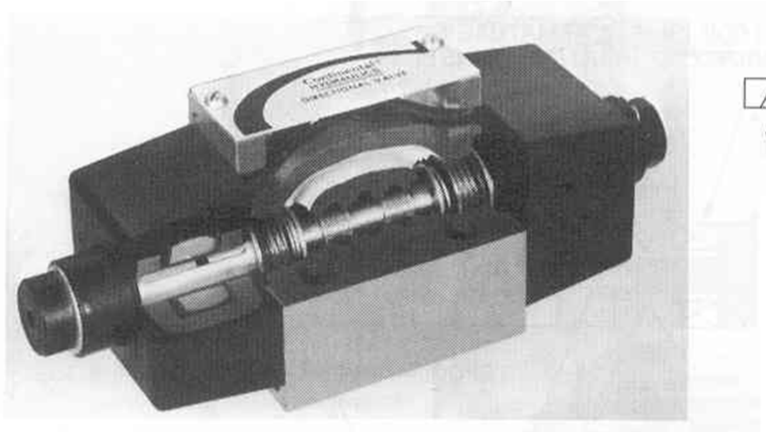


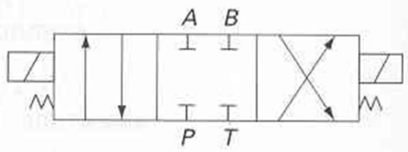
Hydraulic Systems : Single Stage Electrohydraulic Servovalve



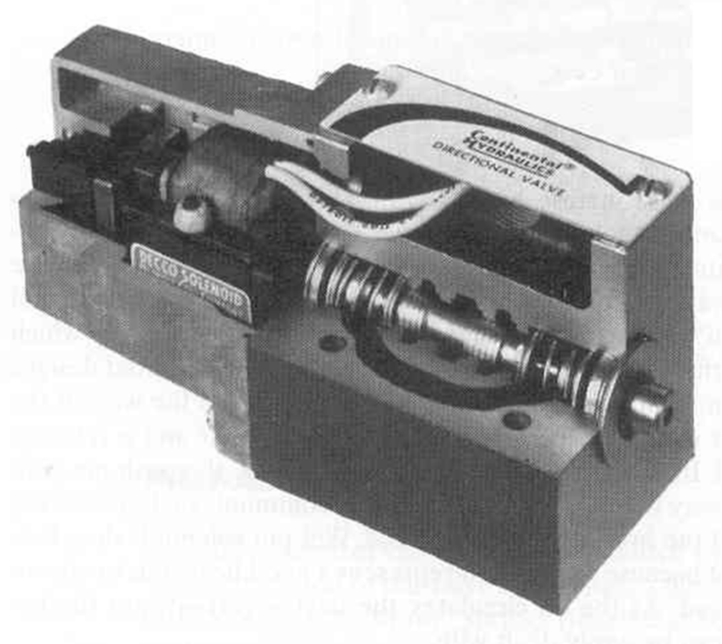
Schematic of a single stage electrohydraulic servovalve connected to a motor with inertia load

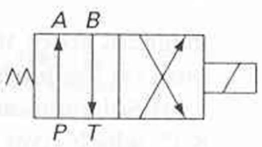
Solenoid-Actuated Valves





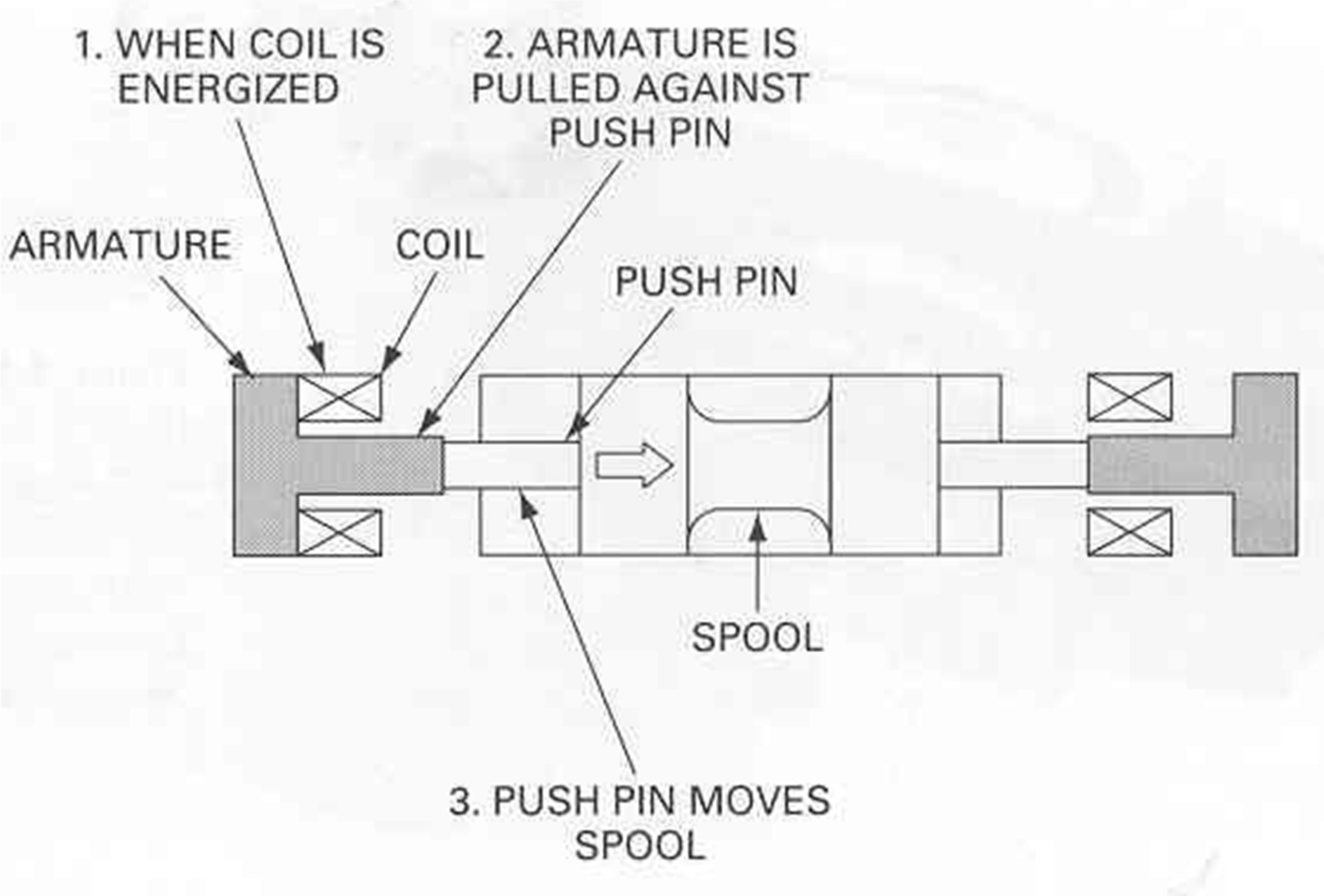
❖ Solenoid-actuated, three-position, spring-centered, four-way, directional control valve



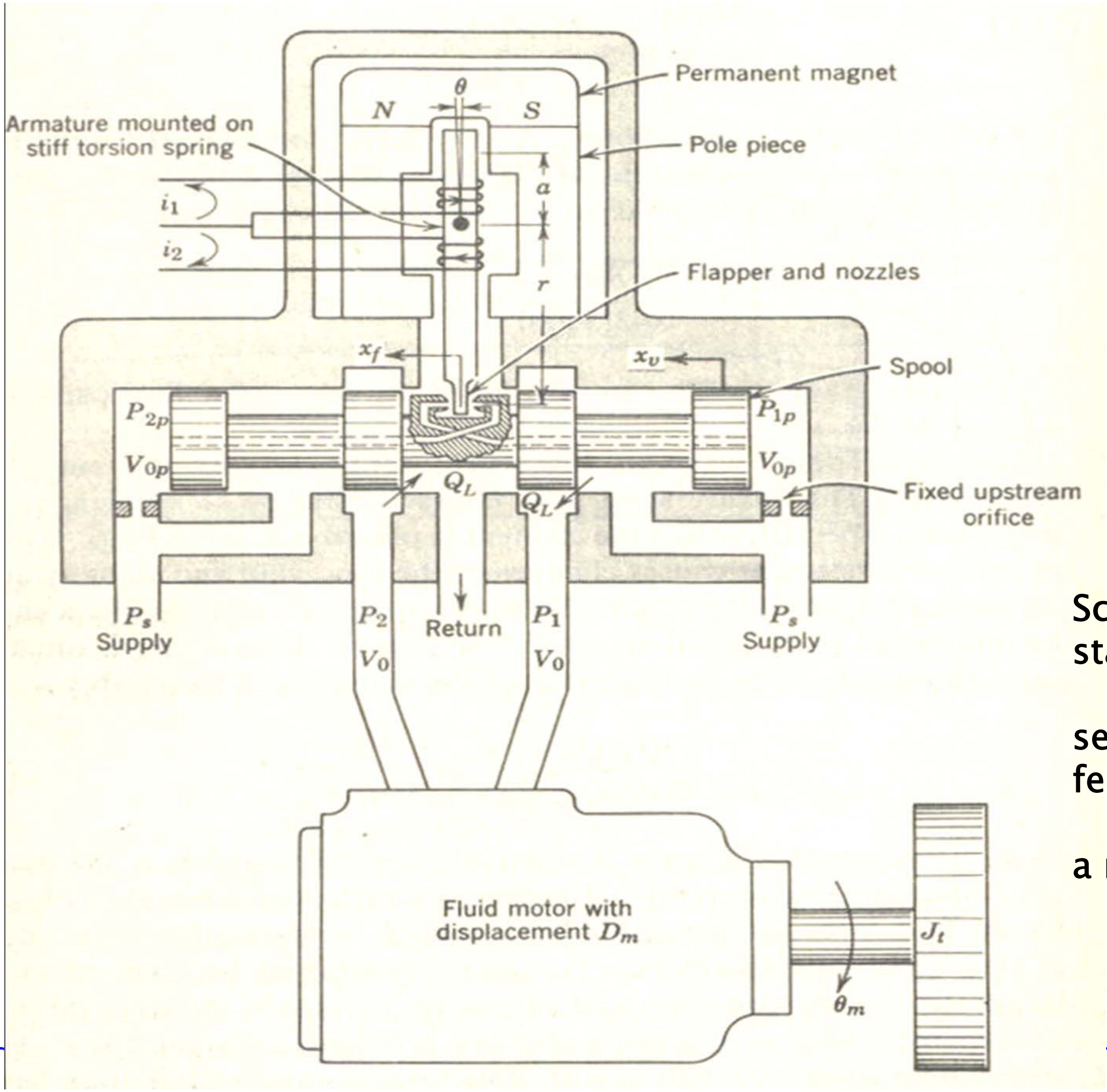


❖ Single solenoid-actuated, two-position, spring-offset, four-way, directional control valve

Operation of Solenoid to Shift of Valve



Hydraulic Systems : Two-stage Electrohydraulic Servovalve

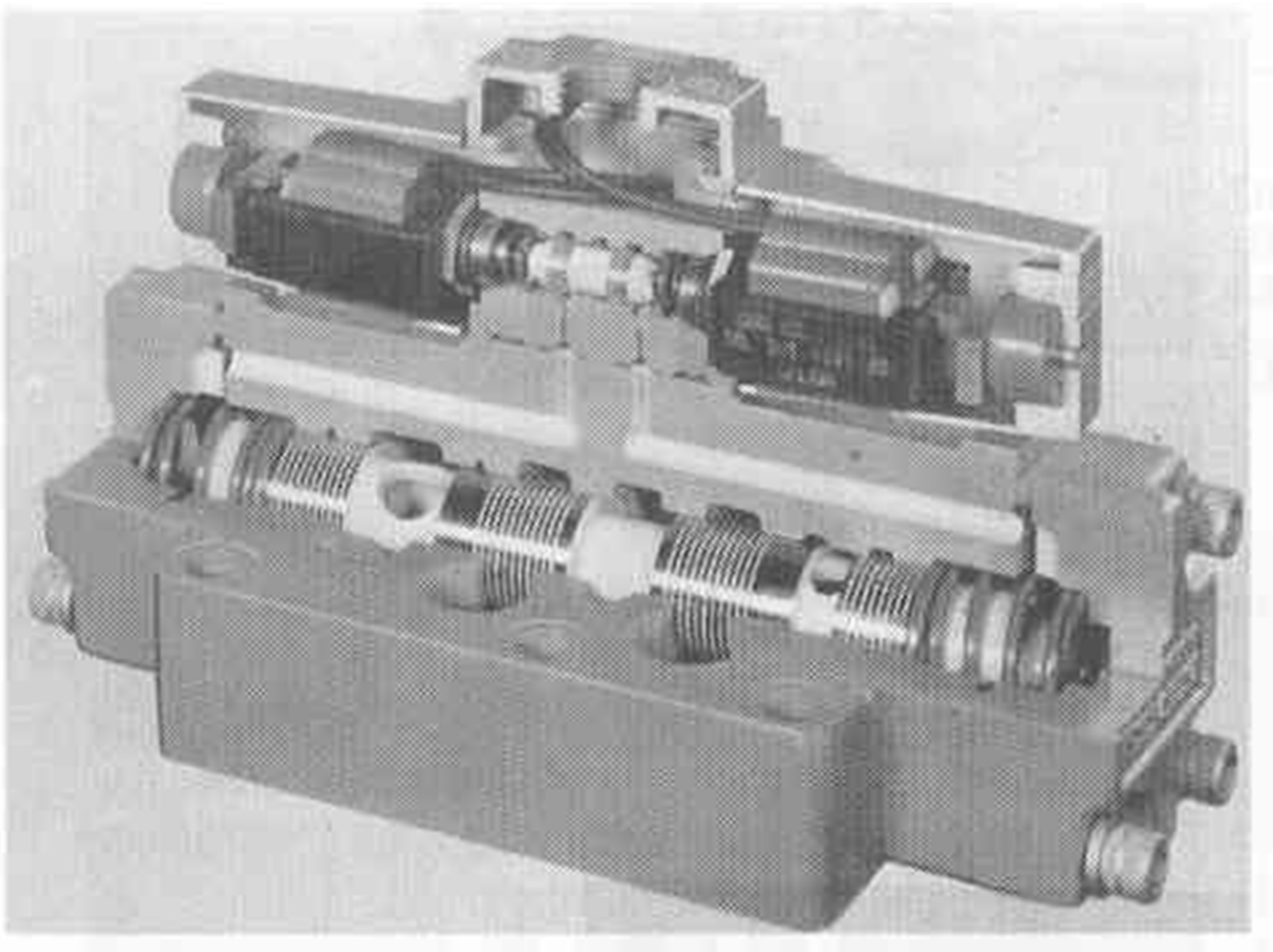


Schematic of a two-stage electrohydraulic

servovalve with force feedback controlling

a motor with inertia load

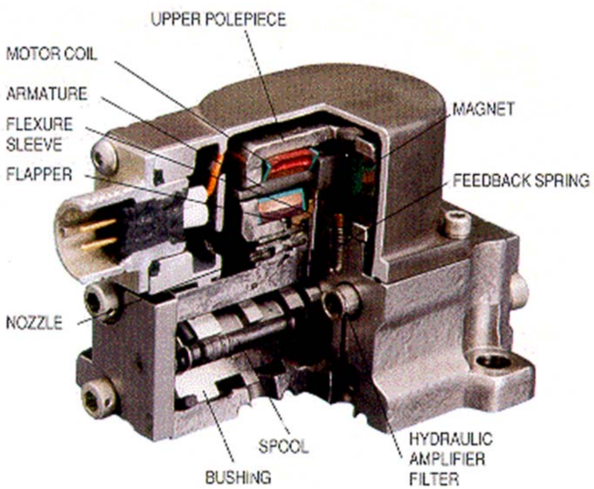
Solenoid-controlled, Pilot-operated Valve

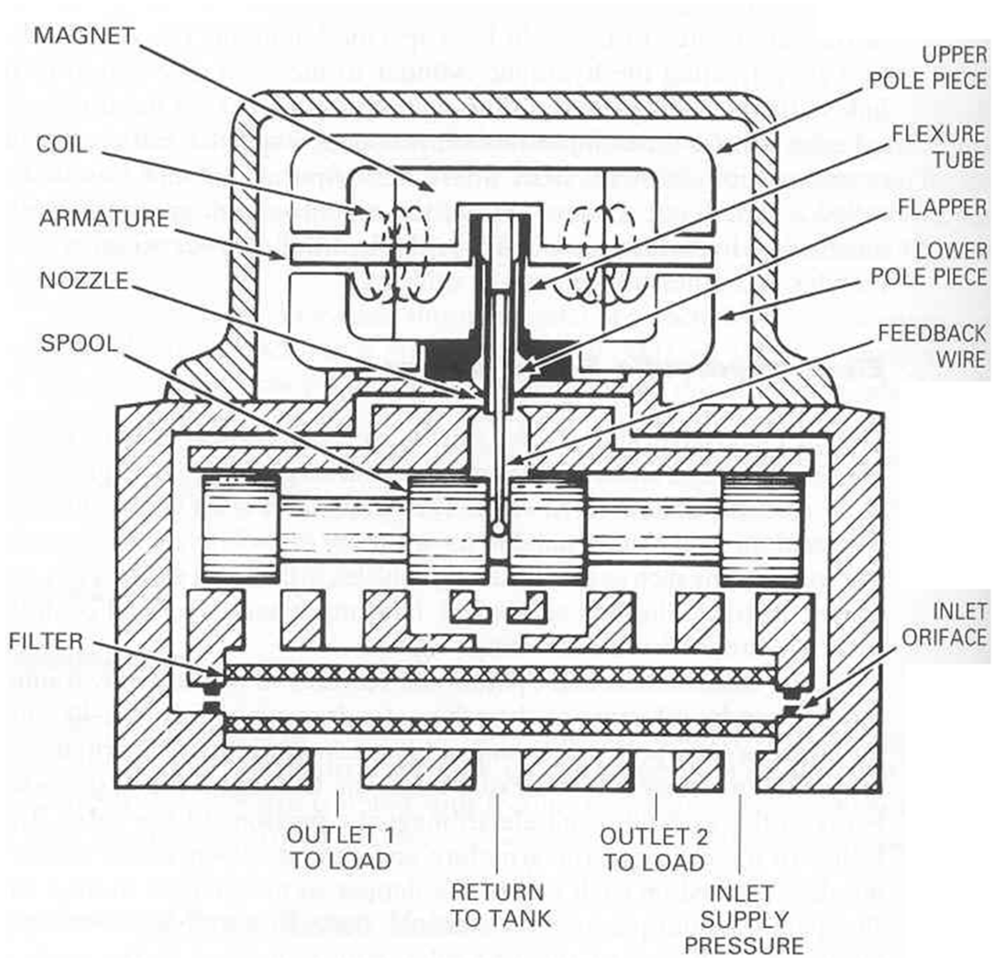


Servo Valve Structure

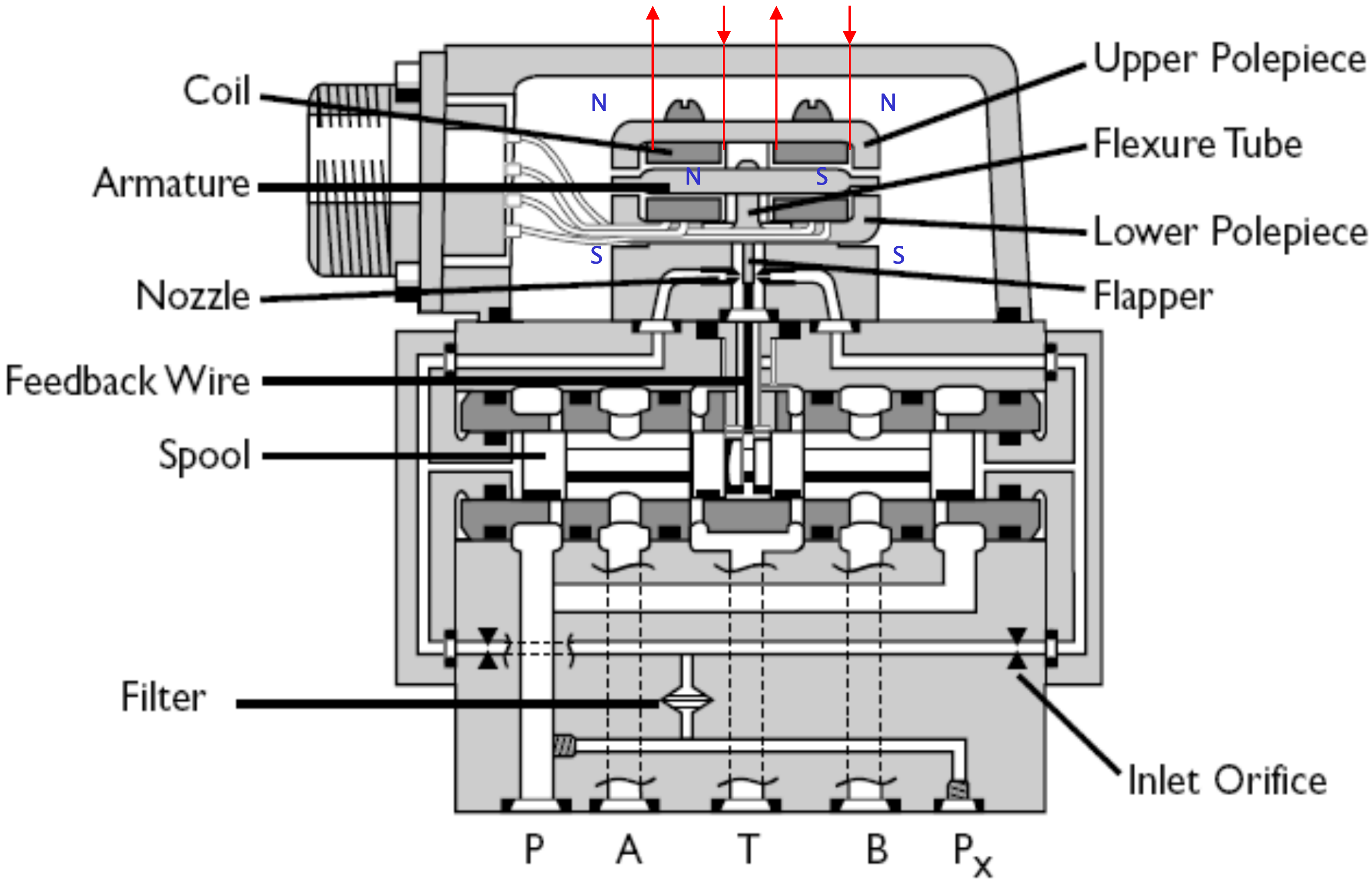
❖Moog 760 Series

❖Moog 30 Series



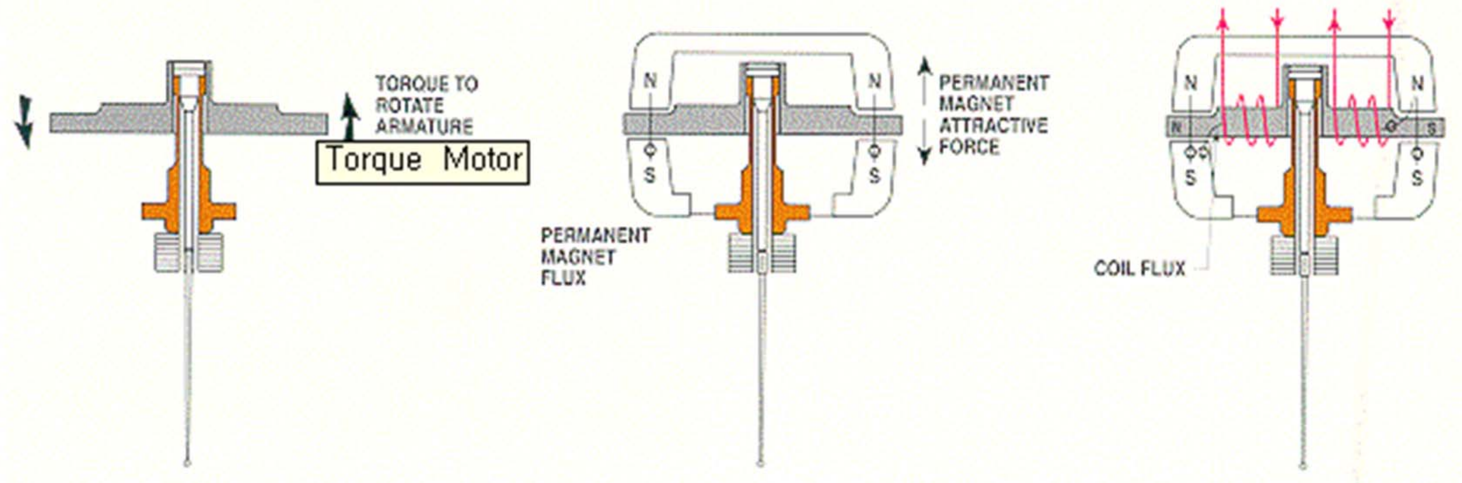


Operation of Servo Valve

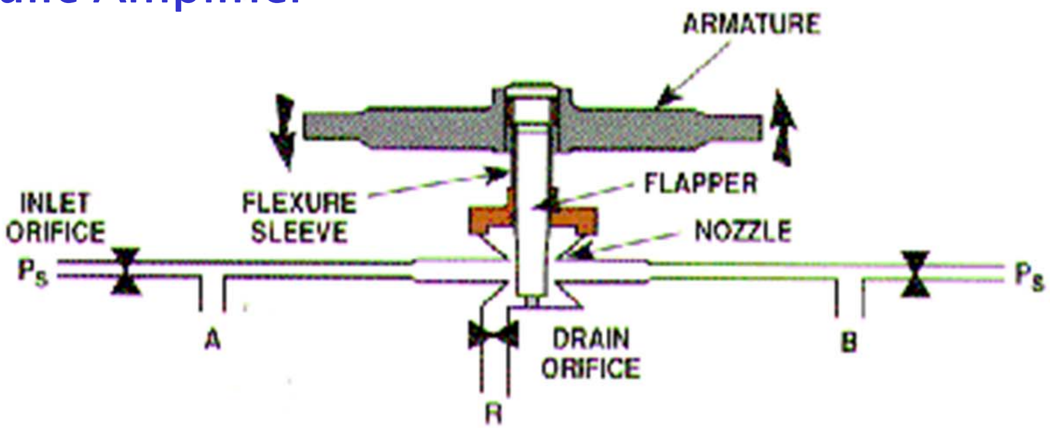


Operation of Servo Valve: Torque Motor

❖Torque Motor

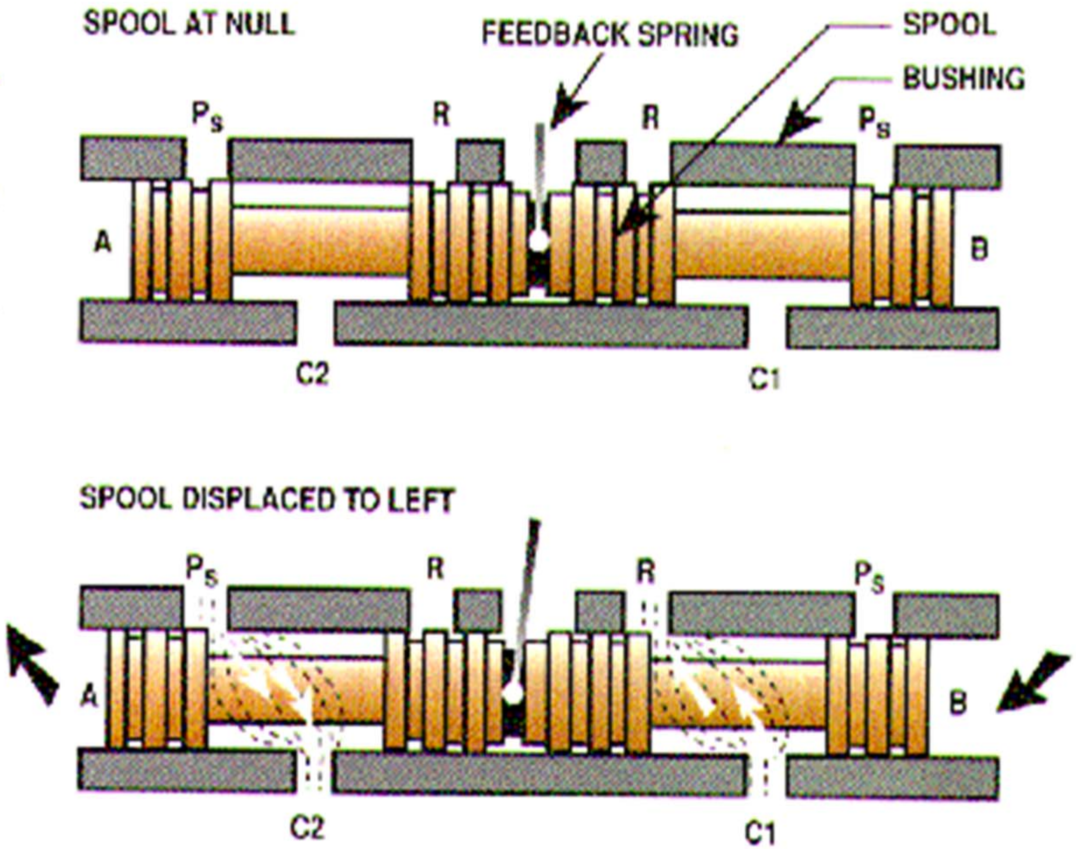


❖Hydraulic Amplifier



Operation of Servo Valve: Valve Spool

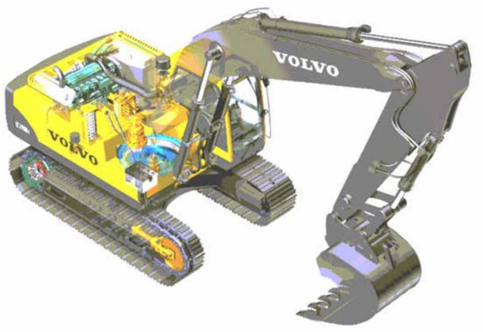
❖Valve Spool

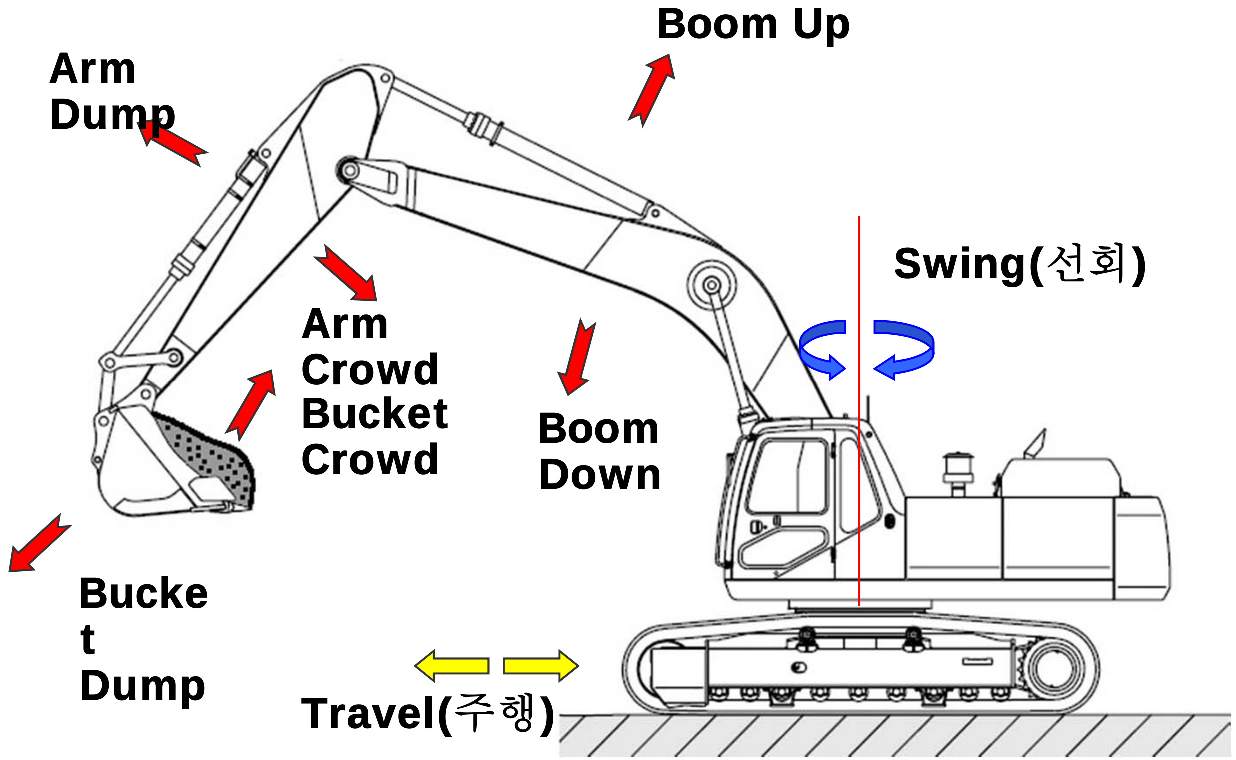


Valve Spool Schematic

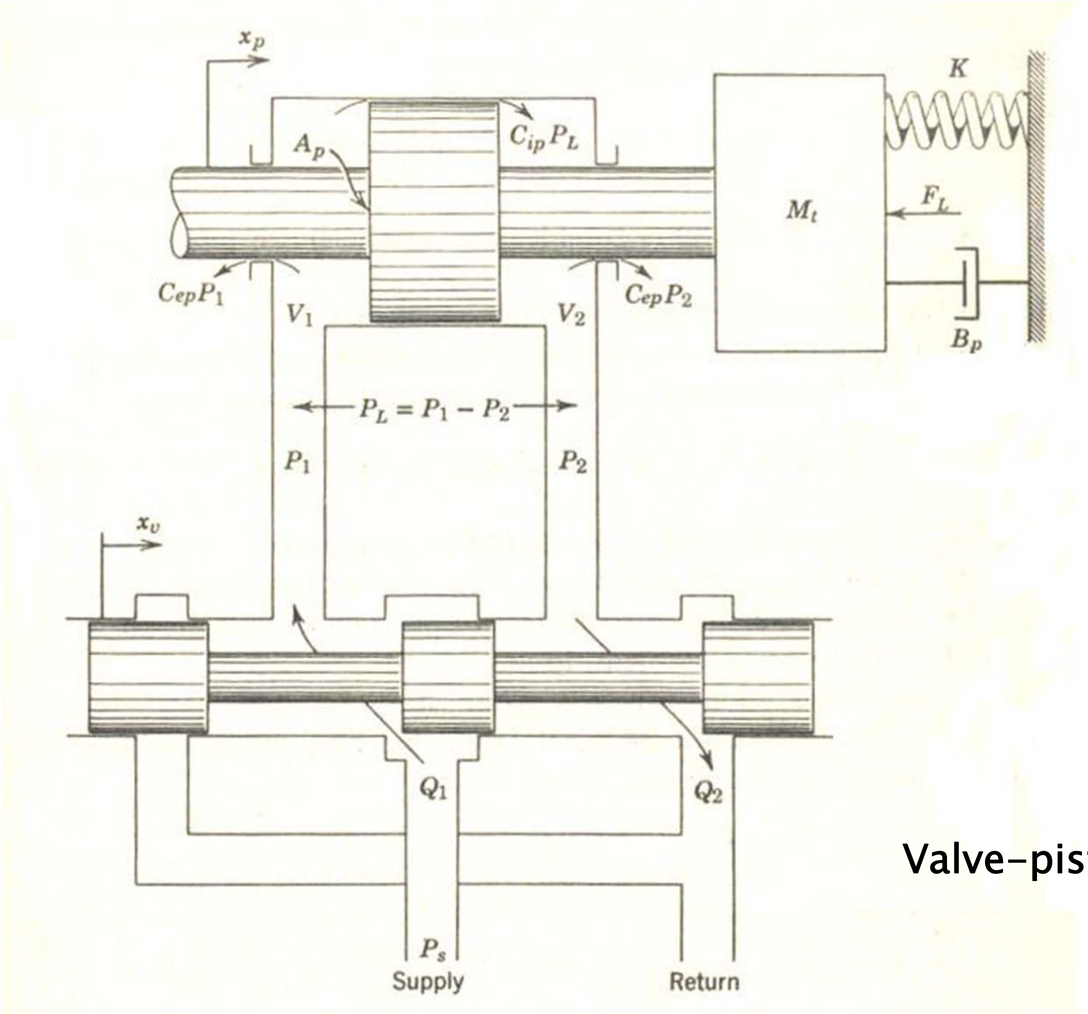
Hydraulic Servo Systems

Hydraulic Excavator



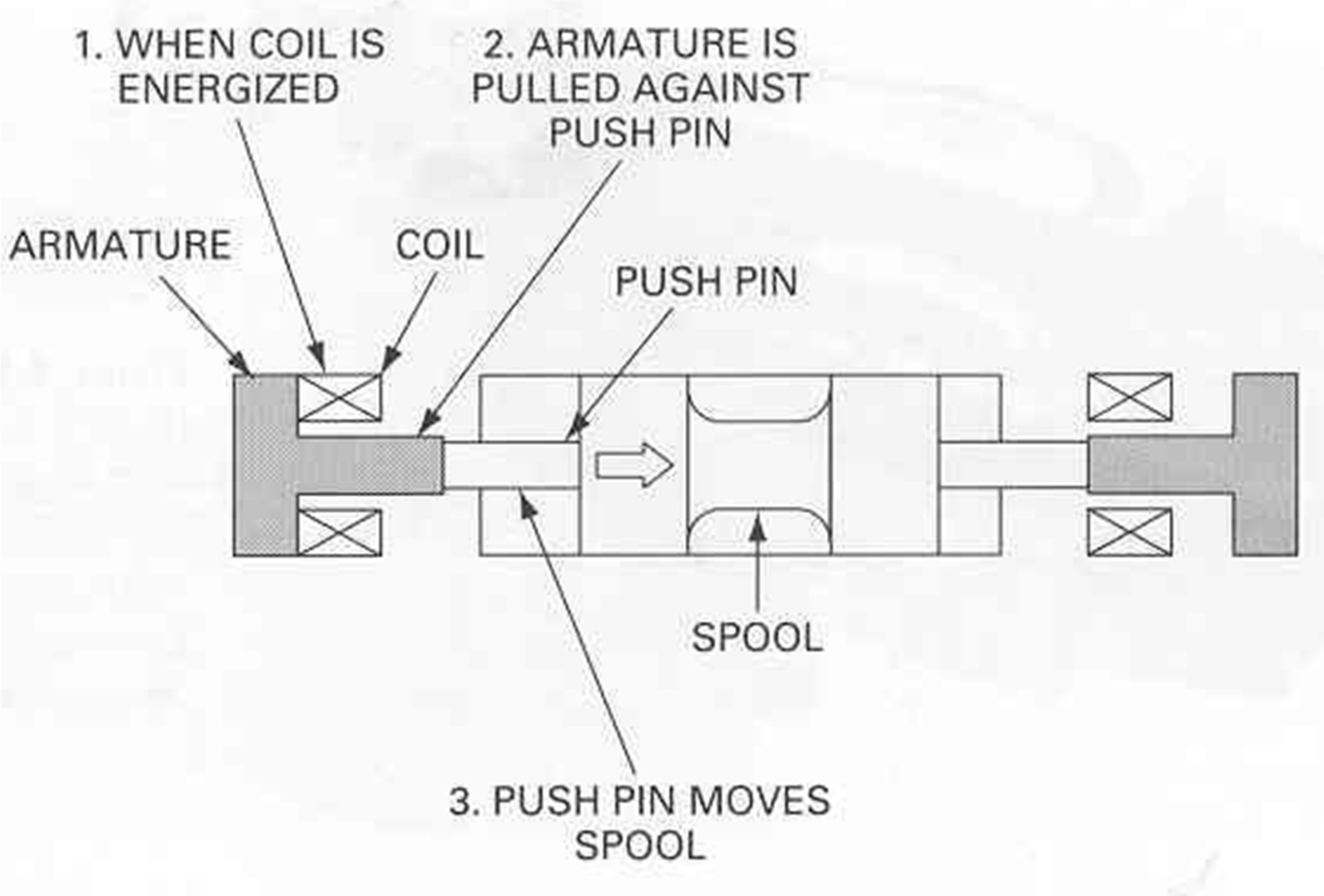


Hydraulic Systems : Valve-piston Combination

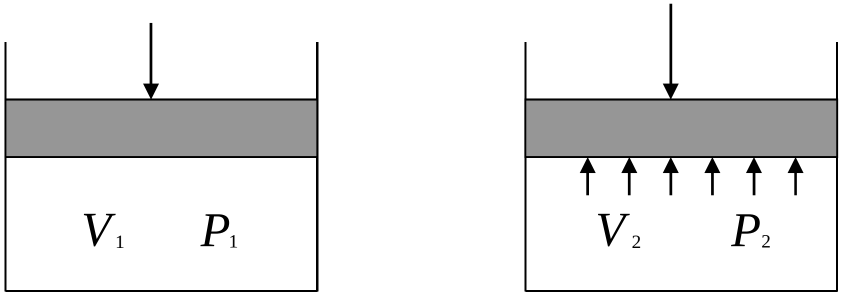


Valve-piston combination

Operation of Solenoid to Shift of Valve



Hydraulic Servo System : Compressibility



$$\frac{\Delta V}{V_1} = \beta \, \Delta P$$

$where, \;\; \beta = Compressibility$

$$PV = mRT$$

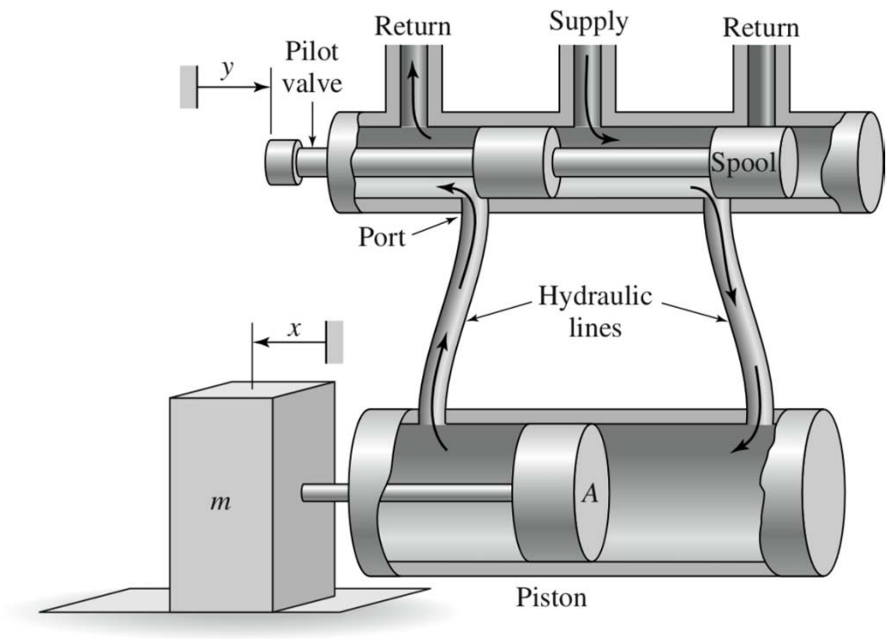
$$\frac{1}{\beta} = \frac{\Delta P}{\Delta V / V}$$

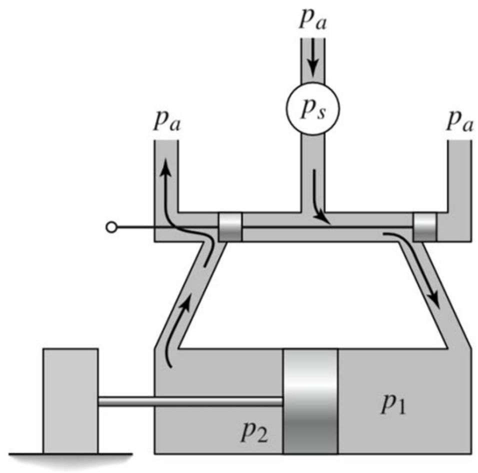
$$\Delta P = dP \;\;, \qquad \Delta V = V_1 - V_2 = -(V_2 - V_1) = -dV$$

$$\frac{1}{\beta} = -V\frac{dP}{dV} = K_B \qquad ;\ Bulk \;\; modulus$$

$$dP = -\frac{1}{\beta}\cdot\frac{1}{V}dV = -K_B\frac{1}{V}dV$$

$$\frac{dP}{dt} = -K_B\cdot\frac{1}{V}\cdot\frac{dV}{dt}$$


Return
Supply
Return
Pilot
valve
y
Spool
Port
Hydraulic
lines
x
m
A
Piston


p_a
p_s
p_a
p_a
p_1
p_2

Basic Modeling of Dynamic Cylinder

- **Generalized Flow - Continuity equation**

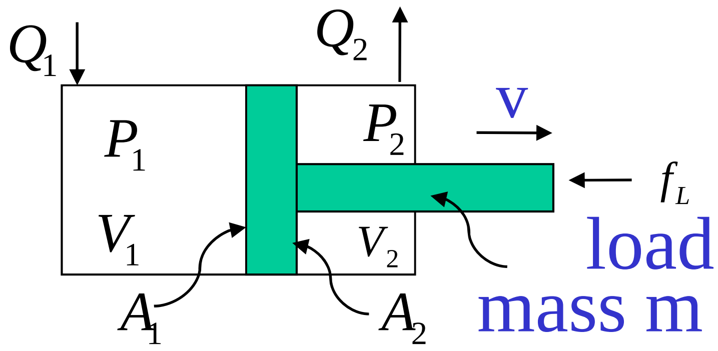
$$Q_1 - 0 = \frac{dV_1}{dt} + \frac{V_1}{\beta_e}\frac{dP_1}{dt}$$

$$0 - Q_2 = \frac{dV_2}{dt} + \frac{V_2}{\beta_e}\frac{dP_2}{dt}$$

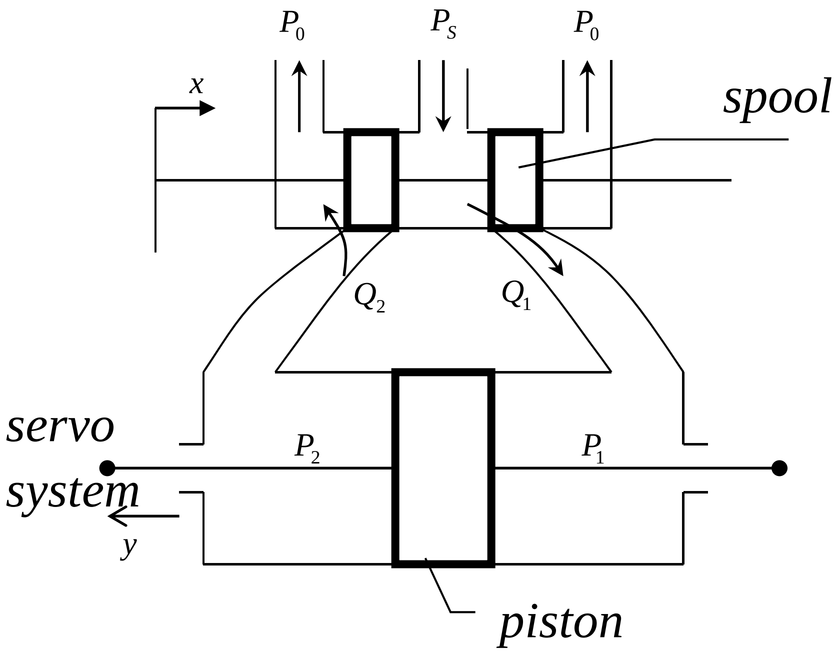
$$Q_1 = A_1 u + \frac{V_1}{\beta_e}\frac{dP_1}{dt} \qquad -Q_2 = -A_2 u + \frac{V_2}{\beta_e}\frac{dP_2}{dt}$$

- **Equation of motion**

$$P_1 A_1 - P_2 A_2 = m\frac{dv}{dt} + bv + f_L$$



Hydraulic Servo System



P_S : *supply pressure*

$$Q_1 = C_d \cdot a \cdot x\sqrt{\frac{2}{\rho}(P_S - P_1)} \qquad \left[m^2/s\right]$$

a : *area gradient*, x : *displacement*

ρ : *density*, C_d : *discharge coefficient*

$$Q_2 = C_d \cdot a \cdot x\sqrt{\frac{2}{\rho}(P_2 - P_0)}$$

$$= C_d \cdot a \cdot x\sqrt{\frac{2}{\rho}P_2} \qquad (P_0 \approx 0)$$

Hydraulic Servo System

no leakage, no compressibility

$$Q_1 = Q_2 \quad \rightarrow \quad P_s - P_1 = P_2 \quad \rightarrow \quad P_s = P_1 + P_2$$

$$P_L = \Delta P = P_1 - P_2 \quad \rightarrow \quad P_S + P_L = 2P_1, \quad P_S - P_L = 2P_2$$

$$\rightarrow \quad P_1 = \frac{P_S + P_L}{2}, \quad P_2 = \frac{P_S - P_L}{2}$$

$$Q = Q_1 = Q_2 = C_d \cdot a \cdot x\sqrt{\frac{2}{\rho}\frac{P_S - P_L}{2}} = C \cdot x\sqrt{P_S - P_L}$$

$$Q = A_P \cdot \frac{dy}{dt} = C \cdot x\sqrt{P_S - P_L}$$

$$\frac{dy}{dt} = C \cdot x\sqrt{P_S - P_L}$$

Hydraulic Servo System

$$\frac{d\bar{y}}{dt} = C \cdot \bar{x}\sqrt{P_S - \overline{P_L}}, \qquad y = \bar{y} + \Delta y, \qquad x = \bar{x} + \Delta x, \qquad P_L = \overline{P_L} + \Delta P_L$$

$$\frac{dy}{dt} = f(\bar{x}, \overline{P_L}) + \frac{\partial f}{\partial x}\Big|_{\bar{x}\ \overline{P_L}} \cdot (x - \bar{x}) + \frac{\partial f}{\partial P_L}\Big|_{\bar{x}\ \overline{P_L}} \cdot (P_L - \overline{P_L})$$

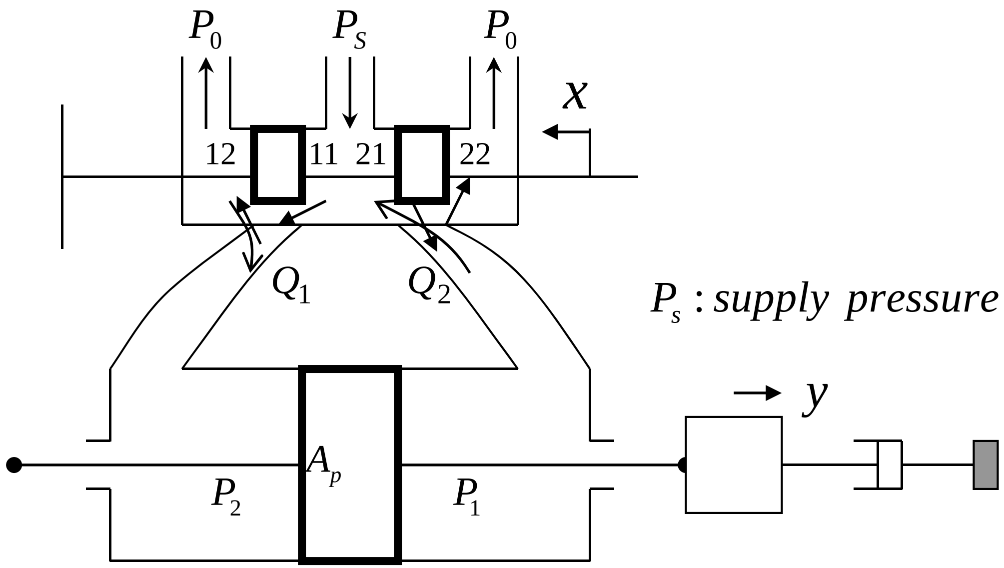
$$= \frac{d\bar{y}}{dt} + C\sqrt{P_S - \overline{P_L}} \cdot (x - \bar{x}) + \left(-\frac{1}{2}C\bar{x}\frac{1}{\sqrt{P_S - \overline{P_L}}}\right) \cdot (P_L - \overline{P_L})$$

$$if \quad \bar{x} = 0, \qquad \overline{P_L} = 0, \qquad \frac{d\bar{y}}{dt} = 0$$

$$\frac{dy}{dt} = C\sqrt{P_S} \cdot x = K_1 \cdot x$$

$$\therefore\ T.F = \frac{Y(s)}{X(s)} = \frac{K_1}{S}$$

Hydraulic Servo System



$x = 0\ , \qquad A = A_0$

Flow equations :

$$Q_{11} = C_d\,(A_0 + ax)\sqrt{\frac{2}{\rho}(P_S - P_1)}$$

$$Q_{12} = C_d\,(A_0 - ax)\sqrt{\frac{2}{\rho}(P_1 - 0)}$$

$$Q_1 = Q_{11} - Q_{12}$$

$$Q_{21} = C_d\,(A_0 - ax)\sqrt{\frac{2}{\rho}(P_S - P_2)}$$

$$Q_{22} = C_d\,(A_0 + ax)\sqrt{\frac{2}{\rho}(P_2 - 0)}$$

$$Q_2 = Q_{22} - Q_{21}$$

A_p

Hydraulic Servo System

Assume no leakage $\quad Q_1 = Q_2$

$$\dot{y} = 0,$$

$$\frac{dP}{dt} = -\frac{1}{\beta} \cdot \frac{1}{V} \cdot \frac{dV}{dt}$$

$$= -\frac{1}{\beta}\frac{1}{V_1}(-Q_1)$$

$$\dot{y} \neq 0,$$

$$\frac{dp_1}{dt} = \frac{1}{\beta}\frac{1}{V_1}(Q_1 - A_p\dot{y}), \qquad \frac{dp_2}{dt} = \frac{1}{\beta}\frac{1}{V_2}(-Q_2 + A_P\dot{y})$$

Equation of motion : $\quad m\ddot{y} = A_p(p_1 - p_2) - b\dot{y}$

$$m\ddot{y} + b\dot{y} = A_p(p_1 - p_2)$$

Hydraulic Servo System Model

$$m\ddot{y} + b\dot{y} = A_p(p_1 - p_2)$$

$$\frac{dp_1}{dt} = \frac{1}{\beta}\frac{1}{V_1}(Q_1 - A_p\dot{y})$$

$$\frac{dp_2}{dt} = \frac{1}{\beta}\frac{1}{V_2}(-Q_2 + A_P\dot{y})$$

$$Q_1 = Q_{11} - Q_{12} = \left\{C_d\,(A_0 + ax)\sqrt{\frac{2}{\rho}(P_S - P_1)}\right\} - \left\{C_d\,(A_0 - ax)\sqrt{\frac{2}{\rho}(P_1 - 0)}\right\}$$

$$Q_2 = Q_{22} - Q_{21} = \left\{C_d\,(A_0 + ax)\sqrt{\frac{2}{\rho}(P_2 - 0)}\right\} - \left\{C_d\,(A_0 - ax)\sqrt{\frac{2}{\rho}(P_S - P_2)}\right\}$$

$$Q_1 = Q_2$$

Hydraulic Servo System : Linearization

$$Q_1 = Q_2 \qquad \Rightarrow \qquad Q_1 - Q_2 = 0$$

$$C_d(A_0 + ax)\left\{\sqrt{\frac{2}{\rho}(P_S - P_1)} - \sqrt{\frac{2}{\rho}P_2}\right\} - C_d(A_0 - ax)\left\{\sqrt{\frac{2}{\rho}P_1} - \sqrt{\frac{2}{\rho}(P_S - P_2)}\right\}$$

$when \;\; x = 0,$

$$C_d ax\left\{\sqrt{\frac{2}{\rho}(P_S - P_1)} - \sqrt{\frac{2}{\rho}P_2} + \sqrt{\frac{2}{\rho}P_1} - \sqrt{\frac{2}{\rho}(P_S - P_2)}\right\}$$

$$+ C_d A_0\left\{\sqrt{\frac{2}{\rho}(P_S - P_1)} - \sqrt{\frac{2}{\rho}P_2} - \sqrt{\frac{2}{\rho}P_1} + \sqrt{\frac{2}{\rho}(P_S - P_2)}\right\} = 0$$

To make an identical equation, $P_s - P_1 = P_2, \quad P_1 = P_s - P_2 \quad \Rightarrow \quad P_s = P_1 + P_2$

let $P_L = P_1 + P_2, \qquad \Rightarrow \qquad P_1 = \dfrac{P_s + P_L}{2}, \qquad P_2 = \dfrac{P_s - P_L}{2}$

Hydraulic Servo System : Linearization

$$Q_1 = C_d(ax + A_0)\sqrt{\frac{1}{\rho}(P_s - P_L)} - C_d(A_0 - ax)\sqrt{\frac{1}{\rho}(P_s + P_L)}$$

$$Q_L = Q_L(x, P_L)$$

Operating point : $x = 0, \quad p_L = 0$

$$Q_1(x, p_L) = Q_1(0,0) + \frac{\partial Q_1}{\partial x}\bigg|_{x=0,\, p_L=0}(x - 0) + \frac{\partial Q_1}{\partial p_L}\bigg|_{x=0,\, p_L=0}(p_L - 0) + \cdots$$

$$\frac{\partial Q_1}{\partial x}\bigg|_{x=0,\, p_L=0} = 2C_d \cdot a\sqrt{\frac{1}{\rho}p_S} = K_1$$

$$\frac{\partial Q_1}{\partial p_L}\bigg|_{x=0,\, p_L=0} = -C_d \cdot A_0 \frac{1}{\sqrt{\rho \cdot p_S}} = -K_2$$

$$Q = K_1 x - K_2 p_L = Q_2 = Q_L$$

$$p_L = p_1 - p_2, \qquad \frac{dp_L}{dt} = \frac{dp_1}{dt} - \frac{dp_2}{dt}$$

Hydraulic Servo System

$$m\ddot{y} + b\dot{y} = A_p p_L$$

$$\frac{dp_1}{dt} = \frac{1}{\beta}\frac{1}{V_1}(Q_L - A_p\dot{y}) = \frac{1}{\beta}\frac{1}{V_1}(K_1 x - K_2 p_L - A_p\dot{y})$$

$$\frac{dp_2}{dt} = \frac{1}{\beta}\frac{1}{V_2}\left\{-(K_1 x - K_2 p_L) + A_p\dot{y}\right\}$$

$$let\ \ V_1 = V_2$$

$$\frac{dp_L}{dt} = \frac{1}{\beta}\frac{1}{V}(2K_1 x - 2K_2 p_L - 2A_p\dot{y})$$

$$\therefore \frac{Y(s)}{X(s)} = \frac{(\ \ )}{(\ \ )} \qquad (\ \ ): cubic\ equation\ \ form$$

Hydraulic Servo System

Simplification : No compressibility, No leakage

$$m\ddot{y} + b\dot{y} = p_L A_p$$

$$Q_L = K_1 x - K_2 p_L = A_p \dot{y} \qquad \Rightarrow \qquad p_L = \frac{1}{K_2}(K_1 x - A_p \dot{y})$$

$$\Rightarrow \quad m\ddot{y} + \left(b + \frac{A_p{}^2}{K_2} \right)\dot{y} = A_p \frac{K_1}{K_2} x$$

$$\therefore \frac{Y(s)}{X(s)} = \frac{K}{s(Ts+1)}$$

$$K = \frac{K_1 A_p}{K_2 b + A_p{}^2}, \quad T = \frac{mK_2}{K_2 b + A_p{}^2}$$

End of Hydraulic systems 8-1