Hydraulic Servo and Related Systems

ME4803 Motion Control

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Hydraulics is Especially critical to the Mobile Equipment Industry

References

1. Norvelle, F.D. *Fluid Power Control Systems*, Prentice Hall, 2000.
2. Fitch, E.C. and Hong I.T. *Hydraulic Component Design and Selection*, BarDyne, Stillwater, OK, 2001.
3. Cundiff, J.S. *Fluid Power Circuits and Controls*, CRC Press, Boca Raton, FL, 2002.
4. Merritt, H.E. *Hydraulic Control Systems*, John Wiley and Sons, New York, 1967.
5. *Fluid Power Design Engineers Handbook*, Parker Hannifin Company (various editions).

The Strengths of Fluid Power
(Hydraulic, to a lesser extent pneumatic)

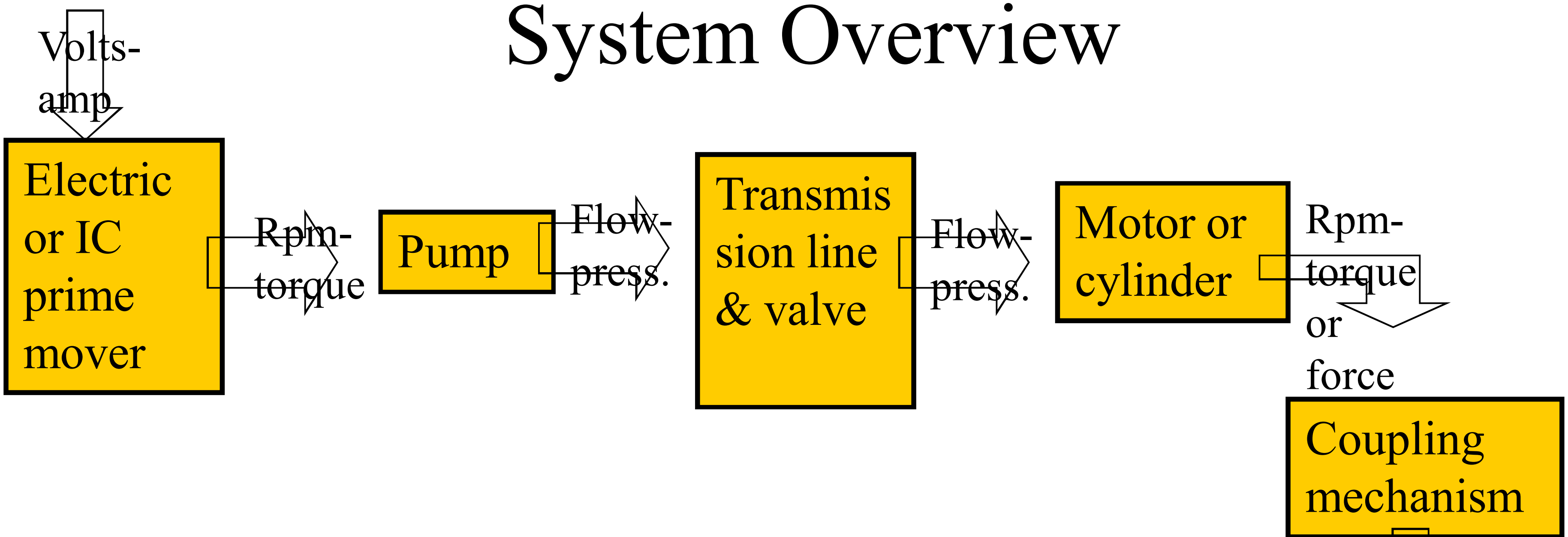
- High force at moderate speed
- High power density at point of action
  - Fluid removes waste heat
  - Prime mover is removed from point of action
  - Conditioned power can be routed in flexible a fashion
- Potentially “Stiff” position control
- Controllable either electrically or manually
  - Resulting high bandwidth motion control at high forces

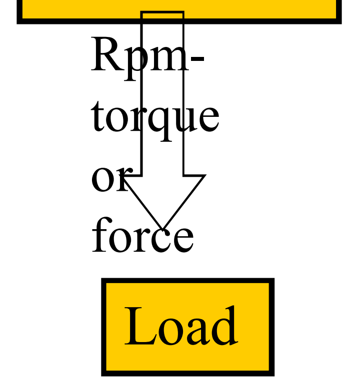
- NO SUBSTITUTE FOR MANY HEAVY APPLICATIONS

Difficulties with Fluid Power

- Possible leakage
- Noise generated by pumps and transmitted by lines
- Energy loss due to fluid flows
- Expensive in some applications
- Susceptibility of working fluid to contamination
- Lack of understanding of recently graduated practicing engineers
  - Multidisciplinary
  - Cost of laboratories
  - Displaced in curriculum by more recent technologies

System Overview





- The system consists of a series of transformation of power variables
- Power is either converted to another useful form or waste heat
- Impedance is modified (unit force/unit flow)
- Power is controlled
- Function is achieved

Simple open-loop open-center circuit

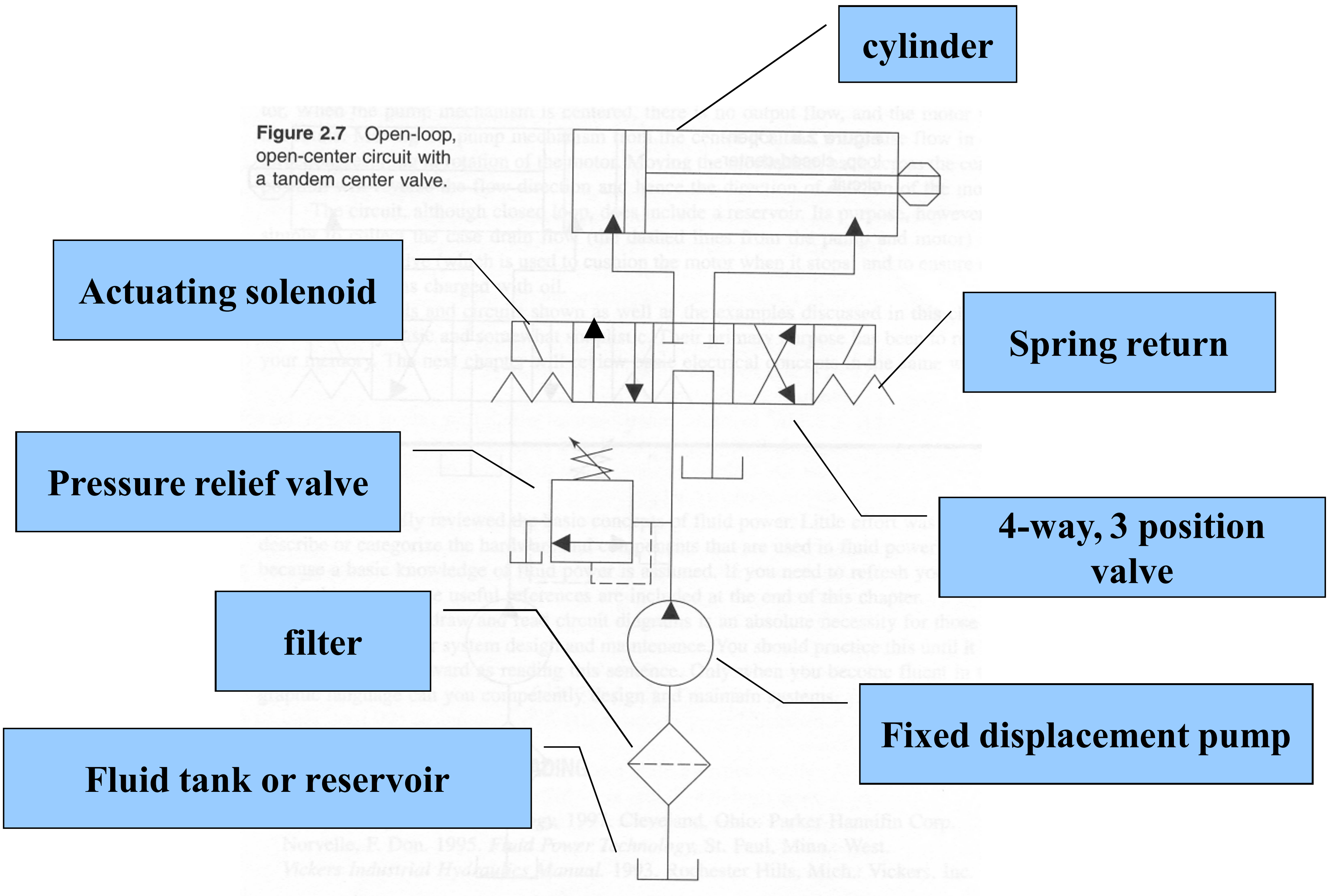


Figure 2.7 Open-loop, open-center circuit with a tandem center valve.

Simple open-loop closed-center circuit

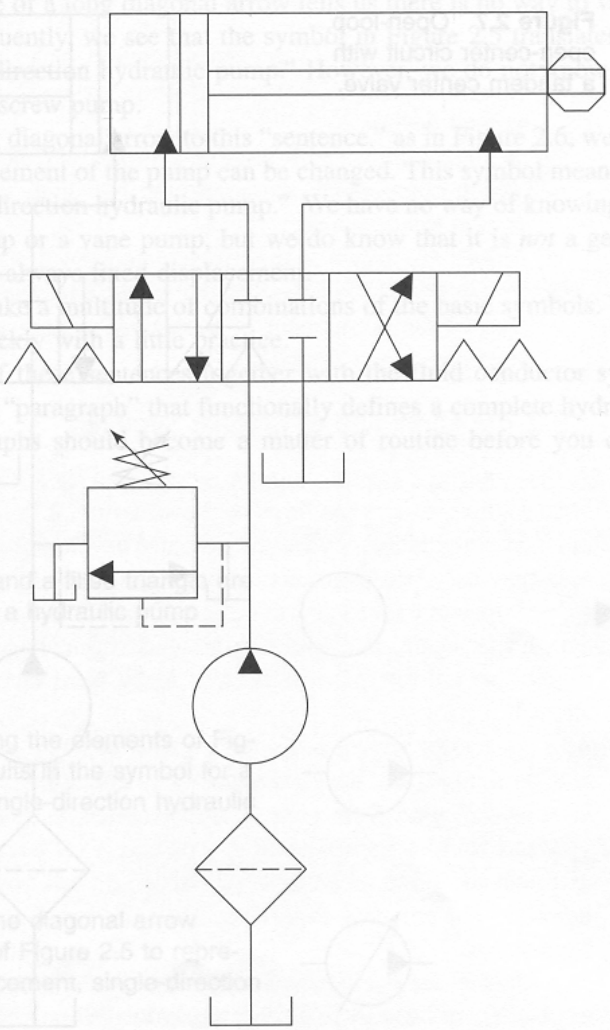


Figure 2.8 Open-loop, closed-center circuit.

Closed-loop (hydrostatic) system

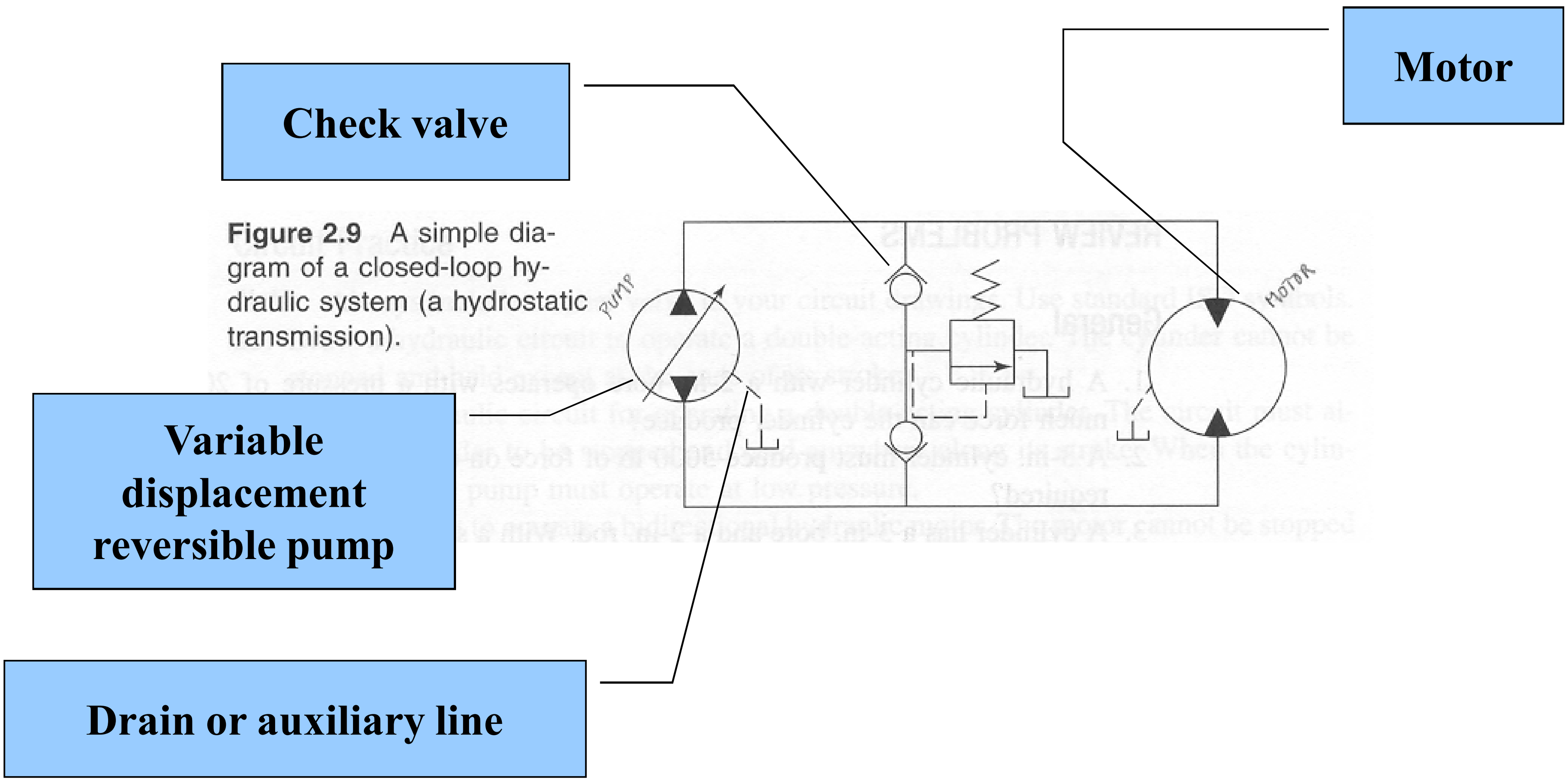
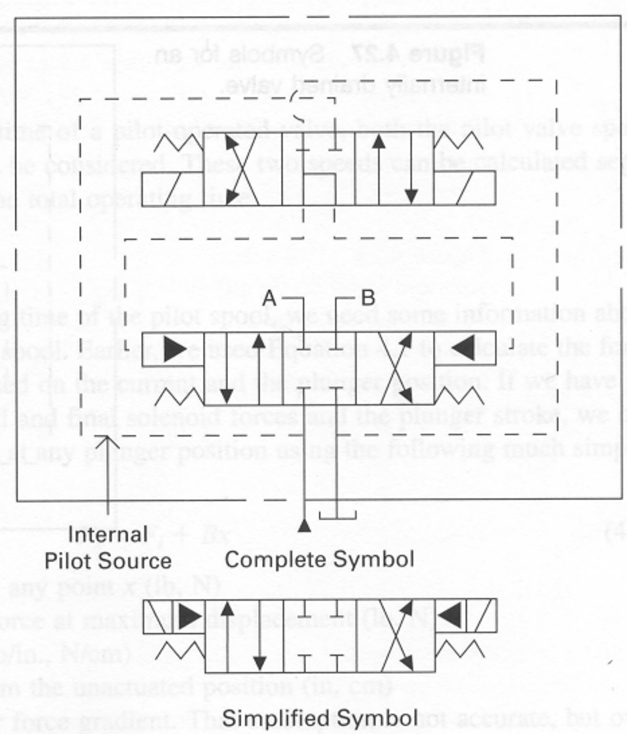


Figure 2.9 A simple diagram of a closed-loop hydraulic system (a hydrostatic transmission).

Pilot operated valve

Figure 4.25 Symbols for an internally piloted valve.



Proportional Valve

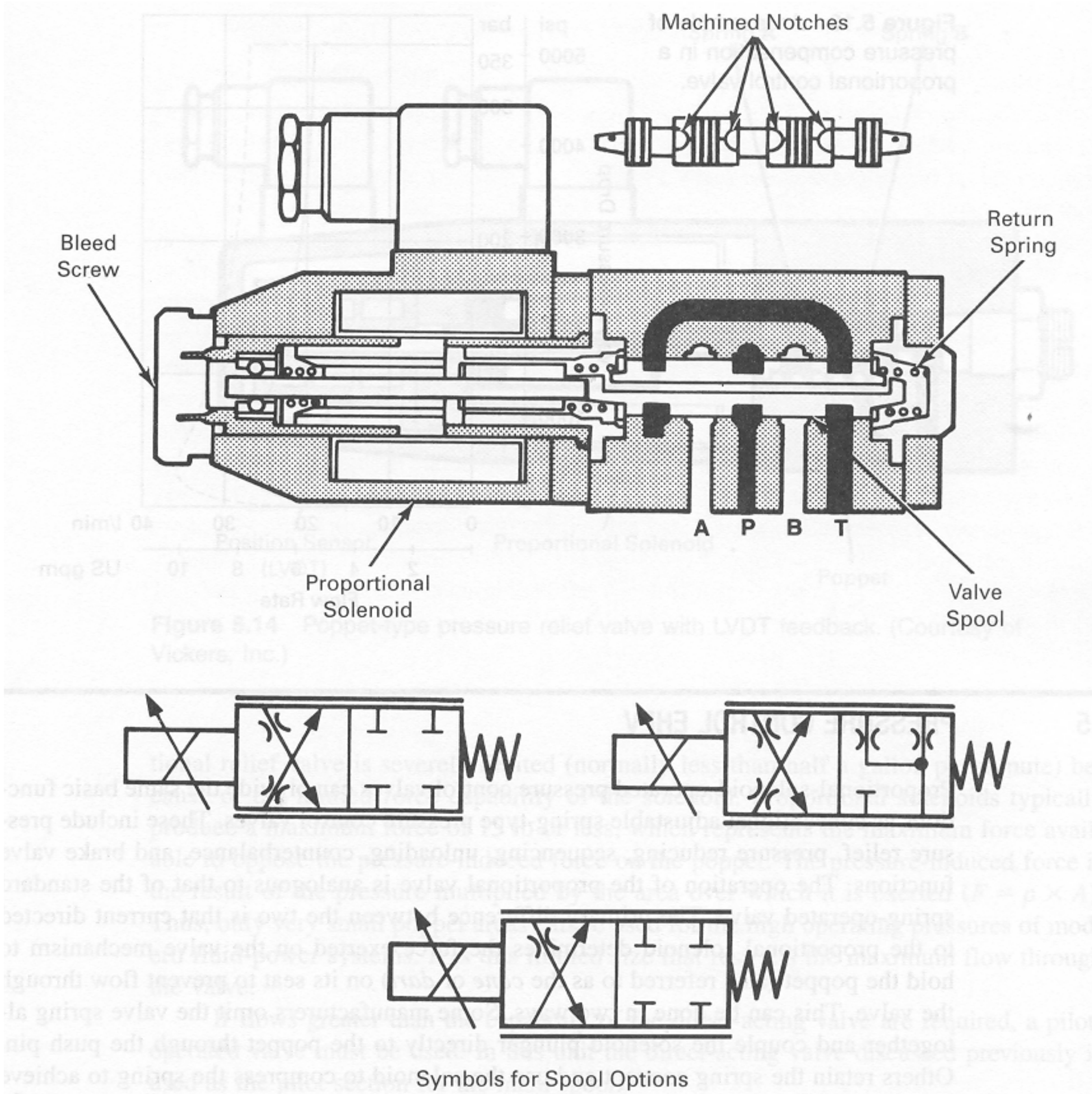


Figure 5.12 Cutaway drawing of a proportional throttle valve. (Courtesy of Vickers, Inc.)

Basic Operation of the Servo Valve
(single stage)

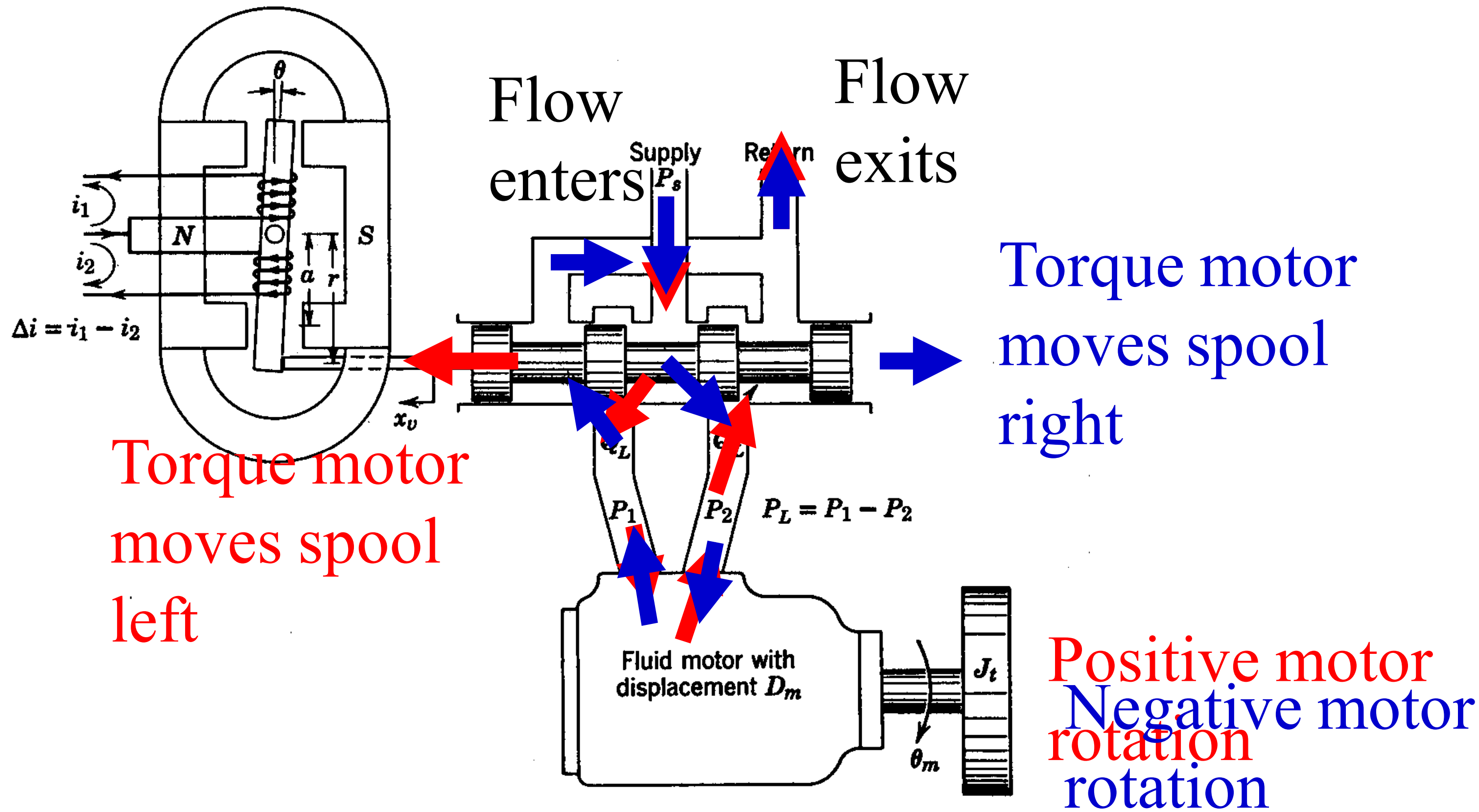


Figure 7-11 Schematic of a single stage electrohydraulic servovalve connected to a motor with inertia load.

Orifice Model

$$Q = C_d A_o \sqrt{\frac{2}{\rho} \Delta p}$$

C_d = orifice flow discharge coef.

A_o = orifice flow area = $w\,x$

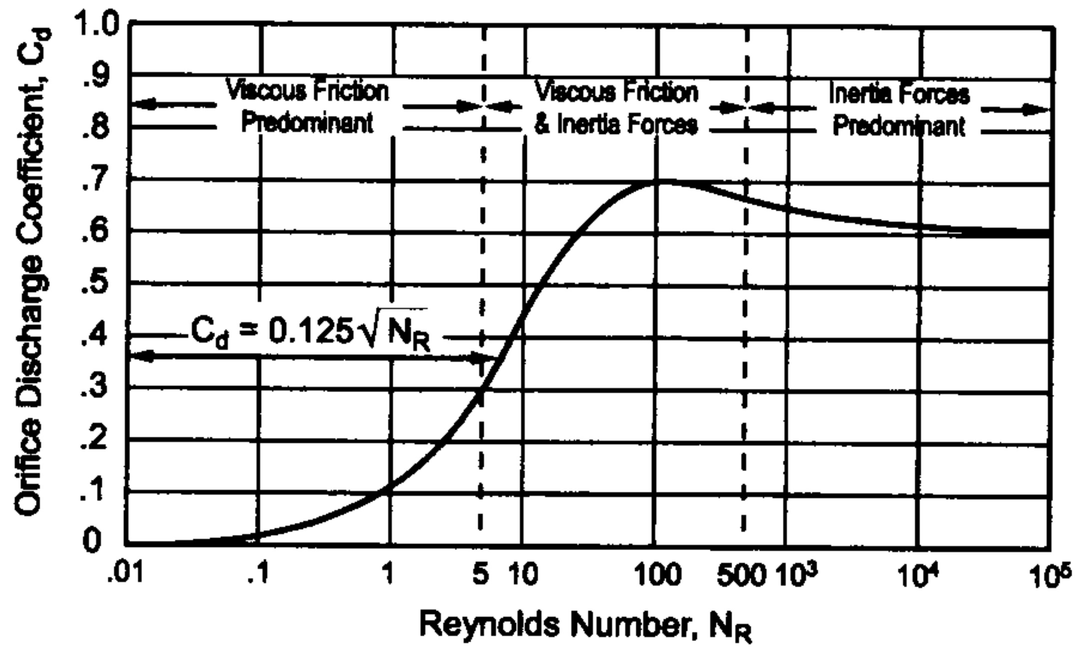


Figure 2-9. Orifice Discharge Coefficient versus Reynolds number.

4 Way Proportional Spool Valve Model

- Spool assumptions
  - No leakage, equal actuator areas
  - Sharp edged, steady flow
  - Opening area proportional to x
  - Symmetrical
  - Return pressure is zero
  - Zero overlap
- Fluid assumptions
  - Incompressible
  - Mass density ρ

$$q_1 = q_2$$

$$q_1 = C\sqrt{p_s - p_1}\,x, \quad C = \text{a constant}$$

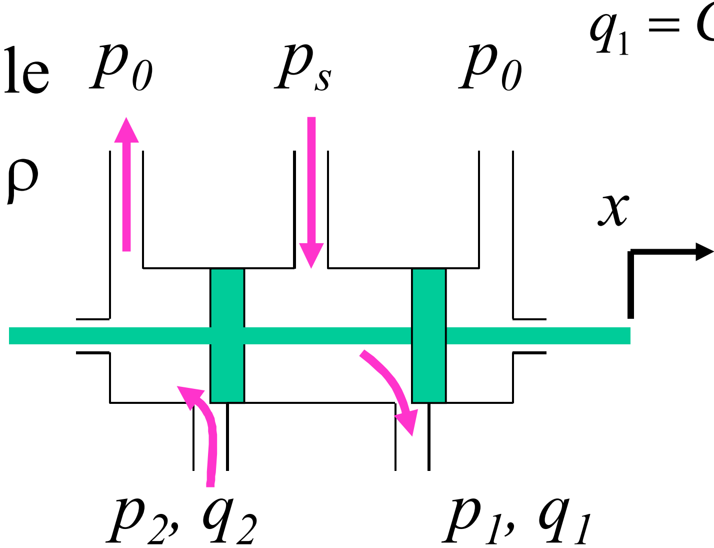
$$q_2 = C\sqrt{p_2 - p_0}\,x$$

$$p_s - p_1 = p_2 - p_0 = p_2$$

$$\text{Load pressure}: \ \Delta p = p_1 - p_2$$

$$\text{so}: \ p_1 = \frac{p_s + \Delta p}{2}; \quad p_2 = \frac{p_s - \Delta p}{2}$$

$$q_1 = C\sqrt{p_s - p_1}\,x = C\sqrt{\frac{p_s - \Delta p}{2}}\,x$$



Dynamic Equations (cont.)

Expand in a Taylor series to first order to linearize

$$q_1 = \bar{q}_1 + \left[\left. \frac{\partial q_1}{\partial x} \right|_{\substack{x=\bar{x} \\ \Delta p=\Delta\bar{p}}} \right](x - \bar{x}) + \left[\left. \frac{\partial q_1}{\partial \Delta p} \right|_{\substack{x=\bar{x} \\ \Delta p=\Delta\bar{p}}} \right](\Delta p - \Delta\bar{p}) + \cdots \quad \text{(high order terms)}$$

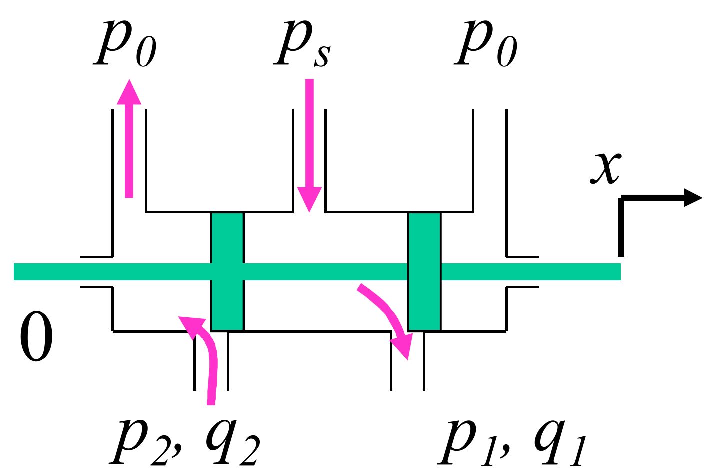
Taking partial derivatives :

$$\left. \frac{\partial q_1}{\partial x} \right|_{\substack{x=\bar{x} \\ \Delta p=\Delta\bar{p}}} = C\sqrt{\frac{p_s - \Delta\bar{p}}{2}}; \qquad \left. \frac{\partial q_1}{\partial \Delta p} \right|_{\substack{x=\bar{x} \\ \Delta p=\Delta\bar{p}}} = \frac{C}{2\sqrt{2}\sqrt{p_s - \Delta\bar{p}}}\,\bar{x}$$

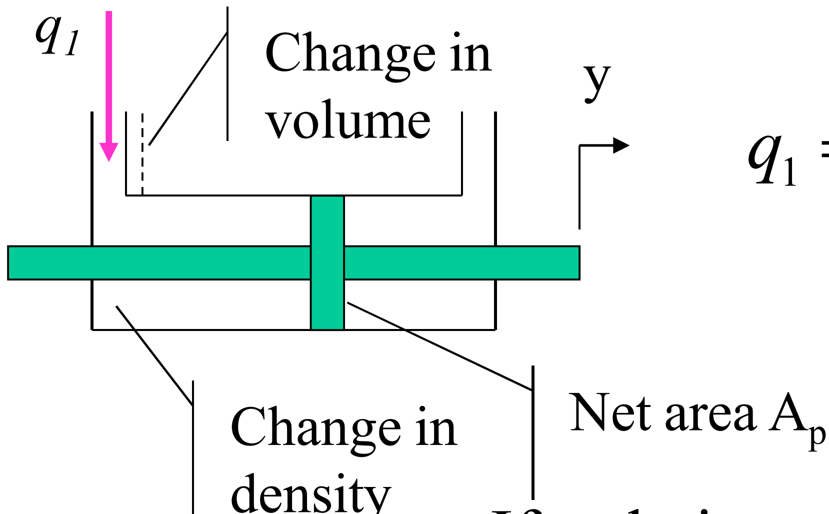
Choose operating point, commonly

$\bar{x} = 0; \quad \Delta\bar{p} = 0,$ at which $\bar{q}_1 = 0$

$$\left. \frac{\partial q_1}{\partial x} \right|_{\substack{x=\bar{x} \\ \Delta p=\Delta\bar{p}}} = C\sqrt{\frac{p_s}{2}} = K_1; \qquad \left. \frac{\partial q_1}{\partial \Delta p} \right|_{\substack{x=\bar{x} \\ \Delta p=\Delta\bar{p}}} = K_2 = 0$$



Dynamic Equations: the Actuator



$$q_1 = K_1 x + K_2 \Delta p = K_1 x = A\dot{y}$$

If truly incompressible:

•Specification of flow without a response in pressure brings a causality problem

•For example, if the piston has mass, and flow can change instantaneously, infinite force is required for infinite acceleration

•Need to account for change of density and compliance of walls of cylinder and tubes

Compressibility of Fluids and Elasticity of Walls

$$\text{Bulk modulus: } \beta = \rho \frac{\partial p}{\partial \rho} \quad \left[\frac{N}{m^2} \right]$$

$$\frac{1}{\beta} = \frac{1}{\rho} \frac{d\rho}{dp} = \frac{1}{\rho} \frac{d(M/V)}{dp}$$

For the pure definition, the volume is fixed.

$$dM = q\, dt; \quad dp = \left(\frac{\beta}{\rho V} \right) q\, dt$$

More useful here is an effective bulk modulus that includes expansion of the walls and compression of entrapped gasses

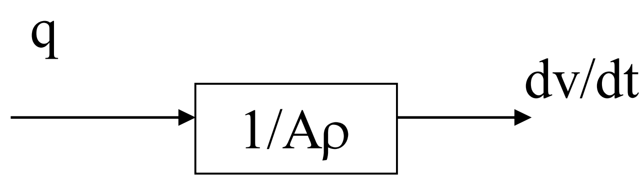
$$\frac{1}{\beta_{eff}} = \frac{1}{\rho} \frac{d(M/V)}{dp} = \frac{1}{\rho} \left[\frac{1}{V} \frac{\partial M}{\partial p} - \frac{M}{V^2} \frac{\partial V}{\partial p} \right]$$

Using this to solve for the change in pressure

$$dp = \left(\frac{\beta_{eff}}{\rho V} \right) dM = k \dot{M}\, dt = k\, q\, dt$$

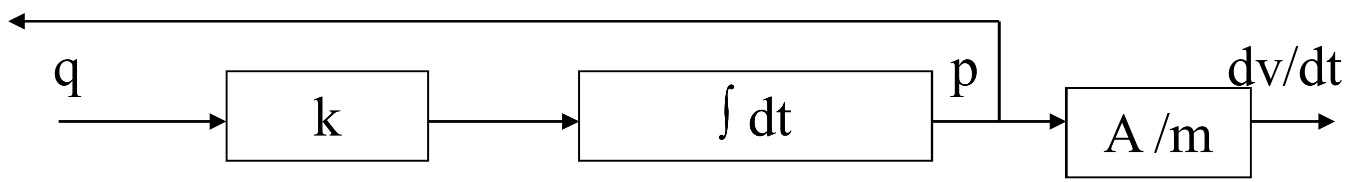
Choices for modeling the hydraulic actuator

With no compliance or compressibility we get actuator velocity v as



With compliance and/or compressibility combined into a factor k

And with moving mass m



Manufacturer's Data: BD15 Servovalve on HAL

Specifications

Rated Flow @ 1000 PSI ΔP	3.78–151 LPM (1.0 – 40 GPM)
Linearity	≤ 5%
Hysteresis	≤ 3%
Threshold	≤ 0.5%
Fluid	Mineral oil, 60–225 SSU, max. 1000 SSU
Oper. Temp. (Ambient)	–1 to 106°C (30 to +225°F)
Pressure Gain	3% of spool shift
Null Shift with Temperature with Supply Pressure	< ± 2% per 38°C (100°F) < 2% per 69 Bar (1000 PSI)
Quiescent Flow (Std. Spool Lap)	BD15 – 1.5–2.1 LPM (.40–.55 GPM) BD30 – 2.1–3.78 LPM (.55 – 1.0 GPM)
Step Response Input	Model / Typical Step Response Input: BD15 10 to 90%, 26 ms; BD30 10 to 90%, 30 ms
Pressure Ranges	For optimum performance, Parker Servo Valves are designed to operate within specific system supply pressure ranges. System Supply Pressure: 180–210 Bar (2600–3000 PSI); 138–172 Bar (2000–2500 PSI); 95–133 Bar (1400–1950 PSI); 68–90 Bar (1000–1300 PSI); 48–66 Bar (700–950 PSI); 14–45 Bar (200–650 PSI); 0–210 Bar (0–3000 PSI) External Pilot
Filtration	SAE Class 3 or better, ISO Code 15/12
Protection Class	NEMA 1 (IP54)

$$Q = K\sqrt{\Delta P}$$

Q = Control flow, cubic inches/sec
K = Valve constant
ΔP = Valve pressure drop

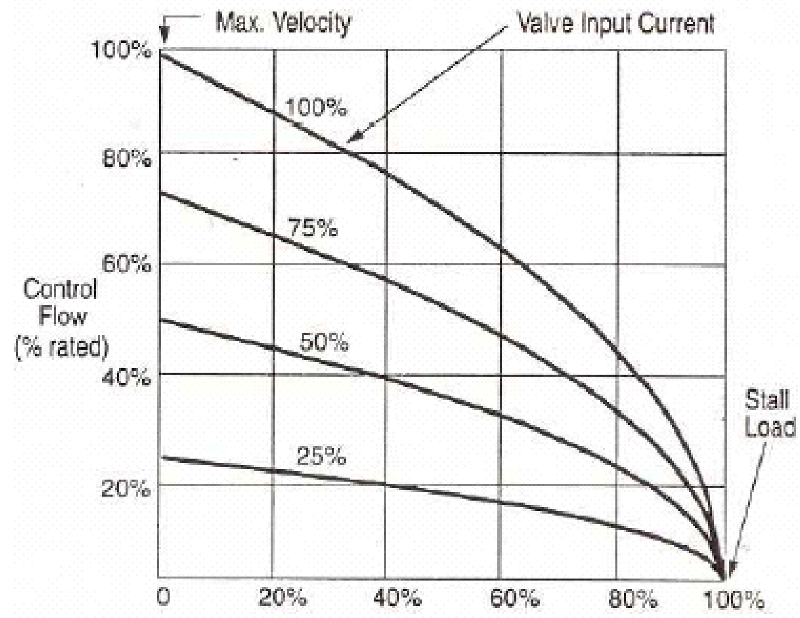
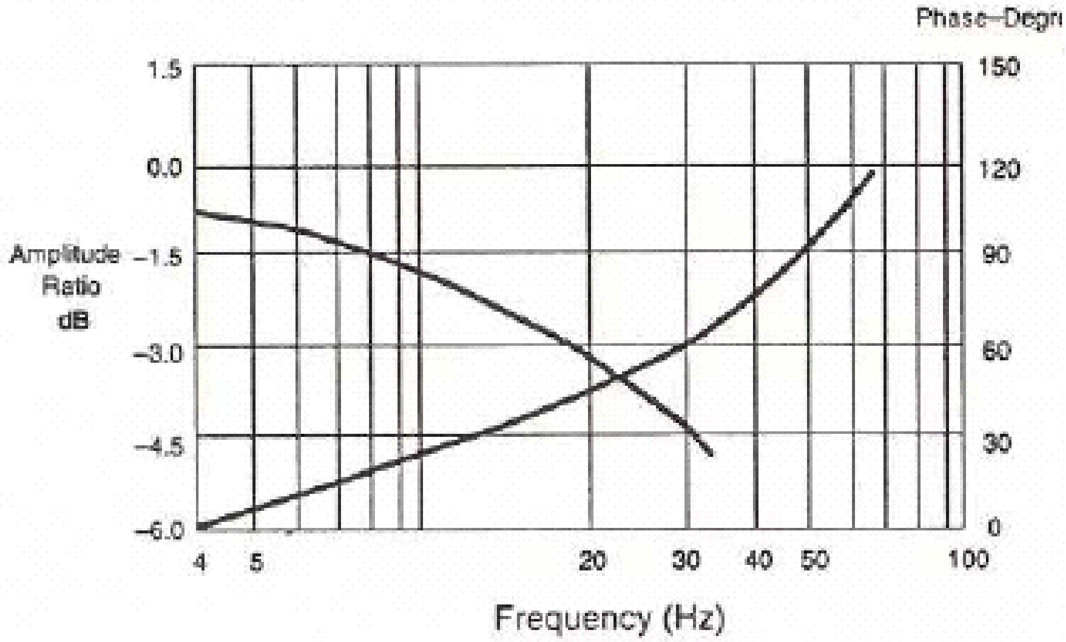


Figure 1. Change in flow with current and load pressure

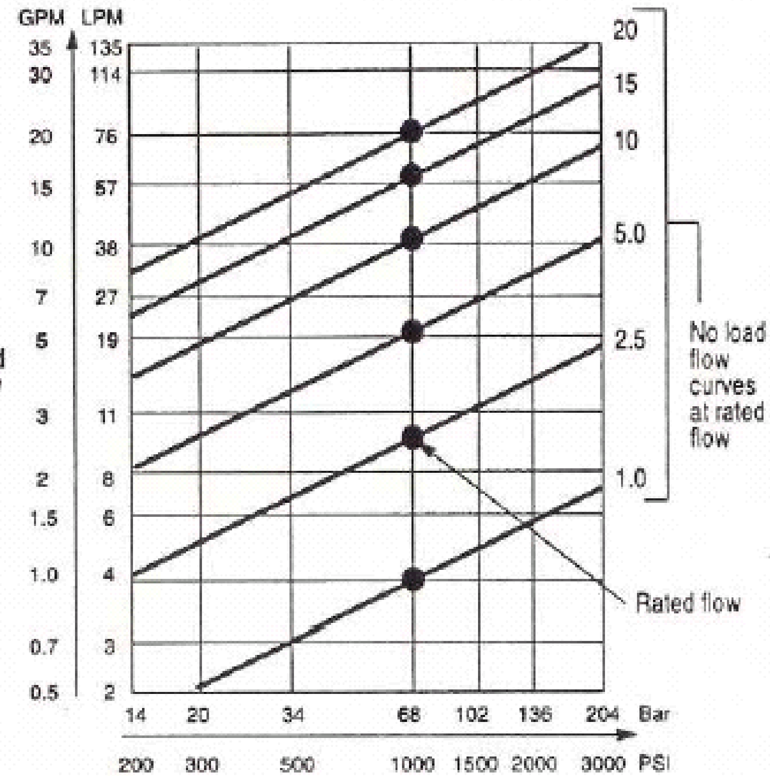
Manufacturer’s Data: BD15 Servovalve on HAL

Typical Response Curves



BD15 Typical Frequency
Response at ± 25% Input Current

Pressure Drop Curves



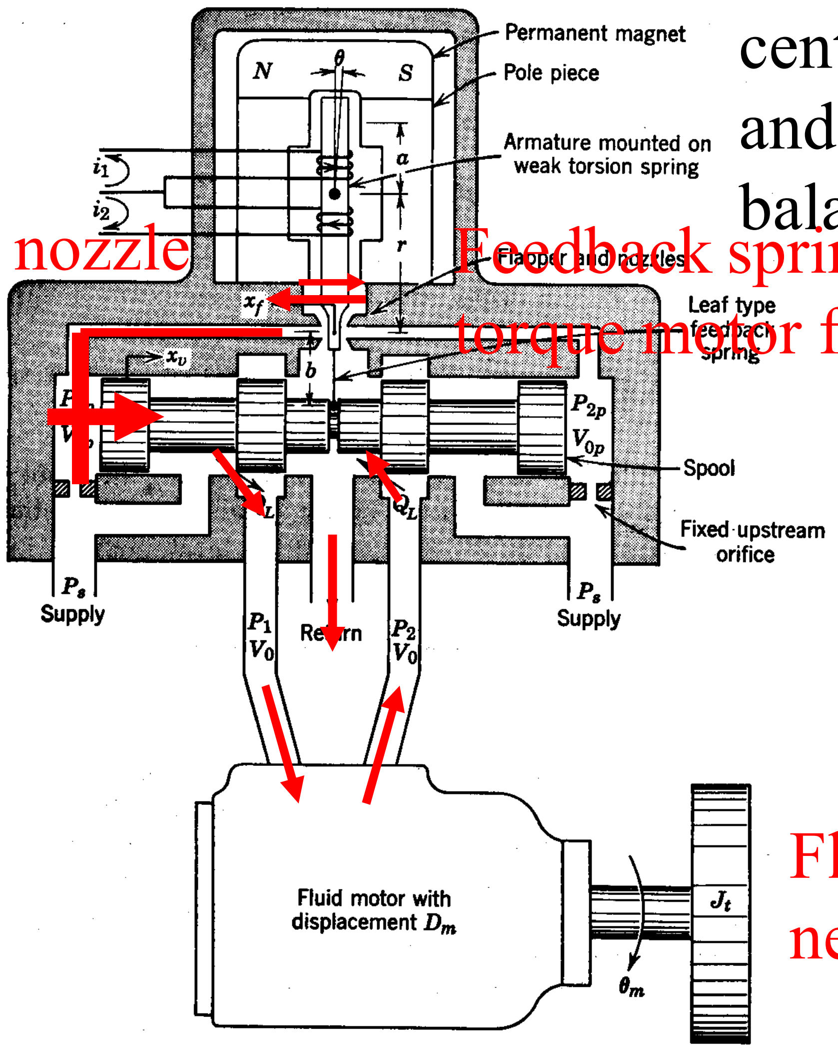
Two-stage Servo Valve

With flapper centered the flow and pressure is balanced

Torque motor rotates flapper, obstructs left nozzle

Feedback spring balances torque motor force

Pressure increases
Spool is driven right

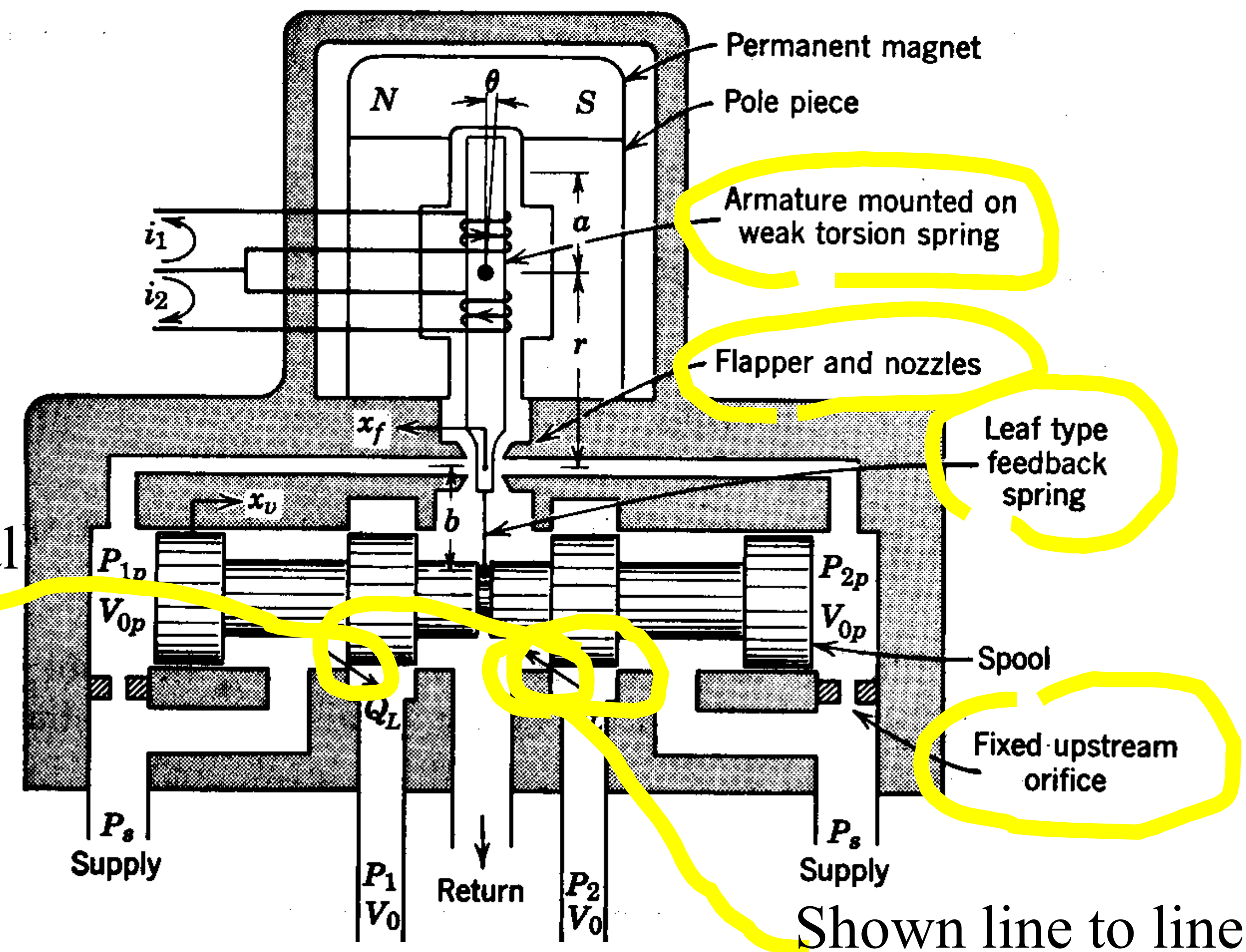


Flow gives negative rotation

Figure 7-17 Schematic of a two-stage electrohydraulic servovalve with force feedback controlling a motor with inertia load.

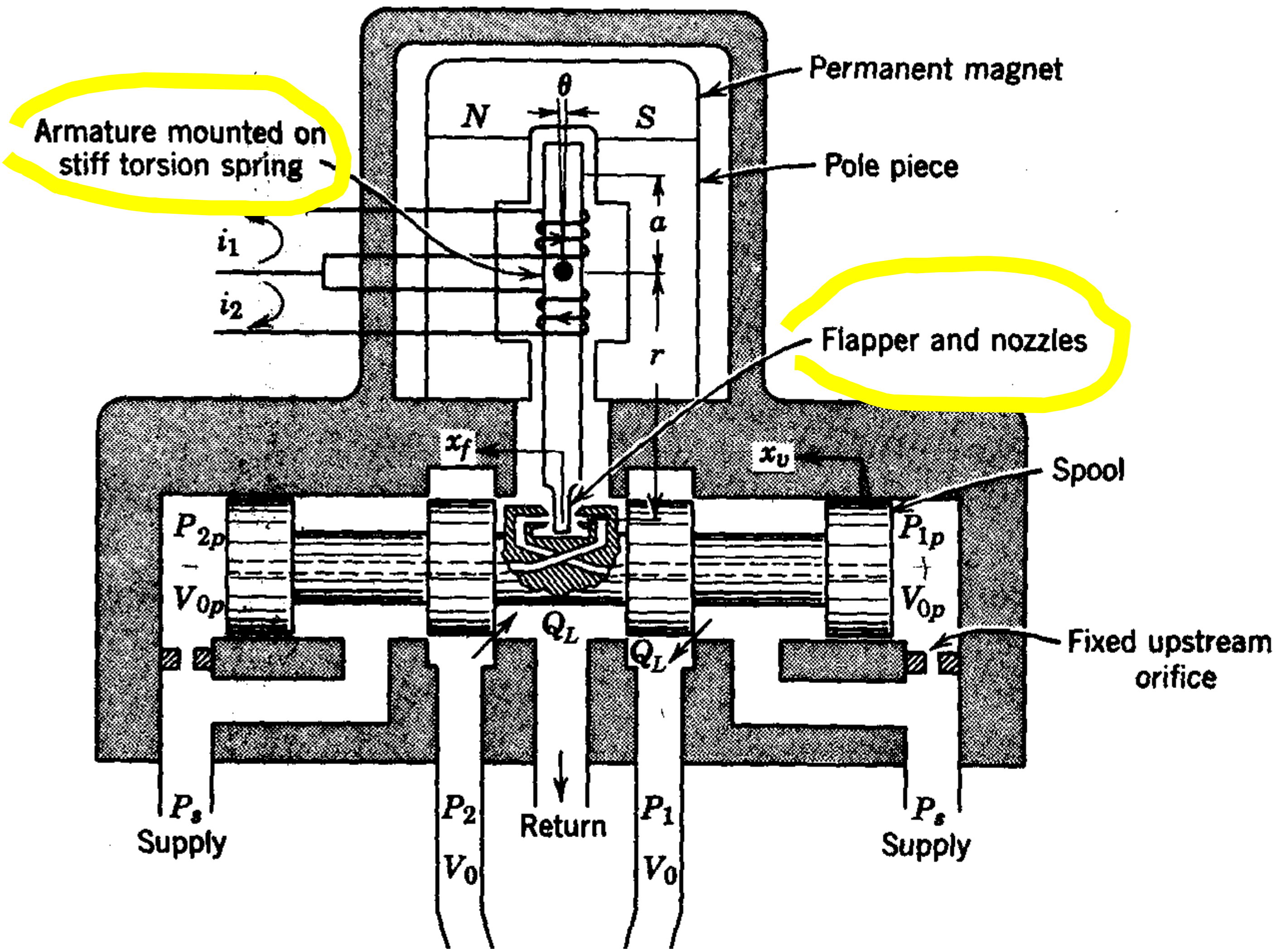
Details of Force Feedback Design

2 Sharp edged orifices, symmetrical opening



Shown line to line; no overlap or underlap

Another valve design with direct feedback

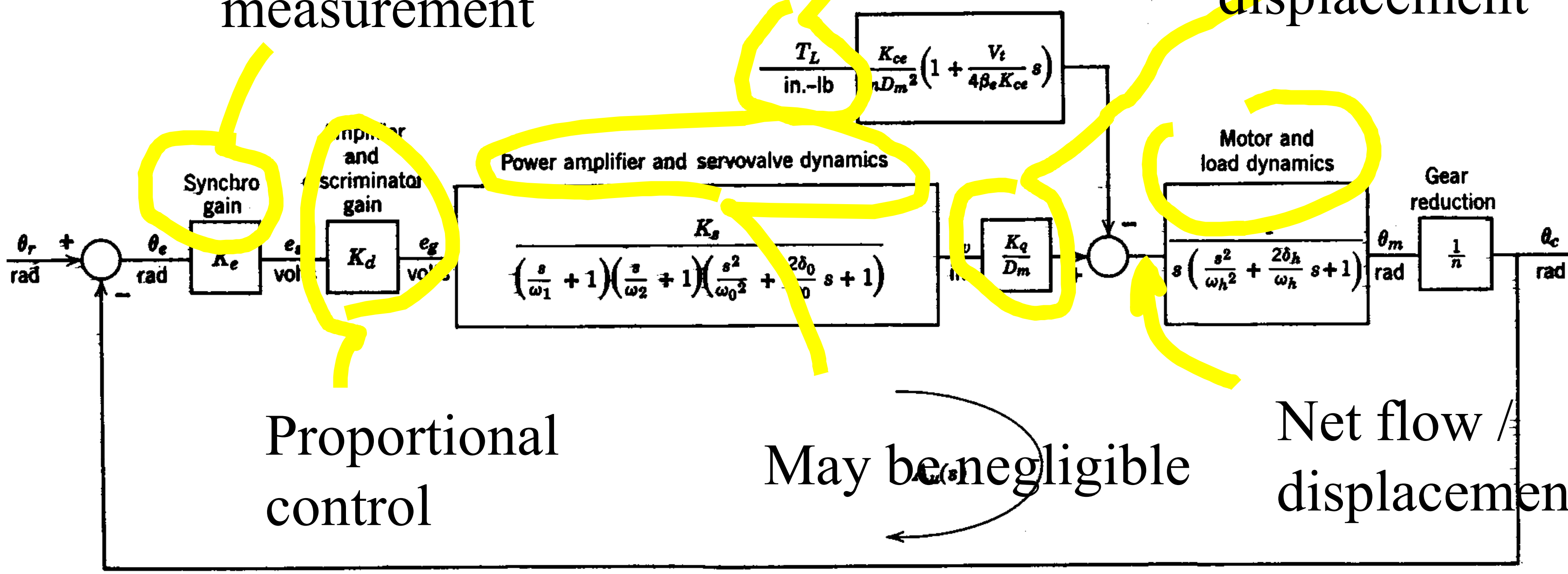


Position Servo Block Diagram

Position measurement

Load torque

Flow gain / motor displacement



Proportional control

May be negligible

Net flow / displacement

Figure 8-5 Block diagram of electrohydraulic position control servo.

Design of some components (with issues pertinent to this class)

- The conduit (tubing) is subject to requirements for
  - flow (pressure drop)
    - 2 to 4 ft/sec for suction line bulk fluid velocity
    - 7 to 20 ft/sec for pressure line bulk fluid velocity
  - pressure (stress)
- The piston-cylinder is the most common actuator
  - Must withstand pressure
  - Must not buckle

Design Equations for Fluid Power Systems

- Flow
  - Darcy's formula
  - Orifice flow models
- Stress
  - Thin-walled tubes ($t<0.1D$)
  - Thick-walled tubes ($t>0.1D$)
- Guidelines
  - Fluid speed
  - Strengths
  - Factors of safety (light service: 2.5, general: 3.15, heavy: 4-5 or more)

Darcy's formula from Bernoulli's Eq.

$$\Delta p = f_D \frac{L\rho}{2D_h}\left(\frac{Q}{A}\right)^2$$

Δp = pressure drop along the tube

f_D = friction factor (depends on N_R)

L = tube length

D_h = hydraulic diameter

= 4x (flow section diameter)/(section perimeter)

ρ = fluid mass density

Q = flow rate

A = flow section area

$$N_R = \frac{\rho u D_h}{\mu}$$

u = fluid velocity

or

$$Q = \frac{\pi D_h^4}{128\mu L}\Delta p,\ \mathrm{N_R} \le 2000$$

μ = absolute viscosity

(Hagen - Poiseuille law)

Friction factor for smooth pipes (empirical) from e.g. Fitch

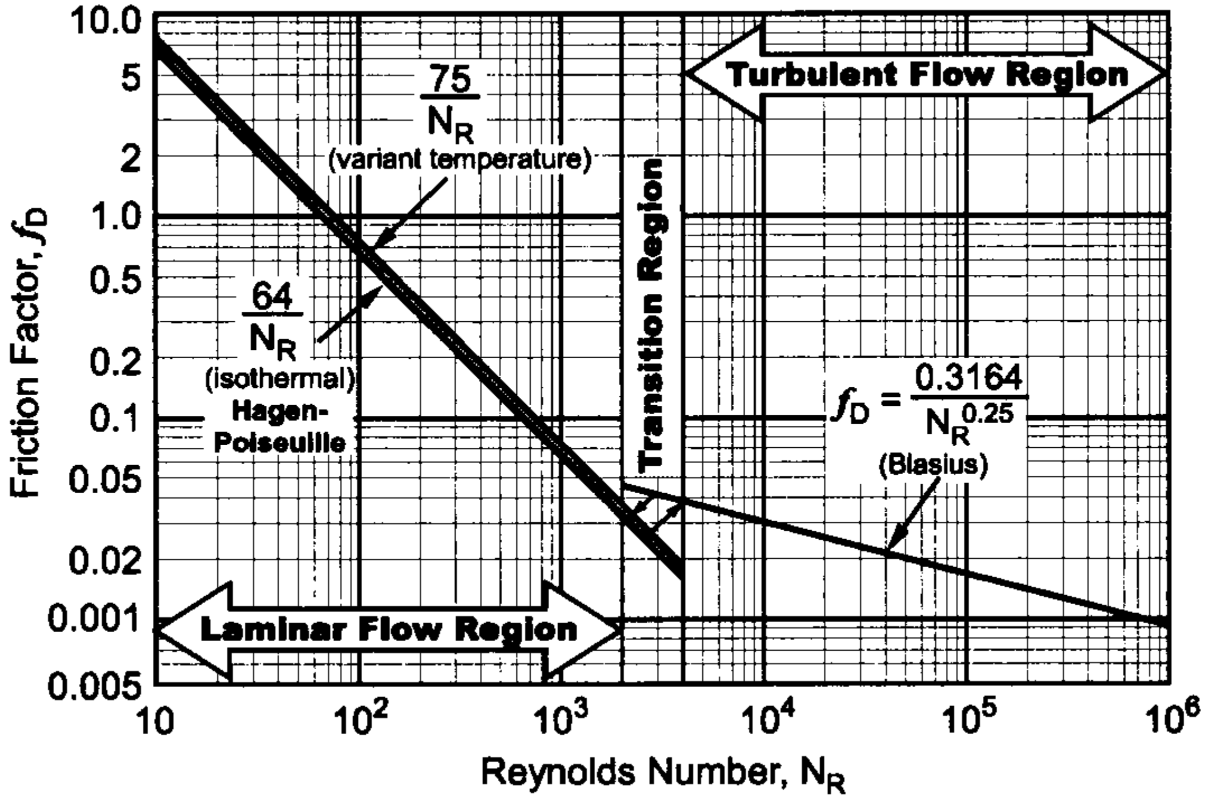


Figure 2-8. Friction Factor for Smooth Conduits.

Orifice Model

$$Q = C_d A_o \sqrt{\frac{2}{\rho} \Delta p}$$

C_d = orifice flow discharge coef.

A_o = orifice flow area

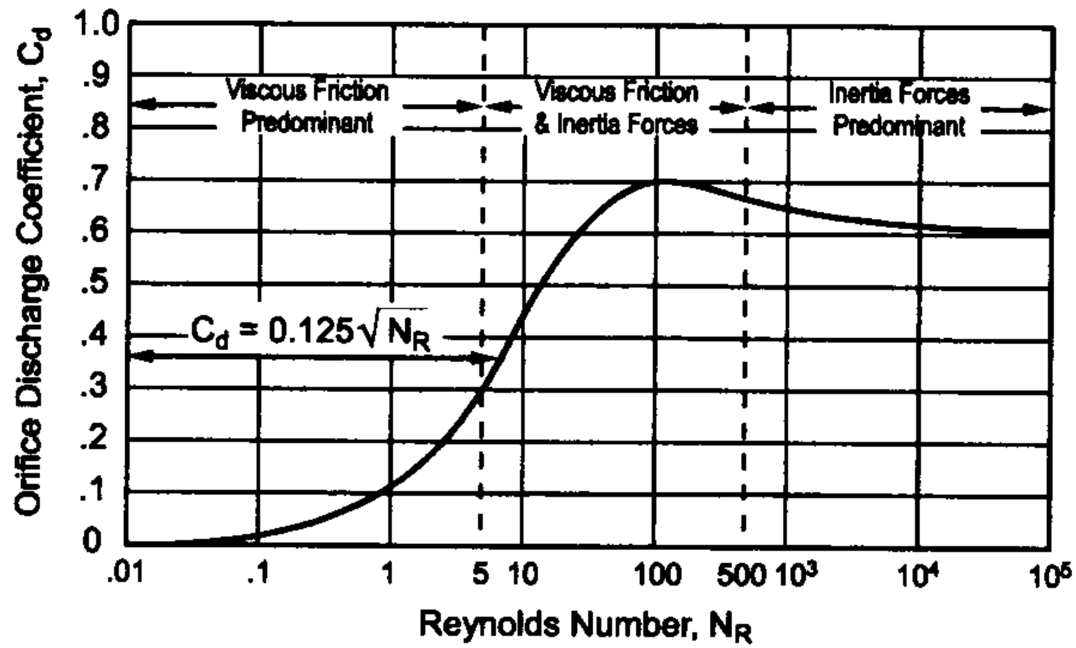


Figure 2-9. Orifice Discharge Coefficient versus Reynolds number.

Buckling in the Piston Rod (Fitch)

- Rod is constrained by cylinder at two points
- Constrained by load at one point
- Diameter must resist buckling
- Theory of composite "swaged column" applies
- Composite column fully extended is A-B-E shown below consisting of 2 segments
  - A-B segment buckles as if loaded by force F on a column A-B-C
  - B-E segment buckles as if loaded by F on DBE
  - Require tangency at B

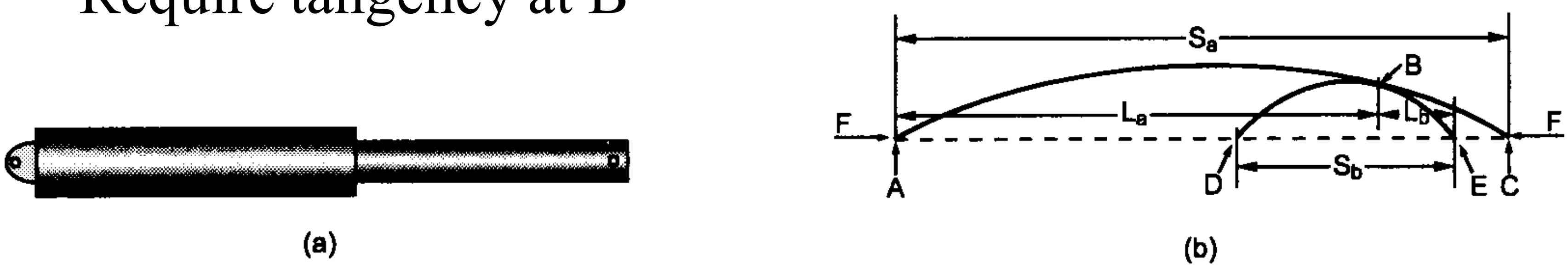


Figure 3-28. Swage-Column Buckling Model.

Cylinder construction (tie-rod design)

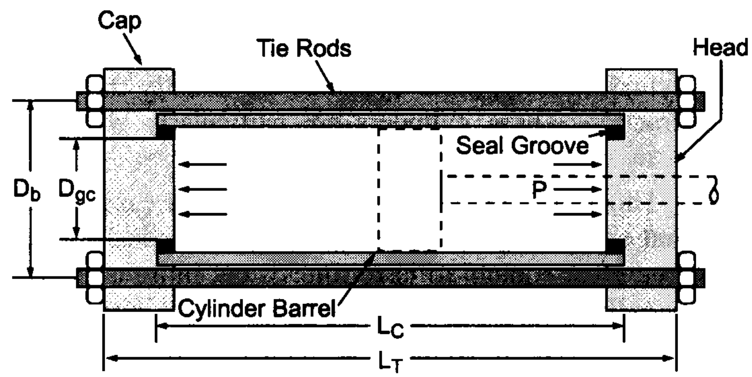


Figure 3-26. Simplified Tie-Rod-Constructed Cylinder.

Resulting loading on cylinder walls

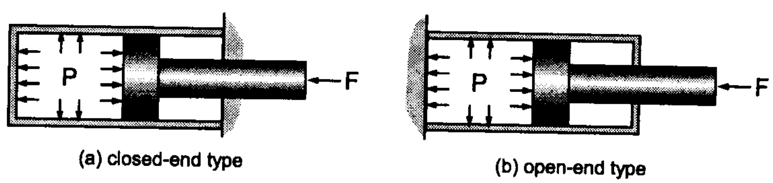


Figure 3-25. Typical Cylinder Mountings which will Result in Open- and Closed-end Cylinder Design.

Applicable wall thickness stress formulas (conduits or cylinders)

- Thin walled cylinders (open, or where only circumferential hoop stress is significant) (Barlow)
- Thick walled cylinders
  - Brittle materials (based on max normal stress) use Lame’s formula
  - Ductile (based on max strain theory)
    - Open end (no axial stress) (Birnie)
    - Closed end (cylinder bears axial stress) (Clavarino)
- Expansion of cylinder based on strain = stress/(Young’s modulus)

Stress formulas

Barlow's formula (thin, open)

$$t = \frac{PD_i}{2s_d}$$

D_i = inside diameter

s_d = design or allowable stress

= strength/(factor of safety * stress concentration)

ν = Poisson's ratio

Clavarino's formula (closed, thick, ductile)

$$t = \frac{D_i}{2}\left(\sqrt{\frac{s_d + (1-2\nu)P}{s_d - (1-\nu)P}} - 1\right)$$

Lame''s formula (brittle)

$$t = \frac{D_i}{2}\left(\sqrt{\frac{s_d + P}{s_d - P}} - 1\right)$$

Birnie's formula (open, thick, ductile)

$$t = \frac{D_i}{2}\left(\sqrt{\frac{s_d + (1-\nu)P}{s_d - (1-\nu)P}} - 1\right)$$

Results of Composite Column Model

Equating the slope of the two column segments at B where they join yields:

$$\sqrt{\frac{I_a}{I_b}}\tan\left(L_a\sqrt{\frac{F}{E_aI_a}}\right)=-\tan\left(L_b\sqrt{\frac{F}{E_bI_b}}\right)$$

$$D_r=\left(\frac{64I_b}{\pi}\right)^{0.25}=\text{diameter of rod}$$

The first equation may be solved iteratively then solve for D_r

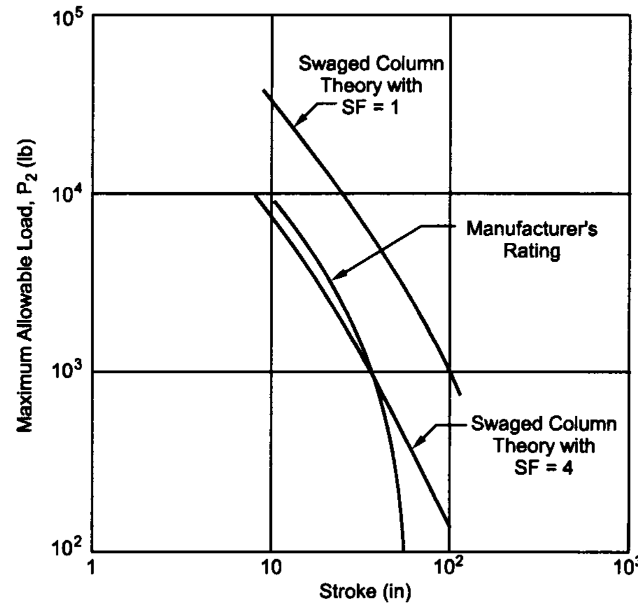


Figure 3-30. Results from Swaged Column Theory vs. Manufacturer's Rating.

Composite column model matches manufacturer's recommendations with factor of safety of 4

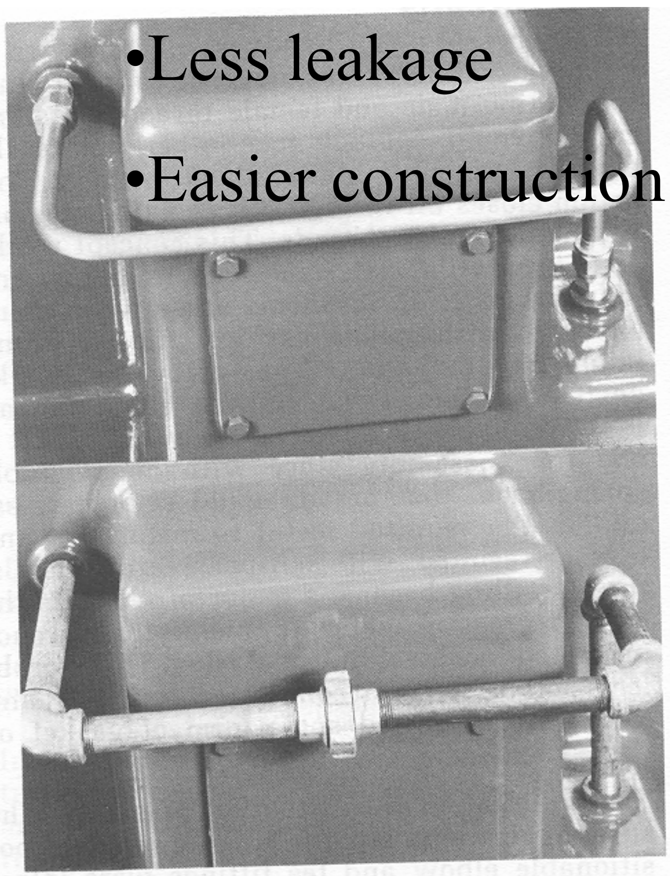
Pressure Specifications

- Nominal pressure = expected operating
- Design pressure = Nominal
- Proof pressure (for test) = 2x Design
- Burst pressure (expect failure) = 4x Design

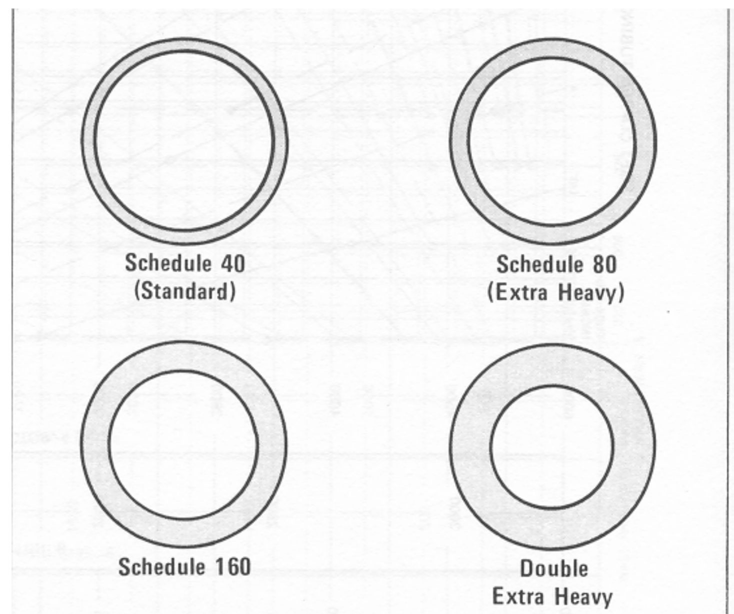
Pipes versus tubes

Tubes are preferred over pipes since fewer joints mean

- Lower resistance
- Less leakage
- Easier construction

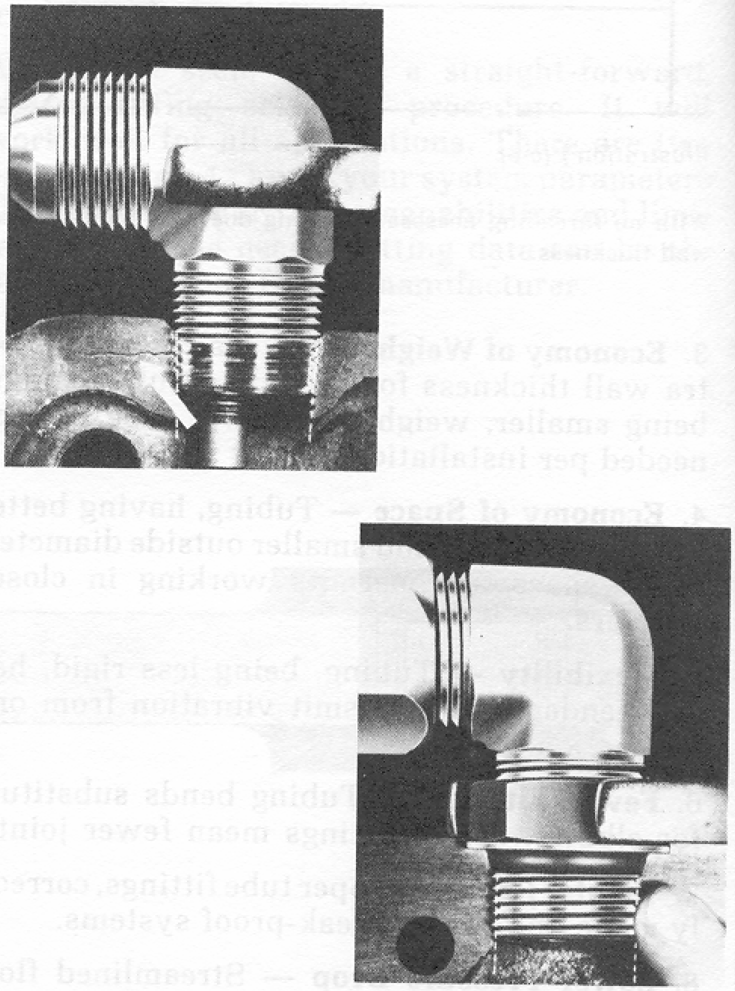


Four joints with tubing; 11 joints with pipe.
illustration j (e-7)



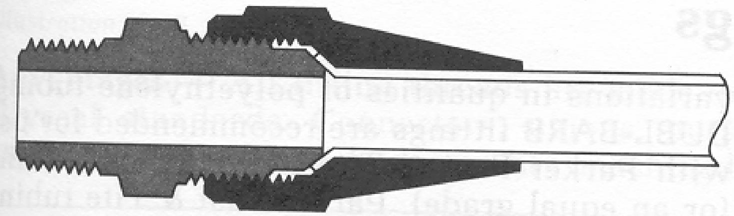
FOR ANY SCHEDULE PIPE, the OD remains the same, but ID decreases as wall thickness increases.

Fittings between tube and other components require multiple seals



Tapered pipe thread at left in cutaway boss; straight thread at right.

illustration j (e-10)



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Flared tube design

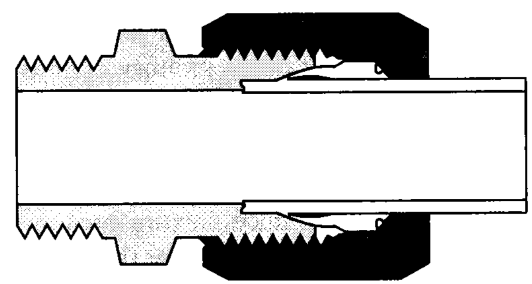

Figure 11-4. The Parker Ferulok Bite Type Fitting.