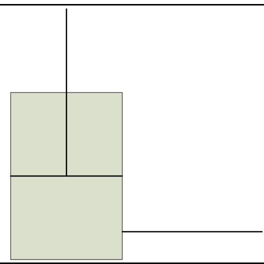
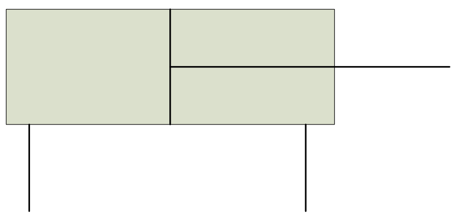
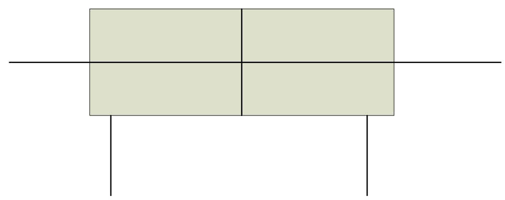
Chapter 6. Hydraulic cylinders/Rams (linear motors), and Lines/fittings

- Transforms the flow of a pressurized fluid into a push or pull of a rod.

6.1 Single Acting Rams

	Gravity, spring, etc. can force piston to return.

6.2 Double Acting Rams (Cylinder)

<u>Standard</u>	<u>Double Rod</u>
- Is a differential cylinder because unequal areas for same inlet flow, speed differential.	Nondifferential cylinder same inlet flow, speed the same

6.3 Construction

Barrel, piston, rod, end caps, seals

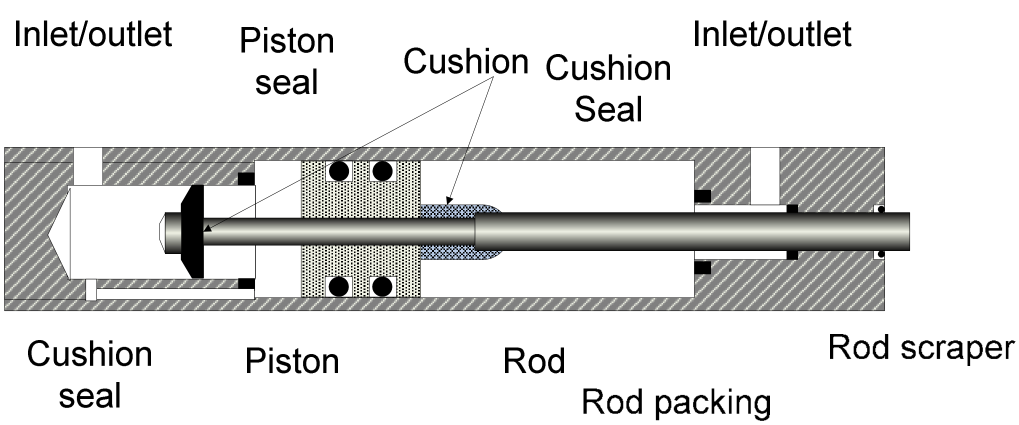


Figure 6.1 Cylinder construction of a cylinder with cushions

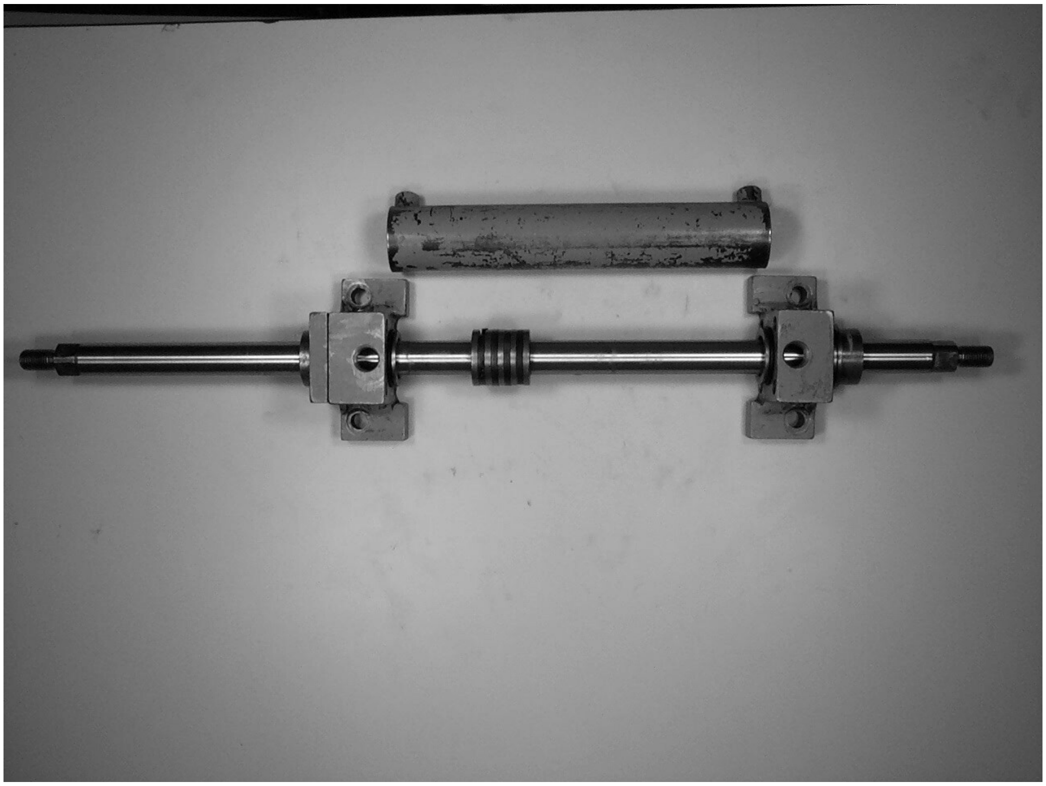


Figure 6.2 Cylinder layout

6.4 Cylinder Ratings

- Cylinders are rated by size and pressure
  Cushions are available to slow piston down at the end of its stroke.
- Stop tubes next to piston provides more support for side bending

6.5. Basic equations:

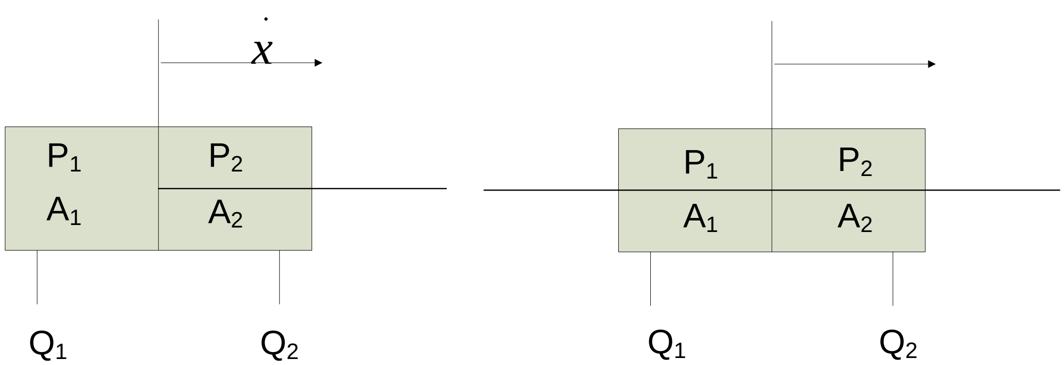


Figure 6.3 Nomenclature

Consider Figure 6.3. The basic describing equations are:

$$P_1\ A_1\ - P_2\ A_2 = \text{"}hydraulic\ \ force\text{"}$$

$$Q_1 = A_1\ \dot{x}$$

$$Q_2 = A_2\ \dot{x}$$

6.6 Limited Rotation Motors

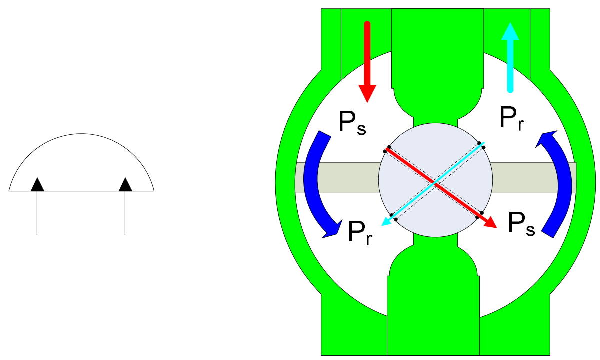


Figure 6.4 Limited-Rotation Dual-Valve Motor

- A double vane provides 180 degrees of motion
- A single vane provides 280 degrees of motion

An intersection variation is a high torque limited rotation motor. This uses many isolation vanes and separation ports but at the expense of rotation. These units are pressure balance by porting. See Figure 6.4.

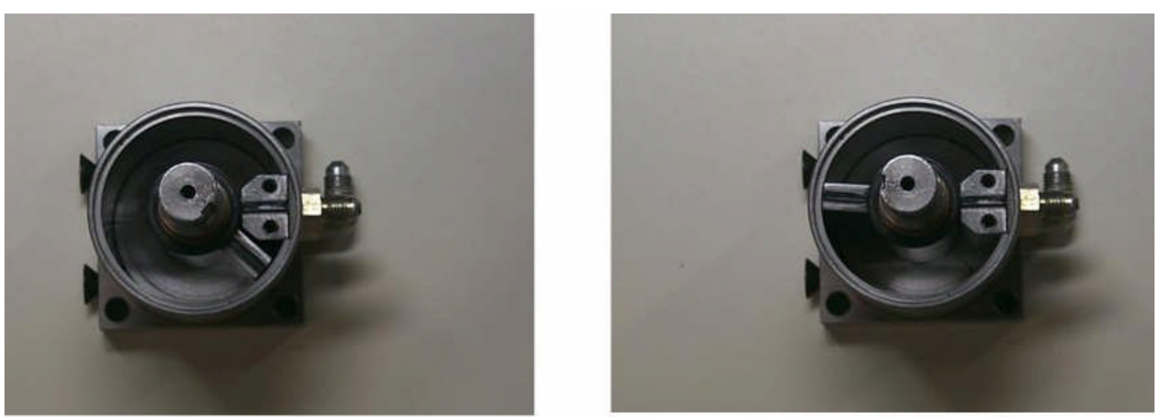


Figure 6.5 Limited rotation single vane motor

6.7 Intensifiers

- Can be used to produce a high pressure, low flow source

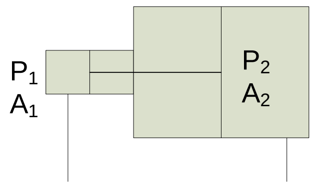


Figure 6.6 Pressure Intensifer

$P_1A_1 = P_2A_2$ $\qquad$ $P_2 = \dfrac{P_1A_1}{A_2}$. Since A_2 is larger than A_1, then $P_2 < P_1$

$\dot{x} = \dfrac{Q_1}{A_1} = \dfrac{Q_2}{A_2}$ Therefore, $Q_2 = \dfrac{Q_1 A_2}{A_1}$. Since A_2 is larger than A_1, then $Q_2 > Q_1$

6.8 Piston Rod Buckling

Consider the following configurations:

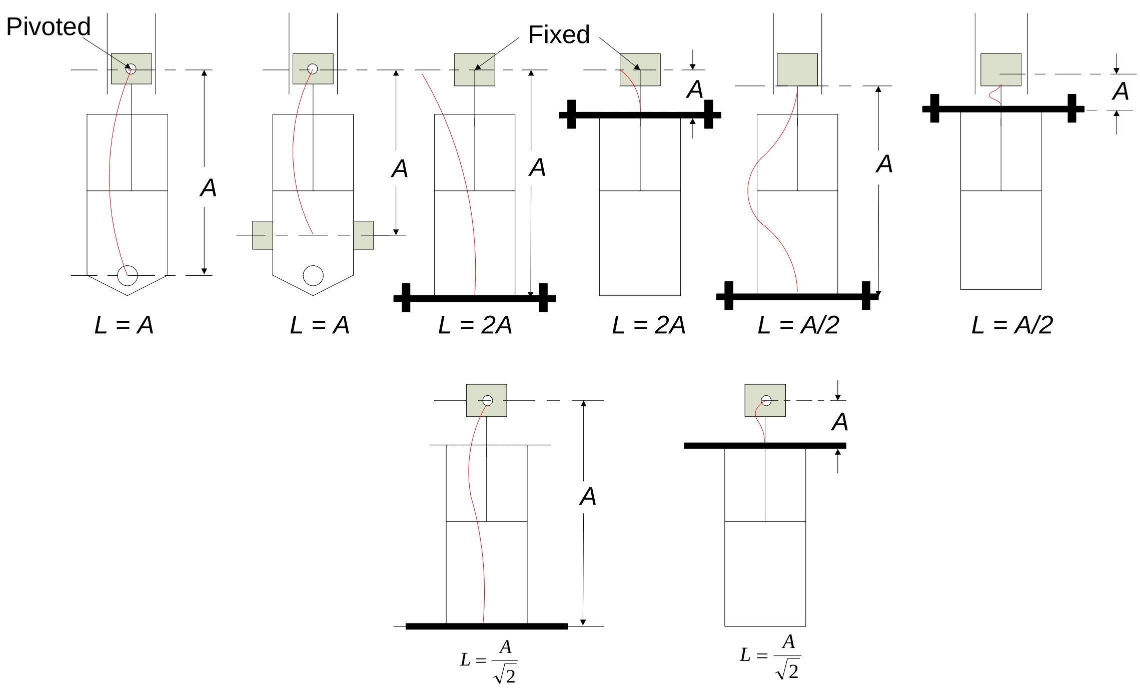


Figure 6.7 L for K factor

Using Euler's Strut Theory:

$$K = \frac{\pi^2 E J}{L^2}$$

Define:

K = buckling load (kg)
E = Mod. of elasticity kg/cm^2
(2.1 x 10^6 kg/cm^2 steel)
J = second moment of area of the piston rod (cm^4)
$= \frac{\pi d^4}{64}$ for solid rod
L = free length (as before) cm

To calculate the maximum safe working thrust, use the relationship

$$F_{max} = \frac{K}{s}$$

where s = safety of factor = 3.5

Recommended cylinder bore and rod size standard BS5785: 1980

Piston Dia (mm)	40	50	63	80	100	125	140	160	180	200	220	250
Rod Dia small (mm)	20	28	36	45	56	70	90	100	110	125	140	160
Large	28	36	45	56	70	90	100	110	125	140	160	180

6.9 Flow resistance in pipes and lines

Some of the largest losses in hydraulic systems are due to undersized hydraulic lines and fittings.

- In general, line losses are a function of the square of the velocity, the length, the diameter, as well as viscosity.
- In general, losses in a line in terms of head loss (ft) is given by

$H_L = f \frac{L V^2}{D\, 2\, g}$, where f is called the friction factor. L is the line length, V the average velocity, D the diameter of the pipe and g, the acceleration due to gravity.

f depends on the Reynolds No. $Re = \rho \frac{V D}{\mu}$

(a) For Re < 2000 (Laminar flow)

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

$$f = \frac{75}{Re}$$ Variation in temperature (adiabatic conditions)

(b) For Re > 2000 (Turbulent flow)

Can be obtained from friction charts (see a standard fluid text) or from $f \cong \frac{.3164}{(Re)1/4}$ (smooth steel)

6.9.1 Calculating Pressure Drops in a Line

$$\Delta P = \frac{\mu \times Q}{18300\ D^4}$$

where μ = viscosity in SSU
Q = flow in GPM (US)
D = inside diameter of pipe (in)
ΔP = pressure drop per foot in psi

If the max velocity in a line is known, then the inside area can be calculated from

$$\text{Area} = \frac{\text{GPM} \times .3208}{\text{velocity}} \quad (\text{in}^2)$$

Recommended maximum line velocities (for acceptable energy losses) [from T.C. Frankenfield, Using Industrial Hydraulics]

Suction Lines:	30.5 to 61 cm/sec	(2 to 4 ft/sec)
Return Lines:	25.4 to 38 cm/sec	(10 to 15 ft/sec)
Working Lines: 3.5 - 21 MPa (500 -3000 psi)	38 to 51 cm/sec	(15 to 20 ft/sec)
Working Lines: 21 - 34.5 MPa(3000-5000 psi)	38 to 76 cm.sec	(15 to 30 ft/sec)

6.9.2 Pressure Losses in Fittings and Valves

As of equal importance to line losses are losses due to elbows, fittings valves, etc.

To assist in approximating losses, the K factor is used.

$H_L = .000510 K\, V^2$ where V is in cm/sec and H_L is in cm

$H_L = .01554 K\, V^2$ where V is in ft/sec and H_L is in ft

K depends on Re, and the restriction cross section.

K for standard valves and fittings are listed in Table 7.1 (it is assumed that the valves are fully opened).

K values for sudden chances in section are shown in Table 7.2 where for enlargement, V = V_1 inlet velocity and for contraction, V_1 = outlet velocity

Many other losses due to such things as strainers and filters occur. These will not be presented here but tabulated data is available for loss consideration.

Table 7.1 K Factors for Standard Valves and Fittings (Typical)

COMPONENT	**INTERNAL DIAMETERS**			
	1.27 cm (1/2)"	2.54 cm (1")	5.1 cm (2")	10.2 cm (4")
GATE VALVE	.36	.3	.25	.21
GLOBE VALVE	9.5	7.9	6.6	5.7
STANDARD ELBOWS				
90^o	.82	.7	.58	.5
45^o	.43	.36	.3	.26
LONG ELBOW				
90^o	.55	.45	.38	.33
STANDARD TEE				
Flow Through	.55	.45	.38	.33
Flow Through	1.7	1.4	1.2	1.0

Table 6.2 K values for Enlargements and Contraction Yeaple, (Industrial Hydraulics)

$K_{enlargement}$	.81	.64	.49	.36	.25	.16	.09	.04	.01
a/A	.1	.2	.3	.4	.5	.6	.7	.8	.9
$K_{contraction}$	.4	.38	.34	.3	.24	.18	.1	.05	.015

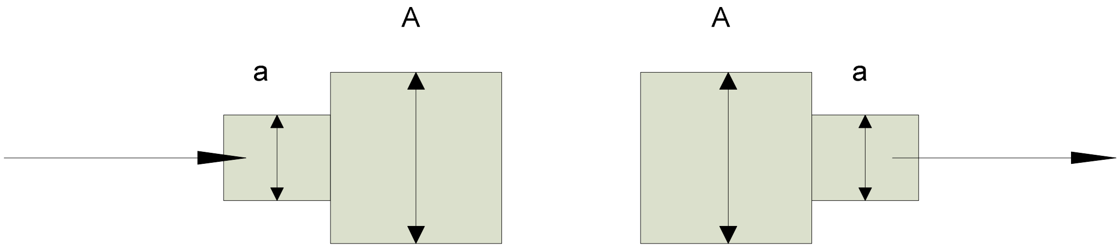


Figure 6.8 Enlargement /contraction