Introduction to Fluid Power

Chapter 1

Dr. Suleiman BaniHani

What is Fluid Power

- Fluid power is the technology that deals with the generation, control, and transmission of power, using pressurized fluids.
- Examples:
  - Automobiles steering, brakes
  - Control planes
  - Drill teeth

Hydraulic Vs. Pneumatic

- Fluid power is called hydraulic when the fluid is liquid and is called pneumatic when the fluid is gas.
- Hydraulic systems use liquids such as:
  - Petroleum oils
  - Synthetic oils
  - Water

Water in Hydraulics

- The first Hydraulic fluid that was used is water because it is readily available.
- However, water has many deficiencies in comparison to hydraulic oils
  - Water freeze more readily
  - Water is not a good lubricant
  - Water tends to rust metal components
- When water hydraulics is used additives are used.
- In spite of these deficiencies there is a renewed effort to return to water in certain applications because
  - Water abundance
  - Nonflammability
  - Environmental cleanliness

Fluid Transportation Vs. Fluid Power

- Fluid transportation system have as their sole objective the delivery of fluid from one location to another tgo accomplish some purpose.
  - Pumping station for pumping water to homes
  - Cross county gas lines
  - Delivery of different fluid in chemical processes
- Fluid power systems are designed specifically to perform work, which is accomplished by a pressurized fluid bearing directly into an operating fluid cylinder, for linear motion, or fluid motor, rotary motion (actuators).

Examples

- The Caterpillar 797B mining truck is the largest truck in the world at 3550 hp.
- It carries 400 tons at 40 mph, uses 900 g of diesel per 12 hr shift, costs about $6M and has tires that are about $60,000 each.

The 797B uses fluid power for many of its internal actuation systems, including lifting the fully loaded bed.

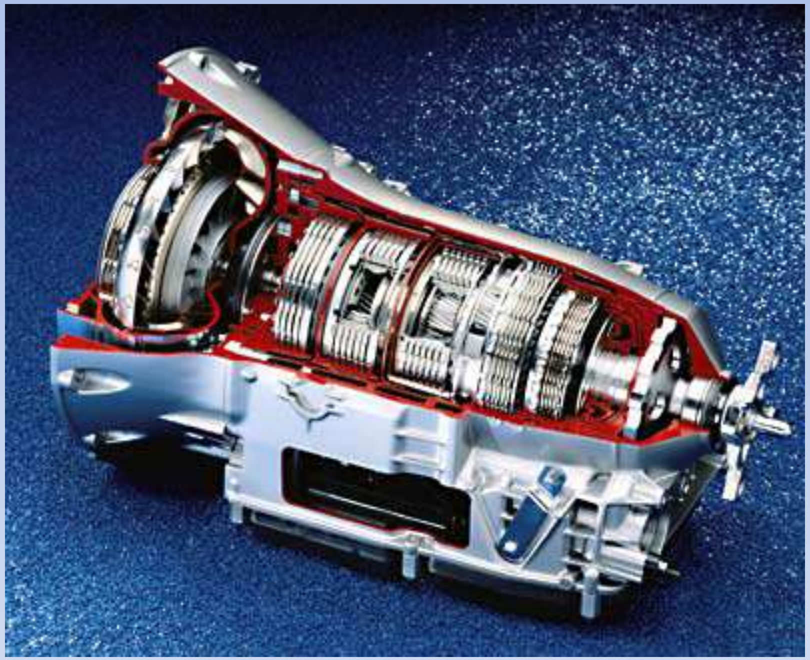
Examples

- The Multi-Axial Subassemblage Testing (MAST) Laboratory is located at the Universtiy of Minnesota and is used to conduct three-dimensional, quasistatic testing of large scale civil engineering structures, including buildings, to determine behavior during earthquakes.

The MAST system, constructed by MTS Systems, has eight hydraulic actuators that can each push or pull with a force of 3910 kN.

Example

- Most automatic transmissions have hydraulically actuated clutches and bands to control the gear ratios. Fluid is routed through internal passageways in the transmission case rather than through hoses
- The dental drill is used to remove small volumes of decayed tooth prior to inserting a filling. Modern drills rotate at up to 500,000 rpm using an air turbine and use a burr bit for cutting. The hand piece can cost up to $800.
- Pneumatic drills are used because they are smaller, lighter and faster than electric motor drills. The compressor is located away from the drill and pressurized air is piped to the actuator.



Advantages

- There are three basic methods for transmitting power: electrical, mechanical, and fluid power.
- Most application uses combination of the three methods for efficient overall system.
- Fluid systems can transmit power more economically over greater distances than can mechanical types. However, fluid systems are restricted to shorter distances than are electrical systems.

Advantages Cont.

- Industry is depending more on automation and the use of fluid power because of
  - Ease and accuracy of control
  - Multiplication of force
  - Constant torque or force
  - Simplicity, safety, economy
  - Instantly reversible motion
  - Automatic protection against overloads
  - Infinitely variable speed control
  - Highest power per weight ratio

Disadvantages

- Oil leakage, hence hydraulic systems must be properly designed and installed.
- Pipelines can burst due to excessive pressure
- In pneumatic systems components such as compressed air tank and accumulators must be properly selected to handle the system maximum air pressure
- Noise.

Functions of Fluid Power Systems

- Fluid power systems perform five functions during operation:
- – Energy conversion
- – Fluid distribution
- – Fluid control
- – Work performance
- – Fluid maintenance

Structure of Fluid Power Systems

- Fluid power systems are structured using component groups that perform specific system functions:
  - Power unit group
  - Actuators group
  - Conductors group
  - Control valves group
  - Fluid maintenance group

Power unit group

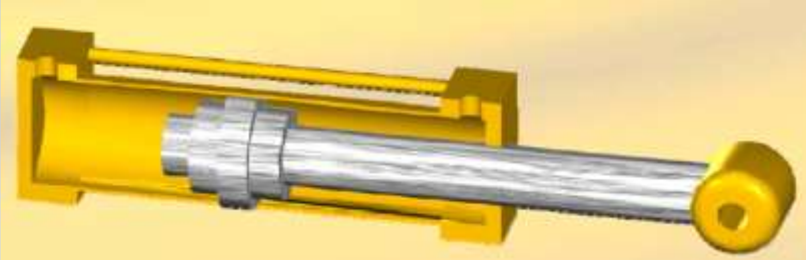
Deals primarily with energy conversion

Consists of:

- Prime mover
- Pump or compressor
- Reservoir or receiver

Actuators group

- Performs the work of the system
- Consists of both cylinders and motors



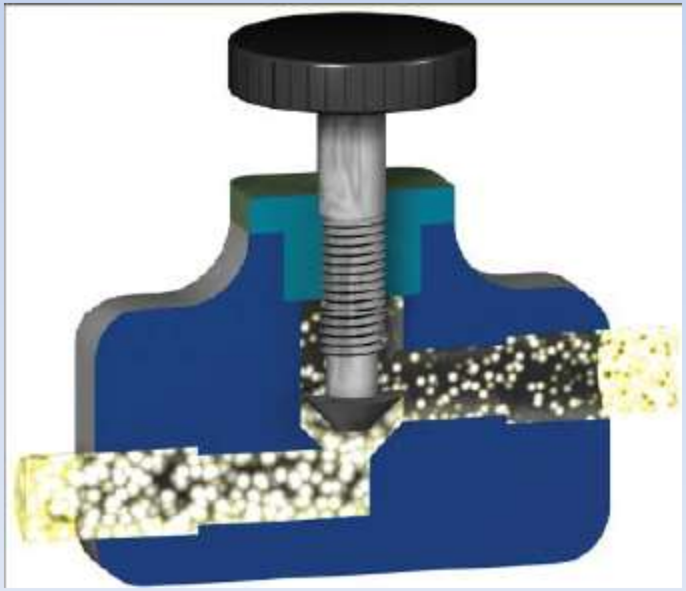
Conductors group

- Conductors distribute fluid throughout the system
- Consists of:
  - Pipes
  - Tubes
  - Hoses

Control valves group

- Controls fluid pressure, flow direction, and flow rate

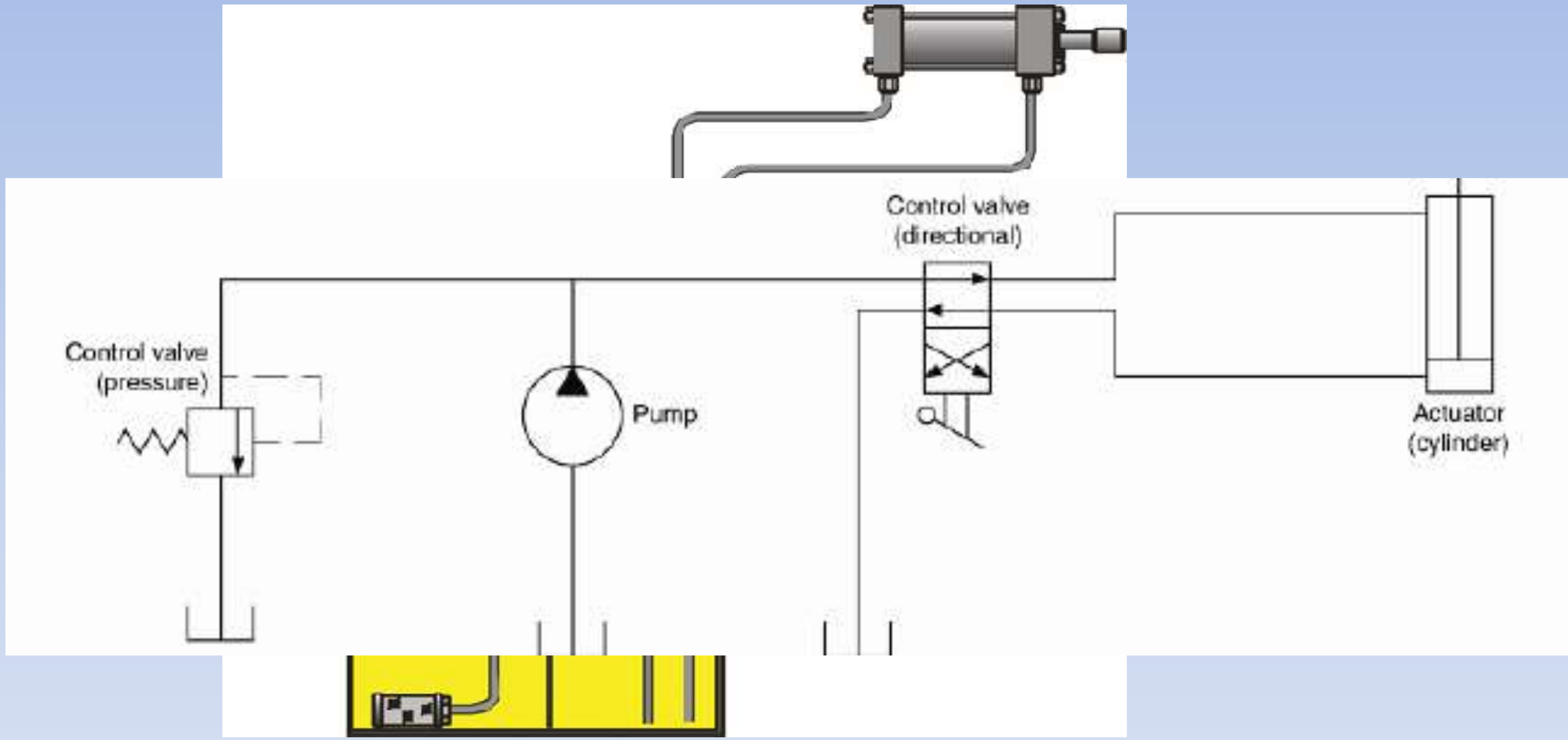
– Three groups of valves:
  - Directional control valves
  - Pressure control valves
  - Flow control valves



Fluid maintenance group

- Maintains system fluid by removing dirt, moisture, and excessive heat
- Filters and other devices are used to perform these functions

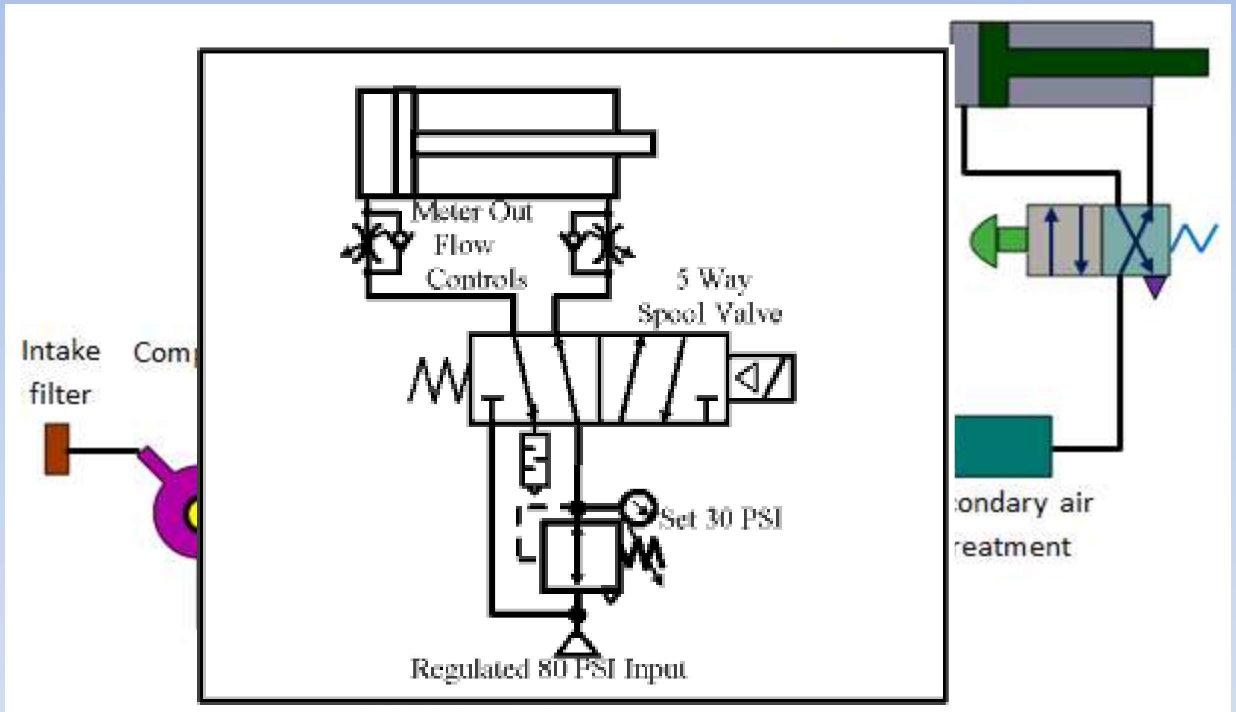
Components of Fluid power system



Components of Fluid power system

- Hydraulic system
  - A tank (reservoir) to hold the hydraulic oil.
  - A pump to force the oil through the system.
  - An electrical motor or other power source to drive the pump.
  - Valves to control the oil direction, pressure, and flow rate.
  - An actuator (cylinders or motors) to convert the pressure of the oil into mechanical force or torque to do useful work.
  - Piping which carries the oil from one location to another.

Components of Pneumatic system



Pneumatic System

- Pneumatic system has the following main components
  - An air tank to store a given volume of compressed air.
  - A compressor that compress the air that comes directly from the atmosphere.
  - An electrical motor or other prime move to drive the compressor.
  - Valves to control the air direction, pressure, and flow rate.
  - Actuators
  - Piping to carry the pressurized air from one location to another.

Hydraulic system operation

– Movement of oil originates at the pump
– Low pressure at the pump inlet causes oil to pass through a filter as it flows from the reservoir into the pump
– High pressure at the pump outlet forces oil to
– the directional control valve and on to the actuator
– System work is performed by the actuator
– Pressure control valves limit pressure in the system
– – Flow control valves control the speed of actuator movement
– Oil is returned to the reservoir to be recirculated through the system

Pneumatic system operation

- Movement of air begins at the compressor
- As air moves into the system from the atmosphere,
- it is:
  - Filtered
  - Compressed
  - Stored in the receiver under pressure
- Pressurized air is distributed to system workstations

At the workstation:

- A pressure regulator sets working pressure
- A filter and lubricator provide final conditioning
- Air then moves through a directional control valve and on to an actuator
- System work is performed by the actuator
- During system operation, flow control valves control the speed of actuator movement
- Air is discharged back into the atmosphere after passing through the system

Physical Properties of Hydraulic Fluids

Chapter 2

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Hydraulic Fluids

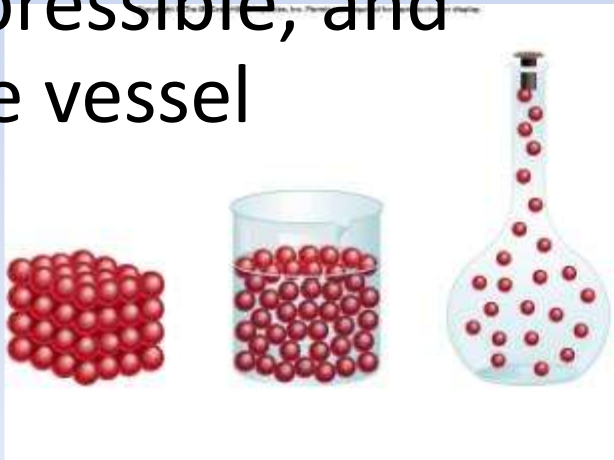
- The most important material in a hydraulic system is the working fluid itself.
- The fluid characteristic is crucial to the performance and life of the equipments.
- Most modern fluids are complex compounds to achieve their demanding task.
- A hydraulic fluid has the following primary functions
  - Transmit power
  - Lubricate moving parts
  - Seal clearances between mating parts
  - Dissipate heat.
- In addition it must be inexpensive and readily available.

Properties of hydraulic fluid

- Good lubricity
- Ideal viscosity
- Chemical stability
- Compatibility with system materials
- High degree of incompressibility
- Fire resistance
- Good heat transfer capability
- Low density
- Foam resistance
- Nontoxicity
- Low volatility

Fluids: Liquids and Gases

- Liquids: A liquid is a fluid that, for a given mass, will have a definite volume independent of the shape of its container.
- Liquids are considered to be incompressible so that their volume doe not change with pressure change.
- Gases: are fluids that are compressible, and their volume will vary to fill the vessel containing them.



Air: as fluid power

- Air is the only gas commonly used in fluid power systems because it is inexpensive and available.
- Air desirable features
  - Fire resistance
  - Not messy
  - Can be exhausted into the atmosphere
- Disadvantages of using air versus using hydraulic oil
  - Can not be used for accurate positioning or rigid holding
  - Sluggish
  - Corrosive, contains oxygen and water
  - Lubricant must be added
  - Pressures greater than 250 psi are typically not used, explosion danger.

Fluid Properties

- Specific weight
- Density
- Specific gravity
- Viscosity

Weight Versus Mass

- Weight: is the amount of force a certain body, solid or fluid, are pulled toward the center of gravity and is proportional to the object mass

$$F = W = mg$$

$F =$ force in units of lb

$W =$ weight in units of lb

$m =$ mass of object in unit of slugs

$g =$ proportionality constant called acceleration of gravity, equals $32.2\, ft/s^2$

Key Concepts

- Density:
  - Amount of mass per unit volume

    ρ = mass/volume
  - Density is a fluid property and slightly dependent on temperature
  - Units: kg/m^3, g/cm^3, slugs/ft^3
- Specific Volume:
  - Inverse of density

    $\nu = 1/\rho$, m^3/kg

Specific Weight

- Specific weight is the weight per unit volume
- It was found that the weight of $1ft^3$ of water is 62.4lb

$$Specific\ weight = \frac{weight}{volume}$$

or

$$\gamma = \frac{\mathrm{W}}{\mathrm{V}}$$

$$\gamma = specific\ weight(lb/\ ft^3)$$

$$W = weight(lb)$$

$$V = volume(\ ft^3)$$

- Most oils have specific weight that vary from a low of $55lb/ft^3$ to a high of $58lb/ft^3$

Key Concepts

- Specific Gravity (SG)
  - Is the specific weight of the fluid divided by the specific weight of water, or
  - Is the ratio of density of fluid to the density of reference fluid (usually water) at the same temperature.

$$(SG)_{oil} = \frac{\gamma_{oil}}{\gamma_{water}} = \frac{\rho_{oil}}{\rho_{water}}$$

  - Ex. Find the specific gravity of air, where $\gamma_{air} = 0.0752 lb / ft^3$

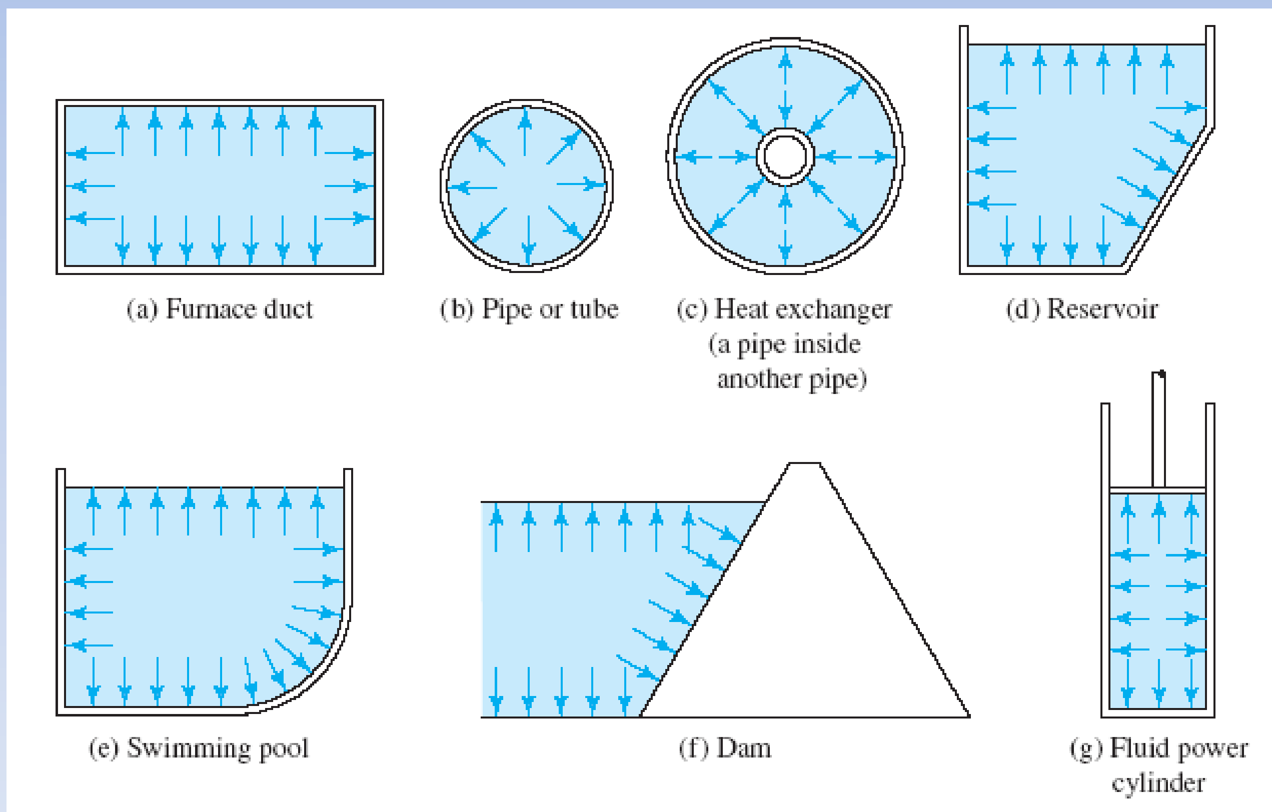
At 68F and atmospheric pressure

Equations for Property Calculations

- Circular Area: Area = $\pi/4*D^2$
- Weight: $w = m*g$
- Density: $\rho = m/V$
- Specific Weight: $\gamma = w/V$
- Specific gravity: $SG=\rho/\rho_{water}$

Key Concepts

- Pressure:

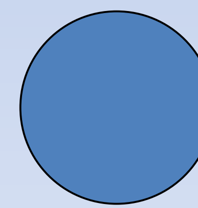


psi

and

container

$\pi/4*D^2$



re=Force/Area

si or Pa (SI)

Head

- Head pressure is the pressure developed by a fluid due to its own weight.
- A 1 ft^3 of water develop a pressure at the base of 0.433psi

$$p = \gamma H$$

$$p = \text{pressure at the bottom of liquid column}$$

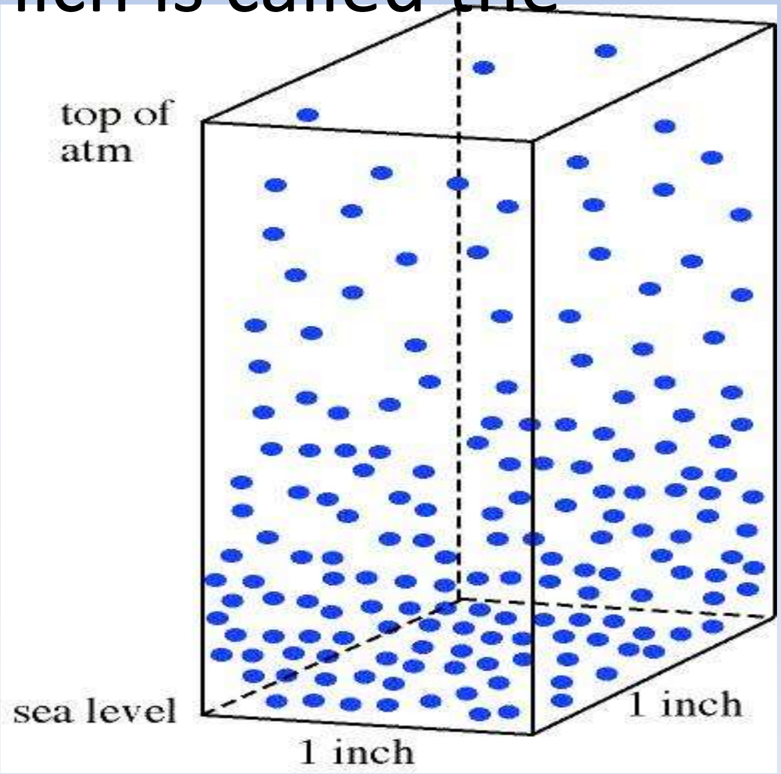
$$\gamma = \text{specific weight of liquid}$$

$$H = \text{liquid column height or head}$$

- Ex: find the pressure at the skin of a diver on 60ft depth of water (26psi gama_water=0.0361lb/in^3)

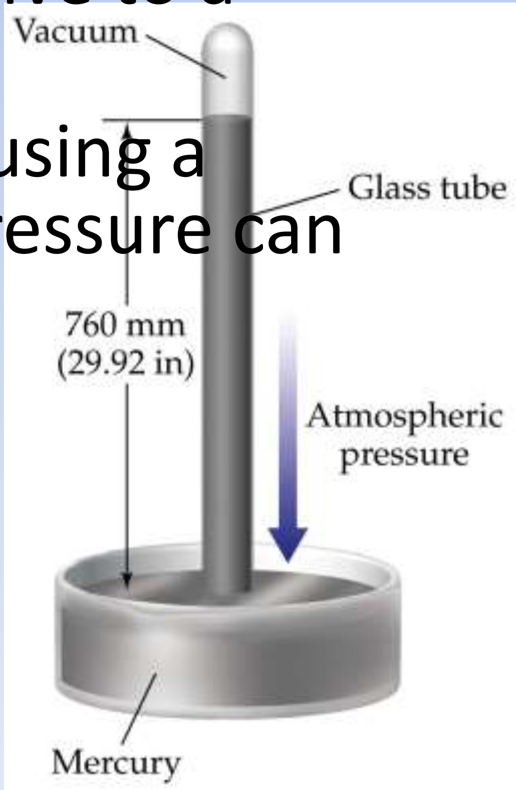
Atmospheric Pressure

- The column of air with cross sectional area of 1 in^2 weighs about 14.7lb and produce a pressure of 14.7lb/in^2 (which is called the atmospheric pressure)



Gage and Absolute Pressure

- Gage Pressure are measured relative to atmosphere pressure, 14.7psi or 14.7 psia or 0 psig.
- Absolute pressure are measured relative to a vacuum.
- Atmosphere pressure are measured using a barometer, where the atmospheric pressure can hold a column of mercury 30 in high



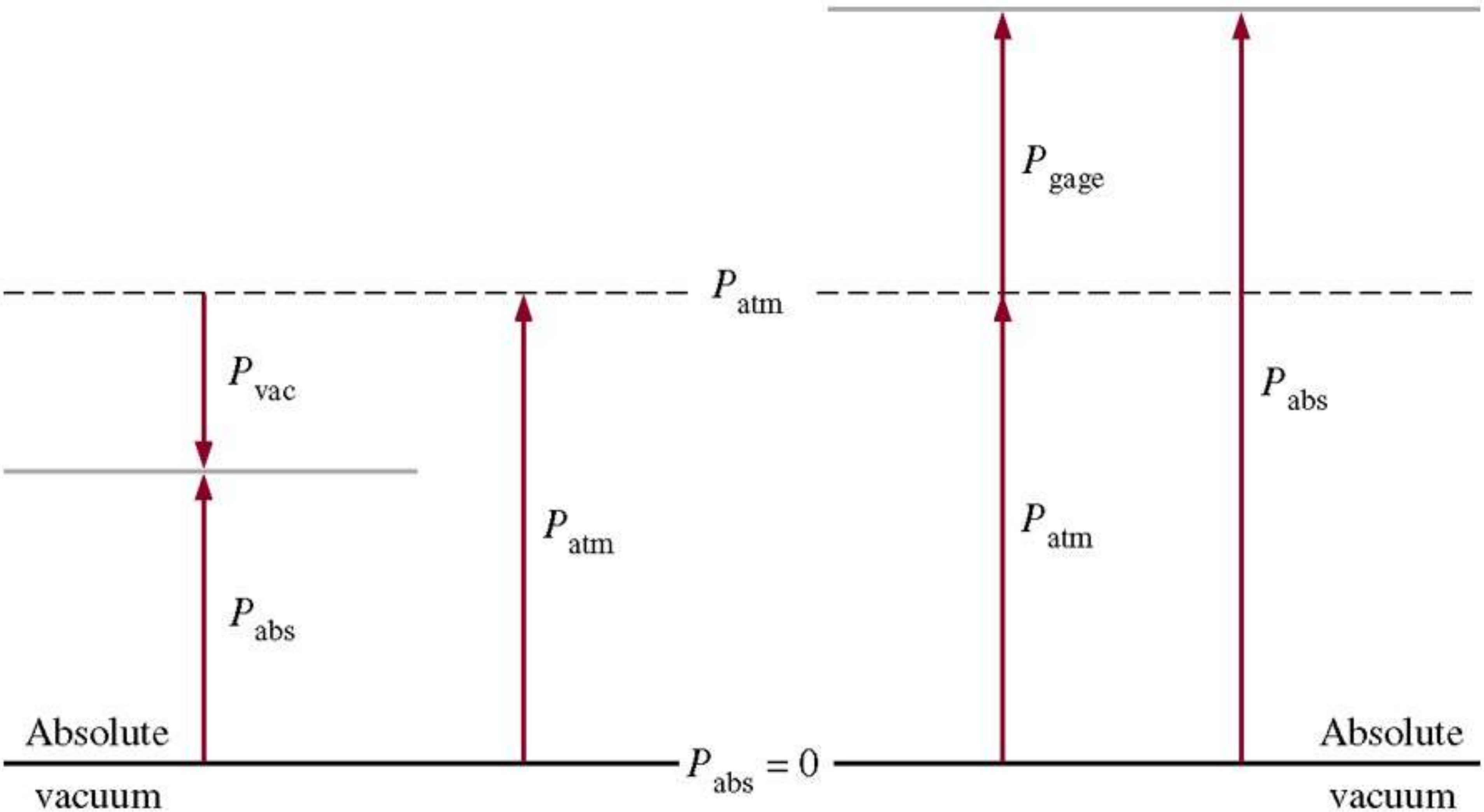
$$\gamma_{mercury} = 0.490 lb / in^3$$

- $$p = \gamma H$$

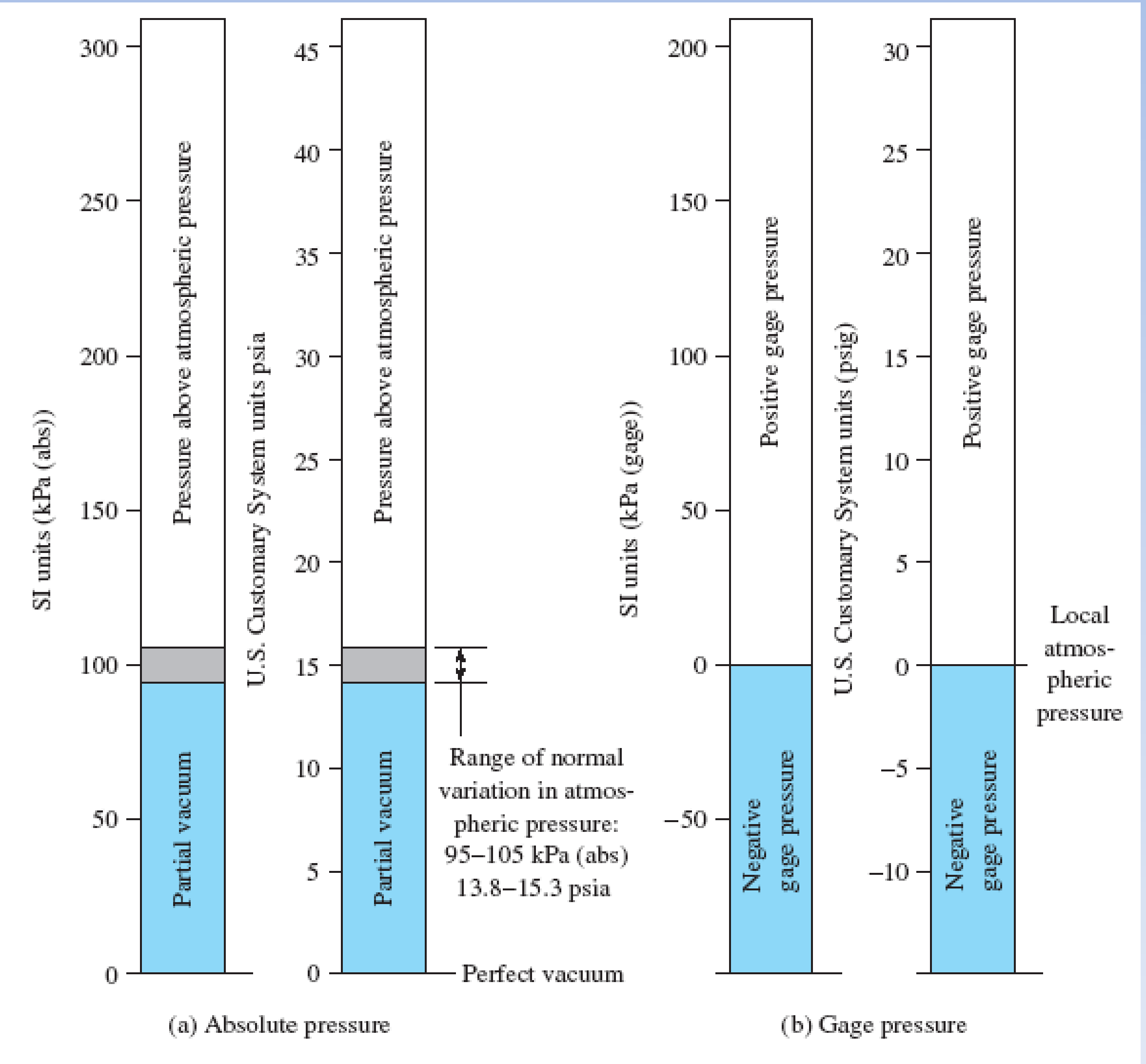
$$14.7 lb / in^2 = 0.490 lb / in^3 \times H(in)$$

$$H = 30 in \text{ of mercury}$$

Pressure Scale



Absolute and Gage Pressure



Pressure

- Pressure:
  - Absolute = Gage + Atmospheric*
  - psia = psig + 14.7 psia
  - *14.7 psia at sea level

Example

- Express 225 kPa (abs) as a gage pressure. The local atmospheric pressure is 101 kPa.
- Express a pressure of -6.2 psig as an absolute pressure

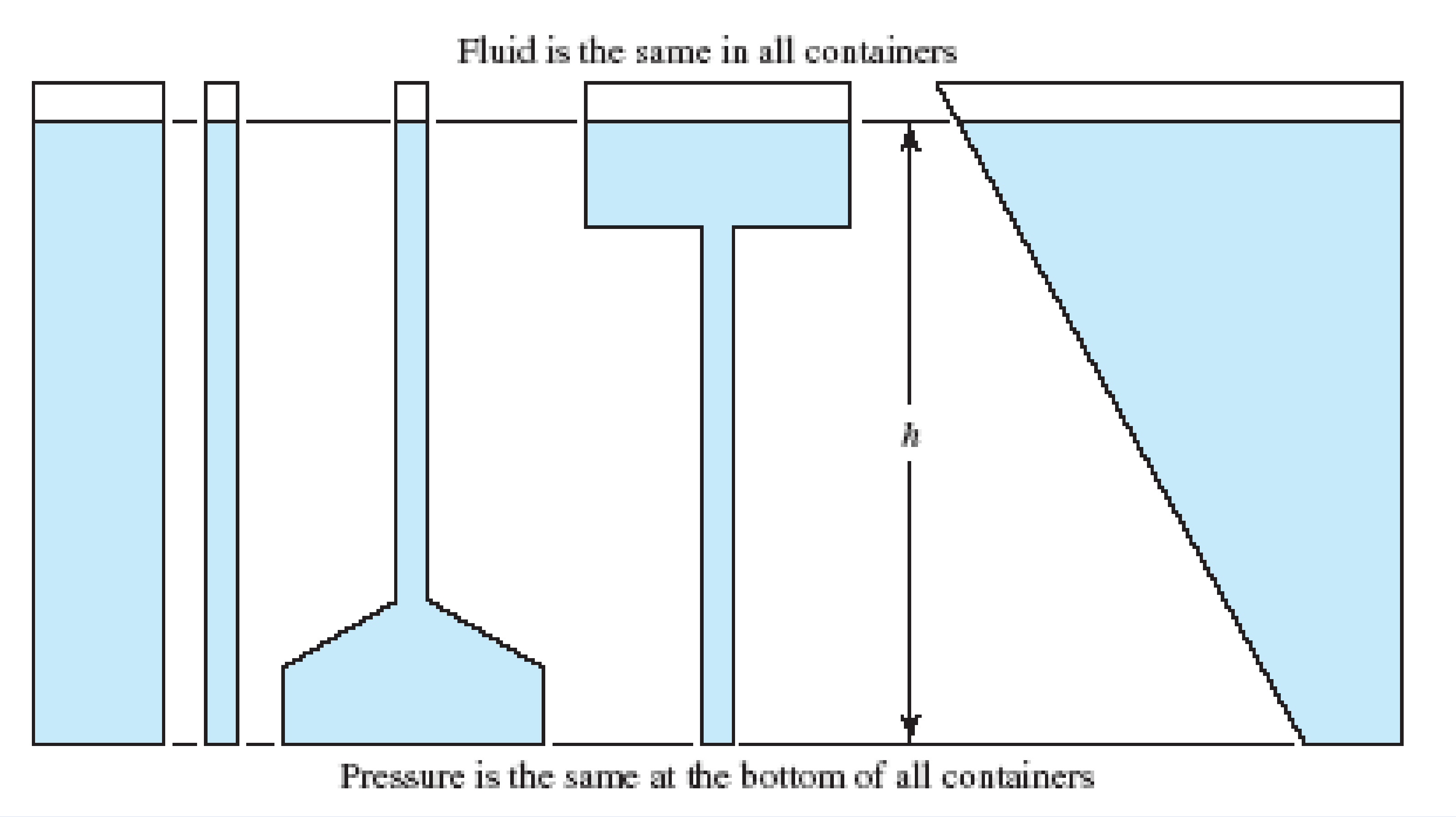
SI Units System

- In the SI metric system the units of measurement is as follow
  - Length is meter (m) =39.4 in =3.28 ft
  - Mass is kilogram (kg)= 0.0685 slugs
  - Force is newton (N) = 0.225lb
  - Time is second (s)
  - Temperature is the degree Celsius (°C) $=(T_F-32)/1.8$ or $T_F=1.8T_C+32$
  - Pressure is pascals (Pa)$=1N/m^2$ =0.000145 psi

Units of Pressure

- 1 bar = 10^5 Pa = 0.1 MPa = 14.5 psi
- 1 atm = 101,325 Pa
- 1 atm = 1.012325 bars
- 1 mm Hg = 0.13333 kPa
- 1 atm = 14.696 psi

Pascal’s Paradox



Pressure Measurement Devices

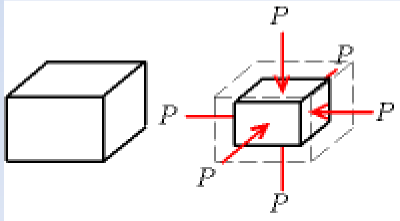
Pressure Transducer

http://www.omega.com/

Bulk modulus

- The bulk modulus (β) of a substance measures the substance's resistance to uniform compression. It is defined as the pressure increase needed to cause a given relative decrease in volume. Its base unit is the pascal.

$$\beta = \frac{-\Delta p}{\Delta V / V}$$



Bulk modulus

- Table bellow shows the Values for bulk modulus for selected liquids at atmospheric pressure and 68°F (20°C).

Liquid	Bulk Modulus (psi)	Bulk Modulus (MPa)
Ethyl alcohol	130 000	896
Benzene	154 000	1 062
Machine oil	189 000	1 303
Water	316 000	2 179
Glycerine	654 000	4 509
Mercury	3 590 000	24 750

Example

- A 10 in^3 sample of oil is compressed in a cylinder until the pressure is increased from 100 to 2000 psi. If the bulk modulus is 250000 psi find the change in volume of the oil.

$$\Delta V = -V\left(\frac{\Delta p}{\beta}\right) = -10\left(\frac{1900}{250000}\right) = -0.076 in^3$$

- This is 0.76% decrease in volume, which shows that oils are highly incompressible.

Viscosity

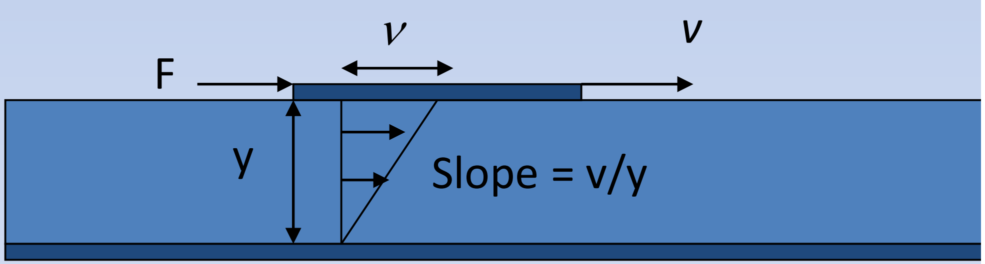
- The most important property of a hydraulic fluid is viscosity.
- It is a measure of fluid resistance to flow.
- Too high viscosity results in:
  - High resistance to flow, sluggish operation.
  - Increase in power consumption due to friction loss
  - Increase in pressure drop through valves and lines
  - High temperatures caused by friction.
- Low viscosity
  - Increase in oil leakage past seals.
  - Excessive wear due to breakdown of oil film between mating parts.

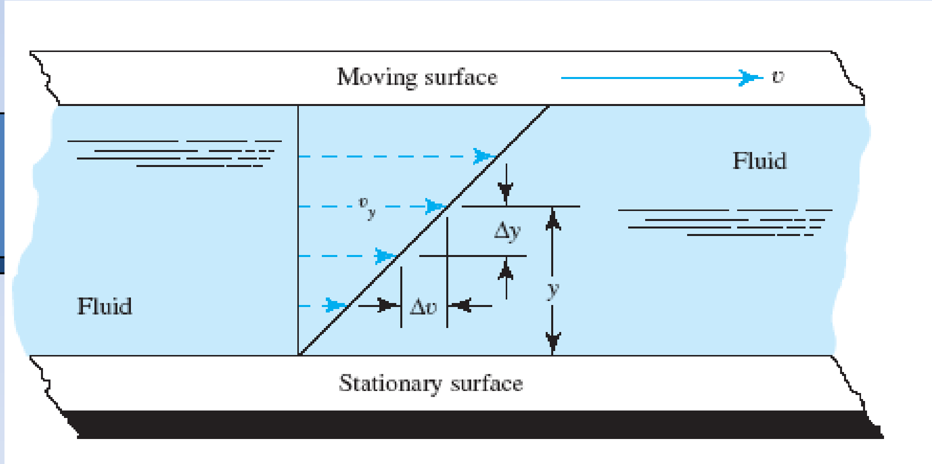
Viscosity

- Dynamic (absolute) Viscosity

  μ = Shear Stress/Slope of velocity profile

  $$\mu = \frac{F/A}{v/y} = \frac{\tau}{v/y}$$ = lb.s/ft^2 or N.s/m^2 or dyn.s/cm^2(poise), 1dyn=10^{-5}N





Units: cP (centipoise)=10^{-2}posie, mPa.sec

Unit of Dynamic Viscosity

Unit System	Dynamic Viscosity Units
International System (SI)	$N{\cdot}s/m^2$, $Pa{\cdot}s$, or $kg/(m{\cdot}s)$
U.S. Customary System	$lb{\cdot}s/ft^2$ or $slug/(ft{\cdot}s)$
cgs system (obsolete)	poise = $dyne{\cdot}s/cm^2$ = $g/(cm{\cdot}s)$ = 0.1 $Pa{\cdot}s$
	centipoise = poise/100 = 0.001 $Pa{\cdot}s$ = 1.0 $mPa{\cdot}s$

Viscosity

- Kinematic Viscosity: absolute viscosity divided by density.

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

- Units: ft^2/s,m^2/s, cm^2/s=stoke, cS (centistokes) mm^2/s

Unit System	Kinematic Viscosity Units
International System (SI)	m^2/s
U.S. Customary System	ft^2/s
cgs system (obsolete)	stoke $= cm^2/s = 1 \times 10^{-4}\ m^2/s$
	centistoke $=$ stoke/100 $= 1 \times 10^{-6}\ m^2/s = 1\ mm^2/s$

Variation of Viscosity with Temperature

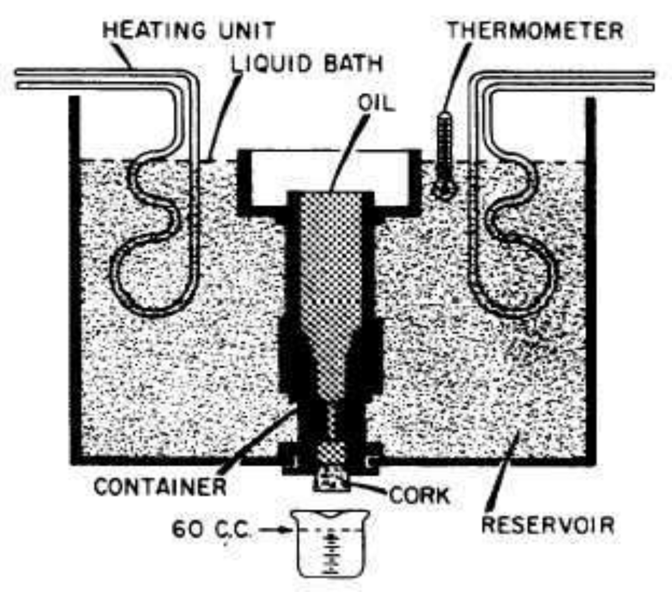
Fluid	Temperature (°C)	Dynamic Viscosity ($N\cdot s/m^2$ or $Pa\cdot s$)
Water	20	1.0×10^{-3}
Gasoline	20	3.1×10^{-4}
SAE 30 oil	20	3.5×10^{-1}
SAE 30 oil	80	1.9×10^{-2}

Saybolt Viscometer

- Saybolt viscometer, measures the time in seconds needed to fill a 60 cm^3 glass capillary through a standard orifice at a specific temperature. This time is called SUS and is related to kinematic viscosity through the following empirical formula

$$v(cS) = \begin{cases} 0.226t - \dfrac{195}{t}, t \le 100\, SUS \\ 0.220t - \dfrac{135}{t}, t > 100\, SUS \end{cases}$$

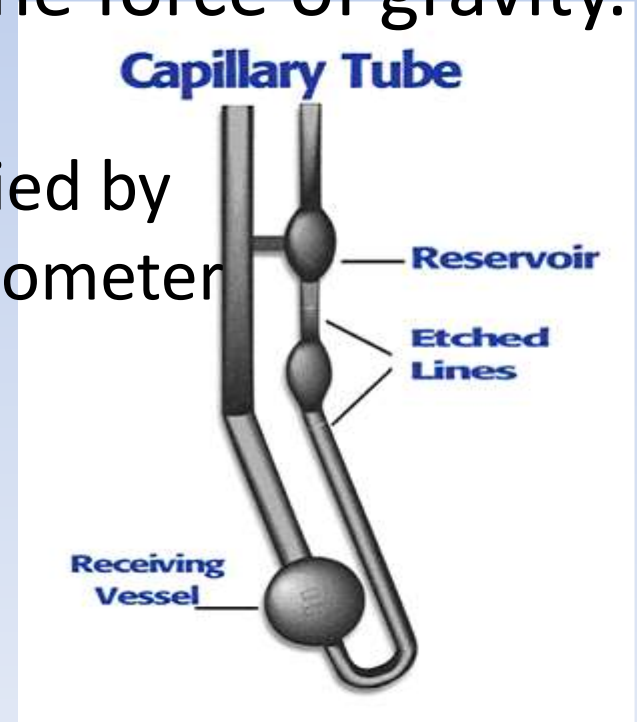
SUS, Saybolt Universal Seconds



Capillary Viscometers

- Capillary tube viscometers measures the time required for a known amount of oil to flow through a small diameter (capillary) tube at a known temperature under the force of gravity.

The time in seconds is then multiplied by the calibration constant for the viscometer to obtain the kinematic viscosity.



Viscosity Index

- The viscosity index (V.I.) of an oil is a number that indicates the effect of temperature changes on the viscosity of the oil.
- A high viscosity index exhibits a small change in viscosity with temperature. A fluid with a low viscosity index exhibits a large change in viscosity with temperature.
- An ideal oil for most purposes is one that maintains a constant viscosity throughout temperature changes.

Viscosity Index Calculation

$$VI = \frac{L - U}{L - H} \times 100$$

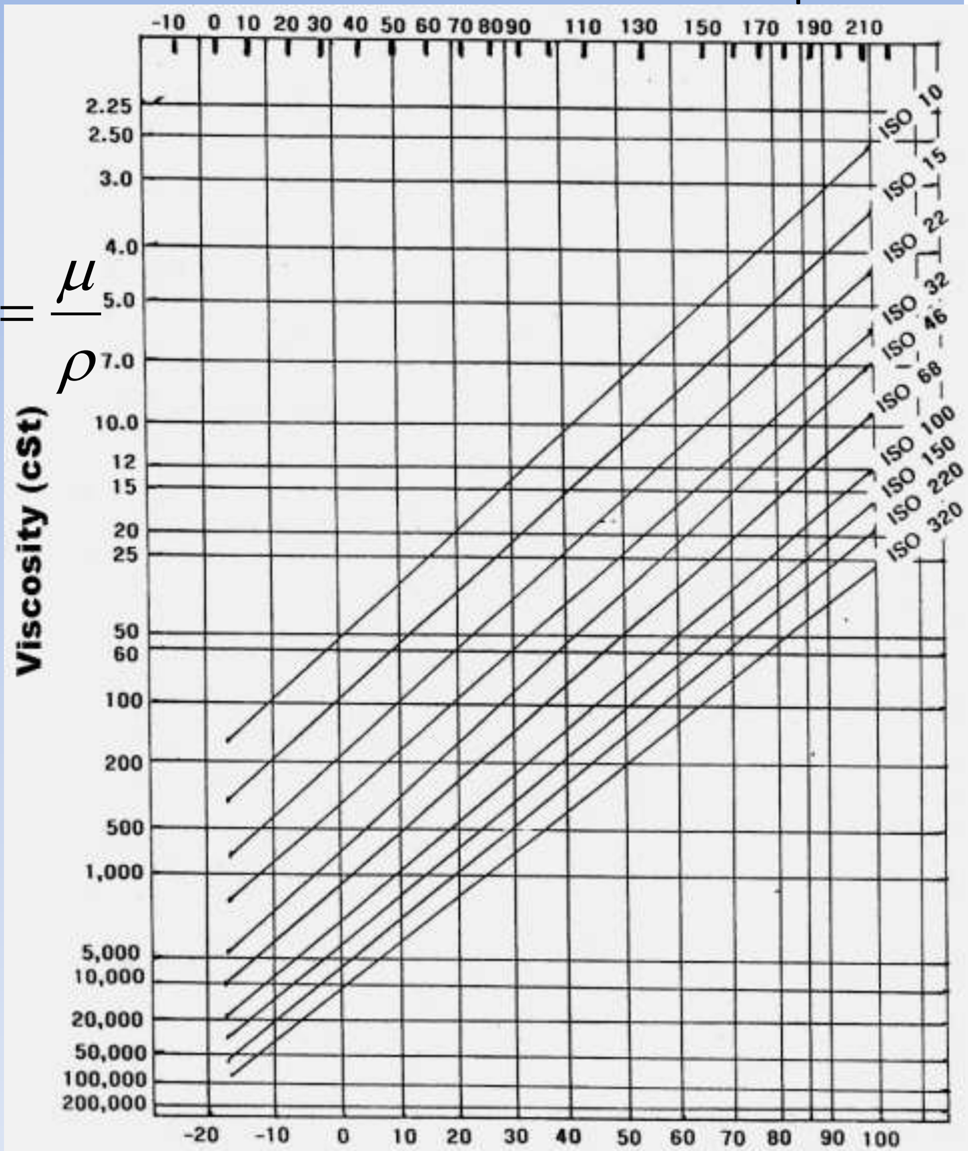
- U= Kinematic viscosity of unknown oil at 40°C
- L= Kinematic viscosity of a standard oil, 0-VI oil, at 40°C
- H= Kinematic viscosity of a standard oil, 100-VI oil, at 40°C

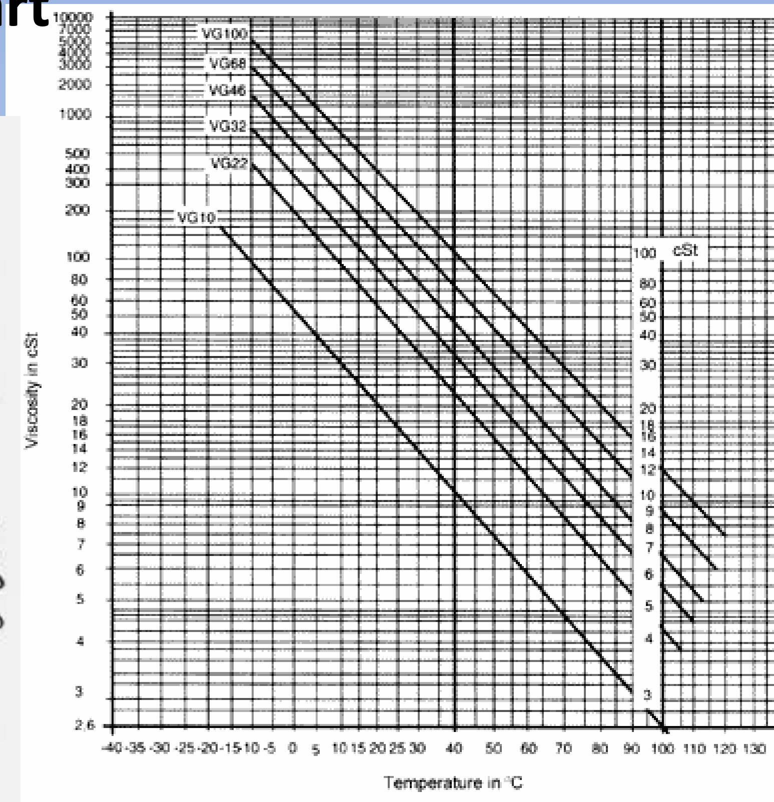
- L and H are reference oils chosen with a unique feature that they have the same viscosity as the unknown oil at 100°C
- All kinematic viscosity values are in the unit of mm^2/s

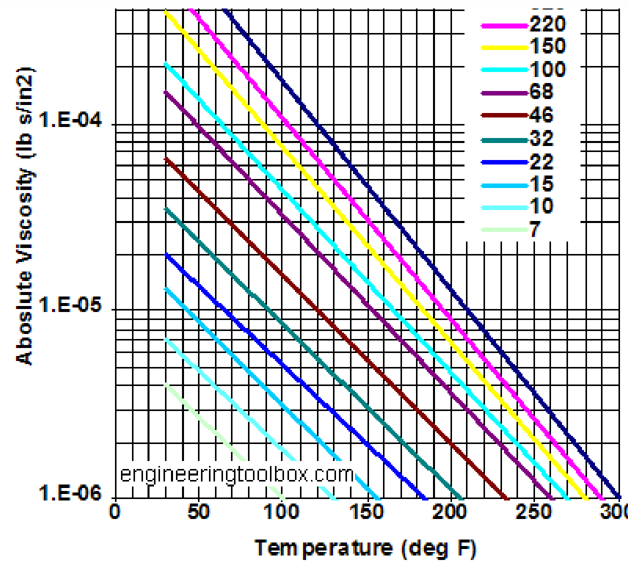
Viscosity Chart

Temp. ° F

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







Temp.° C

http://www.klassenhydraulics.com/Reference/viscositychart.htm

SAE Viscosity Grade

- SAE International has developed a rating

SAE Viscosity Grade	Low Temperature—Dynamic Viscosity: Cranking Condition* (cP) Max. at (°C)	Low Temperature—Dynamic Viscosity: Pumping Condition# (cP) Max. at (°C)	High-Temperature Kinematic Viscosity at 100°C (cSt)+: Min.	High-Temperature Kinematic Viscosity at 100°C (cSt)+: Max.
0W	6200 at −35	60 000 at −40	3.8	—
5W	6600 at −30	60 000 at −35	3.8	—
10W	7000 at −25	60 000 at −30	4.1	—
15W	7000 at −20	60 000 at −25	5.6	—
20W	9500 at −15	60 000 at −20	5.6	—
25W	13 000 at −10	60 000 at −15	9.3	—
20	—	—	5.6	<9.3
30	—	—	9.3	<12.5
40	—	—	12.5	<16.3
40	—	—	12.5	<16.3
50	—	—	16.3	<21.9
60	—	—	21.9	<26.1

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Note: 1 cP = 1 mPa·s ; 1 cSt = 1 mm^2/s

* Using ASTM Standard D 5293

\# Using ASTM D 4684

\+ Using ASTM D 445

◇ Using ASTM D 4683, D 4741, or D 5481

⊥ When used in these multiviscosity grades: 0W-40, 5W-40, 10W-40

When used in single-grade SAE 40 and in these multiviscosity grades: 15W-40, 20W-40,

SAE Viscosity Grade	Maximum Temperature for Dynamic Viscosity of 150 000 cP* (°C)	Kinematic Viscosity at 100°C (cSt)#: Min.	Kinematic Viscosity at 100°C (cSt)#: Max.
70W	−55	4.1	—
75W	−40	4.1	—
80W	−26	7.0	—
85W	−12	11.0	—
80	—	7.0	<11.0
85	—	11.0	<13.5
90	—	13.5	<24.0
140	—	24.0	<41.0
250	—	41.0	—

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Note: 1 cP = 1 mPa·s; 1 cSt = 1 mm^2/s

* Using ASTM D 2983

\# Using ASTM D 445

ISO Viscosity Grade

- The standard designation includes the prefix ISO VG followed by a number representing the nominal kinematic viscosity in temperature of 40°C.

Grade ISO VG	Kinematic Viscosity at 40°C (cSt) or (mm^2/s)		
	Nominal	Minimum	Maximum
2	2.2	1.98	2.40
3	3.2	2.88	3.52
5	4.6	4.14	5.06
7	6.8	6.12	7.48
10	10	9.00	11.0
15	15	13.5	16.5
22	22	19.8	24.2
32	32	28.8	35.2
46	46	41.4	50.6
68	68	61.2	74.8
100	100	90.0	110
150	150	135	165
220	220	198	242
320	320	288	352
460	460	414	506
680	680	612	748
1000	1000	900	1100
1500	1500	1350	1650
2200	2200	1980	2420
3200	3200	2880	3520

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Hydraulic Fluid for Fluid Power System

- Fluid power systems use fluids under pressure to actuate linear or rotary devices used in construction equipment, industrial automation systems, agricultural equipment, aircraft hydraulic systems, automotive braking systems, and many others.
- There are several types of hydraulic fluids in common use, including

1. Petroleum oils
2. Water–glycol fluids
3. High water-based fluids (HWBF)
4. Silicone fluids
5. Synthetic oils

Temperature scales

Kelvin Scale

- Triple point of water at P = 1 atm

  T = 273.15 K
- Steam point of water at P = 1 atm

  T = 373.15 K

Celsius Scale

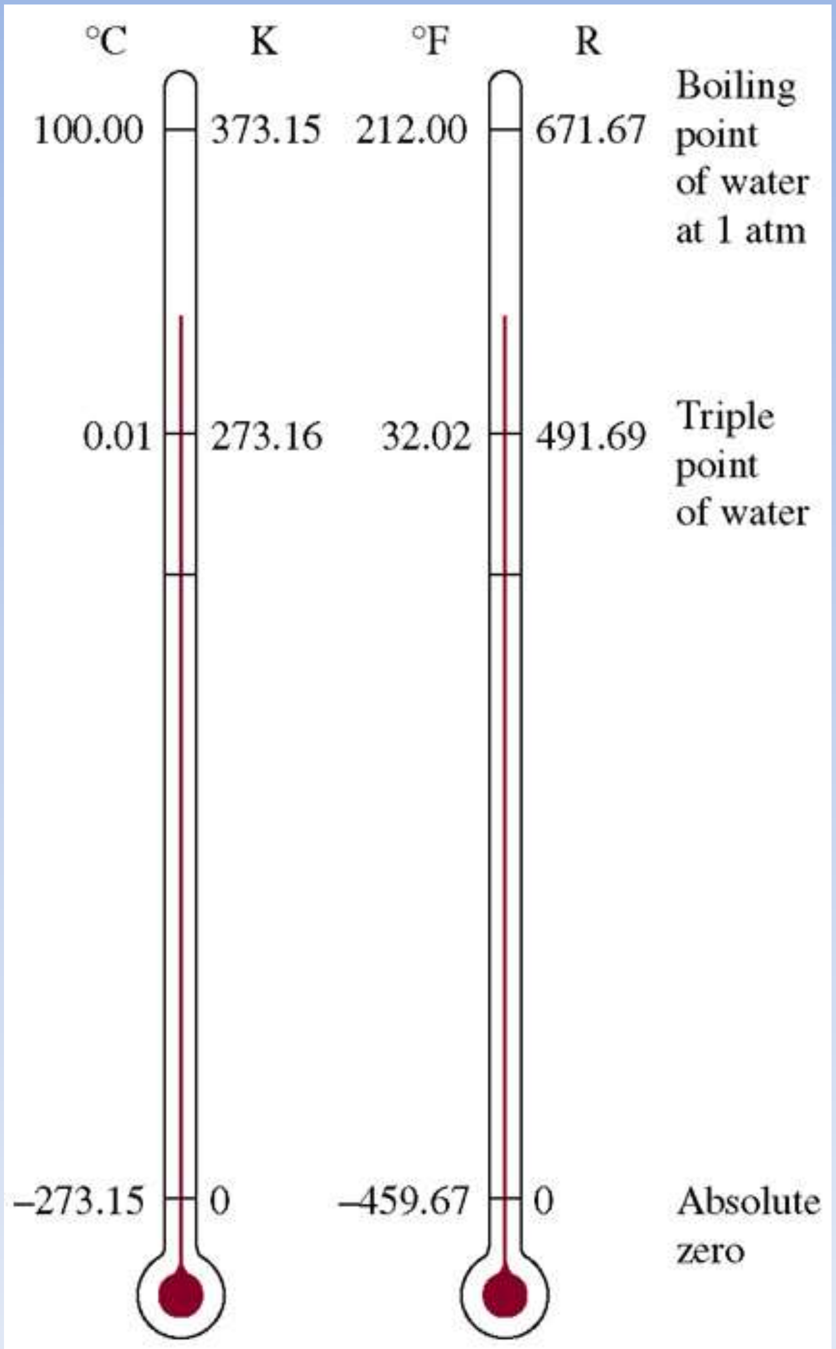
- Triple point of water at P = 1 atm

  T = 0 ^{o}C
- Steam point of water at P = 1 atm

  T = 100 ^{o}C
- 1 ^{o}C = 1 K

Temperature

- T(K)=T(° C) + 273.15, K stands for Kelvin
- T(R) =T(° F)+ 459.67, R stands for Rankine


°C
K
°F
R
100.00
373.15
212.00
671.67
Boiling point of water at 1 atm
0.01
273.16
32.02
491.69
Triple point of water
–273.15
0
–459.67
0
Absolute zero

Example

- Express 25 °C in K, °F and R.
- Express 425 R in °C, °F and K.
- If ΔT is 25 °C, express the same temperature difference in K, °F and R.

Temperature Measurement Devices

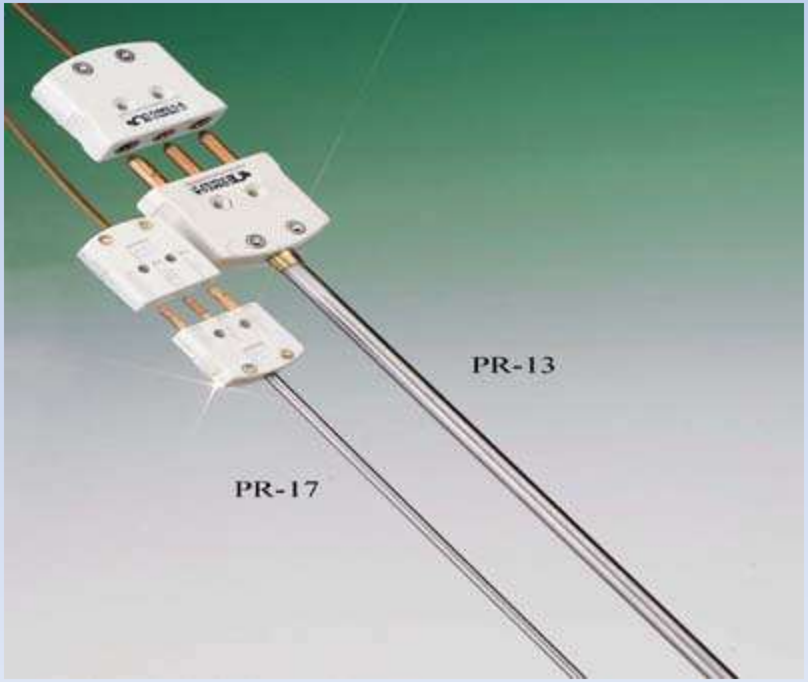
- Glass Thermometer: ± 1.0 ° C, cheapest
  - High precision (± 0.1 ° C) thermometer can cost up to $400 or more
- Thermocouple Wire: ± 0.5 ° C, but inexpensive
- Thermistor and RTD: ± 0.1 ° C, but expensive

Temperature Measurement Devices

- Thermocoupe Wire: ± 0.5 ° C, but inexpensive
- Thermistor and RTD: ± 0.1 ° C, but expensive

Thermocouple Probe:

RTD and Thermistor Probe:



http://www.omega.com/

Examples

- Find the weight of a body having a mass of 4 slugs?
- W=mg=4 slugs * 32.2 ft/s^2=129 lb
- If this body volume is 1.8 ft^3 Find its specific weight?

$$\gamma = \frac{W}{V} = \frac{129lb}{1.8\,ft^3} = 71.6lb/\,ft^3$$

- Find the density of the body?

$$\rho = \frac{m}{V} = \frac{4slugs}{1.8\,ft^3} = 2.22slugs/\,ft^3$$

$$or$$

$$\rho = \frac{\gamma}{g} = \frac{71.6lb/\,ft^3}{32.2\,ft/s^2} = 2.22slugs/\,ft^3$$

Examples

- Air at 68°F and under atmospheric pressure has a specific weight of 0.0752lb/ft^3 find its specific gravity?

$$(SG)_{air} = \frac{\gamma_{air}}{\gamma_{water}} = \frac{0.0752 lb / ft^3}{62.4 lb / ft^3} = 0.00121$$

- Find the pressure on a shin diver who descendent to 60 ft in fresh water?

$$p = \gamma \times H = 62.4 (lb / ft^3) \times 60\, ft = 3744 lb / ft^2 (psf)$$
$$p = 0.0361 (lb / in^3) \times (60 \times 12)(in) = 26.0 lb / in^2 (psi)$$

Examples

- How high would be a the barometer tube if water is used instead of mercury?

$$p = \gamma H$$
$$14.7(lb/in^2) = 0.0361(lb/in^3) \times H(in)$$
$$H = 407in = 34\,ft$$

- Convert a -5psi pressure to an absolute pressure?
- absolute pressure= -5.0 + 14.7 = 9.7 psia

Example

- Find the absolute pressure on the skin of the driver 60 ft deep in fresh water?
- Absolute pressure= 26.0 + 14.7 = 40.7 psia
- An oil has a specific weight of 56 lb/ft^3 , determine the specific weight in N/m^3?

$$\gamma(\frac{N}{m^3}) = \gamma \frac{lb}{ft^3} \times \left(\frac{N}{0.225 lb}\right) \times \left(\frac{3.28\, ft}{m}\right)^3 = 157\gamma \frac{lb}{ft^3}$$

$$= 157 \times 56 = 8700 N / m^3$$

Examples

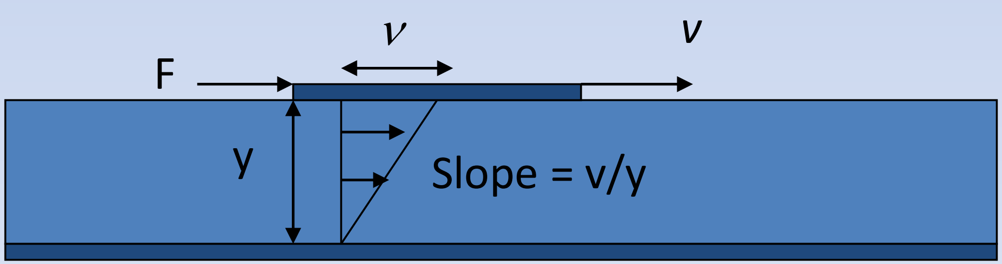
- At what temperature are the Fahrenheit and Celsius values equal?
- T(°F)=T(°C)
- 1.8T(°C)+32=T(°F)=T(°C)
- T(°C)=-32/0.8=-40°C
- Hence -40°C =-40°F

Examples

- An oil has a viscosity of 230 SUS at 150 °F find the corresponding viscosity in centistokes and centipoise. The specific gravity of the oil is 0.9?
- $$v(cS) = 0.220t - \frac{135}{t} = (0.220)(230) - \frac{135}{230} = 50cS$$
$$\mu(cP) = SG \times v(cS) = 0.9 \times 50 = 45cP$$

Example

- In the figure bellow the moving plate is 1 m^2 and the oil film is 5 mm thick. A 10 N force is required to move the plate at a velocity of 1m/s> find the absolute viscosity of the oil in terms of N.s/m^2 and cP.
- $\mu = Fy/vA = 10N*0.005m/(1m/s*1m^2) = 0.05N.s/m^2$
- $\mu = 10*10^5 dyn*0.5cm /(100m/s*100cm*100cm) = 0.5 dyn.s/cm^2 = 0.5 poise = 50cP$



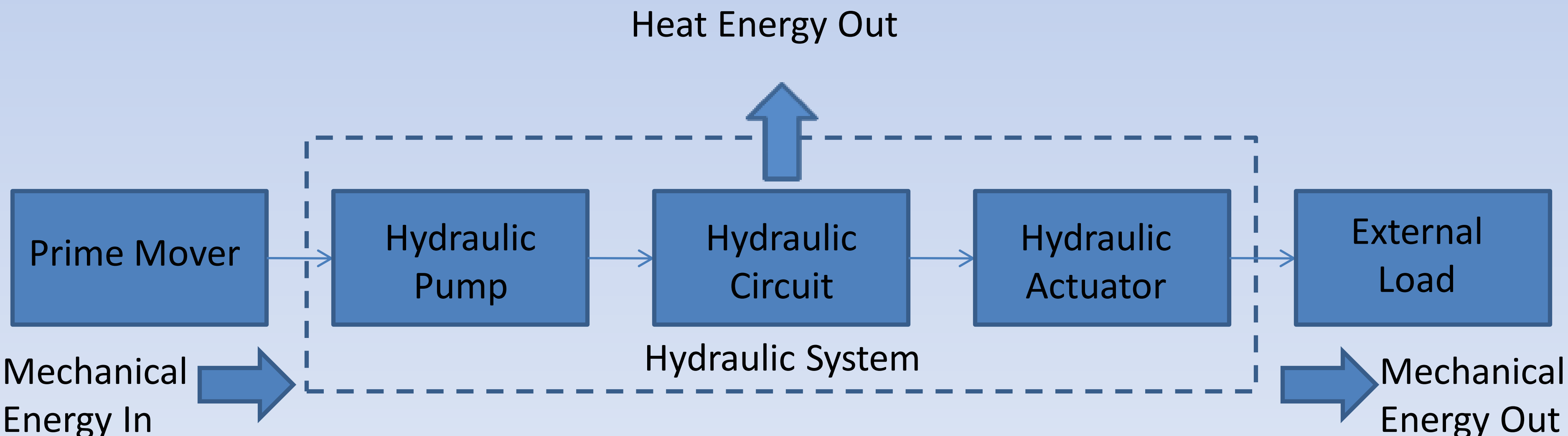
Energy and Power in Hydraulic Systems

Chapter 3

Dr. Suleiman BaniHani

Energy

- Energy is defined as the ability to perform work.
- Hydraulic systems are used to transfer energy.



Mechanical Energy In = Mechanical Energy Out + Energy Loss

Review of Mechanics

- Newton's Laws of Motion
  - Every object in a state of uniform motion tends to remain in that state of motion unless an external force is applied to it.
  - A body of mass m subject to a net force '*F*' undergoes an acceleration '*a*' that has the same direction as the force and a magnitude that is directly proportional to the force and inversely proportional to the mass.
  - The mutual forces of action and reaction between two bodies are equal, opposite and collinear.

Linear Motion

- If a body experiences linear motion, it has a linear velocity, which is defined as the distance traveled divided by the corresponding time.

$$v = \frac{s}{t}$$

- *s* = distance (in, ft, m)
- *t* = time (s, min)
- *v* = velocity (in/s, in/min, ft/s, m/s)

Linear Motion Cont.

- If the body's velocity changes, the body has an acceleration, defined as the change in velocity divided by the corresponding time.
- According to Newton's law a force is needed to change velocity.

$$a = \frac{\Delta v}{\Delta t}$$

$$F = ma$$

- F = force (lb, N)
- a = acceleration (in/s^2, ft/s^2)
- m = mass (slugs)

Cont.

- If a force acts on a body and moves the body through a distance in the direction of the applied force then work has been performed on the body

$$W = FS$$

  - *F* = force (lb)
  - S = distance (in, ft)
  - W = work (in.lb , ft.lb)

- Power is the rate of doing work or expanding energy

$$power = \frac{FS}{t} = Fv$$

F = force (lb)
v = velocity (in/s, ft/s)
Power (in.lb/s) or hp=550 ft.lb/s = 33000 ft.lb/min

HorsePower

- Horsepower was created by James Watt.

- A horse could raise a 150-lb weight at an average velocity of 3.67ft/s.
- The rate of work done (power = 150 lb * 3.67 ft/s = 550 ft.lb/s = 1 hp

Example

- A person exerts a 30 lb force to move a hand truck 100 ft in 60 s.

a. How much work is done?

b. What is the power delivered by the person?

Sol.

a. W = FS = (30 lb)(100 ft) = 3000 lb.ft

b. Power = FS/t = (3000 lb.ft)/ (60 s) = 50 lb.ft/s

HP = (50 lb.ft/s)/(550 lb.ft/s / 550 hp) = 0.091 hp

Angular Motion

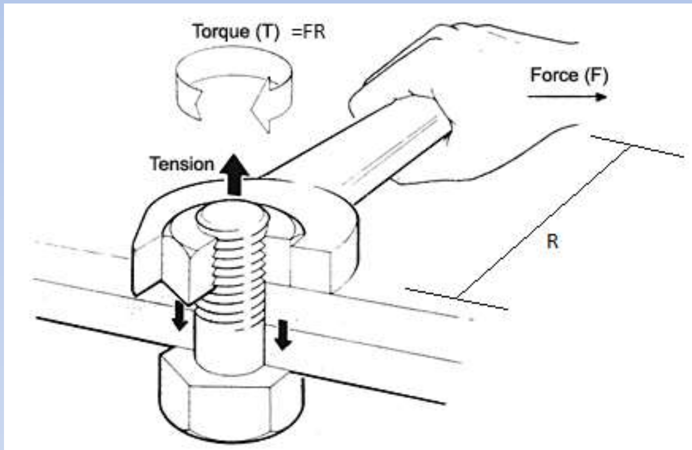
- The turning or twisting force applied to a shaft
- A force is applied to a wrench,

  The force has a moment arm R

  relative to the center of the nut.

  Thus, the force create a torque T

  about the center of the nut.



- The moment arm R is measured from the center of the nut perpendicular to the line of action of the force.

$$T = FR$$

$$F = force\,(lb)$$

$$R = moment\,arm\,(in \text{ or } ft)$$

$$T = torque\,(in.lb \text{ or } lb.ft)$$

Angular Motion Cont.

- Power: rate of doing work

$$\mathrm{HP} = \frac{TN}{63000}$$

$$T = \text{torque } (lb.in)$$

$$N = \text{rotation speed (rpm)}$$

$$\mathrm{HP} = \text{torque horsepower or brake horsepower}$$

- Ex. How much torque is delivered by a 2-hp , 1800 rpm hydraulic motor?
- Sol.

$$\mathrm{HP} = \frac{TN}{63000}$$

$$2 = \frac{T(1800)}{63000} \Rightarrow T = 70\ in.lb$$

Efficiency

- Efficiency is defined as the output power divided by input power.

$$\eta = \frac{output\ \ power}{input\ \ power}$$

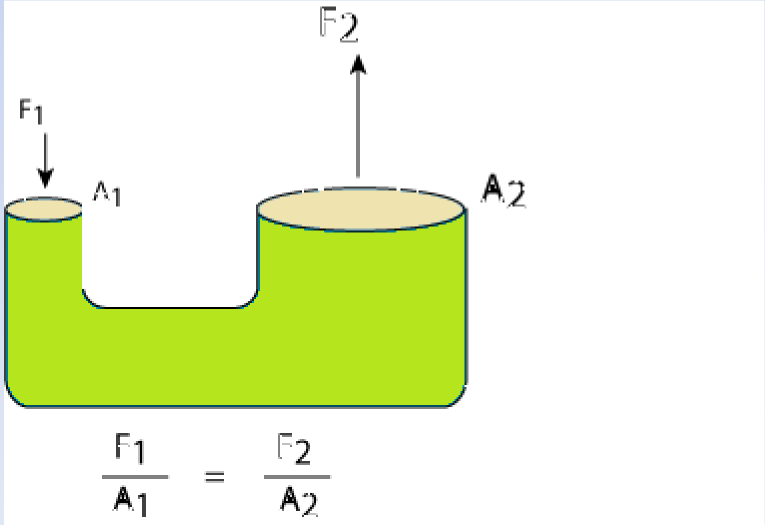
- Ex. An elevator raises 3000 lb through a distance of 50 ft in 10 s. If the efficiency of the system is 80%. How much input horsepower is required by the elevator hoist motor?
- Sol. $output\ \ power = \frac{(3000lb)(50\,ft)}{10s} = 15000\,ft.lb/s \Rightarrow \frac{15000}{550} = 27.3hp$

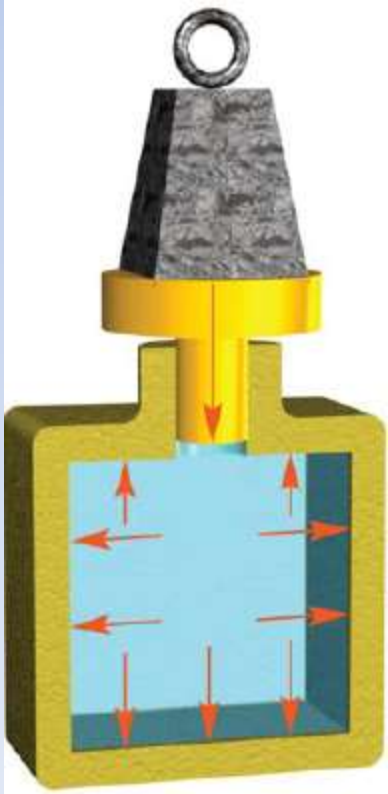
$$\eta = \frac{output\ \ power}{input\ \ power} \Rightarrow 0.80 = \frac{27.3hp}{input\ \ power}$$

$$input\ \ power = 34.1hp$$

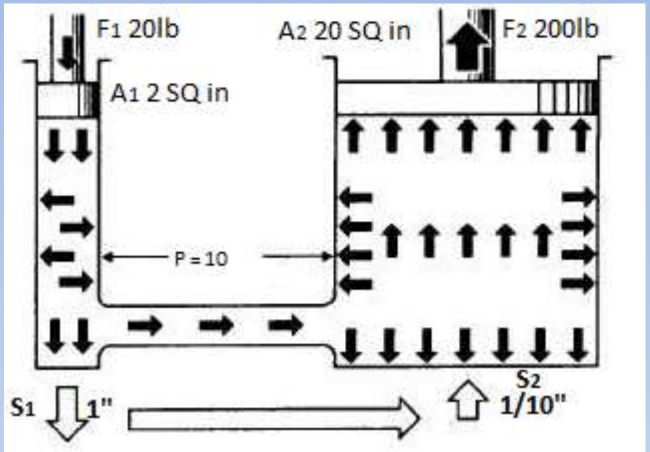
Pascal’s Law

- Pascal’ law can be stated as “*pressure applied to a confined fluid is transmitted undiminished in all directions through the fluid and acts perpendicular to the surface in contact with the fluid*”.





Simple Hydraulic Jack



- $P = F/A = 20$ lb /2 in^2 = 10 psi
- Pascal's law states that the pressure is constant.
- $P_1 = P_2 = P = 10$ psi
- $F_2 = P \times A_2 = 10$ (psi) x 20 in^2 = 200 lb

$$\frac{F_1}{A_1} = \frac{F_2}{A_2} \quad \text{or} \quad \frac{F_2}{F_1} = \frac{A_2}{A_1}$$

- Assuming incompressible liquid, cylinderical volume displacement are equal
  - $V_1 = V_2$ Hence $A_1S_1=A_2S_2$ or $\frac{S_2}{S_1} = \frac{A_1}{A_2} = \frac{F_1}{F_2}$, and $F_1S_1 = F_2S_2$

Example

A Hydraulic jack has the following data

A_1=2in^2, A_2=20in^2, S_1=1in, and F_1=100lb

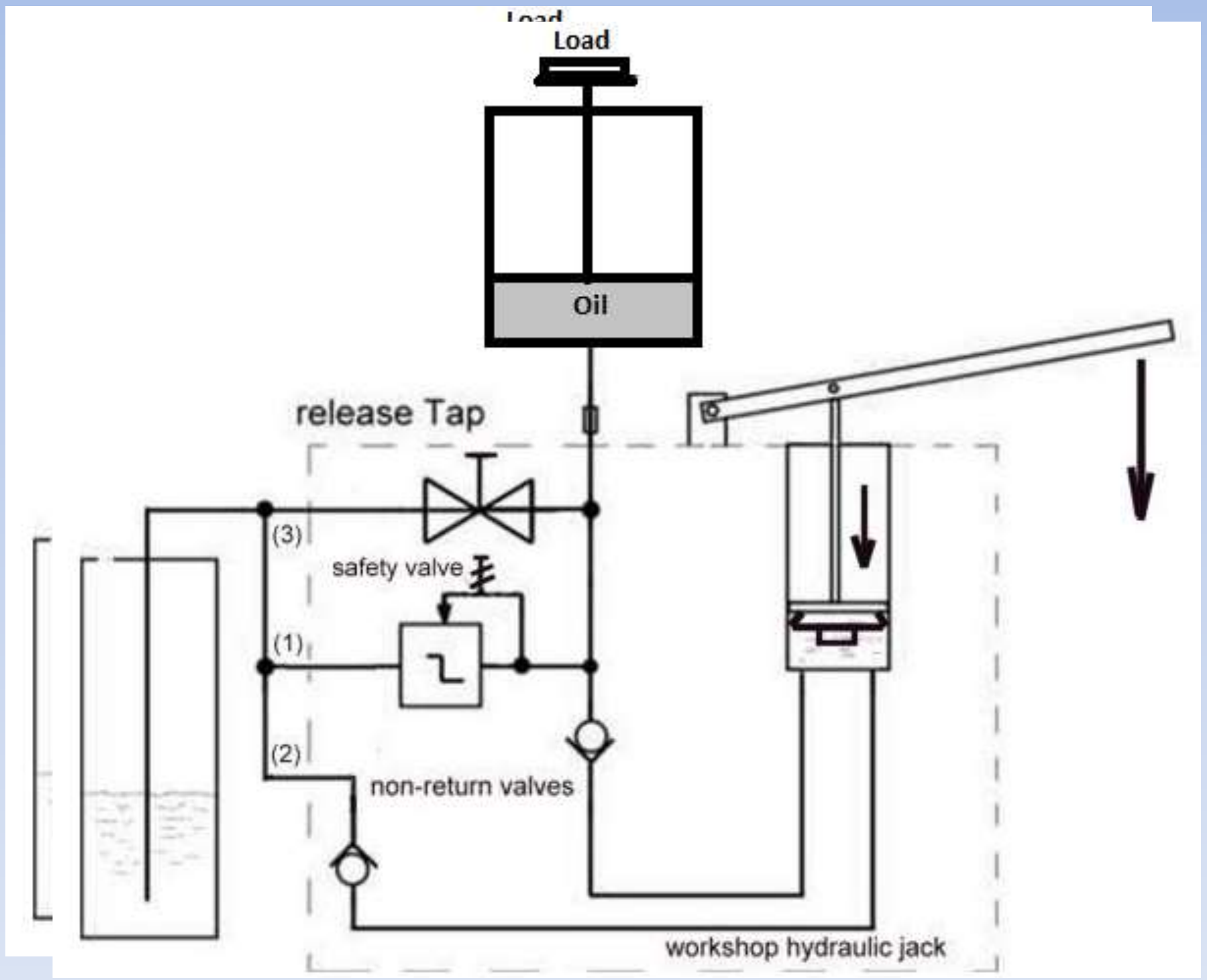
Find;

F_2, S_2, input energy, and output energy?

Sol.

- P= F_1/A_1=100lb/2in^2=50psi
- F_2=P x A_2=50(psi)x20(in^2)=1000lb
- $V_1=V_2=A_1S_1$=2(in^2)x1(in)=2in^3=A_2S_2=20(in^2)S_2
- S_2=0.1in
- Input Energy = F_1S_1=100lbx1in=100lb.in
- Output Energy=F_2S_2=1000lbx0.1in=100lb.in

Hand Operated Hydraulic Jack



Example

- An operator using a hydraulic jack like the one shown before, makes one complete cycle per second, intake and power. The pump cylinder has a 1in diameter piston and the load cylinder has a 3.25 in diameter piston. If the average hand force is 25lb during the power stroke. The hand lever is 2in from the pump to the pivot, and 6 in from the pump to the hand force input.
- Find;

1. How much load can be lifted
2. How may cycles are required to lift the load 10 in assuming no leakage? The pump piston has 2 in stroke.
3. What is the output HP assuming 100% efficiency?
4. What is the output HP assuming 80% efficiency?

Example Cont.

- Sol.
- $M_1 = M_2$,Hence F_{pump}x 2 in = F_{hand}x8 in

  F_{pump}=25x8/2=100lb

  $P=F/A=F_{pump}/A_{pump}$=100lb/((π/4)(1^2)in^2)=127psi

  $F_{load}=PxA_{load\ piston}$=127psi x ($\pi$/4)(3.25)^2 in^2=1055lb
- $(AxS)_{pump\ piston}$ x (no. of cycles)= $(AxS)_{load\ piston}$

  π/4(1)^{2}in^2 x 2 in x (no. of cycles) = π/4(3.25)^2 in^2 x 10 in

  1.57 in^3 x (no. of cycles) = 82.7 in^3

  no. of cycles=52.7
- Output Power =FS/t=1055lb x (10/12ft)/52.7s = 16.7 ft.lb/s

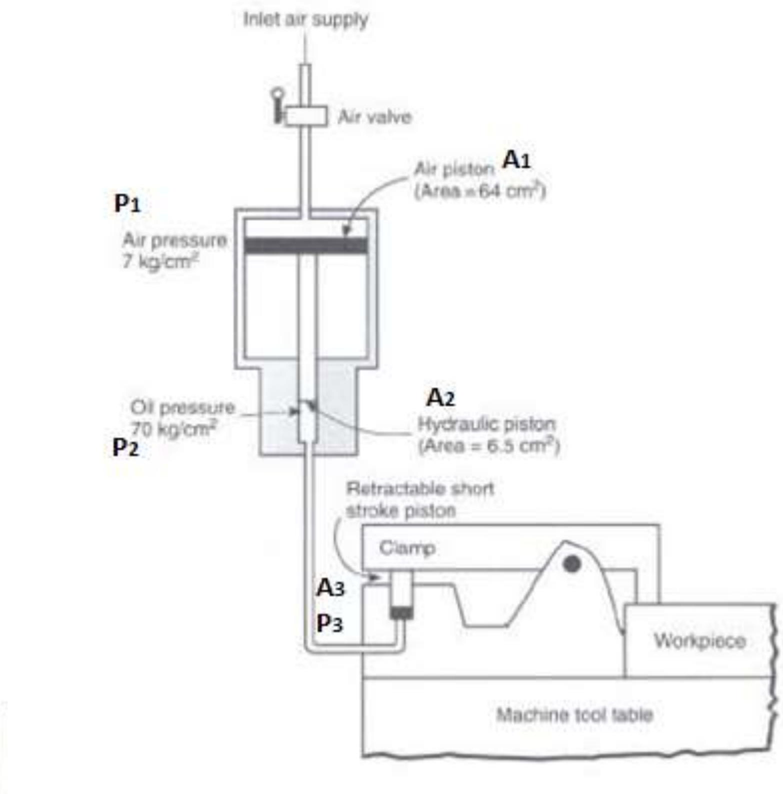
  HP=16.7/550=0.030hp
- 80% efficiency HP = 0.030x0.80=0.24hp

Air to Hydraulic Pressure Booster

Example:
Inlet pressure =100psi, air piston area =20in^2,
Oil piston area = 1in^2 , load piston area = 25 in^2
Find the load carrying capacity?

Sol.

Booster input force = Booster output force
100 psi x 20 in^2 = P_2 x 1 in^2
P_2= 2000 psi
P_2=P_3= 2000psi
F_{load}=2000 psi x 25 in^2 = 50,000 lb



...ing application of an air-to-hydraulic booster

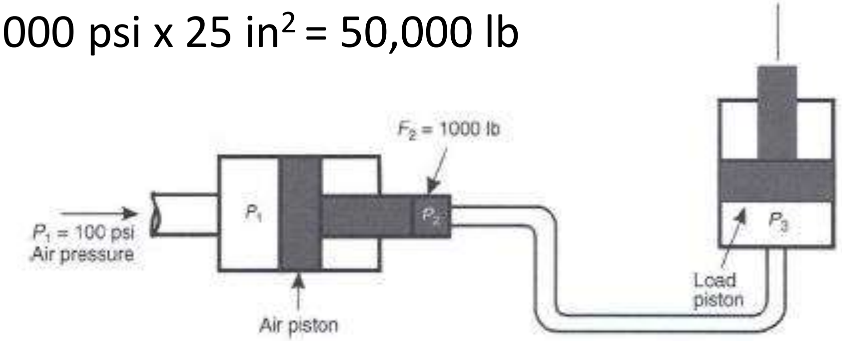
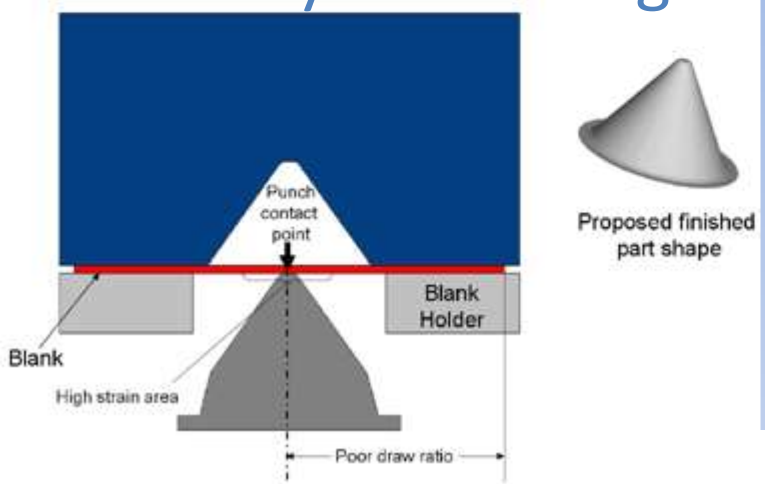
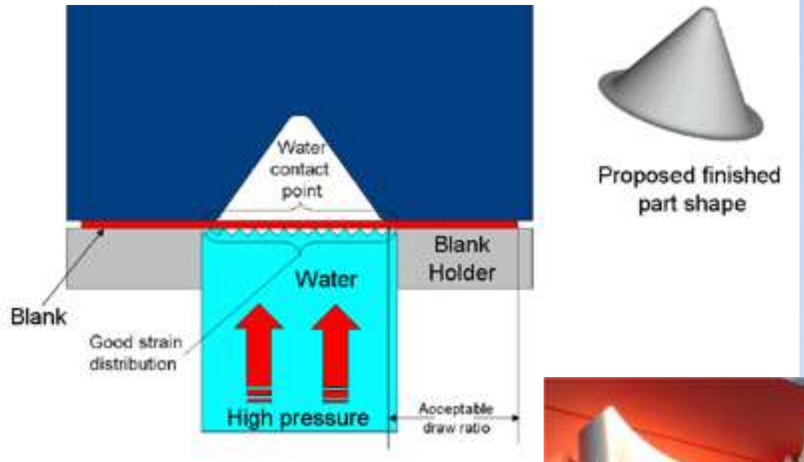


Figure 2.7(a)
An air-to-hydraulic system

Hydroforming

Sheet hydroforming





Pontiac's sheet hydroforming display at the 2006 Detroit International Auto Show. Shown are the Solstice hood outer and decklid outer.

- **The advantages of sheet hydroforming are as follows:**

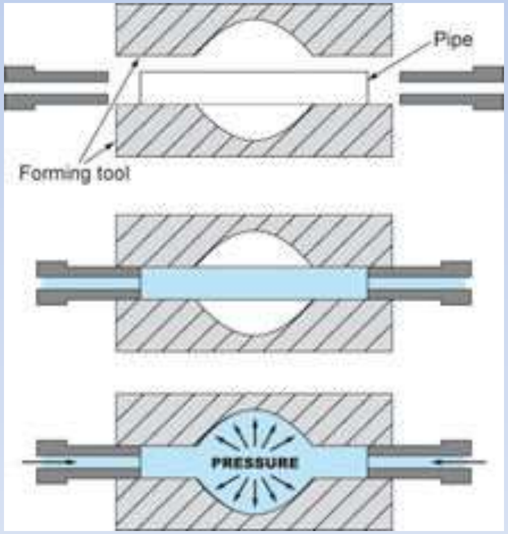
1. **Requires fewer operations to make certain part geometries**
2. **Does not require lower or upper draw punch or cavity**
3. **Uses water, a widely available resource**
4. **Forces material to distribute stretch or strain more evenly**
5. **Reduces springback**
6. **Reduces material consumption**
7. **Forms higher strength materials**

- **The disadvantages of sheet hydroforming are as follows:**

1. **Requires expensive equipment**
2. **Cycle times are generally poor**
3. **Operators often get wet**

Tube Hydroforming

- A tube will be placed in a die, then water will be put in under so high pressure that the tube will form itself according to the die.



Conservation of Energy
Bernoulli's Equation

- Energy cannot be created or destroyed, just transformed.
- The total energy of a system remains constant.
- Three forms of energy in fluid system:
  - Potential due to elevation
  - Kinetic due to velocity
  - Flow energy or pressure energy due to pressure

Potential Energy

- Due to the elevation of the fluid element

$$EPE = WZ = mgz$$

Where,

W= weight of fluid element

z = elevation with respect to a reference level

Pressure Potential Energy

- Flow work or pressure energy
- Amount of energy necessary to move a fluid element across a certain section against pressure

$$PPE = W\frac{P}{\gamma} = m\frac{P}{\rho}$$

Where,

p = pressure on the fluid element

Kinetic Energy

- Due to the velocity of the fluid element

$$KE = W\frac{v^2}{2g} = m\frac{v^2}{2}$$

Where,

v = average velocity of the fluid element

Total Energy and Conservation of Energy Principle

- E = PPE + EPE + KE

$$E = W\frac{P}{\gamma} + W \cdot z + W\frac{v^2}{2g}$$

- Two points along the same pipe:

  $E_1 = E_2$

- Bernoulli's Equation:

The total energy of the system is constant, unless energy is added using a pump or removed using a motor or friction, but it can be transformed from one form to another.

$$\frac{Wp_1}{\gamma} + Wz_1 + \frac{Wv_1^2}{2g} = \frac{Wp_2}{\gamma} + Wz_2 + \frac{Wv_2^2}{2g}$$

$$\boxed{\frac{P_1}{\gamma} + z_1 + \frac{v_1^2}{2g} = \frac{P_2}{\gamma} + z_2 + \frac{v_2^2}{2g}}$$

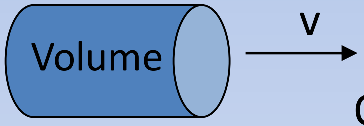
All units are in ft, hence
Z is called elevation head
P/γ is called pressure head
$V^2/2g$ is called velocity head

Flow of Fluids

- Example of fluid flow systems
  - Fire sprinkler system
  - Water distribution system in the house, city etc.
  - Fluid Power System
    - Hydraulics
    - Pneumatics
  - **Thermal systems** (brines, chilled water, steam, etc.)

Definitions

- Volume (Volumetric) Flow Rate
  - Q = Cross Sectional Area*Average Velocity of the fluid
  - Q = A*v



Q = Volume/Unit time
Q = Area*Distance/Unit Time

- Weight Flow Rate
  - $w = \gamma * Q$
- Mass Flow Rate
  - $M = \rho * Q = Q/v$ where v is specific volume

Units and Conversion Factors

- Q: m^3/sec, ft^3/sec
- W: N/sec, lb/sec
- M: kg/sec, slugs/sec
- Volume Flow Rate:
  - 1 L/min = 0.06 m^3/h
  - 1 m^3/sec = 60,000 L/min
  - 1 gal/min = 3.785 L/min
  - 1 ft^3/sec = 449 gal/min

Key Principles in Fluid Flow

- Continuity for any fluid (gas or liquid)
  - Mass flow rate In = Mass Flow Rate out
  - $M_1 = M_2$
  - $\rho_1 * A_1 * v_1 = \rho_2 * A_2 * v_2$
  - $w_1 = w_2$

    $\gamma_1 A_1 v_1 = \gamma_2 A_2 v_2$

- Continuity for liquids, the density is constant.
  - $Q_1 = Q_2$
  - $A_1 * v_1 = A_2 * v_2$

Example

- If d_1 and d_2 are 50 mm and 100 mm, respectively, and water at 70° C is flowing at 8 m/sec in section 1, determine: v_2, Q, W, M, water density at 70° C=977.5kg/m^3 at 1 atm.

$$A_1 v_1 = A_2 v_2$$

$$\frac{\pi}{4}(d_1)^2 v_1 = \frac{\pi}{4}(d_2)^2 v_2$$

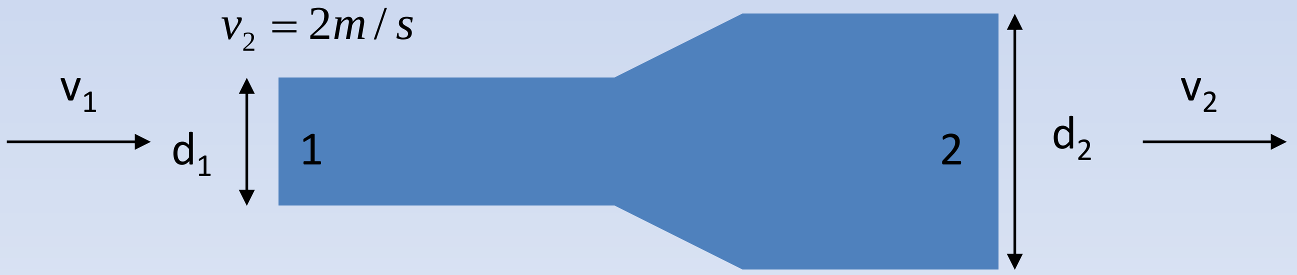
$$(0.05)^2 8 = (0.1)^2 v_2$$

$$v_2 = 2m/s$$

$$Q = Av = \frac{\pi}{4}(0.05)^2 \times 8 = \frac{\pi}{4}(0.1)^2 \times 2 = 0.015708 m^3/s$$

$$w = Q\gamma = 150.6 N/s$$

$$M = Q\rho = 15.3 kg/s$$



Hydraulic Cylinder Power

- How to determine the area of the piston required?

  $PA=F_{load}$

  $A=F_{load}/P$

- What is the pump flow rate required to drive the cylinder in a specific time?

  Cylinder volumetric displacement $V_D=AxS$

  Flow rate of the pump $Q=V_D/t=(Axs)/t=Axv$

  where v = piston velocity

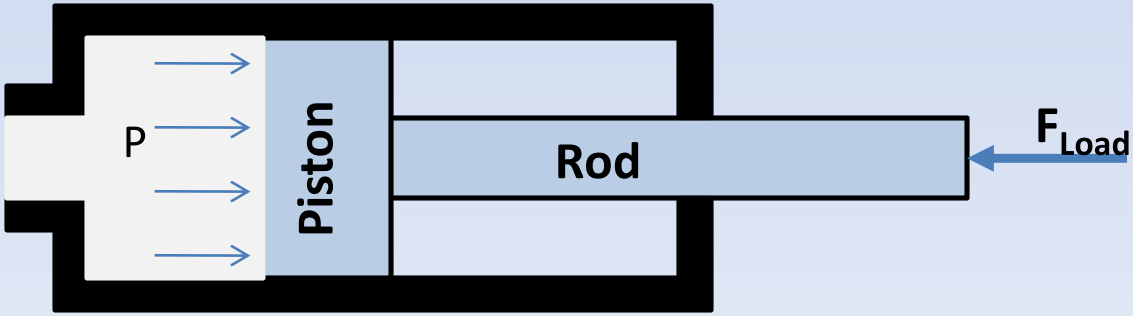
- How much hydraulic horsepower does the fluid deliver to the cylinder?

  $Energy=(F)(S)=(PA)(S)$, $\quad Power=energy/t=(PA)(S)/t=P(Av)=PQ$

  $Hydraulic\ Power(ft.lb/s)=P(lb/ft^2)Q(ft^3/s)$

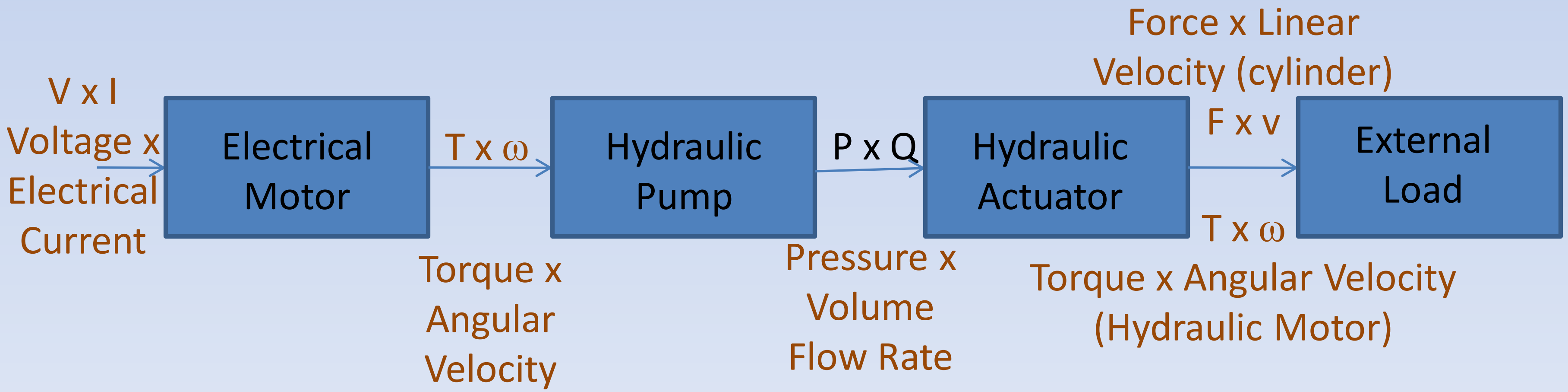
  $Hydraulic\ Horsepower\ HHP= P(lb/ft^2)Q(ft^3/s)/550$

  $HHP= P(psi) \times Q(gpm)/1714$



Power Conversion

- Power analogy between electrical, mechnical, and hydraulic systems.



Example

- A hydraulic cylinder is to compress a car body to a bale size in 10 s. The operation requiers a 10 ft stroke and a 8000 lb force. If a 1000 psi pump is used, and assuming a 100% efficiency,
  - a) The required piston area,
  - b) The necessary pump flow rate
  - c) The hydraulic horsepower delivered to the cylinder
  - d) The output horsepower delivered by the cylinder

$$a.\ A = \frac{F_{Load}}{p} = \frac{8000 lb}{1000 lb / in^2} = 8 in^2$$

$$b.\ Q(ft^3/s) = \frac{A(ft^2) \times S(ft)}{t(s)} = \frac{\left(\frac{8}{144}\right)10}{10} = 0.0556\, ft^3/s = 24.9 gpm, \quad 1\, ft^3/s = 449 gpm$$

$$c.\ HHP = \frac{1000 \times 24.9}{1714} = 14.5 hp$$

$$d.\ OHP = HHP \times \eta = 14.5 \times 1.0 = 14.5 hp$$

Cont.

- Solve the same example assuming a friction loss of 100lb and a leakage of 0.2gpm

$$a.\ A = \frac{F_{Load} + F_{Friction}}{p} = \frac{8000lb + 100lb}{1000lb/in^2} = 8.1in^2$$

$$b.\ Q_{theoretical}(ft^3/s) = \frac{A(ft^2) \times S(ft)}{t(s)} = \frac{\left(\frac{8.1}{144}\right)10}{10} = 0.0563\,ft^3/s = 25.2gpm$$

$$Q_{actual} = Q_{theoretical} + Q_{leakage} = 25.2 + 0.2 = 25.4gpm$$

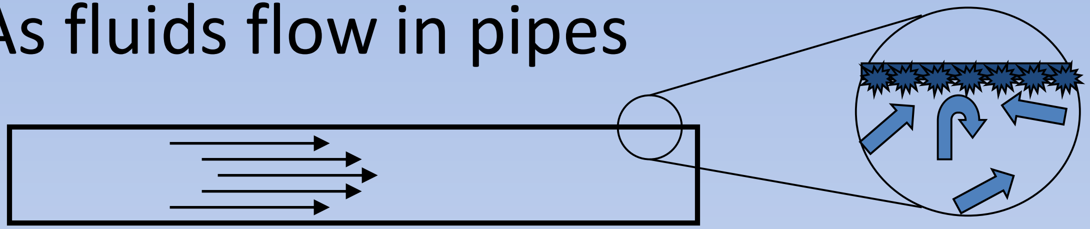
$$c.\ HHP = \frac{1000 \times 25.4}{1714} = 14.8hp$$

$$d.\ OHP = \frac{F(lb) \times v(ft/s)}{550} = \frac{8000 \times 1}{550} = 14.5hp$$

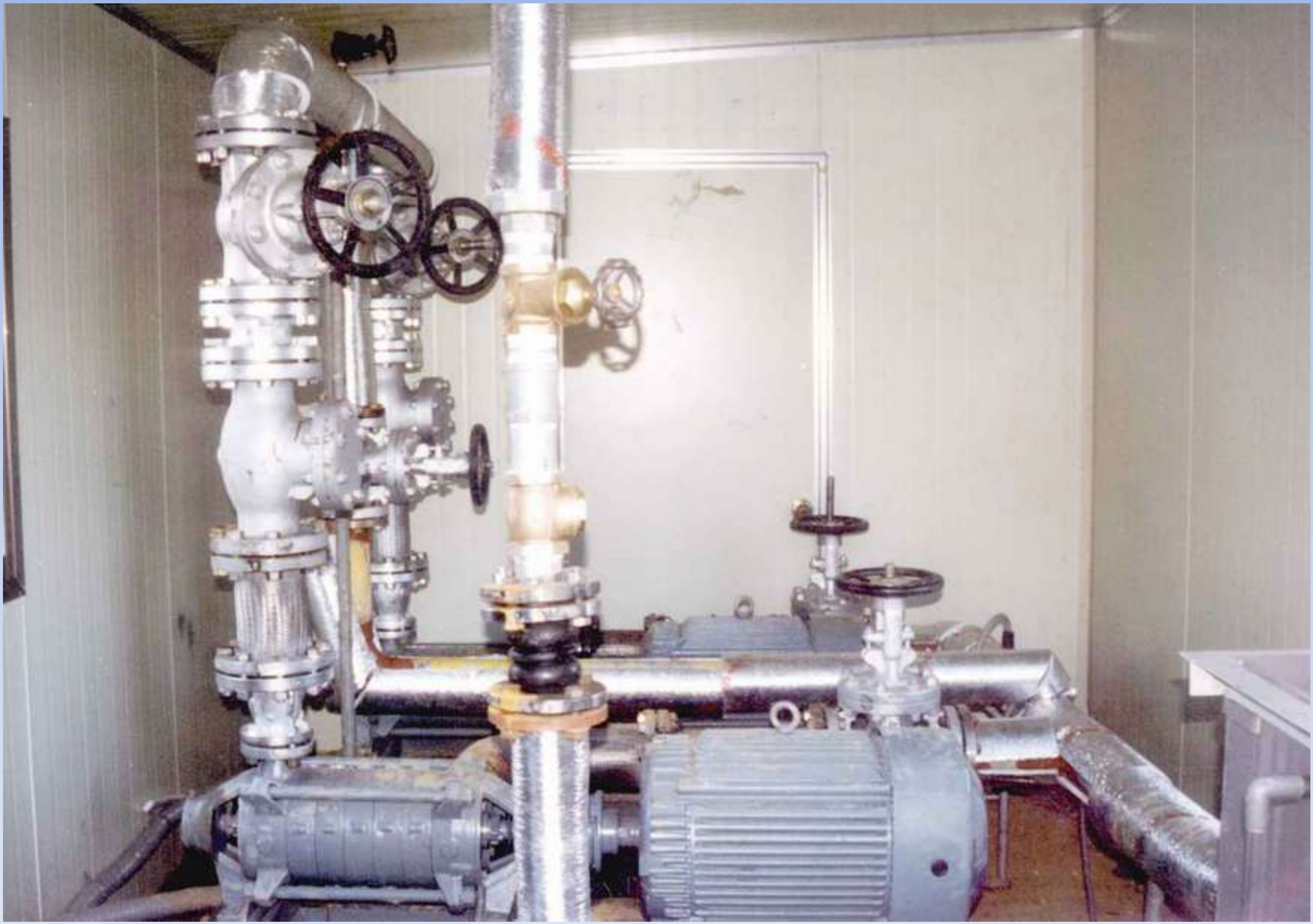
$$\eta = \frac{OHP}{HHP} = \frac{14.5}{14.8} = 0.980 = 98.0\%$$

Real Systems

- Friction losses: As fluids flow in pipes



- Minor losses: due the presence of valves, elbows, pipe entrance, etc.
- Motors: Turbines, actuators, etc. **take energy from fluid**
- Pumps: **Put energy into the fluid**
- The Bernoulli equation <u>*does not*</u> take these losses or gains into account




10. 15. 2000

Energy Equation

- $\text{Energy}_{in} = \text{Energy}_{out}$
- Energy_{in} + Gains - Losses = Energy_{out}

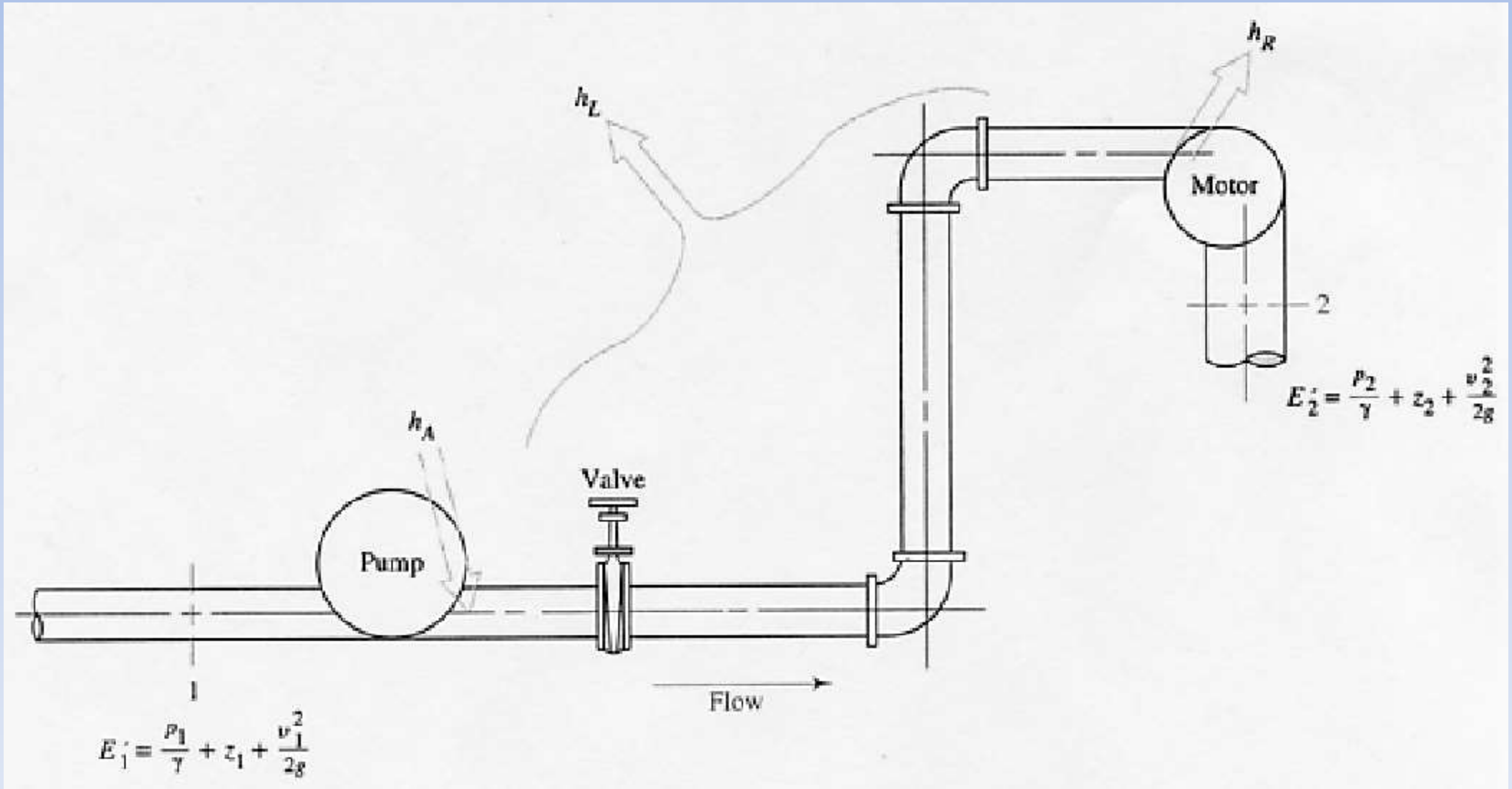
$$\frac{P_1}{\gamma} + z_1 + \frac{v_1^2}{2g} + \boxed{h_A - h_R - h_L} = \frac{P_2}{\gamma} + z_2 + \frac{v_2^2}{2g}$$

$$\frac{P_1}{\gamma} + z_1 + \frac{v_1^2}{2g} + h_A = \frac{P_2}{\gamma} + z_2 + \frac{v_2^2}{2g} + h_R + h_L$$

- h_A = Energy added to the fluid by a **pump** $\quad h_p(ft) = \dfrac{3950 \times (HHP)}{Q(gpm) \times SG}$
- h_R = Energy removed from the fluid by **motors**, etc.
- h_L = Energy losses due to **friction and minor losses**

Conservation of Energy

$$\frac{P_1}{\gamma} + z_1 + \frac{v_1^2}{2g} + h_A - h_R - h_L = \frac{P_2}{\gamma} + z_2 + \frac{v_2^2}{2g}$$



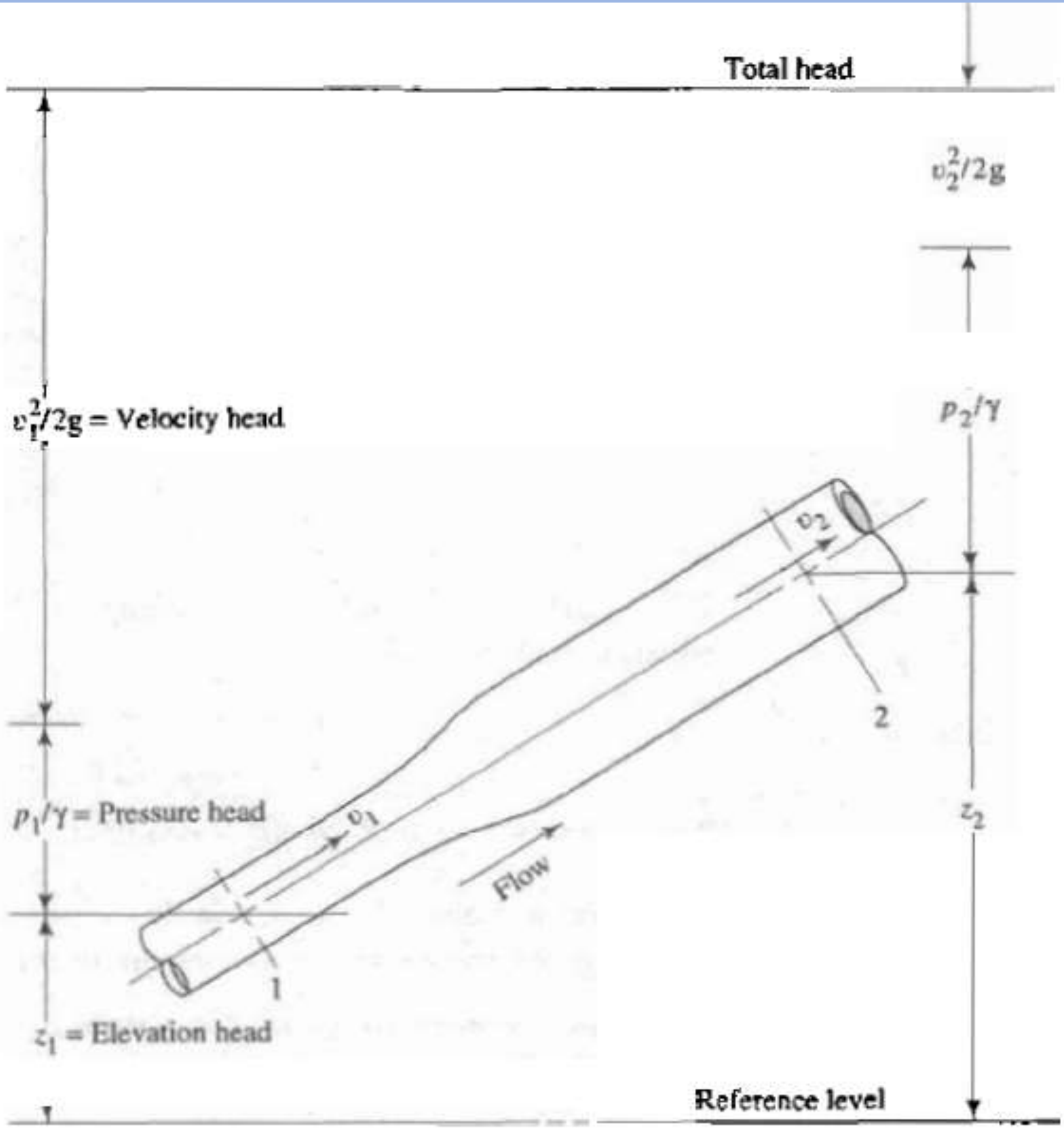
Heads

$$\frac{P}{\gamma} = \Pr essure\_Head$$

$$z = Elevation\_Head$$

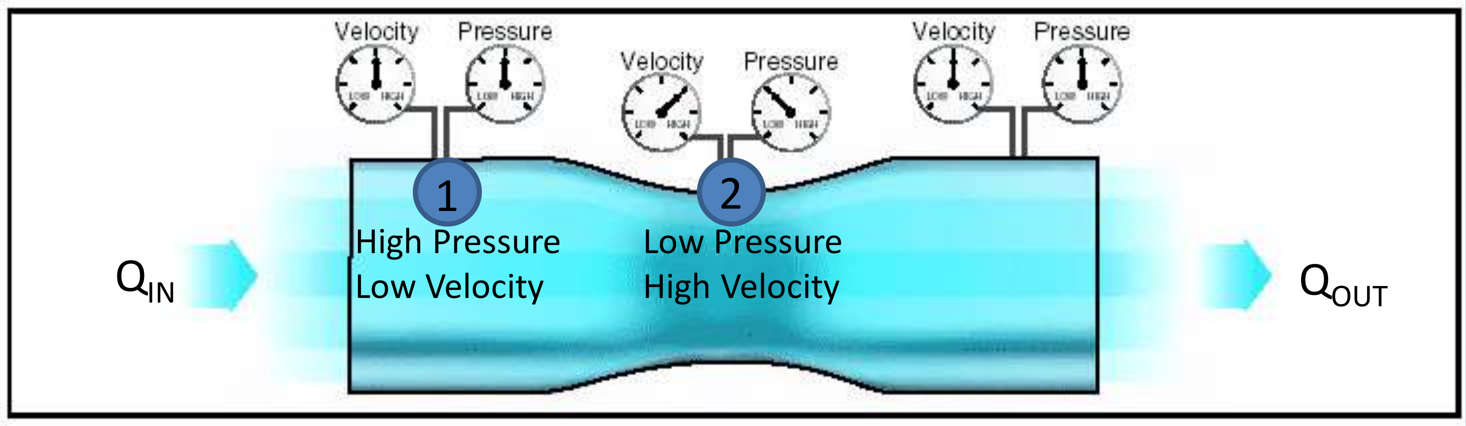
$$\frac{v^2}{2g} = Velocity\_Head$$

$$\frac{P}{\gamma} + z + \frac{v^2}{2g} = Total\_Head$$



Total head
$v_2^2/2g$
$v_1^2/2g$ = Velocity head
p_2/γ
v_2
2
p_1/γ = Pressure head
v_1
Flow
z_2
1
z_1 = Elevation head
Reference level

Venturi Effect



$$\frac{p_1}{\gamma} + \frac{v_1^2}{2g} = \frac{p_2}{\gamma} + \frac{v_2^2}{2g}$$

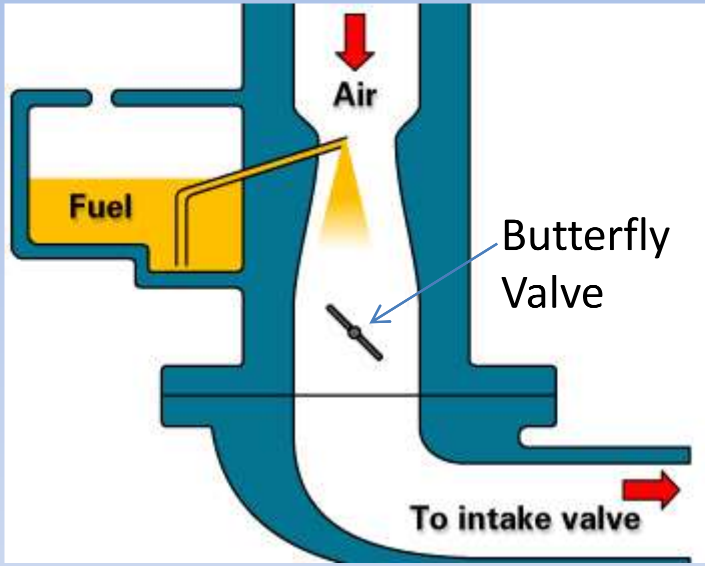
$$p_1 - p_2 = \frac{\gamma}{2g}(v_2^2 - v_1^2)$$

Q is constant through the pipe, therefore v_2 is higher than v_1 Then $p_2 < p_1$

Increase in KE decrease in PPE

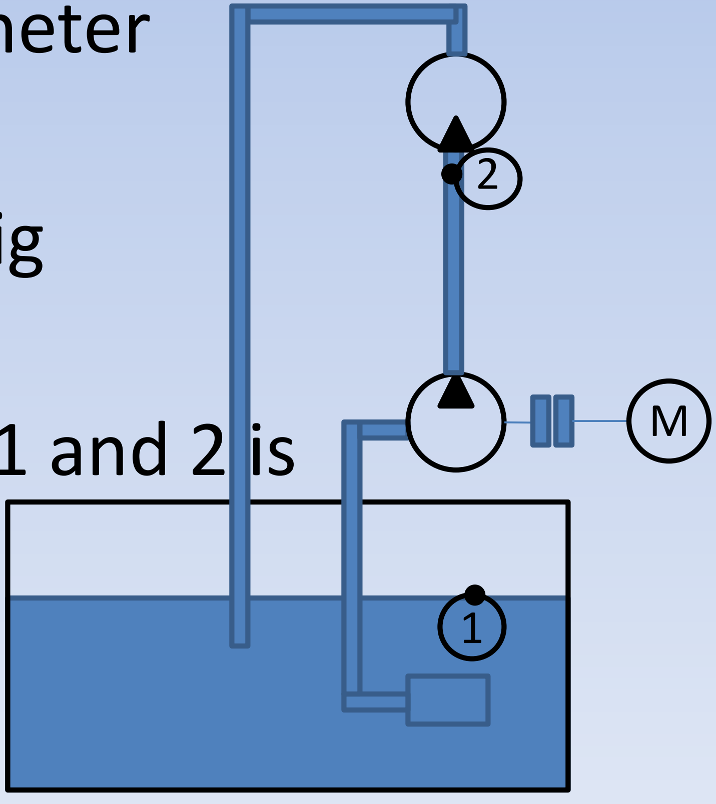
Carburetor

- When air is being drawn into the cylinders of the engine, it passes through a venturi in which there is an inlet connected to a source of fuel vapor. The lower pressure of the air causes some of the higher pressure fuel vapor to be pushed into and mixed with the stream of air.



Example

- The pump is adding 5hp to the system
- Pump flow is 30 gpm
- The pipe has a 1 in inside diameter
- The SG of the oil is 0.9
- The pressure at point 1 is 0 psig
- H_L from 1 to 2 is 30 ft of oil
- Elevation difference between 1 and 2 is 20ft
- Find the pressure at point 2



Example Solution

$$Z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + h_p - h_m - h_L = Z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

Assuming $h_m = 0, v_1 = 0, Z_2 - Z_1 = 20\,ft, h_L = 30\,ft$, and $P_1 = 0\,gage, \gamma = (SG)\gamma_{water} = (0.9)62.4 = 56.2\,lb/ft^3$

$$Z_1 + 0 + 0 + h_p - 0 - 30 = Z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g} \Rightarrow \frac{P_2}{\gamma} = Z_1 - Z_2 + h_p - 30 - \frac{v_2^2}{2g}$$

$$h_p = \frac{(3950)(5)}{(30)(0.9)} = 732\,ft$$

$$v_2 = \frac{Q}{A} = \frac{\frac{30}{449}(ft/s^2)}{\frac{\pi}{4}\left(\frac{1}{12}\right)^2 (ft^2)} = 12.2\,ft/s$$

$$P_2 = \left(732 - \frac{(12.2)^2}{2(32.2)} - 50\right)(SG)\gamma_{water} = 38200\,lb/ft^2 = 265\,psig$$

Torricelli's equation

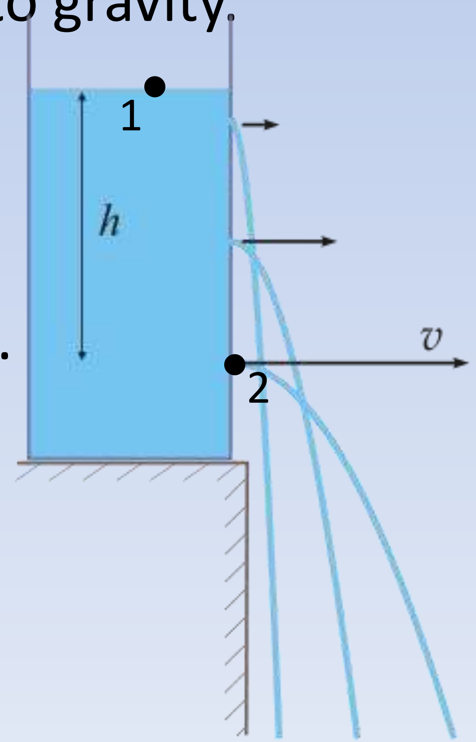
- Torricelli's law states that the speed of efflux, *v*, of a fluid through a sharp-edged hole at the bottom of a tank filled to a depth *h* is the same as the speed that a body (in this case a drop of water) would acquire in falling freely from a height *h*, i.e. , $v=(2gh)^{0.5}$ where *g* is the acceleration due to gravity.

$$Z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = Z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

- $P_1=P_2$ = 0 psig
- The area of the fluid is large that the velocity is zero
- $H_p=H_m=0$, assuming ideal fluid, $H_L=0$, no friction.

$$Z_1 + 0 + 0 + 0 - 0 - 0 = Z_2 + 0 + \frac{v_2^2}{2g}$$

$$Z_1 - Z_2 = h = \frac{v_2^2}{2g} \Rightarrow v_2 = \sqrt{2gh}$$



Example

- For the Torricelli's system let h=36ft and the diameter of the opening is 2 in, find
  - The jet velocity
  - The flow rate
  - Solve for a viscous liquid with h_L=10ft

  Sol.

$$1-v=\sqrt{2gh}=\sqrt{2(32.2\,ft/s^2)(36\,ft)}=48.3\,ft/s$$

$$2-Q=Av=\frac{\pi}{4}\left(\frac{2}{12}\,ft\right)^2\times 48.3\,ft/s=1.05\,ft^3/s=(449\times 1.05)gpm=471gpm$$

$$3-v=\sqrt{2g\left(h-h_L\right)}=\sqrt{2(32.2\,ft/s^2)(36\,ft-10\,ft)}=40.9\,ft/s$$

$$Q=Av=\frac{\pi}{4}\left(\frac{2}{12}\,ft\right)^2\times 40.9\,ft/s=0.89\,ft^3/s=(449\times 0.89)gpm=400.6gpm$$

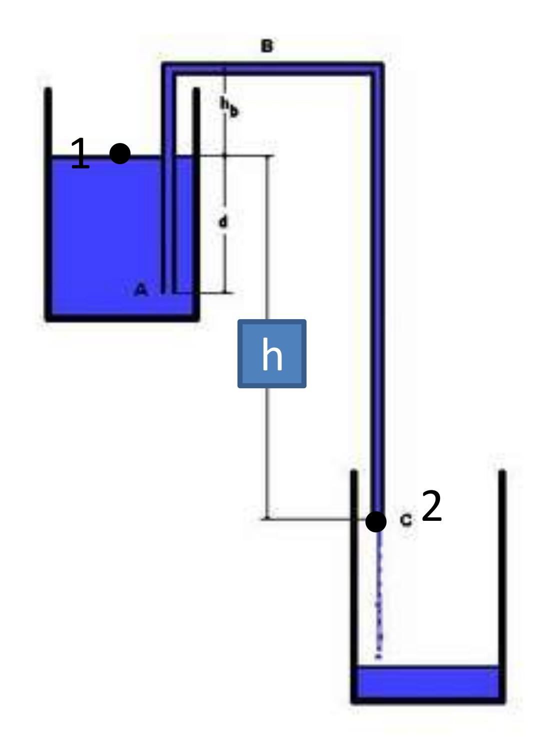
Siphon

- $P_1 = P_2 = 0$ psig
- $v_1 = 0$, large container
- $H_p = H_m = 0$
- $Z_1 - Z_2 = h$

$$Z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + H_p - H_m - H_L = Z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

$$Z_1 + 0 + 0 + 0 - 0 - H_L = Z_2 + 0 + \frac{v_2^2}{2g}$$

$$v_2 = \sqrt{2g(h - H_L)}$$



Example

- For the Siphon system shown h=30ft,h_L=10ft,and U-tube inside diameter=1in. Find *v* and Q
- Sol

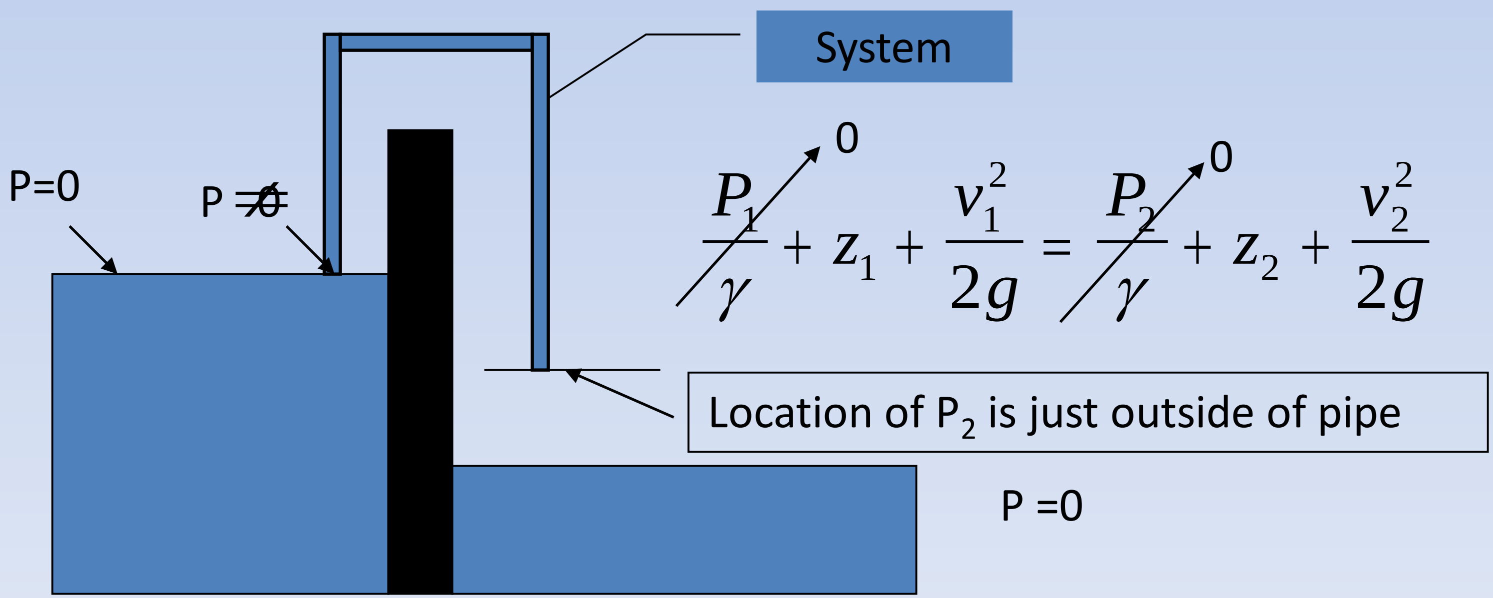
$$v = \sqrt{2g(h - h_L)} = \sqrt{2(32.2)(30 - 10)} = 35.8\, ft/s$$

$$Q = Av = \frac{\pi}{4}\left(\frac{1}{12}\, ft\right)^2 (35.8\, ft/s) = 0.195\, ft^3/s$$

$$Q(gpm) = 449Q(ft^3/s) = 87.6 gpm$$

Tanks, Reservoirs and Nozzles Exposed to the atmosphere

- When a fluid is exposed to the atmosphere at both ends of the system, the gauge pressure is zero at both ends and the pressure head can be cancelled from the equation

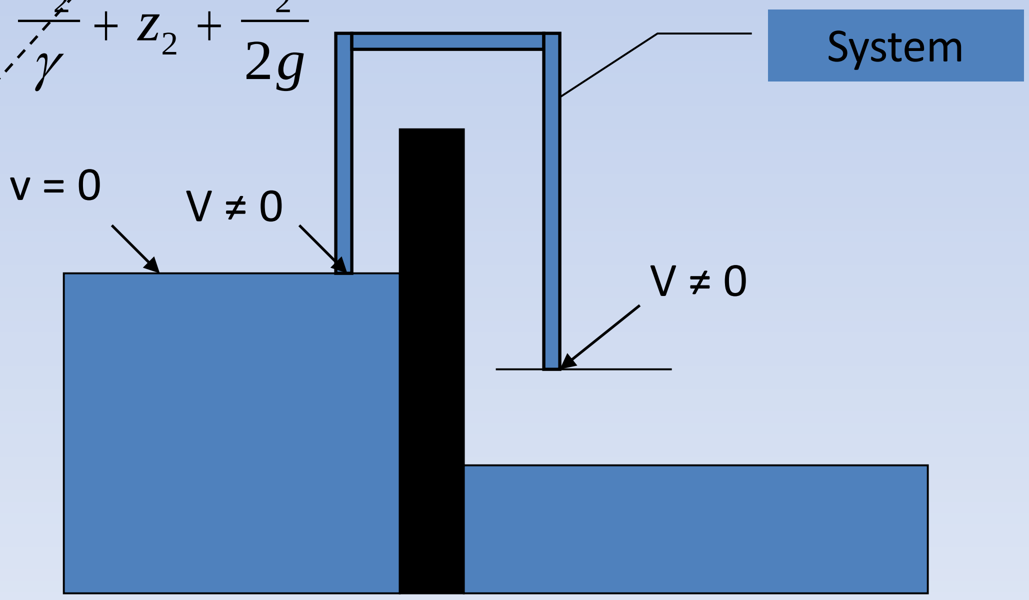


$$\cancelto{0}{\frac{P_1}{\gamma}} + z_1 + \frac{v_1^2}{2g} = \cancelto{0}{\frac{P_2}{\gamma}} + z_2 + \frac{v_2^2}{2g}$$

Tanks, Reservoirs and Nozzles Exposed to the atmosphere

- The velocity head at the surface of tank or reservoir is considered to be zero and it can be cancelled from the equation

$$\cancelto{0}{\frac{P_1}{\gamma}} + z_1 + \cancelto{0}{\frac{v_1^2}{2g}} = \cancelto{0}{\frac{P_2}{\gamma}} + z_2 + \frac{v_2^2}{2g}$$

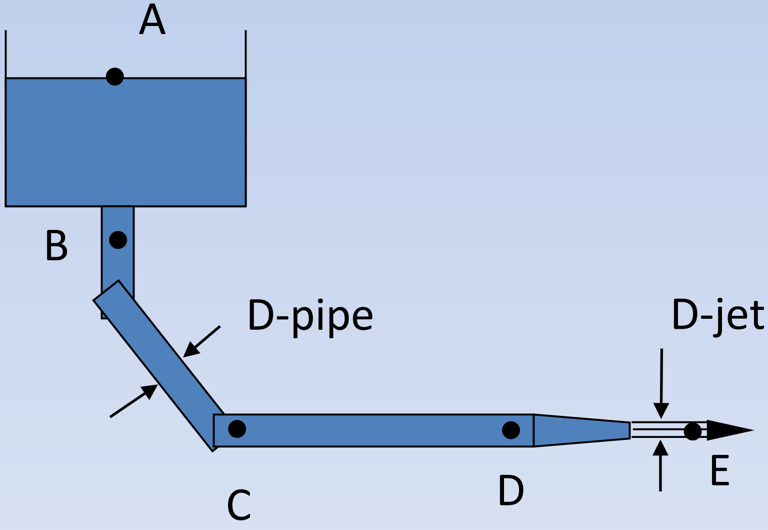


Tanks, Reservoirs and Nozzles Exposed to the atmosphere

When a fluid is exposed to the atmosphere at both ends of the system, the gauge pressure is zero at both ends and the pressure head can be cancelled from the equation

$$\cancelto{0}{\frac{P_A}{\gamma}} + z_A + \frac{V_A^2}{2g} = \cancelto{0}{\frac{P_E}{\gamma}} + z_E + \frac{V_E^2}{2g}$$

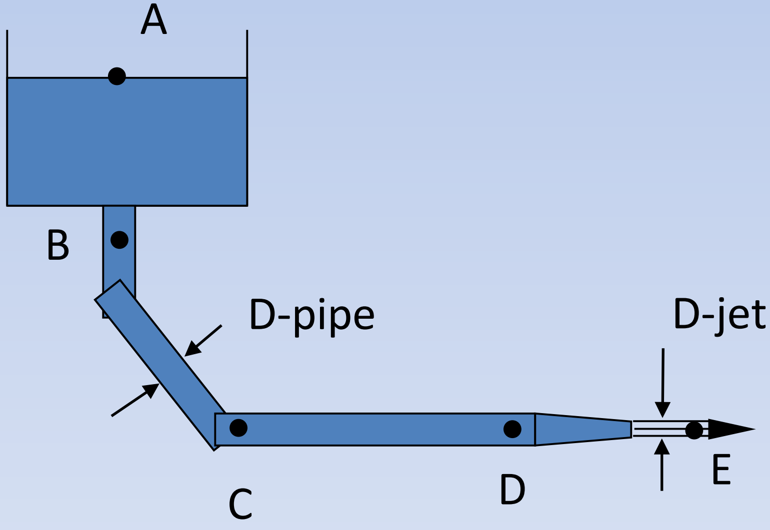
Location of P_E is just outside of pipe



Tanks, Reservoirs and Nozzles Exposed to the atmosphere

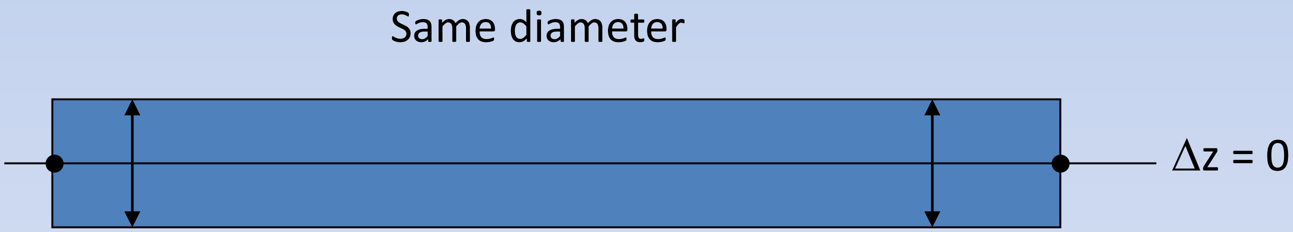
The velocity head at the surface of tank or reservoir is considered to be zero and it can be cancelled from the equation

$$\cancelto{0}{\frac{P_A}{\gamma}} + z_A + \cancelto{0}{\frac{V_A^2}{2g}} = \cancelto{0}{\frac{P_E}{\gamma}} + z_E + \frac{V_E^2}{2g}$$



Pipe or Canal at the same elevation

- If two points of interest of a pipe of constant diameter are at the same elevation, the potential and velocity heads can be cancelled from the equation



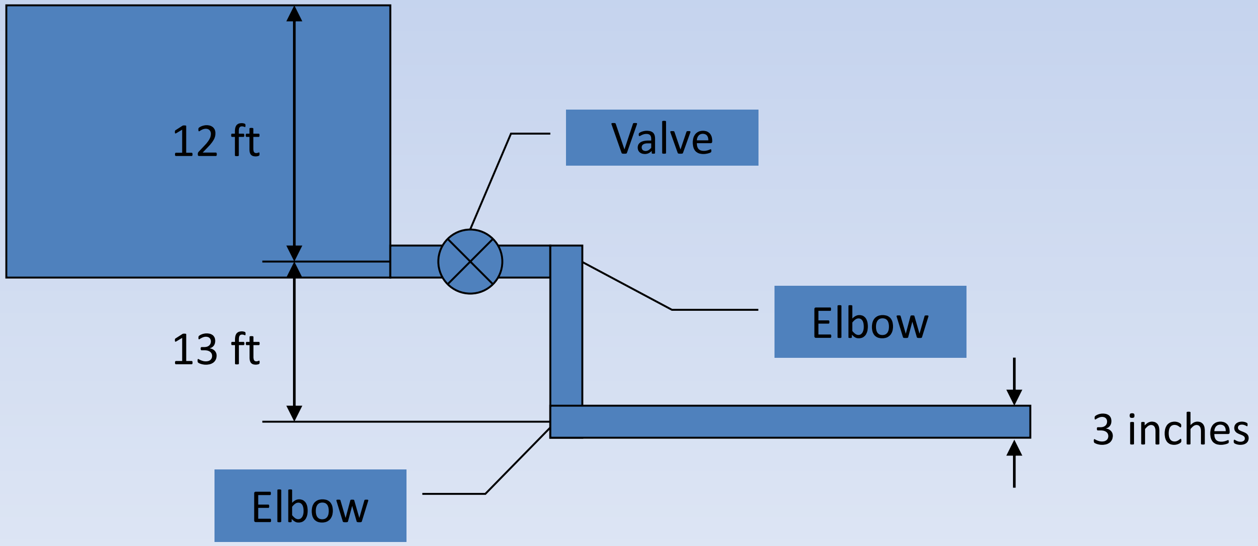
$$\frac{P_1}{\gamma} + \cancelto{0}{z_1} + \cancelto{0}{\frac{v_1^2}{2g}} = \frac{P_2}{\gamma} + \cancelto{0}{z_2} + \cancelto{0}{\frac{v_2^2}{2g}}$$

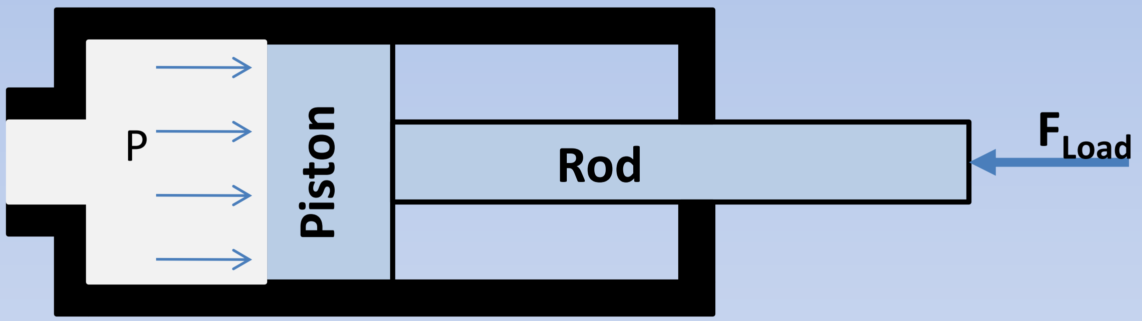
SI Metric Systems

- Energy=1N x 1m=1 N.m= 1J
- Power = Work/time= 1 W= 1 J/1 S =1 N.m/s
- Hydraulic Power (W)= p(N/m^2) x Q (m^3/s)
- H_p (m)= pump hydraulic power (W)/(γ(N/m^3) x Q(m^3/s))
- Torque Power or Brake Power(KW) = T (N.m) x ω (rad/s) /1000 = T (N.m) x N (rpm) /9550
- Q (m^3/s) =A(m^2) x v(m/s)
- 1 m^3/s = 15,800 gpm large quantity hence Lsp is used
- Liters per second (Lps), 1L=0.001m^3

Example

- Water flows from a reservoir at 1.2 ft^3/sec. Calculate the energy lost from the system due to valves, elbows, pipe entrance and fluid friction.




P
Piston
Rod
F_{Load}

Frictional Losses in Hydraulic Pipelines

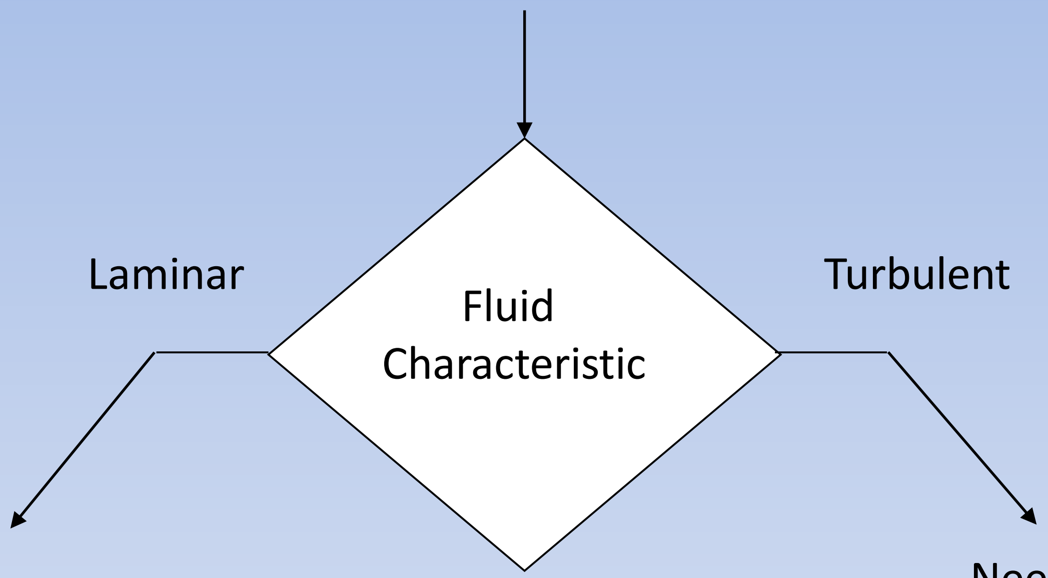
Chapter 4

Dr. Suleiman BaniHani

Fluid Losses (h_L)

- Frictional Losses (due to fluid friction in pipes)
- Minor Losses (due to valves, fittings, etc.)
- How to calculate fluid losses?
  - Need to identify type of flow
    - Laminar or Turbulent?
  - Must know flow conditions and piping system specifications (size, length, etc.)

Fluid Losses

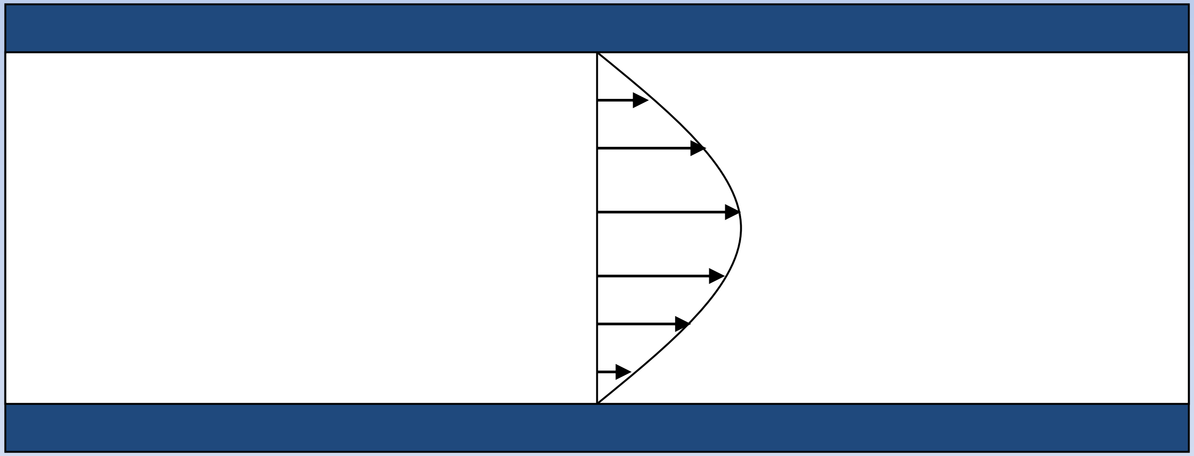


Need to know:
Velocity
Pipe Diameter
Viscosity
Density

Need to know:
Velocity
Pipe Diameter
Viscosity
Density
<u>Roughness</u>

Laminar Flow

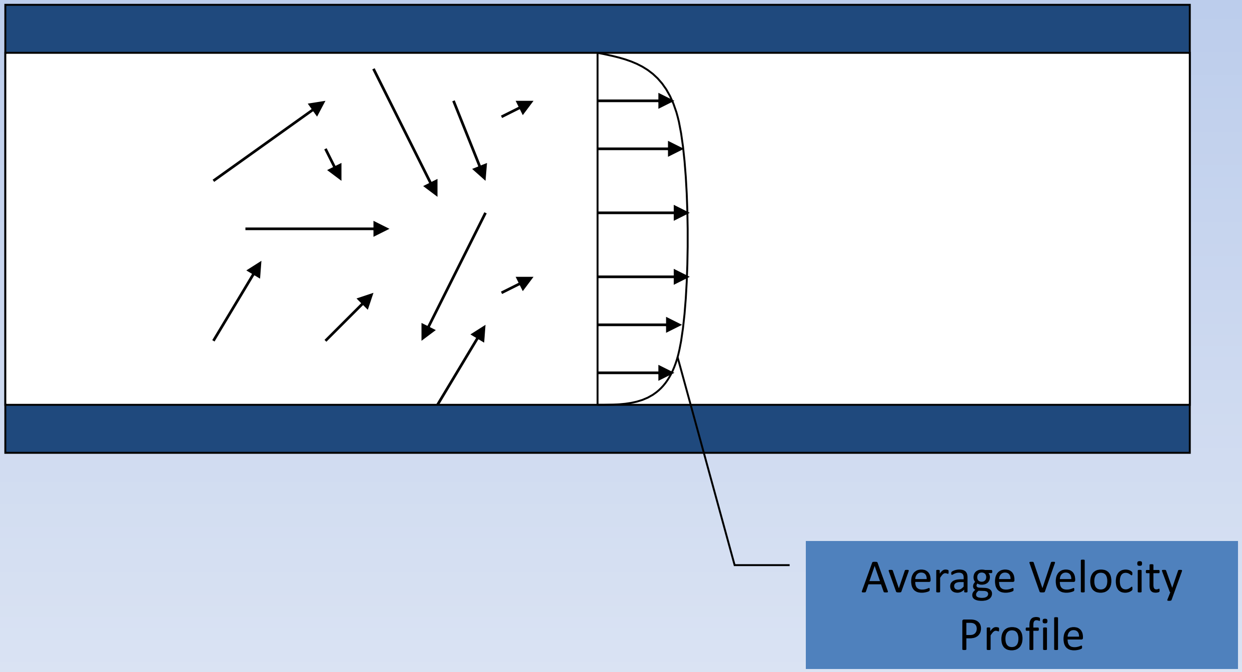
- Streamline flow, smooth velocity profile



$$U = 2v_{AVG}\left[1-\left(\frac{r}{r_o}\right)^2\right]$$

Turbulent Flow

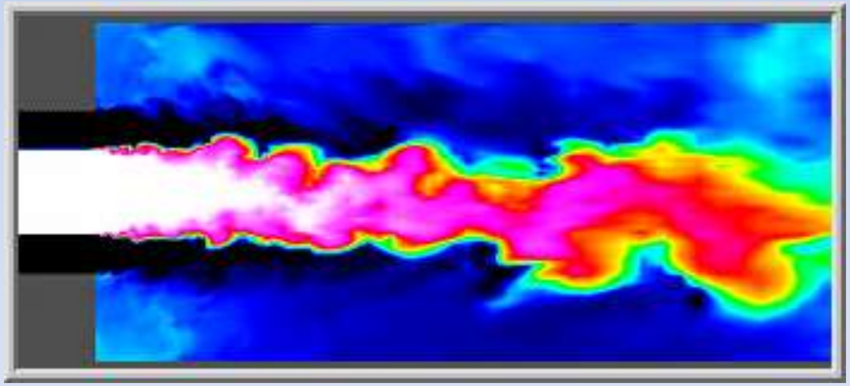
- Fluid particles randomly fluctuate along the streamwise direction



Laminar vs. Turbulent

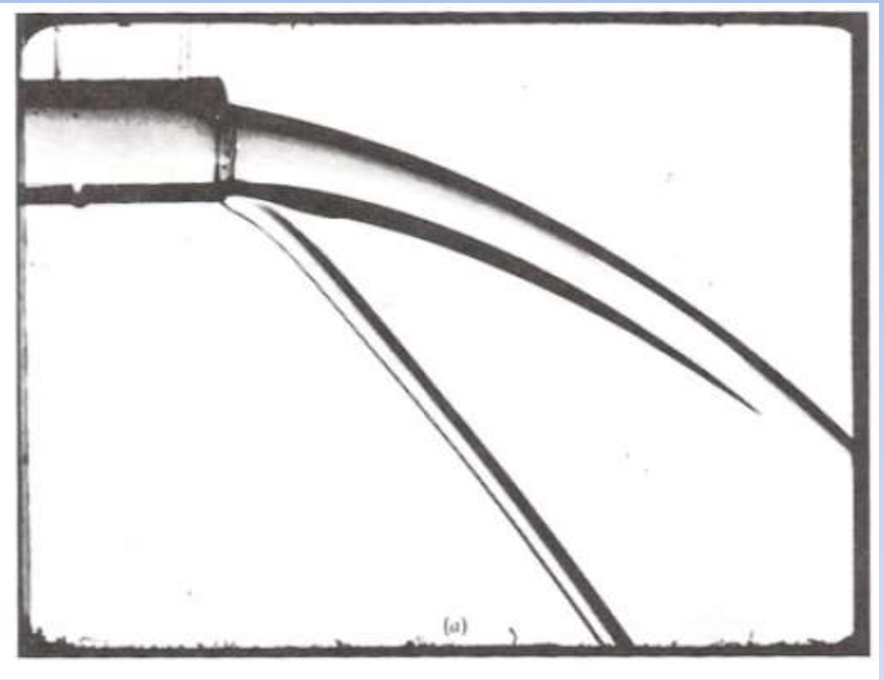
Laminar

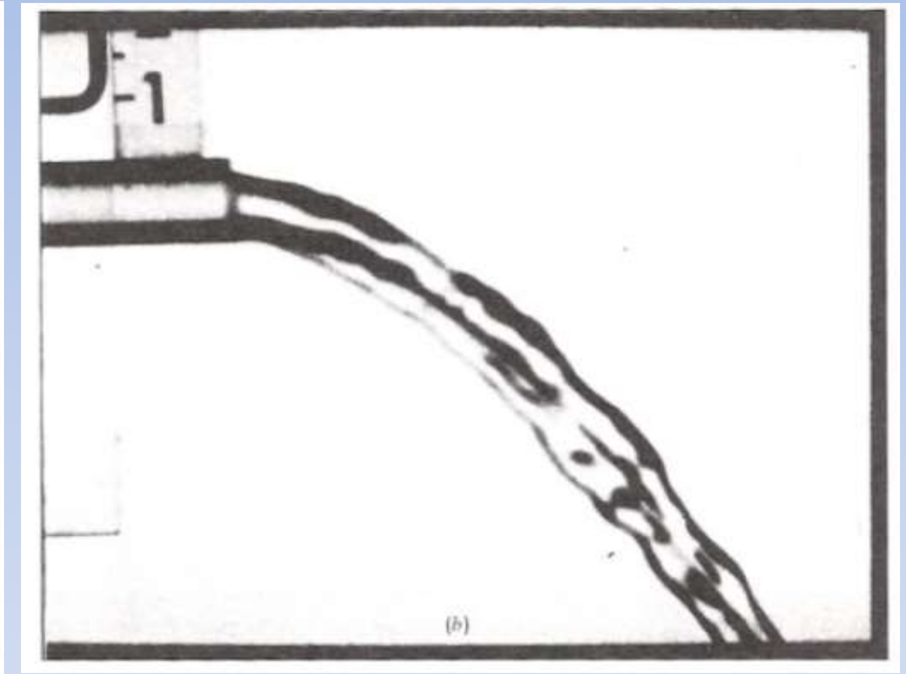
Turbulent



http://www.engineering.uiowa.edu/fluidslab/gallery/images/turb6im.gif

Laminar vs. Turbulent





Reynolds Number

$$Re = \frac{Inertia\_Forces}{Viscous\_Forces}$$

$$\mathrm{Re} = \frac{v * D_{PIPE} * \rho}{\mu} = \frac{v * D_{PIPE}}{\nu}$$

Re < 2000 Laminar Flow

Re > 4000 Turbulent

2000 < Re < 4000 Critical Region or Transitional

Example

- The kinematic viscosity of hydraulic oil 100cS. If the fluid is flowing in a 1 in diameter pipe at a velocity of 10ft/s, what is the Reynolds number?
- N_R=(7740)(10)(1)/100 = 774
- Oil(ν=0.001m^2/s) is flowing in a 50mm diameter pipe at a velocity 5 m/s, what is the Reynolds number?
- N_R= 5(0.05)/0.001= 250

Example

- Water flowing at 285 L/min, D_{PIPE} = 0.02524 m, Area = 5.017×10^{-4} m^2, $\nu = 4.11 \times 10^{-7}$ m^2/sec (kinematic viscosity). Is the flow laminar or turbulent?

The Loss Head

- The loss head H_L in a system consists of two components:
  1. Losses in pipes
  2. Losses in valves and fittings

- Energy loss due to friction in pipes can be found using Darcy's equation.

Energy loss due to friction

- Due to flowing fluid
- Proportional to velocity head:

$$\boxed{h_L \alpha \ \frac{v^2}{2g}}$$

- Proportional to the ratio Length of Pipe/Diameter of Pipe (L/D)
- Darcy equation:

$$h_L = f \frac{L}{D} \frac{v^2}{2g}$$

h_L = Energy loss due to friction, N-m/N or lb-ft/lb

L = length of flow stream or pipe, m

D = pipe diameter, m

v = average fluid velocity, m/sec

f = friction factor (dimensionless)

Friction Loss in Laminar Flow

- Fluid friction is independent of surface roughness for laminar flow
- Hagen-Poiseuille Equation:

$$h_L = \frac{32\mu \cdot L \cdot v}{\gamma \cdot D^2} \rightarrow \mathrm{Re} < 2000$$

$$Yields:$$

$$f = \frac{64}{\mathrm{Re}} \rightarrow$$ Laminar Flow

Example

- For a system the kinematic viscosity of hydraulic oil 100cS. If the fluid is flowing in a 1 in diameter pipe at a velocity of 10 ft/s system find the head loss due to friction in units of psi for a 100 ft length of pipe. The oil has a specific gravity of 0.9

$$N_R = \frac{(7740)(10)(1)}{100} = 774$$

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

$$\frac{64}{774}\left(\frac{100}{\frac{1}{12}}\right)\left(\frac{10^2}{62.4}\right) = 154\text{ft}$$

- H_L has a units of ft.lb/lb which means 154 ft. lb of energy is lost by each pound of oil as it flows through 100 ft of pipe.

$$P_L = \gamma H_L = (SG \times \gamma_{water}) \times H_L =$$

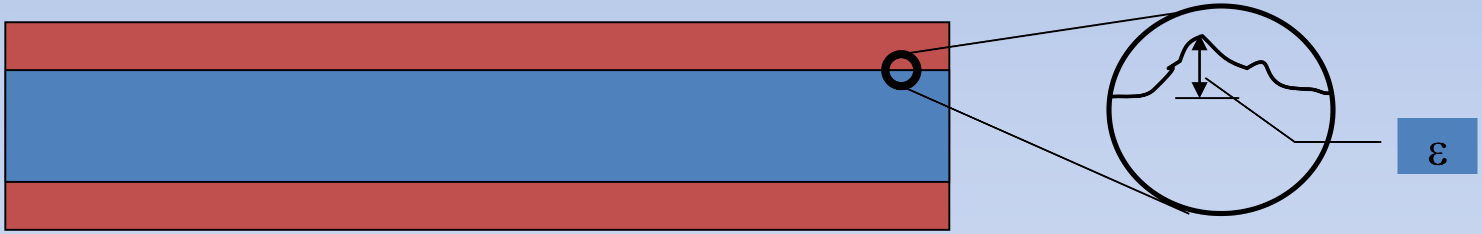
$$(0.9 \times 0.0361) lb / in^3 \times (12 \times 154) in = 60\, psi$$

Example

- Determine the energy loss if glycerine at 25 °C flows 30 m through a 150 mm diameter pipe with an average velocity of 4.0 m/sec.

Friction Loss in Turbulent Flow

- Does depend on surface roughness!
- Surface roughness is expressed as: $\underline{\varepsilon}$



Part of Table 8.2:

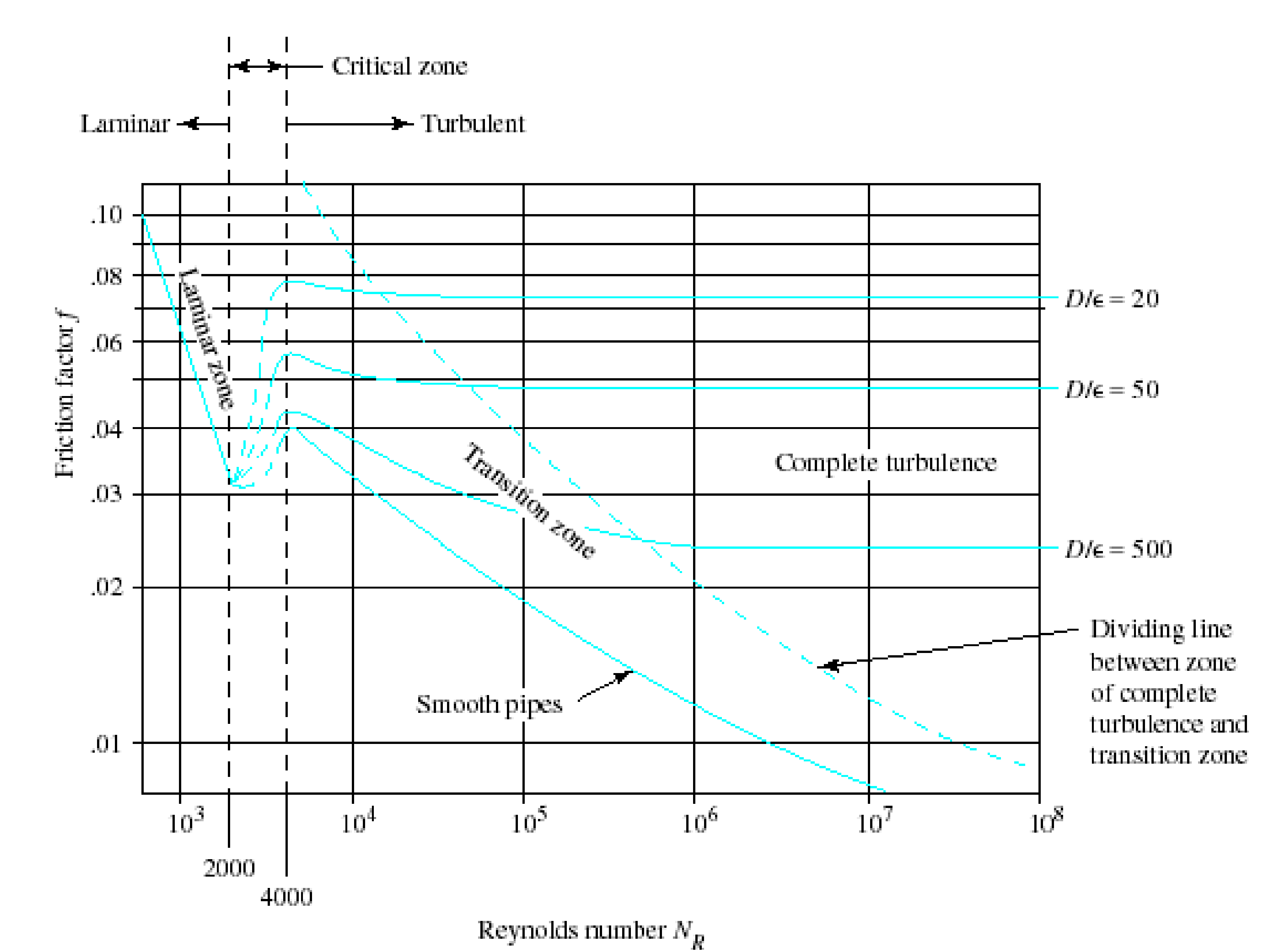
Material	Roughness, ε, m
Glass	Smooth (very small)
Copper, brass	1.5×10^{-6}
Commercial Steel	4.6×10^{-5}
Concrete	1.2×10^{-4}

Material	Roughness ϵ (m)	Roughness ϵ (ft)
Glass	Smooth	Smooth
Plastic	3.0×10^{-7}	1.0×10^{-6}
Drawn tubing; copper, brass, steel	1.5×10^{-6}	5.0×10^{-6}
Steel, commercial or welded	4.6×10^{-5}	1.5×10^{-4}
Galvanized iron	1.5×10^{-4}	5.0×10^{-4}
Ductile iron—coated	1.2×10^{-4}	4.0×10^{-4}
Ductile iron—uncoated	2.4×10^{-4}	8.0×10^{-4}
Concrete, well made	1.2×10^{-4}	4.0×10^{-4}
Riveted steel	1.8×10^{-3}	6.0×10^{-3}

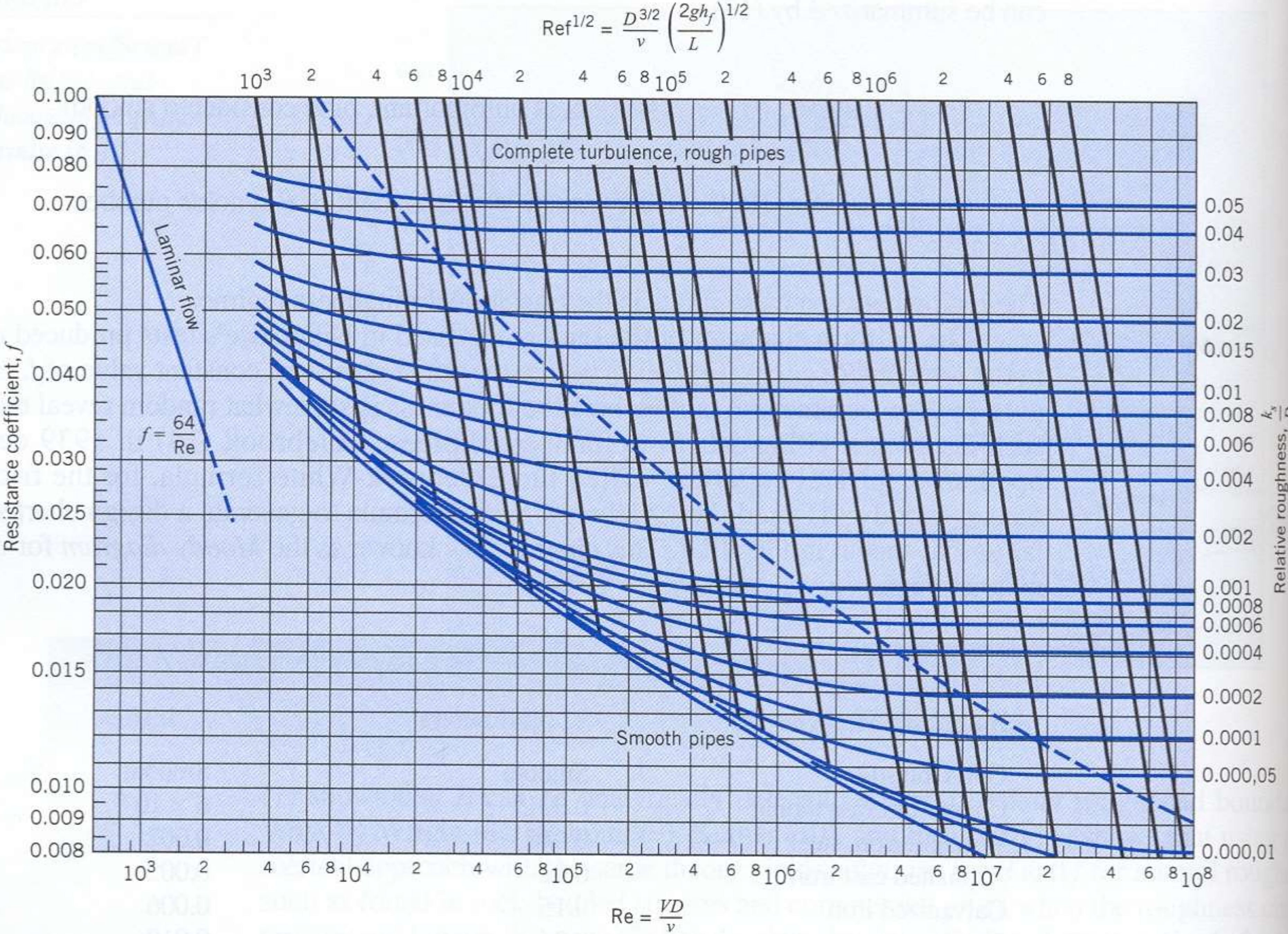
Friction Loss in Turbulent Flow

- f = f(Re, ε, D)
- Moody chart shows f as a funtion of Re and relative roughness = ε/D

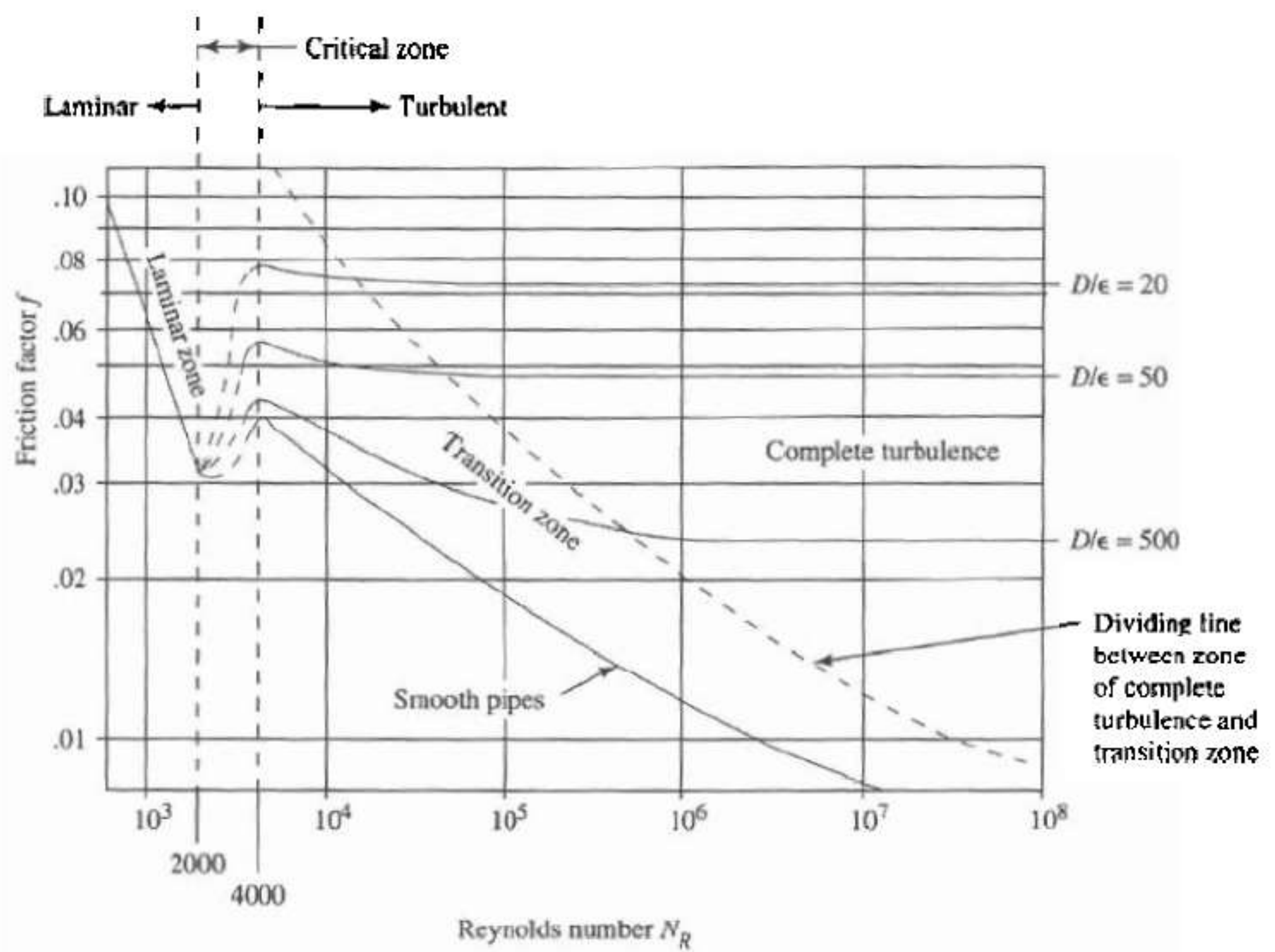
Moody Chart



Critical zone
Laminar
Turbulent
.10
.08
.06
.04
.03
.02
.01
Friction factor f
Laminar zone
Transition zone
Complete turbulence
$D/\epsilon = 20$
$D/\epsilon = 50$
$D/\epsilon = 500$
Dividing line between zone of complete turbulence and transition zone
Smooth pipes
10^3
10^4
10^5
10^6
10^7
10^8
2000
4000
Reynolds number N_R



$\mathrm{Re}f^{1/2} = \frac{D^{3/2}}{\nu}\left(\frac{2gh_f}{L}\right)^{1/2}$
10³ 2 4 6 8 10⁴ 2 4 6 8 10⁵ 2 4 6 8 10⁶ 2 4 6 8
Complete turbulence, rough pipes
Laminar flow
$f = \frac{64}{\mathrm{Re}}$
Smooth pipes
Resistance coefficient, f
0.100
0.090
0.080
0.070
0.060
0.050
0.040
0.030
0.025
0.020
0.015
0.010
0.009
0.008
Relative roughness, $\frac{k_s}{D}$
0.05
0.04
0.03
0.02
0.015
0.01
0.008
0.006
0.004
0.002
0.001
0.0008
0.0006
0.0004
0.0002
0.0001
0.000,05
0.000,01
10³ 2 4 6 8 10⁴ 2 4 6 8 10⁵ 2 4 6 8 10⁶ 2 4 6 8 10⁷ 2 4 6 8 10⁸
$\mathrm{Re} = \frac{VD}{\nu}$



Critical zone
Laminar
Turbulent
Friction factor f
.10
.08
.06
.04
.03
.02
.01
Laminar zone
Transition zone
Complete turbulence
D/ϵ = 20
D/ϵ = 50
D/ϵ = 500
Dividing line between zone of complete turbulence and transition zone
Smooth pipes
10^3
10^4
10^5
10^6
10^7
10^8
2000
4000
Reynolds number N_R

Observations about the Moody Chart

- When Re is constant, as D/ε increases, f decreases

- For constant D/ε, f decreases with increasing Re

- In fully turbulent flow, $f \neq f(\mathrm{Re})$

Example

- The kinematic viscosity of a hydraulic oil is 50 cS. If the oil flows in a 1 in diameter commercial steel pipe find the friction factor if
  a) The velocity is 10ft/s
  b) The velocity is 40ft/s

A. $N_R = \frac{(7740)(10)(1)}{50} = 1548 = 1.548 \times 10^3$ Laminar flow $f=64/N_R$ = 0.042

B. $N_R = \frac{(7740)(40)(1)}{50} = 6192 = 6.192 \times 10^3$ Turbulent flow, we need ε/D

$$\frac{\varepsilon}{D} = \frac{0.0018in}{1in} = 0.0018$$

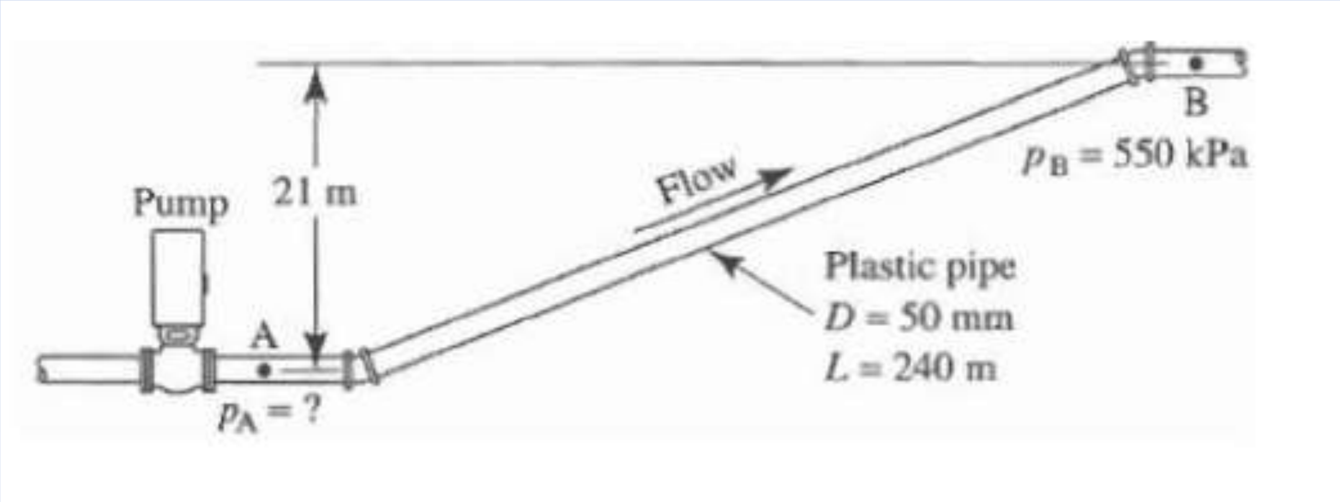
f=*0.036*

Example

- Determine the friction factor (f) if water at 160 °F is flowing at 30 ft/sec in an uncoated ductile iron pipe having an inside diameter of 1 in.
- Repeat, but this the water is flowing at 0.45 ft/sec

Example

- Benzene at 50 °C (sg = 0.86) is pumped from A to B where the pressure is 550 kPa. A pump is located 21 m below point B, and the two points (A & B) are connected by a 240 m plastic pipe with an inside diameter of 50 mm. Q = 110 L/min. Calculate P_A.



Valves and Fittings Losses

- So far we have considered losses in straight sections of pipe
- Other component also cause losses in a piping system
  - Sudden or gradual enlargements of flow path
  - Sudden or gradual contractions of flow path
  - Exit or entrance losses
  - Elbows
  - Valves
  - Diffusers

Energy loss due to Valves and Fittings

- Majority of losses due to valves and fittings
- Proportional to velocity head:
- Can be approximated :

$$\boxed{h_L \alpha \frac{v^2}{2g}}$$

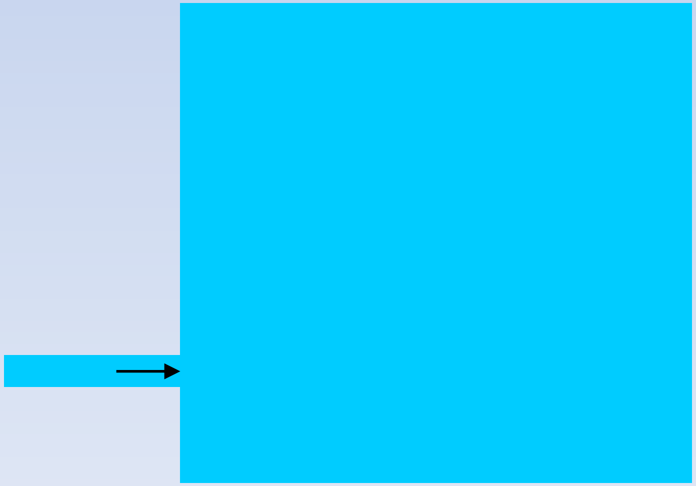
$$H_L = K\left(\frac{v^2}{2g}\right)$$

h_L = Energy loss due to friction, N-m/N
or lb-ft/lb
v = average fluid velocity, m/sec
K = K factor constant (loss coefficients)

Exit Losses

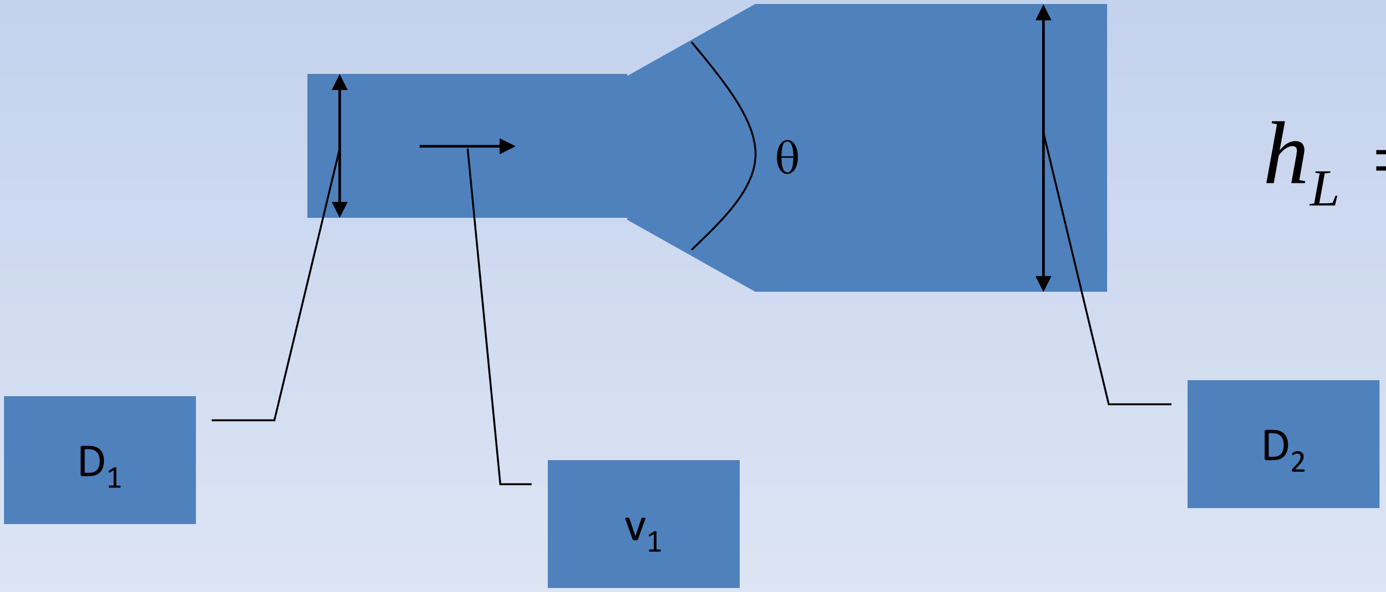
- Take place when fluid flows from a pipe into a large tank or reservoir
- Velocity decreases from a certain value to close to zero.
- K = 1.0 for all cases (exit losses only)

$$h_L = 1.0\left(\frac{v^2}{2g}\right)$$



Minor Losses: Gradual Enlargement

- Gradual enlargement cause less energy losses than sudden enlargements
- Less separation between streamlines



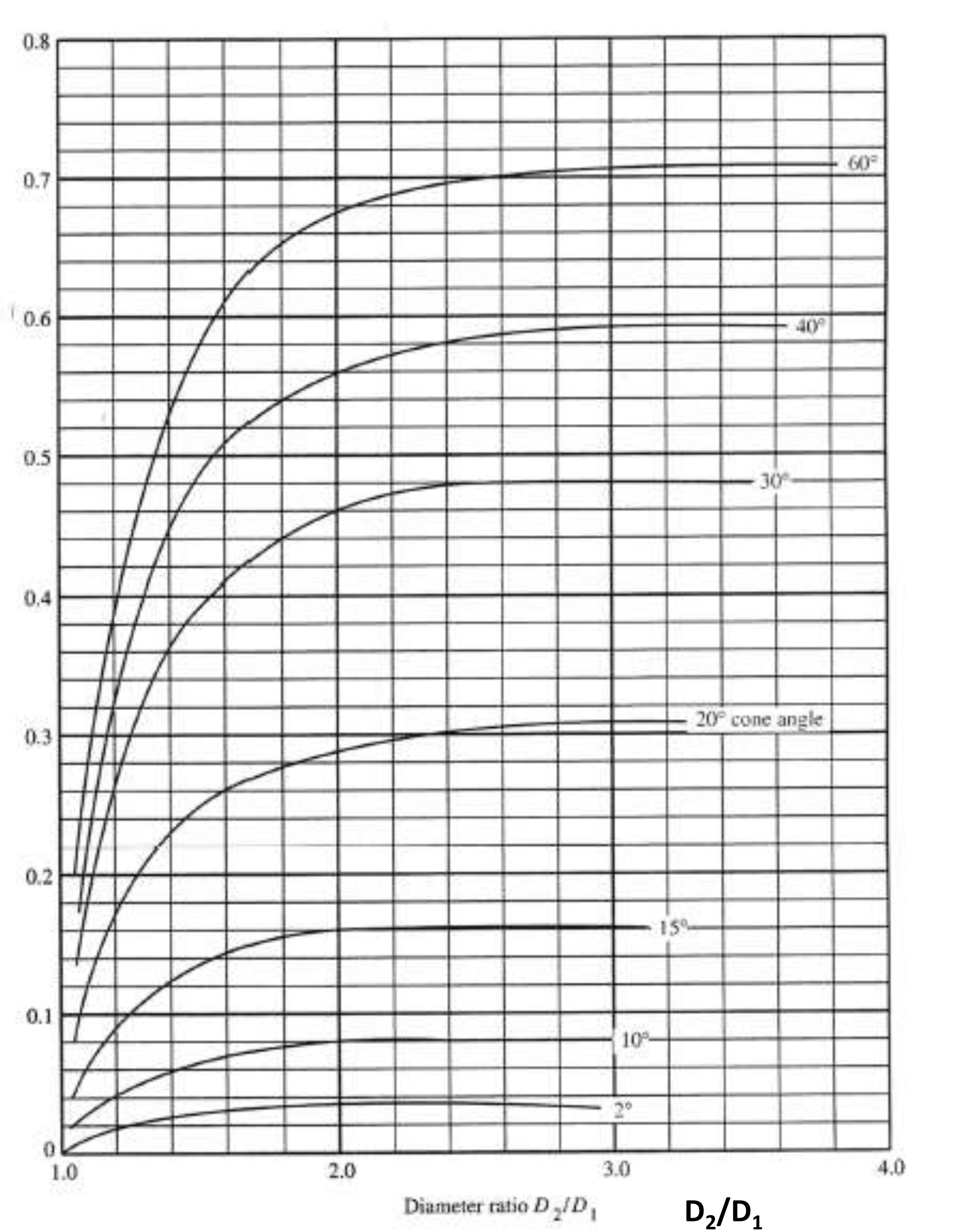
$$h_L = K\frac{v_1^2}{2g}$$

Minor Losses: Gradual Enlargement

- K depends on D_1, D_2 and cone angle θ

$$h_L = K\frac{v_1^2}{2g}$$

Figure 10-5: K for Gradual Enlargement



D_2/D_1

Example

- Determine the energy loss when $v_1 = 3.32$ m/sec, $D_1 = 25.33$ mm, $D_2 = 73.8$ mm, and the cone angle is 30°.

Gradual Enlargement

- Gradual enlargement used to recover pressure, such as diffusers

$$\frac{P_1}{\gamma} + z_1 + \frac{v_1^2}{2g} - h_L = \frac{P_2}{\gamma} + z_2 + \frac{v_2^2}{2g}$$

Ideal:

$$\Delta P = P_2 - P_1 = \gamma\left[\frac{\left(v_1^2 - v_2^2\right)}{2g}\right]$$

Real:

$$\Delta P = P_2 - P_1 = \gamma\left[\frac{\left(v_1^2 - v_2^2\right)}{2g} - h_L\right]$$

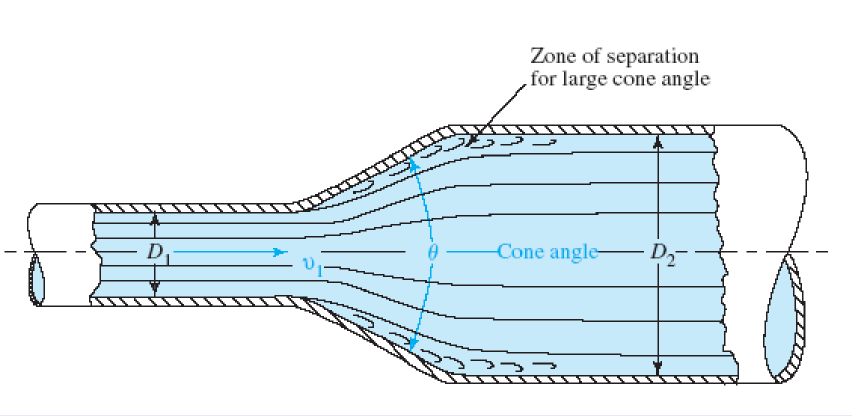


TABLE 10.2 Resistance coefficient—gradual enlargement

D_2/D_1	Angle of Cone θ											
	2°	6°	10°	15°	20°	25°	30°	35°	40°	45°	50°	60°
1.1	0.01	0.01	0.03	0.05	0.10	0.13	0.16	0.18	0.19	0.20	0.21	0.23
1.2	0.02	0.02	0.04	0.09	0.16	0.21	0.25	0.29	0.31	0.33	0.35	0.37
1.4	0.02	0.03	0.06	0.12	0.23	0.30	0.36	0.41	0.44	0.47	0.50	0.53
1.6	0.03	0.04	0.07	0.14	0.26	0.35	0.42	0.47	0.51	0.54	0.57	0.61
1.8	0.03	0.04	0.07	0.15	0.28	0.37	0.44	0.50	0.54	0.58	0.61	0.65
2.0	0.03	0.04	0.07	0.16	0.29	0.38	0.46	0.52	0.56	0.60	0.63	0.68
2.5	0.03	0.04	0.08	0.16	0.30	0.39	0.48	0.54	0.58	0.62	0.65	0.70
3.0	0.03	0.04	0.08	0.16	0.31	0.40	0.48	0.55	0.59	0.63	0.66	0.71
∞	0.03	0.05	0.08	0.16	0.31	0.40	0.49	0.56	0.60	0.64	0.67	0.72

Source: King, H. W., and E. F. Brater. 1963. *Handbook of Hydraulics*, 5th ed. New York: McGraw-Hill, Table 6–8.

Sudden Contraction

- Sudden contraction causes energy losses

$$h_L = K\frac{v_2^2}{2g}$$

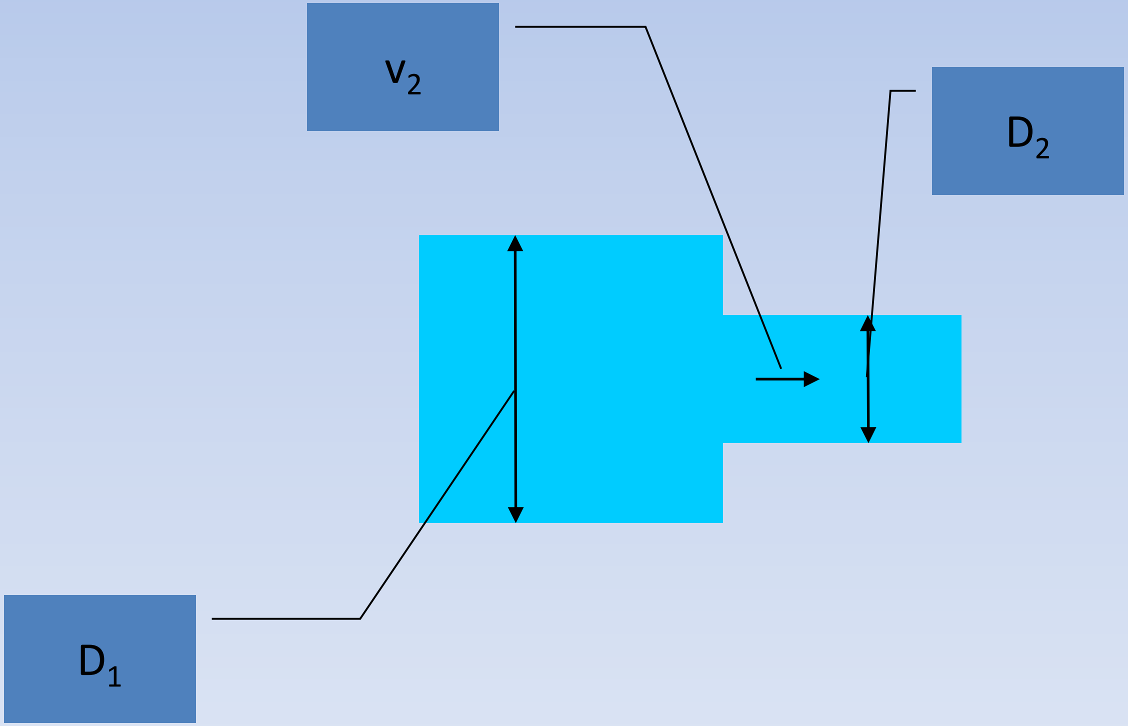
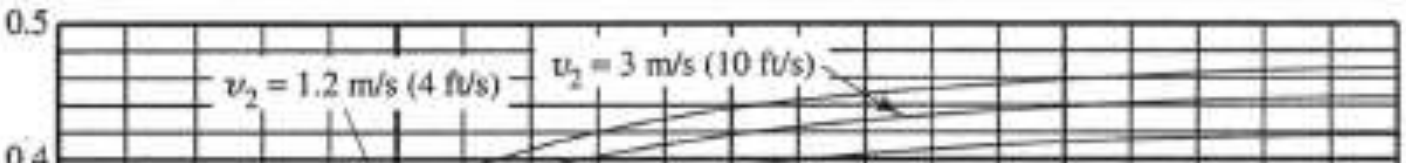


Figure 10-7 Resistance coefficient for sudden contractions



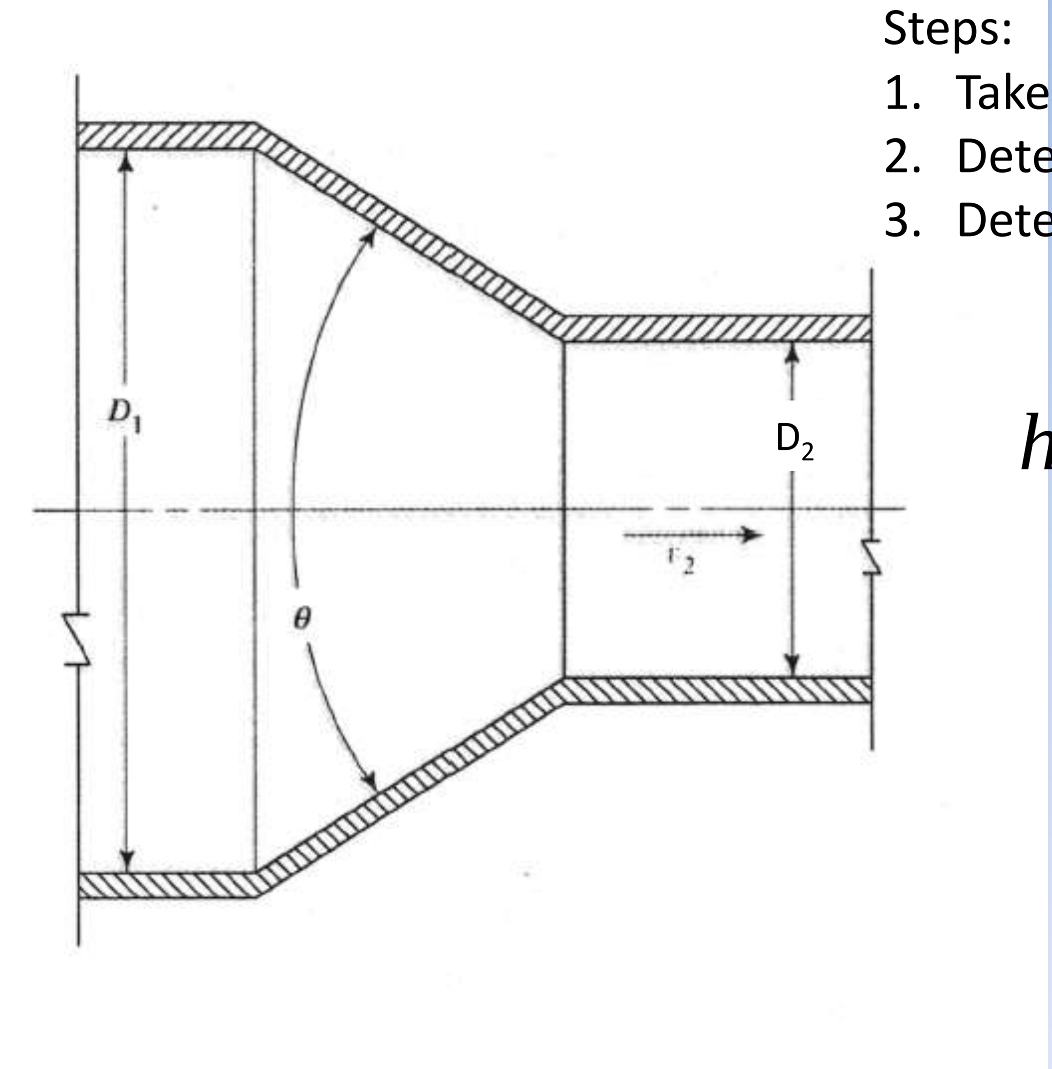
	Velocity v_2								
D_1/D_2	**0.6 m/s**	**1.2 m/s**	**1.8 m/s**	**2.4 m/s**	**3 m/s**	**4.5 m/s**	**6 m/s**	**9 m/s**	**12 m/s**
1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
1.1	0.03	0.04	0.04	0.04	0.04	0.04	0.05	0.05	0.06
1.2	0.07	0.07	0.07	0.07	0.08	0.08	0.09	0.10	0.11
1.4	0.17	0.17	0.17	0.17	0.18	0.18	0.18	0.19	0.20
1.6	0.26	0.26	0.26	0.26	0.26	0.25	0.25	0.25	0.24
1.8	0.34	0.34	0.34	0.33	0.33	0.32	0.31	0.29	0.27
2.0	0.38	0.37	0.37	0.36	0.36	0.34	0.33	0.31	0.29
2.2	0.40	0.40	0.39	0.39	0.38	0.37	0.35	0.33	0.30
2.5	0.42	0.42	0.41	0.40	0.40	0.38	0.37	0.34	0.31
3.0	0.44	0.44	0.43	0.42	0.42	0.40	0.39	0.36	0.33
4.0	0.47	0.46	0.45	0.45	0.44	0.42	0.41	0.37	0.34
5.0	0.48	0.47	0.47	0.46	0.45	0.44	0.42	0.38	0.35
10.0	0.49	0.48	0.48	0.47	0.46	0.45	0.43	0.40	0.36
∞	0.49	0.48	0.48	0.47	0.47	0.45	0.44	0.41	0.38

Source: King, H. W., and E. F. Brater, 1963. *Handbook of Hydraulics*, 5th ed. New York: McGraw-Hill, Table 6–9.

Example

- Determine the energy loss if v_2 = 3.32 m/sec, D_1 = 73.8 mm, D_2 = 25.33 mm

Minor Losses: Gradual Contraction



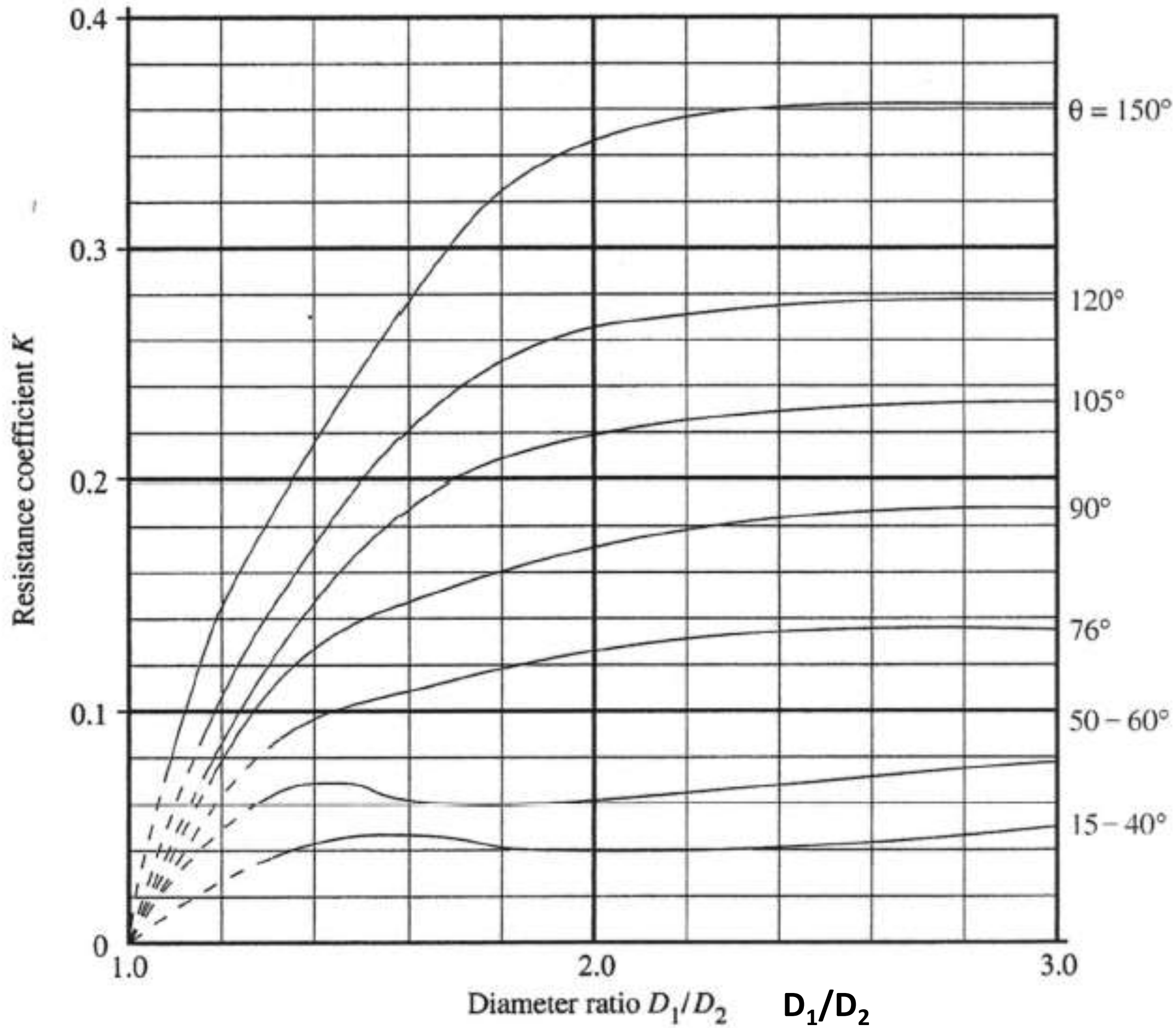
Steps:

1. Take into account D_1, D_2, θ
2. Determine resistance coefficient, K
3. Determine losses, h_L

$$h_L = K\frac{v_1^2}{2g}$$

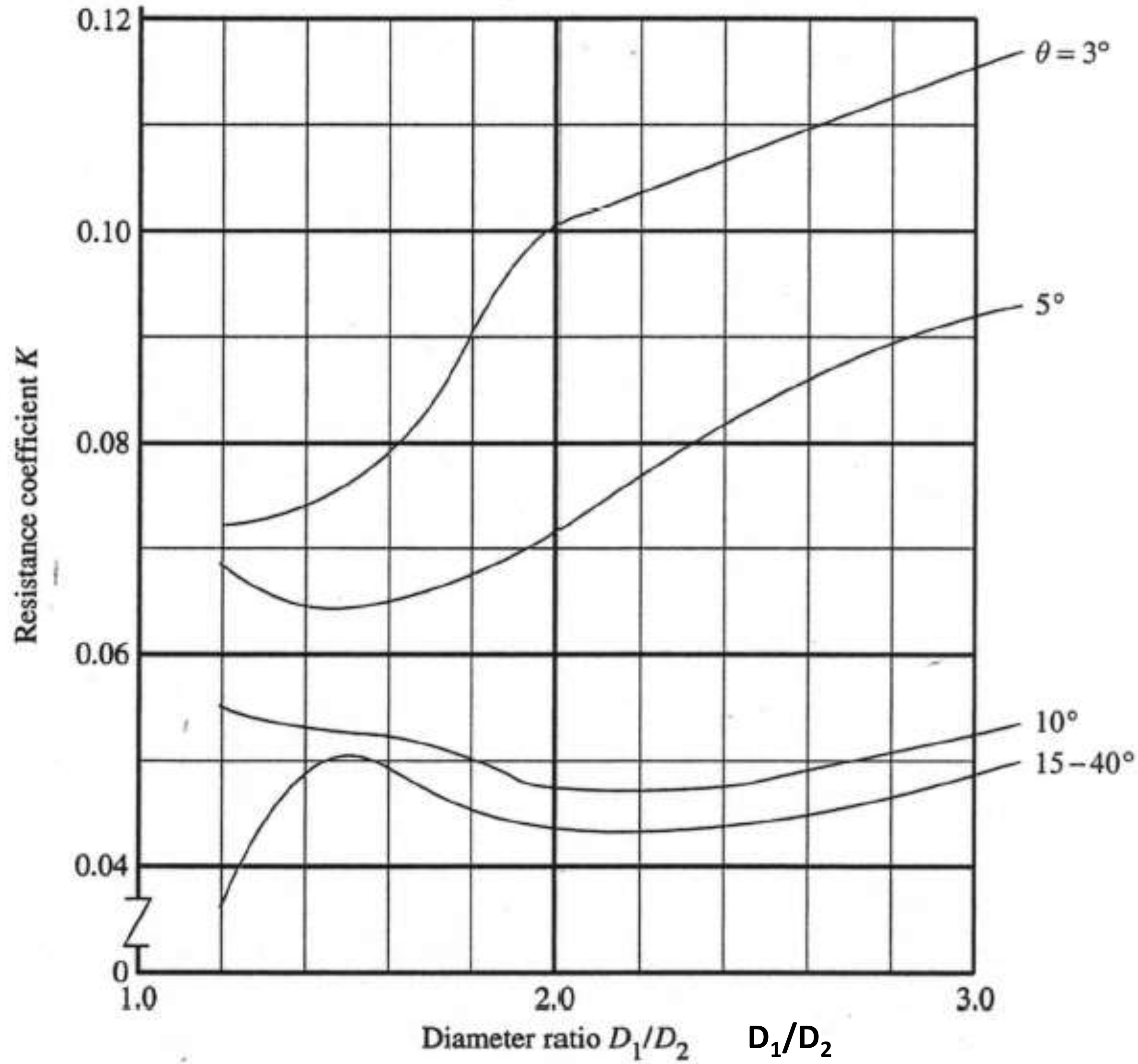
Resistance Coefficient: Gradual Contraction

FIGURE 10.10 Resistance coefficient—Gradual contraction.



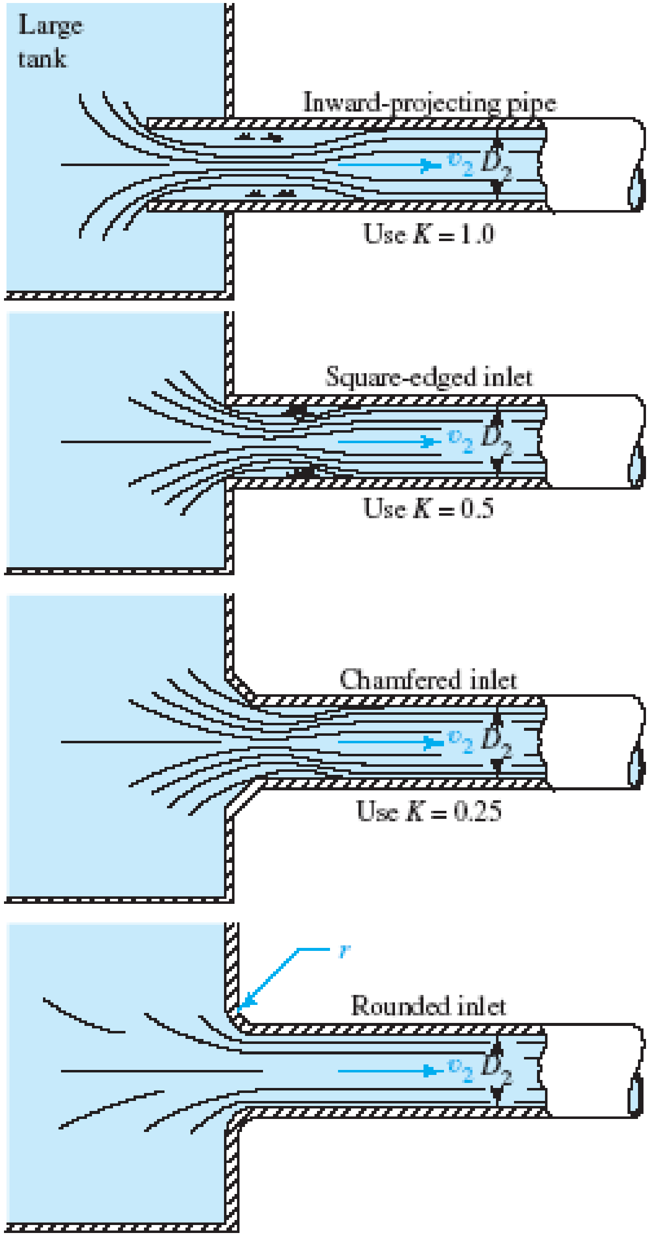
Resistance Coefficient: Gradual Contraction

FIGURE 10.11 Resistance coefficient—Gradual contraction.



D_1/D_2

Entrance Loss



r/D_2	K
0	0.50
0.02	0.28
0.04	0.24
0.06	0.15
0.10	0.09
>0.15	0.04 (Well-rounded)

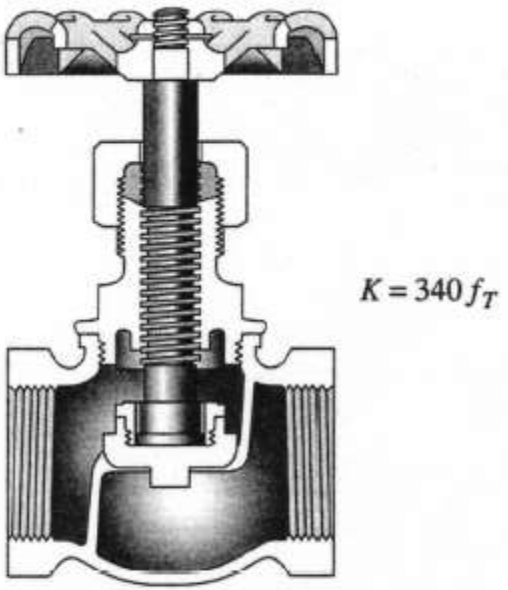
Example

- Determine the energy loss when water flows from a reservoir into a pipe: (a) inward projecting tube and (b) through a well-rounded inlet. $v_2 = 3.32$ m/sec

Valves: Globe Valve

- Relatively inexpensive
- High energy loss for a valve
- Use to control flow

Wide Open	K factor = 10.0
½ Open	K Factor = 12.5



(a)

Valves: Angle Valve

- Relatively inexpensive
- High energy loss for a valve
- Use to control flow
- Similar to globe valve

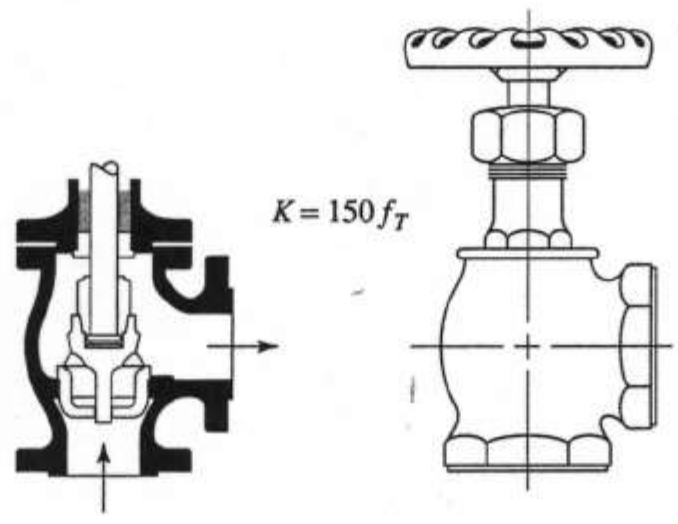
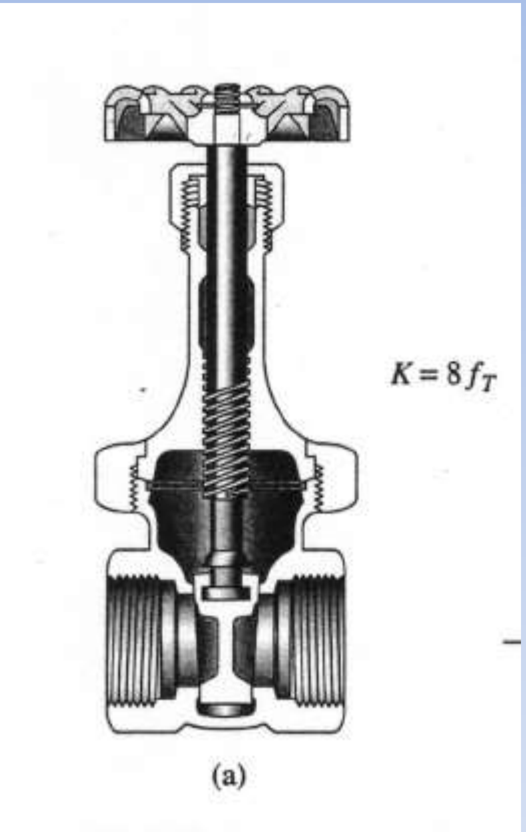


FIGURE 10.15 Angle valve.
(Source: Crane Valves, Joliet, IL)

Valves: Gate Valve

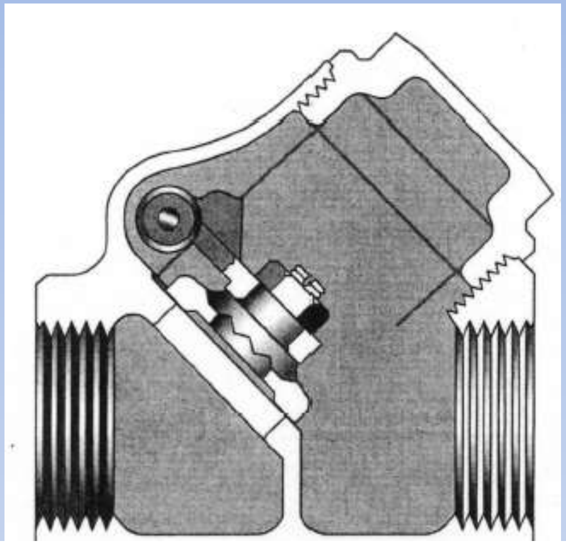
- Relatively expensive
- Low energy loss when compared to globe valve (2.4% of h_L of globe valve)
- Use to control flow

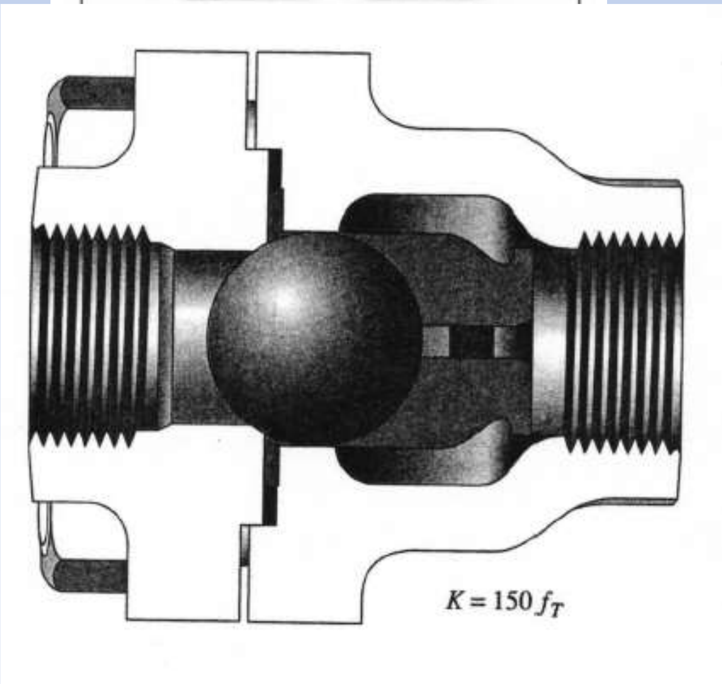
Wide Open	K factor = 0.19
¾ Open	K factor = 0.90
½ Open	K factor = 4.50
¼ Open	K factor = 24.0



Valves: Check Valve

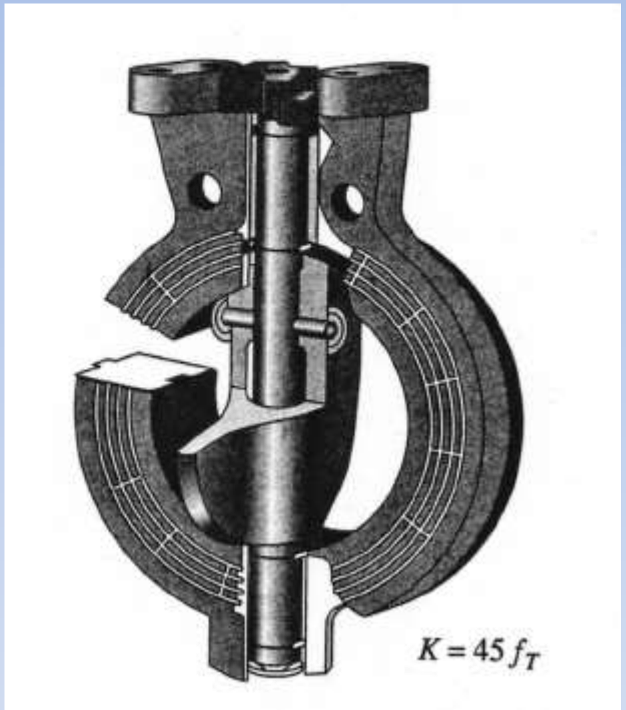
- Used to allow flow <u>only</u> in one direction
- Good for charging fuel tanks, etc.
- Two types: Ball- and Swing-type





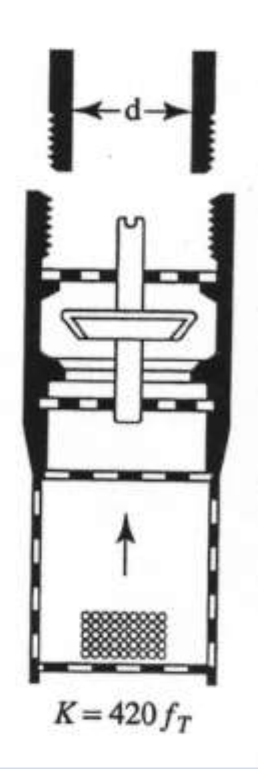
Valves: Butterfly Valve

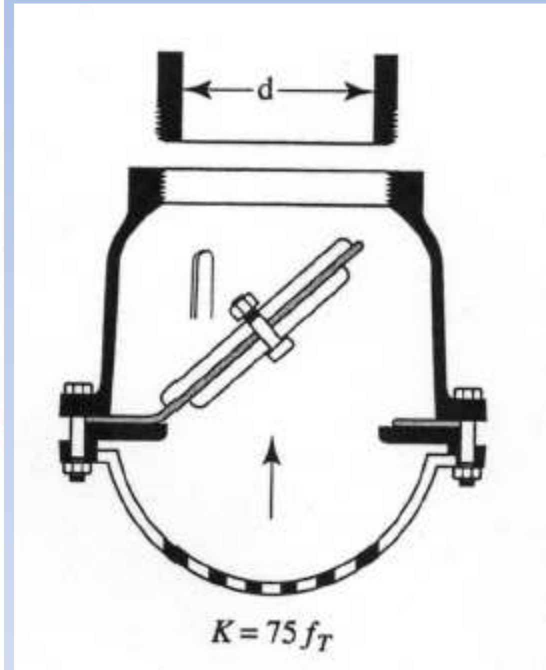
- Used to quickly obstruct flow (on or off by only an one-quarter turn)
- Can be turned on or off by hand or motor



Valves: Foot Valve

- Like a check valve but designed for suction
- Two types: Poppet disc and Hinged disc
- A strainer is attached to it to prevent foreign material from going into piping system





Fittings

Return Bend K factor =2.2

90° elbow K Factor =0.75

45° elbow K Factor =0.42

Standard elbow K factor = 0.9
Standard Tee K factor = 1.8
Ball check valve K factor =4.0

Resistance Coefficients for Valves and Fittings

Type	Equivalent Length in Pipe Diameters L_e/D
Globe valve—fully open	340
Angle valve—fully open	150
Gate valve—fully open	8
—¾ open	35
—½ open	160
—¼ open	900
Check valve—swing type	100
Check valve—ball type	150
Butterfly valve—fully open, 50–200 mm (2–8 in.)	45
—250–350 mm (10–14 in.)	35
—400–600 mm (16–24 in.)	25
Foot valve—poppet disc type	420
Foot valve—hinged disc type	75
90° standard elbow	30
90° long radius elbow	20
90° street elbow	50
45° standard elbow	16
45° street elbow	26
Close return bend	50
Standard tee—with flow through run	20
—with flow through branch	60

Source: Crane Valves, Signal Hill, CA.

Equivalent length technique

- Find a length of pipe that for the same flow rate would produce the same head loss as the valve or fitting
- The length of the pipe is called the equivalent length.

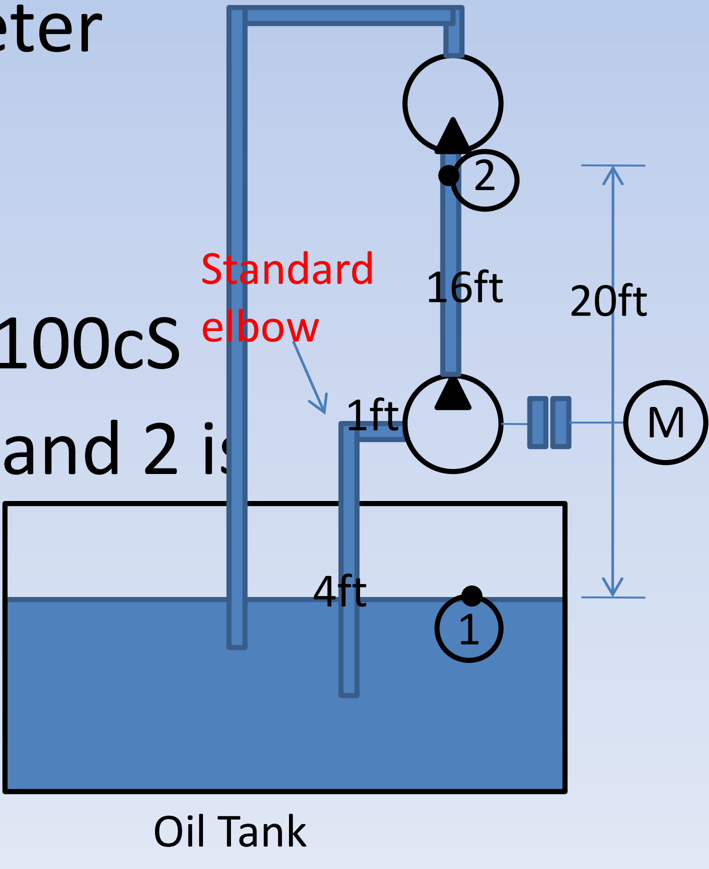
$$H_{L(valve\ or\ fitting)} = H_{L(pipe)}$$

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

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

Example

- The pump is adding 5hp to the system
- Pump flow is 30 gpm
- The pipe has a 1 in inside diameter
- The SG of the oil is 0.9
- The pressure at point 1 is 0 psig
- The kinematic viscosity of oil is 100cS
- Elevation difference between 1 and 2 is 20ft
- Find the pressure at point 2



Example Solution

$$Z_1 + \frac{P_1}{\gamma} + \frac{v_1^2}{2g} + h_p - h_m - h_L = Z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g}$$

Assuming $h_m = 0, v_1 = 0, Z_2 - Z_1 = 20\,ft$, and $P_1 = 0\,gage, \gamma = (SG)\gamma_{water} = (0.9)62.4 = 56.2\,lb/ft^3$

$$Z_1 + 0 + 0 + h_p - 0 - h_L = Z_2 + \frac{P_2}{\gamma} + \frac{v_2^2}{2g} \Rightarrow \frac{P_2}{\gamma} = Z_1 - Z_2 + h_p - h_L - \frac{v_2^2}{2g}$$

$$h_p = \frac{(3950)(5)}{(30)(0.9)} = 732\,ft$$

$$v_2 = \frac{Q}{A} = \frac{\frac{30}{449}(ft/s^2)}{\frac{\pi}{4}\left(\frac{1}{12}\right)^2 (ft^2)} = 12.2\,ft/s \Rightarrow \frac{v_2^2}{2g} = \frac{(12.2)^2}{64.4} = 2.4\,ft$$

$$N_R = \frac{7740v(ft/s) \times D(in)}{\nu(cS)} = \frac{7740(12.2)(1)}{100} = 944 \Rightarrow f = \frac{64}{N_R} = \frac{64}{944} = 0.0678$$

$$H_L = f\left(\frac{L_{eTOT}}{D}\right)\left(\frac{v^2}{2g}\right), \text{where } L_{eTOT} = L_{TOT} + L_{e(std\,\text{elbow})} = 16 + 1 + 4 + \left(\frac{KD}{f}\right)_{(std\,elbow)}$$

$$L_{eTOT} = 21 + \left(\frac{0.9(1/12)\,ft}{0.0678}\right) = 22.1\,ft \Rightarrow H_L = 0.0678\left(\frac{22.1}{1/12}\right)(2.4) = 43\,ft$$

$$P_2 = \left(732 - \frac{(12.2)^2}{2(32.2)} - 63\right)(SG)\gamma_{water} = 37500\,lb/ft^2 = 260\,psig$$

Hydraulic Pumps

Chapter 5

Dr. Suleiman BaniHani

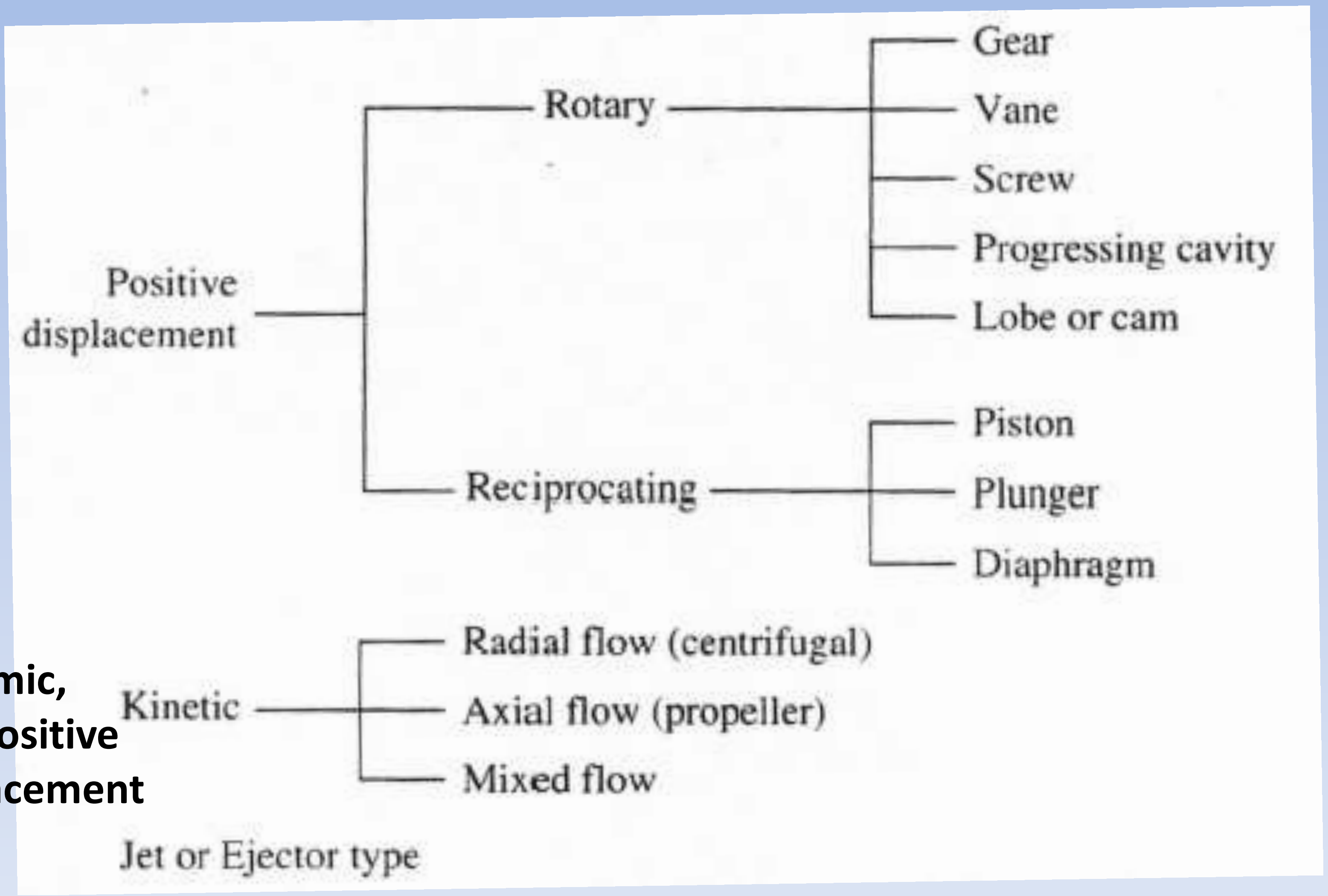
Pumps

$$\frac{P_1}{\gamma} + z_1 + \frac{v_1^2}{2g} + h_A - h_R - h_L = \frac{P_2}{\gamma} + z_2 + \frac{v_2^2}{2g}$$

Spent considerable amount of time quantifying h_L.
Now, we need to learn something about h_A: Pumps

- A pump converts the mechanical energy delivered to the pump by the prime mover, such as an electrical motor, to Hydraulic energy.
- Due to the pump action a partial vacuum is generated at the pump inlet, the atmospheric pressure then forces the fluid into the pump and the pump pushes it to the hydraulic system

Pumps



Dynamic, Nonpositive displacement

Pump Classification

1. Dynamic, nonpositive displacement pumps:
   - Low pressure, high volume flow application.
   - Little use of in fluid power.
   - Normally maximum pressure capability 250-300 psi.
   - Primarily used for fluid transportation.
   - Most common types are centrifugal and axial.

2. Positive displacement pumps
   - Widely used in fluid power.
   - Ejects a fixed amount of fluid into the system per revolution of pump shaft.
   - Capable of overcoming the pressure resulting from mechanical loads and friction.

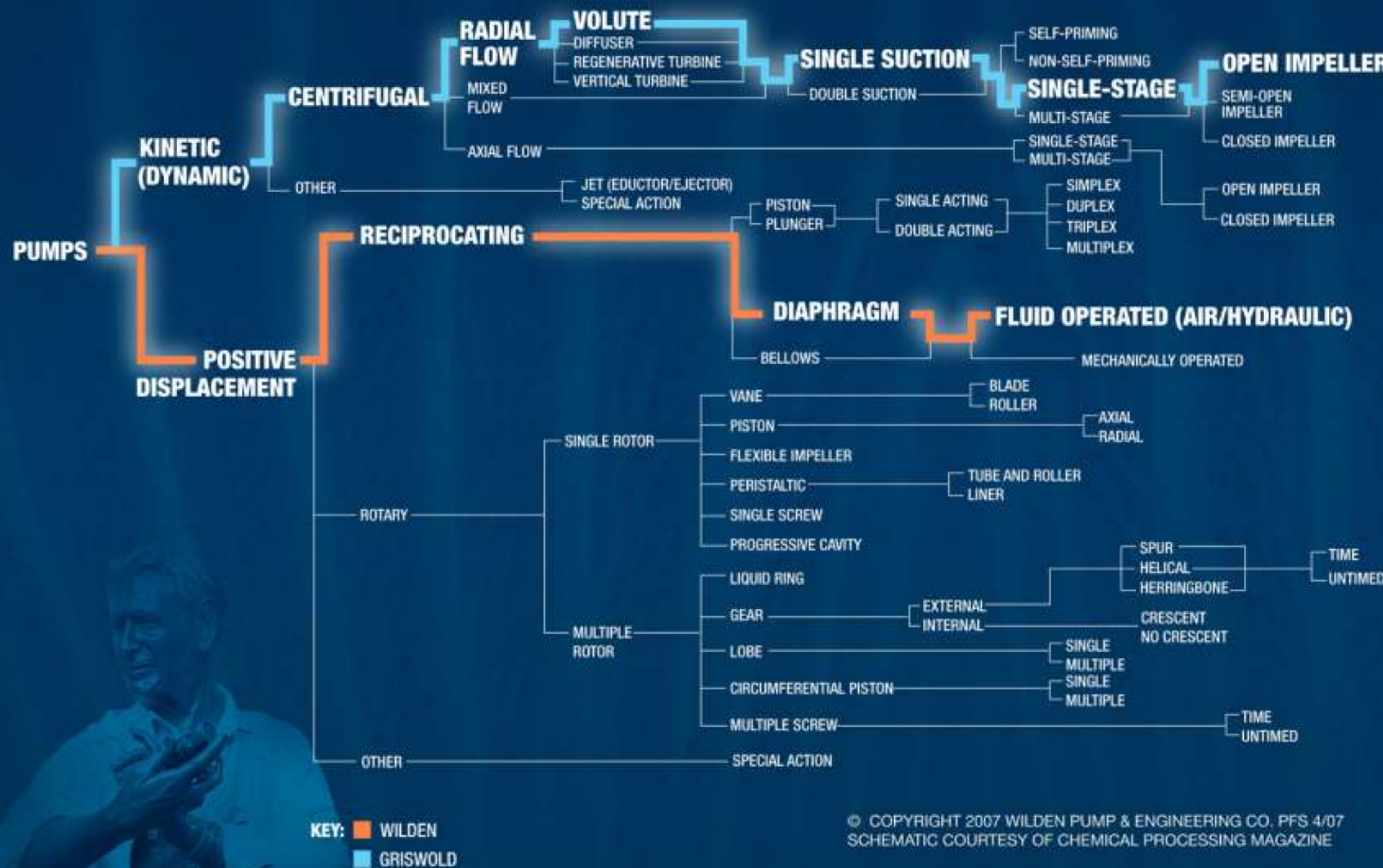
Positive Displacement Pumps Advantages

- High pressure capability (up to 12,000 psi)
- Small compact size.
- High volumetric efficiency.
- Small changes in efficiency throughout the design range of pressure.
- Great flexibility of performance (can operate over a wide range of pressure requirement and speed ranges).

Three main types: Gear, Vane, and Piston.
Many variation within these types, ex. Vane and piston can be either fixed displacement (the amount of fluid ejected per revolution cannot be varied) or variable displacement (in which the flow is changed while the speed of the pump is constant by varying different pump elements)

WILDEN

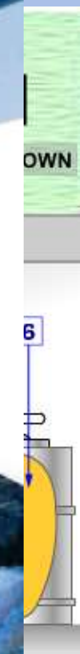
A DOVER COMPANY



Positive Displacement Pumps

– Flow creating device, not pressure.
– Pressure developed dependent upon system characteristics (a pump with open outlet to the atmosphere has a 0 psig pressure while a pump connected to a closed system the pressure will grow until failure), hence a pressure relief valve is used or pressure compensated variable displacement pumps.
– Fixed volume of fluid delivered per rotation

picture courtesy of the Omax Corporation



The Aliko Water Jet Cutting System
A Tool of the Future

Aliko Automation Oy provides its customers with added value via its advanced comprehensive solutions for machining heavy materials. Innovativeness, technological know-how, and expertise, combined with the ability to understand the customer, ensure a reliable and durable system that meets the customer's needs. A comprehensive solution enables rapid and cost-efficient changes and developments in business operations. Aliko is a reliable, long-term partner far into the future.

Durable and Efficient
Developed by Aliko Automation Oy, the abrasive water jet cutting system is excellent for high-precision cutting and for materials that are difficult to machine using traditional methods. The system is particularly designed for water jet cutting.

- Excellent locating precision of +/-0.1 mm: the stable and vibrationless steel structure ensures precision cutting
- Maximum durability: durable silicone seals protect sensitive pieces against humidity and dirt
- Flexible and easy-to-use CNC control allows for the manufacturing of single pieces and industrial series
- Nearly unlimited cutting program length: the maximum program length is equivalent to the storage capacity of the hard disk.

For Heavy Machining
In water jet cutting, a thin water jet is conducted through the nozzles at triple the speed of sound. The jet penetrates almost any material, from steel to glass, from stone to plastic. The ALIKO X–Y desk is particularly designed for heavy machining of thick materials.

Precision Control
The cutting is controlled by the ALIKO CNC precision control, developed by Aliko. The CAD/CAM system used is compatible with the software package.

Abrasive water jet cutting is used in machining hard materials, such as steel, stone, and glass. In abrasive water jet cutting, no heat is transferred onto the material, which also makes the method excellent for cutting thin walls.

Simple User Interface
The ALIKO X–Y desk is controlled with an easy-to-use Windows-based ALIX interface.

ALIX includes several usage-facilitating features, such as groove fixing, the possibility for reverse running, fine-tuning of the cutting speed, and simulated cutting.

All essential cutting parameters can be saved in the material library for reuse.

The cutting unit workstation is easy to connect to the corporate LAN.

A Cost-Efficient Method
Water can be used to cut complex forms, holes, and difficult angles. The method enables the workpieces to be finished at one go, which considerably reduces the amount of re-working. This improves cost-efficiency.

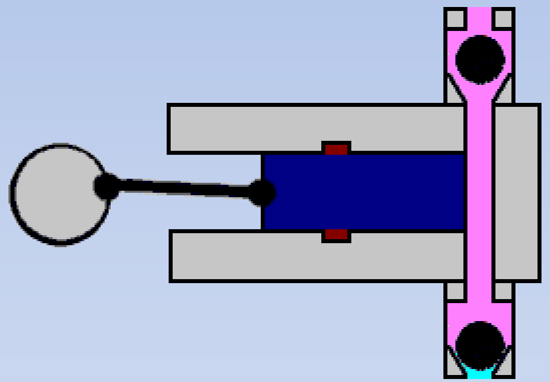
Abrasive water jet cutting is an environmentally-friendly and gentle method. No harmful flue gases are emitted during the cutting process, and distortions or surface damage to the workpiece caused by heat are eliminated. Coated workpieces retain their original surface quality.

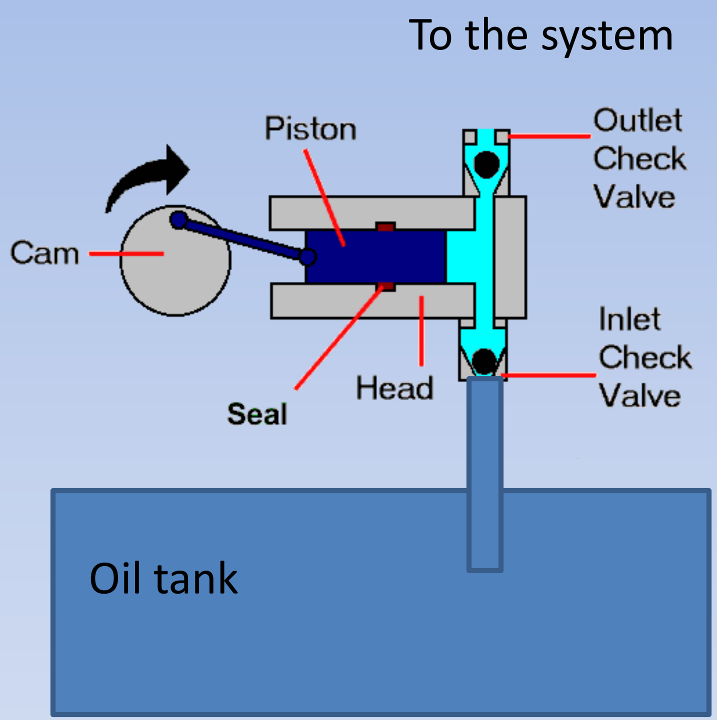
The method suits a wide selection of materials. Hard materials are cut with a mixture of high-pressure water and abrasive sand. Pure water is used in cutting soft and thin materials.

Abrasive Water Jet Cutting
- durable fibrous, layered, and cellular structures
- steel
- titanium
- stone and glass materials
- copper, brass
- aluminium

A 3,000 psi hydraulic pressure from a pump is amplified using a pressure intensifier to a water pressure up to 60,000psi with a diameter of 0.005 to 0.010 in. A piston ration of 1 to 20 is used, conversely the oil flow rate is 20 times the water flow rate.

Pumping Theory



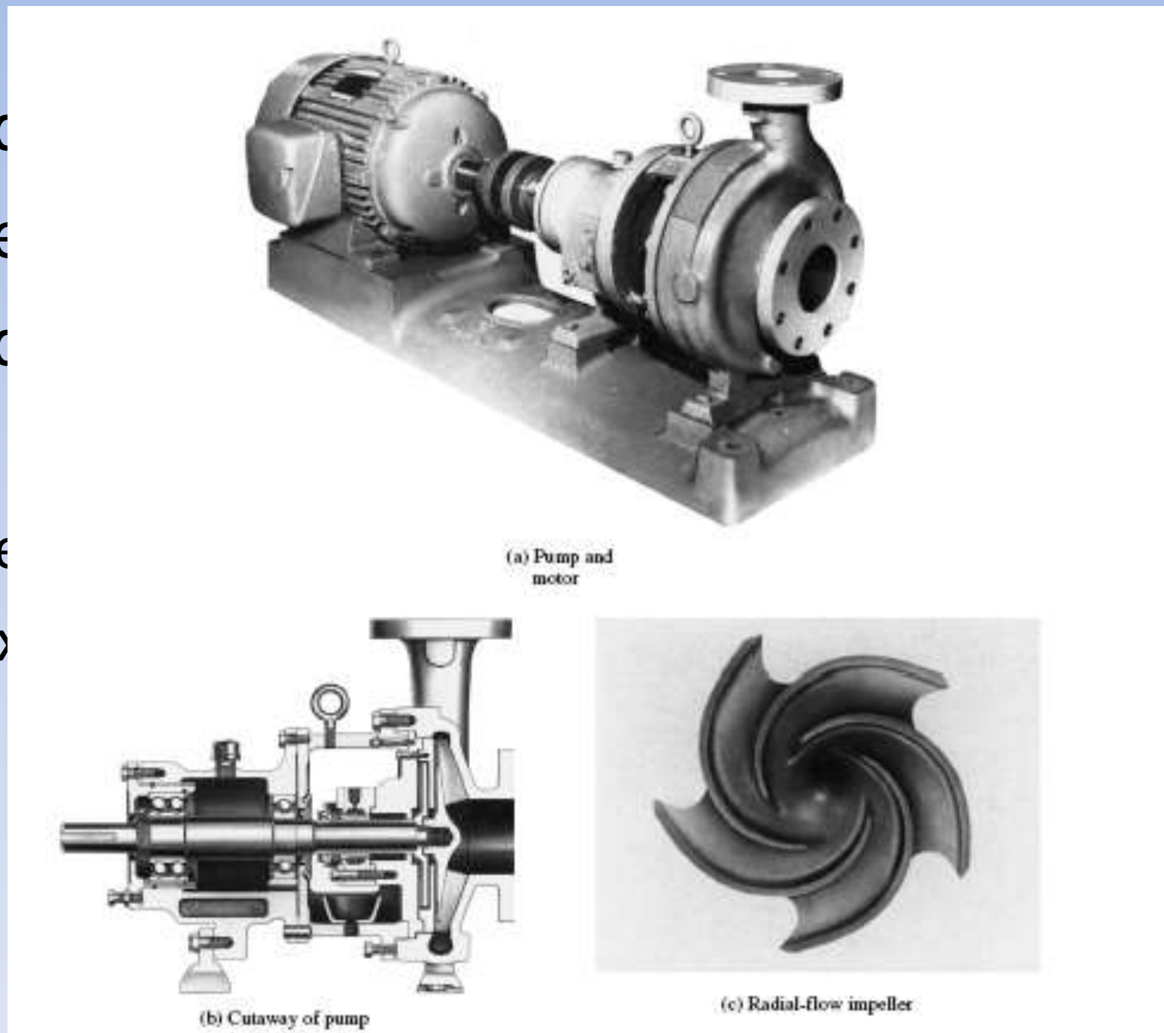


Pump Classification

- Dynamic Pumps (Nonpositive displacement)
  - Centrifugal (Impeller)
  - Axial (Propeller)
- Positive displacement
  - Gear
    - External Gear, Internal Gear, Lope, Gerotor
  - Vane
    - Unbalanced, balanced, Pressure compensated
  - Piston
    - Axial Piston, In-Line Piston, Radial Piston

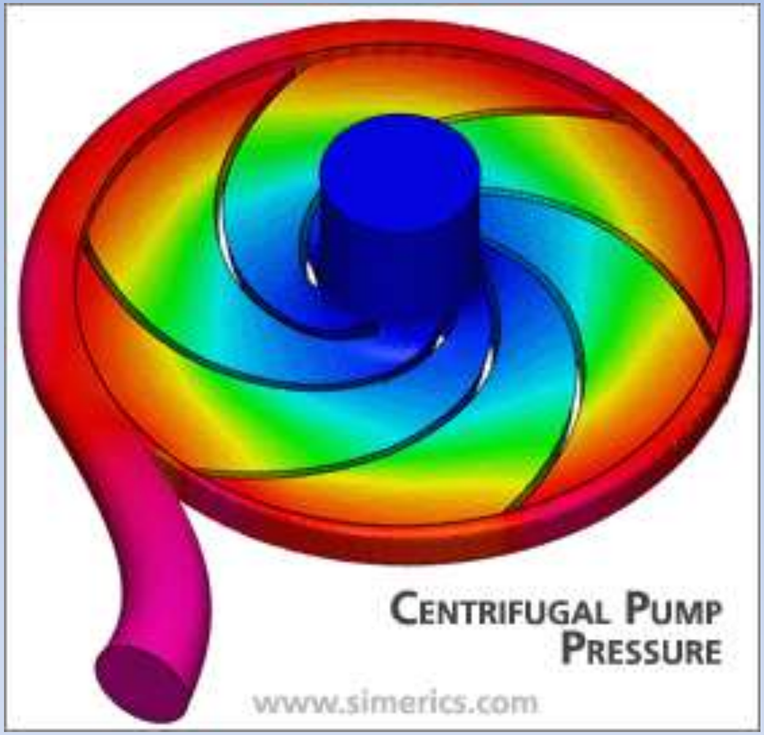
Kinetic (Dynamic) Pumps

- Usec … cations
- Little
- Usec
- Two
  - Ce
  - Ax



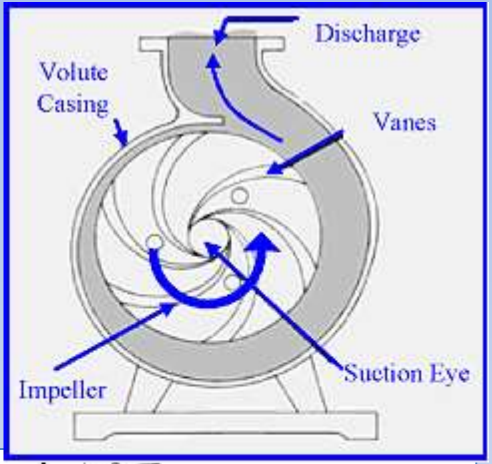
Classification of Pumps

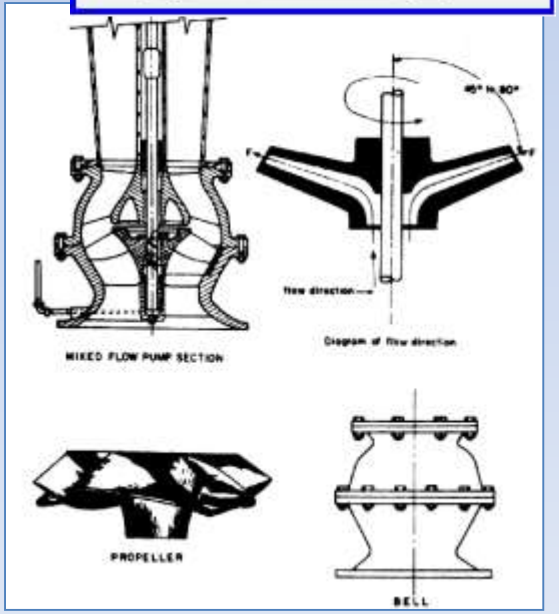
- Centrifugal (Kinetic) Pumps
  - Pressure creating device
  - Converts velocity energy to Pressure Energy (Flow Work)
  - Flow depends upon system characteristics
  - The velocity of the fluid will be the velocity of the impeller at the exit point (discharge)
  - The most popular pump

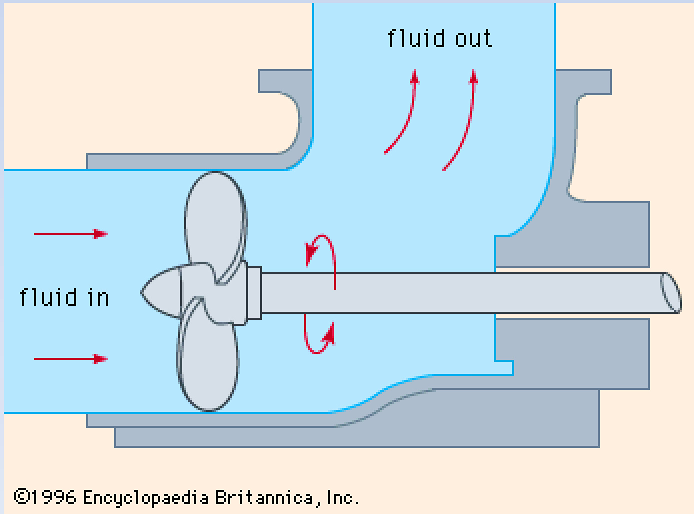


PUMP TYPES

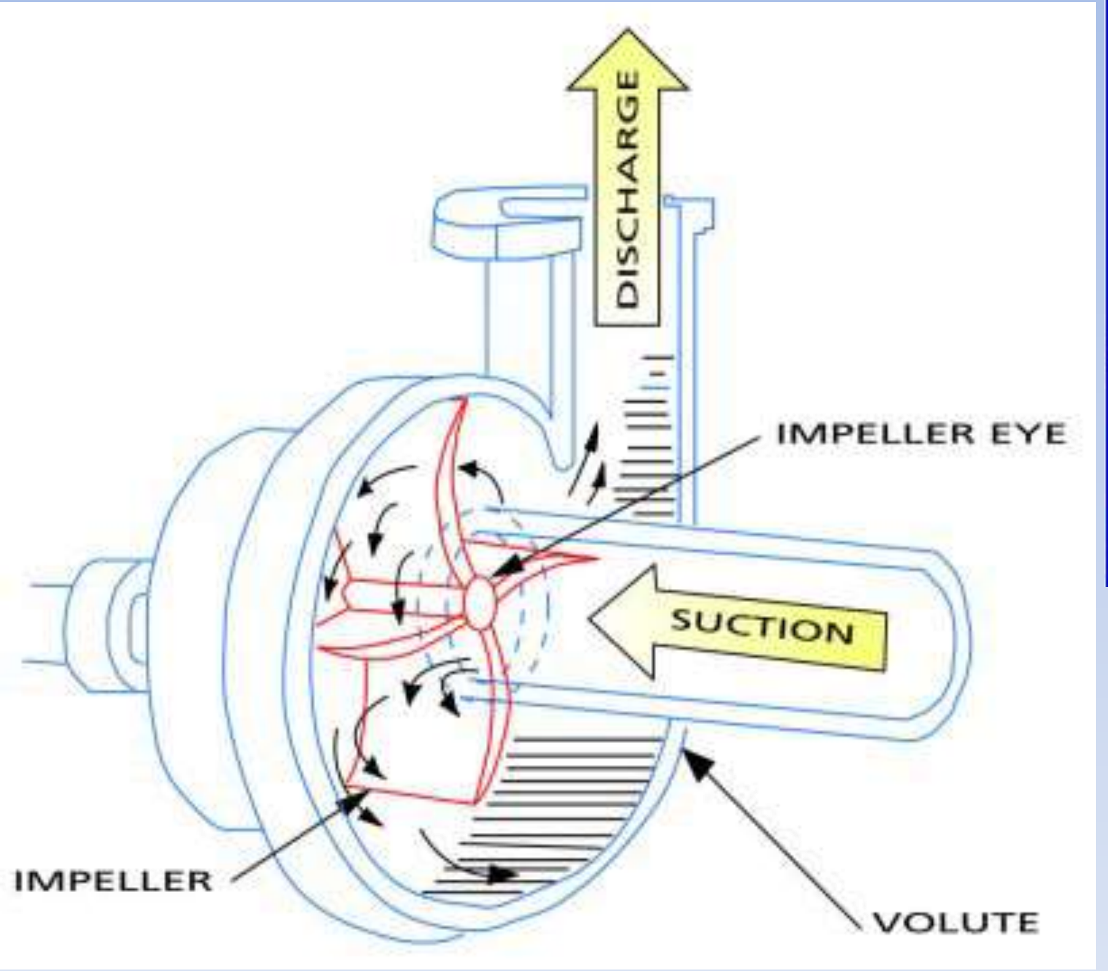
- PUMPS
  -
    - RADIAL FLOW
    - MIXED FLOW
    - AXIAL FLOW

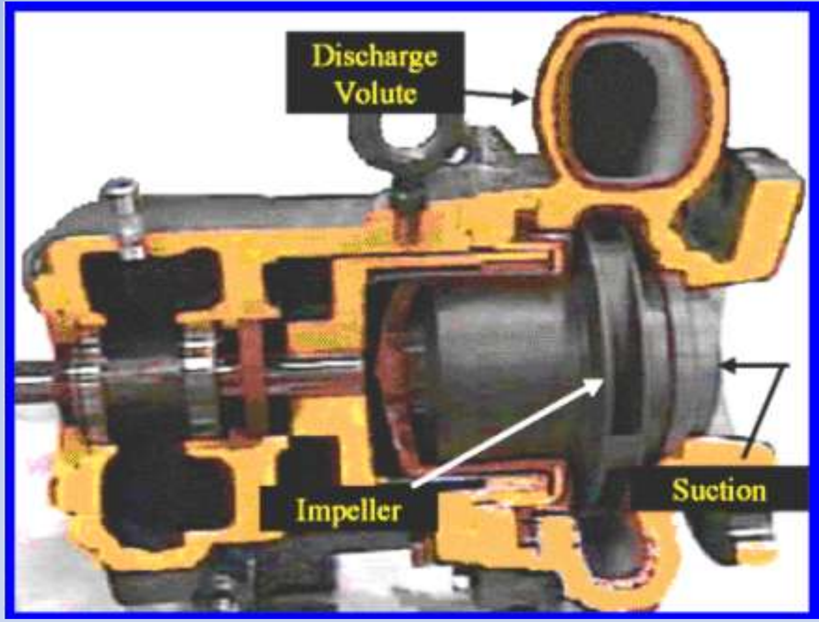






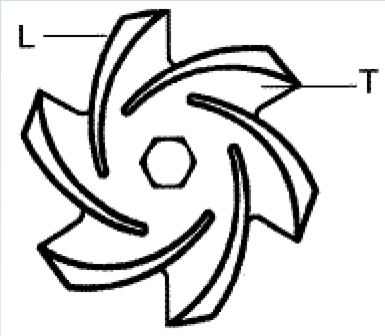
Kinetic Pumps: Centrifugal





Centrifugal: Impeller Design

- **Fully Open Impeller:**
  - Ideally suited for corrosives and abrasives, handles solids and stringy fibers with ease.
  - Allows for simple restoration of clearances when wear takes place.
  - Back pump-out vanes (less shroud material) and/or balance holes reduce pressure on the shaft seal, reduce axial thrust on the bearings.
  - The fluid enters the eye of the impeller where the turning vanes add energy to the fluid and direct it to the discharge nozzle. A close clearance between the vanes and the pump volute prevents most of the fluid from recirculating back to the eye of the impeller.
  - (L) shows the leading edge or higher-pressure side of the impeller. (T) describes the trailing edge of the impeller.
  - Typically screwed to the shaft



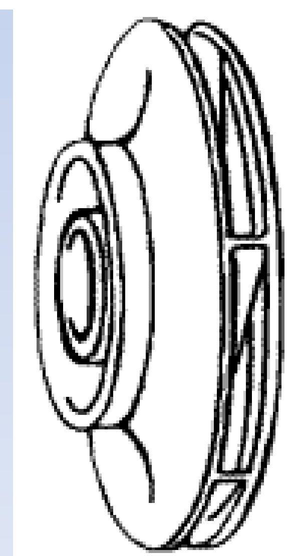
Open Impeller

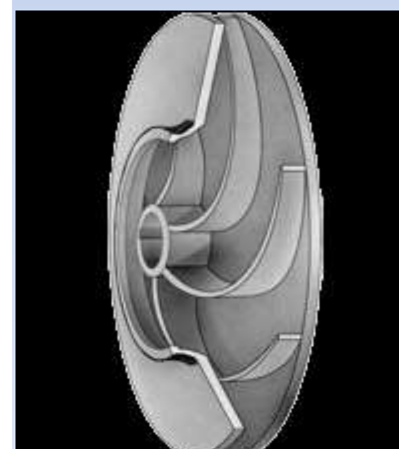
Balance holes

Back Pump Out
Vanes

Centrifugal: Impeller Design

- **Closed Impeller:**
  - The fluid enters the eye of the impeller where the vanes add energy to the fluid and direct it to the discharge nozzle. There is no impeller to volute or back plate clearance to set.
  - Wear rings restrict the amount of discharge fluid that recirculates back to the suction side of the impeller. When this wear ring clearance becomes excessive the wear rings must be replaced.
  - Balance holes are necessary to minimize axial loading.
  - Keyed and bolted to shaft





Closed Impeller

Balance holes

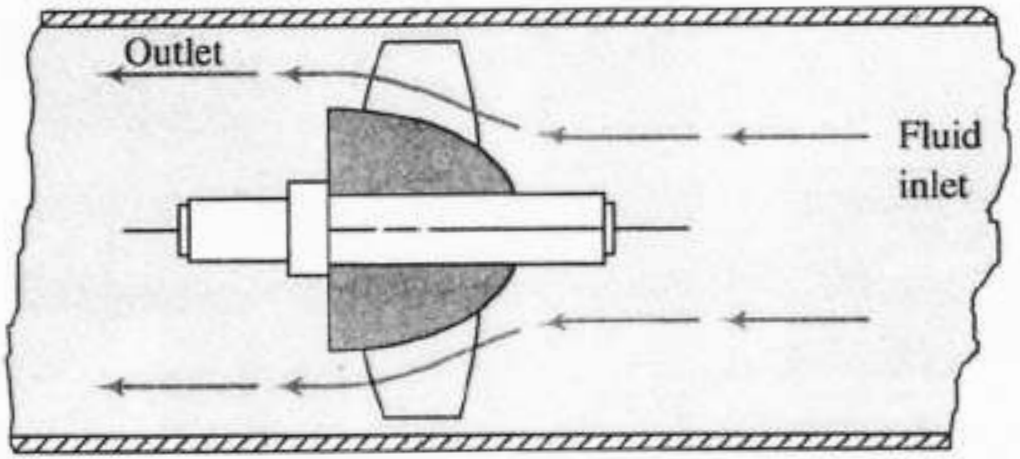
PUMP TYPES

- PUMPS
  - DIFFUSER PUMPS
    - VERTICAL TURBINE
    - HORIZONTAL

PUMP TYPES

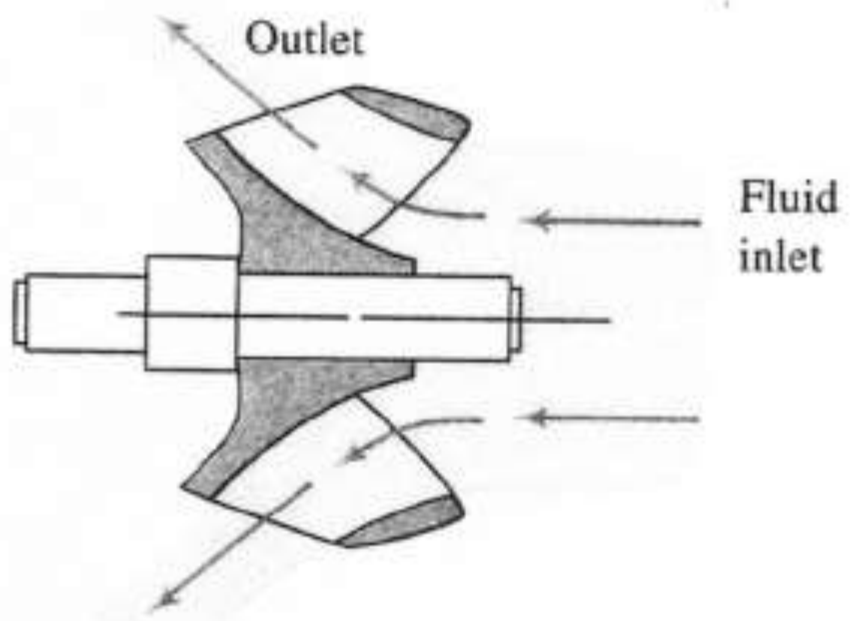
- CONCENTRIC VOLUTE PUMPS
  - REGENERATED TURBINE
  - HIGH SPEED
  - RADIAL VANE
  - RECESSED IMPELLER

Kinetic Pumps: Axial

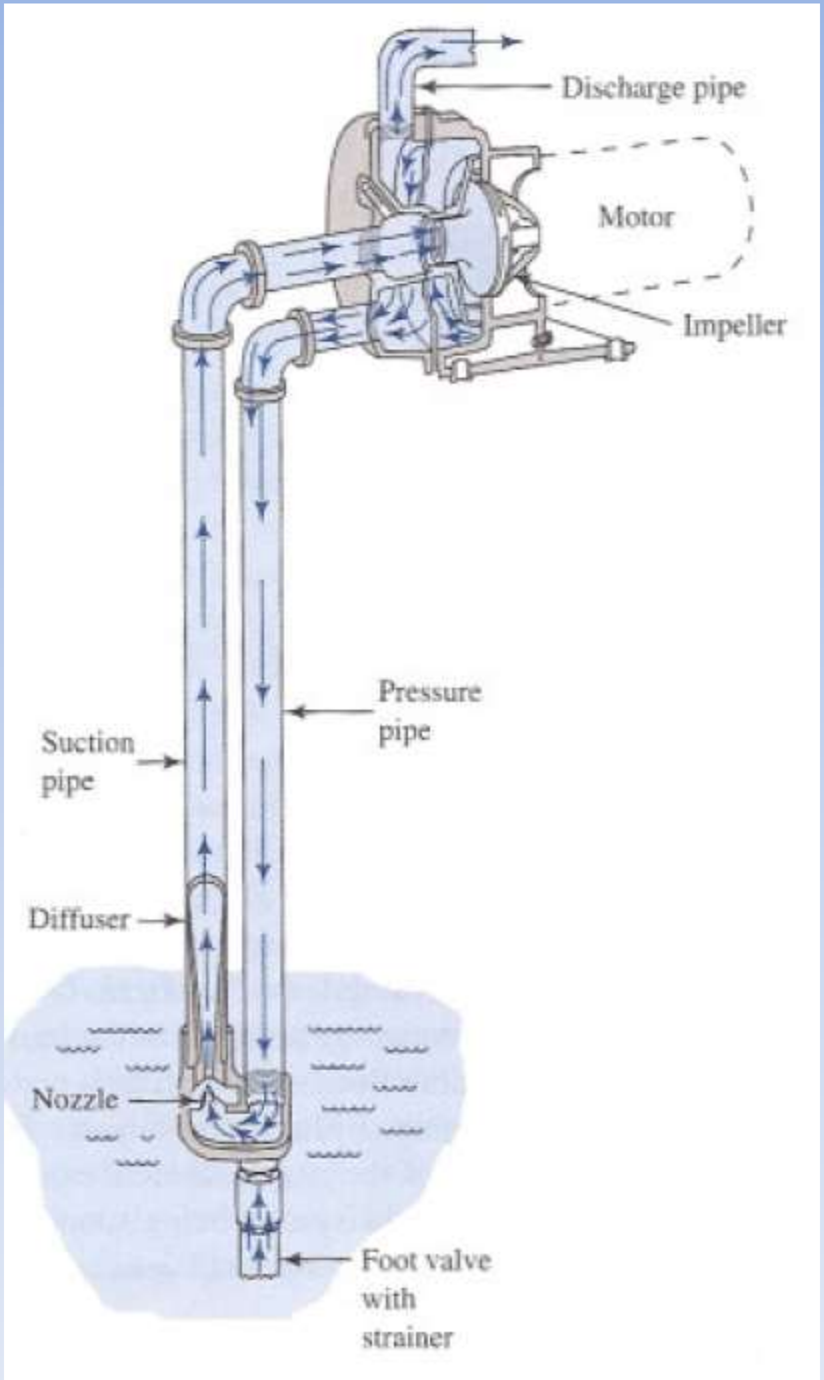


(c) Axial flow impeller (propeller)

Kinetic Pumps: Mixed Flow



(b) Mixed flow impeller



Jet Pumps

Good for dewatering operations

Self-Priming Pumps

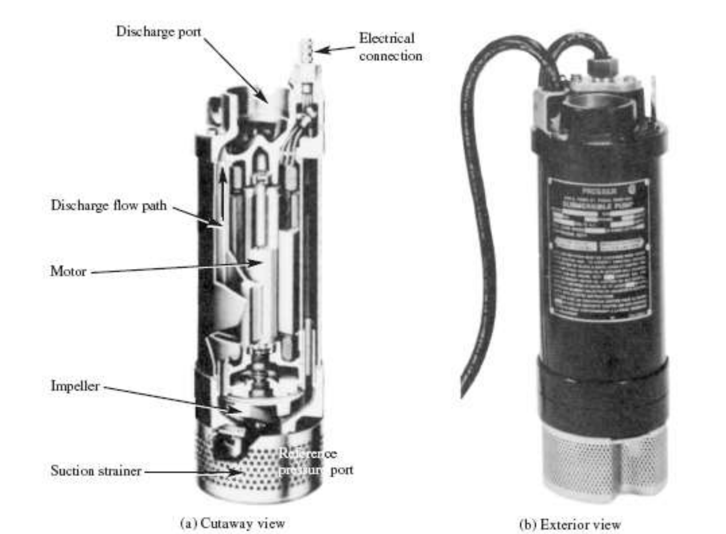
- Priming is required when the inlet side of the pump is initially higher than the fluid level
  – No fluid is in contact with inlet port
- Self-priming pumps
  – Prime the fluid by recirculation until the pump is flooded

http://www.cranepumps.com/

http://www.coleparmer.com/techinfo/techinfo.asp?htmlfile=SelectingLiqPumps.htm

Submersible Pumps

- Entire pump is submerged into fluid
  - Use in sanitary applications, construction sites, etc.
  - The motor is sealed so no water can get into it
  - Pump components must chemically compatible with the fluid being pumped



(a) Cutaway view (b) Exterior view

Efficiencies

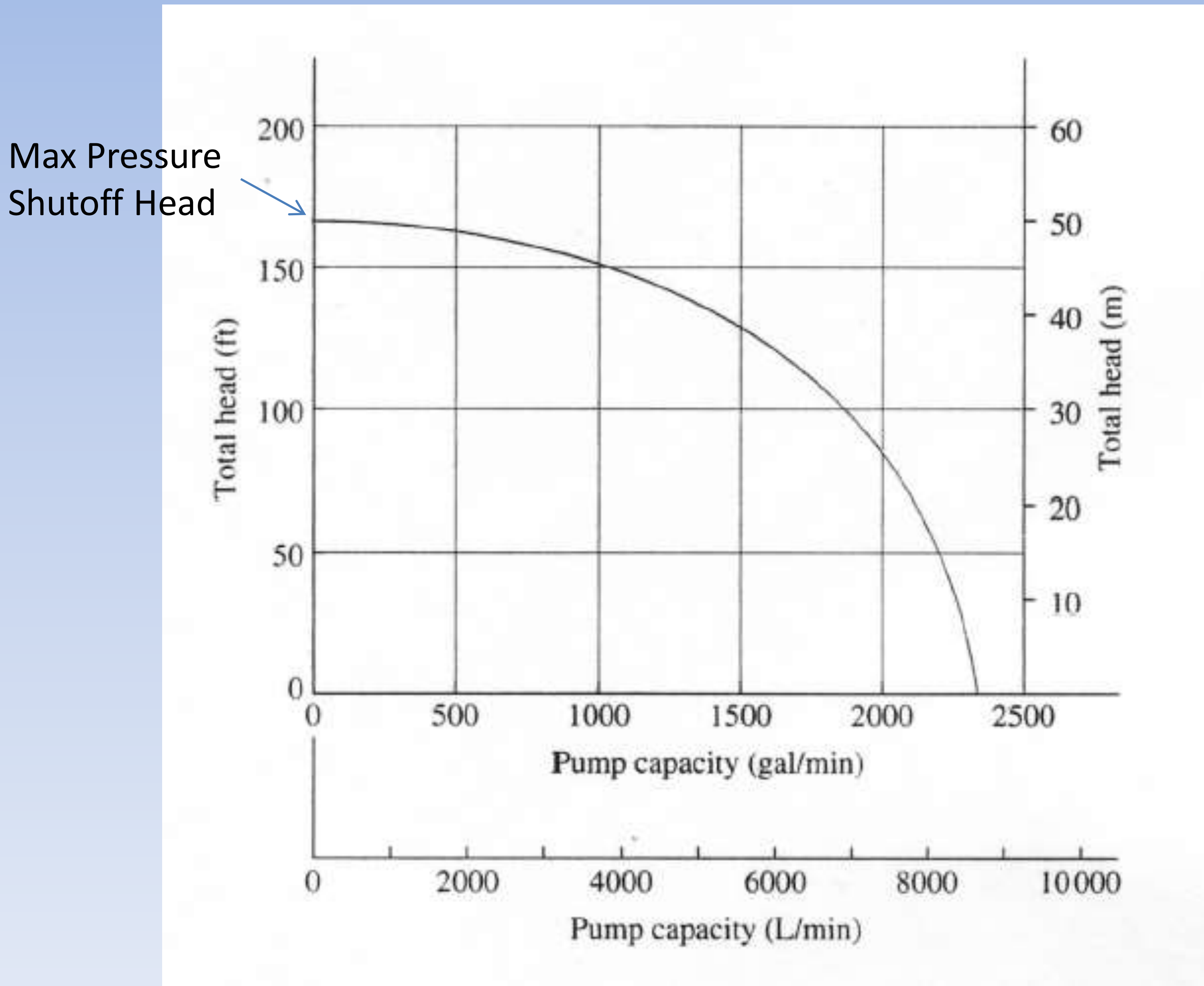
- Volumetric

  $\eta_{volumetric}$ = Actual flow rate/Theoretical flow rate

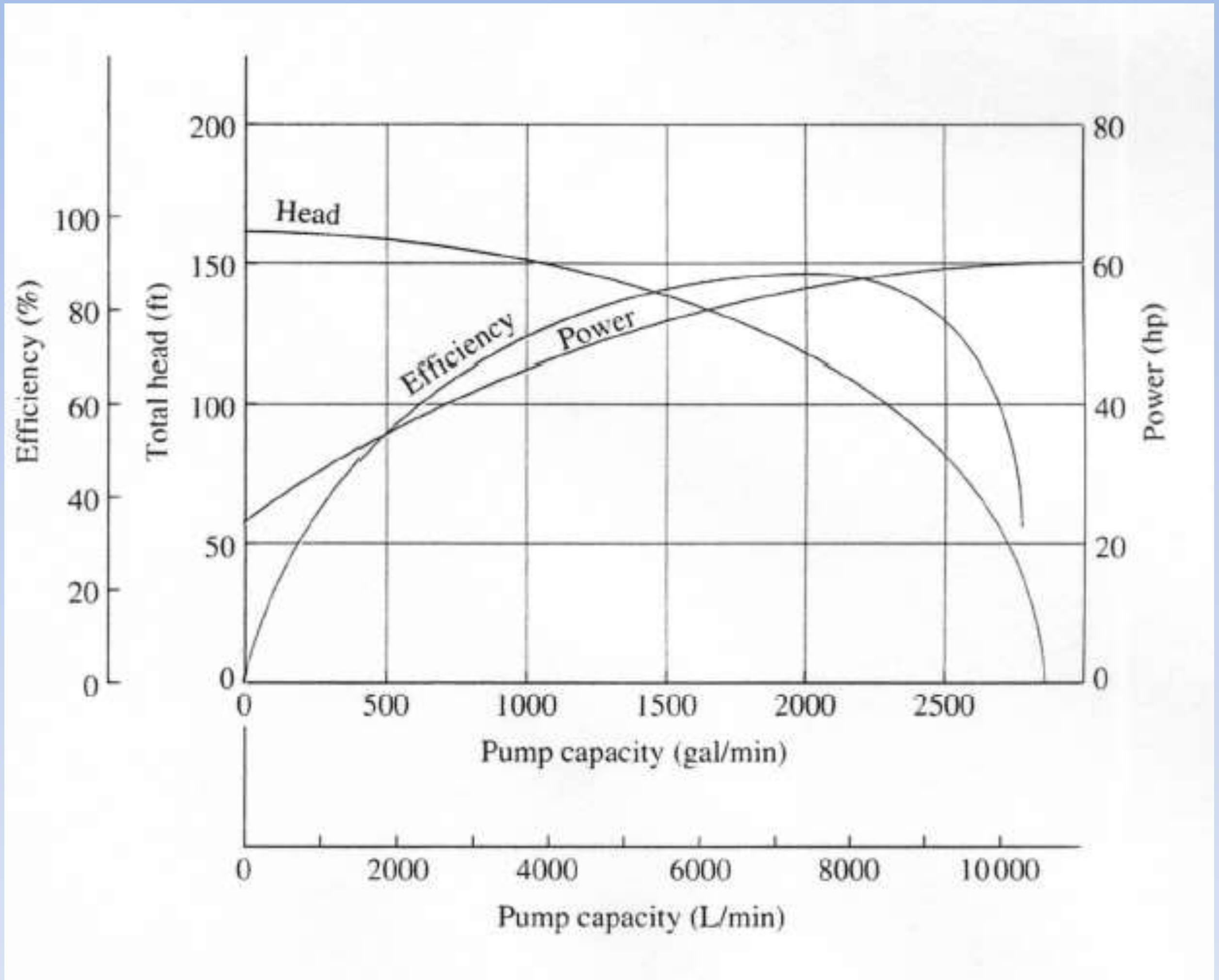
- Overall

  $\eta_{overall}$ = Fluid power/Power delivered to pump

Centrifugal Pumps Performance



Centrifugal Pumps Performance



Affinity Laws for Centrifugal Pumps

- Affinity laws are used to predict how centrifugal pump performance changes with changes in speed (RPM) and impeller diameter
- These relationships, called *affinity laws*, are listed here.
- The symbol *N* refers to the rotational speed of the impeller, usually in revolutions per minute (r/min, or rpm).

Affinity Laws for Centrifugal Pumps

- When *speed varies*:

1. Capacity varies directly with speed:

$$\frac{Q_1}{Q_2} = \frac{N_1}{N_2} \qquad (13\text{–}5)$$

2. The total head capability varies with the square of the speed:

$$\frac{h_{a_1}}{h_{a_2}} = \left(\frac{N_1}{N_2}\right)^2 \qquad (13\text{–}6)$$

Affinity Laws for Centrifugal Pumps

3. The power required by the pump varies with the cube of the speed:

$$\frac{P_1}{P_2} = \left(\frac{N_1}{N_2}\right)^3 \qquad (13\text{–}7)$$

Affinity Laws for Centrifugal Pumps

- When *impeller varies*:

1. Capacity varies directly with impeller diameter:

$$\frac{Q_1}{Q_2} = \frac{D_1}{D_2} \tag{13–8}$$

2. The total head varies with the square of the impeller diameter:

$$\frac{h_{a_1}}{h_{a_2}} = \left(\frac{D_1}{D_2}\right)^2 \tag{13–9}$$

Affinity Laws for Centrifugal Pumps

3. The power required by the pump varies with the cube of the impeller diameter:

$$\frac{P_1}{P_2} = \left(\frac{D_1}{D_2}\right)^3 \qquad (13\text{–}10)$$

- Efficiency remains nearly constant for speed changes and for small changes in impeller diameter.

POSITIVE DISPLACEMENT PUMPS

- Ejects a fixed quantity of fluid per revolution of pump shaft.
- Does not depend on system pressure.
- Widely used in fluid power systems.
- A pressure relief valve is used to protect the pump against overpressure by diverting pump flow back to the hydraulic tanks.
- Can be classified according to the motion of internal elements
  – Rotary
  – Reciprocating

http://www.coleparmer.com/techinfo/techinfo.asp?htmlfile=SelectingLiqPumps.htm

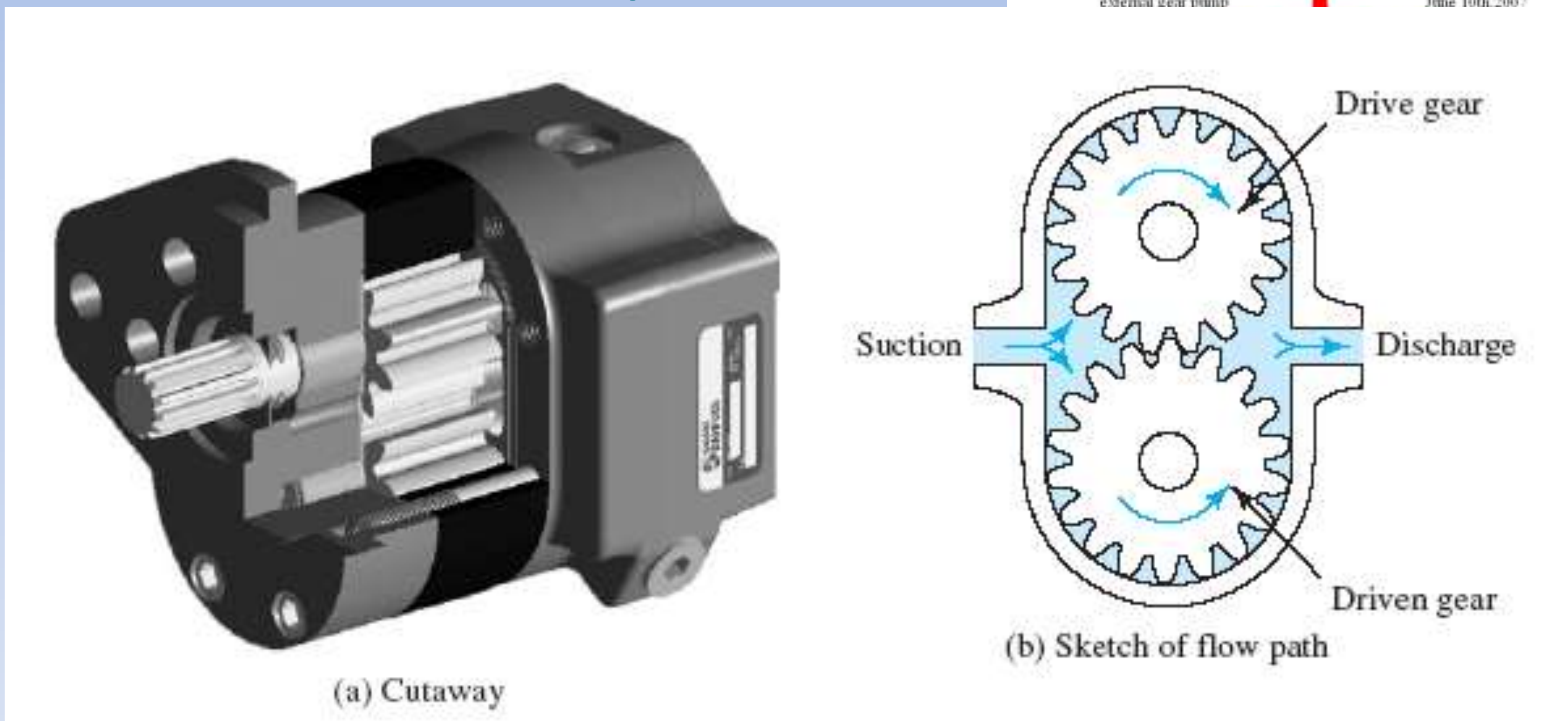
Positive Displacement Pumps

Three main types

- Gear Pumps (fixed displacement only)
  - External gear pumps
  - Internal gear pumps
  - Lobe pumps
  - Screw pumps
- Vane Pumps
  - Unbalanced vane pumps (fixed or variable displacement)
  - Balanced vane pumps (fixed displacement)
- Piston Pumps (fixed or variable displacement)
  - Axial piston pumps
  - Radial Piston pumps.

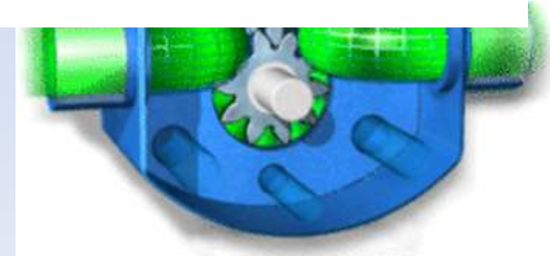
Gear Pumps

- External Gear Pumps



(a) Cutaway

(b) Sketch of flow path



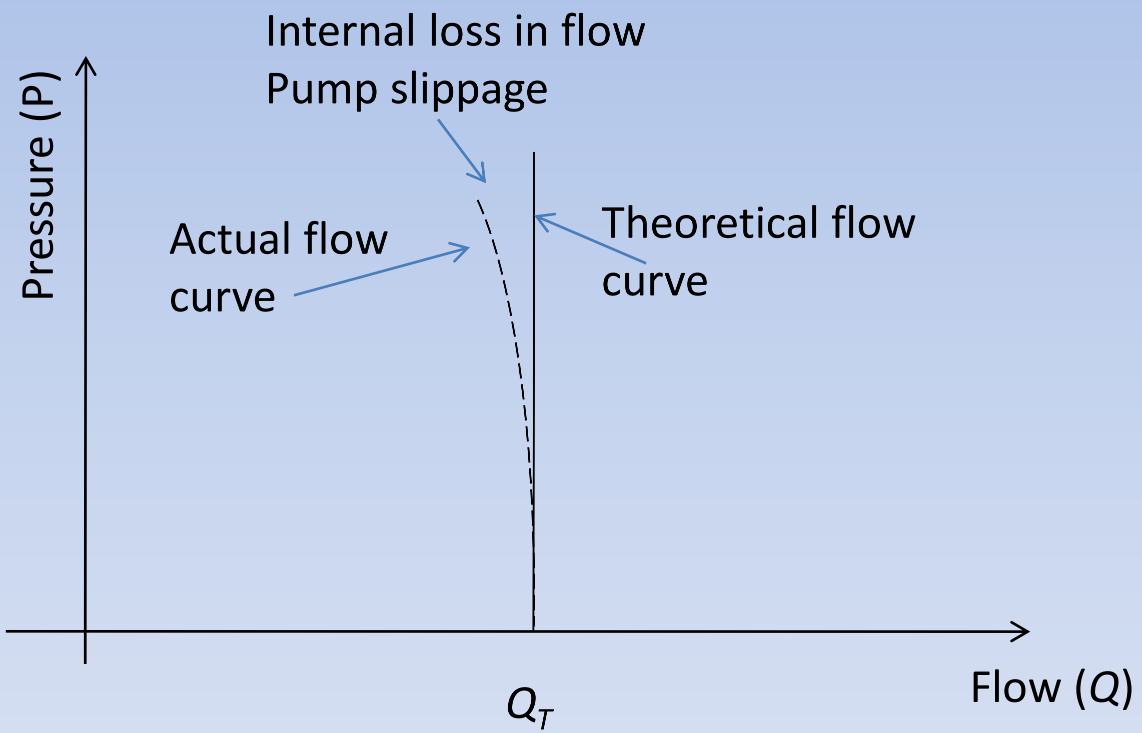
External Gear Pump

- D_o = outside diameter of gear teeth (in,m)
- D_i = inside diameter of gear teeth (in,m)
- L = width diameter of gear teeth (in,m)
- V_D = Displacement volume of pumps (in^3/rev, m^3/rev) = $V_D = \frac{\pi}{4}\left(D_o^2 - D_i^2\right)L$
- N = rpm of pump
- Q_T = theoretical pumps flow rate

$$Q_T(in^3/\min) = V_D(in^3/rev) \times N(rev/\min)$$

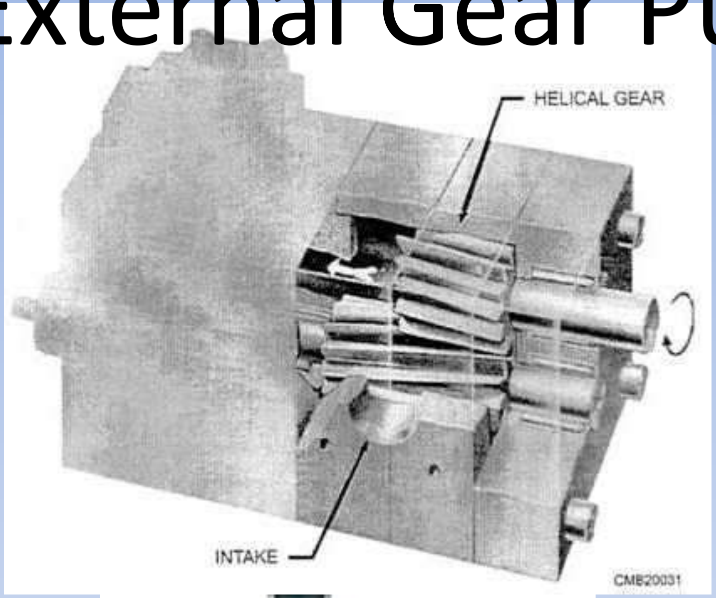
$$Q_T(gpm) = \frac{V_D(in^3/rev) \times N(rev/\min)}{231}$$

External Gear Pump

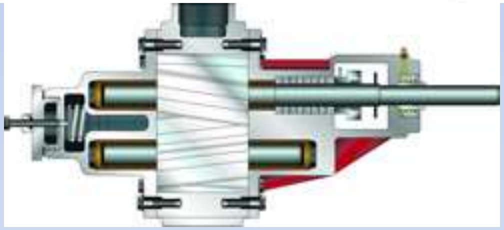


$$\eta_v = \frac{Q_A}{Q_T}$$

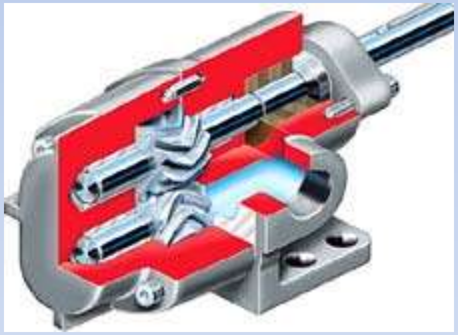
External Gear Pump



advantages:
Shorter packaging
High pressure capability
Good contamination resistance

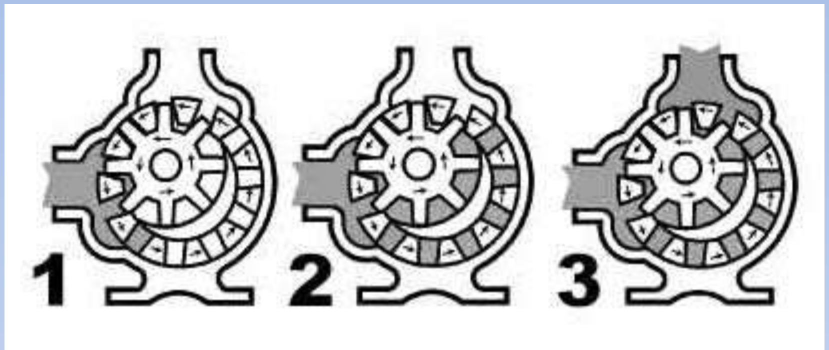


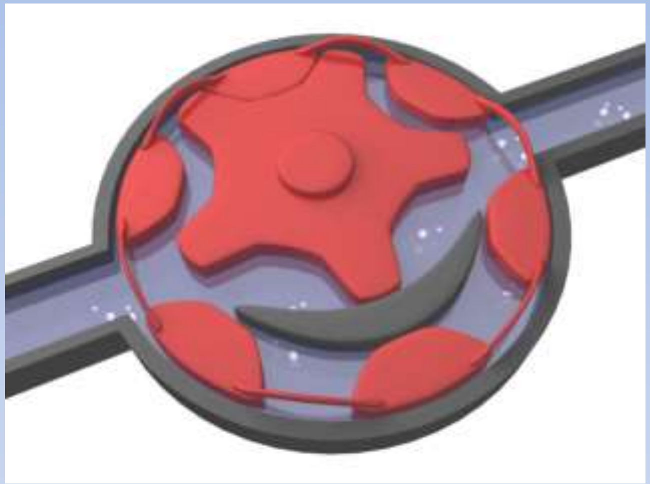
Helical gear pump
Disadvantages: Creates curved flow, Pulsing effect, High stress on bearings, Difficult to clean,No flow stability
Used for low pressure



Herringbone gear pump
Advantages: Precise flow control, No Pulsing, Even distribution, Straight Extrusions, Dimensional stability, Easy cleaning
High pressure up to 3000psi

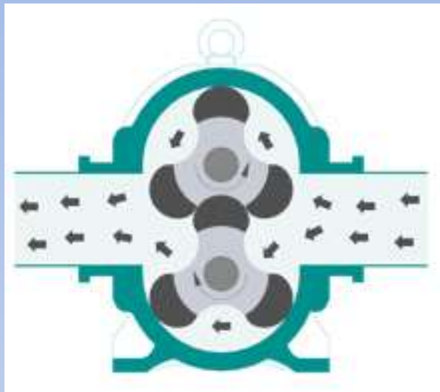
Internal gear Pump

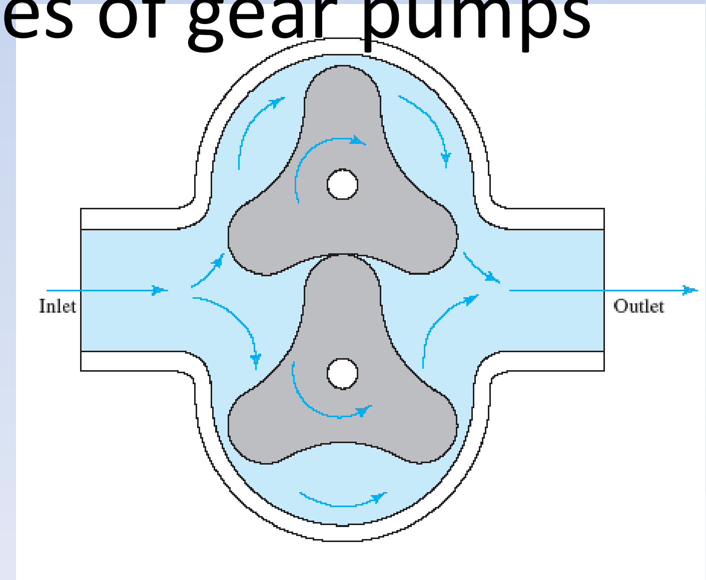


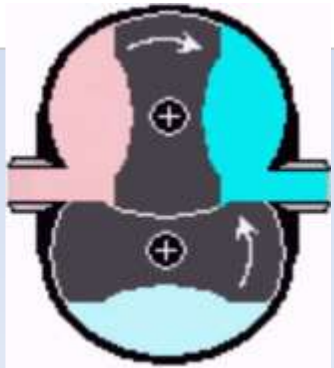


Lobe pump

- Both lobes are driven externally.
- No meshing, less noise
- Great amount of pulsation
- Volumetric displacement is larger than other types of gear pumps

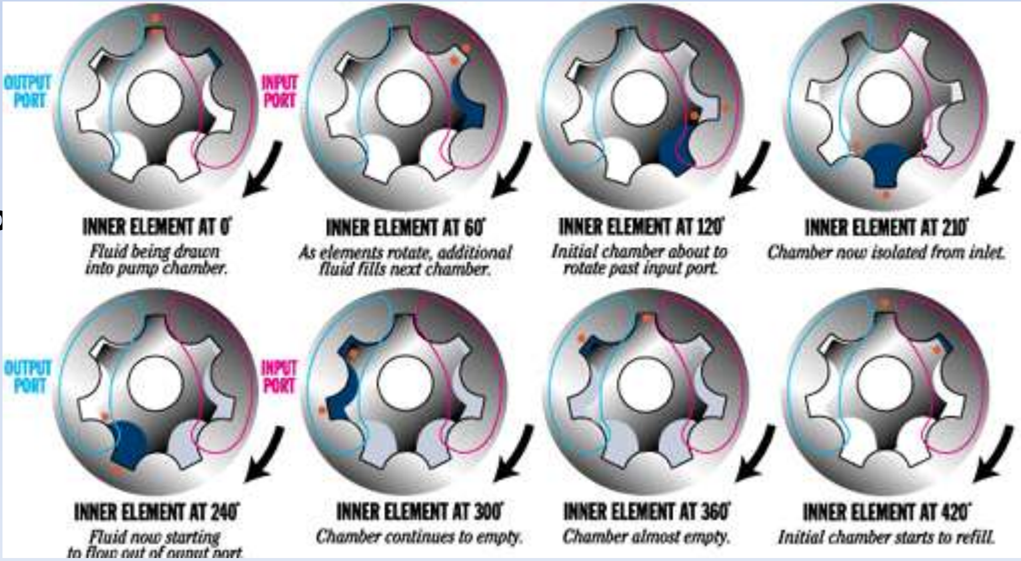






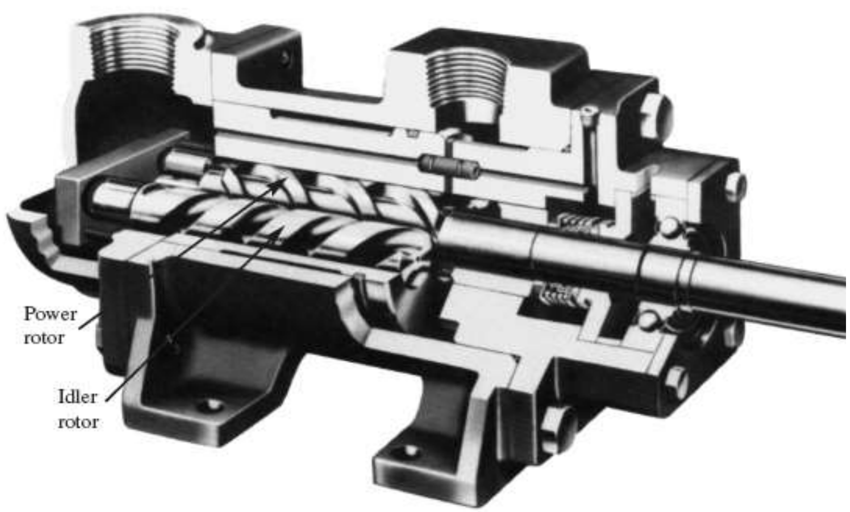
Gerotor pumps

- The inner gear (gerotor) is power driven and draws the outer gear.
- Advantages
    - **High Speed**
    - **Only two moving parts**
    - **Only one stuffing box**
    - **Constant and even discharge regardless of pressure conditions**
    - **Operates well in either direction**
    - **Quiet operation**
    - **Can be made to operate with one direction of flow with either rotation**
- Disadvantages
    - **Medium pressure limitations**
    - **Fixed clearances**
    - **No solids allowed**
    - **One bearing runs in the product pumped**
    - **Overhung load on shaft bearing**

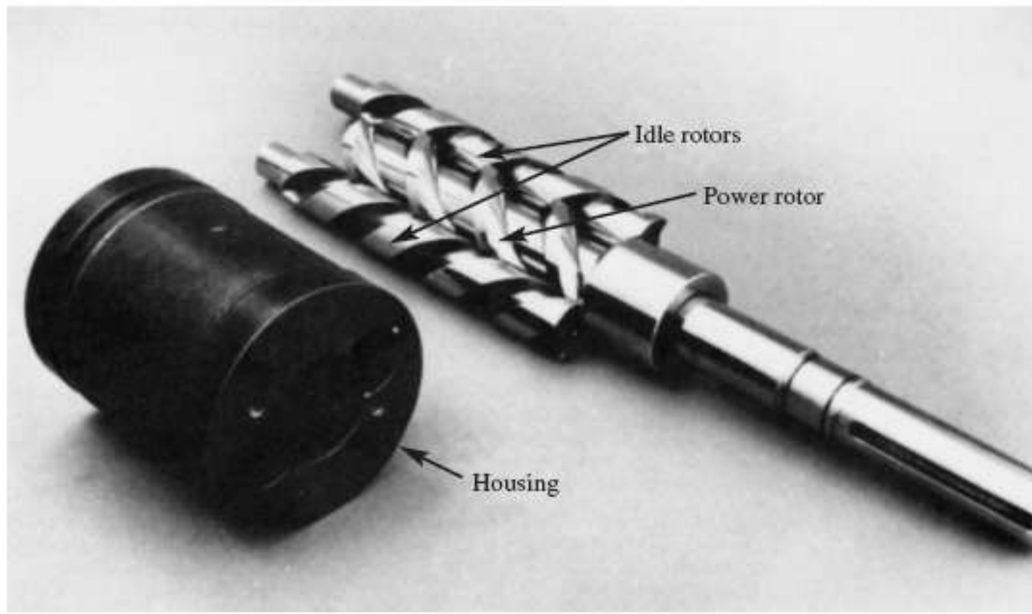


Screw Pump

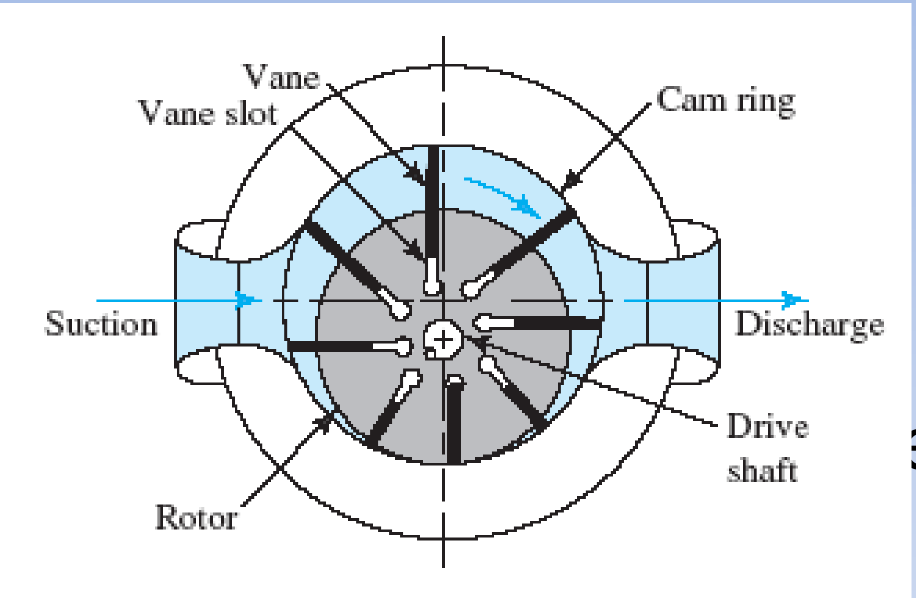
- Deliver nonpulsating flow quietly and efficiently.
- high pressure design up to 3500 psi with output flow 88 gpm.



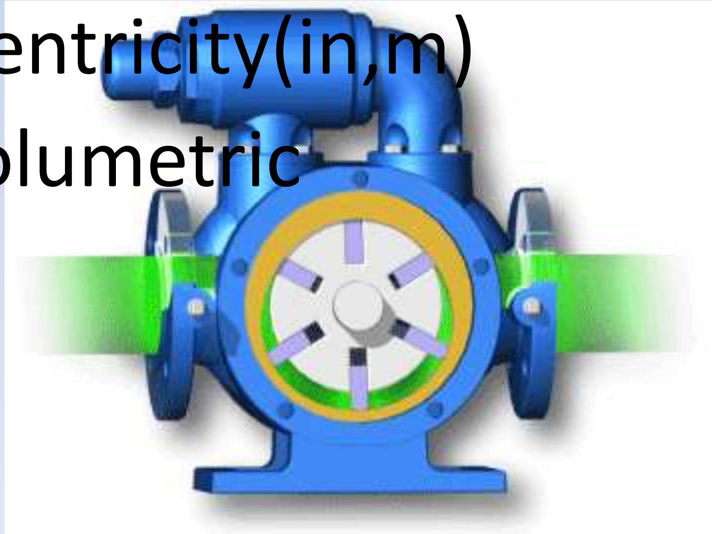
(a) Cutaway of pump assembly



Vane Pumps



- D_C = di
- D_R = di
- L = wic
- V_D = pu ent (in^3,m^3)
- e = ecc
- e_{max} = maximum possible eccentricity(in,m)
- V_{Dmax} = maximum possible volumetric displacement (in^3,m^3)



Vane Pumps

- Maximum possible eccentricity

$$e_{\max} = \frac{D_C - D_R}{2}$$

- Maximum eccentricity produce maximum volumetric displacement

$$V_{D\max} = \frac{\pi}{4}\left(D_C^2 - D_R^2\right)L = \frac{\pi}{4}\left(D_C - D_R\right)\left(D_C + D_R\right)L$$

$$V_{D\max} = \frac{\pi}{4}\left(D_C + D_R\right)\left(2e_{\max}\right)L$$

- The actual volumetric displacement occurs at $e_{max} = e$

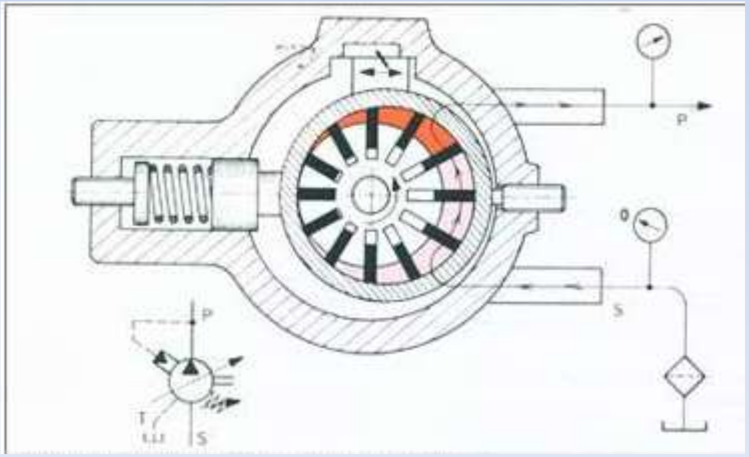
$$V_D = \frac{\pi}{2}\left(D_C + D_R\right)(e)L$$

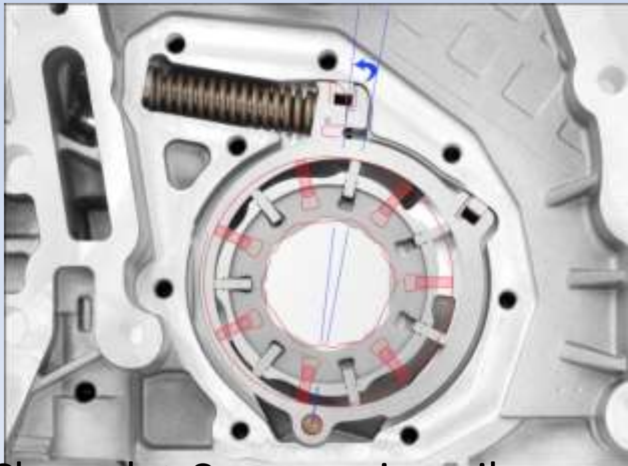
Variable Displacement Vane Pump

- Eccentricity varying mechanically lead to a variable displacement pump
- If pressure is used to move

  the cam ring leading to a

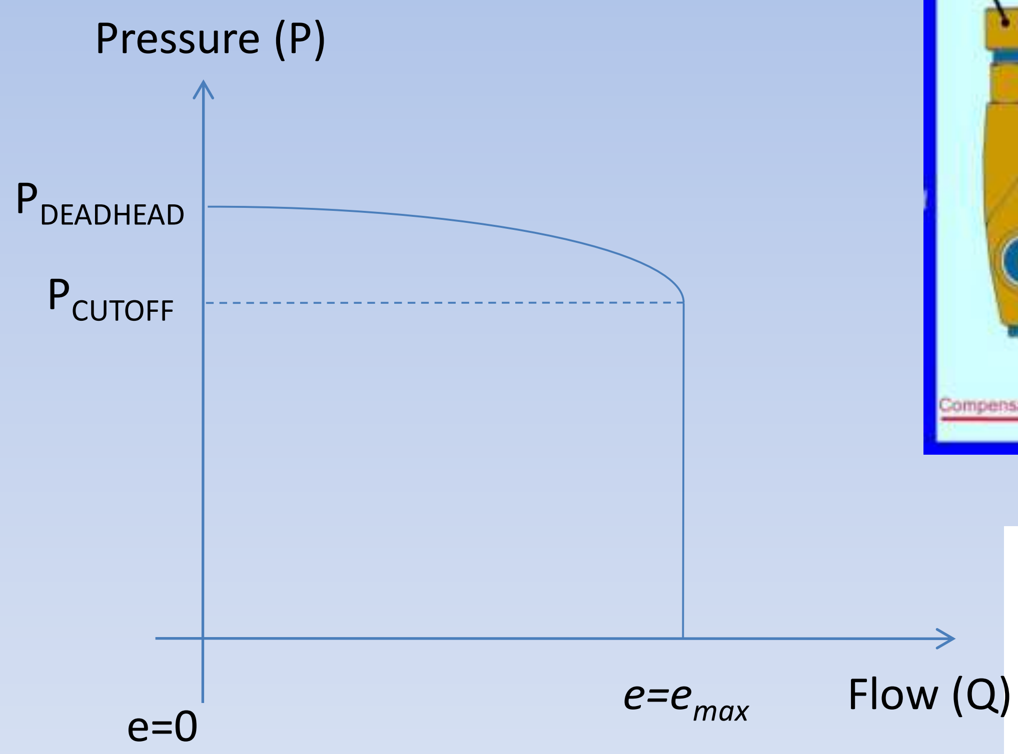
  pressure compensated pump

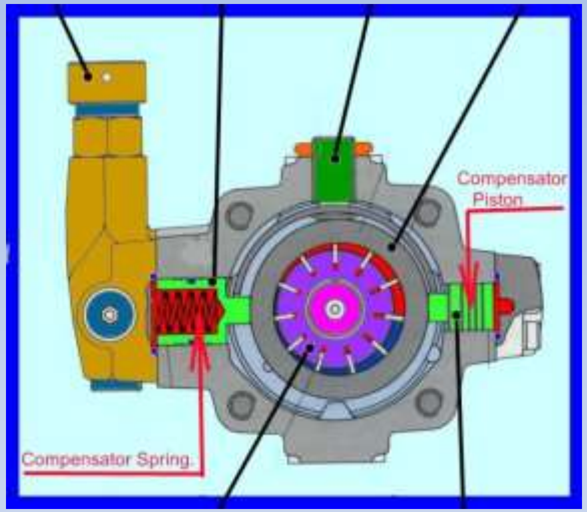




2011 Chevrolet Cruze engine oil pump

Pressure Compensated Vane Pump





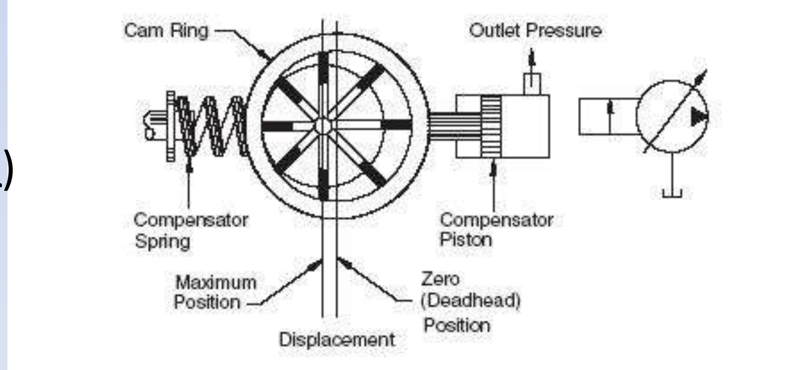


FIGURE 4.9
Pressure-compensated vane pump.

Balanced Vane Pumps

- Two inlets and two outlets
- Complete hydraulic balance
- Can not be designed for variable displacement
- Elliptical cam ring
- Can achieve higher operating pressures

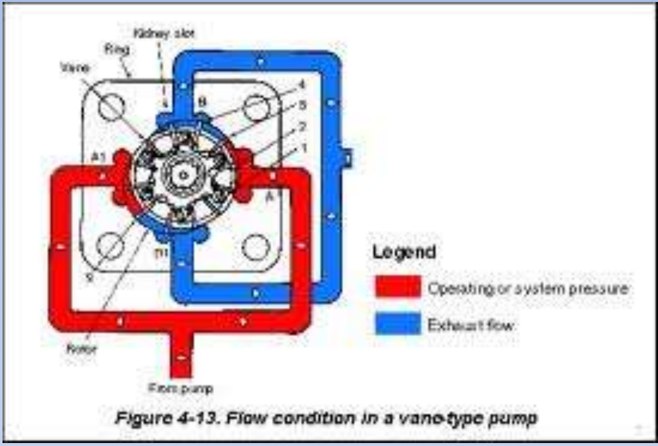


Figure 4-13. Flow condition in a vane-type pump

Piston Pumps

- Two basic types
  - Axial piston pumps: piston that are parallel to the axis of the cylinder block.
    - Bent axis configuration
    - Swash plate design
  - Radial piston pump: pistons arranged radially in the cylinder block.

Axial Piston Pump

- Bent axis design
  - Cylinder block rotate with shaft.
  - The center line of the cylinder block is shifted relative to the shaft.
  - Number of pistons arranged along a circle.
  - The piston rods are connected to the drive shaft flange by ball and socket joints.
  - A universal link connects the block to the drive shaft
  - Volumetric displacement varies with offset angle θ from 0° , no flow, to 30°, maximum flow.

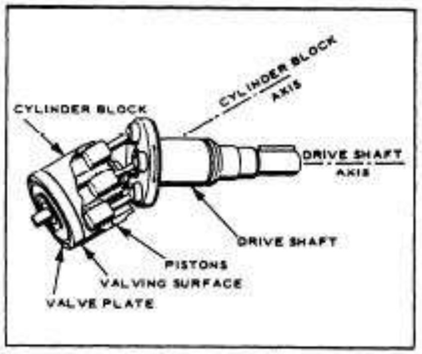


Figure 4-14.—Bent-axis axial piston pump.

Volumetric Displacement

- $\theta =$ offset angle(°)
- S = piston stroke (in,m)
- D = piston circle diameter (in,m)
- Y = number of pistons
- A = piston area (in^2,m^2)
- N = pump speed (rpm)
- Q_T = theoretical flow rate (gpm,m^3/min)

Cont

$$\tan(\theta) = \frac{S}{D} \quad \text{or} \quad S = D\tan(\theta)$$

The total displacement volume

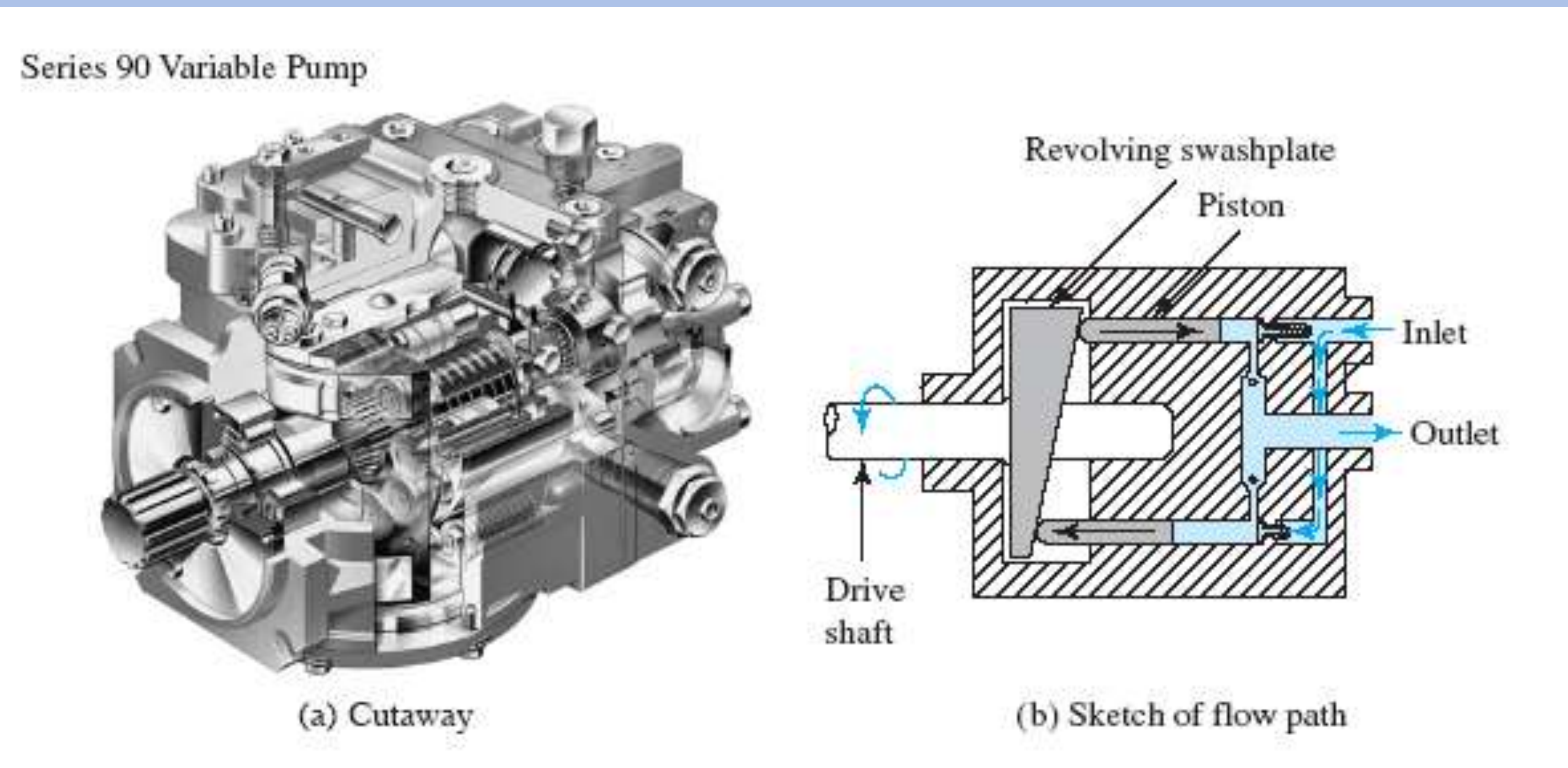
$$V_D = YAS = YAD\tan(\theta)$$

Theoretical flow rate

$$Q_T(gpm) = \frac{DANY\tan(\theta)}{231}$$

In-Line Piston Pump(Swash Plate)

at



(a) Cutaway

(b) Sketch of flow path

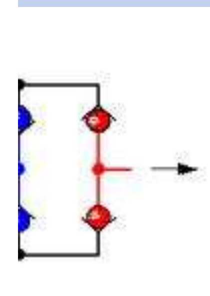
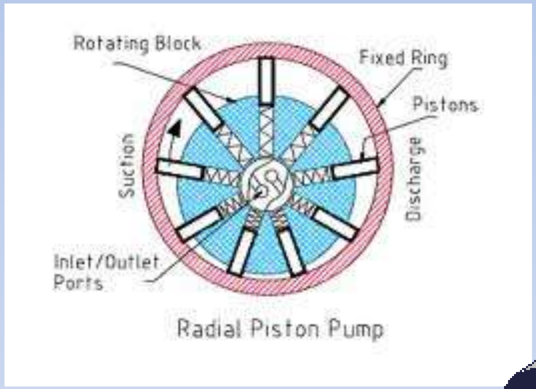
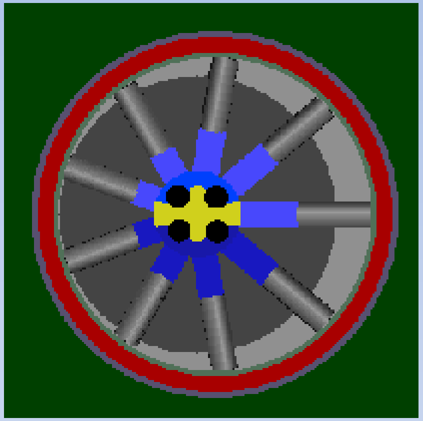
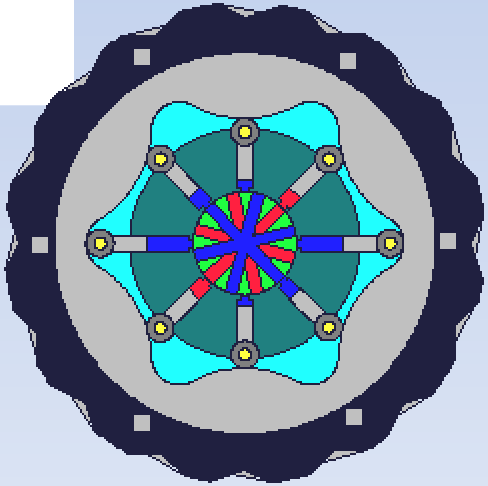


plate angle

Radial piston pump







Pump Performance

- Pump performance is a function of it the precision of its manufacture.
- An ideal pump would have no tolerance, and no friction, which is not feasible.
- Pump efficiency can be measure by three main indices.
  - Volumetric efficiency
  - Mechanical efficiency
  - Overall efficiency

Volumetric Efficiency

- Volumetric efficiency (η_v) indicates the amount of leakage that takes place within the pump

$$\eta_V = \frac{\text{Actual flow rate produced by pump}}{\text{Theoretical flow rate pump should produce}} = \frac{Q_A}{Q_T}$$

- Gear pumps η_v typically run from 80% to 90%
- Vane pumps η_v typically run from 82% to 92%
- Piston pumps η_v typically run from 90% to 98%

Mechanical Efficiency

- Mechanical efficiency (η_m) indicate the amount of energy losses for reasons other than leakage, such as friction in bearing and mating parts, losses due to fluid turbulence.
- Mechanical efficiency typically run from 90% to 95% for most pumps

$$\eta_m = \frac{\text{pump putput power assuming no leakage}}{\text{actual power delivered to pump}}$$

Or

$$\eta_m = \frac{\text{theoretical torque required to operate the pump}}{\text{actual torque delivered to pump}} = \frac{T_T}{T_A}$$

Using english units and horsepower

$$\eta_m = \frac{PQ_T/1714}{T_A N/63{,}000}$$

In metric units, using watts for power

$$\eta_m = \frac{PQ_T}{T_A N}$$

P = pump discharge pressure (psi,Pa)

Q_T=pump theoretical flow rate (gpm, m^3/s)

T_A=actual torque delivered to pump (in.lb, N.m)

N=pump speed (rpm, rad/s)

Overall Efficiency

- Overall efficiency (η_o) the overall efficiency considering all the losses

$$\eta_o = \frac{\text{actual power delivered by pump}}{\text{actual power delivered to pump}}$$

$$\eta_o = \eta_V \times \eta_m$$

for english units, horsepower

$$\eta_o = \frac{PQ_A/1714}{T_A N / 63{,}000}$$

for metric units, watts

$$\eta_o = \frac{PQ_A}{T_A N}$$

Example

- A pump has a displacement volume of 5 in^3 It delivers 20gpm at 1000rpm and 1000psi. If the prime mover input torque is 900 in.lb.
  a. What is the over all efficiency of the pump?
  b. What is the theoretical torque required to operate the pump?

  Sol.

$$a.\ \eta_V = \frac{Q_A}{Q_T},\ \text{where } Q_A = 20gpm \quad \text{and} \quad Q_T = V_D \times N = \frac{5(in^3)\times 1000(rpm)}{231} = 21.6$$

$$\eta_V = \frac{20}{21.6} = 0.926 = 92.6\%$$

$$\eta_m = \frac{PQ_T/1714}{T_A N/63{,}000} = \frac{(1000)(21.6)/1714}{(900)(1000)/63{,}000} = 0.881 = 88.1\%$$

$$\eta_o = \eta_V \times \eta_m = 0.926 \times 0.881 = 0.816 = 81.6\%$$

$$b.\ \eta_m = \frac{\mathrm{T_T}}{\mathrm{T_A}} \Rightarrow \eta_m \times \mathrm{T_A} = 0.881 \times 900 = 793\,lb.in$$

Pump Selection

1. Select the actuator that is appropriate based on the loads encountered.
2. Determine the flow rate requirement.
3. Select the system pressure.
4. Determine the pump speed and select the prime mover.
5. Select the pump type based on application.
6. Select the reservoir and the associated pluming.
7. Consider factor such as noise level, horsepower loss, heat exchanger.
8. Calculate the overall cost of the system.

Typical noise levels for hydraulic pumps

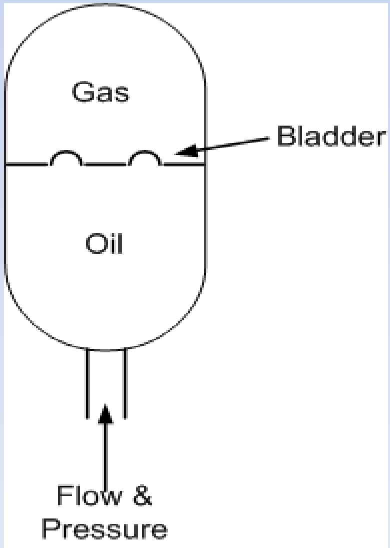
- Typical noise levels for hydraulic pumps are as follows:

Pump Type	**db**
Vane (industrial)	75-82
Axial piston	76-85
Gear (powdered metal)	78-88
Vane (mobile)	84-92
Gear (machined stock)	96-104

Accumulator Analysis

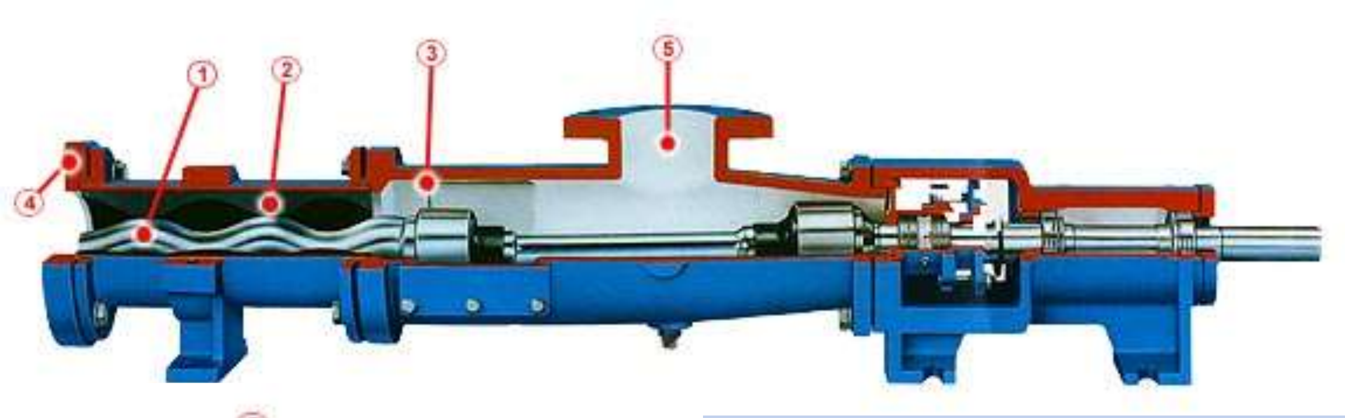
- Accumulator – energy storage devices, energy stored by compressing springs or gases.
  - Temporary demands
  - Average out high frequency demands
  - Damp pump ripples

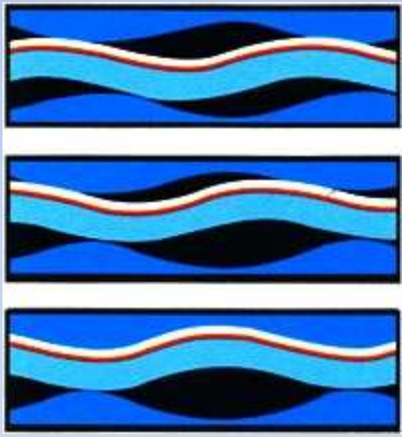
Gas – Oil Accumulators

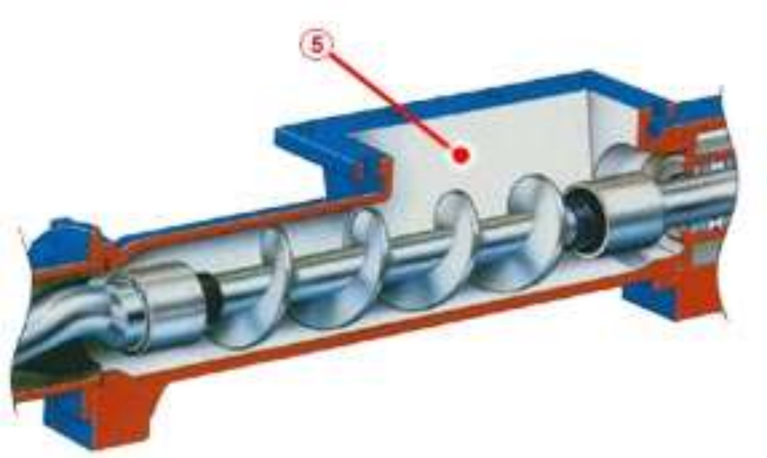


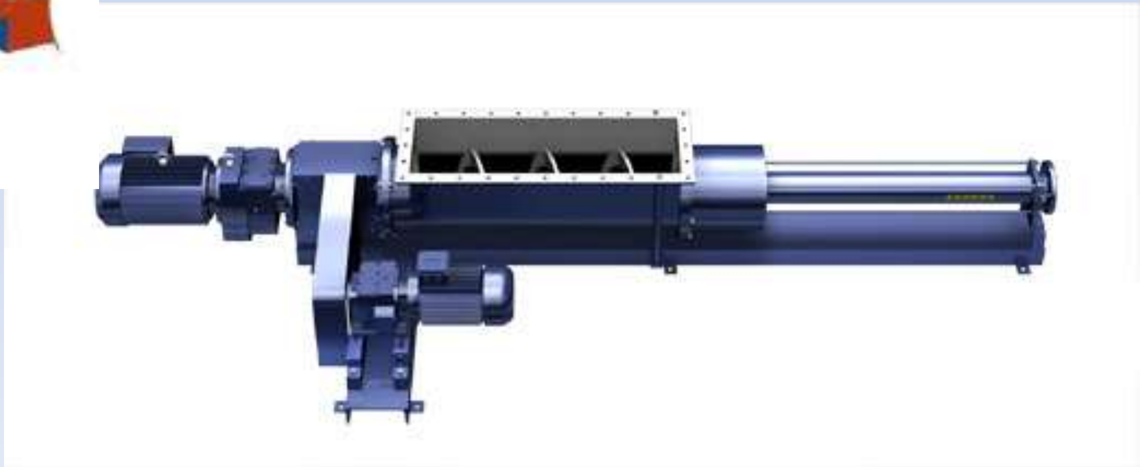
Thanks

Positive Displacement Pump: Progressing Cavity









Positive Displacement Pump: Progressing Cavity

- Advantages
  - Self-Priming/Suction Lift
  - Ability to Vary Capacity
  - Non-Pulsating Flow
  - Generates high pressure due to staging
  - Solids Laden Fluids
  - Abrasive Fluids
  - Handles High Viscosity Applications
  - Handles Shear Sensitive Fluids
  - Runs in Either Direction
  - Accurate Repeatable Flow
  - Open Throat (fluid come into rectangular box)

- Disadvantages
  - Can't Run Dry
  - Length of Pump

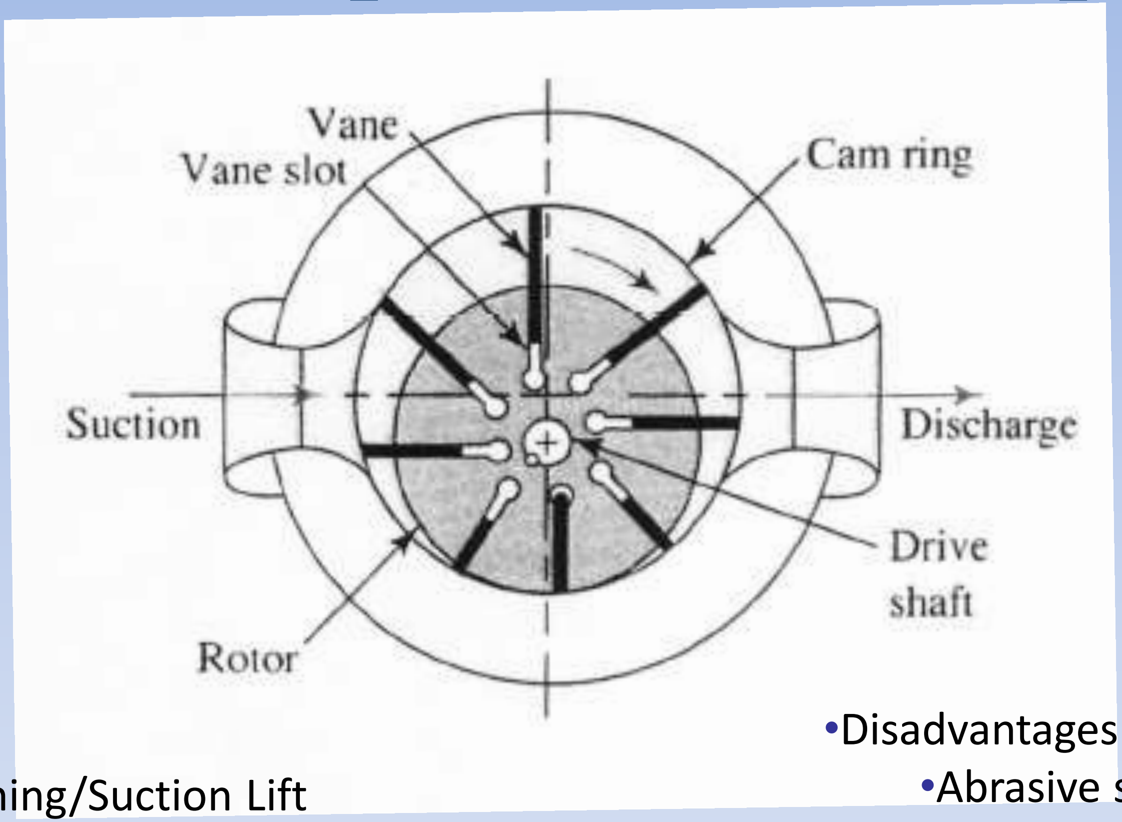
Sizing considerations:

-Solid size smaller than opening and cavity size

-Speed (don't want to overrun material that causes wear and slip)

-Pressure (need to provide enough stages to generate desired pressure)

Positive Displacement Pump: Vane



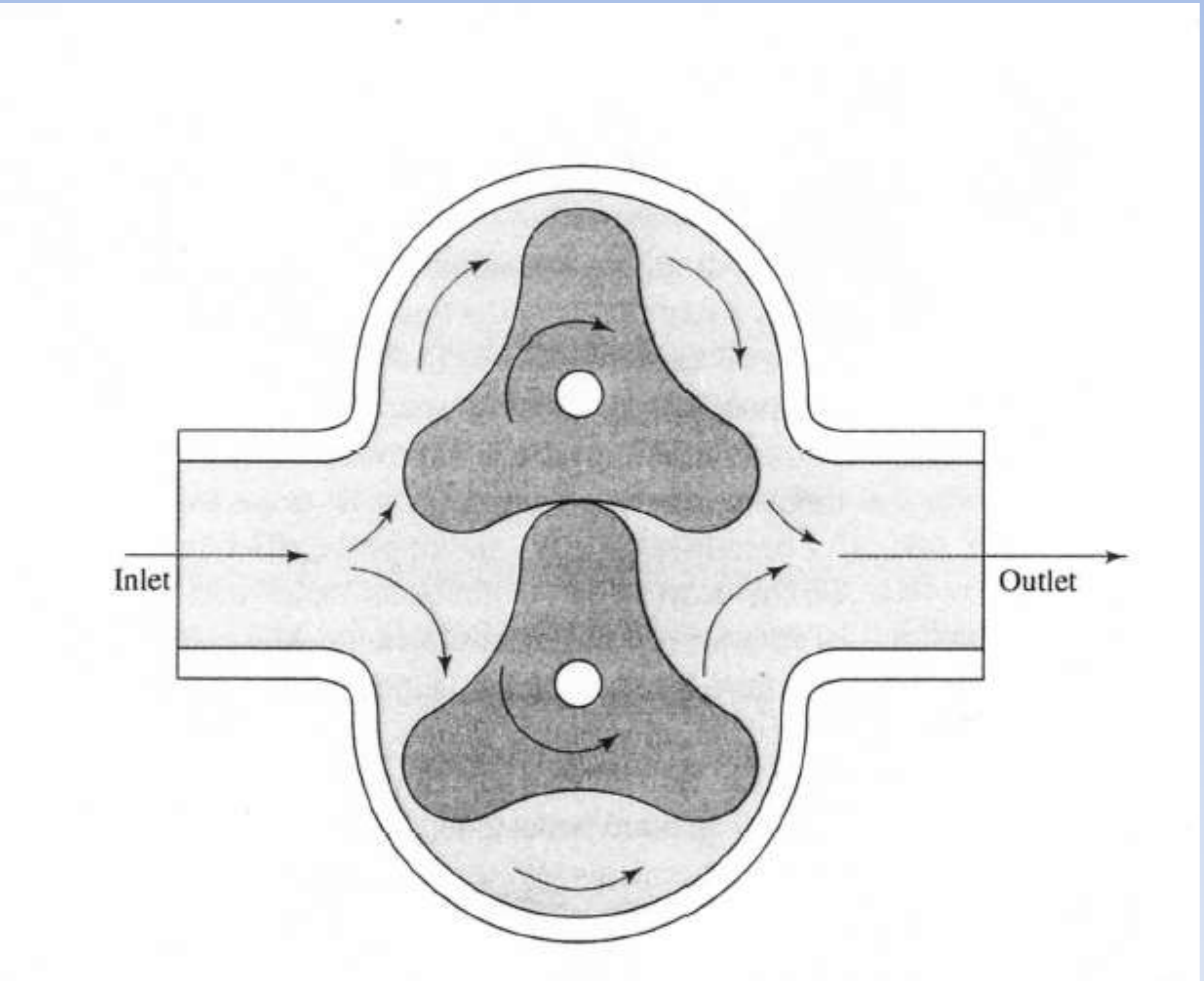
- Advantages:
  - Self-Priming/Suction Lift
  - Ability to Vary Capacity
  - Run Dry Short Time
  - Handles Low Viscosities
  - Handles Abrasive Fluids
  - Handles Shear Sensitive Fluids
  - Self Adjusting Vanes - For Performance

- Disadvantages:
  - Abrasive solids
  - Pressure Capabilities
  - Material Limitations

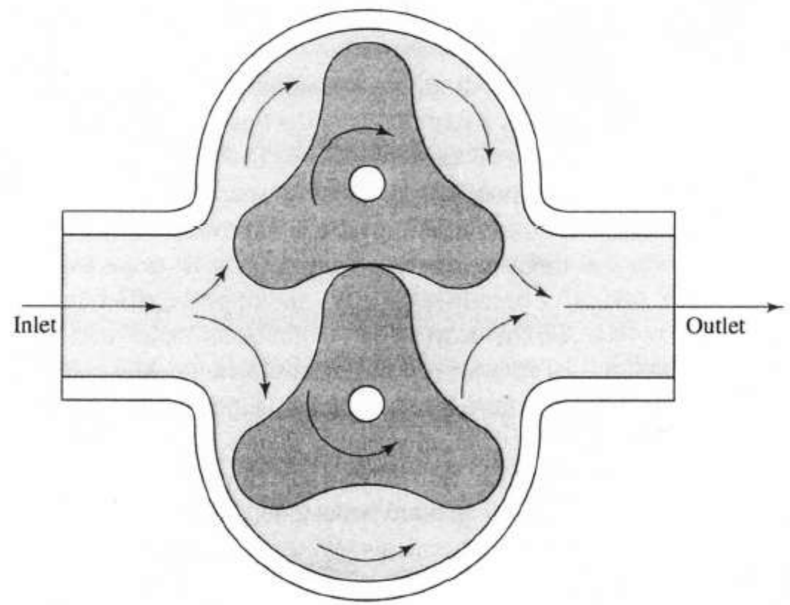
Practical considerations:

-Need to know pump's optimal viscosity range

Positive Displacement Pump: Lobe or Cam



Positive Displacement Pump: Lobe or Cam

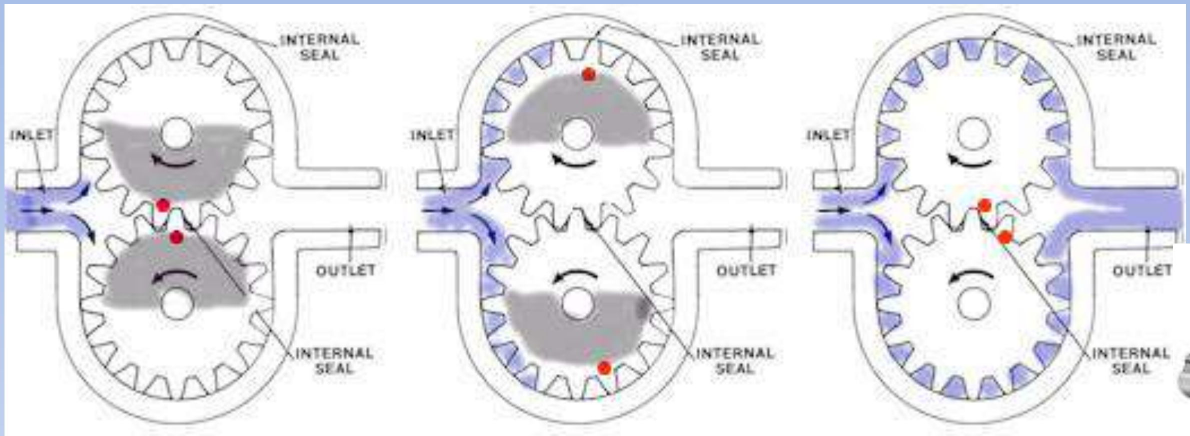


- Advantages:
  - Self-Priming/Suction Lift
  - Ability to Vary Capacity
  - Run in Either Direction
  - Handles High Viscosities
  - Accurate Repeatable Flow
  - Run Dry for a Short Time
  - Non contacting

- Disadvantages:
  - Abrasion Resistance
  - Pressure Capabilities
  - Stuffing Box
  - Jamming
  - Pump Efficiency
  - Non-Pulsating Flow

http://www.vikingpump.com/en/products/RotaryLobe/lobeAnimation.html

Positive Displacement Pump: Gear



http://www.vikingpump.com/en/products/ExternalGear/external.html

- Advantages
  - Self-Priming/Suction Lift
  - Ability to Vary Capacity
  - Non-Pulsating Flow
  - Run Dry for a Short Time
  - High Temperature

- Disadvantages
  - High pressure causes slip
  - Solids
  - Abrasion
  - High Shear
  - Low viscosity fluids (viscosity provides cushioning and lubrication)

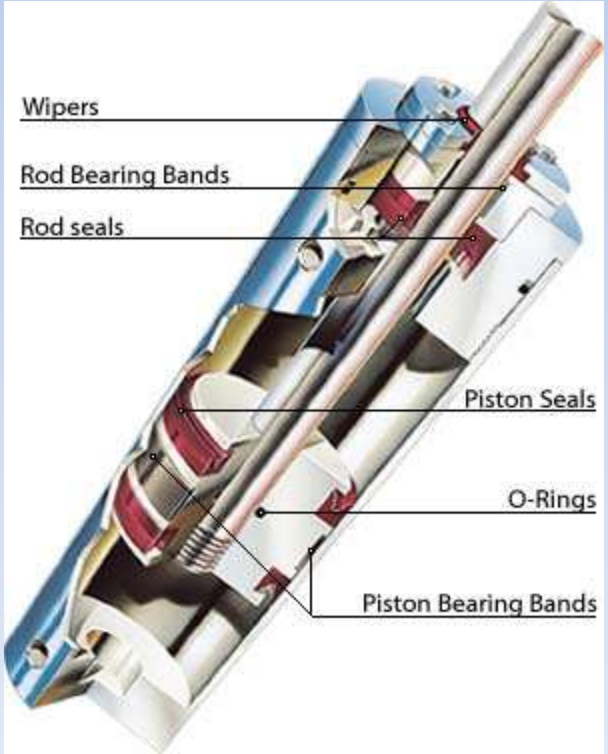
Hydraulic Cylinders and Cushioning Devices

Chapter 6

Dr. Suleiman BaniHani

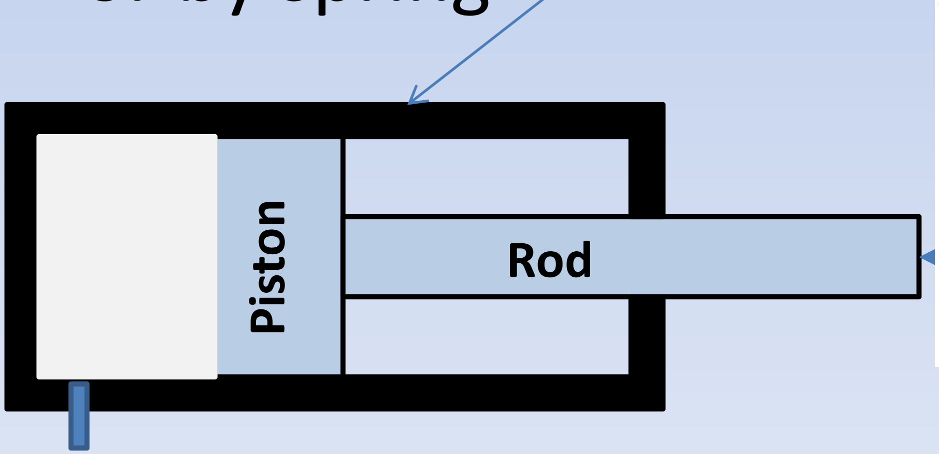
Hydraulic Cylinders

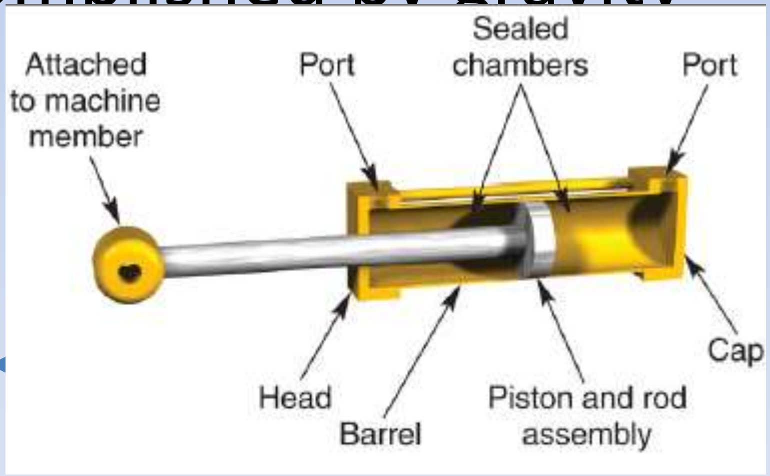
- Hydraulic cylinders or linear actuators extend and retract a piston rod to provide a push or pull force to drive the external load along a straight line path.



Operating Features

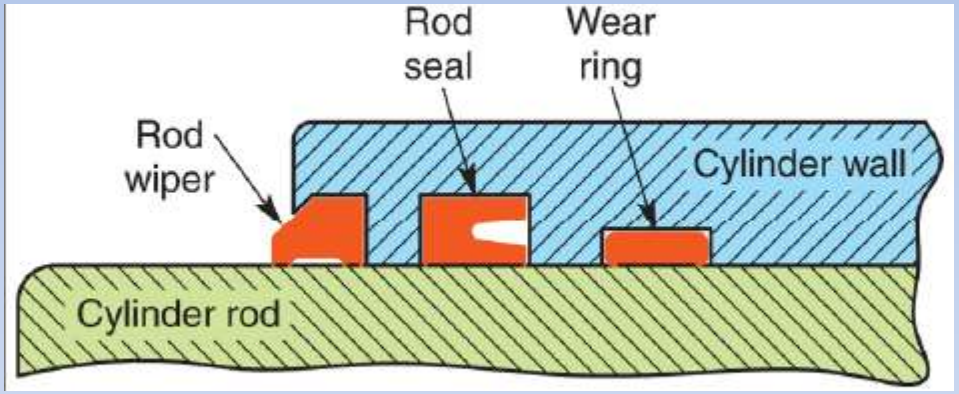
- The simplest type cylinder in the single acting cylinder
- Single acting cylinder can only exert force on extension, retract is accomplished by gravity or by spring

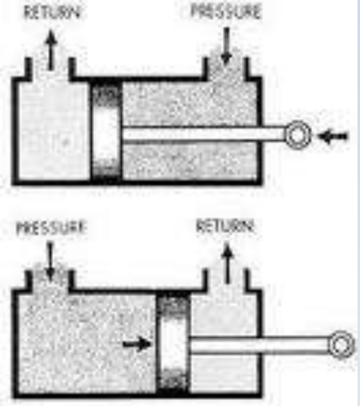


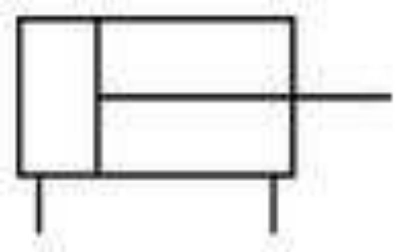


Double Acting Cylinder

- Exert force in both directions, has two input ports.







Cylinder Mountings

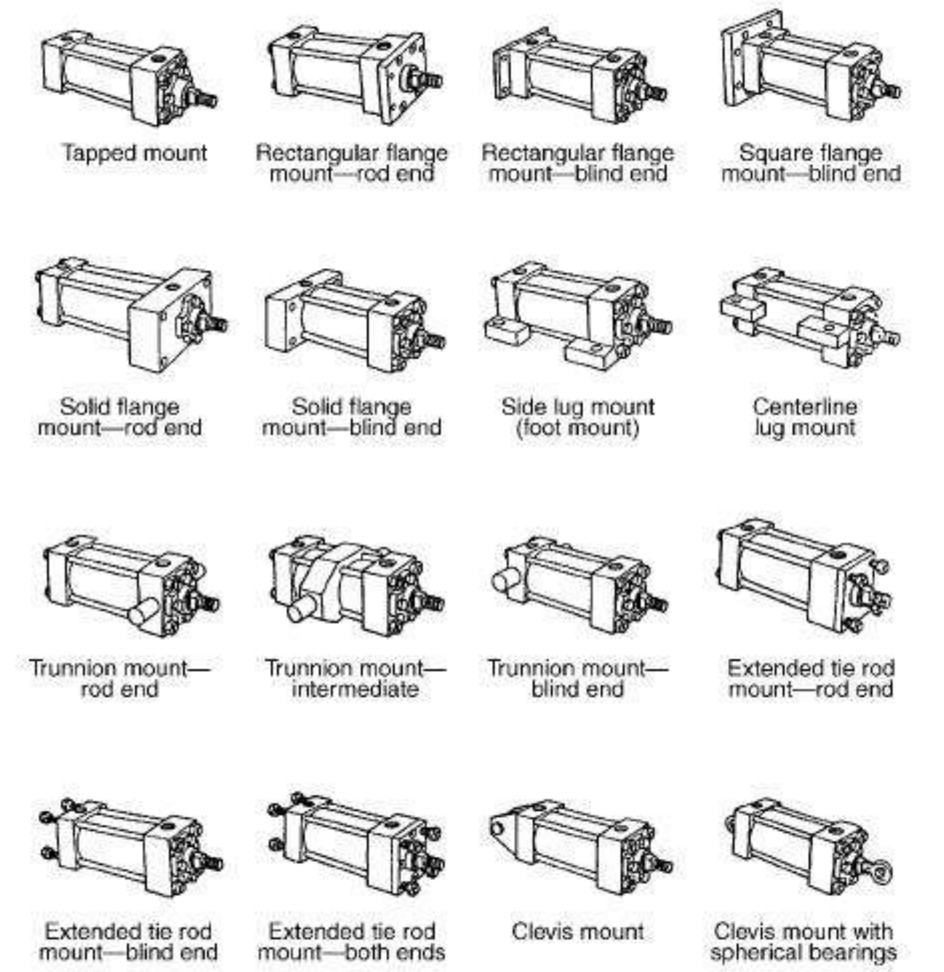
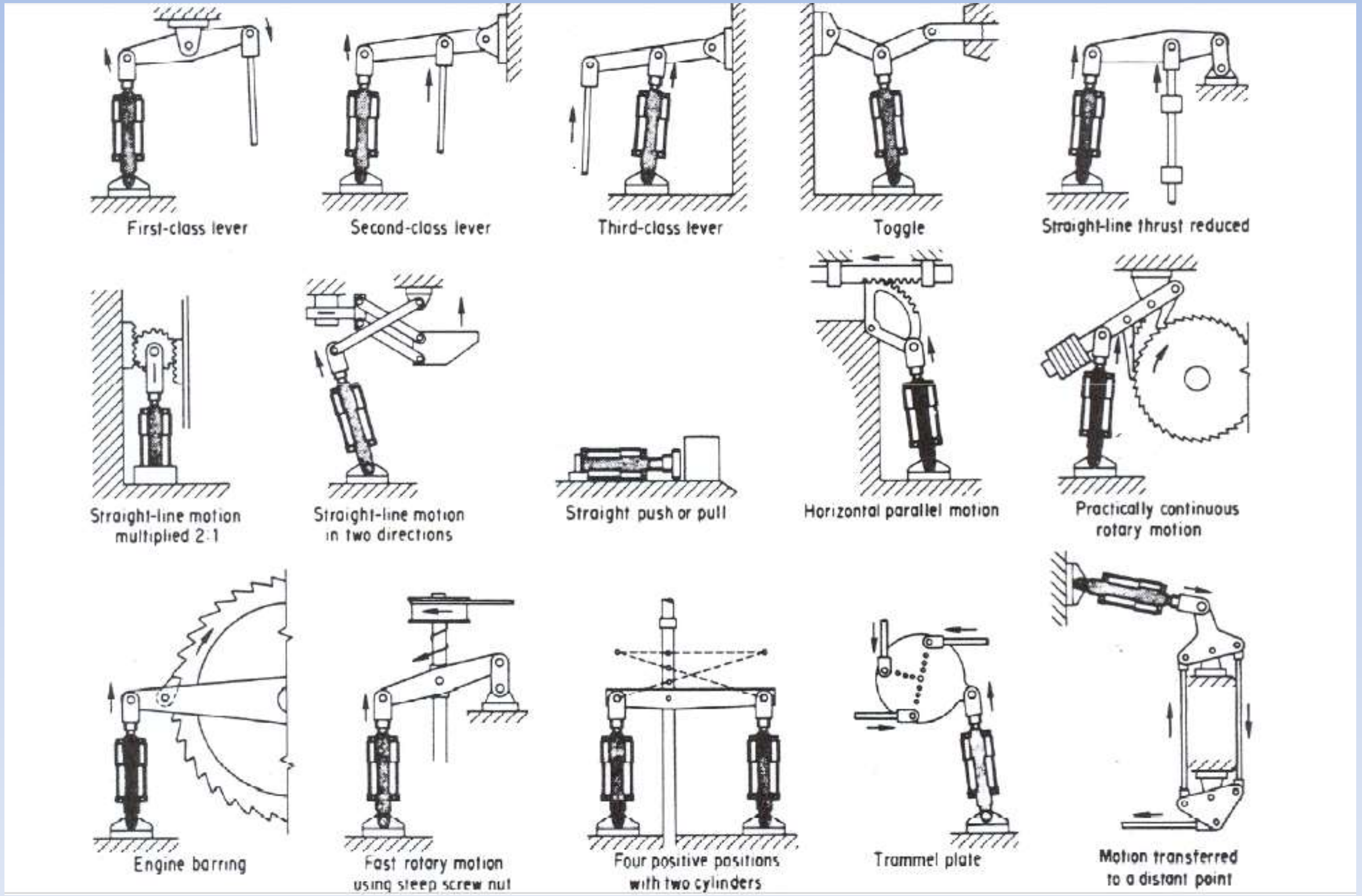


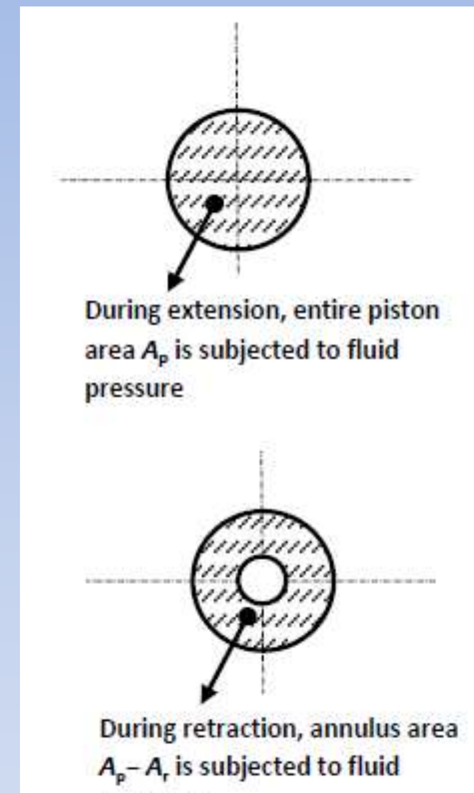
FIGURE 7.15
Partial listing of cylinder mounting methods.

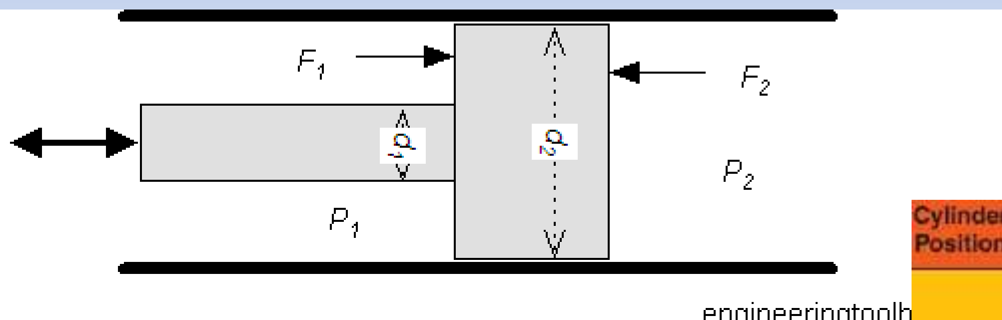
Cylinder Mechanical Linkages

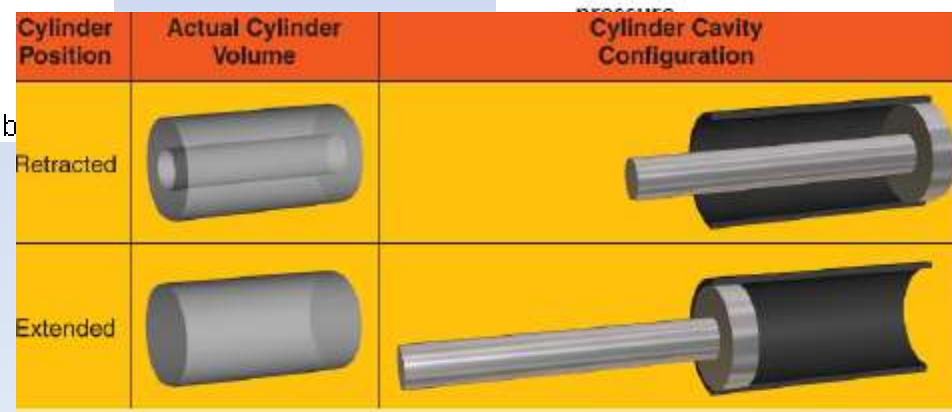


Cylinder Force, Velocity, and Power

- For double acting cylinder the forces (*F*) and velocity (*v*) are not the same in extension and retraction.
- In extension stroke fluid enters the black end of the cylinder through the entire piston area (A_P) while during retraction it only has (A_P -A_r)







Cont.

- For extension

$$F_{ext}(lb) = p(psi) \times A_p(in^2) \quad \text{or } F_{ext}(N) = p(Pa) \times A_p(m^2)$$

$$v_{ext}(ft/s) = \frac{Q_{in}(ft^3/s)}{A_p(ft^2)} \quad \text{or} \quad v_{ext}(m/s) = \frac{Q_{in}(m^3/s)}{A_p(m^2)}$$

For retraction

$$F_{ret}(lb) = p(psi) \times (A_p - A_r)(in^2) \quad \text{or } F_{ext}(N) = p(Pa) \times (A_p - A_r)(m^2)$$

$$v_{ext}(ft/s) = \frac{Q_{in}(ft^3/s)}{(A_p - A_r)(ft^2)} \quad \text{or} \quad v_{ext}(m/s) = \frac{Q_{in}(m^3/s)}{(A_p - A_r)(m^2)}$$

$$Power(HP) = \frac{v_p(ft/s) \times F(lb)}{550} = \frac{Q_{in}(gpm) \times P(psi)}{1714}$$

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

Example

- A pump supplies oil at 20 gpm to a 2 in diameter double acting cylinder. If the load is 1000 lb (extension and retraction) and the rod diameter is 1 in. Find
  a. Hydraulic pressure in extension

$$P_{ext} = \frac{F_{ext}(lb)}{A_P(in^2)} = \frac{1000}{\frac{\pi}{4}2^2} = 318\,psi$$

  b. The piston extension velocity

$$v_{ext} = \frac{Q_{in}(ft^3/s)}{A_P(ft^2)} = \frac{20/449}{3.14/144} = 2.05\,ft/s$$

  c. Cylinder HP during extension

$$HP_{ext} = \frac{v_{ext}(ft/s)\times F_{ext}(lb)}{550} = \frac{2.05\times 1000}{550} = 3.72hp$$

  d. Hydraulic pressure during retraction

$$P_{ret} = \frac{F_{ret}(lb)}{(A_P - A_r)(in^2)} = \frac{1000}{\frac{\pi}{4}\left(2^2 - 1^2\right)} = 425\,psi$$

  e. The piston retraction velocity

$$v_{ret} = \frac{Q_{in}(ft^3/s)}{(A_P - A_r)(ft^2)} = \frac{20/449}{2.355/144} = 2.73\,ft/s$$

  f. Cylinder HP during retraction

$$HP_{ret} = \frac{v_{ret}(ft/s)\times F_{ret}(lb)}{550} = \frac{2.73\times 1000}{550} = 4.96hp$$

Cylinder Loads Due to Moving Weights

- Cylinder has to overcome the weight of the body it is lifting.
- If the body is moved vertically at a constant speed, then the cylinder must exert a force equal to the bodies weight.
- If the cylinder is used to push or pull a body horizontally, it has to overcome the friction force between the body and the surface.
- If a cylinder is to move a body along an inclined surface, it has to over come the friction force and the body weight component in the direction of the cylinder.
- If the motion of the cylinder involve acceleration and deceleration , the inertia force has to be taken into account.

Examples

Find the cylinder force needed to move a 6000lb weight along a horizontal surface with friction coefficient (CF) between the weight and the surface = 0.14

Sol. $F=f=W\times CF=6000\times0.14=840$ lb

Find the force required to lift 6000lb weight along a 30° surace from the horizontal at constant velocity

Sol. $F= W \sin 30°=6000 \times \frac{1}{2} =3000$ lb

Find the force required to lift a 6000 weight vertically, with accelerate the body from rest to 8ft/s in 0.5 sec

Sol. $a= (8ft/s – 0ft/s)/0.5s=16ft/s^2$

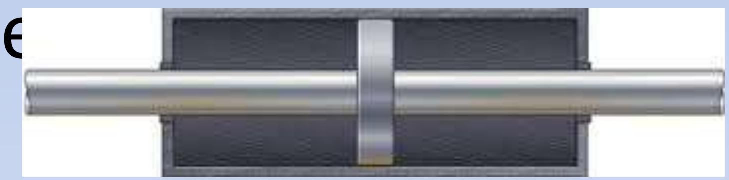
$\Sigma F=m \times a \Rightarrow F_{cyl}-W=m \times a \Rightarrow F_{cyl}=m \times a+W$

$F_{accel}=m \times a=W/g \times a= 6000/32.2 \times 16=2980lb$

$F_{cyl}=6000lb+ 2980lb=8980lb$

Special Cylinder Designs

- Double rod cylinder: The force and speed are the same for both extension and retraction.
- Typically used when the task is performed at either end of the



- Telescopic cylinder: Contains multiple cylinders that slide inside each other.
- Mainly used to minimize retraction length

Cylinder Loading Through Mechanical Linkage

First class liver:

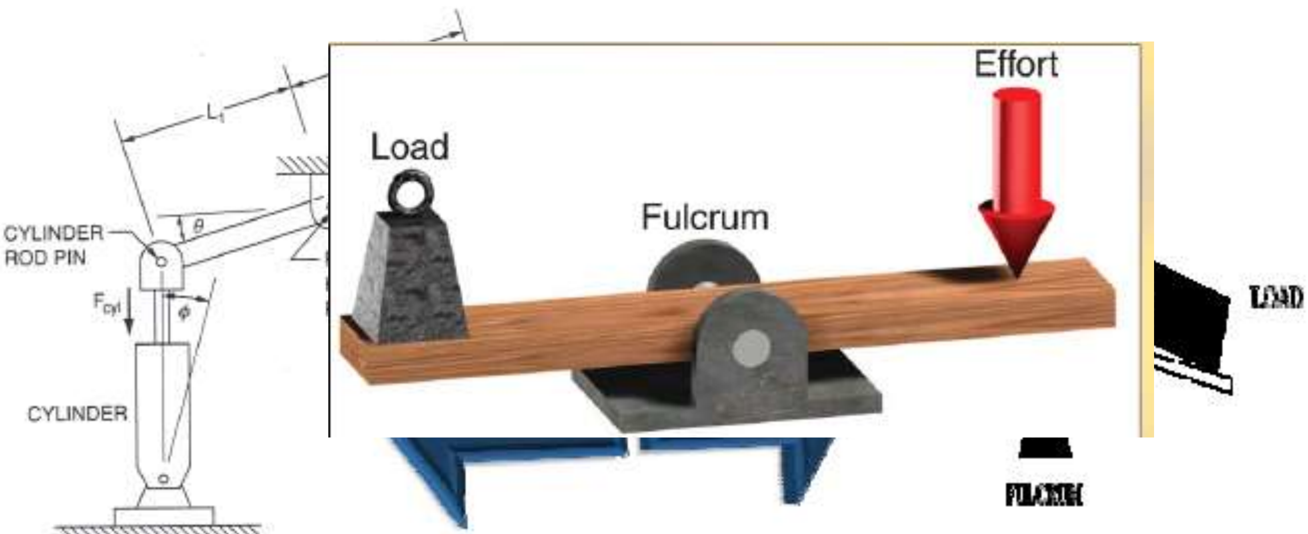


Figure 7-14. Use of a first-class lever to drive a load.

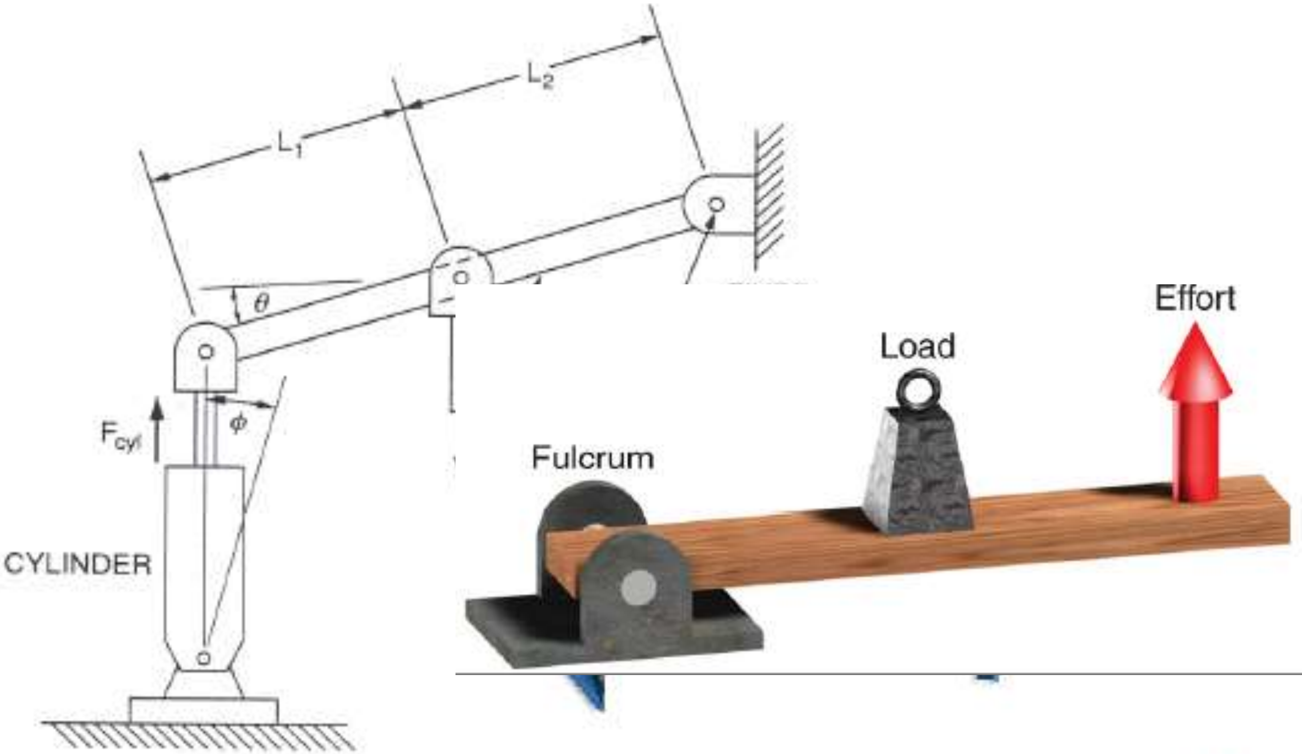
$$F_{cyl} \times L_1 = F_{ld} \times L_2$$

$$F_{cyl} = F_{ld} \frac{L_2}{L_1}$$

Or $$F_{Cyl} = \frac{L_2}{L_1 \cos\phi} F_{Load}$$

The fulcrum lies between the effort and the load. The point on which the bar is resting is known as the fulcrum. The bar is the lever. The muscular power or effort alone would not move the weight. The lever is a simple machine that concentrates the effort or force and makes it possible to perform the work.

Second class lever system:

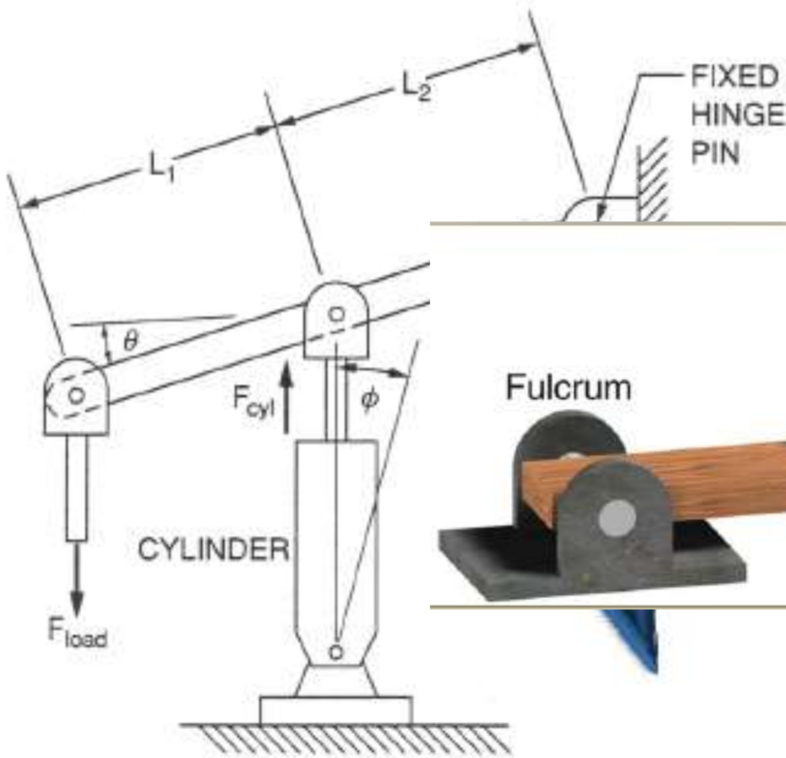


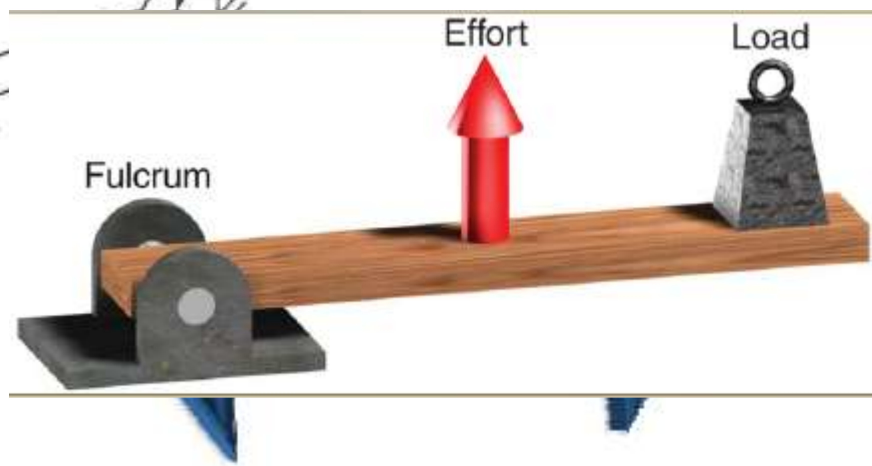
$$F_{cycl} \times (L_1 + L_2) Cos\,\varphi = F_{load} \times L_2$$

$$F_{Cycl} = \frac{F_{load} \times L_2}{(L_1 + L_2) Cos\,\varphi}$$

Imagine lifting a heavy log by means of a lever which has been pushed under the log. It should be noted that the fulcrum is the point where the lever touches the ground, and that the load is between the fulcrum and the point where the effort is exerted. The effort is exerted upwards.

Third Class Lever:





$$F_{Cycl} \times L_2 Cos\,\varphi = F_{load} \times (L_1 + L_2)$$

$$F_{Cycl} = \frac{F_{load} \times (L_1 + L_2)}{L_2 Cos\,\varphi}$$

The third class lever is the strangest ... and the one you use the most!

The fulcrum is once again at one end of the lever, but this time the ***load*** is at the other end, and you apply a force in between.

Example

For the first, second and third class lever system, L_1=L_2=10 in, ϕ=0°, and F_{load}=1000lb.

Find the cylinder fore required for the three cases

Sol.

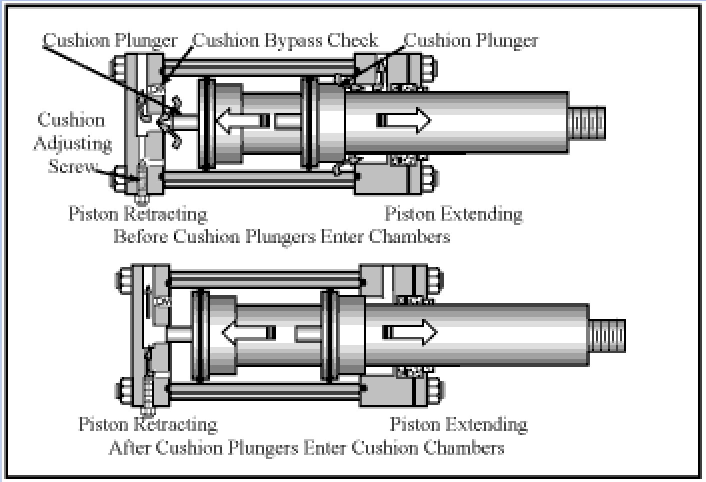
$$\text{Case 1}: F_{Cyl} = \frac{L_2}{L_1 \cos\phi} F_{Load} = \frac{10}{10 \times 1}(1000) = 1000lb$$

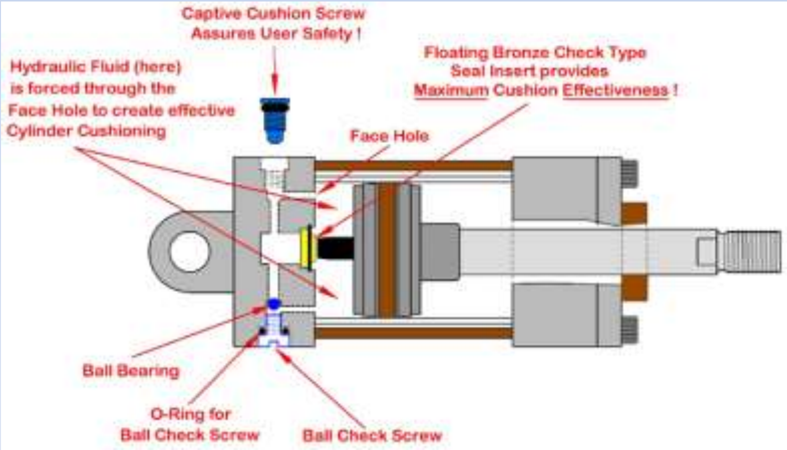
$$\text{Case 2}: F_{Cyl} = \frac{L_2}{(L_1 + L_2)\cos\phi} F_{Load} = \frac{10}{(10+10)\times 1}(1000) = 500lb$$

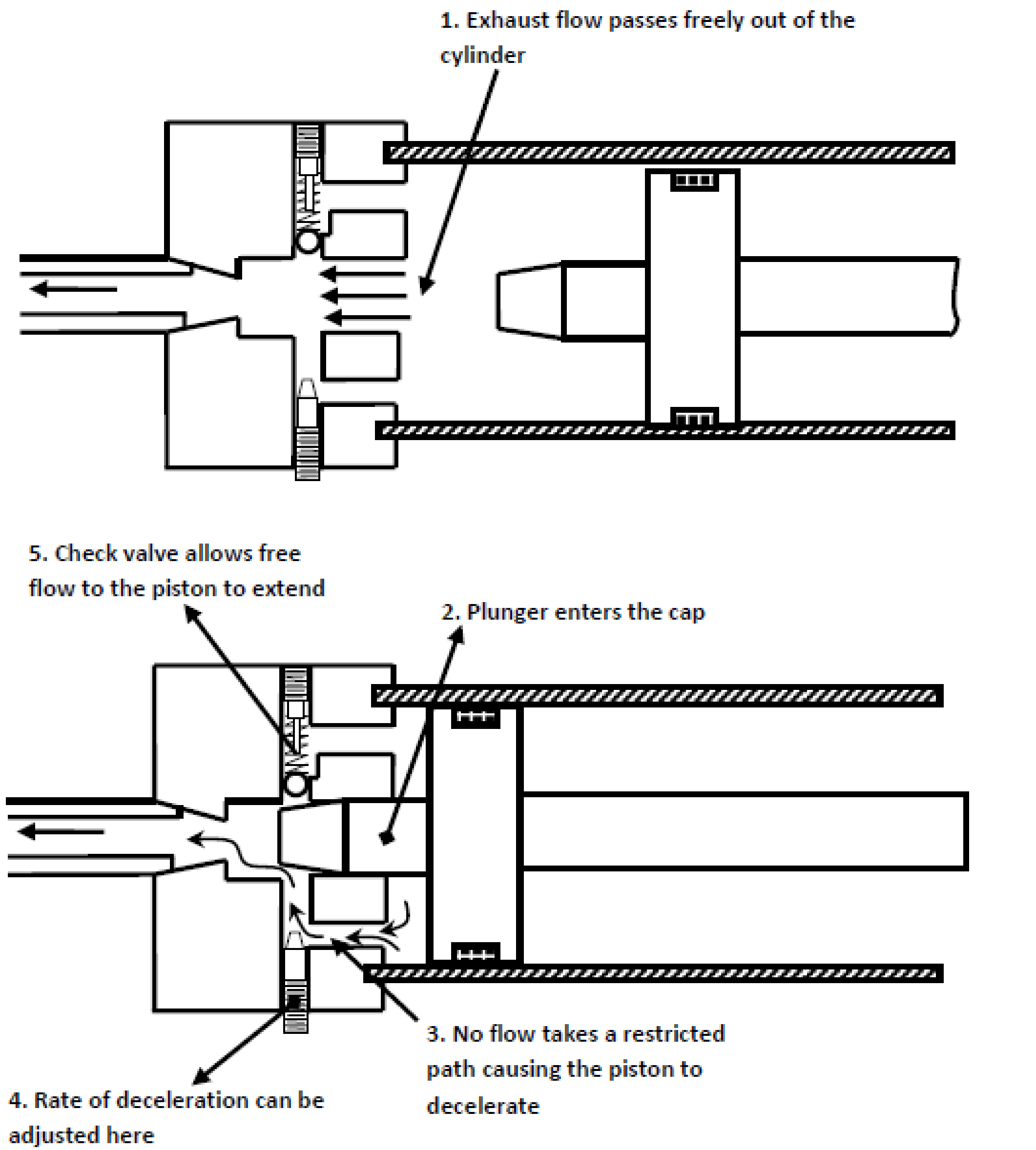
$$\text{Case 3}: F_{Cyl} = \frac{L_1 + L_2}{L_2 \cos\phi} F_{Load} = \frac{10+10}{10 \times 1}(1000) = 2000lb$$

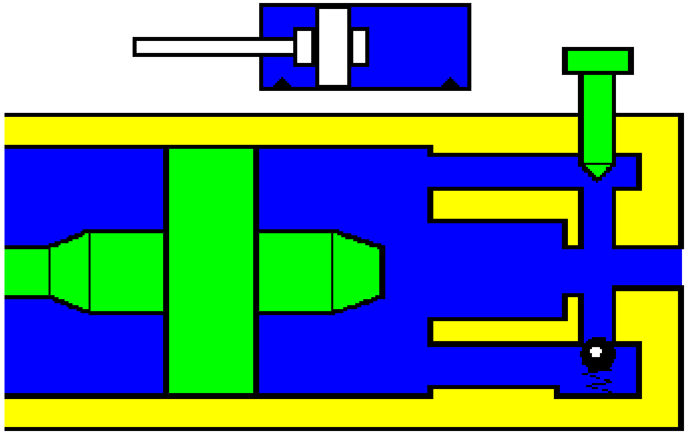
Hydraulic Cylinder Cushions

- Cylinder cushions at the end of the cylinder to slow the piston down near the ends of the stroke.
- At the end of the stroke the oil is force to exhaust from an adjustable opening.
- An imbedded check valve is used to allow free flow in reverse direction.






1. Exhaust flow passes freely out of the cylinder
5. Check valve allows free flow to the piston to extend
2. Plunger enters the cap
3. No flow takes a restricted path causing the piston to decelerate
4. Rate of deceleration can be adjusted here



Example

A pump delivers oil at a rate **18.2gpm** into the blank end of **a 3 in diameter** hydraulic cylinder. The piston contains **a 1 in diameter** cushion plunger that is **0.75 in long**, therefore the piston decelerated over a 0.75 in distance in the extension stroke. The piston drives a **1500lb weight** which slides on a horizontal surface with **friction coefficient (CF=0.12)**.The pressure relief valve of the system is set **to 750 psi**. find the maximum pressure developed by the cushion.

$$\text{Sol.}\quad \text{Piston velocity } v = \frac{Q_{pump}}{A_{piston}} = \frac{18.2/449\,(ft^3/s)}{\frac{\pi}{4}(3)^2 / 144\,(ft^2)} = 0.83\,ft/s$$

$$\text{acceleration } v_2^{\,2} - v_1^2 = 2aS \Rightarrow v_2 = 0, S = 0.75/12(ft) \Rightarrow a = -5.51\,ft/s^2$$

$$\sum F = ma \Rightarrow -p_2(A_{piston} - A_{plunger}) - (CF)W + p_1 A_{piston} = \frac{W}{g}a$$

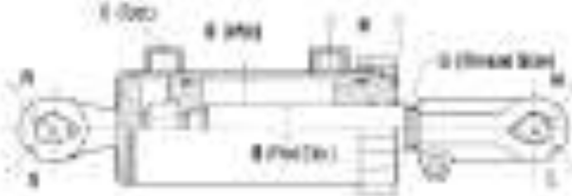
$$p_2 = 856\,psi$$

CHIEF WC 3000 PSI

Engineered and manufactured by Bailey.

Welded Cylinders

Intended Use: Double-acting applications • **Piston:** Ductile iron • **Gland:** Ductile iron • **Tube:** Bored and roller burnished, precision finished steel for extended seal life • **Rod:** Hard chrome plated • **Rod End Mount:** Ductile iron, threaded female clevis with pin and clips • **Base End Mount:** Forged steel female clevis with pin and clips • **Tube Seal:** Buna O-ring with Polythane hybrid backup • **Rod Seal:** Deep polypak with U cup design • **Rod Wiper:** Urethane, snap-in • **Piston Seal:** Hallite 758 provides prolonged wear resistance • **All Seals:** Manufactured to US specifications • **Paint:** Red. Custom colors are available. • **Packaging:** Individually poly bagged.



Dimensional Data in Inches (Millimeters)

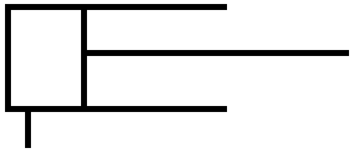
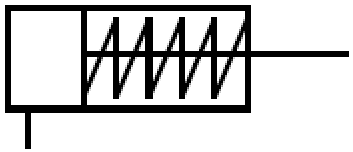
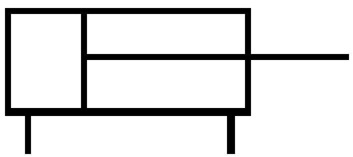
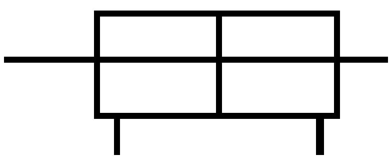
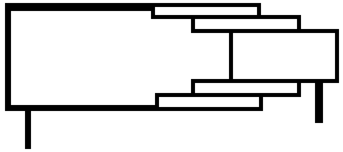
	DIMENSIONS																
BORE	A*	B	C	D	E	G	H	I	J	K	L	M	N	O	Q	R	T
2.000 (51)	10.250 (260)	0.188 (4.8)	3/4-16 SAE	1.125 (28.6)	2.625 (66.68)	2.000 (50.8)	1.124 (28.58)	1.125 (28.6)	2.125 (54.0)	1.812 (46.0)	1.000 (25.4)	1.062 (26.97)	2.125 (54.0)	1.000 (25.4)	1 1/16-12 UNF	1.011 (25.7)	3.125 (79.4)
2.500 (64)	10.250 (260)	0.188 (4.8)	3/4-16 SAE	1.250 (31.8)	2.625 (66.68)	2.500 (63.5)	1.124 (28.58)	1.125 (28.6)	2.125 (54.0)	1.812 (46.0)	1.000 (25.4)	1.125 (28.58)	2.125 (54.0)	1.000 (25.4)	1 1/16-12 UNF	1.011 (25.7)	3.625 (92.1)
3.000 (76)	10.250 (260)	0.188 (4.8)	3/4-16 SAE	1.500 (38.1)	2.625 (66.68)	2.625 (66.7)	1.124 (28.58)	1.125 (28.6)	2.125 (54.0)	1.940 (49.3)	1.125 (28.6)	1.328 (33.73)	1.875 (47.6)	1.000 (25.4)	1 1/16-12 UNF	1.011 (25.7)	4.134 (105.0)
3.500 (89)	10.250 (260)	0.188 (4.8)	3/4-16 SAE	1.750 (44.5)	2.625 (66.68)	2.625 (66.7)	1.124 (28.58)	1.125 (28.6)	2.125 (54.0)	2.000 (50.8)	1.125 (28.6)	1.328 (33.73)	1.875 (47.6)	1.000 (25.4)	1 1/16-12 UNF	1.011 (25.7)	4.488 (114.0)
4.000 (102)	10.250 (260)	0.25 (6.4)	3/4-16 SAE	2.000 (50.8)	2.625 (66.68)	2.625 (66.7)	1.124 (28.58)	1.125 (28.6)	2.125 (54.0)	2.062 (52.37)	1.125 (28.6)	1.500 (38.10)	1.750 (44.5)	1.000 (25.4)	1 1/16-12 UNF	1.011 (25.7)	5.248 (133.3)

*Retracted length is 10.25" (260.4) for 8" (203.2) stroke cylinders.

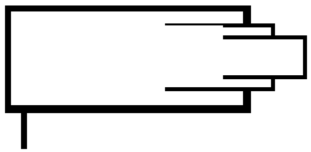
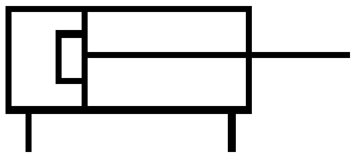
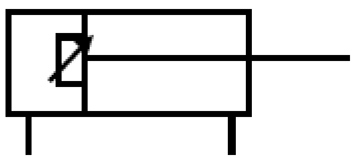
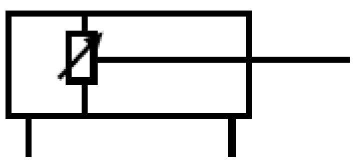
Recommended cylinder bore and rod sizes

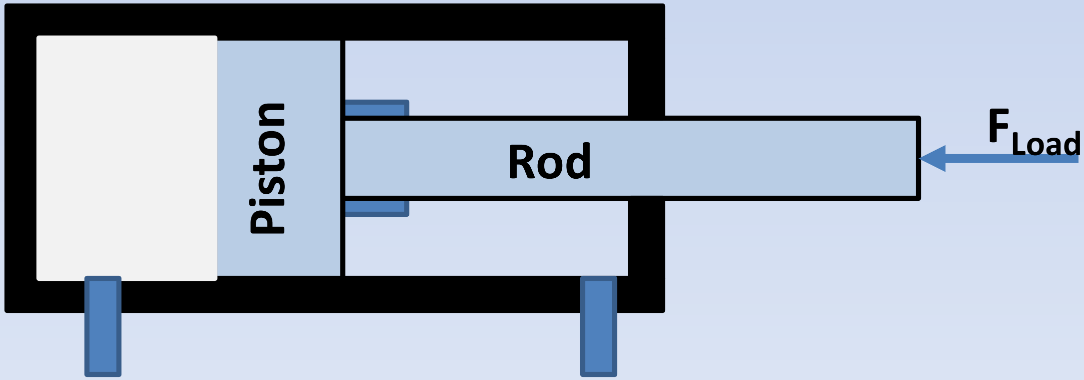
Piston diameter (mm)		40	50	63	80	100	125	140	160	180	200	220	250	280	320
Piston rod diameter (mm)	Small	20	28	36	45	56	70	90	100	110	125	140	160	180	200
	Large	28	36	45	56	70	90	100	110	125	140	160	180	200	220

Graphical symbols of different linear actuators

S. No.	Graphical Symbols	Explanation
1.		Single-acting cylinder with unspecified return
2.		Single-acting cylinder with spring return
3.		Double-acting cylinder –single piston rod
4.		Double-acting cylinder –doublepiston rod
5.		Telescopic cylinder–double acting

Cont.

6.		Telescopic cylinder–single acting
7.		Double-acting cylinder– fixed cushion on one side
1.		Double-acting cylinder–variable cushion on one side
9.		Double-acting cylinder–variable cushion on both sides



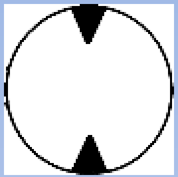
Piston
Rod
F_{Load}

Hydraulic Motors

Chapter 7

Dr. Suleiman BaniHani

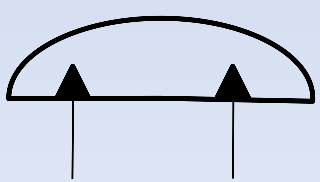
Hydraulic Motor

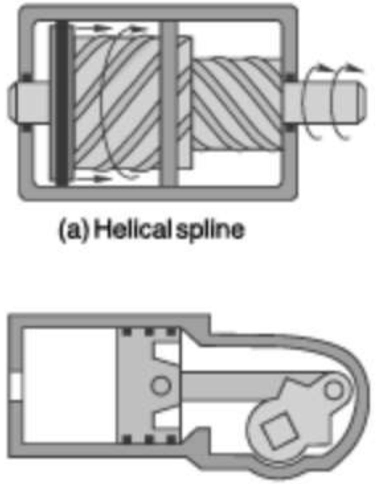


- A hydraulic motor is a mechanical actuator that converts hydraulic pressure and flow into torque and angular displacement (rotation).
- The hydraulic motor is the rotary counterpart of the hydraulic cylinder.
- Conceptually, a hydraulic motor should be interchangeable with a hydraulic pump because it performs the opposite function. However, most hydraulic pumps cannot be used as hydraulic motors because they cannot be backdriven.

Limited Rotation Hydraulic Motors

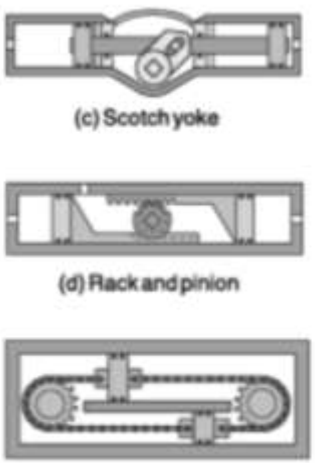
- Limited rotation hydraulic motor, also called oscillation motor or rotary actuator, provides rotary output over a finite angle.
- Produce high instantaneous torque in either direction





(a) Helical spline

(b) Enclosed piston crank



(c) Scotch yoke

(d) Rack and pinion

(e) Piston chain

(c) Double vane

(a) Bladder

(b) Single vane

Single Vane Rotary Actuator

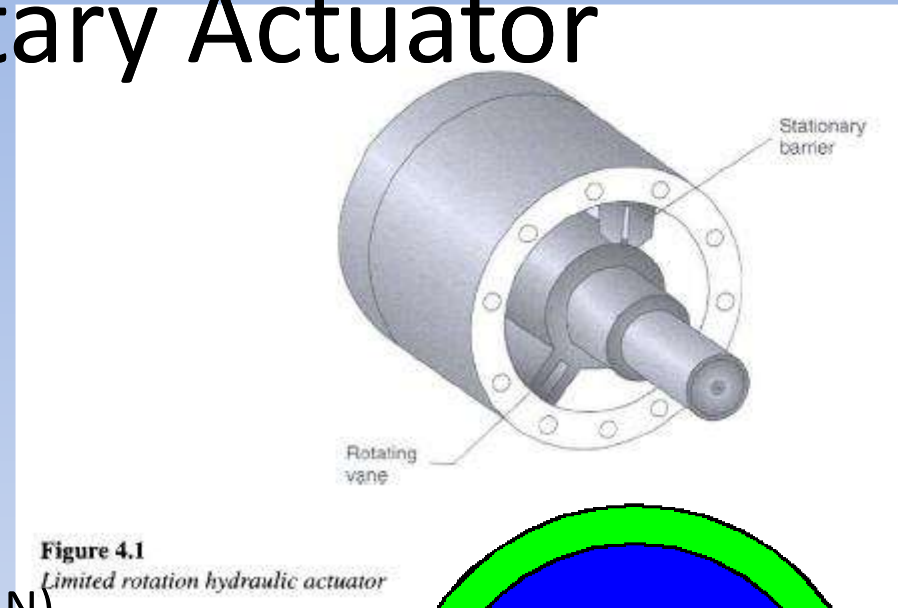
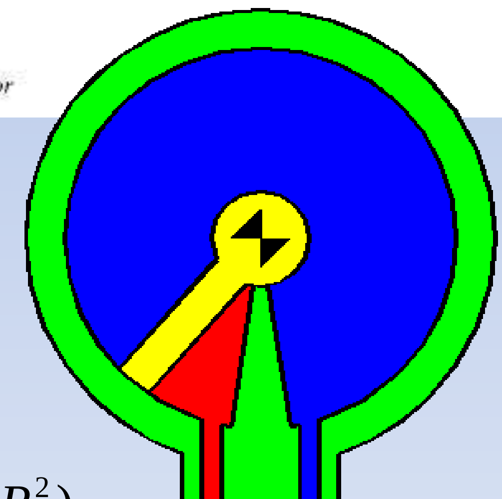


Figure 4.1
Limited rotation hydraulic actuator



Nomenclature

- R_R = outer radius of rotor (in, m)
- R_V = outer radius of vane (in, m)
- L = width of vane (in, m)
- P = hydraulic pressure (psi, Pa)
- F = hydraulic force acting on vane (lb, N)
- A = area of vane in contact with oil (in^2, m^2)
- T = torque capacity (in.lb, N.m)

$$F = PA = P(R_V - R_R)L$$

$$T = F\frac{(R_V + R_R)}{2} = P(R_V - R_R)L\frac{(R_V + R_R)}{2} = \frac{PL}{2}(R_V^2 - R_R^2)$$

$$V_D = \pi(R_V^2 - R_R^2)L \Rightarrow T = \frac{PV_D}{2\pi}$$

Example

A single vane rotary actuator has the following data

Rotor outer radius= 0.5in

Vane outer radius=1.5in

Width of vane=1in

If the torque load is 1000in.lb what pressure must be developed to overcome the load?

Sol.

$$\text{The volumetric displacement } V_D = \pi(R_V^2 - R_R^2)L = \pi(1.5^2 - 0.5^2)1 = 6.28in^3$$

$$P = \frac{2\pi T}{V_D} = \frac{2\pi(1000)}{6.28} = 1000\,psi$$

Applications

- Rotary actuators are used for mixing, dumping, intermittent feeding, screw clamping, continuous rotation, turning over, automated transfer, providing constant tension, and material handling. They are also suitable for turning, toggle clamping, indexing, positioning, oscillating, lifting, opening, closing, pushing, pulling, and lowering.

Hydraulic Motors

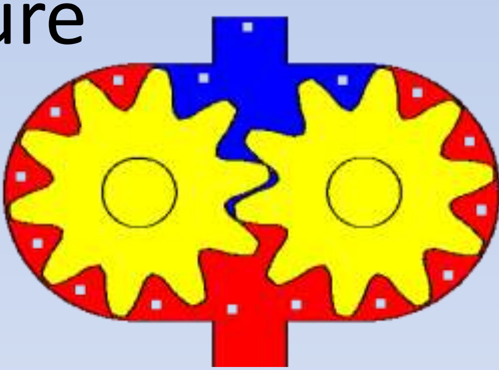
- Hydraulic motors can rotate continuously and have the same basic configuration as pumps.
- Most hydraulic motors have casing drains to protect shaft seals.
- Three basic types of motors
  - Gear
  - Vane
  - Piston

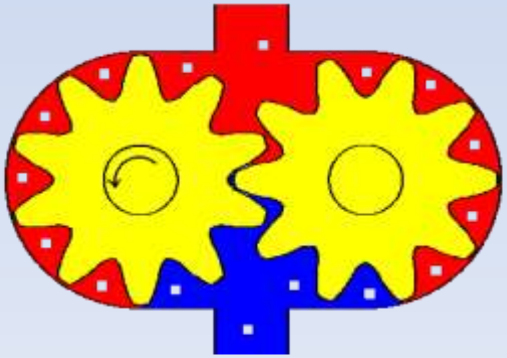
Gear Hydraulic Motor

- Develop torque due to pressure acting on the gear teeth.
- Direction reversal by reversing flow
- Not balanced with respect to pressure
- Large sid
- Limited to 2000psi and 2400rpm,

150gpm.

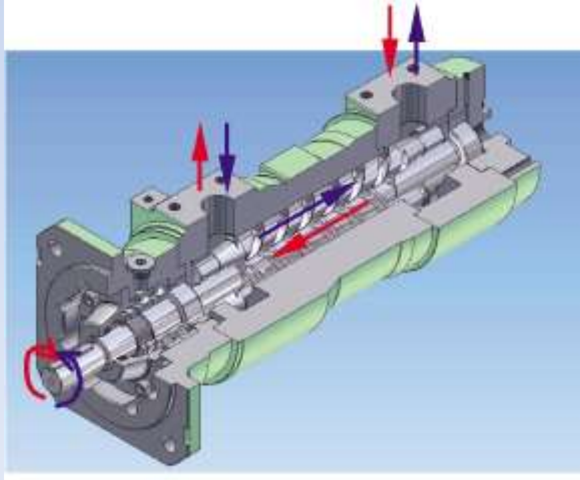
- Simple design, low cost





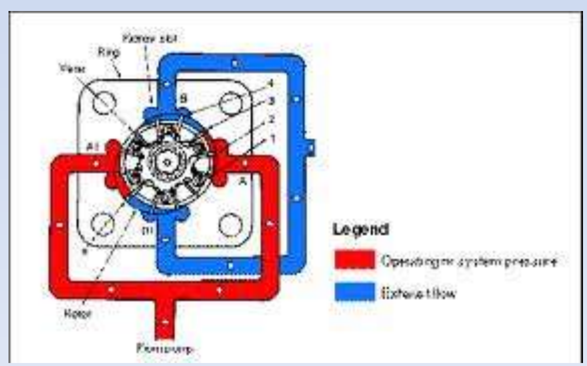
Internal Gear and Screw Motor

- Internal gear motor
  - Greater pressure and speed as well as displacement than external gear motors
- Screw type motor
  - Three meshing screws(a power rotor and two idler rotors)
  - Quite operation
  - Pressure up to 3000psi and V_D up to 13.9in^3



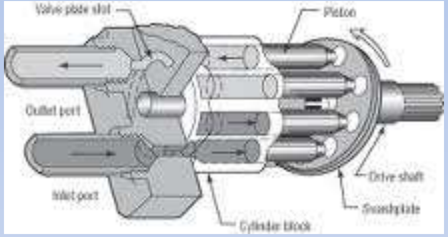
Vane Motors

- Uses springs to force the vanes to follow the surface of the cam (no centrifugal force before rotor moves)
- Vane motors are universally of the balanced type (fixed displacement)
- Low noise and vibration and high energy efficiency



Piston Motors

- Piston motors can be either fixed or variable displacement.
- In-line piston motors (swash plate)



- Axial piston motor (bent axis)
- Axial piston are the most efficient of the three types
- High speed up to 12000rpm and
- Pressure up to 5000psi,
- flow up to 450gpm

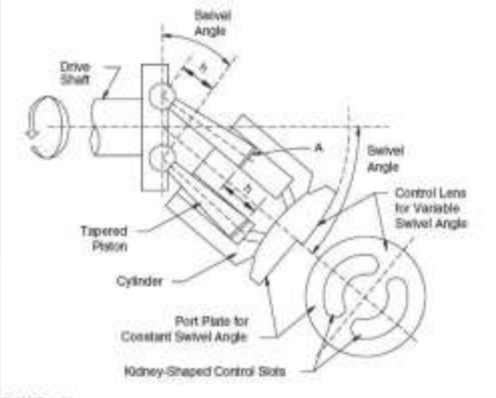


FIGURE 5.11
Diagram of bent axis motor (Dimension A is the area of the individual piston, and h is the stroke.)

Hydraulic Motor Theoretical Torque, Power, and Flow Rate

- Due to frictional losses a hydraulic motor delivers less torque than it should theoretically.
- The theoretical torque for limited rotation hydraulic actuator can be given by $T_T(in.lb) = \frac{V_D(in^3/rev) \times P(psi)}{2\pi}$ or

$$T_T(N.m) = \frac{V_D(m^3/rev) \times P(Pa)}{2\pi}$$

- The theoretical power is given by

$$HP_T = \frac{T_T(in.lb) \times N(rpm)}{63{,}000} = \frac{V_D(in^3/rev) \times P(psi) \times N(rpm)}{395{,}000}$$

$$Theoretical\ \ power(W) = T_T(N.m) \times N(rad/s) = \frac{V_D(m^3/rev) \times P(Pa) \times N(rad/s)}{2\pi}$$

- And theoretical flow rate

$$Q_T(gpm) = \frac{V_D(in^3/rev) \times N(rpm)}{231}$$

$$Q_T(m^3/s) = V_D(m^3/rev) \times N(rev/s)$$

Example

A hydraulic motor has a 5 in^3 volumetric displacement. If it has a pressure rating of 1000 psi and it receives oil from a 10 gpm theoretical flow rate pump, find the motor

a) Speed
b) Theoretical torque
c) Theoretical horsepower

Sol

$$a.\ N = \frac{231Q_T}{V_D} = \frac{231(10)}{5} = 462rpm$$

$$b.\ T_T = \frac{V_D P}{2\pi} = \frac{5(1000)}{2\pi} = 795in.lb$$

$$c.\ HP_T = \frac{T_T N}{63{,}000} = \frac{795(462)}{63{,}000} = 5.83HP$$

Hydraulic Motor Performance

- Motor performance depend on the precision of its manufacturing as well as the maintenance of close tolerances under design conditions.
- Internal leakage(slippage) between inlet and outlet reduce volumetric efficiency.
- Friction between mating parts and fluid turbulence reduce mechanical efficiency.
- Gear motors typically have an overall efficiency of 70% to 75% as compared to 75% to 85% of vane motors and 85% to 95% for piston motors.

Motor efficiencies

- Volumetric efficiency (η_v)The volumetric efficiency of the hydraulic motor is the inverse of the pumps, Hence

$$\eta_v = \frac{\text{theoretica l flow rate motor should consume}}{\text{actual flow rate consumed by motor}} = \frac{Q_T}{Q_A}$$

- Mechanical efficiency (η_m) The mechanical efficiency of the motor is the inverse of a pump

$$\eta_m = \frac{\text{actual torque delivered by motor}}{\text{torque motor should theoretic ally deliver}} = \frac{T_A}{T_T}$$

- Overall efficiency (η_o) is the product of the volumetric efficiency by the mechanical efficiency

$$\eta_o = \eta_v \times \eta_m$$

$$\eta_o = \frac{\text{actual power delivered by motor}}{\text{actual power delivered to motor}}$$

Example

A hydraulic motor has a displacement of 10 in^3 and operate at 1000psi pressure and a speed of 2000rpm. If the actual flow rate consumed by the motor is 95gpm and the actual torque delivered by the motor is 1500 in.lb find

η_v, η_m, η_o, and the actual horsepower delivered by the motor

Sol.

$$\eta_v = \frac{Q_T}{Q_A} = \frac{V_D N / 231}{Q_A} = \frac{10 \times 2000 / 231}{95} = 0.911 = 91.1\%$$

$$\eta_m = \frac{T_A}{T_T} = \frac{T_A}{P \times V_D / 2\pi} = \frac{1500}{1000 \times 10 / 2\pi} = 0.942 = 94.2\%$$

$$\eta_o = \eta_v \times \eta_m = 0.911 \times 0.942 = 0.858 = 85.8\%$$

$$HP_A = \frac{T_A N}{63{,}000} = \frac{1500 \times 2000}{63{,}000} = 47.6hp$$

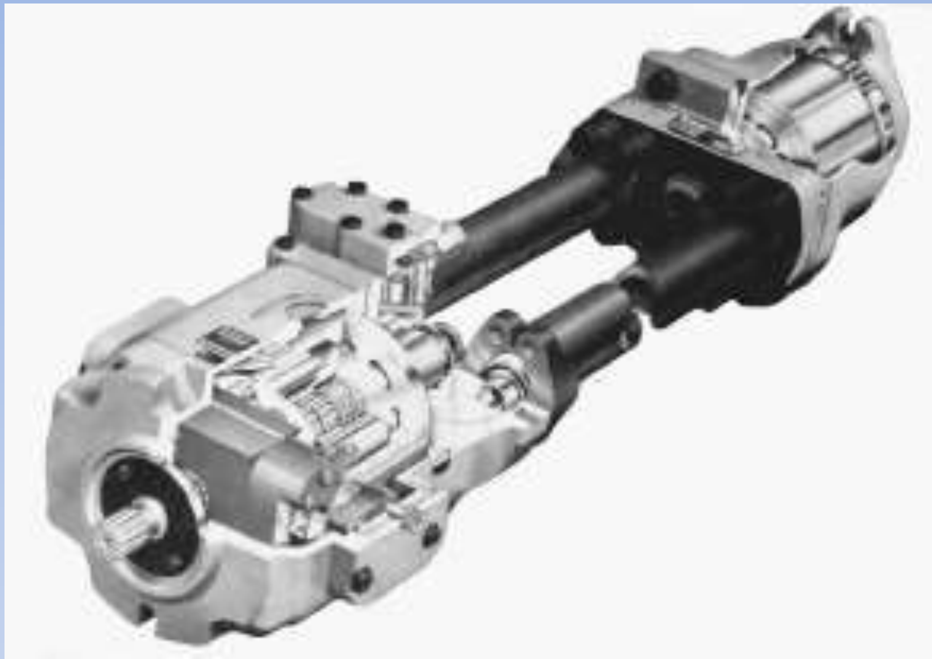
Hydrostatic Transmission

- The primary function of any hydrostatic transmission (HST) is to accept rotary power from a prime mover having specific operating characteristics and transmit that energy to a load having its own operating characteristics.
- HST generally must regulate speed, torque, power, or, in some cases, direction of rotation. Depending on its configuration, the HST can drive a load from full speed in one direction to full speed in the opposite direction, with infinite variation of speed between the two maximums - all with the prime mover operating at constant speed.
- The operating principle of HSTs is simple: a pump, connected to the prime mover, generates flow to drive a hydraulic motor, which is connected to the load.

Hydrostatic Transmission Advantages

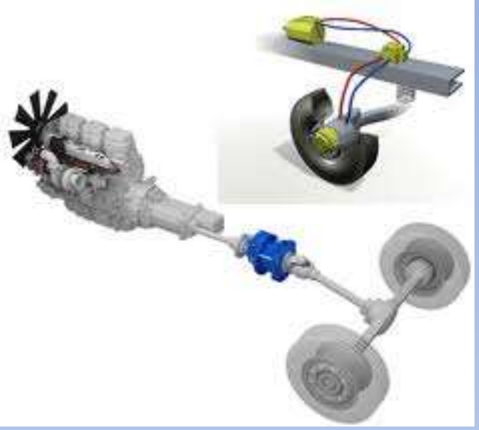
HSTs offer many important advantages over other forms of power transmission.

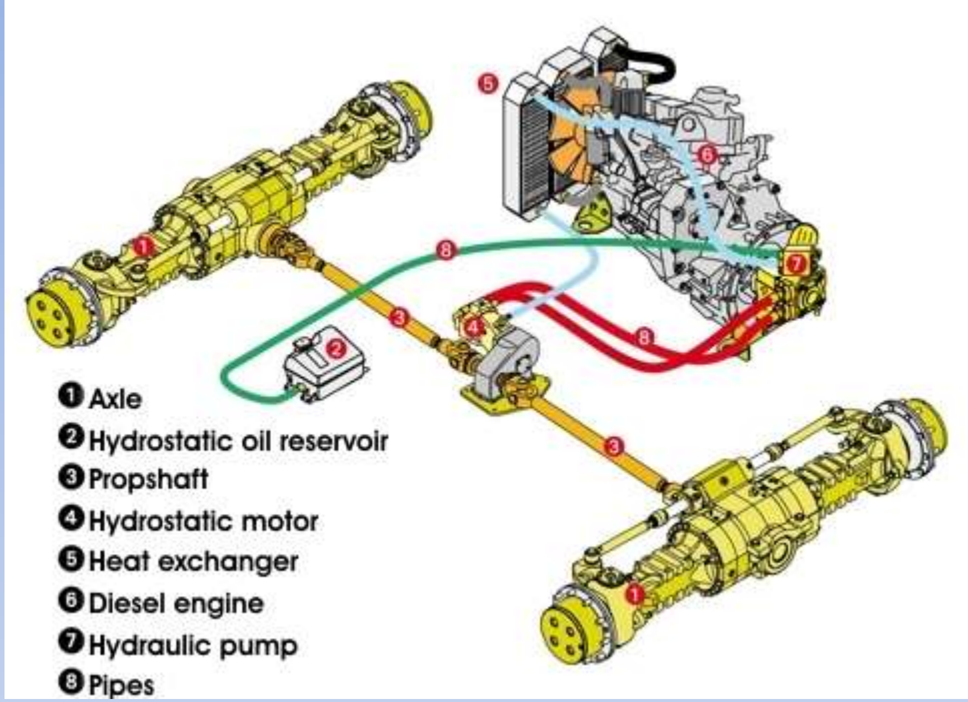
- transmits high power in a compact size
- exhibits low inertia
- operates efficiently over a wide range of torque-to-speed ratios
- maintains controlled speed (even in reverse) regardless of load, within design limits
- maintains a preset speed accurately against driving or braking loads
- can transmit power from a single prime mover to multiple locations, even if position and orientation of the locations changes
- can remain stalled and undamaged under full load at low power loss
- does not creep at zero speed
- provides faster response than mechanical or electromechanical transmissions of comparable rating, and
- can provide dynamic braking.

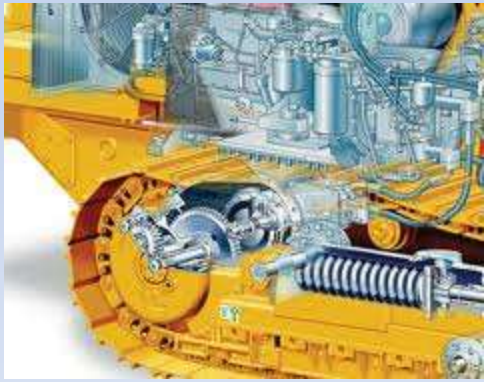


Packaged HST encloses pump, motor, controls, conducting system, and all auxiliary components into a single housing. The unit shown accepts input power from a V-belt drive and transmits power to the load through its output shaft. Packaged HSTs are available in a variety of configurations, many of which bolt directly to an engine.

A typical hydrostatic transmission consists of a variable-displacement pump and fixed-displacement motor connected through metal tubing, hose assemblies, or both. Providing a reservoir (and usually a heat exchanger and filtration system) between the pump and motor forms an open-circuit HST.




1 Axle
2 Hydrostatic oil reservoir
3 Propshaft
4 Hydrostatic motor
5 Heat exchanger
6 Diesel engine
7 Hydraulic pump
8 Pipes



Example

A hydrostatic transmission, operating at 1000psi pressure, has the following characteristics.

Pump	Motor
V_D=5 in^3	V_D=?
η_v=82%	η_v=92%
η_m=88%	η_m=90%
N=500 rpm	N=400 rpm

Find the

- Displacement of the motor
- Motor output torque

Sol

$$Q_{Tpump} = \frac{V_D \times N}{231} = \frac{5 \times 500}{231} = 10.8gpm$$

$$Q_{Apump} = \eta_{vpump} \times Q_{Tpump} = 0.82 \times 10.8 = 8.86gpm$$

$$Q_{Amotor} = Q_{Apump} = 8.86gpm$$

$$Q_{Tmotor} = Q_{Amotor} \times \eta_{vmotor} = 8.86 \times 0.92 = 8.15gpm$$

$$V_{Dmotor} = \frac{Q_{Tmotor}}{N_{motor}} = \frac{8.15 \times 231}{400} = 4.71in^3$$

$$\text{Hydraulic HP delivered to motor} = \frac{\text{P} \times \text{Q}_\text{A}}{1714} = \frac{1000 \times 8.86}{1714} = 5.17hp$$

$$\text{brake HP delivered by motor} = 5.17 \times 0.92 \times 0.90 = 4.28$$

$$\text{Torque delivered by motor} = \frac{\text{HP delivered by motor} \times 63{,}000}{\text{Motor speed}} = \frac{4.28 \times 63{,}000}{400} = 674in.lb$$

Thanks

Hydraulic Valves

Chapter 8

Dr. Suleiman BaniHani

Introduction

- Valves are used primarily to control fluid power system.
- Three types of valves
  - Directional control valves
    - Controls the direction of flow in a hydraulic circuit
      - Check valves, shuttle valves, and two-way, three-way, and four-way directional control valves.
  - Pressure control valves
    - Protect the system from overpressure
      - Pressure relief, pressure reducing, sequence, unloading, and counterbalance valves
  - Flow control valves
    - Control fluid flow rate in various lines of a hydraulic system
      - Non-compensated, and pressure-compensated flow control valves

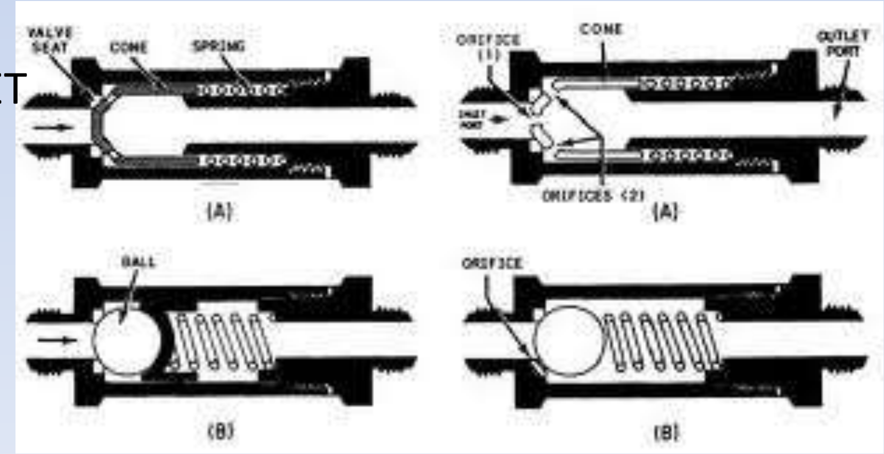
Directional Control Valves

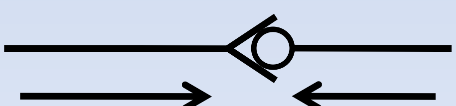
- Used to control the flow direction.
- Contain ports that are external openings where fluid can enter and leave via pipelines
- The number of ports on a directional control valve (DCV) are identified by the term *way.*

Check Valves

- The simplest type of direction control valves is a check valve
- Has tow ports two-way valve
- Its purpose is to permit free flow in one direction and prevent it in the opposite direction

POPPET

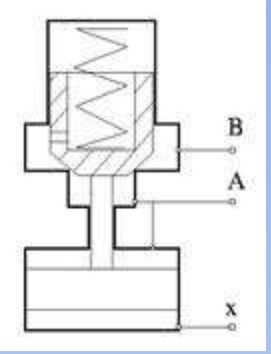




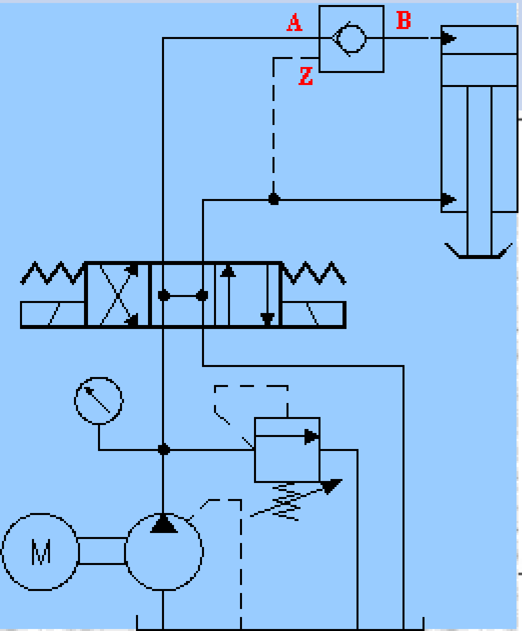
Free-Flow direction

No-Flow direction

Pilot Operated Check Valve



- Permit free flow in one direction but permits flow in the normally blocked direction if pilot pressure is applied at the pilot pressure port
- Used to lock hydraulic cylinder in position



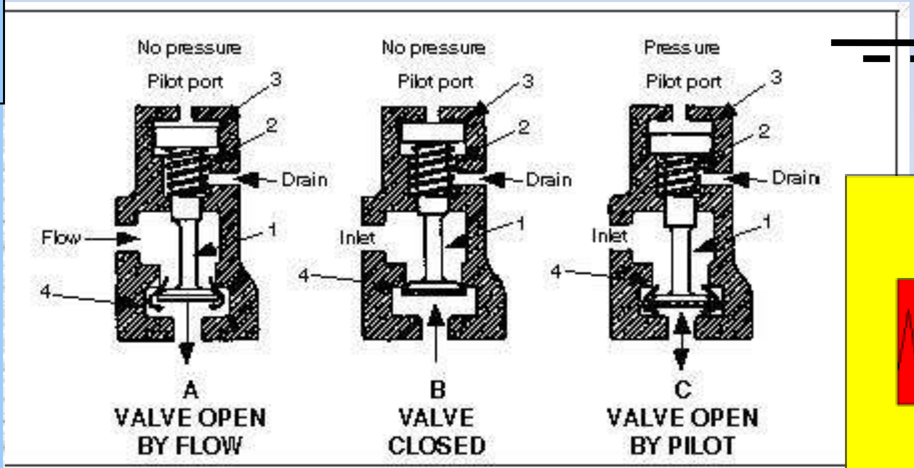
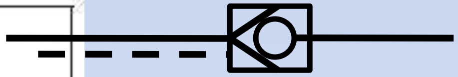
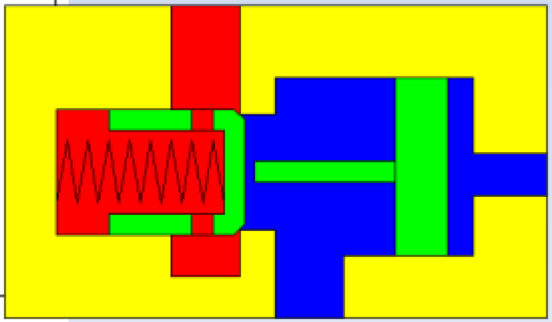


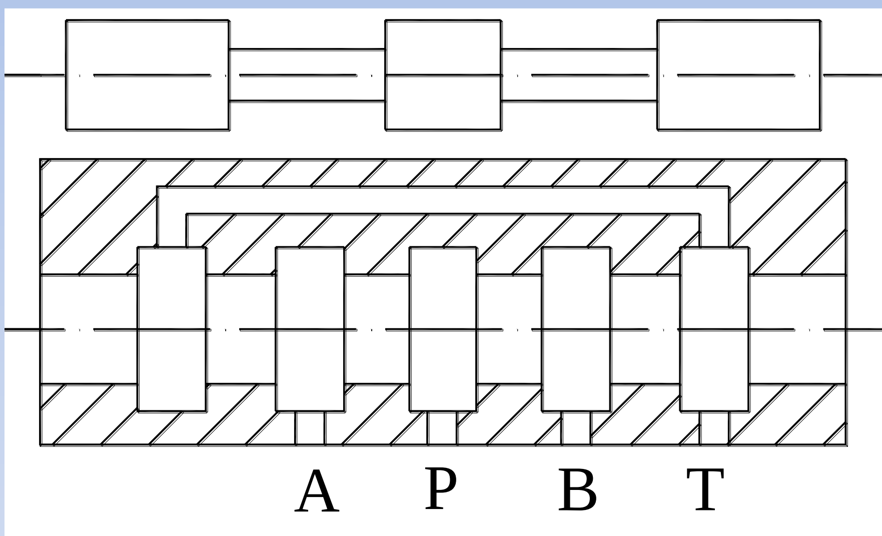
Figure 5-20. Pilot-operated check valve, second type

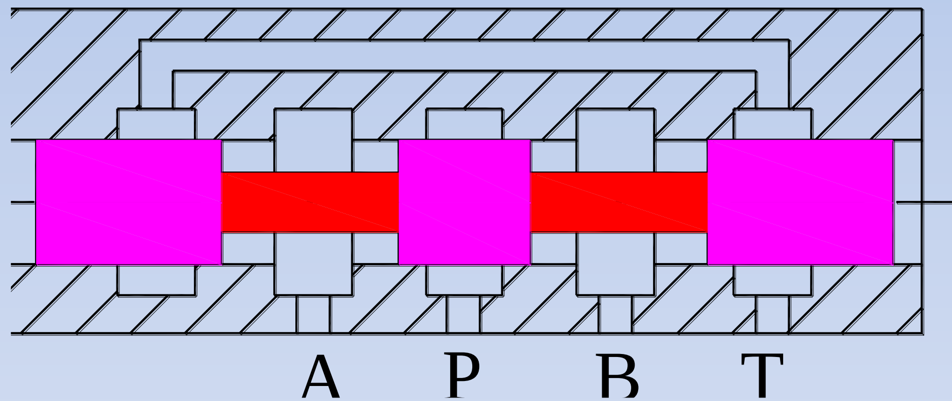




Sliding Spool Valves

Most directional control valves use a sliding spool to change the path of flow through the valve.





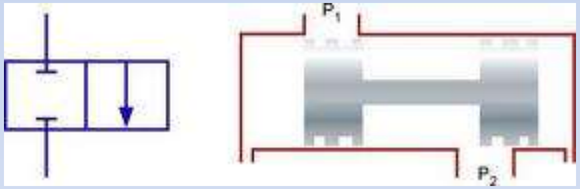
Position： For a given position of the spool, a unique flow path configuration exists within the valve.

Way: The number of "ways" refers to the number of ports in the valve.

Normal/Neutral/center position: The spool is not actuated

Two Way Valves

- A two-way valve is generally used to control the direction of fluid flow in a hydraulic circuit and is a sliding-spool type



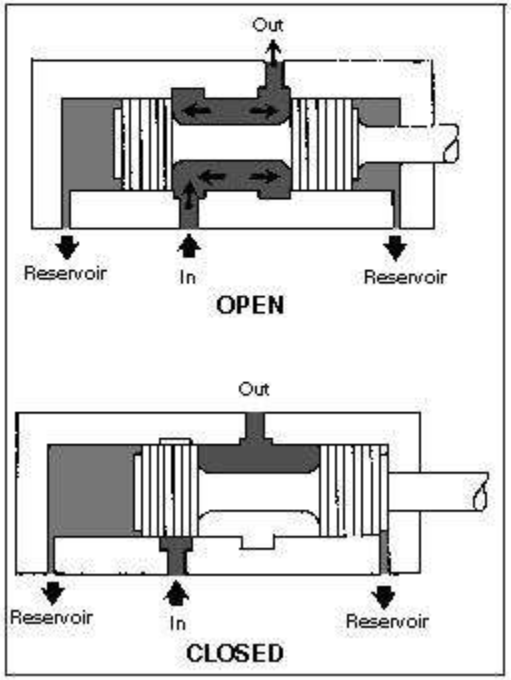


Figure 5-21. Two-way valve

5.3.1 Two-way directional control valves

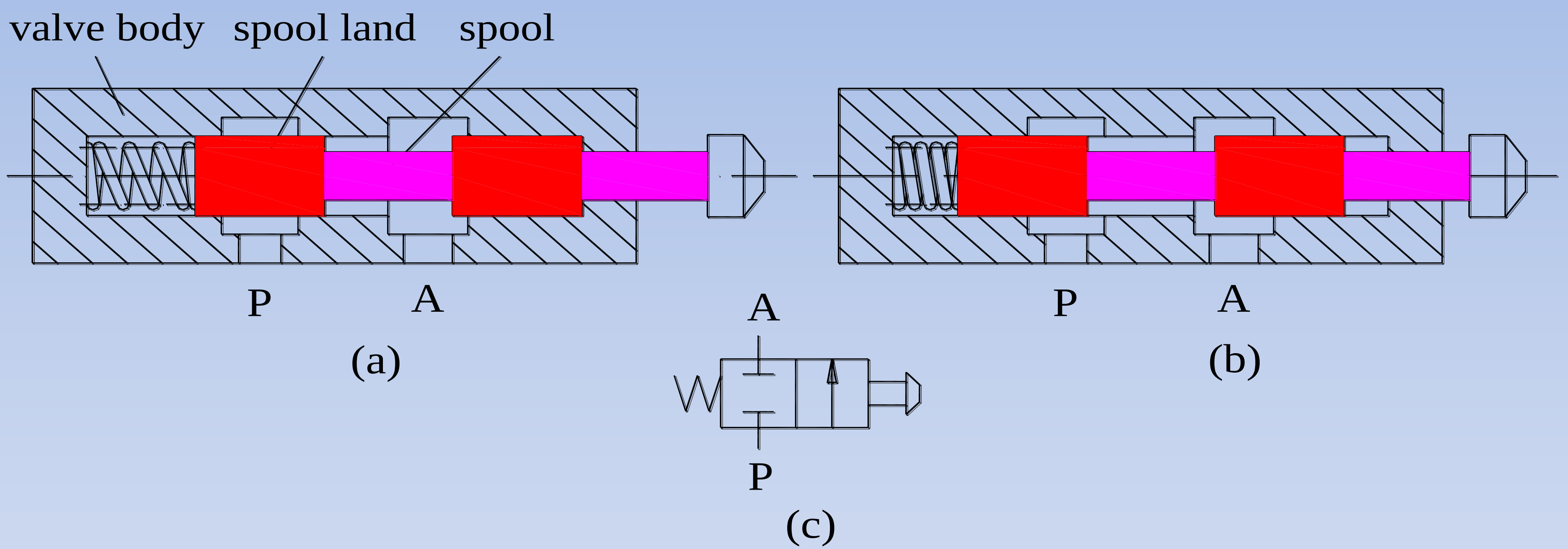


Figure 5-6 Two-way, two-position normally closed directional control valve.
(a) Valve in the normal position (b) Valve actuated (c) Complete graphic symbol

Three Way Valves

- Contains three ports, typically of spool design, usually operate in two or three positions
- The spool can be positioned manually, mechanically, using pilot pressure, or using electrical solenoids.

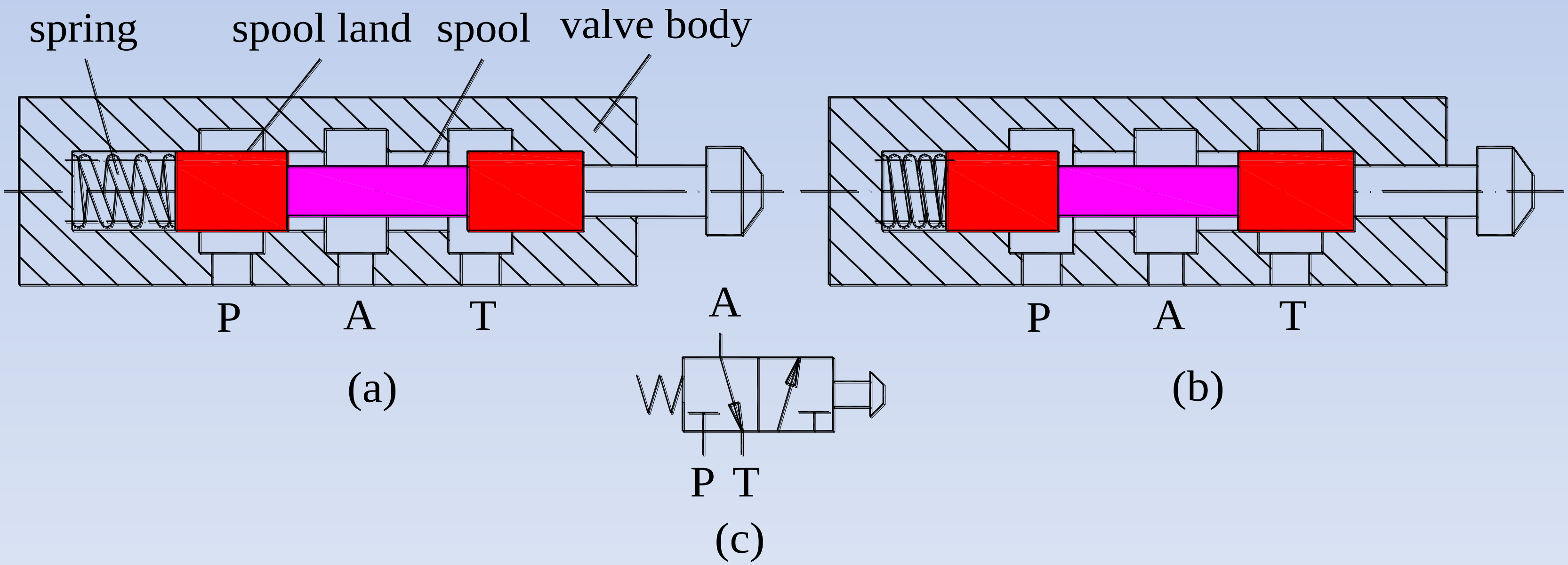
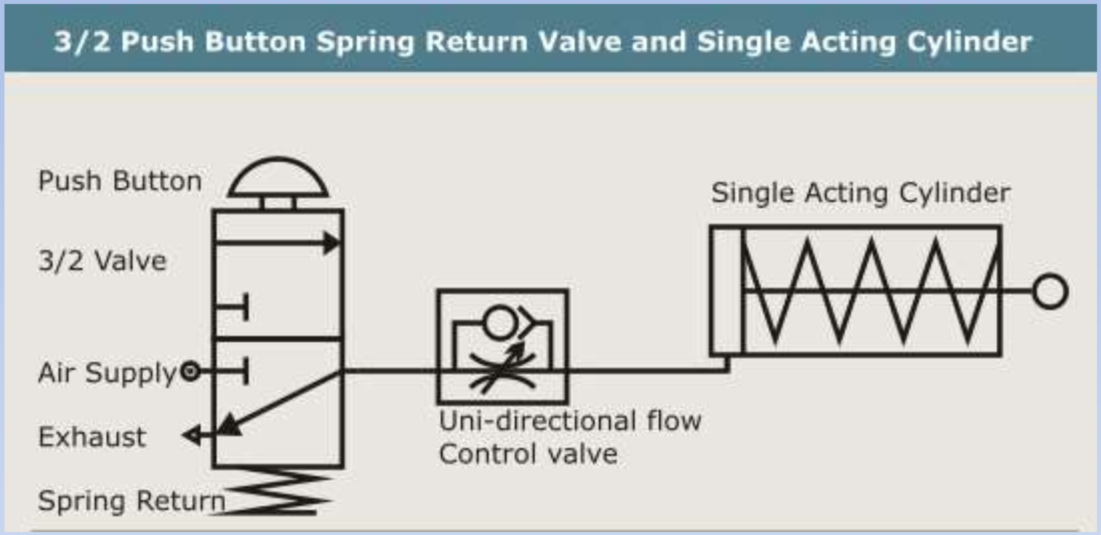


Figure 5-8 Three-way, two-position normally closed directional control valve.
(a) Valve in the normal position (b) Valve actuated (c) Complete graphic symbol

Three-Way Valve with Single Acting Cylinder



Four-Way Valves

- Typically used to control the flow directions to and from double acting cylinders
- If there is more than one tank port, they all count as one port because they all serve the same function

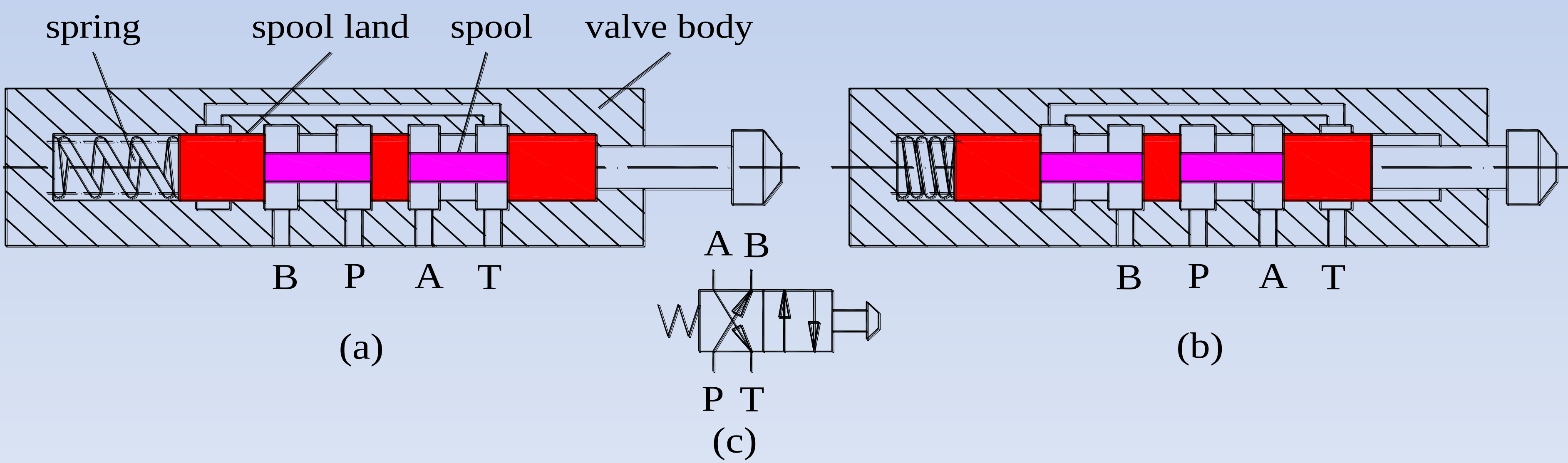


Figure 5-10 Four-way, two-position directional control valve.
(a) Valve in the normal position. (b) Valve actuated. (c) Complete graphic symbol.

direction Control Valve Actuation

Manually-actuated valve

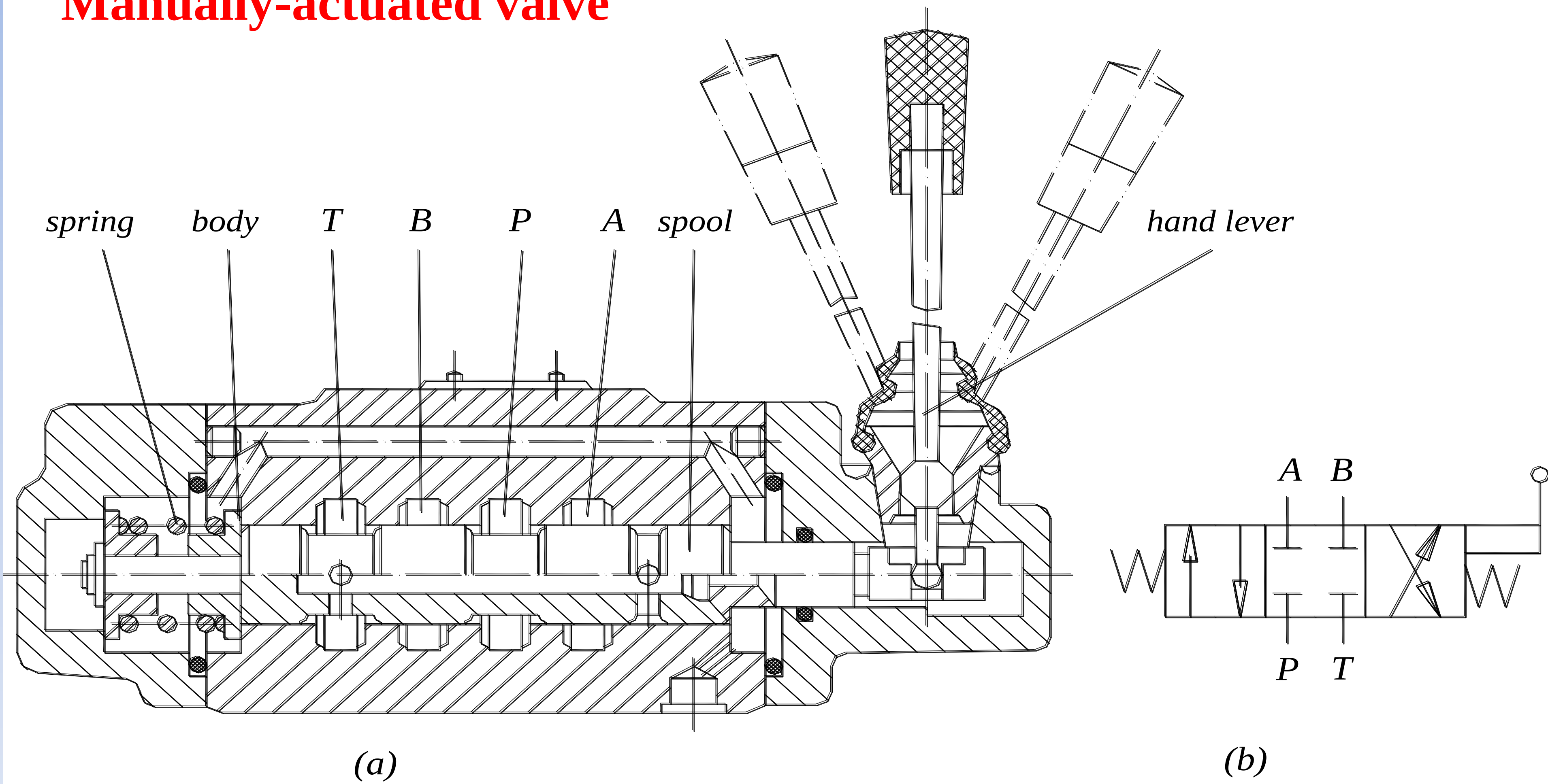


Figure 5-13 Manually actuated, spring-centered, three-position, four-way valve
(a) Construction (b) Complete graphic symbol

Mechanically-actuated valve

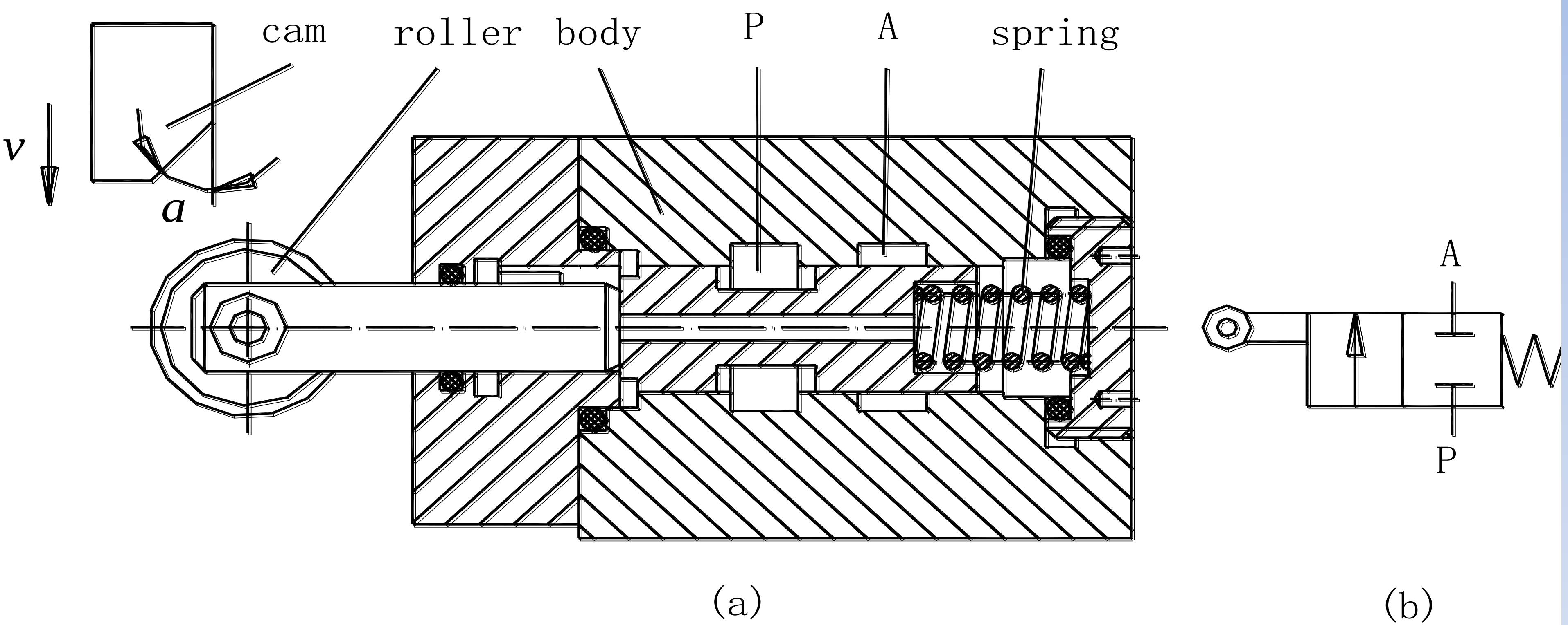


Figure5-14 Mechanically-actuated two-way valve
(a) Construction (b) Complete graphic symbol

Pilot-actuated valve

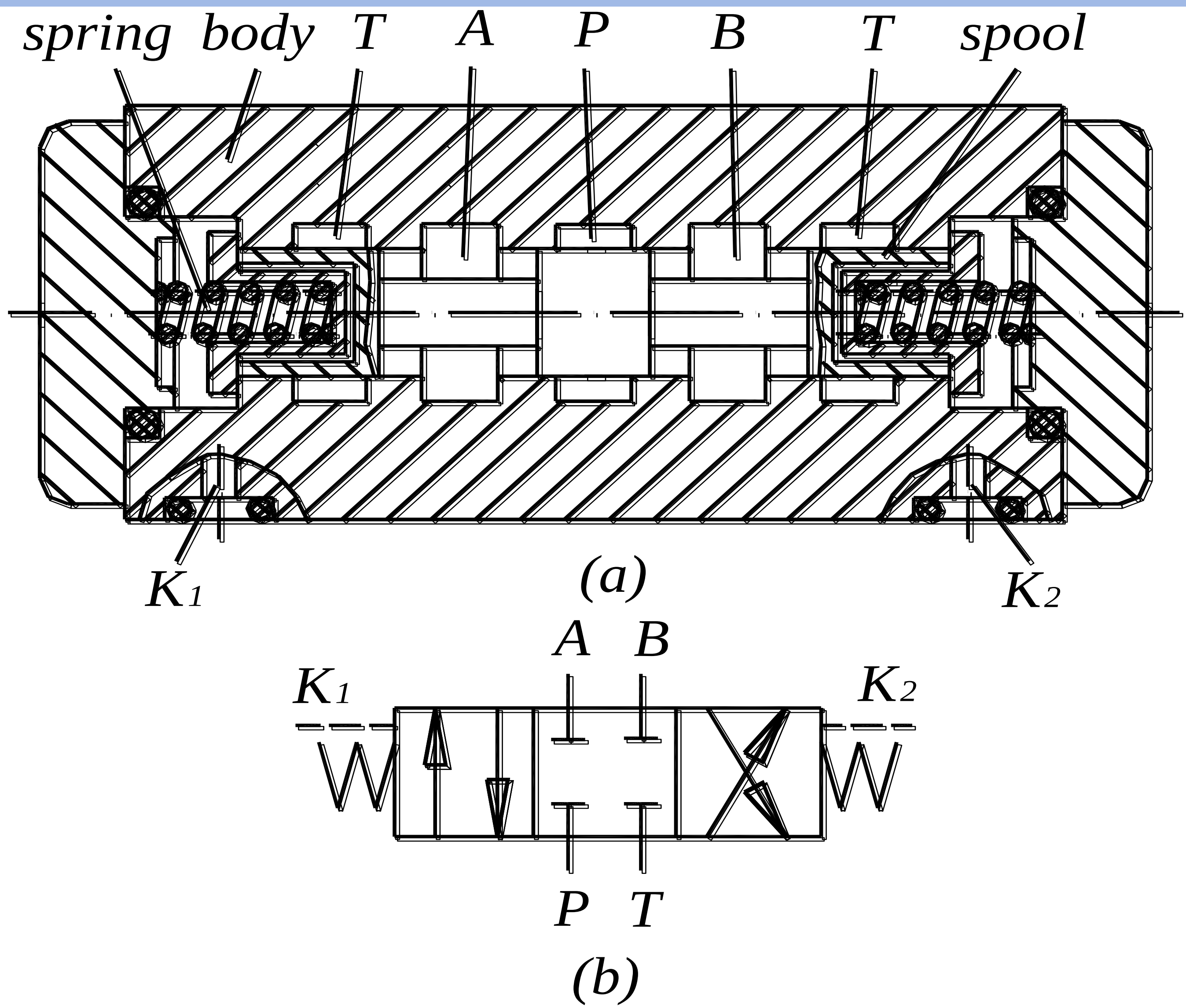


Figure 5-15 Oil pilot-actuated four-way valve

(a) Construction (b) Complete graphic symbol

Solenoid-actuated valve

Two solenoid designs to dissipate heat in the coil; **air gap solenoid** and **wet pin solenoid**. The first type dissipates heat into the air, while the second design has a passageway inside the push pin to the solenoid which allows the oil from the tank to carry the heat away and cool the coil.

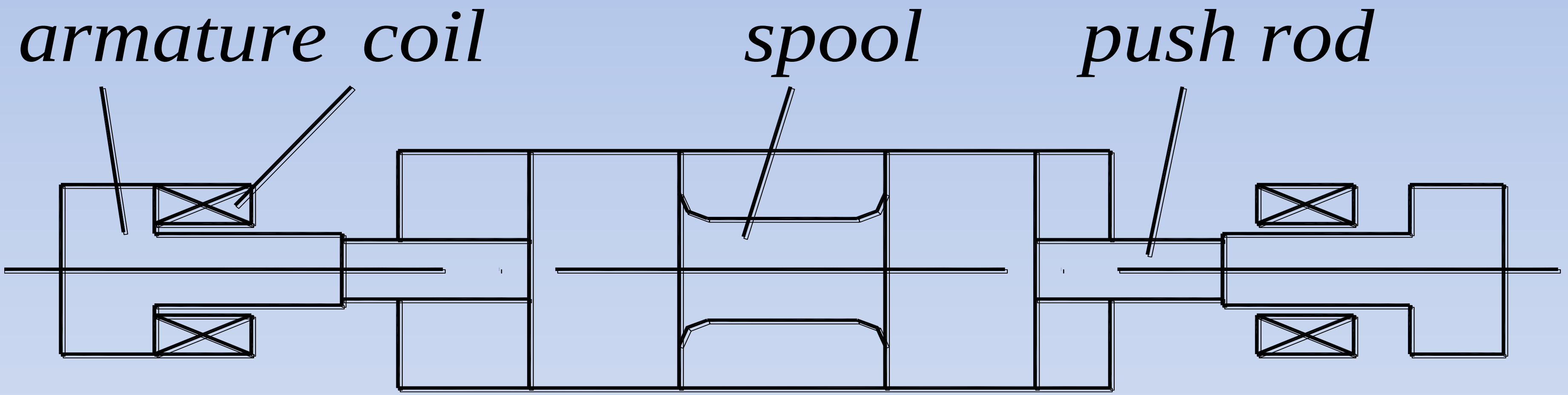
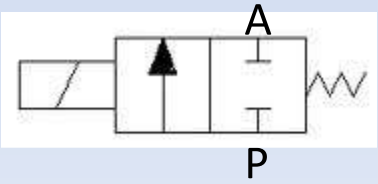
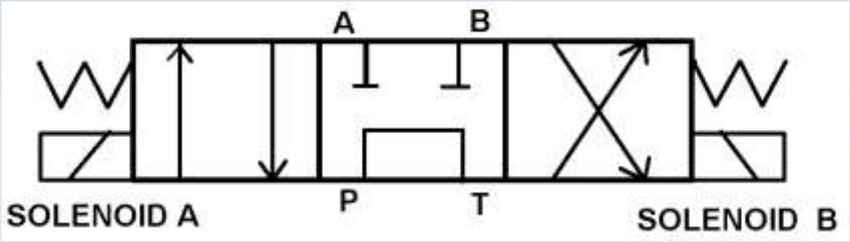


Figure 5-16 Operation of solenoid to shift spool of valve.





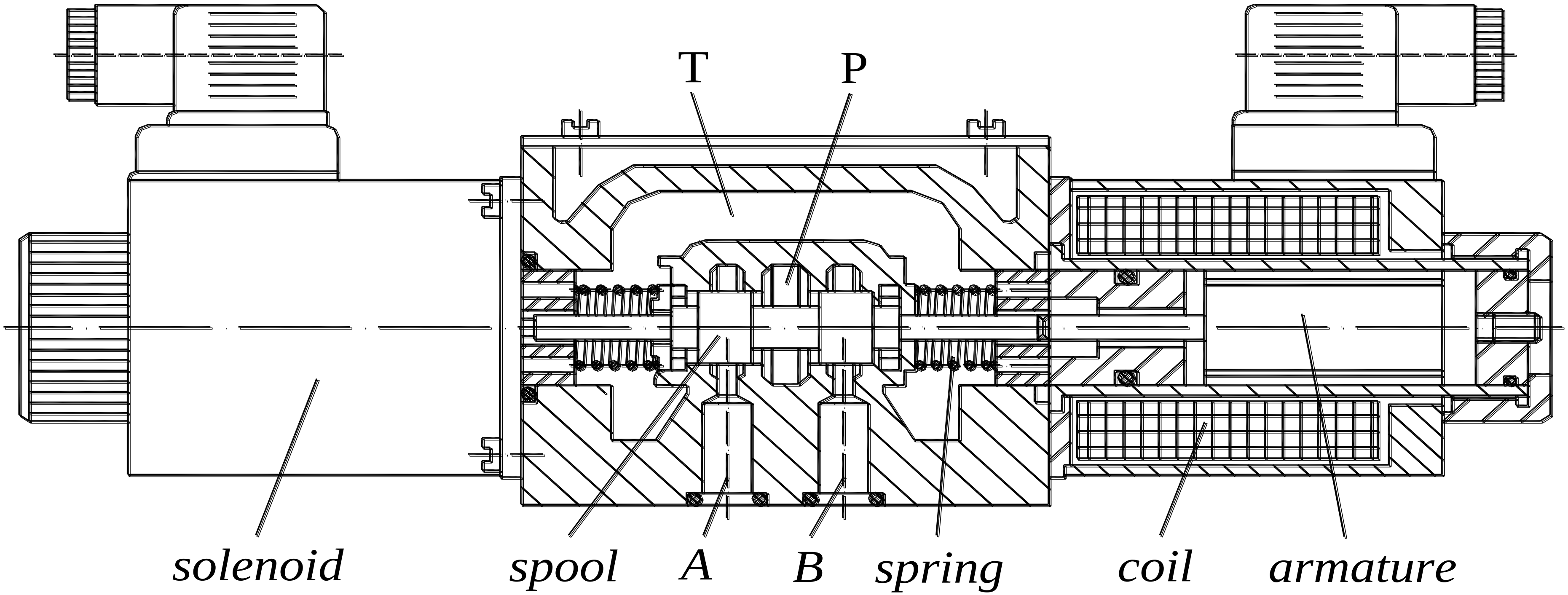


Figure 5-17 Solenoid-actuated directional control valve.

5.4.5 Combination actuation

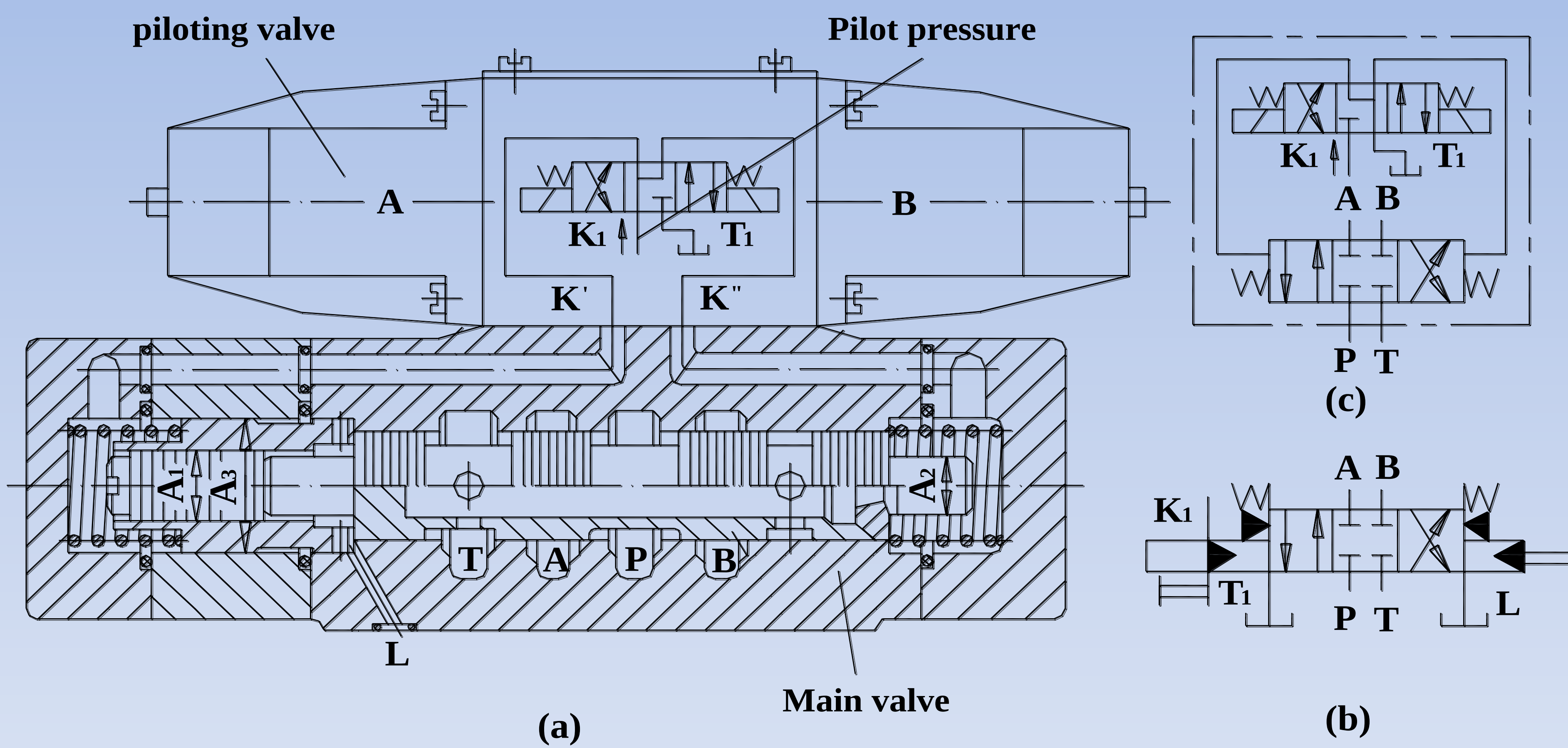
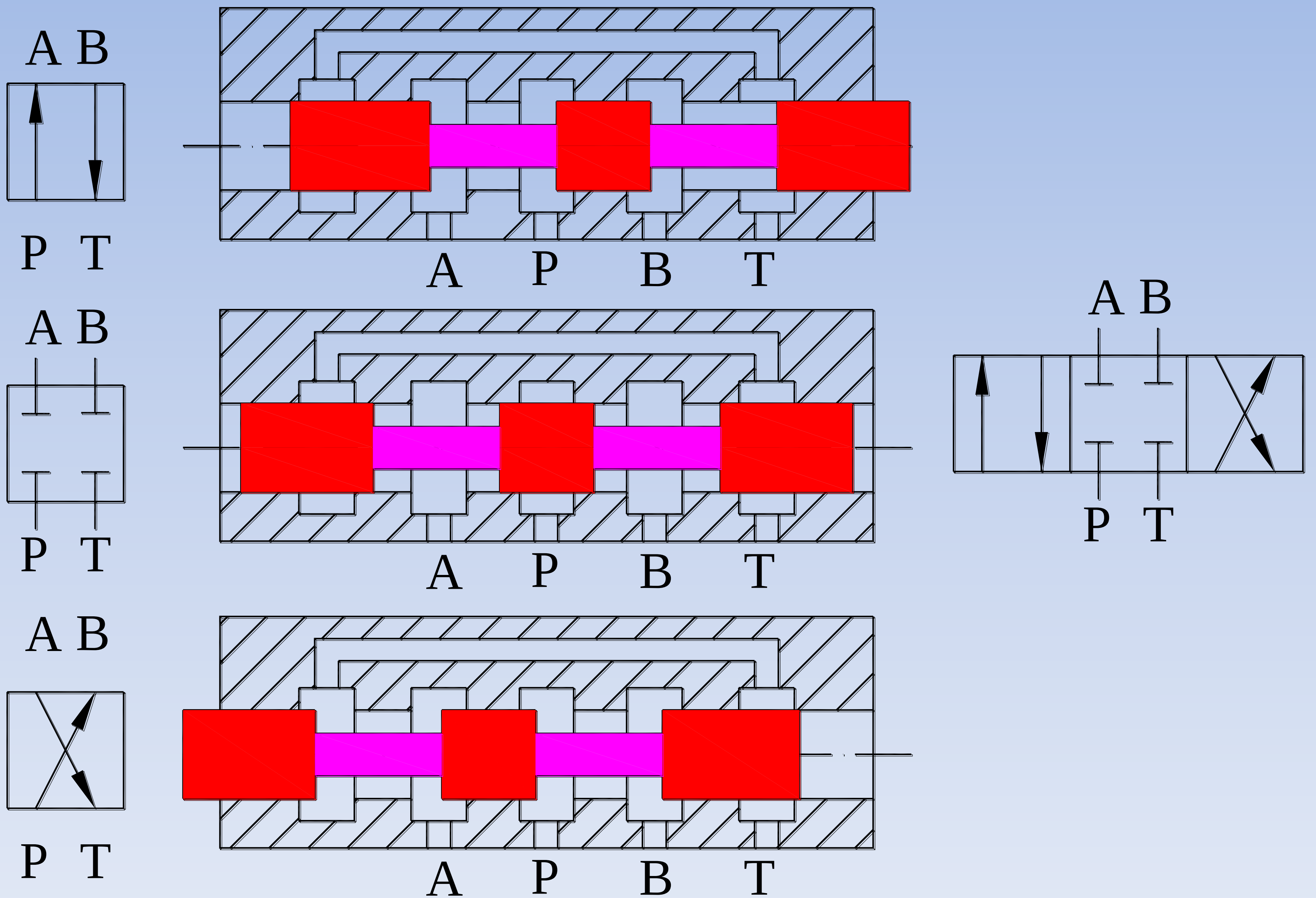


Figure 5-19 Piggy-back directional control package.
(a) Construction (b) Complete graphic symbol

4-way, 3-position directional control valves



5.3.4 Center positions in three-position, four-way valves

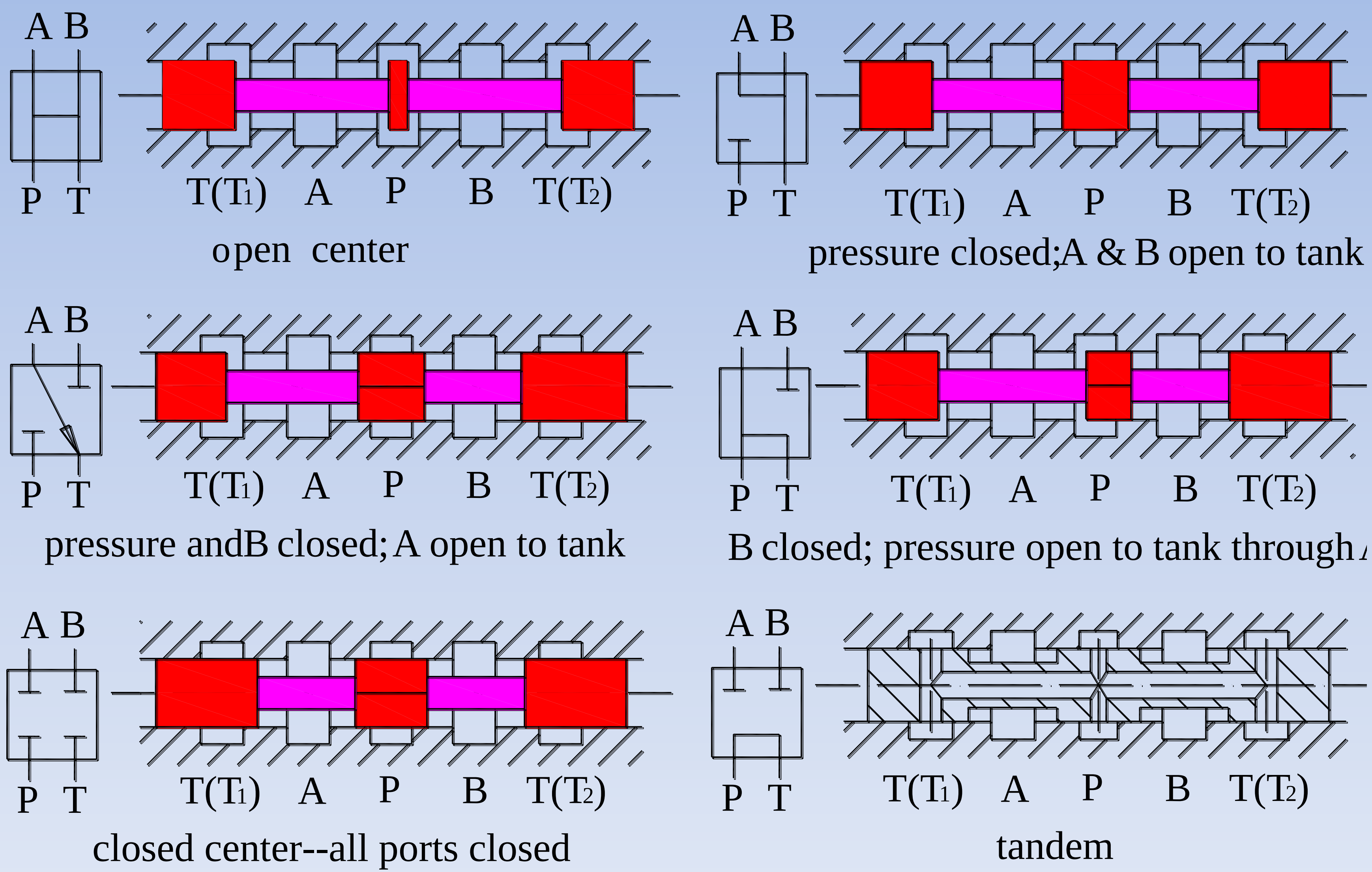


Figure 5-12 Various center flow paths for three-position,four-way valve

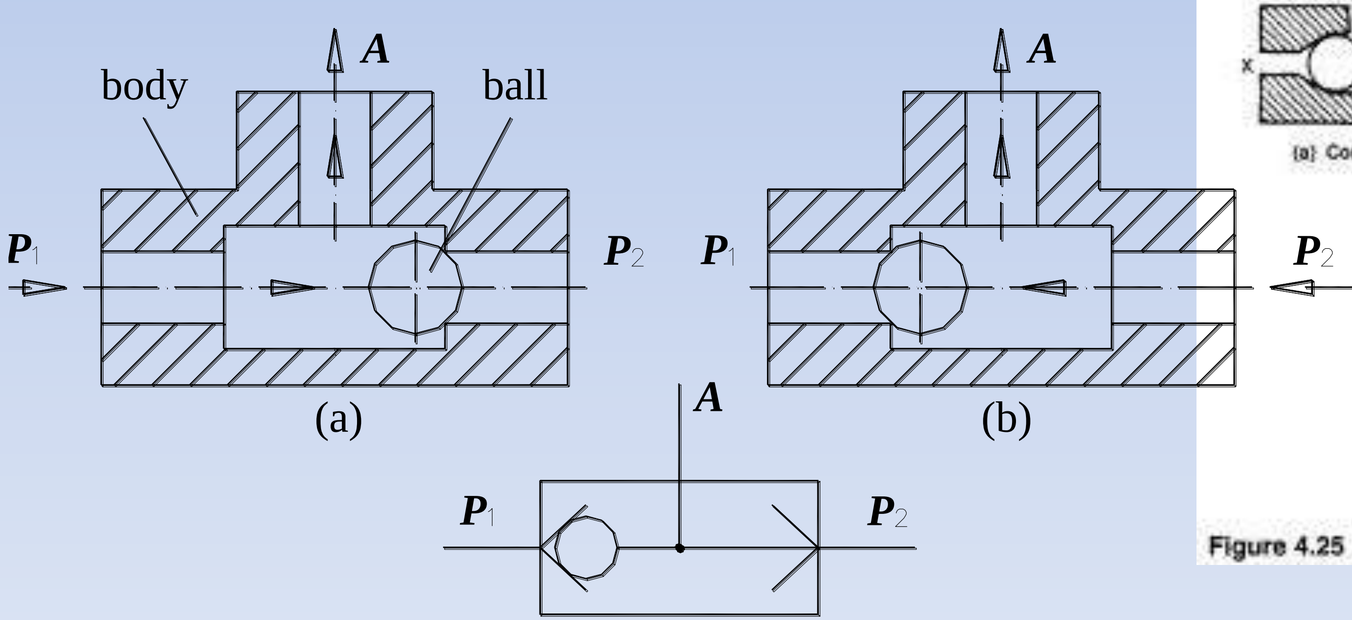
- **The open-center** type connects all ports together. The pump flow can return directly back to the tank at essentially atmospheric pressure, little horsepower is consumed. The actuator(cylinder or motor) can be moved freely by applying an external force.

- **The closed-center** design has all ports blocked. The pump flow can be used for other circuit. The actuator is hydraulically locked. This means it cannot be moved by the application of an external force.

- **The tandem** design also results in a locked actuator. It also unloads the pump at essentially atmospheric pressure.

5.2 Shuttle Valve

A shuttle valve allows two alternate flow sources to be connected to one branch circuit



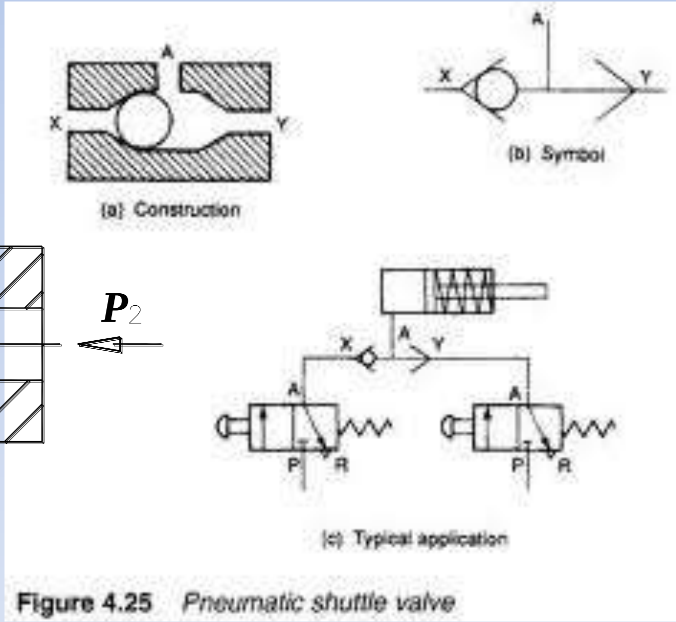
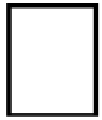
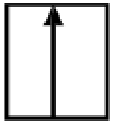
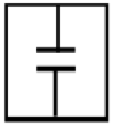
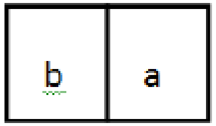
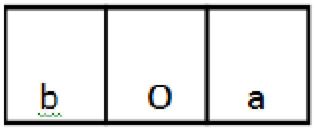


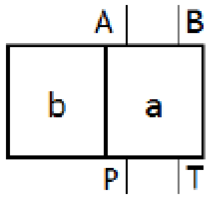
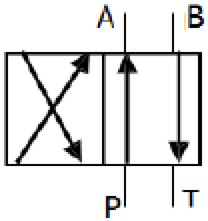
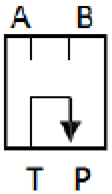
Figure 5-5 Shuttle valve

(a) ***P_1 higher than P_2***. (b) ***P_2 higher than P_1***. (c) Graphic symbol

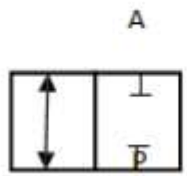
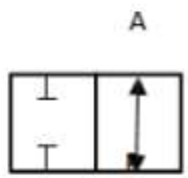
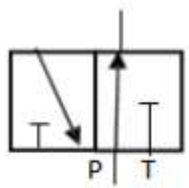
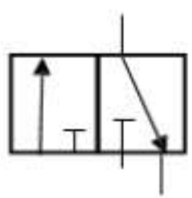
Symbols

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

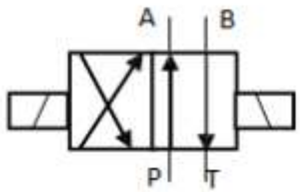
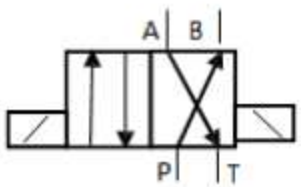
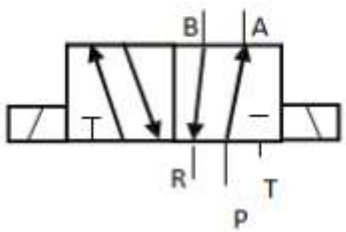
Cont.

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

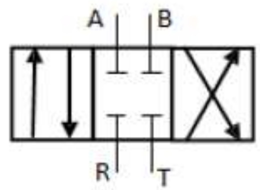
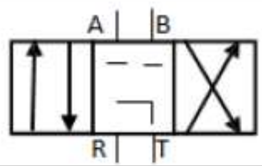
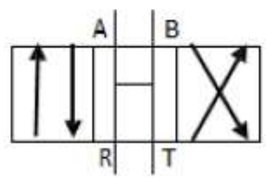
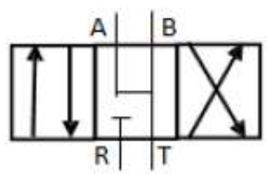
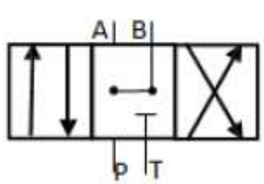
Cont.

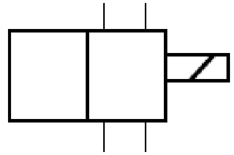
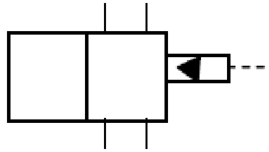
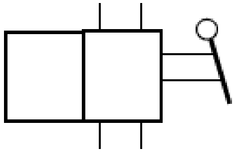
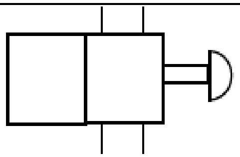
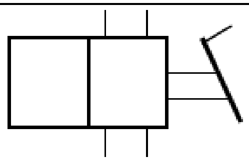
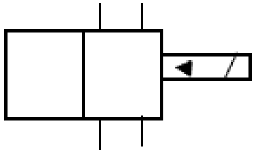
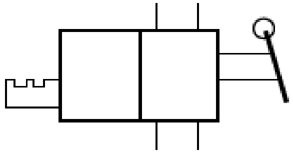
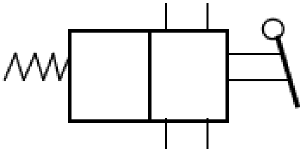
2/2-way valve: 2-ports and 2-position DCV	
	Normally closed position: P is not connected to A. When the valve is not actuated, the way is closed.
	Normally open position: P is connected to A. When the valve is not actuated, the way is open.
3/2 way valve : 3ports and 2 position DCV	
	Normally open position: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

Cont.

<table>
<tr><th colspan="2">4/2-way valve – 4-port and 2-position DCV</th></tr>
<tr><td>
</td><td>P is connected to A
B is connected to T</td></tr>
<tr><td>
</td><td>Position 2:
P is connected to B
A is connected to T</td></tr>
<tr><th colspan="2">5/2-way valve – 5-port and 2-position DCV</th></tr>
<tr><td>
</td><td>Normal position:
P is connected to B
A is connected to R</td></tr>
</table>

Cont.

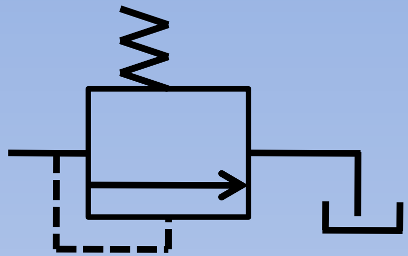
4/3-way valve – 4-port and 3-position DCV	
	P, T, A, B
	Mid-position pump reticulating: P to T, A and B closed
	H-Mid-position closed: P to A, B to T
	Mid-position working lines depressurized: P, A to B to T
	P to A to B, T

	Solenoid operated
	Pilot operated
	Manual operated
	Push button
	Foot operated
	Pilot-operated solenoid
	Two-position detent
	Spring return

Pressure Control Valves

- Pressure control valves are classified as Pressure relief, pressure reducing, sequence, unloading, and counterbalance valves
- The most commonly used pressure control valve is pressure relief valves, it is found practically in every hydraulic system

Pressure Relief Valves



- **Pressure relief valves limit/ maintain the maximum pressure in a hydraulic circuit by diverting pump flow back to the tank.**

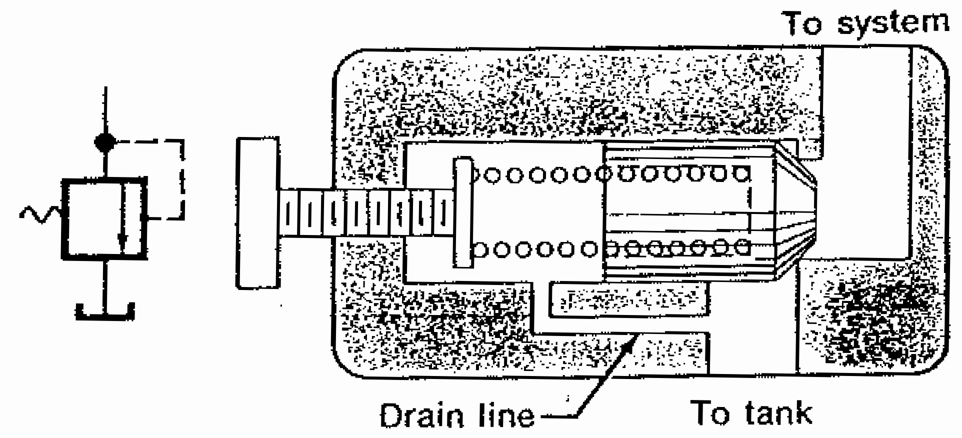
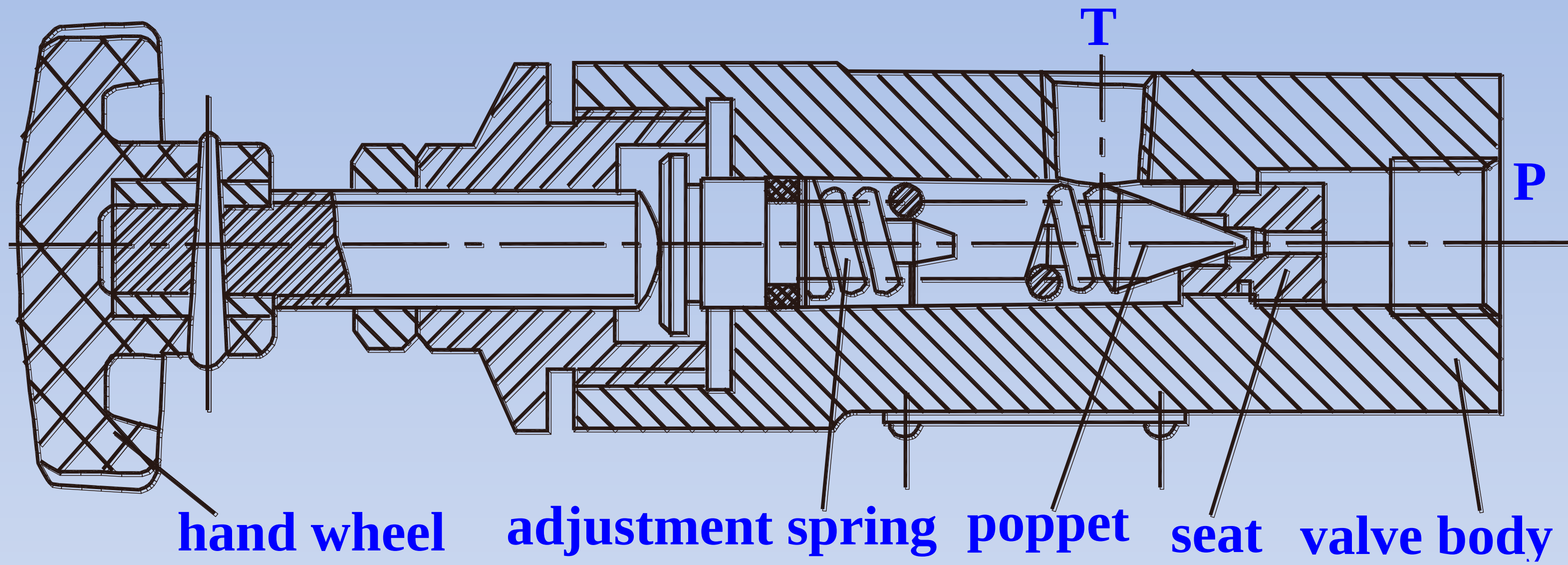


Fig. 2.6. Pressure relief valve regulates system output fluid pressure.

Pressure Relief Valve

1 Structure

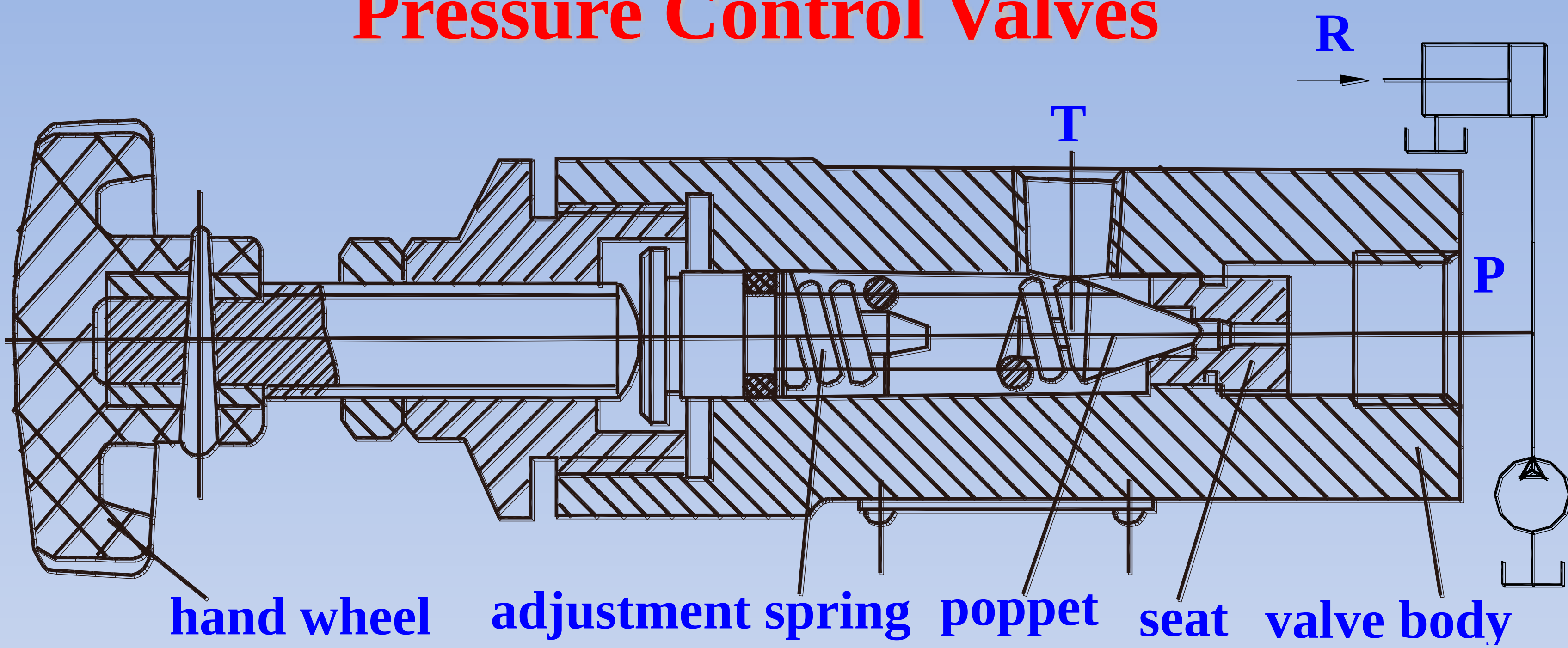


2 Operating process

When the hydraulic force is less than the spring force, the poppet remains on its seat and no flow pass through the valve.

When the hydraulic force is greater than the spring force, the poppet will be forced off its seat, and fluid will flow back to the tank through port *T*.

Pressure Control Valves



3 Equilibrium equation

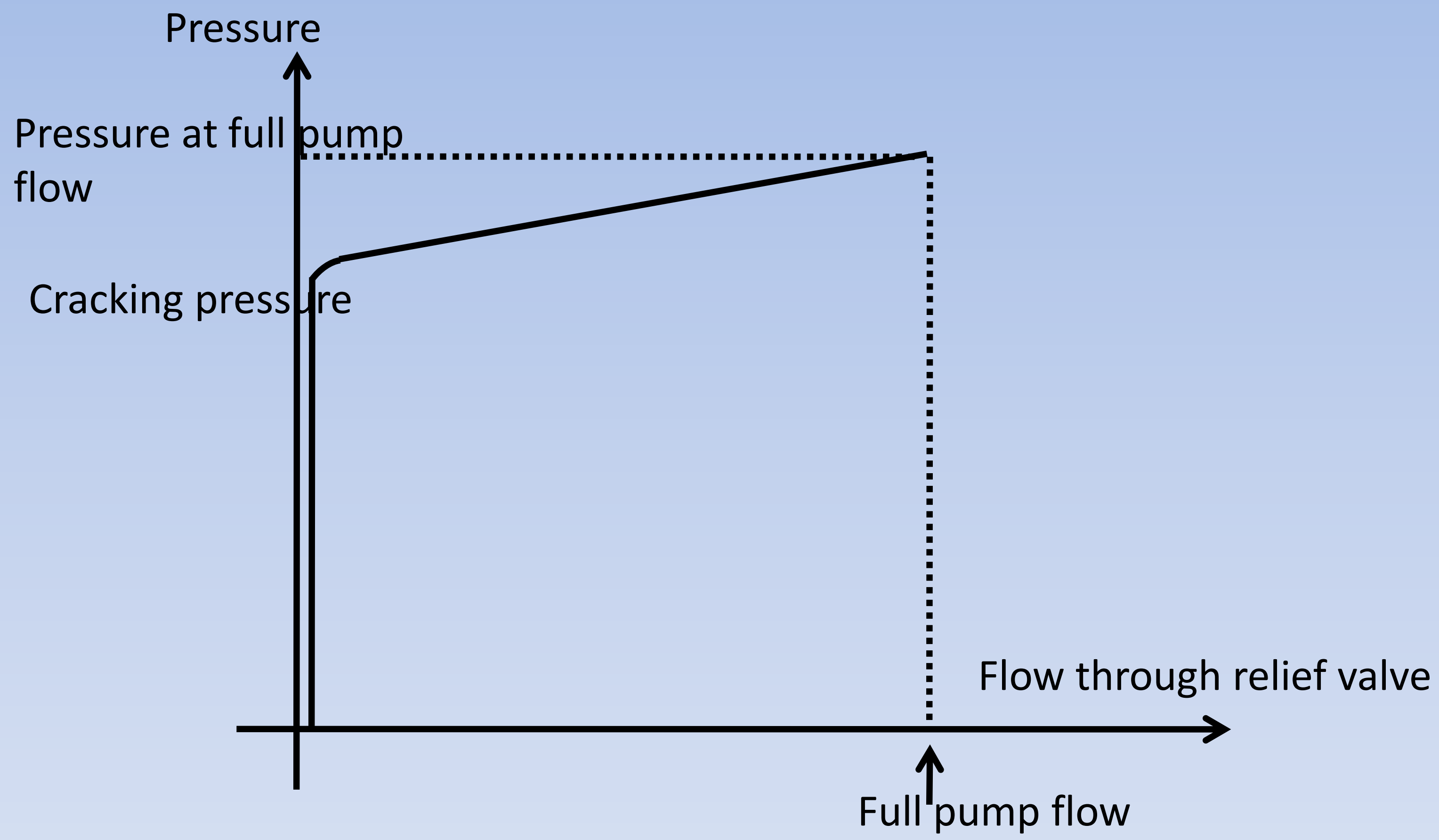
Neglecting the poppet weight, friction force, and flow force, the static force equilibrium equation of the poppet is

$$\Sigma F = F_S - pA = k(x_0 + x) - pA = 0$$

that is $$p = k(x_0 + x) / A$$

If the k is very small and $x_0 >> x$

$$p \approx kx_0 / A = \text{costant}$$


Pressure
Pressure at full pump flow
Cracking pressure
Flow through relief valve
Full pump flow

Example

A pressure relief valve has a poppet 0.75 in^2 area on which the system pressure acts, the spring constant is 2500 lb/in. the spring was initially compressed by 0.20 from the free length. To pass the full pump flow the PRV poppet must move 0.10 in from its fully closed position. Find

a. The cracking pressure

b. Full pump flow pressure

Sol.

$$\text{a. At cracking pressure } F_{spring} = F_{pressure}$$

$$F_{Spring} = kx_0 = 2500 \times 0.2 = 500lb = p \times A_{poppet} = p_{cracking} \times 0.75$$

$$p_{cracking} = 667\,psi$$

$$\text{b. Pressure at full pump flow } F_{spring} = F_{pressure}$$

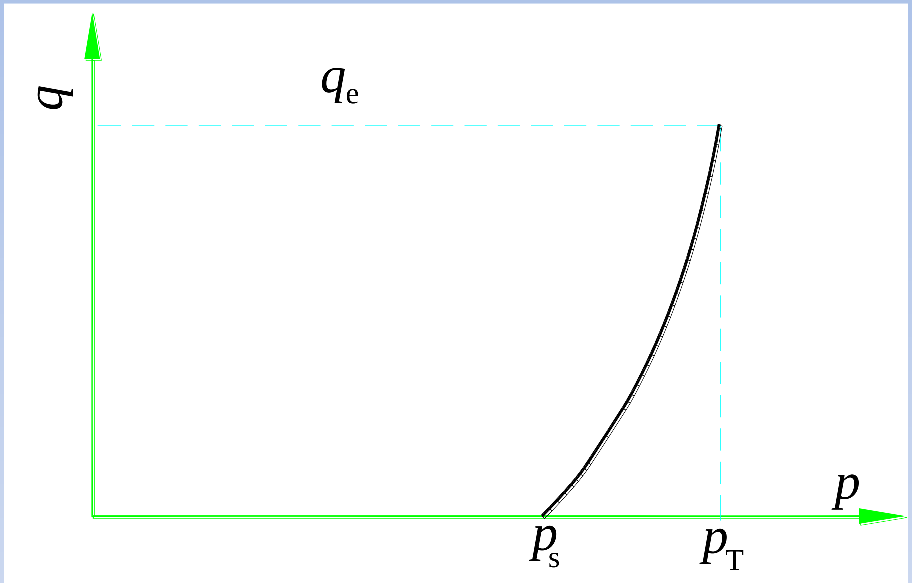
$$F_{Spring} = k(x_0 + x) = 2500 \times (0.2 + 0.1) = 750lb$$

$$= p \times A_{poppet} = p_{fullpumpflow} \times 0.75$$

$$p_{fullpumpflow} = 1000\,psi$$

Chapter 6 Pressure Control Valves

4 Characteristic curve



The relief valve should be able to pass through the overall flow rate of the pump.

5 advantage and disadvantage

The direct-acting pressure relief valve has a simple construction and a high sensitivity; however, it is not suitable for the application of high pressure and high flow rate.

Pressure Control Valves

Pilot-operated pressure relief valves (Compound pressure relief valve)

View and constitution

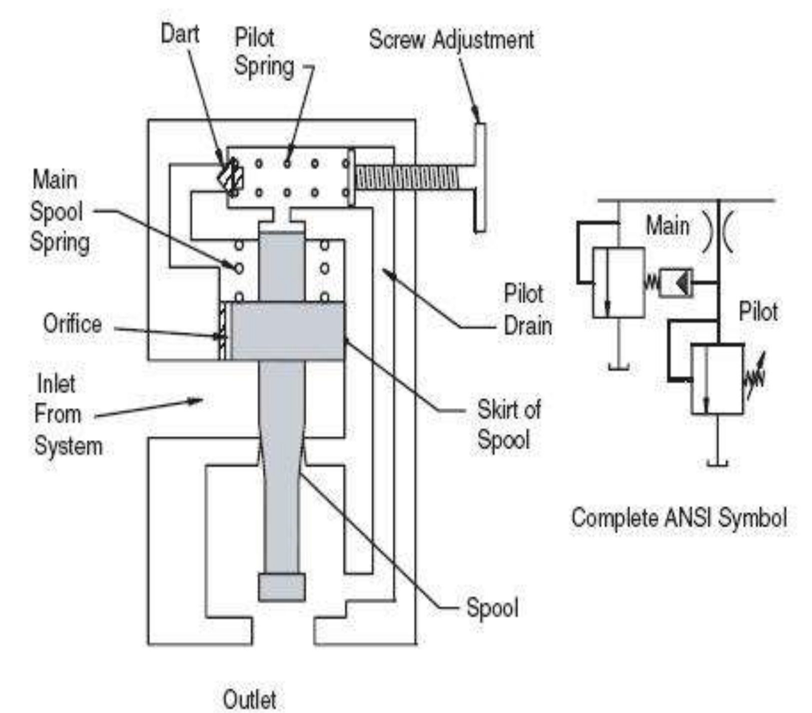
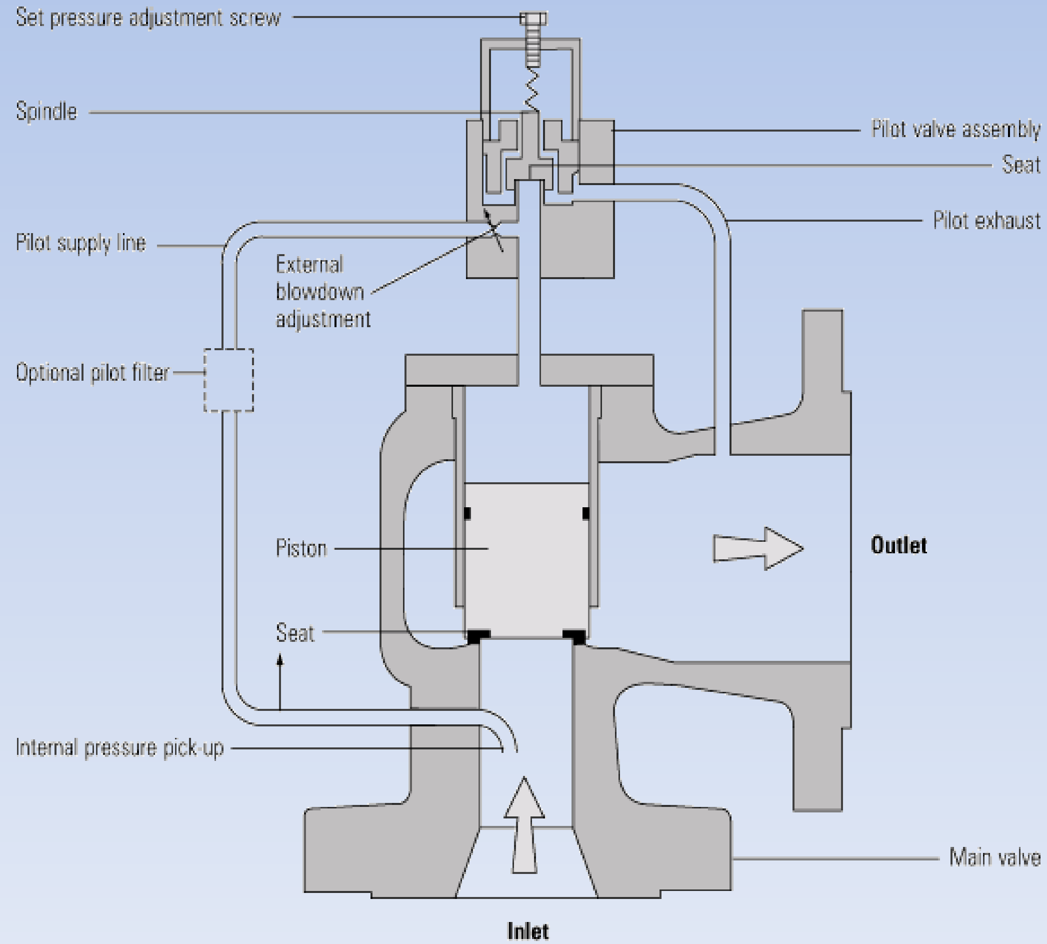
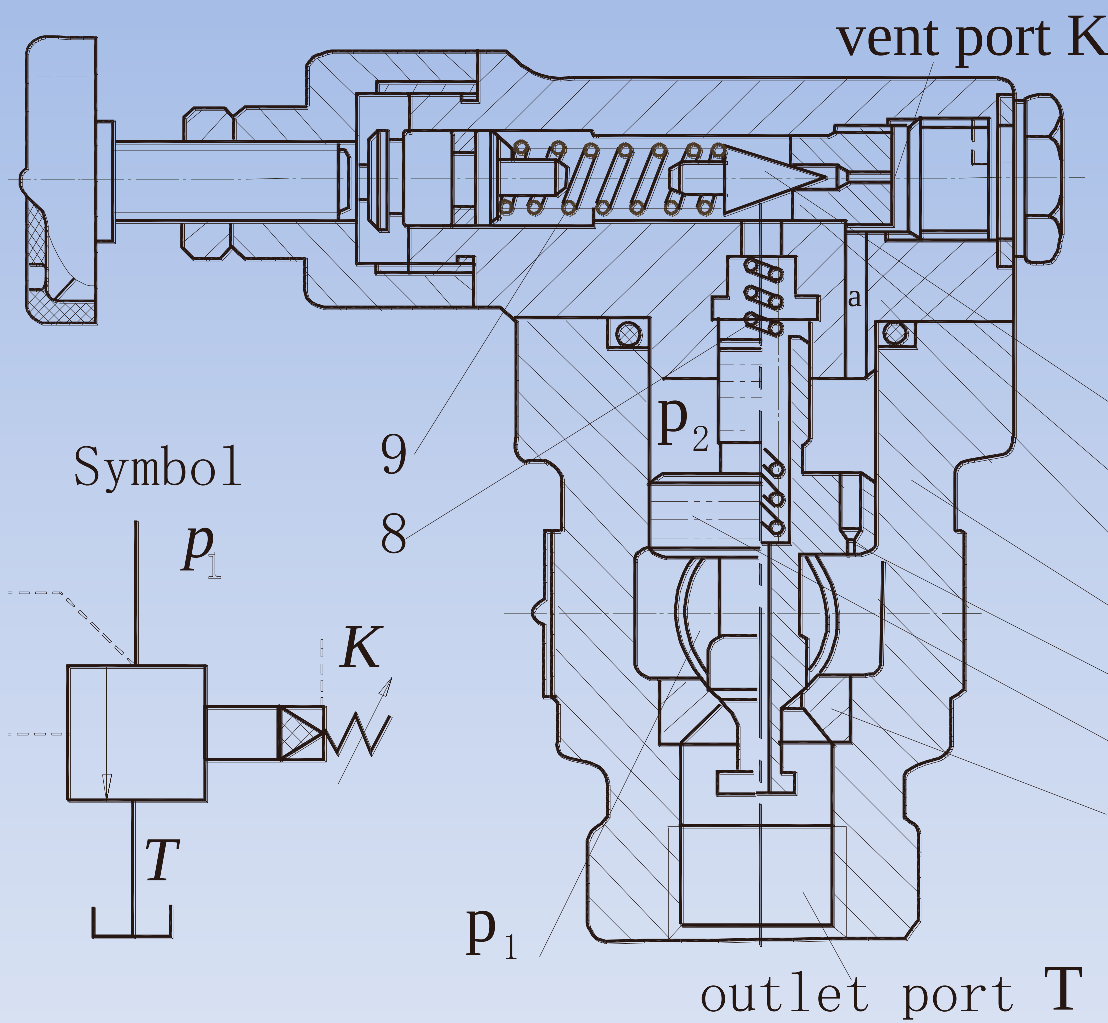


FIGURE 3.8
Functional diagram of pilot-operated relief valve. (Reprinted with permission from Parker Hannifin Corp.)



Compound Pressure Relief Valve



1 Schematic

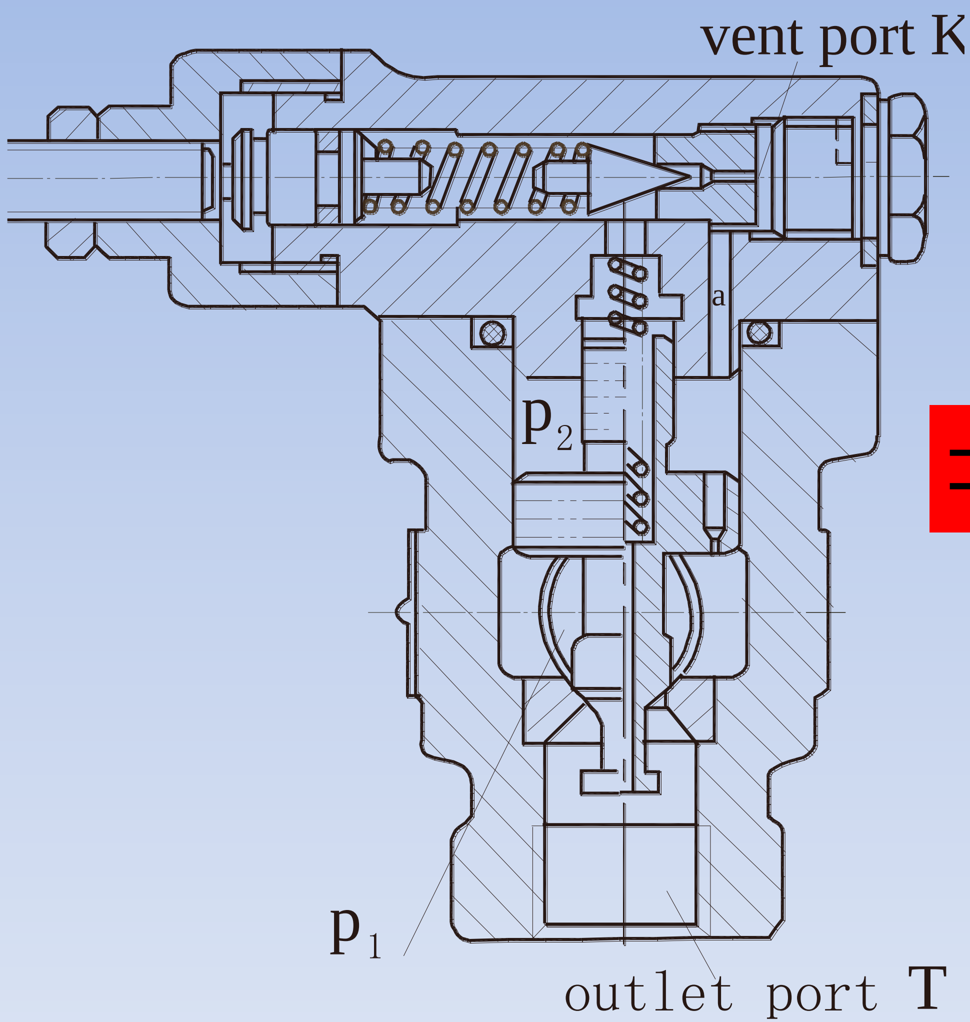
1-pilot valve;
2-poppet seat;
3-valve cap;
4-valve body;
5-orifice;
6-main spool;
7-main valve seat;
8-main spring;
9-adjustment spring

Port P_1 is connected to the pump line.

Port T is connected to the tank.

$$p_1 \xrightarrow{5} p_2 \xrightarrow{a} K \xrightarrow{central\text{–}hole} T,\ p_1 \rightarrow T$$

Compound Pressure Relief Valves



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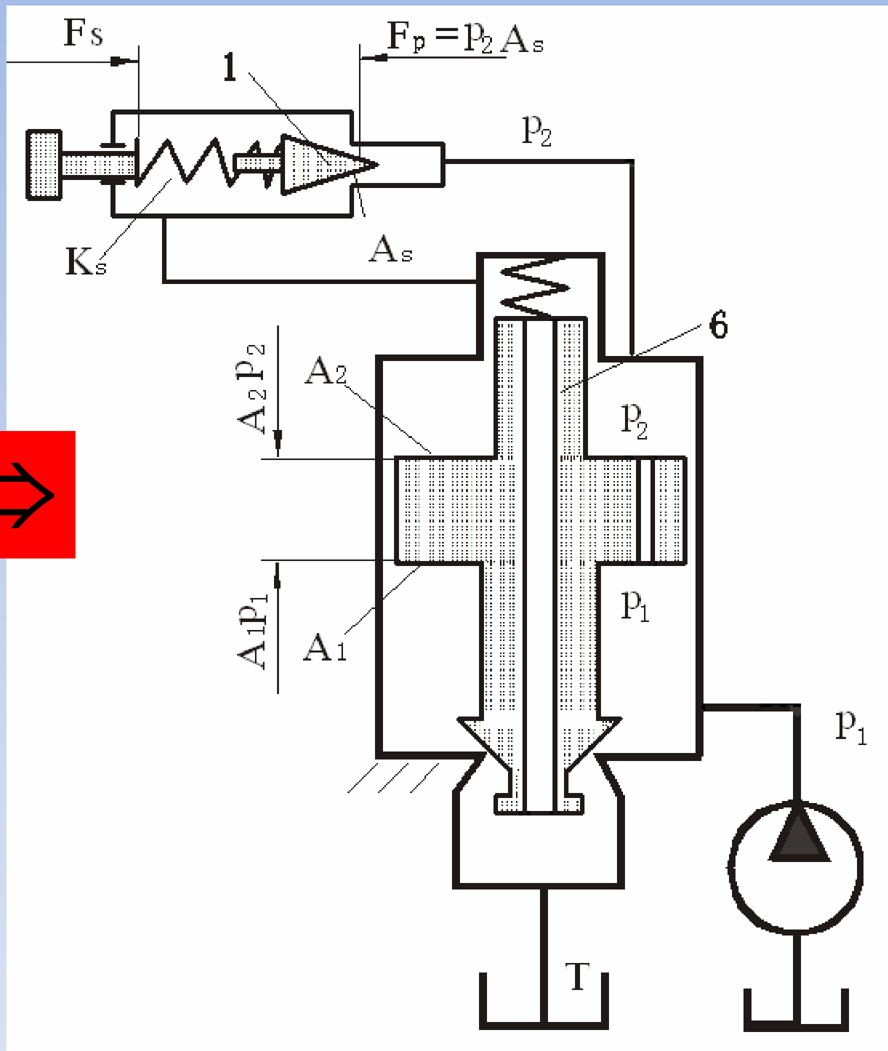
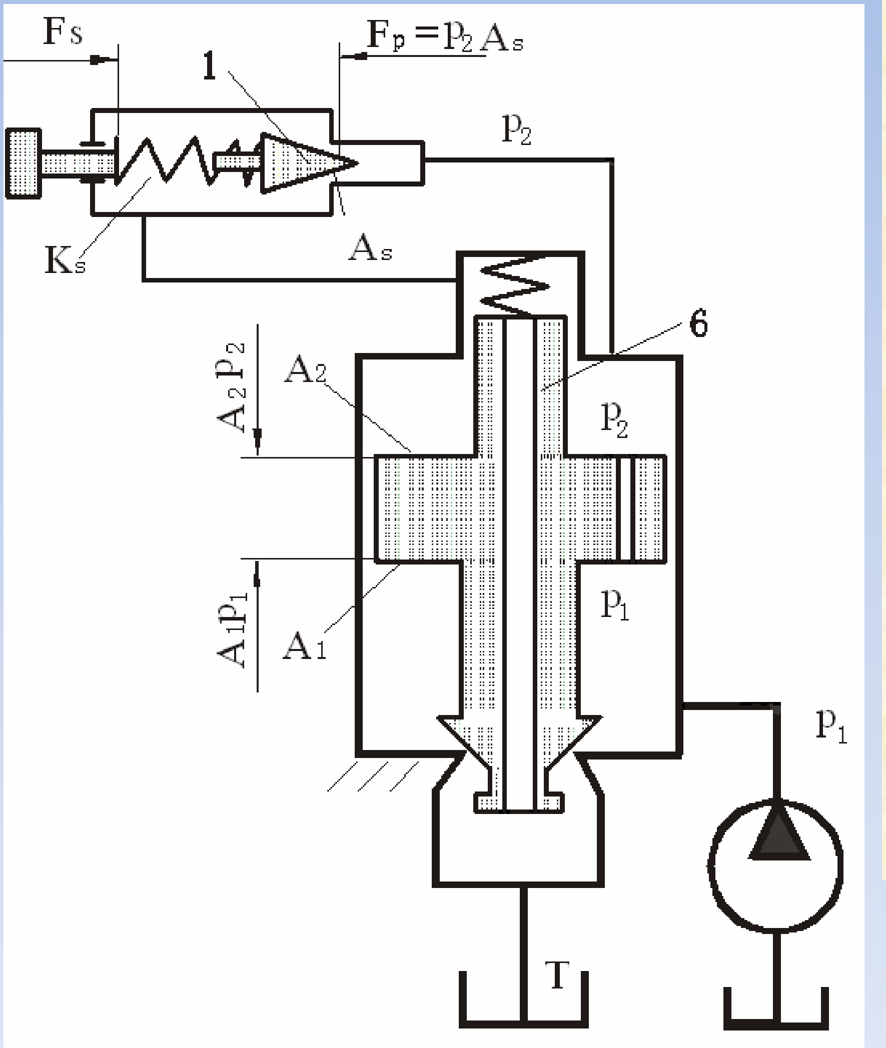


2 Operating process



As soon as the systematic pressure reaches the setting, it will force the pilot poppet off its seat.

A small amount of flow begins to go through the pilot line and the piston orifice back to tank.
(pilot valve opening)

When the systematic pressure further rises, the opening of the pilot valve further increases. As a result, the pressure drop of the camping orifice causes the piston to lift off its seat and the flow goes directly from the pressure port to the tank. (both valve opening)

Measure and control quantity: inlet pressure

3 Equilibrium equation

Neglecting the poppet weight, friction force, and flow force, the static force balance equation of the poppet is

Pilot valve: $$\sum F_x = F_s - pA = K_x(x_0 + x) - p_2 A_s = 0$$

$$p_2 = K_x(x_0 + x) / A_s$$

if the K_x is very small and $x_0 >> x$,

thus $p_2 \approx K_x x_0 / A_3$ **（constant）**

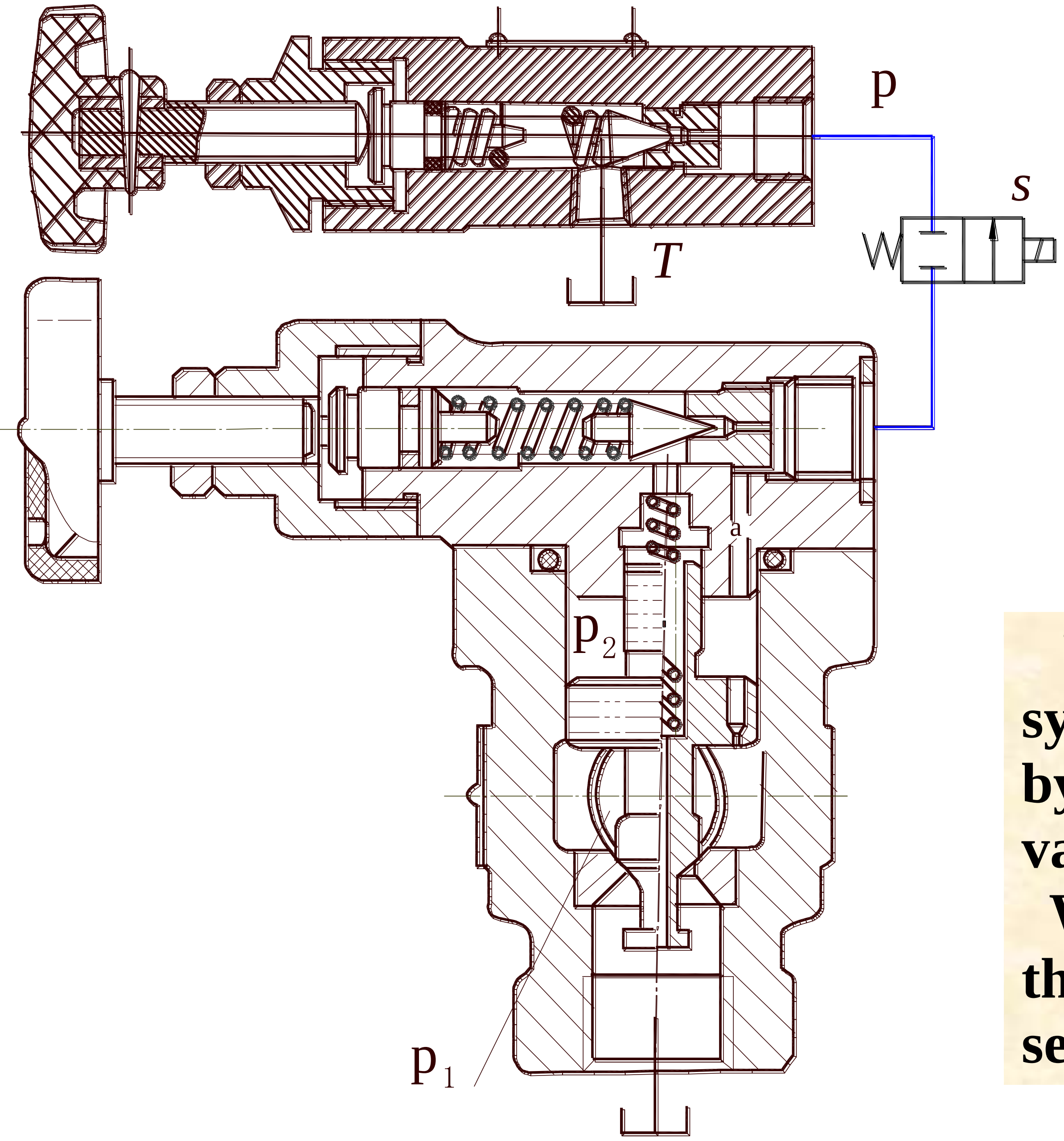
main valve: $\Sigma F_y = K_y(y_0 + y) - p_1 A_1 + p_2 A_2 = 0$

That is $p_1 = K_y (y_0 + y)/A_1 + p_2 A_2 / A_1$

if the K_y is very small and $y_0 >> y$,

thus $p_1 \approx K_y y_0 / A_1 + (K_x x_0 / A_s) A_2 / A_1$ **（constant）**

4 Remotely adjusting pressure



The pilot relief valve is set for the maximum pressure that the circuit is designed.

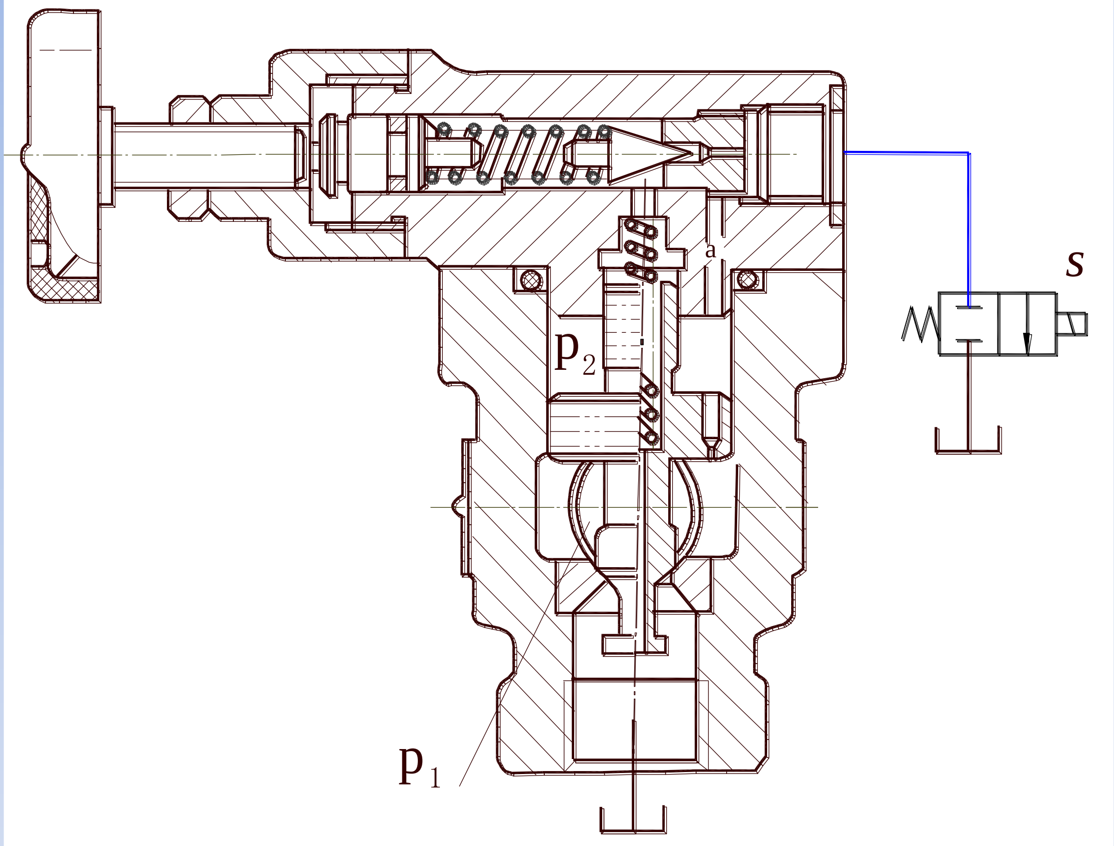
The remote relief valve is set to a lower pressure dictated by the current operating parameters.

When s is energized, the systematic pressure is set by the remote pilot relief valve

When s is de-energized, the systematic pressure is set by the pilot relief valve.

4 Unloading the pump

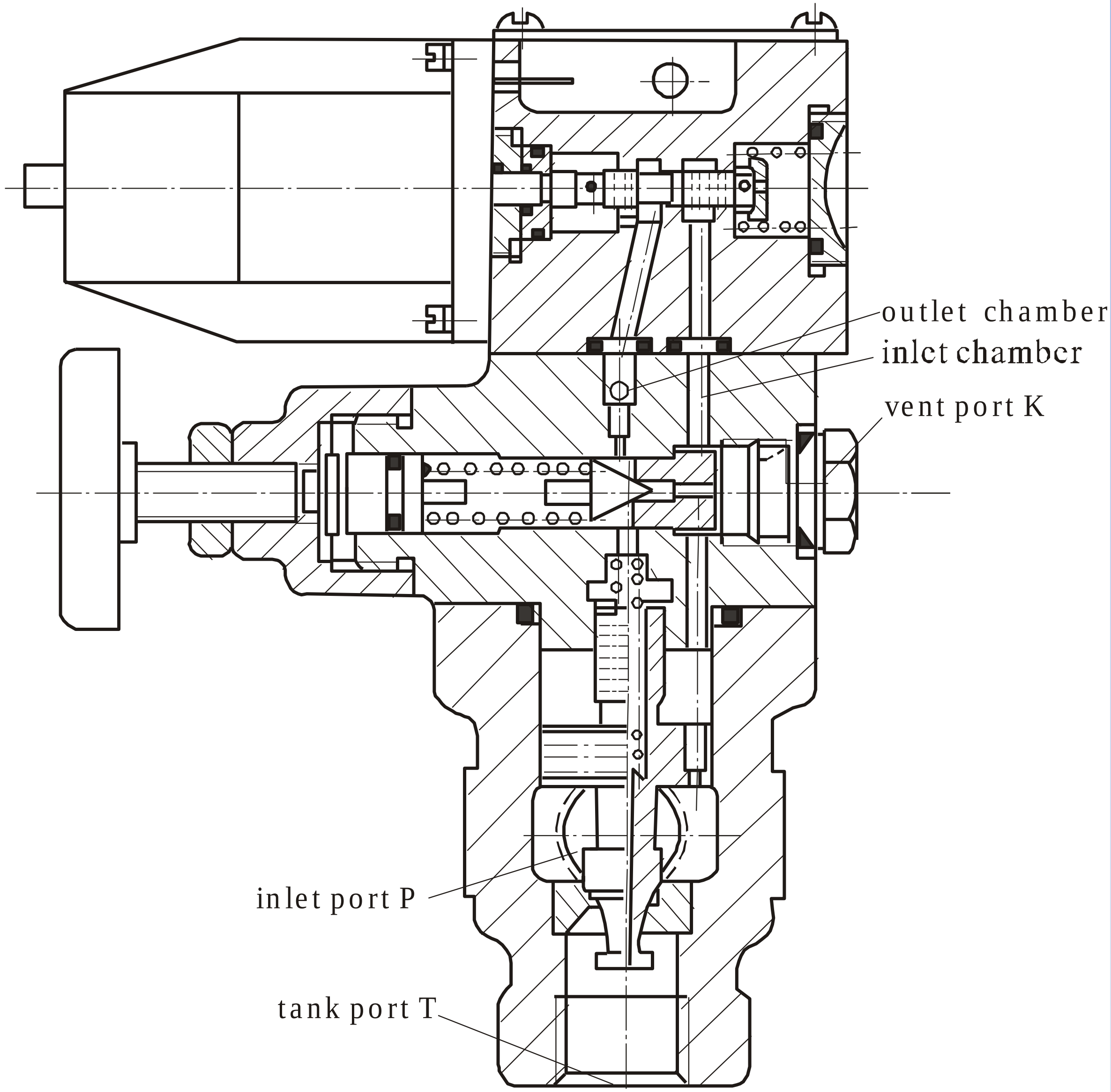
When the vent port is connected to tank via a solenoid directional control valve, the system will be able to be unloaded.
(only the main valve opening)



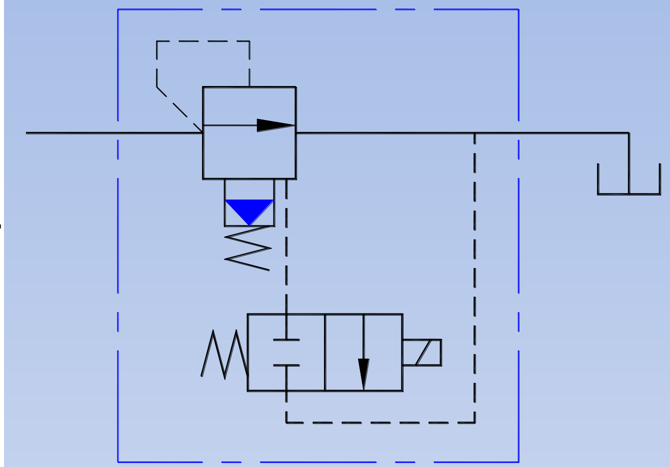
5 advantage

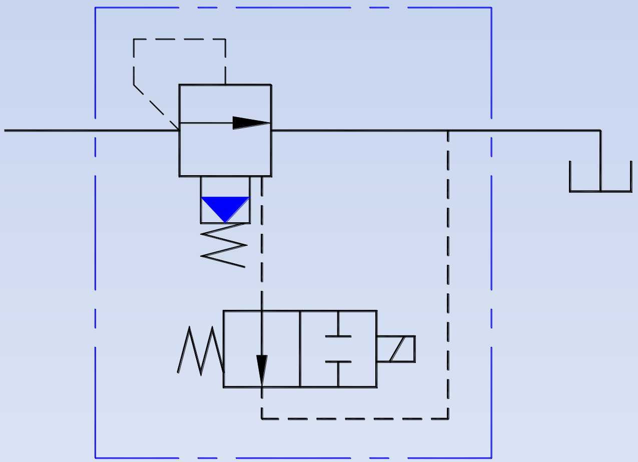
The pilot-operated relief valve usually is smaller than a direct-acting relief valve for the same flow and pressure ratings.

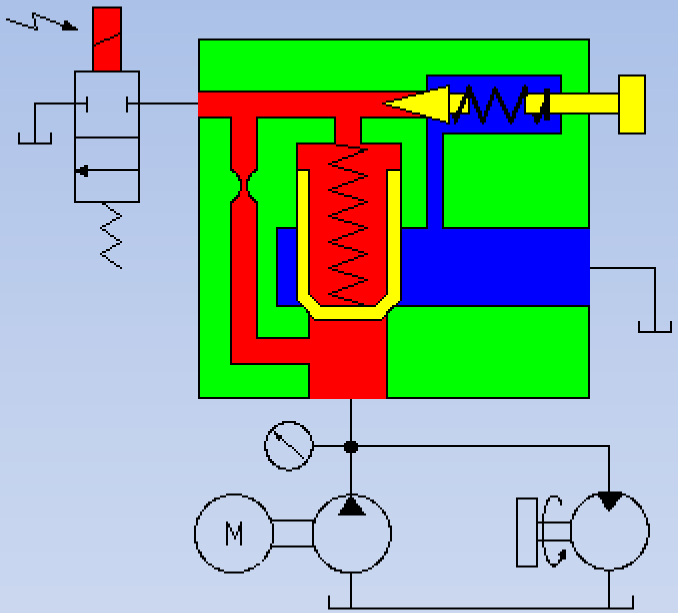
Solenoid pressure relief valves



(a)



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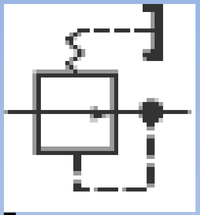
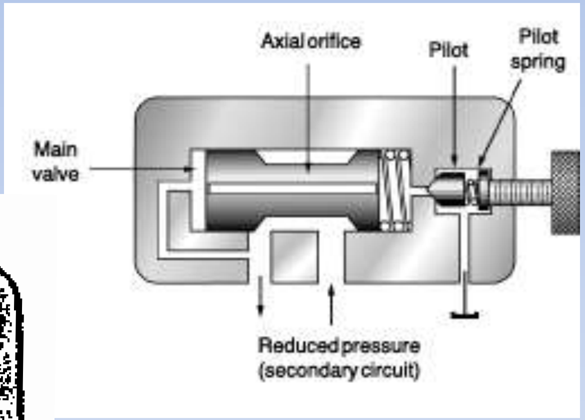
Pressure reducing valves



The pressure reducing valve maintains a reduced pressure level in a branch circuit of a hydraulic system.

Pressure reducing valve with pilot oil from outlet



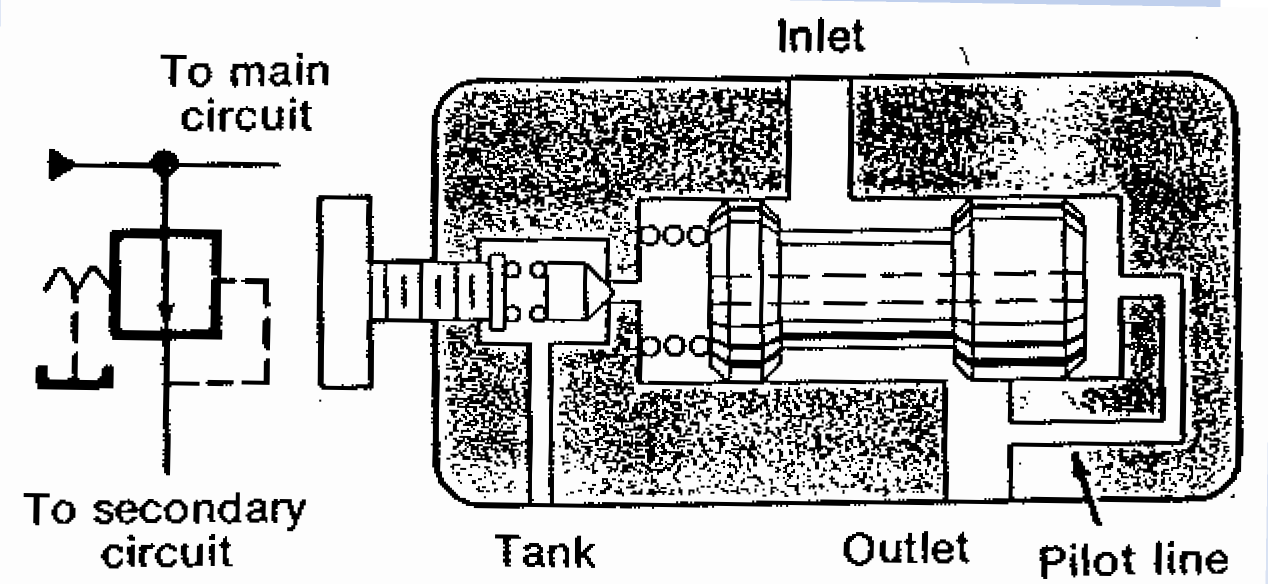
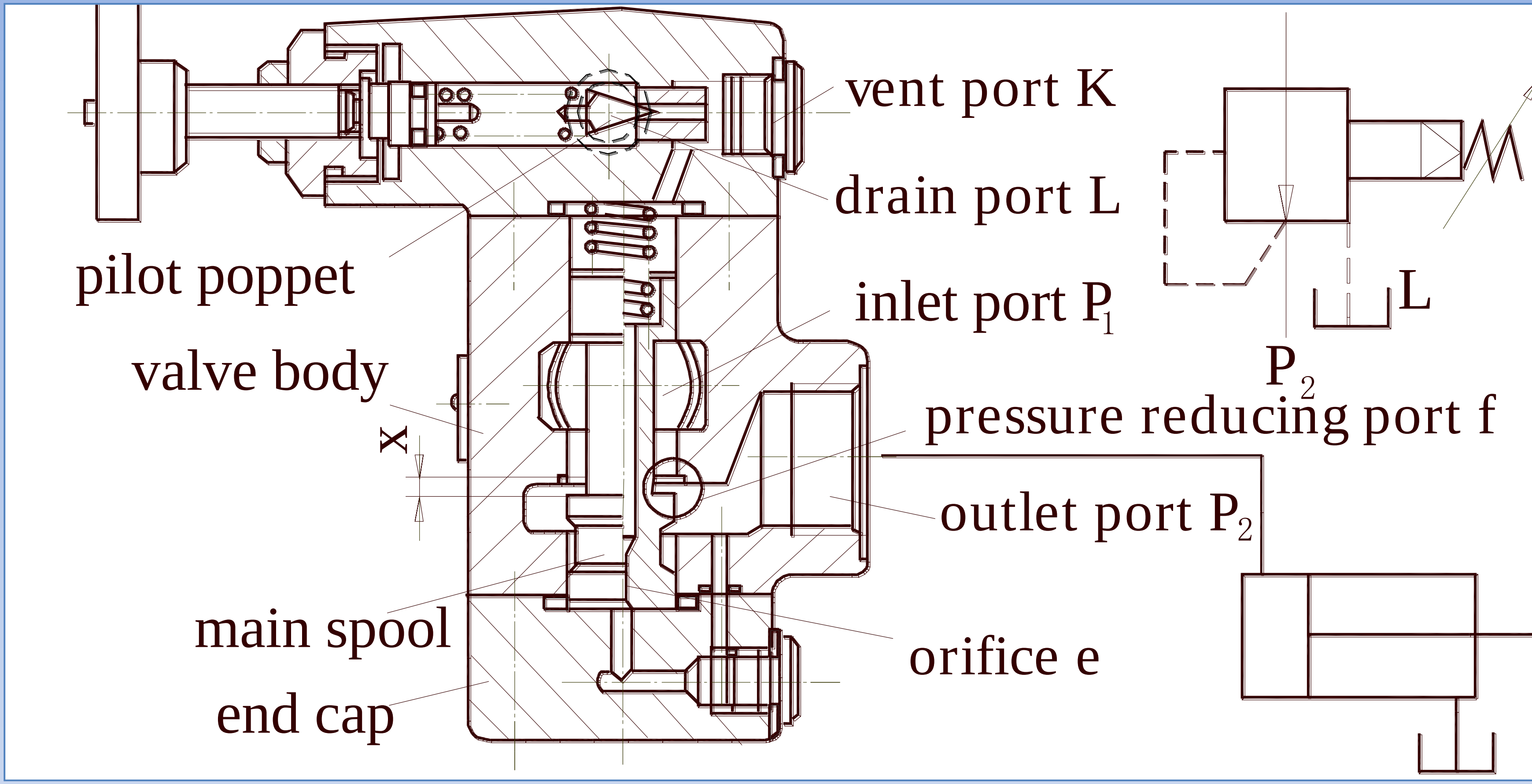


Fig. 2.9. Pressure reducing valve allows one branch of a circuit to operate at a lower pressure than the main system.



Pilot valve: $\Sigma F_x = F_s - \mathrm{p}A = -K_s(x_0+x) - \mathrm{p}_3 A_s = 0$

That is $\mathrm{p}_3 = K_s\,(x_0+x)/A_s$

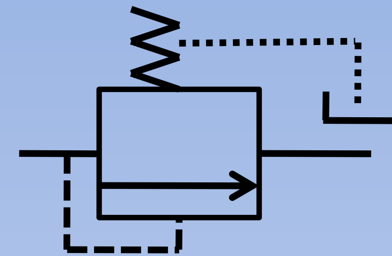
if the K_x is very small and $x_0 >> x$, $\mathrm{p}_3 \approx K_s x_0/A_s$（constant）

main valve $\Sigma F_y = K(y_0+y) - \mathrm{p}_2 A + \mathrm{p}_3 A = 0$

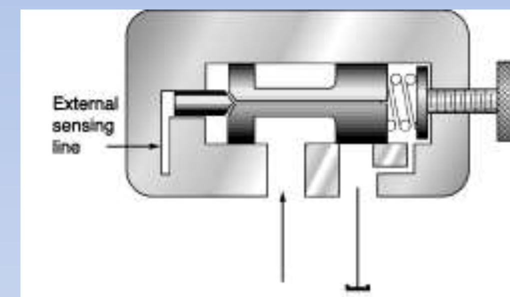
That is $\mathrm{p}_2 = K\,(y_0+y)/A + \mathrm{p}_3$

if the K is very small and $y_0 >> y$, $\mathrm{p}_2 \approx K y_0/A + K_s x_0/A_3$（constant）

Unloading Valve



- Direct the pump flow to the tank directly by applying a remote pilot pressure.



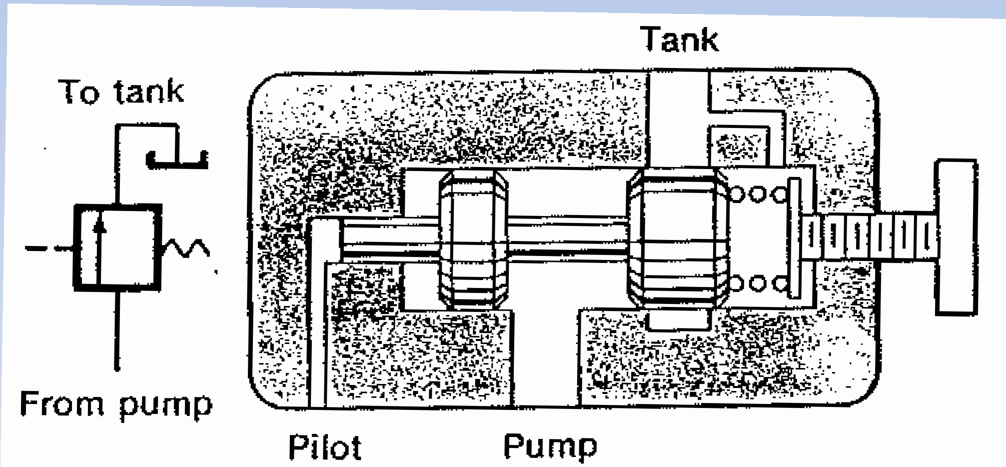
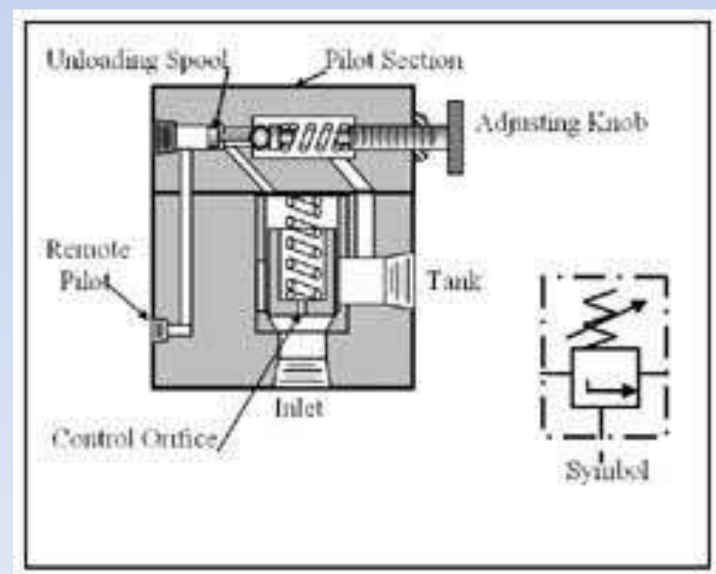


Fig. 2.7. Pressure unloading valve unloads pump output to tank at low pressure when high pressure flow is not required.



Example

- A pressure relief valve setting is 1000 psi, compute the horsepower loss across the valve if all the flow is returned to the tank at 20 gpm

$$HP = \frac{P \times Q}{1714} = \frac{1000(20)}{1714} = 11.7hp$$

- If an unloading valve is used to unload the pump and the unloading pressure is 25psi find the wasted hydraulic horsepower

$$HP = \frac{P \times Q}{1714} = \frac{25(20)}{1714} = 0.29hp$$

Sequence Valve

- Cause the hydraulic system to operate in a pressure sequence

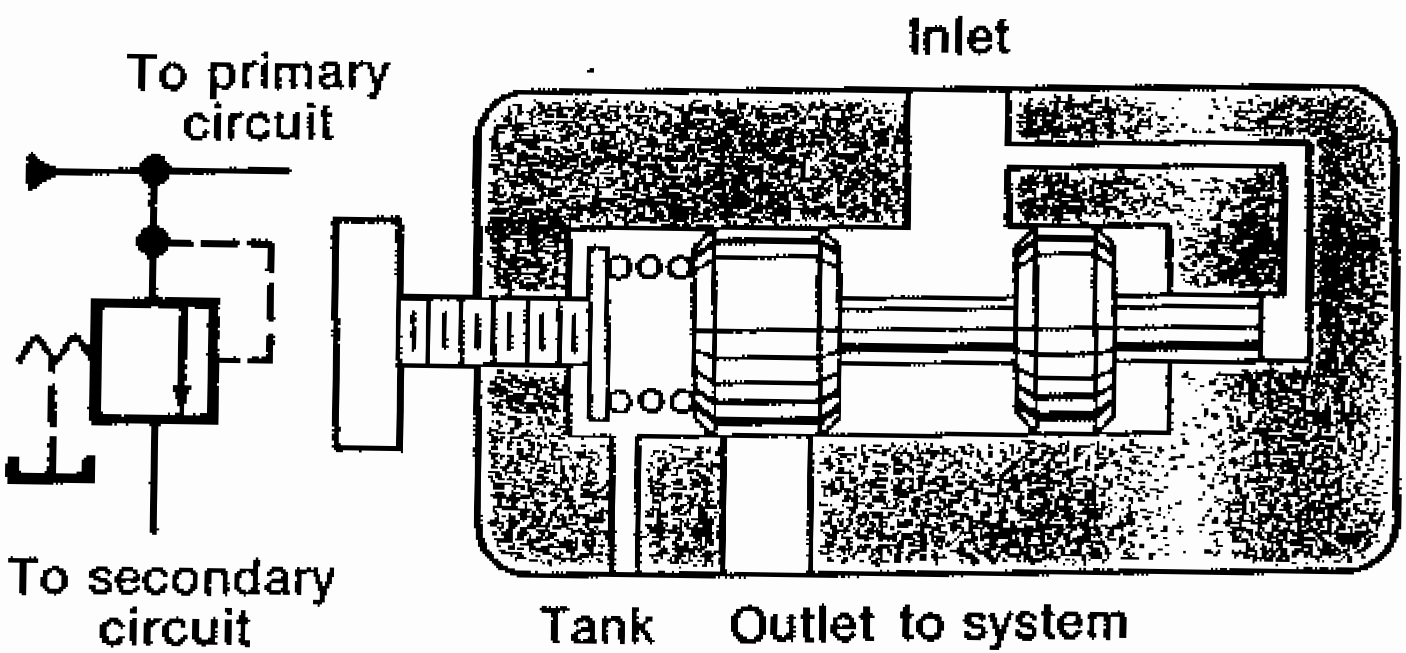


Fig. 2.8. Sequence valve prevents fluid from entering one branch of a circuit until a preset pressure is reached in the main circuit.

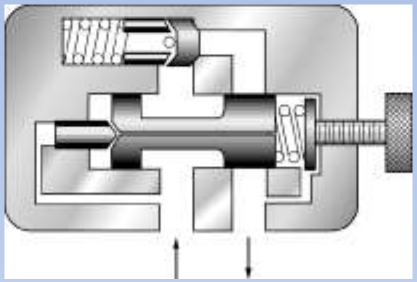
Counter balance valve

- Prevents the cylinder to retract

Due to its own weight

- The valve setting should be higher

than the weight



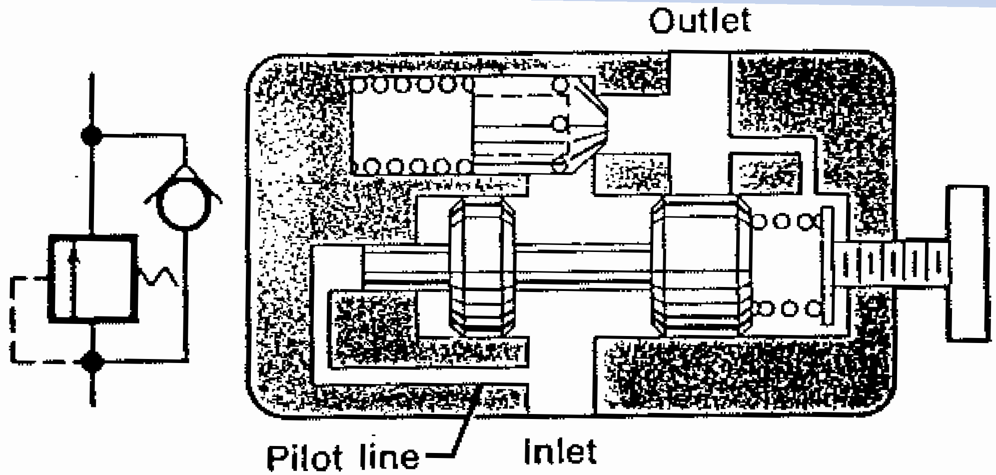
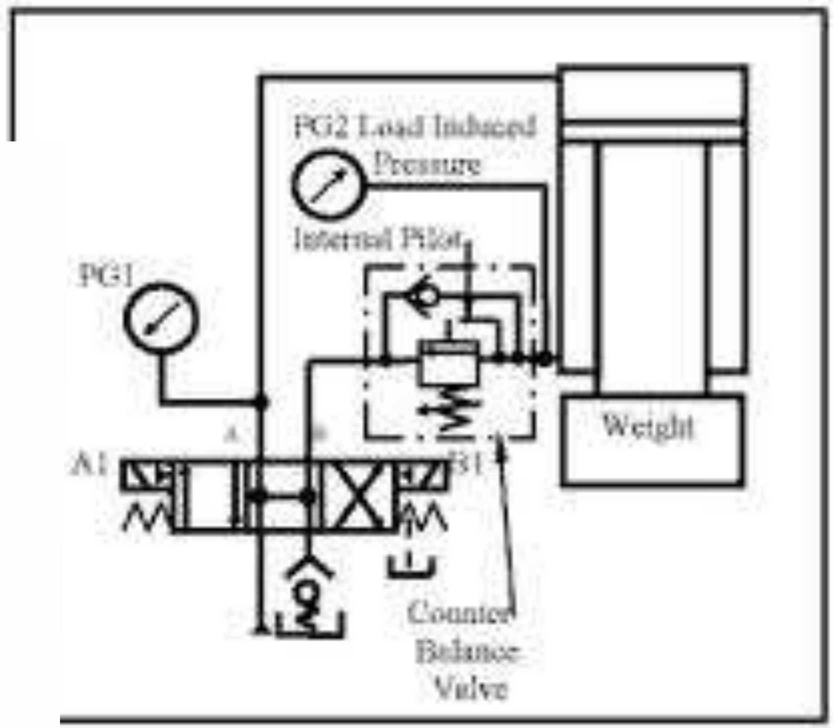


Fig. 2.10. Counterbalance valve holds fluid pressure in part of a circuit to counterbalance weight on external force.



Flow Control Valves

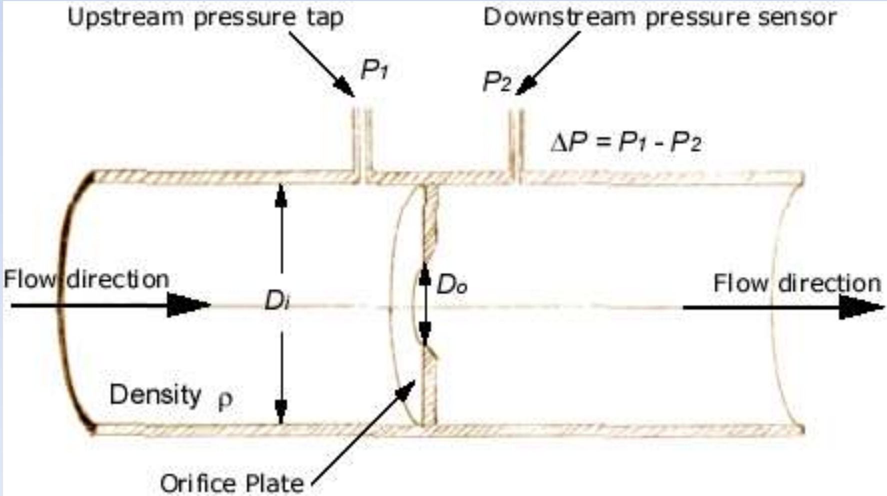
Orifice as a Flow Meter or Flow Control Device

- By measuring the pressure drop (ΔP), we can find the flow (Q)

$$Q = 38.1 CA \sqrt{\frac{\Delta P}{SG}} \text{ English Units, or}$$

$$Q = 0.0851 CA \sqrt{\frac{\Delta P}{SG}} \text{ Metric Units}$$

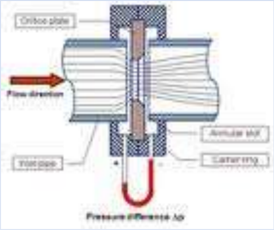
Q = flow rate (gpm,Lpm)
C = flow coefficient (C= 0.8 for sharp edge orifice , C=0.6 for square edge orifice
A = area of orifice opening (in^2,mm^2)
$\Delta P = P_1 - P_2$ = pressure drop across orifice (psi, kPa)
SG= specific gravity of flowing fluid



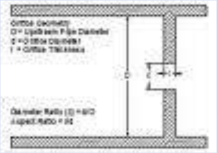
Example

- The pressure drop across a sharp edge orifice is 100psi the orifice is 1 in diameter, and the fluid has a SG=0.9. Find the flow rate
- Sol.

$$Q = 38.1CA\sqrt{\frac{\Delta P}{SG}} = 38.1(0.8)\left(\frac{\pi}{4}\times 1^2\right)\sqrt{\frac{100}{0.9}} = 252gpm$$

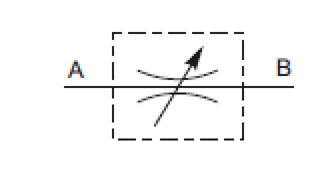


Sharp edge

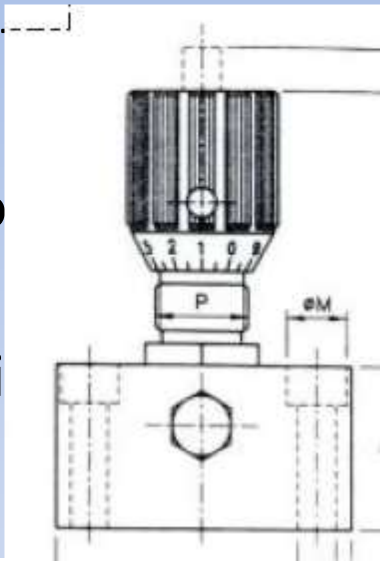


Square edge

Needle Valve



- Flow control valves are used to regulate the speed o
hydraulic actuators.
- Needle valves are designed to give fine control of flo
diameter piping.
- For a given position a needle valve behaves as an ori



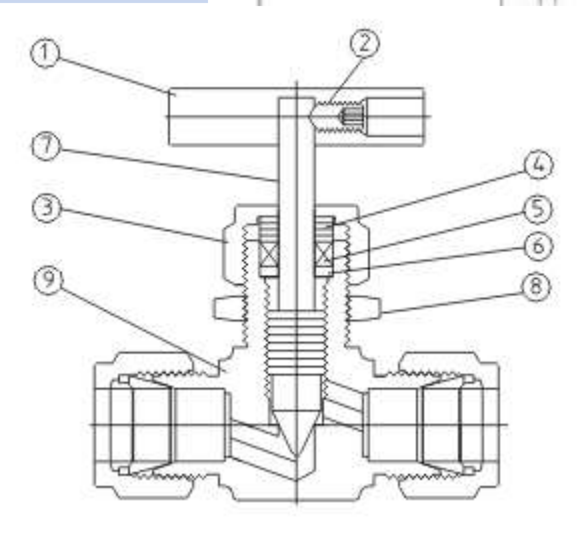
$$Q = C_v \sqrt{\frac{\Delta p}{SG}}$$

$Q =$ volume flow rate (gpm, Lpm)

$C_v =$ capacity orifice (gpm/$\sqrt{\text{psi}}$, Lpm/$\sqrt{\text{kPa}}$)

$\Delta p =$ pressure drop across the valve (psi, kPa)

SG = specific gravity of the liquid



C_v capacity coefficient is defined as the flow of water in gpm that wi
pressure drop of 1 psi or in Lpm at 1 kPa in metric which is determir
fully open position and listed by the manufacturer as rated C_v

Example

- A flow control valve experience a pressure drop of 100 psi (687 kPa) for a flow rate of 25 gpm (94.8 Lpm). The fluid is a hydraulic oil with SG =0.9. Find the capacity coefficient.

$$Q = C_v \sqrt{\frac{\Delta p}{SG}}$$

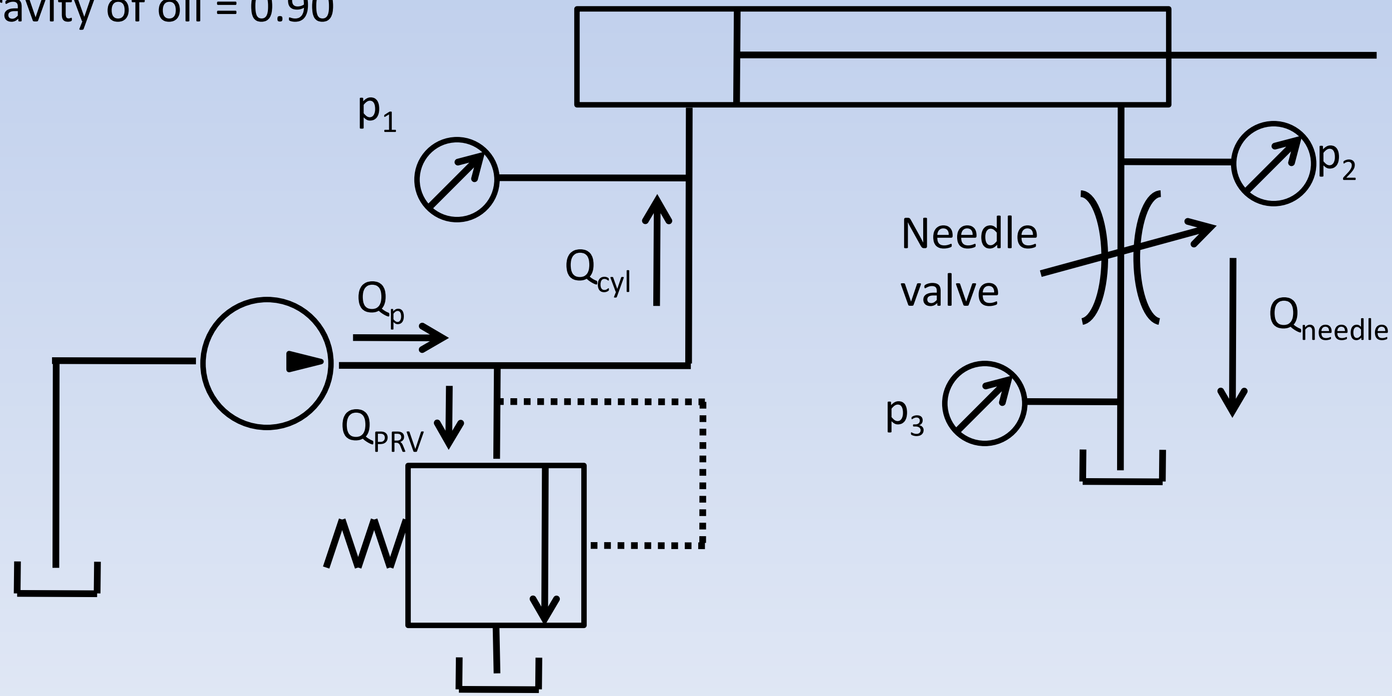
$$25 = C_v \sqrt{\frac{100}{0.90}} \Rightarrow C_v = 2.37\, gpm/\sqrt{psi} \quad \text{or} \quad 3.43\, \mathrm{Lpm}/\sqrt{\mathrm{kPa}}$$

Example

A needle valve is used to control the extension speed of a hydraulic cylinder. The valve is placed at the outlet of the cylinder as shown bellow.

Find the capacity coefficient of the needle valve if

1. The desired cylinder speed is = 10 in/s
2. Cylinder piston diameter = 2 in (area =3.14 in^2)
3. Cylinder rod diameter = 1 in (area = 0.79 in^2)
4. Cylinder load = 1000lb
5. Pressure relief valve (PRV)setting = 500 psi
6. Specific gravity of oil = 0.90



Ex. Cont.

- Sol.
- The piston speed is determined by the flow of the pump when the PRV is closed. By closing the Needle valve, p_2 increase, raising p_1 to the PRV setting and directing part of the flow to the tank.

$$p_1 = 500\,psi$$

$$p_1 A_1 - F_{load} = p_2 A_2$$

$$500\,psi \times 3.14 in^2 - 1000 lb = p_2 \times (3.14 - 0.79) in^2$$

$$p_2 = 243\,psi$$

The flow to the cylinder should produce a 10 in/s speed

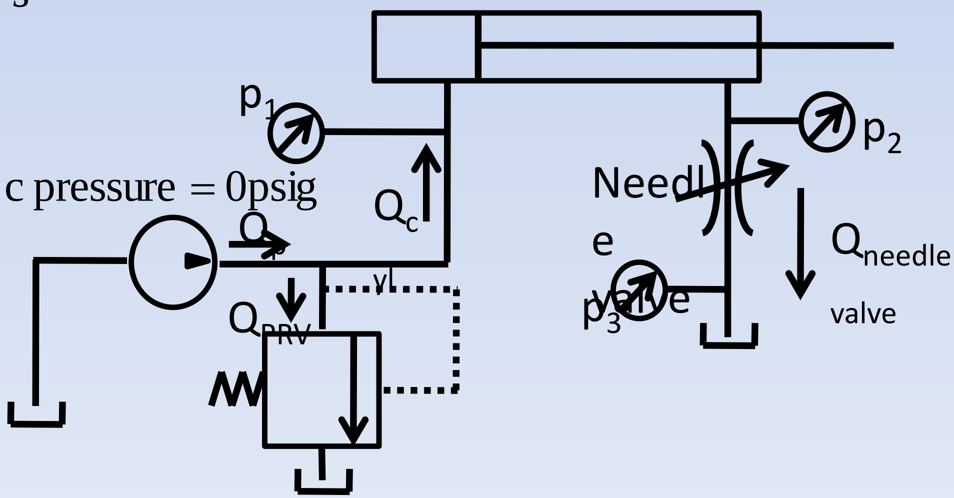
$$Q = A_2 v_{cyl} = (3.14 - 0.79) in^2 \times 10 in/s = 23.5 in^3/s$$

$$= 23.5 in^3/s \times \frac{1 gal}{231 in^3} \times \frac{60 s}{1 \min} = 6.1 gpm$$

The pressure after the valve p_3 is the atmospheric pressure = 0psig

The pressure drop across the valve is p_2

$$C_v = \frac{Q}{\sqrt{\Delta p / SG}} = \frac{6.1}{\sqrt{243/0.9}} = 0.37\,gpm/\sqrt{psi}$$

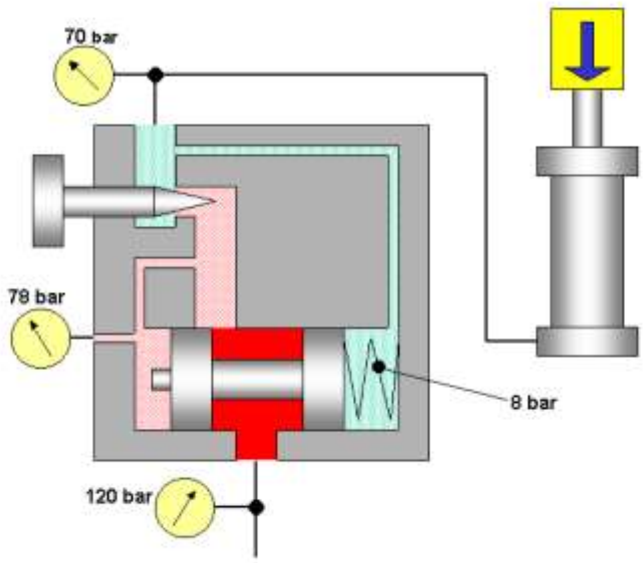


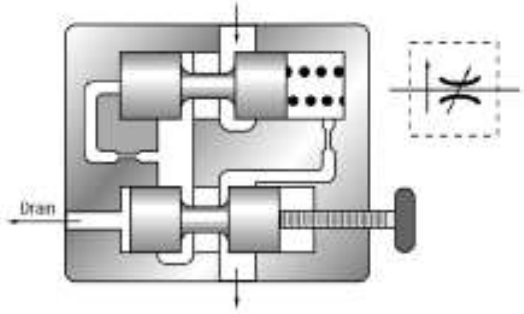
FLOW CONTROL VALVE, NON-PRESSURE COMPENSATED

- Is usually used when the system pressure is relatively constant and the motoring speed is not too critical.
- Assume constant flow and constant pressure drop.

FLOW CONTROL VALVE, PRESSURE COMPENSATED

- Maintains a constant pressure drop across the valve.
- The pressure drop setting can be modified by an external knob.





Servo Valves

- Servo valve is a directional valve that has infinitely variable positioning capability.
- It controls the direction and amount of flow.
- It is coupled with a feedback sensing device which allow for accurate control of position, velocity, and acceleration of an actuator.
-

Mechanical type servo valve

The sequential operation of which occurs as follows:

- The input or command signal is the turning of the steering wheel.
- This results in movement of the valve sleeve, which ports oil to the actuator (steering cylinder).
- The piston rod moves the wheels through the steering linkage.
- The valve spool is attached to the linkage, thereby moving it.

When the valve spool has moved far enough, it cuts off the oil flow through the cylinder. This stops the motion of the actuator.

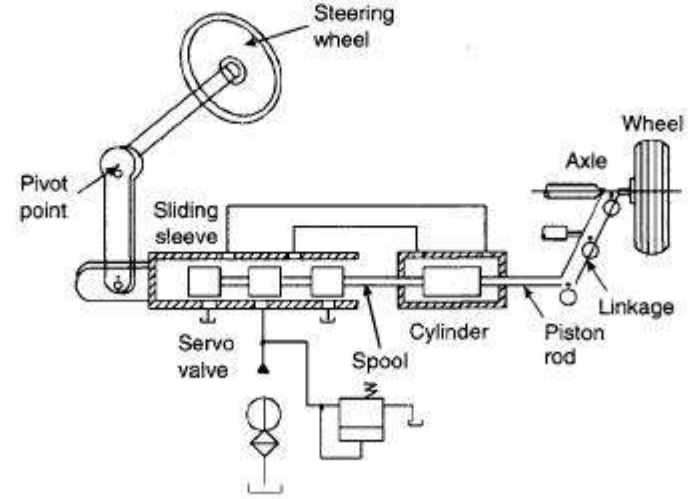


Figure 10.10
Mechanical hydraulic servo system

Electrohydraulic Servo Valve

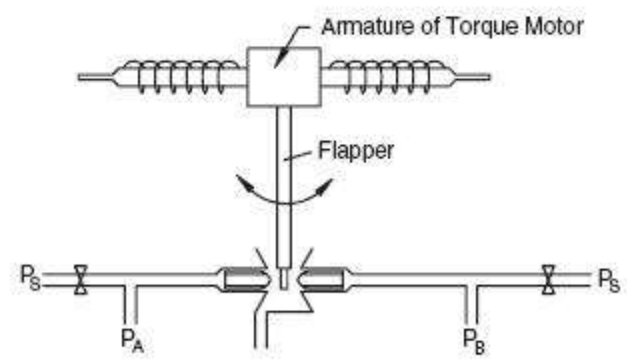


FIGURE 11.10
Diagram of double flapper nozzle as a first stage for a two-stage servo valve (second stage not shown). Reprinted with permission from Electrohydraulic Servo Systems, James E. Johnson, Penton Media, Inc., Cleveland, Ohio.

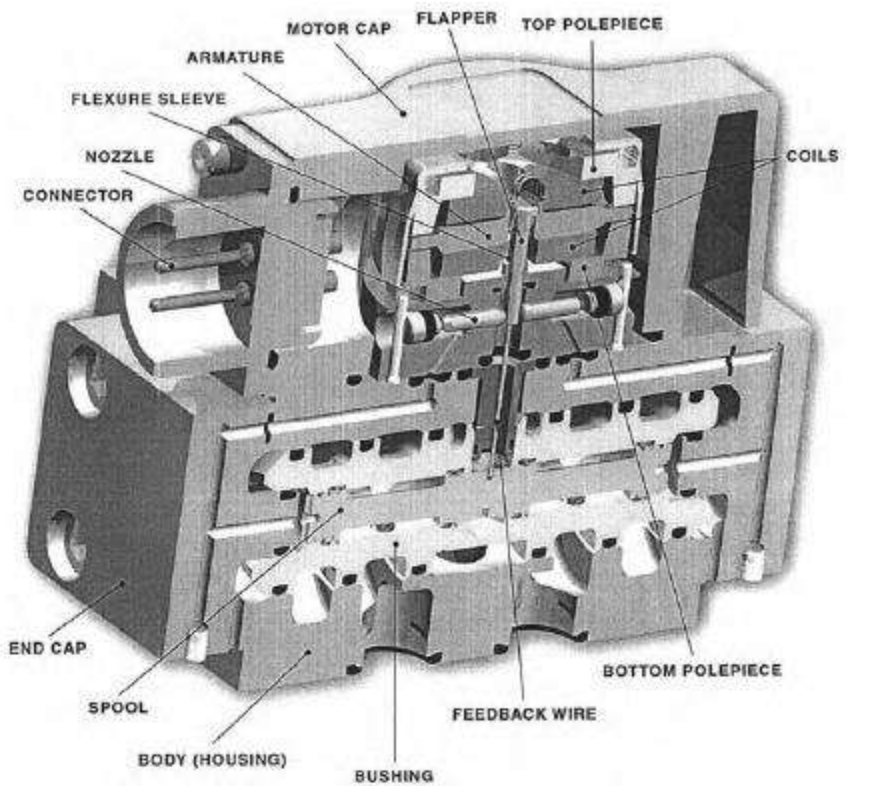
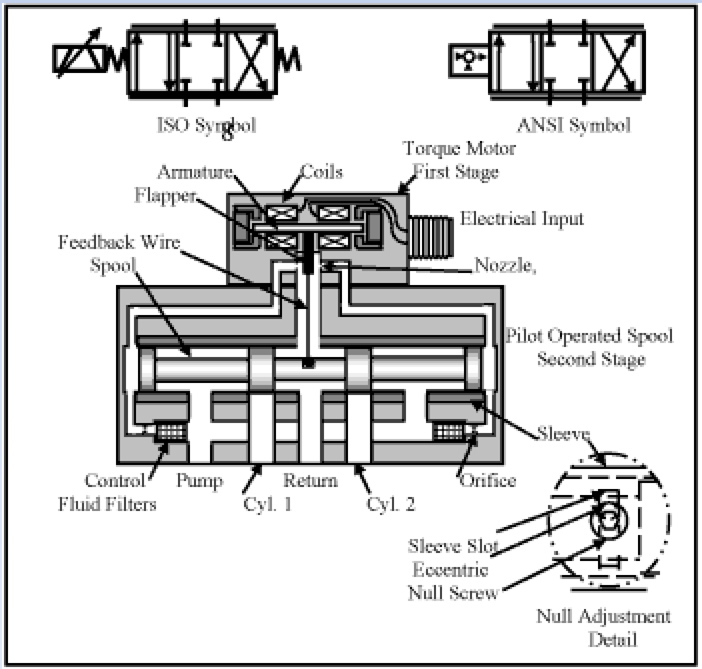


FIGURE 11.11
Cutaway of two-stage servo valve with double flapper nozzle for a first stage. Courtesy of Moog Inc.

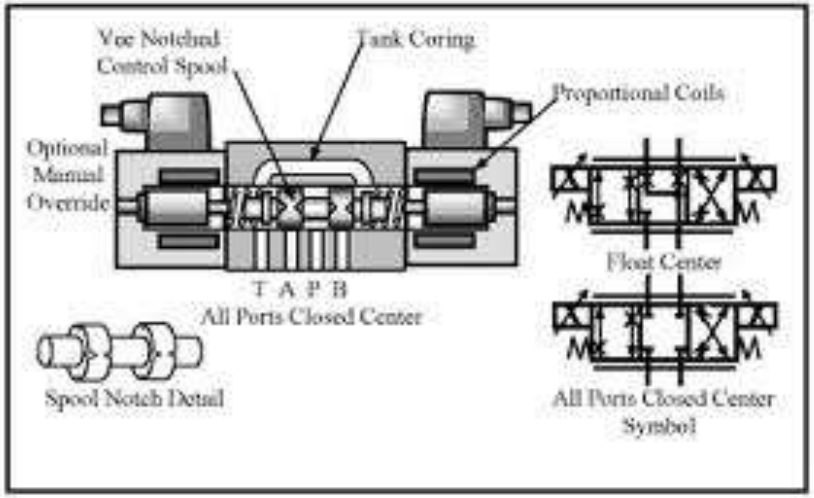


Electrohydraulic Servo Valve

Supply pressure is supplied to the points identified with Ps. Fluid flows across the fixed orifices and enters the center manifold. Orifices are formed on each side between the flapper and the opposing nozzles. As long as the flapper is centered, the orifice is the same on both sides and the pressure drop to the return is the same. Pressure at A equals the pressure at B, and the spool is in force balance. Suppose the torque motor rotates the flapper clockwise. Now, the orifice on the left is smaller than the orifice on the right, and the pressure at A will be greater than the pressure at B. This pressure difference shifts the spool to the right. As the spool shifts, it deflects a feedback spring. The spool continues to move until the spring force produces a torque that equals the electromagnetic torque produced by the current flowing through the coil around the armature. At this point, the armature is moved back to the center position, the flapper is centered, the pressure becomes equal at A and B, and the spool stops. The spool stays in this position until the current through the coil changes. Because of the feedback spring, the spool has a unique position corresponding to each current through the coil ranging from 0 to rated current. At rated current, the spool is shifted to its full open position.

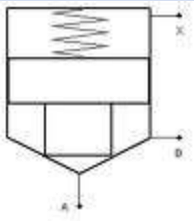
Proportional Control Valves

- Used to move actuators in a precise speed.
- produce a very accurate force to perform the work for which they were designed.
- Physically, proportional valves appear the same as their on/off solenoid counterparts. The big difference is in the way their solenoid coils perform. Proportional coils operate on DC current and produce varying force with varying voltage.



Cartridge Valves

- Cartridge valves can be a pressure, directional, and flow control valves that screw into a threaded cavity.
- Screw-in cartridge or Slip-in cartridge valves
- Advantages :Reduce number of fittings, reduce oil leakage, lower system installation and cost, reduce service time, and smaller space.




Hydraulic Fuses

- Direct flow to tank when the maximum pressure is exceeded.
- Used as safety valve with pressure compensated variable displacement pumps.
- If it opens it has to be replaced for reuse.
- A metallic membrane will rupture to allow flow.

Chapter 6 Pressure Control Valves

6.4 Sequence valves

Sequence valves cause a hydraulic system to operate in a pressure sequence.

They are used to control the order of various actuators of a hydraulic system.

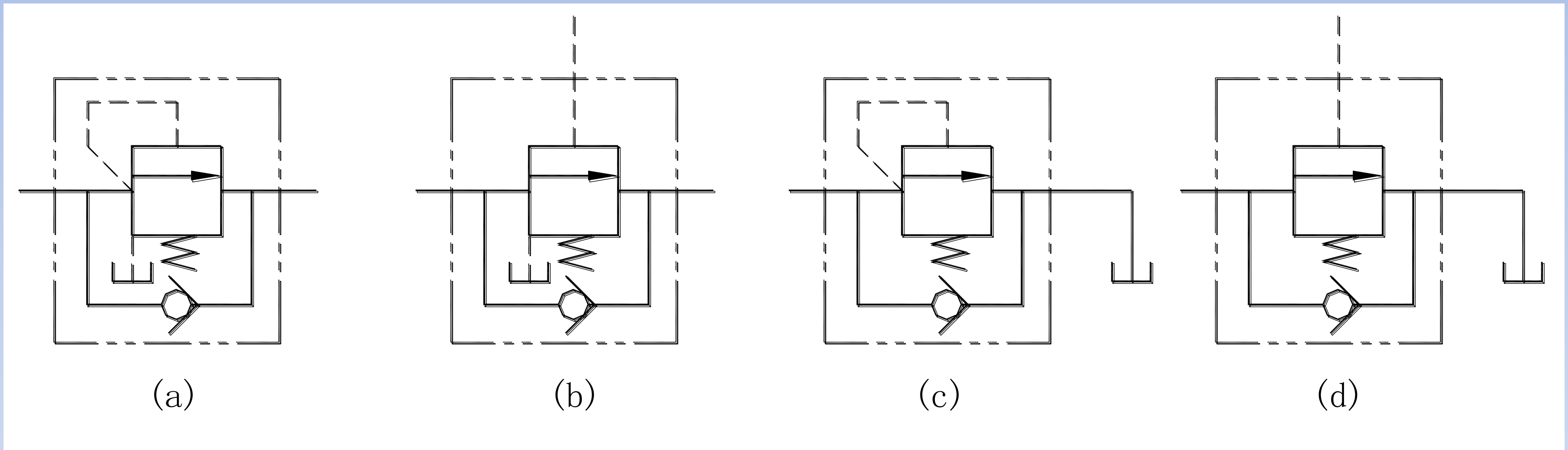
As soon as the inlet pressure reaches a preset pressure value, the sequence valve will open and let oil pass to a secondary circuit.

Sequence valves have two types—direct-acting and pilot-operated.

They can also be classified as internal control and external control types according to where the control pressure is from.

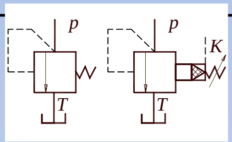
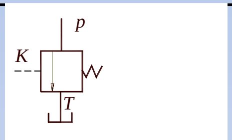
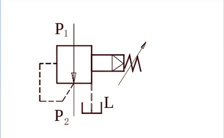
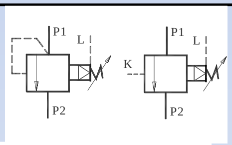
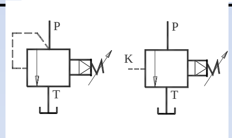
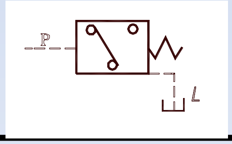
Chapter 6 Pressure Control Valves

One-way sequence valve and counterbalance valve



(a) One-way internal control sequence valve
(b) One-way external control sequence valve
(c) One-way internal control counterbalance valve
(d) One-way external control counterbalance valve

Summary

No.	Pressure valve	Measure and control quantity	Outlet port to	leakage	symbol
1	Pressure relief	Inlet pressure	tank	to have no drain port	
2	Unloading valve	Pilot pressure	tank	to have no drain port	
3	Pressure reducing valve	Outlet pressure	actuator	to have a drain port	
4	Sequence valve	Inlet/pilot pressure	actuator	to have a drain port	
5	Counterba lance	Inlet/pilot pressure	tank	to have no drain port	
6	Pressure switch	Inlet port pressure			

Thanks

Hydraulic Circuit Design and Analysis

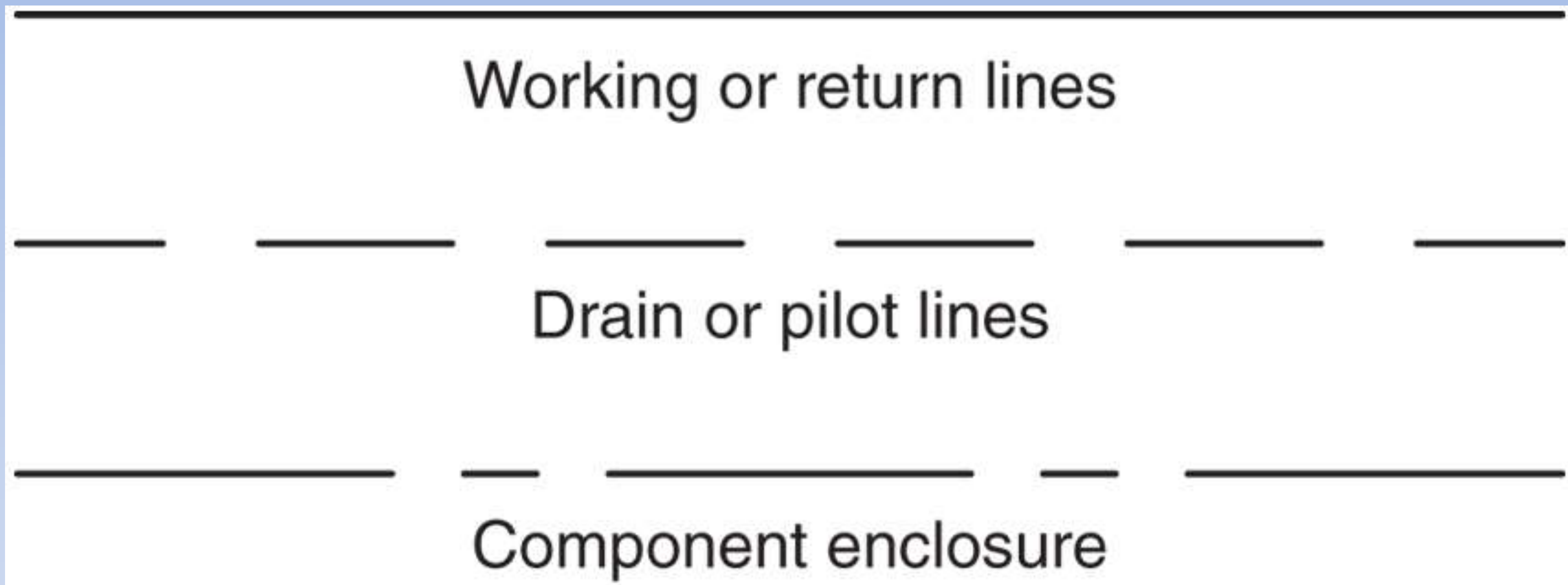
Chapter 9

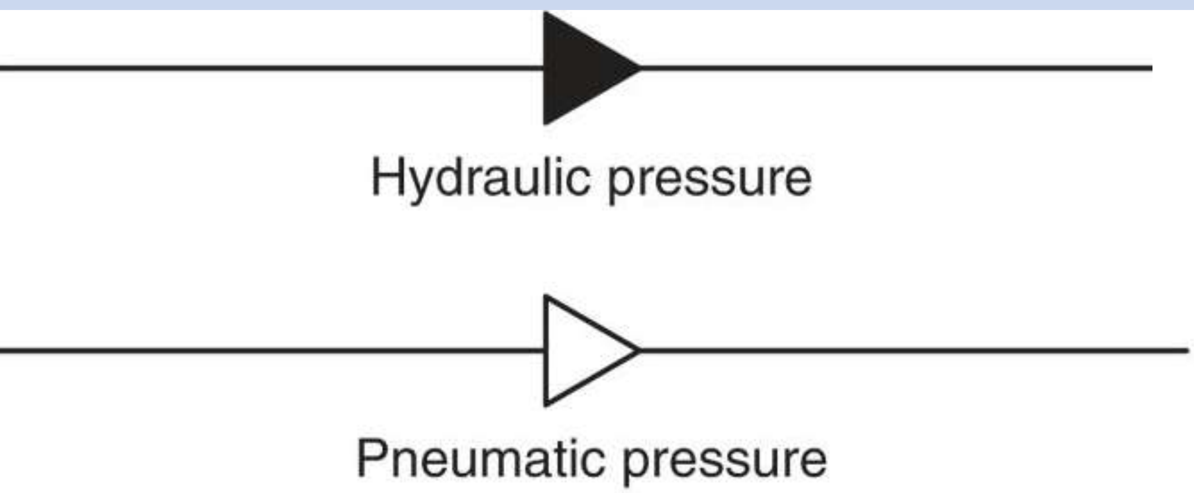
Dr. Suleiman BaniHani

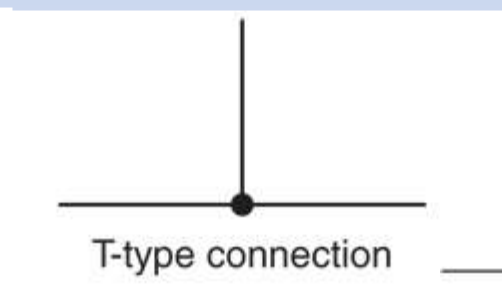
Hydraulic Circuits

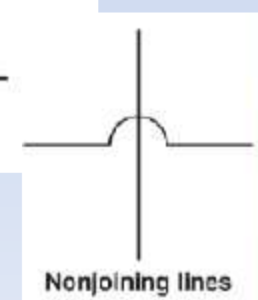
- A hydraulic circuit is a group of components such as pumps, actuators, control valves, and conductors arranged so that they perform a useful task.
- A hydraulic circuit has three basic considerations
  - Safety of operation
  - Performance of desired function
  - Efficiency of operation

Fluid Power Symbols and Circuit Diagrams

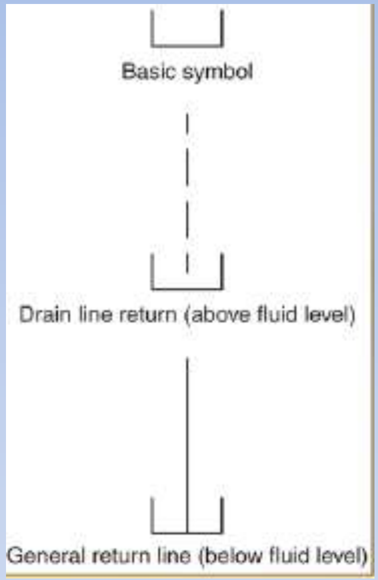


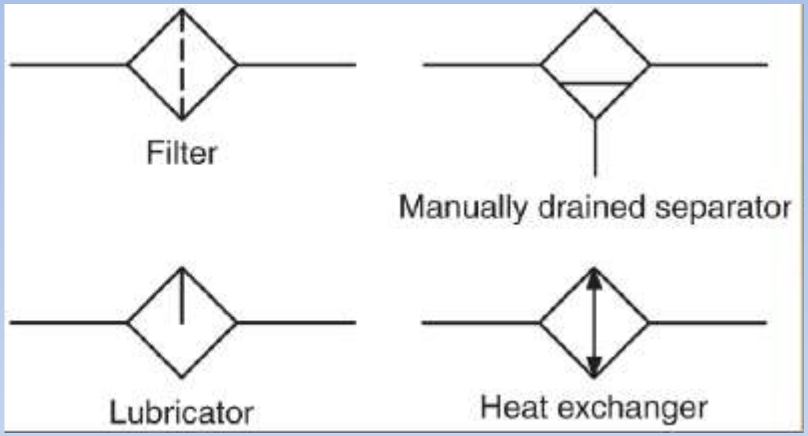


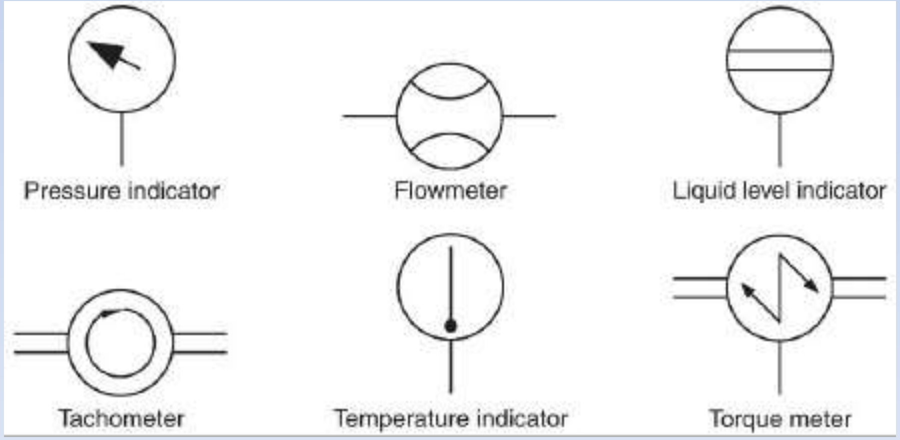




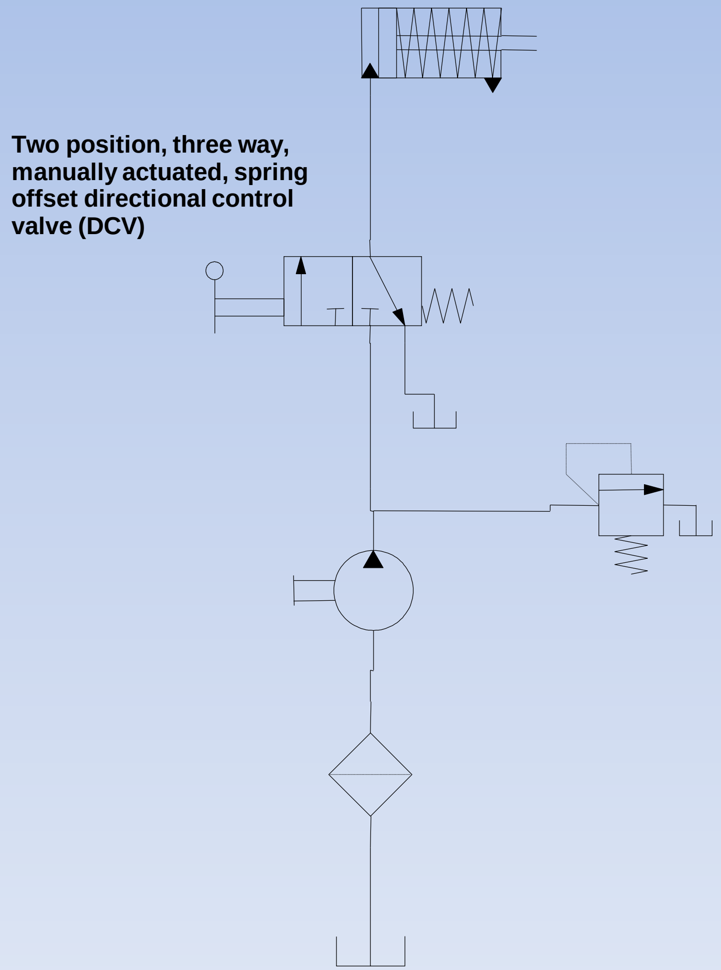
Fluid Power Symbols and Circuit Diagrams





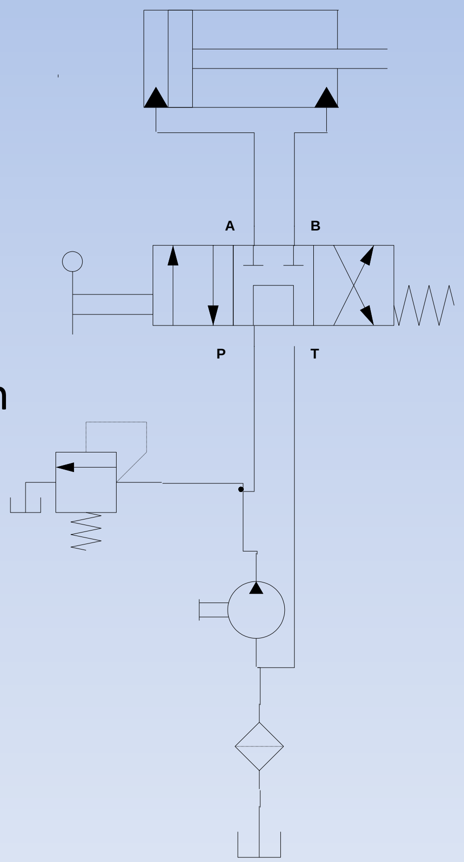


Control of Single Acting Hydraulic Cylinder



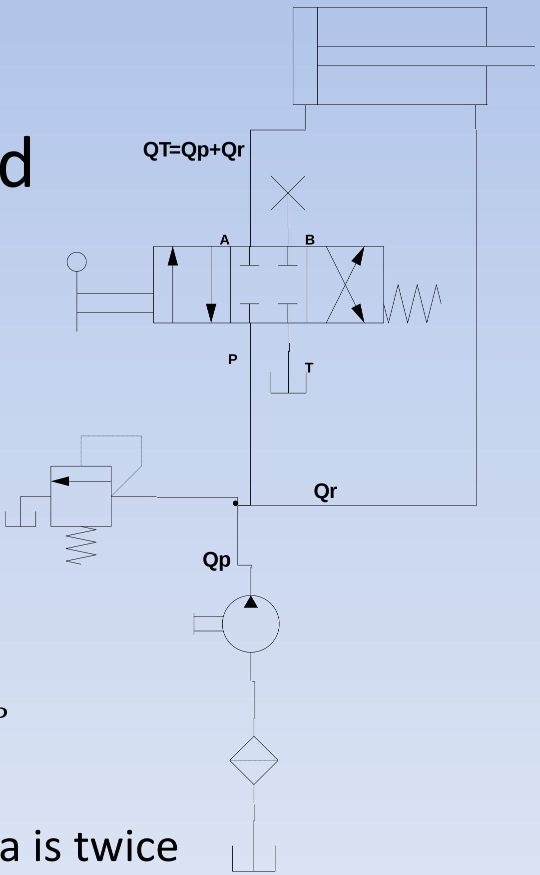
Control of Double Acting Hydraulic Cylinder

Extension Speed is lower that retraction speed. However extension force is greater



Regenerative Cylinder Circuit

- Used to speed up extension of a double acting hydraulic cylinder
- In extension the flow from the rod end regenerates with the pump flow, increasing the flow going into the cylinder.
- Cylinder extension speed



$$Q_T = Q_P + Q_R$$

$$A_P v_{ext} = Q_P + (A_P - A_r) v_{ext} \Rightarrow A_P v_{ext} - A_P v_{ext} + v_{ext} A_r = Q_P$$

$$v_{ext} = \frac{Q_P}{A_r}, \text{On the other hand}$$

$$v_{ret} = \frac{Q_P}{A_p - A_r} \Rightarrow \frac{v_{ext}}{v_{ret}} = \frac{A_p - A_r}{A_r} = \frac{A_p}{A_r} - 1$$

If the piston area is twice the rod area, then both speeds will be equal

Load Carrying Capacity of Regenerative Circuit

- The load carrying capacity of regenerative cylinder during extension is less than that obtained with regular double acting cylinder.

$$\sum F = 0$$

$$PA_p - P\left(A_p - A_r\right) = F_{load} \Rightarrow F_{load} = PA_r$$

Example

- A double acting cylinder is connected in regenerative circuit. The cracking pressure of the pressure relief valve is 1000psi, the piston area is 25 in^2 and the rod area is 7 in^2 , the pump flow rate is 20 gpm. Find the cylinder speed, load carrying capacity and power delivered to the load (assuming the load equals the load carrying capacity)during
  a. Extension Stroke
  b. Retraction Stroke

Solution

a.

$$v_{ext} = Q_p \big/ A_r = \frac{20gpm\left(231in^3/1gal\right)\left(1\min/\ 60s\right)}{7in^2} = 11.0in/s$$

$$F_{load-ext} = pA_r = 1000(psi)7(in^2) = 7000lb$$

$$Power_{ext} = F_{load-ext} \times v_{ext} = 7000lb \times 11.0in/s = 77{,}000lb.in/s = 11.7hp$$

b.

$$v_{ret} = Q_p \big/ \left(A_p - A_r\right) = \frac{20 \times \dfrac{231}{60}}{25-7} = 4.28in/s$$

$$F_{load-ret} = p\left(A_p - A_r\right) = 1000lb/in^2\left(25-7\right)in^2 = 18{,}000lb$$

$$Power_{ret} = F_{load-ret} \times v_{ret} = 18{,}000lb \times 4.28in/s = 77{,}000in.lb/s = 11.7hp$$

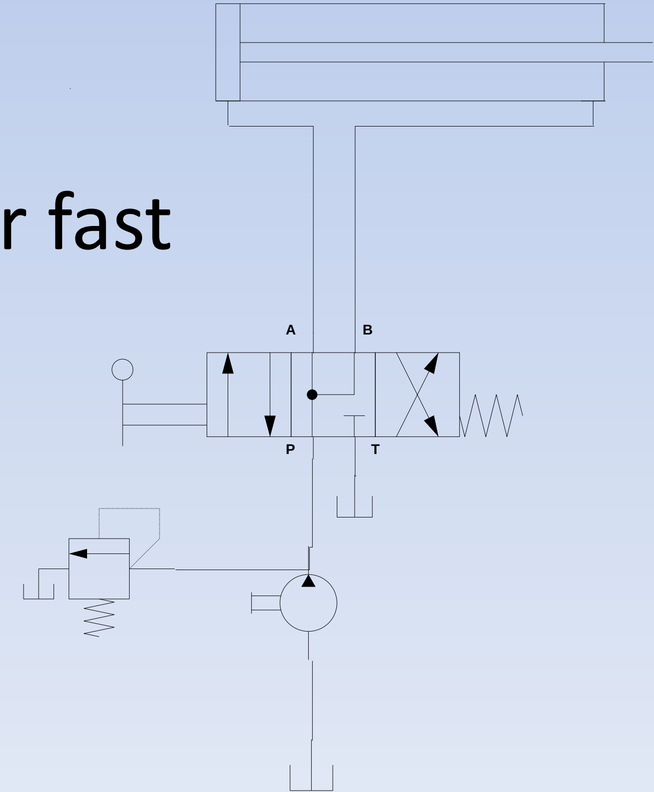
The hydraulic horsepower delivered by the pump for both cases can be found as follow

$$HP_{pump} = \frac{P(psi) \times Q_p(gpm)}{1714} = \frac{1000 \times 20}{1714} = 11.7hp$$

Drilling Machine

- Center position give rabid spindle advance
- Left envelope slow feed(extension) during drilling.
- Right envelope retract piston.

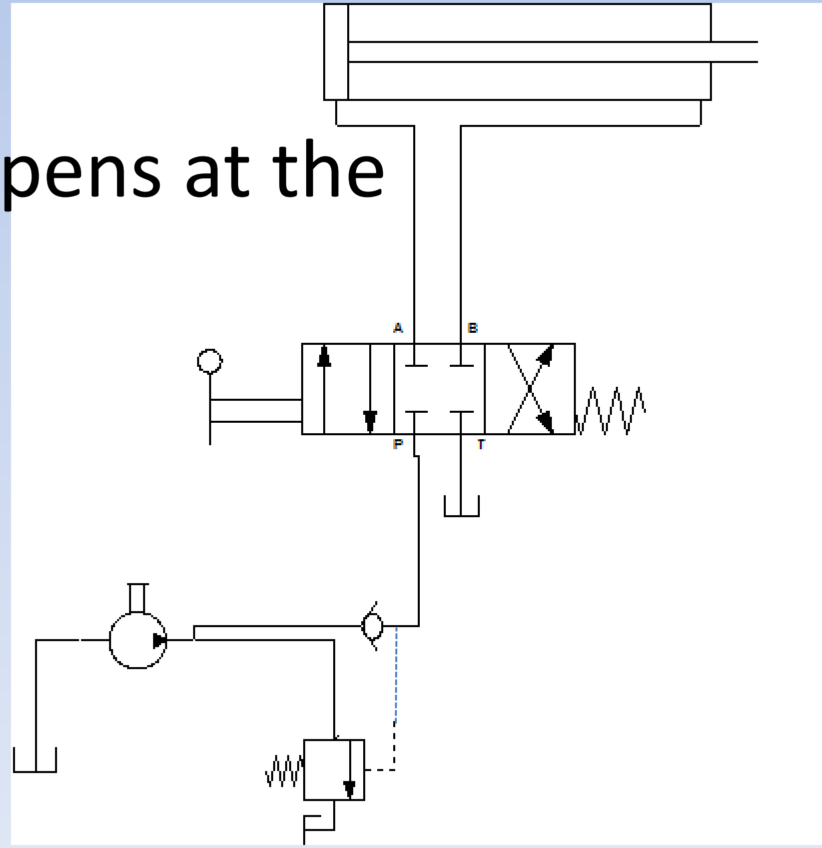
Center position is regenerative for fast extension.



Pump Unloading Circuit

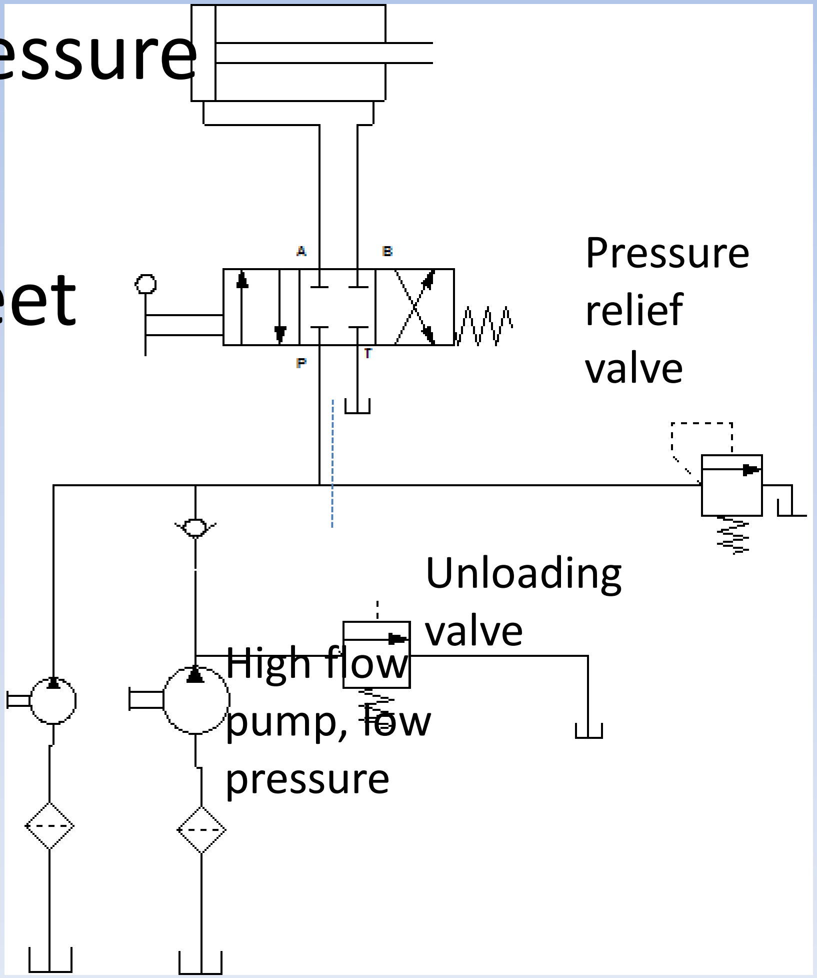
- The unloading valve opens when the cylinder reaches the end of the extension and retraction strokes
- Also the unloading valve opens at the

center position



Double Pump Hydraulic System

- Uses a high pressure, low flow pump in conjunction with a low pressure high flow pump.
- Typical application is a sheet metal punch press.



Example

For the double pump system, what should be the pressure setting of the unloading valve and pressure relief valve under the following conditions.

a. Sheet metal punching requires a 2000lb force.
b. Hydraulic cylinder is 1.5 in Dia., 0.5 in rod Dia.
c. During rapid extension a friction loss of 100 psi occur in the line from the high flow pump to the blank end of the cylinder, and 50 psi from the rod end to the tank (this loss is negligible during punching)
d. Assume that the unloading and pressure relief valves setting are 50% higher than the pressure required to over come the loads.

Sol.

- Unloading valve setting.

The back pressure force on the cylinder is given as

$$F_{back\ pressure} = P_{exhaust}\left(A_p - A_r\right) = 50\frac{lb}{in^2} \times \frac{\pi}{4}\left(1.5^2 - 0.5^2\right)in^2 = 78.5lb$$

The blank end pressure to over come the back pressure force is

$$P_{cyl\ blankend} = \frac{F_{back\ pressure}}{A_p} = \frac{78.5}{\pi/4\left(1.5^2\right)} = 44.4\,psi$$

Thus the unloading valve setting should be

$$1.50\left(100 + 44.4\right)psi = 217\,psi$$

- Pressure relief valve setting

$$P_{relief\ valve} = 1.50\left(\frac{2000lb}{\frac{\pi}{4}1.5^2 in^2}\right) = 1.50 \times 1132 = 1698\,psi$$

Example

For the system of the previous example the poppet of the pressure relief valve must move 0.10 from its fully closed position in order to pass the full pump flow at the PRV setting. The poppet has a 0.75 in^2 area on which the pressure acts. Assuming that the pressure relief valve cracking pressure should be 10% higher than the pressure required to over come the hydraulic cylinder punch operation. Find

a. Spring constant of the compression spring of the PRV
b. Initial compression of the spring from its free length condition.

Solution

A. At full pump flow pressure (PRV setting), the spring force equal the hydraulic force acting on the poppet

$$kS = P_{PRV\ setting} A_{poppet} = 1698\,psi \times 0.75 in^2 = 1274 lb$$

where S is the full compression of the spring $S = (l + 0.1)$

$$k(l + 0.1) = 1274 lb \Rightarrow kl + 0.1k = 1274 lb \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

At the cracking pressure of the relief valve the spring force equals the hydraulic force

The cracking pressure is $1.10 \times \left(\dfrac{2000 lb}{\dfrac{\pi}{4} 1.5^2 in^2} \right)$

$$kl = p_{cracking} A_{poppet} = (1.10 \times 1132) \times 0.75$$

$$kl = 934 lb \Rightarrow \text{Sub. in (1)} \Rightarrow 934 + 0.1k = 1274$$

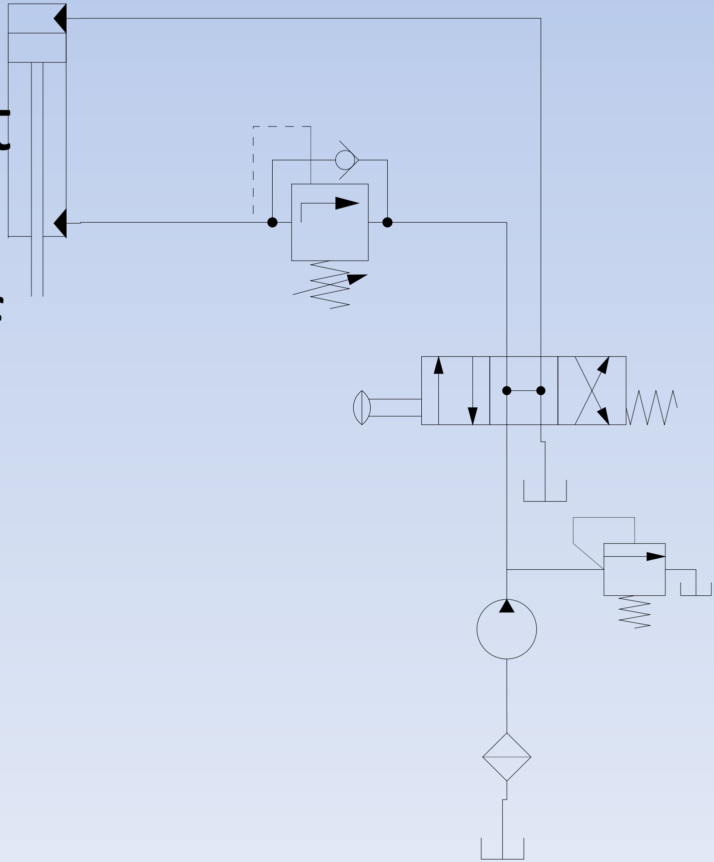
$$k = 3400 lb / in$$

B. The initial compression is given by

$$kl = 934 lb \Rightarrow l = \frac{934 lb}{3400 lb / in} = 0.275 in$$

Counter Balance Valve

- Used to keep a vertically mounted hydraulic cylinder in the upward position while the pump in idling
- The CBV is set somewhat above the pressure required to prevent the vertical cylinder from descending due to the weight of the load. Usually 30% above the pressure of the weight.

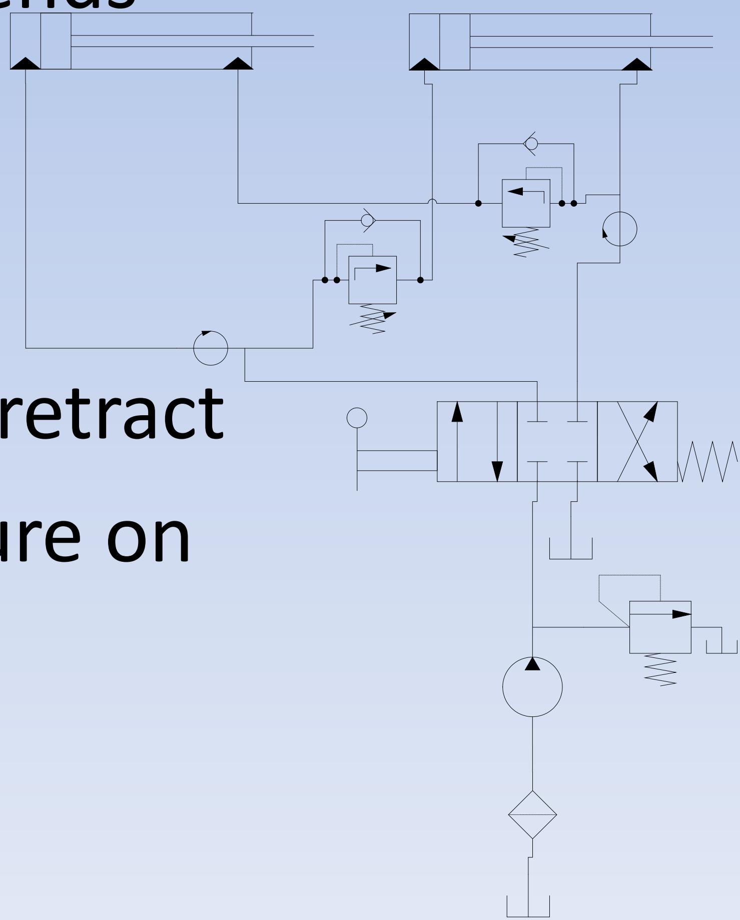


Hydraulic Cylinder Sequencing Circuit

- When the DCV is shifted to the left envelop the left cylinder extends and after full extension, the pressure in the line increase extending the right cylinder.

In retraction the right cylinder retract

after full retraction, the pressure on the sequence valve increase retraction the left cylinder.



Reciprocating of Double Acting Cylinder by Means of a Sequence Valve

- The sequence valves senses the end of the stroke and gives a pilot signal to

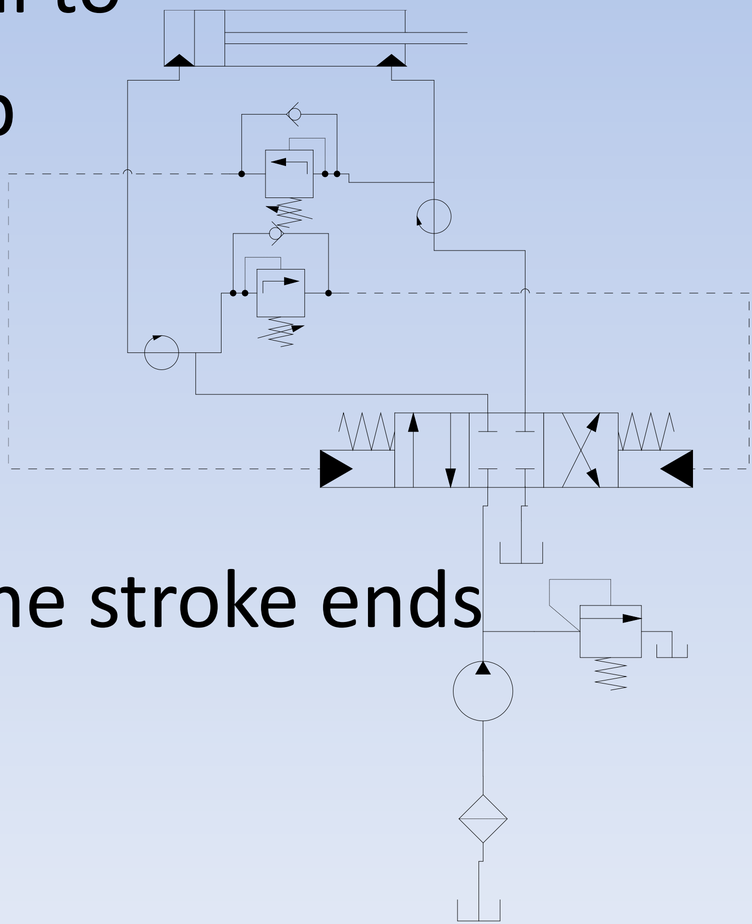
shift the DCV to other envelop

The Check valve is used to

allow the pilot line to drain to

the tank and prevent the

actuation of the DCV before the stroke ends

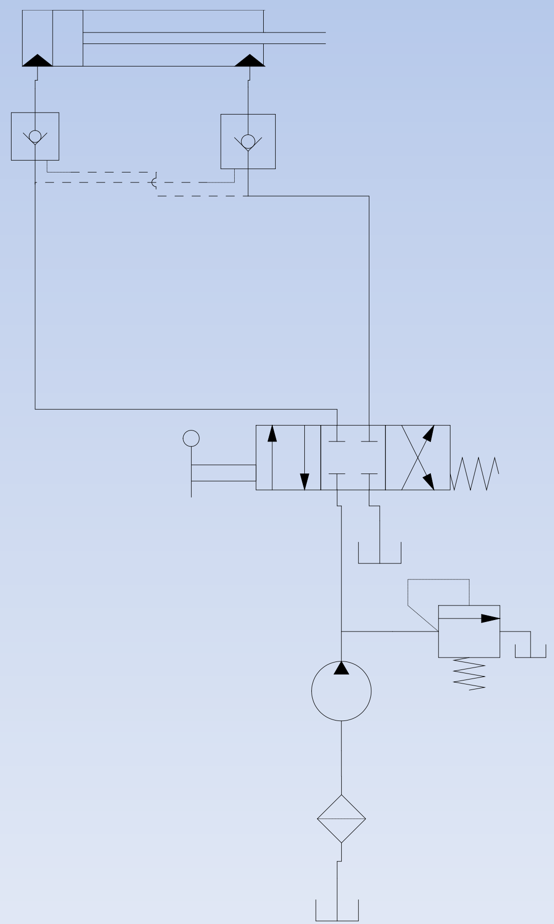


Locked Cylinder Using Pilot Check Valves

- Locking a double acing cylinder using a pilot operated check valve.
- Prevents the cylinder from

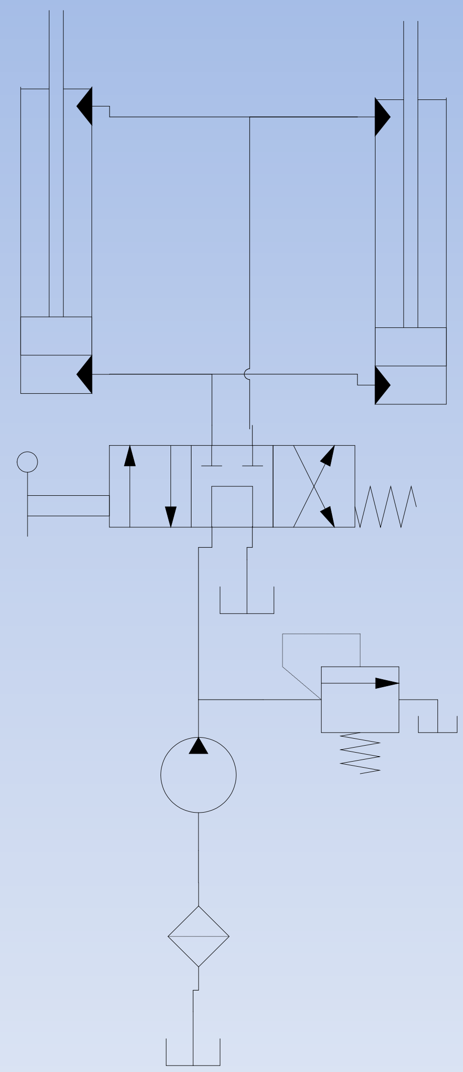
moving due to external force.

-



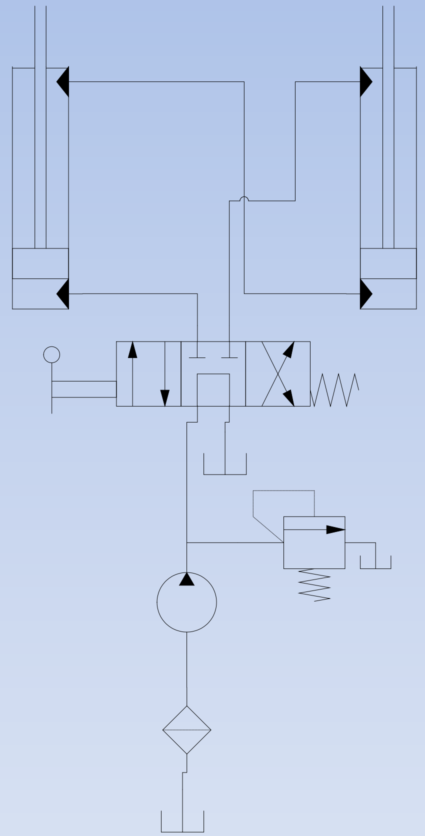
Cylinder Synchronizing

Two double acting cylinders connected in parallel, not synchronized due to different loading conditions of the two cylinders.



Cylinder Synchronizing

Two cylinders connected in series for synchronizing of operation the flow existing from the first cylinder is the input to the second cylinder.



$$Q_{out(cyl1)} = Q_{in(cyl2)}$$

$$(A_{p1} - A_{r1})v_{cyl1} = A_{p2}v_{cyl2}$$

For synchronization $v_{cyl1} = v_{cyl2}$, hence

$$(A_{p1} - A_{r1}) = A_{p2}$$

The areas should be adjusted for precise synchronizing

The pump should be able to overcome the load on both cylinders

$$p_1A_{p1} - p_2(A_{p1} - A_{r1}) = F_1$$

$$p_2A_{p2} - p_3(A_{p2} - A_{r2}) = F_2$$

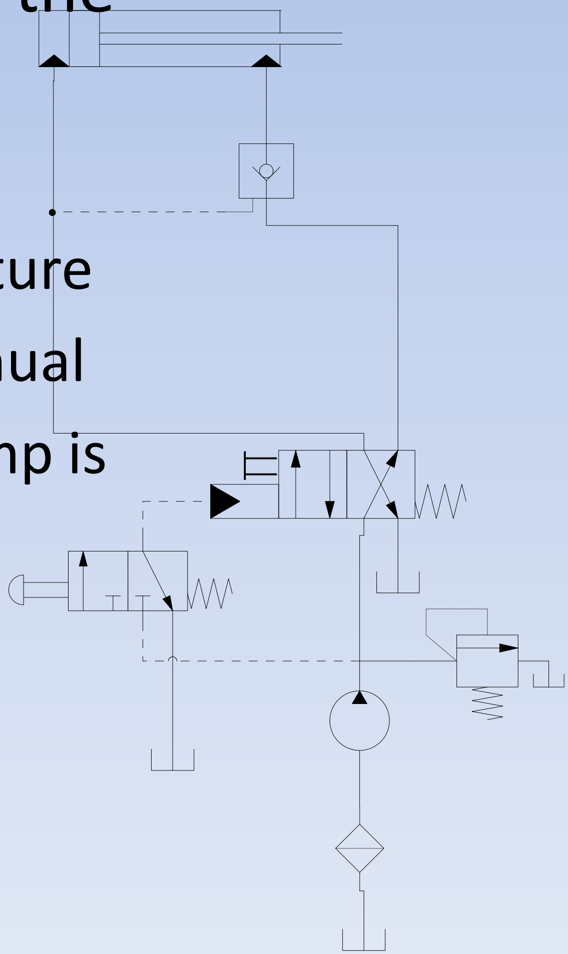
However $p_3 = 0$, adding the two equations and $(A_{p1} - A_{r1}) = A_{p2}$

$$p_1A_{p1} = F_1 + F_2$$

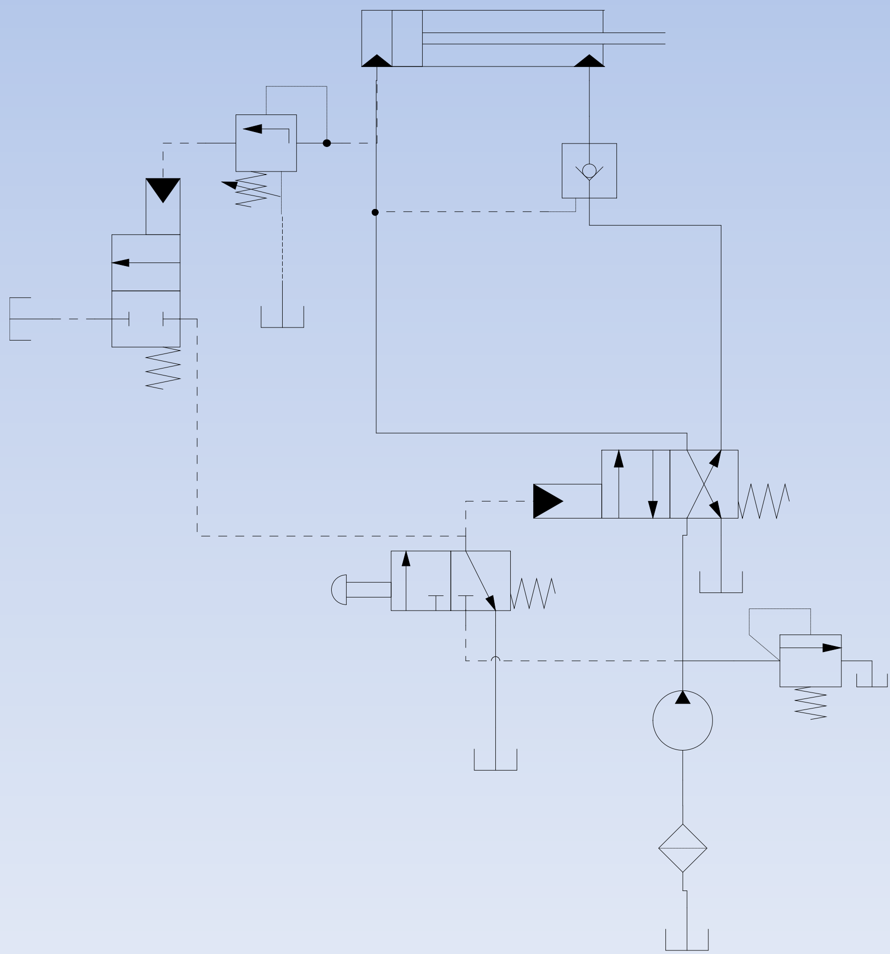
Fail Safe Circuits

Protection from Inadvertent Cylinder Extension.

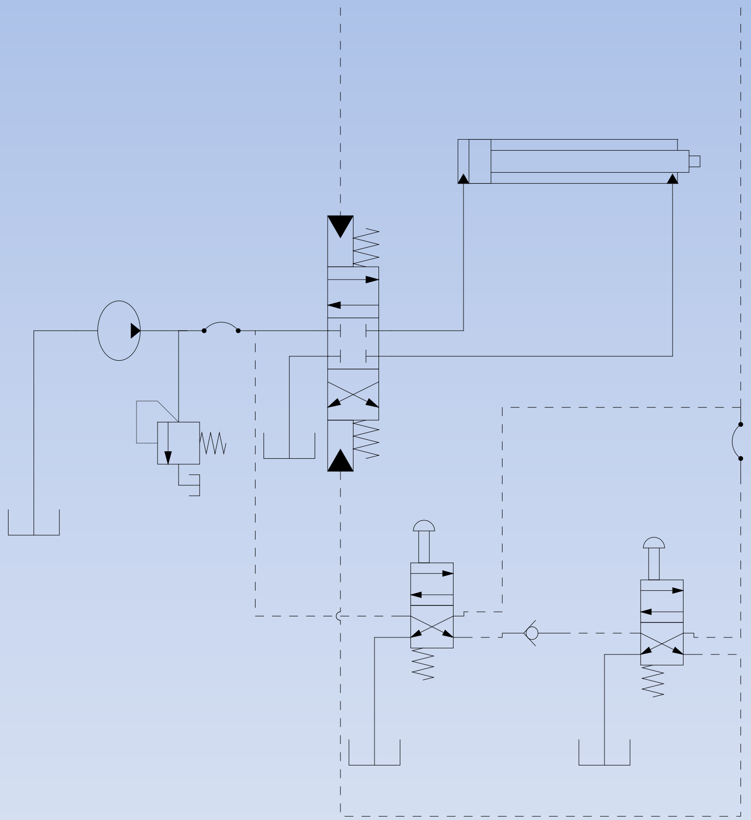
- Fail safe circuits are those designed to prevent injury to the operator or damage to the equipment.
  - The circuit shown prevent the cylinder from falling in case of hydraulic line rapture or someone accidently operate the manual override on the pilot DCV when the pump is not operating.



Fail Safe System with Overload Protection



Two Handed Safety System



Speed Control of Hydraulic Cylinder

- Meter In: Extension Speed Analysis

$$Q_{cyl} = Q_{pump} - Q_{PRV}$$

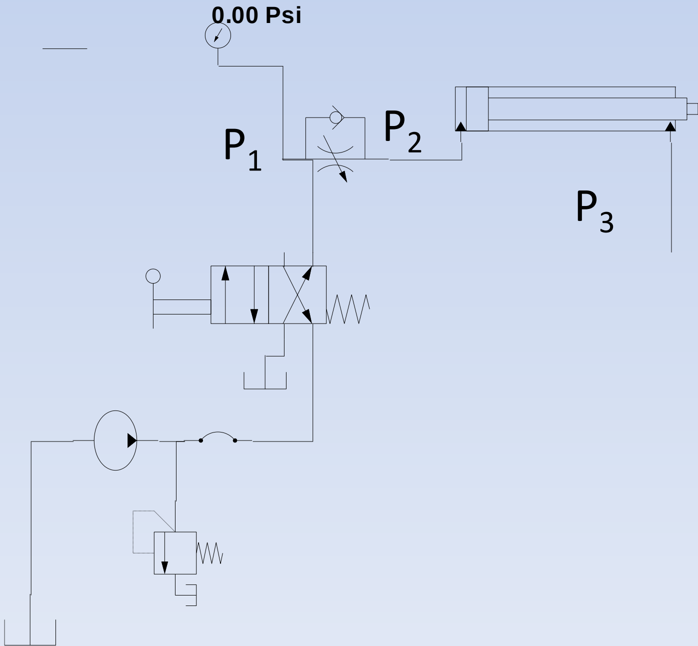
$$Q_{FCV} = C_v \sqrt{\frac{\Delta P}{SG}} = C_v \sqrt{\frac{P_1 - P_2}{SG}}$$

P_1=PRV setting
Ignoring the back pressure P_3

$$P_2 A_{Piston} = F_{Load}$$

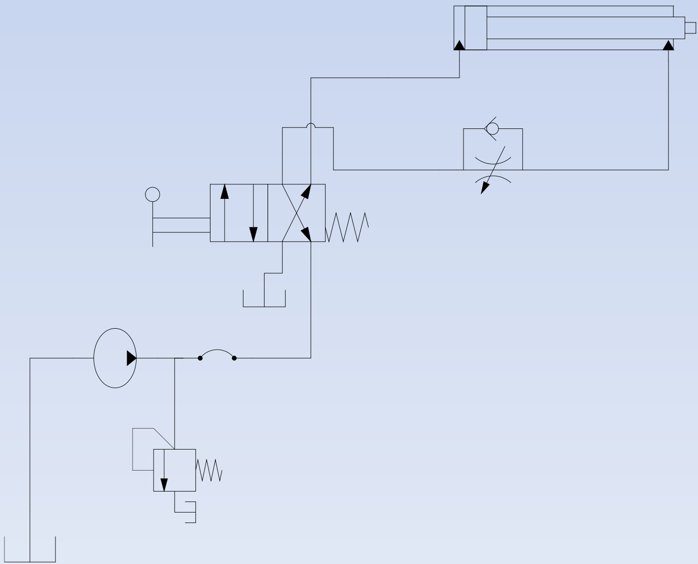
$$v_{cyl} = Q_{cyl} / A_{Piston} = Q_{FCV} / A_{Piston}$$

$$v_{cyl} = \frac{C_v}{A_{piston}} \sqrt{\frac{P_{PRV} - F_{Load} / A_{Piston}}{SG}}$$



Meter Out Speed Control of a Hydraulic Cylinder

- Meter in systems are primarily used when the external load opposes the direction of the motion of the cylinder.
- Disadvantage of meter out is excessive pressure build up at the rod end of the cylinder and heat generation.



Example

A meter in system has the following parameters.

$$C_v = 1.0 gpm / \sqrt{psi}$$

$$D_{piston} = 2in \Rightarrow A_{piston} = 3.14 in^2$$

$$F_{load} = 4000 lb$$

$$SG_{oil} = 0.90$$

$$\text{Pressure relief valve setting} = 1400\, psi$$

Determine the cylinder speed

$$Q_{cyl} = C_v \sqrt{\frac{P_{PRV} - F_{Load} / A_{Piston}}{SG}} = 1.0 \sqrt{\frac{1400 - 4000 / 3.14}{0.90}} = 11.8 gpm = 45.4 in^3 / s$$

$$v_{cyl} = \frac{45.4 in^3 / s}{A_{piston}(in^2)} = \frac{45.4}{3.14} = 14.5 in / s$$

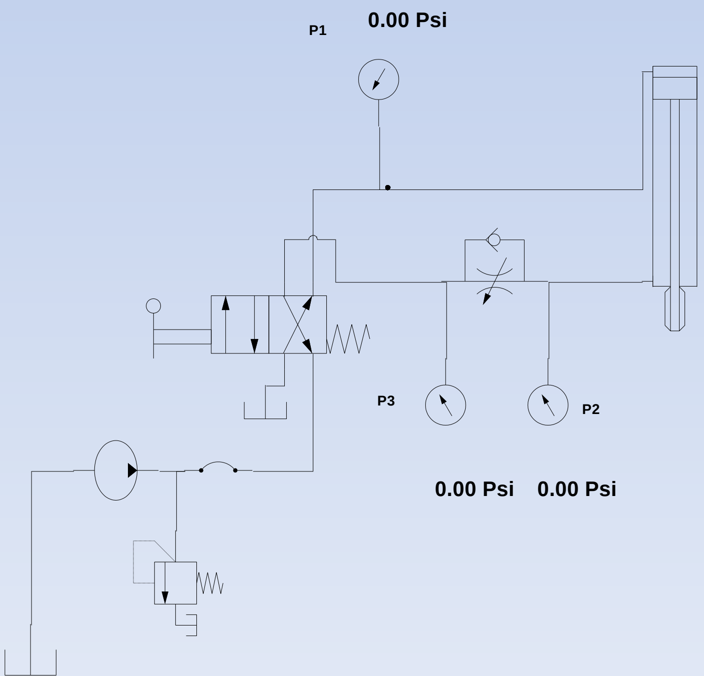
Example

For the meter out system shown find the pressure at the pressure gages during constant speed extension of the cylinder for

a) No load

b) 20,000N load

The cylinder piston diameter is 50 mm and rod is 25 mm, and the PRV setting is 10 Mpa



Solution

- During extension the cylinder flow rate is less than the pump flow rate due to the flow through the PRV and the P_1=PRV setting (neglecting losses)=10 MPa

a.

$$F = 10\times10^6 \, {}^{N}\!/_{m^2} \times \frac{\pi}{4}(0.05m)^2 = 19,600N$$

At No load with constant speed $\sum F = 0$

$$P_1 A_P = P_2 (A_p - A_r)$$

$$19,600N = P_2 \times \frac{\pi}{4}\left(0.05^2 - 0.025^2\right)m^2$$

$$P_2 = 13.3MPa$$

$$P_3 = 0$$

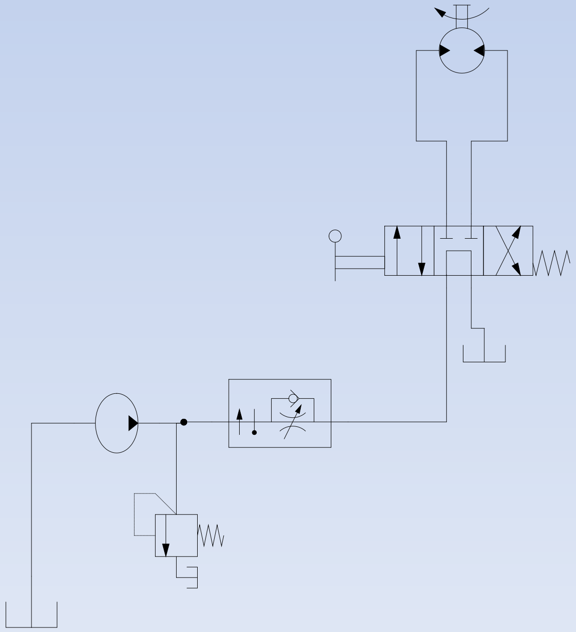
b.

For the case of 20,000N load

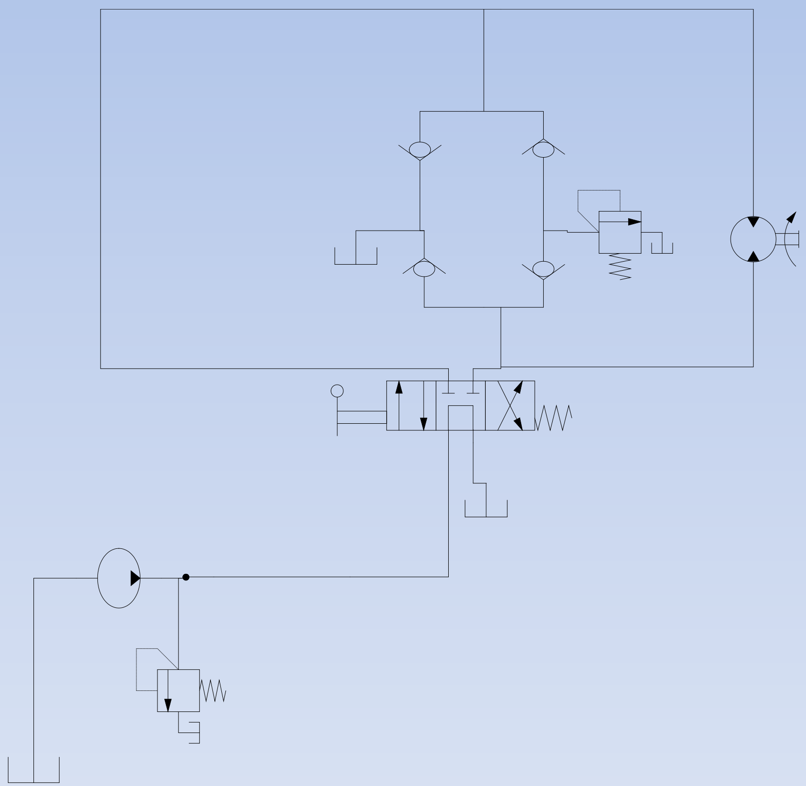
$$19,600N + 20,000N = P_2 \times \frac{\pi}{4}\left(0.05^2 - 0.025^2\right)m^2$$

$$P_2 = 26.9MPa$$

Speed Control of Hydraulic Motor



Hydraulic Motor Braking System



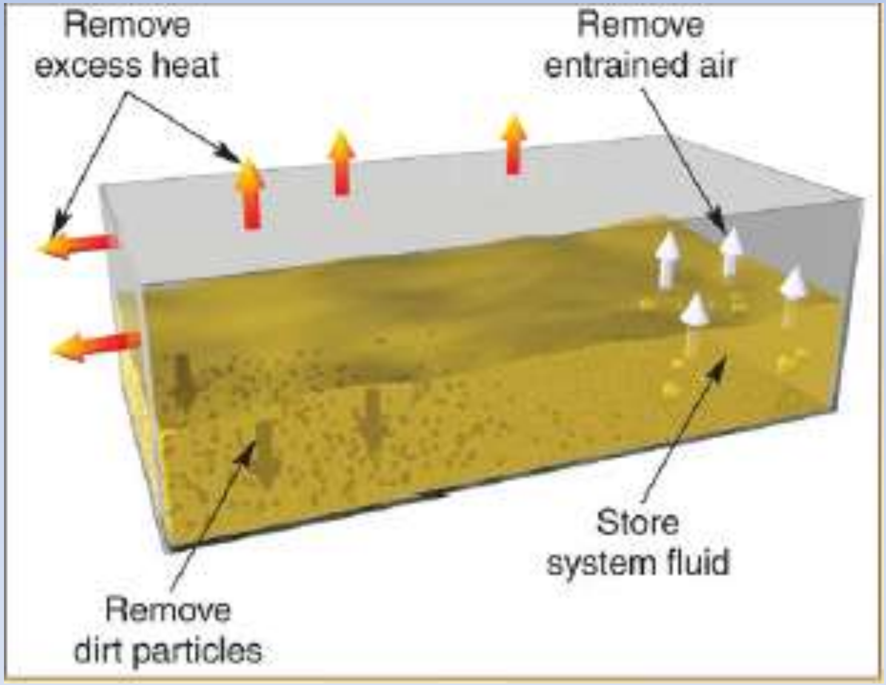
Hydraulic Conductors and Fittings

Chapter 10

Dr. Suleiman BaniHani

Reservoir

- The reservoir serves many functions



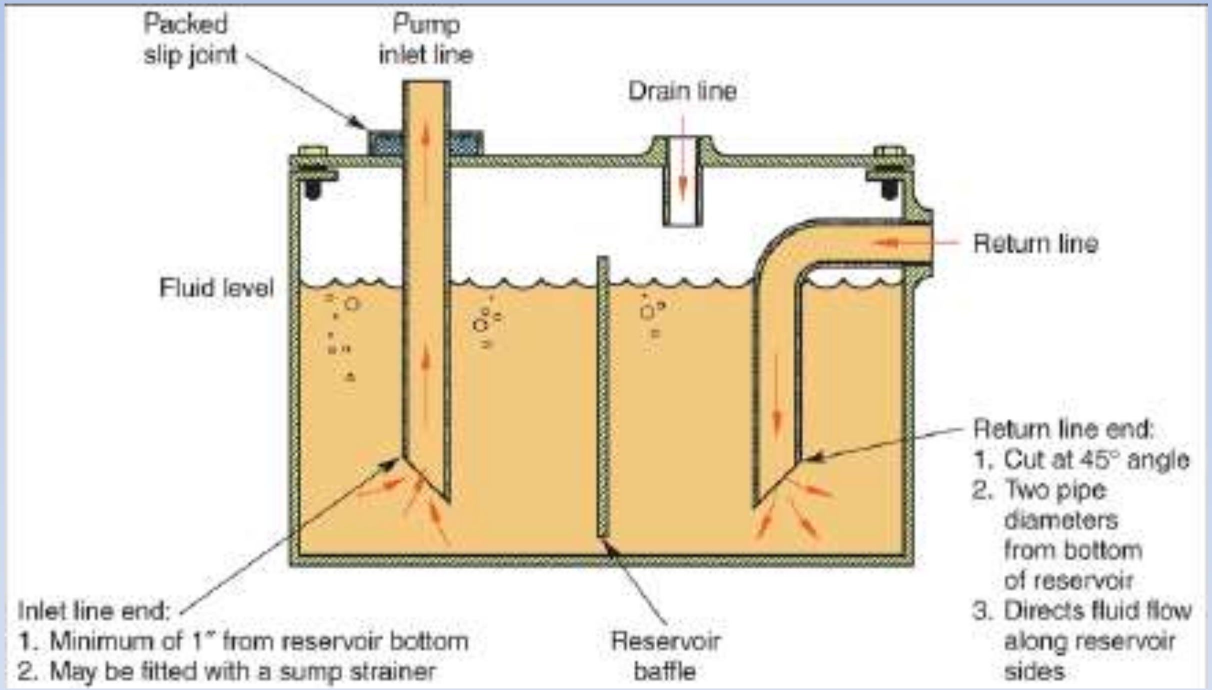
Reservoir

- The primary purpose of the reservoir is to hold the system fluid not currently in use in the system
- Other important functions of the reservoir are:
  - Remove heat
  - Separate solid particles
  - Release air from fluid
  - Separate water from fluid

Reservoir

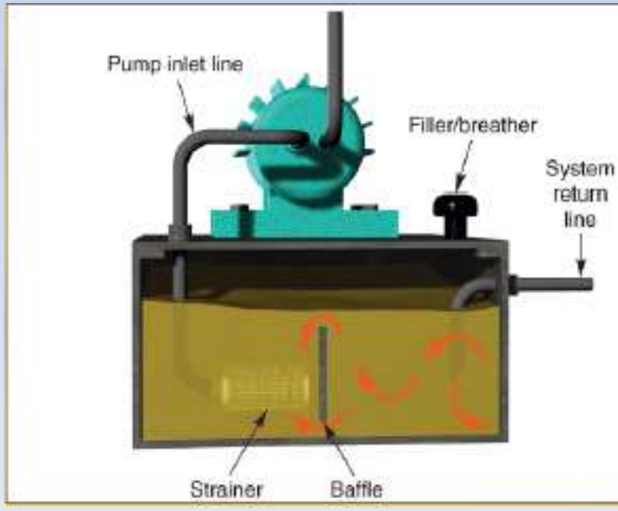
- The typical hydraulic system reservoir is a rectangular, covered steel tank
- The tank is typically fitted with:
  - Pump inlet line
  - System fluid return line
  - Drain line
  - Filler cap
  - Air breather
  - Fluid-level indicator

Reservoir


Packed slip joint
Pump inlet line
Drain line
Return line
Fluid level
Return line end:
1. Cut at 45° angle
2. Two pipe diameters from bottom of reservoir
3. Directs fluid flow along reservoir sides
Inlet line end:
1. Minimum of 1" from reservoir bottom
2. May be fitted with a sump strainer
Reservoir baffle

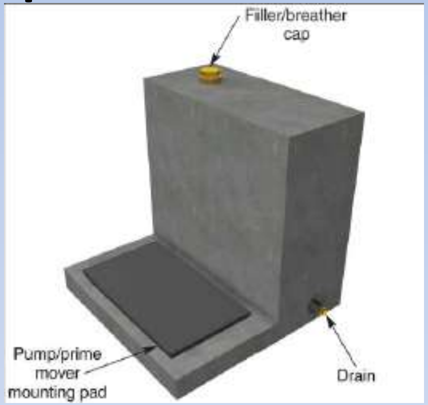
Reservoir

- Baffles are used in the interior of reservoirs to direct flow to maximize the distance the fluid must travel between the return line and the pump inlet line
  - Slows the movement of the fluid
  - Increases cooling
  - Increases separation of solid particles, air, and water



Reservoir

- L-shaped and overhead reservoir designs may be used in systems where positive pressure is needed on the pump inlet line
- Overhead reservoir
- In special situations, the system reservoir may be:
  - Cavities in large machines
  - Gear cases in mobile equipment



Reservoir

- As a general rule, the capacity of the reservoir should be three times the rated flow of the pump
  - Fixed installations may be higher
  - Mobile applications, where weight and space are factors, may be less

Conductors

- In Hydraulics systems, the fluid flows through a distribution system consisting of conductors and fittings.
- These conductors and fittings must be properly designed.
- Hydraulic systems use four main types of connectors:
  - Steel Pipes
  - Steel Tubing
  - Plastic Tubing
  - Flexibale Hoses

Conductors

- Conductors must have:
  - Adequate strength to withstand high system pressures
  - Low flow resistance to assure low energy loss during system operation
  - A design that allows economic installation and low maintenance

- Note: Conductors must not only withstand normal system operating pressure, but also hydraulic shock pressures
  - Shock pressures result from kinetic energy in the system when:
    - Directional control valves are shifted to reverse the movement of a load or heavy machine member
    - Actuators encounter sudden load changes

Which Connector to Use

- The choice of the connector depends on
  - The system pressure
  - Flow rate
  - Environmental conditions
    - Type of fluid
    - Operating Temperatures
    - Vibration
    - Relaive motion between connected components

Conductors

- Low flow resistance requires a conductor with an inside diameter large enough to allow the needed volume of fluid to move through the line within recommended fluid velocities

Restriction to Flow	Basic Flow Characteristic	Contributing Factors
Minimal	 Laminar flow	• Adequate conductor diameter • Recommended flow velocity • Smooth conductor surfaces
Excessive	 Turbulent flow	• Small conductor diameter • Excessive fluid velocity • Inappropriate conductor fittings • Excessive fluid velocity

Conductors

- Flow resistance in a system results from resistance to fluid flow caused by:
  - Surface of the conductor
  - Bends and fittings in the lines
  - Orifices in components
  - Turbulence in the fluid stream
  - Viscosity of the fluid

Conductors

- Fluid flow resistance resulting from fluid movement through conductors and other system components:
  - Lowers the work output of a system
  - Produces heat, which may cause operating problems

Conductors

- Future maintenance must be carefully considered when designing and installing hydraulic system conductors to assure minimal difficulty in removing components for service

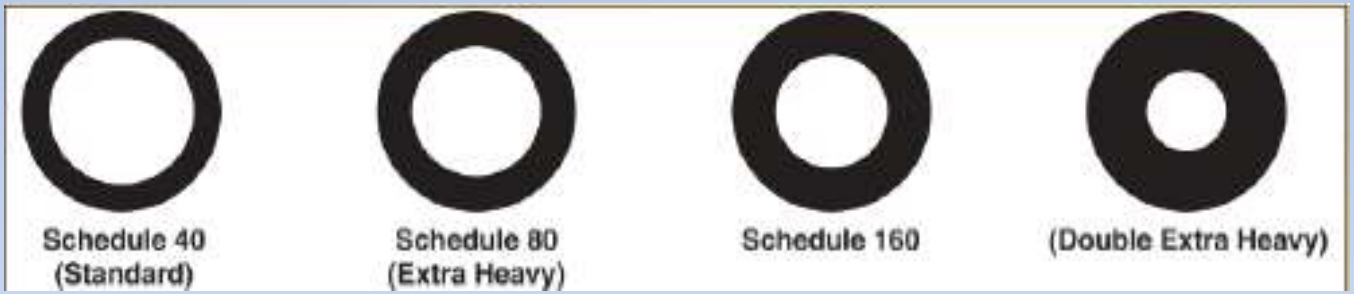
Pipe in a hydraulic system

- Pipe in a hydraulic system should be:
  - Seamless, black pipe
  - ANSI schedule rating of 40, 80, or 160, depending on the maximum pressure expected in the system

Nominal Size	Outside Diameter	Inside Diameter			
		Schedule 40	Schedule 80	Schedule 160	Double Extra Heavy
1/4	.540	.364	.302	—	—
3/8	.675	.493	.423	—	—
1/2	.840	.622	.546	.466	.252
3/4	1.050	.824	.742	.618	.434
1	1.315	1.049	.957	.815	.599
1 1/4	1.660	1.380	1.278	1.160	.896
1 1/2	1.900	1.610	1.500	1.338	1.100
2	2.375	2.067	1.939	1.689	1.503

Pipes

- Schedule number indicates wall thickness



Tubing

- Is a relatively thin-walled, semirigid conductor
- – Can be bent and shaped into lines that provide good flow characteristics with a minimum of visual clutter

Tubing

- The size of tubing is indicated by the actual outside diameter
  - Inside diameter varies according to wall thickness
  - Most tubing is manufactured to the specifications of a standardizing organization such as ANSI or SAE

Hoses

- Hose is a flexible conductor made up of:
  - Inner tube to conduct the fluid
  - Middle layer of reinforcing material for strength
  - Outer protective coating to withstand abrasion and abuse

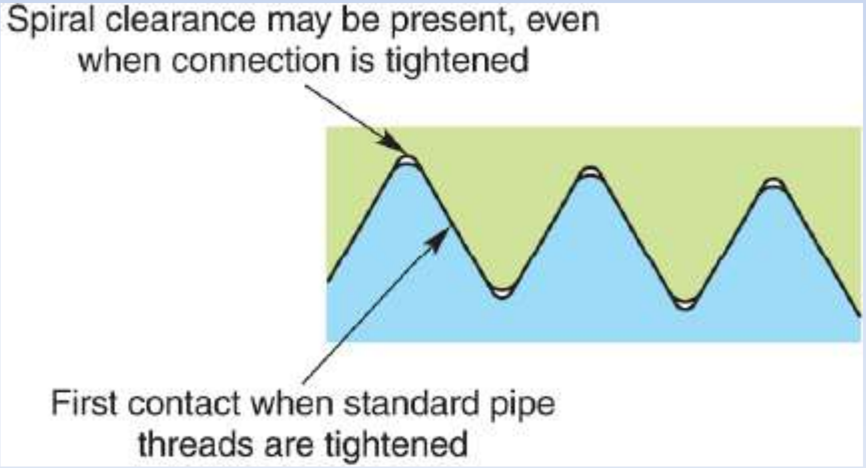
Fittings

- A wide variety of fittings are available to assist in attaching conductors to system components such as:
  - Reservoir
  - Pump
  - Valves
  - Actuators

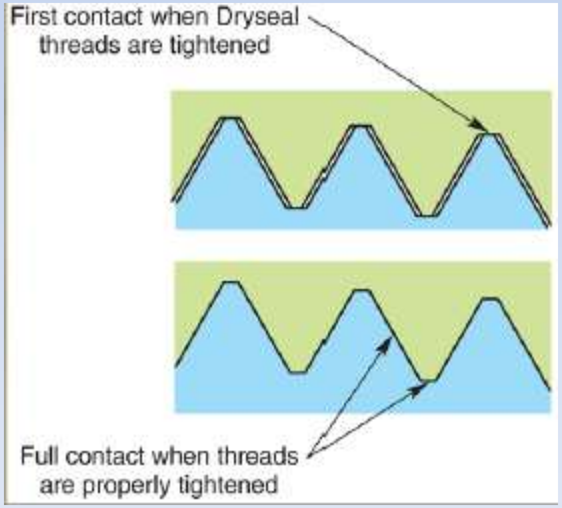
Pipe fittings

- Dryseal standard pipe threads should be used on pipe fittings, rather than standard pipe threads, to assure a tight thread seal that will not leak under high system pressure

Standard pipe threads



Dryseal standard pipe threads



Tubes and Hoses

- Fittings with pipe threads or straight threads sealed with an O-ring or a metal compression washer are typically used to attach tube and hose to hydraulic components
- Tubing is attached to fittings by flaring the tube, compression, soldering, or brazing

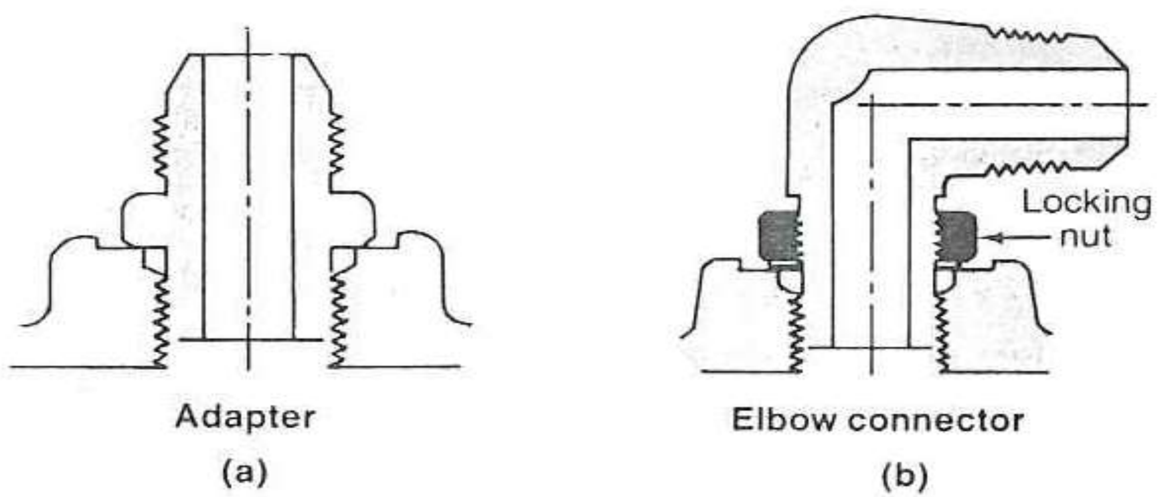


Fig. 25.16. SAE straight thread, fittings are sealed with an O-ring. Elbows can be properly oriented before locking nut is tightened.

Tubes

- Hydraulic tubing is most widely used in mobile applications. Tubing can be either seamless carbon steel or welded steel. Seamless steel tubing is usually annealed to facilitate bending and flaring.

- Tubing is sized by O.D. in contrast to pipe, which is sized by I.D.

- Tubing walls are thin and therefore connections other than cut threads must be used.

Tubes working pressure

Tube OD in.	Tube Wall Thickness, in.													Swagelok Fitting Series
	0.028	0.035	0.049	0.065	0.083	0.095	0.109	0.120	0.134	0.148	0.165	0.180	0.220	
	Working Pressure, psig **Note:** For gas service, select a tube wall thickness outside of the shaded area. (See **Gas Service,** page 2.)													
1/8	8000	10 200												200
3/16	5100	6 600	9600											300
1/4	3700	4 800	7000	9600										400
5/16		3 700	5500	7500										500
3/8		3 100	4500	6200										600
1/2		2 300	3200	4500	5900									810
5/8		1 800	2600	3500	4600	5300								1010
3/4			2100	2900	3700	4300	5100							1210
7/8			1800	2400	3200	3700	4300							1410
1			1500	2100	2700	3200	3700	4100						1610
1 1/4				1600	2100	2500	2900	3200	3600	4000	4600	5000		2000
1 1/2					1800	2000	2400	2600	2900	3300	3700	4100	5100	2400
2						1500	1700	1900	2100	2400	2700	3000	3700	3200

Hoses

- Flexible Conductors (Hose)

  Hoses provide the following advantages :

  1. Overcome severe vibration
  2. Compensate for manufacturing tolerances
  3. Provide freedom of routing conductors
  4. Absorb hydraulic impulse shock and smooth fluid flow

- Hose attached to an actuator with elbow fitting and adapter with pipe threads

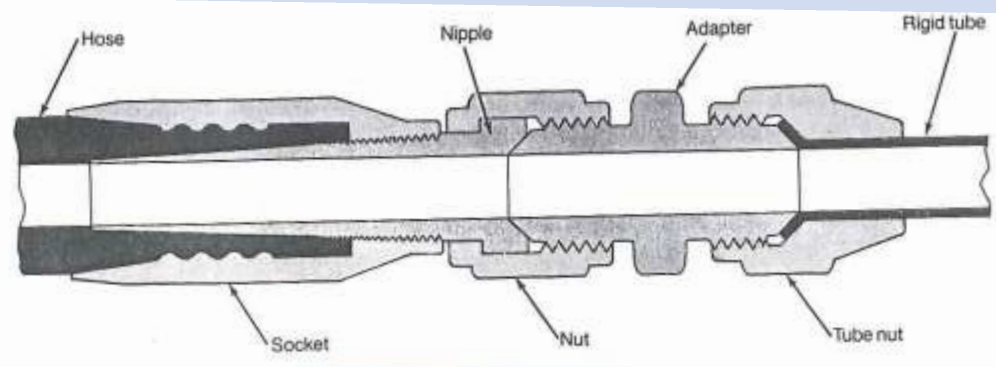


Fig. 25.19. Relationship between IDs based on the dash system of elements in a typical conductor.

Analysis of Circuit and System Operation

- Properly selecting a conductor requires an examination of not only the hydraulic system, but also the mechanisms operated by the system
- Factors that must be considered are:
  - Pressure requirements
  - Flow requirements
  - Vibration
  - Required movements of machine members

Analysis of Circuit and System Operation

- Flow velocity must be carefully considered
- when selecting a conductor for a system
  - Pump inlet line average fluid velocity should not exceed 4 ft/sec
  - Working line fluid velocity should not exceed 20 ft/sec

Recommended Velocity of Flow in Pipe and Tubing

- Many factors affect the selection of a satisfactory velocity of flow in fluid systems.
- Some of the important ones are the type of fluid, the length of the flow system, the type of pipe or tube, the pressure drop that can be tolerated, the devices (such as pumps, valves, etc.) that may be connected to the pipe or tube, the temperature, the pressure, and the noise.

Recommended Velocity of Flow in Pipe and Tubing

- The resulting flow velocities from the recommended pipe sizes in Fig. 6.2 are generally lower for the smaller pipes and higher for the larger pipes, as shown for the following data.

Volume Flow Rate		Suction Line			Discharge Line		
		Pipe	Velocity		Pipe	Velocity	
gal/min	m^3/h	Size (in)	ft/s	m/s	Size (in)	ft/s	m/s
10	2.3	1	3.7	1.1	¾	6.0	1.8
100	22.7	2½	6.7	2.0	2	9.6	2.9
500	114	5	8.0	2.4	3½	16.2	4.9
2000	454	8	12.8	3.9	6	22.2	6.8

Recommended Velocity of Flow in Pipe and Tubing

Recommended Flow Velocities for Specialized Systems

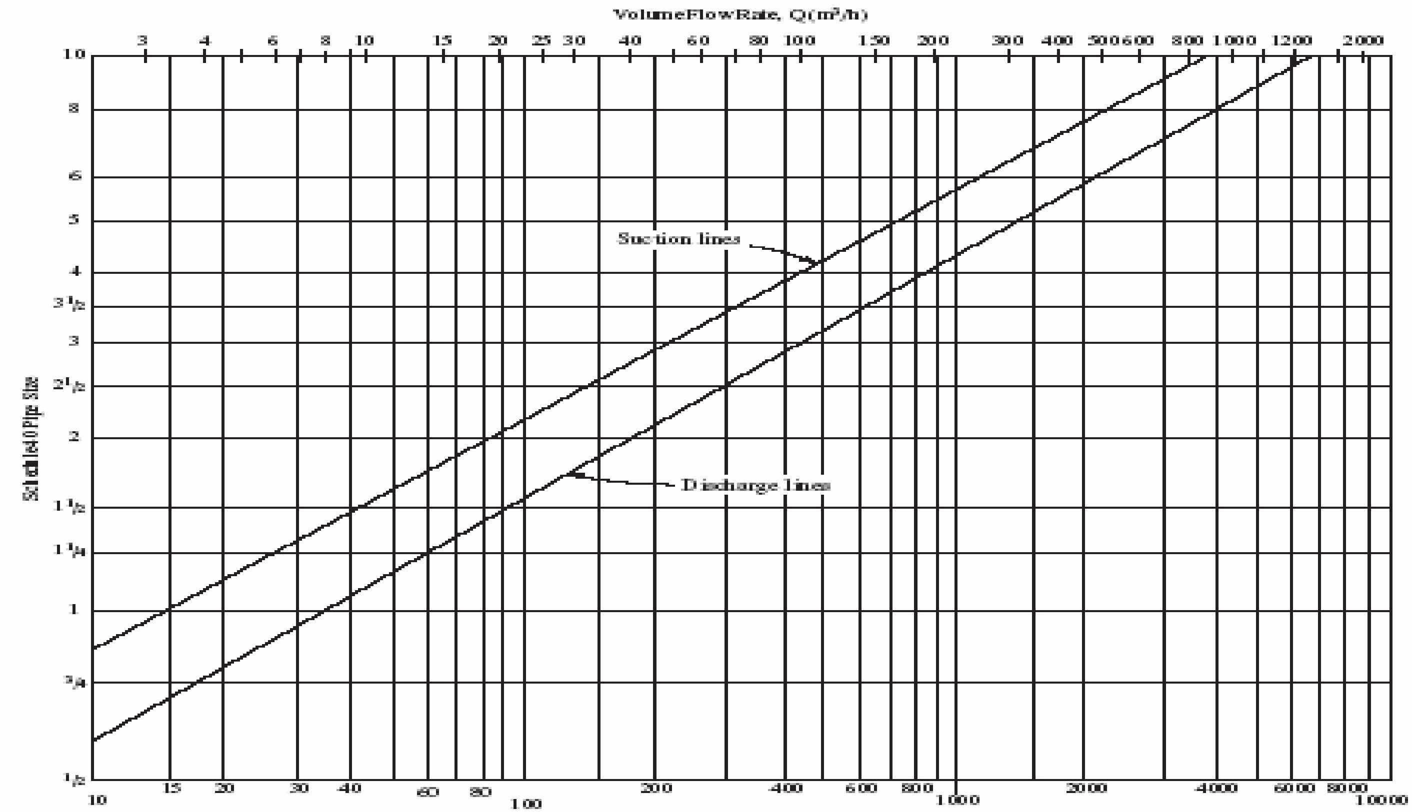


FIGURE 6.2 Pipe-size selection aid.

Recommended Velocity of Flow in Pipe and Tubing

- For example, recommended flow velocities for fluid power systems are as follows:

Type of Service	Recommended Range of Velocity ft/s	m/s
Suction lines	2–4	0.6–1.2
Return lines	4–13	1.5–4
Discharge lines	7–18	2–5.5

Recommended Velocity of Flow in Pipe and Tubing

- The suction line delivers the hydraulic fluid from the reservoir to the intake port of the pump.
- A discharge line carries the high-pressure fluid from the pump outlet to working components such as actuators or fluid motors.
- A return line carries fluid from actuators, pressure relief valves, or fluid motors back to the reservoir.

Example

Determine the maximum allowable volume flow rate in L/min that can be carried through a standard steel tube with an outside diameter of 1.25*in* and a 0.065 in wall thickness if the maximum velocity is to be 3.0 m/s.

Using the definition of volume flow rate, we have

$$Q = Av$$

$$A = 6.356 \times 10^{-4}\,\mathrm{m}^2 \quad \text{(from Appendix G)}$$

$$Q = (6.356 \times 10^{-4}\,\mathrm{m}^2)(3.0\,\mathrm{m/s}) = 1.907 \times 10^{-3}\,\mathrm{m}^3/\mathrm{s}$$

$$Q = 1.907 \times 10^{-3}\,\mathrm{m}^3/\mathrm{s}\left(\frac{60\,000\,\mathrm{L/min}}{1.0\,\mathrm{m}^3/\mathrm{s}}\right) = 114\,\mathrm{L/min}$$

Conductor Installation

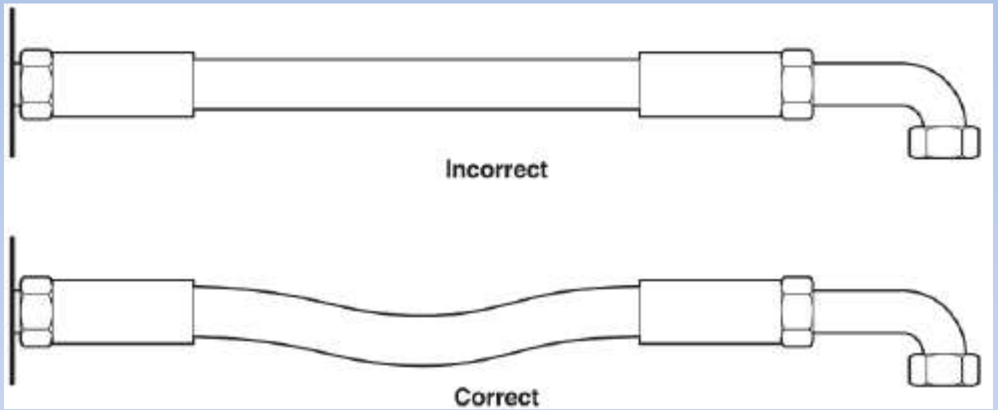
- When installing pipe and tubing, it is important to have the correct lengths
  - Should not be distorted
  - Should not be placed under tension
  - Distortion and tension can result in material fatigue and lead to part failure

Conductor Installation

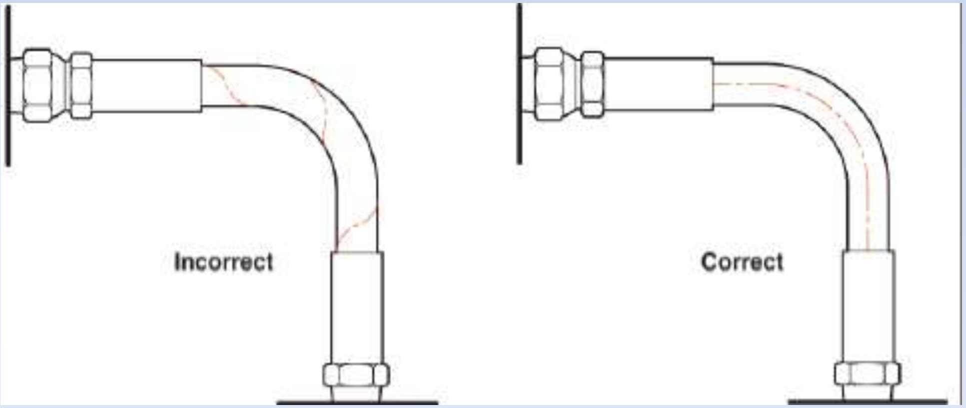
- When installing tubing, the number of fittings in a system can be reduced by bending the tube where possible
- Hand tools and power equipment are availab
- Long lengths of pipe and tubing should be supported by brackets or clamps to secure the conductor
- This will reduce fatigue caused by conductor weight or system vibration le to produce accurate bends

Conductor Installation

- Allow slack in hose when it is installed



- When hose is installed, ensure it is not twisted



MAINTENANCE OF FLUID POWER SYSTEMS

- The following is a list of most common causes of hydraulic system breakdown:
  - Clogged or dirty oil filters.
  - Inadequate supply of oil in the reservoir.
  - Leaking seals.
  - Loose inlet lines that cause the pump to take in air.
  - Incorrect type of oil.
  - Excessive oil temperature.
  - Excessive oil pressure.

The Importance of Cleanliness

- Cleanliness is the first requirement when it comes to servicing hydraulic systems. Keep dirt and other contaminants out of the system. Small particles can score valves, seize pumps, clog orifices and cause expensive repair jobs.
  - Keep the oil clean.
  - Keep the system clean.
  - Keep your work area clean.
  - Be careful when you change or add oil.

- **Importance of Oil and Filter Changes**
- **Draining the System**
- **Cleaning and Flushing the System**
- **Filling the System**
- **Preventing Leaks**
- **Preventing Overheating**
- **Problems Caused By Gases in Hydraulic Fluids**
  - **Free Air**
  - **Entrained Gas**
  - **Dissolved Air**

Air Free

- Keep suction velocities below 1.5 m/s.
- Keep pump inlet lines as short as possible.
- Minimize the number of fittings in the pump inlet line.
- Mount the pump as close as possible to the reservoir.
- Use low-pressure drop-pump inlet filters or strainers.
- Use a properly designed reservoir that can remove entrained air from the fluid before it enters the pump inlet line.
- Use proper oil as recommended by the manufacturer.
- Keep the oil from exceeding the recommended maximum temperature level

Excessive noise

Symptom	Cause	Remedy
Pump noisy	Cavitation	Any or all of the following: ☐ Replace dirty filters. ☐ Wash strainers. ☐ Clean the clogged inlet line. ☐ Clean the reservoir breather vent. ☐ Change the system fluid. ☐ Change to proper pump drive motor speed. ☐ Overhaul or replace the pump. ☐ Check fluid temperature.
	Air in fluid	Any or all of the following: ☐ Tighten leaky inlet connections. ☐ Fill the reservoir to proper level. ☐ Bleed air from the system. ☐ Replace the pump shaft seal.
	Coupling misaligned	All of the following: ☐ Align unit. ☐ Check the condition of seals, bearings and couplings.
	Pump worn or damaged	Overhaul or replace defective parts
Motor noisy	Coupling misaligned	All of the following: ☐ Align unit. ☐ Check the condition of seals, bearings and couplings.
	Motor or coupling worn	Overhaul or replace defective parts
Relief valve noisy	Setting too low or too	Install and adjust pressure gauge

Excessive heat

Symptom	Cause	Remedy
Pump heated	Fluid heated	See symptom "fluid heated"
	Cavitation	Any or all of the following: ☐ Replace dirty filters. ☐ Wash strainers. ☐ Clean the clogged inlet line. ☐ Clean the reservoir breather vent. ☐ Change the system fluid. ☐ Change to proper pump drive motor speed. ☐ Overhaul or replace the pump. ☐ Check fluid temperature.
	Air in fluid	Any or all of the following: ☐ Tighten leaky inlet connections. ☐ Fill the reservoir to proper level. ☐ Bleed air from the system. ☐ Replace the pump shaft seal.
	Excessive load	All of the following: ☐ Align unit. ☐ Check the condition of seals, bearings and couplings. ☐ Locate and correct mechanical binding. ☐ Check for workload in excess of circuit design.
	Pump worn or damaged	Overhaul or replace defective parts
	Relief or unloading valve set	Install and adjust pressure gauge
Motor heated	Fluid heated	See symptom "fluid heated"
	Relief or unloading valve set	Install and adjust pressure gauge
	Excessive loading	All of the following: ☐ Align unit. ☐ Locate and correct mechanical binding. seals, bearings and couplings. ☐ Check for workload in excess of circuit design.
	Motor or coupling worn or	Overhaul or replace defective parts
Relief valve heated	Fluid heated	See symptom "fluid heated"
	Valve setting	Install and adjust pressure gauge
	Worn or damaged valve	Overhaul or replace defective parts
Fluid heated	System pressure too	Install and adjust pressure gauge
	Unloading valve set	Install and adjust pressure gauge
	Fluid dirty or low supply	☐ Change filters. ☐ Check system fluid viscosity, change if necessary. ☐ Fill the reservoir to proper level.
	Incorrect fluid viscosity	☐ Change filters. ☐ Check system fluid viscosity, change if necessary. ☐ Fill the reservoir to proper level.
	Faulty fluid cooling system	☐ Clean the cooler and/or strainer. ☐ Replace the cooler control valve. ☐ Repair or replace the cooler.
	Worn pump, valve, motor,	Overhaul or replace defective parts

Incorrect flow

Symptom	Cause	Remedy
No flow	Pump not receiving fluid	Any or all of the following: ☐ Replace dirty filters. ☐ Clean the clogged inlet line. ☐ Clean the reservoir breather vent. ☐ Change the system fluid. ☐ Overhaul or replace the pump.
	Pump drive motor not operating	Overhaul
	Pump to drive coupling sheared	Check for the damaged pump Replace and align coupling.
	Pump drive motor turning in the wrong	Reverse rotation
	Directional control set in the wrong direction	Any or all of the following: ☐ Check the position of manually operated controls. ☐ Check the electrical circuit on solenoid- operated controls. ☐ Repair or replace pilot pressure pump.
	Entire flow passing over the relief	Adjust part
	Damaged pump	Check for the damaged pump Replace and align coupling.
	Incorrectly assembled pump	Overhaul or replace part
Excessive flow	Flow control set too high	Adjust part
	Yoke actuating device inoperative (variable	Overhaul or replace part
	Rotation per minute (RPM) of pump drive	Replace with correct unit
	Improper size pump used for	Replace with correct unit

Low flow	Flow control set too low	Adjust part
	Relief or unloading valve set too low	Adjust part
	Flow bypassing through the partially open valve	Overhaul or replace part or any or all of the following: ☐ Check the position of manually operated controls ☐ Check the electrical circuit on solenoid- operated controls. ☐ Repair or replace the pilot pressure pump.
	External leak in the system	☐Bleed air from the system.
	Yoke actuating device inoperative (variable displacement pump)	Overhaul or replace part
	RPM of pump drive motor incorrect	Replace with correct unit
	Worn pump, valve motor, cylinder or other	Overhaul or replace part

Incorrect pressure

Symptom	Cause	Remedy
No pressure	No flop	See "incorrect flow", symptom "no flow"
Low pressure	Pressure relief path exists	See "incorrect flow," symptom "no flow" and "low flow"
	Pressure-reducing valve set too low	Adjust part
	Pressure-reducing valve damaged	Overhaul or replace part
	Damaged pump, motor or cylinder	Overhaul or replace part
Erratic pressure	Air in fluid	☐ Tighten leaky connections. ☐ Fill the reservoir to proper level. ☐ Bleed air from the system.
	Worn relief valve	Overhaul or replace part
	Contamination in fluid	Replace dirty filters and system fluid
	Accumulator defective or had lost charge	☐ Check the gas valve for leakage. ☐ Change to correct pressure. ☐ Overhaul if defective.
	Worn pump, motor or cylinder	Overhaul or replace part
Excessive pressure	Pressure-reducing, relief or unloading valve misadjusted	Adjust part
	Yoke actuating device inoperative (variable displacement pumps)	Overhaul or replace part
	Pressure-reducing, relief or unloading valve worn or damaged	Overhaul or replace part

Faulty operation

Symptom	Cause	Remedy
No movement	No flow or pressure	See "incorrect flow"
	Limit or sequence device inoperative or misadjusted	Overhaul or replace part
	Mechanical bind	Locate bind and repair
	No command signal to the servo amplifier	Repair command console or interconnecting wires
	Inoperative or misadjusted servo amplifier	Adjust, repair or replace part
	Inoperative servo valve	Overhaul or replace part
	Worn or damaged cylinder or motor	Overhaul or replace part
Slow movement	Low flow	See "incorrect flow"
	Fluid viscosity to high	☐ Check fluid temperature. ☐ Check system fluid viscosity, change if necessary.
	Insufficient control pressure for valves	☐ See "incorrect pressure"
	No lubrication of machine ways or linkage	Lubricate
	Misadjusted or malfunctioning servo amplifier	Adjust, repair or replace part
	Sticking servo valve	☐ Clean and adjust or replace part. ☐ Check the condition of system fluid and filters.
	Worn or damaged cylinder or motor	Overhaul or replace part
Erratic movement	Erratic pressure	See "incorrect pressure"
	Air in fluid	See "excessive noise"
	No lubrication of machine ways or linkage	Lubricate
	Erratic command signal	Repair command console or interconnecting wires
	Misadjustment of malfunctioning servo amplifier	Adjust, repair or replace part
	Malfunctioning feedback transducer	Overhaul or replace part
	Sticking servo valve	☐ Clean and adjust or replace part. ☐ Check the condition of system fluid and filters.
	Worn or damaged cylinder or motor	Overhaul or replace part
Excessive speed or movement	Excessive flow	See "incorrect flow"
	Feedback transducer	Overhaul or replace part

Thanks