Jazan University
Mechanical Engineering Department

CHAPTER 13

CRITICAL SPEEDS OF SHAFTS

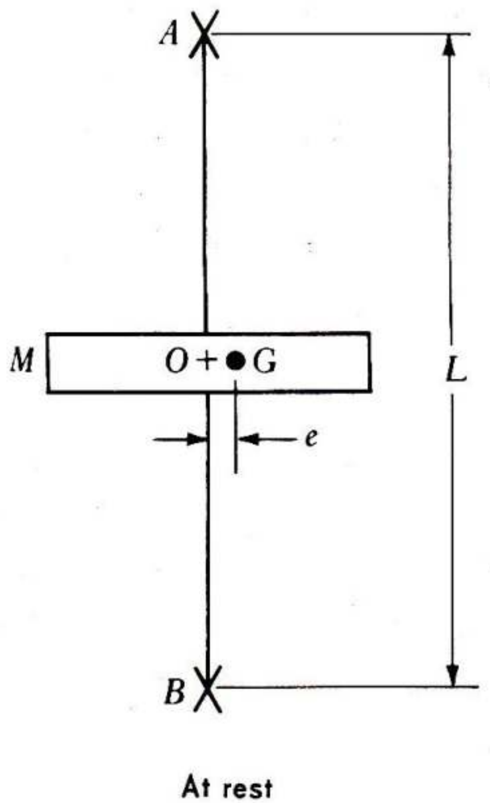
EngM 271 Theory of Machines

Dr. Amr Assie

introduction

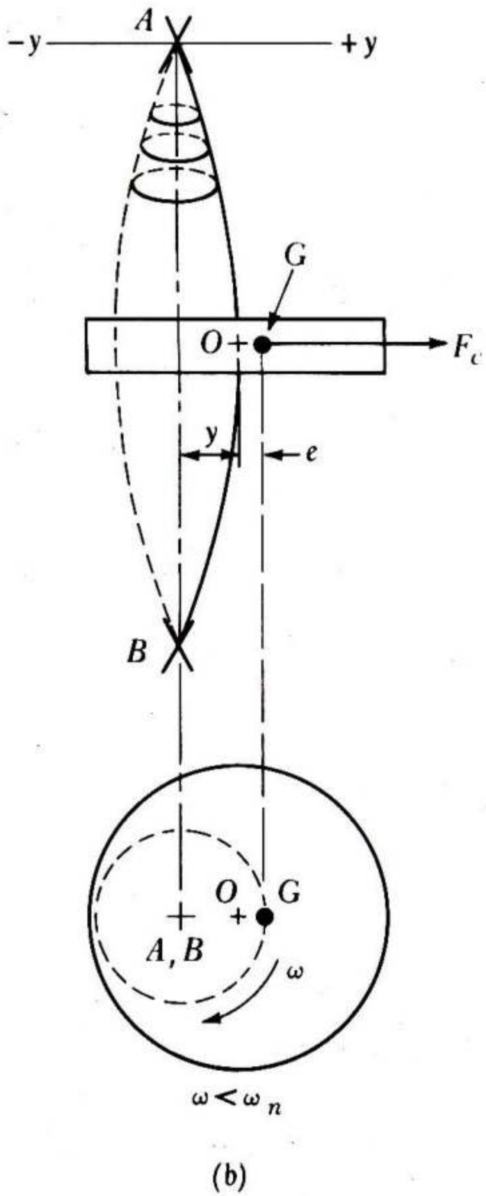
- At certain speeds a rotating shaft or rotor has been found to exhibit excessive lateral vibrations. The angular velocity of the shaft at which this occurs is called a critical speed. At a critical speed the shaft deflections become excessive and may cause permanent deformation or structural damage; for example, the rotor blades of a turbine may contact the stator blades.

- The large shaft deflections occurring at a critical speed cause large bearing reactions and can result in bearing failure or structural damage to the bearing supports. This phenomenon occur even for very accurately balanced rotors. A machine should never be operated for any length of time at a speed close to a critical speed.



(a)

Consider the shaft with the disk of mass M located between the bearings. The mass of the shaft is negligible compared with the mass of the disk. Point O is on the shaft axis, and G is the center of gravity of the disk. Then the distance e is the eccentricity.



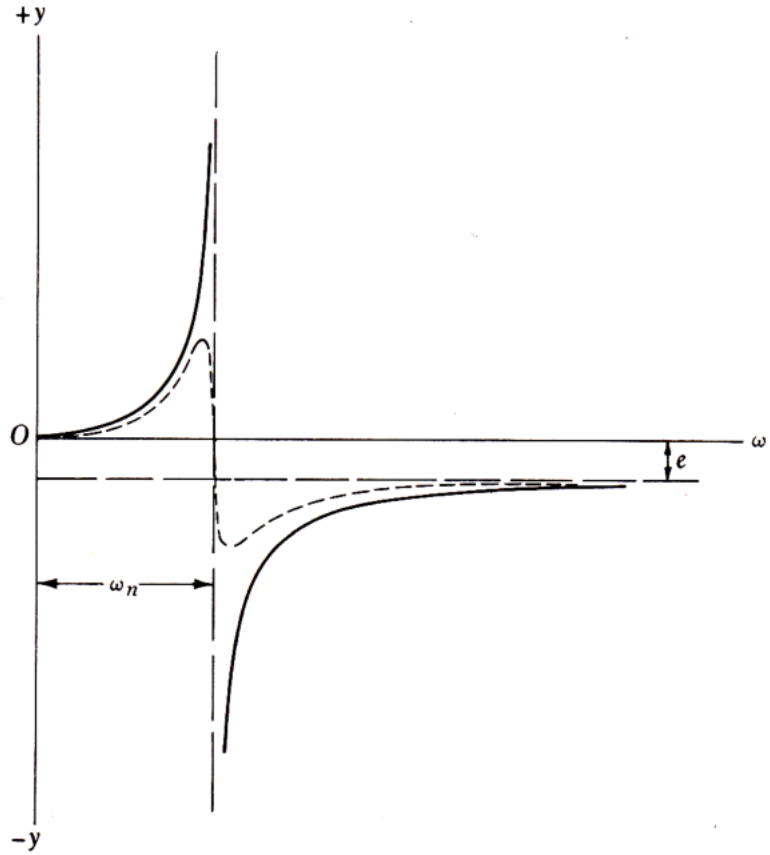
The shaft is rotating and the centrifugal force Fc acts radially outward through G causing the shaft to bend as shown. The centrifugal force is equal to the mass of the disk times the normal acceleration of point G; because the normal acceleration equals the radius of rotation times ω^2

$$F_c = M(y+e)\omega^2$$

Omega is the shaft speed in rad/s and y is the deflection of the shaft where the disk is located.

- The shaft behaves like a spring and for a deflection *y* it exerts a resisting force *ky*, where *k* is the spring constant of the shaft in bending. For equilibrium the resisting force equals the centrifugal force and hence

$$ky = M(y+e)\omega^2 \quad \rightarrow \quad y = \frac{e\omega^2}{(k/M)-\omega^2}$$



Solid curve is the plot of the equation. From the equation we see that when omega is zero, *y* is zero, and when

$$\omega^2 = k / M$$

The denominator becomes zero and *y* becomes infinite. This value of omega is known as the critical speed ω_n

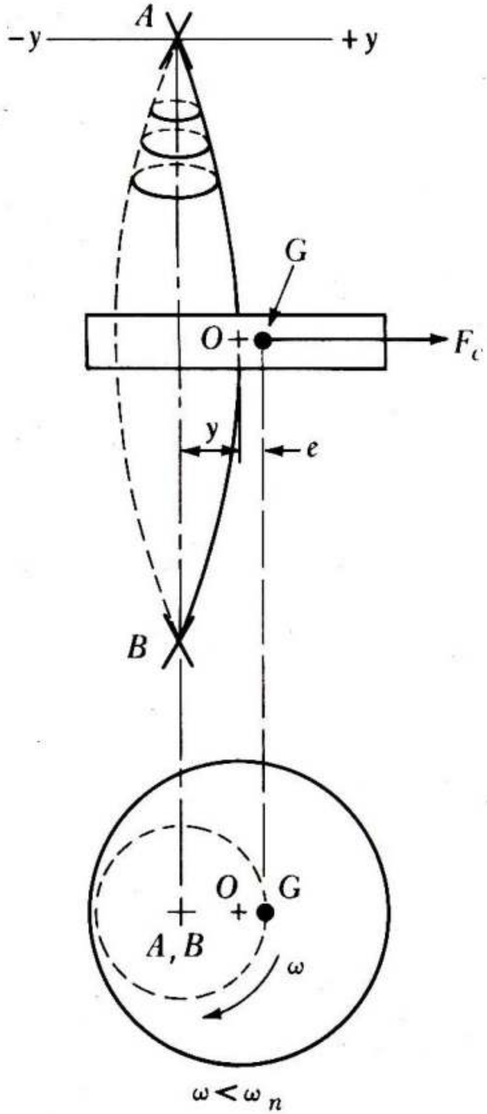
$$\omega_n = \sqrt{\frac{k}{M}} = \sqrt{\frac{kg}{W}}$$

W is the weight of the disk and *g* is the gravity.

$$y = \frac{e\omega^2}{(k / M) - \omega^2}$$

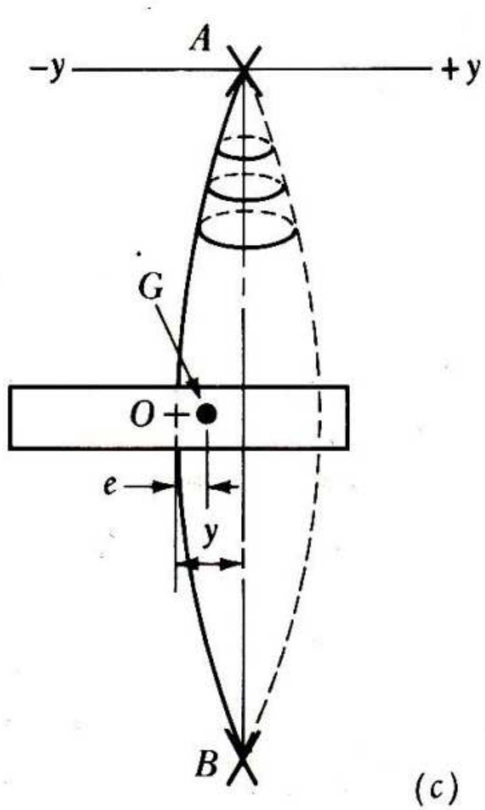
Notice that when $\omega > \omega_n$ the denominator of the equation becomes negative and then y is negative as shown by the right branch of the solid curve. When omega becomes very large y approaches $-e$.

In this analysis damping effect is neglected. Because of the damping the shaft deflection does not become infinite at the critical speed, and a plot of the actual deflection versus omega is indicated by the dotted curve in the plot. The effect of damping alters the critical speed but not appreciably.



(b)

If $\omega < \omega_n$ y will be positive and the configuration of the shaft will be as shown in the figure. If ω is constant, y will be constant, and the centerline of the shaft will remain in fixed bent position whirling around axis AB, describing a surface of revolution which is called whirling.



$\omega > \omega_n$

If $\omega > \omega_n$ y will be negative and hence the shaft bends in the opposite direction. The configuration is then shown in the figure.

It is known from observation that the side of the shaft which is convex below the critical speed becomes the concave side at speed above the critical speed.

For large values of ω, y approaches –e. hence we see that the center of gravity of the disk then approaches the axis of rotation AB. This explains why a rotor operated high above the critical speed operates smoothly.

- At speeds near the critical speed the shaft deflections are large, and from the equation we see that this makes for a large centrifugal force, and hence the forces on the bearings will be large. Because these bearing forces are changing direction with the centrifugal force, the frame of the machine supporting the bearings will be set in vibration.

- Besides the objectionable noise due to vibration, the varying stresses in the bearings and the frame of the machine can result in structural damage. In addition, the large shaft deflections at speeds near the critical speed may cause permanent deformation of the shaft or contact between the rotor and its housing.

- The amplitude of shaft vibration at critical speed reaches dangerous proportions only if time is allowed for the amplitude to buildup; therefore if the machine is accelerated through the critical speed, the magnitudes of the amplitudes can be acceptable. Machines like centrifuges and some high-speed turbines normally operates at speed well above the critical speed and are brought up to their operating speeds by passing quickly through their critical speeds.

$$\omega_n = \sqrt{\frac{k}{M}} = \sqrt{\frac{kg}{W}} = \sqrt{\frac{g}{W/k}} = \sqrt{\frac{g}{y_{st}}}$$

This equation is generally applies to any shaft of negligible weight having a single disk mounted anywhere along the shaft. y_{st} is the static deflection of the shaft at the disk location produced by a force equal to the weight of the disk.

A shaft mounted in ball bearings may be assumed to be simply supported; if it is mounted on journal bearings it may be assumed to be supported as a fixed-end beam.

For a disk of weight *W*, mounted between bearings on a shaft of negligible weight and located distance *a* from the left bearing and a distance *b* from the right bearing and where the bearings are all ball bearings,

$$y_{st} = \frac{a^2 b^2 W}{3EIL} \qquad \omega_n = \sqrt{\frac{g}{y_{st}}} = \sqrt{\frac{3EILg}{a^2 b^2 W}}$$

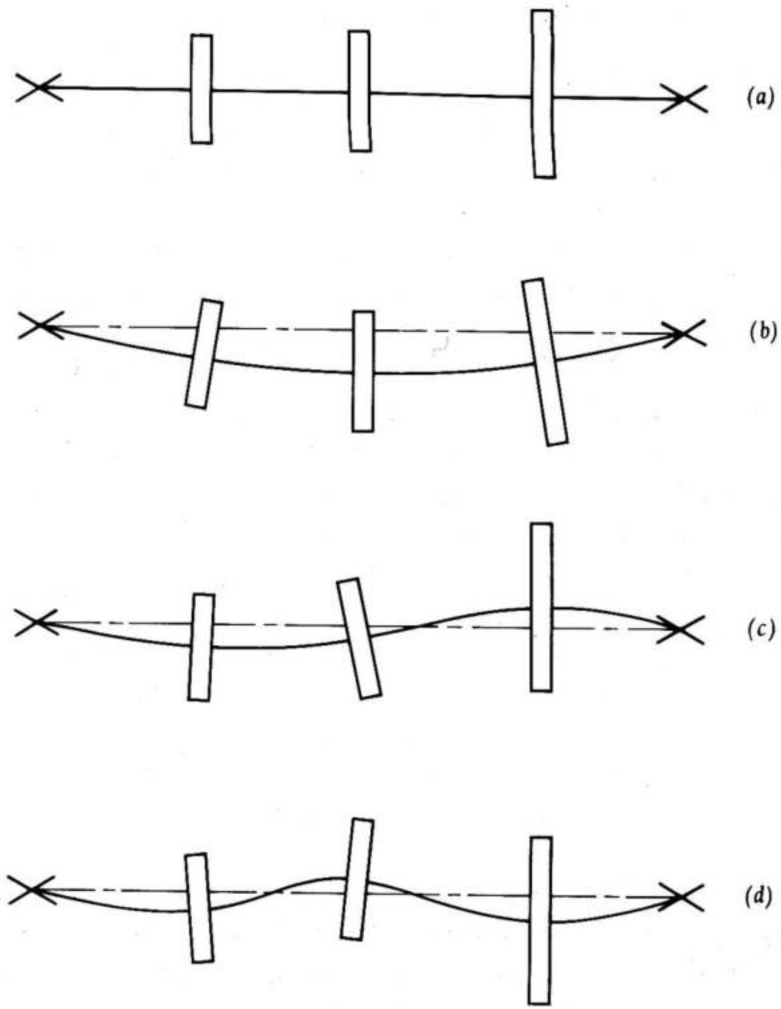
E is the modulus of elasticity, N/m2
I is the moment of inertia of cross sectional area, m4
L is the distance between bearings, m
g is the gravity, m/s2

$$I = \frac{\pi d^4}{64} [\mathrm{m}^4] \qquad d \text{ is shaft diameter} [\mathrm{m}]$$

For the same condition, except using journal bearings (fixed-end boundary condition)

$$y_{st} = \frac{a^3 b^3 W}{3EIL^3} \qquad \omega_n = \sqrt{\frac{g}{y_{st}}} = \sqrt{\frac{3EIL^3 g}{a^3 b^3 W}}$$

SHAFT WITH A NUMBER OF DISKS

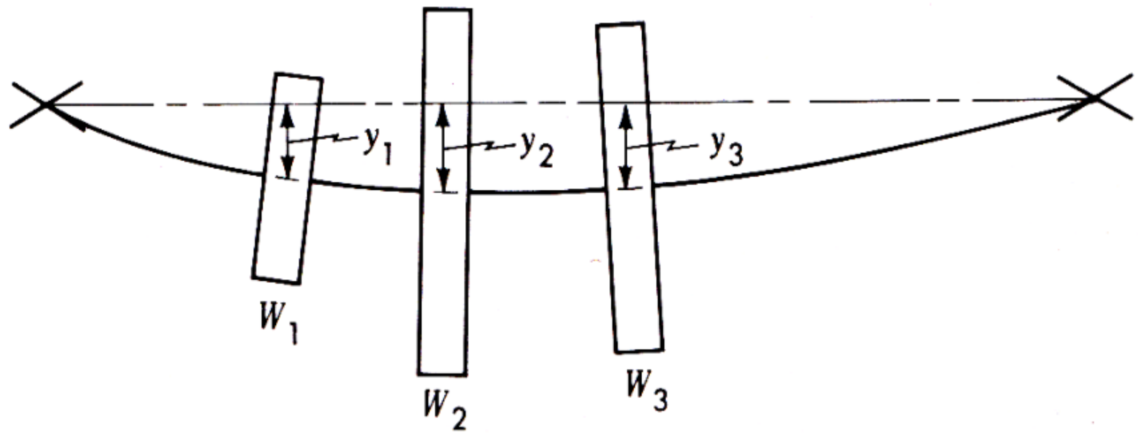


The shaft which has three disks, is representative of a shaft with any number of disks. The shaft may be considered horizontal or vertical because gravity does not effect the critical speed.

The shaft could vibrate transversely in any of the three modes a shown in the figures (b), (c) and (d) and the frequency would be different for each mode. Because the stiffness of the shaft increases as we go from (b) to (c) and to (d), natural frequency increases.

Since the critical speeds correspond to the natural frequencies of transverse vibration, the deflection curves in the figure also represent the configuration of the shafts at its critical speeds. As the number of disks is increased, the number of modes of vibration and the number of critical speeds increases. Usually it is only the lowest critical speed which is of importance because the operating speed of the machine is less than the higher critical speeds.

The fundamental natural frequency of transverse vibration for a shaft with a number of disks can be found by Rayleigh's method. This method is based on the continuous interchange between kinetic and strain energy in the system. Further the method utilizes the assumption that the dynamic deflection curve for the shaft is similar in shape in the static deflection curve.



Consider the shaft in the figure to be of negligible weight and supporting any number of concentrated weights. To illustrate Raleigh's method, let us suppose that there are three weights *W1*, *W2*, and *W3*. The solid line shows the axis of the shaft in its most deflected position as it vibrates transversely with a frequency ω_n. The dynamic deflections *y1*, *y2,* and *y3* may be assumed equal to the static deflections produced by weights *W1, W2* and *W3.*

The kinetic and potential energy for the system is

$$KE = \sum \frac{1}{2} M V^2$$

$$PE = \sum \frac{1}{2} k y^2 = \frac{1}{2} \sum W y \qquad \text{remember } W = ky$$

$$V = y\omega_n \qquad \text{maximum velocity of the disk}$$

$$M = \frac{W}{g} \qquad \text{mass of the disk}$$

$$KE = \frac{1}{2} \frac{\omega_n^2}{g} (W_1 y_1^2 + W_2 y_2^2 + W_3 y_3^2 + \cdots)$$

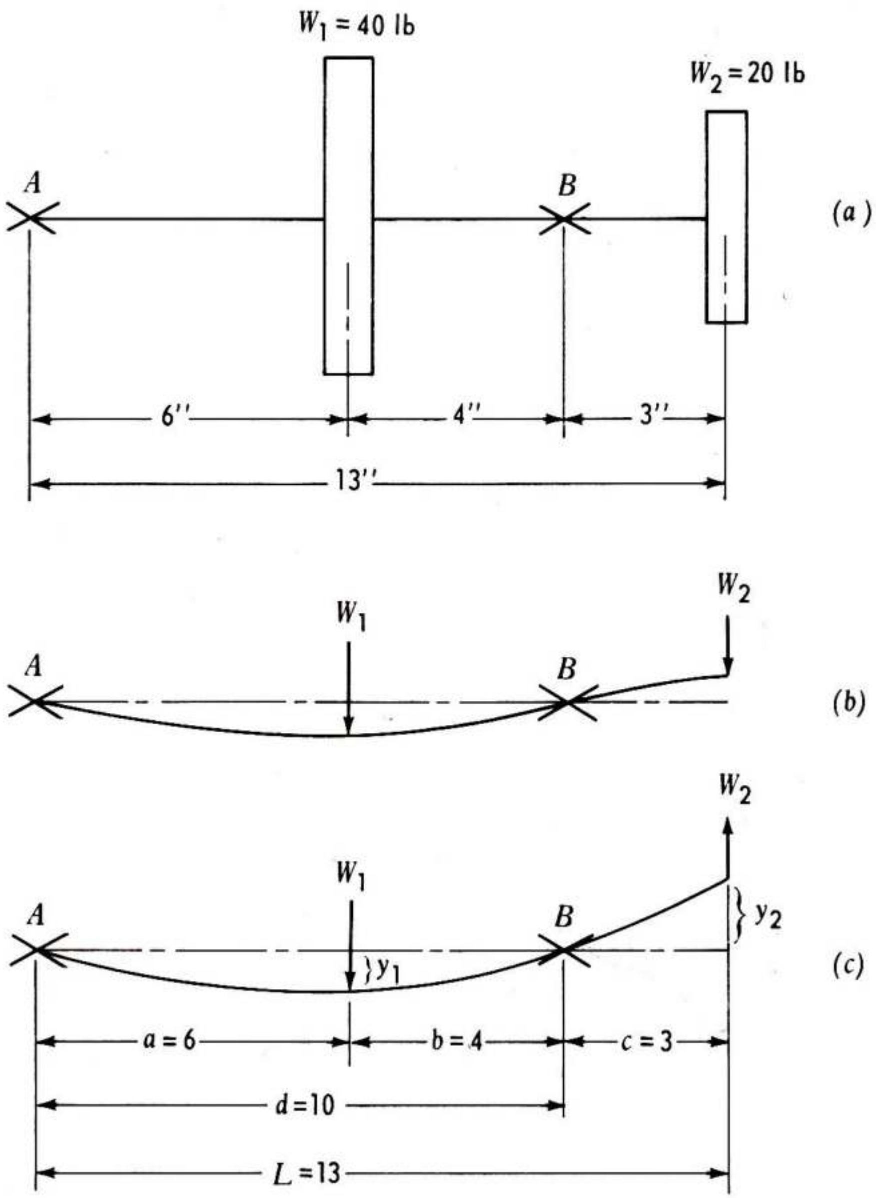
$$PE = \frac{1}{2}(W_1 y_1 + W_2 y_2 + W_3 y_3 + \cdots) \qquad \text{potential energy of the system}$$

$$KE = PE$$

$$\frac{1}{2}\frac{\omega_n^2}{g}(W_1 y_1^2 + W_2 y_2^2 + W_3 y_3^2) = \frac{1}{2}(W_1 y_1 + W_2 y_2 + W_3 y_3)$$

$$\omega_n = \sqrt{\frac{g\sum Wy}{\sum Wy^2}}$$

If the cross section of the shaft is uniform over the length, the static deflections in the equation can be most easily computed by using the equations for beam deflections given in mechanics of materials textbooks and by use of the principle of superposition, which is the deflection at a given point along the shaft is equal to the sum of the deflections at that point as found by considering each load acting by itself.



Consider the shaft in the figure (a) to be simply supported at A and B and to be negligible weight compared with the two concentrated weights *W1* and *W2*.

The shaft is steel and is 25 mm in diameter. It is desired to find the critical speed in rpm by Rayleigh's method. The static deflection is shown in figure (b) but a better approximation to the dynamic deflection curve will be obtained if we use the static deflection curve shown in figure (c) where *W2* is assumed to be acting upward rather than downward.

Deflections for the load *W1* acting alone on the shaft

$$W_1 = 200\text{N}$$
$$W_2 = 100\text{N}$$
$$\text{a} = 15\,\text{cm},\ \text{b} = 10\,\text{cm},\ \text{c} = 7.5\,\text{cm}, \text{L} = 32.5\,\text{cm}$$

$$y_{11} = \frac{W_1 ba}{6dEI}(d^2 - b^2 - a^2) = \frac{200\times 0.1\times 0.15}{6\times 0.25\times EI}(0.25^2 - 0.1^2 - 0.15^2) = \frac{60\times 10^{-3}}{EI}\downarrow$$

$$y_{12} = c\sin\theta_B \approx c\theta_B = \frac{cW_1 ab(d+a)}{6dEI}$$

$$= \frac{0.075\times 200\times 0.15\times 0.1\times(0.25+0.15)}{6\times 0.25\times EI} = -\frac{60\times 10^{-3}}{EI}\uparrow$$

Deflections for the load *W2* acting alone on the shaft

$$y_{21} = \frac{W_2 ca}{6dEI}(a^2 - d^2) = \frac{100 \times 0.075 \times 0.15}{6 \times 0.25 \times EI}(0.15^2 - 0.25^2) = -\frac{30 \times 10^{-3}}{EI} \uparrow$$

$$y_{22} = \frac{W_2 c^2 L}{3EI} = \frac{100 \times 0.075^2 \times 0.325}{3 \times EI} = \frac{61 \times 10^{-3}}{EI} \downarrow$$

By the method of superposition the resultant deflections are

$$y_1 = y_{11} + y_{21} = \frac{60\times10^{-3}}{EI} - \frac{30\times10^{-3}}{EI} = \frac{30\times10^{-3}}{EI} \downarrow$$

$$y_1^2 = \frac{\left(30\times10^{-3}\right)^2}{(EI)^2} = \frac{900\times10^{-6}}{(EI)^2}$$

$$y_2 = y_{12} + y_{22} = -\frac{60\times10^{-3}}{EI} + \frac{61\times10^{-3}}{EI} = \frac{1\times10^{-3}}{EI} \downarrow$$

$$y_2^2 = \frac{\left(1\times10^{-3}\right)^2}{(EI)^2} = \frac{1\times10^{-6}}{(EI)^2}$$

$$\omega_n = \sqrt{\frac{g\sum Wy}{\sum Wy^2}} = \sqrt{\frac{g(W_1 y_1 + W_2 y_2)}{(W_1 y_1^2 + W_2 y_2^2)}}$$

$$= \sqrt{\frac{9.8(200\times 30\times 10^{-3} + 100\times 1\times 10^{-3})EI}{(200\times 900\times 10^{-6} + 100\times 1\times 10^{-6})}} = 18.22\sqrt{EI}$$

$$I = \frac{\pi d^4}{64} = \frac{\pi 0.025^4}{64} = 19.16\times 10^{-9}\,\text{m}^4 \qquad E = 200\,\text{GPa}$$

$$\omega_n = 18.22\sqrt{200\times 10^9 \times 19.16\times 10^{-9}} = 1127.87\ \text{rad/s}$$

Shafts with variable diameter

Rayleigh's equation

$$\omega_n = \sqrt{\frac{g\sum Wy}{\sum Wy^2}}$$

Rayleigh's equation ca be applied to find the critical speed of any rotor shaft if the static deflections are known. Mathematical determinations of these deflections becomes very tedious if there are many loads and especially so if the shaft has different diameters along its length. A combined numerical and graphical procedure for determining the deflections is then convenient.

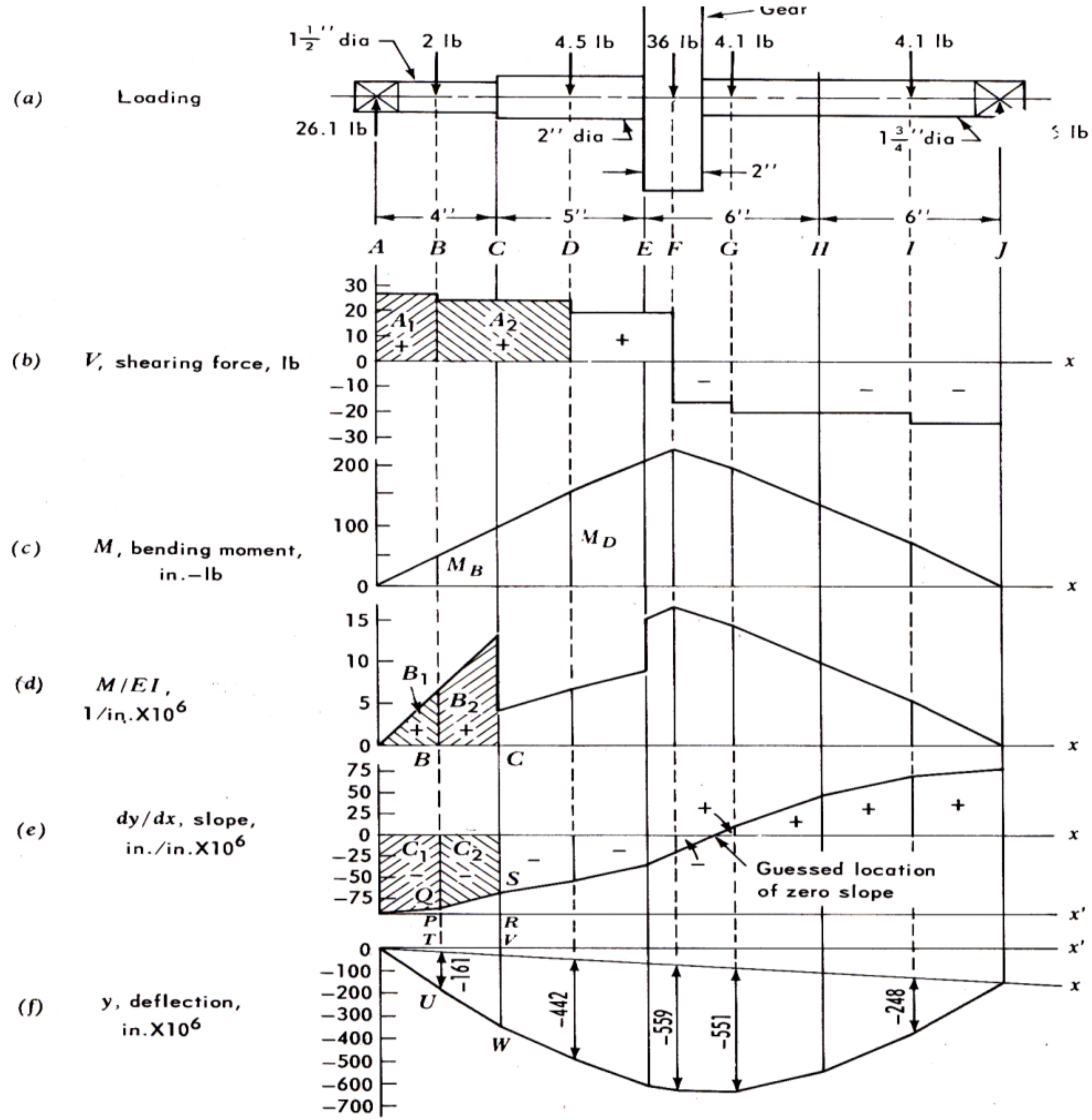
Texts on mechanic of materials show that the basic differential equation which must be solved in order to find the static deflection is

$$\frac{d^2y}{dx^2} = \frac{M}{EI}$$

y = deflection at any distance x from one end of shaft, m
M = bending moment at distance x, Nm
E = modulus of elasticity, N/m2
I = moment of inertia of cross sectional area at distance x, m4

Integrating the equation once we obtain the slope of the elastic curve of the shaft and integrating again, we obtain the deflection curve. Further, starting with the loading, one integration gives the shearing force and a second integration gives the bending moment. Hence starting with the loads, four successive integrations are needed to obtain the deflections.

example



(a) Loading
(b) V, shearing force, lb
(c) M, bending moment, in.–lb
(d) M/EI, 1/in.X10^6
(e) dy/dx, slope, in./in.X10^6
(f) y, deflection, in.X10^6
Gear
2 lb
4.5 lb
36 lb
4.1 lb
4.1 lb
26.1 lb
2'' dia
2''
4''
5''
6''
6''
A B C D E F G H I J
Guessed location of zero slope

Consider the shaft in figure (a) to be of steel and simply supported in bearings as shown. The shaft supports a gear weighing 36 lb; it is desired to find the static deflection curve.

The shaft is divided into a number of segments, and in order to facilitate the integration of the areas under the various curves, they are drawn to scale using any convenient scale along the *x* axis and any convenient scales for coordinates.

In figure (a) the loads are shown. The load at pin B is the weight of the shaft from A to C and the load at D is the weight of the shaft from C to E, etc. By statics the bearing reactions were found to be as shown. The shearing force diagram can then be drawn and appears in figure (b). Next the bending moment diagram is constructed. Since bending moment is the integral of shearing force, M_B equals area A_1, which is 26.1 lb, multiplied by length AB, which is 2 in. Area A_2 is equal to 24.1 lb multiplied by length BD, which is 4.5 in. M_D equals the sum areas A_1 and A_2. Note that through the entire procedure, areas above the *x* axis are considered positive, and those below negative.

The moment of inertia of the cross sectional area for each shaft diameter shown in figure (a) is computed next. Then ordinates for the *M/EI* diagram are obtained by dividing the ordinates in the moment diagram by the value of *E*, which is 30x10^6 psi for steel, and by the value of *I* for the corresponding cross section. Note that at section C the shaft from the bending moment diagram and is then divided by each of the values of *I*, in order to determine the two values of *M/EI* at this section. Similarly, since the shaft diameter changes at *E*, there will be a step in the *M/EI* diagram at this section.

The slope diagram in figure (e) is constructed next by laying off ordinates from an axis x'. Since slope is the integral of *M/EI*, ordinate PQ is equal to area B_1, which equals M_B/EI_B divided by 2 and multiplied by length AB, which is 2 in. PQ is laid off upward from the x' axis because area B_1 is positive. Ordinate RS is equal to area B_1 plus area B_2. we continue in this manner until the entire slope curve is drawn. The ordinates from the x' axis to this curve would represent the true values of slope along the deflection curve if the slope occurs somewhere between the bearings. We can guess the location (distance *x*) of the point of zero slope and in figure (e) draw the *x* axis so that it intersects the slope diagram curve at this location. Then areas lying below the x axis are considered negative and those above positive.

The deflection diagram in figure (f) is constructed by laying off ordinates from an axis x'. Since deflection is the integral of slope, ordinate TU equals area C_1, and because C_1 is a negative area, TU is negative and thus is drawn downward. Ordinate VW equals area C_1 plus area C_2. We continue in this manner laying off ordinates from the x' axis until the entire curve is drawn. At the bearings the deflections must be zero and thus through these points on the deflection curve the *x* axis is drawn as shown. The ordinates from the *x* axis to the deflection curve are then the actual deflections.

Jazan University
Mechanical Engineering Department

CHAPTER 14

HYDRAULIC SYSTEMS

EngM271 Theory of Machines

Dr. Amr Assie

Spring 2019

- Definitions
- Hydraulic circuits
- Hydraulic pumps
- Control valves
- Actuators
- Basic calculations
- Examples of hydraulic circuits
- Pneumatic systems

definitions

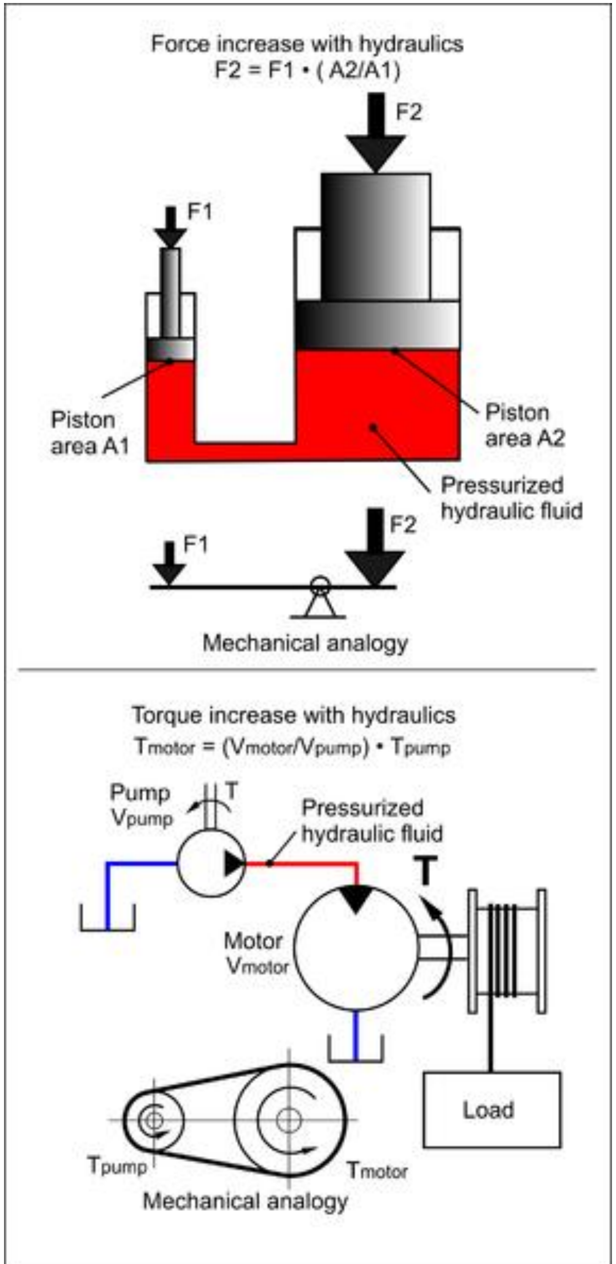
- **Hydraulic machines** are machinery and tools that use hydraulic fluid power to do a simple work.
- Heavy equipment is a common example. In this type of machines, hydraulic fluid is transmitted throughout the machine to various hydraulic motors and hydraulic cylinders which becomes pressurized according to the resistance present. The fluid is controlled manually or automatically by control valves and distributed through hoses and tubes.
- The popularity of hydraulic machinery is due to the very large amount of power that can be transferred through small tubes and flexible hoses, and the high power density and wide array of actuators that can make use of this power.

Excavator

Hydraulic Crane

Force and torque multiplication

A fundamental feature of hydraulic systems is the ability to apply force or torque multiplication in an easy way, independent of the distance between the input and output, without the need for mechanical gears or levers, either by altering the effective areas in two connected cylinders or the effective displacement (cc/rev) between a pump and motor.



Hydraulic transmission

$$\text{pressure} \quad p = \frac{F_1}{A_1} = \frac{F_2}{A_2} \qquad\qquad F_2 = F_1 \frac{A_2}{A_1}$$

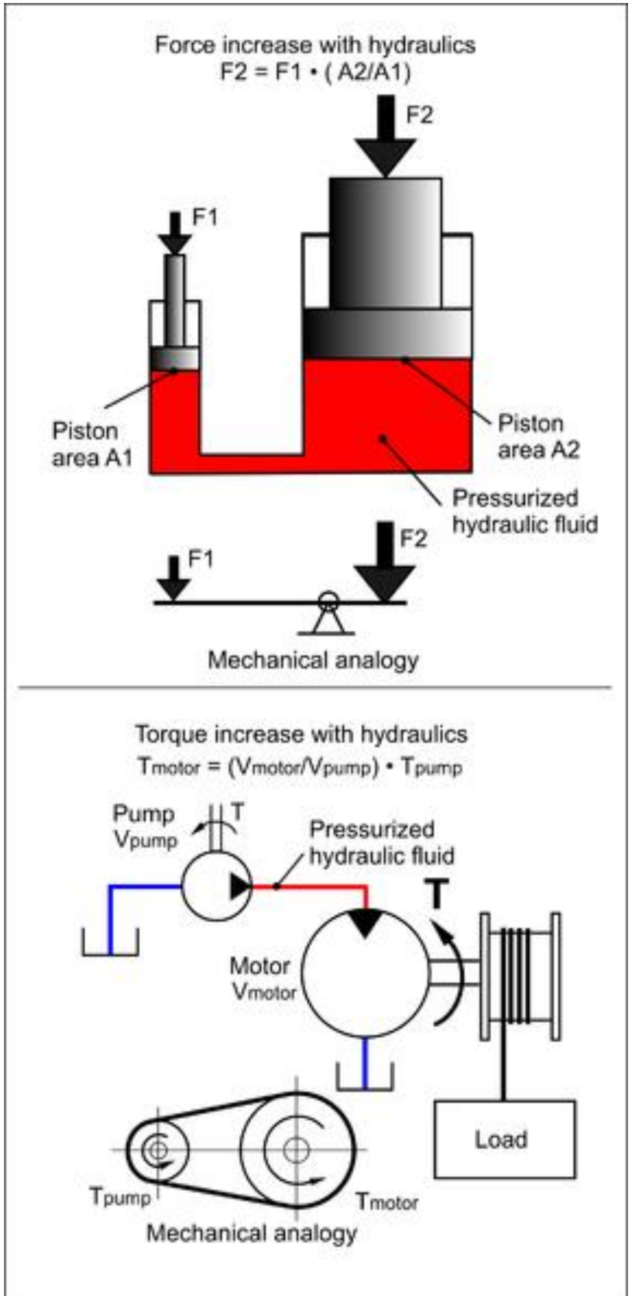
$$A_1 = \pi R_1^2 \qquad\qquad A_2 = \pi R_2^2$$

$$\text{assume} \quad R_2 = 10 R_1$$

$$F_2 = F_1 \frac{A_2}{A_1} = F_1 \frac{\pi R_2^2}{\pi R_1^2} = F_1 \frac{R_2^2}{R_1^2} = F_1 \frac{(10 R_1)^2}{R_1^2} = 100 F_1$$

Volume of oil $\qquad A_1 y_1 = A_2 y_2$

$$y_1 = \frac{A_2}{A_1} y_2 = 100 y_2$$



Hydraulic rotational transmission

$$\text{power } \; T_m \omega_m = T_p \omega_p$$

$$\text{flow rate } \; \mathrm{Q} = \omega_m S_m = \omega_p S_p$$

$$\frac{T_m}{T_p} = \frac{\omega_p}{\omega_m} = \frac{S_m}{S_p}$$

assume motor and pump displacements as

$$S_m = 100\,\mathrm{cm}^3/\mathrm{rad} \quad \text{and} \quad S_p = 10\,\mathrm{cm}^3/\mathrm{rad}$$

$$T_m = \frac{S_m}{S_p} T_p = \frac{100}{10} T_p = 10 T_p$$

$$\omega_m = \frac{S_p}{S_m} \omega_p = \frac{10}{100} \omega_p = \frac{\omega_p}{10}$$

T_m : motor torque, Nm

T_p : pump torque, Nm

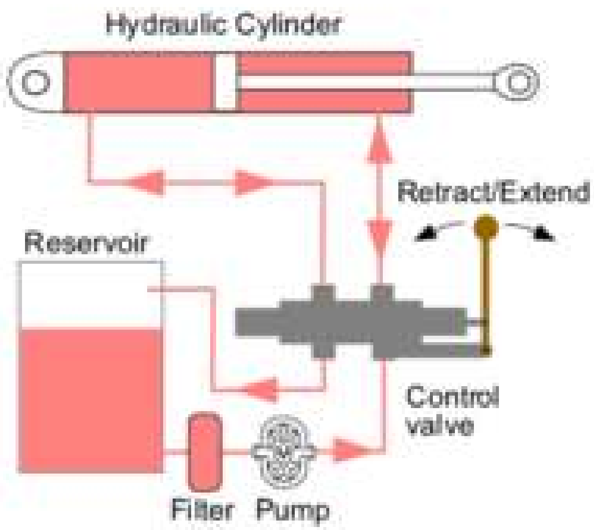
ω_p : pump speed, rad/s

ω_m : motor speed, rad/s

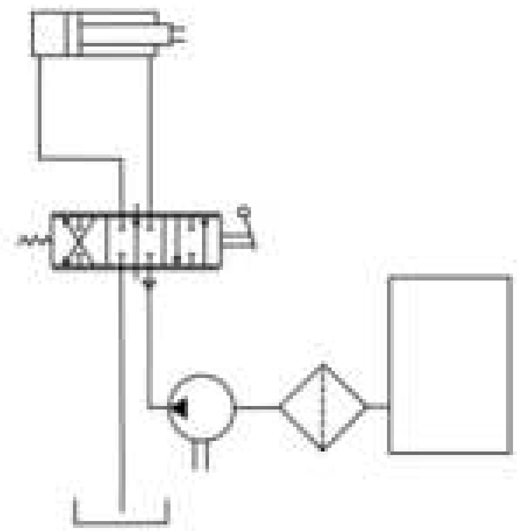
S_p : pump displacement, m^3/rad

S_m : motor displacement, m^3/rad

Hydraulic circuits



A simple *open center* hydraulic circuit



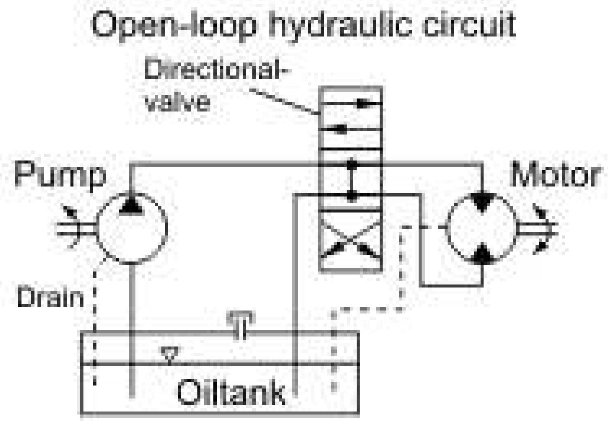
The equivalent schematic circuit

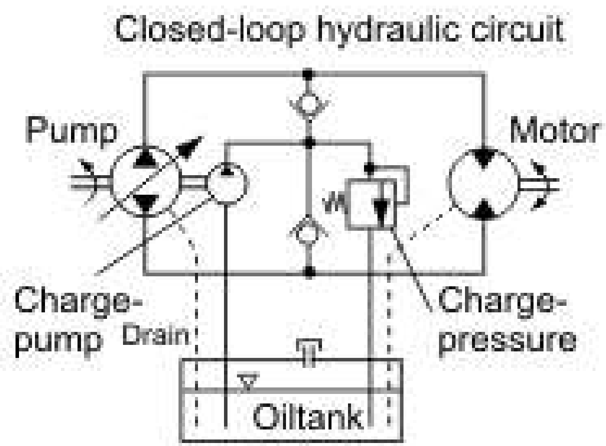
Hydraulic circuits

For the hydraulic fluid to do work, it must flow to the actuator and or motors, then return to a reservoir. The fluid is then filtered and re-pumped. The path taken by hydraulic fluid is called a **hydraulic circuit** of which there are several types.

Open center circuits use pumps which supply a continuous flow. The flow is returned to *tank* through the control valve's *open center*; that is, when the control valve is centered, it provides an open return path to tank and the fluid is not pumped to a high pressure. If the control valve is actuated it routes fluid to and from an actuator and tank. The fluid's pressure will rise to meet any resistance, since the pump has a constant output. If the pressure rises too high, fluid returns to tank through a pressure relief valve. Multiple control valves may be stacked in series. This type of circuit can use inexpensive, constant displacement pumps.

Closed center circuits supply full pressure to the control valves, whether any valves are actuated or not. The pumps vary their flow rate, pumping very little hydraulic fluid until the operator actuates a valve. The valve's spool therefore doesn't need an open center return path to tank. Multiple valves can be connected in a parallel arrangement and system pressure is equal for all valves.





Open-loop circuit: Pump-inlet and motor-return (via the directional valve) are connected to the hydraulic tank.

Closed-loop circuit: Motor-return is connected directly to the pump-inlet. To keep up pressure on the low pressure side, the circuits have a charge pump (a small gear pump) that supplies cooled and filtered oil to the low pressure side.

Hydraulic pumps

Hydraulic pumps supply fluid to the components in the system. Pressure in the system develops in reaction to the load. Hence, a pump rated for 30 MPa is capable of maintaining flow against a load pressure of 30 MPa.

Pumps have a power density about ten times greater than an electric motor (by volume). They are powered by an electric motor or an engine, connected through gears, belts, or a flexible elastomeric coupling to reduce vibration.

Common types of hydraulic pumps to hydraulic machinery applications are;

Gear pump: cheap, durable, simple. Less efficient, because they are constant (fixed) displacement, and mainly suitable for pressures below 20 MPa.

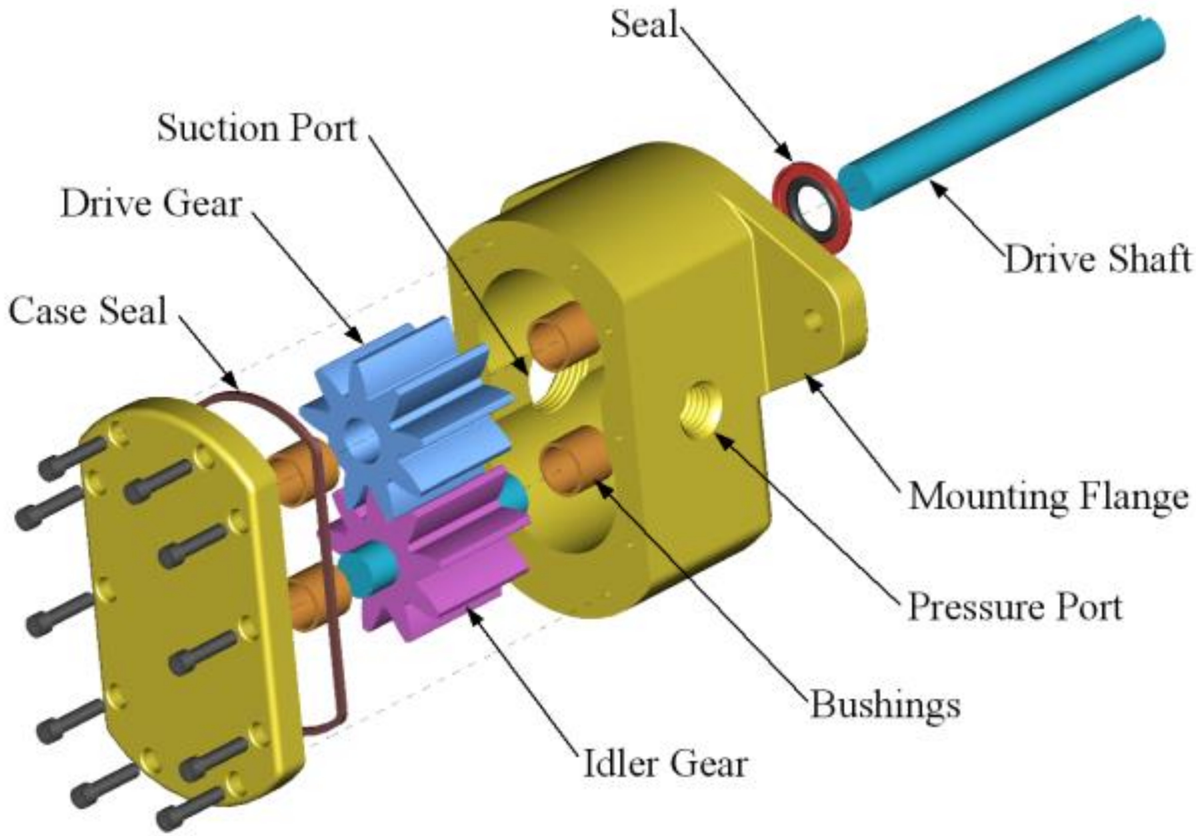
Vane pump: cheap and simple, reliable. Good for higher-flow low-pressure output.

Axial piston pump: many designed with a variable displacement mechanism, to vary output flow for automatic control of pressure. There are various axial piston pump designs. The most common is the swash plate pump. A variable-angle swash plate causes the pistons to reciprocate a greater or lesser distance per rotation, allowing output flow rate and pressure to be varied (greater displacement angle causes higher flow rate, lower pressure, and vice versa).

Radial piston pump is a pump that is normally used for very high pressure at small flows.

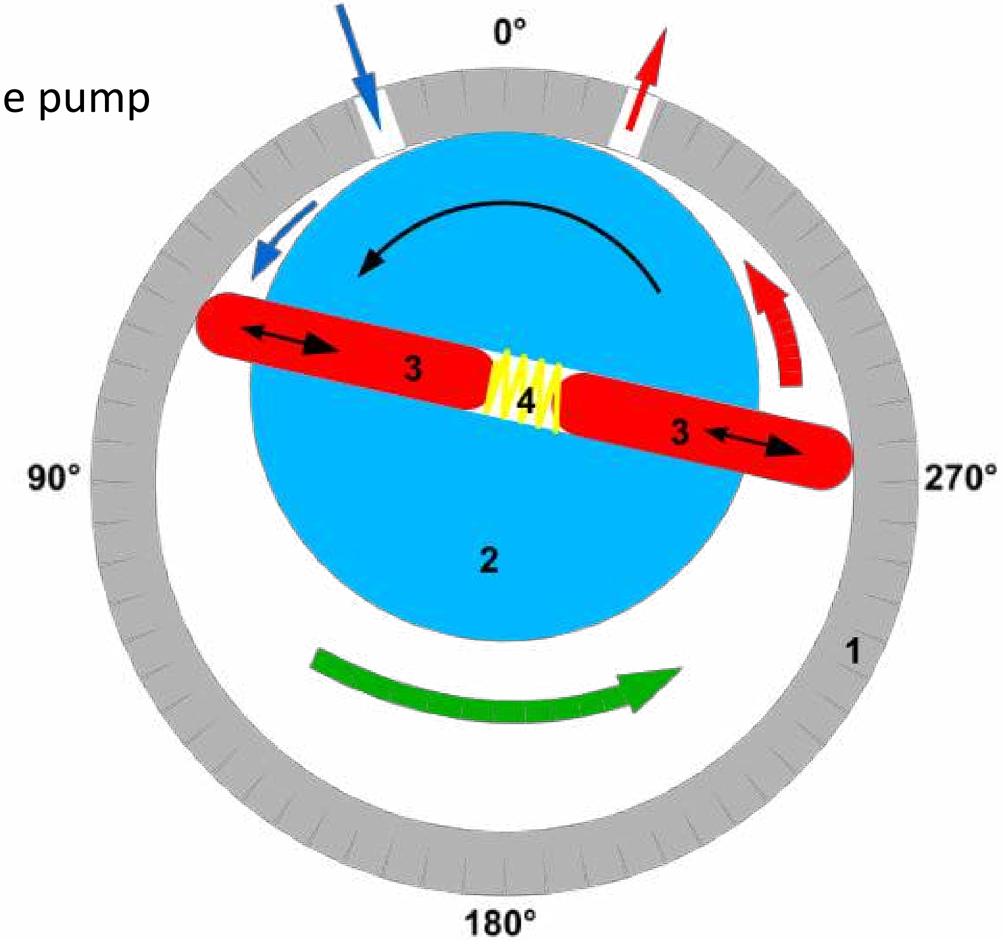
Piston pumps are more expensive than gear or vane pumps, but provide longer life operating at higher pressure, with difficult fluids and longer continuous duty cycles.

exploded view of a gear pump



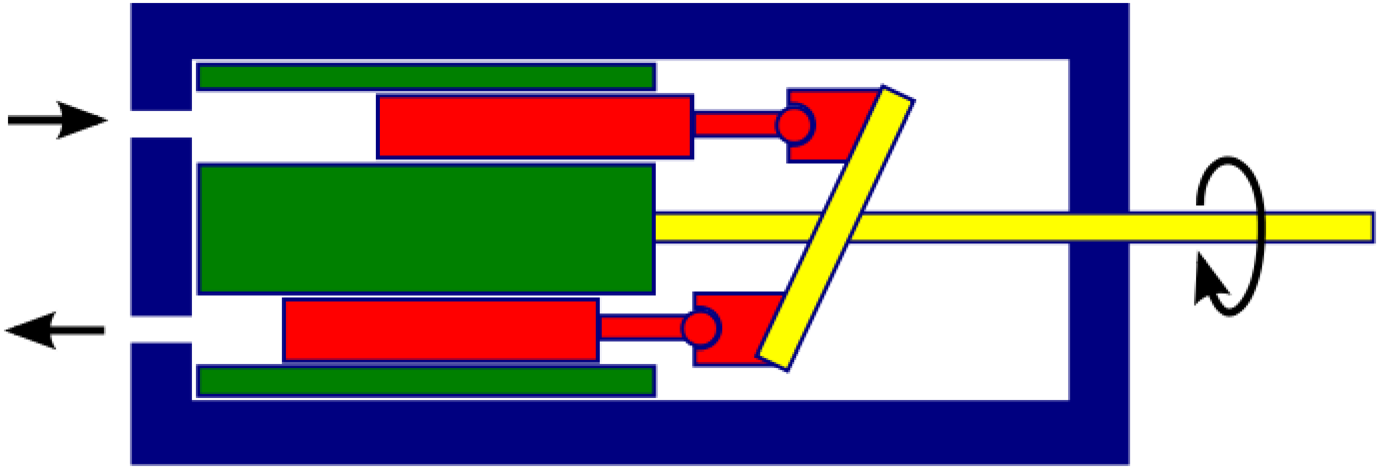
Gear pump: cheap, durable, simple. Less efficient, because they have constant (fixed) displacement, and mainly suitable for pressures below 20 MPa.

Rotary vane pump



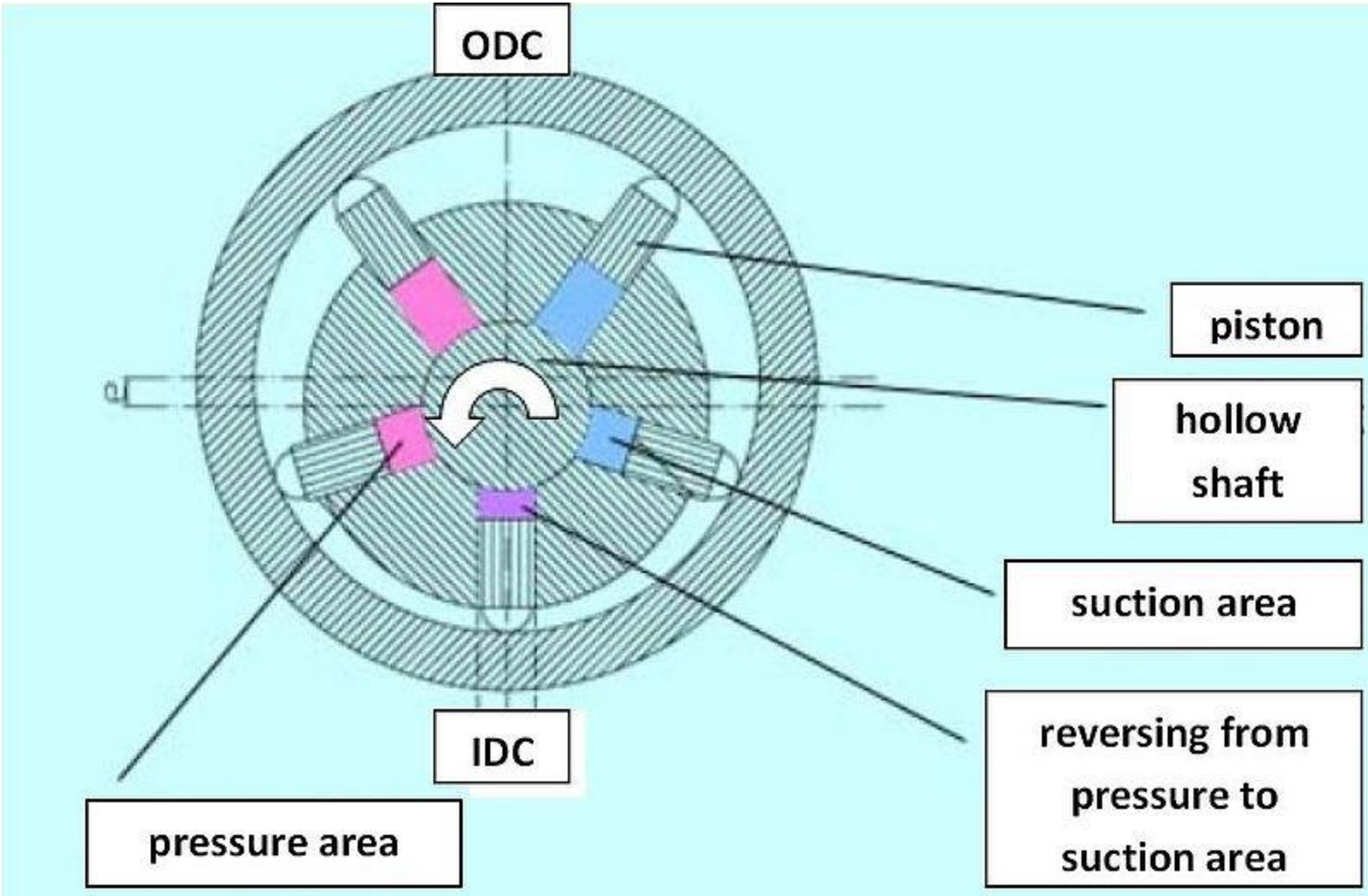
1. body
2. rotor
3. piston valve
4. spring

Vane pump is cheap and simple, reliable. Good for higher-flow low-pressure output.

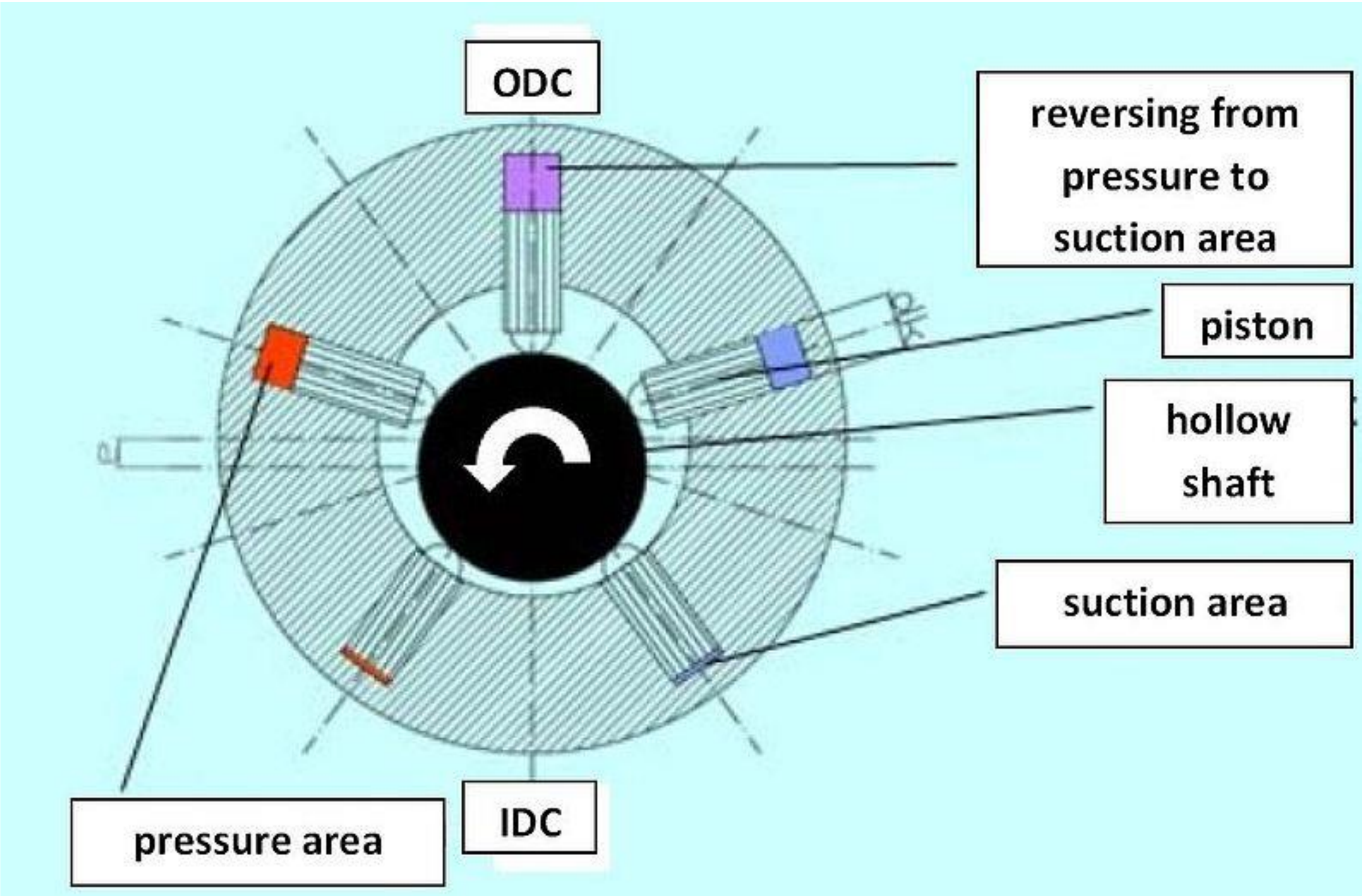


Axial Piston Pump

Inside impinged radial piston pump



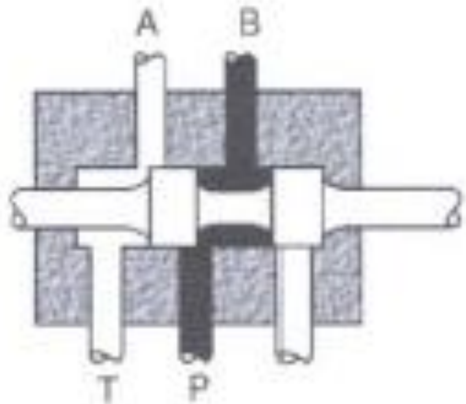
outside impinged radial piston pump



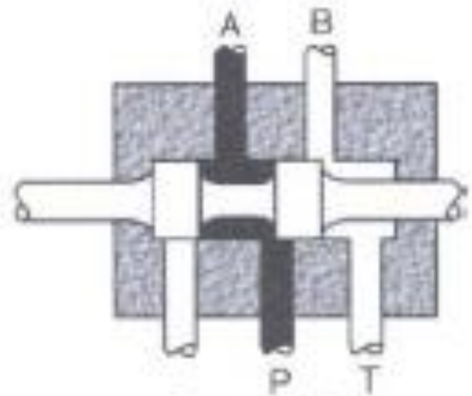
Control valves

manual control valves

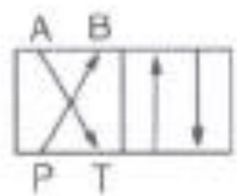
Solenoid valves



Pressure to B;
A to tank



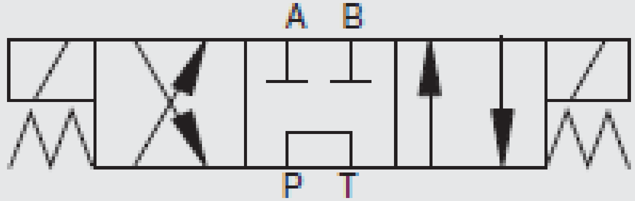
Pressure to A;
B to tank



4 way / 3 position / TANDEM Center

CENTER POSITION: Pessure to Tank

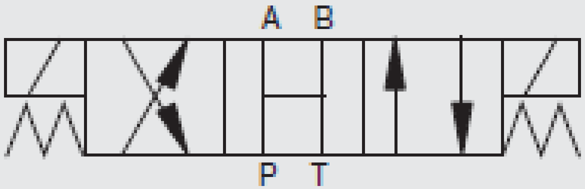
USES: Idles pump in the center position



4 way / 3 position / OPEN Center

CENTER POSITION: All ports to Tank

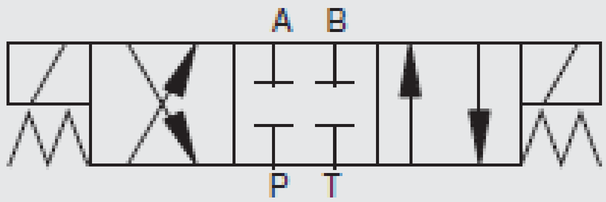
USES: Idles pump in the center position
commonly used with pilot operated check valve



4 way / 3 position / CLOSED Center

CENTER POSITION: All ports blocked

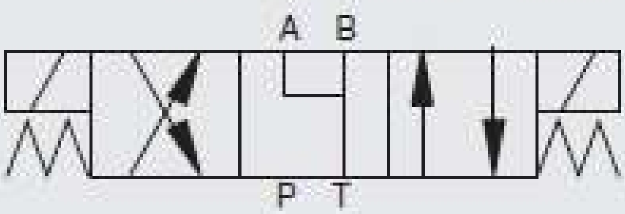
USES: Used in system with multiple valves



4 way / 3 position / FLOAT Center

CENTER POSITION: A & B to to Tank, P blocked

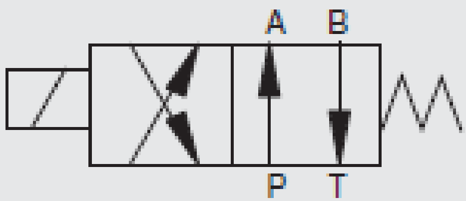
USES: Used in system with mulitple valves with pilot operated check valves



4 way / 2 position

NO CENTER POSITION

USES: Used in systems where cylinders are always either advanced or retracted



Control valves

Directional control valves route the fluid to the desired actuator. They usually consist of a spool inside a cast iron or steel housing. The spool slides to different positions in the housing, intersecting grooves and channels route the fluid based on the spool's position.

The spool has a central (neutral) position maintained with springs; in this position the supply fluid is blocked, or returned to tank. Sliding the spool to one side routes the hydraulic fluid to an actuator and provides a return path from the actuator to tank. When the spool is moved to the opposite direction the supply and return paths are switched. When the spool is allowed to return to neutral (center) position the actuator fluid paths are blocked, locking it in position.

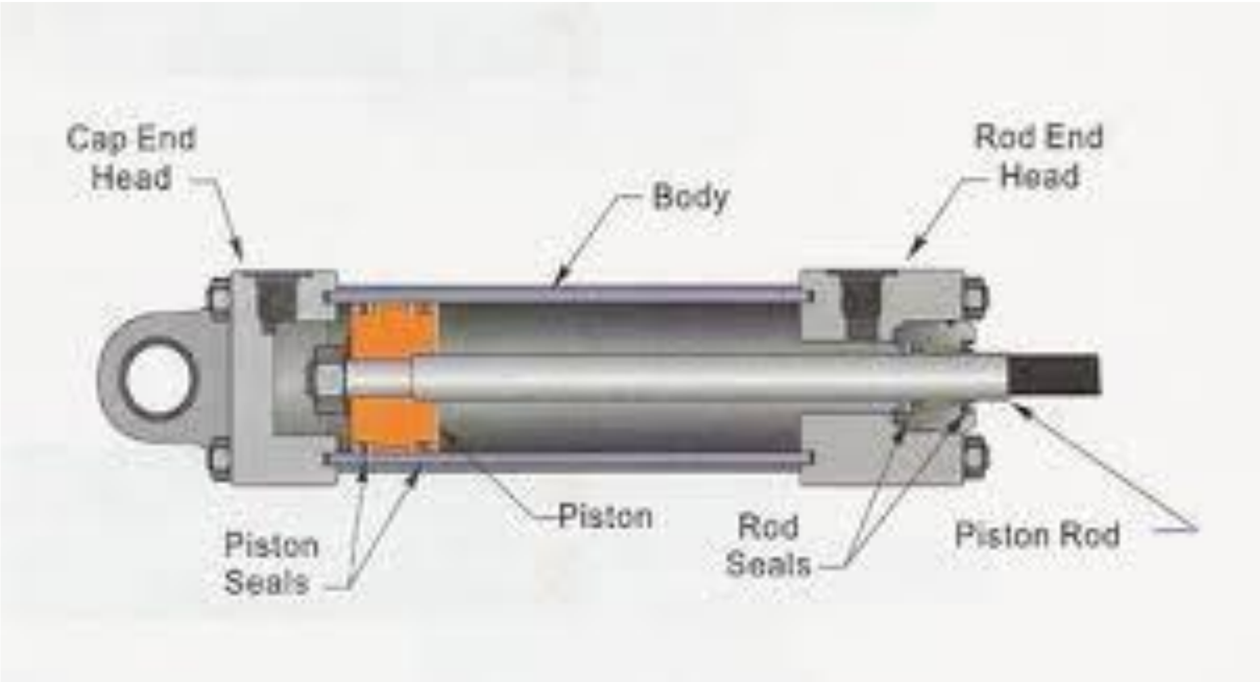
Directional control valves are usually designed to be stackable, with one valve for each hydraulic cylinder, and one fluid input supplying all the valves in the stack.

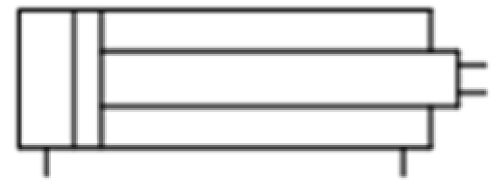
Tolerances are very tight in order to handle the high pressure and avoid leaking, spools typically have a clearance with the housing of less than 25 µm. The valve block will be mounted to the machine's frame with a *three point* pattern to avoid distorting the valve block and jamming the valve's sensitive components.

The spool position may be actuated by <u>mechanical levers</u>, <u>hydraulic pilot pressure</u>, or <u>solenoids</u> which push the spool left or right. A seal allows part of the spool to protrude outside the housing, where it is accessible to the actuator.

The main valve block is usually a stack of off the shelf directional control valves chosen by flow capacity and performance. Some valves are designed to be **proportional** (flow rate proportional to valve position), while others may be simply **on-off.** The control valve is one of the most expensive and sensitive parts of a hydraulic circuit.

Actuators



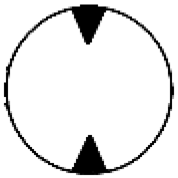


A cut-away of hydraulic cylinder

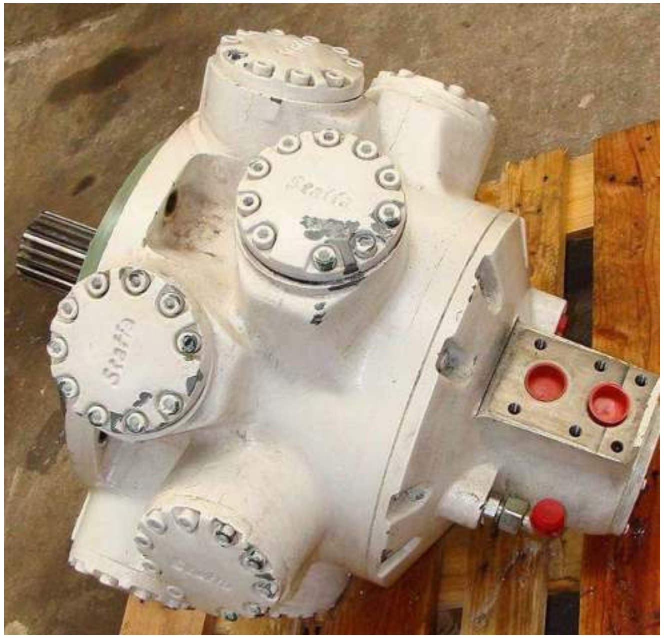
A **Hydraulic cylinder** (also called a **linear hydraulic motor**) is a mechanical actuator that is used to give a unidirectional force through a unidirectional stroke.

Hydraulic cylinders get their power from pressurized hydraulic fluid, which is typically oil. The hydraulic cylinder consists of a cylinder barrel, in which a piston connected to a piston rod moves back and forth. The barrel is closed on each end by the cylinder bottom (also called the cap end) and by the cylinder head where the piston rod comes out of the cylinder. The piston has sliding rings and seals. The piston divides the inside of the cylinder in two chambers, the bottom chamber (cap end) and the piston rod side chamber (rod end). The hydraulic pressure acts on the piston to do linear motion.

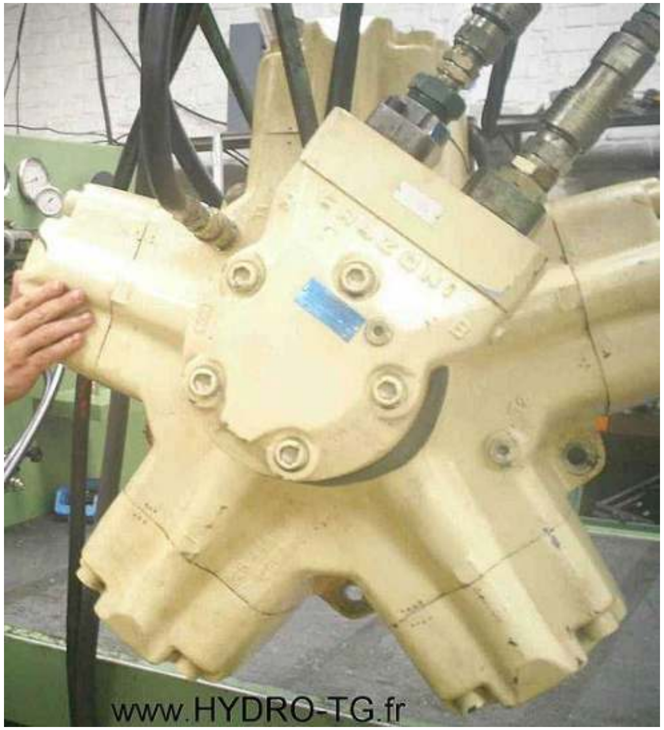
An hydraulic cylinder is the actuator or "motor" side of this system. The "generator" side of the hydraulic system is the hydraulic pump which brings in a fixed or regulated flow of oil to the bottom side of the hydraulic cylinder, to move the piston rod upwards. The piston pushes the oil in the other chamber back to the reservoir.



Symbol of hydraulic motor



Hydraulic motor Staffa



Hydraulic motor Calzone

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

Reservoir

The hydraulic fluid reservoir holds excess hydraulic fluid to accommodate volume changes from cylinder extension and contraction, temperature driven expansion and contraction, and leaks. The reservoir is also designed to aid in separation of air from the fluid and also work as a heat accumulator to cover losses in the system when peak power is used. Design engineers are always pressured to reduce the size of hydraulic reservoirs, while equipment operators always appreciate larger reservoirs. Reservoirs can also help separate dirt and other particulate from the oil, as the particulate will generally settle to the bottom of the tank.

Accumulator

Accumulators are a common part of hydraulic machinery. Their function is to store energy by using pressurized gas. One type is a tube with a floating piston. On one side of the piston is a charge of pressurized gas, and on the other side is the fluid. Examples of accumulator uses are backup power for steering or brakes, or to act as a shock absorber for the hydraulic circuit.

Hydraulic fluid

Also known as *tractor fluid*, hydraulic fluid is the life of the hydraulic circuit. It is usually petroleum oil with various additives. Some hydraulic machines require fire resistant fluids, depending on their applications. In some factories where food is prepared, either an edible oil or water is used as a working fluid for health and safety reasons.

In addition to transferring energy, hydraulic fluid needs to lubricate components, suspend contaminants and metal filings for transport to the filter, and to function well to several hundred degrees Fahrenheit or Celsius.

Filter

Filters are an important part of hydraulic systems. Metal particles are continually produced by mechanical components and need to be removed along with other contaminants.

Filters may be positioned in many locations. The filter may be located between the reservoir and the pump intake. Blockage of the filter will cause cavitation and possibly failure of the pump. Sometimes the filter is located between the pump and the control valves. This arrangement is more expensive, since the filter housing is pressurized, but eliminates cavitation problems and protects the control valve from pump failures. The third common filter location is just before the return line enters the reservoir. This location is relatively insensitive to blockage and does not require a pressurized housing, but contaminants that enter the reservoir from external sources are not filtered until passing through the system at least once.

Hydraulic tubes are seamless steel precision pipes, specially manufactured for hydraulics. The tubes have standard sizes for different pressure ranges, with standard diameters up to 100 mm. The tubes are supplied by manufacturers in lengths of 6 m, cleaned, oiled and plugged. The tubes are interconnected by different types of flanges (especially for the larger sizes and pressures), welding cones/nipples (with o-ring seal), several types of flare connection and by cut-rings. In larger sizes, hydraulic pipes are used. Direct joining of tubes by welding is not acceptable since the interior cannot be inspected.

Hydraulic pipe is used in case standard hydraulic tubes are not available. Generally these are used for low pressure. They can be connected by threaded connections, but usually by welds. Because of the larger diameters the pipe can usually be inspected internally after welding.

Hydraulic hose is graded by pressure, temperature, and fluid compatibility. Hoses are used when pipes or tubes can not be used, usually to provide flexibility for machine operation or maintenance. The hose is built up with rubber and steel layers. A rubber interior is surrounded by multiple layers of woven wire and rubber. The exterior is designed for abrasion resistance. The bend radius of hydraulic hose is carefully designed into the machine, since hose failures can be deadly, and violating the hose's minimum bend radius will cause failure. Hydraulic hoses generally have steel fittings swaged on the ends. The weakest part of the high pressure hose is the connection of the hose to the fitting. Another disadvantage of hoses is the shorter life of rubber which requires periodic replacement, usually at five to seven year intervals.

Basic Calculations

The fluid pushes against the face of the piston and produces a force. The force is

$$F = pA$$

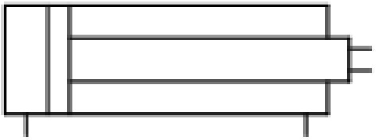
P is the pressure in N/m2=Pa , *A* is the area the pressure acts on in m2.

$$F = pA$$

Force on the full area of the piston

$$F = p(A - a)$$

Force on the rod side of the piston, *a* is the cross sectional area of the rod.



$$flow\ rate\,[m^3/s] = Area\left[m^2\right] \times Velocity\left[m/s\right]$$

$$Q = A\text{v}$$ Full side

$$Q = (A-a)\text{v}$$ Rod side

$$Power = Force\left[N\right] \times velocity\left[m/s\right] = F\text{v}\left[Watt\right]$$

$$Power = pressure[N/m^2] \times Flowrate[m^3/s] = pQ[Watt]$$

$$F\text{v} = pQ$$

Example

A double acting hydraulic cylinder has a bore (diameter) of 100 mm. The rod is 40 mm in diameter and the stroke is 120 mm. It must produce a pushing force of 12 kN. The flow rate available in both directions is 12 liter/min. Calculate;

a) The system pressure needed
b) The force it pulls with same pressure
c) The speed on the outward stroke
d) The speed on the retraction
e) The power used

$$A = \frac{\pi D^2}{4} = \frac{\pi 0.1^2}{4} = 7.85 \times 10^{-3} m^2$$

a) $$p = \frac{F}{A} = \frac{12000}{7.85 \times 10^{-3}} = 1.53 \times 10^6 \frac{N}{m^2} = 1.53 MPa$$

$$a = \frac{\pi d^2}{4} = \frac{\pi 0.04^2}{4} = 1.26 \times 10^{-3} m^2$$

b)

$$F = p(A-a) = 1.53\times10^{6}(7.85\times10^{-3} - 1.26\times10^{-3})$$

$$= 1.53\times10^{6}\times6.59\times10^{-3} = 10.08\times10^{3}\,N \cong 10\,\mathrm{kN}$$

c)

$$Q = \frac{12\,[\mathrm{l/min}]\times10^{-3}}{60\,[\mathrm{s}]} = 200\times10^{-6}\,\mathrm{m^3/s}$$

$$v = \frac{Q}{A} = \frac{200\times10^{-6}}{7.85\times10^{-3}} = 0.0255 = 25.5\,\mathrm{mm/s}$$

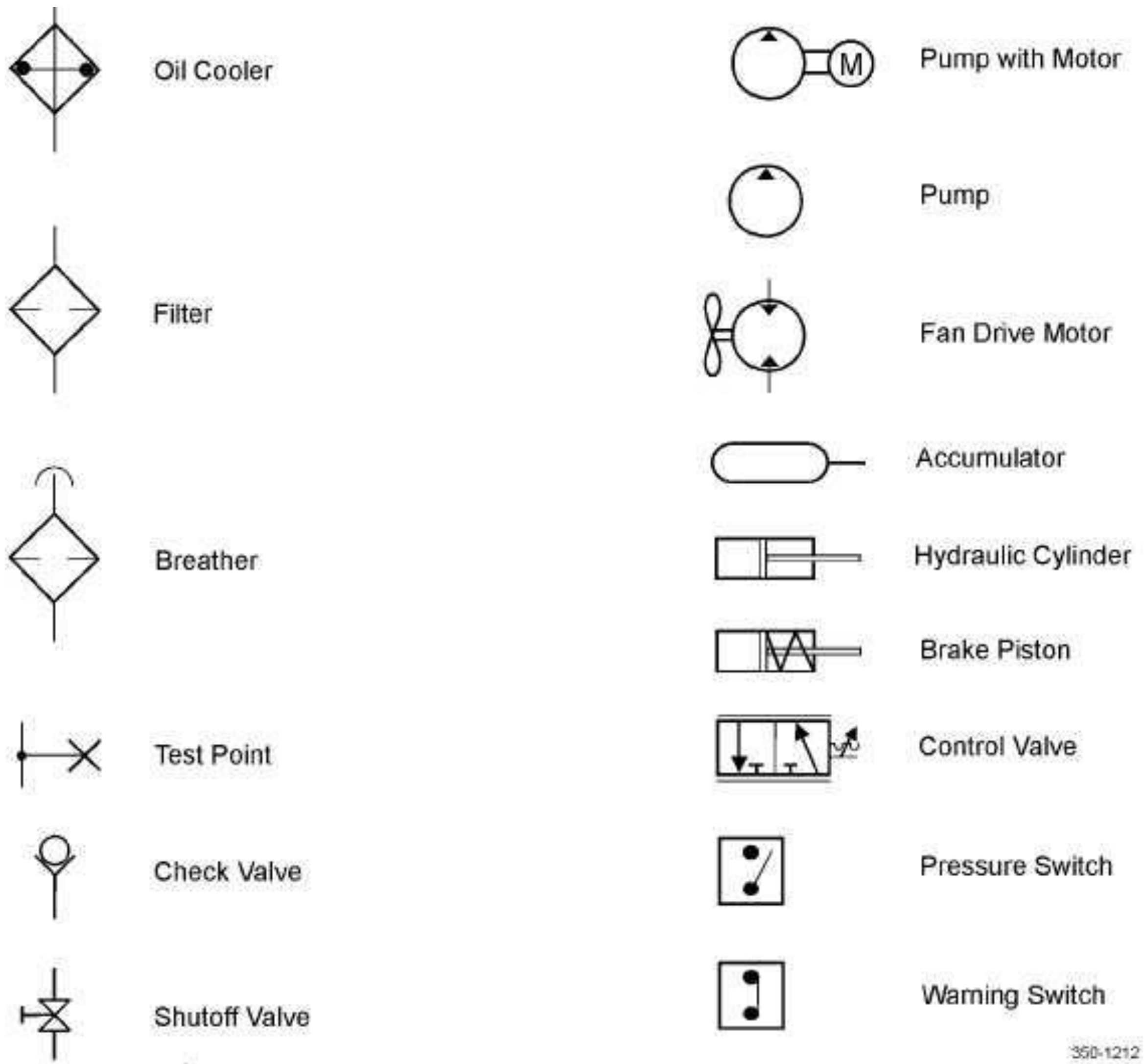
d)

$$v = \frac{Q}{(A-a)} = \frac{200\times10^{-6}}{6.59\times10^{-3}} = 0.030 = 30\,\mathrm{mm/s}$$

e)

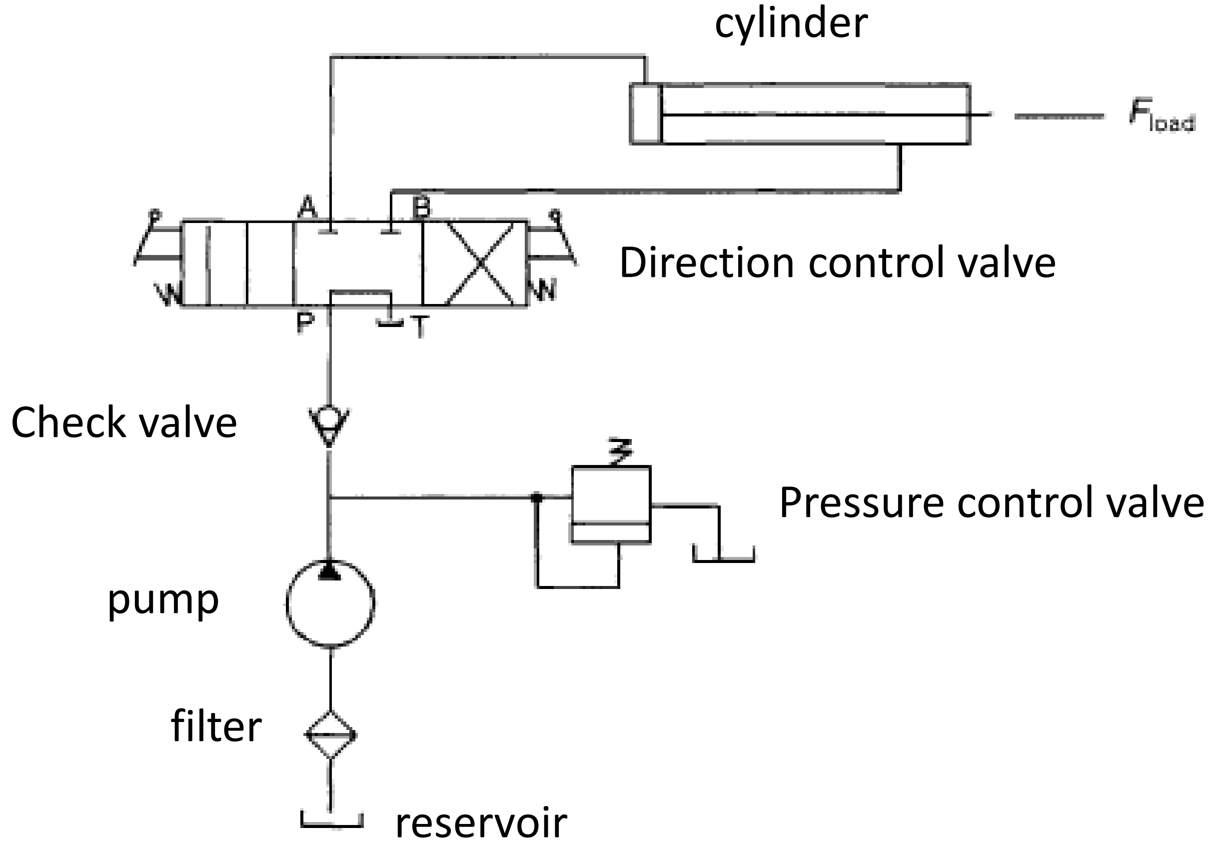
$$P = pQ = 1.53\times10^{6}\times200\times10^{-6} = 306\ \mathrm{W}$$

$$P = F\mathrm{v} = 12000\times0.0255 = 306\ \mathrm{W}$$



Some symbols of a hydraulic system

Example circuit



Control of a double acting hydraulic cylinder

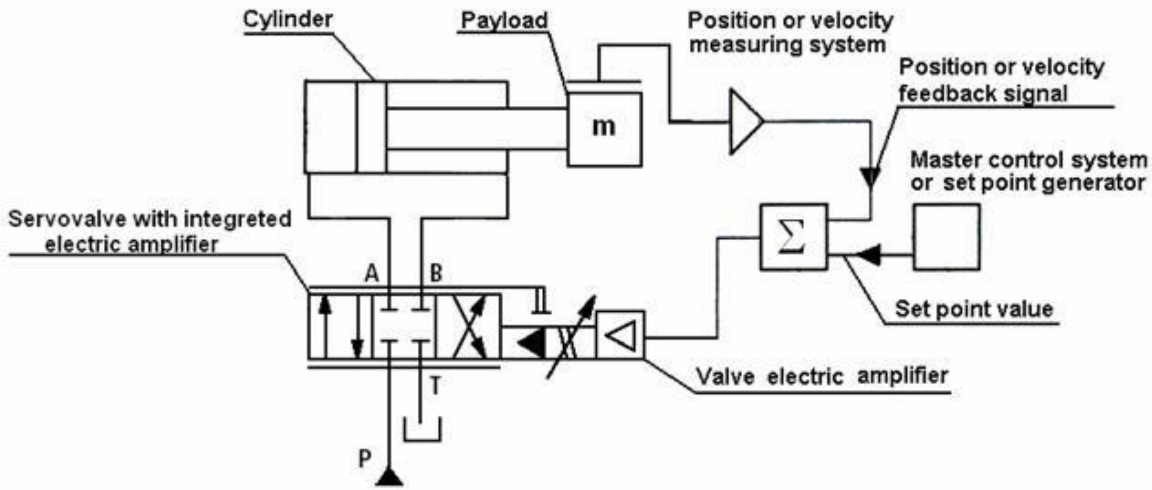
When the four-way valve is in its spring-centered position, the cylinder is hydraulically locked. Also the pump is loaded back to the tank at atmospheric pressure. When the four-way valve is actuated into the flow path configuration of the left envelope, the cylinder is extended against its force load. As oil flows from port P through port A. The oil at the rod end of the cylinder is free to flow back into the reservoir through the four-way valve from port B through port T. The cylinder will not extend if the oil in the rod end is not allowed to flow back to the reservoir.

When the four-way valve is de-activated, the spring-centered envelope prevails, and the cylinder is once again hydraulically locked.

When the four-way valve is actuated in the right envelope configuration, the cylinder retracts, as oil flows from port P through port B. Oil in the blank end is allowed to flow back to the reservoir from port A through port T of the four-way valve. At the end of the stroke, there is no system demand for oil. Therefore the pump flow goes through the relief valve at its set pressure, unless the four-way valve is de-activated. In any event, the system is protected from cylinder overloads.

The check valve prevents the load from retracting the cylinder, while it is being extended using the left envelope flow path configuration.

Example of a hydraulic circuit



Hydraulic Fitness Equipment makes use of hydraulic cylinders instead of adjustable weight increments or free weights. Resistance to the motion is determined by the amount of effort applied and the adjustment sizing of the orifice. This design allows the resistance to be familiarized to an individual's strength level.

hydraulic scraper

Hydraulic Crane

Hydraulic Press

Hydraulic Jack

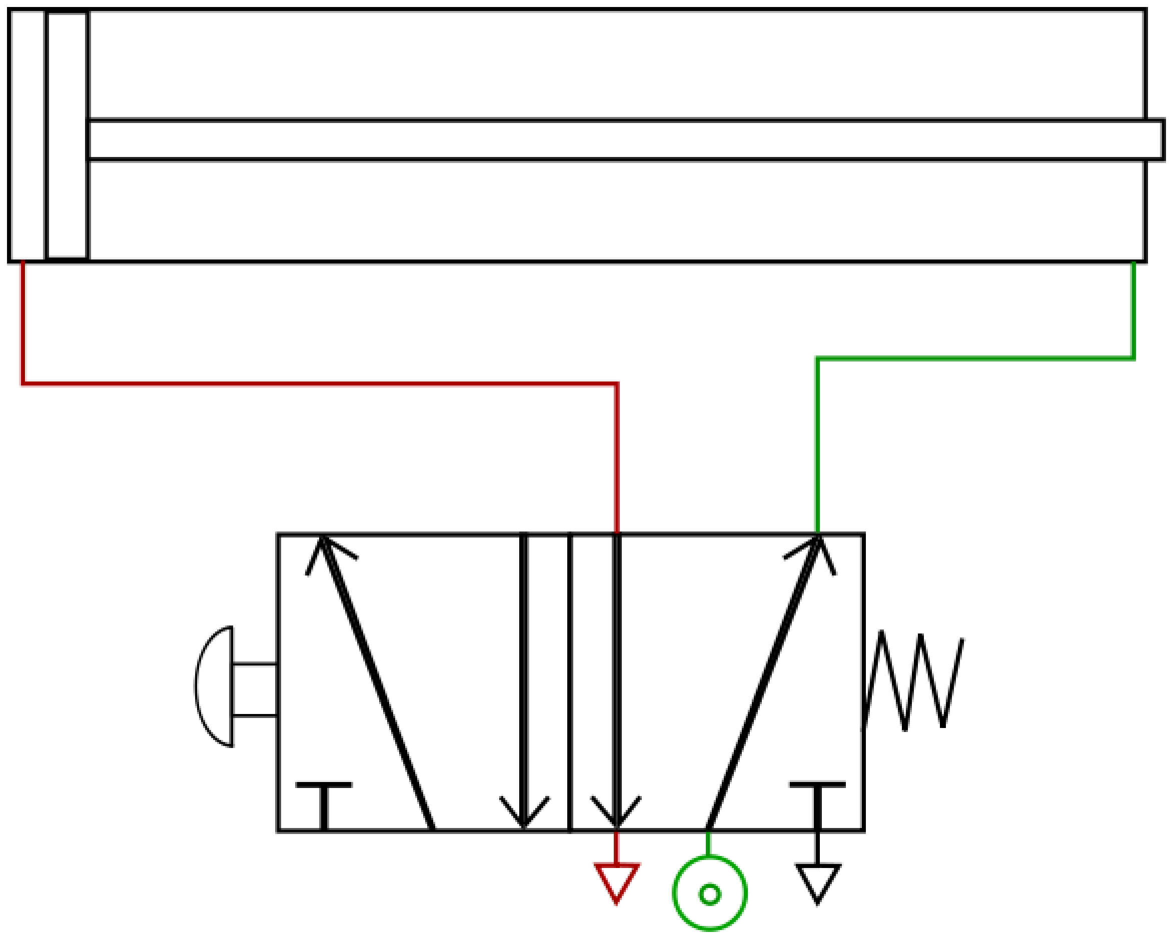
Hydraulic Lift

PNEUMATIC SYSTEMS

Pneumatics is a branch of technology, which deals with the study and application of use of pressurized gas to effect mechanical motion.

Pneumatic systems are extensively used in industry, where factories are commonly plumbed with compressed air or compressed inert gases. This is because a centrally located and electrically powered compressor that powers cylinders and other pneumatic devices through solenoid valves is often able to provide motive power in a cheaper, safer, more flexible, and more reliable way than a large number of electric motors and actuators.

Pneumatics also has applications in dentistry, construction, mining, and other areas.



Both pneumatics and hydraulics are applications of fluid power. Pneumatics uses an easily compressible gas such as air or a suitable pure gas, while hydraulics uses relatively incompressible liquid media such as oil. Most industrial pneumatic applications use pressures of about (0.5 to 0.7 MPa). Hydraulics applications commonly use from 7 to 35 MPa, but specialized applications may exceed 70MPa.

Advantages of pneumatics:

Simplicity of Design And Control

Machines are easily designed using standard cylinders & other components.

Machines operate by simple ON - OFF type control.

Reliability

Pneumatic systems tend to have long operating lives and require very little maintenance.

Because gas is compressible, the equipment is less likely to be damaged by shock. The gas in pneumatics absorbs excessive force, whereas the fluid of hydraulics directly transfers force.

Storage

Compressed gas can be stored, allowing the use of machines when electrical power is lost.

Safety

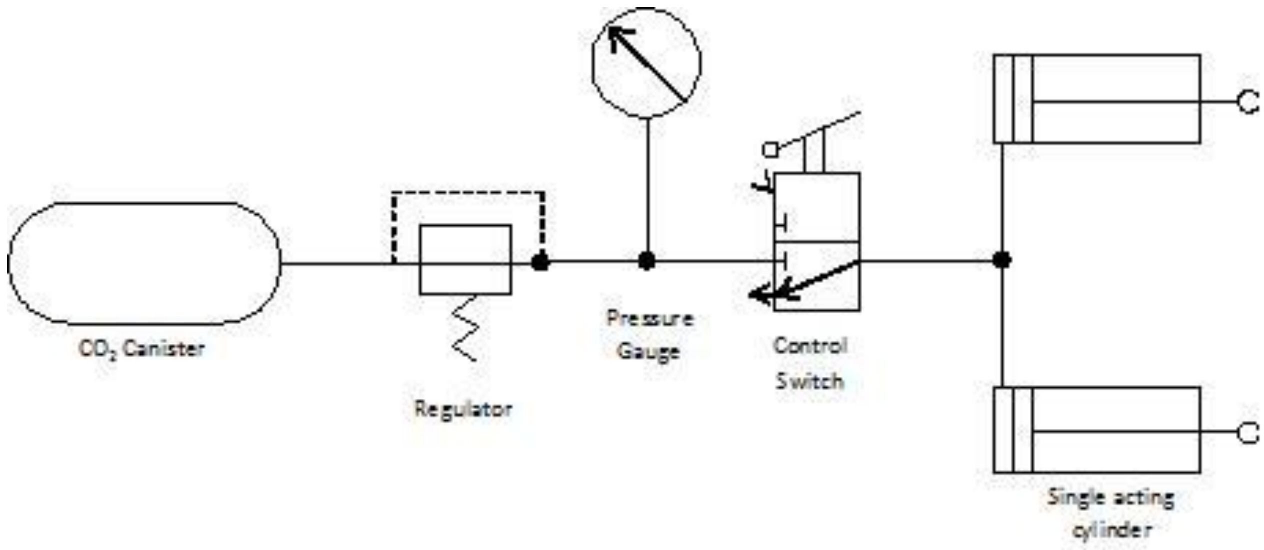
Very low chance of fire (compared to hydraulic oil).

Machines can be designed to be overload safe.

Advantages of hydraulics

Liquid does not absorb any of the supplied energy.

Capable of moving much higher loads and providing much higher forces due to the incompressibility.



CO_2 Canister
Regulator
Pressure Gauge
Control Switch
Single acting cylinder