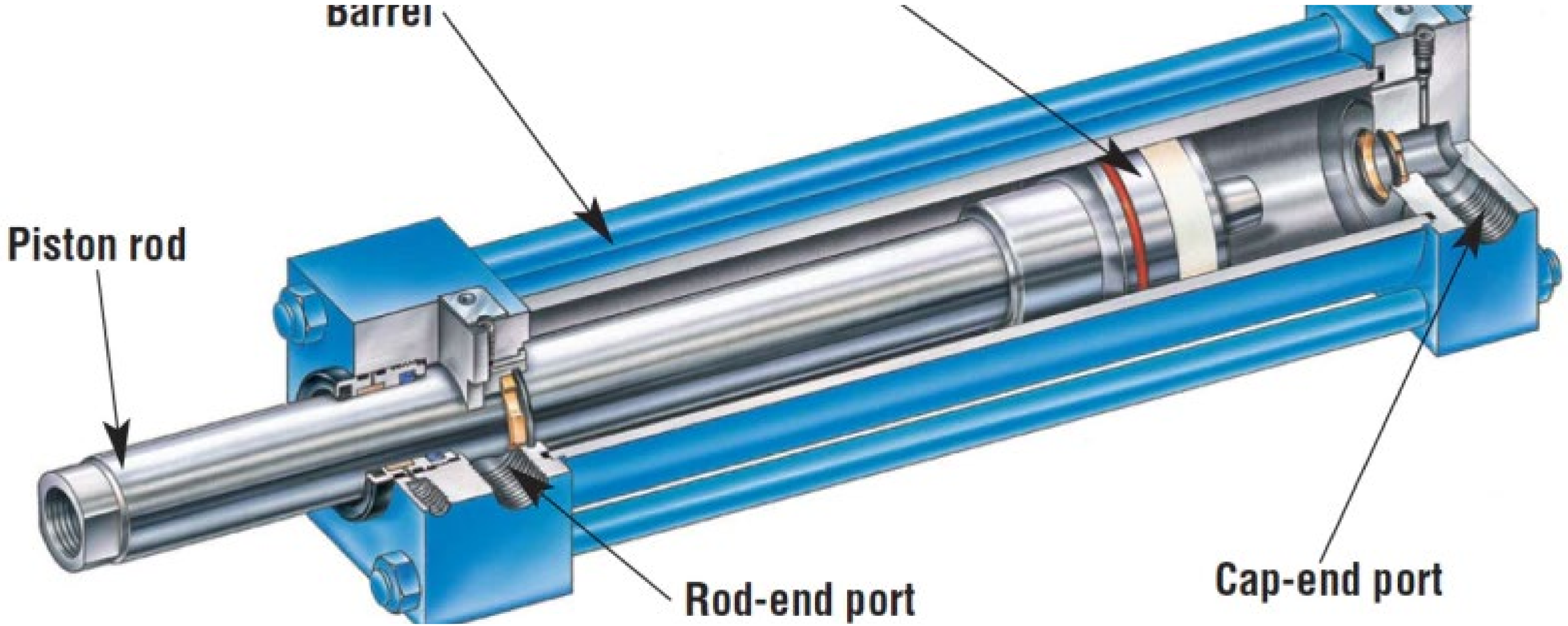


Pneumatics and hydraulics

Hydraulic Cylinders and Cushioning Devices

Dr. Ahmad Al-Mahasneh

Introduction

- Hydraulic cylinders and hydraulic motors extract energy from the fluid and convert it to mechanical energy to perform useful work.
- *Hydraulic cylinders* (also called *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.

- *hydraulic motors* (also called *rotary actuators*) rotate a shaft to provide a torque to drive the load along a rotary path.

Single acting hydraulic cylinder

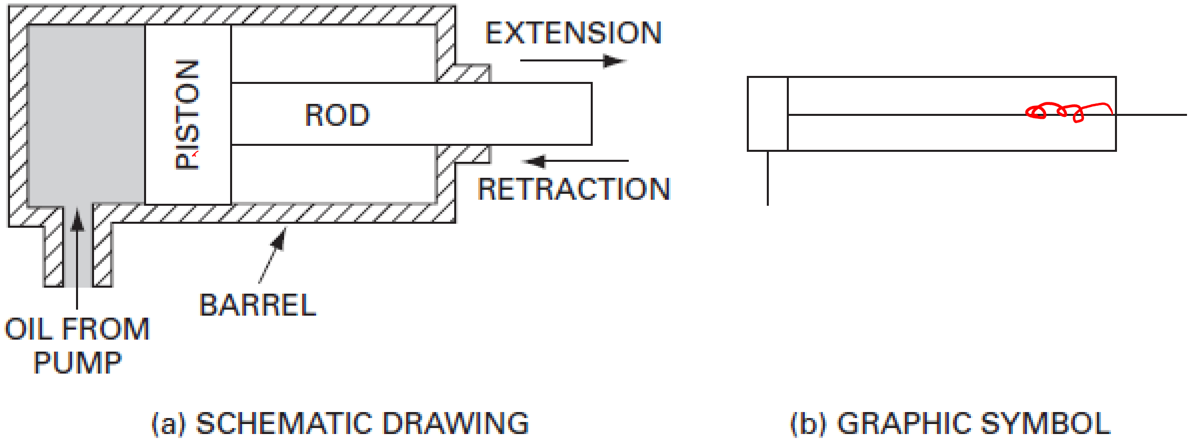
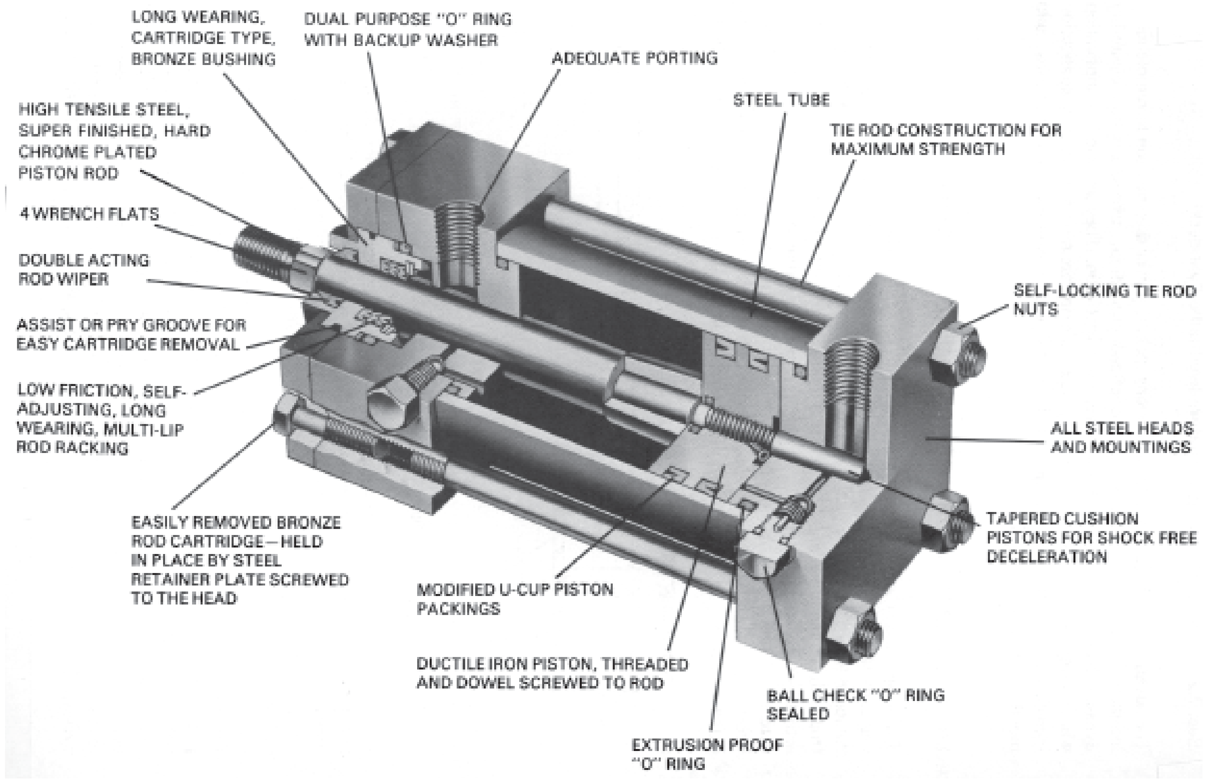


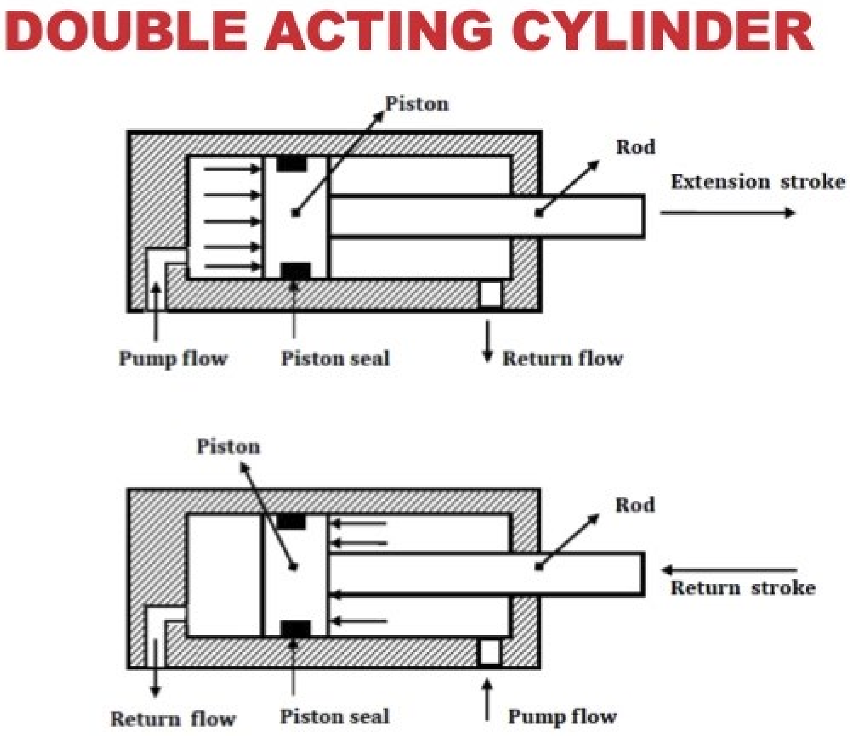
Figure 6-2. Single-acting hydraulic cylinder.

It consists of a

- piston inside a cylindrical housing called a *barrel*.
- Attached to one end of the piston is a rod, which extends outside one end of the cylinder (rod end).
- At the other end (blank end) is a port for the entrance and exit of oil.
- A single-acting cylinder can exert a force in only the extending direction as fluid from the pump enters the blank end of the cylinder. Single acting cylinders do not retract hydraulically.
- Retraction is accomplished by using gravity or by the inclusion of a compression spring in the rod end.

Double acting cylinder





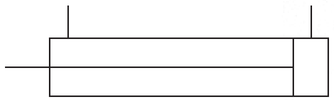


Figure 6-3. Double-actin

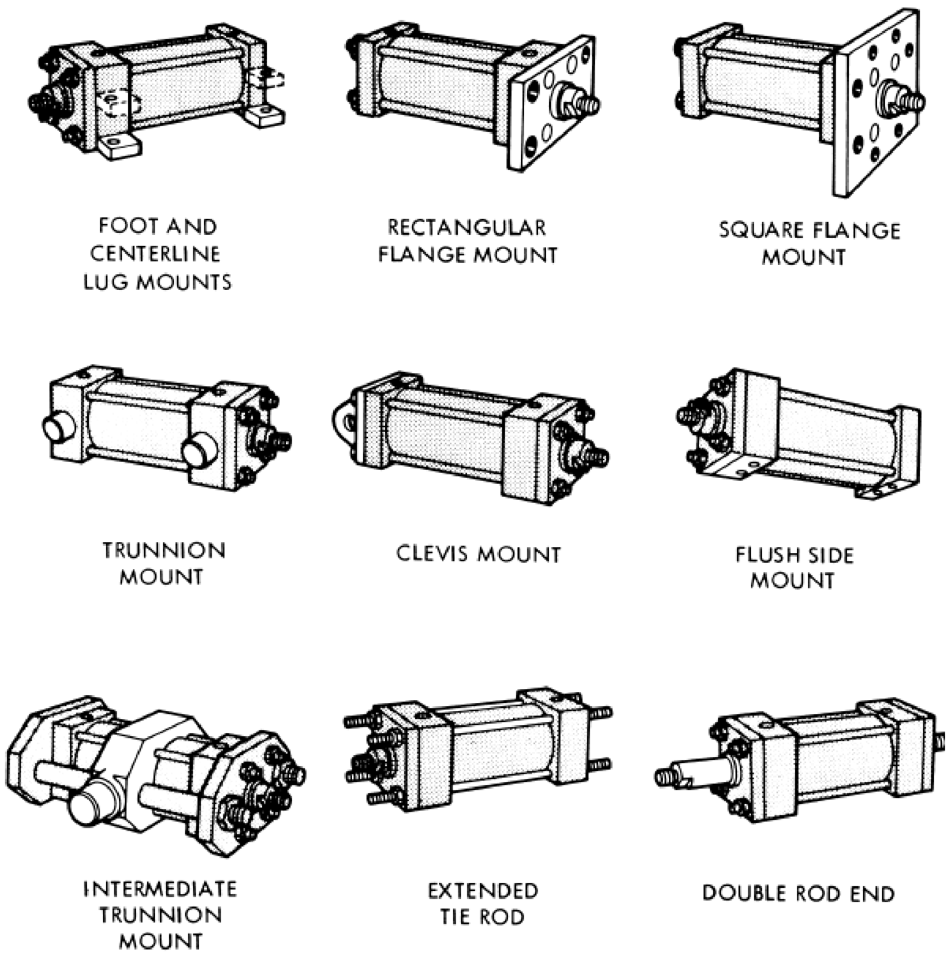


Figure 6-4. Various cylinder mountings. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

CYLINDER MOUNTINGS AND MECHANICAL LINKAGES

MECHANICAL LINKAGEs

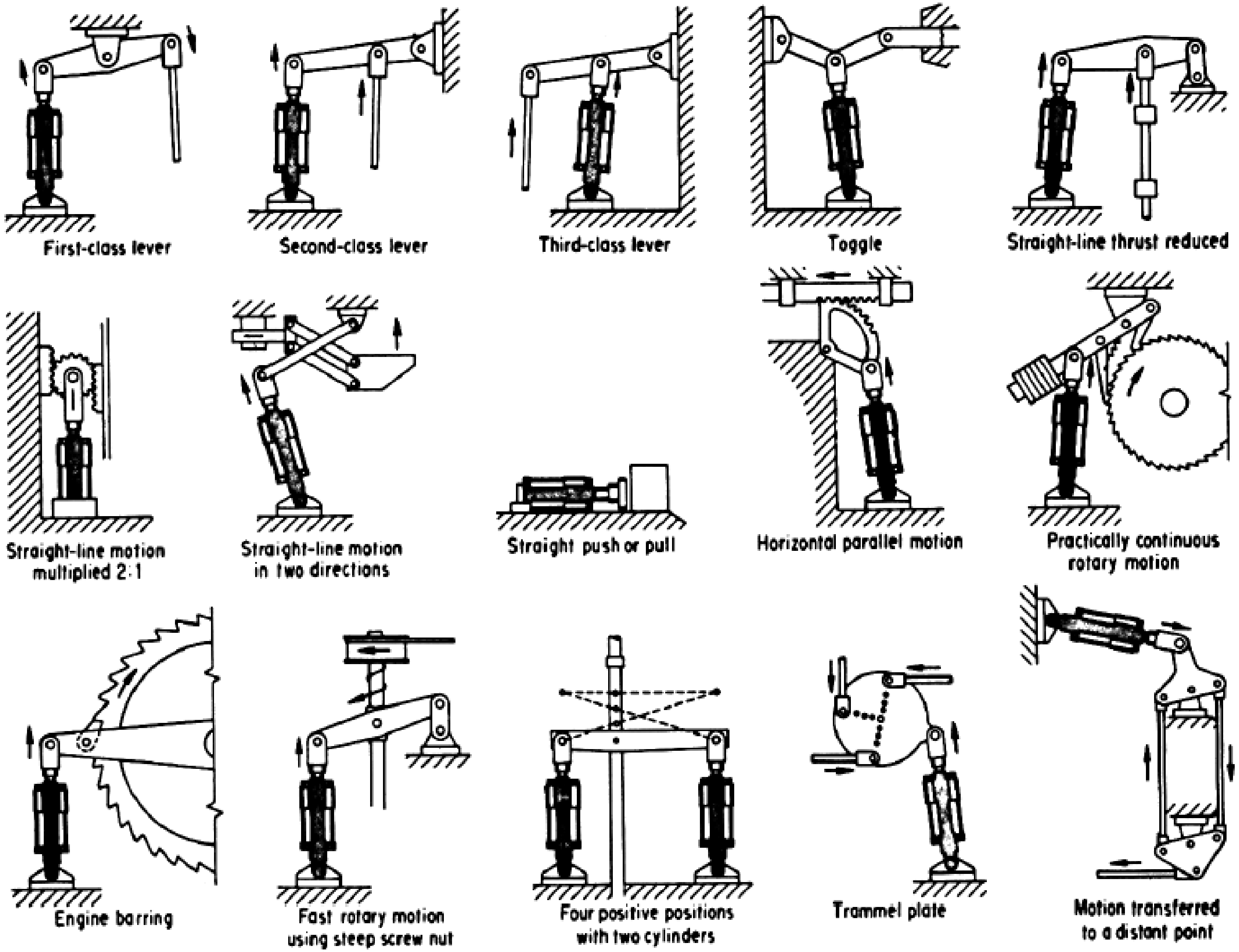


Figure 6-5. Typical mechanical linkages that can be combined with hydraulic cylinders. *(Courtesy of Rexnord Industries, Hydraulic Components Division, Racine, Wisconsin.)*

Cylinder misalignment

- Much effort has been made by manufacturers of hydraulic cylinders to reduce or eliminate the side loading of cylinders created as a result of misalignment.
- It is almost impossible to achieve perfect alignment even though the alignment of a hydraulic cylinder has a direct bearing on its life.
- A universal alignment mounting accessory designed to reduce misalignment problems .
- By using one of these accessory components and a mating clevis at each end of the cylinder, the following benefits are obtained:

1. Freer range of mounting positions
2. Reduced cylinder binding and side loading
3. Allowance for universal swivel
4. Reduced bearing and tube wear
5. Elimination of piston blow-by caused by misalignment

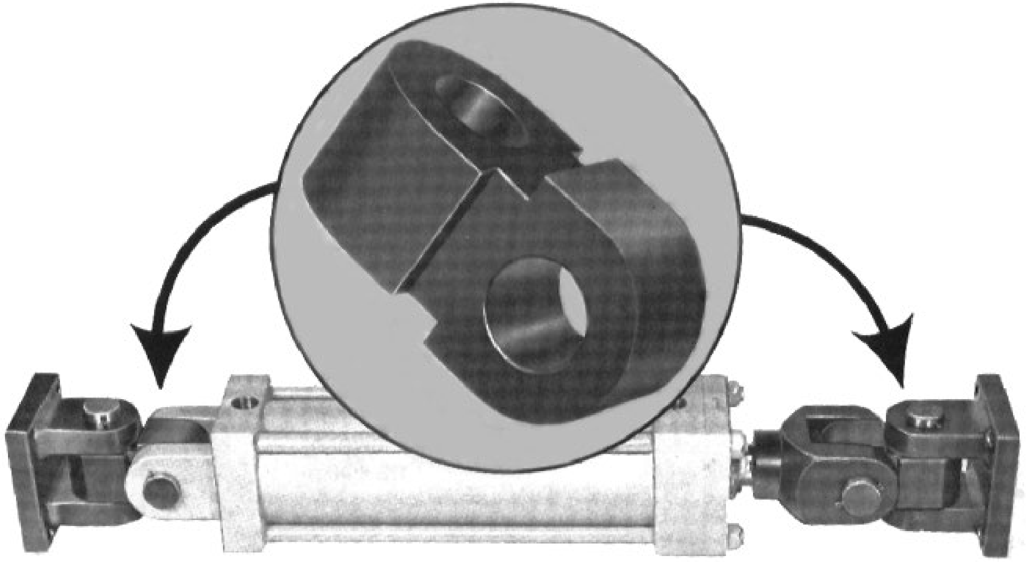


Figure 6-6. Universal alignment mounting accessory for fluid cylinders. *(Courtesy of Sheffer Corp., Cincinnati, Ohio.)*

CYLINDER FORCE, VELOCITY, AND POWER

- The output force (F) and piston velocity (υ) of double-acting cylinders are not the same for extension and retraction strokes.
- This is explained as follows:
- During the extension stroke, fluid enters the blank end of the cylinder through the entire circular area of the piston (A_p).
- However, during the retraction stroke, fluid enters the rod end through the smaller annular area between the rod and cylinder bore (*Ap - Ar*), where *Ap* equals the piston area and *Ar* equals the rod area.
- This difference in flow-path cross-sectional area accounts for the difference in piston velocities.

Extension Stroke

$$F_{ext}(\text{lb}) = p(\text{psi}) \times A_p(\text{in}^2) \quad \textbf{(6-1)}$$

$$F_{ext}(\text{N}) = p(\text{Pa}) \times A_p(\text{m}^2) \quad \textbf{(6-1M)}$$

$$\upsilon_{ext}(\text{ft/s}) = \frac{Q_{in}(\text{ft}^3/\text{s})}{A_p(\text{ft}^2)} \quad \textbf{(6-2)}$$

$$\upsilon_{ext}(\text{m/s}) = \frac{Q_{in}(\text{m}^3/\text{s})}{A_p(\text{m}^2)} \quad \textbf{(6-2M)}$$

Retraction Stroke

$$F_{ret}(\text{lb}) = p(\text{psi}) \times (A_p - A_r)\text{in}^2 \quad \textbf{(6-3)}$$

$$F_{ret}(\text{N}) = p(\text{Pa}) \times (A_p - A_r)\text{m}^2 \quad \textbf{(6-3M)}$$

$$\upsilon_{ret}(\text{ft/s}) = \frac{Q_{in}(\text{ft}^3/\text{s})}{(A_p - A_r)\text{ft}^2} \quad \textbf{(6-4)}$$

$$\upsilon_{ret}(\text{m/s}) = \frac{Q_{in}(\text{m}^3/\text{s})}{(A_p - A_r)\text{m}^2} \quad \textbf{(6-4M)}$$

CYLINDER FORCE, VELOCITY, AND POWER

- The power developed by a hydraulic cylinder equals the product of its force and velocity during a given stroke.
- Power = $p \cdot Qin$.
- Thus, we conclude that the power developed equals the product of pressure and cylinder input volume flow rate for both the extension and retraction strokes.

The horsepower developed by a hydraulic cylinder for either the extension or retraction stroke can be found using Eq. (6-5).

$$\text{Power (HP)} = \frac{v_p(\text{ft/s}) \times F(\text{lb})}{550} = \frac{Q_{in}(\text{gpm}) \times p(\text{psi})}{1714} \qquad \textbf{(6-5)}$$

Using metric units, the kW power developed for either the extension or retraction stroke can be found using Eq. (6-5M).

$$\text{Power (kW)} = v_p(\text{m/s}) \times F(\text{kN}) = Q_{in}(\text{m}^3/\text{s}) \times p(\text{kPa}) \qquad \textbf{(6-5M)}$$

CYLINDER FORCE, VELOCITY, AND POWER

EXAMPLE 6-1

A pump supplies oil at 20 gpm to a 2-in-diameter double-acting hydraulic cylinder. If the load is 1000 lb (extending and retracting) and the rod diameter is 1 in, find

a. The hydraulic pressure during the extending stroke
b. The piston velocity during the extending stroke
c. The cylinder horsepower during the extending stroke
d. The hydraulic pressure during the retraction stroke
e. The piston velocity during the retraction stroke
f. The cylinder horsepower during the retraction stroke

Solution

a.
$$p_{ext} = \frac{F_{ext}(\text{lb})}{A_p(\text{in}^2)} = \frac{1000}{(\pi/4)(2)^2} = \frac{1000}{3.14} = 318 \text{ psi}$$

b.
$$v_{ext} = \frac{Q_{in}(\text{ft}^3/\text{s})}{A_p(\text{ft}^2)} = \frac{20/449}{3.14/144} = \frac{0.0446}{0.0218} = 2.05 \text{ ft/s}$$

c.
$$\text{HP}_{ext} = \frac{v_{ext}(\text{ft/s}) \times F_{ext}(\text{lb})}{550} = \frac{2.05 \times 1000}{550} = 3.72 \text{ hp}$$

or
$$\text{HP}_{ext} = \frac{Q_{in}(\text{gpm}) \times p_{ext}(\text{psi})}{1714} = \frac{20 \times 318}{1714} = 3.72 \text{ hp}$$

d.
$$p_{ret} = \frac{F_{ret}(\text{lb})}{(A_p - A_r)\text{in}^2} = \frac{1000}{3.14 - (\pi/4)(1)^2} = \frac{1000}{2.355} = 425 \text{ psi}$$

Therefore, as expected, more pressure is required to retract than to extend the same load due to the effect of the rod.

e.
$$v_{ret} = \frac{Q_{in}(\text{ft}^3/\text{s})}{(A_p - A_r)\text{ft}^2} = \frac{0.0446}{2.355/144} = 2.73 \text{ ft/s}$$

Hence, as expected (for the same pump flow), the piston retraction velocity is greater than that for extension due to the effect of the rod.

f.
$$\text{HP}_{ret} = \frac{v_{ret}(\text{ft/s}) \times F_{ret}(\text{lb})}{550} = \frac{2.73 \times 1000}{550} = 4.96 \text{ hp}$$

or
$$\text{HP}_{ret} = \frac{Q_{in}(\text{gpm}) \times p_{ret}(\text{psi})}{1714} = \frac{20 \times 425}{1714} = 4.96 \text{ hp}$$

Thus, more horsepower is supplied by the cylinder during the retraction stroke because the piston velocity is greater during retraction and the load force remained the same during both strokes. This, of course, was accomplished by the greater pressure level during the retraction stroke. Recall that the pump output flow rate is constant, with a value of 20 gpm.

CYLINDER LOADS DUE TO MOVING OF WEIGHTS

- The force a cylinder must produce equals the load the cylinder is required to overcome.
- In many cases the load is due to the weight of an object the cylinder is attempting to move.
- In the case of a vertical cylinder, the load simply equals the weight of the object because gravity acts in a downward, vertical direction.
- Sometimes a cylinder is used to slide an object along a horizontal surface. In this case, the cylinder load is theoretically zero. This is because there is no component of the object’s weight acting along the axis of the cylinder (a horizontal direction).
- However, as the object slides across the horizontal surface, the cylinder must overcome the frictional force created between the object and the horizontal surface.
- This frictional force, which equals the load acting on the cylinder, opposes the direction of motion of the moving object.
- If the cylinder is mounted in neither a vertical nor horizontal direction, the cylinder load equals the component of the object’s weight acting along the axis of the cylinder, plus a frictional force if the object is sliding along an inclined surface.
- Thus for an inclined cylinder, the load the cylinder must overcome is less than the weight of the object to be moved if the object is not sliding on an inclined surface.
- The cylinder loads described up to now are based on moving an object at a constant velocity.
- However, an object to be moved at a given velocity is initially at rest.
- Thus the object has to be accelerated from zero velocity up to a steady state (constant) velocity as determined by the pump flow rate entering the cylinder.
- This acceleration represents an additional force (called an *inertial force*) that must be added to the weight component and any frictional force involved.

CYLINDER LOADS DUE TO MOVING OF WEIGHTS

EXAMPLE 6-2

Find the cylinder force F required to move a 6000-lb weight W along a horizontal surface at a constant velocity. The coefficient of friction (CF) between the weight and horizontal support surface equals 0.14.

Solution

The frictional force f between the weight and its horizontal supporting surface equals CF times W. Thus, we have

$$F = f = (CF) \times W = 0.14 \times 6000 \text{ lb} = 840 \text{ lb}$$

EXAMPLE 6-3

Find the cylinder force F required to lift the 6000-lb weight W of Example 6-2 along a direction which is 30° from the horizontal, as shown in Figure 6-8(a). The weight is moved at constant velocity.

Solution

As shown in Figure 6-8(b), the cylinder force F must equal the component of the weight acting along the centerline of the cylinder. Thus, we have a right triangle with the hypotenuse W and the side F forming a 60° angle. From trigonometry we have $\sin 30° = F/W$. Solving for F yields

$$F = W \sin 30° = 6000 \text{ lb} \times \sin 30° = 3000 \text{ lb}$$

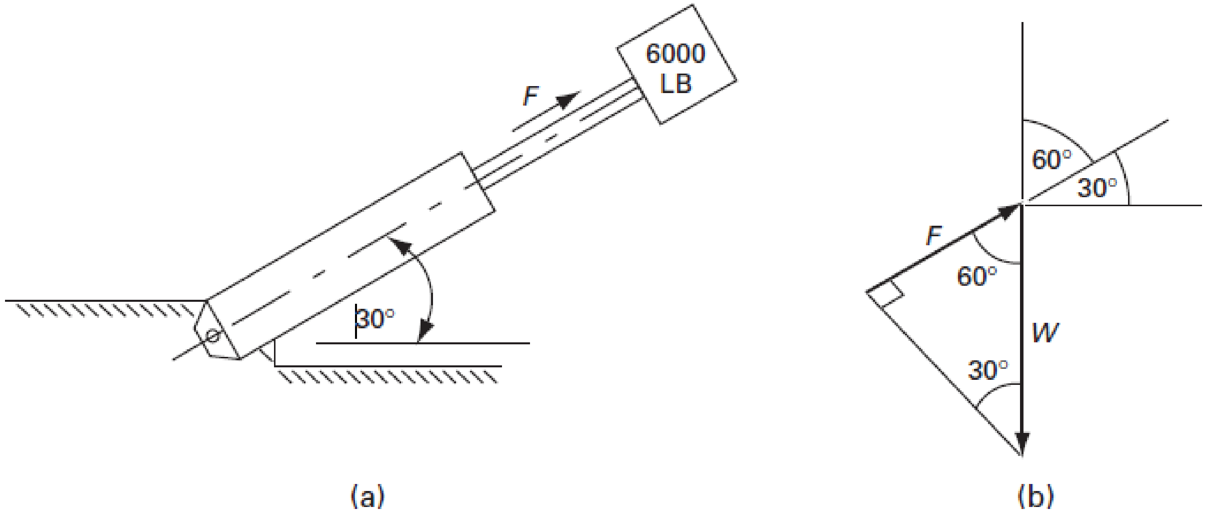


Figure 6-8. Inclined cylinder lifting a load for Example 6-3.

CYLINDER LOADS DUE TO MOVING OF WEIGHTS

EXAMPLE 6-4

The 6000-lb weight of Example 6-2 is to be lifted upward in a vertical direction. Find the cylinder force required to

a. Move the weight at a constant velocity of 8 ft/s
b. Accelerate the weight from zero velocity to a velocity of 8 ft/s in 0.50 s

Solution

a. For a constant velocity the cylinder force simply equals the weight of 6000 lb.

b. From Newton's law of motion, the force required to accelerate a mass m equals the product of the mass m and its acceleration a. Noting that mass equals weight divided by the acceleration of gravity g, we have

$$a = \frac{8 \text{ ft/s} - 0 \text{ ft/s}}{0.50 \text{ s}} = 16 \text{ ft/s}^2$$

Thus, the force required to accelerate the weight is

$$F_{accel} = \frac{6000 \text{ lb}}{32.2 \text{ ft/s}^2} \times 16 \text{ ft/s}^2 = 2980 \text{ lb}$$

The cylinder force F_{cyl} required equals the sum of the weight and the acceleration force.

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

SPECIAL CYLINDER DESIGNS

- Double-rod cylinder in which the rod extends out of the cylinder at both ends. For such a cylinder, the words *extend* and *retract* have no meaning.
- Since the force and speed are the same for either end, this type of cylinder is typically used when the same task is to be performed at either end.
- Since each end contains the same size rod, the velocity of the piston is the same for both strokes

Figure 6-9. Double-rod cylinder. *(Courtesy of Allenair Corp., Mineola, New York.)*

SPECIAL CYLINDER DESIGNS

- Telescopic cylinder contains multiple cylinders that slide inside each other.
- They are used where long work strokes are required but the full retraction length must be minimized.
- One application for a telescopic cylinder is the high-lift fork truck, illustrated
- in Figure 6-11.
- As shown, this lift truck is in the process of accessing materials located inside a second-story storage area of a warehouse, from the outside.

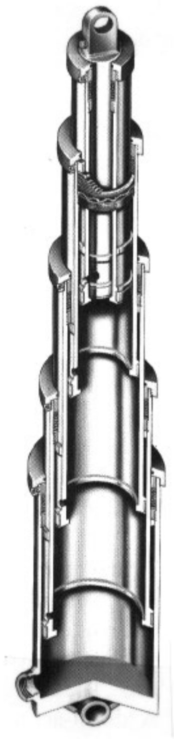


Figure 6-10. Telescopic cylinder. *(Courtesy of Commercial Shearing, Inc., Youngstown, Ohio.)*

Figure 6-11. High-lift truck. *(Courtesy of Mitsubishi Caterpillar Forklift America, Houston, Texas.)*

CYLINDER LOADINGS THROUGH MECHANICAL LINKAGES

first-class lever system, which is characterized by the lever fixed-hinge pin being located between the cylinder and load rod pins.

Note that the length of the lever portion from the cylinder rod pin to the fixed hinge is L_1, whereas the length of the lever portion from the load rod pin to the fixed hinge is L_2.

To determine the cylinder force F_{cyl} required to drive a load force F_{load}, we equate moments about the fixed hinge, which is the pivot point of the lever.

The cylinder force attempts to rotate the lever counterclockwise about the pivot, and this creates a counterclockwise moment. Similarly, the load force creates a clockwise moment about the pivot. At equilibrium, these two moments are equal in magnitude:

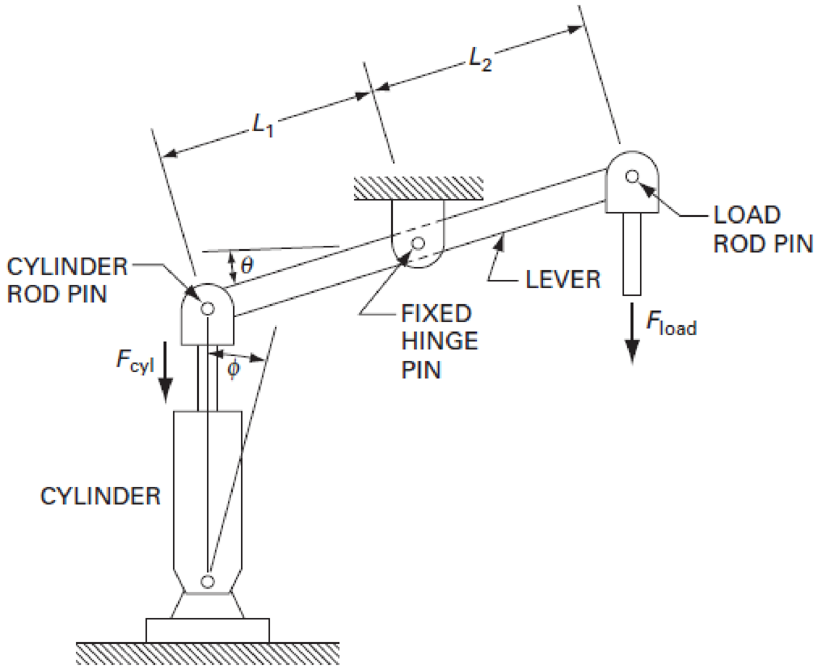


Figure 6-12. Use of a first-class lever to drive a load.

Counterclockwise moment = clockwise moment

$$F_{cyl}(L_1 \cos\theta) = F_{load}(L_2 \cos\theta)$$

$$F_{cyl} = \frac{L_2}{L_1} F_{load} \quad \textbf{(6-6)}$$

CYLINDER LOADINGS THROUGH MECHANICAL LINKAGES

If the centerline of the hydraulic cylinder becomes offset by an angle from the vertical, as shown in Figure 6-12, the relationship becomes:

$$F_{cyl}(L_1 \cos\theta \times \cos\phi) = F_{load}(L_2 \cos\theta)$$

or

$$F_{cyl} = \frac{L_2}{L_1 \cos\phi} F_{load} \qquad \textbf{(6-7)}$$

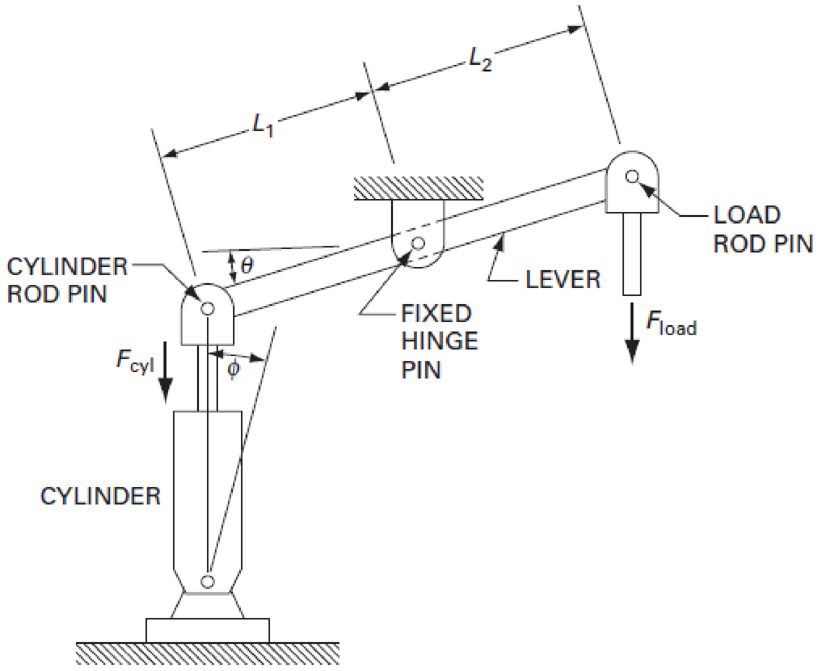


Figure 6-12. Use of a first-class lever to drive a load.

CYLINDER LOADINGS THROUGH MECHANICAL LINKAGES

- Second-class lever system, which is characterized by the load rod pin being located between the fixed-hinge pin and cylinder rod pin of the lever.

- The analysis is accomplished by equating moments about the fixed-hinge pin, as follows:

$$F_{cyl} \cos \phi (L_1 + L_2) \cos \theta = F_{load} (L_2 \cos \theta)$$

or

$$F_{cyl} = \frac{L_2}{(L_1 + L_2) \cos \phi} F_{load} \quad \text{(6-8)}$$

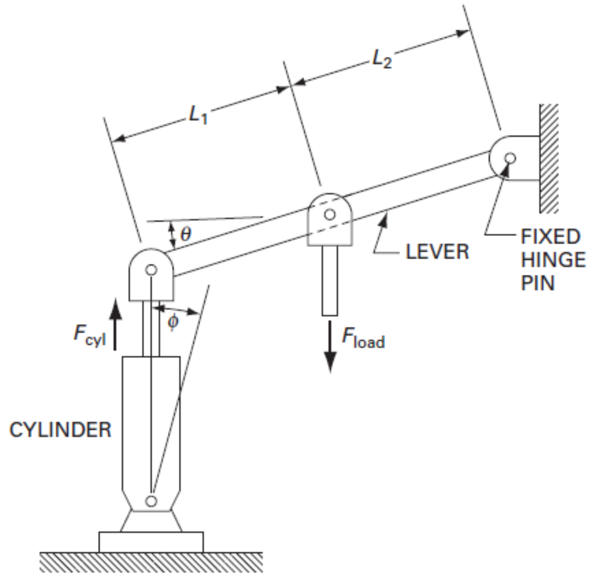


Figure 6-13. Use of a second-class lever system to drive a load.

Comparing Eq. (6-7) to Eq. (6-8) shows that a smaller cylinder force is required to drive a given load force for a given lever length if a second-class lever is used instead of a first-class lever. Thus, using a second-class lever rather than a first-class lever reduces the required cylinder piston area for a given application. Of course, using a second-class lever also results in a smaller load stroke for a given cylinder stroke.

CYLINDER LOADINGS THROUGH MECHANICAL LINKAGES

- Third-class lever system the cylinder rod pin lies between the load rod pin and fixed-hinge pin of the lever.
- Equating moments about the fixed-hinge pin yields

$$F_{cyl}\cos\phi(L_2\cos\theta) = F_{load}(L_1 + L_2)\cos\theta$$

or

$$F_{cyl} = \frac{L_1 + L_2}{L_2\cos\phi}F_{load} \qquad \textbf{(6-9)}$$

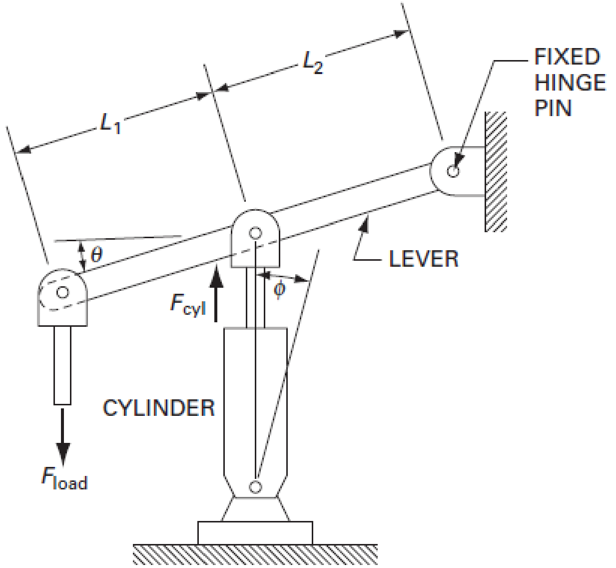


Figure 6-14. Use of a third-class lever system to drive a load.

Examination of Eq. (6-9) reveals that for a third-class lever, the cylinder force is greater than the load force. The reason for using a third-class lever system would be to produce a load stroke that is greater than the cylinder stroke, at the expense of requiring a larger cylinder diameter.

CYLINDER LOADINGS THROUGH MECHANICAL LINKAGES

Example 6-5

For the first-, second-, and third-class lever systems of Figures 6-12, 6-13, and 6-14 the following data are given:

$$L_1 = L_2 = 10 \text{ in}$$

$$\phi = 0°$$

$$F_{\text{load}} = 1000 \text{ lb}$$

Find the cylinder force required to overcome the load force for the

a. First-class lever
b. Second-class lever
c. Third-class lever

Solution

a. Per Eq. (6-7), we have

$$F_{\text{cyl}} = \frac{L_2}{L_1 \cos\phi} F_{\text{load}} = \frac{10}{10 \times 1}(1000) = 1000 \text{ lb}$$

b. Using Eq. (6-8) yields

$$F_{\text{cyl}} = \frac{L_2}{(L_1 + L_2)\cos\phi} F_{\text{load}} = \frac{10}{(10 + 10) \times 1}(1000) = 500 \text{ lb}$$

c. Substituting into Eq. (6-9), we have

$$F_{\text{cyl}} = \frac{L_1 + L_2}{L_2 \cos\phi} F_{\text{load}} = \frac{(10 + 10)}{10 \times 1}(1000) = 2000 \text{ lb}$$

Thus, as expected, the second-class lever requires the smallest cylinder force, whereas the third-class lever requires the largest cylinder force.

HYDRAULIC CYLINDER CUSHIONS

- Double-acting cylinders sometimes contain cylinder cushions at the ends of the cylinder to slow the piston down near the ends of the stroke.
- This prevents excessive impact when the piston is stopped by the end caps.
- Deceleration starts when the tapered plunger enters the opening in the cap. This restricts the exhaust flow from the barrel to the port.
- During the last small portion of the stroke, the oil must exhaust through an adjustable opening.
- The cushion design also incorporates a check valve to allow free flow to the piston during direction reversal.
- The maximum pressure developed by cushions at the ends of a cylinder must be considered since excessive pressure buildup would rupture the cylinder.

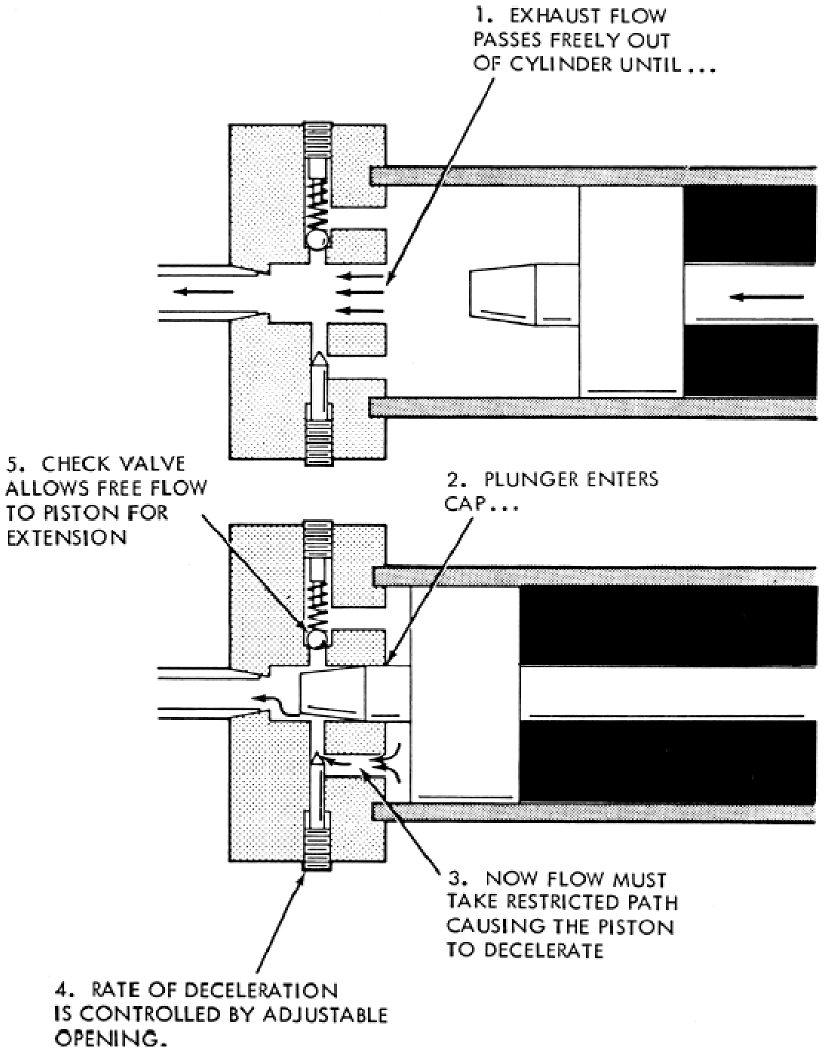


Figure 6-16. Operation of cylinder cushions. *(Courtesy of Sperry Vickers, Sperry Rand Corp., Troy, Michigan.)*

HYDRAULIC CYLINDER CUSHIONS

Example 6-6

A pump delivers oil at a rate of 18.2 gpm into the blank end of the 3-in-diameter hydraulic cylinder shown in Figure 6-17. The piston contains a 1-in-diameter cushion plunger that is 0.75 in long, and therefore the piston decelerates over a distance of 0.75 in at the end of its extension stroke. The cylinder drives a 1500-lb weight, which slides on a flat horizontal surface having a coefficient of friction (CF) equal to 0.12. The pressure relief valve setting equals 750 psi. Therefore, the maximum pressure (p_1) at the blank end of the cylinder equals 750 psi while the cushion is decelerating the piston. Find the maximum pressure (p_2) developed by the cushion.

Solution

Step 1: Calculate the steady-state piston velocity v prior to deceleration:

$$v = \frac{Q_{\text{pump}}}{A_{\text{piston}}} = \frac{(18.2/449)\text{ft}^3/\text{s}}{[(\pi/4)(3)^2/144]\text{ft}^2} = \frac{0.0406}{0.049} = 0.83 \text{ ft/s}$$

Step 2: Calculate the deceleration a of the piston during the 0.75-in displacement S using the constant acceleration (or deceleration) equation:

$$v^2 = 2aS$$

Solving for deceleration, we have

$$a = \frac{v^2}{2S} \tag{6-10}$$

Substituting known values, we obtain the value of deceleration:

$$a = \frac{(0.83 \text{ ft/s})^2}{2(0.75/12 \text{ ft})} = 5.51 \text{ ft/s}^2$$

Step 3: Use Newton's law of motion: The sum of all external forces ΣF acting on a mass m equals the product of the mass m and its acceleration or deceleration a:

$$\Sigma F = ma$$

When substituting into Newton's equation, we shall consider forces that tend to slow down the piston as being positive forces. Also, the mass m equals the mass of all the moving members (piston, rod, and load). Since the weight of the piston and rod is small compared to the weight of the load, the weight of the piston and rod will be ignored. Also note that mass m equals weight W divided by the acceleration of gravity g. The frictional retarding force f between the weight W and its horizontal support surface equals CF times W. This frictional force is the external load force acting on the cylinder while it is moving the weight.

Substituting into Newton's equation yields

$$p_2(A_{\text{piston}} - A_{\text{cushion plunger}}) + (\text{CF})W - p_1(A_{\text{piston}}) = \frac{W}{g}a$$

Solving for p_2 yields a usable equation:

$$p_2 = \frac{(W/g)a + p_1(A_{\text{piston}}) - (\text{CF})W}{A_{\text{piston}} - A_{\text{cushion plunger}}} \tag{6-11}$$

Substituting known values produces the desired result:

$$p_2 = \frac{[(1500)(5.51)/32.2] + 750(\pi/4)(3)^2 - (0.12)(1500)}{(\pi/4)(3)^2 - (\pi/4)(1)^2}$$

$$p_2 = \frac{257 + 5303 - 180}{7.07 - 0.785} = \frac{5380}{6.285} = 856 \text{ psi}$$

Thus, the hydraulic cylinder must be designed to withstand an operating pressure of 856 psi rather than the pressure relief valve setting of 750 psi.

HYDRAULIC CYLINDER CUSHIONS

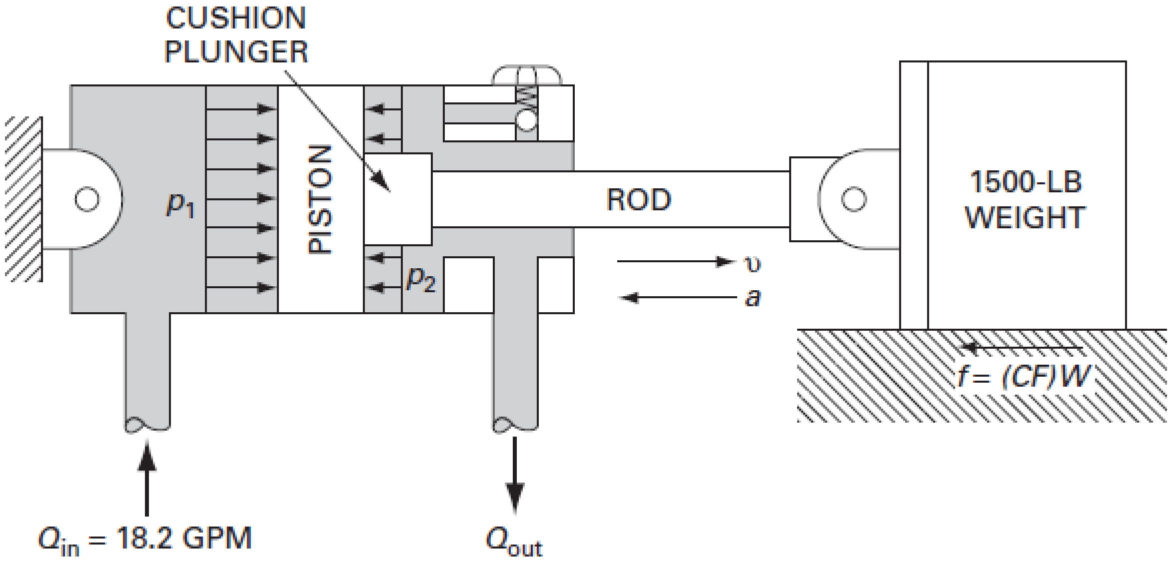


Figure 6-17. Cylinder cushion problem for Example 6-6.

HYDRAULIC SHOCK ABSORBERS

- A hydraulic shock absorber is a device that brings a moving load to a gentle rest through the use of metered hydraulic fluid.
- hydraulic shock absorber can provide a uniform gentle deceleration of any moving load from 25 to 25,000 lb or where the velocity and weight combination equals 3300 in · lb.
- Heavy-duty units are available with load capacities of over 11 million in · lb and strokes up to 20 in.

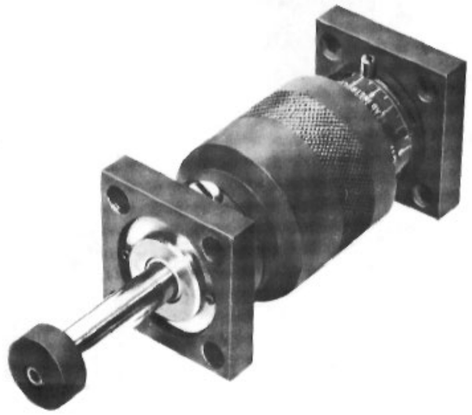
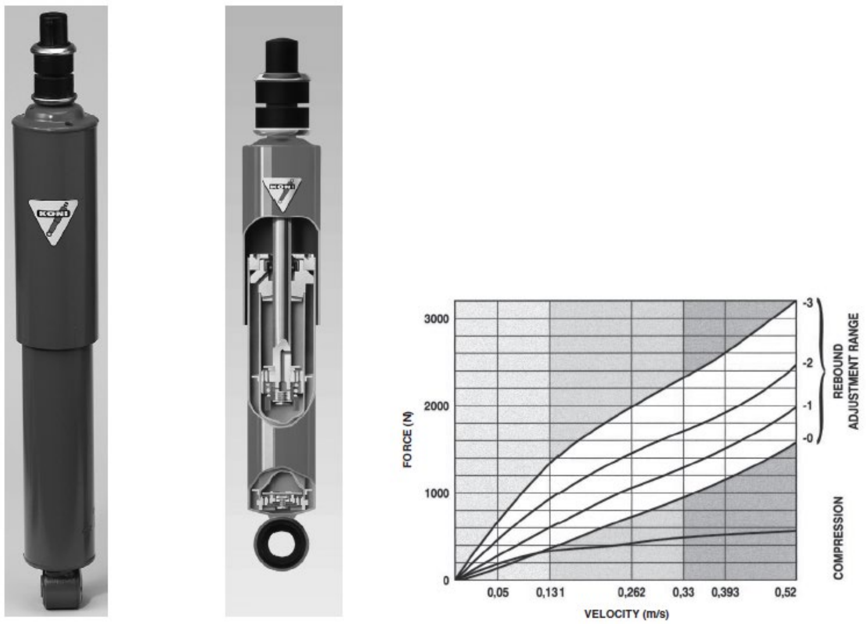


Figure 6-18. Hydraulic shock absorber. *(Courtesy of EGD Inc., Glenview, Illinois.)*

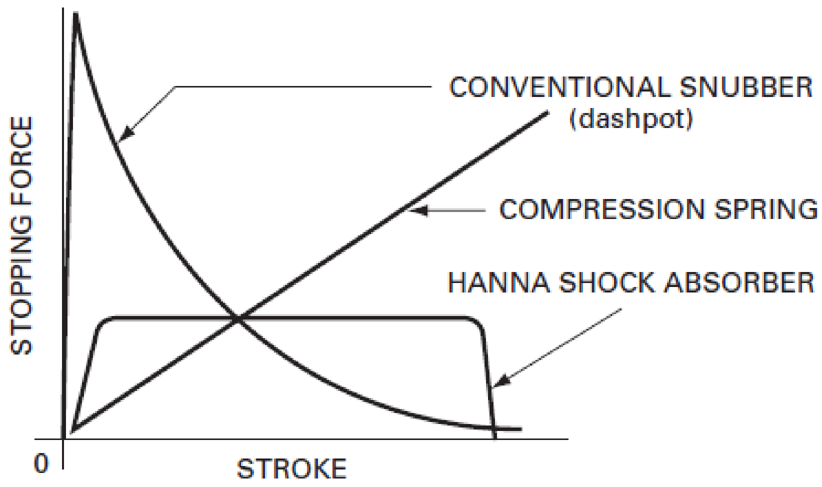


(a) External view. (b) Cutaway view. (c) Force-velocity diagram.

Figure 6-22. Automotive hydraulic shock absorber. *(Courtesy of KONI North America, Hebron, Kentucky.)*

HYDRAULIC SHOCK ABSORBERS

- These shock absorbers are filled completely with oil. Therefore, they may be mounted in any position or at any angle.
- The spring-return units are entirely self-contained, extremely compact types that require no external hoses, valves, or fittings.
- In this spring-returned type a built-in cellular accumulator accommodates oil displaced by the piston rod as the rod moves inward.
- Since the shock absorber is always filled with oil, there are no air pockets to cause spongy or erratic action.
- These shock absorbers are multiple-orifice hydraulic devices.
- The orifices are simply holes through which a fluid can flow.
- When a moving load strikes the bumper of the shock absorber, it sets the rod and piston in motion.
- The moving piston pushes oil through a series of holes from an inner high-pressure chamber to an outer low-pressure chamber.
- The resistance to the oil flow caused by the holes (restrictions) creates a pressure that acts against the piston to oppose the moving load.
- Holes are spaced geometrically according to a proven formula that produces constant pressure on the side of the piston opposite the load (constant resisting force) from the beginning to nearly the end of the stroke.



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