Lecture 14

HYDRAULIC ACTUATORS [CONTINUED]

1. 7 First-, Second- and Third-Class Lever Systems

Many mechanisms use hydraulic cylinders to transmit motion and power. Among these, lever mechanisms such as toggles, the rotary devices and the push--pull devices use a hydraulic cylinder. In this section, the mechanics of cylinder loading used in first-class, second-class and third-class lever systems is being discussed.

1. 7.1 First-Class Lever System

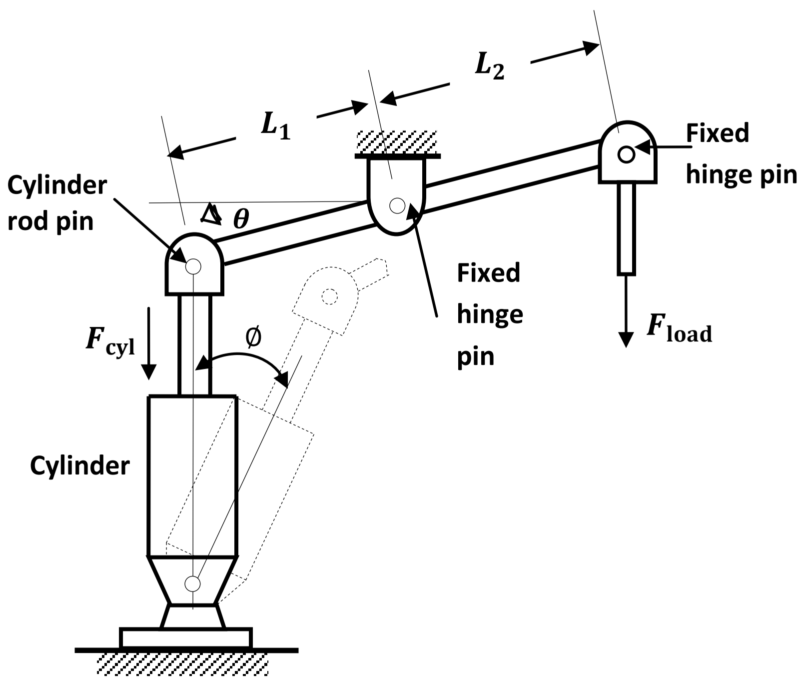


Figure 1. 21 First-class lever system

In this lever system, the fixed-hinge point is located in between the cylinder and the loading point. The schematic arrangement of a first-class lever system with a hydraulic cylinder is shown in Fig.1. 21. In this system, the downward load acts at the lever end. The cylinder has to apply a downward force to lift the load. The cylinder has a clevis mounting arrangement; it pivots about its eye-end center through an angle. However, the effect of this angle (around 10° to 15°) is negligible on the force and hence cannot be considered.

Here, F_{load} = load to be operated, F_{cyl} = load to be exerted by a hydraulic cylinder, L_1 = distance from the rod end to the pivot point, L_2 = distance from the pivot point to the loading point and θ = inclination of the lever measured with respect to the horizontal line at the hinge.

When the load is being lifted, the cylinder force rotates the lever in an anticlockwise direction about the pivot point. Due to this, a moment acts in the anticlockwise direction. At the same time, the force due to the load acting causes a clockwise moment. At equilibrium, the two moments are equal

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

$$F_{cyl} = \frac{L_2}{L_1} F_{load} \qquad (1.\ 3)$$

Suppose the centerline of the hydraulic cylinder tilts by an offset angle ϕ from the vertical; the relationship becomes

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

$$F_{\text{cyl}} = \frac{L_2}{(L_1)\cos\phi} F_{\text{load}} \qquad (1.\ 4)$$

Note 1: If$L_1 > L_2$, the cylinder force is less than the load force and the cylinder stroke is greater than the load stroke.
Note 2: Ifthe inclination of cylinder (ϕ) is less than 10^o,its effect can be ignored in equation (2).

1. 7.2 Second-Class Lever System
In this lever system, the loading point is in between the cylinder and the hinge point as shown in Fig.1. 22.

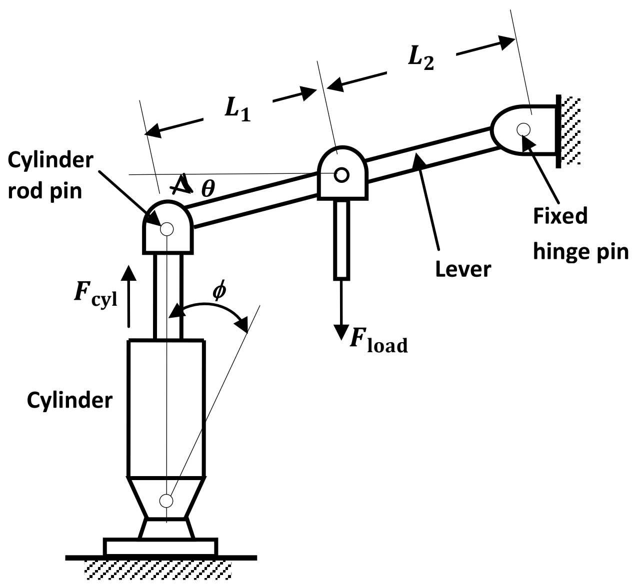


Figure 1. 22 Second-class lever system
Using the same nomenclature discussed under the previous lever systems and equating moments about the fixed-hinge pin, we can write

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

$$F_{\text{cyl}} = \frac{L_2}{(L_1 + L_2)\cos\phi} F_{\text{load}} \quad (1.\ 5)$$

Note 3: Compared to the first-class lever, the second-class lever requires smaller cylinder force to drive the given load force for same L_1 and L_2 and load force. In other words, if we use a second-class lever cylinder, a smaller size cylinder can be used.
Note 4: Compared to the first-class lever, the second-class lever also results ina smaller load stroke for a given cylinder stroke.

1. 7.3 Third-Class Lever System
For a third-class lever system shown in Fig. 1.23, the cylinder rod pin lies between the load road pin and the fixed-hinge pin of the lever.

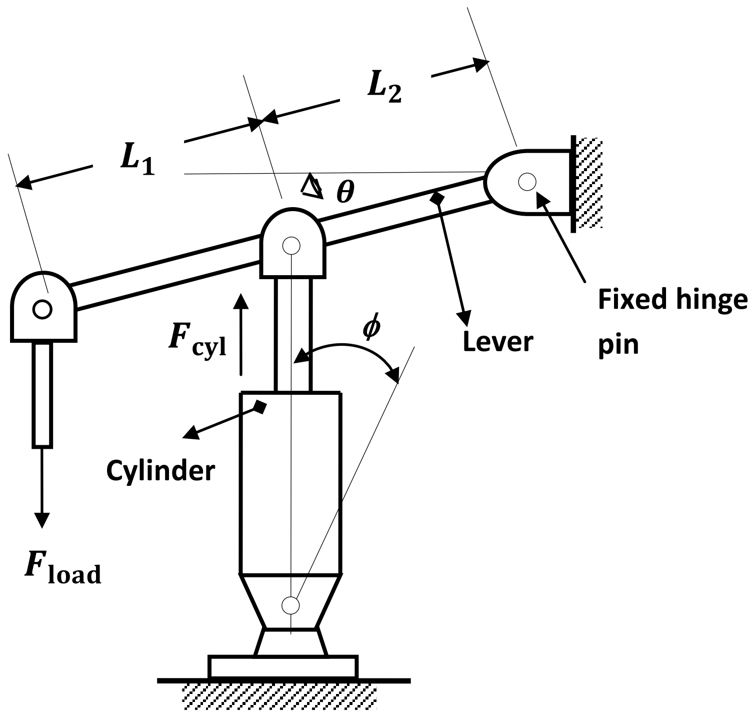


Figure 1. 23 Third-class lever system

Equating moments about the hinge point, we can write

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

$$F_{cyl} = \frac{L_1+L_2}{L_2\cos\phi}F_{load} \qquad (1.\ 6)$$

Note 4: In a third-class lever system**,** cylinder force is greater than load force.
Note 5: In a third-class lever system, load stroke is greater than the cylinder stroke and therefore requires a larger cylinder.

Example 1. 16

Following data are given for the first-, second- and third-class lever systems: $L_1 = L_2 = 25.4$ cm, $\phi = 10°$, $F_{load} = 4444$ N . Compare the cylinder force needed in each case to overcome the load force. Repeat this with $\phi = 5°$ and $10°$.

Solution: First-class lever system

$$F_{cyl} = \left(\frac{L_2}{L_1}\right)\frac{F_{load}}{\cos\phi}$$

$$= \frac{25}{25\cos 0°}4444\,\text{N}$$

$$= 4444\ \ \text{N}$$

Second-class lever system

$$F_{\text{cyl}} = \frac{L_2}{(L_1 + L_2)\cos\phi} F_{\text{load}}$$
$$= \frac{2}{(25+25)\cos 10°} 4444$$
$$= 2222\,\text{N}$$

Third-class lever system

$$F_{\text{cyl}} = \frac{L_1 + L_2}{L_2 \cos\phi} F_{\text{load}}$$
$$= \frac{25+25}{25\cos 10°} 4444$$
$$= 8888\,\text{N}$$

As seen, the cylinder force in the second-class lever system is half of that in the first-class lever system and one-fourth of that in the third-class lever system.

When $\phi = 5°$ and $10°$:

First-class lever

$$F_{\text{cyl}}(\phi = 5°) = \frac{4444}{\cos 5°} = 4461 \text{ N}$$

$$F_{\text{cyl}}(\phi = 10°) = \frac{4444}{\cos 10°} = 4729 \text{ N}$$

Second-class lever

$$F_{\text{cyl}}(\phi = 5°) = \frac{2222}{\cos 5°} = 2231 \text{ N}$$

$$F_{\text{cyl}}(\phi = 10°) = \frac{2222}{\cos 10°} = 2365 \text{ N}$$

Third-class lever

$$F_{\text{cyl}}(\phi = 5°) = \frac{8888}{\cos 5°} = 8922\,\text{N}$$

$$F_{\text{cyl}}(\phi = 10°) = \frac{8888}{\cos 10°} = 9458 \text{ N}$$

Example 1. 17

For the system given in Fig. 1. 24 determine the force required to drive a 1000 N load.

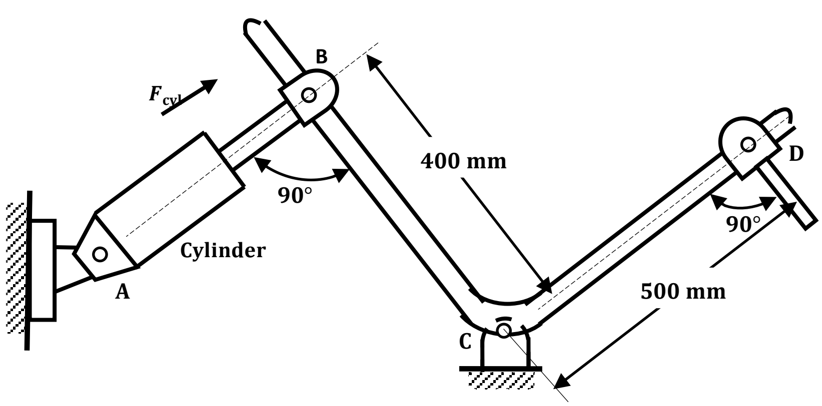


Figure 1. 24

Solution: As seen, it is a first-class lever system. Taking moments about the hinge, we get

$$F_{\text{cyl}} \times \text{BC} = F_{\text{load}} \times \text{CD}$$
$$\Rightarrow F_{\text{cyl}} = F_{\text{load}} \times \frac{\text{CD}}{\text{BC}} = 1000 \times \frac{500}{400} = 1250\ \text{N}$$

Example 1. 18
For the crane system, determine the hydraulic cylinder force required to lift a 2000 N load (Figs. 1. 25 and 1. 26).

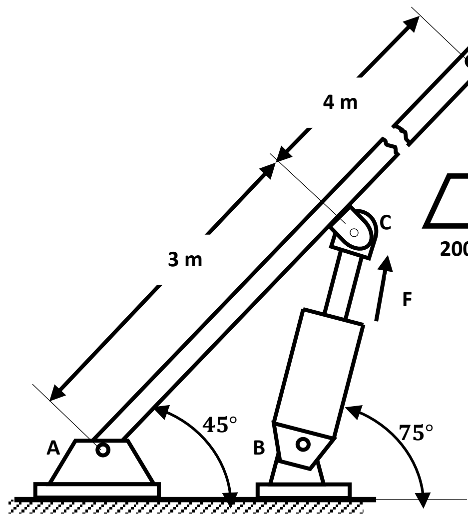


Figure 1. 25

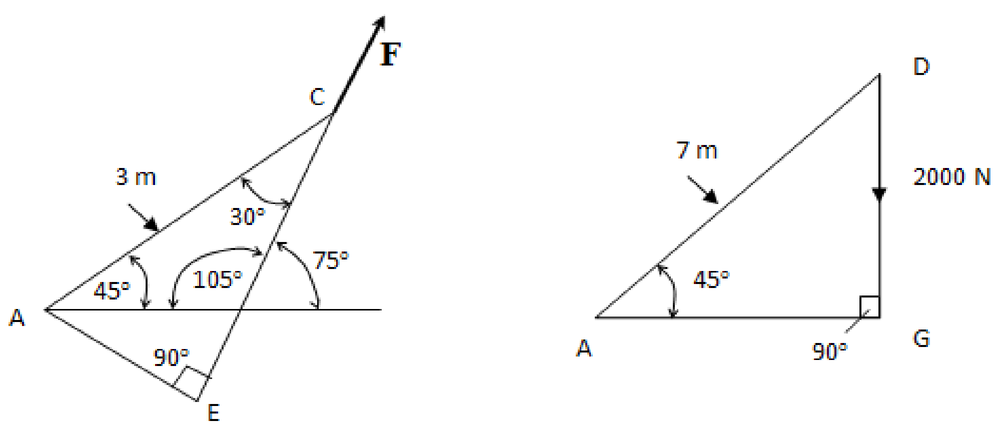


Figure 1. 26

Solution: The given system is a third-class lever system as the cylinder pin lies in between the load rod pin and fixed-hinge pin of the lever. Equating moments about fixed pin A due to the cylinder force *F* and the 2000 N force we get

$$2000 \times \text{Perpendicular dist. AG} = F \times \text{Perpendicular dist. AE}$$

From trigonometry of right-angled triangles, we have

$$\cos 45° = \frac{\text{AG}}{7} \Rightarrow \text{AG} = 7\cos 45° = 4.95\,\text{m}$$

$$\sin 30° = \frac{\mathrm{AE}}{3} \Rightarrow \mathrm{AE} = 3\sin 30° = 1.5\,\mathrm{m}$$
$$2000 \times 4.95 = F \times 1.5$$
$$\Rightarrow F = 6600\ \mathrm{N}$$

Example 1. 19

Figure 1. 27 shows a toggle mechanism. Find the output load force for a hydraulic cylinder force of 1000 N.

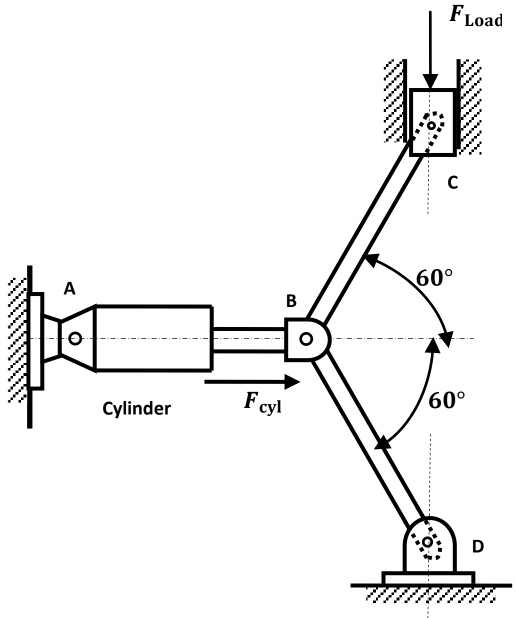


Figure 1. 27

Solution: Setting the sum of the forces on pin C equal to zero (from Newton's law of motion), force (F) = ma = 0 because a = 0 for constant velocity motion, yields the following for the x- and y-axes:

y-axis: $F_{\mathrm{BC}} \sin 60° - F_{\mathrm{BD}} \sin 60° = 0 \Rightarrow F_{\mathrm{BC}} = F_{\mathrm{BD}}$

x-axis: $F_{\mathrm{cyl}} - F_{\mathrm{BC}} \cos 60° - F_{\mathrm{BD}} \cos 60° = 0$

$$F_{\mathrm{cyl}} - 2F_{\mathrm{BC}} \cos 60° = 0$$
$$\Rightarrow F_{\mathrm{BC}} = \frac{F_{\mathrm{cyl}}}{2\cos 60°}$$

Similarly, setting the sum of forces on pin C equal to zero for the y-axis direction yields

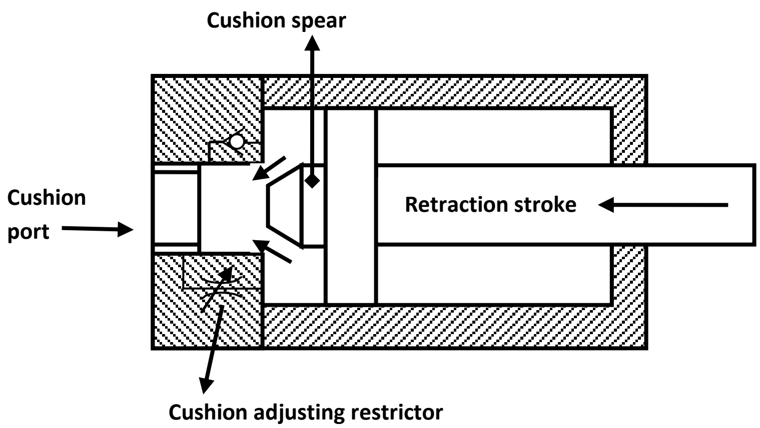
$$F_{\mathrm{BC}} \sin 60° - F_{\mathrm{load}} = 0$$

Therefore, we have

$$F_{\mathrm{load}} = F_{\mathrm{BC}} \sin 60°$$
$$= \frac{\sin 60°}{2\cos 60°} \times F_{\mathrm{cyl}}$$
$$= \frac{\tan 60°}{2} \times 1000 = 866\,\mathrm{N}$$

1. 8 Cylinder Cushions

For the prevention of shock due to stopping loads at the end of the piston stroke, cushion devices are used. Cushions may be applied at either end or both ends. They operate on the principle that as the cylinder piston approaches the end of stroke, an exhaust fluid is forced to go through an adjustable needle valve that is set to control the escaping fluid at a given rate. This allows the deceleration characteristics to be adjusted for different loads. When the cylinder piston is actuated, the fluid enters the cylinder port and flows through the little check valve so that the entire piston area can be utilized to produce force and motion. A typical cushioning arrangement is shown in Fig. 1. 28.



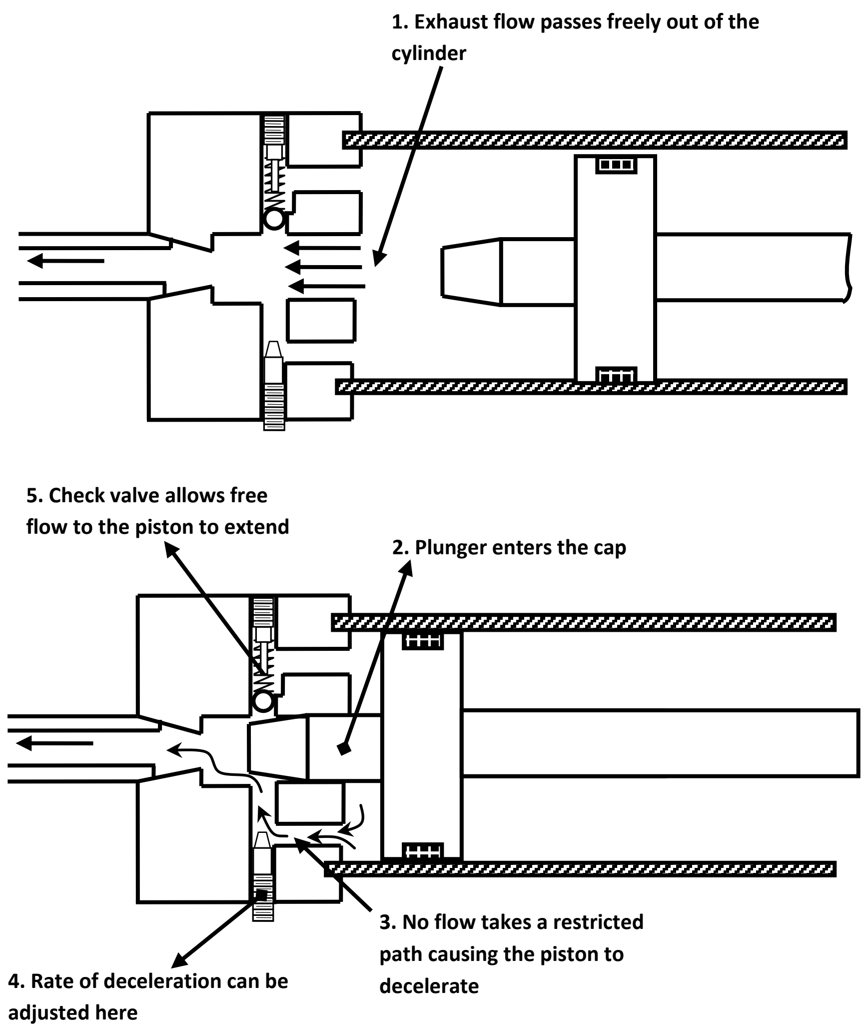


Figure 1. 28 Operation of cylinder cushions

1. 8.1 Cushioning Pressure

During deceleration, extremely high pressure may develop within a cylinder cushion. The action of the cushioning device is to set up a back-pressure to decelerate the load.

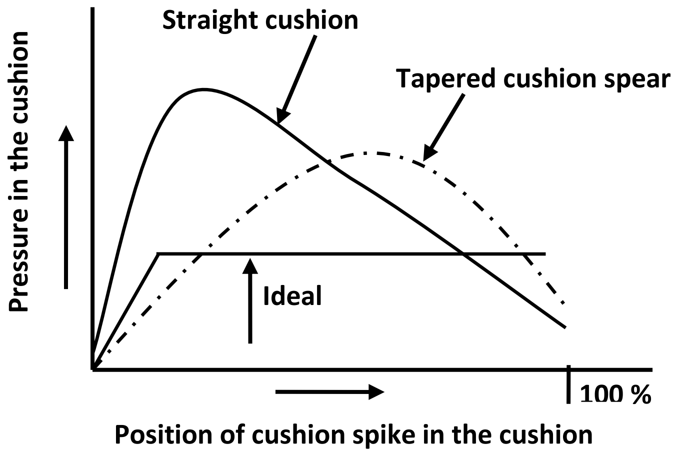


Figure 1. 29 Pressure distribution in cushioning

Ideally, the back-pressure is constant over the entire cushioning length to give a progressive load deceleration. In practice, cushion pressure is the highest at the moment when the piston rod enters the cushion(Fig. 1. 29).Some manufacturers have improved the performance of their cushioning devices by using a tapered or a stepped cushion spear. Wherever high inertia loads are encountered, the cylinder internal cushions may be inadequate but it is possible for the load to be retarded by switching in external flow controls. Deceleration can then take place over a greater part of the actuator stroke.

1. 8.2 Maximum Speeds in Cushioned Cylinders

The maximum speed of a piston rod is limited by the rate of fluid flow into and out of the cylinder and the ability of the cylinder to withstand the impact forces that occur when the piston movement is arrested by the cylinder end plate.

In an uncushioned cylinder, it is normal to limit the maximum piston velocity to 8m/min. This value is increased to 12 m/min for a cushioned cylinder, and 30 m/min is permissible with high-speed or externally cushioned cylinders. Oversize ports are necessary in cylinders used in high-speed applications. In all cases, the maximum speed depends upon the size and type of load. It is prudent to consult the manufacturer if speeds above 12 m/min are contemplated.

When only a part of the cylinder stroke is utilized, cushions cannot be used to decelerate the load. In such cases, it may be necessary to introduce some form of external cushioning especially where high loads or precise positioning is involved.

Example 1. 20

A cylinder has a bore of 125 mm diameter and a rod of 70 mm diameter. It drives a load of 2000 kg vertically up and down at a maximum velocity of 3 m/s. The lift speed is set by adjusting the pump displacement and the retract speed by a flow control valve. The load is slowed down to rest in the cushion length of 50 mm. If the relief valve is set at 140 bar, determine the average pressure in the cushions on extend and retract. (Neglect pressure drops in pipe work and valves.)

Solution: Kinetic energy of load

$$\text{Kinetic energy} = (1/2)\ \text{Mass} \times \text{Velocity}^2$$

$$= (1/2)\ (2000) \times 3^2 = 9000 \text{ N m}$$

Average force to retard load over 50 mm is

$$\frac{\text{Kinetic energy}}{\text{Distance}} = \frac{9000\times10^3}{50} = 180 \text{ kN}$$

The force acting on the load is

$$\text{Load} = 2000 \text{ kg} = 2000 \times 9.81 = 19.6 \text{ kN}$$

Annulus area = $\frac{\pi}{4}(0.125^2 - 0.07^2) = 0.0084 \text{ m}^2$

Full bore area = $\left(\frac{\pi}{4}\right)\times(0.125^2) = 0.0123 \text{ m}^2$

The kinetic energy of the load is opposed by the cushion force and the action of gravity on the load. Cushion pressure to absorb the kinetic energy of load when extending is

$$\frac{(180\times10^3) - (19.6\times10^3)}{(8.4\times10^{-3})} \text{ (N/ m}^2\text{)} = 19.1 \times 10^6 = 191 \text{ bar}$$

When the piston enters the cushion, the pressure on the full bore side of the piston rises to relief valve pressure. This pressure on the full bore side drives the piston into the cushions, and so increases the cushion pressure needed to retard the load. The cushion pressure to overcome the hydraulic pressure on the full bore end is

$$\text{Pressure} \times \frac{\text{Full bore area}}{\text{Annulus area}} = 140 \times \frac{12.3\times10^{-3}}{8.4\times10^{-3}} = 205 \text{ bar}$$

Thus, the average pressure in the cushion on the extend stroke is (190 + 205) = 395 bar.
During cushioning, the effective annular area is reduced as the cushion sleeve enters the cushion. This has been neglected in the calculation, and in practice, the cushion pressure is even greater.
When the load is retracted, forces act on the load. The back pressure owing to the flow control valve in the circuit is minimal once the piston enters the cushion and is neglected in this calculation.
The force in the cushion has to overcome the kinetic energy of the load, the weight of the load and the force due to the hydraulic pressure. The force owing to the hydraulic pressure is

$$\text{Force} = \text{Pressure} \times \text{Annulus area}$$
$$= (140 \times10^5) \times (8.4 \times10^{-3}) \text{ N}$$
$$= 117.6 \text{ kN}$$

Also

$$\text{Cushion force} = 180 + 19.6 + 117.6 = 317.2 \text{ kN}$$

After knowing the force, we can find cushion pressure =

$$\text{Cushion pressure} = \frac{\text{Force}}{\text{Area}} = \frac{317.2}{0.0123} \text{ (kN/ m}^2\text{)} = 25800 = 258 \text{ bar}$$

The average pressure in the cushion retracting is 258 bar. Again this value is somewhat higher as the cushion spike reduces the effective cushion area below that used.

Example 1. 21

A pump delivers oil at a rate of 1.15 LPS into the blank end of the 76.2 mm diameter hydraulic cylinder shown in Fig. 1. 30. The pistons decelerate over a distance of 19.05 mm at the end of its extension stroke. The cylinder drives a 6672 N weight which slides on a flat horizontal surface having a coefficient of friction (CF) equal to 0.12. The pressure relief valve setting equals 51.7125 bar. Therefore, the maximum pressure (p_1) at the blank end of the cylinder equals 51.7125 bar while the cushion decelerates the piston. Find the maximum pressure (p_2) developed by the cushion.

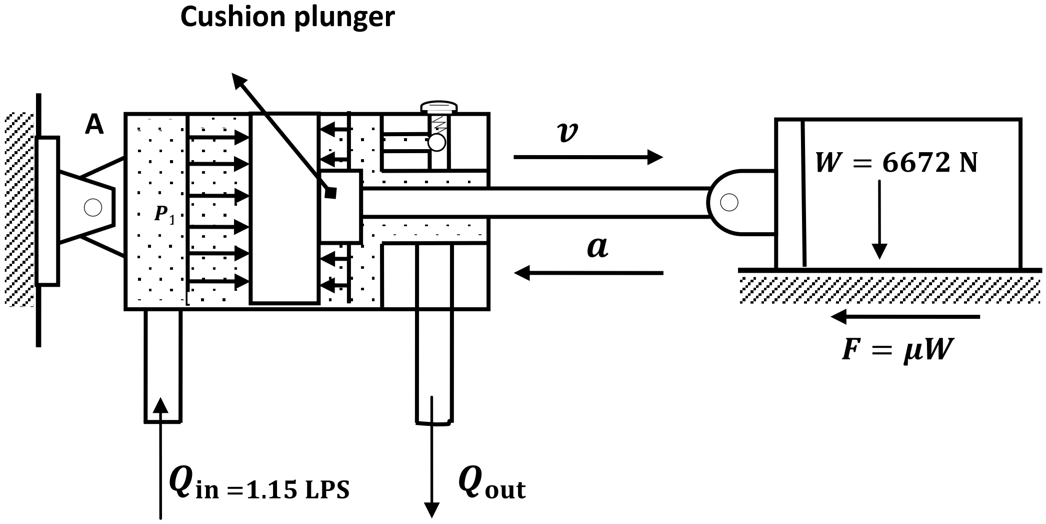


Figure 1. 30

Solution:

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

$$v = \frac{Q_{\text{pump}}}{A_{\text{piston}}} = \frac{\dfrac{1.15}{1000}}{\dfrac{\frac{\pi}{4}(0.0762^2)}{1}} = \frac{1.15}{4.5} = 0.255 \text{ m/s}$$

Step 2: Calculate the deceleration a of the piston during the 19.05 mm displacement S using the constant acceleration or deceleration equation:

$$v^2 = 2as$$

Substituting the values and solving for deceleration we get

$$a = \frac{v^2}{2s} = \frac{0.255^2}{2(19.05 \times 10^{-3})} = 17.06 \text{ m/s}^2$$

Step 3: Using Newton's laws of motion, the net force acting on the system is equated as

$$\sum F = ma$$

Consider the forces that tend to slow down the system as positive forces as we are solving for deceleration. The mass under consideration m is equal to the sum of all the masses of moving members (piston, road and load). Because the weight of the piston and road is small compared to the weight of the load, the weight of the piston and rod is ignored. The frictional forces acting between the weight W and the horizontal support surface equal *coefficient of friction* (CF) times W. This frictional force is the external force acting on the cylinder while it moves the weight.

Substituting into Newton's equations yields

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

$$p_2 = \frac{\left[(6672)\left(\frac{17.06}{9.81}\right)\right] + 51.7125\left(\frac{\pi}{4}\right)(76.2\times10^{-3})^2 - (0.12)(6672)}{\left[\frac{\pi}{4}(76.2\times10^{-3})^2 - \frac{\pi}{4}\left(25.4\times10^{-3}\right)^2\right]} = 59.0212\,\text{bar}$$

Thus, the hydraulic cylinder must be designed to withstand an operating pressure of $59.0212\,\text{bar}$ rather than the pressure relief setting of $51.7125\,\text{bar}$.

1. 9 Cylinder Mountings and Strength Calculations

The types of mounting on cylinders are numerous, and they can accommodate a wide variety of applications. One of the important considerations in selecting a particular mounting is whether the force applied is tensile or compressive. As far as possible, bucking load must be avoided. The ratio of rod length to diameter should not exceed 6:1 to prevent bucking. Alignment of the rod with the resistive load is another important consideration while selecting cylinder mounts. The various kinds of mountings normally used in industries are as follows (for various mounting, refer Fig. 1. 31):

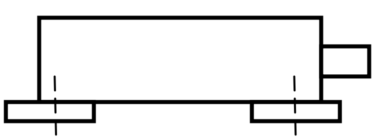
1. **Foot mounting:** It should be designed to give a limited amount of movement on one foot only to allow for thermal or load expansion. That is, the cylinder should be positively located or dowelled at one end only.

2. **Rod-end flange or front flange mounting:** During the extend stroke, pressure in the hydraulic fluid acts on the cylinder-end cap, the force set up being transmitted to the front mounting flange through the cylinder body.

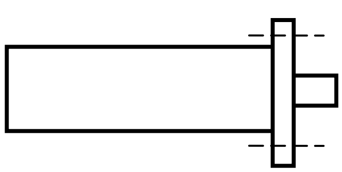
3. **Rear flange, back flange or head-end flange mounting:** No stress is present in the cylinder owing to load on the extend stroke; only hoop stress is present. The load acts through the fluid onto the rear flange.

4. **Trunnion mounting:** It allows angular movement. It is designed to take shear load only. Bearing should be as close to the cylinder body as possible.

5. Eye or clevis mounting: There is a tendency for the cylinder to jack knife under load. Side loading of bearing must be carefully considered.



(a)



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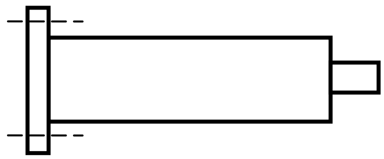

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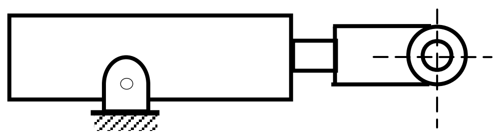

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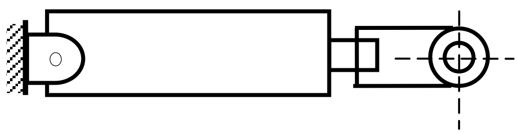

(e)

Figure 1. 31 (a) Foot mounting; (b) rod-end flange or front flange mounting; (c) rear flange, back flange or head-end flange mounting; (d) trunnion mounting; (e) eye or clevis mounting.

1. 9.1 Piston Rod Ends

The piston rod ends can be supplied with a male or female thread according to the manufacturer's specification. Rod-end eyes with spherical bearings are available from some suppliers.

1. 9.2 Protective Covers

These are fitted to protect the piston rod when the piston works in an abrasive environment or when the cylinder is not used for long periods and a heavy deposit of dust accumulates on the rod. The protective covers are of the form of telescope or bellows and completely enclose the rod at all times in the cylinder movement.

Bellows may either be molded or fabricated. Molded bellows are manufactured from rubber or plastic and owing to their construction; they are limited to a contraction ratio of about 4:1. An extended piston rod is required to accommodate the closed length of the bellows. This increases the overall cylinder length and tends to restrict their use to relatively short stroke cylinders.

Fabric covers made of plastic, leather, impregnated cloth or canvas can have a contraction ratio greater than 15:1. When a fabricated cover is used on a horizontal cylinder, it must be supported externally to prevent the cylinder rod from rubbing the cover.

Telescopic covers are made of a rigid material, normally metal, and are used under conditions where fabric covers are inadequate.

1. 9.3 Piston Rod Buckling

A piston rod in a hydraulic cylinder acts as a strut when it is subjected to a compressive load or it exerts a thrust. Therefore, the rod must be of sufficient diameter to prevent buckling. Euler's strut theory is used to calculate a suitable piston rod diameter to withstand buckling. Euler's formula states that

$$F_b = \frac{\pi^2 EI}{L^2}$$

where F_b is the buckling load (kg), E is the modulus of elasticity (kg/cm^2; 2.1×10^6 kg/cm^{2}for steel), I is the second moment of inertia of the piston rod (cm^4; $\pi d^2/64$ for a solid rod of diameter d cm) and L is the free (equivalent) buckling length (cm) depending on the method of fixing the cylinder and piston rod and is shown in Fig.1. 32. The maximum safe working thrust or load F on the piston rod is given by

$$F = \frac{F_b}{S}$$

where S is the factor of safety that is usually taken as 3.5. The free or equivalent buckling length L depends on the method of fixing the piston rod end and the cylinder, and on the maximum distance between the fixing points, that is, the cylinder fully extended. In cases where the cylinder is rigidly fixed or pivoted at both ends, there is a possibility of occurrence of excessive side loading. The effect of side loading can be reduced by using a stop tube inside the cylinder body to increase the minimum distance between the nose and the piston bearings. Refer Fig. 1. 33 for use of a stop tube to minimize side loading. The longer the stop tube, the lower the reaction force on the piston owing to the given value of the side load. Obviously, the stop tube reduces the effective cylinder stroke.

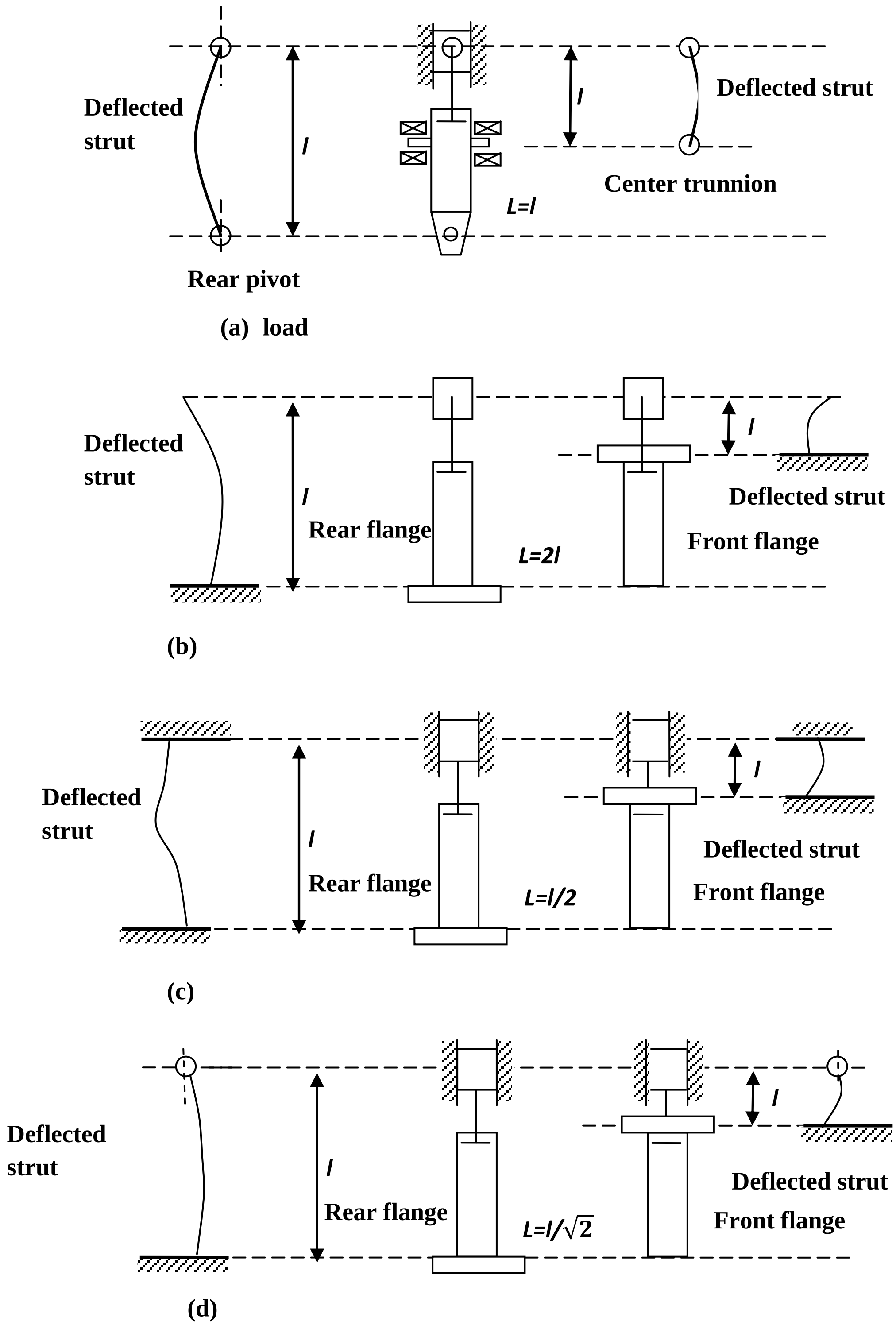

Figure 1. 32 Relationship between the piston rod, free buckling length and method of fixing. (a) Rear pivot and center trunnion mounted, guided pivoted. (b) One end rigidly fixed, free load.(c) One end rigidly fixed, guided load. (d) One end rigidly fixed, pivoted and guided load

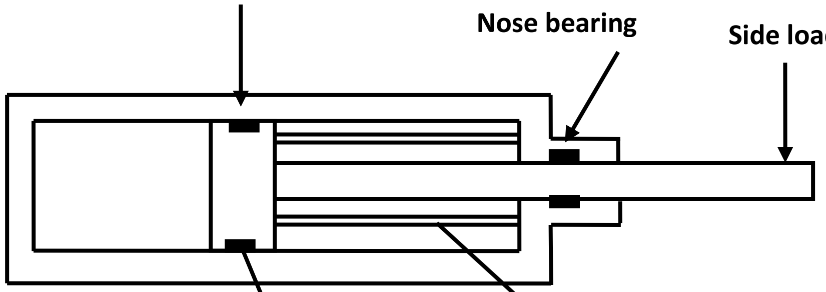

Figure 1. 33 Use of a stop tube to minimize side loading

Objective-Type Questions
Fill in the Blanks

1. An actuator is used to convert the _______ back into the _______.
2. A telescopic cylinder is used when _______ stroke length and _______ retracted length are required.
3. In a push type, the cylinder _______ to lift a weight against the force of gravity by applying oil pressure at the _______.
4. The drawback of tandem cylinder is that it is _______ than a standard cylinder, achieves an equal speed because flow must go to _______.
5. A major problem in the manufacture of through-rod cylinders is achieving _______ and concentricity of cylinder bore.

State True or False

1. Hydraulic actuators are devices used to convert the pressure energy of the fluid into mechanical energy.
2. A telescopic cylinder is used in applications where a large amount of force is required from a small-diameter cylinder.
3. Semi-rotary actuators are capable of limited angular movements that can be several complete revolutions.
4. Single-acting cylinders can exert a force in both the extending and retracting directions.
5. In pull-type gravity, return-type single-acting cylinder, the cylinder lifts the weight by extending.

Review Questions

1. What is the function of a hydraulic cylinder in a hydraulic system?
2. When is a telescoping cylinder used?
3. Explain the operation of tandem-type cylinder and mention its applications.
4. Explain the function of cushioning in cylinders.
5. Why are wiper rings used on cylinder rods?
6. Mention two applications of single-acting cylinders.

7. How does a welded type of cylinder differ from a tie-rod type? Mention the major parts of a tie-rod cylinder.
8. What are the technical specifications of a hydraulic cylinder?
9. Name the materials that are commonly used to manufacture (a) cylinder covers,(b) piston rods,(c) pistons and (d) tie-rods.
10. What is a hydraulic ram?
11. Mention the different types of mountings used in fixing the hydraulic cylinders.
12. What is the difference between a single-acting and a double-acting hydraulic cylinder?
13. Name four different types of hydraulic cylinder mountings.
14. What is a cylinder cushion? What is its purpose?
15. What is a double-rod cylinder? When would it normally be used?
16. What is a telescoping rod cylinder? When would it normally be used?
17. Differentiate between first-, second- and third-class lever systems used with hydraulic cylinders to drive loads.
18. When using a lever system with hydraulic cylinders, why must the cylinder be clevis mounted?
19. What is the purpose of a hydraulic shock absorber? Name two applications.
20. What is a hydraulic actuator?
21. How is a single-acting cylinder retracted?
22. What are the advantages of a double-acting cylinder over the single-acting cylinder?
23. For which applications, a double-rod cylinder is best suited?
24. What are the advantages and disadvantages of a tandem cylinder?
25. Name the types of cylinder mounting.

Answers
Fill in the Blanks

1.Fluid energy, mechanical power
2.Long, short
3.Extends, blank end
4.Longer, both pistons
5.Correct alignment

State True or False

1. True
2. False
3. True
4. False
5. False