Hydraulics and Pneumatics
HYDRAULIC CYLINDER OPERATING FEATURES
eNotes

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HYDRAULIC CYLINDER OPERATING FEATURES

The simplest type of hydraulic cylinder is the single-acting design. It consists of a piston inside a cylindrical housing called a barrel. Attached to one end of the piston 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 accompanied by using gravity or by the inclusion of a compression spring in the rod end.

A graphic symbol implies how a component operates without showing any of its construction details. In drawing hydraulic circuits graphic symbols of all components are used. This facilitates circuit analysis and troubleshooting. The symbols, which are merely combinations of simple geometric figures such as circles, rectangles, and lines, make no attempt to show the internal configuration of a component. However symbols clearly show the function of each component.

A double-acting cylinder can be extended and retracted hydraulically. Thus, an output force can be applied in two directions (extension and retraction). This type of cylinder has working pressure rating of 40 bar to 400 bar depending on the constructional details and application needs. The output force is proportional to the bore area.

The barrel is made of seamless steel tubing honed to a fine finish on the inside. The piston which is made of ductile iron has grooves in which are fitted seals (U-cup packings) to seal against leakage between the pistol and barrel. The ports are located in the end caps, which are secured to the barrel by tie rods. The tapered cushion plungers provide smooth deceleration at both ends of the stroke. Therefore, the piston does not bang into the end caps with excessive impact, which could damage the hydraulic cylinder after a given number of cycles. The static sealing at the ends is achieved by use of ‘O’ Rings.
CYLINDER MOUNTINGS AND MECHANICAL LINKAGES

Various types of cylinder mountings are in existence. This permits versatility in the anchoring of cylinders. The rod ends are usually threaded so that they can be attached directly to the load, a clevis, a yoke, or some other mating device.

Through the use of various mechanical linkages, the applications of hydraulic cylinders are limited only by the ingenuity of the fluid power designer. These linkages can transform a linear motion into either an oscillating or rotary motion. In addition, linkages can also be employed to increase or decrease the effective leverage and stroke of a cylinder.

Much efforts has been made by manufacturers of hydraulic cylinder to reduce or eliminate the side loading of cylinders created as a result of misalignment. It is almost impossible to achieve perfect alignment of a hydraulic cylinder, even though the alignment of the cylinder has a direct bearing on its life.

A universal alignment mounting accessory designed to reduce misalignment problems is developed. By using one of these accessory components and a mating clevis at each end of the cylinder, the following benefits are obtained.

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

A large number of cylinders use the clevis and trunnion concept to avoid side-loads and they are very common in construction machines such as loaders, excavators, dozers etc., as well as material handling equipment such as cranes.
The output force \((F)\) and piston velocity \((v)\) 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 piston rod and cylinder bore \(A_p - A_r\), where \(A_p\) equals the piston area and \(A_r\) equals the rod area. The difference in flow-path cross-sectional area accounts for the difference in piston velocities. Since \(A_p\) is greater than \(A_p - A_r\), the retraction velocity is greater than the extension velocity for the same input flow-rate.

Similarly during the extension stroke, fluid pressure bears on the entire circular area of the piston. However, during the retraction stroke, fluid pressure bears only on the smaller annular area between the piston rod and cylinder bore. This difference in area accounts for the difference in output forces. Since \(A_p\) is greater than \(A_p - A_r\), the extension force is greater than the retraction force for the same operating pressure.

Equations (6-1) through (6-4) allow for the calculation of the output force and velocity for the extension and retraction strokes of 100% efficient double-acting cylinders.

**Extension stroke**

\[
F_{ext}(1b) = p \text{ (psi)} \times A_p \text{(in}^2) \tag{6-1}
\]

\[
F_{ext}(N) = p \text{ (Pa)} \times A_p \text{(m}^2) \tag{6-1M}
\]

\[
v_{ext}(\text{ft/s}) = \frac{Q_{in}(\text{ft}^3/\text{s})}{A_p(\text{ft}^2)} \tag{6-2}
\]

\[
v_{ext}(\text{m/s}) = \frac{Q_{in}(\text{m}^3/\text{s})}{A_p(\text{m}^2)} \tag{6-2M}
\]

**Retraction Stroke**

\[
F_{ret}(1b) = p \text{ (psi)} \times (A_p - A_r) \text{ in}^2 \tag{6-3}
\]

\[
F_{ret}(N) = p \text{ (Pa)} \times (A_p - A_r) \text{ m}^2 \tag{6-3M}
\]

\[
F_{ret}(\text{ft/s}) = \frac{Q_{in}(\text{ft}^3/\text{s})}{(A_p - A_r)\text{ft}^2} \tag{6-4}
\]
\[ v_{\text{ret}}(\text{m/s}) = \frac{Q_{\text{in}}(\text{m}^3/\text{s})}{(A_p - A_r)\text{m}^2} \]  

(6-4M)

The power developed by a hydraulic cylinder equals the product of its force and velocity during a given stroke. Using this relationship and Eqs. (6-1) and (6-2) for the extending stroke and Eqs. (6-3) and (6-4) for the retraction stroke, we arrive at the same result: 

\[ \text{Power} = p \times Q_{\text{in}}. \]

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)

\[ \text{Power (HP)} = \frac{v_p(\text{ft/s}) \times F(1\text{b})}{550} = \frac{Q_{\text{in}}(\text{gpm}) \times p(\text{psi})}{1714} \]  

(6-5)

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

\[ \text{Power (kW)} = \frac{v_p(\text{m/s}) \times F(\text{kN})}{p(\text{kPa})} \]  

(6-5M)

Note that the equating of the input hydraulic power and output mechanical power in Eqs. (6-5) and (6-5M) assumes a 100% efficient hydraulic cylinder.

**SPECIAL CYLINDER DESIGNS**

For a double-rod cylinder in which the rod extends out of the cylinder at both ends, 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. A good example is in the table reciprocation of surface grinding machines.

The telescopic cylinder actually 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. Other examples are in dampers, vertical hoists etc.,

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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. The following example illustrates how to calculate this pressure, which decelerates the piston at the ends of its extension and retraction strokes.

Worked example for cushion pressure calculation:

Problem:
A pump delivers oil at the rate of 18.2 gpm into the blank end of the 3-in-diameter hydraulic cylinder. 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:

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

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

\[ A = \frac{v^2}{2S} \]

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 force \( \Sigma F \) 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 the rod is small compared to the weight of the load, the weight of the piston and rod will be ignored.

Also note that the 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 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

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

Solving for \( p_2 \) yields a usable equation:

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

Substituting known values produces the desired result:

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

\[ p_2 = \frac{257 + 5303 - 180}{5380} = 856 \text{ psi} \]

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7.07 – 0.785

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.