FPE IAT 2 Scheme & Solution

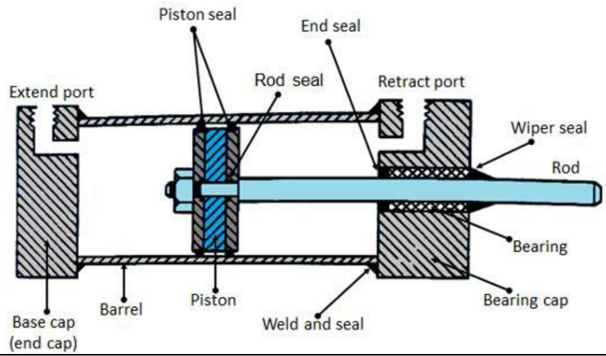
1. Definition of actuator. 2M
   Classification 2 M
   Explanation of 2 types of cylinders with neat diagrams. 6M
2. Double acting cylinder line diagram with differential area 2M
   Derivation showing expressions, Force , velocity and Power during extension and retraction. 3+3+2 M
3. Numerical:
   Use double acting expressions and find Q_{in} 5M
   V_{ret} 5M
4. Hydraulic motor performance
   Volumetric efficiency
   Mechanical efficiency
   Overall efficiency
   Power developed 2.5 *4 = 10M
5. Diagram of cylinder with cushioning 4 M
   Explanation 6M
6. Balanced Vane motor diagram 4 M
   Explanation 6M

Solutions

1. Pumps perform the function of adding energy to a hydraulic system for transmission to some remote point. Fluid power actuators just do the opposite. Actuators extract energy from the fluid and convert it to a mechanical output to perform useful work. A hydraulic or pneumatic system is generally concerned with moving, gripping or applying force to an object. Devices which actually achieve this purpose are called actuators. Hydraulic actuators are installed to drive loads by converting the hydraulic power into mechanical power. The hydraulic actuators are classified into two main groups according to motion type:

   1. Linear actuators (Cylinders), as the name implies, are used to move an object or apply a force in a straight line.
   2. Rotary actuators (Motors) are the hydraulic and pneumatic equivalent of an electric motor.

The main parts of a hydraulic double acting cylinder are: piston, piston rod, cylinder tube, and end caps. These are shown in Figure. The piston rod is connected to piston head and the other end extends out of the cylinder. The piston divides the cylinder into two chambers namely the rod end side and piston end side. The seals prevent the leakage of oil between these two chambers. The cylindrical tube is fitted with end caps. The pressurized oil, air enters the cylinder chamber through the ports provided. In the rod end cover plate, a wiper seal is provided to prevent the leakage of oil and entry of the contaminants into the cylinder. The combination of wiper seal, bearing and sealing ring is called as cartridge assembly. The end caps may be attached to the tube by threaded connection, welded connection or tie rod connection. The piston seal prevents metal to metal contact and wear of piston head and the tube. These seals are replaceable. End cushioning is also provided to prevent the impact with end caps.



2. The output force (F) and piston velocity (v) of double-acting cylinders are not the same for extension and retraction strokes.

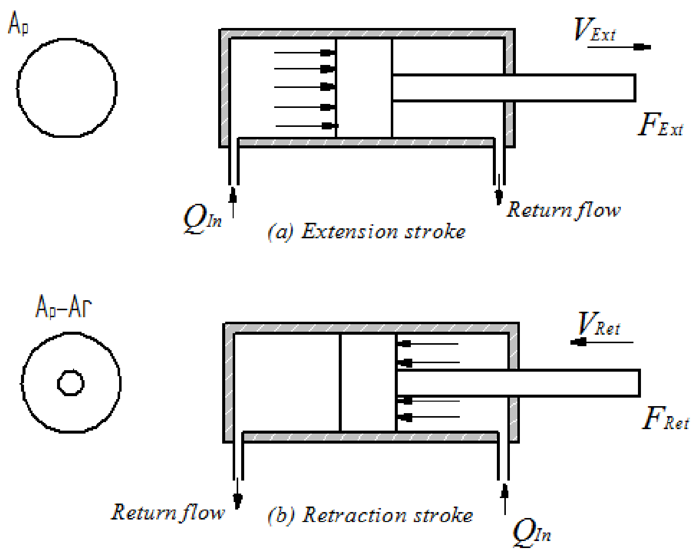


Figure Effective areas during (a) extension strokes and (b) retraction strokes

During the extension stroke shown in Fig. (a),the fluid pressure acts on the entire circular piston area A_p. During the retraction stroke, the fluid enters the rod-end side and the fluid pressure acts on the smaller annular area between the rod and cylinder bore ($A_p - A_r$) as shown by the shaded area in Fig.1.11(b) (A_r is the area of the piston rod). Due to the difference in the cross-sectional area, the velocity of the piston changes. Because A_p is greater than ($A_p - A_r$), the retraction velocity (v_{ret}) is greater than the extension velocity (v_{ext}) for the same flow rate.

During the extension stroke, the fluid pressure acts on the entire piston area (A_r), while during the retraction stroke, the fluid pressure acts on the annular area ($A_p - A_r$). This difference in area accounts for the difference in output forces during extension and retraction strokes. Because A_r is greater than $A_p - A_r$), the extension force is greater than the retraction force for the same operating pressure.

Force and velocity during extension stroke

Force,

$$v_{\text{ext}} = \frac{Q_{\text{in}}}{A_{\text{p}}}$$

Velocity,

$$F_{\text{ext}} = p \times A_{\text{p}}$$

Force and velocity during retraction stroke
Velocity,

$$v_{\text{ext}} = \frac{Q_{\text{in}}}{A_{\text{p}} - A_{\text{r}}}$$

Force

$$F_{\text{ext}} = p \times (A_{\text{p}} - A_{\text{r}})$$

Power developed by a hydraulic cylinder (both in extension and retraction) is
Power Force ×Velocity = F × V
In metric units, the kW power developed for either extension or retraction stroke is

$$\text{Power (kW)} = v_{\text{p}}\,(\text{m/s}) \times F\ (\text{kN})$$

$$= Q_{\text{in}}\,(\text{m}^3/\text{s}) \times p\ (\text{kPa})$$

Power during extension is

$$P_{\text{ext}} = F_{\text{ext}} \times v_{\text{ext}} = p \times A_{\text{p}} \times \frac{Q_{\text{in}}}{A_{\text{p}}} = p \times Q_{\text{in}}$$

Power during retraction is

$$P_{\text{ret}} = F_{\text{ret}} \times v_{\text{ret}}$$

$$= p \times (A_{\text{p}} - A_{\text{r}}) \times \frac{Q_{\text{in}}}{A_{\text{p}} - A_{\text{r}}}$$

$$= p \times Q_{\text{in}}$$

Comparing Equations, we can conclude that the powers during extension and retraction strokes are the same.

Example 1.5

An 8 cm diameter hydraulic cylinder has a 4 cm diameter rod. If the cylinder receives flow at 100 LPM and 12 MPa, find the (a) extension and retraction speeds and (b) extension and retraction load carrying capacities.

Solution:

Let us first convert the flow in LPM to m^3/s before we calculate forward velocity Q_{in}=100 LPM = 100/(1000 × 60) =1/600 m^3/s

Now

D_C = Diameter of cylinder = 8 cm = 8×10^{-2} m

d_r = Diameter of piston rod = 4 cm = 4×10^{-2} m

p = 12 MPa = 12×10^6 N/m^2 or Pa

(a) Forward velocity is given by

$$v_{ext} = \frac{Q_{in}}{A_p} = \frac{1/600}{\pi d^2/4} = 0.3315 \text{ m/s}$$

Return velocity is given by

$$v_{ret} = \frac{Q_{in}}{(A_p - A_r)} = \frac{\frac{1}{600}}{\frac{\pi(d_C^2 - d_r^2)}{4}} = 0.442 \text{ m/s}$$

(b) Force during extension is given by

$$F_{ext} = p \times a_p = 12 \times 10^6 \frac{\pi(8\times10^{-2})^2}{4} = 60318.57 \text{ N}$$

3.

4. A hydraulic motor has a displacement of 164 cm^3 and operates with a pressure of 70 bar and a speed of 2000 rpm. If the actual flow rate consumed by the motor is 0.006 m^3/s and the actual torque delivered by the motor is 170 Nm, find (a) η_V (b) η_M (c) η_o (d) actual power delivered by the motor.

Solution:

(a) We have

$$\eta_v = \frac{\text{Theoretical flow rate the motor should consume}}{\text{Actual flow rate consumed by the motor}} = \frac{Q_T}{Q_A}$$

Now $Q_A = 0.006\ \text{m}^3/\text{s}$. Theoretical flow rate is

$$Q_T = V_D \times N = 164 \times 10^{-6}\ (\text{m}^3/\text{rev}) \times \frac{2000}{60}\ (\text{rev/s}) = 0.0055\ \text{m}^3/\text{s}$$

So volumetric efficiency is

$$\eta_v = \frac{0.0055}{0.006} \times 100 = 91.67\%$$

(b) Mechanical efficiency is given by

$$\eta_m = \frac{\text{Actual torque delivered by the motor}}{\text{Theoretical torque motor should deliver}} = \frac{T_A}{T_T}$$

Theoretical torque,

$$T_T = \frac{p \times V_D}{2\pi} = \frac{70 \times 10^5 \times 164 \times 10^{-6}}{2\pi} = 182.71\ \text{N m}$$

So mechanical efficiency,

$$\eta_m = \frac{170}{182.71} = 93.04\%$$

(c) We have

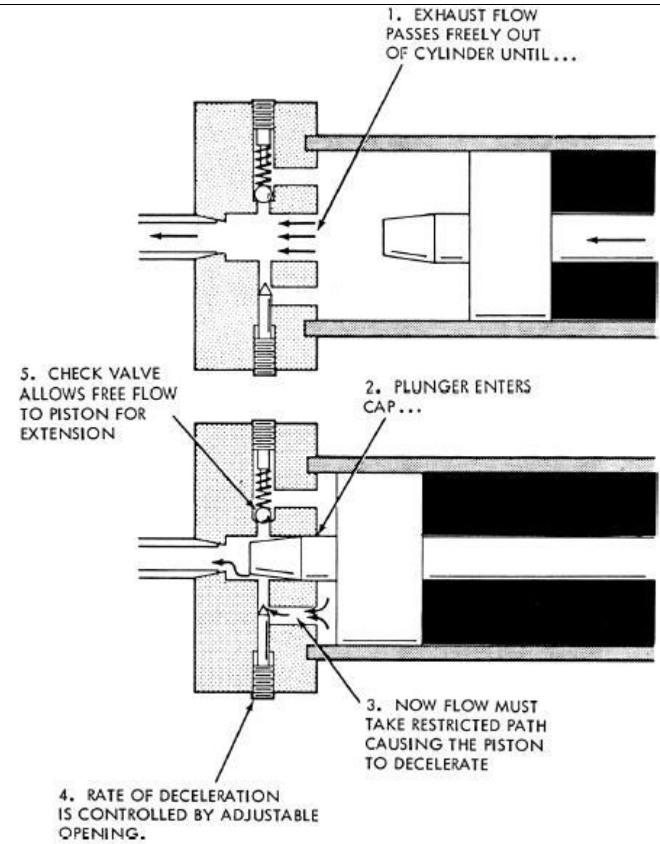
$$\eta_o = \eta_m \times \eta_v = 0.9304 \times 0.9167 = 0.853 = 85.3\%$$

So overall efficiency is 85.3 %.

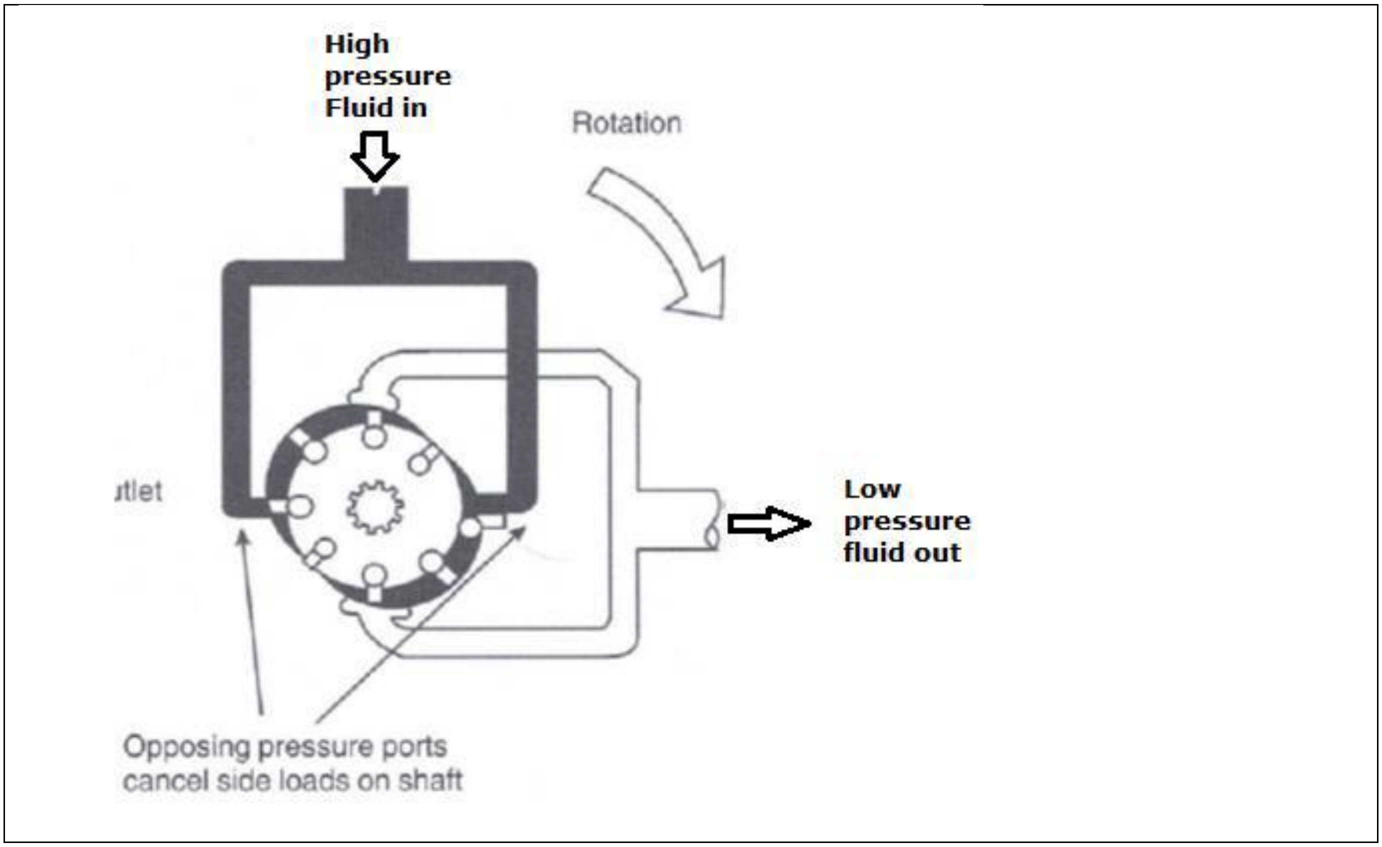
(d) Actual power is

5. The extension and retraction speeds of hydraulic cylinders are managed by controlling the inlet or exit-oil flow rates. When reaching its end position, the piston is suddenly stopped. In the case of high speed and/or great inertia, the sudden stopping of the piston results in a severe impact force. It affects both the cylinder and the driven mechanism. Therefore, a cushioning arrangement might be necessary to reduce the piston speed to a limiting value before piston reaches its end position.

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.



6. Figure shows the balanced vane motor. The radial bearing load problem is eliminated in this design by using a double-lobed ring with diametrically opposite ports. Side force on one side of bearing is canceled by an equal and opposite force from the diametrically opposite pressure port. The like ports are generally connected internally so that only one inlet and one outlet port are brought outside. The balanced vane-type motor is reliable open-loop control motor but has more internal leakage than piston-type.


High pressure Fluid in
Rotation
utlet
Low pressure fluid out
Opposing pressure ports cancel side loads on shaft