

CHAPTER 5

ACTUATORS

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Examples and Exercise

5.1 Introduction

Hydraulic cylinders and motors form the hydraulic circuit component called the actuators. They extract energy from the pressurized fluid and convert it to mechanical energy to perform linear or rotary motions.

Hydraulic cylinders (linear actuators) extend and retract a piston rod to exert a force on an external load along a straight line path.

Hydraulic motors (rotary actuators) rotate a shaft to provide a torque to drive the load along a rotary path. The rotation

Actuator classification is given in [fig-chp5\fig5.1.pptx](#)

5.2 Cylinders

Single acting design is the simplest type of hydraulic cylinder, shown in [fig-chp5\fig5.2.pptx](#) . It can exert a force only in the extending direction. Retraction is accomplished by gravity or spring.

The double acting type delivers force in both directions. Extension and retraction are by hydraulic means. [fig-chp5\fig5.3.pptx](#) shows the design of a double acting hydraulic cylinder. Piston has U-cup packings for sealing. Ports are located in the end caps secured to the barrel by tie rods.

5.2.1 Cylinder loading

Vertical loading – load on cylinder is the weight of the object.

Sliding an object along a horizontal surface-the load is to overcome frictional force.

Lifting a weight in an inclined direction-the cylinder load equals the component of the object's weight acting along the axis of the cylinder, plus the frictional force if the object is sliding along an inclined surface.

Acceleration-additional force the cylinder must generate to get the load from initial velocity of zero to the final steady state velocity.

5.2.2 Cylinder force, velocity and power

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

During extension stroke, fluid enters the blank end of the cylinder through the entire circular area of the piston (A_p).

However, during retraction stroke, fluid enters the rod end through the smaller annular area between the rod and the cylinder bore ($A_p - A_r$).

Where A_p – piston area
 A_r – rod area

Therefore,

Extension stroke :

$$F_{ext}(N) = p(Pa) \times A_p (m^2)$$

$$v_{ext}(m/s) = \frac{Q_{in}(m^3/s)}{A_p(m^2)}$$

Retraction stroke :

$$F_{ret}(N) = p(Pa) \times (A_p - A_r) m^2$$

$$v_{ret}(m/s) = \frac{Q_{in}(m^3/s)}{(A_p - A_r) m^2}$$

5.2.3 Cylinder loadings through mechanical linkages

In many applications the force the cylinder must handle does not act along the axis of the cylinder – the load and the cylinder force are not equal. The two forces will interact through linkages.

There are different types of linkages as shown in

[fig-chp5\fig5.4.pptx](#) which basically 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.

Analysis on how to determine the hydraulic cylinder force required in a linkage will be attempted on first-class, second-class and third-class systems. A similar analysis can be made for the other types of linkages.

First-Class Lever System

In this arrangement the lever fixed-hinge pin is located between the cylinder and the load rod pins

[fig-chp5\fig5.5.pptx](#) Equality of moments (only for vertical force) will result in

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

This will give

$$F_{\text{cyl}} = (L_2/L_1) F_{\text{load}}$$

But as shown in the figure the cylinder is mounted to allow the rod-pinned end travel along a circular path of radius L_2 about its fixed-hinge pin. If the cylinder is offset by an angle φ from the vertical, the equation will change to

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

which will give

$$F_{\text{cyl}} = [L_2/(L_1 \cos \varphi)]F_{\text{load}}$$

Second-Class Lever System

It is characterized by the load rod pin being located between the fixed-hinge pin and cylinder rod pin of the lever [fig-chp5\fig5.6.pptx](#) .

Equating the moments about the fixed-hinge will give

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

Or

$$F_{cyl} = \frac{L_2}{(L_1 + L_2) \cos \varphi} F_{load}$$

Compared to the first-class lever this requires less cylinder force (less piston area), also smaller load stroke for a given cylinder stroke.

Third-Class Lever System

Here the cylinder rod pin lies between the load rod pin and fixed-hinge pin of the lever [fig-chp5\fig5.7.pptx](#)

Again equating moments about the fixed-hinge pin will give

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

or

$$F_{cyl} = \frac{L_1 + L_2}{L_2 \cos \varphi} F_{load}$$

Cylinder force is greater than the load force (requires large piston area). Required where the load stroke must be greater than the cylinder stroke.

A good example of an industrial machine using cylinder loadings through linkages is the excavator shown in [fig-chp5\fig5.8.pptx](#). There are a total of four cylinders to drive the three pin-connected members called the ***boom***, ***stick***, and ***bucket***.

Observe that the two cylinders on the boom act like third-class lever system. For the others the hinges are also moving. To determine the loads on each cylinder, it is necessary to understand the mechanism of the linkages and make a force analysis by incorporating the given external load applied on the bucket.

5.2.4 Hydraulic cylinder cushions

Cushioning devices are provided in the ends of the hydraulic cylinders or as separate units when loads must be decelerated to prevent the excessive impact that can occur. [fig-chp5\fig5.9.pptx](#) shows the cross-section of the piston near the end cap.

Deceleration starts when the tapered plunger enters the opening in the cap and restricts the exhaust flow. During the last small portion of the stroke, the oil must pass through an adjustable opening. For charging the piston a check valve that allows free flow is incorporated in the system.

Deceleration of the load from initial steady velocity of V through a distance of S load is given by

$a = V^2/2S$ and the deceleration force is given by

Newton's second law

$$\Sigma F = ma \quad (m = \text{mass of load})$$

(care must be taken for the consistency of units)

5.2.5 Hydraulic shock absorbers

This is a device that brings a moving load to a gentle rest through the use of metered hydraulic fluid. It is the

The shock absorbers shown in [fig-chp5\fig5.10.pptx](#) are completely filled with oil. The springs serve for return.

The shock absorbers are multiple-orifice hydraulic devices. The orifices are simply holes through which a fluid can flow.

The resistance to the oil flow caused by the holes (restrictions) creates a pressure that acts against the piston to oppose the moving load. [fig-chp5\fig5.11.pptx](#) . The piston progressively shuts off these orifices as the piston and rod move inward. This results in the load decelerating uniformly.

The snubber or dash pot uses a single orifice and has a high peak force at the beginning and the resistance is sharply reduced during the remainder of the stopping distance.

Compression springs have a low initial stopping force and build to a peak at the end of the stroke. When the load is removed the spring bounces back.

5.3 Motors

These are other classes of actuators that extract energy from a fluid and convert it to mechanical energy.

Two types of motors:

- ***Rotary actuators or oscillation motor*** – limited rotation
- ***Hydraulic motor*** – continuous rotation
(gear, vane, and piston configuration)

5.3.1 Limited rotation hydraulic motors

It can rotate clockwise or counterclockwise but through less than one complete revolution.

They are used extensively in industry for actuating clamping devices, material handling, rotating cams for braking mechanisms and many others.

[fig-chp5\fig5.12.pptx](#) shows a few of the operations that require limited rotations.

[fig-chp5\fig5.13.pptx](#) shows means of a restricted achieving rotary motion.

5.3.2 Continuous rotation actuators

These provide sustained rotation in either direction due to application of torque by the pressurized fluid.

There are three basic types of hydraulic motors: gear, vane, and piston

a) Gear Motors

A gear motor develops torque due to hydraulic pressure acting on the surface of the gear teeth as shown in [fig-chp5\fig5.14.pptx](#) . The direction of rotation can be reversed by reversing the direction of flow. Volumetric displacement is fixed. The motor is not balanced with respect to pressure loads, thus producing a large side load.

They are normally limited to 130 bar pressure with operating speed of 2400 rpm and a maximum flow capacity of 570 l/m. Also available are internal gear, ge rotor, and screw type designs.

b)Vane Motors

These develop torque by the hydraulic pressure acting on the exposed surfaces of the vanes. As there is no centrifugal force until the rotor starts to revolve they will need the help of springs for sealing. The operation is shown in [fig-chp5\fig5.15.pptx](#) and the spring loading is shown in [fig-chp5\fig5.16.pptx](#) . These operate at 170 bar with speeds up to 4000 rpm and a maximum flow of 950 l/m.

By in large vane motors are of the balanced type shown in [fig-chp5\fig5.17.pptx](#)

Analysis of Torque capacity

For a single rotating vane

R_R = outer radius of rotor shaft (m)

R_V = outer radius of vane (m)

L = width of vane (m)

p = hydraulic pressure (Pa)

F = hydraulic force acting on vane (N)

A = surface area of vane in contact with oil
(m^2)

T = torque capacity (N.m)

Force on vane:

$$F = pA = p(R_V - R_R)L$$

Torque acting on the mean radius of the vane:

$$T = p(R_V - R_R)L \left(\frac{R_V + R_R}{2} \right)$$

$$= \frac{pL}{2} (R_V^2 - R_R^2)$$

The volumetric displacement can be approximated by

$$V_D = \pi (R_V^2 - R_L^2)L$$

Substituting in the torque equation will give

$$T = \frac{pV_D}{2\pi}$$

The above equation shows that the torque can be increased by increasing the pressure or the volumetric displacement.

c) Piston motors

Here we have, as in pumps, the in-line type and the bent axis type

fig-chp5\fig5.18.pptx &

fig-chp5\fig5.19.pptx

Principles of operation is shown **fig-chp5\fig5.20.pptx** in using the inline type. The tangential component (F_2) of the piston force (F) acts on the swash plate to initiate rotation. This imparts rotary motion to the piston group, cylinder block, and output shaft.

The swash plate type can be of the

Piston motors are the most efficient of the three. Operating speeds of 12000 rpm, pressures of 340 bar and flow rate of 1700 l/m (large piston motors) can be achieved.

5.3.3 Theoretical torque, power, and flow rate

This is for a frictionless operation. The expression is given by
$$T = \frac{V_D (m^3/rev) \times p (Pa)}{2\pi}$$

$$\begin{aligned} \text{Theoretical power (W)} &= T (N.m) \times \omega (\text{rad/s}) \\ &= \frac{V_D \times p \times \omega}{2\pi} \end{aligned}$$

Considering no leakage the theoretical volume flow rate will be:

$$Q_T (m^3/s) = V_D (m^3/rev) \times N (rev/s)$$

5.3.4 Motor efficiencies

Volumetric efficiency (η_v)- Inverse of that of the pump since it uses more flow than it should theoretically due to leakage.

$$\eta_v = \frac{\text{theoretical flow rate}}{\text{actual flow rate}} = \frac{\dot{Q}_T}{\dot{Q}_a}$$

Mechanical efficiency (η_m) - Since additional torque is delivered to overcome friction, the actual torque delivered is lower.

$$\eta_m = \frac{\text{actual torque delivered by motor}}{\text{theoretical torque to be delivered by motor}} = \frac{T_a}{T_T}$$

Actual power delivered by motor = $T_a \omega$

Overall efficiency

This is given by the product of the two efficiencies.

$$\eta_o = \eta_v \eta_m = \frac{\text{actual power delivered by motor}}{\text{actual power delivered to motor}}$$

$$\eta_o = \frac{T_a \omega}{p \dot{Q}_a}$$

5.3.5 Hydraulic motor performance

This topic will deal with volumetric efficiency, considering leakages, and mechanical efficiency which takes into account the frictional losses. Finally the overall efficiency which will involve both efficiencies is discussed.

Range of overall efficiencies:

Gear motors – 70 to 75 %

Vane motors – 75 to 85 %

Piston motors – 85 to 95 %

5.4 Hydrostatic transmission

A system consisting of a hydraulic pump, a hydraulic motor, and appropriate valves and pipes can be used to provide adjustable speed drives for many practical applications. Such a system is called a hydrostatic transmission. A prime mover, electric motor or gasoline engine will be required. Essentially it converts a unidirectional variable speed shaft input to a bidirectional variable speed output.

Applications in tractors, rollers, loaders, and lift trucks.

Some of the advantages include

- Infinitely variable speed and torque in either direction and over the full speed and torque ranges
- Extremely high power-to-weight ratio
- Flexibility and simplicity of design

[fig-chp5\fig5.22.pptx](#) shows a complete heavy duty hydrostatic transmission system with pump, motor and other accessories including heat exchanger.