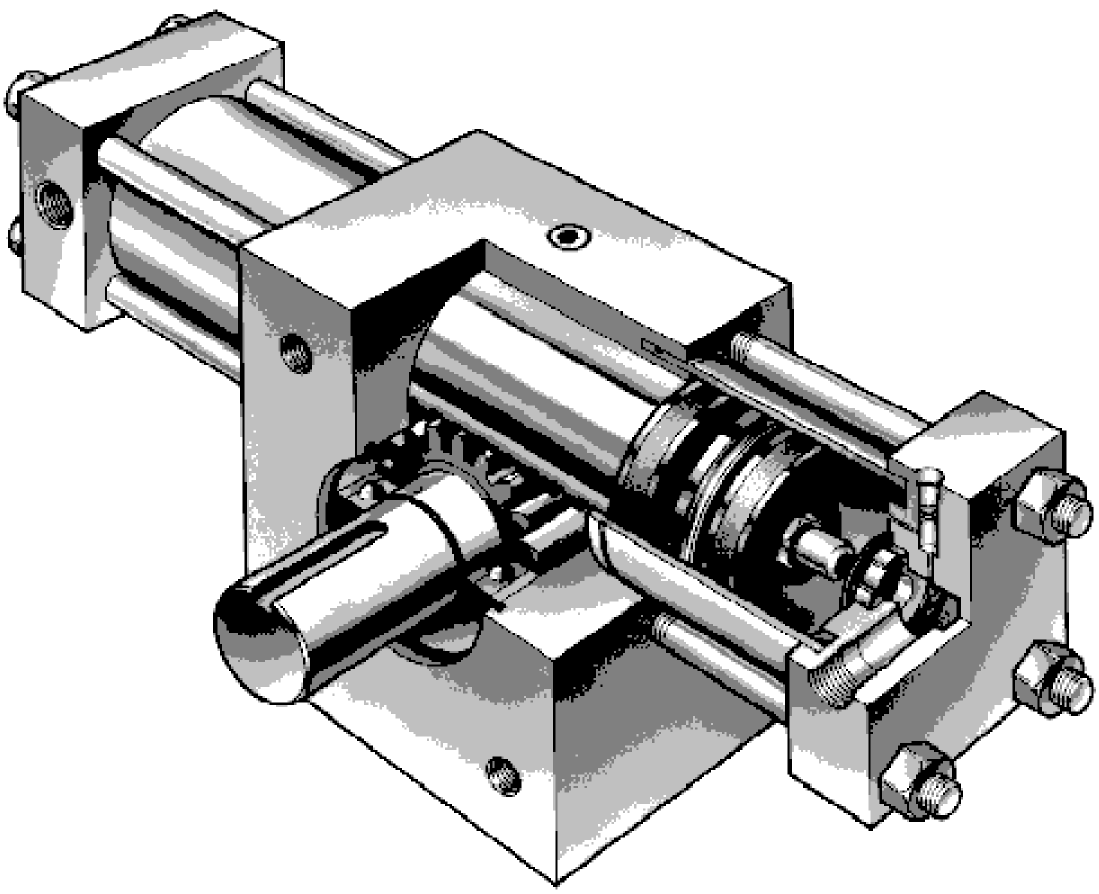
Rotary Actuator Applications Guide

For HTR and PTR/LTR Series Rack and Pinion Rotary Actuators

Bulletin 1230/2-GB



How to Use This Catalogue

This catalogue contains information to assist with the selection and application of Parker Hannifin's HTR and PTR/LTR Series rack and pinion rotary actuators. Used in conjunction with Product Catalogue no. 1220 for the HTR Series and no. 1225 for the PTR/LTR Series, it provides the technical information required to select rotary actuators for a wide range of applications. Pages 4 and 5 contain application information and standard equations for use when selecting an actuator, while worked application examples begin on page 7. The inside back page incorporates an Application Data Check List, and a copy of this should be completed and forwarded with each request for information.

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Key to Symbols

α	– angular acceleration, rad/sec^2
α^*	– angular deceleration, rad/sec^2
a	– linear acceleration, m/sec^2
ω	– angular velocity, rad/sec
v	– linear velocity, m/sec
θ	– angle of rotation [see note below], radians
M	– torque, Nm
F	– linear force, N
f	– friction

[1 radian = 57.3°]

A	– area, m^2
p	– pressure [see note below], N/m^2
r	– radius (r_p = pinion radius), m
J_m	– rotational mass moment of inertia, kgm^2
μ	– coefficient of friction
t	– time, sec
m	– mass, kg
σ	– stress, N/m^2
g	– acceleration due to gravity, 9.81m/sec^2

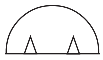
[1 bar = 10^5 N/m^2]

Introduction

A rotary actuator is the most compact device available for producing torque from hydraulic or pneumatic pressure. The symbols for each type are:

Hydraulic

Pneumatic



There are many types of rotary actuators, each offering particular advantages. The three most commonly used types are rack and pinion, vane and helical. Parker Hannifin Series HTR and PTR/LTR rotary actuators are of the rack and pinion type.

Rack and Pinion Rotary Actuators

Rack and pinion actuators consist of a housing to support a pinion which is driven by a rack with cylinder pistons on the ends. Theoretical torque output M is the product of the cylinder piston area A, operating pressure p and the pitch radius of the pinion r_p.

$$M = Apr_p$$

Single, double or multiple rack designs are possible, and overall efficiencies for rack and pinion units average 85-90%. Because standard cylinder components can be used to drive the rack, many standard cylinder features can be incorporated into the rotary actuators, eg: cushions, stroke adjusters, proximity switches and special porting. In addition, virtually leakproof seals allow the actuator to be held in any position under the load, although safety considerations may require the use of a mechanical locking device in certain applications.

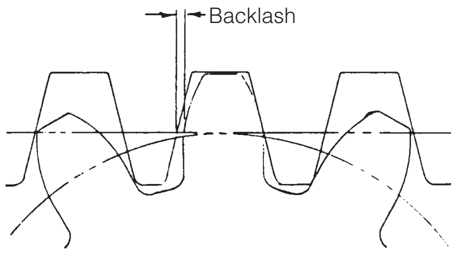
Rack and pinion rotary actuators cover a wide range of torque outputs, from less than 1Nm for pneumatic units to more than 5,000,000Nm from hydraulics. Available rotations range from a few degrees to five revolutions or more. Typical rotary actuator applications are for working pressures of up to 18 bar for pneumatic and 210 bar for hydraulic actuators, with rotations of 90°, 180° or 360°.

Position Repeatability

The positional repeatability of rotary actuators is affected by the inherent backlash found in any gear mechanism. Backlash is the amount by which the width of a tooth space exceeds the thickness of the mating tooth, and can be up to 0.5°. This can be reduced almost to zero by pre-loading the rack into the pinion, but the increased friction results in a corresponding reduction in mechanical efficiency.

Maximum Angular Backlash

See product catalogue for individual specifications.



Bearing Capacity

Because the load ratings of the bearings used to support the pinions are high in comparison to the internal loading of the unit, external load bearing capacity is usually available. This can eliminate the need for machine support bearings, or it can accommodate overhung and thrust loads. A rotary actuator with a hollow, ie: female, pinion shaft may be used in place of a coupling and support brackets, by mounting the actuator directly onto the input shaft

Rotary Actuator Applications

General Industry

- camming
- indexing
- clamping
- braking
- tensioning
- positioning
- tilting
- safety closure systems

Materials Handling

- switching conveyors
- turning and positioning container clamps on lift trucks
- tensioning and guiding
- operating valves
- braking
- lifting

Marine

- opening and closing hatches
- swinging cargo handing gear
- opening and closing fire and collision bulkhead doors
- valve operation
- positioning hydrofoils
- steering control

Robotics

- rotation and positioning

Metals Processing

- upending coils
- turnstiles
- walking beams
- immersion/agitation mechanisms
- rollover devices
- tilting electric furnaces
- crust breaking
- indexing transfer tables
- charging furnaces

Calculating Torque Requirements

Design torque represents the maximum torque that an actuator is required to supply in an application. This maximum is the greater of the **demand torque** and **cushion torque**. If the demand torque exceeds the torque that the actuator can supply, the actuator will move too slowly or will stall. If the cushion torque is too high, the actuator may suffer damage due to excessive pressure. Demand torque and cushion torque are defined below in terms of load, friction and acceleration torques.

M – Torque
The amount of turning effort exerted by the rotary actuator.

M_D – Demand Torque
The torque required from the actuator to do the job. It is the sum of the load torque, friction torque and acceleration torque, multiplied by an appropriate design safety factor to be determined by the designer.

$$M_D = (M_L + M_f + M_\alpha) \times \text{design safety factor}$$

M_L – Load Torque
The torque required to support the weight or force of the load. In example A, the load torque is 2943Nm; in example B, it is zero; in example C, it is 160Nm.

M_f – Friction Torque
The torque required to overcome any friction between moving parts, especially bearing surfaces. In example A, the friction torque is zero for the hanging load; in example B, it is 883Nm for the sliding load; in example C, it is zero for the clamp.

M_α – Acceleration Torque
The torque required to overcome the inertia of the load in order to provide a required acceleration or deceleration. In example A, the load is suspended motionless so there is no acceleration. In example B, the load is accelerated from rest to a specified angular velocity. If the rotational mass moment of inertia about the axis of rotation is J_m and the angular acceleration is α, then the acceleration torque is equal to $J_m\alpha$, in this example 800Nm. In example C, there is no acceleration.

Some values for rotational mass moment of inertia, and some useful equations for determining α, are listed on page 6.

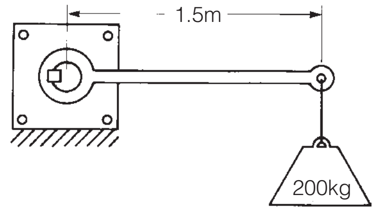
M_C – Cushion Torque
The torque that the actuator must apply to provide a required angular deceleration, α*. Cushion torque is generated by restricting the flow out of the actuator in order to create a back pressure which decelerates the load. Often, this back pressure (deceleration) must overcome both the inertia of the load and the driving pressure (system pressure) from the pump. If system pressure is maintained during cushioning, then cushion torque is the sum of the demand torque and deceleration torque, less the friction torque.

$$M_C = (M_D + M_{\alpha^*} - M_f) \times \text{design safety factor}$$

Warning – rapid deceleration can cause high pressure intensification at the actuator outlet. Always ensure that cushion pressure does not exceed the manufacturer's pressure rating for the actuator, and cross-check with the cushion energy absorption capacity data in the product catalogue.

Demand Torque Examples

Example A
Demand Torque due to Load Torque



$M_D = M_L + M_f + M_\alpha$

$M_L = 200\text{kg} \times 9.81\text{m/s}^2 \times 1.5\text{m} = 2943\text{Nm}$

$M_f = 0$

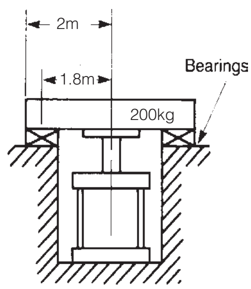
$M_\alpha = 0$

$M_D = M_L = 2943\text{Nm}$

Demand Torque $M_D = 2943\text{Nm}$

Example B
Demand Torque due to Friction and Acceleration Torque

A rotating circular platform with bearing friction. The platform is supported completely by the bearings – there is no load on the shaft.



Platform mass = 200kg
Bearing radius = 1.8m
Angular acceleration α = 2 rad/sec^2
Bearing coefficient of friction μ = 0.25
Rotational mass moment of inertia J_m = 400kgm^2

$M_D = M_L + M_f + M_\alpha$

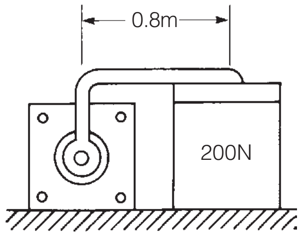
$M_L = 0$

$M_f = 0.25 \times 200\text{kg} \times 9.81\text{m/s}^2 \times 1.8\text{m} = 883\text{Nm}$

$M_\alpha = 400\text{kgm}^2 \times 2\ \text{rad/sec}^2 = 800\text{Nm}$

Demand Torque $M_D = M_f + M_\alpha = 883 + 800 = 1683\text{Nm}$

Example C
Demand Torque due to Load Torque
The clamp exerts a force of 200N at 0.8m.



$M_D = M_L + M_f + M_\alpha$

$M_L = 200\text{N} \times 0.8\text{m} = 160\text{Nm}$

$M_f = 0$

$M_\alpha = 0$

$M_D = M_L = 160\text{Nm}$

Demand Torque $M_D = 160\text{Nm}$

Basic Angle, Velocity and Acceleration Equations

When acceleration α is uniform:

$\theta = \omega_o t + \frac{1}{2}\alpha t^2$

$\theta = \omega_t t - \frac{1}{2}\alpha t^2$

$\omega_t = \omega_o + \alpha t$

$\omega_t = (\omega_o^2 + 2\alpha\theta)^{1/2}$

$\alpha = (\omega_t - \omega_o)/t$

$\alpha = \dfrac{(\omega_t^2 - \omega_o^2)}{2\theta}$

When velocity is constant: $\theta = \omega t$

Where:

t = time

θ = angle of rotation

ω_t = angular velocity at time = t

ω_o = angular velocity at time = o

J_P = rotational mass moment of inertia about an axis parallel to a centroidal axis

J_G = rotational mass moment of inertia about a centroidal axis

m = mass, centre of gravity G

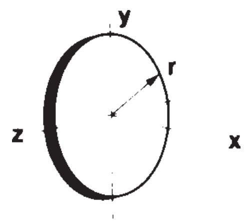
d = distance between axes

Rotational Mass Moments of Inertia

Thin Disk

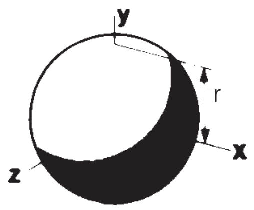
$J_X = \frac{1}{2}mr^2$

$J_y = J_Z = \frac{1}{4}mr^2$



Sphere

$J_X = J_Y = J_Z = \frac{2}{5}mr^2$

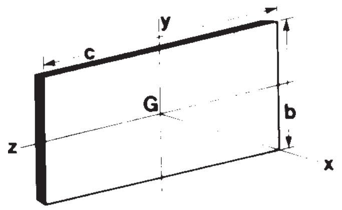


Thin Rectangular Plate

$J_X = \frac{1}{12}m(b^2 + c^2)$

$J_Y = \frac{1}{12}mc^2$

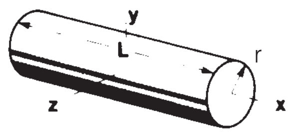
$J_Z = \frac{1}{12}mb^2$



Circular Cylinder

$J_X = \frac{1}{2}mr^2$

$J_Y = J_Z = \frac{1}{12}m(3r^2 + L^2)$

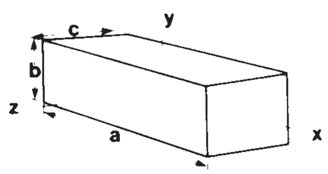


Rectangular Prism

$J_X = \frac{1}{12}m(b^2 + c^2)$

$J_Y = \frac{1}{12}m(c^2 + a^2)$

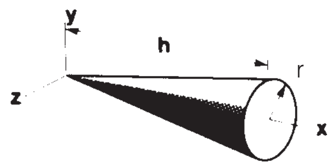
$J_Z = \frac{1}{12}m(a^2 + b^2)$



Circular Cone

$J_X = \frac{3}{10}mr^2$

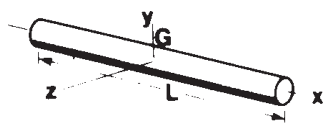
$J_Y = J_Z = \frac{3}{5}m(\frac{1}{4}r^2 + h^2)$



Slender Rod

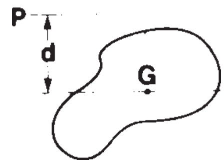
$J_y = J_Z = \frac{1}{12}mL^2$

Assume $J_x = 0$



Parallel Axis Theorem

$J_P = J_G + md^2$



Applied Torque Examples

The application examples and corresponding equations shown on the following pages are supplied to assist the design engineer when selecting a rotary actuator. They should be read in conjunction with the notes opposite.

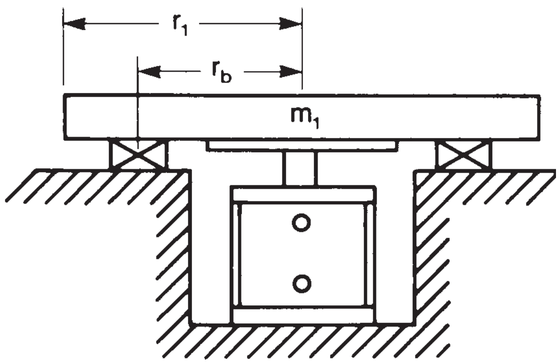
Notes

1 The examples and equations shown below and on the following pages are intended only as a guide. It is the responsibility of the design engineer to verify the accuracy of the equations and to ensure that all performance and safety requirements of the application are met.

2 Unless specified otherwise, the following examples do not take into account system and actuator efficiencies, or the effects of friction.

3 Deceleration torques are based on the assumption that, due to restrictor-type flow controls, the actuator is subject to relief valve pressure during deceleration.

1 Rotary Index Table – No Load



Coefficient of friction = μ
Acceleration = α
Deceleration = α^*
Load = 0
Subject to bearing friction

$M_D = M_L + M_f + M_\alpha$

$M_L = 0$, no load

$M_f = \mu m_1 g r_b$

$M_\alpha = J_1\alpha$

$J_1 = \frac{1}{2} m_1 r_1^2$

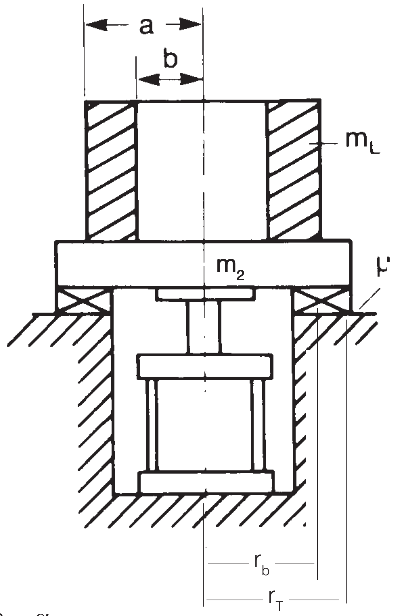
$M_c = M_D + M_{\alpha^*} - M_f$

$M_{\alpha^*} = J_1\alpha^*$

Demand Torque $M_D = \mu m_1 g r_b + \frac{1}{2} m_1 r_1^2 \alpha$

Cushion Torque $M_C = \frac{1}{2} m_1 r_1^2 \alpha + \frac{1}{2} m_1 r_1^2 \alpha^*$

2 Rotary Index Table with Co-axial Load



Acceleration = α
The index table rotates in a horizontal plane, with a cylindrical load.

$M_D = M_L + M_f + M_\alpha$

$M_L = 0$

$M_f = \mu(m_2 g + m_L g) r_b$

$M_\alpha = (J_T + J_L)\alpha$

$J_T = \frac{1}{2} m_2 r_T^2$

$J_L = \frac{1}{2} m_L(a^2 - b^2)$

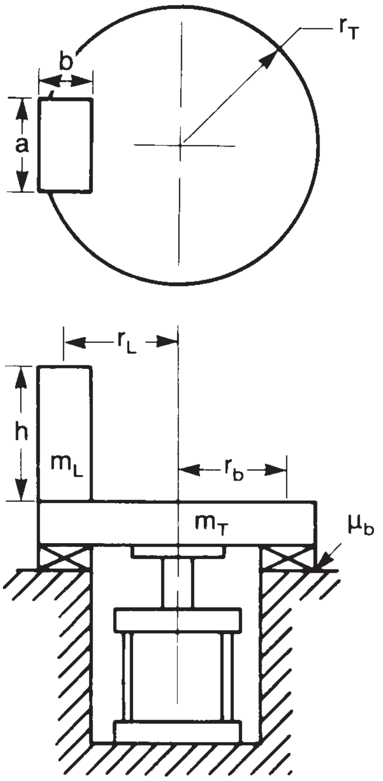
$M_C = M_D + M_{\alpha^*} - M_f$

$M_{\alpha^*} = (J_T + J_L)\alpha^*$

Demand Torque $M_D =$
$\mu(m_2 g + m_L g) r_b + \frac{\alpha}{2}[m_2 r_T^2 + m_L(a^2 - b^2)]$

Cushion Torque $M_C = \frac{1}{2}(\alpha + \alpha^*)[m_2 r_T^2 + m_L(a^2 - b^2)]$

3 Rotary Index Table with Offset Load



Acceleration = α
The index table rotates in a horizontal plane, with a rectangular load.

$M_D = M_L + M_f + M_\alpha$

$M_L = 0$

$M_f = \mu(m_T g + m_L g) r_b$

$M_\alpha = (J_T + J_L)\alpha$

$J_T = \frac{1}{2} m_T r_T^2$

$J_L = \frac{1}{12} m_L (a^2 + b^2) + m_L r_L^2$

$M_C = M_D + M_{\alpha^*} - M_f$

$M_{\alpha^*} = (J_T + J_L)\alpha^*$

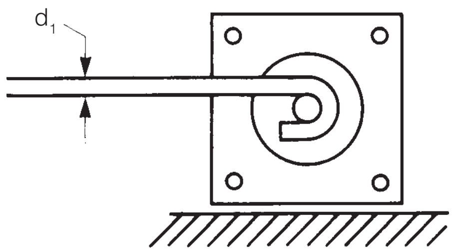
Demand Torque M_D =

$$\mu(m_T g + m_L g) r_b + \alpha[\tfrac{1}{2} m_T r_T^2 + \tfrac{1}{12} m_L (a^2 + b^2) + m_L r_L^2]$$

Cushion Torque M_C =

$$(\alpha + \alpha^*)\,[\tfrac{1}{2} m_T r_T^2 + \tfrac{1}{12} m_L (a^2 + b^2) + m_L r_L^2]$$

4 Wire or Round Tube Bending



Acceleration = α
Maximum yield stress for wire or tube material = σ_y
Section modulus of tubing* = L/C
Outside diameter of wire or tube = d_1
Inside diameter of tube = d_2

* This can be calculated or found in mechanical engineering handbooks.

$M_D = M_L + M_f + M_\alpha$

$M_L = (L/C)\sigma_y$

$M_f = 0$, no external bearings

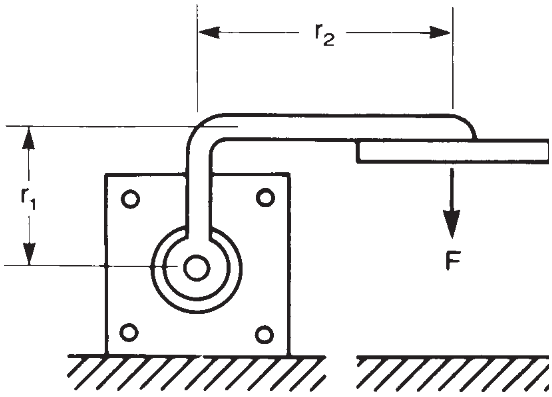
$M_\alpha = 0$, assumed to be zero

$M_D = M_L = (L/C)\sigma_y$

Round Tube –
Demand Torque $M_D = \frac{\pi}{32}\left[\frac{d_1^4 - d_2^4}{d_1}\right]\sigma_y$

Round Wire –
Demand Torque $M_D = \frac{\pi d_1^3 \sigma_y}{32}$

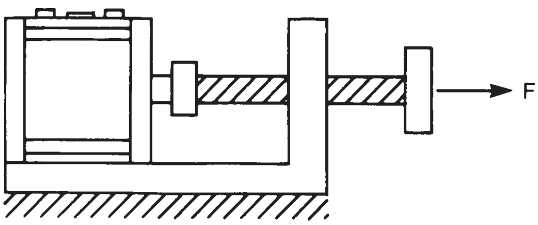
5 Simple Clamp



Acceleration = 0
Friction = 0

Demand Torque $M_D = Fr_2$

6 Screw Clamping

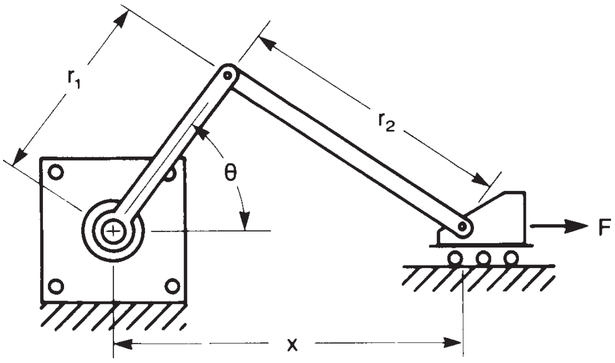


Acceleration = 0
Friction = 0
Pitch of thread = p
Clamping force = F

Note: this example does not take account of losses due to thread friction.

Demand Torque $M_D = \frac{Fp}{2\pi}$

7 Linear Motion, Clamping



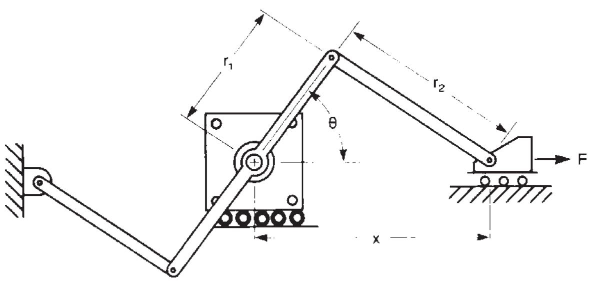
Notes

1 High clamping force is achieved as θ nears 0
2 θ should not be able to become 0
3 F should not exceed the bearing capacity of the actuator

$$\text{Demand Torque } M_D = \frac{Fr_1}{2}\left[\frac{\sin 2\theta}{x - r_1\cos\theta} + 2\sin\theta\right]$$

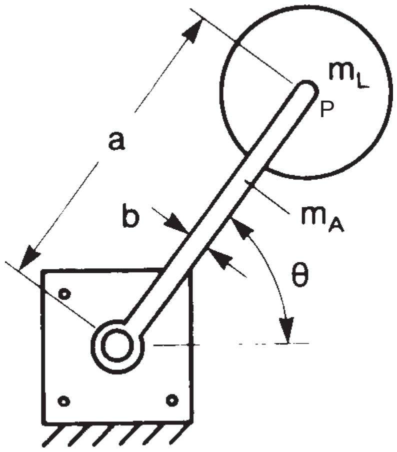
$$\text{Where } x = r_1\cos\theta + \sqrt{r_2^{\,2} - r_1^{\,2}\sin^2\theta}$$

8 Modified Linear Motion, Clamping



This application is similar to that shown at 7 above, except that the bearing strength requirement is reduced.

9 Overcentre Load



The load is rotated through a vertical plane. Load torque M_L is positive or negative, depending on position and direction of rotation.

$$M_D = M_L + M_f + M_\alpha$$

$$\pm M_L = (m_L g + \tfrac{1}{2} m_A g)a \cos\theta$$

$$M_f = 0, \text{ no external bearings}$$

$$M_\alpha = [\tfrac{1}{12} m_A(a^2 + b^2) + \frac{m_A a^2}{4} + m_L a^2]\,\alpha$$

Demand Torque M_{Dmax} =

$$(m_L g + \tfrac{1}{2} m_A g)a + [\tfrac{1}{12} m_A(a^2 + b^2) + \frac{m_A a^2}{4} + m_L a^2]\,\alpha$$

Cushion Torque $M_C = M_{Dmax} + M_{\alpha^*} - M_F$

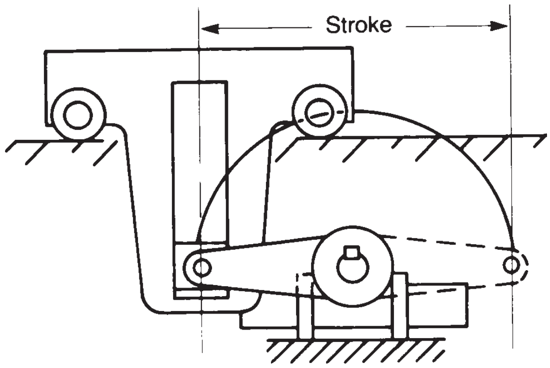
where $M_{\alpha^*} = [\tfrac{1}{12} m_A(a^2 + b^2) + \frac{m_A a^2}{4} + m_L a^2]\alpha^*$

Note: If the mass m_L is not free to turn about point P, then its rotational mass moment of inertia about its own centroid J_L must be added to the equations for M_α and M_{α^*} as follows.

$$M_\alpha = [\tfrac{1}{12} m_A(a^2 + b^2) + \frac{m_A a^2}{4} + J_L + m_L a^2]\alpha$$

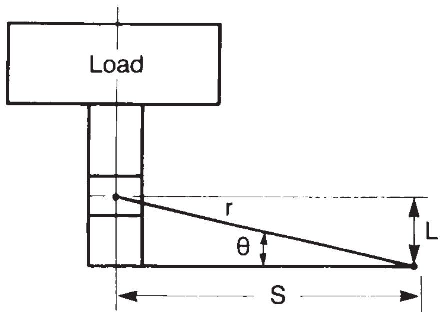
$$M_{\alpha^*} = [\tfrac{1}{12} m_A(a^2 + b^2) + \frac{m_A a^2}{4} + J_L + m_L a^2]\alpha^*$$

10 Harmonic Drive



The harmonic motion linkage shown is a compact, low cost method of producing linear motion with smooth acceleration and deceleration. This property permits the cycle times of applications such as transfer lines to be optimized.

Flow control valves can be adjusted to regulate the smooth acceleration and deceleration necessary to handle fragile parts such as bottles or light bulbs.



M = actuator torque
r = torque arm length
$L = r \sin\theta$
$S = r \cos\theta$
m = mass of load
$g = 9.81 m/sec^2$
f = friction coefficient of load on slideway
f_s = friction coefficient of peg in slide

By using the geometry of the linkage and including the friction of both the load and the slide, equation A on page 11 can be used to determine the required torque. The curves on page 11 give solutions for equation A for various arm lengths, with friction coefficients of 0.05 and 0.25.

Equation A assumes a constant actuator velocity; as the inertia from the moving load tends to drive the actuator during the deceleration phase, it is recommended that the circuit should include pressure compensated flow control valves. It is also recommended that, due to the resistance of the flow control valves plus the load deceleration requirements, figures obtained from the graphs should be doubled to obtain the maximum torque requirement.

Note that, for high friction applications, equation B may be used, with data from the f = 0.05 graph on page 11.

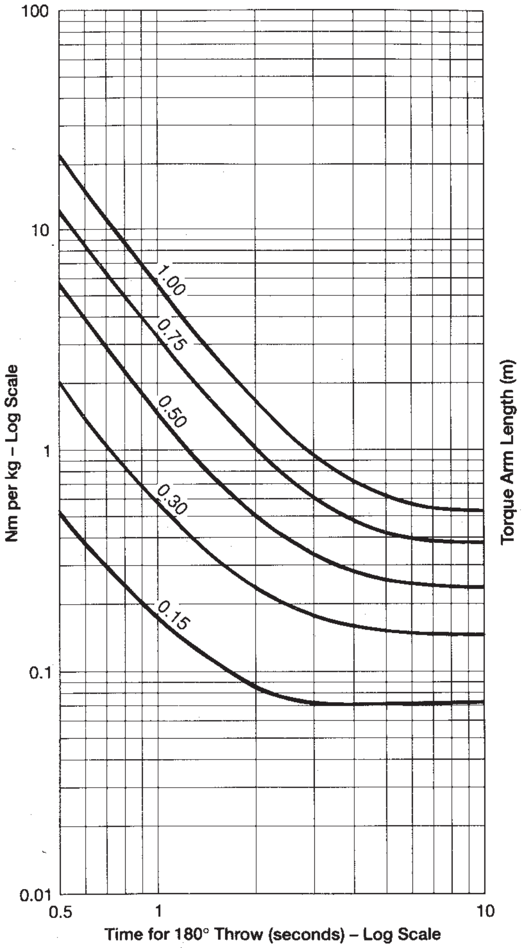
10 Harmonic Drive Cont'd.

Equation A: $\frac{M}{m} = \frac{r}{1+ff_s}(fg\sin\omega t + ff_s g\cos\omega t + r\omega^2\cos\omega t\sin\omega t + f_s r\omega^2\cos^2\omega t)$

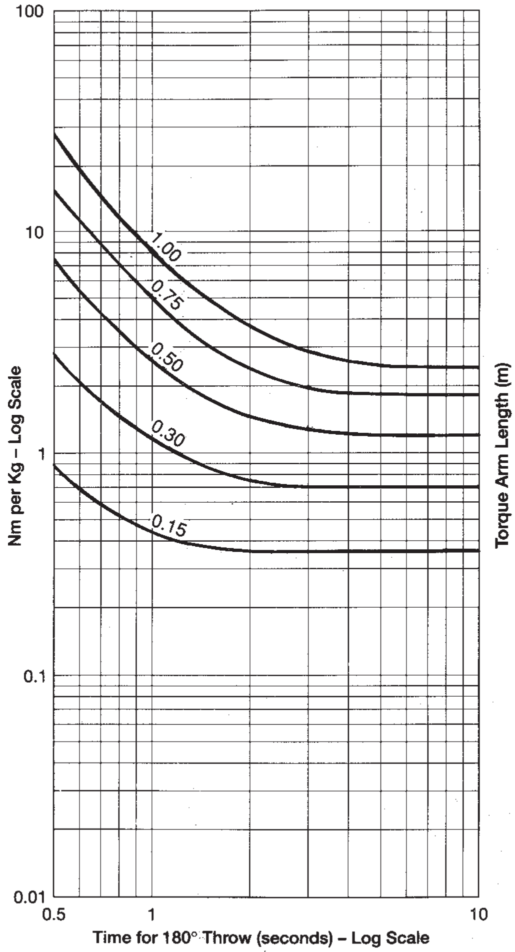
Equation B: M/m (deceleration) = 2M/m (acceleration) - M/m (at 10 second throw time)

Torque Required per Kilogram Moved

Coefficient of Friction = 0.05



Coefficient of Friction = 0.25



Example

A 200kg mass is to be moved over a distance of 1.5 metres in four seconds. Assume that the load is supported on roller bearings with a coefficient of friction of 0.05.

Solution

Torque arm length = 1.5m travel/2 = 0.75m

Using the curves for the 0.05 coefficient of friction, find the point of intersection between the four second line on the x-axis and the curve representing the 0.75 metre torque arm length. The torque required per kilogram moved can then be read off from the y-axis.

Demand Torque M_D = 200 x 0.48 = 96Nm

High Friction Applications

For a high friction application, this becomes:

$M_D = 2M_\alpha - M_{10}$

= (2 x 96)Nm - (0.38 x 200)Nm

= (192 - 76)Nm

Demand Torque M_D = 116Nm

Determining Rotary Actuator Size

Torque is generated by applying a force at a distance from a pivot. In a rotary actuator the force is produced by applying pressure to the piston and the distance is produced by the rack and pinion gears. Duty cycle, torque and pressure are the determining factors when sizing a rotary actuator. The rotary actuator must be able to exceed the duty cycle, deliver sufficient torque to move the load and also to withstand the pressure required to stop it. It should be noted that the back pressure, generated during cushioning, is often greater than system pressure.
A method for determining the smallest rotary actuator, for a given application, is described below.

1. Determine the duty cycle of the actuator.

2. Determine the maximum allowable safe System Pressure, P, that all the system components can withstand. This is not the actual working pressure, which is only determined after an actuator is selected.

3. Calculate the Demand Torque, M_D, required using the following equation (see Page 5).

   $M_D = (M_L + M_f + M_\alpha)$ x design safety factor

4. Calculate the Cushion Torque, M_C, required using the following equation (see Page 5).

   $M_C = (M_D + M_{\alpha^*} - M_f)$ x design safety factor

5. Taking account of the duty cycle, pressure and torque requirement of the application, select an actuator that is larger than your requirements.

6. Calculate the System Operating Pressure based on the Demand Torque, M_D, and the torque/pressure data for the selected actuator. The relief valve must be set at:
   – less than the maximum safe system pressure established in Step 2 above;
   – no greater than the actuators rated working pressure;
   – a high enough value to compensate for pressure drop through lines and valves.

7. Check the flow rate required by the rotary actuator using the calculations on page 13. If the flow rate associated with this is impractical for the application, M_D or M_C can be reduced by:
   – reducing the rotational mass moment of inertia of the load and repeating steps 3-5;
   – increasing the time for acceleration and repeating steps 3-5;
   – increasing the deceleration time and repeating steps 4-5;
   – using an external shock absorber.

   Note: Fluid velocity should be less than 5m/s to reduce cavitation and turbulence in fittings.

8. Check the cushion capacity of the selected actuator against the energy of the application.

Caution

The biggest single cause of rotary actuator failure is the introduction of shock and surge pressures beyond the maximum rated working pressure of the unit.
This failure is most common in actuator systems with any or all of the following criteria:
– Rotational speeds in excess of 10 RPM
– Control of a large mass in the horizontal plane
– Control of a mass moving over centre
– Operation of a long lever arm.

If in doubt copy, complete and forward/fax the Applications Data Check List on page 19 to your Parker Representative.

Calculating Required Pump Flow

The flow rate required for a rotary actuator can be determined from the desired time for rotation and the rotary actuator's displacement, using the equation:

$$Q = V/t$$

Where:

Q = flow rate
V = rotary actuator displacement
t = time to fill displacement

This equation is reproduced in graphical form on page 14, allowing pump flow to be read off from the x-axis.

V_s = specific volume, cm^3/radian

Example 1

A 360° rotary actuator is selected to provide a 340° rotation in 6 seconds. If the rotary actuator's displacement is 1164cm^3, find what flow rate is required from the pump.

Solution

The actuator is only rotating through 340°, so the volume of fluid required for this rotation is:

$$V = 1164 \times 340/360$$

$$= 1099cm^3 \text{ for } 340° \text{ of rotation}$$

$$Q = \frac{V}{t}$$

$$= \frac{1099}{6} \times \frac{60 \text{ seconds}}{1000cm^3}$$

$$= 10.99 \text{ litres per minute}$$

Example 2

A 180° rack and pinion rotary actuator is to accelerate from 0 to a given angular velocity ω during its first 10° of rotation, remain at that angular velocity for the next 150° of rotation, then decelerate back to 0 radians/second during the last 20°. The actuator is to rotate through the total 180° in less than 2 seconds. If the actuator's displacement is 582cm^3, find:

A The angular velocity ω after the first 10° of rotation

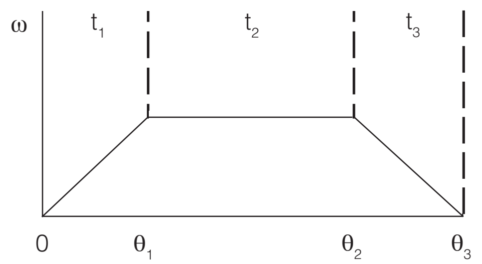
B The pump flow rate required for the rotary actuator

C The pump flow required if the actuator travelled the full 180° in 2 seconds at a constant angular velocity

Solution A

Assume constant acceleration during the first 10° and constant deceleration during the last 20°.

$$2 \text{ seconds} = t_1 + t_2 + t_3$$



$$t_1 = 2\,\frac{\theta_1 - 0}{\omega} = \frac{2\,(10°)}{\omega}\,\frac{\pi}{180°} = \frac{(0.35)}{\omega}$$

$$t_2 = \frac{\theta_2 - \theta_1}{\omega} = \frac{(150°)}{\omega}\,\frac{\pi}{180°} = \frac{(2.62)}{\omega}$$

$$t_3 = 2\,\frac{\theta_3 - \theta_2}{\omega} = \frac{2\,(20°)}{\omega}\,\frac{\pi}{180°} = \frac{(0.70)}{\omega}$$

$$2 \text{ seconds} = \frac{1}{\omega}\,[0.35 + 2.62 + 0.70]$$

$$\omega = 1.83 \text{ radians/second}$$

Solution B

The 180° rotary actuator has a volume displacement of 582cm^3. The cm^3/radian can be expressed as:

$$180° = \pi \text{ radians}$$

$$V_S = V/\theta = 582/\pi \text{ radians}$$

$$V_S = 185.2cm^3/\text{radian}$$

The actuator must be able to rotate at 1.83 radians/second, so the pump flow must be:

$$Q = V_S\omega$$

$$= \frac{(185.2cm^3/\text{radian})\,(1.83 \text{ radians/second}) \times 60 \text{ seconds}}{1000cm^3 \text{ per litre}}$$

$$= 20.3 \text{ litres per minute}$$

Solution C

If the entire 180° were traversed at constant speed in 2 seconds, the pump flow would be:

$$Q = \frac{V}{t}$$

$$= \frac{582}{2} \times \frac{60 \text{ seconds}}{1000cm^3 \text{ per litre}}$$

$$= 17.46 \text{ litres per minute}$$

Note: In this example, it is necessary to take into account the time required for acceleration and deceleration of the actuator, in order to determine the maximum velocity required. It is this maximum velocity of the actuator which will determine the maximum flow required. Equations for velocity and acceleration are shown on page 6.

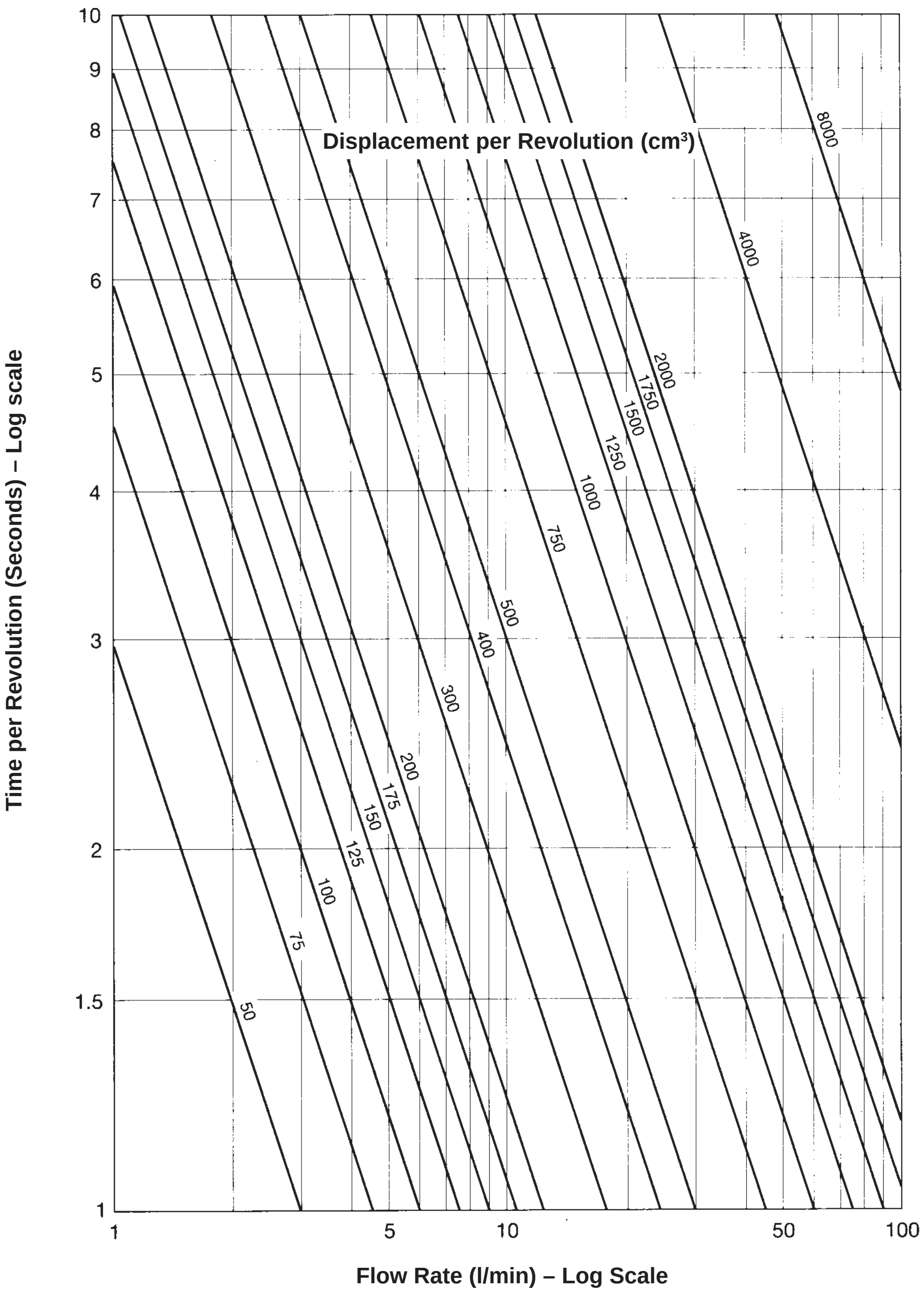
Flow Rate

The flow rate is determined from the equation:

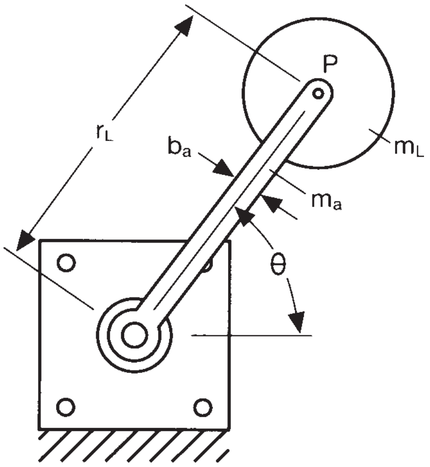
$$t = 0.06 \frac{V}{Q}$$

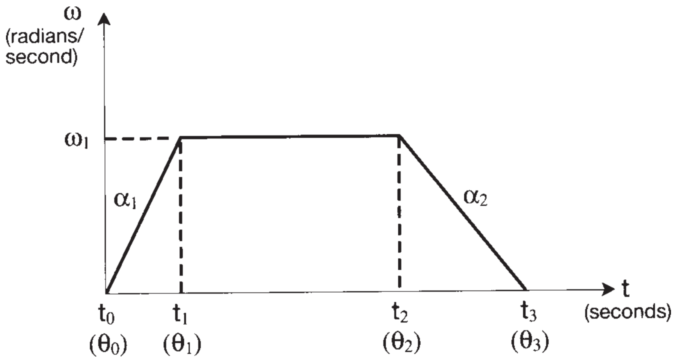
Where: Q = flow rate in litres/minute
V = rotary actuator displacement in cm^3/revolution
t = time in seconds
efficiency per revolution is assumed to be 100%

This equation may be expressed in graphical form, as shown.



Example





Paper roll, 70 kg, pivoted at P.
Lever arm, 6kg (1.5m long, 0.3m wide).

m_L	=	70kg
m_a	=	6kg
r_L	=	1.5m
b_a	=	0.3m
Rotation	=	180°
Time, t	=	4 seconds
Pressure, p	=	50 bar

20° cushions fitted (0.349 radians)

Duty cycle: 21 seconds between rotations, 8 hours per day, five days per week, 50 weeks per year, 10 year actuator life.

θ_0 = 0°	= 0 rad	t_0 = 0 sec	ω_0 = 0 rad/sec
θ_1 = ?	= ? rad	t_1 = ? sec	ω_1 = ? rad/sec
θ_2 = 160°	= 2.793 rad	t_2 = ? sec	ω_2 = ? rad/sec
θ_3 = 180°	= 3.142 rad	t_3 = 4 sec	ω_3 = 0 rad/sec

Assume $\theta_1 = 10° = 0.175$ rad
For this application $\omega_1 = \omega_2$

$$\omega_1 = \frac{2(\theta_1 - \theta_0) + (\theta_2 - \theta_1) + 2(\theta_3 - \theta_2)}{(t_3 - t_0)}$$

$$= \frac{(2 \times 0.175) + (2.793 - 0.175) + (2 \times 0.349)}{4}$$

$$= \frac{3.67}{4} = 0.917 \text{ rad/sec}$$

$$\alpha_1 = \frac{\omega_1 - \omega_0}{t_1 - t_0} = \frac{\omega_1^2 - \omega_0^2}{2(\theta_1 - \theta_0)}$$

$$= \frac{0.917^2 - 0^2}{2 \times 0.175}$$

$$= \frac{0.841}{0.350} = 2.4 \text{ rad/sec}^2$$

$$\alpha_2 = \frac{\omega_3^2 - \omega_2^2}{2(\theta_3 - \theta_2)} = \frac{0^2 - 0.917^2}{2 \times 0.349} = -1.2 \text{ rad/sec}^2$$

$$\alpha^* = -\alpha_2 = 1.2 \text{ rad/sec}^2$$

1. Duty Cycle

$$\left[\frac{8 \times 60 \times 60}{(21 + 4)}\right] \times 5 \times 50 \times 10 = 2.88 \times 10^6 \text{ cycles}$$

Therefore **continuous** duty

2. System pressure, p = **50 bar**

continued overleaf ...

Example Cont'd.

3. Demand torque $M_D = M_L + M_f + M_\alpha$

$$M_L = \left[M_L g r_L + m_a g \frac{r_L}{2}\right]\cos\theta$$

M_{Lmax} at $\cos\theta = 1 \rightarrow \theta = 0°$

$$M_{Lmax} = M_L g r_L + m_a g \frac{r_L}{2}$$

$$= (70 \times 9.81 \times 1.5) + (6 \times 9.81 \times \frac{1.5}{2}) = 1074 \text{ Nm}$$

M_f = 0 Nm, no external friction

$M_\alpha = J_m \alpha$

$$J_m = \frac{1}{12} M_a(r_a{}^2 + b_a{}^2) + M_a \frac{r_L{}^2}{4} + M_L r_L{}^2$$

$$= \left[\frac{1}{12} \times 6 \times (1.5^2 + 0.3^2)\right] + \left[6 \times \frac{1.5^2}{4}\right] + (70 \times 1.5^2)$$

$$= 1.17 + 3.375 + 157.5 = 162 \text{ kgm}^2$$

$$M_\alpha = 162 \times 2.4 = 390 \text{ Nm}$$

M_D = 1074 + 0 + 390 = **1464 Nm**

4. Cushion torque $M_C = M_D + M_{\alpha^*} - M_f$

M_D = 1464 Nm

$M_{\alpha^*} = J_m \alpha^*$

J_m = 162 kgm^2

M_{α^*} = 162 x 1.2 = 194 Nm

M_f = 0, no external friction

M_C = 1464 + 194 - 0 = **1658 Nm**

Assuming a design safety factor of 1.0 for this example, the actuator must be capable of generating this torque at p = 50 bar.

5. Actuator selection for continuous duty, 1658 Nm at 50 bar

HTR75 produces 4500 Nm at 110 bar
(2045 Nm at 50 bar)

6. System operating pressure, p $= \frac{1658}{4500} \times 110$

= **40.5 bar**

7. Specific volume cm^3/rad for HTR 75
= 480 cm^3/rad
= 3016 cm^3/rotation

ω_1 = 0.917 rad/sec = 6.85 secs/revolution

assume peak velocity = 2 x ω_1
= 2 x 0.917
= 1.834 rad/sec = 3.43 sec/rev.

Therefore flow rate, Q $= 480 \times 1.834 \times \frac{60}{1000}$

= **52.8 l/min**

[from 4 seconds per revolution, 3000 cm^3/revolution and the graph on page 14 approx. 55 l/min]

8. Cushion capacity

Assuming downward motion of mass:

$$E = \tfrac{1}{2} J_m \omega^2 + mgR\theta$$

$$= (\tfrac{1}{2} \times 162 \times 0.917^2) + (162 \times 9.81 \times 1 \times 0.349)$$

= 68 + 555 = **623 J**

At 50 bar the HTR75 actuator can absorb 700 Joules of energy.

Therefore the HTR75 actuator can cope with this application.

Circuit Recommendations

Composite Operating Circuit

When designing hydraulic operating circuits for rotary actuators, consideration should be given to the following criteria:

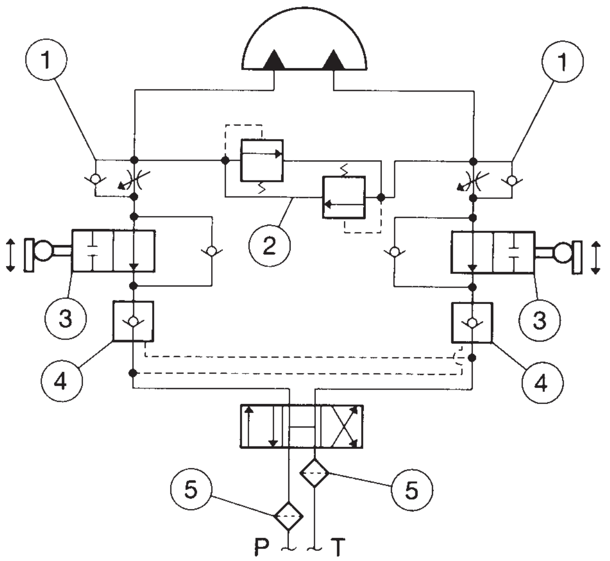
- actuator rotational velocity
- kinetic energy developed
- actuator holding requirements
- system filtration

The composite drawing below shows general recommendations for sample circuitry, and is intended as a guide only. Flow control valves (1) in the meter-out position provide controlled actuator velocity. Care should be taken if the load is to move overcentre, as the combination of load and pump-generated pressure may exceed the actuator rating.

To protect the actuator and other system components from shock pressures caused when the actuator is suddenly stopped in mid-stroke, cross-over relief valves (2) should be installed, as close to the actuator as possible. These relief valves also protect the actuator and system if the load increases and 'back-drives' the hydraulic system.

In applications involving high speeds or heavy loads, the built-up kinetic energy may be too much for cushions to absorb during their 20° of operation. By using cam or lever-operated deceleration valves (3), the deceleration arc can be increased beyond 20° so that the kinetic energy can be absorbed more gradually and without over-pressurizing the actuator. Where there is a need to hold the load in intermediate positions for extended periods of time, pilot operated check valves (4) should be used. These must be fitted with leakproof actuator seals to hold the load in position, as any bypass flow will allow drifting of the load.

Note: For safety reasons, some applications require a mechanical locking device for holding loads over an extended period of time.



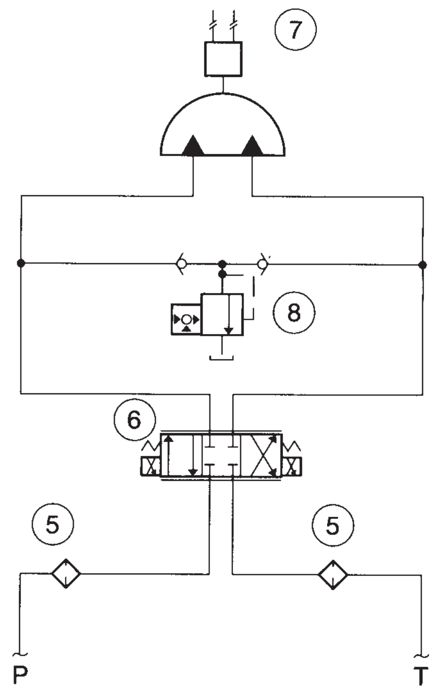
As with most standard hydraulic circuits, rotary actuator applications should have filtration to provide a continuous cleanliness rating in accordance with the ISO 17/14 fluid classification. Filters (5) should be fitted and maintained to ensure this minimum level.

Electro-hydraulic Circuitry

The use of electro-hydraulic components for rotary actuator applications can provide greater system flexibility. The diagram below is a representative circuit showing some possible applications of electro-hydraulic valves. Proportional or servo-control valves (6) can provide precise position, velocity or acceleration control of loads, and 'closing the loop' around a feedback device (7) can provide even greater control and velocity profiles for overcentre or varying loads. Anti-backlash devices allow a high degree of positional control to be obtained from rack and pinion units.

Torque control can also be achieved, by using proportional pressure control valves (8) to vary the 'stall torque' of an actuator, or to provide a torque profile for various machine processes.

All those considerations examined under 'Composite Operating Circuits' above are relevant to electro-hydraulic applications. Cross-over relief valves should be installed if there are to be sudden stops in mid-stroke, and caution should be exercised when running overcentre with high speeds or high inertia loads. Note that the filtration requirements of electro-hydraulic systems are generally higher than those of the actuator components.



Model Selection

The maximum working pressures for HTR, LTR and PTR rotary actuators are 210 bar, 70 bar and 18 bar respectively. Torque outputs at maximum working pressure for all models in each of the three series are shown below.

HTR Series

Torque outputs of HTR Series rotary actuators at 210 bar static duty are as follows:

Single Rack Units

HTR.9	100Nm
HTR3.7	420Nm
HTR5	560Nm
HTR15	1700Nm
HTR22	1700Nm (@ 140 bar max. rating)
HTR75	8500Nm
HTR300	34000Nm

Double Rack Units

HTR1.8	200Nm
HTR7.5	850Nm
HTR10	1130Nm
HTR30	3400Nm
HTR45	3400Nm (@ 140 bar max. rating)
HTR150	17000Nm
HTR600	68000Nm

LTR Series

Torque outputs of LTR Series rotary actuators at 70 bar static duty are as follows:

Single Rack Units

LTR101	40Nm
LTR151	120Nm
LTR201	290Nm
LTR251	440Nm
LTR321	1175Nm

Double Rack Units

LTR102	60Nm (@ 50 bar max. rating)
LTR152	240Nm
LTR202	580Nm
LTR252	880Nm
LTR322	2350Nm

PTR Series

Torque outputs of PTR Series rotary actuators at 18 bar static duty are as follows:

Single Rack Units

PTR101	9Nm
PTR151	29Nm
PTR201	68Nm
PTR251	105Nm
PTR321	280Nm

Double Rack Units

PTR102	18Nm
PTR152	58Nm
PTR202	136Nm
PTR252	210Nm
PTR322	560Nm

Model Numbers and Ordering

Each Parker rack and pinion rotary actuator is assigned a model number consisting of a set of characters. To develop a model number, refer to the 'How to Order' page of the appropriate actuator catalogue and select those characters which represent the features that you require. The Applications Data Check List on page 19 of this Guide provides a framework when establishing a rotary actuator specification and, where further information is necessary, it can be photocopied, completed and faxed to the address shown at the foot of the page.

Maintenance and Spare Parts

Full instructions for the maintenance of Parker's rotary actuators, together with a complete list of the spares available, are contained in the Rotary Actuator Maintenance Bulletins. Please contact your nearest Parker Sales Office, whose address can be found on the rear cover of this guide, for further details.

Parker Hannifin plc
Cylinder Division
Watford, Herts.

Applications Data Check List

(To be completed when requesting further information)

Rotary Actuators

Parker Ref.

Contact Information

Name .... Job Title ....

Company..

Address..

.... Post Code ....

Telephone .... Fax ....

E-mail Address..

Application Details

1 Operation – Hydraulic / Pneumatic ....

2 Operating Pressure – bar ....

3 Operating Temperature – °C ....

4 Design Torque – Nm ....

5 Required Rotation – ° ....

6 Cycle Time – sec. ....

7 Working Life – cycles ....

8 External Bearing Load – kN ....

9 Operating Environment ....

10 Maximum Loading

a Maximum Mass – kg ....

b Rotational Mass Moment of Inertia – kgm^2 ....

c Maximum Rotational Speed – rad/sec

Clock/Anticlockwise ....

d Maximum Rotational Acceleration – rad/sec^2

Clock/Anticlockwise ....

11 Brief Description of Application
(Please supply a sketch if necessary)

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Actuator Details

12 Mounting Style ....

....

13 Type of Shaft ....

....

14 Port Type and Location ....

15 Seals ....

16 Cushioning ....

17 Stroke Adjusters ....

18 Proximity/Feedback Devices ....

....

19 Special Requirements ....

....
....
....

Please photocopy, complete and forward/fax to:
The Product Manager, Rotary Actuators
Parker Hannifin plc.
Greycaine Road, Watford, Herts. WD2 4QA, UK
Tel. 01923 492000 Fax: 01923 210562

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