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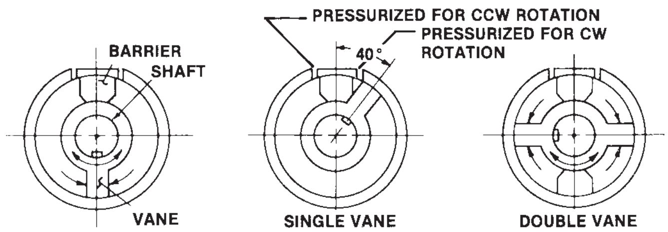
THE BASICS

Rotary actuators convert fluid pressure into rotary power, and develop instant torque in either direction. Basic construction consists of an enclosed cylindrical chamber containing a stationary barrier and a central shaft with vane(s) affixed. Fluid pressure applied to either side of the vane will cause the shaft to rotate.



The output torque developed is determined by the area of the vane, the number of vanes, and the fluid pressure applied. Speed of rotation is dependent on the flow and pressure capacities of the hydraulic system. The majority of actuators are constructed with one or two vanes, but are available with three or more for special applications. The theoretical torque output of a multivane unit is greater by a factor equal to the number of vanes times the torque of a single vane unit at equal pressure. The maximum arc of rotation for any actuator depends on the size and construction of the unit, and will always be less than the number of vanes divided into 360° because of the space occupied by the internal barrier(s). The arc of a single vane is approximately 280°, a double vane 100° and a triple vane 50°.

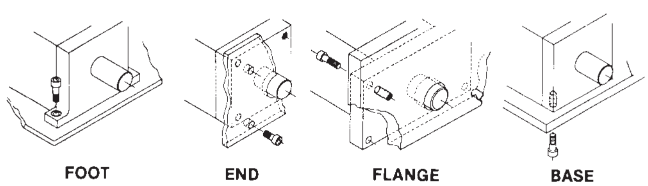
VERSATILITY

Fluid Media — Actuators can be operated on either pneumatic or hydraulic pressure. The fluid can be air, oil, high water base fluid (HWBF), or fire resistant fluid. Actuators can be assembled with special seals and/or internally plated for specific fluids.

Mounting — Actuators can be mounted horizontally, vertically or any angle in between. Models are available with flange, end, base or foot mounting provisions.

Actuators are usually mounted in a stationary position with the shaft rotating, but also can be shaft mounted with the housing portion rotating. Some models require mounting dowels to resist torsional forces. See the specific actuator model for mounting details.

TYPICAL MOUNTINGS



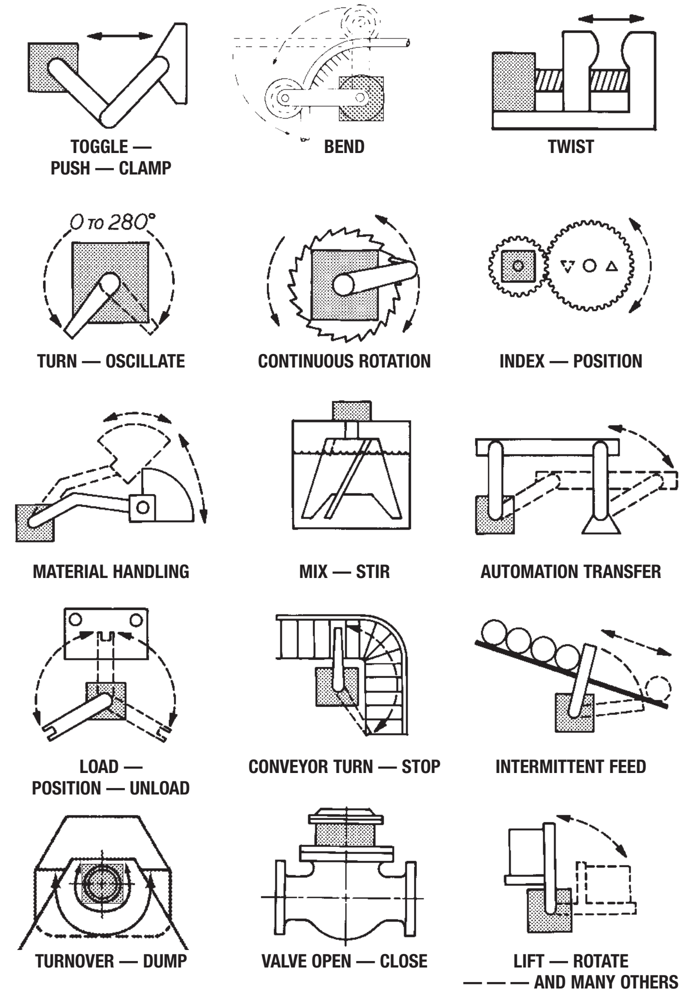
Control — Stopping, starting, acceleration and deceleration of actuators can be controlled by various types of valves in the fluid circuit.

External stops are recommended for most applications, although the arc of oscillation can be controlled by valves or positive internal stops (for light duty applications only).

In most cases special manifolds can be designed to mount servo-valves to the actuators allowing sophisticated control of all functions.

POSSIBLE APPLICATIONS

Rotary actuators are adaptable to a wide variety of uses in many different industries. The sketches shown give only an idea of the various possibilities. Actuators can perform a wide range of operations involving rotary or linear motion.



FACTORS TO CONSIDER WHEN APPLYING ACTUATORS

Service

Light Load — Heavy Load — consider weight of load and distance from actuator shaft.

Bearing Loads — heavy radial loads without external bearing support.

Shock Loads — consider dropped loads or mechanical failure of associated equipment. Also start - stop - jog and other non mechanical contact, hydraulic shock loads.

Rate of Oscillation — time to move load thru required angle. Also consider small angle - high rate applications.

Cycle Frequency — how often actuator is cycled. One cycle per minute, one cycle per week, etc.

External Stops — external stops should be used to limit angular travel as the actuator abutments (shoes) are not designed as mechanical stops.

Operating Press — should not exceed rated pressure of actuator.

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Environmental

Temperature — Hot example – foundry applications.
Cold example – cryogenic equip, outdoor equip.
Dirt — Examples, foundries, construction equipment
Caustic — Examples, valve operators, mixers plating tanks
Humidity — marine applications, outdoor
Vibration — machine tools, test equipment
Radiation — nuclear energy plants
Electricity — welding equipment
Clean — food processing, medical equipment

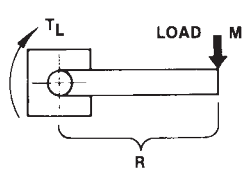
Maintenance

Lubrication — consult factory
Filter Maintenance — especially foundry and construction type applications
Shaft Alignment — close tolerance alignment or flexible couplings
Proper Mounting — rigid support, tight bolts, good coupling fits
Long Term Storage — fill with compatible oil
External Stops — tightness and proper location
Fluid Media Conditioning — water separators, lubricators, oil coolers
Fittings and Hoses — tightness and general condition
Protective Shielding — for high temperature or excessively dirty applications

GENERAL ENGINEERING NOTES

Selection of the proper sized actuator for an application is accomplished by determining the necessary torque to move the load at the required speed, the available fluid pressure and the necessary arc of rotation. Good design practice dictates a nominal over – capacity be designed into the load moving system.

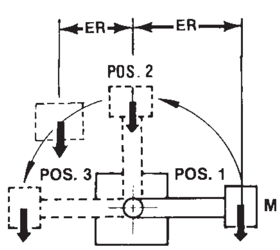
Load torque, T_L (inch pounds) is the resistance to movement of the shaft due to a load force or mass, M, (pounds) acting at a distance, R, (inches) from the center of the shaft rotation. $T_L = MR$.



Motion will occur when the applied torque of the actuator exceeds the load torque. The velocity and acceleration, A, given to the load mass, M, is proportional to the excess torque or force, F.

$$A=\frac{F}{M} \quad \text{or} \quad F=MA$$

Similarly, the load mass once set in motion must be stopped or decelerated with an opposing force F=MA. This deceleration force can be obtained by gradually restricting the flow of fluid to and from the actuator.



Caution:

Actuator should be protected from over pressurization during deceleration.

Lifting a mass in an arc causes the effective radius ER, to vary with the rotational position, becoming minimum at the vertical (90°) position. The load torque due to load force thus decreases from maximum at position 1 to minimum at position 2, and then reverses to aid rotation from position 2 to position 3. Restrictions of fluid flow and control of deceleration pressures is vitally necessary in this type of application.

Calculation of the amount and rate of energy dissipation required to stop a moving mass is possible if the variables such as velocity, mass, time, pressure, viscosity, etc., can be determined. In actual circuits these factors are inter-related and solution is often complex.

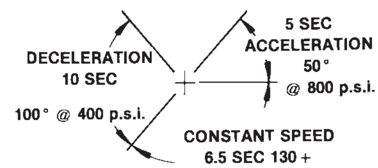
Good general practice requires that more cycle time be allowed for deceleration than for acceleration of a given mass.

A simplified calculation can be made if the assumption is made that the acceleration and deceleration are constant and uniform. The energy required to accelerate the mass must be equal to the energy to decelerate the mass. This simplifies to the following formulas:

Pressure (PSI accel) times Rotation (Degrees accel)= Pressure (PSI decel) times Rotation (Degrees decel)	OR	Pressure (PSI accel) times Time accel= Pressure (PSI decel) times Time decel

Example:

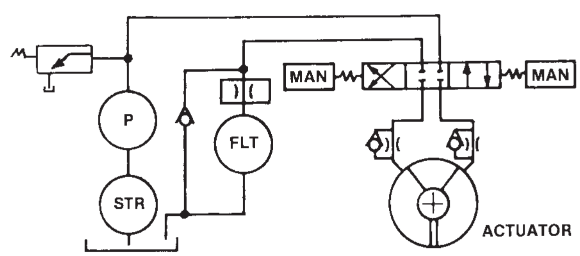
A mass accelerated uniformly for 50° @ 800 psi moves at constant velocity through use of flow-control valves until decelerated in the last 100° in 10 seconds @ 400 psi.



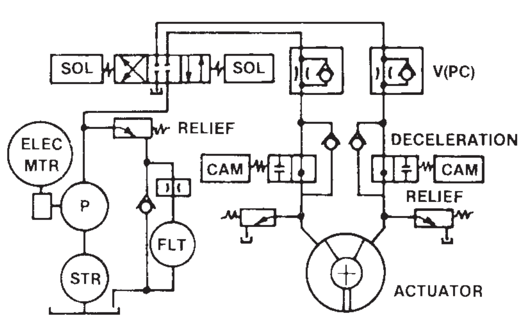
Note, however, that if the driving pressure were not removed during the deceleration period, the total deceleration pressure would be the sum of pressures, and at 1,200 psi could exceed the rating of the unit.

Actuator distributors can provide valuable assistance in solving specific circuit and application problems.

Direction and speed control for **slow speed and light loading** applications can be accomplished with relatively simple fluid circuits using hand- operated 4-way valves.



High speed and/or rapid cycling operation would suggest a commercially available solenoid-operated 4-way directional control valve and flow-control valves for better control of cycle motions, and the addition of fluid cooler, accumulators, and other components directed to specific system requirements.



Severe shock and possible damage to the system can occur on hydraulic applications by sudden or complete restriction of outgoing fluid, which allows the moving mass to generate high surge or transient shock wave pressures which must not exceed the rating of the unit.

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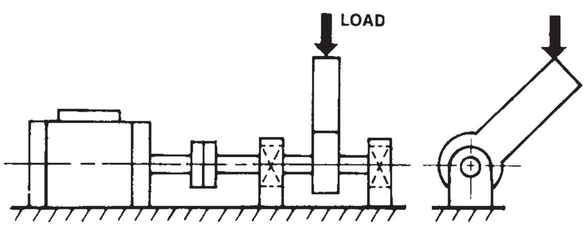
Deceleration valves, actuated by cams or by limit switches, are often used to gradually restrict the fluid and stop the moving mass. Usually, relief valves plumbed as shown, or plumbed from one line to the other in each direction, will limit the generation of surge pressures to a safe value. Cross-port relief manifolds are available for most actuators. If cam valves are used, the cam shape should provide a gentle ramp transition, and the spool should be tapered to provide a gradual closing off of fluid. **As a general rule, external stops, mounted securely to the machine framework, should be used to stop the load. The shaft vanes should not contact the internal stops except under very light loads.**

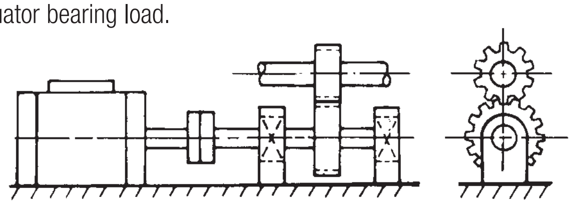
Air bleeding in hydraulic systems is usually not required if actuator is mounted with supply ports upward. In other positions, air will gradually dissolve in the oil and be carried away as the actuator is cycled. Special bleed connections are available as an optional feature on some actuators if specified when ordering.

Internal by-pass flow is always present to a small degree, and increases with increase of pressure. On air applications it must be recognized that on stall-out applications, under air pressure, there will be a small continuous by-pass flow.

Pure torque out-put from the actuator without external radial shaft bending loads is preferred to allow maximum bearing life. An arrangement with a semi-flexible coupling and the load shaft supported by separate bearings is recommended.

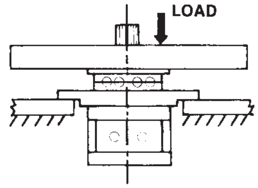


A similar arrangement is advised for power transmission through gears to eliminate gear load and separating forces from aggravating the actuator bearing load.



Where a flexible coupling cannot be used, very accurate alignment of the actuator and associated equipment is essential to prevent undue actuator bearing loading.

End thrust or axial loading of the actuator shaft is not advised. A thrust bearing, and the load driven through a sliding spline (or other means) is recommended to minimize internal wear for maximum actuator life.



Temperature:

Standard actuators, unless otherwise specified, may be operated satisfactorily between minus 30°F and plus 250°F. Operation at higher temperatures requires special seal compounds.

Filtration:

Filtration of operating fluid to the 25 micron range is recommended.

Storage:

Actuators, when stored for any extended period of time, will require additional rust protection. Upon receipt of the actuator, remove port plugs, fill the actuator chambers with clean, mineral-base oil (or other fluid compatible with seal compounds), and replace plugs securely. Cover exterior surfaces with adequate rust-preventive material. Place in a poly bag and seal.

Installation:

Normal machinists' practice and care should be used in installing actuators. As for any oscillating type actuator, the most efficient means of transmitting the torque developed is through multiple tooth, involute spline or SAE 10-B spline. Suitable flange type adapters and straight connectors are covered under "Accessories" in the catalog. These are also available through the local distributor.

System Pressure:

Caution must be exercised in actuator sizing by making allowance for a pressure drop throughout the hydraulic system in which the actuator is installed. If an extensive system of piping, control valves, flow control valves, etc. is present, it is to be expected that full line pressure will not be available at the actuator inlet port.

Angular Velocity:

Angular velocity can be readily controlled by metering the amount of flow of fluid into or out of the actuator ports. Many designs of flow control valves are available on the market for this purpose. If greater flow is required than that available in the selected standard actuator, special larger size ports can be specified within reasonable limits.

Service and Repair:

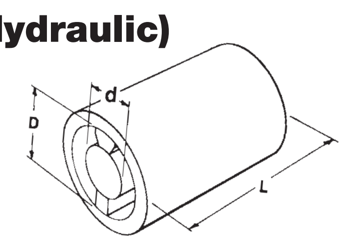
Seals in actuators are readily replaced by qualified personnel trained in hydraulic equipment repair. Interchangeable replacement parts are available from factory. Always specify the serial number and bill of material of unit when ordering spare or replacement parts. Replacement of worn bearings may be accomplished by qualified personnel, but we recommend that such repairs be made by the Factory Repair Department so that units can be reconditioned to meet original performance specifications.

Distributors in principal cities throughout the U.S., Canada, Europe, and Asia can supply you with additional information. If you have any questions, contact your distributor, or the actuator factory.

An overhaul procedure which contains complete instructions for replacement of seals or other worn parts, and an exploded view and parts list for ordering replacement parts, is available from the factory.

Service operations should be performed by competent hydraulic equipment technicians to maintain high manufacturing quality standards.

Basic Formulas (Hydraulic)



L = Body Length (in.)
D = Body I.D. (in.)
d = Hub dia. (in.)
ARC = Degrees of Rotation
N = Number of Vanes
PSI = Lbs/Sq. Inch (Pressure)

Displacement Per Radian = $[N \bullet L(D^2 - d^2)] \div 8$ (in^3/Rad.)

Theoretical Torque = $[N \bullet L(D^2 - d^2) \div 8]$PSI (in-lb)

Actual Torque = Theoretical Torque $\bullet$% efficiency (in-lb)

Total Displacement = $[L \bullet ARC \bullet N \bullet \pi\,(D^2 - d^2)] \div 1440$ (in^3)

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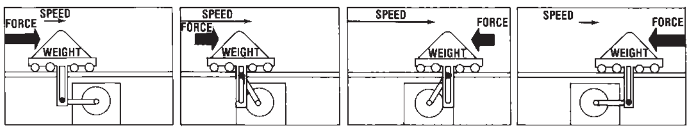
HARMONIC MOTION DRIVES

Applications requiring the linear transfer of a load under controlled acceleration and deceleration are quite common. Within limits, this type of motion can be achieved thru a harmonic motion drive. An actuator driven, scotch yoke arrangement as shown in Figure 1 imparts this type motion. The scotch yoke converts the constant speed rotating motion to a sinusoidal motion producing maximum linear force for acceleration, maximum linear speed thru the middle of the actuator stroke, and maximum decelerating forces to slow and stop the load.



The following equations assume a constant actuator rotational velocity. This is sometimes difficult to achieve, particularly for short cycle times that result in a large load velocity. The inertia of the load will tend to drive the actuator during the deceleration phase. These forces may cause cavitation or physical damage to the actuator. Therefore, under certain conditions the actuator may require external assistance in decelerating the load.

A flow control in the discharge side of the actuator provides this assistance, assuring a positive-pressure throughout the cycle. The added resisting torque resulting from the discharge metering must be added to the driving torque requirement.

Equations of Motion

The equation of motion for a Scotch Yoke mechanism can be developed as follows:

Referring to Figure 1.

(1) $s = r\cos\Theta$

and

(2) $\Theta = \omega t$

Where

ω = angular velocity of crank (link 1). $\frac{rad}{sec.}$

t=time, sec.

r=crank length, in.

s=horizontal movement of load W from midpoint of travel, in.

The velocity of link 2, and thus load W, may be found by differentiating the movement with respect to time.

(3) $v = \frac{d(-s)}{dt} = \frac{d(-r\cos\omega t)}{dt} = r\omega\sin\omega t$

The acceleration of load W is found by differentiating its velocity with respect to time:

(4) $a = \frac{dv}{dt} = \frac{d(r\omega\sin\omega t)}{dt} = r\omega^2\cos\omega t$

Therefore, when the crank rotates at constant angular velocity, the velocity and acceleration of the load can be determined for any position of the crank. Equation (4) indicates that maximum acceleration occurs when $\cos\omega t = 1$ or

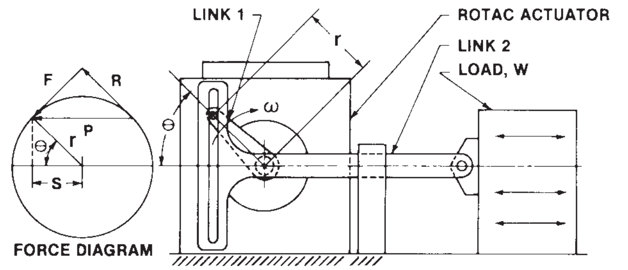
(5) $a\ max. = r\omega^2$

For a 180° crank throw, $\omega = \frac{\pi}{t'}$, where t' represents the time required to transfer the load a distance of 2r. Therefore,

(6) $a\ max. = r\left(\frac{\pi}{t'}\right)^2$

This relation applies for any load W.

FIGURE 1. TYPICAL HARMONIC MOTION DRIVE ARRANGEMENT



Required Torque

Consider an actuator powered Scotch Yoke mechanism moving a load as shown in Figure 1. Assume for simplicity that the system is frictionless. The forces acting on the actuator crank (link 1) are also shown in Figure 1.

(7) $P = \frac{W}{g}a = \frac{W}{g}(r\omega^2\cos\omega t)$

(8) $F = \frac{W}{g}(r\omega^2\cos\omega t)(\sin\omega t)$

(9) $R = \frac{W}{g}(r\omega^2\cos^2\omega t)$

Therefore, the required actuator torque at any time during the cycle is:

(10) $T = (F)r = \frac{Wr^2\omega^2}{g}(\cos\omega t)(\sin\omega t)$

The maximum torque requirement may be found by differentiating equation (10) with respect to time and setting the result equal to 0 as follows:

(11) $\frac{dT}{dt} = \frac{Wr^2\omega^2}{g}\ \frac{d(\cos\omega t\ \sin\omega t)}{dt} = 0$

$\frac{Wr^2\omega^2}{g}\ [\omega\cos^2\omega t - \omega\sin^2\omega t] = 0$

Since $\sin^2\omega t = 1 - \cos^2\omega t$, substitution into equation (11) yields

$\cos^2\omega t = 0.5$

or

$\cos\omega t = \sin\omega t = \sqrt{0.5}$

Therefore, the maximum actuator torque requirement is:

(12) $T\ max. = (.5)\frac{Wr^2\omega^2}{g}$

Recalling that $\omega = \frac{\pi}{t'}$ (t' = time for 180° crank throw)

and $g = 386.4\ in/sec^2$

(13) $T\ max. = \frac{(.5)(\pi)^2}{386.4}XW\left(\frac{r}{t'}\right)^2 = .01277W\left(\frac{r}{t'}\right)^2$ IN-LB with r measured in inches.

This expression may be used to determine the maximum actuator torque requirement for a frictionless system by knowing the load weight, crank arm length and the time required for 180º crank rotation.

In systems where friction must be considered, the required actuator torque will obviously be greater than that given by equation 13. The derivation of torque equations which consider the effects of friction becomes somewhat mathematically involved and will therefore not be repeated here.

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However, by considering only friction of the moving load and neglecting the crank friction forces along the vertical axis (vertical friction forces have little effect on torque) it can be shown that the maximum actuator torque is approximately:

(14) $T\ max. = Wr[.02554 \frac{r}{(t')^2} \cos\omega t + \mu]\sin \omega t$, in-lb

where μ = coefficient of friction of moving load

$$\omega t = \cos^{-1} \left\{-9.788 \frac{\mu(t')^2}{r} + .25 \left[1532.76 \frac{\mu^2(t')^4}{r^2} + 8\right]^{1/2}\right\}$$

ROTATIONAL SPEED OF ACTUATORS/PUMP CAPACITY REQUIRED

For hydraulic operation the time necessary for the actuator to make its travel arc can be figured with reasonable accuracy.

Where:

Arc=amount of rotation required (in degrees).

t=time, in seconds, for the actuator to make its arc of rotation.

Av=Angular velocity, in degrees per minute, for the actuator to make its arc of rotation.

Da=displacement, in cubic inches per radian, of the actuator.

GPM=gallons per minute required to rotate the actuator the specified arc in the specified time.

$$t = \frac{60 \bullet Arc}{Av}$$

$$Av = \frac{13235 \bullet GPM}{Da}$$

Example:

Calculate the time necessary to rotate an actuator 100°, that displaces 3.78 cubic inches per radian, with a five gallon per minute fluid supply.

$$Av = \frac{13235 \bullet GPM}{Da} = \frac{13235 \bullet 5}{3.78} = 17506.6 \text{ degrees per minute}$$

$$t = \frac{60 \bullet Arc}{Av} = \frac{60 \bullet 100}{17506.6} = .343 \text{ seconds}$$

Using the same basic formula, the GPM required to rotate an actuator a specified arc in a specified time can be figured.

$$GPM = \frac{Da \times Av}{13235}$$

Example:

Calculate the necessary pump capacity required to rotate an actuator that displaces 10.9 cubic inches per radian, 180° in .5 seconds.

$$Av = \frac{60 \bullet Arc}{t} = \frac{60 \bullet 180°}{.5} = 21,600 \text{ degrees per minute}$$

$$GPM = \frac{Da \bullet Av}{13235} = \frac{10.9 \bullet 21,600}{13235} = 17.79 \text{ Gallons per minute}$$

SAMPLE PROBLEMS

A few typical Rotac application problems are presented here along with simplified solutions which can be used to approximate the torque requirement for a specific job. These formulas should be used only as a guide in the selection of an actuator since friction and other system characteristics are not considered.

The symbols used in the sample problems are defined as follows:

a, b, ℓ	Dimensional Characteristics of Load, IN.
F	Force, LB.
g	Acceleration of Gravity, (386.4 IN./SEC.^{2})
Jm	Polar (mass) Moment of Inertia, in-lb sec^2
r	Radius, IN. (to the center of gravity of the weight)
t	Time, Sec. (per stroke or 1/2 cycle)
T	Torque, IN.-LB.
m	Mass of Load (Weight ÷ 386.4)
α	Angular Acceleration, RAD./SEC.^{2}
Θ	Angular movement in radians (degrees per stroke ÷ 57.3)

Problem #1

Find the torque required to rotate a rectangular load (horizontally) thru a given arc in a specified time. (See fig. 1)

Solution:

$T = \sum Jm\alpha$

$\sum Jm = Jm_1 + Jm_2$ . . . The sum of all polar mass moments of inertia being rotated.

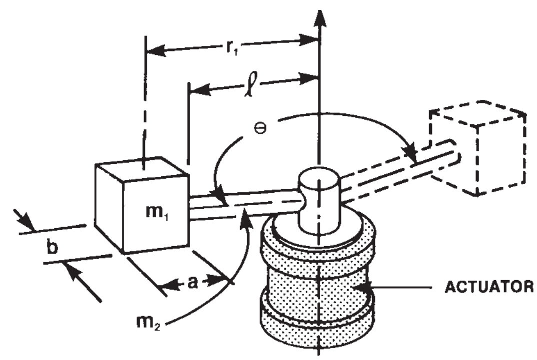

FIGURE 1

$Jm_1 \cong m_1 r_1^2$ (for applications where r is large in comparison to a & b)

$Jm_2 = \frac{m_2\ell^2}{3}$ (for a straight rod or any straight symmetrical shape)

$\alpha = \frac{4\Theta}{t^2}$ (assumes 50% of rotating time for acceleration and 50% for deceleration)

Example #1

Find the torque necessary to rotate a 20 lb. weight, 160°, in .5 seconds. The weight is supported by a 36" long, 3 lb. rod. (a & b are 8.4 inches) (r_1 = 40.2 inches)

$$Jm_1 \cong m_1 r_1^2 = \frac{20}{386.4}(40.2)^2 = 83.64 \text{ in-lb sec}^2$$

$$Jm_2 = \frac{m_2\ell^2}{3} = \frac{[3 \div (386.4)]36^2}{3} = 3.35 \text{ in-lb sec}^2$$

$$\Theta = \frac{160°}{57.3°} = 2.792 \text{ radians}$$

$$\alpha = \frac{4\Theta}{t^2} = \frac{4(2.792)}{.5^2} = 44.67 \text{ radians / sec.}^2$$

$T = \sum Jm\alpha = (Jm_1 + Jm_2)\alpha = (83.64 + 3.35)44.67 = 3885$ in-lb of torque required

Note: If r_1 is small in relation to a & b use: $Jm_1 = m_1\left(\frac{a^2 + b^2}{12} + r^2\right)$

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Example #2
(assume r_1 in example #1=12" all other parameters remain the same)

$$Jm_1 = m_1\left(\frac{a^2+b^2}{12}+r_1^2\right)=\frac{20}{386.4}\left(\frac{8.4^2+8.4^2}{12}+12^2\right)=8.06 \text{ in-lb sec}^2$$

$$\ell = r_1 - (a\div 2) = 12 - (8.4\div 2) = 7.8$$

$$Jm_2 = \frac{m_2\ell^2}{3}\ \frac{[3\div(386.4)]7.8^2}{3} = .157 \text{ in-lb sec}^2$$

α = same as previous (44.67)

$T = \sum Jm\alpha = (Jm_1 + Jm_2)\alpha = (8.06 + .157)\ 44.67$
= 367 in-lb of torque required

Problem 1A:
Find the torque required to lift a weight and rotate it vertically thru a specified arc in a specified time.

Solution:

$$T = \sum(Jm\alpha + (Wr\cos\Theta_s))$$

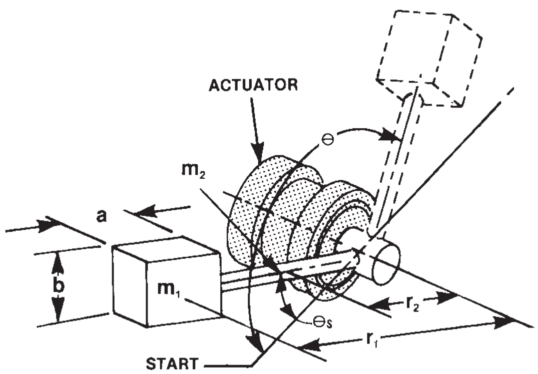


FIGURE 2

Note: $Jm\alpha$ is the torque required to move the load without the effect of gravity.
$Wr\cos\Theta$ is the torque resulting from the effect of gravity on the load. The torque required changes as the angle changes, the maximum requirement at horizontal, lessening to zero at the vertical. The torque value is negative past vertical, gravitational forces actually aiding in producing torque.

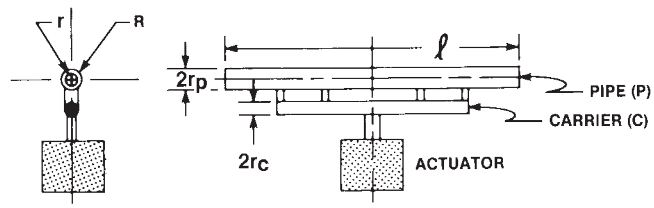
Example #3
Find the torque required if the load in example # 1 is rotated vertically. Assume the starting angle (Θ_S) is 20°.

Assume:
$Jm\alpha = T = 3885$ in.-lb. (from example #1) $w_1 = 20$ lb., $w_2 = 3$ lb., $r_1 = 40.2$, $r_2 = 20.1$

$T = \sum(Jm\alpha + (w_1 r_1 + w_2 r_2)\cos\Theta_S)$
$= 3885 + (20 \bullet 40.2 + 3 \bullet 20.1)\cos 20° = 4697$ in.-lb. required at start.

$Tmax = \sum[Jm\alpha + (w_1 r_1 + w_2 r_2)]$
$= [3885 + (20\bullet 40.2 + 3\bullet 20.1)] = 4749$ in-lb

Problem 2:
Find the torque required to rotate a thin hollow pipe about its transverse axis through a given angle in a specified time.



Solution:

$$T = Jm\alpha = (Jm_c + Jm_p)\ \alpha$$

For thin-walled pipe

$$Jm_p = \frac{m}{2}\left(r_p^2 + \frac{\ell_p^2}{6}\right)$$

For thick-walled pipe

$$Jm_p = \frac{m}{4}\left(R_p^2 + r_p^2 + \frac{\ell_p^2}{3}\right)$$

For solid-circular bar

$$Jm_c = \frac{m}{12}\left(3r_c^2 + \ell_c^2\right)$$

Assume:
50% (t) for acceleration
50% (t) for deceleration

Therefore,

$$\alpha = \frac{4\Theta}{t^2}$$

Example:
Assume:
Carrier: — 1" dia. x 12" long steel bar (2.7 Lb.)
Pipe: — 2.88 I.D. x 3.00 O.D. x 36" long (steel) (6 Lb.)
Rotate pipe 180° in 2 secs.

$$m = \frac{W}{386.4}$$

$$T = (Jm_p = Jm_c)\ \alpha$$

$$Jm_p = \frac{m}{2}\left(r_p^2 + \frac{\ell_p^2}{6}\right) = \frac{.0155}{2}\left(1.44^2 + \frac{36^2}{6}\right) = 1.690 \text{ in-lb sec}^2$$

$$Jm_c = \frac{m}{12}\left(3r_c^2 + \ell_c^2\right) = \frac{.007}{12}\left(3(.5)^2 + 12^2\right) = .084 \text{ in-lb sec}^2$$

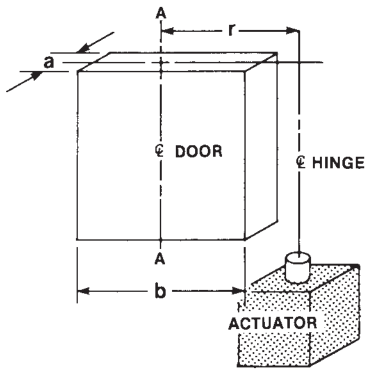
$$\alpha = \frac{4\Theta}{t^2} = \frac{4(180\div 57.3)}{2^2} = \frac{4(3.14)}{4} = 3.14 \text{ rad/sec.}^2$$

$T = (1.690 + .084)\ 3.14 = 5.57$ in.-lb. torque required

Engineering Data

ENGINEERING DATA

Problem 3:
Find the torque required to open or close a door through a given angle in a specified time.



Solution:

$$T = Jm \text{ ℄ hinge } \alpha$$

$$Jm_{A\text{-}A} = \frac{m}{12}(a^2 + b^2)$$

$$Jm \text{ ℄ hinge} = Jm_{A\text{-}A} + mr^2$$

Assume:

50% (t) for acceleration
50% (t) for deceleration

Therefore,

$$\alpha = \frac{4\Theta}{t^2}$$

Example:
Find the torque necessary to open a 350 Lb. door 100° in .8 secs.
Assume:
door: a = 4", b = 36", r = 22", w = 350 Lb.

$$m = \frac{W}{386.4}$$

$$T = Jm \text{ ℄ hinge } \alpha$$

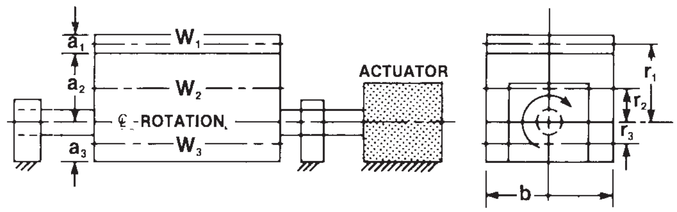
$$Jm_{A\text{-}A} = \frac{m}{12}(a^2 + b^2) = \frac{.906}{12}(4^2 + 36^2) = 99.06 \text{ in-lb sec}^2$$

$$Jm \text{ ℄ hinge} = Jm_{A\text{-}A} + (mr^2) = 99.06 + (.906\,(22^2)) = 537.56 \text{ in-lb sec}^2$$

$$\alpha = \frac{4\Theta}{t^2} = \frac{4(100 \div 57.3)}{.8^2} = \frac{6.98}{.64} = 10.91 \text{ rad./sec}^2$$

$$T = Jm \text{ ℄ hinge } \alpha = 537.56\,(10.91) = 5864.12 \text{ in-lb. torque required}$$

Problem 4:
Find the torque required to rotate several plates of various thicknesses through a given angle in a specified time.



Solution:

$$T = Jm \text{ ℄ Rotation } \alpha = \sum [(Jm_1 + Jm_2 + Jm_3)\,\alpha + (w_1r_1 + w_2r_2 + w_3r_3)]$$

$$Jm_1 = \frac{M_1}{12}(a_1{}^2 + b_1{}^2) + m_1r_1{}^2$$

$$Jm_2 = \frac{M_2}{12}(a_2{}^2 + b_2{}^2) + m_2r_2{}^2$$

$$Jm_3 = \frac{M_3}{12}(a_3{}^2 + b_3{}^2) + m_3r_3{}^2$$

Assume:

50% (t) for acceleration
50% (t) for deceleration

Therefore,

$$\alpha = \frac{4\Theta}{t^2}$$

Example:
Rotate three plates as shown, 180º in 2 secs.
Assume:

w_1: a_1 =.5", b_1 =6" weight=10 Lb., r_1 =5.25
w_2: a_2 =5", b_2 =6" weight=100 Lb., r_2 =2.5
w_3: a_3 =2", b_3 =6" weight=40 Lb., r_3 =1.0

$$m = \frac{W}{386.4}$$

$$T = Jm \text{ ℄ Rotation } \alpha = \sum (Jm_1 + Jm_2 + Jm_3)\,\alpha$$

$$Jm_1 = \frac{m_1}{12}(a_1{}^2 + b_1{}^2) + m_1\,r_1{}^2 = \frac{.026}{12}(.5^2 + 6^2) + .026\,(5.25)^2$$
$$= .795 \text{ in-lb sec}^2$$

$$Jm_2 = \frac{m_1}{12}(a_2{}^2 + b_2{}^2) + m_2\,r_2{}^2 = \frac{.259}{12}(5^2 + 6^2) + .259\,(2.5)^2$$
$$= 2.94 \text{ in-lb sec}^2$$

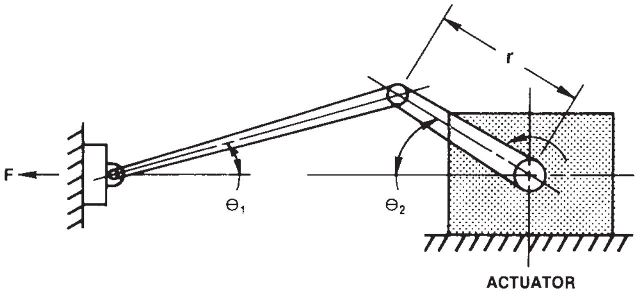
$$Jm_3 = \frac{m_3}{12}(a_3{}^2 + b_3{}^2) + m_3\,r_3{}^2 = \frac{.104}{12}(2^2 + 6^2) + .104\,(1.0)^2$$
$$= .451 \text{ in-lb sec}^2$$

$$\alpha = \frac{4\Theta}{t^2} = \frac{4(180 \div 57.3)}{2^2} = \frac{4(3.14)}{4} = 3.14 \text{ rad/sec.}_2$$

$$T = \sum [(Jm_1 + Jm_2 + Jm_3)\,\alpha + (w_1r_1 + w_2r_2 + w_3r_3)]$$
$$= [(.795 + 2.94 + .451)\,3.14 + (10 \times 5.25 + 100 \times 2.5 + 40 \times 1)]$$
$$= 355.64 \text{ in-lb torque required}$$

ENGINEERING DATA

Problem 5:
Find the torque required to produce a given force as shown in the figure below.



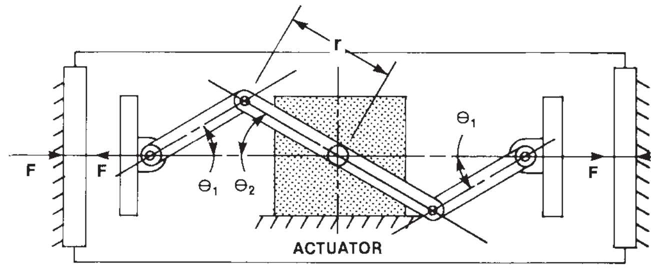
Solution:

$$T = \left[\frac{Fr}{\cos\Theta_1} \sin(\Theta_1 + \Theta_2) \right]$$

Design Notes:
1. The design should be such that angles Θ_1 and Θ_2 are not permitted to go to zero degrees.

2. Force, F, must be less than the bearing capacity of the actuator.

Problem 6:
Find the torque required to produce a given force in a typical die closer application.



Solution:

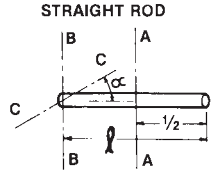
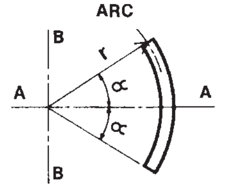
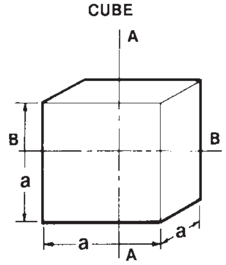
$$T = \left[\frac{2Fr}{\cos\Theta_1} \sin(\Theta_1 + \Theta_2) \right]$$

Design Notes:
1. The design should be such that angles Θ_1 and Θ_2 are not permitted to go to zero degrees.

2. Force, F, may be greater than the bearing capacity of the actuator since it is transmitted through the linkage, and not to the bearing.

REFERENCE DATA

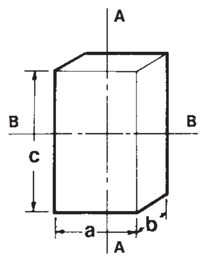
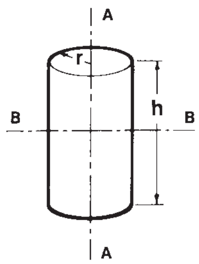
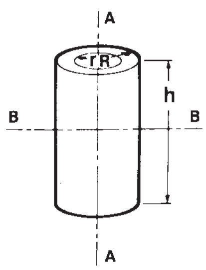
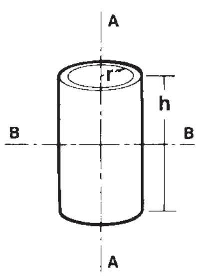
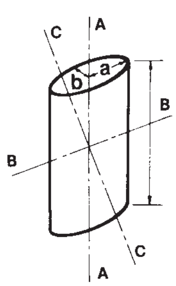
PROPERTIES OF VARIOUS SOLIDS*

Solids	Polar mass Moment of inertia, Jm	Radius of gyration, K
STRAIGHT ROD 	$J_{AA} = \frac{m\ell^2}{12}$ $J_{BB} = \frac{m\ell^2}{3}$ $J_{CC} = \frac{m\ell^2 \sin^2\alpha}{3}$	$K_{AA} = \frac{\ell}{\sqrt{12}}$ $K_{BB} = \frac{\ell}{\sqrt{3}}$ $K_{CC} = \ell\sqrt{\frac{\sin\alpha}{3}}$
ROD BENT INTO A CIRCULAR ARC 	$J_{AA} = \frac{mr^2}{2}\left[1 - \frac{\sin\alpha\cos\alpha}{\alpha}\right]$ $J_{BB} = \frac{mr^2}{2}\left[1 + \frac{\sin\alpha\cos\alpha}{\alpha}\right]$	$K_{AA} = r\sqrt{1/2\left(1 - \frac{\sin\alpha\cos\alpha}{\alpha}\right)}$ $K_{BB} = r\sqrt{1/2\left(1 + \frac{\sin\alpha\cos\alpha}{\alpha}\right)}$
CUBE 	$J_{AA} = J_{BB} = \frac{ma^2}{6}$	$K_{AA} = K_{BB} = \frac{a}{\sqrt{6}}$

* All axes pass through the center of gravity unless otherwise noted. W = total weight of the body. $m = \frac{W}{386.4}$

REFERENCE DATA

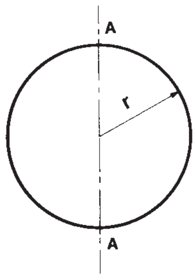
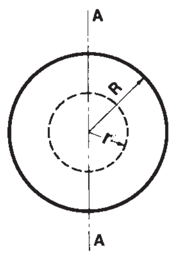
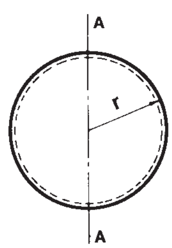
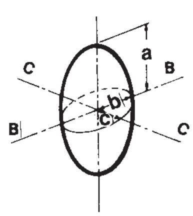
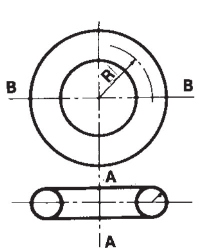
PROPERTIES OF VARIOUS SOLIDS* (CONTINUED)

Solids	Polar mass Moment of inertia, Jm	Radius of gyration, K
RECTANGULAR PRISM 	$J_{AA} = \frac{m(a^2 + b^2)}{12}.$ $J_{BB} = \frac{m(b^2 + c^2)}{12}.$	$K_{AA} = \sqrt{\frac{a^2 + b^2}{12}}.$ $K_{BB} = \sqrt{\frac{b^2 + c^2}{12}}.$
RIGHT CIRCULAR CYLINDER 	$J_{AA} = \frac{mr^2}{2}.$ $J_{BB} = \frac{m(3r^2 + h^2)}{12}.$	$K_{AA} = \frac{r}{\sqrt{2}}.$ $K_{BB} = \sqrt{\frac{3r^2 + h^2}{12}}.$
HOLLOW RIGHT CIRCULAR CYLINDER 	$J_{AA} = \frac{m(R^2 + r^2)}{2}.$ $J_{BB} = \frac{m(R^2 + r^2 + \frac{h^2}{3})}{4}.$	$K_{AA} = \sqrt{\frac{R^2 + r^2}{2}}.$ $K_{BB} = \sqrt{\frac{3R^2 + 3r^2 + h^2}{12}}.$
THIN HOLLOW CYLINDER 	$J_{AA} = mr^2.$ $J_{BB} = \frac{m}{2}\left(r^2 + \frac{h^2}{6}\right).$	$K_{AA} = r.$ $K_{BB} = \sqrt{\frac{6r^2 + h^2}{12}}.$
ELLIPTICAL CYLINDER 	$J_{AA} = \frac{m(a^2 + b^2)}{4}.$ $J_{BB} = \frac{m(3b^2 + h^2)}{12}.$ $J_{CC} = \frac{m(3a^2 + h^2)}{12}.$	$K_{AA} = \sqrt{\frac{a^2 + b^2}{2}}.$ $K_{BB} = \sqrt{\frac{3b^2 + h^2}{12}}.$ $K_{CC} = \sqrt{\frac{3a^2 + h^2}{12}}.$

* All axes pass through the center of gravity unless otherwise noted. W = total weight of the body. $m = \frac{W}{386.4}$

REFERENCE DATA

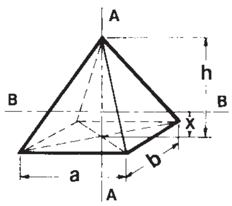
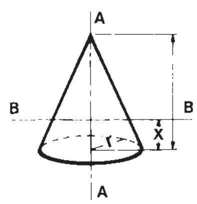
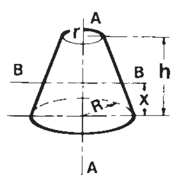
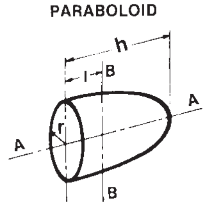
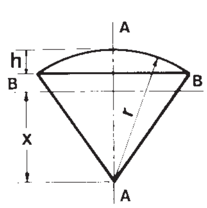
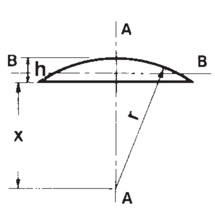
PROPERTIES OF VARIOUS SOLIDS* (CONTINUED)

Solids	Polar mass Moment of inertia, Jm	Radius of gyration, K
SPHERE 	$J_{AA} = \frac{2mr^2}{5}$.	$K_{AA} = \frac{2r}{\sqrt{10}}$.
HOLLOW SPHERE 	$J_{AA} = \frac{2m}{5}\left(\frac{R^5 - r^5}{R^3 - r^3}\right)$.	$K_{AA} = 2/5 \frac{R^5 - r^5}{R^3 - r^3}$.
THIN HOLLOW SPHERE 	$J_{AA} = \frac{2mr^2}{3}$.	$K_{AA} = \frac{2r}{\sqrt{6}}$.
ELLIPSOID 	$J_{AA} = \frac{m(b^2 + c^2)}{5}$. $J_{BB} = \frac{m(a^2 + c^2)}{5}$. $J_{CC} = \frac{m(a^2 + b^2)}{5}$.	$K_{AA} = \sqrt{\frac{b^2 + c^2}{5}}$. $K_{BB} = \sqrt{\frac{a^2 + c^2}{5}}$. $K_{CC} = \sqrt{\frac{a^2 + b^2}{5}}$.
TORUS 	$J_{AA} = m\left(R^2 + \frac{3R^2}{4}\right)$. $J_{BB} = m\left(\frac{R^2}{2} + \frac{5r^2}{8}\right)$.	$K_{AA} = 1/2\sqrt{4R^2 + 3r^2}$. $K_{BB} = \sqrt{\frac{4R^2 + 5r^2}{8}}$.

* All axes pass through the center of gravity unless otherwise noted. W = total weight of the body. $m = \frac{W}{386.4}$

REFERENCE DATA

PROPERTIES OF VARIOUS SOLIDS* (CONTINUED)

Solids	Distance to center of gravity, x	Polar mass Moment of inertia, Jm	Radius of gyration, K
RIGHT RECTANGULAR PYRAMID 	$x = \frac{h}{4}$.	$J_{AA} = \frac{m\,(a^2 + b^2)}{20}$. $J_{BB} = \frac{m}{20}\,(b^2 + \frac{3h^2}{4})$.	$K_{AA} = \sqrt{\frac{a^2 + b^2}{20}}$. $K_{BB} = \sqrt{1/80\,(4b^2 + 3h^2)}$.
RIGHT CIRCULAR CONE 	$x = \frac{h}{4}$.	$J_{AA} = \frac{3mr^2}{10}$. $J_{BB} = \frac{3m}{20}\left(r^2 + \frac{h^2}{4}\right)$.	$K_{AA} = \frac{3r}{\sqrt{30}}$. $K_{BB} = \sqrt{3/80\,(4r^2 + h^2)}$.
FRUSTRUM OF RIGHT CIRCULAR CONE 	$x = \frac{h\,(R^2 + 2Rr + 3r^2)}{4\,(R^2 + Rr + r^2)}$	$J_{AA} = \frac{3m}{10}\,\frac{(R^5 - r^5)}{(R^3 - r^3)}$.	$K_{AA} = \sqrt{3/10\,\frac{(R^5 - r^5)}{(R^3 - r^3)}}$.
PARABOLOID 	$x = 1/3h$.	$J_{AA} = \frac{mr^2}{3}$. $J_{BB} = \frac{m}{10}\,(3r^2 + h^2)$.	$K_{AA} = \frac{r}{\sqrt{3}}$. $K_{BB} = \sqrt{1/10\,(3r^3 + h^2)}$.
SPHERICAL SECTOR 	$x = 3/8(2r - h)$.	$J_{AA} = \frac{m\,(3rh - h^2)}{5}$.	$K_{AA} = \sqrt{\frac{3rh - h^2}{5}}$
SPHERICAL SEGMENT 	$x = \frac{3}{4}\,\frac{(2r - h)^2}{(3r - h)}$. For half sphere $x = 3/8\ r$.	$J_{AA} = m\left(r^2\,\frac{3rh}{4} + \frac{3h^2}{20}\right)\frac{2h}{3r - h}$.	$K_{AA} = \sqrt{\frac{I}{W}}$.

* All axes pass through the center of gravity unless otherwise noted. W = total weight of the body. $m = \frac{W}{386.4}$

REFERENCE DATA

DEFINITIONS, ABBREVIATIONS AND SYMBOLS

ABBREVIATIONS:

BTU.	British Thermal Unit — 1 BTU = Heat required to raise temperature of one pound of water 1°F.
ºC	Degrees Centigrade
CAL.	Calorie — 1 CAL. = Heat required to raise temperature of one gram of water 1°C.
C. C.	Cubic Centimeter
CU. FT.	Cubic Foot
CU. IN.	Cubic Inch
°F	Degrees Fahrenheit
FPS.	Feet per second
FT.	Feet (foot)
GAL.	U.S. Gallon
GPM.	Gallons per minute
HP.	Horsepower = Work at rate of 33,000 FT. LB./MIN.
IN.	Inch(es)
IPS	Inches per second
°K	Degrees Kelvin
LB.	Pound(s)
MIN.	Minute(s) of time
PSI	Pounds per square inch
REV.	Revolutions (of shaft or pump)
SEC.	Second(s) of time
SP. GR.	Specific Gravity — Ratio of the weight of a body to the weight of an equal volume of water at 4°C or other specified temperature.
SP. HT.	Specific Heat — Ratio of heat required to raise a unit weight of a substance 1°F. to the amount of heat required to raise an equal weight of water 1°F. at a certain temperature. (Hydraulic oil is approx. 0.45.)
SP. WT.	Specific weight or weight density = LB./CU.FT.; LB./CU. IN. or grams/C.C.
SQ. IN.	Square inch(es)

SYMBOLS:

A	Area
a	Linear acceleration (FPS^2), rate of change of velocity
α	Angular acceleration (Radians per SEC.^{2})
C	Compressibility of oil (CU. IN.)
D	Density, mass per unit volume
E	Energy
F	Force, (LB.) an influence which produces or tends to produce, motion or change of motion.
f	Coefficient of friction
g	Acceleration of gravity (IPS^2) = 386.4 at sea level
H_E	Elevation Head
H_g	Mercury
H_P	Pressure head (static)
H_v	Velocity head
L	Gallons per minute (GPM)
M	Mass = $\frac{W}{386.4}$; or a mass which, with an unbalanced force of 1 LB. acting upon it, would have an acceleration of|1|IPS^2.
M_f	Mechanical friction
N	Revolutions per minute (RPM)
ΔP	Pressure differential (DROP)
P	Pounds per square inch (PSI)
r	Arm (torque), radius in inches
T	Torque (inch-pounds)
U	Velocity (FPS) rate of change of distance (length)
V	Volume (CU. IN.)
W	Weight (LB.) force which gravitation exerts on a material body.

CONVERSION TABLES

TORQUE
IN-LB x .1130 = N-m
N-m x 8.851 = IN-LB
N-m x 9.807 = Kgf-m
Kgf-m x 86.799 = IN-LB

PRESSURE
PSI x .06895 = BAR
BAR x 14.5 = PSI
Kpa x .1450 = PSI
PSI x 6.895 = Kpa

VOLUME
Cubic Inches x 16.39 = CU. CMS
CU. CMS x .06102 = Cubic Inches
Gallon x 3.785 = Liter
Liter x .264 = Gallon
Gallon x 3785 = CU.CMS
CU. CMS x .0002642 = Gallon

MASS
Kg x 2.2046 = Lbs
Lbs x .4536 = Kg

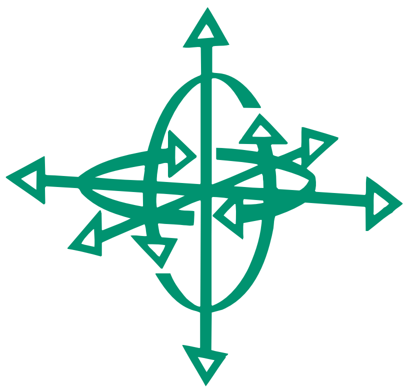
POWER
Hp x .7457 = Kw

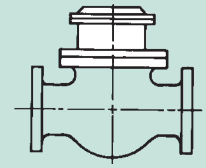
REFERENCE DATA

DEFINITIONS BY FORMULAS

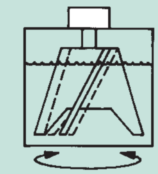
ACCELERATION	$a = \frac{F}{M} = \frac{Fg}{W}$ From $F = Ma$ and $M = \frac{W}{g}$ α = Radians/SEC.^{2} = $\frac{\text{Degrees/SEC.}^2}{57.3}$
FORCE	$F = AP$
FRICTION	$M_f = W \text{ x} f$ Note: Static (or breakaway) friction coefficient is greater than kinetic (or moving) friction coefficient
GRAVITY	g = 386.4 in. / SEC. ^{2} (at sea level)
HORSEPOWER	$HP = \frac{FU}{550} = \frac{LP}{1714} = \frac{TN}{63{,}025}$
MASS	$M = \frac{W}{g}$ or, at sea level, $= \frac{W}{32.2}$, or $= \frac{W \text{ (grams)}}{980}$ or $\frac{W}{386.4}$ **NOTE:** Mass is constant regardless of altitude.
ORIFICE AREA	See pressure drop
PRESSURE	$P = \frac{F}{A}$ (consistent units)
PRESSURE DROP	For oil hydraulic systems, the following will approximate pressure drop thru "short orifice" (1/4 to 1/2-inch long-length not over 3 times diameter) $\Delta P = \frac{0.001056L^2}{A^2}$ For specified pressure drop: A (required) = $\frac{0.0325L}{\sqrt{\Delta P}}$
RADIAN	Arc (of circle) = Length of radius (see velocity, angular) In degrees $= \frac{360}{2\pi} = \frac{180}{\pi} = 57.3°$
SPRING RATE	$\frac{F}{\text{Distance compressed (or stretched) where distance is from the free length.}}$
TORQUE	$T = F \text{ x } r = \frac{HP \text{ x } 63.025}{N} = \frac{\text{CU. IN. /REV x } P}{2\pi}$
VELOCITY, Angular Flow	Radians/SEC. = $\frac{\text{Degrees/SEC.}}{57.3}$ $U = 0.321 \frac{L}{A}$

FLOW RATE DATA GUIDE TO SIZE

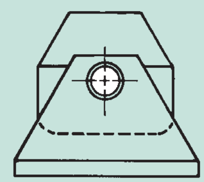




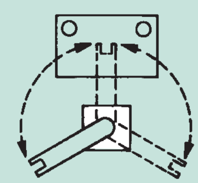
VALVE OPEN—CLOSE



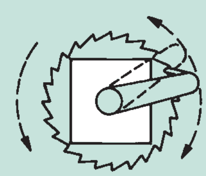
MIX—STIR



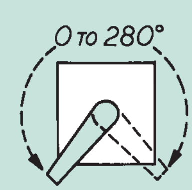
TURNOVER—DUMP



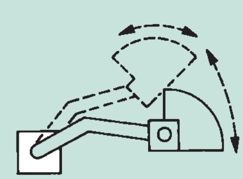
LOAD—POSITION—UNLOAD



CONTINUOUS ROTATION



TURN—OSCILLATE



MATERIAL HANDLING

FLOW RATE FORMULAS

GMP = 3.117 AV

RPS = $\frac{.0333\,AV\Theta}{D}$

Rad/Sec = 2 π (RPS)

GPM = Gallons per minute

RPS = Revolutions per second

Rad/Sec = Radians per second

Where:
A = Port area (in^2)
V = Flow velocity in feet per sec.
Θ = Amount of rotation (degrees)
D = Total displacement of actuator (in^3)

Engineering Data

FLOW RATE DATA

	MODEL	SAE STRAIGHT THREAD PORT SIZE	PORT DIAMETER TUBE I.D. ①	PORT AREA (IN^2)	ACTUATOR DISPLACEMENT IN^3 TOTAL	IN^3 RADIAN	FLOW RATE AND ANGULAR VELOCITY AT 10 FPS OIL VELOCITY GPM	RAD/SEC	RPS	TIME (SEC.) PER STROKE	FLOW RATE AND ANGULAR VELOCITY AT 15 FPS OIL VELOCITY GPM	RAD/SEC	RPS	TIME (SEC.) PER STROKE
MEDIUM PRESSURE	MPJ-11-1V	3/8-24	.117	.0107	.835	.178	.33	7.22	1.15	.65	.50	10.87	1.73	.43
	-2V				.557	.357		3.52	.56	.45		5.43	.86	.29
	MPJ-22-1V	1/2-20	.187	.0275	3.820	.815	.86	4.07	.65	1.16	1.28	6.10	.97	.77
	-2V				2.560	1.631		2.02	.32	.78		3.03	.48	.52
	MPJ-32-1V	7/8-14	.435	.1493	9.2	1.88	4.63	9.47	1.51	.52	6.95	14.2	2.26	.34
	-2V				6.6	3.78		4.72	.75	.37		7.07	1.13	.25
	MPJ-34-1V	7/8-14	.435	.1493	18.4	3.76	4.63	4.74	.75	1.03	6.95	7.10	1.13	.69
	-2V				13.0	7.44		2.40	.38	.73		3.59	.57	.49
	MPJ-63-1V	1 1/16-12	.532	.2223	53.3	10.90	6.93	2.44	.39	1.99	10.39	3.67	.58	1.33
	-2V				38.0	21.77		1.22	.19	1.42		1.84	.29	.95
	MPJ-84-1V	1 5/16-12	.760	.4537	127.4	26.07	14.14	2.09	.33	2.34	21.21	3.13	.50	1.56
	-2V				91.0	52.14		1.04	.17	1.63		1.57	.25	1.11
	MPJ-105-1V	1 5/8-12	1.01	.8012	253.3	51.83	24.97	1.85	.29	2.63	37.96	2.78	.44	1.76
	-2V				181.0	103.71		.93	.15	1.85		1.39	.22	1.26
	MPJ-116-1V	1 7/8-12	1.26	1.247	412.9	84.50	38.87	1.77	.28	2.76	58.30	2.66	.42	1.84
	-2V				295.0	169.04		.88	.14	1.98		1.33	.21	1.31
	MPJ-128-1V	1 7/8-12	1.26	1.247	588.4	120.41	38.87	1.24	.20	3.93	58.30	1.86	.30	2.62
	-2V				420.3	240.83		.62	.10	2.78		.93	.15	1.87
	SS-1-1V	7/16-20	.152	.0182	5.86	1.20	.57	1.82	.29	2.69	.85	2.72	.43	1.79
	-2V				4.19	2.40		.91	.14	1.92		1.36	.22	1.28
	SS-4-1V	9/16-18	.245	.0472	18.62	3.81	1.47	1.48	.24	3.29	2.20	2.23	.35	2.19
	-2V				13.29	7.62		.74	.12	2.35		1.11	.18	1.57
	SS-8-1V	9/16-18	.245	.0472	39.09	8.00	1.47	.71	.11	6.91	2.20	1.06	.17	4.60
	SS-12-1V	3/4-16	.334	.0876	60.84	12.45	2.73	.84	.13	5.79	4.10	1.27	.20	3.86
	-2V				43.46	29.90		.42	.07	4.13		.63	.10	2.76
	SS-25-1V	7/8-14	.435	.1493	43.46	24.90	4.63	2.00	.32	2.44	6.95	3.01	.48	1.62
	SS-40-1V	1 5/16-12	.760	.4537	195.46	40.00	14.14	1.36	.22	3.59	21.21	2.04	.32	2.39
	-2V				139.62	80.00		.68	.11	2.56		1.02	.16	1.71
	SS-65-1V	1 5/16-12	.760	.4537	317.63	65.00	14.14	.84	.13	5.83	21.21	1.26	.20	3.89
	-2V				226.88	130.00		.42	.07	4.17		.63	.10	2.79
	SS-130-1V	1 5/8-12	1.010	.8012	635.25	130.00	24.97	.74	.12	6.61	37.46	1.11	.18	4.40
	-2V				453.75	260.00		.37	.06	4.72		.55	.09	3.15
HIGH PRESSURE	HS-1.5-1V	9/16-18	.245	.0472	7.33	1.50	1.47	3.77	1.30	.93	2.20	5.66	.90	.86
	HS-2.5-1V	9/16-18	.245	.0472	12.22	2.50	1.47	2.26	.36	2.16	2.20	3.39	.54	1.44
	HS-4.0-1V	9 16-18	.245	.0472	19.55	4.00	1.47	1.41	.23	3.46	2.20	2.12	.34	2.30
	HS-6.0-1V	3/4-16	.334	.0876	31.08	6.36	2.73	1.65	.26	2.95	4.10	2.48	.39	1.97
	HS-10-1V	3/4-16	.334	.0876	46.62	9.54	2.73	1.10	.18	4.43	4.10	1.65	.26	2.96
	HS-15-1V	3/4-16	.334	.0876	73.30	15.00	2.73	.70	.11	6.97	4.10	1.05	.17	4.65
	SS-.2A-1V	—	.125	.0123	.98	.20	.38	7.34	1.17	.67	.57	11.02	1.75	.44
	SS-.5A-1V	3/8-24	.117	.0107	2.20	.45	.34	2.87	.46	1.71	.50	4.30	.68	1.14
	-2V				1.57	.90		1.43	.23	1.22		2.15	.34	.81

NOTE: ① INLET HOLE DIA. IN MPJ-22 & SS-.2A UNITS

	MODEL	SAE STRAIGHT THREAD PORT SIZE	PORT DIAMETER TUBE I.D. ①	PORT AREA (IN^2)	ACTUATOR DISPLACEMENT IN^3 TOTAL	ACTUATOR DISPLACEMENT IN^3 RADIAN	FLOW RATE AND ANGULAR VELOCITY AT 20 FPS OIL VELOCITY GPM	RAD/SEC	RPS	TIME (SEC.) PER STROKE	FLOW RATE AND ANGULAR VELOCITY AT 25 FPS OIL VELOCITY GPM	RAD/SEC	RPS	TIME (SEC.) PER STROKE
MEDIUM PRESSURE	MPJ-11-1V -2V	3/8-24	.117	.0107	.835 .557	.178 .357	.67	14.95 7.22	2.30 1.15	.33 .22	.83	18.10 9.05	2.88 1.44	.26 .17
	MPJ-22-1V -2V	1/2-20	.187	.0275	3.820 2.560	.815 1.631	1.71	8.13 4.04	1.29 .64	.56 .39	2.14	10.16 5.06	1.62 .80	.46 .31
	MPJ-32-1V -2V	7/8-14	.435	.1493	9.2 6.6	1.88 3.78	9.26	18.95 9.43	3.02 1.50	.26 .19	11.58	23.68 11.79	3.77 1.88	.21 .15
	MPJ-34-1V -2V	7/8-14	.435	.1493	18.4 13.0	3.76 7.44	9.26	9.47 4.79	1.51 .76	.52 .36	11.58	11.84 5.99	1.88 .95	.41 .29
	MPJ-63-1V -2V	1 1/16-12	.532	.2223	53.3 38.0	10.90 21.77	13.85	4.89 2.45	.79 .39	1.00 .71	17.32	6.11 3.06	.97 .49	.80 .57
	MPJ-84-1V -2V	1 5/16-12	.760	.4537	127.4 91.0	26.07 52.14	28.28	4.18 2.09	.66 .33	1.17 .84	35.35	5.22 2.61	.83 .42	.94 .67
	MPJ-105-1V -2V	1 5/8-12	1.01	.8012	253.3 181.0	51.83 103.71	49.95	3.71 1.85	.59 .30	1.32 .94	62.43	4.64 2.32	.74 .37	1.05 .75
	MPJ-116-1V -2V	1 7/8-12	1.26	1.247	412.9 295.0	84.50 169.04	77.73	3.54 1.77	.56 .28	1.38 .99	97.16	4.43 2.21	.70 .35	1.10 .79
	MPJ-128-1V -2V	1 7/8-12	1.26	1.247	588.4 420.3	120.41 240.83	77.73	2.49 1.24	.40 .20	1.97 1.40	97.16	3.11 1.55	.49 .25	1.57 1.12
	SS-1-1V -2V	7/16-20	.152	0.182	5.86 4.19	1.20 2.40	1.13	3.63 1.81	.58 .29	1.35 .96	1.41	4.54 2.27	.72 .36	1.08 .77
	SS-4-1V -2V	9/16-18	.245	.0472	18.62 13.29	3.81 7.62	2.94	2.97 1.49	.47 .24	1.65 1.17	3.67	3.71 1.86	.59 .30	1.32 .94
	SS-8-1V	9/16-18	.245	.0472	39.09	8.00	2.94	1.41	.23	3.45	3.67	1.77	.28	2.76
	SS-12-1V -2V	3/4-16	.334	.0876	60.84 43.46	12.45 29.90	5.46	1.69 .84	.27 .13	2.89 2.07	6.83	2.11 1.06	.34 .17	2.31 1.65
	SS-25-1V	7/8-14	.435	.1493	43.46	24.90	9.26	4.01	.64	1.22	11.58	5.01	.80	.97
	SS-40-1V -2V	1 5/16-12	.760	.4537	195.46 139.62	40.00 80.00	28.28	2.72 1.36	.43 .22	1.80 1.28	35.35	3.40 1.70	.54 .27	1.44 1.03
	SS-65-1V -2V	1 5/16-12	.760	.4537	317.63 226.88	65.00 130.00	28.28	1.68 .84	.27 .13	2.92 2.08	35.35	2.09 1.05	.33 .17	2.33 1.67
	SS-130-1V -2V	1 5/8-12	1.010	.8012	635.25 453.75	130.00 260.00	49.96	1.48 .74	.24 .12	3.30 2.36	62.43	1.85 .92	.29 .15	2.64 1.89
HIGH PRESSURE	HS-1.5-1V	9/16-18	.245	.0472	7.33	1.50	2.94	7.54	1.20	.65	3.67	9.43	1.50	.52
	HS-2.5-1V	9/16-18	.245	.0472	12.22	2.50	2.94	4.52	.72	1.08	3.67	5.66	.90	.86
	HS-4.0-1V	9/16-18	.245	.0472	19.55	4.00	2.94	2.83	.45	1.73	3.67	3.54	.56	1.38
	HS-6.0-1V	3/4-16	.334	.0876	31.08	6.36	5.46	3.31	.53	1.48	6.83	4.13	.66	1.18
	HS-10-1V	3/4-16	.334	.0876	46.62	9.54	5.46	2.20	.35	2.22	6.83	2.76	.44	1.77
	HS-15-1V	3/4-16	.334	.0876	73.30	15.00	5.46	1.40	.22	3.49	6.83	1.75	.28	2.79
	SS-.2A-1V	—	.125	.0123	.98	.20	.77	14.69	2.34	.33	.96	18.36	2.92	.27
	SS-.5A-1V -2V	3/8-24	.117	.0107	2.20 1.57	.45 .90	.67	5.73 2.87	.91 .47	.85 .61	.84	7.16 3.59	1.14 .57	.68 .47

ABBREVIATIONS GPM - GALLONS PER MINUTE FPS - FEET PER SECOND RPS - REVOLUTIONS PER SECOND RAD/SEC - RADIANS PER SECOND

SINGLE VANE UNITS — GUIDE TO SIZE

Note: Having determined pressure and torque requirements for your applications, use this guide to size to assist in model selections. See appropriate catalog for complete data.

MODEL	SEE PAGES
MP	MP-1 — MP-4
SS	SS-1 — SS-4
HS	HS-1 — HS-4
Rotating	R-1 — R-4

Torque Inch Lbs																	
750000																	
700000																	
650000																	
600000																	
550000																	
500000																	
450000																	
400000																	
350000																	SS-130
300000																SS-130	
250000														SS-130	SS-130		
225000													SS-130				
200000												SS-130					
175000											SS-130						SS-65
150000																SS-65	
125000										SS-130				SS-65	SS-65		
100000									SS-130 MP-128			SS-65	SS-65				SS-40
90000															SS-40	SS-40	
80000								SS-130			SS-65			SS-40			
70000								MP-128	MP-116	SS-65			SS-40				
60000												SS-40				SS-25	SS-25
50000							SS-130	MP-116	SS-65		SS-40			SS-25	SS-25		
40000							MP-128	SS-65	MP-105	SS-40			SS-25				HS-15
30000							MP-116	MP-105	SS-40		SS-25	SS-25		HS-15	HS-15	HS-15 SS-12	SS-12
20000					SS-130	SS-130	SS-65	SS-40	SS-25 MP-84	SS-25	HS-15	HS-15	HS-15 SS-12	SS-12	HS-10 SS-12	HS-10	HS-10 SS-8
15000				SS-130	MP-116	MP-128	SS-40 MP-105	SS-25 MP-84		HS-15	SS-12	HS-10 SS-12	HS-10	SS-8 HS-10	SS-8	HS-6 SS-8	HS-6
12250						SS-65 MP-116			HS-15	SS-12	HS-10	SS-8	SS-8	HS-6	HS-6		
10000			SS-130	MP-128	SS-65		SS-25	HS-15	SS-12	HS-10	SS-8	HS-6					SS-4 HS-4
7500		SS-130		SS-65 MP-116		SS-40 MP-105	MP-84	SS-12	HS-10 MP-63	SS-8	HS-6			HS-4 SS-4	HS-4 SS-4	HS-4 SS-4	
5000	SS-130		SS-65 MP-128	SS-40 MP-105	SS-40 MP-105	SS-25	HS-15 SS-12	HS-10 MP-63 SS-8	SS-8 HS-6	HS-6	HS-4 SS-4	SS-4 HS-4	SS-4 HS-4	HS-2½	HS-2½	HS-2½	HS-2½

MID PRESSURE — UP TO 1500 PSI   HIGH PRESSURE — UP TO 3000 PSI

4500				SS-65	MP-116							HS-4			HS-2 1/2					
4000				MP-128			SS-25	MP-84	HS-10 MP-63	HS-6		SS-4							HS-1 1/2	
3500			SS-65		SS-40		MP-84		SS-8		HS-4							HS-1 1/2		
3000						SS-25		HS-15			SS-4 MP-34		HS-2 1/2	HS-2 1/2		HS-1 1/2	HS-1 1/2		SS-1	
2500				SS-40 MP-116	MP-105	MP-84	HS-15	SS-12	HS-6	SS-A HS-4		HS-2 1/2			HS-1 1/2		SS-1	SS-1		
2000			SS-40 MP-128		SS-25	HS-15	SS-12	HS-10		MP-34	HS-21/2			HS-1 1/2	SS-1	SS-1				
1750				SS-25 MP-105				SS-8 MP-63	HS-4					SS-1						
1500						SS-12	HS-10 MP-63		SS-4	HS-2 1/2	MP-32	HS-1 1/2	HS-1 1/2 SS-1							
1250			SS-25 MP-116		HS-15 MP-84	HS-10	SS-8	HS-6	MP-34		HS-1 1/2	SS-1								
1000				HS-15	SS-12	SS-8 MP-63	HS-6		HS-2 1/2	HS-1 1/2 MP-32	SS-1									
900			MP-105	MP-84				HS-4												
800			HS-15	SS-12	HS-10	HS-6		SS-4		SS-1						SS-5A				
700					SS-8		HS-4		MP-32						SS-5A					
600			SS-12	HS-10	MP-63		SS-4	MP-34	HS-1 1/2		MP-22			SS-5A						
500			HS-10	SS-8	HS-6	SS-4 HS-4	MP-34	HS-2 1/2	SS-1	MP-22			SS-5A							
400			SS-8 MP-84	HS-6			HS-2 1/2					SS-5A						SS-2A	SS-2A	
300			HS-6	MP-63	HS-4 SS-4	HS-2 1/2 MP-34		HS-1 1/2 MP-32	MP-22		SS-5A				SS-2A	SS-2A	SS-2A			
200			HS-4 SS-4	HS-4 SS-4	HS-2 1/2 MP-34	HS-1 1/2	SS-1 HS-1 1/2 MP-32	SS-1		SS-5A		SS-2A	SS-2A	SS-2A						
100			HS-2 1/2 MP-63	HS-1 1/2 MP-34	HS-1-1/2 SS-1 MP-32	SS-1 MP-32	MP-22	MP-22	SS-5A	SS-2A	MP-11 SS-2A									
80			HS-1 1/2	SS-1				SS-5A	SS-2A	MP-11										
60			SS-1 MP-34	MP-32		MP-22	SS-5A													
40					MP-22	SS-5A		SS-2A	MP-11											
35																				
30			MP-32				SS-2A													
25						SS-2A		MP-11												
20				MP-22			MP-11													
15						MP-11														
10																				
5			MP-22	MP-11	MP-11															
0			MP-11																	
PSI ➡	20	40	60	80	100	150	200	250	500	750	1000	1250	1500	1750	2000	2250	2500	2750	3000	

MID PRESSURE UP TO 1500 PSI **HIGH PRESSURE** UP TO 3000 PSI

DOUBLE VANE UNITS — GUIDE TO SIZE

Note: Having determined pressure and torque requirements for your applications, use this guide to size to assist in model selections. See appropriate catalog for complete data.

MODEL	SEE PAGES
MP	MP-1 — MP-4
SS	SS-1 — SS-4
HS	HS-1 — HS-4
Rotating	R-1 — R-4

Torque Inch Lbs																				
750000																				
700000																			SS-130	
650000																		SS-130		
600000																	SS-130			
550000																SS-130				
500000																				
450000															SS-130					
400000														SS-130						
350000													SS-130						SS-65	
300000												SS-130					SS-65	SS-65		
250000											SS-130					SS-65				
225000															SS-65				SS-40	
200000											MP-128			S-65				SS-40		
175000										SS-130			SS-65							
150000										MP-128	MP-116	SS-65			SS-40	SS-40	SS-40			
125000														SS-40						
100000									SS-130	MP-116	SS-65		SS-40							
90000									MP-128	SS-65	MP-105	SS-40								
80000																				
70000											SS-40								SS-12	
60000								SS-130	SS-65 MP-116		MP-105							SS-12		
50000										SS-40						SS-12	SS-12			
40000							SS-130	MP-128	MP-105		MP-84			SS-12	SS-12					
30000						SS-130	MP-128	SS-65 MP-116	SS-40	MP-84			SS-12							
20000					SS-130	MP-128	SS-65 MP-116		MP-84		SS-12	SS-12							SS-4	
15000				SS-130		SS-65 MP-116	SS-40 MP-105	SS-40 MP-105		SS-12	MP-63					SS-4	SS-4	SS-4		
12250			SS-130		SS-65 MP-128					MP-63				SS-4	SS-4					
10000					MP-116	SS-40 MP-105			SS-12				SS-4							
7500				MP-128	SS-40		MP-84	MP-84	MP-63			SS-4								
5000			SS-65	SS-40 MP-116	MP-105	MP-84		SS-12		SS-4	SS-4 MP-34					SS-1	SS-1	SS-1	SS-1	

MID PRESSURE (light shading) **UP TO 1500 PSI** **HIGH PRESSURE** (dark shading) **UP TO 3000 PSI**

PSI ➡	20	40	60	80	100	150	200	250	500	750	1000	1250	1500	1750	2000	2250	2500	2750	3000
4500			SS-40 MP-128				SS-8			MP-34					SS-1				
4000																			
3500				MP-105		SS-12		MP-63	SS-4					SS-1					
3000			MP-116		MP-84		MP-63		MP-34		MP-32		SS-1						
2500												SS-1							
2000					SS-12	MP-63				MP-32	SS-1								
1750			MP-105	SS-12 MP-84				SS-4								SS-5A			
1500									MP-32	SS-1					SS-5A				
1250			SS-12		MP-63		SS-4	MP-34			MP-22			SS-5A					
1000						SS-4	MP-34		SS-1	MP-22		SS-5A	SS-5A						
900			MP-84																
800				MP-63							SS-5A								
700					SS-4	MP-34													
600								MP-32	MP-32	SS-5A									
500				SS-4			MP-32	SS-1											
400			SS-4 MP-63		MP-34		SS-1		SS-5A										
300						SS-1 MP-32		MP-22			MP-11								
200				MP-34	SS-1 MP-32		MP-22	SS-5A		MP-11									
100			SS-1 MP-34	SS-1 MP-32		SS-5A MP-22	SS-5A		MP-11										
80																			
60			MP-32		MP-22			MP-11											
40				MP-22			MP-11												
35																			
30						MP-11													
25																			
20					MP-11														
15																			
10			MP-22																
5				MP-11															
0			MP-11																

MID PRESSURE UP TO 1500 PSI **HIGH PRESSURE** UP TO 3000 PSI

NOTES