



216 Rotary Actuators

Heavy duty, torque generating devices designed
for precise servo-control applications

THE **216 ROTARY ACTUATORS** ARE HEAVY-DUTY TORQUE GENERATION DEVICES THAT OPERATE UNDER PRECISION SERVOVALVE CONTROL. WHEN MATCHED WITH MTS SERVOVALVES, TRANSDUCERS, AND CLOSED-LOOP CONTROL ELECTRONICS, THEY ARE IDEALLY SUITED FOR TESTING MATERIALS, STRUCTURES, AND COMPONENTS.



Description

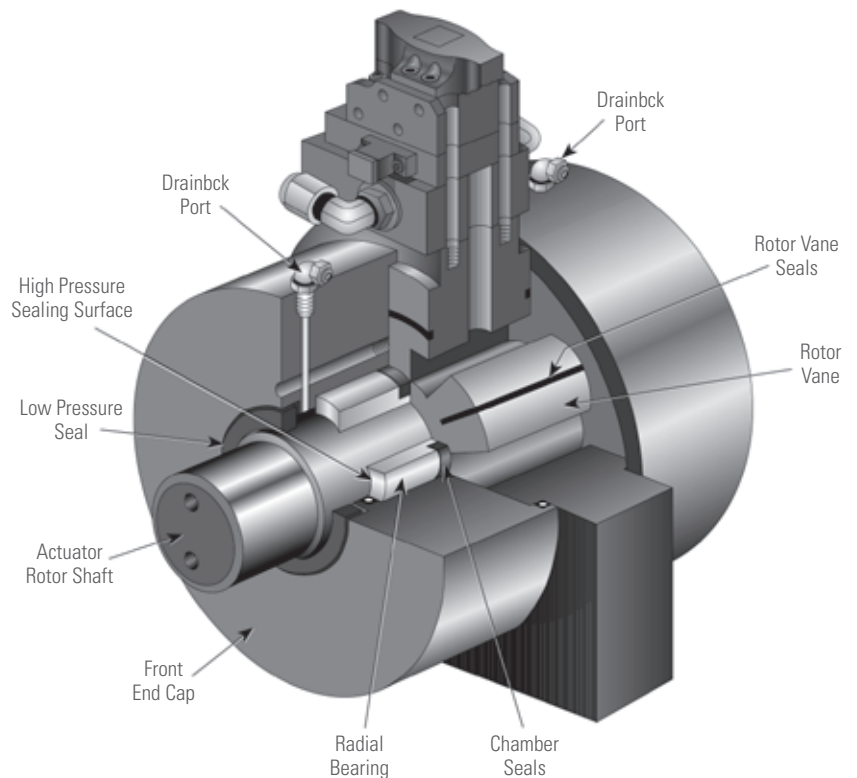
The double-vane rotor design accommodates full force application through 100° of rotation. Matched reaction brackets, diaphragm flexures, and reaction bases allow you to create versatile test configurations that you can easily reconfigure. Available instrumentation options include angular displacement transducers (ADT), torque cells, and differential pressure cells (ΔP).

Features

- » Actuators are available in force ratings from 200,000 to 730,000 lbf-in. (22 597 to 82 479 N-m)
- » Heavy-duty bearings accommodate significant radial and axial loads
- » Precision chamber and rotor design ensures zero actuator backlash during torque reversals
- » Available with a complete line of fixturing and angular displacement transducer options
- » Servovalve ports are compatible with standard MTS servovalve manifolds
- » Balanced double vane rotor design reduces friction and provides continuous torque output through 100° of rotation
- » Separate actuator mounting and fixture attachment equipment accommodate diverse test articles and configurations
- » Diaphragm flexures are available to reduce excessive thrust and side loads exerted on the test specimen

Components

- » **Chamber Seals** – Cap seals between the actuator cylinder and end caps prevent hydraulic fluid from flowing around the rotor vane seals.
- » **Radial Bearing** – Precision bearing designed to tolerate the high radial and axial forces that can be encountered during testing.
- » **Front End Cap** – Provides a modular carrier for the actuator seals, bearings, and drainback ports.
- » **Actuator Rotor Shaft** – Machined steel alloy shaft with replaceable rotor vanes.
- » **High Pressure Seal** – Low friction seal design prevents high pressure hydraulic fluid from flowing out of the cylinder.
- » **Rotor Vane** – Bolt on vane design equipped with non-metallic seals.
- » **Rotor Vane Seals** – Replaceable seals ensure a minimum of leakage between chambers.
- » **Low Pressure Seal** – U-shaped seal/wiper ring prevents external leakage and protects against internal contamination.
- » **Drainback Port** – Collection port for fluid allowed to flow through the high pressure seals and bearing. Fluid is returned to the hydraulic power supply.



216 Rotary Actuator Cut-Away View

Fixturing Options

There are a variety of options for the Series 216 Rotary Actuators. The following photograph shows a typical test system containing a rotary actuator and the available optional components. Each of the options is described below.

Reaction Base

A reaction base is used with the rotary actuator for two purposes: (1) the machined flat surface and edge ensure accurate alignment at any actuator position; (2) it provides a rigid structure which can react the high torques generated by the rotary actuator.

Flange Adapter

The flange adapter (located behind the diaphragm flexure in the photograph) is secured to the actuator rotor shaft by two concentric lock ring assemblies. It provides a coupling surface between the actuator and the specimen or diaphragm flexure.

Diaphragm Flexures

Diaphragm flexures are used at both ends of the specimen if large axial and angular deflections are generated during testing. If reaction forces exceed stated actuator operating limits, diaphragm flexures help reduce the thrust and side loads reacted by the actuator and test specimen.

Reaction Bracket

Reaction brackets attach securely to the reaction base or T-slot table and provide a mounting surface for the torque cell. Each reaction bracket is designed to restrain a specific model torque cell.

Torque Cell

A torque cell provides a precise electrical feedback signal that is proportional to the torque applied to the specimen. For more information on MTS torque cells, refer to the appropriate MTS product specification.

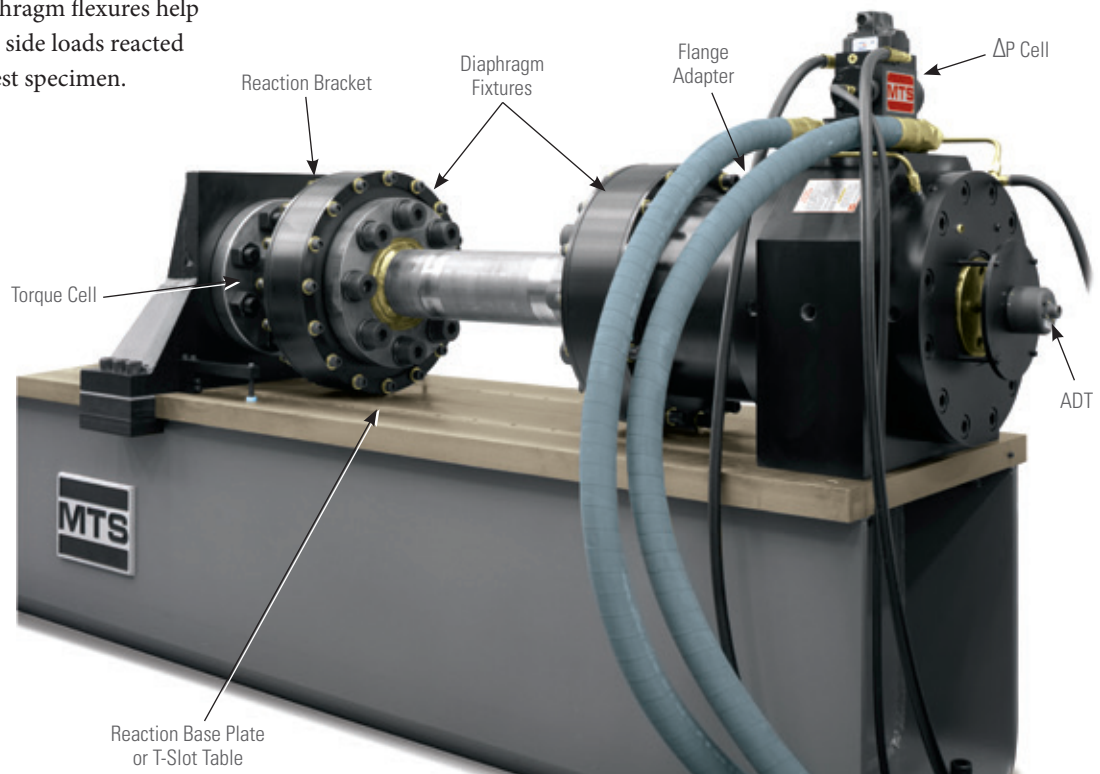
ADT

An angular displacement transducer (ADT) connected to the rear shaft of the actuator produces a DC electrical signal that is proportional to the angular

position of the actuator. Rotation of the actuator will generate a feedback signal from the ADT to a transducer conditioner. Rotation is continuous with no reactive torque induced. The ADT is a precision differential capacitor coupled to a solid state oscillator, demodulator, and amplifier to yield DC input/DC output performance.

Differential Pressure Cell

The differential pressure (ΔP) cell is a single-unit, dual port, bonded strain gage pressure sensor. Depending on the specific application, the ΔP cell is used to stabilize or control actuator force output. The ΔP cell (located beneath the servovalve) provides a feedback signal to monitor fluid pressure within the actuator housing. For more information on MTS ΔP cells refer to the appropriate MTS product specification.



Specifications

The 216 Rotary Actuators are available in three models. Table 1 lists the actuator performance characteristics by model number. Table 2 (following page) lists the specifications for the basic actuator assembly shown in the accompanying

figure. Subsequent tables list the specifications for the actuator options. The letters in parenthesis indicate values used in calculations performed in the *Testing Considerations* section (page 11) of this specification.

Table 1. Actuator Specifications

Model	Dynamic* Torque Rating		Displacement	
	lbf-in.	N-m	in. ³ /rad	cm ³ /rad
216.10	200,000	22,600	70	1,147
216.20	397,000	44,900	137	2,245
216.30	730,000	82,500	252	4,130

Model	Maximum Thrust Load*		Maximum Load on Front Bearing (W)		Maximum Side Load† (P)	
	kip	kN	kip	kN	kip	kN
216.10	3.8	16.9	12.7	56.5	5.7	25.4
216.20	7.2	32.0	17.6	78.3	7.7	34.3
216.30	15.0	66.7	28.8	128.1	13.1	58.3

Model	Maximum Velocity Into Vane Stops ^{††}		Rotational Inertia of Actuator		Dynamic Seal Friction ^{††}	
	U.S. Customary rad/sec	Metric rad/sec	lbm-in. ²	kg-m ²	lbf-in.	N-m
216.10	$w = \frac{867}{\sqrt{J}}$	$w = \frac{1486}{\sqrt{J}}$	725	0.212	2,500	282.5
216.20	$w = \frac{1166}{\sqrt{J}}$	$w = \frac{19.95}{\sqrt{J}}$	2,180	0.637	4,500	508.4
216.30	$w = \frac{1580}{\sqrt{J}}$	$w = \frac{27.02}{\sqrt{J}}$	6,350	1.858	8,500	960.4

* Structural fatigue rating. Rated at differential pressure of 2900 psi (203 kg/cm²).

† Applied at end of output shaft and (M) equal to 0.0 lbf-in.

†† If values are excessive, additional internal or external cushions are required. Contact MTS Systems Corporation.

‡ Where w = rotational velocity in rad/sec and J or I = rotational inertia in lbm-in.² or kg-m² including inertia from rotary actuator, flange, flexure, and 1/2 of test specimen.

‡‡ Average test data at 1500 psi (10.3 MPa) strut pressure.

Specifications subject to change. Contact MTS for specifications critical to your needs.

Table 2. Actuator Dimensions

Model	Weight		A		B		C	
	lb	kg	in.	mm	in.	mm	in.	mm
216.10	750	340	5.118	130.0	14.00	356	16.00	406
216.20	1300	590	6.299	160.0	17.75	451	20.00	508
216.30	2700	1230	7.874	200.0	22.00	559	27.00	686

Model	D		E		F		G	
	in.	mm	in.	mm	in.	mm	in.	mm
216.10	6.86	174	2.62	76	6.50	165	12.00	304.8
216.20	7.55	186	2.87	82	8.00	203	16.00	406.4
216.30	8.26	210	4.14	105	8.50	216	23.00	584.2

Model	H		I	J		s		u*	
	in.	mm		in.	mm	in.	mm	in.	mm
216.10	4.50	114.3	3/4 - 10	2.00	50.8	7.35	186.7	9.12	231.6
216.20	5.00	127.0	1-8	2.25	57.2	8.30	210.8	10.75	273.0
216.30	6.00	152.4	1-8	2.25	57.2	10.00	254.0	12.00	304.8

*Not including specimen adapter.

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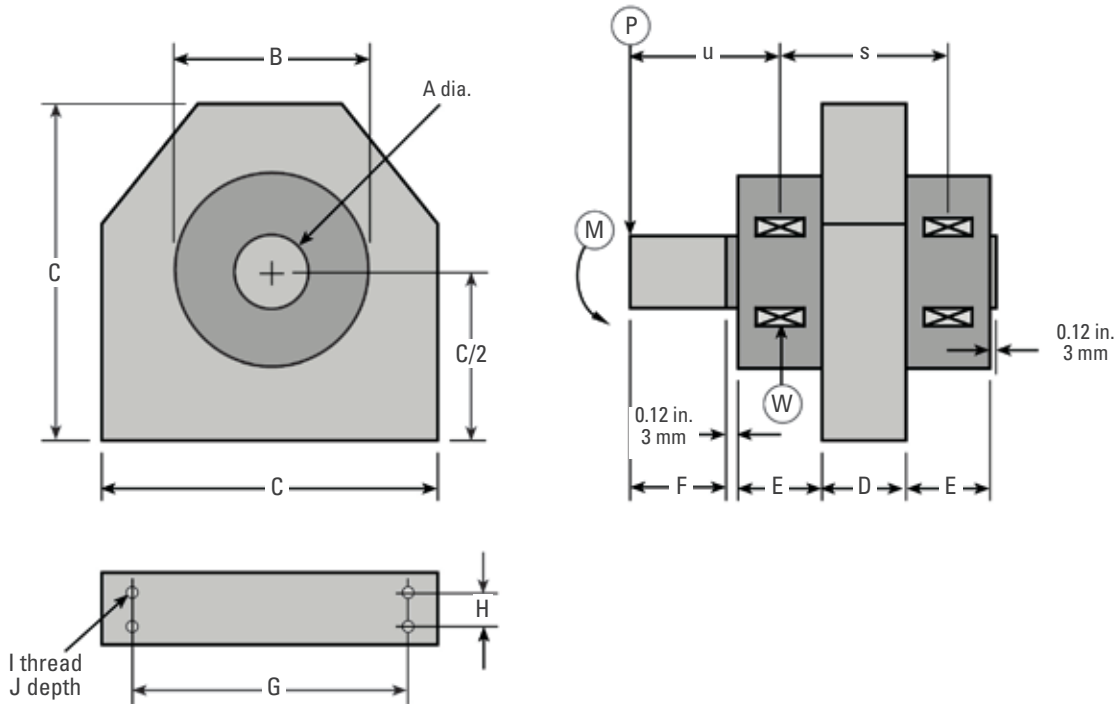


Table 3. Diaphragm Flexure Dimensions and Ratings

Model	A		B		C		D	
	lb	kg	in.	mm	in.	mm	in.	mm
216.10	13.00	330.2	14.25	361.9	15.75	400.0	4.05	102.9
216.20	15.00	381.0	17.25	438.1	19.75	501.6	5.22	132.6
216.30	17.00	431.8	21.25	539.7	23.75	603.2	5.46	138.7

Model	E		F	G		Max Thrust Load*	
	in.	mm	in.-Thd	in.	mm	lbf	N
216.10	0.80	20.3	1-8	11.00	279.4	1,700	7,500
216.20	1.01	25.6	1 1/2 - 6	12.00	304.8	2,000	8,900
216.30	1.01	25.6	1 1/2 - 6	14.00	355.6	1,800	8,000

Model	Maximum Deflection		Maximum Bending Moment* (M_{F1})	Maximum Angular Deflection. (θ_{F1})		Rotational Inertia	
	in.	mm	lbf-in.	Nm	Radians	lbm-in ²	kg-m ²
216.10	0.05	1.3	5,000	560	0.002	3,338	0.98
216.20	0.13	3.3	6,600	750	0.005	10,296	3.01
216.30	0.39	9.9	8,000	900	0.017	21,112	6.18

* Thrust load and bending moment are interdependent. If bending moment = 75% of rating, bending moment must not exceed 25% of rating, etc.

Specifications subject to change. Contact MTS for specifications critical to your needs.

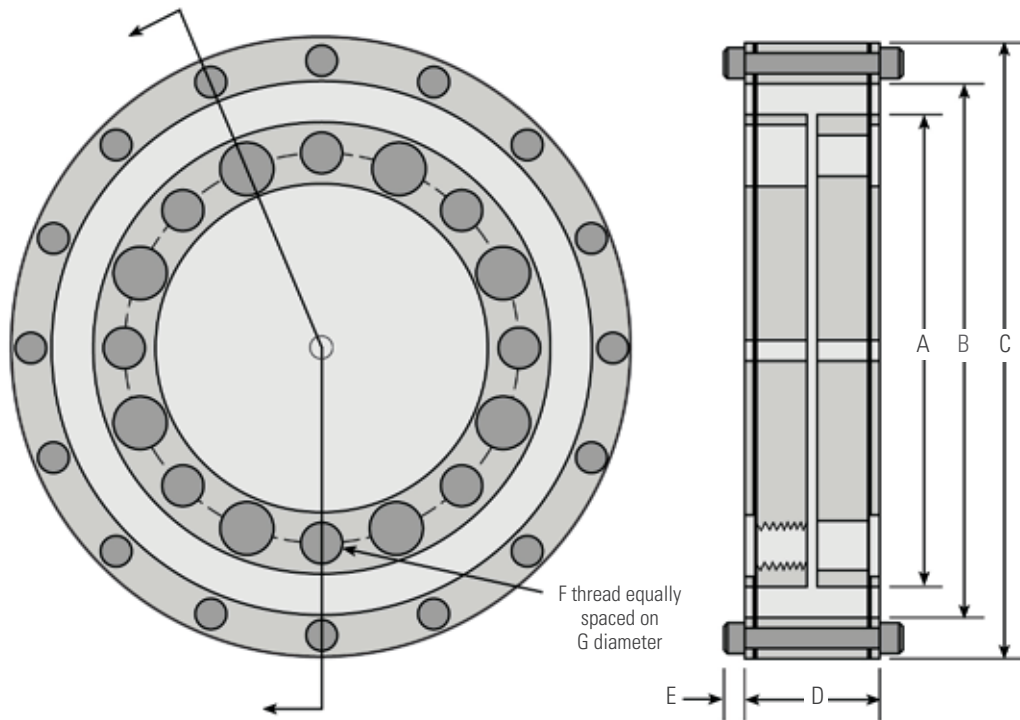


Table 4. Flange Adapter Dimensions and Ratings

Model	A		B		C		D	E	
	in.	mm	in.	mm	in.	mm	in.-Thd	in.	mm
216.10	5.118	130.0	14.00	355.6	6.25	158.7	1-8	3.25	82.5
216.20	6.299	158.2	15.00	381.0	7.62	193.5	1 1/2 - 6	4.00	101.6
216.30	7.874	200.0	17.00	431.8	8.25	209.5	1 1/2 - 6	4.00	101.6

Model	F		G		H		Rotational Inertia	
	in.	mm	in.	mm	in.	mm	lbm-in. ²	kg-m ²
216.10	1.031	26.2	1.00	25	11.00	279.4	6,550	1.92
216.20	1.531	38.9	1.00	25	12.00	304.8	10,400	3.04
216.30	1.531	38.9	1.00	25	14.00	355.6	18,300	5.35

Specifications subject to change. Contact MTS for specifications critical to your needs.

Table 5. Reaction Base Dimensions and Ratings

Model	Length		Width		Height		Max Space*		Weight		Torsional Stiffness† (K _t)	
	in.	mm	in.	mm	in.	mm	in.	mm	lb	kg	lbf-in./rad	N-m/rad
216.10	120	3048	24	610	26	660	75.3	1913	2000	1200	0.66 x 10 ⁹	0.075 x 10 ⁹
216.20	120	3048	27	686	29	737	67.5	1714	3600	1600	1.40 x 10 ⁹	0.160 x 10 ⁹
216.30	120	3048	32	813	34	864	60.5	1537	5700	2600	2.80 x 10 ⁹	0.320 x 10 ⁹

* Maximum space between mounting surfaces of actuator output flange and torque cell (with the MTS reaction bracket upporting the torque cell). (excludes diaphragm flexures.)

† Torsional stiffness over entire length. Stiffness increases proportionately as the actuator and reaction bracket are moved toward each other.

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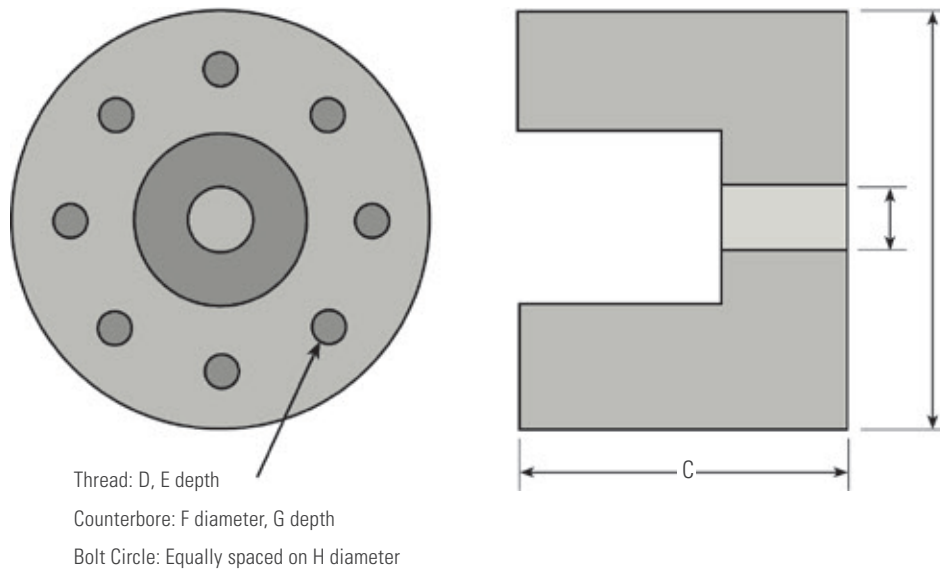


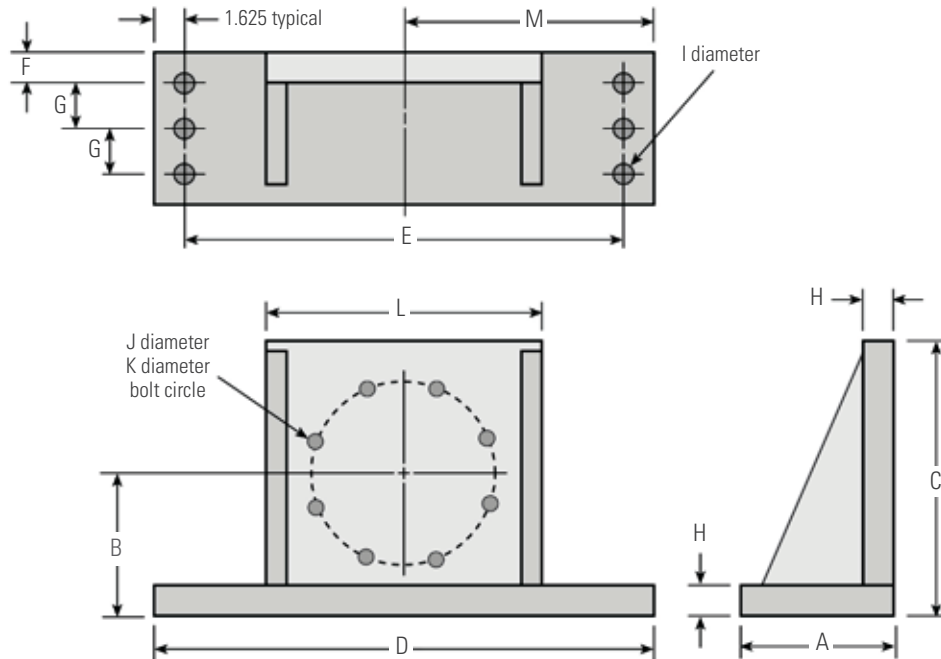
Table 6. Reaction Bracket Dimensions

Model	A		B		C		D	
	lb	kg	in.	mm	in.	mm	in.	mm
216.10	7.00	177.8	8.00	203.2	16.00	406.4	29.25	742.9
216.20	8.00	203.2	10.00	254.0	20.00	508.0	32.25	819.1
216.30	9.00	228.6	13.50	342.9	27.00	685.8	37.25	946.1

Model	E		F		G		H	
	lb	kg	in.	mm	in.	mm	in.	mm
216.10	26.00	660.4	1.50	38.1	2.00	50.8	1.50	38.1
216.20	29.00	736.6	2.00	50.8	2.00	50.8	2.00	50.8
216.30	34.00	863.6	2.50	63.5	2.00	50.8	2.00	50.8

Model	I		J		K		L		M	
	lb	kg	in.	mm	in.	mm	in.	mm	in.	mm
216.10	1.031	26.2	1.062	27.0	11.00	279.4	16.00	406.4	14.50	368.3
216.20	1.031	26.2	1.562	39.7	12.00	304.8	18.00	457.2	16.00	406.4
216.30	1.031	26.2	1.516	38.5	14.00	355.6	26.00	660.4	18.50	469.9

Specifications subject to change. Contact MTS for specifications critical to your needs.



Testing Considerations

This section describes factors that must be considered when using the Series 216 Rotary Actuators in a test system. It includes sample calculations that should help you determine the constraints on your planned testing scenario.

The thrust and side loads that may be encountered during testing are generally the result of the following factors:

- » Specimen shortening or lengthening due to torsional force
- » Specimen shortening or lengthening due to temperature
- » Misalignment of the test specimen when initially mounted
- » Reaction base or T-slot table twisting
- » Permanent deformation of the specimen due to torsional force

Thrust loads

Table 1 lists the maximum allowable thrust load that can be applied to the actuator rotor shaft. Because thrust loads can be induced by a wide variety of experimental conditions, this specification will not

attempt to define or predict the forces that can result from specific testing situations. If there is a possibility that the maximum thrust load rating of the actuator will be exceeded during testing, steps should be taken to minimize the load. One way of reducing the effect of thrust loads on the actuator bearings is to install diaphragm flexures.

As an example of thrust loads (which are a function of specimen geometry and material), increase the temperature of a steel shaft 1 in. (25.4 mm) in diameter and 50 in. (1,270 mm) in length by 40°F (22°C). The increase in specimen temperature causes the shaft to expand by approximately 0.012 in. (0.305 mm). If the shaft is mounted in a force train using a 216 Rotary Actuator, the shaft expansion would exert a resultant force of 6,000 lbs. on the actuator bearings. To confine the resultant force to an acceptable maximum requires the addition of flexure diaphragms to the force train.

Use the values from Table 3 to complete the following formula:

$$\frac{\text{Maximum thrust load}}{\text{Maximum deflection (of flexure)}} = \text{Flexure stiffness}$$

Multiply the flexure stiffness by the amount of specimen expansion to determine the thrust load imposed on the actuator bearings. Ensure that the resultant trust load does not exceed the maximum allowable thrust load listed in Table 1.

Side loads

Side loads, which are normally induced by specimen misalignment and/or reaction base or T-slot table compliance, may be active at the same time thrust loads are active. If the specimen is soft, such as a length of rubber hose, side loads are relatively small. This is because the specimen bends easily and exerts little resistance to the deflection caused by reaction base twisting.

However, if the specimen is stiffer, then the increased resistance of the specimen to bending exerts substantial side loads on the specimen, actuator bearings and torque cell due to the restraining characteristics of the test setup. As in the test setup for thrust loads, diaphragm flexures can be used to reduce the side loads to a practical limit.

Definition of Mathematical Terms

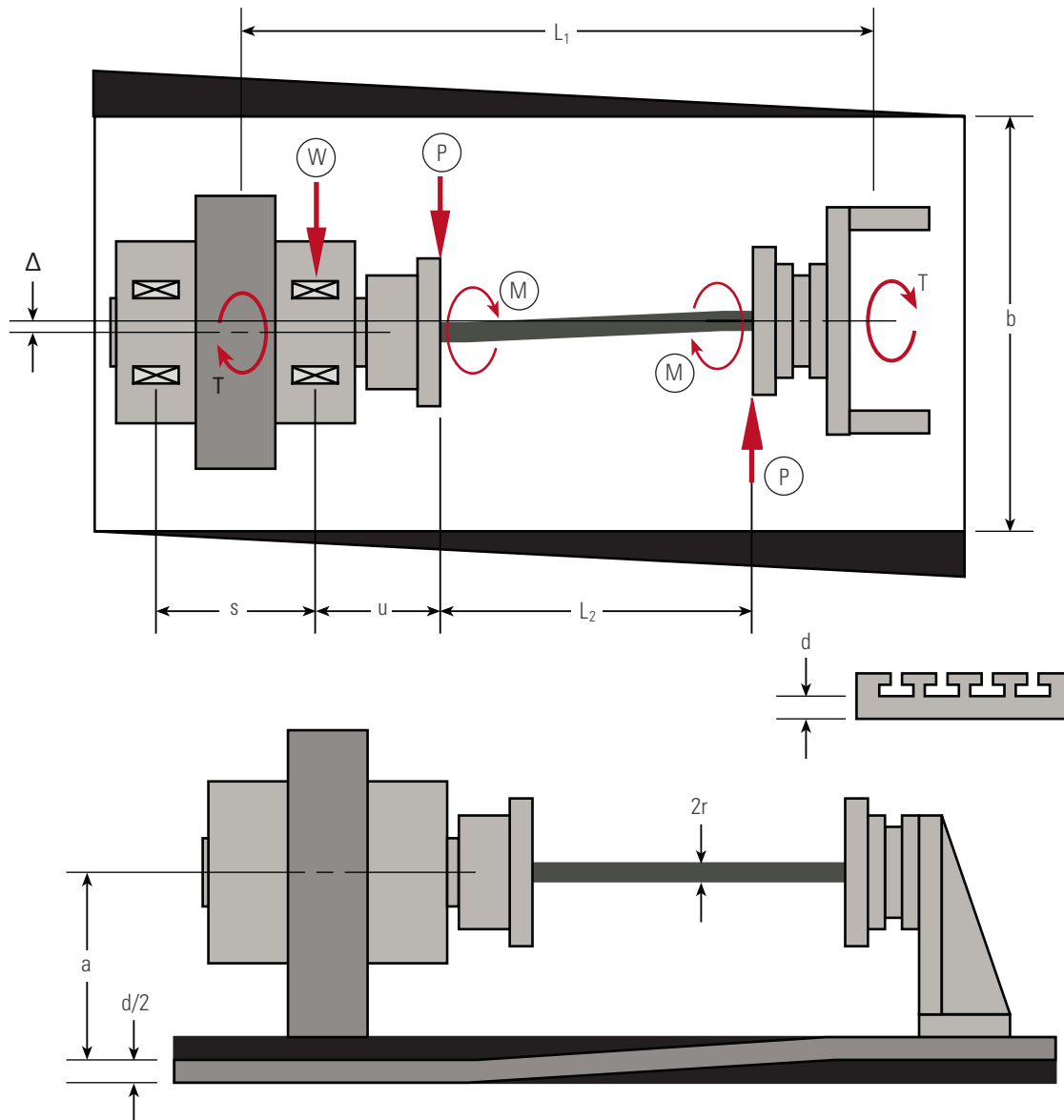
This section lists and defines the mathematical terms that are required to complete the sample calculations. The terms are listed in alphabetical order and defined in both U.S. Customary and SI Metric units of measure.

Table 7. Mathematical Terms

Term	Definition
a	= Distance from actuator's center line to center of reaction base's solid height (in.) (mm).
b	= Width of reaction base (in.) (mm).
d	= Thickness of reaction base (in.) (mm). Measurement of solid metal only. Do not include T-slot depth.
E_s	= Modulus of elasticity of the reaction base or T-slot table, shear (lb/in. ²) (N/m ²).
E	= Modulus of elasticity of the specimen, tension (lb/in. ²) (N/m ²).
I	= Moment of inertia ($\pi r^4/4$) for a round solid (in. ⁴) (mm ⁴).
K_1	= Torsional stiffness of a thin flat plate (lbf-in./rad) (N-m/rad).
K_2	= Lateral stiffness of a solid cylindrical specimen (lbf/in.) (kN/mm).
K_{F_1}	= Angular horizontal stiffness of actuator and reaction bracket (lbf-in./rad) (N-m/rad).
K_{F_2}	= Lateral stiffness of diaphragm flexures (lbf-in./rad) (N-m/rad).
L_1	= Length of reaction base or T-slot table subjected to twisting (in.) (mm).
L_2	= Length of test specimen (in.) (mm). <i>Do not include specimen adapter plates unless their compliance is equal to or greater than that of the specimen.</i>
M	= Bending moment on test specimen (lbf-in.) (N-m).
M_1	= Bending moment on test specimen fitted with flexure diaphragms (lbf-in.) (N-m).
M_{F_1}	= Maximum horizontal bending capacity of flexures (lbf-in.) (N-m).
P	= Side load imposed on test specimen and actuator (lbf-in.) (N-m).
r	= Radius of test specimen (in.) (mm).
s	= Distance between front and rear bearings (in.) (mm).
S_B	= Bending stress on test specimen due to reaction base twisting (psi) (N/m ²).
T	= Applied torque (lbf-in.) (N-m).
u	= Distance from front bearing to specimen (in.) (mm). Include specimen adapter plates if they are less compliant than the specimen.
W	= Load on front actuator bearing (lbf) (kN).
Δ	= Centerline offset between actuator and reaction bracket mountings due to twisting of reaction base or T-slot table (in.) (mm).
θ	= Angle of flex imposed on actuator and reaction bracket flexure (radians).
θ_{F_1}	= Maximum horizontal angular deflection of flexures (radians).

Side Load Calculations when Excluding Flexures

If diaphragm flexures will not be used in the rotary actuator test system, special attention should be paid to the side loads that will be imposed on the specimen and actuator by twisting of the reaction base or T-slot table. The following side load calculation procedure is used to determine side loads due to the reaction base or T-slot table torsional compliance. When side loads are unacceptable as determined from the procedure, optional components are required in the force train to reduce the load imposed on the actuator and torque sensor.



Loads on an Actuator and Specimen due to Reaction Base Twist (Excluding Thrust Loads)

Sample Calculation

The preceding figure illustrates the forces and measurements pertinent to the calculations. Refer to Tables 1 and 2 for dimensions and ratings of the Model 216.10 Rotary Actuator used in the example.

The following procedure uses sample values. When performing the calculations to determine the anticipated test forces, the values appropriate to your specific test should be substituted for the sample values. In addition, the example uses U.S. Customary units of measure. The equivalent SI metric units of measure are defined in Table 7.

The following values are supplied for the sample calculations:

- a = 11 in. (*distance from actuator center line to reaction base center*)
- b = 20 in. (*width of reaction base*)
- d = 6 in. (*thickness of reaction base*)
- L₁ = 51 in. (*length of reaction base subjected to twisting*)
- L₂ = 10 in. (*length of test specimen and specimen adapter plates*)
- r = 1.75 in. (*radius of test specimen*)
- s = 7.35 in. (*distance between front and rear actuator bearings*)
- T = 150,000 lbf-in. (*applied torque*)
- u = 9.12 in. (*distance from front actuator bearing to specimen*)
- E_s = 12 x 10⁶ lb/in.² (*modulus of elasticity of the reaction base or T-slot table*)

1. Calculate the side load (P) imposed on the test specimen and actuator bearing as a result of reaction base twist using the following formula:

$$P = \frac{\frac{K_2}{K_1} aT}{1 + \frac{K_2}{K_1} a^2}$$

The values of K₁ and K₂ can be found using the following equations:

A. If you are using an MTS reaction base, select the appropriate value for K₁ from Table 5. Otherwise, calculate the value of K₁, the torsional stiffness of a thin flat plate, by using the following formula*:

$$K_1 = \frac{ES \left[(0.333 - (0.21 \times \frac{d}{b})) \right] bd^3}{L_1}$$

$$\begin{aligned} \text{then: } K_1 &= \frac{12 \times 10^6 \text{ lb/in.}^2 \left[(0.333 - (0.21 \times \frac{6}{20})) \right] (20 \text{ in.}) (6 \text{ in.})^3}{51 \text{ in.}} \\ &= 274.4 \times 10^6 \frac{\text{lbf-in.}}{\text{rad}} \end{aligned}$$

* In the formula, $[0.333 - (0.21 \times \frac{d}{b})]$ is used in place of J (polar momentary inertia) due to warpage that occurs in thin flat plates under torque.

B. Calculate the value of K₂, the lateral stiffness of a solid cylindrical specimen, by using the formula:

$$K_2 = \frac{12 EI}{L_2^3}$$

where: E = 29 x 10⁶ lb/in.²

$$I = \frac{\pi r^4}{4}$$

r = 1.75 in.

L₂ = 10 in.

$$\begin{aligned} \text{then: } K_2 &= \frac{12 (29 \times 10^6 \text{ lb/in.}^2) \frac{\pi (1.75)^4}{4}}{(10)^3} \\ &= 2.563 \times 10^6 \text{ lbf/in.} \end{aligned}$$

C. Substitute the calculated values for K₁, K₂, and the example values into the original equation to compute the side load (P).

$$P = \frac{\frac{K_2}{K_1} aT}{1 + \frac{K_2}{K_1} a^2}$$

where: a = 11 in.

T = 150,000 lbf-in.

$$\begin{aligned} \text{then: } P &= \frac{\frac{2.563 \times 10^6 \text{ lbf-in.}}{274.4 \times 10^6 \text{ lbf-in./rad}} (11 \text{ in.}) (150,000 \text{ lbf-in.})}{1 + \frac{2.563 \times 10^6 \text{ lbf-in.}}{274.4 \times 10^6 \text{ lbf-in./rad}} (11 \text{ in.})^2} \\ &= \frac{15411.625}{2.1301859} = 7,234.873 \text{ lbf} \end{aligned}$$

The value of 7,234.873 lbf is the side load (P) imposed on the test specimen and actuator by reaction base twist.

2. Calculate the bending moment (M) on the test specimen by using the following formula:

$$M = \frac{PL_2}{2}$$

$$\begin{aligned} \text{where: } M &= \frac{(7,234.873 \text{ lbf}) (10 \text{ in.})}{2} \\ &= 36,174.365 \text{ lbf-in.} \end{aligned}$$

The value of 36,174.365 lbf-in. is the bending moment exerted on the actuator shaft and specimen.

3. Calculate the load on the front actuator bearing (W) by using the following formula:

$$W = P \left\{ 1 + \frac{u}{s} \right\} + \frac{M}{s}$$

Substitute the calculated value for P , M , and the example constants into the equation and compute the load on the front actuator bearing (W) given:

$$s = 7.35 \text{ in.}$$

$$u = 9.12 \text{ in.}$$

$$W = 7,234.873 \text{ lbf} \left(1 + \frac{9.12 \text{ in.}}{7.35 \text{ in.}} \right) + \frac{36,174.365 \text{ in.}}{7.35 \text{ in.}}$$

$$s = 21,133 \text{ lbf}$$

4. Compare the result of step 3 with the specific model actuator bearing capacity in Table 1.

In the sample problem, the actuator bearing capacity of a Model 216.10 Rotary Actuator is 12,700 lbf. The calculated force on the actuator bearing is 21,133 lbf. This force is significantly greater than the actuator bearing capacity. Operating the actuator with such a large force applied to the bearing causes additional wear of the actuator, reduces its service life, and can cause premature failure of the test specimen due to the additional stress.

5. Calculate the stress (S_B) induced in the specimen due to reaction base twist by using the following formula:

$$S_B = \frac{Mr}{I}$$

where: $I = \frac{\pi r^4}{4}$

$$= \frac{\pi(1.75)^4}{4} = 7.366 \text{ in.}^4$$

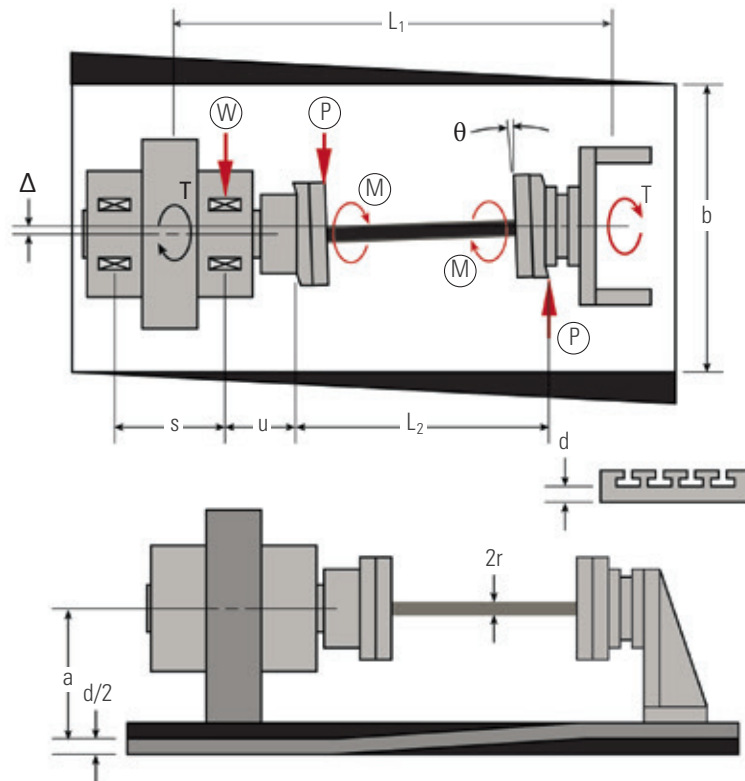
then: $S_B = \frac{36,174.365 \text{ lbf-in.} (1.75 \text{ in.})}{7.366 \text{ in.}^4}$

$$= 8,594.24 \text{ psi}$$

The value, 8,594.24 psi represents the amount of stress experienced by the specimen due to reaction base twist. In a torsion test, stress caused by reaction base or T-slot table twist should typically be zero or as close to zero as possible. In the sample calculation, the excessive stress imposed on the test specimen can invalidate the test results or cause premature failure of the specimen. To reduce these loads requires the use of diaphragm flexures or a stiffer mounting surface.

Side Load Calculations when Using Flexures

If the values derived from the calculations in the previous example indicate that diaphragm flexures must be used to reduce side loads to acceptable levels, then the following calculations should be completed to ensure that the selected diaphragm flexures are adequate. In addition, this subsection contains the equations necessary for calculating the stress experienced by the specimen when diaphragm flexures are installed in the test system.



Force Train with Diaphragm Flexures Mounted at Both Ends of Test Specimen

Sample Calculation

The following procedure uses the values derived from the previous sample calculations. The preceding figure illustrates the forces and measurements pertinent to the calculations. Refer to Tables 1 and 2 for dimensions and ratings of the Model 216.10 Rotary Actuator used in the example.

The following sample values will be used in the equations:

$a = 11$ in. (*distance from actuator center line to reaction base center*)

$K_1 = 274.4 \times 10^6$ lbf-in./rad (*torsional stiffness of thin flat plate*)

$L_1 = 51$ in. (*length of reaction base subjected to twisting*)

$L_2 = 18.1$ in. (*length of specimen, adapter plates, and flexures*)

$M_{F_1} = 15,000$ lbf-in. (*maximum horizontal flexure bending capacity*)

$\theta_{F_1} = 0.002$ rad (*maximum horizontal flexure angular deflection*)

$s = 7.35$ in. (*distance between front and rear actuator bearings*)

$T = 150,000$ lbf-in. (*applied torque*)

$u = 9.12$ in. (*distance from front actuator bearing to specimen*)

1. Calculate the center line offset (Δ) between actuator and reaction bracket due to reaction base twist by using the following formula:

$$\Delta = \frac{Ta}{K_1}$$

where: $a = 11$ in.

$$K_1 = 274.4 \times 10^6 \frac{\text{lbf-in.}}{\text{rad}}$$

$$T = 150,000 \text{ lbf-in.}$$

then:
$$\Delta = \frac{(150,000)(11 \text{ in.})}{274.4 \times 10^6}$$

$$= 0.00601 \text{ in.}$$

2. Calculate the angle of flex (θ) imposed on each diaphragm flexure by using the following formula:

$$\theta = \frac{\Delta}{L_2}$$

where: $L_2 = 18.1$ in.

then:
$$\theta = \frac{0.00601 \text{ in.}}{18.1 \text{ in.}}$$

$$= 0.000332 \text{ rad}$$

3. Compare the maximum horizontal angular deflection of the diaphragm flexures ($\theta_{F_1} = 0.002$ rad) with the calculated angle of flex imposed on one diaphragm flexure ($\theta = 0.000601$ rad) to determine if the flexures are adequate.

$$\theta < \theta_{F_1}$$

In the case of the sample calculation, the flexures are adequate. If the flexures are not adequate, additional flexural capability is required or reaction base stiffness must be increased. Consult MTS Systems for assistance.

4. Calculate the lateral stiffness (K_{F_1}) of the diaphragm flexures by using the following formula:

$$K_{F_1} = \frac{M_{F_1}}{\theta_{F_1}}$$

where: $M_{F_2} = 5,000$ lbf-in.
 $\theta_{F_1} = 0.002$ rad

then:
$$K_{F_1} = \frac{5,000 \text{ lbf-in.}}{0.002 \text{ rad}}$$

$$= 2.5 \text{ rad} \times 10^6 \text{ lbf-in./rad}$$

5. Calculate the bending moment (M_1) that is applied to the test specimen when equipped with flexures.

$$M_1 = (K_{F_1})(\theta)$$

$$M_1 = (2.5 \times 10^6 \text{ in.-lb/rad})(0.000332 \text{ rad})$$

$$= 830 \text{ lbf-in.}$$

6. Calculate the load (W) on the front actuator bearing when diaphragm flexures installed by using the following formula:

$$W = P \left(1 + \frac{u}{s}\right) + \frac{M_1}{s}$$

where: $P = \frac{2M_1}{L_1} = \frac{2(830 \text{ lbf-in.})}{18.1 \text{ in.}} = 46 \text{ lbf}$

$$u = 9.12 \text{ in.}$$

$$s = 7.35 \text{ in.}$$

$$L_2 = 18.1 \text{ in.}$$

$$M_1 = 1502.5 \text{ lbf-in.}$$

then:
$$W = 46 \text{ lbf} \left(1 + \frac{9.12 \text{ in.}}{7.35 \text{ in.}}\right) + \frac{830 \text{ lbf-in.}}{7.35 \text{ in.}}$$

$$= 216.0 \text{ lbf}$$

7. Calculate the stress (S_B) induced in the specimen due to reaction base twist by using the following formula:

$$S_B = \frac{M_1 r}{I}$$

where: $M_2 = 830$ lbf-in.

$$r = 1.75 \text{ in.}$$

$$I = 7.366 \text{ in.}^4$$

then:
$$S_B = \frac{830 \text{ lbf-in.} (1.75 \text{ in.})}{7.366 \text{ in.}^4}$$

$$= 197.2 \text{ psi}$$

Rotational Inertia

This section describes how to calculate the total rotational inertia of the 216 Rotary Actuator, specimen, and optional equipment. If the total rotational inertia exceeds recommended levels and the actuator is allowed to rotate until the rotor vane makes contact with the rotor vane stops, then the flange adapter may rotate on the actuator shaft or the actuator may be damaged.

To determine if the internal actuator rotor vane stops are adequate, the total rotational inertia (J_T) must be determined for the rotating mass. J_T equals the sum of the calculated J for the specimen, plus known J for the actuator, flange, and flexures.

Table 8 provides the rotational inertia values for the actuator and optional components. Table 9 lists the maximum allowable rotational inertia values in both U.S. Customary and SI metric units for each available actuator/servo valve combination. The following procedure describes how to calculate the total rotational inertia.

Rotational Inertia Equations

1. Calculate the total rotational inertia by using the following formula:

$$J_T = J_R + J_F + J_D + J_S$$

where: J_R = rotational inertia for actuator
 J_F = rotational inertia for flange adapter options
 J_D = rotational inertia for diaphragm flexure
 J_S = rotational inertia value for specimen configurations (step 2)

2. To determine J_S refer to Substeps A, B, and C and select the formula appropriate to the specimen configuration.

Refer to the figure on page 18 and note that m is equal to the mass of the specimen.

- A. If the specimen is a regular *solid* mass, use the following formula to calculate J_S :

$$J_S = \frac{1}{2} m r^2$$

where: $m = \pi r^2 L \rho$
 L = length
 ρ = density

- B. If the specimen is a regular *hollow* mass, use the following formula to calculate J_S :

$$J_S = \frac{1}{2} m (r_0^2 + r_1^2)$$

where: $m = \pi (r_0^2 + r_1^2) L \rho$
 L = length
 ρ = density

- C. If the specimen is an *offset* mass, use the following formula to calculate J_S :

$$J_S = m r^2$$

Table 8. Rotational Inertia for Actuator Components

Model	Rotary Actuator (J_R)		Flange Adapter (J_F)		Diaphragm Flexure (J_D)	
	lbm-in. ²	kg-m ²	lbm-in. ²	kg-m ²	lbm-in. ²	kg-m ²
216.10	725	2.12	6,500	1.92	2,700	0.79
216.20	2,180	6.37	10,400	3.04	7,900	2.30
216.30	6,350	18.43	18,300	5.35	14,300	4.20

3. After calculating the total rotational inertia (J_T), compare the value to the maximum allowable J_T for the specific actuator and servovalve combination indicated in Table 9. If the maximum allowable J_T exceeds the maximum levels specified in Table 9, then steps must be taken to control actuator motion and limit servovalve pressure. Contact MTS Systems Corporation for information on available actuator cushions and cross port relief valves.

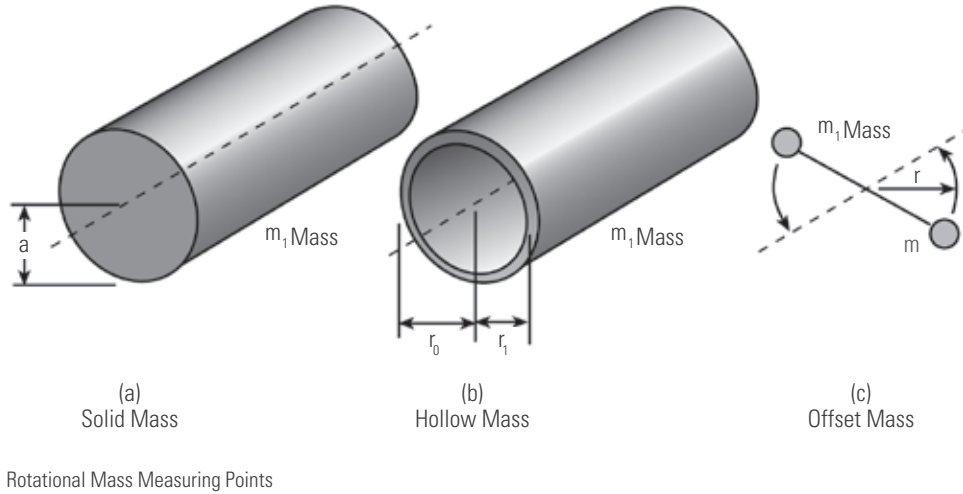


Table 9. Maximum Allowable Rotational Inertia (J) When Using Only Internal Actuator Rotor Vane Stops

U.S. Customary

Model	Servovalve Flow		Max J_T for Actuator Model (lbm-in. ²)		
	Rated (gpm)	Peak* (gpm)	216.10	216.20	216.30
252.25	15	26	367,592	2,546,602	15,821,000
252.31	25	43	134,392	931,045	5,784,381
252.32	40	70	50,712	351,327	2,182,718
252.33	60	103	23,422	162,268	1,008,136
256.05	50	87	32,830	227,441	1,413,042
256.09	90	156	10,210	70,738	439,458
256.18	180	311	2,569	17,798	110,579

SI Metric

Model	Servovalve Flow		Max J_T for Actuator Model (lbm-in. ²)		
	Rated (l/m)	Peak (l/m)	216.10	216.20	216.30
252.25	56.77	98.40	70.11	443.95	1854.42
252.31	94.62	162.75	25.63	162.31	677.98
252.32	151.39	264.94	9.67	61.25	255.83
252.33	227.09	389.83	4.47	28.29	118.16
256.05	189.24	329.28	6.26	39.65	165.62
256.09	340.63	590.43	1.95	12.33	51.51
256.18	681.26	1177.07	0.49	3.10	12.96

*Flow through the valve at 3,000 psi (ΔP). Using reduced system pressures (ΔP) will decrease peak flow $Q_{peak} = Q_{rated} \sqrt{\frac{\Delta P}{1000}}$.
Decreasing peak flow will allow an increase in acceptable inertia (J). Refer to Table 1 for the maximum velocity into vane stops where $W = \frac{Q_{peak} \times 3.85 \text{ in.}^3/\text{sec}}{\text{Displacement in.}^3/\text{rad}}$.

Ordering Information

The standard Series 216 Rotary Actuator assembly includes the actuator, ADT, servovalve manifold (standard MTS Model 256 type), flange adapter, and one diaphragm flexure. The following optional equipment is also available:

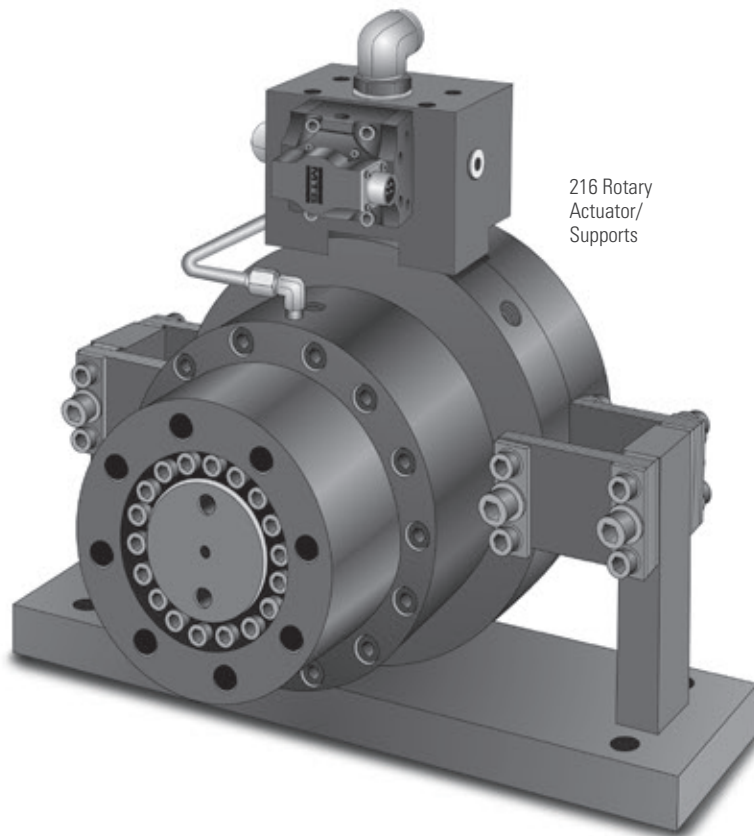
- » Diaphragm flexure
- » Torque cell
- » Reaction bracket
- » Reaction base

When ordering, please indicate the desired actuator force rating and specify the desired options.

Related Products

Refer to the following MTS Product Specifications for information on products that are commonly used in conjunction with the Series 216 Rotary Actuator:

- » Series 252 Servovalves
- » Series 256 Servovalves
- » Models 660.22 and 660.23 Differential Pressure Cells
- » Series 633 Reaction Torque Sensors



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