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Pneumatic Cylinders

Rotary Actuators

For installation, maintenance and replacement parts information, go to www.parker.com/actsafety.

Safety Guide for Selecting and Using Hydraulic, Pneumatic Cylinders and Their Accessories

WARNING: ⚠ FAILURE OF THE CYLINDER, ITS PARTS, ITS MOUNTING, ITS CONNECTIONS TO OTHER OBJECTS, OR ITS CONTROLS CAN RESULT IN:
- **Unanticipated or uncontrolled movement of the cylinder or objects connected to it.**
- **Falling of the cylinder or objects held up by it.**
- **Fluid escaping from the cylinder, potentially at high velocity.**

THESE EVENTS COULD CAUSE DEATH OR PERSONAL INJURY BY, FOR EXAMPLE, PERSONS FALLING FROM HIGH LOCATIONS, BEING CRUSHED OR STRUCK BY HEAVY OR FAST MOVING OBJECTS, BEING PUSHED INTO DANGEROUS EQUIPMENT OR SITUATIONS, OR SLIPPING ON ESCAPED FLUID.

Before selecting or using Parker (The Company) cylinders or related accessories, it is important that you read, understand and follow the following safety information. Training is advised before selecting and using The Company's products.

1.0 General Instructions

1.1 Scope – This safety guide provides instructions for selecting and using (including assembling, installing, and maintaining) cylinder products. This safety guide is a supplement to and is to be used with the specific Company publications for the specific cylinder products that are being considered for use.

1.2 Fail Safe – Cylinder products can and do fail without warning for many reasons. All systems and equipment should be designed in a fail-safe mode so that if the failure of a cylinder product occurs people and property won't be endangered.

1.3 Distribution – Provide a free copy of this safety guide to each person responsible for selecting or using cylinder products. Do not select or use The Company's cylinders without thoroughly reading and understanding this safety guide as well as the specific Company publications for the products considered or selected.

1.4 User Responsibility – Due to very wide variety of cylinder applications and cylinder operating conditions, The Company does not warrant that any particular cylinder is suitable for any specific application. This safety guide does not analyze all technical parameters that must be considered in selecting a product. The hydraulic and pneumatic cylinders outlined in this catalog are designed to The Company's design guidelines and do not necessarily meet the design guideline of other agencies such as American Bureau of Shipping, ASME Pressure Vessel Code etc. The user, through its own analysis and testing, is solely responsible for:
- Making the final selection of the cylinders and related accessories.
- Determining if the cylinders are required to meet specific design requirements as required by the Agency(s) or industry standards covering the design of the user's equipment.
- Assuring that the user's requirements are met, OSHA requirements are met, and safety guidelines from the applicable agencies such as but not limited to ANSI are followed and that the use presents no health or safety hazards.
- Providing all appropriate health and safety warnings on the equipment on which the cylinders are used.

1.5 Additional Questions – Call the appropriate Company technical service department if you have any questions or require any additional information. See the Company publication for the product being considered or used, or call 1-800-CPARKER, or go to www.parker.com, for telephone numbers of the appropriate technical service department.

2.0 Cylinder and Accessories Selection

2.1 Seals – Part of the process of selecting a cylinder is the selection of seal compounds. Before making this selection, consult the "seal information page(s)" of the publication for the series of cylinders of interest.

The application of cylinders may allow fluids such as cutting fluids, washdown fluids etc. to come in contact with the external area of the cylinder. These fluids may attack the piston rod wiper and or the primary seal and must be taken into account when selecting and specifying seal compounds.

Dynamic seals will wear. The rate of wear will depend on many operating factors. Wear can be rapid if a cylinder is mis-aligned or if the cylinder has been improperly serviced. The user must take seal wear into consideration in the application of cylinders.

2.2 Piston Rods – Possible consequences of piston rod failure or separation of the piston rod from the piston include, but are not limited to are:
- Piston rod and or attached load thrown off at high speed.
- High velocity fluid discharge.
- Piston rod extending when pressure is applied in the piston retract mode.

Piston rods or machine members attached to the piston rod may move suddenly and without warning as a consequence of other conditions occurring to the machine such as, but not limited to:
- Unexpected detachment of the machine member from the piston rod.
- Failure of the pressurized fluid delivery system (hoses, fittings, valves, pumps, compressors) which maintain cylinder position.
- Catastrophic cylinder seal failure leading to sudden loss of pressurized fluid.
- Failure of the machine control system.

Follow the recommendations of the "Piston Rod Selection Chart and Data" in the publication for the series of cylinders of interest. The suggested piston rod diameter in these charts must be followed in order to avoid piston rod buckling.

Piston rods are not normally designed to absorb bending moments or loads which are perpendicular to the axis of piston rod motion. These additional loads can cause the piston rod to fail. If these types of additional loads are expected to be imposed on the piston rod, their magnitude should be made known to our engineering department.

The cylinder user should always make sure that the piston rod is securely attached to the machine member.

On occasion cylinders are ordered with double rods (a piston rod extended from both ends of the cylinder). In some cases a stop is threaded on to one of the piston rods and used as an external stroke adjuster. On occasions spacers are attached to the machine member connected to the piston rod and also used as a stroke adjuster. In both cases the stops will create a pinch point and the user should consider appropriate use of guards. If these external stops are not perpendicular to the mating contact surface, or if debris is trapped between the contact surfaces, a bending moment will be placed on the piston rod, which can lead to piston rod failure. An external stop will also negate the effect of cushioning and will subject the piston rod to impact loading. Those two (2) conditions can cause piston rod failure. Internal stroke adjusters are available with and without cushions. The use of external stroke adjusters should be reviewed with our engineering department.

The piston rod to piston and the stud to piston rod threaded connections are secured with an anaerobic adhesive. The strength of the adhesive decreases with increasing temperature. Cylinders which can be exposed to temperatures above +250°F (+121°C) are to be ordered with a non studded piston rod and a pinned piston to rod joint.

2.3 Cushions – Cushions should be considered for cylinder applications when the piston velocity is expected to be over 4 inches/second.

Cylinder cushions are normally designed to absorb the energy of a linear applied load. A rotating mass has considerably more energy than the same mass moving in a linear mode. Cushioning for a rotating mass application should be reviewed by our engineering department.

2.4 Cylinder Mountings – Some cylinder mounting configurations may have certain limitations such as but not limited to minimum stroke for side or foot mounting cylinders or pressure de-ratings for certain mounts. Carefully review the catalog for these types of restrictions.

Always mount cylinders using the largest possible high tensile alloy steel socket head cap screws that can fit in the cylinder mounting holes and torque them to the manufacturer's recommendations for their size.

2.5 Port Fittings – Hydraulic cylinders applied with meter out or deceleration circuits are subject to intensified pressure at piston rod end.

The rod end pressure is approximately equal to:

$$\frac{\text{operating pressure x effective cap end area}}{\text{effective rod end piston area}}$$

Contact your connector supplier for the pressure rating of individual connectors.

3.0 Cylinder and Accessories Installation and Mounting

3.1 Installation

3.1.1 – Cleanliness is an important consideration, and cylinders are shipped with the ports plugged to protect them from contaminants entering the ports. These plugs should not be removed until the piping is to be installed. Before making the connection to the cylinder ports, piping should be thoroughly cleaned to remove all chips or burrs which might have resulted from threading or flaring operations.

3.1.2 – Cylinders operating in an environment where air drying materials are present such as fast-drying chemicals, paint, or weld splatter, or other hazardous conditions such as excessive heat, should have shields installed to prevent damage to the piston rod and piston rod seals.

3.1.3 – Proper alignment of the cylinder piston rod and its mating component on the machine should be checked in both the extended and retracted positions. Improper alignment will result in excessive rod gland and/or cylinder bore wear. On fixed mounting cylinders attaching the piston rod while the rod is retracted will help in achieving proper alignment.

3.1.4 – Sometimes it may be necessary to rotate the piston rod in order to thread the piston rod into the machine member. This operation must always be done with zero pressure being applied to either side of the piston. Failure to follow this procedure may result in loosening the piston to rod-threaded connection. In some rare cases the turning of the piston rod may rotate a threaded piston rod gland and loosen it from the cylinder head. Confirm that this condition is not occurring. If it does, re-tighten the piston rod gland firmly against the cylinder head.

For double rod cylinders it is also important that when attaching or detaching the piston rod from the machine member that the torque be applied to the piston rod end of the cylinder that is directly attaching to the machine member with the opposite end unrestrained. If the design of the machine is such that only the rod end of the cylinder opposite to where the rod attaches to the machine member can be rotated, consult the factory for further instructions.

3.2 Mounting Recommendations

3.2.1 – Always mount cylinders using the largest possible high tensile alloy steel socket head screws that can fit in the cylinder mounting holes and torque them to the manufacturer's recommendations for their size.

3.2.2 – Side-Mounted Cylinders – In addition to the mounting bolts, cylinders of this type should be equipped with thrust keys or dowel pins located so as to resist the major load.

3.2.3 – Tie Rod Mounting – Cylinders with tie rod mountings are recommended for applications where mounting space is limited. The standard tie rod extension is shown as BB in dimension tables. Longer or shorter extensions can be supplied. Nuts used for this mounting style should be torqued to the same value as the tie rods for that bore size.

3.2.4 – Flange Mount Cylinders – The controlled diameter of the rod gland extension on head end flange mount cylinders can be used as a pilot to locate the cylinders in relation to the machine. After alignment has been obtained, the flanges may be drilled for pins or dowels to prevent shifting.

3.2.5 – Trunnion Mountings – Cylinders require lubricated bearing blocks with minimum bearing clearances. Bearing blocks should be carefully aligned and rigidly mounted so the trunnions will not be subjected to bending moments. The rod end should also be pivoted with the pivot pin in line and parallel to axis of the trunnion pins.

3.2.6 – Clevis Mountings – Cylinders should be pivoted at both ends with centerline of pins parallel to each other. After cylinder is mounted, be sure to check to assure that the cylinder is free to swing through its working arc without interference from other machine parts.

4.0 Cylinder and Accessories Maintenance, Troubleshooting and Replacement

4.1 Storage – At times cylinders are delivered before a customer is ready to install them and must be stored for a period of time. When storage is required the following procedures are recommended.

4.1.1 – Store the cylinders in an indoor area which has a dry, clean and noncorrosive atmosphere. Take care to protect the cylinder from both internal corrosion and external damage.

4.1.2 – Whenever possible cylinders should be stored in a vertical position (piston rod up). This will minimize corrosion due to possible condensation which could occur inside the cylinder. This will also minimize seal damage.

4.1.3 – Port protector plugs should be left in the cylinder until the time of installation.

4.1.4 – If a cylinder is stored full of hydraulic fluid, expansion of the fluid due to temperature changes must be considered. Installing a check valve with free flow out of the cylinder is one method.

4.1.5 – When cylinders are mounted on equipment that is stored outside for extended periods, exposed unpainted surfaces, e.g. piston rod, must be coated with a rust-inhibiting compound to prevent corrosion.

4.2 Cylinder Trouble Shooting

4.2.1 – External Leakage

4.2.1.1 – Rod seal leakage can generally be traced to worn or damaged seals. Examine the piston rod for dents, gouges or score marks, and replace piston rod if surface is rough.

Rod seal leakage could also be traced to gland wear. If clearance is excessive, replace rod bushing and seal. Rod seal leakage can also be traced to seal deterioration. If seals are soft or gummy or brittle, check compatibility of seal material with lubricant used if air cylinder, or operating fluid if hydraulic cylinder. Replace with seal material, which is compatible with these fluids. If the seals are hard or have lost elasticity, it is usually due to exposure to temperatures in excess of 165°F (+74°C). Shield the cylinder from the heat source to limit temperature to 350°F (+177°C) and replace with fluorocarbon seals.

4.2.1.2 – Cylinder body seal leak can generally be traced to loose tie rods. Torque the tie rods to manufacturer's recommendation for that bore size.

Excessive pressure can also result in cylinder body seal leak. Determine maximum pressure to rated limits. Replace seals and retorque tie rods as in paragraph above. Excessive pressure can also result in cylinder body seal leak. Determine if the pressure rating of the cylinder has been exceeded. If so, bring the operating pressure down to the rating of the cylinder and have the tie rods replaced.

Pinched or extruded cylinder body seal will also result in a leak. Replace cylinder body seal and retorque as in paragraph above.

Cylinder body seal leakage due to loss of radial squeeze which shows up in the form of flat spots or due to wear on the O.D. or I.D. – Either of these are symptoms of normal wear due to high cycle rate or length of service. Replace seals as per paragraph above.

4.2.2 – Internal Leakage

4.2.2.1 – Piston seal leak (by-pass) 1 to 3 cubic inches per minute leakage is considered normal for piston ring construction. Virtually no static leak with lipseal type seals on piston should be expected. Piston seal wear is a usual cause of piston seal leakage. Replace seals as required.

4.2.2.2 – With lipseal type piston seals excessive back pressure due to over-adjustment of speed control valves could be a direct cause of rapid seal wear. Contamination in a hydraulic system can result in a scored cylinder bore, resulting in rapid seal wear. In either case, replace piston seals as required.

4.2.2.3 – What appears to be piston seal leak, evidenced by the fact that the cylinder drifts, is not always traceable to the piston. To make sure, it is suggested that one side of the cylinder piston be pressurized and the fluid line at the opposite port be disconnected. Observe leakage. If none is evident, seek the cause of cylinder drift in other component parts in the circuit.

4.2.3 – Cylinder Fails to Move the Load

4.2.3.1 – Pneumatic or hydraulic pressure is too low. Check the pressure at the cylinder to make sure it is to circuit requirements.

4.2.3.2 – Piston Seal Leak – Operate the valve to cycle the cylinder and observe fluid flow at valve exhaust ports at end of cylinder stroke. Replace piston seals if flow is excessive.

4.2.3.3 – Cylinder is undersized for the load – Replace cylinder with one of a larger bore size.

4.3 Erratic or Chatter Operation

4.3.1 – Excessive friction at rod gland or piston bearing due to load misalignment – Correct cylinder-to-load alignment.

4.3.2 – Cylinder sized too close to load requirements – Reduce load or install larger cylinder.

4.3.3 – Erratic operation could be traced to the difference between static and kinetic friction. Install speed control valves to provide a back pressure to control the stroke.

4.4 Cylinder Modifications, Repairs, or Failed Component – Cylinders as shipped from the factory are not to be disassembled and or modified. If cylinders require modifications, these modifications must be done at company locations or by The Company's certified facilities. The Actuator Division Engineering Department must be notified in the event of a mechanical fracture or permanent deformation of any cylinder component (excluding seals). This includes a broken piston rod, tie rod, mounting accessory or any other cylinder component. The notification should include all operation and application details. This information will be used to provide an engineered repair that will prevent recurrence of the failure.

It is allowed to disassemble cylinders for the purpose of replacing seals or seal assemblies. However, this work must be done by strictly following all the instructions provided with the seal kits.

Operating Fluids and Temperature Range
Fluidpower cylinders are designed for use with pressurized air, hydraulic oil and fire resistant fluids, in some cases special seals are required.

Standard Seals (class 1)
Class 1 seals are what is normally provided in a cylinder unless otherwise specified. They are intended for use with fluids such as: air, nitrogen, mineral base hydraulic oil or MIL-H-5606 within the temperature range of -10°F (-23°C) to +165°F (+74°C). Generally they are nitrile except for piston rod seals in hydraulic cylinders. However the individual seals may be nitrile (Buna-N) enhanced polyurethane, polymyte, P.T.F.E. or filled P.T.F.E.

Water Base Fluid Seals (class 2)
Generally class 2 seals are intended for use with water base fluids within the temperature of -10°F (-23°C) to +165°F (+74°C) except for High Water Content Fluids (H.W.C.F.) in which case Class 6 seals should be used. Typical water base fluids are: Water, Water-Glycol, Water-in Emulsion, Houghto-Safe 27, 620, 5040, Mobil Pyrogard D, Shell Irus 905, Ucon Hydrolube J-4. These seals are nitrile. Lipseal will have polymyte or P.T.F.E. back-up washer when required. O-rings will have nitrile back-up washers when required.

Ethylene Propylene (E.P.R.) Seals (class 3)
Class 3 seals are intended for use with some Phosphate Ester Fluids between the temperatures of -10°F (-23°C) to +130°F (+54°C). Typical fluids compatible with E.P.R. seals are Skydrol 500 and 700. E.P.R. are Ethylene Propylene. Lipseals will have a P.T.F.E. back-up washer when required. O-rings will have EPR back-up washers when required. Note: E.P.R. seals are not compatible with mineral base hydraulic oil or greases. Even limited exposure to these fluids will cause severe swelling. P.T.F.E. back-up washer may not be suitable when used in a radiation environment.

Low Temperature Nitrile Seals (class 4)
Class 4 seals are intended for low temperature service with the same type of fluids as used with Class 1 seals within the temperature range of -50°F (-46°C) to +150°F (+66°C). Lipseals will have leather, polymyte or P.T.F.E. back-up washers when required. O-rings will have nitrile back-up washers when required.

Fluorocarbon Seals (class 5)
Class 5 seals are intended for elevated temperature service or for some Phosphate Ester Fluids such as Houghto-Safe 1010, 1055, 1120; Fyrquel 150, 220, 300, 350; Mobile Pyrogard 42, 43, 53, and 55. Note: In addition, class 5 seals can be used with fluids listed below under standard service. However, they are not compatible with Phosphate Ester Fluids such as Skydrols. Class 5 seals can operate with a temperature range of -10°F (-23°C) to +250°F (+121°C). Class 5 seals may be operated to +400°F (+204°C) with limited service life. For temperatures above +250°F (+120°C) the cylinder must be manufactured with non-studded piston rod and thread and a pinned piston to rod connection. Class 5 Lipseals will have P.T.F.E. back-up washers when required. O-rings will have fluorocarbon back-up when required.

Warning ⚠
The piston rod stud and the piston rod to piston threaded connections are secured with an anaerobic adhesive which is temperature sensitive. Cylinders specified with Class 5 seals are assembled with anaerobic adhesive having a maximum temperature rating of +250°F (+74°C). Cylinders specified with all other seal compounds are assembled with anaerobic adhesive having a maximum operating temperature rating +165°F (+74°C). These temperature limitations are necessary to prevent the possible loosening of the threaded connections. Cylinders originally manufactured with class 1 seals (Nitrile) that will be exposed to ambient temperatures above +165°F (+74°C) must be modified for higher temperature service. Contact the factory immediately and arrange for the piston to rod and the stud to piston rod connections to be properly re-assembled to withstand the higher temperature service.

Lipseal Pistons
Under most conditions lipseals provide the best all around service for pneumatic applications. Lipseals with a back-up washer are often used for hydraulic applications when virtually zero static leakage is required. Lipseals will function properly in these applications when used in conjunction with moderate hydraulic pressures. A high load piston option is recommended when operating at high pressures and especially with large bore hydraulic cylinders.

Water Service
For pressures up to 250 psi 2A series cylinders can be modified to make them more suitable for use with water as the operating medium. The modifications include chrome-plated cylinder bore; cadmium-plated head, cap and piston; chrome-plated 17-4 stainless steel piston rod; chrome plated cushion sleeve or cushion spear.

Warranty
Parker Hannifin will warrant cylinders modified for water or high water content fluid service to be free of defects in materials or workmanship, but cannot accept responsibility to premature failure due to excessive wear due to lack of lubricity or where failure is caused by corrosion, electrolysis or mineral deposits within the cylinder.

Pre-Lubricated Air Cylinders
Parker Hannifin air cylinders are factory pre-lubricated with Lube-A-Cyl applied to seals, piston, cylinder bore, piston rod and gland surfaces, provides for normal cylinder operations with lubricated air.

Non-Lubricated Air Cylinders
For heavier duty operation, Series 2AN is recommended for non-lubricated air service. These cylinders are originally lubricated at the factory and, with the rounded lip seal design, typically require no additional lubrication for most applications.

Class No.	Typical Fluids	Temperature Range
1 (Standard) (Nitrile Polyurethane)	Air, Nitrogen	-10°F (-23°C) to +165°F (+74°C)
2 Optional Water Base Fluid Seal	Water, Water-Glycol, H.W.C.F. — See Class 6 below. Water-in-Oil Emulsion Houghto-Safe, 271, 620, 5040 Mobil Pyrogard D, Shell Irus 905 Ucon Hydrolube J-4	-10°F (-23°C) to +165°F (+74°C)
3 Special (E.P.R.) (At extra cost) Note: (E.P.R.) seals are not compatible with Hydraulic Oil	Some Phosphate Ester Fluids Skydrol 500, 7000	-10°F (-23°C) to +130°F (+54°C)
4 Special (Nitrile) (At extra cost)	Low Temperature Air	-50°F (-46°C) to +150°F (+66°C)
5 Optional (At extra cost) (Fluorocarbon Seals) Note: Fluorocarbon seals are not suitable for use with Skydrol fluid, but can be used with hydraulic oil if desired	High Temperature Houghto-Safe 1010, 1055, 1120 Fyrquel 150, 220, 300, 550 Mobil Pyrogard 42,43,53,55	See above paragraph on fluorocarbon seals for recommended temperature range.

Fundamental Cylinders

Standard Double-Acting Cylinders
Power stroke is in both directions and is used in the majority of applications.

Single-Acting Cylinders
When thrust is needed in only one direction, a single-acting cylinder may be used. The inactive end is vented to atmosphere through a breather/filter for pneumatic applications, or vented to reservoir below the oil level in hydraulic application.

Double-Rod Cylinders
Used when equal displacement is needed on both sides of the piston, or when it is mechanically advantageous to couple a load to each end. The extra end can be used to mount cams for operating limit switches, etc.

Spring Return, Single-Acting Cylinders
Usually limited to very small, short stroke cylinders used for holding and clamping. The length needed to contain the return spring makes them undesirable when a long stroke is needed.

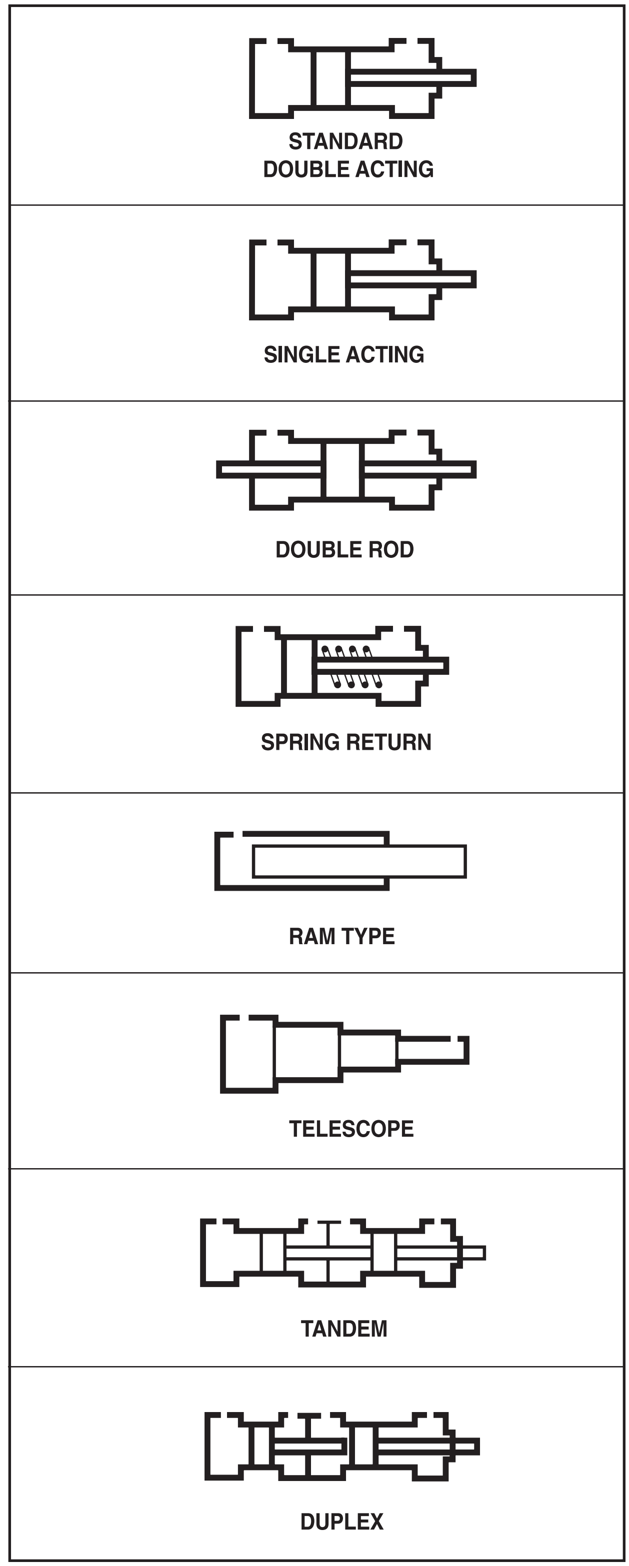
Ram Type, Single-Acting Cylinders
Containing only one fluid chamber, this type of cylinder is usually mounted vertically. The weight of the load retracts the cylinder. They are sometimes know as “displacement cylinders”, and are practical for long strokes.

Telescoping Cylinders
Available with up to 4 or 5 sleeves; collapsed length is shorter than standard cylinders. Available either single or double-acting, they are relatively expensive compared to standard cylinders.

Tandem Cylinders
A tandem cylinder is made up of two cylinders mounted in line with pistons connected by a common piston rod and rod seals installed between the cylinders to permit double acting operation of each. Tandem cylinders allow increased output force when mounting width or height are restricted.

Duplex Cylinders
A duplex cylinder is made up of two cylinders mounted in line with pistons not connected and with rod seals installed between the cylinders to permit double acting operation of each. Cylinders may be mounted with piston rod to piston (as shown) or back to back and are generally used to provide three position operation.

Illustration B29



Calculation of Cylinder Forces – Inch Based Product

Theoretical Push and Pull Forces for Pneumatic Cylinders

Push Force and Displacement

Cylinder Bore Size (inch)	Piston Area (in^2)	Cylinder Push Stroke Force in Pounds at Various Pressures (PSI)						Cu. Ft. Free Air at 80 lbs. Pressure, Required to Move Max. Load 1 Inch	Displacement Per Inch of Stroke (ft^3)
		25	50	65	80	100	250		
1	0.785	20	39	51	65	79	196	0.00293	0.0004542
1 1/2	1.767	44	88	115	142	177	443	0.00659	0.0010225
2	3.14	79	157	204	251	314	785	0.01171	0.0018171
2 1/2	4.91	123	245	319	393	491	1228	0.01830	0.0028414
3 1/4	8.30	208	415	540	664	830	2075	0.03093	0.0048032
4	12.57	314	628	817	1006	1257	3143	0.04685	0.0072743
5	19.64	491	982	1277	1571	1964	4910	0.07320	0.0113657
6	28.27	707	1414	1838	2262	2827	7068	0.10541	0.0163599
7	38.49	962	1924	2502	3079	3849	9623	0.14347	0.0222743
8	50.27	1257	2513	3268	4022	5027	12568	0.18740	0.0290914
10	78.54	1964	3927	5105	6283	7854	19635	0.29280	0.0454513
12	113.10	2828	5655	7352	9048	11310	28275	0.42164	0.0654513
14	153.94	3849	7697	10006	12315	15394	38485	0.57389	0.0896643

Deductions for Pull Force and Displacement

Piston Rod Dia. (inch)	Piston Rod Area (in^2)	Piston Rod Diameter Force In Pounds At Various Pressures (PSI) — To determine Cylinder Pull Force or Displacement, deduct the following Force or Displacement corresponding to Rod Size, from selected Push Stroke Force or Displacement corresponding to Bore Size in table above.						Cu. Ft. Free Air at 80 lbs. Pressure, Required to Move Max. Load 1 Inch	Displacement Per Inch of Stroke (ft^3)
		25	50	65	80	100	250		
1/2	0.196	5	10	13	16	20	49	0.00073	0.0001134
5/8	0.307	8	15	20	25	31	77	0.00114	0.0001776
1	0.785	20	39	51	65	79	196	0.00293	0.0004542
1 3/8	1.49	37	75	97	119	149	373	0.00554	0.0008622
1 3/4	2.41	60	121	157	193	241	603	0.00897	0.0013946
2	3.14	79	157	204	251	314	785	0.01171	0.0018171
2 1/2	4.91	123	245	319	393	491	1228	0.01830	0.0028414
3	7.07	177	354	460	566	707	1767	0.02635	0.0040914
3 1/2	9.62	241	481	625	770	962	2405	0.03587	0.0055671
4	12.57	314	628	817	1006	1257	3143	0.04685	0.0072743
4 1/2	15.90	398	795	1033	1272	1590	3975	0.05929	0.0092013
5	19.64	491	982	1277	1571	1964	4910	0.07320	0.0113657
5 1/2	23.76	594	1188	1544	1901	2376	5940	0.08857	0.0137500

General Formula

The cylinder output forces are derived from the formula:

$$F = P \times A$$

Where F = Force in pounds.

P = Pressure at the cylinder in pounds per square inch, gauge.

A = Effective area of cylinder piston in square inches.

Free Air refers to normal atmospheric conditions of the air at sea level (14.7 psi). Use above cu. ft. free air required data to compute CFM required from a compressor at 80 psi. Cu. ft. of free air required at other pressures can be calculated using formula below.

$$V_1 = \frac{(P_2 + 14.7)\,V_2}{14.7}$$

Where V_1 = Free air consumption per inch of stroke (cubic feet).

V_2 = Cubic feet displaced per inch of stroke.

P_2 = Gauge pressure required to move maximum load.

Calculation of Cylinder Forces – Metric Based Product

General Formula

The cylinder output forces are derived from the formula:

$$F = \frac{P \times A}{10}$$

Where F = Force in N
P = Pressure at the cylinder in Bar
A = Effective area of cylinder piston in square mm.

Prior to selecting the cylinder bore size, properly size the piston rod for tension (pull) or compression (push) loading (see the Piston Rod Selection Chart).

If the piston rod is in compression, use the 'Push Force' table below, as follows:

1. Identify the operating pressure closest to that required
2. In the same column, identify the force required to move the load (always rounding up).
3. In the same row, look over to the cylinder bore required.

If the cylinder envelope dimensions are too large for the application, increase the operating pressure, if possible, and repeat the exercise.

If the piston rod is in tension, use the 'Deduction for Pull Force' table. The procedure is the same but, due to the reduced area caused by the piston rod, the force available on the 'pull' stroke will be smaller. To determine the pull force:

1. Follow the procedure for 'push' force as described previously.
2. Using the 'Deduction for Pull Force' table, identify the force indicated according to the rod and pressure selected.
3. Deduct this from the original 'push' force. The resultant is the net force available to move the load.

If this force is not large enough, repeat the process and increase the system operating pressure or cylinder diameter if possible. For assistance, contact your local authorized Parker distributor.

Push Force

Cylinder Bore Size (mm)	Piston Area (mm²)	Cylinder Push Force (N) at Various Pressures in Bar			
		1	5	7	10
6	28	3	14	20	28
8	50	5	25	35	50
10	79	8	39	55	79
12	113	11	57	79	113
14	154	15	77	108	154
16	201	20	101	141	201
20	314	31	157	220	314
25	491	49	245	344	491
32	804	80	402	563	804
40	1,257	126	628	880	1,257
50	1,963	196	982	1,374	1,963
63	3,117	312	1,559	2,182	3,117
80	5,027	503	2,513	3,519	5,027
100	7,854	785	3,927	5,498	7,854
125	12,272	1,227	6,136	8,590	12,272
160	20,106	2,011	10,053	14,074	20,106
200	31,416	3,142	15,708	21,991	31,416

Deduction for Pull Force

Piston Rod Size (mm)	Piston Rod Area (mm²)	Reduction in Force (N) at Various Pressures in Bar			
		1	5	7	10
4	13	1	6	9	13
6	28	3	14	20	28
8	50	5	25	35	50
10	79	8	39	55	79
12	113	11	57	79	113
16	201	20	101	141	201
20	314	31	157	220	314
25	491	49	245	344	491
32	804	80	402	563	804
40	1,257	126	628	880	1,257

Single rod type, fluid power cylinders are commonly available in 20 standard mounting styles ranging from head or cap end mounts to intermediate mounts. Many mounting styles are also available in double rod type cylinders. Refer to NFPA Std. B93.15-1981 or Parker air or hydraulic cylinder catalogs for detailed description.

Standard mounting styles for fluid power cylinders fall into three basic groups. The groups can be described as follows.

Group 1 – Straight line force transfer with fixed mounts which absorb force on cylinder centerline.

Group 3 – Straight line force transfer with fixed mounts which do not absorb force on cylinder centerline.

Group 2 – Pivot force transfer with pivot mounts which absorb force on cylinder centerline and permit cylinder to change alignment in one plane.

Cylinder mounting directly affects the maximum pressure at which the fluid power cylinder can be used, and proper selection of mounting style will have a bearing on cylinder operation and service life. Whether the cylinder is used in thrust or tension, its stroke length, piston rod diameter and the method of connection to load also must be considered when selecting a mounting style.

Many pneumatic cylinders are offered for use with air pressure up to 250 psi. The industrial tie rod types, known as NFPA cylinders, with square heads and caps, plus mountings lend themselves to standardized mounts which are similar in appearance for air cylinders.

Because of the all steel construction, Parker 2A/2AN air cylinders have a design factor of better than 4:1, and the various mounts can be used without limitations up to the cylinder manufacturer's maximum rated pressure.

Straight Line Force Transfer (Group 1)

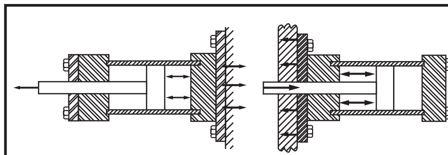
Cylinders with fixed mounts (Group 1) which absorb the force on centerline are considered the best for straight line force transfer. Tie rods extended, flange or centerline lug mounts are symmetrical and allow the thrust or tension forces of the piston rod to be distributed uniformly about the cylinder centerline. Mounting bolts are subjected to simple tension or simple shear without compound forces, and when properly installed damaging cylinder bearing sideloading is kept to a minimum.

Tie Rods Extended are considered to be of the centerline mount type. The cylinder tie rods are designed to withstand maximum rated internal pressure and can be extended and used to mount the cylinder at cap or head end. This often overlooked mounting will securely support the cylinder when bolted to the panel or machine member to which the cylinder is mounted. The torque value for the mounting nuts should be the same as the tie rod nut torque recommended by the cylinder manufacturer. Cylinders are available with tie rod extended both ends. In such applications one end is used for mounting and the opposite end to support the cylinder or to attach other machine components.

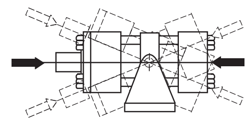
Tie rod mount cylinders may be used to provide thrust or tension forces at full rated pressures.

Tie rods extended head end (Parker Style TB), cap end (Parker Style TC) or extended both ends (Parker Style TD) are readily available and fully dimensioned in Parker cylinder product catalogs.

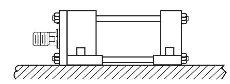
Flange Mount cylinders are also considered to be centerline mount type and thus are among the best mounts for use on straight line force transfer applications. The machine designer has a choice of mounting styles at each end, such as head rectangular flange (Style J), head square flange (Style JB), cap rectangular flange (Style H), and cap square flange (Style HB). Selection of a flange mounting style depends, in part, upon whether the major force applied to the load will result in compression (push) or tension (pull) stresses of the cylinder piston rod. Cap end mounting styles are recommended for thrust loads (push), while head end mounting styles are recommended where the major load puts the piston rod in tension (pull).



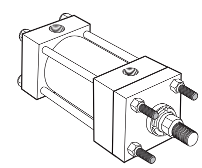
Group 1



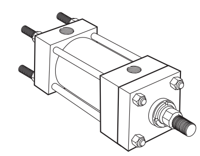
Group 2



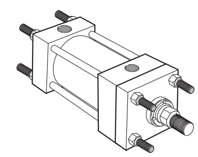
Group 3



Tie rods extended head end, Style TB

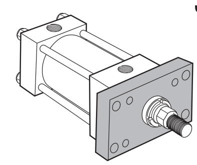


Tie rods extended cap end, Style TC

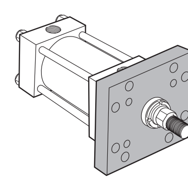


Tie rods extended both ends, Style TD

J



JB



Flange mounts are best used when end face is mounted against the machine support member. (Fig. 1) This is especially true where head rectangular flange type (Style J) is used with major load in tension. In this mode, the flange is not subjected to flexure or bending stresses, nor are the mounting bolts stressed to unusually high levels. The use of head rectangular flange (Style J) mount with major load in compression (see Fig. 2) is not recommended except on reduced pressure systems. The use of Style J mount in compression subjects the flange to bending and the mounting bolts to tension stresses, which could result in early fatigue failure. For maximum allowable pressure with Style J head rectangular mount used for compression (push) or rear face of flange mounted, see pressure rating in product catalogs for medium- or heavy-duty hydraulic cylinders. For applications where push forces require full rated system pressure, head square flange (Style JB) mounts are recommended.

Cap flange mounts are also best used when end face is mounted against the machine support member. The use of cap rectangular flange mount, Style H, is not recommended on applications where the major load is in tension (pull) except at reduced pressure. For maximum allowable pressure with cap rectangular flange, Style H, used in tension application (pull) or front of flange mounted, see maximum pressure rating in product catalogs for medium- and heavy-duty hydraulic cylinders.

For applications where pull forces involved require full rated system pressure, cap square flange, Style HB mounts are recommended.

Straight Line Force Transfer (Group 3)

Side Mount cylinders are considered to be fixed mounts which do not absorb force on their centerline. Cylinders of this group have mounting lugs connected to the ends, and one style has side tapped holes for flush mounting. The plane of their mounting surfaces is not through the centerline of the cylinder, and for this reason side mounted cylinders produce a turning moment as the cylinder applies force to the load. (Fig. 4) This turning moment tends to rotate the cylinder about its mounting bolts. If the cylinder is not well secured to the machine member on which it is mounted or the load is not well-guided, this turning moment results in side load applied to rod gland and piston bearings. To avoid this problem, side mount cylinders should be specified with a stroke length at least equal to the bore size.

Shorter stroke, large bore cylinders tend to sway on their mountings when subjected to heavy loads, especially side end lug or side and angle mounts. (Fig. 5)

Side mount cylinders are available in several mounting styles, such as side lug (Style C), Side tapped (Style F), side end lug (Style G*) and side end angle (Style CB*). Of these, the side lug mount its the most popular and reliable, since the mounting lugs are welded to head and cap to form an integral unit at each end.

Side tapped mount is the choice when cylinders must be mounted side by side at minimum center-to-center distance. Another narrow side mount style is the side end lug mount which has lugs threaded to the tie rods. Thus the end lugs serve a dual function of holding the cylinder together and act as a means of mounting. This mounting style should be used only on medium- to light-duty applications, because the end lugs are subjected to compound stresses which could result in early failure.

*Not available for 2A/2AN Series.

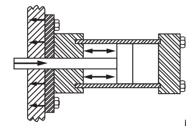


Fig. 1

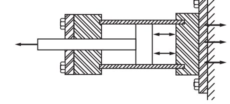
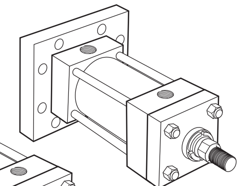
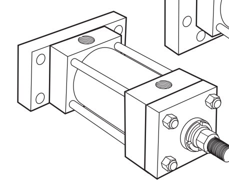


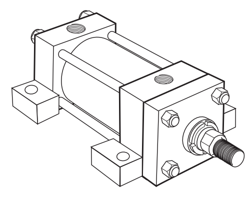
Fig. 2



HB



H



C

Fig. 4

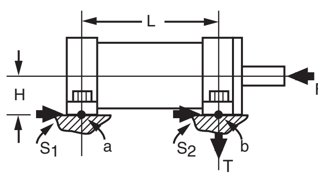
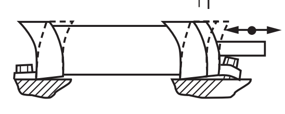
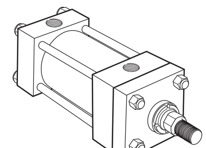
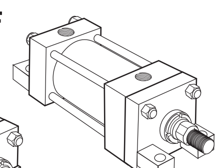


Fig. 5

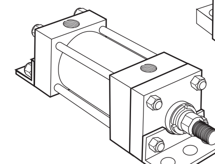




F



G



CB

The side end angle mount is also a narrow mount type, but is the weakest of the side mount styles. Its use should be limited to a maximum pressure of 500 psi and minimum stroke length of two times the bore size. For pressure rating of longer strokes, consult the cylinder manufacturer.

Consideration should also be given to design of the machine frame used to support cylinders non-centerline mount, since stronger members are often required to resist bending moments. (See Fig. 6)

Side mount cylinders depend wholly on the friction of their mounting surfaces in contact with the machine member to absorb the force produced. Thus the torque applied to the mounting bolts is an important consideration. Since the mounting bolts are the same diameter as the tie rods for a given cylinder, it is recommended that the torque applied to the mounting bolts be the same as the tie rod torque recommended by the cylinder manufacturer for the given bore size.

For heavy loads or high shock conditions, side mounted cylinders should be held in place to prevent shifting by keying or pinning. A shear key, consisting of a plate extending from side of cylinder, can be supplied on most cylinders. (Fig. 7) This method may be used where a keyway can be milled into a machine member. It serves to take up shear loads and also provides accurate alignment of the cylinder.

Side lug mounts are designed so as to allow dowel pins to be used to pin the cylinder to the machine member. Pins, when used, are installed on both sides of the cylinder but not at both ends. (See Fig. 8)

The use of a separate shear key is fairly common. It should be placed at the proper end of the cylinder to absorb the major load. (see Fig. 9)

Side mount cylinders should not be pinned or keyed at both ends. Changes in temperature and pressure under normal operating conditions cause the cylinder to increase (or decrease) in length from its installed length and therefore must be free to expand and contract. If pinned or keyed at both ends, the advantages of cylinder elasticity in absorbing high shock loads will be lost. (Fig. 10)

If high shock loads are the major consideration, the cylinder should be mounted and pins or shear key so located as to take full advantage of the cylinder's inherent elasticity. For major shock load in tension, locate key at rear face of head or pin the head in place. For major shock load in thrust, pin cap in place or locate key at front face of cap.

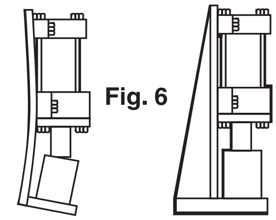


Fig. 6

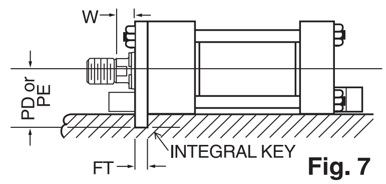


Fig. 7

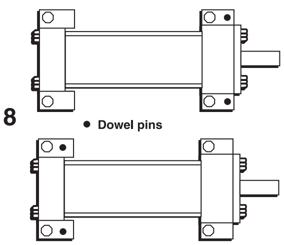


Fig. 8

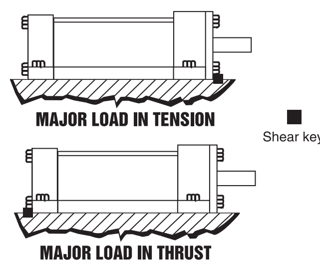


Fig. 9
Right

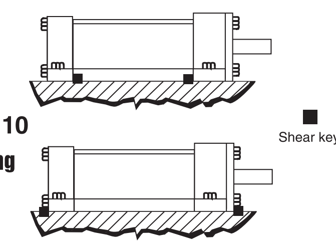


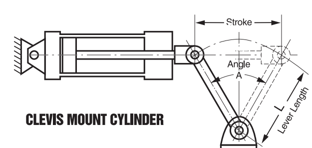
Fig. 10
Wrong

Pivot Force Transfer (Group 2)

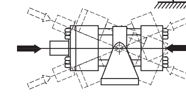
Cylinders with pivot mounts which absorb force on centerline should be used on applications where the machine member to be moved travels in a curved path. There are two basic ways to mount a cylinder so that it will pivot during the work cycle: clevis or trunnion mounts, with variations of each. Pivot mount cylinders are available in cap fixed clevis (Style BB), cap detachable clevis (Style BC*), cap spherical bearing (Style SB), head trunnion (Style D), cap trunnion (Style DB), and intermediate fixed trunnion(Style DD).

Pivot mount cylinders can be used on tension (pull) or thrust (push) applications at full rated pressure, except long stroke thrust cylinders are limited by piston rod column strength. See Piston Rod Selection Chart on page 15.

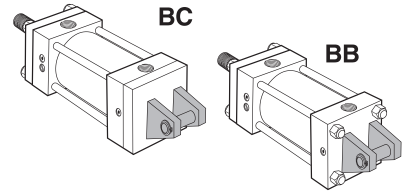
Clevis or single ear mounts are usually an integral part of the cylinder cap (though one style is detachable) and provide a single pivot point for mounting the cylinder. A pivot pin of proper length and of sufficient diameter to withstand the maximum shear load developed by the cylinder at rated operating pressure is included as a part of the clevis mount style. The fixed clevis mount, Style BB, is the most popular of the pivot force transfer types and is used on applications where the piston rod end travels in a curved path in one plane. It can be used vertically or horizontally or any angle in between. On long stroke push applications it may be necessary to use a larger diameter piston rod to prevent buckling or stop tube to minimize side loading due to "jackknife" action of cylinder in extended position. Fixed clevis mount cylinders will not function well if the curved path of piston rod travel is other than one plane. Such an application results in misalignment and causes the gland and piston bearing surfaces to be subjected to unnecessary side loading. For applications where the piston rod will travel in a path not more than 3° either side of the true plane motion, a cap spherical bearing mount is recommended. A spherical bearing rod eye should be used at rod end. Most spherical bearing mounts have limited pressure ratings. Consult cylinder manufacturer's product catalog.



CLEVIS MOUNT CYLINDER



TRUNNION MOUNT CYLINDER



*Not available for 2A/2AN Series.

Cap detachable clevis mounts are usually used for air service. Cap detachable clevis mounts are longer, centerline of pivot pin to shoulder of piston rod, than fixed clevis mount in any given bore size. They are most often specified to avoid port relocation charges. Application parameters are the same as described for fixed clevis mounting.

Trunnion mount cylinders are a second type of pivot mounts used on applications where the piston rod travels in a curved path in one plane. Three styles are available – head trunnion (Style D), cap trunnion (Style DB) and intermediate fixed trunnion (Style DD). Trunnion pins are designed for shear loads only and should not be subjected to bending stresses. Pillow blocks, rigidly mounted with bearings at least as long as the trunnion pins, should be used to minimize bending stresses. The support bearings should be mounted as close to the head, cap or intermediate trunnion shoulder faces as possible.

Cap end trunnion mounts are used on cylinder applications similar to fixed clevis mounts, and the same application data applies.

Head trunnion mount cylinders can usually be specified with smaller diameter piston rods than cylinders with pivot point at cap end or at an intermediate position. This is evident in data shown in piston rod selection chart on page 15. On head end trunnion mount, long stroke, cylinder applications consideration should be given to the overhanding weight at cap end of cylinder. To keep trunnion bearing loading within limits, stroke lengths should be not more than 5 times the bore size. If cylinder stroke is greater than 5 times the bore size and piston speed exceeds 35 ft/minute, consult factory.

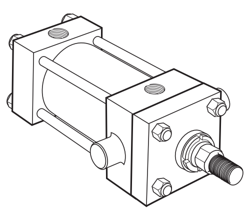
Intermediate fixed trunnion mount is the best of the trunnion mount types. The trunnion can be located so as to balance the weight of the cylinder, or it can be located at any point between the head or cap to suit the application. It is of fixed design, and the location of the trunnion must be specified (XI dimension) at time of order. The location cannot be easily changed once manufactured.

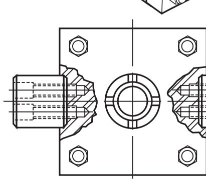
Thrust exerted by a pivot transfer cylinder working at an angle is proportional to the angle of the lever arm which it operates. In Fig. 12 that vector force, T, which is at right angle to the lever axis, is effective for turning the lever. The value of T varies with the acute angle A between cylinder centerline and lever axes. To calculate effective thrust T, multiply cylinder thrust by the power factor shown in table below.

Accessories

Rod clevises or rod knuckles are available for use with either fixed or pivot mount cylinders. Such accessories are usually specified with pivot mount cylinders and are used with pivot pin centerline in same axis as pivot pin centerline on cylinder. Pivot pins for accessories must be ordered separately.

Pin size of rod clevis or rod knuckle should be at least equal in diameter to the pin diameter of the cap fixed clevis pin for the cylinder bore size specified. Larger accessories are more costly and usually result in a mis-match of pin diameters, especially when used with oversize piston rods.





Removable Trunnion Pins

Removable trunnion pins are a convenience when machine structures or confined space prohibit the use of separate pillow blocks situated close to the cylinder sides. Parker offers a removable pin design in 1-1/2" through 8" bores sizes. (See following table for recommended maximum operating pressure.) Mounting pin diameters and lengths are identical to those in Mounting Styles D and DB for any given bore size. These removable trunnion pins can be provided on the cap end (Style DBR) of Series “2A” cylinders with any rod diameter. They can also be provided on the head end (Style DR) of cylinders with standard rods.

Pressure Ratings – Removable Trunnion Pin Mounting

Bore Size	1"	1½"	2"	2½"	3¼"	4"	5"	6"	8"
Std. Pressure Rating (PSI)	—	250	250	250	250	250	150	200	125
Extreme Pressure Rating	—	450	400	275	375	250	150	200	125
Hydraulic Pressure (PSI)	—	750	700	450	625	400	250	325	200

Spherical Bearing Mount

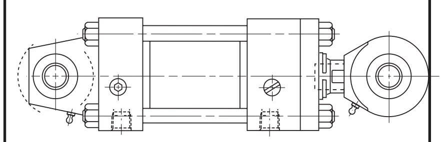
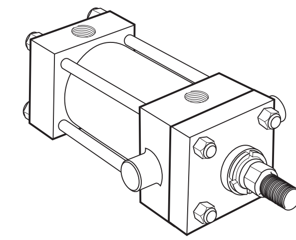
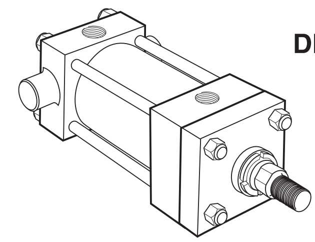


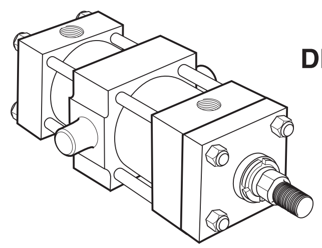
Fig. 11



D



DB



DD

Clevis Mount Cylinder

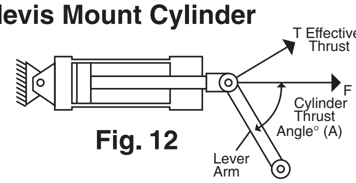


Fig. 12

Power Factor Table

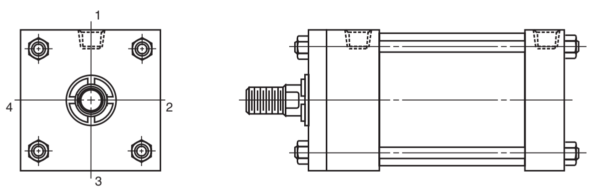
Angle A Degrees	Pwr. Factor (SIN A)	Angle A Degrees	Pwr. Factor (SIN A)
5	0.087	50	0.766
10	0.174	55	0.819
15	0.259	60	0.867
20	0.342	65	0.906
25	0.423	70	0.940
30	0.500	75	0.966
35	0.573	80	0.985
40	0.643	85	0.996
45	0.707	90	1.000

Ports

Parker hydraulic and pneumatic cylinders can be supplied with S.A.E. straight O-ring ports or N.P.T.F. pipe thread ports. For the type of port recommended and port size, see respective product catalogs. If specified on your order, extra ports can be provided on the sides of heads or caps that are not occupied by mountings or cushion valve on all cylinders except Series C and S.

Standard port location is position 1 as shown on line drawings in product catalog and Figure 1 below. Cushion adjustment needle and check valves are at positions 2 and 4 (or 3), depending on mounting style. Heads or caps which do not have an integral mounting can be rotated and assembled with ports at 90°or 180° from standard position. Mounting styles on which head or cap can be rotated at no extra charge are shown in Table A below. To order, specify by position number. In such assemblies the cushion adjustment needle and check valve rotate accordingly, since their relationship with port position does not change.

Figure 1



Head (Rod) End **Head** **Cap**

Table A

Mounting Style	Head End	Cap End
T, TB, TC, TD, H, HB, J, JB, DD	1, 2, 3 or 4	1, 2, 3 or 4
BB, DB, HH	1, 2, 3 or 4	1 or 3
D, JJ	1 or 3	1, 2, 3 or 4
C, F	1	1

Applies to Series 2A/2AN.

Straight Thread Ports

The S.A.E. straight thread O-ring port is recommended for hydraulic applications. Parker will furnish this port configuration at positions shown in Table A on previous page. This port can also be provided at positions other than those shown in Table A at an extra charge. S.A.E. port size numbers are listed next to their N.P.T.F. pipe thread counterparts for each bore size in the respective product catalogs. Size number, tube O.D. and port thread size for S.A.E. ports are listed in Table C.

Table C

S.A.E. Straight Thread "O" Ring Ports

Size No.	Tube O.D. (In.)	Thread Size	Size No.	Tube O.D. (In.)	Thread Size
2	1/8"	5/16 - 24	12	3/4"	1 1/16 - 12
3	3/16"	3/8 - 24	—	—	—
4	1/4"	7/16 - 20	16	1"	1 5/16 - 12
5	5/16"	1/2 - 20	20	1 1/4"	1 5/8 - 12
6	3/8"	9/16 - 18	24	1 1/2"	1 7/8 - 12
8	1/2"	3/4 - 16	32	2"	2 1/2 - 12
10	5/8"	7/8 - 14	—	—	—

Note: For the pressure ratings of individual connectors, contact your connector supplier.

Cylinder Port Options

Option "T" SAE Straight Thread O-Ring Port. Recommended for most hydraulic applications.

Option "U" Conventional NPTF Ports (Dry-Seal Pipe Threads). Recommended for pneumatic applications only.

Option "R" BSPP Port (British Parallel Thread). ISO 228 port commonly used in Europe.

Option "P" SAE Flange Pots Code 61 (3000 psi). Recommended for hydraulic applications requiring larger port sizes.

Option "B" BSPT (British Tapered Thread).

Option "G" Metric Straight Thread Port similar to Option "R" with metric thread. Popular in some European applications.

Option "Y" ISO-6149-1 Metric Straight Thread Port. Recommended for all hydraulic applications designed per ISO standards.

Ports can be supplied at positions other than those shown in Table A at an extra charge. To order, specify port position as shown in Figure 1.

International Ports

Other port configurations to meet international requirements are available at extra cost. Parker cylinders can be supplied, on request, with British standard taper port (BSPT). Such port has a taper of 1 in 16 measured on the diameter (1/16" per inch). The thread form is Whitworth System, and size and number of threads per inch are as follows:

Table D

British Standard Pipe Threads

Nominal Pipe Size	No. Threads Per Inch	Pipe O.D.
1/8	28	.383
1/4	19	.518
3/8	19	.656
1/2	14	.825
3/4	14	1.041
1	11	1.309
1 1/4	11	1.650
1 1/2	11	1.882
2	11	2.347

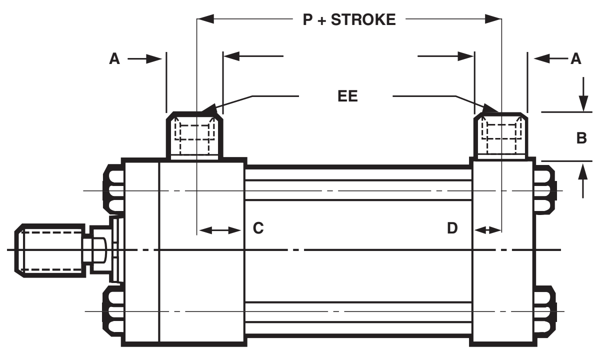
British standard parallel internal threads are designated as BSP and have the same thread form and number of threads per inch as the BSPT type and can be supplied, on request, at extra cost. Unless otherwise specified, the BSP or BSPT port size supplied will be the same nominal pipe size as the N.P.T.F. port for a given bore size cylinder.

Metric ports options G or Y can also be supplied to order at extra cost.

Oversize Ports

Oversize NPTF or SAE straight thread ports can be provided, at an extra charge, on pneumatic and hydraulic cylinders. For ports one size larger than standard for steel cylinders, welded port bosses which protrude from the side of the head or cap are supplied. For dimensions, see drawings and tables below. Cylinders which are equipped with cap end cushions and ordered with one size oversize ports having hydraulic fluid flow exceeding 25 ft./sec. All cylinders ordered with double oversize ports should always be ordered with a "solid cushion" at cap end.

Cylinders which are connected to a meter out flow control with flow entering the cap end of a cylinder provided by an accumulator may also experience damage to the cushion bushing due to high instantaneous fluid flows. This condition can be eliminated by using a meter in flow control or "solid cushions" at cap end.

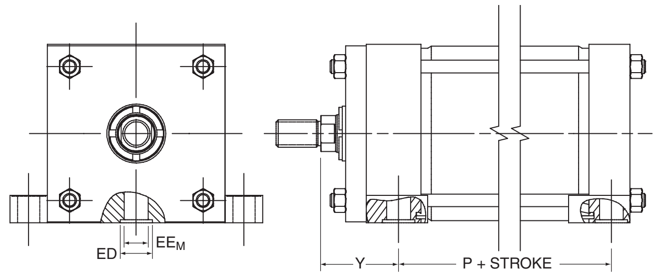


Oversize NPTF Port Boss Dimensions
Series 2A/2AN Cylinders

Bore	EE (NPTF)	A (Dia.)	B	C	D	P
1	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$\frac{9}{16}$	$\frac{1}{2}$	$2\frac{1}{16}$
$1\frac{1}{2}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{15}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$2\frac{3}{16}$
2	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{15}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$2\frac{3}{16}$
$2\frac{1}{2}$	$\frac{1}{2}$	$1\frac{1}{8}$	$\frac{15}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$2\frac{5}{16}$
$3\frac{1}{4}$	$\frac{3}{4}$	$1\frac{3}{8}$	1	$\frac{11}{16}$	$\frac{5}{8}$	$2\frac{9}{16}$
4	$\frac{3}{4}$	$1\frac{3}{8}$	1	$\frac{11}{16}$	$\frac{5}{8}$	$2\frac{9}{16}$
5	$\frac{3}{4}$	$1\frac{3}{8}$	1	$\frac{11}{16}$	$\frac{5}{8}$	$2\frac{13}{16}$
6	1	$1\frac{3}{4}$	$1\frac{3}{16}$	$\frac{15}{16}$	$\frac{3}{4}$	$3\frac{3}{16}$
7-8	1	$1\frac{3}{4}$	$1\frac{3}{16}$	$\frac{15}{16}$	$\frac{3}{4}$	$3\frac{5}{16}$
10	$1\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{1}{8}$	1	$4\frac{1}{4}$
12	$1\frac{1}{4}$	$2\frac{1}{4}$	$1\frac{5}{16}$	$1\frac{1}{8}$	1	$4\frac{3}{4}$
14	$1\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{9}{16}$	$1\frac{1}{4}$	$1\frac{1}{8}$	$5\frac{1}{2}$

Manifold Ports

Side mounted cylinders, Style C can be furnished with the cylinder ports arranged for mounting and sealing to a maniforld surface. The ports are drilled and counterbored for O-ring seals which are provided. With these specifications, the mounting is designated Style CM or KCM.



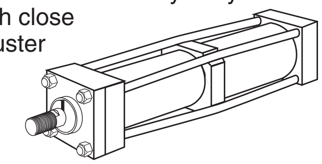
Series 2A/2AN Cylinders

Bore	Rod Code	Rod. Dia. (MM)	Y±1/32	P±1/32	EEM	ED
1	All	All	$1\frac{15}{16}$	$2\frac{1}{8}$	$\frac{3}{8}$	$\frac{11}{16}$
$1\frac{1}{2}$	1	$\frac{5}{8}$	2	$2\frac{1}{8}$	$\frac{1}{2}$	$\frac{13}{16}$
	2	1	$2\frac{3}{8}$			
2	1	$\frac{5}{8}$	2	$2\frac{1}{8}$	$\frac{1}{2}$	$\frac{13}{16}$
	2	$1\frac{3}{8}$	$2\frac{5}{8}$			
	3	1	$2\frac{3}{8}$			
$2\frac{1}{2}$	1	$\frac{5}{8}$	2	$2\frac{1}{4}$	$\frac{1}{2}$	$\frac{13}{16}$
	2	$1\frac{3}{4}$	$2\frac{7}{8}$			
	3	1	$2\frac{3}{8}$			
	4	$1\frac{3}{8}$	$2\frac{5}{8}$			
$3\frac{1}{4}$	1	1	$2\frac{7}{16}$	$2\frac{5}{8}$	$\frac{5}{8}$	$\frac{15}{16}$
	2	2	$3\frac{1}{16}$			
	3	$1\frac{3}{8}$	$2\frac{11}{16}$			
	4	$1\frac{3}{4}$	$2\frac{15}{16}$			
4	1	1	$2\frac{7}{16}$	$2\frac{5}{8}$	$\frac{5}{8}$	$\frac{15}{16}$
	2	$2\frac{1}{2}$	$3\frac{5}{16}$			
	3	$1\frac{3}{8}$	$2\frac{11}{16}$			
	4	$1\frac{3}{4}$	$2\frac{15}{16}$			
	5	2	$3\frac{1}{16}$			
5	1	1	$2\frac{7}{16}$	$2\frac{7}{8}$	$\frac{5}{8}$	$\frac{15}{16}$
	2, 6 & 7	$3\frac{1}{2}$, $2\frac{1}{2}$ & 3	$3\frac{5}{16}$			
	3	$1\frac{3}{8}$	$2\frac{11}{16}$			
	4	$1\frac{3}{4}$	$2\frac{15}{16}$			
	5	2	$3\frac{1}{16}$			
6	1	$1\frac{3}{8}$	$2\frac{13}{16}$	$3\frac{1}{8}$	$\frac{7}{8}$	$1\frac{3}{16}$
	2, 5, 6, 7	4, $2\frac{1}{2}$, 3 & $3\frac{1}{2}$	$3\frac{7}{16}$			
	3	$1\frac{3}{4}$	$3\frac{1}{16}$			
	4	2	$3\frac{3}{16}$			
7 - 8	1	$1\frac{3}{8}$	$2\frac{13}{16}$	$3\frac{1}{4}$	$\frac{7}{8}$	$1\frac{3}{16}$
	2, 5, 6, 7, 8, 9 & 0	$5\frac{1}{2}$, $2\frac{1}{2}$, 3, $3\frac{1}{2}$, 4, $4\frac{1}{2}$ & 5	$3\frac{7}{16}$			
	3	$1\frac{3}{4}$	$3\frac{1}{16}$			
	4	2	$3\frac{3}{16}$			
10	1	$1\frac{3}{4}$	$3\frac{1}{8}$	$4\frac{1}{8}$	$1\frac{3}{16}$	$1\frac{1}{2}$
	3	2	$3\frac{1}{4}$			
	4, 5, 6, 7, 8, 9 & 0	$2\frac{1}{2}$, 3, $3\frac{1}{2}$, 4, $4\frac{1}{2}$, 5 & $5\frac{1}{2}$	$3\frac{1}{2}$			
12	1	2	$3\frac{1}{4}$	$4\frac{5}{8}$	$1\frac{3}{16}$	$1\frac{1}{2}$
	3, 4, 5, 6, 7, 8 & 9	$2\frac{1}{2}$, 3, $3\frac{1}{2}$, 4, $4\frac{1}{2}$, 5 & $5\frac{1}{2}$	$3\frac{1}{2}$			
14	All	All	$3\frac{13}{16}$	$5\frac{1}{2}$	$1\frac{9}{16}$	$1\frac{7}{8}$

Stroke Data

Parker cylinders are available in any practical stroke length. The following information should prove helpful to you in selecting the proper stroke for your cylinder application.

Stroke Tolerances – Stroke length tolerances are required due to build-up of tolerances of piston, head, cap and cylinder body. Standard production stroke tolerances run +1/32" to -1/64" up to 20" stroke, +1/32" to -.020" for 21" to 60" stroke and +1/32" to -1/32" for greater than 60" stroke. For closer tolerances on stroke length, it is necessary to specify the required tolerance plus the operating pressure and temperature at which the cylinder will operate. Stroke tolerances smaller than .015" are not generally practical due to elasticity of cylinders. If machine design requires such close tolerances, use of a stroke adjuster (below) may achieve the desired result.



Tie Rod Supports

Rigidity of Envelope – The pre-stressed tie rod construction of Parker cylinders has advantages in rigidity within the limits of the cylinder tube to resist buckling. For long stroke cylinders within practical limits, Parker provides exclusive TIE ROD SUPPORTS (see table below) which move the tie rod centerlines radially outward (US patent number 3011844).

Standard tie rod supports are kept within the envelope dimensions of the head and cap, and generally do not interfere with mounting a long cylinder.

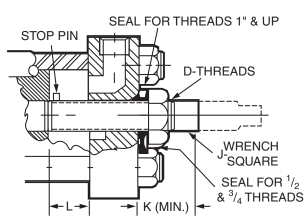
Number of Supports Required	Bore	Stroke (inches)											
		36	48	60	72	84	96	108	120	132	144	156	168
	1	—	1	1	1	2	Consult Factory						
	1½	—	—	1	1	1	2	2	2	3	3	3	4
	2	—	—	—	1	1	1	1	2	2	2	2	3
	2½	—	—	—	—	—	1	1	1	1	1	2	2
	3¼	—	—	—	—	—	—	—	1	1	1	1	1
	4	—	—	—	—	—	—	—	—	—	1	1	1

Note: 5" through 14" bore sizes — no supports required.

Stroke Adjusters

Stroke Adjusters – For the requirement where adjusting the stroke is specified. Parker has several designs to offer, one of which is illustrated below. This is suitable for infrequent adjustment and is economical.*

Bore Size Series 2A, MA, 3L	D	J	K	L (Max.)
1 1/2, 2	1/2 - 20	5/16	15/16	5
2 1/2, 3 1/4, 4	3/4 - 16	7/16	1 1/4	8
5, 6	1 - 14	5/8	1 11/16	9
8	1 1/2 - 12	15/16	2 1/8	18
10	2-12	1 5/16	2 11/16	20
12, 14	2 1/2 - 12	1 11/16	3 1/8	20
—	3-12	2	3 1/4	20
—	3 1/2 - 12	2 3/8	3 1/2	20



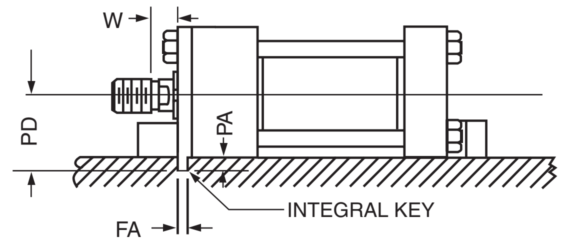
Here a "retracting stroke adjuster" must be called for in specifications, and the length *of the adjustment must be specified*.

Where frequent adjustment or cushions at the cap end are required, other designs are available according to application needs. Please contact Actuator Division for more information.

*Infrequent is defined by positioning the retract stroke in a couple of attempts at original machine set up. The frequent stroke adjuster is recommended for adjustments required after the original equipment has been adjusted by the original machine manufacturer.

Thrust Key Mountings

Thrust key mountings eliminate the need of using fitted bolts or external keys on side mounted cylinders. Parker cylinders in mounting styles CP and FP can be provided with the gland retainer plate extended below the mounting side of the cylinder (see illustration below). This extended retainer plate can then be fitted into a keyway milled into the mounting surface of the machine member. This is referred to as the "P" Modification of any side mounting style.



Series 2A and 2AN

Bore	Dim. FA	Dim. PA	Dim. PD Mtg. Styles CP, FP
1	.312 +.000 -.002	3/16	15/16
1 1/2			1 3/16
2			1 7/16
2 1/2			1 11/16
3 1/4	.562 +.000 -.002	5/16	2 3/16
4			2 9/16
5			3 1/16
6	.687 +.000 -.002	3/8	3 5/8

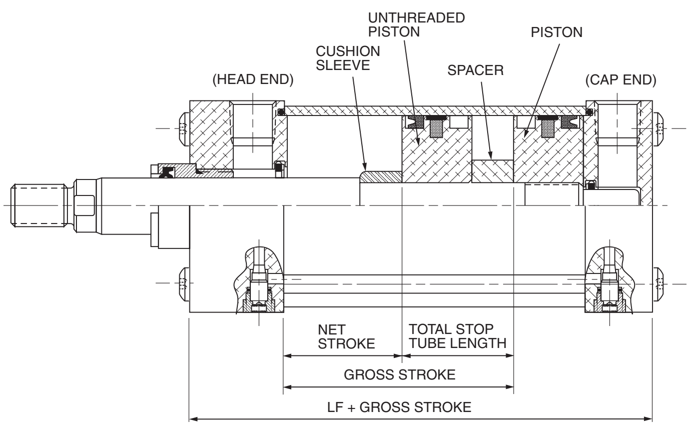
Parker
Parker Hannifin Corporation
Actuator Division
Wadsworth, Ohio USA

Stop Tubing

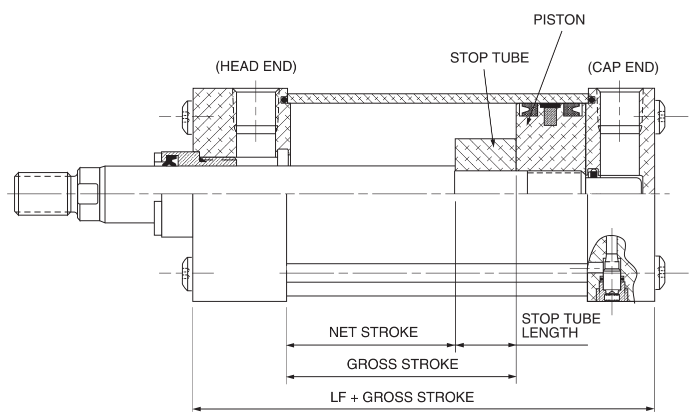
Long stroke cylinders, fixed or pivot mounted, tend to jack-knife or buckle on push load applications, resulting in high bearing loading at the rod gland or piston. Use of a stop tube to lengthen the distance between the gland and piston when cylinder rod is fully extended is recommended to reduce these bearing loads. The drawing below shows stop tube construction for fluid power cylinders. Refer to chart on next page to determine stop tube length.

When specifying cylinders with long stroke and stop tube, be sure to call out the net stroke and the length of the stop tube. Machine design can be continued without delay by laying in a cylinder equivalent in length to the NET STROKE PLUS STOP TUBE LENGTH, which is referred to as GROSS STROKE.

Refer to the next page to determine stop tube length.



Double piston design is supplied on air cylinders with cushion head end or both ends.



This design is supplied on cushioned cap or non-cushioned cylinders.

Cushion Selection

Cushions are required when cylinder piston rod speed exceeds 4" per second.

Mounting Classes

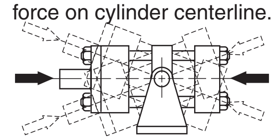
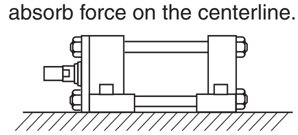
Standard mountings for fluid power cylinders fall into three basic groups. The groups can be summarized as follows:

Group 1 – Straight Line Force Transfer with fixed mounts which absorb force on cylinder centerline.

Group 2 – Pivot Force Transfer. Pivot mountings permit a cylinder to change its alignment in one plane.

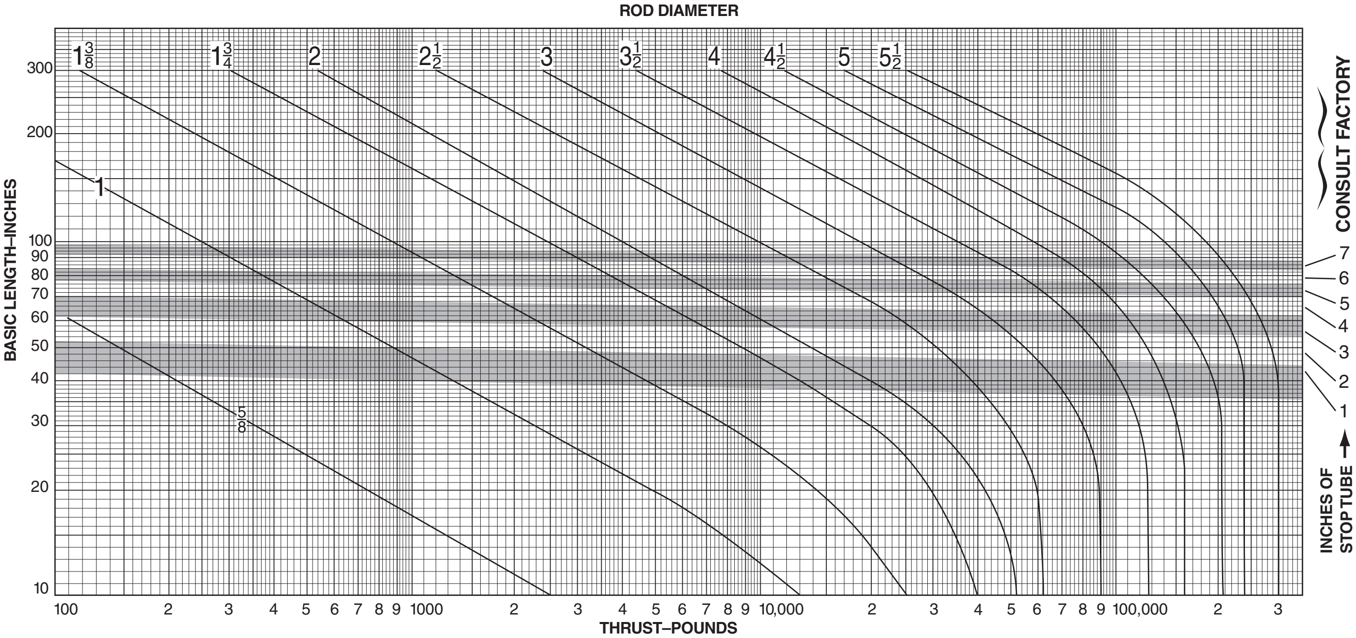
Group 3 – Straight Line Force Transfer with fixed mounts which do not absorb force on cylinder centerline.

Because a cylinder's mounting directly affects the maximum pressure at which the cylinder can be used, the charts below should be helpful in the selection of the proper mounting combination for your application. Stroke length, piston rod connection to load, extra piston rod length over standard, etc. should be considered for thrust loads. Alloy steel mounting bolts are recommended for all mounting styles, and thrust keys are recommended for Group 3.

	GROUP 1
	FIXED MOUNTS which absorb force on cylinder centerline. 
HEAVY-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	Mtg. Styles TC Mtg. Styles TB
MEDIUM-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	Mtg. Styles H, HB Mtg. Styles J, JB
LIGHT-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	Mtg. Styles J, JB Mtg. Styles H, HB
	GROUP 2
	PIVOT MOUNTS which absorb force on cylinder centerline. 
HEAVY-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	**Mtg. Styles DD, D** Mtg. Styles BB, BC*, DD, D, DB
MEDIUM-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	Mtg. Styles BB, BC*, BE Mtg. Styles BB, BC*, BE
	GROUP 3
	FIXED MOUNTS which do not absorb force on the centerline. 
HEAVY-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	Mtg. Styles C Mtg. Styles C
MEDIUM-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	Mtg. Styles F Mtg. Styles F
LIGHT-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	Mtg. Styles CB*† Mtg. Styles CB*†

* Not available for 2A/2AN Series.
† Mounting style CB recommended for maximum pressure of 150 psi.

Piston Rod — Stroke Selection Chart



How to Use the Chart

The selection of a piston rod for thrust (push) conditions requires the following steps:

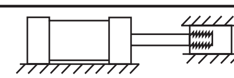
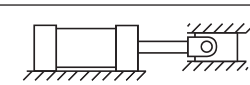
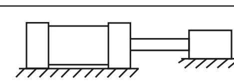
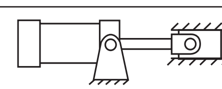
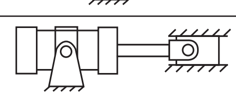
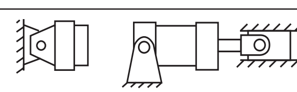
1. Determine the type of cylinder mounting style and rod end connection to be used. Then consult the chart below and find the "stroke factor" that corresponds to the conditions used.
2. Using this stroke factor, determine the "basic length" from the equation:

   Basic Length = Actual Stroke x Stroke Factor

   The graph is prepared for standard rod extensions beyond the face of the gland retainers. For rod extensions greater than standard, add the increase to the stroke in arriving at the "basic length."
3. Find the load imposed for the thrust application by multiplying the full bore area of the cylinder by the system pressure.
4. Enter the graph along the values of "basic length" and "thrust" as found above and note the point of intersection:
   - A) The correct piston rod size is read from the diagonally curved line labeled "Rod Diameter" next *above* the point of intersection.
   - B) The required length of stop tube is read from the right of the graph by following the shaded band in which the point of intersection lies.
   - C) If required length of stop tube is in the region labeled "consult factory," submit the following information for an individual analysis:
     1) Cylinder mounting style.
     2) Rod end connection and method of guiding load.
     3) Bore, required stroke, length of rod extension (Dim. "LA") if greater than standard, and series of cylinder used.
     4) Mounting position of cylinder. (Note: If at an angle or vertical, specify direction of piston rod.)
     5) Operating pressure of cylinder if limited to less than standard pressure for cylinder selected.

Warning ⚠

Piston rods are not normally designed to absorb bending moments or loads which are perpendicular to the axis of piston rod motion. These additional loads can cause the piston rod end to fail. If these types of additional loads are expected to be imposed on the piston rods, their magnitude should be made known to our Engineering Department so they may be properly addressed. Additionally, cylinder users should always make sure that the piston rod is securely attached to the machine member.

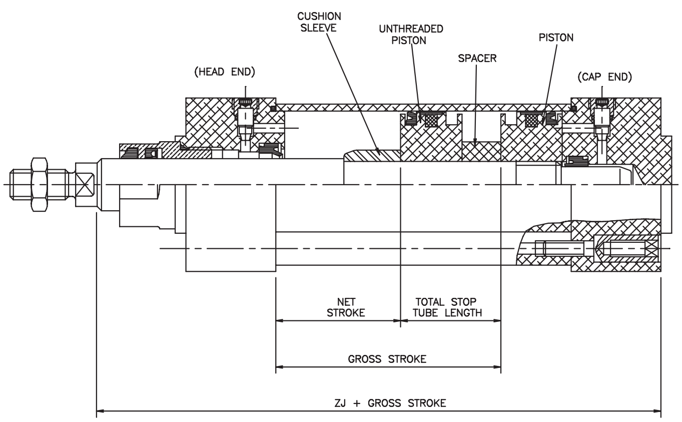
Recommended Mounting Styles for Maximum Stroke and Thrust Loads	Rod End Connection	Case		Stroke Factor
Groups 1 or 3 Long stroke cylinders for thrust loads should be mounted using a heavy-duty mounting style at one end, firmly fixed and aligned to take the principal force. Additional mounting should be specified at the opposite end, which should be used for alignment and support. An intermediate support may also be desirable for long stroke cylinders mounted horizontally. See page 11 under "Tie Rod Supports — Rigidity of Envelope" for a guide. Machine mounting pads can be adjustable for support mountings to achieve proper alignment.	Fixed and Rigidly Guided	I		0.50
	Pivoted and Rigidly Guided	II		0.70
	Supported but not Rigidly Guided	III		2.00
Group 2 Style D — Trunnion on Head	Pivoted and Rigidly Guided	IV		1.00
Style DD — Intermediate Trunnion	Pivoted and Rigidly Guided	V		1.50
Style DB — Trunnion on Cap or Style BB — Clevis on Cap	Pivoted and Rigidly Guided	VI		2.00

Stop Tubing

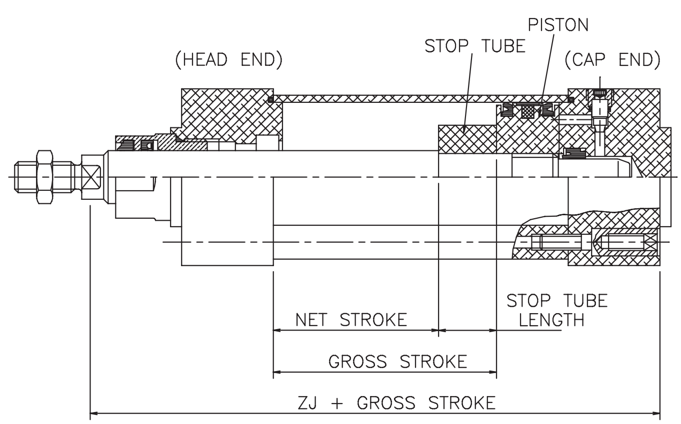
Long stroke cylinders, fixed or pivot mounted, tend to jackknife or buckle on push load applications, resulting in high bearing loading at the rod gland or piston. Use of a stop tube to lengthen the distance between the gland and piston when cylinder rod is fully extended is recommended to reduce these bearing loads. The drawing below shows stop tube construction for fluid power cylinders. Refer to piston rod/stroke selection chart to determine stop tube length.

When specifying cylinders with long stroke and stop tube, be sure to call out the net stroke and the length of the stop tube. Machine design can be continued without delay by laying in a cylinder equivalent in length to the NET STROKE PLUS STOP TUBE LENGTH, which is referred to as GROSS STROKE.

Refer to piston rod/stroke selection chart to determine stop tube length.



Double piston design is supplied on air cylinders with cushion head end or both ends.



This design is supplied on cushioned cap or non-cushioned cylinders.

Cushion Selection

Cushions are required when cylinder piston rod speed exceeds 100mm per second.

Mounting Classes

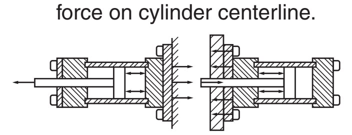
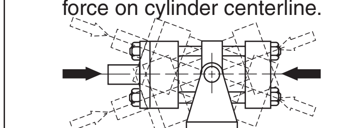
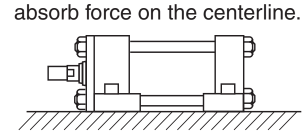
Standard mountings for fluid power cylinders fall into three basic groups. The groups can be summarized as follows:

Group 1 – Straight Line Force Transfer with fixed mounts which absorb force on cylinder centerline.

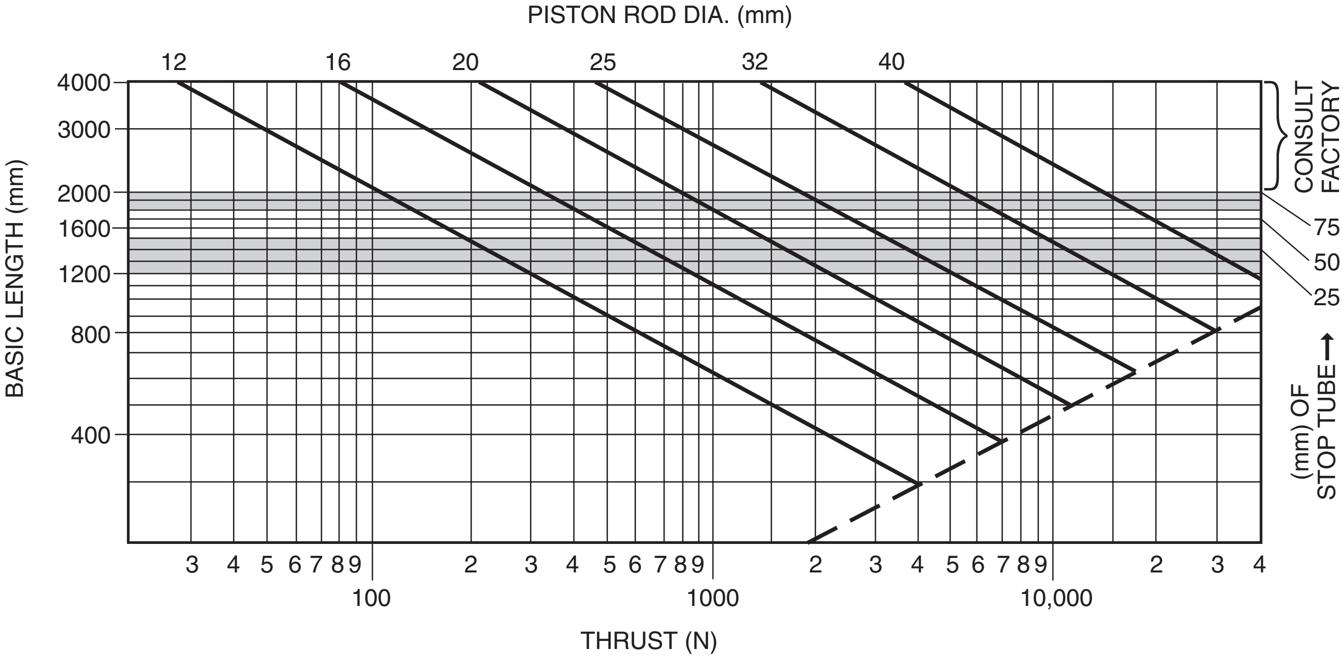
Group 2 – Pivot Force Transfer. Pivot mountings permit a cylinder to change its alignment in one plane.

Group 3 – Straight Line Force Transfer with fixed mounts which do not absorb force on cylinder centerline.

Stroke length, piston rod connection to load, extra piston rod length over standard, etc. should be considered for thrust loads. Alloy steel mounting bolts are recommended for all mounting styles for Group 3.

	GROUP 1
	FIXED MOUNTS which absorb force on cylinder centerline. 
HEAVY-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	ISO MX2 Mtg. (Style TC) ISO MX3 Mtg. (Style TB)
MEDIUM-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	ISO MF2 Mtg. (Style H) ISO MF2 Mtg. (Style J)
LIGHT-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	ISO MF2 Mtg. (Style H) ISO MF2 Mtg. (Style J)
	GROUP 2
	PIVOT MOUNTS which absorb force on cylinder centerline. 
HEAVY-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	ISO MP2, MT4 Mtg. (Styles BC, DD)
MEDIUM-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	ISO MP2, MP4, MP7, MP6 Mtgs. (Styles BC, BE, BJ, SB)
	GROUP 3
	FIXED MOUNTS which do not absorb force on the centerline. 
LIGHT-DUTY SERVICE For Thrust Loads ______ For Tension Loads ______	ISO MS1 Mtg. (Style CB)

Piston Rod – Stroke Selection Chart



How To Use The Chart

The selection of a piston rod for thrust (push) conditions requires the following steps:

1. Determine the type of cylinder mounting style and rod end connection to be used. Then consult the chart below and find the "stroke factor" that corresponds to the conditions used.
2. Using this stroke factor, determine the "basic length" from the equation:

   Basic Length = Actual Stroke x Stroke Factor

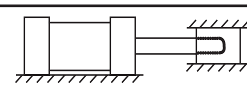
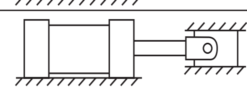
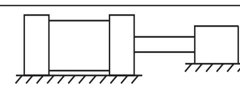
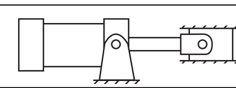
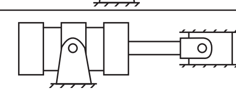
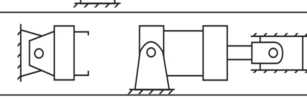
   The graph is prepared for standard rod extensions beyond the face of the head. For rod extensions greater than standard, add the increase to the stroke in arriving at the "basic length."
3. Find the load imposed for the thrust application by multiplying the full bore area of the cylinder by the system pressure.
4. Enter the graph along the values of "basic length" and "thrust" as found above and note the point of intersection:

a) The correct piston rod size is read from the diagonally curved line labeled "Rod Diameter" next *above* the point of intersection.

b) The required length of stop tube is read from the right of the graph by following the shaded band in which the point of intersection lies.

c) If required length of stop tube is in the region labeled "consult factory," submit the following information for an individual analysis.

1) Cylinder mounting style.
2) Rod end connection and method of guiding load.
3) Bore, required stroke, length of rod extension (Dim. WH) if greater than standard, and series of cylinder used.
4) Mounting position of cylinder. (Note: if at an angle or vertical, specify direction of piston rod.)
5) Operating pressure of cylinder if limited to less than standard pressure for cylinder selected.

Recommended Mounting Styles for Maximum Stroke and Thrust Loads	Rod End Connection	Case	Stroke Factor
Groups 1 or 3 Long stroke cylinders for thrust loads should be mounted using a heavy-duty mounting style at one end, firmly fixed and aligned to take the principal force. Additional mounting should be specified at the opposite end, which should be used for alignment and support. An intermediate support may also be desirable for long stroke cylinders mounted horizontally.	FIXED AND RIGIDLY GUIDED	I 	0.50
	PIVOTED AND RIGIDLY GUIDED	II 	0.70
	SUPPORTED BUT NOT RIGIDLY GUIDED	III 	2.00
Group 2 Style MP7 (BJ)	PIVOTED AND RIGIDLY GUIDED	IV 	1.00
Style MT4 (DD)	PIVOTED AND RIGIDLY GUIDED	V 	1.50
Style MP2 (BC), MP4 (BE), MP6 (SB)	PIVOTED AND RIGIDLY GUIDED	VI 	2.00

Parker Hannifin Corporation
Actuator Division
Wadsworth, Ohio USA

Cushion ratings for **Air Cylinders Only** are described in **Table b-2** and **Graph b-1**. To determine whether a cylinder will adequately stop a load without damage to the cylinder, the weight of the load (including the weight of the piston and the piston rod from **Table b-1**) and the maximum speed of the piston rod must first be determined. Once these two factors are known, the **Kinetic Energy Graph** may be used. Enter the graph at its base for the value of weight determined, and project vertically to the required speed value. The point of intersection of these two lines will be the cushion rating number required for the application.

To determine the total load to be moved, the weight of the piston and rod must be included.

Total Weight = weight of the piston and non-stroke rod length (column 1) + weight of the rod per inch of stroke x the inches of stroke (Column 2) + the load to be moved.

Series 2A/2AN Cylinder Weight Table

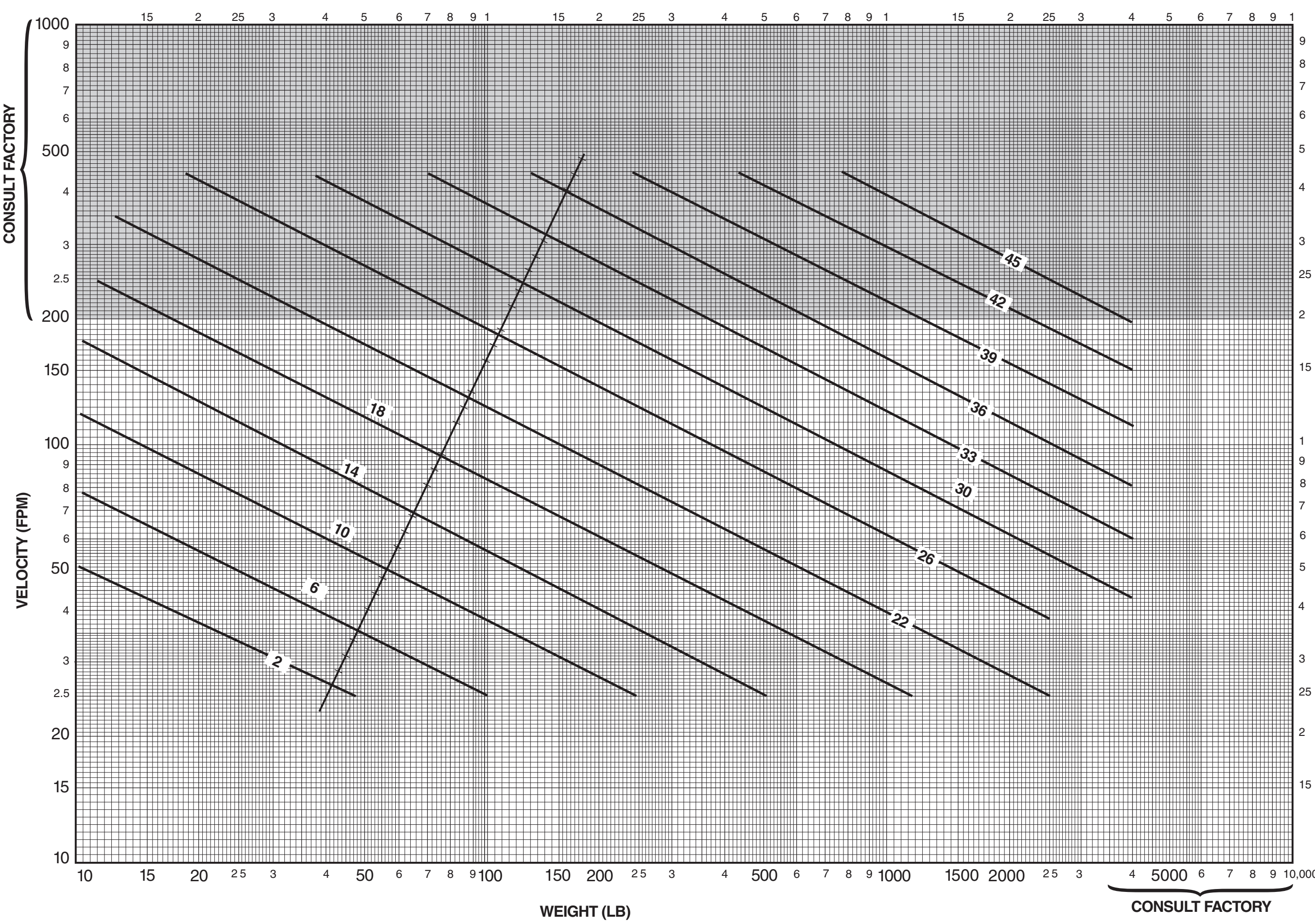
Table b-1

Bore Dia.	Column 1 Basic Wgt. (lbs.) for Piston & Non-Stroke Rod	Rod Dia.	Column 2 Basic Wgt. (lbs.) for 1" Stroke
1 1/2	1.5	5/8	0.087
2	3.0	1	0.223
2 1/2	5.4	1 3/8	0.421
3 1/4	8.3	1 3/4	0.682
4	14.2	2	0.89
5	29.0	2 1/2	1.39
6	41.0	3	2.0
8	89.0	3 1/2	2.73
10	115.0	4	3.56
12	161.0	5	5.56
14	207.0	5 1/2	6.73

Example: *a 3 1/4" bore cylinder, having a 1" diameter rod and 25" stroke; load to be moved is 85 pounds. Total load to be moved is then 8.3 lbs. + 0.223 lbs./in. x 25 in. + 85 lbs. or a total of 99 lbs.*

Kinetic Energy Graph – Air Cylinders

Graph b-1



Now refer to **Table b-2** and find the cushion ratings, using bore size and rod diameter of the cylinder selected. If a simple circuit is used, with no meter out or speed control, use the “Rating with No Back Pressure” column values. If a meter out or speed control is to be used, use the “Rating with Back Pressure” column values. If the cushion rating found in **Table b-2** below is **greater** than the number determined in **Graph b-1,** then the cylinder will stop the load adequately. If the cushion rating in **Table b-2** is **smaller** than the number found in **Graph b-1**, then a larger bore cylinder should be used. In those applications where back pressures exist in the exhaust lines, it is possible to exceed the cushion ratings shown in **Table b-2**. In these cases, consult the factory and advise the amount of back pressure.

Air Cylinder Cushion Ratings Table

Table b-2

Bore Dia.	Rod Dia.	Rating with No Back Pressure	Rating with Back Pressure
1 1/2	Cap End	12	17
	5/8	8	14
	1	3	8
2	Cap End	14	20
	5/8	12	18
	1	9	15
	1 3/8	6	11
2 1/2	Cap End	17	23
	5/8	14	20
	1	14	19
	1 3/8	12	18
	1 3/4	8	13
3 1/2	Cap End	21	26
	1	18	24
	1 3/8	17	23
	1 3/4	16	22
	2	13	19
4	Cap End	23	28
	1	20	27
	1 3/8	20	26
	1 3/4	19	25
	2	17	23
	2 1/2	17	22
5	Cap End	26	31
	1	23	28
	1 3/8	23	28
	1 3/4	22	28
	2	20	26
	2 1/2	19	25
	3	18	24
	3 1/2	15	20
6	Cap End	26	31
	1 3/8	26	31
	1 3/4	26	31
	2	24	29
	2 1/2	24	29
	3	22	28
	3 1/2	21	27
	4	20	26
7	Cap End	28	33
	1 3/8	28	33
	1 3/4	28	33
	2	26	31
	2 1/2	25	30
7	3	24	30
	3 1/2	24	30
	4	23	29
	4 1/2	22	28
	5	21	27
8	Cap End	29	35
	1 3/8	29	35
	1 3/4	29	34
	2	27	33
	2 1/2	26	32
	3	26	32
	3 1/2	26	32
	4	25	31
	5	23	29
	5 1/2	22	28
10	Cap End	33	39
	1 3/4	32	38
	2	31	37
	2 1/2	31	36
	3	30	36
	3 1/2	30	36
	4	30	36
	5	28	34
	5 1/2	27	33
12	Cap End	35	41
	2	33	39
	2 1/2	33	38
	3	33	38
	3 1/2	32	38
	4	32	38
	5	31	36
	5 1/2	31	36
14	Cap End	38	43
	2 1/2	37	42
	3	36	42
	3 1/2	36	41
	4	36	41
	5	35	40
	5 1/2	34	40

Inch Based Cylinders
Air Requirement Per Inch of Cylinder Stroke

The amount of air required to operate a cylinder is determined from the volume of the cylinder and its cycle in strokes per minute. This may be determined by use of the following formulae which apply to a single-acting cylinder.

$$V = \frac{3.1416\ L\ D^2}{4} \qquad C = \frac{f V}{1728}$$

Where: V = Cylinder volume, cu. in.
L = Cylinder stroke length, in.
D = Internal diameter of cylinder in.
C = Air required, cfm
f = Number of strokes per minute

The air requirements for a double-acting cylinder is almost double that of a single-acting cylinder, except for the volume of the piston rod.

The air flow requirements of a cylinder in terms of cfm should not be confused with compressor ratings which are given in terms of free air. If compressor capacity is involved in the consideration of cylinder air requirements it will be necessary to convert cfm values to free air values. This relationship varies for different gauge pressures.

Thrust (pounds) = operating pressure x area of cylinder bore.

Note: *That on the "out" stroke the air pressure is working on the entire piston area but on the "in" stroke the air pressure works on the piston area less the rod area.*

Graph b-2 and **b-3** offer a simple means to select pneumatic components for dynamic cylinder applications. It is only necessary to know the force required, the desired speed and the pressure which can be maintained at the inlet to the air preparation system. The graphs assume average conditions relative to air line sizes, system layout, friction, etc. At higher speeds, consider appropriate cushioning of cylinders.

The general procedure to follow when using these graphs is:

1. Select the appropriate graph depending upon the pressure which can be maintained to the system – **Graph b-2** for 100 psig and **Graph b-3** for 80 psig.
2. Determine appropriate cylinder bore. Values underneath the diagonal cylinder bore lines indicate the maximum recommended dynamic thrust developed while the cylinder is in motion. The data in the table at the bottom of each graph indicates available static force applications in which clamping force is a prime consideration in determing cylinder bore. Please reference table number b-3 and b-4 for approximate thrust developed at a given operating pressure.

Graph b-2

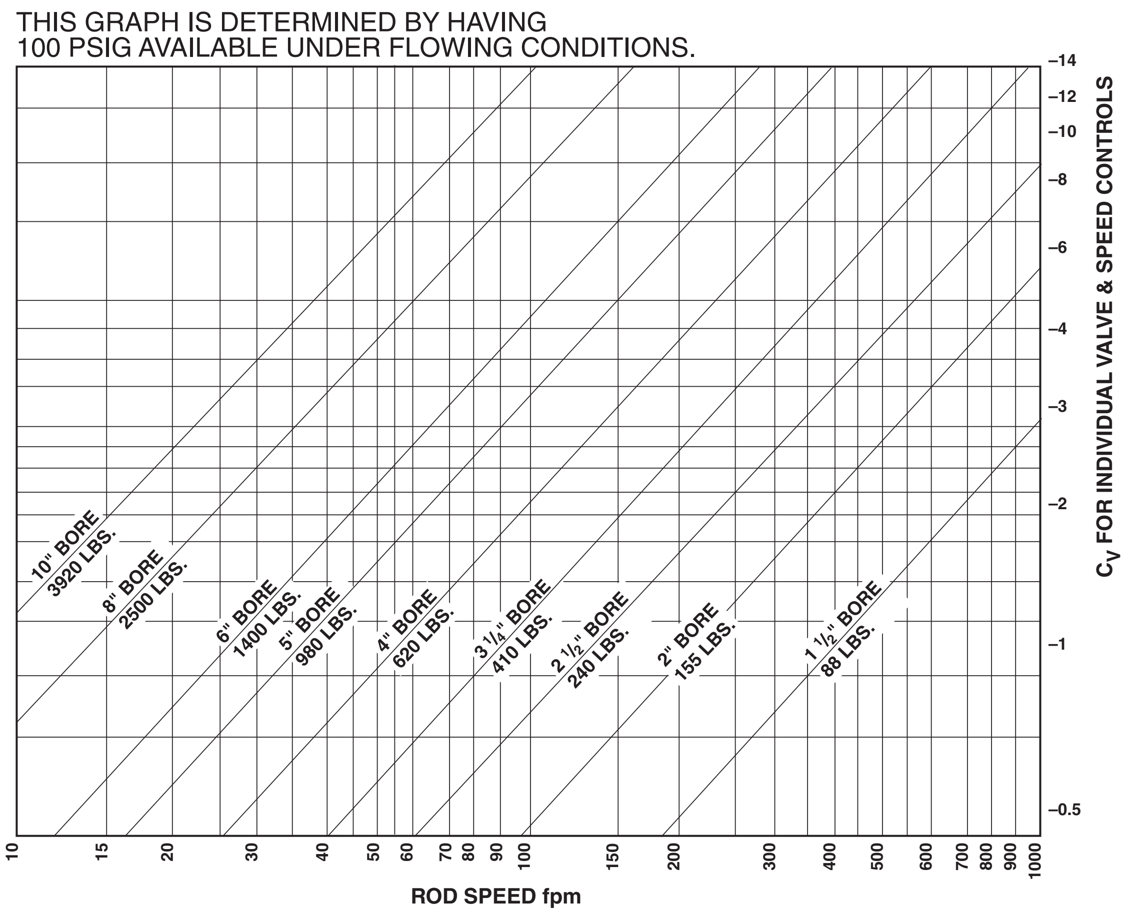


Table b-3
Thrust Developed

Bore Size	1½"	2"	2½"	3¼"	4"	5"	6"	8"	10"
Dynamaic Thrust (lbs.)	88	155	240	410	620	980	1400	500	3920
Static Thrust (lbs.)	177	314	491	830	1250	1960	2820	5020	7850

3. Read upward on appropriate rod speed line to intersection with diagonal cylinder bore line. Read right from intersection point to determine the required C_v of the valve and the speed controls. Both the valve and speed controls must have this C_v.

The following examples illustrate use of the graphs:

Example 1: *Assume it is necessary to raise a 900-pound load 24 inches in two seconds. With 100 psig maintained at the inlet to the air preparation system, use* **Graph b-2**. *The 5-inch bore cylinder is capable of developing the required thrust while in motion. Since 24 inches in two seconds is equal to 60 fpm, read upward on the 60 fpm line to the intersection of the 5-inch bore diagonal line. Reading to the right indicates that the required valve and speed controls must each have a C_v of over 1.9.*

Example 2: *Assume similar conditions to Example 1 except that only 80- psig will be available under flowing conditions. Using* **Graph b-3***, a 6-inch bore cylinder is indicated. Read upward on the 60 fpm line to the intersection point. Interpolation of the right-hand scale indicates a required valve and speed control C_v of over 2.8.*

Example 3: *Assume similar conditions to Example 1 except that the load is being moved in a horizontal plane with a coefficient of sliding friction of 0.2. Only a 180-pound thrust is now required (900 lb. x 0.2). Consult* **Graph b-3**. *The 2½ inch bore cylinder will develop sufficient thrust, and at 60 fpm requires a valve and speed control C_v of about 0.5.*

Graph b-3

THIS GRAPH IS DETERMINED BY HAVING
80 PSIG AVAILABLE UNDER FLOWING CONDITIONS.

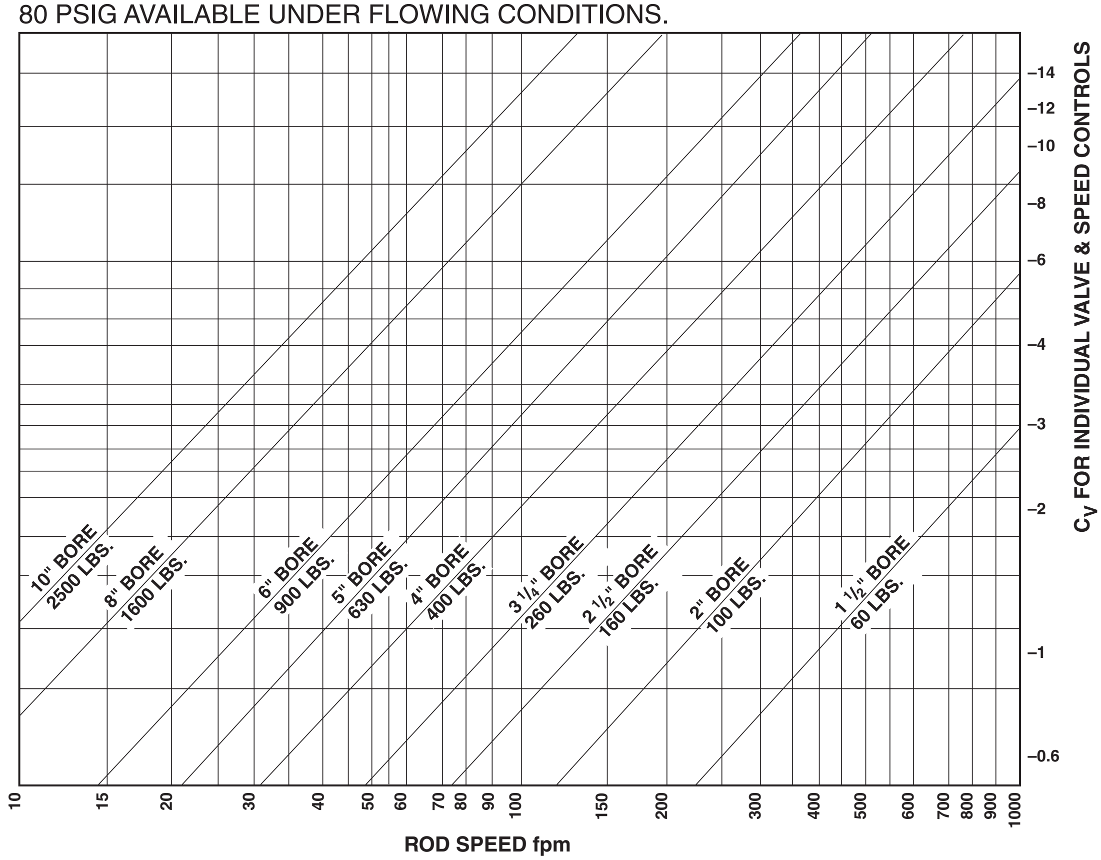


Table b-4
Thrust Developed

Bore Size	1½"	2"	2½"	3¼"	4"	5"	6"	8"	10"
Dynamic Thrust (lbs.)	60	100	160	260	400	630	900	1600	2500
Static Thrust (lbs.)	141	251	393	663	1000	1570	2260	4010	6280

Rod End Data

Rod end dimension symbols as shown comply with the National Fluid Power Association dimensional code. The following chart indicates the symbols used in this catalog.

Description	Symbol
Thread diameter and pitch	KK or CC
Length of thread	A
Length of rod extension from face of gland retainer to end of retracted rod	LA or LAF (Male Thread) W or WF (Female Thread)

Three rod ends for Parker cylinders are offered as shown on the dimension pages of this catalog. They are Parker styles 4, 8 and 9, and all three are optional without price penalty. Styles 4 and 8 are supplied with high strength rolled thread studs on piston rods through 2" diameter. Longer studs in Parker standard sizes are available, see table below.

Warning ⚠

Piston rods are not normally designed to absorb bending moments or loads which are perpendicular to the axis of piston rod motion. These additional loads can cause the piston rod end to fail. If these types of additional loads are expected to be imposed on the piston rods, their magnitude should be made known to our Engineering Department so they may be properly addressed. Additionally, cylinder users should always make sure that the piston rod is securely attached to the machine member.

On occasion cylinders are ordered with double rods. In some cases a stop is threaded onto one of the piston rods and used as an external stroke adjuster. This can cause a potential safety concern and can also lead to premature piston rod failure. The external stop will create a pinch point and the cylinder user should consider appropriate use of guards. If an external stop is not parallel to the final contact surface it will place a bending moment on the piston rod. An external stop will also negate the effect of a cushion and will subject the piston rod to an impact loading. These two (2) conditions can cause piston rod failure. The use of external stroke adjusters should be reviewed with our Engineering Department.

Piston Rod End Threads

Standard piston rod end thread lengths are shown as dimension "A" in Catalog dimension pages. Special rod end threads which are two times standard length can be supplied at a small extra cost. Available thread lengths are shown in the table below. To order, add suffix "2" to piston rod model number code and specify as Style #42 or Style #82.

Optional Piston Rod End Studs

Piston Rod Dia.	Rod End Thread Style #42		Rod End Thread Style #82	
	Thread Dia. & Pitch (KK)	Length (= 2 × A)	Thread Dia. & Pitch (CC)	Length (= 2 × A)
5/8	7/16-20	1 1/2	1/2-20	1 1/2
1	3/4-16	2 1/4	7/8-14	2 1/4
1 3/8	1-14	3 1/4	1 1/4-12	3 1/4
1 3/4	1 1/4-12	4	1 1/2-12	4
2	1 1/2-12	4 1/2	1 3/4-12	4 1/2

International Rod End Threads

Piston rod threads to meet international requirements are available at extra cost. Parker cylinders can be supplied with British standard fine (W) or metric (M). To order, specify in model number. For dimensions, consult factory.

Special Rod Ends

If a rod end configuration other than the standard styles 4, 8 and 9 is required, such special rod ends can be provided. The designation "Style 3" is assigned to such specials and is incorporated in the cylinder model number. To order, specify "Style 3" and give desired dimensions for KK; A; LA, LAF, W, or WF. If otherwise special, send a dimensioned sketch.

Special Assemblies from Standard Parts

Each dimensioned drawing in this catalog has position numbers shown on the end view to identify the four sides of the cylinder. These aid in communications and simplify the writing of specifications that cover changes in port positions, etc. Following are several suggested special assemblies that can be made up from standard parts.

a) By calling out the position numbers for the desired locations for head and cap ports, many mounting styles can be assembled with ports located at 90° or 180° from standard. In such special assemblies, the cushion needle and check valves are also repositioned since their relation with the port position does not change.

b) The cushion needle valve is interchangeable with the check valve in the cylinder heads. The cushion needle valve can be assembled on side position 4 with the check valve on side 2 for most mounting styles when the port is in the standard side position 1.

   On mounting styles D, DB and DD, the cushion needle valves are provided only on the side position 3 on the head or cap which accommodates the mounting. The opposite head or cap can be rotated.

c) Standard mountings in different combinations can be provided: for example Style J mounting on head end with Style C on the cap end. This would be made up from standard parts and would be designated Model (bore size) **JC**-2AU14A (stroke).

Single-Acting Cylinders

Double-acting cylinders are supplied as standard. They can also be used a single-acting cylinders where fluid force is applied to only one side of the piston, with the load or other external forces acting to "return" the piston after pressure is exhausted.

Spring-Returned, Single-Acting Cylinders – Single-acting, spring-returned models can also be provided. Load conditions and friction factors must be considered in supplying the proper spring for the application. In addition, it is necessary that information be supplied as to which side of the piston the spring should act upon. Specify "Spring to return piston rod" or "Spring to advance piston rod."

On longer stroke spring-returned cylinders, it is recommended that tie rod extensions be specified on the cylinder end in which the spring is located so that the cap or head against which the spring is acting can be "backed-off" slowly until compression of the spring is relieved. In such cases it should also be specified that the tie rod nuts be welded to the tie rods at the opposite end of the cylinder to further insure safe disassembly.

Consult factory when ordering spring-returned cylinders.

Modifications

The following modifications can be supplied on most Parker cylinders.

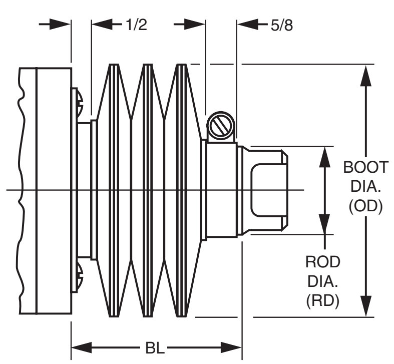
Metallic Rod Wiper

When specified metallic rod wipers can be supplied instead of the standard synthetic rubber wiperseal. Recommended in applications where contaminants tend to adhere to the extended piston rod and would damage the standard wiper or wiperseal. Installation of metallic rod wiper does not affect cylinder dimensions. It is available at extra cost. Please contact Actuator Division for more information.

Rod End Boots

Most Parker cylinders have a hardened bearing surface on the standard piston rod to resist external damage, and are equipped with a high efficiency wiper to remove external dust and dirt. Exposed piston rods that are subjected to contaminants with air hardening properties, such as paint, should be protected. In such applications, the use of a collapsing cover should be considered. This is commonly referred to as a "boot". Calculate the longer rod end required to accommodate the collapsed length of the boot from the following data.

RD	1/2	5/8	1	1 3/8	1 3/4	2	2 1/2	3	3 1/2	4	5	5 1/2
OD	2 1/4	2 1/4	2 5/8	3	3 3/8	3 3/4	4 3/8	5 1/8	5 5/8	6 1/4	7	7 1/2
LF	.13	.13	.13	.13	.13	.13	.13	.10	.10	.10	.10	.10



To determine extra length of piston rod required to accommodate boot, calculate:

BL = Stroke x LF + 1 1/8"

2A/2AN cylinders:
BL + std LA (male rod end) or W (female rod end) dimension = length of piston rod to extend beyond the retainer.

2MA cylinders:
BL + std LAF (male rod end) or WF (female rod end) dimension = length of piston rod to extend beyond the head face.

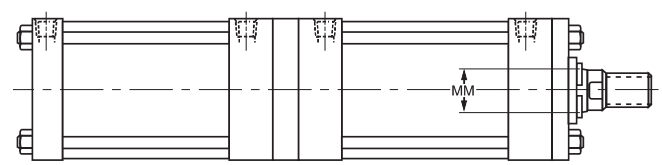
P1D cylinders:
BL + std WH dimension = length of piston rod to extend beyond the head face.

Note: Please compare the Boot OD size to the standard E dimension per desired cylinder series and bore. This may be critical for foot mounted cylinders.

Rod Boots are available for many other cylinder series. Please contact the Actuator Division for rod boot options.

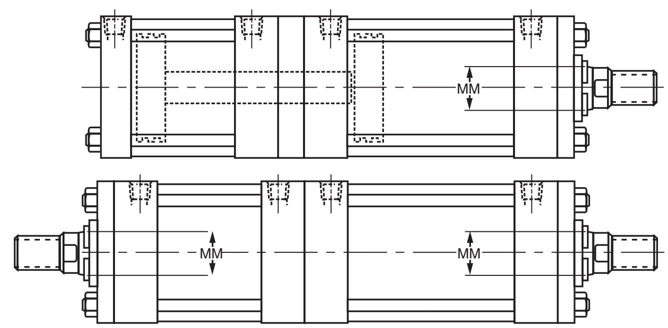
Tandem Cylinders

A tandem cylinder is made up of two cylinders mounted in line with pistons connected by a common piston rod and rod seals installed between the cylinders to permit double acting operation of each. Tandem cylinders allow increased output force when mounting width or height are restricted. Please contact Actuator Division for more information.



Duplex Cylinders

A duplex cylinder is made up of two cylinders mounted in line with pistons not connected and with rod seals installed between the cylinders to permit double acting operation of each. Cylinders may be mounted with piston rod to piston (as shown) or back to back and are generally used to provide three position operation. Please contact Actuator Division for more information.



Design torque

Design torque represents the maximum torque that an actuator must supply in an application. This maximum is the greater of the Demand Torque or the Cushion Torque. If the demand torque exceeds what the actuator can supply, the actuator will either move too slowly or stall. If the cushion torque is too high, the actuator may be damaged by excessive pressure. Demand torque and cushion torque are defined below in terms of load, friction, and acceleration torque.

Equations for calculating demand torque and cushion torque for some general applications are provided on the following pages.

T - Torque

The amount of turning effort exerted by a rotary actuator.

T_D - Demand Torque

This is the torque required from the actuator to do the job and is the sum of the load torque, friction torque, and acceleration torque, multiplied by an appropriate design factor. Design factors vary with the applications and the designers' knowledge.

Equation 4-3) $T_D = T_\alpha + T_f + T_L$

T_L - Load torque

This is the torque required to equal the weight or force of the load. For example, in Fig. 4-8a, the load torque is 563 N•m *(5000 lb-in.)*; in Fig. 4-8b the load torque is zero; in Fig. 4-8c the load torque is 563 N•m *(5000 lb-in.)*. The load torque term is intended to encompass all torque components that aren't included in the friction or acceleration terms.

T_f - Friction torque

This is the torque required to overcome friction between any moving parts, especially bearing surfaces. In Fig. 4-8a, the friction torque is zero for the hanging load; in Fig. 4-8b the friction torque is 775 N•m *(6880 lb-in)* for the sliding load; in Fig. 4-8c the friction torque is zero for the clamp.

Equation 4-4) $T_f = \mu W r$

T_a - Acceleration Torque

This is the torque required to overcome the inertia of the load in order to provide a required acceleration or deceleration. In Fig. 4-8a the load is suspended motionless so there is no acceleration. In Fig. 4-8b, the load is accelerated from 0 to some specified angular velocity. If the mass moment of inertia about the axis of rotation is I and the angular acceleration is a, the acceleration torque is equal to Ia. In Fig. 4-8c there is no acceleration.

Some values for mass moment of inertia are given in Table 4. Some useful equations for determining a are listed in Table 5. Equation 5 below shows the general equation for acceleration torque.

Equation 4-5) $T_\alpha = I\alpha$

T_C - Cushion Torque

This is the torque that the actuator must apply to provide a required deceleration. This torque is generated by restricting the flow out of the actuator (meter-out) so as to create a back pressure which decelerates the load. This back pressure (deceleration) often must overcome both the inertia of the load and the driving pressure (system pressure) from the pump. See applications.

Equation 4-6) $T_c = T_\alpha + \dfrac{P_r V}{\theta} - T_f \pm T_L$

The friction torque T_f reduces the torque the actuator must apply to stop the load. The load torque T_L may add to, or subtract from the torque required from the actuator, depending upon the orientation of the load torque. For example, a weight being swung upward would result in a load torque that is subtracted.

Warning: Rapid deceleration can cause high pressure intensification at the outlet of the actuator. Always insure that cushion pressure does not exceed the manufacturer's pressure rating for the actuator.

KE – Kinetic Energy ($\frac{1}{2} J_m \omega^2$)

This is the amount of energy that a rotating load has. The rotator must be able to stop the load. All products have kinetic energy rating tables. Choose the appropriate deceleration option (i.e., bumper, cushions, shock absorbers, etc.) that meets or exceeds the kinetic energy of the load.

Pages 8-10 excerpted from the Parker Hannifin Design Engineers Handbook.

Demand Torque Examples

A) Example of load torque

The load is held motionless as shown.

$T_D = T_\alpha + T_f + T_L$

$T_\alpha = 0$

$T_f = 0$

$T_L = (500 \text{ lb})(10 \text{ in}) = 5{,}000 \text{ lb-in}$

$T_D = 5{,}000 \text{ lb-in}$

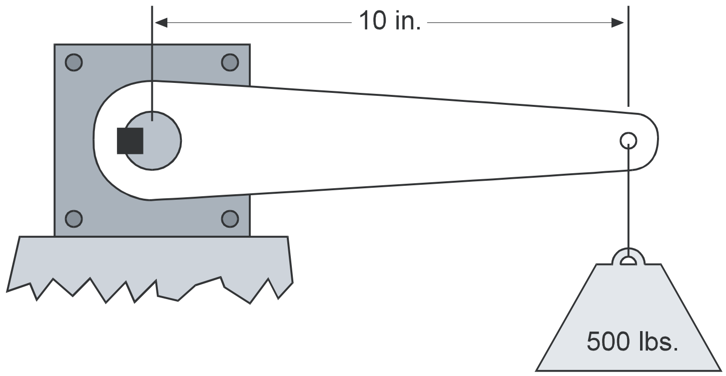


Figure 4-8a

B) Due to friction and acceleration

The 500 lb rotating index table is supported by bearings with a coefficient of friction of 0.25. The table's acceleration a is 2 rad/sec^2. The table's mass moment of inertia I is 2,330 lb-in-sec^2.

$T_D = T_\alpha + T_f + T_L$

$T_\alpha = I\alpha = (2{,}330 \text{ lb-in-sec}^2)(2/\text{sec}^2) = 4{,}660 \text{ lb-in}$

$T_f = \mu W r_b = 0.25\ (500 \text{ lb})(55 \text{ in}) = 6{,}880 \text{ lb-in}$

$T_L = 0$

$T_D = 4{,}660 \text{ lb-in} + 6{,}880 \text{ lb-in} = 11{,}540 \text{ lb-in}$

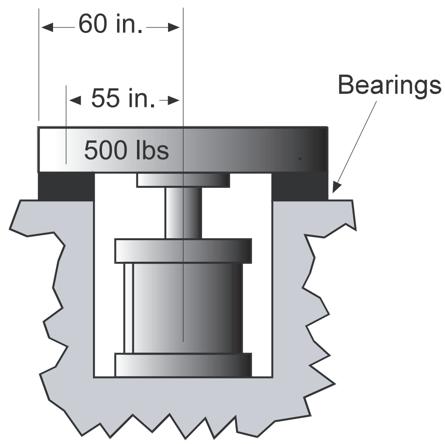


Figure 4-8b

C) Load torque example

$T_D = T_\alpha + T_f + T_L$

$T_\alpha = 0$

$T_f = 0$

$T_L = (500 \text{ lb})(10 \text{ in}) = 5{,}000 \text{ lb-in}$

$T_D = 5{,}000 \text{ lb-in}$

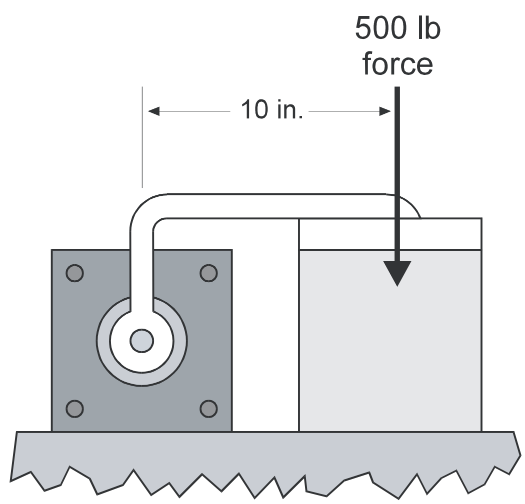


Figure 4-8c

Torque Selection

Parker rotary actuators provide output torque up to 10,000 lb-in. The chart to the right shows the nominal torque output range of various actuator models at 100 PSI.

Caution:

This chart is intended as a guide only. Refer to actual product data in this catalog before specifying an actuator. Factors such as pressure rating, rotation, and actual torque output may be affected by specific product details and options.

Nominal Torque at 100 PSI

Output Torque (lb-in)	Rotation < 95°		Rotation > 100°	
	Vane Models	Rack & Pinion Models	Vane Models	Rack & Pinion Models
10000		HP10		HP10
9000				
8000				
7000				
6000				
5000		HP4.5		HP4.5
4000				
3500				
3000	PRN800D	B6714		B6714
2500		PTR322		PTR322
2000	PV46D			
1750				
1500	PRN800S		PRN800S	
1250	PV44D	PTR321, B6713		PTR321, B6713
1000				
900	PRN300D	PTR252	PV46	PTR252
800				
700	PV36D, PV42D			
600		PTR202, B6712	PV44	PTR202, B6712
500	PV42D, PRN150D	PTR251		PTR251
400	PV33D, PRN300S		PV36, PRN300S	
300		PTR201	PV42	PTR201
250		PTR152, XR20		PTR152, XR20
200	PRN150S		PV33, PRN150S	
150	PV22D, PRN50D	PTR151, B6711		PTR151, B6711
100	PRN30D	XR15		XR15
80		PTR102	PV22	PTR102
60	PRN50S		PRN50S	
40	PRN30S	PTR101, XR10	PRN30S	PTR101, XR10
35	PV11D			
30				
25	PRNA20S		PRNA20S	
20	PV10D	XR07		XR07
15	PRNA10S		PV11, PRNA10S	
10		XR05	PV10	XR05
5	PRNA1S, PRNA3S		PRNA1S, PRNA3S	
0				

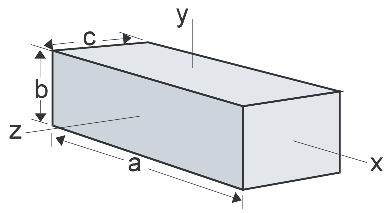
Table 4: Mass Moments of Inertia

Rectangular prism

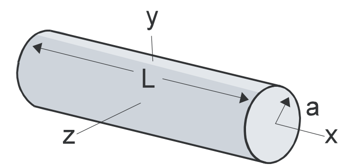
$$I_x = \frac{1}{12} m\left(b^2 + c^2\right)$$

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

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



Circular cylinder



$$I_x = \frac{1}{2} ma^2$$

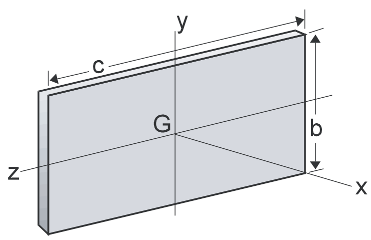
$$I_y = I_z = \frac{1}{12} m\left(3a^2 + L^2\right)$$

Thin rectangular plate

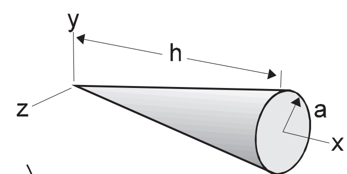
$$I_x = \frac{1}{12} m\left(b^2 + c^2\right)$$

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

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



Circular cone

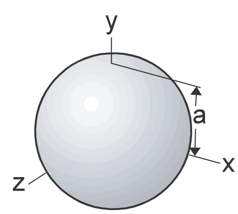


$$I_x = \frac{3}{10} ma^2$$

$$I_y = I_z = \frac{3}{5} m\left(\frac{1}{4}a^2 + h^2\right)$$

Sphere

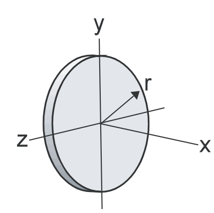
$$I_x = I_y = I_z = \frac{2}{5} ma^2$$



Thin disk

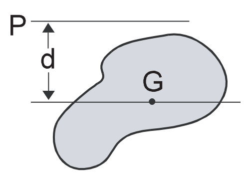
$$I_x = \frac{1}{2} mr^2$$

$$I_y = I_z = \frac{1}{4} mr^2$$



Parallel Axis Theorem:

$$I_p = \bar{I} + md^2$$



I_p = Mass moment of inertia about an axis parallel to a centroidal axis
$\bar{I}$ = Mass moment of inertia about a centroidal axis
m = Mass
d = Distance between axes

When acceleration is constant:

$$\theta = \omega_0 t + \frac{1}{2}\alpha t^2$$

$$\theta = \omega_0 t + \frac{1}{2}\omega_t t$$

$$\omega = \omega_0 + \alpha t$$

$$\omega = \left(\omega_0^2 + 2\alpha\theta\right)^{1/2}$$

$$\alpha = \frac{2\theta}{t^2}$$

$$\alpha = \frac{(\omega_t - \omega_0)^2}{2\theta}$$

$$\alpha = \frac{(\omega_t - \omega_0)}{t}$$

When velocity is constant:

$$\theta = \omega t$$

t = *time*
θ = *angular position*
ω_t = *angular velocity at time = t*
ω_0 = *angular velocity at time = 0*
α = *angular acceleration*

BASIC VELOCITY, ACCELERATION, KINETIC ENERGY AND TORQUE EQUATIONS

Equations below are based on triangular velocity profile.

$$\omega_{max} = .035 \times \frac{\Theta}{t}$$

$$\alpha = \frac{\omega_{max}^2}{\left(\frac{\Theta}{57.3}\right)}$$

$$\alpha = \frac{\omega_{max}}{(t/2)}$$

$$K.E. = 1/2\ J_m\ \omega^2$$

$$T_a = \alpha \times J_m$$

$$T_f = W \times U_s \times \left(\begin{array}{c}\text{Distance from pivot point to}\\ \text{center of external bearings}\end{array}\right)$$

$$T_L = \left(\begin{array}{c}\text{Torque arm length}\\ \text{to C.G. of load}\end{array}\right) \times WL \times \cos(\phi)$$

Where ϕ = Angle between torque arm and horizontal plane

Where:

- Θ = Angle of Rotation (Degrees)
- t = Time to rotate through Θ (sec)
- ω = Angular velocity, radians/sec
- α = Angular accelerations (radians/sec^2)
- W_L = Weight of load (lbf)
- T_a = Torque to accelerate load (lb-in)
- U_s = Coefficient of static friction
- J_m* = Rotational mass moment of inertia (lb-in-sec^2)
- T_f = Torque to overcome friction (lb-in)
- T_L = Torque to overcome effects of gravity

*Use "I" values from Table 4.

Coefficients of Friction

Material*	μ_s	μ_k
Steel on Steel	0.80	0.40
Steel on Steel (lubricated)	0.16	0.03
Aluminum on Steel	0.45	0.30
Copper on Steel	0.22	0.22
Brass on Steel	0.35	0.19
Teflon on Steel	0.04	0.04

*dry contact unless noted

Force Conversion Factors

Multiply value A by conversion factor in table to calculate value B.

B A	oz	lbf	N	kg(f)
oz	1	0.0625	0.2780	0.0284
lbf	16	1	4.4482	0.4536
N	3.5970	0.2248	1	0.1020
kg(f)	35.2740	2.2050	9.8068	1

Torque Conversion Factors

Multiply value A by conversion factor in table to calculate value B.

B A	oz-in	lb-in	lb-ft	Nm
oz-in	1	0.0625	5.2083E-03	7.0616E-03
lb-in	16	1	0.0833	0.1130
lb-ft	192	12	1	1.356
Nm	141.61	8.8507	0.7376	1

Rotational Inertia Conversion Factors

Multiply value A by conversion factor in table to calculate value B.

B A	oz-in^2	oz-in-sec^2	lb-in^2	lb-in-sec^2	lb-ft^2	lb-ft-sec^2	kg-m^2	kg-m-sec^2	kg-cm^2	kg-cm-sec^2
oz-in^2	1	2.5900E-03	6.2500E-02	1.6190E-04	4.3403E-04	1.3490E-05	1.8290E-05	1.8650E-06	1.8290E-01	1.8650E-04
oz-in-sec^2	3.8610E+02	1	2.4130E+01	6.2500E-02	1.6760E-01	5.2080E-03	7.0620E-03	7.2010E-04	7.0620E+01	7.2010E-02
lb-in^2	1.6000E+01	4.1442E-02	1	2.5900E-03	6.9444E-03	2.1583E-04	2.9260E-04	2.9840E-05	2.9260E+00	2.9840E-03
lb-in-sec^2	6.1767E+03	1.6000E+01	3.8610E+02	1	2.6810E+00	8.3333E-02	1.1300E-01	1.1520E-02	1.1300E+03	1.1520E+00
lb-ft^2	2.3040E+03	5.9666E+00	1.4400E+02	3.7300E-01	1	3.1080E-02	4.2140E-02	4.2970E-03	4.2140E+02	4.2970E-01
lb-ft-sec^2	7.4129E+04	1.9201E+02	4.6333E+03	1.2000E+01	3.2175E+01	1	1.3560E+00	1.3824E-01	1.3560E+04	1.3824E+01
kg-m^2	5.4675E+04	1.4160E+02	3.4176E+03	8.8496E+00	2.3730E+01	7.3746E-01	1	1.0190E-01	1.0000E+04	1.0190E+01
kg-m-sec^2	5.3619E+05	1.3887E+03	3.3512E+04	8.6806E+01	2.3272E+02	7.2338E+00	9.8135E+00	1	9.8130E+04	1.0000E+02
kg-cm^2	5.4675E+00	1.4160E-02	3.4176E-01	8.8496E-04	2.3730E-03	7.3746E-05	1.0000E-04	1.0191E-05	1	1.0190E-03
kg-cm-sec^2	5.3619E+03	1.3887E+01	3.3512E+02	8.6806E-01	2.3272E+00	7.2338E-02	9.8135E-02	1.0000E-02	9.8135E+02	1

Length/Distance Conversion Factors

Multiply value A by conversion factor in table to calculate value B.

B A	in	ft	mm	cm	m
in	1	0.0833	25.4	2.54	0.0254
ft	12	1	304.8	30.48	0.3048
mm	0.03937	0.00328	1	0.1	0.001
cm	0.3937	0.03281	10	1	0.01
m	39.37	3.281	1000	100	1