Data Sheet

Frelon lined linear bushings

Frelon is a compound of Teflon and fillers developed for improved performance over traditional bearings. It provides low wear, low friction self-lubrication and high strength.

It is chemically inert and self-lubricating, although additional lubrication may also be used.

Materials

Bearing linear ______________________________Frelon

Bearing shell__________6061-T6 Aluminium anodised with nickel acetate seal

Pillow blocks______________6063 - T6 aluminium with clear anodised seal

Transfer process of line to shaft

The interaction of the Frelon material and the shafting creates a natural microscopic transfer process of the Frelon to the running surface. The valleys in the surface finish are filled in with Frelon material during the initial break-in period. This creates the self-lubricating condition of Frelon riding on Frelon. The break-in period will depend on several criteria.

1. Preparation of the shafting prior to installation. It is best to clean the shafting with a 3-in-1 type oil before installing the bearings. This ensures that the surface is ready to receive a full transfer of material.
2. Speed, load, and length of stroke specific to the application. Typically the initial transfer process will take approximately 50-100 strokes of continuous operation. The running clearance on the bearing will increase and average of .0002 to .0005in. depending on the length of the stroke and surface requiring the transfer.
3. How often the shafting is cleaned. If the shafting is cleaned regularly, increased wear will be seen in the bearings. This due to the transfer process being performed over and over again.

CAUTION: Do not repeatedly clean the shafting with alcohol! This will remove the previously transferred material entirely and increase the wear to the bearing liner.

Chemical Resistance

The Frelon liner has almost universal chemical inertness. Only molten sodium and flourine at elevated temperatures and pressures show any signs of attack. It is approved for use with liquid oxygen, N_2O_2, hydrazine, UDMH, hydrocarbon fuels, high strength hydrogen peroxide, etc.

The table given below applies only to the bearing shell and pillow block materials. The table is provided as a reference only. The data given will be affected by factors such as temperature, PV, degree of contact, strength of solution, etc. In each specific application, it is always advisable to conduct specific testing to determine suitability of use. This table only addresses general corrosion, NOT galvanic, SCC, or other types of corrosion. Corrosion rates are at room temperature unless noted.

Standard and hardcoat data only applies when the coating is intact. If the coating is worn through or damaged, an area of galvanic and pitting corrosion will be created. Then use the bare aluminum data.

E = <.002in per year S = <.050in per year

G = <.020in per year U = >.050in per year

Chemical	Bare Aluminum	Standard & Hardcoat Anodised Aluminum	316 Stainless Steel
Acetic Acid, 20%	G	G	E
Acetone	E	E	E
Ammonia, anhydrous	E	E	E
Ammonium hydroxide, 10%	U	U	E
Ammonium chloride, 10%	U	U	G
Ammyl acetate (122°F/50°C)	E	E	E
Barium hydroxide	U	U	G
Beer	E	E	E
Boric acid solutions	E	E	G
Butane	G	G	G
Calcium chloride, 20%	G	G	G
Calcium hydroxide, 10%	G	G	G
Carbon dioxide	E	E	G
Carbon monoxide	E	E	E
Chlorine gas, dry	G	G	G
Chlorine gas, wet	U	U	U
Chronic acid, 10%	G	E	E
Citric acid, 5%	E	E	E
Ethyl acetate	E	E	G
Ethyl alcohol	E	E	G
Ethylene glycol	E	E	G
Ferric chloride, 50%	U	U	U
Formic acid - Anhydrous	E	E	E
Gasoline	G	G	G
Hydrochloric acid, 20%	U	U	U
Hydrochloric acid, 35%	U	U	U
Hydrocyanic acid, 10%	G	G	G
Hydrofluoric acid - dilute	U	U	U
Hydrofluoric acid, 48%	U	U	U
Hydrogen	E	E	E
Hydrogen peroxide - dilute	E	E	G

Chemical	Bare Aluminum	Standard & Hardcoat Anodised Aluminum	316 Stainless Steel
Hydrogen sulfide, dry	G	E	E
JP-4	G	G	G
Kerosene	G	G	G
Lactic acid, 10%	G	G	E
Magnesium chloride, 50%	U	U	G
Mercury	U	U	E
Methyl alcohol	G	G	G
Methyl ethyl ketone	G	G	G
Methylene chloride	E	E	G
Mineral oil	G	G	G
Naphtha	G	G	G
Nitric acid, 70%	U	U	E
Phosphoric acid, 10%	U	U	E
Sodium chloride	U	U	G
Sodium hydroxide, 20%	U	U	E
Sodium hypochlorite, 20%	G	G	U
Sodium peroxide, 10%	G	G	G
Steam (see water)	-	-	-
Sulfur dioxide, wet	U	U	G
Sulfur dioxide, dry	G	G	G
Sulfur trioxide	G	G	G
Sulfuric acid, 50%	U	U	U
Sulfurous acid	G	G	E
Toluene (122°F/50°C)	E	E	E
Turpentine	G	E	E
Water, demineralized	G	E	E
Water, distilled	U	S	G
Sea Water	G	E	G
Water, sewage	U	S	G
Xylene	G	G	G
Zinc chloride solutions	U	U	G

Bearing shells use 6061-T6 aluminum alloy which is known to have the best corrosion resistance of the high strength aluminum alloys. The sulfuric bath anodising and nickel acetate sealing provide the best corrosion resistance available in anodised coatings. They can withstand a rigorous 14 day exposure in a 5% salt spray solution at 96°F per military specifications without significant damage. With the coating intact, it is considered to be inert in most fluids with pH values of 5 to 8.

Hardcoat anodising provides the same chemical resistance, but is applied to a .002 thickness providing a more durable surface that will stand up to greater abuse. However, if the coating is penetrated, it also will have the resistance reduced.

This resistance allows the bearings to be used in many specialised applications e.g.

Submerged applications -

The bearings will employ the fluid as a lubricant showing increased velocities and wear life - especially effective in oils and non-salt water.

Vacuums/outgassing/clean rooms -

Due to self-lubrication, low outgassing and a minimum of particulate, Frelon bearings are ideal in clean rooms and vacuums.

Lubrication

Although Frelon bearings are self-lubricating, in certain applications additional lubrication may be required. It is difficult to predict these applications which should be selected on an application by application basis. Typically, hand-driven, low horsepower applications or lightly contaminated environment may require oil lubrication.

Do not use motor oil or any oil with additives.

These quickly cause stick-slip reactions in the bearing. 3 in 1 oils are best in preparing for a proper transfer of Teflon to the shafting and are an excellent cleaning oil.

For long-term lubrication, high-speed applications or for lowest friction possible running, greasing is recommended.

Do not use a moly filled or other type filled greases.

These act as a lapping compound on the ID of the bearing and increase wear dramatically.

Do not fill all of the running clearance with grease - apply only a thin layer.

If grease is used and does not work, remove the bearing from the housing, wipe as much grease as possible away from the ends of the bearing and start to fill with 3 in 1 oil for cleaning the liner. To speed the cleaning process. apply forced air to the bearing through the hole and continue using oil lubrication.

Classes of plane bearings

Frelon bearings are in a class of bearings known as plane bearings, meaning that they have no rolling elements. There are three classes of plane bearings:

Class I - Require an outside source of lubrication. (Oil, grease, etc.)

Class II - Lubrication is impregnated within the walls of the bearing. (Bronze, powder metal, etc.) Typically these bearings require an added lubricant also.

Class III - Self-lubricating bearings. Do not require added lubricants.

Frelon bearings are a Class III plane bearing and are self-lubricating.

Rating a plane bearing

Plane bearing performance capacity is rated by 'PV'.

'P' - pressure or load in pounds per square inch (psi) or kilograms per square centimetre (kg/cm^2)

'V' - velocity or surface speed in meters per minute (m/min)

'PV' - pressure velocity value

Frelon bearings maximum parameters

'P' = 1500 psi or $105.45 kgf/cm^2$

'V' = 42.672m/min (dry)

Significantly more with proper lubrication

'PV' = $215 kgf/cm^2$ x m/min

All three parameters must be met in order for the bearing to operate properly.

Formulas for ratings

Pressure is over the projected area of load:

$$A = L \times d$$

$$P = \frac{w}{A} \text{ psi (or } kg/cm^2\text{)}$$

Velocity:

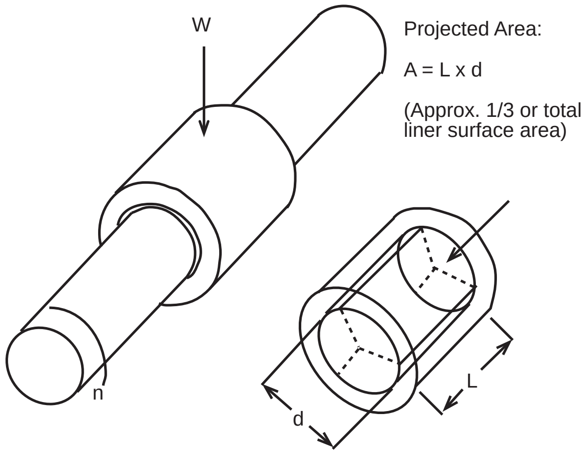
Linear = total distance travelled in one minute

Rotational velocity:

$$V = \frac{x\ d\ x\ n}{12} \quad m/min$$

Pressure velocity value (PV):

$PV = P \times V\ kg/cm^2 \times m/min$



W = max. load
L = bearing length
d = shaft diameter
A = projected contact area
n = rotational speed (rpm)

PV equivalents

	Inch	Technical Metric	Intl' Metric (SI)
Load	1psi	$.0703kgf/cm^2$	$.0069N/mm^2$
Velocity	1ft/min	.3048m/min	.0058m/sec
PV	1PV	.0214PV	.000036PV
Simplicity MAX. PV	10,000	215	0.36

Load capacity (pressure)

Depending upon the material used, a plane bearing's load capacity can greatly exceed a rolling element bearing. There are three basic reasons for this:

1. The area of surface contact with the shaft is far greater than rolling element bearings which have point-to-point contact with a given number of balls.
2. A rolling element bearing must be oriented properly for the ball tracks to carry the load adequately while a plane bearing can be mounted in any orientation.
3. Only 1 or 2 of the tracks in a rolling element bearing will actually carry any of the load applied.

Frelon bearings have a thin liner that is bonded to a metal shell at the molecular level allowing the load to be transferred throughout the bearing. This gives it an advantage over other plane bearings of solid plastic or polymer materials. These other materials will tend to 'cold flow' under pressure. 'Cold flow' is a term that means to deform or lose shape. The idea is similar to pressing your finger into a bar of soap - material will move or deform as pressure is applied.

Linear surface speeds (velocity)

In typical applications, speed is a known quantity and easily converted. Typically meters per minute are used. The most important factor that speed (along with friction) produces is heat buildup. This is not a critical factor in most linear applications because the heat is dissipated over the length of travel and it does not affect the bearing. Short stroke or extremely high speed applications may see the affects of heat buildup in thermal expansion and the bearing ID locking on the shaft.

Rotational Surface Speeds (velocity)

A new set of problems is presented in rotary applications. The heat buildup is retained in one location. Friction, thermal expansion, and running clearances become critical issues. Actual running temperature becomes a moving target as the bearing ID closes with thermal expansion due to operation. The friction increases due to more surface contact, and more heat is created causing more thermal expansion. Because of these types of conditions and reactions, the heat being removed cannot balance the heat being produced. If the balance is not found the bearing can fail. It is always best to first conduct application testing.

Wear rate vs. life expectancy

Rolling element linear bearings' life expectancy is usually expressed in total metres. Rolling element rotary bearings' life expectancy is expressed in hours of operation. They are also rated for average (L-50) and minimum (L-10) life. L-50 life is the average life which can be expected from 50% of rolling element bearings. In other words, 50% will not reach the average life expectancy. L-10 life is the minimum life (1/5 the average life) expected from 90% of rolling element bearings. In other words, 10% will not reach the minimum life expectancy. Theoretically they could fail upon installation.

Plane bearings are not rated by a life expectancy, but by wear rate of the I.D. Wear is dependent greatly upon the proper application of the bearing and material used. If it is not properly applied, it will fail.

Failure, however, is subjective depending upon specific application requirements. .002 inches running clearance may not be acceptable in one application where another may be able to run a bearing until the liner is worn through, rotate it 30 degrees, and continue to run This broad range makes it difficult to determine life expectancy.

Wear testing has been conducted by Pacific Bearing Co. in conjunction with the University of Illinois School of Engineering. These tests have resulted in data for an application running at 100ft/min. with 10psi loading. The average radial wear under these conditions after 1,000,000 inches of travel without lubrication was .001 inches. This is typical of most applications in which Frelon bearings are used properly. If the bearings are not applied properly, excessive, unnecessary wear may result.

Factors that contribute to wear life

1. Proper mating of shaft and linear materials.
2. Surface finish 8-16RMS (.20- .40mm) is required. Peaks in the surface that are polished to a radius provide the best running surface. Sharp peaks in the finish will be like fine lapping compound wearing the I.D. of the bearing.

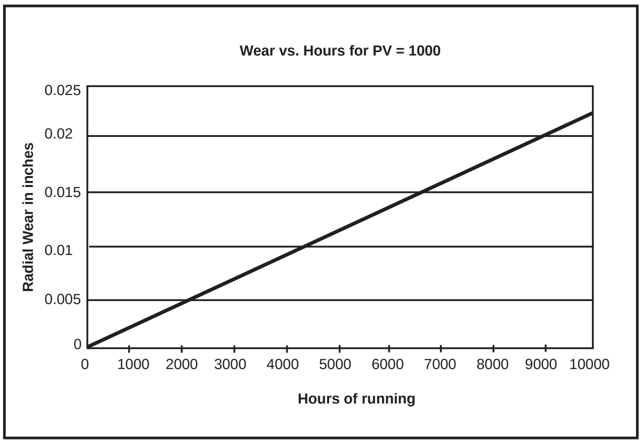
Note: Shafting damaged by use with ball bearings can be salvaged and used with Simplicity bearings. Spin in a lathe and polish with sand papers in this order: 120 grit, 180 grit, and 300 grit. This will also remove sharp peaks in the surface finish.

3. Surface speed - at high speeds, heat buildup will affect linear wear.
4. Proper transfer process of the liner to the shaft.
5. Lubrication - proper lubrication can greatly improve the wear rate of a bearing. At the same time improper lubrication can increase wear and failure.
6. Load - increased load will increase wear, but it is NOT a linear relationship. As a rule of thumb, if everything in a application remains the same and the load is decreased by 1/2, the relationship would be a cubic one or the depth of wear times 1/3.
7. Contamination - while migrating into the bearing and embedding into the linear, certain types of contamination may over time cause increased wear to the liner.

Note: This is not an all inclusive list. There are many, many more factors within an application that can affect wear to different degrees. These are the major issues and the first things to address in a design.

Wear rate (cont.)

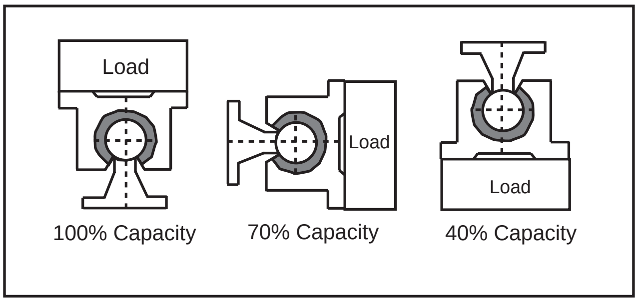
The chart below is a good guideline for radial wear vs. hours of running. It is applicable to dry running application (no lubrication) without binding, cantilever, or moment loads. It is also assuming that the above factors are all optimized for the Simplicity bearings. Wear vs. time is used as opposed to wear vs. distance because two different applications could have the same PV value with very different velocities. In a given amount of time, one would travel farther than the other, but they would both wear approximately the same.



Open bearings orientation

Simplicity bearings can operate in any orientation

- Load capacities will vary on open bearings depending on the orientation in which they are being used.



Application information

Cantilevered load and counterbalances (Figures 1 and 2)

- Maximum 2:1 ratio
- 1 × = bearing separation on same shaft
- 2 × = distance from shaft to load or force
- Example : If 2 × equals 10cm then 1 × must be at least 5cm.

CAUTION: BINDING will occur if the 2:1 ratio is exceeded!!

- This principle is NOT load dependent! It is NOT due to edge loading. It is also NOT dependent on the driving force used! The bearings will bind whether hand or mechanically driven. This principle is a product of friction.

Working through the following equation will explain why this is a product of friction:

P = Force being applied

L = distance out from shaft that P is being applied

s = centre to centre spacing of bearings

f = resultant force on bearings by shaft

F = friction force on each bearing

μ = coefficient of friction (about 0.25 when not moving)

Balance the moments:

$f*s = L*P$

$L/s = f/P$

Calculate friction force:

$F = f*\mu$

Note: Total friction force pushing up is 2*F.

To lock up the slide, the total friction force must be equal to (or greater than) P.

$P = 2*F = 2*f*\mu$

Substitute for P:

Note: The forces drop out of the equation

$L/s = f/(2*f*\mu) = 1/(2*\mu) => L/s = 1/(2*\mu)$

Assume static coefficient of friction is 0.25 ($\mu = 0.25$) then L/s = 2. That is the ratio 2:1 ratio.

There may be other factors that add to the braking effect, but the coefficient of friction is the main cause.

Note: Proper lubrication can help to drop friction and extend the 2:1 ratio.

Figure 1

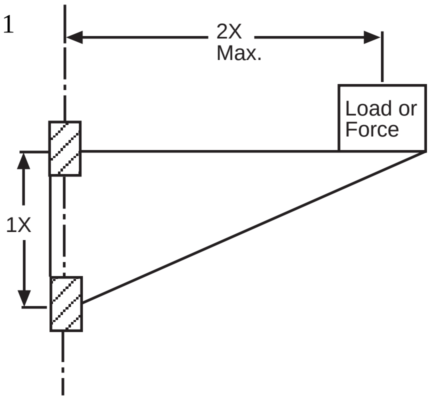
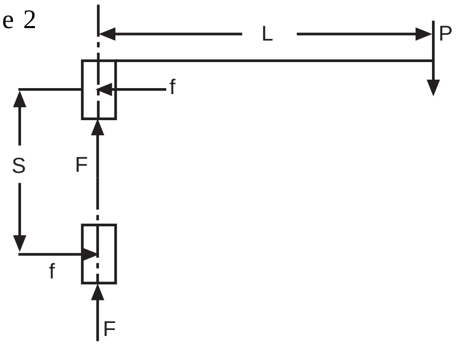


Figure 2



Application information (cont.)

Counterbalances (Figure 3)

If holding the 2:1 ratio is not possible, one method of preventing binding problems is using a counter balance.

Foe efficient counter balances in horizontal applications, use this formula:

$$M*Y = W*Z$$

Notes: To avoid problems when running without mass

(M) Z = 1-1/2s

- W can be calculated - load on bearing will be:

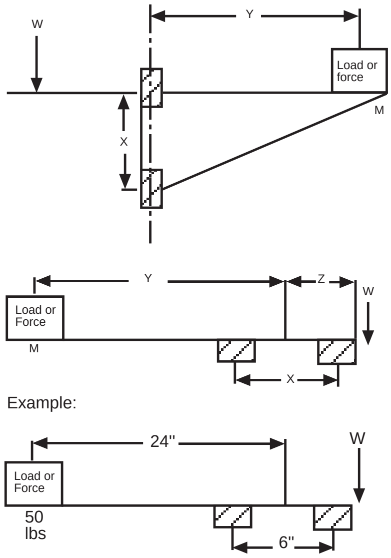
$$\frac{M + W}{\text{\# of bearings}}$$

Example: 50 * 24 = W * Z (Z = 1-1/2*6 = 9)

$$W = \frac{50 * 24}{9} = 133\text{lbs}$$

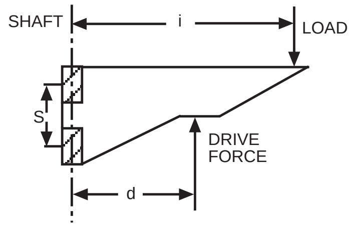
$$\text{Load per bearing} = \frac{50 + 133}{4} = 45.75\text{lbs./bearing}$$

Figure 3



Cantilever loads and drive force location without counterbalance (Figure 4)

Figure 4



d = distance from shaft to Drive Force

l = distance from shaft to the load centre of gravity

s = centre to centre spacing of the bearings on the shaft

(If non-self-aligning, then outside to outside distance should be used).

L = l/s = Load Force Ratio

D = d/s = Drive Force Ratio

- General rule:

1. Drive Force Ratio (D) should never be larger than 2. A drive force ratio (D) larger than 2 can cause the slide to lock up.
2. Load Force Ratio (L) can be larger than 2, but as this ratio increases, the drive force required to move the slide increases dramatically. Load force ratio (L) larger than 4 are not recommended.
3. If the slide is occasionally operated unloaded, use the distance to the slide's cebter of gravity as the distance to the load (l).

- Vertical application:

1. If L is between 0 and 2, the lowest drive forces occur when the value of D is about 90% of L

   D = .9 × L). However, D values between 0 and L will work fine.
2. If L is between 2 and 4, use this equation : D = 4 - L

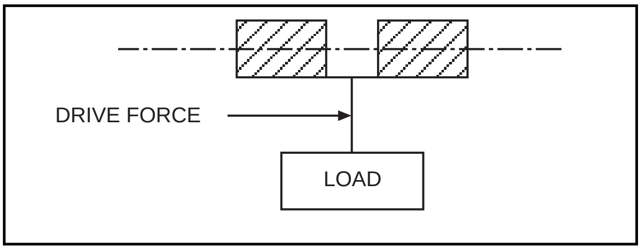
- Horizontal applications:

1. For best results, the drive force should be applied as close to the shaft as possible no matter what the value of the Load Force Ratio (L) is.

- Hanging or 'Top Heavy' horizontal applications with high acceleration rates:

1. If your application will have high acceleration forces, use this formula for the value of the Drive Force Ratio:

   $D = 0.8 \times L \times \sqrt{a}$ where a is acceleration in g's.

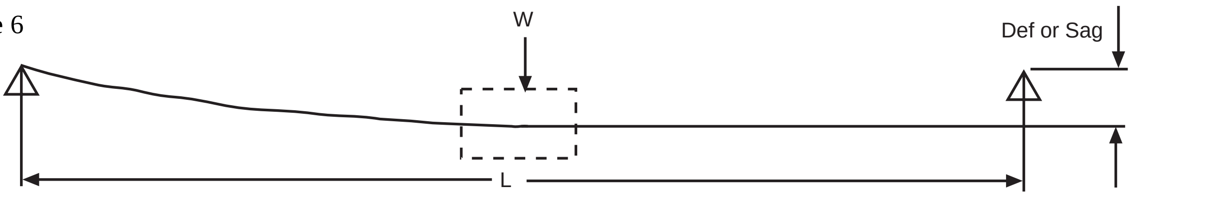


Application information (cont.)

Shaft deflection

In applications where a support rail is not used, shaft deflection can become critical in the function of the bearing. If deflection is greater than the misalignment capabilities of a standard pillow block, binding can occur. Solutions would be to increase shaft & bearing size (to lessen the amount of deflection) or use an open bearing configuration with a support rail. Follow the formulas below to check shaft deflection and sag.

Figure 6



Formula for inch and metric shafting deflection
Total shaft deflection in horizontal applications:

Tot. Def = Def + Sag

$Def = w \times L^3/D$

$Sag = L^4/S$

Def = Pure deflection due to load at centre of shaft (mm)

Sag = Deflection of shaft due to it's own weight (mm)

L = Shaft unsupported length (mm)

w = load being applied at centre of shaft (lbs. or N)

D = Deflection coefficient ($D = 48 \times E \times l$)

S = Sag coefficient ($S = E \times 1 \times 384/(5 \times sw)$)

Notes: $l = p \times diam^4/64$

$sw = L \times p \times diam^2/4 \times density$

E = Modulus of Elasticity (Young's modulus)

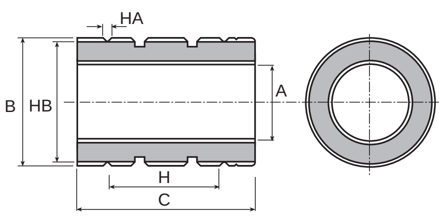
Shaft Dia.	Hardened steel D	Hardened steel S	Stainless steel D	Stainless steel S
3/16in	8.4×10^{4}	1.7×10^{7}	8.0×10^{4}	1.6×10^{7}
1/4in	2.67×10^{5}	3.1×10^{7}	2.54×10^{5}	2.9×10^{7}
3/8in	1.35×10^{6}	6.9×10^{7}	1.29×10^{6}	6.5×10^{7}
1/2in	4.27×10^{6}	1.23×10^{8}	4.06×10^{6}	1.16×10^{8}
5/8in	1.04×10^{7}	1.92×10^{8}	9.92×10^{6}	1.81×10^{8}
3/4in	2.16×10^{7}	2.77×10^{8}	2.06×10^{7}	2.61×10^{8}
1in	6.83×10^{7}	4.92×10^{8}	6.5×10^{7}	4.63×10^{8}
1 - 1/4in	1.67×10^{8}	7.69×10^{8}	1.59×10^{8}	7.24×10^{8}
1 - 1/2in	3.46×10^{8}	1.11×10^{9}	3.29×10^{8}	1.04×10^{9}
2in	1.09×10^{9}	1.97×10^{9}	1.04×10^{9}	1.85×10^{9}
2 - 1/2in	2.67×10^{9}	3.07×10^{9}	2.54×10^{9}	2.9×10^{9}
3in	5.53×10^{9}	4.43×10^{9}	5.27×10^{9}	4.17×10^{9}
4in	1.75×10^{10}	7.87×10^{9}	1.66×10^{10}	7.41×10^{9}

Shaft Dia.	Hardened steel D	Hardened steel S	Stainless steel D	Stainless steel S
5mm	2.94×10^{8}	3.12×10^{11}	2.8×10^{8}	2.94×10^{11}
6mm	6.11×10^{8}	4.5×10^{11}	5.81×10^{8}	4.24×10^{11}
8mm	1.93×10^{9}	8.0×10^{11}	1.84×10^{9}	7.53×10^{11}
10mm	4.71×10^{9}	1.25×10^{12}	4.48×10^{9}	1.18×10^{12}
12mm	9.77×10^{9}	1.8×10^{12}	9.3×10^{9}	1.69×10^{12}
13mm	1.35×10^{10}	2.11×10^{12}	1.28×10^{10}	1.99×10^{12}
14mm	1.81×10^{10}	2.45×10^{12}	1.72×10^{10}	2.31×10^{12}
16mm	3.09×10^{10}	3.2×10^{12}	2.94×10^{10}	3.01×10^{12}
1 - 1/2in	3.46×10^{8}	1.11×10^{9}	3.29×10^{8}	1.04×10^{9}
2in	1.09×10^{9}	1.97×10^{9}	1.04×10^{9}	1.85×10^{9}
2 - 1/2in	2.67×10^{9}	3.07×10^{9}	2.54×10^{9}	2.9×10^{9}
3in	5.53×10^{9}	4.43×10^{9}	5.27×10^{9}	4.17×10^{9}
4in	1.75×10^{10}	7.87×10^{9}	1.66×10^{10}	7.41×10^{9}

Note: The information given in this data sheet is for reference only. The user must test specific applications.

Dimensions

Closed metric bushings



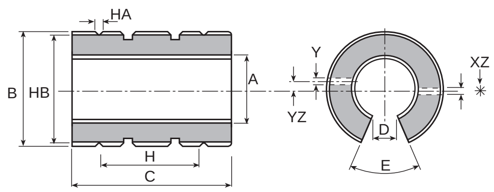
Technical specification

RS stock no.	Nominal Size mm	A-Bearing I.D. Min.	A-Bearing I.D. Max.	B - O.D. Min.	B - O.D. Max.
217-9697	12	12.066	12.093	21.979	22
217-9704	16	16.066	16.093	25.979	26
217-9710	20	20.096	20.129	31.975	32
217-9726	25	25.096	25.129	39.975	40

C-length Min.	C-length Max.	Concentric Max. mm	Weight Kg
31.746	32	0.0254	0.017
35.746	36	0.0254	0.028
44.746	45	0.0254	0.054
57.746	58	0.0254	0.109

Effective Surface Area (sq.cm)	Max. Static Load (kg)	Max. PV (m/min, kg/sq.cm)	Max. Speed (m/min) Dry	Max. Speed (m/min) Lubricated
3.8	403	215	42.6	122
5.8	605	215	42.6	122
9	945	215	42.6	122
14.5	1523	215	42.6	122

Open metric bushings



Technical specification

RS stock no.	Nominal Size mm	A-Bearing I.D. Min.	A-Bearing I.D. Max.	B - O.D. Min.	B - O.D. Max.
217-9732	12	12.066	12.093	21.979	22
217-9754	20	20.096	20.129	31.975	32
217-9760	25	25.096	25.129	39.975	40

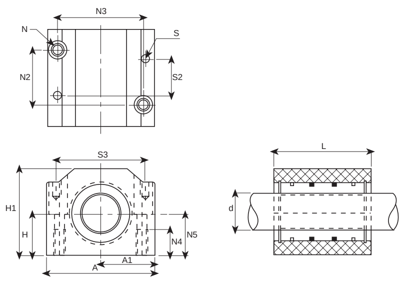
C-length Min.	C-length Max.
31.746	32
35.746	36
44.746	45
57.746	58

D-Slot width Min.	E-Slot Angle	X-Net Hole Dia.	XZ-Net Hole locate
7.6	78°	3	1.35
10.4	78°	2.2	0
10.8	60°	2.2	0
13.2	60°	3	0

Y-Ret. Hole Dia.	YZ-Ret. Hole Locate	Weight Kg
3	7	0.0156
3	0	0.0213
3	0	0.0439
3	-1.51	0.0893

RS stock no.	Max. Static Load (kg)	Max. PV (m/min, kg/sq.cm)	Max. Speed (m/min)	
			Dry	Lubricated
217-9732	403	215	42.6	122
217-9754	945	215	42.6	122
217-9760	1523	215	42.6	122

Metric pillow blocks - closed including bushing



Technical specification

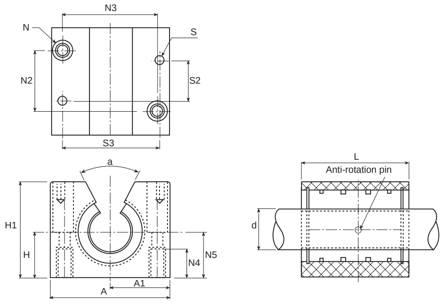
RS stock no.	Nom I.D.	H Centreline +/- .015	H1 Height	A Width	A1 Centreline +/- .013
217-9782	12	18	35	43	21.5
217-9798	16	22	42	53	26.5
217-9805	20	25	50	60	30
217-9811	25	30	60	78	39

L Length	N Bolt	N2	N3	N4	N5	S	S2
39	M5	23±0.15	32±0.15	11	16.5	4	32
43	M6	26±0.15	40±0.15	13	21	4	35
54	M8	32±0.15	45±0.15	18	24	5	45
67	M10	40±0.15	60±0.15	22	29	6	20

S3	Max. Load Rating (Kg)	Assy. Weight (Kg)
34	403	0.118
42	605	0.200
50	945	0.329
64	1523	0.655

For further loading information, see data on Metric Bushing - closed.

Metric pillow blocks - Open including bushing



Technical specification

RS stock no.	Nom I.D.	H Centreline +/- .015	H1 Height	A Width	A1 Centreline +/- .013
217-9827	12	18	28	43	21.5
217-9849	16	22	35	53	26.5
217-9855	20	25	42	60	30
217-9861	25	30	51	78	39

L Length	N Bolt	N2	N3	N4	N5	S	S2
39	M5	23±0.15	32±0.15	11	16.5	4	32
43	M6	26±0.15	40±0.15	13	21	4	35
54	M8	32±0.15	45±0.15	18	24	5	45
67	M10	40±0.15	60±0.15	22	29	6	20

a	Max. Load Rating (kg)	Assy Weight (kg)
66°	403	0.096
68°	605	0.162
60°	945	0.267
60°	1523	0.536

Note: For further loading information see data on Metric Bushing - Open