Engineering Guide Introduction

Avon Bearings Corporation produces large diameter bearings for a broad variety of applications. Founded in 1969 as Formmet Corporation we changed our name to SIFCO Bearings in 1981, and finally changed our name to Avon Bearings Corporation in 1988.

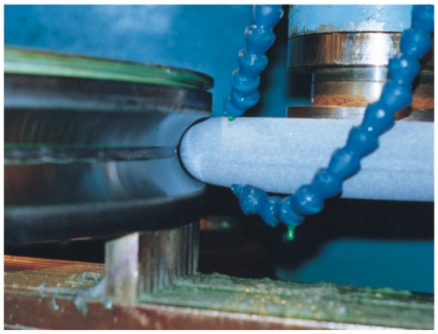
The bearings we manufacture range in size from 10 inches to 240 inches outside diameter. They are used in construction machinery, logging machines, aerial baskets, medical equipment, military vehicles, radar antennas, and a variety of industrial equipment. In terms of both quality and delivery, we believe we've achieved a record as the most dependable supplier in the big bearing industry.

This is particularly remarkable since we've also traditionally offered the shortest lead times in our industry. We're pleased to be regular suppliers to a sizable number of Fortune 500 firms and a great number of smaller companies. We invite you to consult with our customers to verify these claims.

This design guide and catalog describes our most popular types of bearings. Other styles of Avon bearings are mentioned in the closing pages of this catalog and additional information on them is available upon request.

Although we've attempted to make this catalog self-explanatory, we hope you will call upon our sales and engineering people to assist in choosing and designing bearings to suit your needs. We are as near as your telephone or computer. Our application engineering service is normally free and without obligation.

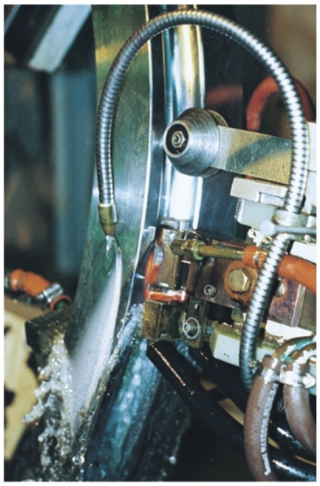
Avon bearings are produced at a twelve acre manufacturing complex in Avon, Ohio, near Cleveland. All machining and assembly operations are performed in a temperature controlled facility.



Raceways precision ground to proper geometry, optimizing load-carrying capability.

Shaping an external gear

Special CNC machining centers drill bolt holes for mounting.



Induction Hardening Raceway

Bearings are manufactured in accordance with Avon's stringent manufacturing and quality systems to assure adherence to O.E.M. specifications.

Avon Bearings Application Data Sheet

Fax: 440.871.2503

Company: ______________________ Phone: ______________ Date: __________

Name: ______________________ Fax: ______________ Desired Response

Address: ______________________ E-Mail: ______________ Date: __________

Application: ______________________

Axis of Rotation: ☐ Vertical ☐ Horizontal

Bearing in Compression or Tension under load? ☐ Compression ☐ Tension

Gear: ☐ External ☐ Internal ☐ No Gear

Rotational Data: ☐ Positioning Only ☐ Slewing With ______ Degrees of Angle ☐ Continuous at ______ R.P.M.

Drive Torque at bearing ______ ft.-lbs.

Number of Drives: ______

BEARING LOADS

LOAD NUMBER	TYPE AND DIRECTION OF LOAD	UNIT	MAXIMUM OPERATING LOADS	MAXIMUM TEST LOAD	EXTREME SHOCK LOAD (NON-OPERATING)
1	Axial or Thrust load (Parallel to axis or rotation)	LBS.			
2	Radial load (Perpendicular to axis of rotation and without gear loads)	LBS.			
3	Moment load resulting from axial or thrust loads	FT.-LBS.			
4	Moment load resulting from radial loads	FT.-LBS.			
5	Moment resulting from axial and radial loads (3 + 4)	FT.-LBS.			
Identify loads which occur simultaneously for each condition. Should simultaneous loads vary from the maximum conditions specified above, please specify the new value.			☐ 1 ____ ☐ 2 ____ ☐ 3 ____ ☐ 4 ____	☐ 1 ____ ☐ 2 ____ ☐ 3 ____ ☐ 4 ____	☐ 1 ____ ☐ 2 ____ ☐ 3 ____ ☐ 4 ____

Note:* For continuous rotation applications, please attach details of dynamic load conditions, percentage of operating time and R.P.M. for each condition as well as required L_{10} life. *See page 2-8 for reference.

BEARING DIMENSIONS:

	Preferred	Max./Min.
Outside Diameter:	______	______
Inside Diameter:	______	______
Overall Height:	______	______

TEMPERATURE RANGE:

Normal Operating: ______

Minimum Temperature: ______

Maximum Temperature: ______

OPERATING ENVIRONMENT:

Describe environment and characteristics

______

Application Data: ***(Special environmental conditions, required accuracies, certification or inspection requirements, material tests, etc.)***

______

Design Data: ***(Preferred gear pitch, bolt size and configuration, type and method of lubrication; reference drawings, etc.)***

______

Bearing Design Support

Free Design Support

Avon Bearings offers design engineering support which is normally free and without further obligation. In order to respond with the most technically and economically effective design, comprehensive application data is needed.

Your Input:

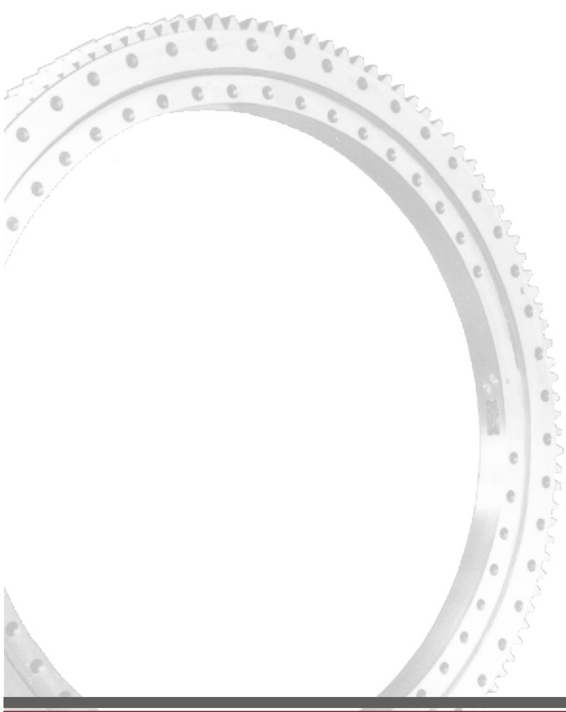
Here are suggestions regarding your input and methods to forward your data to Avon Bearings.

- Please see the Avon Bearings application data sheet on page 2-19. Photocopy, complete and mail or fax to Avon Bearings.
- Visit our website at www.avonbearings.com and select “Customize my bearing” and the application data sheet will appear. Simply complete and your data will be e-mailed directly to us.
- Should your design data already exist, please e-mail to design@avonbearings.com or fax.

Our response:

Upon receipt of your data, we will respond with engineering information generally including the following:

- Drawing of the suitable bearing
- Static capacity chart including bolt and gear capacities
- Dynamic analysis along with stiffness values for specific load conditions



Loading Plug Location

Virtually all Avon Series M, H, T and R turntable bearings are loaded by means of a radial hole in the non-geared race. After loading, this hole is closed with a closely fitting plug and locked with a taper pin to prevent movement. It is preferable that this plug be located in a relatively lightly loaded zone as depicted in Figure 1 on page 2-13.

Seals

Most Avon bearings are provided with neoprene or polyurethane seals. The external type is preferred due to its superior protection, but if there is insufficient ring offset, internal seals can be used. When bearings are relubricated, the excess lubricant extrudes past the seals. The resultant collar of grease along with the seals provides a high degree of protection against dry contaminants. For applications requiring continuous rotation or

Environmental Limitations

Standard Avon bearings may be routinely applied in temperatures ranging from – 10°F to +160°F. (– 23°C to + 71 °C). Operation beyond these extremes may require changes of lubricant and non-metallic material used in seals and some separators. Sub zero operations also require consideration of impact resistance of ring material. Elevated temperatures require consideration of possible material hardness reduction. Please refer to Avon Engineering for all applications outside the – 10°F to +160°F range. Most bearings are equipped with contact seals which are aided by a collar of grease extruding at the seals. In extremely wet or abrasive environments, auxiliary sealing and/or special lubrication procedures may be required. Please refer to Avon Engineering for assistance in this area.

Frictional Torque

Frictional torque of large diameter bearings is difficult to predict precisely. Bearing torque is very much affected by the flatness of the mounting structure, by the allowable deformation of the mounting structure, and of course by the load. In most applications, the major determinant of necessary rotational torque is the torque necessary to accelerate the rotating mass. However, in applications where frictional torque is critical, please contact Avon Engineering for design assistance.

An estimate of frictional torque can be calculated using the formula below:

$$T = \frac{\mu}{2}(4.4M + AD + 2.2RD)$$

when:

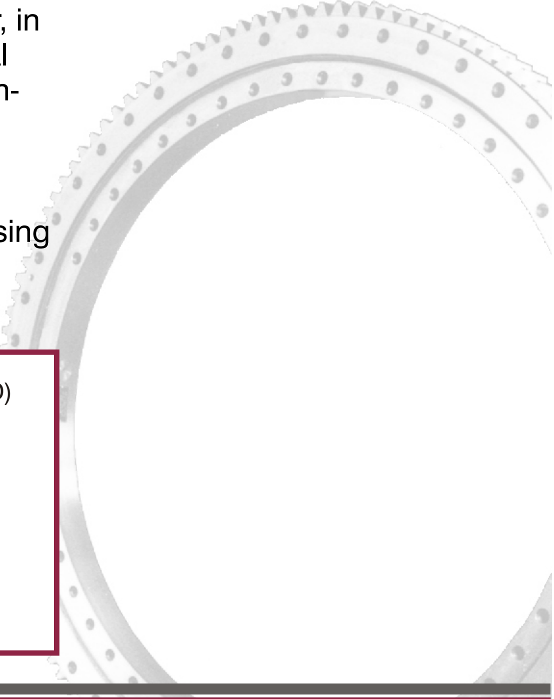
- **T** = torque (ft-lb)
- **M** = moment load (ft-lb)
- **A** = thrust load (lb)
- **R** = radial load (lb)
- **D** = raceway diameter (ft)

Frictional Torque Coefficients

Description	Bearings Series	Turning Coefficient
Single-row ball Four-point contact	M, H, T	μ = .006
Two-row ball	D	μ = .004
Single-row Cross roller	R	μ = .004
Three-row roller	TR	μ = .003

Please note the torque values derived from this equation are estimates. It is not uncommon for actual torque values of the same bearing design to vary significantly. Avon Engineering recommends a safety factor up to 5 be applied to any estimate to insure the proper drive torque is available.

For more accurate estimates, contact Avon Engineering.



Lubrication, Maintenance and Storage

Lubrication

The bearing raceway should be lubricated with a heavy-duty extreme pressure grease. A list of suggested manufacturers and trade names is shown below for your convenience. Please note that these are not the only manufacturers of this type of grease.

SUPPLIER	TRADE NAME
Mobil	Mobilux EP Grease
Gulf Oil	Gulfcrown EP Grease
Shell	Alvania EP Grease

Minimum re-lubrication, whether or not the bearing is used, is every six months. Slow rotating, intermittently used equipment should be re-lubricated at least after every 100 hours of use. Equipment that is turning continuously or that operates in an adverse environment, should be re-lubricated after every 8 hours of use, or more often if required.

While adding grease, the bearing should be rotated to spread the grease throughout.

Lubrication of the Gear

If the bearing has an integral gear, the gear should be coated with an appropriate grease. Mobil Gear Lube 275 or Gearite HVY supplied by the Union Oil Company are some suggested greases for manually applied exposed gearing.

If the gear is enclosed, protected or shrouded (except enclosed gear boxes) the same type of grease as is used for the bearing raceway may be used for the gear.

Because the meshing action of the gear teeth and pinion teeth tends to push the grease out of the critical areas, the gear should be lubricated more often than the bearing. Small amounts of the lubricant should be applied to the point of mesh between the gear and the pinion.

Slow rotating, intermittently used gears should be re-greased every eight hours, more often for moderate to fast rotating or continuous operating gears. The gear should be rotated while lubricating for even distribution.

Maintenance

General Maintenance

Periodically perform a visual inspection at a minimum of every six months. Inspect the seals to ensure they are properly inserted into their grooves and that they are wholly intact and preventing contaminants from entering the bearing.

Bolts should be checked periodically to ensure proper pre-tension. Improperly pre-tensioned bolts can fail, causing damage to equipment and/or harm to human life.

Bearing Storage

Store bearings in the horizontal position in the wrapping and containers in which they were shipped. Do not store outdoors or in an environment that is not temperature controlled.

If a bearing is not used within one year, carefully unwrap it and apply new grease, rotating the bearing to distribute the grease evenly throughout. Cover the outer metal surfaces with a rust preventative coating and re-wrap the bearing for storage.

For detailed maintenance instructions, please request Avon Bearings SPEC #BES 5-1-050.

Maximum Clearance Values

TABLE 3

MAXIMUM ALLOWABLE CLEARANCE (INCHES) FOR SINGLE ROW, 4 PT. CONTACT BALL BEARINGS WITH THE EXCEPTION OF SERIES M BEARINGS.

BALL DIAMETER	.75"	1.00"	1.25"	1.50"	1.75"	2.00"	2.25"	2.50"	2.75"	3.00"
RACEWAY DIAMETER										
40 INCHES	0.047	0.050	0.053	0.067	0.074	0.087				
50 INCHES		0.054	0.057	0.071	0.078	0.090	0.093			
60 INCHES		0.057	0.060	0.072	0.081	0.092	0.097			
70 INCHES			0.062	0.074	0.082	0.094	0.101	0.102	0.104	0.106
80 INCHES			0.064	0.078	0.084	0.098	0.102	0.104	0.107	0.109
90 INCHES				0.082	0.087	0.101	0.104	0.107	0.111	0.112
100 INCHES				0.083	0.091	0.103	0.108	0.109	0.112	0.114
110 INCHES				0.084	0.093	0.105	0.111	0.113	0.114	0.116
120 INCHES				0.088	0.095	0.108	0.113	0.114	0.118	0.120

Clearances identified above are axial clearances with the bearing mounted on a given unit of equipment and reflect nominal loading from the upper structure and counterweights, if any.

TABLE 4

MAXIMUM ALLOWABLE CLEARANCE (INCHES) ROLLER BEARINGS

ROLLER DIAMETER	.625"	.875"	1.00"	1.25"	1.50"	1.75"	2.00"	2.25"	2.50"
RACEWAY DIAMETER									
40 INCHES	0.011	0.012	0.013	0.014	0.015				
50 INCHES	0.015	0.016	0.016	0.017	0.019	0.021	0.022		
60 INCHES	0.018	0.019	0.019	0.021	0.023	0.024	0.025		
70 INCHES			0.021	0.023	0.024	0.025	0.027		
80 INCHES			0.023	0.025	0.026	0.027	0.029	0.032	0.035
90 INCHES			0.024	0.026	0.028	0.029	0.030	0.033	0.036
100 INCHES			0.026	0.028	0.030	0.031	0.032	0.035	0.038
110 INCHES				0.030	0.031	0.032	0.033	0.037	0.040
120 INCHES				0.032	0.033	0.034	0.035	0.039	0.042

Clearances identified above are axial clearances with the bearing mounted on a given unit of equipment and reflect nominal loading from the upper structure and counterweights, if any.

Before the equipment is put into operation measure and record the tilting clearance of the bearing. This will establish a base measurement for subsequent repeat measurements, which should be taken periodically in order to determine the amount of wear that has occurred in the bearing. The wear present in the bearing raceway is evidenced by the change in the axial motion of the bearing.

Tilting clearance can be measured on equipment that allows both a positive and negative application of moment loads. Measuring points as depicted in Figure 3 should be permanently marked around the circumference while the center of gravity of the load is kept in a specified position. Measurements are then taken between the lower mating structure and the bearing mounted to the upper structure, as shown in Figure 4.

Take the measurements as close as possible to the outer bearing race to eliminate mounting surface deflection as a possible contributor to clearance measurement.

Using dial gages that have an accuracy of at least 0.001", begin by applying the maximum backward moment and set the dial gages to zero. Next, apply a forward turning moment. Record the first measurement.

At least once every twelve months repeat the measurement procedure, using the same loads as originally used, and compare the readings with the base values originally recorded. The difference between the two represents the wear that has occurred in the bearing. Shorten the time intervals in between measurements as the wear increases.

Tables 3 and 4 on page 2-15 show the maximum allowable clearance values. If wear values are exceeded, contact Avon Bearings to discuss bearing repair or replacement recommendations.

Figure 3

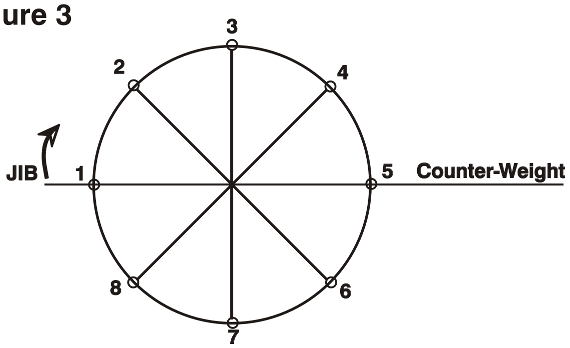
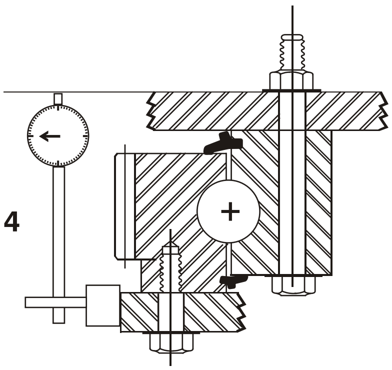


Figure 4



Bearing Preparation

Avon Bearings are covered with a rust preventative/ preservative coating in order to protect them during shipment and storage. This coating must be removed from the bearing before mounting. An all-purpose industrial strength cleaner can be used.

Remove any nicks or burrs from the bearing that may have been caused during shipping and handling, using a hand file.

Clean the bearing, including pilots and mounting holes, one last time to ensure that they are free of foreign materials.

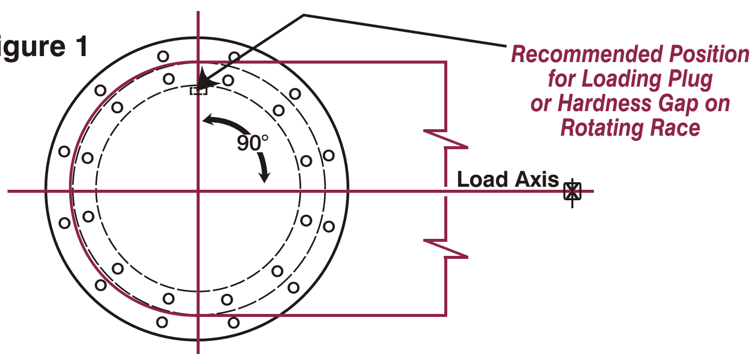
Mounting

Use eye bolts in the mounting holes for handling and moving the bearing. If slings are used, use only nylon web type or equivalent in order to avoid damage to the bearing.

Lift the bearing into position for mounting. Position the loading plug and the spot marked with a “G” on the opposite race 90 degrees from the loading axis whenever possible as depicted in Figure 1. These are the unhardened “soft spots” in the induction hardened raceways. They should be located perpendicular to the axis of principle loading.

Mount the bearing to both surfaces of the machine, aligning it with the bolts. Always use Grade 8 bolts or better. Do not distort the bearing’s races to align it with the bolts. Do not tighten bolts at this time. First, check the bearing torque by rotation and record the results.

Figure 1



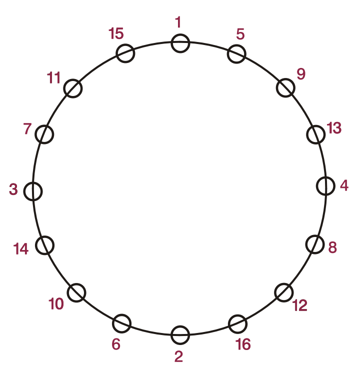


Figure 2

If the gear backlash is adjusted by means of locating the bearing with respect to a fixed drive system, it should be done at this time. The point of minimum gear clearance on the bearing is marked on the face of the gear teeth with paint.

Tighten all bolts by the method recommended and to the level of preload as instructed by the bolt manufacturer. The preferred technique when tightening the bearing mounting bolts would be to tighten in the order depicted in Figure 2.

Continue tightening bolts until all bolts are tightened and the bearing is uniformly secured to the structure. Note that improper bolting can cause failure of the equipment.

Recheck the bearing torque and compare it to the first reading. If the torque of the mounted bearing has increased significantly over the first reading, the bearing is being distorted. This condition needs to be corrected before continuing. Periodic torque checks should be taken during the bearing installation and compared to the first reading. There should be no significant difference.

Adjustable drive gears can now be adjusted for proper backlash.

For detailed installation instructions, request Avon Bearings SPEC #BES 5-2-050.

To transmit loadings approaching the bearing's capacity, it is essential that the bearing be mounted to an adequately flat, rigid mounting structure. To achieve proper flatness, it is normally necessary to machine the mounting surface after all fabrication has been completed.

Bearings with diameters up to approximately 2 feet are often mounted to simple plate structures, utilizing the stiffness of the plate to transfer loadings from major structural elements into the bearing and vice versa. Figures 2-12A and 2-12C show such arrangements.

As bearing diameter increases, the required plate thickness tends to become prohibitive in cost and weight so that the plate stiffness must be supplemented by gusseting as shown in Figures 2-12B and 2-12D.

The vast majority of large bearings are operating successfully mounted to structures such as have been described above. It can be argued, however, that regardless of how well gusseted a structure is, there will be some load concentration at the points where the load-introducing members intersect the bearing diameter. The ideal structure is a cylinder, with sufficient depth to isolate the bearing mounting surface from all localized force concentrations. A rule of thumb is for the required cylinder depth to be **60%** of its diameter.

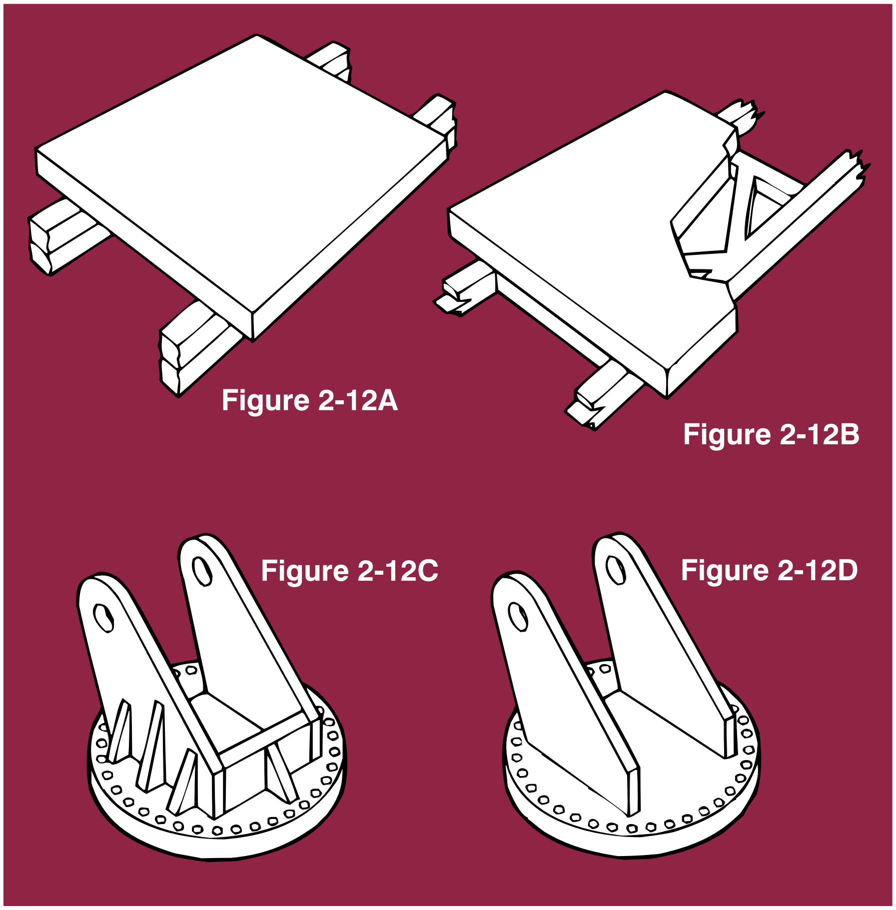


Figure 2-12A

Figure 2-12B

Figure 2-12C

Figure 2-12D

Mounting Structure Preparation

The equipment mounting surfaces must be machined flat and any weld spatters, chips and/or contaminants removed. Unpainted surfaces are required and mounting surface must be rigid.

Though shimming is not recommended, if shimming is required, shim material must be the same as the mounting structure material.

Note that excessive distortion caused by non-flat mounting surfaces or insufficiently rigid structures will result in higher torque and localized loading, most likely leading to reduced bearing life.

TABLE 1

Initial Mounting Surface Error within 90° Arc

Raceway Diameter	Single Row Ball Bearings	Roller Bearings
Up to 40"	0.006"	0.004"
Up to 60"	0.007"	0.005"
Up to 80"	0.009"	0.006"
Up to 100"	0.010"	0.007"
Up to 120"	0.012"	0.008"

TABLE 2

Raceway Diameter	Maximum Deflections Under Peak Operating Loads
40"	0.024"
60"	0.030"
80"	0.040"
100"	0.052"
120"	0.064"

For detailed mounting structure requirements request Avon Bearings SPEC #BES 5-2-051.

Most Avon bearings are equipped with integral gearing. The material used in the bearing rings is well suited for high strength gearing. Virtually any configuration of gearing can be provided. The most popular tooth forms are 20° stub, full depth and module.

Most designers prefer the stub tooth for low speed, intermittently rotated applications. The stub tooth provides greater beam strength than the full depth and module tooth.

For continuously operating applications, particularly with higher speed, the full depth tooth form provides the advantage of higher contact ratio, resulting in somewhat smoother, quieter operation and greater resistance to fatigue failure of the tooth surfaces.

Internal and external gears can be cut to precision levels meeting AGMA Class 10.

Gear Hardness

Avon Series M as well as a few series H, T and R bearings are produced with normalized rings providing gear tooth hardness in the range of 180-248 BHN. Nearly all other Avon bearings utilize heat treated rings, mostly with core hardness of 248-302 BHN. A few are produced with core hardness ranges of 285-321 BHN. This latter value is the greatest hardness which can be conveniently and economically cut with standard gear cutting equipment. (The term core hardness is used to differentiate from raceway hardness, which is produced by secondary hardening operations.)

For heavily loaded equipment which has a high rotational speed, the bearing gear can be provided with a secondary heat treating on the surface, typically producing surface hardness of approximately 55 Rc. The surface hardening increases the gear capacity from a standpoint of surface endurance and is desirable in high duty cycle applications. If the gearing utilizes full root radii and if the surface hardening is uniform throughout the root, there is some increase in the beam strength. As noted above, this high hardness is of no value unless precise gear and pinion alignment is assured and the gearing is protected from heavy shocks. Avon can produce any of the variations in gear profile and metallurgy discussed above.

Backlash Allowance

In order to provide for unavoidable dimensional variation, thermal expansion, and lubricant clearance, the teeth in one or both gears in the set are cut somewhat thinner than nominal. This thinning is referred to as“backlash allowance”. In the case of high ratio gearing, normally only the gear is thinned, while the pinion is kept at nominal thickness in order to maximize its tooth strength. The tooth thickness on geared Avon bearings is specified on outline drawings which are available on request. Please contact Avon Bearing Engineering for assistance in the determination of proper assembled backlash.

Gear Capacity

Tangential tooth force capacity is shown in the dimension tables for each catalog bearing. These capacity values are based on static bending strength according to the Lewis Equation with a bending stress value of 25 percent of ultimate tensile strength. Experience has shown this to be a practical limitation for low speed intermittent operation gear loading. This value should not be exceeded in routine acceleration or deceleration. This rating allows for some reasonable amount of gear misalignment resulting from pinion shaft deflection, etc. When such misalignment exists, some permanent deformation of the gear teeth may result, however not to the extent that it interferes with normal function. Occasional peak loads, up to 200 percent of the rated load, can usually be tolerated, albeit with some additional localized deformation. These gear ratings consider only bending strength. On gear sets which will frequently be exposed to heavy loading, fatigue resistance should also be considered. Tooth bending fatigue and surface endurance may be calculated according to procedures published by the American Gear Manufacturers Association (AGMA). In order to make meaningful fatigue calculations it is essential to provide a realistic estimate of expected duty cycle showing frequency of operation at various load levels. It is also essential that the extent of gear misalignment be accurately estimated. If you can provide this data, Avon will be pleased to perform gear fatigue calculations.

Mounting Holes Figure 2-10A

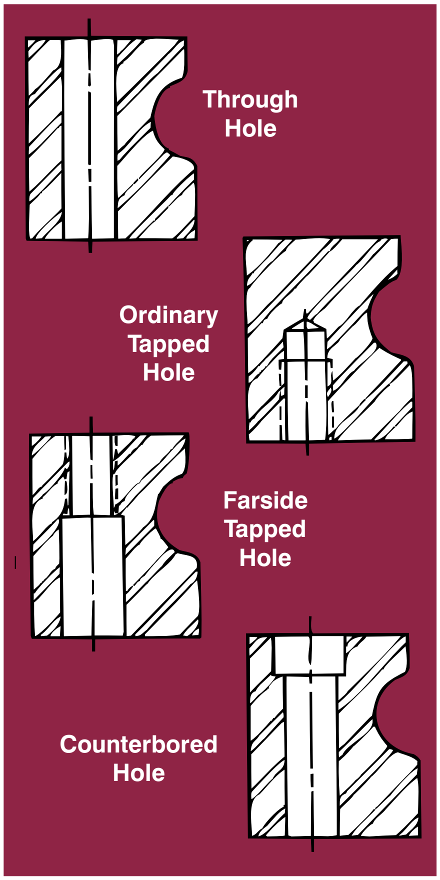


Figure 2-10A illustrates typical mounting hole arrangements. From an engineering standpoint, the ideal means of mounting is by bolts passing completely through the bearing ring and the mounting structure. For cyclically loaded applications, it is necessary to have relatively long fasteners. Long fasteners retain a higher percentage of their original prestress after the seating of the threads and the other surfaces in the joint. Active fastener length should be a minimum of 4 times the fastener diameter. The use of through bolts makes it relatively easy to obtain this minimum ratio. Hex head bolts and hex nuts with hardened flat washers under each permit the use of generous clearance holes in both bearing ring and mounting structure, while still providing substantial load bearing area under the washers.

Counterbored holes, are occasionally necessary to avoid interference with other components. In this case, the load bearing area under the heads is reduced and the allowable clearance in the mounting holes may be reduced, requiring somewhat more precise hole locations.

Tapped holes require the most precise hole locations because the total location tolerance in both mating holes must be accommodated by the clearance in the through hole.

Unless the mounting structure is exceptionally thick, ordinary tapped holes often result in unacceptably short fasteners. Far-side tapped holes, Figure 2-10C, provide for increased active fastener length and are recommended when through holes cannot be used.

Figures 2-10B through 2-10E illustrate the variation in active fastener length in various mounting arrangements. If a machine is equipped with a built-up deck structure, it is advantageous to bring the fasteners through the structure to maximize their length.

Figure 2-10E shows a method of increasing active fastener length often used as a field repair to eliminate loosening problems caused by inadequate fastener length.

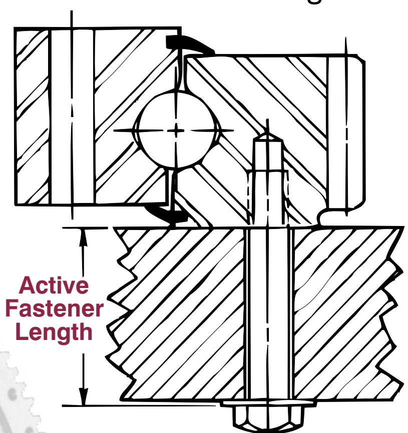


Figure 2-10B

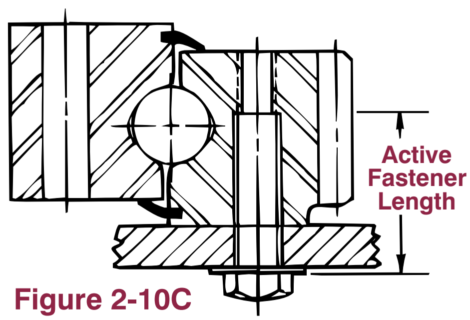


Figure 2-10C

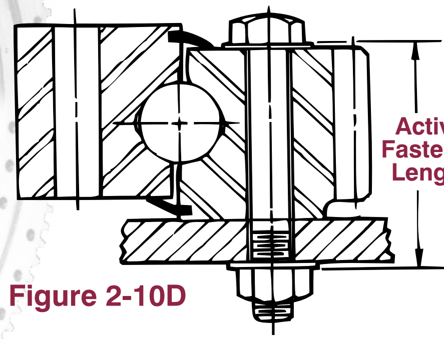


Figure 2-10D

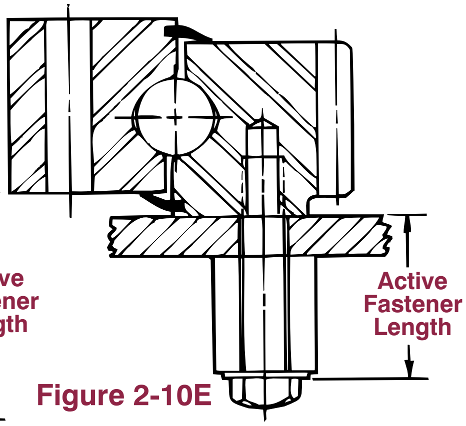


Figure 2-10E

Recommended Fastener Types

All calculations in this section are based on SAE Grade 8 strength values. Fasteners of this strength or greater are recommended for all applications of bearings with significant tilting moment loading. Fasteners should be tightened to provide prestress of 75-80% of proof load. For applications subject to repetitive loading, the active fastener length should be at least 4 times diameter. Coarse thread fasteners are recommended. Avon Bearings recommends that fasteners be identified with means of traceability to the manufacturer as well as the standard SAE grade code.

Duty Cycle

This table provides maximum recommended fastener load values for each of the following conditions:

Fatigue Loading - A consistently high moment loading applied day in and day out. Examples of fatigue loading would be industrial machine tool and other dynamic applications.

Intermittent Loading - An occasional high moment loading applied at inconsistent intervals, such as truck mounted cranes. This load rating should be limited to 20,000 cycles. If the bearing bolts are to be subjected to more than 20,000 cycles, fatigue load rating should be utilized.

Maximum Loading - Under this loading, a one time test load can be undertaken. This load will cause the fastener material to approach its stress limit or the beginning of joint separation to occur. ***CAUTION: THIS LOADING SHOULD NEVER BE EXCEEDED.***

Washers

Avon Bearings recommends that hardened steel flat washers be used under all bolt heads and nuts.

Nuts

Nuts should be of the same quality and grade as bolts.

Compressive Mountings

All designs in this catalog presume that the bearing is mounted in compression. If bearing is to be mounted in tension, please refer to Avon Engineering, and note that in addition to requiring additional fasteners, substantial changes in internal geometry may be required for tension mounted applications.

Maximum Fastener Loads (MFL)
Based on SAE Grade 8 Bolts

Bolt Size	Fatigue Loading	Intermittent Loading	Maximum Loading
1/4-20	1018	1696	2181
5/16-18	1677	2795	3593
3/8-16	2480	4133	5314
7/16-14	3402	5669	7289
1/2-13	4540	7568	9730
9/16-12	5824	9707	12,480
5/8-11	7232	12,053	15,497
3/4-10	10,688	17,813	22,903
7/8-9	14,784	24,640	31,680
1-8	19,392	32,320	41,554
1 1/8-7	24,416	40,693	52,320
1 1/4-7	31,008	51,680	66,446
1 3/8-6	36,960	61,600	79,200
1 1/2-6	44,960	74,933	96,343
1 3/4-5	60,800	101,333	130,286

Static and Dynamic Load Analysis

Ideally, the load spectrum should be determined or estimated, as in the following example:

Condition	Speed	Thrust Load	Radial Load	Moment Load	% Operating Time
1	12 RPM	30,000 lbs.	2,000 lbs.	60,000 ft.-lbs.	5%
2	18 RPM	24,000 lbs.	1,500 lbs.	50,000 ft.-lbs.	39%
3	18 RPM	8,000 lbs.	1,000 lbs.	20,000 ft.-lbs.	25%
4	30 RPM	2,400 lbs.	1,000 lbs.	25,000 ft.-lbs.	30%
5	12 RPM	40,000 lbs.	2,500 lbs.	65,000 ft.-lbs.	1%
6	Various	30,000 lbs.	3,000 lbs.	100,000 ft.-lbs.	Less than 1%

Static Load and Capacity

Condition No. 6 is the peak loading. In most applications this maximum load occurs momentarily. It is usually induced by shock of some form, rapid acceleration or deceleration. If the bearing is on mobile equipment, it may be desirable to compute the peak load which comes from traveling over rough surfaces. With vehicles moving at substantial speed, forces resulting from pitching and rolling must be considered. If the application is a crane-like device, consideration must be given to stopping a load while it's being caught before it reaches the ground. If the bearing is used for steering on a multiple wheeled device, consideration must be given to times when the wheel or truck sees more than its share of the load due to uneven terrain. In general, any time a machine element stops, starts, or changes direction, forces are produced which, if passed through the bearing, should be considered. The first consideration in sizing a bearing is making certain that the **static** raceway capacity is adequate to handle the peak loading. The static capacity must be considered regardless of whether the bearing is rotating or stationary when the peak load occurs. If the bearing is inadequate in capacity to handle these loads, indentations or grooves will be formed in the raceway. This is referred to as brinnelling. In some slow speed applications, slight brinnelling may not be harmful. However, overloads that produce indentations will lead to rough operation and very short bearing life. Overload may also, of course, lead to structural failure. In bearings subjected to continuous or frequent rotation, even slight brinnelling will shorten raceway life.

Dynamic Load and Capacity

Based on a load spectrum as shown above, a prediction can be made for the life of a specified bearing or, more precisely, it is possible to predict the probability that an individual bearing will survive a specified load spectrum. The dynamic raceway capacity deals with the resistance to fatigue flaking or spalling of the raceway and rolling element surfaces. The basic parameters were determined statistically after years of research into the relationship between contact stresses and material properties.

The L_{10} life of a bearing is the predicted life that 90% of a group of bearings will meet or exceed when subjected to the same loads and operating conditions. The average life for ball bearings will be approximately 5 times their L_{10} life and may be referred to as L_{50} life. If the above information on loading can be developed and stated along with the life requirement and dimensional limitations, Avon Engineering can promptly recommend bearings to meet these requirements. The design of the mounting structure has a substantial impact on the bearing and capacity and must also be considered. Please refer to the page on mounting structures, 2-12.

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BALL BEARING ANALYSIS

Avon Part Number:	*2074B1*	**Customer:**	*ABC Corporation*
Date:	*10/18/00*	**Reference No.:**	*1234*
Application:	*Crane*		
Run By:	*RLS*		

BEARING ANALYSIS

I. Bearing Description

Ball Diameter (inches)	Pitch Diameter (inches)	No. of Balls (approx.)	Contact Angle (degrees)	Bearing IDC (inches)
2	74.02	107	60	0.0000

No. of Rows	Two Row Path Center Distance	Pair Configuration
1	0	DB

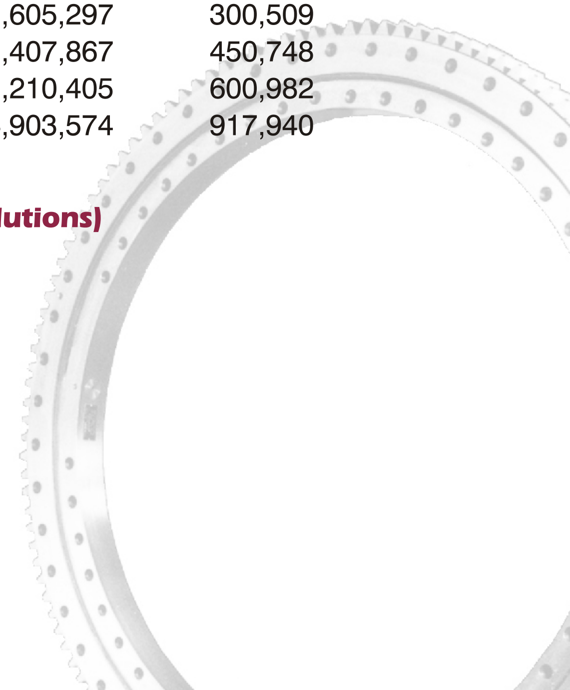
II. Bearing Analysis Results

A. Static Capacity

	Axial (lbs.)	Moment (ft.-lbs.)	Radial (lbs.)
1. Frequent Static Loading	1,301,240	1,605,297	300,509
2. Intermittent Heavy Duty	1,951,797	2,407,867	450,748
3. Theoretical Stress Limit	2,602,328	3,210,405	600,982
4. ABMA 1990	3,974,797	4,903,574	917,940

B. L10 Dynamic Capacity (at 1 Million Revolutions)

1. Radial: 77,431 (lbs.)
2. Axial: 232,184 (lbs.)



NOTE: *The above table is intended to indicate the general capacity of this bearing under a single loaded axis and is not intended for any particular application. Contact Avon Engineering for multi-axis computer analysis of your required loadings.*

Static Capacity Charts

The example shows the format of a Bearing Static Capacity Chart which is routinely supplied by Avon with proposals. The capacity values outlined in this chart should serve as a guide only. In order to determine the optimum bearing choice for a given application, a load spectrum as described on page 2-8 is needed. In many applications, however, it is impractical or even impossible to generate the necessary data. Accordingly, a system of bearing ratings has been developed based upon common industry practices, as well as experience, which provides a basis for bearing selection for most applications.

(See page 2-7, Section A 1)

Frequent Static Loading:
This rating applies to applications continuously rotating yet with peripheral speeds below 300 surface feet per minute. The Frequent Static Load rating is approximately 1.5 times less than the Intermittent Heavy Duty Load rating and about one-half of the Theoretical Stress Limit rating. Common applications which should use this rating as a guideline for bearing selection include canning and bottling equipment, industrial turntables, stretch wrapping equipment, amusement rides and coil winders.

(See page 2-7, Section A 2)

Intermittent Heavy Duty:
This rating is appropriate for applications which do not rotate continuously but rather slew through a finite range. The average bearing loading typically is about 50% of the rated load. The Intermittent Heavy Duty Load rating is about 1.33 times less than the Theoretical Stress Limit rating. Common applications which should use this rating as a guideline for bearing selection include cranes, excavators,

(See page 2-7, Section A 3)

Theoretical Stress Limit: This rating is the highest loading to which bearings should be subjected under the most adverse conditions of shock, overload, etc. The Theoretical Stress Limit Load rating is approximately two times the Frequent Static Load rating and about 1.33 times the Intermittent Heavy Duty Load rating.
All of the ratings shown in this engineering guide are the Theoretical Stress Limit ratings.

(See page 2-7, Section A 4)

ABMA 1990: This rating is based upon maximum compressive stress levels of 609,000 psi at the interface between the most heavily loaded ball and the raceway (580,000 psi for rollers) as defined by the American Bearing Manufacturer’s Association (ABMA). Exceeding these stress values theoretically will result in permanent deformation of 0.0001" per inch of rolling element diameter. The ABMA 1990 rating presumes an infinitely rigid mounting structure, perfectly flat mounting surfaces and a raceway which is through-hardened versus the conventional case-hardened designs. The ABMA 1990 rating is simply for informational purposes only and should not be considered in the bearing selection process.

Industry Standards:
There is no accepted industry-wide standard for rating the capacity of large diameter bearings. As a result, claimed capacities of identical bearings may vary from one manufacturer to another. It is also common to recommend bearings with loading more or less than the appropriate rating due to specific application data. It is vital to contact Avon Engineering for such special situations requiring this attention.

Calculating Loads

Loading on most bearings can be resolved into thrust and/or radial loading as shown in Figures 2-5A and 2-5B.

Most large diameter bearings also transmit moment loading which is resolved into thrust in one direction on one side of the bearing and thrust in the opposite direction at the other side. See Figure 2-5C. Most large diameter bearing applications involve low speed, intermittent rotation. As a consequence, most big bearings are selected based upon their static raceway capacity. Static raceway capacity refers to the ability of the bearing's raceway to resist permanent deformation when under load.

Bearings which will be subjected to frequent or continuous rotation may require consideration of dynamic raceway capacity. The dynamic raceway capacity is the ability of the bearing to resist fatigue failure of the raceway surfaces and/or rolling elements after prolonged rotation.

All of the ratings in this catalog and all of the load calculation procedures are based upon static raceway capacity limitation. Applications which require continuous or frequent rotation should be referred to Avon Engineering. Avon will provide dynamic raceway capacity data and life calculations. See Page 2-8.

Figure 2-5A Thrust (or Axial) Load

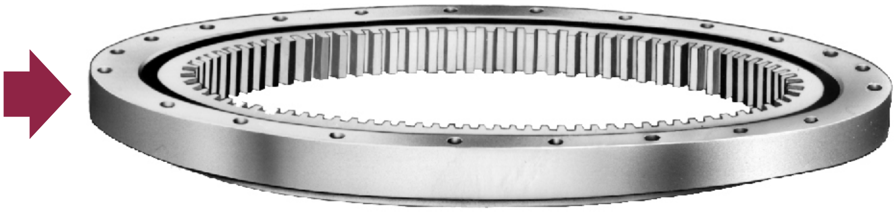


Figure 2-5B Radial Load

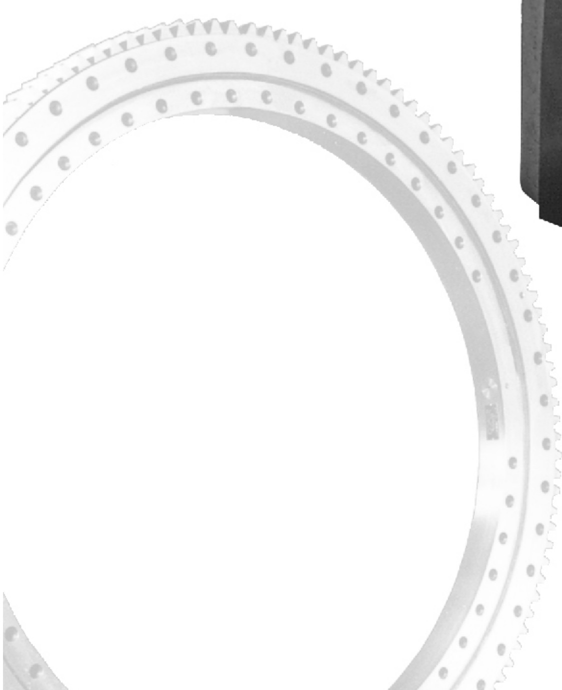
Figure 2-5C Moment Load

Engineering Guide

This catalog contains sufficient information to select bearings for most low speed, intermittent rotation applications. Raceway capacities are provided for your guidance.

If your application requires continuous rotation or peripheral speeds over 300 surface feet per minute, please request application assistance from Avon Engineering . Avon can provide modifications to catalog designs to cover high speed usage or can recommend completely different bearing styles when required.

All of the bearings described in this catalog are equipped with mounting holes and most of them are also equipped with integral gearing. Modifications to mounting hole configuration and gearing to suit specific design needs can be easily accommodated.



Applications

Utility and Municipalities

Aerial lifts, fire trucks, utility trucks, wastewater treatment, digger derricks, wind turbines

Military

Search radar, turrets, communication antennae, missile launchers

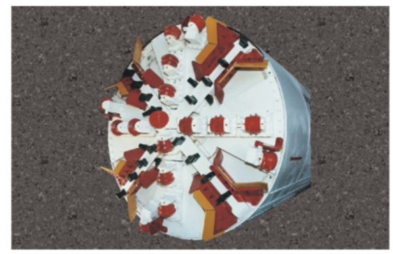
Miscellaneous

Amusement rides, scrap handling cranes, marine cranes, machine tools, railway and aerial lifts

Applications

Construction

Mobile cranes, excavators, tunnel borers, concrete pumps, tower cranes, truck cranes



Logging

Log loaders, delimbers, feller bunchers, debarkers

Mining

Stacker-reclaimers, shovels, draglines

AVON BEARINGS

ENGINEERING GUIDE and CATALOG

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