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Guidelines for Selecting Spherical Roller Bearings for Heavy Movable Structures

by

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Guidelines for selecting Spherical Plain Bearings and Spherical Roller Bearings

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Abstract:

Several criteria should be examined in the selection of bearings for bridge applications. Although selection criteria are weighed differently for each individual bridge application certain criteria are common to all. This paper will present methods for evaluating these criteria for both Spherical Plain Bearings and Spherical Roller Bearings and thus serve as a means of comparing these two different types of bearings.

Both bearing types will be compared using the following criteria: 1.) load carrying capacities, both static and dynamic, 2.) speed limits, 3.)frictional torque, 4.) misalignment capabilities, 5.) false brinelling, 6.) maintenance, 7.) costs and availability.

Guidelines For Selecting Spherical Roller Bearings For Heavy Movable Structures

By Gill Detweiler, SKF USA Inc.

Introduction

This paper presents criteria for selecting large spherical roller bearings for heavy, low speed applications. The Following criteria will be considered: 1) Static and Dynamic Loads, 2) Bearing Speed Limits, 3) Frictional Torque, 4) Misalignment Capabilities, 5) False Brinelling, 6) Maintenance, 7) Cost and Availability.

Selecting Spherical Roller Bearings

1. Static and Dynamic Loads

Table 1. gives the various Capacities for a selected list of common large spherical roller bearings as defined by ABMA (American Bearing Manufacturers Association) [1], AASHTO (American Association of State Highway Transportation Officials) [2], and AREA (American Railway Engineering Association) [3].

The ABMA Dynamic Basic Radial Load Rating, C, is defined as the constant stationary equivalent radial load which a rolling bearing could theoretically endure for one million revolutions [1]. It has been adjusted to account for the improvements in steel quality that have occurred in the last few decades since the calculation was defined. Most major bearing manufacturers have made this adjustment in approximately, but not the same way, so that there may be slight differences between capacities for different bearing manufacturers. The theoretical bearing life is calculated by:

$$L_{10} = [16{,}667(C/P)^{10/3}]/n$$

Where:

L_{10} = Basic Rating Life in hours

C = Basic Dynamic Radial Load Rating in pounds or newtons

P = Dynamic Equivalent Radial Load in pounds or newtons

n = revolutions per minute

For spherical roller bearings under pure radial load the Dynamic Equivalent Radial Load is equal to the applied load. For bearings subjected to combined axial and radial loads the Dynamic Equivalent Radial Load may be determined using factors defined in the ABMA Standards or any major bearing manufacturer's catalog. For bearings operating at very low speed or subjected to heavy shock loads the bearing size should be based on the static rating rather than the dynamic rating.

The Basic Static Load Rating, Co, is defined by ABMA to approximate the load which will result in a permanent, plastic deformation of the bearing raceway and roller that is 0.0001 of the roller diameter in depth. For smooth, vibration-free applications the Static Equivalent Bearing Load should be less than 1/2 of the Basic Static Radial Load

Rating for downwardly loaded applications and less than 1/4 for upwardly loaded applications. For downward applications where there is pronounced shock loading the Static Equivalent Bearing Load should be less than 1/3 of the Basic Static Radial Load Rating.

Bearing number	Bearing size, bore x OD x width	Adjusted Basic Dynamic Load Rating, C	ABMA Basic Static Load Rating, Co	AREA/ AASHTO Maximum Allowable Force	Minimum arc of rotation
	mm	kips	kips	kips	degrees
23284 CAK	420x 760x 272	1,420	2,610	307	20
23292 CAK	460x 830x 296	1,660	3,080	363	20
232/500 CAK	500x 920x 336	2,200	4,110	488	18.9
232/530 CAK	530x 980x 355	2,500	4,590	551	20
232/560 CAK	560x 1030x 365	2,590	4,950	581	20
232/600 CAK	600x1090x388	2,950	5,730	668	18.9
231/530 CAK	530x 870x 272	1,600	3,150	359	17.1
231/670 CAK	670x 1090x 250	2,450	5,040	576	16.4
230/600 CAK	600x 870x 200	1,180	2,560	285	12.4
230/750 CAK	750x 1090x 250	1,900	4,180	465	12.4
230/850 CAK	850x 1220x 272	2,100	4,860	539	12
239/710 CAK	710x 950x 180	1,070	2,700	290	9.5
239/800 CAK	800x 1060x 195	1,270	3,220	345	9
239/900 CAK	900x 1180x 206	1,450	3,820	410	8.6
24064 CCK30	320x 480x 160	558	1,150	129	13.3
24156 CCK30	280x 460x 180	600	1,150	133	16.4
24168 ECACK30	340x 580x 243	1,050	1,950	228	18.9
24184 CAK30	420x 700x 280	1,290	2,560	296	17.1

Table 1. Bearing ratings according to ABMA, AASHTO, AREA and minimum arc of rotation for a selected list of common large spherical roller bearings. The AASHTO and AREA maximum allowable load is calculated according to references [2] and [3].

2. Bearing Speed Limits

For rolling bearings the maximum speed limits are usually well above the speed required for heavy movable structures.

3. Frictional Torque

Frictional torque in roller bearings is generated by load-dependent friction (caused by the microslip that results from deformation of the raceways and rollers), load-independent friction (caused by the viscous flow of the lubricant in the rolling and sliding contacts), and external friction from the seals contacting a rotating surface. In low speed applications where the speed is less than about 20 rpm, the load-independent friction can be ignored. Seal friction is beyond the scope of this paper and must be defined by the seal manufacturer; however, it can be estimated that the friction caused by the seals in large spherical roller bearings at slow speeds equals roughly one to four times the load-dependent friction depending on the number and type of seals.

Load dependent friction is calculated as follows:

$$M_1 = f_1(P_1{}^a)(d_m{}^b)$$

Where:

M_1 = load-dependent friction in newton millimeters

f_1 = a factor that depends on bearing type (see Table 2.)

P_1 = the load (see below) in newtons

d_m = bearing mean diameter = 0.5 (bore + outside diameter) in millimeters

a,b = exponents (see Table 2.)

For: $F_r / F_a > Y_2$; $P_1 = F_r[1+.35(Y_2F_a/F_r)^3]$ For most large trunnion and counterweight sheave bearings F_r / F_a will exceed Y_2; if not consult a bearing manufacturer.

Where: F_r = applied radial load in Newtons

F_a = applied axial load in Newtons

Y_2 = a factor that depends on bearing (see Table 2.)

Bearing series (First 3 digits in the number)	f_1	Exponents a	Exponents b	Y_2
232	0.00045	1.5	-0.1	2.9
231	0.00035	1.5	-0.1	3.4
240	0.0008	1.5	-0.2	3.3
241	0.001	1.5	-0.2	2.5

Table 2. Exponents and factors for calculating load-dependent friction [5] (pp 46 - 50).

4. Misalignment Capabilities

Dynamic misalignment in Spherical Roller Bearings is accommodated by the spherical inner surface of the outer ring of the bearing. It is theoretically restricted at the point where the rolling surface of the rollers starts to protrude beyond the rolling surface of the outer ring (see Figure 1) However it should be cautioned that in most applications the misalignment capabilities of the seals is a tighter constraint. The misalignment capability of the bearings is given in Table 3.

Bearing series (First 3 digits in the number)	Bearing misalignment capability Degrees +/-
232	2.5
231	1.5
240	2.0
241	2.5

Table 3. Misalignment capabilities of spherical roller bearings by series [5] (p 443).

5. False Brinelling

False Brinelling is a microscopic oxidation of two surfaces in intimate contact caused by small sliding motion between the surfaces over a relatively long period of time. Considerable research literature exists on Fretting [4]. Such research will provide the capability to simulate real conditions and predict this phenomenon in future applications. The reduction in bearing service life may be small since loads are relatively low in most heavy movable structure applications and the only detriment would be an increase in vibration which may be insignificant at the very low speeds common to these applications.

6. Maintenance

Bearing maintenance is especially critical in heavy movable structure applications because of the anticipated long service life, safety hazards, and potential for extreme costs in time and money should a failure occur. To assure long service life, maintainability must be designed in from the initial conception. Following are some important initial considerations:

a. The seals should have ample space for replacement and a replacement set should be on hand. A typical seal arrangement is shown in Figure 1.

b. The required surface finish and roundness for seal contact surfaces should be specified to assure that they meet the seal manufacturer's recommendation.

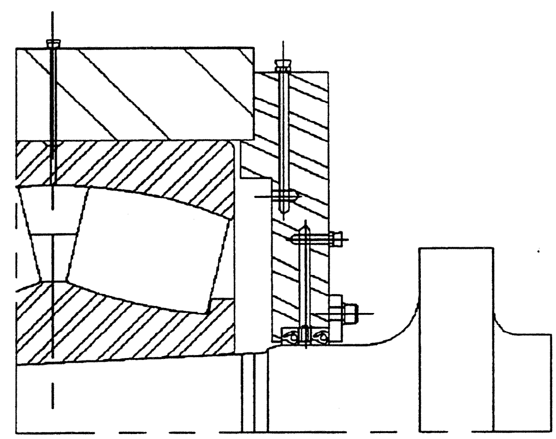

Figure 1. A typical bearing seal arrangement

c. The performance of any seal will be improved with better alignment so the best possible alignment should be achieved.

d. For most heavy movable structures, double seals are a wise investment. There should be provision for applying grease between the seals so the seal lip contact can be lubricated independently of the bearing. A double seal is shown in Figure 2.

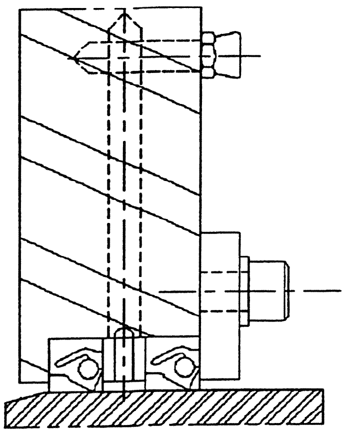

Figure 2. Typical Double Seal Arrangement

e. The grease should be selected early in the design stages and should be specified on the drawings. Compatibility with the seals and possible contaminants should be considered. Extreme Pressure additives are generally desirable and viscosity should be as high as possible considering the full expected temperature range of operation because speeds are rarely not high enough to develop full elastohydrodynamic lubrication (EHL).

f. Since rotational speeds are slow and operation is intermittent, neither temperature or vibration monitoring will normally be adequate to evaluate bearing condition. The preferred monitoring procedure is to take initial and regular periodic grease samples at the bearing contact area and at the seals. If there is a problem, analysis of the grease samples will reveal metal wear particles from the bearing and ingress of contaminant at the seals so that remedial action can be taken before the bearing is damaged.

g. With oscillating movement, false brinneling can be reduced if the angle of oscillation is greater than the arc separating adjacent rollers. This will allow fresh lubricant to be distributed to the rolling surface with each cycle. The minimum arc of rotation is given in Table 1.

h. Proper mounting of the bearing is critical to performance since it sets radial internal clearance and shaft-to-bearing inner ring interference. Radial internal clearance, the diametrical clearance between the rollers and the bearing outer ring, determines the number of rollers which will share the load and as such has an influence on bearing life. The interference between the bearing inner ring and the shaft eliminates fretting corrosion by keeping these surfaces in intimate contact during rotation. The preferred method of mounting is to drive the bearing onto a tapered shaft while measuring radial internal clearance and/or axial drive up to assure that both radial internal clearance and shaft interferences are achieved. Figure 3. shows a typical tapered bore mounted bearing.

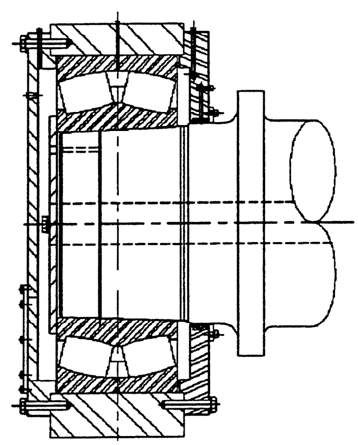

Figure 3. Shows a typical tapered bore bearing mounted on a tapered shaft.

7. Cost And Availability

Large spherical roller bearings are produced at only a few locations in the world. The extremely small tolerances and strict metallurgical controls necessitate volume runs for economically competitive manufacture. Tight schedules and economic factors that exist today can no longer permit these bearings to be custom designed and manufactured. As a result, designers and end users will increasingly select sizes that are produced in volume to obtain bearings at reasonable prices. The bearings listed in Table 1. are in common usage in the paper and/or steel industries and if not always readily available are at least tooled for production and are manufactured on a regular basis by major producers. Many of these bearings are in inventory in the United States. When selecting large spherical roller bearings it is wise to consult major bearing manufacturers to assure product quality for dependability and long service life.

Large bearing housings, on the other hand, are generally manufactured on a small lot basis so that special features like heavy load capability, special mounting, special lubrication, ventilation devices, etc. can usually be incorporated if they can be designed in at the early stages.

Conclusion

There are numerous things to consider in selecting large bearings. Although this paper explains many of them, questions will arise that application engineers, available at most competent bearing manufacturers, should be able to answer.

REFERENCES

[1] Load Ratings and Fatigue Life for Roller Bearings; American National Standard AFBMA Standard, ANSI/AFBMA Std. 11-1990 (July 17, 1990)

[2] Standard Specifications for Movable Highway Bridges; American Association of State Highway and Transportation Officials, Inc. (1988)

[3] Manual for Railway Engineering; American Railway Engineering Association (1993)

[4] Fretting Tests on Four Greases, Heavy Movable Structures, Inc., The Sixth Biennial Symposium, Herguth Laboratories for SKF Condition Monitoring, (1996)

[5] General Catalog 4000 US, SKF USA Inc., 1100 First Ave, King of Prussia, Pennsylvania (1991)