1.4 Bearing Loads

The first step in sizing a suitable ball bearing for a given application is the determination of the loads which it has to support. In this section, we list some of the most frequently occurring mechanical configurations and the bearing loads imposed by them.

(a) Radial Shaft Load Between Bearings

P = radial load
R_1, R_2 = bearing loads
l_1, l_2 = distances from radial load to bearings

$$R_1 = \frac{l_2 P}{l_1 + l_2} \qquad (1)$$

$$R_2 = \frac{l_1 P}{l_1 + l_2} \qquad (2)$$

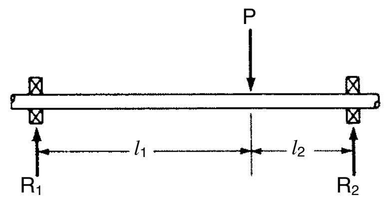


Fig. 1-6 Radial Load Between Bearings

(b) Overhung Radial Load

Notation same as in paragraph **(a)**.

$$R_1 = \frac{l_2 P}{l_1 - l_2} \qquad (3)$$

$$R_2 = \frac{l_1 P}{l_1 - l_2} \qquad (4)$$

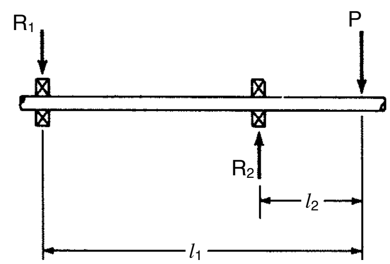


Fig. 1-7 Overhung Radial Load

For cases other than those shown above, the rules of static distribution of loads on a beam should be considered. The shaft which is supported by bearings is nothing else but a beam subjected to forces which result in radial loading of bearings.

1.5 Determination of Bearing Size

(a) Basic Definitions

In the course of many years of experience with ball bearings and extensive testing, it has been found that the prediction of the load capacity of a ball bearing is a statistical event related to the fatigue life of the bearing. This makes the sizing of ball bearings more difficult than that of many other machine elements.

A basic phenomenon in ball bearings is that ball bearing life has been found to be inversely proportional to the cube of the bearing load. This means that when the load is doubled, the life expectancy of the bearing is reduced by a factor of eight. This phenomenon has been studied extensively and has led to the adoption of an industry-wide national standard for rating ball bearings pioneered by the American Bearing Manufacturers Association (formerly Anti-Friction Bearing Manufacturers Association, Inc.), 1200 19th Street, N.W., Suite 300, Washington, D.C. 20036-2433.

The following represents a summary of the load rating of ball bearings of less than one inch in diameter, according to ANSI-AFBMA Standard 9-1978: “Load Rating and Fatigue Life for Ball Bearings” – reprinted with the permission of the American National Standards Institute, Inc., 11 West 42nd Street, 13th Floor, New York, N.Y. 10036.

Ball bearings were formerly rated on the basis of the compressive stress in the most heavily loaded ball. Except for static loads, experience has shown that the actual cause of failure is fatigue. Fatigue characteristics are thus used for load rating and are dependent to a large extent on experimental results.

The life of a ball bearing is the life in hours at some known speed, or the number of revolutions, that the bearing will attain before the first evidence of fatigue appears on any of the moving elements. Experience has shown that the life of an individual ball bearing cannot be precisely predicted. Fatigue characteristics are thus used for load ratings.

Even if ball bearings are properly mounted, adequately lubricated, protected from foreign matter, and are not subject to extreme operating conditions, they can ultimately fatigue. Under ideal conditions, the repeated stresses developed in the contact areas between the balls and the raceways eventually can result in fatigue of the material which manifests itself as spalling of the load carrying surfaces. In most applications, the fatigue life is the maximum useful life of a bearing. This fatigue is the criterion of life used as the basis for the first part of this standard.

The material in the standard which follows assumes bearings having nontruncated contact area, hardened good quality steel as the bearing material, adequate lubrication, proper ring support and alignment, nominal internal clearances, and adequate groove radii. In addition, certain high-speed effects such as ball centrifugal forces and gyroscopic moments are not considered.

The following nomenclature and definitions are used in life testing of bearings. A multitude of identical bearings are tested under same conditions:

RATING LIFE is the life at which 10 percent of bearings have failed and 90 percent of them are still good. This value is designated as L_{10} and is expressed in millions of revolutions.

LIFE of an individual ball bearing is the number of revolutions (or hours at some given constant speed) designated as L which the bearing runs before the first evidence of fatigue develops in the material of either ring (or washer) or of any of the rolling elements.

MEDIAN LIFE is the life at which 50 percent of bearings failed and 50 percent are still good. It is designated as L_{50}, which is generally not more than five times the RATING LIFE, L_{10}.

BASIC LOAD RATING "C" for a radial or angular contact ball bearing is the calculated, constant, radial load which a group of apparently identical bearings with stationary outer ring can theoretically endure for a RATING LIFE of one million revolutions of the inner ring. For a thrust ball bearing, it is the calculated, constant, centric, thrust load which a group of apparently identical bearings can theoretically endure for a RATING LIFE of one million revolutions of one of the bearing washers. The basic load rating is a reference value only of the base value of one million revolutions RATING LIFE having been chosen for ease of calculation. Since applied loading as great as the basic load rating tends to cause local plastic deformation of the rolling surfaces, it is not anticipated that such heavy loading would normally be applied.

(b) Determination of Basic Load Rating

The basic load rating C for a rating life of one million revolutions for radial and angular contact ball bearings, except filling slot bearings, with balls not larger than 1 in. diameter, is given by the equation:

$$C = f_c(i \cos \alpha)^{0.7} Z^{2/3} D^{1.8} \quad \text{(lbs.)} \qquad (5)$$

where:

- i = number of rows of balls in the bearing
- α = nominal angle of contact (angle between line of action of ball load and plane perpendicular to bearing axis)
- Z = number of balls per row

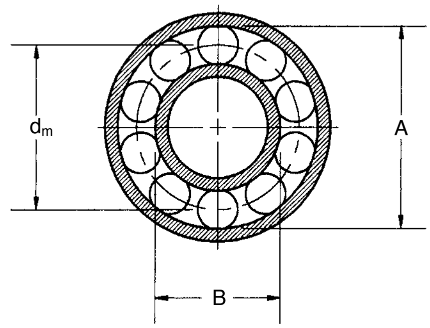
D = ball diameter
f_c = a constant from **Table 1-2**, as determined by the value of $(D \cos \alpha)/d_m$
d_m = pitch diameter of ball races

NOTE: For balls larger than 1 inch diameter, the exponent for D is 1.4.

To get a better feel for the meaning of one million revolutions, it is attained in 8 hrs at a speed of 2,084 rpm. Most ball bearings, however, may have intended life many times exceeding one million revolutions.

In the above formula, d_m represents the pitch diameter of the ball races. It can be expressed as follows:

$$d_m = \frac{A - B}{2} + B = \frac{A + B}{2} \qquad (6)$$



A and B are dimensions as shown. However, assuming that inner ring and outer ring wall thicknesses are the same, A becomes outside diameter, and B the bore of the bearing.

Values of f_c are shown in **Table 1-2** for different values of $(D \cos \alpha)/d_m$.

RATING LIFE L_{10} in millions of revolutions for a ball bearing application can be calculated from:

$$L_{10} = \left(\frac{C}{P}\right)^3 \qquad (7)$$

where:
C = the basic load rating as previously defined
and P = the load.

(c) Illustrative Examples

Example 1

Consider an ABEC 3 single row, radial ball bearing having 10 balls of 1/16" diameter, 0.300" inner race diameter and 0.452" outer race diameter in a single shield configuration.

$\alpha = 0°$ (radial bearing)
Z = 10 (number of balls)
D = 1/16" (ball diameter)

and $d_m = \frac{1}{2}(0.330 + 0.452) = 0.391"$ (pitch diameter of ball races).

Therefore, $\left(\frac{D \cos \alpha}{d_m}\right) = \frac{0.062 \times 1}{0.391} = 0.16$

From **Table 1-2** this value yields (from third column) a value of f_c=4530. Substituting these values in Equation (5) for C, we obtain:

$$C = 4530 \times 1 \times 10^{2/3} \times 0.062^{1.8} = 143 \text{ lbs}$$

This means that for a load of P = 143 lbs, the rating life of this ball bearing will be one million revolutions and 90% of a group of such ball bearings will be expected to complete or exceed this value.

Table 1-2*   Values of f_c

$\frac{D \cos \alpha}{d_m}$	Single Row Radial Contact; Single & Double Row Angular Contact, Groove Type(1)		Double Row Radial Contact Groove Type		Self-Aligning	
	Metric(2)	Inch(3)	Metric(2)	Inch(3)	Metric(2)	Inch(3)
0.05	46.7	3550	44.2	3360	17.3	1310
0.06	49.1	3730	46.5	3530	18.6	1420
0.07	51.1	3880	48.4	3680	19.9	1510
0.08	52.8	4020	50.0	3810	21.1	1600
0.09	54.3	4130	51.4	3900	22.3	1690
0.10	55.5	4220	52.6	4000	23.4	1770
0.12	57.5	4370	54.5	4140	25.6	1940
0.14	58.8	4470	55.7	4230	27.7	2100
0.16	59.6	4530	56.5	4290	29.7	2260
0.18	59.9	4550	56.8	4310	31.7	2410
0.20	59.9	4550	56.8	4310	33.5	2550
0.22	59.6	4530	56.5	4290	35.2	2680
0.24	59.0	4480	55.9	4250	36.8	2790
0.26	58.2	4420	55.1	4190	38.2	2910
0.28	57.1	4340	54.1	4110	39.4	3000
0.30	56.0	4250	53.0	4030	40.3	3060
0.32	54.6	4160	51.8	3950	40.9	3110
0.34	53.2	4050	50.4	3840	41.2	3130
0.36	51.7	3930	48.9	3730	41.3	3140
0.38	50.0	3800	47.4	3610	41.0	3110
0.40	48.4	3670	45.8	3480	40.4	3070

NOTES:

(1) a. When calculating the basic load rating for a unit consisting of two similar, single row, radial contact ball bearings, in a duplex mounting, the pair is considered as one, double row, radial contact ball bearing.

b. When calculating the basic load rating for a unit consisting of two, similar, single row, angular contact ball bearings in a duplex mounting, "Face-to-Face" or "Back-to-Back", the pair is considered as one, double row, angular contact ball bearing.

c. When calculating the basic load rating for a unit consisting of two or more similar, single angular contact ball bearings mounted "in Tandem", properly manufactured and mounted for equal load distribution, the rating of the combination is the number of bearings to the 0.7 power times the rating of a single row ball bearing. If the unit may be treated as a number of individually interchangeable single row bearings, this footnote (1) c. does not apply.

(2) Use to obtain C in newtons when D is given in mm.

(3) Use to obtain C in pounds when D is given in inches.

Suppose now it is desired to determine the "L" life of this bearing when operating at 200 rpm and a load of 50 lbs, the life being evaluated in hours of operation.

Let the life in hours be denoted by L, and let N denote the rpm of the bearing. We then have:

$$L = \frac{10^6 L_{10}}{60\ N} = \left(\frac{C}{P}\right)^3 \frac{10^6}{60\ N} \qquad (8)$$

Substituting N = 200, P = 50 and C = 143 into Equation (8), we obtain L = 1949 hours.

NOTE: L_{10} is bearing life in millions of revolutions; L is bearing life in hours.

A table showing required life at constant operating speed has been given by N. Chironis ("Today's Ball Bearings", *Product Engineering,* December 12, 1960, pp. 63-77, table on p. 68). This table is reproduced below with the permission of McGraw-Hill Book Company, New York, N.Y.

Table 1-3 Required Life at Constant Operating Speed
(data from SKF Industries)

Type of Machine	Life in Hours of Operation
Instruments and apparatus which are only infrequently used. *Ex.:* demonstration apparatus, devices for operation of sliding doors.	500
Aircraft Engines.	500–2000
Machines for service of short duration or intermittent operation, where service interruptions are of minor importance. *Ex.:* hand tools, lifting tackle in machinery shops, hand-driven machines in general, farm machinery, assembly cranes, charging machines, foundry cranes, household machines.	4000–8000
Machines for intermittent service where dependable operation is of great importance. *Ex.:* auxiliary machines in power stations, conveying-equipment in production lines, elevators, general-cargo cranes, machine tools less frequently used.	8000–12000
Machines for 8-hour service which are not always fully utilized. *Ex.:* machines in general in the mechanical industries, cranes for continuous service, blowers, jackshafts.	20000–30000
Machines for continuous operation (24-hour service). *Ex.:* separators, compressors, pumps, mainline shafting, roller beds and conveyor rollers, mine hoists, stationary electric motors.	40000–60000
Machines for 24-hour service where dependability is of great importance. *Ex.:* pulp and paper machines, public power stations, mine pumps, public pumping stations, machines for continuous service aboard ships.	100000–200000

In order to provide data for larger size bearings as well as additional examples, **Table 1-4** is given.

Table 1-4 Dimensions and Basic Load Ratings for Conrad-Type Single-Row Radial Ball Bearings

Bearing No.	Bore		Outside Diameter		Width		Balls		Capacity, lbs	
	mm	inch	mm	inch	mm	inch	No. Z	Dia. D	Dynamic C	Static P_{st}
102			32	1.2598	9	0.3543	9	3/16	965	550
202	15	0.5906	35	1.3780	11	0.4331	7	1/4	1340	760
302			42	1.6535	13	0.5118	8	17/64	1660	930
103			35	1.3780	10	0.3937	10	3/16	1040	640
203	17	0.6693	40	1.5748	12	0.4724	7	5/16	1960	1040
303			47	1.8504	14	0.5512	6	3/8	2400	1240
104			42	1.6535	12	0.4724	9	1/4	1620	980
204	20	0.7874	47	1.8504	14	0.5512	8	5/16	2210	1280
304			52	2.0472	15	0.5906	7	3/8	2760	1530
105			47	1.8504	12	0.4724	10	1/4	1740	1140
205	25	0.9843	52	2.0472	15	0.5906	9	5/16	2420	1520
305			62	2.4409	17	0.6693	8	13/32	3550	2160
106			55	2.1654	13	0.5118	11	9/32	2290	1590
206	30	1.1811	62	2.4409	16	0.6299	9	3/8	3360	2190
306			72	2.8346	19	0.7480	8	1/2	5120	3200
107			62	2.4409	14	0.5512	11	5/16	2760	2010
207	35	1.3780	72	2.8346	17	0.6693	9	7/16	4440	2980
307			80	3.1496	21	0.8268	8	17/32	5750	3710
108			68	2.6772	15	0.5906	13	5/16	3060	2450
208	40	1.5748	80	3.1496	18	0.7087	9	1/2	5640	3870
308			90	3.5433	23	0.9055	8	5/8	7670	5050
109			75	2.9528	16	0.6299	13	11/32	3630	2970
209	45	1.7717	85	3.3465	19	0.7480	9	1/2	5660	3980
309			100	3.9370	25	0.9843	8	11/16	9120	6150
110			80	3.1496	16	0.6299	14	11/32	3770	3260
210	50	1.9685	90	3.5433	20	0.7874	10	1/2	6070	4540
310			110	4.3307	27	1.0630	8	3/4	10680	7350
111			90	3.5433	18	0.7087	13	13/32	4890	3950
211	55	2.1654	100	3.9370	21	0.8268	10	9/16	7500	5710
311			120	4.7244	29	1.1417	8	13/16	12350	8660
112			95	3.7402	18	0.7087	14	13/32	5090	4560
212	60	2.3622	110	4.3307	22	0.8661	10	5/8	9070	6890
312			130	5.1181	31	1.2205	8	7/8	14130	10100
113			100	3.9370	18	0.7087	15	13/32	5280	4950
213	65	2.5591	120	4.7244	23	0.9055	10	11/16	10770	8460
313			140	5.5118	33	1.2992	8	15/16	16010	11600
114			110	4.3307	20	0.7874	14	15/32	6580	6080
214	70	2.7559	125	4.9213	24	0.9449	10	11/16	10760	8740
314			150	5.9055	35	1.3780	8	1	18000	13260

Example 2:

Find the value of C for a 207 radial bearing.

Solution:

By **Table 1-4**: $d_m = \frac{1}{2}(2.8346 + 1.3780) = 2.1063$ in

$$\frac{D\cos\alpha}{d_m} = \frac{0.4375}{2.1063} = 0.208$$

By **Table 1-2**: $f_c = 4550$

By **Table 1-4**: $D = \frac{7}{16} = 0.4375$ in

$\log D = 9.64098 - 10$
$1.8 \log D = 9.35376 - 10$
$D^{1.8} = 0.2258$
$Z = 9, \quad Z^{2/3} = \sqrt[3]{9^2} = 4.327$

From Equation (5) for C: $C = 4550 \times 4.327 \times 0.2258 = 4440$ lbs, load for 1 million revolutions with 90 percent probability that it will be attained or exceeded.

(d) Relationship between Load and Number of Revolutions

In some cases, it is needed to determine the new value of the permitted loading when the number of revolutions N is changed.

Experimentally, it was proven that:

$$\frac{N_1}{N_2} = \frac{P_2^3}{P_1^3} \qquad (9)$$

where N is number of revolutions and P is radial load.

Furthermore, it was established that

$$10^6 C^3 = N_1 P_1^3 = N_2 P_2^3 = N_3 P_3^3 \text{ ... is a constant,}$$

or subsequently:

$$N_1 = \frac{10^6 C^3}{P_1^3} \qquad (10)$$

It has to be made clear that C is the basic load rating in lbs for a rating life of 1 million revolutions, and this fact establishes the above relationship.

If a bearing has a rating life expressed in number of revolutions designated by N, the life of the bearing expressed in hours, designated by L, can be found from:

$$N = 60\, n\, L$$

where n is the actual speed in rpm of the bearing.

Example 3

For Example 2 where we found C = 4440 lbs, find the radial load P_1 for a rating life of 500 hours, at 1500 rpm.

$$P_1^3 = \frac{10^6\ C^3}{N_1} = \frac{10^6\ C^3}{60\ n\ L}$$

Apply: C = 4.440 lbs, n = 1500 rpm, and L = 500 hrs

$$P_1^3 = \frac{10^6 \times 4.440^3}{60 \times 1500 \times 500} = 1.945 \cdot 10^6$$

$$P_1 = 10^2 \times \sqrt[3]{1.945} = 1250 \text{ lbs}$$

(e) Combined Axial and Radial Loads

This condition is dealt with by ANSI-AFBMA Standard 9-1978 which defines the combined load to be expressed as:

$$P = C_1\ (X \cdot i \cdot F_r + Y \cdot F_a) \qquad (11)$$

Table 1-5 Shock and Impact Factors

Type of Load	C_1
Constant or steady	1.0
Light shocks	1.5
Moderate shocks	2.0
Heavy shocks	3.0 and up

where value C_1 is a service factor which is shown in **Table 1-5**.

In the above equation:

i = race rotation factor equal 1 for inner ring rotation, 1.2 for outer ring rotation.

F_r and F_a are radial and axial components, respectively, of the load.

X and Y are factors to be used as shown in **Table 1-6**.

NOTE: Y is the axial or thrust factor determined from the value of

$$\frac{F_a}{i\ Z\ D^2}$$

Table 1-6 Values of X and Y

Bearing Type				Single Row Bearings (Fa/Fr) > e		Double Row Bearings (Fa/Fr) ≤ e		Double Row Bearings (Fa/Fr) > e		e
				X	Y	X	Y	X	Y	
	$\frac{F_a}{C_o}$	$\frac{F_a}{i Z D^2}$ Newtons	lbf							
Radial Contact Groove Ball Bearings	0.014	0.172	25		2.30				2.30	0.19
	0.028	0.345	50		1.99				1.99	0.22
	0.056	0.689	100		1.71				1.71	0.26
	0.084	1.03	150		1.56				1.55	0.28
	0.11	1.38	200	0.56	1.45	1	0	0.56	1.45	0.30
	0.17	2.07	300		1.31				1.31	0.34
	0.28	3.45	500		1.15				1.15	0.38
	0.42	5.17	750		1.04				1.04	0.42
	0.56	6.89	1000		1.00				1.00	0.44
	$\frac{F_a}{C_o}$	$\frac{F_a}{i Z D^2}$ Newtons	lbf							
Angular Contact Groove Ball Bearings with Contact Angle 5°	0.014	0.172	25	For this type use the X, Y and e values applicable to single row radial contact bearings.			2.78		3.74	0.23
	0.028	0.345	50				2.40		3.23	0.26
	0.056	0.689	100				2.07		2.78	0.30
	0.085	1.03	150				1.87		2.52	0.34
	0.11	1.38	200			1	1.75	0.78	2.36	0.36
	0.17	2.07	300				1.58		2.13	0.40
	0.28	3.45	500				1.39		1.87	0.45
	0.42	5.17	750				1.26		1.69	0.50
	0.56	6.89	1000				1.21		1.63	0.52
10°	0.014	0.172	25		1.88		2.18		3.06	0.29
	0.029	0.345	50		1.71		1.98		2.78	0.32
	0.057	0.689	100		1.52		1.76		2.47	0.36
	0.086	1.03	150		1.41		1.63		2.20	0.38
	0.11	1.38	200	0.46	1.34	1	1.55	0.75	2.18	0.40
	0.17	2.07	300		1.23		1.42		2.00	0.44
	0.29	3.45	500		1.10		1.27		1.79	0.49
	0.43	5.17	750		1.01		1.17		1.64	0.54
	0.57	6.89	1000		1.00		1.16		1.63	0.54
15°	0.015	0.172	25		1.47		1.65		2.39	0.38
	0.029	0.345	50		1.40		1.57		2.28	0.40
	0.058	0.689	100		1.30		1.46		2.11	0.43
	0.087	1.03	150		1.23		1.38		2.00	0.46
	0.12	1.38	200	0.44	1.19	1	1.34	0.72	1.93	0.47
	0.17	2.07	300		1.12		1.26		1.82	0.50
	0.29	3.45	500		1.02		1.14		1.66	0.55
	0.44	5.17	750		1.00		1.12		1.63	0.56
	0.58	6.89	1000		1.00		1.12		1.63	0.56
20°				0.43	1.00		1.09	0.70	1.63	0.57
25°				0.41	0.87		0.92	0.67	1.41	0.68
30°				0.39	0.76	1	0.78	0.63	1.24	0.80
35°				0.37	0.66		0.66	0.60	1.07	0.95
40°				0.35	0.57		0.55	0.57	0.98	1.14
Self-aligning Ball Bearings				0.40	0.40 cot ∞	1	0.42 cot ∞	0.65	0.65 cot ∞	1.5 tan α

(1) Two similar, single row, angular contact ball bearings mounted "Face-to-face" or "Back-to-back" are considered as one, double row, angular contact bearing.
(2) Values of X, Y and e for a load or contact angle other than shown in **Table 5-5** are obtained by linear interpolation.
(3) Values of X, Y and e shown in **Table 5-5** do not apply to filling slot bearings for applications in which ball-raceway contact areas project substantially into the filling slot under load.
(4) For single row bearings, when Fa/Fr ≤ e, use X = 1 and Y = 0.

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Example 4

For a bearing dealt with in Example 2, assume that it carries a combined load of 400 lbs radially and 300 lbs axially at 1200 rpm. The outer ring rotates, and the bearing is subjected to moderate shock. Find the rating life of this bearing in hours.

Solution:

$$\frac{F_a}{i\,Z\,D^2} = \frac{300}{9 \times 0.4375^2} = 174$$

$Y = 1.50$

$C_1 = 2$

$P = 2(0.56 \times 1.2 \times 400 + 1.5 \times 300) = 1440$ lbs equivalent radial load

$$N = \frac{10^6\,C^3}{P^3} = 60\,n\,L$$

$$L = \frac{10^6\,C^3}{60\,n\,P^3} = \frac{10^6 \times 4440^3}{60 \times 1200 \times 1440^3} = 410 \text{ hr,}$$ life with 90 percent probability that it will be attained or exceeded.

NOTE: The impact load on a bearing should not exceed the static capacity as given by **Table 1-4** or the race may be damaged by Brinelling from the balls. This load may be exceeded somewhat if the bearing is rotating and the duration of the load is sufficient for the bearing to make one or more complete revolutions while the load is acting.

Example 5

What change in the loading of a ball bearing will cause the expected life to be doubled?

Solution:

Let N_1 and P_1 be the original life and load for the bearing. Let N_2 and P_2 be the new life and load.

Then: $N_2 = 2N_1$

By Equation (9):

$$P_2^3 = \frac{N_1\,P_1^3}{N_2} = \frac{N_1\,P_1^3}{2N_1} = 0.5\,P_1^3$$

$$P_2 = \sqrt[3]{0.5} \times P_1 \quad \text{or}$$

$$P_2 = 0.794\,P_1$$

Hence a reduction of the load to 79 percent of its original value will cause a doubling of the expected life of a ball bearing.

(f) Variable Loading of Bearings

Ball bearings frequently operate under conditions of variable load and speed. Design calculations should take into account all portions of the work cycle and should not be based solely on the most severe operating conditions. The work cycle should be divided into a number of portions in each of which the speed and load can be considered as constant.

Suppose P_1, P_2,... are the loads on the bearing for successive intervals of the work cycl. Let

N_1 be the life of the bearing, in revolutions, if operated exclusively at the constant load P_1. Let there be N_1' applications of load P_1. Then N_1'/N_1 represents the proportion of the life consumed in this portion of the cycle.

Let N_2 be the life of the bearing, in revolutions, if operated exclusively at load P_2. Let there be N_2' applications of load P_2. Then N_2'/N_2 represents the proportion of the life consumed by load P_2.

A corresponding statement can be made for each portion of the work cycle. The sum of these proportions represents the total life of the bearing or unity. Then:

$$\frac{N_1'}{N_1} + \frac{N_2'}{N_2} + \frac{N_3'}{N_3} + \ldots = 1 \qquad (12)$$

Let N_c be the life of the bearing under the combined loading. Let $N_1' = \alpha_1 N_c$ where α_1 represents the proportion of the total life, consumed under load P_1. In a similar way, $N_2' = \alpha_2 N_c$, $N_3' = \alpha_3 N_c$, and so on. Substitution in Equation (12) yields:

$$\frac{\alpha_1}{N_1} + \frac{\alpha_2}{N_2} + \frac{\alpha_3}{N_3} + \ldots = \frac{1}{N_c}$$

Using Equation (10):

$$N_1 = \frac{10^6\, C^3}{P_1^3}, \quad N2 = \frac{10^6\, C^3}{P_2^3}, \ \ldots \text{ and so on.}$$

Combining these last two equations we can obtain:

$$\frac{1}{N_c} = \frac{\alpha_1\, P_1^3}{10^6\, C^3} + \frac{\alpha_2\, P_2^3}{10^6\, C^3} + \ldots$$, or multiplying both sides of the equation by $10^6\, C^3$

$$\frac{10^6\, C^3}{N_c} = \alpha_1\, P_1^3 + \alpha_2\, P_2^3 + \ldots \qquad (13)$$

From previous definition of α it is obvious that $\alpha_1 + \alpha_2 + \ldots$ must equal unity. The application of this equation will be demonstrated by the following examples.

Example 6

A ball bearing is to operate on the following work cycle:

Radial load of 1400 lbs at 200 rpm for 25% of the time
Radial load of 2000 lbs at 500 rpm for 20% of the time
Radial load of 800 lbs at 400 rpm for 55% of the time

Total rpm is to be 1100.

Additional conditions:

The inner ring rotates; loads are steady. Find the minimum value of the basic rating load C for a suitable bearing for this application if the required life is 7 years at 4 hours per day.

	Assumed interval, min	rpm	In assumed interval, rev.
P_1 = 1400 lbs	0.25	200	50
P_2 = 2000 lbs	0.20	500	100
P_3 = 800 lbs	0.55	400	220
	1.00		370 rpm

Since both the load as well as the speed for the particular load varies, we have to establish the actual work cycle per minute.

The following table should be constructed:

Then $\alpha_1 = \frac{50}{370}, \ \alpha_2 = \frac{100}{370}, \ \alpha_3 = \frac{220}{370}$

A working year is assumed to consist of 250 days.

Total life duration of the bearing expressed in hours will become 7 x 250 x 4 = 7000 hours, whereas this expressed in number of revolutions becomes:

$$N_c = 7000 \times 60 \times 370 = 1554 \times 10^5 \text{ revolutions.}$$

Inputting this data in the formula (13), previously derived in **1.5 (f)**:

$$\frac{10^6 C^3}{N_c} = \alpha_1 P_1^3 + \alpha_2 P_2^3 + \alpha_3 P_3^3 \ldots,$$

we obtain: $\frac{50}{370} \times 1400^3 + \frac{100}{370} \times 2000^3 + \frac{220}{370} \times 800^3 = (3708 + 21622 + 3044) \times 10^5$

$$\frac{10^6 C^3}{N_c} = 28374 \times 10^5$$

$$\frac{C^3}{N_c} = 2837.4$$

$$C^3 = 2837.4 \times N_c = 2837.4 \times 1554 \times 10^5 = 44093 \times 10^7$$
$$C = 7610 \text{ lbs}$$

In order to choose the appropriate bearing, we refer to **Table 1-4** from which we find that a bearing such as No. 308 should be satisfactory, keeping in mind there is but a 90 percent probability that the required life will be attained or exceeded.

Example 7

A 306 radial ball bearing with inner ring rotation has a 10-sec work cycle as follows:

For 2 seconds	For 8 seconds
$F_r = 800$ lbs	$F_r = 600$ lbs
$F_a = 400$ lbs	$F_a = 0$ lbs
n = 900 rpm	n = 1200 rpm
Light shock	Steady load

Find the rating life of this bearing in hours and in years of 250 working days of 2 hours each.

Solution:

Since the bearing chosen is No. 306, from **Table 1-4**:

Z = 8, D = 0.5 and i = 1.

$$\frac{F_a}{i\,Z\,D^2} = \frac{400}{1 \times 8 \times 0.5^2} = 200$$

From Table 1-6 for this value of 200, a value for Y will be 1.45 and X will be 0.56.

From Equation (11) and **Table 1-5**, for the combined axial and radial loads with light shock and 2-second duration:

$$P_1 = C_1 (X\ i\ F_r + Y\ F_a) = 1.5\ (0.56 \times 1 \times 800 + 1.45 \times 400)$$

$$P_1 = 1542 \text{ lbs (equivalent radial load)}$$

Since P_2 is a pure radial load:

$$P_2 = F_r = 600 \text{ lbs}$$

The number of revolutions for the 2-second time duration will be:

$$\frac{900}{60} \times 2 = 30$$

whereas for the 8-second time duration will be:

$$\frac{1200}{60} \times 8 = 160$$

The combined total number of revolutions in 10 seconds is:

$$30 + 160 = 190$$

then,

$$\alpha_1 = \frac{30}{190} = \frac{3}{19}, \qquad \alpha_2 = \frac{160}{190} = \frac{16}{19}$$

From formula (13)

$$\frac{10^6\ C^3}{N_c} = \alpha_1\ P_1^3 + \alpha_2\ P_2^3$$

Using C = 5120 in **Table 1-4** for bearing No. 306:

$$\frac{10^6 \times 5120^3}{N_c} = \frac{3}{19} \times 1542^3 + \frac{16}{19} \times 600^3 = 578.9 \times 10^6 + 181.9 \times 10^6 = 760.8 \times 10^6$$

$$N_c = \frac{5120^3}{760.8} = \frac{134\,218 \times 10^6}{760.8} = 176 \times 10^6 = L_{10} \times 10^6$$

This is the number of revolutions the bearing will endure. The total number of revolutions during the 10-second operation was established as being 190. Therefore, the number of revolutions per minute will be:

$$n = \frac{190}{10} \times 60 = 1140 \text{ rpm}$$

From Equation (8):

$$L = \frac{10^6 L_{10}}{60 \times n} = \frac{175 \times 10^6}{60 \times 1140} = 2558 \text{ hours}$$

This expressed in years of operation will become

$$\frac{2558}{2 \times 250} = 5.12$$ years of life with 90 percent probability of service, assuming 2 hours of operation per day

(g) Static Loading of Bearings

Up to this point we have been dealing with dynamic loading of bearings. This is the condition when there is relative motion between the rings of the bearings and the balls that are rotating. If this is not the case, as a result of static concentrated loads of the balls against the races, the depressions of the balls into the races will gradually enlarge, and permanent indentations will remain. The static capacity is ordinarily defined as the maximum allowable static load that does not impair the running characteristics of the bearing to make it unusable.

This permanent deformation under the balls is known as Brinnelling and takes place at moderate to high loads. The magnitude of the permissible load is found by methods given in the standards. Calculations for the bearings of **Table 1-4** have been made and are shown in the column headed P_{st}.

When very smooth and quiet operation is required, the loading should be no more than about one-half the static capacity.

Back and forth rotation of the shaft through small angles can cause early failure of bearings unless the load is very light. Lubrication is difficult because the oil or grease may not be replenished back of a ball or roller before the motion is reversed.

(h) Effect of Increased Confidence Levels

When a bearing is installed, there is no way of knowing whether it is one of the 90 percent that are good or one of the 10 percent that will not attain the rating life. In other words, one can have but 90 percent confidence that the bearing will achieve or exceed its rating life, usually designated L_{10}.

In some cases, a greater degree of reliability is required. The expected life will of course be reduced as the reliability is made higher. Let an adjusting factor a_1 be taken such that life L_n is equal to $a_1 L_{10}$. Factors a_1 for different values of the reliability are given in **Table 1-7**. Life L_{10} is the rating life.

Table 1-7 Constants for Designing at Different Confidence Levels

Reliability (%)	L_n	Life Adjustment Factor, a_1
90	L_{10}	1.00
95	L_5	0.62
96	L_4	0.53
97	L_3	0.44
98	L_2	0.33
99	L_1	0.21