ME 314 - Engineering Design : Mechanical Components
Lecture 21

Note Title

11.9 Failure of Rolling-Element Bearings

REBs fail by surface fatigue as described in Chapter 7, even if they are carefully installed, properly lubricated, kept free of foreign material, and are not overloaded. This is, of course, due to the repeating Hertzian stresses that occur on all surfaces of contact in the inner and outer raceways and on the rolling elements.

Since the Hertzian stresses are usually more than the endurance limit of the material, the bearing has a limited life. Failure is considered to occur when either raceway or balls (rollers) exhibit the first pit.

Typically the raceway fails first and the bearing will give an audible indication that pitting has begun. This will then lead to spalling or fracture of the rolling elements and possible jamming and damage to other connected elements.

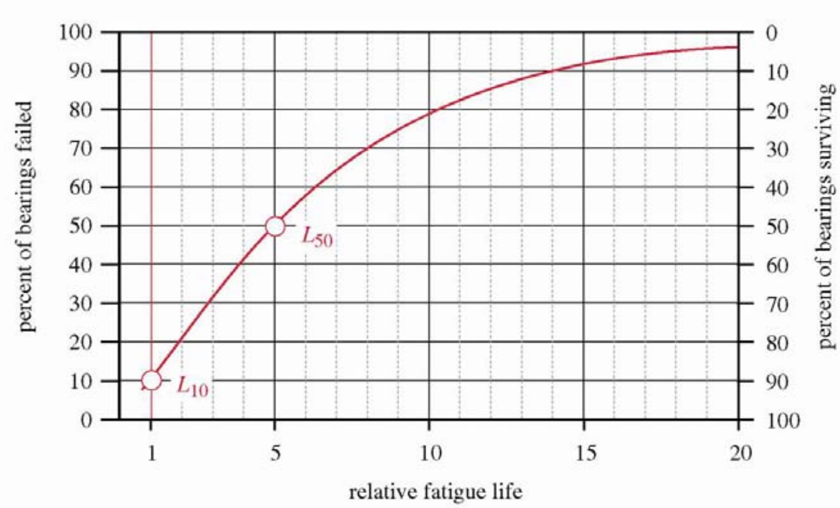
"Life" of a bearing is defined as the number of revolutions, or the number of hours of operation at a constant speed, before failure occurs.

Bearings are typically rated based on the life that 90% of a random sample of bearings of that size can be expected to reach or exceed at their design load.

In other words, 10% of the batch can be expected to fail at that load before the design life is reached. This is called the L_{10} life. Some manufacturers refer to this as B_{90} or C_{90}. We mainly use this parameter and the applied loads to select bearings.

The L_{10} life is taken as the reference in Fig. 11-22 which shows the percentage of bearing failures (survivals) as a function of relative fatigue life. The curve is linear until L_{50}. After that point the curve is highly nonlinear:

@ $5L_{10}$, 50% of bearings are running
@ $20L_{10}$, only 3% of bearings are still running



Figure

Typical Life Distribution in Rolling-Element Bearings *Adapted from SKF USA Inc.*

If we test a large number of same size bearings, we will find that their fatigue lives have a skewed distribution (i.e., not Gaussian). The mathematical formula for this distribution was proposed by the Swedish engineer, W. Weibul.

Based on Weibul's distribution, the AFBMA has recommended the life adjustment reliability factor, K_R, given in Table 11-5.

This factor is applicable to both ball and roller bearings. The life L for failure percentages other than the reference life, L_{10} (corresponding to 10% failure) can be calculated by multiplying L_{10} by K_R. Thus

P%	R%	K_R
50	50	5.0
10	90	1.0
5	95	0.62
4	96	0.53
3	97	0.44
2	98	0.33
1	99	0.21

Table 11-5
Reliability Factors *R* for a Weibull Distribution Correspondingto the Probability of Failure *P*.

$$L_P = K_R L_{10} \qquad (11.19)$$

where L_P expressed in millions of revolution, is the life corresponding to a probability of failure P; and K_R is the corresponding reliability factor given in Table 11-5.

11.10 Selection of Rolling-Element Bearings

For any application, we first choose the type of bearing that is most suitable for that application by using the information given in Figure 11-21. Next, we select an appropriate size bearing depending on the magnitude of applied loads and the desired fatigue life.

Basic Dynamic Load Rating C

The Basic Dynamic Load Rating C is defined as the radial load that will give a life of one million revolutions of the inner race. This load C is so large that it produces plastic yielding of contacting surfaces so it should never be applied. It is just used as a reference value by AFBMA. Extensive testing by bearing manufacturers has shown that

$$L_{10} = \left(\frac{C}{P}\right)^3 \quad \text{for ball bearings} \qquad (11.20a)$$

$$L_{10} = \left(\frac{C}{P}\right)^{10/3} \quad \text{for roller bearings} \qquad (11.20b)$$

where

L_{10} = Fatigue life in millions of revolutions

P = Radial applied load

From these and Eq. 11.19 we obtain:

$$L_p = K_R \left(\frac{C}{P}\right)^3 \quad \text{for ball bearings} \qquad (11\text{-}20c)$$

$$L_p = K_R \left(\frac{C}{P}\right)^{10/3} \quad \text{for roller bearings} \qquad (11.20d)$$

The values of load C for each bearing is specified in bearing manufacturers' catalogs (see Fig. 11-23).

Basic Static load Rating C_0

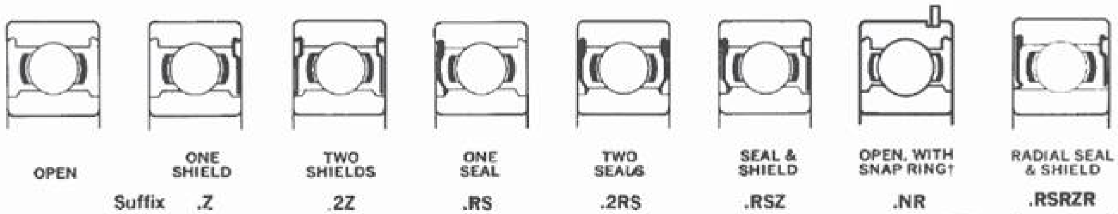
Bearing manufacturers also publish a Basic Static Load Rating C_0 for each bearing, calculated according to AFBMA standards. C_0 is the load that will produce a total permanent deformation in the raceway and rolling element at any contact point of 0.0001 times the diameter d of the rolling element. It usually takes a load of $8C_0$ or larger to fracture a bearing. The stresses required to produce this 0.0001d static deformation in steel are quite high; about 4 GPa (580 kpsi) in roller and 4.6 GPA (667 kpsi) in ball bearings. Fig. 10-23 shows values of C_0 for various bearings.

Modified Bearing Life Rating

All the life rating equations we discussed above consider only Hertzian contact stresses. ASME and ISO have recently adopted a new standard ISO 281/2 for the calculation of rolling-element bearing life which also considers the effects of friction, hoop stress from press fits, lubrication, etc. The stresses due to these effects are then used to define an equivalent von Mises stress and a stress-life factor A_{SL} which is then applied to L_{10} value of Eq. (11-20):

$$L_{ASME} = A_{SL} L_{10} \qquad (11\text{-}21)$$

This gives a more accurate result than (11.20). Calculation of A_{SL}, however, is complicated (see Ref. [30]) and in the text A_{SL} =1 is assumed.



This configuration only shown to illustrate new standard enclosures. Some bearings are now being converted.

BEARING NUMBER*	BOUNDARY DIMENSIONS BORE mm	BORE inch	O. DIAM. mm	O. DIAM. inch	WIDTH mm	WIDTH inch	SNAP RING DIMENSIONS inches H	S	t	MAX. FILLET RADIUS Shaft & Hsg. inch	APPROX. WEIGHT lb.	S_L LIMITING SPEED ‡ rpm	C DYNAMIC LOAD RATING lb.	C_o STATIC LOAD RATING lb.
6300	10	.3937	35	1.3780	11	.4331	.125	1.562	.044	.025	.13	22000	1400	850
6301	12	.4724	37	1.4567	12	.4724	.125	1.625	.044	.040	.15	20000	1700	1040
6302	15	.5906	42	1.6535	13	.5118	.125	1.821	.044	.040	.20	18000	1930	1200
6303	17	.6693	47	1.8504	14	.5512	.141	2.074	.044	.040	.25	16000	2320	1460
6304	20	.7874	52	2.0472	15	.5906	.141	2.276	.044	.040	.34	14000	3000	1930
6305	25	.9843	62	2.4409	17	.6693	.195	2.665	.067	.040	.58	11000	3800	2550
6306	30	1.1811	72	2.8346	19	.7480	.195	3.091	.067	.040	.83	9500	5000	3400
6307	35	1.3780	80	3.1496	21	.8268	.195	3.406	.067	.060	1.07	8500	5700	4000
6308	40	1.5748	90	3.5433	23	.9055	.226	3.799	.097	.060	1.41	7500	7350	5300
6309	45	1.7717	100	3.9370	25	.9843	.226	4.193	.097	.060	1.95	6700	9150	6700
6310	50	1.9685	110	4.3307	27	1.0630	.226	4.587	.097	.080	2.50	6000	10600	8150
6311	55	2.1654	120	4.7244	29	1.1417	.271	5.104	.111	.080	3.30	5300	12900	10000
6312	60	2.3622	130	5.1181	31	1.2205	.271	5.498	.111	.080	3.81	5000	14000	10800
6313	65	2.5591	140	5.5118	33	1.2992	.304	5.892	.111	.080	4.64	4500	16000	12500
6314	70	2.7559	150	5.9055	35	1.3780	.304	6.286	.111	.080	5.68	4300	18000	14000
6315	75	2.9528	160	6.2992	37	1.4567	.304	6.679	.111	.080	6.60	4000	19300	16300
6316	80	3.1496	170	6.6929	39	1.5354	.346	7.198	.122	.080	9.53	3800	21200	18000
6317	85	3.3465	180	7.0866	41	1.6142	.346	7.593	.122	.100	11.00	3400	21600	18600
6318	90	3.5433	190	7.4803	43	1.6929	.346	7.986	.122	.100	11.60	3400	23200	20000
6319	95	3.7402	200	7.8740	45	1.7717	.346	8.380	.122	.100	13.38	3200	24500	22400
6320	100	3.9370	215	8.4646	47	1.8504	—	—	—	.100	16.34	3000	28500	27000
6321	105	4.1338	225	8.8582	49	1.9291	—	—	—	.100	17.8	2800	30500	30000
6322	110	4.3307	240	9.4488	50	1.9685	—	—	—	.100	21.0	2600	32500	32500
6324	120	4.7244	260	10.2362	55	2.1654	—	—	—	.100	32.3	2400	36000	38000
6326	130	5.1181	280	11.0236	58	2.2835	—	—	—	.12	40.1	2200	39000	43000
6328	140	5.5118	300	11.8110	62	2.4409	—	—	—	.12	48.1	2000	44000	50000
6330	150	5.9055	320	12.5984	65	2.5590	—	—	—	.12	57.8	1900	49000	60000

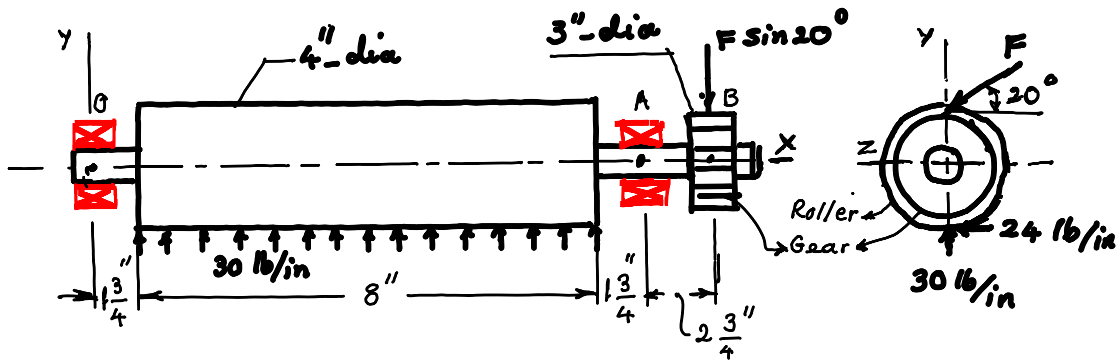
*Bearing numbers listed are for open bearings only. For shields, seals and snap rings, add suffix or prefix indicated below bearing diagram. Eg. 6300.Z, 6300.RS, 6300.NR, etc. Check availability of closures for larger sizes.
†Snap ring bearings available with shields or seals. Add both suffixes. Eg. 6300.ZNR, etc.
‡For grease lubricated bearings without seals. For other conditions, see Page 114.
For mounting data, shaft and housing fits and shoulder diameters, see Pages 124-132.

Figure 11-23

Dimensions and Load Ratings for 6300 Series, Medium, Metric, Deep-Groove (Conrad-type) Ball Bearings *Courtesy of FAG Bearings Corporation, Stamford, Conn.*

Example 1. Selection of Ball Bearings for Radial Load

Example 1: Shown in the figure is a gear-driven squeeze roll which mates with an idler roll, not shown. The roll is designed to exert a normal force of 30 lb/in of roll length and a pull of 24 lb/in on the material being processed. The roll speed is 300 rev/min, and an L_{10} life of 30 kh is desired. Select suitable deep-groove (Conrad-type) ball bearings to be mounted at O and A. Use the same-size bearings.

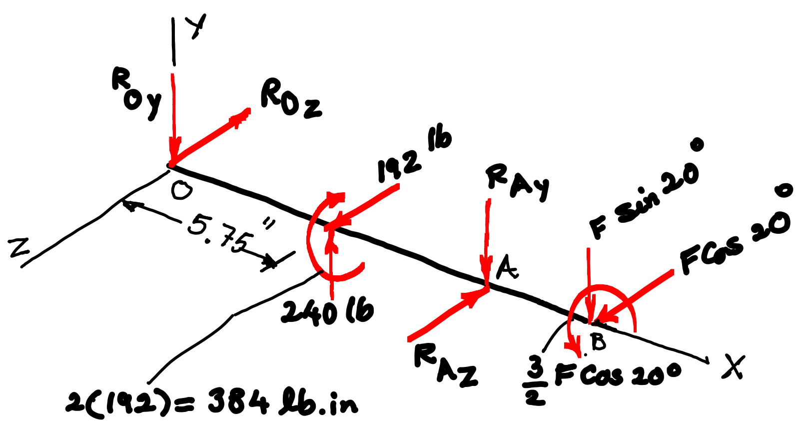


Solution

1. Determine the loads applied to the bearings

Normal force applied by the squeeze roll $= (30\ \text{lb/in})(8\ \text{in}) = 240\ \text{lb}$

Pull along the z-axis applied by the squeeze roll $= (24\ \text{lb/in})(8) = 192\ \text{lb}$



$\Sigma F_x = 0$: satisfied - no axial force

$\Sigma F_y = 0$: $-R_{Oy} + 240 - R_{Ay} - F\sin 20^\circ = 0$

$\Sigma F_z = 0$: $R_{Oz} - 192 + R_{Az} - F\cos 20^\circ = 0$

$\Sigma M_x = 0$: $-384 + \frac{3}{2} F\cos 20^\circ = 0$

$\Sigma M_y = 0$: $-192(5.75) + R_{Az}(8+2\times 1.75) - F\cos 20^\circ (8+2\times 1.75+2.75) = 0$

$\Sigma M_z = 0$: $240(5.75) - R_{Ay}(8+2\times 1.75) - F\sin 20^\circ (8+2\times 1.75+2.75) = 0$

Solve the above 5 equations for the 5 unknowns, R_{Oy}, R_{Oz}, R_{Ay}, R_{Az}, and F. We find

$R_{Oy} = 142$ lb , $R_{Oz} = 35$ lb ,

$R_{Ay} = 5$ lb , $R_{Az} = 413$ lb , F= 272 lb

Radial Force on Bearing A : $R_A = \sqrt{(413)^2 + (5)^2} = 413$ lb

$\therefore$ Radial Force on Bearing O : $R_O = \sqrt{(142)^2 + (35)^2} = 146$ lb

We select same size bearings at A and O based on the 413 lb.

2. Bearing L_{10} Design Life

3. Calculate the minimum basic load rating required for the desired L_{10} life using Eq. 11.20c :

$$L_p = B_R \left(\frac{C}{P}\right)^3$$

Choose Bearing No.

Bore=

O.D.=

Width=

Weight=

static load rating

Combined Radial and Thrust Loads

If both radial and thrust loads are applied to a bearing, an equivalent load P must be calculated for use in Eq. 11.20. The AFBMA recommends:

$$P = XVF_r + YF_a \qquad (11.22a)$$

where

P= Equivalent load for use in Eq. 11.20.
F_r = Applied constant radial load
F_a = Applied constant axial load
V = A rotation factor
X = A radial factor
Y = A thrust factor

Values of V, X, and Y are defined by bearing manufacturers in tables and figures such as Figure 11-24. Bearing types such as cylindrical rollers, that cannot support thrust loads are not included in this table.

The axial force can be neglected if the ratio F_a / VF_r is less than a factor "e" which is also specified in Table 11-24. Hence,

$$\text{if} \quad \frac{F_a}{VF_r} \leq e \quad \text{then } X = 1 \text{ and } Y = 0 \qquad (11.22b)$$

Factors V, X, and Y for Radial Bearings

Bearing Type			In Relation to the Load the Inner Ring is: Rotating	Stationary	Single Row Bearings 1) $\frac{F_a}{VF_r} > e$		Double Row Bearings 2) $\frac{F_a}{VF_r} \leqq e$		$\frac{F_a}{VF_r} > e$		e
			V	V	X	Y	X	Y	X	Y	
3)	4) $\frac{F_a}{C_0}$	5) $\frac{F_a}{i Z D_w^2}$									
Radial Contact Groove Ball Bearings	0.014	25				2.30				2.30	0.19
	0.028	50				1.99				1.99	0.22
	0.056	100				1.71				1.71	0.26
	0.084	150				1.55				1.55	0.28
	0.11	200	1	1.2	0.56	1.45	1	0	0.56	1.45	0.30
	0.17	300				1.31				1.31	0.34
	0.28	500				1.15				1.15	0.38
	0.42	750				1.04				1.04	0.42
	0.56	1000				1.00				1.00	0.44
20°					0.43	1.00		1.09	0.70	1.63	0.57
25°					0.41	0.87		0.92	0.67	1.44	0.68
30°			1	1.2	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	0.93	1.14
Self-Aligning Ball Bearings			1	1	0.40	0.4 cot α	1	0.42 cot α	0.65	0.65 cot α	1.5 tan α
Self-Aligning and Tapered Roller Bearings			1	1.2	0.40	0.4 cot α	1	0.45 cot α	0.67	0.67 cot α	1.5 tan α

1) For single row bearings, when $\frac{F_a}{VF_r} \leqq e$ use $X = 1$ and $Y = 0$.

For two single row angular contact ball or roller bearings mounted "face-to-face" or "back-to-back" the values of X and Y which apply to double row bearings. For two or more single row bearings mounted "in tandem" use the values of X and Y which apply to single row bearings.

2) Double row bearings are presumed to be symmetrical.

3) Permissible maximum value of $\frac{F_a}{C_0}$ depends on the bearing design.

4) C_0 is the basic static load rating.

5) Units are pounds and inches.

Values of X, Y and e for a load or contact angle other than shown in the table are obtained by linear interpolation.

Figure 11-24

V, X, and *Y* Factors for Radial Bearings *Courtesy of SKF USA Inc.*

Example 2. Selection of Ball Bearings for Combined Radial and Thrust Loads

Select a ball bearing for an industrial machine intended for operation at 1800 rpm under radial and thrust loads of 268 lb (1200 N) and 337 lb (1500 N), respectively. Find a suitable size, Conrad-type, deep-groove bearing to give an L_{10} life of 110 million revolutions. The inner ring of bearing rotates.