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Rated bearing life according to its reliability

Đánh giá tuổi thọ vòng bi theo độ tin cậy của nó

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ABSTRACT

Currently, the bearing calculations are based on two criteria: low speed n <10 rpm (or stationary) bearings are calculated according to the static load capacity to avoid residual deformation of the working surface; roller bearings working at high or relatively high speeds n ≥ 10 rpm calculated according to dynamic load capacity, to avoid fatigue peeling.

The paper presents how to choose a bearing based on dynamic load, including many influencing factors such as: reliability, axial force due to contact angle, impact of shock force.

Keywords: bearings; balls; rollers; radial load.

TÓM TẮT

Hiện nay các tính toán ổ lăn dựa theo hai tiêu chí: các ổ lăn làm việc với vận tốc thấp n < 10vg/ph (hoặc đứng yên) được tính theo khả năng tải tĩnh để tránh biến dạng dư bề mặt làm việc; các ổ lăn làm việc với vận tốc cao hoặc tương đối cao n ≥ 10 vg/ph được tính theo khả năng tải trọng động, để tránh tróc vì mỏi.

Bài báo trình bày cách chọn ổ trục dựa trên tải trọng động, bao gồm nhiều yếu tố ảnh hưởng như: độ tin cậy, lực dọc trục do góc tiếp xúc, tác động của lực xung kích.

Từ khóa: ổ lăn; ổ bi; ổ đũa; tải trọng hướng tâm.

1. INTRODUCTION

A bearing allows relative motion of joined elements. One of the elements can be fixed. Bearings can act as support for shaft and withstand radial and/or axial loads. The rolling elements, balls, rollers, or needles divide the bearing rings. The rolling bearings operate at lower starting friction with a coefficient of friction of μ = 0.001÷0.003. Other advantages are easy lubrication and replacement in case of failure. Some disadvantages of rolling bearings are collapse to large loads, higher costs, and noise.

Methods of calculating bearings according to different criteria are currently not available because these standards involve many random factors which are difficult to determine. Because of this, the bearings were standardized and mass produced so the calculation and testing determined the load capacity of each bearing type, type and size. When designing the machine, it is not necessary to design the bearing, but only need to calculate and select the standard roller according to the conventional formula. The rolling bearing selection method is also standardized.

The paper presents with an overview of bearing types; then we note that bearing life cannot be described in deterministic form. We introduce the invariant, the statistical distribution of life, which is strongly Weibullian.1 There are some useful deterministic equations addressing load versus life at constant reliability, and we introduce the catalog rating at rating life.

The reliability-life relationship involves Weibullian statistics. The load-life-reliability relationship, combines statistical and deterministic relationships giving the designer a way to move from the desired load and life to the catalog rating in one equation.

Ball bearings also resist thrust, and a unit of thrust does different damage per revolution than a unit of radial load, so we must find the equivalent pure radial load that does the same damage as the existing radial and thrust loads. Next, variable loading, stepwise and continuous, is approached, and the equivalent pure radial load doing the same damage is quantified. Oscillatory loading is mentioned. Having the tools to find the proper catalog ratings, we make decisions (selections), we perform a design assessment, and the bearing reliability is quantified.

The commonly used term is bearing life, which is applied to either of the measures just mentioned. It is important to realize, as in all fatigue, life as defined above is a stochastic variable and, as such, has both a distribution and associated statistical parameters. The life measure of an individual bearing is defined as the total number of revolutions (or hours at a constant speed) of bearing operation until the failure criterion is developed. Under ideal conditions, the fatigue failure consists of spalling of the loadcarrying surfaces. The American Bearing Manufacturers Association (ABMA) standard states that the failure criterion is the first evidence of fatigue.

The rating life is a term sanctioned by the ABMA and used by most manufacturers. The rating life of a group of nominally identical ball or roller bearings is defined as the number of revolutions (or hours at a constant speed) that 90 percent of a group of bearings will achieve or exceed before the failure criterion develops. The terms minimum life, L_{10} life, and B_{10} life are also used as synonyms for rating life. The rating life is the 10th percentile location of the bearing group's revolutions-to-failure distribution.

2. FORCE ANALYSIS

An angular contact or radial thrust ball bearing with the contact angle α is depicted in Fig. 1. The contact angle for a radial ball bearing is null [2, 3, 4].

A transverse load, F_r, is acting perpendicular to the shaft axis of the angular contact bearing. The force that acts on the ball is

$$F = \frac{F_r}{cos\alpha} \quad (1)$$

The thrust or axial load is

$$F_a = F_t = F sin\alpha \quad (2)$$

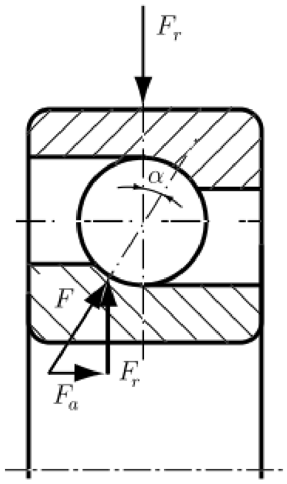

Figure 1. Radial thrust (angular contact) ball bearing

3. BEARING LOAD LIFE AT RATED RELIABILITY

When nominally identical groups are tested to the life-failure criterion at different loads, the data are plotted on a graph as depicted in Fig.2 using a log-log transformation. To establish a single point, load F_1 and the rating life of group one $(L_{10})_1$ are the coordinates that are logarithmically transformed. The reliability associated with this point, and all other points, is 0.90. Thus we gain a glimpse of the load-life function at 0.90 reliability. Using a regression equation of the form

$$FL^{1/a} = constant \quad (3)$$

the result of many tests for various kinds of bearings result in

- a = 3 for ball bearings
- a = 10/3 for roller bearings (cylindrical and tapered roller)

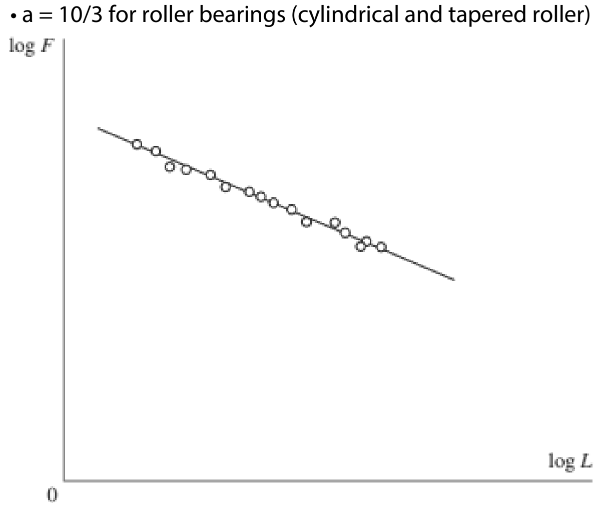

Figure 2. Typical bearing load-life log-log curve.

A catalog load rating is defined as the radial load that causes 10 percent of a group of bearings to fail at the bearing manufacturer's rating life. We shall denote the catalog load rating as C_{10}. The catalog load rating is often referred to as a Basic Dynamic Load Rating, or sometimes just Basic Load Rating, if the manufacturer's rating life is 10^6 revolutions. The radial load that would be necessary to cause failure at such a low life would be unrealistically high. Consequently, the Basic Load Rating should be viewed as a reference value, and not as an actual load to be achieved by a bearing.

The geometry of the bearings and the standard forces are found in catalogues where the bearings are tabulated. Next a procedure to select the rolling bearing is presented [1].

The life of a rolling bearing represents the number of revolutions or the period of time at a constant angular velocity prior to the initial sign of material failure. [1] shows the rated capacity, C, corresponding to a standard life of $L_R = 9.10^7$ revolutions, and a constant radial load.

Bearing application required life is calculated with

$$L = L_R \left(\frac{C}{F_r}\right)^{10/3} \quad (4)$$

where the radial load for the application is F_r. The required rated capacity for the application is

$$C_{req} = F_r \left(\frac{L}{L_R}\right)^{3/10} \quad (5)$$

The standard life denoted by L_{10} or B_{10} is the life of the bearing that corresponds to a reliability of r = 90% or 10% failure. The median life is five times the standard life. Eqs. (3) and (4) for ball and roller bearings are improved with a reliability factor K_r [1]

$$L = K_r L_R (C/F_r)^{3.33} \quad (6)$$

$$C_{req} = F_r \left(\frac{L}{K_r L_R}\right)^{0.3} \quad (7)$$

where

$$K_r = \begin{cases} 1.00 \text{ for 90\% reliability, L10,} \\ 0.62 \text{ for 95\% reliability, L5,} \\ 0.53 \text{ for 96\% reliability, L4,} \\ 0.44 \text{ for 97\% reliability, L3,} \\ 0.33 \text{ for 98\% reliability, L2,} \\ 0.21 \text{ for 99\% reliability, L1.} \end{cases} \quad (8)$$

The influence of the axial force is adjusted with an radial equivalent force, F_e [1]

$$L = K_r L_R (C/F_e)^{3.33} \quad (9)$$

$$C_{req} = F_e \left(\frac{L}{K_r L_R}\right)^{0.3} \quad (10)$$

The radial equivalent load for ball bearings with the contact angle α = 0⁰ is

$$F_e = \begin{cases} F_r & \text{for } 0.00 < F_a/F_r < 0.35\,, \\ F_r \left[1 + 1.115\left(\frac{F_a}{F_r} - 0.35\right)\right] & \text{for } 0.35 < F_a/F_r < 10.0\,, \\ 1.176F_a & \text{for } F_a/F_r > 10.0 \end{cases} \quad (11)$$

For angular ball bearings with the contact angle $\alpha = 25^0$, the radial equivalent load is

$$F_e = \begin{cases} F_r & \text{for } 0.00 < F_a/F_r < 0.68\,, \\ F_r \left[1 + 0.87\left(\frac{F_a}{F_r} - 0.68\right)\right] & \text{for } 0.68 < F_a/F_r < 10.0\,, \\ 0.911F_a & \text{for } F_a/F_r > 10.0 \end{cases} \quad (12)$$

The influence of a shock force is taken into consideration by an application factor K_a [1, 2]:

$$L = K_r L_R \left(\frac{C}{K_a F_e}\right)^{3.33} \quad (13)$$

$$C_{req} = K_a F_e \left(\frac{L}{K_r L_R}\right)^{0.3} \quad (14)$$

The application factor for a ball bearing is

$$K_a = \begin{cases} 1 & \text{for uniform force, no impact,} \\ 1.0 \div 1.1 & \text{for precision gearing,} \\ 1.1 \div 1.3 & \text{for commercial gearing,} \\ 1.2 & \text{for poor bearing seals,} \\ 1.2 \div 1.5 & \text{for light impact,} \\ 1.5 \div 2.0 & \text{for moderate impact,} \\ 2.0 \div 3.0 & \text{for heavy impact.} \end{cases} \quad (15)$$

The application factor for a roller bearing is

$$K_a = \begin{cases} 1 & \text{for uniform force, no impact,} \\ 1 & \text{for gearing,} \\ 1.0 \div 1.1 & \text{for light impact,} \\ 1.1 \div 1.5 & \text{for moderate impact,} \\ 1.5 \div 2.0 & \text{for heavy impact.} \end{cases} \quad (16)$$

The following suggestions for the life of the bearing, are given table 1 [1, 2]:

Table 1. Bearing-Life Recommendations for Various Classes of Machinery

Type of Application	Life, 10^3 h
Instruments and apparatus for infrequent use	Up to 0.5
Aircraft engines	0.5 ÷ 2
Machines for short or intermittent operation where service interruption is of minor importance	4 ÷ 8
Machines for intermittent service where reliable operation is of great importance	8 ÷ 14
Machines for 8-h service that are not always fully utilized	14 ÷ 20
Machines for 8-h service that are fully utilized	20 ÷ 30
Machines for continuous 24-h service	50 ÷ 60
Machines for continuous 24-h service where reliability is of extreme importance	100 ÷ 200

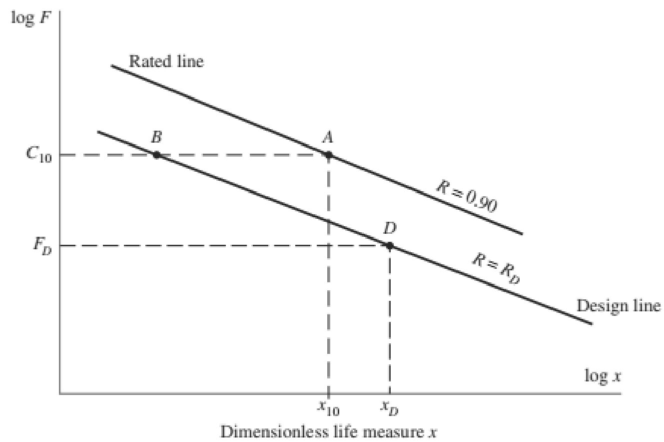

Figure 3. Constant reliability contours.

To assist the designer in the selection of bearings, most of the manufacturers' handbooks contain data on bearing life for many classes of machinery, as well as information on load-application factors. Such information has been accumulated the hard way, that is, by experience, and the beginner designer should utilize this information until he or she gains enough experience to know when deviations are possible. Table 1 contains recommendations on bearing life for some classes of machinery.

Relating Load, Life, and Reliability

This is the designer's problem. The desired load is not the manufacturer's test load or catalog entry. The desired speed is different from the vendor's test speed, and the reliability expectation is typically much higher than the 0.90 accompanying the catalog entry. Figure.3 shows the situation. The catalog information is plotted as point A, whose coordinates are (the logs of) C_{10} and $x_{10} = L_{10}/L_{10} = 1$, a point on the 0.90 reliability contour. The design point is at D, with the coordinates (the logs of) F_D and x_D, a point that is on the $R = R_D$ reliability contour. The designer must move from point D to point A via point B as follows. Along a constant reliability contour (B D).

Point A represents the catalog rating C_{10} at $x = L/L_{10} = 1$. Point B is on the target reliability design line R_D, with a load of C_{10}. Point D is a point on the desired reliability contour exhibiting the design life $x_D = L_D/L_{10}$ at the design load F_D.

4. EXAMPLES

Fig.4 depicts a countershaft with two rigidly connected gears 1 and 2. The angular speed of the countershaft is 200 rpm. The force on the countershaft gear 1 at *P* is $\mathbf{F}_P = F_{Py}\,\mathbf{j} + F_{Pz}\,\mathbf{k} = 1500\mathbf{j} + 1500\tan(20^0)\mathbf{k}$ N, and the force on the gear 2 at *R* is $\mathbf{F}_R = F_{Ry}\,\mathbf{j} + F_{Rz}\,\mathbf{k} = -2000\tan(20^0)\,\mathbf{j} - 2000\mathbf{k}$ N. The pitch radius of gear 1 is $OP = R_1 = 0.20$ m and the pitch radius of gear 2 is $C_R = R_2 = 0.15$ m. The distance between the bearings is $s = 0.20$ m and the distance between the gear and bearing is $l = 0.15$ m. The gear system is a part of an industrial machine intended for 8-hour service, but not every day. Select identical 300 series ball bearings for *A* and *B*.

Solution

The forces and their position vectors are given by:

```
FPx=0 (N);
FPy= 1500(N);
FPz= FPy*tan(20*pi/180);
FP_= [FPx, FPy, FPz];
external force FP_ = [0  1500.000  545.955] (N)
FRx = 0 (N);
FRz = -2000 (N);
FRy = FRz*tan(20*pi/180) (N);
FR_= [FRx, FRy, FRz];
external force FR_ = [0   -727.940   -2000.000] (N);
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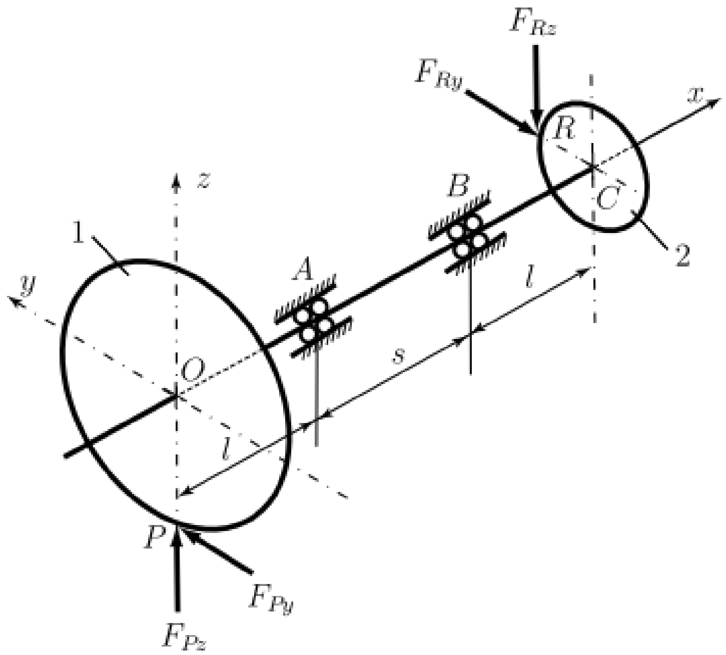

Figure 4. Force diagrams acting on the bearing

The unknowns are the bearing loads at *A* and *B*:

syms FAy FAz FBy FBz

FA_ = [0, FAy, FAz];

FB_ = [0, FBy, FBz];

The bearing reaction force at *B* is calculated using the sum of the moments about *A*:

sum MA_ = rAP_ x FP_ + rAB_ x FB_ + rAR_ x FR_ = 0_ =>

MA_ = cross(rP_-rA_, FP_) + cross(rB_-rA_,FB_) + cross(rR_-rA_,FR_);

FB_ = [0, FBys, FBzs];

FBr = sqrt(FB_(2)^2+FB_(3)^2);

radial force FBr = 4586.789 (N)

The bearing reaction force at *A* is calculated using the sum of the forces for the system:

sum F_ = FA_+FB_+FP_+FR_= 0_ =>

FA_ = -FP_-FB_-FR_;

FAr = sqrt(FA_(2)^2+FA_(3)^2);

FA_ = [-0.000 -3170.955 -2455.422] (N)

radial force FAr = 4010.493 (N)

The radial force at bearing *B* is higher than that at bearing *A*, and the selection of the bearing is based on this force FBr = 4586.789 (N):

if FAr > FBr

Fe = FAr;

else

Fe = FBr;

end

radial equivalent force Fe = FBr = 4586.789 (N)

From table 1 the design bearing life for machines for 8-hour service, but not every day, is selected (conservatively) *DL* = 20000 h. The life *L* corresponding to the 200 rpm rotation of the shaft is:

DL = 20000 (h) : design life

n = 200 (rpm)

L = n * DL * 60;

life L = 2.400e+08 (rev)

The life adjustment is represented by the reliability factor *Kr*, given by Eq. (8), *Kr* = 0.33 for 98% reliability. The application factor for commercial gearing is selected (conservatively) from Eq. (15), *Ka* = 1.3. The life corresponding to rated capacity is *LR* = $90(10)^6$. The required value of rated capacity for this application is given by:

Ka = 1.3; for commercial gearing

Kr = 0.33;

LR = 90 * 10^6;

Eq. (14): Creq = Ka*Fe*(L/(Kr*LR))^0.3*10^-3;

rated capacity: Creq = 11.161 (kN)

From [1] with 11.16 kN for 300 series we select C = 12.6 kN and d = 40 mm bore. From [1] with 40 mm bore and 300 series the bearing number is 308.

A bearing number 308 is selected for both bearings *A* and *B*.

5. CONCLUDE

The paper is intended to complement courses on the calculation of bearing selection methods in machine components and is a reference for mechanical engineers. The article introduces the American method of calculating the selection of a roller compared to the current method of choosing a roller bearing in Vietnam.

The reliability-life relationship involves Weibullian statistics. The load-life-reliability relationship, combines statistical and deterministic relationships giving the designer a way to move from the desired load and life to the catalog rating in one equation.

REFERENCES

[1] R.C. Juvinall, K.M. Marshek, Fundamentals of Machine Component Design, 5th edition, John Wiley & Sons, New York, 2010.

[2] R. Budynas, K.J. Nisbett, Shigley's Mechanical Engineering Design, 9th edition, McGraw-Hill, New York, 2013.

[3] Nguyễn Trọng Hiệp (2003). *Chi tiết máy, tập 2,* Nhà xuất bản giáo dục.

[4] Trịnh Chất (2005). Cơ sở thiết kế máy & *Chi tiết máy,* Nhà xuất bản Khoa học và kỹ thuật Hà Nội.