4 Load Rating and Life

Even in bearings operating under normal conditions, the surfaces of the raceway and rolling elements are constantly being subjected to repeated compressive stresses which causes flaking of these surfaces to occur. This flaking is due to material fatigue and will eventually cause the bearings to fail. The effective life of a bearing is usually defined in terms of the total number of revolutions a bearing can undergo before flaking of either the raceway surface or the rolling element surfaces occurs.

Other causes of bearing failure are often attributed to problems such as seizing, abrasions, cracking, chipping, gnawing, rust, etc. However, these so called "causes" of bearing failure are usually themselves caused by improper installation, insufficient or improper lubrication, faulty sealing or inaccurate bearing selection. Since the above mentioned "causes" of bearing failure can be avoided by taking the proper precautions, and are not simply caused by material fatigue, they are considered separately from the flaking aspect.

Usually, the load exerted on the main spindle of a machine tool is relatively small compared to the dynamic rated load on the bearing. Therefore, the fatigue life of a bearing seldom poses a problem. The following operating conditions, rather than a bearing's rating life, can significantly affect the bearing functions (running accuracy, rigidity, heat generation, etc.) and require special consideration.

(1) High speed operation.

(2) Heavy preload.

(3) Large bending of the shaft.

(4) Large temperature difference between the inner and outer rings.

4.1 Basic rating life and basic dynamic load rating

A group of seemingly identical bearings when subjected to identical load and operating conditions will exhibit a wide diversity in their durability. This "life" disparity can be accounted for by the difference in the fatigue of the bearing material itself. This disparity is considered statistically when calculating bearing life, and the basic rating life is defined as follows. The basic rating life is based on a 90% statistical model which is expressed as the total number of revolutions 90% of the bearings in an identical group of bearings subjected to identical operating conditions will attain or surpass before flaking due to material fatigue occurs. For bearings operating at fixed constant speeds, the basic rating life (90% reliability) is expressed in the total number of hours of operation.

The basic dynamic load rating is an expression of the load capacity of a bearing based on a constant load which the bearing can sustain for one million revolutions (the basic life rating). For radial bearings this rating applies to pure radial loads, and for thrust bearings it refers to pure axial loads. The basic dynamic load ratings given in the bearing tables of this catalog are for bearings constructed of TPI standard bearing materials, using standard manufacturing techniques. Please consult TPI for basic load ratings of bearings constructed of special materials or using special manufacturing techniques.

The relationship between the basic rating life, the basic dynamic load rating and the bearing load is given in formula:

$$L_{10} = \left(\frac{C_r}{P}\right)^p$$

$$L_{10h} = \frac{10^6}{60n}\left(\frac{C_r}{P}\right)^p$$

where，

p=3··············for ball bearings

L_{10} : Basic rated life 10^6 revolutions

L_{10h} : Basic rated life, hour

C_r : Basic dynamic rated load, N or kgf

P : Equivalent dynamic load, N or kgf

n : Rotational speed, rpm

When several bearings are incorporated in machines or equipment as complete units, all the bearings in the unit are considered as a whole when computing bearing life

$$L = \frac{1}{\left(\frac{1}{L_1^{e}} + \frac{1}{L_2^{e}} + \cdots + \frac{1}{L_n^{e}}\right)^{1/e}}$$

where,

L : Total basic rating life of entire unit, h

L_1 , L_2 …L_n : Basic rating life of individual bearings, 1, 2,⋯n, h

e = 10/9..........For ball bearings

When the load conditions vary at regular intervals, the life can be given by formula.

$$L_m = \left(\frac{\Phi_1}{L_1} + \frac{\Phi_1}{L_1} + \cdots \frac{\Phi_j}{L_j}\right)^{-1}$$

where,

Lm : Total life of bearing

Φ_j : Frequency of individual load conditions

($\Sigma \Phi_j$ = 1)

L_j : Life under individual conditions

4.2 Adjusted rating life

The basic bearing rating life (90% reliability factor) can be calculated by the formula mentioned. However, in some applications a bearing life factor of over 90% reliability may be required. To meet these requirements, bearing life can be lengthened by the use of specially improved bearing materials or manufacturing process. Bearing life is also sometimes affected by operating conditions such as lubrication, temperature and rotational speed.

Basic rating life adjusted to compensate for this is called "adjusted rating life," and is determined by using the formula according to ISO 281.

$$L_{na} = a_1 a_2 a_3 \left(\frac{C}{P} \right)^p$$

where,

L_{na} : Adjusted rating life in millions of revolutions(10^6)

a_1 : Reliability factor

a_2 : Bearing characteristics factor

a_3 : Operating conditions factor

4.3 New bearing life formula

According to the conventional Lundberg-Palmgren (L-P) theory, a stress that governs rolling fatigue is considered, that is, a maximum dynamic shear stress τo that is exerted, at a depth of Zo from the rolling contact surface, in a plane parallel with the rolling contact surface. The probability of survival S of a volume V illustrated in Fig. 4.1 that is subjected to N times of stress application is determined by the formula below according to the Weibull theory.

$$\ln \frac{1}{S} \propto \frac{N^e \tau_0^c}{z_0^h} V$$

For many applications, prediction of bearing fatigue life using the Lundberg-Palmgren theory life modified by serial multiplication of life adjustment factors, is substantially inaccurate. This is especially true for bearings in appplications involving relatively light loads; e.g., applications in which the maximum Hertz stress is less than 1400 MPa (approximately 200 kpsi).

The Ioannides-Harris (I-H) theory, which is an extension of the Lundberg-Palmgren theory, introduced the concept of a fatigue limit stress. Under the same Weibull weakest link theory, the probability of survival ΔV_i of a volume ΔS_i sufficiently large to contain many defects, can be generally expressed as:

$$\ln \frac{1}{\Delta S_i} \propto \frac{N^e (\sigma_{VM,i} - \sigma_{VM,\lim})^c \Delta V_i}{z_i^h}$$

A major feature of the Ioannides-Harris theory is that it compares the total stress at any point in the rolling component to the fatigue limit. The total stress at each point is calculated using the Von Mises stress. The life of each ball-raceway contact is determined by numerical integration of the field of Von Mises stresses. Bearing fatigue life is determined by statistical combination of the rolling component lives. If this value is equal to or less than 0, then the elemental volume will not experience fatigue. Fig. 4.2 illustrates schematically the load-life relationship between L-P and I-H theories. Fig. 4.3 shows the stress volume resulting from rolling contact according to I-H theory.

Fig. 4.1 Stress volume resulting from rolling contact according to L-P theory

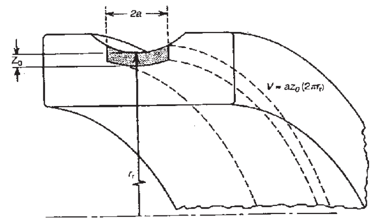


Fig. 4.2 Load-life comparison between L-P and I-H theories

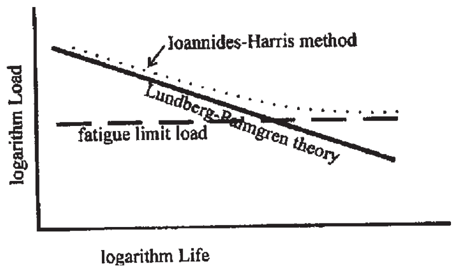
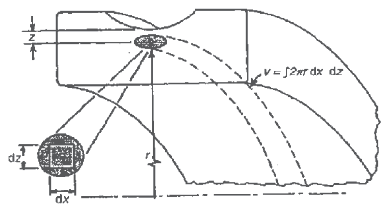


Fig. 4.3 Stress volume resulting from rolling contact according to I-H theory



Lundberg and Palmgren state that the volume under stress is proportional to the volume of the cylindrical ring. This proportionality is only valid when simple Hertz contact is applied to a smooth surface. The L-P theory also does not account for the effect of temperature on lubrication and hence on shear stresses. A number of shortcomings of the L-P theory became apparent in the decades.

The stress-life method based on I-H theory for prediction of bearing fatigue life considers the integrated effect of all stresses acting on the rolling component surfaces together with those stresses within the component. The latter stresses may be caused by ring rotation, mounting of the inner ring on the shaft or the outer ring in the housing, heat treatment of the component, or surface forming and finishing processes. In addition to the Hertz stresses, which tend to be the stresses of greatest influence on bearing fatigue life, stresses are caused by shearing of the lubricant in the contacts and friction between asperities on mating surfaces when lubricant film thickness is insufficient to completely separate the surfaces. Moreover, dents in the rolling surfaces caused by hard particle contamination result in stress concentrations which augment both the Hertz and surface shear stresses. This was accomplished using the analytical TH-BBAN computer program developed by the concept of a stress-life to fulfill the requirement for the interdependency of the various fatigue life-influencing factors. TH-BBAN calculates fatigue life of each ball-raceway contact is accomplished by evaluation of a life integral according to the total or actual stress condition compared to the life integral corresponding to the simple Hertz stress application shown as follows:

$$L_{nM} = A_1 A_{SL}\left(\frac{C}{P}\right)^p$$

where

$$A_{SL} = \frac{L_{actual}}{L_{LP}} = \frac{u\left\{\int_V \frac{(\sigma_{VM,i} - \sigma_{VM,\lim})^c}{z^h} dV\right\}^{1/e}_{actual}}{u\left\{\int_V \frac{(\sigma_{VM,i})^c_{LP}}{z^h} dV\right\}^{1/e}_{LP}}$$

A_{SL} is called the contact stress-life factor in TH-BBAN. As an alternative to the life formula, ISO 281:2007 established the bearing life equation format as follows:

$$L_{nM} = A_1 A_{ISO} L_{10}$$

L_{nM} : the basic rating life modified for a reliability (100-n)%

A_1 : the reliability-life factor

A_{ISO} : the integrated life factor, including material, lubrication and contamination effects(ISO 4406 cleanliness code adopted)

4.4 Static load rating and allowable axial load

When stationary rolling bearings are subjected to static loads, they suffer from partial permanent deformation of the contact surfaces at the contact point between the rolling elements and the raceway. The amount of deformity increases as the load increases, and if this increase in load exceeds certain limits, the subsequent smooth operation of the bearings is impaired.

It has been found through experience that a permanent deformity of 0.0001 times the diameter of the rolling element, occurring at the most heavily stressed contact point between the raceway and the rolling elements, can be tolerated without any impairment in running efficiency.

The basic static load rating refers to a fixed static load limit at which a specified amount of permanent deformation occurs. It applies to pure radial loads for radial bearings and to pure axial loads for thrust bearings. For ball bearings, the maximum applied load value for contact stress occurring at the rolling element and raceway contact points is 4200Mpa or 428kgf/mm^2.

Generally the static equivalent load which can be permitted is limited by the basic static rating load as stated above. However, depending on requirements regarding friction and smooth operation, these limits may be greater or lesser than the basic static rating load.

This is generally determined by taking the safety factor So given in Table 3.3 and formula into account.

$$S_o = \frac{C_o}{P_{o\max}}$$

where,

S_0 : Safety factor

C_0 : Basic static load rating, N {kgf}

P_0 : Static equivalent load, N {kgf}

Table 4.1 Minimum safety factor values So

Operating conditions	Ball bearings	Roller bearings
High rotational accuracy necessary	2	3
Normal rotating accuracy necessary (Universal application)	1	1.5
Slight rotational accuracy deterioration permitted (Low speed, heavy loading, etc.)	0.5	1

Note: When vibration and/or shock loads are present, a load factor based on the shock load needs to be included in the Po max value.

A greater axial load can be exerted on a main spindle bearing on a machine tool allowing for tool changes while the machine is stationary. When an angular contact ball bearing is subjected to a larger axial load, the contact ellipse between its rolling elements and raceway surface can overflow the raceway surface. The maximum allowable load that does not cause such problems is defined as the "allowable axial load." The allowable axial load is reached when 1)The end of contact ellipse on the raceway surface reaches the shoulder of either an inner or outer ring. 2)The contact surface pressure on the raceway surface reaches 4200Mpa or 428kgf/mm2 in either the inner or outer ring raceway. The allowable axial load for each bearing is found in the precision bearing tables.

4.5 Bearing life for high speed application

For high speed applications, the effects of ball centrifugal forces and gyroscopic moments needs to be included. The force and moment equilibrium equations for the bearing inner ring are solved for the bearing axial, radial, and angular deflections. If the bearing has a complement of Z balls, then a system of 4Z+5 equations is solved numerically using the Newton-Raphson method.

For the analysis including the determination of ball friction forces and speeds, in addition to the 5 force and moment load equilibrium equations for the inner ring, the torques acting on the cage in the plane of bearing rotation is balanced, and cage speed is determined. In this case a system of 9Z+6 equations are solved numerically. TPI's HSE high speed type angular contact ball bearings are optimally designed with their internal configuration to accommodate both low frictional 24 heat or ball skidding effect and high rigidity by using TH-BBAN.

4.6 Life for hybrid bearings

When calculating the rating life for hybridbearings, the same life values can be used as for all-steel bearings. The ceramic balls in hybrid bearings are much harder and stiffer than the all-steel bearings. Although this increased level of hardness and stiffness creates a higher degree of contact stress between the ceramic ball and the steel raceway, extensive experience and testing shows that in typical machine tool applications, the service life of a hybrid bearing is significantly longer life than that of an all-steel bearing. The reasons for this are: 1) low density minimizes centrifugal and inertial forces; 2) low surface adhesive wear is reduced by the lower affinity to steel; and 3) better surface finish enables the bearing to maximize the effects of the lubricant.

References：

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T. A. Harris and M. H. Kotzalas, "Rolling Bearing Analysis: Advanced Concepts of Bearing technology", pp.209~258, 5th Ed., CRC Press, (2007).

5 Bearing Preload and Rigidity

5.1 Stiffness of spindle

System rigidity in machine tool applications is extremely important because the magnitude of deflection under load determines machining accuracy. Bearing stiffness is only one factor that influences system stiffness, others include shaft diameter, tool overhang, housing stiffness number, position and type of bearings. For axial stiffness of spindles, bearing stiffness plays an important role of it. Giving preload to a bearing results in the rolling element and raceway surfaces being under constant elastic compressive forces at their contact points. This has the effect of making the bearing extremely rigid so that even when load is applied to the bearing, radial or axial shaft displacement does not occur.

If high radial rigidity of bearing is needed, cylindrical roller bearings are normally used. In contrast to angular contact ball bearing, they provide more surface contact and gross sliding and are not suitable for very high speed applications. For axial loading applications, angular contact ball bearings are normally used. Their larger contact angle type provide higher axial rigidity. The stiffness of this type also depends on number and size of balls. Recently, the ceramic material silicon nitride Si_3N_4 is used for precision ball bearings. The radial rigidity of this hybrid bearings is approximately 15% higher because of the higher Young’s modulus. As mentioned in 4.5, TPI’s HSE type angular contact ball bearings are optimally designed with their internal configuration to accommodate both low ball skidding effect and high rigidity by using TH-BBAN.

5.2 Bearing preload

The preload method is divided into fixed position preload and constant pressure preload as shown in Fig. 5.1. The fixed position preload is effective for positioning the two bearings and also for increasing the rigidity. Due to the use of a spring for the constant pressure preload, the preloading amount can be kept constant, even when the distance between the two bearings fluctuates under the influence of operating heat and load.

Fig. 5.1 Preloading methods for bearings