FATIGUE LIFE ANALYSIS OF A RAILWAY BEARING USING TAGUCHI METHOD

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ABSTRACT

An axle bearing is one of the most important components to guarantee the service life of a rail car. In order to ensure the stable and reliable bearing life, it is essential to estimate the fatigue life of an axle bearing under the loading conditions. The fatigue life of a bearing is affected by many parameters such as material properties, heat treatment, lubrication conditions, operating temperature, loading conditions, bearing geometry, the internal clearance of bearing, and so on. Because these factors are so complicatedly related to each other, it is very important to investigate the effects of these factors on the axle bearing life. This paper presents the process of estimating the fatigue life of a railroad roller bearing, which takes into account geometric parameters of the bearing in the life calculation. The load distributions of the bearing were determined by solving numerically force and moment equilibrium equations with Lundberg's approximate model. This paper focuses on analyzing the effects of bearing geometric parameters on the fatigue life using Taguchi method.

Keywords: fatigue life, internal geometric parameter, railway axlebox, Taguchi method, tapered roller bearing unit.

1 INTRODUCTION

An axlebox is the linking design element between the rotating wheelset and the railway vehicle. An axlebox bearing has always been a critical component in the reliability of railway vehicles. In order to ensure the stability and reliability of the bearings, it is essential to estimate the fatigue life of an axle bearing under the loading conditions. Tapered roller bearing units are widely used in railway axleboxes due to the high-load capacity against axial loads as well as radial loads.

The fundamental theories of rolling element bearings were early established by Lundberg, Palmgren, and Harris [1]. The application of these theories was the origin of analytical methods to calculate the internal load distribution in a rolling bearing and subsequently to predict its stiffness and fatigue life. With the rapid development of computers, various investigations [2–5] to estimate the load distribution of a bearing have been carried out considering the effect of complex loading conditions as well as the internal bearing geometry. However, most studies are too complicated to use in the industrial field and have focused on the investigation of bearing characteristics under the assumption that the values of external loads acting on the bearing are known. In practice, it is hard to know accurately the bearing load values in many cases.

In this paper, the relationships between external forces, bearing loads, displacements, and load distributions of a bearing unit were investigated. The fatigue life of a tapered roller bearing unit was calculated based on Lundberg's approximate model which considers the influence of the internal geometric parameters of a bearing unit. Then, the effects of internal geometric parameters on the fatigue life of the bearing unit are investigated by using Taguchi method.

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2 FATIGUE LIFE OF A BEARING UNIT

2.1 Load-deflection relationships of a rolling bearing

Generally, when a tapered roller bearing with contact angle α is subjected simultaneously to a radial and an axial load, Fig. 1 shows the relative radial displacement δ_r and the axial displacement δ_a between bearing rings [1]. At any angular position ψ measured from the most heavily loaded rolling element, the approach of the rings can be expressed as

$$\delta_\psi = \delta_a \sin\alpha + \delta_r \cos\alpha \cos\psi. \quad (1)$$

The maximum relative deflection at $\psi = 0$ is given by

$$\delta_{\max} = \delta_a \sin\alpha + \delta_r \cos\alpha = \frac{2\varepsilon}{2\varepsilon - 1}\delta_a \sin\alpha \quad (2)$$

where ε is the load distribution factor

$$\varepsilon = \frac{1}{2}\left(1 + \frac{\delta_a \tan\alpha}{\delta_r}\right). \quad (3)$$

In condition of pure axial deflection for standard tapered roller bearings made of the bearing steel [1], the axial deflection can be approximated as

$$\delta_{a0} = \frac{0.000077}{\sin\alpha}\frac{Q_{\max}^{0.9}}{l_e^{0.8}} = \frac{\delta_{\max}}{\sin\alpha} \quad (4)$$

where $Q_{\max}$ is the maximum roller normal load (N), and l_e is the effective roller length (mm).

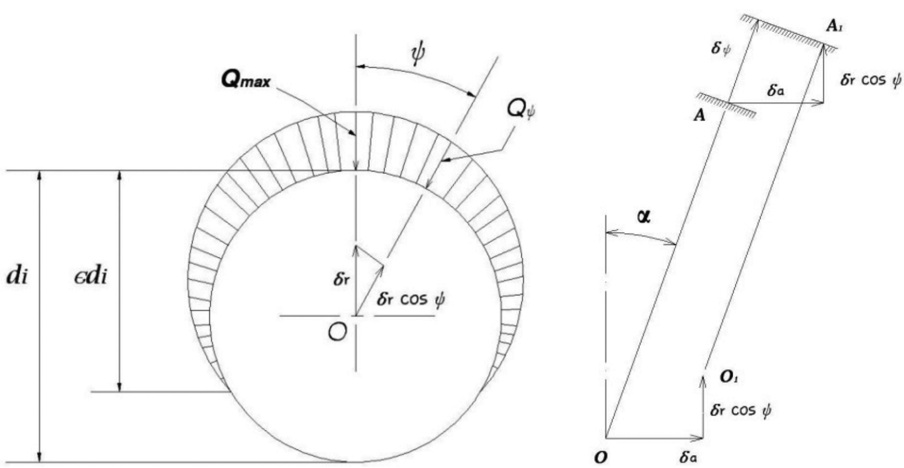

Figure 1: Rolling bearing deflection due to combined radial and axial loading.

From eqns (2) and (4), the axial deflection δ_a can be derived as

$$\delta_a = \frac{0.000077}{\sin\alpha}\frac{Q_{max}^{0.9}}{l_e^{0.8}}\frac{2\varepsilon - 1}{2\varepsilon}. \quad (5)$$

Using load integrals, applied axial bearing load can be expressed as

$$F_a = Z\sin\alpha Q_{max}\, J_a(\varepsilon) \quad (6)$$

where Z is the number of rollers. J is defined as

$$J_a(\varepsilon) = \frac{1}{2\pi}\int_{-\psi_l}^{\psi_l}\left[1 - \frac{1}{2\varepsilon}(1-\cos\psi)\right]^{1.11} d\psi. \quad (7)$$

Applied radial bearing load can be expressed as

$$F_r = Z\cos\alpha Q_{max}\, J_r(\varepsilon) \quad (8)$$

where

$$J_r(\varepsilon) = \frac{1}{2\pi}\int_{-\psi_l}^{\psi_l}\left[1 - \frac{1}{2\varepsilon}(1-\cos\psi)\right]^{1.11} \cos\psi\, d\psi \quad (9)$$

$$\psi_l = \cos^{-1}(1-2\varepsilon). \quad (10)$$

Consequently, if bearing loads F_a and F_r are given, Q_{max} and ε can be calculated numerically from eqns (6) and (8). Using them, δ_r and δ_a can be acquired from eqns (3) and (5).

2.2 Equilibrium conditions of a tapered roller bearing unit

Figure 2 shows a typical tapered roller bearing unit used in railway vehicles. From the SKF guide principles, which are widely used in railway industry [6, 7], the equivalent radial axlebox load can be calculated as

$$K_r = f_0 f_{rd} f_{tr} G \quad (11)$$

where f_0 is the payload factor, f_{rd} is the dynamic radial factor, f_{tr} is the dynamic traction factor, and G is the axlebox load.

The equivalent axial axlebox load can be calculated as

$$K_a = f_0 f_{ad} G \quad (12)$$

where f_{ad} is the dynamic axial factor.

Considering the loading conditions, the following forces and moment equilibrium equations can be derived.

$$K_r - F_{ro} - F_{ri} = 0 \quad (13)$$

$$K_a + F_{ao} - F_{ai} = 0 \quad (14)$$

$$l_d K_a + l_c F_{ro} - \frac{l_c}{2} K_r = 0 \quad (15)$$

where l_d is the axial axlebox load distance (mm) (Fig. 2), and l_c is the distance between bearing load centres (mm).

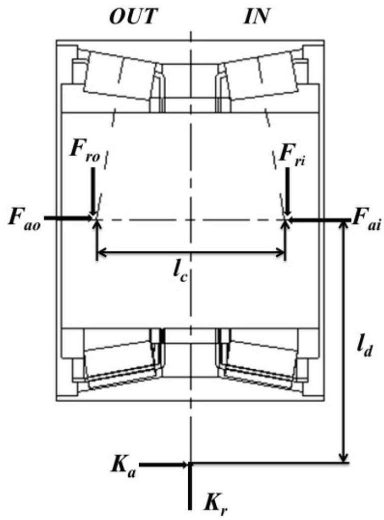

Figure 2: Axle bearing unit with applied loads.

In order to obtain the four unknown bearing loads, one more equation can be used from the geometric constraint which is assumed that total axial clearance is unchanged after loading.

$$\delta_{ao} + \delta_{ai} = \delta_0 \tag{16}$$

where δ_0, δ_{ao}, and δ_{ai} are the initial axial clearance, the outer bearing axial deflection, and the inner bearing axial deflection, respectively.

All bearing loads can be obtained by solving eqns (13)–(16).

2.3 Fatigue life of a tapered bearing unit

The fatigue life of a rolling bearing is calculated from the following formula:

$$L_{10} = \left(\frac{C}{P}\right)^{10/3} \times 10^6 \text{ (rev.)}. \tag{17}$$

where C is the basic dynamic capacity of a rolling bearing (N).

The equivalent dynamic bearing load P is

$$P = \frac{J_r(0.5)J_1(\varepsilon)}{J_1(0.5)J_r(\varepsilon)}F_r \tag{18}$$

where

$$J_1(\varepsilon) = \left\{\frac{1}{2\pi}\int_{-\psi_l}^{\psi_l}\left[1-\frac{1}{2\varepsilon}(1-\cos\psi)\right]^{4.4} d\psi\right\}^{1/4}. \tag{19}$$

Because a tapered roller bearing unit has two bearings, the total fatigue life of a bearing unit is given by

$$L_{10\_tot} = \left(L_{10\_in}^{-1.125} + L_{10\_out}^{-1.125}\right)^{-8/9}. \tag{20}$$

3 EFFECTS OF INTERNAL GEOMETRIC PARAMETERS ON THE FATIGUE LIFE

The fatigue life of a rolling bearing with the same external dimensions can be varied with internal geometric parameters. The effects of four geometric parameters; effective roller length, roller diameter, pitch diameter of the bearing, and number of rollers, on the fatigue life of a tapered roller bearing unit were investigated using Taguchi method. Taguchi method is based on an orthogonal array arrangement which reduces the number of experiments and the cost for the necessary information [8]. A signal-to-noise ratio (*S/N*) is used in the objective function to analyze the variation of the parameters. For the fatigue life of the bearing unit, the-larger-the-better type of the objective function is taken and is defined as

$$S/N = -10\log\left(\frac{1}{n}\sum_{i=1}^{n}\frac{1}{y_i^2}\right) \tag{21}$$

where y_i is the observed data, and n is the number of experiments in a trial. A greater *S/N* ratio shows the best performance. Relative influences of the parameters on the response can be analyzed with the sum of squares (*SS*) which is defined as:

$$SS = \sum\left(S/N - \overline{S/N}\right)^2 \tag{22}$$

where

$$\overline{S/N} = \frac{1}{n}\sum S/N. \tag{23}$$

Three levels of a parameter were chosen to investigate a trend of the response to the parameter variation. Consequently, L_9 was employed as an orthogonal array because it is suitable for four parameters with three levels. The specifications of the axlebox and the tapered roller bearing unit used in this study are shown in Table 1 and 2 shows four parameters with three levels which were selected considering the implementable range of each parameter.

Table 1: Specifications of the axlebox and the bearing unit.

Axlebox	
Axlebox load, *G* (kN)	98.7
Wheel diameter (mm)	860
Payload factor	1.0
Dynamic radial factor	1.3
Dynamic axial factor	0.12
Dynamic traction factor	1.05
Axial axlebox load distance (mm)	16.5
Tapered roller bearing unit	
Outside diameter (mm)	207
Bore diameter (mm)	131.75
Width (mm)	152.4
Contact angle (deg)	10
Distance between load centres (mm)	114.8
Basic dynamic capacity (kN)	635

The fatigue life of a bearing unit was calculated according to the orthogonal array and is listed in Table 3. The initial axial clearance was assumed 0. Table 4 shows the analysis of each *S/N* and *SS* of parameters. The effects of four parameters on the fatigue life of the bearing unit are shown in Fig. 3.

Table 2: Parameters and their levels.

Parameters		Level 1	Level 2	Level 3
Effective roller length (mm)	l_e	40.7	41.2	41.7
Mean roller diameter (mm)	d_r	17.12	18.4	19.35
Effective pitch diameter (mm)	D_p	164.4	166.4	168.4
Number of rollers	Z	22	23	24

Table 3: Values of the fatigue life with L_9 array.

No.	l_e	d_r	D_p	Z	Unit life ($\times 10^4$km)	*S/N* (dB)
1	40.7	17.12	164.4	22	24.4	107.7
2	40.7	18.4	166.4	23	36.2	111.2
3	40.7	19.35	168.4	24	49.1	113.8
4	41.2	17.12	166.4	24	31.1	109.9
5	41.2	18.4	168.4	22	33.3	110.4
6	41.2	19.35	164.4	23	46.1	113.3
7	41.7	17.12	168.4	23	28.7	109.2
8	41.7	18.4	164.4	24	43.2	112.7
9	41.7	19.35	166.4	22	42.3	112.5

Table 4: *S/N* and *SS* of parameters.

Parameters	Level	S/N	Mean	SS
	1	110.9		
l_e	2	111.2	111.2	0.151
	3	111.5		
d_r	1	108.9	111.2	9.347
	2	111.4		
	3	113.2		
D_p	1	111.2	111.2	0.005
	2	111.2		
	3	111.1		
Z	1	110.2	111.2	1.786
	2	111.2		
	3	112.1		

From Fig. 3, the mean roller diameter is the most significant parameter affecting to the fatigue life of a bearing unit. The next parameter is the number of rollers. Other parameters have a little effects on the fatigue life. The fatigue life can be decreased by increasing the effective pitch diameter. However, the fatigue life can be increased by increasing other parameters.

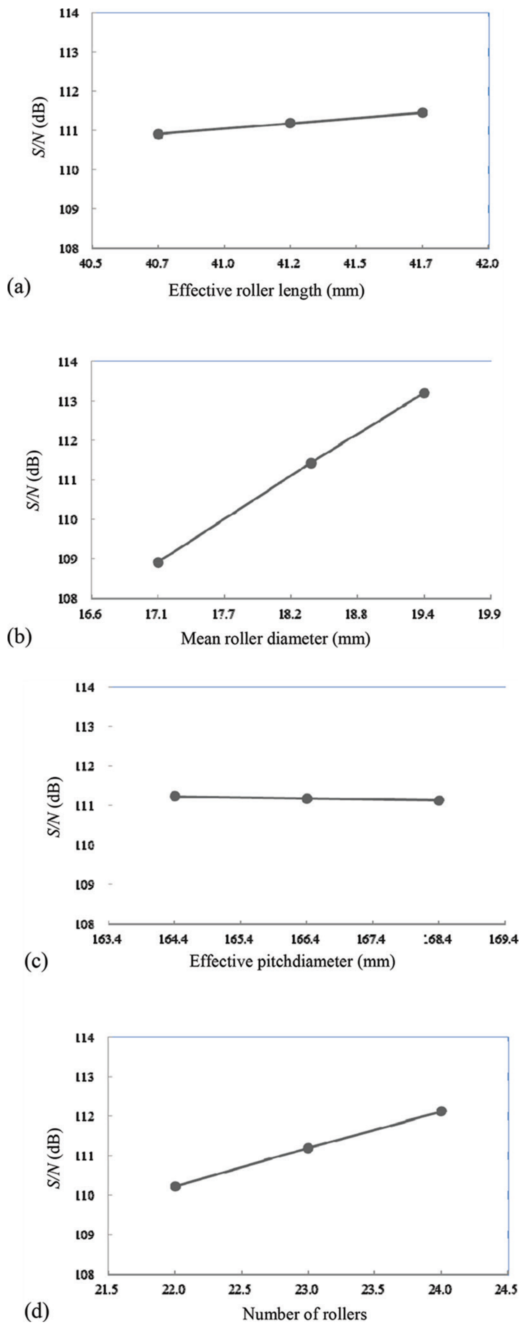

Figure 3: Effects of four parameters on the fatigue life of the bearing unit.

4 CONCLUSIONS

An axlebox bearing unit is the critical component for the reliability of railway vehicles. In order to investigate the fatigue life of a bearing unit, equilibrium conditions were formulated on the basis of Lundberg's approximate model. The fatigue life of bearing units can be varied with internal geometric parameters even if bearing units have all the same external dimensions. Taguchi method has been applied to determine the effects of the internal geometric parameters on the fatigue life. The effects of four internal geometric parameters on the fatigue life are examined by L_9 orthogonal array. From the results, the mean roller diameter is the most significant parameter and the number of rollers is the next. The effective roller length and the effective pitch diameter of the bearing unit effect a little on the fatigue life. The fatigue life of the bearing unit can be increased by increasing all parameters except the effective pitch diameter.

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