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Vibration analysis for tapered roller bearing

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Abstract— **This paper investigated the vibration behavior of a tapered roller bearing (TRB). A two degree-of-freedom vibration model was adopted for estimating the radial vibrations of TRBs subjected to angular misalignment and unbalance force due to the rotor eccentricity. The bearing stiffness used in the system equations of motion was determined from a five degree-of-freedom model of TRBs. The system equations of motion for the bearing were numerically solved using the Runge-Kutta algorithm. The bearing vibration responses in frequency and time domains were obtained and discussed considering the effects of rotational speed, axial preload, and angular misalignment.**

Keywords— **tapered roller bearings, thermal expansion, heat generation, induced preload**

I. Introduction

Rolling bearing is an essential component that is used in almost all rotating machines. Bearing vibration has a significant contribution to the noise and vibration of a rotating system. Thus, much effort has been made to reduce the vibration of rolling bearing. Reduction of vibration in rolling bearing definitely improves overall performance of the rotating system, for example, surface finish of the machined elements, accurate transmission of gear or belt systems, reducing premature failure in components, and thus saving the maintenance cost. Among many kinds of rolling bearings, due to the ability to sustain a combination of large radial and axial loads, tapered roller bearings (TRBs) are widely adopted for high load applications such as heavy-duty machine tool spindles, aircraft gearboxes, and gas turbines [1].

Investigation on vibration in roller bearings has been performed by many researchers [2-7]. However, most studies analyzed the bearing vibration in relation to the bearing geometric errors and loading. Only simple axial loading without misalignment angle has been considered for TRB vibration investigation. In practice, angular misalignment is unavoidable, and may be initiated by various factors such as shaft deflection, off-square shaft shoulder, out-of-line mounting, etc. Therefore, it is of great necessity to address the effect of angular misalignment on the vibration characteristics of TRBs.

This paper dealt with the vibration response of TRBs caused by unbalance. A computational model for TRB radial vibration was presented. Investigation was made for the effects of rotational speed, axial preload and angular misalignment on the vibration of TRBs. The computational results were provided in both time and frequency domains.

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II. Modeling of TRB

A. Bearing equations of motion

A two degree-of-freedom vibration model is used in this study. The governing equations of motion for the TRB inner ring may be written as

$$[M]\ddot{u}+([C]+[G])\dot{u}+[K]u=\{F_u\} \tag{1}$$

where the displacement and unbalance force vectors are defined, respectively, as,

$$u=\{x \quad y\}^T \tag{2}$$

$$\{F_u\}=\{me\omega^2\cos\omega t \quad me\omega^2\sin\omega t\}^T . \tag{3}$$

Here m and e denote the equivalent rotor mass supported by the TRB and the mass eccentricity, respectively. ω is the angular velocity of the bearing. M, C, and K are the mass, damping, and stiffness matrices given by

$$[M]=\begin{bmatrix} m & 0 \\ 0 & m \end{bmatrix} \tag{4}$$

$$[C]=\begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix} \tag{5}$$

$$[K]=\begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix} \tag{6}$$

where k_{ij} and c_{ij} ($i,j = x,y$) represent the stiffness and damping coefficients of TRB. A quasi-static model is used to determine the stiffness matrix of TRB as mentioned in the next section. The stiffness is estimated for TRB under radial load and axial preload. Regarding the bearing damping, only diagonal damping coefficients c_{xx} and c_{yy} are introduced to quickly obtain the steady state vibration, whereas the off-diagonal coefficients c_{xy} and c_{yx} are neglected.

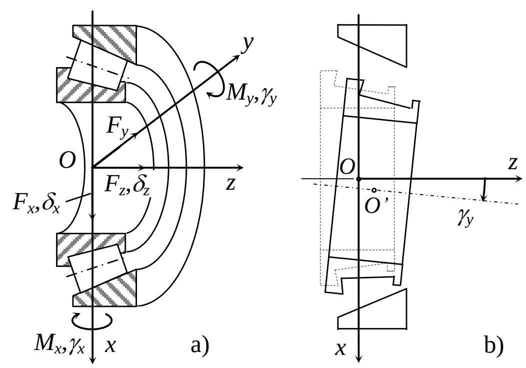

Figure 1. Quasi-static TRB model: a) Bearing loads and displacements, b) Bearing angular misalignment

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B. TRB stiffness calculation

For a general scenario, the TRB is loaded by a five degree--of-freedom load vector $\{F\}^T = \{F_x, F_y, F_z, M_x, M_y\}$. The corresponding displacement vector is written by $\{\delta\}^T = \{\delta_x, \delta_y, \delta_z, \gamma_x, \gamma_y\}$. Fig. 1(a) shows the bearing geometry with loading and displacements. Bearing angular misalignment is taken into account, which is assumed to be independent of loading and vibration as shown in Fig. 1(b). Calculation of bearing stiffness has been detailed in the recent papers by the authors [7-9]. In general, the equilibrium of all rollers needs to be found first to estimate the contact forces and moments at the roller-race contacts. Then, the global equilibrium equations of inner ring can be established based on the contact loads at all rolling elements found in the previous step. These equations are solved to give the bearing displacement. Because all the equation concerned with roller and inner ring equilibrium are nonlinear, an iterative method such as Newton-Raphson method should be used. The bearing stiffness matrix, which represents the load-displacement relationships, can be found by taking partial derivative of load vector with respect displacement vector. Then, the fully occupied (5x5) stiffness matrix is derived as

$$[k_b] = \left[\frac{\partial\{F\}}{\partial\{\delta\}^T}\right] = \begin{bmatrix} k_{xx} & k_{xy} & k_{xz} & k_{x\theta_x} & k_{x\theta_y} \\ k_{yx} & k_{yy} & k_{yz} & k_{y\theta_x} & k_{y\theta_y} \\ k_{zx} & k_{zy} & k_{zz} & k_{z\theta_x} & k_{z\theta_y} \\ k_{\theta_x x} & k_{\theta_x y} & k_{\theta_x z} & k_{\theta_x\theta_x} & k_{\theta_x\theta_y} \\ k_{\theta_y x} & k_{\theta_y y} & k_{\theta_y z} & k_{\theta_y\theta_x} & k_{\theta_y\theta_y} \end{bmatrix} \tag{7}$$

Substituting the stiffness coefficients in eq. (7) into eq. (1), the dynamic equations of motion for the bearing would be available. By solving eq. (1) using the Runge-Kutta method, the radial displacements, x and y of the TRB center can be determined as a function of time. Overall calculation for bearing vibration is demonstrated in Fig. 2.

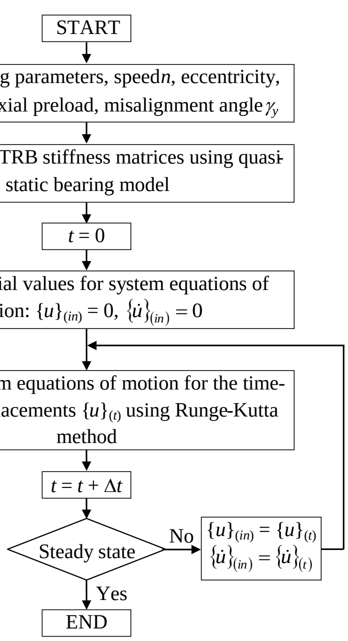

Figure 2. Time domain simulation for TRB vibration

C. Computational results

TABLE I. PARAMETERS OF TRB 30206-A

Parameter	Symbol	Value
Bore diameter	d (mm)	30.00
Outer diameter	D (mm)	62.00
Pitch diameter	d_m (mm)	45.92
Total width	B (mm)	17.25
Number of rows	h	1
Number of rollers	Z	17
Damping coefficient	c_{xx} and c_{yy} (Ns/m)	100

TABLE II. ROTOR PARAMETERS

Parameter	Symbol	Value
Mass	m (kg)	200
Eccentricity	e (mm)	0.001

The TRB vibration is calculated using MATLAB® with varying bearing load, speed, and misalignment. A sample TRB 30206-A is considered whose parameters listed in Tables 1. Table 2 displays the rotor mass parameter used for simulation.

Effect of rotational speed on TRB vibration

Fig. 3 illustrates the vertical and horizontal vibrations of the aligned TRB in time domain. In general, radial load against the rotor mass introduces anisotropic behavior in lateral vibrations of the TRB. Consequently, the vertical and horizontal vibration responses are clearly distinguishable as seen in Fig. 3.

Fig. 4 shows the frequency spectrum of the aligned TRB vibration as a function of the rotational speed. The rotational speed varies from 250 to 5000 rpm, while the axial preload remains constant at F_z = 1000N. Increasing rotational speed increases the vibration amplitude, resulting from increasing unbalance force acting on the bearing. There are two major peak frequencies: the first frequency is equal to unbalance frequency of rotor ($f_r = n/60$) and the second frequency is almost constant and independent of rotational speed. This is the natural frequency of the TRB.

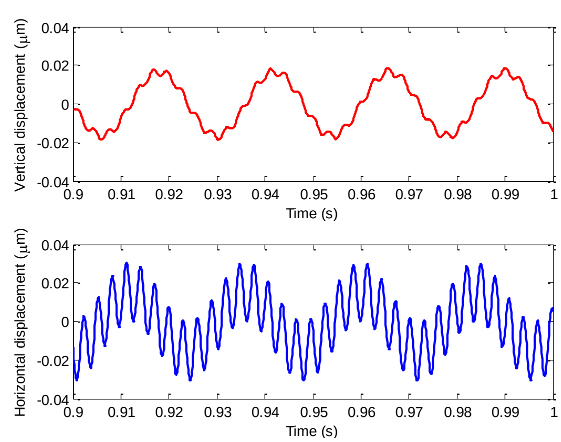

Figure 3. Time domain vibration response of aligned TRB (n = 2500 rpm, F_z = 1000 N)

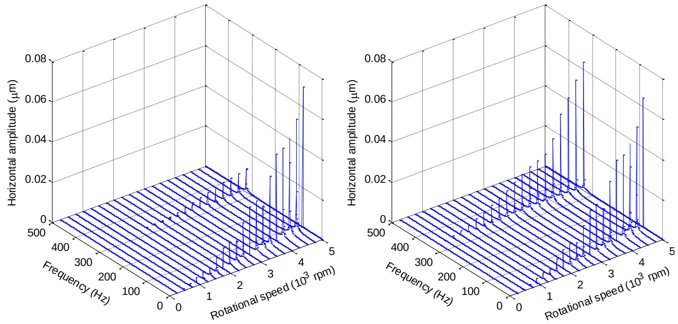

Figure 4. Effect of rotational speed on frequency spectrum of the aligned TRB vibration (F_z = 1000 N)

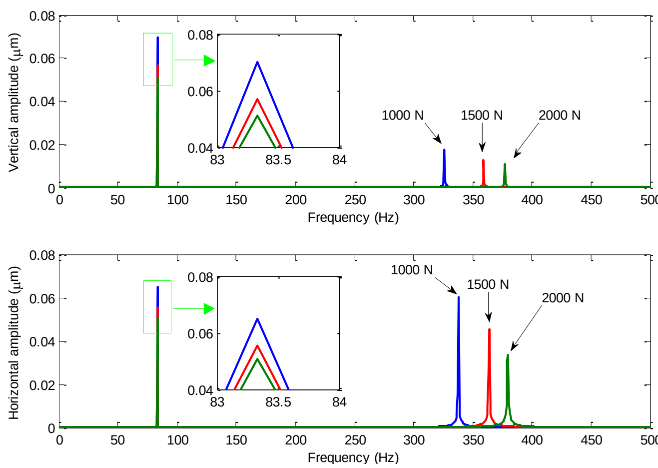

Figure 5. Effect of axial preload on frequency spectrum of the aligned TRB vibration (n = 5000 rpm)

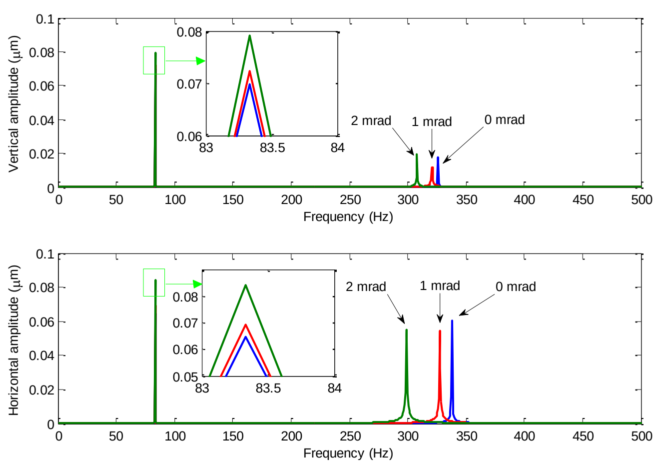

Figure 6. Effect of angular misalignment on frequency spectrum of the TRB vibration (n = 5000 rpm, F_z = 1000 N)

Effect of axial preload on TRB vibration

Fig. 5 shows the effect of axial preload on vibration behavior of the aligned TRB running at a rotational speed of 1000 rpm. This figure shows that preload does not change the first frequency, whereas it increases the second vibration frequency. Therefore, it can be deduced that axial preload alters the second peak frequency, natural frequency.

Effect of angular misalignment on TRB vibration

The effect of angular misalignment on frequency spectrum of the TRB is shown in Fig. 6. The bearing preload and rotational speed are 1kN and 5000rpm, respectively. It is observed that the first frequency, which is caused by the unbalance force, is unchanged for all misaligned TRBs. However, the second vibration frequency (natural frequency) significantly reduces with increasing angular misalignment.

Moreover, the second frequency of the horizontal vibration gets lowered down more significantly than that of the vertical vibration as angular misalignment increases. This implies that the angular misalignment in TRB softens the radial stiffness of TRB and that the stiffness in the orthogonal to the misaligned direction is more significantly reduced than the other.

III. CONCLUSIONS

A simulation of the TRB vibration with regard to rotational speed, preload and angular misalignment effects has been performed. There existed two dominant frequencies of the TRB vibration. The first frequency was caused by the unbalance, and the second frequency was a natural frequency that is influenced by bearing stiffness variation due to axial preload or misalignment angle. Increasing axial preload can enhance vibration characteristics of the TRB, whereas increasing angular misalignment or rotational speed leads to substantial increase of the TRB vibration. Bearing axial preload and angular misalignment should be considered during bearing assembly and loading for obtaining adequate bearing dynamic stiffness, so as to reduce the bearing vibration.

Acknowledgment

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