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The effect of angular misalignment on the running torques of tapered roller bearings

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ABSTRACT

The occurrence of angular misalignment can considerably change bearing characteristics. However, the angular misalignment effect on the running torques for tapered roller bearings (TRBs) has not been thoroughly investigated. This paper presents a comprehensive formula to provide the running torques for TRBs with angular misalignment between inner and outer races. Simulations were performed to investigate the effects of angular misalignment on TRB torques for two representative preloading methods, constant force preload and constant displacement preload. Angular misalignment under the constant displacement preload was found to increase the running torques under mild preload and to slightly decrease the running torques under heavy preload. Under the constant force preload, the running toques consistently decrease with an increase in angular misalignment.

1. Introduction

Tapered roller bearings (TRBs), which have tapered rollers and races, can withstand a combination of high radial and axial loads. Therefore, TRBs have been widely used in heavy-load applications, for example, heavy-duty spindles, train and automobile wheel hubs, aircraft and gearboxes. The running torques for TRBs are usually high due to their long roller–race contact lines and the large sliding friction between the roller end and guide flange. To reduce the TRB running torques, various issues may need addressed, including bearing geometry, surface quality of rolling elements, cooling, lubrication, misalignment, and preloading condition. This paper investigates the running torque of TRBs with angular misalignment.

Bearing running torque implies a rotational resistance of a bearing. Starting torque is often discriminated from the running torque. The starting torque is defined as the frictional moment that must be overcome to cause a bearing to rotate. The running torque is the moment required to maintain a constant rotational speed of a bearing. Since the running torque of a bearing relates directly to the power loss of a rotating system, it is crucial to study the bearing running torque in both design and operation to increase the system efficiency.

Early studies on bearing torque [1,2] considered only Coulomb friction from the sliding contacts. Palmgren [3] first included the effect of lubricant viscosity in a torque formula based on experimental measurements of torque for ball and roller bearings. Reichenbach [4] introduced the concept of spinning action of the ball with respect to the race, as a major component of overall resistance in ball bearings. Later, with the development of the elasto-hydrodynamic lubrication (EHL) theory, the effect of lubricant film thickness on the friction torque in rolling element bearings was taken into account. Pioneering studies that addressed the effect of EHL-based lubricant film thickness on the bearing friction torque were performed by Harris [5] and Zaretsky et al. [6]. Because of the increasing demand for higher speed and higher loading of rolling element bearings, studies on bearing torque under various lubricating and loading conditions have been conducted. Schuller et al. [7] investigated the effect of rotational speed on the power loss of a high-speed, jet-lubricated ball bearing subjected to axial and radial loads by measuring the running torque. Nishimura and Suzuki [8] presented the friction torque responses of solid-lubricated ball bearings for use in vacuum conditions. Kanatsu and Ohta [9,10] analyzed the running torque of an axially loaded deep groove ball bearing lubricated with a polymer lubricant. The effect of lubrication oil and grease on the friction torque of thrust ball and cylindrical roller bearings was investigated by Cousseau et al. [11,12] and Fernandes et al. [13,14]. Recently, Olaru et al. [15] presented a theoretical model to predict the friction torque of thrust ball bearings. The model accuracy was confirmed by experimental testing of a modified thrust ball bearing with three balls and no cage.

Regarding the running torque of TRBs, there are several analytical models, which, however, have little association with angular

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misalignment. Palmgren [3] developed a roller bearing torque formula that consists of torque components arising from the applied load and viscosity. However, this torque formula fails to provide the accurate running torque for TRBs subject to high axial load and high rotational speed [16].

Witte [17] developed a semi-empirical formula for predicting the running torque of TRBs under axial and radial loads. This formula is represented by a single term expression that is the product of several constants and exponential terms, including speed, lubricant viscosity, and external load. The TRB geometric effect on the running torque was reflected in terms of a coefficient, but detailed bearing geometry information was not required. The load, speed, and viscosity exponents were derived using curve fitting on experimental torque measurements. Because bearing designs have been appreciably improved for higher efficiency, the constants in Witte's torque formula are no longer applicable. In addition, with increasing usage of high-speed computers along with abundant bearing databases, it is of great concern to formulate bearing torques reflecting all possible factors.

Aihara [16] proposed a running torque formula for TRBs that is based on an experiment-assisted theoretical approach. The total TRB torque was defined as the summation of the EHL rolling resistance in roller–raceway contacts, the sliding friction between the guide flange and roller end contact, and the fluid-film friction. The calculated torque agreed well with that from experimental measurements in the case of axially preloaded TRBs. Later, Zhou and Hoeprich [18] extended Aihara's approach to provide an analytical formula for TRB torque calculations. To reduce the calculation efforts and validate the torque formulation for different TRB designs, they employed a numerical data curve-fitting method and established the relationship for load, material, and speed parameters with rolling resistance forces for isothermal EHL line contacts. However, the running torque of TRBs with angular misalignment has not been studied in the above-mentioned studies.

Harris [19] introduced the roller bearing friction torque formula based on calculations of friction heat generation. The slicing technique was used in which the actual contact length was divided into a finite number of laminas. The friction heat generated over each lamina was evaluated to provide the total heat generation from the entire contact area. This method can be applied for torque computations in misaligned roller bearings. Recently, Houpert [20] presented a new formula for the TRB running torque. In this study, the TRB running torque was assumed to consist of hydrodynamic rolling forces in roller–raceways, elastic rolling moments at the contact regions between the roller and raceways, and rib–roller end friction moments. The effect of angular misalignment on the TRB torque, however, has not been addressed.

Most studies on TRB torque estimations have confined themselves to properly aligned TRBs under axial or combined radial and axial loading. In realistic applications, angular misalignment is unavoidable due to many factors [19]. The occurrence of misalignment would introduce considerable changes in contact characteristics of rollers and raceways, such as contact length and load distribution in rolling elements. These may consequently alter the bearing friction torque. However, the angular misalignment effect has not been adequately included in the TRB torque formulas available in the literature.

This paper aims to improve the existing TRB torque formula by taking into account the effect of angular misalignment. Based on the TRB dynamic model presented in Tong and Hong [21,22], the roller contact characteristics, such as the contact forces between the roller and raceways, the contact forces between the roller and flange, the actual contact length between the roller and raceways, and the distance between the flange and roller end contact, are analyzed. Calculation of a TRB dynamic model is performed with respect to the two most commonly employed preloading methods, constant displacement and constant force preloading methods. Then, a modified torque equation that is applicable to misaligned TRBs is proposed and compared with the existing formulas in the literature. The computational results for TRB running torques are presented with regard to angular misalignment, axial and radial loads, and preload method. The obtained results are rigorously discussed.

2. TRB equilibrium

In order to estimate TRB torque, the contact forces between the roller and races should be determined in advance. Calculation of these forces is based on solving the bearing dynamic equations relevant to the equilibrium of the rollers and inner ring. For an aligned TRB under pure axial force F_z, the contact forces can be approximated as described by Aihara [16]. However, such a situation is very rare in actual bearing applications. For a general loading condition, as shown in Fig. 1(a), the inner ring of the TRB is assumed to be loaded by an external load vector

$$\{F\}^T = \{F_x, F_y, F_z, M_x, M_y\}, \tag{1}$$

and the corresponding inner ring displacement vector is

$$\{\delta\}^T = \{\delta_x, \delta_y, \delta_z, \gamma_x, \delta_y\}. \tag{2}$$

Considering the TRB cross-section at a particular roller of location angle ϕ, as indicated in Fig. 1(b), because the roller is displaced from its initial position by $\{\nu\}^T=\{\nu_r,\nu_z,\psi\}$, the roller contact forces Q_i, Q_e, and Q_f are generated, as illustrated in Fig. 1(c). Here, the subscripts i, e, and f denote the inner raceway, outer raceway, and flange, respectively. Q_i and Q_e can be calculated using the well-known slicing method [19,21–23]. In this method, the roller–raceway contact region is divided into n_s slices, and the total contact force is calculated by the summation of the contact forces in the individual slices q_k. It should be noted that the slice contact force is not uniformly distributed, but depends on the roller and raceway profiles. Thus, when forces Q_i and Q_e are moved to the middle of the nominal contact length, as illustrated in Fig. 2, moments M_i and M_e are induced consequently. Then, the contact forces and moments are described as

$$Q_a = \sum_{k=1}^{n_s} q_k = \sum_{k=1}^{n_s} c \times \delta_k^{10/9} \Delta l, \qquad a = i, e \tag{3}$$

$$M_a = \sum_{k=1}^{n_s} q_k \times l_k = \sum_{k=1}^{n_s} c \times \delta_k^{10/9} \Delta l \times l_k, \qquad a = i, e \tag{4}$$

The force Q_f is determined by the classical Hertzian contact theory between the flat flange and spherical roller end, as

$$Q_f = c_f \times \delta_f^{1.5} \tag{5}$$

where c and c_f indicate the contact constants for the roller-to-raceways and the roller-to-flange, respectively, which depend on the material and geometry at the contact. δ_k and δ_f represent the contact compressions between the roller and raceway and the roller and flange, respectively. h_k denotes the compression drop at slice k due to a modified roller profile. The width and axial position of slice k are Δl and l_k, respectively.

Then, the roller equilibrium equations are obtained with inclusion of the inertial effects, such as centrifugal force F_c and gyroscopic moment M_g. The roller equations are then solved to provide the roller displacements and the reactive forces Q_i, Q_e, and Q_f.

The resultant inner race contact loads at all rollers and the external load $\{F\}$ give the global equilibrium equations of the inner ring. The inner ring displacement vector $\{\delta\}$ is finally obtained by solving the global equations. The detailed descriptions for the roller load computation and dynamic equations of TRB, including