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Contact trajectory of angular contact ball bearings under dynamic operating condition

H. Xi [a,b], H.Y. Wang [c], W. Han [b,d], Y. Le [b,d], H. Xu [a,b], W. Chen [a], S.N. Xu [b], F.C. Wang [a,d,e,*]

[a] iBT Group, School of Mechanical Engineering, Xi'an Jiaotong University, China
[b] Institute of Bearing Manufacturing Technology of Shizuishan, Ningxia, China
[c] School of Mechanical Engineering, Swansea University, United Kingdom
[d] School of Mechanical Engineering, Wuhan University of Science & Technology, China
[e] Institute of Bearing Technology, United Manufacturing Group, United Kingdom

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ABSTRACT

A contact trajectory model of ball bearings under the dynamic condition was developed by the means of the development of the integrated multi-body dynamics and multi-freedom kinematics with the multi-interfacial contact mechanics. Subsequently, the general methodology was employed to investigate the dynamic contact performance of ball bearings, particularly for the contact trajectory along the raceways due to each ball against the inner and outer raceways. It was found that the radial loading and inner ring misalignment could lead to the deviation of the contact trajectory on the raceways. The variation of contact angle could bring into the correspondence with the deviation of the contact trajectory on the raceways. The oscillation frequency of the contact trajectory on the outer raceway was higher than one on the inner raceway either under the axial and radial loading or misalignment condition.

1. Introduction

Ball bearings are widely applied to various machineries in most branches of industry with low friction and suitability for high speed, especially for rotary machines due to their ability to sustain forces along the radial and axial directions. Generally, a ball bearing comprises of the inner and the outer rings with a group of balls which remained in the pockets of a cage. It is well known that angular contact ball bearings could be identified as one typical ball bearing. For instance, an angular contact hybrid ball bearing consists of the inner and outer rings as well as the cage with a group of ceramic balls. Since the better dynamic contact performance of each ball against the raceways of ball bearings can be achieved by design optimization and manufacturing technology, the high precision angular contact hybrid ball bearings can be often used in the high speed spindle of machine tools or the electric main spindle under the operating condition. Generally, the dynamic and kinematic performance of ball bearings under the operating condition is much dependent on the internal contact behavior of each ball against the inner and the outer raceways of ball bearings, such as contact loading, contact deformation and contact position etc. It is evidently believed that the contact performance of ball bearings could have a significant effect not only on the dynamic performance such as vibration and noise of high speed spindle, but also on the manufactured surface quality of working pieces by the machine tools. Therefore, it was necessary to understand the contact features of each ball against the raceways of ball bearings under the loading and motion condition by the means of the dynamic contact modeling, for example the contact point distribution could lead to the contact trajectory on the bearing surfaces. However, the study of contact point trajectory of each ball against the raceways of ball bearings under the dynamic condition has not been found in the previous research works yet. The study of contact point distribution and trajectory of ball bearings should be one significant topic of the dynamic and kinematic problems, which was possibly helpful to carry out design optimization, failure evaluation and wear track analysis of ball bearings [1–3].

An early contact point analysis of ball bearings was carried out only by simplifying the inner and the outer rings without motions, hence each ball without the orbital rotation in order to observe the contact path on the ball surface of bearings only under the pure spinning and the regular precession condition [4]. The influence of the change of contact angle and the orbital rotation period of ball bearings during transient motion and dynamic loading condition on the contact point distribution and the contact trajectory along

* Corresponding author at: iBT Group, School of Mechanical Engineering, Xi'an Jiaotong University, China.
E-mail address: f.c.wang@hotmail.co.uk (F.C. Wang).

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Nomenclature

$O\text{-}XYZ$ inertial/space coordinate system
$o_r\text{-}x_r y_r z_r$ inner raceway-fixed coordinate system
$o\text{-}x_a y_a z_a$ azimuth coordinate system
$o\text{-}x_b y_b z_b$ ball-fixed coordinate system
$o_1\text{-}x_{c1} y_{c1} z_{c1}$ contact coordinate system of outer ring
$o_2\text{-}x_{c2} y_{c2} z_{c2}$ contact coordinate system of inner ring
$\boldsymbol{r}_b$ (x, r, θ) vector of mass center of ball in the inertial space cylindrical-coordinate system
$\boldsymbol{r}_r$ (x_r, y_r, z_r) vector of mass center of inner ring in the inertial space coordinate system
ω $(\omega_{xa}, \omega_{ya}, \omega_{za})$ three-dimensional angular velocity of ball in the $x_a y_a z_a$ coordinate system
ω $(\omega_{xb}, \omega_{yb}, \omega_{zb})$ three-dimensional angular velocity of ball in the $x_b y_b z_b$ coordinate system
ω $(\omega_{xr}, \omega_{yr}, \omega_{zr})$ three-dimensional angular velocity of inner ring in the $x_r y_r z_r$ coordinate system
O mass center of the ball
o_1 contact point of ball against outer raceway
o_2 contact point of ball against inner raceway
m_b mass of ball
m_r mass of inner ring
I_b inertia moment of ball
F_{ball} $(F_{brox}, F_{broy}, F_{broz})$ forces acting on ball by outer ring
F_{ball} $(F_{brix}, F_{briy}, F_{briz})$ forces acting on ball by inner ring
F_{race} $(\sum F_{rbixk}, \sum F_{rbiyk}, \sum F_{rbizk})$ sum of forces acting on inner ring by balls
M_{ball} $(M_{brox}, M_{broy}, M_{broz})$ moments acting on ball by outer ring
M_{ball} $(M_{brix}, M_{briy}, M_{briz})$ moments acting on ball by inner ring
$F_{contact}$ (F_{xc}, F_{yc}, Q) forces acting on the contact point by the ring in the $x_c y_c z_c$ coordinate system
F_a axial force acting on inner ring
F_r radial force acting on inner ring
(η, β, γ) three Euler attitude parameters
n step number of numerical simulations.
$[A_{ib}]$ transformation matrix from O-XYZ to $x_b y_b z_b$
$[A_{ai}]$ transformation matrix from $o\text{-}x_a y_a z_a$ to O-XYZ
$[A_{c1a}]$ transformation matrix from $o_1\text{-}x_{c1} y_{c1} z_{c1}$ to $o\text{-}x_a y_a z_a$
$[A_{c2a}]$ transformation matrix from $o_2\text{-}x_{c2} y_{c2} z_{c2}$ to $o\text{-}x_a y_a z_a$
$[A_{ir}]$ transformation matrix from O-XYZ to $o_r\text{-}x_r y_r z_r$
$[r_o]$ vector of locating ball center relative to contact point of ball

Superscripts

r vector in the inner raceway-fixed coordinate system
a vector in the azimuth coordinate system
b vector in the ball-fixed coordinate system
c vector in the contact coordinate system
[…] matrix

the raceways has not been studied yet. The study of the dynamic contact point distribution and the corresponding contact trajectory needed to resolve the similar problems to the latitude and the longitude coordinates of contact points of each ball against the raceway, which were described as the function of the three Euler angles and their time derivatives. In addition, under the assumption of the pure spinning and the regular precession, the study of the operating attitude of each ball of ball bearings under the dynamic condition was assumed as the static equilibrium method so that it could not be applied to effectively and accurately evaluate the positions of the dynamic contact points of ball bearings during motion. Therefore, it was obviously necessary to simultaneously resolve the fully time dependent dynamic problem with the transient motion and the contact mechanics of ball bearings, included the attitude kinematics and the variable contact angle as well as the orbital period in order to predict the dynamic contact trajectory of each ball against the inner and the outer raceways of ball bearings.

It is obviously seen that the contact trajectory model of ball bearings under the dynamic condition should be developed by integrating the dynamic and kinematic methods with the corresponding contact mechanics of ball bearings. A significant progress of studies of dynamics and kinematics of ball bearings, including the corresponding contact and tribological problems, has been made in the last few decades [5–24]. The studies can be mainly classified into two categories: one for the equilibrium model which consists of force and moment equilibrium equations, and the other for the corresponding dynamic model which was developed on the basis of the integration of classical differential equations of motion of each bearing element [5]. A quasi-static equilibrium model was early developed for the study of motion and loading of ball bearings in terms of ball inertia and its effect on the frictional resistance resulting from interfacial slip at contact area [6]. Subsequently, the raceway control hypothesis was significantly proposed to simplify the calculation of speeds, in which the motions of rolling and spinning were assumed only to occur on one of contact interfaces of each ball against either the inner or the outer raceway of ball bearings, while only pure rolling motion existed on the other raceway without spinning and sliding [7]. Since then, the raceway control hypothesis and the ball-race traction model were popularly employed in the governing model for the study of the orbital motion, rotation, spinning of balls of ball bearings under the operating conditions of either the axial or the axial and radial loads, respectively [8,9]. It is well known from these studies that the rotation, spin, gyroscopic motion due to gyroscopic moment could take place in an angular contact ball bearing during the given operating condition. Recently, the relationship between the equilibrium equation of the gyroscopic torque of a rolling element and the friction coefficient of contact interface between the rolling element and groove was analyzed using a quasi-static model of high-speed angular contact ball bearing [10]. However, since the constrained ball motion assumption was taken for these studies, the dynamic model of ball bearings could not be used to carry out a fully time-dependent analysis of transient motions of ball bearings under the operating condition.

A generalized dynamic model with the consideration of each ball with six degrees of freedom in addition to elastohydrodynamic contact traction/slip has been developed soon [11–13]. Meanwhile, a similar dynamic simulation model of rolling bearings with less calculation time by means of parallel solution technology was developed [14,15]. The equations of forces and motions of the ball of angular contact ball bearings were solved in the cage-fixed coordinate systems to enhance the computing efficiency in another analytical model [16]. It was worth mentioning that kinematics of ball was investigated with the consideration of the local friction forces on the contact interfaces which were obtained using elastohydrodynamic lubrication theory [17,18]. However, the rotational motion equations of each ball was usually considered in one separate coordinate frame which was fixed neither in the space coordinates nor fixed in the body-fixed coordinates of each ball of bearings in most of the dynamic models