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Theoretical and experimental vibration studies of lubricated deep groove ball bearings having surface waviness on its races

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ABSTRACT

A dynamic model has been presented to study vibrations of deep groove ball bearings having waviness on surface of bearing races. Masses of shaft, races, balls and housing have been considered in modelling. Stiffness and damping due to non-linear Hertzian contact and lubricant film have been incorporated in dynamic model. The coupled solution of governing equations of motions is obtained using Runge Kutta method. The vibrations velocity spectra of bearing housing computed through dynamic model have been validated with experimental results, which were generated using healthy and defective deep groove ball bearings. The effect of governing parameters like radial load, shaft rotational speed, waviness order, waviness amplitude on amplitude of waviness defect frequency have been investigated broadly using both theoretical and experimental results. Severe vibration amplitudes have been observed in presence of waviness order equal to number of balls. Good correlations between theoretical and experimental results have been observed. The present study would give researchers new insights into dynamic behaviour of shaft bearing system.

1. Introduction

The rolling element bearings are mostly used in all the rotating machinery ranging from domestic appliances to heavy industrial applications. The probability of functioning these machineries in industries greatly depends on the reliability of supporting bearings. It is essential to mention here that defective machine components produce vibrations, which destroy/disturb functionality of machine. Rolling bearing in any machine is a major source of vibrations due to varying compliance, time varying contact forces, radial clearance, and bearing defects (local and distributed). Therefore, it is essential to study the vibrations of shaft-bearing system to check the influence of various sources of vibrations. The vibration analysis helps in condition monitoring, predictive maintenance of machinery and design modifications of rolling element bearing manufacturers for reducing the vibration level. Actual vibrations generated by bearings can be captured through proper sensors, while vibrations can also be simulated through proper dynamic model. The accuracy of vibrations obtained through any dynamic model depends on the operating parameter considered during modelling. The vibrations captured practically or theoretically are analysed in time domain and frequency domain for defect detection in machine components.

Authors of review papers [1,2] have summarised the dynamic models to predict vibrations of rolling element bearing with considering various distributed defects like off size rolling elements, waviness of balls and races, surface roughness, misalignment of races and eccentric races due to manufacturing errors. The contact forces between races and rolling elements vary nonlinearly due to the manufacturing imperfections or distributed defect on bearing races. The premature surface fatigue failure of bearing occurs due to this nonlinearity of contact forces, results in reduction of bearing life.

Yhland [3] has studied measurement of geometrical imperfection in form of waviness of bearing races and their effect on bearing vibrations. Tallian and Gustafsson [4] have noticed that the certain orders of bearing waviness (number of waves per circumference of raceway) have predominant effect on vibration generation in machine. Meyer et al. [5] have proposed an analytical model for flexural vibration of the stationary race under axial load due to presence of waviness on the moving race or unequal ball diameter. However, researchers of references [5,6] have found higher vibration amplitude for outer race waviness as compared to inner race waviness. Wardle [6,7] has also derived the theoretical model to predict the vibration forces produced by waviness of the rolling surfaces (inner race, outer race and ball) of thrust loaded ball bearings and validate the predicted results with measured vibration spectra obtained from experimentation. Wardle [6,7] have observed that the harmonics in bearing spectrum change

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Nomenclature

A_0	initial amplitude of waviness
A_m	maximum amplitude of waviness
C	damping coefficient, N-sec/mm
C_r	internal radial clearance
D	pitch diameter of the bearing, mm
d	ball diameter, mm
d_{in}	diameter of inner race, mm
d_{out}	diameter of outer race, mm
d_s	diameter of shaft, mm
E	Young's modulus, N/mm^2
EHL	elastohydrodynamic lubrication
K	stiffness, N/mm
M_{bi}	mass of i^{th} ball, gm
M_h	total mass of housing with mass of outer race, kg
M_s	total mass of shaft with mass of inner race, kg
N_b	number of balls in bearing
N_w	number of waves (waviness order)
P_d	diametaral clearance, mm
Q	radial load applied on the test bearing, N
t	time coordinate, sec
W	normal load acting on ball, N
X, Y	deflection along the axes, mm
Δ	contact deflection in radial direction, mm
δ^*	dimensionless contact deflection
η	loss factor (depends on the material)
ν	Poisson's ratio
ω	angular velocity, rad/s
θ	angular position of the ball, rad
φ	contact angle of ball with race waviness

Subscript

b	ball
c	cage
eq	equivalent
h	housing
i	ball number
in	inner race
out	outer race
s	shaft
t	time coordinate

with waviness order. Moreover, Ohta and Sugimoto [8] captured axial and radial vibrations for outer ring of tapered roller bearings at various shaft rotational speed. The peaks occurred in spectra of radial vibrations and axial vibrations are found different [8].

A multi degree of freedom dynamic model to study vibration generated by bearing having distributed defect on its races have been presented by Tandon and Choudhury [9,10]. Authors have included masses of shaft, inner race, outer race, housing and linear stiffness for bearing in their suggested model. The expression for excitation forces generated by race waviness has been derived using Fourier coefficients. While, Akturk [11] has assumed the sinusoidal wave with radial clearance for consideration of race waviness in dynamic model of angular contact ball bearing. The authors have concluded that severe vibration occurs when number of waves (N_w) equal to number of ball (N_b) and $N_w = N_b \pm 1$ [11,12]. Researchers [3–12] have observed the peak amplitude of vibration at $N_b*\omega_c$ and its harmonics in case of outer race waviness and at $N_b*(\omega_s–\omega_c)$ for an inner race waviness with its harmonics. However, it is difficult to identify the bearing fault only from the vibration spectra at the initial stage. In such cases both the frequency information and magnitude information of vibration spectra should be used for reliable bearing defect detection [13].

More sophisticated dynamic models with consideration of centrifugal force, gyroscopic moments, diametric clearance, external load, frictional moment, lubrication and waviness direction for precise defect detection have been suggested by many researchers [14–18]. Jang and Jeong [14,15] have proposed the 5 degree of freedom dynamic model and concluded that the centrifugal force and gyroscopic moments of the ball play important role on principal vibration frequencies for rotor-bearing system. Govardhan et al. [16] have observed additional sidebands at shaft frequency in vibration spectra of defective bearing under dynamic loading condition. Moreover, the load dependent frictional moment significantly enhances the amplitudes of vibrations in comparison to load independent frictional moment irrespective to values of waviness amplitude and waviness order [17]. Cao and Xiao [18] have also established comprehensive dynamic model and carried out the case studies to illustrate the effect of loading conditions and surface waviness defect on the vibration responses of the spherical roller bearing.

Ono and Okada [19] have studied the effect of bearing outer race waviness in combination of shaft imbalance and radial clearance on frequency response of shaft bearing system. Authors observed dominant frequency peaks at $q*N_{b*}\omega_c$ and its multiple harmonics in absence of shaft imbalance. They have also noticed the dominant frequency peaks based on waviness order in presence of radial clearance. Liu and Shao [20] have presented a sophisticated 2-DOF dynamic model in which race waviness has been incorporated through combined time-varying deflection excitation and time-varying contact stiffness excitation. The authors have studied the vibrations generated by a lubricated roller bearing with a uniform and a non-uniform surface waviness on its races. Authors have concluded that the contact stiffness between roller and races rises with increase of the number of waves and the waviness amplitude. Authors have also noticed the variation in amplitude of uniform and non-uniform waviness on bearing races. Rahnejat and Gohar [21] have studied theoretically the vibration generated by a rigid shaft supported by lubricated ball bearings considering shaft unbalance and inner race surface waviness. In case of inner race waviness, authors observed the frequency peaks at product of number of waves and shaft speed. While, Harsha et al. [22] have investigated theoretically, the effect of inner and outer race waviness on the vibration characteristics of a rotor bearing system with consideration of rotor unbalance. Xing et al. [23] have presented non-Hertzian contact model to study vibration due to waviness of spherical roller bearing considering external load and self-aligning contact angle. The effect of the waviness is reduced with the rise in magnitude of self-aligning contact angle.

Along with the loading conditions the presence of lubricant viscosity and rotational speed also play significant role in dynamic conditions of shaft bearing systems by changing the stiffness and damping parameters of the systems [24–27]. Therefore, researchers [28–30] have computed lubricant film thickness between rolling elements and races. Sopanen and Mikkola [28,31] have added the lubricant film thickness into elastic deformation to find the total elastic deformation. Moreover, authors have found that the diametric clearance has significant influence on natural frequencies and vibration response of the system. The stiffness and damping characteristics of isothermal elasto-hydrodynamic mixed lubricated point contact are evaluated numerically considering surface roughness effect including asperity contact load by Sarangi et al. [30].

Literature review reveals that majority of the researchers have developed vibration models for bearings with race waviness considering either combination of masses of shaft and balls or masses of shaft and housing. The authors of the present paper could not notice any dynamic model for the vibration study of a defective deep groove ball bearing which include masses of the shaft, balls, and housing, along with