Dynamic Behavior Analysis of Rolling Element Bearing

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Abstract. Dynamic analysis of rolling element bearing is important due to the increasing demand in running accuracy of rotor bearing. A lot of research work has been done on rolling element bearing as a structural element of the machine and vibration generation source. In this study, an improved model is designed to analyse vibrations in a ball bearing during run-up. The integration technique is use to simulate the response of defective bearings. Dynamic response of bearing has been simulated to study the effect of parameters such as defect width, depth and radial clearance. The results are observed as the peak frequency amplitude decreases with increase in defect width, 40% decrease for 133% increase in width, whereas it increases with increase in defect depth, 75% increase for 133% increase in depth.

1. Introduction

There is huge demand of special purpose bearing with zero noise. So the noise and vibrations generated during running condition should reduced. The perfect geometrically designed bearings also produce noise and vibrations. It is generated due to limited number of balls, unbalanced rotor force, distributed and localized defects. Machinery upkeep in such state is incredibly expensive yet it is necessary in order to keep machines functioning. Externalities arising from operation and the environment make existing solutions to counter the problem unacceptable. As such, infrastructure maintenance poses a huge cost, so there is a strong demand for reducing such costs. The optimal resolution to the problem is analyze it in development stage and within that, process, anomaly detection, is the most crucial aspect.

Recent advances in materials and production techniques have enabled the development of bearings for very high speeds. One such application is the gas turbines which normally operates at very high speeds which results in a considerable increase in stress levels in bearings. The dynamics of bearings suitable for such an application becomes difficult because the rolling elements are slipping due to centrifugal forces that act on it. Despite such difficulty, the researchers are working to control the dynamic behavior of the bearing for high-performance applications. The dynamic modeling of the bearings comes in the sphere of non-linear systems.

The analysis of the vibrations is becoming important to get better running accuracy. Development in rotor dynamics has led to high speed and low weight rotors, which has made higher order effects such as non-linearity significant and it is essential that these effects are taken at the design level, as at high speed the failure can be very dangerous.

2. Dynamic Behaviour of Rolling Element

The bearing dynamic behaviour is analysed with localized defect variation. The sources of non-linearity are due to the Hertzian contact, source of parametric excitation, localized defect and internal
radial clearance. The non-linearity is observed due to fixed number of rolling elements which are rotating at different velocities with respect to the guiding races generate a time-varying stiffness component. The present study is an attempt to analyse effects of defect parameters and radial clearance on the bearing response.

Defects may arise in rolling element bearings while manufacturing or during use. A variety of different techniques have been employed for the detection of defective bearings. Methods involving the design parameters and simulation of defects provide an accurate approach to analyse dynamic behaviour of rolling element bearing.

The good geometrical design generate noise and vibration in running condition. This is due to highly non linear behaviour of bearing structure that bear the external load. And these rolling elements are rotating at speed different from the guiding races which produce a time-varying stiffness coefficient which results in vibrations [1]. These vibrations are significantly increased if there is a defect in the bearing. Defects in bearings can be mainly divided into two groups, localized and distributed defects. Distributed defects consist of off-sized rolling elements, misaligned races, surface roughness and waviness [2]. The source of the generation of these defects is generally an error in manufacturing processes [3]. The spalls, pits, and surface cracks of the races and the rolling elements are considered as local defect. One of the major modes of failure of the rolling element bearings is the formation of spall on the races, in this case, thin flanges of metal come off the surface due to excessive rolling pressure [4]. When a defect on the race comes in contact with a rolling element, it results in sudden changes in the contact stresses. This produces vibrations in the bearing.

A lot of work has been done on bearing defect detection [5-7]. Yamatoto [12] tried to investigate the effects of radial clearance in a bearing and found that as the clearance is increased, the amplitude of critical speed decreases. Sankarvelu [13] tried to determine the dynamic parameter of a rolling element bearing through an experiment. Tiwari [14-15] tried to study the non-linear behaviors introduced by the radial clearance of a rolling element bearing and also tried to analyze the characteristics of an unbalanced rotor. Tandon and Choudhury [4] have written an extensive review of the different types of acoustic and vibration techniques that are used while detecting vibrations in rolling element bearings. But there are very few works that have presented a mathematical model for simulating rolling element bearings having localized defects and most of the models that have been proposed have tried to observe the effects of waviness of the races and rolling elements. [8-11]. Harsha [16] analysed rotor supported borings response in simulated for waviness and radial internal clearance condition. He concluded that the bearing response was associated with the inner and outer race ball pass frequency and that the serious vibrations occurred when the number of rolling elements equaled the waves on the outer race. Harsha [17-18] also tried to observe the dynamic behaviors of rolling element and developed a rotor dynamic model.

3. Proposed Model
In this section, we have proposed a model to study the dynamic response of a bearing. First, the total energy of the system is calculated for all the components of the bearing. And then the governing equations for the rolling element bearing have been derived using the Lagrange equation. The schematic diagram of a rolling element bearing is shown in Fig. 1.

While modeling, it is considered that the bearing acts as a spring-mass system in which the rolling contacts are acting as non-linear springs as shown in Fig. 2. Since the contact forces will act only when the rolling elements are in contact with the guiding races, so the respective spring forces will only act when the instantaneous length of the springs is smaller than its normal length, i.e. when the springs are in compression. Otherwise, the rolling elements will not be in contact with the races and the contact stress will be equal to zero.
Designing a real rolling element bearing is very complex, so in order to simplify the modeling of the bearing some assumptions have been made:

- All the bearing components and the rotor are rigid, i.e. there is no bending.
- All the parts of the bearing, the races, and the rolling elements can move in the bearing’s plane, there is no movement in the perpendicular plane.
- The rotation of the rolling elements about their own axis is absent. So, it can be said that the rolling elements and cage do not interact.
- Deformations occur due to the Hertzian theory of elastic contact. Small elastic deformations of the rolling elements, the inner and outer races have been taken into account, but plastic deformations have been neglected.
- The angular velocity of the cage remains constant.

There is negligible damping of rolling elements. Presence of a small amount of lubrication and friction is responsible for the damping. It is very hard to estimate the amount of damping caused by these sources because of the presence of another unnecessary damping present in the system.

There is no slip condition. In other words, there is enough friction present between the rolling elements and the guiding races that the rolling elements do not slip while they are rolling. This condition can help in deriving a relation between the cage velocity and the inner race velocity. It has been considered that the temperature remains constant during bearing operation. So, all the effects that change in temperature may have caused such as, thermal expansion of bearing components, change in viscosity of the lubricant, etc. have been neglected.
The cage ensures the constant angular separation (β) between the rolling elements. Hence, there is no interaction between rolling elements. In addition, at any given instant, some of the rolling elements will be in contact with both races. Therefore,

The presence of the cage ensures constant angular separation between the rolling elements, so the rolling element angles can be related as,

\[ \text{(1)} \]

Therefore,

Now, the equations of motion that describe the dynamic behavior of the rolling element bearing can be derived using the Lagrange equation,

\[ \text{(2)} \]

Where \( V, T, f, p \) are the potential energy, the kinetic energy, generalized external force vector, and generalized co-ordinate vector respectively.

In order to calculate the total potential-energy and the kinetic-energy, the system can be divided into components, and the potential and kinetic energies of those components can be calculated.

So, the potential and kinetic energies can be subdivided into contributions from the various components i.e., the inner race, the rolling elements, the outer race, and the rotor.

The total potential energy of the system (\( V \)) is the sum of the potential energy of the rolling elements, the inner and outer race, the rotor and the springs. Hertzian Theory of elasticity has been used in order to calculate the amount of deformation at the contacts-

\[ \text{(3)} \]

Similarly, the total kinetic energy of the system (\( T \)) is calculated by summing over the kinetic-energy of the inner race, the rolling elements, the outer race, and the rotor.

\[ \text{(4)} \]

4. Effect of Different Parameters on Bearing Response

The results are simulated for localized defects. The effect of different bearing and defect parameters on the dynamic response of a rolling element bearing are plotted for different size of fault as the width of the defect, depth of the defect, and radial Internal Clearance. The effect of these defects has been observed on the dynamic response of a rolling element bearing by carrying out a series of simulations by just changing the value of that parameter.

The effect of the width of the rectangular spall defect on the dynamic response of a rolling element bearing has been observed by carrying out simulations of 3 bearings having similar physical properties, only the width of the defects of these bearings is different as shown in table 1. These bearings have rectangular spall defect of width 0.015, 0.025 and 0.035 radians on their inner race.

| Table 1. Peak Frequency Amplitude for Defect width. |
|-----------------------------------------------|
| Defect Width (in radians) | 0.015 | 0.025 | 0.035 |
The effect of the depth of the rectangular spall defect on the dynamic response of a rolling element bearing has been observed by carrying out simulations of 4 bearings having similar physical properties, only the depth of the defects of these bearings is different as shown in table 2. These bearings have rectangular spall defect of depth 0.003, 0.005, 0.007 and 0.01 mm.

| Defect depth (in mm) | 0.003 | 0.005 | 0.007 | 0.010 |
|----------------------|-------|-------|-------|-------|
| Peak Frequency Amplitude | $10 \times 10^{-4}$ | $8 \times 10^{-4}$ | $6 \times 10^{-4}$ |       |

The effect of the radial internal clearance on the dynamic response of a rolling element bearing has been observed by carrying out simulations of 5 bearings having similar physical properties, only the clearance of these bearings is different. These bearings have clearance of 0.02, 0.0205, 0.021, 0.0215 and 0.0220 mm. The peak frequency amplitude is shown in table 3.

| Radial Clearance (in mm) | 0.0200 | 0.0205 | 0.0210 | 0.0215 | 0.0220 |
|--------------------------|--------|--------|--------|--------|--------|
| Peak Frequency Amplitude | 109.7  | 121.6  | 131.1  | 138.3  | 145.4  |

5. Conclusions

In this study, the dynamic behaviour is analysed with variation in design parameters of a rolling element bearing having a localized defect. First, we tried to analyze response of a healthy bearing to find out the effect of parametric excitation. In the next step, we tried to introduce a rectangular spall defect on the inner race of a healthy bearing and tried to analyze the behavior of that bearing. The analysis shows that the defective bearing gives a peak at around 116 Hz which is very close to the Inner Race Ball Pass Frequency (IBPF).

The defect is simulated to see its effect on the behavior of a bearing. The effects of change in defect width, height, and radial internal clearance have been investigated by running simulations on bearings having different values for these parameters. The results of those simulations are as follows:

- Effect of defect width: The peak frequency amplitude decreases as the width of the defect increases.
- Effect of defect depth: The peak frequency amplitude increases as the depth of the defect increases.
• Effect of Radial Clearance: The peak frequency variation is proportional to radial clearance of a bearing.

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