A REVIEW ON BEARING CONDITION MONITORING AND DIAGNOSIS THROUGH VIBRATION AND ARTIFICIAL INTELLIGENCE TECHNIQUES

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ABSTRACT

The important sources of vibration generation considered here are as the non-linear stiffness/damping and localized defects of the contacting surfaces of bearing elements. It also covers non-linear dynamic mathematical modeling of cylindrical roller bearing for localized defects. The numerical integration technique Newark-has been used for solution of the system equations. The parameter of study for cylindrical roller bearings is rotor speed. Results have been analyzed by Fast Fourier Transform (FFT), phase plot and Poincare maps. focus on use of multi-parameters, synthesis of fault features and optimization of feature sets, in order to improve fault diagnostics accuracy, which would thus enhance machinery reliability, availability, safety, and reduce maintenance costs. The complex and non-stationary vibration signals with a large amount of noise make fault detection of rolling element bearings very challenging, especially at the early stage. Daubechies wavelet is popular for smoothing of signals so, it is chosen for reducing the background noise from raw vibration signal. This research investigates the possibilities of improving machinery diagnostics accuracy based on time domain features and various intelligence techniques like artificial neural network, support vector machine.

***Keywords**: Cylindrical Roller Bearing, Artificial Intelligence, Condition Monitoring .*

INTRODUCTION

Machinery is used for the providing services, manufacturing and all most all activity surrounding to the human being. Turbines, Pumps and Compressors are playing key roles as rotating machinery in power plants, oil refineries, chemical engineering plants, air plane engine and most of all the industries. Bearing is most commonly used important component to be found in all rotating machinery. It is found form bicycle to turbo machinery and helicopter. Bearing is not only providing relative motion between support bodies and rotating part but also transmit the load to the base.

There are many reasons for the deterioration and damage of bearings. Main causes of failure are the high rotating speeds, over load and harsh working conditions. Catastrophic failure of bearing failures without warning can damage human lives and properties. Sometimes personal and economical loses due to bearing failure accident cannot be fulfilled over the period of time. To understanding the behavior of bearing during operation it is very important to develop mathematical model for real application. One such application of rolling element bearing is the high speed rotary gas compressor, which operates at very high speeds resulting in considerable increase in stress levels in bearings. The dynamics of bearing for such an application becomes difficult because of centrifugal forces acting on the rolling elements, material properties and the slipping of the rolling elements as they roll on the race. Despite such difficulty, it is very important to model the dynamic behavior of the bearings for high performance applications. The vibration during operation. Even new and geometrically perfect bearing is also generating the vibration. This is due to the finite number of the bearing element and changing of the load zone. The other possible sources of the rolling element bearing vibrations are the defects of the rolling bearings, unbalance load and misalignments. Therefore it is very important to clear understanding of vibration signals associated with rolling element bearings. Vibration Analysis has proven to be a powerful tool in assessing the health of a machine.

BEARING

Bearings are further classified as per their geometrical change in shape for the load requirement as Radial, Angular and Trust bearings. Rolling bearings are widely used bearing because of wide range of availability, easy to use and versatility. Wide ranges of bearings are being designed for carrying both radial and thrust loads. Well designed and precisely manufactured bearings can use for wide range of operating condition of verity load and rotating speed. Now a day, rolling bearings are easy to maintain due to advancement in materials and lubrication and bearing design.

Bearing Defects

There are two types of defects occur in rolling element bearing: one local defects, In which defect crack pits, spells, defects generated due to fatigue are considered. And second is distributed defect.

Common Causes of Bearing Failures

There are many causes for developing defects in different parts of bearing. It is difficult to identify exact cause of bearing defects. Moreover one or more following reasons are responsible to damage the bearings. If it is taken care about all the causes it is improve the operation of bearing, reduce catastrophic failure and increases the life of bearing

BEARING CONDITION MONITORING

In present era of competition, any industries cannot survive if there break down time is high due to failure of any component of machinery. To maintain the good condition of machinery, industries used different maintenance techniques. The purpose of maintenance is keeping machinery and plant at high reliability, safe and consistent. This will result industries profitable. The basic types of maintenance techniques are breakdown maintenance and preventive maintenance. Breakdown maintenance was not used by industries due to west of time and money. Preventive maintenances further classified as maintenance on fixed time or routine maintenance, opportunity maintenance and condition based maintenance.

Condition monitoring is one of most power full preventive maintenance techniques. The machine condition monitoring knows what to look for. Successful diagnosis is having the ability to measure different parameter lively and to correlate the output results with known failure mechanisms. Condition monitoring of machines provides knowledge about the present condition, deterioration rate and performance of machines. Any degradation in performance can be detected and preventive action taken at the appropriate time to avoid sudden failures. This is carried out by monitoring such parameters as wear debris in oil, vibration, acoustic emission and measuring temperature etc. The changes in measuring parameters of monitoring systems help in the detection of the development of faults, diagnosis of causes of problem and anticipation of failure. Maintenance corrective actions can be planned accordingly. The application of condition monitoring in bearing results in saving in maintenance costs, improved availability of machinery and safety. Online or offline condition monitoring is done in industries

LITERATURE REVIEW

[1]. Then Gupta et al. (1977) have derived differential equation of motion for an angular thrust loaded ball bearing about its initial conditions and ball mass center[2].Furthermore, Sunnersjo (1978) was the one who has reported theoretical and experimental work on non-linear model of rolling bearings supporting a horizontally balance rotor with a constant vertical radial load. The non-linearity introduced was due to Hartzian contact stress, radial internal clearance and parametric effect owing to varying compliance[3]. To counter this effect, Rahenjatet al. (1979) have investigated the axial profile and effect of roller misalignment for taper roller bearing. Also represented numerical method for the pressure distribution for taper roller[4]. Later, Mayer et al.

(1980) have considered distributed defect in rolling bearing systemand shows time-variation contact forces which exist between the rolling element and race ways of the bearing[5].

White et al. (1984) have modeled equation of motion for rolling element bearing with considering two degree of freedom and subjected to external excitations. In which outer race and inner race considered as rigidly mounted on shaft and housing. Bearing system has been modeled as spring, mass and damping system[6]. Rahenjat et al. (1988) have made model for rigid shaft and radial deep groove ball bearings. The model includes the consequences of unbalance and waviness on the bearing surface. Frequency spectra and phase plane plots had been pretended, which helps in understanding the non-linear dynamics of the system[7]. Researchers have contributed gradually increasing nonlinearity in model and degree of freedom of whole bearing systems. In these extension, the support ball bearings used in precision spindle of machine have modeled by Matsubara et al. (1988) with considering bearing as a piecewise linear springs and External concentrated load were provided on the both end of the spindle baering[8]. Aini et al. (1990) have developed simulation model for precision machine tool spindle, supported by a pair of lubricated angular contact ball bearings under moment loading at bearing support. They prepared five degree of freedom model and determined the system exciting frequencies and compared them with those obtained from the same model under dry contact conditions. It was found that the oil film contributed little to the overall frequency response[9,10].

Tiwari and Vyas (1995) have estimated the non-linear stiffness parameter of rolling element bearings in a rotor bearing system. The theoretical results wereequated with experimental findings[11]. Tiwari and Vyas (1997) have further extended their work on a balance rotor to estimate bearing parameters in a non-liner rotor bearing system undergoing small residual imbalance forces along with random forces[12]. Akturk et al. (1997) have investigated the effect of preload on bearing and number of ball on the shaft vibration. Amplitude of vibrations was considerably reduced if bearing is preloaded and numbers of balls are correctly selected it means both parameters greatly affected dynamic behavior of bearings[13].

Another class of bearing defects may be categorized as point or local defects. This includes cracks, pits and spalls in the running surfaces, as well as practical contamination of the bearing lubricant. These kinds of defects manifest themselves in the bearing vibration signal as vibratory transients, which result from discontinuities in the contact forces during rotating of bearing.

NEWMARK-β METHOD

Several techniques are available for solving nonlinear equations iteratively. For finding out transient response through nonlinear differential equation of motion, Newmark-β method is used iteratively at every time increment is obtained. The Newmark –β method based on acceleration varies linearly between two instants of time. The velocity and displacement are calculated from acceleration based on two parameter α and β. The equation for velocity and displacements are given as,

Algorithm based on Newmark Beta method:

[A] Initial Computations:

1. Form stiffness $[K]$, mass $[M]$ and damping $[C]$ matrices

2. Initialize{ } $\{\dot{}\}$ $\{\ddot{}\}$

3. Select time step t, parameters α and β, and calculate integration constants,

, , , , , ( )

4. Form effective stiffness matrix:

[b] For each time step:

1. Calculate effective force vector at time *t* + *t*

SIMULATION OF MODEL

Initial condition for new mark-β method

The initial condition and size of time step are very important in time iterative dynamic solution process. For solving above equations, time step t = 10^{-5} is given stable output. The output is very sensitive about the initial values also. So, for stability and accuracy the following input at t=0 is given. Initial displacement in vertical and horizontal direction is x_0= 1μm, y_0 =1μm. Initial velocity in respective directions are $\dot{x}_{0}$ = o, $\dot{y}_0$ =0. To observe the effect of rotational speed and localized defect, few outputs at various speeds were obtained in vertical and horizontal directions through simulated mathematical model and it is behavior is discussed. The responses are obtained in frequency domain with Poincare maps. The simulation frequency is compared with calculated frequencies of existing model for same operating condition.

NJ 305 bearing is considered for simulation of mathematical model. Same bearing is used for experimental work. For calculation of characteristic fault frequency bearing geometrical and operational data is required. The geometrical data of bearing is given.

3.3.2 Response of Inner race defect

The responses are obtained for 0.5 mm localized defect on inner race surface of bearings at 1000 rpm and 1800 rpm rotational speed of balance rotor. Horizontal and vertical responses of outer race defect are analyzed for frequency spectrum, displacement and velocity time plot and phase plot of velocity-displacement.

With considering no slip between inner race and roller, the inner race defect frequency is calculated by equation B.1 of Appendix B. Where ω_2 is rotational frequency of inner race or shaft, d is the ball diameter and D is the pitch diameter of bearing and N_b is the number of rolling elements and α is the contact angle.

For the bearing geometry and rotor speed 1000 rpm, the significant components are the rotor frequency (ω_2) 16.6 Hz, Varying compliance frequency (VC) 66.21 Hz and the ball passage frequency on the outer race (ω_{bpfi}), 100.45 Hz as per the calculation based equation B.1. The vertical displacement response of vibration appears at ω_{bpfi} (100 Hz) is 5.2 μm as shown in figure 3.7. Other peaks aptitudes are also appear at 7.8 μm and 6.9 μm at ($2\omega_{bpfi}$) 190 Hz and (ω_2)15Hz respectively. And same horizontal amplitude 5,2μm peak appears at (ω_{bpfi}) 100 Hz, ($2\omega_{bpfi}$) 190 Hz and (ω_2)15Hz are shown in Fig. 3.8. The chaotic behavior is observed as periodic from the observation of phase plot for horizontal and vertical response at 1000 rpm.

Similarly, at 1800 rpm speed, the significant components are the rotor frequency (ω_2) 30 Hz, Varying compliance frequency (VC) 119 Hz and the ball passage frequency on the inner race (ω_{bpfi}), 180.81 Hz as per the calculation based on equation B.1. Responses of mathematical modeling in vertical and horizontal directions shown are in Fig. 3.9 and 3.10 respectively. The vertical displacement response of vibration appears at (ω_{bpfi}) 200 Hz is 5.9 μm. And other peaks aptitudes are also appear at 4.8 μm and 3.6 μm at (VC) 115 Hz and

(ω_2) 30 Hz respectively. Horizontal peak of 5.0 μm appears at characteristic fault frequency

(ω_{bpfi}) 200 Hz and at (VC) 110 Hz. Peak is also appear at rotational frequency (ω_2) 27 Hz,

its magnitude is 5.1 μm. The chaotic behavior is observed as periodic for the horizontal and vertical response.

Response of Outer race defect

Horizontal responses of outer race defect are analyzed for 1400 and 1800 rotational speed with outer race defect size is 0.5 mm. outer race characteristic defect frequency is calculated as per equation (B.2) of Appendix B.

For rotor speed1400 rpm, the significant components are calculated as, the rotor frequency ($\omega_{2)}$ 23.3 Hz, and the ball passage frequency of the outer race (ω_{bpfo}), 92.7 Hz.

The peak amplitude of horizontal response appears at the (ω_{bpfo}) 93 Hz is 5.0μm, as shown in Fig. 3.11. Due to interaction with rotational frequency and fault frequency, the other peak of 4.7μm is appears ($\omega_{bpfo}+3\omega_2$) 169 Hz. The chaotic behavior is observed as period.

The significant component of outer race fault at 1800 rpm are rotor frequency (ω_2) 30 Hz, and the ball passage frequency of the outer race (ω_{bpfo}), 119.2 Hz. The maximum peak appears at (nearer to ω_{bpfo}) 110 Hz and ($\omega_{bpfo}+ \omega_2$) 145 Hz is 2.85 μm. The other peak appears nearer shaft rotation frequency (ω_2) 25 Hz.

From the calculated frequency and characteristic fault frequencies appear in simulated results are very nearer. For the inner race defect the frequencies appears at harmonics of fault frequency also. And for the outer race defect frequencies appears at sum of the characteristic fault frequency and rotational frequency of rotor or inner race.

CONCLUSION

From the experimental work and mathematical model, investigation of a roller, due to localized defect and combined defects the succeeding findings may be predicted. With increment in defect size the amplitude of vibrations is increased and then decreases due to “self-Peening’ effect. And, with increment of rotational speed with localized defect the characteristics defect frequency also shifted on FFT diagram and the behavior of operation is change from periodic to chaotic. For individual localized defects on inner race, roller and outer race, the peaks appear at characteristics fault frequencies. The characteristic fault frequencies appear with interaction with case frequency and ball passage frequencies in combined localized faults.

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