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CONDITION MONITORING OF LOCALIZED OUTER RACE DEFECTS OF TAPER ROLLER BEARINGS USING VIBRATION ANALYSIS

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ABSTRACT
Defects in bearings may arise during use or during the manufacturing process such as crack damage, spalling, corrosion, fatigue failure, etc. The detection of the defects is important for condition monitoring as well as quality inspection of bearings. Vibration analysis is the most common method used in monitoring applications since a local defect produces successive impulses at every contact of defect. The aim of this paper is to focus on investigating the position of the local defect on the outer race is changed 0°, 45° and 180°. The different angular defect position is studied at zero clearance, $\varepsilon = 0.5$, by using the statistical parameters RMS. It can be concluded that the acceleration RMS ratio is more effective in detecting the defect for acceleration response.

KEYWORDS
Bearings, Faults, Vibration analysis, RMS.

INTRODUCTION
The defects and malfunction of mechanical components of one of the major challenges that causes many technical and economic problems for industrial systems. Gear faults typically occur in the teeth of a gear mechanism due to surface wear, fatigue, cracks, or pitting. Most gear Fault diagnosis and detection techniques which are based on signal response methods via acoustic emission transducers and vibration sensors have been widely investigated. A model-based gear fault detection method is preferable to identify the gear defects using transmission error. A parametric model of a gear model was established to estimate the transmission error, [1]–[3]. The belt drive experimental equipment was performed to obtain realistic vibration signals under different operating conditions. The pulley-belt system faults like unbalance, misalignment, and mis-cogs, that are dedicated by vibration analysis technique, [4]–[6]. Among the other mechanical components, researchers pay great attention to the rolling element bearings due to their unquestionable industrial importance to study

the behavior of vibration analysis for monitoring bearing health condition in practice, a test rig had been built, [7, 8].

The dynamic problem of a simple supported beam subjected to a constant force moving at a constant speed is discussed, [9]. The dynamics of beams on an elastic foundation and subjected to moving loads are studied by using the finite element method, where the foundation has been modeled by springs of variable stiffness, [10], [11]. High-speed-rotors have been simulated with the multi-body simulation software package MSC/ADAMS, [12 - 14]. The nonlinear dynamic behaviors of an unbalanced rotor system supported on ball bearings with Alford force are investigated, [15]. A numeric approach has been developed to estimate the effect of Hertzian and non-Hertzian contact parameters in ball bearings, [16 - 18]. A computing code was developed in Borland Delphi and Visual Fortran to study these cases. ABAQUS/Standard was used to examine the Hertzian contact stress in two-dimension models before extending the study to three-dimension models, [19]. A 3-D model of deep groove ball bearing was built by using APDL language embedded in the finite element software ANSYS. The obtained results have good consistency with that of the Hertzian theory, [20]. The effect of the displacement of inner race and the friction coefficient on fatigue, wear, penetration and generated Hertzian stress was studied, [21]. The contact stress of large diameter ball bearings was studied using analytical and numerical methods, [22].

Vibration analysis is among the most common methods used in the monitoring applications since a defect produces successive impulses at every contact of defect and the rolling element, and the housing structure is forced to vibrate at its natural modes. By using the estimated frequency, a simple notch filter removes the frequency component so that further detail in the vibration signal may be analyzed, [23], [24]. The synchronous averages are used to examine the calculation of the envelope signal of the high-frequency vibration produced by rolling element bearings with spalling damage, [25]. The effect of local defects on the nodal excitation functions is modeled. Simulated vibration signals are obtained, [26]. RMS values are obtained in the time domain and the high frequency resonance technique is used in the frequency domain. Both inner and outer race defects were artificially introduced to the bearing using electrical discharge machining, [27]. Time domain analysis, frequency domain analysis and spike energy analysis have been employed to identify different defects in bearings, [28]. The defect was detected using off the shelf portable vibration analysis hardware and software, [29]. A dynamic simulation method is proposed to study ball bearing with local defect based on the coupling of the piecewise function and the contact mechanism at the edge of the local defect, [30]. The finite element model of the spindle can predict the acceleration time responses due to the excitation, [31]. The external vibration has been imparted to the housing of the test bearing through electromechanical shaker, [32]. Therefore, spectrum analyses are conducted at specified test durations to predict defect locations, [33]. The study revealed that ultrasound technique is demonstrably superior to vibration acceleration measurements for detecting incipient defects in low-speed bearings, [34]. A simple time series method for bearing fault feature extraction using singular spectrum

analysis of the vibration signal is proposed, [35, 36]. A finite element contacts mechanics bearing model is established based on a contact algorithm suited to high-precision elastic, [37]. A method is presented for calculating and analyzing the quasi-static load distribution and varying stiffness of a radially loaded double row bearing with a raceway defect of varying depth, length, and surface roughness, [38]. The resonance frequency in the first vibration mode of mechanical system was studied, [39]. A mathematical model for the ball bearing vibrations due to defect on the bearing race has been developed, [40]. Finite element model can be effectively used to differentiate between vibration signatures for defects of different sizes in the bearing, [41]. The vibration response of healthy and defected bearing is compared [42]. The simulated vibration pattern has similar characteristics with results from experimental results, [43]. Mathematical expressions were derived for inner race, outer race and rolling element local defects [44]. The validity of the proposed model verified by comparison of frequency components of the system response with those obtained from experiments. The vibrations generated by deep groove ball bearings having multiple defects on races was studied, [45].

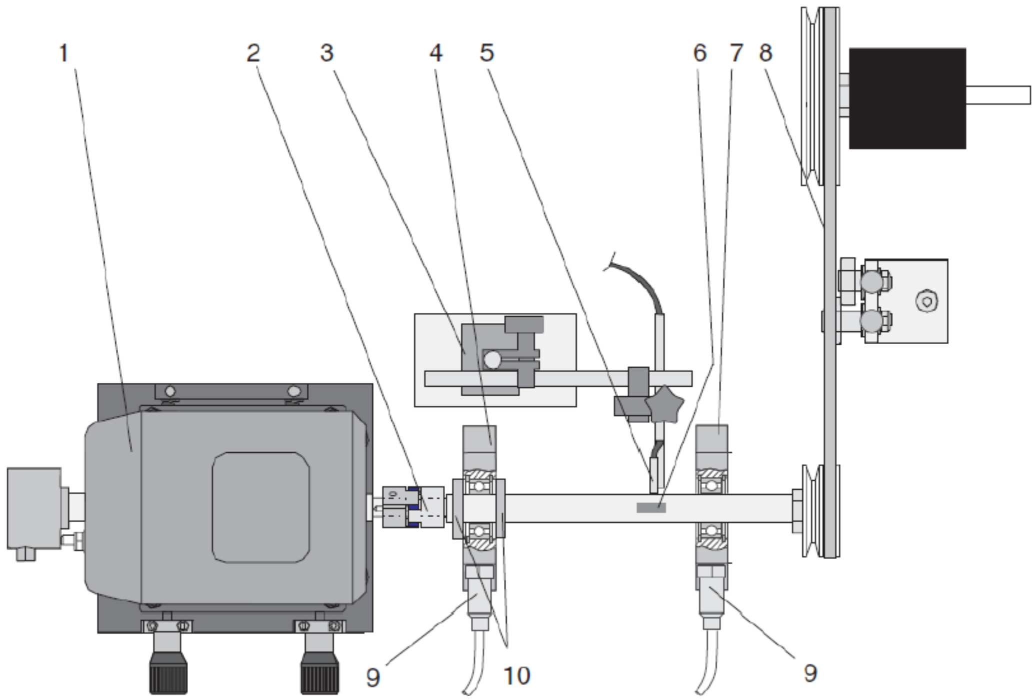
The main point of this work is to develop a procedure that can be used to design the condition monitoring of rolling element bearings by vibration measurements. Also, to study the effect of the size of the defects and their places on the vibration curve, which leads to the ease of identification of the defect and the possibility of maintenance at the right time.

EXPERIMENTAL

An experimental setup is employed in this research to collect the vibration signals to study the vibration signatures generated by incipient bearing defects. A sketch of the test rig and the equipment used in collecting the data shown in Fig. 1. The system is driven three-phase asynchronous motor with cage rotor, with the speed up to 6000 r/min. A control unit can control the shaft rotation speed. An optical encoder used for shaft speed measurement. Thus, the shaft rotation speed is known directly from the reading of this a control unit. An elastic claw coupling utilized to damp out the high-frequency vibration generated by the motor. Two ball bearings fitted into the solid housings. Accelerometers (IMI Sensors- 603C01) mounted on the housing of the tested bearing to measure the vibration signals along two directions. A variable load applied by a belt drive. A data acquisition card (BMC USB-AD16F) employed for signal collection.

To study the effect of the size of the defects and their places on the vibration curve, which leads to the ease of identification of the defect and the possibility of maintenance at the right time. The defects were created at the outer race, as illustrated in Figure 2, with various widths of 0.1, 0.2, 0.3, and 0.4 mm. While the position of the local defect on the outer race is changed from 0˚ to 45˚ and 180 ˚. The different angular defect position is studied at zero clearance, ε = 0.5, by using the ratios RMS. The analytical results describe the alteration of the bearing acceleration, for vertical direction. Figure 3 gives the defect positions and load distribution with no clearance condition. All the bearings were operated for 10 min at the same speed from

1000 rpm to 3000 rpm ,500 rpm step for each and under the constant radial load of 100 N, 200 N and 300 N with three deferent defect positions 0° to 45° and 180 °.



1. Drive unit
2. Elastic claw coupling.
3. Magnetic clamp with steel plate.
4. Bearing block.
5. Reference sensor.
6. Short shaft with reflective mark.
7. Bearing block.
8. Belt drive.
9. Acceleration sensor.
10. Setting ring.

Fig. 1 Schematic representation of the test rig.

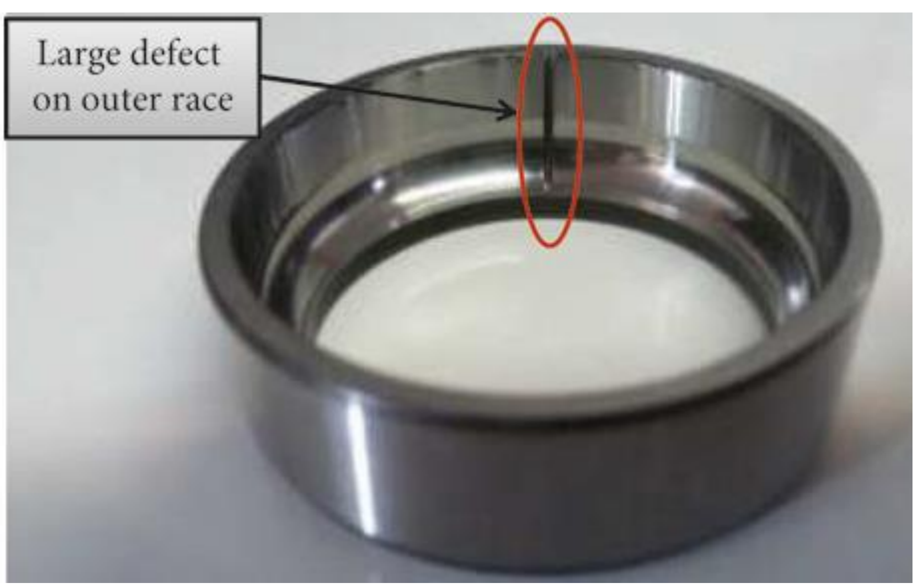


Fig. 2 Defects located in outer race of the bearing.

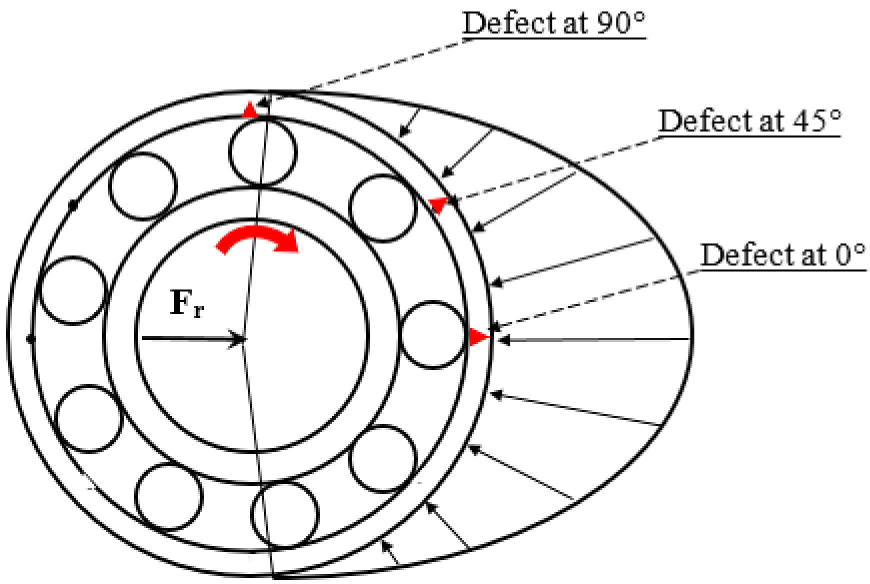


Fig. 3 Positions of localized defects, zero clearance, ε = 0.5

RESULTS AND DISCUSSION

1.1 The effect of the size of defect and the load on the bearing

In this section the position of the local defect on the outer race is changed from 0˚ to 45˚ and 180 ˚. The different angular defect position is studied at zero clearance, ε = 0.5, by using the ratios RMS. The analytical results describe the alteration of the bearing acceleration. The vibration data was carried out under a shaft rotational speed from 1000 rpm to 3000 rpm, and 100 N load. For vertical direction. It can be seen form Figures 4-7 that the RMS-ratio "RMS" at any point, in the vertical direction; give high result with the defect located at 0˚. While the low result is given with the defect located at 180˚, The error that occurs is due to different installation, especially at high speeds. Moreover, it can be adopted that the un-defected bearing represents the lowest value of RMS under different operating conditions. It can be pointed out that defects in the loading area are easy to detect, as they are clear values to be noticed. While this advantage is reduced for defects located outside the loading area. This may be attributed to an impulse that occurs when a rolling element passes and strikes the fault position at loaded area is significant than that occurs when a rolling element passes and strikes the fault position at un-loaded area.

1.2 The effect of the size of defect and rotational speed

In this section the position of the local defect on the outer race is changed from 0˚ to 45˚ and 180 ˚. The different angular defect position is studied at zero clearance, ε = 0.5, by using the ratios RMS. The analytical results describe the alteration of the bearing acceleration. The vibration data was carried out under load from 100 N to 300 N, and a shaft rotational speed of 1000 rpm. For vertical direction. It can be seen form Figures 8-11 that the RMS-ratio "RMS" at any point, in the vertical direction; give high result with the defect located at 0˚. While the low result is given with the

defect located at 180°, the error that occurs is due to different installation, especially at high speeds.

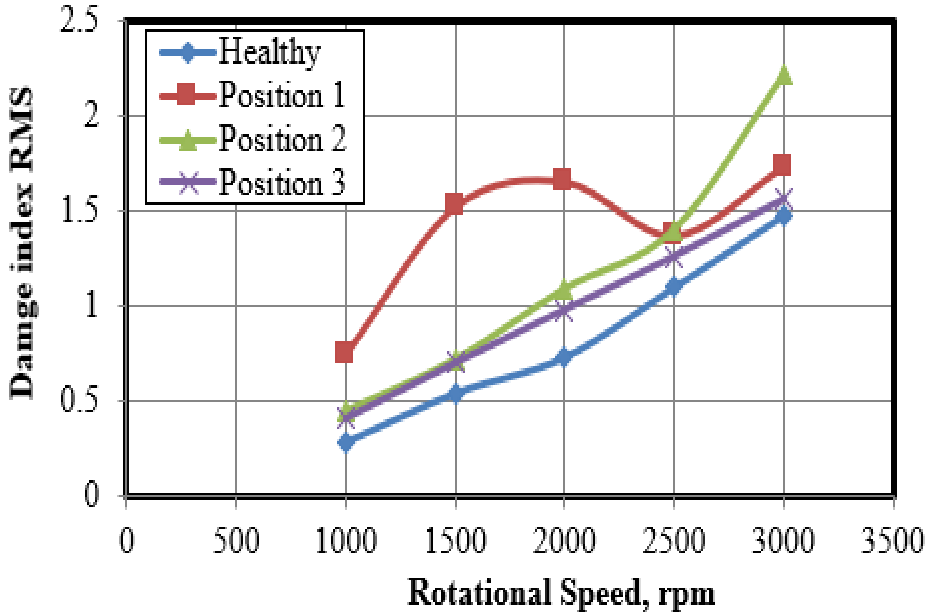


Fig. 4 RMS index for different positions of defect size of 0.1 mm versus rotational speeds.

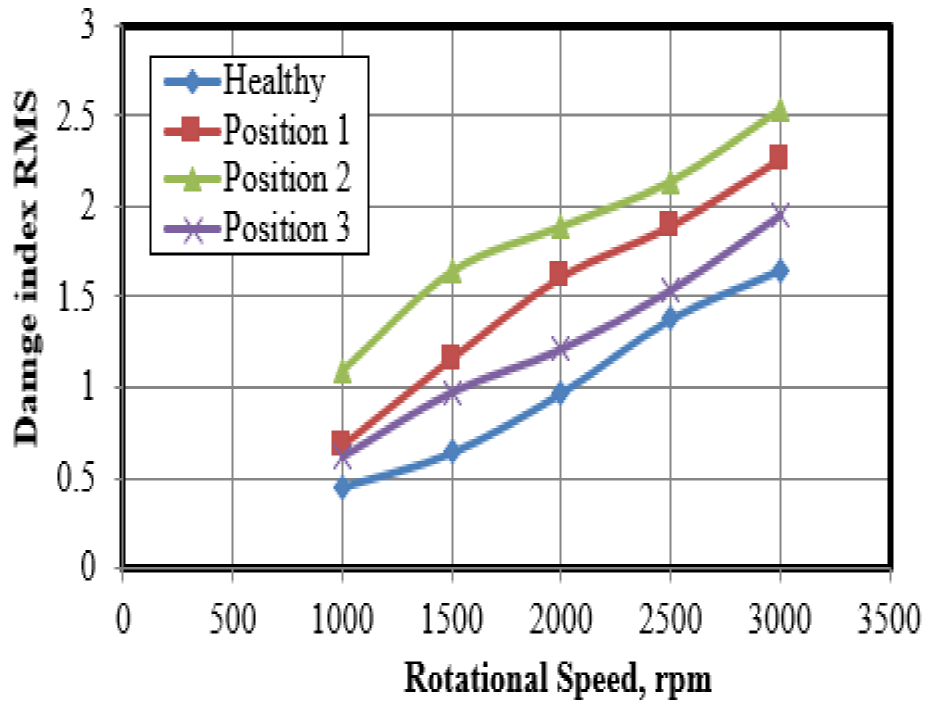


Fig. 5 RMS index for different positions of defect size of 0.2 mm versus rotational speeds.

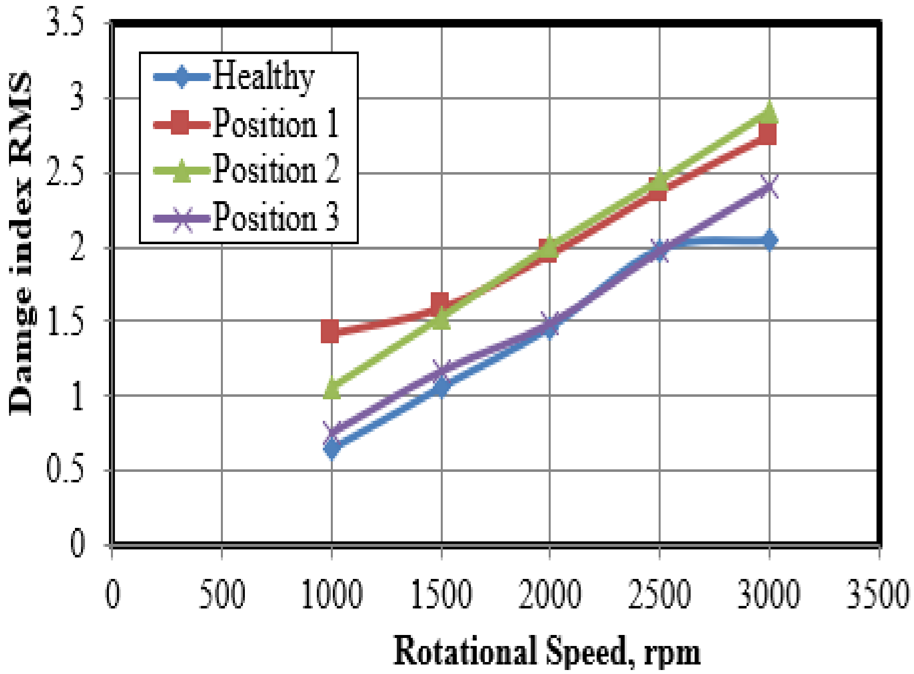


Fig. 6 RMS index for different positions of defect size of 0.3 mm versus rotational speeds.

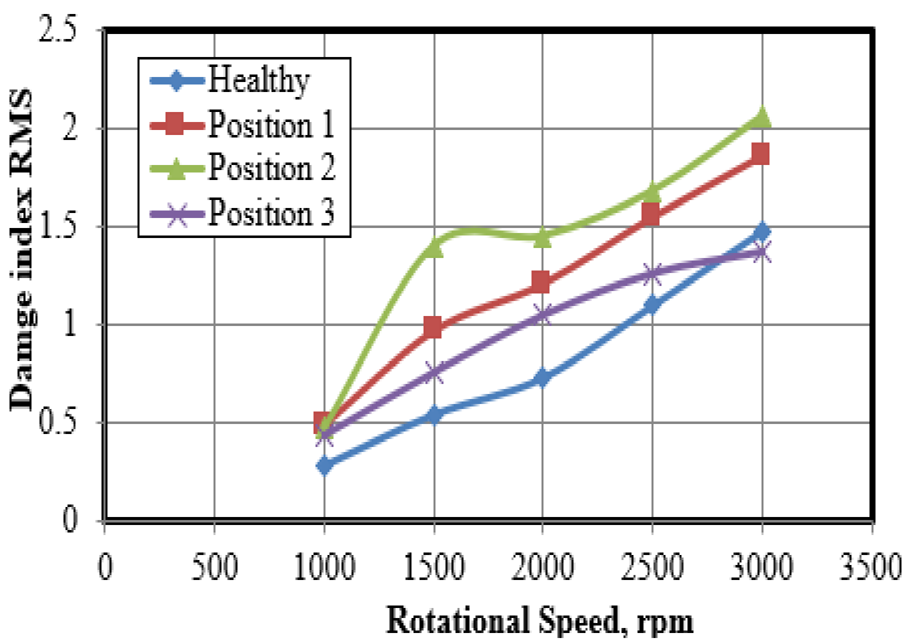


Fig. 7 RMS index for different positions of defect size of 0.4 mm versus rotational speeds.

The statistical parameters for the acceleration responses in vertical direction are performed. It can be seen form that the RMS parameter at vertical direction give high result with the defect located at 90˚. Furthermore, a local defect at 180˚ give the low

result and the defects locate at 45˚ give the same result. It can be concluded that the horizontal is a good sensor location to detect a local defect at 0˚ according to the RMS result. And the vertical is a good sensor location to detect a local defect at 90˚. The defects located at the radial load distribution area have more effect than the defects located at the unloaded area.

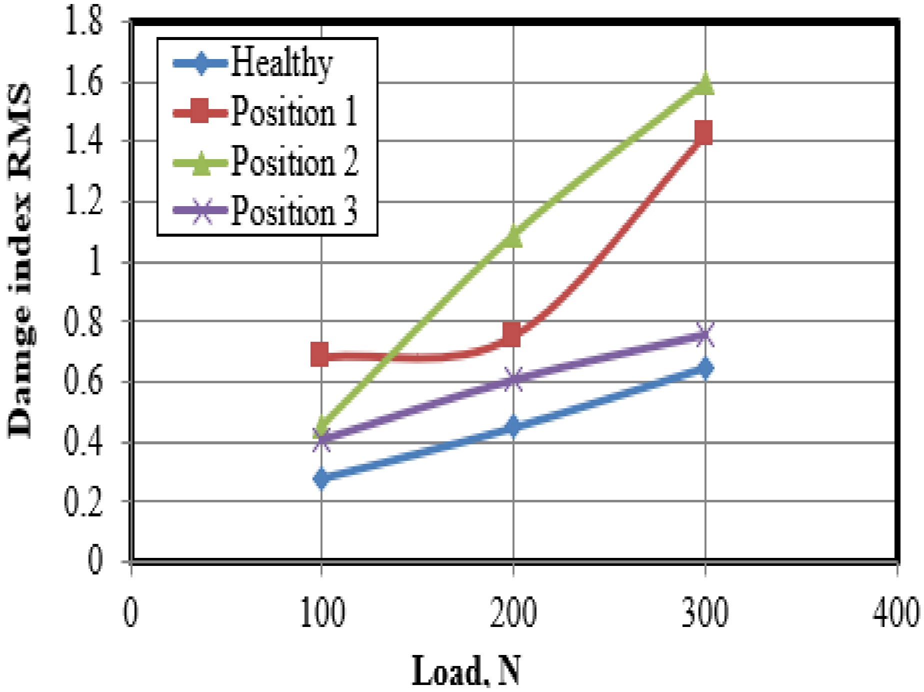


Fig. 8 RMS index for different positions of defect size of 0.1 mm versus radial loads.

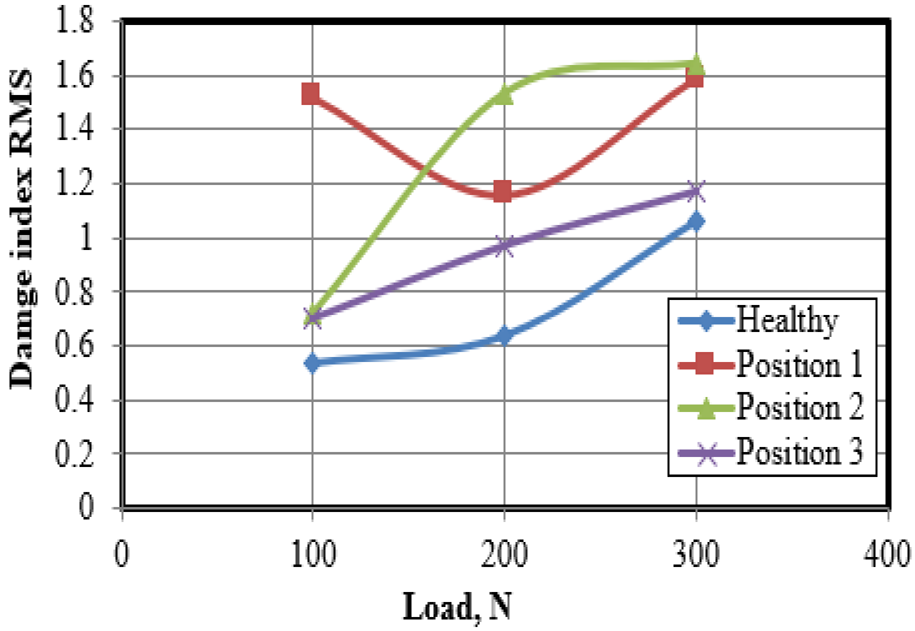


Fig. 9 RMS index for different positions of defect size of 0.2 mm versus radial loads.

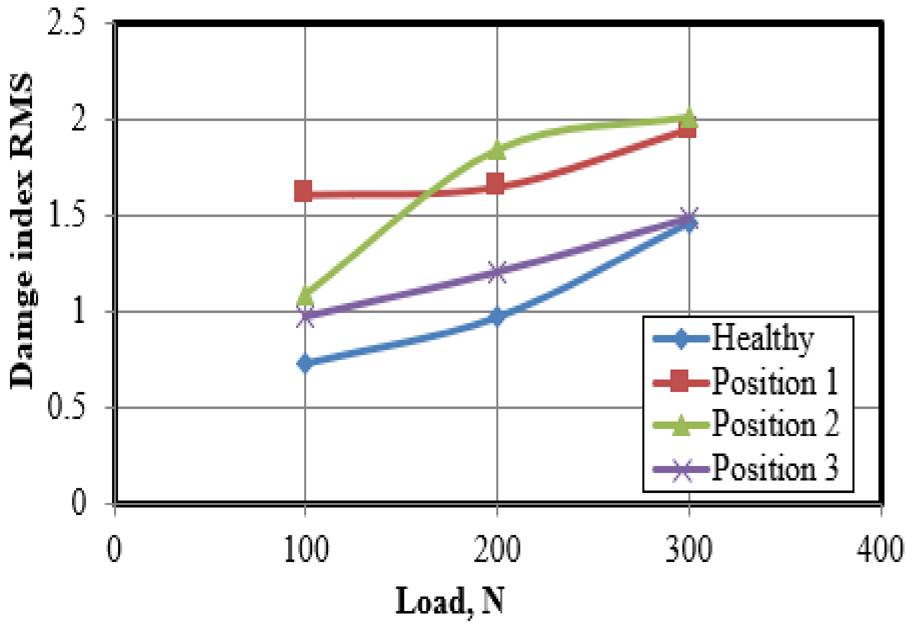


Fig. 10 RMS index for different positions of defect size of 0.3 mm versus radial loads.

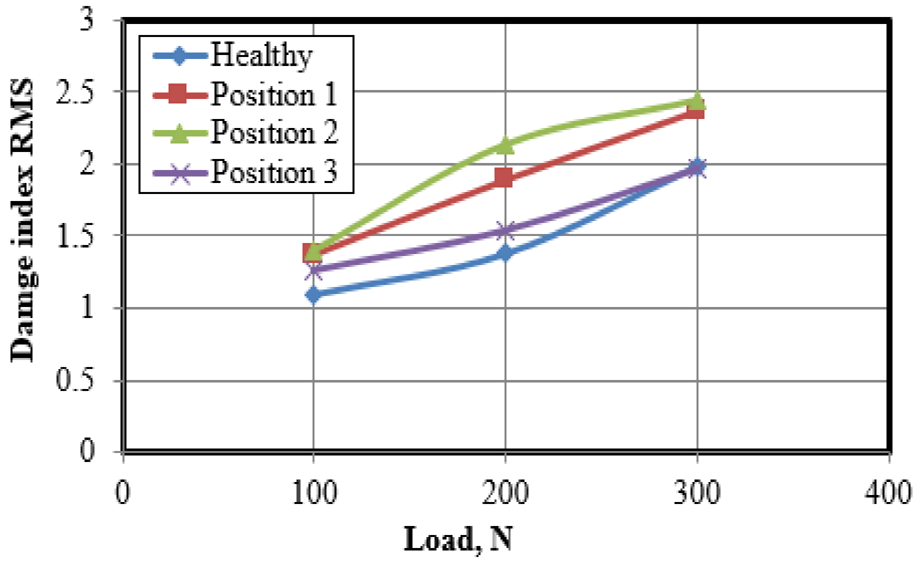


Fig. 11 RMS index for different positions of defect size of 0.4 mm versus radial loads.

CONCLUSIONS

The experiments work studies multi defects in taper roller bearings, the following conclusions can be drawn:

- **The experiments show how much the value of the damage index has changed due to changes in the value of load and speed.**
- **The RMS-ratio "RMS" at any point, in the vertical direction; give high result with the defect located at 0˚. While the low result is given with the defect located at 180˚.**

- **An error occurs due to different installation especially at high speeds and loads.**
- **It's became possible to estimate the defect size in the outer race of taper roller bearing (32004) using the experiment results.**

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