The importance of bearing stiffness and load when estimating the size of a defect in a rolling element bearing

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ABSTRACT

The change in the static stiffness of a bearing assembly is an important discriminator when determining the size of a defect in a rolling element bearing. In this paper, the force-displacement relationships for defective bearings under various static radial loadings at various cage angular positions are analytically estimated and experimentally measured and analyzed. The study shows that the applied load has a significant effect on the static stiffness variations in defective rolling element bearings. The experimental measurements of the effect of the defect size on the varying stiffness of the bearing assembly, which has not been shown previously, provides valuable knowledge for developing methods to distinguish between defective bearings with defects that are smaller or larger than one angular ball spacing. The methods and results presented here contribute to the wider experimental investigation of the effects of loadings on the varying static stiffness of defective bearings and its effects on the measured vibration signatures. A large data set was obtained and has been made publicly available.

Keywords: rolling element bearing, varying stiffness, defective bearing, defect size estimation, bearing spall
1. Introduction

Defects in bearings are commonly categorized into distributed and localized defects. Distributed defects encompass the whole contact surface and are usually the result of manufacturing errors; such as waviness, surface roughness or rolling elements with different diameters, \(^1\), \(^2\). Localized defects do not cover the entire contact surface, but occur during the operational lifespan of the bearing. The most common reason for the failure of rolling element bearings are the formation of surface defects during operation, forming spalls, dents and pitting defects. Spall defects are the most common defects that occur and are initiated from surface or sub-surface cracks that form in bearing components; and these cracks grow eventually causing a large flake of material to be removed from the contact surface in a bearing assembly over time. These cracks occur due to insufficient lubrication and high contact stresses between the rolling element and the raceways. While dent defects are caused by contaminates being pushed into the contact surfaces or high impulsive loading of the bearing plastically deforming the contact surface, pitting defects are caused by adhesion forces that cause surface grains to be removed from the contact surfaces \(^3\). This paper considers square shape defects of various dimensions on the outer-raceway. These defects influence the static force-displacement relationships of the bearing assembly and the resulting vibration signature. Defects smaller than the angular spacing of rolling elements are referred to as line-sparl defects while larger defects, with multiple rolling elements in the defect zone, are referred to as extended defects.

When estimating the defect size of an extended defect, there is an aliasing issue that occurs as the vibration response of the extended defect is similar to that of a smaller line spall defect. It was analytically shown by Petersen et al. \(^4\), \(^5\), that the change in the stiffness of the bearing assembly as a rolling element enters and exits a defect can be used to distinguished between a line spall and an extended defect as the change in stiffness causes the natural frequencies of the bearing to change. It was found that the stiffness for the bearing assembly is less for
a bearing with an extended spall, as compared with a line spall defect, as the rolling elements enter and exit the defect area. This characteristic was not experimentally verified by Petersen et al.\textsuperscript{4, 5}, and is one of the new contributions of this paper. It is shown in this paper that the applied load alters the static stiffness in defective bearings, and must be included to determine an accurate estimate of the defect size.

Conventional severity assessment methods in vibration condition monitoring consider the measured vibration levels of a bearing housing over time. A bearing is classed as faulty when the level exceeds a nominated threshold. Alternatively, it is possible to estimate the physical size of a defect by using the measured vibration signal, without using historical data, and is the focus of the work in this paper.

Previous studies on the vibration responses of a bearing with a spall defect on both the inner and outer raceways show that when a rolling element travels through a spall defect, it generates two distinct vibration signatures when the rolling element enters and exits the spall defect\textsuperscript{4, 6-8}, as shown in Figure 1. As a rolling element enters a defect and begins to unload, a low-frequency vibration is generated (Events 1 to 2). Similarly, when a roller exits the defect and becomes reloaded by the inner and outer rings, a low frequency vibration is generated (Event 4). Just before a rolling element exits the defect, it will typically strike the rings and generate a high-frequency impact vibration signature that excites the bearing’s assembly resonance frequencies (Event 3)\textsuperscript{7, 9-13}. The process of re-loading the rolling element between the raceways causes parametric excitations due to rapid changes in the bearing stiffness\textsuperscript{5}. From the analytical work presented by Petersen et al.\textsuperscript{4, 5}, the frequency of the oscillation when the rolling element becomes unloaded and when the rolling element begins to reload (Events 2 and 4) will change for an extended defect when compared to the frequency response of a bearing with a line spall defect.
Several defect size estimation methods have been suggested previously for bearings with line spall-defects based on detecting the time separation between entry and exit events from the vibration signal \(^7,^9\). These previous defect size estimation methods underestimate the size of a defect that is larger than, at least, one ball angular spacing. The possibility of using stiffness variations in defective bearings as a measure to distinguish between extended defects and line-spall defects was suggested by Petersen et al. \(^4,^5\), but there was no further investigation. They presented an analytical formulation of the varying static stiffness and load distribution of a ball bearing assembly as a function of the cage angular position. Petersen et al. \(^4,^5\) analyzed the static stiffness variation in defective bearings with rectangular shaped outer raceway defects of varying circumferential extent and similar depths without experimental validation. In the work conducted here, the effects of the applied load on the static stiffness in defective bearings, which was ignored previously, are investigated and the importance of including the effect of load in developing a reliable defect size estimation method is demonstrated.
The total deformation in rolling element bearings is composed of the deformation of the local rolling elements in the load zone between the inner and outer raceways and the global deformation of the inner and outer raceways. The stiffness of the bearing depends on the angular position of the rolling elements, which changes as the rolling elements move around the bearing. Consequently, the bearing stiffness varies with the number of load carrying rolling elements in the load zone, thus the stiffness of the bearing is a function of cage position and load. The static stiffness of a healthy ball bearing has been studied analytically, numerically, and experimentally by researchers. However, the change in the static stiffness of a defective bearing as a function of cage angular position has only been studied analytically and numerically by Petersen et al. without experimental validation and analysis, and is addressed in the work presented here. A bearing test rig, shown in Figure 2, was used to measure the vibration of defective bearings. The vibration of the “floating” bearing housing was measured using accelerometers, and the change in the relative displacements between the inner and outer raceways in the vertical and horizontal directions were measured using eddy current proximity probes. Deep-groove ball bearings with line-spall and extended defects machined using electro-discharged-machining (EDM) were installed on the test rig and the force-displacement curves were obtained by measuring the relative displacement signals for a range of loads. Static stiffness curves were obtained and analyzed for two cage positions. The vibration responses of the two defective bearings were measured and a new method is proposed that can be used to distinguish the two defect sizes. It will be shown by experimental and analytical analyses that the applied load alters the low-frequency response, and can be used as an indicator to identify extended defects.

The main contributions of this paper are (1) an experimental measurement and analysis of the varying static stiffness in defective bearings with various defect lengths and loads, which validates previous analytical formulation of the static varying stiffness and load distribution of a ball bearing assembly as a function of cage angular position, (2) an analysis of the load on the varying static stiffness in bearings with extended defects which shows the effects of
the applied load on the static stiffness in defective bearings, which was ignored previously, (3) an improved approach to distinguishing extended defects and line spalls using the low frequency variation in stiffness.

This paper is structured as follows: Section 2 describes the test rig and the measurement method used in the experiments; Section 3 presents the analytical and experimental outcomes, and detailed analysis of the static stiffness in defective bearings; Section 4 presents vibration measurements of the defective bearings and analysis of the characteristic frequency variation and Section 5 summarizes the findings of this research.

2. Test equipment and measurements

Figure 2 shows the test-rig used in this study, which is a modified Spectra Quest Inc. bearing simulator. The test bearings were fitted into a custom-made bearing housing at the end of the shaft. The radial load is applied by a hydraulic jack to the custom-made bearing housing. An electric motor, controlled by a variable frequency drive, is fitted onto the drive-end of the shaft and is supported by two back-to-back tapered roller bearings.

Figure 2: Top view of the modified Spectra Quest bearing simulator used in this study.
Figure 3, shows the schematic of the custom-made bearing housing and the loading mechanism. The housing accommodates accelerometers (Bruel & Kjaer type 4393) to measure vibration and eddy current proximity probes (Micro-Epsilon type EPU05-C3) to measure the relative displacement between the inner and outer raceways of the test bearing. The shaft speed was measured using a tachometer, and the applied load to the test bearing is measured using a load cell. A National Instruments (NI) CompactDAQ system with two NI 9234 modules were used to acquire 10 seconds of data and the signal was sampled at a frequency of 25.6 kHz.

Ball bearings (Rexnord ER16K) were tested that have 9 balls, ball diameter of 7.94 mm, a pitch diameter of 39.32 mm, and a contact angle of 0°. Defective bearings were prepared by electric discharge machining rectangular shape defects on the outer raceway with various circumferential lengths across the full axial extent of the outer raceway. The defects vary in size by exactly one angular ball spacing. Table 1 lists the bearings and the geometry of defects tested in this study, and Figure 4 shows photographs of the defects examined in this study.
Table 1: Test bearings and geometry of defects.

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<th>Test bearing</th>
<th>Defect size</th>
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<td>Angular extent (degrees)</td>
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<tr>
<td>TB2</td>
<td>55.8</td>
</tr>
<tr>
<td>DFB</td>
<td>Defect-free bearing</td>
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Figure 4: Photographs of the machined defects used in the experiments, where (a) is the defect with an angular extent of 15.8° and (b) is the defect with an angular extent of 55.8°.

Each defective bearing was installed into the bearing housing such that the center of the defect was aligned with the direction of the horizontal radial load. Two test positions were chosen such that the minimum or the maximum possible number of rolling elements were located in the load zone. The position of the rolling elements in each test case with regards to the location and the angular extents of the defects are shown in Figure 5. At Test Position 1, (TP1), the loading axis passes through one rolling element and at Test Position 2, (TP2), the loading axis passes in between two rolling elements. The cage angular position for each test position differs by 20° (360° / 9 balls / 2 = 20°).
Figure 5: Schematic view of the positions of the rolling element for the two test position, where the cage angular position for each test position differs by 20°. The dark and light grey sectors show the 15.8° and 55.8° angular extents for Test Bearings 1 and 2, respectively. Test position 1 shows one rolling element inside the defect for both test bearings. Test position 2 shows zero rolling elements inside the defect for Test Bearing 1 and two rolling elements inside the defect for Test Bearing 2.

The testing procedure and the setup are designed to eliminate potential errors in measuring the force-displacement relationship caused by the geometry imperfections of the inner raceway or the shaft. A narrow machined surface, in a non-load bearing location, was placed on the inner rings of all the test bearings to provide a reference measurement surface for the eddy current proximity probes, as shown in Figure 6. Consequently the relative distance measurements between the inner and the outer raceways do not contain measurement variations due to out-of-roundness, or surface faults pre-existing in the inner ring, which can occur during manufacturing processes when the bearing components are made and assembled.
Figure 6: A Rexnord ER16K bearing assembly showing the machined surface on the inner ring that provides a reference surface for the eddy current proximity probes.

3. **Static stiffness analyses**

3.1 **Experimental analysis**

Analyses of the static measured signals using the test rig are presented in this section. The static measurements were conducted on the test bearings by applying various static loads at the two test positions described in the previous section. The aim was to determine the load-deflection factor for the analytical analyses and to verify the model by measuring the relative displacement between the inner and outer raceway as a function of the applied static load to provide insights into the static stiffness behavior of the test bearings as the defect grew in size. Figure 7 shows the obtained force-displacement relationships for the three test bearings: TB1, TB2, and the defect free bearing.

![Figure 7: Force-displacement relationship for TB1, TB2 and the defect free bearing, at the two Test positions 1 (TP1) and 2 (TP2), as described in Section 2.](image)

The slope of each force-displacement curve in Figure 7 represents the static stiffness of the bearing assembly at a particular test position. The slopes for TB2 are lower at both test positions when compared with the results from the DFB case. From Figure 7 it is shown that the bearing assembly stiffness of TB2 at TP2 changes non-linearly as the load increases. Studying the force-displacement relationship of TB2 at TP2 is particularly important as a
load of 1500 N causes the rolling elements in the defect to contact the inner and outer rings and the slope of the force-displacement curve dramatically increases.

The experimental force-displacement results can be explained as following:

- The force-displacement relationships for each tested bearing vary for the two test positions, TP1 and TP2. The variation is due to the angular position of the cage in each test bearing, and is the smallest for the defect-free case, DFB, and the largest for the bearing with the extended defect, TB2. These variations are due to the difference in the number of rolling elements carrying the static load and the magnitude of the applied static load.

- The resemblance between the force-displacement relationships for TB1 at TP2 and the DFB at TP2, suggests similar non-linear static stiffness at TP2 for both defective test bearings. Therefore at TP2, the applied load and the defect size have no effect on the static stiffness, as there is no ball in the defect area in TB1.

- The force-displacement relationships for TB1 only deviate from the static stiffness of the DFB at TP1 when a rolling element is positioned inside the defect.

- Force-displacement relationships for TB2 at TP1 and TP2 deviate entirely from the force-displacements for the DFB as at least one rolling element is positioned inside the defect at any given angular position of the cage. Therefore, a lower static stiffness for the extended defect, TB2, is expected at any cage angular position when compared with a healthy bearing.

- Force-displacement relationship curves for TB1 and TB2 are similar at TP1, whereas the force-displacement relationship curves vary for TP2. This suggests that the dynamic stiffness of the two defective bearings are not expected to be similar.
The results from the force-displacement tests confirm that the static stiffness does vary with the applied load on the bearing and angular position of the rolling elements, which is aligned with the Petersen et al. hypothesis that was based on theoretical predictions. It was found that the slope of the force-displacement curve for TB2 at TP2 will greatly increase when the applied load is greater than 1500 N to a similar slope to that of the DFB and TB1. Therefore, the change in the natural frequency of the bearing assembly between TB1 and TB2 may be too small at high load to determine if the defect is an extended spall defect, this is investigated in the later sections of this paper.

3.2 Analytical analysis

In this section, the analytical formulation presented by Petersen et al. is implemented to study the effects of the circumferential extent of a defect on the static stiffness under varying loads, and the number of the load carrying rolling elements in the defect zone for the test cases introduced in Section 2. In the work presented here, the effect of the circumferential extent of a defect on the static stiffness of a bearing is examined for a range of applied loads, which differs from the previous work by Petersen et al. that only examined one static load.

The parameter value for the load–deflection factor for the Hertzian contacts used in the model is $7 \times 10^9$ N/m$^{1.5}$ for TB1 and the DFB, which was calculated using the method found in Harris. Since the model presented by Petersen et al. is unable to accurately model the sudden increase in the bearing stiffness from TP2 to TP1 for TB2, the model was adapted by using three parallel non-linear springs that approximated the force-displacement curves of TB2, as shown in Figure 7. The bearings were subjected to static loads ranging from 0 N to 3000 N in the horizontal direction. The analytical formulation of the load distribution and the varying stiffness of a ball bearing assembly presented by Petersen et al. does not include the finite size of the rolling elements. Therefore, the path of rolling elements has to be modified to account for the gradual de-stressing and re-stressing of a rolling element at the entry and exit points of the defect. Figure 8 presents the defect depth profiles for rectangular shaped defects of 100 μm and circumferential extents of 15.8° and 55.8°, the
actual and modelled defect depth profiles are indicated by the solid and dashed lines, respectively.

Figure 8: Defect profiles and the modeled rolling element paths for the square shaped defects with a depth of 100 μm and a circumferential extent of 15.8° (black) and 55.8° (gray); (dashed line) actual defect depth profile; (solid line) modeled rolling element path profile, graph taken from Petersen et al.

Figure 9 (a) and (b) shows the variation of the estimated number of balls that carry the static load as a function of the applied load and cage angular position for TB1 and TB2, respectively. As a result of the applied load and the number of balls supporting the load, the corresponding displacement of the outer raceway relative to the inner raceway will also vary. Knowing the force-displacement relationship, the equivalent bearing assembly stiffness can be determined, as shown in Figure 9 (c) and (d) for TB1 and TB2, respectively.
Figure 9: Contour plots showing the number of balls carrying load and radial bearing stiffness as a function of applied load and cage angular position with a rectangular shaped defect on the outer raceway. (a) and (b) show the number of balls carrying the static load for 15.8° (TB1) and 55.8° (TB2) defects respectively, and plots (c) and (d) show the corresponding radial bearing stiffness for 15.8° and 55.8° defects respectively.

The stiffness and the number of rolling elements carrying the static load vary periodically for both defective bearings, with the fundamental period defined by the rolling element angular spacing of 40°. The number of load carrying rolling elements and the varying stiffness as a function of the applied load is similar for both TB1 and TB2 at the cage angular position of 20°. This similarity is also valid for the cage angular positions neighboring $\varphi_c = 20°$ at which only one rolling element is positioned in the defect.

Compared with the defect with the smaller circumferential extent, the stiffness and the number of load carrying rolling elements in the extended defect (TP2) suddenly increases at the cage angular position 0° above a load of 500 N, as highlighted in Figure 9(b) and (d).
the load applied to the bearing increases, eventually the displacement of the outer raceway causes the rolling elements within the defect zone that were unloaded with an applied load under 500 N, to come into contact with the raceways again and start to become loaded. This load is called the “critical-load” in this study. When the applied load is above 1500 N all the rolling elements in the load zone for TB2 become loaded, as shown in the force-displacement curve in Figure 7.

At loads less than the critical-load, when the circumferential extent of the defect is greater than the ball angular spacing, as is the case of TB2, one or more balls are positioned in the defect at any one time. Therefore, the stiffness is always equal to or less than the stiffness of the bearing with a smaller circumferential defect, as is the case of TB1, depending on the cage angular position. When the rolling elements positioned in the defect start to take load at loads higher than the critical-load, the varying stiffness of the bearing with extended circumferential defect increases dramatically and becomes closer to the stiffness values of the case with smaller defects.

Since the presence of the extended defect reduces the number of loaded balls compared with the bearing with smaller defects, the balls positioned outside the defect zone in the load zone carry more load. Hence, the local relative contact deformations of the load carrying rolling elements in the larger defect are greater than the smaller defects. Consequently, if the same applied load is applied to a bearing with an extended defect and another with a small defect, the relative displacement between the inner and outer rings is greater in the case of the extended defect, when compared with smaller defects. This force-displacement relationship can also be explained by the fact that bearings with an extended defect have a lower static stiffness at loads under the critical-load, when compared with bearings with smaller defects, as seen in Figure 9. Therefore, this change in the bearing assembly stiffness is not only due to the size of the defect but also due to the applied load. These analyses show the importance of determining the critical load when the characteristic frequency changes in
the bearing are being monitored. The analytical model presented in this paper has been made publicly available at FigShare 16.

4. Dynamic analyses

4.1 Vibration measurement

The bearing test rig, shown in Figure 2, was used to measure the vibration response of two bearings with manufactured rectangular defects TB1 and TB2, which vary in size by exactly one angular ball spacing. Each bearing was tested with an applied radial load of 500 N and 3000 N at a shaft speed of 600 rpm. The vibration response in the horizontal direction (the direction of the loading axis) is measured using an accelerometer mounted on the bearing housing. Figure 10 shows the experimental vibration results in the horizontal direction as a function of the cage angular position for the two defective bearings. The figure shows the sizes (in degrees) of the defects on each test bearing. It can be seen that the angular extents between any two successive entry and exit events appear nearly identical for both TB1 and TB2 despite TB1 having a defect size of 15.8° and TB2 having a defect size that is 40° larger at 55.8°. This is because the balls will enter and exit the defects in TB1 and TB2 with the same period as defects that vary in size by exactly one angular ball spacing. Hence, the vibration signatures appear identical for the small and large defect, and therefore, current defect size estimation methods are unable to accurately estimate the size of the larger defect in bearing TB2. A method is shown in this paper how to differentiate a line spall from an extended spall defect by using knowledge of the dynamic stiffness and measurement of the resonance frequencies of a bearing assembly to differentiate the size of the defects.
Figure 10: Experimentally measured acceleration in the horizontal x-direction (see Figure 3) of bearings TB1 (15.8° defect size) and TB2 (55.8° defect size) as a function of angular cage position at 500 N of load and a shaft speed of 600 rpm. The corresponding sizes of the defects are indicated on the figures.

4.2 Detection of dynamic stiffness variations

In this section, the effects of the stiffness variation due to defect size variation and static load variation on the measured vibration responses are investigated. The purpose of these experimental analyses is to justify the hypothesis that the low-frequency variations of the entry and exit vibration events can be used as a feature to distinguish defects that vary in size by exactly one angular ball spacing. Test bearings are loaded with a 500 N and 3000 N load to investigate the effect of load. These loads were selected to be smaller and greater than the critical load estimated in section 0 and experimentally shown in section 3.1.

The low-frequency vibration events that occur when a rolling element enters and exits a defect, as discussed in Section 1, are due to the stiffness changes of the bearing assembly. It has been shown in this paper, both analytically and experimentally, that the stiffness of the bearing assembly is noticeably different when comparing line spall and extended defects. The change in the bearing assembly stiffness can be detected by inspecting the characteristic frequency of the low-frequency events. This can be achieved by using time-frequency signal-processing techniques, as shown below.
Firstly, peaks of the entry and exit transient events are detected on the measured vibration data. Then, the entry and exit events are separated and individually captured using a window with the length of 1155 samples that starts 155 samples before a detected event’s peak. These captured events are then ensemble averaged together, and the spectrograms of the resultant ensemble averaged signals created using a Discrete Gabor Transform (DGT)\(^\text{17, 18}\). A DGT is a Short Time Fourier Transform (STFT) with a Gaussian window. Other windowing techniques can be used in the STFT, such as a Hanning or Hamming window, and will produce similar results when compared to the DGT. However, for the purpose of this paper a Gaussian window was chosen. Figure 11 (a) to (d) and (m) to (p) show the 20 ms of the 30 events captured from the measured vibration signals of the test bearings, TB1 and TB2, under 500 N and 3000 N static loads. Figures 12(e) to (h) and (q) to (t) show the ensemble averages of the related captured signals and their standard deviation (SD). Figures 12 (i) to (l) and (u) to (x) show the spectrograms of the resultant ensemble averaged signals. The vibration energy densities on the DGT spectrograms for each event are normalized.
Figure 11: Comparison of the acceleration results in the horizontal x-direction (see Figure 3) and their spectrogram using the Discrete Gabor Transform (DGT) for TB1 and TB2, under 500 N and 3000 N loads. Where, (a) to (d) and (m) to (p) show 20 ms the captured 30 entry and exit events, (e) to (h) and (q) to (t) show the ensemble average and its standard deviation for the captured events, (i) to (l) and (u) to (x) show the spectrograms for the calculated entry and exit mean signals.

The observed low-frequency entry and exit events shown in Figure 11, for both the acceleration results and the spectrograms, are caused by varying stiffness excitations of the rigid body modes that occur when a ball enters or exits the defect (also called parametric excitation) 17. Some comments about the spectrograms in Figure 11 are as follows:

The characteristic frequencies of entry events appearing on the spectrograms are generally lower than the characteristic frequency of the exit events for every individual test bearing. These results are consistent with the theoretical analysis and experimental measurements of static stiffness of the test bearings presented in Sections 3.1 and 0.

The difference between the characteristic frequencies of the entry events in TB1 compared to TB2 under varying loads, are that the characteristic frequencies are higher in the case of TB1 (compare Figure 11 (i) and (k) to (u) and (w)). This phenomenon can be explained by the fact that when the rolling elements of TB2 enter the defect with an applied load of less than 500 N they do not carry any load in the defect area, whereas the same rolling elements will carry some load when entering the defect zone when the applied load is greater than 500 N. Hence, TB1 will be stiffer at higher loads than TB2. Under similar loading conditions, the dominant characteristic frequencies of the entry and exit events are higher in the case of TB1 when compared to TB2, as shown when compared to Figure 11 (i) with (u), (j) with (v), (k) with (w) and (l) with (x). However, the entry events of TB1 and TB2 under 500 N have the largest difference between the dominant characteristic frequencies among the rest, as shown in Figure 11 (i) and (u).

It has been shown in Sections 3.1 and 0 that the static stiffness of the bearing assembly with the larger defect TB2 at position 2 is lower for applied loads less than the critical-load
when compared with the static stiffness of the rigid body modes of TB1. As a result, the rigid body modes of the test bearing have lower dominant characteristic frequencies.

The dominant characteristic frequencies of the entry and exit events can be confirmed using the acceleration response as shown in Figure 12, where (a) is the ensemble average of the entry event and (b) is the ensemble average of the exit of TB1 at a shaft speed of 600 rpm with an applied load 500 N. Figure 12 (a), shows that the time taken for the rolling element to unload as it enters the defect is approximately 1.8 ms and the period of the oscillation once the rolling element has completely unloaded is approximately 0.9 ms, which equates to approximate frequencies of 550 Hz and 1100 Hz, respectively. These frequencies are the dominant frequencies in the entry event, as highlighted in Figure 11 (i). When this method is repeated for the exit event the dominant frequency was found to be 1100 Hz, as presented in Figure 12 (b) and this dominant frequency is highlighted in Figure 11 (j).

When the method is repeated on TB2 at a shaft speed of 600 rpm with an applied load of 500 N, the periods of the key vibration signatures are now longer, thus the frequencies have decreased. Figure 13, shows that the periods of the entry event have approximately doubled (1.8ms), thus the characteristic frequency is now similar to the time taken for the rolling element to unload when entering the defect, which is a frequency of 550 Hz and is highlighted in Figure 11 (v). These changes in the characteristic frequencies are due to a change in the bearing assembly stiffness and are consistent with the changes shown in the spectrograms in Figure 11.
Figure 12: The ensemble average of the entry and exit events of TB1 at a shaft speed of 600 rpm with an applied load of 500 N, highlighting the periods of the key vibration signatures.

Figure 13: The ensemble average of the entry and exit events of TB2 at a shaft speed of 600 rpm with an applied load of 500 N, highlighting the periods of the key vibration signatures.

These experimental investigations demonstrate how the characteristic frequencies of the low-frequency events change with respect to the defect size and the applied load. It is possible to use these characteristics to distinguish between two defects that vary in size by exactly one angular ball spacing when estimating the size of the defect. Further, it has been demonstrated that the applied load must be considered for such analyses. When the applied load is above the critical-load, the shift of the characteristic frequencies of the low-frequency entry and exit events due to the presence of the extended defect may not be significant enough to be detected clearly, as the bearing assembly stiffness is similar to that of TB1. Successful
implementation of the detection of the change in the characteristic frequencies of the low-frequency entry and exit events to distinguish between a line spall and an extended spall defect requires determining the critical-load.

4.3 Detection of the critical load

In this section, the critical load of TB2 was determined using various loads at a shaft speed of 600 rpm to determine when the dominant characteristic frequency switches from approximately 500 Hz to 1 kHz. It was found that the critical load is between an applied load of 500 N and 600 N as estimated in the analytical analysis, as shown in Figure 9. Figure 14 (a) and (b) shows the spectrogram of the entry and exit events of TB2 with an applied load of 500 N; and Figure 14 (c) and (d) are the entry and exit events with an applied load of 600 N. It can be seen that the bearing characteristic frequencies increased greatly when the applied load was increased to 600 N; and the same can be seen in Figure 15, where the time period of the vibration signatures in the acceleration response have decreased when the applied load was increased to 600 N.
Figure 14: Spectrogram of the entry and exit event of TB2, where (a) and (b) are the spectrogram of the spectrogram of the entry and exit of TB2 had an applied load of 500 N at a shaft of 600 rpm; and (c) and (d) are the spectrogram of the entry and exit events of TB2 with an applied load of 600 N at a shaft speed of 600 rpm.
Figure 15: Acceleration response of the entry and exit event of TB2, where (a) and (b) are the acceleration response of the entry and exit of TB2 had an applied load of 500 N at a shaft speed of 600 rpm; and (c) and (d) are the acceleration response of the entry and exit of TB2 with an applied load of 600 N at a shaft speed of 600 rpm.

4.4 Detection of the dynamic stiffness variation as speed varies

This section will present the results of the variation of the dynamic stiffness of the bearing assembly as the shaft speed varies. Figure 16, shows the spectrogram of the entry and exit events of TB1 as the shaft speed was varied from 300 rpm to 750 rpm with an applied load of 500 N and 3000 N, where the magnitudes of the frequencies have been normalized. It was found that as the speed of the shaft increased the characteristic frequency related to the stiffness of the bearing assembly remained constant at about 1 kHz, shown by the arrows in Figure 16. However, the dominant frequency of the exit event decreases as the load increases; the low-frequency acceleration of the rolling element being reloaded as it is forced out of the defect is more prominent than the higher characteristic frequency of the bearing. Figure 17
(a) and (b), shows the ensemble average of the exit event at a shaft speed of 300 rpm with an applied load of 500 N and 3000 N, respectively; and it can be seen that the high-frequency characteristic frequency at approximately 1 kHz is still apparent but is no longer the dominant frequency of the exit event.

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<td></td>
<td>Exit</td>
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Figure 16: Spectrogram of the ensemble averaged of the entry and exit event of TB1 at varying loads and speeds.
Figure 17: Comparison of the exit events of TB1 when the load is varied from 500 N to 3000 N at a shaft speed of 600 rpm, where (a) is TB1 with an applied load of 500 N and (b) is TB1 with an applied load of 3000 N.

Figure 18 shows the dominant frequencies of the entry and exit events of TB2 as the speed of the shaft was varied from 300 rpm to 750 rpm with an applied load of 500 N and 3000 N. It was found that with a load that is less than the critical load, the dominant frequencies of the entry and exit event are lower than the dominant frequencies in TB1; even when the speed of the shaft varies.
Figure 18: Spectrogram of the entry and exit event of TB2 at varying loads and speeds.

The characteristic frequency of the reloading of the rolling element during the event are lower for TB2 than that of TB1, as the trailing edge of the defect has been worn from a sharp trailing edge to a shallower slope, as shown in Figure 19. This occurs due to high contact stresses between the rolling element and the raceway, and the contact stresses are higher in TB2 as there are less rolling elements supporting the applied load, thus the defect grows at a faster rate.
4.5 Recommendations for condition monitoring

In this section, recommendations are provided to implement a defect size estimation method utilizing the high-frequency characteristic frequency entry event as a roller enters a defect. According to the discussion presented in this study, the applied load has a significant effect on the bearing assembly stiffness and consequently the characteristic frequencies of the bearing, and hence determining the critical load is helpful. The critical load can be estimated using the method discussed in Section 4.20. The minimum acceptable depth of a defect should be defined to determine the critical-load of the bearing under condition monitoring. If the minimum acceptable depth of a defect is not known, defective bearings should be tested under loads close to their minimum recommended load carrying capacity. Spectrograms, should be constructed for the entry and exit events separately in accordance with the method described earlier in Section 4.2. The presence of the following characteristics in the spectrograms is an indication of existing extended defects:

1. Lower dominant characteristic frequencies of entry and exit events on the spectrograms of the vibration data under similar loads as the defect grows. This analysis requires historical trend data of the bearing being monitored and the bearing should be tested under similar loads.

2. Significant increase in the dominant characteristic frequencies of the entry events by increasing the applied load when compared with the increase of the dominant
characteristic frequency of the exit events. It should be possible to test a bearing at two different loads. This analysis does not require historical data of the bearing being monitored.

Note that while the first characteristic might be detectable in the power spectra density results, as shown in simulations by Petersen et al.\textsuperscript{4}, detection of the second characteristic is only possible by performing time-frequency analysis, similar to the one presented in this paper. This proposed method, which is an original contribution of this paper, does not require historical data for comparison, unlike the method suggested in Petersen et al.\textsuperscript{4}.

5. Conclusions

This paper investigates the relationship between the applied load and the relative displacement between the raceways of defective rolling element bearings with outer raceway defects. Analytical estimations and experimental measurements of the static stiffness show that stiffness variation is a function of load, defect size and cage angular positions. Experimental analysis of the bearing stiffness variations in this paper confirms that the static stiffness decreases in the loaded direction at cage angular positions where rolling elements are positioned in the defect. Furthermore, it is shown that the static stiffness in defective bearings with extended defects are more sensitive to applied loads than cage angular positions when the change in the bearing assembly stiffness is compared to a bearing with smaller defects. It was shown in this study that the stiffness of the bearing assembly in defective bearings changes dramatically when the applied load is above the critical-load. The critical-load depends on the depth and size of the defect; and the contact stiffness between the rolling element and the raceways. In a defective bearing, once the applied load reaches the critical-load, the number of load carrying rolling elements will increase as there is sufficient deflection of the assembly to cause the unloaded rolling elements in the defect zone to regain contact with the inner and outer raceways and support the applied load.

The feasibility of using the variation of the low-frequency entry event to distinguish defects that vary in size by exactly one angular ball spacing was investigated experimentally
by measuring the vibration response of two defective bearings. The experimental findings presented here, involving a range of applied loads to the bearing, confirm the Petersen et al. hypothesis that was based on theoretical predictions for a single load. Detection of the changes in the characteristic frequencies of the low-frequency vibration events is done using time-frequency analyses. Careful time-frequency analyses of different defective bearings under different applied loads showed the feasibility of this method in distinguishing the two defect sizes. It is shown that a successful implementation of this method to distinguish the two different defect sizes requires analysis to determine the critical-load introduced in this research. Recommendations for condition monitoring methods are given based on the analyses presented in this study. The new method proposed in this paper does not require historical data, whereas previously suggested methods require historical data. The complete analytical model and experimental data with the MATLAB script to view the data are publicly available at FigShare.
References


