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Analytical validation of an explicit finite element model of a rolling element bearing with a localised line spall

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ABSTRACT

In this paper, numerically modelled vibration response of a rolling element bearing with a localised outer raceway line spall is presented. The results were obtained from a finite element (FE) model of the defective bearing solved using an explicit dynamics FE software package, LS-DYNA. Time domain vibration signals of the bearing obtained directly from the FE modelling were processed further to estimate time–frequency and frequency domain results, such as spectrogram and power spectrum, using standard signal processing techniques pertinent to the vibration-based monitoring of rolling element bearings. A logical approach to analyses of the numerically modelled results was developed with an aim to presenting the analytical validation of the modelled results. While the time and frequency domain analyses of the results show that the FE model generates accurate bearing kinematics and defect frequencies, the time–frequency analysis highlights the simulation of distinct low- and high-frequency characteristic vibration signals associated with the unloading and reloading of the rolling elements as they move in and out of the defect, respectively. Favourable agreement of the numerical and analytical results demonstrates the validation of the results from the explicit FE modelling of the bearing.

1. Introduction

Studying the vibration response characteristics of rolling element bearings, not only defective [1–28] but also non-defective (ideal) [29–32], has been a subject of significant interest to many researchers due to their wide usage in rotating machinery across various industries. While non-defective bearings can produce cyclic vibrations [29–32], also referred to as variable compliance vibrations [33], which are caused by the cyclic variation of the stiffness of a bearing assembly as a varying number of rolling elements support the applied load, the presence of various defects within rolling element bearings produces undesirable vibrations that can eventually lead to machinery breakdown if not diagnosed in time. Since the development of the first analytical model in 1984 [1], often referred to as a *classical model* in the literature [34], several authors have developed analytical [1–20] and FE models [21–28] to simulate the vibration of bearings with localised defects on the raceways (both inner and outer) and rolling elements. The analytical models can be categorised [34] into periodic impulse-train [1–4], quasi periodic impulse-train [5–8], and nonlinear multi-body dynamic models [9–20], whereas FE models [21–28] are dynamic and have been solved using commercial software packages [35].

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The periodic impulse-train models [1–4] simulate force impulses that are considered to be periodically generated by a localised point spall. For stationary outer race defects [3,4], these models produce impulses of equal amplitude with similar characteristics (shape, height, and width), but for rotating inner race and rolling element defects [1–4], the impulses were modulated according to the static load distribution in bearings [28,36]. Random fluctuations in the period of impulses due to defects [5] that are caused due to the slip of rolling elements were added to the periodic impulses-train models to develop quasi-periodic (or aperiodic) impulse-train models [5–8].

Using valuable insights from the impulse-train models [1–8], nonlinear, multi-body dynamic models [9–20] were later developed to relatively better simulate the vibration response of bearings with localised defects. These models incorporated parts of a bearing and associated housing that is contrary to the impulse-train models. For simplification, the multi-body models [9–20] considered various assumptions, such as: 1) inner and outer rings could not bend or flexurally deform due to their rigid coupling with shaft [9–20] and bearing housing [9–17,19,20], respectively, except the model in Ref. [18], which considered the outer ring as deformable, 2) rolling elements within a bearing were either excluded or considered to have no mass [9–11,13–16,19] except in Refs. [12,17,18,20], 3) slip of the rolling elements was ignored [9–12,14–17] leading to simulating periodic impulses, whereas some [13,18–20] consider the slippage, 4) except the models in Refs. [17,20], centrifugal forces acting on the rolling elements as they rotate during the operation of a bearing were ignored [9–16,18,19]. The main emphasis of the multi-body models was to generate vibration time-traces, and subsequently perform an envelope analysis [37] to primarily predict the defect-related frequency components for model validation purposes. Although validated, the aforementioned assumptions led to the accuracy of the multi-body models being compromised in the sense that they could only partially predict the vibration characteristics of defective bearings. For instance, the models accurately estimated the values of the bearing defect frequencies; but not the amplitudes of the defect-related impulses [4,11,12,15,16]. While the difference between the predicted and measured vibration amplitudes has been reported as high as 60,000% [16], for some models [4,11,12], the predicted instantaneous amplitudes were corrected to match experimental measurements with no explanation or justification provided. In summary, from the review [34] of the multi-body models, no single publication addresses all the aforementioned limitations. In the work presented in this paper, no assumptions such as those mentioned above were considered during the FE modelling of a rolling element bearing. Furthermore, a majority of the multi-body models could not predict the low-frequency vibration characteristic signatures pertinent to the *unloading* of the rolling elements on their entrance in a defect [25,28]; this *unloading event* can also be referred to as *de-stressing*. Based on previous experimental [25,28,38] and numerical [25,28] findings, some authors [19,20] have adapted their models to generate the low-frequency event. However, in the FE model presented here, the low-frequency unloading event was generated without the application of any specific conditions during the simulation, and the presence of low-frequency characteristic signatures pertinent to this event is demonstrated through the time–frequency analysis, which is lacking in the existing multi-body modelling.

A few researchers [21–24] have also developed FE models of ball bearings with localised raceway defects. In FE modelling and methods using commercially available software packages [35], a number of assumptions can be minimised leading to better accuracy in results compared to analytical models. However, certain values and choices for parameters in FE methods still need judicious assumptions and problem-related inputs like material model and properties, damping, boundary conditions and loads in addition to adequate meshing of a model. A recent review [34] of these models has highlighted that the accuracy of the FE models [21–24] was compromised because either the outer ring or its outer surface was modelled as rigid similar to the aforementioned multi-body models [9–20]. This artificially over-stiffened the bearing structure that led to unrealistic (high) instantaneous acceleration levels of the order of 4000 g [23] and 15,000 g [24] compared to the corresponding experimental measurements of 100 g and 10 g in Refs. [23] and [24], respectively. The significant mismatch between the predicted and measured amplitude levels reported for the multi-body models [2,4,11,12,14–16] remains a problem with the existing FE models. Furthermore, several errors and ambiguities associated with the FE models [21–24] and results have been discussed in Ref. [34]. An extensive review of existing impulse train [1–8], multi-body [9–20], and FE [21–24] models that simulate the vibration response of bearings with local defects has recently been published by the authors of this paper [34].

In this paper, results from an explicit FE modelling of a rolling element bearing that has a localised outer raceway line spall are presented. The development of the FE model was presented by the authors of this paper in an earlier publication [25]. Therefore, the model is not fully described here; however, its brief description is provided for completeness and convenience. Unlike previous models in the literature, the bearing components in the FE model in question [25], from which the results are presented here, were modelled as flexible parts, which provides relatively better representation of the stiffness of the bearing, hence its structural vibration response. Had the parts of the bearing been modelled as rigid instead of flexible, it would have caused artificial over-stiffening of the bearing structure that would have consequently compromised the accuracy of the modelling results. Contrary to previous models [1–16,18,21–24], the explicit FE model in question [25] accurately predicts the low-frequency characteristic vibration signatures pertinent to the unloading of the rolling elements when they move into the defect, in addition to the predominantly high-frequency *reloading* of the rolling elements when they move out of the defect [25,28]; this *reloading* event can also be referred to as *re-stressing*. The vibration time-traces obtained directly from the FE modelling of the defective bearing were processed further to estimate time–frequency and frequency domain results, such as spectrogram and power spectrum. Through the time domain analysis, the accurate bearing kinematics, defect frequencies, predicted by the FE model, are discussed. Time–frequency analysis of the results is presented to demonstrate the generation of the distinct low-frequency unloading and predominantly high-frequency reloading events. For the frequency domain analysis, the most commonly used technique, envelope analysis [37], was used to show that demodulated power spectrum shows fundamental defect frequency and harmonics. The presentation of the results from various analysis along with relevant discussion leads to the analytical vali-