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Christopher Niezrecki *Editor*

Structural Health Monitoring and Damage Detection, Volume 7

Proceedings of the 33rd IMAC, A Conference and Exposition on Structural Dynamics, 2015

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Editor
Christopher Niezrecki
Department of Mechanical Engineering
University of Massachusetts Lowell
Lowell, MA, USA

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Preface

Structural Health Monitoring & Damage Detection represents one of ten volumes of technical papers presented at the 33rd IMAC, A Conference and Exposition on Structural Dynamics, 2015, organized by the Society for Experimental Mechanics, and held in Orlando, Florida February 2–5, 2015. The full proceedings also include volumes on Nonlinear Dynamics; Dynamics of Civil Structures; Model Validation and Uncertainty Quantification; Dynamics of Coupled Structures; Sensors and Instrumentation; Special Topics in Structural Dynamics; Experimental Techniques, Rotating Machinery & Acoustics; Shock & Vibration Aircraft/Aerospace, Energy Harvesting; and Topics in Modal Analysis.

Each collection presents early findings from experimental and computational investigations on an important area within Structural Dynamics. Structural Health Monitoring is one of these areas. Topics in this volume include:

Structural Health Monitoring
Damage Detection
Numerical Modeling

The organizers would like to thank the authors, presenters, session organizers, and session chairs for their participation in this track.

Virginia Tech, USA A. Wicks

Contents

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Chapter 1
Bearing Faults Simulations Through a Parametric Model of a Gearbox

S. Cinquemani, F. Rosa, and E. Osto

Abstract The paper deals with a project aimed to improve the reliability of a condition monitoring system applied on gearboxes installed on rolling mills. In this context, to properly set up the algorithm, it is necessary to have measurements associated both to standard operating conditions and to malfunctioning. Since the experimental determination of the latter is obviously not cost and time effective, they can be simulated by means of numerical models of the mechanical system in several operating conditions. The outputs generated, corresponding to different fault conditions of the more critical elements of the system, will provide a useful data base to tune the algorithm of condition monitoring.

Keywords Condition monitoring • Bearing • Mechanical transmission • Rolling mill • Parametric model

1.1 Introduction

Gearboxes for rolling mills are complex machines that cannot be treated as commodities, since they are parts of a complex drive system that in case of failure could seriously affect the plant productivity [1–3]. The root causes of a geared system failure can sometimes be quite different from the appearing ones. From this point of view, an early detection of improper operating conditions, as condition monitoring can provide, gives a good chance to plan extraordinary maintenance to prevent sudden stop of production and to identify primary failure causes instead of secondary ones. It worth to be mentioned that preventive maintenance to solve primary failure causes implies negligible cost compared to ones related to secondary failure causes (e.g. bearing replacement vs. gear replacement).

Bearings represent a typical source of gearboxes failures or improper operation [4, 5]. Also for bearing, due to the vast number of different failure modes, specific standards have been introduced as ISO 15243. Gradual deterioration of the operating behaviour is normally the first signal of bearing damage. Failures due to poor ordinary maintenance (lack of lubrication, for example) and improper mounting are relatively infrequent, and very often lead quickly to machine downtime. On the other hand, depending on the operating conditions, a few weeks, or under some circumstances, even a few months, may pass from the time damage begins to the moment the bearing actually fails because of the contact fatigue damaging mechanisms. This typical progressive evolution of the damage makes bearings especially suitable for continuous condition monitoring applications. From the point of view of failure detection, the main effects of bearing damage impacts on operating temperatures, lubricant contamination and vibrations. In principle, all these information can be used for condition monitoring application, but the techniques based on the analysis of vibration signals are the most efficient as they can provide an early identification of the specific bearing involved, of the part of the bearing affected by damage and on the degree of the damage itself, thanks to the information coming from frequency and amplitude data. In particular damages related to contact fatigue affecting the races or the rolling elements can be detected and identified [6]. Moreover they are suitable to be modelled by means of models which can provide preliminary information in terms of expected frequencies and amplitudes in the signal analysis.

Condition monitoring algorithms are based on signals in different working conditions. To properly set up the algorithm, it is necessary to have measures associated both to standard operating conditions and to malfunctioning. The latter, not being experimentally determinable, can be simulated by developing numerical models of the machine under varying

S. Cinquemani (✉) • F. Rosa
Department of Mechanical Engineering, Politecnico di Milano, Via La Masa 1, 20156 Milano, MI, Italy
e-mail: simone.cinquemani@polimi.it

E. Osto
Siemens VAI Metals Technologies Srl, Via Luigi Pomini, 92, Marnate, Italy

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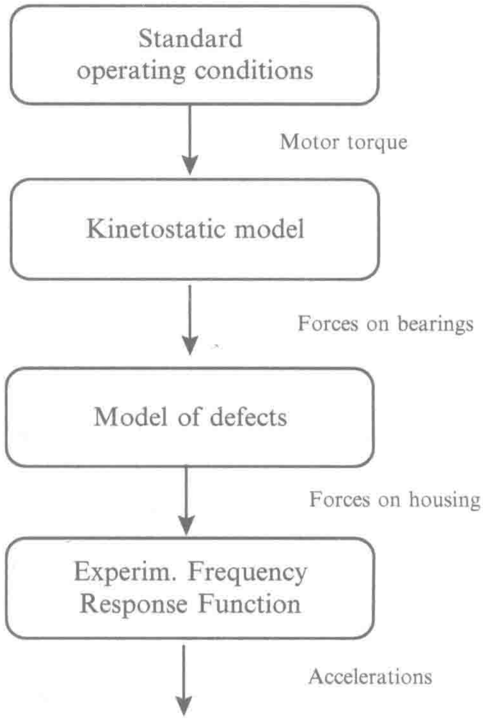

Fig. 1.1 Overview of the steps to generate virtual signals corresponding to malfunctioning

conditions [7]. The outputs generated, corresponding to different fault conditions associated with the main common failures of the elements that constitute the transmission, will provide a useful data base to properly set the algorithm of condition monitoring.

The paper deals with the description of such approach (Fig. 1.1). Virtual signals related to malfunctioning can be obtained using the real working conditions (torque and motor speed) as the inputs of the model thus implementing the dynamics associated with the malfunctioning of different bearings. The output of the model consists on the harmonic forces acting on the bearing housing. Through experimental transfer functions of the gearbox, virtual accelerations can be calculated.

The paper is structured as follows. Section 1.2 describes the kinetostatic model of the mechanical transmissions. Section 1.3 introduces the mathematical model of main failures of bearings. For the sake of clarity, the presented only cylindrical roller bearings model is here presented, but the reader will understand that this approach can be easily replicated for spherical, tapered and ball bearings. Section 1.4 introduces how to obtain the frequency response function of the gearbox, while Sect. 1.5 shows how the approach can be usefully applied on a real test case. Finally conclusions are drawn in Sect. 1.6.

1.2 Kinetostatic Model of the Transmission

Transmissions installed on rolling mills can be generally divided into two groups: with parallel axis (Fig. 1.2) and with orthogonal axis (Fig. 1.3). For each group a set of parameters has been defined to describe the transmission (i.e. mechanical and geometrical features, type of bearings, gears, etc.).

To be able to manage different transmissions, a reduced model with lumped parameters is developed according to the parameterization carried out. Forces on bearings (F_X, F_Y, F_z) can be computed according to [3], as a function of the main features of the transmission and of the operating conditions (i.e. motor torque and angular speed). Then, the displacements of the inner ring of each bearing are calculated with respect to applied forces, as the mathematical models of defects need this input to generate the virtual accelerations.

Unfortunately, while the direct relationship between displacement of the shaft and forces transmitted by the bearing exists [2, 8], the inverse relationship can not be solved analytically. For this reason an iterative procedure has been implemented.

Even if many bearing types (cylindrical rollers, double row tapered rollers, double row self-aligning, and four-point-contact bearings) have been analysed and modelled, only cylindrical roller bearings will be presented hereafter in order to give an overview of the adopted approach. Each model is furthermore capable to simulate the more common bearing defects: inner and outer race pits and cages wear resulting in an uneven roller distribution.

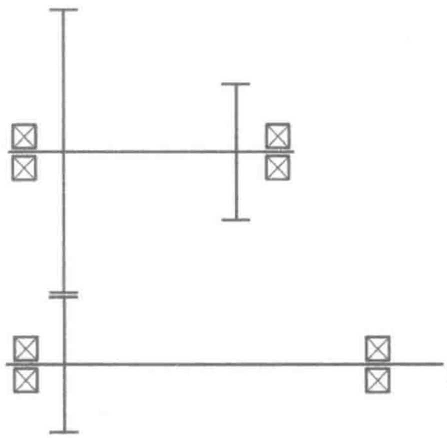

Fig. 1.2 Parallel axis

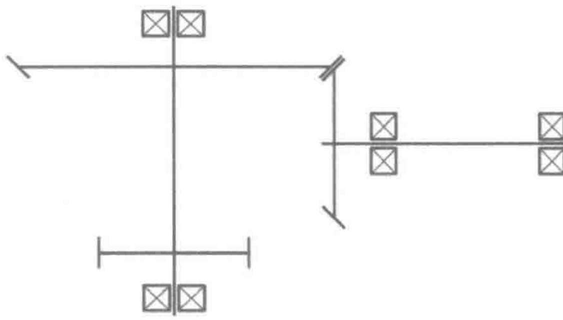

Fig. 1.3 Orthogonal axis

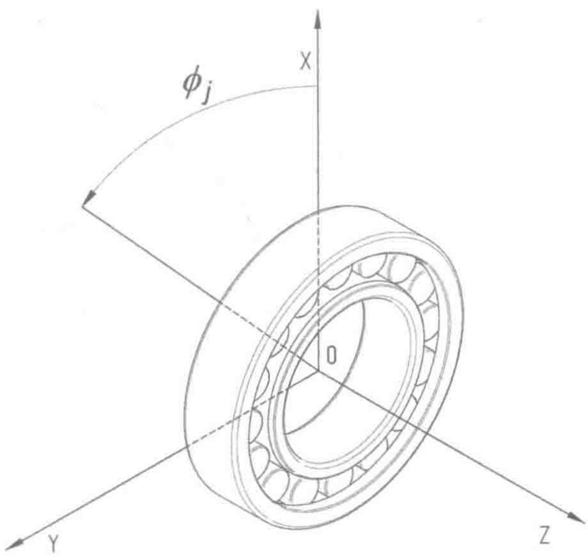

Fig. 1.4 Cylindrical roller bearing model

In order to ease the integration of these models with the global model of the transmission and to derive an explicit analytical formulation, it was assumed that outer race is fixed, while inner race rotates with the shaft and hence undergoes to the same translational displacements of the shaft section and that bending deformations of inner and outer rings are negligible with respect to roller-race contact deformations, that are hence the only deformations of bearing components considered to determine bearing global stiffness.

Adopting the reference system shown in Fig. 1.4, in a cylindrical roller bearing without any defect, for a given displacement of the inner race $\underline{\Delta} = \Delta_x \, \underline{u}_x + \Delta_y \, \underline{u}_y$, the total radial contact deformation of the generic *j-th* roller at angular position ϕ_j is therefore:

$$\delta_j = \Delta_x \, cos \, \phi_j + \Delta_y \, sin \, \phi_j - c$$

where c is the radial clearance of the bearing in operating conditions. It is worth observing that, because of the geometrical nature of this expression, it is an actual contact deformation only if δ_j is positive. The next step consists in determining the load-deformation relationship.

According to Harris and Kotzalas [4], on the basis of laboratory testing of crowned rollers loaded against raceways, Palmgren [6] experimentally determined the following equation for contact deformation δ *[mm]*:

$$\delta = 3.84 \cdot 10^{-5} \cdot Q^{0.9} / \, l^{0.8}$$

where l is roller effective length *[mm]*, and Q is the normal total force between a rolling element and a raceway *[N]*.

This expression has been adopted since it allows deriving a direct and explicit relationship between the applied load and the consequent displacements, even if it does not allow taking into account some aspects (i.e. curvature variations and ratios). The numerical constant value for specific roller crowning and/or materials can anyhow be determined by means of experimental tests. By solving this equation with respect to Q and taking into account that each roller is in contact with both races, it is possible to derive an "equivalent stiffness":

$$K_{eq} = k_b \cdot 2^{-n}$$

where $n = 10/9$ for roller bearings and $k_b = 7.7652 \cdot 10^4 \cdot l^{8/9}$ is the "single stiffness".

As a result, the load acting on a single roller and its deformation are related by the following equation:

$$Q_j = \gamma_j \cdot K_{eq} \cdot \delta_j{}^n$$

where $\gamma_j = 1$ if $\delta_j > 0$, and is zero otherwise.

The final step is the summation of components of forces Q_j, in order to determine the force exerted by the outer ring on its seat as a consequence of inner race displacement $\underline{\Delta}$. This approach have been further extended to the so called "lamina model" [9], in order to take also into account roller profile modifications [9].

At the end, for each bearing, it is possible to find out a combination of displacements along the three axes (X, Y, Z) and forces transmitted along the same three directions (F_X, F_Y, F_Z). This look-up table can be calculated offline for all the bearings in the database and represents a theoretical stiffness of each bearing, as it relates displacement to forces.

Once the look-up table has been calculated, the displacement of the inner ring of the bearing can be obtained iteratively finding the best combination of forces (F_X, F_Y, F_Z) that matches with the forces applied on the bearing ($\underline{F}_X$, $\underline{F}_Y$, $\underline{F}_z$). It is important to note that the accuracy of the solution is strictly related to the resolution of the look up table. If the combinations are few, the result of the algorithm could be far from the real solution.

1.3 Models of Defects on Bearings

Pits presence on a raceway has been introduced as a reduction of roller contact deformation, i.e. as an increase of bearing radial clearance. In practice, this means that the expression of δ_j modifies as follows

$$\delta_j = \Delta_x \cos \phi_j + \Delta_y \sin \phi_j - c - \beta_j(t) \cdot C_d$$

where C_d is pit depth and $\beta_j(t)$ is a Boolean time dependant function that is equal to 1 only if the theoretical contact point between roller and inner or outer race at time t is within pit circumferential extension. It is worth observing that this function depends also on which race the pit is located, since the inner race rotates, and hence the absolute position of a pit on it is not constant. To reduce the number of variable, a simple relationship between the depth (C_d) and the width (Δ_d) of the defect has been imposed as shown in Fig. 1.5

To evaluate the effects of damages on bearings, it is possible to set different values of C_d, and then of Δ_d.

It's important to note that models of defects are based on the idea that a roller, when it is over a defect, loses some of the load that can transmit. The limit condition occurs when the roller completely loses the contact with the ring and, therefore, it is no more able to transmit any force. In this condition (easy to be reached for bearing under small a load), even increasing the depth of the defect an increases in force associated with the defect would not be observed.

Coming to cage wear, it was introduced by assuming that its effect on bearing global model can be considered by changing the circumferential positions (with respect to their evenly spaced theoretical positions) of the points where each roller exerts

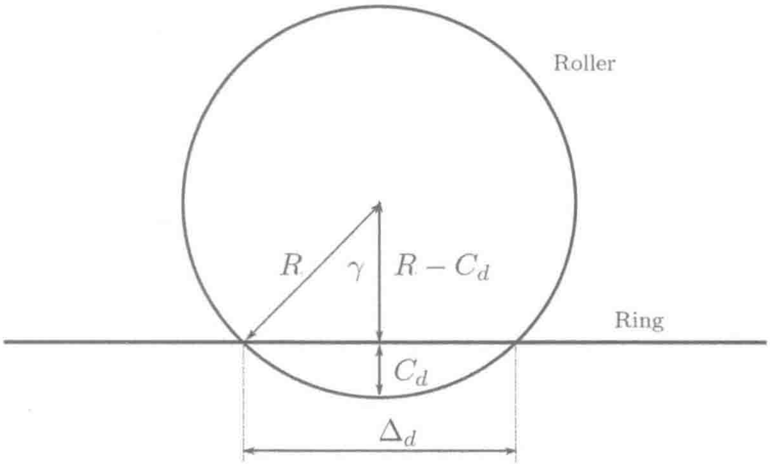

Fig. 1.5 Geometrical relationship between the depth (C_d) and the width (Δ_d) of the defect

its resultant forces on the rings. From a mathematical point of view, after having defined the maximum circumferential displacement (ε_{wc}) of these points, this assumption leads to define a distribution of these displacements in order to locate all of them. If this type of defect is introduced, this implies that angles ϕ_j are no longer evenly spaced, as assumed in the model of a "healthy" bearing.

Similar considerations have been made for the other types of bearings taking into account also, if necessary, the relationships between axial and radial components, in order to derive appropriate analytical models.

1.4 Experimental Frequency Response Function

The relationship between forces applied on the housing and the corresponding virtual accelerations is obtained experimentally.

Each test consists on hitting the chassis with a dynamometric hammer and measuring the corresponding accelerations through a three axis accelerometer. As a matter of fact, an impulsive test allows driving all the resonances of the system. This means that, with a single test, it is possible to evaluate the dynamics of the system in a wide range of frequency. As the hammer contains a load cell, a measurement of the applied force can be collected. Signal have been acquired using a Daq board Ni 9234 with a sample frequency of 51.2 kHz. Figure 1.6 shows time histories of the force applied to the chassis and the corresponding measured acceleration. The impulsive test is repeated 10 times to improve the quality of the frequency response function, increasing the signal to noise ratio through the so called linear averaging technique.

Knowing the forces applied and the accelerations of the chassis it is possible to estimate the frequency response function of the structure. Figure 1.7 shows, for example, the frequency response function between the force applied on the chassis along the vertical direction, and the corresponding acceleration along the same axis. It is important to highlight that the virtual accelerations resulting from this work, are strictly related to the experimental transfer functions of the transmission.

Information collected in the frequency response function are extremely useful. Once the forces transmitted to the chassis, due to a defect on a bearing, are known from the mathematical model, the corresponding accelerations can be easily obtained simply multiplying the forces (in frequency domain) to the frequency response function of the system. For example, being $TF_{X,B1}(f)$ the frequency response function of the chassis, close to bearing B1, along the X axis and $F_{X,B1}(f)$ the forces along the same direction estimated by the model for a defect on bearing B1, it results:

$$\ddot{x}_{B1}(f) = TF_{X,B1}(f) \cdot F_{X,B1}(f)$$

where $\ddot{x}_{B1}(f)$ is the acceleration measured along x axis due to a defect on bearing B1.

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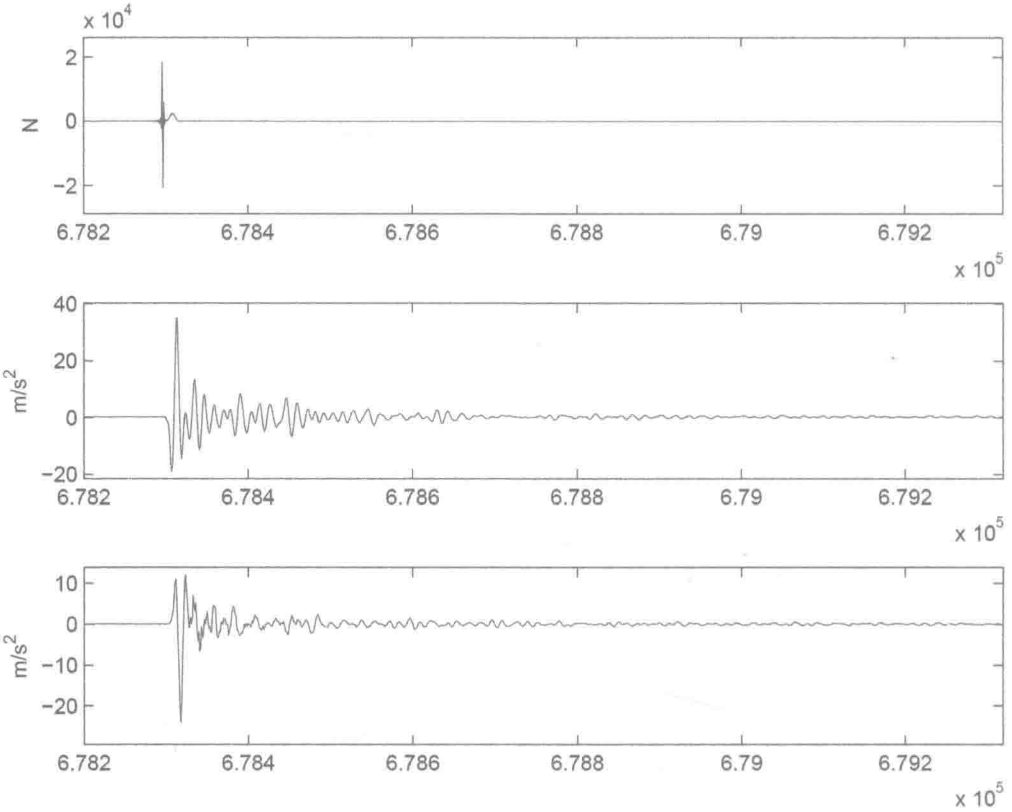

Fig. 1.6 Force applied to the chassis and corresponding measured acceleration

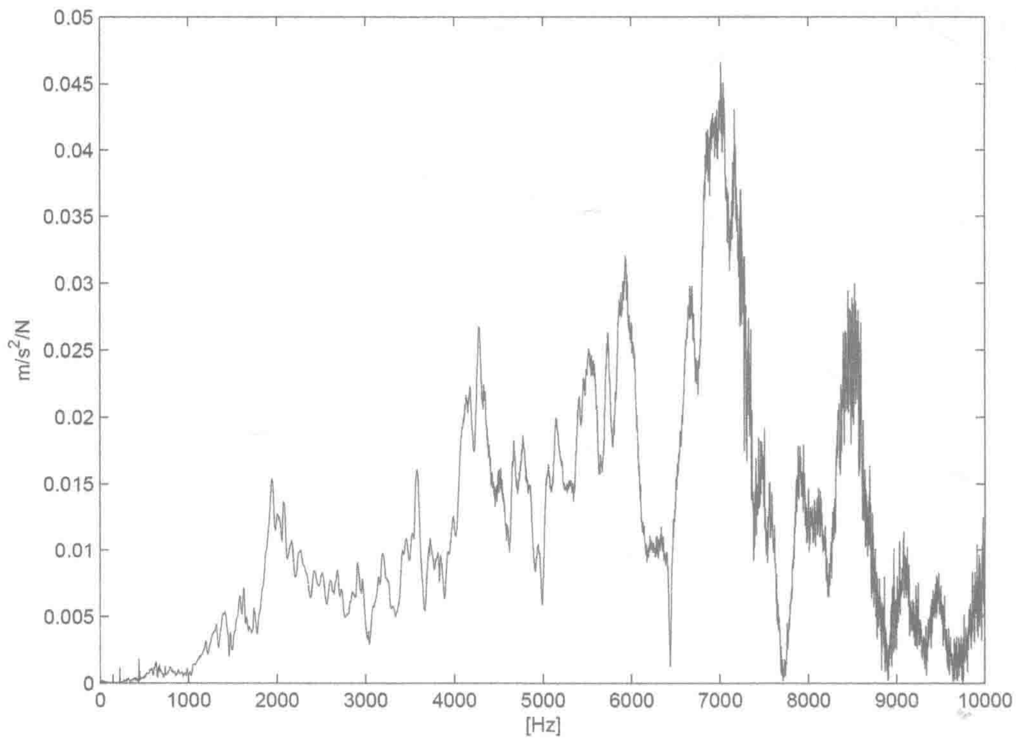

Fig. 1.7 Frequency response function

1.5 Test Case

The parametric model descripted in the paper has been used to simulate failures on bearing of a real mechanical transmission installed on a rolling mill. The layout of the transmission to be analysed is depicted in Fig. 1.8. The transmission is used in a rolling mill plant and it has a speed ratio of 0.072. Shaft 0 is connected to the motor, while shafts 2, 3 are linked by spindles to the rolls of the rolling mill. Speed reduction is obtained from the first two stages of the transmission, while the third one (whose ratio is equal to 1) is used to split the power to the two rolling rolls. Bearings configuration is resumed in Table 1.1.

According to different operating conditions and different kind of defects, the model can evaluate the forces applied to the chassis along X, Y directions and the corresponding accelerations.

Let's consider the tapered roller bearing B1. Figures 1.9 and 1.10 respectively show the forces applied along X direction and the corresponding accelerations. Numerical analysis have been carried out considering motor torque: $T_M = 14$kN, motor angular speed $\omega_M = 500$ rpm, Cd = 0.1 mm for defect on the outer ring, Cd = 0.1 mm for defect on the inner ring, Cd = 3.2 mm for defect on the cage.

Information on virtual accelerations will be useful to train the condition monitoring algorithm.

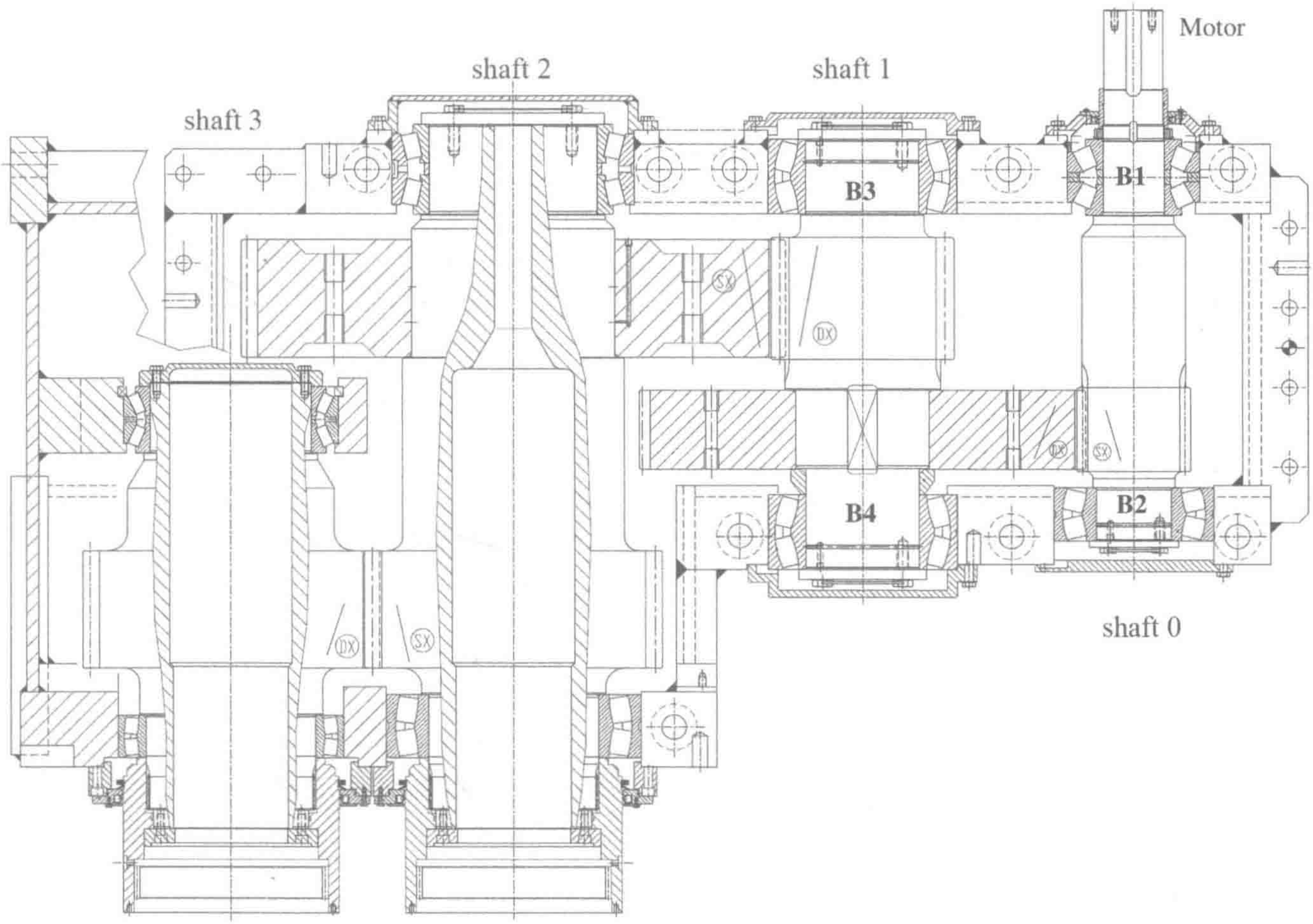

Fig. 1.8 Layout of the considered mechanical transmission

Table 1.1 Configuration of bearings

B1	B2	B3	B4
Tapered roller b	Spherical b	Spherical b	Spherical b
SKF 31326	SKF 22332	SKF 24148	SKF 24148

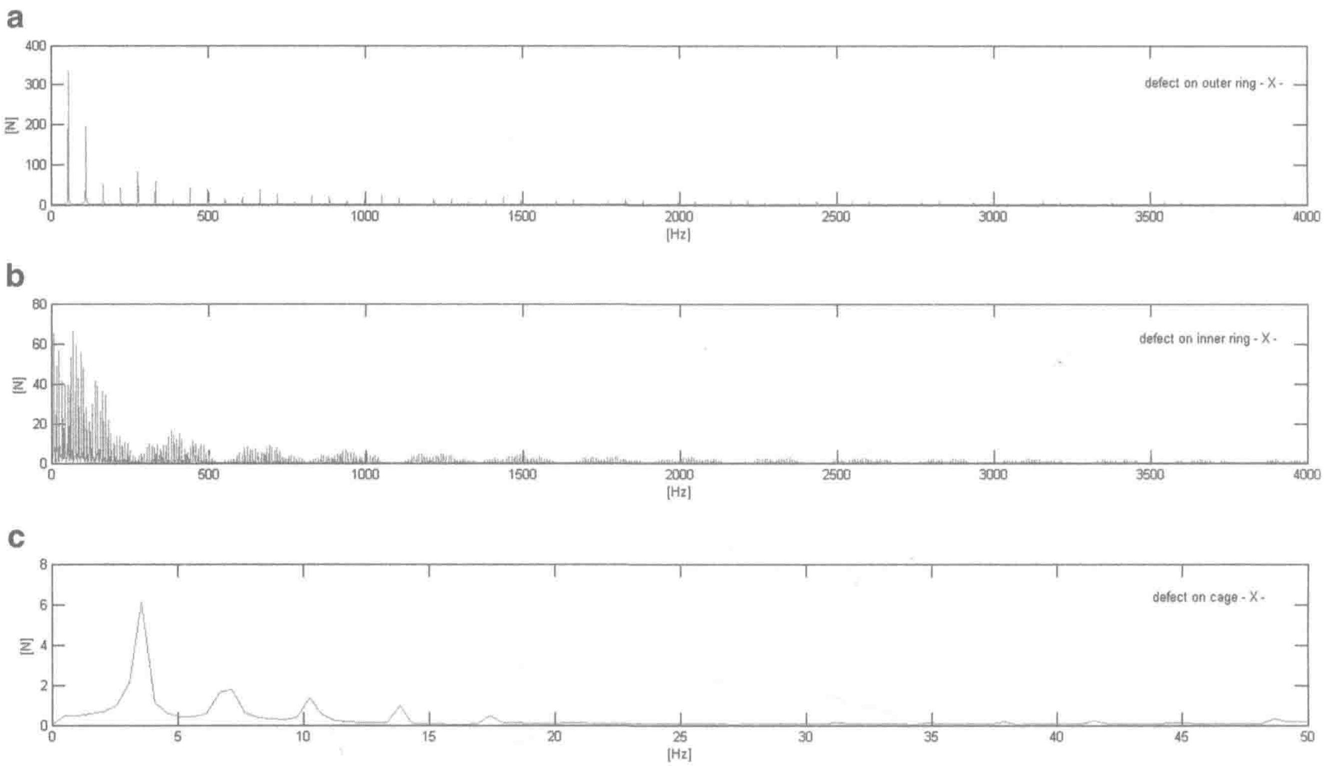

Fig. 1.9 Forces applied on the chassis along X direction ((**a**) defect on the outer ring, (**b**) defect on the inner ring, (**c**) defect on the cage)

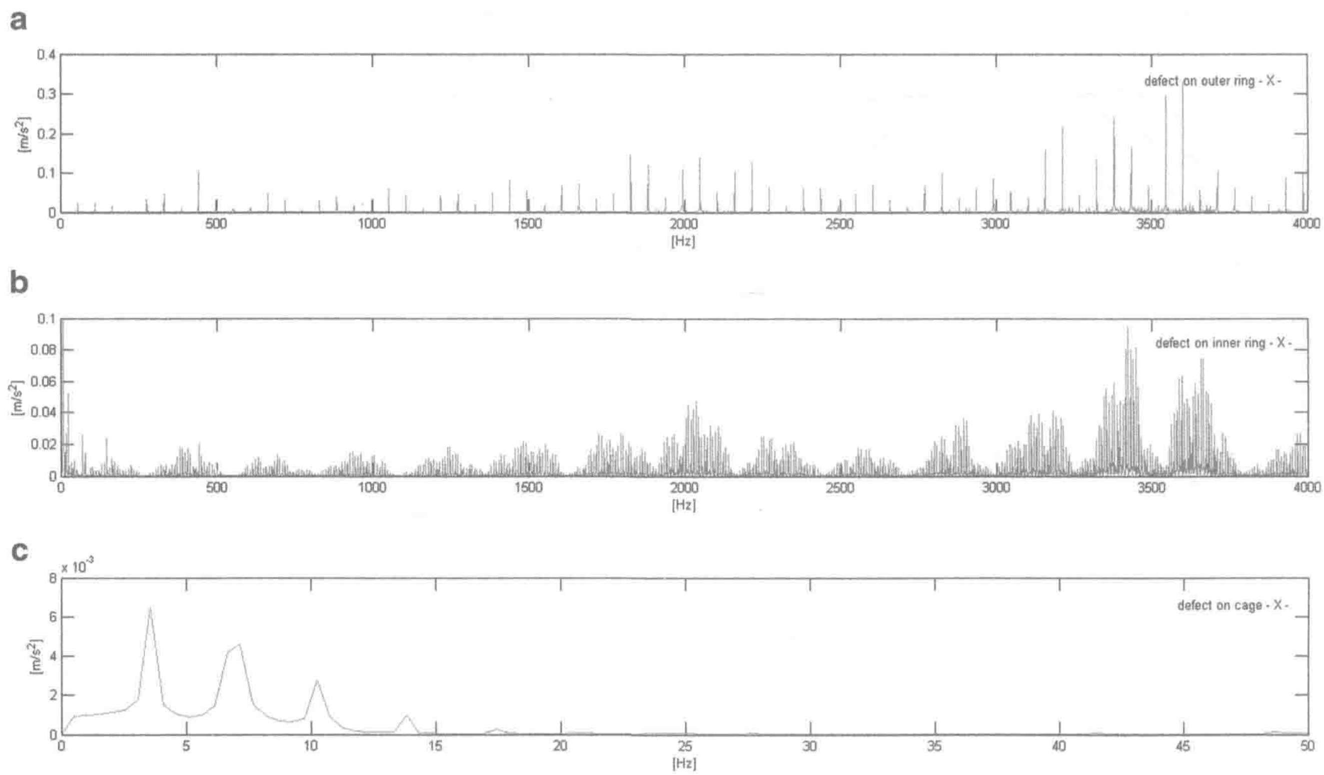

Fig. 1.10 Virtual accelerations of the chassis along X direction ((**a**) defect on the outer ring, (**b**) defect on the inner ring, (**c**) defect on the cage)

1.6 Concluding Remarks

This activity is part of a project aimed to develop a system to monitor the working conditions of mechanical transmissions installed on rolling mills. In this context, to properly set up the algorithm, it is necessary to have measures associated both to standard operating conditions and to malfunctioning. The letters, not being possible to effectively determine them experimentally, have been simulated by developing numerical models of the machine under varying conditions.

The outputs generated, corresponding to different fault conditions of the more critical components of the transmission, provide a useful data base to properly set the condition monitoring algorithm.

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Chapter 2
Sensitivity Evaluation of Subspace-Based Damage Detection Method to Different Types of Damage

S. Allahdadian, C.E. Ventura, P. Andersen, L. Mevel, and M. Döhler

Abstract In this paper we investigate a damage detection technique based on the subspace method by applying it to an existing bridge structure model. A reference state of the structure is evaluated using this technique and subsequently its modal parameters are indirectly compared to the current state of the structure. There are no modal parameters estimated in this method. A subspace-based residual between the reference and possibly damaged states is defined independently from the input excitations employing a χ^2 test and then is compared to a threshold corresponding to the reference state. This technique is applied to a model of the bridge structure located in Reibersdorf, Austria. The structure is excited randomly with white noise at different locations and the output data is generated at typical locations instrumented and measured in a bridge. Various damages are simulated in different elements and the sensitivity of the method to each type and ratio of damages is assessed. This evaluation is performed by comparing the prediction of the damage state using this technique and the simulated damage of the structure. It can be inferred from the results that in general the statistical subspace-based damage detection technique recognizes most of the damage cases, except the ones with insignificant change in the global dynamic behaviour.

Keywords Damage detection • Subspace method • Health monitoring • Bridge damage • Statistical damage detection

2.1 Introduction

Structural health monitoring is regarded as the main tool in assessing the functionality of existing structures. The importance of these techniques and researches becomes obvious by considering that failure of a structure can result in catastrophic lost. During past decades extensive researches have been done in the literature in order to investigate an ideal nondestructive damage detection technique. Nondestructive damage detection techniques can be categorized into two groups based on their requirements [1]: (I) local techniques, which need access to all parts of the structure or the location of damage if known, and (II) global damage techniques which use vibration data to evaluate global dynamic characteristics of the structure. In the latter method there is no need to know or have access to the location of damage in priori. The dynamic characteristics used in these methods are usually natural frequencies, mode shapes and damping values.

In order to detect if a damage has occurred in a structure, evaluation of these dynamic characteristics can be avoided by using statistical approaches, e.g. statistical subspace-based damage detection technique (SSDD) [1–5]. The damage can be detected by comparing a statistical model from the possibly damaged structure to the one obtained from a reference state. In other words a subspace based residual function between these states is defined and compared using a χ^2 test. In this way there is no need to estimate the natural frequencies and mode shapes of the structure, making this approach capable of being used in real-time monitoring of structures. In [3] and [6] it is investigated that this approach can also perform robustly under ambient excitations with changing statistics.

S. Allahdadian (✉) • C.E. Ventura
University of British Columbia, Vancouver, Canada
e-mail: saeid@civil.ubc.ca; ventura@civil.ubc.ca

P. Andersen
Structural Vibration Solutions A/S, Aalborg, Denmark
e-mail: pa@svibs.com

L. Mevel • M. Döhler
Inria, Centre Rennes—Bretagne Atlantique, Campus de Beaulieu, 35042 Rennes, France
e-mail: laurent.mevel@inria.fr; michael.doehler@inria.fr

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