Auralisation of machine structure-borne sound due to rolling bearings

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Introduction

The interest on bearing diagnosis techniques is increasing due to the necessity of early evaluation of their surface condition and remaining operating time. The use of auralisation as a tool for machine diagnosis has not yet been investigated. Further, the techniques to track the development of degradation of bearings are not suitable to deliver information for very early stages of use (actually, before the occurrence of pitting or other material removal occurs). This work concentrates on the early stages of degradation and presents the results of the auralisation of machine structure-borne sound coming from a rolling bearing. To do so, one needs the transfer functions from the excitation point to a vibration sensor mounted on the machine surface (already described in [1]) and a proper physical model of the contact in bearings to calculate the time evolution of the excitation produced by this source [2]. This model was adapted to correspond to the case of rolling bearings and its dynamical behaviour was simulated with the actual excitation signal in dependence to parameters such as rotational velocity, radial load in bearing and roughness profile of the surfaces, among others. Then, convolving the impulse response of the previously measured transfer functions with the simulated vibration of the bearing, it is possible to auralise the machine vibration for different surface conditions of 4-ball and 3-cylindrical rolling bearings. The resulting signals are then compared with measurements taken with accelerometers on the running experimental machine.
The final aim is to try to run the bearings until different surface conditions are achieved and to see if these alterations would lead to sensitive changes on the vibration signal and on the simulated excitation signal. The use of auralisation can be helpful as a diagnosis tool, as it is known that some specialists are able to "hear and feel" if a machine is running improperly.

Measured transfer functions

The measurement of the transfer path between the point where the excitation occurs, i.e., in the bearing, and the point where the measurements are taken (usually at the machines housing) is made with the help of a special actuator that substitutes one of the rolling elements. The details concerning this measuring technique can be found in [1].

Nevertheless, it should be mentioned that all measurements of the transfer paths were made with the machine mounted, but not running. It means that so far no information about the dynamics of the machine, the influence of temperature or from other vibration sources (like the couplings and the two other support bearings) as well as the effect of lubrication is incorporated in the transfer functions.

Physical model of contact

The main objective of the physical modelling of this problem is to try to simulate the actual excitation signal coming from the contact between the rolling bodies and the races in the bearing. This simulation should take into account parameters like the radial load imposed to the bearing, rotational speed, type of rolling element, the geometry of the bearing, the presence of lubrication among others.

The physical model is broadened over the approach described in [2] and will be described in depth in a future publication. It is now part of the computational package SAMBA (Structural Acoustic Model for Bearing Analysis). The theory was extended to accomplish the contact between every type of surface without a priori assumptions about the height distribution of asperities. This is especially important if one is interested in calculating the actual vibration generated by this contact and also in studying the changing of the characteristics of the surface with time.

The first step is to create an equivalent rough surface through the sum of the measured roughness profiles of the partners in contact. The resulting surface will retain information about all the surfaces in contact and will be assumed to be representative of the actual state of the bearings' surfaces. This equivalent surface is supposed to be compressed by a smooth flat surface and the objective is to find the distribution of pressure and contact points that would come out due to the action of the radial load imposed to the bearing. This is done by simultaneously solving the equation for the pressure distribution along the nominal theoretical contact area and the equation for the distribution of the contact. This can be related to the movement of the centre of mass of one rolling body and can be repeated to cover all the extension of the rough profile. In this sense, one has a vector with the successive displacements made by the centre of mass of this element.

Construction of the excitation signal

Measurements according to the standard DIN EN ISO 4287 were made to cover the whole extent of the rolling path (inner and outer ring) and rolling elements. The single roughness profiles were then combined in the way shown in Figure 1. The surfaces of small length were repeated to cover the whole extension of the outer ring. The summation creates the equivalent rough surface that will serve as input to the physical model developed. The model furnishes the displacement, velocity and acceleration time evolution of the vibration produced in the bearing. This result, however, corresponds to the movement of only one rolling element. In order to produce the total excitation signal, one has to combine the influences of all rolling bodies. Assuming a stationary process and that the signal of all the rolling bodies rolling over the same surface will be a delayed version of the signal already calculated, one can obtain the final excitation signal by a combination of them (see Figure 1 (below)).

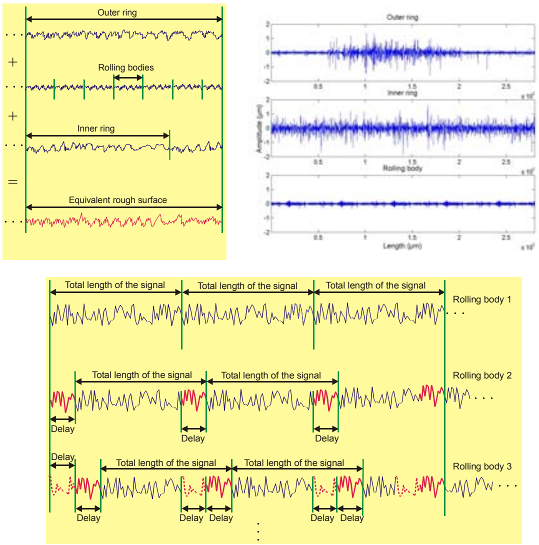

Figure 1: Scheme of the combination of the measured roughness profiles to form the equivalent rough profile (upper left). Profiles of a stationary outer ring and moving inner ring (upper right). Combination of the dynamic signal of all rolling bodies to form the time evolution of the excitation signal (lower).

Measurement conditions

Seven bearings were tested under different running times (see Table 1). At normal speeds and optimal lubrication, the degradation of the bearings would take more than 5000 hours. This is not practicable, so that other ways to speed up the degradation of the surfaces were used. Limited by the experimental apparatus to a velocity of 720 rpm and discarding the use of particle injection or extreme radial loads, the alternative to speed up their degradation was to run the bearings under non-optimal lubrication conditions.
Table 1 below resumes the tests performed with the ball and cylindrical rolling bearings.

Name	Radial load (kN)	Rotation (RPM)	Running time
Cylind. rolling bearing 1	32	720	9h 16’ 14’’
Cylind. rolling bearing 2	16	720	1h 13’ 01’’
Cylind. rolling bearing 3	16	720	4h 23’ 15’’
Ball bearing 1	16	720	0h 7’ 59’’
Ball bearing 2	16	720	4h 54’ 17’’
Ball bearing 3	16	720	1h 09’ 34’’
Ball bearing 4	16	720	0h 15’ 36’’

Table 1: Test conditions of the investigations with the 7 rolling bearings.

Discussion and Conclusions

The source modelled has a characteristic distribution of energy over the frequency. By the measurements, the influence of the vibration of all moving parts are present. Not to mention the temperature, responsible for example for the deformation of the races and a different distribution of load compared to the static case. Frequencies under 3kHz do not come from the rough contact in bearings, while the presence of lubrication on the 2 supporting bearings is, among other things, responsible for the extra damping and other dynamic effects that influence the results.

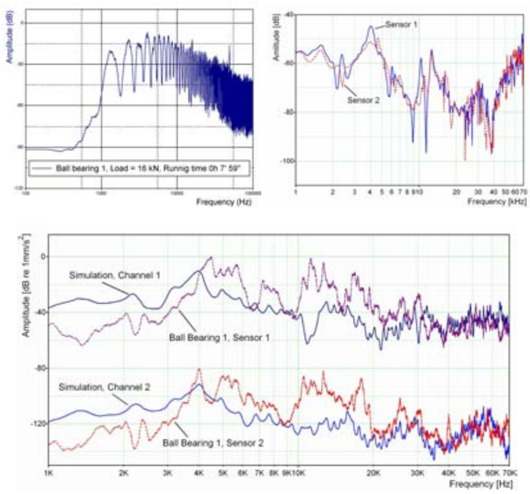

Figure 2: Excitation spectrum (upper left). Transfer function (upper right). 2-channel simulation and measurement (lower). The data have an offset to improve the visualisation.

This work presents some results of the auralisation of structure-borne sound of an experimental rotating machine concentrating on the identification of variations of the vibratory behaviour of different types of bearings in different states of degradation. Its use as a diagnosis tool for malfunctions in machines presents some challenging scientific and technological questions. At first, the difficulty to obtain the transfer functions from the source to the measuring point under operation conditions and secondly, the difficulty of separating and/or modelling all vibration sources in a real machine.

The results show that, in principle, it is possible to identify variations between the bearings and their influences in the corresponding spectra. However, further development is necessary to improve the determination of the transfer functions under more realistic circumstances, i.e., while running in normal operational conditions and to model, or at least, to separate the influence of all the vibration sources that contributes to the measured signal. Additionally, further measurements and simulation of lubricated bearings have to be made in order to correspond more closely to real operational conditions. The author is grateful and acknowledges the support of the Deutsche Forschungsgemeinschaft.

References

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[2] Guimarães, J.H.D.: Equivalent surface approach in modelling of rough elastic contact in rolling bearings, DAGA’03, Aachen, 2003.