Guimarães - Portugal
acústica 2004

RECENT DEVELOPMENTS IN AURALISATION OF STRUCTURE-BORNE SOUNDS FOR BEARING DIAGNOSIS

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ABSTRACT: In very early stages of operation of bearings, the monitoring techniques based on deterministic patterns on the vibration signal of a machine fail to deliver relevant information to access the condition of bearings surfaces. Further, the use of auralisation of structure-borne sound for diagnosis purposes has not yet been investigated in depth. Nevertheless, the acoustic signal turns out to show audible differences not only for the type of bearing under test, but also reflects the changing of the conditions of the surfaces in contact (inner and outer rings and the rolling bodies) with time.
This work presents the results of the auralisation of structure-borne sound in machines. These signals correspond to the vibration measured on the machine's housing generated by cylindrical and ball bearings under similar conditions and different running times. Previously measured transfer functions and a suitable physical model of the rough contact of surfaces, adapted to the case of bearings, furnish the basis for the simulation. This is compared with measurements in a experimental machine. The methodology used here shows to be sensitive to small variations of the state of the surfaces of bearings in early stages of operation and states an alternative for machine diagnosis.

1. INTRODUCTION

The majority of the techniques used in vibration analysis of machines is based on the identification of deterministic patterns on the vibration signal. In the case of bearing monitoring, those deterministic behaviours are closely connected to defects on races or rolling elements and the rotational frequency at which the machine runs (e.g. the impact produced when a rolling body rolls over a damaged bearing race). By the proper analysis of the vibration signal, it is possible to identify the existence and severity of a damage and to try to predict the remaining lifetime of the bearing.

However, there is also interest in assessing the development of the condition of the bearing before a surface damage, like a 'pitting', actually occurs. Before this occurrence, the changes in the vibration of the machine are very small and non-deterministic, however it should be assessed in order to make possible the evaluation of the actual condition of this machine element. More than that, it would be desirable to assess the development (change) of the condition of the surface and its influence on the vibration measured on the machine's housing. Once this influence is understood and correctly modelled, one can try to go the way back, i.e., to evaluate the bearing surface condition only through the analysis of the vibration of the machine.

It is well known that some persons are able to tell if a machine is ‘running well’ or not. Whether coming from the long time spent working with a certain machine or from the experience on mechanical venues, these persons can ‘hear’ a machine or use a screw driver gently touching his/her ears and the machine housing to ‘feel’ its vibration and say if something is wrong. This raises the questions: Would it be possible to simulate the structure-borne sound of a running machine and to verify differences in this signal due to different conditions of some of its part? Would this information be relevant and/or useful as a diagnosis tool?

To try to answer these questions, one needs to determine the transfer functions from the excitation point (the bearing itself) to a vibration sensor mounted on the machine surface (already described in [5]) and a proper physical model of the contact in bearings to calculate the time evolution of the excitation produced by this source [6, 7]. 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.

2. MEASUREMENTS OF THE TRANSFER FUNCTIONS

Figure 1 shows the assemblage for the measurements of the transfer functions and one example of the measured transfer function for one ball bearing at 16 kN of radial load.

Figure 1: *Schematic positioning of sensors and actuator (left). Built in actuators of ball - detail of fixing (centre) and measured transfer function for 16 kN of radial load.*

The first step to determine the transfer function in a mounted machine was to build a special transducer to substitute one of the rolling elements in the bearing [5]. The transducer was made with two piezo ceramics and was then built into the bearing and this was mounted in the machine (see Figure 1). In this way, it possible not only to extend the frequency range of the excitation but also to measure the transfer function with a mounted machine, which corresponds closer to the real situation of operation.

Under the effect of an applied voltage, the piezo ceramic responds with a force that excites the machine. This signal propagates through the machine and is captured by the two sensors positioned over its housing. With the use of sine sweeps as excitation signals and a rigorous calibration procedure, 34 different mounting positions (different angles over the perimeter of the bearing) of the actuator on the bearing under 4 different radial loads were measured for both cylindrical and ball bearing. In the applications described in the following sections, a mean transfer functions over the 34 positions was considered.

3. THE EXCITATION SIGNALS DUE TO ROLLING CONTACT

The physical-mathematical model developed uses the basis of the Hertzian theory of contact [3] adapted to the case of bearings. The details involved in this modelling were partially explored in [6, 7]. It has the advantage over the approaches of [1, 2] of using the information about the actual state of the surface instead of assuming a statistical distribution of asperities. This is important as the final goal is to accompany the degradation of the surfaces. Therefore, the model should be able to calculate the excitation due to every type of surface and not make prior assumptions about its roughness. The model also extends the approach, combining the surfaces to form the total excitation signal generated by this source as explained later in this paper.

Apart from the type, geometry and loading conditions imposed to the bearing, the model requires a rough surface as input parameter for the calculations. A computational package, called SAMBA (**S**tructural **A**coustic **M**odel for **B**earing **A**nalysis), calculates the displacement, i.e., the spatial movement in the contact area [6]. This is actually what is being searched as it represents the spatial displacement that results from the rough interaction of all contact partners, nominally, the races and rolling bodies.

The displacement signal (d) represents the equilibrium position of one imaginary flat surface that compresses an equivalent rough profile. This profile, in its turn, is the combination of the measured rough surfaces that are put in contact. The position of this imaginary surface represents, therefore, exactly the equilibrium position that would be found if the rough surfaces would be put in contact. As the calculation is done throughout the whole profile, the result is a successive state of equilibrium reached by the surfaces when their asperities interact with each other. Furthermore, as d is related to a fixed reference, it describes the movement of the centre of mass of the rolling body due to its elastic interaction with the rolling races. Considering also the velocity at which the matching surfaces touch each other, one can create a time signal that represents the evolution of the displacement. By direct derivation one obtains the velocity and the acceleration signals that finally represents the excitation signal imposed to the machine due to this specific source.

It is important to notice that the excitation signal generated corresponds to the movement created by the elastic deformation experimented by the rolling elements and the races and does not considers their rigid body movement caused by the dynamic behaviour of the axis, geometry of the bearings, clearances etc. In this sense, the modelling intends to describe how is the excitation imposed by this specific source (interaction of elastic rough surfaces) to the machine.

3.1. Creation of an excitation signal

To obtain the roughness profiles, measurements according to the standard ISO 4287 were made to cover the whole extension of the rolling path (inner and outer ring) and rolling elements. The roughness profile of the rolling element and inner ring were repeated to cover the whole extension of the outer race and then all the profiles were summed in order to make an equivalent rough surface whose length is equivalent to one complete turn of the rolling element over the outer ring (see Figure 2) [6, 7].

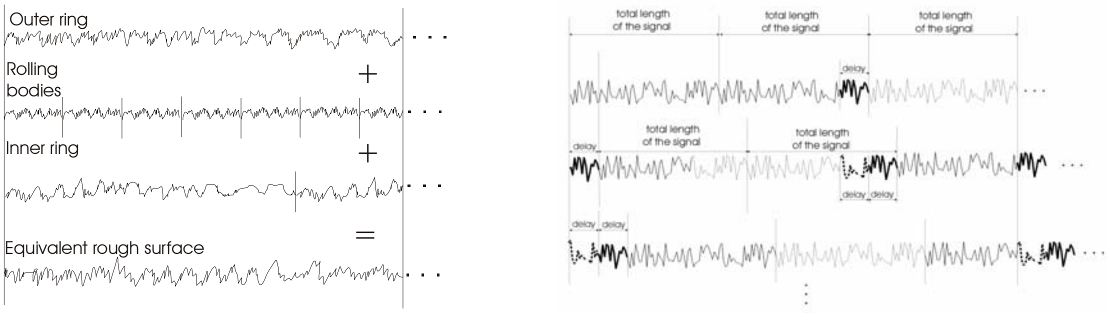

Figure 2 - *Creation of the equivalent rough surface(left) and combination of the signal produced by each rolling bearing(right).*

This equivalent rough surface can be used as input to the model and results on the acceleration signal produced by one rolling element in contact with inner and outer rings.

If no slip or spinning is considered, the velocity of the rolling element and of the inner ring is the same at the point of contact as the outer ring remains fixed. Provided the dynamical compatibility between moving parts is maintained, it is easy to prove that [4]

$$n_R = \frac{1}{2}\frac{d_m}{D} n_i \left(1 - \left(\frac{D}{d_m}\right)^2\right) \qquad \text{RPM .} \quad (1)$$

The pitch diameter d_m is the distance between the centre of the rolling bodies, D is the diameter of the rolling body and n_i and n_r (in RPM) are the rotational velocities of the moving inner and outer ring, respectively. The velocity at which the surfaces will match, is finally $v = n_R D/2$. For a shaft rotating at 720 rpm the value of v is 224.19 mm/s for the ball bearing and 194.11 mm/s for the cylindrical bearing.

It is important to notice that the excitation signal calculated describes only this vibration source and represents one complete running of one rolling body over the entire length of the outer ring. One can further assume that the measured rough profiles are representative of the state of the surface so that the signal produced by the contact of the other rolling elements is a delayed version of the previous one. In this way, knowing the number of rolling elements, one can sum up delayed signals and construct the total excitation that is imposed to the machine by this source. Figure 2 above resumes the process.

4. MEASUREMENTS AND RESULTS

Table 1 below resumes the tests performed with the ball and cylindrical rolling bearings and the corresponding running time and operation conditions.

Table 1: *Test conditions and running time of the investigations with the 7 rolling bearings.*

Name	Radial load (kN)	Rotation (rpm)	Running time
Cylindrical rolling bearing 1	32	720	9h 16’ 14’’
Cylindrical rolling bearing 2	16	720	1h 13’ 01’’
Cylindrical 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’’

Under normal operation conditions and optimal lubrication, the degradation of the bearings would take too long (over 5000 hours of nominal life). In order to speed up the degradation of the surfaces it was decided to run the bearing under non-optimal lubrication conditions, leading to fast enough alteration of the surfaces and allowing the test of various bearings. The final aim was to try to run the bearings until different conditions and to see if these alterations would lead to sensible changes on the vibration signal and on the simulated excitation signal. They should, however, not exceed the limit where material removal occurs, as the main interest lies actually before such phenomena take place. For them other diagnosis techniques are available. The physical model was also adapted to this special case of dry contact of surfaces.

From the total running time of the cylindrical rolling bearing 1, 6h 56’ 38’’ where with a little bit of oil and taken under different radial loads (16 and 32 kN) as test. The rest of the time was run without oil. Nevertheless, it should be considered as an exception as during the tests, it was noticed that the cage was a little bit damaged and brass particles coming from it were found on the races when it was dismounted for the roughness measurements. The ball bearing 2, after dismounted, showed ‘pittings’ over its load zone. The material removal was so strong that the attempt to measure its roughness could damage the measuring needle. It was, therefore, discarded. The other bearings where in what can be called ‘run state’ and their surfaces could be measured and the data used for the simulations.

All of the bearings lacked oil during the tests. The tests were performed either with the small amount of oil used in the package to avoid oxidation (the case of the cylindrical rolling bearings) or they where totally cleaned and some drops of machine oil where added before assembling them (the case of the ball bearings). No further lubrication was provided during the tests. It means that the amount of oil provided was not enough to provide a constant and thick enough lubrication film for safe operation without metal-metal contact. However it provided enough lubrication to avoid direct metal-metal contact during running-in and during the first small plastic deformations of the material of the races and rolling bodies that occurs in the initial operation.

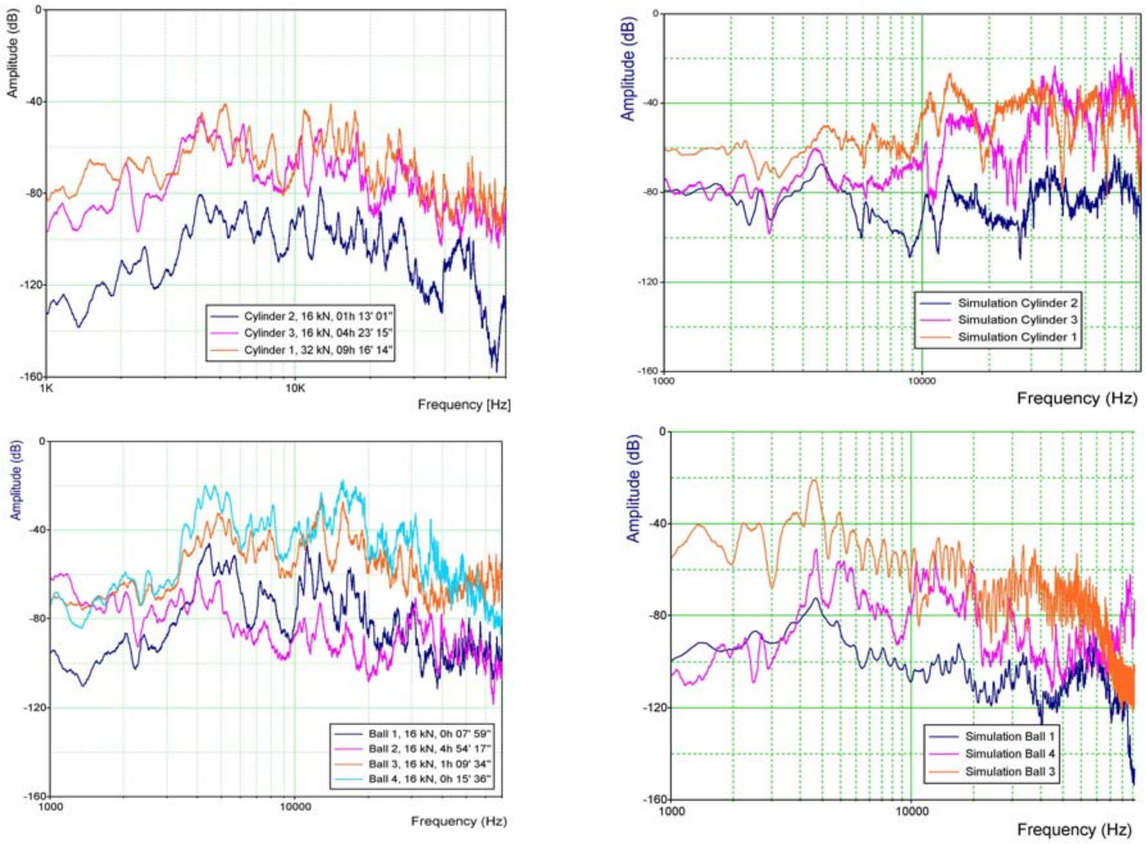

Figure 5: *Measurements (left)and simulations (right) and for 720 RPM and 16 kN radial load.*

It is difficult to describe the state of the surfaces by visual inspection. One notice that the races have been used and altered and that the effect of temperature in the contact was big as the colour of the metal parts have changed, also due to the burning of the small amount of oil. However, only the measurement of its roughness can indicate the difference between them.

Figures 5 show the results of the measurements together with the simulations of ball and cylindrical bearings.

5. DISCUSSION

One important aspect to be noticed is that the transfer functions where measured under constant temperature (around 22^{o}C) and in non-operating condition. The tests, however, where taken with the machine in movement at a constant rotation. In this condition, without lubrication, the temperature increases very much and has reached values of $40^{o}C$ on the extern part of its housing (meaning that on the bearing it was much higher than that). The result is that the races deform and the same situation of contact and shape of races present during the measurement of the transfer functions are not more attended. It can be seen in the graphics as a displacement of some peaks toward higher frequencies. That is also why some differences on the position of some peaks are noticed. This is especially remarkable in the ball bearings, where the area of contact is smaller and therefore more susceptible to variations and deformations of the races due to the influence of temperature.

The general tendency of the simulations follows well the behaviour of the measurements. This is an indication that the signal of this is specific source was predominant among other sources of noise. Firstly because of the lack of lubrication, implying a high level of vibration of this bearing compared to the other bearings and other sources present on the machine. Secondly, because the measurements were made directly over a special case made to hold the test bearing and to permit the application of the radial load. This special case is independent of the machine housing and moves in relation to it.

Nevertheless, one should notice that the measured signals represent all the possible sources of noise and all vibratory effects possible acting together and being captured by the sensors. Therefore, effects of damping, amplifications and influence of other sources are not considered on the physical model, but can be present on the measurements. This can explain, for example the deviations between simulations and measurements on the higher frequency range, noticed in some of the tests performed and the existence or lack of some resonances. Additionally, the presence of two lubricated symmetrical spherical roller bearings as support bearings is responsible for the general damping of the measured results and contributes with other characteristic frequencies.

As mentioned before, the signals presented here are the result of the convolution of the excitation coming from the bearing with the measured mean impulse response. Like the measured signal, they represent the structure-borne sound taken directly on the housing of the machine. That is why, if these signal are made audible, it would not correspond to the real auditory sensation of a noisy machine, as in this last case, one has also the influence of the ambient. They represent the audible frequency content of the vibration of the machine. However, hearing the structure-borne sound also brings very interesting information as differences between the bearings are also present in the audible signal.

All the signals were low-pass filtered and down sampled to 44100 Hz and then transformed to WAV files. By carefully analysing the data, one can notice the clear difference between them concerning the different stages of degradation.

6. CONCLUSION

The methodology presented showed that this method produces coherent results. Simulations were able to reproduce the principal behaviour expressed by the measurements, showing the usefulness of this methodology in recognising vibration features of machines.

Alterations due to running conditions explain the deviations between measured and simulated signals. Nevertheless, further development concerning modelling should be undertaken to refine the results and include other non-modelled phenomena.

Further attempts should also consider tests with different running conditions such as rotational frequency and radial loads. Also, different artificially created conditions of the surfaces could be tested in order to see its influence on the simulated and measured vibration signal. More tests are needed to exactly determine the potentiality and usability of this technique in the machine diagnose field in the assessment of other kinds of equipments, but these first results shows that, at least under experimental conditions, the methodology presents promising results. It is still to be developed a methodology to solve the inverse problem, i.e., the determination of the bearings condition throughout the measurement of the machine vibration and the use of the transfer function.

Also, as final remark, the possibility to auralise the vibration of a machine and the transformation of this signal into sound examples are spotted. From them, come many questions to be investigated in future works.

ACKNOWLEDGEMENT

The author acknowledges the support of the German Research Foundation (DFG).

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