Equivalent surface approach in modelling of rough elastic contact in rolling bearings

João Henrique Diniz Guimarães
Institut für Technische Akustik, RWTH Aachen; Email:ique@akustik.rwth-aachen.de

Introduction

Rolling bearing failures are one of the major causes of machine break down. Traditional diagnosis methods are based on the measurement of vibration on the housing of the machine under running conditions. Throughout correct processing and interpretation of the signals, existing defects in races and rolling bodies can be reasonably identified. However, in many cases, early diagnosis and determination of the developing of the wear of the bearing surfaces is desirable.

The knowledge about the mechanisms of generation of vibration in machines is important for condition monitoring and diagnostic. One of the causes of vibration in rolling bearings concerns the interaction between its moving parts. The contact established between the mating surfaces is responsible for the excitation signal imposed by this specific source to the machine. This excitation signal will propagate through the machine and can be measured by accelerometers on its housing.

The final aim of this study is to try to correlate the vibration of the machine with the condition of the bearing. To do so, two things are necessary: the transfer function between source and receiver, described elsewhere [[1]], and a physical modelling of the contact between rolling elements and bearing races. This work deals specifically with the last one.

Theoretical background

Rough contact

Classical theory of contact (Hertz theory [[2]]) models the interaction of two compressed surfaces as a flat contact, where a parabolic distribution of pressure takes place. However, in the case of rough surfaces, the contact is not more distributed over the whole nominal contact area, but is concentrated on some points, i.e. the tips of the touching asperities (Fig. 1). At the points where the contact occurs, the pressure can be very different from the one predicted by the classical theory.

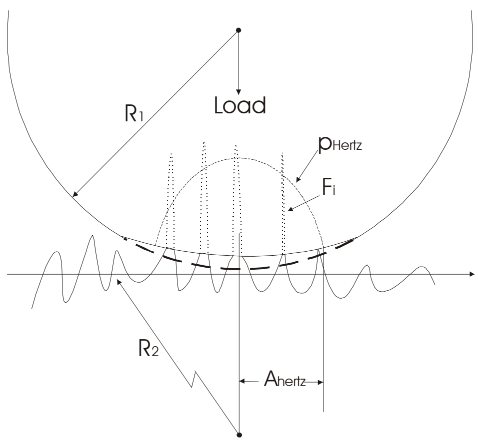

Fig. 1: Rough contact between surfaces.

Nevertheless, due to its small size, it is assumed that the contact at the top of each asperity follows the Hertz theory and the forces (F) and areas (A) at a point **i** are given below in eq.1 and 2:

$$F_i = \tfrac{4}{3} E' r_i^{1/2} \delta_i^{3/2} \qquad \text{eq. 1}$$

$$A_i = \pi \, r_i \, \delta_i \qquad \text{eq. 2}$$

E' is the composition of the elasticity modulus of the metallic surfaces in contact, r_i and δ_i are respectively the radius and the elastic displacement of the i-th asperity.

The equivalent surface approach

This procedure consists in the creation of a rough profile that is the sum of the profiles of the surfaces in contact (Fig. 2). This new profile, at its turn, reflects the situation of the contact, as the higher points (higher height asperities) represent the places where the contact will be established.

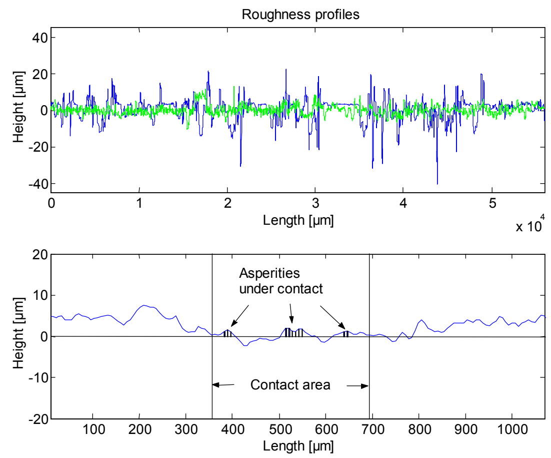

Fig. 2: Combination of roughness profiles showing the compressed asperities within the nominal contact area.

The calculation procedure considers that, within a certain area of integration, the asperities above a certain reference will be compressed by a flat surface. This area of integration represents the area where the contact would possibly be established. It can be different from the nominal contact area predicted by the Hertz theory and depends on factors like the roughness and the curvature of the surfaces in contact. The actual contact area is not known a priori, so that the calculation of the distribution of pressure and area must be done simultaneously, as they are dependent of each other. The sum of the F_i's must clearly be equal to the total load applied to the body. Through the repetition of the calculation within the whole equivalent profile, one obtains the elastic displacement of the contacting surfaces.

The rolling process

As previously mentioned, within the nominal contact area, only some points are in contact. For a given bearing geometry, type and rotational frequency one can calculate the velocity at which the surfaces match to each other and construct a velocity as well as an acceleration vector. These represent the excitation signal that is imposed to the machine due to the contact of rough surfaces.

For easiness of handling, one hypothesis also assumed by [3,4] is that this contact points are randomly distributed within this contact area and that the contact is developed in a quasi-stationary way. Assuming also that the height of the asperities follow a Gaussian distribution, one can calculate statistical quantities that are representative of a Gaussian rough profile. These parameters are the surface density of summits $\boldsymbol{\eta}$, the standard deviation of the probabilistic distribution of summit heights $\boldsymbol{\sigma}_s$ and the deterministic radius of the mean spherical summit caps **r**. In this way one has an easy-to-handle way to describe roughness and retains information about number, distribution and shape of the asperities. Further assuming that the mean duration and force amplitude at each contact are the same and that each contact works as an elastic incomplete impact, the interaction between the surfaces can be thought to be a flow of binary states (contact/no contact). The Power Spectrum of this flow of impulses has the form of a sinc function (sin(x)/x). The interdependence of the parameters involved in the sinc function is complex. They relate the mean amplitude and duration of the statistically distributed contacts to the rolling velocity, total load applied to the bearing, shape of the rolling bodies and surface profiles and the material characteristics.

Results

Fig. 3 shows the power spectrum of the velocity on the surface of contact for two different rough profiles. The second one was low pass filtered in order to simulate the influence of a better superficial finishing. It is seen that the effect of the higher asperities, responsible for part of the high frequency content, is diminished.

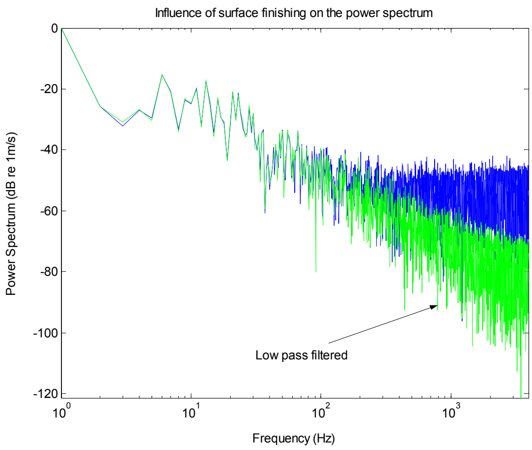

Fig. 3: Power spectrum of the velocity of two different rough profiles.

The graphic below (see Fig. 4) shows the power spectrum of the statistically distributed load flow mentioned in the previous section. It compares the situation of a cylindrical and a spherical rolling body under the same load (8kN). As expected, one can see that the level is higher for the sphere as the load is distributed in a smaller area compared to the cylinder, where the length also plays a role. Considering the frequency content, it is observed that the position of the first minimum is also lowered for the cylinder.

When considering one type of rolling body ate once, it is possible to analyse the influence of the other parameters to the power spectrum of the statistical distribution of asperities, i.e. on the shape of the sinc function shown in Fig.4. The increase of the load applied to the bearing and rolling bodies increase the amplitude, but lowers the position of the first minimum. This can be explained by the fact that the asperities interaction time increases due to the increase of the real contact area. The greater correspondence between contacting surfaces increases the number of asperities in contact.

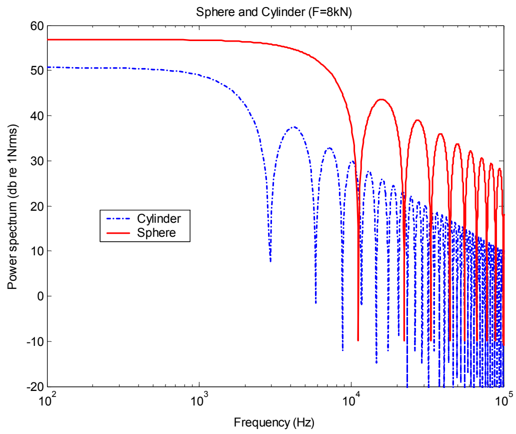

Fig. 4: Power spectrum of spherical and cylindrical rolling body under the load of 8 kN.

The increase of the radius of the rolling body leads to a decrease in amplitude and frequency content because the curvature of the surfaces as well as the bulk deformation influences the distribution of points in contact. On the other hand, the increase of the cylinder length only shows lower amplitudes, but no change in frequency content. Despite the greater area to distribute the load, the mean time of interaction of the asperities is not altered. Finally, when the modulus of elasticity is altered, one can see a decrease in level and on the position of the first minimum. It works as if a softer material is inserted between the surfaces in contact, which can be thought to be a lubrication film between them. The explanation is that the presence of lubricant reduces the influence of the low-lying asperities.

Discussion and conclusion

This approach showed to be useful to describe the contact of elastic surfaces. The statistical distribution of asperities shows a good compromise between easiness to handle and precision of approach. However, it should be noticed that not every surface has a rough profile that follows a Gaussian distribution of heights. The relationship between the parameters involved is complex, however the results reproduce well the expected results. The calculation of every detail in a moving contact patch is highly sensible to the input rough profile, due to its random characteristic and non-repeatability. Nevertheless, the final interest of this work is also to obtain the exact excitation signals occurring at the contact between the surfaces. These are to be convolved with the transfer functions and compared to the actual vibration signal measured on the surface of the machine [1]. In this sense, the determination of the actual excitation signal is important, or at least any statistical time signal representative of the excitation imposed to the machine due to this specific source. A deeper study of the excitation signal is in course. Further development also involves the physical modelling of plastic deformation of asperities and the presence of lubricants, since they correspond closer to the actually operation condition in bearings. The author is grateful to DFG for the support to this project.

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[1]| Guimarães *et al.* Forum Acusticum Sevilla 2002

[2]| Johnson, K. L. Contact Mechanics, 1985

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