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Dependency of bearing noise properties on surfaces lubrication

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1. Introduction

Noise of technical equipment usually assumed to be negative effect. Noise affects personnel and brings its negative effect of operating machinery and equipment. Technical equipment with rotational movement has different types of bearings, which are known as technical noise generators.

Modern materials in bearings create some specific problems with noise intensity and this paper touches some aspects of noise generation in steel and ceramic bearings. To make research of such task it is necessary to built a model of noise generation and to find parameters of noise generation mechanism.

Noise of bearings is created by vibrating solid bodies, which is transmitting further to air. It is known [1] that sound filters solid body vibrations due to its own physical properties of wave media.

Vibration of bearing components has different nature, but for operable ones the main part of noise in 2 – 5 Hz area belongs to friction noise [2-8].

2. Aim and formulation of research

This research is intended to find dependency between bearing material hardness, lubrication of bearing and vibration characteristics as well as acoustic noise.

The research presented consists from several parts. The first step is to accept model of friction noise generation in a bearing, based on tribologic properties of bearing surfaces and other sources of vibration are neglected. This statement is used only in case of experimental setup; in real machinery it can't be used directly. Noise generation hypothesis is presented in Fig. 1. There are two cases of vibration generation - when two surfaces are sliding (as shown in Fig. 1, a), and direction of surfaces movement is opposite in tangential direction or when the surfaces are rolling and direction of surfaces movement is coincident in tangential direction.

In this paper only rolling mode is assumed.

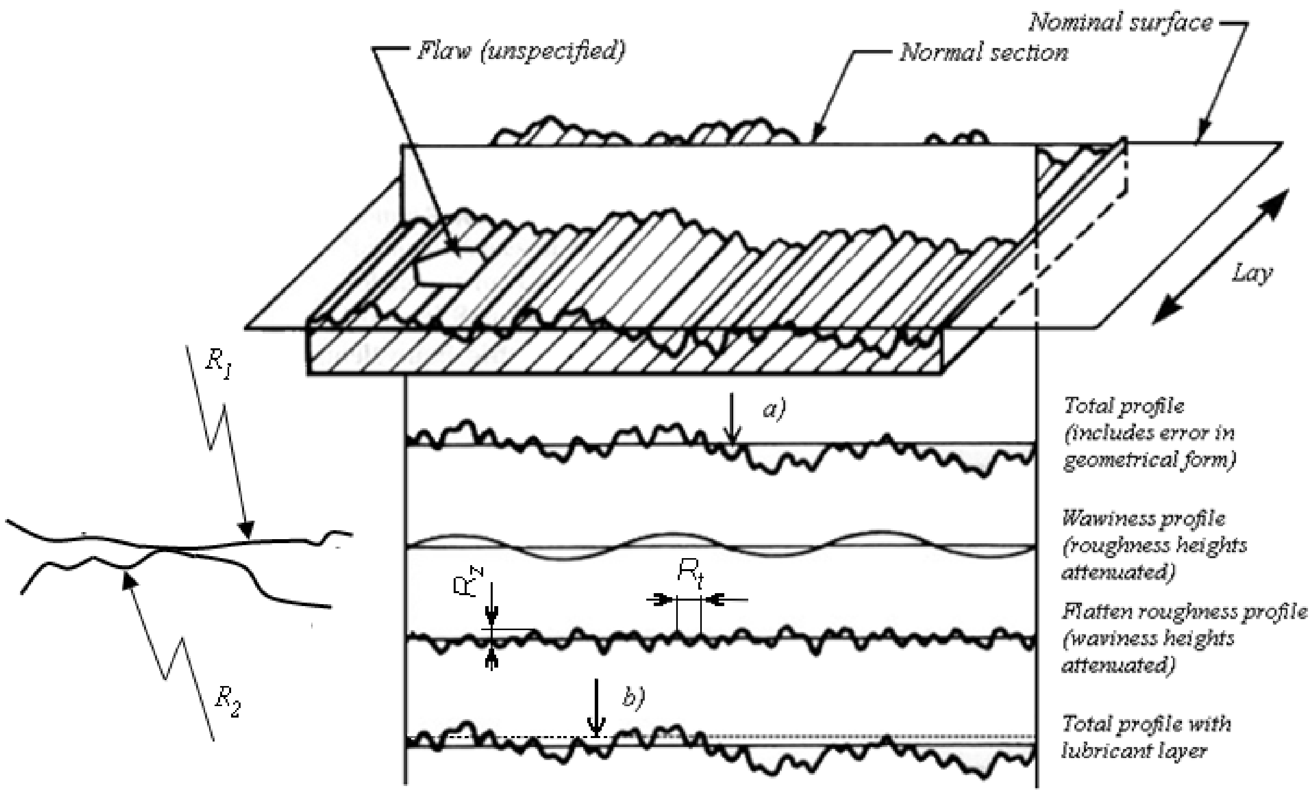


Fig. 1 Model of friction noise generation: a) surface roughness interaction, b) lubricant interference to average roughness pitch and noise frequency, R_1, R_2 – radii of contacted surfaces, R_t – average pitch of surface roughness, R_z – average height of surface roughness

Surface of contact area are random peaked and only statistically evaluated surface pitch R_t and average roughness R_z are available to evaluate in this model. So, piece of some surface projection, showed on Fig. 1, b can be accessed directly by another surface to support force, or support surface is modified by lubricant film, which is represented by two positions of oil surface as “thin oil” and “thick oil”.

These surfaces are interacting with peaks, “sticking out” from lubricant film with their own R_z and R_t.

This creates different conditions in excitation of vibrations when the surfaces are moving in coinciding tangentiall directions (rolling) or in opposite tangentiall directions (sliding).

Then for a single surface cross section is possible to write the following dependencies

$$f_{sl} = \frac{\omega_1 R_1 + \omega_2 R_2}{R_t} \tag{1}$$

where f_{sl} is average statistical frequency in sliding; ω_1, ω_2 are angular speed of surfaces; R_1, R_2 are radii of rotation for these surfaces; R_t is equivalent pitch for a roughness.

In case of rolling average theoretical frequency can be expressed as

$$f_{sl} = \frac{\omega_1 R_1}{R_t} = \frac{\omega_2 R_2}{R_t} \tag{2}$$

Dependencies (1) and (2) are very rough estimation of kinematical excitation in contact of friction noise, which separates modes of movement of contact surfaces cross-sections. Real frequencies are generated from real surface relief, so the number of frequencies in huge, but center values follows this model [8 - 9].

This model has also more modifications, but the main ideas can be accepted also in slightly simplified form [10 - 11].

3. Modelling of contact

In order to look on contact area behavior, FEM model of one side bearing contact, which allows to evaluate distribution of stresses in contact and shows active area in noise generation was created.

As the basis for this model a bearing No. 208 was taken, but the model can be used in other applications. Contact model was created using SolidWorks software and FEM analysis was performed on CosmosWorks 2010. FEM model (Fig. 2) was created using one fragment from bearing No. 6208. The fragment of outer ring was taken with angular size, which corresponds to another neighboring ball contact places. For contact modeling here was chosen contact type “surface –surface”, while in the model little gap was created and bonding place was unknown. Because of ball shape, restrictions in both horizontal axis directions were fixed by creation axis, only vertical ball movement was allowed. Outer ring fragment was fixed in all 3 axes direction on the outer surface. In order to run such model, special stabilizing spring was used.

All elements were taken as 1st order tetrahedron, contact was assumed to be Coloumb’s. Load of the ball was applied in the area of estimated contact area of other ring, size – 80% of bearing radial load, according the shape of free body diagram.

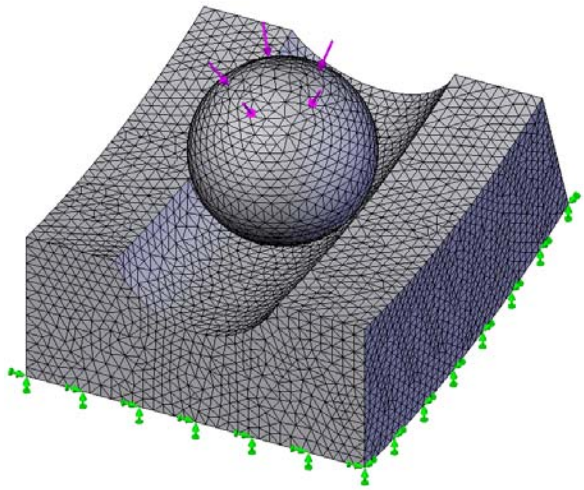


Fig. 2 FEM model outer ring fragment and ball of the bearing

Solution of this task is presented in Fig. 4, where inner surface of the bearing and ball contact surface are shown. Configuration of contact on ring running surface and ball are slightly shifted from the axis of symmetry due to global friction in the contact.

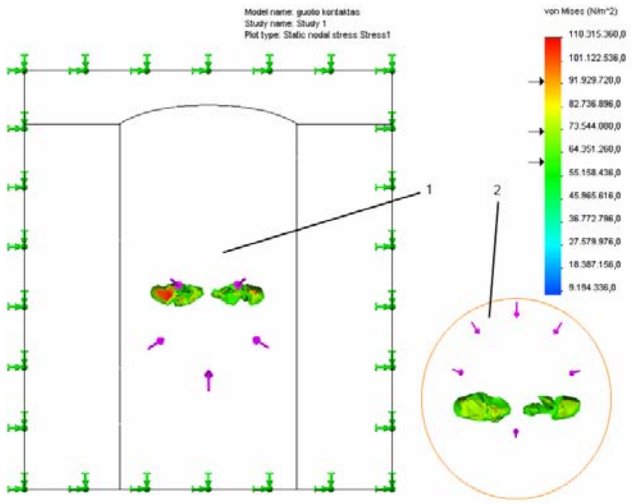


Fig. 3 Contact stress distribution in the outer ring (ball is removed), when the load is 35% of radial load: *1* – outer ring fragment, *2* – ball

The analysis of stresses and displacements shows slight change in stresses in the area contact proves model of friction noise, because great gradient in stresses would create another scenario in vibration excitation.

4. Methodology of research

This experimental research was performed in Braunschweig technical university (Germany) for ceramic bearings and in Hannover Leibnitz technical university (Germany) was made extensive steel bearing research.

Initially contact area of the bearing was measured for roughness. Profile of contact, which is shown in Fig. 4, was evaluated for the main parameters and statistically proved values or R_t and R_z were defined. These values were basic in the definition of desired noise frequency range. It is necessary to take into account, that statistically proved data on surface of contact differs within certain tolerance and phase of vibration from touching roughness profile is also unknown. The frequency of resulting vibration from outer ring of the bearing was recorded and further analyzed. Spectrum of vibration accelerations is a result of such research and the main information supplier.

In case of nonlubricated bearing, ceramic bearing with very similar surface parameters was tested in another test rig. This test rig was intended to record acoustic pressure. Duration of the test in this test rig was very short in order to avoid heating. Partial axial load was applied, because installation of ceramic bearing was not tightening enough the outer ring and there was internal gap. Loads on ceramic bearing and steel bearing were made proportional to their maximum load (80% of it) and axial load was assumed not influential.

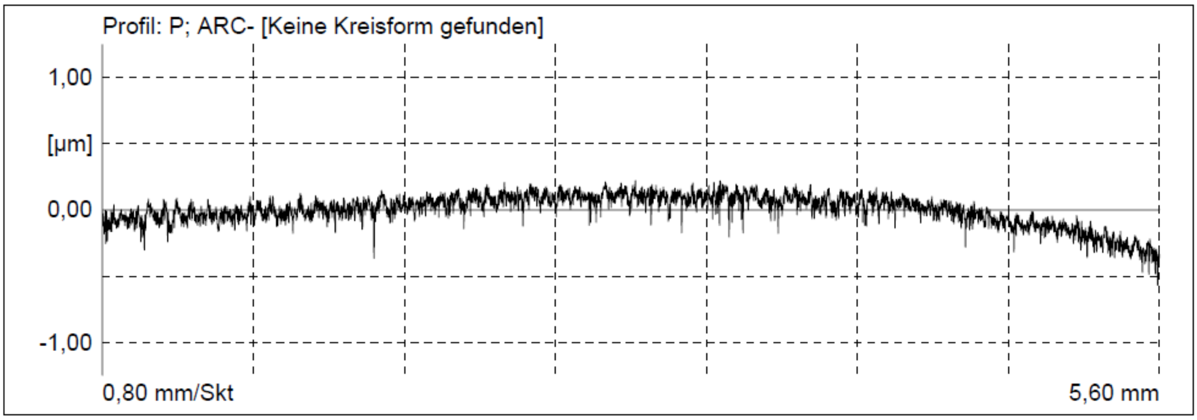


Fig. 4 Bearing internal surface profile: Ra = 0.04µm, Rz = 0.41 µm, Rq = 0.05 µm, Rp = 0.12 µm, Rt = 0.52 µm

Bearings (steel and ceramic) were built into special setups and rotated with corresponding load and rotational speed. Because of different design of setup and bearing size, rotational frequency was taken so, that linear rotational velocity of the rolling bodies should be the same – 600 rpm for steel bearing and 1000 rpm for ceramic bearings.

5. Results

Results of performed research are shown below graphically. Fig. 5 shows steel bearing vibration acceleration signal in time. This sample was lubricated by industrial lubricant t9.

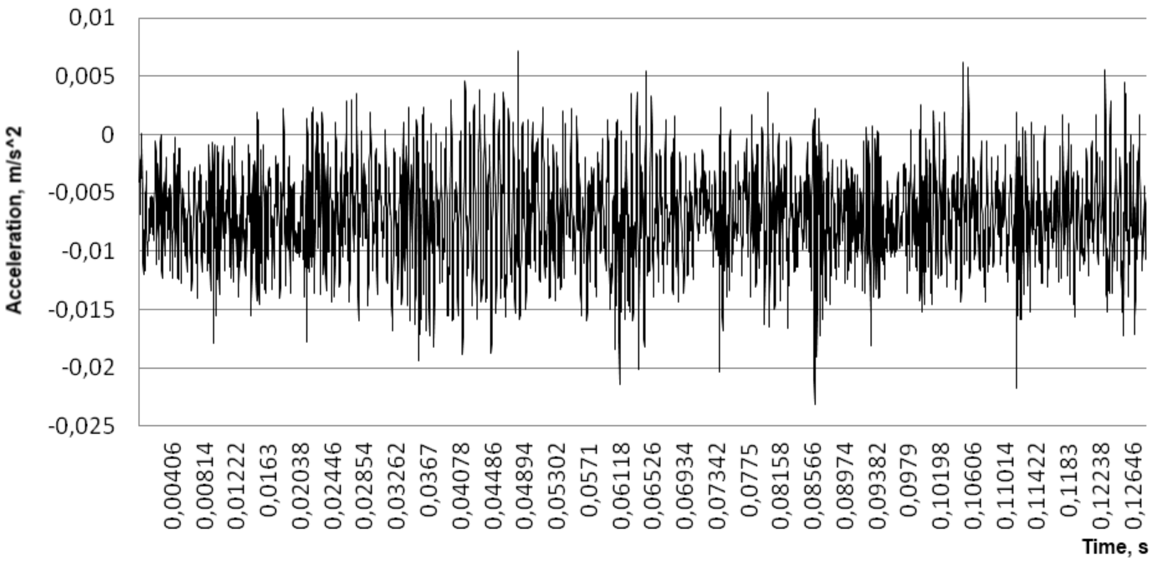


Fig. 5 Vibration data of steel bearing, lubricated with t9, shaft rotation 600 rpm

Vibration signal output from the steel bearing was recorded; signal discretion is 0.00002 s, which corresponds 50 kHz of sampling rate. From such signal frequency spectrum was calculated using FFT. As it is seen from vibrational spectrum (Fig. 6), the main frequency peak is about 3000 Hz and 7000 Hz as the second harmonics and low ½ subharmonic on 1500 Hz. Higher frequency range in spectrum has low accuracy because of sampling rate. These frequencies do not fit to ball pass or inner bearing ring revolution rate.

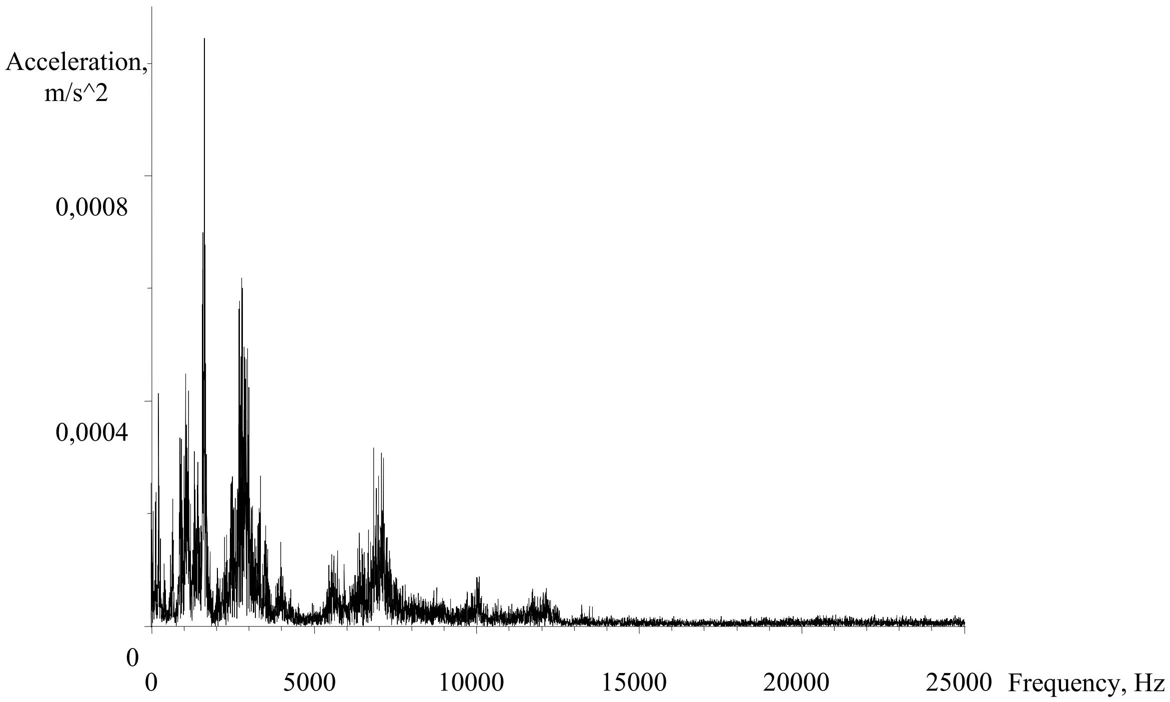


Fig. 6 Spectrum of steel bearing 308 vibration accelerations, bearing lubricated with t9 lubricant, radial load 150 N, rotational speed 600 rpm

In case of lubricant t68 (Fig. 7), the spectrum was shifted to higher side and the values of vibration amplitudes are significantly higher, what means lower film thickness, as stated in initial model. The main frequency range shifted to 2600 Hz, correspondingly the second harmonics to 7200 Hz. During these tests lubricant film thickness was not measured, but the frequency change for more than 10% shows that the number of sticking roughness peaks correspondingly increased about the same number.

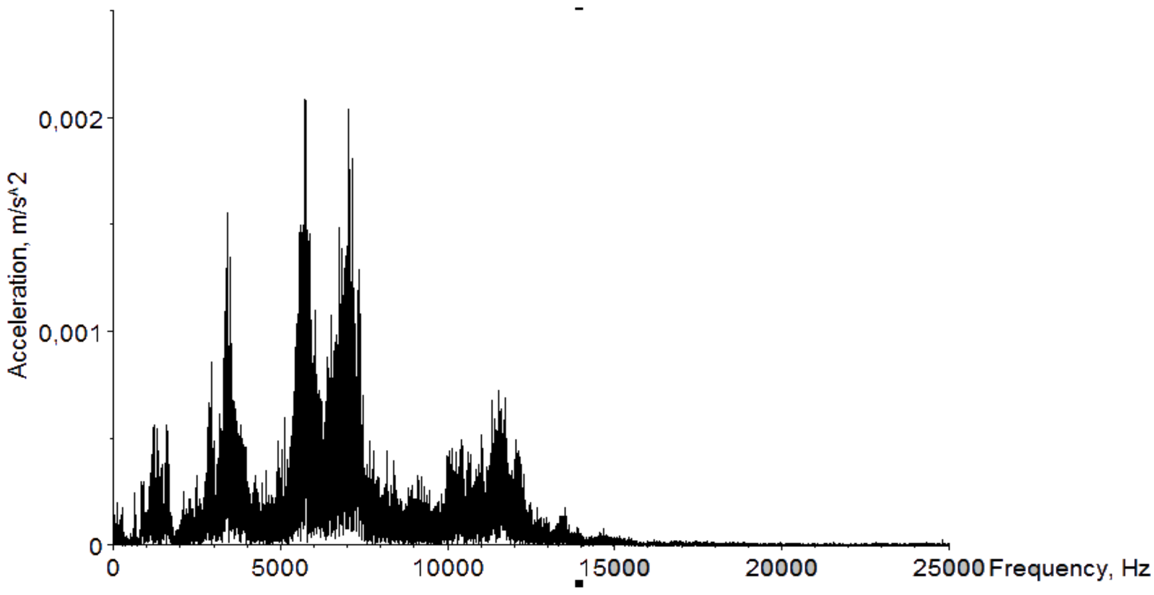
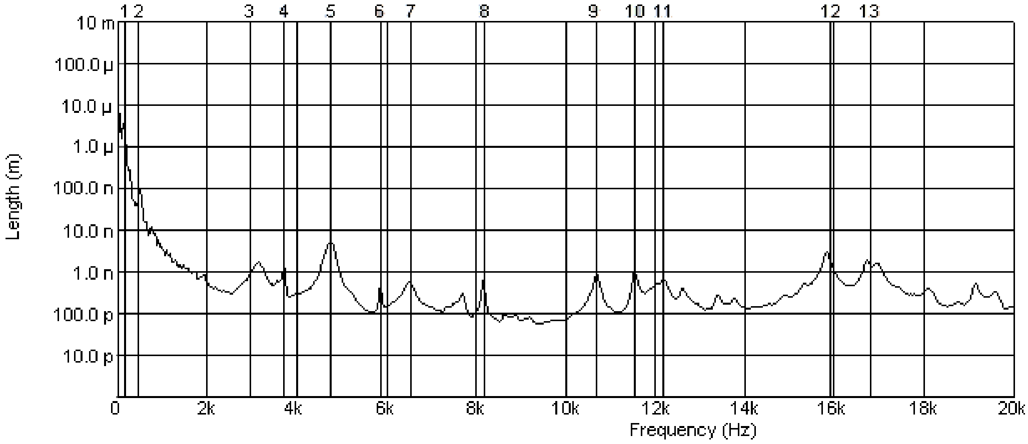
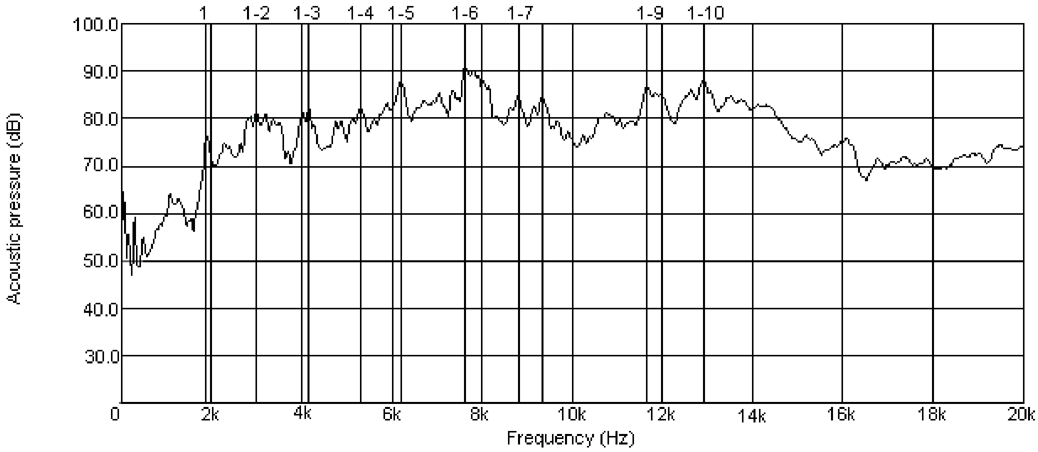


Fig. 7 Spectrum of vibration of steel bearing lubricated with t68, radial load 150 N, rotational speed 600 rpm

In case of ceramic bearing analysis (the measuring of outer ring vibration was made using laser sensor) another software was used. As it is possible to see in processed spectrum, ceramic bearings create much higher values of vibrations (Fig. 8, a) and correspondingly higher acoustic pressure. Higher level of vibration amplitudes is caused due to higher hardness of bearing material. Absence of lubricant in contact enables to touch much more peaks in contact area and energy of elementary impact between roughness peaks makes much higher.



a



b

Fig. 8 Vibration amplitude a) and noise b) of ceramic bearing test: material Si_3N_4-TiN^9, Ra = 0.1, 1000 rpm, radial load – 300 N, axial load – 25 N

Result of such investigation evidently show that more accurate model for vibration frequency evaluation is necessary, because the presence of prescribed frequency in the range of 3000 Hz and below has coinciding sub harmonics and higher harmonics.

6. Conclusions

After analysis of different types of bearings, it is possible to state, that material of a bearing and lubrication influence on bearing vibrations and bearing noise is significant. Harder materials for bearings, such as ceramics, without the use of lubricant increase noise values and vibration energy and only soft media between ceramic parts can make effect of vibration energy decrease. Correspondingly, in case of soft media, like thick oil film, increase resistance moment in the bearing and cause higher heating. In case of oil film breakage, dry friction due to dry surface increase energy of vibration (expressed in acceleration amplitude shift to higher frequency range) and higher values of local stresses (contact in surface peaks) develop heat and surface damage. As particular conclusion it is possible to state:

1. Difference in dynamic characteristics of bearing vibration in the same surface interaction velocity with bearings from different material is defined by surface hardness and particularly rolling/sliding values.

2. Lowering of bearing vibration and noise with different lubricant properties are directly influenced by lubricant film efficient thickness and surface “sticking roughness”, load in the bearing can be expressed in the terms of efficient film thickness;

3. Rolling and momentum sliding mode of rolling elements inside bearing is characterizing by vibration spectrum frequencies significantly.

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PAVIRŠIŲ TEPIMO ĮTAKA GUOLIŲ KELIAMAM TRIUKŠMUI

R e z i u m ė

Straipsnyje pateiktas lyginamasis eksperimentinis skirtingos medžiagos ir skirtingai tepamų guolių keliamo triukšmo ir virpesių charakteristikų tyrimas. Specialiuose eksperimentiniuose stenduose buvo tiriami plieniniai ir keraminiai vienodai apkrauti rutuliniai guoliai, išmatuoti jų keliamų virpesių ir akustinio triukšmo parametrai ir nustatyta keliamų virpesių parametrų priklausomybė nuo paviršių tepimo bei tepalo medžiagos. Tyrimo metu nustatytas virpesių dažnio mažėjimas tepant tirštesniu tepalu. Iškelta ir patikrinta tribologinė reiškinio prigimties hipotezė.

Gautas virpesių dažnių spektro juostos apie 2 kHz poslinkis dažnių žemėjimo kryptimi, tepant tirštesniu tepalu, rodo pradinės prielaidos pagrįstumą. Palyginamasis akustinis keraminio guolio triukšmas spektras, esant analogiškam sausam guolio riedėjimo paviršiui ir toms pačioms darbo sąlygoms, turi dar aukštesnius dažnius ir didelį triukšmo intensyvumą iki 20 kHz dažnio.

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DEPENDENCY OF BEARING NOISE PROPERTIES ON SURFACES LUBRICATION

S u m m a r y

This paper presents the investigation on bearing noise and vibration characteristics, when they are made from different material and lubricated with different materials. Experimentals research was performed on special workbenches, where ceramic and steel ball bearings were tested with equal loads and rotational speed. Parameters of their vibrations and acoustic noise were measured and dependencies of noise or vibration frequency lubricating material were found. There was found the decrease of resulting frequencies when thick oil was used. Hypothesis of tribological nature of the effect was raised and verified.

Decrease of resulting frequencies in vibration spectrum of about 2 kHz when thicker lubricant was used proves initial prescriptions. Noise of ceramic bearing without lubrication and with similar surface parameters and testing conditions is generated in higher frequencies and high noise intensities are observed up to 20 kHz.

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