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THE INFLUENCE OF LUBRICANTS ON THE DURABILITY OF ROLLER BEARINGS

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Abstract: *The paper describes the determination of various factors lubricants that affect the durability of rolling bearings or general rolling contact. Their knowledge is a prerequisite for use in calculating the life of roller bearings from two perspectives. Above all, it will be possible to determine more accurately the durability of rolling bearings or rolling contact with all the economic consequences. The deepening of this information will enable purposeful and controlled increase of rolling bearing durability.*
Developing a reliable calculation of the life and durability of roller bearings or other machine nodes with rolling contact is a long-term global problem that has recently received considerable attention. The need for more accurate calculations with respect to operating conditions and desired reliability of machinery and equipment is enforced by constant rapid pace of development of most industries as well as engineering. In this situation it is necessary to give designers such calculation methods and materials that enable optimal use of the properties of roller bearings in rotating systems of machinery and equipment.

Key words: *lubricant, durability, lifetime, reliability, roller bearing.*

1. INTRODUCTION

Developing a reliable calculation of the life and durability of rolling bearings or other machine knots with rolling contact is a long-term global problem that has recently received considerable attention. The need for more accurate calculations with respect to operating conditions and desired reliability of machinery is enforced by constant rapid pace of development of most industry branches and also engineering. In this situation it is necessary to give designers such calculation methods and materials that enable optimal use of the properties of rolling bearings in rotating systems, machinery and equipment.

The cases where bearings limit work ability, durability and reliability of the machines can often be encountered in engineering practice. It appears that the tribological point of view is far from optimal as bearing design, as well as its use, especially in tough operating conditions. In many cases, small structural changes, changes in working conditions, direction and nature of load, lubrication and cooling, can significantly increase the technical life of the bearing and the entire, often very complex machine. The same conclusions can be made also on the basis of long-term monitoring of test results bearings on testing laboratories and testing on test analogs which model the conditions of rolling contact.

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The required solution is found at tribology, which is the scientific discipline concerned with the contact surfaces in relative motion and practices with respect to these. Analysis of the current state of design calculations and calculations of service life and reliability of rolling bearings shows that while in recent years sharply increased demands on bearing, calculations themselves are essentially unchanged for 80 years.

When designing the bearing knots the criterion of rolling contact fatigue (pitting) from contact tension defined according to the Hertz theory prevails. In this way the calculated life does not respond in many cases the real. Especially in modern extremely loaded bearings (high and low rotational speed, high temperature, high load, etc.) the bearing life ends very early, often due to defects other than fatigue damage (pitting). Very often it is a failure which processes cannot be even verbally described. It can be difficult to mathematically model the emergence and spread, or even predict their sudden or destructive disorder. It is because one of the primary factors delimiting operability bearings - lubrication effect of the contact is not yet included in the phase of bearing calculation. Namely, there are elastohydrodynamic (EHD) effects in the lubricated contact areas which determine the size of the forces of interaction between the rolling elements and the amount of friction. In operating conditions, but especially at testing laboratories, there are also cases where the bearing lifetime is much longer than it would correspond to the existing calculation methods. In particular, the phenomenon is dominant where tested durability of bearings. That causes especially significantly better lubrication at which rolling elements are separated by EHD oil layer [1–3].

2. CALCULATION OF ROLLER BEARING DURABILITY

Traditional equations of rolling bearing life, which was proposed on the basis of extensive research, was based on the information currently available and assumed the use of the bearing steel quality. Equation (1) applied to the 90% probability of survival S, respectively for 10% probability of failure P_P [1–3].

$$L_{10} = \left(\frac{C}{F_e}\right)^m . \quad (1)$$

Traditional life equation does not reflect deviations that were set at inception date. The equation does not stand up well in the complex operating conditions, and does not calculate the durability of the survival probability greater than 90%. With the above conditions adjusted according to the literature 2 the traditional life equation (1) was adjusted on the modified life equation (2) [1–3].

$$L_{na} = a_1 \cdot a_2 \cdot a_3 \cdot \left(\frac{C}{F_e}\right)^m , \quad (2)$$

where

L_{na} – modified basic durability of the desired probability of survival (n indicates the probability of failure in %),
a_1 – coefficient of probability of survival,
a_2 – coefficient of bearing material,
a_3 – coefficient of the operating conditions, in particular lubrication.

Equation (2) in this form takes into account new factors, particularly the impact resistance of the lubricant layer, which is nowadays becoming a structural material (the part touching the surface). The equation also counts with new bearing materials which are homogeneous and have a smaller amount of inclusions and other defects than before and also influences the production technology. Equation (2) also allows the calculation bearings are more likely to survive. The positive effect of the lubricant factor (2) is seen in the lubricant layer thickness, which separates touching surfaces see Fig. 2. It cannot be considered to what extent the consideration lubricant in the bearing will fulfill its purpose from the minimum thickness of elastohydrodynamic lubrication layer. The effectiveness of elastohydrodynamic layer also depends on the surface roughness of surfaces in contact. The relationship between the minimum lubricating film thickness and surface roughness can be expressed using the parameter lubrication, which is given by equation (3) [1–4].

$$\lambda = \frac{h_{\min}}{\sqrt{\left({R_{q_1}}^2 + {R_{q_2}}^2\right)}} , \quad (3)$$

where
R_{q1}, R_{q2} - initial roughness of functional areas expressed by mean square deviation from the mean line profile.

Equation (3) can be modified and preferably expressed in the form (4) [1–4].

$$\lambda = \frac{h_{\min}}{\sqrt{1.11 \cdot \left({R_{a1}}^2 + {R_{a2}}^2\right)}} , \quad (4)$$

where
R_{a1}, R_{a2} – initial roughness of functional areas expressed the arithmetical mean deviation from the mean line profile.

Parameter of lubrication can have different values and by its size it can be stated that if [1–4]:
λ = 3 or more, while loaded mostly elastohydrodynamic lubrication film prevails,
λ = 1; about 30% of the total load is transmitted by asperities of the surface profile, which tore lubricating film,
λ = 0.4; the load is not transferred by lubricating film, but directly by surfaces of solids (fatigue damage and wear).

Effect of elastohydrodynamic lubrication layer thickness on life of bearing is very high. The resulting durability in all cases of fatigue damage of roller bearings, for the desired quantile is given by (5) [1–4].

$$L_n = L_{ndef} + L_{n\lambda} , \quad (5)$$

where
L_n – a durability for quantile n,
L_{ndef} – a durability due to pre-existing surface and subsurface defects,
$L_{n\lambda}$ – a durability caused by the interaction of surface unevenness.

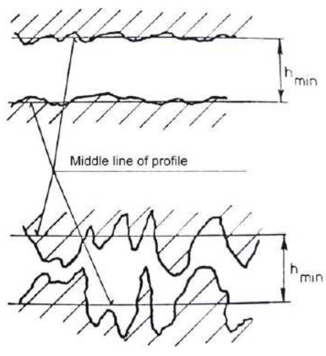

Fig. 1. Minimum thickness of the lubricating film at different surface integrity [1].

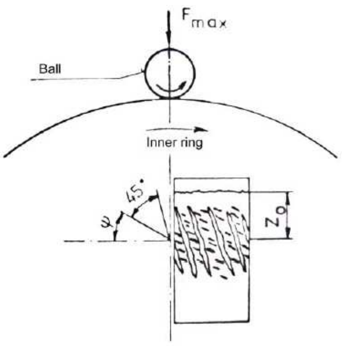

Fig. 2. Determination of the transformation angle ψ [1].

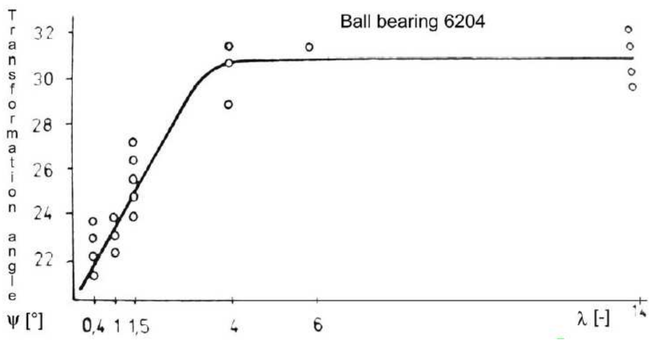

Fig. 3. The dependence of the transition angle ψ of the lubrication parameter [1].

Size parameter of lubrication affects not only the level of interaction touching the surface. Its size also depends on the character of changes in the structure of the material below the surface of the cyclically loaded rolling contact. The slope of the 30 ° light stripes that are formed below the surface at a depth of maximum shear stress changes with the size parameter of lubrication. Angle 30 °of light stripes towards the tangent to the direction of rolling motion is characterized by the transformation angle ψ, which is called as Martin angle Fig. 2 [1–4].

Plot of the transformation angle ψ example to the lubrication parameter was evaluated for ball bearing and is shown in Fig. 3 [1–4].

3. ELASTOHYDRODYNAMIC THICKNESS OF OIL FILM

The lifetime and durability of the roller bearing affects a large number of factors. The elastohydrodynamics is a field that deals with situations in which the elastic deformation of the surrounding objects plays an important role in the process of hydrodynamic lubrication. Among machine parts in general there are two kinds of deformations. In the first case, the contact geometry is overall deformation of the elastic portion to which the force acts, see Fig. 4. In the latter case the normally distributed stress at the contact point causes local elastic deformations which are significant in comparison with the thickness of the lubricating film, Fig. 5. The difference between deformations lies in the fact that the first type of deformation is relatively insensitive to the size and distribution of stress in the contact area, while the second

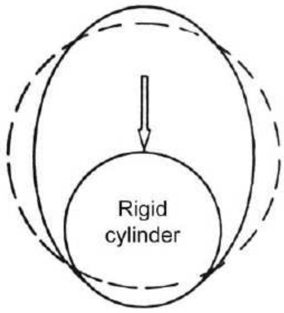

Fig. 4. Ways of elastic deformation.
a) The total deformation of elastic part [1].

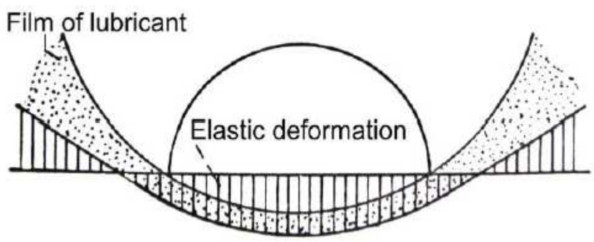

Fig. 5. Ways of elastic deformation.
b) Local deformation of contact [1].

type is closely related to local stress conditions. elastohydrodynamic lubrication can be briefly described as a condition in which the elastic deformation of the surrounding objects plays an important role in the process of hydrodynamic lubrication [1, 5–7].

In this case, there are two important phenomena that are not in the conventional hydrodynamic theory included. These are [1, 5–7]:

- the influence of high pressure on the liquid lubricant viscosity,
- substantial local deformation of elastic bodies.

These two factors greatly influence the geometry of the lubricating film, which conversely changes the pressure distribution in the contact. Complex interaction between state oil film and elastically deformed part shows the interferogram of point contact (touching ball on the plane), which is shown in Fig. 6 [1, 5–7].

Basically, it is a fact that the balance between hydrodynamic pressure in the fluid and elastic pressure in the bodies must occur, so that the common solution of the fluid flow equations and elasticity affects elasto-

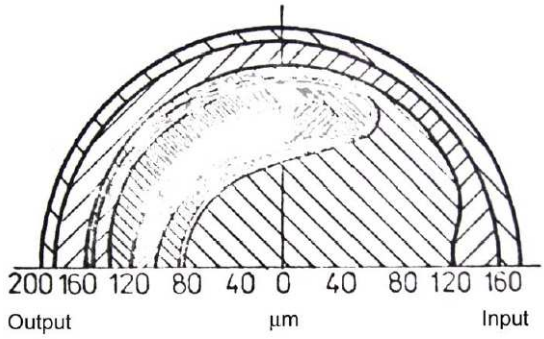

Fig. 6. Interferogram of elastohydrodynamic point contact [1].

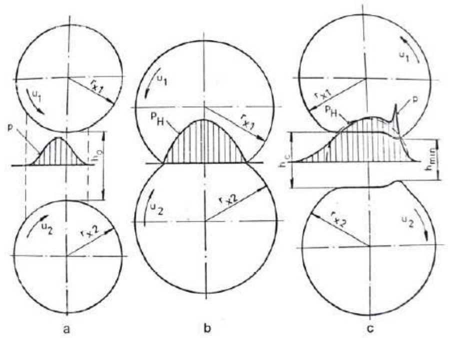

Fig. 7. Pressure distribution during rolling contact between two rolls when the calculation is performed on basis [1]:
a) Hydrodynamic Theory (Martin) - rigid bodies, iso-viscous lubricant,
b) Hertz theory - elastic body, metal contact (without lubricant),
c) Elastohydrodynamic theory - elastic bodies, the lubricant is Newtonian fluid.

hydrodynamic conditions at the contact point. The starting point of elastohydrodynamic theory falls to 1916, when Martin received the relation (6) for the minimum thickness of the lubricating film [1, 5–7].

$$H_o = \frac{h_o}{R_x} = 4.9 \cdot \frac{U}{W}, \tag{6}$$

where
H_o – dimensionless parameter of hydrodynamic oil film thickness,
U – dimensionless parameter of speed,
W – load.

Applied solution is shown in Fig. 7. Equation (6) can be used to calculate the thickness of the oil film between the face of a roller element and a flange at conical, cylindrical, and spherical roller bearings, where areas of hydrodynamic lubrication exist [1, 5–7].

The dimensionless parameter of speed U and load W are defined as follows [1, 5–7]:

$$U = \frac{\eta_o \cdot V}{E' \cdot R_x}, \tag{7}$$

$$W = \frac{\overline{F}}{E' \cdot R_x}, \tag{8}$$

where
$\overline{F} = \frac{F}{2a}$; Reduced modulus of elasticity:
$E' = \frac{2}{\frac{1-\mu_1^2}{E_1} + \frac{1-\mu_2^2}{E_2}}$.
Index 1 and 2 apply to body 1 and 2.

Relative speed V is determined by the relation:

$$V = \sqrt{v^2 + u^2}, \tag{9}$$

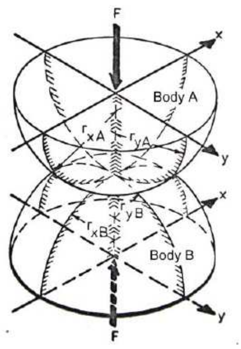

Fig. 8. Geometry of contact of two elastic bodies [1].

where

$$v = \frac{1}{2} \cdot (v_1 + v_2);\ u = \frac{1}{2} \cdot (u_1 + u_2).$$

The reduced radius R_x and R_y at the point of contact is determined by the radii of curvature of the functional surfaces in Fig. 8 according to:

$$R_x = \left(\frac{1}{r_{x_1}} + \frac{1}{r_{x_2}} \right)^{-1};\ R_y = \left(\frac{1}{r_{y_1}} + \frac{1}{r_{y_2}} \right)^{-1},$$

where
R_x – reduced radius in the direction of movement,
R_y – reduced radius perpendicular to the direction of movement.

Based on long-term examination (analysis included the effect of pressure on viscosity) in 1949 Grubin obtained an approximate relationship (10) to calculate the oil film thickness in a high elastic contact, under the isothermal condition [1, 5–8]:

$$H_{\min} = 1.95 \cdot U^{0.727} \cdot G^{0.727} \cdot W^{-0.091}, \tag{10}$$

where
$H_{\min}$ – dimensionless parameter of minimum film thickness,
G – dimensionless material parameter is calculated as follows: $G = \alpha \cdot E'$,
α – pressure coefficient of viscosity of lubricating oil.

Current state of elastohydrodynamic contact solution from the perspective of calculating the thickness of the lubricating film performed by Hamrock and Dowson [3, 4]. In calculation the numerical solution of the body elastic deformation using numerical methods for iterative procedures was included. The procedure is in the idea and the expression of pressure coefficient of lubricating oil viscosity α [5, 6] according to equation (11).

$$\alpha = \left[\int_0^{P \to \infty} \frac{\eta_o}{\eta_{(p,T)}} \cdot dp \right]^{-1} = \frac{1}{p_{iv,as}}. \tag{11}$$

Relation (11) initially determined the fictitious asymptotic iso viscous pressure $p_{iv,as}$, where the pressure coefficient of viscosity is the inverse. Contrary to equation (10) the modified dimensionless load parameter defined by equation (12) was used [1, 5–8].

$$W = \frac{F}{E^{\prime} \cdot R_x^2}. \tag{12}$$

Thanks to equation (12) the dimensionless parameter of the material G may then be expressed in the form (13) [1, 5–8].

$$G = \alpha \cdot E^{\prime} = \frac{E^{\prime}}{p_{iv,as}}. \tag{13}$$

From the above equations (11), (12), (13) and substituting into equation (10) the equation (14) was obtained, which provides calculation of the minimum oil film thickness [1, 5–8].

$$H_{\min} = 3.63 \cdot U^{0.68} \cdot G^{0,49} \cdot W^{-0.073} \cdot \left(1 - e^{-0.68 \cdot \varphi}\right). \tag{14}$$

Calculation of the oil film thickness at the center of the contact area is given by (15) [1, 5-8].

$$H_c = 2.69 \cdot U^{0.67} \cdot G^{0.53} \cdot W^{-0.067} \cdot \left(1 - e^{-0.73 \cdot \varphi}\right). \tag{15}$$

Meaning of the above parameters is shown in Fig. 9.

Equations (14) and (15) apply to the point and linear contacts, it also includes ellipse parameters φ that can be determined from equation (16) [1, 5–8].

$$\varphi = 1.0399 \cdot \left(\frac{R_y}{R_x}\right)^{0.636}. \tag{16}$$

Testing, in which field the examined contact works, whether at the hydrodynamic or elastohydrodynamic, this can enable four dimensionless parameters [3-6], which were designed to test Johnson operating conditions linear contact of two bodies and are given by formulas (17), (18), (19), (20) [1, 5-8].

Viscosity parameter is:

$$g_v = \sqrt{\left(\frac{\alpha^2 \cdot \overline{F}^3}{\eta_o \cdot V \cdot R_x^2}\right)}. \tag{17}$$

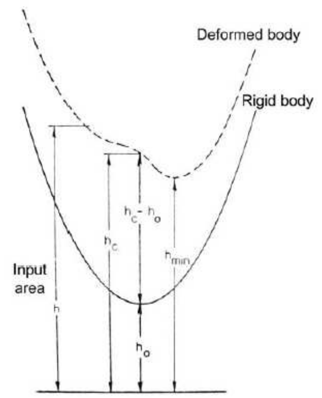

Fig. 9. Oil film thickness of elastohydrodynamic contact [1].

Parameter of elasticity:

$$g_p = \sqrt{\left(\frac{\overline{F}^2}{\eta_o \cdot V \cdot E^{\prime} \cdot R_x}\right)}. \tag{18}$$

Parameter of load transient area:

$$g_1 = \sqrt{\left(\frac{\alpha^2 \cdot \overline{F} \cdot E^{\prime}}{2 \cdot \pi \cdot R_x}\right)}. \tag{19}$$

Speed parameter of the transition zone:

$$g_2 = \sqrt[4]{\frac{\alpha^4 \cdot E^{\prime 3} \cdot \eta_o \cdot V}{R_x}}. \tag{20}$$

If the inequality: $g_v < 1.0$, $g_p < 0.6$, then it is area of hydrodynamic lubrication. Otherwise, this is an area of elastohydrodynamic lubrication. Out of this fact the transition region of the partial elastohydrodynamic lubrication determine inequality: $1.0 < g_1 < 100$, $1.5 < g_2 < 100$. These four parameters are valid also for the point contact of two bodies, if the ratio of normal load in the contact area F per unit length of the contact area is used [1, 5–8].

4. CALCULATION OF ELASTOHYDRODYNAMIC LUBRICATION CONDITIONS FOR TESTED BEARINGS

When determining elastohydrodynamic lubrication in roller bearings the contact between the most loaded rolling element and the corresponding ring is analyzed. At the single row ball bearings it is the inner ring. At the double row self-aligning ball bearings on the contrary it is the outer ring. In fact, it is always the place where the most severe contact conditions usually arises and where fatigue damage (pitting) occurs. First, it is always necessary to determine the normal force acting between the most loaded rolling element and functional flat ring. On Fig. 10 there is shown a distribution of the radial load for double row self-aligning ball bearing. Force acting in the normal direction $F_{1,2\max}$ can be calculated for two-row self-aligning ball bearing from (21) [1, 6–10].

$$F_{1,2\max} = \frac{F_r}{J_r \cdot 2 \cdot z \cdot \cos\alpha}, \tag{21}$$

where

z – the number of rolling elements in one row,

J_r – radial integral (dimensionless coefficient).

Numerical values of the radial integrals are calculated depending on the load bearing parameter ε that expresses the relation (22) [1, 6–10].

$$\varepsilon = \frac{\delta_o}{2 \cdot \delta_o + V_r}, \tag{22}$$

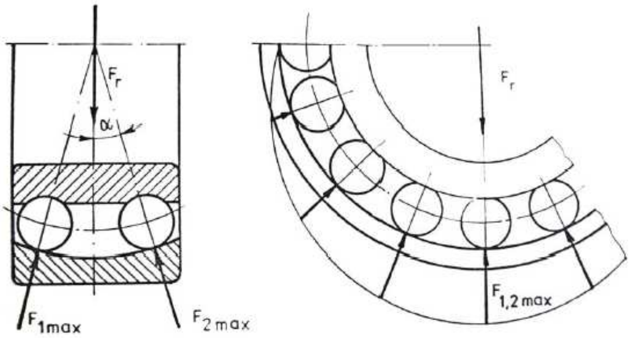

Fig. 10. Radial load distribution in the bearing [1].

where

δ_o – the total elastic displacement,

V_r – radial clearance of the bearing.

Radial clearance V_r is determined by measuring after bearing assembly. Overall elastic displacement in the radial direction can be calculated based on the known empirical relationships. For double row self-aligning ball bearings can use the following formula (23) [1, 6–10].

$$\delta_o = \sqrt[3]{\frac{6.98 \cdot 10^{-4}}{\cos\alpha} \cdot \frac{F_{1,2\max}^2}{d_o}}, \tag{23}$$

where

$F_{1,2\max}$ – the biggest load of the rolling element,

d_o – the diameter of the rolling element,

α – contact angle.

Flexible displacement of single row ball bearing can be calculated from equation (24) [1, 6–10].

$$\delta_o = \sqrt[3]{\frac{4.36 \cdot 10^{-4}}{\cos\alpha} \cdot \frac{F_{\max}^2}{d_o}}. \tag{24}$$

The value of the radial integral can be numerically calculated according to equation (25) [1, 6–10].

$$J_r = \frac{1}{2 \cdot \pi} \cdot \int_{-\psi}^{+\psi} \left[1 - \frac{1}{2 \cdot \varepsilon} \cdot (1 - \cos\psi)\right]^{\frac{3}{2}} \cdot \cos\psi \cdot d\psi. \tag{25}$$

Analogously, from equation (26) it is also possible calculate the normal force acting between the most affected contact for single row the ball bearing [1, 6–10].

$$F_{\max} = \frac{F_r}{J_r \cdot z \cdot \cos\alpha}. \tag{26}$$

For single row radial bearings can be used in practice simplified formula (27) designed by Stribeck, which already includes the effect of normal radial clearance [1, 6–10].

$$F_{\max} = \frac{5 \cdot F_r}{z}. \tag{27}$$

For the calculation it is also necessary to know the geometry of the contact surfaces, surface structure and their mutual speed. Equally knowledge of the parameters of the lubricant is important. Outside the dynamic viscosity primarily knowledge of pressure coefficient of viscosity or fictitious asymptotic iso viscous pressure is necessary [1, 6–10].

5. TESTING OF THE INFLUENCE OF ADDITIVES IN LUBRICANTS

At AXMAT stations, see Fig. 11, experiments were performed that were designed to test the effect of additives in lubricants to contact fatigue and determine the sensitivity of methods for monitoring the AE on test sample in the application of additives in the lubricant. At the same time the suitability of AE parameter was verified on detection of pitting.

The subject of the test:
Flat AXMAT pattern mat. E295 (1.0050);
Grease: Mogul LV3;
Ingredient: Methane F1.5.

Type of test:

Tests were conducted on adjusted AXMAT stations on conventional samples. Bearings were loaded statically at a constant speed. During test the vibration, temperature and sensed acoustic emission were monitored.

Test parameters:
The load: 5000 N;
RPM: 1380 1/min;
The test period − until a pitting.

Tests process:

Intentionally sample of steel (E295) that with its property is not suitable for machine nodes exposed to contact loading was used.

To record test contact lubricated with grease without additive, sustained increase in the emission events expressed as the number of counts and RMS process is shown, see Fig. 12. During the test phase the classical focal damage was recorded, steady mode did not occur.

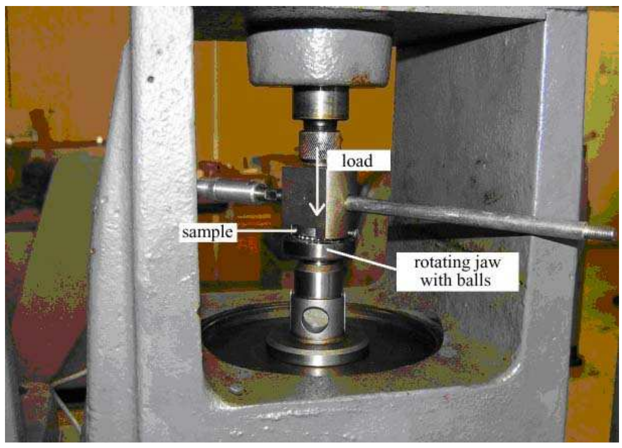

Fig. 11. AXMAT station.

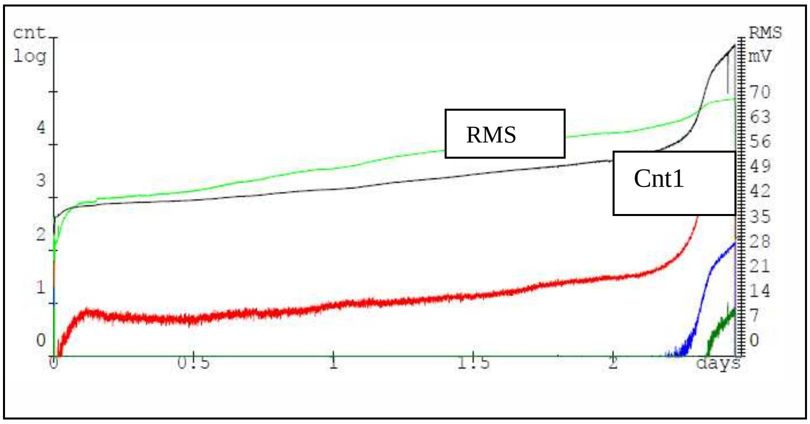

Fig. 12. Test record without additives.

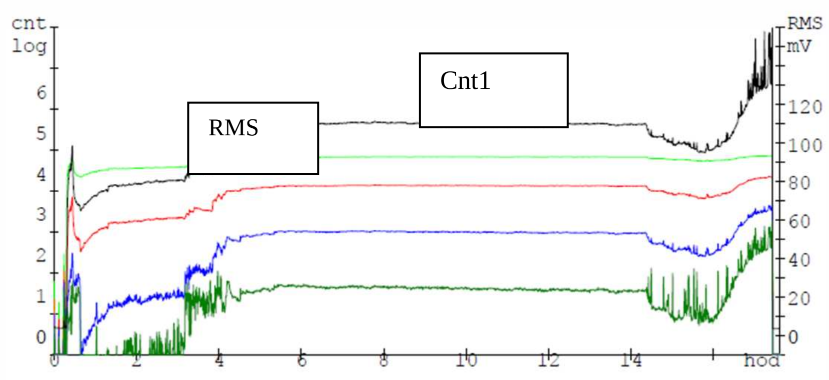

Fig. 13. Test record with additives.

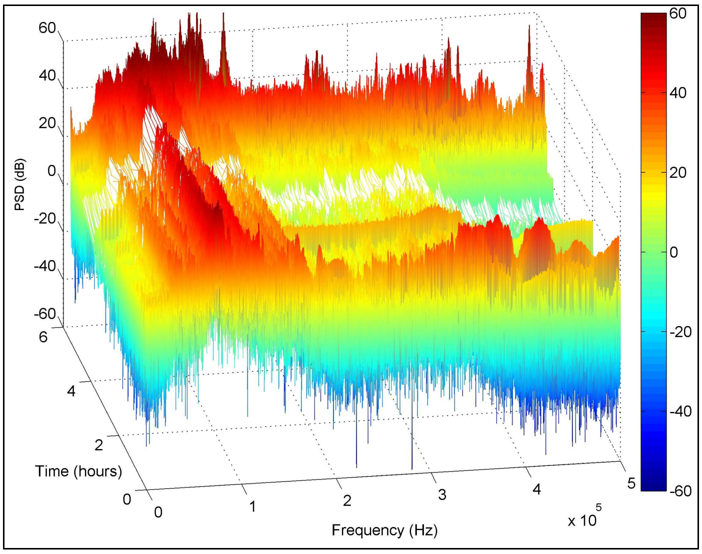

Fig. 14. Signal intensity versus time.

The test record with the use of additive Metanova F1.5 additive shows its positive effect, see Fig. 13. At the record a delay is noticeable, when there is no formation of new emission events, the experiment is carried out in steady mode.

6. THE DEMONSTRATION OF ADVANCED METHODS OF SIGNAL PROCESSING OF ACOUSTIC EMISSIONS

Example of basic record at the Test analyzer IPL is shown in Fig. 14, where the signal frequency [kHz] is on the *x* axis and the test time [min] is at the *y* axis and the degree of signal intensity [dB] shows the color scale. This particular record does not contain an initial trial run. If we make cuts parallel to the time axis, we can get during certain selected signal frequency (e.g. peak) versus time, the test procedure, see Fig. 15.

7. CONCLUSION

Currently, knowledge as to the effect of additives on the durability of rolling contact is at a relatively low

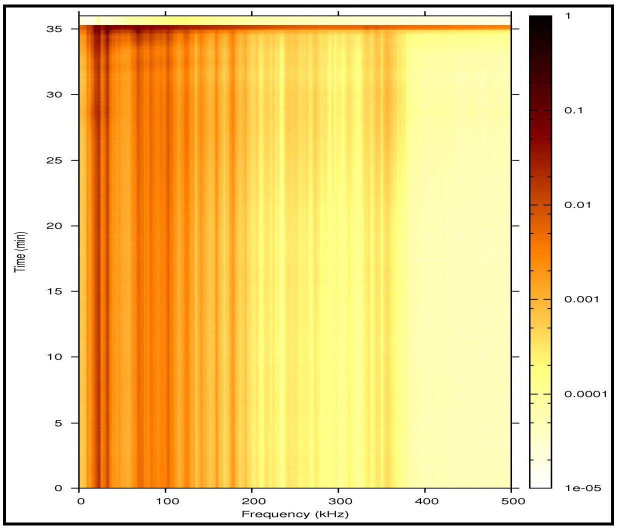

Fig. 15. Sample record of AE signal obtained by IPL analyzer.

level. There is only a small number of works that deal with this problem. Achieved experimental results, which are results of various tests, are compared in many cases only with difficulty. It is caused mainly by different test conditions, a small number of experimental results, usage of different test equipment, etc. With these conditions then no exception is in completely different results, where in one case additives increase durability and in another case it is reduced. This can be caused by different contact pressure, different parameters of lubrication λ, or other chemical composition additives. There may also be other reasons, for example, slide, water content in the lubricant, etc. Very few authors point to the effect of additives in statistical testing.

Based on previous research, it can be concluded that the additives in most cases reduce the durability of rolling contact, if they are contained in mineral oils. At synthetic oils, the effects can be quite opposite. Current knowledge of effects of additives would argue these sub-conclusions:

- Chemical characteristics of the lubricant and additives affect the course of the durability empirical distribution.
- Effect of chemical composition of the lubricant and additives depends on the maximum contact stress of lubrication parameter – λ, sliding and lubricant temperature.
- Experiments to determine the influence of the chemical composition of lubricants and additives should be carried out under the same conditions, particularly as operating conditions of the investigated case. It is also necessary to also take into account the influence of lubricant contamination and the possible presence of water in the lubricant.

From the above it is evident that effects of additives on the contact fatigue should be further examined experimentally. It is also necessary to pay attention to tribological test conditions which must be stable and their description must be an integral part of the experiment results.

The test demonstrated the ability of AE method to distinguish differences in the behavior of the contact surfaces with the same load conditions, with only incremental changes – in this case, the application of additives. Vibration sensor did not notice a change in test mode.

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