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Performance Evaluation of Rolling Element Bearings Based on Tribological Behaviour

Jerzy Nachimowicz and Marek Jałbrzykowski

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http://dx.doi.org/10.5772/51144

1. Introduction

It is a well-known fact that the appropriate test method could be one of the crucial conditions of the correct evaluation of a simulated process. One might presume that in some cases in which the subject matter is well-researched, only widely recognized methods are applied. However, in less familiar cases, or where one needs to evaluate some unusual values, the question of the appropriate methodology becomes far more significant. There are a number of papers in literature dealing with the analysis of the same phenomenon or process where tests were based on completely different methodological approach. As a result most papers quoted different results which caused serious controversy. In addition, it appears that even when the research was conducted according to the same procedure, but at different research units the results were still divergent.

As far as ball bearings are concerned there are numerous research methods to test these elements. For instance, using a four-ball pitting apparatus (T-03), a pitting ball-bearing tester (T-06), Amsler device or a ball-bearing testing platform. Most standards describe research methodology with the main focus on using particular testing devices, disregarding to a certain extent the very essence and purpose of the analysed object. The official standard for ball bearings in Poland is PN- 89/M-86410, however, it lacks a description of research methodology related to the external function of a ball bearing in „true-to-life" conditions. The description contains regulations concerning controlling the particular components of a ball bearing e.g. radial and axial run-out deviation, radial and axial clearance without paying any special attention to the working conditions of a ball bearing. Still, the standard allows applying other research methods which require prior acceptance of both the manufacturer and recipient of ball bearings.

In this context, the authors of the present section conducted tests on two-set ball bearings according to two different methods: their own methodology and the methods

recommended by the manufacturer (FŁT PLC in Kraśnik - Poland). Should add that all of the test, later in this chapter bearings are factory items and their designations are in accordance with PN, PN-EN and ISO standards. All tests were performed for three replicates at each point.

1.1. The metods of investigations

Tribological tests were conducted with a widely-used friction tester SMT-1. Fig. 1 shows the apparatus and the structure of the friction node.

Laboratory tests were conducted in two stages. In the first stage tribological tests were conducted according to the authors' own methododology. Next the tests were conducted according to the method applied in FŁT in Kraśnik (Poland). These tests were meant to compare the two methods and evaluate their influence on test results. It should be noted that the load is strictly radial, without a longitudinal component which has been verified experimentally, and this is due to the construction SMT1.

1.1.1. Authors' own methodology (MB –1)

In this section the following test parameters were assumed: constant normal pressure N = 350 [kG] ≈ 3433 [N], changeable rotational speed of the bearing : n_1= 300 [rpm], n_2= 600 [rpm], n_3= 1200 [rpm]. Because of changeable rotational speed it was assumed that the number of bearing's cycles will serve as the unit of its operation. Since the tests were treated as pilot ones, at this stage tests were conducted at 20,000 cycles of bearing's operation. It should be noted that before each measurement the bearing was cooled in a stream of air for 4 hours in normal conditions. All the measurements were recorded using a computer system.
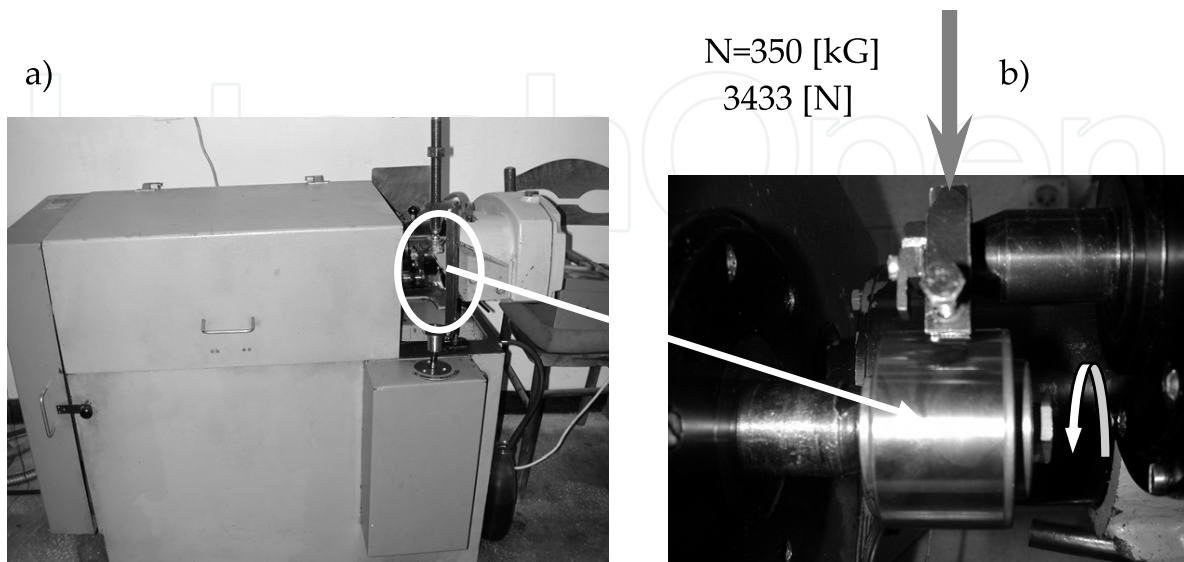
a)

N=350 [kG]
3433 [N]

b)

Figure 1. SMT-1 friction tester a) general view, b) friction node

1.1.2. Tests recommended by FŁT in Kraśnik (MB-2)

Before the tests were started, in the initial stage, normal force of N= 350 [kG] ≈ 3433 [N] was applied to the bearing. Next the apparatus was switched on for t=30 [min] at the speed of n= 1000 [rpm]. After that the rotational speed was increased to n= 1400 [rpm] and as soon as the system became stable the moment of friction was measured (M_t). Then, the rotational speed was decreased by 200 [rpm] in each step, and on each occasion after the friction node became stable the measurements M_t were taken.

1.2. Test results and discussion

Due to the extensive research material, the present section does not contain the results of initial tests obtained with MB-1 and MB-2 methods, it only includes their description.

As far as MB-1 method is concerned, it should be noted that its results indicate diversification of tribological characteristics of the examined elements in relation to rotational speed. There is also a certain analogy in the course of characteristics for the two groups of the evaluated bearings. In the initial stage of the process the maximum value of friction moment was obtained (for all rotational speeds). In the subsequent stages it decreases and, in the final stages, the resistance of motion becomes stable. One should note that at the beginning of the process the highest value of the moment of friction was obtained at the speed of n = 1200 [rpm] while the lowest at n = 300 [rpm]. In the final stage of the process the trend is reversed.

In the case of MB-2 method general diversification of the moment of friction was also noticed. It is logical considering different conditions of the external function. However the lowest value of the moment of friction was recorded for the lowest rotational speed n = 200 [rpm], while the highest for n = 1000 [rpm]. Further measurements, at higher rotational speeds, indicate the decrease of the value of the moment of friction. However, the most controversial conclusions are drawn from the comparative analysis of the two research methods.

Fig. 2 and 3 shows selected comparative graphs of the courses and values of the moment of friction obtained in tests according to MB-1 and MB-2 methods described above.

The graphs above provide data concerning completely divergent results in the conducted tests. The analysis of the data in fig. 2. (MB-1 method) proves that both at the speed of n = 300 [rpm], and especially n = 1200, throughout the whole of the research cycle, one notices distinctly higher values of the moment of friction for the bearings from the 134-781TNG-2RS group. At the same time there are lower values of resistance to motion for the bearings from the CBK 441TNG group. Since these bearings have different dimensions of working elements one could assume that different values of the moment of friction is a natural phenomenon. When tests were carried out according to the other method similar trends in friction characteristics were expected. However, when the results obtained with MB-2 were analysed, quite contrary to expectations, they were in direct opposition to those obtained with MB-1 method.
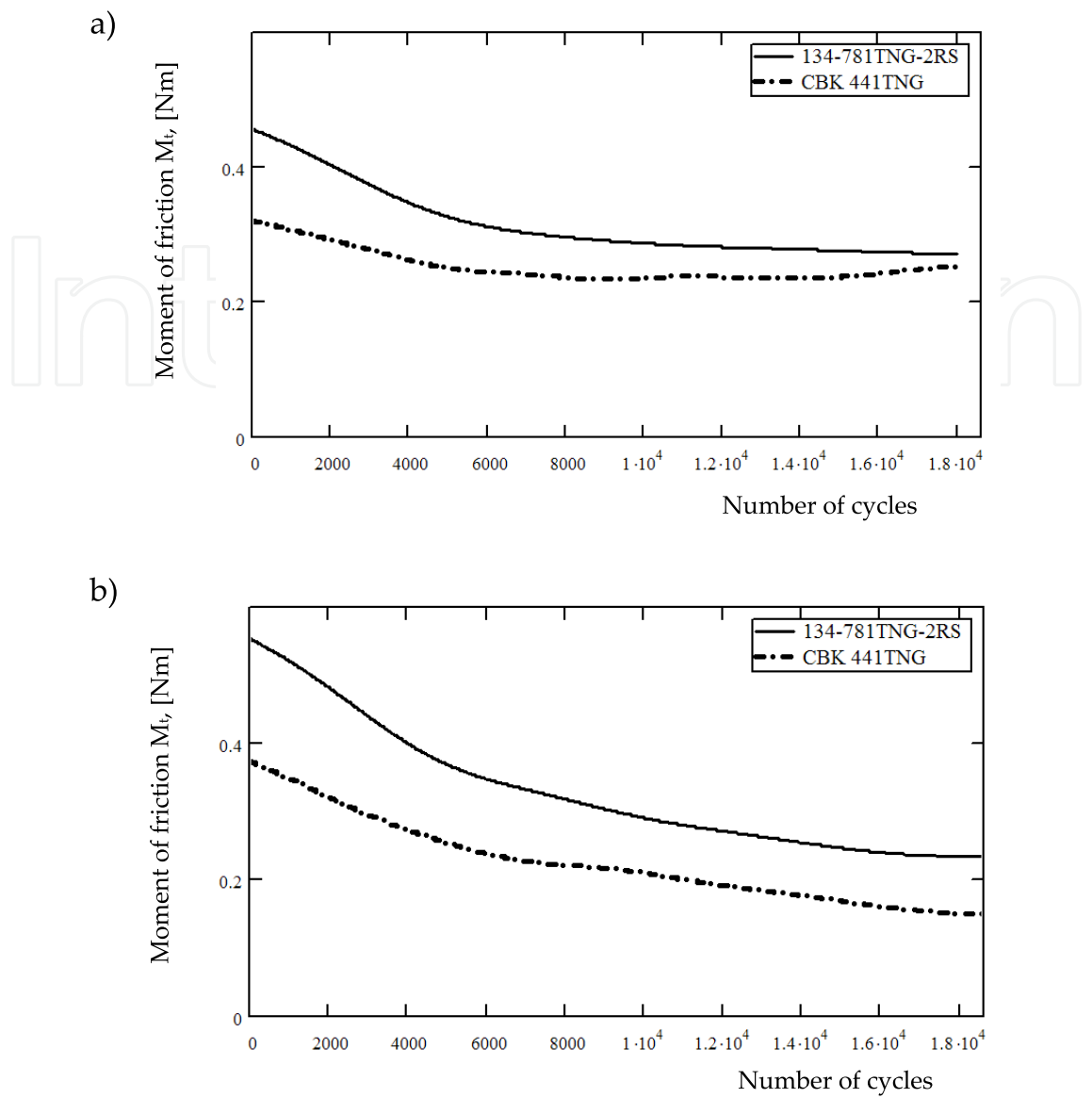
a)

b)

Figure 2. The course of the moment of friction in the function of time for two groups of bearings at different rotational speeds (MB1 method): a) 300 [rpm], b) 1200 [rpm]
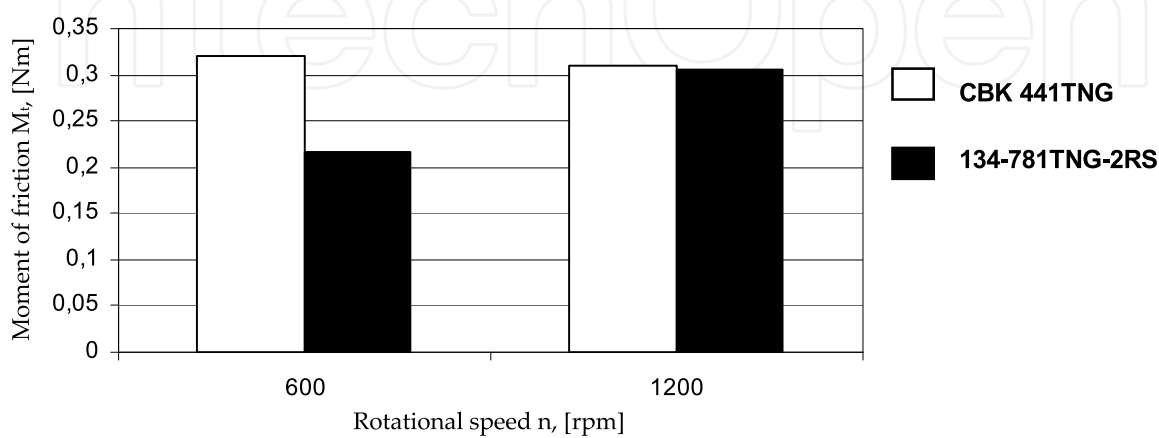

Figure 3. The values of the moment of friction in the function of rotational speed – MB-2 method
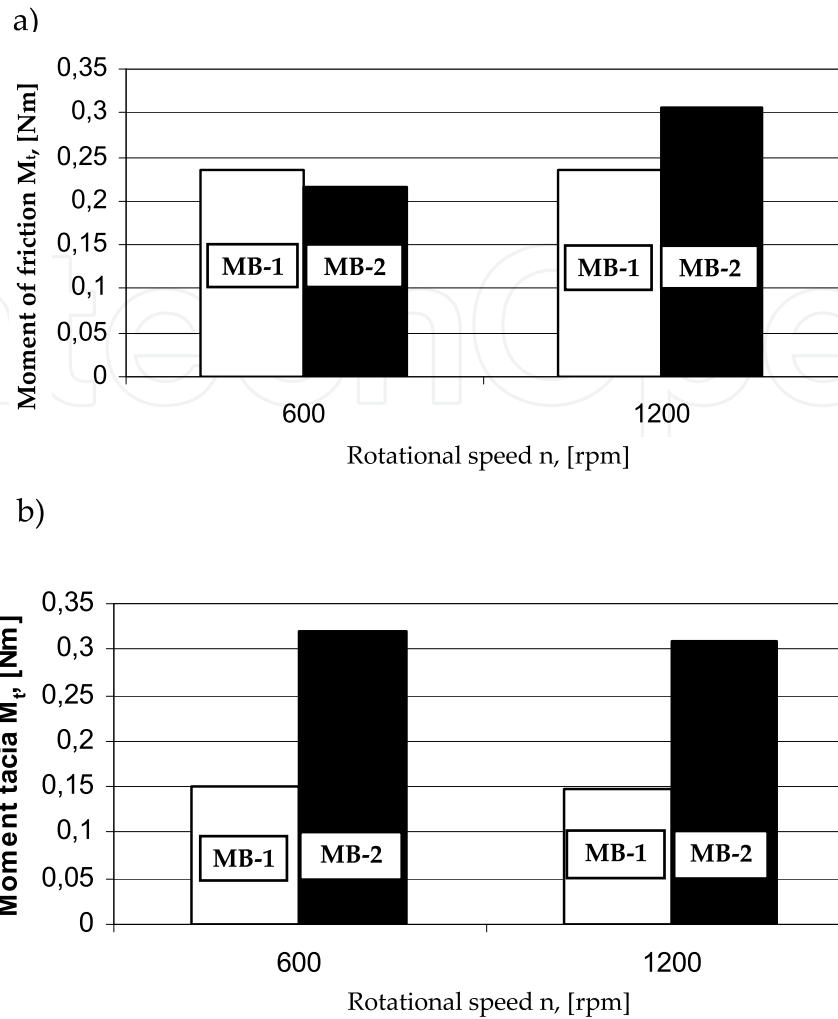
a)

b)

Figure 4. Comparison of two research methods MB-1 and MB-2 (bearings) a) 134-781TNG-2RS b) CBK 441TNG

This comparison is shown in fig.4. Fig. 4a refers to the test results for the bearings from the 134-781TNG-2RS group, fig. 4b is related to the bearings from the CBK 441TNG group. No doubt, the results for the two groups of bearings are completely different. It should be noted that, in the 134-781TNG-2RS group, the differences in the recorded values of the moment of friction range from several to twenty-odd per cent, while in the CBK 441TNG group the differences are even two-fold. Moreover, one should note that in the case of bearings from the 134-781TNG-2RS group at the speed of n = 600 [rpm] slightly higher values of the moment of friction were registered with MB-1 than with MB-2 method. In turn, at the speed of n = 1200 [rpm] the situation is reversed. That means that when tribological tests are carried out, especially on such elements, one should bear in mind the significance of the selected research method. Choosing an arbitrary method may result in, as the evidence discussed in the present section suggests, obtaining entirely different or even unexpected test results. In consequence it may lead to wrong conclusions related to the tribological properties of such elements. At the same time one should remember it is often a complex task to select the right method that can reflect the real process of technical objects exploitation closely enough.

2. Selected energetic aspects of needle bearings work performance

Shafts on needle bearing can work at heavy load accurately lateral. Durability of such a bearing is guaranteed by suitable selection materials which are made (roughness, microhardness, etc.) and optimum lubrication conditions. The factors specified above have influence on reliable work and life of needle bearings.

Wear equation presented in energetic form describes real processes occurring under various external influences on materials which lead to their damage and create wear products. The initial form of energetic approach to wear processes is as follows:

$$E_{y\partial} = A_{wn}/V \tag{1}$$

where:

$E_{y\partial}$ - specific energy of wear material of volume V when wear products are created, A_{wn} - the work of external forces affecting the volume in time t or at N cycles of load.

It is assumed that:

$$A_{wn} = A_i N = A_i \omega t \tag{2}$$

where: A_i – external work under single load of volume V, ω - frequency of load.

Equations (1) and (2) result in wear equation in the following form:

$$V = \frac{A_i}{E_{y\partial}} \omega t \tag{3}$$

Equation (3) describes the wear when wear products are produced at a uniform rate i.e. linear wear.

For non-linear wear:

$$V = \frac{A_i}{E_{y\partial}} f(\omega t) \tag{4}$$

where: the function f (ω t) may be presented (apart from the linear one) as power, exponential, logarithmic and as other relations.

In dynamic character of load (impact, hydro-abrasive, cavitation wear) the equation (4) should consider the velocity of impact:

$$V = \frac{A_i \vartheta_i}{E_{y\partial} \vartheta_{kr}} f(\omega t) \tag{5}$$

where:

ϑ_i - the average value of velocity of impact load, ϑ_{kr} - critical velocity of impact which causes damage of wear material , $A_i\vartheta_i$ – the flux of energy which enters a material in the course of separate impact, $E_{y\partial}\vartheta_{kr}$ – critical density of strain energy – critical density of strain power.

Assuming $A_i\vartheta_i = w$ and $E_{y\partial}\vartheta_{kr} = W_{kr}^*$, we obtain:

$$V = \frac{w_i}{W_{kr}^*} f\left(\omega t\right) \qquad (6)$$

The numerator and denominator of this equation are reduced after the singularity of a given type of wear is taken into account. The flux of external energy w should be expressed by the friction force in external friction. Fig. 5 shows the model of distribution of the energy of plastic deformation.
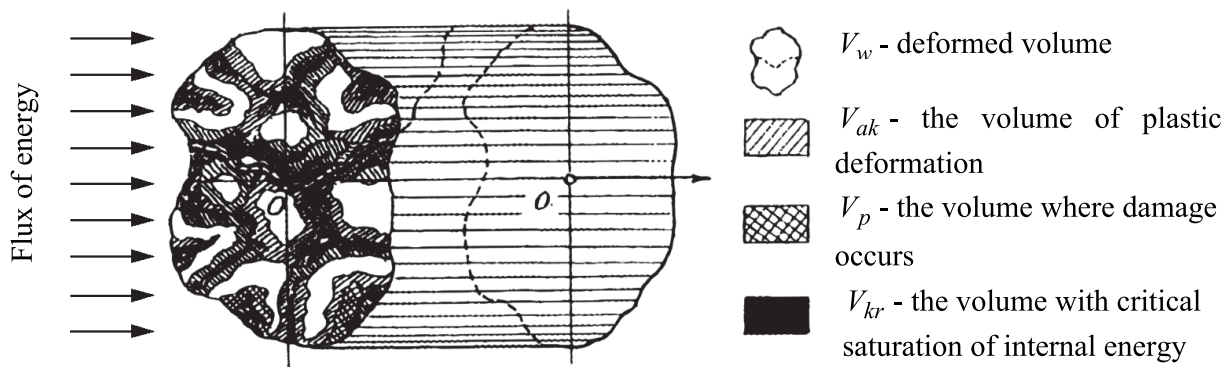

V_w - deformed volume

V_{ak} - the volume of plastic deformation

V_p - the volume where damage occurs

V_{kr} - the volume with critical saturation of internal energy

Figure 5. The model of the energy flux distribution in the deformed volume

The criterion of resistance to wear W_{kr}^* should consider the properties of materials (mechanical, physical, chemical and others), causing damage in given conditions of external influences. The value W_{kr}^* depends on the character of external load and assumes the averaged values in the worn out volume (* the symbol of averaging).

By presenting ϑ_{kr} as the sum of components of elastic and plastic deformation, ϑ_{kr}^{spr} ϑ_{kr}^{pl} respectively, we shall obtain:

$$W_{kr}^* = \frac{1}{3} E_{y\partial}^* \vartheta_{kr} = \frac{1}{3} E_{y\partial}^* \left(\vartheta_{kr}^{spr} + \vartheta_{kr}^{pl}\right) \qquad (7)$$

The specific strain energy E_{ya}^*, irreversibly absorbed by the material at the instant of creating wear products, can be determined according to the durability or hardness graph as an area below the curve $\sigma(e_i)$ or $H(e_i)$, restricted on the right side by the damage deformation e_p determined in given wear conditions with the method of micro-hardness directly on the worn out surface of a material.

The elastic component $\vartheta_{kr}{}^{spr}$ in the equation (7) is defined with the velocity of the elastic wave in deformation c_0 and elastic deformation e_{spr}:

$$\vartheta_{kr}^{spr} = c_0 e_{spr} = \sqrt{\frac{E}{\rho_M} e_{spr}^2} \tag{8}$$

where:

E – the module of elasticity, ρ_M - density of a material being used. Plastic component $\vartheta_{kr}{}^{pl}$ in (7) depends on the velocity of plastic deformation wave c_{pl} and total plastic deformation accumulated in the course of wear:

$$\vartheta_{kr}^{pl} = \int_{e_{spr}}^{e_p} c_{pl} de = \int_{e_{spr}}^{e_p} \left(\frac{d\sigma/de}{\rho_M} \right)^{0.5} de \tag{9}$$

where: $d\sigma/de$ - local slope of tangent to a curve of stretching determined with stress – strain coordinates (σ - ε).

By expressing the right-hand side of equations (8) and (9) as the energy of brittle E_{kr} and plastic E_{pl} damage we shall obtain:

$$\vartheta_{kr} = \sqrt{\frac{2E_{kr}}{\rho_M}} + \sqrt{\frac{2E_{pl}}{\rho_M}} \tag{10}$$

where: $E_{kr} = E_{kr} = \sigma_b^2/2E$, $E_{pl} = \left(\sigma_b - \sigma_T \right)^2 /2D$, σ_b, σ_T - strength of a material and the limit of plasticity, D – module of consolidation (tg of inclination angle of tangent to a curve $\sigma(e)$ within the range of plastic deformations, with linear character of deformation consolidation of a material. The critical velocity of impact, connected with singularities of spreading in materials undergoing wear, elastic and plastic deformation waves, features resistance to material wear at dynamic, high-velocity stress and it can be expressed by durability characteristics of worn-out volumes, as shown in equations (8) – (10).

Test data analyses proves that in conditions of stretching at impact (impact on the front surface of a rod) there exists an involution dependence (between ϑ_{kr} i σ_b):

$$\vartheta_{kr} = const \sigma_b^n \tag{11}$$

where: n = 2,5 for steel with $\sigma_b \le 500$ [MPa]

const – a constant determined empirically, considering the specific test conditions or the proportional coefficient between the discussed characteristics.

Taking into account (11) the criterion of resistance to wear for iron-based alloys with $\sigma_b \le 500$ [MPa], it will prove that ϑ_{kr} and σ_b cubed are proportional:

$$W_{kr}^* \approx const \vartheta_{kr}^3 \approx const \sigma_b^3 \qquad (12)$$

The particular dependencies (12) are of significant practical interest with a view of predicting the use of materials and coating. By substituting equation (10) for (7) we shall obtain:

$$W_{kr}^* = \frac{1}{3} E_{y\partial}^* \left(\sqrt{\frac{2E_{kr}}{\rho_M}} + \sqrt{\frac{2E_{pl}}{\rho_M}} \right) \qquad (13)$$

In this equation, the total energy-consumption of a material E_{ya}^* and its components E_{kr} and E_{pl} can be determined according to a microhardness (endurance) graph. If the graphs of hardness or endurance cannot determined, then the evaluation of resistance of materials to wear at external impact is possible as a result of analogy to W_{kr}^* criterion.

Still, it is observed that when the stiff impact stress is reduced and when external influence resemble static conditions, W_{kr}^* criterion will assume very simple forms. For instance it can be expressed in reduction of indicator in the power at ϑ_{kr} and σ_b or hardness (12), [5, 6, 8].

2.1. The method of investigation

Fig. 6 shows the scheme of the tribotester for the estimation of needle bearing. In this case, used the bath lubrication. Bearing designation are given in Fig. 6.
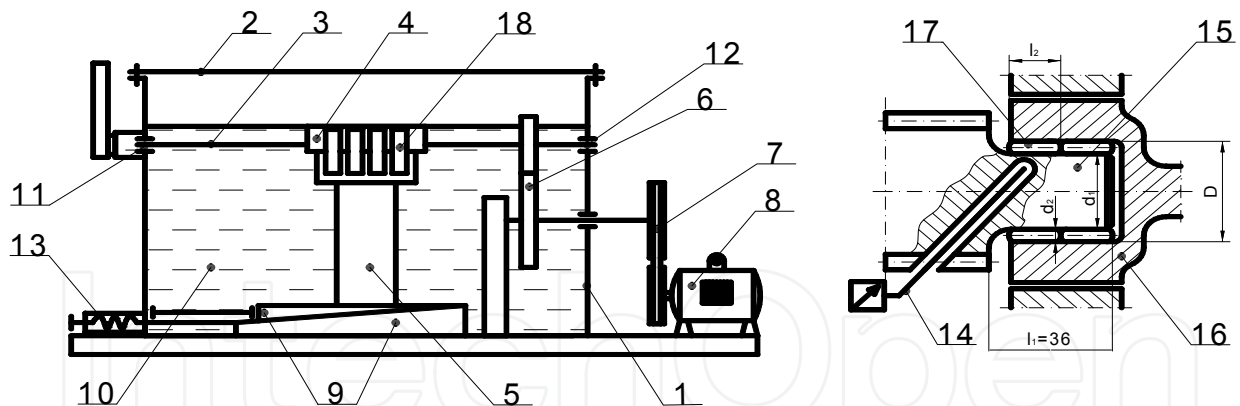

Figure 6. The scheme of the tribotester: 1- frame, 2- cover, 3- investigated shaft, 4- reducer sleeve, 5- cylinder, 6- toothed gear, 7- transmission belt, 8- electric motor SZJe 3,5 [kW], 9- wedges, 10- oil, 11- needle bearing K28x33x13, 12- needle bearing K40x45x17, 13- screw gear, 14- thermoelement NiCrNi, 15- shaft neck d= 28 [mm], 16- hub, 17- needle roller ϕ2,5 [mm], l2= 13 [mm], 18- bearing head Parameters of investigation:surface stresses, p = 950 [MPa], rotational speed, n = 1500 [rpm].

2.2. The results of the investigations

Fig. 7 shows the influence of energetic criterion on the wear of a shaft neck in the function of change of material and lubricating parameters.
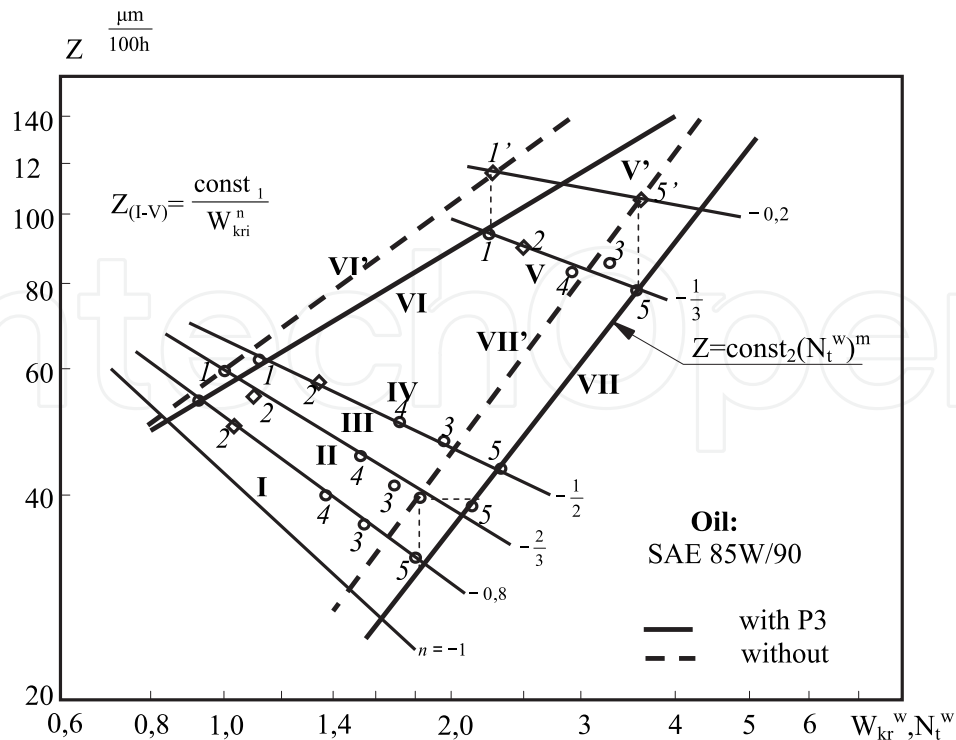
Figure 7. Wear intensity of a shaft neck – energetic criterion: Nt^w – relative density of friction power; W_{kr}^w – relative critical density of strain power ; a shaft neck: 1 – 16MnCr5, 2 – 20MoCrS4 + carburising; welding : 3-80G; 4-35X5M1,5; 5-50X10GC1,5; P3 – addition to Acorox 88 oil, needle bearing K 40 x 45x17

The analysis of test data shows that at general resistance to wear criterion for materials at external friction there occurs average critical density of friction power W_{kr}^* , in worn out volumes.

If chemical reactions, surface-active substance and other factors have significant influence on the processes of creating various secondary structures and products, then determining adequate reliability of criteria of resistance to wear proves to be a difficult problem, for instance at low-intensity oxidizing wear of a tribo-coupling. Resistance criteria in such conditions might be: dislocation density in the thin surface layer of a material, activating energy of chemical reactions and durability characteristics [5 ,9, 10].

The list of materials resistance to wear criteria – invariants of universal criterion W_{kr}^* (critical density of strain power) indicates huge variety and complexity of processes of materials surface damage.

It ought to be noted that the properties which characterize deformed areas of materials at the moment of their destruction (the initial stage of creating wear products) show reactions of these areas to external influence. In order to point those reactions in the desired direction, in this case – to ensure high resistance to abrasion (endurance) of its elements, it is necessary to make the right choice of known materials or produce new ones, having structure of the highest resistance to cracking resulting from the influence of external (exploitation) factors. This means that materials science involving problems of tribotechnology should be based on the analysis of microstructures.

3. The influence of geometrical parameters on the friction process in the needle bearing

The construction of the rolling bearing was initially based on the assumption that the friction loss during the bearing work is significantly smaller than during the sliding. However, during the work of the bearing in the operating conditions there exists both the bearing and the sliding friction. Different factors result in appearance of resistance to motion while the bearing is operating:

- Hysteresis of deformations;
- Interior friction in lubricant;
- Sliding and microsliding caused by deformations, the geometry of contact area and the movement caused by the gyroscope moment;
- Sliding between the bearing cage, the bearing elements and the bearing ring;
- Sliding in the bearing seal.

When bodies are deformed in the operating conditions, the phenomenon of pure bearing exists if the cooperating elements possess the same diameter, length, the properties of the material and parallel axes. Also, the roughness of the cooperating surfaces should be minimal. In such a case, when there is no lubricant, there appear only some losses caused by hysteresis of deformations [15]. This stems from the fact that the difference between the length of the contact arc and the corresponding arcs before deformations is identical for both bodies. As a result, there is no mutual sliding of the surfaces in the deformation area.

If the curvatures of the cooperating surfaces are different, in the elastic deflection the length of the contact arc for both bodies is identical, whereas before the deflection it was different. Consequently, the deformations in the contact area are accompanied by microsliding; if, however, the speed of both cooperating surfaces is identical, their mutual movement is called „bearing".

In a typical situation, in the process of bearing of two bodies with different peripheral speed there occurs the bearing with sliding; such a situation is the subject of examination here. The aim is to estimate of the extent to which the sliding friction matters in the overall balance of motion resistance.

3.1. The analysis of contact areas of the needle bearing elements

In the place of contact of two elastic bodies pressed against each other with some force, some contact stresses within a certain field of mutual contact occur. They reach significant values even in the situation when the pressing force is relatively small, which, as a consequence, may lead to exceeding the acceptable limit of the material effort. This is of paramount importance during the work of needle bearings which are under considerable load. Figure 8b shows a situation when the axis of the needle and the axis of the shaft neck are parallel. The stresses that occur (Fig. 8 a) are evenly spread along the length of the needle, and the area of contact between the two elements equals the field of the ellipse of the length which is the same as the length of the needle and the width 2b calculated by means of Hertz's solution:

$$b = \sqrt{P\frac{D_1 \cdot D_2}{D_1 + D_2}\left(\frac{1-v_1^2}{E_1} + \frac{1-v_2^2}{E_2}\right)} \tag{14}$$

where P is the strength per each unit of the rollers length; D_1, D_2 are the diameters of the shaft neck and the needle respectively; v_1, v_2 are Poisson's figures; E_1, E_2 are Young's modulus. The indexes *1,2* refer to the roller *1* (the shaft neck) and *2* (the needle roller) respectively.

In the case when the axes of the touching rollers (the shaft neck and the needle) are not parallel (Fig. 8 c), the whole load and the stress connected with it concentrate in a relatively small point [13].
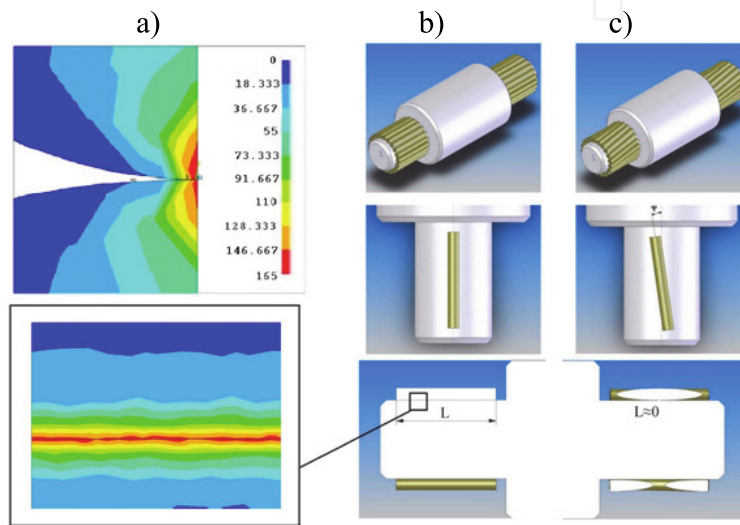

Figure 8. The pictorial diagram of the roller bearing: a) the spread of stresses for the connecting rollers with parallel axes, b) the axes of the parallel elements, c) the axes of the elements shifted by an angle

The area of the friction surface of the elements of bearing is influenced not only by the adequate mutual positioning of the cooperating elements, but also by the changeable relation of the diameters of the needle and the roller, changeable values of Young's modulus, Poisson's ratio of the materials used, and the change of load.

The basic theoretical perspective assumed while calculating contact stresses and the width of the contact area between co-working bodies was based on Hertz's theory, drawing on the following premises: the contacting elements are made from a homogenous, isotopic material; they are limited by the smooth surfaces with a regular curvature; and within the contact area some deformations occur [11, 12, 14].

The analysis of the change in the contact area of the elements of needle bearing presented below was prepared on the basis of all the premises of Hertz's theory concerning the work of two rollers (the shaft neck, the needle roller) with parallel axes, working under static load in dry environment. In the examination of the factors which have an impact on the measure of contact area of the elements cooperating in the form of 'the shaft neck – the needle roller' interaction is particularly important as the contact area influences the change in the moment of motion resistance in the process of bearing.

In the first test eight different needles of 1,4 [mm] up to 2,5 [mm] in diameter were juxtaposed with four rollers of 10, 20, 30, 40 [mm] in diameter. The contact area of the cooperating elements was calculated taking as an assumption the constant load of 200 [N/mm]. Different combinations of elements (needles and rollers) were tested. The bigger the diameter of the needle, the bigger the contact area becomes – this tendency can be measured by means of power equations (Fig. 9.).
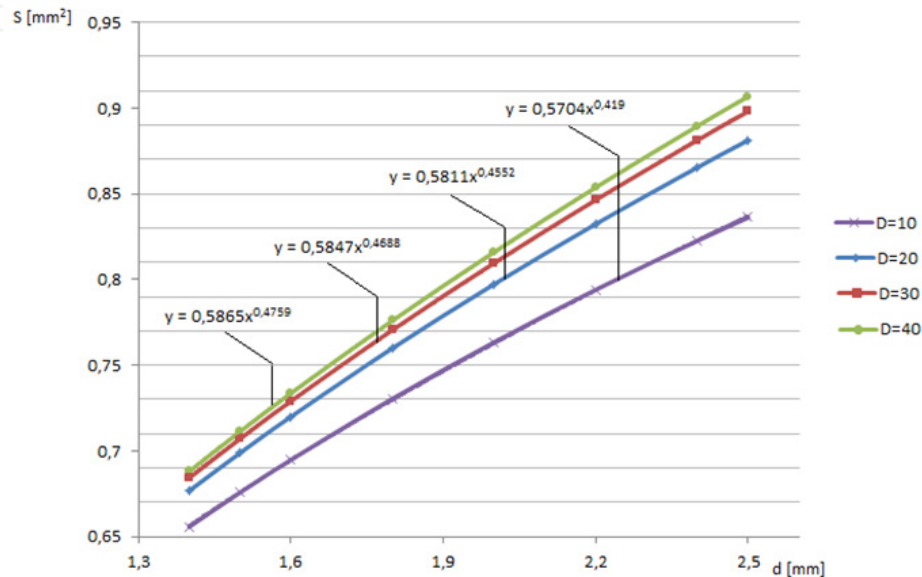

Figure 9. The influence of the needle roller diameter (d) and the shaft neck diameter (D) on the contact area (S) of the elements of bearing.

In the case demonstrated above, the biggest changes in the contact area can be observed while putting together the needles of bigger diameters with the given set of rollers, i.e. for the needle of 1,4 [mm] in diameter cooperating with the rollers of 10 and 40 [mm] in diameter the contact area increased by approximately 5%, while for the needle of 2,5 [mm] in diameter cooperating with the same rollers the contact area increased by approximately 9 [%].

3.2. The mathematical model of the friction process in the needle bearing

The state of the present studies, the information in scientific publications and the catalogues of leading producers of bearings show that the total moment of resistance in the work of a bearing can be understood as the sum of the elemental moments originating in the bearing friction, the sliding friction, the bearing seal friction and the friction occurring in oil environment.

However, in order to estimate the moment of resistance in the friction sliding we suggest the mathematical model in which the total moment of different types of resistance to motion comprises firstly the moment resulting from the bearing friction, calculated as the product of the load – N and the rolling friction ratio - f, and the moment originating from the sliding friction, taking into account load, the sliding friction - μ and the radius of the roller – r (2).

$$M_{Tc} = N \cdot f + N \cdot \mu \cdot r \qquad (15)$$

The recommended examination on a stand (the friction machine SMT-1) makes it possible to obtain the values of the moment of resistance created in the process of sliding friction from the total moment of resistance to motion in the bearing. The needle bearing with the outer ring is mounted directly on the shaft neck which is an interior bearing track for the needles. The outer ring is stationary, and the bearing is loaded with radius force. During the work of the roller turning with the rotational speed of 50-100 rotations per minute there exists only the resistance connected with bearing friction. The observations provided by a torque meter help estimate the moment of bearing friction and the value of the ratio f. Then, if the number of rotations of the roller is increased, in the total moment there will appear the already mentioned sliding friction, the value of which can be calculated as the difference between the total moment of resistance obtained in the trail work of the bearing and the value of bearing resistance established earlier.

Such an examination helps to estimate the extent to which the bearing friction and sliding friction matter in the overall balance of motion resistance. The next step in the examination may be to check the influence of load with constant rotational speeds, and the influence of the change in the lubricant environment.

3.3. The analysis of the friction process as the function of the initial slackness

The bearing slackness is a very important geometrical parameter which determines the intensity and the type of wear. In order to estimate the influence of slackness on the process of wear the examination was carried out for the nominal slackness of 0.065 [mm] and other slacknesses achieved as a sum of the nominal one and the value of reduction of the shaft neck diameter. The associations received in this way made it possible to examine the process of friction with slacknesses of 0.1, 0.2, 0.3, 0.4, 0.5 [mm] respectively. A hundred-hour examination was carried out (on the friction machine SMT-1) for each value of slackness in bearing; next, the micro-structure of the friction surface was examined under an optic microscope with magnification X300 (Fig. 10 b). On the basis of the results received it can established that the enlargement of slackness between the elements in the friction pair leads to a considerable enlargement of the intensity in the shaft neck wear, which, in turn, results in the worse durability of the connection.

Fig. 11. shows the kinetic relations between the wear of the shaft neck and different slacknesses in the function of work time. The lines 1 and 2, corresponding to the slacknesses 0,5 [mm] and 0,4 [mm] respectively, indicate the group with a relatively high rate of wear. A significantly smaller rate of wear is represented by the lines 4-7, corresponding to the slacknesses 0,065 [mm] – 0,3 [mm] respectively (Fig. 11). The dotted line 3, typified as the critical slackness $h = 0,325$ [mm] characterizes the shift from the less intensive, mechano-chemical one, to the more intensive one. The increase of the slackness from the critical $h = 0,325$ [mm] to 0,4 [mm] results in the intensity which is approximately twice bigger. With the slackness 0,5 [mm] the durability of the needle bearing decreases

by 50 [%], if compared with the durability with the critical slackness. Fig. 10 shows the wear of the shaft neck understood as function of the initial slackness. Point A (Fig. 10 a) as the critical slackness with which the friction pair works in the conditions of normal wear was achieved as a result of crossing the tangents led to the curve of wear in areas I and III [13, 15].
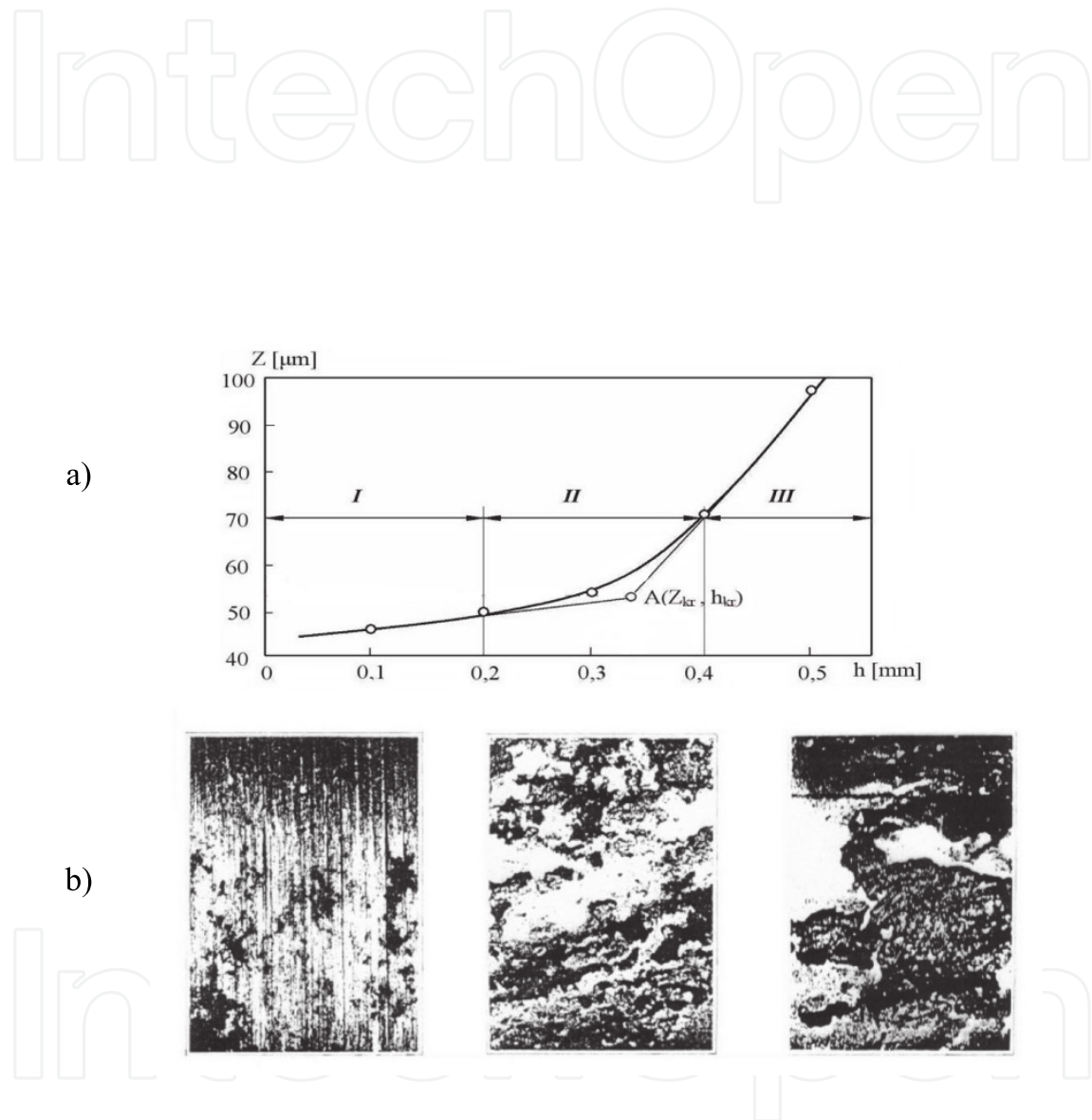
a)

b)

Figure 10. Demonstration of the shaft wear: a) the shaft wear in the function of initial slackness where: I is normal wear, II is mechano-chemical wear, III is pathological wear, b) the structure of the worn surface of the shaft neck x 300
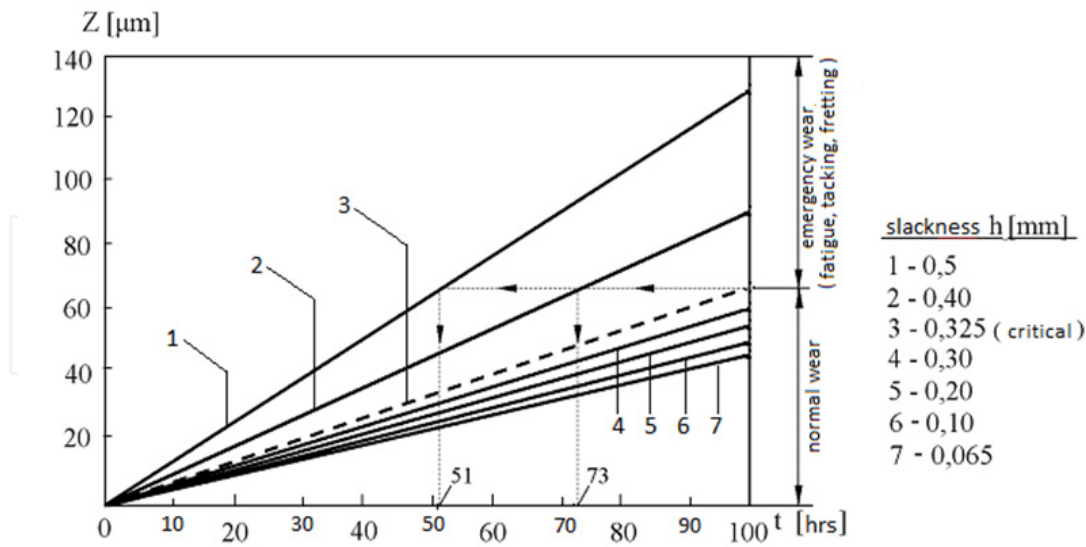
Figure 11. The kinetic relations in the wear of the needle bearing elements (the operating conditions, different slacknesses in association)

In real conditions, when the surfaces of the bodies in direct contact are uneven, and anisotropy of top layer occurs, the problem of body contact becomes a more complicated one. Further examination will focus on the analysis of friction surface of the needle bearing elements in real conditions, in the lubricated environment and under kinematic load.

4. The analysis of friction surfaces of the needle bearing elements

This part gives an analysis of the change in the contact area of elements cooperating under static load in dry environment when two rollers with parallel axes – the needle rollers and the shaft neck – work against each other [16, 17]. Since the nominal contact areas are relatively small – especially when compared with the overall dimensions of the aforementioned elements – pressure in contact areas are significant and accompanied by concentration of stresses. Hertz's solution made it possible to establish the nature of mutual interaction between the bodies, namely the width of contact area and maximum pressure. Next, the area of contact surface was measured, taking into consideration various diameters of the needle and the shaft, various values of Young's modulus and Poisson's ratio and diversified load. The above mentioned will be borne in mind while examining the wear of bearing elements in the energetic perspective. Deformed areas of friction surfaces are characterised by specific features which directly influence the durability of the elements of a friction pair. This, in turn, means that it is essential to choose appropriate friction materials and to analyse their geometry.

4.1. Description of the stand

The examination stand (Fig. 12) consists of an apparatus responsible for loading *F* the bearing, the electric motor (1) with a fluent regulation of rotation speed 0 – 1500 rpm, the torque meter

(2) responsible for constant measurement of the friction moment, the rotation sensor: the shaft (7), the needle roller around the axis of the shaft (6) and its axis of symmetry (5), the thermo-visional camera (8), the converter processing the above-mentioned parameters (9) and the computer for recording data (10). The needle bearing with a fixed, immobile outer ring is placed on the shaft neck and loaded with radius force *F*. The motor drives the shaft neck up to a certain speed and then the rotation sensors, the torque meter and the camera record the rotation speed of the shaft and the needle, the moment of friction and the heat emitted.
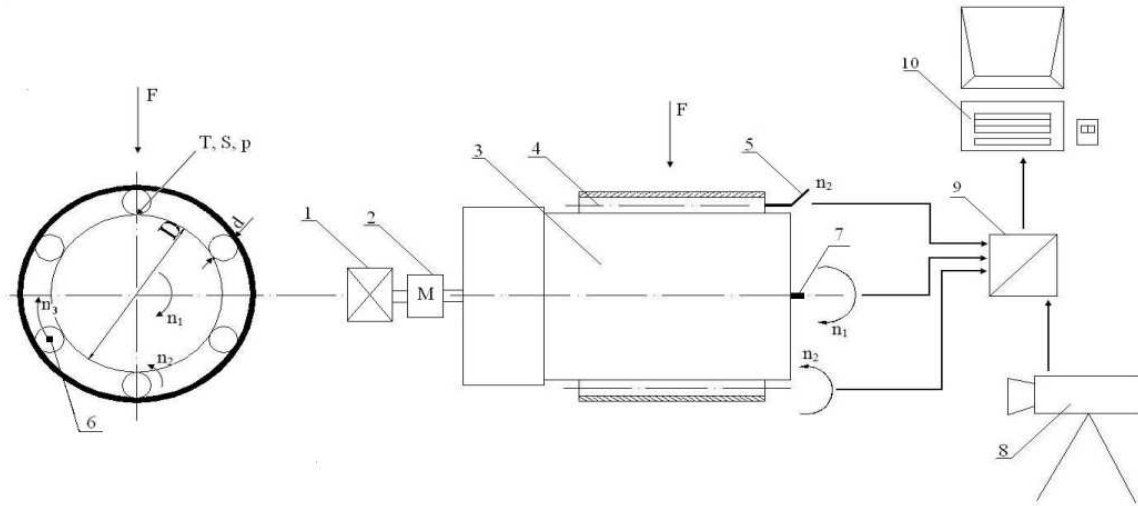

Figure 12. The examination stand: 1 – the source of energy, 2 – the torque meter, 3 – the shaft neck of diameter *D*, 4 – the needle roller of diameter d, 5 – the rotation sensor n2 of the needle against the symmetry axis, 6 – the rotation sensor n3 of the needle around the axis of the roller, 7 – the rotation sensor n1 of the roller, 8 the thermo-visional camera, 9 – the converter, 10 – the computer for processing data

4.2. The analysis of contact areas of friction elements

In the place of contact of two elastic bodies pressed against each other with some force, some contact stresses within a certain field of mutual contact occur. They reach significant values even in the situation when the pressing force is relatively small, which, as a consequence, may lead to exceeding the acceptable limit of the material effort. This is of paramount importance during the work of needle bearings which are under considerable load. Needle bearing requires an adequate positioning of the needles in relation to the shaft neck; in the right position – i.e. when the axes of the elements are parallel – the contact area of these bodies equals the area of the ellipse of the length which is the same as the length of the needle and the width $2b$ calculated by means of Hertz's solution:

$$b = \sqrt{P \frac{D_1 \cdot D_2}{D_1 + D_2} \left(\frac{1 - v_1^2}{E_1} + \frac{1 - v_2^2}{E_2} \right)},$$

where P is the strength per each unit of the rollers' length, D_1, D_2 are the diameters of the shaft neck and the needle respectively, v_1, v_2 are Poisson's E_1, E_2 are Young's modulus. The indexes 1,2 refer to the roller 1 (the shaft neck) and 2 (the needle roller) respectively.

The area of the friction surface of the elements of bearing is influenced not only by the adequate mutual positioning of the cooperating elements, but also by the changeable relation of the diameters of the needle and the roller, changeable values of Young's modulus, Poisson's ratio of the materials used, and the change of load.

The basic theoretical perspective assumed while calculating contact stresses and the width of the contact area between co-working bodies was based on Hertz's theory, drawing on the following premises: the contacting elements are made from a homogenous, isotopic material; they are limited by the smooth surfaces with a regular curvature; and within the contact area some deformations occur (Dietrich, 2003). The analysis of the change in the contact area of the elements of needle bearing presented below was prepared on the basis of all the premises of Hertz's theory concerning the work of two rollers (the shaft neck, the needle roller) with parallel axes, working under static load in dry environment. In the examination of the factors which have an impact on the measure of contact area of the elements cooperating in the form of "the shaft neck – the needle roller" interaction is particularly important as the contact area influences the change in the moment of movement resistance in the process of bearing.

In the first test eight different needles of 1,4 [mm] up to 2,5 [mm] in diameter were juxtaposed with four rollers of 10, 20, 30, 40 [mm] in diameter. The contact area of the cooperating elements was calculated taking as an assumption the constant load of 200 [N/mm]. Different combinations of elements (needles and rollers) were tested. The bigger the diameter of the needle, the bigger the contact area becomes – this tendency can be measured by means of power equations (Fig. 9).

In the case demonstrated above, the biggest changes in the contact area can be observed while putting together the needles of bigger diameters with the given set of rollers, i.e. for the needle of 1,4 [mm] in diameter cooperating with the rollers of 10 and 40 [mm] in diameter the contact area increased by approximately 5 [%], while for the needle of 2,5 [mm] in diameter cooperating with the same rollers the contact area increased by approximately 9 [%].

Fig. 13. shows the influence of Young's modulus E, Poisson's ratio v and the roller diameter D on the area S under a constant load. The range of Young's modulus assumed in the test was identical with the one meant for the materials used in the construction of shafts and bearings. The increase in the contact area of cooperating elements when E and v are changed is not significant; in the situation illustrated in Fig. 13, the increase approximates to mere 0.8 [%] when E is doubled. This is connected with the assumed load which is transmitted onto the whole length of the needle (the axes of the shaft and the needle roller are parallel). In case of materials that are characterized by Young's modulus of 200 [GPa] and more, the assumed load of 200 [N/mm] does not enforce any deformation in the cooperating elements, i.e. the needle of 2 [mm] in diameter and a set of rollers of 20, 30, 40 [mm] in diameter.
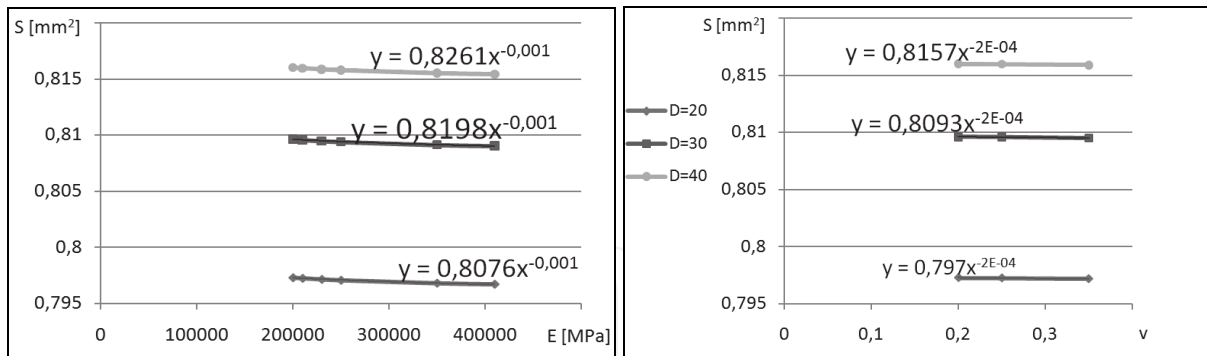
Figure 13. The influence of Young's modulus – E of the material, Poisson's ratio – v, the diameter of the shaft neck (D) on the contact area of the elements of bearing (S)

Assuming that the task of loading is an indispensable factor in creating stresses and deformations in the needle roller and the shaft neck, the influence of radius strength P on the contact area becomes obvious.

From the analysis of the diagram in Fig. 13 it is clear that the increase in the area S is directly proportional to the growing strength P. In the test a needle roller of 2 [mm] in diameter (E = 210 [GPa], v = 0.3) was used as well as a set of four rollers of 10, 20, 30, 40 [mm] (E = 200 [GPa], v = 0.25) in diameter. The diagram of the increase in the contact area with the load of bearing by radius force ranging from 200 to 450 [N/mm] is reflected by power function (Fig. 14) which is connected z hardening of the contact area under the influence of growing force.
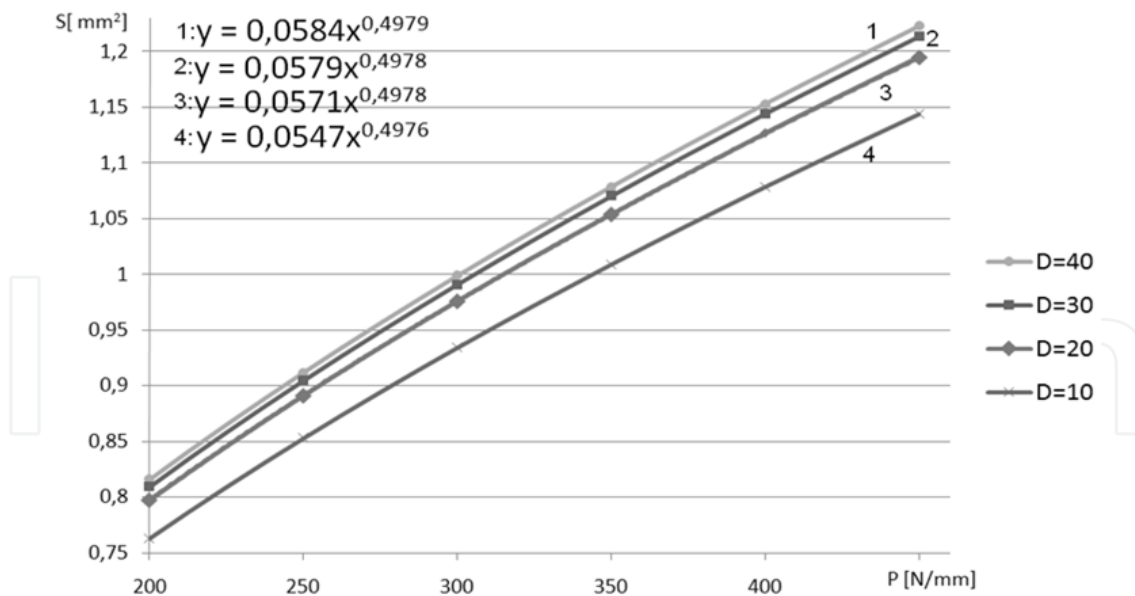

Figure 14. The influence of load (P) and the diameter of the shaft neck (D) on the contact area of the elements of bearing (S)

To model the process of friction, to calculate the rotational speed of the shaft and the needle roller, and to estimate the importance of slide resistance in the overall process of friction, the following examination stand was used [18,19].

5. Summary

The work refers to different aspects of research methodology and the difficulties in the interpretation of the durability and reliability of the bearings. The proposition of own methods of research and empirical descriptions was done. The material presented in the work was divided into four separate subsections: on research methodology, work, geometry, and the impact of these elements on the friction and wear. All tests were performed for three replicates at each point. The investigations were conducted with using different tribological testers and tribological stands. Should add that all of the test, later in this chapter bearings are factory items and their designations are in accordance with PN, PN-EN and ISO standards. All tests were performed for three replicates at each point.

On the basis of the test results the following conclusions have been made:

The results of tribological tests indicate different friction characteristics of the evaluated elements. Bigger resistance to motion is noticed at the beginning of the test. Afterwards it becomes stable – at minimal level, at the end of the process. The data included in this section indicate entirely different test results obtained with two different methods. This may lead to totally different conclusions concerning the tribological properties of tested elements. The data presented in fig. 2, 3 and 4 clearly indicate the lack of correlation between MB-1 and MB-2 methods applied in the tests. This means that conducting a scientifically reliable evaluation of tribological properties of such elements requires applying one method. However, the results obtained with this method may not reflect the actual friction nodes when one is unable to faithfully reproduce their actual working conditions. The conducted examination and simulations have demonstrated the changes in the contact area of friction pair elements when two rollers with parallel axes (i.e. the needle roller and the shaft neck) press against each other. Examination of the factors that can have an influence on the contact area of cooperating elements, i.e.: changes in the needle roller diameter and the change in the shaft diameter, are of particular importance because the contact area alters the resistance moment of motion in the bearing. The suggested mathematical model and the examination on a stand help to estimate the extent to which the bearing friction and the sliding friction matter in the overall balance of motion resistance. While examining the shaft neck wear in the function of initial slackness, the conditions of movement from the normal wear to the pathological one were established by means of setting up the critical parameters: the slackness, the intensity of wear in the function of load. It has been observed that the movement from the normal wear to adhesion is characterized by a gradual increase of the intensity of the elements wear. The conducted examination and simulations have demonstrated the changes in the contact area of friction pair elements when two rollers with parallel axes (i.e. the needle roller and the shaft neck) press against each other. Examination of the factors that can have an influence on the contact area of cooperating elements, i.e.: changes in the needle roller diameter and the change in the shaft diameter, changes in the value of Young's modulus, Poisson's ratio of the materials used and changes of load are of particular importance because the contact area alters the resistance moment of movement in bearing. The analysis of the above-mentioned relations in the energetic perspective made it possible to conclude that deformed areas of friction surfaces are

characterized by specific properties which influence the durability of the elements of the friction pair. This means that the materials meant for friction pairs should be properly selected and their geometry carefully analyzed. While examining the shaft neck wear in the function of initial clearance, the conditions of movement from the normal wear to the pathological one were established by means of setting up the critical parameters: the clearance, the intensity of wear in the function of load. It has been observed that the movement from the normal wear to adhesion is characterized by a gradual increase of the intensity of the elements wear. The examination was carried out for the nominal clearance of 0.065 mm and other clearances achieved as a sum of the nominal one and the value of reduction of the shaft neck diameter. The associations received in this way made it possible to examine the process of friction with clearances of 0,1, 0,2, 0,3, 0,4, 0,5 [mm] respectively. A hundred-hour examination was carried out for each value of clearance in bearing; next, the micro-structure of the friction surface was examined under an optic microscope with magnification ×300, which lead to the distinction of three different surfaces of wear characteristic for normal, mechano-chemical and pathological wearing (Fig. 10 b). On the basis of the results received is can established that the enlargement of clearance between the elements in the friction pair leads to a considerable enlargement of the intensity in the shaft neck wear, which, in turn, results in the worse durability of the connection. Fig. 10 a shows the wear of the shaft neck understood as function of the initial clearance. Point A as a critical clearance with which the friction pair works in the conditions of normal wear was achieved as a result of crossing the tangents led to the curve of wear in areas I and III (Nachimowicz et al., 2007). In real conditions, when the surfaces of the bodies in direct contact are uneven, and anisotropy of top layer occurs, the problem of body contact becomes a more complicated one. Further examination will focus on the analysis of friction surface of the needle bearing elements in real conditions, in the lubricated environment and under kinematic load.

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