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Preliminary communication

Influence of Housing Material and Geometry on Thermal Stability of Threaded Spindle Bearing Assembly

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Abstract: The accuracy of guiding a spindle is greatly influenced by the thermal load of the bearing. Increased thermal load implies deformation of the bearing parts which is directly reflected in the work quality of the machine system. For this reason, the aim of this paper is to analyse the influence of the material and the geometry of the housing on the temperature field of the threaded spindle bearing assembly, which was realised by "radially mounting" of axial angular contact ball bearing ZKLN. This paper represents a new approach for thermal analysis of the special bearings, type ZKLN which are not involved in the relevant standards for the determination of the reference speed. At the same time the value of this paper finds itself in finding of the guideline at engineering practice which is very important at early design phase of the systems with threaded spindle. The analysis was carried out using experimental and numerical methods. Combination of the mentioned methods gave result which was used to define appropriate guideline regarding influence of housing material and geometry on thermal stability of the ZKLN bearing.

Keywords: FEA thermal analysis; housing; threaded spindle; ZKLN axial conatct ball bearing

1 INTRODUCTION

In accordance with the development of tool materials and the increase in cutting speed, the design of cutting tools and machine tools and processing systems has changed, creating the conditions for using the available possibilities of modern tool materials, both in terms of increasing the cutting speed, and in terms of productivity, economy, accuracy, and processing quality. The quality of the final product largely depends on the quality, i.e. machine system precision. This implies the accuracy of the work of the tool (tool heads and guide spindles) but also the accuracy of guiding the workpiece, which depends on the accuracy of the threaded spindle of the machine table. The operating conditions of new generation machine systems are based on the highest productivity, i.e. on as much final products as possible in a unit of time. In order to meet the basic condition, it is necessary to increase the processing speed significantly, which further implies an increase in speed. As the speed rises, the level of friction in the bearing increases, resulting in a significant bearing load on the bearing assembly. At the moment when the thermal load exceeds the limit value, excessive elastic deformations of the bearing parts occur, which then quickly turns into plastic deformations. After plastic deformation, the system loses optimum (projected) accuracy of operation, which in time leads to a significant loss of quality of the final product. In order to avoid excessive thermal loading of the bearing and the threaded spindle, the correct choice of housing and bearing should be taken into consideration at the design stage of the machine system. To this end, large world-renowned bearing manufacturers offer a wide range of different types of bearings, designed specifically for this purpose. Roller bearings intended for mounting of the threaded spindles are classified as axial bearings. In this paper, the bearing ZKLN2557-2Z, of the German manufacturer Schaeffler, which produces this series of bearings under the INA brand, was taken as a representative. The aim of this study is to research the influence of material type and housing geometry on the temperature field of the bearing assembly. The idea is that the obtained results provide a guideline that, at the design stage, can define relevant parameters that will help to optimize the bearing assembly. To this end, the paper presents a thermal numerical analysis of the mentioned bearing, which is combined with housings of various current materials, as well as the effect of changing the housing dimension (considered only for the steel housing) on the bearing assembly temperature field. The housing materials tested in this paper are steel, aluminum, gray cast iron (EN-GJL250), and mineral cast. Previous research has in many cases focused on the thermal stability of bearings used in tool heads and guide spindles. The starting point for the study of thermal stability is certainly the determination of the reference speed (thermal limit speed). This procedure is shown in DIN732 part one and part two [1]. The causes of heat load have been examined in various aspects. Thus, Li et al. in [2] gave an analysis of heat generation in a angular contact ball bearing for different values of preload, number of revolutions, and contact angle. Truong et al. calculated the warm spread and obtained the stiffness matrix for the ball bearings with oblique contact [3]. Ngo et al in [4] combine the inverse method with a high-speed spindle ball bearing model to determine the constant-bias bearing characteristics under real operating conditions. The conclusion of their study is that the stiffness of the bearing, as well as the thickness of the lubricants, change nonlinearly with the increase in the speed of the spindle, while the thermal effects have a significant effect on the thickness of the lubricant film. Chen et al. in [5], using thermal and quasi-static analysis, calculated the heat between the high-speed ball bearing elements (parts) as well as the total heat taking into account the rpm, load, and other parameters. Their research has shown that in addition to the number of revolutions, the axial load and the curvature coefficient of the inner ring have the greatest influence on heating. Zhang et al. mentioned in [6] optimize the thermal boundary conditions applied in the thermal FEA analysis of tool spindles using the hybrid artificial bee colony (ABC) method a surface reaction model. This optimization greatly improves the accuracy of the tool spindle simulation. Zhenjun et al. combine an optimization algorithm and experimental results propose a new numerical calculation method used to identify heat sources [7]. Andrei et al. provide an overview of studies on the dynamics of the bearing-spindle system [8]. Dong et al. in their work [9] present a method for determining the optimal bearing bias in a wide range of speeds that is applicable in the design of machine tool spindles. In [10, 11] authors test the influence of velocity on the dynamic behavior of a ball bearing and determine the optimum bearing bias using a combination of fatigue model and internal load distribution. Zhang et al. provide a comparative model that analyses the stiffness of angular contact ball bearings that are preloaded by different preload mechanisms [12]. Wang et al. propose a dynamic model for the sliding test in angular contact ball bearings. This model also considers the interaction between the balls and the rolling path as well as the cages and the lubricant [13]. Zhaohui et al. propose a five-degree-of-freedom model to analyze the effect of cage clearance on heating [14]. Krstic et al. presented a thermal FEA analysis of a bearing assembly realised by axial angular conatct ball bearing [15], while an experimental study of the thermal limit number of revolutions of ZKLF type axial ball bearing was presented in [16]. As can be seen, there are many analyses which try to describe thermal stability and behavior of the spindle-bearings. Most of them refer to the standard spindle-bearings, so the idea of this analysis is to introduce new sub-group which is different constructively in comparison with the standard spindle-bearings. Different bearing construction requires a completely new approach which will be presented in this paper. On the other hand this kind of bearings (ZKLN) are not involved in the relevant standard which can be used by thermal load evaluation, so that is the second reason why the analysis in this paper was carried out. When the thermal load is known, the last question is: what are the influnce parameters? The answer to this question will be given in this paper. At the same time it will be a guideline for the further engineering practice.

2 BEARING CONSTRUCTION FOR "RADIAL" THREADED SPINDLE MOUNTING

Since the threaded spindle bearing is a highly demanding task in mechanical engineering, bearing manufacturers have developed specialized series of bearings that will successfully meet all the requirements of this application. In this paper, the bearing ZKLN2557-2Z, of the German manufacturer Schaeffler, will be presented. In order to understand further analysis better, it is necessary to become familiar with the construction of this bearing.

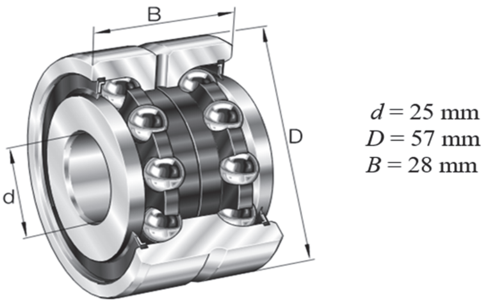

Figure 1 Construction of bearing ZKLN2557-2Z [17]

The main features of this bearing are the two-piece inner ring through which the bearing is further preloaded with the help of a special tightening nut, as well as a larger contact angle of the rolling bodies and a rolling path of 60. The bigger contact angle ensures that the bearing can transmit a greater axial force, which is characteristic of the threaded spindles. Fig. 1 shows the bearing structure of the ZKLN. The corresponding precise nut for preloading is AM25 from the same manufacturer. Its construction is shown in Fig. 2.

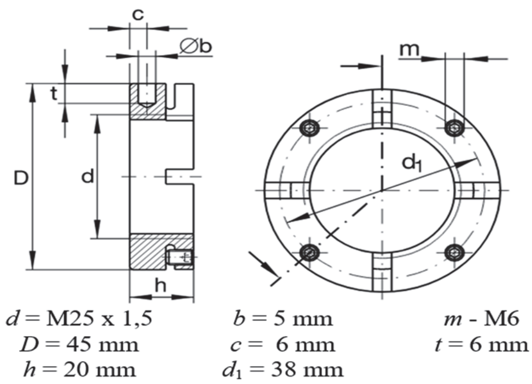

Figure 2 Construction of precision nut for preloading of the bearing [17]

As the bearing structure was shown, it can be clearly concluded that the system to be explored in this paper is expanded by additional elements relating to the precision preloading nut, the cover that closes the housing, the housing, and the screw connection that fixes the housing cover to the housing. Given the additional elements of the system, a completely new approach in the analysis of the temperature field of the bearing assembly is expected in advance. Previous research has been based on standard bearings that did not include additional elements introduced by the ZKLN type bearing.

3 EXPERIMENTAL SETUP AND NUMERICAL SIMULATION

In order to determinate a reference speed for the bearing ZKLF2575-2Z, as the start point experimental research was carried out. The experiment was carried out through the following phases: mounting and prestressing of the bearing on the shaft (prestressing realised through precision locknut and disc springs), measurement of the speed, friction torque, temperature of the inner and outer ring and the environment as well as prestressing force in axial direction. Test rig is shown in Fig. 3.

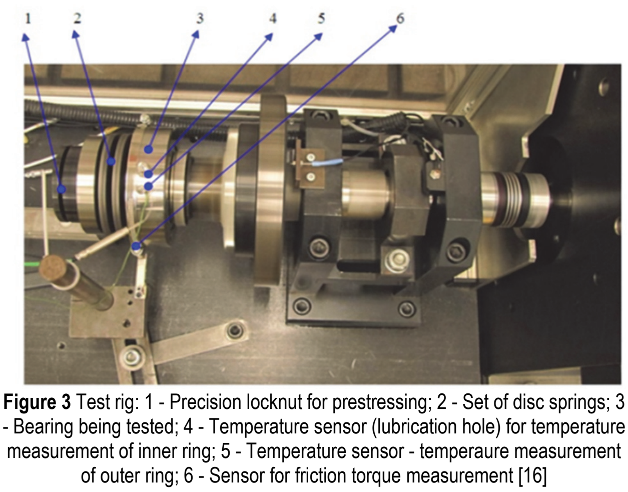

Figure 3 Test rig: 1 - Precision locknut for prestressing; 2 - Set of disc springs; 3 - Bearing being tested; 4 - Temperature sensor (lubrication hole) for temperature measurement of inner ring; 5 - Temperature sensor - temperaure measurement of outer ring; 6 - Sensor for friction torque measurement [16]

As it is shown in Fig. 3 test bench consists of the electric motor and shaft which will simulate a threaded spindle. At the end of the shaft the bearing was fastened with a precision locknut (axial force F = 1945 N according to Schaeffler) and additionally prestressed with disc springs. Torque is transmitted from the electric motor via the shaft to the inner ring of the bearing. The outer ring was stationary and used for installing of a measurement sensor for friction torque, temperature of the outer ring and inner ring (through the lubrication hole). All mentioned sensors are conected to the main data acquisition device by appropriate cables. Data acquisition device is equipped with software for automatic data collection (values for temperature and total friction torque) which records data in real time. Besides data acquisition this device provides visualization of the measurement process. Since the bearings were sealed on both sides and factory lubricated (total amount of fat L192 was 1,9 g) , it was necessary to first distribute the grease, i.e. form an even grease film on the contact surface. This was realized with two stages of number of revolutions lasting 15 minutes each (500 and 1000 rpm). After that, each sample was gradually loaded in the speed range 0 - 8000 rpm. Each load level lasted 30 minutes. The last one corresponding to the maximum number of revolutions lasted 5 minutes. Additional condition after each load cycle was a constant prestressing force, which was measured after each cycle. Number of the tested samples was four. Experimental results are shown in Fig. 4. Abscissa shows the speed (number of revolutions) and ordinate shows friction torque as well as the temperature of the outer ring. The main condition for determining the reference speed is that the bearing is in a state of temperature equilibrium (temperature of the outer ring of 70 °C [1]). From the diagram, Fig. 4, it can be seen that none of the samples reached reference temperature and that is why it was not possible to apply heat balance as an insturment for determination of the reference speed.

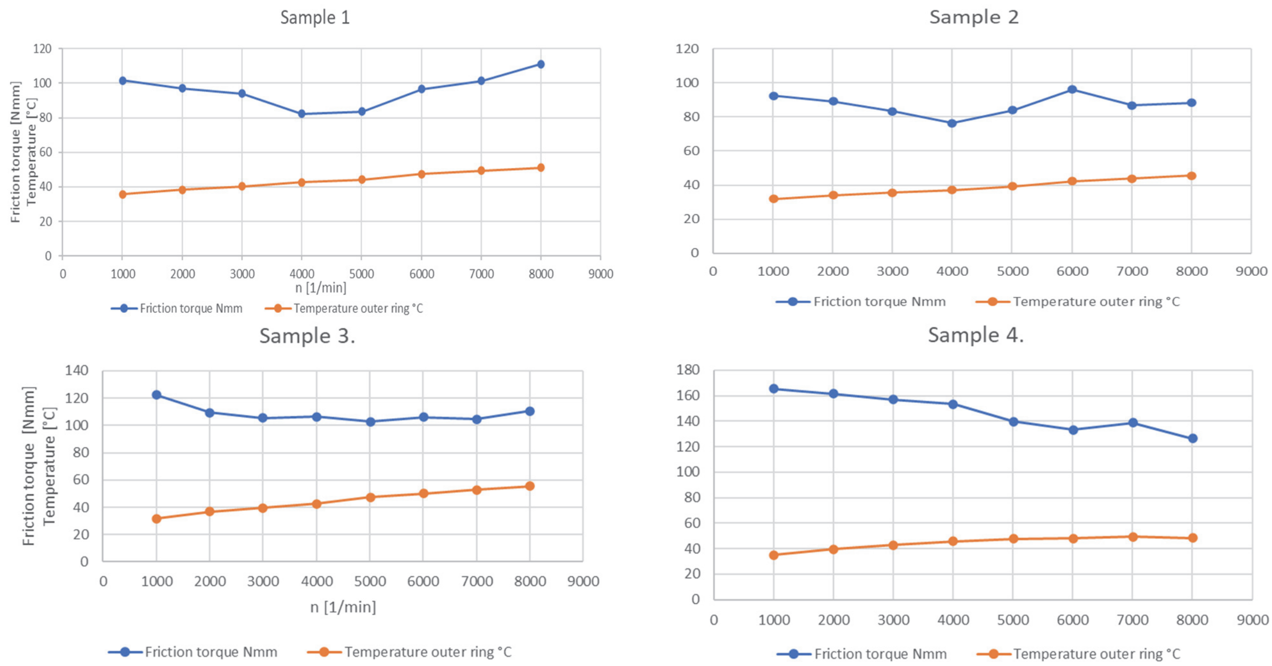

Figure 4 Experimental results

Since the main reference condition (temperature of the outer ring of 70 °C) was not fulfilled, it was necessary to make additional simulation (numerical analysis) in order to get thermal load of the bearing. The logical question is why the temperature of the outer ring did not meet this refernce condition? The reason for that is the absence of a sufficiently large axial force as well as the absence of screw connection (limitations of the test rig). That is why it was necessary to make additional simulation. At the beginning a bearing type had to be defined which will have the same reference surface as the sample by experimental research as well as simpler construction so the simulation can be more precisely carried out. Regarding these preconditions the type ZKLN was chosen. As the bearing type was chosen, next step was defining of bearing assembly as well as carrying out the analysis. Knowing the temperature field, and with it the thermal load of the bearing, the question arises, which also becomes the goal of this paper, and that is, whether the type of material from which the housing is made can affect the reduction of the thermal load of the bearing and to what extent? An additional question is whether the geometry or the dimension of the housing affects the temperature field of the bearing assembly? Since the task has been defined, the research methodology will follow which will answer the questions asked. The research plan is defined in the following stages/phases.

1st Phase - the formation of a geometric model of the bearing for the case of "radial" mounting of the bearing ZKLN2557-2Z.

2nd Phase - Determination of the operating speed i.e. generated amount of heat for which there is a temperature equilibrium (outer ring temperature of 70 °C [1]).

3rd Phase - Determination of the temperature field of the bearing model with the housing made of the following materials: Steel,Aluminum, Gray cast iron EN-GJL250 and Mineral cast.

4th Phase - In the case of the model with steel housing additional simulation with a modified housing dimension (larger housing).

ABAQUS/CAE, version 6.9 - 3 was used for numerical analysis. An axially symmetrical thermal FE model with DCAX3 elements was used.

3.1 Thermal Simulation

3.1.1 Geometric Model of the Bearing Assembly

The geometric model of the bearing assembly was made by using the bearing ZKLN2557-2Z given in Fig. 5. For ease of interpretation, Fig. 5 shows the described parametric model.

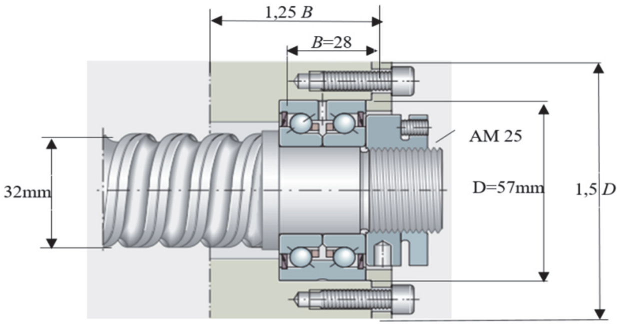

Figure 5 Bearing model defined parametrically

3.1.2 Determination of Generated Heat Quantity

In order to determine the temperature field of the bearing assembly, it is necessary to know the process of generating thermal energy. The main cause of heat in the bearing is the bearing friction inside of the bearing generated on the contact surfaces between the rolling elements and the rolling path, the same as the friction inside the lubricant and the friction on the bearing edges as well as seals. In order to establish the correlation between the quantities that govern friction, the term - frictional torque is defined. Previous research has shown that the total friction, i.e. the total frictional torque in the bearing consists of the frictional torque in function of speed and the frictional torque in function of load. In the physical interpretation, the frictional torque which is a function of speed describes the friction that occurs inside the lubricant, while the frictional torque which is a function of load describes the friction that is conditioned by real forces (mechanical load). The mathematical form of the total frictional torque in the bearing is [18]:

$$M = M_0 + M_1 \tag{1}$$

respectively, where: M is the total friction torque, M_0 is frictional torque as a function of speed and M_1 frictional torque as a function of load.

The frictional torque in function of speed is described by the following equation [18]:

$$M_0 = f_0 \cdot 10^{-7} \cdot (\nu \cdot n)^{2/3} \cdot d_m^3 \tag{2}$$

where f_0 is the bearing factor for frictional torque as a function of speed, taking into account the type of bearing and the method of lubrication [18], ν is kinematic viscosity of lubricant at operating temperature i.e. viscosity of oil or base oil at operating temperature in case of grease lubrication [18], n is operating speed, $d_m = \dfrac{d+D}{2}$ is mean bearing diameter.

The frictional torque as a function of load is determined by the following correlation [18]:

$$M_1 = f_1 \cdot P_1 \cdot d_m \tag{3}$$

where: f_1 is bearing factor for frictional torque as a function of load, which takes into account the magnitude of the load [18], P_1 is decisive load for frictional torque as a function of load [18].

When the total friction torque is known, then the total thermal friction load in the bearing can be determined as in [19]:

$$N_{Fr} = \omega \cdot M = \frac{\pi \cdot n}{30} \cdot M \cdot 10^{-3} \tag{4}$$

For the reference conditions based on Eq. (1) to Eq. (4), the total thermal load of the "threaded spindle" system takes the following form:

$$N_{Fr} = \frac{\pi \cdot n}{30} \cdot 10^{-3} \cdot \begin{bmatrix} f_{0r} \cdot 10^{-7} \cdot (\nu_r \cdot n_{\vartheta r})^{2/3} \cdot d_m^3 + \\ + f_{1r} \cdot P_{1r} \cdot d_m \end{bmatrix} \tag{5}$$

where: $n_{\vartheta r}$ = 2350 rpm is operating speed (half of the value n_G in [21]), for the first iteration, f_{0r} = 4 is bearing factor for frictional torque as a function of speed, depends on bearing type and lubrication for reference conditions [22], ν_r = 22 mm²/s is kinematic viscosity of oil or base oil at grease for reference conditions and reference bearing temperature ϑ_r (greasing, a value from [1]), $f_{1r} = 0{,}001 \cdot \left(\dfrac{F_a}{C_0}\right)^{0{,}33} = 0{,}000371$ is bearing factor for frictional torque as a function of load depends on bearing type and load for reference conditions [21], $F_a = P_{1r} = 0{,}05 \cdot C_0 = 0{,}05 \cdot 55000 = 2750$ N is decisive-equivalent dynamic bearing load for reference conditions (reference load) for determination of frictional torque as a function of load [21] and d_m = 41 mm is mean bearing diameter.

3.1.3 Defined Simulation Model and Thermal Analysis Results

The simulation model is given in Fig. 6, Fig. 7 and Fig. 8. When determining the heat transfer coefficient related to radiation, the emission of dark areas ε = 0,8 was adopted, while in the part related to convection, the heat transfer coefficient was determined by the Nusselt number. In order to describe the conditions prevailing on the contact surfaces as realistically as possible in the numerical analysis, it is for them the model as such mid-air (FE - Spec. 000.2) [23]. FEA thermal analysis is performed iteratively according to the algorithm given in Fig. 9.

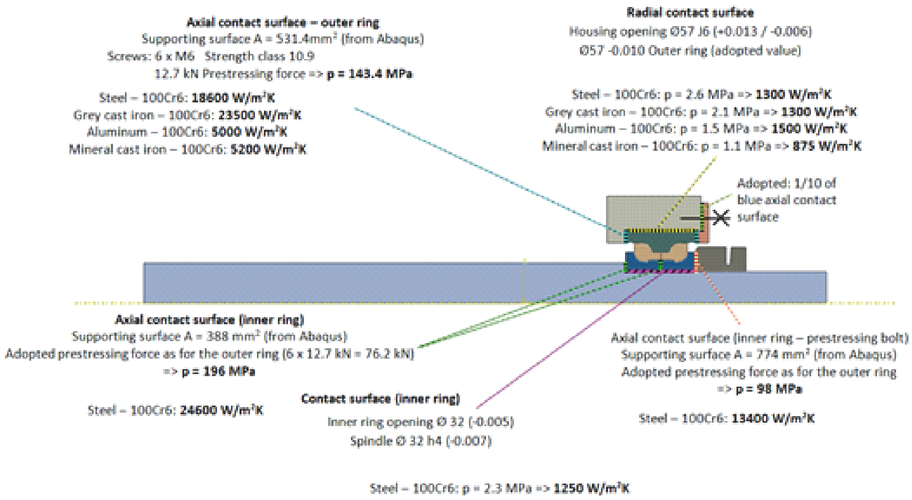

Figure 6 Contact surfaces in the bearing assembly [20]

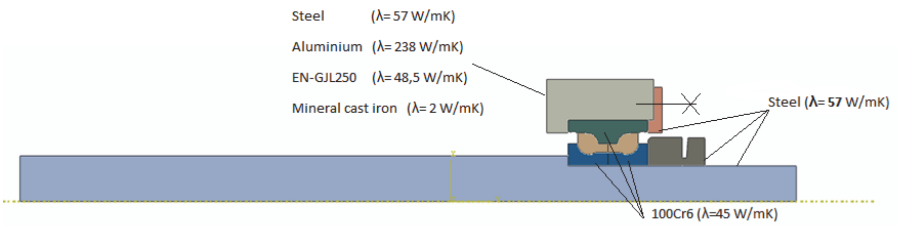

Figure 7 Thermal conductivity of materials in the bearing assembly

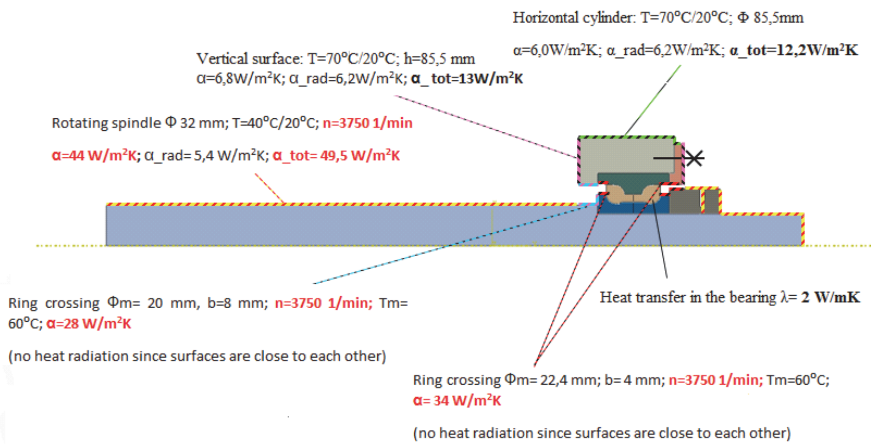

Figure 8 Heat transfer coefficients of the reference surfaces in the bearing assembly

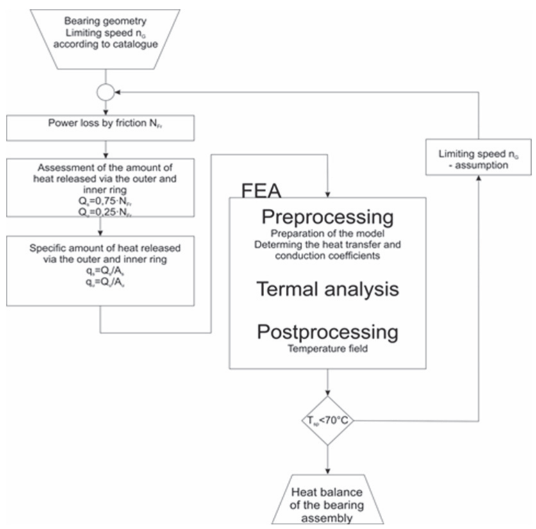

Figure 9 Algorithm for thermal analysis of threaded spindle bearing assembly

For the first iteration, the operating speed $n_{\vartheta r} = n_G / 2 = 2350$ rpm was accepted [21]. With each subsequent iteration, the bearing operating speed increases until the temperature of the outer ring reaches 70 °C. For each value of $n_{\vartheta r}$, the heat generated through the outer and inner rings is determined, and ¾ of the generated amount of heat is transferred through the outer ring to the housing, and the rest of the heat via the inner ring to the threaded spindle. From the condition [1] that the equilibrium temperature is reached when the temperature of the outer ring of the bearing reaches 70 °C, it was necessary to do a couple of iterations of the numerical simulation. By gradually increasing the operating speed, and for the number of revolutions $n_{\vartheta r} = 3750$ rpm, a generated amount of heat was obtained by friction $N_{Fr} = 36$ W which resulted in an outer ring temperature of about 69 °C, which is technically acceptable for further analysis of the temperature field. The amount of heat dissipated through the outer ring:

$$\dot{Q}_a = \frac{3}{4} N_{Fr} = \frac{3}{4} \cdot 36 = 27 \text{ W} \tag{6}$$

respectively, the inner ring:

$$\dot{Q}_i = \frac{1}{4} N_{Fr} = \frac{1}{4} \cdot 36 = 9 \text{ W} \tag{7}$$

For the surface of the external rolling path A_a = 1788 mm^2 the heat flux is:

$$\dot{q}_a = \frac{\dot{Q}_a}{A_a} = \frac{27000}{1788} = 15{,}1 \text{ mW/mm}^2 \tag{8}$$

That is, for the inner rolling path A_i = 1420 mm^2 the heat flux is:

$$\dot{q}_i = \frac{\dot{Q}_i}{A_a} = \frac{9000}{1420} = 6{,}3 \text{ mW/mm}^2 \tag{9}$$

The thermal load of the bearing for which the heat balance exists is given in Fig. 10. The results of the thermal analysis are given in Fig. 11 to Fig.15.

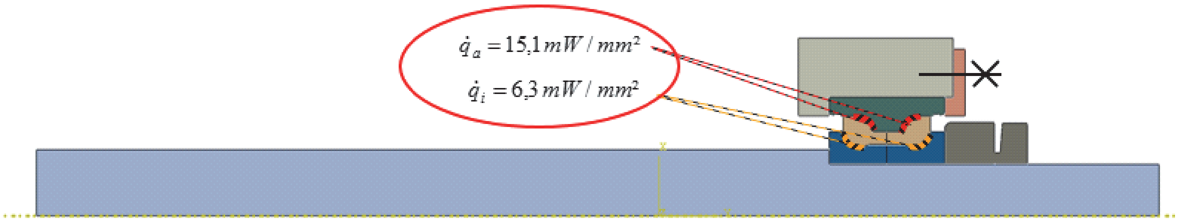

Figure 10 The thermal load of the bearing assembly for which the heat balance exists

Figure 11 Bearing assembly's temperature field with steel housing

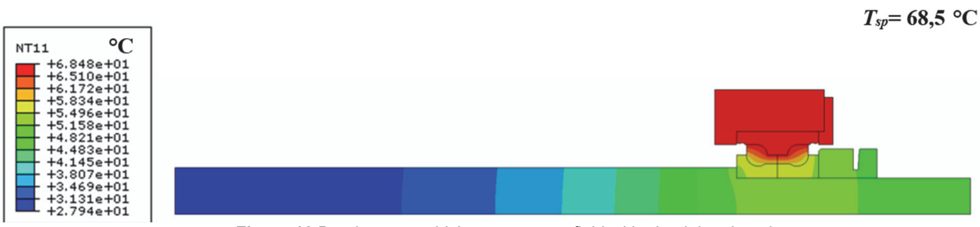

Figure 12 Bearing assembly's temperature field with aluminium housing

Figure 13 Bearing assembly's temperature field with grey cast housing

Figure 14 Bearing assembly's temperature field with mineral cast housing

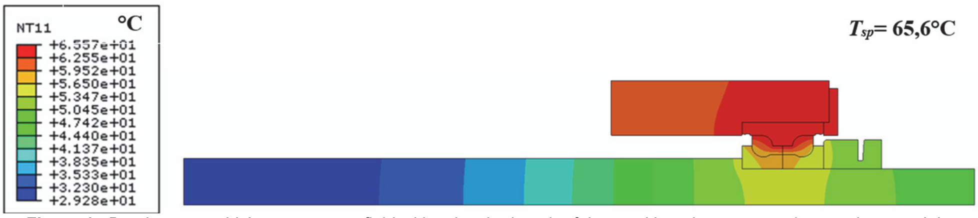

Figure 15 Bearing assembly's temperature field with twice the length of the steel housing compared to previous models

4 DISCUSSION OF RESULTS

As per conclusion of research, the obtained results were edited. By comparing them, a conclusion was obtained, which is also the answer to the question, or better the research goal was gained. The results obtained differed by about 2 °C. This data answers the question of whether the type of material affects the temperature of the bearing assembly. The answer is that the type of housing material does not have a significant effect on the temperature load of the threaded spindle bearing assembly. For this reason, the type of housing material cannot be one of the factors for optimizing the bearing assembly's temperature field because it obviously does not have a significant effect on it. Mineral cast is an ideal material for load-bearing construction wherever dynamic and at the same time, very precise movement is required, with clear technological, economic, and environmental advantages over steel or cast iron. In the case of the application of this material for the bearing housing, a more favorable temperature distribution to the machine body was observed. Also, in the case of cast iron housing, the temperature of the outer ring of the bearing was approximately 70 °C, which is optimal for bearing operation because at this temperature heat balance is established, so the amount of newly generated heat energy is equal to the amount of heat removed from the bearing (from the aspect of thermal stability). The second part of the analysis dealt with the examination of the influence of the geometry (dimensions) of the housing on the thermal load. This part concerned the case with a housing made of steel. The basic model is compared to a model that has a twice as big (longer) case. Comparing the obtained results, a difference of 4 °C was found. For this reason, it is concluded that even the geometry (size) of the housing is not an influential factor through which the optimization of the temperature field could be performed. One of the advantages of a larger housing is better i.e. more favorable temperature distribution to the machine body.

5 CONCLUSION

As is presented in chapter 4, material and geometry of housing do not have significant influence on thermal stability of threaded spindle bearing assembly. This conclusion generates a new question what is the influence parameter on thermal stability of this kind of bearing assembly? In order to give an answer to this question it is necessary to carry out an additional research but now with a similar bearing construction with "axial mounting" which contains additional elements - screw connection. After that it will be possible to compare obtained results which will indicate the main influencing parameter on the thermal stability of this kind of bearing assembly. At the end of this paper, it can be concluded that neither the geometry nor the type of material of the housing affect the thermal stability. It should be noted that the conclusions drawn in this study are valid for the model as well as the conditions stated in this analysis and that this analysis should primarily be understood in a qualitative sense.

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