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Preliminary communication

Thermal Load Analysis of the Threaded Spindle Bearing in Terms of the Housing Construction for the Case of "Axial Assembly" of the Axial Angular Contact Ball Bearing

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Abstract: Modern engineering increasingly requires certain guidelines/procedures for faster, simpler and more efficient optimization of machine tool systems even in the design phase. Accordingly, the task of this paper has been defined, which is focused on the analysis of the impact of the machine housing construction on the thermal loads of the threaded spindle bearing. The focus of the work is on the bearing performed using an axial angular contact ball bearing of the ZKLF type from the German manufacturer Schaeffler. The result of this research gave a guideline to designers to optimize the thermal load of the bearing of the threaded spindle made using the mentioned type of bearing even in the design phase.

Keywords: ball bearing; housing; thermal load; threaded spindle

1 INTRODUCTION

Large machine tool systems for large-scale and mass production are most often composed of one or more technological lines that are specific, i.e. optimal way of connecting each other. In order for the entire system to function properly, and with that to provide a final product of the expected quality, it is necessary that each segment of the system, i.e. each line, should be maximally functional at all times in terms of processing accuracy, i.e. part of the technology for which it is intended. In order to fulfil this requirement, it is necessary that the system during operation has a constant level of guidance quality. This primarily refers to the accuracy of guiding the leading spindles (tool heads) of automatic lines, as well as the guiding of the work table, i.e. preparation. In this research, emphasis was placed on the accuracy of guiding those parts of machine systems that have a threaded spindle in their subassembly. The most common systems that include a threaded spindle are: mobile work tables, robots, manipulators, actuators, etc.

When talking about the accuracy of screw spindle guidance, the working speed is also an unavoidable topic. Modern production systems have a basic requirement, which is the highest possible operating speed. However, the fulfilment of this requirement from the aspect of the threaded spindle extends the requirement not only to the spindle but also to its housing. This is directly related to the thermal stability of the threaded spindle and housing. Higher operating speeds of machine tools cause a higher number of revolutions of the screw spindle, which further causes increased friction, which causes the appearance of higher generated thermal energy. The newly created thermal energy now represents an additional load factor for the spindle and bearing. As long as the thermal load is within the prescribed limits, it does not represent a danger for the machine system, but when this limit is exceeded, significant elastic or plastic deformations of the bearing and spindle occur, which ultimately causes a change in the accuracy of guidance, which is directly reflected in the quality of the final product.

In the case of machine tools, an important place is occupied by the thermal load of the housing. The goal of each bearing is to ensure safe and stable operation of tool heads or screw spindles, which means that in addition to the transmission of radial and axial forces, the transmission of a large number of revolutions is also expected. With regard to the construction of the transmission machine elements, i.e. bearings, increased friction that generates thermal energy is expected, which becomes an additional load factor that must be kept within prescribed limits so that the machine system is operational in an area that does not threaten its thermal stability. Otherwise, the system is exposed to thermal overload, which results in a permanent loss of processing accuracy of the machine system.

For this reason, a lot of scientific papers have been published on the topic of thermal load research on bearings. For practical application, the starting point is the standard DIN 732 part 1 and 2 [1]. Liu et al. in [2] propose an improved spindle modelling model that takes into account the interaction of temperature and thermal deformation using heat source updates and boundary conditions. Xu et al. in [3] using a five-degree-of-freedom model study the nonlinear dynamic response and stability conditions of a spindle designed for high speeds at the bearing and tool location. The subject of research by Zhang et al. in [4] is uneven heat generation conditioned by external load on the thermal characteristics of the bearing. The evaluation of power fluctuation in the contact zone was done by analysis of variance. The conclusion of their work is that a suitable non-uniform preload of the working spindle can affect the reduction of the bearing temperature by max. 15.4%. Dexing et al. in [5] propose a new thermal model for predicting the temperature of the machine tool lead spindle bearing. This model takes into account structural constraints (radial and axial), assembly constraints as well as coolant/lubricant. Finally, using the "muti-node" model, an integrated comprehensive thermal network model was generated for bearing temperature prediction. Than et al. in [6] propose a new algorithm for the analysis of nonlinear thermal characteristics of the lead spindle bearing. This algorithm shows that the preload and bearing stiffness vary nonlinearly with temperature. Wang et al. in [7] analyse a model of an angular contact ball bearing subjected to combined radial, axial loading. In the analysis, the hypothesis of "raceway control" was not used, but instead the control equation of the angle of inclination of the ball was introduced. The model showed that in angular contact ball bearings, especially at a higher number of revolutions, the effect of centrifugal forces is significant. Yan et al. in [8] present a dynamic model of a ball bearing with six degrees of freedom of the balls, three degrees of freedom of the cage and five degrees of freedom of the inner ring. In this model, the influence of cage parameters on air-oil and thermal dissipation was investigated. The parameters of the cage here mean the way of guiding, the gap and the shape of the pocket. The conclusion of this analysis is that at a large number of revolutions, suitable parameters of the cage are extremely important for the proper distribution (dissipation) of heat and air-oil inside the bearing opening. Zhao et al. in [9] provide a simulation of a machining centre spindle system that operates with high precision and high speed. The result of the simulation gives the heat dissipation of the cooling system of the spindle as well as the corresponding thermal deformations. Ma et al. in [10] propose a method of modelling the thermal characteristics of the spindle. This model takes into account thermal contact resistance which is based on morphological characterization and mechanical properties. The paper concluded that the stiffness of the bearing as well as the thermal contact resistance have an influence on the thermal characteristics of the spindle. Krstić et al. in [11] give a presentation of the experimental investigation of the limit number of revolutions of the axial ball bearing for the bearing assembly of the threaded spindle, while in [12] they give a proposal for the thermal analysis of the bearing assembly of the threaded spindle. Also, Krstić and Milčić in [13] show the thermal balance of the bearing, which was performed using axial angular contact ball bearings. Takafumi et al. in [14] carry out a three-dimensional measurement of the movement of the ball and, taking into account the heat generation process as well as the deformation of the balls, propose the construction of an axial angular contact ball bearing. The influence of different preload mechanisms on the stiffness of angular contact ball bearings is investigated by the model proposed by Zhang et al. in [15]. Gheorghiță et al. in [16] provide a summary of research in the field of modelling and optimization of tool head spindle bearing assembly.

The paper presents the results of research into the influence of the spread of heat generated in the ZKLF2575-2Z bearing through the bearing housing. The ZKLF 2575-2Z bearing is marketed by the German manufacturer Schaeffler under the brand name "INA". This bearing is included in the group of axial angular contact ball bearings intended for a bearing assembly of the threaded spindles. As it is an unconventional bearing construction, a new approach in thermal analysis was developed to solve this problem. By construction, it specifically means the type of housing material and the dimensions of the housing itself. The whole analysis carried out for the case of "axial mounting" of the bearing ZKLF2575-2Z.

2 THERMAL LOAD OF THE THREADED BEARING ASSEMBLY

The parametric model of the threaded spindle bearing ZKLF 2575-2Z, which is the subject of the research, is given in Fig. 1.

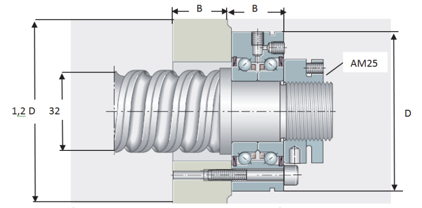

Figure 1 The model of the bearing assembly performed using the bearing ZKLF 2575-2Z

The threaded spindle bearing assembly consists of a bearing (Fig. 2), a precision preload nut (Fig. 3), a direct screw connection that serves to fix the bearing to the machine housing, as well as the machine housing.

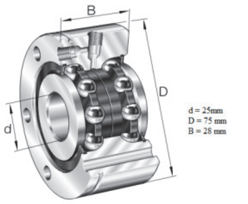

Figure 2 View of bearing ZKLF 2575-2Z [17]

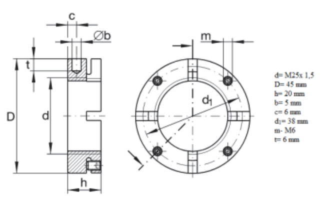

Figure 3 View of AM25 nut for bearing preload [17]

The specificity of the ZKLF type bearing series is, first of all, the two-part inner cage, the larger angle of contact between the balls and the rolling path, which is 60°, the precise nut through which the bearing is additionally tightened and pre-stressed, as well as the screw connection. All these additional specificities require a new approach in the thermal analysis of the bearing system. The two-part cage ensures that the bearing is additionally pre-stressed, which increases the stiffness of the bearing during operation. The extremely large angle of contact between the balls and the rolling path increases the possibility of compensation and transmission of larger axial forces, which favours the loading of the bearing itself, which is predominantly loaded with axial forces that are largely generated by the associated recirculation nut of the spindle, and the rest is due to the forces that arose as a result resistance during workpiece processing. In order to ensure the highest possible accuracy of spindle guidance, it is an increased class of bearing manufacturing accuracy (class P4).

As a result of friction in the bearing, heat is generated, which is determined according to Eq. (1):

$$N_{Fr} = \omega \cdot M = 10^{-3} \left[\frac{\pi \cdot n}{30} \right] \cdot M, \text{ W} \tag{1}$$

n - Number of revolutions, min^{-1}.
M - The total friction moment in the bearing, Nmm.

The total friction moment in the bearing consists of two moments: the friction moment within the lubricant and the friction moment depending on the load. It can be described by Eq. (2).

$$M = M_0 + M_1, \text{ Nmm} \tag{2}$$

M_o - Friction moment inside the lubricant, Nmm.
M_1 - Load-dependent friction torque, Nmm.

The friction moment inside the lubricant is described by Eq. (3).

$$M_0 = f_0 \cdot 10^{-7} \cdot (v \cdot n)^{\frac{2}{3}} d_m^3, \text{ Nmm} \tag{3}$$

f_0 - coefficient that takes into account the type of bearing and the method of lubrication (table in [20]).
v - operating viscosity of oil or base oil for grease (diagram in [20]), mm²/s.
n - number of revolutions, min^{-1}.
$d_m = \dfrac{d+D}{2}$ - mean bearing diameter, mm (according to [18]).

The load-dependent friction torque has the mathematical form shown in Eq. (4).

$$M_1 = f_1 \cdot P_1 \cdot d_m, \text{ Nmm} \tag{4}$$

f_1 - loading coefficient, which takes into account the size of the load (table in [20])
P_1 - relevant bearing load (table in [20]), N
$d_m = \dfrac{d+D}{2}$ - mean bearing diameter, mm (according to [18]).

Substituting Eq. (2) to Eq. (4) into Eq. (1) gives the final equation for the total thermal load of the bearing due to friction in the bearing:

$$N_{Fr} = 10^{-3} \left[\frac{\pi \cdot n_{\vartheta r}}{30} \right] \cdot \left[10^{-7} f_{0r} \left(v_r n_{\vartheta r} \right)^{\frac{2}{3}} d_m^3 + f_{1r} P_{1r} d_m \right], \text{W} \tag{5}$$

The subscript "r" in Eq. (5) indicates that the equation is valid for the reference conditions specified in [1].

3 THERMAL ANALYSIS OF THREADED SPINDLBEARING ASSEMBLY

The thermal analysis of the bearing assembly of the threaded spindle is carried out by an iterative procedure in order to determine the number of revolutions of the threaded spindle for which the basic reference condition from [1] will be fulfilled, which is the temperature of the outer ring of the bearing is 70 °C, which is also the temperature at which the thermal balance is established. When the heat balance of the threaded spindle bearing assembly is established, it means that the amount of heat generated due to friction in the bearing is equal to the amount of heat removed from the bearing. Thermal analysis was done in the CAE software ABAQUS, version 6.9-3.

For the sake of easier explanation, the methodology of thermal analysis of the bearing assembly of the threaded spindle realized by the bearing ZKLF 2575-2Z is shown by the algorithm (Fig. 4).

A thermally symmetric model composed of DCAX3 elements was used to model the bearing assembly.

A: Pre-processing includes:

- Modelling of the bearing system ZKLF 2575-2Z.
- Definition of bearing housing material properties (thermal conductivity coefficients) for: steel, aluminium, gray-cast iron (EN-GJL250) and mineral cast iron.
- Determination of the number of revolutions of the threaded spindle $n_{\vartheta r}$ for the first iteration of the thermal analysis. It was adopted that the number of revolutions of the screw spindle for the first iteration is equal to half of the limit number of revolutions $n_{\vartheta r} = n_G/2 = 2350$ min^{-1} [18].
- Determination of the amount of heat generated in the bearing due to friction according to Eq. (5).
- Assessment of the distribution of the total load from friction (heat flux) in the bearing (empirical; ¾ heat energy drain via the outer ring and ¼ via the inner ring). Here, the values for the total friction load were taken for each individual cycle separately, and each cycle was initially determined by the number of revolutions $n_{\vartheta r}$.
- Determination of the specific heat flux corresponding to the inner and outer rolling paths for each cycle separately.
- Determination of the total heat transfer coefficients for the material and contact surfaces. In the pre-processing part, special emphasis is placed on the present heat transfer mechanisms. In this connection, in the analysis for the emissivity coefficient of dark surfaces, $\varepsilon = 0.8$ was taken, while the heat transfer coefficients were obtained from the Nusselt number. According to the recommendation in [19], the contact surfaces are described by the model for air (FE-Spec. 0009.2).

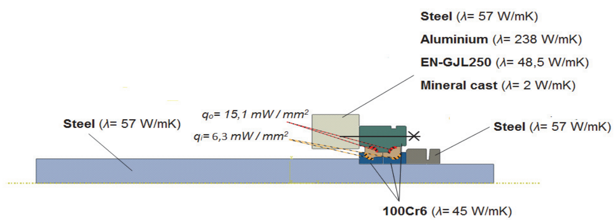

Figure 5 Specific heat flux and thermal conductivity coefficients

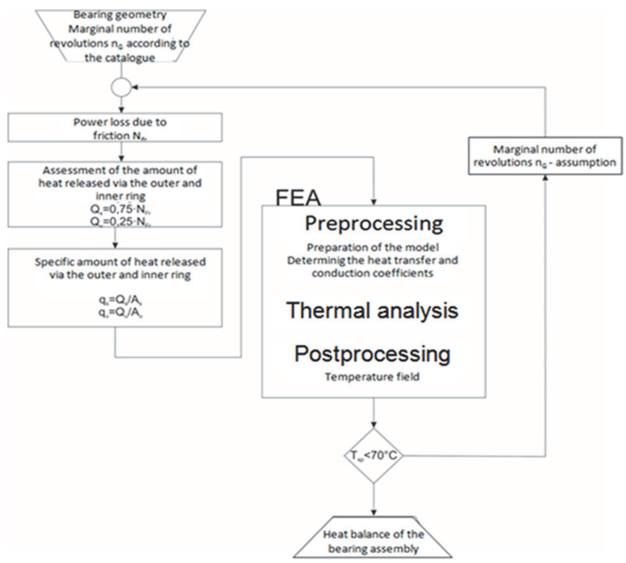

Figure 4 Algorithm of thermal analysis of the bearing assembly of the threaded spindle with bearing ZKLF 2575-2Z [12]

The next task is to determine the heat transfer coefficient for the contact surfaces. This is a very sensitive part of the analysis, considering that modelling the contact surface in simulation is very difficult. An elastic model with air as the medium (FE-Spec. 0009.2) [19] was used as the most favourable for modelling contact surfaces. The obtained heat transfer coefficients for the contact surfaces are shown in Fig. 6.

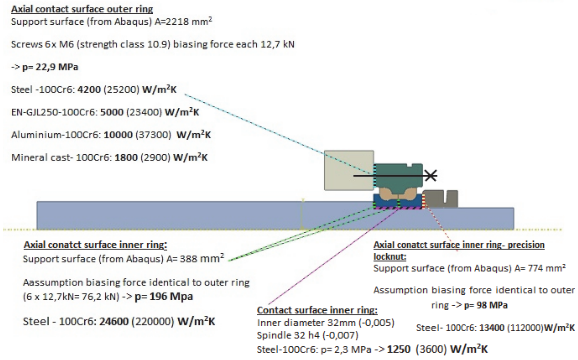

Figure 6 Heat transfer coefficients for contact surfaces

Since the heat transfer coefficients for the contact surfaces have been determined, in the next step the heat transfer coefficients for the other elements of the system (seats) are determined (Fig. 7).

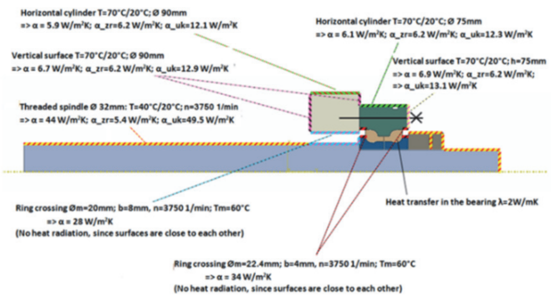

Figure 7 Determined heat transfer coefficients for the bearing assembly elements

B: Solving process.

C: Post-processing.

The result of the thermal analysis is the temperature distribution in the bearing system.

The result of the thermal analysis is the temperature distribution (temperature field) of the bearing assembly system for each variant of the housing made of different materials (steel, aluminium, gray cast iron, mineral cast iron) (Fig. 8 to Fig. 11).

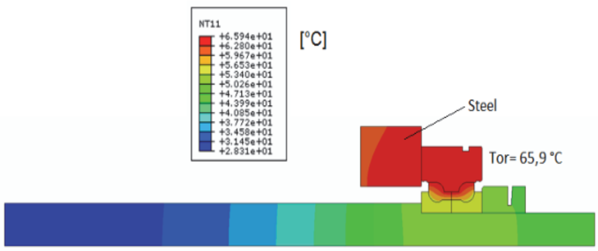

Figure 8 Temperature distribution for the model with steel housing

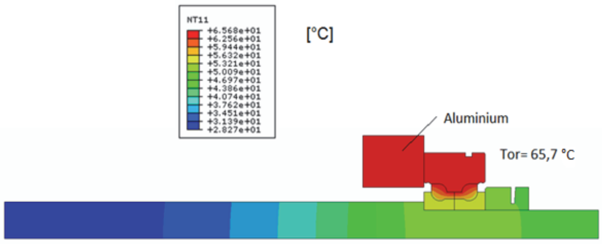

Figure 9 Temperature distribution for the model with aluminum housing

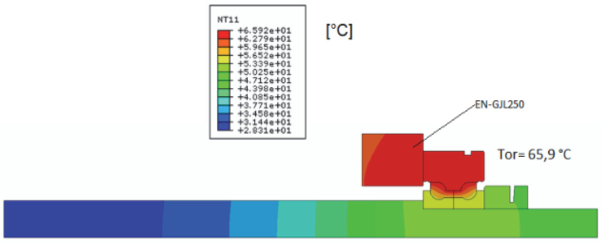

Figure 10 Temperature distribution for the model with EN-GJL250 gray cast iron housing

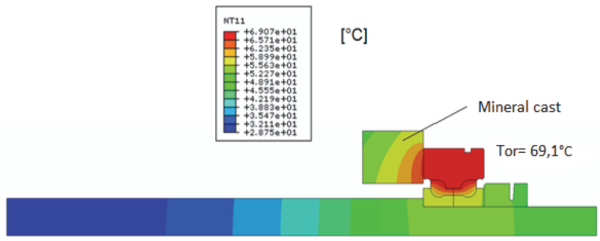

Figure 11 Temperature distribution for the model with mineral cast housing

It is of interest to investigate whether the changed dimensions of the housing affect the temperature field and to what extent. That is why an additional simulation was done, which took into account a twice-longer housing (Fig. 12). This simulation was done only for the steel housing.

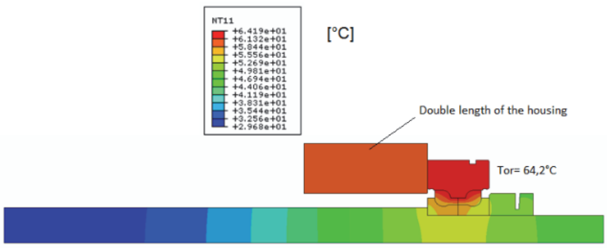

Figure 12 Temperature distribution for a model with twice the length of the steel housing

4 DISCUSSION OF RESULTS

The goal of this research is to obtain certain guidelines that will help designers in the design phase to assess and optimize the thermal load of the threaded spindle bearing assembly. The guideline primarily refers to the influence of the construction of the machine housing on the temperature field of the bearing. In the physical interpretation of the guidelines, it answers the question of whether the choice of a certain material from which the housing will be made can be influenced, i.e. whether it is possible to optimize the temperature field, which would achieve better thermal stability of the bearing and thereby ensure the accuracy of guiding the threaded spindle, which is a very important condition when designing the machine system.

For this reason, the analysis presented in this paper was carried out. The analysis is based on the thermal simulation of the threaded spindle bearing model, which was performed using the ZKLF 2575-2Z bearing. In the first part of the research, an iterative procedure was carried out on a thermal model with an axial angular contact ball bearing, the construction of which is simpler (ZKLN2557-2Z) with the aim of easier determination of the number of revolutions and the total load from friction that correspond to the reference condition from [1], which is the temperature of the outer ring of 70 °C. This is the so-called balance temperature at which heat balance can be applied to the bearing assembly system. When the system is in a state of thermal balance, the influence of the construction on the temperature field can be seen (here it means the type of material and the size of the housing), which is the goal of this analysis. The iterative procedure was carried out on a model with a simpler construction, due to the fact that it is structurally and dimensionally very similar to the ZKLF 2575-2Z bearing (here it primarily means the geometry of the bearing, which is important for thermal analysis identical geometry in terms of the surfaces of the outer and inner rolling paths - identical specific flux) and at the same time it is a simpler deposition model, so such a model is more suitable for the implementation and control of the iterative procedure. Since the geometry of the rolling paths is identical for both bearing models, the application of the results obtained under the reference conditions from [1] from one to the other model is fully justified.

The output data of the first part of the research, the total frictional load and the number of revolutions, become the input/start data for the second part of the research, which was carried out on the bearing model ZKLF 2575-2Z. Since the input data were obtained under the reference conditions from [1] and correspond to the state of thermal balance, the model was prepared for determining the required thermal quantities that will correspond to the ZKLF 2575-2Z bearing. With that, the new model is completely thermally defined and ready for post-processing, which resulted in a spectrum of thermal fields for infectious variants of the model. A schematic representation of the spectrum of temperature fields is given in Fig. 13.

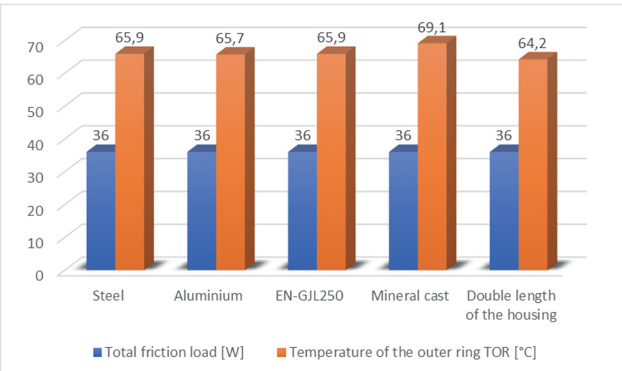

Figure 13 Spectrum of temperature fields

The interpretation of the spectrum is presented in the conclusion of this research.

5 CONCLUSION

Fig. 13 shows the spectrum of temperature fields, which is given depending on the total friction load, the type of machine housing material, and the size of the housing.

The first part of the spectrum refers to the type of housing material. The total temperature difference of the outer ring is approximately 3 °C, which indicates that the type of housing material does not significantly affect the temperature field of the housing, so it can be concluded that the type of housing material is not a relevant influencing factor that could be used to optimize the thermal load of the threaded spindle bearing assembly. It can be seen that the temperature of the outer ring is lower than the reference 70 °C, but it can be seen in pictures (8 - 10) that the temperature distribution through the housing is unfavourable, which indicates the fact that the machine can be additionally heated by the housing. In the case of mineral cast housing, two facts can be observed. The first is that the temperature of the outer ring is approximately 70 °C, which means that the bearing will have an optimal operating mode in terms of thermal stability because it will always be in a state of temperature balance. Another fact that can be observed is the significantly more favourable temperature distribution through the housing. This gives the conclusion that this type of housing is favourable for reducing the thermal load of the machine due to a more favourable temperature distribution.

The last column in the spectrum in Fig. 13 describes the effect of housing size on the temperature field. Only the steel housing is considered here. The conclusion is that housing size has no significant influence on the temperature field of the bearing assembly. The temperature is about 6 °C lower than the reference one, but this is due to the changed size of the housing (longer housing), which affects the increase of the reference surface over which heat is transferred. It can also be seen in Fig. 12 that the temperature distribution is unfavourable, which affects the additional heat load of the machine.

The conducted research as well as the research in [21] give the conclusion that the type of material does not have a significant influence on the temperature field of the bearing, and for that reason it is not a relevant parameter for thermal load optimization. Finally, the conclusion is that the reference surface is a relevant parameter for thermal load optimization. By increasing the reference surface, the thermal image of the bearing is improved. The reference surface here means the total surface in the system through which the active heat transfer from the bearing assembly will be realized.

Acknowledgements

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