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A THERMODYNAMICAL APPROACH TO THE CONDUCTIVE COMPOSITE SLIDE BEARING

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ABSTRACT

Thermodynamic of conductive composite materials is very complex, to calculate each parameter we need to have good knowledge of the necessary methods, because there is a lot of parameters that we need to consider. In this paper we will first explain the basic principle of thermal load on the composite bearing, than calculations of the heat flow for the sliding bearing / sleeve system and of the sleeve temperature. The lifetime of a composite bearing depends also on the sleeve heat flow and the sleeve thermal expansion but also on the thermal expansion of the outer housing of the sliding bearing. Therefore, in this paper we will present and explain step by step all the important thermodynamic parameters to ensure good tribological understanding of the bearing / sleeve system. In various papers we can see that mechanical properties of composite materials significantly depend on the working temperature when it comes to avoiding bearing failure, therefore it is good to calculate the maximum working temperature. The purpose of this paper is to explain the key parameters of avoiding the damage of the composite slide bearing by overheating.

Keywords: *Composite slide bearing, bearing / sleeve system, conductive composite, thermal load.*

1. INTRODUCTION TO THE THERMAL LOAD ON THE BEARING

The thermal load can be defined as the increase in bearing temperature resulting from the conversion of mechanical friction energy into heat due to the relative motion between the sliding bearing and the antibody under the action of mechanical loading.

Studies show that 84% to 91% of the generated friction energy is converted into heat. The rest of the friction energy is used for the fracture process, deformation of the material and transformation of the structure of the material [1]. In figure 1, we can see the area of contact in the sliding bearing / sleeve system and the area of heat generation.

Thermal overload can result in the melting of the bearing material in the case of plastomers or the separation of the matrix from the reinforcement, in the case of duromer composites. Another characteristic of both materials is the excessive deformation of the material, which leads to an increase in the contact surface area between the sliding bearing and the sleeve, thus causing excessive heat generation. Ultimately, this results in a rapid buckling of the bearing material [2].

Composite bearing materials that operate in dry working conditions without additional lubrication by conventional methods like oil or grease lubrication, have a relatively high friction factor, so that at high speed, high temperature can be generated, which can be greater than the allowed operating temperatures of the bearing material. Therefore, the bearing may overheat, which can lead to a functional failure of the sliding bearing [3]. Bearing materials designed for dry working conditions in most cases they are designed to operate at high loads at very low circumferential speeds [4], [5].

The calculation of the temperature at the area of contact of the sliding bearing and the sleeve can be obtained through the heat flow rate released from friction losses, i.e. from the force required to overcome friction, as we can see in equation (1), (2).

$$P_r = \dot{Q} \tag{1}$$

$$\mathrm{P}_r = F_T \cdot v_r = M_T \cdot \omega_r = \mu \cdot p \cdot r \cdot B \cdot D \tag{2}$$

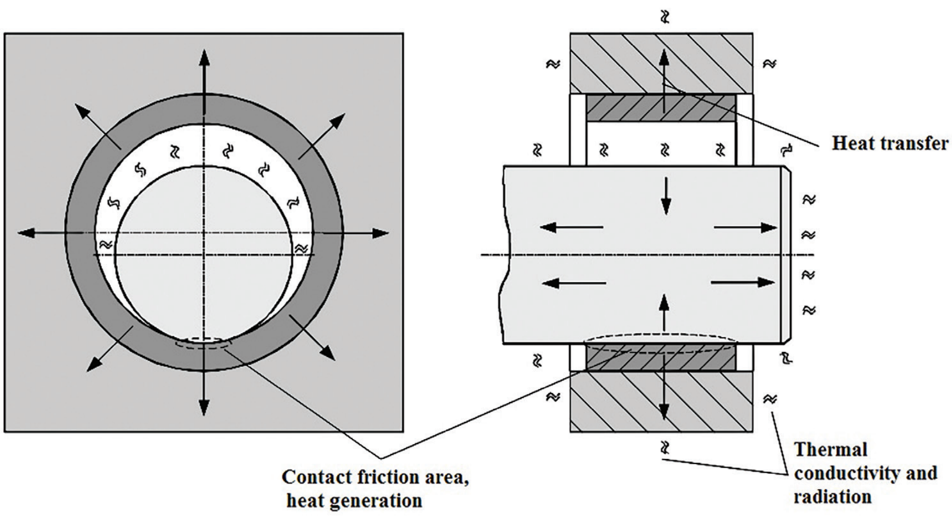

Figure 1 *Heat generation in system bearing /sleeve*

This calculation method according to equation (1) is suitable for orientation calculation, but in order to accurately calculate the heat flow rate, the friction factor needs to be determined experimentally.

It is more complicated to determine the heat dissipation according to equation (3).

$$\dot{Q}_{uk} = \dot{Q}_p + \dot{Q}_k + \dot{Q}_z \quad (3)$$

Heat is transferred to the sliding bearing / sleeve system in three different ways, namely:

• Conduction (*p*)

• Convection (*k*)

• Radiation (*z*)

The transfer of heat flow from the system to the environment is crucial for design of the sliding bearing geometry as well as for the proper selection of the bearing material. Equations (4) to (6) are the basic expressions for heat transfer.

$$\dot{Q}_p = A \cdot \frac{\lambda}{\delta}\left(\vartheta_1 - \vartheta_2\right) \quad (4)$$

$$\dot{Q}_k = A \cdot \alpha \cdot \left(\vartheta_1 - \vartheta_2\right) \quad (5)$$

$$\dot{Q}_z = C_{12} \cdot A \cdot \left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right] \quad (6)$$

There are some disadvantages of the sliding bearing / sleeve system, which are [6]:

• The exact determination of the heat transfer is very difficult to determine without experimental studies

• Due to the unbalanced heat conduction, the temperature gradient depends on the heat direction

• It is very difficult to determine the exact values of the contact surface area between the bearing and the surrounding bearing surface, just by determining the temperature gradient in the contact area [7].

• The thermal conductivity coefficient λ for composite materials depends on many parameters that affect its value. Most often, the coefficient of thermal conductivity is determined experimentally.

It is very difficult to describe a mathematical description of the heat flow for the sliding bearing / sleeve system by conventional methods, but including an experimental part in the mathematical description significantly increases the cost of research, which is difficult to do for practical real systems [8], [6]. Therefore, a simplification of the mathematical formula for the calculation of heat flow is used. The heat flow rate in the sliding bearing / sleeve system depends on their structural characteristics and the heat transfer coefficient *k*.

Equation (7) can be used to describe the heat flow rate of the sliding bearing / sleeve system using the heat transfer coefficient.

$$\dot{Q} = k \cdot A \cdot \Delta T \tag{7}$$

$$k = \frac{1}{\frac{1}{\alpha_a} + \frac{\delta}{\lambda} + \frac{1}{\alpha_b}} \tag{8}$$

Equation (8) describes the heat transfer coefficient that is equal to the reciprocal of the sum of the specific thermal resistances. It should be noted that the magnitude $1/\alpha$ represents the specific thermal resistance of the heat transfer through the boundary layer, and δ/λ indicates the specific thermal resistance of the heat conduction through the wall [9].

According to research, the heat transfer coefficient for the polymeric bearing / sleeve system is from 1 W/m^2K to 10 W/m^2K depending on the proportion of metal-based alloying elements, while for the composite bearing / sleeve system without the addition of metal-based alloying elements or carbon black, it ranges from 0.9 W/m^2K to 1.5 W/m^2K [2].

Increase of the friction factor results in an increase of the bearing temperature ΔT, which can be expressed by the equation (9).

$$\Delta T = \frac{\mu \cdot p \cdot v \cdot B \cdot D}{k \cdot A} \tag{9}$$

2. THERMODYNAMIC CALCULATION OF THE HEAT FLOW FOR THE SLIDING BEARING / SLEEVE SYSTEM

In order to calculate the required clearance in the tribological system of the sliding bearing / sleeve, it is necessary to set up a thermodynamic model of the system from the parts in contact, and to determine the temperatures of the individual parts. Then we can assess the influence on the change of their geometric sizes, which we can see in figure 2.

Simplifications and assumptions:

- The coefficient of thermal expansion is constant.
- The sliding speed, ambient temperature, bearing force and friction factor do not change.
- Stationary heat transfer is required.
- In general, the thermal conductivity coefficient λ is temperature dependent, $\lambda = \lambda(T)$, and in the case of composite materials (eg, duromer composite) and depends on the direction of the fibers.
- Radiation and convection are not considered within the sliding bearing / sleeve system.
- According to Detter [10], the sleeve is uniformly heated in contact with the sliding bearing when the radial force is applied to the bearing.
- In the axial direction there is a fall in the temperature of the sleeve, the farther we are from the sliding bearing the lower will be the temperature.
- The temperature difference between the sliding bearing and the outer sliding bearing housing will be measured during testing in our experiment.
- The temperature of the sleeve will be measured near the contact with the sliding bearing.

In order to determine the required clearance of the sliding bearing, it is necessary to determine the relative clearance ψ_E before installing the bearing in the housing, as well as the relative clearance that occurs when operating the bearing marked with the mark ψ_W.

In order to determine the above relative radiations, the following material characteristics must be known.

- Heat thermal diffusivity α_1 to α_3
- Thermal conductivity coefficient λ_1 to λ_3
- Air heat thermal diffusivity α_{Air}
- Ambient temperature T_O

- Heat transfer coefficient k
- Width of sliding bearing B
- Friction factor μ
- Normal force on bearing F_N
- Sliding speed v

Slide bearing diameter D_a, the inside diameter DK and the outer diameter of the bearing housing DK_v as we can see in figure 2.

The relative installation clearance ψ_E is determined in an iterative manner, starting from the outer diameter of the shaft d and the inner diameter of the sliding bearing D in cold state before operation and comparing it to the released heat flow during the operation of the bearing, which causes a decrease in the clearance.

The selected value of relative installation clearance ψ_E must be sufficiently large so that the sliding bearing can continuously operate in the most unfavorable operating conditions, i.e. the clearance is reduced to the minimum size allowed due to the influence of one or more operating parameters.

3. CALCULATION OF THE SLEEVE TEMPERATURE

The temperature of the T_W sleeve can be calculated using the equation (10) for which we need to know the following values: heat transfer coefficient k, surface area A, friction factor μ, normal force F_N, circumferential velocity v, and ambient temperature T_O.

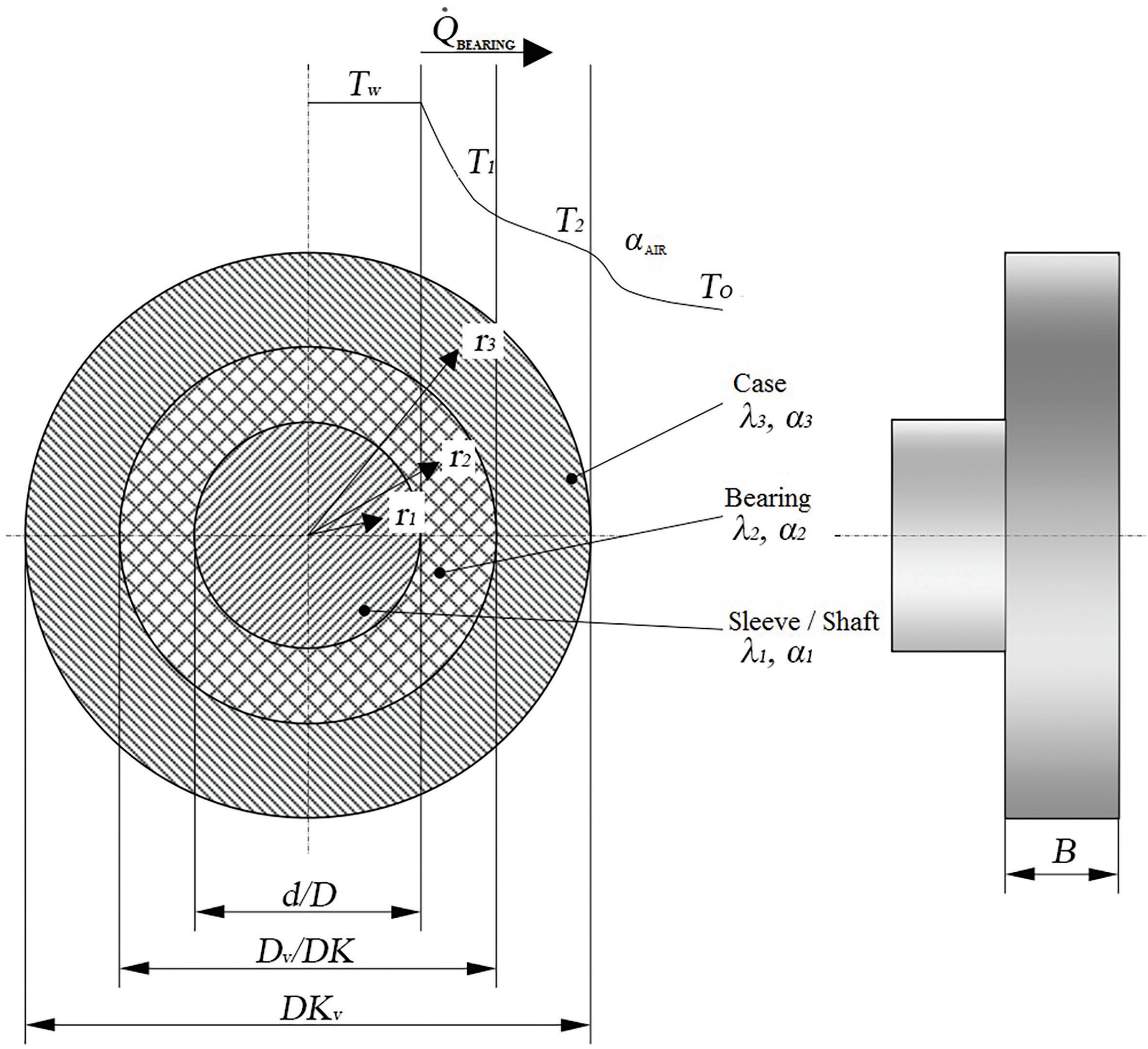

Figure 2 *Thermodynamic model of the sliding bearing / sleeve system*

$$T_W = \frac{\mu \cdot F_N \cdot v}{k \cdot A} + T_O \quad (10)$$

The friction factor for the case of the sliding bearing / sleeve is mostly obtained experimentally with the device for testing the sliding bearings.

A review of the literature [10] notes a number of different studies regarding the determination of the friction factor of a durometric composite in contact with a steel sleeve, depending on the sliding speed, on the load and without lubrication. The friction factor ranges from 0.18 to 0.32.

The heat transfer coefficient and the contact surface of the two bodies describe the thermal state of the sliding bearing / sleeve system. According to the literature [12], [13] studies have shown that for a polymeric sliding bearing / sleeve system the amount of heat transfer coefficient multiplied by a given contact area ranges from 1 W / K to 10 W / K, depending on the type of bearing material. In cases where carbon fibers are present in the composite, the thermal conductivity coefficient increases, while it decreases in the case of glass fiber composites, for which the thermal conductivity coefficient is lower.

Studies [12], [14] have showed that for duromer composites, due to their anisotropic property, depending on the direction in which the heat impacts the fibers, we will have a different heat conductivity coefficient λ as we can see in fig. 3.

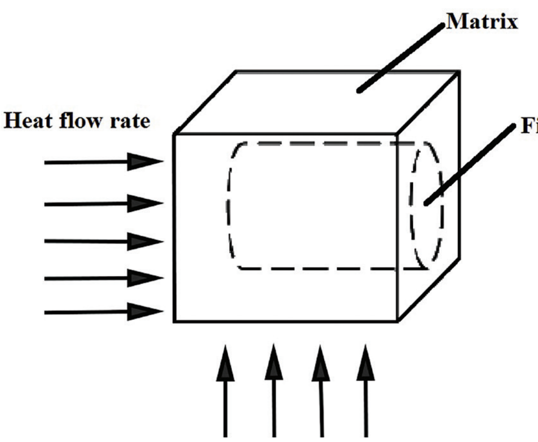

***Figure 3** Effect of heat flow on a fiber of composite material*

The theoretical coefficient of thermal conductivity from composite depends on the arrangement of fibers and reinforcers. For evenly spaced fibers within an amplifier, as in the case of the test composite material under study, we can observe two cases of thermal conductivity coefficient.
In the case where the heat flow rate acts in the parallel direction of the composite fibers than we have a parallel coefficient of conductivity $\lambda_{\perp}$ of heat which can be seen in fig. 4 under a), and in the case of perpendicular heat flow rate $\lambda_{\perp}$ to the fibers there is a vertical coefficient of conductivity of heat as can be seen in fig. 4 under b)

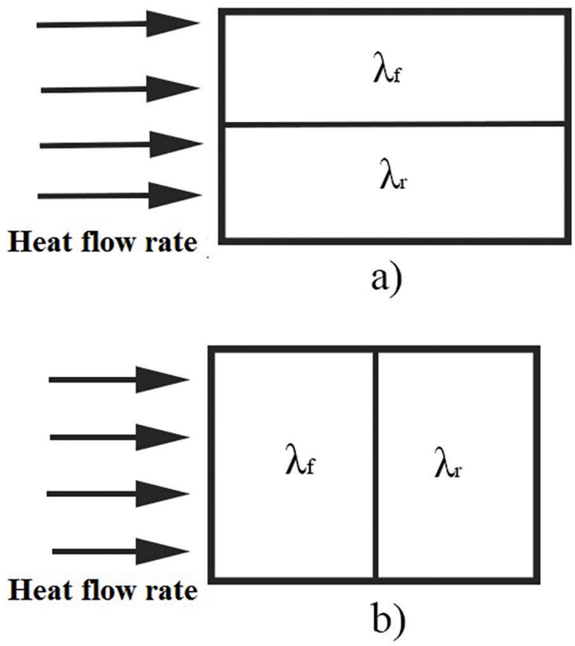

***Figure 4** Parallel and vertical effect of heat flow on the matrix and composite reinforcement*

We can calculate the parallel thermal conductivity coefficient λ_{II} using the equation (11).

$$\lambda_{II} = V_f \cdot \lambda_f + (1 - V_f) \cdot \lambda_r \quad (11)$$

The vertical conductivity coefficient $\lambda_{\perp}$ can be calculated using the equation (12).

$$\lambda_{\perp} = \frac{1}{\frac{V_f}{\lambda_f} + \frac{1 - V_f}{\lambda_r}} \quad (12)$$

Equations (11) and (12) denote λ_r of the conductivity coefficient of the matrix, i.e. fiber composite, for glass fibers of 1.2 W/m K, while the symbol λ_r indicates the conductivity coefficient of the epoxy reinforcer, 0.27 W/m K, and the symbol $\lambda_\perp$ denotes a fiber volume fraction of approximately 0,556 [12].

For example tested durometer composite in this paper was a Norden marine 605 bearing material, the vertical thermal conductivity coefficient $\lambda_\perp$ is set at 0.296 W/m K.

The thickness of the bearing wall is of great importance for the heat transfer to the bearing housing. The greater the thickness of the bearing wall, the smaller the heat transfer from the bearing to the housing, all of which can cause a difficult cooling of the sliding bearing.

4. CALCULATION OF THE SLEEVE HEAT FLOW AND THE SLEEVE THERMAL EXPANSION

According to standard R310DE 2950 [15] the heat flow of the sliding bearing drawn through the steel shaft can be described by the equation (13)

$$\dot{Q}_{SLEEVE} = K_1 \cdot \frac{\pi \cdot B \cdot d}{z} \cdot \lambda_\perp \cdot \Delta T + K_2 \cdot \frac{2\pi \cdot d^2}{4 \cdot d} \cdot \lambda_R \cdot \Delta T \qquad (13)$$

The dimensionless factors K1 and K2 describe the extent to which the heat flow rate is transferred from the sliding bearing to the shaft depending on the bearing material and bearing geometry, and according to R310DE 2950 [15] for polymer composites they amount is K1 = 0.5 and K2 = 0.0416.

The thermal conductivity coefficient for the duromer composite bearing material is d_w=0.296 W/m K, while for the stainless steel sleeve AISI 316 it is d_w=17 W/m K.

According to the literature [15] the thermal expansion of the sleeve, outside diameter d_w of the sleeve, can be described by the equation (14).

$$d_w = \alpha_1 \cdot (T_W - T_O) \cdot d_p + d_p \qquad (14)$$

In the equation (14), description α_1 indicates the coefficient of elongation of the material of stainless steel, from which the sleeve is made.

The operating temperature of the sleeve is indicated by T_W, the ambient temperature d_p, and the initial diameter of the sleeve is indicated by d_p.

5. THERMODYNAMIC CALCULATION OF HEAT FLOW FROM SLIDING BEARING

The total friction force is described by the equation (15).

$$P_R = \mu \cdot F_N \cdot v \qquad (15)$$

In order to calculate the heat flow of the sliding bearing $\dot{Q}_{BEARING}$, it is first necessary to calculate the total thermal resistance of the sliding bearing of the duromer composite. Thermal resistance $R_{k,sum}$ describes the resistance to the heat transfer and is opposite to the coefficient of the heat transfer *k*.

Depending on the thickness of the wall (δ), the surface of the wall (A), and the characteristics of the material, of which the heat transfer coefficient (λ_k) is the most important in this case, we can describe the equation (16) with the thermal resistance R_k [1]

$$R_k = \frac{\delta}{\lambda_k \cdot A} \qquad (16)$$

As we can see in the equation (14), the thermal resistance increases with the thickness of the wall and decreases with the increase of the wall surface and the increase of the heat transfer coefficient.

The difference in temperature between the inner and outer walls (ΔT) can be described as a product of thermal resistance R_k as we can see in Equation (17).

$$\Delta T = R_k \cdot \dot{Q}_{BEARING} \qquad (17)$$

Analog to Ohm's law in electrical engineering described by the equation (18), the equation (17) is also known as Ohm's law of heat conduction.

This can be explained as a change in temperature, in our case the temperature drop is equal to the voltage, the thermal resistance is equal to the electrical resistance, while the heat flow rate is equal to the current strength [1].

$$U = R \cdot I \tag{18}$$

Depending on the number of walls that are in contact with each other, the thermal resistance can be summed up as a parallel connection of resistors as in electrical engineering [1] which is described by the equation (19).

$$\frac{1}{R_{k,sum}} = \frac{1}{R_{\lambda 1}} + \frac{1}{R_{\lambda 2}} + \frac{1}{R_{\lambda 3}} + \cdots + \frac{1}{R_{\lambda n}} \tag{19}$$

By formulating Ohm's law for the tube wall, thermal resistance [1] can be described by the equation (20).

$$R_k = \frac{\ln \frac{D}{D_v}}{2 \cdot \pi \cdot B \cdot \lambda_k} \tag{20}$$

In order to calculate the thermal resistance of the pipe walls for the particular case of the test bed, which can be seen in the equation (21), it is necessary to know the heat transfer coefficient of the composite bearing material (λ_2), the outer steel housing of the bearing (λ_3) and the length of the sliding bearing (B).

$$R_{k,sum} = \frac{\ln \frac{DK_{v,k}}{DK_k}}{2 \cdot \lambda_3 \cdot \pi \cdot B} + \frac{\ln \frac{D_{v,k}}{D_k}}{2 \cdot \lambda_2 \cdot \pi \cdot B} \tag{21}$$

The sliding bearing heat flow rate can be calculated using the equation (22).

$$\dot{Q}_{BEARING} = \frac{(T_W - T_O)}{R_{k,sum}} \tag{22}$$

6. CALCULATION OF THERMAL EXPANSION OF THE OUTER HOUSING OF THE SLIDING BEARING

In order to calculate the thermal expansion of the housing, it is necessary to determine the temperature of the cushion from the inner walls of the housing T_1 to the outer sides of the housing T_2 as can be seen in figure 2.

The temperature T_2, i.e. the temperature with the outer sides of the housing, can be described by the equation (23). It is important to increase the outdoor temperature to about the ambient temperature T_O [9].

$$T_2 = \frac{\dot{Q}_{BEARING}}{\pi \cdot B \cdot \alpha_{Air} \cdot DK_{v,k}} + T_O \tag{23}$$

The temperature of the inside housing T_1, which is in contact with the outer diameter of the sliding bearing, can be calculated using the equation (24).

$$T_1 = \frac{\dot{Q}_{BEARING} \cdot \ln \frac{DK_{v,k}}{DK_k}}{2 \cdot \pi \cdot B \cdot \lambda_3} + T_2 \tag{24}$$

The assumption of the calculation of the test system, i.e. the outer bearing housing, is stationary heat conduction [1] with a sufficiently small error for non-stationary heat conduction. With non-stationary heat conduction, the temperature changes over time, so we do not have a regular form of linear temperature drop but rather a non-linear temperature drop.

7. CONCLUSION

As presented in this paper, the thermodynamic calculation can be very complex. Therefore, it is recommended to calculate all the parameters from step 1 to 6 (Calculation of the thermal expansion of the outer housing of the sliding bearing) to avoid all potential failures. As described, before proceeding with the calculation, it is necessary to precisely gather all the relevant values necessary for calculation.

It is important to understand that on the market we have a lot of different composite materials that have different thermal properties, and lose mechanical properties in high temperature, which can lead to failure of the composite bearing. If we don't know some of the necessary values, it is recommended to make specific thermodynamic experiments to ensure the exact calculation of values. For future research, it would be good to investigate the change of the contact area between the slide bearing and the sleeve at higher temperatures.

8. REFERENCES

[1.] H. Windisch, Thermodynamik Ein Lehrbuch für Ingenieure, München: Oldenbourg Wissenschaftsverlag GmbH, 2008.

[2.] J. P. PASCAULT, H. SAUTEREAU i J. VERDU, THERMOSETTING POLYMERS, New York: Eastern Hemisphere Distribution, 2002.

[3.] M. Gruender , »High-PV Wear Study of Six High Performance Wear Grade Engineering Plastics,« 2012.

[4.] S. S. Trelleborg, Orkot hydro Berings, Bearing design Guide, TSS, 2013.

[5.] Thordon, Hydro-turbine Bearings, Thordon Bearings, 2007.

[6.] P. Stephan, K. Schaber, K. Stephan i F. Mayinger, Thermodynamik, Band 1: Einstoffsysteme Grundlagen und technische Anwendungen, Berlin: Springer-Verlag, 2006.

[7.] A. Miszczak i A. Czaban , »SURFACE TOPOGRAPHY OF SLIDE JOURNAL BEARINGS,« Journal of KONES Powertrain and Transport, svez. 18, pp. 279-285, 2011.

[8.] K. Stierstadt, Thermodynamik Von der Mikrophysik zur Makrophysik, Berlin: Springer-Verlag, 2010.

[9.] D. Labuhn i O. Romberg, Keine Panik vor Thermodynamik!, Wiesbaden: Friedr. Vieweg & Sohn Verlag, 2006.

[10.] H. Detter, »Berechnungshinweise für Gleitlager im Trockenlauf der Gleitpaarung Kunststoff-Stahl,« Schmiertechnik + Tribologie, Sves.. 107-113, 1975.

[11.] L. SHI, Heat Transfer in the Thick Thermoset Composites, Netherlands: by Uitgeverij BOXPress || Proefschriftmaken. nl, 2016.

[12.] T. Owens, »THERMAL MANAGEMENT IN FIBER-REINFORCED COMPOSITE APPLICATIONS,« 2009.

[13.] D. Cugnet, . C. Hauviller, A. Kuijper, V. Parma i . G. Vandoni, »THERMAL CONDUCTIVITY OF STRUCTURAL GLASS/FIBRE EPOXY COMPOSITE AS A FUNCTION OF FIBRE ORIENTATION,« Grenoble, 2002.

[14.] N. N, »Kunsttoffgleitlager berechnung nach norm R310DE 2950,« Bosch Rexroth AG, 2010, pp. 1-19.

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