19th Brazilian Congress of Thermal Sciences and Engineering
November 06th-10th, 2022, Bento Gonçalves - RS - Brazil

ENC-2022-0719 MATHEMATICAL MODEL OF HEAT DISSIPATION IN ROLLING BEARINGS

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Abstract.
This work aims to predict the temperature in the bearing from the thermal modeling of a Rolling bearing. The model is based on the law of conservation of energy in a transient regime, whose equations are based in the thermal resistance network. In this model, the bearing was discretized at 39 nodes and has major detail in the inner region of the bearing. A bearing friction model is used to predict the torque and power dissipated in the bearing in the form of thermal energy. The model results were compared with measured values in a Rolling Bearing Power Loss Test Rig of the Universidade Tecnológica Federal do Paraná. The experimental rig makes it possible to measure temperature, torque, shaft rotation and load on the deep groves rolling bearing as a function of time. Eighteen 6-hour tests were performed on 6310ZZ rolling bearings lubricated with Polyrex EM 103 Mobilith grease and MT33 SKF grease. The system was subjected to rotations between 350rpm and 1800rpm and radial loads up to 1000N. An additional analysis in experimental results about of increase temperature are performed. The Results show as the friction torque model predicts a torque well below the measured values. Using measured torque measurement results as the model boundary condition, a good comparison was observed between calculated and measured temperature values in both transient and steady state. In all tests performed, the maximum difference between the calculated and measured temperatures was below 2.6ºC in steady state.

Keywords: *Temperature Prediction, Rolling Bearings, Thermal resistance model.*

1. INTRODUCTION

In a background where about 45% of the global electricity demand is consumed by electric motors and represents a power consumption of energy greater than 7000 TW.h/year, there is a need for a detailed understanding of the mechanisms of power loss and energy consumption usage. In this way, guaranteeing the best possible performance in the transmission of axis movement is a challenge that represents one of the greatest opportunities for energy savings and reduction of carbon emissions, given its primary role in today's use (Weis et al., 2021).

Rolling bearings are key parts for fixing rotational shafts that in addition to supporting high loads and rotations, also provide additional damping to mechanical vibrations. The interactions of the bearing's internal components (lubricant, rolling element, cages, raceways and seal) cause power loss in the form of heat due to friction (Almeida e Fonseca, 1997). The excess of temperature can reduce the lubricant's viscosity and change the way it flows when in operation, altering the lubrication regime, and can even be boundary lubrication, a situation in which there is no lubricant separating the surfaces. As a result, there is an increase in friction losses, which results in even higher operating temperatures, and consequently a reduction in the useful life of the lubricant and the rolling bearing itself (Liu et al., 2020). These are one of the main causes of effect failures in rotation systems and are evident in bearings operating at high speeds (Sanchez Garrido et al., 2021). Therefore, in order to optimize the efficiency and service life of bearings, research related to friction and temperature prediction is expressly necessary.

As the world advances in search of progressive technological evolution, the improvement and development of techniques in the areas of engineering are increasingly constant. The branch of research involving an estimation of frictional power loss in bearings and a temperature prediction has been increasingly developed.

The Rolling Bearing industry developed the first rolling bearing power loss prediction has made from the mid-1950s with work by Arvid Palmgren, director of technical department at bearing manufacturer SKF from 1937 to 1955, describing a model capable of estimating friction torque in bearings (Palmgren, 1959). SKF currently recommends using a proprietary empirical friction torque model that considers friction losses based on a kinematic analysis of the interaction between lubricant and bearing (Bearing friction, slip and drag), bearing and seal (Seal friction) and is widely used as references in current works related to temperature prediction (Gonçalves et al., 2017; Kanazawa et al., 2020; Krishnan e Nayani, 2021).

With the emergence of the estimation of friction losses in bearings, works related to temperature prediction also appeared, such as of Parker (1984) describing one of the first works carried out to estimate the temperature of oil-lubricated rolling bearings involving the use of numerical methods, through a model represented by 51 nodes and a prediction with assertiveness of 92% accuracy (analysis performed in % in this period) for tests at high temperature ranging between 420K and 480K. In more recent times, it has been observed that the use of numerical methods has been accepted and widely used through the modeling of thermal resistances to represent the rolling bearing (Neurouth et al., 2014), making it possible to consider the thermal capacitance to compare the estimate with the record of temperature increase in the transitory regime in bearings (Takabi, 2015).

The Thermal Resistance Model is a technique that makes it possible to estimate the temperature distribution in a system, through energy balance and the discretization of its parts (Volumes) in thermal resistances. The assembly of this balance in discrete volumes, results in a linear system of equations that allows the separation of the variables related to the thermal resistances and the amount of heat stored in each volume and whose solution is the estimated temperature.

The possibility of estimating internal temperatures in the bearing in a localized way allows the integration of new variables of the estimation of new models of frictional torque, such as the dilation of the parts by the increase in temperature (De-xing et al., 2018) as long as it is satisfied the necessary level of discretization for the modeling of this phenomenon to be adequate (Zheng; Chen, 2020).

In this context, this work aims to estimate the bearing temperature using the thermal resistance model in in the "Rolling Bearing Power Loss Test Rig" (RBPLR) at the Universidade Tecnológica Federal do Paraná (UTFPR). This rig allows testing different sets of bearings and lubricants, in this work the Bearing 6310ZZ with two different greases was subjected to imposition of different loads and rotations. The model elaborated considers the source of heat generation by friction measured in the bearing by a torque cell, heat dissipation by forced convection on the rotating shaft and natural convection along the non-moving part of the bearing and ambient temperature measured by thermocouple. The objective is to verify if the thermal resistance model is suitable for predicting the temperature of bearings by introducing the data obtained from the heat generation source directly into the model, instead of using a theoretical friction torque prediction model. At the end, the results obtained by the prediction using the thermal resistance network and the experimental test using thermocouples were confronted.

2. PROBLEM DESCRIPTION

Rolling bearings are elements that demand high precision in their design as they are made to have good loading capacity, work at high rotations, low friction torque and low vibration. As shown in Figure 1, they consist of four parts: the outer ring, the inner ring, cage and a set of rolling elements (ball or rollers).

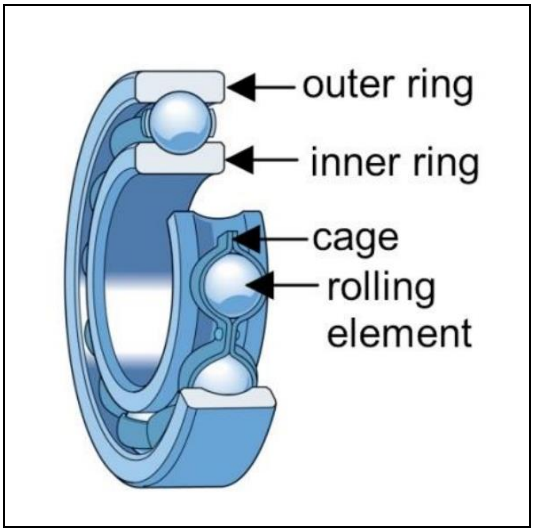


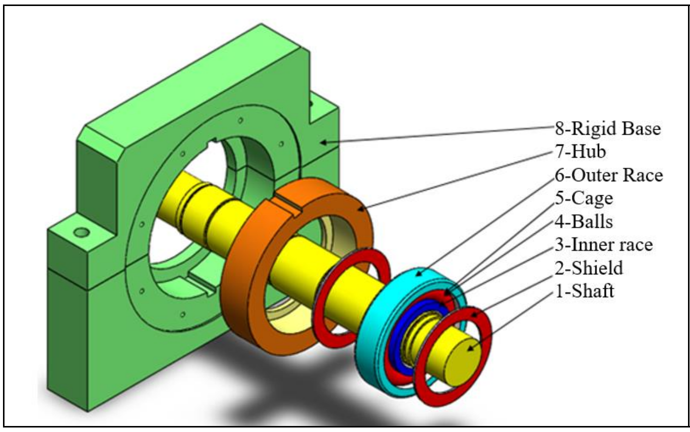
Figure 1. Components of a deep groove ball bearing.
Available from: (Nabhan et al. 2015).

The rings have channels or tracks that serve as a guide to guide the movement of the rolling elements, which are equally spaced from each other by the cage (Carvalho, 2010). Inside, they are lubricated to reduce the friction of moving parts in contact, with oil or grease, being the most common grease in 80% to 90% of bearings because it has properties that make it difficult for lubricant to leak into the environment and prevent external contaminants from entering. inside the bearing (Lugt e Baart, 2012). When the bearing is in operation, heat generation occurs in the region of contact between ball, fluid film and bearing race. The separation of solid surfaces occurs due to the hydrodynamic effect generated in the lubricant due to the movement between the surfaces. This effect prevents surface wear if there is sufficient speed or viscosity for lift to occur and when the deformation of surfaces by a load is considered, this lubrication is called Elastohydrodynamic (EHD) (Campos, 2004).

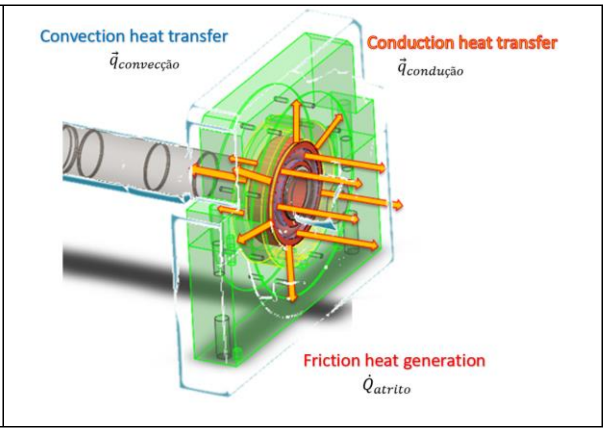
3. THEORETICAL FOUNDATION

The method of temperature prediction using thermal resistances is based on identifying the basic thermal elements as sources of heat and thermal resistances by dividing the volumes where heat conduction occurs and connecting by nodes of similarity of electrical resistances. The elements linked together form a network that makes it possible to calculate temperature differences at the nodes.

The housing bearing basically consists of 8 different parts as shown in Figure 2 (a): Shaft (1-Yellow), Shield (2-Red), Inner race (3-Dark Blue), Balls (4-Red), Cage (5–Red), Outer Race (6-Light Blue), Hub (7–Orange), Rigid Base (8–Green). In the analysis of the heat exchange of the bearing, it is verified that the heat generated by friction in the contact between the bearing race and the ball propagates along the entire bearing and dissipates along its surface, as shown in Figure 2 (b).



(a)



(b)

Figure 2. (a) Components of a housing bearing. (b) analysis of heat transfer in a housing bearing.

3.1 Heat Generation by Friction

The SKF friction torque model (2003) is a model that establishes that the friction torque in the bearing "M_{SKF}" is the result of the sum of four corresponding parts that can be calculated: "M_{rr}" refers to losses due to resistance to the rotation of the element rolling element (in N.mm), "M_{Sl}" referring to the friction generated by the sliding of the rolling element (in N.mm), "M_{Seal}" related to the sealing friction of the seal that keeps the lubricant in the bearing (in N.mm) and "M_{drag}" represents the drag friction generated by the rolling elements and cage as they pass through the oil bath (in N.mm). In this work only shielded grease lubricated bearings were tested, in this case according to SKF the last two terms are null. Thus, the friction torque on the bearing in this work is defined by Equation (1):

$$M_{SKF} = M_{rr} + M_{Sl} \tag{1}$$

The power loss in the shaft is related through the product of the friction torque and the angular velocity, represented by Equation (2).

$$Q_{atrito} = \omega . M \tag{2}$$

where "Q_{atrito}" represents the power loss due to friction torque in the bearing, "ω" represents the angular speed of rotation of the shaft and "M" represents the torque due to all friction phenomena in the bearing.

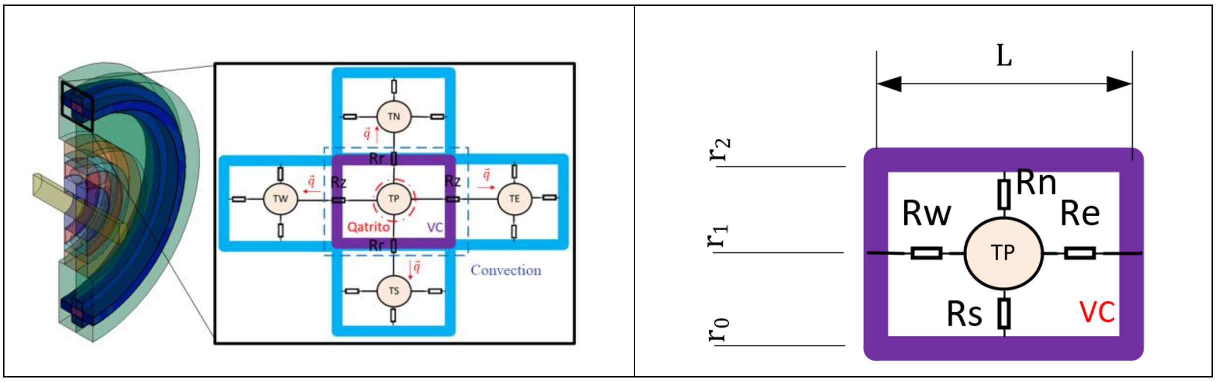
The portions of heat generation related to ball-to-race contact have slightly different amounts, both in ball to outer race and inner race contact. (Li et al., 2018). In order to simplify the process of calculating the division of dissipated heat parcels, an approximation based on the Merritt (1962) through the relation (3) and (4):

$$Q_1 = M_{atrito}.\,\omega.\frac{R_2.}{(R_1+R_2)}. \tag{3}$$
$$Q_2 = M_{atrito}.\,\omega.\frac{R_1.}{(R_1+R_2)}. \tag{4}$$

where "Q_1" represents the power generated by friction in the bearing contact on the inner race, "Q_2" represents the power generated by friction in the bearing contact on the outer race, "M_{atrito}" represents the friction torque generated by the resistance to rotational movement in the Rolling Bearing, "ω" represents the angular speed of rotation of the shaft, "R_1" represents the radius of the inner diameter of the outer race of the bearing and "R_2" represents the radius of the outside diameter of the bearing's inner race.

3.2 Thermal Resistance Model

The geometry of housing bearing is considered to be cylindrical (Figure 3a) and therefore, the conduction thermal resistances in cylindrical coordinates can be determined from Fourier's Law by dividing the bearing into discrete elements (Figure 3b).



(a) (b)

Figure 3. (a) Generic control volume in Housing bearings. (b) definition of thermal resistances for a generic control volume.

The energy balance of the equation is applied to each of these discrete volumes (Equação 5) and it is verified that the heat conduction and convection phenomena can occur both in the $\dot{E}_e$ term of heat flow input and in the term $\dot{E}_s$ in the energy flow output depending on the location of the discrete volume. Since the only heat source is the heat generation by friction $\dot{Q}_{atrito}$ in the rolling bearing, we can rewrite equation (5) through expression (6).

$$\frac{\partial E}{\partial t} = \dot{E}_g + \dot{E}_e - \dot{E}_s \tag{5}$$

$$\frac{\partial E}{\partial t} = \dot{Q}_{atrito} + [q_{conduction} + q_{convection}]_e - [q_{conduction} + q_{convection}]_s \tag{6}$$

The rate of energy transferred between adjacent control volumes is defined by the thermal resistances corresponding to the direction of heat flow where heat exchange can occur. The discrete control volumes will be defined by the inner radius "r_0", average radius "r_1", outer radius "r_2" and bearing race length "L", according to the dimensions shown in Figure 3(b). The four conduction resistances defined in Figure 3(b) can be calculated by:

$$R_e = R_w = \frac{2k\pi\left({r_0}^2 - {r_i}^2\right)}{L} \tag{7}$$

$$R_n = \frac{\ln(r_2/r_1)}{2\pi kL} \tag{8}$$

$$R_s = \frac{\ln(r_1/r_0)}{2\pi kL} \tag{9}$$

where "R_e" and "R_w" represent the thermal resistance from the center of the volume to the "East" and "West" sides, "R_n" represents the thermal resistance from the center of the volume to the top "North" and "R_s" represents the thermal resistance from the center of the volume to the bottom "South".

In an analogous way, the thermal resistances referring to the heat exchange of heat by convection in the control volume "R_{conv}" can be formulated from Newton's Law of Cooling (Equation 10):

$$R_{conv,d} = \frac{1}{h.A_d} \tag{10}$$

The heat exchange caused by the convection of the moving shaft and the ambient air increases with the increase in the imposed rotation speed. The model proposed by Becker (1963) for calculating the heat exchange coefficient is accepted in several works (Takabi and Khonsari 2015; Jiang and Lin 2022). The calculation of the average Nusselt number "$\overline{Nu}$" for axes moving with constant velocity is obtained through expression (11).

$$\overline{Nu} = 0{,}133\ {Re_r}^{2/3}.{P_r}^{1/3}\ \text{(para } 800 \leq Re_r \leq 100000\text{)} \tag{11}$$

The heat exchange for the rest of the fixed parts of the bearing occurs only by natural convection. The natural convection coefficient adopted is a constant value of 5W/(m².K), as an approximate initial value and that will be readjusted with greater precision in future works.

3.3 Temperature Calculation

Substituting the calculation of thermal resistances defined by equations (7), (8), (9) and (10) in equation (6) and considering the transient term applied in each, we obtain the energy balance from the formulation of thermal resistances in generalized form for each discrete bearing control volume defined by equation 12:

$$\rho C \forall \frac{dT_P}{dt} - (Q_{atrito}) = \sum \left[\frac{1}{R_d}\right] T_d - \left[\sum \left[\frac{1}{R_d}\right] + \left[\sum \left(\frac{1}{h.A_d} + \frac{1}{R_d} \right) \right] \right] T_P + \sum \left[\left(\frac{1}{h.A_d} + \frac{1}{R_d} \right) \right] T_{\infty_d} \tag{12}$$

where "ρ" represents the density of the material, "C" represents the heat capacity, "∀" represents the control volume where the balance is being applied, "T_P" represents the temperature at the internal node of the control volume, "t" represents the time variable, "T_d" represents the temperature of adjacent control volumes, "T_{∞_d}" represents the temperature of the fluid when there is convection in some direction of the analyzed discrete control volume.

The temperature distribution in each control volume is found through the numerical solution of the differential system established by the equation that is associated with the thermal resistance network. In this model, at least two types of boundary conditions must be considered as a temperature imposed on some node (such as the axis and ambient air) and an imposed heat generation (such as the generation of heat by friction Q_{atrito}). Although the differential system is a two-dimensional problem, the matrix form of this system is characterized by a five-diagonal matrix and variable diagonal band due to unstructured discretization, causing a part of the matrix to have a variable diagonal band. More details on the variable diagonal band thermal resistance matrix can be found in the work by Maliska (1995).

In this way, the differential system at each instant of time reduces the problem to a solution of a system of linear algebraic equations where each term can be separated in the form of expression (13):

$$\{Q\} = [A]\{T\} \tag{13}$$

where "$[A]$" represents a matrix of the inverse of the thermal resistances, "$\{T\}$" represents a vector of temperatures that we want to calculate and $\{Q\}$ represents a vector of heat rate in each control volume.

There are several ways to solve the linear system by finding. The solution method of this system for a nearly pentadiagonal matrix of size NxN recommended by the MATLAB software is the partial pivoting LU matrix decomposition method. More information on this linear solution method can be found in the work of Chapra (2002).

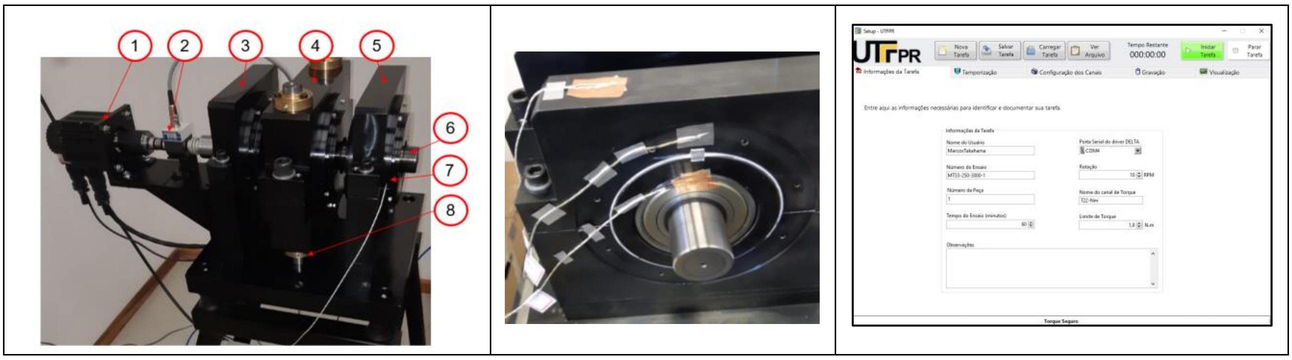
4. MATERIALS AND METHODS

The study related to temperature prediction in grease lubricated bearings was done by through modeling a bearing through a capacitive-resistive thermal circuit implemented in MATLAB and validated through experimental tests on the rig “Rolling Bearing Power Loss Test Rig” (RBPLR) of the Federal Technological University of Paraná (UTFPR).

In this chapter, the “Rolling Bearing Power Loss Test Rig” is presented as described in section 4.1. The thermal resistance model adopted for application in rolling bearings is presented in section 4.2.

4.1 Rolling Bearing Power Loss Test Rig

The “Rolling Bearing Power Loss Test Rig” (RBPLR) test bench shown in Figure 4 (a) is intended to evaluate the performance of bearings and greases under different load and speed conditions. The rig is equipped with a Servomotor (1) responsible for maintaining the rotation on the main axis (6). The rotational movement imposed by the shaft is transmitted to the bearings located in the bearings (3, 4 and 5) and the torque generated by the resistance to movement of these bearings torsionally deform the torque cell (2), and from this deformation the torque measurement is performed. Thermocouples (7) are positioned in the bearing and adjacent regions, as show Figure 4 (b) allow the recording of the temperature evolution in each test. The bearings are radially loaded by means of a screw that pushes the floating center housing upwards. The Load Cell (8) records the applied load.



(a) (b) (c)

Figure 4. (a) Rolling Bearing Power Loss Test Rig. (b) Positioning of the thermocouple for Rolling bearing temperatures and (c) Interface of control on LabVIEW

Temperature data is recorded by type K thermocouple and National Instruments signal acquisition board with module NI-9211. The data from the load transducers HBM C9C and torque T20WN/2Nm are recorded by a National Instruments module NI-9218. The limitations of the tests on this bench are associated with the limitations of the torque and force transducers, and the servomotor. Thus, the maximum measurable torque is 2N.m, the maximum applicable load is 10 kN and the maximum rotation speed is 3000 rpm.

An algorithm developed in a LabVIEW environment, according to the interface shown in Figure 4 (c), controls the speed of the servomotor and also records the data obtained by the boards that capture the signals from the transducers in real time.

Tests were performed on 6310ZZ (Double Shielded) bearings using Polyrex EM 103 (Polyurea) grease for higher speeds and MT33 (Mineral Oil and Lithium) grease for lower speeds. 18 tests were carried out in 9 different operating conditions varying: rotation speed, load applied to the bearing and use of thermal paste in the assembly. In the tests performed, the grease used for rotation tests up to 600rpm was MT33, while for rotations above 900 rpm, EM 103 grease was used. Table 1 provides details on how each assay was performed.

Table 1. Experimental conditions for carrying out 18 rolling tests.

Test Condition	Tests	Rotation	Radial Force in Rolling Bearing	Grease	Thermal paste in assembly	Initial temperature in tests
1	1	350rpm	0N	MT33	Não	15.2°C
2	2	350rpm	540N	MT33	Não	19.1°C
	3	350rpm	540N	MT33	Não	18.0°C
3	4	350rpm	790N	MT33	Não	15.0°C
4	5	550rpm	540N	MT33	Não	18.0°C
	6	550rpm	540N	MT33	Não	17.6°C
	7	550rpm	540N	MT33	Não	17.6°C
	8	550rpm	540N	MT33	Não	16.0°C
	9	550rpm	540N	MT33	Não	14.2°C
5	10	600rpm	790N	MT33	Não	15.5°C
	11	600rpm	790N	MT33	Não	13.9°C
6	12	900rpm	1000N	EM103	Sim	22.1°C
7	13	1000rpm	460N	EM103	Sim	20.4°C
	14	1000rpm	460N	EM103	Sim	20.3°C
	15	1000rpm	460N	EM103	Sim	19.3°C
	16	1000rpm	460N	EM103	Sim	17.8°C
8	17	1000rpm	460N	EM103	Não	20.8°C
9	18	1800rpm	1000N	EM103	Sim	22.2°C

4.2 Theoretical model of thermal resistances in the housing bearing

The thermal resistance model applied to housing bearing is shown in Figure 5. The domain is represented by 49 discrete volumes, the 39 circles represent the nodes defined for the discretization of the model and between each node the connected thermal resistances are represented. Table 2 shows the location of heat sources at the friction and convection model nodes.

Table 2. Location of heat sources in the thermal resistance model applied to housing bearing.

Nodes	Phenomena	Model
T2, T8, T9, T13	Forced Convection	$R_{conv,d} = \frac{1}{h.A_d}$, $\overline{Nu} = 0{,}133\ {Re_r}^{2/3}.{P_r}^{1/3}$
T14, T16, T17, T21, T22, T26, T27, T28, T29, T30, T31, T32, T33	Natural Convection	$R_{conv,d} = \frac{1}{h.A_d}$, h =5W/(m².K)
T34 e T35	Heat Generation by friction torque	$Q_1 = M_{atrito}.\omega.\frac{R_2.}{(R_1+R_2)}$ $Q_2 = M_{atrito}.\omega.\frac{R_1.}{(R_1+R_2)}$.

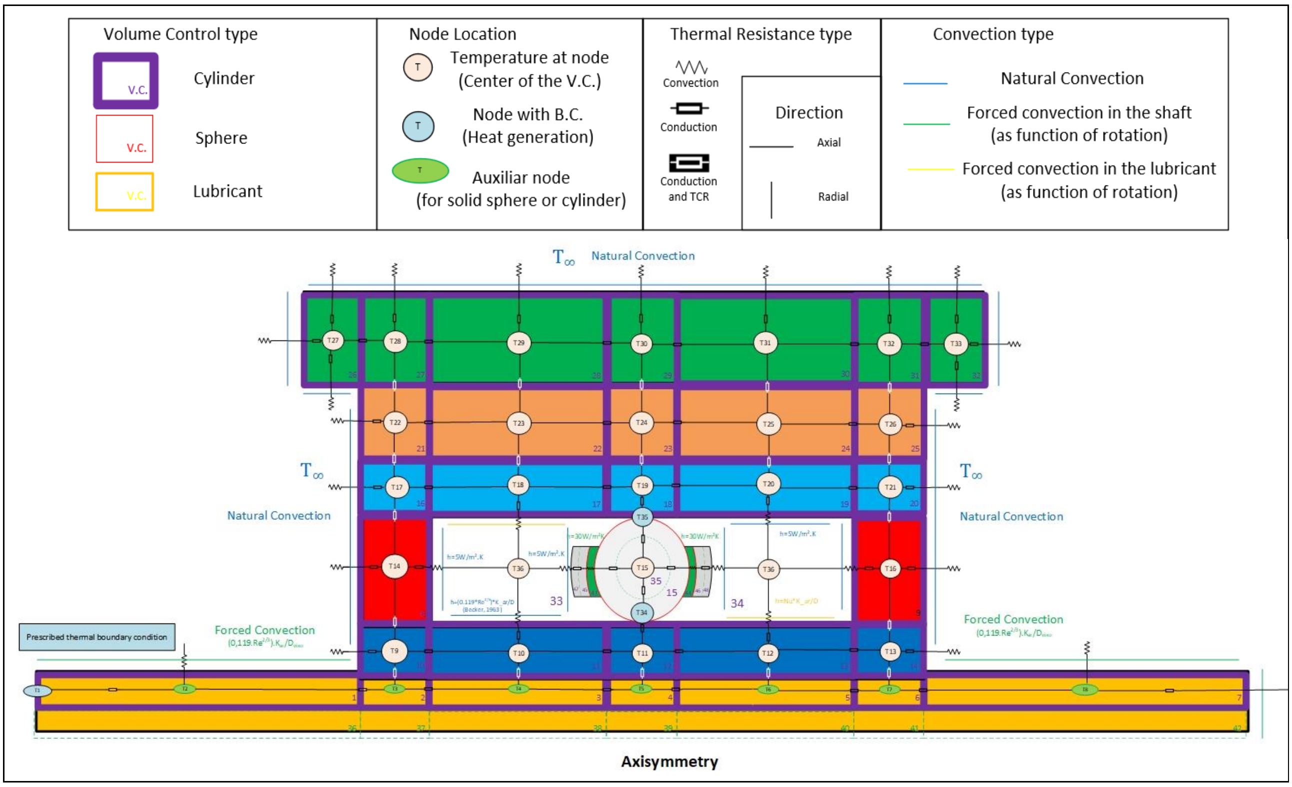
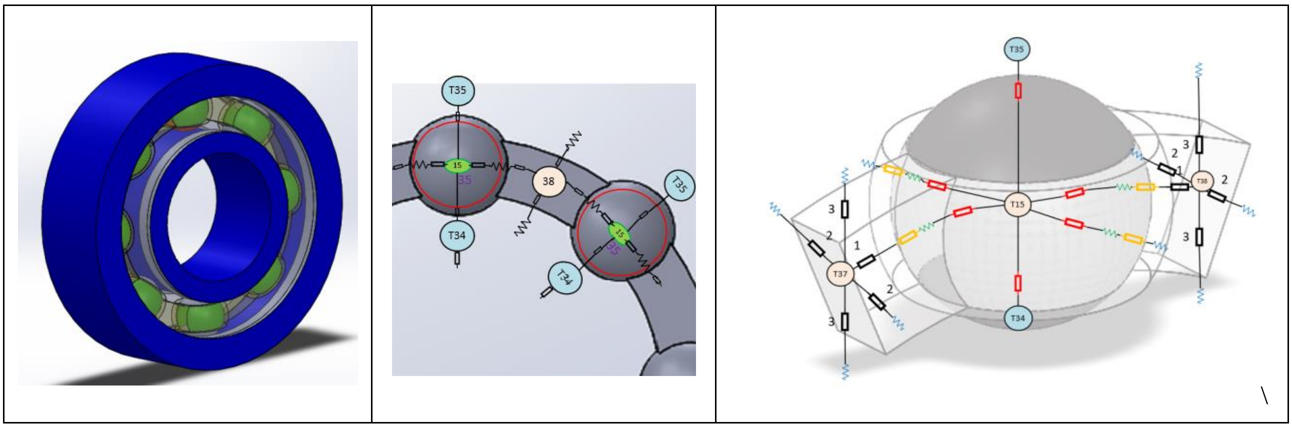


Figure 5. Network model of thermal resistances applied to the Housing bearing.

The network of thermal resistances applied to Housing bearing also has a detailing in the internal region of Rolling Bearing. As the representative drawing of the bearing is shown in Figure 6 (a). The spheres are considered to be equidistant from each other (symmetric) and exchanging heat in the cage, as shown in Figure 6 (b). In Figure 6 (c), the thermal resistances that make up a single sphere in symmetry condition are presented. In red are the thermal resistances by conduction in the ball, in green, the thermal resistances by convection of the lubricant film around the ball, in yellow, the thermal resistances of conduction of the housing that holds the ball in the bearing cage, in black, the thermal resistances of conduction of the lateral support of the cage and, finally, in blue, the thermal resistances of convection of the fins in contact with the air inside the bearing.



(a) (b) (c)

Figure 6. (a) Bearing model, (b) symmetry of thermal resistances and (c) detail of the bearing symmetry profile.

The model is validated by substituting the measured torque instead of the torque model calculated. The temperature calculated by this method will be compared with the value measured by thermocouples.

5. RESULTS

Results for each test condition were obtained after a 6 hour test period. This time was defined as the temperature at which the bearing reached the permanent temperature regime (defined as a variation of less than 1ºC/hour). The Figure 7 illustrates by means of a bar graph, the values recorded for each test condition of table 1 side by side, the black bars are the measured temperatures, the red bars the temperatures the calculated by the proposed model using the measured torque as input, and the green bars the temperatures calculated by the model using the calculated the SKF Friction Torque Model as input. It can be seen that when the results are placed side-by-side, the temperature calculation by the proposed model (in red) is closer to the real temperature (in black) than when using the theoretical model (in green).

Deviations between model and temperature measurement can be better analyzed in Figure 8, where each calculated value is displayed on the “y” axis and its corresponding value experimentally measured with a thermocouple on the “x” axis. The graph shows the points calculated by the proposed method using the friction torque measured in red and the linear regression of these points on the red line. The same was done using the calculated friction torque, whose results are presented by the points in green and its tendency represented by the green line of its linear regression. The thin black line is a regression of the measured temperature values only, where the dots at the top represent temperatures above the expected, and at the bottom, temperatures below the expected.

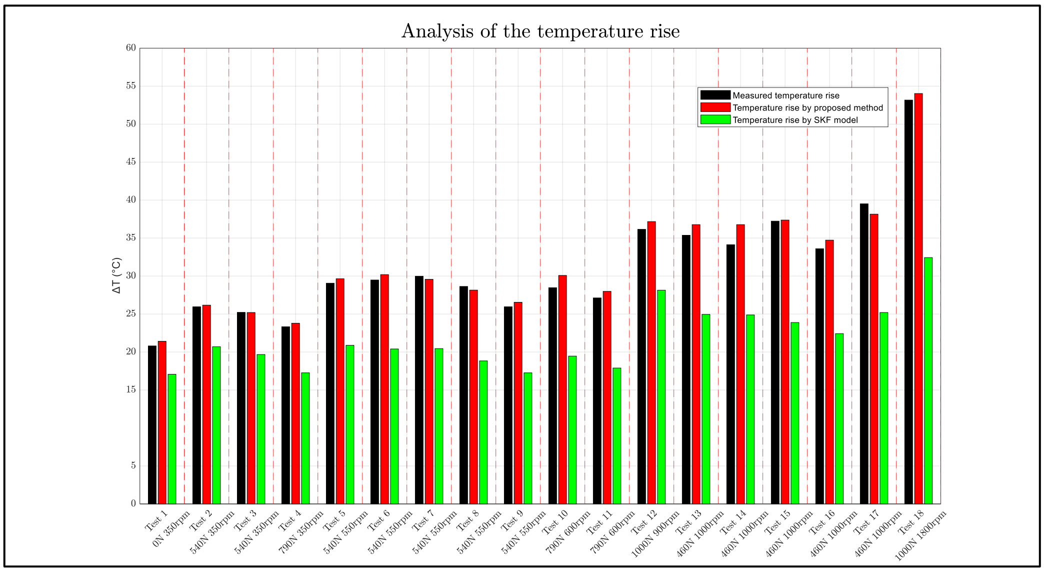


Figure 7. Comparison of Temperatures in each test.

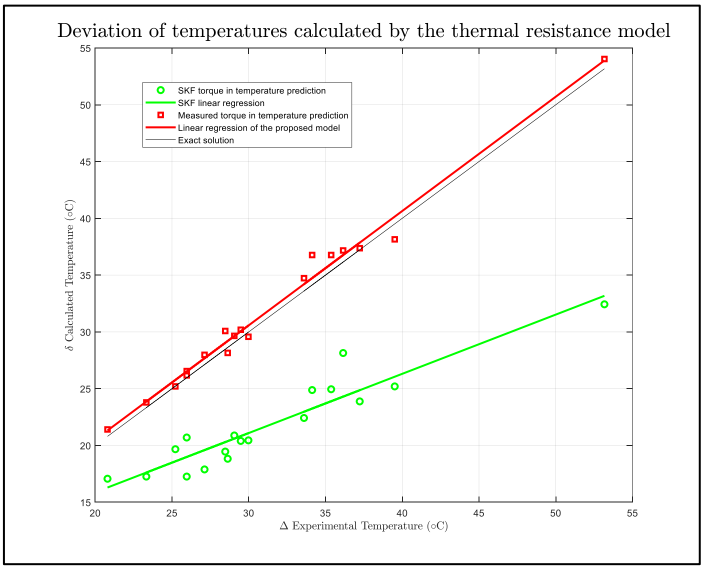


Figure 8 – Temperature deviation between calculated values and measured values.

6. CONCLUSIONS

The results presented about the "Rolling Bearing Power Loss Test Rig" show that is sensitive to the increase in temperature as expected (introducing greater load or rotation) and it is possible to repeat experiments according to the results presented.

Based on the experimental results shown in Figure 7 and Table 1, the following conclusions were reached:

• The the load increase in the bearing tests showed significant effects on the temperature increase of the bearings in the experimental tests and was evidenced by the temperature increase in the bar graph of the first 3 test conditions (Test condition 1, 2 and 3).

• Comparison of Test Conditions 2 and 4, 3 and 5, 6 and 9, show that the rotational speed causes Temperature increase in the Rolling Bearing. This is evidence that speed can have a direct influence on heat generation as stated in equation 2 where heat generation can be determined by the product of angular velocity and frictional torque. In addition, the speed can influence the friction torque, however more studies on this analysis should be carried out to verify which factor is more influential.

• Test repetitions in Test Conditions 3 and 7 show that there are conditions to obtain repeatability of results and the initial test temperature variation shows that it almost does not influence the temperature increase at the end of the test

• Comparison of Test Condition 7 and 8 shows that the use of thermal grease in the bench assembly indicates an improvement in heat dissipation where Test 17 without thermal grease shows a higher temperature compared to the other 4 tests without thermal grease (Tests 13, 14, 15 and 16).

The proposed method for calculating temperatures using thermal resistances, using torque recorded by the torque cell in tests as a source of heat generation with ambient temperature measurement proved to be accurate. The maximum temperature deviation in relation to the experimental results was 2.6ºC in Figure 7. It is also verified in Figure 8 that the proposed methodology (red points) is presented around the ideal line (black line), and its tendency is presents parallel to the ideal line, indicating that the model presents a good correlation between the calculated temperatures and the measured temperatures. However, the same does not occur when the SKF Torque model is introduced in the temperature prediction model, both the points and the trend indicate that there is a need for improvement.

Regarding Friction Torque, the SKF Friction Torque model is not efficient under the test conditions presented. For this reason, the temperature estimates using the thermal resistance network are well below the expected. While the equations of the model developed to estimate the friction torque were prepared considering oil as a Newtonian lubricant, in the experimental tests outside greases are used as a lubricant. For this case, the torque calculation of this model considers only the viscosity of the base oil that makes up the grease, so a possible cause of this deviation is the fact that the grease has a higher viscosity at low shear rates.

It is concluded that the methodology to validate the temperature prediction through a model consisting of thermal resistances is more assertive when considering the actual values measured in its energy balance. The advantage of this method is that the uncertainty arising from the heat source is eliminated, proving that the model's discretization is sufficient and that its simplifying assumptions do not affect the evaluation of the result.

In this way, knowing that the thermal resistance model is adequate, the limitation of the temperature prediction is restricted to the estimation of heat generation. The next steps of this project focus on improving the estimation of the friction torque model in search of a fully theoretical and assertive temperature prediction model. In future works, new friction prediction models using grease can be developed with the objective of significantly reducing this error and thus establishing a fully theoretical temperature prediction model using thermal resistance networks.

7. ACKNOWLEDGEMENTS

The authors are thankful to Aleksander Kokot and M.sc. Fernando André Lindroth Dauner for their discussion. The authors gratefully acknowledge WEG S.A. Company and Conselho Nacional de Desenvolvimento Científico e Tecnológico (CNPq) (process 141839/2020-1) for the financial support.

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