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EXPERIMENTAL ANALYSIS OF FRICTION TORQUE LOSS IN GREASE LUBRICATED DEEP GROOVE BALL BEARINGS

Atilio Silva

Diego Martins

Marcos Hiroshi Takahama

Cezar Otaviano Ribeiro Negrão

Tiago Cousseau

Universidade Tecnológica Federal do Paraná –Campus Curitiba R. Dep. Heitor Alencar Furtado, 5000 -Cidade Industrial De Curitiba, Curitiba.

atilio@alunos.utfpr.edu.br

diegoaguilar@alunos.utfpr.edu.br

marcostakahama@alunos.utfpr.edu.br

negrao@utfpr.edu.br

tcousseau@utfpr.edu.br

Abstract. *Ninety percent of rolling bearing are grease lubricated. Besides that, rolling bearing friction torque and life predictions only take into account grease base oil properties. In this context, the objective of this research is to compare friction torque loss and temperature using the SKF friction model, and compare it with experimental tests. To investigate the accuracy of the model, tests were conducted on the Rolling Bearing Power Loss Test Rig - RBPLR at the Universidade Tecnológica Federal do Paraná. The test rig allows for controlling the rotation and load imposed on the bearing, while measuring frictional loss and temperature. In total, nine eight-hour tests were carried out on 6310ZZ deep groove ball bearings lubricated with GJN grease, with speeds of 900 rpm, 1350 rpm, and 1800 rpm, and loads of 100 N, 250 N, 500 N, 750 N, and 1000 N. Experimental results show the SKF friction torque model underestimates the measured values in the analyzed interval of load and speed. The dependence of friction torque loss with load and speed was shown to be nearly, which agrees with sensibility with the predictions in a different scale.*

Keywords: *Bearings, friction torque, lubricating greases, theoretical and experimental results.*

1. INTRODUCTION

A precise energy efficiency model of rolling bearings allows an appropriate selection of components through the combination of rolling bearing and grease. However, the physical modeling of the lubricated contact effect remains a challenge, as demonstrated in the literature (Wu et al., 2022). The friction on rolling bearings originates from the interactions on the lubricated point contact between the surfaces of the internal components of the rolling bearing rolling elements, cages, rails, and seals) and friction is transformed into heat (Almeida and Fonseca, 1997).

The manufactures of rolling bearings currently recommends the use of SKF's friction torque model (2003), based on experimental tests (Liu et al., 2022) based on results of experimental tests using different combinations of various oil-lubricated and different geometry of rolling bearings. In this model, friction losses are a sum of empiric's models of separated kinematic analysis of the interaction between lubricant and bearing and are widely used as a reference in works related to temperature prediction (Gonçalves et al., 2017; Kanazawa et al., 2020; Krishnan and Nayani, 2021).

The SKF model (2003) recommends use of the friction torque model for grease-lubricated bearings as well, considering the base oil composition as an input parameter. As verified by Cann, P. M. E. (1996) through experiments, the concentration of thickener at the contact inlet is practically zero for high shear rates, which explains the possibility of using the Hamrock and Dowson model (1977) to predict the film thickness capable of separating the metal surfaces, considering only the viscosity of the base oil. However, the literature shows that there are discrepancies of up to 500% (Kanazawa et al., 2020; Ianus et al., 2016; Cousseau, 2013) in some studies when comparing the SKF model (2003) with experimental tests on grease-lubricated bearings.

In this context, this work aims to compare the torque loss due to friction and the temperature rise using the SKF friction model (2003) to compare experimental tests of rolling bearings grease lubricated at different speeds and loads to verify the accuracy of the model at low speeds (low shear rates in theory). This study was do using Rolling Bearing Power Loss Test Rig - RBPLR at the Universidade Tecnológica Federal do Paraná. The test rig allows for controlling the rotation and load imposed on the bearing while measuring friction loss and temperature. In total, nine eight-hour tests were performed on 6310ZZ/C3 deep groove ball bearings lubricated with GJN grease, with speeds of 900 rpm, 1350 rpm, and 1800 rpm, and loads of 100 N, 250 N, 500 N, 750 N, and 1000 N.

2. THE SKF FRICTION TORQUE MODEL

The bearings are machine elements with the purpose of support of rotational shafts and enable the transmission of rotational motion. As shown in Figure 1, the fundamental parts consist in: the inner ring, the outer ring, the rolling element (balls or rollers), and a cage to fix equal position between the rolling elements.

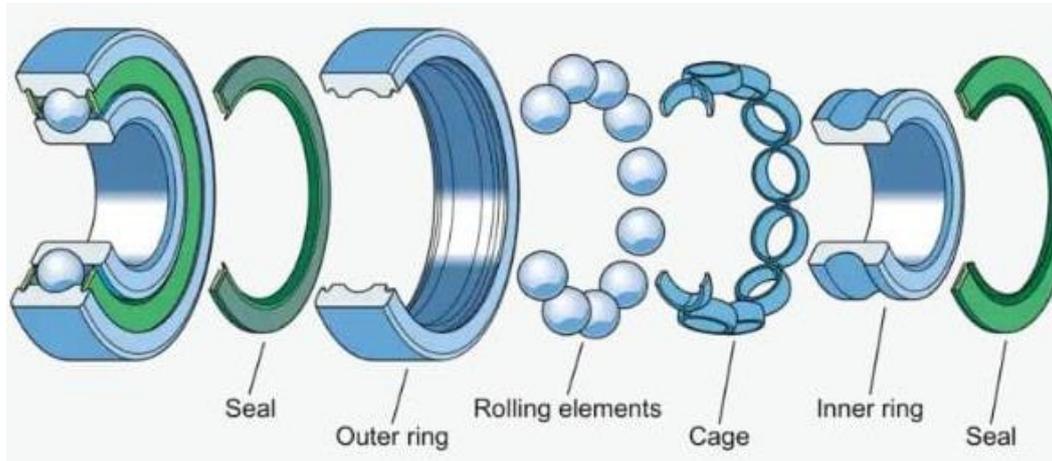


Figure 1. Fundamental parts of a Rolling bearings.
 Available from: (SKF General, 2003)

The SKF model (2003) “ 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 (N.mm), “ M_{sl} ” referring to the friction generated by the sliding of the rolling element (N.mm), “ M_{seal} ” related to the sealing friction of the seal that keeps the lubricant in the bearing (N.mm) and “ M_{drag} ” represents the drag friction generated by the rolling elements and cage as they pass through the oil bath (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):

$$M_{skf} = M_{rr} + M_{sl} \quad (1)$$

The rolling friction moment M_{rr} occurs due to the sliding effect of the bearing ball when the lubricating fluid depletes on the raceway and due to the heating of the fluid caused by entry shear. The heating of the fluid caused by entry shear occurs due to an excess of lubricating fluid. During bearing operation, a small portion of the lubricant occupies the space between the ball and the raceway, generating a lubricating film. The excess lubricant accumulates, tending to form a counterflow to the normal flow of the lubricant, which is responsible for heating and shearing the lubricating film, thereby increasing the contact friction. The friction related about the translational sliding motion of rolling element on the raceways lubricated and represented by M_{sl} and with a significant portion of total of energy loss (Cousseau, 2013).

3. EXPERIMENTAL PROCEDURE

The Rolling Bearing Power Loss Test Rig (RBPLR), shown in Figure 2a, measure the friction torque and temperature of bearings under different load and speed conditions. The test rig is equipped with a servo motor (1) responsible for maintaining rotation on the main shaft (6). The rotational motion imposed on the shaft is transmitted for the rolling bearings located in the housing bearings (3, 4, and 5), and the torque generated by the resistance to the motion of the friction in these bearings causes torsional deformation in the torque cell (2), from which torque measurements are obtained. Thermocouples (7), positioned on the bearing and adjacent regions as shown in Figure 4 (b), allow for the recording of temperature evolution in each test. The bearings are radially loaded by means of a screw that pushes the central housing, which is floating, upwards. The load cell (8) records the applied load.

The torque cell (2) is strategically positioned at the output of the servo motor because the motor provides the rotational motion to the analyzed system, and since the only resistance to motion is due to the friction of the bearings attached to the shaft, the torque cell only measures only the bearing related friction torque.

The load cell (8) is positioned on the housing of the floating housing (4) in order to quantify the force exerted on the bearings by a screw located just below it.

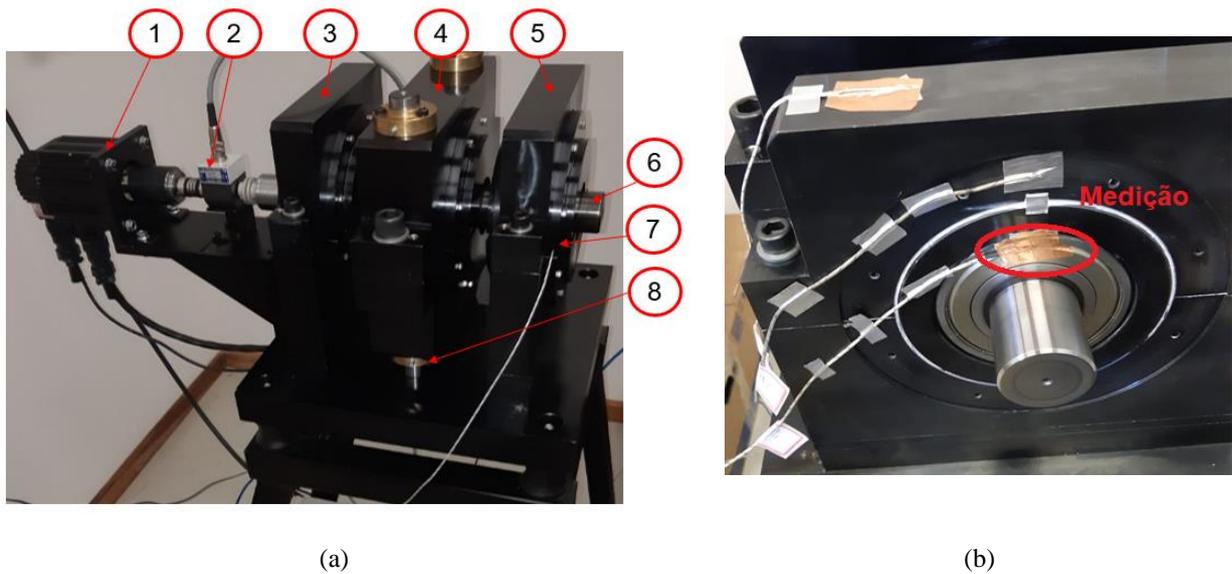


Figure 2. (a) The “Rolling Bearing Power Loss Test Rig”. (b) Position of the thermocouple for temperature measurement in the rolling bearing.
Available from: (Takahama et al. 2022)

The temperature of the bearing is recorded by a thermocouple set and an NI-9211 National Instruments acquisition board. The signals from the HBM C9C load transducers and T20WN/5N.m torque transducers data are recorded by a National Instruments brand board, NI-9218 module. The limitations of the tests on this bench are associated with the limitations of the torque and force transducers, as well as the servo motor. Therefore, the maximum measurable torque is 5 N.m, the maximum applicable load is 10 kN, and the maximum rotational speed is 3000 rpm. The experimental procedure was limited to tests speed at 1800 cause a high vibration was verified when the rotation is above 2000 rpm probably because the rotation of the shaft is nearly of natural frequency.

During the tests, in the transient state, there is a reduction in the measured value of the friction torque by the torque cell during the first hours of the test. This decrease is related, among other factors, to the spreading of the grease inside the bearing and the heating that reduces the viscosity of the grease. Therefore after 8 hours of rig working, the analysis of the results is considered for the last 30 minutes of the test, as shown in the Figure 3, where it is assumed that the torque and temperature stabilize (steady-state).

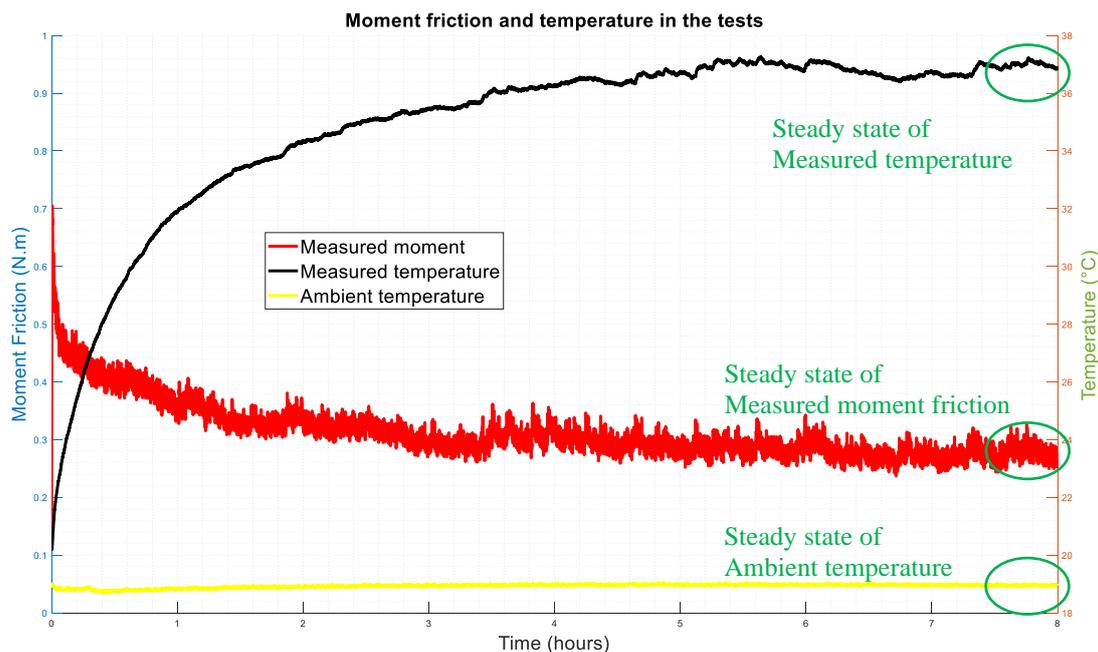


Figure 3. Measured torque and temperatures at the steady-state.

The tests were performed on 6310ZZ/C3 bearings (double shielded) using GJN greases (polyurea). Eighteen tests were conducted, varying the rotational speed at 900 rpm, 1350 rpm, and 1800 rpm, and the applied load of 100 N, 250 N, 500 N, 750 N, and 1000 N on the bearing. Table 1 provides the information.

Table 1. Experimental conditions of 18 test of the Rolling Bearing.

Ambient temperature on the measurement					
Speed (RPM)	Loads				
	100N	250N	500N	750N	1000N
900	17.6 °C	17.5 °C	18.2 °C	19.1 °C	19.3 °C
1350	18.4 °C	17.7 °C	18.2 °C	20.0 °C	19.0 °C
1800	18.2 °C	18.6 °C	18.3 °C	19.8 °C	18.8 °C

4. RESULTS

The measured data for friction torque and increase of temperature at the end of each bearing test on the test bench are presented in Tables 2 and 3, respectively. The increase of temperature value considers the difference between the measured value on the housing and the ambient temperature during the test.

Table 2. Experimental results for friction moment.

Measured friction moment					
Speed (RPM)	Loads				
	100N	250N	500N	750N	1000N
900	0.134	0.150	0.138	0.145	0.148
1350	0.113	0.126	0.144	0.141	0.173
1800	0.131	0.139	0.143	0.157	0.161

Table 3. Experimental increase of temperature.

Increase of temperature					
Speed (RPM)	Loads				
	100N	250N	500N	750N	1000N
900	10.1	11.7	10	10.6	11.2
1350	13.4	15.4	14.5	13.4	18
1800	18.1	18.1	19.4	18.4	19.9

In the following subsections, the behavior of friction moment and temperature, sensitive to changes in the controlled operating condition, is showed graphically to analyze the sensibility about the increase load and speed.

4.1 Analysis of measured friction moment

The graph in Figure 4 was generated by considering each line corresponds to a controlled constant load and each point in this line a variation of a controlled bearing rotation in each of the conducted experiments measured for the friction moment values. The experimental values represented by the lines in the graph were compared with the theoretically obtained values using SKF's friction moment prediction model, represented by the dashed lines. The color of the line in the graph refers to the same bearing load value, both for the theoretical prediction and the experimental data.

It is noticeable that, for the experimental evaluation conditions used, SKF's friction moment model deviates from the experimentally measured values. For the data with lower rotation speeds (900 rpm), it can be observed that the measured moment does not increase predictably with the increase in the applied load on the bearing, which may be related to the component not being designed for operation at such low speeds. For higher rotation speeds (1800 rpm),

the increase in load shows a higher predictability of the friction moment values, and the theoretical data appear to be more consistent with the experimental data, although to lesser value.

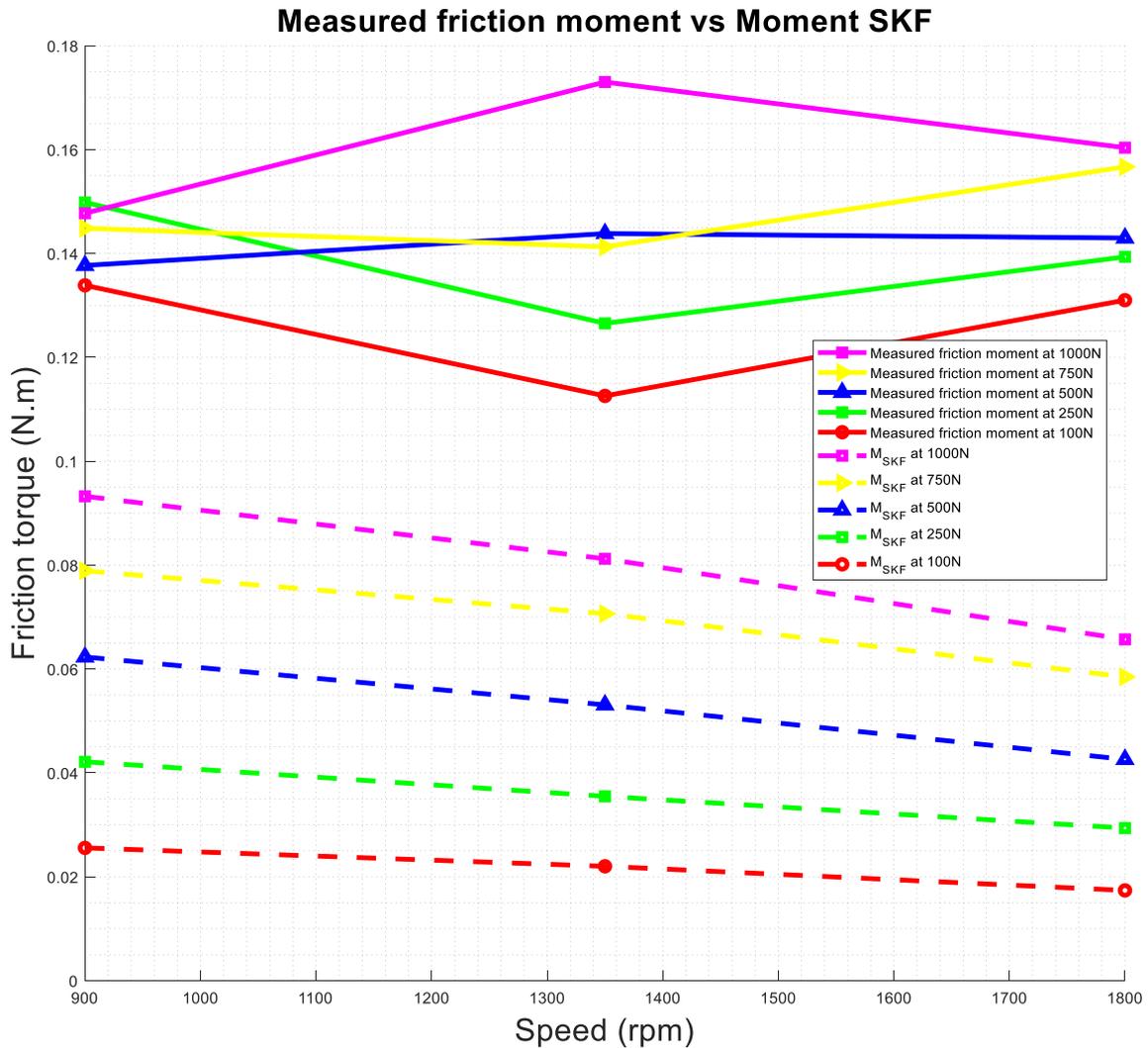


Figure 4. Measured friction moment vs SKF model friction moment.

4.2 Analysis of increase temperature

The graph in Figure 5 considers each color line as a controlled constant speed in each experiment and relating the controlled load to the temperature rise (by Δ temperature). It can be observed that for these experimental conditions, the temperature rise increases significantly with the variation in rotation, represented by the separation and non-crossing of lines, and has a slight sensitivity to the increase in load.

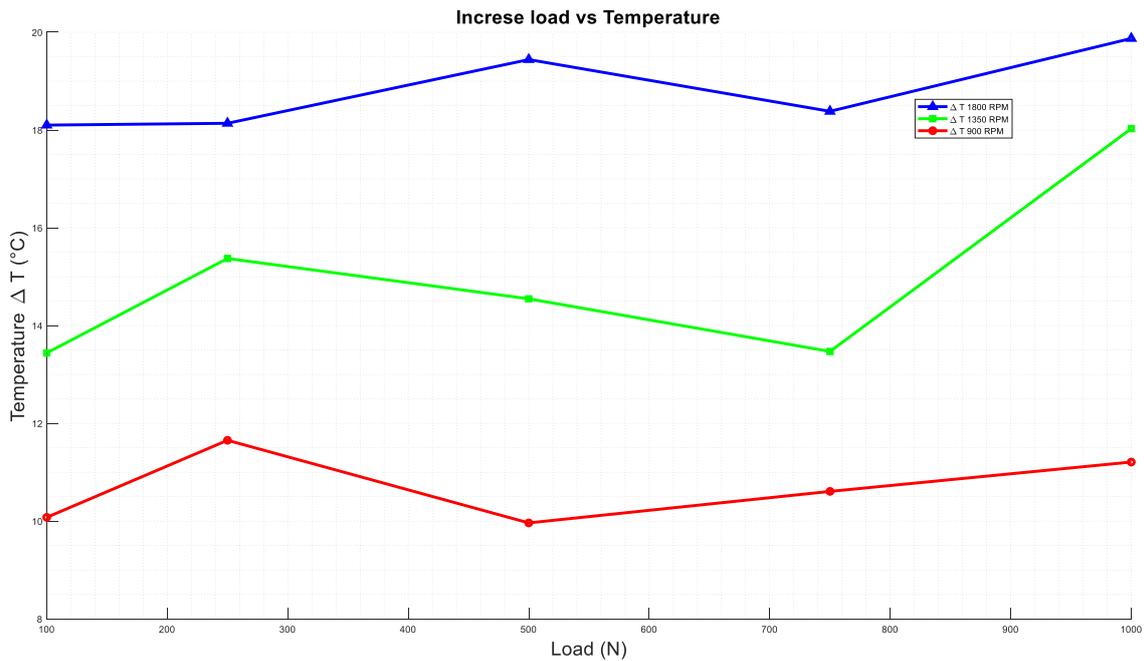


Figure 5. Experimental increase load and temperature rise behavior.

4.3 Relation between measured moment friction and temperature

The graph in Figure 6 was generated by considering each line as a controlled constant speed in each experiment and relating the measured friction moment to the temperature increase (Δ temperature). The most left point indicate the first load of 100N and the sequence to right the increase of load for each experiments. It can be observed that the temperature rise increases significantly with the variation in rotation, represented by the separation of lines and non-crossing. The temperature increase shows a slight sensitivity to the increase in load for rotations above 1350 rpm, as seen by the slight upward trend in temperature with increasing load at 1350 rpm and 1800 rpm (green and blue lines, respectively). At 900 rpm, the friction moment appears to be unpredictable at some points.

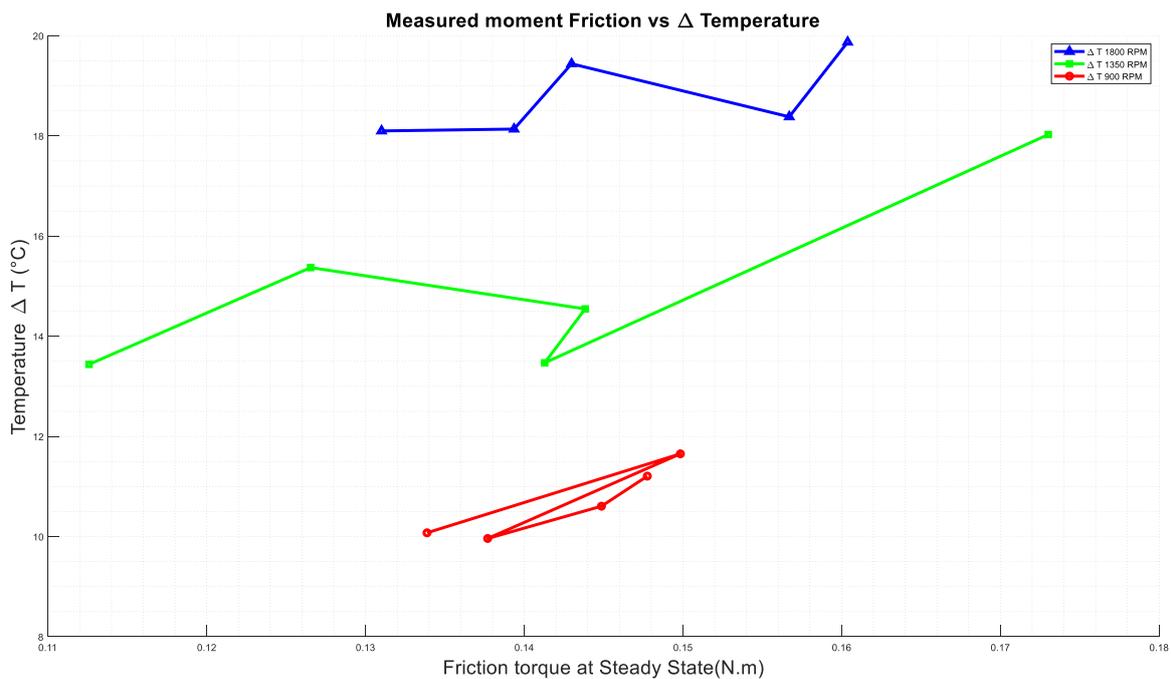


Figure 6. Measured friction moment and temperature rise behavior (Δ temperature).

5. CONCLUSIONS

The presented results show that increasing the load or speed on the 6310-ZZ/C3 bearing lubricated with GJN grease, tested on the "Rolling Bearing Power Loss Test Rig", leads to an increase in friction torque and temperature.

Based on the experimental results shown, the following conclusions is reached:

- An increase in rotational speed under constant load, as shown in Figure 4, the measured friction torque for this grease-lubricated bearing is higher (between 7.5x and 1.6x) compared to the SKF friction model. Furthermore, the reduction in friction torque variation with increasing speed is smaller in the experimental results than in the theoretical model. For these speeds (900 rpm, 1350 rpm, and 1800 rpm) and loads (100 N, 250 N, 500 N, 750 N, and 1000 N), it is expected by the theoretical model that moment friction increases with increasing load and decreases with increasing speed. While most of the measured values adhere to this rule, the test conditions of 900 rpm and 250N; 1350 rpm and 750N deviated from this premise.

- The increase in bearing temperature in each test is more sensitive to speed increase (vertical separation of lines) than to the load imposed on the bearings (from left to right), as shown in Figure 5. As observed in the same figure, there is a slight trend of temperature increase in the three lines, and this increase does not necessarily occur for all cases of load increase.

- In the graph in Figure 6 it is evident that the variation in speed (vertical separation of lines) has a greater influence on temperature increase than friction torque. This temperature increase is related to the heat power generated by friction (W), which can be calculated by multiplying the friction torque (N.m) and angular velocity (rad/s) (Takabi and Khonsari, 2016; Harris, 2006). The angular velocity for these experimental conditions starts from a base value of 900 rpm and is multiplied by 1.5x and 2.0x, significantly increasing the heat generation. The variation in friction torque of about 10% for these experimental conditions has a much smaller influence on the generated heat.

The SKF friction torque model proved inadequate for predicting friction torque in grease-lubricated bearings with loads of up to 1000N and speeds of 1800 rpm. SKF recommends using a minimum load of 1% of the ball load, which for the 6310ZZ/C3 bearing is 650N. This minimum load is related to failure (fatigue, slippage, adhesive wear of the raceways, or cage damage), but not directly to the friction prediction model. Regarding the lubricant properties, the torque calculation in this model only considers the viscosity of the base oil that makes up the grease. Therefore, a possible cause of this deviation is the fact that the grease has higher viscosity at low shear rates (Wozniak et al., 2023).

In conclusion, based on the experimental tests, the friction torque model proves inadequate for predicting friction torque, despite the bearing tests showing temperature increase results sensitive to changes in operating conditions such as speed and load, as expected physically.

Finally, this work highlights the need for the development of new friction prediction models that consider the properties of grease and adequately predict friction in grease-lubricated bearings. Most greases exhibit non-Newtonian fluid behavior, such as pseudoplastic fluids (Wozniak et al, 2023) and that viscosity decrease with increase the shear rate.

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