ANNALS OF THE ORADEA UNIVERSITY
Fascicle of Management and Technological Engineering
ISSUE #1, MAY 2015, http://www.imtuoradea.ro/auo.fmte/

ON THE MEASUREMENT PROCEDURE FOR TESTING FRICTION IN BEARING BOXES

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***Abstract*—**The aim of this research is friction measurements on the bearing boxes of a testing rig for parallel axes transmissions. Friction in the transmission without bearings is calculated by subtracting the bearing friction from the global friction. This is why it is of maximal importance in the correct evaluation of friction losses to have accurate data on the friction on bearings. This paper is presenting the main specific aspects of the measurement procedure that must be carefully considered for accurate results. Finally, the friction torque in bearing boxes is presented depending on rotational speed, load and lubricating oil temperature.

***Keywords*—**Bearing boxes, friction, testing procedure

I. INTRODUCTION

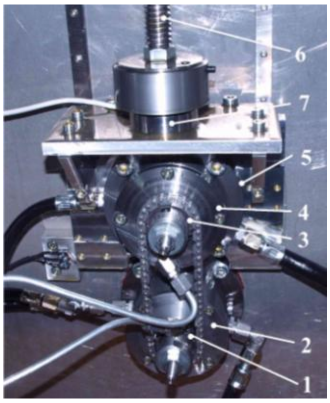
TRIBOLOGICAL performance improvement of automobile engine systems, can generate a lot of benefits: reduced fuel and oil consumption; increased engine power output; reduction in harmful exhaust emissions; improved durability, reliability and engine life; reduced maintenance requirements and longer service intervals [1]. In the analysis of [2], the following factors are taken into account: car energy consumption, driving cycle effects, friction loss distribution, tribo-contact friction levels today and the future, global fuel consumption today and potential savings. The part of the fuel energy devoted to mechanical power to overcome friction can be subdivided into groups based on data from [1, 2, 3, 4, 5]: – 35% (12–45%) to overcome the rolling friction in the tire–road contact, – 35% (30–35%) to overcome friction in the engine system, – 15% (7–18%) to overcome friction in the transmission system, and – 15% (10–18%) to overcome friction in the brake contact . Very few experimental results on chain friction have been published. Reference [3] proposes a technique for precisely measuring sliding loss in the timing chain and the loss in the guides of an engine, using an equipment consisting in a full engine, in order to separate timing chain system losses into components. The test rig can simulate different temperature conditions and allows a good replica of the real engine conditions.

The object of the present paper and current research is testing of friction on the existing bearing boxes of a testing rig for chain or belt drives. The bearing boxes are components of a testing rig and the final purpose of the research will be the evaluation of friction losses in parallel axes transmissions, depending on the tensioning force, speed, temperature and quality of lubrication.

Fig. 1. presents an image of the chain rig together with a functional diagram. A flexible torsion coupling transmits the torque from the engine to the driving sprocket. The driving sprocket (1) is mounted on the input shaft, which is part of the lower bearing box (2). The driven sprocket (3) is mounted on the output shaft, which is part of the upper bearing box (4). The upper bearing box is fixed on a sliding carriage (5) which allows vertical adjustment of the driven sprocket, ensuring the tensioning force. A rotating nut attached to an electric engine controlling axial displacement of a screw (6) accomplishes the self-adjusting control. The screw is connected to the upper bearing box sliding carriage by a force sensor (7). Specific sensors and devices measure the input shaft speed and input torque (determined by the frictions in chain and bearing boxes), the chain tensioning force, the temperature and pressure of the oil for the bearing boxes lubrication and for the chain lubrication. Testing rig devices control the input shaft rotational speed, tensioning force and temperature of the bearing boxes lubrication oil.

The measured input torque is a sum of all the frictional torques in the transmission: from the bearing boxes and from the chain transmission. Friction in chain transmission without bearings is calculated by subtracting the bearing friction (sum of friction in the two bearing boxes) from the global friction. It is of maximal importance in the correct evaluation of experimental measurements on the rig to have accurate data on the bearings friction.

The transmission uses two bearing boxes. Both bearing boxes are consisted of: one radial ball bearing taking radial force and possible axial force on both directions; one radial cylindrical roller bearing taking the most important radial force and sealing rings at both ends. Lubrication is based on oil flow. The rig gives the possibility to measure friction torque, by controlling and measuring radial loading at one end of the shaft, rotational speed, oil temperature and pressure.



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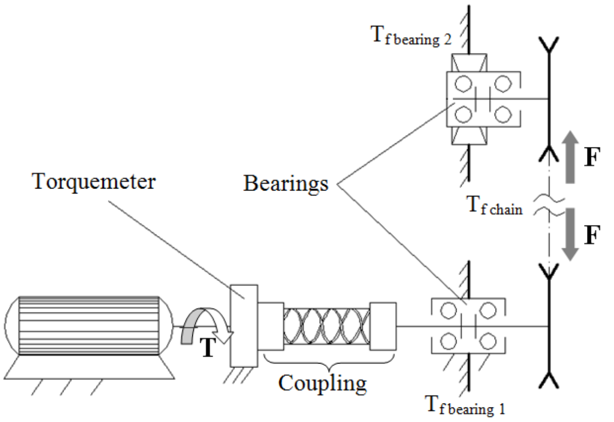

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Fig. 1. Chain transmission mounted on test rig: a) front view; b) functional diagram.

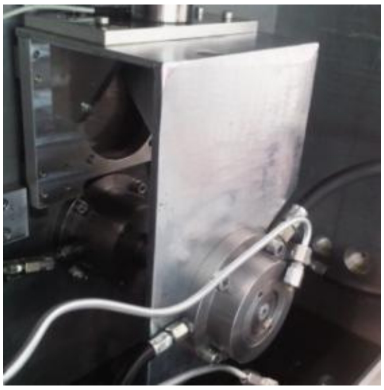
II. Measurement Procedure

A. Measurement Device

The details of the bearing friction measurement device are presented in [6]. It is adapted on the chain rig presented in Fig. 1. Fig. 2. shows an image of the device together with a functional diagram. The upper and lower bearing boxes are coaxially mounted.

The two bearing shafts are connected through a mobile coupling. Both bearing boxes are radial loaded with force F positioned as in the chain transmission tensioning situation. The load is applied through the tensioning system of the testing rig and a rigid element, mounted between the sliding carriage and the upper bearing box. The constructive solution gives the possibility of adjustment of the coaxially position of the two shafts. With this device, the measurement of bearing friction is performed in the same conditions of running as in the case of transmission testing. This is why the same tensioning, lubrication and drive systems and their measurement devices are used (see Fig. 1.).

For a given construction of bearing boxes (type of bearings, clearances, arrangement, dimensions, type of lubrication) bearing friction theoretically [7], [8] depends on load (F), rotational speed (n) and viscosity of lubricant (ν), which depends on temperature. Measurements have been performed for the following range of variables: load F = 1…3kN, rotational speed n = 1000…5000 rpm, lubricant temperature t = 35…50°C.



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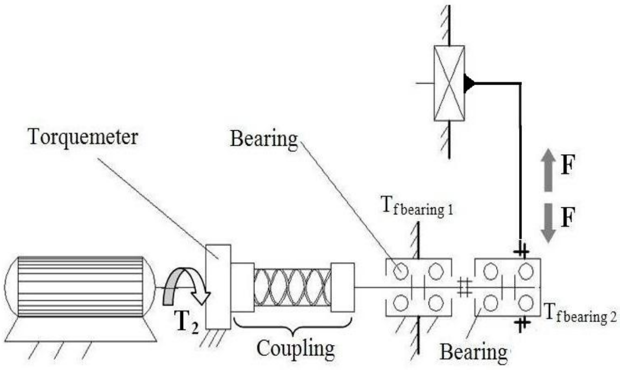

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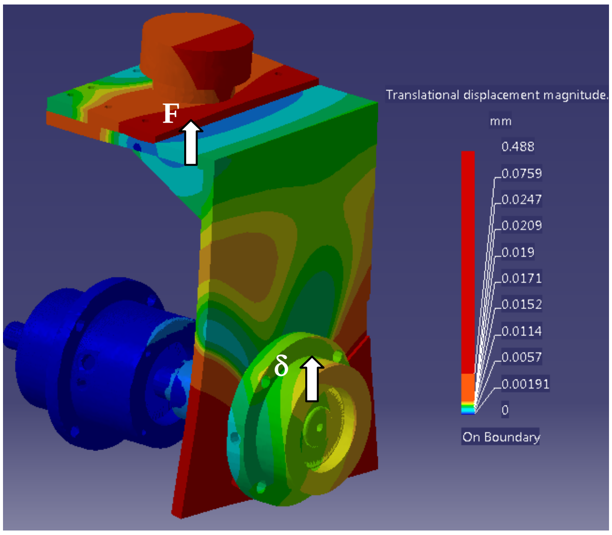
Fig. 2. Bearing friction measurement device: a) front view; b) functional diagram.

B. Preparation of the Rig

The most important part of preparation of the rig for bearing friction measurements is the adjustment of the coaxially position of the two shafts and check for clearances and elastic deformations. In order to reduce to minimum the friction losses in the mobile coupling linking the two bearing boxes, the precision of coaxially position must be as higher as possible. 3D modelling of the system is an instrument for optimization, allowing easy analysis and simulation of different load cases [9].

Fig. 3. shows the results of FEM analysis giving translational displacements and stresses on the device loaded with a force F = 1 kN. The resulted displacement at the level of the end of the output shaft is 0.015 mm.

Fig. 4. shows the measured deformations at the level of casing of the output bearing box as presented in Fig. 3. a, depending on load F. The translation at small loads is coming from taking the clearances from the system.



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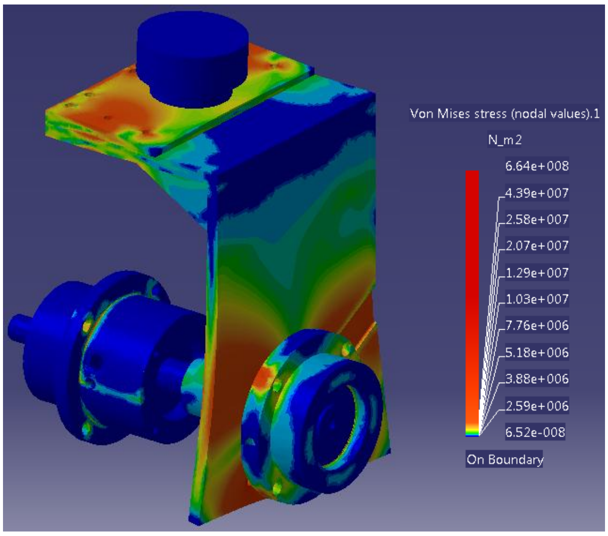

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Fig. 3. FEM analysis: a-translational displacements, b-Von Misses stress

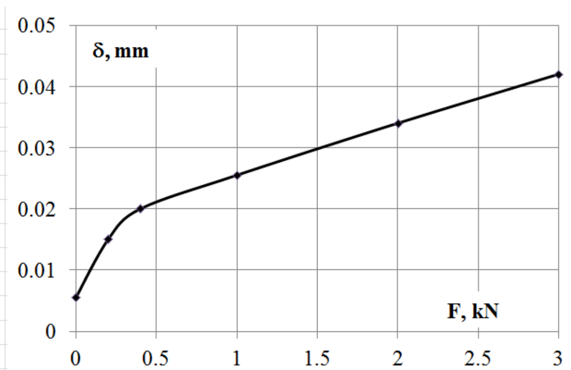


Fig. 4. Measured deformations

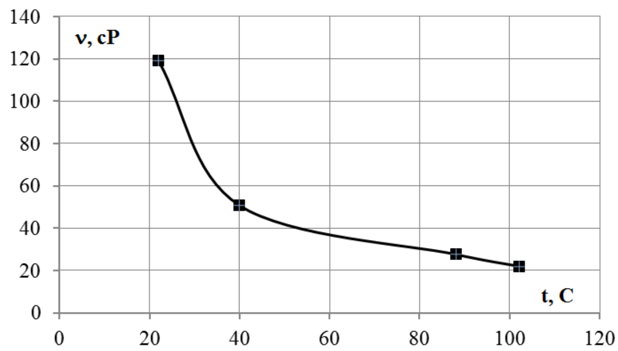
C. Time Depending Friction

Friction tests on chain or belt drive can take hundreds of hours. This is why time depending bearing friction is another aspect to consider. In order to evaluate bearing friction as precise as possible and maintain friction vs. time variation in reasonable limits two elements have been considered: 1 – lubricant deterioration (time depending viscosity and acidity of lubricant) and 2 – running in process and normal wear process influence on friction.

Lubricant parameters must be monitored in time. The oil used in testing is a multigrade 5W30 mineral oil. Mineral oils inevitably oxidize during service and cause significant increases in friction and wear which affects the performance of the machinery. The main effect of oxidation is a gradual rise in the viscosity and acidity of oil.

Measurements of kinematic viscosity of the mineral oil vs. temperature and running time have been performed. Fig. 5. a presents temperature depending kinematic viscosity for fresh oil and Fig. 5. b presents time depending kinematic viscosity of the oil.

It can be seen that viscosity starts to increase after 500 hours of running affecting mostly high temperature kinematic viscosity of the oil. If the friction tests are performed in the first 300 hours, changes in oil viscosity should not influence bearing friction measurements.



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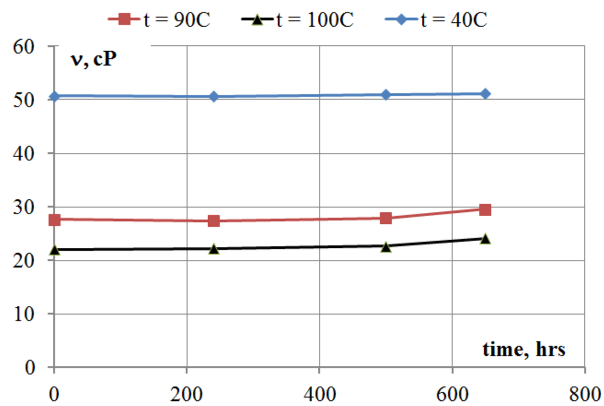

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Fig. 5. Kinematic viscosity: a- versus temperature, b- versus time

Acidity index depending on time is presented in Fig. 6. for two ranges of oil temperature. Speed of deterioration depends mostly on time of running at high temperatures.

The case of higher temperatures (60…100°C), appropriate to chain drives, shows an important increase of acidity index after only 200 hours of running.

In the case of medium temperatures (35…50°C) acidity index has a slowly increase of approximately 5% during 500 hours of running. This is the range of temperatures on which the bearings of the testing rig run.

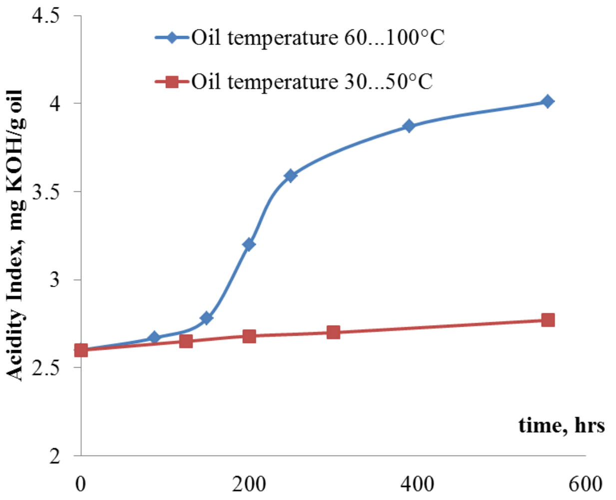

Fig. 6. Acidity index versus time

Running in is performed for 50 hours, at a speed n = 1800 rpm, with an applied load F = 1 kN, with lubricant temperature in the range t = 40…45°C, with continuous measurement of friction torque. Fig. 7. shows time depending friction torque (T_f) during running in process. Friction torque drops relatively fast during the first hours and then keeps dropping slowly showing that wear rate is stabilized at low value. Continuous dropping of friction torque shows that bearing friction must be re-evaluated from time to time. In our testing of parallel transmissions friction, bearing friction is re-evaluated at 300 hours of running. Differences of up to 5% have been found.

D.Testing Program

The rig offers the possibility of controlling and programing variation of rotational speed and also checking the state of lubricant temperature and pressure. For constant loading and temperature (F = ct., t = ct.) tests are based on the program based on the diagram presented in Fig. 8. It shows time depending rotational speed for one cycle of testing.

The first stage of the program is consisted of slowly increase and decrease of rotational speed. Increase and decrease of rotational speed in the range of 2000…4000 rpm is made in steps of 10 min. The reason for this stage is mainly to check the stability of the rig since the values from these measurements cannot be used in order to characterize the steady state of running.

Fig. 9. shows the measured friction torque depending on rotational speed during the processes of acceleration and deceleration. As seen from Fig. 9., there is a visible difference between measured torque during acceleration and deceleration since inertia of the rotating elements adds value to the measured friction torque during acceleration, while it reduces value during deceleration. This means that a correct evaluation of rotation depending friction torque should not be made during periods of acceleration or deceleration. The diagram from Fig. 8. also shows that there might be some vibration issues at a rotational speed of approximately 2300 rpm, more visible at deceleration.

The next stages of the program are consisted in steps of constant rotational speed. The first step is usually longer since it must check and adjust the oil temperature and also stabilize the temperature distribution on all the elements of the rig. The time for each rotational speed step, presented in the testing program from Fig. 8. is minimum 250 seconds but in some cases of low loads and high temperatures, we recommend longer periods.

Fig. 10. shows time depending measured friction torque for steps of constant rotational speed. It can be seen from Fig. 10. that the role of these steps is to stabilize the system and induce the steady state where the measurements are made. There is a general trend of stabilizing by decreasing the value of the friction torque in time. This stabilizing is faster for lower speeds.

The readings that count in evaluation of bearing friction are only the one of the steady state period. In this experimental research the measurements of friction torque considered in steady state is an average of the last quarter of each step of constant rotational speed.

III. RESULTS

Tests determining global friction torque on bearing boxes (T_f) have been developed for tensioning forces of 1kN and 3kN, rotational speed in the range 1000 ... 5000 rpm, oil temperature of 35ºC and 50ºC. Fig. 11. shows together the measured friction torque on bearing boxes (T_f), at oil temperatures of 35ºC, respectively 50ºC, for tensioning force of 1 kN and 3kN, depending on rotational speed.

The results show the increase of bearing friction with rotational speed and load and also with decrease of oil temperature (increase of viscosity). The trends are according to the theoretical models from [8]. The differences between experimental and theoretical results have been analyzed in [10], confirming that the experimental evaluation of friction in bearing boxes as a whole have been correctly performed.

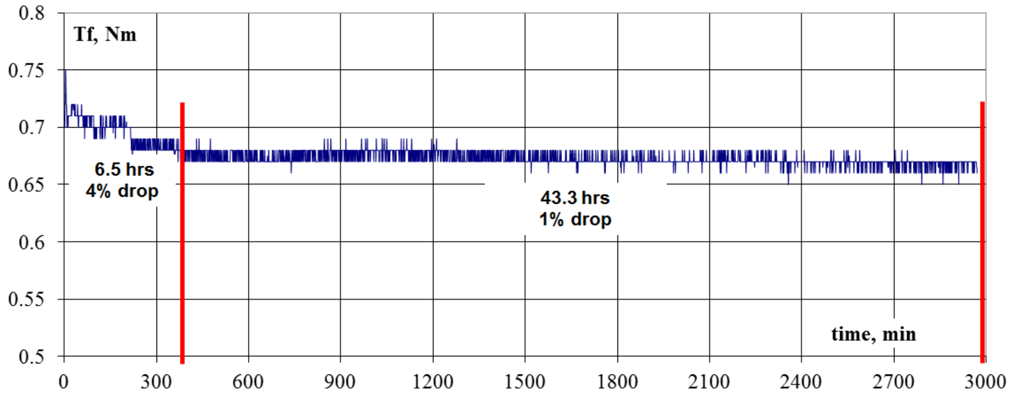

Fig. 7. Running in process

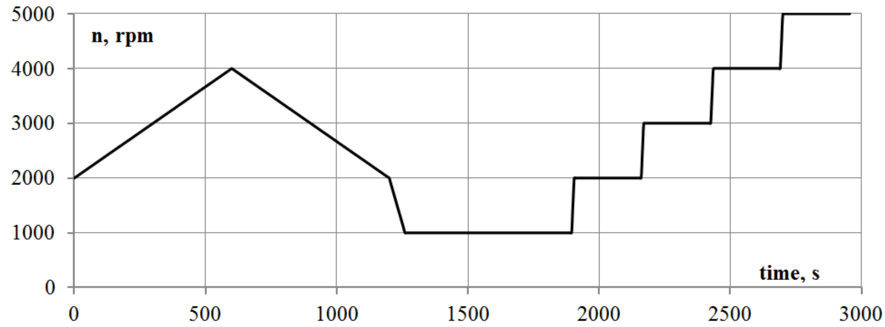

Fig. 8. Testing program

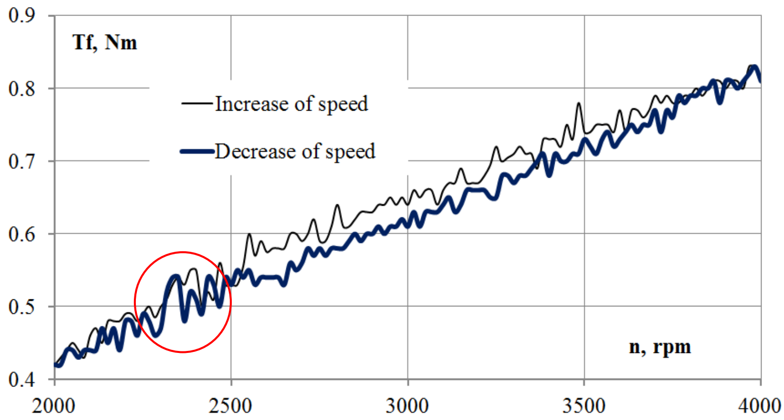

Fig. 9. Friction torque during acceleration and deceleration

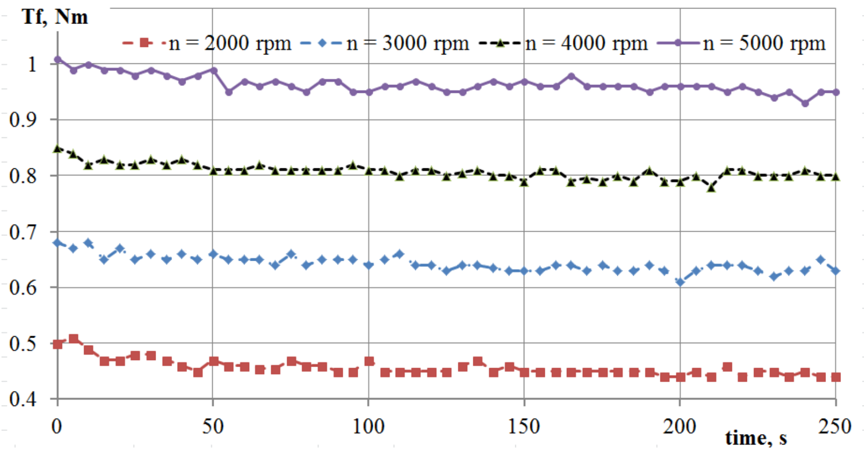

Fig. 10. Time depending friction torque for constant speed, load and oil temperature

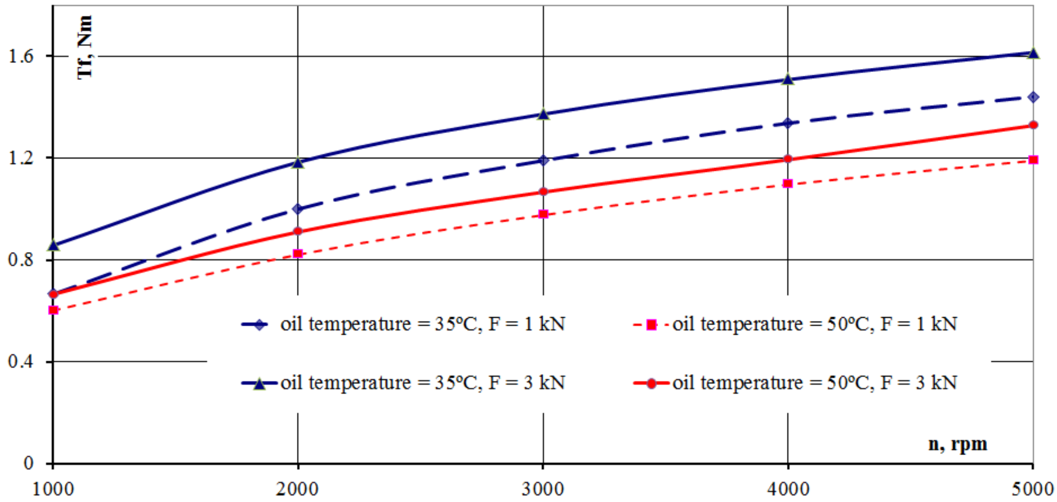

Fig. 11. Measured friction torque on bearing boxes depending on rotation, for constant tensioning force and oil temperature

IV. Conclusion

Evaluation with high accuracy of bearing boxes friction is difficult. For an existing construction (type and dimensions of bearings, type and dimensions of sealing elements, type of lubrication system and type of lubricant, position of bearings, sealing elements and loads) the remaining parameters involved in bearing boxes friction are: values of external loads; rotational speed and temperature depending kinematic viscosity of oil. Even if the theoretical models of individual bearing friction calculus are comprehensive, based on years of experiments and can be applied to bearing boxes, they cannot be considered as highly accurate. Author's opinion is that only experimental measurements of bearing boxes friction, copying exactly the conditions of functioning give accuracy of results. Several aspects like deformations, time depending parameters, measurement conditions and strictly following procedures have maximal importance. The experimental measurements presented in this research show the influence of load, speed and oil temperature on bearing boxes friction. This is just the first step in correct evaluation of friction in transmissions with parallel axes (chains, belts).

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