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A New Device Proposed for the Industrial Measurement of Rolling Bearing Friction Torque

Mateusz Wrzochal[1,*] – Stanisław Adamczak[1] – Ryszard Domagalski[1] – Grzegorz Piotrowicz[2] – Sylwester Wnuk[2]

[1] Kielce University of Technology, Poland
[2] Fabryka Lozysk Tocznych-Krasnik S.A., Poland

This article presents the general assumptions and the mechanical design of the new device for measuring the friction torque of rolling bearings, as well as a preliminary evaluation of the indications on the basis of which further detailed analyses and corrections of the prototype will be made. The device presented in the manuscript is dedicated to quality control as part of its support for rolling bearing plants.
Keywords: rolling bearings, friction torque, industrial measurement, quality control

Highlights
- *To support the rolling bearing quality control process, an industrial bearing friction torque measuring device was designed.*
- *A device was made on the basis of the proposed concept.*
- *The principle of operation and the role of the individual components of the innovative measuring device for rolling bearing testing are presented.*
- *The results of preliminary tests assessing the fulfilment of initial assumptions are presented.*

0 INTRODUCTION

Rolling bearings are a common component of machines and mechanical devices, which explains their great popularity among the interests of researchers throughout the world [1] to [3]. Often topics are taken up in relation to a specific bearing application [4] and [5]. Research from a typical production point of view is conducted less frequently [6] to [8]. It can be observed that problems related to bearing measurement under industrial conditions are generally solved by bearing producers individually; the information is confidential and, as such, it is not shared in the sector. It was thus very difficult for the researchers involved in the project to find any detailed descriptions of or patents for specialist measurement equipment to provide solutions to the specific problems encountered by Fabryka Lozysk Tocznych-Krasnik S.A., a leading Polish manufacturer of roller bearings. Since the main applications of roller bearings are in the mechanical equipment and automotive industries, the focus is on their quality. Credible measurement results obtained under laboratory conditions are required for reference by individual and industrial users all over the world to confirm the products they buy are of the best quality.

A rolling bearing can be assessed on the basis of a number of certain quantities. The friction torque, together with vibrations and durability, is one of the most important parameters determining the quality of bearings and their suitability for the intended applications. The friction torque contains a great deal of information about the bearing, whether in terms of its design, the quality of its mating parts, its cleanliness or certain lubricant properties [9] and [10]. Given the widespread use of bearings in the automotive, machinery or household appliance industries, a need is being generated to reduce friction in rolling bearings, which is a key measure in the quest to improve efficiency, reduce energy consumption, and protect the environment. Work in this direction focuses on isolating the factors that increase frictional resistance in a bearing and seeking to minimize their impact. At the same time, this creates the need to design devices that can accurately measure bearing friction torque and its subtle changes in relation to fluctuations in the factors that increase it [11].

The possibility of precise measurement of friction torque occurring in rolling bearings, as well as the knowledge of the dependence of their values on the conditions in which the bearings operate, including such factors as rotation and load, allows machine and equipment designers to optimize the selection of bearings in specific design nodes, while also enabling bearing manufacturers to assess their quality and choose the right directions for improving their designs [12] and [13].

Research on friction torque is especially important for nodes for which power losses are of high importance. Similar studies are undertaken by many research centres, as well as by leading bearing manufacturers [14] to [16]. They focus on isolating factors causing an increase in frictional resistance

*Corr. Author's Address: Kielce University of Technology, al. Tysiąclecia Państwa Polskiego 7, 25-314, Kielce, Poland, wrzochalm@gmail.com

in bearings and striving to reduce the influence of those factors [17]. All these efforts create the need to build devices that allow precisely measuring the tested bearings' friction torque and its subtle changes depending on the fluctuation of factors influencing its growth [18] to [20]. In the available sources, one can find both descriptions of test rigs measuring the friction torque of bearings intended for research purposes, as well as commercial offers of companies producing professional measuring equipment [21] to [23]. Industrial measuring devices at the disposal of companies producing rolling bearings are, however, an uncommon object of scientific research.

This article presents a new industrial device for testing the bearing friction torque of rolling bearings at the stage of their production. The device is characterized by an innovative design that allows for testing a wide range of bearing dimensions, applying significant axial loads, additional measurement of the mounting width of the bearings, and most importantly, testing the cone bearings.

1 FRICTION TORQUE OF ROLLING BEARINGS

Friction torque is defined as the resistance of the bearing when attempting to rotate one ring in relation to the other, and it depends to a greater or lesser extent on the following: type, variant and dimensions of the bearing, load values and its direction, rotational speed, type and properties of the lubricant and the method of lubrication. Friction losses in rolling bearings are caused, inter alia, by deformations at the contact between the rolling element and raceway, internal friction of the lubricant, slips and micro-slips, cage friction, as well as friction on seals [9] and [14].

The theoretical friction torque can be calculated from a well-established formula present in both older and more recent publications [3], [11], and [12]:

$$M = M_0 + M_1 , \tag{1}$$

where M_0 is a section of the equation independent of the load [N·mm]:

$$\begin{aligned} M_0 &= f_0 \cdot 10^{-7} \cdot (v \cdot n)^{2/3} \cdot d_m^3 \quad &&\text{for} \quad v \cdot n \geq 2000, \\ M_0 &= f_0 \cdot 10^{-7} \cdot 160 \cdot d_m^3 \quad &&\text{for} \quad v \cdot n < 2000, \end{aligned} \tag{2}$$

where f_0 is the coefficient depending on the type of bearing and lubrication conditions, selected on the basis of tables; v oil kinematic viscosity, [mm²·s⁻¹]; n rotational speed, [min⁻¹]; d_m bearing pitch diameter [mm].

M_1 is the section of the equation dependent on the bearing load, [N·mm]:

$$M_1 = f_1 \cdot P \cdot d_m , \tag{3}$$

where P is equivalent load, [N]; and f_1 factor depending on the type and size of the bearing and the permissible static load coefficient.

It should be mentioned that Eq. (2), as it appeared in the publications, is simplified to some extent. For example, in Eq. (2) for $v \cdot n < 2000$ the component that is 160 has a unit corresponding to the expression $(v \cdot n)^{2/3}$. In addition, Eq. (2) assume that the density of the lubricant is equal to about 0.9 kg/dm³. Therefore, the unit of density, which does not appear explicitly, compensates for the other units, which enables obtaining a unit that is identical to the friction torque.

There are other models proposed by industrial units, such as Timken FAG and SKF [24] to [26], which have extended the basic model with additional empirical components representing, for example, seal friction, friction resulting from the lubrication system,

Table 1. Table of values for the f_0 and f_1 factors

Bearing type	f_0			f_1
	Oil mist lubrication	Oil bath or plastic grease	Oil bath lubrication under pressure	Oil bath lubrication under pressure
Deep groove ball bearing	0.7 to 1	1.5 to 2	3 to 4	$0.0009 \cdot \left(\frac{P_0}{C_0}\right)^{0.55}$
Angular contact ball bearings	1	2	4	$0.0003 \cdot \left(\frac{P_0}{C_0}\right)^{0.4}$
Cylindrical roller bearings	1 to 1.5	2 to 3	4 to 6	0.00025 to 0.0003
Needle roller bearings	3 to 6	6 to 12	12 to 24	0.0004 to 0.0005
Spherical roller bearings	2 to 3	4 to 6	8 to 12	0.0001 to 0.0005
Tapered roller bearings	1.5 to 2	3 to 4	6 to 8	0.0004 to 0.0005

C_0 nominal static load capacity, and P_0 equivalent static load grease compression or splash, or have formulated other relations for determining the friction torque, taking into account the structure and specific operating conditions of these bearings. However, none of these detailed proposed new models is universal; moreover, for the same bearing type and the same operating conditions, sometimes very divergent results are obtained using different models, as presented in [11]. In future it will be possible to attempt to establish a new mathematical model with the presented device, but it requires many test runs.

2 GENERAL DESIGN OF A SYSTEM FOR MEASURING THE FRICTION TORQUE OF ROLLING BEARINGS

The test rigs used to measure the friction torque, depending on the purpose of the objects tested, are suitable for tests with axial or transverse loads applied. Due to the technical difficulties related to the construction of devices used for torque measurement with simultaneous application of both axial or transverse loads, devices able to perform such tests are rare. All devices used to measure the friction torque in rolling bearings have components necessary to perform the testing procedures on the given test rig. Fig. 1 shows a general breakdown of the components (subassemblies) that make up an industrial friction torque measurement device.

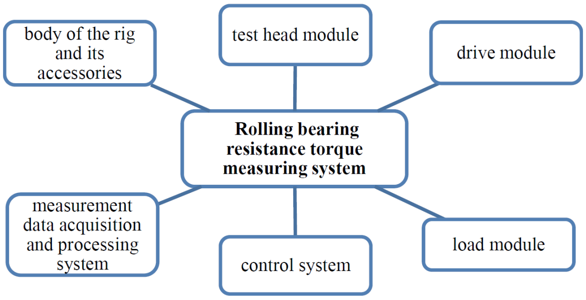


Fig. 1. *Design features of the measuring system used to test the friction torque of rolling bearings*

The test head must ensure repeatable testing conditions for the tested bearing. This is a precondition to ensure reliable comparison of results. The most dangerous situation for the node with the tested bearing is the introduction of interactions that go beyond the measurable load values in the directions controlled by the system supervising the testing process, especially the forces that may act in such a way that the bearing rings overlap each other. Solutions introduced by the testing equipment manufacturers are aimed at obtaining the best possible co-axiality and/or perpendicularity of the mated subassemblies and, where possible, to introduce a margin for connection flexibility to prevent excessive stiffness of the system. The key issue of the test head assembly design is the measuring method of the friction torque value generated by the tested bearing, especially that the measurement must be taken at different loads. The assembly transmitting the torque value to the measuring system must do so in a lossless manner or at least with a controlled loss included in the result obtained.

The spindle drive of the torque gauge should accurately reproduce the rotations set by the control system and keep them stable during the measurement. Manufacturers of torque-measuring devices typically use one of two spindle and test head power transmission systems. Some designs have the motor in line with the spindle, while other designs transmit the drive through a belt transmission. Each of the designs has its advantages and disadvantages. The advantage of placing the motor in line with the spindle is the possibility of reading the rotation of the tested bearing directly, especially if a servo drive is used as the drive unit. The introduction of a belt transmission, especially in applications requiring low revolutions and intended for measuring torques of high values, enables obtaining the required torque at lower engine power and significantly reduces the impact of its natural vibrations on the spindle and, consequently, on the test head. The selection of the motor depends on the implemented spindle speed control system.

The test load of the bearing during the test are generated, depending on its size, by gravity, a pneumatic system, or a hydraulic system. The design of the loading mechanism is to ensure that the axial (or radial) force at the value set by the test process control system is transferred to the tested bearing. Various designs of the loading systems are used, and their selection depends mainly on the value of applied force. When high value axial loads are required by the test procedure, the force is generally transmitted to the tested bearing via the spindle. When transverse loads are applied, the load is transferred to the bearing through its housing associated with the outer ring. Gravity systems can be used for a limited range of loads. Advantages of the systems based on gravity are the simple structure and high repeatability of the implementation of the set parameters. The disadvantage, however, is the problematic changing of load value and limited range of application. Systems based on pneumatic actuators are much more convenient to control but require more complex equipment and appropriate infrastructure.

Less frequently used hydraulic systems require a power station and are selected when high test loads are required.

The control and monitoring module is installed to ensure the safe operation of the device and to supervise the course of all processes carried out by the testing device. As far as the testing process carried out on the designed test rig is concerned, it is particularly important to maintain and record the parameters determining the conditions in which the measurements are taken. These include the rotational speed of the bearing and the magnitude of the applied load. The control system responsible for implementation of the testing process must effectively cooperate with the software and the driver's software by sending the information from monitoring and sensors. It must also properly react to commands resulting from procedures sent for execution. This control system is partially located directly on the test rig (sensors and amplifiers), and partially in the control cabinet equipped with an operation panel, together with the rig power supply equipment.

The test software is used to perform two basic functions. The first is transferring to the control and monitoring system the procedures necessary for the implementation of the test program, its execution and reacting to irregularities by correcting them or stopping the test rig. The second is collecting information about conducted measurements, their analysis, development, visualization of results, and preparing the test reports.

Like all measuring devices, the torque measuring device should be equipped with a stable body and anti-vibration feet. The body houses accessories such as media conditioning systems, power supply components, and control and measurement systems.

3 CONSTRUCTION OF A NEW DEVICE FOR MEASURING THE FRICTION TORQUE OF BEARINGS

The device that is the subject of this research, is one of the eight test systems built at Fabryka Lozysk Tocznych - Krasnik S.A. The new measuring practices that have been implemented allow for an improvement in the quality of the manufactured bearings and for the production of non-standard bearings, increasing the capacity of machines and mechanical devices. The new device allows controlling the friction torque: a parameter that has not been tested, as it has not been among the acceptance criteria for rolling bearings. The new device allows testing the friction torque of the bearings in any configuration of axial load and rotational speed.

For a more complete overview of the operational parameters of a tapered roller bearing, besides the knowledge of resistances that it generates at a certain axial load and rotational speed, it is important to know the changes of its mounting height. These changes are caused by the displacement of contact points of rolling elements with the ring raceways within the elastic limits of these elements under the influence of external loads imposed on the bearing. The mounting height of the tapered roller bearing, defined as the distance of the plane of the large section of the inner ring face from the plane of the large section of the outer ring face, is one of the basic parameters determining the correct mounting of a tapered roller bearing in a bearing node. It aids in determining and assuring the preload of the bearing, or possibly of the bearing unit when mounting them in a specific node, and also on the correct correction of these parameters when the node is in operation. The measurement of changes in the mounting height of tapered roller bearings as a function of load is an innovative concept for this class of industrial devices.

The design work performed allowed for the selection of 5 original design solutions. These solutions apply to both the entire test rig and its individual universal mechanisms, which can be used in other devices of a similar structure. The innovation of the presented test rig, which allows for:

- execution of cone bearing tests (similar devices are mainly used for ball bearings),
- increasing the number and range of test parameters,
- obtaining higher measurement accuracy.

The work on the bearing friction torque is important in terms of environmental protection, as striving to minimize frictional resistance increases the overall efficiency of the devices, thus reducing CO_2 emissions to the atmosphere.

It is not possible to identify a specific design principle. The starting point was the specific features and parameters that the device had to meet. The design process itself took several years. Many different solutions for individual nodes were considered, and the variant finally presented in the article was decided upon. The possibility to measure this parameter in the factory will result in significant technological progress, because the very appearance of the possibility of checking the friction torque is feedback for designers and technicians and a signal to increase the efficiency of rolling bearings.

The new device for measuring the friction torque of cone bearings is based on several plates, mounted on four columns. Some of these plates are fixed in place with clamping blocks, while others move on linear bearings. A pneumatic actuator located on a fixed top plate drives the module that houses a spindle and a drive. This movement enables coupling and decoupling the drive with the test rig accessories. Under the movable plate with the pneumatic table, there is a force gauge measuring the load applied to the bearing. The central part of this device is a rotary table supported with an air bearing. This solution ensures the rotation of the tabletop with minimal losses. A measuring table is attached to the tabletop. The mechanism located next to the pneumatic table contains two opposing force sensors. A clamping bar moves between them, which is rigidly connected to the measuring table. The tested bearing is mounted in the special test fixture intended for a given type of bearing. The inner ring of the bearing is seated in the lower part of the fixture, which is connected to the measuring table, while the outer ring is seated in the upper part of the fixture, which is coupled with the spindle during the test.

The rotation of the outer ring (driven from the outside) causes the rolling elements to roll on the raceways, and the friction inside the bearing causes the inner ring to rotate freely, attempting to spin the entire measuring table. The pressure bar presses the sensor with a force proportional to the frictional resistance caused by the bearing operation. Higher resistance of the tested bearing (e.g., poor workmanship or a factory defect) causes the greater force to be indicated by the force sensors. The critical level of friction torque generated by a given bearing type is defined by internal company standards or requirements imposed by the customer.

3.1 Measuring Table Assembly

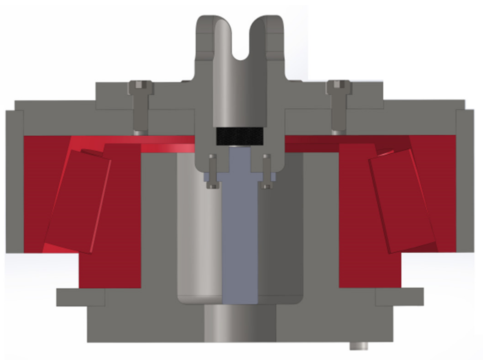
The measuring table assembly is the central part of the device where the actual measurement of the tested object is taken. The plate on which the measuring table assembly is placed is movable and is mounted on the columns by means of linear ball bearings, which enable it to move freely in the vertical direction. This is required to ensure the proper functioning of the device, because the pressure the plate generates allows the measurement of the force acting on the tested bearing. The pneumatic table on which the measuring table assembly with the tested bearing and the necessary accessories is mounted is designed not to resist the inner ring of the tested bearing during the test, thus allowing for a lossless measurement of the friction torque generated by the bearing loaded with high axial force, which is forced to rotate at the required rotational speed by means of the outer ring. A frame is mounted on the pneumatic table, allowing for the alignment and levelling of the measuring table assembly in relation to the loading-driving system. The lower part of the test fixture, having direct contact with the inner ring of the tested bearing, is connected to the frame by means of a steel pin. The outer ring of the bearing is located in the upper part of the fixture, which is coupled to the drive during the measurement. Due to this design, the rotation of the outer ring caused by the resistance created in the bearing causes the spontaneous rotation of the pneumatic table. The table has been adapted to test an extensive range of bearings. The measurement table assembly with the test accessories model is presented in Fig. 3. In addition to the measuring table assembly, a friction torque measurement subassembly is installed on the same plate, enabling the performance of the most important task for his device (Fig. 4). The friction torque is measured by means of two force gauges installed in opposite directions, which enables the measurements to be taken in both directions of rotation of the tested bearing. The pressure is transferred through the pressure bar attached to the base of the measuring table. It exerts pressure on one of the force gauges (depending on the rotation direction of the bearing) with a force proportional to the resistance of the tested bearing. The value of this force, multiplied by the arm length, which is the distance from the force gauge contact point with the measuring bar to the pneumatic table axis of rotation is the value that determines the friction torque of the tested bearing. The design of the subassembly protects the force gauges against overload, which could damage them if the tested object reaches a friction torque greater than the maximum assumed (e.g., as a result of bearing seizure). The protection is provided by a system of tilting arms on which force gauges and adjustable fenders are installed, limiting the possible movement of the pressure bar. The maximum pressure, and thus the load on the force gauges, controls the position of the overload protection spring.

An additional function of the device is the ability to measure the change in the width of the cone bearing caused by the applied load (displacement between the bearing rings in the axial direction). The displacement sensor positioning subassembly installed in the test zone allows monitoring changes in the height of the bearing as a function of the axial load during the test. The subassembly is fixed to the frame plate so as to be stationary in relation to its axis. Its cylindrical part is filled in the alignment hole and forms the basis for attaching the test accessories to the bearing. The subassembly is shown in Fig. 5. The design of the subassembly is based on an eddy-current sensor and allows measurements to be made with an accuracy of 1 μm. The worm gear that drives the sensor support screw is responsible for the precise positioning of the sensor. A sensor is mounted in the head of the load-bearing screw. The axial plays in the worm wheel and plays in the screw thread are removed by springs. By one full turn of the worm wheel, the sensor moves approximately 0.2 mm vertically. This ratio allows for precise zero adjustment of the sensor position in relation to the bearing outer ring before starting the test. The assembly is rigidly fixed to the bottom of the test accessories and detects displacement of the shaft connected with the top of the test accessories. The measuring shaft, the displacement of which is formally measured, is shown in the cross-section in Fig. 3a.

3.2 Force Gauge Relieving Lever Assembly

The plate under the measuring system plate is permanently attached to the torque-measuring unit columns by means of clamping blocks, to which it is fixed by means of a sleeve with a flange. The sleeves must ensure the hole and the outer diameter are in line so as not to introduce errors in the spacing of the holes of all plates that constitute the structure of the device. A precise force gauge is installed on this plate to measure the test load during the test. A ball encapsulated in a sleeve with a conical bore presses the dynamometer. This sleeve is screwed to the underside of the movable plate with the entire measuring system.

Under the plate, there is a lever mechanism to relieve the force gauge during breaks between tests and when the device is not powered. This subassembly is constructed in such a way as to enable the lifting of the table above the pressure gauge by means of a set of springs, which exert pressure on it with appropriate force by means of a circular cam and two pins. When the device is being operated, the pneumatic actuator counteracts the pressure of the springs and lowers the spindles, freeing the measuring table and allowing measurement of the pressure. Due to the large gear ratio, the spring package and the actuator operate in the range of forces ten times smaller than the mass of the table with the test accessories. The use of such



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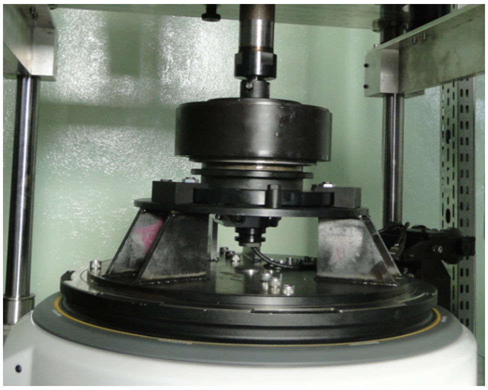

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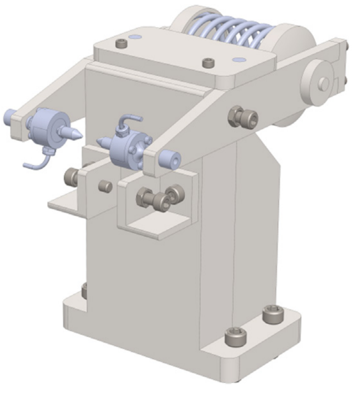
Fig. 3. *a) Cross-section of the measuring accessories assembly, and b) measuring table assembly*



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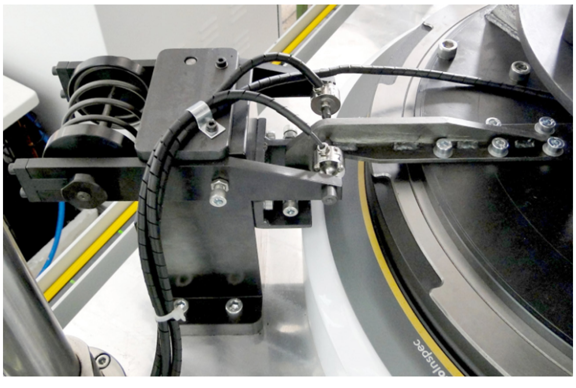

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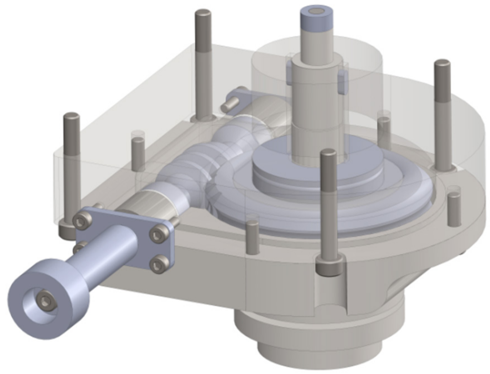
Fig. 4. *a) Model of the torque measuring unit, and b) measurement of the friction torque*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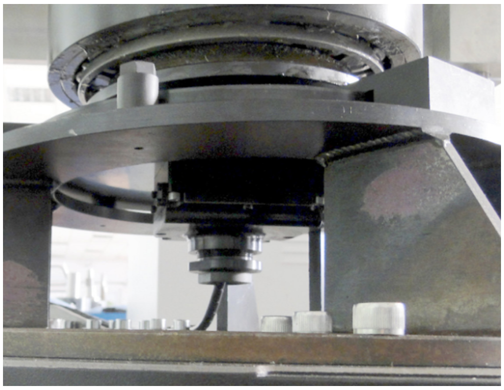
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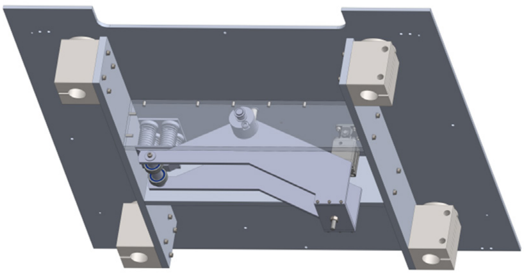
Fig. 5. *a) Model of the assembly height measuring unit, and b) location of the assembly height measuring unit*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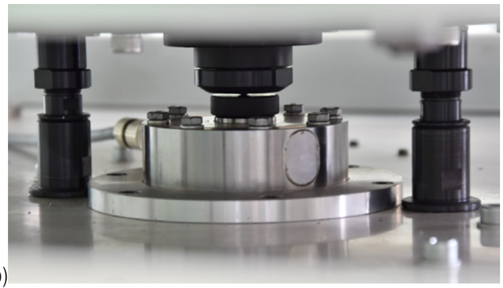
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Fig. 6. *a) Model of the lever relieving the force gauge, and b) measurement of the axial force*

a mechanism increases the durability and technical efficiency of the force gauge.

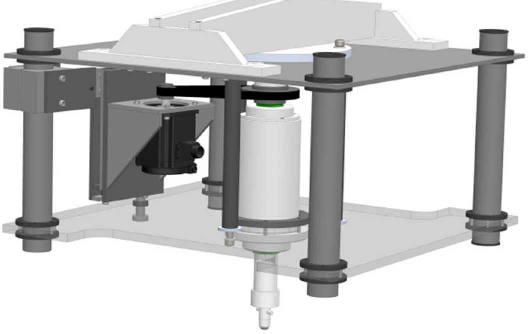
3.3 The Power and Load Transmission Unit

The power and load transmission unit plays a role of a complete power unit and an indirect load transmission unit from the pneumatic actuator. This unit has a form of a cage and includes a drive consisting of a spindle equipped with a motor and a belt transmission, which is simultaneously an intermediate element transmitting the load exerted by the pneumatic actuator to the tested bearing. Four tubes guide the assembly along the columns. They link three load-bearing posts with load-bearing plates, forming a compact integrated unit. The tube guides are supported by four pairs of linear ball bearings. Very important parameters determining the correct functioning of this unit are the equal length of the posts and the separation provided by the bearing tubes and, as in the case of all six supporting plates, the appropriate spacing of the holes leading to the columns. The power unit is fully assembled into the power and load transmission unit. The spindle, mounted in the centre of the bottom plate of the assembly, is inserted from the bottom and screwed with a ring attached from above.

The servomotor is suspended on a rotatable mounted support, which has a wide range of adjustment its position in relation to the spindle. The tensioner is mounted on a bearing tube with a block clamp. Fig. 7 shows the cage model and the spindle with the drive.

To ensure safety, the cage forming the power and load transmission unit is suspended on the relief assembly, which lifts it up and keeps it in a safe position in any emergency (power outage, compressed air outage, overload of the measuring system or emergency shutdown by the personnel). The general view of the assembly is shown in Fig. 8. The relief assembly consists of a set of four special spring shock absorbers.

The assembly is also suspended to a set of loads that is designed to apply the loads predetermined by the program to the tested bearing In addition to ensuring work safety, the relief assembly is designed to keep the spindle at a level that allows for the assembly and disassembly of bearing samples with the test accessories and to disengage the package of accessories with the sample when the test is finished or in emergency cases.



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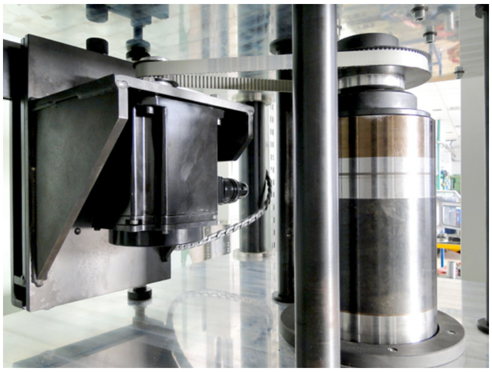

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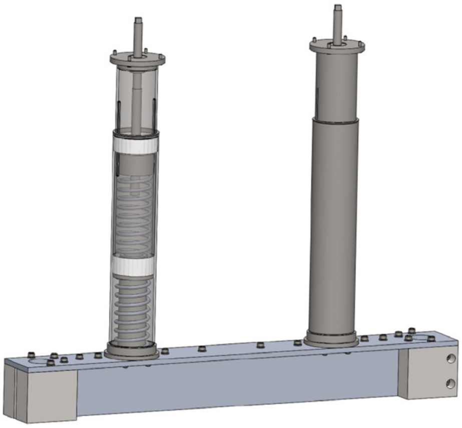
Fig. 7. *a) Model of the power and load transmission unit, and b) drive unit with the spindle*



a)

b)

Fig. 8. *a) Shock absorber model, and b) shock-absorbing suspension of the power and load transmission unit*

3.4 Load Assembly

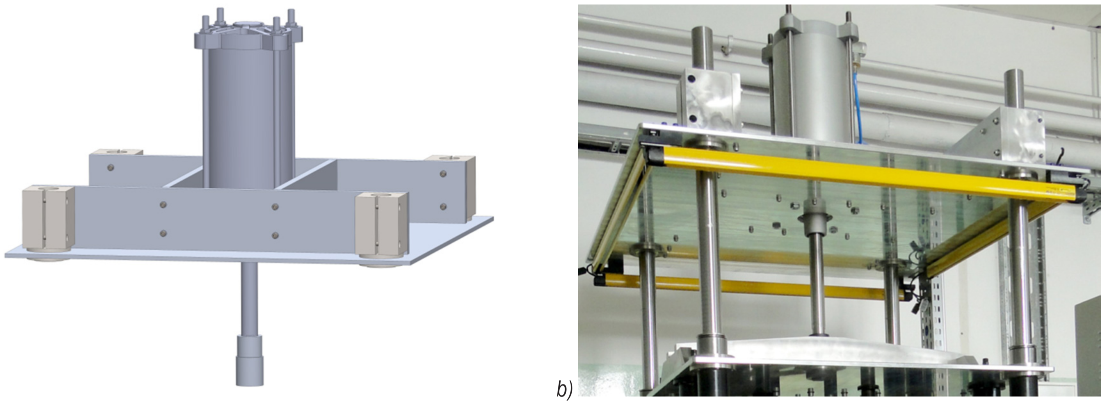
The last, upper support plate is permanently connected to the columns by means of clamping blocks similarly to the two lower support plates, The plate encapsulates the entire structure of the device from above, and at the same time, serves as a platform for mounting the pneumatic actuator and creates a set of axial loads with the cylinder. This unit is coupled with the structure of the power transmission system, so as to ensure the pressure exactly along the axis of the unit. Fig. 9 shows design of the assembly. In order to couple the spindle with the measuring system and set the required force, the actuator on the top plate must first overcome the force exerted by the shock absorber springs.

Additionally, in Fig. 9b, one can see a chain of light curtain panels. Appropriate pairs of the panels are attached to the main plate of the force gauge relieving lever assembly. Interruption of the invisible light beam flowing between the pairs of the curtain panels results in the automatic stopping of the upper actuator and lifting the power and load transmission unit mechanically.

4 TESTS AND TEST PROGRAM PREPARATION

After the first launch of the test rig tests were carried out during which the following activities were performed:

- The mass, force and displacement sensors were calibrated.
- Adjustments were made to the height setting of the clamps blocking the plates in order to provide as much space as possible for the replacement of the test accessories while using the greatest possible displacement of the upper actuator.
- The degree of parallelism of the stationary plates and the degree of concentricity of the central holes (in plates, measuring table, spindle shaft, etc.) were checked using the coordinate measuring device (mobile measuring arm).
- The operation of the mechanical units of the device was checked.



a) b)

Fig. 9. *a) Load assembly model, and b) location of the top plate with light curtains*

- The reaction of assemblies and executive subassemblies to commands of control systems sent by software and directly from the control panel was tested.
- The reaction of the safety systems to simulated threats was checked.
- The measurement sensors indications to the given loads were checked.

The activities performed above proved that the constructed test rig used for measuring the friction torque value in rolling bearings meets the initial assumptions and the general requirements for this device class. Fig. 10 show a direct screen view of the device software, which presents the measurement data on its panel in this manner. The graph is illustrative and is intended to enable the operator to quickly assess the repeatability of the results. However, the way the results shown in Fig. 10 is presented does not contribute much to their analysis in terms of evaluation of repeatability of the results obtained on a particular device. It only shows the behaviour of a specific bearing in response to changes to external parameters. In order to evaluate the repeatability of the test procedures, the results were compiled on Fig. 11 in such a way that the trend in their behaviour was visible as a function of subsequent measurements.

The task of the test program was to establish a forecast of the test results values. Therefore, the test program was prepared for a medium-sized bearing that can be tested for the maximum loads achieved on the rig. The obtained results were verified on the basis of friction torque measurement analysis for a cone roller bearing type 33213. The test program is presented below:

- The tests lasted one week; during one day two measurements of one bearing were taken, several hours apart. The tested bearing had a chance to return to the previous condition before the next test, which should help ensure the highest possible repeatability of the measurement conditions.
- Each measurement consisted of three stages, each of which had a different rotational speed: 50 rpm, 150 rpm, and 250 rpm. For each of these speeds, the axial force that loaded the bearing was changed four times: 200 daN, 500 daN, 800 daN, 1100 daN.
- Each measurement program was therefore executed as follows: after mounting the bearing in the housing and placing it on the device, the bearing was loaded with an initial force of 50 daN. The spindle then accelerated the bearing outer ring housing to 50 rpm. After reaching the set rotations, the bearing was loaded with a force of 200 daN. Once the torque value was stabilized, the measurement was taken. In the next step, the load was increased to 500 daN, and the measurement was taken again. The same steps were repeated for the load values of 800 daN and 1100 daN. An analogous sequence of measurements with different axial forces was executed after increasing the speed to 150 rpm, and then to 250 rpm. Once the test was completed, the bearing was removed from the housing.

5 THE RESULTS OF THE FRICTION TORQUE TEST

By using above methodology, a series of 10 friction torque measurement results were obtained for one cone roller bearing type 33213, at 12 different combinations of rotational speed and load. The example of obtained results presented in Fig. 10 refer to the one measurement while Fig. 11 refers to whole series of 10 measurements.

When analysing the results, it good to know the hypothetical value of the friction torque, which is calculated on the basis of Eqs. (1) to (3). By substituting the data on the 33213 bearing geometry, the viscosity of the lubricants used, the selected coefficients characteristic for the cone roller bearings, as well as the test parameters in the form of rotational speed and axial load, the theoretical values of the friction torque values presented in Table 2 were obtained. The calculated values are indicative, because the friction torque is very difficult to determine unambiguously by a theoretical calculation. This is mainly due to the fact that the coefficients f_0 and f_1 are determined empirically, and these are not unambiguous values but only a certain range of the very general three groups of lubricants. Secondly, the model described by Eqs. (1) to (3) is the basic model and one of the few models used for the calculations. As already mentioned, original formulas have been developed by SKF, FAG and Timken **[24]** to **[26]**, and the resulting predicted theoretical value of the resisting torque may differ significantly between the formulas. This is very well illustrated in **[11]**.

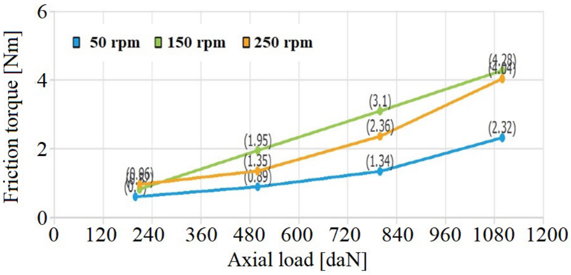


Fig. 10. *Example chart of the friction torque value changes as a function of load for various rotational speeds*

In addition to the measurement results, Table 2 shows the theoretical friction torque values for the bearing 33213. The viscosity of the oil used was assumed to be 220 mm²/s (based on the manufacturer's data). The analysis of the obtained data on the tested torque allows the following conclusions to be drawn:

Functions for a specified rotational speed value are corresponding (correlated). The only difference is the friction torque value, which always increases consistently when the load is increased. The exception is the function for the force of 1100 daN, which sometimes does not follow the trend set by the previous load values. Presumably, the bearing has inadequate operating conditions for such a high load if lubricated with a lubricant of relatively low viscosity. As a result, the friction torque behaviour is unstable.

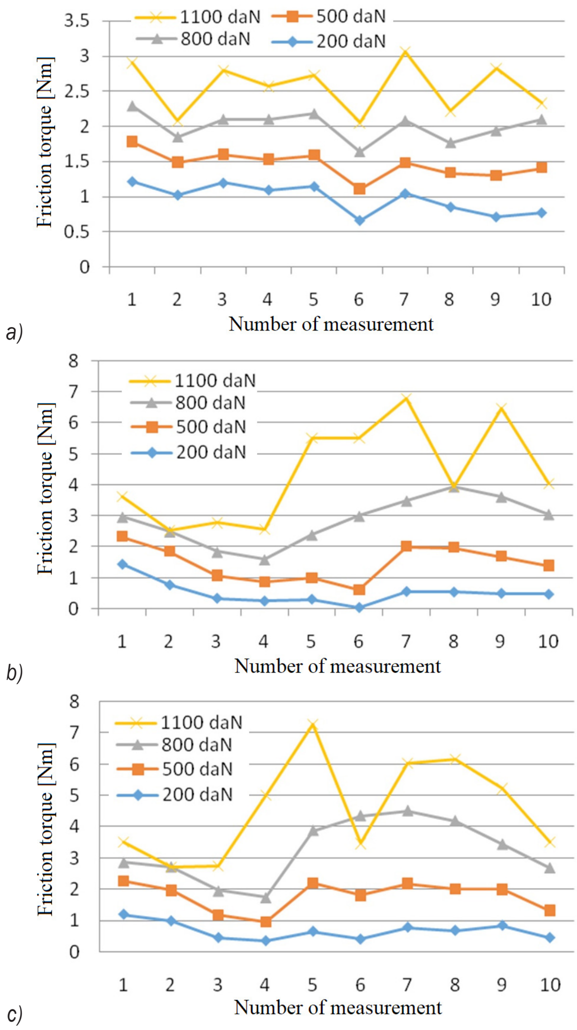


Fig. 11. *Bearing friction torque measurement series; a) rotational speed 50 rpm, b) rotational speed 150 rpm, and c) rotational speed 250 rpm*

The phenomenon of consistent duplication of the function curve when using the same rotational speed may also indicate the impact of the bearing fixture mounting method, incorrect zeroing of sensors or changes on of the bearing mating surface that occurred in previous tests (especially if the lubrication did not effectively fulfil its function).

The variabilities of the results are rising together with the rotational speed increase. The smallest variations were noticed for the rotational speed of 50 rpm. The measurement uncertainty depends mainly on the stability of the tested bearing, and not on the testing device itself. This is confirmed by the increase in the dispersion of the results along with the growing research parameters, creating more and more difficult conditions for maintaining the elasto-hydrodynamic film. The curves obtained during the tests show that the most stable operation of the bearing is at 50 rpm, so it can be roughly assumed that the dispersion obtained for this speed is the dispersion closest to that dispersion caused by the testing device itself.

Table 2. *Results of 10 bearing 33213 measurements at different combinations of test parameters, together with a comparison with theoretical values*

Rotational speed [rpm]	Axial load, [daN]	Theoretical friction torque, [Nm]	Friction torque of bearing 33213 in the presence of 220 mm²/s viscosity oil, [Nm]		
			Arithmetic mean	Root-mean-square deviation	maximum/minimum value
50	200	0.36	0.97	0.2	1.21/0.66
50	500	0.77	1.46	0.16	1.78/1.11
50	800	1.19	1.97	0.18	2.29/1.64
50	1100	1.6	2.56	0.36	3.06/2.05
150	200	0.44	0.54	0.21	1.46/0.06
150	500	0.86	1.49	0.52	2.33/0.62
150	800	1.27	2.83	0.8	3.93/1.6
150	1100	1.69	4.37	1.67	6.78/2.53
250	200	0.51	0.68	0.22	1.19/0.35
250	500	0.92	1.78	0.47	2.26/0.95
250	800	1.34	3.23	1.04	4.51/1.74
250	1100	1.76	4.57	1.65	7.28/2.72

In conclusion, the variability of the result of the rolling bearing friction torque measurement seems to be rational in a significant part of the obtained data. However, in order to finally decide on the reliability of the obtained results, the experiments should be repeated. This will be particularly important after adjustments have been made to the alignment or design corrections have been made.

6 CONCLUSIONS AND RECOMMENDATIONS

The performed tests showed that the obtained accuracy of the performed measurements meets the requirements of the assumptions adopted at the test rig designing stage.

The measures include:

- additional calibration of measuring sensors,
- conducting a cycle of tests aimed at optimizing the methodology of sensors zeroing before starting the measurement process,
- carrying out tests using various lubricants in order to optimize their selection for the bearing operating conditions assumed for the test,
- checking whether the function for specific rotational speeds is consistently reproduced at various loads (without dismantling of the bearing),
- performing tests on a series of bearings of different sizes and on a larger number of samples,
- checking the possibility of measuring the friction torque on a bearing with the outer ring slidably mounted.

Finally, it is worth paying attention to the level of values obtained in the test. The torque values obtained empirically are consistently greater than those obtained through calculation. This is obviously due to not taking into account all the factors affecting the bearing friction torque, such as cage friction, the geometric structure of the mating surfaces, the dimensional and shape accuracy of the bearing elements or even the lubricant purity.

Future experimental works should be aimed at determining as many factors as possible influencing the result of the friction torque measurement, simultaneously being aware that some of these factors will be caused by the measuring device itself, which is very often overlooked. Numerical simulations of the friction torque result of tapered roller bearings may also prove useful in further device evaluation studies **[27]**.

From an academic point of view, the most important thing is that the innovative solutions within the presented devices have been automatically implemented in industrial conditions as a prototype of a stand for inspecting newly manufactured bearings. The design work carried out has therefore contributed to increasing the base with new solutions for science and technology. In addition, many of the mechanisms are universal (lever, positioning mechanism, damper, pressure) and can be applied to other devices of this type.

It could also be important to investigate the influence of dynamics on the result of friction torque measurement. Eq. (2) cited in Chapter 2 suggests that as long as the quotient $v \cdot n$ is less than or equal to 2000, the friction torque does not depend on the rotational speed (no influence of dynamics on the measurement result). In other cases, the resisting torque is a function of the rotational speed (dynamics effects the measurement result).

However, measurement practice shows that the influence of dynamics can be greater even for low ranges of the product $v \cdot n$. In addition, the variability of the result over time under unchanging measurement conditions should also be investigated, as well as the reproducibility of the friction torque result.

The most difficult problem, however, is to assess the accuracy of the friction torque measuring device.

There are no standards with known friction torque. There are also no reference devices for measuring friction torque that can provide a reference for other devices. At the current level, the only possibility is to assess the accuracy of force sensors that directly measure the bearing's resistant force. From a metrological point of view, however, this is not a satisfactory solution, as the accuracy is affected by additional factors such as resistance or the rigidity of other components of the measuring system.

Further work should therefore focus on estimating uncertainty in the measurement of the friction torque, attempts to develop bearing standards with a known friction torque and comparative procedures of various devices and design solutions to select a reference stand.

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