ISSN (Print) 1226-1750
ISSN (Online) 2233-6656
Journal of Magnetics 25(1), 48-55 (2020) https://doi.org/10.4283/JMAG.2020.25.1.048

Experimental Studies of Magnetic Fluid Seals and Their Influence on Rolling Bearings

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(Received 28 November 2019, Received in final form 27 February 2020, Accepted 10 March 2020)

There are many seals designed for rolling bearings. One interesting perspective is the application of magnetic fluid seals for this purpose, because they are an alternative to commonly-used solutions and combine the advantages of contactless seals (e.g. labyrinths) and contact seals (e.g. rubber lip seals), therefore they have low resistance moment and high tightness. This publication describes rolling bearings and a magnetic fluid seal system in terms of the maximum sealing pressure, which was compared to the results of numerical simulations, and the influence of the magnetic field on the bearing torque was also investigated. The tests were carried out for various types of seal stage shapes. Different magnetic circuit configurations have been considered i.e. the main magnetic flux passing through the seals or rolling bearings (steel or ceramic). Studies have shown that a magnetic field increases the torque in the bearing and magnetic fluid seals may reduce the torque because part of the magnetic flux passes through the seal elements. This result is important because it shows that it's possible to reduce the negative effects of variable magnetic fields on rolling bearings by using additional elements of the magnetic circuit.

Keywords : magnetic fluid seal, ferrofluid, rolling bearing, eddy currents

1. Introduction

There are many types of rolling bearings and they play a key role in the functioning of many machines. They can work under different conditions and modern rolling bearings can reach rotational speeds of up to 20000 1/s [1]. Working at high rotational speeds and high loads is related to the occurrence of friction, which generates heat [2]. This affects the thermal deformation of the bearing elements [3] and their wear [4]. In order to increase their durability, various lubricants, such as plastic lubricants [5] or gear oils [6] are used. Suitable selection of the seal is also a key element that affects durability, because its purpose is to prevent the entrance of impurities and moisture into the region of rolling elements or to prevent the leakage of grease from shielded rolling bearings [7].

An interesting idea is the use of magnetic fluids (ferrofluids) both in seals for rolling bearings or as lubricants [8]. A magnetic fluid is a substance consisting of nanoparticles with magnetic properties, which are coated with a surface-active compound to prevent their agglomeration. A carrier liquid has no magnetic properties and examples include water and oil. A magnetic fluid has unique properties and changes its rheological properties under the influence of a magnetic field [9]. It can also be maintained at a selected location in the region where the magnetic field gradient occurs, with this property mainly being used in seals. The advantages of this seal are low friction torque, high tightness and durability. The principle of operation is based on the creation of a liquid barrier and they are used primarily in vacuum technology. However, they are becoming more widely used in other applications, such as valves [10], miniature seals in water environments [11] or fast-rotating devices [12]. The seals that are currently available are usually drive units, i.e. complete, assembled units containing: a body, magnetic fluid seal, rolling bearings and a shaft or sleeve, which are connected to moving parts of machines or devices by couplings. This means that such solutions often require a relatively large assembly space.

Magnetic fluid seals may be smaller and have dimensions similar to typical seals and these constructions are mainly intended for rolling bearings. There are types mounted directly in the outer ring [13]. However, this requires a

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special type of bearing, which must be wider. Another type is a construction mounted next to the bearing [14] and, in this case, there are more design solutions. Application examples are magnetic fluid seals for rolling bearings in hard disks (HDD) or fans for cooling electric motors [15]. Their aim is to protect against dust particles, impurities and humidity. These constructions are relatively simple and usually consist of a permanent magnet and pole piece with a one-seal stage on which the ferrofluid is placed.

It should be noted that there are many design solutions that have not been tested thoroughly and are only in the form of patents. There are also few test results in publications describing the interaction between the rolling bearing and magnetic fluid seal.

One of the main objectives of the publication was to determine the maximum seal pressure (critical pressure -p_{CR}), after which there is seal failure and leakage. The aim of the research was to determine the difference in critical pressure between the measurements and when calculating this parameter by using simulation results.

The second key objective was to determine the influence of the magnetic field coming from the seal on the rolling bearing. The negative effects of a variable magnetic field are known because, in some modern drive systems, the rolling bearings of electric machines may be damaged within a few months. This phenomenon is often a result of electrical discharge currents in bearings, which are induced in the motor due to the magnetic flux asymmetries in the motor, asymmetrical, non-shielded cabling, rotor eccentricity or fast-switching frequency converters [16]. This phenomenon is known as electrical discharge machining (EDM), and the currents are associated with a modulated drive or e.g. residual currents from electric locomotives [17]. Damage caused by electrical erosion is revealed as micro craters, which leads to micro cratering known as fluting [18]. One solution to this problem is grease with high electrical conductivity but, in this case, for high frequencies this grease is the cause of a capacitor creation. When the voltage reaches a certain limit, the capacitor is discharged and a capacitive discharge current occurs. Another solution is to use electrically insulated bearings coated by an aluminum oxide layer which acts as a resistor and provides insulation against electric currents.

In the case of magnetic fluid seals, the magnetic field source in bearings is a permanent magnet. In the absence of movement, the magnetic field has a constant value and affects only the mutual attraction forces of ferromagnetic elements. In the case of rotational velocity, a variable magnetic field appears due to the movement of the elements in the bearing. This magnetic field generates eddy currents, which in turn generate an induced magnetic field that counteracts the primary magnetic field changes. There are currently no specific scientific studies confirming the harmful effect of eddy currents on bearing performance. This phenomenon definitely affects the rolling bearing's torque. This is important in the case of precision mechanics devices or measuring devices with rotating elements, because more and more devices are currently being manufactured in which we have to take into account friction losses in rolling bearings and seals. One example could be sealing applications used in the aerospace industry and by the military in optoelectronic devices. In this work, the magnetic field influence on a rolling bearing was measured as the torque change of the rolling bearing.

2. Research Stand and Methodology

The scheme of the test stand is shown in Fig. 1. The main components are: a rolling bearing (pos. 8) mounted on a pivot (pos. 12), a shield (pos. 10) with magnetic fluid (pos. 11). The magnetic field source is a permanent magnet N42 (pos. 3) polarized in the axial direction.

In the system, there are two main magnetic fluxes Φ1 and Φ2. The first magnetic flux passes through the first magnetic pole (pos. 7), then the shield, the magnetic fluid, the pivot, an air gap and the second magnetic pole (pos. 4). The flux Φ2 passes through the magnetic poles, the sleeve (pos. 9), the bearing, the pivot and an air gap. Some elements (pos. 1) are non-magnetic. All parts are placed in the housing (pos. 2), which is also non-magnetic. In order to ensure temperature stabilization during the study, a coolant at a suitable temperature flows through the channels (pos. 6).

A rolling bearing 6204 without seals and grease was chosen with the following dimensions: inside diameter: 20 mm, outside diameter: 47 mm, width: 14 mm. Steel and ceramic bearings were selected. Due to the testing at different temperatures, a steel bearing with clearance C3 (bigger than normal) was selected.

The sleeve (pos. 9) in some tests was made of steel (ferromagnetic material) and in others of aluminum (paramagnetic material). In the case of the ceramic bearing, only a steel sleeve was used. The sleeve thickness was 1.5 mm.

Three magnet volumes were used in the studies. The volumes V_1, V_2 and V_3 for the aluminum sleeve correspond respectively to the constant (DC) magnetic induction values: 11.9, 26.9 and 46.6 mT. For the steel sleeve, this value is: 9.2, 21.9 and 41.8 mT. These values were measured in places shown in Fig. 1(e) by a hall transverse probe (pos. 13). The lower value in the case of the steel

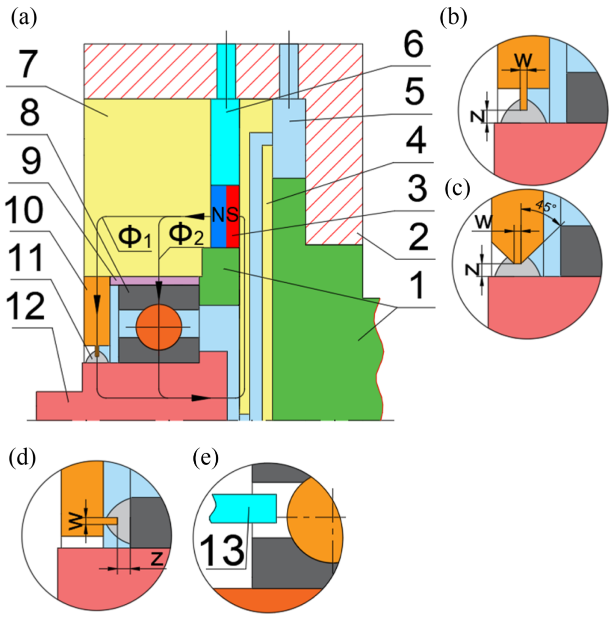

Fig. 1. (Color online) Elements of the test stand.

sleeve is a result of the fact that the magnetic flux is closed more through the rolling bearing elements and the measurement was made on the outside of the bearing.

Three shields with different seal stages were used. The first stage (Fig. 1(b)) has a rectangular shape and z = 0.1 mm, w = 0.6 mm, the second (Fig. 1(c)) has a trapezoidal shape and z = 0.1 mm, w = 0.6 mm, the third (Fig. 1(d)) has a rectangular shape and z = 0.3 mm, w = 0.6 mm, but is directed to the inner ring of the bearing.

The critical pressure tests were carried out for three shields, two rolling bearings and two sleeve materials. This parameter was measured in such a way that air (pos. 5) was slowly increased in the test chamber until there was a sudden break in the magnetic fluid continuity. The pivot rotational speed was zero.

Other tests involved torque measurement. For this purpose, a torque transducer was connected to a pivot (pos. 12) by using a clutch that was made of non-conductive current and non-magnetic material. The rotational speed was changed during 300 s in the range 0 to 50 1/s, which corresponds to 2.6 m/s linear velocity. The aim of the research was to determine the torque change for different shield variants and to determine how the magnetic flux influences the rolling bearing torque.

One magnetic fluid was selected for testing purposes. This fluid has the saturation magnetization 30.66 kA/m, density 1.299 g/ml, and dynamic viscosity 0.5 Pas at 25 °C. The volume of the magnetic fluid in the seal in all tests was 25 μl.

3. Numarical Simuation Results

Numerical simulations were performed to determine the magnetic field distribution according to the method described in a previous publication [19].

Numerical simulations are very convenient as they allow the value of the magnetic field in the seal gap region to be determined, an appropriate magnetic field source to be selected or the seal structure to be optimized [20]. It should be noted that the rolling bearing does not meet the axially symmetrical condition, but in order to reduce computation time this model was adopted [21]. This is the most commonly used model for the numerical calculation of magnetic fluid seals.

In order to conduct numerical simulations, the ANSYS 19 program was used. Zero boundary conditions of Dirichlet were introduced, i.e. it was assumed that the direction of the magnetic flux was parallel to the edges of the analyzed area. The mesh size was concentrated in the seal gap area.

In the case of ferromagnetic elements, a B-H magnetization curve, like for steel, was assumed. In the case of magnetic circuit elements, such as air, aluminum and magnetic fluid, the relative magnetic permeability was 1. The distributions of magnetic field in the seal gap region for different sleeves and shields, magnet volume V_2 are shown in Fig. 2, where (R) refers to the rectangular stage, (T) to the trapezoidal and (RB) to the rectangular stage directed to an inner ring. The results are not presented for the (T), (R) and (RB)-steel sleeve, because, according to the simulations, the main magnetic flux passes through the rolling bearing and the maximum magnetic induction value in the seal gap in this case does not exceed 0.1 T. A similar value is obtained in (RB) and the ceramic bearing

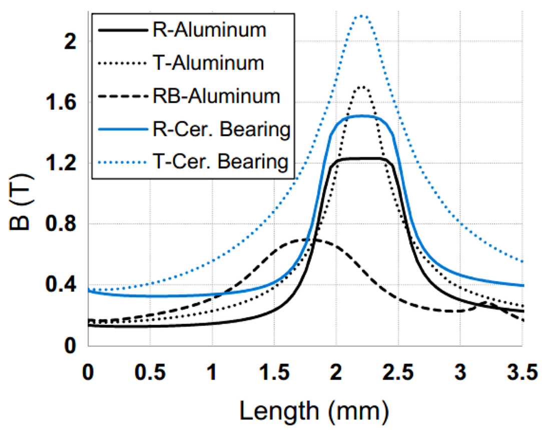

Fig. 2. (Color online) The magnetic induction in the seal gap for the steel and ceramic bearings, different sleeve materials and different seal shields.

due to the non-magnetic properties of the ceramic material.

The results of numerical analyses can be successfully used to calculate critical pressure in the case of a one-stage seal. The difference in this case compared to experiments generally does not exceed 15 %. The reason for the inaccuracy is mainly because of the magnetic circuit element properties and the ideal geometry adopted for simulations. In most cases, the simplified model for calculating this pressure takes the form:

$$\Delta p_{CR} = M_S \Delta B_{\max-\min} \qquad (1)$$

The value is determined mainly by the saturation magnetization (M_S) of the ferrofluid and the magnetic induction difference between ($B_{\max}$) and ($B_{\min}$). These magnetic induction values are located on the two sides of the ferrofluid ring [23]. The simplifications of the equation describing critical pressure are: the omission of the influence of the gravitational field, and surface tension forces. Another condition is that the magnetic fluid is in a state of magnetic saturation, but this is fulfilled in most cases. This formula refers to static cases, but can be used with sufficient accuracy to calculate the critical pressure for seals that have a linear velocity below 10 m/s.

It should be noted that the rolling bearing is not subjected to radial force. The bearing is loaded only with axial force which is the result of the magnetic attraction of the pivot to the magnetic pole. Based on the simulation results for different rolling bearing variants, the value is about 200 N. In this case, a steel bearing is loaded as P/C = 0.015, where P is the load applied to the bearing and C is the dynamic load rating. As a general rule, the minimum load on ball bearings should be 0.01C. The lack of a dominant factor, in this case the load, leads to an increase in the percentage impact on the movement resistance of other factors, such as: rotational speed, oil viscosity, ambient temperature or clearance in the bearing.

4. Critical Pressure Test Results

The results of critical pressure measurements for the temperature 25 °C and magnet volume V_2 are shown in Fig. 3(a). In the case of a steel bearing and aluminum sleeve, higher values are obtained for the trapezoidal (about 12 %) than for the rectangular stage. It should be noted that, when we compare these results when a steel sleeve is mounted, p_{CR} does not differ by more than 5 %. This is contrary to the simulation result, where the value of magnetic induction is several times smaller. An increase (about 25 %) in the critical pressure value is observed for the (RB) shield when the sleeve material is changed from aluminum to steel. The ceramic bearing, due to its non-

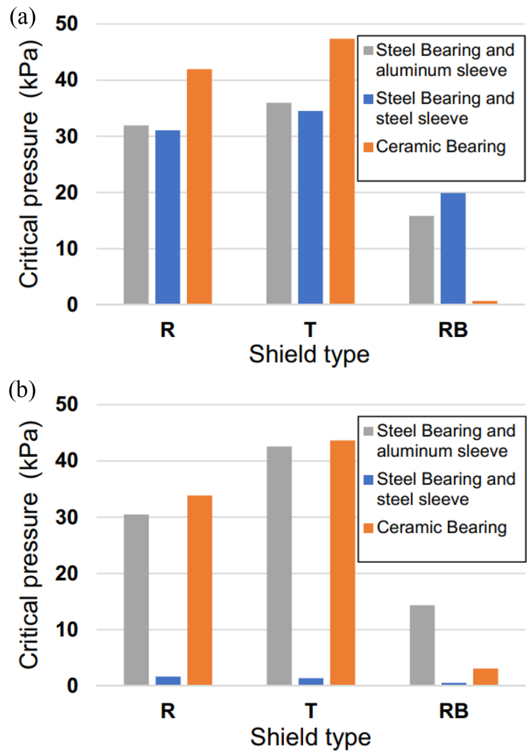

Fig. 3. (Color online) Critical pressure values for steel and ceramic bearings, different sleeve materials and different shields: (a) measurement result, (b) simulation result.

magnetic properties, increases the magnetic flux in the seal gap region and this translated into an increase in the value of p_{CR} by about 31 % for the (R) shield and the (T) shield.

The critical pressure was calculated by using equation (1) and the results are shown in Fig. 3(b). When comparing differences between the measurement and simulation, the results for the steel bearing and the aluminum sleeve (R, T, and RB shields) and also for the ceramic bearing (R and T shields) are not greater than 19 %.

An interesting result was obtained in the case of the ceramic bearing and the (RB) shield. Despite the fact that the bearing had non-magnetic properties, a liquid ring was created. Tests have shown that the critical pressure value was only 0.68 kPa, compared to the simulation results of 3.06 kPa.

In the case of the steel bearing and steel sleeve, for all cases, magnetic induction less than 0.1T was obtained. The reason for this was that the axially symmetrical model adopted for the simulation meant that the rolling elements were not a ball, but a ring. This caused the main magnetic flux to close through the bearing, not the seal. Therefore, for all cases, the p_{CR} value was below 1.6 kPa. The bearing could be modeled as a 3D element, but the problem is the small seal gap value (0.1 mm) compared to the dimensions of the entire seal. This means that the model has a large number of finite elements and this increases the calculation time. Simulations of seal 3D models have been used so far only with the shaft eccentricity being taken into account [22, 23].

5. Torque Test Results

In the case of research, a torque transducer with an operating range of 2 Nm (TM305-Magtrol) was chosen. Accuracy class < 0.1 % causes that the value of measurement error was about 2 Nmm. Each experiment was conducted at least three times to check if the results are repeatable. The charts show the average values. The torque results in the case of the steel bearing, two sleeve materials, two temperatures (25 °C and 60 °C) and magnet volume V_2 are shown in Fig. 4. Higher torque values are obtained for the steel sleeve, due to the fact that the magnetic flux Φ2 increases in this case and, by magnetizing steel elements, higher force occurs between two nearby surfaces. An important trend in all studies occurs. For almost all tests, we obtain a higher value of torque in the case of bearings without sealings. In turn, the torque friction coming from the magnetic fluid seal should add up to the bearing torque. According to calculations based on the equations described in [24], the seal torque for the rotational speed 50 1/s should be about 2 Nmm. The observed phenomenon is the result of reducing the magnetic flux Φ2 in the rolling bearing because of the magnetic fluid seal. This effect can be explained as follows; the value of the magnetic field influences the force of mutual attraction of bearing elements and when the magnetic flux value decreases, eddy currents also decrease and thus drag force on the moving elements of the rolling bearing is lower. A bigger difference appears at 60 °C. The highest torque decrease for the aluminum sleeve is observed for the (T) shield (for v = 2.6 m/s, it was 23 %) and for the steel sleeve for the (RB) shield (for v = 2.6 m/s, it was 25 %).

Confirmation that the magnetic fluid seal presence reduces the magnetic flux Φ2 and thus the torque in the rolling bearing can be seen in Fig. 5(a). Measurements in this case refer to the ceramic bearing (Ceramic B.) and (T) shield for two temperatures (25 °C and 60 °C). In this case, no differences are observed between results.

The results indicate that the temperature influence is a

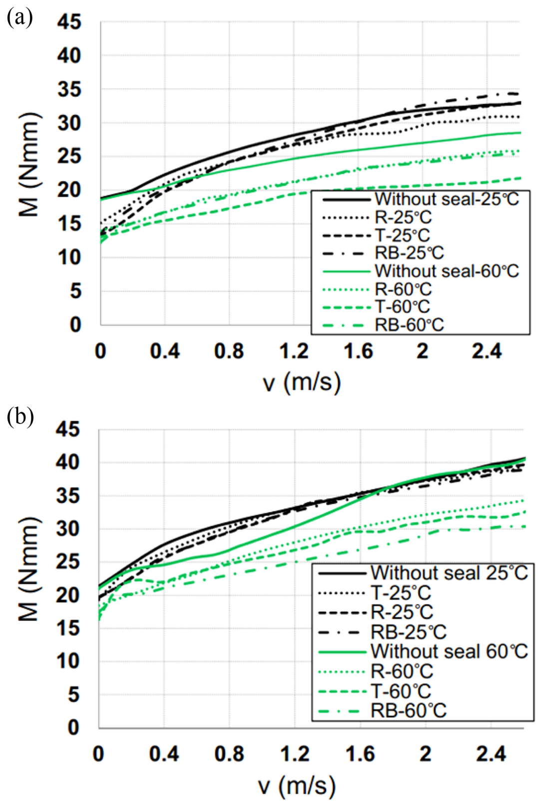

Fig. 4. (Color online) Torque vs. linear velocity for two temperatures: (a) aluminum sleeve, (b) steel sleeve.

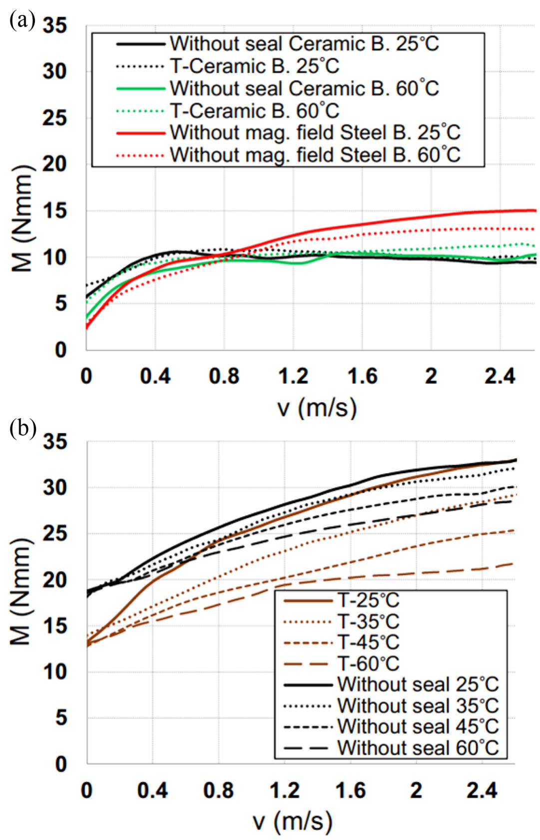

Fig. 5. (Color online) Torque vs. linear velocity: (a) two temperatures, the (T) shield and ceramic bearing and steel bearing without magnetic field, (b) four temperatures.

key issue. Tests for four temperatures and the (T) shield and aluminum sleeve are shown in Fig. 5(b). A noticeable torque decrease is observed from 35 °C and for v = 2.6 m/s it is 11 %, and 24 % for 60 °C. The decrease in the torque value in the magnetic field with increasing temperature presented in Fig. 5(b) may be due to the fact that the temperature rise causes a decrease in the electrical conductivity. This in turn causes a reduction in eddy currents and thus torque.

A summary of the torque values for two temperature, linear velocity v = 2.6 m/s and for different configurations the steel rolling bearing-sleeve is shown in Fig. 6.

In the case of the steel bearing without a magnetic field, some torque also decreases due to the temperature increase. The temperature rise from 25 °C to 60 °C means that the torque decreases by about 2 Nmm. The reason for this is mainly related to a sliding friction reduction which is larger than normal, because no lubricant is present in the bearing. As the temperature rises, the clearance in the bearing decreases and thus increases rolling friction, which causes bearing torque reduction.

From the results, it can also be seen that use of the shield (T) for the temperature 25 °C did not decrease the torque, while this value decreased at 60 °C. This could lead to the conclusion that this trend is the result only of a reduction in eddy currents due to the decrease in conductivity. Another possible reason may be related to the seal gap height decrease, as temperature increases. This affects the value of the magnetic flux Φ1. The seal gap height after assembly differed from the nominal one by about 0.02 mm on the circumference, mainly due to manufacturing inaccuracies. This parameter was measured for a non-rotating pivot and 25 °C. Measurement of this was made with a feeler gauge. To confirm this, more

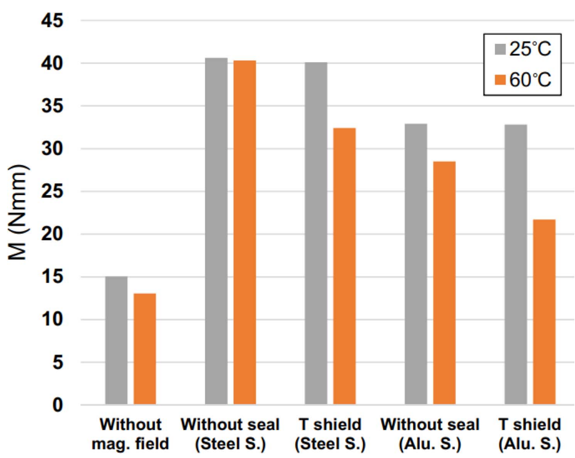

Fig. 6. (Color online) Torque for the steel bearings, different sleeve materials and the (T) shield.

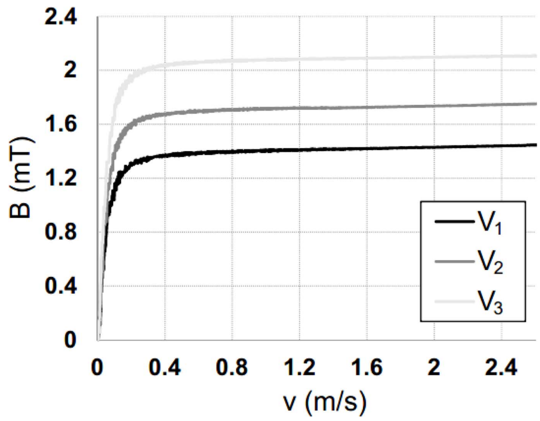

Fig. 7. Magnetic induction RMS value for different magnet volumes and aluminum sleeve.

accurate gap height testing would be required, especially during pivot rotation. However, this was difficult due to the measuring method.

As further research has shown, the value of the magnetic field is also important here. In the next stage, the

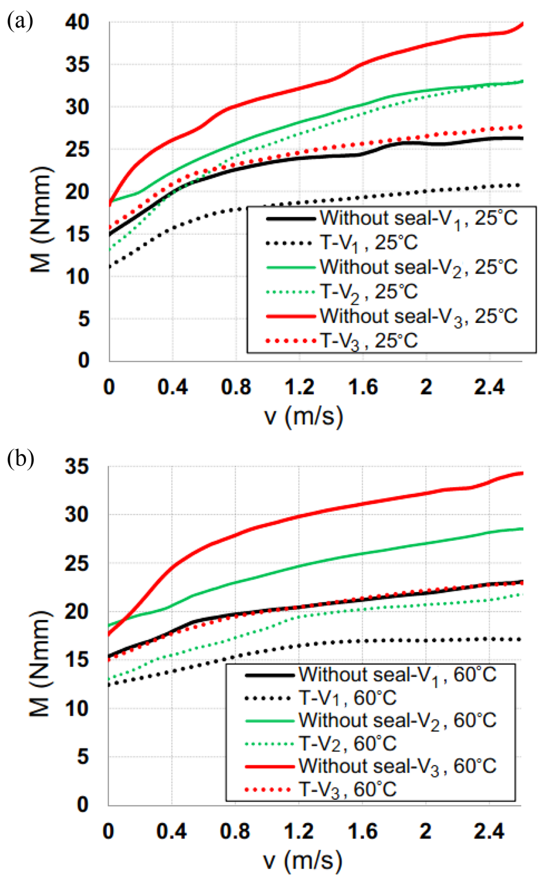

Fig. 8. (Color online) Torque vs. linear velocity for different magnetic fields, aluminum sleeve: (a) Temperature 25 °C, (b) Temperature 60 °C.

influence of the magnetic field value (magnet volume V_1, V_2 and V_3) is investigated. In this case, the AC magnetic induction value is measured and the RMS parameter as a function of velocity for three magnet volumes is shown in Fig. 7. These values are measured for an aluminum sleeve at the place shown in Fig. 1(e) by a hall transverse probe (pos. 13). Similar trends but lower values are observed for the steel sleeve. In the velocity range 0-0.5 m/s, a fast increase in the magnetic induction is observed and then the value stabilizes. The increase in the magnet volume from V_1 to V_2 for v = 2.6 m/s increases the value by 21 %, and in the case of V_3 by 45 %. Magnetic induction was also carried out for various temperatures in the range from 25 °C to 60 °C and the difference between the results was no more than 2 %.

The results which relate to the aluminum sleeve and (T) shield at two temperatures are shown in Fig. 8. The increase in the magnet volume leads to a torque difference between the application of the magnetic fluid seal, and using only the bearing increases this. When the shield is present, the torque decrease is: V_1-20 %, V_3-30 % (for 25 °C) and V_1-25 %, V_3-33 % (for 60 °C). In case of tests not presented in the publication for the steel sleeve, torque decrease was only: V_1-4 %, V_3-13 % (for 25 °C).

6. Conclusions

Magnetic fluid seals can be successfully used as protective seals for rolling bearings. They can be an alternative to other seals due to their simple construction. The solution from Fig. 1(d) is particularly interesting because the gap height z does not depend on the radial run out.

The results of numerical simulations based on the axisymmetric model can be used for design in the case of magnetic fluid seals for rolling bearings. When modeling a rolling bearing, it should be assumed that it is made of a material with non-magnetic properties. Modeling the rolling bearing as an axisymmetric element and as a ferromagnetic material means that most of the magnetic flux would pass through this element. In real bearings, this is not always the case. This is mainly due to the fact that the contact between the rolling elements and inner and outer rings is through concentrated contacts.

The magnetic field, which is an integral part of the seal, affects the torque in the rolling bearing. In the case of the conducted tests, additional rotation resistance resulted from the formation of eddy currents and mutual attraction forces of ferromagnetic elements. Elements of the seal and, in this case, the shield can reduce this effect, due to directing part of the magnetic flux in the region of the seal gap. The torque reduction in the bearing is larger with temperature rises and, additionally, this effect could probably also be increased by using more than one seal stage. However, this phenomenon was not observed in all cases, because this is influenced by the value of the magnetic field and the conditions at which eddy currents are created (e.g. conductivity). The smallest effect was observed in the case of the steel rolling bearing and steel sleeve, because in this case the magnetic flux passing through the bearing is the highest and the sealing gap is a high magnetic resistance region in the magnetic circuit.

One of the main advantages of magnetic fluid seals is their low friction moment and high tightness. The results show that the friction can be 2-3 times greater when the magnetic field affects the rolling bearing. If the values are compared with other types of seals, it turns out that, for example, 40.3 Nmm torque (steel sleeve, 25 °C, v = 2.6 m/s, T shield) is a high value. Calculating the theoretical values for typical seals at the same operating parameters and including bearing torque (15 Nmm) we obtain: Radial shaft seal - 54.8 Nmm, NBR Contact seal on one side (RSH) - 42.8 Nmm, NBR Low-friction seal on one side (RSL) - 16.6 Nmm. These results show that, when designing magnetic fluid seals, the main magnetic flux passing through the bearing reduces their main advantage.

Measurements using a probe showed the presence of a variable magnetic field in the bearing. The percentage increase in the RMS magnetic induction value for different magnet volumes corresponds approximately to the increase in torque (for aluminum sleeve). When we consider the rolling bearing itself and v = 2.6 m/s, the increase from V_1 to V_2 corresponds to the RMS value increase by 21 %, and the increase from V_2 to V_3 is 20 %. In the case of the results from Fig. 8(a), torque increase is, respectively: 18 % and 21 %, and for Fig. 8(b) it is: 24 % and 20 %. For further research a better place for measuring magnetic induction would be the air gap between the pivot and second magnetic pole, because the influence of the seal shield on the magnetic flux could also be measured.

Acknowledgements

This work is financed by AGH University of Science and Technology, Faculty of Mechanical Engineering and Robotics, research program No. 16.16.130.942.

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