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Research Article

Experimental study on the feasibility of alternative materials for tilting pad thrust bearings operating in transition to mixed friction

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Abstract: In hydrodynamic bearings traditional bearing alloys: Babbitts and bronzes are most frequently utilized. Polymer sliding layers are sometimes applied as a valuable alternative. Hard diamond-like carbon (DLC) coatings, which are also considered for certain applications may show some advantages, as well. Although material selection is of secondary importance in a full film lubrication regime it becomes important in mixed friction conditions, which is crucial for bearings with frequent starts and stops. Experimental research aimed at studying the performance of fluid film bearings in the specific operating regime, including the transition to mixed friction, is described in the paper. The tests were carried out on four tilting pad bearings of different material compositions: Steel/bronze, DLC/steel, steel/polyether ether ketone (PEEK), and steel/Babbitt. The tests comprised stopping under load and reproduction of the Stribeck curve by decreasing rotational speed to very low values, and observing the changes of friction force during the transition to mixed friction regime. Analysis of the transition conditions and other results showed clear differences between the tested bearings, illustrating the feasibility of less popular material compositions for bearings operating in specific conditions. More specifically, the DLC/steel bearing was demonstrating superior performance, i.e. lower friction, transition to mixed friction occurring at higher load, and more stable performance at start-stop regime over the other tested bearings.

Keywords: tilting pad thrust bearing; experimental research; start-stop regime; polyether ether ketone (PEEK); diamond-like carbon (DLC) coating; metallic bearing alloys

1 Introduction

In full film bearings, although traditional bearing alloys are predominantly used, polymer sliding layers are also utilized in some applications more and more frequently, which is described in some papers [1–7], but they remain relatively uncommon in industrial practice. The operation of metal–polymer lubricated contact attracts the attention of the researchers, also in the transient conditions [8, 9].

On the other hand, hard coatings based on diamond-like carbon (DLC) are applied in the machine industry in many friction pairs [10], showing high wear resistance and very low friction, especially in mixed friction conditions, e.g. [11]. Tests of DLC coatings in full film conditions are relatively rare, but also in this regime, low wear, and low friction occur [12]. Low friction losses in lubricated contact of DLC-coated parts are sometimes attributed to slip conditions occurring on the DLC-lubricant interphase [13, 14].

The limited number of experimental results of the performance of tilting pad thrust bearings made from different materials was the basis for presenting these results. The presented research started with the DLC bearing tests, described earlier in Ref. [14], later a bronze bearing comparative research was carried out

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with the use of a bearing with identical parameters. The other two bearings, polyether ether ketone (PEEK) and Babbitt, were slightly different in dimensions but tested similarly. These results were partly presented in Ref. [15]. The result of the previous investigations gathered and the measurements carried out for a bronze bearing are systematically compared in this paper, to present results comprising four similar tilting pad thrust bearings utilizing different friction pairs: two, more traditional, metallic-bronze/steel and babbitt/steel pairs, and two, being modern alternatives—PEEK/steel and DLC/steel. The whole research program included also long-time steady-state tests, as a general test of bearing feasibility, but these are not reported in this paper.

The presented research aimed to study transition conditions to mixed friction in tilting pad thrust bearings made from different materials. Such conditions are important for the bearing operation because they usually occur during starts and stops and also occur on the verge of failure. Although there are papers in which polymer materials used in sliding bearings are tested in transient conditions, e.g. [8, 9], there are not many papers that describe experiments at the component level i.e., on complete thrust bearings and compare bearings with different sliding layers. Bouyer et al. [16] compared steady state operation and start-ups of two small tilting pad bearings differing in the sliding surface material. One of the bearings was lined with PEEK and the other with Babbitt. In steady-state operation, the polymer lined bearings demonstrated lower friction and also smaller film thickness, most probably due to a higher temperature. During the start-ups, maximum torque in a polymer lined bearing was considerably smaller than that in a babbitt lined bearing. The friction coefficient during the start-up stage of journal bearing made of bronze and Babbitt was investigated and compared by Bouyer and Fillon [17]. Results of their measurements proved that the Babbitted bearing operated with higher friction than the bronze one, however the difference in friction coefficient (COF) decreased with the increase of the bearing load. Han et al. [18] carried out a comparison of a journal bearing losses in water lubrication. The tested bearings were made from three materials: Rubber, PEEK, and polyamide. The obtained results confirmed lower friction and speed at the transition from fluid to mixed lubrication regimes for PEEK and polyamide bearing than for the rubber bearing.

Contrary to the tests at the component level, the tests of materials at the sample level are relatively frequent. For example, Kalin et al. [19, 20] investigated different lubrication regimes for DLC coatings in comparison to steel surfaces. The observed advantages of DLC coatings were connected with their lower wettability. McCarthy and Glavatskih tested several polymer materials and Babbitt in block-on-ring tribometer configuration under transient conditions of operation [21]. They found out, that Babbitt performed poorly in comparison with the polymers with the COF almost twice that of the composites. It should however be noticed, that there is a limited possibility to transform the results obtained at the sample level directly to the component level due to differences in sliding couple configuration, dimensions, and operating conditions.

2 Experimental

2.1 Test rig

Experimental testing was carried out at a specialized thrust bearing test rig at Gdańsk University of Technology. The test rig was described in more detail in Ref. [14]. Its main specifications are shown in Table 1.

The design of the test head is shown schematically in Fig. 1, and the photo of the test head with the open front cover, showing the test bearing is presented in Fig. 2. Two thrust bearings are mounted in the housing 5—front bearing 1 and back bearing 3. The thrust collar 2 fixed to the shaft is their counterpart. The axial force of up to 90 kN is applied to the back bearing by a ring-shaped hydraulic piston 4—hydraulic pressure in the piston is used to evaluate the axial

Table 1 Specification of the test rig.

Parameter	Value
Bearing outer diameter	Up to 200 mm
Bearing axial load	Up to 90 kN
Rotational speed of the motor	Approximately 20–4,500 rpm
Lubrication mode	Flooded housing, controlled flow to each bearing

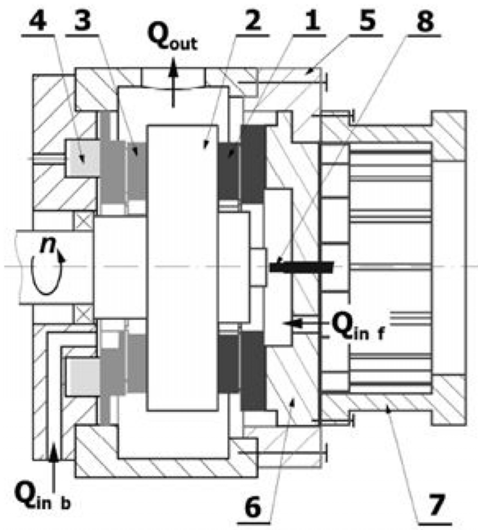

Fig. 1 Cross section of the bearing test rig: 1-front bearing, 2-collar, 3-back bearing, 4-load piston, 5-bearing housing, 6-plate, 7-torque meter, 8-distance sensor [23]. Reproduced with permission from Ref. [23], © The authors, 2018.

Fig. 2 Photograph of the bearing test rig with an open cover showing the front test bearing.

force acting on the test bearing. The front bearing is fixed to the base 6, which is attached to a spring structure being the main part of a special torque meter 7. The torque meter allows for precise measurements of a very small friction torque in the presence of substantial axial loads. Its original design was described in detail in Ref. [22]. The shaft is driven by an electric motor with an inverter to change and control the rotational speed. The rotational speed is also measured. Lubricating oil is delivered to both bearings (see $Q_{in\ f}$ and $Q_{in\ b}$ in Fig. 1), and drained from the top of the housing (Q_{out} in Fig. 1), to ensure that the housing is flooded. The axial position of the shaft related to the test bearing film thickness is measured with the use of a proximity probe 8 installed in line with the shaft axis.

Test conditions of the bearings provide a broad area of the operational envelope of the industrial bearings, where specific loads can reach 10 MPa and sliding speeds can reach 35 m/s. A frequency converter controls the driving motor of 30 kW power and 3,000 rpm of nominal speed. The range of operational speeds was adequate for this research, especially the possibility of maintaining the low rotational speed of the shaft at a stable level of about 20 rpm, as well as the possibility of precise torque measurements, which were important for low-speed tests and Stribeck curve tests. Due to changes introduced in the design of the lubricating system, different methods of oil supply can be tested.

2.2 Test bearings and instrumentation

Two thrust bearings are mounted in the test rig. They operate in virtually identical operating conditions but only in the test bearing the friction torque is measured during the tests. The main specifications of the bearings used for the research are presented in Table 2.

Altogether four bearings were tested—one of them incorporated a DLC-steel material combination, with tilting pads made from steel and a thrust collar lined with a thin DLC layer [10]. The second bearing had tin bronze (11.6% of Sn and 2.2% of Ni–CuSn12Ni2) pads operating against a steel collar. The other two bearings differed slightly in dimensions and comprised PEEK vs steel and Babbitt (SnSb8Cu4) vs steel material combinations. The details of the test bearings and lubrication are collected in Table 2 and the bearings are shown in Fig. 3.

Both types of bearings were extensively instrumented with numerous thermocouples installed in the pads. The layout of the thermocouples is shown in Fig. 4.

In Table 3 coordinates of selected thermocouples are specified, the readings of these thermocouples

Table 2 Test thrust bearing data.

Parameter	Bearing no. 1	Bearing no. 2	Bearing no. 3	Bearing no. 4
Material combination	Babbitt SnSb8Cu4 (acc. to ISO 4381) vs steel collar	PEEK vs steel collar	Tin-bronze CuSn12Ni2 (acc. to EN 1982) vs steel collar	Steel pad vs collar lined with DLC
Outer diameter	192 mm	192 mm	175 mm	175 mm
Inner diameter	103 mm	103 mm	95 mm	95 mm
Mean diameter	147.5 mm	147.5 mm	135 mm	135 mm
Pad thickness	14 mm	14 mm	14 mm	14 mm
Number of pads	8	8	8	8
Pivot type	Off-set edge support	Off-set edge support	Symmetric edge support	Symmetric edge support
Lubricant (supplied at 40 °C, 15 L/min per bearing)	ISO VG-32	ISO VG-32	ISO VG-68	ISO VG-68

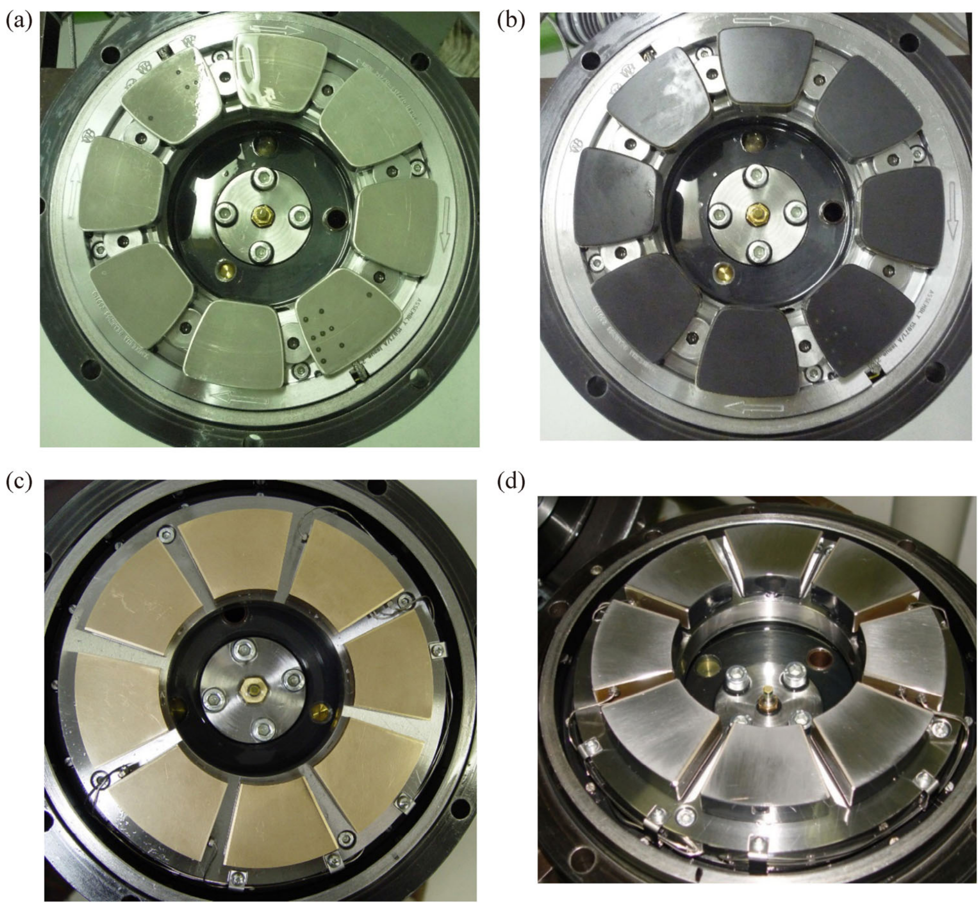

Fig. 3 Photographs of the test bearings. (a) Bearing no. 1 (Babbitt); (b) bearing no. 2 (PEEK); (c) bearing no. 3 (bronze); and (d) bearing no. 4 (steel pad/DLC collar).

Table 3 Specification of selected thermocouples coordinates.

Symbol	Radial relative *Rp*	Tangential relative Θ	Distance from the sliding surface	Remarks
T6	75.3%	66.3%	0 mm	Bearings nos. 1 and 2
TF_A2, TF_B2	75%	75%	3 mm	Bearings nos. 3 and 4

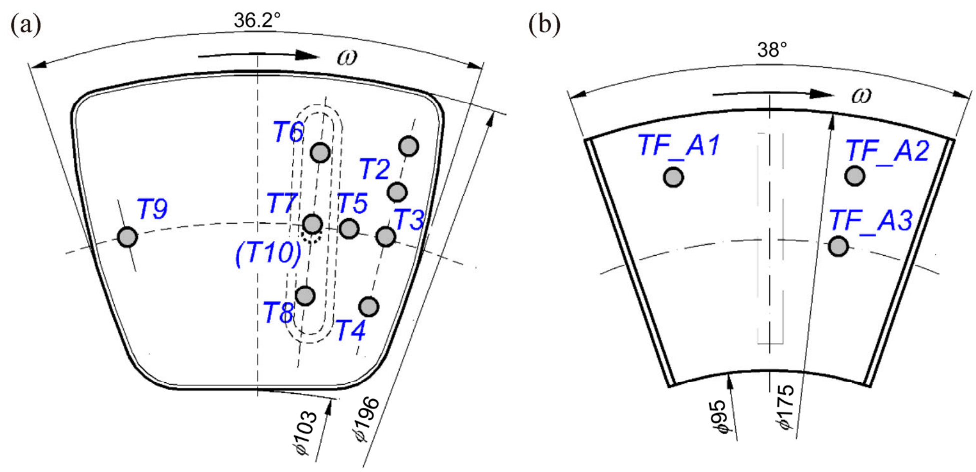

Fig. 4 Arrangement of the temperature sensors in the test bearings. (a) Bearings nos. 1 and 2; and (b) bearings nos. 3 and 4.

located close to the point of maximum pad temperature will be used to show the representative maximum bearing temperature.

2.3 Test conditions

To show and compare all four bearings' performance in severe transient operating conditions the results of two tests are discussed here. One was the start-stop test and the other was the test aimed at reproducing the transition from full film to mixed lubrication conditions. The conditions of the tests described here are specified in Table 4.

Start-stop tests were run with the start at no-load condition because the driving system did not generate adequate torque to start under load, while the stopping was carried out under full load. Similar tests of start-stop were repeated one hundred times with each of them lasting for 5 min.

The transition to mixed friction test was carried out at 5 or 3.65 MPa of specific load, as shown in Table 4, and a stepwise decrease of the rotational speed until the increase of friction coefficient, and also temperature, was observed. To maintain steady-state conditions at all speed steps, each speed step lasted between 5 and 10 min.

3 Results of start-stop tests

Friction torque measured during cycles #1, #50, and #100 of the start-stop tests for all tested bearings is shown in Fig. 5 together with the load and speed. In each of the bearings, except the bronze one, the course of parameters over the whole test period looks very similar. The measured friction torque in a bronze bearing differs significantly between cycles for the stopping phase (Fig. 5(c), ~285 s). In this bearing, it is clear that the smallest friction during stopping occurred in cycle #1. In addition, close to the transition point from fluid to mixed lubrication, the course of friction torque in this bearing changed between cycles from smoothly shaped at the beginning (cycle #1), to scattered in the next shown tests (#50 and #100).

Comparing trends of certain parameters between particular cycles one could observe some distinct changes. During stopping under load, the bearing was experiencing the transition from a full film

Table 4 Specification of test conditions.

Test	Conditions		
	Specific load	Rotational speed	Remarks
Start–stop	3.65 MPa (brgs #1 and #2) 5 MPa (brgs # 3 and #4)	0 ↗ 500 rpm ↘ 0	Stop under load, 100 cycles, ~ 5 min each
Transition to mixed friction	3.65 MPa (brgs # 1 and #2) 5 MPa (brgs # 3 and #4)	1,000 rpm ↘ 25 rpm	~10 min at each speed step

lubrication regime to mixed friction regime, and the analysis of the course of certain quantities shows the conditions at which the transition occurred.

An enlarged view of the changes of friction torque and rotational speed at the end of the stopping in cycles #1, #50, and #100 for all tested bearings is shown in Fig. 6. Due to the failure of the shaft speed tachometer, the speed was not measured for the bronze bearing. From all start-stop cycles, two sets of values were extracted—Minimum COF during stopping and the rotational speed at which this minimum COF occurred. A simple algorithm was used to select the minimum value of the friction torque. The friction torque was recalculated to obtain the COF, based on the mean diameter of each bearing (see Table 2), and the load.

In these investigations, the minimum measured friction torque during the stopping phase of the cycle was assumed to be a transition point between fluid and mixed friction in the bearing. However, numerical simulations show that the COF keeps decreasing with the speed, even after the asperity contact has occurred [24, 25]. According to this, the minimum COF occurs already in the regime of mixed lubrication, shortly after the occurrence of the first asperity contact. From a practical point of view, however, defining the transition point at minimum measured friction torque allows us to easily evaluate the transition point in experimental data. It should be also noted, that according to the results of simulations, the occurrence of minimum friction and the first asperity contacts is very close.

The changes of two parameters are shown below: The first is the value of minimum COF measured during the stopping period, just before the stopping of the shaft (Fig. 7). The value of the COF was gradually decreasing in the course of the tests for bronze bearing—at the beginning, it was approximately 0.0023 and at the end of the tests this value dropped to 0.0016—most probably as the result of the bearing running-in process. In the tests of DLC bearing, the COF was considerably smaller—approximately 0.0012 and no substantial changes were observed during the tests. No considerable changes in minimum COF

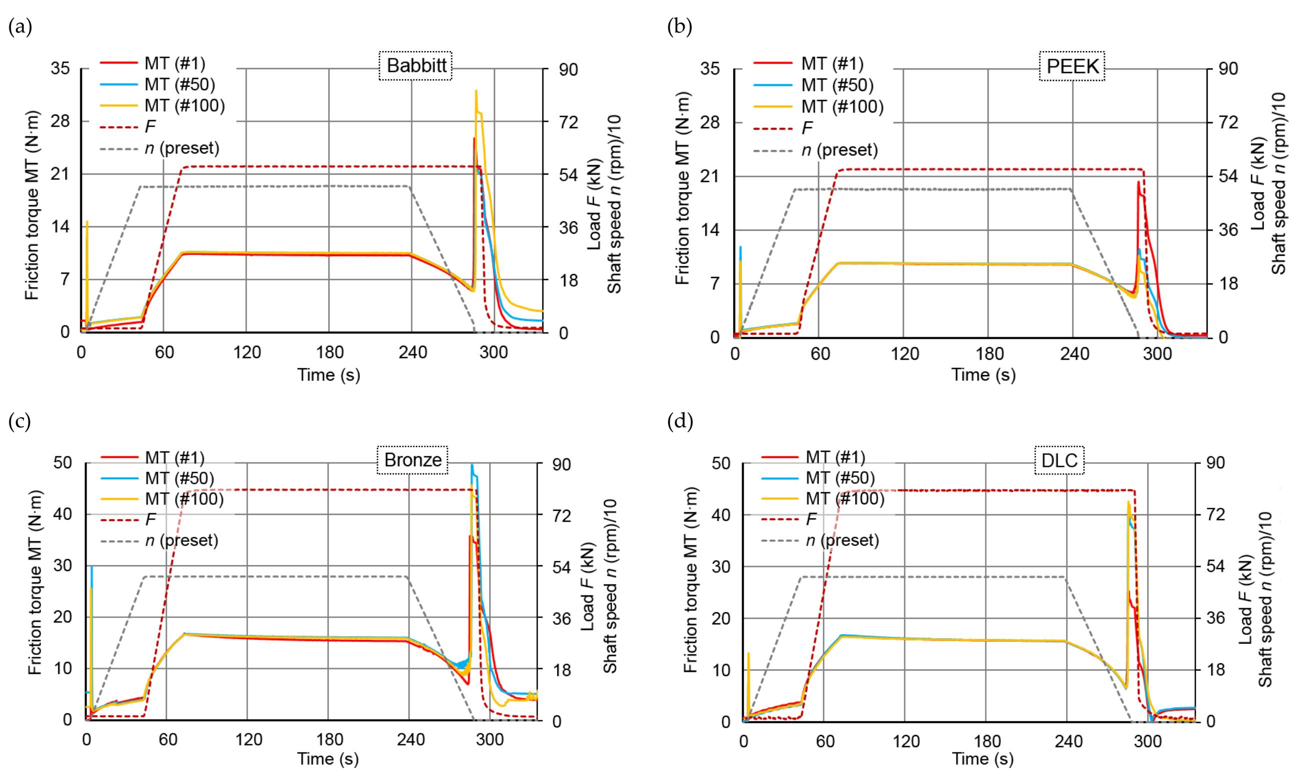

Fig. 5 Results of friction torque in test bearings in test cycles nos 1, 50, and 100 as a function of time, axial force *F* (kN), and shaft speed *n* are also plotted to show the load and speed conditions at each start-stop cycle. (a) Babbitt bearing; (b) PEEK bearing; (c) bronze bearing; and (d) DLC bearing.

during the subsequent cycles were observed for Babbitt and PEEK lined bearing with the value for PEEK (0.00125) being slightly smaller than that for Babbitt (0.0013).

The other quantity showing parameters at which transition to mixed friction occurred was the rotational speed at which the minimum torque was observed (Fig. 8). In this case, the largest value and at the same time the largest scatter can be observed in a bronze bearing. The speed was varying throughout the tests

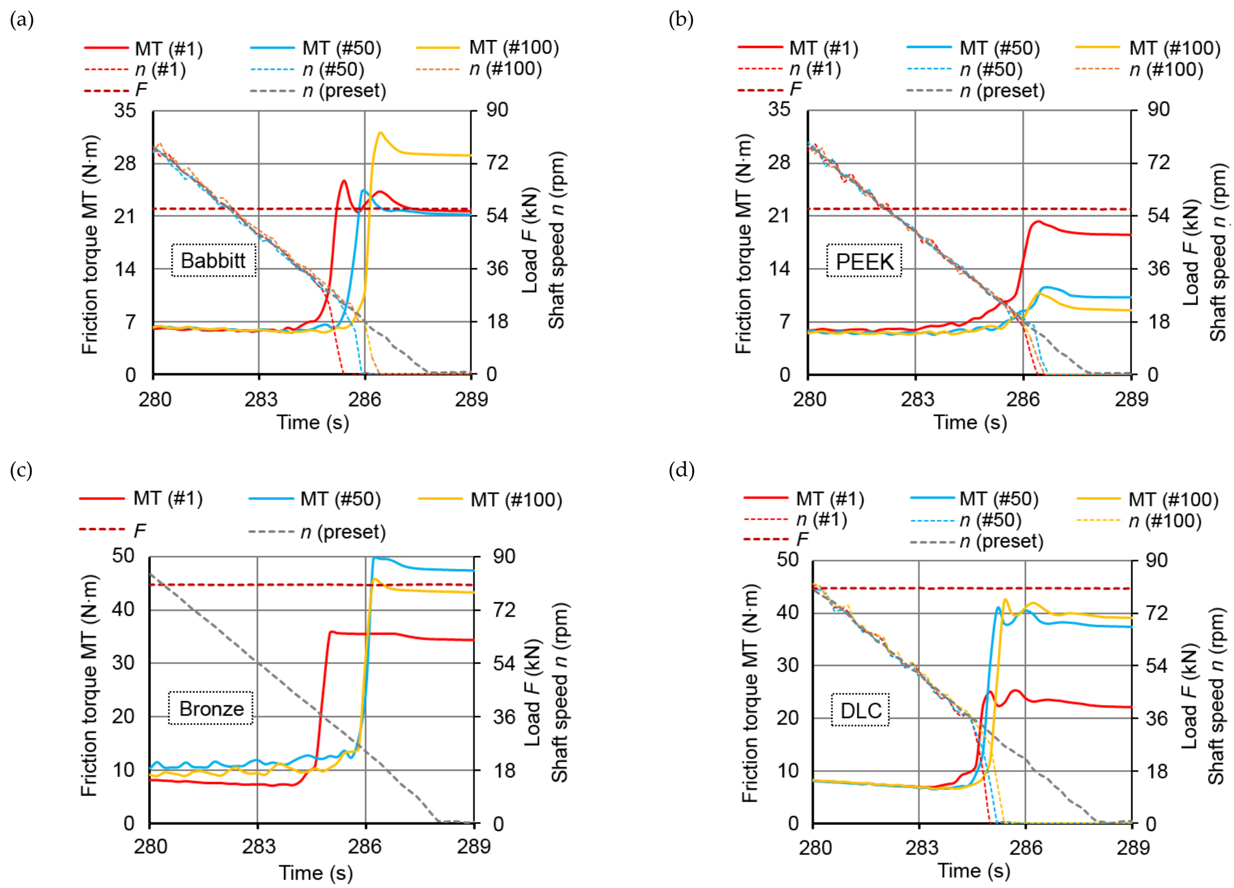

Fig. 6 Results of friction torque as a function of time, at the end of start-stop test cycles nos. 1, 50, and 100. (a) Babbitt bearing; (b) PEEK bearing; (c) bronze bearing; and (d) DLC bearing.

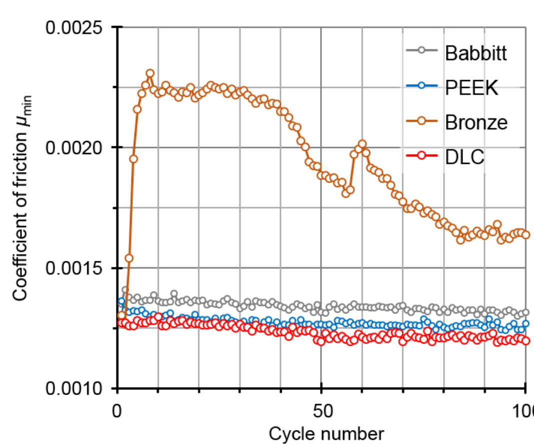

Fig. 7 Comparison of minimum COF measured before the stop of the shaft during the shut-down stage of each cycle.

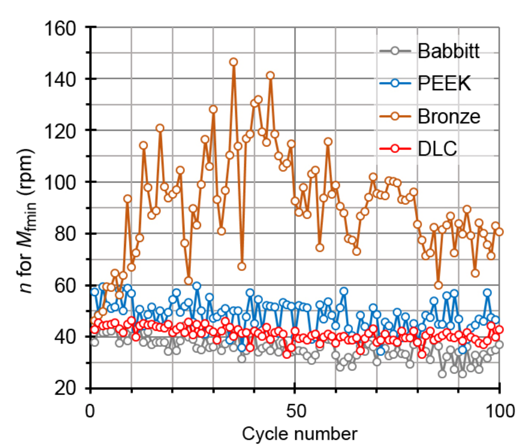

Fig. 8 Comparison of shaft speed (rpm), at which minimum friction torque was measured.

reaching stability at approximately 80 rpm almost at the end of the tests, after 80's cycle. The most stable value throughout the whole series of tests was observed in a DLC–steel material combination. In this bearing, the transition speed was equal to 40 rpm and was stable throughout the tests, which also shows better stability of bearing parameters and the ability to operate in a fluid film regime at a lower rotational speed.

4 Results of Stribeck curve tests

The second test was aimed at experimental evaluation of the parameters of transition to mixed friction, it was carried out by a stepwise decrease of shaft rotational speed, starting from 1,000 rpm, down to approximately 20–25 rpm. Bearing specific load was kept constant at 5 or 3.65 MPa, as shown in Table 4. Obtained results of COF were presented as a function of dimensionless Hersey number, which makes them independent from different load values and, what is also important, possible to compare between each other. The change of parameters during the test for all analyzed bearing material configurations is shown in Fig. 9.

Looking at the differences between all bearings one can observe:

- Larger changes in COF after the change of speed for DLC and bronze bearings;
- the increasing scatter of COF value with decreasing collar speed;
- the largest scatter of COF in the Babbitt bearing and the smallest in the DLC bearing;
- quite clear signs of COF decrease during one step of speed at the smallest speeds, looking like running in process;
- large peaks of COF at the beginning of some speed steps in the bronze bearing.

Based on these experiments Stribeck curves were created. As discussed above, the minimum value of COF occurs soon after the first asperities contact and is often considered as the point of transition from full film to mixed lubrication—this value is analyzed in this paper.

To plot a Stribeck curve, the COF was calculated, and at each load step the Hersey number, which is a dimensionless parameter showing bearing load [26, 27], was evaluated based on load, rotational speed, and instantaneous oil viscosity:

$$Hs = \frac{\eta \times n}{p_{\mathrm{av}}} \tag{1}$$

where Hs is Hersey number; η is oil dynamic viscosity (Pa·s) at the measured bearing temperature, i.e., T6 or TF_A2 depending on the tested bearing; n is rotational speed (1/s), and p_{av} is bearing specific load (Pa).

The graphs in Fig. 9 show real data during the test and illustrate one of the problems of the Stribeck curve reproducing, namely, COF value. The COF has a certain scatter and changes even at one level of load, so it is necessary to use an average value. In this case, the average was calculated from the values after the COF value dropped and stabilized. Another question may be what value of oil viscosity should be used in Eq. (1), however in this research, due to the installation of thermocouples in the bearings, bearings temperature was known (Fig. 10), and this temperature was used to evaluate oil viscosity. It is worth noting that due to low sliding speed and low COF, the heat generated in the oil film was small, and due to this temperature differences at particular speed steps were also small, lower than 0.1 °C, as shown in Fig. 10 illustrating the bearings temperature vs Hs number.

The result of the COF plotted as a function of the Hersey number (Stribeck curve) in all four bearings is shown in Fig. 11, in this figure one can observe when the transition from full film lubrication to mixed friction occurred. It is clear that DLC/steel combination performed best out of all four tested bearings with the minimum COF equal to 0.0012, and smaller by more than 10%, and transition to mixed friction shifted towards lower values of Hersey number—demonstrating the ability to operate in fluid film lubrication regime at a lower value of Hersey number, i.e. at more severe operating conditions. This result is consistent with the outcomes of the start-stop tests. Out of all four bearings, the Babbitt/steel combination was the most prone to enter a mixed lubrication regime—it occurred at a much higher value of the Hersey number and the highest value of the COF—equal to approximately 0.0016. Bronze bearing did

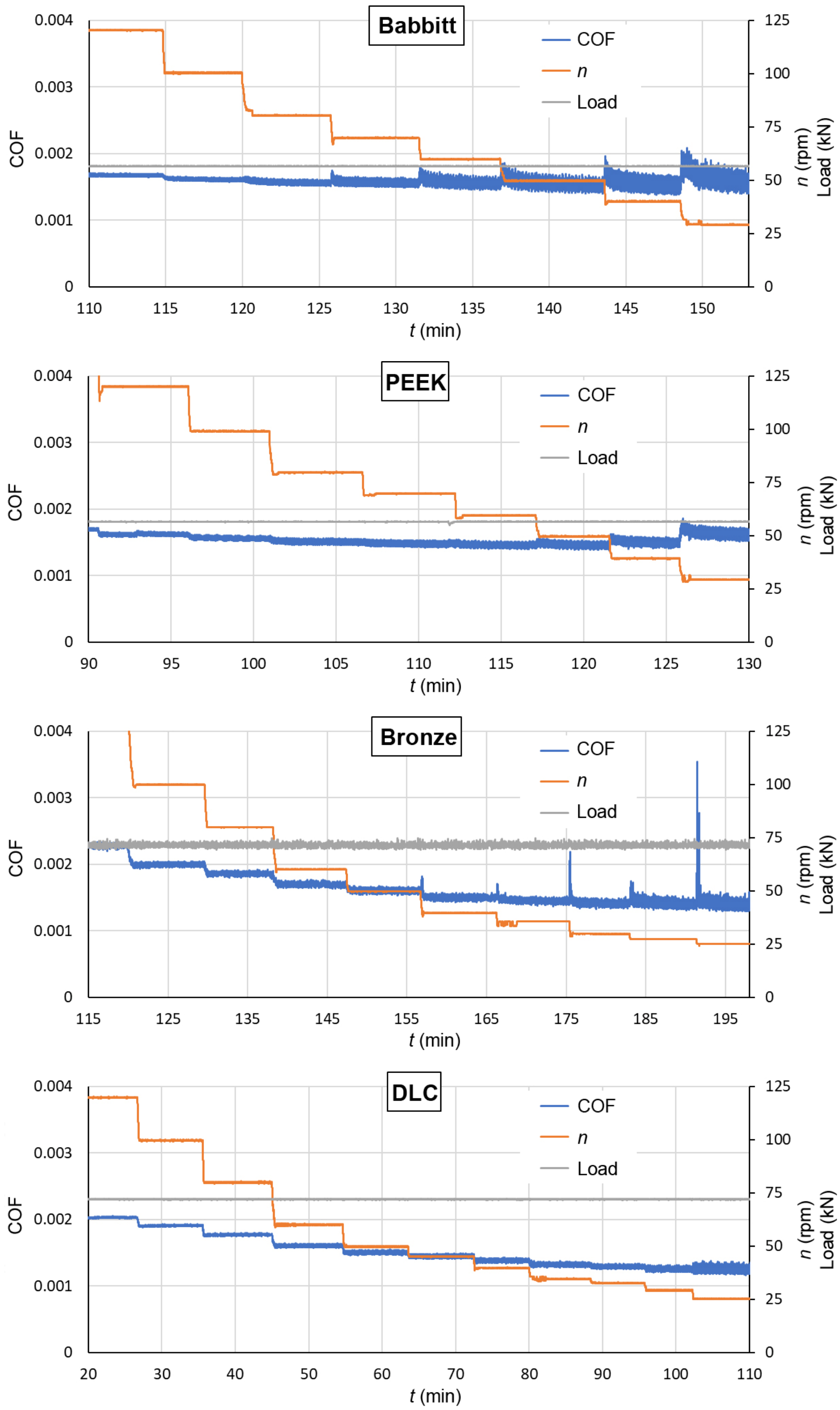

Fig. 9 Changes of COF in time during transition tests, shaft speed n and constant bearing load P are also plotted.

not demonstrate a very clear transition to mixed friction and the COF was equal to 0.0014. In the bearing lined with PEEK, the transition was very clear and occurred at a bigger value of Hersey number than that of DLC bearing and the COF was slightly lower than 0.0015.

5 Conclusions

The main conclusions of the research can be summarized as follows:

Transition to mixed friction in DLC bearing was occurring at higher bearing load (lower value of

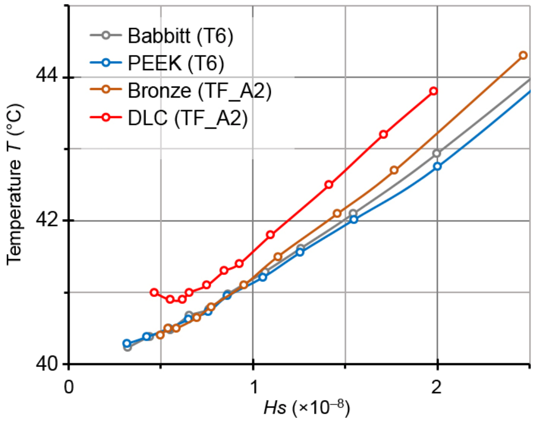

Fig. 10 Bearing temperature as a function of *Hs*.

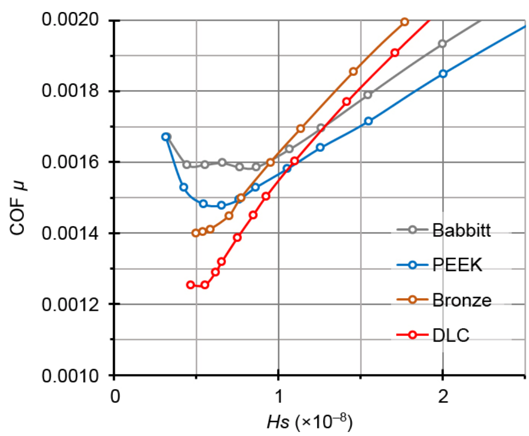

Fig. 11 COF as a function of *Hs*.

Hersey number parameter). There were two main indications of DLC bearing superiority over the other bearings:

1) the lowest friction in the moment of transition to mixed friction and fewer irregularities and peaks of torque, which were also occurring at lower speeds;

2) lower minimum torque occurring during stopping of the shaft in start–stop tests, and much lower speed at which this stop was caused.

The other three bearings performed slightly worse, with a higher friction coefficient (COF) and the transition occurring at a larger Hersey number.

These results are interesting, especially bearing in mind that the DLC steel combination is not usually used in fluid film bearings, as opposed to the metal-based combinations of bronze–steel and babbitt–steel. The other non-standard combination, which was polyether ether ketone (PEEK) steel, performed well, with a low COF, both in start- stop tests and the Stribeck curve test.

Although the DLC lining performed very well in the tests, its potential wider application in fluid film bearings would require thorough testing aimed at its feasibility, one of the dangers impairing the service life and reliability is delamination of the lining. On the other hand, DLC coatings are quite extensively used in roller element bearings and other various automotive applications, so the coating technique is already well-developed.

Declaration of competing interest

The authors have no competing interests to declare that are relevant to the content of this article.

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