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Tribological properties of MoS_2 particles as lubricant additive on the performance of statically loaded radial journal bearings

Hasan Baş[*1]

[1]*Karadeniz Technical University, Department of Mechanical Engineering, Türkiye*

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Abstract

It is very important to reduce power losses for plain bearings used in industry and especially in automobiles. In recent years, inorganic compounds have been added to the engine oils to reduce friction in plain bearings and increase their performance. In this study, the effects of additive oil obtained by adding 1% by weight Molybdenum Disulfide (MoS_2) to the base hydraulic (or lubricating) oil (Shell Tellus 10) on the statically loaded radial journal bearing performance were experimentally investigated. As a result of experiments, it was determined that MoS_2 added engine oil showed less friction behavior by forming a better oil film compared to the base oil at increasing bearing load and temperatures, and therefore exhibited less wear and power loss.

1. Introduction

Hydrodynamic bearings are used in high-speed rotary device [1]. The radial bearings, which carry load in the radial direction and work on the principle of the hydrodynamic lubrication, are one of the common types of the bearings used in industry. They are especially used in environments where cushioning of vibrations and resistance and silence to vibratory forces are essential. Journal bearings have a great effect on reducing power losses in mechanical systems. The friction factor and thus the power loss is a function of viscosity, load, and speed. There was a great deal of research at that time trying to find the best combinations of materials and lubricants that would give the lowest coefficient of friction.

The use of suitable lubricants to reduce the friction and wear of mechanical contacts is very important for power loss [2]. To reduce maintenance period and power loss of rotary machine, some of additive oils have been added to base mineral oil in various industries. To improve lubricant oil properties, additives have been used. These contain antiwear additives, extreme pressure additives, viscosity control additives, film-forming additives, and sediment control additives [3]. Recently, attention has been directed towards nanoparticles since they have unique properties when compared to their majority equivalents.

Studies in the use of nanoparticles as additives have shown to reduce friction and wear [4,5]. The choice of appropriate lubricant has a special effect on the performance of the bearings. The major losses that immerge in a car engine can be friction between moving parts. The main role of the lubricant is to retain the surface of the two metals dampish, that separates them from each other by creating a suitable layer on the surfaces with friction, and the heat and created abrasive particles will be annihilated [6]. A good lubricant should have a high flash point, low pour point and high viscosity index (VI). Nano-additive oil-based lubricants expose wonderful lubrication performance for tribological applications [7,8]. Due to their proper lubrication properties, the use of MoS_2 nanoparticles as solid lubricants has been highly regarded, recently. MoS_2 nanoparticles has a hexagonal crystalline structure.

* **Corresponding Author**

*(bas@ktu.edu.tr) ORCID ID 0000-0002-5653-3813

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The self-lubricity properties are related to the large area between the S-Mo-S sandwich layers and the weak Van der Waals forces and the pure positive charge at the surface, resulting in propagation of electrostatic repulsion. The layers exist with weak molecular forces and can easily slide over each other, [6, 9-12]. The effect of MoS_2 nano-additives on the frictional properties of the lubricating oil was studied by Rajendhran et al [13]. The results divulged that 0.5 wt% nano-additives added to the pure oil could distinctly improve the tribological properties. Numerous studies have been carried out to investigate the tribological properties of MoS_2 nano-doped oil-based lubricants. The results demonstrated that the lubricant characteristics of the prepared nano-fluids is attractively improved with the combine of MoS_2 nanoparticles [12, 14-19].

There are three lubrication regimes for the tribology of the journal bearing, which are generally defined as boundary, mixed and full film lubrication, firstly obtained by the McKee brothers [20]. In this graph, there is usually a friction coefficient on the vertical axis, and only velocity or an $\eta n/P$ dimensionless expression can be on the horizontal axis.

The variation of the "friction coefficient [μ]" in the journal bearing versus the "bearing parameter $\eta n/P$ " is shown in Fig. 1. Where n is the speed of the shaft (rev./s), p is the nominal pressure of the bearing (N/m^2), and η is the dynamic viscosity of the lubricant (N.s/m^2).

The friction coefficient, the viscosity of the lubricant used in the bearing, the strain applied to the system by the external load carried by the bearing, and the relative speed difference between the sliding speeds of the surfaces can instantly be seen in the graph. With the help of this graphic, practical information can be obtained about whether the bearing works in the critical operating region, that is, in the region close to metal-metal contact.

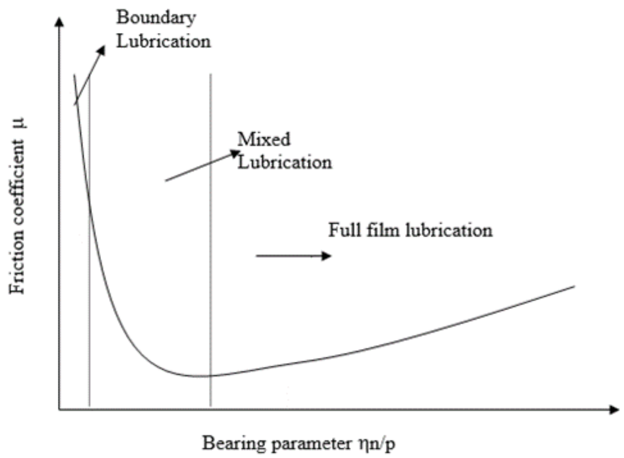

Figure 1. A Schematic Stribeck curves showing three lubrication regimes

Since the minimum film thickness is generally dependent on surface roughness, bed load and relative velocity, an acceptable minimum value is not shown. However, in order to avoid metal-to-metal contact and to keep friction losses at a minimum level, %20 increase in the sum of the surface roughness can be determined as the minimum value. In this study, a measurement system was used that shows the metal-to-metal contact between the bearing and the shaft during the tests. This study experimentally compares the effects of MoS_2 on the base engine oil (Shell Tellus 10) in the statically loaded radial journal bearing. In addition, the effect of the MoS_2 concentration particle's on the tribological behavior of lubricants is explored.

To determine the friction behavior, the bearing system was modified and constructed. The structure of the particles was characterized by scanning electron microscopy (SEM).

2. Method

To investigate the effects of additive engine oil on bearing performance, a journal bearing test rig designed specifically for this research. All experiments were carried out using the base engine oil (Shell Tellus 10) and by adding 1% by weight Molybdenum Disulfide (MoS_2) to the base engine oil at the flow rate of 55 cm^3/min. The optimum mixture of nanoparticle was determined according to the results obtained from studies in the literature [21-23]. Accordingly, molybdenum disulfide was added to the oil at 1% wt.

The bearing oil temperatures were measured using a thermometer at the outlet of the bearing and dynamic viscosity values were taken from the calibration chart of viscosity-temperature for this type of oil, given Table 1.

Table 1. Typical physical characteristics of the base oil, Shell Tellus 10

Shell Tellus C Oil		
Viscosity Garde (ISO 3448)		10
ISO Oil Type		HL
Viscosity cSt	40 °C	10
	@ 0 °C	2.5
Viscosity Index (IP34)		70
Density @ 15 °C (IP 160) kg/l		0.877
Falsh Point (IP34) (PMCC) °C		166

Tests carried out two different oil temperature, each test oil temperature remained in the range 25-30 °C and 50-60 °C with matching dynamic viscosity 0.02 and 0.01 N.s/m^2, respectively. After running-in (initial) process, and final (the end of working at 253N and 553N; 1.22 and 2.66 MPa pressures) surface roughness of the journal bearings and the shaft was measured by using surface roughness tester (Mahr-Germany Perthen Perthometer) in pursuant of DIN 4768. Surface roughness parameters such as center line average (Ra), average peak to valley height (Rz) and maximum peak to valley height (Rmax) were measured by using a driving unit at a tracing speed of 0.5 mm/s with traversing and cut-off lengths, 4.8 mm. In order to read measured roughness values, the device was connected to a measuring indicator. The average values of five roughness measurements were taken. The surface roughness results before and after the tests are given in Table 2. As seen in Table 2, the roughness of the bearing and shaft surfaces changed very little during the tests.

Table 2. Surfaces roughness values before and after experimental studies

Before (Bearing)		After (Bearing)		Before (Shaft)		After (Shaft)	
Ra (µm)	Rz (µm)	Ra (µm)	Rz (µm)	Ra (µm)	Rz (µm)	Ra (µm)	Rz (µm)
0,25	1,45	0,18	0,92	0,36	1,17	0,34	2,02

2.1. Bearing Material

In order to minimize wear in plain bearings, the selection of the bearing material working with the shaft is very important [24]. For this reason, containing tin (6-40%), aluminum-based ZA-27 plain bearing was used as the bearing material in the experiments. To determine the chemical compositions of the alloy, the atomic absorption method was used. The results of chemical analysis of the plain bearing and measured MoS_2 particles size range were given in Table 3.

Table 3. Chemical compositions of the alloys and measured MoS_2 particles size range

Alloy	Chemical composition (Weight %)				Size range of MoS_2 particles (±1µm)
	Zn	Al	Cu	Mg	
ZA-27	70,8	27.2	2.01	0,02	3,5-11

2.2. Physical Methods

Experimental test rig of the journal bearing is given in Fig. 2 as; a) schematic representation, b) photo and c) technical drawing with dimensions. The rig consists of a rigid tubular steel of square profile frame equipped with a pneumatically operated pressurized oil supply, a filtered oil tank, a modified electronic circuit with shaft and bearing housing, a direct current variable speed control unit, assembly of split journal bearing and electronic instrumentation for indicating shaft speed and motor current. The apparatus operates a 220-240 V mono phase electrical supply and an air supply at a minimum pressure of 0.6 MPa. The pressurized oil system composed of a double acting pneumatic cylinder driving a hydraulic oil cylinder in a reciprocating manner. Integrated in the system are two over pressure relief valves, which limit the oil supply pressure to the bearing and hydrostatic pad to approximately 0.15 and 1.6 MPa respectively.

In this experimental test rig, the hydrostatic pad is constructed using mild-steel, and accurately lapped to fit the journal bearing housing with suitable tolerance. Without affecting the sensitivity of friction torque measurements, 11 oil jets used for separate the hydrostatic pad and outside of the bearing cap, enables the bearing load to be transmitted to the bearing, which is increase up to maximum 553 N. Oil leakage from the bearing and hydrostatic pad is collected in the drip tray and returned to the reservoir. The test oil is collected in another tray and returned to the gap between shaft and bearing via pump. The journal bearing housing is made of iron and has a correctly ground bore to support thin-walled bearings used in automobile engines. The sleeve was made of steel hardened to 722 HB.

The rotational speed could be adjusted in the range of 0-1100 rpm (0-2.88 m/s) using a direct current speed control unit.

In order to be able to measure the measurements in the boundary and mixed friction regions in detail at the test duration, the stepper motor and equipment's were mounted to the system using one-way clutch gear, Fig 2-b. The effective bearing length was taken as 18 mm (Fig. 2-c) to operate at a pressure of 2.66 MPa because the maximum allowable load of the test rig was 553 N. The diameter of shaft and thin-walled journal bearing are 50.760 and 50.850 mm, respectively. The bearings had a bearing clearance of 45 µm, Fig. 2-c.

By using a Wheatstone bridge circuit with strain gauges on the torque plate, the friction torque on the journal bearing was measured. Two strain gauges on the bridges were used as active gauges, others used as passive gauges. The torque signal obtained from the bridge circuit was calibrated and transmitted to the recorder device and this enabled monitoring and recording the signals. Using strain gauges signals, the recorded signal values were then converted into friction torque considering the dimension of bearing. Before executing tests, the journal bearing surfaces were exposed to running-in process under a pressure of 1.2 MPa at a constant rotational shaft speed (500 rpm, 1.3 m/s) for 10 minutes. After this process, experimental tests were carried out at rotational speeds ranging from 0 rpm to 300 rpm (0.78 m/s) for 5 minutes at each shaft speed. The pressures of hydrostatic pad and lubricating oil were selected as 1.2 and 0.1 MPa respectively. To investigate the effects of additive engine oil on bearing performance, base engine oil (Shell Tellus 10) and MoS_2 nano particles were used by adding 1% by weight to the base oil. The structure of the particles was characterized by scanning electron microscopy (SEM), shown in Fig. 3. The MoS_2 nanoparticles have a platelet-like shape with an average diameter of 7.2 µm, (Table 3). Purity of the nanoparticles is 99%. The average particle size and density of the MoS_2 additive at 25°C are 3.20 µm and 5.06 g/cm^3, respectively.

After each experiment, the bearing and shaft surfaces were cleaned with acetone-isopropanol mixture and the tests were prepared for initial conditions. The results were evaluated taking into consideration the formation of a boundary lubrication, mixed lubrication and liquid film lubrication. Experiments were carried out up to 300 rpm; 553 and 353 N bearing loads. Additive rates were selected as 1wt%. The running time of the tests were implemented in the range of 600-700s and 20-55°C under 70% relative humidity. Information on test conditions is also given in Table 4.

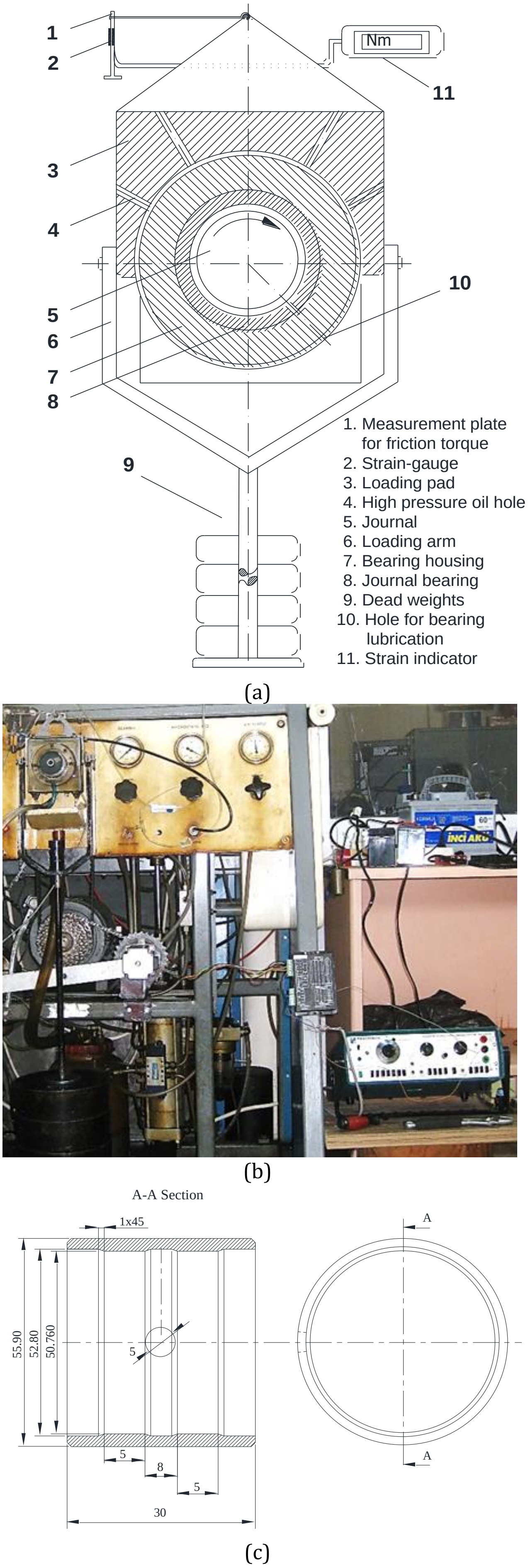

Figure 2. The journal bearing test rig; (a) Schematic, (b) Photo (c) Dimensions of the plain bearing

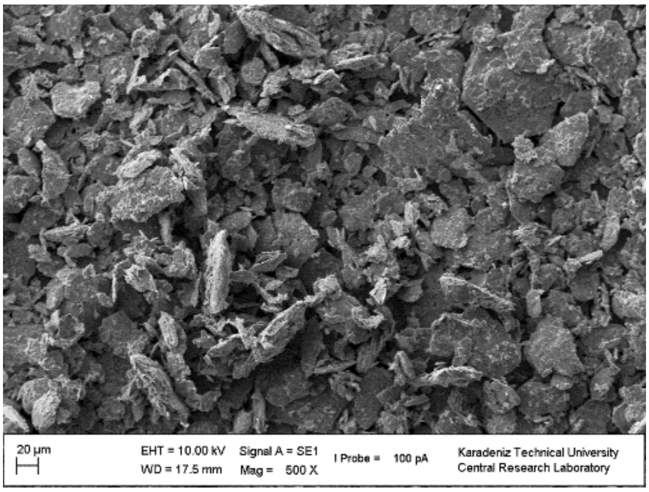

Figure 3. SEM morphologies of the MoS_2 nanoparticles

3. Results and Discussion

In the experiments, the base oil (Shell Tellus 10) and 1 wt.% MoS_2 additive were used to determine the effects of MoS_2 particles on the friction behavior in the bearing. As shown in Fig. 4, using MoS_2 additives, lower friction torque values were obtained in the tests compared to the base oil at 27°C test condition. This difference is even more pronounced in the boundary and mixed friction regions. For example, an average increase of 0.3 and 0.6 Nm is observed in the friction torque at 253N and 553N bearing loads in the boundary friction region, respectively. It is seen that the friction torque values in the bearing increase with the increasing bearing load. As a result, it has been determined that the effect of MoS_2 additive is more dominant in metal-to-metal contact operating conditions than that of the liquid film region. In the bearing tests with different oils performed at 253 N bearing load, the transition speed values are very close to each other and are around 14 rpm. As shown in Fig. 5, using of MoS_2 additives at 253N, the minimum friction torque value was obtained as 0.037 Nm. While using the base oil at the same test conditions, this value was 0.041 Nm. It was detected that MoS_2 particles has a more effects to decrease the friction torques than the base oil in the bearing.

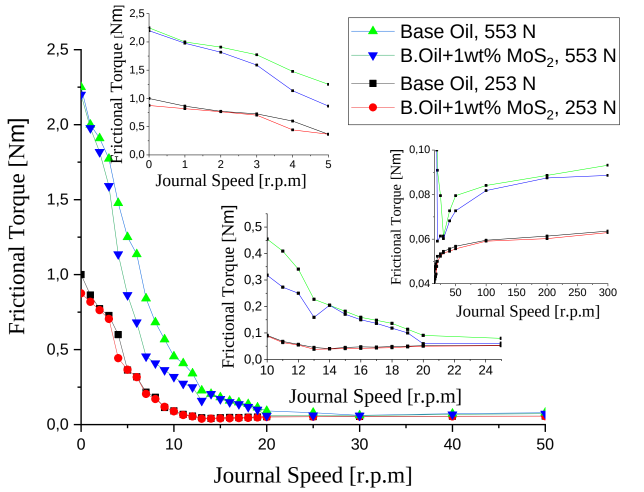

Figure 4. Variation of friction torque with rotational speed in two different oil and load tests

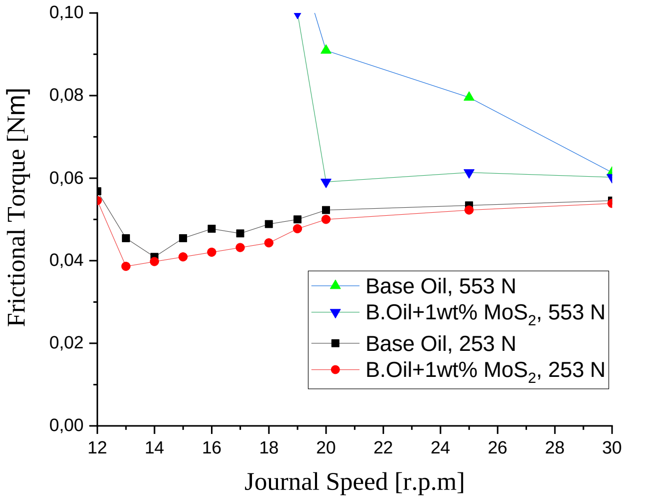

Figure 5. Variation of friction torque with rotational transition speed values in two different oil and load tests

In addition, the rotational transition speed values decrease with the use of MoS_2 additives, hence, frictional torque reaches at the minimum value at 27°C test condition. This means that, MoS_2 additives facilitates the formation of oil film between the bearing and the shaft. Using the base oil at 553 N bearing load, friction torque and rotational transition speed values increases and these values are 0.06 Nm and 30 rpm, respectively. Using MoS_2 additives these values are 0.057 Nm and 20 rpm, as shown in Fiq 5. In the fluid friction region, this difference is approximately 0.02 Nm at low load and 0.01 Nm at high load.

In order to determine the friction coefficient in the plain bearing at 27°C test conditions, firstly, the peripheral friction force values in the bearing were measured, and then, the friction coefficient values were obtained by taking the ratio of this friction force to the bearing load. Variation of frictional coefficient with rotational speed for two different oil (base oil and 1 wt% MoS_2 additive) and load was given in Fig. 6. As shown in Fig. 6, higher friction coefficient values were obtained in the base oil tests compared to the MoS_2 additive oil. These differences were even more in the boundary and mixed friction regions. It is seen that the friction coefficient values formed in the bearing decreases with increasing bearing load. The reason for the decrease may be that, compared to the friction force, the increase in the bearing load is higher, or the internal friction of the oil decreases as a result of the thinning of the oil film between the surfaces. Hence, the effect of surface roughness was effective in terms of friction coefficient under metal-to-metal contact operating conditions.

In addition, the rotational transition speed values decrease with the use of MoS_2 additives, hence, frictional coefficient reaches at the minimum value. Using the base oil at 253 N, the values of transition speed and frictional coefficient are 14 rpm and 0.007, respectively. With the use of MoS_2 additives, the values decreased to 13 rpm and 0.006, respectively. This means that, MoS_2 additives facilitates the formation of oil film between the bearing and the shaft. Using the base oil at 553 N bearing load, friction coefficient and rotational transition speed values increases and these values are 0.015 and 30 rpm, respectively. Using MoS_2 additives these values are 0.009 and 20 rpm, respectively, as shown in Fig. 7. In the fluid friction region, this difference is approximately 0.001 Nm at low load and 0.015 Nm at high load.

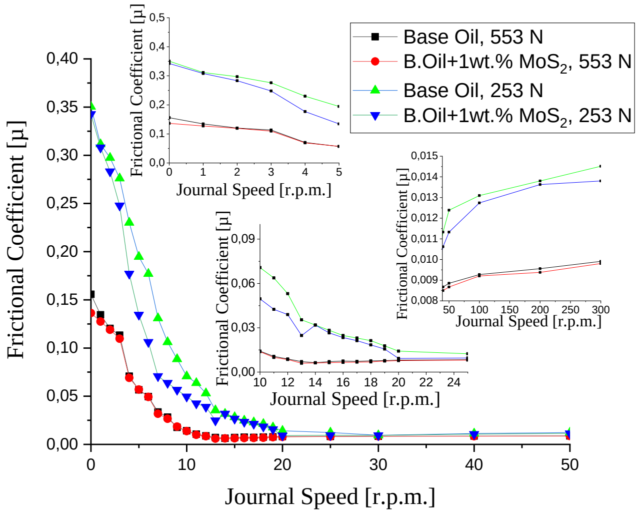

Figure 6. Variation of frictional coefficient with rotational speed in two different oil and load tests

Using the MoS_2 particles as additives, the effect of oil temperature on the friction torque is given in Fig. 8. As shown in Fig. 8, higher friction torque values were obtained at higher oil temperatures in the tests.

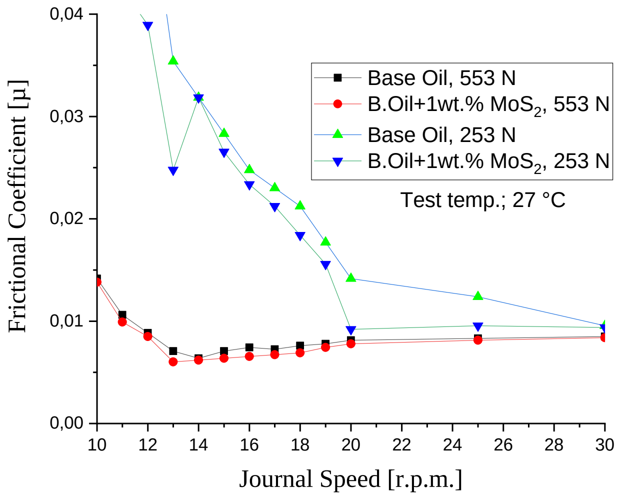

Figure 7. Variation of friction coefficient with rotational transition speed values in two different oil and load tests

Compared to the base oil, tests using oil with MoS_2 additives reveal lower temperature differences. This difference decreasing with the load As a result, it has been determined that the effect of surface As In the fluid friction region, the friction torque in the bearing decreases with the increasing oil temperature. While the friction torque decreases due to increasing temperature, the transition speed is increases. With the increase of temperature in the fluid friction region, the friction torque also decreases, while the transition speed increases. As a result, it has been determined that the effect of oil temperature is effective in terms of friction torque under metal-to-metal contact operating conditions.

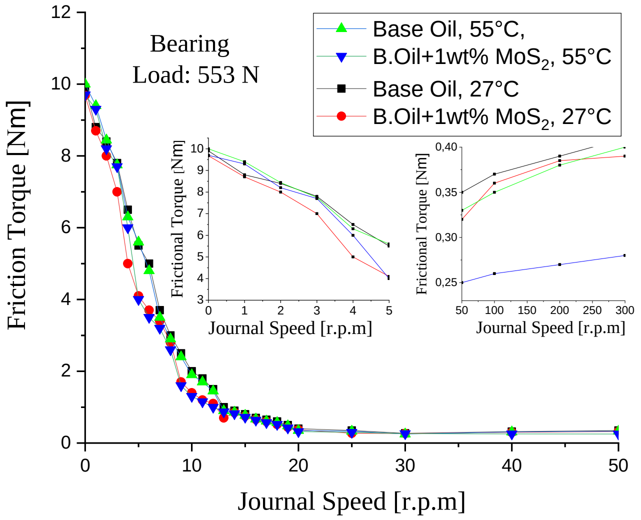

Figure 8. Variation of friction torque with speed using two different oil and temperature tests

The effects of oil temperature with different oil (base oil and 1 wt.% MoS_2 additive) on the friction coefficient were given in Fig. 9. As shown in Fig. 9, higher friction coefficient values were obtained in the base oil tests compared to the MoS_2 additive oil at 55 °C test conditions. These differences were more in the boundary and mixed friction regions. It was seen that the friction coefficient values formed in the bearing decreases with the increasing bearing load at high working oil temperatures, 55°C. The reason for the decrease may be that compared to the friction force the increase in the bearing load is higher, or the internal friction of the oil decreases as a result of the thinning of the oil film between the shaft and bearing surfaces. Hence, the surface roughness was effective in terms of friction coefficient under metal-to-metal contact operating conditions.

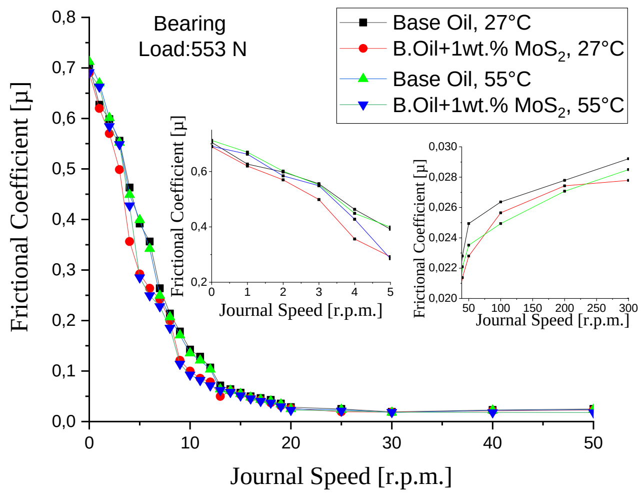

Figure 9. Variation of frictional coefficient with speed in two different oil and temperature at 553 N

Additionally, variation of frictional coefficient with transient speed in two different oil and temperature at 553 N test were detailed in Fig.10.

The rotational transition speed values decrease with decreasing oil temperature for both of MoS_2 additives and base oil. But using MoS_2 additives these decreases were more than that of the base oil. It can be said that, oil viscosity decreased with increasing temperature, and oil film formation becomes easier. Friction coefficient reaches to the minimum value when transient rotational speed has occurred. Using the base oil at 553 N, the values of transition speed and frictional coefficient were 30 rpm and 0.018, respectively, at the 55°C test condition. With the use of MoS_2 additives at the same oil temperature, friction coefficient value decreased to 0.016. Using the base oil at 27 °C test condition, friction coefficient and rotational transition speed values decreased and these values were 0.021 and 30 rpm, respectively. Using MoS_2 additives these values were 0.017 and 20 rpm, respectively. Hence, increasing with oil temperature, oil film formation between shaft and bearing becomes more difficult.

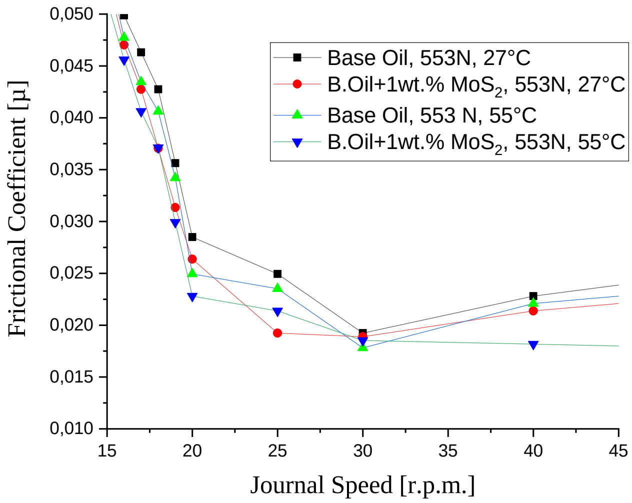

Figure 10. Variation of frictional coefficient with transient speed in two different oil and temperature at 553 N

4. Conclusion

The aim is to further reduce friction by adding inorganic compounds as additives to the engine oils used between the shaft and the bearing. The results of this study show that all the influential factors such as base oil, additive oil, test pressure and test speed have a significant effect on controlling the frictional behavior of thin-walled plain bearings. In this study, by adding 1% by weight Molybdenum Disulfide (MoS_2) to the base (Shell Tellus 10) engine oil, the effects of the MoS_2 additives on the bearing performance were examined and the following results obtained.

1. Compared to the base oil, the MoS_2 additives exhibits less friction behavior at increasing bearing load and temperatures.
2. MoS_2 additives is more effective in the mixed friction region than that of liquid film region. As a results, it is possible to evaluate whether tribofilm formation may occur in boundary or mixed lubrication on the surfaces of the shaft and bearing.
3. The friction reduction effect of MoS_2 additives are increasing with higher bearing load.
4. As a result of these tests, owing to MoS_2 additives facilitates the formation of oil film between the bearing

and shaft, it is concluded that MoS_2 additives reduces wear and increases journal bearing life.

Conflicts of interest

The authors declare no conflicts of interest.

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