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EXPERIMENTAL STUDYING THE EFFECT OF WATER TEMPERATURE ON FRICTION PROPERTIES OF MARINE PROPELLER SLIDING BEARING

Hiep Xuan Trinh[1,*], **Ngoc Bich Nguyen**[1], **Sinh Truong Nguyen**[1]
[1]*Le Quy Don Technical University, Hanoi, Vietnam*
*E-mail: hieptx@mta.edu.vn

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Abstract. This paper presents the effect of water's temperature on the friction properties of materials used in marine propeller sliding bearing. Copper-Rubber and Copper-Capron, two common pairs of material in the shaft water-lubricated polymer bearing were chosen to conduct experiments with the pin-on-disc model. Various conditions including water temperature, stress, and sliding velocity were examined, their results showed that in the range 30 °C to 100 °C of water temperature, the frictional coefficient of both friction pairs were unchanged under the small stress and low sliding velocity (0.3 MPa and 0.9 m/s). While in the case of stress and sliding velocity were both high (0.6 MPa and 1.5 m/s), it increased significantly in a certain transition temperature range. This temperature range of the pair Copper-Rubber and Copper-Capron is 50 °C to 60 °C and 80 °C to 90 °C, respectively. The experiment's results also pointed out that in these transition temperature ranges, the friction coefficient of two pairs was slightly influenced by the change in sliding velocity, whereas the stress change has an important impact on its values. Nonetheless, when the water temperature was below the transition range, the effect of the stress change on the friction coefficient was not significant. Thus, high water temperature is the main reason for the friction coefficient's increase rather than the increase of the stress. This work is expected to broaden the understanding of the friction behavior of the water-lubricated polymer bearing.

Keywords: tribology, friction coefficient, marine sliding bearing, polymer material.

1. INTRODUCTION

In the marine ship's structure, the propeller shaft plays an important role. It has a function to transmit power and torque from the motor to the propulsion device and inversely, receives the propeller thrust to push the hull in a certain direction [1,2].

Fig. 1 shows a schematic of the propeller shaft-driven system, the propeller shaft is placed on a pair of sliding bearings. In these bearings, the shaft sleeve is contacted directly to the bearing bush under liquid lubricated conditions by water or oil. Generally,

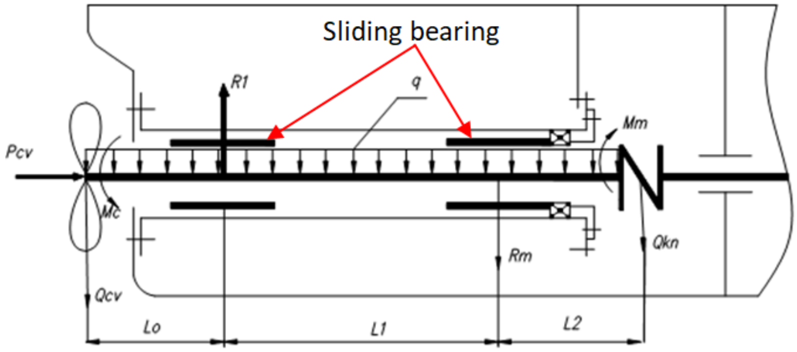

Fig. 1. Schematic of marine propeller shaft driven system

the shaft sleeve is made of Copper alloy such as Babbitt alloys while the bearing bush is fabricated by Woods, polymer materials as Rubber, Capron, and Textolit [3]. Recently, the shaft water-lubricated polymer bearings have been more and more popular application, thanks to its following advantages: simple structure, easy operation, low cost, and environmentally friendly in a comparison with the classical oil-lubricated bearings [4–6]. The structure of sliding bearing has a great influence on the performance of drive power [7]. The failures of sliding bearings such as the wear of bush or shaft sleeve and blocked shaft due to overheating are the main reasons for the marine propeller's damage [8, 9]. Many studies focused on the bearing's structure optimization to enhance the efficiency and service life of the propeller shaft-driven system. For instance, in [7], Litwin investigated the influence of the main design parameters as choosing a suitable bearing clearance, designing optimum bush geometry, and selecting a proper bush material on the life-time of ship propeller shaft water-lubricated bearing. The author in [10], proposed the new composite seal for better performance of marine bearing technology. The tribology behavior of water-lubricated bearing was also studied in [11–13], the authors mainly focused on the evaluation of friction coefficient and wear with different properties of materials such as the thickness, hardness of compound rubber [11], the physical properties of water [12] and the component of bronze-graphite composites [13]. These investigations were conducted under normal temperatures. Because almost bearing bushes are made of polymer materials with poor heat absorption, the heat generated in the contact zone is mainly absorbed by the lubricating flow. In the marine ship operation, the bumping system maybe is clogged, leading to the lack of lubrication flow and insufficient cooling, resulting in increasing the temperature in the sliding bearing. Consequently, the overheating phenomenon will occur in the contact zone, causing the change in the tribological manner which is followed by some sliding bearing failures like the rapid wear process of polymer bushing. Hence, the effect of increased water temperature on the tribological behavior of the water-lubricated polymer bearing needs to address thoroughly. Many publications reported the effect of temperature on the tribological behavior of polymer materials [14–16]. Though, in these researches, the authors considered the tribology manner with other lubricant conditions. In terms of the marine sliding bearing's operation under the lack of flowing water, Litwin and Dymarsky conducted experimental research to identify how different bearing types operate under the condition of no lubricant flow, by investigating the increased temperature of polymer bush under the water temperature rising [17]. However, the effect of temperature on the friction coefficient of the water-lubricated polymer sliding bearing has not been considered yet, especially with Rubber and Capron as bearing bush material. The issues that how the frictional coefficient changes as well as the effect of contact stress and sliding velocity under increasing water temperature in the polymer sliding bearing are still open questions. Finding the answers to these questions is the purpose of this study.

In this paper, we present experimental research on the influence of water temperature on the friction coefficient with the different contact stress and sliding velocities in the propeller polymer sliding bearing. The remainder of this paper is organized as follows. In Section 2, the experimental specimens and the experiment setup, data acquisition is described. Section 3 presents the experimental results and discussion. Finally, our paper is concluded with a description of future work.

2. MATERIAL AND METHOD

In this research, the experiments were carried out on a pin-on-disc model under various testing conditions. The pin is an upper specimen, fabricated by Babbitt alloy that is the most popular Copper alloy to make the shaft sleeves [3]. The disc is fabricated with Rubber and Capron, these are widely used polymer for the bearing bush. The disc samples are cut from real bearings bush of the propeller shaft system (as Fig. 2).

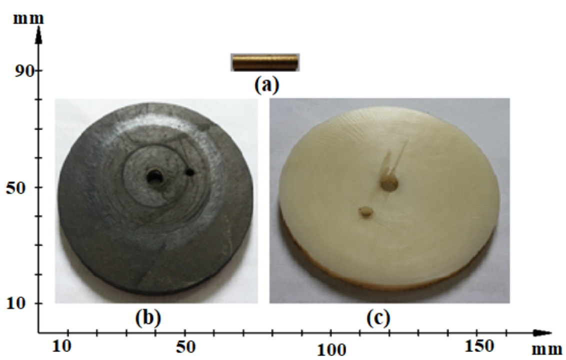

Fig. 2. Image of fabricated specimens (a) Copper pin, (b) Rubber disc, (c) Capron disc

The dimension of the pin is $\Phi 4.6 \times 20$ mm cylinder, and that of the disc is $\Phi 70 \times 5$ mm. The images of the upper and bottom specimens (pin and disc) are shown in Fig. 2.

Fig. 3 shows a schematic of the friction test using pin-on-disc mode. The experiments were conducted on the Universal Micro Material Tester (UMT) system (CETR-USA)[1], which can be used effectively for the tribological testing of various materials and accommodates both the upper and lower samples of practically any shape.

[1] https://www.bruker.com/products/surface-and-dimensional-analysis/tribometers-and-mechanical-testers/umt-tribolab/overview.html

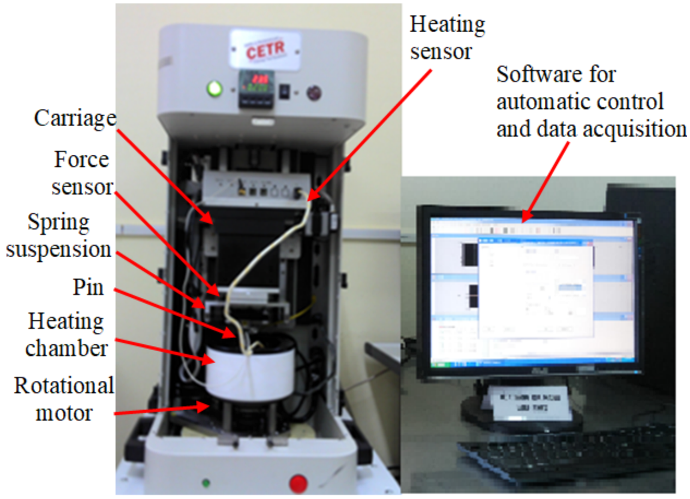

Fig. 3. Experimental setup

The upper specimen was attached to the spring suspension through an adapter. The suspension was added directly to the 2D force sensor with a measured range of 0.5 N to 50 N and 1 mN revolution. A block of the upper specimen, an adapter, suspension, and force sensors, was connected to a carriage, and then the carriage was connected to a vertical linear motion system that provided the precise movement with 5.10^{-5} mm position revolution. The load sensor provides feedback to the vertical motion controller and actively adjusts the sample position to ensure a constant specific load during testing.

The lower specimen is the disc, which is connected to a rotational motion drive that can provide various rotational speeds. The disc is fixed to the specimen table of the lower rotational motor by a screw. The specimen table is placed in a closed chamber, enabling the disc is submerged in water lubrication. The chamber also allows the water temperature changes from the environment temperature to up 150 °C by an integrated heated device. The temperature value is set up, controlled, and recorded automatically through a temperature sensor. The experiments have fully automated PC-based motor-control and data-acquisition with software so the testing data can be acquired, calculated, displayed instantly in real-time, and saved.

Before testing, the contact surface of the upper and bottom specimens was polished with abrasive papers. The following experimental data is collected: Fz - Normal force, Fx - Friction force, COF - Friction coefficient, TR - water temperature, Z - Carriage position. The contact load is setup though the value of normal force Fz, the corresponding stress σ can be calculated as the following:

$$\sigma = Fz/s, \tag{1}$$

where s is the cross-sectional area of the pin.

The friction coefficient COF is automatically calculated by the software using the measured values of friction force Fx and normal force Fz as the following equation:

$$COF = Fz/Fx. \tag{2}$$

The value of sliding velocity v is determined as below:

$$v = \omega r, \tag{3}$$

where ω is the rotation speed of the discs, that is set up automatically by the experiment system. r is the distance between the disc center and the contact position of the pin and the disc (Fig. 4).

The values of the normal force and rotation speed were set up based on the magnitude of the stress and sliding speed in the real working conditions of sliding bearings [4, 6, 11].

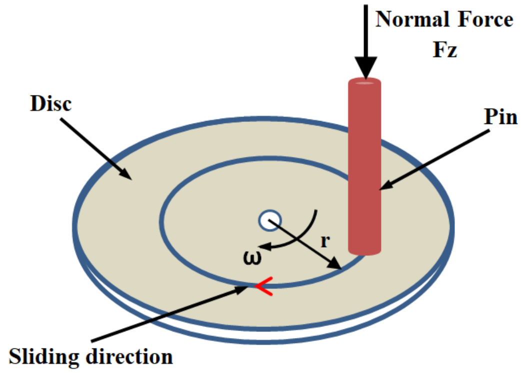

Fig. 4. Schematic of Pin-on-disc model

3. EXPERIMENTAL RESULTS AND DISCUSSION

The experiments were organized into three stages. The first stage was focused on finding the conditions, including the stress, sliding velocity, and transition range of temperature that cause the significant change in the value of friction coefficient. In the next stage, the experiments were conducted under the transition temperature range with constant values of either the stress or sliding velocity, and the other was varied to clarify their role on the change of friction coefficient. Finally, in the third stage, the temperature was remained below the transition range, and the stress value was high. This stage was conducted to highlight the effect of temperate on the value of friction coefficient. The experiment results of each stage are presented and analyzed as follows:

The experiment results of the first stage are shown in Figs. 5–8. Fig. 5 shows the result data of the friction pair Copper-Rubber under the experimental condition 1 of normal force 5 N (corresponding stress of 0.3 MPa), rotation speed 300 rpm (corresponding sliding velocity of 0.9 m/s), the water temperature increases from 30 °C to 100 °C.

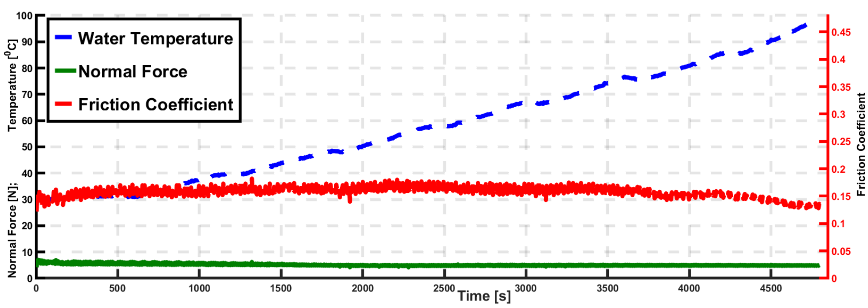

Fig. 5. Experimental results of friction pair Copper-Rubber under condition of stress of 0.3 MPa, rotation speed of 0.9 m/s, water temperature increases from 30 °C to 100 °C

In experimental condition 2, the normal force is 10 N (corresponding stress of 0.6 MPa), rotation speed 500 rpm (corresponding sliding velocity of 1.5 m/s), the water temperature increases from 30 °C to 100 °C. The result data of the friction pair Copper-Rubber is shown in Fig. 6.

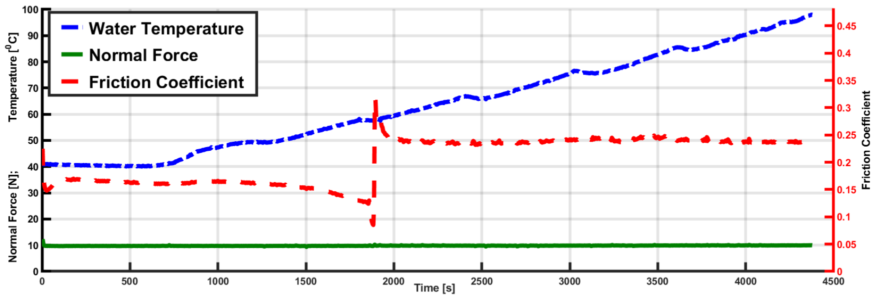

Fig. 6. Experimental results of friction pair Copper-Rubber under the condition of stress of 0.6 MPa, the rotation speed of 1.5 m/s, water temperature increases from 30 °C to 100 °C

Similarly, the friction pair Copper-Capron was also investigated with the same above conditions, and experimental results are shown in Figs. 7 and 8.

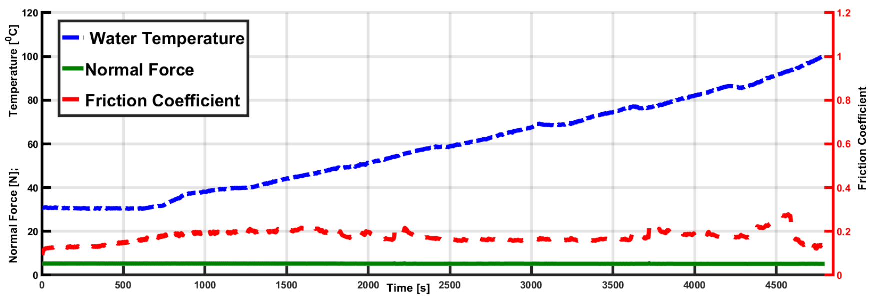

Fig. 7. Experimental results of friction pair Copper-Capron under the condition of stress of 0.3 MPa, the rotation speed of 0.9 m/s, water temperature increases from 30 °C to 100 °C

With the friction pair Copper-Rubber, the results in Fig. 5 shows that under the experimental conditions of the lower magnitudes of stress (0.3 MPa), sliding velocity (0.9 m/s), when the water temperature increases from 30 °C to 100 °C, the value of friction coefficient is relatively stable. Whereas Fig. 6 presents that with the conditions of the higher stress (0.6 MPa) and sliding velocity (1.5 m/s), the friction coefficient increases suddenly at a temperature range of 50 °C to 60 °C. As a result, the friction coefficient of the pair Copper-Rubber is not affected by the water temperature increasing from 30 °C to 100 °C if the magnitudes of stress and sliding velocity are small. Nonetheless, it is affected greatly at the temperature range of 50 °C to 60 °C with higher stress and sliding speed.

Similarly, with the friction pair Copper-Capron, the values of friction coefficient plotted in Fig. 7 are also not significantly changed. Whereas its value has a noticeable increase

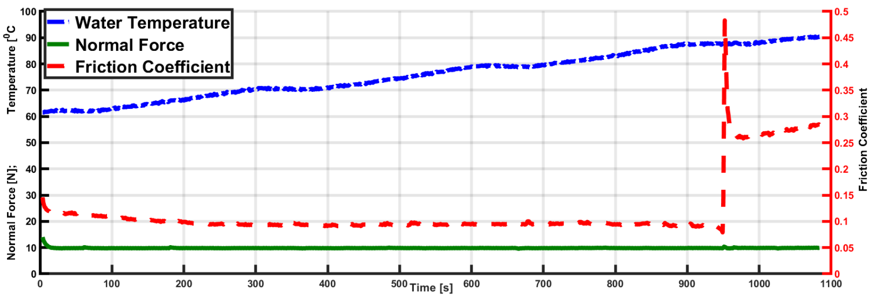

Fig. 8. Experimental results of friction pair Copper-Capron under the condition of stress of 0.6 MPa, the rotation speed of 1.5 m/s, water temperature increases from 60 °C to 100 °C.

at the water temperature range of about 80 °C to 90 °C. This indicates that under the small values of stress and sliding velocity, the friction coefficient of the pair Copper-Capron is also not affected by the increase of water temperature, which is similar to the pair of Copper-Rubber. Whereas with higher stress and sliding speed, it is significant changed at the water temperature range of 80 °C to 90 °C (Fig. 8). This phenomenon can be explained that under the high values of stress and sliding velocity when the temperature reaches to transition range, the molecular structure of the surface layer of the friction pairs was significantly modified, resulting in the major change of the contact manner and tribology's properties. The transition temperature range that causes the significant change of frictional behavior of the pairs Copper-Rubber and Copper-Capron is 50 °C to 60 °C and 80 °C to 90 °C, respectively. It also means the friction coefficient of the pair Copper-Capron is more thermal stable than that of the friction pair Copper-Rubber.

The average values of the friction coefficient of two friction pairs under various conditions are shown in Figs. 9 and 10. Those figures also indicate that the friction coefficient of two friction pairs has a noticeable increase at these transition temperature range. Besides, the friction coefficient values of the friction pair Copper-Capron are higher than that of the pair Copper-Rubber in the same conditions.

In short, the results of the first stage's experiment have confirmed that the friction coefficients of both friction pairs are increased under the transition temperature range with high values of the stress and sliding velocity. Then, to investigate the effect of each parameter, in the next experiment stage, we conducted more experiments with the change of either stress or sliding velocity under the transition temperature range of two friction pairs. The experiments were carried out with various values of stress and sliding velocity. Each test was repeated four times with a new set of specimens, and averages of four repeated tests were reported.

Firstly, to clarify the effect of sliding velocity, the normal load is constant at the high value of 10 N (corresponding stress of 0.6 MPa), and the rotational speed is varied with various values of 200 rpm, 300 rpm, 500 rpm, 700 rpm, 1000 rpm (corresponding sliding velocity of 0.6 m/s, 0.9 m/s, 1.5 m/s, 2.1 m/s, respectively).

Secondly, the role of stress on the change of friction coefficient is investigated by the experiments with the increased normal load from 5 N to 30 N (corresponding stress from

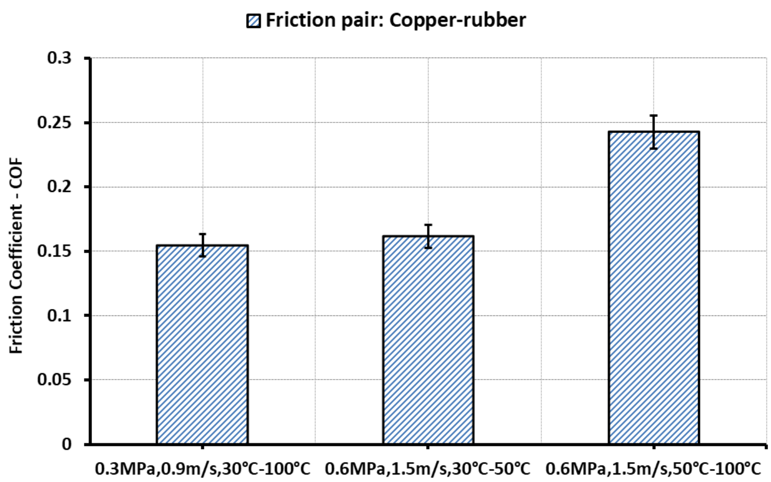

Fig. 9. The friction coefficient of friction pair Copper-Rubber under various conditions.

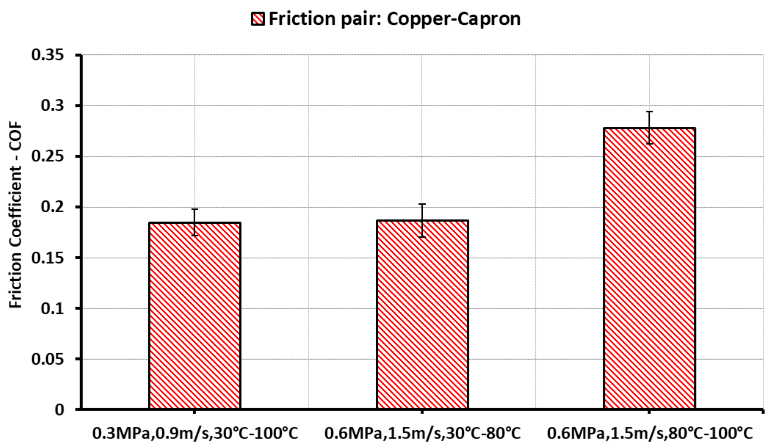

Fig. 10. The friction coefficient of friction pair Copper-Capron under various conditions.

0.3 MPa to 1.8 MPa) and the sliding velocity is unchanged at the magnitude of 1.5 m/s. These experiments are conducted under the temperature rises from 50 °C to 60 °C with the pair Copper-Rubber and 80 °C to 90 °C with the pair Copper-Capron.

The values of friction coefficient in the cases of the changed sliding velocity are presented in Fig. 11 and Fig. 12 shows it in the case of various stress. It can be seen that under the transition temperature range, both friction pairs have a relatively constant friction coefficient when sliding velocity changes. In contrast, it noticeably fluctuates in the case of stress increases. Specifically, the friction coefficient of the pair Copper-Rubber has a significant increase at the stress of 0.6 MPa, and then it rises sharply at the higher stress

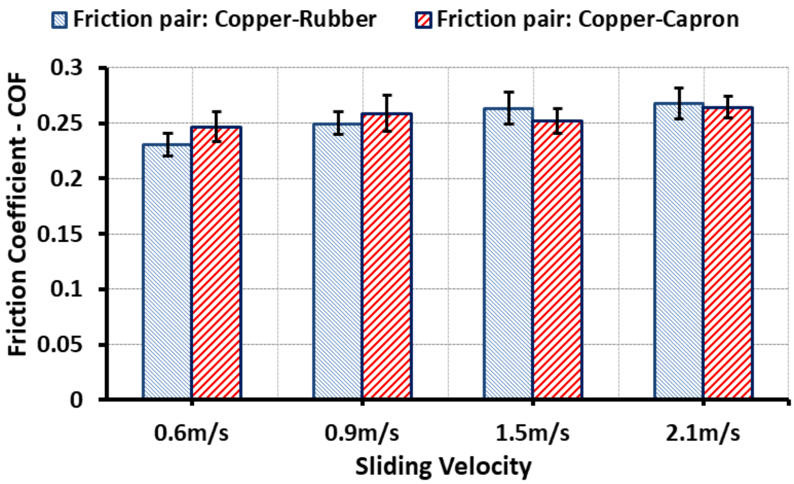

Fig. 11. The friction coefficient of friction pairs when sliding velocity increases, the stress of 0.6 MPa under the transition temperature range (From 50 °C to 60 °C of the pair Copper-Rubber and from 80 °C to 90 °C of the pair Copper-Capron)

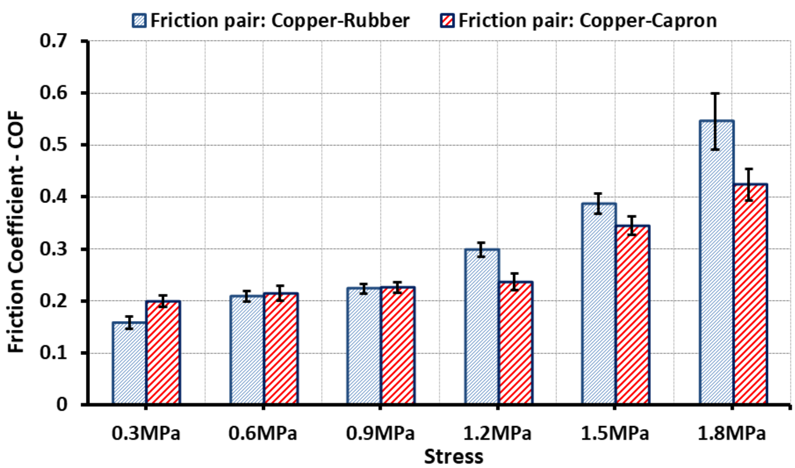

Fig. 12. The friction coefficient of friction pairs when stress increases, the sliding velocity of 1.5 m/s under the transition temperature range (from 50 °C to 60 °C of the pair Copper-Rubber and from 80 °C to 90 °C of the pair Copper-Capron)

of 1.2 MPa. With the pair of Copper-Capron, the friction coefficient just increases dramatically at the stress of 1.5 MPa. Thus, at the transition temperature range (from 50 °C to 60 °C of the pair Copper-Rubber and from 80 °C to 90 °C of the pair Copper-Capron) with both frictional pairs, the effect of sliding velocity's change on the friction coefficient is slight, while the stress is an important factor. The strong rise in the friction coefficient of Copper-Capron occurs at higher stress than Copper-Rubber pair.

From the results of the second experimental stage, we acknowledged that the rise of stress and temperature has a significant effect on the change of the friction coefficient. Then, to indicate which parameter is the main factor among those parameters, in the third stage, the experiments are conducted with conditions of the stress range from 0.3 MPa to 1.8 MPa and the constant sliding velocity of 1.5 m/s, while the water temperature is below the transition range. Specifically, its value is under 50 °C and 80 °C for the case of friction pair Copper-Rubber and Cooper-Capron, respectively. The experimental results with these conditions are shown in Fig. 13. The experimental data in Fig. 13 shows that with these conditions, the friction coefficients of both friction pairs are not changed significantly. It means with the water temperature is under the transition temperature values (less than 50 °C of friction pair Copper-Rubber and 80 °C of Cooper-Capron) the effect of the increased stress on the friction coefficient is not significant.

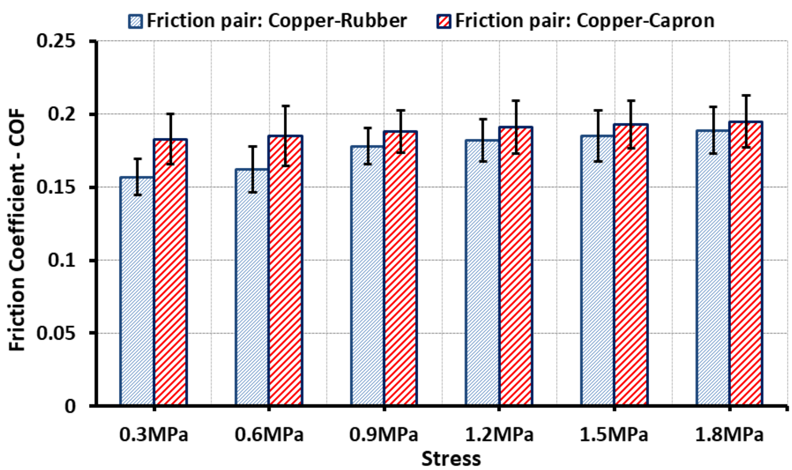

Fig. 13. The friction coefficient of friction pairs when stress increases, the sliding velocity of 1.5 m/s, with water temperature is under the transition temperature values (less than 50 °C of friction pair Copper-Rubber and 80 °C of Cooper-Capron)

Thus, from the data of Figs. 12 and 13, we concluded that the effect of high water temperature is the main reason for the friction coefficient's increase rather than the rise of the normal load.

4. CONCLUSIONS

In this paper, we investigated the effect of the water temperature on the tribology behavior of the propeller sliding bearing. Two friction pairs: Copper- Rubber, and Copper-Capron are chosen to conduct experiments with the pin-on-disc model. By analyzing the experimental results, we have the main conclusions.

Firstly, when the water temperature increases from 30 °C to 100 °C, the frictional coefficient of both friction pairs is stable under the condition of small stress (0.3 MPa) and sliding velocity (0.9 m/s), while it has a big increase at the transition temperature range under higher stress and sliding velocity (0.6 MPa and 1.5 m/s, respectively). And the transition temperature range that causes the significant change of frictional behavior of the pairs Copper-Rubber and Copper-Capron is 50 °C to 60 °C and 80 °C to 90 °C, respectively. In the same conditions, the friction coefficient of the pair Copper-Rubber is smaller than that of the pair Copper-Capron.

Secondly, in the transition water temperature range, the friction coefficient of both friction pairs is slightly influenced by the change of sliding velocity, whereas the stress change has the significant impact on its values. And under the transition temperature range, the stress value causes a significant increase in the friction coefficient with the friction pair Copper-Rubber and Cooper-Capron is 0.6 MPa and 1.5 MPa, respectively.

Finally, the effect of high water temperature is the main reason for the friction coefficient's increase rather than the rise of the normal load. Thus, the effect of water temperature on the friction properties must be considered in the design and operating of the water-lubricated sliding bearing, especially in the case of high-stress conditions.

In the future, we will focus on finding new materials to enhance the efficiency and working life of the propeller sliding bearing.

REFERENCES

[1] Y. Bai and W.-L. Jin. *Marine structural design*. Elsevier, (2016).

[2] J. Carlton. *Marine propellers and propulsion*. Butterworth-Heinemann, (2018).

[3] L. Roldo, I. Komar, and N. Vulić. Design and materials selection for environmentally friendly ship propulsion system. *Strojniški vestnik – Journal of Mechanical Engineering*, **59**, (2013), pp. 25–31. https://doi.org/10.5545/sv-jme.2012.601.

[4] W. Litwin. Marine propeller shaft bearings under low-speed conditions: water vs. oil lubrication. *Tribology Transactions*, **62**, (2019), pp. 839–849. https://doi.org/10.1080/10402004.2019.1625991.

[5] R. L. Orndorff. Water lubricated rubber bearings, history and new developments. *Naval Engineers Journal*, **97**, (1985), pp. 39–52. https://doi.org/10.1111/j.1559-3584.1985.tb01877.x.

[6] W. Litwin. Water-lubricated bearings of ship propeller shafts - problems, experimental tests and theoretical investigations. *Polish Maritime Research*, **16**, (2009). https://doi.org/10.2478/v10012-008-0055-z.

[7] W. Litwin. Influence of main design parameters of ship propeller shaft water-lubricated bearings on their properties. *Polish Maritime Research*, **17**, (2010), pp. 39–45. https://doi.org/10.2478/v10012-010-0034-z.

[8] G. Vizentin, G. Vukelic, L. Murawski, N. Recho, and J. Orovic. Marine propulsion system failures—A review. *Journal of Marine Science and Engineering*, **8**, (2020). https://doi.org/10.3390/jmse8090662.

[9] B. Primorac and J. Parunov. Review of statistical data on ship accidents. In *Maritime Technology and Engineering III*, CRC Press, (2016), pp. 809–814, https://doi.org/10.1201/b21890-106.

[10] A. Ford. New composite seal and bearing technology for better performance. *Wartsila Tech J*, **1**, (2012), pp. 34–39.

[11] Y.-Q. Wang, X.-J. Shi, and L.-J. Zhang. Experimental and numerical study on water-lubricated rubber bearings. *Industrial Lubrication and Tribology*, **66**, (2014), pp. 282–288. https://doi.org/10.1108/ilt-11-2011-0098.

[12] G. Gao, Z. Yin, D. Jiang, and X. Zhang. Numerical analysis of plain journal bearing under hydrodynamic lubrication by water. *Tribology International*, **75**, (2014), pp. 31–38. https://doi.org/10.1016/j.triboint.2014.03.009.

[13] G. Cui, Q. Bi, S. Zhu, J. Yang, and W. Liu. Tribological properties of bronze–graphite composites under sea water condition. *Tribology International*, **53**, (2012), pp. 76–86. https://doi.org/10.1016/j.triboint.2012.04.023.

[14] R. F. King and D. Tabor. The effect of temperature on the mechanical properties and the friction of plastics. *Proceedings of the Physical Society. Section B*, **66**, (9), (1953). https://doi.org/10.1088/0370-1301/66/9/302.

[15] A. Jean-Fulcrand, M. A. Masen, T. Bremner, and J. S. S. Wong. Effect of temperature on tribological performance of polyetheretherketone-polybenzimidazole blend. *Tribology International*, **129**, (2019), pp. 5–15. https://doi.org/10.1016/j.triboint.2018.08.001.

[16] F. Zheng, M. Lv, Q. Wang, and T. Wang. Effect of temperature on friction and wear behaviors of polyimide (PI)-based solid–liquid lubricating materials. *Polymers for Advanced Technologies*, **26**, (8), (2015), pp. 988–993. https://doi.org/10.1002/pat.3516.

[17] W. Litwin and C. Dymarski. Experimental research on water-lubricated marine stern tube bearings in conditions of improper lubrication and cooling causing rapid bush wear. *Tribology International*, **95**, (2016), pp. 449–455. https://doi.org/10.1016/j.triboint.2015.12.005.