Experimental Study on Water Lubrication Characteristics of Full-Flat Thrust Bearing with Partial Water-Repellent Surface*

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Abstract
In this study, a flat thrust bearing is developed by active exploiting the slip flow generated on a water-repellent surface. In order to produce a load, this bearing has a structure that generates a pressure flow by using the discontinuity of shear flow rate between a water-repellent surface and an untreated or a hydrophilic surface. This structure results in the completely flat bearing that has no geometrical variations in its surface. Friction tests were carried out with respect to the water-repellent thrust bearing that consists of three water-repellent parts and untreated or hydrophilic parts. The results verified that this partial water-repellent thrust bearing functioned well with a low and stable friction coefficient (less than 0.002). And lubrication surfaces had no damages at all because both surfaces are separated with fluid film which enables to avoid the solid contact. The friction coefficient became lower for the bearing having large difference of contact angle (thus, shear flow rate) between both parts. Furthermore, a friction coefficient for the bearing of which water-repellent part and hydrophilic part had same area was the lowest, and friction coefficient became larger than its value in case of not only wide water-repellent part but also narrow one. These results suggest a possibility that this thrust bearing operates by the same mechanism as a conventional bearing under fluid lubrication.

Key words: Slip Flow, Water-Repellent, Thrust Bearing, Water Lubrication, Friction Reduction, Glass

1. Introduction
In many sliding bearings not involving the approach of two surfaces, pressure is generated using the discontinuity of the shear flow rate, which is generated by arranging the thinner film thickness region in outlet direction of bearing. In particular, for parallel sliding surfaces such as those used in thrust bearings that supports the load by using two apparently parallel surfaces, the pressure cannot be generated without the arranging of convergent film region such as groove having spiral or herringbone configurations where film thickness becomes thin in the flow direction. However, pressure is expected to be generated, even for parallel sliding surfaces with smooth and flat surfaces, if there is a discontinuity in the shear flow rate in the flow direction. For example, the shear flow rate becomes discontinuous at the boundary between a sliding surface on part of which slip flows are generated and a surface where there is no or little slip flow; therefore, a bearing structure
can be formed for flat and dense glass or ceramic surfaces without a removal processing.

Researchers have examined the slip flow of low density fluids such as air for many years and have clarified that the slip velocity depends on the velocity gradient and the ease of slip at the surface. Slip flows on porous lubricating surfaces have been extensively studied. For example, stable low-frictional behavior was confirmed for a flat porous thrust bearing that porous surface on which slip flow easily occurs and dense surfaces on which slip flow hardly occurs were arranged alternately. The effectiveness of surface slip has been demonstrated by a theoretical calculation carried out with the aim of improving the characteristics of mechanical seals and journal bearings, and by reduced friction for a ring-on-plate equipment. However, the relationship between the increase in load capacity and the decrease in friction has not been experimentally confirmed to the best of our knowledge.

Surface slip flow is also observed on water-repellent (WR) and oil-repellent surfaces. Watanabe and coworkers determined the velocity distribution and measured the flow resistance in a rectangular duct with a highly WR surface having a static contact angle with a water drop of 140°. They demonstrated that there is slip on a highly WR surface with a large θ and that the slip velocity is proportional to the shear stress on the surface. They also reported that the reduction in the friction coefficient of a duct owing to the slip is as high as 22% for a square duct. In a friction examined by a setup having a gap between a highly-WR rotating disk (equivalent to that of the thrust bearing used in this study) and a fixed surface, a reduction in resistance of up to 45% was observed. The slip at the WR surface and the resulting reduction in friction are expected to enable the behavior as a bearing, realized at a mirror-smooth and flat parallel sliding surface, and to enable the operation of the bearing with low friction. Previously, we fabricated disk-shaped test pieces, in which three fan-shaped WR regions and three isometric fan-shaped nontreated glass regions are alternately placed (3-WR test pieces), and operated them under a water-lubricated condition, thus confirmed the feasibility of the test pieces.

It is considered that the characteristics of the fluid lubricated bearing investigated in this study are affected by the difference in shear flow rate between WR and nontreated region, if assuming that this bearing is operated in the fluid lubrication region by the above-mentioned mechanism. In this study, we focused on the fact that the ease of slip on the bearing surface depends on θ and we examined its effect by using various types of 3-WR test piece of which the difference in the shear flow rate of two regions was varied by changing the θ in WR regions. With this bearing structure, no effective pressure is expected to be generated when the entire surface of the test piece is untreated (or hydrophilic treated) or WR, because there is no difference in the shear flow rate along the sliding direction. Therefore good bearing characteristics are considered to be obtained only for the sliding surfaces with partial WR region such as those of the 3-WR test pieces. In this study, we confirmed this phenomenon by using test pieces of which the rate of area (β) comprising WR regions against the entire area of test piece is different, and determined the optimal β giving the best bearing characteristics. And then the operation mechanism of flat thrust bearings that have partial WR regions and their feasibility as the bearings are discussed based on those results.

2. Experimental equipment and test pieces

In the experiment a thrust bearing tester (Fig. 1), in which no pressure can be generated when test pieces that have not been subjected to partial WR treatment were applied, was employed. As shown in Fig. 1, the smooth and flat surface of a fixed test piece faces the smooth and flat surface of a rotating test piece fixed at the center of rotating water tank. The center of shaft for loading by a spring force is adjusted to agree with the center of rotating...
water tank. The fixed test piece which is coaxial to loading-shaft is pressed onto the rotating test piece by the shaft with the tip for pivot support, and the friction force is measured from the friction torque. The friction force \( F \) is defined as the tangential force acted to an equivalent radius \( R \) (\( R=13.3 \) mm in this test piece), at which the torque is equivalent to that calculated by summing all micro torques over the entire friction surface. The circumferential velocity of the rotating test piece with radius \( R \) is assumed to be the sliding velocity \( V \) (\( V=\frac{\pi RN}{1000} \), \( N \) is the number of rotations per second). The water purified by ion exchange was used as a lubricant. Before the experiment, the water tank and test pieces were cleaned and new purified water was supplied to the water tank as a lubricant.

The diameter and thickness of each fixed test piece are 40 and 5 mm respectively, and the pieces are made of soda-lime glass. As shown in Fig. 1, the fixed test pieces are 3-WR test pieces, in which three WR regions and three nontreated (NT) or hydrophilic treated (HT) regions are alternately arranged. The water repellency of the WR regions and the percentage of the WR region were varied in the experiment. A glass test piece for which no WR treatment was carried out (F-NT test piece) and a test piece for which WR treatment was carried out over the entire surface (F-WR test piece) were also used in the experiment for comparison. The rotating test pieces fixed on the rotating water tank were either F-NT test piece made of the same material as the fixed test piece, i.e., soda-lime glass, or the test piece for which hydrophilic treatment had been carried out over the entire surface (F-HT test piece).

A fluorine-based WR agent, perfluoroalkyl silazane, which has a low surface energy and relatively high water repellency, was used for the WR treatment. Fan-shaped regions excluding the WR regions on the ultrasonically cleaned test piece were masked with a weakly adhesive ultrathin sheet before processing. The test piece was then subjected to WR treatment using a spin coater and dried for 1 h. After the removal of the adhesive ultrathin sheet, the test piece was again subjected to ultrasonic cleaning in purified water and placed in a thermostatic chamber at 80°C for 1 h, then stored in a desiccator. The procedure for preparing the test pieces subjected to hydrophilic treatment instead of WR treatment is explained in section 4.

Figure 2 shows photographs of (a) a 3-WR test piece after exposure to humidified air, (b) a water drop on the WR region, and (c) a water drop on the NT region. In the WR region, \( \theta (=110^\circ) \) is approximately fourfold that in the NT region (\( \theta=30^\circ \)), demonstrating relatively high water repellency. Figure 3(a) shows a surface profile at the boundary between the NT and WR regions measured using a surface roughness meter. At this resolution, no difference in the surface roughness of the two regions is observed and the surface is considered to be
almost flat, including the boundary area. However, it has undulation of 0.2-0.3 μm in radial direction. When the surface was observed by atomic force microscope (AFM), many nanosize protrusions with an average height of 20 nm were observed on the WR region; however, the arithmetic mean roughness Ra was low and did not exceed 6 nm.

The sliding velocity V was set at 0.15 m/s so that the sliding surface was completely immersed in the lubricant in the rotating water tank even when a centrifugal force was applied to the lubricant. A fixed test piece was placed on a rotating test piece via the purified water to ensure a lubricating film that is sufficiently thick for use as a bearing. Only the weight of the fixed test piece and its holder (a total of 4 N) was applied to the rotating test piece. The rotation speed was increased to the predetermined sliding velocity V and then a predetermined load W was applied. W was in the range of 5-100 N (surface pressure, p_m=4-80 kPa) and was increased in a stepwise manner with increments of 5 N. The change in the friction characteristics over 5 min at different values of W and the surfaces of the test pieces after the experiment were observed. The temperature of the purified water was 24±1°C throughout the experiment.

3. Friction characteristics of each test piece

Figure 4 shows the friction behaviors of 3-WR, F-NT and F-WR test pieces for a light load of W=10 N. The friction coefficient of the F-NT test piece fluctuates significantly and is high (μ=0.02) even under this light load. The friction coefficient of the F-WR test piece is
lower than that of the F-NT test piece and fluctuates by 0.003. In contrast, the friction coefficient $\mu$ of the 3-WR test piece is $\approx 0.012$, approximately half that of the F-NT test piece, demonstrating reasonably stable friction behavior with limited fluctuation. The superior friction behavior of the 3-WR test piece was also observed when the load was doubled ($W=20$ N), as shown in Fig. 5. The friction coefficient of the F-WR test piece markedly increases to $\mu \approx 0.6$ after load application and fluctuates significantly.

Even at the F-NT test piece, it can be operated at a relatively low friction under a light load as explained above. It is considered that the pumping effect\textsuperscript{16} by the microscopic fluctuation in lubrication film thickness due to the undulation of disk surface (0.2-0.3 $\mu$m) is the major reason for this friction behavior, because the lubrication film thickness of F-NT test piece measured by ultrasonic technique is almost same in a whole area of test piece\textsuperscript{17}. However, operation at a relatively low friction is only possible when the load is small. As a result of the increased contact between solids caused by an increase in the load, the friction coefficient significantly increases ($\mu=0.2$), as explained later. A similar tendency was also observed for the F-WR test piece. However, the friction coefficient of the F-WR test piece at $W=10$ N, for which a lubricating film was ensured, was more stable and lower than that of the F-NT test piece owing to the slip at the WR surface.

Figure 6 shows the relationship between the bearing characteristic number and the average friction coefficient (bearing characteristic curves) for all the $W$ values examined in this study. In the figure, the friction coefficients of three F-NT, F-WR and 3-WR test pieces (a total of nine test pieces) are shown. The abscissa represents the bearing characteristic number $N/p_m$ ($\eta$ is the viscosity of the purified water, $N$ is the number of rotations per second, and $p_m$ is the average surface pressure). The superior characteristics of the 3-WR test piece can be observed; a low friction coefficient is maintained even under the severe operating condition of a low bearing characteristic number, demonstrating the improved
lubrication characteristics of the bearing with the proposed structure. For example, a friction coefficient of \( \mu \approx 0.004 \) is obtained at the maximum load \( W=100 \text{ N} \), and the friction behavior is highly stable for 3 min as shown in the inset of Fig. 6. This is considered to be attributable to the generation of pressure due to the discontinuity of the shear flow rate between the \( WR \) and \( NT \) regions.

The friction coefficient of the \( F-NT \) test piece is the highest among the three test pieces because the glass surface of the fixed test piece is in direct contact with the glass surface of the rotating test piece. Friction coefficient \( \mu \) markedly increased to \( \approx 0.2 \) at \( W=25 \text{ N} \), at which the experiment was stopped. The friction coefficient of the \( F-WR \) test piece is lower than that of the \( F-NT \) test piece and is similar to that of the \( 3-WR \) test piece when subjected to a low load. However, \( \mu \) increased to 0.6 at \( W=20 \text{ N} \), at which the experiment was stopped. It is the main reason of such phenomenon that \( F-NT \) and \( F-WR \) test pieces do not have a mechanism to generate the pressure, unlike in the \( 3-WR \) test piece.

Figure 7 shows photographs of the test pieces after the experiment shown in Fig. 6. The friction coefficient of the \( F-NT \) and \( F-WR \) test pieces markedly increased during load application and the surfaces of both the fixed and rotating test pieces were markedly damaged. In contrast, the surface of the \( 3-WR \) test piece was durable to \( W=100 \text{ N} \), which is fivefold the load applied to the \( F-NT \) and \( F-WR \) test pieces, and maintained the surface as

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**Fig. 6** Bearing characteristic curves

**Fig. 7** Photographs of test pieces after experiment
before the experiment and no trace of contact between two solid surfaces was observed. In addition, the $\theta$ values in the WR and NT regions of the 3-WR test piece were similar to those before the experiment shown in Fig. 2.

4. Influence of water repellency on friction characteristic

Compared with the friction coefficient of the $F$-NT and $F$-WR test pieces, the friction coefficient of the 3-WR test piece is low and stable over a wide range of bearing characteristic numbers. Figure 8 shows a schematic diagram of the velocity profile in a bearing with regions where slip is generated (slip region) and not generated (nonslip region), which are placed along the sliding direction. Pressures (flows) are generated by the discontinuity of the shear flow rate between the two regions as explained in the introduction. The slip velocity $U_S$ shown in the left region in Fig. 8 is the totalled velocity in the shear flow $U_S$ and the slip velocity $U_SP$, which is the pressure flow generated to the opposite direction for $U_S$. The magnitude of the generated pressure depends on the difference in the shear flow rate between the left slip region and the right nonslip region (in other words, the difference between the areas of the left trapezium determined by $U_S$ and the right triangle) because the generated pressure is determined so that the flow rate in the left slip region is equal to the flow rate in the right nonslip region. The larger difference in the shear flow rate generates the higher pressure, and it increases the thickness of the lubricating film required for supporting the same load.

![Fig. 8 Schematic diagram of velocity profile in bearing](image)

Static contact angle $\theta$ is approximately 30° in the NT regions of the 3-WR test piece, as shown in Fig. 2(c). The dynamic contact angle, which indicates the ease of movement of water drops on a surface, generally depends on $\theta$ which varies with a surface energy due to van der Waals force. So it is considered that the friction characteristic is improved as the difference in shear flow rate between two regions increases with decreasing of $\theta$ (increase...
of surface energy) in the outlet side in Fig. 8. Figure 9 shows the friction characteristic curves for 3-WR test pieces in which only the NT regions are hydrophilic treated by plasma processing to change them into HT regions with $\theta \approx 10^\circ$. In the figure, the friction characteristic curve for the 3-WR test piece with NT regions ($\theta \approx 30^\circ$) shown in Fig. 6 is also plotted by a dotted line for reference. For all three measurements, the friction coefficients of the 3-WR test piece with HT regions having high surface energy are lower than those of that with NT regions having lower surface energy (more slippy than HT surface) for a wide range of bearing characteristic numbers. The friction coefficient of the 3-WR test piece with HT regions at the maximum load, $W=100$ N, is less than half that of the 3-WR test piece with NT regions.

From the above findings, the friction characteristics are considered to strongly depend on the difference in the shear flow rate between the slip region and nonslip region (or difficult-to-slip region) or the difference in $\theta$ for the two regions. Next, we fabricated 3-WR test pieces with HT regions ($\theta \approx 10^\circ$) obtained by the above procedure and adjusted WR regions obtained by WR treatment and plasma processing to control $\theta$ in the WR regions, and the friction characteristics of the fabricated 3-WR test pieces with different values of $\theta$ in the WR region were compared. The procedure for fabricating the 3-WR test pieces was as follows. The surface of the glass disk was subjected to ultrasonic cleaning in purified water.
then hydrophilic treatment was carried out using a plasma ion bombarder. Then the entire surface was subjected to WR treatment using a spin coater and dried for 1 hour in a thermostatic chamber at 80℃. Surfaces with contact angle θ=30, 60, 90 and 110° were fabricated by changing the discharge current in the range of 0-8 mA (maximum applied voltage = 700 V) under the constant hydrophilic treatment time of 15 second at a gas pressure of 25 Pa. The three fan-shaped regions that were not hydrophilic treated were masked with a weak adhesive ultrathin sheet before processing, and hydrophilic treatment was carried out again to obtain the 3-WR test piece schematically shown in Fig. 10, and it was then stored in a desiccator after ultrasonic cleaning and drying. Here, no difference in the value of θ before and after masking was observed. In addition, F-HT test pieces (θ≈10°) were used for the rotating test pieces.

Figure 11 shows the behavior of friction coefficient for the four 3-WR test pieces (θ≈30, 60, 90, 110°) at (a) W=10 N, (b) 45 N and (c) 100 N. Lower friction coefficients were obtained for test pieces with larger θ. When W is 10 N, the friction coefficient of the 3-WR test piece with θ≈110° is one-fifth of that of the test piece with θ≈30°. No significant difference was observed between the friction coefficient of the 3-WR test piece with θ≈90° and that of the test piece with θ≈110°, however, the friction coefficient of the 3-WR test piece with θ≈110° at W=100 N is lower, stable and scarcely fluctuates, as shown in Fig. 11(c). Figure 12 shows the friction characteristic curves of all the test pieces examined in the above experiment including those shown in Fig. 11. In the figure, good characteristics of the 3-WR test pieces with larger θ are observed over a wide range of bearing characteristic numbers.

The acceleration of water drops (30 mg) falling down on the surface of a polymer thin film tilted by 35° is lower for stronger hydrophilic surfaces with smaller static contact angle such as θ≈30 or 60° compared with the WR surfaces with larger one (θ≈90 or 110°). This fact indicates that slips scarcely occur on the surfaces with small θ and that the difference in the shear flow rate on the surface with θ≈10, 30 and 60° is small, leading to decrease in the load capacity. This may reduce the film thickness being required to support a load, and generate the contact between two solid surfaces.

Watanabe and coworkers measured the distribution of velocity in rectangular ducts and confirmed that slips were generated on a highly WR surface of a duct with θ≈140°, although slips were not observed on the surface of an acrylic duct with θ≈80°, demonstrating that θ can serve as an indicator of the ease of slip on a surface. Slips are considered to occur on the WR surfaces with θ≈90 or 110°, considering that the slip velocity is proportional to the
shear stress on the surface\(^{(13)}\) and that the surface shear stress observed in this experiment is about 1000 times larger than that reported by Watanabe et al.

The friction coefficient of the 3-WR test piece with \(\theta = 60^\circ\) scarcely fluctuated (Fig. 11) until the experiment was quitted by sudden increase of friction, unlike that of the F-NT and F-WR test pieces shown in Figs. 4 and 5. This test piece is considered to be under a nearly fluid-lubricated condition similar to the 3-WR test pieces with \(\theta \approx 90\) or 110\(^\circ\). Therefore it is thought that the friction coefficient becomes lower when the difference in \(\theta\) between WR and HT region is larger, as mentioned above.

![Fig. 13 Photographs of test pieces after experiment](image)

Figure 13 shows photographs of the test pieces after the experiment. When \(\theta\) was 60\(^\circ\), the friction sharply increased during loading and the surfaces of both the fixed and rotating test pieces were damaged. In contrast when \(\theta\) was 90 and 110\(^\circ\), the surfaces of the test pieces after the experiment were the same as those before the experiment, and no trace of contact between two solid surfaces was observed. This result indicates that the test pieces with \(\theta = 90\) and 110\(^\circ\) were operated under a fully fluid-lubricated condition. For the test pieces with \(\theta = 90\) and 110\(^\circ\), the \(\theta\) values in the WR and HT regions were hardly changed by the experiment.

5. Influence of area ratio of water-repellent region on friction characteristics

For the F-HT test piece (\(\beta = 0\)) and F-WR test piece (\(\beta = 1\)), no pressure is generated and load cannot be supported, because no difference in the shear flow rate is generated in the lubrication surfaces. When 0<\(\beta<1\), the proposed structure functions as a thrust bearing and there is an optimal \(\beta\) that produces the maximum load capacity in this range. Here, we used a unit bearing model, which does not consider the side leakage, having a slip region (corresponding to the WR region) and a nonslip region (corresponding to the HT region). And we examined the relationship between maximum pressure \(p_M\) (or load capacity) and WR area ratio \(\beta = (0.25, 0.5, 0.75)\) which is a percentage of water-repellent region to the total length of the unit bearing. Figure 14 shows only the positive pressure part in the slip and nonslip regions, and the shear and pressure flows shown separately. To investigate the balance between the flow rates in the slip region (entrance side) and the nonslip region (exit side) of the proposed bearing, only the dependence of the pressure flow on \(\beta\) should be considered because the difference in the shear flow rate between the two regions is not
affected by $\beta$. The pressure gradients of the two regions are the same at $\beta=0.5$ because the length of the slip region is the same as that of the nonslip region. This case is used as a reference and it is assumed that no slip caused by the pressure flow occurs on the entrance side (the effect of this slip is discussed later).

The equation $(p_M/\beta B-p_{M05}/0.5B)+(p_M/(1-\beta)B-p_{M05}/0.5B)=0$ consists for the difference between the flow rate in the entrance and exit sides at a certain $\beta$ and that of $\beta=0.5$. Here, $B$ is the length of the positive pressure region in unit bearing (half the length of the unit bearing), $p_M$ is the maximum pressure at an arbitrary $\beta$ and $p_{M05}$ is the maximum pressure at $\beta=0.5$. From this equation, $p_M/p_{M05}=4\beta(1-\beta)$ is obtained. For example, for a bearing with a long slip region ($\beta=0.75$), maximum pressure is $p_M=(3/4)p_{M05}$. This also consists for a bearing with a short slip region ($\beta=0.25$) when the influence of the slip caused by the pressure flow is not taken into consideration. However, in practice, there is a small influence of the slip caused by the pressure flow on the entrance side. At $\beta=0.25$, the pressure gradient on the entrance side is larger than that at $\beta=0.5$, and the load capacity is smaller because the flow rate in the bearing is balanced at a lower $p_M$ as a result of the increased pressure flow into the inflow direction due to the slip.

As explained above, the load capacity of the proposed thrust bearing depends on $\beta$ even when the difference in the shear flow rate between the slip and nonslip regions is the same. The thickness of the lubricating film required to support the same load is thickest at $\beta=0.5$ and it becomes thin with $\beta=0.75$ and $\beta=0.25$ sequentially. Therefore, the friction coefficient becomes highest at $\beta=0.25$ and it becomes sequentially lower as $\beta=0.75$ and $\beta=0.5$. Next, the influence of $\beta$ on the friction coefficient is examined with using three 3-WR test pieces ($\beta=0.25, 0.5$ and $0.75$; $\theta=110^\circ$ in the WR regions and $\theta=10^\circ$ in the HT regions).

Figure 15 shows the behaviors of the friction coefficient during 5 min under the condition of $W=10$ N, 50 N and 100 N for the above 3-WR test pieces. If only the ease of slip on the surface affected the friction characteristics, the friction coefficient of the test piece with a large slip region ($\beta=0.75$) would have been the lowest. However, the friction coefficient is highest at $\beta=0.25$ for all the values of above $W$, and it lowered as $\beta=0.75$ and then $\beta=0.5$ sequentially. This order is consistent with that in the above discussion. It is considered that the friction coefficient is lowest and most stable at $\beta=0.5$, at which the shear stress on the fixed surface is lowest owing to the increased thickness of the lubricating film. It is considered that the friction coefficient is lowest and most stable at $\beta=0.5$, at which the shear stress on the fixed surface is lowest owing to the increased thickness of the lubricating film. In addition, the friction coefficient at $\beta=0.75$ is lower than that at $\beta=0.25$ because the area of the slip regions at $\beta=0.75$ is large and the shear resistance is lower owing to the large area.

Figure 16 shows the friction characteristic curves of all the test pieces examined in the
above experiment including those shown in Fig. 15. Similar to the friction behavior shown in Fig. 15, the friction coefficient is highest when \( \beta = 0.25 \), and it becomes sequentially lower as \( \beta = 0.75 \) and then \( \beta = 0.5 \). However, the surface of the test piece at \( \beta = 0.25 \) after the experiment is not damaged, even though the friction coefficient is larger than that at other \( \beta \) and more fluctuated (Figs. 15 and 16). From this result, it can be judged that the difference of friction coefficient depending on \( \beta \) shown in Fig. 16 is attributable to the difference in the bearing characteristics under a full fluid lubrication condition.

6. Conclusions

The influence of the static contact angle \( \theta \) for water drop and of the percentage \( \beta \) of the area comprising water-repellent, for the friction characteristic of bearing with smooth and flat surface of which the water-repellent region and the nontreated (or hydrophilic treated region) were arranged alternately was investigated. As a result, the following points were clarified.
(1) In case of both test pieces for which the water-repellent treatment was carried out over the entire surface and it was not carried out at all, the friction coefficient was high and lubrication surfaces were markedly damaged, because the supply of the lubricant to lubrication surfaces was extremely insufficient.

(2) The friction coefficient of three fan-shaped water-repellent test pieces of which partial water-repellent was carried out demonstrated the low value (0.002) with considerably small fluctuation. Then they exhibit a good lubrication characteristic even under the severe operating condition of a low bearing characteristics number, and lubrication surfaces were similar to those of before the experiment without the damage.

(3) The friction of three fan-shaped water-repellent test pieces becomes lower when the difference in static contact angle $\theta$ between a water-repellent region and non-water-repellent region was large, and stable friction behavior was maintained.

(4) Frictional coefficient of this thrust bearing was different by water-repellent area ratio $\beta$ even if the difference in shear flow rate is the same. Then it became lower in order of $\beta = 0.25, 0.75, 0.5$, and lowest friction coefficient was demonstrated near $\beta = 0.5$ at which a water-repellent area was equal to a non-water-repellent area.

The influence of such $\theta$ and $\beta$ was able to be explained qualitatively from the continuity of the flow rate that the difference in shear flow rate between water-repellent region and non-water-repellent region and the pressure flow which varies with pressure gradient and wall surface slip were considered.

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