Friction Characteristics of Vane for a Balanced Vane Pump*

Yoshiharu INAGUMA**

This article describes the friction characteristics at a vane tip, which causes primary loss of torque in a balanced vane pump, through the measurement of the coefficient of friction at the vane tip using cylindrical test rings having various values of inner surface roughness. Especially, the influence of operating conditions such as operating pressures, rotational speeds and oil temperature on the friction characteristics is experimentally investigated. Under the situation that the coefficient of friction becomes lower by lessening the surface roughness in the region of less than 0.7 µmRz, the coefficient of friction increases with an increase in the pressure acting on the vane in this region. However, both a rise in oil temperature and a decrease in the rotational speed (the vane sliding speed) make the coefficient of friction higher independently of the surface roughness. In addition, a mathematical model for the friction characteristics at the vane tip under a fixed pressure condition is proposed by using the sliding speed of the vane and the oil temperature instead of the viscosity of oil as parameters. It can well simulate the relationship between the coefficient of friction and the operating conditions for each ring with a different value of the surface roughness.

**Key words:** Fluid power system, Hydraulic vane pump, Vane tip, Tribology, Friction, Operating condition

1. Introduction

A balanced vane pump has been widely used for a hydraulic power steering\(^1\) and a continuously variable transmission\(^2,3\) in a vehicle because it has an advantage of low pressure ripples and low noise in addition to its compactness and lightweight. However, the balanced vane pump generally has a disadvantage that the friction torque of vanes is significant\(^4\) because the vanes are always pushed to an inner surface of a cam ring by the delivery pressure. In order to improve the efficiency in such a system, it is desired to reduce the friction torque in the pump.

The author theoretically revealed how a ratio of the cam lift to the vane thickness and the friction at the vane tip affect the mechanical efficiency in the balanced vane pump\(^5\). In addition, reduction in the friction in the vane under a constant oil temperature was verified by smoothing the inner surface of the cam ring\(^6\). As a means to reduce the friction torque of the vane, to lessen the surface roughness of the cam contour was also attempted by using coated vanes in an actual vane pump\(^6\). For measurement of the coefficient of friction, a cylindrical ring with a constant inner radius is practical and several studies using vanes with or without coatings at the vane tip were reported\(^4,5,7-10\). However, the influence of the operating conditions, especially the oil temperature, on the friction characteristics at the vane tip has been unclearly investigated. In addition, the effectiveness of lessening the surface roughness for the reduction in the coefficient of friction should be closely investigated.

In this work, the influence of the operating conditions including the oil temperature on the coefficient of friction at the vane tip is clarified for cylindrical test rings with various values of inner surface roughness. Then, a practical mathematical model to simulate the changes in the coefficient of friction at the vane tip is presented for the test rings with the various values of the surface roughness.

2. Nomenclature

\[ b \] : rotor width (㎜)
\[ C_0 \] : characteristic constant (−)
\[ F_v \] : vane force \((=wbp)\) (N)
\[ N \] : rotational speed of rotor \((\text{min}^{-1})\)
\[ p_1 \] : inlet pressure of test apparatus (MPa)
\[ p_2 \] : outlet pressure of test apparatus (MPa)
\[ R_c \] : inner radius of cylindrical test ring (㎜)
\[ T \] : driving torque (Nm)
\[ T_0 \] : friction torque independent of \(p_1\) (Nm)
\[ T_v \] : friction torque of vane (Nm)
\[ v \] : sliding velocity of vane tip \((=R_c\omega)\) (m/s)
\[ v_0 \] : characteristic constant (m/s)
\[ w \] : vane thickness (㎜)
\[ z \] : number of vanes (−)
3. Experiment

For understanding the friction in sliding parts of a hydraulic pump, it is very important to measure the coefficient of friction by using parts with similar material and shapes to the actual pump parts under the same sliding condition. Figure 1 shows a cross-sectional view of the test apparatus for measuring the coefficient of friction at a vane tip. In the apparatus based on an actual vane pump, a cylindrical test ring with a constant inner radius is used instead of a cam ring. The rotor and the vanes are from the actual vane pump, and the rotor has ten vanes. Two side plates made of ferro-sinterd alloy without heat treatment have no port to communicate with pump delivery and have vane back pressure grooves to lead the delivery pressure to the bottom of the rotor vane slot. The vane tips slide on the inner surface of the test ring with the loads due to the delivery pressure.

Figure 2 shows the dimensions of the test parts. The inner radius of the test ring \( R_c \) corresponds to the small radius of the actual cam ring. In the previous study, it was found that the inner surface roughness affected significantly the coefficient of friction \( \lambda_0 \). Therefore, seven test rings having different values of the inner surface roughness were prepared in order to investigate the friction characteristics in detail. The test rings A to E were made of the same material as the actual cam ring and the test rings F and G were made of high-speed tool steel, the same as the vane. The test rings had inner surfaces finished by grinding or lapping. The surface roughness was measured in four parts per ring at right angles to the sliding direction of the vanes, and their values are presented in Table 1. The surface roughness becomes finer in alphabetical sequence for the test rings. The vanes with two kinds of thickness \( w \) were finished by barrel polishing to roughness of about 0.3 \( \mu m \) at their tips. Because the surface of the test rings and vanes were harder than Hv 650, its roughness hardly changed after the test.

![Fig. 1 Cross-sectional view of test apparatus](image1)

![Fig. 2 Specifications of test rings and vanes](image2)

### Table 1 Surface roughness of test rings

<table>
<thead>
<tr>
<th>Test ring</th>
<th>Surface roughness (( \mu m ) Rz)</th>
<th>Values at four sections (average)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>1.398, 1.468, 1.460, 1.520 (1.462)</td>
<td></td>
</tr>
<tr>
<td>B</td>
<td>0.546, 0.980, 1.460, 0.513 (0.697)</td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>0.550, 0.502, 0.477, 0.532 (0.515)</td>
<td></td>
</tr>
<tr>
<td>D</td>
<td>0.319, 0.369, 0.520, 0.398 (0.402)</td>
<td></td>
</tr>
<tr>
<td>E</td>
<td>0.177, 0.252, 0.257, 0.177 (0.216)</td>
<td></td>
</tr>
<tr>
<td>F</td>
<td>0.108, 0.124, 0.094, 0.136 (0.116)</td>
<td></td>
</tr>
<tr>
<td>G</td>
<td>0.116, 0.098, 0.055, 0.105 (0.104)</td>
<td></td>
</tr>
</tbody>
</table>

Figure 3 shows the experimental system. After a regulation of the inlet pressure at \( p_1 \), oil from a feed pump was introduced to the vane bottoms through the vane back pressure grooves of the side plate in the test apparatus. The inside of the test ring was filled with the oil of the outlet pressure \( p_2 \) equal to the atmospheric pressure. The vanes slide touching their tips to the inner surface of the test ring by \( p_1 \). The oil temperature \( \theta \) was measured at the inlet of the test apparatus. The properties of the hydraulic fluid, commercial mineral oil, are given in Table 2.

For the apparatus having the test ring with no cam lift, the driving torque \( T \) becomes the total friction torque of the shaft, rotor and vanes. In the test, the torque was measured twice at the same operating condition. After setting the pump speed \( N \) and the oil temperature \( \theta \) at a measuring point within \( \pm 10 \text{ min}^{-1} \) and \( \pm 3 \text{ °C} \), respectively, the driving torque (total friction torque) \( T \) was measured in the process to increase the inlet pressure \( p_1 \) and to decrease it again. During the measurement of \( T, p_1 \) was maintained within \( \pm 59 \text{ Pa} \).
0.02 MPa. Through the entire test, the maximum value of the difference between the two measured values of $T$ at the same condition was 0.08 Nm. In this work, the mean value of the two measured values was adopted as $T$.

### 4. Experimental results and discussion

#### 4.1 Coefficient of friction

Figure 4 shows examples of the relationships between the pressure $p_1$ and the total friction torque $T$ in the cases of the test rings A and G. The test ring A has a relatively rough surface roughness. Figure 4⒜ shows that $T$ has a linear increase against $p_1$ for each $N$. Then, for the change in $T$ against $p_1$, the Y-intercept, i.e. $T$ at $p_1=0$, is denoted as $T_0$. Then, $T$ is divided into two parts: $T_0$ independent of $p_1$, and $T_n$ dependent on $p_1$, as expressed by the following equation.

$$T = T_n + T_0 \quad (1)$$

In these components, $T_n$ becomes the friction torque of the vane pushed on the test ring inner surface. In the test ring G with an extremely fine inner surface, $T$ is considerably low compared with that of the test ring A at the same $p_1$, as shown in Fig. 4⒝. However, the change in $T$ against $p_1$ is non-linear. For the non-linear $p_1$-$T$ characteristics, the Y-intercept, $T_0$, should be determined by using $T$ in a low region of $p_1$, as shown in Fig. 4⒝. In this work, $T_0$ was determined through a linear approximation using three values of $T$ at $p_1$ less than 1 MPa. The maximum error between two measured $T$ for each value of $T$ is shown in Fig. 4. In the test ring G having the lowest friction characteristics in all the test rings, although the ratio of dispersion to $T$ becomes greater because of their low values, the mean values would have no problem to discuss $p_1$-$T$ characteristics for the test rings.

#### 4.2 Surface roughness

Table 2 shows the properties of oil (equivalent to ISO VG 32).

<table>
<thead>
<tr>
<th>Temperature (℃)</th>
<th>40</th>
<th>60</th>
<th>80</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density $\rho$ (kg/m³)</td>
<td>855</td>
<td>842</td>
<td>829</td>
</tr>
<tr>
<td>Viscosity $\mu$ (pPa·s)</td>
<td>0.0293</td>
<td>0.0153</td>
<td>0.0089</td>
</tr>
</tbody>
</table>

Figure 5 shows the relationship between $N$ and $T_0$ for the test rings A and G as examples. When the pump operates, both the vane force caused by the pressure $p_1$ and the centrifugal force of the vane act on the cam contour. The centrifugal force depends on the square of the rotational speed $N$ independently of the pressure. Using a method of dividing $T$ into $T_0$ and $T_n$ shown in Fig. 4, the friction torque of the vane due to the centrifugal force is considered to be included in $T_0$. When the centrifugal force is significant, $T_0$ would increase non-linearly with increasing $N$. The actual change in $T_0$ against $N$ was almost linear.

Because $T_0$ is important for the analysis of the coefficient of friction at the vane tip, the changes in $T_0$ according to the operating conditions, especially oil temperature, were investigated for each test ring. Figure 6 shows the relationships between $p_1$ and $T_0$ of typical five test rings at the rotational speed of the rotor $N$ of 1500 min⁻¹ for three
kinds of oil temperature $\theta$. To make it easily seen, Fig. 6 is divided into two. In all the test rings, only the test ring A with the average surface roughness greater than 1 $\mu$mRz has a linear change in $T_n$ against $p_1$, including the results at other $N$. It was already reported that a test ring with a finer surface roughness has lower $T_n$. However, it is a notable fact that $T_n$ increases with an increase in oil temperature for all the test rings including the test rings B and F.

By dividing $T$ into $T_0$ and $T_n$ the effect of the centrifugal force is considered to be eliminated in $T_n$. Then, $T_n$ can be expressed by the following equation.

$$T_n = \lambda zwR_c p_1$$  (2)

From the measured $T_n$, $\lambda$ can be reversely estimated as follows.

$$\lambda = T_n / (zwR_c p_1)$$  (3)

In this experiment, $\lambda$ means the average value of the coefficient of friction for ten vanes. Figure 7 shows the relationships between $p_1$ and $\lambda$ calculated from $T_n$ in Fig. 6 for the five test rings. The torque error of $\pm 0.04$ Nm corresponds to the error of $\pm 0.01$ at $p_1 = 1$ MPa or $\pm 0.002$ at $p_1 = 6$ MPa in $\lambda$. The magnitude of this error would enable to discuss the difference in the coefficient of friction for the various test rings, as shown in Fig. 7. Except for the test ring A, $\lambda$ becomes higher with increasing $p_1$. In particular, the test ring C with a surface roughness of about 0.5 $\mu$mRz has a distinct increase in $\lambda$ with increasing $p_1$. As the surface roughness becomes smaller, the increasing rate of $\lambda$ against $p_1$ becomes smaller. For all the test rings, it is clear that $\lambda$ increases with rising oil temperature. The same results were obtained at other rotational speed $N$. In the results shown in Fig. 7, it could be considered that the vane slides under the condition near the boundary lubrication for the test ring A and under the mixed lubrication for the other test rings.

When a test ring has an extremely fine inner surface i.e. the test ring G and makes a condition near hydrodynamic lubrication, an increasing rate of $\lambda$ against $p_1$ and the influence of oil temperature become small.
Although the previous study presented that the roughness of the sliding surface affected the coefficient of friction \( \lambda \), the influence of the operating conditions on the relationship between the surface roughness and \( \lambda \) was insufficiently clarified. Figure 8 shows the influence of the pressure \( p_1 \) on it under the condition of \( N = 1500 \text{ min}^{-1} \) and \( \theta = 60^\circ\text{C} \). From this figure, it would be interesting that \( \lambda \) increases with an increase in \( p_1 \) for the test rings B to E and that the change in \( \lambda \) becomes small for the test ring A with the roughest inner surface of 1.5 \( \mu \text{mRz} \) and the test ring G with an extremely fine inner surface of 0.2 \( \mu \text{mRz} \). This tendency appears remarkably at a high rotor speed or a low oil temperature.

![Figure 8](image)

**Fig. 8** Relationship between surface roughness and \( \lambda \) (Influence of pressure)

For the influence of the rotor speed \( N \) and oil temperature, the results under a high pressure condition, i.e. \( p_1 = 5.88 \text{ MPa} \), are shown. Figure 9 shows the influence of \( N \) on the relationship between the surface roughness and \( \lambda \) at \( \theta = 60^\circ\text{C} \) for five kinds of \( N \). In the range of the practical \( N \), \( \lambda \) becomes lower with an increase in \( N \) independently of the surface roughness.

![Figure 9](image)

**Fig. 9** Relationship between surface roughness and \( \lambda \) (Influence of rotational speed)

Figure 10 shows the influence of oil temperature on the relationship between the surface roughness and \( \lambda \). As seen from this figure, \( \lambda \) increases with rising oil temperature for each rotor speed \( N \) independently of the surface roughness. Also for the surface roughness greater than about 0.7 \( \mu \text{mRz} \), with which the vane slides under the condition near the boundary lubrication, \( \lambda \) remains constant despite an increase in the surface roughness for each oil temperature. The similar results were obtained in other cases of \( N \).

![Figure 10](image)

**Fig. 10** Relationship between surface roughness and \( \lambda \) (Influence of oil temperature)

For the friction at the vane tip, the oil film thickness at the vane tip formed by sliding of the vane would be an important factor. Based on the elastohydrodynamic lubrication (EHL) theory, the oil film thickness at the vane tip was calculated\(^{12, 13}\). According to the calculated results, the oil film became thinner with rising oil temperature (decreasing the viscosity of oil) or with a decrease in the sliding velocity of the vane\(^{12, 13}\). Furthermore, it was calculated that the coefficient of friction increased with an increase in oil temperature\(^{13}\). When the oil film becomes thin, the area of metal-to-metal contact between the ring surface and the vane tip increases. Then, it could be inferred that the coefficient of friction increases because of a decrease in the support caused by the oil film for the vane force. In a study on the friction loss of hypoid gears, it was revealed that a higher oil temperature of lubricant brought a greater increase in the temperature on the gear tooth surface due to an increase in the heat generated by the friction loss\(^{14}\).

### 4.2 Modeling of friction characteristics

From Fig. 7, it is found that the pressure and the oil temperature affect the coefficient of friction \( \lambda \). Furthermore, it could be understood from the difference in the slope of \( T_n \) against \( p_1 \) in Fig. 4 that the rotational speed of the rotor, namely the sliding velocity of the vane, affects \( \lambda \). Then, it is attempted to represent \( \lambda \) using the non-dimensional parameter \( S \), which is used to represent the friction characteristics in plain bearings and denoted as follows.

\[
S = \frac{\mu v}{(F_c/b)} = \frac{\mu v}{(wbp_1/b)} = \frac{\mu v}{(wp_1)} \tag{4}
\]

In Eq. (4), the oil temperature and the rotational speed of...
the rotor $N$ are taken into consideration as the viscosity of oil $\mu$ and the sliding velocity of the vane $v$, respectively.

Figure 11 shows the changes in $\lambda$ against $S$ indicated on a logarithmic scale. In this figure, the values of $\lambda$ for five kinds of test rings, five kinds of $N$ and three kinds of oil temperature are plotted all together. The data of $\lambda$ at the same oil temperature are related together using three kinds of lines. An obvious fact is that the change in values of $\lambda$ can be roughly classified by the test rings with a different surface roughness. In the cases of the test rings D, E and G with a finer inner surface, it might be possible to roughly represent the change in $\lambda$ against $S$ with one curve because of their much lower values as a whole. The representation of $S-\lambda$ characteristics using one curve, however, would be difficult for the test rings A and C.

![Figure 11 Changes in $\lambda$ against $S$ for various operating conditions](image)

Because the friction at high pressures is important in the actual pump, the data at $p_1 = 5.88$ MPa were selected from Fig. 11 and plotted again in Fig. 12. In comparison with Fig. 11, the figure becomes straightforward. However, it is considered to be still ineffective to represent the change in $\lambda$ using $S$ for the variations of $N$ and the oil temperature.

The previous study\(^*\) reported that the friction torque characteristics in hydraulic pumps including a vane pump could be accurately represented by the following equation.

$$
\Delta T = \frac{C_{\alpha} V_0}{1 + (\omega / \omega_0)} \left( 1 + C_\omega \frac{\theta - \theta_0}{\theta_0} \right) \Delta p + (C_\omega \mu + C_v) V_0 \omega + T_c
$$

In Eq. (5), $V_0$ is the theoretical pump displacement, $\Delta p$ is the pressure differential across the pump, and $C_{\alpha}, \omega_0, \alpha, C_\omega, C_v, C_\omega$ and $T_c$ are pump constants independent of $\omega$, $\Delta p$ and $\theta$. In addition, $\theta_0$ is a typical working oil temperature used as the standard in the test. By comparing Eq. (5) with Eq. (1) and changing $\Delta p$ to $p_1$, the following equations can be deduced.

$$
T_n = \frac{C_{\alpha} V_0}{1 + (\omega / \omega_0)} \left( 1 + C_\omega \frac{\theta - \theta_0}{\theta_0} \right) p_1
$$

$$
T_n = (C_\omega \mu + C_v) V_0 \omega + T_c
$$

In Eq. (6), $1 + (\omega / \omega_0)$ is an additional term representing the change in $T_n$ against $\omega$, and $1 + C_\omega(\theta - \theta_0)/\theta_0$ is an additional term representing that against $\theta$. At this time, $C_{\alpha} V_0$ in Eq. (6) corresponds to $\lambda_0 \omega^2 b R$, in Eq. (2). Seeing Eq. (2), the only changeable parameter is $\lambda$. Hence, by introducing $\lambda_0$ independent of $\omega$ and $\theta$ as a constant, and following Eq. (6), $\lambda$ can be rewritten as $\lambda^*$ as expressed in the following equation.

$$
\lambda^* = \frac{\lambda_0}{1 + (\omega / \omega_0)} \left( 1 + C_\omega \frac{\theta - \theta_0}{\theta_0} \right)
$$

$$
= \frac{\lambda_0}{1 + (v / v_0)} \left( 1 + C_\omega \frac{\theta - \theta_0}{\theta_0} \right)
$$

In Eq. (8), $v (= R \omega)$ and $v_0 (= R \omega_0)$ are finally used, because the use of the sliding velocity of the vane $v$ is better than that of the angular velocity of the rotor $\omega$ to discuss the friction.

Figure 13 shows the change in $\lambda$ for $p_1 = 5.88$ MPa. In this figure, not $S$ but the sliding velocity of the vane $v$ is used as the abscissa, and both $\omega$ and $N$ corresponding $v$ are also indicated. In comparison with Fig. 12, this figure makes it easier to understand the influence of the oil temperature on the relationship between $v$ and $\lambda$ for the test rings with various values of surface roughness. It was attempted to represent the changes in $\lambda$ for five test rings using $\lambda^*$ expressed by Eq. (8).

As seen in Fig. 13, Eq. (8) can also represent the change in $\lambda$ against the various values of $v$ and the oil temperatures for the various test rings. The result in the test rings B and F are also similar although they are not shown. Representing the change in $\lambda$ using $\theta$ is considered to be better than that using the viscosity of oil $\mu$. The values of constants in Eq. (8) to simulate the changes in $\lambda$ for the five test rings in Fig. 13

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are presented in Table 3. Regrettably, the equation to represent perfectly the change in $\lambda$ for all the operating conditions of $v$, $p_1$ and $\theta$ was not constructed because the increasing rate of $\lambda$ against $p_1$ changes complicatedly according to the inner surface roughness, as seen from Fig. 7. As a result, it was difficult to add the effects of the surface roughness and $p_1$ to the equation representing the change in $\lambda$. However, Eq. (8) can well represent the change in $\lambda$ under a fixed $p_1$ condition, and is very useful to understand the behavior of the coefficient of friction.

4.3 Friction Characteristics for extended vane force

A test ring with an extremely fine inner surface produces an excessively low value of the coefficient of friction $\lambda$. It would be interesting to know how the fine surface roughness keeps low friction for increasing the vane force due to the pressure. Therefore, by using the test ring F with a fine surface roughness of 0.116 µmRz (average), the influence of the increase in the vane force $F_v$ on $\lambda$ is investigated. By using a vane with a thickness $w$ of 1.8 mm, the vane force denoted as $F_v = wbp_1$ can be increased.

Figure 14 shows the change in $T_n$ against $F_v$ at oil temperature of 60°C for the vanes with a thickness $w$ of 1.4 and 1.8 mm. At $N=500$ and 1500 min$^{-1}$, the change in $T_n$ against $F_v$ are almost identical independently of $w$. At $N=3000$ min$^{-1}$, however, the values of $T_n$ for $w=1.8$ mm become much higher than those for $w=1.4$ mm at the same $F_v$ in the region of $F_v>130$ N.

Figure 15 shows the relationships between $F_v$ and $\lambda$ for the vane of $w=1.8$ mm. As seen from Fig. 15a) for $N=1500$ min$^{-1}$, $\lambda$ increases gradually with an increase in $F_v$ and the oil temperature $\theta$.

Figure 15b) shows the changes in $\lambda$ at $N=3000$ min$^{-1}$, and $\lambda$ becomes higher in the region of $F_v>130$ N for all the oil temperatures. Regrettably, the cause of an abrupt increase in

<table>
<thead>
<tr>
<th>Test ring</th>
<th>$\lambda_0$ (m/s)</th>
<th>$v_0$ (m/s)</th>
<th>$\alpha$</th>
<th>$C_v$</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>0.2004</td>
<td>6.680</td>
<td>0.2547</td>
<td>0.0731</td>
</tr>
<tr>
<td>B</td>
<td>0.1958</td>
<td>0.628</td>
<td>0.1612</td>
<td>0.0863</td>
</tr>
<tr>
<td>C</td>
<td>0.1134</td>
<td>1.362</td>
<td>0.3819</td>
<td>0.2649</td>
</tr>
<tr>
<td>D</td>
<td>0.1002</td>
<td>1.130</td>
<td>0.6439</td>
<td>0.6925</td>
</tr>
<tr>
<td>E</td>
<td>0.1389</td>
<td>0.068</td>
<td>0.4441</td>
<td>0.1560</td>
</tr>
</tbody>
</table>

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at only $N=3000$ min$^{-1}$ was not clarified. However, this fact suggests that a thinner vane should be used in order to maintain a low friction condition.

5. Conclusions

In this work, the friction characteristics of a vane for various operating pressures, rotational speeds and oil temperatures were experimentally investigated by using cylindrical test rings having various values of inner surface roughness. As a result, the following conclusions were drawn.

For the test rings with an inner surface roughness less than 0.7 $\mu$mRz, the coefficient of friction decreased with lessening the surface roughness but increased with increasing the pressure acting on the vane. For the test rings with a rougher inner surface greater than 1 $\mu$mRz, the coefficient of friction is independent of the pressure. With an increase in the oil temperature and a decrease in the vane sliding speed, the coefficient of friction increased independently of the surface roughness.

A proposed mathematical model using the sliding speed of the vane and the oil temperature instead of the viscosity of oil as parameters could well represent the friction characteristics at the vane tip for each test ring with a different value of the surface roughness.

In addition, in the test ring having low friction due to an extremely fine inner surface, the coefficient of friction at a high vane speed became higher at a greater vane force independently of the oil temperature.

References