The Lubrication Characteristics of the Vane Tip Under Pressure Boundary Condition of Oil Hydraulic Vane Pump

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The Lubrication Mode of line contacts between the vane and the camring in an oil hydraulic vane pump has been investigated. First, the variations of the radial force of a vane were calculated from previous measurements of dynamic internal pressure in four chambers surrounding a vane. Next, the lubrication modes were distinguished with Hooke's chart, which is an improvement over Johnson's chart. Finally, the influence of the boundary conditions in the lubrication region on the fluid film lubrication was examined by calculating the film pressure distributions. The results showed that the lubrication mode of the vane tip exists in the rigid-variable-viscosity region, and that discharge pressure higher than 7 MPa greatly affects the oil film pressure in the small and the large arc section because of the Piezo-viscous effect.

Key Words : Inlet Pressure, Outlet Pressure, Vane Pump, Vane Tip, Line Contact, Lubrication Mode

1. Introduction

In an oil hydraulic vane pump, the tribological characteristics of the vane tip is regarded as the most important region in the vane pump. For this reason, these characteristics are examined to design or construct a vane tip suitable for high velocity and pressure with improved durability. To analyze the tribological characteristics in this region, the normal acting force between the vane and the camring in relative sliding motion under high load at high speed conditions should be obtained. The overall acting force can be calculated with the values of hydraulic, inertial, and viscosity forces, obtained from experiments under various operational conditions. Therefore, when the specification of a pump is known, the normal force acting on the vane can be easily obtained.

Table 1 shows Johnson's chart, which classifies the lubrication regions of the line contact under concentrated load into four groups : R-I (Rigid-Isoviscosity), R-V (Rigid-Variable viscosity), E-I (Elastic-Isoviscosity), and E-V (Elastic-Variable viscosity). However, many regions have not been verified yet, because when the load is high and the viscosity effect is predominant, the solution of the analysis tends to diverge. In the contact region between the vane tip and the camring, the acting force should be small to reduce friction and wear,

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Table 1 Lubrication region of concentration load contact

Region	Elastic Effect	Pressure Viscous Effect	Applications
R-I	Insignificant	Insignificant	Roller bearing with low load, Circular arc type thrust bearing
R-V	Insignificant	Significant	Cylindrical bearing with medium load, Taper roller bearing, Piston ring-liner
E-I	Significant	Insignificant	Seal, Tire, Articulation
E-V	Significant	Significant	Ball bearing, Gear & cam

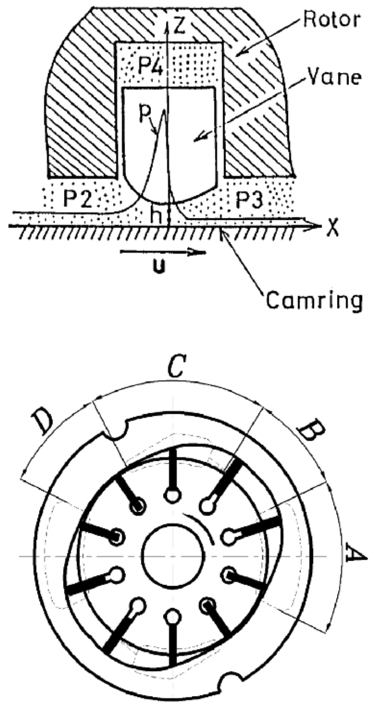
but the film thickness should be thin to reduce oil leakage. To overcome these conflicting conditions, there is one way to decrease contact area by reducing the curvature radius of the vane tip.

In this paper, using the pump series applied to D-2882 (standard test method for indicating wear characteristics of petroleum and non-petroleum hydraulic fluids in a constant volume vane pump) of ASTM (American Society of Testing Materials), we examined the lubrication modes between the vane tip and the camring. In addition, we wanted to ensure that the lubrication characteristics of the suction port, large arc region, delivery port and small arc region were adequate under regular operational conditions, and to understand the lubrication characteristics according to the inlet pressure boundary conditions, which exist in these regions.

2. Lubrication Modes of the Line Contact Region

The vane tip and the camring of an oil hydraulic vane pump contact each other in the form of a line. Figure 1 shows the cross sectional area of the vane pump. Line contact is mainly lubricated by Elasto-Hydrodynamic Lubrication (EHL), but for various of conditions, there are 4 lubrication modes, as shown in Table 1.

In the E-V region, the Reynolds' equation and the equation of the elastic deformation of the material are used simultaneously to analyze lubrication characteristics, but in doing so, the equations become non-linear so an analytical solution can not be obtained. However, the test formula obtained from numerical solutions under specific conditions can be obtained in a dimensionless form.



A : Suction port B : Large arc region
C : Delivery port D : Small arc region

Fig. 1 Line contact model of vane and position of vane chamber

In the application of the existing numerical test equation of EHL, obtained by substituting the E-V region with the R-V region or R-I region, or the R-I region with the R-V region, a large calculation error can occur. Therefore, when the mode is not known, the lubrication characteristics must be analyzed with the Reynolds' equation and the equation of elastic deformation, simultaneously, considering the pressure-viscous effect.

The normal acting force varies as the rotor revolves 1 cycle, from nearly zero to the maximum.

Therefore, vast calculation is required. If the calculation method is not changed according to the lubrication mode, the error can become large and the vast calculation could be useless when there is almost no elastic deformation or viscosity change.

3. Lubrication Modes of the Vane Tip

The lubrication modes of the vane tip was analyzed assuming that there is enough oil film with sufficient pressure around the inlet and the outlet, and that the surfaces of the vane and the camring are perfectly smooth, so that they are under a perfect hydrodynamic lubrication status. The pressure — viscosity coefficient, α, was $2.2\times10^{-8}\,m^2/N$ as that of a standard grade of mineral oil. The dimensionless elasticity and viscosity parameters, g_e, g_v, which represented by the horizontal and the vertical axis of Johnson's chart, respectively, were calculated by using the normal force acting on the vane, obtained from the operational conditions and the measured pressure of the pump (Hooke, 1977 ; Johnson, 1970). The lubrication modes of the standard vane pumps for the vane pump wear test of ASTM, presented in Table 2, were examined.

Figure 2 shows lubrication modes for each region of the straight vane type pumps, whose the standard specification is presented in Table 2 (Cho et al., 2005). As shown in the figure, all the pumps are gathered close to the line of $g_v = g_e^{1.5}$. All regions except the delivery port, which has no load, are in the R-V region, as well. However, the large arc regions deviate slightly from the line to the left (where g_e is small) of the suction regions. When operated near the line of $g_v = g_e^{1.5}$ with the half of the discharge pressure, g_e decreases to half, and when rpm is reduced to half, g_e decreases to the Root Mean Square (RMS) under regular operating conditions. Therefore, if the regular discharge pressure is set and the rpm is reduced to half, they approach the boundary between the R-V and the E-V region.

From the above, at the vane tips of all regions except the delivery ports with no load, lubrication

Table 2 Capacity of vane pump (straight vane)

No	Delivery flow (cm³/rev)	Re'f Pressure (MPa)	Max. rpm (rev/min)
1	25.8	7	1800
2	113.0	7	1500
3	16.8	14	1200
4	61.1	14	1200
5	153.0	14	1200

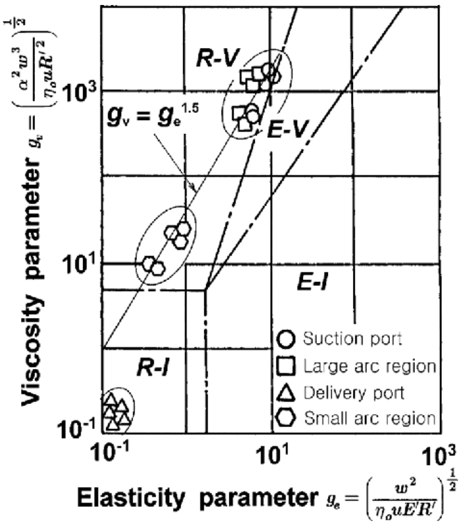


Fig. 2 Lubrication mode in a straight vane type pump

modes are not like the typical elastohydrodynamic lubrication (E-V region) of sliding bearings, gears and so on, and under such conditions, viscosity changes, and elastic deformation acts to a similar extent in the boundary region of the R-V and the E-V region of suction ports and large arc regions.

4. Effect of Pressure Boundary Conditions on the Large and the Small Arc Regions

The lubrication mode diagram presented in Fig. 2, used for distinguishing the lubrication modes of the vane tip, has been drawn based on the existing calculation of EHL, which was carried out under the assumption that the inlet and the outlet pressure are equal to the atmospheric pressure. In addition to replace these boundary conditions, the normal acting force is calculated based on the definition of the normal acting force in the work of Jung and Jung (1998). However, the pre-chamber pressure is the discharge pressure in a large arc region, and the post-chamber pressure is the discharge pressure in a small arc region. Therefore, when the vane comes to the large and the small arc regions, the results must be reviewed.

The effects of the pressure boundary conditions on the lubrication characteristics according to the lubrication modes under the above assumption are examined assuming the materials are rigid because the lubrication modes of the large and the small arc region are in the R-V region.

4.1 In case of the small arc region

Figures 3～6 are the calculation examples in which the inlet pressure is the suction pressure (atmospheric pressure) and the outlet pressure is the discharge pressure under the assumption that the pure normal load applied to the vane is fixed.

Figure 3 shows the influence of the outlet pressure on the pressure distribution of the lubrication film, when the pressure — viscosity coefficient $\alpha=0$. The slopes of both of the inlet and the outlet curves increase as the outlet pressure increases.

Figure 4 shows the influence of the outlet pressure on film thickness, when the pressure — viscosity coefficient $\alpha=0$. If the viscosity changes according to pressure are not considered, the slope of the pressure affects the film thickness which decreases as the slope of the pressure increases.

Figure 5 shows the influence of the outlet pressure on the pressure distribution of the lubrication film of the mineral lubricant ($\alpha=2.2\times10^{-8}$ m^2/N), which has the pressure — viscous effect. The slopes of both of the inlet and the outlet curves increase as the outlet pressure increases. And also, it is more effect of the slope of the pressure than that in Fig. 3. This is because the viscosity change due to the pressure — viscous affects on the slope of the pressure.

Figure 6 shows the influence of the outlet pressure on the film thickness, when the pressure — viscosity coefficient $\alpha=2.2\times10^{-8}$ m^2/N. When the

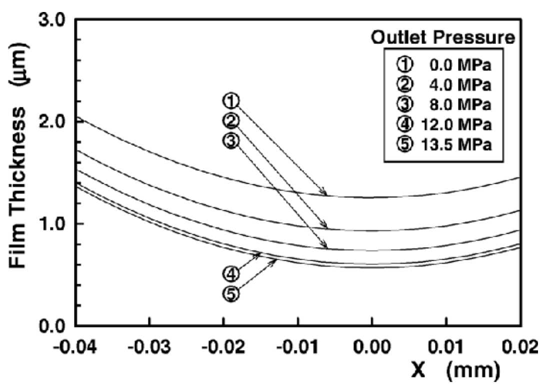


Fig. 4 Influence of outlet pressure on the film thickness (at $\alpha=0$)

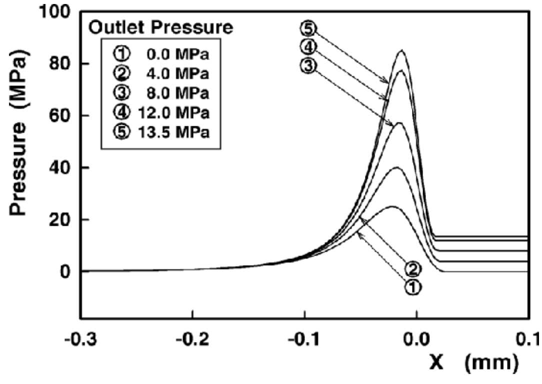


Fig. 3 Influence of outlet pressure on the pressure distribution of lubrication film (at $\alpha=0$)

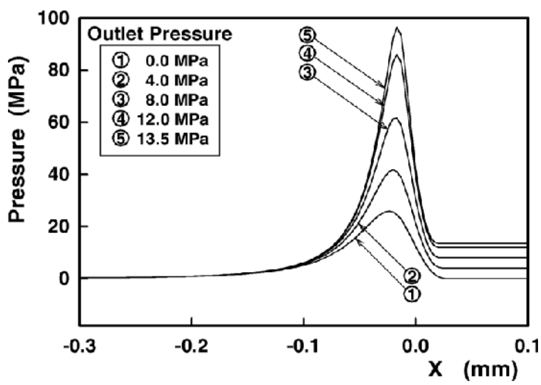


Fig. 5 Influence of outlet pressure on the pressure distribution of lubrication film (at $\alpha=2.2\times10^{-8}$ m^2/N)

viscosity changes due to pressure is considered, the film thickness decreases on the whole because the viscosity increases as the pressure of the fluid film increases, and when the film thickness does increase, the increase is relatively less than the amount of decrease because the increase in the slope of the pressure.

4.2 In case of the large arc region

Figures 7～10 are the calculation examples in which the inlet pressure is the discharge pressure and the outlet pressure is the suction pressure (atmospheric pressure) under the assumption that the pure normal load applied to the vane is fixed.

Figure 7 shows the influence of the inlet pressure on the pressure distribution of lubrication film, when the pressure — viscosity coefficient $\alpha=0$. The slope of the inlet pressure is almost the same as the inlet pressure but the slope of the outlet pressure increases as the inlet pressure increases.

Figure 8 shows the influence of the inlet pressure on the film thickness, when the pressure — viscosity coefficient $\alpha=0$. The slope of the pressure affects the film thickness, which decreases as the slope of the pressure increases.

Figure 9 shows the influence of the inlet pressure on the pressure distribution of lubrication film of the mineral lubricant ($\alpha=2.2\times10^{-8}\,\mathrm{m^2/N}$), which has the pressure — viscous effect. The slope of the inlet pressure is almost the same as the inlet pressure but the slope of the outlet pressure increases as the inlet pressure increases. This

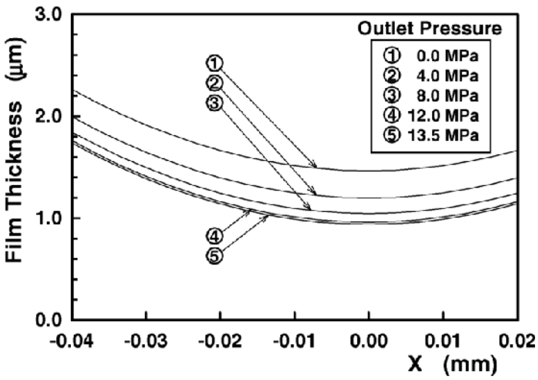


Fig. 6 Influence of outlet pressure on the film thickness (at $\alpha=2.2\times10^{-8}\,\mathrm{m^2/N}$)

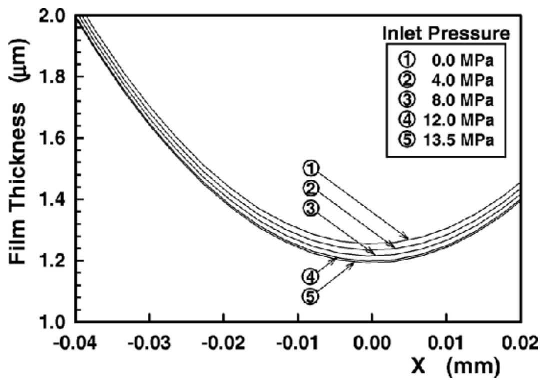


Fig. 8 Influence of inlet pressure on the film thickness (at $\alpha=0$)

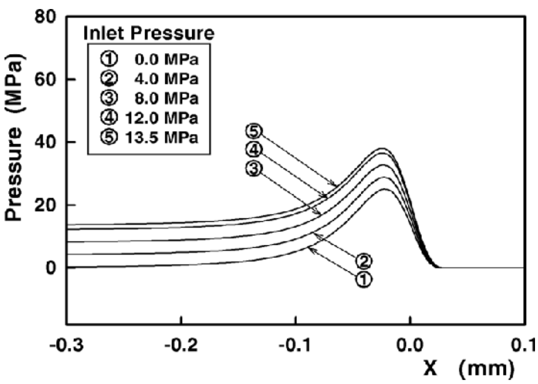


Fig. 7 Influence of inlet pressure on the pressure distribution of lubrication film (at $\alpha=0$)

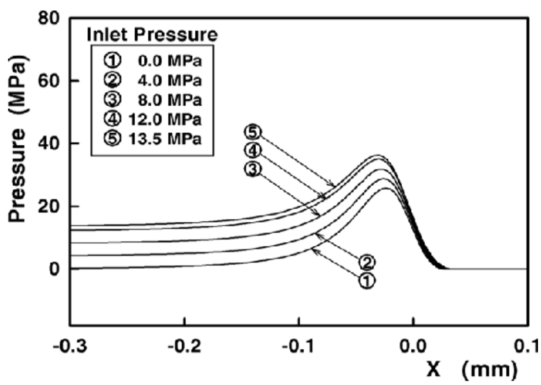


Fig. 9 Influence of inlet pressure on the pressure distribution of lubrication film (at $\alpha=2.2\times10^{-8}\,\mathrm{m^2/N}$)

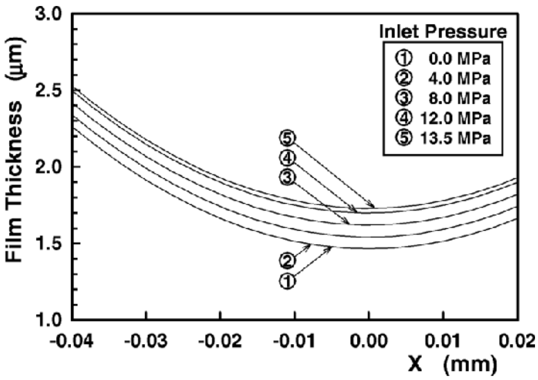


Fig. 10 Influence of inlet pressure on the film thickness (at $\alpha=2.2\times10^{-8}\ m^2/N$)

is because the value of integral by x is the same under the assumption that the pure normal load applied to the vane is fixed.

Figure 10 shows the influence of the inlet pressure on the film thickness, when the pressure — viscosity coefficient $\alpha=2.2\times10^{-8}\ m^2/N$. If the viscosity changes due to the pressure is considered, the film thickness decreases due to the increase of the slope of the pressure in the outlet, but the film thickness, on the whole, increases because of the increase of the viscosity due to the increase of the pressure in the film.

Therefore, from the results above, the slope of the pressure in the fluid film and the film thickness are affected greatly by the pressure boundary conditions of the inlet and outlet regions. And, when the pressure-viscous effect is considered, the amount of change in the film thickness is decided by the interaction between the slope of the pressure and the viscosity.

5. Conclusions

This study on the lubrication modes between the vane and the camring of an oil hydraulic vane pump was carried out by assuming that the lubricational regions of the vane tip are in hydrodynamic lubrication. The following results were obtained.

(1) For the straight vane type vane pumps, even under the most severe conditions, the vanes are in the vicinity of the boundaries of the R-V and E-V regions, and are not in the so-called typical elastohydrodynamic lubrication (Hard EHL), as thought before. Therefore, to analyze the lubrication characteristics of vane pump, the suitable analysis methods to analyze each of lubrication regions are required.

(2) The shape of pressure distribution is slightly affected by the pressure-viscous effect, but the fluid film shape is affected remarkably. Therefore, pressure-viscous effect must be considered in the analysis of the lubrication characteristics.

(3) The pressure boundary conditions at the inlet and outlet of both of the small and large arc regions have greatly affected the slopes of the pressure and film thickness distributions, and specifically what analysis considering the pressure conditions at the inlet and outlet is required.

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