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The Lubrication Characteristics of the Vane Tip Under Inlet Pressure Boundary Conditions for an Oil Hydraulic Vane Pump

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The lubrication modes of line contact between the vane and the camring in an oil hydraulic vane pump have been investigated. First, variations of the radial acting force of a vane were calculated from previously measured results of the dynamic internal pressure in four chambers surrounding a vane. Next, distinctions of the lubrication modes were made using Hooke's chart, which represents an improvement over Johnson's chart. Finally, the influence of boundary conditions in the lubrication region on fluid film lubrication was examined by calculating film pressure distributions. The results show that the lubrication modes of the vane tip are a rigid-variable viscosity region. This region discharges pressure higher than 7 MPa, and exerts a great influence on oil film pressure in the large arc section due to the Piezo-viscous effect.

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

Nomenclature

G	: Dimensionless material parameter $=\alpha E'$
$g_E = g_3$	: Dimensionless elastic parameter
$g_V = g_1$	: Dimensionless viscosity parameter
H	: Dimensionless oil film thickness $=h_0/R'$
h	: Oil film thickness
h_{min}	: Minimum oil film thickness
P_1	: Intra vane chamber pressure
P_2	: Pre-chamber Pressure
P_3	: Post-chamber pressure
P_4	: Bottom chamber pressure
p	: Oil film pressure
R'	: Equivalent radius $=\dfrac{R_c R_v}{R_c - R_v}$
R_c	: Curvature radius of the camring
R_v	: Curvature radius of the vane tip
u	: Mean velocity $=U_v/2$
U	: Dimensionless velocity parameter
U_v	: Velocity of the vane
w	: Load per unit length
W	: Dimensionless load parameter
α	: Pressure viscosity index
η	: Viscosity
η_0	: Viscosity at the atmospheric pressure

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1. Introduction

In the case of the oil hydraulic vane pump, the tribological characteristics of the vane tip, regarded as the most important region in a vane pump, should be clearly examined to make it suitable for high velocity and pressure, and to improve its durability. In order to analyze the tribological characteristics in this region, the normal acting force, caused between the vane and the camring in a relative sliding motion under a high load at high speed conditions, should be obtained. In obtaining the force by the pressure of all acting forces at various conditions, the overall acting force can be obtained from the terms, of which are the hydraulic, inertial and viscosity forces, obtained from the experiment, which measure the abnormal pressures of various positions of Jung, J. Y. and Jung, S. H. (1998). Therefore, when the specification of a pump is known, the normal force acting on the vane is easily obtained.

Figure 1 shows Johnson's chart, which classifies the lubrication regions of line contact under a concentrated load into four groups, R-I (Rigid-Isoviscosity), R-V (Rigid-Variable viscosity), E-I (Elastic-Isoviscosity), E-V (Elastic-Variable viscosity), however, the slant-lined region has not been verified thus far. Analysis of the lubrication characteristics has been carried out with objects such as sliding bearings, gears, cams and so on. There are not many parts like these in machine elements, and the solutions with the pressure viscosity effect tend to diverge. Regarding the contact region between the vane tip and the camring, the acting force should be small to reduce friction and wear, but the film thickness should be thin to reduce leakage. To overcome these conflicting conditions, the contact area should be decreased by reducing the curvature radius of the vane tip.

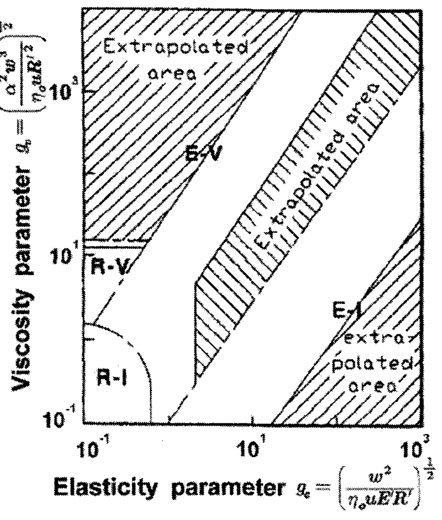
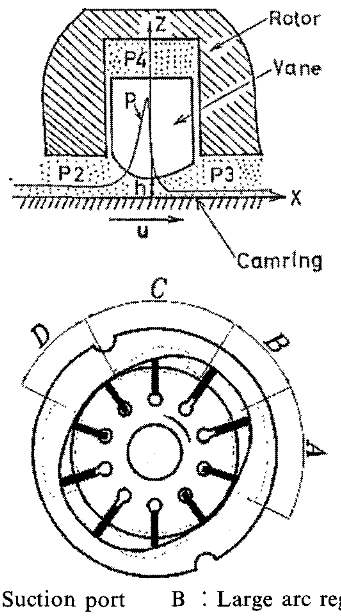


Fig. 1 Lubrication mode on Johnson's chart

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

2. Lubrication Modes of the Line Contact Region

The vane tip and camring of an oil hydraulic vane pump contact in the form of a line. Figure 2



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

Fig. 2 Model of vane-tip section & Position of vane chamber

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

Table 2 Non-dimensional display of EHL

Non-dimension	Dowson-Higginson	Blok-Moes	Greenwood-Johnson
Film thickness	$H=\frac{h_{min}}{R}$	$N_1=HU^{-1/2}$	$\bar{h}=\frac{HW}{U}=N_1N_2=\frac{h_{min}w}{\mu_0 uR}$
Normal load	Load parameter $W=\frac{w}{E'R}$	$N_2=WU^{-1/2}$	Elasticity parameter $g3=N_2=\frac{g_1}{\sqrt{2}\pi g_2}=\frac{w}{(\mu_0 E'R)^{1/2}}$
Pressure viscous effect	Material parameter $G=\alpha E'$	$N_3=GU^{1/4}$	Viscosity parameter $g_1=\frac{GW^{3/2}}{U^{1/2}}N_2^{3/2}N_3=\frac{\alpha w^{3/2}}{(\mu_0 u)^{1/2}R}$ $g_2=G\left(\frac{W}{2\pi}\right)^{1/2}=N_3\left(\frac{N_2}{2\pi}\right)^{1/2}=\alpha\left(\frac{wE'}{2\pi}\right)^{1/2}=\alpha p_0$
Velocity	Velocity parameter $U=\frac{\mu_0 u}{E'R}$	—	—

shows the cross sectional area of a vane pump. The lubrication of line contact is mainly of Elasto-Hydrodynamic Lubrication (EHL), but with variation of conditions, there are 4 modes as shown in Table 1. Hydrodynamic lubrication of the contact region under a concentrated load is classified into four lubrication modes. However, since the material of the vane and camring is rigid, the mode of these are not in the E-I region (called soft EHL).

In the E-V region, the Reynolds' equation and the equation of the elastic deformation of the material are needed simultaneously in order to solve it, but in doing so, it becomes non-linear such that an analytical solution can not be obtained. However, the test formula obtained from numerical solutions under some specific conditions could be acquired as a dimensionless form. Table 2 shows the dimensionless parameters, which have been used as the line of contact. Blok-Moes presented the relation among lubrication modes and three dimensionless parameters (N1 : oil film, N2 : load, N3 : viscosity).

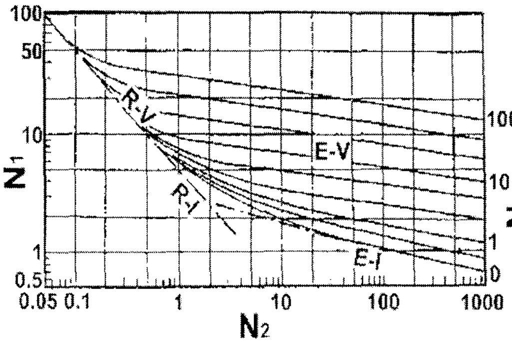


Fig. 3 Relation of Lubrication modes (Blok-Moes chart)

As presented in Fig. 3, if applying the existing numerical test equation of EHL, obtainded by substituting the E-V region with R-V region or R-I region, or the R-I region with the R-V region, it should be realized that a large calcula-

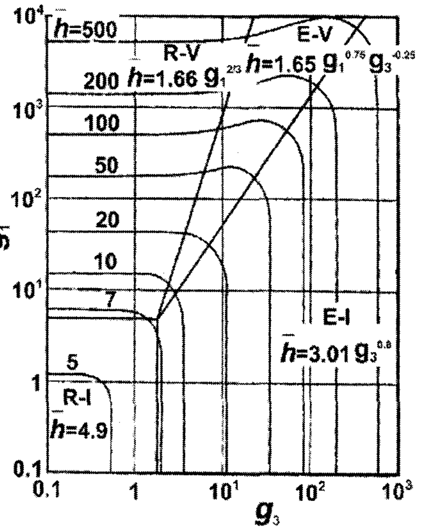


Fig. 4 Distinct diagram of lubrication mode

tion error could occur.

Therefore, when the mode is not known, it must be solved with a Reynolds' equation, and the equation of elastic deformation, simultaneously, considering pressure-viscous effect.

Normal acting force varies, while the rotor revolves 1 cycle, from nearly zero to the maximum so that a vast calculation is required. If the calculation method is not altered according to each lubrication mode, the error becomes large and it could be useless in cases when there are nearly no elastic deformation, or viscosity changes.

In the R-V region, where g_v is large, the solution is easy to diverge, such that it is relatively hard to solve. In this paper, therefore, the lubrication modes have been discriminated as shown in Fig. 4, improved by Hooke, from the graph in which Greenwood-Johnson had divided EHL regions using two parameters, g_v (viscosity parameter), g_e (elasticity parameter) as presented in Table 2. Thus, the lubrication modes have been examined by calculating g_v, g_e from the lubrication conditions of the vane tip and by plotting them in the chart.

3. Lubrication Modes of the Vane Tip

The analysis was conducted assuming that there

Table 3 Capacity of vane pump (intra vane)

No	Delivery flow (cm^3/rev)	Re'f Pressure (MPa)	Max. rpm (rev/min)
1	95.0	17.5	1800
2	156.0	17.5	1800
3	52.5	17.5	2700
4	75.8	14	2000
5	97.6	17.5	2500
6	156.0	17.5	2200

Table 4 Capacity of vane pump (straight vane)

No	Delivery flow (cm^3/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

is enough oil film with sufficient pressure around the inlet and outlet, and that the surfaces of the vane and camring are perfectly smooth, so that it is under a perfect hydrodynamic lubrication status. The viscosity index was a standard grade of mineral oil, $\alpha=2.2\times10^{-8}\,m^2/N$. The dimensionless elasticity and viscosity parameters, g_e, g_v, which are the horizontal and the vertical axis of Johnson's chart, respectively, were calculated using the normal force acting on the vane, obtained from the operational conditions and the pressure of the pump measured. The lubrication modes of the standard types for the vane pump wear test of ASTM, presented in Tables 3 and 4, were examined.

3.1 Lubrication modes of the intra vane type pump

Figure 5 shows lubrication modes for each region of the intra vane type pumps, with the standard specification presented in Table 3. As shown in the figure, all the pumps are gathered close to $g_v=0.5g_e^{1.5}$, and they are distributed, in order of the large size of the suction port, large

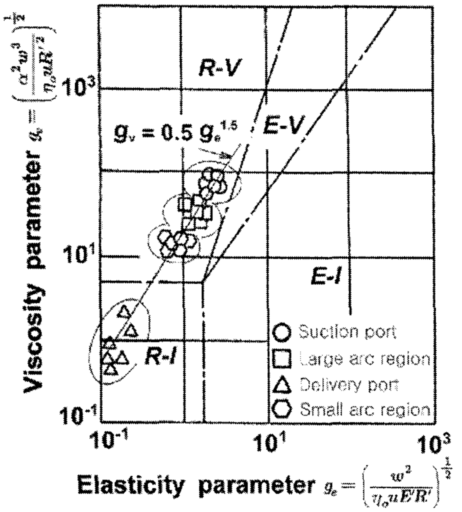


Fig. 5 Lubrication mode of each section in an intra vane pump

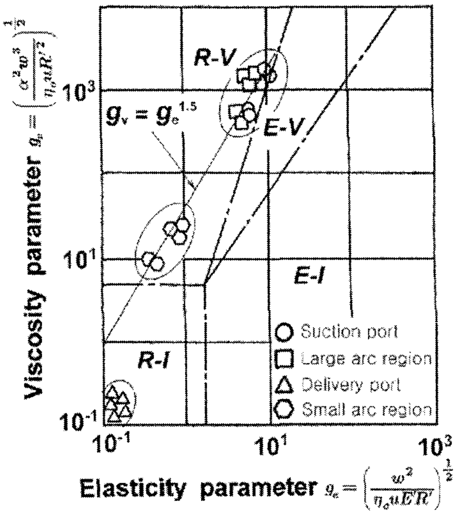


Fig. 6 Lubrication mode of each section in a straight vane pump

arc region, and small arc region, in the R–V region within the range of g_v from 10 to 100. Therefore, when discharge pressure or RPM becomes half, each region becomes half, or the Root Mean Square (RMS) of g_e around the line of $g_v=0.5\, g_e^{1.5}$. They are all close to EHL, but no one comes to the region. The lubrication region of the suction port, which has the most severe lubricating conditions, is closest to the most standard hydrodynamic lubrication. The delivery port in which the acting normal force is applied only as the centrifugal force, is in the R–I region, like sliding bearings.

3.2 Lubrication modes of the straight vane type pump

Figure 6 shows lubrication modes of the straight vane type pumps as presented in Table 4 as the form presented in Fig. 5. As in the cases of the intra vane type pumps, they are also gathered around the line of $g_v=0.5g_e^{1.5}$. All regions except the delivery port, which has no load, are in the R–V region too. However, the large arc regions deviate slightly from the line to the left (where g_e is small) of the suction regions. Differences among the suction ports, large arc regions and the small arc regions, compared with the intra vane type pumps, are about ten times larger. When operated near the line of $g_v=0.5_e^{1.5}$ with half the discharge pressure, they become half of g_e, and when RPM is reduced to half, they become the Root Mean Square (RMS) of g_e, with regular operating conditions.

Therefore, if the regular discharge pressure is set and RPM is reduced to half, they more closer to the boundary between the R–V and E–V regions. From above, lubrication modes at the vane tips of all regions except for the delivery ports with no load are not like the typical elastohydrodynamic lubrication (E–V region) of sliding bearings, gears and so no, in which viscosity changes and elastic deformation occur at a similar extent ; in case of intra vane type pumps, it is in the R–V region, and in case of straight vane type pumps, it is in the R–V and E–V region when the vane is in the scuction ports and large arc regions. Calculation of this region is so hard that it has not been calculated, and there are no examples so far.

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

The lubrication mode diagram presented in Fig. 5, used for discriminating lubrication modes of the vane tip, has been drawn based on the existing calculation of EHL, which was carried out assuming the inlet and outlet pressure are equal to the atmospheric pressure. In addition to this diagram, the normal acting force is calculated, in order to replace these boundary conditions, based on the definition of the normal acting force in Jung, J. Y. and Jung, S. H.(1998). However, the pre-chamber pressure is discharge pressure in the case of a large arc region, and the post-chamber pressure is discharge pressure in the small arc region. In the definition Jung, the pressure distribution of oil film, when the boundary conditions of the inlet and outlet pressures are not atmospheric pressure, is based on the assumption that each inlet and outlet pressure was discriminated as the location of the minimum film thickness as a boundary when the boundary conditions are atmospheric pressure. Therefore, when it comes to the large and small arc regions, the results must be reviewed.

4.1 In the case of the small arc region

When assuming that if the outlet pressure of the lubricational film region is higher than atmospheric pressure like in the small arc region, the pressure is sufficiently lower than the maximum pressure caused in the oil film, the film breaks by negative pressure produced right behind where the minimum film thickness is, and the film thickness at the region becomes relatively small, so that Swift-Stiber (Reynolds) boundary conditions are applied as when the outlet pressure is equal to the atmospheric pressure. Therefore, in the small arc region, where the outlet pressure becomes discharge pressure when the inlet pressure is nearly atmospheric pressure, the effect of the outlet pressure on the pressure of oil film can be neglected, and it is thought that the assumption above is valid.

4.2 In the case of the large arc region

According to the results of lubrication modes with the above assumption, effects of the pressure boundary conditions are examined assuming the materials are rigid, because the lubrication modes of the large and small arc regions are in the R-V region. However, when calculated as rigid, the solutions diverge such that they could not be obtained, so calculatable conditions are examined here.

Figures 7 and 8 are calculation examples when the lubricational surfaces are "rigid", the clearance is "fixed", the inlet pressure is discharge pressure, and the outlet pressure is suction pressure (atmospheric). When the viscosity index is zero, that is, when there is no pressure viscosity effect, the overall pressure can only be increased by the inlet pressure, as illustrated in Fig. 7. In addition, when the viscosity index is that of the mineral lubricants ($\alpha=2.2\times10^{-8}\,m^2/N$) which have a pressure-viscous effect, the maximum pressure of oil film according to each inlet pressure

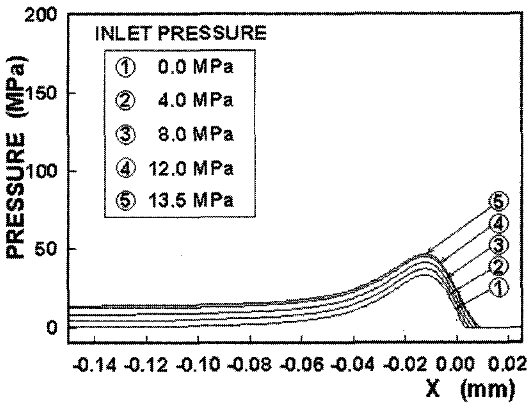


Fig. 7 Influence of inlet pressure affect on the pressure distribution of lubrication film (at $\alpha=0$)

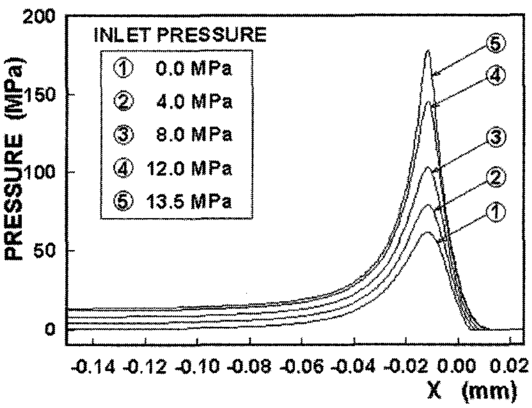


Fig. 8 Influence of inlet pressure affect on the pressure distribution of lubrication film (at $\alpha=2.2\times10^{-8}\,m^2/N$)

has many more differences than the pressure differences of the inlet as presented in Fig. 8. In particular, when the discharge pressure is above 7 MPa, it greatly influences the overall oil film pressure, compared to the maximum oil pressure. This is because the Piezo-viscous effect has a nonlinear character.

Figures 9 and 10 are calculation examples according to the value of the pressure viscosity index, α, when the lubricational surfaces are "rigid", the clearance is "fixed", the inlet pressure is discharge pressure and the outlet pressure is suction pressure (atmospheric). In Fig. 9, when the inlet pressure is zero, the effect of the pressure viscosity index is small, and in Fig. 10, when the inlet pressure is not zero, the effect of the pressure viscosity index is large, and the larger the inlet pressure is, the larger the Piezzo-viscous effect.

Figures 11 and 12 are calculation examples according to the temperature of lubricant, when the lubricational surfaces are "rigid", the clearance is "fixed", the inlet pressure is discharge pressure and the outlet pressure is suction pressure (atmospheric). In Fig. 11, when the inlet pressure is zero, as the temperature rises, the viscosity is reduced so that the maximum pressure is low, and the change of maximum pressure according to the temperature is also small. In Fig. 12, when the inlet pressure is not zero, the effect of the temperature of lubricant is rather

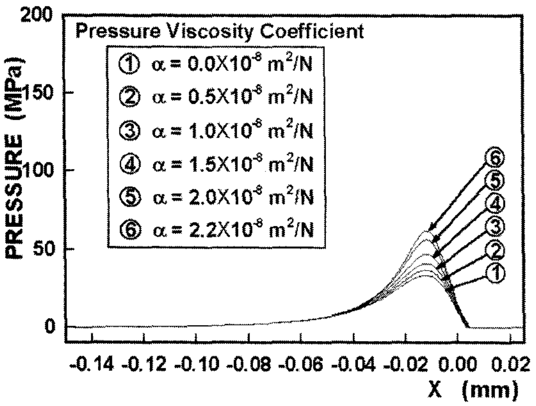


Fig. 9 Influence of pressure viscosity coefficient affect on the pressure distribution of lubrication film (at Inlet Pressure=0)

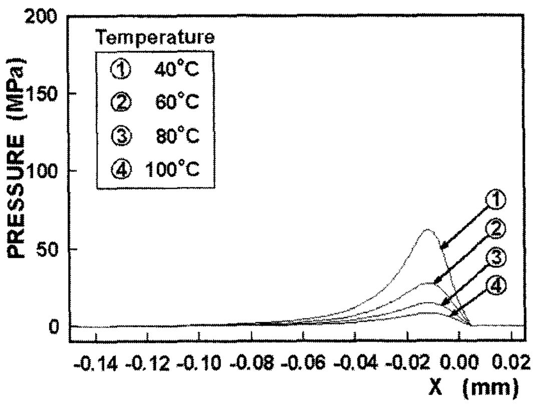


Fig. 11 Influence of temperature affect on the pressure distribution of lubrication film (at Inlet Pressure=0)

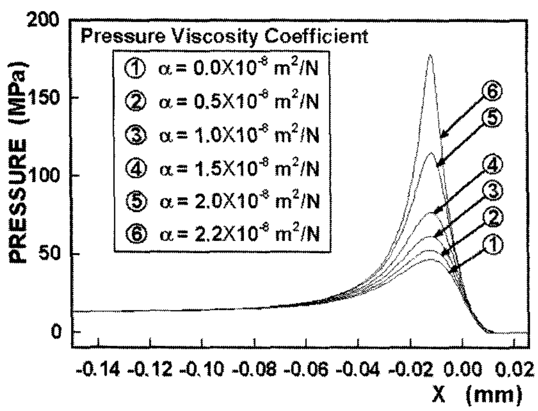


Fig. 10 Influence of pressure viscosity coefficient affect on the pressure distribution of lubrication film (at Inlet Pressure=13.5 MPa)

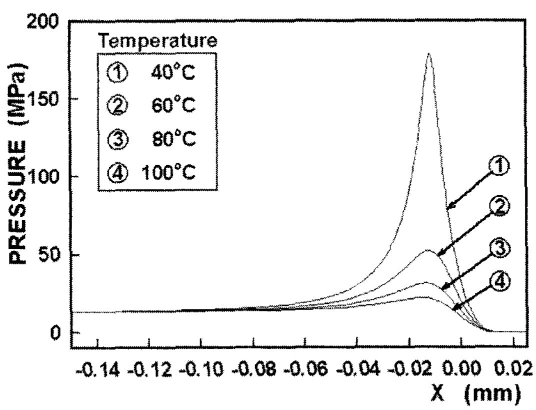


Fig. 12 Influence of temperature affect on the pressure distribution of lubrication film (at Inlet Pressure=13.5 MPa)

large, and the larger the inlet pressure is, the larger the effect of the temperature on the maximum pressure.

Accordingly, the assumption above cannot be applied to the large arc region and an analysis considering the pressure boundary conditions is required. In the region where g_v is large, if elastic deformation is not considered the solution tends to diverge, and even if the region is rigid, the analysis for this region must consider elastic deformation.

5. Conclusions

As a result of this study on the lubrication modes between the vane and the camring of an oil hydraulic vane pump, carried out assuming the lubricational regions of the vane tip are in hydrodynamic lubrication, the following results were obtained.

(1) In the case of the intra vane types, lubrication modes of the suction ports, the large and small arc regions except the delivery ports with an acting force of nearly zero, are in the R-V region. In the case of the straight vane types, even if it is under the most severe conditions, they are in the vicinity of the boundary of the R-V and E-V region, and are not in the so called typical elastohydrodynamic lubrication (Hard EHL) as guessed before, which shows the remarkable effect of viscosity changes and elastic deformation.

(2) The modes of the suction port and the large arc region under the regular conditions are in the R-V region, and because it is the region where the nondimesional parameter g_v is large, the calculation of the lubrication characteristics is so hard that it has not been calculated completely thus far.

(3) Regarding the lubrication characteristics of the large arc region, it is under great influence of the boundary conditions of the inlet pressure by the Piezo-viscous effect, and the effects of the temperature and the pressure viscosity, according to the inlet pressure, is clear. Therefore, analysis by the charts of Johnson or Hooke have some problems, so that a new method of analysis considering inlet pressure conditions is required.

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