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STUDY ON THE OIL SUPPLY SYSTEM FOR ROTARY COMPRESSORS

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ABSTRACT

Research has been undertaken to clarify shaft oil pump mechanisms and oil supply network systems for rotary compressors. Numerical expressions were developed for each part of the rotary compressor, (such as drive shaft oil pump and journal bearings grooves.) in order to confirm that the calculated values agree with the experimental results. Finally, a computer program to evaluate the oil supply system performance under steady conditions for rotary compressors was developed.

INTRODUCTION

Reliability of journal bearings in rotary compressors is one of the most important factors when considering rotary compressor efficacy. Many studies have been completed that deal with journal bearings. However, it is assumed in most of studies that a sufficient amount of oil can be supplied for journal bearings. There are few studies, however, about the oil supply system necessary for lubricating each part in the compressor. Therefore, an attempt has been made to clarify the characteristics of the oil supply system for rotary compressors.

This paper presents the following.

(1) A theoretical analysis for elements of an oil supply system;
(2) Experimental results for elements of an oil supply system; and
(3) An explanation of the development of a computer program to evaluate the oil supply system

STRUCTURE AND LUBRICANT PRINCIPLES

Fig. 1 shows the cross sectional view of a hermetic type rotary compressor. An outline of the oil supply system is as follows. A centrifugal force caused by rotation of the drive shaft pumps up oil from a lower oil chamber through a cap hole in the shaft end. The drive shaft has three discharge holes which are located in the upper journal bearing, the lower journal bearing and the rolling piston. The oil discharged from the upper journal bearing hole and the lower journal bearing hole is led to each journal bearing groove. The oil returns to the lower oil chamber again after lubrication of journal bearings. The oil discharged from the hole of the rolling piston lubricates the journal bearing between the eccentric shaft and the inner side of rolling piston. Finally, the oil leaks into the cylinder through the slight space which consists of the rolling piston sides and the journal bearing flat faces.

In order to analyze the oil supply system for rotary compressors, it is necessary to know the characteristics of flow rates and pressure losses for each part.

The characteristics of compressor parts requested to be analyzed are shown as follows.

(1) The characteristics of the centrifugal pump in the shaft;
(2) The characteristics of the inlet cap hole in the shaft;
(3) The characteristics of the discharge hole in the shaft;
(4) The characteristics of the journal bearing grooves.

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Centrifugal pump in shaft

A pumping head created by shaft rotation can be explained as a function of the relationship between centrifugal force and pressure. The pumping head is

$$ P = \frac{\rho}{2} (r_0 \omega)^2 $$

(1)

where, $P$ is the pump head, $\rho$ is the density of oil, $r_0$ is the radius of the shaft and $\omega$ is the angular velocity of the shaft.

In terms of cap hole pressure drop, the open area of compressor's hole is unable to support the pumping pressure. Therefore, the oil flows down through the hole. The pressure distribution of the open area can be determined by taking into account of inflow and outflow through the hole. The pumping head $P$ considering the cap hole is

$$ P = \frac{\rho}{2} (r_0 \omega)^2 - \Delta P_c $$

(2)

where, $\Delta P_c$ is the pressure drop of the cap hole.

The equation for the fluid velocity, $u_c$, is as follows:

$$ u_c = \sqrt{\frac{2}{\rho} \left( \Delta P_c - \frac{\rho}{2} (r \omega)^2 \right)} $$

(3)

The integration of eqn(3) in terms of radius $r$, gives the flow rate $Q_c$.

$$ Q_c = \int_0^r 2\pi u_c \, dr $$

$$ = \frac{2\pi}{3\omega} \left( \frac{2\Delta P_c}{\rho} \right)^{3/2} \left[ 1 - \left( \frac{\rho \omega^2 r^2}{2\Delta P_c} \right)^{3/2} \right] $$

(4)

On the assumption that the amount of inflow oil is equal to the amount of outflow oil, the pressure drop $\Delta P_c$ is given as eqn(5).

$$ \Delta P_c = \frac{\rho}{4} (r_0 \omega)^2 $$

(5)

Substituting of eqn(5) in eqn(2), the total pumping head is written as

$$ P = \frac{\rho}{2} (r_0 \omega)^2 - \frac{\rho}{4} (r_0 \omega)^2 $$

(6)

Discharge oil port in drive shaft

Discharge holes in the shaft pump are regarded as orifices. The relation between the pressure difference $\Delta P_d$ and the volumetric discharge flow rate $Q_d$ is...
\[ Q_d = C_d \cdot A_d \sqrt{\frac{2 \Delta P_d}{\rho}} \]  

(7)

where, \( A_d \) is the area of the discharge hole and \( C_d \) is the discharge coefficient, a function of Reynolds number \( Re \). The schematic of \( C_d \) is shown in fig.2.

In this case, Reynolds number \( Re \) is defined as follows:

\[ Re = \frac{d_s \cdot \sqrt{2 \Delta P_d}}{\nu} \]  

(8)

where, \( d_s \) is the diameter of the discharge hole and \( \nu \) is the kinematic viscosity of oil. Actually, the discharge coefficient, \( C_d \), was measured experimentally when the shaft speed is equal to zero.

**Journal bearing groove**

Fig. 3 shows the sketch of a journal bearing. Journal bearing groove can be expressed by two models: (1) Pressure drop due to groove and shaft surface friction loss, and (2) The viscous pumping effect caused by the spiral groove. With regard to pressure drop, Hagen-Poiseuille flow pattern can be applied.

Eqn(9) represents the Hagen-Poiseuille equation,

\[ Q_{fg} = \frac{\pi d^4}{128 \mu \ell} \Delta P_g \]  

(9)

where, \( d \) is the diameter of the groove, \( \ell \) is the length of the groove, \( \mu \) is the viscosity of oil and \( \Delta P_g \) is the pressure difference.

Because the cross section of the groove is not a circle, a hydraulic diameter is used instead of a diameter. A hydraulic diameter is defined as four times length of a cross sectional area divided by a circumference of the section.

A hydraulic diameter, \( 4m \), is

\[ 4m = \frac{4 A_g}{s} \]  

(10)

where, \( A_g \) is the cross section area of the groove and \( s \) is the circumference length of the groove.

Substituting the hydraulic diameter \( 4m \) in eqn(9), for diameter \( d \), eqn(11) is created;

\[ Q_{fg} = \frac{2 \pi m^4}{(\mu \ell)} \Delta P_g \]  

(11)

In terms of the viscous pumping model, Couette flow patterns to the spiral groove of the journal bearing can be applied. When the groove has an axial direction angle \( \phi \) rotating shaft velocity toward the direction of the shaft must be taken into account. In this case, the volume of the Couette flow is given as the product of the groove area and the average speed. The average speed \( V_g \) toward the groove direction is presented in eqn(12).
The flow volume of Couette flow \( Q_{sp} \) is

\[
V_g = \left( \frac{r_1 \omega}{2} \right) \sin \phi \quad (12)
\]

The flow volume of Couette flow \( Q_{sp} \) is

\[
Q_{sp} = V_g \cdot A_g = A_g \cdot \left( \frac{r_1 \omega}{2} \right) \sin \phi \quad (13)
\]

Combining eqn(11) and eqn(13) gives an equation of the relationship between the flow rate \( Q_g \) and the pressure loss \( \Delta P_g \) in the journal bearing groove.

\[
Q_g = Q_{fg} + Q_{sp} = \frac{2 \pi \rho \nu}{(\mu)} \Delta P_g + A_g \left( \frac{r_1 \omega}{2} \right) \sin \phi \quad (14)
\]

Oil supply network

In order to evaluate the oil supply network system, an equivalent electrical circuit composed. From this circuit, non-linear equations for the closed circuit loops were obtained. Using a numerical analysis called the Newton-Raphson method, these equations were solved.

**EXPERIMENTAL APPARATUS**

Fig.4 shows the cross sectional view of the experimental apparatus. The apparatus consists of drive shaft parts and oil flow line parts. The shaft is driven by an electric motor and its rotating speed is measured by an optical digital tachometer. The oil tank with its head control device is joined to the lower oil chamber. The discharge oil pressure in the measurement chamber is measured by a manometer. The discharge flow rate is measured by a meter-glass. To find the density and the viscosity of the oil, the oil temperature is measured by a thermal-couple.

**EXPERIMENTAL RESULTS AND DISCUSSION**

**Centrifugal shaft pump**

The experimental data of the centrifugal shaft pump compared with eqn(6) is shown in fig.5. The experimental results agree with the theoretical ones. From those results, it can be confirmed that the pumping head of the centrifugal shaft pump is given by the outer radius of the shaft.

Fig.6 shows the characteristics of the pressure drop in the cap hole at a flow rate of zero. The pressure drop in the cap hole is evaluated in eqn(5).

**Cap hole**

Fig.7 shows the relationship between pressure loss and the cap hole flow rate. Solid lines in the graph describe the single orifice theory and interrupted lines describe the double orifice theory. The experimental results with cap holes having diameter of 4.8 mm and 2.4 mm correspond to hypothesized lines. However results for cap holes a diameter of 6.8 mm disagree with a single orifice theory. Therefore, it can be considered that the small hooks of a cap, as shown in fig.8, act as a flow resistance. Flow resistance was evaluated as an imaginary orifice. In this case, the diameter of the imaginary orifice size is estimated at around 7.0 mm. The interrupted line in fig.7 is calculated using this value and the theoretical curve correlated well with the experimental data. In this case, the flow coefficient is 0.61, equal to the value of a thin-bladed orifice.
Discharge coefficient

Fig. 9 shows the results of the discharge coefficient. From this result, it was confirmed that the discharge coefficient was changed by Reynolds number. Therefore, an experimental expression to the discharge coefficient \( C_d \) can be written as follows.

\[
\begin{align*}
C_d &= 0.0082 \cdot Re & (0 \leq Re \leq 37.3) \\
C_d &= 0.329 \cdot \log(Re) - 0.188 & (37.3 \leq Re \leq 400) \\
C_d &= 0.668 - 3.36 \times 10^3 \cdot (Re - 400) & (400 \leq Re \leq 2000) \\
C_d &= 0.61 & (Re > 2000)
\end{align*}
\] (15)

From now on, the discharge coefficient presented in eqn(15) is used when the volumetric flow rate of discharge is calculated by the pressure difference. However, eqn(15) is not established in general, but can only be assumed when the shaped its orifice is similar to the tested orifice.

Discharge holes

Fig. 10 shows the relationship between pressure loss and volumetric flow rate in the discharge holes. The characteristics of the discharge are changed due to rotating speed. It is impossible to explain this phenomenon from eqn(7).

It has been hypothesized that this phenomenon was caused by the fluid motion in the discharge chamber. Fig. 11 shows the schematic of the discharge. The flow pattern in the discharge chamber is assumed to be the Couette flow in the concentric pipes. The distribution of the velocity in the discharge chamber is given as eqn(16).

\[
V_o(r) = A \cdot r + B \cdot \frac{1}{r}
\] (16)

\[
A = \frac{r_o^2 \omega}{r_e^2 - r_o^2}, \quad B = \frac{r_e^2 r_o \omega}{r_e^2 - r_o^2}, \quad r_e, r_o: \text{radius of chamber}
\]

The oil jet from the discharge hole is deflected by the fluid motion in the chamber and diffuses to the chamber flow. However, a sharing layer between the oil jet and the chamber flow is generated when the oil is discharged from the hole. Therefore, the discharge velocity keeps the initial velocity based on the pressure difference. The effective discharge area is decreased by the oil jet deflection. The effective discharge area \( A_{eff} \) is given as eqn(17).

\[
A_{eff} = A_d \cdot \cos \theta
\] (17)

The deflection angle \( \theta \) is defined in eqn(18).

\[
\cos \theta = \frac{u_d}{\sqrt{u_d^2 - r_o \omega^2}} = \sqrt{\frac{2 \Delta P_d}{\rho}} \sqrt{\frac{2 \Delta P_d}{\rho + r_e^2 \omega^2}}
\] (18)
The relationship between pressure loss and volumetric flow rate is given as eqn(19).

\[ Q_d = C_d \cdot A_{\text{eff}} \sqrt{\frac{2 \Delta P \cdot d}{\rho}} \]  

(19)

The solid lines in fig.10 are calculated in eqn(19). It is possible to explain the experimental results using eqn(19). However, eqn(19) must be treated as an experimental equation, because there is no empirical evidence of the sharing layer and the deflection angle.

Journal bearing grooves

The relationship between pressure loss and volumetric flow rate in grooves of journal bearings are shown in fig.12. There is a correlation between the theoretical results given in eqn(13) and the experimental ones. The volumetric flow rate with the spiral angle of 37 deg. is about three times as much as the flow rate with the spiral angle of 0 deg. From these results, it is possible to evaluate the characteristics of journal bearing grooves. The spiral groove greatly increases the oil supply velocity capability.

Development of computer program

A computer program to evaluate the oil supply system for rotary compressors was developed, by integrating the characteristics of the elements provided by the above-mentioned analysis and experiments. The volumetric flow rate of oil and oil pressure loss for each part can be computed.

MEASUREMENT OF AMOUNT OF OIL SUPPLY IN ROTARY COMPRESSOR

Measurement apparatus and measurement method

Under stable conditions, it is difficult to measure the amount of oil supply for each component. Therefore, only the total amount of the oil supply in a rotary compressor was attempted to be measured.

Fig.13 shows a cross sectional view of the apparatus to measure the total amount of the oil supply. The oil is led to the oil flow-meter from the lower oil chamber. Simultaneously, the oil temperature is measured in the oil flow-meter. The oil is led to the center plug which sets under the lower journal bearing after passing through the oil flow-meter. Since the inner side of the plug is separated from the lower oil chamber, the total suction oil by the centrifugal shaft pump is fed through the flow-meter.

Results of measurement

Fig.14 shows the measurement results of the total oil supply. In this measurement, the suction pressure is 0.523 MPa and the discharge pressure is 2.04MPa. The solid line in fig.14 shows the analysis by the computer program for evaluating the oil supply system. Results of the measurement approximate the analyzed values. However, there is a small difference. This can be considered to be caused by the estimation error of the oil viscosity. In this case, the oil on the way to the center plug is cooled by air. For this reason, the calculated viscosity is estimated to be higher than the real viscosity in the rotary compressor.
CONCLUSION

The followings are some conclusions as a result of this study:

(1) The pumping head of the centrifugal shaft pump is determined by the shaft diameter and the cap hole diameter in the shaft end.

(2) The orifice flow theory can be applied to the characteristics of the cap hole and the discharge hole.

(3) The groove of the journal bearing can be measured by the pipe line, and the pumping effect is caused in the spiral groove.

(4) The measurement results of the amount of the oil supply in the real rotary compressor approximate the analysis value which is calculated by the computer program. Estimating the oil viscosity is the most important point in the evaluation of the oil supply system.

This study may be utilized to aid in the design of rotary compressors.

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Fig. 1 Sectional views of Comp.  Fig. 2 Schematic of discharge coefficient
Fig. 3 Sketch of journal bearing

Fig. 4 Experimental apparatus

Fig. 5 Pressure of centrifugal shaft pump

Fig. 6 Pressure drop in cap hole
Fig. 7 Cap flow rate

Fig. 8 Outward motion of cap

Fig. 9 Measured discharge coefficient

Fig. 10 Flow rate in discharge hole
Fig. 11 Illustration of discharge

Fig. 12 Characteristics of journal bearing groove

Fig. 13 Measurement apparatus for total amount of oil supply

Fig. 14 Measured total amount of oil supply