1998

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ABSTRACT

This Paper has studied the oil supply characteristics to scroll compressors for package air conditioners. Scroll compressor has four major lubricating parts: the lower journal bearing, the main journal bearing, the orbiting scroll journal bearing and the thrust bearing between the orbiting scroll and the main frame. It is very important to supply the required flow rate of oil in order to prevent these lubricating parts from wearing. Namely if the required flow rate were not supplied to these lubrication parts, the locking phenomena of scroll compressors would frequently occur. So we had to study the characteristics of the oil supply system for the reliability of scroll compressors. In this study we estimated theoretically the supply flow rate, pressures, resistance and the required flow rate according to several design parameters of the oil supply system. In this simulation equivalent electrical circuit was used, and we compared these results with the experimental data in order to improve the accuracy of simulation.

NOMENCLATURE

Parameter

- B Oil Groove Width
- C Clearance , Coefficient
- D Diameter
- H Oil Groove Height, Oil Film Thickness
- N revolution
- P Pressure
- Q Flow Rate
- R Radius , Resistance , Roughness
- T Temperature

Subscript

- gc Rotational Flow of Oil groove
- gp Pressure Difference of Oil Groove
- i Without Rotation
- m Minimum Oil Film Thickness
- n Necessary Flow Rate
- o Orifice
- p Pressure Difference Flow
- r Rotation
- s Leakage Flow

Greek

- β Groove Angle
- ε Eccentric Ratio
- λ Oil Film Parameter
- μ Viscosity
- ξ Shape Coefficient
- π Pi
- ρ Density
- ω Angular Velocity

INTRODUCTION

Scroll compressor is consisted of a motor, a shaft, fixed and orbiting scrolls, and main and lower frames that support the shaft. It is superior to rotary and reciprocating type compressors in efficiency, noise and vibration due to its structural advantages. But it is rather difficult to supply enough oil to the lubricating parts because these parts are located at the upside of compressor. So it is much more important to supply oil properly to these parts in term of reliability. Oil plays important roles in compressors, such as the prevention of gas leakage at the compression chamber, the lubrication of the lubricating parts and the cooling of the lubricating parts heated by friction heat. If oil is oversupplied to the lubricating parts, the oil discharged into cycle increases. The poor supply of oil results in the temperature rise of bearings. So it is required to design the optimum oil supply system to guarantee high reliability and efficiency of scroll compressors.

Until now many studies about the oil supply system of scroll compressors have been done, but only few take the whole system as an object of analysis. In this study, we take into accounting the whole oil supply system and we are to design the reliable oil supply system that the supply flow rate can satisfy the required flow rate for the lubrication and the cooling of the lubricating parts.

Theoretical Development

Lubrication Parts

As shown in Fig.1, scroll compressors have four major lubricating parts: the orbiting scroll journal bearing, the main journal bearing, the main thrust bearing and the lower journal bearing. Out of the oil pumped up through the inner hole of shaft from the base reservoir by the centrifugal force, some flows out through the upper oil discharge hole of shaft by the centrifugal force between the inner and the outer radius of shaft and others gets to the upper end of shaft. The oil passing the upper oil discharge hole of shaft lubricates the main journal bearing. It goes up along the groove and clearance of the main journal bearing except a little oil discharge in the downward, and then flows into the oil discharge hole of the thrust surface and the orbiting space of orbiting scroll at main frame. Oil transported to the upper end of the shaft flows in two mid path - through the gap between the slide bush and the shaft pin and through the clearance and groove of the O/S journal bearing. And then the oil flowing through the two mid path is transported to the orbiting space of orbiting scroll at main frame. And some of the oil joining in the orbiting space of orbiting scroll at main frame returns to the base reservoir through oil return pipe and others is supplied to the thrust bearing.

Design Flow

We are going to design the safe oil supply structure by calculating both the supply flow rate and the required flow rate for the lubrication and the cooling effect of bearings. At first in the calculation of the supply flow rate we calculate the viscosity of oil by inputting the geometric dimensions such as the shaft and bearing, and the operation condition such as temperature, pressure, rotation speed and mixing characteristics of oil and refrigerant. And by solving simultaneous equations consisted of pressure and resistance constants of each oil path we determine whether the oil pumping capacity can supply the upper lubricating parts with oil. To ensure the safety ratio of the supply flow rate to the required flow rate, we can change the pumping capacity and improve the oil supply structure.

In calculating the required flow rate, the input data is geometric dimensions of bearings, bearing loads calculated from the performance analysis and operation condition such as pressure, inlet temperatures of bearing, rotation speed, etc.. We use the mean value of the inlet and the outlet temperature of bearing to calculate oil viscosity in bearings. And we calculate the characteristics of bearings such as pressure, velocity, eccentric ratio, sommerfeld number and minimum film thickness. And we calculate the heat loss that results from the mechanical friction of bearings. By reducing gradually the mean temperature of bearing we calculate the cooling flow rate to cool the heat of friction at bearings and the necessary flow rate to form the minimum oil film thickness thicker than the limited film thickness. From the necessary and cooling flow rates, we take the larger one as the required flow rate.

Equivalent Electrical Circuit

Fig.2 shows the electrical circuit equivalent to the oil supply system to calculate flow rates. In this case, we substitute current for supply flow rate[Q] transported from point to point, potential difference for pressure difference[$\triangle$ P] and electrical resistance for supply resistance. And we formulate 1st order equation , $Ri \cdot Qi = \triangle Pi$ for the ith path and n simultaneous 1st order equations, $\sum Rij \cdot \sum Qij = \sum \triangle Pij$ for the jth closed loop composed of several paths. So the piping network has 8 closed loops.

$$L1: Q1 \cdot R(1\sim3)+(Q1-Q2) \cdot R(3\sim4)+(Q1-Q2-Q3) \cdot R(4\sim5)-Q8 \cdot R(21\sim25)=P(1\sim3)+P(3\sim4)+P(4\sim5)-(21\sim25) \quad (1)$$

$$L2: Q2 \cdot R(3\sim10)+(Q2-Q5-Q6) \cdot R(11\sim14)+(Q2-Q6) \cdot R(14\sim15)+Q2 \cdot R(15\sim16)-(Q3-Q4) \cdot R(6\sim16) \quad (2)$$

$$-Q3 \cdot R(4\sim6)-(Q1-Q2) \cdot R(3\sim4)=P(3\sim10)+P(11\sim14)+P(14\sim15)+P(15\sim16)-P(6\sim16)-P(4\sim6)-P(3\sim4)$$

$$L3: Q3 \cdot R(4\sim6)+Q4 \cdot R(6\sim8)+(Q2+Q3-Q7) \cdot R(8\sim9)-(Q1-Q2-Q3) \cdot R(4\sim5)=P(4\sim6)+P(6\sim8)+P(8\sim9)-P(4\sim5) \quad (3)$$

$$L4: (Q2+Q3-Q4-Q7) \cdot R(16\sim8)-Q4 \cdot R(6\sim8)+(Q3-Q4) \cdot R(6\sim16)=P(16\sim8)-P(6\sim8)+P(6\sim16) \quad (4)$$

(5)

$$L5: Q5 \cdot R(12\sim14)-(Q2-Q5-Q6)\cdot R(11\sim14)=P(12\sim14)-P(11\sim14) \quad (6)$$
$$L6: Q6 \cdot R(13\sim15)-(Q2-Q6)\cdot R(14\sim15)-Q5 \cdot R(12\sim14)=P(13\sim15)-P(14\sim15)-P(12\sim14) \quad (7)$$
$$L7: Q7 \cdot R(16\sim17)-(Q2+Q3-Q7)\cdot R(8\sim9)-(Q2+Q3-Q4-Q7)\cdot R(16\sim8)=P(16\sim17)-P(8\sim9)-P(16\sim8) \quad (8)$$
$$L8: (Q1+Q8)\cdot R(1\sim21)+Q8 \cdot R(21\sim25)=P(1\sim21)+P(21\sim25)$$

In equations above, (1~2) means resistance and pressure difference between point 1 and point 2 and resistance and pressure difference are constants. Before solving 8 simultaneous 1st order equations, it is much more important to calculate constants in each path. We can calculate constants by defining the oil supply characteristics.

Centrifugal Pump

The pumping capacity of centrifugal pump is concluded from pressure rise due to the rotation of shaft and propeller, gravity and pressure drop due to the resistance of oil path. Pressure Difference in pumping hole is

$$\triangle P = -\rho G (Z - Zo) + \rho \omega^2 R^2/2 + 0.1 \omega^2 R^2 \quad (9)$$

Resistance in pumping hole is given as eqn (10).

$$R = \triangle P / Q = 8 \mu L/\pi R^4 \quad \text{for Linear Pipe} \quad (10)$$
$$R = (1 + \xi_1 + \xi_2 + ...) V^2 / 2 G \quad \text{for Curved \& Enlarged Pipe}$$
$$R = C_0 \mu / \pi D^3 \quad \text{for Orifice}$$

Journal Bearing

As shown in Fig.3(a), journal bearing has the oil supply characteristics such as the natural flow rate owing to the rotation of shaft, the forced flow rate due to the pressure difference between the oil discharge hole and ends of bearing, the flow rate caused by oil groove. The natural flow rate includes the necessary flow rate at the point that the maximum surface pressure acts to form the minimum oil film thickness thicker than the limited oil film thickness during the rotation of shaft

$$Qn = 2 \pi R Cr N L (1 - \varepsilon^2) / (2 + \varepsilon^2) \quad (11)$$

And the leakage oil flow rate through ends of bearing.

$$Ql = \pi N D L \varepsilon \quad (12)$$

The forced flow rate generated by pressure difference between oil supply hole and ends of bearing is

$$Qp = \pi R Cr^3 (1 + 1.5 \varepsilon^2) \triangle P / (6 \mu L) \quad (13)$$

The flow rate of oil groove includes the flow rate generated by shaft rotation,

$$Qgc = V B H / 2 = \pi N D B H \cos \beta / 2 \quad (14)$$

and the flow rate generated by pressure difference

$$Qgp = \triangle P d H^2 A / 2 \mu F L \quad (15)$$

Thrust Bearing

As shown in Fig.3(b), when orbiting scroll receives the axial gas force as it orbits having eccentricity, it has a property to orbit inclined. An Equation about velocity and flow rate is obtained by applying boundary conditions on Navier-Stokes Equation about the bearing flow surface. The flow rate of thrust bearing has the oil supply property determined by surface pressure difference, rotation and geometric coefficient in case of non-circle. The flow rate without rotation effect is,

$$Qi = \pi H^3 \triangle P / 6 \mu \operatorname{Ln}(R_2 / R_1) \quad (16)$$

The flow rate with rotation effect is,

$$Qr = \pi \rho \omega 2 H^3 (R_2^2 - R_1^2) / 40 \mu \operatorname{Ln}(R_2 / R_1) \quad (17)$$

Required Flow Rate

It is required for bearings to operate on the safe operating area in surface pressure, velocity and bearing life. Particularly in association with oil supply, it is required that the minimum oil film thickness is thicker than the limited oil film thickness, and that the maximum bearing temperature is lower than the limited temperature. The limited oil film thickness is determined for the oil film parameter (λ) associated with surface roughness and state of assembly to be within the fluid lubrication range. Maximum bearing temperature is determined by the temperature which doesn't cause oil property change by heating and mechanical performance drop of bearing(under 150°C), and the limit of temperature rising between bearing inlet and outlet. The required flow rate is the flow rate that satisfies these two conditions.

RESULTS AND DISCUSSION

Fig.4 shows the comparison of the simulation and the experiment results on the flow rate according to oil levels. In case of no discharge hole above thrust surface the amount of Oil supply is measured in the experiment and calculated in the simulation. The simulation has the similar results with the experiment. In the experiment, the flow rate is increased by the rotation of propeller at all oil levels, and the pressure is risen as much as 540Pa by the rotation of propeller in the simulation result. The driving force of propeller is the function of oil density, velocity and diameter. Therefore, an equation below is obtained.

$$\triangle P = 0.1\, \omega^2 R^2 \qquad (19)$$

After the investigation on the simulation program as before, a simulation is performed in the compressor test condition [ARI]. Fig.5 is the simulation result on the flow rate of each lubricating part according to the frequency. Operating condition is 6.37Kgf/㎠, and 75 ℃. Flow rate to each lubricating part is the biggest in orbiting scroll journal bearing, and in the order of main journal bearing, thrust bearing and lower journal bearing. Fig.6 shows the minimum oil film thickness, the necessary flow rate to form the limited oil film thickness, and the cooling flow rate according to temperature rise in main journal bearing. Temperature rise in the bearing means mean temperature difference of bearing inlet and outlet. The rising of bearing temperature means the decrease of the minimum oil film thickness, the necessary flow rate to form the minimum oil film thickness and the cooling flow rate to cool bearings. Therefore, it is necessary to secure the required oil flow rate to secure proper oil film thickness and to prevent temperature from rising. We can know that the required flow rate with bearings depends on the cooling flow rate rather than the necessary flow rate to form oil film. Table 1 shows the safety factor of oil supply system of each bearing. As the cooling flow rate is bigger than the necessary flow rate to form oil film, the cooling flow rate becomes the required flow rate with bearings. On the whole, the supplied flow rate secures a safety factor more than 2 against the required flow rate.

CONCLUSIONS

1. The centrifugal force of propellers was determined by the experiment and simulation.
2. The required flow rate of each lubricating part was determined by the cooling flow rate rather than the necessary flow rate to form oil film.
3. The design parameter for the safety design of oil supply system was determined.

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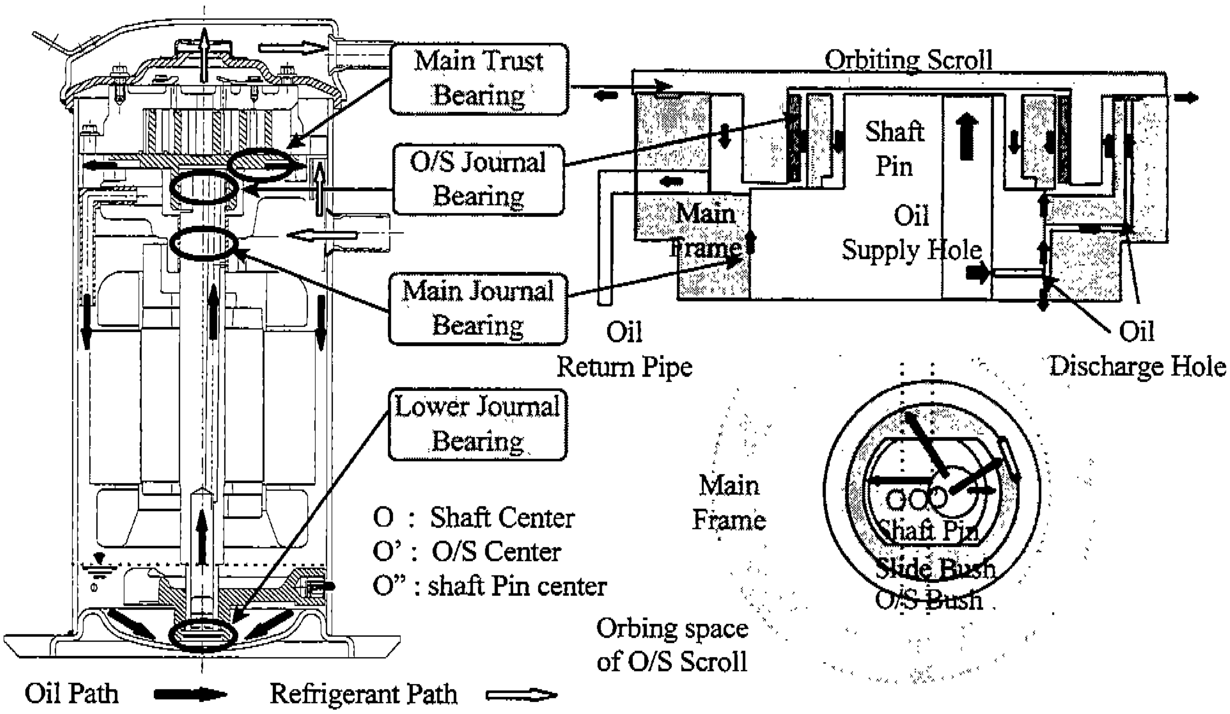


Fig. 1 Compressor Cross Section

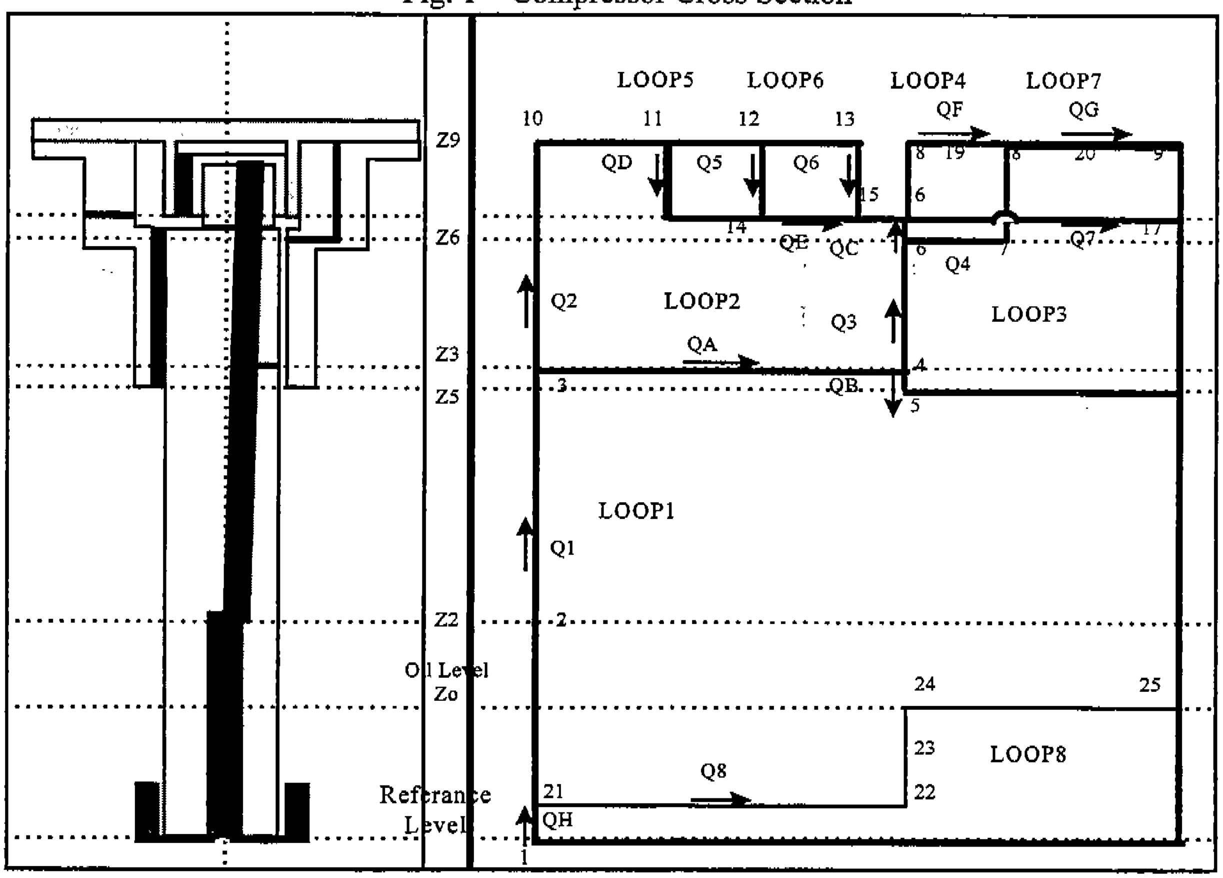
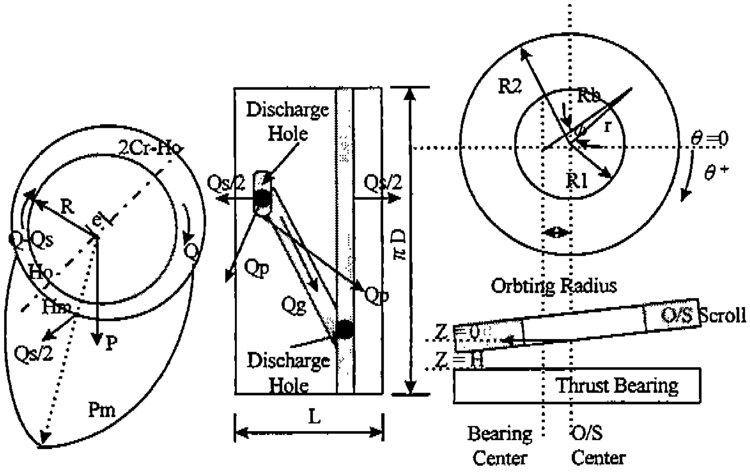


Fig.2 Equivalent Electrical Circuit



a) Journal Bearing b) Thrust Bearing

Fig.3 Bearing Cross Section

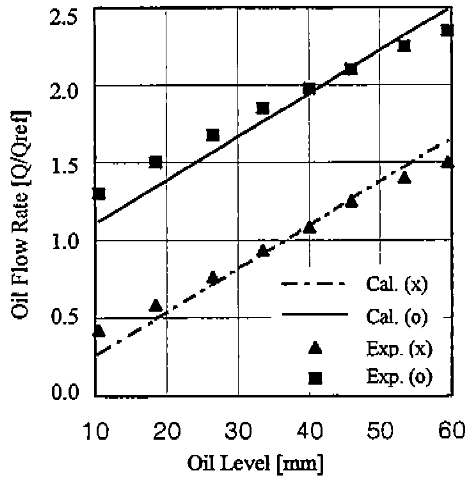


Fig.4 Flow Rate vs Oil Level By Propeller

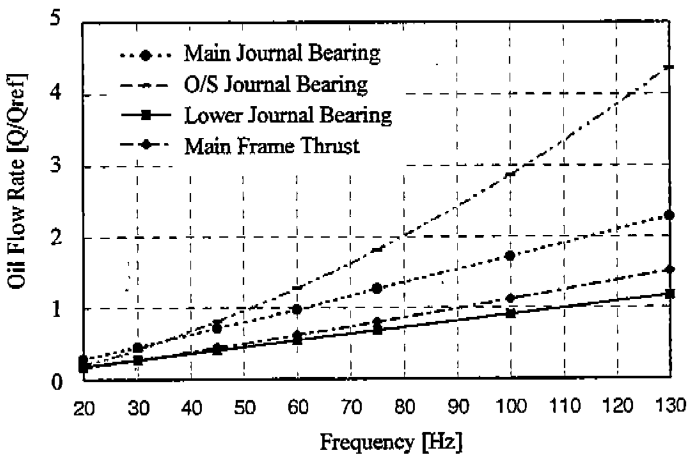


Fig. 5 Flow Rate vs Frequency in Each Bearing

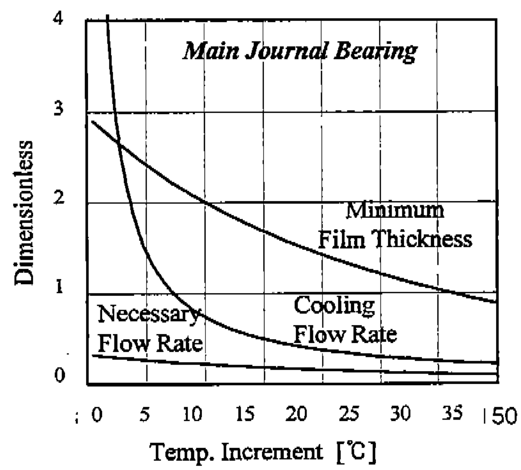


Fig.6 Required Flow Rate in 60Hz

Table 1 Safety Factor of Oil Supply System in 60 Hz

	Force	Sommerfeld Number	Power Loss	Necessary Flow Rate	Cooling Flow Rate	Required Flow Rate	Oil Supply Flow Rate	Safety Factor
Main Journal	3120	0.0129	36.4	0.33	0.49	0.49	1.14	2.33
O/S Journal	2710	0.0106	31.1	0.24	0.48	0.48	1.23	2.57
Lower Journal	405	0.0435	6.2	0.15	0.18	0.18	0.53	2.92
Main Thrust	1912	-	65.0	0.38	0.42	0.42	0.80	1.91