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H. Cho
J. T. Chung

B. K. Yoo
J. T. Chung

Y. Kim
J. T. Chung

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Honghyun Cho and Byoung-Kil Yoo, Graduate Student, Korea University.
Seoul, 136-701, KOREA; E-mail: h2cho@korea.ac.kr

*Yongchan Kim, PhD, Associate Professor, Dept. of Mechanical Engineering, Korea University.
Seoul, 136-701, KOREA; Tel.: 82-2-3290-3366; Fax: 82-2-921-5439
E-mail: yongckim@korea.ac.kr *Author for Correspondence

Jin Taek Chung, PhD, Associate Professor, Dept. of Mechanical Engineering, Korea University.
Seoul, 136-701, KOREA; E-mail: jchung@korea.ac.kr

ABSTRACT

An analytical study was carried out to investigate the performance of an oil supplying system of a variable speed compressor using a commercial CFD program. The simulations for the oil supplying system with the oil and air mixture were performed by varying compressor speed from 40 Hz to 90 Hz. Comparing the predicted with the measured data on the modified scroll compressor validated the simulation model. The predicted results were consistent with the test data with a maximum deviation of 12.8%. The oil flow rate significantly increased with a rise of compressor speed due to a higher oil flow rate from the swing pump and a greater centrifugal force on the oil gallery. This analytical method was proved to be very useful to determine the oil flow rate through the oil gallery and to design an optimized oil gallery for a variable speed scroll compressor.

NOMENCLATURE

H	:	cylinder height of swing pump	u	:	velocity vector
Q_{swp}	:	oil flow rate of swing pump	V	:	volume
R_1	:	piston radius of swing pump	w_v	:	vane width of swing pump
R_2	:	cylinder radius of swing pump	x	:	position vector
r_c	:	piston eccentric ratio of swing pump	z	:	vane length of swing pump

Greek letters

μ	:	viscosity [kg/m·s]
α	:	volume ratio
Θ	:	orbiting angle [rad]
ρ	:	density [kg/m^3]

Subscripts

i, j	:	vector tensor
q	:	qth phase in multi-phase
g	:	groove channel

INTRODUCTION

A proper lubrication in a scroll compressor is very important to achieve higher reliability and performance. A scroll compressor includes four major lubrication parts [1]: an orbiting scroll journal bearing, a main journal bearing, a main thrust bearing, and a lower journal bearing. A swing pump and oil gallery are core parts in the oil pumping system of a low-pressure type scroll compressor. The lubrication oil pumped from an oil reservoir by the swing pump is supplied to several lubrication parts through the oil gallery, which is located inside of a crankshaft. An analysis and optimum design of the oil supplying system is required to develop an energy efficient compressor with a higher reliability and durability.

Most previous analytical studies on the oil supplying system have conducted for a constant speed scroll compressor with an assumption of single-phase flow [2-5]. A fundamental study on the characteristics of the oil supplying system with a variation of operating conditions and geometry of oil gallery are limited in the open literature [6]. The objective of this work is to obtain performance data of the oil supplying system for a variable speed scroll compressor using a CFD simulation method. To validate the simulation results, the scroll compressor was tested with a variation of compressor speed and oil temperature.

MODELING AND EXPERIMENTS

Oil Supplying System

Fig. 1 shows the schematic of the oil supplying system in the variable speed scroll compressor. The swing pump, which is a rolling piston type, is located at the lower end of the crankshaft. The swing pump provides the oil to the oil gallery located inside of the crankshaft by passing through the sections of 1 and 2 as shown in Fig. 1. The oil gallery consists of a vertical tube, and its eccentricity becomes larger with an increase of the distance between the swing pump and a lubrication point. Since the swing pump does not have sufficient head to provide the oil to each lubrication part, an extra driving head is generated by the eccentricity of the oil gallery using a centrifugal force with a rotation of the crankshaft. The oil is supplied to the sub journal bearing at the lower side of the crankshaft and the main journal bearing at the upper side of the crankshaft. The oil at the exit port of 6 as shown in Fig. 1 is utilized to obtain the cooling and lubrication effects for the orbiting scroll journal bearing and the thrust journal bearing. In addition, the oil entering into a compression chamber plays an important role in preventing tangential leakage by sealing tip clearance. In this study, the simulations were conducted by setting the oil gallery in the crankshaft as a boundary.

Oil Flow Rate in the Swing Pump

The oil flow rate of the swing pump is calculated by using geometric analysis, as shown in Fig. 2. The volume of the swing pump can be expressed as

$$V(\theta) = H\left[\pi(R_2^2 - R_1^2) - \frac{1}{2}\left\{R_2^2\theta - R_1^2\theta_1 - (R_2 - z)r_c \sin\theta + w_v z\right\}\right] \quad (1)$$

Therefore, the variations of volume, vane length, and θ_1 with respect to orbiting angle (θ) can be given by

$$\frac{dV}{d\theta} = -\frac{1}{2}H\left\{R_2^2 - R_1^2\frac{d\theta_1}{d\theta} + \frac{dz}{d\theta}r_c \sin\theta - (R_2 - z)r_c \cos\theta + w_v \frac{dz}{d\theta}\right\} \quad (2)$$

$$\frac{dz}{d\theta} = r_c \sin\theta + \frac{r_c^2 \sin\theta\cos\theta}{\sqrt{R_1^2 - r_c^2 \sin^2\theta}} \quad (3)$$

$$\frac{d\theta_1}{d\theta} = 1 + \frac{r_c \cos\theta}{\sqrt{R_1^2 - r_c^2 \sin^2\theta}} \quad (4)$$

The oil flow rate of the swing pump is determined by the following equation.

$$Q_{swp} = -\frac{dV}{dt} = -\frac{d\theta}{dt}\frac{dV}{d\theta} = -w\frac{dV}{d\theta} \quad (5)$$

CFD Modeling

To simplify the analysis of the oil gallery, the following assumptions are made in the present model [4]: (1) the oil is in incompressible laminar flow, (2) there is no phase change in the oil, (3) the heat transfer between oil and air is negligible, and (4) the properties of a working fluid inside of a control volume are constant.

Based on the measured dimension for the oil gallery, a three-dimensional CAD model was developed to write input data for the grid generation program. The generated grid for this study is shown in Fig. 3.

The VOF (volume of fluid) model was used to analyze two-phase mixture of oil and air in the oil gallery. Governing equations of the oil supplying system were derived from continuity and momentum equations. The governing equations are given by

$$\frac{\partial \alpha_q}{\partial t} + u_i \frac{\partial \alpha_q}{\partial x_i} = 0 \tag{6}$$

$$\frac{\partial}{\partial t}\rho u_j + \frac{\partial}{\partial x_i}\rho u_i u_j = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_i}\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) + \rho g_j + F_j \tag{7}$$

The rotating reference frame and moving mesh method considering a rotation of the crankshaft was used in the numerical simulation. The velocity at the inlet of the oil gallery was determined by using the outlet velocity of the swing pump. The boundary condition at the outlet of the oil gallery was evaluated from the exit pressure of the oil gallery. A body-forced weighted method was used in the pressure interpolation. The continuity and momentum equations were solved by the PISO (pressure-implicit with splitting of operators) method, which tends to have fast convergence at non-steady state conditions. In this study, the FLUENT [7] that is a commercial CFD simulation program was used. The CFD simulations were performed at compressor speeds of 40, 60, 80, and 90 Hz and oil temperature of 40°C.

Experimental Apparatus and Test Method

In order to validate the numerical model, the oil flow rates at the exit ports of 3, 5 and 6 as shown in Fig. 1 have to be measured. However, in this study, the oil flow rate at the outlet port of the section 6 was measured because the measuring ports in the journal bearing (exit ports of 3 and 5) may significantly affect the performance of the compressor. To measure the oil flow rate at the exit port of 6, the compressor upper part and orbiting scroll were separated from the compressor body.

Fig. 4 shows the schematic of test setup for oil flow rate measurement. Two oil ports were installed in the setup: one was the port for oil supply to the reservoir at the bottom of the compressor, while the other port was designed for the measurement of the discharged oil from the outlet of 6. The oil temperature supplied to the reservoir of the compressor was adjusted by using a constant temperature bath for the oil tank. The amount of the discharged oil from the exit port of 6 was measured for 10 minutes at 20-second interval. The oil used in this study was the SUNISO 3GS.

RESULTS AND DISCUSSION

Table 1 shows the comparison of the measured oil flow rate with the predicted values from the CFD simulation. The deviations between the predicted and the measured oil flow rates to the scroll journal bearing were 12.8, 8.5, 8.9, and 11.3% at 40, 60, 80, and 90 Hz, respectively. These deviations are due to some errors in the assumptions of the model, and the uncertainties of the measurements.

Fig. 5 shows the measured oil flow rate to the scroll journal bearing with a variation of oil temperature. The oil flow rate slightly increases with an increase in oil temperature due to a rise of the viscosity. The oil flow rate increases by 4.8% at a frequency of 40 Hz as the oil temperature increases from 40 to 60°C. The oil flow rate linearly increases with a rise of compressor speed due to an increase of oil pumping rate by the swing pump and an addition of centrifugal force. Generally, the influence of oil temperature on the oil flow rate is relatively small, while the effect of the compressor speed is significant.

Since the model was validated with the measured data, the simulations were performed with a variation of compressor speed and oil temperature. The predicted free surface of the oil at each cross section is shown in Fig. 6. The bottom section of the oil gallery is fully filled with the oil. As the oil moves toward the upper section of the oil gallery, the oil exists at the outside of the tube due to an increase in centrifugal force. As the compressor speed increases, the oil film becomes thinner. When the compressor speed is high, the sufficient oil is fed into the upper section of the oil gallery even though the eccentricity of the oil gallery is small.

Fig. 7 shows the predicted velocity fields at the outlets of the sub journal bearing and the main journal bearing. Since the oil gallery of the sub journal bearing is not eccentric, the velocity field in the sub journal bearing rotates centrically with respect to the axis of the crankshaft. The velocity at the outlet of the sub journal bearing forms complicated velocity vectors having a movement toward one direction. The velocity fields in the main journal bearing show an eccentric profile due to an eccentricity of the oil gallery. The velocity at the outlet of the main journal bearing represents large velocity vectors due to a higher centrifugal force.

Figs. 8 through 10 show the variation of the predicted oil flow rate at each outlet with respect to compressor speed. For a lower compressor speed, since the oil flow rate provided by the swing pump as well as the centrifugal force are relatively small as compared to a higher compressor speed case, more time is required to establish steady state operation of oil pumping into each bearing. In addition, it takes relatively long time to provide the oil into each outlet at low compressor speed. It takes 0.5 second to initiate oil circulation into the scroll journal bearing and the main journal bearing at a compressor speed of 40 Hz, while it takes less than 0.3 second for the frequencies above 60 Hz. It is notable that there is an overshoot of oil flow rate in the scroll journal bearing at 90 Hz. This is due to an excessive supply of oil at high frequency. When the oil is supplied redundantly, the friction loss may be increased due to an increase of solubility with a refrigerant.

An optimum design of the oil supplying system as a function of compressor speed is very important to achieve higher reliability and durability of the compressor. The present results show the trends of the performance of the oil supplying system with the oil and air mixture as a function of operating parameter. Therefore, a further study is required on the oil pumping system with the oil and refrigerant mixture.

CONCLUSIONS

The performance of the oil supplying system for the variable speed scroll compressor was investigated by using the commercial CFD program. The simulations were performed with a variation of compressor speed and oil temperature. Generally, the oil flow rate linearly increased with an increase of compressor speed. For a higher frequency, excessive oil can be provided to the oil gallery, which can yield a higher friction loss. The predicted results were consistent with the test data with a maximum deviation of 12.8%. The analytical model can provide a useful tool to design an optimum oil supplying system for a variable speed compressor.

ACKNOWLEGEMENTS

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Table 1 Comparison of the predicted with the measured oil flow rate at T_{oil}=40°C

Compressor speed (Hz)	Oil flow rate (ml/s)		Accuracy (%)
	Simulated	Measured	
40	6.64	7.63	12.8
60	9.83	10.75	8.5
80	12.30	13.51	8.9
90	12.75	14.37	11.3

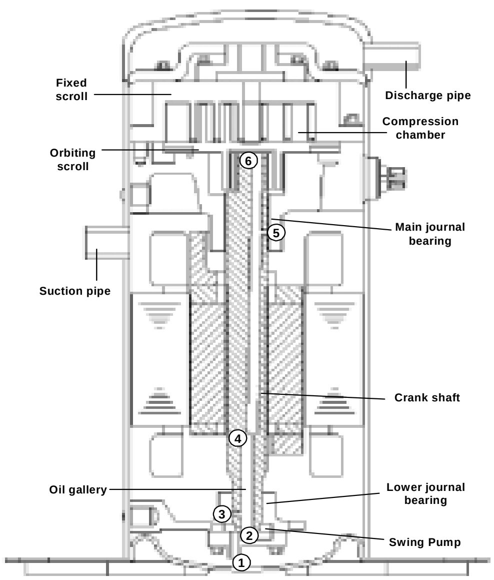


Figure 1: Schematic of the oil supplying system

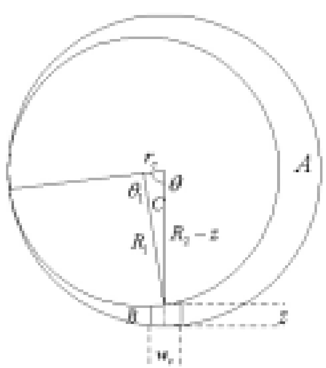


Figure 2: Geometry of cylinder and piston for The swing pump

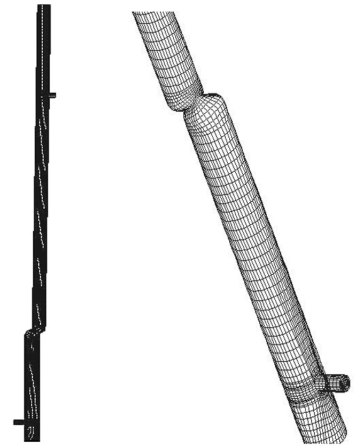


Figure 3: Grid system of the oil gallery

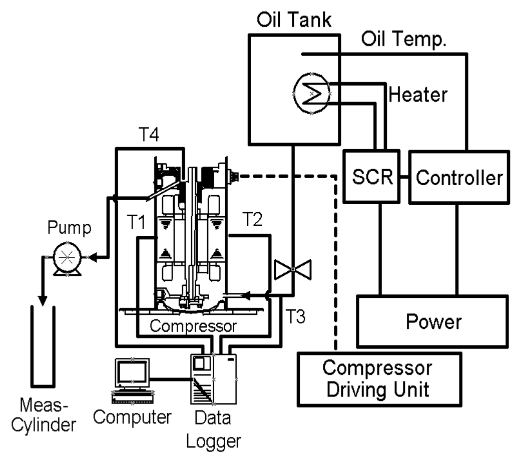


Figure 4: Schematic of the test setup

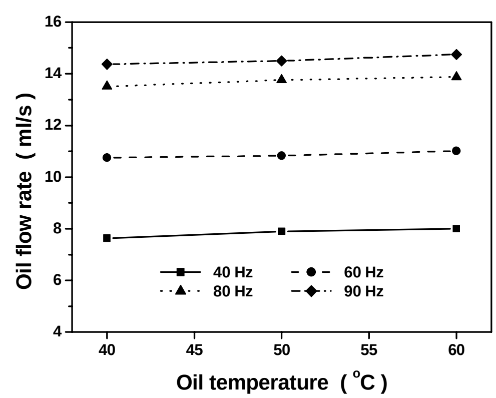


Figure 5: Oil flow rate to scroll journal bearing with oil temperature

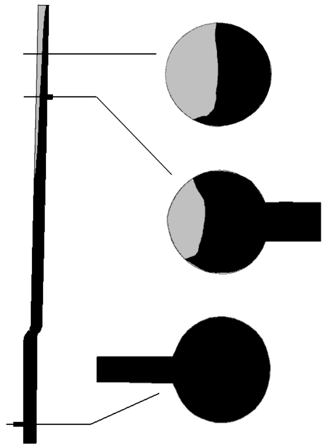
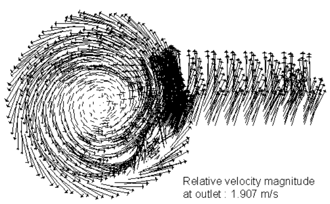
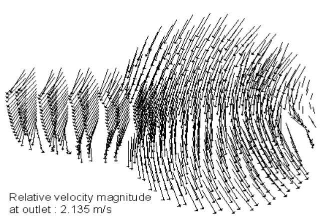


Figure 6: Shape of oil free surface at 60 Hz



(a) Sub journal bearing



(b) Main journal bearing

Figure 7: Outlet velocity fields at 60 Hz

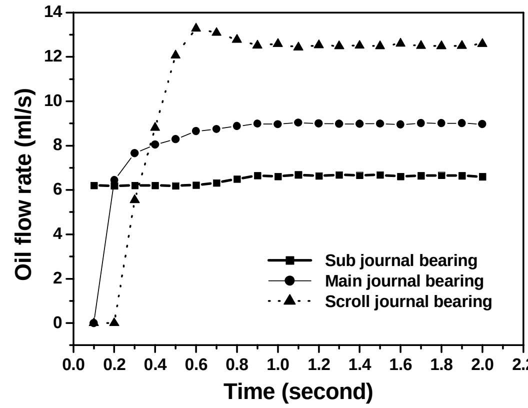


Figure 8: Oil flow rate to the bearings at 40 Hz

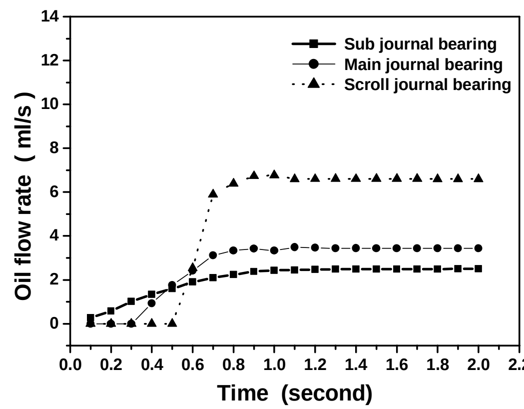


Figure 9: Oil flow rate to the bearings at 60 Hz.

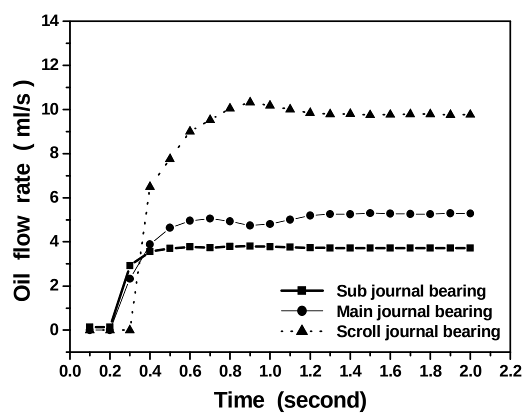


Figure 10: Oil flow rate to the bearings at 90 Hz