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INVESTIGATION ON THE OIL SUPPLY SYSTEM OF A SCROLL COMPRESSOR

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ABSTRACT

The startup process of the scroll compressor oil supply system is important for scroll compressor reliability and performance. Since the bearings are running dry when a scroll compressor starts, the bearing dry time is an important design parameter as well. When a scroll compressor starts, the oil sump, bearings, and shaft interact with each other and form an oil supply system. The working mechanism of this oil supply system is complex and dynamic. To characterize the oil supply system and achieve high reliability of scroll compressors, a numerical study is embarked. The model integrates the oil sump, oil gallery inside the shaft, and bearing system together to define the major features of the oil supply system. Two operating parameters are studied using the numerical model developed. The first parameter is the oil depth inside the oil sump. When oil is pumped into the gallery and bearings, the oil sump depth changes gradually with time. This change of depth impacts the lubrication capacity of the oil system. Since the sump has an open surface, the surface shape is a function of time and waves form inside the compressor. These surface waves interact with oil flow inside the gallery and bearings. It is necessary to include them in the analysis to determine the performance characteristics of the oil supply system. The second parameter is the fluid viscosity. The change in oil viscosity is caused by refrigerant absorbed in the oil and heat transfer between the motor and oil. A total of four cases are analyzed comparatively. The physical mechanisms controlling oil supply rate are defined. The oil supply rate and the dry periods of the bearings are obtained. The results can be used to design and optimize the oil supply system of scroll compressors.

1. INTRODUCTION

The oil flow inside the oil pump and sump is a complex physical process. When a scroll compressor starts, oil remains inside the sump. The bearings and pump are dry. As the motor rotates, the centrifugal force gradually overcomes gravity and surface tension. The pressure generated by the centrifugal force inside the flow field drives the oil into the oil gallery. The shape of the oil surface is determined by the balance of gravity, centrifugal force, and surface tension as well as gas flow above the oil surface. The rising oil level eventually reaches the bearings and ports. When the oil pressure overcomes the resistance of the bearings and ports, the oil starts flowing into bearings and ports in the system. The oil supply rates to bearings are resulted from design features and working conditions of oil pump and bearings. The design objective of the oil supply system is to provide sufficient oil to the bearings and minimize the bearing dry period.

Kim *et al.* (1998) showed the oil flow rate is sensitive to the shape of the oil gallery inside the crank shaft. Cho *et al.* (2002) and Drost *et al.* (1992) found the oil flow rate changes with temperature (viscosity). Drost *et al.* (1992) also obtained the data that indicates the oil flow rate is a function of sump oil level. CFD analyses (Cho *et al.*, 2002 and Bernardi, 2000) for the oil flow inside the gallery indicate that the oil supply system may not be filled with oil. These studies reveal the complex nature of the flow physics inside the scroll compressor oil supply system and suggest the further study of the system. The key elements that need to be included in the oil supply system study are the oil sump, bearing system, vent, and ports.

The current investigation of the scroll compressor oil supply system starts from the fundamental physics of the oil pumping flows. The study includes the interactions of the oil pump, sump, as well as the bearing system. The

physics in the analysis covers gravity, centrifugal force, and surface tension. The details of the oil supply system geometry are included. The study characterizes the physical process controlling the oil supply system and determines the effect of the design and operating parameters on the oil supply system performance.

2. METHODOLOGY

The oil supply system is an integrated part of a scroll compressor. Figure 1(a) shows the schematic of a scroll compressor with the oil supply system. The oil supply system is located in the low-pressure side of the scroll compressor. Oil is pumped up from oil sump and feeds the lower main, upper main and orbiting bearings. There are a few ports connected to the oil gallery to send gas out and discharge the dirt in the system. Figure 1(b) shows the flow domain in the oil supply system. The lower main and orbiting bearings are end-port fed and the upper main bearing is side-port fed.

The oil supply system includes three parts: oil sump, oil gallery, and bearings. The multiphase flow regime shows distinguishable features during the startup process in different parts of the system. The refrigerant gas-oil flow contains both continuous and dispersed gas and oil phases. To simulate this process, the flow inside the oil supply system is treated as a heterogeneous Eulerian-Eulerian multiphase flow. A commercial solver based on the finite volume method is used for this analysis. The oil in the shaft, including flows in the gallery and feed holes, is calculated in a rotating frame of reference. The oil sump is calculated in a stationary frame of reference. All the field quantities are obtained at each time step. The oil supply rates to bearings and ports are also calculated at each time step. The shaft starts at 0 rpm and gradually accelerates to 3500 rpm, its maximum speed. Equation 1 describes the motor speed variation during the startup process:

$$\omega = \begin{cases} 170525 * t^3 - 57687 * t^2 + 7378.8 * t, & \text{when } \omega < 367 \text{ rad/s} \\ 367, & \text{when } \omega > 367 \text{ rad/s} \end{cases} \quad (1)$$

The rotating speed of the oil pump is shown Figure 2 as a function of time when motor starts at time equals zero.

The initial condition of the numerical simulation is chosen as the start time of the shaft rotation. The oil level is constant in the sump and shaft when the transient simulation starts. The calculations continue until the oil flow reaches its steady state and is no longer a function of time.

To characterize the performance of the oil supply system at difference working conditions, a total of four cases with different viscosity and sump oil levels are analyzed. The runs are designed as 2^2 factorial. The parameters of the cases analyzed are listed in Table 1.

Case	1	2	3	4
Viscosity, cp	5	26.4	5	26.4
Oil level, cm	5	5	10	10

Table 1. Parameters of the cases analyzed.

Since the flow inside the bearings has a smaller scale than the flow in other parts of the oil supply system, the oil and gas flows inside the bearings are simulated separately. The total of three bearings is simulated. These bearings can be classified into two categories. The upper main bearing is simulated as a side-port fed bearing (Figure 3(a)). The lower main and orbiting bearings are simulated as end-port fed bearings (Figure 3(b)). Since both gas and oil go through the bearings, the oil and gas are simulated separately to obtain the flow characteristics in side-port fed and end fed bearings. The obtained flow characteristics are used in the startup process simulation.

Figure 1. Schematic of the scroll compressor (a), the geometry of the oil supply system (b), and location of vent and ports (c).

Figure 2. Rotating speed of scroll compressor oil pump.

Figure 3. Side-port fed bearing (a) and end-port fed bearing (b).

3. FEATURES OF FLOW FIELD

The driving forces of the flow inside the oil supply system are centrifugal and gravitational forces as well as surface tension. These forces induce different features of the flow field during the oil supply system working process.

The first notable feature of the oil flow inside the oil sump is the wave generation and dispersion (Figure 4). When flow rate is low at the beginning of the pumping process, the wave is stable with a distribution of the wavelength proportional to the distance from the shaft. The amplitudes of the waves are similar. As flow rate increases, the waves generated near the pump inlet are suppressed. The oscillations in the region near the inlet disappear due to the strong damping induced by the large mean velocity. The amplitude of the waves decays exponentially as the waves propagate from the shaft.

In the shaft, the oil goes up along the circumference of the pickup tube. After oil reaches above the lower main bearing, the oil stays in the one side of the gallery that has a larger radius. The oil first rises into the gallery from the oil sump located at the bottom of the compressor. When the oil takes more space in the top cavity the oil starts to flow back down into the oil gallery to fill the rest of the space. The wave in front of the oil flow inside the gallery can be seen in the Figure 5. The space left in the gallery is located in the upper cavity and lower part of the gallery. This space is eventually filled up simultaneously. It is interesting that the empty space is not filled bottom up.

Since the vent is located in the side of the gallery that has a smaller radius, the oil with higher density is spun away by centrifugal force from this region (Figure 5). There is a clearly defined interface between the spaces occupied by gas and oil. The vent is connected to the gas pocket during the startup process. If there is gas evaporating from the oil, the vent provides an exit for gas to leave the gallery.

The oil filling process of the annulus beneath the lower main bearing provides a good example for the complexity of the flow field. The oil enters the two feed holes along the wall of the pickup tube. The feed holes are partially filled with oil. Because the shaft rotates counter clockwise, the oil fills one side of the feed holes. As the shaft rotates, the oil gradually fills the annular space (Figure 6). Since there is a port located at the annulus, the rotation of the two feed holes causes the oscillations inside the annulus. As a result, the oil flow through the lower the main bearing and the port shows a periodic pattern with a frequency of 117 Hz (Figure 7 and Figure 8).

Figure 4. Wave pattern and surface velocity distribution at (a) 0.1695 second, (b) 0.475 second, (c) 0.972 second, (d) 1.082 second.

Figure 5. Oil fraction as a function of time inside the gallery.

Figure 6. Oil filling process of the annulus beneath the lower main bearing.

4. EFFECTS OF VISCOSITY AND OIL SUMP DEPTH

The oil flow rates through bearings and ports are plotted as functions of time (Figure 7 and Figure 8). The oil flow rates with the same depths and different viscosity values are shown in the plots for comparison. To generalize the results and to measure the relative change, the oil flow rate is plotted as a non-dimensional quantity. The reference flow rate is the oil flow rate to the orbiting bearing in case 3 (Table 1). The impact of viscosity of the oil on the pumping process can be shown in Figure 7 and Figure 8. The overall features of the oil flow rates to the bearings show that the orbiting bearing enjoys the largest oil supply rate while the upper main and lower main bearings obtain similar amounts of oil per unit time. When the shaft starts to rotate, the order of the bearings receiving the oil is the lower main, upper main, and orbiting. The length of dry time ranges from 0.2 to 0.5 second. The oscillations

caused by the oil feed holes on the shaft can be seen in the oil supply rate to the lower main bearing and oil flow rate through the port.

After the oil flow reaches its steady state, the oil supply rates to bearings are larger for smaller viscosity value. These results qualitatively agree with the experimental results obtained by Cho *et al.* (2002) and Drost *et al.* (1992). Since their test data was obtained without an orbiting bearing, a direct or quantitative comparison is difficult. However, both tests and numerical data point to the same direction: the variation of the oil supply rate to bearing caused by the viscosity change is small compared to the magnitude of the oil flow rate. Conceptually, lower viscosity makes it easier for the oil to flow in the gallery and inside the bearings. This provides a physical ground for the results obtained. A lower viscosity also reduces the bearing dry time (Figure 7 and Figure 8). The magnitude of the dry time reduction is also in the same order as the total oil supply rate to the bearings. The features of the oil supply curves are similar for the upper main and orbiting bearings. The oil supply rates for the lower main bearing are different for two viscosity values. The oil supply rate for higher viscosity (26.4 centipoises) takes longer to reach its maximum value.

The viscosity also changes the oil flow rate through the ports in the system. The oil flow rate through the port at the annulus below the lower main bearing shows different features. The higher viscosity value makes more oil pass through the port. This partially results from the lower flow rates through the bearings. The resistances through the bearings are high and the oil takes the least resistance route. The most noticeable feature of oil flow through the port is the periodic oscillation. This phenomenon is caused by the rotation of the feed holes. The magnitude of the oscillation is bigger for lower viscosity cases.

There is very little oil going through the vent (Figure 7 and Figure 8) since the vent is located on the inner side of the oil gallery. The centrifugal force is proportional to the radial distance of the oil surface from the shaft centerline and works against the flow entering the vent. Since the total pressure of the oil flow is generated by the centrifugal force, the pressure of the oil stream near the vent entrance can not overcome the centrifugal force before the oil reaches the top cavity. Therefore the oil will not enter the vent before the pressure in the system exceeds the centrifugal force at the entrance of the vent. When oil eventually fills the cavity to the top of the shaft, the oil flow to orbiting bearing is saturated. The pressure starts to build up due to the larger radial distance from the centerline of the shaft. The static pressure near the vent entrance gradually approaches the centrifugal force at the same location. The oil starts to flow into the vent. Figure 7 through Figure 8 indicate the time when oil enters the vent after the oil supply rate to the orbiting bearing reaches its plateau. For the high viscosity cases, the oil enters the vent earlier since the resistance to the oil flow entering orbiting bearing is higher and pressure builds up quicker in the oil supply system.

When the oil level in the sump becomes higher, the oil supply rates to the bearings increase. The oil flow through the vent and port of the oil supply system also increases when the oil level goes higher in the oil sump. These findings are in line with the experimental results obtained by Drost *et al.* (1992). Their data shows that the total oil flow rate increases with the oil depth in the sump. The overall profiles of oil supply rates are similar for the two depths analyzed. However, the bearing dry time is significantly prolonged. The time for oil flow through the port to reach its plateau increases 42%. The overall profiles of the oil flow through the vent and port are similar. The wear and tear of the bearings need to be assessed to obtain a better reliability of the bearing system.

Figure 7. Oil flow rate through bearings (a) and ports (b) at the sump oil depth of 10 cm.

Figure 8. Oil flow rate through bearings (a) and ports (b) at the sump oil depth of 5 cm.

5. CONCLUSIONS

A CFD model for oil supply system of the scroll compressors is developed. This model includes all components of the oil supply system and flow physics. The results provide physical insights for the mechanisms controlling the bearing supply system of scroll compressors.

Lower viscosity of the oil that results from refrigerant absorption in the oil and higher temperature increases bearing oil supply rate. The total amount of the oil pumping through the system is also larger.

A lower oil level in the sump reduces the bearing oil supply rate and prolongs the bearing dry time significantly.

NOMENCLATURE

$\frac{Q}{t}$	flow rate	ω	angular speed
	time	ω_{ref}	reference value

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