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The Characteristics of Tangential Leakage in Scroll Compressors for Air-conditioners

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Abstract

Although the leakage loss occupies a considerable portion of the total loss in a scroll compressor, studies on the leakage phenomena under the actual operating conditions are insufficient because of the complicated shape of a scroll compressor. Therefore in this present study, we executed two researches for obtaining basic data about the tangential leakage phenomena at radial clearance. Firstly, experiments were performed to find the characteristics of the tangential leakage between pockets in a scroll compressor under actual operating conditions. The quantities of tangential leakage between pockets could be measured with orbiting angle, operating condition, and radial clearance between the fixed and the orbiting scroll. Especially, the flow coefficient was yielded from the experimental values. The flow coefficient value of tangential leakage in a scroll compressor was always constant under the condition of constant value of oil concentration regardless to the shape of leakage path, the radial clearance, and the pressure condition. Later, simulations which employed this flow coefficient were conducted to examine the influence of tangential leakage on the performance of a scroll compressor under various conditions. Through this analysis, we were able to find out the quantity of tangential leakage under actual operating conditions and the influence of tangential leakage on the performance of a scroll compressors.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Leakage area (m²)</td>
</tr>
<tr>
<td>C</td>
<td>Flow coefficient</td>
</tr>
<tr>
<td>k</td>
<td>Specific heat ratio</td>
</tr>
<tr>
<td>m₈</td>
<td>Theoretical mass of refrigerant on completion of suction (kg)</td>
</tr>
<tr>
<td>m₈_leak</td>
<td>Mass of tangential leakage to suction pocket during the suction process (kg)</td>
</tr>
<tr>
<td>m₈ideal</td>
<td>Ideal mass flow rate of refrigerant (kg/sec)</td>
</tr>
<tr>
<td>m</td>
<td>Mass flow rate of refrigerant by leakage loss (kg/sec)</td>
</tr>
<tr>
<td>m₈</td>
<td>Mass flow rate to test compressor inlet (kg/sec)</td>
</tr>
<tr>
<td>m₈₁</td>
<td>Leakage mass flow rate at high-pressure part (kg/sec)</td>
</tr>
<tr>
<td>m₈₁₂</td>
<td>Leakage mass flow rate at middle-pressure part (kg/sec)</td>
</tr>
<tr>
<td>m₈₂₅</td>
<td>Leakage mass flow rate at low-pressure part (kg/sec)</td>
</tr>
<tr>
<td>m₈₆</td>
<td>Bypass mass flow rate at P₆ pocket bypass valve (kg/sec)</td>
</tr>
<tr>
<td>m₈₁</td>
<td>Bypass mass flow rate at P₁ pocket bypass valve (kg/sec)</td>
</tr>
<tr>
<td>m₈₂</td>
<td>Bypass mass flow rate at P₂ pocket bypass valve (kg/sec)</td>
</tr>
<tr>
<td>P₆</td>
<td>Pressure of upstream (Pa)</td>
</tr>
<tr>
<td>P₁</td>
<td>Pressure of downstream (Pa)</td>
</tr>
<tr>
<td>R</td>
<td>Gas constant of refrigerant (kJ/kg · mol · K)</td>
</tr>
<tr>
<td>Tₘ</td>
<td>Temperature of upstream (°C)</td>
</tr>
</tbody>
</table>
Introduction

The higher standard of living makes not only a high performance of air-conditioner but also lower noise and vibration important. A scroll compressor, which meets these needs is built in the room and packaged air-conditioners. The scroll compressor attains the lower noise and reduced vibration by virtue of low driving torque variation. Also the scroll compressor has many merits; lower pressure loss during the suction process, little re-expansion, and a simplicity of structure. But the scroll compressor has a difficulty in machining the scroll wrap compared with a reciprocating compressor and a rotary compressor. And special mechanism is needed for preventing the leakage of high-pressure refrigerant. The back-pressure mechanism and the tip-seal mechanism are applied for preventing the radial leakage at axial clearance and the variable radius crank mechanism is used for the tangential leakage at radial clearance.

Because leakage loss occupies a considerable portion of the total loss in a scroll compressor, researches on the mechanism for reduction of leakage have been made. But studies on the leakage under the actual operating conditions are insufficient because of the complicated shape of a scroll compressor. Therefore, in this research we made a test-compressor which has a little changes in real compressor and realized a real leakage under the actual driving condition. And we developed the indirect flow measurement method because of impossibility of measuring the leakage directly. Tangential leakage mass flow rate in radial clearance is measured by this method and flow coefficient is picked out from the experimental values. Using this flow coefficient simulation was conducted to examine the influence of tangential leakage on the performance of a scroll compressor under various conditions.

Experiment

An experiment was performed to find out the characteristics of the tangential leakage between pockets in a scroll compressor under actual operating conditions. Fig.1 is a schematic of experimental apparatus. The experimental apparatus is consisted of three parts; an air-conditioner part, a mass flow measuring device parts and a test compressor part. The air-conditioner part is a device to supply high pressure and temperature of refrigerant to a test compressor. Heat transfer rate of the condenser with revolution of fan controls discharge pressure and the area of expansion valve controls suction pressure. The mass flow measuring device part is made for measuring the flow rates at many parts with one mass flow meter. And the test compressor has the same structure as real scroll compressor except a driving motor which is removed. Fig.2 is a sectional view of the test compressor. The radial clearance is controlled by changing the eccentricity of crank pin, and the axial clearance became nearly zero by perfect contact between the tips of orbiting scroll and the fixed scroll. So we can obtain the only tangential leakage at the radial clearance.

The indirect flow measuring method is described at Fig.3. Pressure in a pocket is controlled by adjusting bypass (m₁, m₂) valve to set pressure conditions. And the leakage flow rates between the pockets can be calculated by the continuity equation. Then, the pressure condition at each pocket for the experiment is arranged at Table 1. In a pressure condition, the value of P₁ and P₂ increase with the orbiting angle but the value of P₃ and P₄ is constant because P₃ and P₄ is always adjusted to the suction pressure and the discharge pressure, respectively. And in an orbiting angle, P₁ and P₂ have the same value because the scroll compressor has fixed compression ratio.

Theoretical analysis

At the radial clearance of the scroll compressor, the leakage of high-pressure refrigerant due to the difference of pressure between two pockets can be theoretically obtained by using the flow coefficient equation of steady state one-dimensional compressible fluid, considering the refrigerant as a single phase isentropic nozzle flow.

Generally, the flow rate increases with the increase of pressure difference. But in case of the fixed upstream temperature and pressure and the reducing downstream pressure, choking phenomenon occurs. When the
radial leakage path is considered as a converging nozzle, the leakage at the clearance is governed by Eq.(1). In the Eq.(1), the factors which influence on the quantity of leakage are leakage area, pressure ratio between upstream and downstream \((P_{up}/P_{up})\), and conditions\((P_{up}, T_{up})\) of upstream.

\[
\dot{m} = \begin{cases} 
\frac{CAP_{up}}{2k} \left( \frac{2k}{(k-1)RT_{up}} \left( \frac{P_{dn}}{P_{dn}} \right)^{\frac{2k}{k+1}} - \left( \frac{P_{dn}}{P_{up}} \right)^{\frac{2k}{k+1}} \right) & \left( \frac{P_{dn}}{P_{up}} \right) \geq \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}} \\
\frac{CAP_{up}}{k} \frac{k+1}{k-1} \left( \frac{2}{(k-1)RT_{up}} \right)^{\frac{k+1}{k-1}} & \left( \frac{P_{dn}}{P_{up}} \right) \lt \left( \frac{2}{k+1} \right)^{\frac{k}{k-1}}
\end{cases}
\]

(1)

In the present research, the flow coefficient can be acquired by putting the experimental values, that is the mass flow rate, temperature, and the leakage area at the radial clearance into the Eq.(1). Then, using the acquired flow coefficient the influence of the tangential leakage on the performance of the scroll compressor such as volumetric efficiency, indicated work, and so on.

Generally, the definition of the volumetric efficiency of a compressor is like the Eq.(2), the ratio of the flow rate in the actual mass flow rate of the cycle to the ideal mass flow rate.

\[
\eta_v = 1 - \frac{m_{\text{leak rad}} + m_{\text{leak tan}} + m_{\text{heat}}}{m_{\text{ideal}}}
\]

(2)

For a convenience of the analysis on the effect of tangential leakage, if we assume that there is no leakage at the axial clearance and no heat transfer in the suction process, and we define \(m_i\) and \(m_{\text{leak,tan}}\) as theoretical mass of refrigerant on completion of suction and mass of tangential leakage to suction pocket during the suction process respectively, the Eq.(2) can be converted into the Eq. (3). So we intended to analyze influence of the tangential leakage by using the Eq.(3).

\[
\eta_v = 1 - \frac{m_i - m_{\text{leak,tan}}}{m_i}
\]

(3)

**Result and Discussion**

**Results of Experiment**

The radial clearance was measured with thickness gauge of 10\(\mu\)m order. Table 2 shows the actual measured clearance. The clearance of the sample B is 50\(\mu\)m larger than that of sample A. Fig.3 describes that the tangential leakage increases with the radial clearance shown at Table 2. It can be confirmed by experiment that the quantity of the leakage is proportional to the radial clearance as known as Eq.(1).

Fig. 4 shows the leakage mass flow rate with discharge pressure and orbiting angle in the case of the sample A. Because the change of the discharge pressure has no effect on the compression process due to the fixed compression ratio of the scroll compressor, \(m_{1s}\) and \(m_{12}\) before discharge starting angle (orbiting angle 80°) are not affected by the change of discharge pressure. But, after discharge starting angle, \(P_1\) pocket (high pressure pocket) and \(P_a\) pocket are merged, and then the pressure of \(P_1\) pocket is equal to the pressure of \(P_a\) pocket. Therefore, \(m_{12}\) after discharge starting angle and \(m_{1s}\) increase with the rising discharge pressure due to the
increase of the density and the pressure difference. On the other hand, because the pressures of the $P_1$ and the $P_d$ pocket are same after discharge starting angle, there is meaningless in $\dot{m}_d$.

Flow Coefficient

To obtain the flow coefficient, we used the result of choking experiment. The choking experiment is to measure the leakage flow rate between $P_1$ and the $P_d$ pockets under the condition of the fixed pressure of $P_d$ pocket (21kg/cm²G) and the reduced pressure of $P_1$ pocket (from 20kg/cm²G to 6kg/cm²G).

The result of the choking experiment is shown at Fig.5. The flow coefficients are able to be obtained by putting the experimental values into the Eq.(1), and the values of flow coefficient were 0.115 and 0.106 for sample A and sample B respectively. As we can see at Fig.5, the results of theoretical calculation using the average (0.110) of two flow coefficients coincide with experimental values within the limit of 5% error.

Then, the theoretical result from calculation putting the obtained flow coefficient (0.110), the leakage area, pressure and temperature into Eq.(1) is shown at Fig.6. This result has a good agreement with experimental value in Fig.4 within the limit of 10% error. So we could conclude that the flow coefficient of tangential leakage is always constant under the condition of constant oil concentration regardless to shape of leakage path, radial clearance, and pressure condition.

Effect of Tangential Leakage

The simulation using the flow coefficient were conducted to analyze the influence of the tangential leakage on the performance of a scroll compressor under the various operating conditions. The condition for simulation was the same as the standard compressor performance test condition of ARI (American Refrigeration Institute), and the parameters are leakage factors, such as radial clearance, discharge pressure, suction pressure, and suction gas temperature.

Fig.7 shows the leakage mass flow rate with the orbiting angle at each radial clearance (10μm, 20μm, and 30μm), and Fig.8 shows the volumetric efficiency with the radial clearance. According to the increase of the radial clearance, the leakage flow rate also increases linearly. This result makes the volumetric efficiency decrease with the increase of tangential clearance.

Fig.9 shows the leakage mass flow rate with the orbiting angle at the radial clearance of 20μm at each discharge pressure (1.866MPa, 2.161MPa, and 2.455MPa). Before discharge starting angle, there is no difference in the leakage flow rate. So the volumetric efficiency, which is defined as Eq.(3), has little difference. After discharge starting angle, however, the leakage flow rate has a great difference owing to the discharge pressure. The leakage flow rate increases with the increasing discharge pressure because of the increase of upstream density. The variation of discharge pressure has no effect on the volumetric efficiency, but the difference of the leakage flow rate could influence on the power consumption. Fig.10 shows the indicated work ratio with discharge pressure. When the discharge pressure increases, the leakage flow rate increases, so the power consumption increases also.

The variation of suction pressure causes the variation of the leakage flow rate. In Fig.11 the leakage flow rate with orbiting angle at each suction pressure (0.4936MPa, 0.5916MPa, 0.6897MPa). Before discharge starting angle, there is a large difference in the leakage flow rate. When the suction pressure is high, there is much leakage flow rate due to the high upstream density. So it could be thought that the volumetric efficiency would be low. When the theoretical mass of refrigerant on completion of suction ($m_i$ in Eq.(3)) increases at the high suction pressure, however, the volumetric efficiency is not nearly affected by the variation of suction pressure as shown in Fig.12. As an example, in the case that the suction pressure increases from 0.4936MPa to 0.5916MPa, the volumetric efficiency increases 0.01%.

Fig.13 shows the leakage mass flow rate with orbiting angle at the radial clearance of 20μm at each suction gas temperature (7.2°C, 18.3°C, 35°C). The quantity of leakage flow rate becomes higher over all the orbiting angle at a low suction gas temperature because the density becomes higher at the low temperature and same
pressure. Fig.14 represents the volumetric efficiency with suction gas temperature. In the case of the lower suction gas temperature, the volumetric efficiency becomes higher. Although the leakage flow rate increases according to the decreasing suction gas temperature, the rate of increase of the theoretical mass of refrigerant on completion of suction is higher than that of increase of the leakage flow rate as shown in Eq.(3), thereafter the volumetric efficiency increases.

**Conclusions**

In this present study, the indirect flow measuring method was developed in order to measure the leakage flow rate at the clearance of scroll compressor under the actual operating conditions, and the simulation was conducted using the flow coefficient obtained by experiment. Finally, following conclusions were obtained about the influence of the tangential leakage on the scroll compressor.

(1) From the result of choking experiment, the flow coefficient was obtained, and this value was always constant, 0.11, regardless to the shape of leakage path, the radial clearance, and the pressure condition at a given oil concentration.

(2) In a scroll compressor, the increasing radial clearance makes the linear increase of tangential leakage flow. This causes the volumetric efficiency decrease as a result.

(3) The load variation affects the tangential leakage, but has little effect on the volumetric efficiency.

(4) The variation of suction gas temperature has an effect on the leakage flow and performance of a scroll compressor. In the case of lower suction gas temperature, the leakage flow rate becomes higher, but volumetric efficiency becomes lower owing to the increase of theoretical mass of refrigerant on completion of suction.

**Acknowledgement**

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**References**

(1) Jean-Luc Caillat, Shimaeo Ni, Michael Daniels, 1988, "A computer model for scroll compressors," Proc. of Int. Compressor Engineering Conf. at Purdue, pp.47~55


(4) Tadashi Yanagisawa, Cheng Min Chi, Mitsuhiro Fukuta, Takashi Shimizu, 1990, "Optimum operating pressure ratio for scroll compressor" Proc. of Int. Compressor Engineering Conf. at Purdue, pp.425~433


### Tables and Figures

<table>
<thead>
<tr>
<th>Orbiting angle (°)</th>
<th>Pressure conditions ((P_p/P_d)) [kg/cm²]</th>
<th>Clearances (μm)</th>
</tr>
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<tr>
<td></td>
<td>(5/18)</td>
<td>(5/21)</td>
</tr>
<tr>
<td>0</td>
<td>(5/5/13/18)</td>
<td>(5/5/13/21)</td>
</tr>
<tr>
<td>45</td>
<td>(5/6/14/18)</td>
<td>(5/6/14/21)</td>
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<tr>
<td>180</td>
<td>(5/8/18/18)</td>
<td>(5/8/21/21)</td>
</tr>
<tr>
<td>270</td>
<td>(5/10/18/18)</td>
<td>(5/10/21/21)</td>
</tr>
</tbody>
</table>

Table 1: Experimental conditions for pocket pressure with orbiting angle

<table>
<thead>
<tr>
<th>Contact points</th>
<th>Clearances (μm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner contact point</td>
<td>105</td>
</tr>
<tr>
<td>Middle contact point</td>
<td>55</td>
</tr>
<tr>
<td>Outer contact point</td>
<td>35</td>
</tr>
</tbody>
</table>

Table 2: Measured radial clearances according to contact points at the reference angle (0°)

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**Fig. 1** Schematic of experimental apparatus

**Fig. 2** Exampled diagram of the indirect flow measuring method

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Fifteenth International Compressor Engineering Conference at Purdue University, West Lafayette, IN, USA – July 25-28, 2000
Fig. 3 Leakage mass flow rate with orbiting angle and radial clearance at (a) inner contact point, (b) middle contact point, and (c) outer contact point.

Fig. 4 Leakage mass flow rate with discharge pressure and orbiting angle at each contact point (sample A).

Fig. 5 Comparison of leakage mass flow rate between experimental and theoretical values.

Fig. 6 Leakage mass flow rate with orbiting angle at each discharge pressure by simulation (sample A).

Fig. 7 Leakage mass flow rate with orbiting angle at each radial clearance.

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Fig. 8 Volumetric efficiency with radial clearance

Fig. 12 Volumetric efficiency with suction pressure

Fig. 9 Leakage mass flow rate with orbiting angle at each discharge pressure

Fig. 13 Leakage mass flow rate with orbiting angle at each suction gas temperature

Fig. 10 Indicated work ratio with discharge pressure (radial clearance = 20 μm)

Fig. 14 Volumetric efficiency with suction gas temperature

Fig. 11 Leakage mass flow rate with orbiting angle at each suction pressure