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INFLUENCE OF OIL-REFRIGERANT SOLUBILITY ON THE PERFORMANCE OF ROTARY COMPRESSORS

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ABSTRACT

Utilization of non-CFC refrigerants urges compressor engineers to develop new refrigeration oils compatible with the CFC alternatives. One of the most important characteristics of the oil is miscibility with refrigerant and good miscibility is usually acceptable. This study tries to identify influence of oil-refrigerant solubility on the performance of refrigerant compressors. Flow rate and compression power of a rotary compressor were measured by using four kinds of oil having different solubility of refrigerant. As a result, volumetric efficiency and total efficiency of the compressor were higher in the case of the oil having low solubility than in the case of the oil having high solubility at low speed operation of the compressor. The tendency was supported by the theoretical analysis of leakage loss through clearance on roller faces.

INTRODUCTION

Refrigeration oil is a must in refrigerant compressors to keep high efficiency and to insure high reliability of the compressor. In the process of utilization of non-CFC refrigerants, many efforts have been done to develop new refrigeration oils compatible with the CFC alternatives because conventional mineral and alkylbenzene oils are less miscible with the CFC alternatives /1/. On the other hand, in recent years, Sumida /2/ reported a possibility to use a refrigeration oil having low solubility of non-CFC refrigerant but having good lubricating characteristic. Under these circumstances, it is needed to clear how the solubility of refrigerant in the oil influences performance of refrigerant compressors.

In general, viscosity of refrigeration oil becomes low by dissolution of refrigerant in the oil and the dissolved refrigerant dissociates (flashes) from the oil when the oil is exposed to low pressure in the compressor, which degrades lubricating characteristics of the oil and affects leakage loss in the compressor. In this study, performance of a rotary compressor is experimentally measured by using refrigeration oils having good or bad miscibility with refrigerant, and influence of oil-refrigerant solubility on the flow rate of the compressor is theoretically analyzed.

EXPERIMENT

Figure 1 shows a schematic view of an experimental rotary compressor (displacement: 13.5 cm³/rev, inner dia. of casing: 122.4 mm, inner dia. of cylinder: 54 mm, height of cylinder: 23.8 mm, outer dia. of roller: 46.8 mm, inner dia. of roller: 30.3 mm). The compressor is equipped with a large discharge casing including an oil separator in order to minimize quantity of refrigeration oil which circulates with refrigerant in a refrigeration cycle. An oil reservoir is connected to the compressor in order to control a level of the oil stored in a bottom casing of the compressor. A piezo-electric pres-
sure transducer is mounted on a cylinder to measure pressure change in a compression chamber of the cylinder and C-C thermocouples are inserted in the compressor to measure temperatures of oil in the casing and refrigerant in a suction port.

In experiments, the compressor was connected to an experimental refrigeration cycle and was operated under steady state conditions. Then flow rate of refrigerant and electric power input of the compressor were measured with a rotameter and an electric power meter respectively. R134a and R22 were used as working fluid of the compressor and four kinds of oil shown in table 1, having different miscibility with refrigerant, were used. Table 2 shows steady state operating condition of the compressor. Rotational speed of the compressor was changed by an inverter-controlled power source in a range of 30 - 70 1/s.

**THEORETICAL ANALYSIS**

When refrigeration oil in the compressor contains refrigerant in solution, performance of the compressor is influenced mainly by (1) drop in viscosity of the oil and (2) dissociation of vapor refrigerant from the oil. The both affect lubricating characteristic and leakage characteristic of the compressor. In this study, leakage flow through clearance on the roller face is taken as an example and its influence on volumetric efficiency of the compressor is analyzed theoretically.

In the rotary compressor, mass flow rate, \( q_m \), of oil which leaks from the inside of the roller to a suction chamber in the cylinder through face clearance of the roller is expressed as follows like a radial flow between two discs.

\[
q_m = \pi \left( \delta_1^3 + \delta_2^3 \right) \frac{(P_d - P_s)}{[12 \nu \log(r_o/r_i)]} \tag{1}
\]

where, \( \delta_1 \) and \( \delta_2 \): clearances on upper and lower faces of the roller, \( P_s \) and \( P_d \): suction and discharge pressures, \( \nu \): kinematic viscosity of the oil, \( r_i \) and \( r_o \): inner and outer radii of the roller. When the leaked oil is exposed to suction pressure in the suction chamber, some vapor refrigerant is dissociated from the oil according to the difference of refrigerant solubility in the oil. Then mass, \( q_r \), of the vapor refrigerant dissociated from the oil per unit time is given by

\[
q_r = X q_m \tag{2}
\]

where \( X = (a_i - a_o)/(1 - a_o) \), \( a_i \) and \( a_o \): solubilities of refrigerant inside and outside the roller. In the suction chamber, the leaked oil having high temperature heats suction refrigerant by its heat capacity. At that time, equation of heat balance is expressed as follows.

\[
c_1(1 - X)q_m(t_i - t_a) = c_g G_{ch}(t_a - t_s) + R q_r \tag{3}
\]

where, \( c_1 \) and \( c_g \): specific heats of oil and suction refrigerant, \( t_i \): temperature of oil inside the roller, \( t_s \): temperature of suction refrigerant, \( t_a \): temperature of refrigerant after the heat balance, \( G_{ch} \): ideal mass flow rate of suction refrigerant, \( R \): heat quantity required to dissociate unit mass of vapor refrigerant. The balanced temperature, \( t_a \), is easily obtained from the equation (3).

Volumetric efficiency, \( \eta_v \), taking account of the dissociation of vapor refrigerant and the heating effect of suction refrigerant is expressed as follows.

\[
\eta_v = 1 - \frac{q_r}{G_{ch}} - (1 - v_s/v_a) \tag{4}
\]
where, \( v_a \) and \( v_9 \): specific volumes of suction refrigerant corresponding to temperature \( t_a \) and \( t_9 \). In the equation (4), the second and third terms in the right hand express drop of volumetric efficiency due to the dissociation and the heating by the leaked oil respectively.

**RESULTS AND DISCUSSION**

Influence of Solubility on Volumetric Efficiency

Figure 2 shows experimental relationship between volumetric efficiency, \( \eta_v \), and rotational speed, \( N \), of the experimental compressor which was operated under the combination of refrigerant 134a and four kinds of oil. In experiments, solubility of refrigerant in each oil was measured by sampling the oil from the bottom of the compressor and the result is shown in Table 3. Oils 1 and 2 have relatively low solubility, but oils 3 and 4 have high solubility. In the figure 2, at the high rotational speed, volumetric efficiency has little difference between the four oils. But at the low rotational speed, there exists some difference of the volumetric efficiency and the efficiencies for the oils 1 and 2 having low solubility are larger than those for the oils 3 and 4 having high solubility. In the figure, temperature, \( t_a \), measured in the suction port is illustrated as one of factors which affect the efficiency. The temperature increases with decreasing the rotational speed and difference of the temperature between the four oils exceeds more than 10 °C. In addition, volumetric efficiency, \( \eta_{vh} \), which only counts for the change of specific volume due to the raised suction temperature, \( t_a \), is also shown in the figure 2, and \( \eta_{vh} \) for the oil 1 is about 3 % higher than that for the oil 4 at the low speed operation. Though the illustrated efficiency, \( \eta_{vh} \), doesn't account for the whole effect of the suction gas heating, it may represent a certain degree of the heating by the leaked oil.

Figure 3 shows theoretical results of volumetric efficiency, \( \eta_v \), calculated from the equation (4) by taking account of leakage loss through the clearance on the roller face. In the figure, \( \eta_{vv} \) is volumetric efficiency calculated by taking account of the dissociation only. Difference of the \( \eta_v \) and \( \eta_{vv} \) corresponds to the effect of the suction gas heating whose temperature is illustrated by \( t_a \) in the figure. With decreasing the rotational speed, \( N \), the volumetric efficiency, \( \eta_v \), decreases because leakage mass per one revolution of the compressor increases, and decreasing rates of the efficiency for the oils 3 and 4 are larger than those for the oils 1 and 2.

In the theoretical calculation, physical properties of the oils shown in Table 3 were used. Solubility of refrigerant at the discharge pressure is a measured value and the value was used as that for the oil inside the roller. On the other hand, solubility at the suction pressure and kinematic viscosity are estimated values under the assumption that these values for each oil change in the same manner as those for the conventional mineral oil 2. In addition, dissociation heat per unit mass of vapor refrigerant was given by difference of enthalpy between a saturated vapor refrigerant at suction pressure and a saturated liquid refrigerant at discharge pressure.

With respect to relation between the oil-refrigerant solubility and the volumetric efficiency, results of the figures 2 and 3 indicate that the efficiencies for the oils 3 and 4 having high solubility is lower than those for the oils 1 and 2 having low solubility. But that relation between the solubility and the efficiency cannot be easily concluded from the figures because kinematic viscosity of each oil is different as shown in Table 3 and leakage flow rate of the oil is not the same. Therefore, additional results for a reference oil 4' in which no refrigerant dissolves but whose vis-
cosity is equal to that of the oil 4 in the table 3 are shown in the figure 3. \( \eta_v \) for the oil 4' is a few percent higher than that for the oil 4, which indicates clearly that the refrigerant dissolved in the oil affects the volumetric efficiency badly.

In the above theoretical calculation, it was assumed that the dissociation of vapor refrigerant from the oil is ideally performed by the difference of the refrigerant solubility and that the heat balance between the leaked oil and the suction refrigerant is performed perfectly. But in the practical compressor, those are not the case, and effects of the refrigerant dissociation and the suction heating become smaller with increasing the rotational speed because a time elapsed one revolution of the compressor becomes shorter. In fact, experimental results in the figure 2 show a tendency that difference of the volumetric efficiency between the four oils decreases with increasing the rotational speed of the compressor.

Influence of Solubility on Power Efficiency and Total Adiabatic Efficiency

Figure 4 shows experimental results of power efficiency, \( \eta_p \), and total adiabatic compression efficiency, \( \eta_t \), when the compressor was operated with R134a. The efficiencies were defined by equations (5) and (6).

\[
\eta_p = \frac{L_{th}}{L_{in}} = \frac{G_{th} \Delta h}{L_{in}} = \eta_i \eta_m \eta_{mt}
\]

\[
\eta_t = G \frac{\Delta h}{L_{in}} = \frac{G}{G_{th}} \frac{G_{th} \Delta h}{L_{in}} = \eta_v \eta_p
\]

where, \( L_{in} \): electric power input, \( L_{th} \): ideal adiabatic compression power, \( \Delta h \): specific enthalpy change (specific work) with isentropic compression, \( G \): refrigerant mass flow rate, \( \eta_i \), \( \eta_m \) and \( \eta_{mt} \): indicated, mechanical and motor efficiencies of the compressor. In the figure 4, power efficiencies for the oils 3 and 4 are higher than those for the oils 1 and 2 in general. The difference is examined from the viewpoint of such components of the power efficiency as indicated, mechanical and motor efficiencies.

Concerning the indicated efficiency, pressure change in the cylinder was measured in the experiments. Generally speaking, the measured pressure curves for the four oils had little difference from each other though the pressure rise in the compression process was slightly earlier for the oil having higher solubility at the low speed operation. This means that difference of the oil has little influence on the indicated work, namely, the indicated efficiency. On the other hand, motor efficiency of the compressor is almost the same not depending on the oil at each rotational speed. Therefore, difference of the power efficiency due to the oil difference at each rotational speed in the figure 4 is understood by difference of the mechanical efficiency. In general, mechanical loss in the compressor increases with increasing the oil viscosity. This supports such a tendency shown in the figure 4 that the power efficiency for the oil 1 having high viscosity is lower than that for the oil 4 having low viscosity.

On the other hand, the total adiabatic efficiency, \( \eta_t \), in the figure 4 has the same tendency as the volumetric efficiency, \( \eta_v \), in the figure 2 though the difference of the total efficiency between the oils is less than that of the volumetric efficiency at each rotational speed. In the figure 4, total efficiencies for the oils 1 and 2 having low refrigerant solubility is about 2% higher than those for the oils 3 and 4 having high solubility at the low speed operation.

Efficiency in Case of R22

Figures 5 and 6 show experimental volumetric efficiency, \( \eta_v \), power efficiency,
\[ \eta_p, \text{ and total adiabatic efficiency, } \eta_t, \text{ when the compressor was operated with R22. In general, changing tendency of these efficiencies are almost the same as those of the figures 3 and 4 for R134a. The volumetric efficiency, } \eta_v, \text{ for the oil 1 having relatively low solubility (13 \% of R22) is higher than that for the oil 3 having high solubility (26 \%) at the low speed operation, and the total efficiency, } \eta_t, \text{ for the former oil is slightly higher than that for the latter oil.} \]

CONCLUSIONS

Influence of oil-refrigerant solubility on the performance of the rotary compressor was investigated theoretically and experimentally. Experiments using four kinds of oil having different solubility of refrigerant indicated that the volumetric efficiency and the total adiabatic efficiency for the oil having low solubility are higher than those for the oil having high solubility at the low speed operation. From the viewpoint of refrigerant flow rate and power consumption of the compressor, it is favorable to use the oil having low solubility of refrigerant, but the further study is needed from the viewpoint of lubricity and reliability of the compressor.

REFERENCES

(3) Yanagisawa, T. and Shimizu, T., "Leakage Losses with a Rolling Piston Type Rotary Compressor (II. Leakage Losses through Clearances on Rolling Piston Faces)", Int. J. Refrig., 8-3 (1985), 152.

Table 1 Properties of pure oil

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Oil 1</th>
<th>Oil 2</th>
<th>Oil 3</th>
<th>Oil 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base material</td>
<td>Mineral</td>
<td>Mineral</td>
<td>PAG</td>
<td>Ester</td>
</tr>
<tr>
<td>Kinematic viscosity (mm²/s) at 40 °C</td>
<td>34.9</td>
<td>30.1</td>
<td>34.2</td>
<td>30.7</td>
</tr>
<tr>
<td>at 100 °C</td>
<td>5.7</td>
<td>4.3</td>
<td>7.8</td>
<td>5.2</td>
</tr>
</tbody>
</table>

Table 2 Operating condition of compressor

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R134a</th>
<th>R22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction pressure (MPa[gage])</td>
<td>0.20</td>
<td>0.49</td>
</tr>
<tr>
<td>Discharge pressure (MPa[gage])</td>
<td>1.37</td>
<td>1.96</td>
</tr>
<tr>
<td>Suction temperature (°C)</td>
<td>10</td>
<td>15</td>
</tr>
<tr>
<td>Oil temperature (°C)</td>
<td>80</td>
<td>90</td>
</tr>
</tbody>
</table>

Figure 1 Schematic view of experimental rotary compressor
Table 3 Solubility of R134a and kinematic viscosity

<table>
<thead>
<tr>
<th>Specimen</th>
<th>Oil 1</th>
<th>Oil 2</th>
<th>Oil 3</th>
<th>Oil 4 (Note)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solubility (wt %)</td>
<td>5.1</td>
<td>6.5</td>
<td>17.7</td>
<td>17.9</td>
</tr>
<tr>
<td>at discharge pressure</td>
<td>0.5</td>
<td>0.6</td>
<td>2.3</td>
<td>2.3</td>
</tr>
<tr>
<td>at suction pressure</td>
<td>6.5</td>
<td>5.1</td>
<td>3.8</td>
<td>2.9</td>
</tr>
</tbody>
</table>

Figure 2 Experimental volumetric efficiency (R134a)

Figure 3 Theoretical volumetric efficiency (R134a)

Figure 4 Experimental power efficiency and total efficiency (R134a)

Figure 5 Experimental volumetric efficiency (R22)

Figure 6 Experimental power efficiency and total efficiency (R22)