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Leakage Tests of Wet CO$_2$ Gas with Oil-Mixture in Scroll Compressors and Its Use in Simulations of Optimal Performance

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ABSTRACT

This study presents empirical values of the friction factor for leakage flows of the dry and wet CO$_2$ gas with oil-mixture through small axial clearance between the orbiting and fixed scrolls of scroll compressors. An oil mixture device with venture pipe was manufactured to permit experiments on leakage flows of CO$_2$ gas containing oil mist. The wet refrigerant gas with an oil mixture rate flowed from a pressurized vessel to the atmosphere through the small axial clearance with a thin rectangular cross-section. The pressure decay in the pressurized vessel due to leakage was measured at a variety of initial pressure up to 3 MPa. The Darcy-Weisbach equation for incompressible viscous fluid flow through the thin rectangular cross-section was applied to calculate the leakage mass flow rate, thus simulating the pressure decay characteristics, where the empirical friction factor was determined and plotted on a Moody diagram. As a result, it was shown that the wet CO$_2$ gas leakage flows also can be represented by the friction factor which takes on a larger value than for similar flows using dry CO$_2$ gas. Subsequently, the empirical friction factor was incorporated into computer simulations for both wet and dry CO$_2$ gas to calculate the scroll compressor efficiencies, thus addressing the oil mixture effect on the performance.

1. INTRODUCTION

In recent years, scroll compressors, because of their low vibration, low noise and high efficiency, have become an increasingly popular choice for compressors not only in the conventional refrigerant air-conditioning systems but also in the air-conditioning systems using CO$_2$ as the refrigerant. The scroll compressors have small axial and radial clearances between the orbiting and fixed scrolls. The leakage of the compressed refrigerant gas through these small

![Image](https://via.placeholder.com/150)

Figure 1. Refrigerant leakage flows in scroll compressors: (a) cross-sectional view; (b) leakage flow through radial clearance; (c) leakage flow through axial clearance.
clearances has a strong detrimental effect on the volumetric efficiency. In order to carry out accurate performance simulations of scroll compressors for efficiency optimization, it is necessary to establish a reliable method of calculating the leakage flow through these clearances. However, it is noted here that the theoretical method for the refrigerant gas leakages should be kept as simple as possible, since the computer simulations of the resultant performance include many complicated procedures to calculate the mechanical, compression and volumetric efficiencies. Previous studies by Ishii et al (1996a, 2005) present a very simple method for calculating the refrigerant gas leakage flows through the axial and radial clearances in a scroll compressor based on the Darcy-Weisbach equation for incompressible, viscous fluid flow through the thin rectangular cross-section, thus representing the gas leakage flows with the friction factor. First, very simple experiments for the leakage flow through the axial and radial clearances were conducted for the dry refrigerants of R22, R410A and CO2. Secondly, the leakage flow through the small clearances in scroll compressors was calculated by the Darcy-Weisbach equation to determine the empirical friction factors.

In the actual scroll compressors, however, the refrigerant gas is mixed with the oil for its sealing function. Some researches show that the optimal oil mixture rate is about 5%, thus resulting in the optimal performance of air conditioning system. In the present study, as the first trial, an oil mixture device with venture pipe was manufactured to permit very simple experiments on leakage flows of the wet CO2 gas containing oil mist. The wet gas with an oil mixture rate, initially pressurized in a large vessel, flowed out to the atmosphere through the axial clearance with a thin rectangular cross-section, where the pressure decay due to gas leakage was measured at a variety of initial pressure up to 3MPa. The Darcy-Weisbach equation for incompressible, viscous fluid flow was applied to the leakage flow through the axial clearance, thus calculating the leakage mass flow rate. Subsequently, the measured pressure decay was carefully simulated by assuming a polytropic process, where the empirical friction factor was determined and plotted on a Moody diagram. The empirical friction factor was compared with that for the dry CO2 gas, thus addressing the oil sealing effect. Finally, the determined empirical friction factor for the wet CO2 gas was incorporated into computer simulations to calculate the volumetric, mechanical and compression and resultant overall efficiencies, where the calculated efficiencies were compared with those for the dry CO2 gas to address the oil mixture effect on the scroll compressor performance.

2. EXPERIMENTAL SET-UPS

In the scroll compressors, the refrigerant gas leaks through the axial clearance $\delta_a$ and the radial clearance $\delta_r$ due to the pressure difference between compression chambers, as shown in Fig. 1. In order to address the leakage characteristics, an axial clearance model sandwiched with thrust plates was made, as shown in Fig. 2, where (a) shows the plane view and (b) the A-A cross-sectional view. The leakage path has the streamwise length of 4.0mm and the depth of 15.0mm. The leakage clearance height $\delta_a$ was carefully adjusted to 10µm by using thickness gauges with a width of 2.5mm, as shown in Fig. 2(b). Therewith, the leakage net width is 10 mm.

![Figure 2. Axial Clearance model for gas leakage tests in scroll compressors: (a) plane view; (b) cross-sectional view.](image-url)
The layout of leakage experiments is shown in Fig. 3, where an oil mixture device consisted of the oil container and the venture pipe is presented on the left-hand side between the CO₂ gas tank and the reservoir tank. The oil container was made with the transparent acrylic resin, designed to maintain the adequate strength against high pressure CO₂ gas, so that the mixture amount of oil can be visually checked. The oil is inhaled into the venturi pipe due to its low pressure, thus being mixed into the dry CO₂ gas flow. In order to control the oil mixture speed at its best condition, the diameter of the 7mm-long path from the oil container to the venture pipe were adjusted at 1mm. A small electric fan is installed in the reservoir tank to agitate the oil mist. The refrigerant oil of VG5 was specially compounded for the present leakage tests, at a lower viscosity on the normal operation of actual scroll compressors.

First of all, the oil was supplied into its container with Valve 1 closed, and the high and low pressure chambers and the reservoir tank were made at vacuum with Valves 2 and 3 closed. Next, Valve 4 closed and Valve 1 opened, Valve 2 was opened to make the wet CO₂ gas with oil mist into the reservoir tank. At the specified pressure in the reservoir tank, Valve 2 closed and the low pressure chamber was suddenly vented to the atmosphere with Valve 3 opened. The reservoir tank as high pressure chamber decreases in pressure due to leakage through the test clearance, and its time-dependent pressure decay was measured by a semiconductor pressure transducer. Some amount of oil mist adheres to the inside of reservoir tank. Therewith, the mass of oil mist contained in the breathed-out gas was measured to calculate the oil-to-CO₂ mixture ratio.

### 3. LEAKAGE TEST RESULTS

Measured data of pressure decay in the high pressure chamber, due to leakage through a 10μm axial clearance, are

![Figure 4. Pressure decay in the high pressure chamber for leakage flow through 10 μm axial clearances.](image-url)
shown in Fig. 4, where the initial pressure was adjusted at 6 different values ranging from 0.6 to 3.0MPa, chosen as being representative of the pressure difference between adjacent compression chambers in scroll compressors. The solid lines are for the wet CO$_2$ gas and the dashed lines are for the dry CO$_2$ gas. After the sudden connection of the low pressure chamber to the atmosphere at time $t=0$, the pressure in the high pressure chamber decays due to gas leakage, approaching atmospheric pressure (about 0.1MPa). The pressure decay is significantly slower in the wet gas (solid lines) than in the dry gas (dashed lines), thus representing the oil sealing effect against the leakage flows through small axial clearance. In the present leakage tests, the oil mixture mass ratio was less than about 2.8%.

4. CALCULATIONS OF PRESSURE DECAY AND EMPIRICAL FRICTION FACTOR

The following analysis presents a very simple method for calculating the refrigerant gas leakage flows through the axial and radial clearances in a scroll compressor, based on the Darcy-Weisbach equation for incompressible, viscous fluid flow through the thin rectangular cross-section (see Ishii 1996a, 2005).

When the axial clearance leakage flow through the thin rectangular cross-section is assumed to be an incompressible viscous flow, the following relation can be derived from the conservation of momentum principle:

$$\frac{P - P_a}{\rho g} = \lambda_a \frac{L}{4 m} \cdot \frac{u_m^2}{2 g}$$

(1)

where the frictional force acting on the wall outside the leakage passage was neglected, since it is far smaller than the frictional forces on the leakage passage area. This expression is known as the Darcy-Weisbach equation when written for a pipe flow, and indicates that the pressure drop ($P - P_a$) through the leakage channel with a length of $L$ is basically determined by the friction factor $\lambda_a$ of the channel surface. The average leakage flow velocity is represented by $u_m$. Since the axial clearance height, $\delta_a = 10 \mu m$, is very small compared with the channel depth $W=10$ mm, the hydraulic mean depth, $m$, is given by $\delta_a / 2$. The hydraulic diameter $d$ of a circular pipe with a pressure drop equivalent to that of the rectangular cross-section channel is given by $4m$ ($d \equiv 4m$).

Given values of the friction factor $\lambda_a$ and the pressure difference ($P - P_a$), the mean leakage flow velocity $u_m$ can be calculated from Eq. (1), and the mass flow rate $\dot{M}$ can be calculated by

$$\dot{M} = \rho \delta_a W u_m$$

(2)

This flow produces the following pressure decay $\Delta P$ in the high pressure chamber over a small time $\Delta t$:

$$\Delta P = \frac{P_0}{G_0^n} \cdot n \cdot G^{n-1} \cdot \dot{M} \cdot \Delta t$$

(3)

where the pressure decay is assumed to be a polytropic process with exponent $n$. $P_0$ represents the initial pressure. $G$ represents the residual refrigerant mass in the high pressure chamber, which can be calculated by subtracting the total leakage mass from the initial refrigerant mass $G_0$:

$$G = G_0 - \int_0^{\Delta t} \dot{M} dt$$

(4)

Using Eqs. (4) and (5), the pressure $P$ in the high pressure chamber can be calculated successively.

The friction factor $\lambda_a$ can be given by the formula of Nikuradse:

$$\lambda = 0.0032 + \alpha \cdot Re^{-0.5}.$$  

(5)

which corresponds to fully turbulent flow. The Reynolds number $Re$ is defined by
\[ \text{Re} \equiv \frac{4 \mu u_m}{\delta \rho} = \frac{2 \delta_s u_m}{\mu / \rho} \]  
(6)

where the equivalent diameter \( d (= 4m) \) is chosen as the representative length. The viscosity coefficient \( \mu \) is given in Table 1, as a function of pressure at 18°C. Using a polytropic exponent \( n \) of 1.30 for CO\(_2\), the density \( \rho \) can be calculated as

\[ \rho = \rho_0 \left( \frac{P}{P_0} \right)^{\frac{1}{n}} \]  
(7)

When the coefficient \( \alpha \) and exponent \( \beta \) in Eq. (5) for the friction factor \( \lambda \) are assigned the values given in the second plot in Figs. 5 and 6, the pressure decay simulated for CO\(_2\) gas shows close agreement with the measured decay, as shown by the dotted lines in the first diagram in Figs. 5 and 6, respectively. Using the mean values of the plotted data for \( \alpha \) and \( \beta \), the friction factor \( \lambda \) for dry and wet CO\(_2\) gas leakage flows through the axial clearance can be given by

\[ \lambda_a = 0.0032 + 0.35 \cdot \text{Re}^{-1.52} \text{ for dry CO}_2 \text{ gas leakage flow,} \]
\[ \lambda_a = 0.0032 + 0.30 \cdot \text{Re}^{-1.22} \text{ for wet CO}_2 \text{ gas leakage flow.} \]  
(8)

As shown in the third diagram of Figs. 5 and 6, the leakage flow velocity \( u_m \) is significantly smaller than the sonic speed for CO\(_2\) (about 250 m/s), yielding a Mach number less than 0.3 and justifying the treatment of the flow as incompressible. The tested maximum Reynolds numbers are about 5800 for dry CO\(_2\) and about 4900 for wet CO\(_2\), as shown in the last diagram of Figs. 5, 6.

![Figure 5](image1.png)

**Figure 5.** Simulated pressure decay, fitted values of friction factor, flow velocity and Reynolds number of dry CO\(_2\) gas leakage flow through axial clearance \( \delta_a=10\mu m \).

![Figure 6](image2.png)

**Figure 6.** Simulated pressure decay, fitted values of friction factor, flow velocity and Reynolds number of wet CO\(_2\) gas leakage flow through axial clearance \( \delta_a=10\mu m \).
The empirical friction factors for a leakage clearance height of 10 µm are denoted by the solid lines on the Moody diagram in Fig. 7. It is clearly shown here that the friction coefficient for wet CO₂ gas leakage is significantly larger than for dry CO₂ gas leakage. This result significantly suggests that the oil-mist sealing effect of the gas leakage through small clearance can be represented with the friction factor.

5. CALCULATIONS OF COMPRESSOR EFFICIENCIES

With the empirical friction factors, the oil-mist sealing effects on the compressor efficiencies can be quantitatively addressed by undertaking computer simulations developed to optimize scroll compressor performance by Ishii et al. (2002a,b, 1996b). The friction factors used in the simulations were all assumed to follow the relationship (8), given

\[ \lambda = \frac{0.032 + 0.35R e^{-1.52}}{0.032 + 0.30R e^{-1.22}} \]

Figure 7. Empirical friction factors on Moody diagram, for wet and dry CO₂ gas leakage flows through axial clearance with the height of δₐ = 10µm.

The empirical friction factors for a leakage clearance height of 10µm are denoted by the solid lines on the Moody diagram in Fig. 7. It is clearly shown here that the friction coefficient for wet CO₂ gas leakage is significantly larger than for dry CO₂ gas leakage. This result significantly suggests that the oil-mist sealing effect of the gas leakage through small clearance can be represented with the friction factor.

Table 2. Major specifications and mechanical constants of scroll compressor.

<table>
<thead>
<tr>
<th></th>
<th>Dry CO₂</th>
<th>Wet CO₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>Suction volume</td>
<td>Vₛ [cm³]</td>
<td>4.25</td>
</tr>
<tr>
<td>Cooling capacity</td>
<td>[kJ/h]</td>
<td>727</td>
</tr>
<tr>
<td>Operating speed</td>
<td>[rpm]</td>
<td>3498</td>
</tr>
<tr>
<td>Involute base circle</td>
<td>rₕ [mm]</td>
<td>1.4 ~ 2.8</td>
</tr>
<tr>
<td>Scroll height</td>
<td>B [mm]</td>
<td>9.7 ~ 3.6</td>
</tr>
<tr>
<td>Scroll thickness</td>
<td>t [mm]</td>
<td>3.0</td>
</tr>
<tr>
<td>Cylinder disameter</td>
<td>D [mm]</td>
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<tr>
<td>Volume ratio</td>
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<td>Pressure ratio</td>
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<td>Specific heat ratio</td>
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<tr>
<td>Suction temperature</td>
<td>Tₛ [°C]</td>
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<tr>
<td>Suction pressure</td>
<td>Pₛ [MPa]</td>
<td>3.5</td>
</tr>
<tr>
<td>Discharge pressure</td>
<td>Pₐ [MPa]</td>
<td>9.00</td>
</tr>
<tr>
<td>Axial clearance</td>
<td>δₐ [µm]</td>
<td>3.0</td>
</tr>
<tr>
<td>Radial clearance</td>
<td>δₐ [µm]</td>
<td>6.0</td>
</tr>
<tr>
<td>Empirical fric. factor</td>
<td>axial</td>
<td>0.0032 + 0.35Re^{-1.52}</td>
</tr>
<tr>
<td></td>
<td>radial</td>
<td>0.0032 + 0.30Re^{-1.22}</td>
</tr>
<tr>
<td>Moment of Crankshaft</td>
<td>lₐ[kg·m²]</td>
<td>0.107 ~ 0.114</td>
</tr>
<tr>
<td>Orbiting scroll mass</td>
<td>mₐ[kg]</td>
<td>0.116 ~ 0.112</td>
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<tr>
<td>Oldham ring mass</td>
<td>mₐ[kg]</td>
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<tr>
<td>Crankshaft radius</td>
<td>rₐ[mm]</td>
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</tr>
<tr>
<td>Crankpin radius</td>
<td>rₐ[mm]</td>
<td>8.0</td>
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<tr>
<td>Fric. coef. at oldham ring</td>
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<td>0.055</td>
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<tr>
<td>Fric. coef. at thrust bearing</td>
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<td>0.011</td>
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<td>Fric. coef. at crankpin</td>
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<td>0.011</td>
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<tr>
<td>Fric. coef. at crank journal</td>
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<td>0.0033</td>
</tr>
<tr>
<td>Fric. coef. at ball bearing</td>
<td></td>
<td></td>
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</table>

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in Table 2 also, both for the axial and radial clearances. Also shown in Table 2 are the major specifications for the CO₂ compressors. The suction volume and compression ratio were 4.25 cm³ and 2.07, resulting in the cooling capacity of 727 kJ/h at the rated values of mean crankshaft speed, suction temperature, suction pressure and discharge pressure given in Table 2. The suction and discharge pressures were 3.5 and 9.0 MPa. The scroll thickness and cylinder diameter, based on the current scroll compressor designs, were kept at 3.0 mm and 67.54 mm, respectively. The clearance between the orbiting and fixed scrolls was kept at 3.0 µm for the axial direction and 6.0 µm in the radial direction. The crankshaft moment of inertia, I₀, was adjusted depending upon the necessary driving shaft power, and the orbiting scroll mass m₀ depended on the scroll height B. The coefficients of Coulomb friction at each pair of moving compressor elements were kept at from 0.055 to 0.0013, as measured by friction tests. Calculations were made for the involute base circle radius r₀ from 1.4 mm to 2.8 mm, which results in a scroll height B ranging from 9.7mm to 3.6mm. First, the leakage flow velocity and leakage mass flow rate were calculated for both the axial and radial clearances using the empirical friction factor. Then the compressed gas pressure was calculated. From the pressure, the net leakage mass \( \Delta G \) during one revolution of the crankshaft was calculated, as shown in Fig. 8(a), where the dashed line is for the dry CO₂ gas and the solid line is for the wet CO₂ gas. The abscissa is the involute base circle radius r₀. Since the friction factor for the wet CO₂ gas is larger than that for the dry, the net leakage mass \( \Delta G \) becomes smaller for the wet CO₂ gas than for the dry. The improvement for the wet CO₂ gas is about 10 % compared with for the dry CO₂ gas. As a result, the volumetric efficiency \( \eta_v \) is improved by about 1.1% for the wet CO₂ gas, as shown in Fig. 8(b). The compressed gas pressure can not be meaningfully affected by the gas leakage, thus resulting in the high compression efficiency \( \eta_c \) exceeding 90% for both the dry and wet CO₂ gas, as shown in Fig.8(c). The mechanical efficiency \( \eta_m \) also exhibits its highest level of 90% for both the dry and wet CO₂ gas, as shown in Fig.8(d). As a result, the overall efficiency \( \eta \), clearly dominated by the volumetric efficiency, is improved by 0.9% at its optimal performance, as shown in Fig.8(e). This improvement is of course due to the oil-mist sealing effect. The involute base circle radius r₀ affording the optimal compressor efficiency takes the same value of 2mm for both the dry and wet CO₂ gas.

6. CONCLUSION

The oil-mixture device was manufactured in order to address the oil-mist sealing effect. Using this device, leakage flow experiments were conducted for both the dry and wet CO₂ gas, flowing through the axial clearances with a height of 10µm between the orbiting and fixed scrolls of scroll compressors, where the pressure drop due to leakage from 3 MPa at maximum was measured over a range of Reynolds number up to \( 1.2 \times 10^4 \). The Darcy-Weisbach equation for incompressible, viscous fluid flow through the thin rectangular channels was used to calculate the pressure drop of the leakage flow. Empirical values for the friction factor were determined by comparing the calculated pressure drop with the measured values, thus showing that the oil-mist sealing effect of the gas leakage
through small clearance can be represented by the friction factor $\lambda$. The empirically determined friction factors were then plotted on a Moody diagram, being found to be significantly dependent on the oil to CO$_2$-gas ratio in mass.

Subsequently, the empirical friction factors were incorporated into computer simulations for CO$_2$ scroll compressors using both the dry and wet CO$_2$ gas with the same leakage clearance height of 3$\mu$m for the axial and 6$\mu$m for the radial clearances. The objective was to address the efficiency improvement due to the oil sealing effect and to determine the optimal involute base circle radius $r_b$ in compressor efficiency. The optimal overall efficiency was 76.4% at $r_b=2.0$ mm for the wet CO$_2$ gas, which is 0.9% higher than for the dry CO$_2$ gas.

In the present leakage tests, it was quite hard to increase the oil to CO$_2$ gas ratio to its sufficiently large value of about 5% yielding the optimal compressor efficiency. In the near future, more excellent oil mixture device will be developed to conduct leakage experiments for more wide range of oil-gas ratio. The relation between the oil-gas ratio and the friction factor $\lambda$ will be determined in more detail.

**NOMENCLATURE**

- $d$: Equivalent diameter, m
- $G, G_0$: Mass of refrigerant gas, kg
- $H$: Channel height, m
- $L$: Leakage channel length, m
- $m$: Hydraulic mean depth, m
- $M$: Leakage mass flow rate, kg·s$^{-1}$
- $n$: Polytropic exponent
- $P, P_0, P_1$: Absolute pressure of refrigerant gas, Pa
- $P_a$: Atmospheric pressure, Pa
- $u_m, u_0$: Average flow velocity, m·s$^{-1}$
- $W$: Leakage channel depth, m
- $\alpha$: Coefficient of friction factor
- $\beta$: Index of friction factor
- $\delta, \delta_a$: Leakage clearance, $\mu$m
- $\lambda, \lambda_a$: Friction factor
- $\mu$: Coefficient of viscosity, $\mu$Pa·s
- $\rho, \rho_0$: Density of refrigerant gas, kg/m$^3$

**REFERENCES**


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