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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

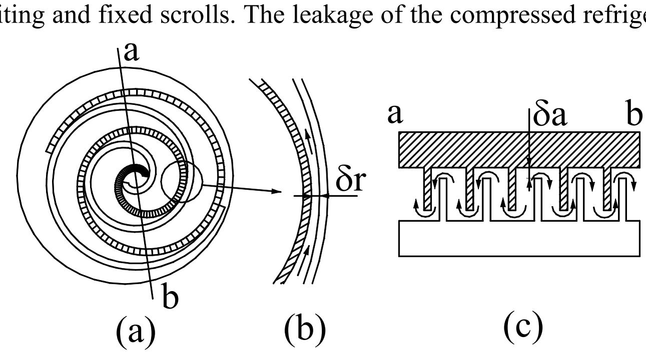

Figure 1. Refrigerant leakage flows in scroll compressors: (a) cross-sectional view; (b) leakage flow through radial clearance; (c) leakage flow through axial clearance.

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 CO_2. 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 CO_2 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 CO_2 gas, thus addressing the oil sealing effect. Finally, the determined empirical friction factor for the wet CO_2 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 CO_2 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 δ_a and the radial clearance δ_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 δ_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.

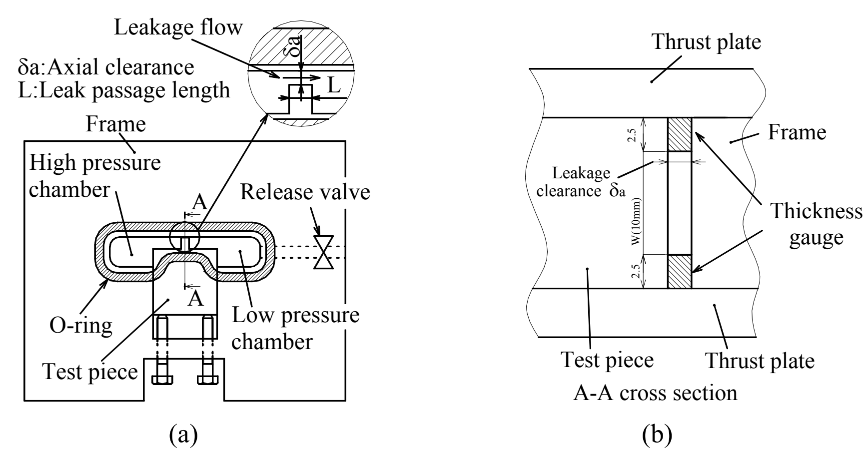

Figure 2. Axial Clearance model for gas leakage tests in scroll compressors:
(a) plane view; (b) cross-sectional view.

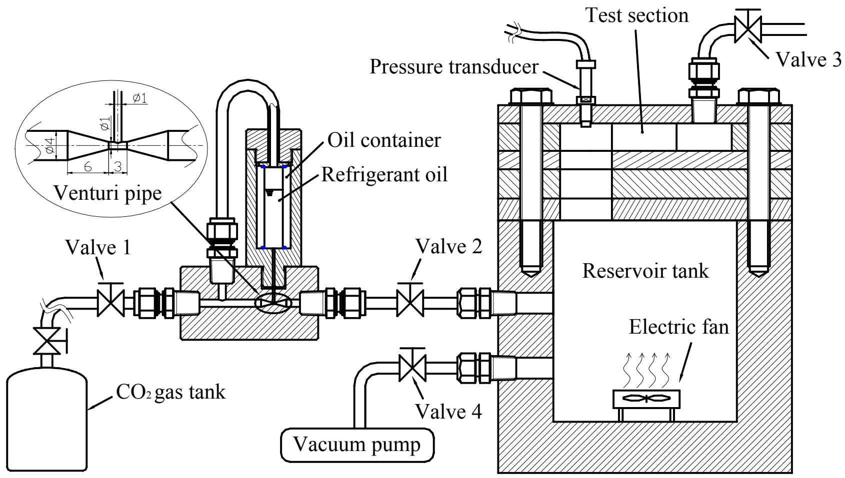

Figure 3. Layout of leakage experiments with oil-mixture device.

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 CO_2 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 CO_2 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 CO_2 gas flow. In order to control the oil mixture speed at its best condition, the diameter of the ?mm-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 CO_2 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-CO_2 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

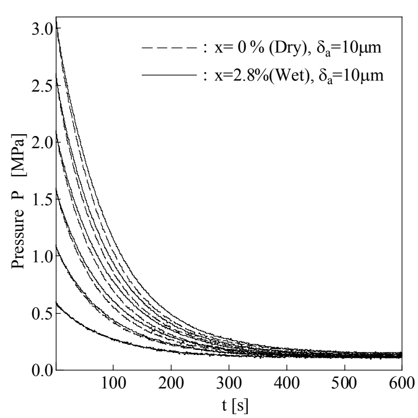

Figure 4. Pressure decay in the high pressure chamber for leakage flow through 10 μm axial clearances.

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:

$$\frac{P - P_a}{\rho g} = \lambda_a \frac{L}{4m} \cdot \frac{u_m^2}{2g} \tag{1}$$

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 λ_a of the channel surface. The average leakage flow velocity is represented by u_m. Since the axial clearance height, δ_a = 10μm, is very small compared with the channel depth W=10 mm, the hydraulic mean depth, m, is given by δ_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 ($\equiv d$).

Given values of the friction factor λ_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

$$\dot{M} = \rho \delta_a W u_m \tag{2}$$

This flow produces the following pressure decay ΔP in the high pressure chamber over a small time Δt:

$$\Delta P = \frac{P_0}{G_0^{\,n}} \cdot n \cdot G^{n-1} \cdot \dot{M} \cdot \Delta t \tag{3}$$

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:

$$G = G_0 - \int_0^t \dot{M} dt \tag{4}$$

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

The friction factor λ_a can be given by the formula of Nikuradse:

$$\lambda = 0.0032 + \alpha \cdot Re^{-\beta}. \tag{5}$$

which corresponds to fully turbulent flow. The Reynolds number Re is defined by

$$\mathrm{Re} \equiv \frac{4mu_m}{\mu/\rho} = \frac{2\delta_a u_m}{\mu/\rho} \tag{6}$$

where the equivalent diameter d (= 4m) is chosen as the representative length. The viscosity coefficient μ 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 ρ can be calculated as

$$\rho = \rho_0 \left(P / P_0 \right)^{\frac{1}{n}} \tag{7}$$

When the coefficient α and exponent β in Eq. (5) for the friction factor λ_a 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 α and β, the friction factor λ_a for dry and wet CO_2 gas leakage flows through the axial clearance can be given by

Table 1. Coefficient of viscosity of pressurized CO_2 gas at 18°C.

Pressure [MPa]	Coefficient of viscosity [μPa·s]
0.1	14.49
0.5	14.73
1.0	14.97
1.5	15.20
2.0	15.49
2.5	16.20
3.0	16.92

$$\lambda_a = 0.0032 + 0.35 \cdot \mathrm{Re}^{-1.52} \text{ for dry } CO_2 \text{ gas leakage flow,}$$
$$\lambda_a = 0.0032 + 0.30 \cdot \mathrm{Re}^{-1.22} \text{ for wet } CO_2 \text{ gas leakage flow.} \tag{8}$$

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 250m/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.

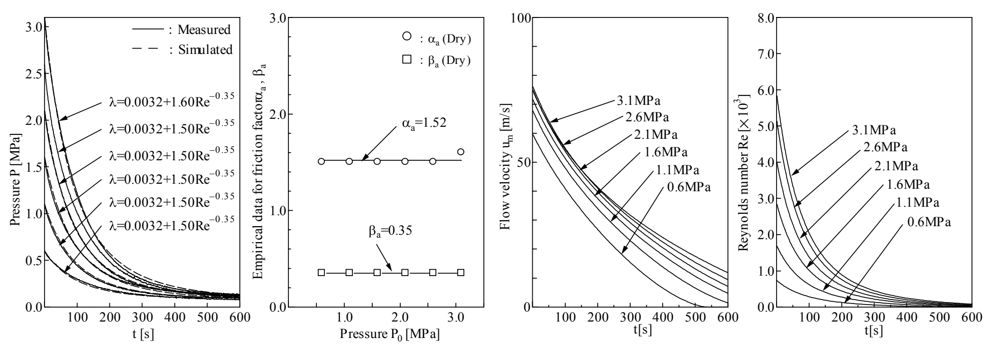

Figure 5. Simulated pressure decay, fitted values of friction factor, flow velocity and Reynolds number of dry CO_2 gas leakage flow through axial clearance δ_a=10μm.

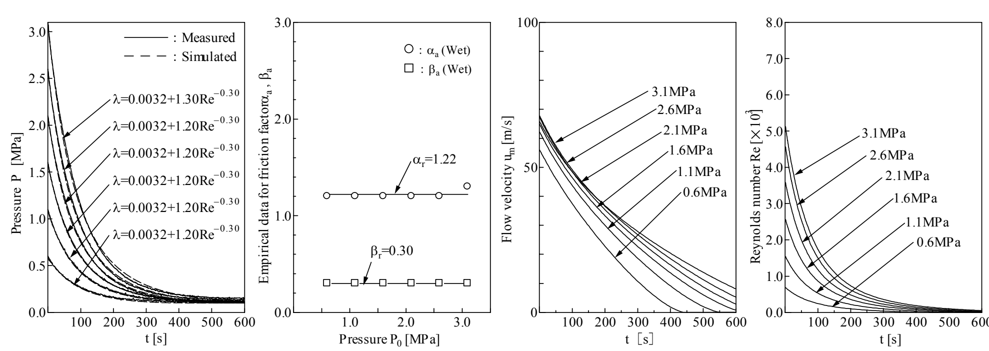

Figure 6. Simulated pressure decay, fitted values of friction factor, flow velocity and Reynolds number of wet CO_2 gas leakage flow through axial clearance δ_a=10μm.

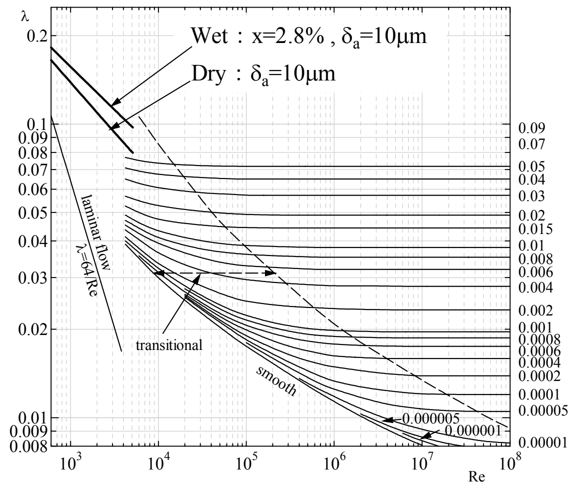

Figure 7. Empirical friction factors on Moody diagram, for wet and dry CO2 gas leakage flows through axial clearance with the height of δ_a=10μm.

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 CO_2 gas leakage is significantly larger than for dry CO_2 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

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

		Dry CO_2	Wet CO_2
Suction volume	Vs [cm^3]	4.25	
Cooling capacity	[kJ/h]	727	
Operationg speed	[rpm]	3498	
Involute base circle	r_b [mm]	1.4 ~ 2.8	
Scroll height	B [mm]	9.7 ~ 3.6	
Scroll thickness	t [mm]	3.0	
Cylinder disameter	D [mm]	67.54	
Volume ratio		2.07	
Pressure ratio		2.57	
Specific heat ratio	κ	1.30	
Suction temperature	Ts [°C]	10.5	
Suction pressure	P_s [MPa]	3.5	
Discharge pressure	P_d [MPa]	9.00	
Axial clearance	δ_a [μm]	3.0	3.0, 0
Radial clearance	δ_r [μm]	6.0	
Empirical fric. factor, axial	λ_a	$0.0032+0.35Re^{-1.52}$	$0.0032+0.30Re^{-1.22}$
Empirical fric. factor, radial	λ_r		
Moment of Crankshaft	I_0[kg·m^2]	0.107~0.114	
Orbiting scroll mass	m_s[kg]	0.116~0.112	
Oldham ring mass	m_0[kg]	0.037	
Crankshaft radius	r_O[mm]	8.0	
Crankpin radius	r_S[mm]	8.0	
Fric. coef . at oldham ring		0.055	
Fric. coef . at thrust bearing			
Fric. coef. at crankpin		0.011	
Fric. coef. at crank journal			
Fric. coef. at ball bearing		0.0013	

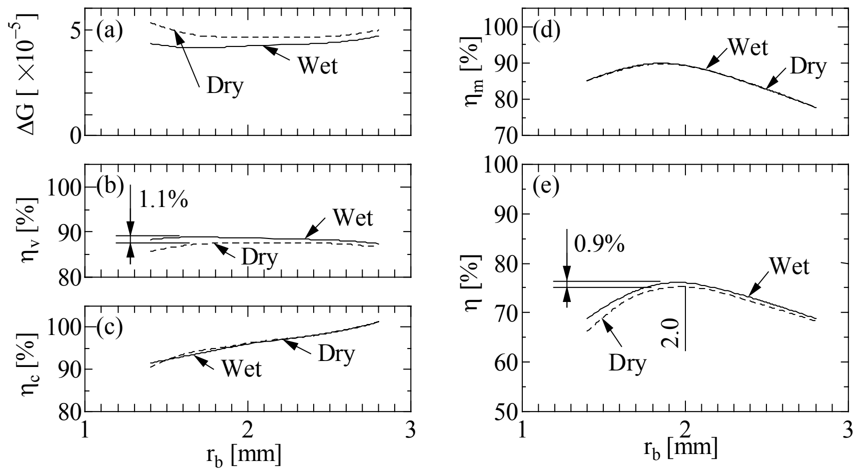

Figure 8. Calculated results for CO_2 scroll compressor: (a) refrigerant gas leakage ΔG; (b) volumetric efficiency η_v; (c) compression efficiency η_c; (d) mechanical efficiency η_m; (e) overall efficiency η.

in Table 2 also, both for the axial and radial clearances. Also shown in Table 2 are the major specifications for the CO_2 compressors. The suction volume and compression ratio were 4.25 cm^3 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, I_0, was adjusted depending upon the necessary driving shaft power, and the orbiting scroll mass m_o 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 r_b 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 ΔG during one revolution of the crankshaft was calculated, as shown in Fig. 8(a), where the dashed line is for the dry CO_2 gas and the solid line is for the wet CO_2 gas. The abscissa is the involute base circle radius r_b. Since the friction factor for the wet CO_2 gas is larger than that for the dry, the net leakage mass ΔG becomes smaller for the wet CO_2 gas than for the dry. The improvement for the wet CO_2 gas is about 10 % compared with for the dry CO_2 gas. As a result, the volumetric efficiency η_v is improved by about 1.1% for the wet CO_2 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 η_c exceeding 90% for both the dry and wet CO_2 gas, as shown in Fig.8(c). The mechanical efficiency η_m also exhibits its highest level of 90% for both the dry and wet CO_2 gas, as shown in Fig.8(d). As a result, the overall efficiency η, 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 r_b affording the optimal compressor efficiency takes the same value of 2mm for both the dry and wet CO_2 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 CO_2 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 1.2×10^4. 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 represented by the friction factor λ. 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 3μm for the axial and 6μm 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 λ 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
$\dot{M}$: Leakage mass flow rate, $kg \cdot s^{-1}$
n: Polytropic exponent
P, P_0, P_1: Absolute pressure of refrigerant gas, Pa
P_a: Atmospheric pressure, Pa
Re: Reynolds number
u_m, u_ϕ: Average flow velocity, $m \cdot s^{-1}$
W: Leakage channel depth, m
α_a: Coefficient of friction factor
β_a: Index of friction factor
δ, δ_a: Leakage clearance, μm
λ, λ_a: Friction factor
μ: Coefficient of viscosity, μPa·s
ρ, ρ_0: Density of refrigerant gas, kg/m^3

REFERENCES

Oku, T., Anami, K., Ishii, N., Knisely, C.W., Yasuda, K., Sawai, K., Sano, K. and Morimoto, T., 2005, Gas Leakage in CO_2 and R22 Scroll Compressors and Its Use in Simulations of Optimal Performance, *International Conference On Compressors And Their Systems 2005*, pp.403-412.

Ishii, N., Yamamoto, S., Sano, K., Sawai, K., Hiwata, A., Nakamoto, T. and Kawano, H., 2002a, Efficiency Simulations of a Compact CO_2 Scroll Compressor and Its Comparison with Same Cooling Capacity R410A Scroll Compressor, *International Compressor Engineering Conference at Purdue*, C22-2.

Ishii, N., Kawamura, S., Yamamoto, S., Sawai, K., Hiwata, A., Nakamoto, T., Kawano, H. and Ting, K.S., 2002b, Efficiency Simulations with Consideration of Heat Losses of R410 Compact Scroll Compressor for Its Optimal Performance, *International Compressor Engineering Conference at Purdue*, C22-3.

Ishii, N., Bird, K., Sano K., Oono, M., Iwamura S. and Otukura T., 1996a, Refrigerant Leakage Flow Evaluation for Scroll Compressors, *International Compressor Engineering Conference at Purdue*, Vol. II, pp.633-638.

Ishii, N., Sakai, M., Sano, K., Yamamoto S. and Otokura T., 1996b, A Fundamental Optimum Design for High Mechanical and Volumetric Efficiency of Compact Scroll Compressors, *International Compressor Engineering Conference at Purdue*, Vol. II, pp.639-644.

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