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# Effects of Surface Roughness upon Gas Leakage Flow through Small Clearances in $CO_2$ Scroll Compressors

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## ABSTRACT

This study presents empirical values of the friction factor for $CO_2$ gas leakage through thin rectangular cross-sectional openings of small axial and radial clearances between the orbiting and fixed scrolls of scroll compressors. Leakage flow experiments are conducted for different relative roughness of leakage channel surface. The pressure drop due to leakage is measured using a maximum pressure of 3 MPa. The Darcy-Weisbach equation for incompressible, viscous fluid flow through the thin rectangular cross-section is applied to calculate the pressure drop characteristics, where the empirical friction factors are determined and plotted on a Moody diagram. As a result, it is shown that the empirical friction factors for both axial and radial clearance leakage flows take on essentially the same values, strongly dependent on the relative roughness of leakage channel surface. Subsequently, the empirical friction factor is incorporated into computer simulations for a $CO_2$ scroll compressor with small cooling capacity to determine its optimal performance. As a result, it is shown that $CO_2$ scroll compressors can achieve higher performance levels by increasing the relative roughness of scroll wrap surfaces.

## 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. Scroll compressors have small axial and radial clearances between the orbiting and fixed scrolls. The leakage of the compressed refrigerant gas through these small clearances has a strong detrimental effect on the volumetric efficiency. In order to carry out accurate performance simulations of scroll compressors, a reliable method of calculating the leakage flow through these clearances is needed.

Previous studies by Ishii *et al.* (1996a) and Oku *et al.*, (2005, 2006) 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 a thin rectangular cross-section, in which the gas leakage flow is characterized in terms of a friction factor. As a result, it was shown that the effect of surface roughness on the friction factor becomes remarkable when the leakage clearance is less than about 10 μm.

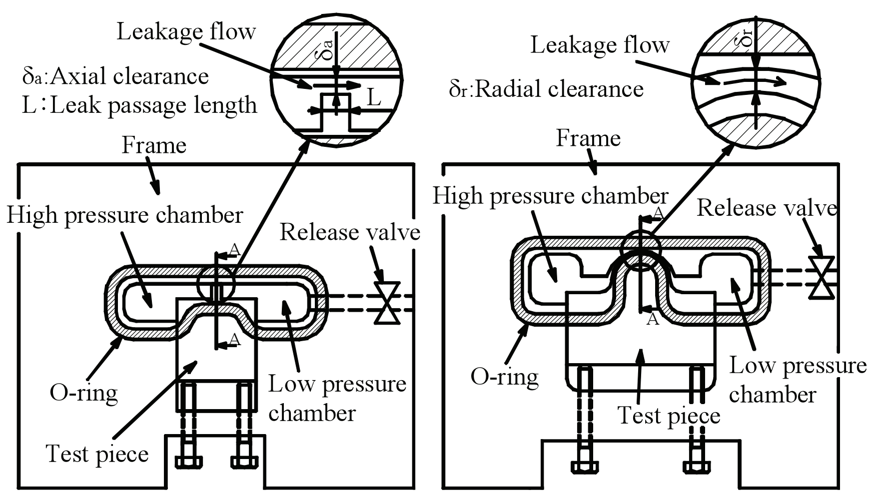
In the present study, very simple experiments are conducted with leakage channel surfaces with known relative roughness to identify more precisely the relationship between surface roughness and friction factor. Initially pressurized $CO_2$ gas in a large vessel is released to the atmosphere through axial and radial clearances, where the pressure decay due to gas leakage is measured for a variety of initial pressures, up to 3.1 MPa. The Darcy-Weisbach equation for incompressible, viscous fluid flow is used to calculate the leakage flow rate through the axial and radial clearances in terms of an unknown friction factor. Subsequently, the measured pressure decay is carefully simulated by assuming a polytropic process and provides an independent measure of mass discharge, from which the empirical friction factor is determined and subsequently plotted on a Moody diagram. The empirical friction factor is then correlated with its corresponding surface roughness, thus identifying the surface roughness effect. Finally, the empirically determined friction factor for various roughnesses is incorporated into computer simulations of a scroll compressor to calculate the volumetric, mechanical, compression and resultant overall efficiencies, thereby identifying the role surface roughness plays in the performance of a scroll compressor.

## 2. EXPERIMENTAL SET-UP

Experiments on leakage flows through axial and radial clearances in scroll compressors have been conducted using the two representative models shown in Figure 1, under the assumption that surface roughness has a significant effect on the leakage flow, with clearances of 10 μm and smaller. Tests were conducted with average surface roughness $\varepsilon$, ranging from 0.2 μm to 0.8 μm.

Figure 1a shows a plane view of the axial clearance model, where the leakage path has a streamwise length of $L$=4.0 mm and a depth of 15.0 mm. The leakage clearance height $\delta_a$ was carefully adjusted to 10 μm using thickness gauges with a width of 2.5 mm. Thus, the net leakage depth becomes 10.0 mm. In addition, an O-ring was attached between the test piece and the thrust plate, and a liquid gasket was also used on the contact surface to entirely eliminate any unintended leakage. Figure 1b shows the model for the radial clearance, where the involute curves of orbiting and fixed scrolls are represented by two circular arcs with different radii. The radii were fixed at 14.4 mm and 11.8 mm, respectively, and the net leakage depth of the radial clearance was fixed at 10.0 mm.

The high pressure chamber on the left side of test section is directly connected to a supply tank with a volume of 860 cm$^3$. The low pressure chamber on the right hand side is opened to atmospheric pressure through a discharge valve. With the discharge valve closed, both the high pressure and low pressure chambers are initially pressurized at a specified pressure. Then the low pressure chamber is suddenly vented to the atmosphere by opening the discharge valve. The pressure in the high pressure chamber decreases due to refrigerant leakage through the test section, and




(a) axial clearance model (b) radial clearance model

Figure 1 Test models for $CO_2$-gas leakage in scroll compressors.

its time-dependent pressure decay is measured by a semiconductor pressure transducer.

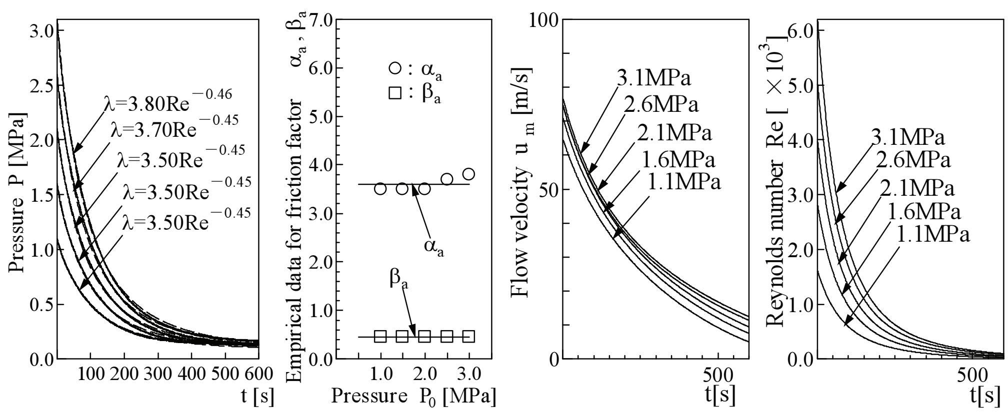
## 3. LEAKAGE TEST RESULTS

Measured pressure decays in the high pressure chamber due to leakage through 10 μm axial and radial clearances with the surface roughness of 0.4 μm are shown in Figures 2a and 3a as solid lines, where four additional test results with different initial pressures from 1.1 MPa to 3.1 MPa are also plotted. The initial pressures were chosen as being representative of pressure differences between adjacent compression chambers in scroll compressors. At time $t$ =0, the release valve opens the low pressure chamber to the atmosphere and then the high pressure decays due to refrigerant leakage, approaching the atmospheric pressure (about 0.1 MPa). The clearance heights in both the axial and radial tests are 10 μm, however the pressure decay through the radial clearance is faster than through the axial clearance. The initial temperature was between 16°C to 21°C, since both the $CO_2$ and the R22 leakage tests were made under the atmospheric exit conditions,

## 4. CALCULATIONS OF PRESSURE DECAY AND EMPIRICAL FRICTION FACTOR

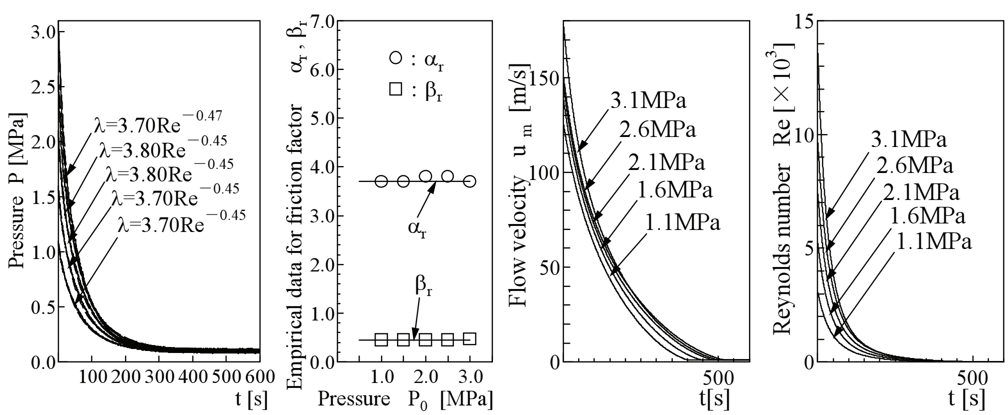
### 4.1 Leakage through the axial clearance channel

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:




(a) (b) (c) (d)

Figure 2 Tested and calculated results for $CO_2$-gas leakage flow through 10 μm axial clearance ($\varepsilon$ =0.4 μm): (a) pressure drops due to leakage; (b) empirical data for friction factor; (c) mean flow velocity; (d) Reynolds number.




(a) (b) (c) (d)

Figure 3 Tested and calculated results for CO2-gas leakage flow through 10 μm radial clearance ($\varepsilon$=0.4 mm): (a) pressure drops due to leakage; (b) empirical data for friction factor; (c) mean flow velocity; (d) Reynolds number.

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

where the friction 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 well-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 leakage passage length and average leakage flow velocity are represented by $L$ and $u_m$, respectively. The hydraulic mean depth is represented by $m$.

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 $Q$ can be calculated by

$$Q=\rho\delta_a W u_m \tag{2}$$

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

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

where the pressure decay is assumed to be a polytropic process with exponent $n$. $P_0$ represents the initial pressure, $G_0$ the initial mass on the high pressure side, and $G$ the mass in the high pressure chamber at any subsequent time.

The friction factor $\lambda_a$ is generally a function of the Reynolds number $Re$. Here the following expression for the friction factor is assumed:

$$\lambda_a=\alpha\cdot Re^{-\beta} \tag{4}$$

which corresponds to fully turbulent flow.

When the coefficient $\alpha$ and $\beta$ in Eq. (4) for the friction factor $\lambda_a$ are assigned the values given in Figure 2b, the simulated pressure decay, shown by the dashed lines in Figure 2a, for the axial clearance with a surface roughness of 0.4 μm shows close agreement with the measured pressure decay. Using the mean values of the plotted data for $\alpha$ and $\beta$, the friction factor $\lambda_a$ for the roughness $\varepsilon$=0.4 μm can be given by

$$\lambda_a=3.60Re^{-0.45} \tag{5}$$

As shown in Figure 2c, the leakage flow velocity $u_m$ is far smaller than the sonic speed for $CO_2$ (about 240 m/s), thus yielding a Mach number less than 0.3 and justifying the treatment of the flow as incompressible. The tested maximum Reynolds numbers were about 6175, as shown in Figure 2d.

Similar leakage tests and simulations were carried out for other conditions of the surface roughness of leakage passage. The friction factors obtained are listed in Table 1.

### 4.1 Leakage through the radial clearance channel

Figure 4 shows the theoretical model for leakage flow through the radial clearance. The compressed $CO_2$ gas leaks from the left side to the right side through the clearance between circular arcs with different radii of R and r, due to the pressure difference ($P$-$P_a$). The clearance height at angle $\phi$ from the minimum clearance $\delta_r$ can be approximated by the following expression:

$$h=R-(R-r-\delta_r)\cos\phi-\sqrt{r^2-(R-r-\delta_r)^2\sin^2\phi} \tag{6}$$

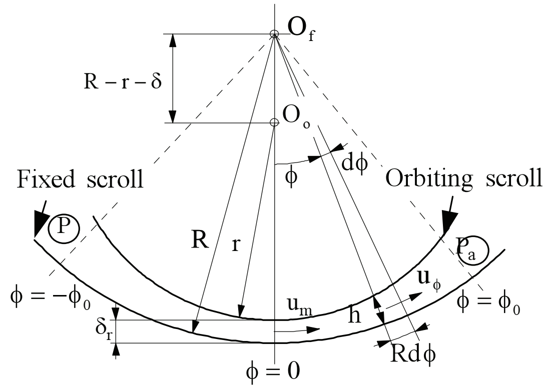



Figure 4 Equivalent flow model for leakage flow through radial clearance.

When the leakage velocity is represented by $u_\phi$ at the position of clearance $h$ and by $u_m$ at the minimum clearance, the following expression can be obtained from the equation of continuity:

$$u_\phi = \frac{\delta}{h} u_m \tag{7}$$

Since the integrated leakage friction caused by the leakage flow $u_\phi$ balances approximately the force due to pressure difference, the following expression can be obtained:

$$\frac{P - P_a}{\rho g} = \int_{-\phi_0}^{\phi_0} \lambda \frac{Rd\phi}{4m_\phi} \cdot \frac{u_\phi^{\,2}}{2g} \tag{8}$$

where $m_\phi$ is the hydraulic mean depth.

Assuming the friction factor can be given in the same form as in Eq. (4), the pressure decay can be calculated from Eqs. (2), (3) and (8). When the coefficient $\alpha$ and $\beta$ in Eq. (4) for the friction factor $\lambda_r$ are assigned the values plotted in Figure 3b, the calculated pressure decay for the radial clearance with surface roughness of 0.4 μm shows close agreement with the measured pressure decay, as shown by the dotted lines in Figure 3a. Using the mean values of the plotted data for $\alpha$ and $\beta$, the friction factor $\lambda_r$ for the roughness $\varepsilon$=0.4 μm can be given by

$$\lambda_r = 3.74\,\mathrm{Re}^{-0.45} \tag{9}$$

As shown in Figure 3c, the leakage flow velocity $u_m$ is larger than for the axial leakage, however $u_m$ is still far smaller than the sonic speed for $CO_2$ (about 240m/s). The tested maximum Reynolds numbers were about 13560, as shown in Figure 3d.

Similar leakage tests and simulations were carried out for other conditions of the surface roughness of leakage passage, resulting in the friction factors listed in Table 1.

The empirical friction factors with a leakage clearance height of 10 μm for varying surface roughnesses are represented by the solid and dashed lines on the Moody diagram in Figure 5, where the data for the tested range of Reynolds number is plotted by the solid line, while the extrapolated values for higher Reynolds numbers are plotted

Table 1 Friction factor to surface roughness

| Surface Roughness ε | Axial clearance $\delta_a$ | Radial clearance $\delta_r$ |
|---|---|---|
| 0.2μm | $\lambda$=3.38Re$^{-0.46}$ | $\lambda$=3.70Re$^{-0.46}$ |
| 0.4μm | $\lambda$=3.60Re$^{-0.45}$ | $\lambda$=3.74Re$^{-0.45}$ |
| 0.6μm | $\lambda$=3.56Re$^{-0.44}$ | $\lambda$=3.66Re$^{-0.44}$ |
| 0.8μm | $\lambda$=3.52Re$^{-0.43}$ | $\lambda$=3.62Re$^{-0.43}$ |

Table 2 Relative roughness

| | Axial clearance | | | | Radial clearance | | | |
|---|---|---|---|---|---|---|---|---|
| Clearance δ (μm) | 10 | | | | | | | |
| Hydraulic mean depth (μm) | 4.995 | | | | | | | |
| Equivalent diameter d (μm) | 19.98 | | | | | | | |
| Absolute roughness ε (μm) | 0.23 | 0.38 | 0.59 | 0.80 | 0.25 | 0.40 | 0.61 | 0.82 |
| Relative roughness ε/d | 0.011 | 0.019 | 0.030 | 0.040 | 0.013 | 0.020 | 0.031 | 0.041 |

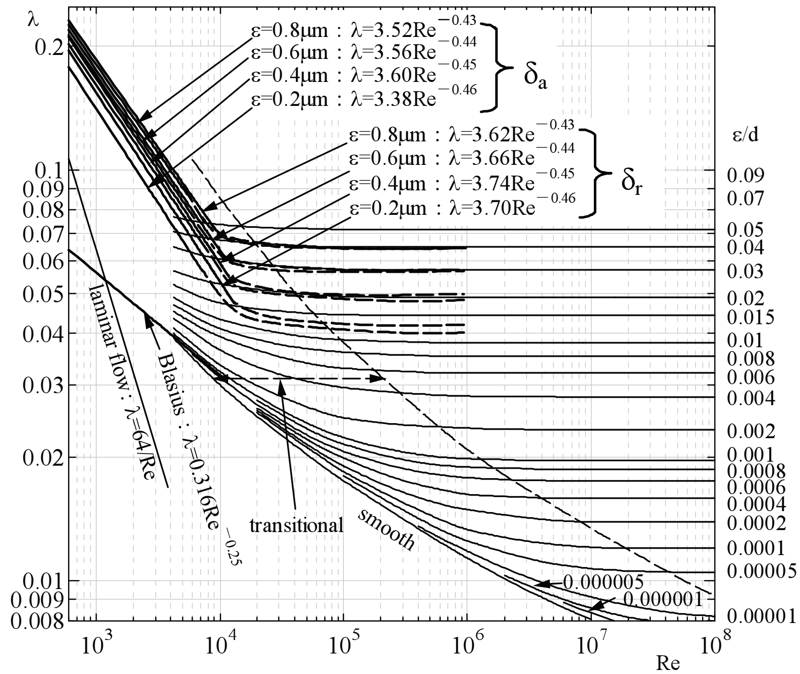



Figure 5. Moody diagram.

by the dashed lines which approach the corresponding relative roughness $\varepsilon / d$ on the right ordinate. The friction factor increases with increasing relative roughness for both the axial and radial clearance leakages.

## 5. CALCULATIONS OF SCROLL COMPRESSOR EFFICIENCY

Computer simulations of $CO_2$ and R410A scroll compressors have been undertaken to optimize compressor performance (see Ishii *et* al., 1996b, 2002a, and 2002b). In order to address the effect of surface roughness on the efficiency, similar computer simulations of efficiency were carried out in the present study for a small capacity scroll compressor with the specifications listed in Table 3. The scroll wraps are assumed to be carefully assembled with an axial clearance height of 3 μm and a radial clearance of 6 μm.

Table 3 Major specifications of Scroll compressor.

| | | | $CO_2$ | |
|---|---|---|---|---|
| Suction volume | | Vs [cc] | 4.25 | |
| Involute base circle radius | | $r_b$ [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 | |
| Specific heat ratio | | κ | 1.3 | |
| Suction temperature | | Ts [℃] | 10.5 | |
| Suction pressure | | $P_s$ [MPa] | 3.5 | |
| Discharge pressure | | $P_d$ [MPa] | 9.0 | |
| Axical clearance | | $\delta_a$ [μm] | 3.0 | |
| Radial clearance | | $\delta_r$ [μm] | 6.0 | |
| Surface roughness | | ε [μm] | 0.2 | 0.8 |
| Empirical fric.factor | Axical | $\lambda_a$ | $3.38Re^{-0.46}$ | $3.52Re^{-0.43}$ |
| | Radial | $\lambda_r$ | $3.70Re^{-0.46}$ | $3.62Re^{-0.43}$ |

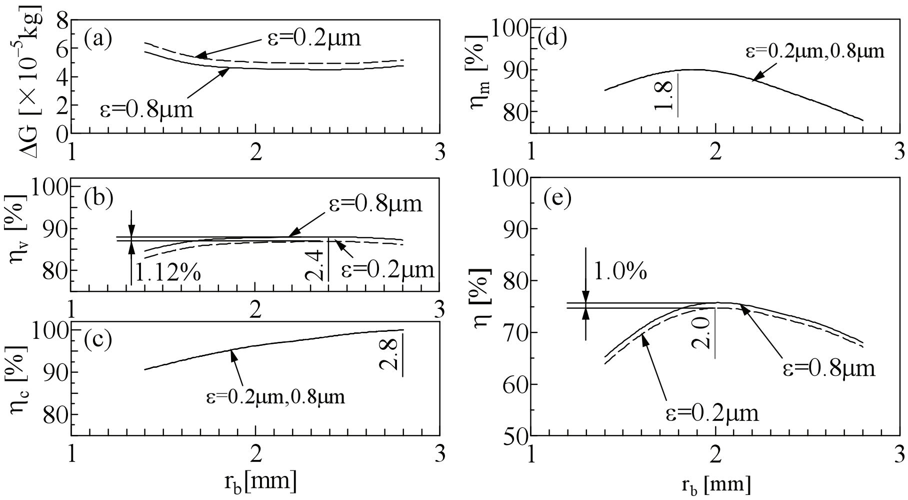



Figure 6. Calculated results : (a) refrigerant gas leakage $\Delta G$; (b) volumetric efficiency $\eta_v$; (c) compression efficiency $\eta_c$; (d) mechanical efficiency $\eta_m$; (e) overall efficiency $\eta$.

Two different surface roughnesses of 0.2 μm and 0.8 μm were assumed in the present efficiency simulations, resulting in the leakage mass flow $\Delta G$, shown in Figure 6a. $\Delta G$ shows a significant decrease with increasing surface roughness. As a result, the volumetric efficiency $\eta_v$ exhibits a maximum value of 86.87 % at $\varepsilon$=0.2 μm and 87.98% at $\varepsilon$=0.8 μm with an involute base circle radius of $r_b$=2.4 mm, as shown in Figure 6b. The volumetric efficiency is significantly proved by 1.12% by increasing the surface roughness from 0.2 μm to 0.8 μm, whereas the compression and mechanical efficiencies, $\eta_c$ and $\eta_m$, are not affected, exhibiting the same features with different roughness, as shown in Figures 6c and 6d. Finally, the overall efficiency $\eta$ exhibits a maximum value of 75.12% for $\varepsilon$=0.2 μm and 76.12% for $\varepsilon$=0.8 μm, as shown in Figure 6e. The overall efficiency is improved by a significant 1.0% by increasing the surface roughness from 0.2 μm to 0.8 μm.

## 6. CONCLUSIONS

Leakage flow experiments were conducted for $CO_2$ gas, flowing through models of the axial and radial clearances between the orbiting and fixed scrolls found in scroll compressors. The clearance height was 10 μm. The pressure drop from an initial pressure as high as 3.1 MPa due to leakage was measured over a range of Reynolds numbers up to 13560 for the radial clearance and 6175 for the axial clearance. The Darcy-Weisbach equation with an unknown friction factor for incompressible, viscous fluid flow through the thin rectangular channels was applied to the pressure drop to determine empirical values for the friction factor. The empirically determined friction factors were then plotted on a Moody diagram.

It was shown that the friction factor λ is strongly dependent on the relative roughness of the leakage channel surface, with the friction factor increasing with increasing surface roughness. Subsequently, the empirical friction factor was incorporated into computer simulations for $CO_2$ scroll compressors with the leakage clearance height of 3.0 μm and 6.0 μm for the axial and radial clearances, respectively, and for two different surface roughness of 0.2 μm and 0.8 μm. The computer simulations showed the effect of the surface roughness was to increase optimal overall efficiency by 1% from 75.12% to 76.12% at $r_b$=2.4 mm as the surface roughness increased from 0.2 to 0.8 μm.

The conclusion is that the volumetric efficiency of a scroll compressor can be significantly improved by increasing the surface roughness. This conclusion will provide a new, and perhaps counterintuitive, concept in the design of components for higher performance scroll compressors.

## NOMENCLATURE

| | | |
|---|---|---|
| $\alpha_a$, $\alpha_r$ | The coefficient portion of a friction factor | (-) |
| $\beta_a$, $\beta_r$ | The index portion of a friction factor | (-) |
| $\delta$, $\delta_a$, $\delta_r$ | Clearance | (μm) |
| $\varepsilon$ | Surface roughness | (μm) |
| $\lambda$ | Friction factor | (-) |
| $\eta$, $\eta_v$, $\eta_m$, $\eta_c$ | Efficiency | (%) |
| $B$ | Scroll height | (mm) |
| $D$ | Cylinder diameter | (mm) |
| $G$ | Mass in high pressure chamber | (kg) |
| $P_s$ | Suction pressure | (MPa) |
| $P_d$ | Discharge pressure | (MPa) |
| $Q$ | Leakage mass flow rate, | ($kg \cdot s^{-1}$) |
| $r_b$ | Involute base circle radius | (m) |
| $t$ | Scroll wrap thickness | (mm) |
| $T_s$ | Suction temperature | ($^oC$) |
| $u_m$, $u_\phi$ | Average flow velocity, | ($m \cdot s^{-1}$) |
| $V_s$ | Suction volume | ($cm^3$) |
| W | Channel depth | (mm) |

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