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ABSTRACT

This study presents computer simulations of the volumetric, mechanical and compression efficiencies of a high-pressure scroll compressor for ammonia, with a suction volume of 192.5 cm³. Computer calculations were made for a number of combinations of the involute base circle radius and the scroll height for a fixed cylinder diameter of 155 mm. As a result, the design dimensions of the scroll configuration leading to the optimum efficiency are presented for this particular cylinder diameter. Based on the computer simulations presented here, a high efficiency scroll compressor for ammonia has been developed and is expected to enter the market in 2008.

1. INTRODUCTION

In recent years, increased environmental concerns, such as global warming and energy utilization, have motivated the use of naturally occurring refrigerants and the optimization of components in the air-conditioning industry. In this context, the use of ammonia as a refrigerant with a zero potential contribution to global warming is attractive. In addition, ammonia has a significant technical advantage in that its heat capacity is 5.9 times larger than that of R22 and 3.9 times larger than that of CO₂, thereby making it especially advantageous when used in systems with large cooling capacities. Based on these environmental and technical issues, the refrigeration and air-conditioning markets have been demanding the development of scroll compressors (generally acknowledged to exhibit high performance and low vibration and low noise levels) for use with ammonia. In order to develop a high efficiency ammonia scroll compressor, a computer simulation of its expected performance was needed to optimize the combination of geometrical dimensions that would yield with maximum performance.

To satisfy this need for a computer simulation, the methodology used in earlier studies (Ishii et al. 1996a,b, 2002b) on optimizing scroll compressor performance for use with R22 and CO₂ was adapted to predict the performance of an ammonia scroll compressor with an approximate cooling capacity of 30 kW. Using this established methodology, the dependencies of the mechanical, compression and volumetric efficiencies on the dimensions of the involute base circle radius and the scroll height for a fixed cylinder diameter of 155 mm were determined. As a result, a high pressure ammonia scroll compressor with maximum performance was developed and is currently being manufactured to supply the market demand.


2. PERFORMANCE SIMULATION

Figure 1 shows a sectional view of an involute type scroll compressor. The scroll wrap profile is determined by the combination of the involute base circle radius \( r_b \), the wrap thickness \( t \) and the wrap height \( B \) for a cylinder diameter \( D \).

Refrigerant leakage through the wrap clearances depends on the combination of these dimensions, as well on the friction loss at the sliding pairs and the gas pressure in the compression chambers. The leakage, in turn, significantly affects the volumetric, mechanical, compression and overall efficiencies.

2.1 Volumetric Efficiency \( \eta_v \)

The volumetric efficiency \( \eta_v \) can be calculated as the ratio of the actual discharge mass flow rate to the theoretical mass flow rate into compressor:

\[
\eta_v = \frac{Q_e}{Q_{th}} = \frac{Q_{th} - Q}{Q_{th}}
\]

where \( Q_e \) is the actual discharge mass flow rate from compressor, which can be calculated by subtracting the leakage mass flow rate, \( Q \), from the theoretical mass flow rate into compressor suction port, \( Q_{th} \). This leakage flow rate through the small clearances caused by the pressure difference between compression chambers in the scroll compressor can be determined using the simple theory by Ishii et al. (1996a) and Oku et al., (2005, 2006). The overall leakage mass flow rate for one revolution of the scroll’s orbiting movement can be calculated by integrating the local differential leakage mass flow rate over the length of the scroll wrap.

Once the scroll wrap profile is determined from the given parameters, such as compressor size, motor power and refrigeration capacity, the volume and the pressure in the compression chambers formed between the orbiting and fixed scrolls can be calculated geometrically. Subsequently, the leakage flow can be calculated.

To calculate the leakage flow through the small clearances in scroll compressors, the Darcy-Weisbach equation for incompressible, viscous fluid flow was applied, using an friction factor \( \lambda \), shown in Figure 2 for \( \text{CO}_2 \). It is of significance to note that the corresponding empirically determined friction factor for R22 has been found to be almost indistinguishable from that for \( \text{CO}_2 \) (Oku et al., 2005). On this basis, it is assumed that the friction factor can

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Figure 1  Sectional view of scroll compressor.        Figure 2  Friction factors of axial and radial leakage flows of \( \text{CO}_2 \), plotted on Moody diagram.
be considered to be independent of the refrigerant itself. With this assumption, the friction factor $\lambda$ for ammonia can be given as:

$$\lambda_a = 3.38Re^{-0.46} \quad \text{and} \quad \lambda_r = 3.70Re^{-0.46}$$  \hspace{1cm} (2)

where $\lambda_a$ is the friction factor for the axial clearance and $\lambda_r$ is that for the radial clearance. The surface roughness was assumed to be 0.2 $\mu$m and the axial and radial clearances to be 10 $\mu$m.

### 2.2 Mechanical Efficiency $\eta_m$

The mechanical friction (between the crankshaft and the crank journal, the crank pin and the orbiting scroll, the orbiting scroll and the Oldham ring, and the orbiting scroll and the thrust bearing) is the major source of power loss in scroll compressors. These mechanical friction loads can be obtained from a dynamic analysis for each pair of machine elements, and then summed to determine the overall friction force.

Dynamic equilibrium analysis yields the equation of motion governing the behavior of the crankshaft rotation given in the following expression (Ishii, et al., 1992):

$$\left( I_o + m_r r_o^2 + m_o r_o^2 \sin^2 \theta \right) \ddot{\theta} + m_o r_o^2 \sin \theta \cos \theta \cdot \dot{\theta}^2 = N - \left\{ F_r r_o + L_s + (f_{s1} + f_{s2}) \cdot r_o \sin \theta \right. $$

$$\left. + \left( f_{r1} + f_{r2} \right) \cdot r_o \cos \theta + (f_{r1} + f_{r2}) \cdot r_o \right\}$$

where, the terms on left side represent the inertia torques and those on the right side represent the motor drive torque $N$ and the torque loads due to the gas compression, the mechanical friction torques at the crankshaft, the crankpin, the Oldham ring and the thrust slide-bearing. The mechanical friction forces are obtained as the product of the resultant force at each pair of compressor elements and the corresponding friction coefficient. The equation of motion (Eq. 3) can be solved numerically for the given torque characteristic of the electric motor and the pressure in the compression chambers to obtain a periodic solution.

Integrating the equation of motion of the crankshaft over the duration of one revolution of the crankshaft, an energy balance can be obtained. The shaft input energy $W_s$, the gas compression energy $W_i$ and the frictional losses $W_f$ are given by the following expressions, respectively:

$$W_s = \int_0^{\pi r} N \dot{\theta} dT \quad \text{and} \quad W_i = \int_0^{\pi r} F_r r_o \dot{\theta} dT$$

$$W_f = \int_0^{\pi r} \left\{ L_s + (f_{s1} + f_{s2}) \right\} r_o \dot{\theta} dT$$

The mechanical efficiency $\eta_m$ can then be calculated as:

$$\eta_m = \frac{W_i}{W_s} - \frac{W_f}{W_s}$$

### 2.3 Compression Efficiency $\eta_c$

Due to the continual re-compression of the leakage flows and the dissipation of energy through the frictional losses associated with the leakage flow, a scroll compressor with leakage requires greater compression power than one with no leakage. Therefore, the compression efficiency $\eta_c$ can be defined by the ratio of the theoretical compression power with no leakage, $E_{th}$, to that for the compressor with leakage, denoted by $\eta_c$, takes on a value that is then less than 1.0:

$$\eta_c = \frac{E_{th}}{E}$$

### 2.4 Resultant Efficiency $\eta$

The overall efficiency $\eta$ can be obtained as the product of the component efficiencies $\eta_v$, $\eta_m$ and $\eta_c$:

$$\eta = \eta_v \times \eta_m \times \eta_c$$
which represents the ratio of the accumulated energy in the discharged refrigerant to the shaft input power.

3. CALCULATED RESULTS OF EFFICIENCIES

The major specifications for computer calculations are shown in Table 1, where the operation conditions and the major dimensions of compressor are specified. The condensation temperature is 55°C and the evaporation temperature is 0°C. The suction volume $V$, cylinder diameter $D$ and wrap thickness $t$ are conventionally fixed at 192.5 cm$^3$, 155 mm and 4.7 mm, respectively. As a result, the scroll wrap height $B$ is determined for a given value of the involute base circle radius $r_b$. The orbiting radius $r_o$ changes from 5.98 mm to 12.26 mm with increasing involute base circle radius $r_b$ from 3.4 mm to 5.4 mm. Both the axial clearance $\delta_a$ and radial clearance $\delta_r$ are fixed at 10 $\mu$m. The friction coefficients, as is usual, are taken to be 0.055 for the thrust slide bearing, 0.011 for the crank pin and crank journal and 0.0013 for the ball bearing at the end of the crank shaft, all which were measured by friction tests on small cooling capacity compressors.

![Figure 3](image)

Figure 3  Calculated results: (a) refrigerant gas leakage; (b) pressure curve; (c) volumetric efficiency $\eta_v$.  

<table>
<thead>
<tr>
<th>Operation conditions</th>
<th>Condensation temperature $T_c$ (°C)</th>
<th>55</th>
</tr>
</thead>
<tbody>
<tr>
<td>Condensation pressure $P_c$ [MPa]</td>
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<td>2.31</td>
</tr>
<tr>
<td>Evaporation temperature $T_s$ (°C)</td>
<td></td>
<td>0</td>
</tr>
<tr>
<td>Evaporation pressure $P_s$ [MPa]</td>
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<td>0.43</td>
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<tr>
<td>Involute base circle radius $r_b$ [mm]</td>
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<td>3.4 ~ 5.4</td>
</tr>
<tr>
<td>Scroll thickness $t$ [mm]</td>
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<td>4.7</td>
</tr>
<tr>
<td>Scroll height $B$ [mm]</td>
<td></td>
<td>28.8 ~ 45.7</td>
</tr>
<tr>
<td>Orbiting radius $r_o$ [mm]</td>
<td></td>
<td>5.98 ~ 12.26</td>
</tr>
<tr>
<td>Suction volume $V$ [cm$^3$]</td>
<td></td>
<td>192.5</td>
</tr>
<tr>
<td>Cylinder disameter $D$ [mm]</td>
<td></td>
<td>155</td>
</tr>
<tr>
<td>Axial clearance $\delta_a$ [μm]</td>
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<td>10</td>
</tr>
<tr>
<td>Radial clearance $\delta_r$ [μm]</td>
<td></td>
<td>10</td>
</tr>
<tr>
<td>Fric. coef. at thrust bearing $\mu_s$ [-]</td>
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<td>0.055</td>
</tr>
<tr>
<td>Fric. coef. at crankpin $\mu_{cp}$ [-]</td>
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<td>0.011</td>
</tr>
<tr>
<td>Fric. coef. at crank journal $\mu_{cj}$ [-]</td>
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<td>0.011</td>
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<tr>
<td>Fric. coef. at ball bearing $\mu_{cb}$ [-]</td>
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<td>0.0013</td>
</tr>
</tbody>
</table>

Table 1  Operation conditions and major dimensions for performance calculations.
First, the leakage flow velocity and leakage mass flow rate were calculated for both the axial and radial clearances. The leakage mass flow rate $\Delta G$ for one cycle of orbiting motion is shown in Figure 3(a), in which the abscissa is the orbiting radius $r_b$. As $r_b$ increases, the pressure difference between the suction chamber and the inner compressed chamber increases, as shown in Figure 3(b), thus resulting in an increase in the refrigerant gas leakages through the radial and axial clearances, $\Delta G_r$ and $\Delta G_a$. The resultant total refrigerant gas leakage $\Delta G$ increases from $3.1 \times 10^{-5}$ kg to $6.6 \times 10^{-5}$ kg as $r_b$ increases from 3.4 mm to 5.4 mm, and results in a corresponding decrease in volumetric efficiency from its maximum value of 93.5% at $r_b = 3.4$ mm to 86% at $r_b = 5.4$ mm, as shown in Figure 3(c).

Figure 4(a) shows the calculated friction power losses. As the orbiting radius $r_b$ increases, the friction loss $W_{c-p}$ at the crankpin and $W_{c-s}$ at the crankshaft decrease gradually, since the wrap height decreases and, hence, the gas loads on the crankpin and crankshaft decrease. To the contrary, as $r_b$ is increased, the friction loss $W_{t-b}$ at the thrust slide-bearing increases, because the chamber bottom area increases and, hence, the gas thrust force on the thrust slide-bearing increases. The resultant friction loss $W_f$ exhibits a convex trend, initially decreasing, then increasing. Figure 4(b) shows the gas compression power, $W_i$, and the shaft input power, $W_s$. From these results, the mechanical efficiency $\eta_m$ can be obtained as shown in Figure 4(c), in which the mechanical efficiency $\eta_m$ exhibits its maximum

![Figure 5](image-url)  
Figure 5  $P-V$ diagram of compression chamber: (a) $r_b=3.8$ mm; (b) $r_b=4.6$ mm; (c) $r_b=5.4$ mm.
value of 92% at \( r_b = 4.0 \text{mm} \).

The \( P-V \) diagrams for \( r_b = 3.8, 4.6, 5.4 \text{ mm} \) are shown in Figures 5(a), (b) and (c), respectively, in which the theoretical gas pressure \( P_T \) is plotted by the dashed line and the actual gas pressure \( P_R \) by the solid line. The sudden increase in pressure, that occurs as the inner chamber connects to discharge chamber was calculated by Dalton’s Law. From these \( P-V \) diagrams, the compression efficiency \( \eta_c \) can be calculated, as shown in Figure 6, in which \( \eta_c \) increases with increasing \( r_b \) and approaches 100% at \( r_b = 5.4 \text{ mm} \). The value of the compression efficiency is governed predominantly by the in-flow leakage from the higher pressure compression chamber, which has a far larger effect that due to leakage out-flow to the suction chamber for the small \( r_b \) range.

Finally, the overall efficiency \( \eta \) can be calculated by Eq. (7), as shown in Figure 7, in which \( \eta \) exhibits its maximum value of 79.9% at \( r_b = 4.6 \text{ mm} \). The scroll wrap profile providing the maximum overall efficiency is shown in Figure 8, where the scroll height was 32.4 mm and the aspect ratio was 0.21.

4. DEVELOPED AMMONIA SCROLL COMPRESSOR

The high pressure ammonia scroll compressor, developed based on the identified optimal scroll wrap dimensions, is shown in Figure 9. The refrigerant oil is contained in the lower shell region and is pumped up with a trochoidal pump installed on the end of the crankshaft through a hole in the crankshaft to lubricate the compression mechanism.
The compression mechanism is driven by a high performance 15 kW IPM (Interior Permanent Magnet) motor with an aluminum coil. This IPM motor provides high power with small space requirements. All materials in the scroll compressor were carefully chosen to resist the corrosive effects of the ammonia. The design pressure is up to 2.6 MPa corresponding to a condensation temperature of 55°C. The suction pipe is directly connected to the suction chamber to avoid heating the suction gas. The compressed gas is cooled by liquid injection and discharged into the compressor shell to separate the refrigerant oil and to cool the motor, and then discharged through the discharge port positioned at the side of the shell.

As a result, an ammonia scroll compressor with a high COP was successfully developed and entered the markets in 2008.

5. CONCLUSION

Efficiency calculations of an ammonia scroll compressor were undertaken to determine the optimal design values for involute base circle radius and scroll height for a fixed cylinder diameter of 155 mm. The condensation temperature, evaporation temperature, and suction volume were fixed at 55°C, 0°C, and 192.5 cm$^3$, respectively. The ammonia leakage flow through axial and radial clearances between the fixed and orbiting scrolls was calculated with application of the friction factor for CO$_2$ gas leakage to obtain the volumetric and compression efficiencies, whereas the friction losses were calculated to determine the mechanical efficiency. As a result, the overall efficiency of 79.9% was found for the involute base circle radius of 4.6 mm. The scroll wrap height and aspect ratio were 32.4 mm and 0.21, respectively. Finally, a high pressure ammonia scroll compressor has been manufactured based on the obtained scroll wrap dimensions and possesses a higher COP than the conventional compressor.

In the present efficiency simulations, it is assumed that ammonia leakage is can be calculated using friction factors for dry CO$_2$ gas leakage. In the near future, leakage tests for ammonia will be performed to determine precisely the appropriate friction factor for ammonia and then incorporated into the analysis to ensure the validity of the present simulation of efficiency.
NOMENCLATURE

- \( B \): Wrap height (mm)
- \( D \): Cylinder diameter (mm)
- \( E, E_{th} \): Gas compression energy (W)
- \( f_{x1}, f_{x2} \): Friction force at fixed slot of Oldham ring (N)
- \( f_{y1}, f_{y2} \): Friction force at Oldham ring (N)
- \( f_{t1}, f_{t2} \): Friction force at thrust bearing (N)
- \( G \): Refrigerant mass (kg)
- \( I_0 \): Moment of inertia of crankshaft (kg·m\(^2\))
- \( L_Q \): Friction torque at crankshaft bearing (N·m)
- \( L_s \): Friction torque at crankpin (N·m)
- \( m_o \): Oldham ring mass (kg)
- \( m_r \): Orbiting scroll mass (kg)
- \( N \): Motor drive torque (N·m)
- \( Q, Q_e, Q_{th} \): Leakage mass flow rate (kg/s)
- \( r_b \): Involute base circle radius (mm)
- \( t \): Wrap thickness (mm)
- \( V \): Suction volume (cm\(^3\))
- \( W_f \): Frictional loss energy (W)
- \( W_i \): Gas compression energy (W)
- \( W_s \): Shaft input energy (W)
- \( \delta_a, \delta_r \): Axial and radial clearance (μm)
- \( \eta \): Overall efficiency (-)
- \( \eta_c \): Compression efficiency (-)
- \( \eta_m \): Mechanical efficiency (-)
- \( \eta_v \): Volumetric efficiency (-)
- \( \lambda_a, \lambda_r \): Friction factor (-)

REFERENCES


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