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Optimum Combination of Parameters for High Mechanical Efficiency of a Scroll Compressor

by

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ABSTRACT

This paper presents an optimum combination of major parameters for high mechanical efficiency of a compact scroll compressor. The frictional power loss at each pair of compressor elements depends mainly upon the corresponding constraint force. The constraint forces depend not only upon the compression pressure and the inertia forces but also upon the major parameters, such as the involute base circle diameter, the scroll depth, the scroll thickness and the number of scroll turns. It is suggested here that the suction volume of the scroll compressor is determined by these major parameters and, subsequently, there are many combinations of major parameters that result in a scroll compressor with the same suction volume. Therefore, the constraint force and the frictional power loss at each element pair change, depending upon the selected combination of major parameters, thereby resulting in a different mechanical efficiency. Thus, selection of the optimum combination of major parameters is necessary to ensure a high mechanical efficiency of the scroll compressor. The mechanical efficiency of a small capacity scroll compressor was calculated, where the number of scroll turns and the scroll thickness were fixed at constant values and the combination of the involute base circle diameter and the scroll height was varied for a fixed suction volume. The calculated results indicate that there is an optimum combination of major parameters. In addition, the physical basis of the optimum combination of major parameters was examined by showing the characteristics of the frictional power loss at each pair.

1. INTRODUCTION

Mechanical efficiency is a major consideration in designing refrigerant compressors used for air conditioners or refrigerators. A slight improvement in mechanical efficiency results in a significant energy savings, since a large number of refrigerant compressors is used all over the world. When designing refrigerant compressors, therefore, one should try to minimize the frictional power loss at each pair of the compressor elements. The frictional power loss depends mainly upon the frictional coefficients and constraint forces at each pair of compressor elements. Factors which have been seriously considered in achieving in a lower frictional coefficient are better selection of highly effective lubricants, devices which depend on a specific lubricating mechanism, introduction of a new sintered material into the refrigerant compressors, and so on. On the other hand, in order to decrease the constraint forces themselves, an optimum combination of dimensions of the compressor elements is being investigated. This study presents the calculated results for an optimum combination of parameters to achieve high mechanical efficiency in a compact scroll compressor.

If the rotary behavior of the crankshaft in refrigerant compressors is carefully analyzed with regard to the fundamental dynamics of machinery, it is possible to calculate precisely the constraint forces. Several comprehensive studies were carried out for the reciprocating compressors [see Refs. 1 & 2], the rolling-piston rotary compressors [3], the scroll compressors [4-6], the variable displacement wobble plate compressors [7] and also for the slider-crank drive Stirling machines [8]. On the basis of these studies, the frictional power losses and the mechanical efficiency were carefully calculated for the rolling-piston rotary compressor [9-11] and the scroll compressor [12 & 13]. It is quite interesting to note here that in a rolling-piston type rotary compressor, the constraint forces depend not only upon the cylinder pressure and the inertia forces but also upon the major dimensions, such as the rolling-piston diameter, the cylinder depth and the cylinder bore, for example. The suction volume of the rolling-piston ro-
tary compressor is determined by these major dimensions and, consequently, there are many combinations of major dimensions that result in a rolling-piston rotary compressor with the same suction volume. Depending upon the selected combination of major dimensions, the constraint force and the frictional power loss at each element pair change, and this results in different mechanical efficiencies. Thus, selection of the optimum combination of major dimensions is necessary to ensure a high mechanical efficiency of the rolling-piston rotary compressor. The mechanical efficiency was calculated for various possible combinations of major dimensions, and it was shown that there is an optimum combination of major dimensions for a rolling-piston rotary compressor [14].

A similar study can be made for the scroll compressors as well. Matsushita developed a vertical scroll compressor with a cooling capacity of 2.4 kW and began installing it in room air conditioners which were put on the market in 1990 [15]. Moreover, a more compact variable speed horizontal scroll-type compressor with a cooling capacity of 1.8 kW was developed and sales began in 1991 [16]. Since any further reduction in scroll compressor size would reduce the efficiency, careful attention was paid to the mechanical efficiency as well. In the case of scroll compressors, the major parameters determining the suction volume are the involute base circle diameter, the scroll depth, the scroll thickness and the number of scroll turns. Depending upon the selected combination of these major parameters, the mechanical efficiency will definitely change. One of the main reasons for the present study was to determine if there is an optimum combination of parameters for scroll compressors as well, and to clarify the physical basis of this. The number of scroll turns and the scroll thickness were fixed at constant values of 2.75 and 4.0 mm, respectively, and the combination of the involute base circle diameter and the scroll height was varied for a fixed suction volume of 10.26 cm³. When calculating mechanical efficiency, reasonable values must be determined for the frictional coefficients. This study made use of the frictional coefficients which were determined numerically in the previous studies [9-11] for a rolling-piston rotary compressor with small cooling capacity. It is well known that the scroll compressor mechanism is quite different from the rolling-piston rotary compressor mechanism. However, if the frictional state of the scroll compressor is assumed to be basically the same as that of the rolling-piston rotary compressor, it would be interesting to examine how the mechanical efficiency is affected by the difference in the compression mechanism itself. An important characteristic of the scroll compressor is its extremely low level of vibrations. In order to monitor the vibration level which will be affected by the selected combination of major parameters, therefore, the rotating speed fluctuation of the crankshaft was also calculated.

2. SUMMARY OF THE PROCEDURE FOR CALCULATING MECHANICAL EFFICIENCY

The cross-sectional view of a compact scroll-type compressor is shown in Fig. 1a. The major power losses are caused by the mechanical friction between the crankshaft and the crank journal, the crank pin and the orbiting scroll, and the orbiting scroll and the thrust bearing. The frictional power loss around the Oldham ring is negligibly small.

![Figure 1. Closed-type scroll compressor: (a) vertical cross-sectional view; (b) horizontal cross-sectional view on A-A' plane.](image-url)
2. SUMMARY OF THE PROCEDURE FOR CALCULATING MECHANICAL EFFICIENCY

The cross-sectional view of a compact scroll-type compressor is shown in Fig. 1a. The major power losses are caused by the mechanical friction between the crankshaft and the crank journal, the crank pin and the orbiting scroll, and the orbiting scroll and the thrust bearing. The frictional power loss around the Oldham ring is negligibly small compared with other losses [5, 12 & 13]. It is assumed that the frictional forces of any pair that is not well lubricated by a thick oil film are subject to Coulomb's law of friction. Therefore, the frictional forces can be obtained by multiplying the constraint forces at each pair of compressor elements by an adequate frictional coefficient. The constraint forces can be derived from the equations of motion of all moving elements. For example, the constraint force components between the crank pin and the orbiting scroll, $S_x$ and $S_y$, are given in the following form composed of the inertia forces, gas forces and frictional forces:

\[
S_x = -(m_b + m_o)x_t + (F_t + f_{11} + f_{12}) \sin \theta + F_r \cos \theta,
\]
\[
S_y = m_b y_t + (F_t + f_{11} + f_{12}) \cos \theta - F_r \cos \theta,
\]

where the main variable is the clockwise orbiting angle from the x-axis of the orbiting scroll, $\theta$. The $x$ and $y$ axes represent the rectangular coordinate shown in Fig. 1b, which is the horizontal cross section denoted by A-A' in Fig. 1a. The coordinate $(x_t, y_t)$ represents the orbiting scroll center $O_m$ shown in Fig. 1b. The constraint force components between the crankshaft and the crank journal are also described by similar expressions. The first terms on the right-hand side of Eq. (1) represent the inertia force of the orbiting scroll (the mass, $m_b$) and the Oldham ring (the mass, $m_o$). The other terms represent the effects of the gas forces and the frictional forces. The tangential and radial components of the compressed gas forces acting on the orbiting scroll are represented by $F_t$ and $F_r$, respectively, as shown in Fig. 1b. The frictional forces at the thrust bearing are represented by $f_{ii}$ (i=1,2). The gas forces $F_t$ and $F_r$ can be reduced to an integral expression proportional to the product of the scroll height $B$ and the involute base circle $r_0$:

\[
F_t, F_r \propto B \times r_0.
\]

If all constraint forces are written so that they are in a form similar to Eq. (1), the equation of motion of the crankshaft (the moment of inertia $l_0$) driven by the motor torque, $N$, can be reduced to the following form:

\[
(I_0 + m_b r_0^2 + m_o r_0^2 \sin^2 \theta) \ddot{\theta} + m_o r_0^2 \sin^2 \theta \cos \theta \dot{\theta}^2 = N - [F_t r_0 + L_0 + L_S + (f_{11} + f_{12}) r_0],
\]

where $r_0$ represents the orbiting radius. $L_0$ and $L_S$ represent the frictional torques at the crank journal and the crank pin, respectively; they are obtained by multiplying the frictional forces by the orbiting radius. If the discharge and suction gas pressures, and the motor torque curve are given, the dynamic rotatory behavior of the crankshaft can be calculated. Using these calculated results, the constraint forces, the frictional forces, and the frictional torques can all be calculated. Thus, the frictional power loss at each pair of compressor elements and, finally, the mechanical efficiency can be calculated.

3. SUCTION VOLUME AND FRICTIONAL COEFFICIENT

For the scroll compressor characterized by involute curves, the area of the crescent-shaped compression chamber is given by a second order polynomial expression with respect to the base circle radius, $r_b$; by multiplying the area of the chamber by the scroll height, $B$, the compression chamber volume is calculated. When the ending involute angle of the scroll curve is 990 degrees, that is, when the number of scroll turns is 2.75, the maximum value of the compression chamber volume, namely, the suction volume, $V_s$, is given by

\[
V_s = 16 \pi^3 B \left( r_b^2 - \frac{7}{8 \pi} r_b - \frac{12}{8 \pi^2} \right),
\]

where $t$ represents the scroll thickness. Here the suction volume is 10.26 cm$^3$, and the scroll thickness is 4.0 mm. Thus, from the above expression, the combination of the scroll height, $B$, and the base circle radius, $r_b$, can be obtained as shown in Fig. 2. It is also apparent from Eq. (4) that the base circle radius increases rapidly as the scroll height approaches zero.

In the frictional loss analysis for a rolling-piston rotary compressor [9-11], first, it was determined experimentally that there is a fluid lubrication between the rolling piston and the crank pin, and a boundary lubrication between the crankshaft and the crank journal, and the blade and the cylinder slot. Subsequently, the oil viscosity coefficient was determined on the basis of oil pressure and temperature experiments, and the two friction coefficients at the crank journal and the blade were adjusted so that
Figure 2. Combination of scroll height and involute base circle radius for the number of scroll turns equal to 2.75.

the calculated values of the mechanical efficiency and the mean rotational speed of the rolling piston agreed with their respective measured values. The calculated values for the frictional coefficients were determined to be 0.013 at the crank journal and 0.083 at the blade. The rolling-piston rotary compressor is well lubricated, for example, by the fluid lubrication between the rolling piston and the crank pin. On the other hand, the scroll compressor has no fluid lubrication, but it also has no poorly-lubricated pair such as the blade and the cylinder slot. We may assume, therefore, that the boundary lubrication between the sliding pairs in the scroll compressor is the same as that for the crank journal in the rolling-piston rotary compressor; the frictional coefficients for the scroll compressor also take on a value of 0.013.

4. CALCULATED RESULTS FOR FRICTIONAL POWER LOSS, MECHANICAL EFFICIENCY AND RotATING SPEED FLUCTUATION

By multiplying each term in the equation of motion (3) by a small rotatory displacement, \( \theta \), and integrating over one revolution of the crankshaft, the motor input power \( W_{\text{shaft}} \), the gas compression power \( W_{\text{gas,c}} \) and each power loss due to frictional forces can be calculated. The mechanical efficiency, \( \eta_m \), is then obtained from the ratio of the gas compression power to the motor input power. In the calculations, the combinations of the base circle radius and the scroll height were varied, but the scroll thickness, the radiiuses of crankshaft and crank pin were fixed at a value of 10 mm and the moment of inertia of the crankshaft was fixed at a value of 0.0422 N/cms². The diameter of the thrust bearing and the mass of the orbiting scroll were adjusted depending on the base circle radius and the scroll height. More detailed specifications of the scroll compressor are given in previous studies (4 & 5).

Assume here that the gas-compression process is subjected to an adiabatic change of specific heat ratio 1.32, which is the value of the super-heated Freon R-22 vapor at a pressure of 1.27 MPa and a temperature of 55.6 °C. The calculated results for the suction pressure of 0.62 MPa and the discharge pressure of 2.17 MPa are shown in Fig.3. The top diagram in Fig.3 clearly shows that there is an optimum combination of the scroll height, \( B \), and the base circle radius, \( r_b \); the mechanical efficiency, \( \eta_m \), takes on a maximum value of 92.5 % when \( B \) is 5.0 mm and \( r_b \) is 2.7 mm. It is interesting to note here that this maximum mechanical efficiency for the compact scroll compressor exceeds the high mechanical efficiency of about 90.3 % for a compact rolling-piston rotary compressor.

In the present calculation, it was assumed that the thrust bearing is so well lubricated that the frictional coefficient takes on a comparatively small value of 0.013, which is the same value as the frictional coefficient at the crank journal of a compact rolling-piston rotary compressor. Now assume that the thrust bearing is not so well lubricated and the frictional coefficient takes on a fairly large value of 0.083, which is the frictional coefficient between the blade and the cylinder slot. The calculated results for this case are shown by a dotted line in the top diagram of Fig.3. The optimum combination at \( B = 6.0 \) mm and \( r_b = 2.5 \) mm is a little different from the previous calculation results. Because of the increased frictional coefficient at the thrust bearing, therefore, the maximum value of the mechanical efficiency decreases to 85.8 %.

As shown in the top diagram of Fig.3, the gas compression power \( W_{\text{gas,c}} \) is al-
most constant, but the motor input power $W_{\text{motor}}$ increases as the combination of the scroll height and the base circle radius gets farther away from the optimum combination. Correspondingly, the mechanical efficiency $\eta_m$ shows a maximum value. The increased motor input power is definitely caused by the increase in frictional power losses, which can be explained as follows: (1) as the scroll height $B$ increases, the frictional power loss $W_{\text{fb}}$ at the thrust bearing does not change but the frictional power losses at the crankshaft and crank pin, $W_{\text{c-a}}$ and $W_{\text{c-p}}$, increase in proportion to the scroll height; (2) as the scroll height $B$ decreases, all the frictional power losses increase rapidly.

As shown in the bottom diagram of Fig.3, the rotating speed fluctuation ratio of the crankshaft, $\alpha$, takes on a constant value of about 0.5 %, even if the combination of the scroll height $B$ and the base circle radius $r_b$ changes. This calculated result suggests that the compressor vibrations are not significantly affected by the combination of $B$ and $r_b$.

5. PHYSICAL BASIS OF THE OPTIMUM COMBINATION

We wished to determine why the frictional power losses at the crank journal, the crank pin and the thrust bearing changed depending upon the combination of the scroll height and the base circle radius. One may conclude from Eq.(1) that the constraint
forces at the crank journal and crank pin are composed mainly of the gas forces (F₁ and F₂) and the inertia force of the orbiting scroll. Since the mass of the Oldham ring is far smaller than that of the orbiting scroll, the inertia force of the Oldham ring can be disregarded here. Incidentally, the gas forces are proportional to the product of the scroll height and the base circle radius, (B•rb), as shown in Eq.(2). On the other hand, the inertia force is naturally proportional to the product of the mass of the orbiting scroll and the orbiting radius, (m₁•r₀), where the orbiting radius r₀ is proportional to the base circle radius r₀:

\[ r₀ = r_b \pi - t. \]  

In the present study, the scroll thickness t takes on a value of 4.0 mm in this expression.

The product (B•r₀) is taken as a representative quantity of the gas forces and the product (m₁•r₀) represents the inertia force. The calculated results are shown in Fig.4. As the scroll height B increases, the inertia force term becomes smaller but the gas force term increases in proportion to B. As the scroll height B decreases, on the other hand, the gas force term becomes smaller but the inertia force term increases rapidly. The two curves cross near B=5 mm. Consequently, as the scroll height gets farther away from B=5 mm, the shaft loads at the crankshaft and crank pin increase, thus resulting in the kind of frictional power losses at the crankshaft and crank pin shown in the middle diagram of Fig.3.

As the scroll height B approaches zero, the base circle radius r_b increases rapidly, as shown in Fig.2. Moreover, the area of the crescent-shaped compression chamber increases in proportion to the square of the base circle radius (see, for example, Eq.(4)). These results suggest that the constraint forces at the thrust bearing, namely the thrust loads, F₁₁ and F₁₂, increase rapidly as the scroll height B decreases. Two examples of the calculated results are shown in Fig.5. The abscissa is the time T over one revolution of the crankshaft. The solid curves show the thrust loads at the optimum combination (B=5 mm and r_b=2.7 mm) and the dotted curves the thrust loads at an unfavorable combination of B=1 mm and r_b=5.2 mm. One may clearly conclude from these calculated results that if the scroll height becomes too small, the thrust loads increase significantly, resulting in the large increase in the frictional power losses at the thrust bearing, as shown in the middle diagram of Fig.3.

6. CONCLUSIONS

In this paper, the number of scroll turns, the scroll thickness and the suction volume were fixed at constant values, and the mechanical efficiency of a compact scroll compressor was calculated for various combinations of the scroll height and the involute base circle radius. The shaft loads at the crankshaft and the crank pin increase due to gas forces as the scroll height becomes larger, while inertia forces as the scroll height becomes smaller. Moreover, the thrust loads at the thrust bearing increase due to
gas forces as the scroll height becomes smaller. Because of these load characteristics, there is definitely an optimum combination of scroll height and involute base circle radius which results in a minimum power loss due to mechanical friction, that is, a maximum mechanical efficiency. The rotating speed fluctuation ratio of the crankshaft is less affected by the combination of these major parameters.

To extend the present study to more practical cases, similar calculations need to be performed for a variety of other combinations of the major parameters of the scroll compressor. Any further reduction in scroll compressor size would reduce the mechanical efficiency. Especially from the standpoint of designing a more compact scroll compressor, therefore, the optimum combination of major parameters is especially important.

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REFERENCES


### APPENDIX: NOMENCLATURE

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<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>B</td>
<td>scroll height (m)</td>
</tr>
<tr>
<td>( f_{11}, f_{12} )</td>
<td>frictional forces at thrust bearing (N)</td>
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<tr>
<td>( F_l, F_r )</td>
<td>tangential and radial gas forces acting on orbiting scroll center (N)</td>
</tr>
<tr>
<td>( F_{t11}, F_{t12} )</td>
<td>constraint forces at thrust bearing (N)</td>
</tr>
<tr>
<td>( I_0 )</td>
<td>moment of inertia of crankshaft (kgm^2)</td>
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<tr>
<td>( L_Q, L_S )</td>
<td>frictional torques at crankshaft and crank pin (N.m)</td>
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<tr>
<td>( m_o )</td>
<td>mass of Oldham ring (kg)</td>
</tr>
<tr>
<td>( m_s )</td>
<td>mass of orbiting scroll (kg)</td>
</tr>
<tr>
<td>( N )</td>
<td>motor torque (Nm)</td>
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<tr>
<td>( r_b )</td>
<td>involute base circle radius (m)</td>
</tr>
<tr>
<td>( r_o )</td>
<td>orbiting radius (m)</td>
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<tr>
<td>( S_x, S_y )</td>
<td>constraint force components at crank pin (N)</td>
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<tr>
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<td>mechanical efficiency (%)</td>
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<td>orbiting angle (degree)</td>
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