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A STUDY ON DYNAMIC BEHAVIOR OF A SCROLL COMPRESSOR

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ABSTRACT

This study discusses in relation to compressor dynamics the suitability of a scroll machine for small-sized refrigerant compressors. For this purpose the calculated results on the crankshaft load fluctuation, the crankshaft rotary behavior, the unbalanced inertia forces, the compressor vibrations and the mechanical efficiency of a scroll-type compressor are presented initially. Secondly, the calculated results are compared with those of four different types of compressors: an one-rolling-piston rotary type, a two-rolling-pistons two-blades rotary type, an one-rolling-piston two-blades rotary type and a single-cylinder reciprocating type.

SYMBOLS

$B$ = height of scroll
$f_{11}, f_{12}$ = frictional force on thrust-bearing plane
$f_r$ = tangential gas force
$f_{t1}, f_{t2}$ = thrust force on thrust-bearing plane
$f_{x1}, f_{x2}, f_{y1}, f_{y2}$ = frictional force on Oldam ring
$I_0$ = crankshaft moment of inertia
$I_x, I_y, I_z$ = compressor moment of inertia
$I_0$ = height of Oldam-slot center
$I_1$ = height of crank-journal center
$I_2$ = height of crank-pin center
$m_e$ = height of orbiting scroll's gravity center
$L_0$ = height of concentred mass $m_e$ from thrust-bearing plane
$L_0$ = frictional torque at crank-journal
$L_3$ = frictional torque at crank-pin
Since air conditioners are often operated in quiet surroundings, the vibration and noise generation of the compressor during steady operation must be minimized. About twelve years ago, the conventional compressors for air conditioners were the reciprocating type. However, recently the one-rolling-piston rotary compressors are used in most low capacity air conditioners. The major reason the reciprocating type compressors were replaced by the rotary type was because of its high volumetric and mechanical efficiency, coupled with its compact, lightweight design. The vibration and noise generation of the rotary compressor has not been so reduced as to satisfy the strict requirement of users sufficiently. The major factor which causes the rotary compressor vibrations is the instantaneous load fluctuation during steady operation. Various methods can be used to overcome this. One is to improve the mechanism of the one-rolling-piston rotary compressor: for instance, to develop a two-rolling-pistons two-blades rotary compressor or an one-rolling-piston two-blades rotary compressor. The other is to design a new mechanism, for instance, a scroll compressor[1-4]. The scroll compressor has many compression chambers which are simultaneously compressed at comparatively low speed.
Therefore, the compressor vibration problems in air conditioners will be considerably alleviated by introducing the scroll compressor.

An important matter is to discover to what degree the introduction of the scroll compressor can alleviate the compressor vibration problems. For this purpose, this study presents an analytical method to evaluate the dynamic behavior of the scroll compressor and the calculated results for a scroll compressor of comparatively small capacity. The calculated results are summarized on the gas-torque fluctuation, the crankshaft rotary behavior (speed fluctuation rate and rotary acceleration), the unbalanced inertia forces, the compressor vibrations and the mechanical efficiency. In discussing the suitability of a scroll machine for a small-sized refrigerant compressor, in relation to compressor dynamics, the calculated results for the small scroll compressor are compared with those of the following four types of compressor of similar capacity. The first type is a "1-rotary," namely, an one-rolling-piston rotary, the second a "2-rotary," namely, a two-rolling-pistons two-blades rotary, the third a "2-blades rotary," namely, an one-rolling-piston two-blades rotary and the fourth a "1-recipro.," namely, a single-cylinder reciprocating type.

DESCRIPTION OF SCROLL COMPRESSOR

Fig. 1 shows the orbiting-scroll configuration which comprises the two involutes defined by

\[ \overline{ST}_I = r_b(\phi - \phi_{10}), \quad \overline{ST}_O = r_b(\phi - \phi_{00}) \]  

in which \( \phi \) is the counterclockwise involute-angle from the horizontal \( x_m \)-axis. The \( x_m-y_m \) coordinate has the origin \( O_m \) at the center of the basic circle of radius \( r_b \).
and moves with the orbiting scroll. The orbiting scroll is mated with the identical scroll as shown in Fig. 2. The non-shadowed scroll is fixed. The basic-circle center \( O \) of the fixed scroll is the origin of the static x-y coordinate. The two scrolls touch and form a series of crescent-shaped chamber between two cover-plates, each fixed to the scrolls. The contact points \( T_{ik} \) and \( T_{Ok} \) \((k=0,1,...)\) appear respectively on the two tangents common to the two basic circles. The subscript \( i \) and \( o \) show respectively the inner and outer contact points of the orbiting scroll. The involute angles \( \phi_i(k) \) and \( \phi_o(k) \) of these contact points are given by

\[
\phi_i(k) = 2\pi (k + \delta) + 3\pi/2 - \theta, \quad \phi_o(k) = 2\pi (k + \delta) + \pi/2 - \theta
\]

\((k=0,1,...)\) (2)

where \( \delta = 0 \) when \( 0 < \theta < \pi/2 - \phi_{o5} \),

\( \delta = 1 \) when \( \pi/2 - \phi_{o5} < \theta < 2\pi \)

\( \theta \) shows the clockwise orbiting angle from the x-axis.

The shadowed scroll orbits around the fixed scroll. The orbiting radius \( r_o \) is determined by

\[
r_o = r_b \pi - t
\]

(3)

where \( t \) is the scroll thickness given by \( r_b (\phi_{o} - \phi_{oo}) \).

The orbiting scroll is driven by a motor through an Oldam coupling. The refrigerant of pressure \( p_o \) is trapped at the periphery of the scrolls and compressed as the crescent-shaped chambers move toward the center. The compressed refrigerant of pressure \( p_o \) is exhausted through the outlet at the fixed-scroll center.

Gas forces acting on the orbiting scroll are reduced to the tangential force \( F_t \) which acts against the clockwise orbiting motion, the radial force \( F_r \) and the clockwise moment \( M_o \) which act on the orbiting-scroll center \( O_m \) (see Fig. 2). They can be obtained by calculating the following forms.

\[
F_t = Br_b \left[ \int_{\phi_j(0)}^{\phi_j(K)} p_j (\phi - \phi_{oo}) \cos(\theta + \phi) \, d\phi \right]
\]

\[
F_t = Br_b \left[ \int_{\phi_j(0)}^{\phi_j(K)} p_j (\phi - \phi_{oo}) \sin(\theta + \phi) \, d\phi \right]
\]

\[
M_o = Br_b^2 \left[ \int_{\phi_j(0)}^{\phi_j(K)} p_j (\phi - \phi_{oo}) \, d\phi \right]
\]

\[
+ tB \left[ -(r_b (\phi_j(0) - \phi_{oo}) - t/2)p_o + \{r_b (\phi_j(K) - \phi_{oo}) + t/2)p_s \right]
\]

where \( B \) represents the height of the scroll. \( K \) is the largest number of \( k \). The subscript \( j \) of the cylinder pressure changes depending on the integral domain, as follows:
Related to the integral along the outer involute,
\[ j = k + 1 \text{ when } \phi_0(k) < \phi < \phi_0(k+1) \quad (k=0,1,\ldots,K-1) \]
\[ j = K + 1 \text{ when } \phi_0(K) < \phi < \phi_0(K) + \pi \] (7)

Related to the integral along the inner involute,
\[ j = 0 \text{ when } \phi_1(0) - \pi < \phi < \phi_1(0) \]
\[ j = k + 1 \text{ when } \phi_1(k) < \phi < \phi_1(k+1) \quad (k=0,1,\ldots,K-1) \] (8)

**FUNDAMENTAL EQUATIONS OF MOTION OF MOVING COMPONENTS**

**Crankshaft System**

A crankshaft model is shown in Fig. 3 where \( m \) represents the total mass of the crank-arm, the crank-pin and the balancers, which is concentrated at the gravity point \((x_*, y_*)\) defined by

\[ x_* = r_0 \cos \theta, \quad y_* = r_0 \sin \theta \] (9)

Consider that the motor drive torque \( N \) acting around the crankshaft induces the reaction forces \( Q_x, Q_y \) on the crankshaft and the reaction forces \( S_x, S_y \) on the crank-pin. Hence, the crankshaft motion in the \( x \) and \( y \) directions is subjected to the following forms:

\[ -m_0 \ddot{x}_e - Q_x + S_x = 0, \quad -m_0 \ddot{y}_e + Q_y - S_y = 0 \] (10)

Representing the moment of inertia of the crankshaft system by \( I_0 \), the equation of rotary motion of the
crankshaft is given by

\[- I_0 \ddot{\theta} = - S_x r_0 \sin \theta - S_y r_0 \cos \theta + N - L_a - L_s = 0 \quad (11)\]

\[L_a = \mu_s \sqrt{Q_x^2 + Q_y^2} r_0, \quad L_s = \mu_s \sqrt{S_x^2 + S_y^2} r_0 \quad (12)\]

Representing the reaction moment on the crankshaft from the crank-journal by \(M_{Q} \) and \(M_{Q_y} \), the equilibrium of moment around the \(x \) and \(y \) axes is given by

\[M_{Q} = - S_x (L_a - L_s) - (-m_x \ddot{x}) (L_a - L_s) = 0, \quad (13)\]

\[M_{Q_y} = - S_y (L_a - L_s) + (-m_y \ddot{y}) (L_a - L_s) = 0 \]

**Orbiting Scroll**

Fig. 4 shows all forces and moments acting on the orbiting scroll. The cross-sectional view along the \(x_m - z_m \) plane is also shown to evaluate the thrust forces. It is so designed that the intermediate pressure between the suction and discharge pressure acts on the outer surface of the orbiting cover-plate and hence the orbiting scroll is pressed up toward the fixed cover-plate. The resultant upward force is shown by \(F_0 \) acting on the center \(O_m \). Therefore, the contact plane of the orbiting and fixed cover-plates forms the thrust bearing surface of radius \(R_{th} \). Consider that the previously defined \(x-y \) and \(x_m-y_m \) planes coincide with this thrust bearing plane.

The reaction forces produced by the crank-pin on the scroll center \(O_m \) are represented by \(S_x \) and \(S_y \) which drive the orbiting scroll. \(F_t \), \(F_r \) and \(M_o \) are the gas forces and moment given by (4) to (6). \(O_1 \) and \(O_2 \) show the reaction forces of the Oldam ring of radius \(R_{or} \). \(f_{v_1} \) and \(f_{v_2} \) are the frictional forces between the Oldam ring and the scroll which are given by the following forms assuming Coulomb’s law of friction.

\[f_{v_1} = \mu_s |O_1|, \quad f_{v_2} = \mu_s |O_2| \quad (14)\]

The resultant thrust force \(F_t \) due to the gas pressure in the cylinders acts on the center of the segment \(O_0 O_m \). The inertia force of the orbiting scroll is represented by \((-m_x \ddot{x}) \) and \((-m_y \ddot{y}) \), where \(x \) and \(y \) are defined by

\[x = r_0 \cos \theta, \quad y = r_0 \sin \theta \quad (15)\]

There are forces acting on the orbiting scroll which would tend to force it from the horizontal plane. In order to evaluate the constraint forces at the thrust bearing surface, the moment \(M_m \) overturning the orbiting scroll and its counter-clockwise angle \(\eta \) from the \(x_m \) axis are derived initially.

\[M_m = \sqrt{M_{x_m}^2 + M_{y_m}^2}, \quad \eta = \tan^{-1}(M_{y_m}/M_{x_m}) \quad (16)\]

where \(M_{x_m} \) and \(M_{y_m} \) representing the moment components around the \(x_m \) and \(y_m \) axes are given by

\[M_{x_m} = - F_y \cdot \frac{r_y}{2} - F \cdot \cos \theta \cdot \frac{B}{2} + F \cdot \sin \theta \cdot \frac{B}{2} + (-m_y \ddot{y})L_s + (f_{v_1} + f_{v_2})L_0 - S_y L_s, \quad (17-1)\]

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\[ M_{mm} = -F_t \cdot x_r/2 + F_t \sin \theta \cdot B/2 + F_r \cos \theta \cdot B/2 \]
\[ + (-m_0 x_r) \cdot \ddot{x} + (O_1 - O_2) \cdot \dot{\theta} + S_x \cdot \dot{\theta} \]  

This overturning moment \( M_{mm} \) must be supported by the thrust bearing surface. In order to simplify the analysis, assume that \( M_{mm} \) is supported at the two representative points on both the thrust-bearing circle and the line perpendicular to \( M_{mm} \) passing \( O_m \). Under this assumption, the reaction thrust-forces \( F_{t1} \) and \( F_{t2} \) at these points are given by
\[
F_{t1} = \frac{-M_{mm}/R_{th} - F_t + F_0}{2}, \quad F_{t2} = \frac{(M_{mm}/R_{th} - F_t + F_0)}{2} 
\]

Therefore the frictional forces \( f_{t1} \) and \( f_{t2} \) at these points are given by the following forms, assuming Coulomb's law of friction.
\[
f_{t1} = \mu_1 |F_{t1}|, \quad f_{t2} = \mu_2 |F_{t2}| 
\]

The direction of these frictional forces is perpendicular to the segment \( O_0M \).

Since the forces acting on the orbiting scroll can be defined, the equation of motion can be derived. The equations of motion in the \( x \) and \( y \) directions are given by
\[
(-m_0 x_r) - S_x + (F_t + f_{t1} + f_{t2}) \sin \theta + F_r \cos \theta - O_1 - O_2 = 0, 
\]
\[
(-m_0 y_r) + S_y + (-F_t - f_{t1} - f_{t2}) \cos \theta + F_r \sin \theta - f_{y1} - f_{y2} = 0 
\]

The moment around the \( z \) axis is balanced by the reaction moment due to the Oldam ring as follows:
\[
M_0 - L_0 - (f_{t1} - f_{t2}) R_{th} \sin (\eta + \theta) = O_1 (R_or + y_r) + O_2 (R_or - y_r) 
\]

Oldam ring

A model of the Oldam ring which performs reciprocating motion along the \( x \) direction is shown in Fig. 5. The mass \( m_0 \) of the Oldam ring is concentrated at its center \( (x_r, 0) \). \( O_1 \) and \( O_2 \) are the reaction forces of the orbiting scroll \( f_{y1} \) and \( f_{y2} \) are the frictional forces given by (14). \( T_1 \) and \( T_2 \) are the reaction forces of the guide slot fixed on the compressor body \( f_{x1} \) and \( f_{x2} \) represent the frictional forces between the guide slot and the Oldam ring, which are given by
\[
f_{x1} = \mu_1 |T_1|, \quad f_{x2} = \mu_2 |T_2| 
\]

By defining the forces acting on the Oldam ring, the equation of motion in the \( x \) direction can be given by
\[
(-m_0 x_r) + O_1 - O_2 + f_{x1} + f_{x2} = 0 
\]

The equilibrium equations of forces in the \( y \) axis and of moment around the Oldam ring center are respectively prescribed by
\[
-T_1 + T_2 + f_{y1} + f_{y2} = 0, \quad O_1 + O_2 - T_1 - T_2 = 0 
\]
EQUATION OF ROTARY MOTION OF CRANKSHAFT

Rearranging the fundamental equations of motion of the moving components, all constraint forces and moments can be derived as a function of the orbiting angle \( \theta \). Substituting the derived constraint-forces \( S_x \) and \( S_y \) into (11), the equation of rotary motion of the crankshaft can be rewritten in the following form.

\[
(I_0 + m_2 r_0^2 + m_3 r_0^2 \sin^2 \theta) \dot{\theta} + m_3 r_0^2 \sin \theta \cdot \cos \theta \cdot \dot{\theta}^2 = N - (F_x r_0 + L_0) + L_0 \left( f_{x_1} + f_{x_2}\right) r_0 \sin \theta + (f_{y_1} + f_{y_2}) r_0 \cos \theta + (f_{z_1} + f_{z_2}) r_0
\]

(25)

The left side represents the moments of inertia. The first moment on the right side is the drive torque and the second represents the load torques. The first load torque is due to gas compression and the others are due to mechanical frictions on each pair of machine components.

EXCITING FORCES AND MOMENTS, AND COMpressor VIBRATIONS

In order to discuss the compressor vibration, the exciting forces and moments of the compressor body should be derived initially. Since the forces and moments acting on the moving components have been indicated in the previous section, it is not difficult to calculate the forces and moments acting on the compressor body. The summation of these forces and moments gives the following expressions, namely, unbalanced forces and moments of inertia.

\[
\begin{align*}
F_x &= -\left( m_2 r_0^2 + m_3 + m_0 \right) x_r - \left( m_0/2 \right) x_r \\
F_y &= -\left( m_2 r_0^2 - m_3 \right) y_r - \left( m_0/2 \right) y_r \\
M_x &= \left( m_2 r_0^2 L_0 + m_3 L_0 - m_0 L_0 \right) y_r + \left( m_0/2 \right) L_0 y_r \\
M_y &= \left( -m_2 r_0^2 L_0 - m_3 L_0 + m_0 L_0 \right) x_r - \left( m_0/2 \right) L_0 x_r \\
M_z &= -\left( I_0 + m_2 r_0^2 \right) \ddot{\theta}
\end{align*}
\]

(26)

in which the best static balancing and the best dynamic balancing have been achieved by choosing optimum values of \( m_2 \), \( r_0 \) and \( I_0 \). \( F_x \), \( F_y \) and \( F_z \) are the forces acting on the point 0. \( M_x \), \( M_y \) and \( M_z \) are the moments around the \( x \), \( y \) and \( z \) axes respectively.

Assume that the natural vibration frequencies of the compressor elastically suspended by coiled springs are fairly small compared with the mean rotation speed of the crankshaft. Thus, the compressor vibrations can be approximately calculated from the following simple expression.

\[
[X] = [M]^{-1} [E][F]
\]

(27)

where

\[
\begin{bmatrix}
X \mid X_0 & [M] = \begin{bmatrix}
W & \cdots & 0 \\
0 & \cdots & 0
\end{bmatrix}, \ [E] = \begin{bmatrix}
1 & \cdots & 0 \\
0 & \cdots & 0
\end{bmatrix}, \ [F] = \begin{bmatrix}
F_x \\
F_y \\
M_x \\
M_y \\
M_z
\end{bmatrix}
\end{bmatrix}
\]

(28)

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SPECIFICATION OF SCROLL COMPRESSOR FOR NUMERICAL CALCULATION

Table 1 gives the specification of scroll configuration. Since the initial angles of the inner and outer involutes with a base circle of radius 0.2069 cm are 0° and -110.77° respectively, the scroll thickness is 0.4 cm and hence the orbiting radius is 0.25 cm from (3). The scrolling number of the involutes is 2.5. Fig.6 shows the calculated area Sj(j=0,1,2) of the identical crescent-shaped compression-chambers. This figure shows the change in area for about two revolutions of the crankshaft. As θ increases, the suction area of 8.796 cm trapped at the periphery of the scrolls decreases linearly until the two identical compression chambers are connected at θ = 117.6°. Assume that the gas-compression process is subjected to the adiabatic change of specific heat ratio 1.32. Thus, the compression-chamber gas pressure p1 shown in Fig.7 can be calculated from Fig.6. Consider that the scroll compressor has no valve at the discharge port. Hence the discharge pressure is approximately determined by the compression-chamber pressure just when the two identical chambers are connected at θ = 117.16°. Thus, the gas pressure increases from the given suction pressure 0.617 MPa to the discharge 2.17 MPa. Finally consider that the intermediate pressure pmd pressing the orbiting scroll up toward the fixed scroll is 0.882 MPa.

Table 2 shows the major mechanical constants of a small-sized scroll compressor presently being developed. Assume that the friction coefficients at each pair of moving components take the same value of 0.027 which was determined on the basis of the previous studies [5,6] carried out using a 1-rotary compressor. The scroll

![Fig.6 Crescent-shaped area Sj](image)

![Fig.7 Gas pressure p1](image)
Tab. 2 Mechanical constants

of scroll compressor

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$I_0$</td>
<td>0.0422 N·cm·s²</td>
</tr>
<tr>
<td>$I_x$</td>
<td>3.519 N·cm·s²</td>
</tr>
<tr>
<td>$I_y$</td>
<td>3.904 N·cm·s²</td>
</tr>
<tr>
<td>$I_z$</td>
<td>0.996 N·cm·s²</td>
</tr>
<tr>
<td>$J_o$</td>
<td>-0.4003 cm</td>
</tr>
<tr>
<td>$J_x$</td>
<td>0.5 cm</td>
</tr>
<tr>
<td>$J_y$</td>
<td>0.7 cm</td>
</tr>
<tr>
<td>$J_z$</td>
<td>4.8 cm</td>
</tr>
<tr>
<td>$M$</td>
<td>8.7 Kg</td>
</tr>
<tr>
<td>$m_o$</td>
<td>0.748 Kg</td>
</tr>
<tr>
<td>$m_e$</td>
<td>0.051 Kg</td>
</tr>
</tbody>
</table>

$m_o = 0.277$ Kg
$r_o = -0.101$ cm
$K_o = 2.85$ cm
$r_e = 1.0$ cm
$R_{th} = 4.1$ cm
$v_e = 10.26$ cm³
$x_o = 0$ cm
$y_o = 0$ cm
$z_o = 6.54$ cm

$\mu = 0.027$

Fig. 8 Motor torque curve

The compressor is driven by the motor torque shown in Fig. 8. The synchronous speed is 3600 rpm.

COMPARISON OF OTHER REFRIGERANT COMPRESSOR TYPES

For clarity of understanding, the calculated results on the scroll-compressor's dynamic behavior may be compared with those of the other types of refrigerant compressor. Comparisons of a 1-rotary, a 2-rotary, a 2-blade rotary and a 1-recipro type compressors are made. Of course, they must be of the same capacity as the scroll compressor, that is, their suction-gas volume should be about 10.26cm³. For comparison of the compressor vibrations and the mechanical efficiency, it should be, moreover, assumed that the major mechanical constants, for instance, the rotating crankshaft moment of inertia, the compressor mass, the compressor moment of inertia, the compressor-gravity coordinate and the friction coefficients take the same values as those in Table 2 of the scroll compressor, respectively.

Please refer to the previous studies [5,6] for the 1-rotary compressor of the same capacity and mechanical constants as the scroll compressor. The 1-rotary compressor is composed of a rolling piston, a blade, a cylinder and a crankshaft. The diameter, the mass and the eccentricity of the rolling piston are 3.24 mm, 0.0741 Kg and 3.3 mm respectively. The blade mass is 0.0113 Kg. The diameter and the depth of the cylinder are 3.9 cm and 2.8 cm respectively. On the basis of this 1-rotary compressor, the 2-rotary and the 2-blade rotary compressors can be developed. Assume that the cylinder spaces are equally divided by setting a thin partition plate at the center of the cylinder depth and the eccentricity directions of the divided two rolling-
pistons are opposite. Hence the two blades perform reciprocating motion in opposite directions. This type of compressor possessing two rolling-pistons and two blades is called "2-rotary." A cross-sectional view of the 2-blade rotary is shown in Fig. 9. The difference from the 1-rotary is that the 2-blade rotary has one extra blade. The two blades perform the same reciprocating motion. Please refer to the previous studies [7-11] for the compressor dynamics on the conventional 1-reciprocating type. The 1-reciproc. compressor of the same capacity has a piston of diameter 2.48cm and mass 0.16Kg, and a 4.24cm long connecting-rod of mass 0.053Kg. The rotating crank radius is 1.06cm.

CALCULATED DYNAMIC BEHAVIOR OF SCROLL COMPRESSOR COMPARED WITH OTHER TYPES OF COMPRESSOR

Substituting the compression-chamber gas pressure shown in Fig. 7 into (4) to (6), the gas forces F_t, F_r and moment M_o acting on the orbiting scroll can be calculated. Hence, the gas torque F_r on the right side of (25) is obtained as shown in Fig. 10 which includes the calculated gas torques of the other four types. The cylinder pressure of the other four types was also calculated assuming the adiabatic change of specific heat ratio 1.32. The same pressure 0.617MPa as the scroll compressor was chosen for the suction pressure. The average gas-load for the other four types should be, of course, the same as the scroll compressor. Therefore the discharge pressure of the other four types was so adjusted that the gas compression-work per crankshaft-revolution takes the same value 9.30Nm as the scroll compressor. It is important to find the fluctuation magnitudes of the gas torques from Fig. 10. They are shown on the second line of Table 3. The standard decibel values in parentheses are those of the 1-rotary.
compressor which is, at present, most commonly used for air-conditioning appliances. The gas torque fluctuation of the scroll compressor is the lowest -22.8 dB. This level is lower by 12.1 dB compared with the 2-rotary which is the second lowest. This is an excellent characteristic of the scroll compressor, which is results from the fact that many compression chambers are compressed simultaneously. The third lowest is the 2-blade rotary, the forth is the 1-rotary and the worst is the 1-recipro. type.

The crankshaft motion of the scroll compressor can be numerically calculated from (25), applying a repeated calculation method. The calculated results of the average crankshaft-speed and the speed fluctuation ratio are shown in Table 3. The crankshaft-speed fluctuation ratio of the scroll compressor is very small. Its value of 0.50 is about half that of the 2-rotary type. The calculated $\dot{\theta}$ is shown in Fig. 11 which includes those of the other four types. Using the calculated $\dot{\theta}$, the exciting moment $M_z$ is calculated from the sixth form of (26) and thus (27) gives the rotation acceleration $\ddot{\theta}$ of the compressor body which is shown in Fig. 12. The peak to peak values of $\dot{\theta}$ and $\ddot{\theta}$ are summarized in Table 3. From these results it is confirmed that the fluctuation levels of $\dot{\theta}$ and $\ddot{\theta}$ are smallest for the scroll compressor and increase by a similar rate as the gas-torque fluctuation becomes larger. It may be concluded

![Fig.11 Rotatory acceleration of crankshaft](image1)

![Fig.12 Compressor-body acceleration around crankshaft](image2)
that the scroll compressor is significantly improved in $\dot{\theta}$ and $\ddot{\theta}$, compared with the 1-rotary type and, even compared with the 2-rotary or the 2-blade rotary. These fluctuation levels are lower by 11dB to 16dB.

Using the calculated $\dot{\theta}$, the exciting forces $F_x$, $F_y$ and moments $M_x$, $M_y$ are calculated from (26) and thus (27) gives the compressor translation-accelerations $\dot{x}_0$, and $\dot{y}_0$ and rotation accelerations $\ddot{\theta}_x$, $\ddot{\theta}_y$. The translation and rotation accelerations are shown by trajectories of vectors $(\dot{x}_0 + \dot{y}_0)$ in Fig.13 and vectors $(\ddot{\theta}_x + \ddot{\theta}_y)$ in Fig.14. The x and y axes correspond to the axes in Fig. 2 for the scroll type and the axes in Fig. 9 for the rotary types. For the 1-recipro., the piston center line coincides with the x axis. These figures represent the degree of static and dynamic balancing and also the crankshaft-speed fluctuation. The vector trajectories of the 1-rotary and the 1-recipro. are comparable to a rough triangle, since their speed fluctuation ratios are fairly large. As the speed fluctuation ratio decreases, like the scroll, and the 2-blade rotary types, the vector trajectories approach a circle. In the case of the 2-rotary type, the static balancing is achieved ideally, i.e. the static unbalanced forces of inertia are reduced to zero. The vector trajectory of a short segment in Fig. 13 results from the second-order unbalanced forces of inertia, and these mainly cause the 2-rotary type's vector trajectory shown in Fig. 14. The well dynamic balancing of the 2-rotary type can also be achieved since the acceleration amplitude caused by the first-order unbalanced moments of inertia is about 0.45 rad/s. The maximum values of each vector-trajectory are summarized in Table 3. The scroll compressor is worse in both static and dynamic balancing even compared
with the 1-rotary or the 2-blade rotary. The major reason is that the scroll compressor has the Oldam coupling ring of larger mass which is about 4.5 times the blade mass of the 1-rotary.

Next consider which type of compressor possesses the most well-balanced characteristic in relation to compressor vibrations. It is evident that the most-commonly used 1-rotary type is fairly good in relation to vibrations, compared with the 1-recipro type. However, the vibration $\Theta_v (=507 \text{ rad/s}^2)$ around the rotating crankshaft is much larger than the others $\Theta_s + \Theta_r$ ($=5.12 \text{ rad/s}^2$). The 2-rotary type is good in relation to vibration $\Theta_v$, but it may not be satisfactory enough for the user's exact requirements. In the case of the scroll type, the vibration $\Theta_v$ is greatly reduced to 35.5 rad/s$^2$ which is of the same order as $\Theta_s + \Theta_r$. From this point of view, it may be concluded that the scroll type possesses a revolutionary well-balanced characteristic in relation to compressor vibrations, though both the static and dynamic balancing is never better than the rotary types.

The calculated values of mechanical efficiency are shown on the last line of Table 3. Compared with the high efficiency of the rotary types, the mechanical efficiency of the scroll type is lower by 10.8% to 12%. This difference results from the following. The first is that the lubrication at the pair of crank-pin and piston is under fluid friction for the rotary types and on the other hand under Coulomb friction for the scroll type. The second is that the scroll type has an extra thrust-bearing supporting a fairly large thrust force, which causes extra energy consumption due to mechanical friction.

<table>
<thead>
<tr>
<th>compressor type</th>
<th>scroll</th>
<th>1-rotary</th>
<th>2-rotary</th>
<th>2-blades</th>
<th>1-recipro</th>
</tr>
</thead>
<tbody>
<tr>
<td>p-p value of gas torque (Nm)</td>
<td>0.31</td>
<td>4.30</td>
<td>1.28</td>
<td>2.07</td>
<td>8.38</td>
</tr>
<tr>
<td>average crankshaft-speed (rpm)</td>
<td>3385</td>
<td>3404</td>
<td>3422</td>
<td>3429</td>
<td>3469</td>
</tr>
<tr>
<td>speed fluctuation ratio (%)</td>
<td>0.50</td>
<td>8.24</td>
<td>1.08</td>
<td>1.78</td>
<td>15.0</td>
</tr>
<tr>
<td>p-p value of $\delta$ (rad/s$^2$)</td>
<td>794</td>
<td>12200</td>
<td>3100</td>
<td>5070</td>
<td>27500</td>
</tr>
<tr>
<td>p-p value of $\Theta_v$ (rad/s$^2$)</td>
<td>35.5</td>
<td>507</td>
<td>132</td>
<td>218</td>
<td>1280</td>
</tr>
<tr>
<td>max. of $\sqrt{\delta_1^2 + \delta_2^2}$ (m/s)</td>
<td>0.94</td>
<td>0.35</td>
<td>0.09</td>
<td>0.37</td>
<td>19.8</td>
</tr>
<tr>
<td>max. of $(\Theta_v + \Theta_r)$ (rad/s$^2$)</td>
<td>14.9</td>
<td>5.12</td>
<td>2.62</td>
<td>5.73</td>
<td>425</td>
</tr>
<tr>
<td>mechanical efficiency (%)</td>
<td>80.0</td>
<td>90.8</td>
<td>91.8</td>
<td>92.0</td>
<td>77.9</td>
</tr>
</tbody>
</table>
CONCLUSION

An analytical method to evaluate the scroll compressor's dynamic behavior was established. Calculation was performed for the comparatively small-sized scroll compressor of suction-gas volume $10.26\, \text{cm}^3$ and the calculated results were compared with those of the 1-rotary, the 2-rotary, the 2-blade rotary and the 1-recipro. type compressors of the same capacity and similar major mechanical constants. We may conclude from this study as follows:

(1) The introduction of the scroll compressor can reduce the main vibration component, that is, the vibration around the rotating crankshaft by $23.1\, \text{dB}$ compared with the 1-rotary type, $11.8\, \text{dB}$ compared with the 2-rotary type, $16.1\, \text{dB}$ compared with the 2-blade rotary type and $30.8\, \text{dB}$ compared with the 1-recipro. type. Therefore it may be concluded that the scroll compressor can significantly alleviate the vibration problems commonly found in the other types of compressors.

(2) The mechanical efficiency of the scroll compressor is lower by $10.8$ to $12.8\%$ than the rotary type compressors. To overcome this bad characteristic, a special device for lubrication is desirable for the scroll compressors.

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