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by

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ABSTRACT

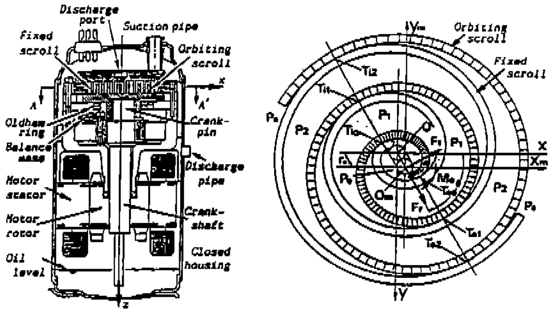
This study presents the calculated mechanical efficiency of a variable speed compact scroll compressor for a wide range of synchronous-speed on an induction motor. The compression mechanism in the scroll compressor is quite different from that in the rolling-piston rotary compressor, and thus it is quite interesting to examine the difference in mechanical efficiency which is caused by the compression mechanism itself. From this point of view, the mechanical efficiency for the scroll compressor was calculated by using the dynamic analysis introduced in our previous studies, and finally the calculated results were compared with those for a rolling-piston rotary compressor with the same cooling capacity. This study concludes that a scroll compressor with optimum dimensions has the same high level of mechanical efficiency as a rolling-piston rotary compressor with the same cooling capacity, and that as the operating speed increases, the mechanical efficiency of the scroll compressor does not decrease as rapidly as that of the rolling-piston rotary compressor.

INTRODUCTION

Since air conditioners must often be operated in quiet surroundings, the vibration and noise generation of the compressor during steady operation must be minimized. Recently the rolling-piston rotary compressor has been used in most low capacity air conditioners, because of its high volumetric and mechanical efficiency, coupled with its compact, lightweight design. However, the vibration and noise generation of the rolling-piston rotary compressor has not yet been sufficiently reduced so as to satisfy the stringent requirements of users. The main cause of rolling-piston rotary compressor vibrations is the instantaneous load fluctuation during steady operation.

The scroll compressor is an alternative to the rolling-piston compressor with its noise problems due to load fluctuation. Analyses of the dynamic behavior and mechanical vibrations of both a constant speed scroll compressor [see Refs. 1 and 2] and a variable speed scroll compressor for refrigeration-capacity control [3] have been performed. These analyses showed that the scroll compressor has superior dynamic behavior in terms of minimal fluctuation in speed and low levels of mechanical vibration which are far below those of a rolling-piston rotary compressor with the same capacity, across the entire range of operating speeds.

Another significant characteristic of the scroll compressor is concerned with the mechanical efficiency. The mechanical efficiency of various capacity scroll compressors driven by constant speed induction motors was presented in a previous study [4]. In this study, the mechanical efficiency of a compact scroll compressor, which is operated over a wide range of synchronous-speed on an induction motor, is calculated using the same dynamic analysis presented in the previous studies [1-3]. The calculated results are compared with those for a rolling-piston rotary compressor with the same cooling capacity. Thus, the scroll compressor, which possesses the superior characteristic of having only low level vibrations over a wide range of operation, is examined from an energy savings standpoint, and it is shown that the scroll compressor has a high mechanical efficiency which is nearly as high as that of the rolling-piston rotary compressor.



(a) Vertical cross-sectional view (b) A-A' cross-sectional view

Fig.1 Scroll compressor

MECHANICAL EFFICIENCY

In order to calculate the mechanical efficiency, the frictional force at each pair of compressor elements must first be calculated. Prior to this, of course, the constraint force at each pair must be calculated. The constraint forces are affected by the gas forces in the cylinder and also by the inertial force of each moving element, and thus they should be derived from the equations of motion of the moving elements of the compressor [1-3]. Based on the derived equations of motion, the equation of rotating motion of the crankshaft can be obtained. Multiplying the crankshaft rotating velocity by the derived equation of motion of the crankshaft and integrating it with respect to the time, an energy equilibrium equation can be obtained. From the equation of motion of the crankshaft and the energy equilibrium equation, the periodic dynamic rotation of the crankshaft can be determined numerically. Thus, the constraint forces and the frictional forces which include the effects of the inertial forces can be calculated precisely. Finally, the shaft input power, the gas compression power and the power losses due to friction can be calculated, leading to an understanding of the mechanical efficiency and its physical basis.

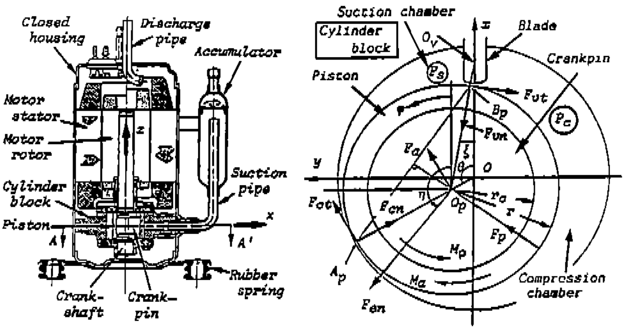
Scroll Compressor and Brief Description of the Dynamic Analysis

A vertical cross-sectional view of the scroll compressor is shown in Fig.1a. The compression device consisting of the fixed and orbiting scrolls is located in the upper portion of the closed housing. The shaded part is the orbiting scroll and the unshaded part the fixed scroll. This compression device is driven by an induction motor having a wide range of synchronous speeds. The induction motor is located in the lower portion of the closed housing. The horizontal A-A' cross-section of this compression device is shown in Fig.1b. The crankshaft center coincides with the center O of the involute basic circle of the unshaded fixed scroll. The center of the involute basic circle of the shaded orbiting scroll is represented by O_m. The distance between these two points represents the orbiting radius r_0, which is determined by the involute base circle radius and the scroll thickness [see Eq.(3) in Ref.2]. The static coordinates are represented by X-Y-Z with the origin at the point O. The shaded scroll orbits around the crankshaft center O. The clockwise orbiting angle from the X-axis is represented by θ.

The gas pressures (p_0, p_1, p_2, and p_s indicated in Fig.1b) in the crescent-shaped compression chambers act on the orbiting scroll. They can be represented by the tangential force F_t which acts against the clockwise orbiting motion, and the radial force F_r, and the clockwise moment M_0, which act on or around the orbiting-scroll center O_m. The loading of the gas forces F_t and F_r is carried by the crank journal and the clockwise moment M_0 by the constraint moment from the Oldham ring. The moment around the crankshaft center O which is given by $F_t r_0$ is the torque for gas compression. The motor torque N has to overcome this gas compression torque which fluctuates in the course of one revolution. Thus, the crankshaft undergoes a dynamic rotation which is subject to the following equation of motion [see Eq.(25) in Ref.2]:

$$(I_0+m_s r_0^2+m_0 r_0^2\sin^2\theta)\,\ddot{\theta}+m_0 r_0^2\sin^2\theta\cos\theta\cdot\dot{\theta}^2 = N-\{F_t r_0+L_Q+L_S+(f_{t1}+f_{t2})r_0\} \quad (1)$$

where the terms due to the frictional forces at the Oldham ring are omitted due to their negligible effect on the dynamic rotation of the crankshaft. The terms on the left represent the torques due to the inertial forces. I_0 is the moment of inertia of the whole



(a) Vertical cross-sectional view (b) A-A' cross-sectional view

Fig.2 Rolling-piston rotary compressor

crankshaft, m_s the mass of the orbiting scroll and m_o the mass of the Oldham ring. The terms on the right represent the torques due to the motor, the gas compression and the frictional forces. L_Q and L_S are the frictional torques at the crank journal and at the crankpin, respectively. f_{ti} $(i=1,2)$ represents the frictional forces at the thrust bearing. Based on Coulomb's law of friction, these frictional torques and forces can be given [see Eqs.(12) and (19) in Ref.2].

Multiplying the crankshaft rotating velocity $\dot{\theta}$ by Eq.(1) and integrating it over one revolution of the crankshaft, the following energy equilibrium equation can be obtained [4]:

$$W_{shaft} = W_{g\text{-}co} + (W_{c\text{-}jo} + W_{c\text{-}pn} + W_{t\text{-}br}) \tag{2}$$

where the shaft input energy W_{shaft} , the gas compression energy $W_{g\text{-}co}$, the frictional losses $W_{c\text{-}jo}$ at the crank journal, $W_{c\text{-}pn}$ at the crankpin and $W_{t\text{-}br}$ at the thrust bearing, are given by the following expressions, respectively:

$$W_{shaft} = \int_0^{t_0} N\dot{\theta}dT, \qquad W_{g\text{-}co} = \int_0^{t_0} F_t r_0 \dot{\theta}dT,$$

$$W_{c\text{-}jo} = \int_0^{t_0} L_Q\dot{\theta}dT, \quad W_{c\text{-}pn} = \int_0^{t_0} L_S\dot{\theta}dT, \quad W_{t\text{-}br} = \int_0^{t_0} (f_{t1}+f_{t2})r_0\dot{\theta}dT \tag{3}$$

Time is represented by T and the period for one revolution of the crankshaft by t_0. The energy equilibrium equation (1) means that the energy supplied by the motor is entirely consumed by the gas compression energy and the frictional losses at the crank journal, the crankpin and at the thrust bearing.

Rolling-Piston Rotary Compressor and Brief Description of the Dynamic Analysis

The structure of a rolling-piston rotary compressor is shown in Fig.2a. The compression device is located in the lower portion of the closed housing and the motor in the upper portion. The A-A' cross section of the compression device is shown in Fig.2b. The x-y coordinates represent the static coordinate system having its origin at the crankshaft center O. The x-axis is along the center line of the blade. The crankpin center O_p with eccentricity e rotates around the crankshaft center O. Its counterclockwise rotating angle from the x-axis is represented by θ. The crankshaft is driven by the motor torque N and the crankpin moves against the fluctuating gas force F_p caused by the gas pressure in the suction and compression chambers. Thus, the dynamic behavior of the rotating crankshaft is subject to the following equations of motion [see Ref.5 and Eq.(3) in Ref.(6)]:

$$\begin{aligned}&(I_c+m_pe^2)\,\ddot{\theta} - e\frac{\sin(\theta+\xi)}{\cos\xi}(m_v\ddot{x}_v+F_s)\\ &= N - [[e\sin\frac{\theta+\xi}{2}F_p - e\frac{\sin(\theta+\xi)}{\cos\xi}(F_{qx})] + M_p + M_s + e\frac{\sin(\theta+\xi)}{\cos\xi}(F_{gt1}+F_{gt2})\end{aligned} \tag{4}$$

where the terms due to the frictional forces at the blade tip, the thrust bearing and so on, are omitted due to their negligible effect on the dynamic rotation of the crankshaft. The terms on the left represent the torques due to the inertial forces and the spring force F_s , where x_v represents the displacement of the blade, I_c the moment of inertia of the whole crankshaft, m_p and I_p the mass and the moment of inertia of the rolling piston, and m_v the blade mass. The first term in the brackets on the right represents the torque necessary for gas compression, where F_{qx} represents the resultant gas force acting on the blade. The other terms in the brackets represent the frictional torque M_p at the crankpin, the torque M_s at the crank journal, and the torque due to the frictional forces F_{gti} (i=1,2) between the blade and the cylinder slot. In the rolling-piston rotary compressor, the frictional surface between the rolling piston and the crank pin is so well lubricated by oil that a fluid lubrication is formed. Thus, it is possible to express the frictional torque M_p on the basis of Sommerfeld's fluid lubrication theory of journal bearings [see Eq.(9) in Ref.6]. The crank journal, however, cannot be as well lubricated as the surface between the rolling piston and the crankpin and, furthermore, the frictional surfaces between the blade and the cylinder slot are also rather poorly lubricated by oil which is mixed into the refrigerant. Therefore, based on Coulomb's law of friction, the frictional torque M_s and the frictional forces F_{gti} (i=1,2) can be given [see Eqs.(10) and (11) in Ref.6].

Multiplying the crankshaft rotating velocity $\dot{\theta}$ by Eq.(4) and integrating it over one revolution of the crankshaft, the following energy equilibrium equation can be obtained [6]:

$$W_{shaft} = W_{g\text{-}co} + (W_{p\text{-}cp} + W_{c\text{-}jo} + W_{b\text{-}cy}) \tag{5}$$

where the shaft input energy W_{shaft} is given by the same expression as the first one in (3). The gas compression energy $W_{g\text{-}co}$, the energy losses due to friction $W_{p\text{-}cp}$ at the crankpin, $W_{c\text{-}jo}$ at the crank journal and $W_{b\text{-}cy}$ between the blade and the cylinder slot, are given by the following expressions:

$$W_{g\text{-}co} = \int_0^{t_0} \{e \sin\frac{\theta+\xi}{2} F_p + e\frac{\sin(\theta+\xi)}{\cos\xi} F_{qx}\}\dot{\theta}\,dT\,,$$

$$W_{p\text{-}cp} = \int_0^{t_0} M_p\dot{\theta}\,dT\,,\quad W_{c\text{-}jo} = \int_0^{t_0} M_s\dot{\theta}\,dT\,,\quad W_{b\text{-}cy} = \int_0^{t_0} e\frac{\sin(\theta+\xi)}{\cos\xi}(F_{gt1}+F_{gt2})\dot{\theta}\,dT \tag{6}$$

Calculation of Mechanical Efficiency

The mechanical efficiency η_m can be defined as the ratio of the gas compression power $W_{g\text{-}co}$ to the shaft power W_{shaft}:

$$\eta_m = \frac{W_{g\text{-}co}}{W_{shaft}} \times 100\ (\%) \tag{7}$$

As the energy losses due to friction increase, the shaft input energy increases because of the relationship given in (2) or (5), and thereby the mechanical efficiency decreases.

NUMERICAL CALCULATIONS

Frictional Coefficients

When calculating the mechanical efficiency, reasonable values must be determined for the frictional coefficients. This study makes use of the frictional coefficients which were determined numerically in the previous studies [6-8] for a rolling-piston rotary compressor with small cooling capacity. As is well known, the scroll compressor mechanism is quite different from the rolling-piston rotary compressor mechanism. It is interesting to examine how the mechanical efficiency is affected by the difference in the compression mechanism itself. From this viewpoint, the mechanical efficiency of the scroll compressor should be compared with a rolling-piston rotary compressor with the same cooling capacity, but the frictional state of the scroll compressor must be assumed to be basically the same as that of the rolling-piston rotary compressor.

In the frictional loss analysis for a rolling-piston rotary compressor [6-8], the following conclusions were made: the oil viscosity coefficient determined on the basis of oil pressure and temperature experiments is 2.08 mP_as; the two frictional coefficients at the crank journal and at the blade have a significant effect on the mechanical efficiency and the rotating speed of the rolling piston. Thus the steady dynamic behavior of the moving elements was calculated numerically from (4) and (5), by adjusting the two frictional coefficients so that the calculated values of the mechanical efficiency from (7) and the mean rotational speed of the rolling piston as expressed in

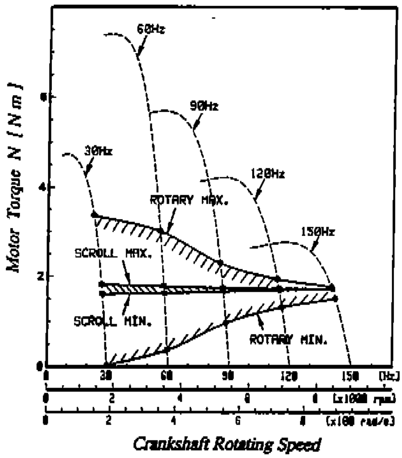


Fig.3 Motor torque curves

the equation of motion (4) in Ref.6 agree with the measured values. The calculated values for the frictional coefficients were 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 crankpin. 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 friction between the sliding pairs in the scroll compressor has the same level of boundary lubrication as the crank journal in the rolling-piston rotary compressor. Thus, it can be determined that the frictional coefficients for the scroll compressor also take on a value of 0.013.

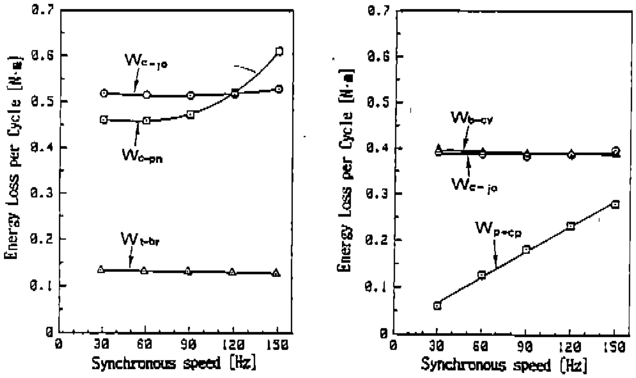
Motor torque, gas pressure and mechanical constants

The motor torques for five synchronous speeds from 30 to 150 Hz are shown by dotted lines in Fig.3, while the solid curves show, for reference, the range of the motor torque fluctuation which was calculated in our previous study [3]. In the numerical calculations, these motor torque curves are approximated by a 4th-order polynomial expression for the rotating speed $\dot{\theta}$. Assuming that the gas compression process is governed by an adiabatic change of specific heat ratio of 1.32, which is the value of the superheated Freon (R-22) vapor at a pressure of 1.32 MPa and a temperature of 55.6 °C, the pressures in the compression chambers can be calculated. The suction pressure is 0.617 MPa and the discharge pressure is 2.17 MPa.

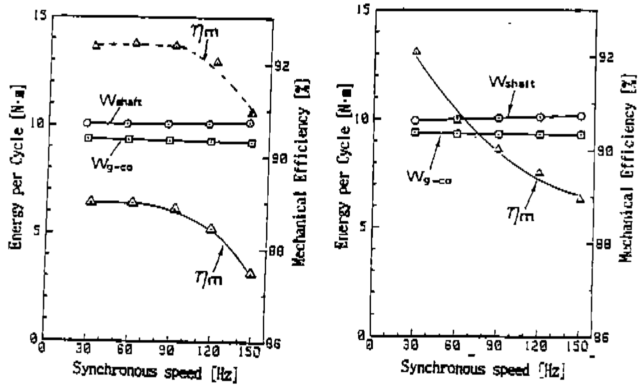
The major mechanical constants of the two small cooling capacity compressors having a suction volume of 10.26 cm^3 are shown in Table 1. A significant factor in the mechanical constants is the mass m_s of the orbiting scroll, which takes on a comparatively large value of 0.277 kg. The crankshaft moments of inertia I_0 for the scroll compressor and I_C for the rolling-piston rotary compressor are the same. Both compressors have a reciprocating element: the Oldham ring in the scroll compressor and the blade in the rolling-piston rotary compressor. The mass m_o of the Oldham ring is about 5 times larger than the mass m_v of the blade. This difference is, of course, caused by the structural difference between the scroll compressor and the rolling-piston rotary compressor.

Calculated Results

The dynamic behavior of the crankshaft can be calculated numerically from the expression (1) for the scroll compressor or from (4) for the rolling-piston rotary compressor as follows: the initial values of the rotating angle θ and the instantaneous rotating speed $\dot{\theta}$, and the crankshaft mean rotating speed which is a little smaller than each synchronous speed are given first, and thus θ, $\dot{\theta}$ and $\ddot{\theta}$ over one period are calculated. The initial value of $\dot{\theta}$ is modified if θ does not increase by one revolution 2π in one period, and the mean rotating speed of the crankshaft is modified if the energy equilibrium equation (2) or (5) is not satisfied. Then the same numerical calculation is



(a) Scroll compressor (b) Rolling-piston rotary compressor
Fig.4 Frictional losses



(a) Scroll compressor (b) Rolling-piston rotary compressor
Fig.5 Mechanical efficiency

repeated over one period again. Thus, the periodic dynamic behavior of the crankshaft can be obtained.

By using the calculated results for the dynamic behavior of the crankshaft, the constraint force and the frictional force at each pair of the compressor elements can be calculated. Finally, the energy losses due to friction given by (3) for the scroll compressor and by (6) for the rolling-piston rotary compressor can be calculated, as shown by the solid curves in Fig.4. It should be noted from these calculated results that

(1) as the synchronous speed of the motor increases, the two frictional losses $W_{c\text{-}pn}$ and $W_{p\text{-}cp}$ at the crankpin only increase;
(2) the frictional loss $W_{c\text{-}pn}$ for the scroll compressor increases at a much slower rate in the lower synchronous speed range than the frictional loss $W_{p\text{-}cp}$ for the rolling-piston rotary compressor, and the increases with the synchronous speed is much more gradual overall.

These two factors result in the mechanical efficiency η_m shown in Fig.5. As the synchronous speed of the motor increases, the mechanical efficiency for both compressors decreases, but the rate of decrease for the scroll compressor is much less, in the synchronous speeds less than about 120 Hz than that for the rolling-piston rotary compressor. The mechanical efficiency of the scroll compressor is a little less over the entire range of synchronous speeds, than that of the rolling-piston rotary compressor. However, by selecting the optimum combination of the major dimensions, such as the involute basic circle radius and the cylinder depth [9], the mechanical efficiency can be vastly improved, as shown by the dotted curve in Fig.5a. In this case, the mechanical efficiency of the scroll compressor exceeds that of the rolling-piston rotary compressor over the entire range of synchronous speed.

Discussion for calculated results

The mechanical efficiency characteristics shown in Fig.5 depend of course upon the frictional losses W_{c-pn} and W_{p-c}, at the crankpin which are shown in Fig.4. Let us examine the physical basis for these frictional losses which vary depending upon the operating speed.

In the case of the scroll compressor, the mass of the orbiting scroll is comparatively large, as shown in Table 1. Thus, the large centrifugal force of the orbiting scroll must be constrained by the crankpin. Therefore, as the operating speed of the crankshaft increases, the effect of the centrifugal force upon the constraint force on the crankpin increases in proportion to the square of the operating speed, and thus the rate of the frictional loss W_{c-pn} at the crankshaft increases in proportion to the operating speed. When the operating speed is lower, however, the gas force has the same level of fluctuation, regardless of the operating speed, and this becomes the major factor dominating the constraint force on the crankpin. Thus, the frictional loss W_{c-pn} at the crankshaft changes slightly, in the lower range of the synchronous speed. As the operating speed increases, however, the effect of the centrifugal force upon the constraint force exceeds that of the gas force.

In the rolling-piston rotary compressor as well, the centrifugal force of the rolling piston must be carried by the crankpin, but a fluid lubrication is formed between the rolling piston and the crankshaft. Therefore, the frictional torque at the crankpin which is caused by the oil viscosity, is independent of the gas force, rather it is a function of the relative sliding speed of the rolling piston with respect to the crankpin. Thus, the frictional torque at the crankpin increases linearly with the operating speed.

CONCLUSIONS

This study was concerned with the mechanical efficiency of a variable speed compact scroll compressor having a suction volume of 10.3 cm^3. A numerical calculation based on the dynamic analysis [1-4] of the moving elements of the compressor was carried out for a compact scroll compressor driven at various synchronous speeds from 30 Hz to 150 Hz, under the standard JIS conditions for gas pressure (0.61 MPa for the suction pressure, 2.17 MPa for the discharge pressure). The calculated results for the mechanical efficiency and the power losses due to friction were compared with those for a rolling-piston rotary compressor with the same cooling capacity. The conclusions of this study can be summarized as follows:.

(1) As the operating speed of the compressor increases, the mechanical efficiency decreases, but the rate of decrease for a scroll compressor is far less than that for a rolling-piston rotary compressor.

(2) The mechanical efficiency of the scroll compressor was a little less than the high mechanical efficiency of the rolling-piston rotary compressor. By selecting the optimum combination of major dimensions in the scroll compressor, however, the mechanical efficiency can be improved to the extent that it exceeds the high mechanical efficiency of the rolling-piston rotary compressor.

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