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Optimum Combination of Dimensions for High Mechanical Efficiency of a Rolling-Piston Rotary Compressor

by

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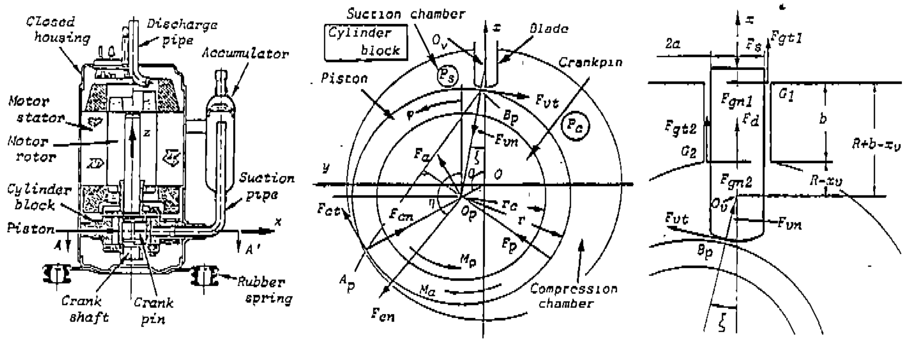
ABSTRACT

The suction volume of a rolling-piston type rotary compressor is determined on the basis of the major dimensions, such as the rolling-piston diameter, the cylinder depth and the cylinder bore. It becomes clear that there are many combinations of major dimensions that yield a rolling-piston rotary compressor with the same suction volume. This paper presents the mechanical efficiency for various combinations of major dimensions and an optimum combination diagram showing which combination ensures the best mechanical efficiency. In order to examine the physical basis for the optimum combination of major dimensions, the frictional power loss at each pair of compressor elements is shown.

INTRODUCTION

The rolling-piston rotary compressor, used in most low capacity air conditioners, exhibits superior performance in terms of high mechanical efficiency. The primary reason for this high efficiency is the rolling piston. Fluid lubrication occurs between the rolling piston and the crank pin, resulting in a substantial decrease in the sliding speed of the blade relative to the rolling piston (see Refs. [1] and [2]). Thus, the rolling piston effectively decreases the frictional power losses between the crank pin and the blade [2, 3].

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 cylinder pressure and the inertial forces but also upon the major dimensions, such as the rolling-piston diameter, the cylinder depth and the cylinder bore. It is suggested here that the suction volume of the rolling-piston rotary compressor is determined by these major dimensions and, consequently, there are many combinations of major dimensions that yield a rolling-piston rotary 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 dimensions, and this results in different mechanical efficiencies. Thus, selection of the optimum combination of major dimensions is necessary to ensure a high mechanical efficiency for the rolling-piston rotary compressor. The mechanical efficiency was calculated for various possible combinations of major dimensions, and it could be shown that there is an optimum combination of major dimensions. In addition, this paper presents an optimum combination diagram which shows which combination ensures the best mechanical efficiency. In addition to this diagram, the physical basis for the optimum combination of major dimensions is examined by showing the characteristics of the frictional power loss at each pair.



(a) Cross section of the compressor (b) A-A' cross section (c) Frictional forces around the blade

Fig. 1 Rolling-piston rotary compressor

ROLLING-PISTON ROTARY COMPRESSOR

Fig. 1a shows the construction of a rolling-piston rotary compressor which is widely used for air conditioners having a refrigerating capacity of 1,755 kcal/h. The motor stator and the cylinder block are fixed inside the closed housing which is supported by three rubber springs on a base. The refrigerant R-22 is sucked into the cylinder through the accumulator. The compressed refrigerant is discharged inside the closed housing and transferred to a condenser through the discharge pipe on the top of the closed housing. The motor is a single phase induction motor with a synchronous speed of 3,600 rpm and power of 0.55 kW.

As the A-A' cross-sectional view in Fig. 1b shows, the rolling-piston rotary compressor is not an ideal rotary machine, since it has a reciprocating blade which partitions the cylinder into suction and compression sections. The blade mass is comparatively small, however, and thus the vibration level of the rolling-piston rotary compressor, except for around the rotating crankshaft [4-6], is much lower than that of the reciprocating compressor[7-12]. In addition to lower vibrations, the rolling-piston rotary type has excellent mechanical efficiency [1-3, 13, 14]. The part compressing the refrigerant is soaked in high pressure lubricating oil. Thus, gas leakage from the compression chamber is prevented by the oil seal, and moreover, since the lubricating oil is pressed into the narrow space between the rolling piston and the crank pin, power losses between the crank pin and the blade, which is tightly pressed against the rolling piston, are effectively decreased.

MECHANICAL EFFICIENCY AND FRICTIONAL COEFFICIENTS

In order to calculate the mechanical efficiency, the frictional force at each pair of compressor elements must first be calculated. Prior to this, however, the reaction force at each pair must be calculated. In this regard, it should be noted that the reaction forces can be derived from the equations of motion of the moving elements of the compressor [4]. Based on the derived equations of motion, the equation of rotating motion of the crank shaft can be written in the following expression [2]:

$$
\begin{aligned}
&(I_c+m_p e^2)\ddot{\theta} - e\,\frac{\sin(\theta+\xi)}{\cos\xi}(m_v\ddot{x}_v + F_s) \\
&= M_m - [\{e\sin\left(\frac{\theta+\xi}{2}\right)F_p - e\,\frac{\sin(\theta+\xi)}{\cos\xi}(-F_{qx})\} + M_p + M_s \\
&\quad + e\{\cos(\theta+\xi) + \tan\xi\,\sin(\theta+\xi)\}F_{vt} + e\,\frac{\sin(\theta+\xi)}{\cos\xi}(F_{gt1}+F_{gt2})\,] \qquad (1)
\end{aligned}
$$

The terms on the left represent those due to the inertial forces and the spring force F_s, where θ is the rotating angle of the crank shaft, x_v the displacement of the blade and e the eccentricity of the crank pin, as shown in Fig. 1. I_c is the moment of inertia of the crankshaft, m_p the rolling-piston mass and m_v the blade mass. M_m on the right side represents the motor torque. The first term in the brackets represents the torque necessary for gas compression. F_p and F_{qx} represent the resultant gas forces acting on the rolling piston and the blade, respectively, as shown in Fig.1b. The other terms represent the frictional torque M_p between the rolling piston and the crank pin, the torque M_s at the crank journal, the torque due to the frictional force F_{vt} between the blade and the rolling piston, and the torque due to the frictional forces F_{gt1} , F_{gt2} (see Fig.1c) between the blade and its guide slot.

Multiplying the crankshaft rotating velocity θ by Eq. (1) and integrating it with respect to the time t, the following energy equation can be obtained:

$$W_{shaft} = W_{gas} + W_{Mp} + W_{Ms} + W_{Fvt} + W_{Fgt} \tag{2}$$

This equation means that the energy supplied by the motor must be entirely consumed by the gas compression and the frictional forces. Thus, the mechanical efficiency η_m can be defined by the following expression:

$$\eta_m = \frac{W_{gas}}{W_{shaft}} \times 100\ (\%) \tag{3}$$

When calculating the mechanical efficiency, it is important to evaluate the frictional coefficients precisely. This study makes use of the frictional coefficients which were determined numerically in the previous studies [1-3], as follows: it was determined, first, 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. Secondly, it was assumed that the friction between the blade and the rolling piston, and the blade and the cylinder slot are exposed to boundary lubrication. In this case, there is a little more boundary friction than at the crank journal, since these pairs of machine elements are lubricated poorly by oil which is mixed into the refrigerant. Therefore, the unknown factors for the computer simulations of the power losses due to mechanical friction are the following: the oil viscosity coefficient of the fluid lubrication and the two frictional coefficients of the two cases of boundary lubrication. The oil viscosity coefficient was determined on the basis of oil pressure and temperature experiments. The remaining unknown factors have a great effect on the mechanical efficiency and the rolling-piston rotational speed. Because of the importance of these factors, the next step was to measure the mechanical efficiency and the rolling-piston mean rotational speed. Then the steady dynamic behavior of the moving elements was calculated numerically from (1) and (2), by adjusting the two frictional coefficients so that the calculated values of the mechanical efficiency from (3) and the rolling-piston mean rotational speed agree with the measured values. The calculated frictional coefficients were 0.013 for the crank journal and 0.083 around the blade.

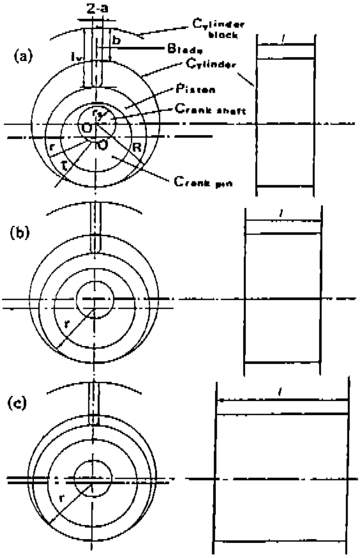
CALCULATED RESULTS

Combination of Major Dimensions

The suction volume V_s of the rolling-piston rotary compressor is given by the following simple expression:

$$V_s = \pi (R^2 - r^2) l \tag{4}$$

where R represents the cylinder radius, r the rolling-piston radius and l the cylinder depth. It is apparent from this expression that there are various combinations of the cylinder radius, the rolling-piston radius and the cylinder depth, for the same suction volume. For instance, Fig.2 shows a schematic explanation for the relationship between the rolling-piston radius r and the cylinder depth l. The cylinder radius R is kept at a constant value. As the rolling-piston radius increases from (a) to (c), the suction area between the cylinder and the rolling piston decreases, and thus the cylinder depth l increases, since the suction volume is kept at a constant value. Here we assume the suction volume to be 10.26 cm^2. When changing the rolling-piston radius, the cylinder depth changes, as shown in Fig.3, and the cylinder radius R as a parameter takes on the values 1.6, 1.95, 2.7 and 3.5 cm.



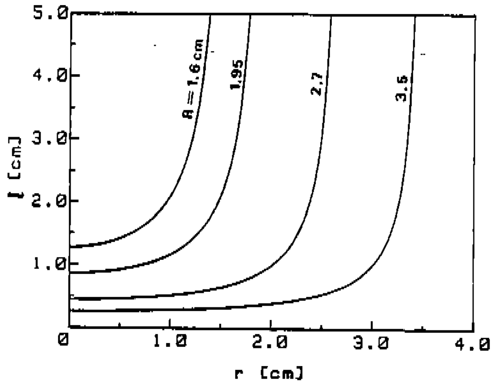


Fig.2 Schematic explanation of the relation between rolling-piston radius r and cylinder depth l

Fig.3 Cylinder depth l vs. rolling-piston radius r (where the suction volume is kept at 10.26 cm^3)

Specifications of the Compressor

Here consider the rolling-piston rotary compressor driven by the motor torque shown in Fig.8 of the previous study [4]. The synchronous speed of the motor is 3,600 rpm. The measured gas pressures in the suction and compression chambers are shown in Fig.9 of the same study [4]. The suction pressure is 0.617 MPa and the discharge pressure 2.17 MPa. It is a matter of course that the piston mass, the blade mass, the blade length and so on change, depending upon the combination of major dimensions. They were calculated based on the major dimensions, but it was assumed for convenience that the blade thickness takes a constant value of 3.2 mm, the rolling-piston thickness 4.2 mm and the blade slot length 14.7 mm. The moment of inertia of the whole crankshaft was kept at a constant value of 0.0422 $N\cdot cm\cdot s^2$.

Mechanical Efficiency for a Constant Cylinder Radius

When the specifications necessary for numerical calculation are given, such as the driving torque, the cylinder pressures, major mechanical constants and the frictional constants, the equation of motion (1) of the crankshaft can be calculated numerically. The frictional forces and moments in Eq.(1) are of course affected by the inertial forces, and thus an iterative calculation method was adopted to calculate Eq.(1). The mean rotating speed of the crankshaft is determined by the energy equilibrium equation (2). As a result, the drive shaft power W_{shaft} and the gas compression power W_{gas} can be obtained, and thus the mechanical efficiency can be calculated from (3).

The calculated results for the drive shaft and gas compression power and the mechanical efficiency are shown in the upper diagram of Fig.4, in which the cylinder radius is kept at a constant value of 1.95 cm. It is significant to note that the mechanical efficiency curve indicated by η_m shows a maximum value of 91.14 % at the rolling-piston radius r of 1.5 cm and that the mechanical efficiency decreases rapidly on either side of this optimum value. As seen from the curve indicated by W_{shaft}, the shaft power (motor input power) increases as the mechanical efficiency decreases. This is of course caused by the increase in power losses due to friction, as shown in the lower diagram of Fig.4. As the rolling-piston radius increases, (1) the frictional power loss W_{Fgt} between the blade and the cylinder slot decreases, since the stroke $2(R-r)$ of the reciprocating blade decreases; (2) the frictional power losses W_{Ms} at the crank journal and W_{Mp} at the crank pin increase, since the shaft loads (for example, F_p and F_{qx}; see Fig.1b) due to cylinder pressure increase because of the increase in the cylinder depth. As a result of these two factors, the minimum power loss due to mechanical friction appeared at the rolling-piston radius of 1.5 cm for the constant cylinder radius of 1.95 cm.

The vertical dotted line represents a value (1.56 cm) for the rolling-piston radius r which was calculated using the following expression:

$$r = (R + t + r_s)/2 \qquad (5)$$

where t represents the rolling-piston thickness and r_s the crankshaft radius (see Fig.2). When the rolling-piston radius is smaller than this value, the inner ring of the rolling piston crosses the crankshaft, and thus it becomes a little difficult to mount the rolling piston on the crankpin.

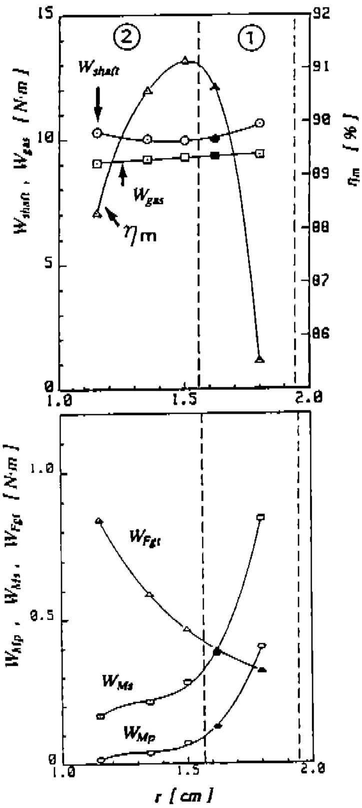
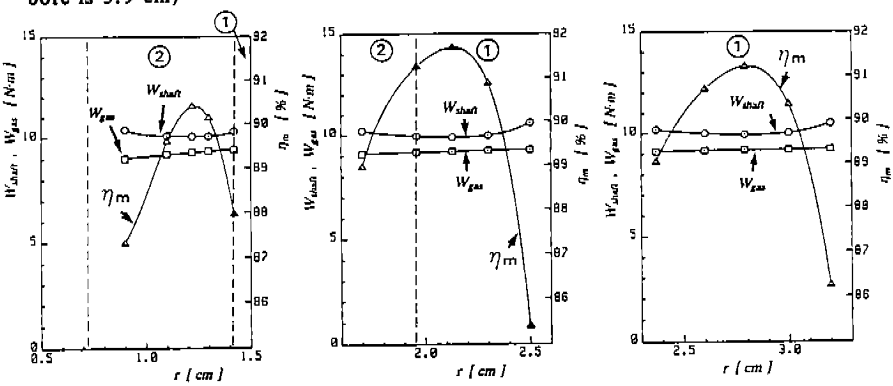


Fig.4 Mechanical efficiency η_m , shaft power W_{shaft}, gas compression power W_{gas} and frictional power losses for various combinations of the rolling-piston radius r and the cylinder depth l (where the cylinder bore is 3.9 cm)

Effects of the Cylinder Radius upon the Mechanical Efficiency

Similar calculations were carried out for cylinder radii of 1.6, 2.7 and 3.5 cm. The cylinder depth for each cylinder radius is shown in Fig.3. The corresponding curves for power and mechanical efficiency for these three radii are found in Fig. 5a, b, and c, respectively. A comparison of Fig.4 (top) and Fig. 5a shows that when the cylinder radius is smaller than 1.95 cm, the maximum value for mechanical efficiency decreases, and, furthermore, the slope of the efficiency curve becomes steeper. When the cylinder radius is larger than 1.95 cm, the mechanical efficiency curve becomes wider (slope decreases), and the maximum mechanical efficiency increases first (see Fig 5b where the cylinder radius is 2.7 cm) and then decreases (see Fig.5c where the cylinder radius is 3.5 cm).

The maximum value for mechanical efficiency and the optimum rolling-piston radius leading to the highest mechanical efficiency are summarized in Fig.6. Based on this optimum combination diagram of the major dimensions of the rolling-piston compressor with a cylinder volume of 10.26 cm^3, it may be suggested that the best choice for the cylinder radius, the rolling-piston radius and the cylinder depth is 2.25, 1.75 and 1.63 cm, respectively. However, the maximum mechanical efficiency curve is very wide, as seen from Fig.6, and hence various design criteria could be included in the selection of the combination of major dimensions.



(a) Cylinder radius 1.6 cm (b) Cylinder radius 2.7 cm (c) Cylinder radius 3.5 cm

Fig.5 Mechanical efficiency η_m , shaft power W_{shaft} and gas compression power W_{gas} for various combinations of the rolling-piston radius r and the cylinder depth l

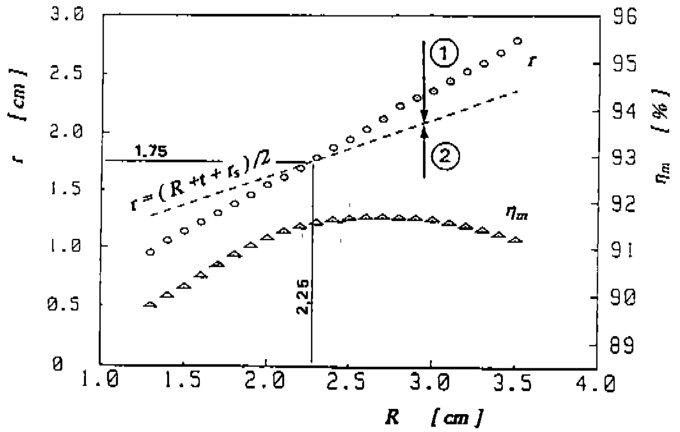


Fig. 6 Optimum combination diagram of major dimensions and maximum mechanical efficiency η_m

CONCLUSIONS

A large number of rolling-piston type rotary compressors are used all over the world. Therefore, a small increase in the mechanical efficiency of one compressor results in a great amount of energy saving. From this viewpoint, even a small improvement in mechanical efficiency is significant. Various approaches will be used to increase mechanical efficiency. In this study, the problem was approached from the dynamics of machinery standpoint, and the effect of the combination of major dimensions upon the mechanical efficiency was examined by calculating numerically the dynamic behavior of the moving elements of the compressor. The conclusions from this study can be summarized as follows:

(1) Given the same suction volume, as the rolling-piston radius increases, the power losses arising around the reciprocating blade decrease, and, conversely, those arising around the crankshaft increase. Thus, for a given cylinder radius, there appears to be an optimum combination of rolling-piston radius and the cylinder depth.

(2) Numerical calculations for various cylinder radii could be used to construct an optimum combination diagram of major dimensions, such as the cylinder radius, the rolling-piston radius and the cylinder depth.

Based on the calculated results from optimum combination diagrams for various suction volumes, Matsushita is continuing to develop a series of compressors with higher mechanical efficiency.

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