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N. Ishii
Osaka Electro-Communication University

K. Bird
Matsushita Compressor Corporation of America

S. Yamamoto
Matsushita Electric Industrial Co.

H. Matsunaga
Matsushita Electric Industrial Co.

K. Sano
Matsushita Electric Industrial Co.

See next page for additional authors

Authors
N. Ishii, K. Bird, S. Yamamoto, H. Matsunaga, K. Sano, and M. Hayashi

A FUNDAMENTAL OPTIMUM DESIGN FOR HIGH MECHANICAL AND VOLUMETRIC EFFICIENCY OF COMPACT ROTARY COMPRESSORS

by

Noriaki Ishii[1], Kenichi Bird[2], Shuichi Yamamoto[3], Hirosi Matsunaga[3]
Kiyosi Sano[4] and Masaki Hayashi[5]

[1] Professor, Faculty of Engineering, Osaka Electro-Communication University, Neyagawa- city, Osaka 572, Japan
Tel: +81-720-20-4561; Fax: +81-720-20-4577; E-mail: ishii@isc.osakac.ac.jp
[2] Engineer, Technical Center, Matsushita Compressor Corp. of America One Panasonic Way, P. O. Box 1418, Mooresville, NC 28115,
Tel: (704)664-8184; Fax: (704)664-9395
[3] Senior Engineer, Compressor Division
[4] Senior Staff Engineer, Air Conditioning Research Laboratory, Matsushita Electric Industrial Co., Ltd.(Panasonic) Noji-cho, Kusatsu-shi, Shiga 525, Japan.
Tel: +81-775-67-9801, Fax: +81-775-61-3201
[5] Undergraduate Student, Faculty of Engineering, Osaka Electro-Communication University.

ABSTRACT

This paper presents a fundamental optimum design which yields a high compressor performance in mechanical and volumetric efficiency of a rolling-piston rotary compressor. The frictional power losses at each pair of machine elements were calculated by an analytical method revealing the dynamic behavior of rolling-piston rotary compressors, and the refrigerant leakage from axial and radial clearances was calculated by the incompressible and viscous theory assuming an entire turbulent leakage flow. Computer calculations were made for a number of combinations of the major dimensions for various suction volumes. Calculations for the mechanical and volumetric efficiency resulted in an optimum combination of major dimensions for various suction volumes of the rotary compressor.

INTRODUCTION

The suction volume of rolling-piston type rotary compressors 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 the major dimensions that yield a rolling-piston rotary compressor with the same suction volume. Depending upon the combination of the major dimensions, the constraint force at each pair of the compressor elements changes and as a result the power loss due to mechanical friction changes. Based on such viewpoint, the mechanical efficiency was calculated for various combinations of the major dimensions to present an optimum combination chart diagram which yields a high performance in mechanical efficiency, by Ishii *et al.* (1990 /1/). Similar calculations have been made for scroll compressors by Ishii *et al.* (1990 /2/; 1992 /3/; 1994 /4/).

In addition to the mechanical efficiency, the volumetric efficiency also has to be calculated, since the leakage flow path area between the compressor elements changes depending upon the combination of the major dimensions. For the scroll compressors, a simple method to evaluate refrigerant leakage flows through the axial and radial clearances between scroll compressor elements was developed by Ishii *et al.* (1996 /5/), based on the incompressible and viscous turbulent flow theory. The developed simple method was applied to a scroll compressor to calculate its volumetric efficiency by Ishii *et al.* (1996 /6/).

Studies for refrigerant leakage flow evaluation can be naturally applied to rotary compressors which have leakage flows through the axial clearance between the blade and the thrust plate and through the radial clearance between the piston and the cylinder, quite similar to those in the scroll compressors. In this study, first, the volumetric efficiency for a fixed suction volume and cylinder bore is calculated for a number of combinations of the major dimensions. Secondly, the product of the volumetric and mechanical efficiencies are calculated to figure out the net efficiency. Finally an optimum combination of the major dimensions, yielding a high net efficiency, is calculated for various suction volumes of the rotary compressor, to insist that such a fundamental optimum design calculation is quite significant, especially when designing a small capacity compressor.

COMBINATION OF MAJOR DIMENSIONS

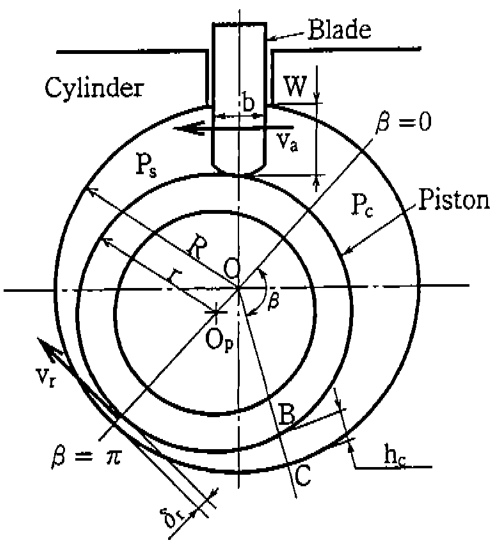

Figure 1. Configuration of a rolling-piston rotary compressor, and leakage flows through axial and radial gaps.

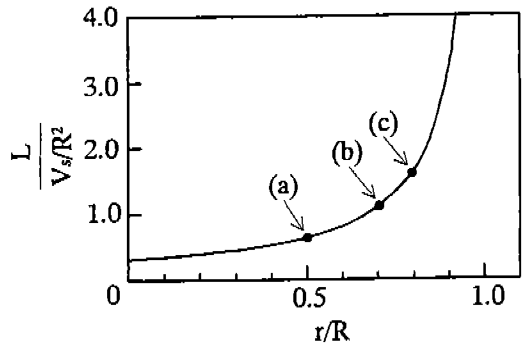

Figure 2. Characteristic curve for combination of major dimensions of a rolling-piston rotary compressor.

A configuration of the rolling piston rotary compressor is shown in Figure 1, where the cylinder radius is represented by R and the rolling piston radius by r. The suction volume V_s is given by

$$V_s=\pi(R^2-r^2)L \quad \text{or} \quad \frac{L}{V_s/R^2}=\frac{1}{\pi\{1-(r/R)^2\}}, \tag{1}$$

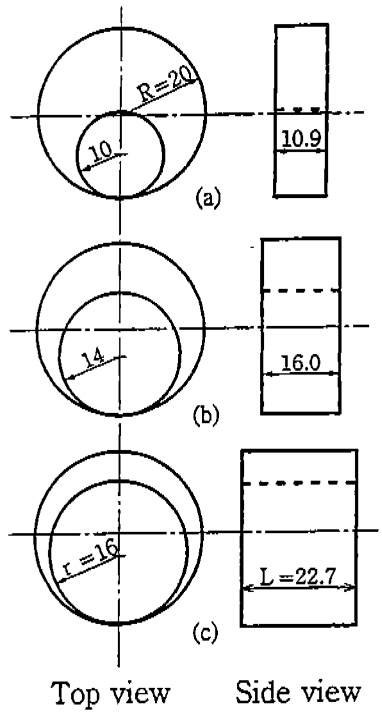

Figure 3. Schematic explanation of combination of the piston radius and the cylinder depth, for the cylinder radius of 20 mm and the suction volume of 10.26 cc.

where L is the cylinder depth. The characteristic curve for combinations of the major dimensions, such as R, r and L, is shown in Figure 2, where the ordinate is namely a reduced cylinder depth. Schematic explanation of combination of the piston radius and the cylinder depth is given in Figure 3, where the suction volume was fixed at 10.26 cc and the cylinder radius at 2 cm. As the piston radius increases, the suction area between the cylinder and the rolling piston decreases, and thus the cylinder depth L increases. The combinations shown in diagrams (a) to (c) were plotted on the characteristic curve in Figure 2.

CALCULATIONS FOR VOLUMETRIC EFFICIENCY

As was shown in Figure 1, the leakage flow through the axial gap δ_a between the blade and the thrust plate is represented by its velocity v_a and that through the minimum radial gap δ_r between the piston and the cylinder is represented by its velocity v_r. Based on the fundamental theory for incompressible and viscous flow through a circular pipe, the leakage velocities caused by the pressure difference of P_c in the compression chamber and P_s in the suction chamber can be calculated by the following expressions:

$$\frac{P_c - P_s}{\rho g} = \lambda \frac{b}{2\delta_a} \frac{v_a^2}{2g} \text{ for axial gap leakage; } \int_{\beta_s}^{\beta_c} \lambda \frac{R d\beta}{2h_c} \frac{v_r^2}{2g} \text{ for radial gap leakage,} \tag{2}$$

where b represents the blade thickness, ρ the refrigerant specific mass and g the gravity acceleration. h_c represents the radial gap height given by a function of the angle β as an integral variable, as shown in Figure 1. Assuming an entire turbulent flow for the refrigerant leakages, the friction factor λ is given by

$$\lambda = 0.35 Re^{-0.35}, \tag{3}$$

where the Reynolds number Re is defined by

$$R_e = \frac{2\delta_a v_a}{\mu/\rho} \text{ for axial gap leakage; } \frac{2h_c v_r}{\mu/\delta} \text{ for radial gap leakage,} \tag{4}$$

where μ is the viscosity coefficient. If the leakage velocities are calculated, the leakage mass flow rate and the net leakage mass during one period can be calculated, and finally the volumetric efficiency can be calculated.

The evaluation method for leakage flows, here introduced, is simple and it has been confirmed by Ishii *et al.* (1996 /5/) that the evaluation method presents results in good agreement with experimental results.

CALCULATED RESULTS

Volumetric Efficiency

In numerical calculations, the blade thickness was fixed at 3.2 mm, and the leakage path was assumed to be 10 μm in height, both for the axial gap and the minimum radial gap. The viscosity coefficient μ was given by a function of pressure in the compression chamber. The specific heat ratio of the compressed refrigerant was assumed to be 1.32, the suction pressure P_s to be 0.62 MPa, the discharge pressure to be 2.17 MPa, the suction temperature to be 18 °C. The specific mass of refrigerant in the suction chamber was 24.71 kg/m^3. The synchronized speed of the drive motor was at 3600 rpm.

One of calculated results for the volumetric efficiency is shown in Figure 4, where the suction volume V_s was fixed at 10.26 cc and the cylinder radius R at 2 cm. As the piston radius r increases, the leakage mass through the axial gap, M_B, decreases, while that through the radial gap, M_{c-p}, increases, thus resulting in the net leakage mass M showing its minimal value at r=1.58 cm and the resulting in volumetric efficiency η_v showing its maximal value of 93.4 %. The cylinder depth L for the optimum was 2.2 cm.

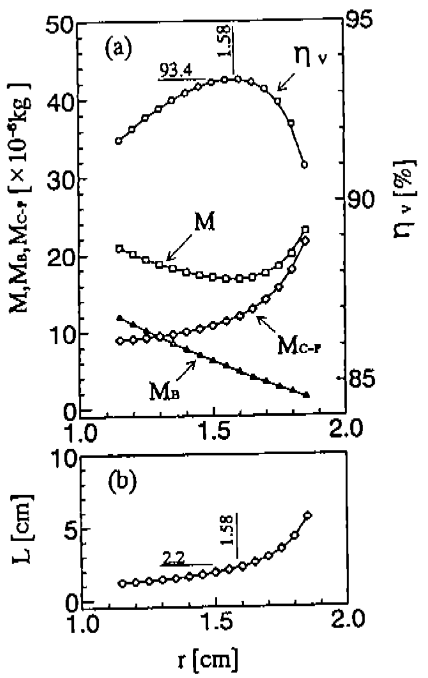

Figure 4. Volumetric efficiency for the suction volume of 10.26 cc and the cylinder radius of 2.0 cm.

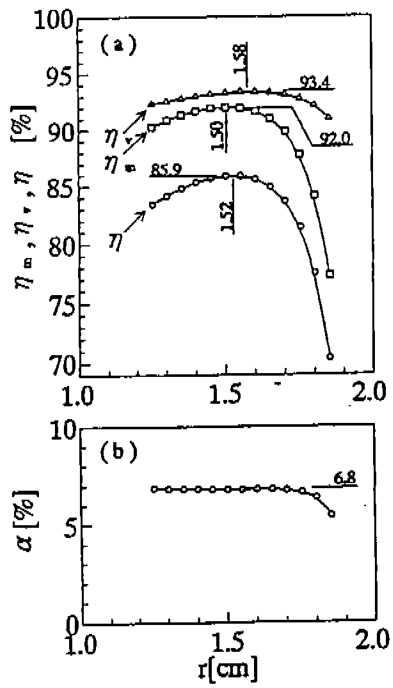

Figure 5. Mechanical efficiency and net efficiency for the suction volume of 10.26 cc and the cylinder radius of 2.0 cm.

Mechanical Efficiency and Net Efficiency

The frictional coefficient for the rotary compressor was assumed to be 0.013 for the crank journal and 0.083 for around the blade, which were calculated on the basis of experimental results by Ishii *et al.* (1998 /7/). In numerical calculations, the crankshaft diameter was adjusted so that the stress due to shaft load is kept at a constant value. The mechanical efficiency η_m calculated for the same suction volume and cylinder radius is shown in Figure 5, where the maximal value of 92.0 % appears at r=1.5 cm. The product of mechanical and volumetric efficiencies yields the net efficiency η, showing its maximal value of 85.9 % at r=1.5 cm. The crankshaft speed fluctuation ratio was about 6.8 %.

Similar calculations for the suction volume V_s of 10.26 cc were made for a number of the cylinder radius R, and each peak value for the volumetric, mechanical and net efficiencies was plotted, as denoted by solid lines in Figure 6, where the abscissa is the cylinder radius R. The optimized volumetric efficiency η_v continuously increases with enlarging the cylinder radius, while the optimized mechanical efficiency η_m shows its maximal value of 92.4 % at R=2.69 cm. Consequently, the optimized net efficiency exhibits its maximal value of 88.0 % at R=2.97 cm. If net efficiency results in 87.6 %, a little smaller than 88.0 %. This difference in optimized net efficiency is small, but becomes significantly large especially when designing a small capacity compressor. The optimized net efficiency calculated for the suction volume V_S of 2.5 cc is shown by a dotted line in the same figure, showing its maximum value of 82.7 % at R=2.44 cm. This maximal vale is significantly larger than 81.0 % at R=1.76 cm, based on the mechanical efficiency alone.

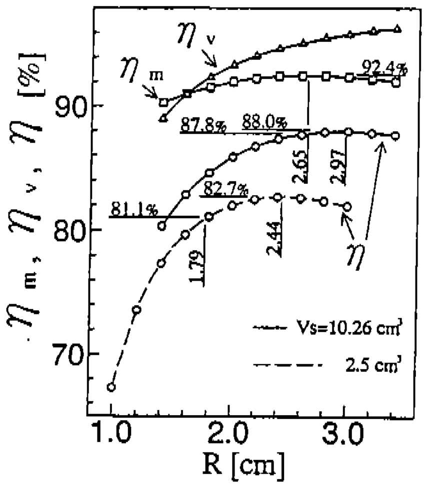

Figure 6. Maximal values of volumetric efficiency, mechanical efficiency and net efficiency, versus cylinder radius, for the suction volume of 10.26cc.

Optimum Combination of Major Dimensions and Resulting Net Efficiency

An optimum design chart diagram for combinations of the major dimensions in rolling-piston rotary compressors is presented by solid lines in Figure 7a, where the abscissa is the suction volume V_S from 2.5 cc to 10.26 cc. The solid lines show optimum combinations based on the mechanical efficiency alone, showing a large difference from those based on the net efficiency.

The maximal net efficiency resulting from the optimum combination is shown in Figure 7b, where the solid line is the result based on the net efficiency and the dotted is based on the mechanical efficiency alone. It is well know from this figure that the difference in maximal net efficiency significantly increases, as the suction volume decreases.

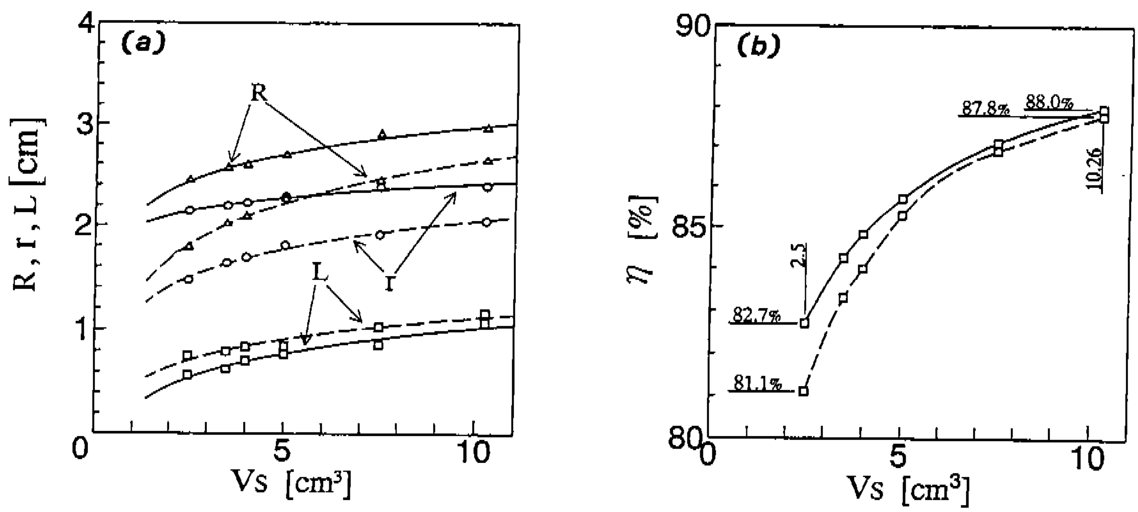

Figure 7. Fundamental optimum design chart diagram: (a) optimum combination of major dimensions for various suction volumes; (b) maximal net efficiency.

CONCLUSIONS

A fundamental optimum design chart diagram for combination of the major dimensions, yielding the maximal net efficiency of the mechanical and volumetric efficiencies, was presented for a rolling-piston rotary compressor which has a leakage gap of 10 mm at both axial ends of the blade and between the piston and the cylinder. Optimum design calculations like as presented in this study are quite significant, because a large number of rolling-piston type rotary compressors are used all over the world, and a small increase in net efficiency of one compressor results in a great amount of energy saving. Based on optimum design chart diagrams for various frictional coefficients and leakage gaps and for various operating conditions, Matsushita is continuing to develop a series of compressors with highest net efficiency.

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