1984

Performance Analysis of Rolling Piston Type Rotary Compressor for Household Refrigerators

S. Nagatomo
H. Sakata
M. Tago
H. Hattori

Follow this and additional works at: https://docs.lib.purdue.edu/icec

https://docs.lib.purdue.edu/icec/466

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
PERFORMANCE ANALYSIS OF ROLLING PISTON TYPE ROTARY COMPRESSOR FOR HOUSEHOLD REFRIGERATORS

Shigemi Nagatomo
Hirotugu Sakata
Masato Tago
Hitoshi Hattori
Toshiba Major Appliance Products Engineering Laboratory,
Yokohama City, Japan

ABSTRACT

This paper concerns the performance analysis in rolling piston type hermetic compressors used in domestic refrigerators. An experimental compressor was provided for measuring the pressure in its cylinder so that the indicated work could be estimated accurately. In particular, the following two items were tested and analyzed on account of their great influence on the compressor efficiency. One is the axial clearance of the rolling piston, the so called roller clearance (1). The other is the discharge port length (2).

The results of these investigations can be summarized as follows:
(1) It was clarified that as roller clearance became larger, volumetric efficiency was reduced and compressor intake power increased, i.e. total compressor efficiency was reduced considerably.
(2) It could be confirmed that the quantity of over-compression was about twice that of wiredrawing and had little to do with the discharge port length.
(3) If discharge port length became large, the quantity of residual gas in the cylinder increased: in consequence compression efficiency tended to decrease.
(4) On the bases of the above mentioned analysis, we were able to design a refrigerator compressor more suitable than the original, i.e. a compressor with an efficiency ratio increase of about 15%.

1. INTRODUCTION

In recent years, rolling piston type rotary compressors have gradually been utilized for domestic refrigerators, because of their special features of (1) high efficiency, (2) small size and light weight, and (3) simple mechanism.

Some research for efficiency improvement has already been reported regarding this type of compressor for air-conditioners (1)(2)(3)(4), but in regards to the efficiency improvement of compressors used in refrigerators research has not yet been performed (5)(6). Because fundamental data was very poor, we were not able to design a highly efficient compressor. To obtain a highly efficient compressor, compression efficiency as well as mechanical and motor efficiency had to be improved. It was necessary to measure the indicated work exactly in order to clarify these three areas. So, in this paper, the indicated work was obtained in detail by measuring the cylinder pressure with three transducers which were mounted at suitable location in the cylinder, so that compression and mechanical efficiency respectively could be exactly estimated. On the other hand, motor efficiency was obtained by measuring the motor performance.

Research in regards to roller clearance to efficiency have already been reported. However, in this paper, when this clearance varies, influences on compressor performance are clarified, and furthermore, behavior of P-V diagram is clarified experimentally in detail.

It has become clear that re-expansion of residual gas in the discharge port decreases compressor efficiency. However, cylinder pressure increase, caused by residual gas re-expansion, has not yet been measured and has not yet been calculated exactly. In this paper, we are able to clarify them.
2. TEST COMPRESSOR AND MEASURING METHOD

2.1 Test Compressor

A hermetic type rolling piston rotary compressor was provided for experiment. In its shell, a 2-pole induction motor and a compression mechanism were housed and connected with each other by a shaft. The compression mechanism was supported between the main-bearing and the sub-bearing. The discharge port and discharge valve were mounted on the same side as the main-bearing. A flange case was provided for easy experiment.

2.2 Measuring Method

(1) Pressure-Volume diagram measurement

Transducers mounting and measuring system for measuring the cylinder pressure and the valve motion is shown in Fig. 1. Cylinder pressures were measured with piezo type pressure transducers which were mounted on three different places: A(θ=31°, Suction gas pressure), B(θ=230°, Suction and compression gas pressure), C(θ=353°, compression gas pressure) as shown in Fig. 1. Discharge valve behaviour was measured by an eddy current probe mounted on the center of the valve stopper. The crank angle was also detected with the use of eddy current probe, measuring the signal dispatched from the motor rotor. At the same time these signals were amplified and stored in the multi-channel wave memorizer. After that, they were displayed on the display monitor, while memorized in the floppy disk. Then, these memorized data were computed by using our wave analyzing program.

Pressure variation, valve motion and crank angle were drawn on a diagram. After that the indicated work, over-compression loss, wiredrawing loss, revolution speed etc. were calculated.

(2) Efficiency measurement

The refrigerating capacity was measured with a gas calorimeter developed by us, and the input power of the compressor was measured with a wattmeter. Fig. 2 shows the schematic diagram of the gas calorimeter used for efficiency measurements. Table 1 shows the test conditions.

3. RESULTS OF EXPERIMENTAL AND THEORETICAL ANALYSIS

3.1 Roller Clearance Analysis

(1) Experimental results

Fig. 3 shows the P-V diagram with the varying roller clearance. The pressure of the compression process and also the indicated work increase as the clearance increases. Fig. 4 and Fig. 5 show the compressor performance. The indicated efficiency decreases and the volumetric efficiency also decreases, as the clearance increases.

The reasons were thought to be as follows:
(a) Pressure rise during the compression process, and then increase in indicated work as the clearance increased were caused by the oil leakage to the suction side of the cylinder from the inside of the roller through the axial clearance of the rolling piston. The oil in the inside of the roller is under the condition of high temperature and high pressure, and contains refrigerant gas.
(b) Similarly, on account of the oil leakage, the suction gas was reduced in quantity, so that volumetric efficiency became small.

Roller clearance influence on compression efficiency and volumetric efficiency was larger in the rotary compressor used in refrigerators than that used in room air-conditioners. Because, the former leakage area per stroke volume was larger than the latter.

3.2 Discharge Port Analysis

A measured sample of gas pressure in the cylinder, rotation timing and valve behaviour during one rotation is shown Fig. 6. In Fig. 6, pressure variation in the suction process is shown by wave (A), by which wiredrawing loss was estimated. Wave (B) was measured by the pressure pick-up at θ=230°. Gas pressure in the cylinder at the end of the suction process could be measured by wave (B). By wave (C), gas pressure on the discharge process and over-compression loss could be measured. Four kinds of discharge port lengths were used in the experiment. Its conditions are shown in Table 1. The P-V diagram shown in Fig. 3 was obtained by measuring the gas pressure in the cylinder. The calculated results of over-compression loss and wiredrawing loss are shown in Fig. 7.
The following results were obtained by these experimental analysis:
(a) The ratio of over-compression loss against indicated work was about 4.5% - 6%.
(b) The ratio of winding loss against it was about 2.5% - 3.5%.
(c) Over-compression loss and winding loss did not depend on discharge port length.

Next, re-expansion influence for residual gas in the discharge port on compressor performance was investigated. Pressure variation in the suction side of the cylinder and the first stage of the compression process against angular displacement is shown in Fig. 8. Wave (B) shows that pressure in the cylinder suddenly rises immediately after the roller has passed through the discharge port (1 in Fig. 8(b)), because compressed gas in the discharge port was expanding again. In wave (A), it was observed that this sudden pressure rise occurred at about $\Delta \theta = 9^\circ$ delay in angular displacement from wave (B).

In this case, the sound velocity is about 150 m/s because of the use of R-12 refrigerant gas, and the delay angle of rotation $\Delta \theta$ is calculated at about $9^\circ$ by substituting the pressure wave propagating time from position (B) to (A). The results meet the experimental values relatively well.

$$\Delta \theta = \frac{L \cdot N}{a} \quad (1)$$

where

$L$: distance from (A) to (B)
$N$: rotation speed
$a$: sound velocity

In wave (A), its pressure peak was reached just at the time when the roller was passing through position A (2 in Fig. 8(b)).

It was considered that the pressure rise in the cylinder from 1 to 2, $\Delta P_s$, was caused by the flow of the residual gas from the discharge port into the cylinder and then re-expanded again. Fig. 8(a) shows the pressure variation in the cylinder with short discharge port length, while Fig. 8(b) shows that with the long one.

When comparing them, it was found that the pressure rise $\Delta P_s$, at the actual compression starting point (2 in Fig. 8(b)) with short discharge port length was smaller than that with the long one.

Pressure rise $\Delta P_s$ caused by flowing residual gas into the cylinder was expressed by the following equation which was already reported by the authors (3).

$$\Delta P_s = k \cdot P_d \cdot \frac{P_d}{P_s} \frac{1}{k} \cdot \frac{V_{cp}}{V_{st}} \quad (2)$$

where

$k$: Specific ratio
$P_d$: Discharge pressure
$P_s$: Suction Pressure
$V_{cp}$: Top clearance volume
$V_{st}$: Stroke volume

Fig. 9 shows pressure rise $\Delta P_s$ calculated by Eq. (2) compared with experimental values. The results meet the experimental values relatively well. Compressor performance via various discharge port lengths are shown in Fig. 10. It was clarified that residual gas volume increased, and pressure rise became large at the compression starting point as the discharge port length increased. As a result, compressor input power increased, accordingly its efficiency decreased.

4. PERFORMANCE IMPROVEMENT

Authors had already reported a highly-efficient rotary compressor for air-conditioners with efficiency of 72.4%. In this paper, by using the same method as mentioned above, the rotary compressor performance for refrigerators was improved.

Its results are shown in Table 2 and can be summarized as follows:
(1) By improving the discharge port length and the diameter to be optimum, loss could be reduced from 8.1% to 6.8%.
(2) By improving roller and blade clearance to optimum, leakage loss could be reduced from 7.2% to 3.9%.
(3) By improving the diameter and clearance of the main bearing, the sub-bearing and the crank shaft bearing are optimum based on journal bearing analysis. As a result, mechanical loss could be reduced from 10.6% to 8.2%.
(4) Compressor efficiency could be improved from 48.2% to 55.2%. As a result, a highly-efficient rotary compressor for refrigerators could be produced.

5. CONCLUSION

(1) Using a rotary compressor for refrigerators, a maximum compressor efficiency of 55.2% was theoretically and experimentally obtained.
(2) Over-compression loss, winding loss and re-expansion loss could be theoretically and experimentally clarified.
It was found that the roller clearance had great influence on compressor efficiency, i.e. reducing its clearance contributed to increasing compressor efficiency.

REFERENCES

(1) P. Pandeya: "Rolling piston type rotary compressors with special attention to friction and Leakage", '78 Purdue Comp. Tech. Conf.
Fig. 1 Transducer mounting and measuring system

Fig. 2 Structure of Gas Carolimeter (GCM)
Fig. 3 Measured P–V diagram

Table 1 Testing conditions

<table>
<thead>
<tr>
<th>Schedule</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Ps$</td>
<td>0.133 MPa</td>
</tr>
<tr>
<td>$Ts$</td>
<td>305 K</td>
</tr>
<tr>
<td>$Pd$</td>
<td>1.08 MPa</td>
</tr>
<tr>
<td>$100 V, 60$ Hz, 57.7 R.P.S.</td>
<td>Compressor output: 110 W</td>
</tr>
<tr>
<td>Refrigerant</td>
<td>R–12</td>
</tr>
</tbody>
</table>

Fig. 4 Roller clearance effects on compressor performance

Fig. 5 Roller clearance effects on volumetric efficiency
Valve displacement near discharge port center

Rotation timing

Gas Pressure in cylinder

Pressure (MPa)

Valve lift

Fig. 6 Measured Pressure variation and valve behaviour

Fig. 7 Discharge port length effects on over-compression and wiredrawing loss

Fig. 8 Discharge port gas re-expansion effects on suction pressure variation
Fig. 8 Comparison of measured and calculated values of pressure rise

Table 2 Consumption power before and after performance improvement

<table>
<thead>
<tr>
<th></th>
<th>After (%)</th>
<th>Before (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isentropic work</td>
<td>55.2</td>
<td>48.2</td>
</tr>
<tr>
<td>Wiredrawing loss</td>
<td>2.3</td>
<td>2.3</td>
</tr>
<tr>
<td>Over-compression loss</td>
<td>3.8</td>
<td>4.0</td>
</tr>
<tr>
<td>Re-expansion loss</td>
<td>3.0</td>
<td>4.1</td>
</tr>
<tr>
<td>Leakage loss and etc</td>
<td>3.9</td>
<td>7.2</td>
</tr>
<tr>
<td>Mechanical loss</td>
<td>8.2</td>
<td>10.6</td>
</tr>
<tr>
<td>Motor loss</td>
<td>23.6</td>
<td>23.6</td>
</tr>
<tr>
<td>Total</td>
<td>100.0</td>
<td>100.0</td>
</tr>
</tbody>
</table>

Fig. 10 Discharge port length effect to compressor performance