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M. Fujiwara
Muroran Institute of Technology

T. Kazama
Muroran Institute of Technology

PERFORMANCE IMPROVEMENT OF A RECIPROCATING AIR MICROCOMPRESSOR

Mitsuru Fujiwara and Toshiharu Kazama
Muroran Institute of Technology, Hokkaido, Japan

ABSTRACT

A previously developed reciprocating air microcompressor having a 1-mm(0.04 in)-diameter piston, an inlet port that did not have a moving valve, and an outlet port equipped with a miniature reed valve, was capable of a maximum discharge pressure of 80 kPa (gage, 11.6 psig) at a piston frequency of 100 Hz. We have improved on this compressor performance, and have attained a maximum discharge pressure of 100 kPa (gage, 14.5 psig) with a volumetric efficiency of 90 % by supplying oil to the cylinder. The amount of the oil needed was as small as one drop per hour. By analyzing the compressor performance while focusing on the effect of leakage, we found that the performance improvement was due to the sealing effect of the oil preventing leakage through the discharge reed valve.

INTRODUCTION

In recent years many studies have been done on miniaturized fluid machinery and related components[1,3,5]. However, little work has been done on microcompressors. Development of a practical microcompressor is important, though, because it will be a step towards realizing micro-pneumatic systems and micro-refrigerators.

Positive-displacement microcompressors are the most promising because the velocity of working fluid must be low in miniaturized machines. We previously developed a reciprocating air microcompressor having a 1-mm(0.04 in)-diameter piston[2]. However, the volumetric efficiency was significantly lower than that predicted by a numerical simulation. The lower efficiency was probably caused by internal leakage, but, the specific causes were not clear.

In this paper, possible causes of the lower efficiency are investigated. As a measure to improve performance, we supplied oil to the cylinder and the effectiveness of this measure is also discussed.

DESIGN AND FABRICATION OF THE MICROCOMPRESSOR

The compressor used in the present study was designed and fabricated fundamentally as described in our previous paper[2]. We briefly review the main points here.

A schematic view of the microcompressor is shown in Fig. 1. The piston diameter is 1 mm(0.04 in) and the stroke is 3 mm(0.12 in). The inlet port is located on the cylinder bore near the bottom and does not have a moving valve. The port is closed when the piston covers it and is opened when the piston's top edge is below the top of the port contour. The outlet port is equipped with a miniature reed valve and this port is located at the top of the cylinder . The reed valve is

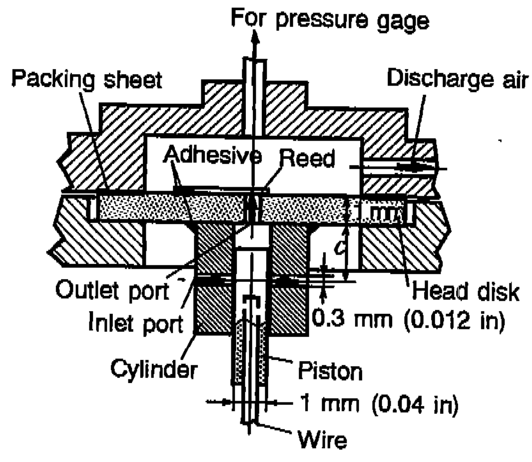


Figure 1 Schematic view of microcompressor

opened and closed in response to the pressure in the cylinder. The advantages of this design is its simple fabrication and ample space for the port mountings.

The cylinder is made of stainless steel and was fabricated with a wire-cut electric-spark machine. The needle of a commercial rolling-contact bearing was used as the piston. The mean radial clearance is about 6 μ m (0.24 μ in), as calculated from the measured diameters of the piston and the cylinder bore. The reed of the outlet valve is made of 15-μ m (0.59 μ in)-thick polyethylene film.

Before the test, the expected volumetric efficiency was determined theoretically using a numerical model. The following factors were considered in the model:

(1) volume change due to piston movement
(2) air entering and leaving through the inlet and outlet ports
(3) air leakage through the clearance between the piston and cylinder.

Valve dynamics and leakage through the reed valve were not taken into account. Mass flow through the inlet and outlet ports was calculated using the equation of flow through an orifice. The leakage mass flow through the clearance between the piston and cylinder was calculated using Grinnell's formula[4] as follows:

$$m = \frac{bh^3}{24\mu RTl}\left(P_1^2 - P_2^2\right) \qquad (1)$$

where,

- m: mass flow rate
- b: width of clearance
- h: height of radial clearance
- μ : viscosity
- R: gas constant
- T: temperature
- l: clearance length
- P_1: entrance pressure
- P_2: outlet pressure

The theoretical performance of the prototype compressor was calculated using the values shown in Table 1, and the results are shown in Fig.2. The volumetric efficiency decreases as the

Table 1 Values used for the calculation

Piston diameter	1 mm (0.04 in)
Stroke	3 mm (0.12 in)
Inlet port diameter	0.3 mm (0.012 in)
Outlet port diameter	0.5 mm (0.02 in)
Radial clearance	6 μm (0.24 μin)
Piston frequency	40, 60, 80, and 100 Hz
Clearance volume ratio	0.12
Suction pressure	0 kPa(gage, 0 psig)
Discharge pressure	20, 40, 60, 80 and 100 kPa(gage) (2.9, 5.8, 8.7, 11.6 and 14.5 psig)

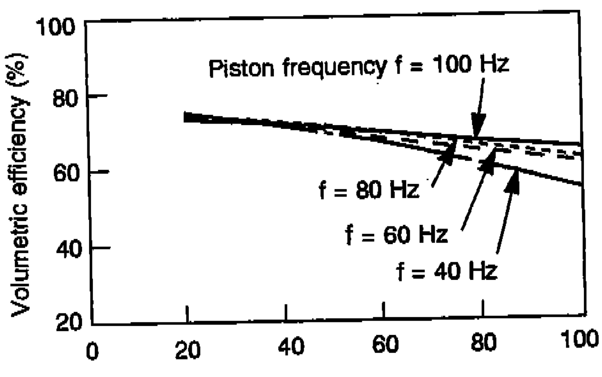


Figure 2 Calculated volumetric efficiency of the microcompressor from the simulation

discharge pressure increases. This is especially so at lower piston frequencies and higher discharge pressures, because the leakage mass increases in proportion to the time required for the compression cycle. Thus leakage is an especially important factor affecting volumetric efficiency. Nonetheless, our simulation indicated that a discharge pressure exceeding 100 kPa (gage, 14.5 psig) is attainable with this compressor.

PERFORMANCE TEST

The details of the measurement apparatus and the method used for the performance test were described in our previous paper[2]. The piston was oscillated sinusoidally by a cylindrical cam with a flat-face follower driven by a DC motor. The rate of discharged air volume was determined by replacing the water in the measurement cylinder. The tests were performed under either of two conditions: oil-free or with oil supplied.

The test results for the first case (without oil) are shown in Fig.3 for various piston frequencies. The volumetric efficiency decreased significantly as the discharge pressure rose and the piston frequency was reduced. This indicates that leakage is the dominant factor affecting the performance of the compressor. The maximum discharge pressure attained under the oil-free condition was 80 kPa (gage, 11.6 psig). This was much lower than what was predicted from the numerical simulation.

Fig.4 shows the volumetric efficiency when oil was supplied to the cylinder. The amount of oil was as small as one drop per hour. The efficiency was greatly improved compared with that in Fig.3, and exceeded 80 % over the entire ranges of frequency and discharge pressure in these tests. A maximum discharge pressure of 100 kPa (gage, 14.5psig) was obtained with a volumetric efficiency of 90 %.

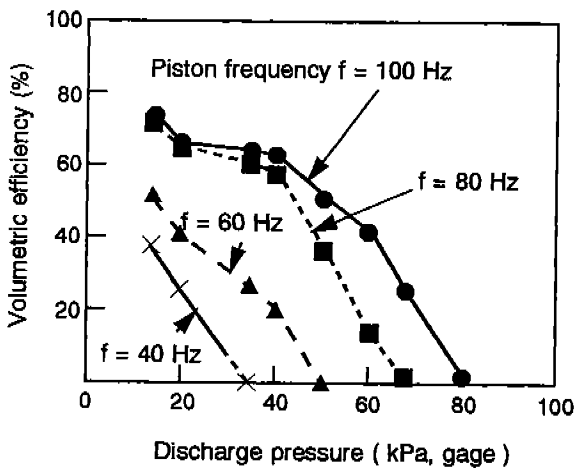


Figure 3 Experimental volumetric efficiency at various piston frequency (without oil)

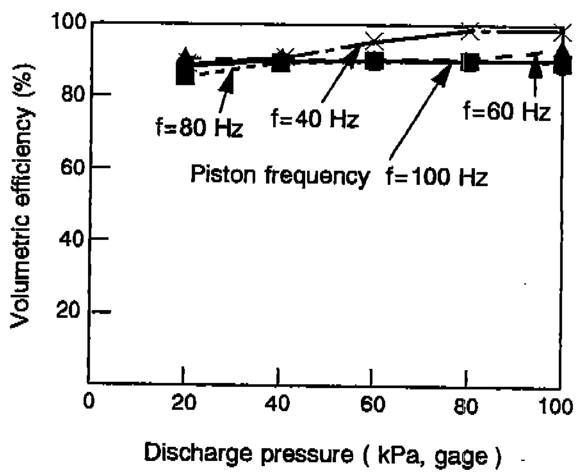


Figure 4 Experimental volumetric efficiency when oil is supplied

DISCUSSION

The numerical model we used assumed oil-free operation, but, our test results under the oil-free condition differed considerably from the numerical prediction.

The following may explain this disagreement:

(1) Grinnell's formula may not be valid for calculating the leakage mass flow rate through the clearance between the piston and cylinder.
(2) The leakage through the discharge reed valve may not be negligible and may greatly affect the efficiency.

We evaluated the validity of the reasons as follows.

First, we checked Grinnell's formula through a component test. Fig.5 schematically shows the leakage-test apparatus. High-pressure air was introduced into one end of a test passage. The leakage air volume was determined in the same way as in the compressor performance test. Two sample passages were used for the tests. The height of the radial clearance in these samples, calculated from diameters measured with a micrometer and a cylinder gage, is shown with length l in Table 2.

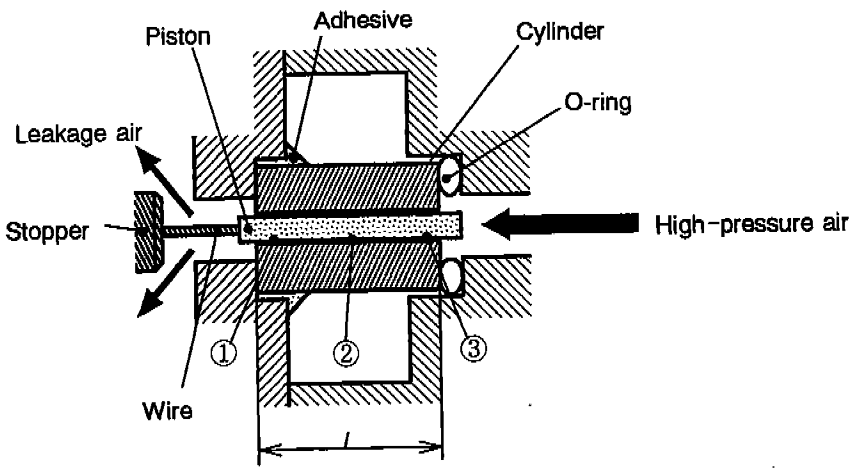


Figure 5 Test of leakage through the clearance between piston and cylinder

Table 2 Height of radial clearance calculated from measured diameters

Sample No.	l (Fig.5)	Position (Fig.5) ①	②	③	Mean
1	5.00 mm (0.197 in)	0.0047 mm (0.00019 μin)	0.0045 mm (0.00018 μin)	0.0047 mm (0.00019 μin)	0.0046 mm (0.00018 μin)
2	6.00 mm (0.236 in)	0.0062 mm (0.00024 μin)	0.0055 mm (0.00022 μin)	0.0068 mm (0.00027 μin)	0.0062 mm (0.00024 μin)

The results of the leakage tests are shown in Fig.6. Based on Grinnell's formula, the equivalent height of the radial clearance was calculated from the leakage rates for each sample passage. The height of the radial clearance is given by,

$$h = \sqrt[3]{\frac{24\mu RTlm}{b\left(P_1^{2} - P_2^{2}\right)}} \tag{2}$$

The calculated heights were 0.0040(0.00015 in) mm for sample 1 and 0.0048(0.00019 in) mm for sample 2. Comparing these values with the values in Table 2, we can see that Grinnell's model is valid for calculating the mass of leakage through the clearance.

For the second possibility, we studied the effect of valve leakage on the compressor performance by numerical simulation. We assumed that there was a small clearance between the reed and the valve seat (Fig.7), even when the valve should have been tightly closed. We call this the reed-valve clearance.

We calculated the effect of the leakage on the volumetric efficiency. In this model, the leakage flow rate was given by the following equation:

$$q_m = CA\sqrt{2\rho(P - P_d)} \qquad (3)$$

where,

q_m: mass flow rate
C: discharge coefficient
A: leakage area
ρ: density in cylinder
P: pressure in cylinder
P_d: pressure in discharge chamber

C is assumed to be 0.6 and A is defined by,

$$A = \pi d_d h \qquad (4)$$

where,

d_d: discharge port diameter
h: clearance height between reed and valve seat

The volumetric efficiencies were calculated for oil-free operation with a constant piston frequency of 100 Hz and various clearance heights (Fig.8). The clearance height greatly affects the compressor performance. This suggests that the greatly improved volumetric efficiency when oil was supplied was mainly caused by reduced air leakage through the reed valve clearance due to the sealing effect of oil.

The oil around the reed valve is thought to behave as shown in Fig.9. When the valve is open and the air is discharged, the oil may stick to the reed and the valve seat, and not be

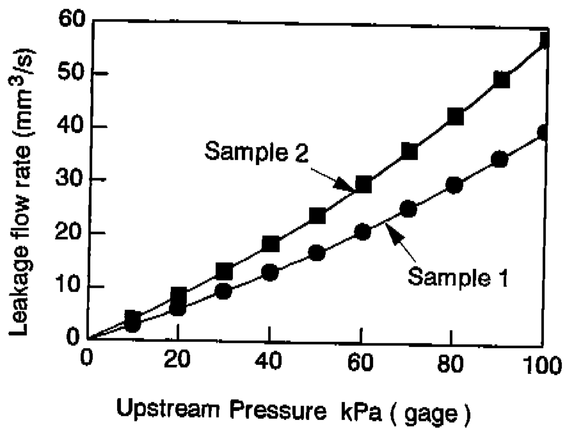


Figure 6 Results of leakage test

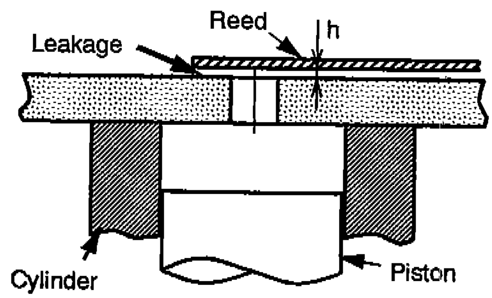


Figure 7 Reed valve clearance

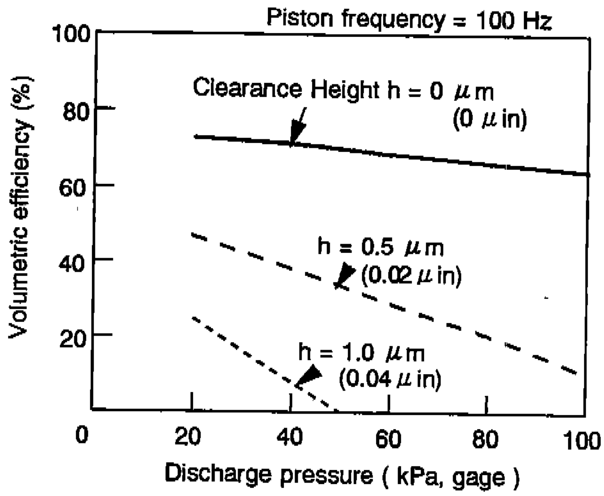
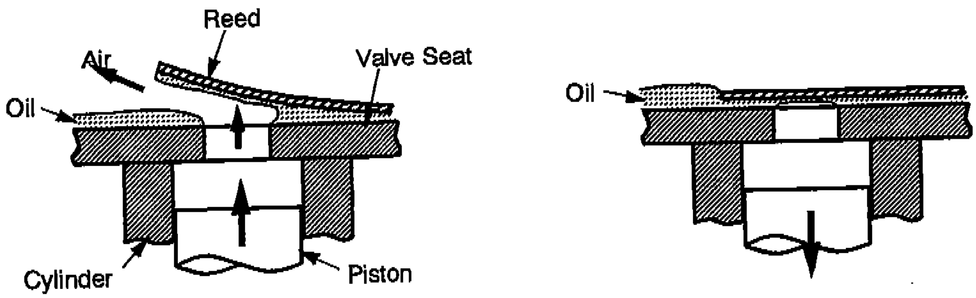


Figure 8 Effect of reed valve clearance on volumetric efficiency (simulation)

blown away. But, when the valve is closed, the reed valve clearance becomes very small and the oil cannot flow back into the cylinder because the oil's viscosity is very high. Thus, the oil can stay around the reed for a long time. The persistence of the oil appears to be a size effect of the small construction.

On the other hand, the oil in the clearance between the piston and cylinder is likely to quickly flow out. Therefore, the sealing effect of the oil there would quickly be eliminated.



(a) When the valve is open. (b) When the valve is closed.

Figure 9 Behavior of oil in the reed valve

CONCLUSIONS

The performance of a reciprocating air microcompressor was improved by supplying oil to the cylinder. A maximum discharge pressure of 100 kPa (gage, 14.5 psi) with a volumetric efficiency of 90 % was obtained even when the amount of oil was as small as one drop per hour. We attribute the improvement to the sealing effect of the oil preventing air leakage through the reed valve.

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REFERENCE

[1] Fuhr,G., Schnelle,T. and Wagner,B., Travelling wave-driven microfabricated electro-hydrodynamic pumps for liquid, Journal of Micromechanics and Microengineering, Vol.4, No.4, 1994, pp.217-226

[2] Fujiwara,M., Kazama,T. and Gunji,Y., Reciprocating air microcompressor, Proceedings of the 1996 International Compressor Engineering Conference at Purdue, 1996, pp.43-48

[3] Goll,C., Bacher,W., Büstgens,B., Maas,D., Menz,W. and Schomburg,W.K., Microvalves with bistable buckled polymer diaphragms, Journal of Micromechanics and Microengineering, Vol.6, No.1, 1996, pp.77-79

[4] Grinnell,S.K., Flow of a compressible fluid in a thin passage, Transactions of the ASME, May, 1956, pp.765-771

[5] Ilzhöfer,A., Ritter,B. and Tsakmakis,Ch., Development of passive microvalves by the finite element method, Journal of Micromechanics and Microengineering, Vol.5, No.3, 1995, pp.226-230