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A STUDY ON THE PERFORMANCE IMPROVEMENT OF A ROTARY COMPRESSOR

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ABSTRACT

Of many elements influencing the performance of a rotary compressor, the valve system plays an important role. In order to evaluate the performance, some basic experimental studies were made on the discharge coefficients and the proper dimensions of the valve system. Also, since the valve systems have previously been studied by using rather simplified models, some efforts were made in this study to describe the valve system with more refined model. A valve stopper, two-plated valve and a valve port are modeled as contacting elements in finite element analysis. Also initial configuration of the upper valve plate is considered. Results of the analysis were the deflection of a valve plate as a function of a rotation angle, mass flow history through the valve port, capacity of a compressor, the velocity losses, compressing work and the quantity of back flow.

1. INTRODUCTION

In order to evaluate the performance of a rotary compressor, the valve system and compressing mechanism should be considered.[1,2] The movement of a reed valve is influenced by many conditions: the valve port type, the shape of the valve plate, the height of the valve stopper and etc. In other words, the dynamic behavior of the reed valve is determined by its mass, stiffness, initial deflection and forces acting on it according to the behavior of a reed valve. [3]

Some characteristics of valve ports of various shapes for performance evaluation should be determined by experiments. Mass flow rate and forces on valve are determined by the port type. Also, two-plated valve has an advantage in sealing tightly the valve port before opening the valve because of the flexibility of a lower valve plate and the pushing force of a upper valve plate toward the valve port. For these reasons, two-plated valve is normally used for a rotary compressor. While the valve plate is in contact with the valve stopper, the stopper must be deflected. These two problems can be analyzed by modeling them as contacting element. [4]

2. BASIC EXPERIMENTAL STUDIES ON VALVE PORT SYSTEM

In order to evaluate the performance (the capacity, compressing work, velocity loss, Kal/W) of the compressor, the proper shapes and dimensions of the valve and port system should be determined in advance. The proper dimensions of the valve port and stopper can be studied by the experiments on discharge coefficients under various conditions. In case of considering pressure loss in compressor valves, steady one-dimensional flow as in ducted flow is usually assumed.[5] In this experiment, the maximum velocity through a model valve port is 87m/s and so equations of incompressible flow can be used. Because in this range there are no difference between valves based on incompressible flow theory and those based on compressible flow theory, which means that one may assume the density of the fluid be constant. Discharge coefficient C_d is defined by the ratio of the effective mass flow rate $\dot{m}$ and the theoretical mass flow rate $\dot{m}_{th}$.

$$C_d = \frac{\dot{m}}{\dot{m}_{th}} \qquad \dot{m}_{th} = A\sqrt{2\rho(P_1 - P_2)}$$

A : port area
$P_1 - P_2$: pressure difference
ρ : density

2.1. Experimental set-up and performance

The real size of a valve port is so small that many difficulties are expected in experiments. Thus large models are made for easy manipulation. And air is used instead of refrigerant. The problems arising due to the difference between a real model and a testing model, can be handled by similitude. Reynold number Re is generally used. The kinematic viscosity of air at $20^{o}C$ and at 1 bar is $1.5\times10^{-5}m/s$ and that of refrigerant at $125^{o}C$ and at 12 bar is $0.3498\times10^{-6}m/s$. However, in actual compressors, the refrigerant is mixed with the lubricant. This makes the refrigerant more viscous. Transition from laminar and turbulent boundary layer occurs at $Re_{crit}=0.333\times10^{5}$ depending on general turbulent level in oncoming flow. Surface roughness of the testing model should not affect Re_{crit}. A schematic diagram of the experimental set-up is shown in Fig.1.

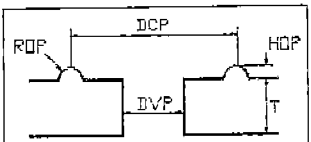
E1 is made to consider the effect of a cylinder. In real compressor, the valve port is partly covered with the cylinder. E2 handles the distance between a valve port and a valve plate. The handling equipment is a micrometer. E3 measures the mass flow rate. From the differance between the total pressure and the static pressure measured at one point in the duct, one can calculate the velocity at that point. In the same way, one can calculate at other points over the cross section and by integrating velocity over the cross section, one can calculate the the mass flow rate. To analyze the experimental data, a simple computer program was developed. The pressure is measured by the digital manometer which is connected with the A/D convertor to transfer the signal to the microcomputer. The signal is sampled 300 times at $0.5sec$ rate. These data are used after averaging. The pressure at each side of a valve port is obtained in the same way. This discharge coefficient, the average velocity, Reynold's number and things like that can be obtained. Table 1 shows dimensions of testing valve port models.

2.2 Discharge coefficient and proper shape

The discharge coefficients corresponding to valve port types and distance between the valve port and the valve plate are shown in Fig.2, Fig.3, and Fig.4. The calculation of the theoretical mass flow rate was based on the port areas. Fig.2 shows that there exists optimal value of the diameter of contacting part (DCP). The comparisons according to the ratio of port area and port plate thickness are shown in Fig.3. Except the case that the thickness is so thin that the port plays like another orifice, the thicker the valve plate is, the more the pressure loss is. Fig.4 shows the discharge coefficients as a function of the radius of protrusion (ROP). The discharge coefficient of port 6 compared with that of port 1 is lower when the distance is below $1.7mm$ and over $2.7mm$. This is very disadvantageous condition. During most of valve opening time, a valve plate contacts with the valve stopper. This indicates the discharge coefficient near the valve stopper height is important. And Bernoulli effect plays greater role in port 6 than in port 1 because port 6 has greater radius of protrusion.

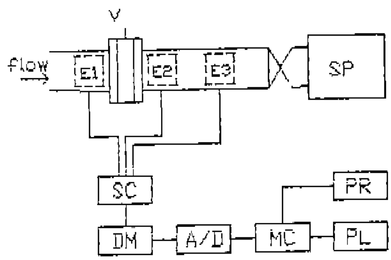
In the real compressor, a valve port is slightly covered with the guttered cylinder to discharge the

Table.1 Dimensions of testing valve port models



DVP : diameter of valve port (mm)
DCP : diameter of contacting part (mm)
T : thickness of valve port plate (mm)
ROP : radius of protrusion (mm)
HOP : height of protrusion (mm)

Port NO.	DVP	DCP	T	ROP	HOP
Port 1	26	60	30	3	5
Port 2	26	55	30	3	5
Port 3	26	65	30	3	5
Port 4	23	60	30	3	5
Port 5	26	60	39.3	3	5
Port 6	26	60	30	6	6



V : Testing compressor valve
E1 : Equipment of cylinder
E2 : Equipment of controlling valve lift
E3 : Equipment of measuring mass flow rate
SP : Suction pump
SC : Channel selector
DM : Digital manometer
A/D : A/D converter
MC : Microcomputer
PR : Printer
PL : Plotter

Fig.1 Schematic diagram of experimental set-up

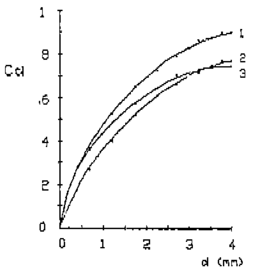


Fig.2 Discharge coefficient (port 1, port 2, port 3)

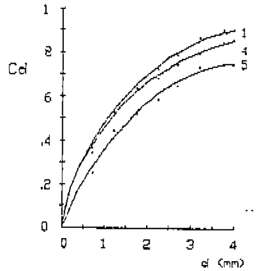


Fig.3 Discharge coefficient (port 1, port 4, port 5)

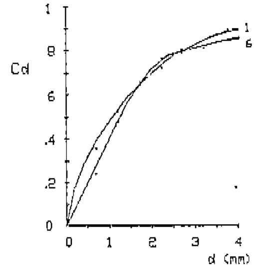


Fig.4 Discharge coefficient (port 1, port 6)

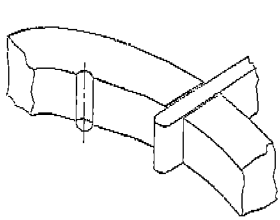


Fig.5 Setting of port and cylinder

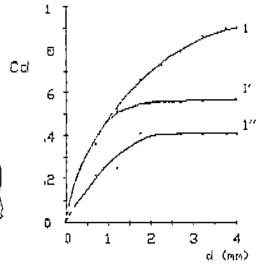


Fig.6 Discharge coefficient (port 1, port $1'$, port $1''$)

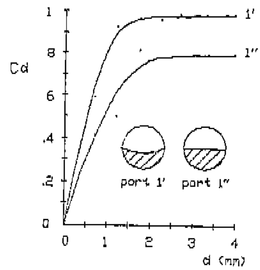


Fig.7 Discharge coefficient (port $1'$, port $1''$)

remaining gas as shown in Fig.5. And guttered part should not influence the clearance volume significantly. Fig.6 shows the discharge coefficients of port 1, port $1'$ and port $1''$ based on the flow area of port 1. Again represented are the discharge coefficients of port $1'$ and port $1''$ based on each flow area in Fig.7. One can notice that port $1'$ is better than port $1''$ and that the shape of the cylinder is important. One, of course, must consider the covered valve port to determine the height of a valve stopper. The discharge coefficients measured in experiment are used to estimate the performance of a compressor.

3. DYNAMIC ANAYSIS OF A REED VALVE

Two-plated valve, the valve stopper and the valve port can be modeled as in Fig.8. Upper truss elements have material nonlinearity and gap. So within the gap the truss elements have no action to the beam elements but beyond the gap, the truss elements act as a stopper. Likewise valve port is modeled as two lower truss elements. The difference between the fixing position and the position of valve port seat can be considered by using gap. Two plates are modeled with beam elements and the contact elements. Such model can be handled by ADINA (Automatic Dynamic Incremental Nonlinear Analysis) program. Pressure upon valve plate according to the behavior of valve plate were supplied by subroutine IUSER and USERSL. Some calculated results are shown in Fig.9. The number of contacting time is different in two cases. It should be small to avoid the valve plate distortion and impact fatigue. It should be considered in determining stiffness and preload of the valve plate. And also the following results are obtained. Fig. 10 shows the flow chart of the algorithm.

1) The effect of friction between two plate is negligible.
2) The deformation of a valve stopper is negligible

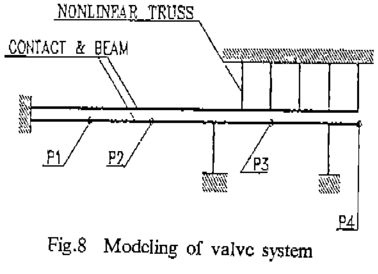
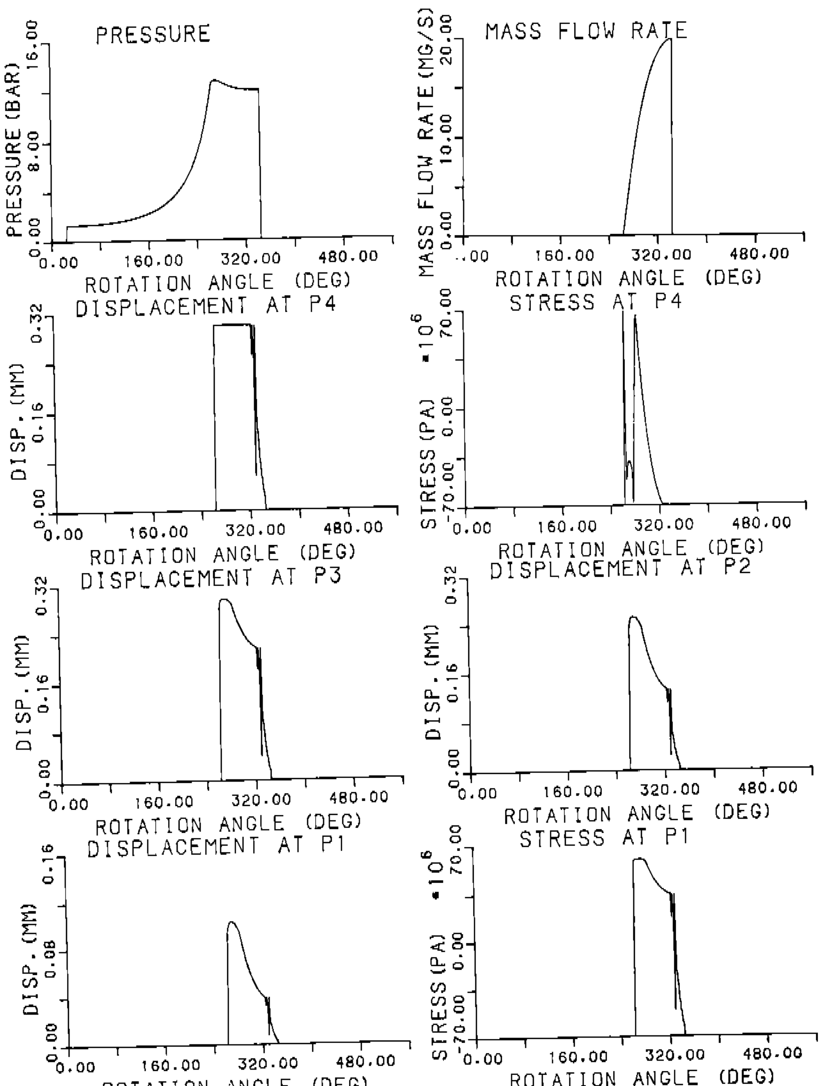
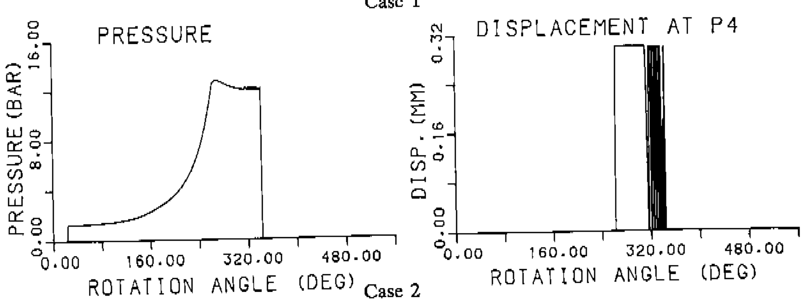


Fig.8 Modeling of valve system



Case 1



Case 2

Fig.9 The results of analysis on valve plates

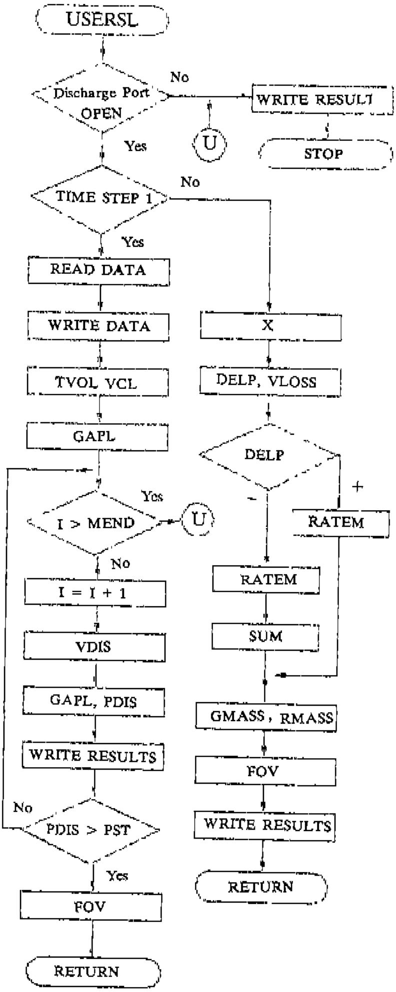


Fig.10 Flow chart of the algorithm

Table 2 Comparison between port 1 and port 2

Port NO.	Capacity (Kcal/h)	Comp. work (Watt)	Vel. loss (Watt)
port 1	139.085	94.598	1.2896
port 2	139.823	95.142	1.6362

Table 3 Effect of valve port size

Dia. of port (mm)	Capacity (Kcal/h)	Comp. work (Watt)	Vel. loss (Watt)	Capacity Work
2.0	143.13	96.548	2.992	1.4379
2.3	141.24	95.786	2.141	1.4423
2.6	139.09	94.598	1 290	1.4506
2.9	136.66	93.903	1.121	1.4381
3.2	133.97	93.477	0.941	1.4189

Table 4 Effect of preload

Preload (N)	Capacity (Kcal/h)	Comp. work (Watt)	Vel loss (Watt)
0.2	139.085	94.590	1.2856
0.5	139.085	94.598	1.2896
0.8	139.085	94.611	1.2936
0.11	139.085	94.629	1.2986

4. DISCUSSIONS ON THE OPTIMAL VALVE SYSTEM

4.1 Valve port type

Energy losses and velocity losses are affected by the valve port. These losses also cause the compressor to increase the compressing work. The comparision between port 1 and port 2 defined in Table 1, is shown in Table 2. Compressing work shown in Table 2 is the value obtained without considering the effect of friction in compressing mechanism and efficiency of the motor. So in real case, more compressing work is needed and the difference between two cases can be lager. Since port 2 has a lower discharge coefficient, it needs more compressing work and has greater velocity losses.

4.2 Valve port size

The larger the valve port size is, the more the compressing work and the velocity losses reduce greatly. But if one makes a valve port area larger than a certain limit, the efficiency of the compressor decreases. Because increasing the valve port area means increasing the clearance volume. This is why the optimal valve port size exists. It can be calculated only with the help of the computer simulation. Table 3 shows the calculated data.

4.3 Determination of preload

The reed valve consists of two plates. Its fixing position is lower than the port seat. The lower valve plate should be pressed down by the upper plate to contact with the valve seat perfectly. To do that, the upper valve plate has to be curved. The efficiency for various preload is given in Table 4. As preload increases, compressing work increases. However it can be noted that the effect of preload is very little. It can be easily understood when one think the compressor is working in high pressure condition. But it affects the behavior of valve plate, that is, it affects the number of contacting time of a valve plate. It is closely related to the distortion of the valve plate, which reduce the performance of the rotary compressor. And the force by the upper valve plate is applied on the valve tip. If one increases the pressing force, the valve plate will contact the other part of the valve port seat. Let this critical force as F_{min}. And if one continues to increase the force, there will be gap between the valve plate and the valve port seat again. This limiting force is defined as F_{max}. Using ADINA, one can find out that $F_{min} = 0.17N$ and that $F_{max} = 1.8N$. Considering the behavior of valve plate and Table 4, $0.5N$ is recommended within that range. In fabricating the valve plate, it is needed that one should calculate the radius of curvature of the upper valve plate to have $0.5N$ preload. $60mm$ is the calculated radius of curvature of the valve plate using ADINA.

5. CONCLUSIONS

Some basic experimental studies were carried out to determine the proper shapes and dimensions of valve and port system of the rotary compressors. Also the dynamic analysis of the valve plates were presented by employing complex contact elements. Finally the performance of the compressor were numerically predicted for various valve and port dimensions.

ACKNOWLEDGEMENTS

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