A Study for Improvement on High Pressure Multistage Reciprocating Compressor

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A STUDY FOR IMPROVEMENT ON HIGH PRESSURE MULTISTAGE RECIPROCATING COMPRESSOR

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ABSTRACT

Characteristics for the multistage high pressure reciprocating compressor, which obtains over 30MPa, have been studied. The simulation of flows in the connecting pipes between cylinder heads and the inside of them are performed by using Computational Fluid Dynamics, in which the compression condition is assumed one dimensional, compressible and unsteady. Then the performance characteristics are comprehensively studied by comparing the numerical results with those of experiments. Consequently, an improvement in thermal efficiency of the compressor has been accomplished.

NOMENCLATURE

A	Area	P	Pressure
a	Sound velocity	Q	Leakage flow rate
c	Damping coefficient	q	Quantity of heat
F	Fluid force ($= \Delta P \cdot A$)	r	Cylinder radius
H	Cylinder height	t	Time
k	Spring constant	T	Gas temperature inside cylinder
L	Length	u	Velocity
m	Valve mass	V	Cylinder Volume
M	Mass of gas in cylinder	X	Axial coordinate of pipe
n	Polytropic exponent	x	Displacement
Greek Letters			
δ	Side clearance between cylinder and piston	μ	Coefficient of friction
		ρ	Density
κ	Specific heat ratio	τ	Shearing stress ($= \mu \cdot (du/dr)$)
Subscripts			
cyl	Inside of cylinder	*s*	Suction
d	Discharge	*top*	Top dead center
pipe	Inside of pipe	*0*	Initial value
port	Suction and Discharge port	*valve*	Minimum sectional area

INTRODUCTION

The multistage reciprocating compressors have potential in attaining higher pressure with higher thermal efficiency than single stage ones because their cylinders are arranged in series so that they could be applied widely in many industries. However, the improvement of these kinds of compressors is still demanded to meet the stringent requirements for better performance. For example, in order to increase the gas flow rate for one compressor, a higher volumetric efficiency is most important. Because the higher the volumetric efficiency, the less the suction flow losses and the better the utilization of pressure pulsation occurring at the suction-discharge system. Many research works have been reported on this topic in reciprocating internal combustion engines, which have similar induction structure and working sequence with reciprocating compressors. However, rarely literatures on such topic are available for multistage compressors till now.

In this report, the influence of discharge pressure and the effect of working fluid on the four stage reciprocating compressor are conducted by using the original numerical simulation program. The possibility of applying this simulation program to estimate the compressor performance is also verified. Furthermore, the internal flow of the connecting pipe and the cylinder head are analyzed, and the higher thermal efficiency compressor has been accomplished using this program.

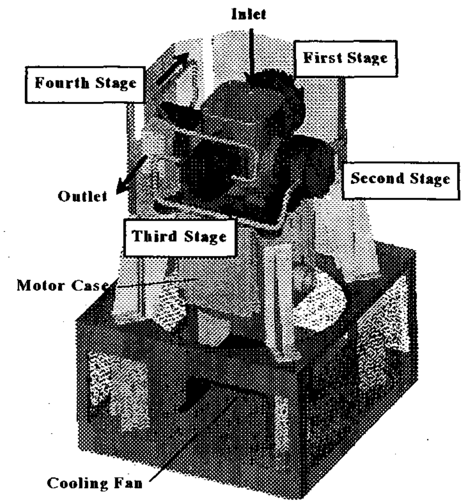

Figure 1. Solid model of the compressor

MODEL OF CONSTRUCTION ELEMENTS

To analyze comprehensively, the simulation model for performance characteristics of four-stage compressor is divided into three elements: the compression space, the suction-discharge system and the connecting pipe. The solid model of the compressor is shown in Figure 1, and the flow diagram of main routine is shown in Figure 2.

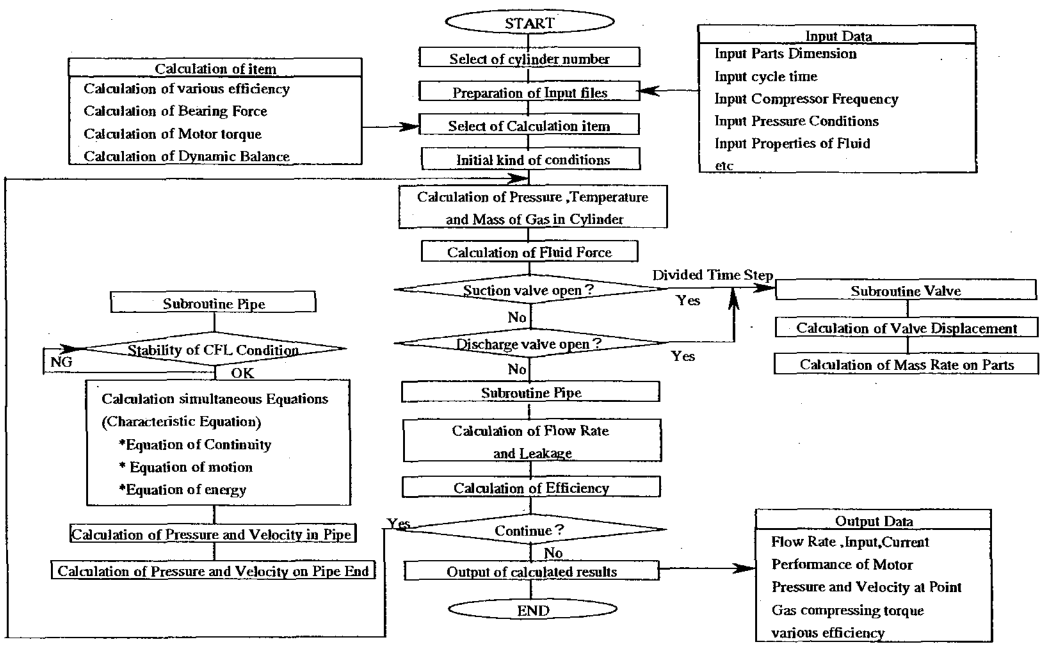

Figure 2. Flow diagram of main routine

Model of Compression Space

In the simulation model of the compression space, a variation of the cylinder volume is calculated from the piston position, then a discharge pressure and temperature are calculated using given suction pressure and temperature in the each process of intake, compression and discharge respectively, shown as Equations (1) and (2). The compression condition for simulation is treated as the polytropic compression. And it is assumed that the re-expansion of the gas in the clearance volume into the cylinder makes the internal pressurizing of the cylinder pressure during the intake process. Accordingly, an increase in the internal pressure ΔP is given by Equation (3).

$$P_{cyl} = \left(\frac{M}{V_{cyl}\,\rho_0}\right)^n \quad (1) \qquad T_{cyl} = T_0\left(\frac{P_{cyl}}{P_0}\right)^{\frac{n-1}{n}} \quad (2) \qquad \Delta P = P_s\left\{\frac{V_{top}(\rho_d - \rho_s)}{(V_0 + V_{top})\rho_s}\right\} \quad (3)$$

Model of Suction-Discharge System

Figures 3, 4 and 5 show structure of the suction-discharge system, suction valve geometry and discharge valve geometry respectively. As illustrated in Figures 3 and 4, calculating area is divided into three domains and four domains respectively. In the calculation, the pressure and the density are assumed to be constant at each domain. And the position of the valves is calculated by using the New Mark β method, which is based on the equation of motion shown as Equation (4), in consideration of a spring constant k and a damping coefficient c.

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = F \quad (4)$$

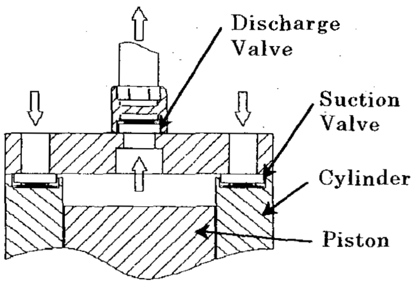

Figure 3. Structure of suction-discharge system

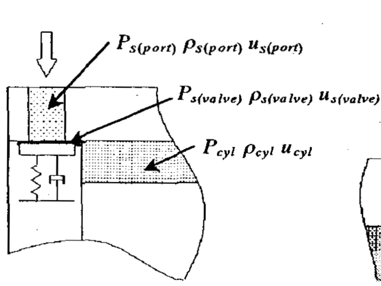

Figure 4. Suction valve geometry

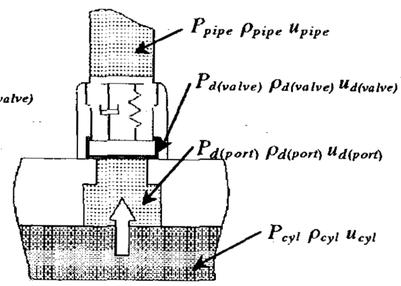

Figure 5. Discharge valve geometry

The pressure, the flow velocity and the other properties around the periphery of valves are obtained from the calculated results of the gas flow in the cylinder and they are defined as the time function. Since the mass flow rate per unit time is constant at each domain of the suction-discharge system, we can get the equation as following.

The equations of continuity

<Suction Intake>
$$\rho_{cyl}\frac{dV_{cyl}}{dt} = \rho_{s(valv)}u_{s(valve)}A_{s(valve)} \quad (5)$$

<Discharge>
$$\rho_{cyl}\frac{dV_{cyl}}{dt} = \rho_{d(port)}A_{d(port)}u_{d(port)} \quad (6)$$

The equations of energy

<Suction Intake>
$$\frac{\kappa}{\kappa-1}\frac{P_{cyl}}{\rho_{cyl}} = \frac{\kappa}{\kappa-1}\frac{P_{s(valve)}}{\rho_{s(valve)}} + \frac{1}{2}u_{s(valve)}^2 = \frac{\kappa}{\kappa-1}\frac{P_{s(port)}}{\rho_{s(port)}} + \frac{1}{2}u_{s(port)}^2 \quad (7)$$

<Discharge>
$$\frac{\kappa}{\kappa-1}\frac{P_{cyl}}{\rho_{cyl}} = \frac{\kappa}{\kappa-1}\frac{P_{d(port)}}{\rho_{d(port)}} + \frac{1}{2}u_{d(port)}^2 = \frac{\kappa}{\kappa-1}\frac{P_{d(valve)}}{\rho_{d(valve)}} + \frac{1}{2}u_{d(valve)}^2 \quad (8)$$

The isoentropic equation

<Suction Intake>
$$\frac{P_{s(port)}}{\rho_{s(port)}^{\kappa}} = \frac{P_{s(valve)}}{\rho_{s(valve)}^{\kappa}} \quad (9)$$

<Discharge>
$$\frac{P_{cyl}}{\rho_{cyl}^{\kappa}} = \frac{P_{d(port)}}{\rho_{d(port)}^{\kappa}} = \frac{P_{d(valve)}}{\rho_{d(valve)}^{\kappa}} \quad (10)$$

Model of Connecting Pipe

With the assumption of the compressible unsteady flow in the connecting pipe, the simulation is performed by using the method of characteristic. The equations of continuity, motion and energy, which are transformed into the characteristic differential equations, are expressed as follows.

$$dP - a^2 d\rho = (\kappa - 1)q\rho dt \quad (11) \qquad dP + \rho a du = (\kappa - 1)q\rho dt \quad (12) \qquad dP - \rho a du = (\kappa - 1)q\rho dt \quad (13)$$

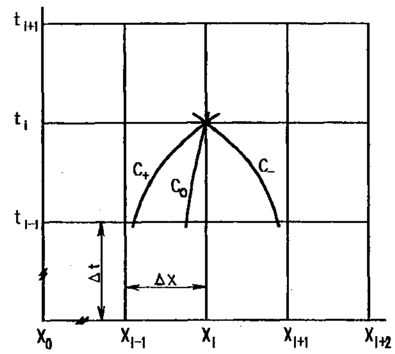

Figure 6. Meshing example of *X-t plane*

The values among the mesh points in *X-t* plane are obtained by proportion, and the pressure, the flow rate and the density in the pipe are given by using Equations (11), (12) and (13) at every time step. Figure 6 shows the meshing example of *X-t* plane. The boundary conditions at the end of the connecting pipe are adopted by both the isoentropic equation and the conservation equation of energy for the compressible quasi-steady flow.

Case 1 : Outflow from Cylinder to Connecting Pipe

In order to consider the dynamic characteristic of the valves, the position that have the minimum sectional area, which is calculated in the model of the suction-discharge system, is set as the reference point. Therefore, considering the two processes: from the discharge port to the minimum sectional area and from the minimum sectional area to the end of the connecting pipe, the equation of continuity, the conservation equation of energy and the isoentropic equation are expressed as follows.

$$\rho_{d(valve)}\, u_{d(valve)}\, A_{d(valve)} = \rho_{pipe}\, u_{pipe}\, A_{pipe} \tag{14}$$

$$\frac{\kappa}{\kappa-1}\frac{p_{d(port)}}{\rho_{d(port)}} = \frac{\kappa}{\kappa-1}\frac{p_{d(valve)}}{\rho_{d(valve)}} + \frac{1}{2}u_{d(valve)}^2 = \frac{\kappa}{\kappa-1}\frac{p_{pipe}}{\rho_{pipe}} + \frac{1}{2}u_{pipe}^2 \tag{15}$$

$$p_{d(port)} \big/ \rho_{d(port)}^{\kappa} = p_{d(valve)} \big/ \rho_{d(valve)}^{\kappa} \tag{16}$$

Case 2 : Inflow from Connecting Pipe to Cylinder

On the condition for the inflow from the connecting pipe to the cylinder, the equation of continuity, the conservation equation of energy and the isoentropic equation are expressed as follows.

$$\rho_{pipe}\, u_{pipe}\, A_{pipe} = \rho_{d(valve)}\, u_{d(valve)}\, A_{d(valve)} \tag{17}$$

$$\frac{\kappa}{\kappa-1}\frac{p_{pipe}}{\rho_{pipe}} + \frac{1}{2}u_{pipe}^2 = \frac{\kappa}{\kappa-1}\frac{p_{d(valve)}}{\rho_{d(valve)}} + \frac{1}{2}u_{d(valve)}^2 \tag{18}$$

$$p_{pipe} \big/ \rho_{pipe}^{\kappa} = p_{d(valve)} \big/ \rho_{d(valve)}^{\kappa} \tag{19}$$

where $u_{pipe} = 0$, for the valves are closed. And $P_{d(valve)}$ is assumed to be equal to $P_{d(port)}$ in this calculation. And in the discrimination for the stability of the solution, Courant-Friedrichs-Lewy (CFL) is adopted, shown as follows.

$$\Delta x > (|u| + a)\Delta t \tag{20}$$

As a time step Δt is decided in advance, ΔX is decided to satisfy this condition. And the calculations are performed by using the characteristic differential equations and each boundary condition.

Calculation of Leakage at side clearance between cylinder and piston

In consideration of the balance of pressure that a fluid flow through the minute clearance between the cylinder and the piston is estimated. The leakage into the cylinder is given by the following equation.

$$H\delta(P_{cyl} - P_s) = 2(H + \delta)L\tau \tag{21}$$

Thus, the leakage becomes:

$$Q = \int_0^{\delta} (uH)dr = (H\delta^3)/(6\mu L)(P_{cyl} - P_s) \tag{22}$$

Here, u is defined by:

$$u = (P_{cyl} - P_s)/(4\mu L)(\delta^2 - r^2) \tag{23}$$

CALCULATED RESULTS AND DISCUSSION

The overall characteristics of the compressor has been calculated and studied to evaluate the performance using the simulation models. The non-lubricated four-stage reciprocating compressor is used for verification. Table 1 shows the specifications of the compressor and the initial conditions for the characteristic calculation, in which five types of working fluids, namely Nitrogen, Carbon Dioxide, Argon, Helium and Air are applied. The characteristics are compared keeping the discharge pressure of the final stage between 10 - 30 MPa at various compressor rotating speeds.

Table 1. Specifications and initial conditions

Input Data	Reference Value
First Stage	
Cylinder diameter	0.080m
Suction port diameter × ports number	0.008m × 6
Discharge port diameter × ports number	0.008m × 4
Second Stage	
Cylinder diameter	0.033m
Suction port diameter × ports number	0.008m × 4
Discharge port diameter	0.008m
Third Stage	
Cylinder diameter	0.022m
Suction port diameter × ports number	0.008m × 2
Discharge port diameter	0.008m
Fourth Stage	
Cylinder diameter	0.013m
Suction port diameter	0.008m
Discharge port diameter	0.008m
Stroke	0.016m
Fluid	N_2/CO_2/Ar/He/Air
Rotating speed	900～2400min^{-1}
Suction pressure	0.15MPa
Discharge pressure	10.0～30.0MPa

Pressure Traces at each Stage

Figures 7 through 10 show the calculation results of the pressures in the first through fourth cylinders and of the discharge pressures under the rotating speed of 1800 min^{-1}. Available results of basic experiments performed using Nitrogen as the working fluid are added to these figures. The calculation results are described in the following.

- At the second through fourth stage, the displacement of the suction valve fluctuates three or four times. And at the first stage, there is hardly fluctuations at the displacement of the suction valve.

- The Discharge pressure fluctuates two or three times with the movement of the discharge valve at each stage, in which the figures show relatively good conformity between the calculation and the experiment results.

Therefore, the development of 3-D simulation model that is needed to perform the calculation in cases where multiple valves are installed will be conducted in the future. However, it can be concluded that the simulation program enables the calculation of the pressure balance, the evaluation of the performance characteristics and efficiency of the compressor.

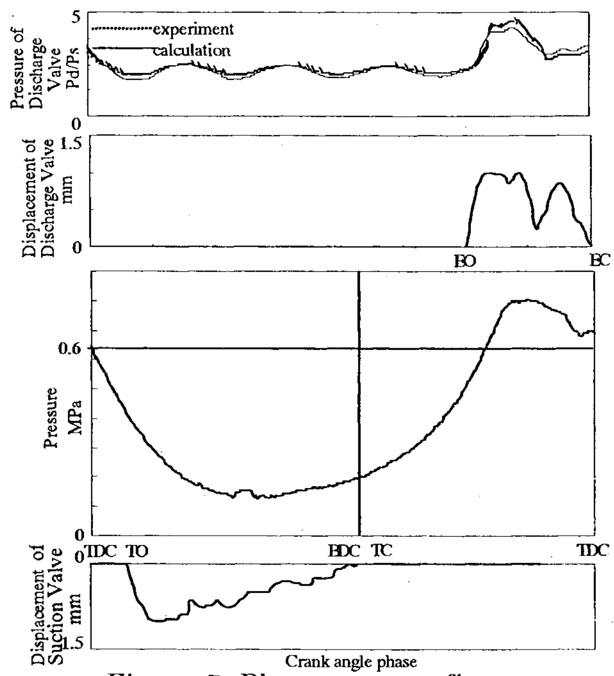

Figure 7. Phenomena at first stage

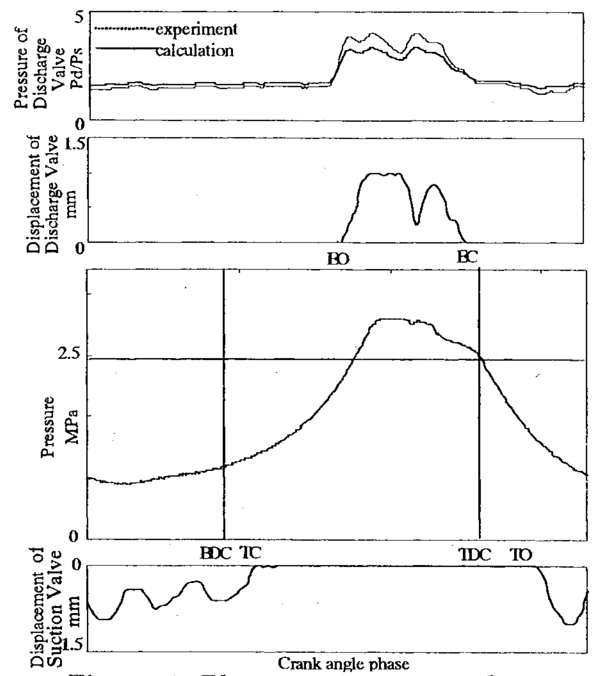

Figure 8. Phenomena at second stage

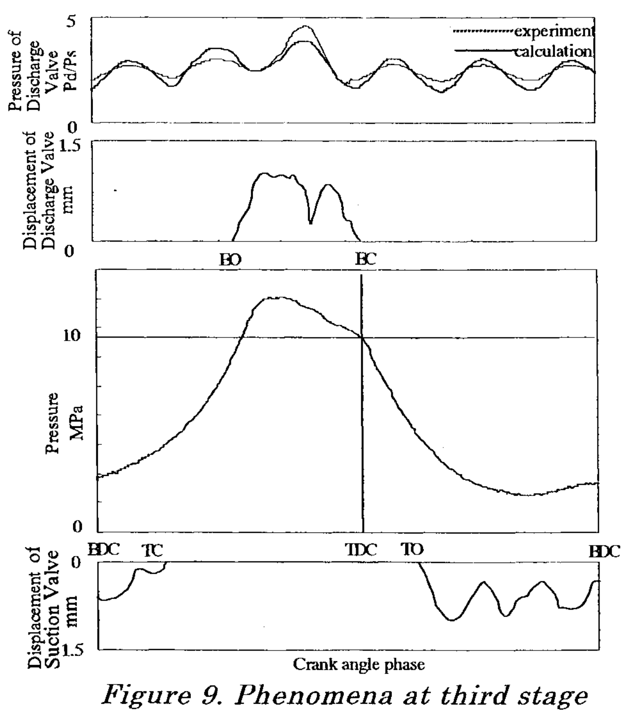

Figure 9. Phenomena at third stage

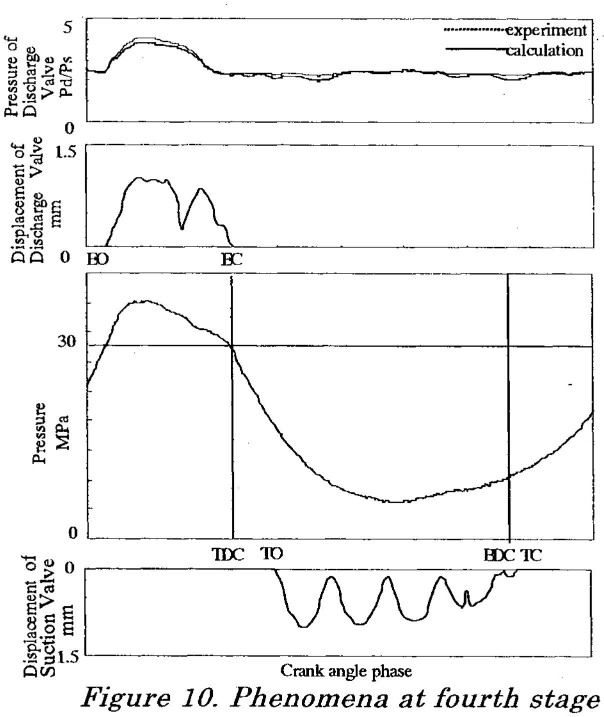

Figure 10. Phenomena at fourth stage

The pressure in the connecting pipe is shown in Figure 11, which shows that a phase lag occurs periodically. The cause may due to the New Mark β method used to solve the equation of motion. However, the amplitude of the pressure is well in agreement with the experiment results.

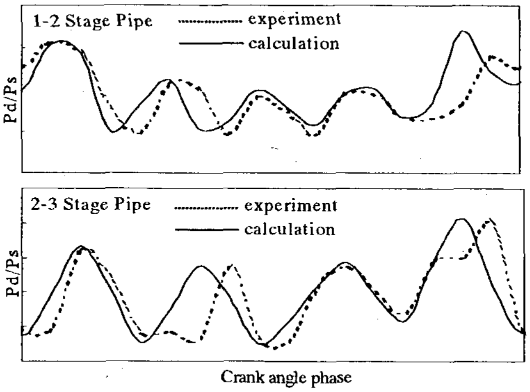

Figure 11. Pressure in connecting pipe

Effect of Discharge Pressure

The construction element affected mostly by the change of the discharge pressure is the discharge valve section at each stage. Figure 12 shows the time-series of discharge valve lift . There is a tendency that the higher the discharge pressure rises, the later the discharge valve starts to open. This is a phenomenon common to any stage, and the tendency at the first and second stages is less than at the third and fourth stages. The reason is considered that the first and second stages are affected by the suction pressure of the compressor more easily than by the change of the discharge pressure at the forth stage.

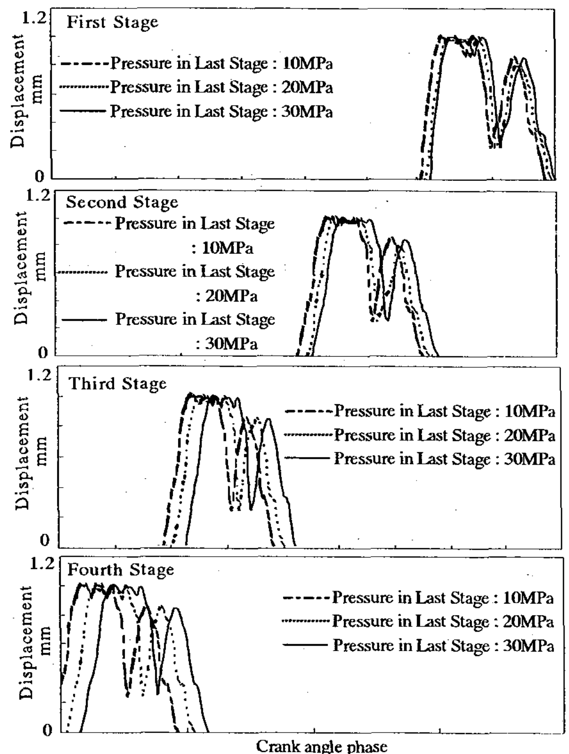

Figure 12. Discharge valve open dynamics

Influence of Working Fluids

The calculated results obtained from the simulation program with Nitrogen, Carbon Dioxide, Argon, Helium, and Air as working fluid is shown in Figure 13. Since it is assumed that the compression condition in this simulation is polytropic compression, the pressure and temperature in each cylinder are affected by specific heat ratio only. Therefore, the result show that monoatomic molecule gas such as Argon and Helium, which have higher specific heat ratios than others, indicates the pressure and temperature increases 5% and 6% in each cylinder respectively. The result of calculations show that Helium, having a low molecular weight and a high specific heat ratio, is the lowest in flow rate. However, it is only approximately 10% lower than nitrogen, which means sufficient flow characteristics. Therefore, it has been clarified that this compressor has a structure which assures sufficient flow rate in cases where various types of working fluids are applied.

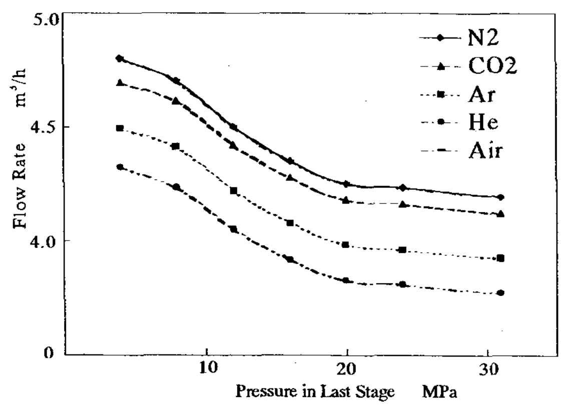

Figure 13. Gas flow rate of the various working fluids

Improvement in Performance

Based on the calculation results from

the simulation model, the improvement of intake efficiency at the suction system is studied. The calculation is performed by changing the piping length of the suction system and suction port diameter focusing on the mass flow rate through each of the multiple suction ports, which are provided in the suction system. From the calculated result for the suction system, the specification in which the ratio of the piping length to the suction port diameter is kept almost equal has been proposed. The suction system is illustrated in Figure 14, and the calculated result is shown in Figure 14. The improved specification is found to allow the mass flow rate to be almost equal in comparison with the ordinary specification showing uneven mass flow rate in each of the suction ports, and the higher thermal efficiency compressor has been developed using these results.

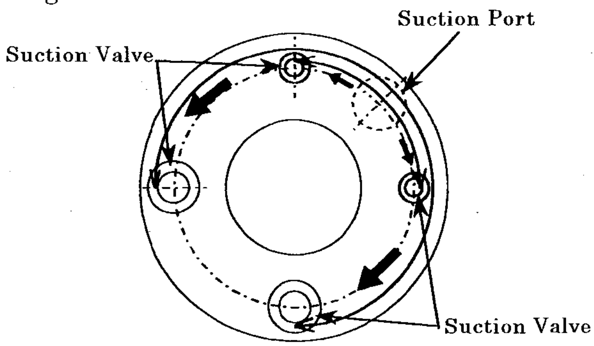

Figure 14. Suction system

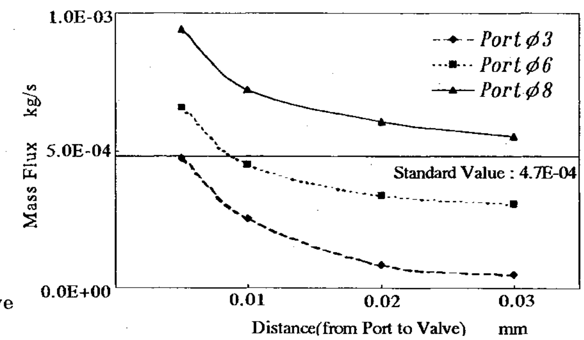

Figure 15. Mass flux of suction port

CONCLUSIONS

The original numerical simulation program proposed here gives the following conclusions.

1. This simulation model could conduct the investigation on the performance characteristics of the compressor, and is effective for the development of compressors suitable for wide applications.
2. For Nitrogen, Carbon Dioxide, Argon, Helium, and Air of which properties are employed in this simulation model, the sufficient performance characteristics of the compressor can be confirmed from the calculated results.
3. The high efficiency has been realized by equalizing the ratio of the piping length of the suction system to the suction port diameter in each of the multiple suction ports.

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