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OPTIMAL SIZING OF THE DISCHARGE PORT AREA OF A RECIPROCATING COMPRESSOR UTILIZING COMPUTER SIMULATION TECHNIQUE

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ABSTRACT

Computer simulation of the thermodynamic cycle of the cylinder process of compressors provides valuable information for compressor engineers. For example, the valve losses and the gas pulsation losses can be estimated from the simulation results. In general, the results of the computer simulation suggest that increasing the discharge port area always reduces the related valve loss and thus helps improve the efficiency of the compressor. However, in reciprocating compressors, a larger discharge port area tends to increase the clearance volume which may adversely affect the efficiency. Therefore, changing the discharge port area may sometimes give an unexpected result. In this work, a design procedure is proposed to determine the optimum discharge port area which balances these two conflicting effects by utilizing computer simulation program. A possible experimental method is also suggested to obtain empirical parameters necessary to use the proposed design procedure.

INTRODUCTION

A compressor simulation program is a very useful tool for compressor designers. For example, thermodynamic efficiency, refrigerant mass delivery per cycle, valve flow losses and gas pulsation losses of a prototype compressor can be predicted from the simulation result [1, 2]. However, some types of losses such as electric motor losses or friction losses can not be identified from the simulation result. The basic assumption of this study is that such losses are able to be identified and classified into two types; one which is proportional to the P-V work and the other which is approximately constant. In this work, the first type is called variable loss and the second is called constant loss.

Accepting the above assumption, the total energy consumption of a compressor per one cycle may be considered as the combination of the P-V work, constant loss and variable loss as follows:

\[ W_a = W_{pv} + W_c + W_v \]

where, \( W_a \) is the total energy consumption, \( W_{pv} \) is the P-V work, \( W_c \) is the constant loss, \( W_v \) is the variable loss.

There have been many theoretical and experimental efforts to identify compressor losses according to their causes [3, 4, 5]. In these studies, the valve loss at the discharge valve port has been found to be one of the major thermodynamic losses. A straightforward use of the simulation program suggests that the discharge port area should be made as large as possible to minimize this loss. Because the size of a port cannot be increased infinitively, it has been recommended to design the port size so that the average Mach number of the flow is less than 0.2 [3]. In practice, the volume occupied by the discharge port is usually the largest portion of the clearance volume in a reciprocating compressor. Therefore, enlarging the discharge port area may not always improve the efficiency because it also increases the clearance volume of a reciprocating compressor. A possible procedure to find an optimal balance of these two effects is proposed in this work.
The clearance volume in a reciprocating compressor may be thought to be composed of two parts as follows:

\[ V_c = V_{c1} + V_{c2} \]  

(2)

where, \( V_c \) is the total clearance volume, \( V_{c2} \) is the volume formed by the discharge port and \( V_{c1} \) is the contribution from all the others. Because \( V_{c1} \) is usually not possible to be estimated accurately, it is taken as 20% of \( V_{c2} \) corresponding to the initial port design in this work.

The compressor simulation program is used to simulate P-V diagram of the prototype as the discharge port size varies. At each design, the refrigerant mass delivery per cycle and the area of the P-V diagram are calculated. Figure 1 shows a typical simulated P-V diagram. The dash line and dash-dot line in Figure 1 indicate the pressures inside the discharge cavity and the suction cavity. For the purpose of demonstration, an extremely small discharge port area is chosen as the initial design.

The performance efficiency index used in this work is defined as the ratio of ideal specific energy consumption to the actual specific energy consumption [6].

\[ \eta_p = \frac{W_i}{m_i} \left/ \frac{W_{ai}}{m_a} \right. \]  

(3)

where, \( W \) and \( m \) are the energy consumption and the mass delivery of the compressor per one cycle. The subscripts 'i' and 'a' stand for the ideal and the actual operating conditions. The ideal energy consumption is defined as the area enclosed in the P-V diagram of an isentropic process at a given operating condition with zero clearance volume. Therefore,

\[ m_i = \rho_s \times V_{sw} \]  

(4)

\[ W_i = m_i \times (h_d - h_s) \]  

(5)

where, \( \rho_s \) is the refrigerant density at the suction pressure, \( V_{sw} \) is the sweep volume of the compressor, \( h_s \) and \( h_d \) are specific enthalpies of the refrigerant gas at the suction and discharge conditions.

As explained in the previous section, two types of losses are considered. At first, the variable loss is defined as:

\[ W_v = \varphi_1 \times W_{pv} \]  

(6)

where, \( \varphi_1 \) is a proportional constant that has to be determined empirically. Then, the constant loss is defined by referencing to the ideal energy consumption:

\[ W_c = \varphi_2 \times W_i \]  

(7)

where, \( \varphi_2 \) is also a empirical constant. Substituting (6), (7) into (1) and then into (3), the performance efficiency index becomes:

\[ \eta_p = \frac{m_i \times W_i}{m_i \times \left(1 + \varphi_1\right) \times W_{pv} + \varphi_2 \times W_i} \]  

(8)
The ratio of the performance index of the current prototype to that of an alternative design can be written as follows:

\[
\eta = \frac{m^*}{m_a} \cdot \frac{W_{wp}^* + \alpha \times W_i}{W_{wp}^a + \alpha \times W_i}
\]

(9)

where, superscripts '*' and 'a' refer to the original design and an alternative design, and \( \alpha = \frac{\phi_2}{1 + \phi_1} \). If \( \eta \) is greater than 1, the alternative design may be considered as a better design than the initial. All the parameters except \( \alpha \) in Equation (10) can be either calculated by Equation (4) and (5) or estimated by the computer simulation. In this work, \( \alpha \), while it should be obtained by measurements in actual practice, is taken three values as 20%, 30% and 40% for the purpose of demonstration.

Figure 2 shows a possible use of this concept. In the figure, the vertical axis is the performance efficiency ratio \( \eta \), and horizontal axis is the ratio of the discharge port area of the alternative design to that of the initial design. For example, the figure shows that the optimum size of the discharge port is approximately 1.5 times of the current design when \( \alpha \) is taken as 30%. Assuming that \( \alpha \) was estimated correctly, this discharge port size would provide the best overall efficiency.

**SUGGESTED EXPERIMENTAL PROCEDURE TO OBTAIN \( \alpha \)**

It is necessary to find the value of \( \alpha \) (or \( \phi_1 \) and \( \phi_2 \)) to use the procedure proposed above. The proportional loss factor \( \phi_1 \) can be understood as similar to typical mechanical or electrical losses. The constant loss \( \phi_2 \) is mainly related to the re-expansion of the clearance volume. For example, the P-V diagram would form a single line if the discharge pressure becomes very high. Even in such a case, the compressor would consume some energy to overcome the frictional and electric losses. A possible experimental procedure to obtain \( \alpha \) can be suggested as follows:

- At each of several different discharge pressure conditions, the P-V work is simulated and the actual total energy consumption is measured.
- The relationship between the simulated P-V work and the measured actual energy consumption may be plotted as shown in Figure 3. The relation may be approximated using the best fitting straight line.
- \( W_c \) is estimated from the intersection with the vertical axis, which gives the value of \( \phi_2 \).
- \( \phi_1 \) is estimated from the slope of the curve.

The test will have to be done using discharge pressures in a reasonably wide range so that the data of the measured total energy and the simulated P-V work will cover a relatively wide range (Figure 3).

**CONCLUSION**

A possible procedure is proposed for an optimal design of the discharge port area considering not only the effect of the valve flow loss but also the effect of the clearance volume re-expansion. The method assumes that a reliable P-V diagram simulation program and related empirical test data are available. A possible method for the experiment to obtain such empirical data is suggested.

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LIST OF REFERENCES


Figure 1. Typical P-V Diagram from Computer Simulation
Figure 2. Optimal Design of Discharge Port Area

Figure 3. A Possible Experimental Procedure to classify Losses