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To Improve the Performance of a Refrigeration Compressor by Optimizing Piston Stroke and Cylinder Diameter

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ABSTRACT

During the operation of a refrigeration compressor, electric power is transported from electric source to the refrigerant. Since there is a large amount of power required for the operation, it is necessary to save energy through the performance improvement. For this purpose, mathematical models consisting of nonlinear ordinary differential and algebraic equations are developed and applied for the simulation and optimization.

The objective function used in optimization has various forms. However, from the viewpoint of available energy, the compressor effectiveness might be a useful term to be used as a part of objective function for the performance improvement of a refrigeration compressor. Both the heat transfer between the refrigerant and the cylinder wall, and the throttling of flow in the suction and discharge passages have to be considered during the optimization as there is a close relationship among the effectiveness, the heat transfer, and the throttling of flow. The changes of cylinder diameter, d, and piston stroke, s, affect the heat transfer area of cylinder wall and the flow area of valves. Therefore, it is necessary to optimize these two dimensions for the maximization of an objective function F.

\[ F = E_{01} \lambda \]  

where \( E_{01} \) is the compressor effectiveness, and \( \lambda \) is volumetric efficiency.

INTRODUCTION

To evaluate the compressor performance, there are two kinds of thermodynamic analysis, the first law analysis, and the exergy analysis based on the second law of thermodynamics. The first law analysis shows the quantitative relationship among heat, work and internal energy. However, the exergy analysis gives us informations about the losses of available energy due to irreversibilities.

Actual compressor processes are shown in Fig.1.

In Fig.1, the area surrounded by processes represents available energy, \( w \), supplied to the refrigerant in one cycle.

A refrigeration cycle with actual compressor processes is shown in Fig.2. \( e_1, e_2, e_3 \) and \( e_4 \) represent the exergy of the refrigerant at point 1, 2, 3 and 4. The difference, \( (e_1 - e_2) \), represents the change in exergy during the suction process 1-2. \( (e_2 - e_3) \) is the change in exergy of the refrigerant during the process 2-3. \( (e_3 - e_4) \) indicates the change in exergy during the discharge process 3-4. The overall loss of available energy of the refrigerant, \( e_{01} \) in one cycle is given as follows:

\[ e_{01} = w + (e_1 - e_2) + (e_2 - e_3) + (e_3 - e_4) = w - (e_3 - e_1) \]  

where \( e_1 \) and \( e_4 \) are the exergy for steady flow and are given as

\[ e_1 = h_1 - T_0 s_1 \]  
\[ e_4 = h_4 - T_0 s_4 \]

where \( h \) is enthalpy, \( T_0 \) is temperature of surroundings, and \( s \) is entropy.

The available energy loss mainly results from the following reasons:

a) Throttling of flow in suction and discharge valves

b) Periodic heat transfer between the refrigerant and the wall.

The leakage of the refrigerant can also affect available energy loss. However, it is assumed that no leakage of the refrigerant occurs during the processes.

The compressor effectiveness is defined as[1].
For actual processes, $E < 1$. It means that a part of work supplied to the refrigerant is lost in quality due to irreversibilities. Therefore, $E$ is a useful tool indicating the compressor performance.

On the other hand, it is assumed that the actual piston displacement is fixed during the optimization. Therefore, the volumetric efficiency, $\lambda$, which is the ratio of effective piston displacement to the actual piston displacement (see Fig.1), can be used to indicate the cooling capabilities of compressors designed for optimization. Since a small value of $\lambda$ means a small capability, people dislike a compressor with a very small value of $\lambda$, even though it has high effectiveness. Hence, the term $\lambda$, can also be used to indicate the compressor performance.

From the discussion above, obviously an objective function used in optimization, should consist of two terms, the compressor and the volumetric efficiency. Therefore, the objective function, $F$, is given as

$$F = E \cdot \lambda$$  

(EFFECTIVE FLOW AREA)

The loss of available energy due to the throttling in suction and discharge valves relates to the effective flow area of valve, $A_e$.

$$A_e = K \cdot A$$  

(8)

where  $K$- flow coefficient  
$A$- flow area,  
(see Fig.3)

$$A = A_1 + A_2$$

Area $A$ depends on the valve plate displacement, $L$. Since the flow area changes with time, the valve displacement-time history is a very important factor to the available energy loss. "A-type history" shown in Fig.4 results in more loss of available energy as compared to "B-type history", even through the valve lift, $L_a$, is nearly two times the valve lift, $L$. Fortunately, with the help of mathematical models and computer simulations[2] [3] [4] [5], "A-type history" can be avoided. Therefore, it is assumed that during the optimization, all of the valves have "B-type history". In addition to this, it is found that for the optimization an assumed history, "C- type history" (see Fig.5), can be used to replace "B-type history". The replacement reduces the available energy loss by calculation. In other words, some of the available energy can be gained from the history replacement. However, calculation shows that only 2~5 percent of available energy loss occurring during the suction and discharge processes is reduced due to the history replacement. Since the gain is quite small, and is received by all the compressors designed during the optimization, it is reasonable to accept the above replacement for the simplification of mathematical models and for time saving in calculation.

Area $A$ also depends on the valve lift, $L_{\text{max}}$, and the average diameter of valve plate, $d_v$.

Two assumptions made on $L_{\text{max}}$ and $d_v$ are:

a) The valve lifts are the same for all of the valves designed during the optimization, as the angular velocity of crank, $\omega$, and the dimensionless parameter, $P_{\text{e}} \cdot d / \omega$, are fixed [6].

b) The diameter of valve plate, $d_v$, is proportional to the cylinder diameter, $d$.

HEAT TRANSFER RATE

Heat transfer takes place due to the temperature difference between the refrigerant and the wall. Generally speaking, heat is transferred from the refrigerant to the wall during the compression process and is transferred from the wall to the refrigerant during the re-expansion process. However, investigations show that both heat-addition and heat-rejection can be found in one process. For example, the re-expansion process shown in $T$-$S$ diagram, Fig.6, can be divided into two parts, the first part 1-2, and the second part 2-3. During the first part, heat is withdrawn from the refrigerant due to the higher refrigerant temperature compared to the wall temperature. Then, in the second part, heat is added to the refrigerant as the refrigerant temperature is lower than the wall temperature.

Some of the investigations were made on heat transfer in a cylinder. Adair, Qvale and Pearson [7] developed a correlation for instantaneous heat transfer rate in a cylinder. The correlation was given as follows:

$$\text{Nu} = \text{A Re}^n \text{Pr}^m$$

(9)

where $A = 0.053$

$n = 0.8$

$m = 0.6$

Adair used a swirl velocity of the refrigerant in a cylinder, $\omega_g$, to calculate the Re number.

$$\omega_g = \begin{cases} 2(1.04 + \cos 2\theta), & \frac{3\pi}{2} < \theta < \frac{\pi}{2} \\ (1.04 + \cos 2\theta), & \frac{\pi}{2} < \theta < \frac{3\pi}{2} \end{cases}$$
Lee and Smith [8] calculated the heat transfer rate in a cylinder by using the First Law of Thermodynamics,

\[ Q = \dot{U} + \int_{\text{in}}^{\text{out}} \left[ \sum_{i} \left( h_{i} \cdot n_{i} \right) + w \right] \]

where
- \( Q \): rate of the heat flow into the cylinder
- \( \dot{U} \): rate of the change of internal energy of the refrigerant in the cylinder
- \( h \): enthalpy of the refrigerant
- \( m \): rate of the refrigerant flow
- \( w \): rate of the work done by the refrigerant in the cylinder

Since the constants in correlation (9) are different for different types of compressor, an investigation [9] was made to find out the suitable values of \( A, n, m \) for a refrigeration compressor, which is the object to be improved in this paper.

The temperature of cylinder head, \( t_{\text{wc}} \), temperature of cylinder surface, \( t_{\text{ww}} \), temperature of piston, \( t_{p} \), speed of crank, \( n \), suction temperature of refrigerant, \( t_{s} \), pressure ratio, \( \varepsilon \), and the ratio of instantaneous piston displacement to actual piston displacement, \( s \), have been measured.

It was found that the temperatures, \( t_{\text{ww}} \) and \( t_{\text{wc}} \), increase with the pressure ratio, \( \varepsilon \) (see Fig. 7 and Fig. 8). The relationships between temperature, \( t_{\text{ww}} \), and suction temperature, \( t_{s} \), are shown in Fig. 9.

The correlations for average \( t_{\text{ww}} \), \( t_{\text{wc}} \) and \( t_{p} \) are given

\[ t_{\text{ww}} = 24.32 + 0.719 t_{s} + 11.235 \varepsilon \]
\[ t_{\text{wc}} = 13.64 + 0.719 t_{s} + 11.235 \varepsilon \]

The correlation used in Ref. [9] for prediction of heat transfer rate has the same form as correlation (9). However, the following two adjustment have been made:
- The constants \( A, n, m \) are
  \( A = 0.75 \)
  \( n = 0.8 \)
  \( m = 0.6 \)
- The value of \( A \) here is about ten times the one in correlation (9).

b) In stead of the swirl velocity of the refrigerant, \( \omega_{3} \), in correlation (9), the following \( \omega_{3} \) is suggested:

\[ \omega_{3} = \begin{cases} \omega_{1} & 0 < \phi < \frac{\pi}{2} \\ \omega_{4} & \frac{\pi}{2} < \phi < \frac{3\pi}{2} \\ \omega_{5} & \frac{3\pi}{2} < \phi < \pi \end{cases} \]

Fig. 11 shows the heat transfer rates predicted from correlation of ref. [7], ref. [8], [9] and ref. [10] and the heat transfer rate obtained from experiment [9], [10].

The heat transfer from discharge refrigerant to suction refrigerant in valve passage is not taken into consideration in this paper due to the following reasons:

a) Only a little information concerning with the heat transfer rate in valve passage has been presented, and it is difficult to be used in our work due to the complexity of valve passage.

b) The neglect of heat transfer in valve passage could cause trouble in computation of available energy of the refrigerant. However, it is hopeful that the trouble can be reduced since the neglect is made for all of the compressor optimized.

The best way, of course, is to make an investigation on heat transfer in the valve passage of the compressor optimized, and it is planned by authors to do such an investigation in the near future.

A refrigeration compressor is used as an object for the performance improvement. Two variables to be optimized are stroke, \( s \), and cylinder diameter, \( d \). Since there is a relationship among the actual piston displacement, \( v \), stroke, \( s \), and cylinder diameter, \( d \),

\[ V = \frac{\pi}{4} s^{2} d \]

and \( V \) is assumed to keep constant during the optimization, the above two variables can be reduced to a new variable, \( \psi \) (\( \psi = s/d \)).

The objective function, \( f \), changes as shown in Fig. 11. Between points a and b, \( f \) almost keeps constant. Since large \( \psi \) indicates large cooling capability, the best choice for optimum \( \psi \) is \( \psi_{b} \).

CONCLUSION

The first law of Thermodynamics announces that when energy in one form disappears, an equal quantity of energy in another form appears. The First Law of Thermodynamics does not attempt to designate whether or not a process is ideal. However, the Second Law of Thermodynamics indicates that different types of energy have different abilities to do work. Hence they are not equivalent. A part of available energy supplied to the refrigerant in one cycle losses due to the irreversible processes such as the throttling of flow in valve passage, the heat transfer between the cylinder wall and the refrigerant. Therefore, from the point of view of available energy, the compressor effectiveness is an important factor to be considered during the optimization.

An objective function consisting of the compressor effectiveness and the volumetric efficiency is given. The objective function is used for the performance improvement of a refrigeration compressor.
REFERENCES


Fig. 1 p-V Diagram showing actual compressor processes
V- actual piston displacement
V'-effective piston displacement

Fig. 2 A Refrigeration Cycle with actual compressor processes
Fig. 3 Schematic Diagram of valve geometry

Fig. 4 Valve Displacement-Time History

Fig. 5 The Actual and Assumed Displacement-Time Histories

Fig. 6 Re-expansion Process on T-S Diagram

Fig. 7 Relationship between temperature of cylinder surface, $t_{cw}$, and pressure ratio, $\mathcal{E}$
Fig. 8 Relationship between temperature of cylinder head, \(t_{wc}\), and pressure ratio, \(\xi\).

Fig. 9 Relationship between temperature of cylinder surface, \(t_{ww}\), and suction temperature, \(t_{s}\).

Fig. 10 Heat Transfer Rate calculated from correlations of Ref.[7], Ref.[8] and Ref.[9].

Fig. 11 Objective Function, \(f\), versus Variable, \(\psi\).