The development of the CS valve used to the variable swash plate compressor for a vehicle air-conditioning system

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Abstract. Since the variable swash plate type compressor used in the vehicle air-conditioning system is mounted on the engine and uses the power of the vehicle, many studies have been conducted on improving the performance of the compressor in order to increase the fuel efficiency of the vehicle. This study was also carried to improve compressor performance through the control room & suction muffler (CS) valve which is an on-off valve located between the control room and suction muffler. The swash plate angle can be controlled by the control room pressure adjusted with control gas. The flow of control gas starts from compressed high-pressure gas which is in the discharge muffler, flows into the control room through the external control valve (ECV) and then flows out to suction muffler through the CS hole which is located between the control room and suction muffler. Such this control method is using the compressed discharge refrigerant so that the compressor power is increased when the amount of the control gas is increased. Therefore, in this study, the CS valve has been developed to reduce the compressor power by minimizing the amount of control gas and it has been confirmed that the compressor power was reduced about 5~10% by the compressor operation speed.

1. Introduction

Among various convenience devices for a vehicle, the air-conditioning system which has been applied since the 1970s has become an essential convenience device. Such as Figure 1, the vehicle air-conditioning which is consisted of condenser, compressor, and Heating Ventilation Air-Conditioning (HVAC) is located at front part of a vehicle, and there are thermal expansion valve and evaporator in the HVAC system. Refrigerant R134a is used at the vehicle air-conditioning system until now, but alternative refrigerant R1234yf begins to be used for environment protection.

In terms of developing the vehicle air-conditioning system, it is focused on improving efficiency because reducing the power consumed on A/C system means improving fuel economy for vehicle. Figure 2 shows the effect of the air-conditioning system on the fuel efficiency of the vehicle. The reduced gas mileage ratio by the air-conditioning system is 9% per year, and a maximum of 23% in the summer. In the parts of the air-conditioning system, the compressor using power of the system for 53% per annum and 70% in the summer. This means that the compressor efficiency affects the fuel efficiency of the vehicle. Therefore, to reduce the compressor power, a fixed swash plate type compressor which is On-Off controlled has been replaced by external controlled variable swash plate compressor (EVDC) whose stroke volume is controlled proportionally by an external signal. As a
result, the fuel consumption rate of the vehicle can be reduced by about 4~5% by using the EVDC [1]. Both the fixed swash plate and externally controlled variable swash plate compressor are shown in Figure 3. The study of the variable compressor is mainly focused on the dynamic analysis of a variable mechanism [2] and performance improvement [3, 4]. The purpose of this study is also to improve the compressor performance by developing CS valve.

Figure 1. Air-conditioning system in a vehicle

Figure 2. Comparison of fuel consumptions of the air-conditioning system and parts

(a). Fixed type swash plate compressor

(b). Variable swash plate compressor

Figure 3. Schematic of compressor for vehicle air-conditioning system
2. Variable Compressor

2.1. Mechanism of variable swash plate compressor
The swash plate angle is changed by two types of movement of the swash plate. One is the movement of swash plate center, along the shaft. The other one is the movement of the swash plate arm, along the lug guide surface. This means that the rotation center of swash plate changes continuously as the swash plate angle, and the rotation center is the cross point of the vertical lines of lug plate guide surface and shaft at each contact point with the swash plate, as shown in Figure 4, Rotation center.

In Figure 4, the $F_s$ is the centrifugal force of the rotation swash plate, the $F_p$ is inertia force of the reciprocating piston and $F_g$ is gas force by the pressure difference between each cylinder, $p_{(n)}$, and control room, $p_c$. For the forces acting on the swash plate, the moment equation is equation (1) and this equation must be zero to maintain the swash plate angle. If $\sum M$ is greater than zero, the swash plate angle is decreased. Otherwise, the swash plate angle is increased.

Since the $M_s$ is a function of gas force, it can be expressed equation (2), and the swash plate angle ($\beta$) can be expressed as equation (3) from equation (1) and (2). In equation (3), $A_p$ is the piston section area and $e_{(n)}$ is the distance from the rotation center to each cylinder, these are the design parameters. And also $p_{(n)}$ is each cylinder pressure which is a function of suction and discharge pressure, and $M_s$ and $M_p$ are a function of operating speed. Therefore, the swash plate angle ($\beta$) can be controlled by the control room pressure, $p_c$.

\[
\sum M = M_s - M_g - M_p \quad (1)
\]

\[
M_g = A_p \sum_{n=1}^{N_c} [e_{(n)} \cdot (p_{(n)} - p_c)] / \cos(\beta) \quad (2)
\]

\[
\beta = \cos^{-1} \left( \frac{A_p \sum_{n=1}^{N_c} [e_{(n)} \cdot (p_{(n)} - p_c)]}{M_s - M_p} \right) \quad (3)
\]

Figure 4. Force modelling on the swash plate

2.2. Control room pressure control
In the control room of Figure 5, since the leakage gas flows into the control room from the cylinder through the clearance between piston and cylinder, the CS hole is needed to discharge leakage gas from the control room to the suction muffler. And the control room pressure $P_c$ is controlled by the control gas which is flows into the control room from the discharge muffler through the ECV. Because the open area of ECV is a function of an external signal, the control room pressure can be controlled by the external signal. From the result of the compressor evaluation according to the current variation in Figure 5, where $p_s$ and $p_d$ are the suction and discharge pressure, the compressor is operated at the max stroke volume when the current of ECV is changed from 0.75A to 0.68A, and when the control valve current is lower than 0.68A, the compressor torque is decreased while the control room pressure is increased, the condition in which the inclination angle of the swash plate is controlled is referred to as a variable condition. In the variable condition, the compressor stroke volume can be controlled according to the required cooling capacity of the air-conditioning system. In conclusion, the external variable compressor can control the stroke volume of a compressor by the external signal of the control valve.
Figure 5. Control gas flow and controllability test result

2.3. Compressor performance by CS hole size
Equation (3) can be expressed as a function of the control room pressure as in equation (4). When the control pressure is defined as $p_c - p_s$, equation (4) becomes a function of control pressure as equation (5).

\[
p_c = \left[ \sum_{n=1}^{N_{c}} \left( e_{(n)} \cdot p_{(n)} \right) - \cos(\beta) \left( M_s - M_p \right) \right] \left/ \sum_{n=1}^{N_{c}} \left[ e_{(n)} \right] \right]
\]

\[
p_c - p_s = \left[ \sum_{n=1}^{N_{c}} \left( e_{(n)} \cdot \left( p_{(n)} - p_s \right) \right) - \cos(\beta) \left( M_s - M_p \right) \right] \left/ \sum_{n=1}^{N_{c}} \left[ e_{(n)} \right] \right]
\]

At the moment the compressor starts to operate, $p_{(n)}$ is very small since the compressor starts from a minimum angle, therefore, the control pressure $p_c - p_s$ has to be small, $p_c \approx p_s$, to increase the swash plate angle. To reduce the control pressure at the moment of compressor start, the size of the CS hole should be large enough. The other hands, at the variable condition, since the control gas flows out to suction muffler from the control room through the CS hole, when the CS hole becomes larger, the amount of control gas is also increased to maintain the control room pressure, and the compressor power is increased because the control gas is compressed refrigerant.

Figure 6 shows the performance and initial operating delay test results by the CS hole size, diameter. As for performance test results, when compressor is operated in a variable condition at the cooling capacity of 1.16kW (1000kcal) at 1000rpm, it can be confirmed that the required compressor power is reduced 12% for Ø1.9mm and 23% for Ø1.5mm based on the power of CS hole Ø2.4mm. However, in the initial operating delay test, the delay time is 5 seconds when the CS hole is Ø1.9mm, but the operation delay time is increased to 25 seconds when the CS hole is Ø1.5mm. As a result, the smaller CS hole can reduce the compressor power, but it is impossible to reduce the CS hole to Ø1.5mm or less due to the operation delay problem. Therefore, the CS hole of the compressor researched in this paper is Ø1.9mm.
3. CS valve design

As discussed in chapter 2.3, the smaller the CS hole can reduce the compressor power, but the initial operation problem of the compressor has been identified. In our study, the CS valve has been developed as a way to reduce the CS hole while improving the initial operability.

3.1. CS valve design

In Figure 5, when the compressor operates in variable condition, it is found that the control room pressure is higher than the suction pressure about 50 kPa, control pressure. And, in Figure 6 (b), when the compressor starts, the control room pressure is also the same as the suction pressure since the discharge pressure and suction pressure is the same. Therefore, it is possible to think the CS valve structure that can be opened when the control room pressure is same with suction pressure, and closed by the pressure difference about 50 kPa over between suction and control room pressure.

The CS valve structure is shown in Figure 7. In variable compressor with CS valve, the base CS hole is reduced from Ø 1.9mm to Ø1.5mm, and CS valve is mounted on the cylinder and operated by the pressure difference between in the control room and suction muffler. Since the maximum opening area of the CS valve has been designed to correspond to a diameter of Ø1.5mm, the total CS hole effective diameter is Ø2.1mm (Ø1.5mm + Ø1.5mm) when the CS valve is opened. And when the CS valve is closed, the CS hole diameter is Ø1.5mm which is the base CS hole diameter.

The equation (6) is a force dynamics of the CS valve, where $A_v$ is the cross-section area of the CS valve, $k$ is the spring coefficient, $y_0$ is the assembled compressed length of the spring, and $y$ is the displacement of the displacement of the rod. The CS valve open pressure, $p_c - p_s$, is equation (7) from the equation (6), and the design dimension $A_v$, $k$, $y_0$, $y$ can be determined by the control pressure. The valve fully closed pressure is fixed to be 40 kPa which is lower than the minimum control pressure of 50 kPa.

In conclusion, the CS valve is able to improve the initial operating delay by discharging the leakage gas when the compressor starts, and when the compressor is operating at variable condition, the CS valve is closed and operates with small CS hole size, so that the compressor power can be reduced.

\[
F_{gas} - F_{spring} = A_v \cdot (p_c - p_s) - k \cdot (y_0 + y) = 0
\]  

\[
control \ pressure \ (40 \ kPa) = (p_c - p_s) = \frac{k \cdot (y_0 + y)}{A_v}
\]
3.2 CS valve design verification

Figure 8 shows a test rig that can verify the function of the CS valve by measuring the volume flow rate of nitrogen gas through the CS valve according to the pressure increasing.

Before testing the CS valve, the volume flow rate according to pressure was measured for the CS hole Ø 2.1mm and Ø 1.5mm respectively, and the measured result was compared with the CS valve. As a result, the flow rate is almost the same as the CS hole Ø 2.1mm until pressure 27kPa and it is equal to CS hole Ø 1.5mm at 40kPa or higher. This means that the CS valve starts to close at 27kPa and is fully closed at 40kPa which is exactly the design target pressure.

4. Test result

4.1 Compressor performance test result

Table 1 shows the performance test results for the compressor equipped with a CS valve. The performance test was conducted by compressor performance test rig which can control suction and discharge pressure, suction temperature, sub cool, compressor speed, and cooling capacity. And the stroke volume of the variable compressor is controlled by the external valve according to the cooling capacity of the test rig at the variable condition.

In the test condition, suction and discharge pressures are 300 (2) and 1570kPa (15kgf/cm²G), respectively, cooling capacity is 2.9kW, and the compressor power was measured according to the compressor speed. In the test results, applying the CS valve, the compressor power is reduced by 8%, 10%, 6%, and 5%, at 1500, 2500, 3500 and 4000 rpm, respectively.
Table 1. Performance test results of compressor in this study

| Operating condition | Required compressor power [kW] | Ratio |
|---------------------|---------------------------------|-------|
|                     | Ø 1.9mm                         | CS valve |
| \( p_s \): 300kPa  | 1.500rpm                        | 1.56   | 1.44   | 92% |
| \( p_d \): 1,570kPa| 2,500rpm                        | 1.71   | 1.54   | 90% |
| \( T_{sh} \): 10 \(^{\circ}\)C| 3,500rpm                        | 1.75   | 1.65   | 94% |
| \( T_{sc} \): 5 \(^{\circ}\)C | 4,000rpm                        | 1.82   | 1.73   | 95% |

4.2 Initial operation test
Figure 9 shows the compressor initial operational test results in the actual air-conditioning system. When the compressor operates at 800rpm and ambient temperature 35deg, as already mentioned, the compressor whose CS hole is Ø 1.9mm operates within 5 seconds after compressor start, and the operation delay is 25 seconds for CS Ø 1.5mm. On the other hand, the variable compressor with CS valve was operated within 3 seconds since its CS hole total open diameter is Ø 2.1mm. This result shows that the initial operating delay can be solved by the CS valve.

![Figure 9. Initial operation test results](image-url)

5. Conclusions
In this study, CS valve has been introduced in order to reduce the compressor power by minimizing the amount of control gas. The following conclusions are obtained through the development of the CS valve to reduce the power in a variable compressor.

(1). In the compressor performance test results, it was confirmed that the required power of the compressor is reduced due to the decrease of the control gas amount as the CS hole is decreased when the compressor operates at the variable condition (same cooling capacity).

(2). In the initial operation test, the operation delay of the compressor occurs when the CS hole is below Ø 1.5mm. Therefore, it is impossible to reduce the CS hole to Ø 1.5mm or less.

(3). The CS valve operated by the pressure difference between the control room and suction muffler was developed to control the CS hole size according to the operating of the compressor, and the function of CS valve has been verified by the CS valve test rig.

(4). A variable compressor applying CS valve is confirmed to reduce compressor power about 5~10 % according to the compressor speed without operating delay problem when the compressor starts.
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