Study on the CO₂ electric driven fixed swash plate type compressor for eco-friendly vehicles

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Abstract. The purpose of this study is to experiment and to performance analysis about the electric-driven fixed swash plate compressor using alternate refrigerant(R744). Comprehensive simulation model for an electric driven compressor using CO₂ for eco-friendly vehicle is presented. This model consists of compression model and dynamic model. The compression model included valve dynamics, leakage, and heat transfer models. And the dynamic model included frictional loss between piston ring and cylinder wall, frictional loss between shoe and swash plate, frictional loss of bearings, and electric efficiency. Especially, because the efficiency of an electric parts(motor and inverter) in the compressor affects the loss of the compressor, the dynamo test was performed. We made the designed compressor, and tested the performance of the compressor about the variety pressure conditions. Also we compared the performance analysis result and performance test result.

1. Introduction

Since the end of the 19th century, carbon dioxide was used a refrigerant in ship’ refrigerators and other stationary systems. After CFCs and HCFCs refrigerants, which became major refrigerants, was introduced, and carbon dioxide were declined from the market. However, environmental issues as the ozone depletion and global warming was caused by using CFCs and HCFCs refrigerants. Lorentzen et al.[1] rediscovered the carbon dioxide as a suitable refrigerant. Carbon dioxide is a good refrigerant which is environmentally friendly and has excellent thermal properties.[2] Since he reasoned with several possible trans-critical cycles with carbon dioxide for vehicle air condition system, many studies have been conducted on the compressors and systems for carbon dioxide refrigerant. [3-6]

Many studies have been carried out on the steady-state simulation models for reciprocating compressors. Many analysis models for electric driven compressor did not consider or calculated the efficiency of the motor using simple relationship. However, because the electric driven compressor for vehicles is equipped with a motor and an inverter therein, it has to consider the accurate efficiency of electric parts in the compressor. And the efficiency of electric parts is changed by the rotational speed and the compression ratio.
This study considered the efficiency of electric parts in the electric driven fixed swash plate compressor including a motor and an inverter for a more accurate performance analysis. The efficiency of electric parts is carried out by the dynamo test and the test results are used in the performance analysis. The efficiency of the electric parts is measured with the rotational speed by the load through the dynamo test. The power of the compressor is calculated by a polynomial relationship using the dynamo test results. And the performance analysis results are compared with the performance evaluation results of the prototype compressors using CO₂.

2. Numerical simulation model

![Fig. 1 Schematic of the geometry and kinematics of the fixed swash plate](image)

A schematic of the geometry and kinematics of the fixed swash plate type compressor is shown in fig.1. The Swash plate is fixed on the crank-shaft and the piston with the shoes installed on the swash plate. The piston reciprocates in accordance with the rotation of the shaft, wherein the piston stroke, $S$ is expressed as a function of the shaft angel, $\theta$ as follow:

$$S(\theta) = PCR \tan \beta (1 + \cos \theta)$$

Where, $PCR$ is the pitch circle radius of the pistons and $\beta$ is the angle of swash plate.

The volume inside the cylinder is calculated by:

$$V(\theta) = S(\theta) \cdot A_p + V_{dead}$$

Where, $V_{dead}$ is the dead volume inside the cylinder and $A_p$ is the area of the piston.

Taking the cylinder enclosure as the boundary of the control volume. Based in the law of mass conservation, the mass change of refrigerant in the cylinder is determined:

$$\frac{dm_c}{dt} = \frac{dm_{suc}}{dt} + \frac{dm_{li}}{dt} - \frac{dm_{dis}}{dt} - \frac{dm_{lo}}{dt}$$

The conservation equation of energy for the cylinder contents can be written by the first law of thermodynamics.

$$\delta Q = \delta U + \delta W - [dE_i - dE_o]$$

Therefore, the temperature in the cylinder as a function of time:
\[
dT_c = \frac{dQ_c}{dt} - \frac{dW_f}{dt} + \left[ \frac{dm_{ac}}{dt} \cdot (h_{suc} - h_c) + \frac{dm_{li}}{dt} \cdot (h_l - h_c) \right] + m_c \cdot \frac{dv_c}{dt} \left[ v_c \cdot \left( \frac{\partial P_c}{\partial T_c}, T \right) - \left( \frac{\partial h_c}{\partial T_c} \right)_T \right] \\
\]

\[
m_c \cdot \left[ \left( \frac{\partial h_c}{\partial T_c} \right)_v - \left( \frac{\partial P_c}{\partial T_c} \right)_T \cdot v_c \right]
\]

The heat transfer from the refrigerant between the gas and the cylinder wall is calculated using the heat transfer coefficient, proposed by Fernanda et al.[12]

\[
\frac{dQ}{dt} = hA_{heat}(T_{cyl}(t) - T_s)
\]

\[
h = \frac{k}{D_p}aRe^bPr^c
\]

Where \(T_s\) is surface temperature of the compression chamber walls, \(A_{heat}\) is the area of the gas chamber, including the piston and end cap, \(T_{cyl}\) is the average gas temperature in the cylinder, \(Nu\) is the Nusselt number, \(h\) is the convective heat transfer coefficient, \(k\) is the thermal conductivity of the cylinder, \(Pr = \frac{C_p \mu}{k}\) is the Prandtl number, \(Re\) is the Reynolds number, \(D_p\) is the characteristic length (Cylinder diameter), and \(a, b, c\) are constants. The characteristic velocity for the Reynolds number is the piston mean velocity.

\[
V_p = 2Sf
\]

\[
V_c = \frac{|\hat{m}(t)|}{\rho(t)A_p}
\]

Where, \(f\) is the rotational speed of the compressor, \(\hat{m}\) is the mass flow rate through the valves, and \(\rho(t)\) is the gas density in the cylinder.

Due to the distinct phenomena taking place in the compression cycle, different definitions for the Reynolds number \(Re = \frac{\rho V}{\mu}\) and different values for \(a, b,\) and \(c\) were specified for the compression, discharge, expansion and suction processes, as indicated in Table 1.

| Process   | Reynolds number |
|-----------|----------------|
| Suction   | \(Re = \frac{\rho(t)DV_p}{\mu(t)} + \frac{2V_p}{\mu(t)}V_c(t)^{1.4}\) |
| Compression | \(Re = \frac{\rho(t)DV_p}{\mu(t)}\) |
| Discharge | \(Re = \frac{\rho(t)DV_p}{\mu(t)}V_c(t)^{0.8}\) |
| Expansion | \(Re = \frac{\rho(t)DV_p}{\mu(t)}\) |

In convective heat transfer, the Churchill–Bernstein equation is used to estimate the surface averaged Nusselt number for a cylinder in cross flow at various velocities. This equation is named after Stuart W. Churchill and M. Bernstein, who introduced it in 1977. This equation is also called the Churchill–Bernstein correlation.[11]
\[ \bar{N}_{ud} = 0.3 + \frac{0.62 Re_d^{1/2} Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \left[ 1 + \left( \frac{Re_d}{282000} \right)^{5/8} \right]^{4/5} Pr Re_d \geq 0.2 \]

The Churchill–Bernstein equation can be used for any object of cylindrical geometry in which boundary layers develop freely, without constraints imposed by other surfaces.

The swash plate compressor is used for the inlet valve and discharge valve, the opening and closing action of the valve has a significant effect on the performance of the compressor. This study were modeled as vibration problems of one degree of freedom model to analyze the behavior of the valve such as respectively for the inlet and outlet valve.

\[ \ddot{y} = \frac{\Delta P A_f}{m} - 2\xi \omega_n \dot{y} - \omega_n^2 y \]

Where, \( y \) is the displacement of the valve, \( m \) is the mass of the valve, \( c(=2m\xi\omega_n) \) is the damping coefficient, and \( k(=m\omega_n^2) \) is the stiffness. \( \Delta P \) is the pressure difference across the valve (the difference between the upstream pressure and the downstream pressure). \( A_f \) is the effective force area of the valve.

The mass flow rate through the discharge valve and the suction valve is usually expressed that would occur for an isentropic flow condition of an ideal gas.

\[
\frac{dm_{value}}{dt} = C_{value} A_e P_u \sqrt{\frac{2k}{(k-1)RT_u}} \left( \frac{P_d}{P_u} \right)^\frac{k}{k-1} - \left( \frac{P_d}{P_u} \right)^\frac{k+1}{k-1}, \quad \left( \frac{P_d}{P_u} \right) > \left( \frac{k+1}{2} \right)^\frac{k}{k-1}
\]

\[
\frac{dm_{value}}{dt} = C_{value} A_e P_u \sqrt{\frac{2k}{(k-1)RT_u}} \left( \frac{k+1}{2} \right)^\frac{k}{1-k} - \left( \frac{k+1}{2} \right)^\frac{k+1}{1-k}, \quad \left( \frac{P_d}{P_u} \right) < \left( \frac{k+1}{2} \right)^\frac{k}{1-k}
\]

Where, the coefficient \( C_{value} \) is 0.58 for the suction valve and 0.6 for the discharge valve.[10] \( A_e \) is the effective flow area of the valve.

Gas leakage through the piston ring is usually expressed that would occur for an isentropic flow condition of an ideal gas.

\[
\frac{dm_{gap}}{dt} = C_{gap} A_{gap} P_u \sqrt{\frac{2k}{(k-1)RT_u}} \left( \frac{P_d}{P_u} \right)^\frac{k}{k-1} - \left( \frac{P_d}{P_u} \right)^\frac{k+1}{k-1}, \quad \left( \frac{P_d}{P_u} \right) > 0.54
\]

\[
\frac{dm_{gap}}{dt} = C_{gap} A_{gap} P_u \sqrt{\frac{k}{RT_u}} \left( \frac{2}{k+1} \right)^\frac{k+1}{k-1}, \quad \left( \frac{P_d}{P_u} \right) \leq 0.54
\]

Where the discharge coefficient \( C_{gap} \) is assumed to be 0.86.[9] The subscripts u and d is defined as upper and down, respectively.

The torque due to the frictional force between the swash plate and the shoes relative to the normal force by the internal pressure in the cylinder.

\[ T_{SS} = PCR \mu F_N \]

Also it is added to the inertia force caused by the mass of the piston and the shoes.
\[ T_{f,t} = \mu_{PCR} \left( A_p \sum_{n=1}^{N} \left( \frac{P_{cly} - NP_S}{\cos \beta} \right) + (m_p + 2m_s) \sum_{n=1}^{N} S''(\theta)_N \right) \]

Thus, a compressor power by the internal pressure and the inertia force as follows:

\[ P_f = 2\pi \frac{N}{60} T_{f,t} \]

The compressor power by the frictional resistance between the piston ring and the cylinder can be expressed by the speed of the piston and the friction force of the piston ring due to in-cylinder pressure.[7] The friction force between the piston ring and the cylinder is represented by the area and pressure on the pressure.

\[ F_{p,f} = \mu_{p,f} P_{cly} A_p \]

\[ P_{ring} = \frac{3F_{p,f} S'}{2\pi} \]

The compressor power by the bearings is obtained by calculating the force on each bearing to the shaft as the force and moment balance equations.

\[ \sum F_y = 0, \quad F_y + F_{r,1} + F_{r,2} = 0 \]
\[ \sum F_z = 0, \quad F_z + F_{thrust} = 0 \]
\[ \sum M = 0, \quad F_z P_{CR} + F_y S' z + F_{r,1} l_{r,1} + F_{r,2} l_{r,2} + F_{thrust} r_{thrust} = 0 \]

The force on each of the radial bearings of both end of the shaft is as follows:

\[ F_{r,2} = \frac{(P_{CR} - r_{thrust}) F_z + \left( S - l_{r,1} \right) F_y}{l_{r,1} - l_{r,2}} \]

\[ F_{r,1} = -(F_y + F_{r,2}) \]

The compressor power by the frictional resistance of the radial bearings is indicated by the rotational angular velocity and the friction torque.

\[ T_{r,1} = \mu_{r,1} r_{r,1} F_{r,1} \]
\[ T_{r,2} = \mu_{r,2} r_{r,2} F_{r,2} \]
\[ T_{thrust} = \mu_{thrust} r_{thrust} F_{thrust} \]
\[ T_{total} = T_{r,1} + T_{r,2} + T_{thrust} \]

Thus, the compressor power by the bearings is as follows:

\[ P_{bearing} = 2\pi \frac{N}{60} T_{total} \]

Therefore, the shaft power of the compressor can be determined by:

\[ P_{shaft} = P_f + P_{bearing} + P_{ring} + P_{comp} \]
The compressor input power can be determined by the shaft power and the efficiency of the electric parts (inverter and motor). Then, the compressor input power is derived as follows:

\[
P_{input} = \frac{P_{shaft}}{\eta_{elec}}
\]

Bin Yang [7] is to calculate the efficiency of the motor by Hubacher’s study [8]. It was proposed to the iterative method by the input power. However, this equation can’t be applied to any motors. In this study for a more accurate prediction, the power loss by the efficiency of the electric parts can be determined by the dynamo test results.

3. Dynamo Test

Fig. 2 is a device configuration for the dynamo test of the motor and the inverter. The equipment has the dynamometer (Magtrol, 1WB65) with accuracies of ±0.3% respectively and the controller for measuring a torque and giving a load on the test motor. It is installed the test inverter to drive the test motor. And the can board and the computer to translate the can communicate instructions and responses was installed. The power meter was installed to measure the power before and after the inverter with accuracies of ±0.1% respectively. It can calculate the efficiency of the inverter, and it is possible to calculate the efficiency of the motor from the dynamo test result. The dynamometer is measured the torque, the phase current and the phase voltage of the test motor according to the load at constant speed. Dynamometer test method is as follows. The motor and inverter to provide for the test is drive at a constant rotational speed. The dynamometer increases the torque by the break. This acts as a load on the motor. It measures the input power of the motor according to increase of the load at a constant rotational speed. The torque to be applied in the dynamometer is the shaft power of the compression section. Therefore, we can calculate the efficiency of the motor and inverter according to the various shaft load at each rotational speed.

![Fig. 2 Schematic of dynamo test rig](image)

The dynamo test results can be represented by the efficiency of the electric parts according to the shaft power at the rotation speed. In case of low speed, the efficiency of the motor and inverter was lower. The efficiency is measured up to 78.9% at 2000rpm. And it is measured up to 84.4% at 3000rpm. As the increase of the rotational speed, its efficiency is increased. In more than 6000rpm measured more than 90% efficiency.

The efficiency of the electric parts was approximated by the following 4th order polynomial equation. Table 2 is constants of the 4th order polynomial equations which approximates the shaft power and the efficiency of the electric parts according to each rotational speed.
\[ \eta_e = a + bP_{\text{shaft}} + cP_{\text{shaft}}^2 + dP_{\text{shaft}}^3 + eP_{\text{shaft}}^4 \]

Therefore, the efficiency of the electric parts can be calculated by the compression power (shaft power) in the calculated performance analysis of the mechanical parts. The total power of the electric driven compressor can be calculated in consideration of the electric parts efficiency. Thus, the compressor performance contributed more accuracy by using the electric parts efficiency.

Table 2. Constants of the efficiency equation for the electric parts

| Speed [rpm] | 2000 | 3000 | 4000 | 5000 | 6000 | 7000 | 8000 | 8600 |
|------------|------|------|------|------|------|------|------|------|
| C          | 6.6933E-01 | 7.3146E-01 | 6.1707E-01 | 7.0600E-01 | 5.4996E-01 | 6.6170E-01 | 6.1125E-01 | 5.4863E-01 |
| b          | 4.7397E-04 | 2.3404E-04 | 4.4717E-04 | 2.3303E-04 | 4.7095E-04 | 2.5128E-04 | 3.3202E-04 | 3.9498E-04 |
| c          | -6.1928E-07 | -1.7103E-07 | -2.8080E-07 | -1.0872E-07 | -2.2525E-07 | -9.7034E-08 | -1.4086E-07 | -1.6658E-07 |
| d          | 3.1113E-10 | 5.1302E-11 | 7.4179E-11 | 2.2118E-11 | 4.5657E-11 | 1.6752E-11 | 2.6103E-11 | 3.0924E-11 |
| e          | -5.7696E-14 | -5.9301E-15 | -7.1889E-15 | -1.7049E-15 | -3.3388E-15 | -1.1062E-15 | -1.7830E-15 | -2.1119E-15 |

4. Experimental equipment and electric driven compressor using R744

To evaluate the performance of the prototype compressor using the R744 was constructed hot-gas cycle. A schematic of the test rig for R744 compressor is shown in Fig. 3. The electric driven compressor is used which includes an AC/DC converter for operating the compressor. The experimental setup utilized thermocouples, pressure transducers, mass flow meter for refrigerant and oil, another mass flow meter for separated oil and power meter with accuracy of ±0.1 K, ±0.33%, ±0.25%, ±0.25%, ±0.3% respectively.

![Fig. 3 Schematic of the hot-gas cycle for the test of compressor](image)

Test conditions are fixed at suction pressure 3.92MPa, suction temperature 30℃. Discharge pressure is changed at 7.85, 9.81, 11.77MPa. And discharge temperature, mass flow rate, and input power of compressor are measured by the each sensor.

The compressor using the test is new developed prototype compressor of the electric driven compressor for CO2. The specifications of the compressor used in the experiment were shown in the table 3. The development of a fixed swash plate type compressor is a conventional compression method for using the vehicles. The compressor was equipped with a speed-adjustable BLDC motor. The inverter for motor driving was installed outside to the compressor. It is made up of three pistons and stroke volume of 4.7cc/rev. Its top speed is 8600rpm. In order to prevent leakage for the use of high-pressure refrigerant, it applied the piston ring. The clearance controller was installed in the compressor to consider the effects of dead volume in the small capacity. This can minimize the influence of dead volume because of the
structure to adjust the top clearance. It was placed in a thrust bearing with motor rearward to support the axial load of the shaft by the high-pressure refrigerant in the cylinder. Lubricating oil for the compressor is applied the PAG oil. An inverter for driving a motor was used as the high-efficiency inverter applying the maximum torque control. The motor and inverter efficiency through a dynamo test was measured previously.

Table 3 Specifications of electric driven compressor for CO₂

| Model       | DEC47     |
|-------------|-----------|
| Type        | Electric driven compressor (3 Piston) |
| refrigerant | R744 (CO₂) |
| Capacity    | 4.7 cm³/rev |
| Motor       | BLDC motor (max. 7N.m) |
| Input Voltage | 330VDC    |
| Max. Speed  | 9,000 rpm |

5. Comparison of numerical and experimental results

An electric driven CO₂ compressor with fixed swash plate was tested using the hot gas system according to the pressure ratio and the speed of compressor. The experimental results are compared to the theoretical analysis results.

Fig. 4 show the deviations between measured values and predicted values for the discharge temperature of the compressor. The red solid line is predicted value, the blue dotted line represents an around of the analysis result of ±5%. And black squares with error bar are experimental results. The suction pressure and temperature is 40 kgf/cm² and 30°C. The discharge pressure is 80, 100, 120 kgf/cm². The discharge temperature increase with the pressure ratio. The analysis model predicts the discharge temperature with the error percentage of ±5%.

Fig. 5 show the deviations between measured values and predicted values for the mass flow rate of the compressor. The mass flow rate increase with the speed of compressor and the pressure ratio. The analysis model predicts the discharge temperature with the error percentage of ±5% too.
Fig. 5 Comparison of experiment and predicted mass flow rate

Fig. 6 show the deviations between measured values and predicted values for the compressor input power. And it show the result to calculate the efficiency of the motor by Hubacher’s study[8]. The input power of compressor increase with the speed of compressor and the pressure ratio. The shaft power by the electric parts transforms a portion of the compressor input power. As the comparison results, the exact efficiency of the electric parts for electric driven compressor can provide more accurate prediction. The predicted input power of the compressor agrees quite well with the experimental value, as shown in Fig. 6.

The accurate prediction result is because the correct efficiency of the electrical components in the compressor by measurement with the load of the compression unit and the rotational speed of compressor through the dynamo test. In general, the industry using a BLDC motor and inverter applied the efficiency of 90% and 95%. In past studies often apply a simple one formula. But the BLDC motor and inverter causes the larger error when interpreting this one formula because it is designed for maximum efficiency out at the design speed and load. In the case of the compressor for the vehicles, the motor is designed to have high efficiency come at a rotational speed of 6000 ~ 8000rpm, shaft power of 2000 ~ 4000W. This is designed in accordance with the driving conditions of the electric driven compressor within the vehicles.
6. Conclusions
A comprehensive simulation model for an electric driven compressor using CO$_2$ for eco-friendly vehicle is presented. This model consists of a compression model and dynamic model. Compression model included valve, leakage, and heat transfer models. And Dynamic model included frictional loss between piston ring and cylinder wall, frictional loss between shoe and swash plate, and frictional loss of bearings. Especially, the efficiency of an electric parts(motor and inverter) in the compressor affects the loss of the compressor.

In this study, for more accurate predictions, its efficiency was formulation through dynamo test. The efficiency of the electric parts was approximated by the following 4th order polynomial equation. It is constants of the 4th order polynomial equations which approximates the shaft power and the efficiency of the electric parts according to each rotational speed. The efficiency of the electric parts can be calculated by the compression power (shaft power) in the calculated performance analysis of the mechanical parts. The total power of the electric driven compressor can be calculated in consideration of the electric parts efficiency. Thus, the compressor performance contributed more accuracy by using the electric parts efficiency.

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