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# Design and Development of Experimental Facility for Supercritical $CO_2$ Two-Stage High-Pressure Ratio Centrifugal Compressor

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## 1. Introduction

Supercritical $CO_2$ ($sCO_2$) power generation differs from conventional steam power generation in that it uses $sCO_2$ as the working fluid instead of steam, operating at high temperatures and pressures to drive a turbine. The supercritical state refers to a condition where a substance exists above its critical temperature and pressure, exhibiting properties of both liquid and gas simultaneously. The critical point of $CO_2$ is 31.1°C and 73.8 bar. Compared with conventional steam power generation, $sCO_2$ power systems offer advantages such as higher efficiency and compactness, making them a promising power conversion system for advanced nuclear reactors [1].

Korea Atomic Energy Research Institute (KAERI) has been conducting research on $sCO_2$ power generation technology, developing the Supercritical $CO_2$ Integrated Experimental Loop (SCIEL) and a 500 kW-class waste heat recovery $sCO_2$ power system prototype to develop power systems in the kilowatt to several hundred kilowatt range [2-3]. Additionally, centrifugal compressors for these systems have been developed [4].

The $sCO_2$ power generation system based on the Brayton cycle must operate at high temperatures and pressures to achieve high thermal efficiency. Therefore, to maximize system efficiency, the development and operation of compressors with a high-pressure ratio of 2.5 or higher are required. Previous studies have focused on a single-stage compressor with a lower pressure ratio, limiting the overall system efficiency in $sCO_2$ power systems. This study aims to overcome these limitations by developing a two-stage centrifugal compressor with a pressure ratio of 2.6. To achieve this, a counter-rotating impeller structure was adopted to minimize axial thrust. The mass flow rate remains similar to that of previous studies.

This paper provides a detailed explanation of the design of the $sCO_2$ two-stage high-pressure ratio compressor experimental facility and key design considerations.

## 2. Overview of the Multi-megawatt-class $sCO_2$ Power System

A 6.5 MWth molten salt reactor with a turbine inlet temperature 630°C was selected as the heat source. To enhance reliability and efficiency, two independent power conversion systems were designed, and the most efficient recompression cycle was applied. The efficiency parameters of the compressor, turbine, and heat exchanger were determined, and a cycle analysis was conducted to establish the design requirements for the compressor. The development of a two-stage high-pressure ratio centrifugal compressor is a critical step toward implementing a multi-megawatt $sCO_2$ power system, as it ensures stable operation and high efficiency.

## 3. Description of Experimental Facility for the $sCO_2$ Compressor

This section explains the major components of the test loop, including the compressor, control valve, precooler, piping design requirements, flow meter, thermometer, pressure gauge, and power analyzer, and considerations for selecting measurement equipment.

### *3.1. Compressor*

The compressor is one of the key components of the $sCO_2$ power system, directly impacting system efficiency and stability. Since compression efficiency increases when the inlet conditions approach the critical point, the compressor inlet pressure and temperature were set at 76.9 bar and 34.0°C. Additionally, the compressor was designed with a mass flow rate of 11.7 kg/s, a pressure ratio of 2.6, and an efficiency of 85%.

A centrifugal compressor was adopted, and to address axial thrust and rotor vibration concerns, a two-stage structure was designed.

With a specific speed (Ns) of 0.6, the designed rotational speed is 40,000 rpm, with the first-stage (low-pressure) specific speed at 0.6 and the second-stage (high-pressure) specific speed at 0.535. To derive a flow-pressure ratio curve (compressor performance map), a high-speed inverter and motor were incorporated to adjust the rotational speed.

*3.2. Precooler*

The precooler, which regulates the fluid temperature before compression, utilizes a printed circuit heat exchanger (PCHE). The maximum heat capacity is set to 517 kW, which is 150% of the capacity required for rated operation, considering the margin. The cooling water inlet temperature was set at 25°C with a flow rate of 15 kg/s. Pressure drop within the $sCO_2$ region was limited to 1.5 bar, and this value was determined based on existing $sCO_2$ power generation experimental data. The pressure drop reflects the total loss value, including PCHE channel region, header, and nozzles.

Precooler dimensions were set at 850 mm (L) × 427.8 mm (W) × 400 mm (H), while meeting the required heat capability.

*3.3. Control Valve*

The control valve is located downstream of the compressor and functions to reduce the high pressure at the compressor outlet to a lower pressure during the compressor performance test. The fluid conditions for the applied control valve are:

- Fluid temperature: 92.4°C
- Inlet/Outlet pressure: 200 bar / 76.92 bar
- Flow range: 10–150% of the rated mass flow rate (1.17–17.55 kg/s)

To minimize noise and vibration, a multi-stage, multi-hole piston-type control valve was used. A four-stage perforated cage trim was implemented to suppress cavitation and stabilize flow.

*3.4. Piping*

Considering the mass flow rate of the main compressor (11.7 kg/s) and the system's maximum pressure/temperature (200 bar/92.4℃ at the compressor outlet), piping was selected to meet these design conditions.

For future re-compressor development and power system expansion, SUS316L was chosen as the piping material due to its ability to maintain mechanical integrity even at high temperatures. The maximum allowable thickness of the seamless pipe was calculated according to ASME B31.1 standard [5], and based on this, SUS316L 4-inch SCH160 pipe was selected.

Given the limited installation space, SUS316L SCH160 elbows were used to adjust the piping direction. The elbows and pipes were joined using argon welding. For the connection between straight pipe sections, flanges were applied to ensure ease of maintenance. To maintain stable operation under high-pressure conditions, SUS316L ANSI Class #2500 standard flanges were used.

To measure pressure and temperature, a 1/2-inch pipe was used to create taps for connecting measurement instruments.

*3.5. Measurement Instrument*

To evaluate compressor performance, flow rate, pressure, and temperature were measured in real time based on compressor speed variations.

For $CO_2$ flow measurement, a mass flow meter and a Venturi flow meter were used, while an electromagnetic flow meter was applied for water flow measurement. Pressure was measured using a pressure transmitter, and temperature was measured using an RTD (Resistance Temperature Detector). The power consumption of the compressor was precisely measured using a power analyzer.

To ensure the reliability of the experiment, all measurement instruments were calibrated in advance before installation at each measurement location. Data acquisition was performed using a Data Acquisition System (DAS), with signal lines connected to the DAS input terminals for real-time data collection and analysis.

## 4. Experimental Facility for $sCO_2$ Compressor and Partial load Test Results

*4.1. Experimental Facility Setup*

As shown in Figure 2, a compressor performance testing loop was established by connecting a two-stage compressor, control valve, precooler, and mass flow meter. The previously mentioned instrumentation was installed to complete the compressor performance testing loop.

Fig. 1. Experimental facility for $sCO_2$ two-stage high-pressure ratio centrifugal compressor

*4.2. Preliminary Test Conditions and Results*

The compressor was driven by a high-speed electric motor. Prior to CO2 charging, a dry-run test was conducted, during which the compressor was operated at its rated rotational speed of 40,000 rpm in ambient conditions to verify mechanical integrity.

After charging the system with $CO_2$, an initial performance test was performed under full valve open condition (100%) at a rotational speed of 30,000 rpm without inducing significant pressure build-up.

Subsequent performance tests were conducted under partial load conditions by adjusting the control valve to reduce the outlet pressure. These tests were performed at 24,000 rpm and 27,000 rpm, respectively. The measured values of rotational speed, mass flow rate, pressure ratio, and thermodynamic conditions are summarized in Table 1.

Table I: Compressor Test Results under Partial Load Conditions

| Component | Test 1 (24,000 rpm) | Test 2 (27,000 rpm) |
|---|---|---|
| Control Valve (%) | 41.15 | 45.84 |
| Mass flow rate (kg) | 3.86 | 4.77 |
| Inlet Temperature (℃) | 38.37 | 47.23 |
| Inlet Pressure (MPa) | 8.02 | 9.02 |
| Outlet Temperature (℃) | 52.17 | 61.54 |
| Outlet Pressure (MPa) | 11.16 | 12.27 |
| Pressure ratio | 1.39 | 1.36 |
| Isentropic Efficiency (%) | 47.68 | 35.41 |

### *4.3 Drive loss estimation and shaft power calculation*

To calculate the isentropic efficiency, the net power delivered to the compressor shaft was estimated by subtracting electrical and mechanical losses from the motor input power. These included motor loss (estimated as ~2% of input power), and bearing losses.

Table II: Estimated Power Losses in the Drive System

| Component | Test 1 | Test 2 | Note |
|---|---|---|---|
| Electric Motor input power (kW) | 101.99 | 165.65 | From power analyzer |
| Motor loss (kW) | 2.04 | 3.31 | Approx. 2% of motor input power |
| Bearing loss (kW) | 24.25 | 34.53 | Estimated from previous test data |
| Net power to Compressor shaft (kW) | 75.70 | 127.81 | Used for efficiency calculation |

### *4.4 Comparison with design point and discussion*

The experimental results show that both mass flow rate and efficiency were significantly lower than the design targets (11.7 kg/s, 85 % efficiency). The main reasons for the discrepancies are as follows: First, the tests were performed under partial load conditions (24,000–27,000 rpm), far below the design speed of 40,000 rpm. Second, due to the low temperature of the cooling water, the compressor inlet conditions could not reach the intended design conditions, especially near the critical point. This deviation in inlet conditions affected the compression process and resulted in lower isentropic efficiency and reduced flow performance.

Figure 2 shows the trend of performance across the test cases.

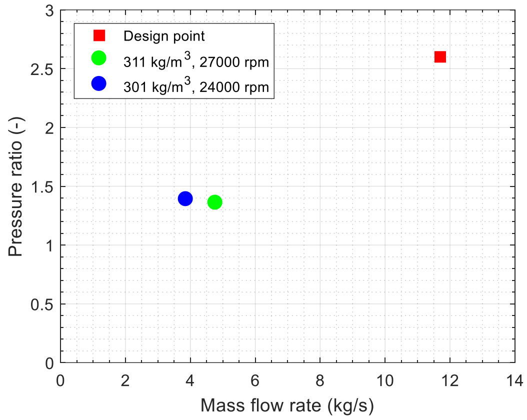

Fig. 2. Preliminary experiment result

In future work, full-load tests will be conducted at 40,000 rpm, and the cooling system will be tuned to meet the target inlet conditions for accurate performance evaluation.

## 5. Conclusion

To develop and test the compressor, a key component of a multi-megawatt class $sCO_2$ power system, a compressor performance test facility was designed, constructed and evaluated. The primary components, including the compressor, precooler, and control valve, were designed and manufactured. Additionally, piping materials and specifications suitable for the operating environment were selected and fabricated. Measurement instruments for flow rate, temperature, and pressure were chosen, and a Data Acquisition System (DAS) was implemented for real-time data collection.

Future studies will focus on further optimizing the compressor design, conducting full-load performance tests, and exploring its applicability to large-scale $sCO_2$ power generation systems.

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