Aerodynamic Characteristics of a Centrifugal Compressor Working in Supercritical Carbon Dioxide

Motoaki Utamura\textsuperscript{a,*}, Taro Fukuda\textsuperscript{b}, Masanori Aritomi\textsuperscript{a}

\textsuperscript{a}Tokyo Institute of Technology, 2-12-1 Ookayama, Meguro-ku, Tokyo 152-8550, Japan
\textsuperscript{b}Daiichi System Engineering Co. Ltd., 1-16-30 Meiekiminami, Nakamura-ku, Nagoya 450-0003, Japan

Abstract

Development of a closed cycle gas turbine with supercritical carbon dioxide as a working fluid is underway to generate power from low-range or intermediate-range waste heat sources. A demonstration test using a reduced scale turbomachine was conducted and the aerodynamic characteristics of a compressor were examined. A compressor was selected as centrifugal with 30 mm outer diameter and rated rotational speed of 1.7 kHz and mass flow rate of 1.2 kg/s. To reduce compression work, the operating condition at the inlet to compressor was chosen in a supercritical state close to the critical point of 7.38 MPa, 304 K where the compressibility coefficient $z$ becomes markedly small and real gas effect dominant. The experimental range in terms of $z$ is $0.16 < z < 0.6$. The measured pressure ratio and adiabatic efficiency of the compressor are compared with calculations conducted using the Meanline method. The compressor performance in the supercritical liquid-like phase becomes highest in the experiment, which is well simulated by the method. The calculated pressure ratio shows excellent matching with experimental data in the supercritical liquid-like phase. However overestimation is recognized at the off-design point in the supercritical gas-like or subcritical region. Experiments also show that the compressor performance improves with reduction of the compressibility coefficient, which the Meanline method has well predicted.

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Keywords: supercritical state, carbon dioxide, centrifugal compressor, turbine, power cycle

* Corresponding author. Tel.: +81357343293; Fax: +81357343293.
E-mail address: utamura@nr.titech.ac.jp.

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1. Introduction

Gas in a supercritical state close to the critical point might engender the reduction of compression work compared to ideal gas because of the real gas effect. Moreover, carbon dioxide has a critical point at around room temperature. Therefore, it is possible to configure an efficient gas turbine power cycle that functions even at a low or intermediate temperature range. Theoretical studies were reported in the literature of the late 1960s [1, 2].

The cycle would facilitate power conversion from unused energy in a low-temperature region such as waste heat from industry or renewable energy. Because the power cycle is a closed regenerative one, it necessitates unusually high efficiency at the regenerative heat exchanger involved in the cycle. Then realization of the power cycle necessitates a high-performance compact heat exchanger with high temperature and also with high mechanical strength that is resistant to supercritical pressure up to 20 MPa. To meet those requirements, Tokyo Tech developed an X-shaped fin microchannel heat exchanger that is made of plural metal plates with flow channels engraved and integrated by diffusion bonding [3].

Based on these fundamental studies, compressor tests were conducted, confirming stable operation close to the critical point [4]. Then, a demonstration test plant of 10 kW power cycle was built [5] and implemented. Continued power generation of 200 W was realized [6]. The cycle compressor plays a vital role in enhancing cycle thermal efficiency. To this end, it operates in a supercritical state near the critical point of carbon dioxide with critical pressure of 7.38 MPa and critical temperature 304 K, where thermal properties change considerably according to a small change of thermodynamic state. This paper describes the performance characteristics of a centrifugal compressor. Applicability of the Meanline aerodynamic design method [7] to widely various operations from a subcritical to a supercritical state was discussed.

2. Experiment

2.1 Supercritical CO$_2$ power cycle

Figure 1 presents a carbon dioxide phase diagram. The critical point at which a phase change disappears is located at pressure 7.38 MPa and temperature 305 K. The whole of the present power cycle is to be formed in a supercritical state so that the state at the compressor inlet should be located as close as possible to the critical point. That is true because compressor work is reduced most there, which would result in enhanced thermal efficiency of the power cycle. The thermodynamic rationale of selecting carbon dioxide as the working fluid is that the critical temperature exists near room temperature, which benefits easy cooling and effective utilization of degraded heat as a heat source. The typical configuration of the cycle is presented in Fig. 2, called a closed regenerative Brayton cycle. It was practiced in the present demonstration test. Working fluid recirculates the whole cycle, undergoing compression, heating, expansion, cooling, and also generating electricity.
2.1 Test apparatus

Figure 3 portrays a cross section of a rotor, which comprises a permanent magnet, a centrifugal compressor, and a radial inflow turbine. A compressor and a turbine are aligned co-axially back-to-back to reduce net thrust. A synchronous motor/generator was driven by an inverter up to 60 000 rpm (1 kHz). Because the rated speed of the rotor was designed as 100 000 rpm (1.7 kHz), CO₂ gas bearings were adopted as journal and thrust bearings. Differential pressure acting on the wheel was estimated as around 4 MPa. Because the thrust is too great for a tapered-land thrust gas bearing to withstand, some compressed gas was introduced between the thrust bearing and the thrust collar to withstand it. The gas passed around the rotor to reach the back face of the turbine wheel to be sucked at the turbine impeller inlet. Incidentally it functioned to reject heat from the electromagnetic loss of the permanent magnet as well as from windage loss i.e. fluid shear around the rotating cooling rotor. The highest allowable rotor temperature was 180°C because the existence of high-density CO₂ windage loss in a typical operating condition was estimated as 5 kW: 100 times more than that of atmospheric air. Pictures of fabricated turbomachines are presented in Fig. 4. Outer diameters of the aluminum compressor and titanium turbine are 30 mm and 35 mm, respectively. The blade height is 1 mm at the compressor exit, where the diffuser is connected with the collector. All compressor-related measurements were made at external piping that was the inlet for total temperature and total pressure, and the exit for static pressure and temperature. Table 1 shows basic design specifications. Compressor inlet operating conditions were determined to account for both aspects of seeking minimum possible compressibility coefficient and avoidance of liquid formation at the mouth of the compressor, where flow is accelerated. A tradeoff study showed tolerance of 4 K. Given this and the turbine inlet temperature of 550 K and compressor outlet pressure of 12 MPa, the compressor inlet pressure was determined by cycle calculation to obtain maximum thermal efficiency. The CO₂ recirculating flow rate was 1.2 kg/s necessary to give 10 kW power output. Aerodynamic performance was designed to become maximum at the rotational speed of 1.7 kHz.

2.2 Measurement results

Unexpected large windage loss prevented the system from reaching self-sustaining operation. Further access to the critical point at the compressor inlet condition finally led to continuous power generation.

<table>
<thead>
<tr>
<th>Table 1 Major design specifications</th>
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<tbody>
<tr>
<td><strong>Compressor</strong></td>
</tr>
<tr>
<td>inlet press.</td>
</tr>
<tr>
<td>inlet temp.</td>
</tr>
<tr>
<td><strong>Turbine</strong></td>
</tr>
<tr>
<td>inlet press.</td>
</tr>
<tr>
<td>inlet temp.</td>
</tr>
<tr>
<td>Flow rate</td>
</tr>
<tr>
<td>Rotational speed</td>
</tr>
<tr>
<td>Power output</td>
</tr>
</tbody>
</table>

![Fig. 3 Cross section view of a rotor](image1)

![Fig. 4 Pictures of impeller blades](image2)
Figure 5 exhibits a time history of parameters during power generation. At the time of 945 s, the rotational speed of 1.17 kHz produced 1.11 kg/s compressor mass flow rate and pressure ratio of 1.44 at turbine inlet temperature of 264.9°C (538 K). Most are close to design values. However, the power output was 206 W at most. Work balance showed that this arose primarily from windage, as shown in Table 2.

3. Meanline analysis of compressor aerodynamics

3.1 Results

Meanline calculations were conducted based on the experimental data. In general, this calculation predicts performance in each component section along the mean streamline using empirical modeling parameters. Refining such parameters to match measurement data and thereby predict performance with high accuracy is an important task. In this task, the slip factor and impeller relative Mach number ratio (MR2) were investigated as the major impeller modeling parameters evaluated at the design point. MR2 was defined by the ratio of the relative Mach number at the impeller inlet tip and impeller exit (primary zone [7]). This task was executed using the Meanline program “COMPAL” developed by Concepts ETI Inc. The slip factor of 0.9 and MR2 of 1.21 were selected as design points. Figure 6 shows that the predicted efficiencies lie in the band due to temperature measurement error of ±0.1K, although the prediction of pressure ratio agreed well with measurements in widely differing mass flows. Red color shows the condition where model tuning was made.

![Graph of pressure ratio vs. mass flow ratio](image)

**Table 2 Work balance**

<table>
<thead>
<tr>
<th></th>
<th>Generation (kW)</th>
<th>Consumption (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbine</td>
<td>17.4</td>
<td>Windage</td>
</tr>
<tr>
<td>Load</td>
<td>0.1</td>
<td></td>
</tr>
<tr>
<td>Miscellaneous</td>
<td>1.5</td>
<td></td>
</tr>
<tr>
<td>Sum</td>
<td>17.4</td>
<td>Sum 17.4</td>
</tr>
</tbody>
</table>

**Fig. 6(a) Pressure ratio vs. design mass flow ratio, (b) Efficiency vs. design mass flow ratio**
The occasion of the efficiency difference between prediction and experiment is regarded as deriving from the fact that the compressor exit thermodynamic state was located near the pseudo-critical line, where, as portrayed in Fig. 7, a small temperature difference produces a large exit specific enthalpy change. Because of both the measurement error of the compressor exit temperature and limitation of accuracy of Meanline analysis, further improvement of the performance prediction at the design point remains as a future design issue. Figure 8 presents the off-design performance curve, which is divided to four regions: a subcritical region \((z=0.49–0.6)\), a supercritical gas-like region \((z=0.3–0.47)\), a supercritical liquid-like high pressure region \((z=0.253–0.29)\), and a supercritical liquid-like low pressure region \((z=0.215–0.251)\) in which compressibility coefficient \(z\) is defined as \(z=pv/RT\), where \(p\) stands for pressure, \(v\) signifies the specific volume, \(R\) denotes the gas constant, and \(T\) represents temperature. Actually, \(v\) was calculated using the thermophysical property library PROPATH [8]. Results show that the pressure ratio prediction tends to depart from experimental values as the thermodynamic state moves apart from supercritical to subcritical as the mass flow rate decreases. However, the efficiency shows only a very slight change with the flow rate.

3.2 Discussion

The reason for the discrepancy between calculations and experiments in Fig. 8a is regarded as follows. The fluid gains specific work through an impeller passage, as expressed by Euler’s equation:

$$E = \frac{U_2^2 - U_1^2}{2} + \frac{C_2^2 - C_1^2}{2} - \frac{W_2^2 - W_1^2}{2}$$ (1)

In that equation, \(U\) signifies the impeller tip speed with suffices 1 and 2 denoting inlet and exit respectively, \(C\) denotes the absolute velocity, and \(W\) stands for the relative velocity in a rotational coordinate in which static pressure rise depends on the first and third terms. The first term signifies the centrifugal force to raise static pressure at the exit. With an increased amount of the compressibility coefficient, it was observed that \(W_2\) is larger than \(W_1\), so that the third term in Eq. (1) became negative. This caused a decrease in MR2, which means that the impeller relative velocity at the exit is greater than that at the inlet tip. Consequently, the pressure ratio in the subcritical phase \((MR<1)\) is smaller than that in the supercritical liquid-like phase \((MR>1)\).
Next, it must be discussed whether the compressibility coefficient affects aerodynamic performance or not. Figure 9 presents an evaluation result of the compressibility coefficient under two constant flow coefficients. Both the pressure ratio and efficiency decreased concomitantly with increased compressibility coefficient $z$. This fact implies that the compressor performance is enhanced in a supercritical state.

![Fig. 9(a) Pressure ratio vs. design mass flow ratio, (b) Efficiency vs. design mass flow ratio](image)

4. Conclusion

Verification testing of a gas turbine with supercritical fluid of carbon dioxide was conducted. Also conducted were refinement and evaluation of the performance using the Meanline program. The compressor performance in the supercritical liquid-like phase achieved the highest performance in both experiments and predictions by the simulation program. The Meanline program also predicted better matching with experimental data in the supercritical liquid-like phase. However overestimation of the pressure ratio develops as a thermodynamic phase of carbon dioxide shifts from supercritical phase through subcritical phase at compressor inlet. Experiments also showed improvement of compressor performance with reduction of compressibility coefficient, which the Meanline program well simulated.

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References


