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Citation	Energy Procedia ELSEVIER, Volume 14, , page 1149 – 1155
Pub. date	2012,
DOI	http://dx.doi.org/10.1016/j.egypro.2011.12.887
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Energy Procedia 00 (2011) 000–000
Energy Procedia
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2011 2nd International Conference on Advances in Energy Engineering (ICAEE 2011)

Aerodynamic Characteristics of a Centrifugal Compressor working in Supercritical Carbon Dioxide

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Abstract

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

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

1. Introduction

Gas in a supercritical state close to the critical point may lead to the reduction of compression work compared to ideal gas owing to real gas effect. Moreover, carbon dioxide has the critical point around room temperature. Then it is possible to configure an efficient gas turbine power cycle working even in a low/intermediate temperature range. Theoretical studies can be found in the literatures in late 1960s [1,2].

The cycle would facilitate power conversion from unused energy in low temperature region such as waste heat from industry or renewable energy Since the power cycle is a closed regenerative one, it needs unusually high efficiency at regenerative heat exchanger involved in the cycle.

Then realization of the power cycle necessitates a high performance compact heat exchanger with and also with high mechanical strength resistant to supercritical pressure up to 20MPa. To meet the requirement Tokyo Tech has developed 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 basic studies, a series of compressor tests were carried out and stable operation close to the critical point was confirmed [4]. Then, a demonstration test plant of 10kW power cycle was built [5] and implemented. Continued power generation of 200W was realized [6]. In the cycle compressor plays a vital role to enhance cycle thermal efficiency. To this end it operates in supercritical state near the critical point of carbon dioxide with critical pressure7.38MPa and critical temperature 304K where thermal properties change significantly to a small change of thermodynamic state. This paper describes the performance characteristics of a centrifugal compressor used in the experiment. Applicability of meanline design method [7] to a wide range of operation from subcritical to supercritical state was discussed.

Nomenclature

ϕ	Flow coefficient($\equiv m/\rho Au$)
MR2	Relative mach number ratio of impeller inlet tip and exit
Z	Compressibility coefficient($\equiv pv/RT$)

2. Experiment

2.1 Supercritical CO_2 power cycle

Figure 1 presents phase diagram of carbon dioxide. Critical point where phase change disappears is located at pressure 7.38MPa and temperature 305K. The whole of the present power cycle is to be formed in supercritical state so that the state at compressor inlet should be located as close as possible to the critical point. It is because compressor work reduces most there, which would result in enhanced thermal efficiency of the power cycle. Thermodynamic reason of selecting carbon dioxide as working fluid is that the critical temperature exists near room temperature that benefits easy cooling and effective utilization of degraded heat as heat source. Typical configuration of the cycle is shown in Fig.2 called closed regenerative Brayton cycle and was practiced in the present demonstration test. Working fluid recirculates the whole cycle experiencing compression, heating, expansion, cooling and generates electricity.

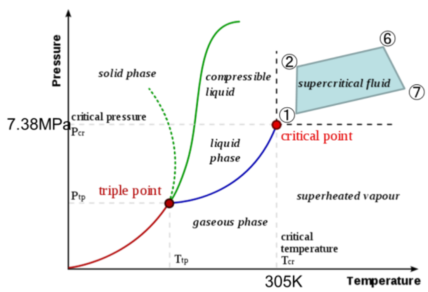


Fig.1 Phase diagram of carbon dioxide and possible location of power cycle

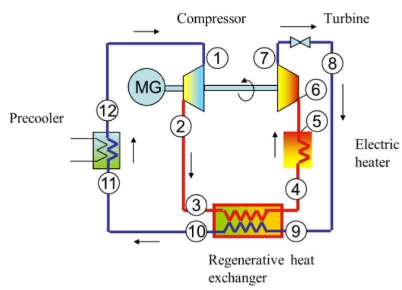


Fig.2 Schematic of test facility

2.2 Test apparatus

Figure 3 illustrates cross section of a rotor which consists of permanent magnet, centrifugal compressor and a radial inflow turbine. A compressor and a turbine are aligned co-axially back to back to reduce net thrust. Synchronous motor/ generator was driven by an inverter up to 60000rpm (1kHz). As the rated speed of rotor is designed at 100000rpm (1.7kHz), CO_2 gas bearings were adopted for journal and thrust bearings. Differential pressure acting on the wheel was estimated to be around 4MPa. As the thrust is too large to withstand by a taper-land thrust gas bearing, a part of compressed gas was introduced between thrust bearing and thrust color to withstand it. The gas passed through around the rotor to reach back face of turbine wheel to be sucked at the turbine impellor inlet. Incidentally it functioned to reject heat from electromagnetic loss of permanent magnet as well as from windage loss i.e. fluid shear around rotating cooling rotor. Highest allowable rotor temperature was 180degC. Because of the existence of high density CO_2 windage loss in a typical operating condition was estimated to be 5kW, ten times more than that of air. Pictures of fabricated turbomachines are presented in Fig.4. Outer diameters of compressor and turbine are 30mm and 35mm and materials are aluminum and titanium respectively. Blade height is 1mm at compressor exit where diffuser is connected with collector. All of compressor related measurements were made at external pipings that were inlet total temperature and total pressure, exit static pressure and temperature.

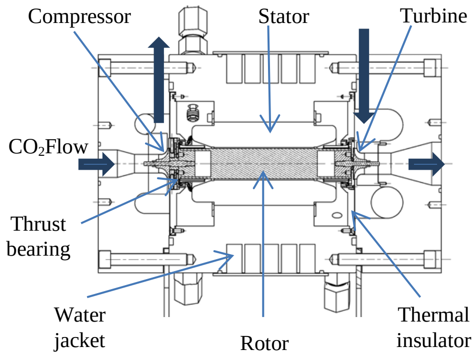


Fig. 3 Cross section view of rotor

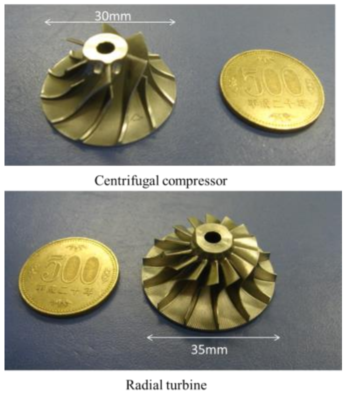


Fig. 4 Picture of impellor blades

Table 1 Basic design specifications

Compressor		
inlet press.	MPa	8.23
inlet tmp.	K	308
Turbine		
inlet press.	MPa	11.9
inlet temp.	K	550
Flow rate	kg/s	1.2
Rotational speed	kHz	1.7
Power output	kW	10

Table 1 shows basic design specifications. Compressor inlet operating conditions were determined to take into account both aspects of seeking minimum possible compressibility coefficient and avoidance of liquid formation at the mouth of the compressor where flow is accelerated. From trade off study, a tolerance of 4K was determined. Given this and turbine inlet temperature of 550K and compressor outlet pressure of 12MPa, compressor inlet pressure was determined by cycle calculation so as to obtain maximum thermal efficiency. CO_2 recirculating flow rate was 1.2kg/s necessary to give 10kW power output. Aerodynamic performance became maximum at the rotational speed of 1.7kHz.

2.3 Measurement results

Unexpected large windage loss prevented the system from reaching self- sustaining operation. Further access to critical point at compressor inlet condition finally led to a continued power generation. Figure 5 exhibits a time history of parameters during power generation. Rotational speed of 1.15kHz produced 1.1kg/s compressor mass flow rate and pressure ratio of 1.44 at turbine inlet temperature of 260degC(533K). Most of them are close to design values. However power output was 100W at most.

Work balance showed that this came primarily from windage and low aerodynamic losses as shown in table 2.

Fig.6 Design point and operation points calculation

Work balance method			
Work balance	kW	Wconsumed	kW
Turbine	13.9	Compressor	10.6
		Windage	4.9
		Power generated	0.09
sum	13.9	sum	15.59

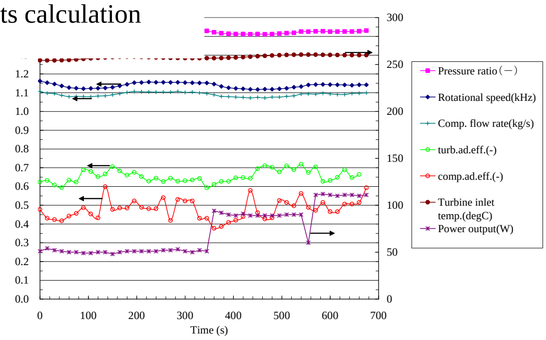
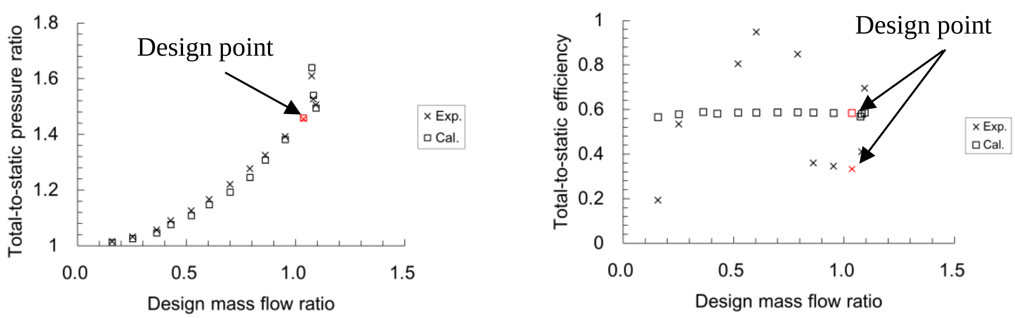


Fig. 5 Trend of parameters

3.Meanline analysis of compressor aerodynamics

3.1 Results

Meanline calculation was carried out based on the experimental data. In general, this calculation predicts performance in each component section along mean streamline using empirical modeling parameters. It is important task to refine such parameters to match measurement data in order to predict performance with very accurate. In this task, 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 relative mach number at impeller inlet tip and impeller exit (primary zone [7]). This task has been executed by meanline program "COMPAL" developed by Concepts ETI. Inc. The slip factor of 0.9 and MR2 of 1.21 were selected at design point. Figure 6 shows that the predicted efficiency at design point shows a significant mismatch with experimental data while the prediction of pressure ratio well agreed with measurements in a wide range of mass flow. The occasion of the efficiency difference between prediction and experiment is considered to come from the fact that the compressor exit thermodynamic state was located near the pseudo critical line where, as shown in Fig.7, tiny amount of temperature difference makes exit specific enthalpy change significantly. Due to both measurement error of compressor exit temperature and limitation of accuracy of Meanline analysis, discussion, further improvement of performance prediction at design point remains future design issue.



(a) Pressure ratio vs design mass flow ratio (b) Efficiency vs design mass flow ratio

Fig.6 Comparison of Meanline prediction with measurements in supercritical states

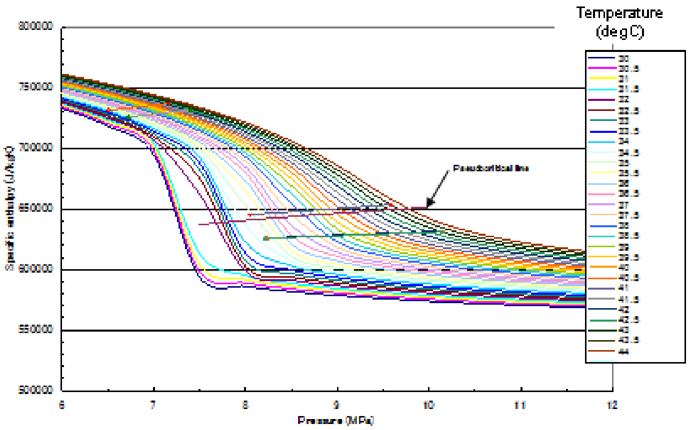
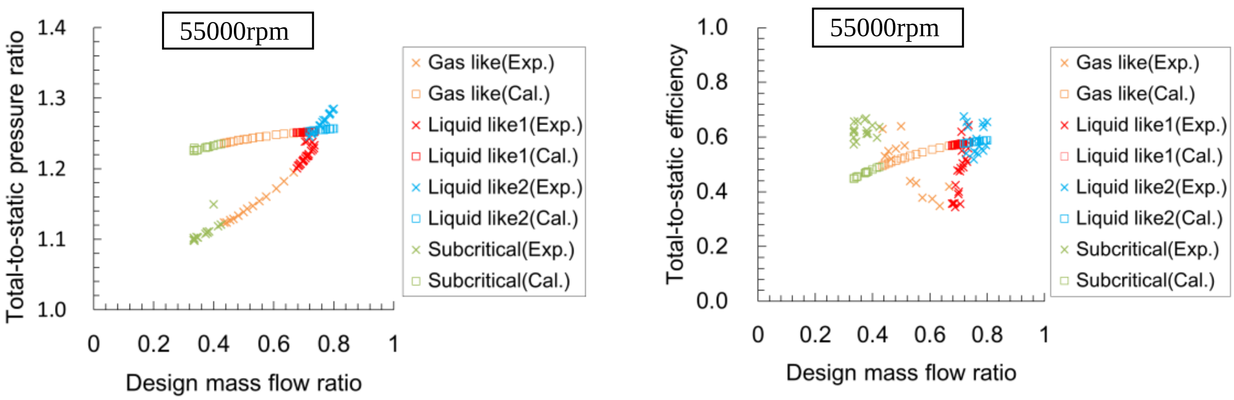


Fig.7 P-h diagram

Figure 8 presents the off-design performance curve, which is divided to 4 areas that are subcritical region (Z=0.49~0.6), supercritical gas like region (Z=0.3~0.47), supercritical liquid like high pressure region (Z=0.253~0.29) and supercritical liquid like low pressure region (Z=0.215~0.251) where compressibility coefficient Z is defined by the following equation:

$$Z = \frac{pv}{RT} \quad (1)$$

where p is pressure, v is specific volume, R is gas constant and T is temperature. V is calculated by thermo-physical property library PROPATH [8]. It shows the pressure ratio prediction tends to depart from experiments as thermodynamic state moves apart from supercritical mass flow rate with reduction of mass flow rate. On the other hand, efficiency shows insignificant change with flow rate.



(a) Pressure ratio vs design mass flow ratio (b) Efficiency vs design mass flow ratio

Fig. 8 Off design performance comparison at constant speed

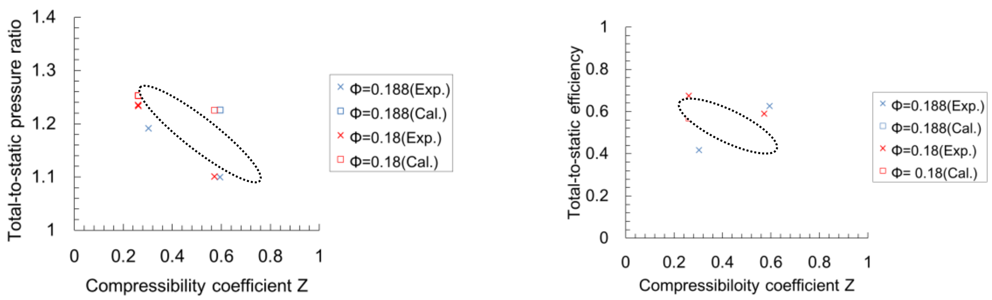
3.2 Discussions

The reason of discrepancy between calculations and experiments in Fig.8a is considered as follows. The fluid gains specific work through an impeller passage is expressed by Euler's equation

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

,where, U is impellor tip speed, C is absolute velocity and W is relative velocity in a rotational coordinate in which static pressure rise depends on first and third terms. The first term means centrifugal force to raise static pressure at exit. With increased amount of compressibility coefficient, it was observed that W2 is bigger than W1, so that the third term in the equation (2) became negative. This results in decrease in MR2 which means impeller relative velocity at exit is faster than that of inlet tip. Hence, the pressure ratio in subcritical phase (MR<1) is smaller than that in the supercritical liquid like phase (MR>1).

Next, it is discussed whether compressibility coefficient affects aerodynamic performance or not. Figure 9 shows an evaluation result of the compressibility coefficient under two constant flow coefficients. Flow coefficient here is defined by the ratio of impeller meridional velocity and rotational speed. Both pressure ratio and efficiency decreased with decrease of compressibility coefficient Z. This implies that the compressor performance enhances in supercritical phase.



(a) Pressure ratio vs design mass flow ratio

(b) Efficiency vs design mass flow ratio

Fig.9 Dependence of compressibility coefficient

4. Conclusion

The verification test of a gas turbine with supercritical fluid of carbon dioxide was carried out. It also conducted that the refinement and evaluation of the performance using Meanline program. The compressor performance in the supercritical liquid like phase achieved the highest performance in both experimental and prediction by the 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 liquid like phase through subcritical phase at compressor inlet. Experiments also showed improvement of compressor performance with reduction of compressibility coefficient, which Meanline program well simulated.

Acknowledgements

Work partly performed under the program of “Strategic Development of Energy Conservation Technology Project” and “Development of Gas Turbine with Super-critical Carbon Dioxide as Working Fluid,” sponsored by NEDO.

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