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Numerical Analysis of Super-critical Carbon-dioxide Flows in a Centrifugal Compressor

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Abstract

A new type of closed-cycle gas turbine which uses supercritical carbon-dioxide (SC-CO_2) as a working fluid was proposed and has been studied in these days. It is supposed that the behavior of SC-CO_2 will be different from that of perfect gas because the thermal properties of SC-CO_2 will change dramatically according to the slight changes of temperature or pressure in compressors. Analysis of the flow field in compressors is needed in order to operate compressors stably and to design blades optimally. In this paper, two-dimensional nozzle flow was simulated in order to confirm that the scheme can analyze the behavior of SC-CO_2 stably. Then the flow field of SC-CO_2 in a centrifugal compressor was simulated under several conditions.

As a result, entropy generation was noticeable near the vortex induced by the tip leakage flow and this caused pressure loss. So it is assumed that pressure loss leads to the dynamic changes of thermal properties and to the shift to the low-density fluid locally.

Background

Gas turbines are widely applied in various field such as jet engines and power generation. Although higher efficiency gas turbine systems are required in terms of the depletion of fossil fuels and the impact of the greenhouse effect gas on the global environment, it is difficult to improve the efficiency of each component of gas turbines dramatically because of the matured technologies.

In recent years, application of exhaust heat is much paid attention to and studied frequently. Usage of Exhaust heat which is low temperature and small-scale is a challenging problem especially. In that case, steam turbines are used conventionally but high efficiencies are not always attained. Moreover there are not any practical technologies which substitute for steam turbines at present.

A new type closed-cycle gas turbine which uses supercritical carbon-dioxide (SC-CO_2) as working fluid was proposed in these days. The advantages of the SC-CO_2 cycle are

・Several heat sources will be available owing to the closed-cycle.
・Higher generating efficiency will be attained in the region below 100,000[kw] class.

The principle of the higher efficiency is accounted for by the compression coefficient Z (Z=pV/RT), which represents the departure from the perfect gas. The compression work will be reduced by operating compressors near the critical point where the compression coefficient is low (Z=0.3～0.4) and the fluids behave like liquids, and then the turbine work will be acquired as usual by operating turbines at the region where the compression coefficient is high (Z=0.9～1.0) and the fluids behave like perfect gas (Fig.1).So as shown in Fig.2 it is thought that the SC-CO_2 cycle will produce bigger output power than conventional gas turbine cycles. It is considered that these advantages will mainly contribute to energy conservation in the division of industry.

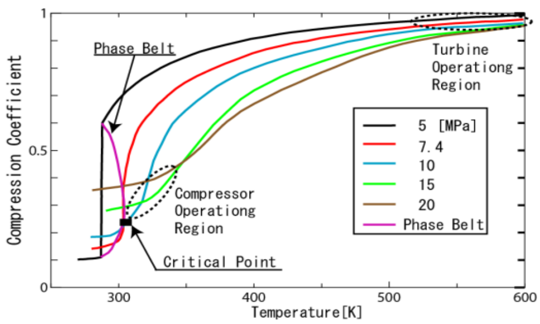


Fig. 1 Compression Coefficient and Operating Regions

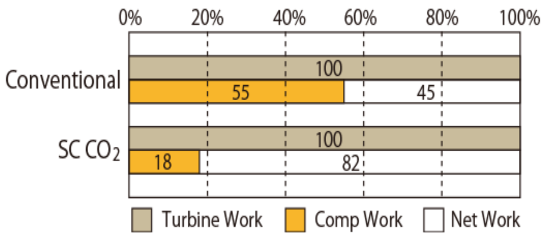


Fig. 2 Compression Work of Conventional Gas Turbines and SC-CO_2 Gas Turbines

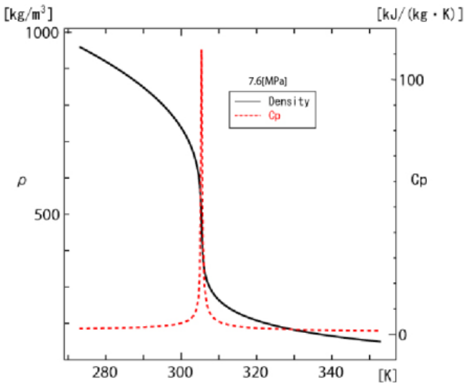


Fig. 3 Density and Cp Distributions near Pseudo-critical Temperature at 7.6[MPa]

Recently, supercritical fluids have been studied actively including the rapid expansion supercritical solution (RESS), but there are few researches focusing on the behaviors of the supercritical fluids in compressors, in which the flow fields become complex. When the thermal properties change dramatically in compressors, it is difficult to apply traditional design methods supposing perfect gas. So it makes a difference to figure out the behavior of the SC-CO_2 in compressors and to get some guidelines of designing a compressor cascade.

Numerical Method

Numerical Scheme

Steady three-dimensional flow simulations were performed with compressible Navier-Stokes equations. The three-dimensional Reynolds-averaged Navier-Stokes equations were discretized in space using a cell center finite volume method and in time using the Euler implicit method with LU-SGS scheme. The inviscid fluxes were evaluated by the SHUS scheme, where the third order MUSCL interpolations were implemented. The viscous fluxes were determined in the central differential manner and the k-ω turbulence model was equipped to estimate the eddy viscosity.

Equation of State

In this paper to capture rapid variations in the thermal properties of s SC-CO_2, Peng-Robinson equation of state[3] (PR EoS) was adopted. The PR EoS is of the form:

$$p = \frac{RT}{V-b} - \frac{a(T)}{V^2 + ubV + wb^2} \quad (1)$$

$$u = 2\ ,\ \ w = -1 \quad (2)$$

$$a = \frac{0.45724R^2T_c^2}{P_c}[1 + f(\omega)(1 - T_r^{0.5})]^2 \quad (3)$$

$$b = \frac{0.07780RT}{P_c} \quad (4)$$

$$f(\omega) = 0.37464 + 1.5423\omega - 0.26992\omega^2 \quad (5)$$

$$T_r = \frac{T}{T_c} \quad (6)$$

Table.1 Critical constant and acentric factor of carbon-dioxide

T_c [K]	P_c [MPa]	V_c [cm^3/mol]	ω	M [g/mol]
304.1	7.377	94.12	0.239	44.01

Viscosity was calculated by the method of chung[1] and thermal conductivity was calculated by the method of Scalabrin[2].

Flows in a Two-dimensional Nozzle

The simulation of two-dimensional inviscid flow through a nozzle was conducted at first in order to verify the numerical scheme.

Model Nozzle

The geometry of the nozzle is determined by equations:

$$A(x) = 2.5 + 3\left(\frac{x}{x_{th}} - 1.5\right)\left(\frac{x}{x_{th}}\right)^2 \quad \text{for } x\ \le x_{th} \quad (7)$$

$$A(x) = 3.5 - \frac{x}{x_{th}}\left\{6 - 4.5\frac{x}{x_{th}} + \left(\frac{x}{x_{th}}\right)^2\right\} \quad \text{for } x\ \ge x_{th} \quad (8)$$

with area of throat : $A_{throat} = 1$
length of nozzle : $x_{max} = 10$
position of the throat : $x_{th} = 5$

Boundary Condition

At inlet boundary, total enthalpy and entropy which were calculated from total pressure 22.0[MPa] and total temperature 400[K] were fixed. At outlet boundary, the static pressure was fixed as

$$p_{outlet} = 0.83049 \times p_{inlet} \quad (9)$$

For these conditions, the density at inlet was equal to the critical density and the flow was subsonic in the convergent part, reached sonic conditions at the throat and expanded in the divergent part with supersonic mach numbers.

Result and Discussion

Figure.4 shows the distribution of the compression coefficient in the nozzle. Compression coefficient became smaller near the throat, namely the real gas effect became larger, and then a shock was present at x = 7 and compression coefficient changed discontinuously. Fig.5 shows the normalized density profiles in the nozzle. Near the throat where the real gas effect was remarkable, the density calculated by PR EoS was higher than by perfect gas EoS, and this

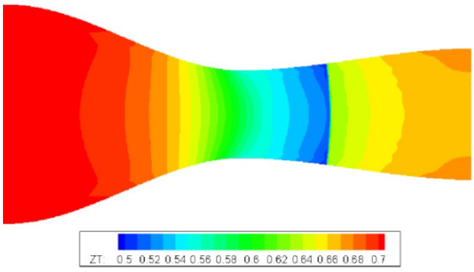


Fig.4 Distribution of Compression Coefficient in two-dimensional nozzle

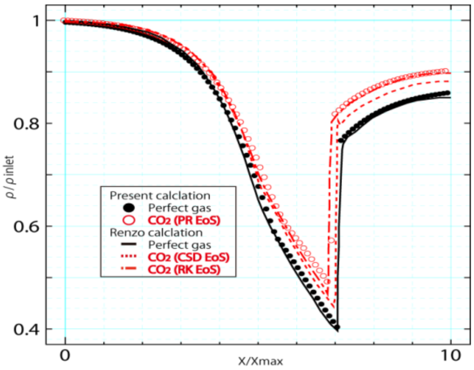


Fig.5 Density profiles for steady nozzle flow

result was very similar to that of Renzo's calculation[4]. So it is thought that the present scheme can simulate stably the flow of SC-CO_2, whose thermal properties change dramatically.

Flows in a Centrifugal Compressor

Model Compressor

As the test compressor, a small compressor which was designed for a verification test in order to gain about 10 [kw] output was adopted. Figure6 shows the test impeller and the vaneless diffuser. The design parameters of the impeller are summarized in Table. The test impeller has 13 blades and the constant tip clearance of 0.15 [mm]. The design pressure ratio is 1.46 and the design adiabatic efficiency is 0.729 at the mass flow rate of 1.2 [kg/sec], which were estimated by an exploratory analysis

Computational Grid

The computational grid is shown in Fig.7. The main flow region is divided into 6 domains and the H-type mesh is adopted except for the C-type grid around the impeller blade. The tip clearance region is divided into 2 regions and the H- and C- type mesh are adopted. The impeller region consists of 459,264 cells and the diffuser region consists of 227,136 cells respectively. The total number of cells is 686,400.

Boundary condition

At the inlet boundary, total enthalpy and entropy, which were calculated from total pressure and total temperature, were fixed. At the exit boundary, the static pressure of the diffuser exit was given equally from hub to tip. No slip and adiabatic conditions were imposed on the casing, hub wall and blade surfaces. As computational domain, one flow passage was adopted and periodic boundary conditions were imposed circumferentially.

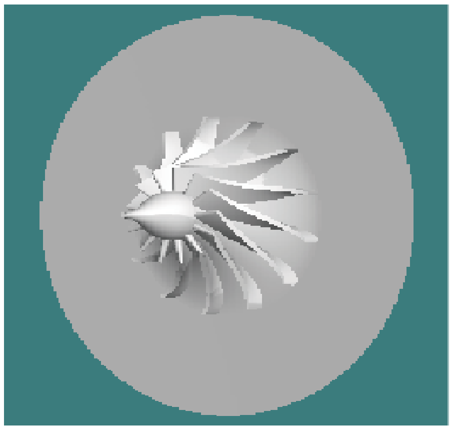
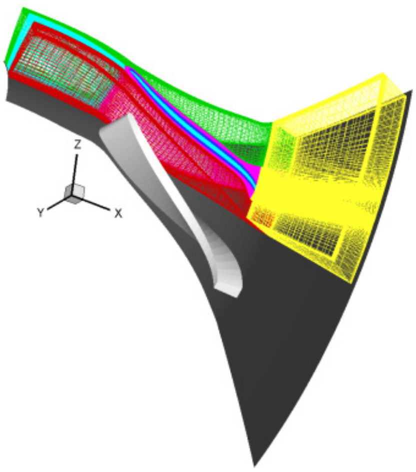


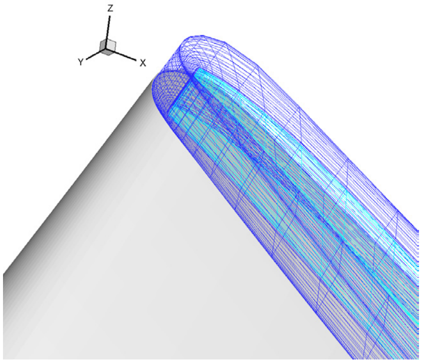
Fig. 6 Model Compressor

Table.2 Design Parameter of Model Compressor

Number of impeller blades	13
Mass flow rate [kg/sec]	1.2
Design wheel speed [rpm]	100000
Inlet hub diameter [mm]	5.23
Inlet tip diameter [mm]	10.46
Outlet impeller diameter [mm]	20
Outlet diffuser diameter [mm]	40
Tip clearance [mm]	0.15
Pressure ratio(out : static ,in : total)	1.46
Adiabatic efficiency(out : static ,in : total)	0.729



(a) Main flow region



(b) Close-up of blade tip near leading edge

Fig. 7 Computational Grid for model compressor

Result and Discussion

In this study, simulations were conducted under several inlet boundary conditions and rotation speeds. The characteristic flow field was observed for a convergent solution where total enthalpy and entropy, which were calculated from total pressure 8.23[MPa] and total temperature 308[K], were fixed at the inlet boundary. In this case, the rotation speed was 100,000[rpm] and the pressure ratio was 1.25 respectively. Figure8, Figure9 and Figure10 show the distribution of the entropy generation, density and the iso-surface of the compression coefficient respectively in a centrifugal compressor. The vortex induced by the tip leakage flow from upstream was observed and entropy generation which leads to pressure loss was noticeable in this region. In the same region, it is also observed that the density of the fluid changed lower. It was assumed that the fluid crossed over the pseudo-critical point locally and shifted to high compression coefficient about Z=0.46 because of the reduction of static pressure. The static pressure rise coefficient on the blade was calculated by eq. (10) and shown in Fig.11.

$$C_p = \frac{P_s - P_{t-in}}{P_{t-in}} \quad (10)$$

P_s : Static Pressure

P_{t-in} : Total Pressure at inlet boundary

The difference of pressures on the pressure and suction sides near the mid code of the tip region where the vortex developed was remarkable and this might have induced intense tip leakage flows.

Conclusion

The flow fields of a two-dimensional inviscid nozzle flow and a centrifugal compressor using $SC\text{-}CO_2$ as working fluid were simulated to figure out the flow field of the behavior of $SC\text{-}CO_2$. The conclusions are summarized as follows.

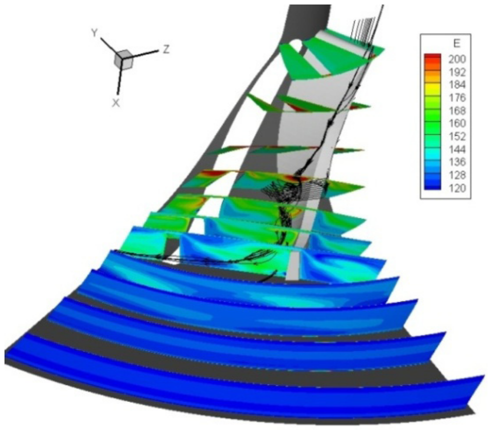


Fig. 8 Distribution of Entropy Generation

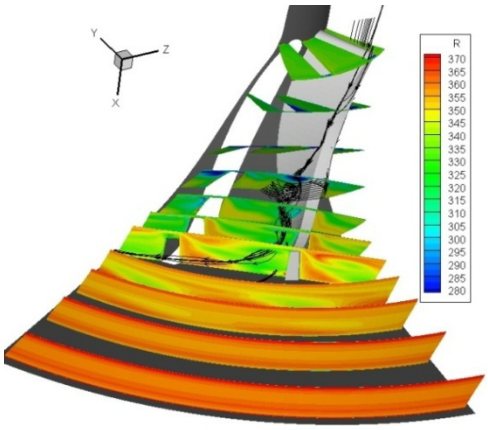


Fig. 9 Distribution of Density

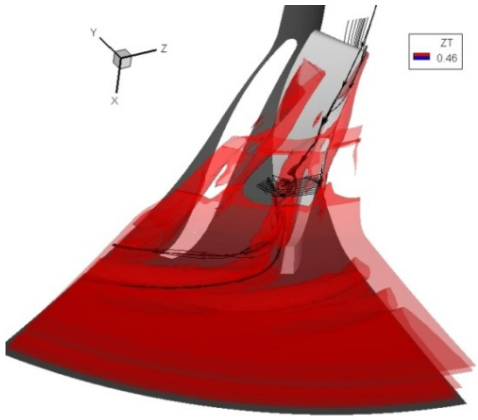


Fig. 10 Iso-surface of Compression Coefficient

1) This scheme can simulate flow fields stably wherever the compression coefficient change rapidly.
2) In a centrifugal compressor, entropy generation was noticeable near the vortex induced by the tip leakage flow and the density of the fluid became lower in the same region under the condition where the inlet boundary conditions were near the critical point.

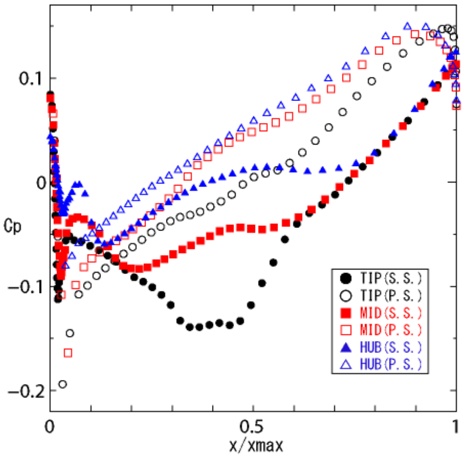


Fig. 11 Iso-surface of Compression Coefficient

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