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Effect of Inlet Conditions on Convective Heat Transfer in Submerged Axisymmetric s-CO_2 Impinging Jet for Next Generation Turbine

Seon Ho Kim
Ph.D Candidate
Yonsei University
SEOUL, Republic of Korea

Taehyun Kim
PhD student
Yonsei University
SEOUL, Republic of Korea

Namkyu Lee
Ph.D Candidate
Yonsei University
SEOUL, Republic of Korea

Minho Bang
Ph.D student
Yonsei University
SEOUL, Republic of Korea

Hyung Hee Cho
Professor
Yonsei University
SEOUL, Republic of Korea

ABSTRACT

The supercritical carbon dioxide, s-CO_2, power cycle has emerged as a promising system for clean and high-efficiency power production. Especially, directly heating system using oxy-fuel combustion technology called Allam cycle is preparing for demonstration as a next generation new concept development system. Since this system operates at a high temperature above 1100°C, like a gas turbine, the design for cooling system is very importance. In this study, the heat transfer characteristics of supercritical carbon dioxide impinging jet with various inlet conditions are investigated. The conditions of impinging jet are submerged, unconfined, fully developed round jet. The jet diameter, *d*, is 25 mm and height to diameter, *h/d*, is 2. Jet Reynolds number is set to 50,000. Simulation range of the inlet pressure which normalized by critical value are from 1.1 to 1.7. The normalized inlet temperature changes from 1.01 to 1.10. As a result, Nusselt number on the target surface varies greatly depending on the surface inlet conditions. When the pressure ratio is low, the distribution of Nusselt numbers is highly volatile with temperature change. Nusselt number on the stagnation point distributes from 250 to 500. However, the distribution of the Nusselt number is merged at the high pressure ratio.

INTRODUCTION

Supercritical power generation systems using direct heating combustion with oxy-fuel is regarded as a promising technology for next power generation. Allam et al [1] suggested the directly fired oxy-fuel supercritical CO_2 power system called Allam cycle. This system uses the CO_2 as a working fluid, and operate in a high pressure and high temperature above the 300 bar and 1100℃. The s-CO_2 turbine inlet temperature nearly reached the operating temperature of the gas turbine. Various cooling systems have been applied to prevent damage of high-temperature components and to improve the system efficiency. According to a report published by IEAGHG [2], Allam cycle also consider the cooling of hot components of s-CO_2 turbine blade. The amount of cooling fluid entering the hot parts of the turbine blade is more than 11% of the flow rate of mainstream. Scaccabarozzi et al. [3] have reported that The turbine inlet temperature is limited due to the amount of the cooling flow. They concluded that cooling systems is the important factors for system efficiency in optimization studies of directed fired oxy-fuel s-CO_2 power generation systems.

Studies on the heat transfer characteristics of s-CO_2 mainly have been focused on internal cooling system. The s-CO_2 turbine blades has been proposed in the form of a circular tube [4].

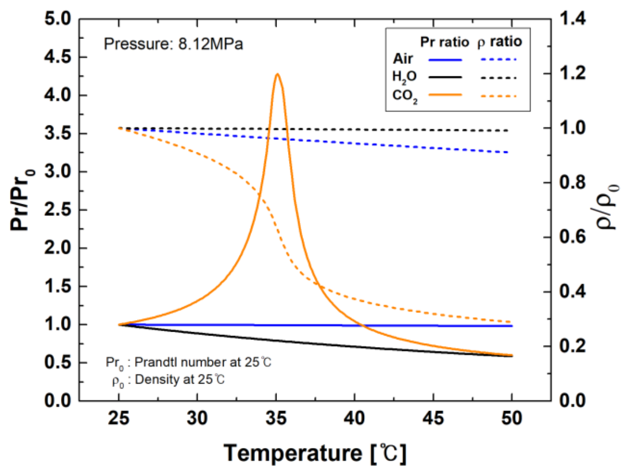

Figure 1 Comparing Prandtl number ratio (Pr/Pr_0) and density ratio (ρ/ρ_0) of various working fluid with temperature change

However, blade cooling using a cylindrical internal forced convection system has a limitation in reducing the surface temperature in response to a high external heat load [5]. Thus, advanced types of cooling system, such as impinging jets, should be considered. That can achieve high levels of heat transfer coefficients. It will be required in the directly fired oxy-fuel s-CO_2 turbine. Heat transfer characteristics of impinging jets have been studied for various parameters: working fluid, Reynolds number (Re), normalized nozzle-to-plate distance (H/D), flow confinement, nozzle geometry, profiles of inlet flow and array jet. Jambunathan et al [6], Viskanta et al [7], Zuckerman and Lior [8], Hong and Cho [9] reviewed the effects and mechanism of heat transfer characteristics with various parameters.

Most of the heat transfer research conducted using air and water. These working fluids have linear property changes with temperature and pressure. It is noteworthy that the change in properties of each working fluid in the single-phase region is not large. However, supercritical fluids are unique substance at a pressure and temperature above its critical point, where distinct gas and liquid phase do not exist. Supercritical fluids have abrupt changes in physical properties near pseudo critical points. Figure 1 shows variation of Prandtl number and density ratio with increasing the temperature at constant pressure. Non-linear characteristics of property for CO_2 appear near pseudo-critical temperature region. Among previous researches, it is that heat transfer enhancement (HTE) and heat transfer deterioration (HTD) are observed depending on the operating and surface conditions between fluid and wall in tube [10]. Therefore, unique heat transfer characteristics are expected to be observed in the impinging jet depending on the heat flux on the wall surface.

This study numerically investigated the heat transfer characteristics of supercritical carbon dioxide, s-CO_2, submerged impinging jet with various inlet conditions. Jet conditions are submerged unconfined fully developed round jet. Heat transfer characteristics of the impinging jet with s-CO_2 in low-pressure ratio is compared with air. Also, Nusselt number according to the change of the inlet conditions of the target surface is analyzed.

RESULTS AND DISCUSSION

Diameter of injection hole, *d*, is 25 mm, and the height of nozzle exit to target surface, *H/d*, is 2. The developing region set to 70*d* to obtain a fully developed velocity profile distribution, and target surface region is 6*d*. In order to simulate the unconfined jet, the analysis domain is extended by 2*d* in the nozzle inlet direction. Baughn and Shimizu [11] measured heat transfer coefficient by experiment in fully developed jet and wall controlled thermal boundary condition (i.e., a uniform heat flux) using air. In this study, setting of domain is based on geometry of Baughn and Shimizu experiments for validate the heat transfer coefficient and physical phenomena such as peak characteristics.

This simulation of impinging jet study is assumed to be axisymmetric two-dimensional compressible, steady state, and turbulent flow. The time-averaged continuous equation, the momentum equation and the energy equation are given by Eq. (1) - (3).

Mass conservation equations:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum conservation equations:

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_l}{\partial x_l}\right)\right] + \frac{\partial}{\partial x_j}\left(-\rho\overline{u_i' u_j'}\right) \tag{2}$$

Energy conservation equations:

$$\nabla \cdot \left(\vec{v}(\rho E + p)\right) = \nabla \cdot \left(k_{eff}\nabla T - \sum_j h_j \vec{J_j} + (\overline{\overline{\tau_{eff}}} \cdot \vec{v})\right) + S_h \tag{3}$$

There are mainly Reynolds number and Nusselt number for using the data analysis. In order to determine the Reynolds number and Nusselt number, bulk mean physical properties are used. The bulk mean physical properties defined as

$$\emptyset_b = \frac{\int \emptyset \rho \left|\vec{v} \cdot d\vec{A}\right|}{\int \rho \left|\vec{v} \cdot d\vec{A}\right|} \tag{4}$$

The bulk mean Reynolds number is defined according to the bulk mean physical properties based on the nozzle exit conditions (i.e., pressure and temperature at the nozzle exit).

$$Re_b = \frac{\rho_b V_d d}{\mu_b} \tag{5}$$

The local heat transfer coefficient is defined as

$$h = \frac{\dot{q_w}}{(T_w - T_b)} \tag{6}$$

The local Nusselt number is that

$$Nu = \frac{hd}{k_b} \tag{7}$$

This simulation solved the steady state Reynolds averaged Navier–Stokes equations (RANS) through the finite control volume method using the commercial package of ANSYS Fluent 17.2. Approximately 0.2 million elements of structured grid have generated in the domain using ICEM CFD 17.2.

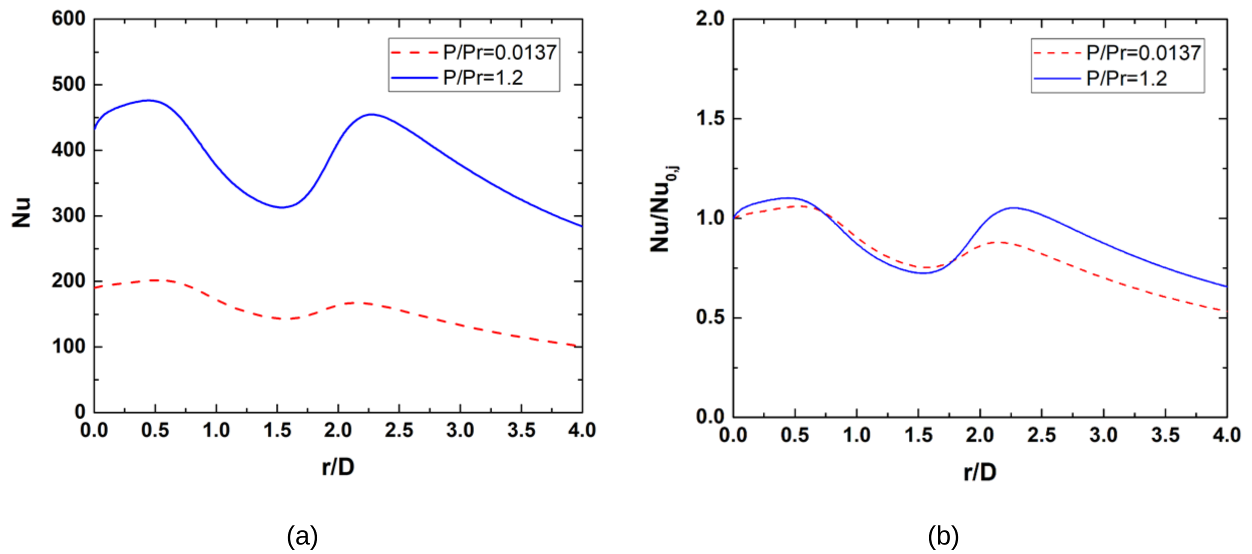

Figure 2 Comparing the heat transfer characteristics between low pressure ratio and high pressure ratio (a) Nusselt number (b) Nusselt ratio based on Nusselt number of stagnation point

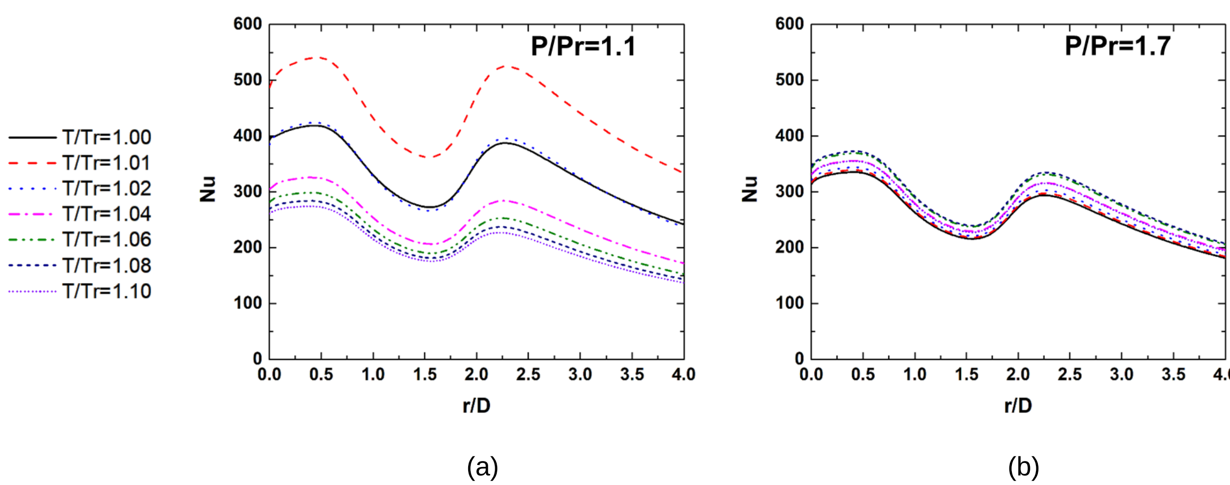

Figure 3 Nusselt number distribution with various inlet conditions (a) case of P/Pr = 1.1 (b) case of P/Pr = 1.7

Several turbulence models such as RNG k-ε, SST k-ϖ, Reynolds stress and v^2-f were applied and compared. In this study, the SST k-ϖ Intermittency transition model is adapted. Real gas models use thermodynamic and transport properties database (REFPROP v9.1 [12]) to evaluate thermodynamic and transport property. Mass flux inlet boundary condition is specified based on the Reynolds number of 50,000. Inlet fluid temperature is from 307.15 K – 334.55 K which is equivalent to temperature ratio (*T/Tr*) 1.01-1.10 based on the critical temperature of CO_2, 304.13K. Outlet conditions set as constant pressure 0.101325 MPa – 12.541 MPa. Target surface set as the no-slip wall boundary condition with constant heat flux. Heat flux range is 1,000 W/m^2. Since the analysis domain is axisymmetric about the central axis, 2D axisymmetric analysis is performed.

Figure 2 shows the heat transfer characteristics between low pressure ratio and high pressure ratio on the target surface. Local Nusselt number is divided by Nusselt number at the stagnation point of each fluid. Near the stagnation region (r/d<1.5) Nusselt number of CO_2 of low pressure ratio condition is similar with air. In the transition region where the flow characteristics change from laminar to turbulence. Therefore, there are two peaks. On the other hand, Nusselt number has a large variation with high pressure ratio. Overall shapes of local Nusselt number have a similar pattern of the low h/d jet. First peak and second peak also appear near r/d~0.5 and r/d~2.0. However, absolute value of local Nusselt number is lager about 2.2 times than low pressure ratio. The peak value is increase as shown in fig.2(b).

Figure 3 shows the effects of inlet conditions of s-CO2 on the heat transfer characteristics on the target surface. the case of constant pressure ratio P/Pr=1.1 is shown in fig.3(a). Local Nusselt number are changed by temperature ratio. Especially, Nusselt number reaches the maximum value at the T/Tr=1.01. In the case of T/Tr = 1.01, the difference of Nusselt number between the first peak and the second peak is under 3%. First peak appear on the r/d = 0.46 and second peak indicate around the r/d= 2.3. However, the value of Nusselt number tends to decrease with increasing T/Tr. When T / Tr = 1.02, Nusselt number is equal to that when T/Tr = 1.0, and then the value decreases sharply. Nusselt number on the stagnation point distributes from 250 to 500. In particular, the ratio of the first peak to the second peak is about 12%, and the position of peak approaches the stagnation point area at the T/Tr=1.10 case. Figure 3(b) shows Nusselt number distribution in the case of P/Pr=1.7. Unlike Fig. 3 (a), it can be seen that the change in Nusselt number is not sensitive to the temperature change. However, it is not a form having the maximum value at a specific temperature ratio, and Nusselt number tends to increase as the T/Tr increases. The heat transfer characteristics due to these pressure changes and temperature changes allude that the closer the inlet flow is to a pseudo-critical point, the greater the variability. When Prandtl number is compared based on the condition of the nozzle outlet, it has a maximum value at P / Pr = 1.1 and T / Tr = 1.01. The inlet Prandtl number determines a certain range of change in physical properties with the temperature at the wall surface due to the influx of heat. The increase of the Prandtl number means that the thickness of the thermal boundary layer becomes thinner than that of the flow boundary layer. This difference in fluid and thermal boundary layer scale may induce the increase or decrease of heat transfer. The results of this study can be used to design the supercritical turbine cooling system as the basis for the heat transfer behavior of s-CO_2.

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