



# Numerical investigation of jet impingement cooling of a low thermal conductivity plate by supercritical pressure carbon dioxide

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## ABSTRACT

Jet impingement cooling is widely used due to its high heat transfer rates, especially within the stagnation region. Jet impingement cooling by supercritical pressure fluids has even higher heat transfer coefficients with the proper working conditions than regular fluids with no burnout. This paper presents a numerical investigation of supercritical pressure carbon dioxide jet impingement cooling of a low thermal conductivity plate. Predictions of 11 RANS turbulence models were first validated against the published experimental data done by our research group. The effects of the inlet temperatures and pressures on the heat transfer were studied numerically for a wide range of heat fluxes with the flow phenomena observed in the experiment further explained by the numerical results. The SST  $k-\omega$  and Transition SST models gave more accurate predictions of the average heat transfer coefficient with differences of less than 15% for the validated case. The maximum average heat transfer coefficient occurred when the inlet temperature was lower than and the surface temperature was slightly higher than the pseudocritical temperature. The maximum average heat transfer coefficient increased with increasing inlet temperature when the inlet temperature was lower than the pseudocritical temperature and decreased with increasing pressure.

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## 1. Introduction

Jet impingement cooling is widely used in industry due to its high heat transfer coefficients, such as for cooling electronic devices and gas turbine blades, paper and textile drying, and for annealing metals. Many experimental and numerical studies [1–7] have been conducted to study the flow and heat transfer mechanism of jet impingement cooling to further enhance the heat transfer rates. Most existing jet impingement cooling studies were conducted at pressures much lower than the thermodynamic critical pressure and the variations of the fluid thermophysical properties were neglected. When the pressure is near the critical pressure, the fluid thermophysical properties vary greatly even for very small temperature changes as shown in Fig. 1 [8]. The large specific heat near the pseudocritical temperature, which corresponds to the maximum specific heat at pressures higher than the critical pressure, greatly enhances the heat transfer rate. This has been verified by many studies on the heat transfer for flows in tubes and porous media at supercritical pressures [9–16]. In addition, at subcritical pressures, the surface can easily dry out

when the heat flux is higher than the critical heat flux (CHF), which results in a dramatic rise of the surface temperature or even burnout. The film boiling can be avoided by keeping the coolant at supercritical pressures. However, there are only a few papers dealing with jet impingement cooling at supercritical pressures to the best of the authors' knowledge.

Rothenfluh et al. [17] experimentally studied the average heat transfer rates for impinging, turbulent, near-critical hot water jets confined by an annular wall for the development and design of a hydrothermal spallation drill bit. Rothenfluh et al. found that the average heat transfer coefficients depended on the plate surface temperature, which was attributed to the large changes in the thermophysical properties near the plate. However, Rothenfluh et al. did not describe the details of the flow field and the local heat transfer characteristics for the jet impingement cooling. Radial thermal conduction in the target wall was also neglected in their experiments. Thus, the detailed local heat transfer characteristics need to be studied further. Chen et al. [18] experimentally investigated jet impingement cooling with carbon dioxide at supercritical pressures. Chen et al. [18] studied the effects of heat flux, jet inlet temperature and mass flow rate on the local heat transfer coefficients with a thermal sensor integrated into the cooled plate developed using MEMS (Micro-Electro-Mechanical Systems) techniques

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## Nomenclature

|                  |                                                            |
|------------------|------------------------------------------------------------|
| $D$              | nozzle diameter, m                                         |
| $E$              | total energy, J/kg                                         |
| $h$              | heat transfer coefficient, W/m <sup>2</sup> K              |
| $k$              | turbulent kinetic energy, m <sup>2</sup> /s <sup>2</sup>   |
| $k_{\text{eff}}$ | effective thermal conductivity, W/m                        |
| $L$              | spacing between the nozzle exit and the cooling surface, m |
| $m$              | mass flow rate, kg/s                                       |
| $p$              | pressure, Pa                                               |
| $q$              | heat flux, W/m <sup>2</sup>                                |
| $r$              | radial coordinate, m                                       |
| $t$              | time, s                                                    |
| $T$              | temperature, K                                             |
| $S_h$            | source term, W/m <sup>3</sup>                              |
| $u$              | time-averaged velocity, m/s                                |
| $u'$             | fluctuating velocity component, m/s                        |
| $v$              | time-averaged velocity, m/s                                |
| $x$              | coordinate axis, m                                         |
| $z$              | axial coordinate, m                                        |

## Greek symbols

|                            |                                        |
|----------------------------|----------------------------------------|
| $\delta_{ij}$              | Dirac delta function                   |
| $\lambda$                  | thermal conductivity, W/m K            |
| $\mu$                      | viscosity, Pa s                        |
| $\mu_t$                    | turbulent viscosity, Pa s              |
| $\nu$                      | kinematic viscosity, m <sup>2</sup> /s |
| $\rho$                     | density, kg/m <sup>3</sup>             |
| $(\tau_{ij})_{\text{eff}}$ | Reynolds stress tensor, Pa             |

## Subscripts

|     |                         |
|-----|-------------------------|
| ave | average                 |
| aw  | adiabatic wall          |
| c   | critical                |
| exp | experiment              |
| in  | inlet                   |
| pc  | pseudocritical critical |
| s   | surface                 |
| sim | simulation              |
| w   | wall                    |

for local temperature and heat flux measurements. Chen et al. [18] found that when the inlet temperature was lower than the pseudocritical temperature, the heat transfer coefficient increased with increasing heat flux and then decreased when the surface temperature became higher than the pseudocritical temperature with further increases in the heat flux. Besides these experimental studies, there have also been a few numerical studies. Kim and Toshio [19] numerically investigated the axisymmetric laminar jet impingement cooling of an isothermal flat surface with supercritical pressure carbon dioxide. Kim and Toshio found that the heat transfer coefficient was higher when the inlet temperature was close to the pseudocritical temperature corresponding to the inlet pressure. Chen et al. [20] numerically studied the heat transfer characteristics of confined round jet impingement cooling of a flat silicon plate at constant heat flux with supercritical pressure carbon dioxide. Their numerical results showed that radial conduction in the silicon plate could not be neglected, especially for high surface heat fluxes with the radial conduction leading to a much more uniform surface temperature distribution. Chen et al. [20] also found that the high specific heat of the fluid near the surface for conditions near the pseudocritical point maximized the average heat transfer coefficient for a given heat flux. However, none of these studies have validated the various turbulent models used to predict the heat transfer for jet impingement cooling at supercritical pressures

through comparisons of the predicted local heat transfer coefficients and measured local heat transfer coefficient data.

This paper describes a numerical investigation of jet impingement cooling with supercritical pressure carbon dioxide of a low-conductivity plate. The predictions by various turbulent models are compared with experiment data for the local heat transfer coefficient [18]. The results given by the turbulence model that best predicted the data was then used to explain the experimental visualization results and to study the effects of the inlet temperature and pressure on the heat transfer rate for various heat fluxes.

## 2. Experiment setup

The experimental investigation studied the flow and heat transfer characteristics of jet impingement cooling with supercritical pressure carbon dioxide [18]. The heater elements were mounted on the backside of a borosilicate glass substrate with a low thermal conductivity of 1.2 W/m K to reduce the radial thermal conduction in the plate that has been reported in the results of Chen et al. [20]. Temperature sensors were then sputtered onto the top side of the glass chip. The chip was 550  $\mu\text{m}$  thick with a uniform heat flux boundary condition applied to the back surface. Supercritical pressure carbon dioxide from the nozzle impinged vertically onto the top surface in the high-pressure vessel shown in Fig. 2 that was full of supercritical pressure carbon dioxide. The flow was visualized through the sapphire windows in the vessel walls. The nozzle diameter was 2 mm and the spacing between the nozzle exit and the cooling surface was 14.6 mm. More experiment setup details can be found in Chen et al. [18].

## 3. Turbulence models

### 3.1. Physical models and thermophysical properties

The flow field in the jet impingement flow with circular nozzle was assumed to be axisymmetric with a 2-D axisymmetric model then used as shown in Fig. 3. The nozzle diameter,  $D$ , was 2 mm and the spacing between the nozzle exit and the cooled surface,  $L$ , was 14.6 mm, which resulted in  $L/D = 7.3$ . The borosilicate glass radius,  $OF$ , was 15 mm while the heating area radius,  $OG$ , was 11.5 mm. The nozzle length,  $BC$ , was 20 mm as in the experiment.

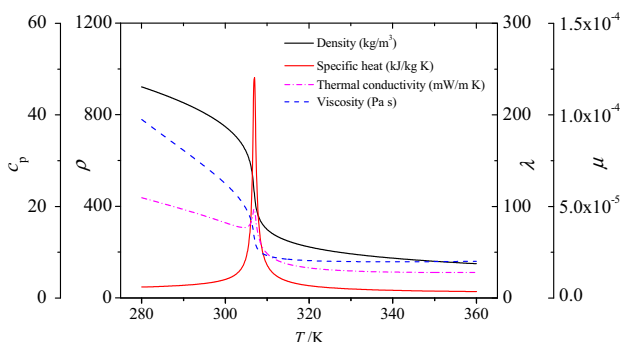


Fig. 1. Carbon dioxide thermal-physical property variations with temperature at 7.85 MPa ( $T_{pc} = 306.65$  K,  $p_c = 7.38$  MPa,  $T_c = 304.13$  K).

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