



# Experimental investigation on heat transfer and pressure drop of supercritical water flows in an inclined rifled tube



Alireza Taklifi, Pedram Hanafizadeh\*, Mohammad Ali Akhavan Behabadi, Abbas Aliabadi

Center of Excellence in Design and Optimization of Energy Systems (CEDOES), School of Mechanical Engineering, College of Engineering, University of Tehran, PO Box: 11155-4563, Tehran, Iran

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## ABSTRACT

This paper presents an investigation of heat transfer behavior and pressure drop characteristics of subcritical and supercritical water flows inside an inclined rifled tube for near-supercritical and supercritical flows. The operating pressure range was from 15 to 28 MPa, the mass flux range was from 600 to 1000 kg/m<sup>2</sup> s and wall heat flux was from 300 to 500 kW/m<sup>2</sup>. The maximum inner diameter of pipe was 19.5 mm and the inclination angle of the tube was 20° with respect to the horizontal plane. The heat transfer to water and pressure drop gradients at various operating pressures for various heat fluxes and mass fluxes were studied. The results show that the effect of mass flux variations on heat transfer at supercritical pressures is more considerable than this effect at subcritical pressures. The same behavior was found for the pressure drop gradient. It was found that as the ratio of the mass flux to the heat flux exceeded 2.46 kg/kW s, heat transfer enhancement occurred near the pseudo critical point for supercritical pressures while maximum heat transfer at subcritical pressures occurred at ratio of the mass flux to the heat flux value of 1.65 kg/kW s for inclined rifled tube.

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## 1. Introduction

Nowadays the efficiency of energy systems is of high importance and it drew the attentions of many OEMs (original equipment manufacturers) in power and energy sector all around the world. Power plants as the main consumers of primary energy resources are at the leading edge of efficiency boost. Within various kinds of electricity generation plants, thermal power plants play a great role in providing the needed demand of power in the world. As a matter of fact ultra super-critical and supercritical thermal power plants are the most efficient kinds of this type [1]. Supercritical flows have many crucial applications in various fields like power engineering for boilers, chemical industries, water treatment (SCWO: super-critical water oxidation) and etc. [2].

Supercritical fluid flows have been investigated widely by many researchers [3–5]. Bringer and Smith [6] performed a set of experiments with a supercritical carbon dioxide flowing in a round tube. They found that the heat transfer is being enhanced at critical region. Miropolskii and Shitsman [7] studied the forced convection heat transfer of supercritical water flow in a round tube and they recommended a modified empirical correlation to predict heat

transfer during turbulent forced convection flows. Hendricks et al. [8] conducted a set of experiments with supercritical hydrogen as the working fluid and offered a mathematical model, based on the idea of pseudo-boiling.

The other investigations of Swenson et al. [9], Ackerman [10], Yamagata et al. [11], Kirillov [12], Pioro and Duffey [13,14] and many others showed that, near the pseudo-critical temperature the heat transfer enhancement might occur at some circumstances.

Modeling the forced convection boiling flows both analytically and numerically has been the topic of many researches. Deissler [15] solved simultaneously the equations of shear stress and heat transfer for a pipe flow by means of a simple analytical method to study the heat transfer and pressure drop characteristics of supercritical fluid flows. He et al. [16] developed an in-house CFD code to simulate the heat transfer to fluids at a supercritical pressure based on the Favre averaging approach. They found that the turbulence models which were previously found closely reproducing mixed convection under conditions of constant properties do not perform well for high pressure supercritical flows. Wen et al. [17] performed a numerical investigation of heat transfer deterioration (HTD) in supercritical water flowing through the vertical tube by using six low-Reynolds number turbulence models and they found that the V2F and SST models perform better than other models in predicting the onset of deterioration.

\* Corresponding author. Tel.: +98 2182084857.

E-mail address: [hanafizadeh@ut.ac.ir](mailto:hanafizadeh@ut.ac.ir) (P. Hanafizadeh).

### Nomenclature

|                 |   |
|-----------------|---|
| $c_p$           | specific heat at constant pressure (J/(kg °C))            |
| $d_{in}$        | hydraulic diameter (m)                                    |
| $d_{out}$       | external diameter (m)                                     |
| $E$             | heating voltage (V)                                       |
| $G$             | mass flux (kg/(m <sup>2</sup> s))                         |
| $h$             | heat transfer coefficient (W/m <sup>2</sup> °C)           |
| $H_f$           | specific enthalpy of bulk fluid (J/kg)                    |
| $\Delta H$      | added enthalpy (J/s)                                      |
| $I$             | heating current (A)                                       |
| $k_w$           | thermal conductivity of rifled tube (W/m <sup>2</sup> °C) |
| $L$             | length of test section (m)                                |
| $Nu$            | Nusselt number  |
| $P$             | pressure (MPa)  |
| $dp$            | pressure drop (kPa)                                       |
| $\Delta P_{lo}$ | frictional pressure drop in single-phase (MPa)            |
| $\Delta P_{tp}$ | frictional pressure drop in two-phase zone (MPa)          |
| $P_{cr}$        | critical pressure (MPa)                                   |
| $q$             | inner wall heat flux (W/m <sup>2</sup> )                  |
| $Q_E$           | heating power (W)   |
| $T_f$           | bulk fluid temperature (°C)                               |
| $T_{in}$        | inlet fluid temperature (°C)                              |
| $T_{out}$       | outlet fluid temperature (°C)                             |
| $t_{wi}$        | inner tube wall temperature (°C)                          |
| $t_{wo}$        | outer tube wall temperature (°C)                          |
| $x$             | vapor quality (–)   |
| DNB             | departure from nucleate boiling                           |

### Greek symbols

|                  |   |
|------------------|---|
| $\eta$           | thermal efficiency of test section            |
| $\kappa$         | thermal conductivity of fluid (W/m K)         |
| $\mu$            | dynamic viscosity of fluid (Pa s)             |
| $\nu$            | specific volume of fluid (m <sup>3</sup> /kg) |
| $\varphi_{lo}^2$ | two-phase frictional multiplier (kPa/kPa)     |

In order to enhance heat transfer to subcritical water flows there have been many active and passive methods such as using flow spoilers, twisted tapes, internal ribs, helical inserts and etc. which have been introduced by various researchers [18–20]. For subcritical water flows different kinds of internally ribbed tubes were used in boilers to delay the DNB (departure from nucleate boiling). This technique was also used for supercritical water heat transfer enhancement [21–24]. Wang et al. [25] showed that, at low heat fluxes, the heat transfer enhancement might occur as the pseudo-critical temperature is approached in the large specific heat region. This region is defined as a region where the specific heat of water at constant pressure is greater than 8.4 kJ/kg K. Fig. 1 shows the variation of thermophysical properties of water at a pressure of 26 MPa.

Mosyak and Hetsroni [26] used infrared thermography technique to study the thermal and hydrodynamic phenomena in intermittent two-phase air–water flow in horizontal and inclined tubes at atmospheric pressure. Ghajar et al. [27] investigated the non-boiling heat transfer of air–water two-phase flow in a slightly inclined tube. For inclined tube flows there are small amount of researches being done for subcritical and supercritical water forced convection boiling, especially when it comes to internally ribbed tubes there would be no references. These kinds of tubes become more interesting when low mass flux conditions are considered. Yin et al. [28] studied the heat transfer to supercritical water in inclined upward smooth tubes. They found that as the pressure increases far from the critical pressure, the amount of deterioration decreases and also they introduced correlations of heat transfer coefficients

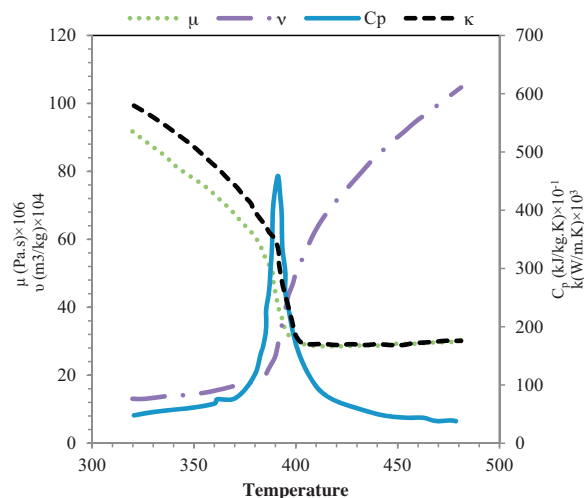


Fig. 1. Thermophysical properties of water at a pressure of 26 MPa.

of the forced-convection heat transfer on the top and bottom of the tube which could be used to predict heat transfer coefficient of spirally water wall in supercritical boilers.

There is no doubt that frictional pressure drop in rifled tubes is much higher than that in smooth tubes. Ackerman [10] reported the frictional pressure drop in adiabatic rifled tube 25% higher than the smooth tube. In this regard, many works have been done by Petukhov and Popov [29], Kolher and Kastner [30], Zdaniuk et al. [31], Chen et al. [32] and they have presented many analytical formulas and empirical correlations for single-phase and two-phase frictional coefficient for both smooth and rifled tubes.

In the current paper, the heat transfer and pressure drop characteristics of forced convection boiling at subcritical and supercritical conditions for an inclined rifled tube are investigated experimentally. The experiments are being performed in a high pressure and high temperature test facility capable of reaching pressures up to 40 MPa. The results describe the heat transfer mechanism and pressure drop gradients in the 2 m long test section for both subcritical and supercritical water flows. As the results express these behaviors for a rifled inclined tube, the possible enhancements due to this inclination angle and internal ribs are discussed among previous works of other researchers.

## 2. Experimental setup

The experiments on heat transfer characteristics and hydrodynamic characteristics of the rifled tube were conducted on a High Pressure Steam–Water Two-phase Flow and Heat Transfer Test Loop (Fig. 2). This test loop can be used to undertake an experimental research project on two-phase flows and heat transfer in large steam boilers, nuclear reactor, steam generators and heat exchangers used in various industrial departments, at various pressures, with different circulating modes and fluid mediums. The capacities of the main parameters for the experimental system are listed in Table 1.

This test loop consists of a deionized water tank, a high-pressure piston pump, a filter, a flowmeter, a heat exchanger, a preheater, an inclined test section, a condenser, a rotameter and some valves.

The heat exchanger is designed for heat recovery and the rotameter is used to visually monitor and measure the flow rate in experimental system, in addition to an accurate mass flowmeter is used for mass flux measurement. The preheater and test section are directly heated by AC power supplies with high current and low voltage. The maximum heating power of the test loop approaches

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