Experimental Investigation on Heat Transfer of Supercritical Pressure Water in Annular Flow Geometry

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Abstract

A supercritical water heat transfer test section has been built at Xi'an Jiaotong University to study heat transfer in annular flow channel. Based on the experimental results, the effects of mass flux and heat flux on heat transfer of supercritical pressure water in vertical annular channel were analyzed. The characteristics and mechanisms of heat transfer enhancement, and that of heat transfer deterioration, were also discussed. Based on a comparison with two identical flow geometries with and without helical wrapped spacer, it was found that the spacer has a positive effect in enhancing the local heat transfer, especially in the pseudo-critical region.

Key words: Supercritical water; Annular flow; Heat transfer; Spacer

1. Introduction

The supercritical pressure water-cooled reactor (SCWR) is one of the six reactor technologies selected for research and development under the Generation Ⅳ international Forum in 2002. It has the potential advantage of minimization of nuclear waste and low capital cost due to its high thermal efficiency and simplifications of the plant system. A SCWR power plant may achieve high thermal efficiency (about 45% vs. about 35% efficiency for advanced LWR). It is operated above the critical pressure of water, where the reactor coolant experiences no phase change (the coolant flowing into the core changes continuously from a lower temperature, high density compressed liquid to high temperature, low density compressed liquid, without any discontinuous phase change in the core). Because of this, the SCWR plant system can be kept simply as the need for many of the traditional LWR components such as the coolant recirculation pumps, pressurizer, steam generator, and steam separator and dryer is eliminated. One of the main features of supercritical water is the strong variation of its thermal-physical properties in the vicinity of the pseudo-critical line (Figure 1). Although operation above the critical pressure eliminates coolants boiling, and the coolants remains single-phase throughout the system, the large variation of thermal-physical properties may result in unusual heat transfer which demands further investigations.

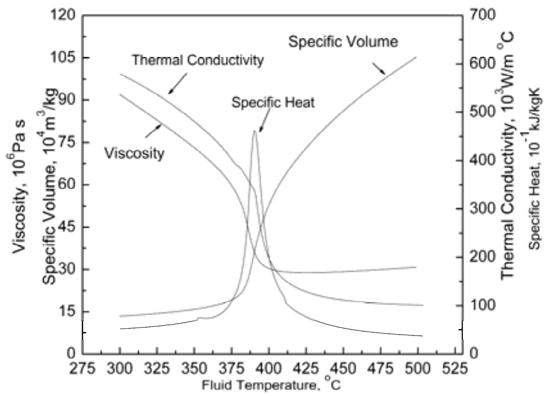
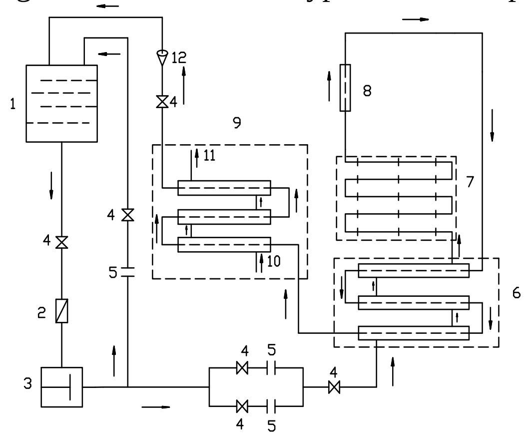


Figure 1: Thermo-physical properties variations of water at 25MPa

There are various design proposals of an SCWR core, the University of Tokyo has studied such reactors in detail since about 1990. Some results have been summarized, e.g. by Oka and Koshizuka (2000). In Europe, the common program High Performance Light Water Reactor (HPLWR), first announced by Heusener et al (2000), has been launched with the main objective to assess the technical and economic feasibility of a high efficiency LWR operating at supercritical pressure. One of the emphases for SCWR is the thermal-hydraulics behavior, especially the heat transfer characteristics of water at supercritical conditions, which differ strongly from that at sub-critical conditions, due to a rapid variation of the thermal-physical properties in the vicinity of the pseudo-critical line. Heat transfer deterioration would occur at high heat fluxes, and low mass fluxes, which leads to a strong reduction in the heat transfer coefficient. Therefore, a profound knowledge of heat transfer characteristics at reactor relevant conditions is necessary for the design work of SCWR. In the 1950s, using supercritical steam-water became an attractive idea for steam generators to increase the thermal efficiency of fossil-fired power plants. Intensive work was done on this subject from 1950's till now. Swenson et al (1965) found that heat transfer coefficients (HTC) has a peak when the film temperature is within the pseudo-critical temperature range, and this peak in HTC decreases with increase in pressure and heat flux. Shiralkar and Geiffith determined the limits for safe operation in terms of maximum heat flux for a particular mass flux with the coolant of supercritical carbon dioxide, they found that the deteriorated heat transfer occurred at a high heat flux relative to the mass flux. Yamagata et al. (1972) found that the HTC increases significantly in the pseudo-critical region for water in vertical and horizontal tubes. He indicated that the heat transfer deterioration was decided by heat flux and mass flux, and gave the limit value of heat flux: $q_c = 0.2G^{1.2}$ for vertical upward tube with an inner diameter of 10mm. Yashida and Mori(2000) stated that the enhancement and deterioration phenomena of heat transfer in the great specific heat region is caused by drastic variations of physical properties with temperature change across the coolant. However, the flow geometries used in previous experiments were mainly restricted to circular tubes. There are few publications devoted to heat transfer in bundle and annuli cooled with water at supercritical pressure. This paper presents the experimental results of the heat transfer characteristics of water in vertical annulus. Experiments were performed within the range of pressure from 23 to 28MPa, mass flux from 700 to 2500 kg/(m^2.s), and heat flux from 200 to 1000kW/m^2. Based on the experimental results, the effects of mass flux, heat flux and pressure on heat transfer of supercritical pressure water in vertical annular channel were analyzed. The characteristics and mechanisms of heat transfer enhancement, and that of heat transfer deterioration, were also discussed.

2. Experimental facility

The experiments were carried out in the High Pressure Steam-water Test Loop in Xi'an Jiaotong University. The schematic diagram of the test loop is shown in Figure 2. Distilled and de-ionized feed water from the water tank is driven through a filter by a high pressure plunger-type pump which is cable of operating at up to 40MPa. The feed water is pre-heated in a heat exchanger and a main pre-heater before flowing into the test section. The pre-heater and the test section are electrically heated by alternating current power supply with maximum heating capacities of 1.0MW and 0.5 MW, respectively. Therefore, we can adjust the test section inlet bulk temperature and heat flux simply by controlling the alternating current power supply. The heat of feed water flowing from the test section was removed by a regenerative heat exchanger and a condenser, and then flowed back to the water tank. The pressure and the mass flux in test section are controlled by adjusting the main valve and bypass valve, respectively.



1: Water tank; 2: Filter; 3: Water pump; 4: Valve; 5: Orifice;
6: Heat exchanger; 7: Preheater; 8: Test section; 9: Condenser;
10: Cooling water inlet; 11: Cooling water outlet; 12: Rotor flow meter

Figure 2: Schematic diagram of the test loop

Two kinds of annular test section are adopted in this experiment: one is a Φ8×1.5mm stainless steel (304) circular pipe within a Φ20×2.0mm circular pipe, with the gap of 4.0mm and the hydraulic diameter of 8mm; another is the same Φ8×1.5mm stainless steel (304) circular pipe within a Φ25×2.5mm circular pipe, with the gap of 6.0mm and the hydraulic diameter of 12mm, respectively. This design permits the use of 6 thermocouples spaced along the inner cladding of the heated rod. The geometry of test section and the thermocouple arrangement are shown in Figure 3 and Figure 4. The electrically-heated test section is 1400mm long from inlet to outlet. The inner pipe and outer piper are thermally insulated to minimize the heat loss. As illustrated in Figure 3, the sealing structure is used on both ends of the test section to guarantee the electric insulation between the inner and outer pipes as well as the test section's hermetic capability under the pressure of 28MPa. Double Sealing structure is used of the test section: flange 2 and outer pipe are wielded together and flat seal structure is adopted between flange 1 and flange 2 with gasket seal. There is a stuffing box between flange 1 and inner pipe and packing seal could be achieved through the hold-down bolt's squeeze to stuffing, graphite. The inner pipe of test section is heated through directly electricity (Low Voltage and High Current) and can achieve the heating condition of variable heat flux. The experiment is carried out via the heavy-current transformer connected to the copper heat sink which is well-matched to both ends of the inner pipe. When the current passes the inner pipe, it produces Joule Heat and heats the inner pipe as well as the water inside it.

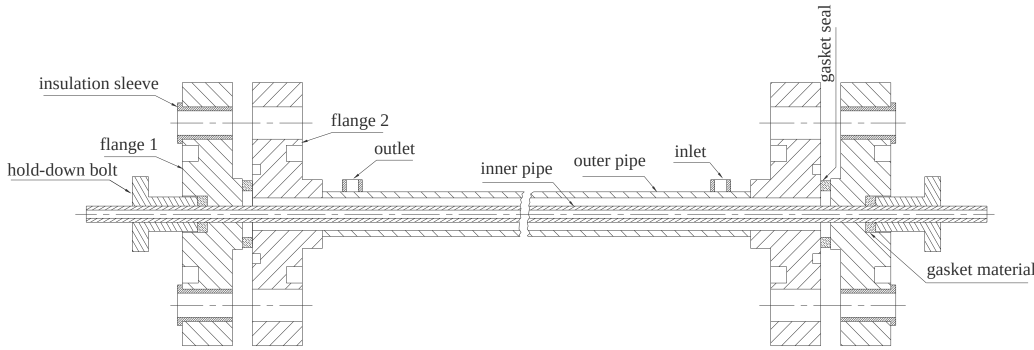


Figure 3: Annular test section geometry

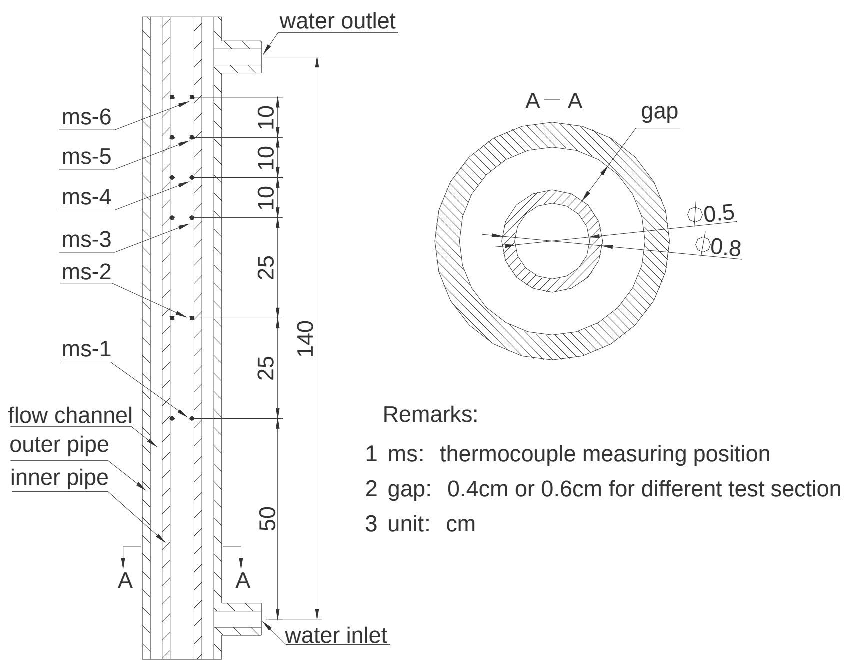


Figure 4: Thermocouple arrangement

The experiment was carried out as follows: adjusted the mass flux, system pressure and the heat flux of the test section to given values, while increasing the heating power of the pre-heater step by step, so the bulk enthalpy of the test section increased correspondingly. The test was finished once the wall temperature was over 700℃ due to heat transfer deterioration or the heating power reached the maximum. Then adjusted the mass flux, system pressure and the heat flux of the test section to another given values and began the next test. All data are collected and recorded by a data acquisition system.

The experiment was performed at steady-state experimental conditions close to operating conditions of an SCWR, within the range of pressure from 23 to 28MPa, mass flux from 350 to 1000 kg/(m^2.s), and heat flux from 200 to 1000kW/m^2.

3. Result and discussion

3.1 Effects of heat flux

Figure 5 show typical results of wall temperature and heat transfer coefficients plotted against bulk enthalpy, respectively, at a pressure of 25MPa, a mass flux of 1000 kg/m^{2}s, and various heat fluxes of 200, 400, 600, 1000 kW/m^2 for the vertical annular flow. Experiment result shows that in low enthalpy and high enthalpy regions which are far away from great specific heat region, heat transfer coefficient is independent of heat flux. However, in the pseudo-critical region, the heat transfer coefficient depends strongly upon the heat flux, and its maximum value increase with decrease of heat flux. It can also be seen from Figure 5 that when the heat flux is relatively low, the heat transfer is enhanced significantly, and no deterioration in heat transfer was seen at high mass flux (1000 kg/m^{2}s) even the heat flux reaches 1000 kW/m^2.

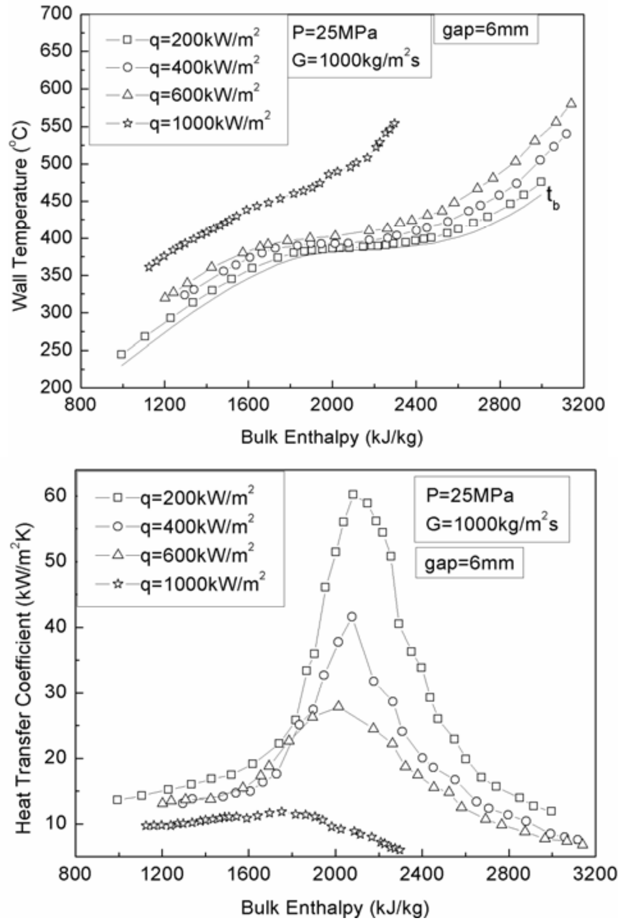


Figure 5: Comparison of wall temperatures and heat transfer coefficients at different heat fluxes.

3.2 Effect of annular flow geometry

In order to better evaluate the influence of gap size on heat transfer characteristic, two annular test sections have been designed and they have the identical geometry structure except the gap size. Figure 6 shows the examples of heat transfer coefficient plotted against bulk enthalpy with the annular gap of 4mm and 6mm, respectively. Experimental results show that at low heat flux relative to the flow rate (q=200 kW/m^2, G=1000 kg/m^{2}s) and normal heat transfer conditions, the two gap size produce different heat transfer coefficients. It can be seen from Figure 6 that the heat transfer coefficient has an obvious enhancement as the gap size increase from 4mm to 6mm, and the difference of heat transfer coefficient between the two test sections has a maximum at a bulk temperature near the pseudo-critical temperature. On the contrary, at a high heat flux relative to the flow rate (q=600 kW/m^2, G=400 kg/m^{2}s) and conditions of deterioration, the heat transfer coefficient for the gap size of 4mm almost has the same value as that for the gap size of 6mm. This experimental result indicates that the effect of gap size on heat transfer characteristic depend strongly on the flow conditions. That is, at similar operating conditions, normal heat transfer is very susceptible to the changes in gap size, whereas conditions of deterioration were found to differ with that.

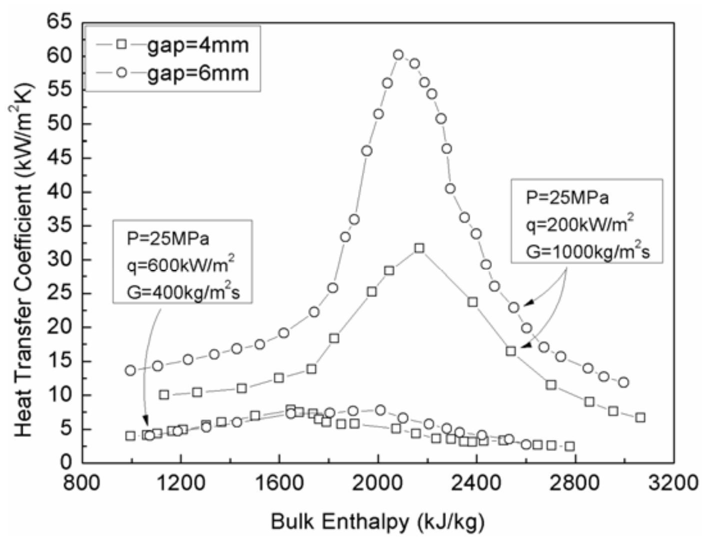
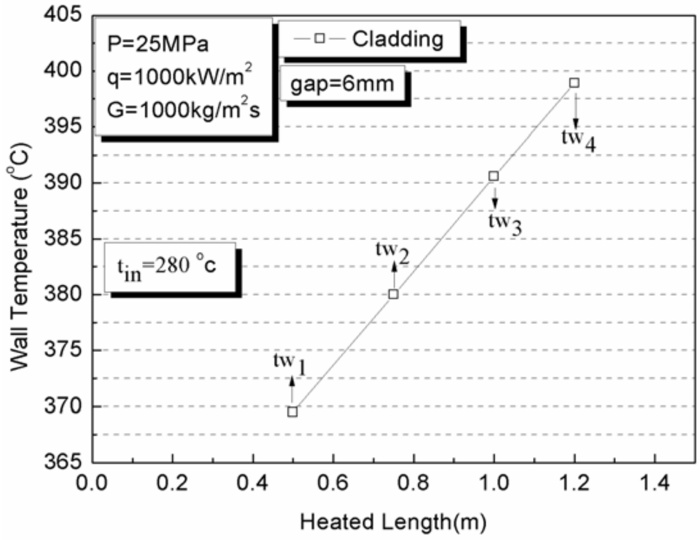


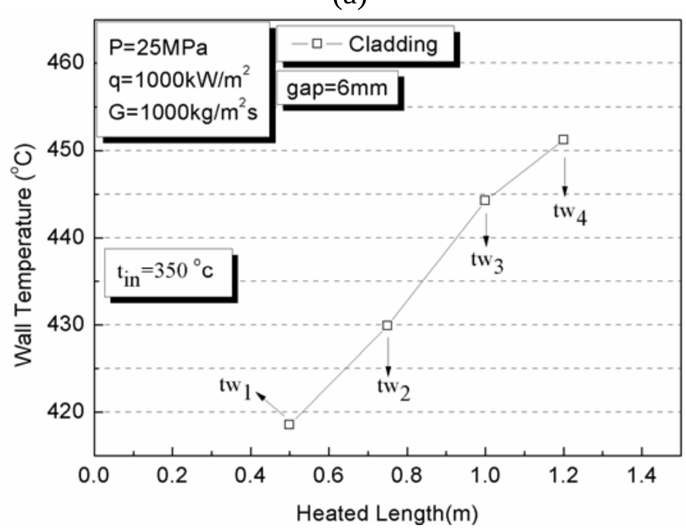
Figure 6: Influence of gap size on heat transfer coefficient

3.3 Deterioration

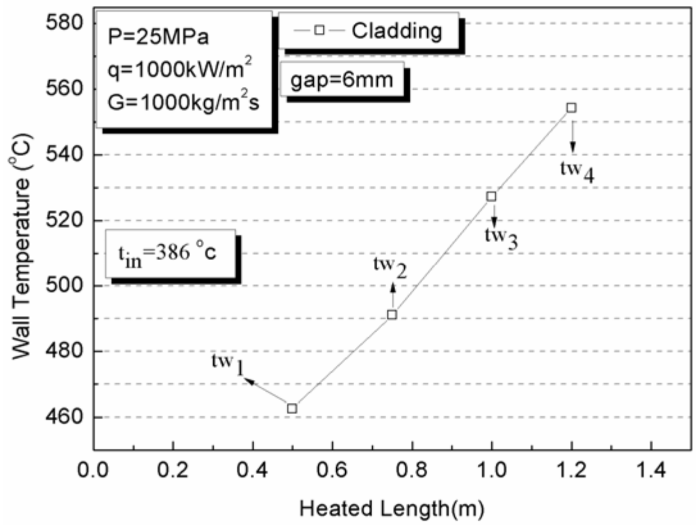
Figure 7 (a)-(c) show axial profile of the cladding temperatures for a heat flux of 1000kW/m^2 and an average mass flux of 1000kg/(m^{2}s) at different preheat temperatures of 280℃, 350℃ and 386℃, simulating the inlet, a mid section and a top section of a core. No deterioration in heat transfer was seen at high mass flux (1000kg/m^{2}s) for our range of heat flux. However, at low mass flux of 345kg/m^{2}s, deterioration in heat transfer was present for a wide range of conditions. Figure 7d show typical results of the cladding temperatures along the axial direction of the heated rod at deteriorated heat transfer conditions. As shown in Figure 7d, deterioration occurred at the location of tw_2, indicated by a local temperature spike of 560℃, compared with the neighboring wall temperature of 420℃ (tw_1) and 470℃ (tw_3), respectively. In our experiment, when deterioration was present, at first, the wall temperature is similar to the expected value, followed by a deterioration (i.e. indicated by spikes in wall temperature) and a recovery back to that expected. The same feature was also found in other previous experiments (Shitsman, 1963; Yamagata et al., 1972; Watts and Chou, 1982).



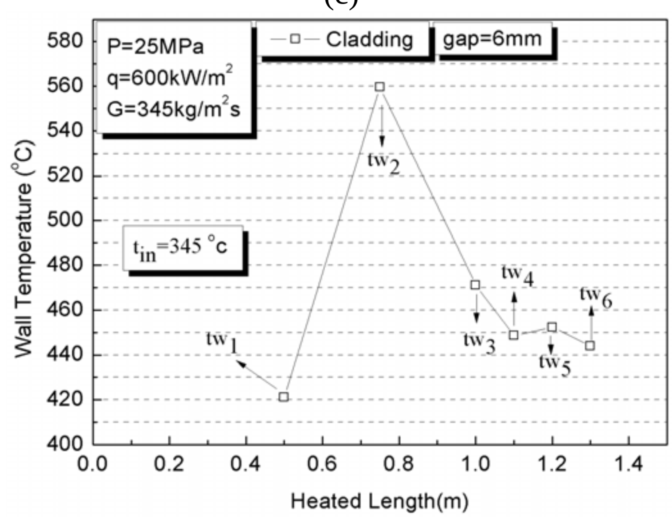
(a)



(b)



(c)



(d)

Figure 7: Wall temperature distributes along heated length

3.4 Effects of spacer

Spacers have been generally used in fuel assemblies so as to maintain an appropriate rod-to-rod clearance. As is known, spacers disturb the flow, enhance turbulence and, subsequently, the heat transfer. The schematic diagram of spacer used in our experiment is shown in Figure 8. The inner heated circular pipe is helically wrapped with a string of ceramic tubes to simulate a fuel pin with a helical wire-wrapped spacer. One pitch of the helical spacer is 5cm. The full length of helical wire-wrapped spacer along the inner pipe is 10cm, that is, two pitches. Besides, one thing needs to be emphasized, that is, the helical wire-wrapped spacer was just arranged exactly at the first measuring position. In other words, there is no spacer arranged in the other five measuring position.

In order to better evaluate the impact of spacer on heat transfer characteristic, we made a comparison of two test sections. One is a bare annular test section, and the other one has the identical geometry structure except there is a spacer positioned exactly the location where the thermocouple arranged there. Measurements were made with and without spacer inserted in the channel. Figure 9 shows typical results of wall temperature and heat transfer coefficients plotted against bulk enthalpy for two test sections with and without spacer, respectively. We can read from Figure 9 that at a bulk temperature which is slightly below the pseudo-critical temperature (T=384℃), the test section with spacer gives a maximum heat transfer coefficient of about 32 kW/m^2k, which is higher than that value of 23 kW/m^2k found for the test section without spacer. This shows, as expected, that the spacer has a positive effect in enhancing the heat transfer, especially in the pseudo-critical region.

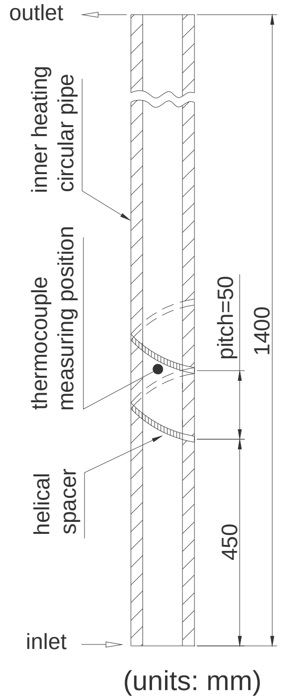


Figure 8: Schematic diagram of spacer

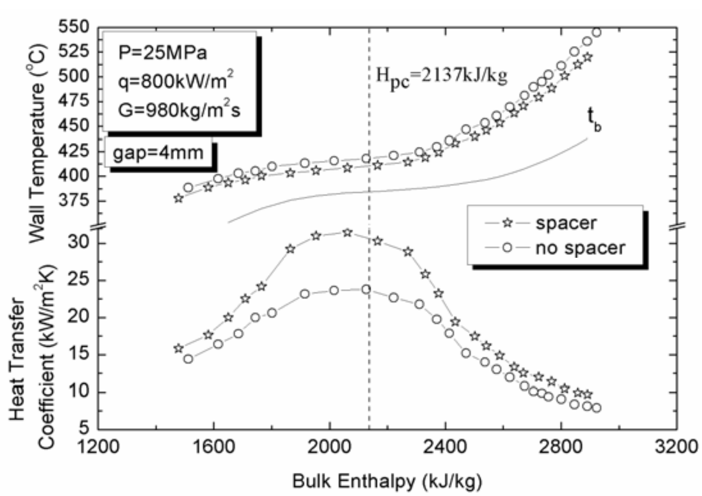


Figure 9: Comparison of wall temperature and heat transfer coefficients with and without spacer.

4. Conclusions

At relatively high mass flux conditions (1000kg/m^{2}s), XJTU data exhibit an improvement in heat transfer near the pseudo-critical temperature region. In low enthalpy and high enthalpy regions which are far away from pseudo-critical point, heat transfer coefficient is independent of heat flux. An increase in heat flux impairs the heat transfer in the region near the pseudo-critical temperature, but no deterioration in heat transfer was seen even the heat flux reaches a relatively high value (1000 kW/m^2).

The effect of gap size on heat transfer characteristic depends strongly on the flow conditions. At similar operating conditions, normal heat transfer is very susceptible to the changes in gap size, whereas conditions of deterioration were found to differ with that.

The spacer has a positive effect in enhancing the local heat transfer, especially in the pseudo-critical region.

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