Heat Transfer And Pressure Drop Characteristics Of Water Flow Boiling In Internally Enhanced Tubes

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ABSTRACT

Flow boiling is a critical process for many thermal conversion processes such as HVAC&R, water heating, power generation, and desalination applications. The goal of this paper is to enhance the in-tube heat transfer for flow boiling of water and thus improve the efficiency of the boiler. This paper presents an experimental investigation of the heat transfer and pressure drop characteristics for water under forced convection and boiling conditions inside both smooth and enhanced tubes. The results in the smooth copper tube are compared with several relevant correlations and the findings have been used as a baseline. The two-phase heat transfer and pressure drop performance of the water flow boiling in the internally enhanced tube is investigated, and the influence of internal surface pattern on the thermal characteristics is discussed. The relations of the heat transfer coefficient with mass flux, heat flux and vapor quality are analyzed, and the findings have been summarized.

1. INTRODUCTION

Flow boiling is an efficient process to transfer heat and has been widely used in various industries for many thermal conversion applications such as HVAC&R, water heating, power generation, and desalination. The major goal of this study is to enhance the in-tube heat transfer for flow boiling of water so that the efficiency of the boiler can be enhanced. It is well known that the in-tube flow boiling heat transfer is contributed by nucleate boiling and convective boiling mechanisms. The effect of surface roughness on nucleate boiling have been explored by numerous researchers and found to enhance the pool boiling heat transfer. Consequently, improving the internal surface pattern is expected to augment the activated nucleation sites and provide a higher heat transfer coefficient for the in-tube flow boiling. In this work, the internal surface of copper tubes was enhanced by using the Magnetic Abrasive Finishing (MAF) technique, and the finished surface was patterned with alternating rough and smooth bands. An infrastructure was developed to measure the in-tube heat transfer coefficient and pressure drop for testing different tube samples under liquid-phase and flow boiling conditions. A baseline case in a smooth copper tube was

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established and validated with the correlations in the literature. The experiments were then conducted in the internally enhanced tube and compared with the smooth tube. The effect of mass flux, heat flux and vapor quality on thermal performance were analyzed and discussed.

2. EXPERIMENTS

2.1 Experimental apparatus

The experimental apparatus employed in this study is designed to study the in-tube heat transfer coefficient and pressure drop of flowing water under liquid-phase and boiling conditions. A schematic diagram of the overall apparatus is presented in Figure 1. The flow loop mainly consists of a reservoir, gear pump, flow meter, pre-test condition section, heat transfer test section, visualization section, and condenser. A 2.5-gallon tank-type water heater serves as the reservoir, where the distilled water is preheated and stored. The water is pumped to the flow loop through a gear pump, and an additional by-pass loop back to the tank is designed for flow rate control. The mass flow rate of water flowing into the main line is measured with a Coriolis flow meter, and a strainer with 150-mesh screen (89 μm) is installed to catch debris and protect the equipment. There are three main sections in the flow loop: pre-heating section, flow development section and heat transfer test section. The pre-heating section is used to provide the desired conditions (temperature or vapor quality) for the heat transfer test section. The heating power is provided and controlled through six 1.5 kW band heaters. The flow development section is a 1 m long copper tube, which enables the water to reach fully developed flow. It is fully insulated to minimize the heat loss, and thus this section is nearly adiabatic. The heat transfer test section is the measurement section for the heat transfer coefficient and pressure drop for tube samples. The heat flux in the test section is supplied and controlled though two 1.5 kW band heaters. After the test section, there is a clear FEP tubing for visualizing the two-phase flow pattern, which helps understanding the simultaneous motion of liquid and vapor flows. Then, the steam-water mixture is condensed in a plate heat exchanger and directed back to the reservoir.

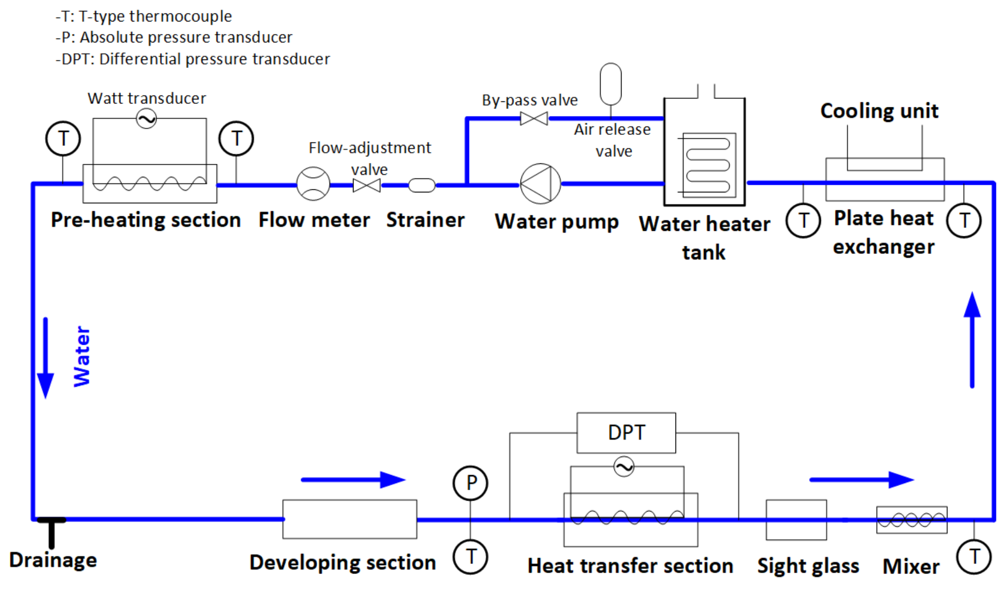


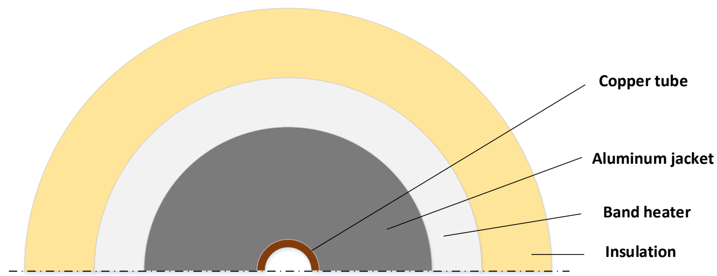
Figure 1: Schematic of test apparatus

2.2 Heat transfer test section

The heat transfer test section consists of a test tube, two half-piece aluminum jackets, and two electric band heaters, as illustrated in Figure 2(a). The test tube is a 3/8" OD copper tube with a heated length of 30 cm. The inner diameter and outer diameters are 7.89 and 9.53 mm, respectively. Around the outer surface are two half-piece cylindrical shape aluminum jackets, which are used to provide a more uniform heat flux condition and temperature distribution. All the gaps between the two aluminum pieces and the copper tube are filled with high thermal conductivity paste to reduce the thermal contact resistance. The required heat flux to the working fluid is generated through two electric band heaters. The heating power of each heater is 1.5 kW, and each heater is insulated with a

1/4" thick ceramic fiber insulating blanket. The built-in blanket comes with three screws for clamping the heat transfer assembly. The heat transfer test section is insulted with a 2" thick fiberglass pipe insulation. Additionally, there are four grooves on each aluminum jacket designed for placing thermocouples, and the locations of the wall temperature measurement and the longitudinal view of the aluminum jacket are illustrated in Figure 2(b). The wall temperatures are measured at four locations along the axis of the heated section with an equal interval of 60 mm. For each location, four thermocouples are equally distributed over the circumference and attached at the outer wall of the test tube. In total, sixteen points on the tube surface are measured and used for the heat transfer calculation. More details regarding the design of the aluminum jacket can be referred to the previous work of Yang and Hrnjak (2018).

(a)



(b)

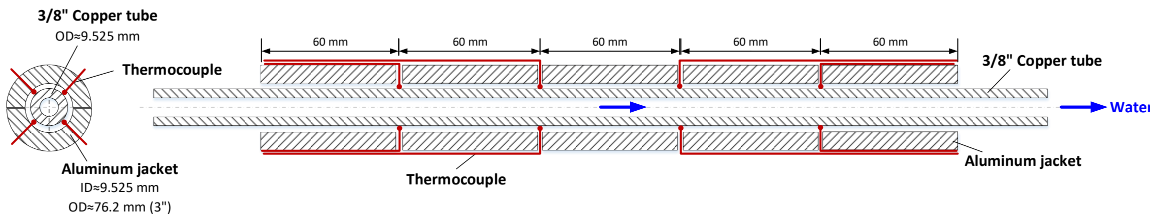


Figure 2: (a) Cross sectional view of the heat transfer test section and (b) locations of thermocouples

2.3 Tube samples

The internal surface of the copper tube is enhanced by using Magnetic Abrasive Finishing (MAF) technique. The surface with alternating rough and smooth bands is created to increase the active nucleation sites during flow boiling. The schematic of the processing principle is illustrated in Figure 3(a). Two NdFeB permanent magnets placed with opposite directions generate the magnetic field needed for attracting a mixture of magnetic particles and abrasive to the finishing area and pressing them against the inner workpiece surface. The abrasive performs micro-cutting on the inner surface of the tube when the tube is rotated at high speed. Manipulating the magnet along the tube axis enables the mixture to follow the magnet motion, thus finishing a wide area of the inner tube surface. The surface pattern can be controlled by adjusting the magnetic field locally, and an example of the finished surface of copper tube is demonstrated in Figure 3(b). Besides, the magnetic particle and abrasive type can be altered to adjust the material removal rate and the surface roughness of the finished surface. The surface roughness of the smooth area and rough area in the current sample are 0.008 μm and 0.73 μm, respectively. The sample tube in this study was manufactured with a two-step process. The internal surface was smoothed and then patterned with rough and smooth bands alternatively across a length of 440 mm, which is schematically shown in Figure 3(c). There are 87 bands along the pattern length, and the rough band width and smooth band width are 1.3 mm and 5 mm, respectively.

2.4 Instrumentation

In the facility, the fluid temperatures were measured using T-type thermocouples. The wall temperatures in the heat transfer test section were measured with K-type thermocouples. All thermocouples were calibrated with an accuracy of ±0.1 °C. The mass flow rate of water was measured by a Coriolis mass flow meter with a range of 0.001 to 0.03 kg/s with an accuracy of ±0.20% of the reading. The absolute pressure of water was determined by an absolute pressure transducer with an accuracy of ±0.31 kPa. Horizontal pressure drop of the tested tubes within the test section were measured by differential pressure transducers with an accuracy of ±5.52 Pa. The electrical power inputs to the pre-heating section and heat transfer sections were determined using watt transducers with 0.5% reading accuracy. The measured data from thermocouples, mass flow meter, pressure transducers, and watt transducer was collected by a data logger every 1 second.

(a) (b)

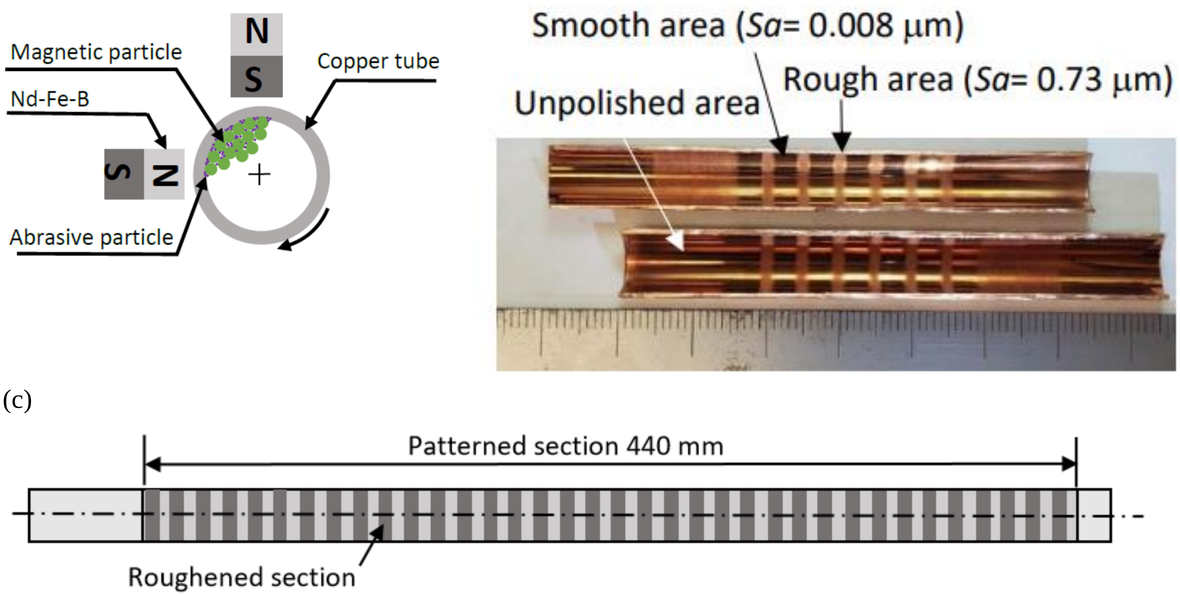


Figure 3: (a) Principle of MAF process, (b) representative polished surface of copper tube and (c) surface pattern of the sample tube

3. DATA REDUCTION

3.1 Heat transfer coefficient

The heat transfer coefficient *HTC* of the water in the test tube is determined as:

$$HTC = \frac{\dot{Q}_{ts}}{A_s\left(T_{wall} - T_{sat}\right)} \tag{1}$$

where A_s is the surface area of the test tube and it is calculated using the inner diameter and the heated length. T_{sat} is the saturation temperature estimated based on the measured saturation pressure P_{sat}, and T_{wall} is the average inner wall temperature. The wall temperature is determined by using the temperature measurement of the sixteen thermocouples on the outer wall of the test tube and is corrected to the corresponding wall temperature at the inner wall surface using 1-D heat conduction equation. The heat transfer rate to the water $\dot{Q}_{ts}$ is calculated based on the energy balance in the heat transfer test section.

$$\dot{Q}_{ts} = \dot{Q}_{elc,ts} - \dot{Q}_{amb,ts} - \dot{Q}_{cond} \tag{2}$$

where $\dot{Q}_{elc,ts}$ is the heating power of the band heaters in the heat transfer test section, and $\dot{Q}_{amb,ts}$ is the heat loss into the environment from the test section estimated through a calibration experiment. The single-phase test with a wide range of inlet water temperatures were conducted to determine the dependence of heat loss on the temperature difference between the water to the surrounding. During the heat loss test, the band heaters in the test section were turned off. The mass flow rate and the temperature change of liquid water within the test section were measured. The overall heat loss coefficient was calculated by the heat transfer rate and the log mean temperature difference between the water and the ambient air. $\dot{Q}_{cond}$ is the axially conductive heat loss through the copper tube wall, which is attributed to the higher wall temperature in the test section than that away from the test section. It is estimated by a finite element method proposed by Jang and Hrnjak (2004).

3.2 Pressure drop

The total pressure drop of two-phase flow inside the test tube is directly measured through a differential pressure transducer. For the evaporation flow, the two-phase pressure drop generally comprises of frictional pressure drop, static pressure drop, and acceleration pressure drop.

$$\Delta P_{tot} = \Delta P_{fric} + \Delta P_{static} + \Delta P_{acc} \tag{3}$$

The acceleration pressure drop in the test tube can be estimated through the change in kinetic energy of flow.

$$\Delta P_{acc} = G^2 \left\{ \left(\frac{(1-x)^2}{\rho_l (1-\alpha)} + \frac{x^2}{\rho_v \alpha} \right)_o - \left(\frac{(1-x)^2}{\rho_l (1-\alpha)} + \frac{x^2}{\rho_v \alpha} \right)_i \right\} \tag{4}$$

where x is vapor quality, α is void fraction, ρ_v is vapor density, ρ_l is liquid density, and G is mass flux.
The void fraction is calculated by the Steiner (1993) version of Rouhani and Axelsson model, as the following:

$$\alpha = \frac{x}{\rho_v} \left[\left(1 + 0.12(1-x)\right) \left(\frac{x}{\rho_v} + \frac{(1-x)}{\rho_l} \right) + \frac{1.18(1-x)\left[g\sigma (\rho_l - \rho_v) \right]^{0.25}}{G \rho_l^{0.5}} \right]^{-1} \tag{5}$$

The static pressure drop is expressed as

$$\Delta P_{static} = \left(\rho_v \alpha + \rho_l (1-\alpha) \right) g l \sin(\theta) \tag{6}$$

where g is the gravitational acceleration, l is the length of the test tube, and θ is the inclination angle of the channel. For a horizontal tube, the change of static head is 0.
Therefore, the two-phase frictional pressure drop can be obtained by subtracting the acceleration term from the measured total pressure drop, which is then used for comparison with some classical correlations.

3.3 Test conditions

During the single-phase and two-phase experiments, the parameters of the operating condition are heat flux, mass flux and vapor quality, which will be defined in this subsection. In the heat transfer test section, heat flux is the heat transfer rate to the fluid per unit area, and it is calculated as the following equation.

$$q_{ts} = \frac{\dot{Q}_{ts}}{\pi D_i L} \tag{7}$$

where D_i is the inner diameter, and L is the heated length of the test tube.
The mass flux of the water in the test tube G is calculated as

$$G = \frac{\dot{m}_w}{A_c} \tag{8}$$

where $\dot{m}_w$ is the mass flow rate of water, and A_c is the cross-sectional area of the test tube.

$$A_c = \frac{\pi D_i^2}{4} \tag{9}$$

The inlet vapor quality of the steam-water mixture at the inlet of the test section is controlled by the band heaters in the pre-heating section. The subcooled water enters the preheating section and is heated until the desired vapor quality is reached. The thermodynamic vapor quality at the inlet of test section can be estimated through the energy balance

$$x_{ts,i} = \frac{\dot{Q}_{pre} - \dot{m}_w C_{p,w} (T_{pre,o} - T_{pre,i})}{\dot{m}_w h_{fg}} \tag{10}$$

where $\dot{Q}_{pre}$ is the heat transfer to the water in the preheating section, $\dot{m}_w$ is the mass flow rate of the water, $C_{p,w}$ is the specific heat of the water at the constant pressure, $T_{pre,i}$ and $T_{pre,o}$ are the temperature at the inlet and outlet of the preheating section, and h_{fg} is the latent heat of the water.

$$\dot{Q}_{pre} = \dot{Q}_{elc,pre} - \dot{Q}_{amb,pre} \tag{11}$$

where $\dot{Q}_{elc,pre}$ is the heating power of the band heaters in the preheating section and $\dot{Q}_{amb,pre}$ is the heat loss into the environment from the preheating section. The heat leak from the preheating section through the insulators were measured through the single-phase calibration experiments. Similarly, the vapor quality of the steam-water mixture at the outlet of the test section is determined with following equation.

$$x_{ts,o} = x_{ts,i} + \frac{\dot{Q}_{w,ts}}{\dot{m}_w h_{fg}} \tag{12}$$

The mean vapor quality of the water in the heat transfer test section is calculated using the arithmetic mean value of outlet and inlet qualities given in Eq. (10) and (12).

$$x_{ts} = \frac{x_{ts,i} + x_{ts,o}}{2} \tag{13}$$

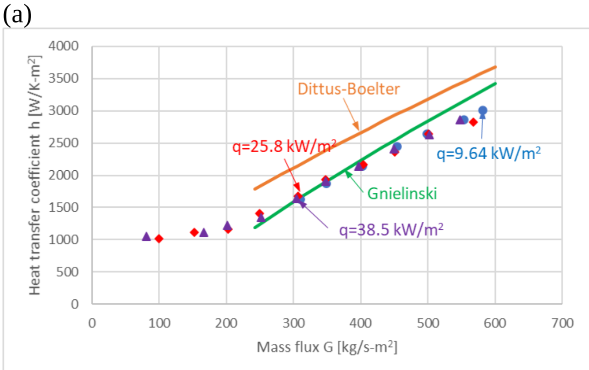
4. RESULTS AND DISCUSSION

In this section, the experimental results of heat transfer coefficient and horizontal pressure drop of water under liquid-phase and two-phase conditions are reported. First, the baseline experiments are carried out in a smooth tube and used for validating the reliability of the experiments. Then, the experiments performed in the internally enhanced tube are presented. The effects of mass flux, heat flux, and vapor quality on heat transfer coefficient and pressure drop are analyzed. Finally, the experimental data acquired in these two tubes are compared and discussed.

4.1 Heat transfer and pressure drop in smooth tubes

The single-phase experiments in a smooth tube were conducted at the inlet temperature of 57°C covering a mass flux range of 100 to 600 kg/s-m^2. Figure 4(a) shows the measured heat transfer coefficient for the smooth tube at the heat fluxes of 9.64 kW/m^2, 25.8 kw/m^2, and 38.5 kW/m^2. It is seen that the heat transfer coefficient increases as the mass flux increases, and the heat flux does not influence the heat transfer of liquid water as expected. The results are also compared with the predicted values from the widely used correlations including Dittus and Boelter (1930) and Gnielinski (1976). For the turbulent region, the experimental data are in line with the Gnielinski's correlation. The correlation of Dittus and Boelter, however, overpredicts the heat transfer coefficient. It might be because the correlation was originally developed and validated for the Re >10000, but the maximum Re in the current experiments is around 6000. Figure 4(b) shows the horizontal pressure drop measurement under adiabatic conditions. The experimental results are compared with the calculated values from Hagen-Poiseuille flow and Blasius equation and show in good agreement. As a result, the test facility is well established and reliable.

(a)



(b)

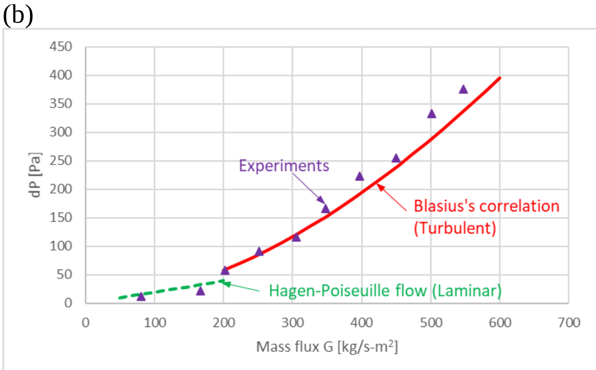
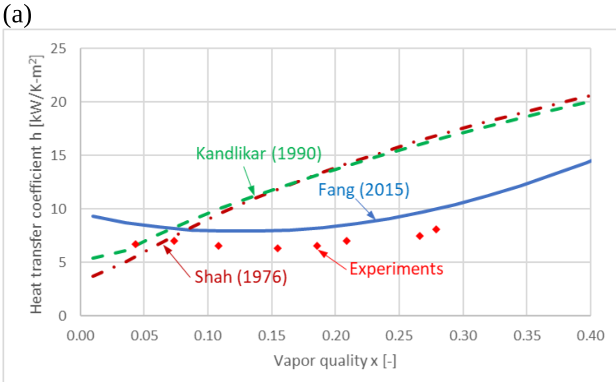


Figure 4: Comparison of the (a) heat transfer coefficient and (b) pressure drop of the liquid water inside smooth tubes with the correlations at different mass fluxes

For the flow boiling experiments, the heat transfer coefficient and pressure drop were measured at q=33 kW/m^2 and G=80 kg/s-m^2 over a vapor quality range of 0 to 0.3. Since the facility is an open loop system, the outlet pressure of the condenser is around 1 atm, and the inlet pressure of the test section varies from 104 kPa to 118 kPa depending on the employed vapor quality. Figure 5(a) shows the comparison of the measured heat transfer coefficient at constant heat flux and mass flux with the correlations of Shah (1976), Kandlikar (1990), and Fang *et al.* (2015). It is seen that the Shah correlation and Kandlikar correlation overpredict the heat transfer coefficient, especially at higher vapor qualities. The reason may be because the above correlations were proposed to fit most of the general fluids and are not the best option to predict the performance of water. The Fang correlation, specifically developed for water, has better prediction accuracies for the current experiments. The pressure drop within the test section were measured under the evaporative condition. In order to compare with the correlations of frictional pressure drop, the acceleration pressure drop attributed to the evaporation is subtracted from the total measurement. Figure 5 (b) compares the frictional pressure drop obtained from the experiments to the correlations including Lockhart–Martinelli (1949), Friedel (1979), and Zhang *et al.* (2010). The experimental results are closer to the predicted values of the Zhang's correlation especially at low vapor quality region. The Zhang's correlation was developed based on the artificial neural network, has a larger data bank of water and therefore a better prediction.

(a)



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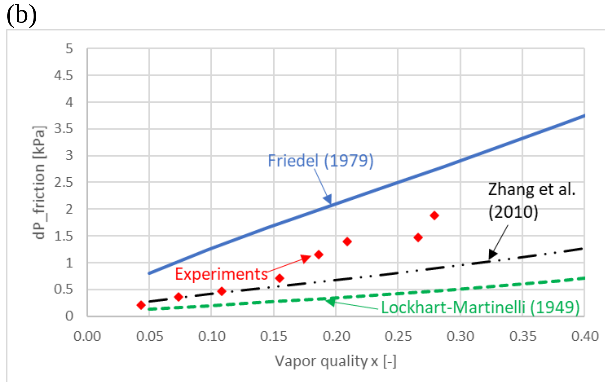

Figure 5: Comparison of the (a) heat transfer coefficient and (b) frictional pressure drop for the flow boiling of water inside smooth tubes with the correlations at various vapor qualities

4.2 Heat transfer and pressure drop in internally enhanced tubes

The flow boiling experiments for the internally enhanced tubes were carried out at various heat fluxes, mass fluxes, and vapor qualities. Figure 6(a) displays the variations of heat transfer coefficient at the heat flux of 34 kW/m^2 with varying vapor quality and mass flux. It is seen that the heat transfer coefficient generally increases with the rise of mass flux especially at higher vapor qualities, in which the convective boiling mechanism is more important. Besides, the trends of the heat transfer coefficient versus vapor quality at the mass fluxes of 80 kg/s-m^2, 150 kg/s-m^2, and 200 kg/s-m^2 are similar. The heat transfer coefficient first decreases with the rise of vapor quality and then increases as the vapor quality is over a certain value. Figure 6(b) shows the dependences of heat transfer coefficient on vapor quality for the constant mass flux of 80 kg/s-m^2 at two heat fluxes (34 kw/m^2 and 78 kw/m^2). There seems to be no effect of heat flux on the heat transfer coefficient in the current test conditions.

The horizontal pressure drop within the internally enhanced tube for flow boiling of water was measured in a diabatic test section. The effects of mass flux and heat flux on total pressure drop are plotted as a function of vapor quality in Figure 7(a) and Figure 7(b), respectively. As shown in Figure 7(a), the pressure drop increases with the increasing of vapor quality by reason of the accelerations of both liquid and vapor inside the tube. Furthermore, the pressure drop increases as mass flux increases owing to the increasing friction as well. Figure 7(b) presents the dependences of the total pressure drop on vapor quality and heat flux. It is found that the heat flux only has little influence on the total pressure drop. The increasing of heat flux leads to a higher vaporization and thus accelerates the fluid more compared to the case of low heat flux.

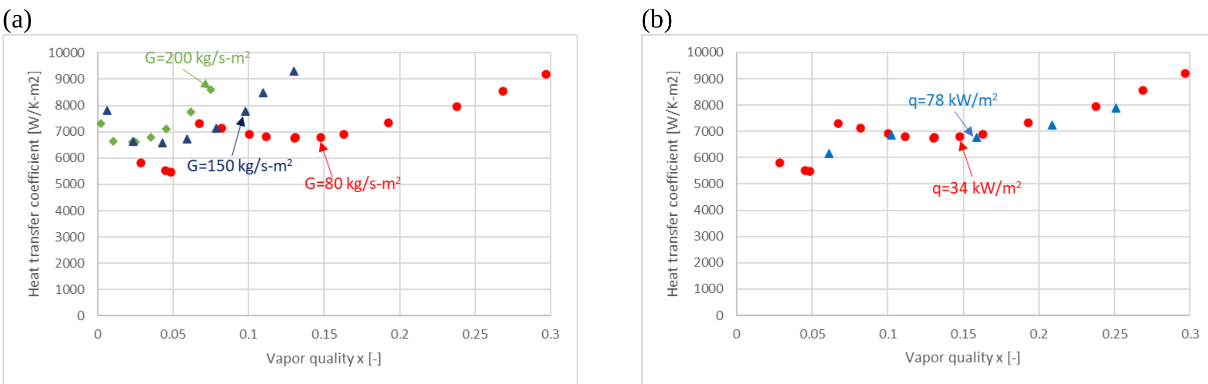

Figure 6: Effects of (a) mass flux and (b) heat flux on the heat transfer coefficient of the flow boiling of water

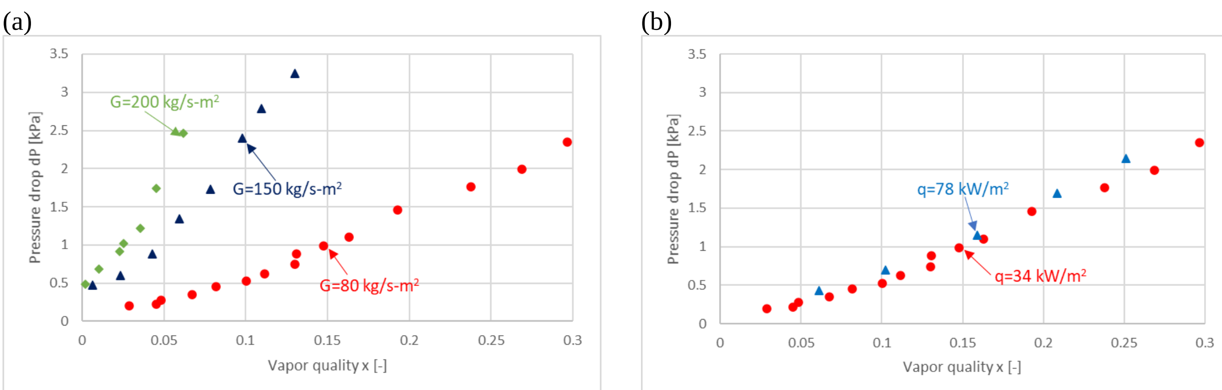

Figure 7: Effects of (a) mass flux and (b) heat flux on the total pressure drop of the flow boiling of water

4.3 Performance comparison of smooth and internally enhanced tubes

Figure 8(a) shows the comparisons of heat transfer coefficient in smooth and internally enhanced tubes under single-phase conditions. The inlet temperature of the heat is set at 57°C, and the heat flux is 34 kW/m^2. As expected, the internal surface pattern does not increase single-phase heat transfer. The measured pressure drop for liquid water under various mass fluxes in the two tubes are also compared, as presented in Figure 8(b). No markable difference between the two tubes are found.

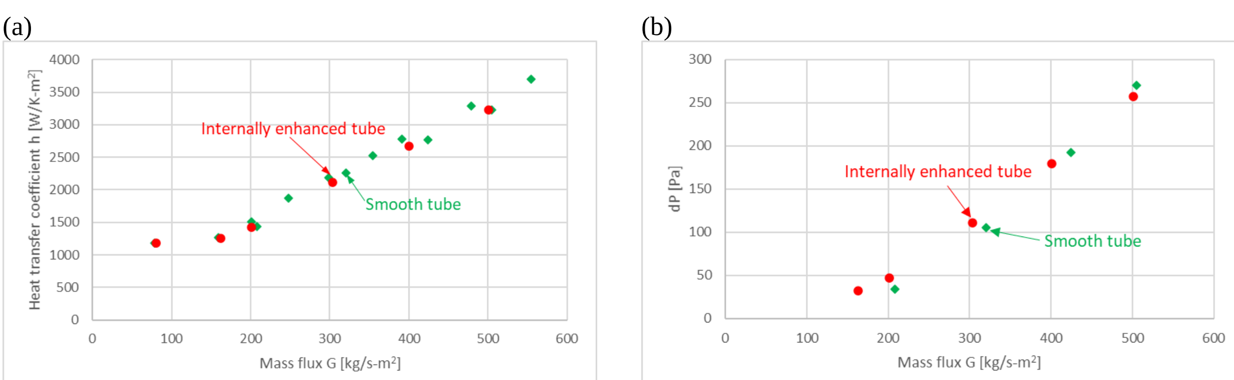

Figure 8: Comparison of (a) heat transfer coefficient and (b) total pressure drop of liquid water in smooth and internally enhanced tube (T=57°C, q=34 kW/m^2)

Figure 9(a) demonstrates the heat transfer measurement in the flow boiling experiments. It is seen that the heat transfer coefficient of the enhanced tube is even lower than that of the plain tube at very lower vapor quality conditions. As the vapor quality increases, the heat transfer enhancement due to the chemical treatment becomes significant. It is around 10% higher than the plain tube as the vapor quality is over 0.2. Figure 9(b) shows the measured pressure drop of flow boiling of water inside the two tubes. In general, their results are very similar. Consequently, the initial flow boiling test indicates the internal chemical treatment slightly improves the heat transfer coefficient without increasing pressure drop. In the future work, higher heat flux conditions will be investigated to better understand the effect of rough surface on the flow boiling heat transfer. Besides, different surface patterns with various band widths and number of bands will be studied.

(a) (b)

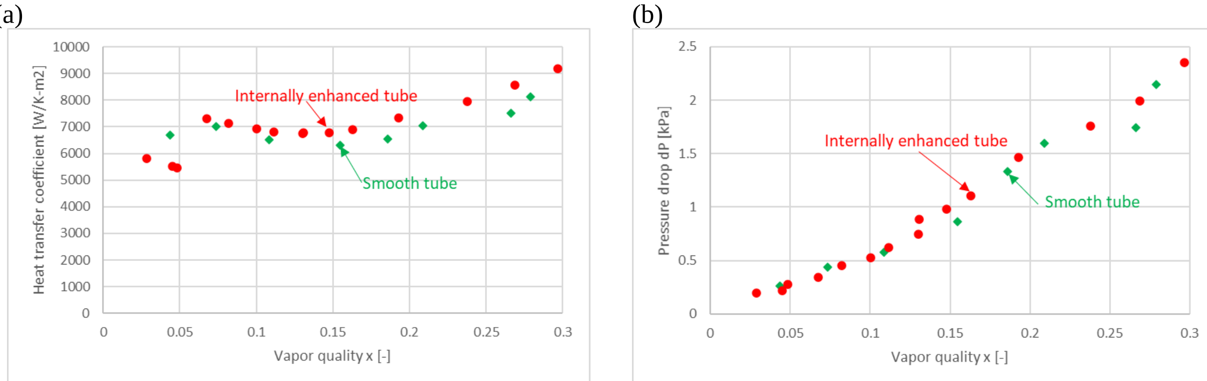

Figure 9: Comparison of (a) heat transfer coefficient and (b) total pressure drop of flow boiling of water in smooth and internally enhanced tube (G=80 kg/s-m^2, q=34 kW/m^2)

5. CONCLUSIONS

This study experimentally investigated the effect of enhanced internal surface on the in-tube heat transfer coefficient and pressure drop for flow boiling of water. The internal surface was patterned with alternating rough and smooth bands with MAF technique. An infrastructure was developed and validated with the classical correlations. The heat transfer coefficient and pressure drop inside the two tubes under liquid-phase and flow boiling conditions were measured and analyzed. The following conclusions can be drawn from this work.

- The test facility is well established, and the measured heat transfer coefficient and the pressure drop of water in the smooth tube show good agreement to the correlations in literature.
- The influences of heat flux and mass flux on the heat transfer coefficient and pressure drop in the internally patterned tube are reported. Mass flux has a stronger influence on the performance than the heat flux for the current test conditions.
- As expected, there is no measurable difference of heat transfer and pressure drop under single phase conditions in the internally patterned tube compared to the smooth tube.
- The preliminary results indicate that the rough surface pattern only has minor influence on the heat transfer of flow boiling of water. It is found that the heat transfer coefficient is around 10% higher than the plain tube as the vapor quality is over 0.2.
- Higher heat flux conditions and different surface patterns with various band widths and number of bands will be further investigated.

NOMENCLATURE

A	area	(m^2)
C_p	specific heat	(J/kg-K)
D_i	inner diameter	(m)
G	mass flux	(kg/s-m^2)
h_{fg}	latent heat	(J/kg)
HTC	heat transfer coefficient	(kW/K-m^2)
L	heated length of the test tube	(m)

l	length of the test tube	(m)
$\dot{m}$	mass flow rate	(kg/s)
P	pressure	(kPa)
$\dot{Q}$	heat transfer rate	(W)
q	heat flux	(kW/m^2)
T	temperature	(°C)
x	vapor quality	(–)

Subscript

amb	ambient
avg	average
cal	calorimeter
cond	conduction
cs	cross section
i	inlet
o	outlet
OD	outer diameter
s	surface
ts	test section
w	water

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