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Condensation and Evaporation Characteristics of Flows Inside Three Dimensional Vipertex Enhanced Heat Transfer Tubes

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Results are presented here from an experimental investigation of condensation and evaporation heat transfer that compares the performance of three dimensional (3-D) enhanced heat transfer tubes to the performance of a smooth surface tube. The equivalent outer diameter of all the tubes was 12.7 mm with an inner diameter of 11.5 mm. Both the inner and outer surfaces of the 3-D tubes are enhanced by a primary enhancement and a background pattern made up by an array of dimples. Experimental runs were performed using R410A as the working fluid, over the quality range of 0.2 – 0.9. The test apparatus included a horizontal, straight test section with an active length heated by water circulated in a surrounding annulus. Constant heat flux was maintained and refrigerant quality varied.
For evaporation, the heat transfer coefficient ratio of the 3-D tubes (comparing the heat transfer coefficient of the enhanced tube to that of a smooth tube) is in the range of 1.1 – 1.80 for a mass flux rate that ranges from 80 to 180 kg/m^{2}s. For condensation, the heat transfer coefficient ratio range is 1.1 - 1.75 for mass flux that ranges from 80 to 260 kg/m^2 s. Frictional pressure drop values for the enhanced tubes show some variation. Heat transfer enhancement on the inner surface of the 3-D tubes increases the surface area and interfacial turbulence; producing flow separation, secondary flows and a higher heat flux from the wall to the working fluid. Enhanced heat transfer tubes are important options to be considered in the design of high efficiency systems. A wide variety of industrial processes involve the transfer of heat energy during phase change and many of those processes employ old designs. These processes are ideal candidates for a redesign that could achieve improved process performance. These three dimensional enhanced tubes recover more energy and provide an opportunity to advance the design of many heat transfer products.

1. Introduction

There is a large amount of heat exchanged during heat transfer processes that involve phase change. Heat transfer involving phase change is required in many industrial applications including air conditioning and refrigeration applications, food production, cryogenics, desalination, electronics cooling, heat recovery and in the power and process industries. Specific process applications include vaporizing, condensing, partial condensing, and freezing. Industrial applications with phase change typically use water-based fluids, hydrocarbons or refrigerants; fluid type is based upon various considerations including the amount of heat that needs to be transferred, operating conditions, and interactions of the fluid components. Approximately 60 % of industrial heat exchangers are two phase devices.
Since the 1930's engineers have tried to improve phase change heat transfer by altering the process surface. Boiling, condensation, and frost formation are quasi-steady, multiphase processes that might be enhanced with the properly designed surface. For example, vigorous vaporization occurs from the boiling of a liquid near a heated surface; some surfaces can be engineered to facilitate increased nucleation with the maximization of the critical heat flux.

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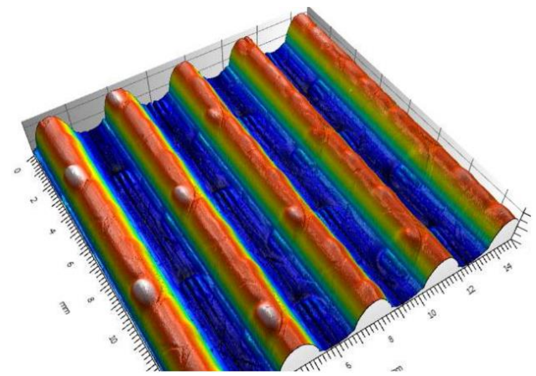
Previous studies have been performed to evaluate the tube side heat transfer and pressure drop characteristics of horizontal micro-fin and some three dimensional tubes in two phase flow. Cavallini et al. (2009) presented a detailed review of condensation heat transfer in smooth and enhanced tubes. They discuss previously reported semi-empirical correlations that are commonly used and accepted for some enhanced tubes to predict the performance of enhanced or non-circular tubes; inaccurate results may be predicted when these correlations and used to predict the heat transfer performance in the newly designed three dimensional tubes. Rollmann et al. (2016) studied the flow boiling of R407C and R410A in a horizontal micro-fin tube with saturation temperature ranging from -30 ^{O}C to 10 ^{O}C; heat flux ranging from 1 to 20 kW/m^2; mass flux in the range of 25 - 300 kg/m^{2}s; with a vapor quality that ranged between 0.1 and 1. Kukulka et al. (2014) present low flow, in tube condensation/ evaporation results for the Vipertex 1EHT tube and a detailed literature review. Similar performance is demonstrated in this study for the three dimensional surface tubes for a wider range of flow conditions.

2. Experimental Set Up

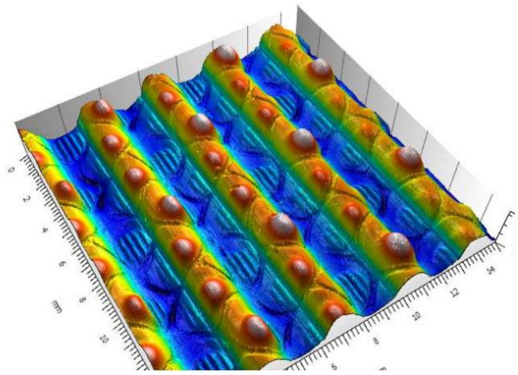
Condensation and evaporation performance characteristics of the Vipertex 1EHT, 2EHT-1 and 2EHT-2 enhanced tubes were investigated. The enhancement surface structure of the tubes (as obtained with an optical profilometer) is shown in Figure 1 for the three dimensional tubes. A schematic diagram of the test apparatus used for this in-tube condensation and evaporation is presented in Figure 2a. The experimental apparatus is composed of four loops: (i) refrigerant circulation loop (including the test section); (ii) pre-heating water circuit - used to heat or cool the refrigerant; this is necessary in order to obtain specified quality of the fluid before it enters the test section; (iii) water circuit - used to control the outlet quality of the tube being tested by changing the water inlet temperature it is; (iv) condensation section - used to sub-cool the refrigerant coming from the outlet of test section. The refrigerant flow loop consists of a 50 L reservoir tank, a digital gear pump, a pressure regulating valve, a mass flow meter, a pre-heater, a test section, a condenser, and sight glasses. A by-pass line from the pump to the reservoir and the regulating valve are used to control the mass flow rate through the refrigerant loop. Between the pump and the pre-heater is a mass flow meter (with an accuracy of ±0.2 % of the reading) and it is used to measure refrigerant flow rates. Sub cooled liquid is electrically heated in the pre-heater to create a two-phase flow with a certain inlet quality at the pre-heater outlet; the fluid then goes into the test section and will be condensed or evaporated. Finally, the two-phase refrigerant is totally condensed and sub cooled in a low temperature, 9 kW alcohol-water bath. A Platinum 100 RTD (with an accuracy of ±0.07 K) and a pressure transducer (with an accuracy of ±0.2 %) is located at the pre-heater inlet and used to determine enthalpy and the thermodynamic state of the fluid. The water circuit includes the annulus, a water thermostat, a centrifugal pump, a control valve, and a magnetic flow meter. A magnetic flow meter (with an accuracy of ±0.35 % of the reading) is used to record the flow rates of water in the annulus of the test section. In order to minimize heat losses, 40 mm thick foam insulation with a layer of 6 mm thick rubber insulation is used to insulate the various components in the test apparatus. This insulation is especially important in the pre-heater and the test section.

Figure 2b shows the straight, 2 m long tube-in-tube horizontal test section where the experimental tube sections (smooth tube and the enhanced tubes) are evaluated. Figure 2c details the cross sectional view of test tube; thermal insulation blanket of the test section; and the preheating section. In order to minimize heat loss through the outer side of the annulus, the entire test section (including the tube-in-tube heat exchanger) is installed in a larger PVC tube. It is then insulated using an insulation foaming agent that provides an insulation layer between the outer wall of double tube heat exchanger and inner wall of PVC tube that is approximately 80 mm thick (see Figure 2c). Total heat loss of the test section can be controlled to less than 3 %; and the heat loss in the test section is an insignificant factor to the final results. The test apparatus, degassing process, measurement procedures, data reduction process and uncertainty analysis that was first performed by Li et al. (2012); it was later modified by Wu et al. (2013) for use in evaporation analysis and by Wu et al. (2014) for condensation analysis over a wider range of mass flux values. This modified apparatus and procedure was utilized in this study. Verification of the test apparatus for single-phase flow and two-phase flow in smooth tubes was previously performed by Wu et al. (2013) for evaporation and Wu et al. (2014) for condensation. Figure 1 provides views of the enhancements on the inner and outer tube surfaces of the 0.50 in (12.7 mm) outer diameter (OD), Vipertex 2EHT-1 (see Figure 1a and 1c), 2EHT-2 (see Figure 1b and 1d) and 1EHT (see Figure 1e and 1f), enhanced copper tubes that have been evaluated in this study. Dimensions of the tubes evaluated include: (a) smooth tube inner diameter of 11.5 mm; (b) Vipertex 1EHT and 2EHT tubes with an inner diameter of approximately 11.5 mm. Enhanced surface structures of the two different 2EHT tubes and the 1EHT tube have the same outer diameter (OD) of 12.7 mm and inner diameter of 11.5 mm.

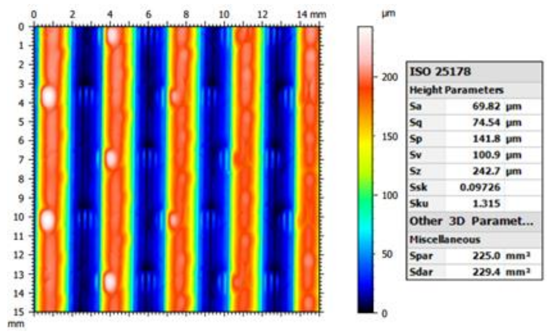
As is shown in Figure 2b, R410A is condensed or evaporated in the test tube as water flows in the annulus of the test section. Temperature, pressure and differential pressure are measured simultaneously during the test process. The measured saturation temperatures are in good agreement (within 0.2 K) with the referenced saturation temperature that is obtained at the measured inlet pressure. Average heat transfer coefficients (HTC) in W/m^{2}K rather than local values are measured due to the annular design of test section and the limitation of measuring points. Pressure drop is limited to the overall pressure drop across the 2 m tube.



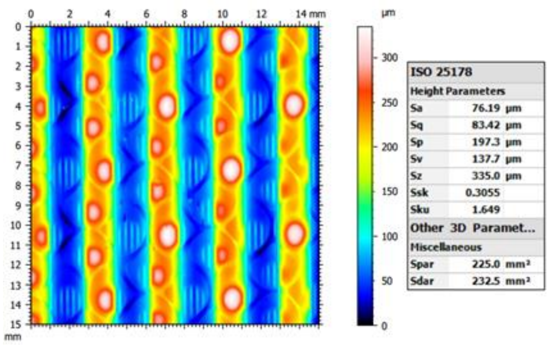
(a) Inner surface of the 2EHT-1 tube



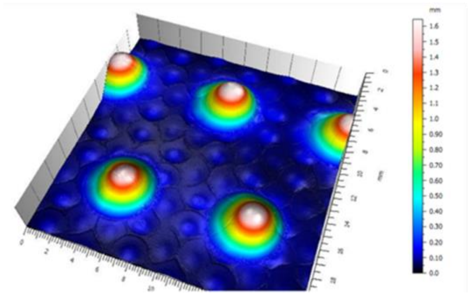
(b) Inner surface of the 2EHT-2 tube



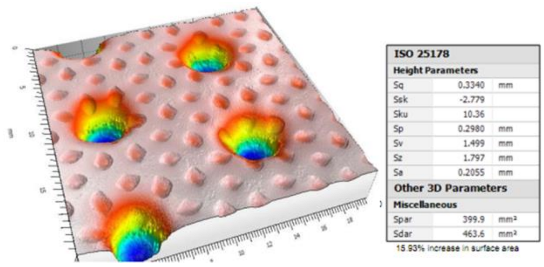
(c) Outer surface of the 2EHT-1 tube



(d) Outer surface of the 2EHT-2 tube



(e) Inner surface of the 1EHT tube



(f) Outer surface of the 1EHT tube

Figure 1: Three Dimensional false color view of the surface enhancement structure

Tubeside condensation and evaporation heat transfer characteristics are studied using refrigerant R410A; at the saturation temperature of 45 ^{O}C and 6 ^{O}C; for mass flux values that range from 80 to 260 kg/m^{2}s (condensation) and 80 - 180 kg/m^{2}s (evaporation); with a quality variation for condensation that ranges from 0.9 (inlet) to 0.2 (outlet) and for evaporation 0.2 (inlet) to 0.9 (outlet). During the experiment, inlet/ outlet quality and saturation temperature were fixed; while mass flux and heat flux varied. Mass flux was determined by the inner tube cross sectional area; heat flux was calculated using surface area obtained from the 3-D profilometer (as shown in Figure 1). Data was collected over 400 s (averaging 20 points, each with interval of 20 s), when deviations of temperature were below 0.1 K; deviations of pressure below 10 kPa; and deviations of quality were below 0.05. Single phase heat transfer evaluation of all the tubes were performed in order to determine the heat transfer enhancement factor on the annuli (water) side of the enhanced tube during tubeside condensation and evaporation testing. In addition, verification of the performance of the insulation of the test section was also performed. Single phase experiments were carried out using a subcooled refrigerant at a mass flow rate of 73 kg/h; with annuli (water) mass flow rates that ranged from 325 to 548 kg/h. Height of the enhancements, primary enhancement shape and the differences in surface area are the major outer surface differences between the three dimensional enhanced tubes.

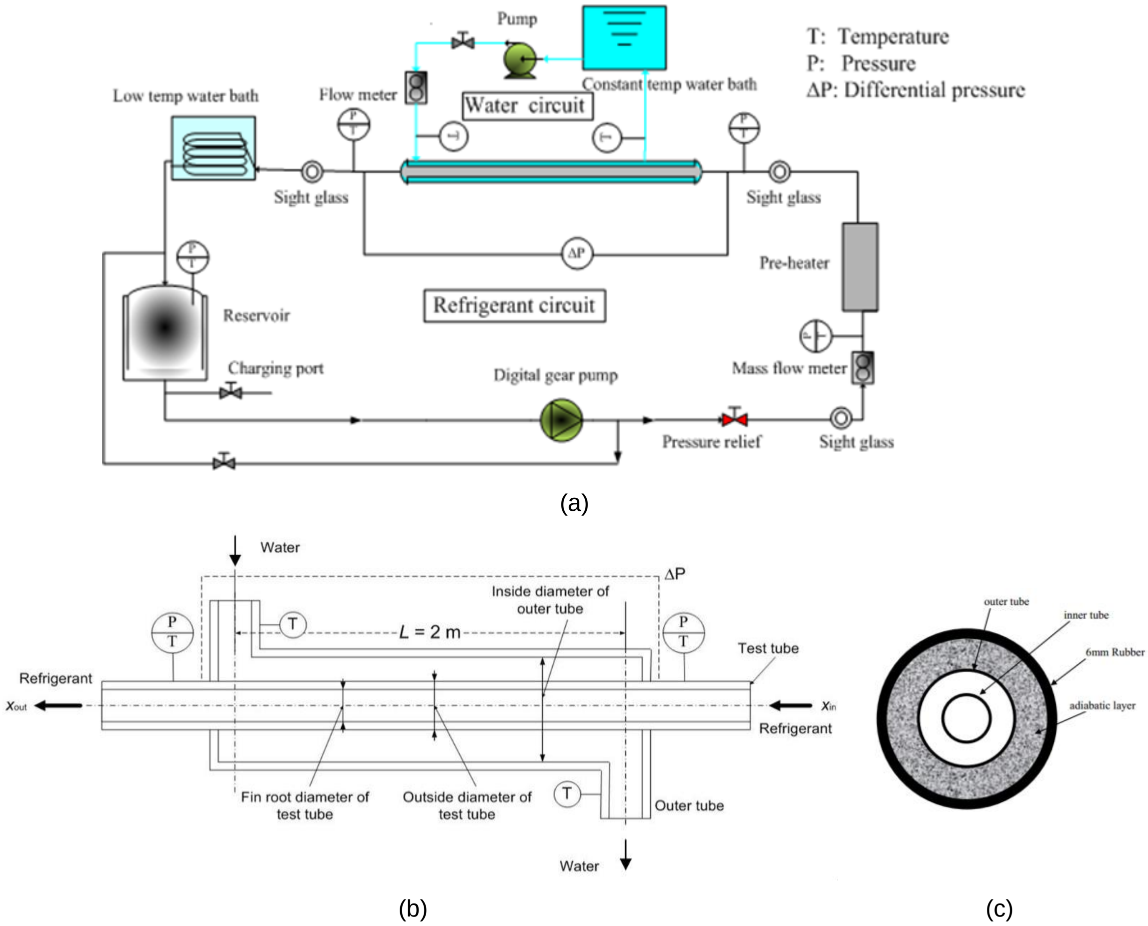
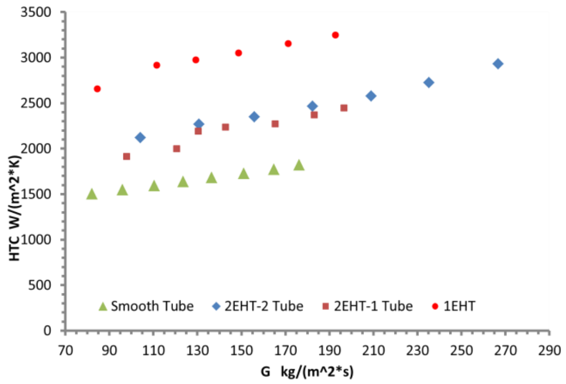


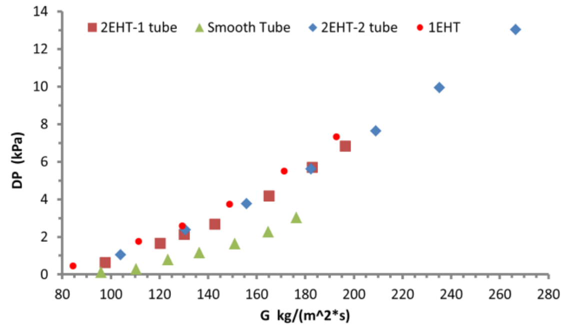
Figure 2: Schematic of (a) the experimental apparatus, (b) test section; and (c) cross sectional view of test section

3. Results

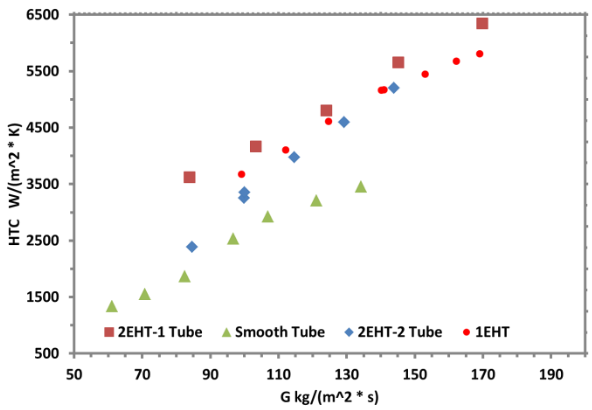
Condensation of R410A in a horizontal, three dimensional surface enhanced tubes (1EHT, 2EHT-1 and 2EHT-2; with a nominal OD of 12.7 mm and ID of 11.5 mm) were evaluated and performance compared to a smooth tube of the same dimensions. Experimental runs were controlled at a saturation temperature 45 ^{O}C; with a mass flux ranging from 80 to 260 kg/m^2 s; with an inlet quality of 0.9 and an exit quality of 0.2. Average values of the heat transfer coefficient for the 2 m long tested tubes are presented in Figure 3. Here it can be seen that the heat transfer coefficient increases with increasing mass flux; however, the rate of increase varies among tubes. Unlike the smooth tube, the three dimensional enhanced tubes (1EHT, 2EHT-1 and 2EHT-2) produce a different variation of heat transfer characteristics. Both of the 2EHT tubes demonstrate a similar variation of the HTC as a function of mass flux; producing uniform increase of HTC with increases to the mass flux. Extended surfaces are produced from an assembly of longitudinal grooves and dimples on both sides of the tube surface (as shown in Figure 1). The repeated formation of the flute shape grooves around the circumference provides the possibility to redistribute the liquid film during an auto-balance process that takes place among the body force, shear force and surface tension for the higher mass flux values of this experiment. Furthermore, the flute shape of the 2EHT-1 and 2EHT-2 tubes promotes condensate drainage from the surface by the surface tension forces. For low mass flux values, the vapor phase condenses on the top of the tubeside surface wrinkle and then as a result of the surface tension, the liquid film moves to the grooves. As mass flux increases, an increase to the heat transfer coefficient takes place as the shear force takes the place of gravity; eventually it dominates the flow regime. In addition to the extended surfaces, the rough background surfaces (due to the small dimples) produces additional disturbances and mixing, i.e. creating secondary flows near the boundary layer. This produces a larger HTC during the convective flow process of the liquid film. All of these factors contribute to a larger (when compared to a smooth tube) condensation HTC found in the enhanced tubes; this is clearly shown in Figure 3a. It can be seen that for higher mass flux rates that the heat transfer coefficients are linearly related to mass flux; since at these higher flows the drainage channels are flooded. The only difference between the 2EHT-1 and 2EHT-2 tubes is the depth of dimple enhancement (on the outer surface) or embossments (on the inner surface). Thus, the heat transfer coefficient curves of both of the 2EHT tubes demonstrate the same tendency and produce only a slight increase in the condensation heat transfer coefficient for the 2EHT-2 tube. Figure 3b shows little variation in pressure drop. Condensation heat transfer results for low flow were compared to 1EHT results from Kukulka et. al (2014) and good agreement is seen. Higher flow rates are new. The new results presented in this study can only be compared to the results of a smooth tube since these three dimensional tubes are novel and no other comparison can be made.



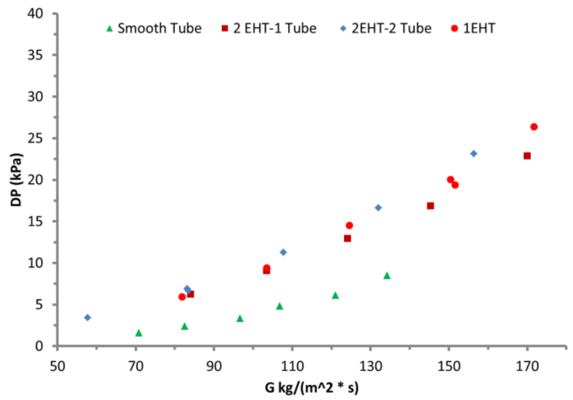
(a) Condensation heat transfer coefficient (HTC) as a function of mass flux (G)



(b) Condensation pressure drop as a function of mass flux (G)



(c) Evaporation heat transfer coefficient (HTC) as a function of mass flux (G)



(d) Evaporation pressure drop as a function of mass flux (G)

Figure 3: Comparison of tubeside condensation (a and b) and evaporation (c and d) for enhanced tubes (1EHT, 2EHT-1 and 2EHT-2) and smooth tubes using R410a

Figure 3c compares the test results of the tube side evaporation heat transfer coefficient with that of a geometrically similar smooth tube at the saturation temperature 6^{O}C; with quality ranging from 0.2 to 0.9. Due to the experimental limitation of pressure drop and the frequency of the digital gear pump (maximum frequency 50Hz), the range of mass flux was limited. A comparison of the evaporation frictional pressure drop of the three dimensional tubes and a smooth tube using R410A is given in Figure 3d. For evaporation, the pressure drop due to momentum makes up 6 - 8% of the pressure drop for the smooth tube, 2.3 - 3.1% for 2EHT-1 and 3.4 - 4.0% for and the 2EHT-2 tubes. An attempt to compare the test results with available correlations was made; however, no previous correlation can accurately predict the results of the three dimensional tubes. Surface structure of the three dimensional tubes produces many turbulent mixing regions which produces a different local temperature distribution of the R410A liquid film than is seen with a smooth tube. The top of the primary enhancement and the background enhancements extend into the liquid film producing a higher heat flow rate of heat flowing from the inner tube wall to the liquid film near the boundary layer. Heat flux (for tubeside evaporation of R410A) of the 2EHT-1 tube ranges from 17.88 to 40.01 kW/m^2, and 20.71 to 43.81 kW/m^2 for the 2EHT-2 tube; compared to 15.26 to 29.53 kW/m^2 for a smooth tube; for a mass flux ranging from 85 to 180 kg/m^2s.

Enhanced ratio results for the two 2EHT tubes are very similar; and are determined from the surface structure details presented in Figure 1. Surface areas of the three dimensional tubes are not very different from each other; therefore the structure of the nucleation sites is not significantly different. Since the heat flux or the wall temperature was set in order to initiate nucleate boiling, the overall heat transfer coefficient is mainly determined from the heat transfer rate of phase change for the given mass flux range; with enhancement ratios for the three dimensional tubes is approximately 1.1 - 1.75 (condensation) and 1.1 – 1.80 (evaporation).

4. Conclusions

An experimental study of the convective condensation and evaporation performance of three dimensional enhanced surface heat transfer tubes and a smooth tube was carried out at the saturation temperatures of 45 ℃ and 6 ℃. Frictional pressure drop and mean heat transfer coefficients were presented, with the following conclusions being made:

(1) Tubeside condensation and evaporation heat transfer characteristics of the 1EHT, 2EHT-1, 2EHT-2 tubes and a smooth tube were presented. For condensation, 2EHT-2 tube produces a higher heat transfer coefficient than the 2EHT-1 tube; 1EHT tubes present a much higher value. For evaporation, all three enhanced tubes produce approximately the same heat transfer coefficient.

(2) Pressure drop characteristic of the three dimensional tubes have also been studied experimentally. For evaporation, a higher frictional pressure drop is found during experimental runs of the three dimensional tubes when compared to smooth tubes.

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