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Experimental Comparison of the Evaporation and Condensation Heat Transfer Coefficients on the Outside of Enhanced Surface Tubes with Different Outer Diameters

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An experimental investigation was performed, and results are presented here for an evaporation and condensation heat transfer study that took place using enhanced heat transfer (1EHT) tubes of different outside diameters. Results were compared to the performance of a smooth surface copper tube. The equivalent outer diameter of the horizontal copper tubes considered were 12.7 mm (0.5 in) and 19.05 mm (0.75 in). The test apparatus included a straight horizontal test section with an active length heated by water. Constant heat flux was maintained and refrigerant quality varied.
Experimental runs were performed using R410A and R134a as the working fluids. Experimental condensation results are presented for a fixed inlet/outlet vapor quality and fixed saturation temperature, while mass flux varied; producing a heat transfer coefficient increase (compared to smooth tube) in the range 54 to 64 %, with a frictional pressure drop increase for the 1EHT tubes in the range of 8 to 31 %. Unexpected evaporation results were found at higher mass flows; producing a decrease (compared to smooth tubes) of the heat transfer coefficient for 1EHT tubes in the range of 100 – 220 %; with an increase of frictional pressure drop values for the 1EHT tubes in the range 5 to 17 %.
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 technology. These processes are ideal candidates for a redesign that could achieve improved process performance. Vipertex 1EHT enhanced tubes recover more energy and provide an opportunity to advance the design of many heat transfer products.

1. Introduction

Heat transfer surface enhancement is widely utilized in industrial applications and is one of the most effective passive techniques used in order to enhance heat transfer. Heat transfer enhancement techniques improve heat exchanger effectiveness. Enhancement is achieved by increasing turbulence, increasing surface area, producing more efficient fluid mixing and creating more efficient condensate drainage. Typical ways to accomplish this is through minor surface modifications that roughen the surface; use surface enhancements that modify the flow and generate secondary flows; and produce an increase to the surface area. Two-phase flow heat exchangers are commonly used in a variety of industries (HVAC, refrigeration, process, etc.); it is often desired to improve the heat transfer in those heat exchangers by utilizing enhanced tubes that can increase the heat transfer coefficient (HTC) and increase performance. Reay (2008) states that “between 1900 and 1955 the average rate of global energy use rose from about 1 TW to 2 TW. Between 1955 and 1999 energy use rose from 2 TW to about 12 TW, and to 2006 a further 16% growth in primary energy use was recorded world-wide.” Over the past ten years’ energy use continued its increase and the increasing trend in energy usage over the next decade is expected to continue. Reay (2008) notes “that there is a need to reduce CO_2 emissions by over 50 % in order to stabilise their impact on global warming. One way in which we can address this is by judicious use of process intensification technology.” He goes on to define process intensification as: ‘‘Any engineering

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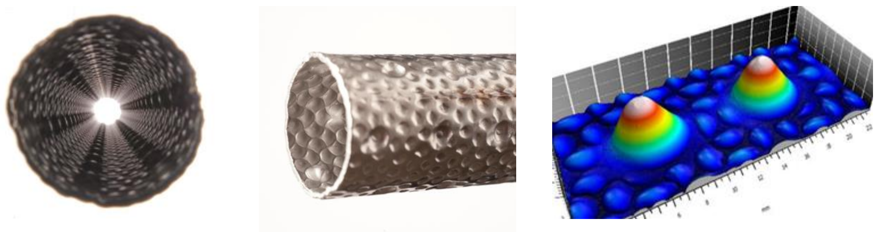
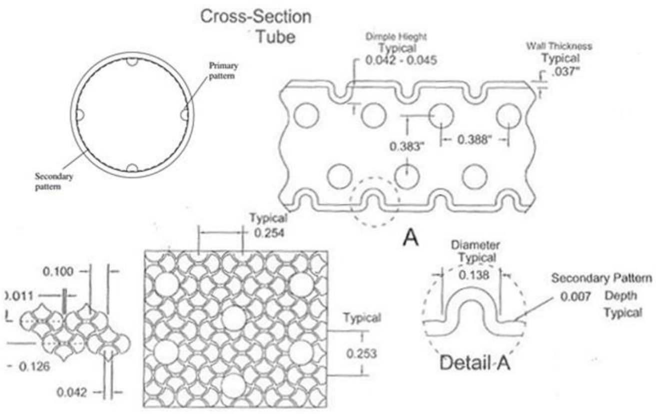
development that leads to a substantially smaller, cleaner, safer and more energy-efficient technology." He continues that it is most often characterised by a huge reduction in plant volume (orders of magnitude); in addition, process intensification is also an important technique in the reduction of greenhouse gas emissions.
Varbanov et al. (2018) discuss concepts and methods for improvement of the efficiency at all stages of energy sourcing, conversion and use. Xie et al. (2019) discusses methods to improve condensation on the condenser wall. Aroonrat and Wongwises (2019) investigated the effect of dimpled depth on the condensation heat transfer coefficient and pressure drop of R-134a flowing inside dimpled tubes. Pan et al. (2013) discuss the significant effect that heat transfer intensification of individual heat exchangers has on capital, maintenance, installation and operating costs. Li et al. (2017) experimentally studied the condensation heat transfer coefficient of R134a inside smooth and micro-fin tubes. Tube diameter was determined to have a small influence on the heat transfer coefficient for the smooth tubes; while the heat transfer coefficient increases with decreasing tube diameter for the micro-fin tubes. Sarmadian et al. (2017) measured and analysed the condensation heat transfer and frictional pressure drops of refrigerant R-600a (isobutane) inside a helically dimpled tube and a plain tube of internal diameter 8.3 mm. Fernardez et al. (2010) carried out an experimental investigation on R417A, R422 and R422D condensation on CuNi Turbo tubes. Fitzgrerald et al. (2012) presented condensation inundation on a low-finned tube and discuss the effects of vapour velocity on inundation angle. There have been numerous studies performed on shell side condensation, however there is no general model that can be used for the prediction of annular side condensation with enhanced tube surfaces since a different heat transfer mechanism takes place. There have been some specialized investigations that investigated the heat transfer performance of tubes with a specialized enhanced surface; however, it is necessary to carry out experimental research to determine the heat transfer characteristics for the newly developed enhanced dimpled tubes.
The types of tubes investigated in the present study include 1EHT, three-dimensional tubes that have been produced using multiple enhancement characters that are made of dimple arrays; details are shown in Figure 1 (a - d). The inner surface of the helically dimpled tube has been designed and reshaped through three-dimensional material surface modifications that consists of both shallow and deep protrusions which are placed evenly in helical directions on the tube wall. These enhanced surfaces are an enhanced hybrid surface that achieves its enhancement by using various dimples with different heights. Previous tube side enhancement studies of enhanced tubes demonstrate that the performance of enhanced tubes exceeds normal smooth tubes in heat transfer performance. However, few condensation heat transfer studies have been conducted on the outside of an enhanced tube and limited investigations have been performed in the changeable spaces for annular side flow condensation. There are no correlations that accurately predicts the performance of different size outer diameter (OD) three dimensional enhanced heat transfer tubes and experimental research is a more accurate representation of heat transfer performance.
The enhanced dimple tubes used in this study have been designed to maximize the heat transfer and to minimize the pressure drop that they produce. Previous research on the 1EHT tube reveals its extraordinary enhancement for heat transfer applications, providing the motivation for additional research on its heat transfer performance in other working conditions.

2. Experimental procedure

Condensation tests were conducted at 318 K saturation temperature; for a range of mass flux from 50 to 100 kg/ (m^2s); with heat flux in the range from 10 to 20 kW/ m^2; 0.8 inlet vapour quality and 0.2 outlet vapour quality. Evaporation tests were conducted at 279K saturation temperature; for a range of mass flux from 40 – 70 kg/ (m^2s); with heat flux in the range from 8 – 15 kW/ m^2; 0.1 inlet vapour quality and 0.6 / 0.8 outlet vapour quality. The schematic diagram of the test apparatus utilized to evaluate the flow boiling heat transfer characteristics of R134A and R410A inside circular tubes is shown in Figure 2 (a-c). A detailed description of the test apparatus and uncertainty is given by Li et al. (2012) and later by Wu et al. (2014).

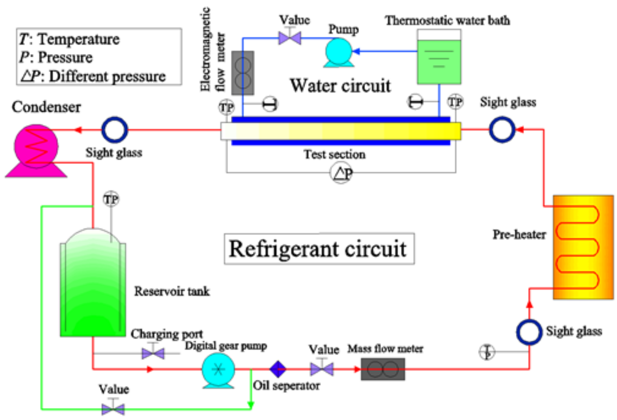


(a) (b) (c)

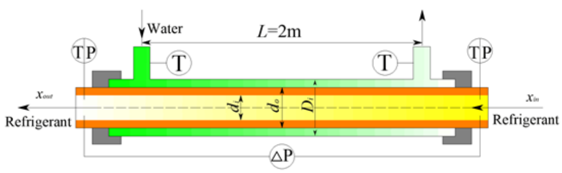


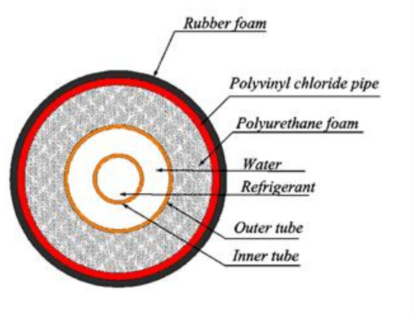
(d)

Figure 1: Surface enhancement of the 1EHT tube: (a) inner surface and (b) outer surface enhancement structure, (c) Non-Contact Profilometer three dimensional representation of the surface enhancement structure (d) details of the enhancement structure.



(a)





(b) (c)

Figure 2: Schematic diagram of the (a) experimental system, (b) test section, (c) cross section of the test section.

3. Results

Experimental condensation results as shown in Figure 3 for a fixed inlet/outlet vapour quality and fixed saturation temperature, while mass flux and heat flux varied; producing a heat transfer coefficient increase (when compared to smooth tubes) in the range of 54 to 64 %, with a frictional pressure drop value increase for the 1EHT tubes in the range of 8 to 31 %. Unexpected evaporation results were found in Figure 4; producing a decrease of the heat transfer coefficient for 1EHT tubes in the range of 100 – 220 %; with frictional pressure drop values increase for the 1EHT tubes in the range 5 to 17 %.

As is shown in Figure 4 the evaporation heat transfer coefficient outside the smooth tube and EHT tube is given; the HTC of the EHT tube is lower than that of smooth tube when the outlet vapour quality is up to 0.8. This is mainly because a larger dryout appears at the top of the outside tube at a higher vapour quality. The outside surface is not in contact with the liquid but directly exchanges heat transfer with the refrigerant vapour. Therefore, this results in heat transfer deterioration and a sharp decrease of the HTC. Moreover, the dryout vapour quality of the EHT tube decreases due to the rough enhanced surface (dimples and secondary petal array), the partial dryout phenomenon will occur earlier than that of the smooth tube, resulting in the HTC of the EHT tube being lower than that of a smooth tube for a high vapor quality. It is uncertain what will happen at other qualities, this is the subject of additional studies.

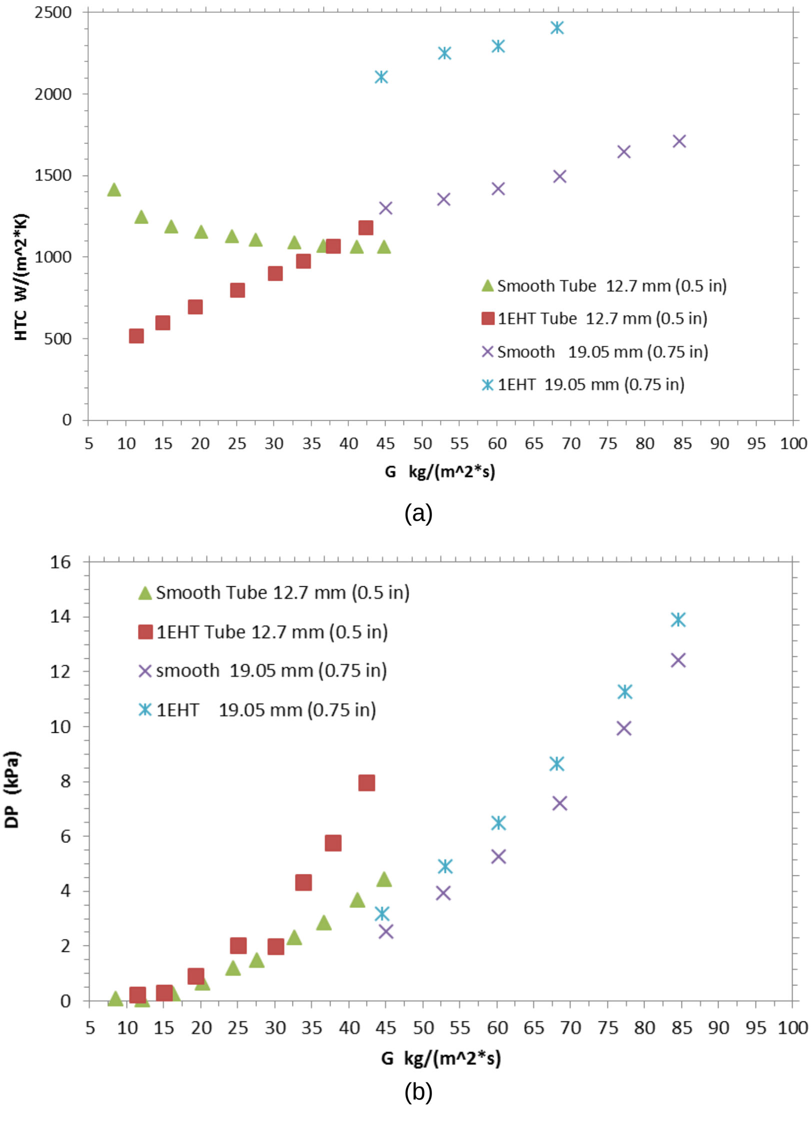
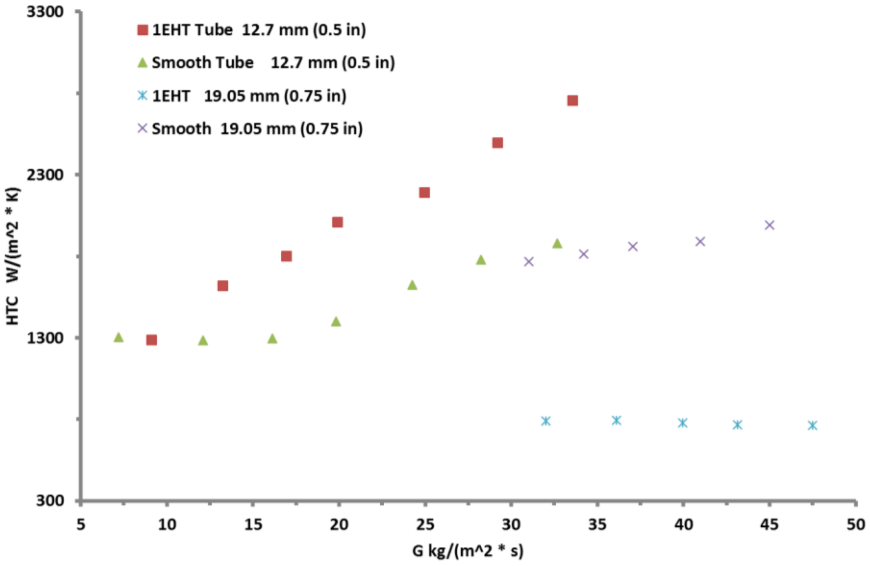
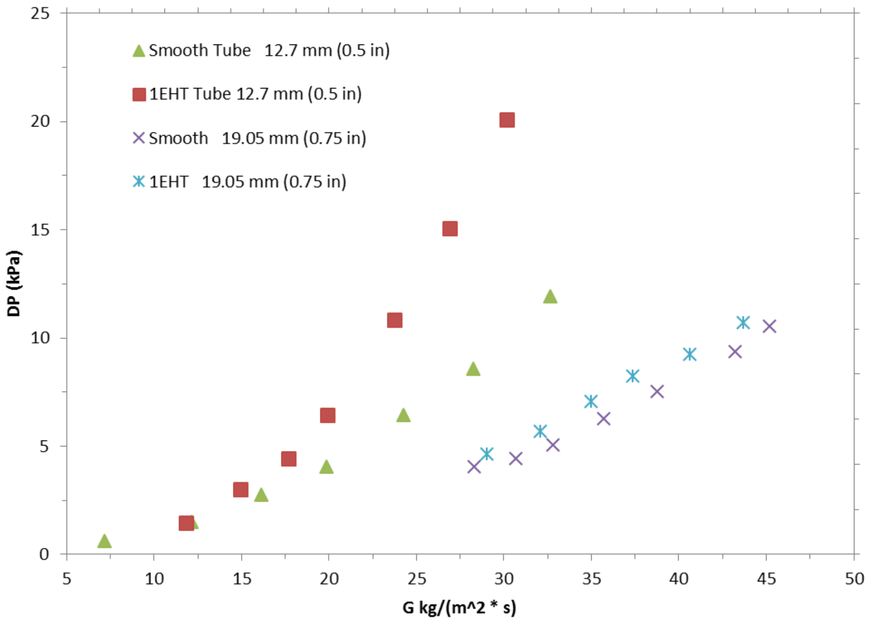


Figure 3: Shellside results for (a) condensation heat transfer coefficient (HTC). (b) pressure drop; as a function of mass flux (G) for the smooth and enhanced tubes of different diameters.



(a)



(b)

*Figure 4**:** Shellside results for (a) evaporation heat transfer coefficient (HTC). (b) pressure drop as a function of mass flux (G) for the smooth and enhanced tubes of different diameters.*

4. Conclusions

An experimental study was performed for evaporation and condensation on the outside tube surface. The effects of saturation temperature, mass flux and vapour quality on the heat transfer coefficient for smooth and enhanced tubes were investigated. These experiments were carried out for condensation tests were conducted at 318 K saturation temperature, 50 - 100 kg/ (m^2s) mass flux, 10 – 20 kW/ m^2 heat flux, 0.8 inlet vapour quality and 0.2 outlet vapour quality. Evaporation tests were performed at a saturation temperature of 279K, 40 – 70 kg/ (m^2s) mass flux, 8 – 15 kW/ m^2 heat flux, 0.1 inlet vapour quality and 0.6 / 0.8 outlet vapour quality.

In summary, there is an uncommon trend for the condensation heat transfer on the outside the smooth tube. As shown in the experimental results, the heat transfer coefficient decreases at first and then flattens out gradually over low mass flux range; then it continues to rise with increasing mass flux. This phenomenon can be explained by the relative significance of gravity, surface tension and inertia force. For the tested tubes, the impact of shear stress is strengthened as the saturated temperature decreases. Consequently, the heat transfer coefficient increases with the larger turbulence effect. The average vapour quality and mass flux has a positive effect on the condensation heat transfer coefficient. Experimental condensation results for a fixed inlet/outlet vapour quality and fixed saturation temperature, while mass flux and heat flux varied; producing an increase to the heat transfer coefficient in the range of 54 to 64 %, with a frictional pressure drop value increase for the 1EHT tubes in the range of 8 to 31 %. Finally, evaporation heat transfer coefficient for smooth tubes is larger than the 1EHT tubes by approximately 100 – 220 %; However, the frictional pressure drop values of 1EHT tubes is larger than the smooth tubes by approximately 5 to 17%. Additional studies at other conditions are currently being studied.

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