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Experimental Investigation to Evaluate the Evaporation and Condensation Heat Transfer Coefficients on the Outside of Vipertex Enhanced Performance 1EHT Tubes

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An experimental investigation was performed, for a wide range of mass flux values and qualities, to evaluate and compare the convective condensation and evaporation on the outside of a smooth tube and the newly developed Vipertex enhanced heat transfer 1EHT tube. Heat transfer enhancement has been an important factor in obtaining energy efficiency improvements in many two phase heat transfer applications. Utilization of enhanced heat transfer tubes is an effective method that is often utilized in the development of high performance thermal systems. Vipertex™ enhanced surfaces, have been designed and produced through material surface modifications which result in flow optimized heat transfer tubes that increase heat transfer. Surface enhancement of the 1EHT tube is accomplished by using a more pronounced primary enhancement (dimple characters) that has been produced over a secondary pattern made up of petal arrays. Heat transfer processes that involve phase-change processes are typically efficient modes of heat transfer; however current energy demands and the desire to increase efficiencies of systems have prompted the development of enhanced heat transfer surfaces that are used in processes involving evaporation and condensation. Vipertex™ was able to develop a series of optimized, three dimensional tubes that enhance heat transfer. This study details the condensation and evaporation results on the outer surface of the Vipertex 1EHT tube.
Results are presented here from an experimental investigation of two phase heat transfer that took place on the outside of a 12.7 mm (0.5 in) O.D. horizontal copper tube. Average evaporation heat transfer coefficients for the outside of the Vipertex 1EHT tube are approximately one to four times greater than those of a smooth tube. However for condensation, the heat transfer is approximately 23 % ~ 65 % of the heat transfer coefficient found for the smooth tube. In both cases the pressure drop increases.

1. Introduction

During the last thirty years a good deal of effort has been devoted to studying methods to enhance the heat transfer and hydraulic performance of heat exchangers. These studies included enhancement of evaporators and condensers that are used in various air-conditioning and refrigeration applications. Developing new kinds of enhanced micro-fin tubes such as helix, cross-grooved or herringbone is an effective approach utilized to enhance the performance of heat exchangers. Compared to a smooth tube, the micro-fin tube can substantially enhance heat transfer coefficients with only a small pressure drop penalty; using surface enhancement tubes in heat exchangers produces more compact and efficient designs.
There have been previous studies that have investigated the inside heat transfer enhancement using micro-fin tubes. Miyara et al. (2000) presents an early experimental study using R410A with herringbone tubes. Wellsandt and Vamling (2005a) carried out an in-tube evaporation investigation of R407C and R410A using a 4 m long micro-fin herringbone tube (outer diameter of 9.35 mm) and compared the measured heat transfer and pressure drop data to predicted results from previously reported correlations.

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Wellsandt and Vamling (2005a) evaluated R134a under the same conditions. Bandarra et al. (2006) studied convective boiling heat transfer and pressure drop of R134a in smooth, standard micro-fin and herringbone copper tubes with a 9.52 mm external diameter. They found that the thermal performance of the herringbone tube is better than that of a standard micro-fin tube for a large range of mass velocities; at the lowest velocities they found the worst performance for the herringbone tube, for qualities larger than 50 %. Additionally they found the herringbone tube to have the highest pressure drop over the entire range of mass velocities and qualities. Afroz et al. (2007) studied the pressure drop of single-phase, turbulent flow inside herringbone micro-fin tubes with different fin dimensions. They developed a general correlation for the single-phase friction factor of a herringbone tube that predicts the experimental data within ±10 %.
Condensation heat transfer on the outside of horizontal tubes plays an important factor in refrigeration, air conditioning and heat pump applications. Heat transfer enhancement is an important consideration for design in order to improve energy efficiency and protect the environment. Currently, enhanced heat transfer tubes are the primary means to raise a system's heat efficiency. The surface structure of enhanced tubes typically is micro-fins, which increases flow turbulence and creates a thin film thickness. Heat transfer enhancement mechanisms of micro-fin tubes have been investigated widely. Cavallini et al. (2009) presented a new simple model to predict the condensation heat transfer coefficient in micro-fin tubes. Doretti et al. (2013) made a review about the condensation flow patterns inside micro-fin tubes and smooth tubes. Miyara et al. (2000) have discussed the flow patterns and enhancement mechanisms in herringbone tubes. Afroz et al. (2007) proposed a generalized prediction correlation of condensation pressure drop inside herringbone tubes. Olivier et al. (2007) performed an experimental study to compare the heat transfer characteristics of a smooth tube, micro-fin tube and herringbone tube.

Figure 1: Surface enhancement structure of the 1EHT tube

The 1EHT enhanced heat transfer tube developed by Vipertex, is a novel kind of tube that was developed by modifying surface geometries (i.e. creating a modified surface that is a combination of larger dimples and smaller petals) which can enhance the heat transfer coefficient on both the inside and outside surface of the tube; its details are shown in Figure 1. The 1EHT enhanced heat transfer tube is neither a classic ''integral roughness'' (little surface area increase) tube, nor an internally finned tube (surface area increase with no flow separation). It can be considered to be more of a hybrid surface that increases surface area and produces flow separation from the dimpled protrusions on the tube. Enhancement of the heat transfer using the 1EHT tube is produced from a combination of increased turbulence, disruption of the boundary layer, secondary flow generation, increased heat transfer surface area and a large number nucleation sites; all leading to an enhanced heat transfer performance for a wide range of conditions. So far little experimental work has been published on this kind of tube. In this study the convective evaporation and condensation heat transfer characteristics of R410A and R22 refrigerant on the outside of a smooth tube and Vipertex 1EHT tube (both tubes with an external diameter of 12.7 mm) was performed at low mass fluxes. As indicated in Wu et al. (2013), helical micro-fin tubes are more effective at low mass fluxes due to the strong surface tension effects that are seen at low mass fluxes. Therefore, the focus of our experimental investigation will be on low mass fluxes.

2. Experimental Set Up

The experimental setup consists of two closed loops: a refrigerant loop which contains the test section and a water circuit which can cool or heat the test section. The refrigerant flow loop consists of a 50 L reservoir tank, digital gear pump with a variable speed motor, pressure regulating valve, mass flow meter, pre-heater, test section, condenser, and sight glasses. A by-pass line from the pump to the reservoir and the regulating valve is used to control the mass flow rate through the refrigerant loop. In order to measure the refrigerant flow rate, a Coriolis Effect mass flow meter (with an accuracy of ±0.2 % of the reading), located between the pump and the pre-heater, is used. The sub-cooled liquid is electrically heated in the pre-heater and produces the required fluid inlet quality at the pre-heater outlet; the fluid then enters the test section where it will be condensed or evaporated. Finally, the two-phase refrigerant is totally condensed and sub-cooled in a 9 kW alcohol-water mixture, low-temperature 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 is utilized in order to indicate the thermodynamic state of the fluid. The water circuit includes a water thermostat, centrifugal pump, control valve, and magnetic flow meter. The magnetic flow meter (with an accuracy of ±0.35 % of reading) is used to determine the flow rates of the water in the annulus of the test section.

The test section includes a straight, 2 m long horizontal test section that includes a counter-flow, double-tube heat exchanger. The refrigerant flows in the annulus, on the outside of the evaporation tube; in order to heat the evaporation tube, water flows inside the test tube. It is insulated using 40 mm thick foam insulation and 6 mm thick rubber insulation that creates a tight fitted. In order to minimize heat losses, the entire test facility is well insulated (especially the pre-heater and the test section). Calculated uncertainities are given in Table 1. Uncertainty of the Platinum RTDs is ±0.1 °C; uncertainty of the differential pressure transducers is ±0.05 % of the set span; and the flow meter uncertainty is ±1.0 % of the flow rate. Single-phase heat-transfer tests have been performed in order to verify the uncertainty in the heat transfer measurement. From the single-phase evaluation of R410A or R22 on the outside of the smooth tube, a heat balance of the test section was performed using the heat exchanged on the water side and the refrigerant side. Figure 2 shows that the test section heat balance can be controlled to ±5 %; this indicates that the heat losses in the experimental apparatus can be ignored. It was determined that the heat transfer average deviation between the water side and refrigerant side is 3.1 %. Therefore, the uncertainty of condensation heat transfer coefficient is ± 5.1 % (Wu et al., 2013).

Table 1: Uncertainty of the primary measurements and dependent quantities

Primary measurements		Dependent quantities	
Diameter	±0.05 mm	Mass flux G, kg m^{-2} s^{-1}	±2.2 %
Length	±0.2 mm	Heat flux q, W m^{-2}	±4.3 %
Temperature	±0.07 K	Vapor quality x	±5.2 %
Electric current	±0.01 A	Frictional pressure drop ΔP_f	±3.9 %
Electric voltage	±1.0 V	Heat transfer coefficient h, W m^{-2} K^{-1}	±15.4 %
Pressure, range: 0-40 bar	±0.2 % of full scale		
Differential Pressure, range: 0-100 kPa	±0.05 % of reading		
Water flow rate, range: 0-12 L min^{-1}	±0.35 % of reading		
Refrigerant flow rate, range: 0-60 kg h^{-1}	±0.2 % of reading		

Dimensions of the smooth tube include an inner diameter of 11.5 mm and an external diameter of 12.7 mm. The 1EHT tube has a primary enhancement dimple over a secondary petal array enhancement; with a nominal inner tube diameter of 11.5 mm and external diameter of 12.7 mm. Parameters of the 1EHT tube are shown in Figure 1 (scale unit in the figure is inch) and includes: (i) primary dimple with a typical diameter of 3.505 mm and a dimple height of 1.067–1.143 mm. (ii) secondary petal-shape pattern with a typical diameter of 2.54 mm and a typical height of 0.178 mm. Inner surface heat-transfer area enhancement of the 1EHT tube enhancement is 1.112 times that of the smooth tube.

Test apparatus, degassing process, measurement methods, data reduction process and uncertainty propagation methods that were adopted in this study followed the process used in Wu et al. (2013). The test apparatus was verified in Wu et al. (2013) for single-phase flow and two-phase flow in smooth tubes. Tubeside condensation and evaporation characteristics of this tube were previously studied in Kukulka et al. (2014). Fouling of the 1EHT tubes were performed by Kukulka et al. (2011).

Evaporation evaluation of R410A on the outside the three tubes were conducted at a saturation temperature of 279 K; for a mass flow range from 10 to 40 kg/h (i.e. mass flux 8-35 kg m^{-2} s^{-1}); 0.1 inlet quality; and an outlet quality of 0.8. Therefore the average vapor quality for all evaporation tests is approximately 0.45.

3. Results

Figure 3a compares the variation of the condensation heat transfer coefficient (HTC) with mass flux for R410A on the outside surface of a smooth tube and the 1EHT tube. In a comparison of the tubes, it was unexpectedly found that the smooth tube had a better heat transfer coefficient (HTC) than an enhanced surface tube. This being due to the retention of condensate on the tube surface causing a reduction in

HTC found for the 1EHT tube; for the range of mass flux in this experiment the 1EHT tube HTC is approximately 23 % ~ 65 % of the HTC found for the smooth tube. As can be seen in Figure 3(a), the HTC of a smooth tube decreases at first and flattens out gradually over the range of mass flux values; while the HTC of the 1EHT tube increases with increasing mass flux. Figure 3(b) presents the relationship between condensation pressure loss and mass flux.

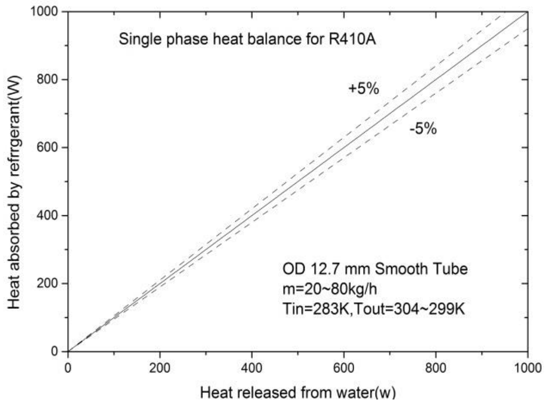
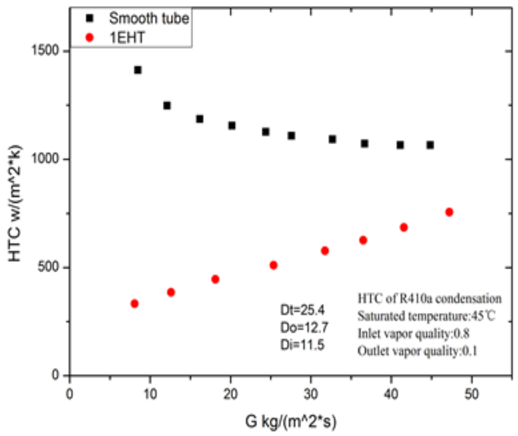


Figure 2: Single-phase Heat Balance

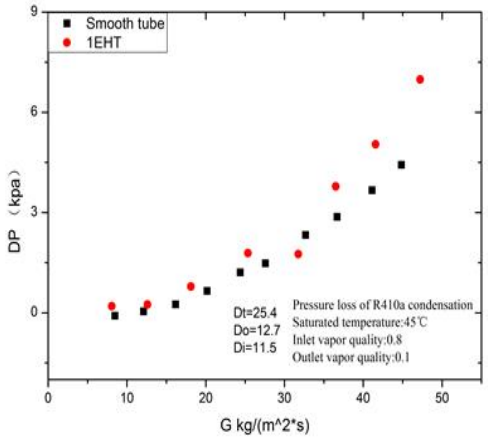
Pressure drop increases with increasing mass flux; both tubes have a similar pressure drop, with the smooth tube showing the lowest pressure loss values. It appears that since the heat transfer coefficient of 1EHT tube is lower than that of a smooth tube that perhaps the liquid drainage is worse for a rough surface for low mass flux values. A further study is needed to confirm the actual cause of this unexpected phenomenon. Figure 3(c) shows the variation of the HTC with flow flux for R22. Once again, the HTC of the smooth tube decreases with increasing flow rate; with the HTC on the outside of the smooth tube being approximately three times the HTC that is found on the outside of the 1EHT tube. In Figure 3(d) it can be seen that the pressure drop for the 1EHT tube (at the flow rate for the minimum HTC difference) is approximately 1.35 times that of the smooth tube. Figure 3(e) shows the evaporation heat transfer coefficient trend of refrigerant R410A on the outside surface of a smooth tube and the 1EHT tube. For both tubes, the evaporation heat transfer coefficient increases with increasing mass flow rate. The evaporation heat transfer coefficient of the 1EHT tube is larger than that of smooth tube; showing an increase in the heat transfer coefficient ratio that ranges from 1.0 to 1.4 (depending on the mass flux); the increase in the heat transfer coefficient is larger than the increase in the 1EHT heat transfer surface area (ratio 1.112). In Figure 3(f) it can be seen that the pressure drop for the 1EHT tube (at the flow rate for the maximum HTC difference) is 1.9 times that of the smooth tube. Figure 3(g) shows the data of the evaporation HTC of R22 on the outside of both the 1EHT tube and the smooth tube. For R22, the evaporation HTC of the 1EHT tube is approximately 4.4 times of HTC in a smooth tube; Figure 3(h) shows that when using the 1EHT tube, the pressure drop increases approximately by a factor of 1.8 to 1.9. An increase in pressure drop slightly increases the cost of operation; additionally the initial cost of the 1EHT tube is approximately 30 % more than a smooth tube. Heat performance of the smooth tube decreases as flow flux increases; however the performance of the 1EHT tube increases with increasing flow flux. Heat transfer increases, for most conditions, are much greater than additional operational costs and initial material cost.

4. Conclusions

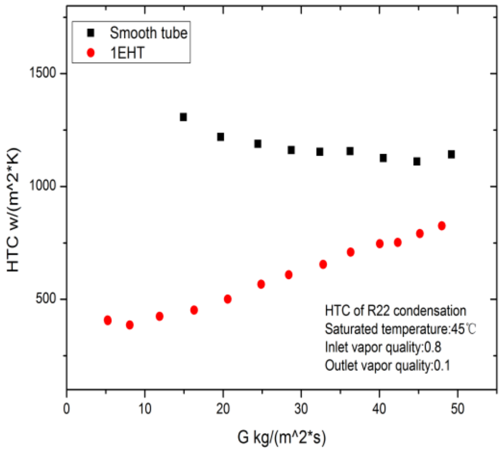
Condensing heat transfer of R410a and R22 on the outside surface of a smooth tube and the 1EHT tube has been experimentally measured over a range of mass flux values ranging from 5 to 50 kg/m^2 s, for specific qualities. In summary the major findings are: (i) Condensation heat transfer coefficient of a smooth tube is higher than that of 1EHT tube for the evaluated conditions. (ii) Condensation heat transfer coefficient of the 1EHT tube increases over the mass flux range; while condensation heat transfer coefficient of the smooth tube decreases over the same range of mass flux. (iii) There is little difference in condensation pressure loss for both the tubes over the range of mass flux evaluated. Studies are needed for a higher range of pressures to determine if these trends continue. Additional studies are also required using other refrigerants and other size tubes. An experimental evaluation of evaporation on the outside


Smooth tube
1EHT
HTC w/(m^2*k)
G kg/(m^2*s)
Dt=25.4
Do=12.7
Di=11.5
HTC of R410a condensation
Saturated temperature:45℃
Inlet vapor quality:0.8
Outlet vapor quality:0.1

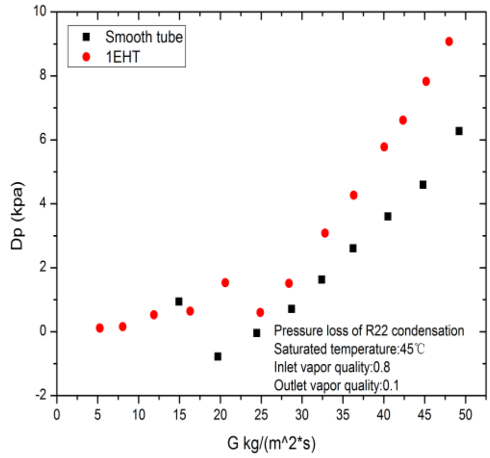
(a)


Smooth tube
1EHT
DP （kpa）
G kg/(m^2*s)
Dt=25.4
Do=12.7
Di=11.5
Pressure loss of R410a condensation
Saturated temperature:45℃
Inlet vapor quality:0.8
Outlet vapor quality:0.1

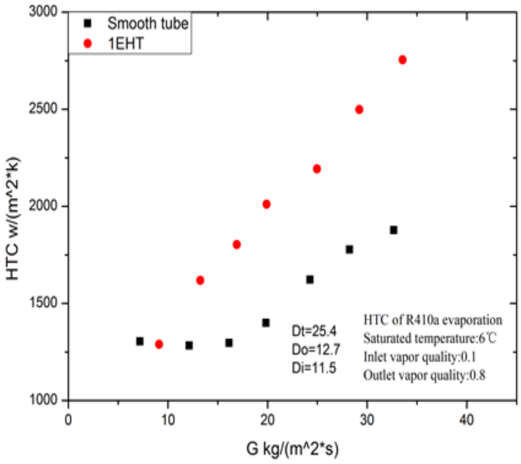
(b)


Smooth tube
1EHT
HTC w/(m^2*K)
G kg/(m^2*s)
HTC of R22 condensation
Saturated temperature:45℃
Inlet vapor quality:0.8
Outlet vapor quality:0.1

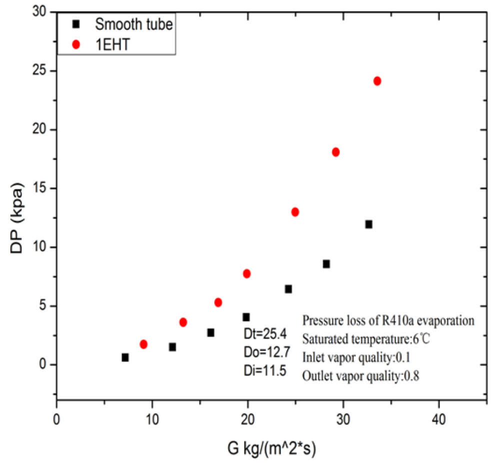
(c)


Smooth tube
1EHT
Dp (kpa)
G kg/(m^2*s)
Pressure loss of R22 condensation
Saturated temperature:45℃
Inlet vapor quality:0.8
Outlet vapor quality:0.1

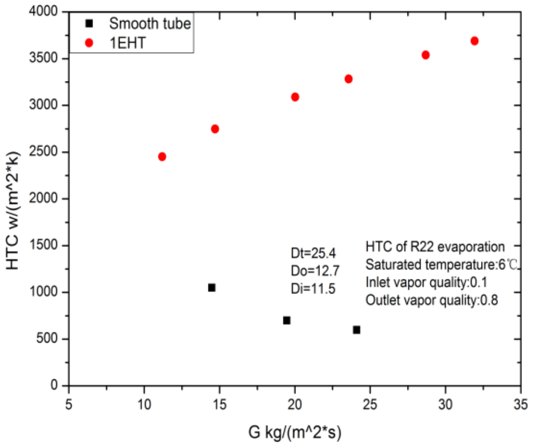
(d)


Smooth tube
1EHT
HTC w/(m^2*k)
G kg/(m^2*s)
Dt=25.4
Do=12.7
Di=11.5
HTC of R410a evaporation
Saturated temperature:6℃
Inlet vapor quality:0.1
Outlet vapor quality:0.8

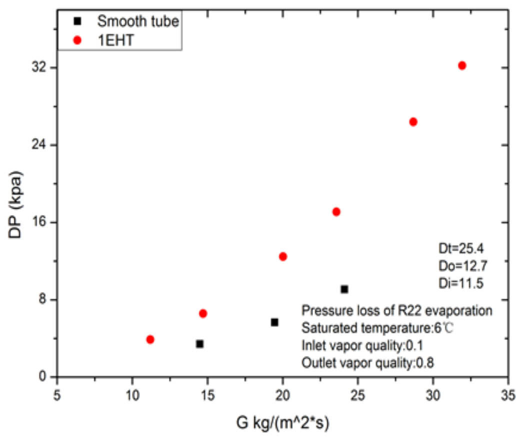
(e)


Smooth tube
1EHT
DP (kpa)
G kg/(m^2*s)
Dt=25.4
Do=12.7
Di=11.5
Pressure loss of R410a evaporation
Saturated temperature:6℃
Inlet vapor quality:0.1
Outlet vapor quality:0.8

(f)



(g)



(h)

Figure 3: (a) Condensation HTC of R410a. (b) Condensation pressure drop of R410a. (c) Condensation HTC of R22.(d) Condensation pressure drop of R22. (e) Evaporation HTC of R410a. (f) Evaporation pressure drop of R410a.(g) Evaporation HTC of R22. (h) Evaporation pressure drop of R22.

surface of a smooth and 1EHT enhanced surface tube has been conducted in this study. The experimental data was obtained at the evaporation temperature of 6 °C; for a refrigerant mass flux range from 8 to 35 kg m^{-2} s^{-1}; inlet quality of 0.1 and outlet quality of 0.8. The objective of the study was to determine how the outside evaporation heat transfer coefficient varies with mass flow for each of the tubes. Evaporation HTC of the 1EHT tube was found to be approximately four times the HTC of a smooth tube. Pressure drop increases with the mass flux for both of the tubes. The current study was conducted at a low mass flow; additional work is necessary at higher mass flows and for a wider variety of tube diameters.

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