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Condensation and Evaporation Characteristics of Vipertex 1EHT Enhanced Heat Transfer Tubes

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Heat transfer enhancement has been an important factor in obtaining energy efficiency improvements in refrigeration and air-conditioning applications. Utilization of enhanced heat transfer tubes is an effective method to be 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. Heat transfer enhancement plays an important role in improving energy efficiency and developing high performance thermal systems. 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 heat transfer and fluid flow results of the Vipertex 1EHT, enhanced heat transfer tube over a range of conditions that involved in-tube evaporation and condensation.
Results are presented here from an experimental investigation of two phase heat transfer that took place in a 12.7 mm (0.5 in) O.D. horizontal copper tube. 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. In-tube evaporation measurements of R22 and R410A are reported for evaporation at 10 °C with mass flow rates in the range of 15 to 40 kg/h. Single phase measurements are reported for mass flow rates from 15 kg/h to 80 kg/h. Condensation tests were conducted at a 47 °C saturation temperature, with an inlet quality of 0.8 and an outlet quality of 0.1. In a comparison to smooth tubes, the local and average heat transfer coefficients for the Vipertex 1EHT tube exceeded those of a smooth tube. Average evaporation and condensation heat transfer coefficients for R22 and R410A in the Vipertex 1EHT tube are approximately two times greater than those of a smooth tube.
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 enhancement has been an important factor in obtaining energy efficiency improvements in refrigeration and air-conditioning applications. Enhanced inner surfaces, are routinely used because they can produce enhanced tubeside heat transfer coefficients with a small pressure drop penalty. The 1EHT tube geometries are neither a classic ‘‘integral roughness’’ (little surface area increase) tube, nor an internally finned tube with a surface area increase and no flow separation. A 1EHT surface is more of a hybrid surface with a surface area increase and flow separation produced by the primary dimple enhancement. This enhanced heat transfer surface creates a combination of: increased turbulence; disruption of the boundary layer; secondary flow generation; increased heat transfer surface area; and a high density of nucleation sites.

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These factors lead to an increase in the heat transfer coefficient for a wide range of conditions; smaller unit footprint; more economic operation costs and a prolonged product life. This is a study of Vipertex 1EHT enhanced tubes that have been evaluated for a wide range of condensation and evaporation conditions. Heat transfer enhancement using the Vipertex EHT series of tubes provides a means to significantly advance the design of many heating and cooling processes.

There have been a source of previous studies that have evaluated different aspects of the current study. Gee and Webb (1980) studied the effect of enhanced tube geometry factors. Liu and Jensen (2001) demonstrated the effect of geometry on heat transfer for enhanced tubes. Christians et al. (2010 a,b) studied film condensation of refrigerants on enhanced tubes. Li et al. (2007) presented a study of micro fin tubes for a wide range of Prandtl Numbers (Pr < 220) and for a range of Reynolds Numbers from 2,500 to 90,000. Zhang and Yuan (2008) presents the development of prediction methods for evaporation heat transfer of refrigerant mixtures in microfin tubes. Boissieux et al. (2000) compares condensation results for three refrigerants and makes a comparison with two previously developed condensation correlations in order to evaluate their validity for use with various refrigerant mixtures. Wu et. al (2013) presented an experimental investigation on the convective vaporization of R22 and R410A inside a smooth tube and compared the performance of five micro-fin tubes for mass fluxes ranging from 100 to 620 kg/m^2s at a saturation temperature of 279 K. Cavallini et al. (2003) presents a comprehensive review of condensation heat transfer inside smooth tubes in different flow regimes, comparing available experimental data for new refrigerants with previously reported prediction models for heat transfer. Zhang et al. (2003) compared evaporation heat transfer coefficients for R417A in internally grooved tubes with a smooth tube. Cavallini et al. (2000) presents a critical review of the correlations necessary to compute heat transfer coefficients and pressure drops for refrigerants condensing inside several commercially available enhanced surface tubes Thome et al. (2003) proposed simplified flow structures in several flow regimes. Zhang et al. (2003) presents results from an experimental study on evaporation heat transfer of R417A and when compared to R22, the evaporation heat transfer coefficients for R417A were lower; additional results from that study found that results for internally grooved tubes were lower than the results of smooth tubes. Kukulka et al. (2011a) evaluated surface geometry of enhanced tubes and that study of enhanced surfaces laid the groundwork for the present study. Kukulka et al. (2011b) evaluated the effect of fouling on the heat transfer of enhanced surfaces.

No previously reported data exists for the enhanced geometry considered in this study therefore a comparison of 1EHT tube data will be made to smooth tube experimental results and previously reported smooth tube heat transfer results from Gnielinski (1976); for single-phase flows in the range $0.5 \le Pr \le 10^6$ and $2{,}300 \le Re \le 5\times10^6$.

2. Experiment

An experimental investigation was performed in order to determine the evaporation heat transfer coefficient (HTC), condensation HTC and pressure drop of a Vipertex 1EHT tube (12.7 mm OD tube) with R22 and R410A refrigerants. Test apparatus is shown in Figure 1 and consists of two closed circuits; a refrigerant circuit which contains the test section; and a water circuit which transfers heat from the test section. Components of the refrigerant loop are the test section, a condenser, a reservoir, a digital gear pump, a mass flow meter, a pre-heater, sight glasses and valves. The test section is a straight, horizontal, counterflow tube-in-tube heat exchanger with a length of 2 m. The refrigerant inlet and exit temperatures were measured using calibrated Platinum RTDs, with an accuracy of ± 0.1 °C. In addition, the saturation pressure was measured using a pressure transducer at the inlet of the test section. Pressure drop across the test section is measured using a differential pressure transducer. Average heat transfer data and pressure drop measurements are taken over the entire length of the test section. The refrigerant exits the test section and goes to a low temperature water bath, where the fluid is condensed and subcooled. The liquid refrigerant is then recirculated through the system by a gear pump. A mass flow meter is located between the pump and the pre-heater to measure the refrigerant flow rates.

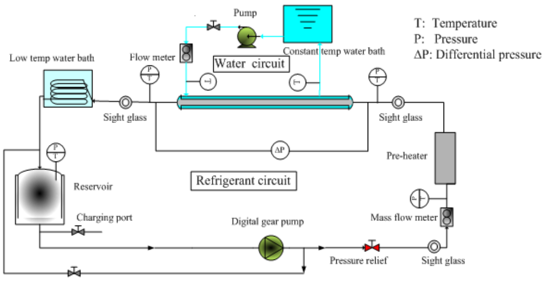


Figure 1: Schematic of Experimental Apparatus

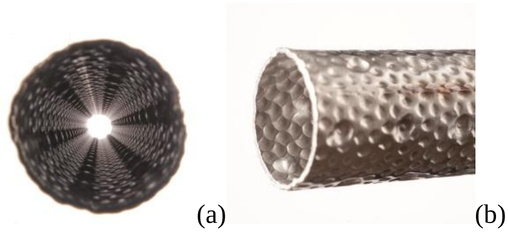


Figure 2: (a) Cross Sectional View (b) Outer Surface of the 1EHT Enhanced Tube

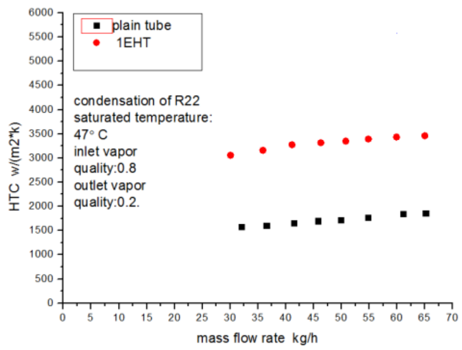


Figure 3: Single-phase HTC of R410a

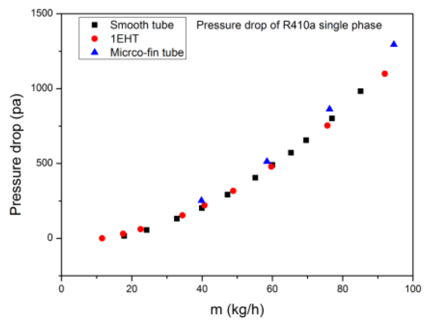


Figure 4: Single-phase Pressure drop of R410a

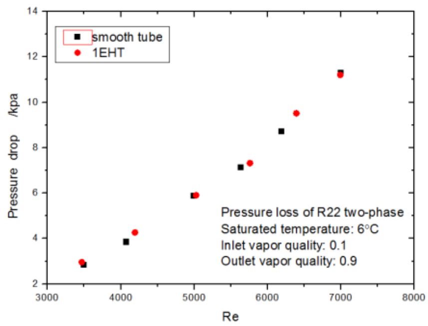


Figure 5: Pressure drop of R22 in two-phase flow

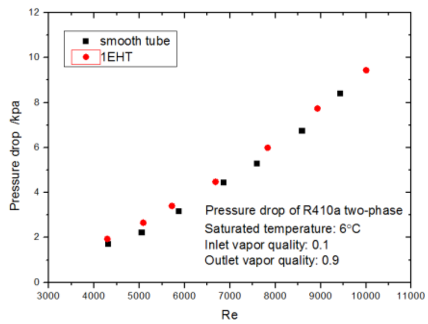
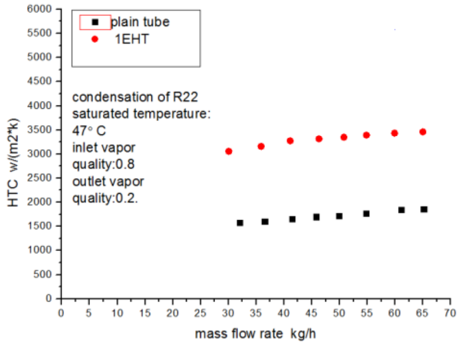


Figure 6: Pressure drop of R410a in two-phase flow



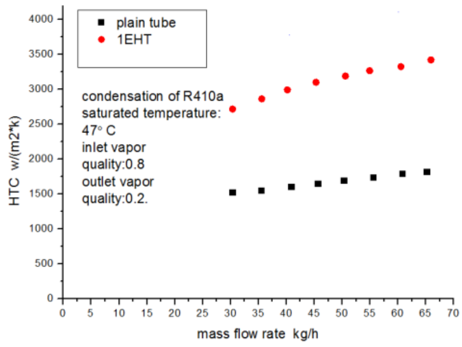
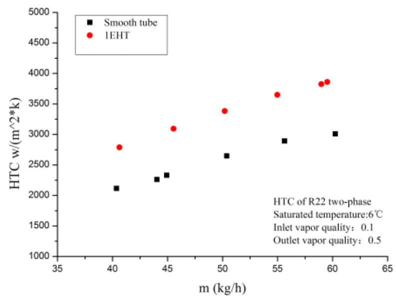


Figure 7: Condensation HTC of R22 *Figure 8: Condensation HTC of R410a*



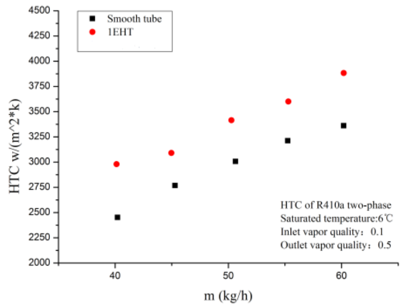


Figure 9: Evaporation HTC of R22 *Figure 10: Evaporation HTC of R410a*

Subcooled liquid enters the pre-heater; then the heated two-phase flow with the specified inlet quality exits the pre-heater outlet. The thermodynamic state of the fluid is obtained by using a platinum RTD and a pressure transducer at the inlet of the pre-heater; this is used in order to obtain the enthalpy. Inlet quality of the refrigerant is obtained from the heat input and the temperature at the pre-heater outlet (or inlet of the test section). The water loop consists of an annulus, a centrifugal pump, a magnetic flow meter, and a constant temperature water bath. Water flows in the annulus side are used to condense (vaporize) the refrigerant. Inlet and exit water temperatures were measured using platinum RTDs and then determine the heat flux. Flow rate is controlled by a valve located after the centrifugal pump. For a fixed inlet quality and fixed quality change, the heat flux increases with mass flux.

Vaporization evaluation in the 1EHT tube (using R22 and R410A) were conducted at a saturation temperature of 279 K, with an inlet quality of 0.1 and an outlet quality of 0.8. Condensation evaluation was conducted at a saturation temperature of 47 ^{o}C, with an inlet quality of 0.8 and an outlet quality of 0.1. Figure 2 provides views of the enhancements on the inner and outer surfaces of the 0.50 in (12.7 mm) outer diameter, Vipertex 1EHT enhanced copper tube that was evaluated in this study.

3. Results

An experimental investigation was performed to determine the evaporation and condensation heat transfer coefficient (HTC) of R22 and R410A inside the Vipertex 1EHT tube (0.5 in OD tube). Condensation tests are performed using R22 and R410a refrigerants in the 1EHT tube; with mass flow rate ranging from 30 to 60 kg/h; for a saturated temperature of 47 ± 0.3°C, with an inlet vapor quality of 0.8 ± 0.03 and outlet vapor quality of 0.2 ± 0.03. Evaporation heat transfer is measured in the 1EHT tube for a saturated temperature of 6 ± 0.03 °C and for a mass flow rate in the range of 25 to 45 kg/h. Results from Figure 3 demonstrate that the single phase HTC of the 1EHT tube is greater than experimental smooth tube results by a factor of approximately 2.2. Additionally smooth tube data fits well with the theoretical values of Gnielinski (1976). Figure 4 compares the pressure drop in the 1EHT tube and smooth tube for R410a single phase flow. Pressure drop in the 1EHT tube increases approximately 8 % when compared to a plain tube. Figure 5 shows a small variation of two phase pressure loss in a comparison of a smooth tube and the 1EHT tube in R22. Figure 6 presents a pressure drop comparison in R410a two-phase flow is presented for a smooth tube and the 1EHT tube. Two phase pressure drop does not vary much in the 1EHT tube when compared to a smooth tube. Since the pressure drop of the 1EHT tube doesn't vary much from that of the smooth tubes for single phase and two-phase flow, it is expected that pressure drop will not vary for the condensation and evaporation flows conducted here.

Figure 7 shows a comparison of the condensation HTC in R22 for the 1EHT tube and a smooth tube. The 1EHT tube provides approximately two times the heat transfer measured in a plain tube. Figure 8 compares the condensation HTC of the 1EHT tube in R410a with the results of a smooth tube. Performance of the 1EHT tube is once again approximately two times that of a smooth tube. As can be seen from Figures 3 - 8 the overall performance of the Vipertex 1EHT tube is much better than a smooth tube.

The evaporation HTC performance of the 1EHT tube is compared to a smooth tube in Figure 9 for R22 and in Figure 10 for R410a. An enhancement multiplier for the evaporation HTC using the 1EHT tube varies between 1.3 and 1.4. The slight variation in multiplier values are due to the property differences of R410a and R22.

4. Conclusions

Evaporation and condensation HTC evaluations for both smooth tubes and in the surface enhanced 1EHT tube have been conducted using both R22 and R410a refrigerants. The 1EHT tube has much better performance than a plain tube, with an enhanced ratio of more than two being shown for single phase, evaporation or condensation. If pressure drop and heat transfer performance is considered in the evaluation of these tubes, a superior performance is demonstrated in the 1EHT tube. The heat transfer coefficient of R22 is less than that of R410A at relatively large mass fluxes since R410A has a larger thermal conductivity. At the same mass flux and quality change, the imposed heat flux of R410A is larger than that of R22 because of its larger latent heat of vaporization. The heat flux needed for the onset of nucleate boiling is lower than that of R22 due to its larger latent heat of vaporization and lower surface tension. Thus, the nucleate boiling component of R410A is larger than R22 in our experimental conditions. Similar to R22 evaporation, the heat transfer coefficient increases as the mass flux increases. This phenomenon may be explained by comparing the fin height (dimple height in the 1EHT tube) and the thickness of the liquid film between the vapor core and the wall. If fin height is far less than the thickness of the liquid film, the fins will be immersed in the liquid. Therefore, the liquid film becomes a thermal resistance and interfacial turbulence effects can be reduced. However, if fin height is far greater than thickness of the liquid film, the increase of heat transfer area due to the increments of the fin height and number of starts is relatively inefficient in heat-transfer enhancement. When the fin height and liquid film height ratio is close to unity, the fin tips will be covered by a very thin liquid film and periodic waves induced by the fins also makes the liquid–vapor interface unstable, which maximizes thermal efficiency and interfacial turbulence, enhancing heat transfer greatly. On the liquid film both vapor shear and surface tension effects occur. The interfacial turbulence induced by fin (dimple) stirs up the liquid–vapor interface, and some fraction of liquid is entrained in the vapor core. Assuming no liquid entrainment, the average film thickness in intermittent and annular flow can be calculated. The heat transfer coefficient of R410A evaporation inside micro-fin tubes is about 1.3 - 1.4 times that of the plain tube.

All this leads to an important and exciting advancement in process design. The patented Vipertex surface enhances heat transfer, conserves energy and minimizes cost. Further studies of new Vipertex surfaces are currently under way.

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