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Development and Evaluation of Enhanced Heat Transfer Tubes for Transitional Reynolds Number Flow

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Heat transfer enhancement plays an important role in improving energy efficiency and developing high performance thermal systems. A wide variety of industrial processes involve the transfer of heat energy and many of those processes employ old technology. These processes would be ideal candidates for a redesign that could achieve improved process performance. Increasing efficiency in process plant operations is always a priority with engineers constantly looking for new ways to reduce energy requirements in process plants. Additionally, there is pressure from the government to reduce energy usage to meet economic and environmental goals. Utilization of an enhanced heat transfer tube is an effective method to be utilized in the development of high performance thermal systems.
In many areas of the world the availability of process water is scarce and the lack of abundant cooling water volume causes major problems in process design. Extreme water availability risks exist across the Middle East and North Africa. Many countries in this region have a growing population and ambitious economic development plans, creating additional demands on water. Of particular importance to the global and regional economy is the use of large quantities of water in the production of oil and chemical products. Water scarcity could also lead to further increases in global oil prices and heightened political tensions to protect water supplies in the future. Use of enhanced heat transfer tubes to decrease process water requirements while at the same time provide higher levels of heat transfer in energy conversion processes are important design considerations. These were some of the goals that were considered when the Vipertex™ EHT series of enhanced tubes were developed.
Enhanced heat transfer tubes must be considered in the design of high efficiency heat exchangers. Their use will allow operations to decrease the required cooling water mass flow rate in order to obtain the required heat transfer rate; allowing the heat exchangers to operate in the transitional flow regime, at flowrates not previously considered with current designs, will save both energy and water. Transition from laminar to turbulent flow for smooth tubes typically is assumed to occur for a Reynolds Number of 2300. In reality, a transition point is not as well defined and for some process conditions actually could occur over a wider range of Reynolds Numbers, typically varying between 2300 and 10,000. Vipertex™ enhanced tubes allow transition to occur earlier than 2300, providing increased heat transfer while at the same time using a smaller volume of cooling fluid.
Vipertex™ enhanced surfaces, have been designed and produced through material surface modifications, which result in flow optimized heat transfer tubes that increases heat transfer through a combination of factors that include: increasing fluid turbulence, secondary flow development, disruption of the thermal boundary layer and increasing the heat transfer surface area. Considerations in the Vipertube™ design (when compared to smooth tubes) include the maximization of heat transfer; minimization of operating costs; and/or minimization of the rate of surface fouling.

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Vipertex™ was able to develop several optimized enhanced heat transfer tubes. This study details the heat transfer and fluid flow results of the Vipertex 1EHT enhanced heat transfer tube for a range of Reynolds Numbers to approximately 18,000. Optimized Vipertex 1EHT tubes produces a region of operation for the inside tube heat transfer performance that is more than 5 times greater than smooth tubes and requires only modest increases in the friction factor. These enhanced tubes recover more energy and provide an opportunity to advance the design of many heat transfer products.

1. Introduction

Enhanced heat transfer surfaces can create a combination of: increased turbulence; disruption of the boundary layer; secondary flow generation and increased heat transfer surface area. These factors lead to an increase in the heat transfer coefficient; smaller unit footprint; more economic operation costs and a prolonged product life. This is a study of Vipertex 1EHT enhanced tubes evaluated for a wide range of conditions that includes the traditional laminar, transitional and turbulent flow regimes. Heat transfer enhancement using the Vipertex EHT series of tubes provides a means to significantly advance many heating and cooling processes, especially at low flows. In the case of a heat exchanger where there is a one-for-one replacement of smooth tubes with enhanced tubes of equal length there is an increase heat transfer for a constant fluid flow rate; however the pumping power of the enhanced tube exchanger is slightly greater due to increased friction. Alternatively, pumping power could be kept constant by reducing the tube-side velocity. Typically, for constant pumping power, the tube length and flow rate would be reduced. Enhanced tubes will reduce pumping power for a constant heat transfer and flow rate. Increases in heat transfer are greater than the friction factor penalty.

Many enhanced heat transfer surfaces increase efficiency in the turbulent region. Associated with those heat transfer increases are substantial increases to the friction factor. For many conditions, the increases in the heat transfer outweigh the increased friction factor penalty; however in some designs the benefit of the increased heat transfer cannot be justified since the increase in the friction factor is so great. In many geographical regions, design requirements for many heat transfer devices specify operations in the low flow region; in or near the traditional smooth tube transitional flow regime. For flows inside a smooth tube, transition from the laminar to turbulent regime occurs at a Reynolds Number of approximately 2300. Vipertex EHT enhanced tubes cause an earlier transition, more heat transfer and can be employed to design more efficient heat exchangers, decreasing the required mass flow rate for a given heat transfer rate and allow heat exchangers to operate at lower flows.

Reynolds Number transition in smooth tubes was observed by Lindgren (1953) to occur in a gradual manner with bursts of turbulence occurring down the length of the tube. Transition was found to be a function of fluid velocity and the distance from the inlet. Tam and Ghajar (1997) present in their study that smooth tube transition occurs in the Reynolds Number (Re) range between 2300 and 10000. They go on to show that, transition is dependent on position and additionally inlet conditions can cause transition to occur at higher Re. Nunner (1956) discusses the effect of tube augmentation on transition. Meyer and Olivier (2011) present results for heat transfer and friction factor for helical finned tubes in the “transition region” for fully developed and developing flows. A variety of enhanced surface studies have been previously performed and include: a study of dimpled tubes by Kalinin et al. (1991) and J. Chen et al. (2001); Wang et al. (2009) presents heat transfer results for internally finned tubes, while Gee and Webb (1980) study the effect of enhanced tube geometrical factors. Liu and Jensen (2001) demonstrate the effect of geometry on heat transfer for enhanced tubes. Vincente et al. (2002) discussed heat transfer and pressure drop of helically dimpled tubes. Christians et al. (2010 a,b) studied film condensation of refrigerants on enhanced tubes. Kukulka et al. (2011a) evaluated surface geometry of enhanced tubes and performed the groundwork for the present study. Wei 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 2500 to 90,000. For high Pr fluids (oil), they present a critical Re of approximately 6000 for enhanced heat transfer; while for a low Pr fluid (water) it is nearly 10,000. Kukulka et al. (2011b) evaluated fouling conditions on the heat transfer of enhanced surfaces.

Previously reported plain tube heat transfer results were used as a basis to compare relative thermal performance gain associated with the Vipertex 1EHT enhanced tubes. Experimental conditions included the traditionally laminar, transitional and turbulent flow regimes. Smooth tube performance was based on previously reported, commonly accepted Nusselt number studies; one for the laminar regime and the other for fully turbulent conditions.

Dittus-Boelter's (1930) classic correlation was used to calculate the inside film coefficient for turbulent conditions. The current study included tubeside fluid flow rates that extended into the fully-developed turbulent flow regime, with conditions evaluated for Reynolds numbers to approximately 10^5. Additionally, a modified form of the Dittus-Boelter evaluated the outside heat transfer coefficient as function of the annulus Reynolds number. For laminar conditions, the laminar Nusselt number was based on one of two sources; one known for accurate predictions in pure forced laminar convection (from Gnielinski as given in the VDI Heat Atlas (1993)) and the second method that considers mixed laminar convection (as reported by Ghajar and Tam, 1994). Hydraulic characteristics of the Vipertex 1EHT tubes were compared relatively to smooth tubes. Measured isothermal friction factors were compared to predictions from the isothermal friction factor correlation presented by Wilson et al. (1922) for plain commercial steel tubes.

2. Experimental Details

Heat transfer and hydraulic characteristics of the enhanced Vipertex EHT1 heat transfer tubes were evaluated. Heating and cooling studies were performed at the Heat Transfer Research, Inc. (HTRI) Research and Technology Center. Figure 1 provides inner and outer surface views of the 0.75 inch (19.05 mm) outer diameter Vipertex 1EHT enhanced stainless steel tube that was evaluated in this study.

For all tests, the heated/cooled length for each tube was 15 ft. (4.572 m). The heat transfer experimental apparatus was configured as a basic horizontal double-pipe heat exchanger through which a single tube could be inserted and centered. Test fluid for inside coefficient evaluation utilized propylene glycol over a wide range of Reynolds numbers (between 250 and 18000 for heating; and between 50 and 20000 for cooling) covering the laminar, transitional and turbulent regimes. Data ports are installed throughout the test apparatus with data being collected digitally at the desired intervals.

(a) (b)

Figure 1: Photographs of the Vipertex 1EHT tube evaluated in this study (a) Interior Cross Section (b) Detailed Outside Surface

During a typical evaluation, the test fluid is pumped from a main holding tank through flow meters and into a high temperature double pipe heat exchanger. After exiting the high temperature test apparatus, the test fluids passes through a set of static mixers and then into a low temperature double pipe heat exchanger. Upon leaving the low temperature test apparatus, the test fluid encounters another set of static mixers and then enters into the shell side of an auxiliary exchanger. After passing through the auxiliary exchanger, the test fluid returns to the tank. Fluid flow rate in the system can be controlled by adjusting the pump motor speed with a variable frequency drive. Saturated steam is used as the heating medium in the high temperature test apparatus. Tempered water from a closed-loop system is used as the cooling medium in the low temperature test apparatus. The temperature of the cooling water through the low temperature test apparatus can be varied between ambient and 250 °F (121.1 °C).

3. Results

Transient heat transfer evaluations were carried out with specific temperature differences for the following conditions: a horizontal tube; with inner fluid heating or cooling; single phase flow; a counterflow configuration; and various Reynolds Numbers. In all tests, the enhanced tubes outperformed smooth tubes under similar conditions. In order to evaluate performance the enhancement ratio is utilized. The enhancement ratio compares overall performance of the enhanced tube to that of a smooth tube. It can be expressed by the following equation:

$$EnhancementRatio = \frac{(hA)_{ENHANCED}}{(hA)_{PLAIN}} \tag{1}$$

where h is the overall heat transfer coefficient and A is the heat transfer surface area.
Figure 2 shows a comparison of heat transfer for Vipertex 1EHT tubes compared to smooth tubes, for interior fluid cooling. A peak in heat transfer occurs for Vipertex 1 EHT tubes in the classically defined "transitional" region. This peak gradually forms in the range of Re from 100 to 2000, with the maximum forming at approximately 800. In this range, the maximum heat transfer enhancement ratio is approximately 6 (see Figure 2). For flows above a Re of 2000, the increase in the heat transfer coefficient of the Vipertex 1EHT tube is almost 100 % more when compared to that of a smooth tube. Figure 3 shows the increase in friction factor that accompanies the increases of heat transfer for the interior fluid cooling case. At very low flows (in the region of Re =100) the maximum friction factor ratio (fENHNACED/fSMOOTH) is approximately 5. Near the region of maximum heat transfer (Re~800) the friction factor ratio decreases to ratio values of approximately 1.25-1.35. This ratio then forms another local maximum of 2.8 for an approximate Re value of 1500. For Re values greater than 3000 the friction factor ratio is almost constant and approximately 1.8. In the heating case we again see a maximum heat transfer (not as large as the cooling case) near a Re of 1000 (see Figure 4). Once again there is a local minimum of the friction factor ratio near a Re of 800, with the ratio then increasing slightly for Re values larger than 2000 (see Figure 5).

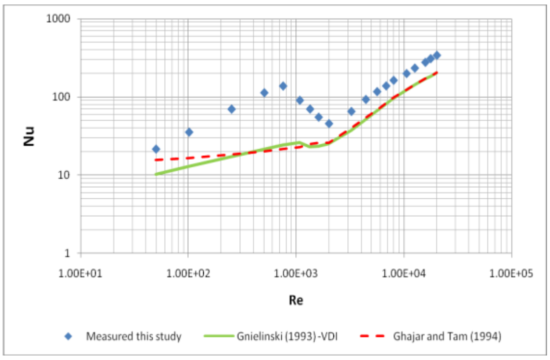

Figure 2 Comparison of the measured / reduced Nusselt number data versus Reynolds number for the Vipertex 1EHT and a smooth tube, in an inside fluid cooling arrangement.

Figure 3 Comparison of the measured / reduced friction factor data versus Reynolds number for the Vipertex 1EHT and a smooth tube, in an inside fluid cooling arrangement.

4. Summary

The purpose of this study was to characterize the thermal- hydraulic performance of an enhanced heat transfer tube that has been enhanced on the inside and outside surfaces. Evaluation was carried out in a double-pipe heat transfer test apparatus that utilized water and propylene glycol. The experiments considered both tube side heating and cooling conditions over a wide range of flow conditions.
A number of conclusions can be drawn from the results of this study:

1. For heating conditions, the 1EHT tube does not appreciably enhance heat transfer in the laminar flow regime beyond what can be expected from mixed convection. It appears to promote an early transition to turbulence at Reynolds numbers on the order of 1300. This is likely due to the surface design of the interior of that tube.

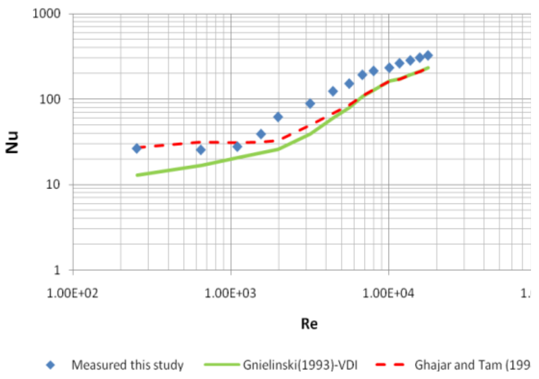

Figure 4 Comparison of the measured / reduced Nusselt number data versus Reynolds number for the Vipertex 1EHT and a smooth tube, in an inside fluid heating arrangement.

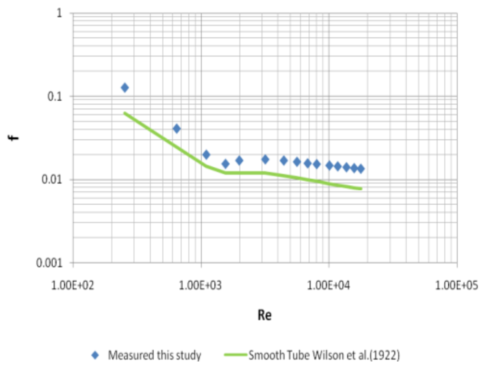

Figure 5 Comparison of the measured / reduced friction factor data versus Reynolds number for the Vipertex 1EHT and a smooth tube, in an inside fluid heating arrangement.

2. In the transition and turbulent flow regimes, the Vipertex 1EHT was found to produce an average two-fold increase in heat transfer relative to plain tubes for heating conditions. The corresponding increase in the isothermal and diabatic friction factors in the same flow regimes suggests that this performance boost is also the result of enhanced turbulence caused by the surface design.
3. The Vipertex 1EHT enhanced tube showed remarkable thermal performance characteristics in the laminar flow regime ($Re \leq 2200$) under cooling conditions. The Nusselt number peaks observed at Reynolds numbers of about 750 for the Vipertex 1EHT tube were on the order of five times greater than the comparable plain tube predictions at the same conditions. Although the underlying phenomena giving rise to it are not fully understood, this peculiar performance characteristic has been verified as being repeatable for the test fluid, tube geometry, and flow conditions considered.

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

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