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The Tube Side Heat Transfer Coefficient for Enhanced Double Tube by Wilson Plot Analysis

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Abstract: The purpose of the investigation presented in this study is to evaluate the tube side single phase heat transfer performance on the EXTEK (Twisted Multi-Head) tube. The geometry of the cross-section for a flow passage has an effect on its convective heat transfer capabilities. For concentric annuli in a double tube heat exchanger, the annular surface enhancement and tube profile enhancement play an important role. EXTEK (Twisted Multi-Head) uses twisted extrusion of a star shape tube for tube profile enhancement. The study was able to develop individual heat transfer coefficient correlations for this new method of enhancement for the turbulent flow regimes. A plain annulus was also investigated for comparison. The Wilson plot method was used to determine the tube side heat transfer coefficients from which the Nusselt type correlations were developed.

Key words: Enhanced double tube, wilson plot

INTRODUCTION

To enhance heat transfer in double tube heat exchangers, Fig. 1, mechanical modification, turbulent promoter (twisted tape), spring and disk are adopted to increase the heat transfer rate by reducing the thermal boundary layer thickness. In the case of mechanical surface enhancement modification, various methods are available, such as attaching extended fins, wires and roped tube on the inside and outside walls of the annulus. However, such mechanical enhancement presents difficulties for manufacturing and its cost of production is higher than that of other methods.

Thus, to improve the thermal performance of the double tube, a simple corrugation method is generally adopted for industrial applications. Although the corrugation geometry can enhance heat transfer by reducing the thermal boundary layer thickness, it will also cause an increase of flow resistance and pressure drop. Hence, many studies have been conducted on enhanced double tubes in order to minimize the pressure drop and maximize the heat transfer performance, review given by Mehta and Rao (1979).

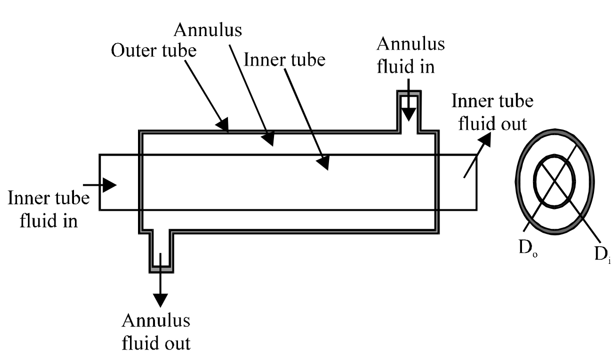

Fig. 1: Schematic diagram of a double tube heat exchanger

Two types of surface enhancement selected for investigation in the present study are a) Plain Annulus, and b) EXTEK (Twisted Multi-Head). EXTEK is a twisted extrusion of a star shaped tube for tube profile enhancement. This study aims to evaluate experimentally both the Plain Annulus and EXTEK tubes for the tube side single-phase heat transfer coefficients for the turbulent flow regime with straight double tube configuration.

LITERATURE REVIEW

Many investigators have devoted their efforts to study the two-phase and single-phase characteristics of enhanced double tube heat exchangers. It is known that the single-phase heat transfer data are of special value for the subcooled region of air-cooled condensers and superheated region of the air-conditioning evaporators, detail review given by Tiruselvam (2009). In addition, the design of water cooling/heating coils (double tube heat

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exchangers) commonly used in ventilators and package air conditioners, requires knowledge of the single-phase heat transfer data. Unfortunately, investigations of the single-phase heat transfer on enhanced annulus are not well correlated. For instance, the microphone single-phase R-113 heat transfer coefficients obtained by Khanpara *et al.* (1987) indicated that $Nu/Pr^{0.4}$ is proportional to $Re^{1.7}$ in the Re range from 6000 to 15,000. However, their R-22 data ($12,000 < Re < 15,000$) are well below the extension of R-113 line. Eckels and Pate (1991) found that the single-phase heat transfer coefficients of the microphone tube are proportional to $Re^{0.8}$. Al-Fahed *et al.* (1993) performed a single-phase experiment on the same tube as Eckels and Pate (1991) and found that the single-phase heat transfer coefficients of the microphone tube are proportional to $Re^{0.7}$. In addition, most of the investigators do not report the heat transfer coefficients for Reynolds number lower than 10,000. However, the design of air-conditioning systems in this range is often encountered. Therefore, it is necessary to clarify the single-phase heat transfer characteristic of enhanced double tube for $Re < 10,000$.

A brief review of the recent literature relevant to the experimental study for enhanced double tube is given in the following. A detailed experimental study performed by Dong *et al.* (2001) on various enhanced tubes have showed that the heat transfer performance of the spirally corrugated tube is 30 to 120% higher than that on the smooth tube. In addition, Zimparov (2001) derived an empirical correlation for heat transfer performance evaluation according to different relative pitches and pressure drop by inserting twisted tapes in the spirally corrugated tube. The results concluded that if the helical flow motion introduced by the twisted tape could be made by the actual tube profile, such a tube will deliver the required high heat transfer with reduced pressure drop. This approach will also be explored in this experimental work.

Initial investigation on the flow development length and fanning friction factor for both the annulus and tube side of these tubes was reported by Tiruselvam (2009). The correlations obtained in Tiruselvam (2009) will be used to evaluate the tube side heat transfer correlation by using the Wilson plot technique as reviewed in Shah (1990). The parallel characteristic of the friction factor (f) and the modified Stanton number (j) as reported by Shah Sekulic (2003) is exploited to achieve the objective of this experimental study.

Test section: This study will investigate two types of tube, i.e. plain surface and tube profile enhancement. The term EXTEK describes the Twisted Multi-Head profile enhanced tube as shown in Fig. 2. The test sections were assembled in straight double tube configuration.

Table 1: Data and dimension of EXTEK tube

Data	Unit
Inner EXTEK tube	
Tube outer surface area	0.09522 $m^2\,m^{-1}$
Tube inner surface area	0.08462 $m^2\,m^{-1}$
Outer perimeter of the tube profile	87.410 mm
Inner perimeter of the tube profile	79.410 mm
Length of the tube section	0.7 m
Tube wall thickness	1.299 mm
Tube flow area (cross section)	187.524 mm^2
Wall surface area (cross section)	103.713 mm^2
Outer steel tube	
Tube outer diameter	25.6 mm
Tube inner diameter	23.0 mm
Wall thickness	1.3 mm

Fig. 2: EXTEK profile enhanced tubes

The EXTEK Twisted Multi-Head tube used in this work consists of an inner EXTEK twisted copper pipe, as shown in Fig. 2, and an outer plain steel pipe. The EXTEK tube uses the twisted tube extrusion profile to induce secondary flows to both the tube and annulus flows. This helical cork screw flow pattern will constantly interrupt the boundary development. The high heat transfer occurring in the disturbed boundary development region will enhance the performance of this type of double tube heat exchanger. The profile dimensions for EXTEK Twisted Tube are given in Table 1. This EXTEK Twisted Tube is designed and manufactured by Zhejiang Co. Ltd. from China.

EXPERIMENTAL FACILITY

A schematic representation of the test facility is shown in Fig. 3. Two different concentric double tube heat exchangers (Plain and EXTEK), were used during the experimental investigation. The test section was operated in a counter flow arrangement with hot water in the annulus and cold water in the inner tube. The usual method to keep the double tube concentric is by employing radial supporting metal pins along the length of the heat exchanger. This method however could not be applied here because of:

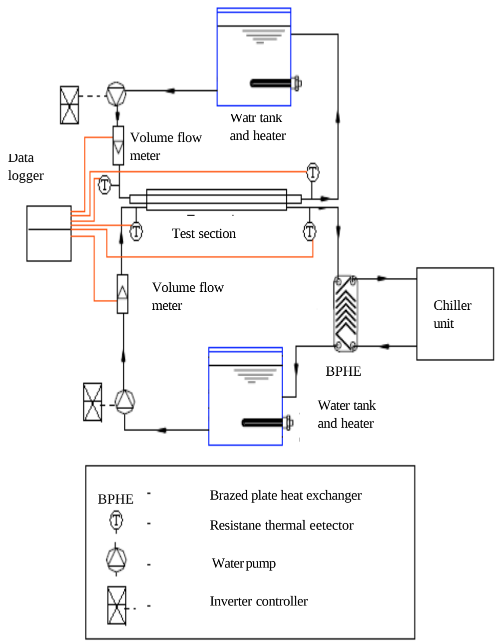

Fig. 3: Schematic diagram of experimental facility

- Small annulus clearance of approximately 2 mm (a = 1.18 for Plain annulus and a = 1.1 for Extek annulus) .Support pins located along the annular axis could restrict the medium flow
- The recommended annulus clearance requirement for installing annular support pins is given by Dirker and Meyer (2005) as a >1.5

Temperature measurements were facilitated by means of Resistance Temperature Detector (RTD - Pt100) at the entry and exit of the medium flow path. The entire test section and RTD measuring points were sufficiently insulated by Superlon pipe insulation and polyurethane enclosure to avoid heat loss to the ambient. Temperature data was captured with the aid of a data logger. Volumetric flow rates were measured by using YOKOGAWA magnetic flow meters. The flow meters were installed upstream to the test section.

By allowing a straight section distance of 1m before entering the flow meters, the chaotic flow patterns generated at tube bends and fittings were decreased. This ensured more accurate flow measurements. The experimental apparatus consist of two circulating water loops, i.e. the cold side and hot side. The cold side water is pumped from the water tank through the centrifugal pump and the flow meter before entering the inner tube of the test section. Any heat picked up from the test section is dispersed to the chiller unit through the brazed plate heat exchanger. The water temperature to the test section is controlled by the submerged water heater. Similarly the hot side water flow uses the same orientation, except for the absence of the heat sink.

Test procedure: Experiments were started by performing the Wilson plot test to evaluate the tube side single phase heat transfer coefficient. This was done with the annular flow rate held constant and the inner tube flow varied through a range of flow rates. The flows on both sides were maintained in the turbulent region while the total heat flux through the system is held constant. After sufficient time was allowed for steady state condition to be established, the inlet and outlet temperatures and flow rate of both fluids were recorded by means of the data logger. It is important to ensure that the energy balance error between both the tube and annulus sides was at a satisfactory low level. A high level of accuracy in the experimental data was thus maintained. More than 95% of the data points exhibited an energy balance error of less than 3% between the inner tube and annular flows. Suspicious data points were reexamined during the analysis process to increase the final accuracy thereof. Information on the experimental data and data sets used for analysis purpose is given in Table 2.

Uncertainties in the experimental data were calculated based on the propagation of error method, described by Kline, McClintok (1953). Accuracy for various measurement devices and water properties are given in Table 3 and 4. Uncertainties in the analysis of the single phase heat transfer coefficient are calculated for various test runs in the smooth and enhanced annulus as a root-sum-square (RSS) method. Experimental results and the associated uncertainties are listed in Table 5.

Data reduction: The overall thermal resistance is evaluated from Eq. 1:

$$U_O = \frac{Q}{LMTD\ A_O} \tag{1}$$

where, Q is the average heat transfer rate of the annulus and tube; Eq. 2:

$$Q = \left(\frac{(\dot{m}Cp\Delta T)_i + (\dot{m}Cp\Delta T)_O}{2}\right) \tag{2}$$

where, ΔT is the temperature rise/drop of water, and the subscripts o and i denotes the annulus and tube side, respectively. In all cases, only those data that satisfy the criteria $|(Q_O - Q_i)/Q| < 0.03$ are taken into consideration in the final data reduction. The log-mean temperature difference LMTD is given by Eq. 3 to 5:

Table 2: Experimental data sets and errors

Heat exchangre test	Number of data points	Nominal hot side mass flow rate (kg^{-1} s) Minimum	Maximum	Energy balance errors (%) Mean	Standard deviance
Plain straight tube $Re^{0.8}$	48	0.1345	0.3404	0.731	0.330
Plain straight tube $Re^{0.733}$	48	0.1335	0.3434	0.449	0.407
EXTEK Straight tube	48	0.1335	0.3464	0.450	0.409

Table 3: Uncertainties of measurement devices

Parameter (Make, Type)	Uncertainties
Water Volume Flow (YOKOGAWA, Magnetic Flow Meter)	±0.08% of reading
Differential Pressure Transmitter (YOKOGAWA, EJA Series)	±0.05% of reading
Pressure Transmitter (YOKOGAWA, EJA Series)	±0.18 psig
Water temperature (CHINO, Pt-100 RTD)	±0.1°C

Table 4: Uncertainties of properties

Predicted properties (Water)	Uncertainties	Source
Density	+0.02%	Wagner and Pruβ (2002)
Isobaric heat capacity	+0.3%	
Viscosity	+0.5%	Kestin *et al.* (1984)
Thermal conductivity	+0.5%	
Predicted properties (Copper)	Uncertainties	
Thermal conductivity	+0.5%	Toulaukian *et al.* (1970)

Table 5: Uncertainty analysis for experimental data

Test Sequence	Heat Transfer Coefficient, h ($W m^{-2}.K$), Turbulent Highest(%)	Lowest(%)
Wilson Plot Plain Straight Tube $Re^{0.8}$	±5.32	±5.04
Wilson Plot Plain Straight Tube $Re^{0.733}$	±4.24	±3.89
Wilson Plot EXTEK Tube	±3.99	±3.95

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln(\Delta T_1 / \Delta T_2)} \tag{3}$$

$$\Delta T_1 = T_{i,\,in} - T_{o,\,out} \tag{4}$$

$$\Delta T_2 = T_{i,\,out} - T_{o,\,in} \tag{5}$$

where, $T_{i,in}$ and $T_{i,out}$ are inlet and outlet temperature of water in the inner tube, and $T_{o,in}$ and $T_{o,out}$ denotes the inlet and outlet temperature of water in the annulus. At the first stage, the data are analyzed by the Wilson plot method and can be described as follows.

The experimentally determined resistance 1/UA of the test tube is related to individual thermal resistance, Eq. 6:

$$\frac{1}{U_O A_O} = \frac{1}{h_O A_O} + R_W + \frac{1}{h_i A_i} \tag{6}$$

where, h_o and h_i represent the average outside and inside heat transfer coefficient, and R_w denotes wall resistance and is given by $R_w = \delta_w/k_w A_w$. In the present calculation, the overall resistance is based on the outer surface area, which is evaluated as $\pi D_o L$, where Do is the outer diameter of the inner tube. Note that the inside heat transfer coefficient is based on nominal inside surface area ($\pi D_i L$). The properties for both streams were calculated using the average of the inlet and outlet bulk fluid temperatures. The tube side heat transfer coefficient h_i is given by Eq. 7:

$$h_i = C_i \frac{k_i}{D_i}\left(\frac{\rho VD}{\mu}\right)_i^M \left(\frac{Cp\mu}{k}\right)_i^{1/3} \tag{7}$$

The correlation form does not include the viscosity ratio to account for the radial property variation, because this effect is very small for the present test range. Therefore, Eq. 6 then becomes:

$$\left(\frac{1}{U_O A_O} - R_W\right) = \frac{1}{C_i\left(\frac{k_i}{D_i}\right) Re_i^M Pr_i^{1/3}} + \frac{1}{h_O A_O} \tag{8}$$

Equation 8 has the linear form of:

$$Y = mX + b \tag{9}$$

$$Y = \frac{1}{U_O A_O} - R_W \tag{10}$$

$$m = \frac{1}{C_i} \tag{11}$$

$$b = \frac{1}{h_O A_O} \tag{12}$$

$$X = \frac{1}{\frac{k_i}{D_i} Re^M Pr^{1/3} A_i} \tag{13}$$

Therefore, with a simple linear regression, the slope of the resulting straight line is equal to $1/C_i$. The interpretation of Eq. 9, which has the linear form, Y = mX + b, if R_w and h_o are constant, is the basis of the Wilson Plot method. In this arrangement, both the annulus heat transfer coefficient and the total heat transfer through the system will be constant. Eq. 9 can then be used to generate a straight line graph which describes the overall heat transfer process across the double tube when the coolant temperature and mass flow rates changes but quantity of heat transfer is constant. As a consequence of this, the internal heat transfer coefficients are balanced at different values so that while the overall heat transfer coefficient varies, the overall heat transfer and annular side heat transfer coefficient remain unchanged. Given such a test series, a line can be plotted as shown in Fig. 4.

TEST RESULTS

In order to validate the experimental apparatus and the testing methods, tests were performed on a straight double tube, with smooth inner and annular surfaces. Figure 5 shows the relationship of X and Y for the validation test. The regression result of the smooth tube yields C_i= 0 0227 which can be rounded up very close to the well-known constant 0.023 of the Dittus-Boelter correlation. The author has chosen Re exponent of 0.8 for validation purpose as such value is used extensively by previous studies, as in Shah (1990). Note that the exponent on Re of 0.8 is not necessary a constant 0.8 as shown in Fig. 5. As addressed by Shah (1990), the Re exponent it is a function of the Prandtl number and Reynolds number. It varies from 0.78 at Pr = 0.7 to 0.9 at Pr = 100 for Re = 50,000 for circular tube.

The author has adapted an approach where the Re exponent of the Nusselt correlation is plus 1 of the Re exponent of the Fanning friction factor. The relevant data extracted from a previous study by the current author in Tiruselvam (2009) is given as:

Straight Plain, Turbulent, Re > 8000:

$$f = 0.091Re^{-0.267} \tag{14}$$

Straight EXTEK, Turbulent, Re > 8500:

$$f = 0.591Re^{-0.41} \tag{15}$$

Based on the Re exponents from the fanning friction factor correlation s, the Wilson plot test was conducted

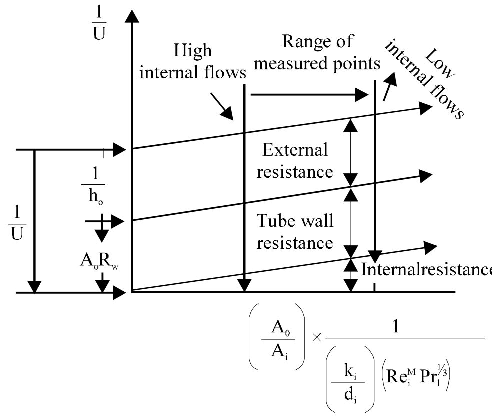

Fig. 4: The wilson plot-general features (Shah, 1990)

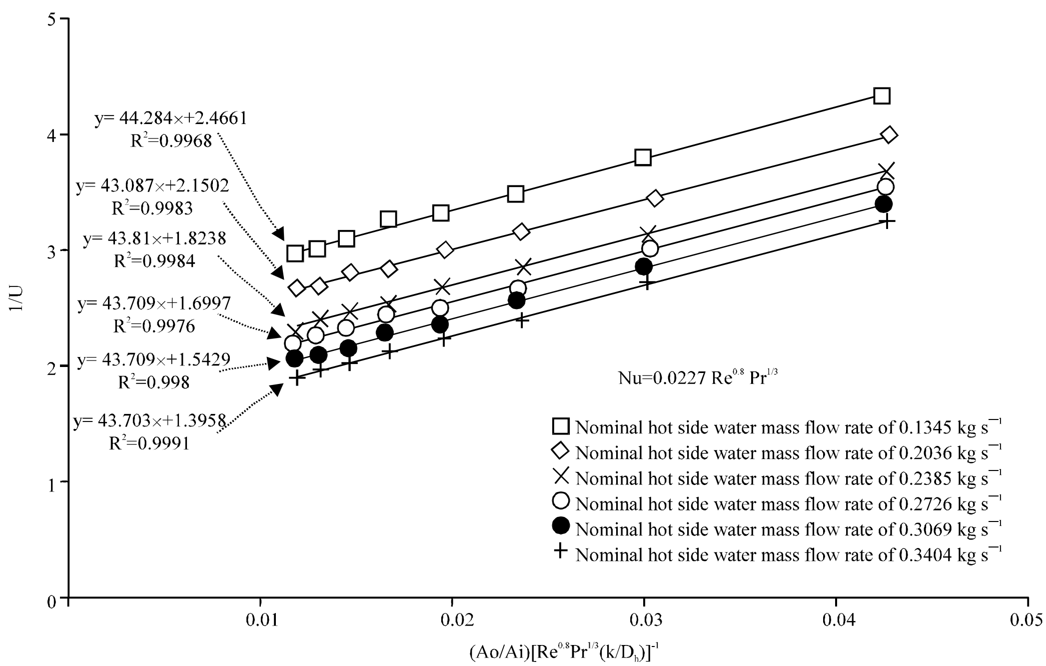

Fig. 5: Wilson Plot analysis for plain straight tube, $Re^{0.8}$

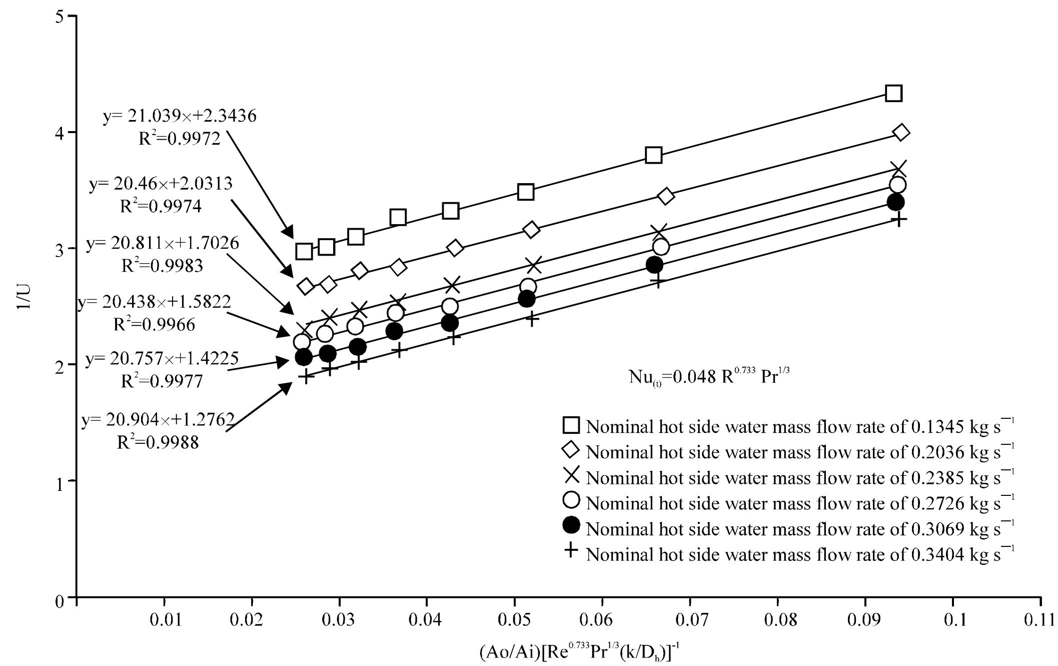

Fig. 6: Wilson Plot analysis for plain straight tube, $Re^{0.733}$

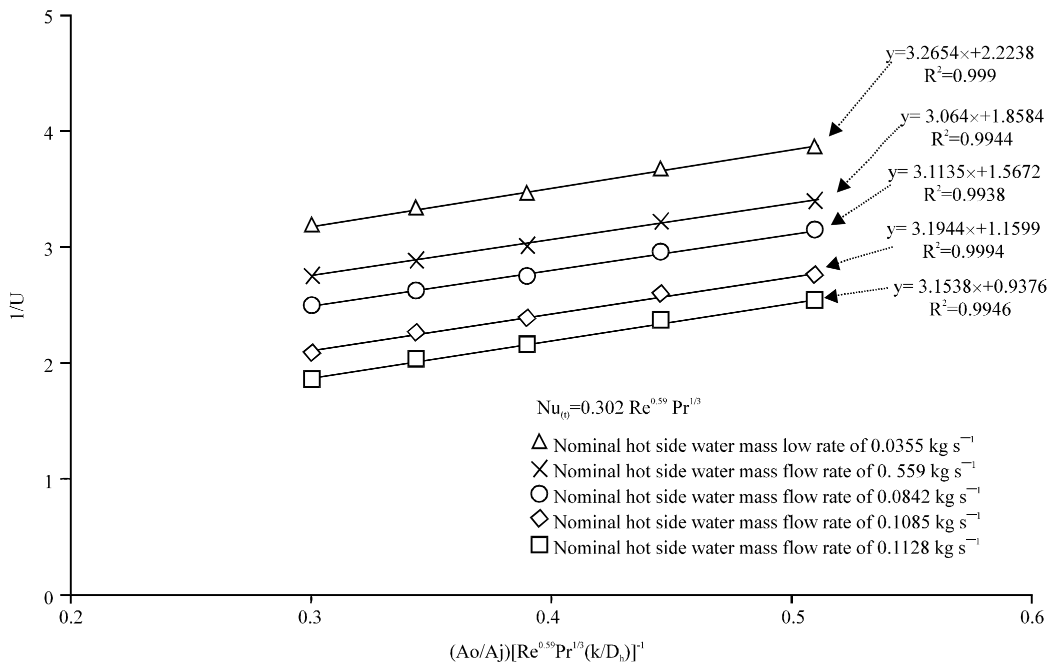

Fig. 7: Wilson Plot analysis for EXTEK straight tube

and the data analyzed for the plain and EXTEK inner tube. The corresponding Wilson plots are shown in Fig. 6-7.

Corresponding to the turbulent flow region of the individual test section, the side Nusselt numbers for single phase heat transfer can be written as: Plain Straight Tube, Turbulent, Re > 8000:

$$Nu = 0.048Re^{0.733}\,Pr^{1/3} \tag{16}$$

EXTEK Straight Tube, Turbulent, Re > 8500:

$$Nu = 0.302Re^{0.59}\,Pr^{1/3} \tag{17}$$

CONCLUSION

Convective heat transfer and pressure drop characteristic for two types of enhanced double tubes are reported in the present investigation. Experiments were conducted in a double tube heat exchanger with water as test fluid in the annulus and the tube side. The two annulus investigated was of Plain and Extek enhanced twisted multi-head profile. The heat transfer coefficients of the inner tube side of the double tube test section were obtained using the standard Wilson plot technique. The Wilson plot test was conducted for the turbulent flow region. An initial test was conducted for the plain annulus to validate the testing method and procedure using Re exponent value of 0.8. After achieving the experimental validation, the test was conducted on the Plain and EXTEK annulus using Re exponent from the Fanning Friction Factor correlation. The Nusselt heat transfer correlation, Eq. (16-17), obtained in this study for the tube side will be used for future study.

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NOMENCLATURE

a	=	Annulus ratio (D_o/D_i) (m)
A_i	=	Nominal inside heat transfer area of the tube (m^2)
A_O	=	Outside heat transfer area of the tube (m^2)
b	=	Intercept on file with ordinate (K/W)
C_i	=	Constant for inside heat transfer correlation, dimensionless
Cp	=	Heat capacity of water (J/kg.K)
D_i	=	Inside diameter of the tube (m)
D_o	=	Outside diameter of the tube (m)
D_h	=	Hydraulic diameter (m)
h_o	=	Heat transfer coefficient on the annulus side ($W/m^2.K$)
h_i	=	Inside heat transfer coefficient, base on A_i ($W/m^2.K$)
j	=	Colburn factor, St.$Pr^{1/3}$, dimensionless
k	=	Thermal conductivity of water ($W/m^2.K$)
L	=	Tube length (m)
LMTD	=	Logarithm mean temperature difference (K)
m	=	Slope of least-square deviation line, dimensionless
$\dot{m}$	=	Average mass flow rate of coolant water (kg/s)
M	=	Reynolds number exponent, dimensionless
Nu	=	Nusselt number (h_iD_i/k), dimensionless
Pr	=	Prandtl number ($\mu Cp/k$), dimensionless
Q	=	Average heat transfer rate (W)
Re	=	Reynolds number based on hydraulic diameter, $\rho VD/\mu$, dimensionless
R_W	=	Wall resistance (K/W)
St	=	Stanton number, dimensionless
$\triangle T$	=	Temperature rise on the water coolant (K)
U_O	=	Overall heat transfer coefficient ($W/m^2.K$)
V	=	Flow velocity (m/s)
X	=	Wilson plot function (K/W)
Y	=	Wilson plot function (K/W)

Greek:

δ_W	=	Wall thickness (m)
μ	=	Dynamic viscosity of water (Pa.s)
ρ	=	Density of water (kg/m^3)

Subscripts:

I	=	Tube side
in	=	Inlet
o	=	Annulus side
out	=	Outlet
t	=	Turbulent flow
w	=	Wall

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