RESEARCH ARTICLE

AN EXPERIMENTAL STUDY ON INFLUENCE OF SURFACE CORRUGATIONS TO HEAT TRANSFER USING DOUBLE PIPE HEAT EXCHANGER

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ABSTRACT

The incorporation of surface irregularities on the heat transfer surface is taking its attention because it enhances the thermo- hydraulic characteristics of the heat exchanger. However, the thermal performance enhancement and pressure drop increment comes parallel with heat transfer enhancement technique. But, in some applications, it is highly appropriate to provide surface roughness where space consideration is more vital to power loss as a result of pressure drop, namely on aerospace industry and nuclear reactors. The objective of this research is to study experimentally on the effect of surface corrugations to heat transfer using double pipe heat exchanger. The fiction factor and heat transfer parameters are obtained in the turbulent regime and the weight age of the both factors are compared with the help of thermal performance factor. The experimental data obtained from WILSON PLOT is validated with the data obtained from empirical relations. One smooth tube of 25 mm diameter and two enhanced tubes of same diameter with V corrugations on the outer surface having surface height, e = 0.3 mm and e = 0.6 mm and pitch, p = 1.3 mm are used. The corrugated tubes are found to be more advantageous than smooth tube for heat transfer applications. The percentage of heat transfer enhancement for tube with e = 0.3 mm is found to be 68 % to 51% and for a tube of e = 0.6 mm, is 90 % to 65 % in the Reynolds number regime of 16700 to 22300.The friction factor for the tube with e = 0.6 mm is found to be greater as compared to tube with e = 0.3 mm.

INTRODUCTION

Heat exchangers are used extensively in many areas like process industries, chemical and petroleum industries, thermal and nuclear power plants, heating ventilation and air conditioning, refrigeration and many other sectors. Since, it occupies a major role in many engineering sectors; the efficiency of its operation is a major concern. The enhancement of effectiveness of heat exchangers permits reduction in size and weight, raises average rate of heat transfer and lowers the difference in temperature between the fluids. Several heat transfer enhancement techniques are adopted to enhance the thermo-hydraulic behavior of the heat exchanger. The essence of enhancing the heat exchanger's thermal performance lies in the facts effecting to material, cost and energy savings, have led to the adaption of numerous approaches termed as *Heat Transfer Augmentation or Enhancement Technique*. The enhancement techniques can be

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categorized into three main types: *Use of nanofluid*, by the use of tabulators like *Inserts* and the *Roughening* of heat transfer surfaces in heat exchanger. In contrast to the augmentation in the heat transfer rate, the heat transfer enhancement technique also escalates largely the friction factor and the required pumping power in the counterpart. However, in some applications, it is highly appropriate to provide surface roughness where space consideration is more vital to power loss as a result of pressure drop. As an example, in case of aerospace applications, the size and weight of the heat exchanger are the significant parameters for design while pumping power comes as secondary parameter. The economic benefit in such applications will not be arising from the increment in the thermal performance but from the reduction of the material that is achieved by increasing thermal performance or in some aspect, allowing for possibility of using low velocity. In case of nuclear reactors, the main focus while designing heat exchangers is to reduce the material quantity because of the concern on absorption of more neutrons with increasing material proportion. The enhancement technique occupies its place in such applications in order to enhance thermo- hydraulic behavior of the

device.(2) The major objective of the present work is to study experimentally the influence of V corrugations made on outside tube surface of inner tube of double pipe heat exchanger to thermo-hydraulic characteristics of heat exchanger. The thermal performance of heat transfer device is evaluated based on heat transfer and friction factor parameters. Also, the experimental data obtained from WILSON PLOT is validated with the data obtained from empirical relations.

Heat Transfer Enhancement Methods

The heat transfer enhancement techniques currently available are generally categorized into three major sets.

- Methods requiring power input: Active methods
- Methods which does not require power input: Passive Methods
- Combination of active and passive Methods: Compound Technique

Roughness Elements Classification

Roughness is defined as the small scale projections or protrusions compared to smooth physical plane surface. On the basis of the physical ground, the artificial roughness can be categorized as:

1. Integral Roughness
2. Superimposed roughness

The *integral roughness* elements are integrated with the primary heat transfer surface. The integral roughness indicates the roughness elements are the integral part of the main surface. The *superimposed roughness* elements can be considered as turbulators. Such roughness elements are not integrated to the heat transfer surface. The different type of inserts like wire inserts, coil inserts are superimposed roughness elements

Experimental setup

The Figure 1 illuminates the experimental setup with its planlayout. The setup consists of double pipe heat exchanger, Rotameter, centrifugal pump, U-Tube manometer, thermocouple, Temperature display unit, overhead tank which is used for supplying cold water, hot water tank with submersion heater and a control unit.

The experimental test section which is one of the components of the experimental setup is a double pipe heat exchanger arranged in a countercurrent flow configuration. It consisted of inner tube made up of Aluminium and outer tube made of CPVC. Total of 3 tubes were tested; one smooth tube for the reference motive and the remaining two were enhanced tubes. The heat transfer fluid used was the normal hard water with the cold fluid being in the annulus side and the hot fluid in the tube side. Two calibrated Rotameters were used in the setup, placed in the cold fluid and the hot fluid side respectively with the flow measuring range of 20-1080 LPH. The cold fluid was drawn from the overhead tank with the assistance of gravity.

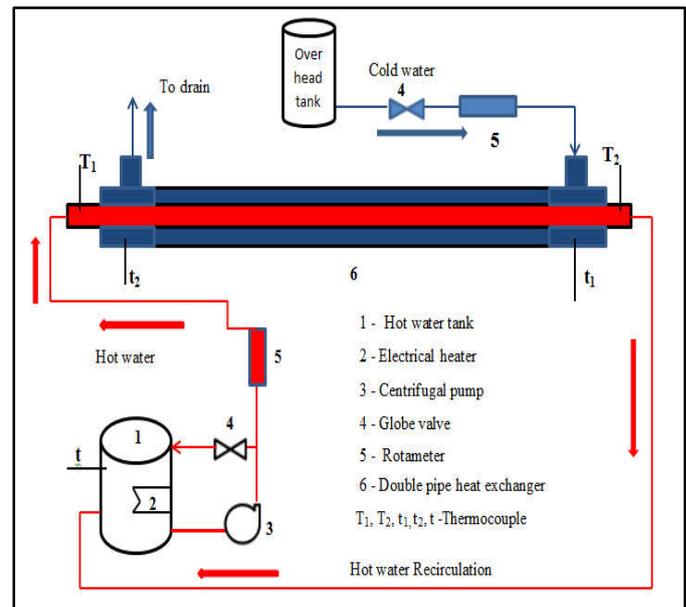The cold water was available with the temperature range: 28-30^0C.

**Fig. 1. Schematic layout of the Experimental Setup**

Fig. 2. Test Experimental Setup

The hot water was supplied from the hot water tank at an approximated temperature of 50-52 ^{0}C with the help of the centrifugal pump of rating 0.5 kW. The hot water was produced with the help of the submersion Nichrome heater of rating 1.5 kW. The bypass valve was facilitated at the discharge of the pump for controlling the flow rate and the bypass was re-circulated to the hot water tank. The U- Tube manometer was placed in order to measure the pressure drop in the cold fluid and the hot fluid side and the two pressure tapings were made at the inlet and outlet section of the heat exchanger and were connected to the U-Tube manometer with mercury as a Manometric fluid. K Type Thermocouple with a temperature measuring range:0-200^{0}C, were placed six in number, one at the hot water tank, one for atmospheric

temperature, two at the inlet and outlet of the hot fluid side and the remaining two at the inlet and outlet of the cold fluid side. The Figure 2 shows the experimental setup used for the heat transfer analysis.

Heat exchanger and Tube geometry

- Heat exchanger: Double pipe heat Exchanger (40 mm * 25 mm) and length = 1.5 m
- Smooth tube: OD = 25 mm, Length= 1.5 mm
- First enhanced tube :Thread depth= 0.3 mm with pitch =1.3 mm
- Second enhanced tube: Thread depth= 0.6 mm with pitch = 1.3 mm
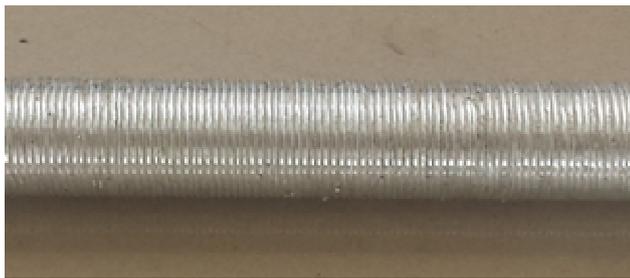
(a)
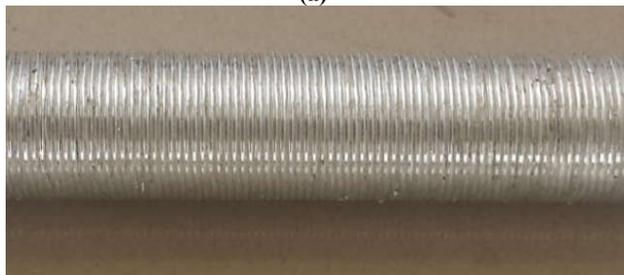
(b)

(c)

Fig. 3. (a) Enhanced tube with Thread depth= 0.3 mm with pitch =1.3 mm (b)Enhanced tube with Thread depth= 0.6 mm with pitch = 1.3 mm (c) Smooth tube with OD = 25 mm, Length= 1.5 mm

Equations used

I. Calculation of Friction Factor:

a.Using Drew, Koo and McAdams within ± 5% (Drew *et al.*, 2013)

$$f = 0.0014 + \frac{0.125}{Re^{0.32}}, \text{ for smooth pipe}$$

$$\frac{1}{\sqrt{f}} = 1.8 \, log \left(\frac{6.9}{Re} + \left(\frac{e/d}{3.7}\right)^{1.11}\right), \text{ for rough pipe (Haaland, 1983)}$$

b.Darcy Equation

$$f = \frac{P}{2} \frac{d_e}{\rho \, L \, V^2}$$

II. Energy Balance Equation

a.$Q = m*C_p*(T_1-T_2)$

III. Heat Transfer Calculations

For, $Re \geq 10000$, $0.7 < Pr < 16700$, $L/D \geq 60$, Sieder Tate Equation (Kothandaraman and Subramanyam, 2015)

a.$Nu = 0.027 \quad Re^{0.8} \quad Pr^{0.33}\left(\frac{\mu}{\mu_w}\right)^{0.14}$

b.For Fully Developed flow, Dittus- Boelter equation is used For,$2500 < Re < 1.25*10^6$, $L/D > 60$ and $0.6 < Pr < 100$

$$Nu = 0.023 \quad Re^{0.8} \quad Pr^n$$

For fluids increasing its temperature, n = 0.4
For fluids decreasing its temperature, n = 0.3

c.$Q = U \quad A_o \quad T_{ln}$

d. $T_{ln} = \frac{(T_1-t_2)-(T_2-t_1)}{\ln\frac{(T_1-t_2)}{(T_2-t_1)}}$

IV. Thermal Performance factor

$$\eta = \frac{\frac{e}{}}{\left(\frac{f_e}{f}\right)^{1/3}}$$

RESULTS AND DISCUSSION

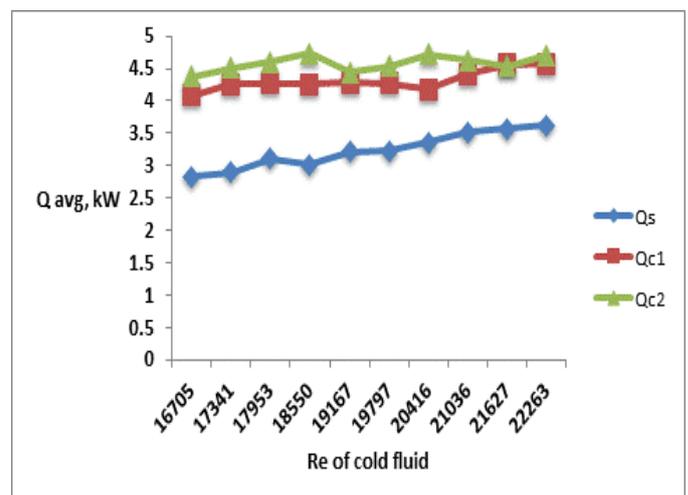The important results drawn from the nature of the experimental data are discussed
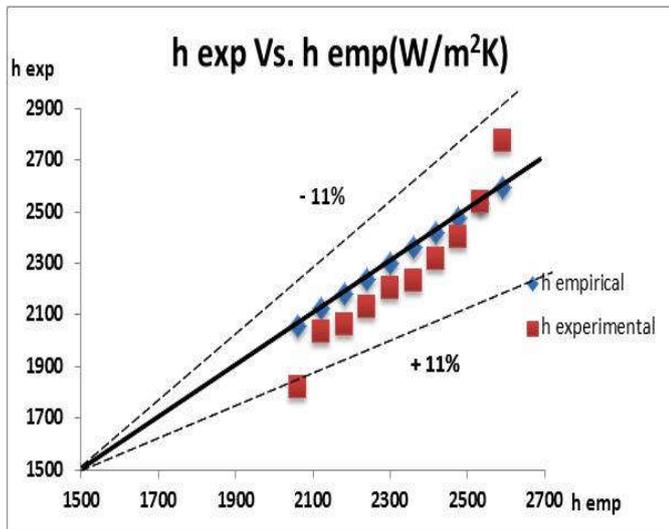
Fig. 4. Q vs. Re of cold fluid

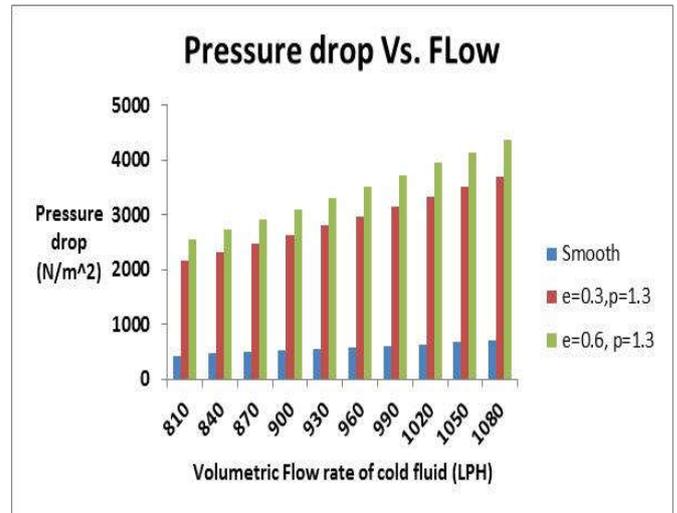Fig. 5. h exp vs. h the or for smooth tube

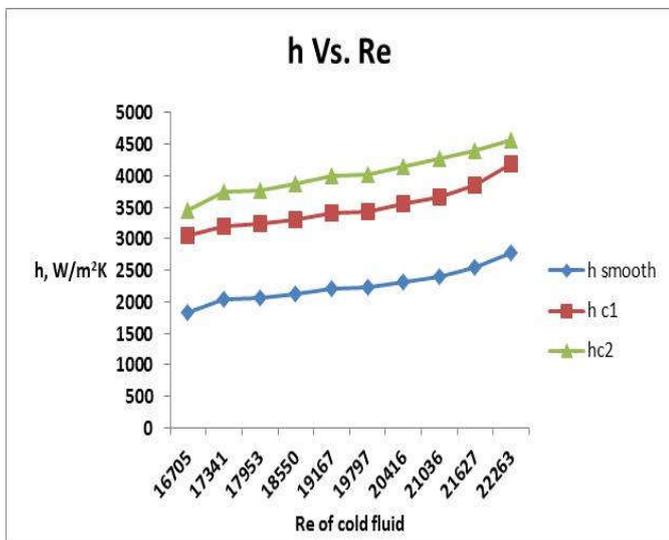Fig. 8. Pressure drop vs. Flow of cold fluid
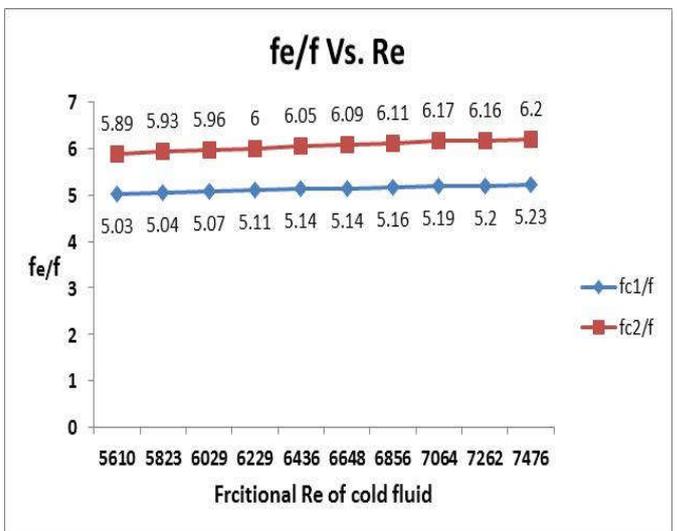

Fig. 6. h vs. Re of cold fluid

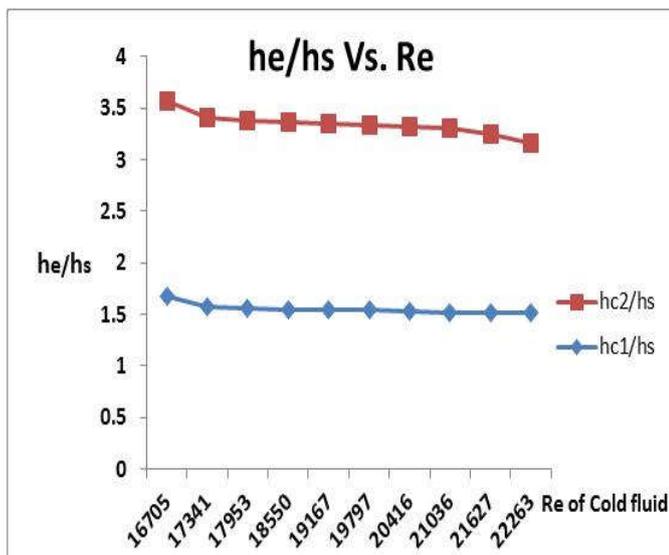Fig. 9. f_e/f vs. Frictional Re of cold fluid
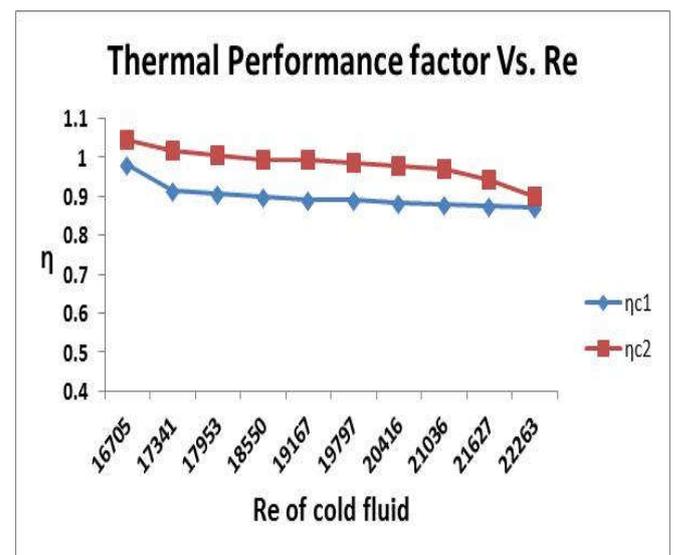

Fig. 7. h_e/h_s vs. Re of cold fluid

Fig. 10. Thermal Performance factor vs. Re

It can be noted from Figure 4 that the average heat transfer rate for smooth tube is increased from 2.8 kW to 3.6 kW, in the range of Re, 16700 to 22300 while for the tube with e = 0.3 mm consisted of heat transfer increment rate from 4.1 kW to 4.57 kW and for a tube with e = 0.6 mm, from 4.4 kW to 4.7 kW. It can be seen from figure 5 for a smooth tube that the range of h_{emp} is from 2060 to 2592 W/m^{2}K while h_{exp} came to be in the range of 1819 to 2773 W/m^{2}K. The theoretical and experimental values are obtained in the range of ± 11% error. The figure 6 showed that the heat transfer coefficient for enhanced tube with e =0.3 mm is increased from 68 % to 51 % in a Reynolds number range of 16700 to 22300 and the maximum enhancement occurred with lowest Re. For a tube with e = 0.6 mm, the heat transfer coefficient is increased from 90 % to 65 % in a given range of Re. The increment in convective coefficient of heat transfer is because of the enhanced thermal and fluid transport properties induced by the thinning of the boundary layer caused by the surface roughness. The thinning of boundary layer is more as the roughness value increases and resulted in more increment in heat transfer coefficient. The Figure 7 illuminated that the enhancement ratio for a given roughness is decreased with increasing Reynolds number. The reason for drastic increment in the pressure drop for enhanced tube as shown in figure 8 is because of high increment in the friction factor caused by the roughness on the surface. The rough tube enhanced the turbulence in the flow with introduction of the secondary viscosity coefficient and induces more shear force and hence more friction factor which leads to the considerably higher pressure drop compared to smooth tube. The figure 10 showed firstly that the thermal performance factor is increased with increase in roughness value ($\eta_{c2}>\eta_{c1}$) and second, the thermal performance factor is declined with rise in Reynolds number. The maximum thermal performance factor for corrugated tube with e = 0.3 mm is 0.981 and for a tube with e = 0.6 mm, the maximum value is 1.047. The mean thermal performance factor is 9.3 % higher for corrugated tube with e = 0. 6 mm as compared to tube with e= 0.3 mm.

Conclusion

The prime objective of this research was to study the influence of surface corrugations to heat transfer in a double pipe heat exchanger. Following conclusions are derived from the experimental study and the nature of the data obtained.

- In a bigger picture, the corrugated tubes are found to be more advantageous than smooth tube for heat transfer applications.
- The variation of empirical and experimental data for smooth tube is found to be in the range of ± 11 %.
- The convective heat transfer coefficient enhancement for tube with e = 0.3 mm is found to be in the range of 68 % to 51 % and for tube with e = 0.6 mm is found to be in the range of 90 % to 65 % in a range of Reynolds number of 16700 to 22300.
- The enhancement in the heat transfer coefficient at low value of Reynolds number is found to be greater. Thus, the heat transfer enhancement ratio is decreased with increasing Reynolds number.
- The friction factor is found to be increased with increasing the roughness value but tends to remain constant at higher regime of Reynolds number.
- The important conclusion that we can make is that, the energy saving is more for corrugated tubes at low value of Reynolds number

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