Effects of Arrangement and Surface Roughness of Transverse Studs on Heat Transfer in Round Tubes

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Abstract: Four heat transfer enhancing tubes (tubes A – D) with transverse studs are fabricated and tested. The studs are regularly welded in the tubes at a pitch of 53.2 mm. The inner and outer diameters of the tubes are 13.3 and 17.3 mm, respectively. Tubes A, B and C use the same threaded studs (M4) as flow disturbing device, but the arrangements of the studs on the tube walls are different. Tube D adopts the same stud arrangement as tube A, but its studs are smooth instead of threaded. Air is the working fluid in the tubes. For the Reynolds number (Re) in the range of 4000 to 23000, fluid friction and heat transfer data of the four heat transfer enhancing tubes are measured and the results are compared with those of a smooth tube. The transverse studs are quite effective for enhancing the heat transfer in the tubes. Regardless of the Re value and stud arrangement, a twofold increase in the convection heat transfer coefficient can be achieved. It is also found that the stud arrangement and the stud surface roughness are insignificant to the heat transfer. A frequent variation in stud direction or a rough stud surface would result in an increase in pressure drop, instead of an increase in heat transfer.

Keywords: Heat transfer enhancement, transverse stud, Darcy friction factor, Nusselt number.

1. INTRODUCTION

Due to simplicity in manufacturing process, many heat exchangers are composed of round tubes. For some applications, the heat transfer performance of these heat exchangers can be largely improved by increasing the internal convection heat transfer coefficients of the tubes. In the past few decades, this demand has inspired various heat transfer enhancement techniques [1, 2]. Roughened surfaces are quite often used for enhancing the heat transfer in round tubes. Webb *et al*. [3] investigated the heat transfer and fluid friction for fully developed turbulent flow in tubes with repeated transverse ribs. Gee and Webb [4], Naphon *et al*. [5] and Ravigururajan and Bergles [6] measured the heat transfer and fluid friction data for tubes with helical rib. The data were correlated with dimensionless characteristic parameters to form empirical equations. San and Huang [7] and Huang *et al*. [8] experimented the heat transfer and fluid friction characteristics for three different fluids flowing in circular tubes with repeated ring-type ribs. Kim *et al*. [9] proposed an optimum design for transverse ribs in circular tubes.

In addition to roughened surfaces, various inserts such as wire coil and twisted tape can also results in a flow disturbing effect in round tubes with an accompanying rise in heat transfer. Uttarwar and Raja Rao [10] experimentally investigated the heat transfer characteristics for helical-wire-coil inserted circular tubes. It was found that the heat transfer enhancement of the wire coil at low Reynolds numbers is more pronounced than that at high Reynolds numbers. Sethumadhavan and Raja Rao [11] proposed optimum rib helical angles for helical-wire-coil-inserted circular tubes. San *et al*. [12] proposed a set of fluid friction and heat transfer empirical equations for helical-wire-coil inserted circular tubes. The equations were expressed as a function of the Reynolds number, dimensionless wire diameter and dimensionless coil pitch. Junkhan *et al*. [13] compared the heat transfer enhancements of three different types of insert. The obtained data are useful in assessing overall performance gains when inserts are used in fire tube boilers. Bas and Ozceyhan [14] investigated the heat transfer enhancement of twisted tape inserts in a circular tube. It was found that the heat transfer enhancement decreases with an increase in the Reynolds number. Promvonge *et al*. [15], Rainieri *et al*. [16] and Zimparov [17] showed that installing wire coil or twisted tape in corrugated tubes can further enhance the in-tube heat transfer. Garcia *et al*. [18] compared the heat transfer performance of corrugated tube to those of dimple tube and wire-coil-inserted tube. At the same heat transfer level, the pressure drop of the wire-coil-inserted tube appears to be higher than the other two. Huang and San [19] measured the boiling heat transfer data for a corrugated tube and a fin-module-inserted tube. It was found that the heat transfer rate in the latter is higher than that in the former.

In this work, a set of transverse studs is proposed for enhancing the heat transfer in circular tubes. To

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verify the effectiveness in heat transfer enhancement due to the transverse studs, four heat transfer enhancing tubes (tubes A–D) plus a smooth tube are tested. The four heat transfer enhancing tubes are different in stud arrangement and stud surface roughness. Through a comparison in fluid friction and heat transfer between the tubes, the enhancement in heat transfer due to the transverse studs is revealed. In addition, it also yields appropriate stud arrangement and stud surface roughness for achieving an effective and efficient heat transfer enhancement in circular tubes.

2. EXPERIMENTAL SETUP AND APPARATUS

The five test tubes and the transverse studs are made of stainless steel. The inner and outer diameters of the tubes are 13.3 mm and 17.3 mm, respectively. In tubes A, B and C, a set of studs with standard M4 thread is welded in the tubes at a pitch of 53.2 mm. The threaded studs are perpendicular to the flow direction and pass through the tube central axis. The arrangements of the studs in the three tubes are different in circumferential angle on tube wall. In tube A, the studs are orderly and spirally distributed along the tube wall and the deviation in circumferential angle between two adjacent studs is 90^{o} (arrangement I, Figure **1a**). In tube B, the deviation in circumferential angle between two adjacent studs is 180^{o} (arrangement II, Figure **1b**). In tube C, for every two studs, the increase in circumferential angle is 90^{o} (arrangement III, Figure **1c**). The stud arrangement in tube D is the same as that in tube A. But the surface of the studs in

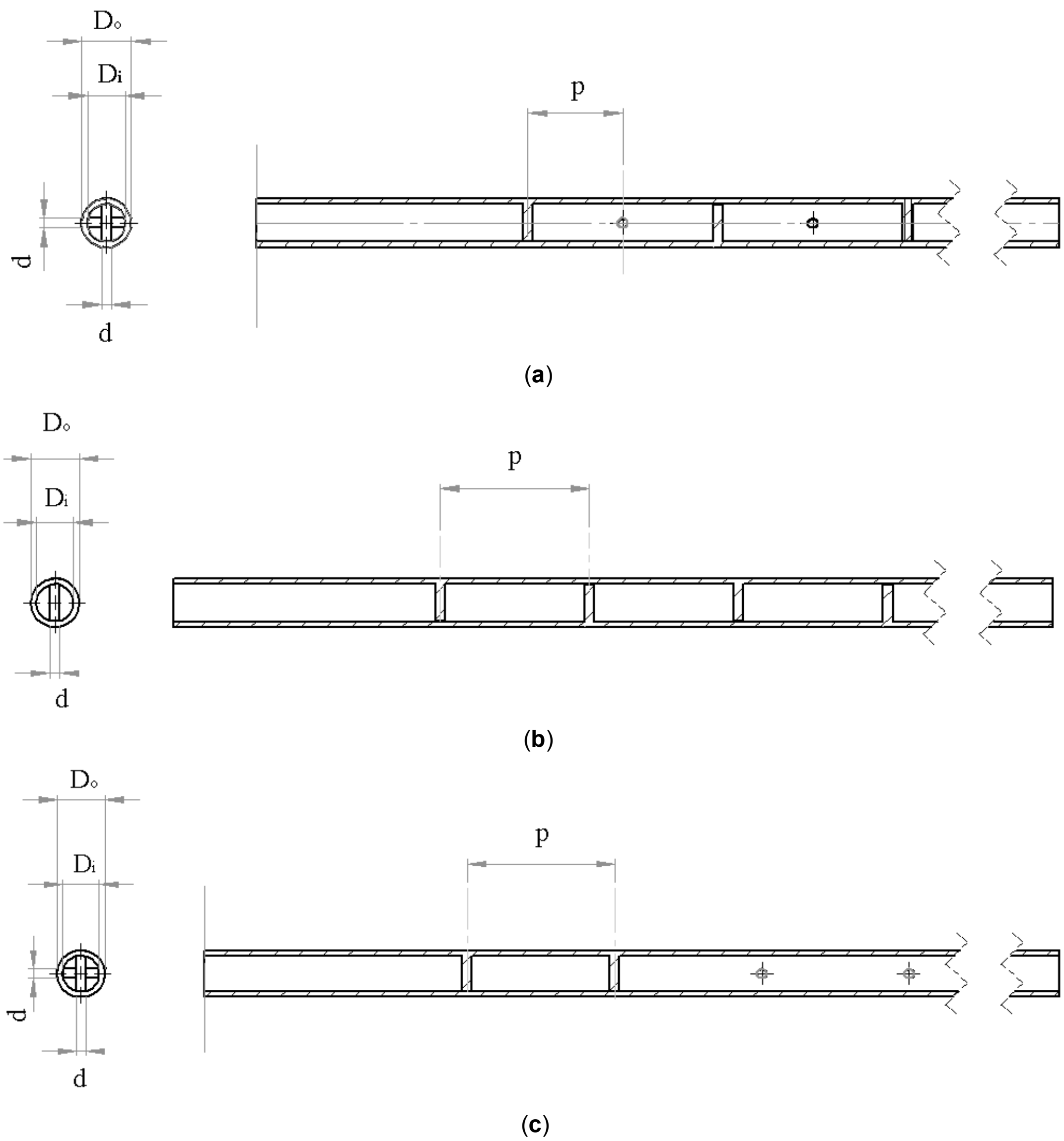

Figure 1: Sectional views of transverse studs in circular tubes (**a**) Type - I stud arrangement. (**b**) Type - II stud arrangement. (**c**) Type - III stud arrangement.

tube D is smooth, instead of threaded. Detailed specifications of the test tubes are listed in Tables **1** and **2**.

Table 1: Specifications of Transverse Studs in Test Tubes

Tube cat.	Diameter (d) mm	Pitch (p) mm	Surface	Arrangement
A	4	53.2	M4 thread	I
B	4	53.2	M4 thread	II
C	4	53.2	M4 thread	III
D	4	53.2	smooth	I
E	0	-	-	-

Table 2: Three Different Arrangements of Transverse Studs

Circumferential angles of studs	
Type I	0^o-90^o-180^o-270^o-0^o (repeated)
Type II	0^o-180^o-0^o (repeated)
Type III	0^o(2)-90^o(2)-180^o(2)-270^o(2)-0^o(2) (repeated)

A double-pipe device is used in the heat transfer measurement (Figure **2**). The inner tube of the device is the test tube which is installed inside an outer tube with inner diameter of 27.8 mm and wall thickness of 3 mm. On each side of the device, the gap between the inner tube and the outer tube is sealed by a tapered rubber sleeve. In the double-pipe device, the effective heat transfer length of the test tube is 1.17 m. Air and water are arranged to flow in the test tube and the annulus, respectively. A pump rated at 375 W is used to circulate the water between the double-pipe device and a water tank. In the tank, the water is heated by using a PID controlled electric resistance heater.

A blower rated at 1.5 kW is used to supply the air passing through the test tube. The rotational speed of the vane in the blower is regulated by using an electricity inverter. This enabled the air volumetric flowrate to be accurately controlled at a designated value. In the experiment, heat transfer and fluid friction data at eight different air volumetric flowrates are measured (40, 60, 80, 100, 120, 140, 160 and 180 L/min). The temperature of the supplied air would increase with the rotational speed of the blower. In order to get a stable temperature, before entering the test tube, the air flow is arranged to pass through a cross-flow air-to-water cooler, in which the inlet temperature of the water is fixed at 25^{o}C.

In the fluid friction measurement, the test tubes are unheated. At low pressure drops, an inclined U tube (Dwyer-424) filled with a red oil is used. At high pressure drops, the inclined U tube is replaced by

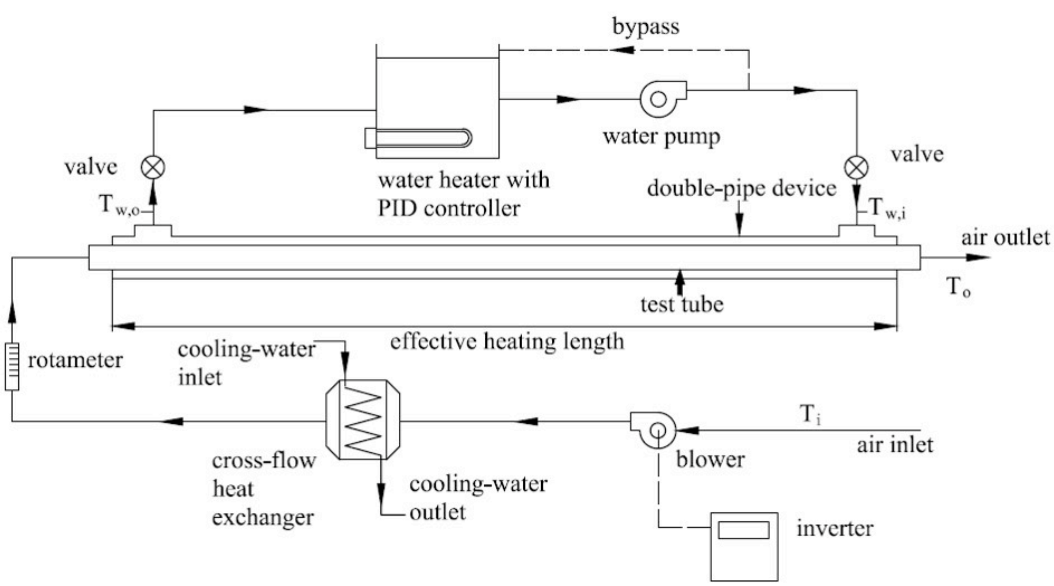

Figure 2: Schematic of heat transfer measurement.

another vertical U tube filled with water. In the heat transfer measurement, four T-type thermocouples are used to measure the inlet and outlet temperatures of the air and the water, respectively. The wire diameter of the thermocouples is 0.1 mm. The signals of the thermocouples are collected by a data recorder (Hioki–LR8431-20) and the air volumetric flowrate is monitored by a rota meter.

3. ISOTHERMAL WALL MODEL AND PERFORMANCE INDEXES

In the experiment, the inner and outer surfaces of the test tubes are considered to be isothermal. For tubes with isothermal inner and outer surfaces, the average in-tube convection heat transfer coefficient (h_i) can be determined using the following equation [9],

$$\frac{T_w - T_o}{T_w - T_i} = \exp[-(\frac{U_i A_i}{\dot{m} c_p})] \tag{1}$$

where $\frac{1}{U_i A_i} = \frac{1}{A_i h_i} + \frac{\ln(D_o / D_i)}{2\pi k_s L} + \frac{1}{A_o h_o}$,

In eq. (1), U_i denotes the overall heat transfer coefficient between the air and the water in the double-pipe device; $[1/A_i h_i]$ is the thermal resistance between the tube inner surface and the air; $[\ln(D_o/D_i)/2\pi k_s L]$ is the thermal resistance due to the tube wall; $[1/A_o h_o]$ is the thermal resistance between the tube outer surface and the water. In the experiment, the volumetric flow rate of the water in the annulus is 22.4 L/min. By using the Dittus-Boelter equation, the water-side convection heat transfer coefficient (h_o) is estimated to be 7007 W/m^2-K. In addition, the arithmetic mean value of the water temperatures at the inlet and outlet is assumed to be the temperature of the water in the annulus (T_w). The maximum temperature difference of the water between the inlet and the outlet is less than 0.2^{o}C.

The Nu enhancement index (r_1) of a heat transfer enhancing tube is defined as the Nu value of the tube divided by the Nu value of a smooth tube. It can be expressed as;

$$r_1 = Nu_{\text{with transverse studs}} / Nu_o \tag{2}$$

The mechanical energy consumption index (r_2) of a heat transfer enhancing tube is defined as the Nu/f value of the tube divided by the Nu/f value of a smooth tube. The r_2 value can be expressed as;

$$r_2 = (Nu/f)_{\text{with transverse studs}} / (Nu/f)_o \tag{3}$$

The r_1 and r_2 values represent the performance of the heat transfer enhancement due to the transverse studs. The larger the r_1 value, the more effective the heat transfer enhancement; the higher the r_2 value, the more efficient the heat transfer enhancement.

4. RESULTS OF FLUID FRICTION AND HEAT TRANSFER MEASUREMENTS

4.1. Fluid Friction and Heat Transfer Data for the Heat Transfer Enhancing Tubes

The f and Nu values for tubes A and E are shown in Figure **3**. Tube E is the smooth tube, while tube A is the tube adopting type I arrangement of the threaded studs. For verifying the accuracy of the experimental apparatus, the f and Nu data for tube E are compared with the f and Nu values predicted by using the Petukhov equation and the Gnielinski equation, respectively. The experimental data agree well with the predicted values.

As shown in Figure **3**, for tubes A and E, the f value decreases with an increase in the Re value, whereas the Nu value increases with the Re value. The Nu value for tube A appears to be much larger than that for tube E. This verifies that the transverse studs are quite effective for enhancing the heat transfer in round tubes. However, the studs also causes a significant increase in the f value. A significant increase in the f value means a large amount of extra mechanical energy needs to be consumed for achieving the heat transfer enhancement.

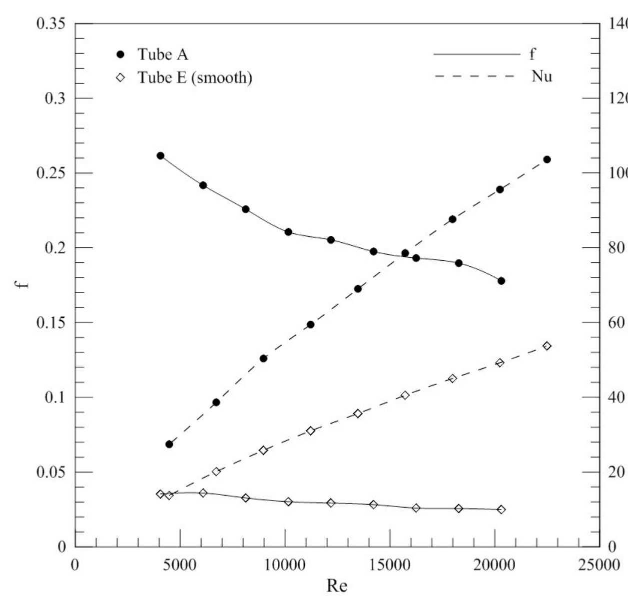

Figure 3: Comparison in f and Nu values between tube A and smooth tube (tube E).

A comparison in f and Nu values between tubes A, B and C is shown in Figure **4**. Tubes A, B and C are different in the arrangements of the threaded studs in the tubes. As shown in the figure, the Nu value for tube C is the largest, whereas the Nu value for tube B is the smallest. However, for the three tubes, the deviation in Nu value is small. This means the effect of stud arrangement on the heat transfer is weak. In the figure, it is also readily noticed that the f values for tube B and tube C are smaller than the f value for tube A, whereas the f value for tube B is close to that for tube C. In Tube A, the directions of the studs consecutively and spirally vary. A frequent variation in stud direction could result in a large increase in the pressure drop.

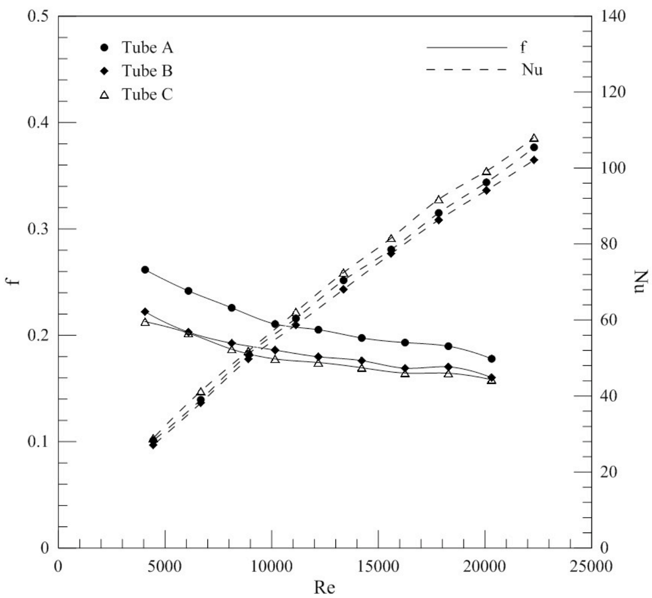

Figure 4: Effect of stud arrangement on the f and Nu values.

A comparison in the f and Nu values between tubes A and D is shown in Figure **5**. The surface of the studs in tube A is threaded, whereas the surface of the studs in tube D is smooth. As shown in the figure, the Nu values for the two tubes are close. But it is quite clear that the f value for tube A is much larger than the f value for tube D. The thread on the stud mainly induces minor disturbances in the core region of the flow instead of near the tube surface. It causes a significant increase in pressure drop. Nevertheless, the heat transfer remains almost unchanged. The effect of thread on the f value increases with a decrease in the Re value. At the Re value in the range of 4000 to 23000, the increase in pressure drop resulting from the threads on the studs is in the range of 12% to 29%.

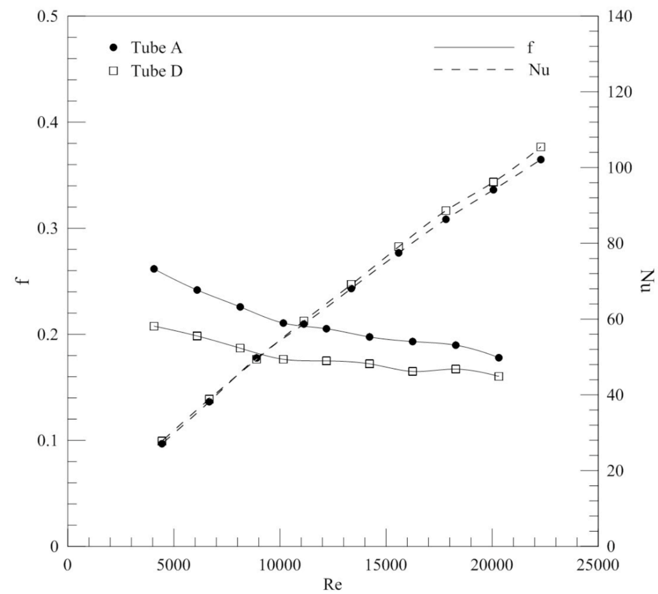

Figure 5: Effect of stud surface roughness on the f and Nu values.

4.2. Performance Indexes for the Heat Transfer Enhancing Tubes

The r_1 values for tubes A, B, C and D are shown in Figures **6-9**, respectively. For tube A, the r_1 value is in the range of 1.93 to 2; for tube B, the r_1 value is in the range of 1.99 to 2.09; for tubes C, the r_1 value is in the range of 2.04 to 2.13; for tube D, the r_1 value is in the range of 1.99 to 2.05. For all the four tubes, even at large Re values, the heat transfer enhancement due to the transverse studs is still quite pronounced. Comparing the r_1 values for the four tubes shows that tube C is slightly superior to the other three.

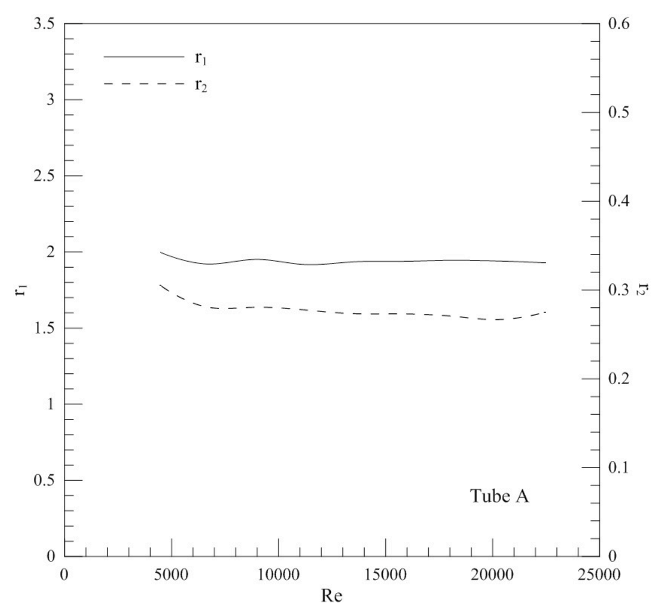

Figure 6: Nu enhancement index and mechanical energy consumption index for tube A.

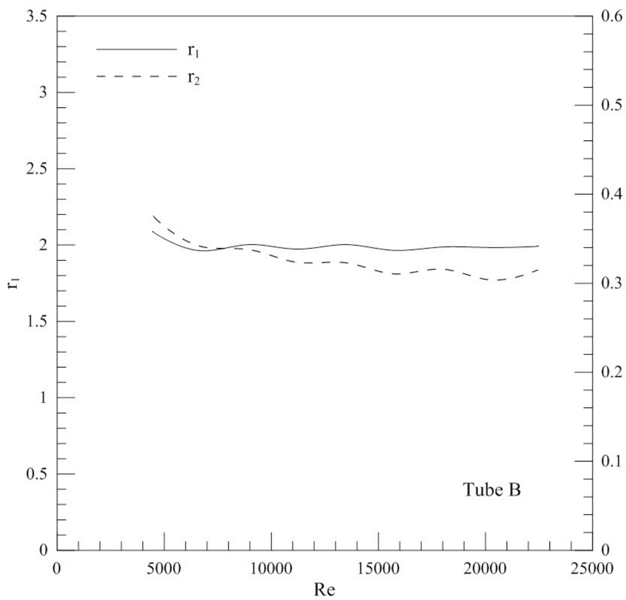

Figure 7: Nu enhancement index and mechanical energy consumption index for tube B.

The r_2 values for tubes A, B, C and D are also shown in Figures **6-9**, respectively. For tube A, the r_2 value is in the range of 0.275 to 0.3; for tube B, the r_2 value is in the range of 0.316 to 0.375; for tubes C, the r_2 value is in the range of 0.327 to 0.4; for tube D, the r_1 value is in the range of 0.316 to 0.394. For all the four tubes, the r_2 value decreases with an increase in the Re value. In addition, it is readily noticed that the r_2 value for tube A is much smaller than the other three. This means, using for heat transfer enhancement, tube A is less efficient than the other three.

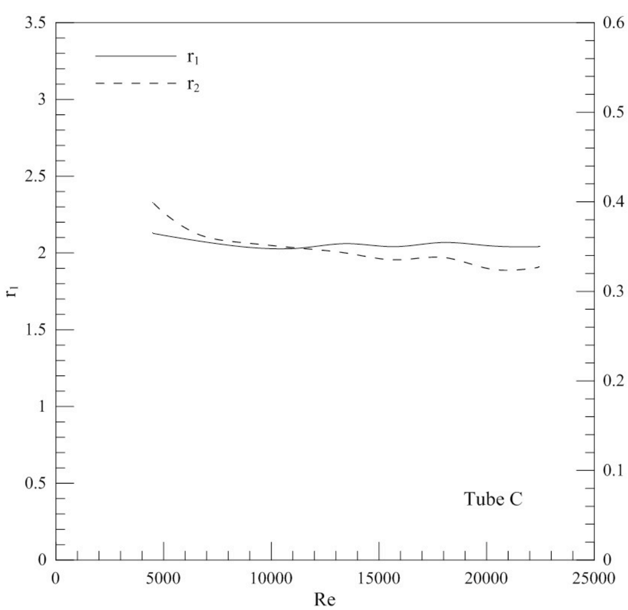

Figure 8: Nu enhancement index and mechanical energy consumption index for tube C.

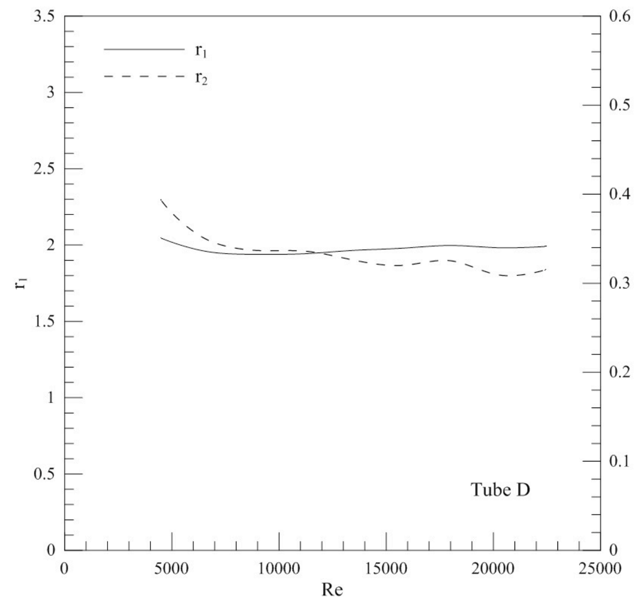

Figure 9: Nu enhancement index and mechanical energy consumption index for tube D.

5. UNCERTAINTY ANALYSIS OF EXPERIMENTAL DATA

An uncertainty analysis for the experimental f and Nu data is performed. In the fluid friction measurement, the maximum errors in pressure drop (P) and mean fluid velocity (V) are estimated to be $\delta(\Delta P)/\Delta P = 0.02$ and $\delta V/V = 0.04$, respectively. Based on the procedure suggested by Kline and McClintock [29], it is found that the uncertainty of the f values is 8.25%. In the heat transfer measurement, the maximum errors in air volumetric flowrate (Q), air inlet and outlet temperatures (T_i and T_o) and inner surface temperature of tube (T_w) are estimated to be $\delta T_i = \delta T_o = \delta T_w = 0.2°C$ and $\delta Q/Q = 0.04$. Using the data, the uncertainty of the Nu values is found to be 5.25%.

CONCLUSIONS

The transverse studs are quite effective for enhancing the heat transfer in circular tubes. Even at large Re values, the heat transfer enhancement is still quite pronounced. At $4000 \leqq Re \leqq 23000$, the r_1 value for the four heat transfer enhancing tubes is in the range of 1.93 to 2.13. It means that, due to the transverse rods, the Nu value is increased twofold.

The thread on the transverse stud is insignificant to the in-tube heat transfer enhancement, instead it could cause a 12% to 29% increase in the f value. As the Re value increases, the rise in the f value due to the thread decreases.

For tubes A, B and C, the deviation in Nu value is minor, whereas the f value for tube A is much larger than the other two. This also causes the r_2 value for tube A to be smaller than the r_2 values for the other two. The three tubes are different in the arrangements of threaded studs. In Tube A, the directions of the studs consecutively vary in the tube. The frequent variation in stud direction causes a significant increase in the pressure drop, but the heat transfer remains almost unchanged.

Based on the experimental result, it is concluded that, for achieving an efficient heat transfer enhancement, the studs need be arranged in the same form as tube C or tube B. In addition, the surface of the studs should be smooth.

NOMENCLATURE

A Surface area of smooth tube, m^2

c_p Constant-pressure specific heat of air, J/kg-K

d Outer diameter of transverse stud, m

D Tube diameter, m

f Darcy friction factor, $\Delta P/[(L/D_i)(\rho V^2/2)]$

h Average convection heat transfer coefficient, W/m^2-K

k Thermal conductivity, W/m-K

L Tube length, m

$\dot{m}$ Mass flowrate, kg/s

Nu Average Nusselt number, h_iD_i/k

p Pitch, m

ΔP Pressure drop, mm-H_2O or kg/m-s^2

Q Volumetric flowrate, m^3/s

r_1 Nu enhancement index, $Nu_{with\ studs}/Nu_o$

r_2 Mechanical energy consumption index, $(Nu/f)_{with\ studs}/(Nu/f_o)$

Re Reynolds number, $\rho VD_i/\mu$

T Temperature, ^oC

U_i Overall heat transfer coefficient based on inner surface area, W/m^2-K

V Mean fluid velocity, m/s

GREEK SYMBOLS

μ Dynamic viscosity, kg/m-s

ρ Density, kg/m^3

SUBSCRIPTS

a Air

i Inlet or inner

o Outlet or outer or smooth tube

s Wall

w Water

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