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Turbulent flow heat transfer and pressure loss in a double pipe heat exchanger with triangular fins

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Abstract
Experimental investigation of heat transfer and friction factor characteristics in a double pipe heat exchanger with triangular fins was studied. The working fluids were air, flowing in the annular pipe, and water through the inner circular tube. The test section is consisting of two parts. The first part is an insulated tube which has been manufactured from Perspex material of (54mm) inner diameter, (2000mm) length and (3mm) thickness. The second part is an internal copper tube without or with triangular copper fins. The smooth copper tube has (2250mm) long and (20mm, 22mm) inner and outer diameter respectively. The triangular fins were made of the copper with thickness of 0.3mm and 10mm height. They were installed on the straight copper tube section in three different cases (32, 27, and 22) mm distance between each two successive fins and (15mm) pitch between each two of fins. Air at various mass flow rates (0.001875 to 0.003133) kg/sec flows through annuli and water at Reynold's numbers ranging from (10376.9 to 23348.03) flows through the inner tube. The inlet cold air and hot water temperatures are 30^{o}C and 70^{o}C, respectively. The experimental results showed an increase in convective heat transfer coefficient by decreasing in distance between two fins and by increasing Reynold's number. This is due to increase in surface area. It was found that (Space=22mm) gives good heat transfer enhancement.

Keywords: Heat exchanger; finned tube; Friction factor.

1. Introduction
Heat exchangers are used in various processes ranging from utilization, conversion and recovery of thermal energy in various industrial, household and commercial uses. Some demotic examples implicate condensation in power, cogeneration plants, cooling in thermal processing of chemicals, agricultural products, sensible heating, steam generation, pharmaceutical, squandering heat recovery and liquid warming in industrialization. Improving heat exchanger's performance can make more frugal design of heat exchanger which can assist to produce energy, cost and material savings relevant to a heat exchange process. The need to raise the thermal performance of heat exchangers, just like that effecting energy, material and cost savings, led up to development and use of many techniques termed as "Heat transfer Augmentation". These techniques are also referred as "Heat Transfer Enhancement" or "Intensification". Augmentation techniques raise convective heat transfer by decreasing the thermal resistance in a heat exchanger. Enhancement techniques basically decrease the thermal resistance in a traditional heat exchanger by rising higher convective heat transfer coefficient without or with surface area increments. As a result of, the volume of a double pipe concentric tube can be incremented, or the pumping power

requirements can be decreased, or the exchanger's operating approach temperature difference can be reduced [1]. Use of heat transfer enhancement techniques leads to increase in heat transfer coefficient but at the cost of increase in pressure drop. So, while designing a heat exchanger using any of these techniques, analysis of pressure drop and heat transfer rate has to be done. Apart from these issues like, long term performance and detailed economic analysis of heat exchanger, should to be studied. To probe high heat transfer rate in an existing or new heat exchanger while taking care of the increased several techniques, pumping power have been suggested in modern years and are talk over [2,3].
Taborek Jerry [4] updated sketchy methods for concentric tube heat exchangers chiefly for longitudinal finned- tubes. Areas and conditions for the generality useful applications were outlined. Computation methods are presented for smooth concentric tube units, as well as finned- tube units. That led to new improvement in the paramount transition region with cut and twisted turbulence promoters. Equations for the average temperature difference for units with flow in series-parallel are also given. Wang et al. [5] accomplished a comparable work of eight finned–tube heat exchangers. They concluded that there is no effect of fin pitch on heat transfer performance for four row coils with Re >1000. But with Re< 1000, the heat transfer performance is highly dependent on fin pitch. An increase of heat transfer performance was found for two rows configuration with decreasing fin pitch. Yu and Tao [6] performed study to compute heat transfer and pressure drop characteristics of concentric tubes with wave–like longitudinal fins at regular axial heat input. This had been accomplished using air as the working fluid on both the entrance and fully-developed regions. The overall heat transfer characteristics for tubes with wave number (8, 12, 18 and 20) were tested at turbulent flow. It was acquired that under the three qualifications all the wave like finned pipes can improve heat transfer and the best wave number was 20. Discussion on the enhancement mechanism was conducted and general correlation was provided. Sahiti et al. [7] performed an experimental study of heat transfer enhancement by pin fins. Considerable enhancement was demonstrated by using small cylindrical pins on the surface of the heat exchanger. Simple relations of convective and conductive heat transfer were used to derive equation that shows the parameters effecting heat transfer enhancement. By employing pin element in their study, it was possible to enhance Nusselt's number compared with smooth tube heat exchanger. Chaudhari et al. [8] studied the influence of finned heat exchanger over a minus finned heat exchanger on overall heat transfer coefficient experimentally. The test rig section is a cross flow heat exchanger (automobile radiator) which built-up, inner the air flow channel (constant duct) and the profile of fin is louvered fin and tube kind. Water was used as coolant through tube (10 passes) with cross section. The fins were made of aluminum and tube is made of M S.the numbers of fins are 900. The overall heat transfer coefficient is studied for both heat exchangers with air velocity variation (3, 4, 5and 6) m/s, and coolant flow (180, 260, 340, 420 and 500) Lit/hr. The experimental results show that the Overall heat transfer rate of finned tube heat exchanger is better than minus finned tube heat exchanger. Now as the air velocity increased, heat transfer rate of finned tube heat exchanger is increments the Nusselt's number is also incremented because Nusselt's number is directly proportional to the heat transfer coefficient. Naphon et al. [9] investigated the influences of different pertinent parameters on the pressure drop and heat transfer characteristics experimentally. The test rig section includes of annular tube horizontal double pipe heat exchanger. The working fluid is water. The test rig section, made from a straight copper tube. The warm water pass through the inside tube, while the cold water pass through the annuli side. The varying of helical rib pitch to diameter (p/di) = (1.05, 0.78, and 0.63).the mass flow rates ranging from (0.01-0.10) kg/s for both warm water and cold water. The experimental results depict that the heat transfer coefficient tends to increment as the helical rib pitch reduced and the friction factors obtained from the tube with and less helical rib pitch are significantly higher than those with and utmost helical rib pitch.

2. Experimental test rig

Schematic diagram of the experimental set-up is shown in Figure 1. Experiments were carried out in a double-pipe heat exchanger. The hot water was flowing on the inner tube-side as shown in Figure 2 (copper tube, Di=20 mm, Do=22mm, L=2 m) and the heat transfer medium was passes, in counter-flow, through the annulus. Cold air flow rate ranging from (0.001875 to 0.003133) kg s^{-1}. Water at Reynold's numbers ranging from (10376.9 to 23348.03) flows through the inner tube. The inlet cold air and hot water temperatures are 30°C and 70°C, respectively. The inner tube is inserted inside the Perspex tube of (54mm) inner diameter and (60mm) outer diameter. The pressure drops across the test section were measured by using inclined U-tube manometer. Two rotameters were provided to indicate the approximate flow rate of the test section. Many of triangular fins are manufactured from copper is shown in Figure 3. They were installed on the external surface of tube by turning technique. Having (10mm) length of base, (10mm) height, (1mm) thickness, (32, 27, and 22) mm distance between each two successive fins and (15mm) pitch between each two fins. Thermocouples were fixed on both ends of the shell, to measure inlet and outlet temperatures on both exchanger sides. Tested heat exchanger consists of single pass type concentric tube heat exchanger.

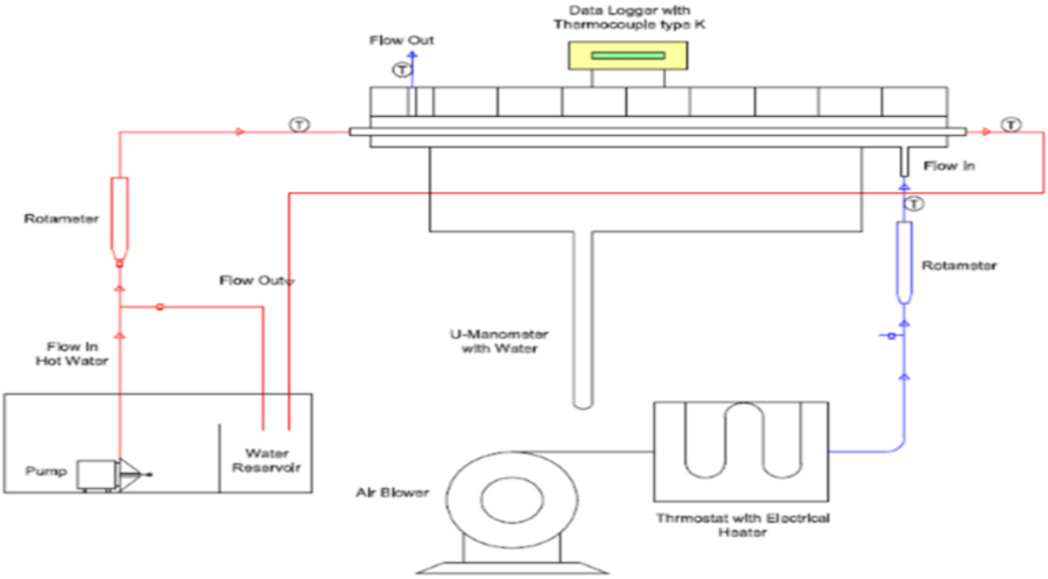

Figure 1. Schematic of test rig.

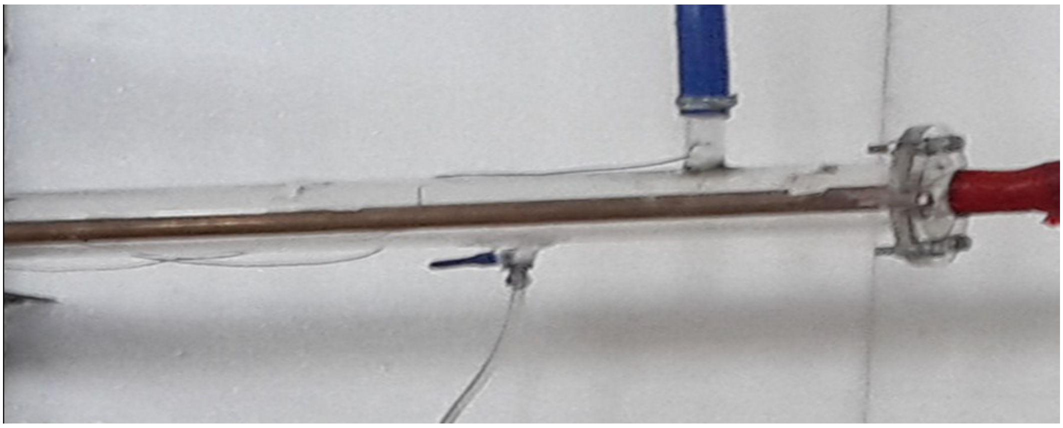

Figure 2. Plain test rig of heat exchanger.

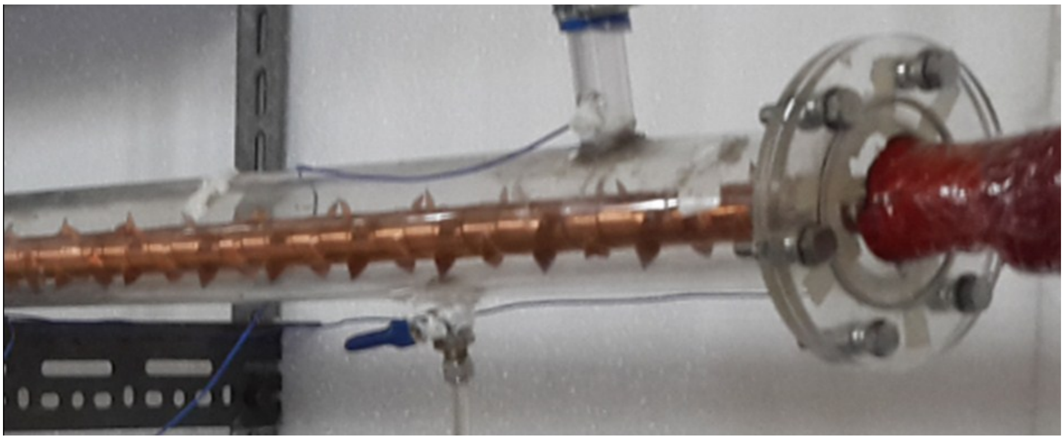

Figure 3. Enhanced test rig of heat exchanger.

3. Experimental procedure

The following procedure has been applied for the experiment after checking water and air sides. Then switching on the electric devices on test rig with water rotating in a cycle until achieving steady state. This is explained as follows:

Adjust air velocity by changing the gate in the blower. Adjust the water flow rate by controlling on both the main and bypass valve. After getting the desired flow rate in both sides and steady state condition, all the temperatures that have been measured carefully. The hot water source is maintained supplying water at 70°C and cold air at 30°C. Heat transfer coefficient is calculated by measuring the inlet and outlet cold air and hot water temperatures. The hot water at about 70°C was provided with a centrifugal pump and passed through the rotameter to the inner tube of heat exchanger. While, the cold air of 30 ºC temperature was pumped to the rotameter before the outer pipe of the heat exchanger. Four cases are covered by this work

Case one: Smooth type heat exchanger seen Figure 2

Case two: Enhanced type one (32mm spacing) is shown in Figure 4.

Case three: Enhanced type one (27mm spacing) is shown in Figure 5.

Case four: Enhanced type one (22mm spacing) is shown in Figure 6.

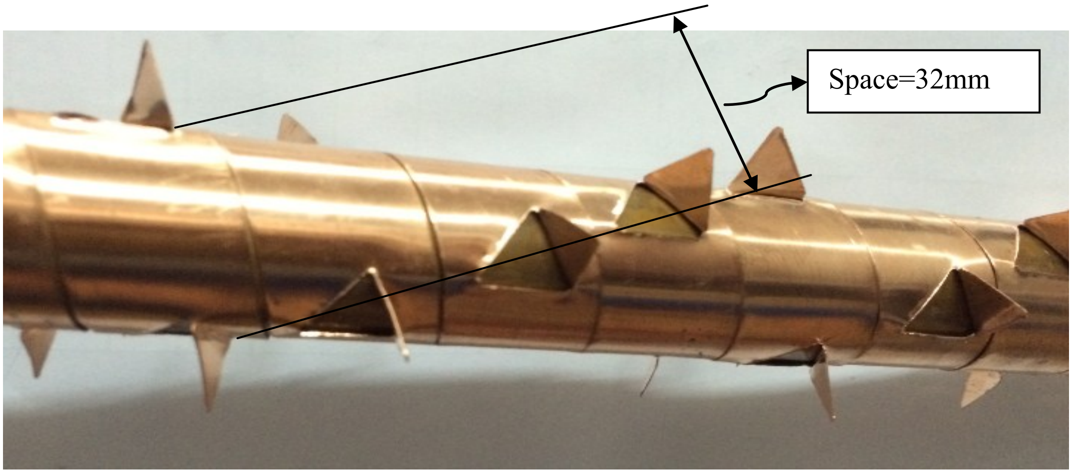

Figure 4. Enhanced type one (32mm spacing).

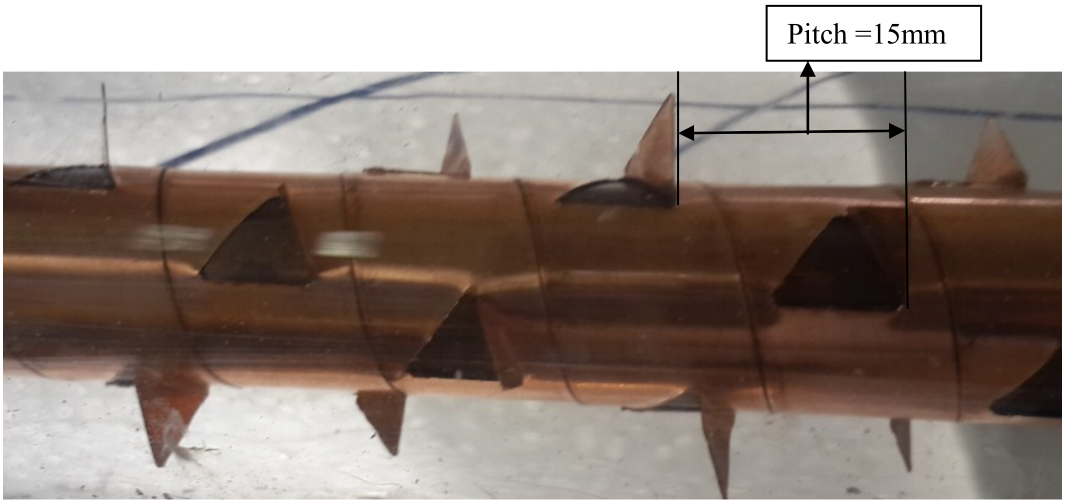

Figure 5. Enhanced type one (27mm spacing).

Figure 6. Enhanced type one (22mm spacing).

4. Data deduction

Heat absorbed by the cold air in the annulus, Qc can be written by

$$Q_c = \dot{m_c} cp_c (T_{co} - T_{ci}) \tag{1}$$

where Q is the heat dissipation (w), m˙ is the mass flow rate (kg/sec), Cp is the Specific heat and T is the temperature (°C). The Subscripts o, i and c is Indicates outer, inner and cold respectively. The heat supplied from the hot water, Qh can be determined by

$$Q_h = \dot{m_h} cp_h (T_{ho} - T_{hi}) \tag{2}$$

where Q is the heat dissipation (w), m˙ is the mass flow rate (kg/sec), Cp is the Specific heat and T is the temperature (°C). The Subscripts o, i and h is Indicates outer, inner and hot respectively. The average value of heat transfer rate, supplied and absorbed by both fluids, is taken for internal convective heat transfer coefficient calculation.

$$Q_{ave} = \frac{Q_c + Q_h}{2} \tag{3}$$

$$Q_{ave} = U A_i \Delta T_{LMTD} \tag{4}$$

$$\mathrm{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{\Delta T_1 - T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{5}$$

where

$$\Delta T_1 = T_{hi} - T_{co} \text{ And } \Delta T_2 = T_{ho} - T_{ci} \tag{6}$$

where U is the overall heat transfer coefficient (W/m^2. °C), A is tube surface area (m^2), ΔTLMTD is the log mean temperature difference (°C). Figure 7 shows the distribution of temperatures in counter flow heat exchanger.

$$A_i = \pi D_i L \tag{7}$$

where D is tube diameter (m),L is the length of tube (m)

The tube-side heat transfer coefficient hi is then determined (by neglecting the thermal resistance in the copper-tube wall) using

$$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_i} \tag{8}$$

where the tube side heat transfer coefficient hi was estimated by using the correlation of Dittus–Boelter [10].

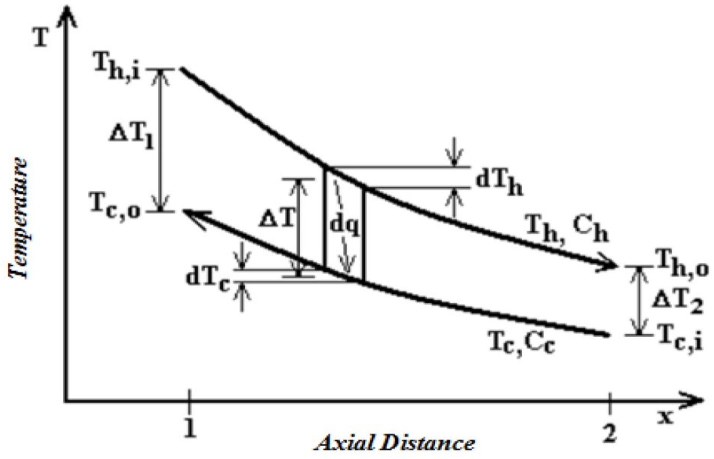

Figure 7. Distribution of temperatures in counter flow heat exchanger.

$$Nu_i = \frac{h_i D_i}{k} = 0.023 Re^{0.8} Pr^{0.3} \tag{9}$$

where k is thermal conductivity (W/m K).

$$h_i = \frac{k Nu_i}{D_i} \tag{10}$$

$$Nu_o = \frac{h_o D_H}{k} \tag{11}$$

where D_H, ($D_H = D_o - D_i$) is tube hydraulic diameter (m).

The local thermal conductivity k of the fluid is calculated from the fluid properties at the local mean bulk fluid temperature. The Reynolds number is based on the different flow rate at the inlet of the test section.

$$Re = \frac{\rho V D}{\mu} \tag{12}$$

where μ is the dynamic viscosity of the working fluid.
Friction factor, f can be written as:

$$f = \frac{\Delta P}{\left(\rho u^2/2\right)\left(L/D_{hu}\right)} \tag{13}$$

where u is mean velocity in the tube (m/sec). All of thermo-physical properties of the water are determined at the overall bulk water temperature. The overall enhancement ratio is defined as the ratio of the heat transfer enhancement ratio to the friction ratio at the same pressure drop. The overall enhancement ratio can be written as:

$$\left(\frac{Nu_f}{Nu_s}\right) = \left(\frac{f_f}{f_s}\right)^{1/3} \tag{14}$$

where Nu_f, f_f, Nu_s and f_s are the Nusselt numbers and friction factors for a tube configuration with and without fins respectively.

5. Validation

It may be noted that prior to actual data collection, the test setup was checked by conducting experiments for a plain tube. The Nusselt number determined from these experimental data were compared with the values obtained from the correlation of Dittus–Boelter [10] for the Nusselt number as show in Table 1. Comparison between the present experimental work and correlation from the previous work of the plain tube is presented in Figure 8. In the figure, results of the present work reasonably agree well with the available correlation within ±8% in comparison within in comparison with Nusselt number correlation of Dittus-Boelter.

Table 1. Experimental and analytical results for different Reynolds number.

Re	Experimental Nu	Analytical Nu
10376.9	45.377	49.662
12971.127	50.276	59.547
15565.35	62.947	68.173
18159.57	70.562	77.843
20753.8	79.989	86.497
23348.03	89.999	95.437

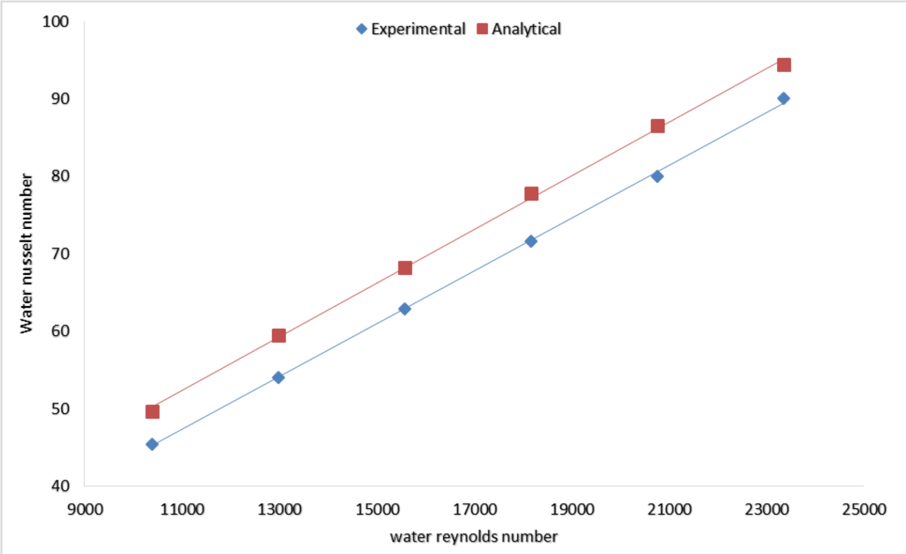

Figure 8. Comparisons of experimental data and empirical correlation of the plain tube for Nu.

6. Results and discussions

A comparison between the Nusselt numbers for water side of finned tube at three different cases (fins density) with plain tube alone is shown in Figure 9. The experimental results indicate that the Nusselt number in finned tube at three cases (case2 to case4) is larger by (73%, 76% and78.8 %) than that of smooth (case1) tube respectively. The increase in Nu may also be presented by calculating the overall enhancement ratio (Nuf/Nus)/ $(ff/fs)^{1/3}$as shown in Figure 10. When used the cases from case2 to case4 (fins density), the ratio of enhancement (Nuf/Nus)/ $(ff/fs)^{1/3}$ is decreased from (9.801 to 9.187) at 6 m^3/hour, (8.978 to 8.145) at 8 m^3/hour, and (8.231 to 7.193) at 10 m^3/hour. This decrease in ratio because increase in friction factor for these cases. Figure 11 shows the relation of the heat exchanger effectiveness with tube side Reynolds number for different cases (fins density). The figure shows that the effectiveness tends to decrease when Reynolds number increases, the air effectiveness decreased by (5%) as a result of the increase in air Reynold's. The air side effectiveness in finned tube with three cases (case2 to case4) is larger by (18%, 22% and26 %) than that of smooth tube (case1) respectively. Figure 12 shows the variation of the friction factor with the air Reynold's number for smooth and triangular finned tube with different cases (fins density). The friction factor tends to decrease (44%) by increasing air Reynold's number. The friction factor (or pressure drop) in the triangular finned tube gradually decreases with increasing distance between each two successive fins (fins density). The friction factor obtained from the tube with less fins distance are significantly higher than those with higher fins distance. Figure 13 depicts the relation between different water Reynold's number and heat dissipation for smooth and triangular finned tube. It was accomplished for different cases (fins density). It can be seen from these figure that heat dissipation of finned tube is higher than that of smooth tube due to increase of surface area. The enhancement of heat dissipation for triangular finned tube with case2, case3and case4 is (3.252 to4.502) (3.502 to4.904) (3.815 to 5.405) times than that of smooth tube (case1) respectively.

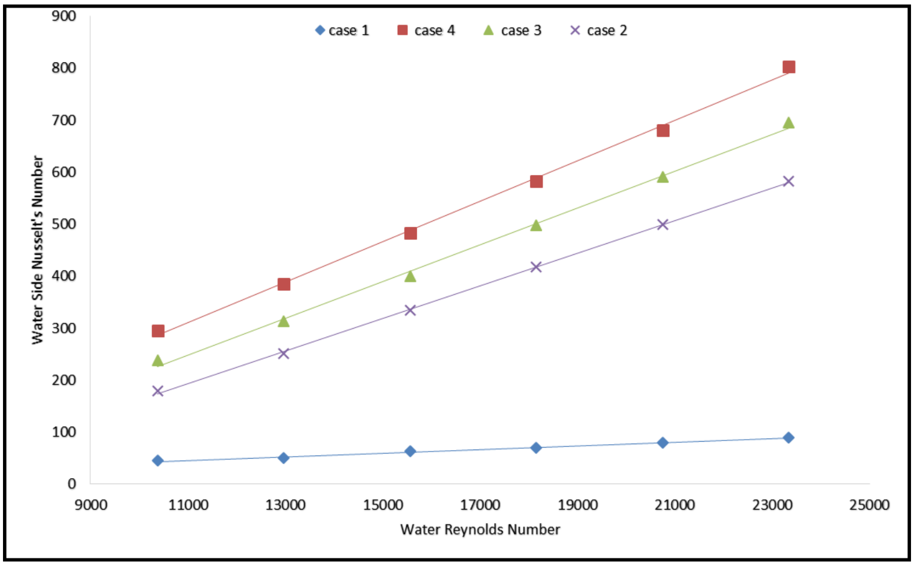

Figure 9. Variation of water Reynold's number with water side Nusselt's number for finned tube.

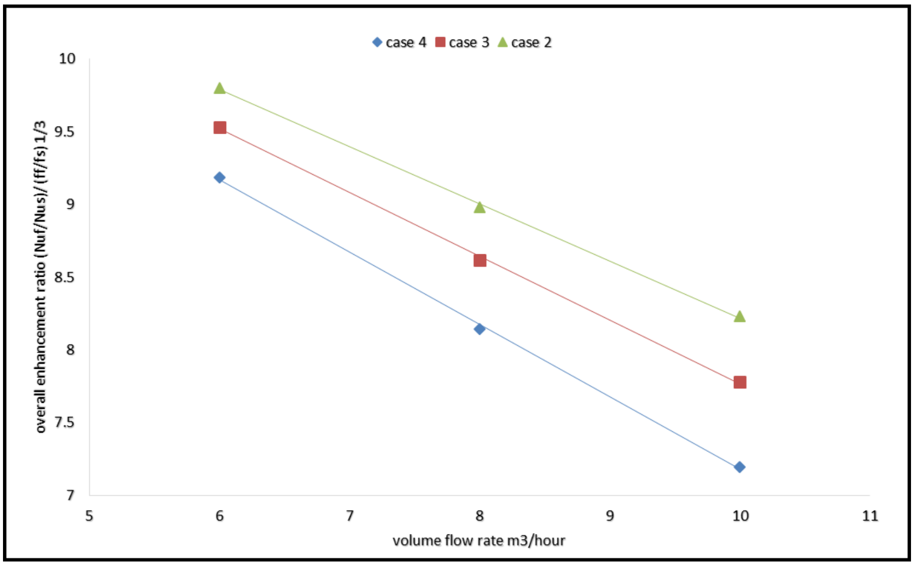

Figure 10. Effect of volume flow rate on overall enhancement ratio for air side.

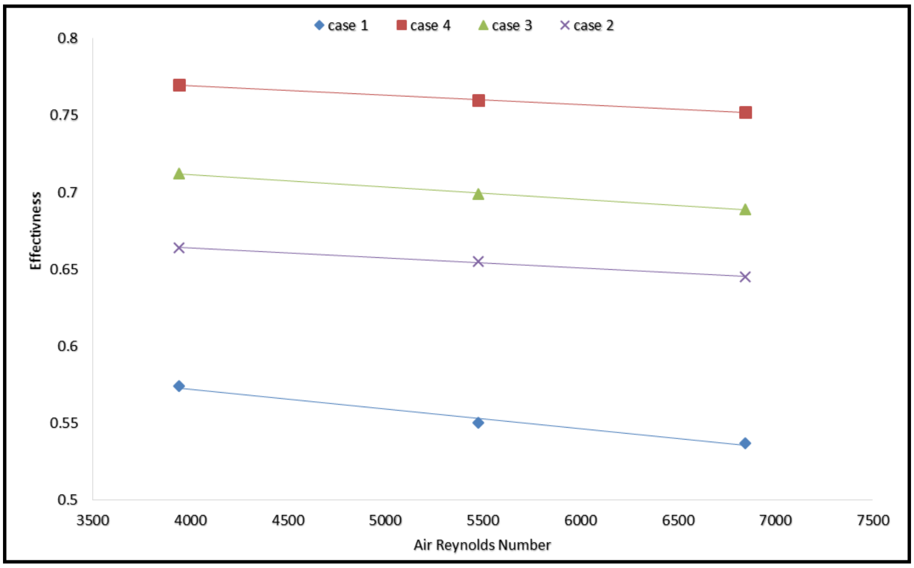

Figure 11. Variation of Air Reynold's number with Effectiveness for finned and smooth tube.

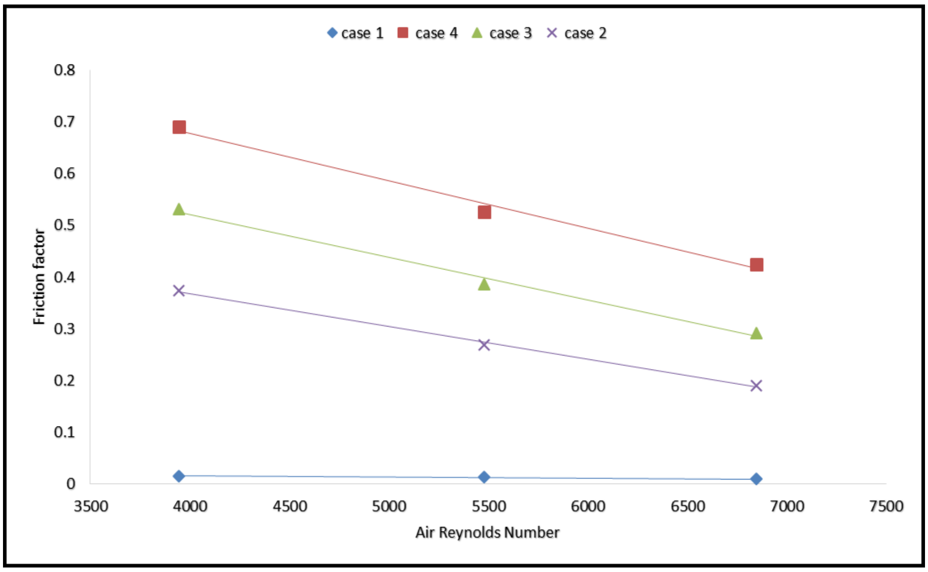

Figure 12. Variation of Air Reynold's number with Friction Factor for finned and smooth tube.

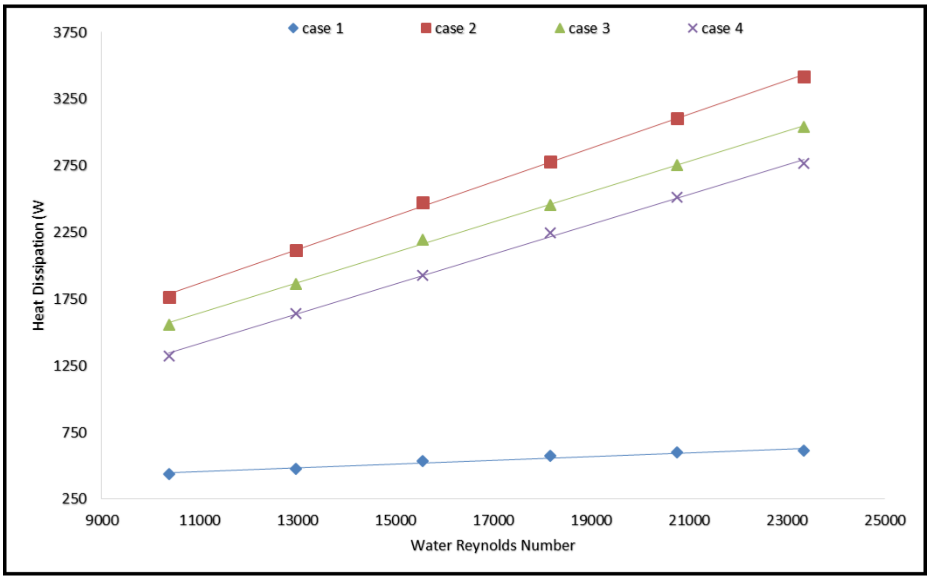

Figure 13. Effect of water Reynold's number on heat dissipation.

7. Conclusion

The heat transfer augmentation is apparent when adopting fins on outer surface of inner tube in present heat exchanger. This enhancement appears clearly in heat dissipation indicating (3.815 to 5.405) (3.502 to4.904) (3.252 to4.502) times than that of smooth tube respectively. The finned tubes provide higher Nusselt numbers than the smooth tube. In the lowest space (space=22mm), the average increment in Nusselt number is about 98% over the smooth tube heat exchanger. The friction factor decreases with Reynolds number increment and the friction factor in the finned tube gradually decreases with increasing distance between two fins (space). Significant enhancement has been found in heat exchanger effectiveness by adopting fins on outer surface of inner tube. Good enhancement for triangular fins has been concluded within the less distance between two fins.

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