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Experimental and numerical investigation to evaluate the performance of triangular finned tube heat exchanger

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Abstract
Experimental and numerical investigation has been performed in this work to evaluate the performance for triangular finned tube heat exchanger. Experimental work included designing and manufacturing of shaped triangular fins from copper material of (10mm) length, (10mm) height, (1mm) thickness, (22 mm) distance between every two fins shaped and (15mm) pitch between each two of fins which are install on the straight copper tube of (2m) length having (20mm) inner diameter and (22mm) outer diameter. The inner tube is inserted inside the Perspex tube of (54mm) inner diameter and (60mm) outer diameter. Cold Air and hot water are used as working fluids in the shell side and tube side, respectively. 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. Performance of (smooth and finned) tube heat exchanger was investigated experimentally. Experimental results showed that the enhancement of heat dissipation for triangular finned tube is (3.252 to4.502) times than that of smooth tube respectively. Numerical simulation has been carried out on present heat exchanger to analyze flow field and heat transfer using COMSOL computational fluid dynamic (CFD) package model. The comparison between experimental work and numerical results showed good agreement.

Keywords: Heat exchanger; Finned tube; COMSOL package.

1. Introduction
Materials and Energy saving considerations as well as economic incentives have lead up to an effort to produce more efficient heat exchange equipment. public thermal hydraulic goals are to decrease the size of a heat exchanger desired for a specified heat duty, to improve the capacity of an existing heat exchangers and to decrease the approach temperature difference for the process streams or to decrease the pumping power. The study of improved heat transfer rendering is denote to as heat transfer enhancement, intensification, or augmentation. In common, this means an increment in heat transfer coefficient [1]. One of the important kinds of double pipe concentric tube is Water to Air heat exchanger. This type of heat exchanger has various application such as: apartment Buildings and Condominiums, Residential Heating, Hybrid Systems, Air Conditioning, Dehumidification. S. Angela Ourivio Nieckele et al. [2] performed study to investigate the mean heat transfer coefficients and friction factors for turbulent flow through annular channels with pin fins attached to the inside wall of the annulus. The measurements were arrangement by averages of a double-pipe heat exchanger. The number of pins in a same cross-section (transversal row) was 8, while the number of transversal rows was 70, totaling 560 pins. The working fluids were air, flowing in the annular duct, and water through the internal circular

tube. The range of the Reynolds number, for the air-side flow, was variation from (13,000 to 80,000). The experiments were performed in an open-loop air flow circuit. The warm fluid (water) operates in a closed-loop circuit and its mass flow rate is controlled by the valve. The water circuit is designed in such way that the heat exchanger works in counter flow mode. The Experiments results showed the for the same Reynolds numbers, the Nusselt numbers for the pinned annulus are roughly 200% more than those for the smooth annular duct. Yu et al. [3] performed to compute the heat transfer and pressure drop characteristics of tubes with internal wave-like longitudinal fins. They conducted two cases for this work were carefully examined, using air as a working fluid. For the tube of type A, since the inner channel of the insertion is not blocked, its flow cross-section area only differs mildly from that without the insertion. While for the tube of type B the cross-section of the inner tube is totally blocked. The test tube is arranged in such a way that its axis is in a horizontal plane. The test tube and fins are made from copper. The wave-like fins are within the annulus and span its full width. There are total 20 waves. The outer tube was electrically heated. Pressure taps were no uniformly distributed in 13 cross-sections along the test tube axis. The temperatures of the test tube surface were measured by 39 copper-constantan thermocouples. The range of Reynold's number was varied from 9*102 to 3.5*103. Results showed that the wave-like fins enhance heat transfer significantly with the blocked case by 36% higher than that of unblocked tube. Mohsen Sheikholeslami etc [4] investigated flow and heat transfer in a horizontal annular tube double pipe heat exchanger is experimentally. The inside tube is made from copper, while the outer tube is made from PVC. The temperature were measured with Stainless steel thermocouples. Pressure losses were determined by using U-manometers that were stuffed with water for the air side. The flow is designed as according to countercurrent flow and turbulent flow. Warm water was passed through the inside pipe, while cold air was flowing through the annulus duct. Water at various volumetric flow rates in the range of from 120 to 200 (Lit/h) flows through the inside tube and in the temperature of water range from 70 to 90 °C. The experimental Results depict that Nusselt number in water side increments with increments of temperature of the water and flow rate of water while opposite trend is observed for Nusselt number in air side. Friction factor increments with increment of inlet temperature and inlet velocity of water. Dong et al. [5] performed study to accomplish the influence of cribriform annular finned–tube on the convective heat transfer of annular finned–tube heat exchanger. The convective heat transfer coefficient was increment by 3.55% and 3.31% and increment in pressure drop by 0.68% and 2.08% for two- hole and four -hole cases respectively. Iqbal et al. [6] investigated of the optimal longitudinal triangular fins (initial shape) close fitting to the outer surface of the inner pipe enclosed. Within a laminar, incompressible and fully- developed flow in the shell of two concentric circular pipes at regular heat flux. The fins are straight, non-porous, regularly distributed about the periphery of the inside pipe and are assumed to be made up of highly conductive material. The fin profile is showed by Piecewise Cubic Hermite Interpolating Polynomial (PCHIP). The results depict that the optimal shape is strongly dependent on the ratio of radii, the number of fins, and the number of control points and the characteristic length. The increment in the Nusselt number up to 138%, 263% and 312% over the traditional fins of trapezoidal, parabolic and triangular shapes for equipollent diameter however 212%, 90% and 59% respectively for hydraulic diameter. Mayank Bhola et al. [7] investigated modeled and simulated for annular tube heat exchanger with and minus insert. Design process for heat exchanger and insert has been carried out in SOLIDWORKS, fluid domain is formed in ANSYS workbench for steel. Inlet temperature of warm water and cold air are (60 - 26) °C respectively. CFD computations were done for three various mass flow rate of air (0.0079, 0.018, and 0.036) kg/s and water (0.0216, 0.36091, and 0.6767) kg/s for both counter flow and parallel flow condition. The profile tape of insert is rectangular insert. The results have illustrative a good acceptable. The inside convective heat transfer coefficient for rectangular insert of this type is roughly 25% higher than for smooth tube. Counter flow configuration is high efficient than parallel flow. And depict increment in heat transfer coefficient of roughly 27%. Shuai and Chang [8]. Performed three–dimensional numerical simulation on integral pin–fin tube heat exchanger using fluent software. They verified that the fluent software on integral pin–fin tube numerical simulation method is feasible. Fluent software was proven to be feasible for numerical simulation on integral pin–fin tube and be effective tools to design and develop heat exchange tubes. Mon and Gross [9] investigated a numerical analysis of pressure drop and heat transfer characteristics on the air side of the circular –finned tube heat exchangers. They studied the influences of fin spacing on (4–row) circular –finned tube parcels in staggered and in–line configurations. The results detection a paramount point of opinion for the flow and the local heat transfer characteristics of the circular –finned tube bundles and the boundary layer improvement on the fin and tube surfaces according to the fin spacing to height ratio. The heat transfer coefficient of the staggered arrangements was increment with height ratio up to (0.32) and then preserved constant. For in–line configuration, the heat transfer coefficient increment over the whole realization range and the pressure drop was reduced for both arrays when height ratio increment. Good agreement was noted of heat transfer with experimental correlations. Mir et al. [10] carried out numerical simulation of studying laminar, forced convection heat transfer in the finned annulus for the case of fully- developed incompressible flow. The simulation was corresponding to thermal boundary condition of regular heat input per unit axial length with peripherally regular temperature at any cross section. They used boundary fitted curvilinear coordinates to overcome the singularities presented by fin tip. In the solution domain, different heat transfer and fluid flow were accomplished for a range of values of the ratio of radii of inside and outer tubes, fin height and number of fins. The results depict a good agreement comparison with the literature results.

2. Numerical simulation

Numerical simulation by using COMSOL (5.0) computational fluid dynamic (CFD) package model has been conducted for performing numerical simulation across the heat exchanger using three–dimensional model. The solution of conservation continuity, momentum and energy equations are used to analyze the flow field inside the heat exchanger. A comparison of heat transfer for (smooth, triangular) finned tube heat exchanger is carried out. AutoCAD is used to draw the geometries of this work with and without consist triangular finned tube heat exchanger with inlet and outlet portions, the outside flow is confined by insulating tube having inlet and outlet portions as shown in Figures 1 and 2.

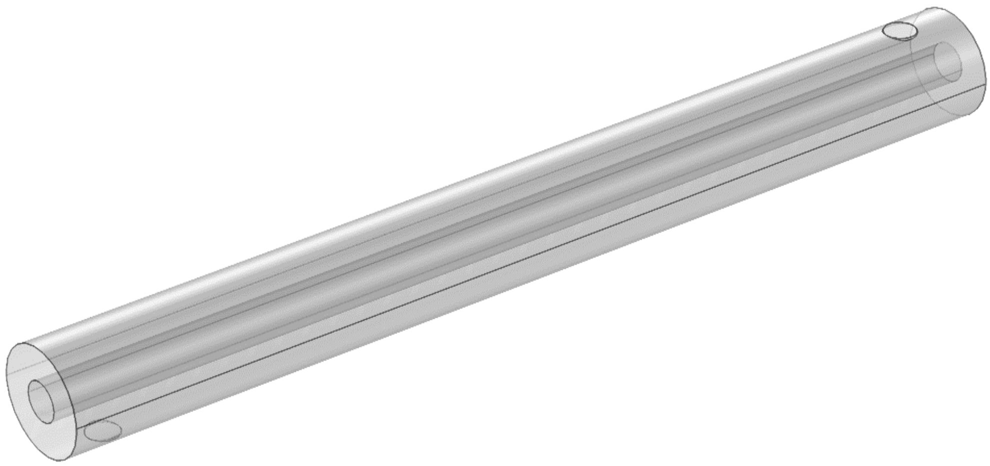

Figure 1. CFD domain of smooth tube heat exchanger

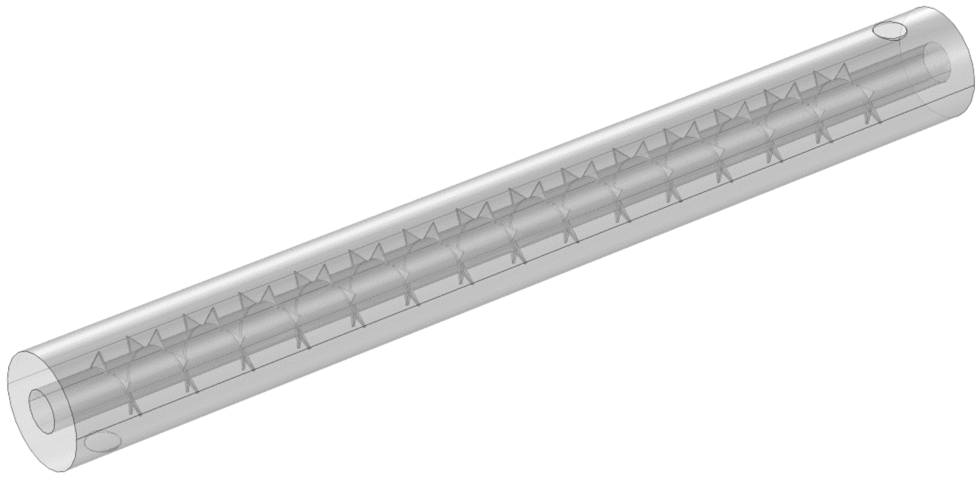

Figure 2. CFD domain of triangular finned tube heat exchanger

3. Governing equations

The governing differential equation for the fluid flow is given by Continuity equation or mass conservation equation, Navier Stokes equation or momentum conservation equation and energy conservation equation [11].

3.1 Continuity equation

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho V)}{\partial y}+\frac{\partial(\rho W)}{\partial z}=0 \tag{1}$$

where (u, v, w) Velocity component in x, y, z directions

3.2 Navier Stokes equation

$$\rho\left(u\frac{\partial u}{\partial x}+V\frac{\partial u}{\partial y}+W\frac{\partial u}{\partial z}\right)=\rho X-\frac{\partial p}{\partial x}+\frac{1}{3}\mu\frac{\partial}{\partial x}\left(\frac{\partial u}{\partial x}+\frac{\partial V}{\partial y}+\frac{\partial W}{\partial z}\right)+\mu\nabla^2 u \tag{2a}$$

$$\rho\left(u\frac{\partial V}{\partial x}+V\frac{\partial V}{\partial y}+W\frac{\partial V}{\partial z}\right)=\rho Y-\frac{\partial p}{\partial y}+\frac{1}{3}\mu\frac{\partial}{\partial y}\left(\frac{\partial u}{\partial x}+\frac{\partial V}{\partial y}+\frac{\partial W}{\partial z}\right)+\mu\nabla^2 V \tag{2b}$$

$$\rho\left(u\frac{\partial w}{\partial x}+V\frac{\partial w}{\partial y}+W\frac{\partial w}{\partial z}\right)=\rho Z-\frac{\partial p}{\partial z}+\frac{1}{3}\mu\frac{\partial}{\partial z}\left(\frac{\partial u}{\partial x}+\frac{\partial V}{\partial y}+\frac{\partial W}{\partial z}\right)+\mu\nabla^2 W \tag{2c}$$

where ρ is the density(Kg/m3), μ is the dynamic viscosity (Kg/ms), ∇ is the vector differential operator

3.3 Energy equation

$$\rho c_p\left(u\frac{\partial T}{\partial x}+V\frac{\partial T}{\partial y}+W\frac{\partial T}{\partial z}\right)=\left(u\frac{\partial P}{\partial x}+V\frac{\partial P}{\partial y}+W\frac{\partial P}{\partial z}\right)+K\nabla^2 T+\mu\emptyset \tag{3}$$

where *Cp* is the specific heat (J/kg-K), *T* is the temperature (K),*Φ* is the Rayleigh dissipation factor

$$\emptyset=2\left[\left(\frac{\partial u}{\partial X}\right)^2+\left(\frac{\partial V}{\partial y}\right)^2+\left(\frac{\partial w}{\partial z}\right)^2\right]+\left[\left(\frac{\partial u}{\partial y}+\frac{\partial V}{\partial X}\right)^2+\left(\frac{\partial V}{\partial z}+\frac{\partial W}{\partial y}\right)^2+\left(\frac{\partial W}{\partial X}+\frac{\partial u}{\partial z}\right)^2\right]\ -\frac{2}{3}\left[\frac{\partial u}{\partial x}+\frac{\partial V}{\partial y}+\frac{\partial W}{\partial z}\right] \tag{4}$$

4. Boundary conditions

Velocity inlet was specified for inner and annuli sides during this study. On the other hand the temperature inlet of the inner tube is (70) °C while in annuli is (30) °C.

5. Mesh generation

Unstructured mesh is used in the present study to discretize the computational domain into a finite number of control volumes by using the finite–volume scheme. Structured mesh is ruled out because it is favorable for easy cases and it becomes insufficient and time consumed for complicated geometries. The model was meshed by using COMSOL (5.0) computational fluid dynamic (CFD) package. The refinement and generation of mesh system are very crucial to predict the heat transfer in sophisticated geometries. Thus both the density and distribution of the mesh lines play distinct roles for accuracy. Complete mesh consists of 39435 domain elements, 5588 boundary elements, and 400 edge elements for plain tube and mesh consists of 165268 domain elements, 14788 boundary elements, and 2857 edge elements for triangular finned tube as shown in Figures 3 and 4.

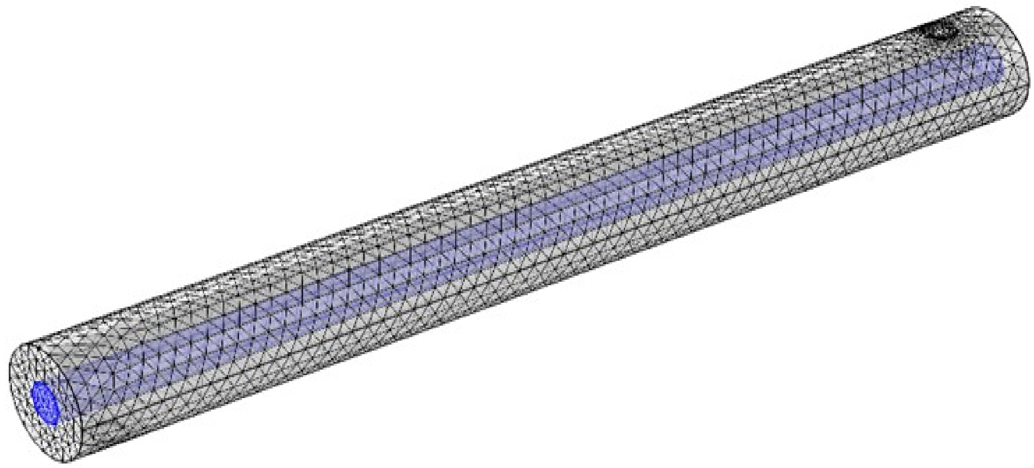

Figure 3. CFD meshing of smooth tube heat exchanger

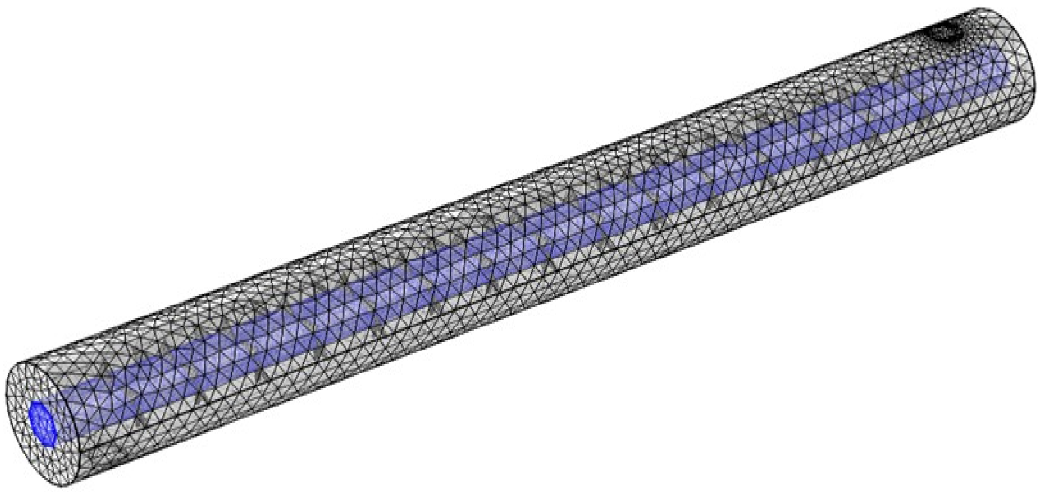

Figure 4. CFD meshing of finned tube heat exchanger

6. Experimental test rig

Figures 5 and 6 show the schematic diagram and photograph of test rig of test section. Test rig consists of test section, air and water supply system, measuring devices and supplements. Various kinds of measuring devices have been used such as digital anemometer, temperature recorder, water flow meter, thermocouples and temperature probes. The test section contains two parts, the first consists annuli tube has been manufactured from Perspex material of (54mm) inner diameter, (2m) length and (3mm) thickness. The second part is an internal copper tube without or with triangular copper fins. The smooth copper tube has (2m) long and (0.02m, 0.022m) inner and outer diameter respectively. Many triangular fins are manufactured from copper, they are install to the external surface of tube by technique having (10mm) length, (10mm) height, (1mm) thickness and (15mm) distance between every pair and (22mm) pitch between each two pairs of fins. A water pump is used for pumping the water in pipes through the water cycle and test section. It has (30 liter/minute) volumetric flow rate and centrifugal blower is employed to provide air for the test section through pipe with the power of 0.75 kW. The flow rates of the water were adjusted with valves and measured with rotameter. The experimental work was repeated for both countercurrent flow modes at various Reynolds numbers of water and air.

7. Validation

The comparison for Nusselt number, the experimental data are in good agreement with existing correlations, which are Dittus–Boelter correlation. It is noted that the Nusselt number differs by up 7%

from the Dittus– Boelter correlation as show in Table 1. In addition, the experimental results of the present study are correlated with Nusselt number as follow,

$$Nu = 0.023 Re^{0.8} Pr^{n} \quad \text{For } 2300<Re<1.25*10^{4} \tag{5}$$

where *Nu* is the Nusselt number, *Re* is the Reynolds number and Pr is the Prandial number

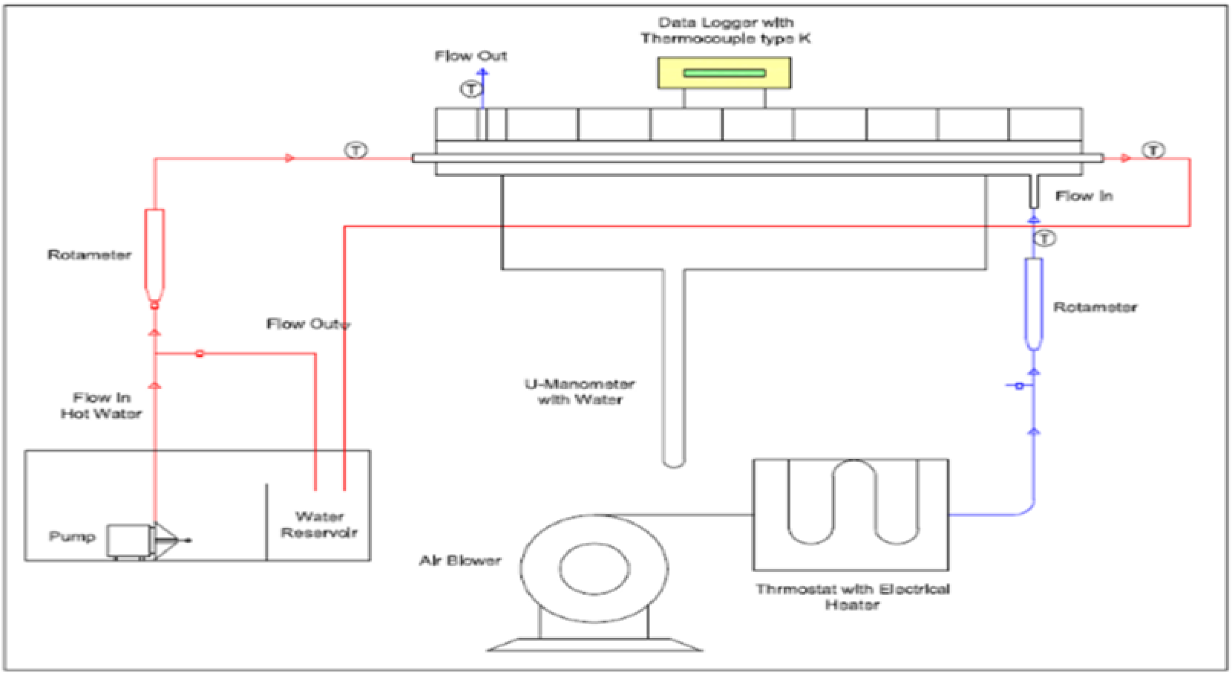

Figure 5. Schematic of test rig

Figure 6. Photograph of test rig

$$Q_h = \dot{m}_h cp_h (T_{ho} - T_{hi}) \tag{6}$$

where Q is the heat dissipation (w), $\dot{m}$ 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.

$$\mathrm{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} = \frac{\Delta T_1 - T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{7}$$

$$\Delta T_1 = T_{hi} - T_{co} \quad \text{and} \quad \Delta T_2 = T_{ho} - T_{ci} \tag{8}$$

where ΔTLMTD is the log mean temperature difference (°C). Figure 7 shows the distribution of temperatures in counter flow heat exchanger.

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

Re	Experimental Nu	Analytical Nu	CFD Nu
10376.9	45.377	49.662	46.086
12971.127	50.276	59.547	51.179
15565.35	62.947	68.173	63.734
18159.57	70.562	77.843	71.895
20753.8	79.989	86.497	81.247
23348.03	89.999	95.437	92.981

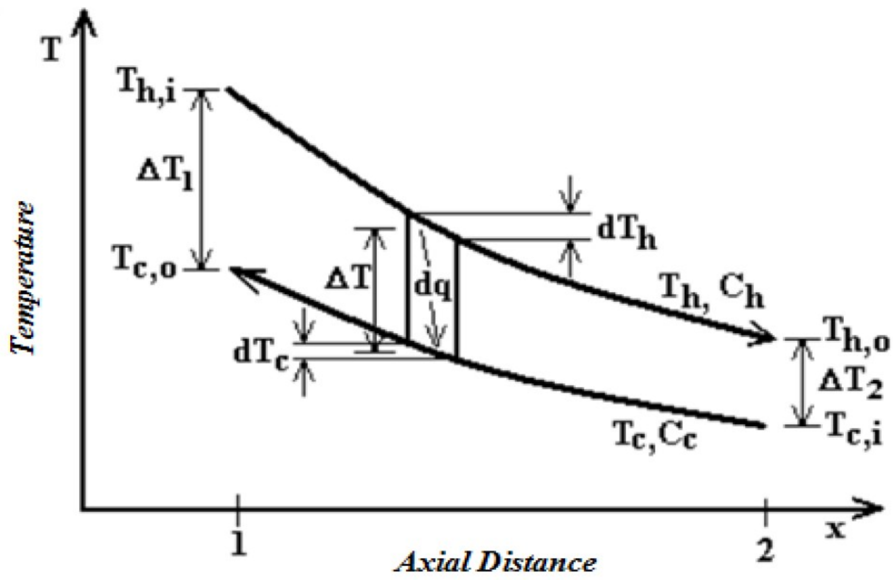

Figure 7. Distribution of temperatures in counter flow heat exchanger

$$h_i = \frac{Q}{A(\Delta T)_{LMTD}} \tag{9}$$

where *hi* is the heat transfer coefficient (W/m^2 K) of the tube, *A* is tube surface area (m^2).

$$\mathrm{Nu}_i = \frac{h_i D}{k} \tag{10}$$

where *D* is tube diameter (m) and *k* is thermal conductivity (W/m K).

8. Results and discussions

Figure 8 shows the effect of different water Reynold's number on the inner Nusselt's number. It can be seen from these figures that water side Nusselt's number increases due to turbulence generated by increasing of water Reynold's number indicating maximum increase of (67%). Figure 9 show the variation of the air side temperature difference with the air mass flow rate for smooth and triangular finned tube. The air temperature difference decreases by (5%) as a result of increase air mass flow rate. The air side temperature difference in finned tube is larger by (25%) than that of smooth tube due to enhancement by increasing the surface area. Figure 10 illustrate the variation of the water side temperature difference with the water Reynold's number for smooth and triangular finned tube. The water side temperature difference tends to decrease (20%) by increasing water Reynold's number. Figure 11 disclose the relation between different water Reynold's number and heat dissipation for smooth and triangular finned tube. It can be seen from Figure 11 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 is (3.252 to4.502) times than that of smooth tube respectively. Figure 12 show the relation between different air Reynold's number and effectiveness for smooth and triangular finned tube The figure show that the effectiveness tends to decrease with increasing Reynolds number, which the air effectiveness decreases by (5%) as a result of the increase in Air Reynold's. The air side effectiveness in finned tube is larger by 18% than that of smooth tube. This because that the effectiveness decease with increasing the hot water mass flow rate. Experimental results of test rig are validated in this study by numerical simulation produced by COMSOL (5.0) computational fluid dynamic (CFD) package model as shown in Figure 13. In this figure, the Nusselt number of inner tube has been illustrated with Reynold's number. Good agreement is observed with 4% difference between experimental and numerical results. Figures14 to 17 shows the temperature and velocity contours of smooth tube heat exchanger at different axial distances. It can be noted that the maximum air and water temperatures appears at Z/d = 1 while the minimum temperatures appears at Z/d = 9. It can be seen also when observing Z/d = middle, that changing water Reynold's number will not have a significant effect on air side. It seems that the air in annuli has the main reason for this behavior. This is because air thermal conductivity is small. Velocity contours in these figures at air side show that air inlet velocity distribution at Z/d = 2, 3, 4 are uniform but at Z/d=1,5are irregular because effect the inlet and outlet zone of air. It will also tend to decrease inside heat exchanger near the walls of inner and outer tubes. Velocity contours in these figures at water side show that the water velocity at the center increases while the velocity gradient near the wall decreases with increasing axial distance. Figures 18 to 21 reveal the temperature and velocity contours of triangular finned tube heat exchanger for specified water Reynold's numbers at various axial distances ratio. Results showed a significant heat transfer augmentation in heat exchanger for air and water sides. The effect of adopting fins on heat transfer enhancement is apparent in both sides. Heat transfer enhancement behavior is clear when increasing water Reynold's number. Increasing mass flow rate of air tends to increase water side temperature difference along axial distance and decreasing in air side temperature difference. Velocity contours in these figures reveal that the air velocity increases at the center. The effect of wall on decreasing the velocity in air and water sides is apparent in these figures.

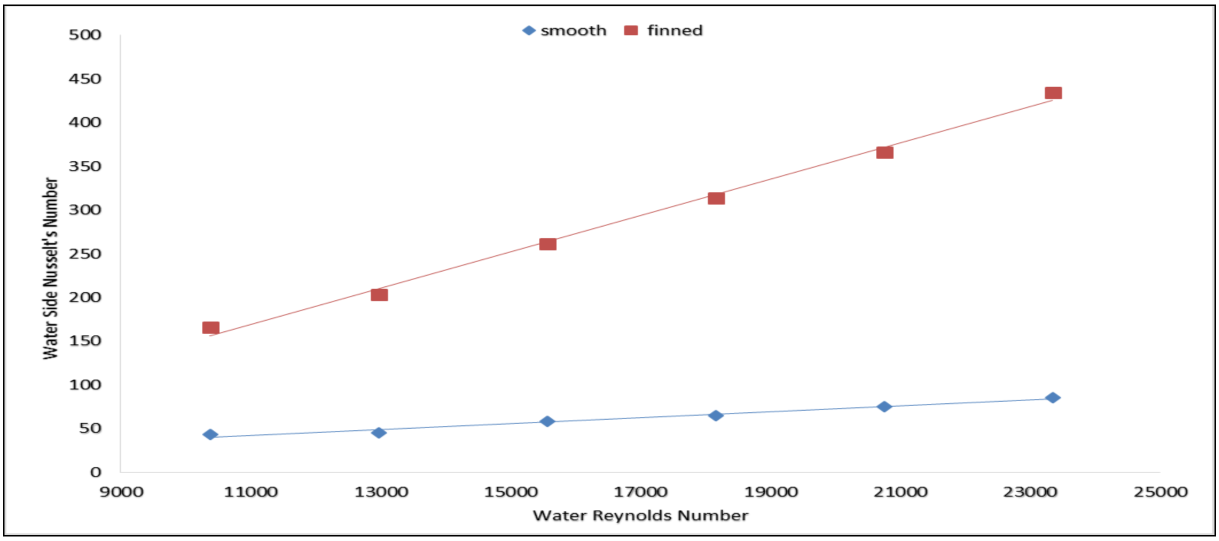

Figure 8. Variation of water Reynold's number with water side Nusselt's number

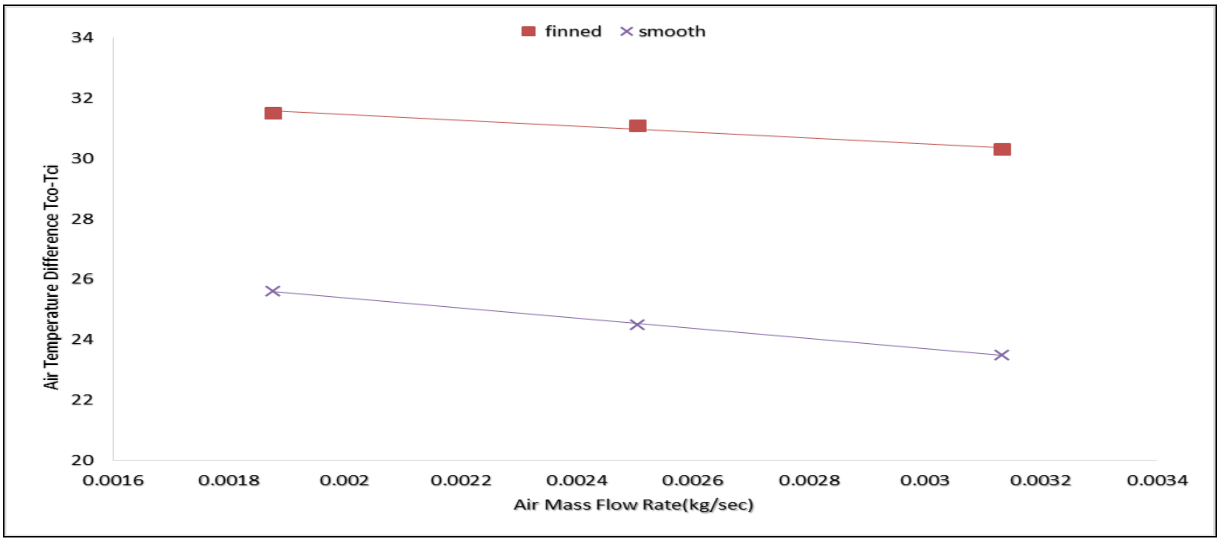

Figure 9. Effect of air mass flow rate on air temperature difference

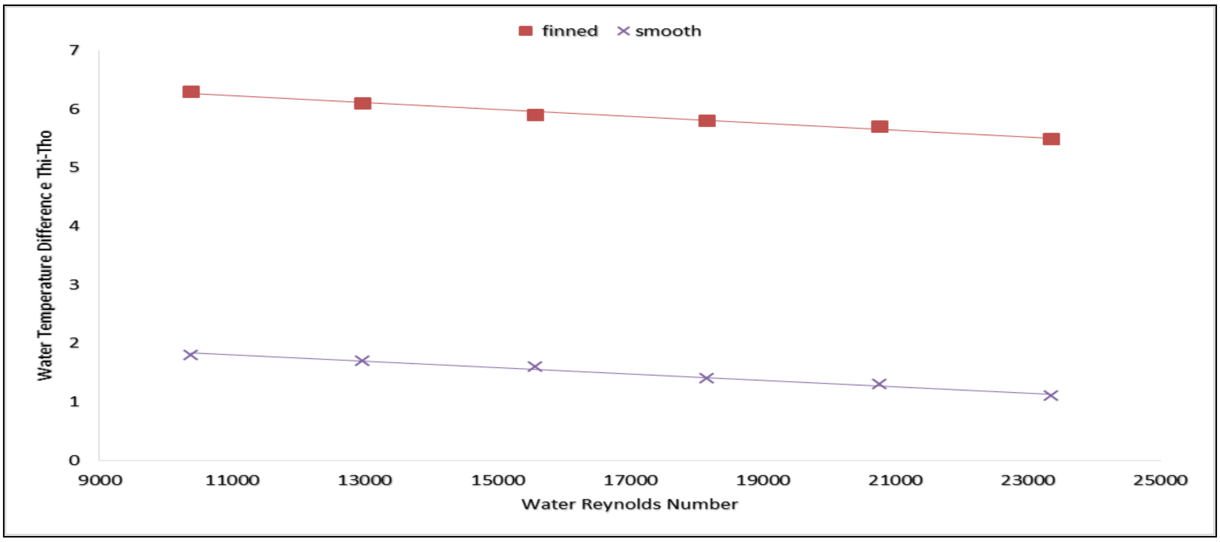

Figure 10. Effect of water Reynold's number on water temperature

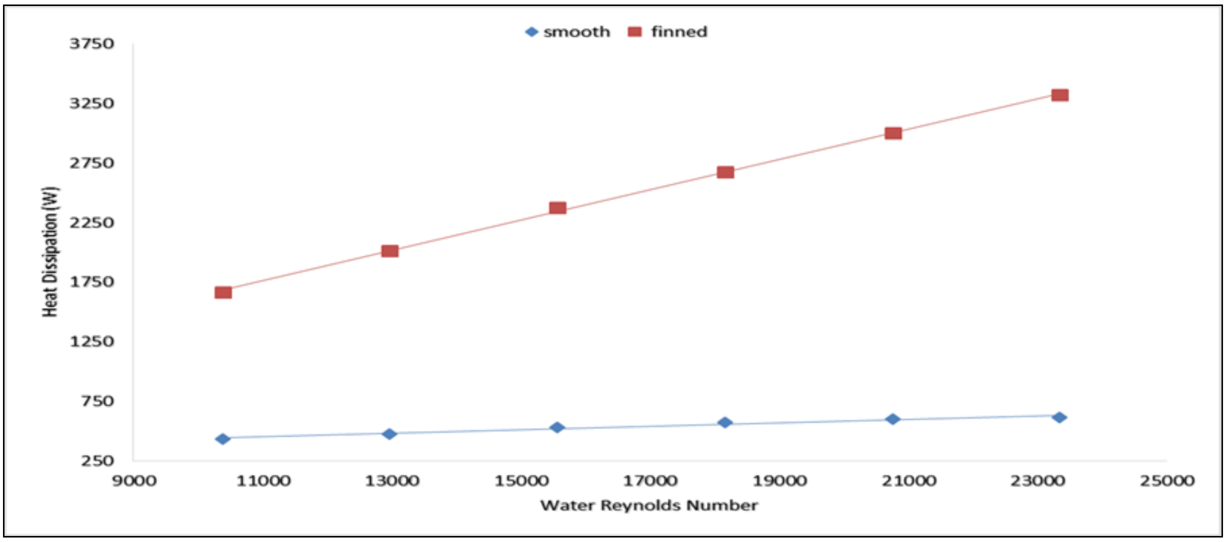

Figure 11. Effect of water Reynolds number on heat dissipation

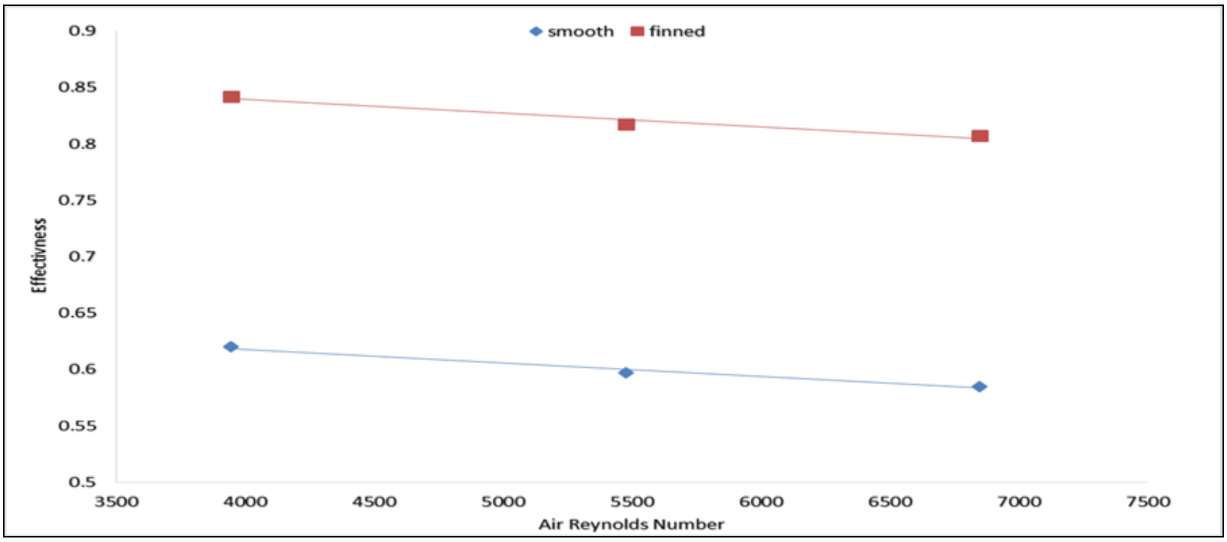

Figure 12. Variation of heat exchanger effectiveness with air Reynolds number

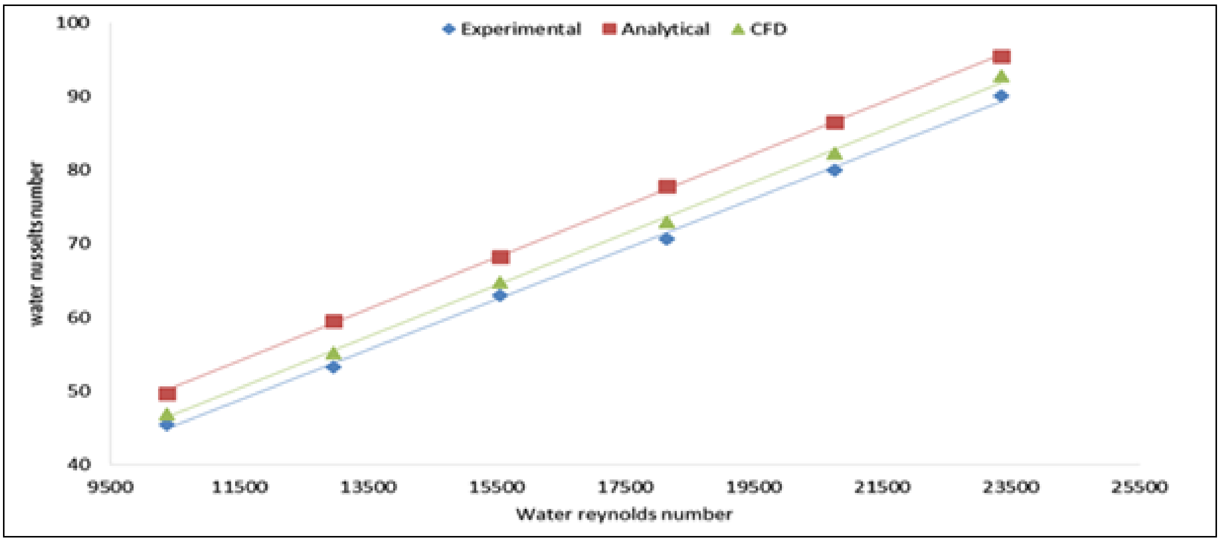

Figure 13. Comparisons of CFD and experimental data and empirical correlations of the plain tube for Nu

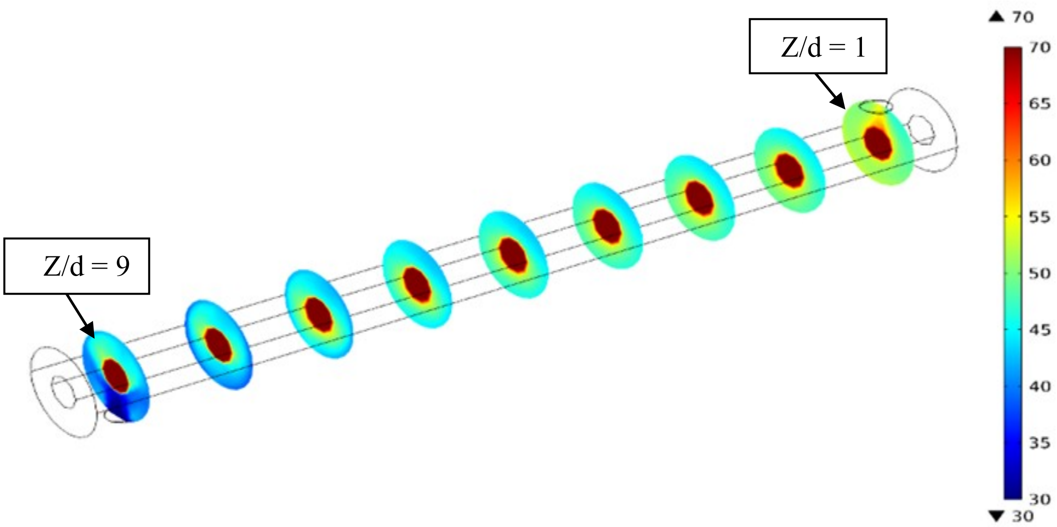

Figure 14. Temperature contours at various axial distances for smooth tube heat exchanger

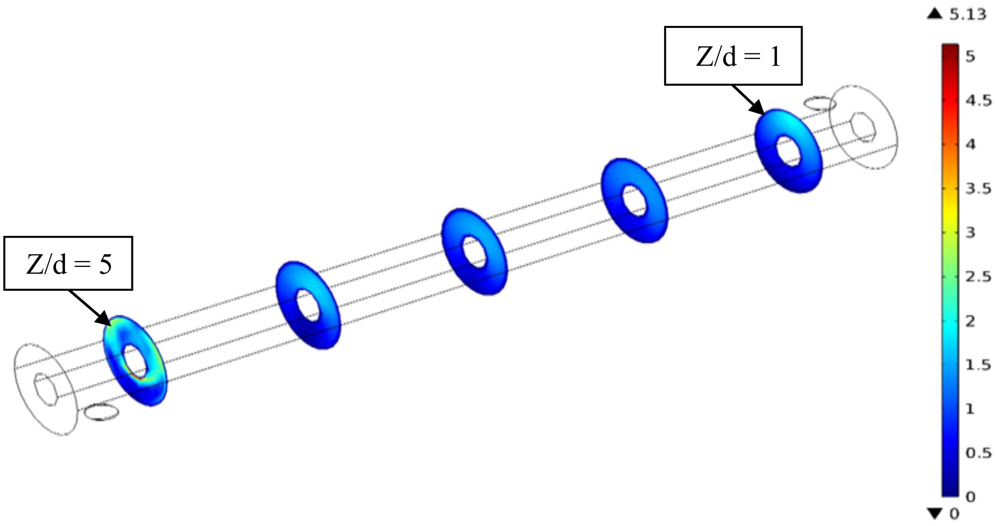

Figure 15. Velocity contours at various axial distances for smooth tube heat exchanger

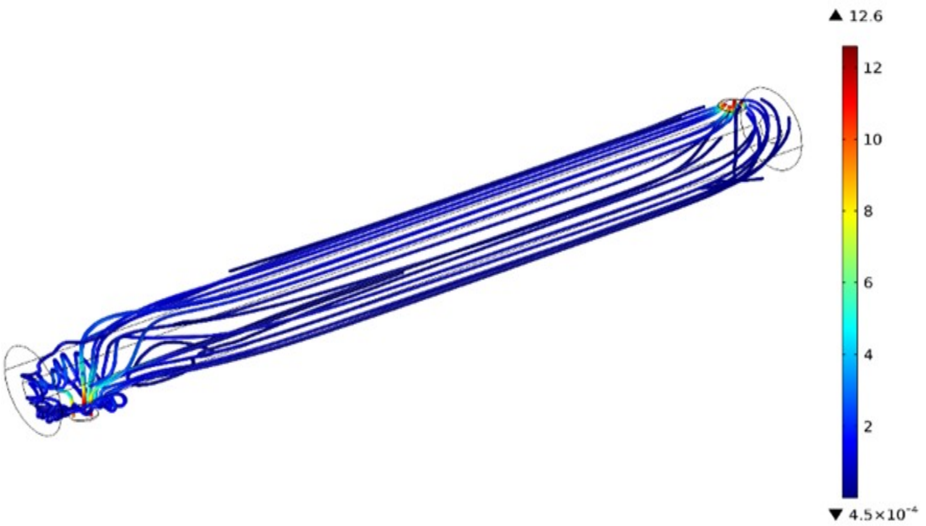

Figure 16. Velocity contours of air side for smooth tube heat exchanger

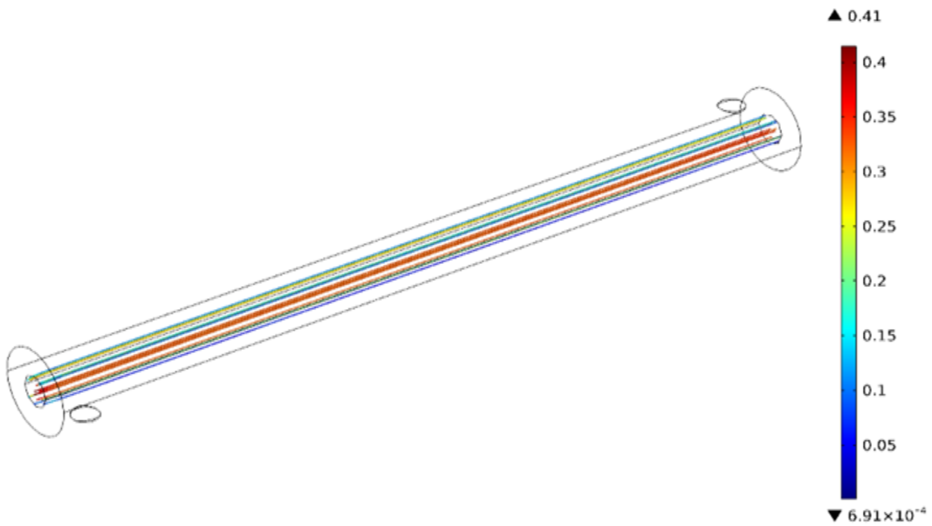

Figure 17. Velocity contours of water side for smooth tube heat exchanger

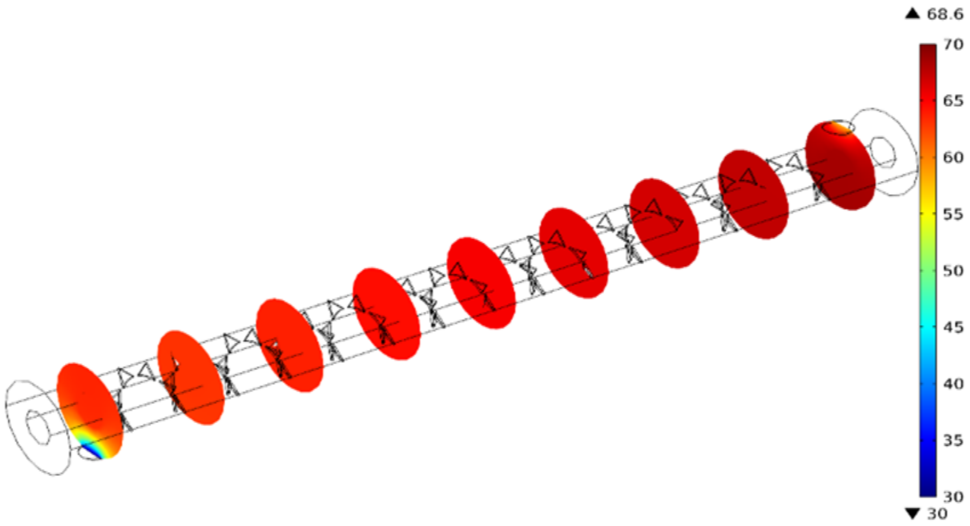

Figure 18. Temperature contours at various axial distances for triangular finned tube heat exchanger

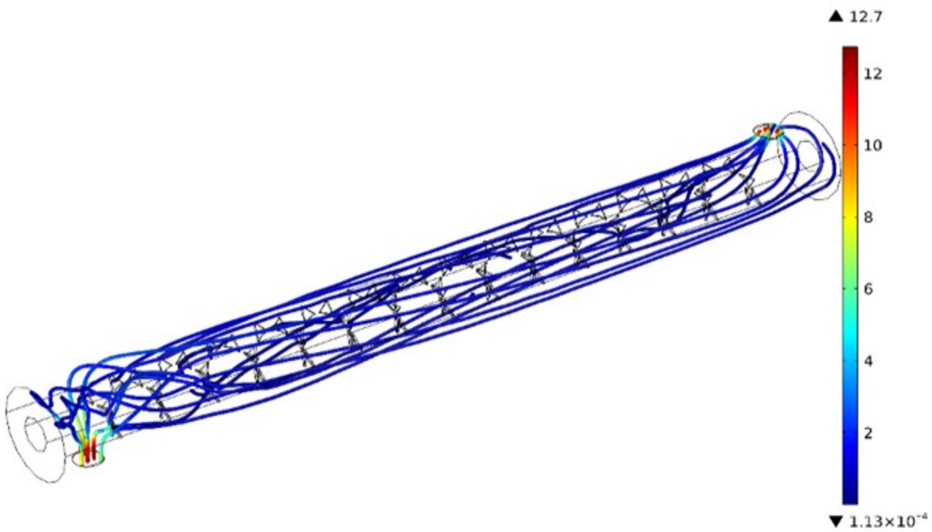

Figure 19. Velocity contours of air side for triangular finned tube heat exchanger

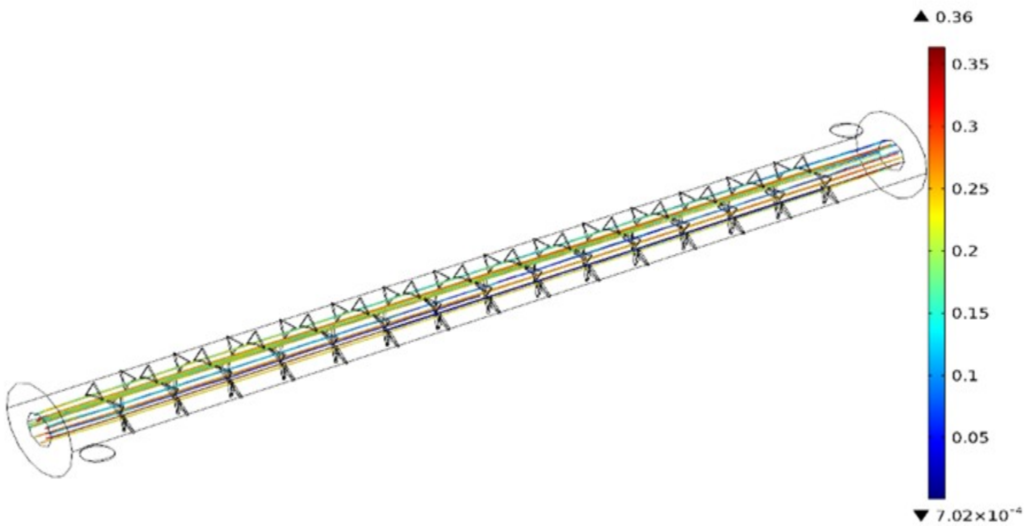

Figure 20. Velocity contours of water side for triangular finned tube heat exchanger

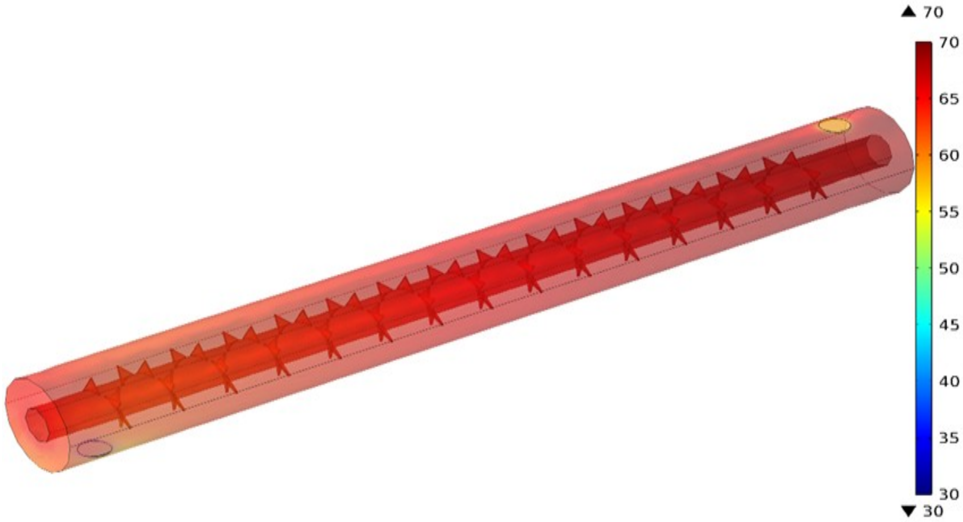

Figure 21. Temperature contours (distribution) at all zone for triangular finned tube heat exchanger

9. Conclusion

Air side temperature difference is directly proportional with the water Reynold's number and decreased by (5%) with increasing the air mass flow rate. Water side temperature difference is directly proportional to the air mass flow rate and decreased by (20%) with increasing the water Reynold's number. Results of numerical simulation showed that adding fins would enhance heat dissipation through heat exchanger. It is noted that heat transfer behavior increases within present model as air mass flow rates and water Reynold's numbers increases. Heat transfer enhancement by adding triangular fins to outer surface tube has been found during numerical simulation. Numerical simulation by using COMSOL (5.0) computational fluid dynamic (CFD) package model. Is successful for predicting both heat transfer and fluid flow in the present heat exchanger. 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 (3.252 to4.502) times than that of smooth tube. Significant enhancement has been found in heat exchanger effectiveness by adopting fins on outer surface of inner tube.

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