University of Anbar

Anbar Journal Of Engineering Science©

journal homepage: http:// http://www.uoanbar.edu.iq/Evaluate/

Numerical Investigation on the Thermal Performance of Double Pipe Heat Exchanger Using Different Shapes of Fins

Asaad K. Ali , Wissam H. Khalil

Department of Mechanical Engineering, College of Engineering, University of Anbar Ramadi, Iraq

PAPER INFO

Paper history:
Received … … …
Received in revised form … … …
Accepted … … …

Keywords:
Double pipe heat exchanger, Extended surface, Convection heat transfer, Thermal hydraulic performance.

ABSTRACT

In this study, a numerical investigation on the thermo-hydraulic performance of the double pipe heat exchanger into heat transfer by different shapes of fins on the outer surface for the inner tube as extended surfaces. The inner and outer diameters of the inner pipe were (16.05 mm), (19.05 mm) respectively, and (34.1 mm), (38.1 mm) for the outer tube. The length of the heat exchanger was (1000 mm). Hot and cold water were used as the working fluid, where the hot water flows inside of the inner one in counter flow with the cold water which flows in the annulus. The inlet temperature for the hot water is (75 ^{O}C) while it is (30 ^{O}C) for the cold. The hot fluid flows at constant rate which is (0.1kg/s) while the cold is varied from (0.1 kg/s to 0.2 kg/s).The study was perform using the known commercial CFD package (ANSYS – FLUNET 15) .The results shows that both (rectangular and triangular) fins enhances the heat transfer coefficient compare with the conventional plain tube .The rectangular fins presents an heat transfer enhancement ratio of (61% to 74%). Using of extended surfaces present a good result in saving energy by enhancing the performance of the double pipe heat exchangers used in petroleum industry.

1. INTRODUCTION

A large portion of energy being consumed in industry processes and the energy resources are to be consumed at a large and alarming rate. Energy conservation is therefore, become an important issue. In many areas of the industries, using of high-performance heat exchanger is one of the promising energy-saving manners.

The high efficient heat exchangers may be obtained by utilization of heat transfer enhancement techniques. In general, heat transfer enhancement creates one or more combinations of the following conditions that are favorable for the increase in heat transfer rate with an undesirable increase in friction: (1) interruption of boundary layer development and rising degree of turbulence, (2) increase in heat transfer area, and (3) generating of swirling and/or secondary flows [1]. Several enhancement techniques have been introduced, such as, treated surfaces, rough surfaces, swirling flow devices, coiled tubes, and surface tension devices [2].

Heat exchangers are an efficient devices constructed for the powerful heat transfer between two fluids of different temperatures. The media may be separated through a solid wall, to prevent mixing, or they may be in direct contact [3]. Heat exchangers are extensively used in food processing industry, dairy industry, biochemical processing, pharmaceuticals, chemical plants and petroleum plants. The use of heat exchangers in bioprocess industry is seem to be everywhere; from high temperature pasteurization to low temperature freezing applications.

Annular or double pipe heat exchanger is one of the simplest type of heat exchangers with a concentric tube like construction with one fluid flowing inside the pipe and another fluid flowing outside the first pipe enclosed within a second pipe circumventing it. The flow of a fluid in a double pipe heat exchanger can either be co-current or concurrent.

The energy from each fluid is exchanged and no extra heat is added or removed [4]. Since the heat in the process is not constant and the heat amount of the fluids is also not constant thus the Heat exchanger must be designed in a way that it is suited for all the cases of heat exchange and the performance is best suited for all conditions. Also the design should be such that the heat exchange is at a particular rate required by the process. Heat exchangers are originally designed to be over sized so that in cases of fouling, the surface of heat exchanger is still large enough to carry out operations. Once cleaned the heat exchanger would be again oversized. For the use of any heat exchanger the proper study of various technical and economical parameters is required such as life of heat exchanger, cost per unit area , Overall Heat Transfer Coefficient , Low heating value of fuel, Effectiveness, Efficiency , Heat Capacities, Annual variation of temperatures of fluid under observation. Various types of heat exchanger follow this general principle.

In the petrochemical industry the installation of different types of heat exchangers has been extensively used for heat recovery tasks or the condensation of solvents in petrochemical industry. The use of double pipe heat exchanger primary surfaces permit to make the heat exchanger design particularly compact and allows to minimize the installation footprint and the supporting civil engineering structure, and therefore to obtain the best return on investment.

In the last decades, many different techniques were used to enhance the heat transfer coefficient [5]. Extended surfaces is one kind of these techniques. This technique is used in many applications and it has two effects the first is increasing the heat transfer area and the second is inducing a swirling flow that making a better mixing which led to an effective method to increase the heat transfer rate through pipes with no need to add any external power (passive techniques).

Numerous works have revealed that tubes with treated surfaces or supplied with extended surfaces are effective devices for enhancing heat transfer rate [6-8]. Rabas et al. [9] reported that the influence of roughness shape and spacing on the performance of three-dimensional helically dimpled tubes. Chen et al. [10] performed an experimental work to study the heat transfer coefficient, friction factor and enhancement index characteristics in treated surfaces (dimpled tube) at different depth/pitch of dimples. Their results showed that the heat transfer rates were enhanced from 25% to 137% at constant Reynolds number, and 15% to 84% at constant pumping power. Vicente et al. [11] reported that the effects of the three dimensional helically dimpled tubes (dimpled height, h/d=0.08 to 0.12 and helical pitch, p/d=0.65 to 1.1). Helical twisted tapes studied by the researchers [12-15], they found that the existence of such techniques present good enhanced in heat transfer due to increase in surface area. The results gave good considerations of using heat transfer enhancement techniques.

The advantages of the extended surfaces as passive enhancement techniques, shown in literature review mentioned previously, motivate us to investigate heat transfer enhancement by using the triangular and rectangular fins as extended surfaces and enhancing device. In the present work, the effects of shape of the fins and mass flow-rate of cold water on the heat transfer coefficient and pressure loss characteristics in the fully developed turbulent flow are examined. The Reynolds numbers for the cold water are ranged from (2500-5000) while the Reynolds numbers for the hot water is kept constant at (8000). The Numerical results of the heat transfer enhancement and pressure loss as well as the empirical correlation for Nusselt number and friction factor are presented.

2. Model Geometry

The dimensions of double pipe heat exchanger used in this numerical investigation is schematically described in (**Figure 1**) and (**Figure 2**). The model has (1000 mm) as the length , the inner and outer diameter for the inner tube is (16.05 mm) and (19.05 mm) respectively, the inner and outer diameter for the outer tube is (34.1 mm) and (38.1 mm) respectively **as shown in figure (1)**. The double pipe heat exchanger is made of copper. The heat exchanger was modified by adding the extended surface (fins) on the outer surface for the inner tube , the configuration of fins are rectangular and triangular shapes **as shown in figure (2)**. The dimensions of fins are (6 mm) as the height and (2 mm) as the base width. In this study the hot and cold water used as working fluid, the hot water flow as counter flow in the inner tube while the cold water flow in the outer tube. Table (1), shows the number of elements that used in each case (plain tube, with triangular fins, and with rectangular fins).

3. Mathematical Formulation and Numerical Methods

In order to solve the Navier–Stokes and energy equations to investigate the effect of double pipe heat exchanger performance both plain conventional one and with extended surfaces, the following assumptions were made:

- ✓ Both fluid flow and heat transfer are in steady state and three dimensional;

- ✓ Fluid is in single phase, incompressible and the flow is Turbulent;
- ✓ Properties of both fluid and pipe material are temperature-independent.

Table (2) and table (3) show the properties of both hot and cold water and the temperature of both them. The governing equations and its boundary conditions in polar coordinates for 3D turbulent incompressible flow for the current problem are **[16]**:

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \frac{1}{r}\frac{\partial}{\partial r}(\rho\, r\, v_r) + \frac{1}{r}\frac{\partial}{\partial \theta}(\rho\, v_\theta) + \frac{\partial}{\partial z}(\rho\, v_z) = 0 \qquad (1)$$

Momentum Equation

The momentum equations can be written as the following:

x-Momentum equation:

$$\rho\left(vr\frac{\partial v_r}{\partial r} + \frac{v_\theta}{r}\frac{\partial v_r}{\partial \theta} - \frac{v_\theta^2}{r} + vz\frac{\partial v_r}{\partial z} + \frac{\partial v_r}{\partial t}\right) = \rho\, gr - \frac{\partial p}{\partial r} + \mu\left[\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(r\, vr)\right) + \frac{1}{r^2}\frac{\partial^2 v_r}{\partial \theta^2} + \frac{\partial^2 v_r}{\partial z^2}\right] \qquad (2)$$

y – Momentum equation:

$$\rho\left(v_r\frac{\partial v_\theta}{\partial r} + \frac{v_\theta}{r}\frac{\partial v_\theta}{\partial \theta} + \frac{v_r v_\theta}{r} + v_z\frac{\partial v_\theta}{\partial z} + \frac{\partial v_\theta}{\partial t}\right) = \rho\, g_\theta - \frac{1}{r}\frac{\partial p}{\partial \theta} + \mu\left[\frac{\partial}{\partial r}\left(\frac{1}{r}\frac{\partial}{\partial r}(r\, v_\theta)\right) + \frac{1}{r^2}\frac{\partial^2 v_\theta}{\partial \theta^2} + \frac{2}{r^2}\frac{\partial v_r}{\partial_\theta} + \frac{\partial^2 v_\theta}{\partial z^2}\right] \qquad (3)$$

z – Momentum equation:

$$\rho\left(v_r\frac{\partial v_z}{\partial r} + \frac{v_\theta}{r}\frac{\partial v_z}{\partial \theta} + v_z\frac{\partial v_z}{\partial z} + \frac{\partial v_z}{\partial t}\right) = \rho\, g_z - \frac{\partial p}{\partial z} + \mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial v_z}{\partial r}\right) + \frac{1}{r^2}\frac{\partial^2 v_z}{\partial \theta^2} + \frac{\partial^2 v_z}{\partial z^2}\right] \qquad (4)$$

Energy Equation

The energy equation can be written in cylindrical coordinates as following:

$$\rho\, Cp\left(\frac{\partial T}{\partial t} + v_r\frac{\partial T}{\partial r} + \frac{v_\theta}{r}\frac{\partial T}{\partial \theta} + v_z\frac{\partial T}{\partial z}\right) = k\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial 2T}{\partial \theta^2} + \frac{\partial 2T}{\partial z^2}\right] + \mu \O \qquad (5)$$

The dissipation function in cylindrical coordinates for incompressible fluid is given by

$$\O = 2\left(\frac{\partial v_r}{\partial r}\right)^2 + 2\left(\frac{1}{r}\frac{\partial v_\theta}{\partial \theta} + \frac{v_r}{r}\right)^2 + 2\left(\frac{\partial v_z}{\partial z}\right)^2 + \left(\frac{\partial v_\theta}{\partial r} - \frac{v_\theta}{r} + \frac{1}{r}\frac{\partial v_r}{\partial \theta}\right)^2 + \left(\frac{1}{r}\frac{\partial v_z}{\partial \theta} + \frac{\partial v_\theta}{\partial z}\right)^2 + \left(\frac{\partial v_r}{\partial z} + \frac{\partial v_z}{\partial r}\right)^2 \qquad (6)$$

A computational fluid dynamic code (FLUENT) is used to calculate flow velocity, pressure, and temperature in the domain of computation. Finite volume method (FVM, was used to convert the governing equations to algebraic equations accomplished using hybrid differencing scheme [**17**]. The SIMPLE algorithm was used to enforce mass conservation and to obtain pressure field **[18]**. This is an iterative solution procedure where the computation is initialized by guessing the pressure field. Then, the momentum equation is solved to determine the velocity components. The pressure is updated using the continuity equation. Even though the continuity equation does not contain any pressure, it can be transformed easily into a pressure correction equation **[17]**. The segregated solver was used to solve the governing integral equations for the conservation of mass, momentum, and energy. Because of the assumption of constant fluid thermo-physical properties and negligible buoyancy, the mass and momentum equations are not coupled to the energy equation. Therefore, the temperature field is calculated by solving the energy equation after a converged solution for the flow field is obtained by solving the momentum and continuity equations. The iterations were continued until the sum of residuals for all computational cells became negligible (less than 10^{-7}) and velocity components did not change from iteration to iteration .

4. Results and discussion

Numerical results obtained from the numerical investigation of the plain tube heat exchanger of heat transfer and fluid flow are compared to other researchers for the conventional plain tube that previously published for the sake of validation of the code used in numerical investigations. Nusselt number is compared with the famous **Dittus-Boelter** correlation ($Nu = 0.023\ Re^{0.8}\ Pr^{0.4}$) **[16]** and the Chinaruk Thianpong et al. [19]. **Figure (4)** show the comparison between the present results of Nusselt number and the correlation , the results shows a good agreement between the two curves. Friction factor results of the program is compared with the results of reference **[19]** as shown in **Figure (5)** ,also the results give a very good agreement .

Numerical results of the pressure drop and friction factor (f) characteristics in finned tubes are presented in Fig. (6) and Fig. (7), respectively. The pressure drop of the plain tube heat exchanger acting alone are also plotted for comparison. Figure (6) shows that the pressure drop increases with increasing Reynolds number for the conventional turbulent tube flow. The increase in pressure drop is due to the blockage occur due the existence of the fins this will led to forming an eddies which create this increase in pressure drop. The increase in pressure drop is greater in the case of the rectangular fins than in triangular fins. Figure (7) shows the numerical results of friction factors of the finned tube in comparison with plain tube heat exchanger. From the figure obvious that the friction factor is decreased as the Reynolds number increases and the triangular fins present a friction factor greater than that of rectangular fins .This results is logical because of the greater velocity of fluid in the rectangular finned tube which led to a smaller value of friction factor ($f = \Delta P/0.5\ \rho\ v^2$). Figure (8) shows the effect of using fins on the pumping power of fluid through the double pipe heat exchanger. As Reynolds number increases the pumping power increases and the increase in treated surfaces in greater than that of plain tube conventional heat exchanger (note that pumping power = Q. ΔP).

The heat transfer coefficient against Reynolds number is presented in Figure (9).It can be shown that the heat transfer coefficient in increase as Reynolds number increases and the increase in rectangular fins is greater than that of triangular fins this is due to the increased in heat transfer area and the increase in secondary flow occurring due to the existence of fins which led to breaking the boundary layer and reduces the thermal resistance to heat transfer. Figure (10) represent the overall enhancement by taken in the same time the effect of heat transfer coefficient and the friction factor effect which is calculated from (O.E.R. = $\frac{\frac{h\ with}{h\ without}}{(\frac{f\ with}{f\ without})^{\frac{1}{3}}}$) **[20]**. The rectangular fins presents an enhancement ratio from (61% to 74%) ,while the heat transfer enhancement ratio in triangular fins are (46% to 56%) .

5. Conclusions

A numerical study of fully developed turbulent flow in a double pipe heat exchanger provided with extended surfaces (triangular and rectangular) fins has been performed. The influences of the shape of fins and varying of cold mass flow-rate (Reynolds number) on the heat transfer rate and friction factor characteristics have also been investigated. The results show that the rectangular extended surfaces present a good heat transfer enhancement ratio than that of triangular fins. The pumping power of the fluid with using extended surface is greater than that of conventional heat exchanger. The empirical correlations for the Nusselt number and the friction factor based on the present numerical data are also presented. The results both for heat transfer and friction factor were compared with the previous published data in the literature and the agreement is very good.

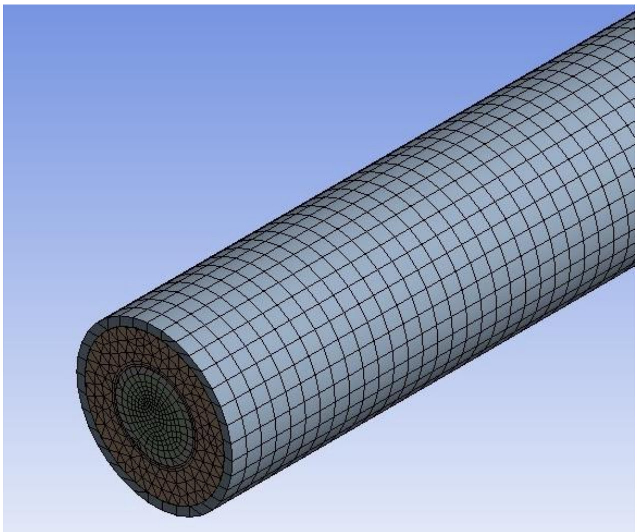


Fig. (1): Double pipe heat exchanger without fins

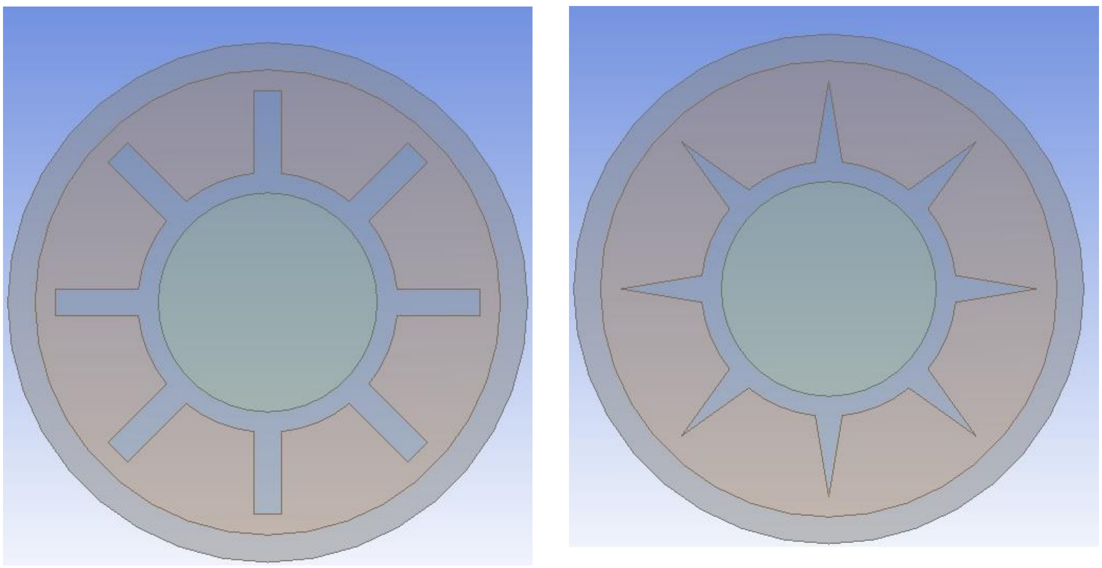


Fig. (2): Double pipe heat exchanger with fins

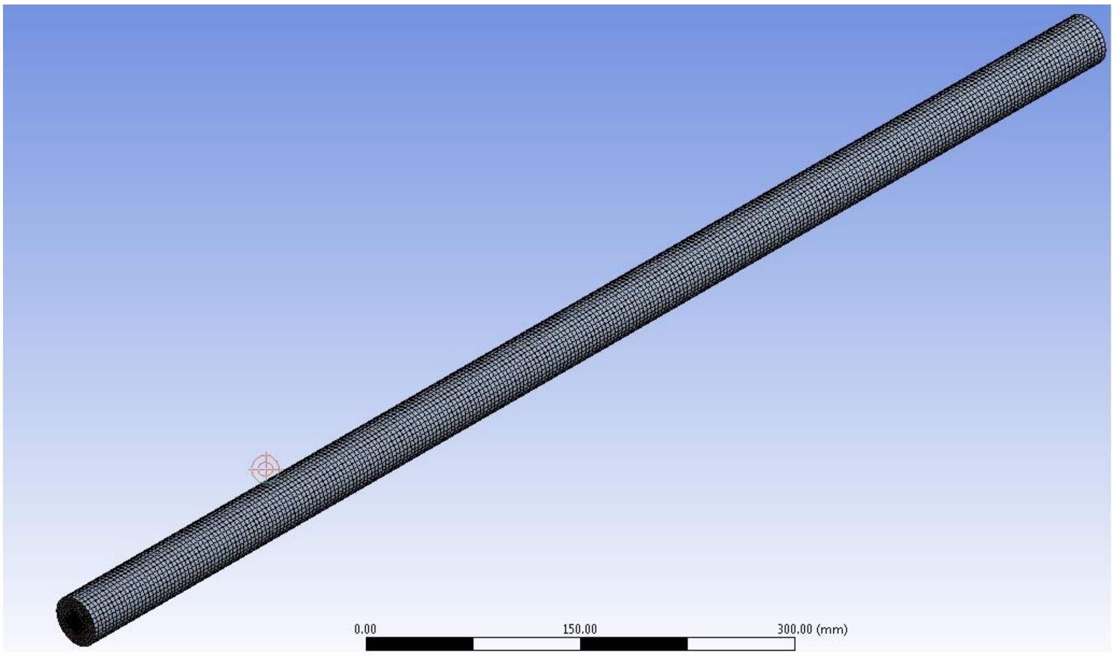


Fig. (3): Mesh for the double pipe heat exchanger

Table (1): Number of element for heat exchanger with and without fins

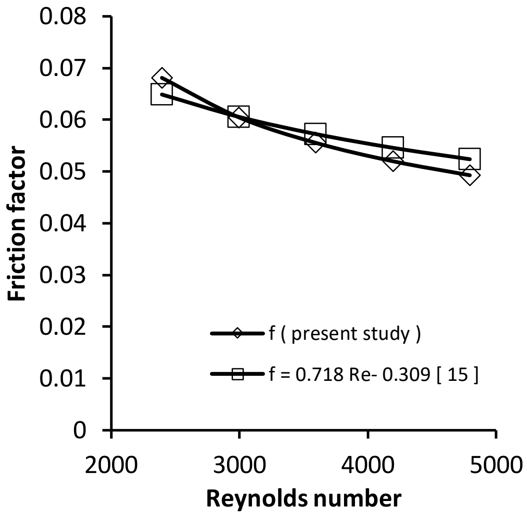
State	Number of elements
Heat exchanger without fin	205840
Heat exchanger with rectangular fins	216874
Heat exchanger with triangular fins	218610

Table (2):The physical properties of pure water

Density ρ (kg/m^3)	998.2
Specific Heat C_p (J/kg.k)	4182
Thermal Conductivity K (W/m.k)	0.6
Viscosity (kg/m.s)	0.001003

Table (3): Boundary condition for inner and outer fluid

Boundary types	Inner tube	Outer tube
Mass flow inlet (kg/s)	0.1	Various from (0.1 – 0.2)
Temperature (^{o}C)	75	30



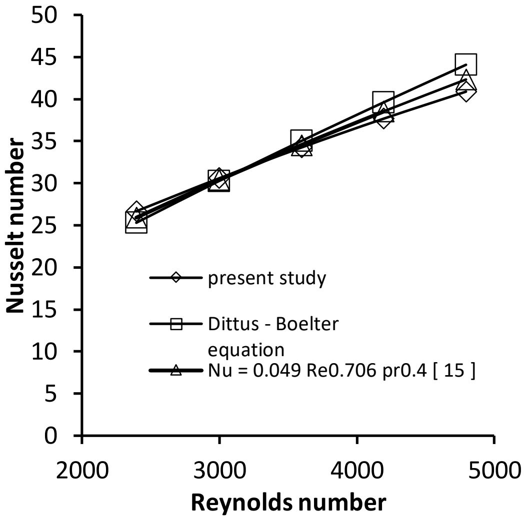


Fig. (4): Reynolds number with Nusselt number for plain heat exchanger

Fig. (5): Reynolds number with Friction friction factor

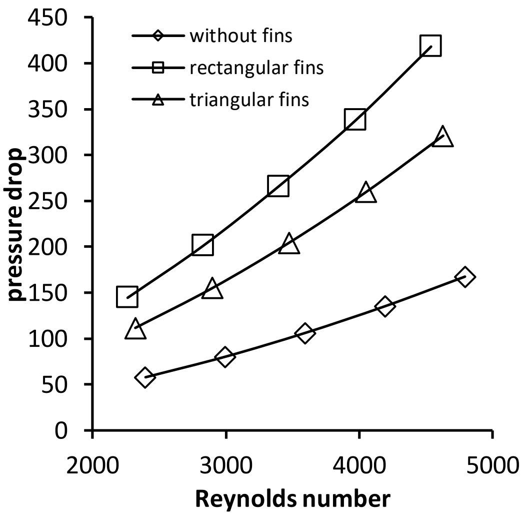


Fig. (6) Reynolds number vs pressure drop

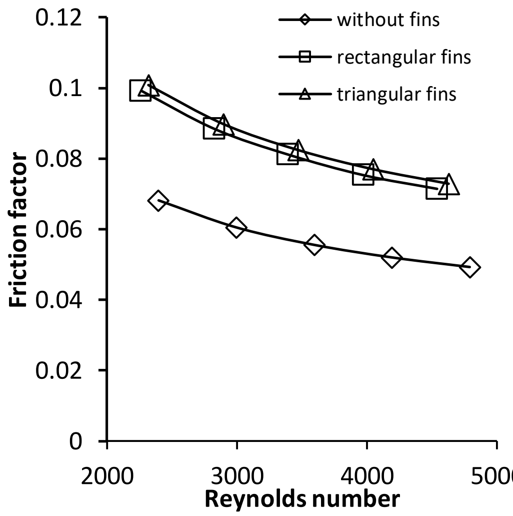


Fig. (7) Reynolds number vs Friction factor

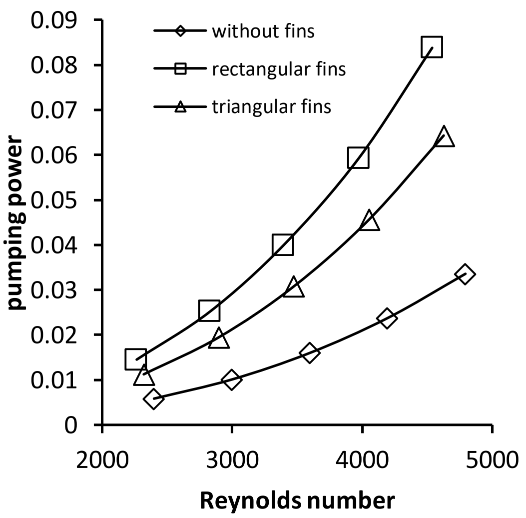


Fig.(8) Reynolds number vs pumping power

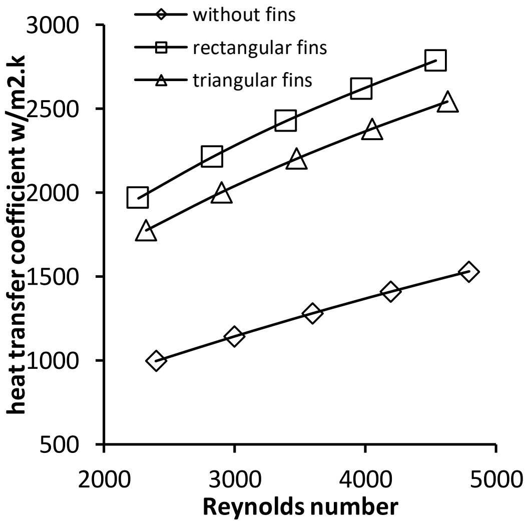


Fig.(9) Reynolds number vs heat transfer coefficient

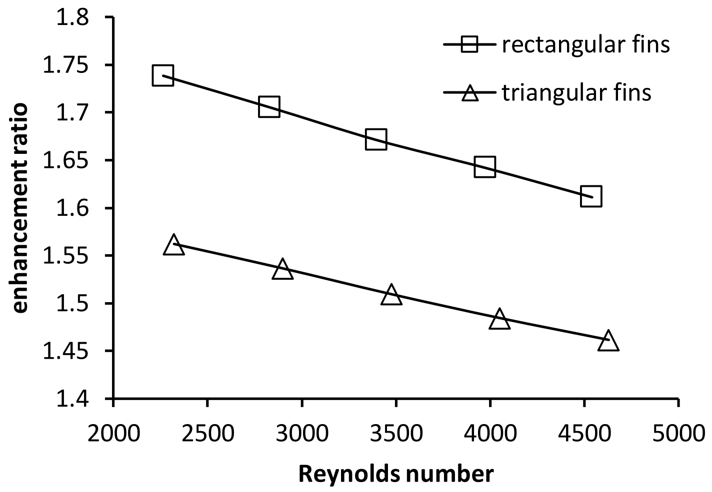


Fig. (10) Reynolds number vs enhancement ratio

References:

1- P. Promvonge, Thermal augmentation in circular tube with twisted tape and wire coil turbulators, Energy Conversion and Management 49 (2008) 2949–2955.

2-A.E. Bergles, ExHFT for fourth generation heat transfer technology, Experimental Thermal and Fluid Science 26 (2002) 335–344.

3-Sadik Kakaç and Hongtan Liu (2002). Heat Exchangers: Selection, Rating and Thermal Design (2nd ed.). CRC Press. ISBN 0-8493-0902-6.

4- P. Raskovic , A. Anastasovski, L. Markosvska,V. Mesko, Process integration in bioprocess industry waste heat recovery in yeast and ethyl alcohol plant, Journal Energy35,2010,704-717.

5- Zimparov, V., "Energy conservation through heat transfer enhancement techniques", Int. J. Energy Res. 26 (2002) 675-696.

6- Heat transfer and pressure drop in a spirally grooved tube with twisted tape insert P. Bharadwaj, A.D. Khondge, A.W. Date International Journal of Heat and Mass Transfer 52 (2009) 1938–1944.

7- V. Ozceyhan, Experimental Investigation of Flow and Heat Transfer in a Rib Inserted Pipe, Erciyes University, Graduate School of Natural and Applied Sciences, Ph.D. Thesis, Turkey, 2002.

8- A. Dewan, P. Mahanta, S.K. Raju, S.K. Kumar, Review of passive heat transfer augmentation techniques, Journal Power Energy 218 (2004) 509–525.

9- T.J. Rabas, R.L. Webb, P. Thors, N.K. Kim, Influence of roughness shape and spacing on the performance of three dimensional helically dimpled tubes, Journal of Enhanced Heat Transfer 1 (1993) 53–64.

10- J. Chen, H. Müller-Steinhagen, G.G. Duffy, Heat transfer enhancement in dimpled tubes, Applied Thermal Engineering 21 (2001) 535–547.

11- P.G. Vicente, A. Garcıa, A. Viedma, Heat transfer and pressure drop for low Reynolds turbulent flow in helically dimpled tubes, International Journal of Heat and Mass Transfer 45 (2002) 543–553.

12- Snehal S. Pachegaonkar, Santosh G. Taji, Narayan Sane "Performance analysis of double pipe heat exchanger with annular twisted tape insert" International Journal of Engineering and Advanced Technology (IJEAT), Vol. 3, February 2014, ISSN: 2249 – 8958.

13- Ramesh Vathare and Omprakash Hebbal "CFD Analysis of Enhancement of Turbulent Flow Heat Transfer in a Horizontal Tube with Different Inserts" International Journal of Engineering Research & Technology, Vol. 2 Issue 8, (2013).

14- Shiva Kumar, K. Vasudev Karanth and Krishna Murthy " Numerical study of heat transfer in a finned double pipe heat exchanger " World Journal of Modelling and Simulation , Vol. 11 , No. 1, pp. 43-54 (2015).

15- C Rajesh Babu and Santhosh Kumar Gugulothu *" CFD Analysis of Heat Transfer Enhancement by Using Passive Technique in Heat Exchanger "* International Journal of Recent advances in Mechanical Engineering , Vol.4, No.3 , (2015).

16- F.P. Incropera, P.D. Witt, T.L. Bergman, A.S. Lavine, Fundamentals of Heat and Mass Transfer, John-Wiley & Sons, 2006 .

17- S.V. Patankar, Numerical Heat Transfer and Fluid Flow, Hemisphere, New York,1980.

18- W.Q. Tao, Numerical Heat Transfer, second ed., Xi’an Jiaotong University Press, Xi’an, China, 2001.
19- Chinaruk Thianpong ,Petpices Eiamsa-ard, Khwanchit Wongcharee, Smith Eiamsa-ard “Compound heat transfer enhancement of a dimpled tube with a twisted tape swirl generator” International Communications in Heat and Mass Transfer 36 (2009) 698–704.
20- Webb R. L. (1981) "Performance evaluation criteria for use of enhanced heat transfer surfaces in heat exchanger design" International Journal of Heat and Mass Transfer 24 , 715 – 726.