The effect of geometrical characteristics of wavy strip turbulator and thermodynamic properties of fluid on exergy loss and heat transfer in a tube in tube heat exchanger

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Abstract

The present work conducts an experimental investigation into the influence of flow, thermodynamic and geometrical characteristics of the wavy strip on exergy loss and dimensionless exergy loss in a double tube heat exchanger. Water was chosen as working fluid with hot water passing the inner tube and cold water passing annulus. Wavy strips with four different angles ranging from 60 to 150 degrees and three widths of 8, 13 and 18 millimeters have been used to conduct the experiments. The result of exergy loss and dimensionless exergy loss for various conditions is presented and on the basis of curve fitting, three empirical correlations are suggested to predict the dimensionless exergy loss in double tube heat exchanger.

Nomenclature

$c_{p,w}$	specific heat at constant pressure J/kg.K
C_{min}	minimum heat capacity W/K
C_{max}	maximum heat capacity W/K
C_r	heat capacity ratio (dimensionless)
E	exergy loss, W
e	dimensionless exergy loss
H	specific enthalpy j/kg
L	length of one wave, m
$\dot{m}$	mass flow rate, kg/s
Nu	Nusselt number
Pr	Prandtl number
Q	heat transfer rate, W
Q_{max}	maximum heat transfer, W
Re	Reynolds number
s	specific entropy, j/Kg.K
T	temperature, K
w	wavy strip width, m
X_p	predicted value
X_a	actual (experimental) data

Greek letters

ε	effectiveness (dimensionless)
θ	wavy strip angle, degree

Subscripts

c	cold
h	hot
i	inner
o	outer
in	inlet
out	outlet
e	environment condition
w	water

Keywords: Double tube heat exchanger, Wavy strip, exergy analyses, dimensionless exergy loss.

1. Introduction

Heat exchangers have extensive applications in cooling systems, power plants, chemical industries, ventilating systems and many other industries. Heat transfer enhancement in the heat exchangers increases the overall system performance and reduces the size of the system. That is the reason why many researches have been done to improve heat transfer in heat exchangers using a variety of active, passive and compound methods during last couple of decades.

Hsieh and Wen [1] presented an experimental investigation into the heat transfer enhancement and pressure drop of a vertical 2-pass tube with three different inserts. Longitudinal, crossed and regularly interrupted strips were inserted in a crossflow heat exchanger. By using the experimental data, four sets of correlations were developed to predict the average Nusselt number and pressure drop factors in the tube with or without the inserted strips.

Yaningsih et al. [2] experimentally investigated the effect of using axial perforated twisted tape in the inner tube of a double pipe heat exchanger on the Nusselt number, friction factor and thermal performance factor. Three different axial pitch ratios were studied. Results indicated that perforating the twisted tape turbulators increases both heat transfer and pressure drop. Also, results show that for a given hot water flow rate, heat transfer and friction factor increase with the decrease of axial pitch ratio.

Bharadwaj et al. [3] studied the heat transfer and pressure drop characteristics of water in a spirally grooved tube equipped with twisted tape turbulators. Experiments were carried out at laminar, transient and fully turbulent flow regimes. The results indicated a non-linear behaviour of friction factor and Nusselt number with Reynolds number and twist ration. It is showed that the direction of the twists influences the thermo-hydraulic characteristics. It is also found that the heat transfer and fluid flow characteristics of a spirally grooved tube were influenced by interactions between momentum and heat transfer in the vicinity of the grooved wall.

Bas and Ozceyhan [4] experimentally investigated the effect of twist and clearance rations on the heat transfer and pressure drop of a twisted tape inserted tube. To obtain the increased heat transfer due to laminar sublayer destruction only, the twisted tapes were placed separately from the tube’s wall. Two clearance ratios and five twist ratios were studied in Reynolds numbers ranging from 5,132 to 24,989. The result showed that the twist ratio has more significant role on enhancement of heat transfer compared to the clearance ratio. It is also shown that the heat transfer and friction factor in the twisted tape inserted tube were higher than the plain tube. Also, it has been showed that the heat transfer and friction factor increased when the clearance and twist ratios decreased.

Hejazi et al. [5] conducted a comprehensive experimental study on the heat transfer enhancement and pressure drop of horizontal twisted tape inserted tubes. Heat transfer coefficients and pressure drop of tubes with four different twisted tape inserts with twist ratios of 6, 9, 12 and 15 were studied during the condensation of R-134a as working fluid. Results showed that the highest heat transfer coefficient achieved in a tube with twist ratio of 6. Also, it has been shown that by considering the effect of pressure drop alongside the heat transfer enhancement, the best performance occurred when the twisted tape with the twist ratio of 9 was employed.

Garcia et al. [6] inserted three wires of different pitch in a smooth tube and studied heat transfer and pressure drop characteristics in laminar and transition flow regimes under uniform heat flux conditions. By using the experimental results two correlations have been suggested to predict the isothermal friction factor and the Nusselt number of each wire. Their results showed that in Reynolds numbers ranging from 700 to 2,500, wire coils inserts have better performance compared to the twisted tape inserts.

Beside the researches on enhancement of the heat transfer in heat exchangers, there is a significant attention in literature on the analysis of exergy in heat exchangers during the last couple of years. Analysis of exergy uses conservation of mass and conservation of energy principles together with the second law of thermodynamics and is a very powerful tool for analyzing, designing and optimizing of the complex thermodynamic systems [7].

Prasad and Shen [8] used exergy analysis method to evaluate the effectiveness of a tubular heat exchanger with coiled-wire inserts. Dimensionless exergy destruction in the heat exchanger was calculated and optimization was performed by minimizing the exergy destruction. It has been mentioned that in an energy system, the optimization of a heat exchanger will not necessarily lead to an optimum result since heat exchanger is one of several subunits. It has also been found that the exergy analysis method is a very useful technique to evaluate the performance of heat exchangers with various enhancement inserts.

Cornelissen and Hirs [9] used a combination of exergy and life cycle analysis to optimal design a heat exchanger. They believe that their method should be implemented by all energy systems that have any kind of trade-off between exergy saving during the operation and exergy use during the construction. They have also mentioned that their method can give the real optimum design point from the conservation of exergy of natural resources viewpoint.

Doohan et al. [10] performed a comprehensive study on the thermal performance of a plate fine heat exchanger with different geometrical parameters. Exergy analysis method was used to achieve high thermal performance with minimum pressure lost in the heat exchanger. The results showed exergy analysis as a powerful technique to optimize the geometrical design parameters. This study also provided useful data for optimized selection of geometrical parameters in cryogenic heat exchanger.

Ipek et al. [11] experimentally analyzed the exergy loss of two newly designed heat exchangers. After performing the experiments for compact heat exchanger and brazed plate heat exchanger, the exergy losses were calculated for each of the heat exchangers and the results showed similar exergy loss for both of the heat exchangers. It is also anticipated that the exergy loss in compact heat exchanger decreases by isolation the heat exchanger.

Sahoo et al. [12] conducted an investigation into the effect of operational parameters on the energetic and exergetic performances of a rectangular fin-tube automotive radiator. Water, ethylene glycol and propylene glycol and their brines were studied and the results showed that water and 25% propylene glycol brine solutions have similar energetic and exergetic performances and 50% higher effectiveness and 7% higher second law efficiency compared to that of 25% ethylene glycol brine. It is also showed that the optimum propylene glycol brine solution has the best performance index, highest heat transfer rate and lower pumping power which lead to a reduction in radiator size, weight and cost, and engine fuel consumption compared to conventional ethylene glycol brine coolant.

Summarizing all mentioned above, it can be concluded that lots of research works have been done on the effects of different kinds of turbulators on heat transfer enhancement and exergy loss in double tube heat exchangers. But, no study has been investigated into the effect of wavy strip width on the performance of a double tube heat exchanger. Also, there has been no research on the exergy loss analysis of a double tube heat exchanger equipped with wavy strip turbulator. Thus, the main scope of this paper is to study of the effect of wavy strip on exergy loss and dimensionless exergy loss by using the result of our experimental work. The effect of flow and thermodynamic properties (cold water flow rate, heat capacity ratio and inlet hot water temperature) on exergy loss and dimensionless exergy loss are also presented.

2. Experimental set-up and procedure

Schematic view of the experimental setup which was used for investigation of the exergy loss and dimensionless exergy loss in a double tube heat exchanger is presented in Fig.1. The experimental set up consists of a test section, cold and hot water tanks, two centrifugal water pumps, data logger, a set of thermocouples, and two rotameters. A horizontal double tube heat exchanger made of two concentric tubes was used for the test section. The inner tube is made of copper with inner diameter of 27.3 mm, while, the outer tube is a galvanized pipe with inner diameter of 29 mm. To minimize the heat losses, the test section was well insulated with glass wool and foam.

Schematic view of the wavy strips is shown in Fig. 2. A bending machine was first used to bend galvanized sheets with width of 8 mm to four angles of 60, 90, 120 and 150 degrees. Then the angle was kept constant at 60 degrees and wavy strips with three different widths of 8, 13 and 18 mm were generated. After bending process the wavy strips were placed along the inner tube of the heat exchanger.

To adjust the inlet hot water temperature to the desired values, two heaters were placed inside the hot water tank and temperature was controlled carefully. After reaching the temperature to the desired value, the control valve was opened and the hot water was pumped out of the hot water tank to pass through the flow meter, test section, and inner tube of the heat exchanger. Meanwhile, an identical pump was used to counter flow the cold water (acting as a cooling medium) through the space between the pipes (annulus). Temperatures and flow rates of the hot and cold water were measured using k-type thermocouples and two calibrated rotameters, respectively. Data was recorded at steady state by a data acquisition system, and all measurement devices were calibrated prior to the tests.

Moffat method [13] utilized to obtain the uncertainties of the measurements. The uncertainties of measured data and the relevant parameters are presented in Table 1.

3. Data processing

Exergy is defined as the maximum theoretical work that can be obtained at the end of a reversible process in which equilibrium with the environment is achieved. In the present study, the amount of heat supplied by the hot water into the test section and the heat absorbed by the cold water assumed equal and therefore, the heat exchanger is assumed adiabatic. After gathering the data in the form of temperatures and flow rates, exergy loss in a double tube heat exchanger can be written as a sum of hot and cold water exergy losses as follow [14 and 15];

$$E = E_h + E_c \tag{1}$$

Exergy changes for hot and cold water can be written as below:

$$E_h = \mathrm{T}_e[\dot{m}_h(\mathrm{s}_{h,o} - \mathrm{s}_{h,i})] \tag{2}$$

$$E_c = \mathrm{T}_e[\dot{m}_c(\mathrm{s}_{c,o} - \mathrm{s}_{c,i})] \tag{3}$$

Substituting Eq. (2) and Eq. (3) into Eq. (1), results in following equation:

$$E = E_h + E_c = \mathrm{T}_e[\dot{m}_h(\mathrm{s}_{h,o} - \mathrm{s}_{h,i}) + \dot{m}_c(\mathrm{s}_{c,o} - \mathrm{s}_{c,i})] \tag{4}$$

For liquids, the entropy change can be expressed in terms of specific heats in constant pressure as below [15-17]:

$$\mathrm{s}_{c,o} - \mathrm{s}_{c,i} = c_{p,w} \ln\frac{\mathrm{T}_{c,o}}{\mathrm{T}_{c,i}} \tag{5}$$

$$\mathrm{s}_{h,o} - \mathrm{s}_{h,i} = c_{p,w} \ln\frac{\mathrm{T}_{h,o}}{\mathrm{T}_{h,i}} \tag{6}$$

By substituting Eq. (5) and Eq. (6) into Eq. (4), the following equation is obtained for the exergy loss:

$$E = \mathrm{T}_e[(\dot{m}_h c_{p,w} \ln\frac{\mathrm{T}_{h,o}}{\mathrm{T}_{h,i}}) + (\dot{m}_c c_{p,w} \ln\frac{\mathrm{T}_{c,o}}{\mathrm{T}_{c,i}})] \quad (7)$$

In the present study, the hot water flow rate is less than cold water flow rate. So, minimum heat capacity is calculated by using the hot water flow rate. Maximum and minimum heat capacities, and heat capacity ratio can be written as follow:

$$C_{min} = min[C_h, C_c], \qquad C_{min} = \dot{m}_h c_{p,w} \quad C_{max} = \dot{m}_c c_{p,w} \quad C_r = C_{min}/C_{max} \quad (8)$$

So, exergy loss can be written as follow:

$$E = \mathrm{T}_e[(C_{min} \ln\frac{\mathrm{T}_{h,o}}{\mathrm{T}_{h,i}}) + (C_{max} \ln\frac{\mathrm{T}_{c,o}}{\mathrm{T}_{c,i}})], \quad (9)$$

Dimensionless exergy loss can be written as follow [15-17]:

$$\mathrm{e} = \frac{E}{\mathrm{T}_e C_{min}} \quad (10)$$

Besides the exergy loss, the effectiveness of a double tube heat exchanger was calculated using the following equation [18]:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{Q}}{C_{min}(\mathrm{T}_{h,i} - \mathrm{T}_{c,i})} = \frac{\dot{Q}}{\dot{m}_h c_{p,w}(\mathrm{T}_{h,i} - \mathrm{T}_{c,i})} \quad (11)$$

4. Results and discussion

4.1 Validation of plain tube

To validate the experimental results, the Nusselt numbers obtained from the experiments of the smooth tube were compared with the Dittus-Boelter (see Eq. 12) [18]. Wilson plots method" as explained in Rose study [19] was used to evaluate the experimental Nusselt number. Fig.3 shows the comparison between the experimental results and the Dittus-Boelter correlation for the Nusselt number of the smooth tube with m_c = 14 L/min and $T_{h,i}$ = 63^o C. As shown in this figure, the experimental results are in a good agreement with those predicted by correlations.

$$Nu = 0.023Re^{0.8}\, Pr^{0.3} \tag{12}$$

4.2. The effect of geometrical characteristics

The experiments were conducted to study the effect of the flow and thermodynamic properties and the geometrical characteristics of the wavy strip turbulators on the exergy loss and dimensionless exergy loss in a double tube heat exchanger. To investigate into the effect of the wavy strip angle on exergy loss and dimensionless exergy loss, the inlet hot and cold water temperatures, the wavy strip width, and the cold water flow rate were kept constant and exergy loss and dimensionless exergy loss were calculated for four angels of 60, 90, 120 and 150 degrees. Then, the effects of the wavy strip width, cold water flow rate, and inlet hot water temperature on the exergy loss and dimensionless exergy loss in a heat exchanger containing a wavy strip with 60-degrees inserts were investigated.

Fig. 4 and Fig. 5 show the variations of the exergy loss and dimensionless exergy loss versus the hot water Reynolds number for four wavy strip angles of 60, 90, 120 and 150 degrees. It was observed from these two figures that both exergy loss and dimensionless exergy loss for the heat exchanger equipped with a wavy strip of 60 degrees are higher than those from the heat exchanger with wavy strips of 90, 120 and 150 degrees and those from the heat exchanger without wavy strips. This can be explained as follows: The heat transfer created by the temperature difference like what happens in a double tube heat exchanger is an irreversible process; more heat transfer means more irreversibility and, therefore, more exergy loss. It is also shown that our proposed correlation for dimensionless exergy loss is in a good agreement with experimental results.

An increase in the width or a decrease in the angle of the wavy strips enhances flow mixing and creates more disrupted boundary layer which improves the heat transfer in the heat exchanger. Therefore, wider wavy strip turbulators with low angles create more irreversibility and exergy loss. In this study, the largest exergy loss occurred in a heat exchanger equipped with a wavy strip having a width of 18mm and an angle of 60 degrees. The exergy loss of the mentioned turbulator is 1.38 to 1.72 times greater than that of a heat exchanger without wavy strips.

The exergy loss value was low at the small Reynolds number, but it increased with an increase in the Reynolds number. It is due to increase in the irreversibility and exergy loss of the heat exchanger which happens when the Reynolds number increases. Dimensionless exergy loss depends on both exergy loss (E) and minimum heat capacity (C_{min}). By increasing the hot water flow rate (Reynolds number), the amount of minimum heat capacity (see Eq. 8) and the exergy loss increase. But, as seen in Fig. 5, the ratio of exergy loss to the minimum heat capacity as the indicator of the dimensionless exergy loss decreases. It shows that the increase in the minimum heat capacity was more than the increase in the exergy loss. To present the dimensionless exergy loss as a function of the Reynolds number and the wavy strip angle, an empirical correlation was presented by curve fitting of experimental results as follow:

$$e = 0.144\mathrm{Re}^{-0.448}\left(\frac{l}{L}\right)^{-0.285} \tag{13}$$

Fig. 6 and Fig. 7 show the variations of the exergy loss and dimensionless exergy loss of the contained wavy strips with three different widths versus the Reynolds number. The results indicate that both exergy loss and dimensionless exergy loss increase with increase in wavy strips width. As shown in these two figures the minimum values occur in the heat exchanger with wavy strip of 8mm width. This is due to the lower flow mixing and heat transfer rate.

Again, curve fitting of experimental result was used to propose a correlation for the dimensionless exergy loss of a double tube heat exchanger as a function of the Reynolds number and wavy strip width as follows:

$$e = 0.2757\text{Re}^{-0.5126}(\frac{w}{L})^{0.117} \tag{14}$$

Fig. 7 also shows that the above mentioned correlation can predict the experimental results very good.

Fig. 8 and Fig. 9 present variations of the dimensionless exergy loss with the effectiveness for a double tube heat exchanger containing wavy strips of various angles and widths. Regarding Eq. 4, the effectiveness of a double tube heat exchanger is a function of the heat transfer rate and the maximum possible heat transfer rate. For a given hot water flow rate, and inlet hot and cold water temperatures, by increasing the wavy strip width or decreasing the wavy strip angle, the heat transfer rate increases and the maximum possible heat transfer rate remains constant. So, the effectiveness of the double tube heat exchanger increases by increasing the wavy strip width or decreasing the wavy strip angle. Higher heat transfer rates result in increasing irreversibility and exergy loss, so by increasing the effectiveness, the dimensionless exergy loss increases.

4.3. The effect of flow and thermodynamic properties

4.3.1 The effect of cold water flow rate

Cold water flow rate variation affects the exergy loss and dimensionless exergy loss of a heat exchanger. In this section, a wavy strip turbulator with a width of 8mm and an angle of 60-degree was placed along the inner tube in the axial direction. The inlet hot and cold water temperatures were kept at 53˚C and 14.5 ˚C, respectively and the effect of the cold water flow rates on the exergy loss and dimensionless exergy loss was studied. Variations of the exergy loss and dimensionless exergy loss versus the Reynolds number for different cold water flow rates were given in Fig. 10 and Fig. 11. As mentioned in Eq. 10, dimensionless exergy loss is a function of minimum heat capacity and exergy loss. By increasing the cold water flow rate, the heat transfer between hot and cold water increases which results in an increase in the irreversibility and the exergy loss. Furthermore, Eq. 8 shows that the minimum heat capacity depends on the hot water flow rate. So, for a given hot water flow rate, the minimum heat capacity does not change. As a result, for a given hot water flow rate, the numerator of Eq. 4 increases by increasing the cold water flow rate while its denominator remains unchanged. Hence, a higher cold water flow rate enhances the dimensionless exergy loss irreversibility and consequently increases the exergy loss and the dimensionless exergy loss.

By curve fitting of the experimental results of dimensionless exergy loss for different cold and hot water flow rates (various heat capacity ratios) Eq. 15 as below was derived.

$$e = 0.0225 Re^{-0.268} C_r^{-0.219}, \tag{15}$$

Where $0.2 < C_r < 1$.

Fig. 12 shows the variation of the dimensionless exergy loss versus Reynolds number for different heat capacity ratios ($C_r = C_{min}/C_{max}$) and Eq. (15). The dimensionless exergy loss at a given Reynolds number, increases with the decrease of C_r, which can be due to the maximum heat capacity ($C_{min} = \dot{m}_h c_{p,w}$) increase or minimum heat capacity ($C_{max} = \dot{m}_c c_{p,w}$) decrement. It is also shown that the proposed correlation is in a very good agreement with experimental results.

4.3.2 The effect of inlet hot water temperature

Besides the geometrical characteristic of wavy strip turbulators, the inlet hot water temperature can also affect the exergy loss. Therefore, in this section, the effect of the inlet hot water temperature on exergy loss and dimensionless exergy loss are investigated. The inlet cold water temperature and cold water flow rate were kept at constant values and experiments were conducted with three different inlets hot water temperatures of 43, 53, and 63˚C. Increase in the temperature difference between hot and cold water, which drives heat transfer, causes an increase in the irreversibility and the exergy loss. An increase in inlet hot water temperature leads to a rise in the temperature difference; hence, the irreversibility and the exergy loss increase. Variations of the exergy loss and dimensionless exergy loss versus the Reynolds number for different inlet hot water temperatures were presented in Figs. 13 and Fig. 14, respectively. The results indicated that both exergy loss and dimensionless exergy loss increase with the increasing of inlet hot water temperature. It should be mentioned that for a given hot water flow rate and inlet cold water temperature, an increasing inlet hot water temperature decreases dynamic viscosity, thus, increases the Reynolds number.

5. Conclusions

In this paper an experimental investigation into the effect of flow and thermodynamic properties and geometrical characteristics of wavy strips on exergy loss and dimensionless exergy loss in a double tube heat exchanger was performed. Using water as the test fluid, the experiments were carried out in a turbulent flow regime ($4,300 < Re < 12,000$). It was found that the exergy loss and dimensionless exergy loss were directly dependent on the width of the wavy strip and inversely dependent on the angle of the wavy strip. For the considered width and angles of wavy strips, the heat exchanger equipped with the wavy strips showed a higher exergy loss and dimensionless exergy loss in comparison with the tube with no wavy strips. It was also realized that the exergy loss of the wavy strip with the angle of 150 degrees and a width of 8 mm is less than others. This observation may be attributed to the lower flow mixing, which leads to less heat transfer and less irreversibility. The results also indicates that both exergy loss and dimensionless exergy loss increase with an increase in the inlet hot water temperature and a decrease in the heat capacity ratio.

References:

[1] S.S Hsieg, M.Y. Wen, Augmentative heat transfer in a vertical 2-pass tube through an air box by use of strip type inserts, International Journal of Heat and Mass Transfer 36(12) (1993) 3043-3057.

[2] I. Yaningsih, T. Istanto, A.T. Wijayanta, Experimental study of heat transfer enhancement in a concentric double pipe heat exchanger with different axial pitch ratio of perforated twisted tape inserts, AIP Conference Proceedings 1717 (2016) 4943436.

[3] P. Bharadwaj, A.D. Khondge, A.W. Date, Heat transfer and pressure drop in a spirally grooved tube with twisted tape insert, International Journal of Heat and Mass Transfer 52(7-8) (2009) 1938-1944.

[4] H. Bas, V. Ozceyhan, Heat transfer enhancement in a tube with twisted tape inserts placed separately from the tube wall, Experimental Thermal and Fluid Science 41 (2012) 51–58.

[5] V. Hejazi, M.A. Akhavan-Behabadi, A. Afshari, Experimental investigation of twisted tape inserts performance on condensation heat transfer enhancement and pressure drop, International Communications in Heat and Mass Transfer 37 (2010) 1376–1387.

[6] A. Garcıa, J.P. Solano, P.G. Vicente, A. Viedma, Enhancement of laminar and transitional flow heat transfer in tubes by means of wire coil inserts, International Journal of Heat and Mass Transfer 50 (2007) 3176–3189.

[7] I. Dincer, M.A. Rosen, Exergy (Second Edition): energy, environment and sustainable development, Elsevier 2013. http://dx.doi.org/10.1016/B978-0-08-097089-9.01001-9

[8] R.C. Prasad, J. Shen, Performance evaluation of convective heat transfer enhancement devices using exergy analysis, International Journal of Heat and Mass Transfer 36(17) (1993) 4193-4197.

[9] R.L. Cornelissen, G.G. Hirs, Thermodynamic optimisation of a heat exchanger, International Journal of Heat and Mass Transfer 42 (1999) 951-959.

[10] R.S. Doohan, P.K. Kush, G. Maheshwari, Exergy based optimization and experimental evaluation of plate fin heat exchanger, Applied Thermal Engineering 102 (2016) 80–90.

[11] O. Ipek, B. Kılıç, B. Gürel, Experimental investigation of exergy loss analysis in newly designed compact heat exchangers, Energy 124 (2017) 330-335.

[12] R.R. Sahoo, P. Ghosh,J. Sarkar, Energy and exergy comparisons of water based optimum brines as coolants for rectangular fin automotive radiator, International Journal of Heat and Mass Transfer 105 (2017) 690–696.

[13] Moffat, Describing the uncertainties in experimental results, Experimental Thermal and Fluid Sciences 1 (1988) 3-17.

[14] S. Khorasani, A. Dadvand, Effect of air bubble injection on the performance of a horizontal helical shell and coiled tube heat exchanger: An experimental study, Applied Thermal Engineering 111 (2017) 676–683.

[15] S.D. Pandey, V.K. Nema, An experimental investigation of exergy loss reduction in corrugated plate heat exchanger, Energy 36 (2011) 2997-3001.

[16] A. Durmuş, Heat transfer and energy loss in a concentric heat exchanger with snail entrance, International communications in heat and mass transfer 29 (2002) 303-312.

[17] E.K. Akpinar, Y. Bicer, Investigation of heat transfer and exergy loss in a concentric double pipe exchanger equipped with swirl generators, International Journal of Thermal Sciences 44 (2005) 598–607.

[18] T.L. Bergman, A.S. Lavine, F.P. Incropera, D.P. Dewitt, Introduction to heat transfer (Sixth Edition), John Wiley & Sons 2011.

[19] J.W. Rose, Heat-transfer coefficients, Wilson plot and accuracy of thermal measurements, Experimental Thermal and Fluid Science 28 (2-3) (2004) 77-86.

Figure captions

Fig. 1 Schematic view of experimental set up

Fig. 2 Schematic view of wavy strips

Fig. 3 Validation of Nusselt number for the plain tube for m_c = 14 L/min and $T_{h,i} = 63^o$ C

Fig. 4 Exergy loss versus Reynolds number for various wavy strip angles

Fig. 5 Dimensionless exergy loss versus Reynolds number for various wavy strip angles

Fig. 6 Exergy loss versus Reynolds number for various wavy strip widths with $\theta = 60^o$

Fig. 7 Dimensionless exergy loss versus Reynolds number for various wavy strip widths with $\theta = 60^o$

Fig. 8 Dimensionless exergy loss versus effectiveness for various wavy strip angles with m_c = 14 L/min and w = 0.8 cm

Fig. 9 Dimensionless exergy loss versus effectiveness for various wavy strip widths with $\theta = 60^o$

Fig. 10 Exergy loss versus Reynolds number for various cold water flow rates

Fig. 11 Dimensionless exergy loss versus Reynolds number for various cold water flow rates

Fig. 12 Dimensionless exergy loss versus Reynolds number for various heat capacity ratios

Fig. 13 Exergy loss versus Reynolds number for various inlet hot water temperatures

Fig. 14 Dimensionless exergy loss versus Reynolds number for various inlet hot water temperatures

Table 1
Uncertainty of parameters

Parameter	Uncertainty
Uncertainty in measurement	
measurement of volume flow rate temperature	±0.5 ^{0}C
measurement of volume flow rate	±4 %
Uncertainty in calculating the result	
Reynolds number	±3.4 %
Exergy loss	±12.3 %
Uncertainty in reading values of table ($C_{p,w}$ etc)	±0.1–0.2 %

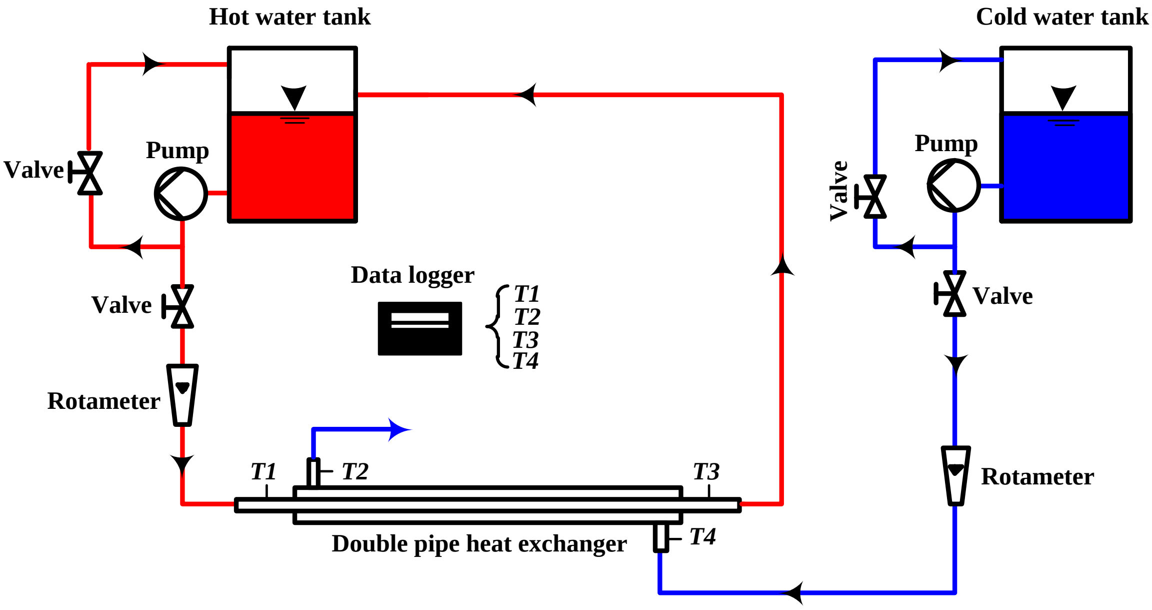

Hot water tank
Cold water tank
Valve
Pump
Pump
Valve
Valve
Valve
Data logger
T1
T2
T3
T4
Rotameter
Rotameter
T1
T2
T3
Double pipe heat exchanger
T4

Fig. 1

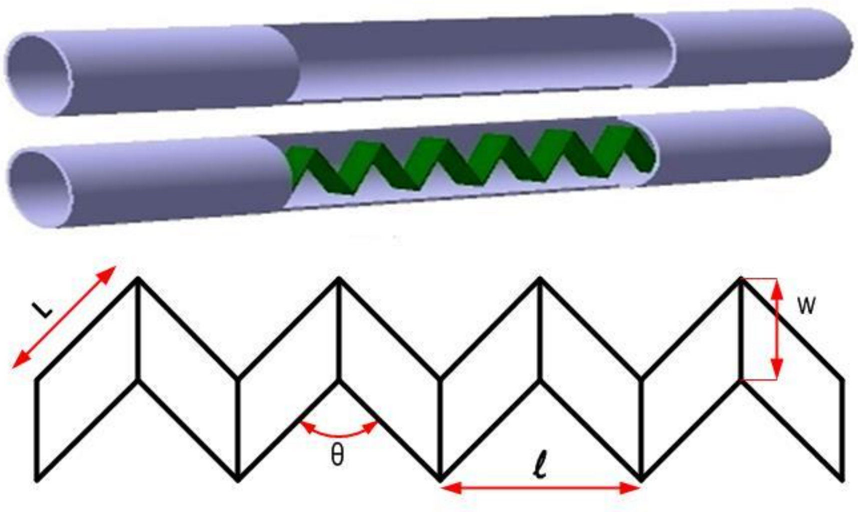


Fig. 2

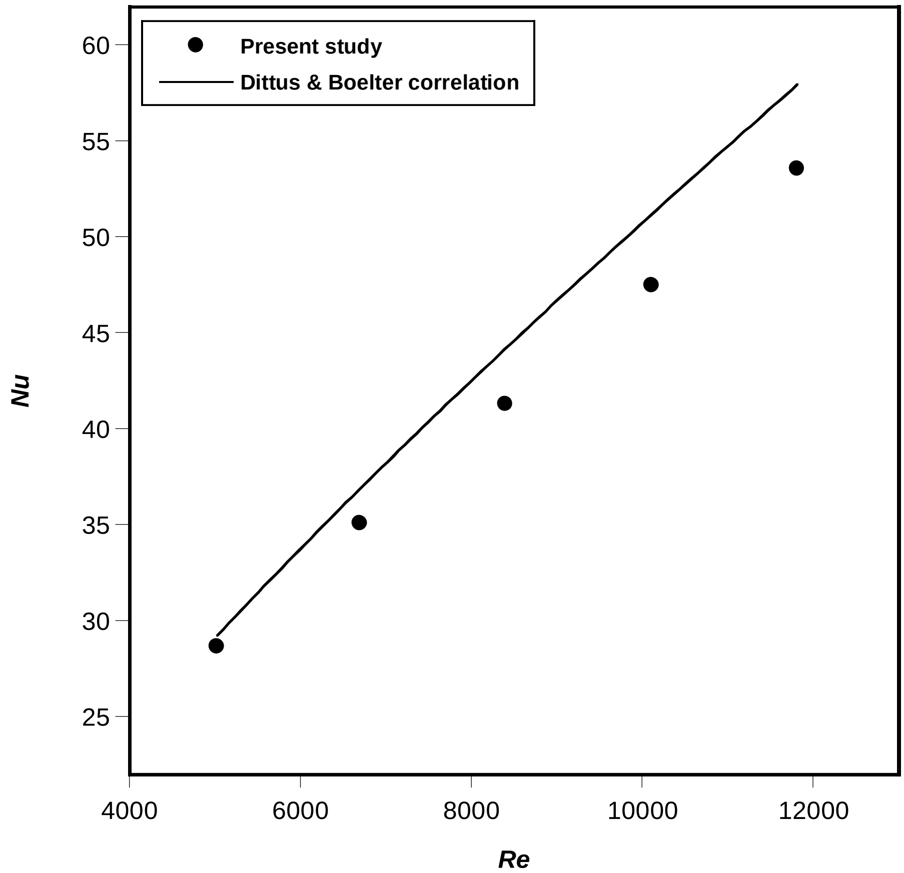

Present study
Dittus & Boelter correlation
60
55
50
45
40
35
30
25
Nu
4000
6000
8000
10000
12000
Re

Fig. 3

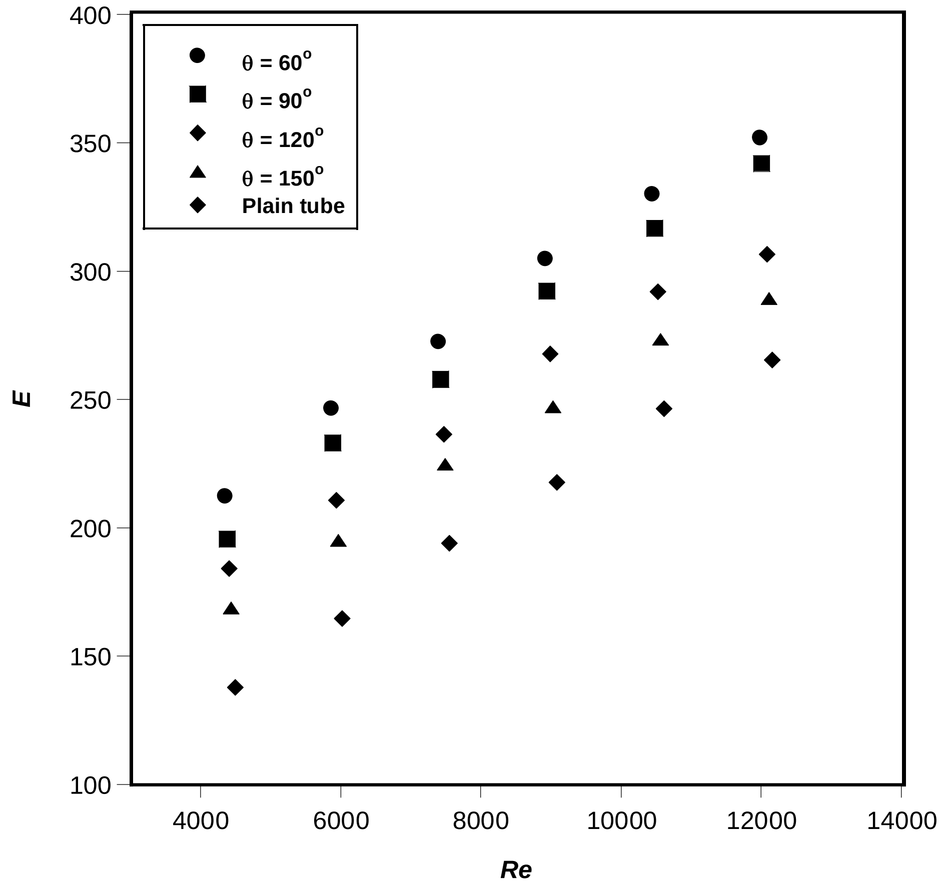

θ = 60°
θ = 90°
θ = 120°
θ = 150°
Plain tube
400
350
300
250
200
150
100
E
4000
6000
8000
10000
12000
14000
Re

Fig. 4

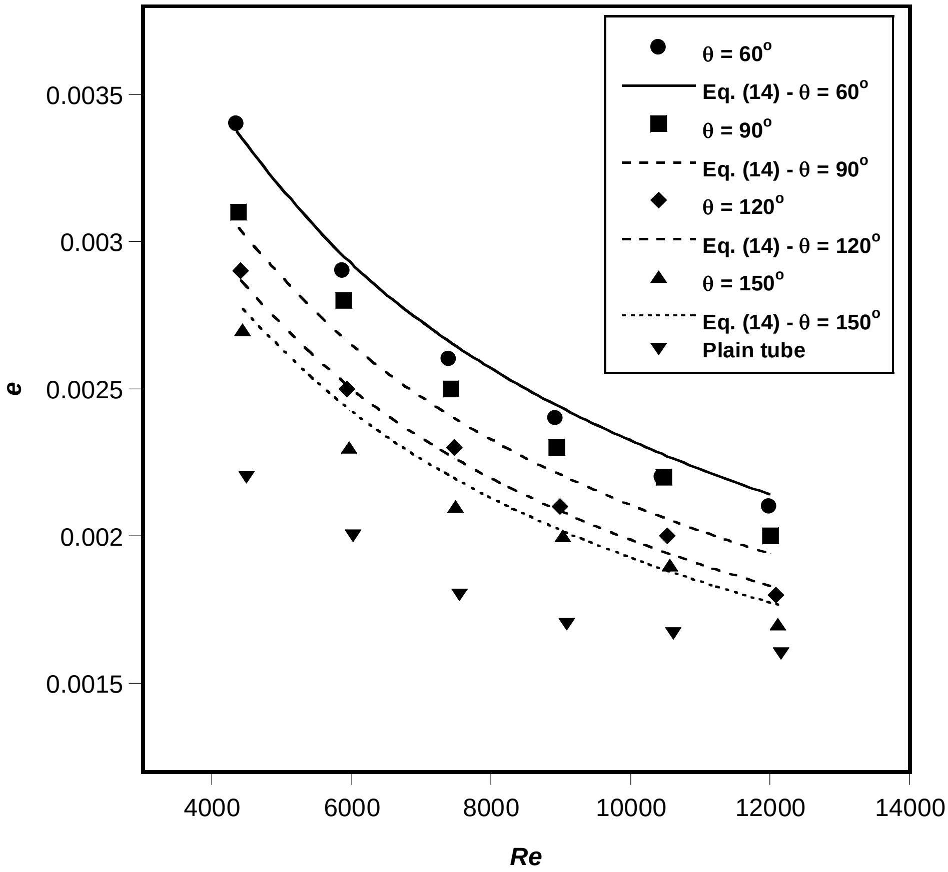
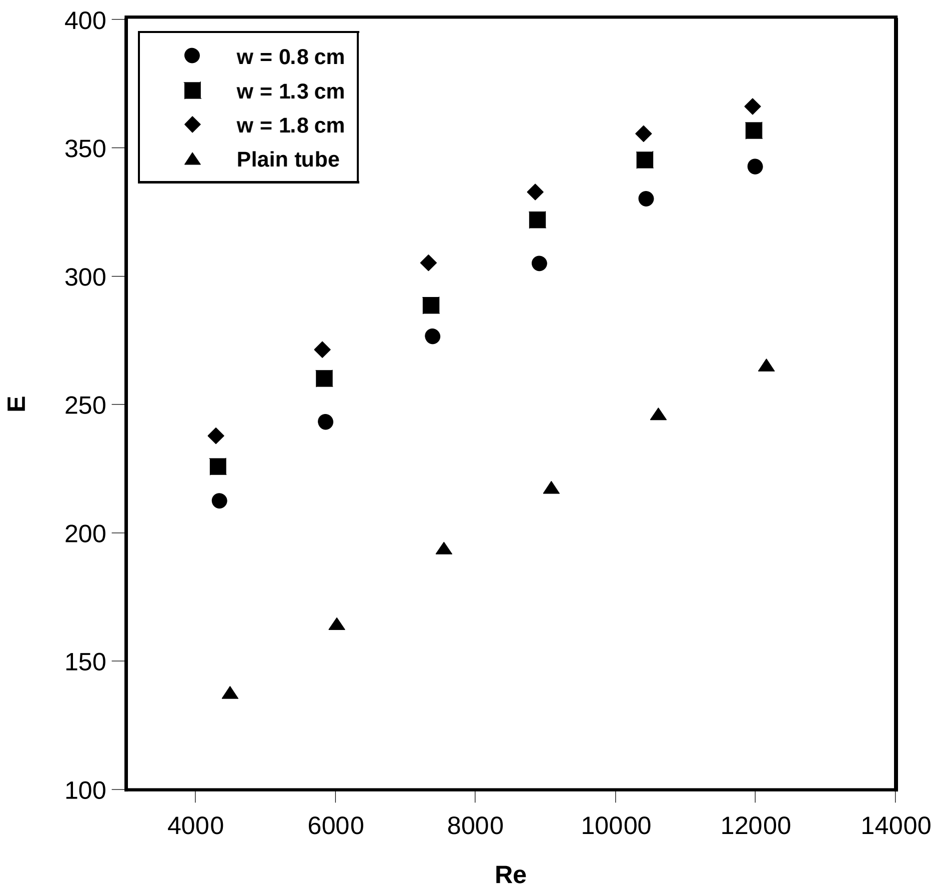
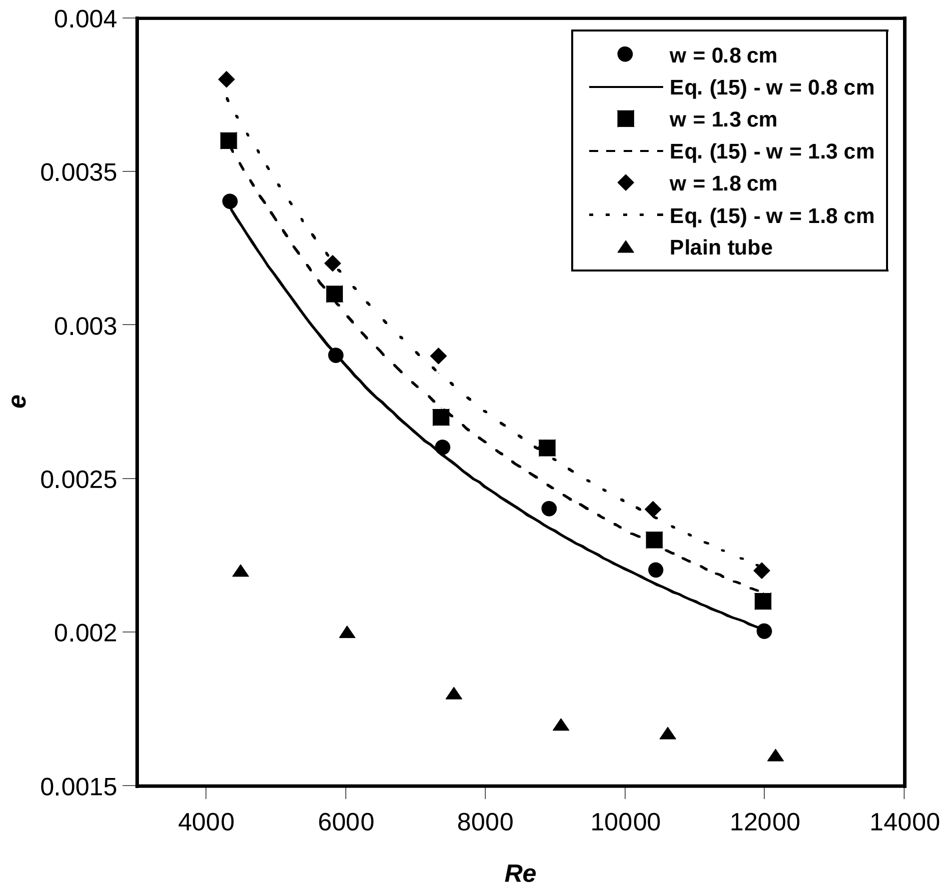

θ = 60°
Eq. (14) - θ = 60°
θ = 90°
Eq. (14) - θ = 90°
θ = 120°
Eq. (14) - θ = 120°
θ = 150°
Eq. (14) - θ = 150°
Plain tube
0.0035
0.003
0.0025
0.002
0.0015
e
4000
6000
8000
10000
12000
14000
Re

Fig. 5



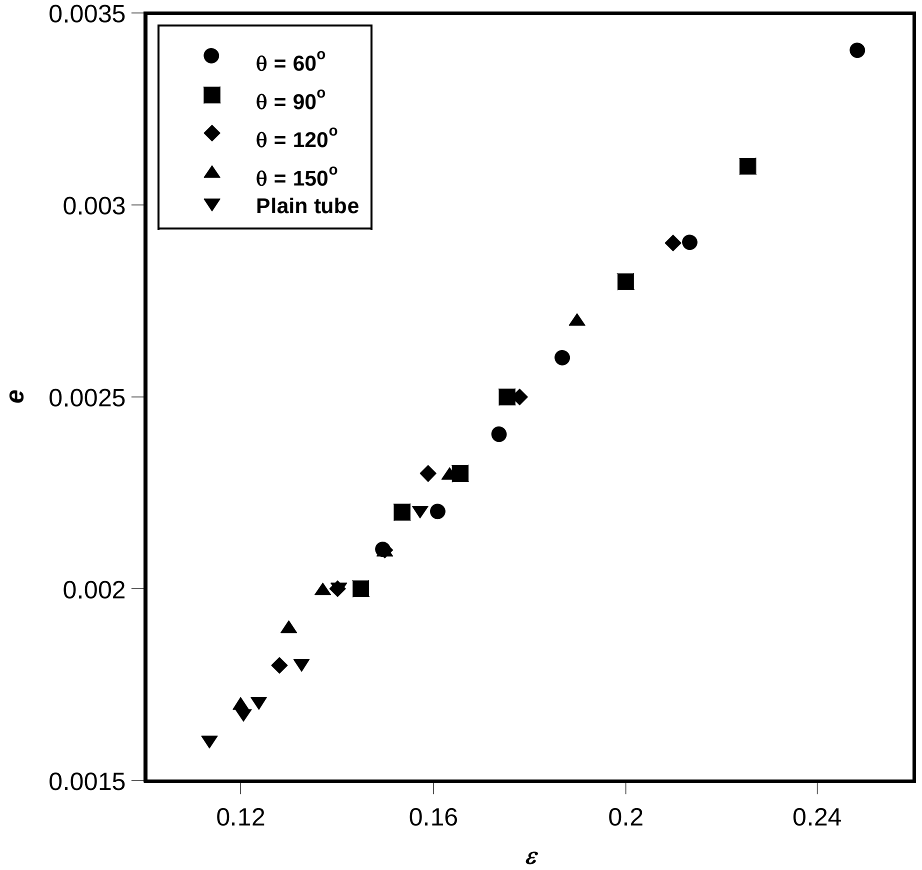
w = 0.8 cm
w = 1.3 cm
w = 1.8 cm
Plain tube
E
Re
100
150
200
250
300
350
400
4000
6000
8000
10000
12000
14000

Fig. 6



0.004
0.0035
0.003
0.0025
0.002
0.0015
e
4000
6000
8000
10000
12000
14000
Re
w = 0.8 cm
Eq. (15) - w = 0.8 cm
w = 1.3 cm
Eq. (15) - w = 1.3 cm
w = 1.8 cm
Eq. (15) - w = 1.8 cm
Plain tube

Fig. 7



0.0035
0.003
0.0025
0.002
0.0015
e
0.12
0.16
0.2
0.24
ε
θ = 60°
θ = 90°
θ = 120°
θ = 150°
Plain tube

Fig. 8

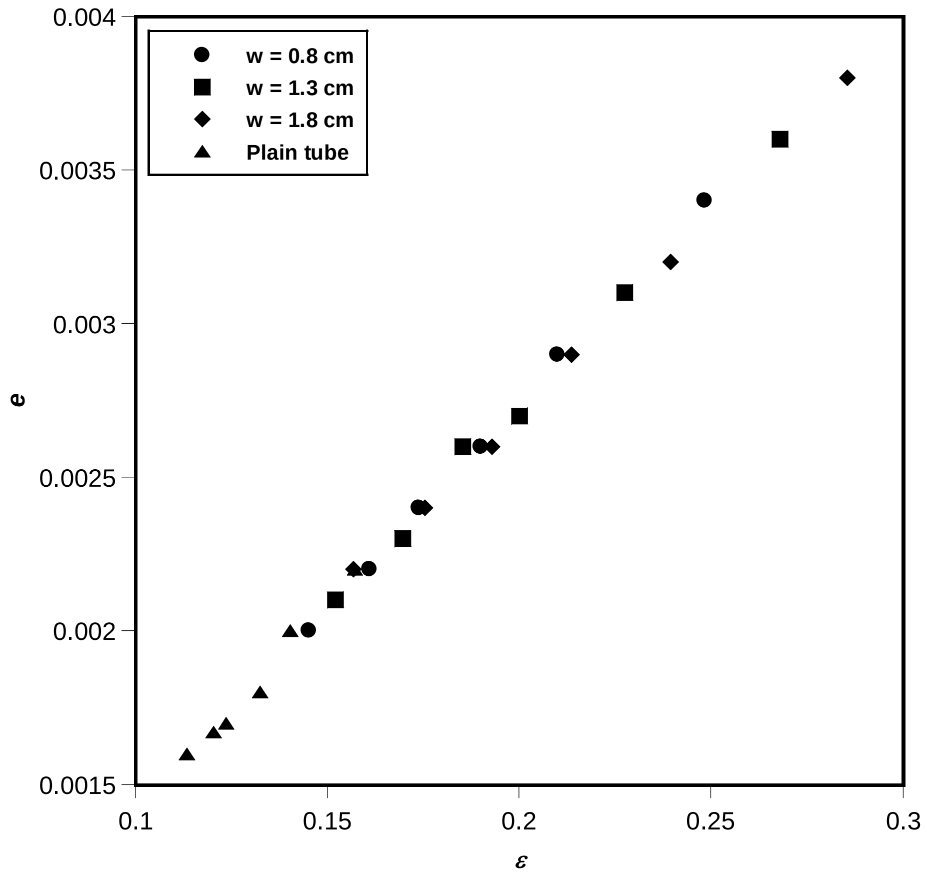

w = 0.8 cm
w = 1.3 cm
w = 1.8 cm
Plain tube
0.004
0.0035
0.003
0.0025
0.002
0.0015
0.1
0.15
0.2
0.25
0.3
e
ε

Fig. 9

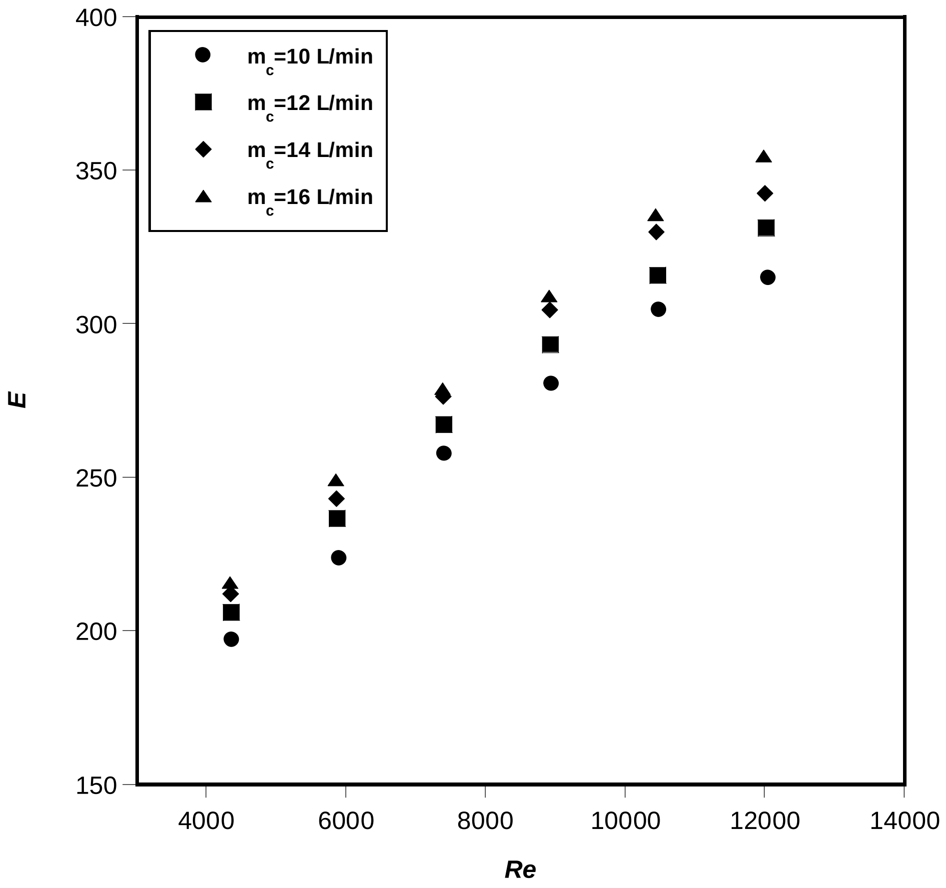

400
350
300
250
200
150
E
4000
6000
8000
10000
12000
14000
Re
m_c=10 L/min
m_c=12 L/min
m_c=14 L/min
m_c=16 L/min

Fig. 10

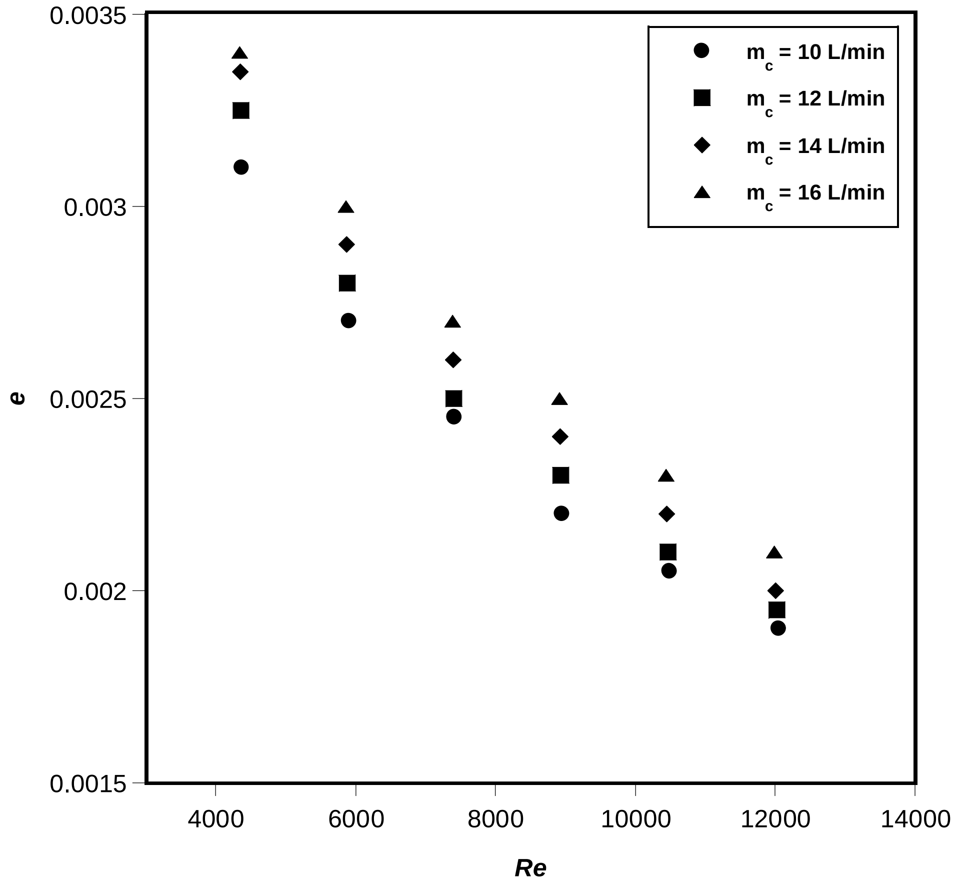
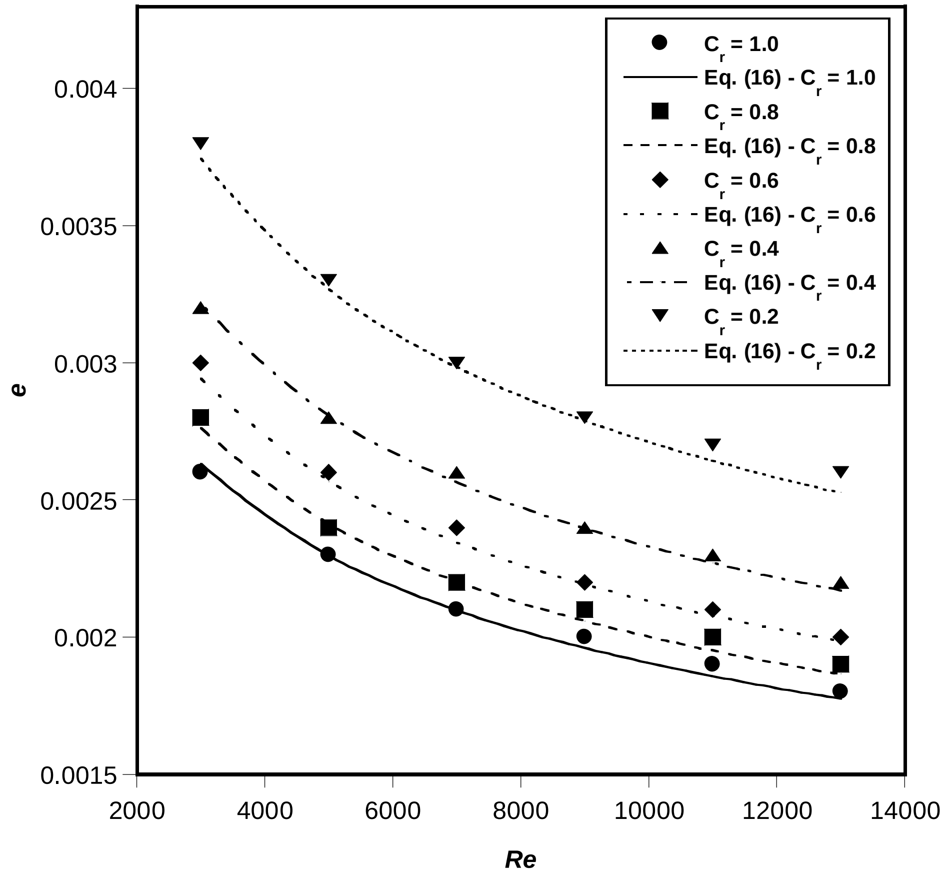
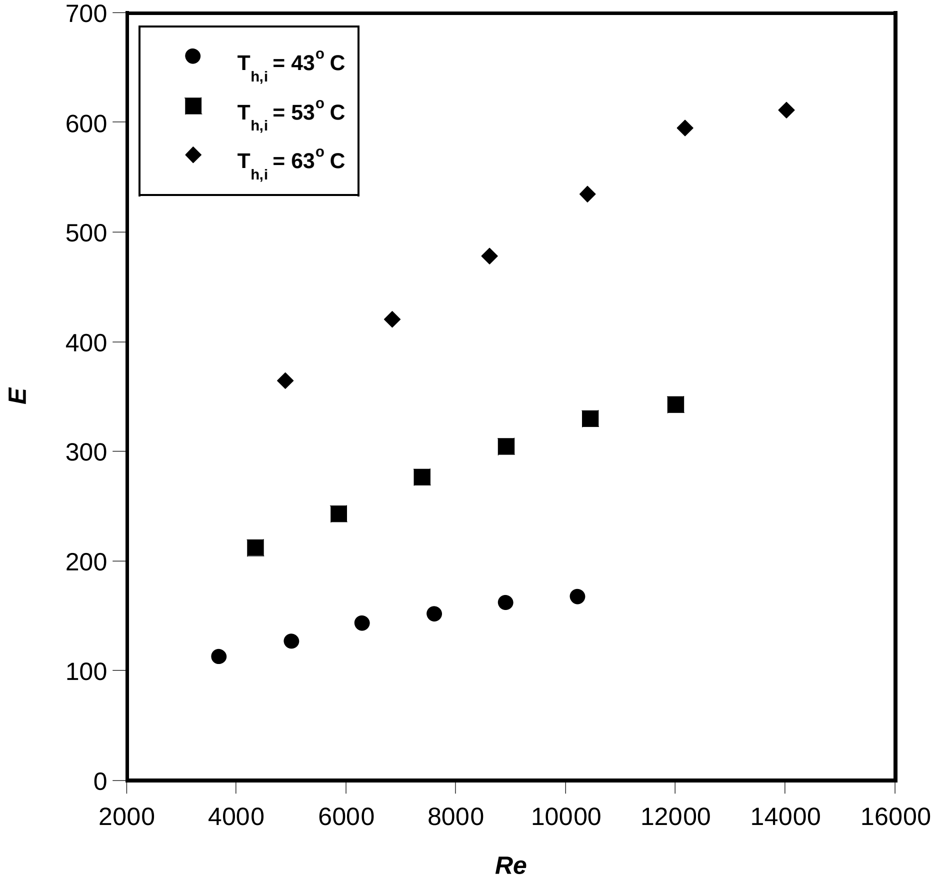

0.0035
0.003
0.0025
0.002
0.0015
e
4000
6000
8000
10000
12000
14000
Re
m_c = 10 L/min
m_c = 12 L/min
m_c = 14 L/min
m_c = 16 L/min

Fig. 11



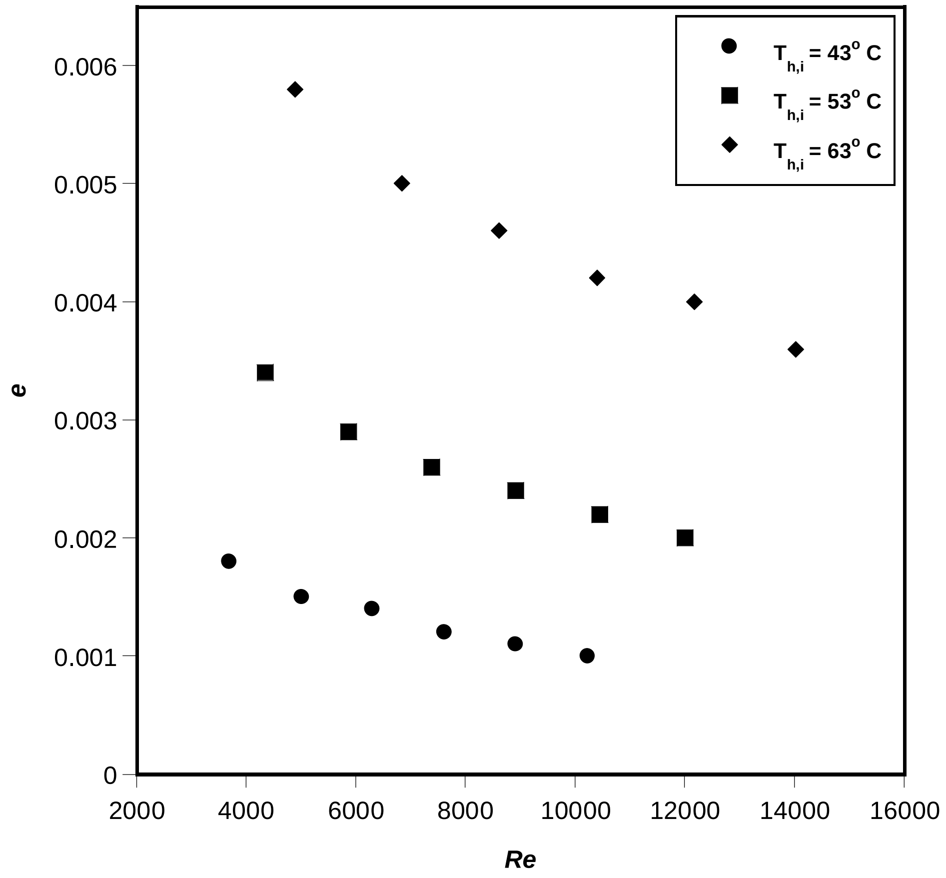
0.004
0.0035
0.003
0.0025
0.002
0.0015
e
2000
4000
6000
8000
10000
12000
14000
Re
C r = 1.0
Eq. (16) - C r = 1.0
C r = 0.8
Eq. (16) - C r = 0.8
C r = 0.6
Eq. (16) - C r = 0.6
C r = 0.4
Eq. (16) - C r = 0.4
C r = 0.2
Eq. (16) - C r = 0.2

Fig. 12



T h,i = 43° C
T h,i = 53° C
T h,i = 63° C
700
600
500
400
300
200
100
0
E
2000
4000
6000
8000
10000
12000
14000
16000
Re

Fig. 13



$T_{h,i}$ = 43° C
$T_{h,i}$ = 53° C
$T_{h,i}$ = 63° C
0.006
0.005
0.004
0.003
0.002
0.001
0
e
2000
4000
6000
8000
10000
12000
14000
16000
Re

Fig. 14