Case Studies in Thermal Engineering 28 (2021) 101587

Contents lists available at ScienceDirect

Case Studies in Thermal Engineering

journal homepage: www.elsevier.com/locate/csite

Thermal, hydraulic, exergitic and economic evaluation of a flat tube heat exchanger equipped with a plain and modified conical turbulator

Heng Chen [a,b,***], Hamdi Ayed [c,d], Riadh Marzouki [e], Faezeh Emami [f,*], Ibrahim Mahariq [g], Fahd Jarad [h,i,**]

[a] *School of Science, Xijing University, Xi'an, Shaanxi, 710123, China*
[b] *Shaanxi Engineering Research Center of Controllable Neutron Source,Xijing University, Xi'an, Shaanxi, 710123, China*
[c] *Department of Civil Engineering, College of Engineering, King Khalid University, Abha, 61421, Saudi Arabia*
[d] *Higher Institute of Transport and Logistics of Sousse, University Sousse, Tunisia*
[e] *Department of Chemistry, College of Science, King Khalid University, Abha, 61421, Saudi Arabia*
[f] *Sustainable Management of Natural Resources and Environment Research Group, Faculty of Environment and Labour Safety, Ton Duc Thang University, Ho Chi Minh City, Viet Nam*
[g] *College of Engineering and Technology, American University of the Middle East, Kuwait*
[h] *Department of Mathematics, Cankaya University, Etimesgut, Ankara, Turkey*
[i] *Department of Medical Research, China Medical University Hospital, China Medical University, Taichung, Taiwan*

ARTICLE INFO

Keywords:
Double tube heat exchanger
Flat tube
Modified conical turbulator
Heat transfer
Pressure drop

ABSTRACT

Increasing use of fossil fuels has led to serious consequences on the environment including global warming, pollution, and increased costs of heating systems. This indicates the importance of improving the efficiency of devices used in heating and air conditioning systems such as heat exchangers. In this study, it was tried to enhance the performance of the heat exchanger by simultaneous use of a flat tube as a central tube of the heat exchanger and dual conic turbulator. In addition to plain conical turbulator, the influence of the conical turbulator with aerodynamic geometry was investigated on the heat transfer and pressure drop. Finally, economic evaluation of the studied cases was done based on the thermal performance enhancement factor (TEF) and the net profit of unit transferred heat load (η_p). Tests were conducted in the hot water flow rate ranging 0.033–0.0678 L/s while cold water flow rate was kept constant at 0.166 L/s. The results indicated that the heat transfers and exergy loss of the flat tube heat exchanger with a dual modified conical turbulator was up to 33% and 30% higher than the plain tube heat exchanger, respectively. Also, a heat exchanger with a modified dual turbulator with convergent embellishment (case B) selected as the economic case and has the highest TEF and η_p . It was revealed that the application of enhanced conical turbulator would lead in the increment of η_p up to 1.26 times.

* Corresponding author.
** Corresponding author. Department of Mathematics, Cankaya University, Etimesgut, Ankara, Turkey.
*** Corresponding author. School of Science, Xijing University, Xi'an, Shaanxi, 710123, China.
E-mail addresses: chenheng@xijing.edu.cn (H. Chen), hayed@kku.edu.sa (H. Ayed), rmarzouki@kku.edu.sa (R. Marzouki), emami.f.s94@gmail.com (F. Emami), ibmahariq@gmail.com (I. Mahariq), fahd@cankaya.edu.tr (F. Jarad).

https://doi.org/10.1016/j.csite.2021.101587
Received 16 July 2021; Received in revised form 29 September 2021; Accepted 23 October 2021
Available online 25 October 2021

1. Introduction

The lack of energy sources and the high cost of those that are accessible have increased the necessity of research into improving the performance of thermal systems. Amongst various thermal systems, the heat exchanger is variously used in the thermal process [1]. In recent years, heat transfer improvement approaches have gotten a lot of attention in the heat exchanger to reduce energy loss, size, and cost of heating systems. Active techniques and passive methods are two types of heat transfer augmentation methods that are widely

Nomenclature

$C_{p,w}$	specific heat at constant pressure, $\frac{J}{kg\ K}$
Nu	Nusslet number, dimensionless
h	Heat transfer coefficient, $\frac{W}{m^2.\ K}$
$\dot{m}$	mass flow rate, $\frac{kg}{s}$
Re	Reynolds number
T	water temperature, K
p	Pressure, Pa
k	turbulent kinetic energy, $\frac{m^2}{s^2}$
Greek symbols	
ε	turbulent dissipation rate
μ	is dynamic viscosity
Δ	difference operator
Subscripts	
c	Cold
h	Hot
i	Inlet
o	Outlet
e	environment condition

used [2]. Using turbulators of various designs in the middle tube of double tube heat exchangers, using enhanced tube and using nanofluids [3,4] as working fluid are the passive approaches for improving thermal performance. In recent years, plain and modified helical and twisted tape turbulators have attracted the attention of researchers. The first studies in the field of twisted tape turbulators were carried by Fahed et al. [5] and Moria et al. [6], who assessed the influence of the turbulator's twisting ratio, width, and pitch on heat transfer and pressure drop. Their research showed that the presence of twisted tape causes swirling flows and enhances the heat transfer coefficient significantly. Eiamsa-ard et al. [7,8] studied the effect of the configuration of plain and modified twisted tape turbulators on the heat transfer and pressure drop of twin-tube heat exchangers in additional investigations. They discovered that the orientation and shape of the twisted tape had a significant effect on the heat exchanger's effeciency. They discovered that the proper orientation of the twisted tape may increase the *Nu* number by up to 90%. Mashoufi et al. [9] investigated heat transfer and pressure drop in a twin tube heat exchanger using a plain and punched twisted tape turbulator. The optimal turbulator was chosen using the TEF factor. They also provided additional correlations for predicting the *Nu* and the friction factor. Eiamsa-ard et al. [10] used a louvered strip turbulator in the twin tube heat exchanger's center tube. Furthermore, the impact of this sort of turbulator on pressure drop was discussed. They claimed that using the stated sort of turbulatore may enhance the *Nu* number by up to 70%. To improve heat transfer, Thejaraju et al. [11] used a para winglet tape air-to-air twin tube heat exchanger. Pourahmad and Pesteei [12,13] studied the effects of a wavy strip turbulator insert on a double tube heat exchanger's efficiency, pressure drop, and exergy loss. Inside the central tube, four distinct angles and three different widths of turbulators were manufactured and implanted. The TEF factor was used to find the best turbulator. Khorasani et al. [14] examined the impact of employing a spring wire turbulator and found that it is quite efficient at increasing the *Nu* number. They also created empirical correlations to estimate the exergy loss, friction factor, and *Nu* number of a helical heat exchanger with a spring wire turbulator. From the perspective of the second law of thermodynamics, Fan et al. [15] did a second law assessment on the impact of perforated twisted tape and reported that utilizing the aforementioned turbulators may greatly improve the heat exchanger's performance. Nakhchi et al. [16] studied the effect of perforated elliptic turbulators on the performance of a twin-tube heat exchanger. regarding their findings, utilizing the aforementioned sort of turbulatores enhanced the heat transfer rate by 217%. Sheikholeslami et al. [17] investigated the impact of nanofluid and twisted tape turbulators in flat and straight tube heat exchangers on heat transfer rate. They implemented the two-phase approach to simulate the behavior of the nanofluids. Kongkaitpaiboon et al. [18] have studied experimentally the effect of using typical conical-ring and perforated conical-ring turbulators on heat transfer and pressure drop in a tube at a constant heat flux. In addition, they chose the most cost-effective option using the thermal performance enhancement factor (TEF). The results indicate the significant effect of using typical conical-ring and perforated conical-ring turbulators on heat transfer and pressure drop. Also, it was found that in the case of perforated conical-ring turbulator, the rate of heat transfers and pressure drop is less than in the case of a typical conical-ring turbulator. The study ranges for the TEF was from 0.55 to 0.91, and the maximum TEF was reported for the perforated conical-ring turbulator. Karakaya and Durmus [19] have examined the effect of using the conical spring turbulators inside a tube with a constant heat flux on heat transfer, pressure drop, and exergy losses. Turbulators were made in three different angles including 30°, 45°, and 60°. Different arrangements of converging conical-ring (C.R.) and diverging conical-ring (D.R.) turbulators and a combination of both were also examined. The results revealed that heat transfer, pressure drop, and exergy losses increase simultaneously using turbulator. The maximum rate of heat transfer, pressure drop, and exergy losses was observed for a 30° turbulator with a D.R. arrangement. In addition, in other studies, the effect of using different arrangements of conical-ring turbulators, including conical-nozzle [20], conical cut-out turbulators [21,22], and a combination of conical-ring and twisted-ring turbulators [23] has been also examined.

Based on the findings of past research, it can be stated that numerous turbulator designs have been used to enhance the heat transfer in the double tube heat exchangers, including wavy strip, twisted tapes, and conical-ring turbulators. They increase pressure drop besides increasing heat transfer. This study tries to present a conical turbulator with new geometry and examine its effect on heat transfer and pressure drop. The innovation of this study lies in how to make aerodynamic the geometry of the conical turbulator. The aerodynamic geometry of the conical turbulator is expected, on the one hand, to reduce the flow turbulence and consequently reduce the turbulator effect on the heat transfer, and on the other hand, reduce the pressure drop. Thus, the thermal performance

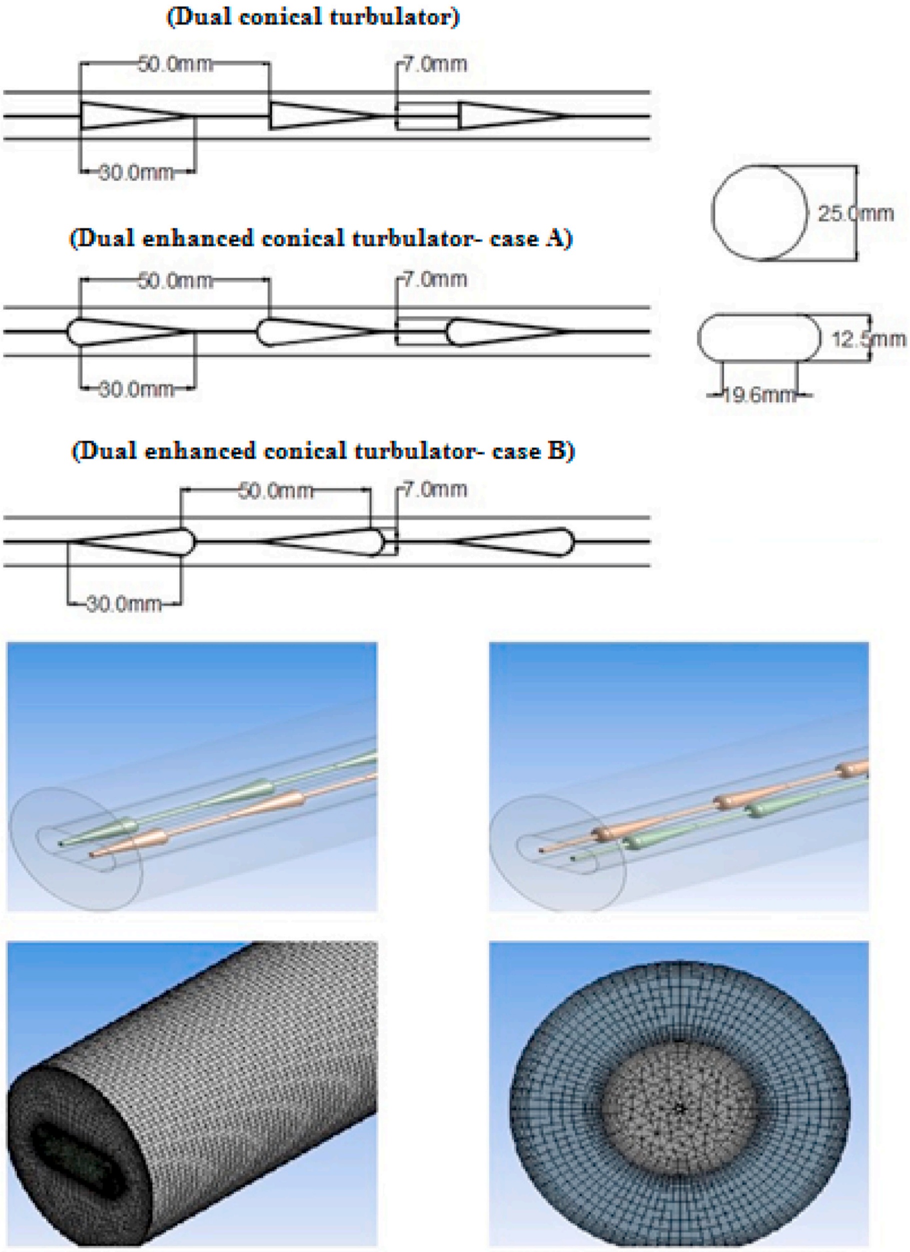

Fig. 1. A view of the plain and flat tube heat exchanger equipped with simple and enhanced dual conical turbulator.

enhancement factor (TEF) and the net profit of unit transferred heat load (η_p) is used to examine the study geometry economically.

2. Numerical modeling

2.1. Physical model

The studied heat exchanger had a length of 1 m. The internal tube was from copper and designed in two modes: plain and flat. The external tube had a diameter of 54 mm and was insulated. The flat tube was designed in such a way that its circumference (i.e. the surface by which the heat transfer occurs) was equal to the circumference of the circular tube. After determining the effect of using a flat tube on heat transfer and pressure drop, dual conical turbulators with a height of 7 mm were installed inside the heat exchanger's central tube, and their impact on heat transfer rate and pressure drop was determined. some modifications were made to the geometry of the dual conical turbulator to optimize it and decline the pressure drop as a result of the presence of this turbulator. Fig. 1 depicts a schematic picture of the investigated modes as well as an example of heat exchanger meshing.

2.2. Governing equations

The working fluid in this article was considered to be water. Water was thought to be an incompressible newtonian fluid. In addition, the working fluid's thermophysical characteristics were believed to be constant. It's worth noting that gravity's effect was ignored in this research. The governing equations of the current research might be expressed as follows, taking into account the aforementioned assumptions [9,24,25].

The continuity equation is as below:

$$\frac{\partial}{\partial x_i}(\rho u_i)=0 \tag{1}$$

The following is a summary of the Momentum equation:

$$\frac{\partial(\rho u_i u_j)}{\partial x_j}=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\mu\frac{\partial u_k}{\partial x_k}\delta_{ij} \tag{2}$$

Moreover, the following is the energy conservation equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_j C_p T-k\frac{\partial T}{\partial x_j}\right)=u_j\frac{\partial p}{\partial x_j}+\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)-\frac{2}{3}\mu\frac{\partial u_k}{\partial x_k}\delta_{ij} \tag{3}$$

In equations (1)–(3), the term μ stands for the dynamic viscosity. Also, the term k denotes the turbulent kinetic energy of the flow. Besides, the term C_p refers to the working fluid's specific heat capacity at a fixed pressure.

In this study, to simulate the turbulence behavior of the flow, the $k-\varepsilon$ model with the enhanced wall function as the wall treatment model was utilized. The mentioned models could be formulated as follows [9,25,26]:

$$\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}(\alpha_k\mu_{eff})\frac{\partial k}{\partial x_j}+G_k-\rho\varepsilon \tag{4}$$

$$\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}(\alpha_\varepsilon\mu_{eff})\frac{\partial\varepsilon}{\partial x_j}+C_{1\varepsilon}\frac{\varepsilon}{k}G_k-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{5}$$

In equations (4) and (5), the terms ε. μ_{eff} and μ_t denote the turbulence dissipation rate, effective viscosity, and turbulence viscosity, respectively. Also, the constants of $C_{1\varepsilon}$and $C_{2\varepsilon}$were considered to be 1.42 and1.68. Besides, the α_k was considered to be equal to α_ε which had a value of 1.393. it should be noted that the term G_k stands for the production of turbulent kinetic energy (TKE) which is derived from the average velocity gradients. Furthermore, the effective viscosity could be calculated as follows.

$$\mu_{eff}=\mu_t+\mu,\ \mu_t=\rho C_\mu\frac{k^2}{\varepsilon} \tag{6}$$

The preceding equations were solved using the finite volume method. For connecting pressure terms to velocity equations, the SIMPLE algorithm was used. Furthermore, for discretizing the diffusion terms the second-order upwind procedure was used. It should be noted that to be certain of the accuracy of the numerical method, the values of the residuals were set to be 10^{-6} for the energy and continuum equations.

2.3. Boundary conditions

The boundary conditions correspond to the constraints and conditions of the subject. The correct selection of the boundary conditions is very crucial to achieve reasonable outcomes from any numerical simulations. The outer wall has been isolated from the surrounding environment in the present study, whereas its inner wall was the heat transfer agent between the heat and cold fluids. The outer wall was isolated by selecting its heat flux as Zero. The inner wall was considered to have coupled heat transfer boundary conditions. It is noteworthy that for the hydrodynamic boundary condition of the inner and outer wall the No-slip boundary condition was selected. The inlet of both the tube and annulus were considered to have constant velocity inlet boundary conditions with constant flow temperatures of 313K and 288K for the hot and cold water flows, respectively. Both hot and cold water flows have their pressure outlet boundary conditions taken into account. It should be noted that both streams' output pressures were set to zero. It's worth noting that the tests were conducted in the hot water flow rate ranging 0.033–0.0678 L/s while cold water flow rate was kept constant at 0.166 L/s.

2.4. Validation of the model and grid independence evaluations

Of those very important parts that should be considered in any numerical solutions, are the validation of the results and mesh independence analysis. Through the present paper, the results of the *Nu* number were validated by the empirical results of the Dittus-Boelter and Gnielinski equation [27]. Furthermore, the findings of the Petukhov [28] equation were used to confirm the friction factor of a simple tube heat exchanger. Figs. 2 and 3 show a comparison of the current study's findings with those in the previous studies. It's worth mentioning that the most significant discrepancy between the *Nu* number and friction factor results was about 14% and 11%, respectively. Also, the mesh independence of the results of the study geometries was examined using six different mesh sizes. The optimal mesh numbers can be determined by calculating the Nusselt number in these meshes. Fig. 4 shows the changes of Nusselt number with the mesh numbers. The criterion for optimal mesh selection was considered to be as reaching the deviation of less than 2% between two consequent Nusselt numbers, giving an acceptable number of meshes. Results indicated that the Nusselt numbers for the last three meshes differ very little. In order to minimize test time, one before the last mesh is selected as the optimal number of meshes.

3. Data processing

3.1. Nusselt number, friction factor

The numerical results are in form of temperature and pressure, to calculate the intended parameters including Nusselt number, friction factor, and thermal efficiency, further calculations are required which will be mentioned in this section. Generally, the Nusselt number may be calculated in two ways. In one of the methods mentioned in the Kongkaitpaiboon et al. [18] and Promvonge et al. study [20,23], the surface temperature of the tube is needed to calculate the Nusselt number. This method is frequently used in studies in which constant heat flux is applied to a tube surface. In the second method used in Promvonge et al. [29] and Wongcharee et al. [30] studies, having input and output temperatures is sufficient to calculate the Nusselt number. This method is frequently used in studies on double tube heat exchangers. In the present study, the second method was used according to the evaluation of a double tube heat exchanger. First, the heat transfer rate between cold and hot water were calculated as follows:

$$q_h = \dot{m}_h c_{p.w} (T_{h.i} - T_{h.o}) \tag{7}$$

$$q_c = \dot{m}_c c_{p.w} (T_{c.o} - T_{c.i}) \tag{8}$$

The difference in the rate of heat transfer between cold and hot water in empirical results was between 3% and 6%. As a consequence, the average cold and hot water heat transfer rates were used in the other calculations to lessen the impact on the results of the final results of the heat transfer rate difference.

$$q_{ave} = \frac{q_c + q_h}{2} = UA_i \Delta T_{LMTD} \tag{9}$$

where the term ΔT_{LMTD} is calculated as following:

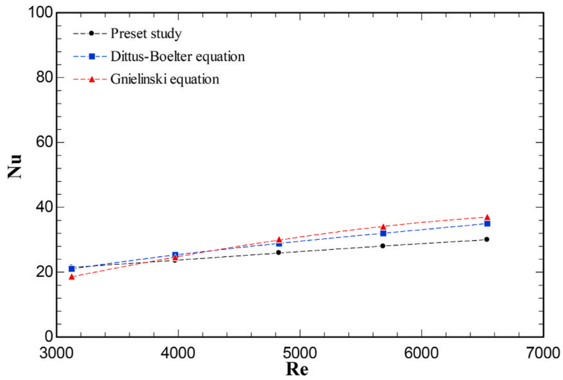

Fig. 2. Validation of Nusselt number.

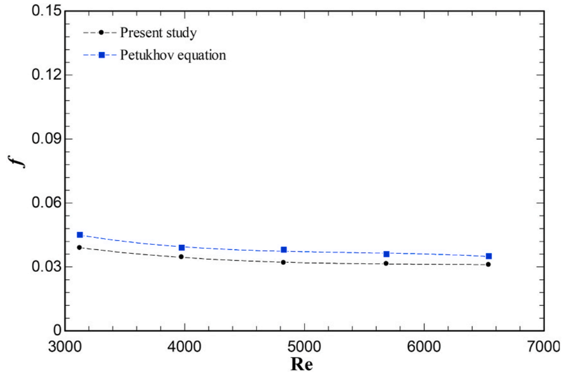

Fig. 3. Validation of friction coefficient.

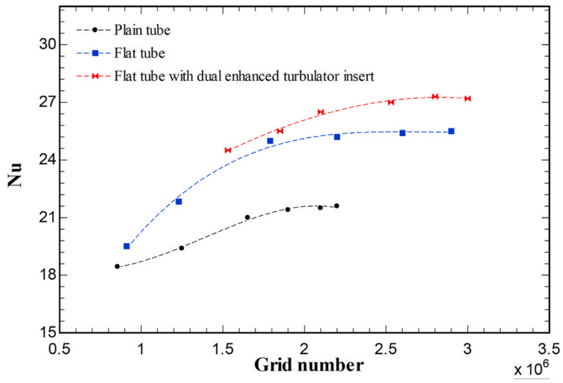

Fig. 4. Variation of Nusselt number with the mesh numbers.

$$\Delta T_{LMTD} = \frac{(T_{h.i} - T_{c.o}) - (T_{h.o} - T_{c.i})}{Ln\frac{(T_{h.i} - T_{c.o})}{(T_{h.o} - T_{c.i})}}, A_i = \pi D_i L, \tag{10}$$

The overall heat transfer coefficients could be measured based on the following equation.

$$1 / U = 1/h_i + A_i \ \ln(D_O - D_i) / 2\pi kL + R_f \tag{11}$$

Through equation (11), the last three-term of the right side was assumed to be a constant term of C1. Consequently, equation (11) could be summarized as follows.

$$1 / U = 1/h_i + C_1 \tag{12}$$

The heat transfer coefficient was discovered to be influenced by the Re number. As a general association between the heat transfer factor and the Re number, the following equation could be assumed [29,30].

$$h_i = CRe^m \tag{13}$$

Through equation (13) the term C is a constant value and the term m stands as the power index.

By Substituting Eq. (13) within Eq. (11), equation (14) could be easily achieved.

$$\frac{1}{U} = \frac{1}{B}Re^{-m} + C_1 \tag{14}$$

Equation (14) denotes that the curve of 1/U in terms of Re^{-m} is a straight line that has a slope of 1/B. reorganizing equation (14) results in equation (15).

$$h_i = 1 \Big/ \left(\frac{1}{U} - C_1\right) \tag{15}$$

As a result, equation (16) is used to compute the Nusselt number.

$$\mathrm{Nu} = h_i D_H / k$$

Where:

$$D_H = 4^*(tube\ area)/(\text{tubr}\ \ perimeter) \tag{16}$$

The quantity k is the heat transfer coefficient of the working fluid in Equation (16), which was computed at the working fluid's mean bulk temperature. Furthermore, at the test section inlet, the Re is calculated by the flow rate of the test section.

$$\text{Re} = \rho \overline{V} D_H \Big/ \mu \tag{17}$$

Where μ is the working fluid's dynamic viscosity.

Also, the friction factor (f) is determined using the following formula:

$$f = 2D_h \Delta P \Big/ \left(L\rho \overline{V}^2 \right) \tag{18}$$

3.2. Therman performance enhancement factor (TEF)

The employment of passive techniques such as turbulators may typically improve the heat transfer rate of the heat exchanger, based on the findings. The pressure drop and the friction factor will increase, too. To simultaneously check the presence of conical turbulators during heat transfer and pressure drop, and to select the most economic case from the point of view of the first law of thermodynamics, the thermal performance enhancement factor (TEF), which can be calculated as follows was used [12,31]:

$$\text{TEF} = \left(Nu / \text{Nu}_{\text{flat}\ \ \text{tub}} \right) / \left(f / \text{f}_{\text{flat}\ \ \text{tube}} \right) \tag{19}$$

3.3. Exergy analysis and exergo-economic performance

The heat exchangers are one of the main parts of many industrial processes as like power generation systems or energy storage systems. On the other hand, when it comes to the optimization of a thermal system, the optimization method according to the second law of thermodynamics has gained lots of attention. Most of these concepts or theories are developed based on the exergy concept. Since any part of the thermal system has its impact on the total exergy analysis of the whole system, the investigation of the exergetic behavior of each part of system has very significant importance through the total investigation of the whole system. The maximum theoretical obtainable work at the end of a reversible process is defined as exergy. Exergy losses in a heat exchanger are due to finite temperature differences between hot and cold water ($E_{\Delta T}$) and fluid friction ($E_{\Delta P}$). After gathering the numerical data such as 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. [32–34];

$$E = E_{\Delta T} + E_{\Delta P} = T_0 \left(S_{gen}^{\dot{\Delta T}} + S_{gen}^{\dot{\Delta p}} \right). \tag{9a}$$

which $\dot{S}$ is entropy generation rate. The entropy generation for incompressible liquids may be described in terms of particular temperatures under constant pressure as follows:

$$S_{gen}^{\dot{\Delta T}} = \left(S_{gen}^{\dot{\Delta T}} \right)_{cold\ water} + \left(S_{gen}^{\dot{\Delta T}} \right)_{hot\ water} = \left(\dot{m}_h c_{p.h} ln \frac{T_{h.o}}{T_{h.i}} \right) + \left(\dot{m}_c c_{p.c} ln \frac{T_{c.o}}{T_{c.i}} \right) \tag{10a}$$

$$S_{gen}^{\dot{\Delta p}} = \left(S_{gen}^{\dot{\Delta p}} \right)_{cold\ water} + \left(S_{gen}^{\dot{\Delta p}} \right)_{hot\ water} = \left(\dot{m}_h \frac{\Delta P_h}{\rho_h} \frac{ln \frac{T_{h.o}}{T_{h.i}}}{T_{h.o} - T_{h.i}} \right) + \left(\dot{m}_h \frac{\Delta P_h}{\rho_h} \frac{ln \frac{T_{h.o}}{T_{h.i}}}{T_{h.o} - T_{h.i}} \right) \tag{11a}$$

Eqs. (10) and (11) are inserted into Eq. (9) to give:

$$E = \text{T}_e \left[\left(\dot{m}_h c_{p.h} ln \frac{T_{h.o}}{T_{h.i}} \right) + \left(\dot{m}_c c_{p.c} ln \frac{T_{c.o}}{T_{c.i}} \right) + \left(\dot{m}_h \frac{\Delta P_h}{\rho_h} \frac{ln \frac{T_{h.o}}{T_{h.i}}}{T_{h.o} - T_{h.i}} \right) + \left(\dot{m}_h \frac{\Delta P_h}{\rho_h} \frac{ln \frac{T_{h.o}}{T_{h.i}}}{T_{h.o} - T_{h.i}} \right) \right] \tag{12a}$$

In addition to evaluating the heat transfer, pressure drop, exergy loss and thermal performance enhancement factor (TEF), the economic study of these methods from the point of view of the second law of thermodynamics is of great importance. However, there are very few studies that have considered this concept. For this purpose, the concepts of net profit per unit transferred heat load (η_p) is used in this study to evaluate the exergo-economic behavior of a double tube heat exchanger equipped with simple and enhanced conical turbulator. The concept of aforementioned parameters of this method was comprehensive explained in the work of Wu et al. [32] study. In this method, the net profit of unit transferred heat load (η_p) can be calculated using following equation:

$$\eta_p = \frac{NPV}{q_{ave}} = \frac{NPV}{\tau \dot{m}_h c_{p.w} \Delta T} \tag{13a}$$

n equation (13), net profit value (NPV) can be calculated as following:

$$NPV = \tau\left(\varepsilon_q E_{\Delta T} - \varepsilon_p E_{\Delta P}\right) - [I_0 - SV(P / F.i.n)](A / P.i.n) \tag{14a}$$

In equation (14), I_0 is the initial investment of the double tube heat exchanger which is proportional to length of tubes and turbulator cost ($I_0 = \lambda_l \times L$). λ_l, is the cost of unit length of heat exchanger and turbulator. The "n" is the life of heat exchanger and turbulator which considered to be equal for all cases. "SV" is the salvage value and was considered to be Zero in this study. The $(P/F.i.n)$ and $(A/P.i.n)$ denote the worth factor and capital recovery factor [43,44], respectively. The τ is the annual operating time.

4. Result and discussion

Flat tube has gained the attention of many researchers. Safikhani et al. [17] compared heat transfer and TEF factor numerically between typical tube and flat tube. The results revealed that the rate of heat transfer in the flat tube is higher than the typical tube in the same conditions. In addition, the TEF factor of the flat tube was greater than that of typical one. Thus, in the present study, the flat tube was used as the central tube of the double tube heat exchanger, then, a dual conical turbulator with a height of 7.2 mm was placed inside the central tube. For a plain tube, flat tube, and flat tube exchanger fitted with a dual conical turbulator, the changes of the Nusselt number vs. the hot water flow rate are illustrated in Fig. 5 A. In the flow inside the tube, the maximum velocity occurs in the center of the tube. In flat tubes, the tube wall is close to the center of the tube and as a result the fluid velocity near the wall is more than a plain tube. This causes the heat transfer coefficient in the flat tube to be higher than the plain tube heat exchanger and as a result the heat transfer rate is higher. Also, the use of conical turbulators increased the flow turbulence. This turbulator guides the fluid toward the wall of the central tube, where heat transfer occurs, and enhances the heat transfer coefficient by creating radial currents. On the other hand, the presence of conical turbulators inside the central tube occupies a certain amount of space inside the tube, hence, decreasing the useful hydraulic diameter inside the central tube. As a consequence, at a given flow rate, the fluid velocity in the presence of the conical turbulator will be higher than in the absence of the turbulator. As a result, the convective heat transfer coefficient and, as a result, the heat transfer rate will rise. As can be seen, a heat exchanger with a flat tube has 1.17 times the heat transfers of a simple heat exchanger. A heat exchanger with a dual conical turbulator also had a Nusselt number that was 1.18–1.33 times that of a simple heat exchanger. When a dual conical turbulator is present, the pressure drop in the heat exchanger rises. The pressure drop of the flat tube heat exchanger with dual conical turbulator inserts was 231–251% greater than that of the simple flat heat exchanger, according to the results (Fig. 6A).

The shape of the turbulator was then changed to a more aerodynamic one to decrease the pressure drop. For this reason, a hemisphere was added to the ends of the cones. Modified dual conical turbulators with two divergent (case A) and convergent (case B) embellishment were inserted into the central tube and the Nusselt number and pressure drop were assessed. Fig. 5 B shows the variation of *Nu* number with hot water flow rate for a plain tube heat exchanger, flat tube heat exchanger, and flat tube heat exchanger equipped with dual enhanced conical turbulator (case A and case B). based on the findings, the heat transfer rate in a heat exchanger with a modified dual turbulator was somewhat lower than the heat transfer rate in a heat exchanger with a simple dual turbulator. However, the aerodynamics of the turbulator geometry significantly reduced pressure drop. In Fig. 6 B, the friction factor is shown against the hot water flow rate for various modes. The friction factor in the presence of a dual conical turbulator is 20–27% higher in instance case A and 27–48% higher in case B, according to the results. Also, for a more complete view of the results, temperature contours and strain lines for different cases are shown in Figs. 7 and 8. As can be seen from these two images, improving the geometry of the turbulators has reduced the separation of current from the surface of the turbulator. Also, the flow turbulence in the presence of the improved turbulator is less than that of the simple turbulator. As a result, using an improved turbulator instead of a simple turbulator, in addition to reducing the pressure drop, also reduces the outlet temperature of the heat exchanger.

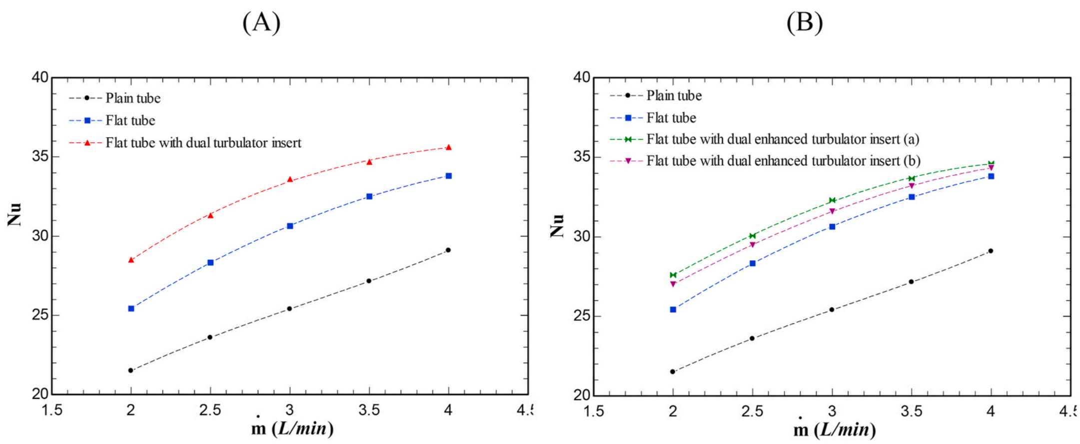

Fig. 5. Variation of *Nu* number with hot water flow rate: (A) For a heat exchanger with flat tube with/without dual conical turbulator; (B) For a heat exchanger with flat tube equipped with modified dual conical turbulator with two divergent (case A) and convergent (case B) embellishment.

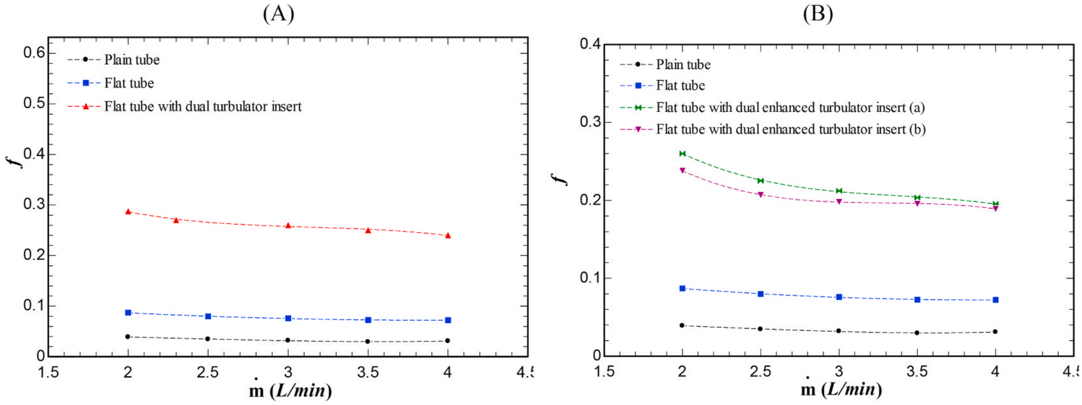

Fig. 6. Variation of friction coefficient with hot water flow rate: (A) for a heat exchanger with flat tube with/without dual conical turbulator; (B) for a heat exchanger with flat tube equipped with modified dual conical turbulator with two divergent (case A) and convergent (case B) embellishment.

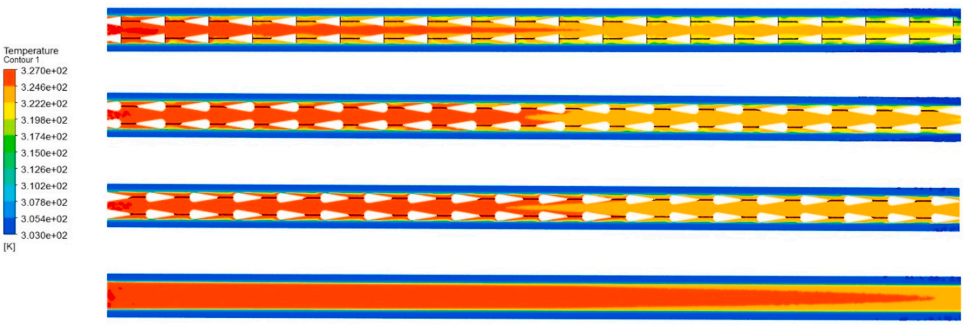

Fig. 7. Temperature contours for different cases.

Fig. 9 shows the exergy loss for a flat tube heat exchanger with and without a dual conical turbulator as a function of hot water flow rates. The heat transfer created by the temperature difference like what happens in a double tube heat exchanger is an irreversible process. The higher heat transfer means higher irreversibility, and subsequently, higher exergy losses. As previously stated, the purpose of using a conical turbulator inside the central tube is to increase the rate of heat transfer and, as a result, increases exergy loss. In the flat tube heat exchanger with dual conical turbulator, modified dual (case A and B), exergy loss increased 17–30%, 15.8–27.3%, and 14.8–25.7% compared to the plain tube heat exchanger, respectively.

The thermal performance enhancement factor (TEF) and the net profit of unit transferred heat load (η_p) were used to the economic study of these methods from the point of view of the first and second law of thermodynamics, respectively. Variation of TEF and η_p with hot water flow rate for the simple and modified conical turbulator is presented in Fig. 10 and Fig. 11, respectively. By considering the results, it is observed that the heat exchanger with a simple dual conical turbulator has the highest heat transfer and maximum pressure drop. Also, a heat exchanger with a modified dual enhanced conical turbulator (case B) has the highest TEF and η_p. In fact, both the first law and the second law of thermodynamics indicate that a heat exchanger with a modified dual enhanced conical turbulator is a more economical case.

5. Conclusions

For increasing the heat transfer of a twin-tube heat exchanger, plain and modified dual conical turbulators were used in this study. Their impacts on the pressure drop and exergy loss were also evaluated in addition to the heat transfer. To select the most economically justified mode, the thermal performance enhancement factor (TEF) and the net profit of unit transferred heat load (η_p) were considered. The results indicated that the use of conical turbulators simultaneous with enhancing the heat transfer can increase the pressure drop as well. The highest heat transfers and exergy loss was observed in the presence of the plain dual turbulator while the system encompassing the modified turbulator (case B) offered the lowest pressure drop. *Nu* number and exergy loss of the flat tube heat exchanger with a dual modified conical turbulator (case B) was up to 33% and 30% higher than the plain tube heat exchanger, respectively. The most economic mode in terms of the TEF and $\eta_p$factor was the flat tube heat exchanger with dual modified conical turbulator (case B). It was revealed that the application of enhanced conical turbulator would lead in the increment of η_p up to 1.26 times. In sum, it can be expressed that in the cases where the goal is the maximal heat transfer regardless of the economic aspects, the

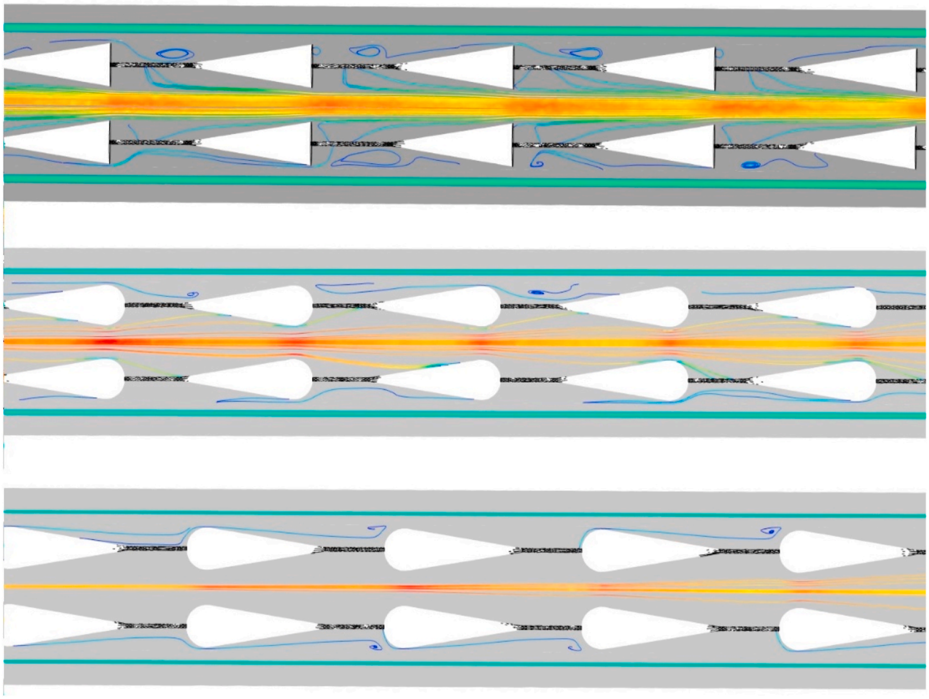

Fig. 8. Streamlines for different cases.

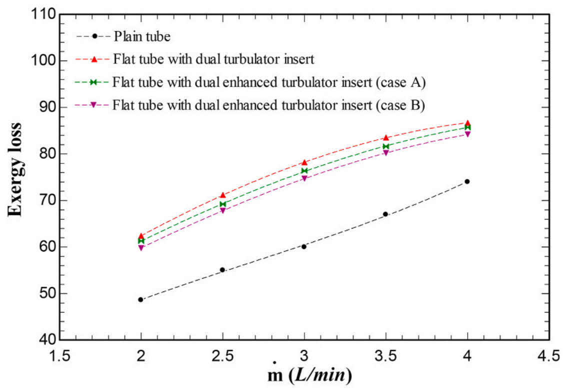

Fig. 9. Variation of exergy loss with hot water flow rate for a heat exchanger with flat tube equipped with modified dual conical turbulator with two divergent (case A) and convergent (case B) embellishment.

flat exchanger with a plain conical turbulator is recommended. When cost is a factor, a flat tube heat exchanger with a modified dual turbulator (case B) is recommended.

CRediT authorship contribution statement

Heng Chen: Resources, Software preparation, Conceptualism, Formal analysis, Investigation. ***Hamdi AYED:*** Writing-review & editing, Resources, Investigation. ***Riadh Marzouki:*** Writing-review & editing, Methodology, Formal analysis, ***Faezeh Emami:*** Writing-original paper & editing, Conceptualization. ***Ibrahim Mahariq***: Resources, Formal analysis. Methodology, Formal analysis. **Fahd Jarad:** Writing-review, Mesh generation, Post processing, Software, Resources.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

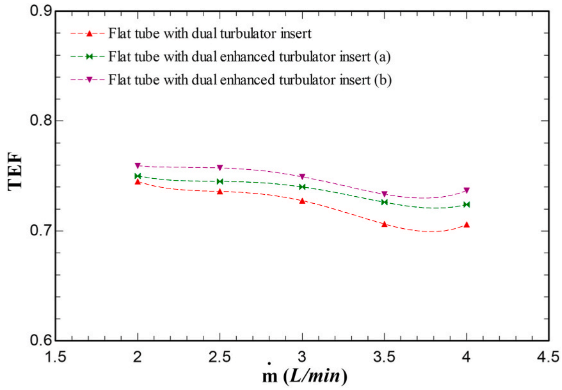

Fig. 10. Variation of TEF with hot water flow rate for different modes.

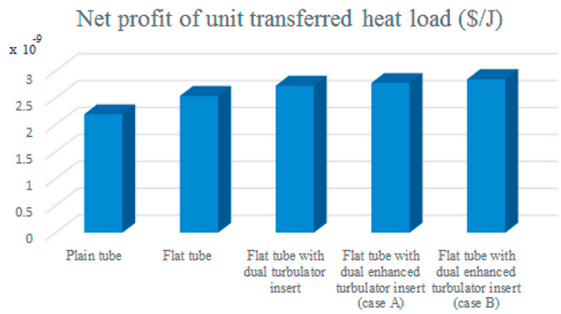

Fig. 11. Variation of η_p with hot water flow rate for different modes.

Acknowledgement

The authors extend their appreciation to the Deanship of Scientific Research at King Khalid University for funding this work through research groups under grant number RGP.1/260/42.

References

[1] S.S. Murshed, M.M. Lopes, Heat Exchangers: Advanced Features and Applications, BoD–Books on Demand, 2017.
[2] M. Chakole, Heat Transfer Enhancement Techniques. With Special Attention to Passive Methods of Heat Transfer Enhancement, Anchor Academic Publishing, 2016.
[3] A. Maleki, A. Haghighi, I. Mahariq, Machine learning-based approaches for modeling thermophysical properties of hybrid nanofluids: a comprehensive review, J. Mol. Liq. (2020) 114843.
[4] R. Daneshfar, A. Bemani, M. Hadipoor, M. Sharifpur, H.M. Ali, I. Mahariq, et al., Estimating the heat capacity of non-Newtonian ionanofluid systems using ANN, ANFIS, and SGB tree algorithms, Appl. Sci. 10 (2020) 6432.
[5] Z. Ayub, S. Al-Fahed, The effect of gap width between horizontal tube and twisted tape on the pressure drop in turbulent water flow, Int. J. Heat Fluid Flow 14 (1993) 64–67.
[6] F.R. Menter, Two-equation eddy-viscosity turbulence models for engineering applications, AIAA J. 32 (1994) 1598–1605.
[7] S. Eiamsa-Ard, P. Promvonge, Thermal characteristics in round tube fitted with serrated twisted tape, Appl. Therm. Eng. 30 (2010) 1673–1682.
[8] P. Samruaisin, S. Kunlabud, K. Kunnarak, V. Chuwattanakul, S. Eiamsa-ard, Intensification of convective heat transfer and heat exchanger performance by the combined influence of a twisted tube and twisted tape, Case Studies in Thermal Engineering 14 (2019) 100489.
[9] N. Mashoofi, S. Pourahmad, S. Pesteei, Study the effect of axially perforated twisted tapes on the thermal performance enhancement factor of a double tube heat exchanger, Case studies in thermal engineering 10 (2017) 161–168.
[10] S. Eiamsa-Ard, P. Promvonge, Performance assessment in a heat exchanger tube with alternate clockwise and counter-clockwise twisted-tape inserts, Int. J. Heat Mass Tran. 53 (2010) 1364–1372.
[11] R. Thejaraju, K. Girisha, S. Manjunath, B. Dayananda, Experimental investigation of turbulent flow behavior in an air to air double pipe heat exchanger using novel para winglet tape, Case Studies in Thermal Engineering 22 (2020) 100791.
[12] S. Pourahmad, S. Pesteei, Effectiveness-NTU analyses in a double tube heat exchanger equipped with wavy strip considering various angles, Energy Convers. Manag. 123 (2016) 462–469.
[13] S. Pourahmad, S.M. Pesteei, M. Mehrabi, 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, Exp. Heat Tran. 32 (2019) 393–409.
[14] S. Pourhedayat, S. Khorasani, H.S. Dizaji, A comprehensive analysis and empirical correlations for Nusselt number, friction factor and exergy destruction of helical tube equipped with spring-wire, Int. J. Therm. Sci. 145 (2019) 106050.
[15] F. Fan, C. Qi, J. Tang, Q. Liu, X. Wang, Y. Yan, A novel thermal efficiency analysis on the thermo-hydraulic performance of nanofluids in an improved heat exchange system under adjustable magnetic field, Appl. Therm. Eng. 179 (2020) 115688.
[16] M. Nakhchi, M. Hatami, M. Rahmati, Experimental investigation of performance improvement of double-pipe heat exchangers with novel perforated elliptic turbulators, Int. J. Therm. Sci. 168 (2021) 107057.

[17] H. Safikhani, F. Abbasi, Numerical study of nanofluid flow in flat tubes fitted with multiple twisted tapes, Adv. Powder Technol. 26 (2015) 1609–1617.
[18] V. Kongkaitpaiboon, K. Nanan, S. Eiamsa-Ard, Experimental investigation of heat transfer and turbulent flow friction in a tube fitted with perforated conical-rings, Int. Commun. Heat Mass Tran. 37 (2010) 560–567.
[19] H. Karakaya, A. Durmuş, Heat transfer and exergy loss in conical spring turbulators, Int. J. Heat Mass Tran. 60 (2013) 756–762.
[20] P. Promvonge, S. Eiamsa-ard, Heat transfer and turbulent flow friction in a circular tube fitted with conical-nozzle turbulators, Int. Commun. Heat Mass Tran. 34 (2007) 72–82.
[21] A. Durmuş, Heat transfer and exergy loss in cut out conical turbulators, Energy Convers. Manag. 45 (2004) 785–796.
[22] H. Ayed, H. Moria, F. Aldawi, N. Farouk, K. Sharma, M.M. Youshanlouei, et al., Thermal, efficiency and power output evaluation of pyramid, hexagonal and conical forms as solar panel, Case Studies in Thermal Engineering 27 (2021) 101232.
[23] P. Promvonge, S. Eiamsa-Ard, Heat transfer behaviors in a tube with combined conical-ring and twisted-tape insert, Int. Commun. Heat Mass Tran. 34 (2007) 849–859.
[24] H. Mohammed, H.A. Hasan, M. Wahid, Heat transfer enhancement of nanofluids in a double pipe heat exchanger with louvered strip inserts, Int. Commun. Heat Mass Tran. 40 (2013) 36–46.
[25] S. Pesteei, N. Mashoofi, S. Pourahmad, A. Roshan, Numerical investigation on the effect of a modified corrugated double tube heat exchanger on heat transfer enhancement and exergy losses, International Journal of Heat and Technology 35 (2017) 243–248.
[26] M. Bahiraei, R. Rahmani, A. Yaghoobi, E. Khodabandeh, R. Mashayekhi, M. Amani, Recent research contributions concerning use of nanofluids in heat exchangers: a critical review, Appl. Therm. Eng. 133 (2018) 137–159.
[27] F. Incropera, D. DeWitt, in: Fundamentals of Heat and Mass Transfer, fifth ed., Wiley, New York, 1996.
[28] B. Petukhov. Advances in Heat Transfer, Volume 6, Chapter Heat Transfer and Friction in Turbulent Pipe Flow with Variable Physical Properties. Academic Press, Inc., New York, NY1970.
[29] P. Promvonge, S. Pethkool, M. Pimsarn, C. Thianpong, Heat transfer augmentation in a helical-ribbed tube with double twisted tape inserts, Int. Commun. Heat Mass Tran. 39 (2012) 953–959.
[30] K. Wongcharee, S. Eiamsa-ard, Heat transfer enhancement by using CuO/water nanofluid in corrugated tube equipped with twisted tape, Int. Commun. Heat Mass Tran. 39 (2012) 251–257.
[31] J. Guo, Y. Yan, W. Liu, F. Jiang, A. Fan, Effects of upwind area of tube inserts on heat transfer and flow resistance characteristics of turbulent flow, Exp. Therm. Fluid Sci. 48 (2013) 147–155.
[32] 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, Appl. Therm. Eng. 111 (2017) 676–683.
[33] S.D. Pandey, V. Nema, An experimental investigation of exergy loss reduction in corrugated plate heat exchanger, Energy 36 (2011) 2997–3001.
[34] A. Akbari, S. Kouravand, S. Pourahmad, A comprehensive study of the second law of thermodynamics for a rotary regenerative heat exchanger with two types of simple and anodized aluminum sheets: an experimental study, in: Energy Sources, Part A: Recovery, Utilization, and Environmental Effects, 2020, pp. 1–16.