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Research Note

Numerical study of using perforated conical turbulators and added nanoparticles to enhance heat transfer performance in heat exchangers

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Abstract. The current study investigates a two-phase turbulent nanofluid flow inside a heat exchanger tube equipped with a novel type of conical turbulators having two parallel rows of holes for the first time. The effects of the number of the conical insert turbulators, number of the holes and volume fraction of the nanoparticles on flow field, average Nusselt number, friction factor, and performance evaluation criterion have been numerically investigated. The results show that the proposed turbulators create vortices and recirculating currents that have significant effect on heat transfer. As the number of the turbulators increases, Nusselt number increases obviously. However, the presence of holes reduces the friction factor and pressure drop that is related to lower resistance in the flow path. In general, the use of perforated conical turbulators improves Performance Evaluation Criterion (PEC) by creating controlled turbulent flows. On the other hand, the use of added nanoparticles also enhances heat transfer. The presented turbulators increase PEC by 43% compared to the smooth tube, if the parameters are determined properly. The maximum PEC of 1.43 is obtained at $M = 8$, $N = 4$, and $Re = 4100$, showing good performance compared to other types of turbulators.

1. Introduction

Due to the increasing energy supply costs, efficient use and prevention of energy waste by replacing old systems with newer technologies, eliminating equipment defects, increasing equipment efficiency by making structural changes [1], or using new technologies [2] have become extremely important [3]. For this purpose, several methods such as nanoparticles [4], cavity [5], porous medium [6], and spherical particles [7] have been proposed to enhance heat transfer. Among these, passive methods have favorable position among techniques of thermal performance enhancement due to the simplicity of implementation and no need for advanced control equipment [8,9]. The use of equipment such as turbulators [10], perforated tube inserts [11], and nanofluids [12] that enhance heat transfer while maintaining performance can reduce the size of heat exchangers [13]. Decreasing equipment size is necessary to minimize costs and safety concerns related to the total fluid volume of the system [14].

Nanofluid is formed by suspending nanoparticles in a pure liquid [15]. Since nanoparticles are extremely fine and have a large specific surface area

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[16–22], they have unique properties such as high thermal conductivity when dispersed in liquids. Such fluids are very different from conventional solid-liquid suspensions in both preparation and properties. By adding a small number of nanoparticles such as copper [23], gold [24], copper oxide, alumina, graphene oxide [25–27], or carbon nanotubes [28] to the fluid, the thermal conductivity of the fluid is increased [29]. Researchers have studied many different aspects of nanofluids including their thermal conductivity, which is unusually high even at low nanoparticle concentrations. One of the first studies in this field was conducted by Pak and Cho [30]. In an empirical study, they investigated heat transfer from AL_2O_3-Water and TiO_2-Water nanofluids in both laminar and turbulent flow regimes and found that the heat transfer coefficient of nanofluid nanoparticles with a volume fraction of 3% was up to 12% higher than water-based fluid at some Reynolds numbers. Javaherdeh et al. [31] numerically studied the thermal and hydrodynamic behaviors of turbulent flow of non-Newtonian nanofluids in a helical double-tube heat exchanger. In their study, aluminum oxide nanoparticles were used. They investigated the effect of Reynolds number and volume fraction of aluminum oxide nanoparticles on heat transfer enhancement. Sharifi Asl et al. [32] used computational fluid dynamics method to simulate heat transfer in non-Newtonian turbulent flow of nanofluid in a horizontal tube. The results of their study illustrate that heat transfer coefficient and Nusselt number are higher in non-Newtonian nanofluid than those in the base non-Newtonian fluid. Heat transfer coefficient and Nusselt number also increase with increasing nanoparticle volume fraction and Reynolds number. Radwan et al. [33] investigated turbulent flow heat transfer and pressure drop of aluminum oxide-water nanofluid empirically. They found that pressure drop for the nanofluid was higher than that for the base fluid. They also observed 25% enhancement in heat transfer coefficient. Thermal behavior of silicon dioxide-Water nanofluid in the Reynolds range of 5000–27000 under constant flux and with inserted equipment inside tube was investigated by Azmi et al. [34]. The results of their study indicate increase in Nusselt number and heat transfer enhancement. Heat transfer and thermal efficiency of heat exchangers with novel turbulators were experimentally investigated by Nakhchi et al. [35]. Kongkaitpaiboon et al. [36] considered the impact of perforated conical rings on the fluid flow in tubes, where a rather significant enhancement in the heat transfer was observed. Natural convection heat transfer of TiO_2 and Al_2O_3 nanofluids in a rectangular cavity while two heated fins are located in the cavity was investigated by Hatami [37]. Nalavade et al. [38] investigated the geometrical influence of perforated twisted tapes on parameters such as heat transfer and friction factor of smooth tubes. The mean heat transfer coefficient enhancement ratio at equal Reynolds number was also obtained.

Another passive method for enhancing heat transfer is the use of turbulators, which has received special attention in recent years [39–42]. One of the most common used turbulators in thermal applications is conical rings, which are among the most widely used equipment for improving heat transfer due to their easy installation, low cost, and high efficiency. This type of turbulators was first used by Yakut et al. [43]. Durmus [44] investigated heat transfer in conical rings using experimental tests. In his study, air was considered as the working fluid, and the performance of the rings was studied in the Reynolds numbers ranging from 15000 to 60000. Promvonge and Eiamsa-Ard [45] studied the heat transfer behavior in tubes with a conical ring and helical turbulators empirically. In another study, Promvonge [46] conducted experimental tests to investigate the effect of conical rings. The results of his study showed that the use of this equipment could increase Nusselt number significantly. Karakaya and Durmus [47] studied the performance of conical spring turbulators in improving heat exchangers heat transfer. Liu et al. [48] studied the characteristics of free convection heat transfer in tubes with conical rings using experimental tests and numerical simulations. Based on their study results, it can be seen that the model presented by them enhances heat transfer while also increasing the friction factor. Sheeba et al. [49] showed that the performance of these turbulators was optimal at a certain cone angle. In one of the most recent studies, Xiong et al. [50] studied the performance of conical rings in double-tube heat exchangers using three-dimensional simulations. They used the k–ε method in their numerical simulations, and showed that applying this type of turbulators could improve the heat transfer coefficient by 4.68% compared to conventional tubes. Ibrahim et al. [51] studied different arrangements of conical rings in heat exchangers and the geometric characteristics effect on thermal performance of the turbulators. Nakhchi and Esfahani [52] investigated the Cu-water turbulent nanofluid flow within a heat exchanger equipped with perforated conical rings. The highest thermal performance of 1.10 was achieved using 1.5% Cu-water nanofluid and perforated conical rings at $Re = 5000$. Mohammed et al. [53] numerically studied the turbulent convection of Al_2O_3, CuO, SiO_2, and ZnO nanofluids flow in a circular pipe prepared with conical ring inserts.

It should also be stated that heat transfer enhancement and thermal analysis are important in designing heat exchangers. Using passive heat transfer enhancement devices such as turbulators and nanoparticles can increase the thermal performance of the heat exchangers. Review of the studies indicates that the performance of tubular heat exchangers containing nanofluids equipped with perforated conical turbulators has not been studied so far using multiphase numerical models. The aim of this study is to supplement the lack of information in this field. In the present study, the effect of the number of the holes located on conical rings on enhancing heat transfer in the presence of nanoparticles is studied. For this purpose, by using a two-phase nanofluid model and computational fluid dynamics method, the performance of this type of turbulators in the Reynolds number ranging between 4000–20000 is evaluated. Independent parameters include the Reynolds number, the geometrical characteristics of the conical turbulators, and the number of holes located on them, the distance of the turbulators (by specifying their number along the tube length), and the volume fraction of TiO_2 nanoparticles. The effects of these parameters on friction factor, Nusselt number, and Performance Evaluation Criterion (PEC) are investigated.

2. Geometry and boundary conditions

The geometry of the model studied in this research is shown in Figure 1. The model includes a heat exchanger tube having length L equal to 1500 mm and inner diameter D equal to 62 mm. The heat exchanger tube is equipped with conical turbulators having two rows of holes, which has been presented for the first time in the current research. The length of the conical rings, their throat diameter, and their thickness are 62 mm, 31 mm, and 2 mm, respectively. Numerical results are extracted for different numbers of conical rings within the length of the tube ($M = 1, 2, 4, 6, 8$) with constant spacing and different numbers of holes in a perimeter row (which is $N = 0, 2, 4, 6, 8$). Water fluid was used as the base fluid and TiO_2 nanoparticles were added with different volume fractions. The TiO_2-water nanofluid enters the tube at a constant temperature ($T_{in} = 300$ K), while the tube wall temperature (T_w) is assumed constant at 350 K. At the inlet of the tube, the velocity-inlet boundary condition is applied. Also, pressure-outlet boundary condition is considered at the tube outlet.

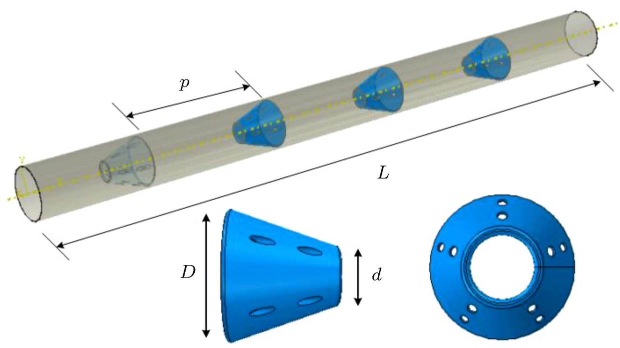

Figure 1. The heat exchanger tube having perforated conical turbulators.

3. Two-phase model

In order to increase the accuracy of modeling, attention to two-phase approaches has increased in recent years. Lotfi et al. [54] investigated Al_2O_3-water nanofluid forced convection flow. They compared Nusselt number results for several correlations of nanoparticles. The equations in the two-phase mixture model are expressed as follows:

Conservation of mass [55]:

$$\nabla . \left(\rho_m \vec{V}_m \right) = 0, \tag{1}$$

in which $\vec{V}_m$ is mean velocity and ρ_m is mass density of the mixture.

Momentum equation [55]:

$$\nabla . \left(\rho_m \overrightarrow{V_m} \overrightarrow{V_m} \right) = -\nabla P + \nabla . \left[\mu_m \left(\nabla \vec{V}_m + \nabla \vec{V}_m^T \right) \right]$$
$$+\rho_m \beta_m g \left(T - T_c \right) + \nabla . \left(\sum_{k=1}^{n} \phi_k \rho_k \vec{V}_{dr,k} \vec{V}_{dr,k} \right). \tag{2}$$

Energy equation [55,56]:

$$\nabla . \left(\sum_{k=1}^{n} \phi_{p,k} V_k \left(\rho_k E_k + P \right) \right)$$
$$= \nabla . \left(k_m \nabla T - C_p \rho_m V T \right). \tag{3}$$

Nanoparticles volume fraction equation [56]:

$$\nabla . \left(\phi_p \rho_p V_m \right) = -\nabla . \left(\phi_p \rho_p V_{dr,p} \right), \tag{4}$$

where ϕ_p, V_m, and $V_{dr,p}$ represent volume fraction, velocity, and drift velocity of the nanoparticles. The subscripts (f, p, and m) represent base fluid, nanoparticles, and the mixture, respectively. Mixture mean velocity, viscosity, density and thermal conductivity are expressed as follows [56]:

$$\left(\rho_m, \lambda_m, \mu_m \right) = \sum_{k=1}^{n} \phi_k \left(\rho_k, \lambda_k, \mu_k \right), \tag{5}$$

$$V_m = \frac{\sum_{k=1}^{n} \phi_k \rho_k V_k}{\rho_m}, \tag{6}$$

$$V_{dr,k} = V_k - V_m, \tag{7}$$

where V_m is the mean velocity. τ and τ_t are defined as follows [56]:

$$\tau = \mu_m \nabla V_m, \tag{8}$$

$$\tau_t = -\sum_{k=1}^{n} \phi_k \rho_k \overline{v_k v_k}. \quad (9)$$

Drift velocity is shown in the following relation as the secondary phase (p) velocity relative to the base phase (f) velocity provided by Nie et al. [57]:

$$V_{pf} = V_p - V_f = \frac{\rho_p d_p^2}{18\mu_f f_{drag}} \frac{(\rho_p - \rho_m)}{\rho_p}$$
$$[g - (V_m.\nabla) V_m]. \quad (10)$$

Drift velocity depends on relative velocity as follows:

$$V_{dr,p} = V_{kp} - \sum_{k=1}^{n} \frac{\phi_k \rho_k V_k}{\rho_m}. \quad (11)$$

Also, the drag function by Schiller and Naumann is expressed as [58]:

$$f_{drag} = \begin{cases} 1 + 0.15 Re_p^{0.687} & \text{for} \quad Re_p \leq 1000 \\ 0.0183 Re_p & \text{for} \quad Re_p > 1000 \end{cases} \quad (12)$$

where $Re_p = V_m d_p / v_m$ is the local Reynolds number of nanoparticles.

The Reynolds and average Nusselt number, friction factor, and PEC, which are dimensionless parameters, are as expressed as follows [58]:

$$Nu_{av} = \frac{hD_h}{k}, \quad (13)$$

where h and k are heat transfer coefficient and thermal conductivity, respectively.

$$Re = \frac{\rho u_m D_h}{\mu}, \quad (14)$$

where u_m is the fluid mean velocity. The tube equivalent hydraulic diameter (D_h) is expressed as:

$$D_h = \frac{4A}{P_h}, \quad (15)$$

where A and P_h are the cross-sectional area and wetted perimeter, respectively.

$$f = \frac{2D_h}{L} \frac{\Delta P}{\rho u_m^2}, \quad (16)$$

where ΔP is the calculated pressure drop between inlet and outlet.

PEC used in this research to determine the thermal performance of heat exchanger in simultaneous presence of both turbulator and nanofluid is calculated as follows:

$$PEC = \frac{(Nu_{av}/Nu_{av,0})}{(f/f_0)^{1/3}}, \quad (17)$$

where Nu_{av} and $Nu_{av,0}$ are the average Nusselt number of the tube in the presence of inserted equipment and the smooth tube, respectively.

4. Simulations

4.1. Model meshing and grid independency

Geometry meshing of the used model for simulations is demonstrated in Figure 2. The grid is concentrated in areas close to the tube walls and turbulators. In addition, high-density mesh has been used near the conical ring holes to capture vortex production and circulating flow. In order to ensure the grid independency, calculations were performed for TiO_2-water nanofluid having volume fraction of 2.5% at the Reynolds number 15000, and the results are given in terms of the average Nusselt number in Figure 3. The changes in Nusselt number for cells number less than 1.4×10^6 are less than 1.17%. Therefore, the use of this grid for two-phase model calculations is appropriate. Similar grids have been used in all calculations of this study.

4.2. Method of solving equations

The simulations are performed using ANSYS Fluent commercial code, and the SIMPLE algorithm is used for velocity and pressure coupling. The second-order upwind method has been selected to discretize momentum and energy equations. In Figure 4, the results from different turbulent models (standard k–ε, RNG k–ε, standard k-omega, and SST k-omega models) are compared to the results of experimental tests from reference [36]. According to the results, the RNG k–e turbulent model provides results with the smallest

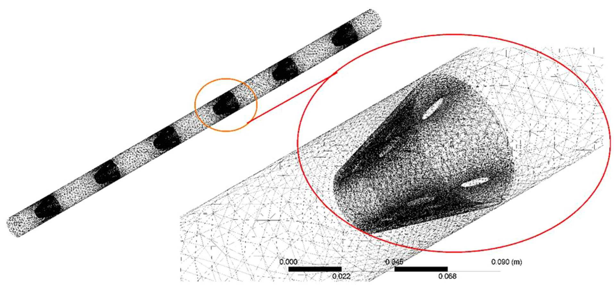

Figure 2. Geometry meshing of the used model.

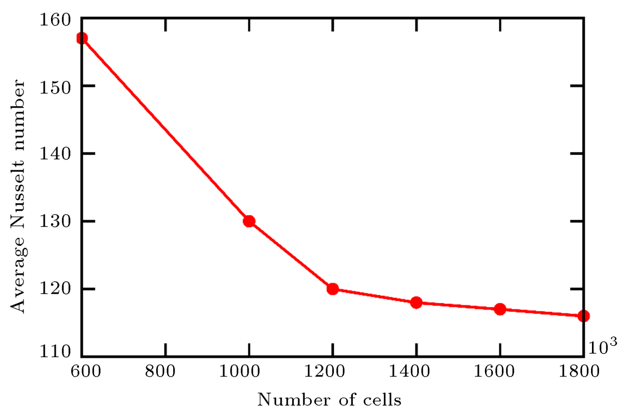

Figure 3. Grid independency investigation for TiO_2-water nanofluid with volume fraction of 2.5% at the Reynolds number 15000.

Table 1. Validation of Nusselt number and friction factor with experimental results of Ref. [32].

	Nu		***f***	
Re	**Present study**	**Ref. [26]**	**Present study**	**Ref. [26]**
4000	41.4	42.7	0.96	0.93
8000	57.1	60.0	0.82	0.74
12000	71.7	74.6	0.59	0.61

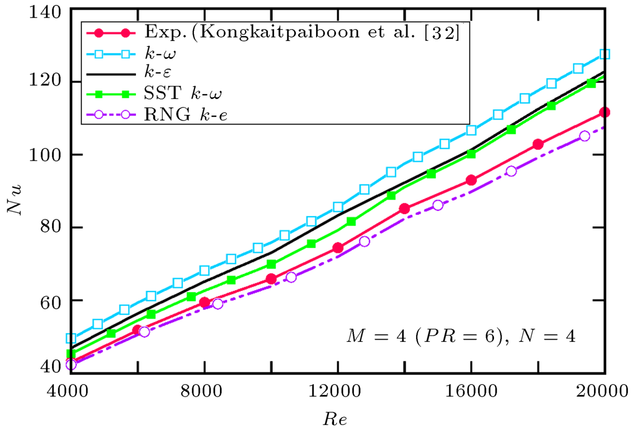

Figure 4. Comparison of the results of different turbulent models with empirical results [32] for $N = 4$.

errors compared with the experimental results. Hence, the k–ε RNG model was used in the present study. The convergence criterion for all the variables is 10^{-6}.

4.3. Validating the results

For validating the numerical model, the friction factor and the average Nusselt number for a tube having conical rings equally spaced 6 times the rings diameter, each having a perimeter row of 4 holes, are compared with the experimental results of Kongkaitpaiboon et al. [36] in Table 1. The results are extracted for the boundary conditions and physical characteristics used in the reference. Validation shows that the numerical model is in acceptable agreement with the experimental results.

5. Results and discussion

5.1. Impact of geometrical characteristics of turbulators

Here, the effect of parameters of perforated turbulators and TiO_2-water nanofluid on thermal performance is studied. First, the effect of number of the holes (N) and the number of turbulators (M) is studied. Figure 5 shows the average Nusselt number changes with the Reynolds number for different numbers of the conical rings and their holes. Also, a comparison is made between the tube equipped with these turbulators and smooth tubes. The results are extracted for TiO_2 nanoparticles with volume fraction of 0.1%. The volume fraction of nanoparticles is intentionally chosen to be very low to compare the effects of the parameters

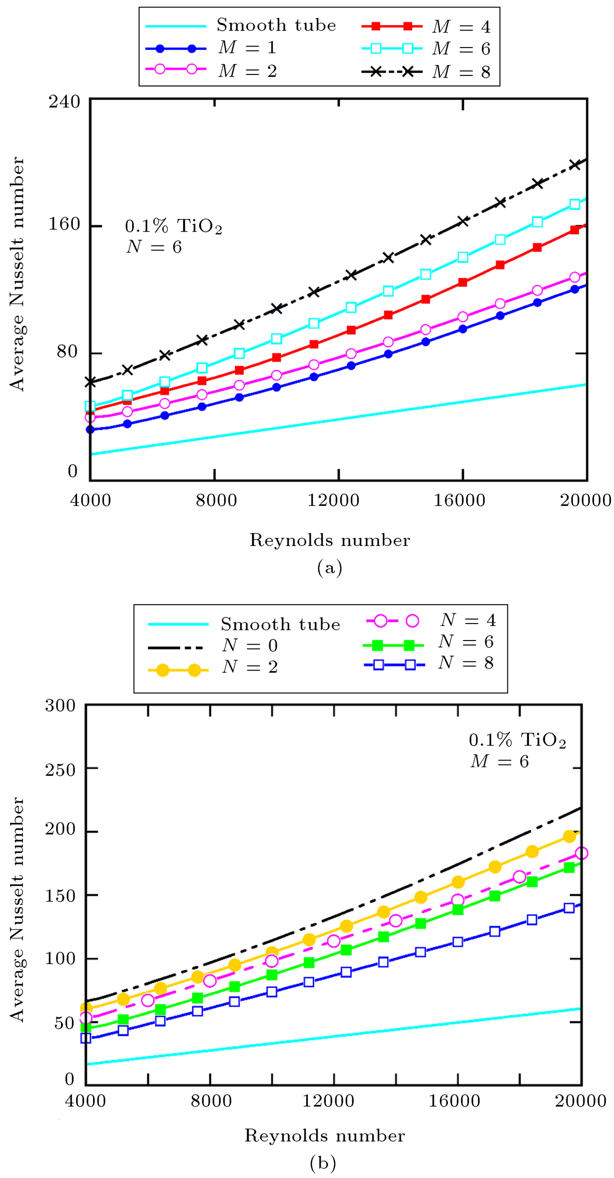

Figure 5. Nusselt number changes with the Reynolds number for: (a) different numbers of conical rings and (b) different numbers of the conical ring holes.

M and N without the interference of the nanoparticle effects. According to the results, by increasing the Reynolds number from 4000 to 20,000, Nusselt numbers in all cases under investigation increase significantly. This is mainly due to the fact that at high Reynolds, the presence of conical rings causes greater turbulence in the fluid flow. As a result, turbulence of the thermal boundary layer increases and the heat transfer is enhanced. The results show that the highest Nusselt number is obtained for $M = 8$ of intact conical rings ($N = 0$), which is about 288% higher than a smooth tube at $Re = 4000$. The results indicate that compared to the smooth tube, for $M = 1$ and $M = 6$, the Nusselt number at the Reynolds number 4000 increases by about 88% and 176%, respectively. As the number of turbulators increases, the average Nusselt number rises. On the other hand, by increasing the number of the holes, the Nusselt number decreases, which is due to lower resistance in the flow and its reduced turbulence. For example, for $M = 6$ and at Reynolds 4000, by increasing the number of the holes from zero to 8, the Nusselt number decreases by about 44%.

The effect of the number of the conical rings and holes located on them on friction factor is shown in Figure 6. Also, the results are plotted for the smooth tube. Upon increasing the number of the turbulators, which leads to a shorter distance between them, friction factor increases significantly. Given that reducing the distance increases turbulence and resistance against the fluid flow, such a result is justifiable. In addition, it is observed that the friction factor in the tube decreases upon increasing the number of the turbulator holes. At $Re = 4000$ for $N = 2$, $N = 8$, and $N = 8$, the friction factor becomes about 7.74, 10.17, and 13.8 times lower, respectively, than the case $N = 0$.

Figure 7 shows the velocity distribution at $Re = 15000$ for $M = 6$ and different numbers of the holes N. According to the results, it is observed that increasing the number of holes reduces the resistance in the flow path and as a result, Nusselt number and friction factor are reduced. In addition, as can be seen, as the number of the holes increases, the velocity of the fluid out of them decreases; as a result, Nusselt number is reduced due to the lower turbulence in the areas around the turbulator. This behavior confirms the reason for increasing the average Nusselt number by decreasing the number of holes, as shown in Figure 5. Figure 8 shows velocity streamlines in the middle of the presented conical rings with 4 and 8 holes at $Re = 15000$. As can be seen in these cases, the maximum velocities are 0.97 and 0.78 m/s, respectively. By increasing the number of the holes from 4 to 8, the maximum velocity is reduced by 20%.

A closer view of what happens in the existence of the conical ring insert is shown in Figure 9. The jet of flow is moved from the middle areas by the holes to the near wall areas, which increases the turbulence, as well as heat transfer to the colder parts of the working fluid. The formed flow vortices disrupt the boundary layer near the tube wall, which again increase heat transfer in the tube under study in the present study. The mentioned secondary flows created by the holes are demonstrated by velocity vectors, while the contour coloring indicates velocity distribution in the vicinity of the holes. The velocity curves in Figure 10 show the velocity profiles at the locations of the turbulator holes. It is observed how the first and second rows of holes redirect the primary flow to the near wall area, which enhances heat transfer from the wall to the flow.

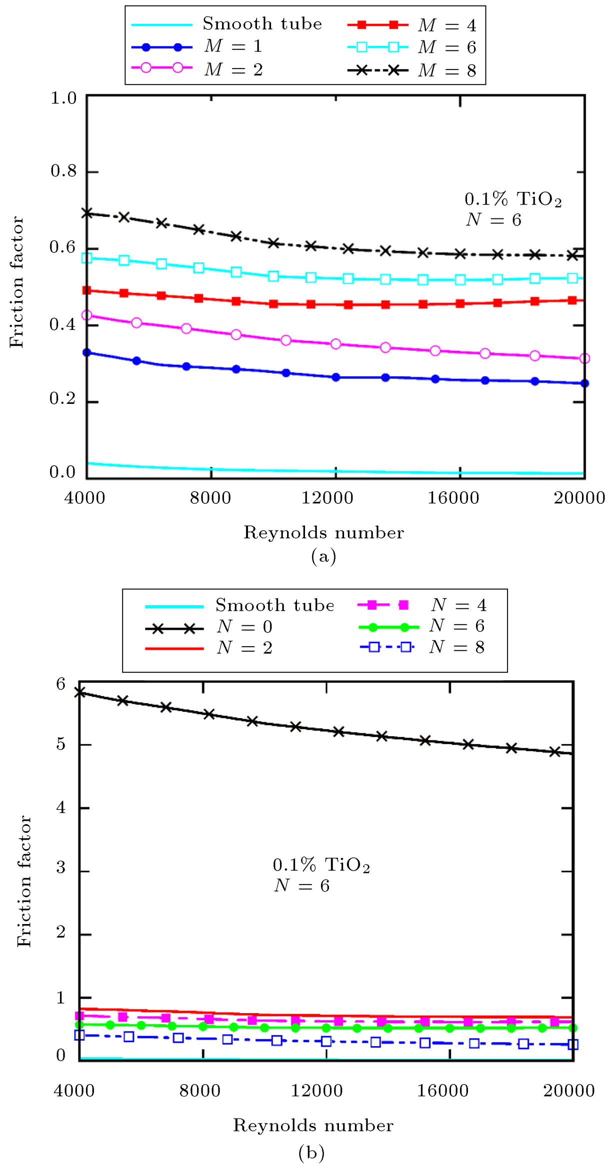

Figure 6. Friction factor variations with the Reynolds number for: (a) different numbers of conical rings and (b) different numbers of the conical ring holes.

5.2. *Effect of TiO_2 nanoparticles*

Figure 11 demonstrates the effect of the TiO_2 nanoparticles volume fraction on Nusselt number for $M = 2$ and $N = 6$. The results show that the presence of nanoparticles leads to an increase in the thermal conductivity of the working fluid and as a result, heat transfer coefficient and Nusselt number increase. By increasing the volume fraction of TiO_2 nanoparticles

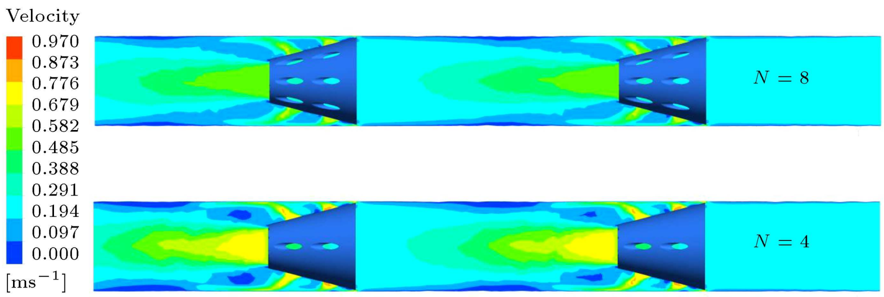

Figure 7. Velocity distribution at $Re = 15000$ for $M = 6$ and different numbers of the holes.

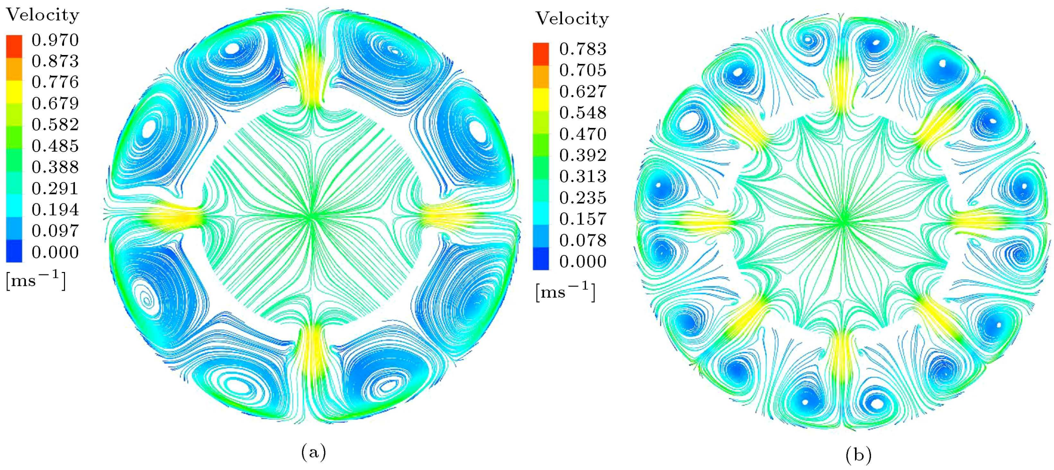

Figure 8. Velocity streamlines in the middle of the conical rings with 4 and 8 holes at $Re = 15000$.

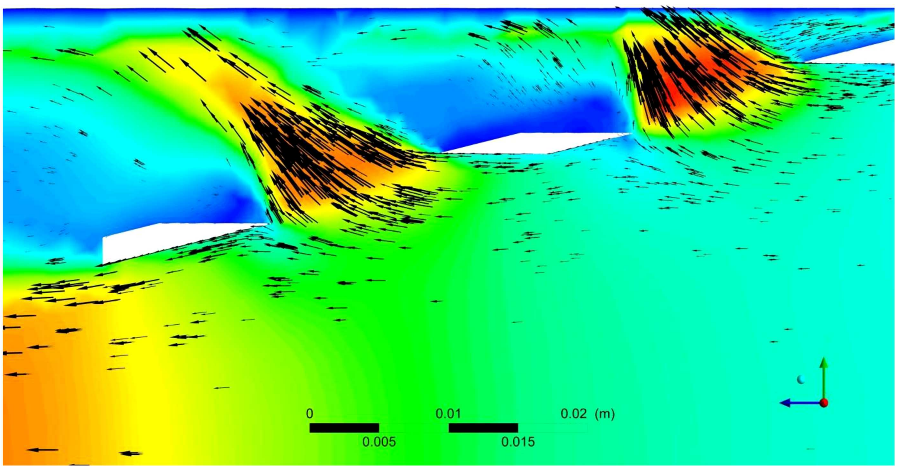

Figure 9. Velocity vectors in the vicinity of the turbulator holes at $Re = 5000$ for $N = 4$.

from 0% to 2.5%, Nusselt number at $Re = 4000$ increases by about 63%. The results show that at Reynolds numbers higher than 6000, effect of the nanoparticles on the increase of Nusselt number increases with Reynolds number changes. One of the reasons can be higher turbulence levels of the fluid flow. Also, the positive effect of Brownian motion can be considered as another reason for increasing Nusselt number. Chaotic and random motions of nanoparticles in the fluid field cause a delay in the development of the thermal boundary layer, which enhances heat transfer coefficient and Nusselt number. It is also observed that increasing the Reynolds number increases Nusselt number. For instance, by increasing the Reynolds number from 4000 to 10000 in the presence of the based fluid and the nanofluid at a volume fraction of 0.5%, the average Nusselt number increases by 58% and 38%, respectively. Comparing the results of the base fluid and the nanofluid having 0.5% volume fraction of the nanoparticles, one can observe that the turbulator

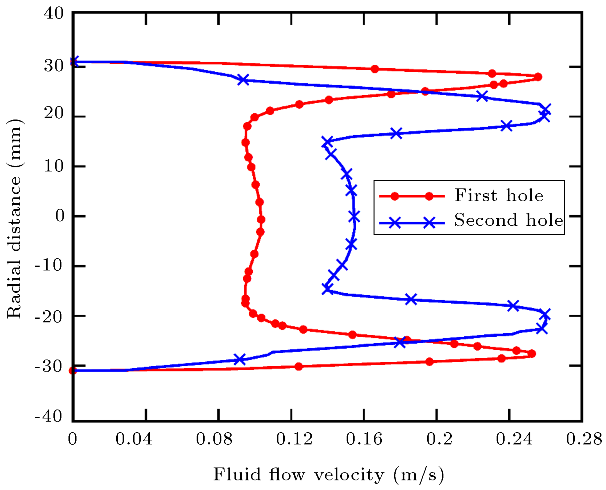

Figure 10. Velocity profiles at the locations of the turbulator holes at $N = 4$ and $Re = 5000$.

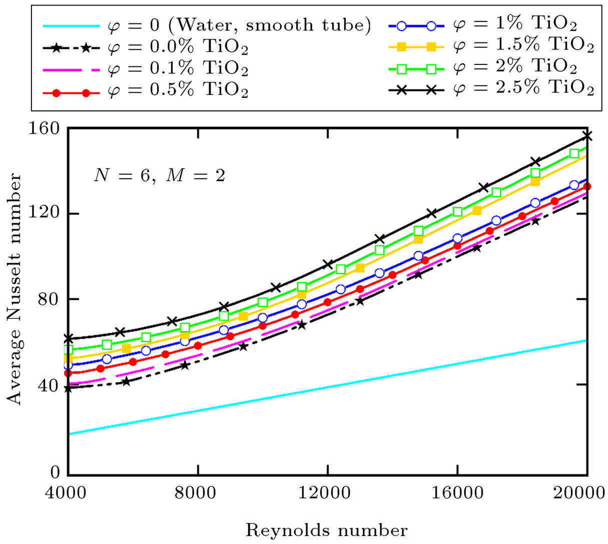

Figure 11. Effect of TiO_2 nanoparticles on Nusselt number of the heat exchanger tube.

presented in the current study has greater effect on increasing the average Nusselt number than the use of the nanoparticles without having the turbulators.

Figure 12 shows friction factor changes in terms of the Reynolds number for water and TiO_2 mixture with different volume fractions of nanoparticles for $M = 2$ and $N = 6$. By increasing the Reynolds number, the friction factor for water and nanofluid decreases. The addition of nanoparticles to the base fluid increases the viscosity and, consequently, raises the value of friction factor. At $Re = 4000$, by increasing the volume fraction of the nanoparticles to 1%, friction factor increases by 20%.

5.3. PEC

Figure 13 shows PEC of the heat exchanger tube presented in the current study for some of the studied cases. Although it was stated that the presented

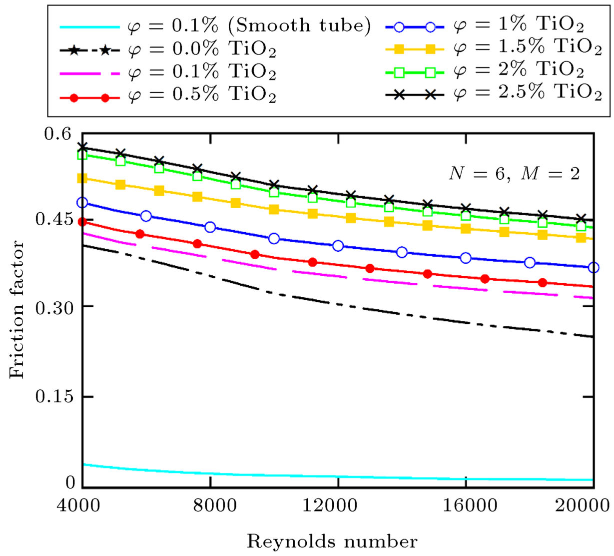

Figure 12. Effect of the TiO_2 nanoparticles on friction factor.

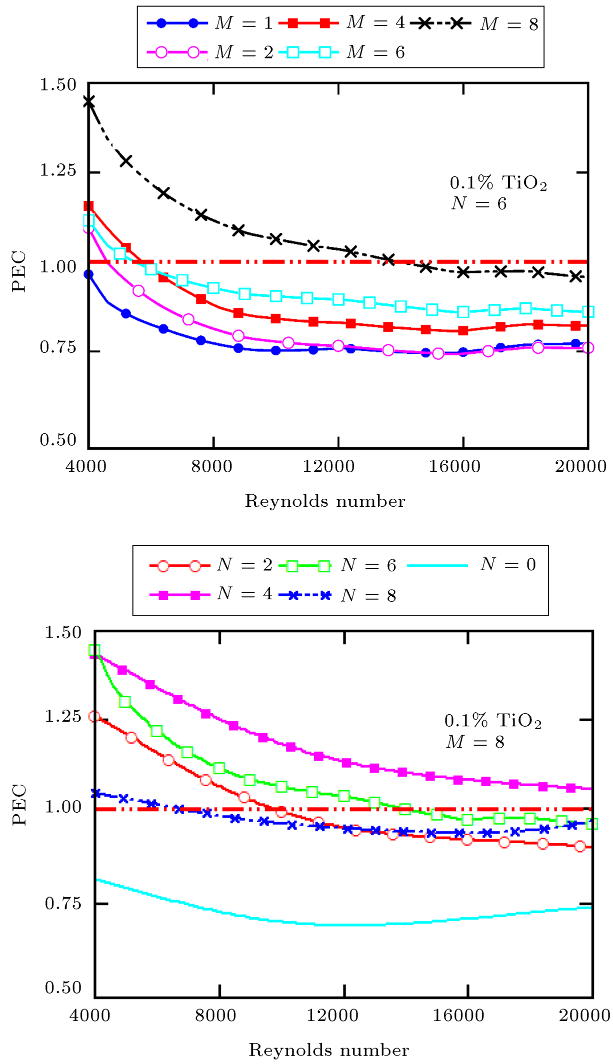

Figure 13. Performance evaluation criterion of the heat exchanger tube.

turbulator, in general, would cause a significant increase in Nusselt number, according to the results of Figure 13, it can be seen that this result is not always true for PEC. Accordingly, depending on the operating conditions of the heat exchanger, there are suitable values for geometric parameters of the turbulator, for which the thermal performance will have the maximum values. For example, at $Re < 14200$ with $M = 8$ and $N = 6$, the turbulators perform well, while at $Re > 14200$, the turbulator performs weaker and PEC is less than one. Also, if $Re > 4600$, the turbulator with characteristics $M = 1, 2, 4, 6$ and $N = 6$ shows negative effect on thermal performance. In addition, it can be seen that at $Re = 4100$, the new presented turbulator with $M = 8$ and $N = 4$ enhances the heat exchanger performance by 43%, which is significant, and is very desirable compared to other types of turbulators.

6. Conclusions

In the present study, by using numerical methods, heat transfer performance of TiO_2-water nanofluid heat exchangers tube equipped with a new type of conical ring turbulators with two rows of holes was investigated. For this purpose, the effect of the number of turbulators, number of the holes located on them, and volume fraction of the nanoparticles ($0 < \phi < 2.5\%$) on friction factor, Nusselt number and thermal performance at the Reynolds numbers 4000 to 20,000 were studied. Nanofluid simulation was performed using a two-phase model. A summary of the important results of the present study is as follows:

- Using perforated conical turbulators enhanced heat transfer by creating recirculating flows in the presence of nanofluids, which disrupt the boundary layer along the heat exchanger tube;
- Nusselt number increases by 4.05, 3.75, 3.31, and 2.25 times at $Re = 4000$ for $N = 0, 2, 4$, and 8, respectively. These types of turbulators have significant effect on increasing Nusselt number. However, increasing the number of the holes reduces the Nusselt number;
- Increasing the number of the turbulator holes reduces the friction factor. At $Re = 4000$, in comparison with the case $N = 0$, it is about 7.74, 10.17, and 13.8 times lower for $N = 2, 6$, and 8, respectively;
- The turbulators presented in this study increase thermal performance by up to 43% compared to smooth tube, if the parameters are determined properly. The maximum Performance Evaluation Criterion (PEC) of 1.43 is obtained for $M = 8$ and $N = 4$ at $Re = 4100$.

Nomenclature

L	Tube length (m)
A	Cross section area (m^2)
f_{drag}	Drag function
$V_{dr,k}$	Drift velocity of the nanoparticles (m/s)
V	Fluid velocity (m/s)
g	Gravity (m/s^2)
h	Heat transfer coefficient
D_h	Hydraulic diameter (m)
V_m	Mean velocity (m/s)
d_p	Nanoparticle diameter (nm)
M	Numbers of conical rings
ρ	Fluid density (kg/m^3)
φ	Volume fraction (%)
N	Numbers of holes
Nu	Nusselt number
PEC	Performance Evaluation Criterion
P	Pressure (pa)
ΔP	Pressure difference (Pa)
Re	Reynolds number
T	Temperature (K)
k	Thermal conductivity (W/mK)
d	Throat diameter (m)
D	Tube inner diameter (m)
P_h	Wetted perimeter
μ	Viscosity (Pa.s)
τ	Shear stress (Pa)

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