Using CFD To Assess The Impact of Helical Baffle On The First-Law And Second-Law Performance of Water-CuO Nanofluid Inside A Hairpin Heat Exchanger

Amin Shahsavar
Kermanshah University of Technology

Davood Toghraie (✉ Toghraee@iaukhsh.ac.ir)
Islamic Azad University

Pouya Bamoon
Islamic Azad University

Research Article

Keywords: Numerical simulation, Heat exchanger, Turbulator, Nanofluid, Irreversibility

DOI: https://doi.org/10.21203/rs.3.rs-877881/v1

License: This work is licensed under a Creative Commons Attribution 4.0 International License.
Read Full License
Using CFD to assess the impact of helical baffle on the first-law and second-law performance of water-CuO nanofluid inside a hairpin heat exchanger

Amin Shahsavar¹, Davood Toghraie*², Pouya Barnoon²

¹Department of Mechanical Engineering, Kermanshah University of Technology, Kermanshah, Iran
²Department of Mechanical Engineering, Khomeinishahr Branch, Islamic Azad University, Isfahan, Iran

*Corresponding author Email: Toghraee@iaukhsh.ac.ir

Abstract

This study is devoted to the numerical assessment of the influence of helical baffle on the hydrothermal aspects and irreversibility behavior of the turbulent forced convection flow of water-CuO nanofluid (NF) inside a hairpin heat exchanger. The variations of the first-law and second-law performance metrics are investigated in terms of Reynolds number (Re), volume concentration of NF (φ) and baffle pitch (B). The results showed that the NF Nusselt number grows the rise of both the Re and φ whereas it declines by boosting with the rise of baffle pitch. In addition, the outcomes depicted that the rise of both the Re and φ results in the rise of pressure drop, while it declines with the increase of baffle pitch. Moreover, it was found that the best first-law performance of the NF belongs to the case B=33.3 mm, φ=2% and Reₙf=10000. Furthermore, it was shown that irreversibilities due to fluid friction and heat transfer augment with the rise of Re while the rise of baffle pitch results in the decrease of frictional irreversibilities. Finally, the outcomes revealed that with the rise of baffle pitch, the heat transfer irreversibilities first intensifies and then diminishes.

Keywords: Numerical simulation; Heat exchanger; Turbulator; Nanofluid; Irreversibility.
1. Introduction

It is well known that the turbulent flow has a higher heat exchange and pumping power than the laminar flow, the former being desirable and the latter undesirable. So, the idea came into the researchers' minds to put equipment in the path of laminar flow and create local turbulence. The idea was very successful and has been widely used in the industry today. This equipment is called turbulator, and so far, various types of turbulators have been introduced and their performance has been studied experimentally and numerically [1-10].

Although the use of turbulators has led to an improvement in the performance of heat transfer systems, this has not prevented researchers from looking for ways to further improve the performance of these systems. One of these amazing techniques, which originated from the low thermal conductivity of heat transfer fluids, is nanofluid (NF). Choi [11] first made these modern fluids and called them NFs. After the introduction of NFs and their amazing thermal properties, much research has been done on their performance in diverse applications [12-15].

The literature inspection shows that the performance of thermal systems with NF coolants equipped with turbulators has been investigated by various researchers. Bellos et al. [16] analyzed the efficacy of oil-CuO NF in a parabolic trough collector equipped with turbulators. They found that using the combination of NF and turbulator causes a 1.54% thermal efficiency improvement. Nakhchi and Esfahani [17] inspected the efficacy of aqueous Cu NF inside a heated tube equipped with perforated conical rings in a turbulent regime. It was reported that using compound NF and turbulator results in a considerable heat transfer intensification. Akyurek et al. [18] experimentally evaluated the forced convection flow of water-Al₂O₃ NF inside a horizontal tube equipped with wire coil turbulator. They utilized two turbulators with different pitches and found that the performance metrics of tube filled with NF without any turbulator is
superior to that of the cases with turbulator. Xiong et al. [19] simulated the efficacy of aqueous CuO NF flowing inside a tube having a compound turbulator. The outcomes portrayed that using turbulator elevates the rate of heat exchange between tube wall and NF. Xiong et al. [20] numerically investigated the forced convection of NF through a pipe equipped with a complex-shaped turbulator. They inspected the consequence of elevating width ratio, flow rate and pitch ratio on the performance features. It was found that the Nusselt number intensifies with boosting pitch ratio of turbulator. In a numerical investigation, Ahmed et al. [21] explored the forced convection of water-Al$_2$O$_3$ and water-CuO NFs inside a triangular duct with a delta-winglet pair of turbulator under turbulent flow regime. They reported the significant effect of using both NF and turbulator on the performance aspects.

The design of a heat transfer system can be done both according to the first-law or the second-law of thermodynamics. If the first-law applies, the system must have the highest overall hydrothermal performance, and if the second-law applies, the system performance must have the least irreversibility. The efficacy of NF flow in turbulator-equipped thermal units has rarely been investigated from a second-law perspective [22-25]. Sheikholeslami et al. [22] inspected the irreversibility features of turbulent flow of aqueous CuO NF inside a pipe equipped with complex turbulators. Li et al. [23] simulated the NF irreversibility in a tube with helical twisted tapes. The thermal irreversibility was found to be declined with elevating the height ratio of turbulator, while the opposite is true for the frictional irreversibility. Farshad and Sheikholeslami [24] analyzed the irreversibility aspects for aqueous Al$_2$O$_3$ NF flow in a solar collector having a twisted tape. The outcomes revealed that the irreversibility diminishes with the increase of diameter ratio. Al-Rashed et al. [25] examined the influence of nanomaterials type on the
irreversibility production of a NF in a heat exchanger. It was reported that the maximum irreversibility belongs to the platelet shape nanomaterials.

The aim of this numerical work is to evaluate the features of turbulent flow of a NF through a hairpin heat exchanger equipped with helical baffles in the annulus side from both the fist and second-law perspectives. The impacts of baffle pitch, $Re$ and $\phi$ on the NF efficacy are assessed. This investigation is the first work on the consequences of using helical baffle on the irreversibility production inside an annulus of a hairpin heat exchanger filled with NF.

2. Problem statement

Fig. 1 gives a demonstrative sketch of geometry under investigation. It is a hairpin heat exchanger with 100 mm length, 10 mm inner tube internal diameter, and 15 mm outer diameter internal diameter. Additionally, the wall thickness for both the inner and outer tubes is 1 mm. Moreover, the baffle pitch varies from 25 mm to 100 mm.

The purpose of using this device is to cool the NF passing through the annulus with the help of water passing through the inner tube. In both water and nanofluid streams, time is of the essence and both streams are in turbulent regime. Both streams enter the device at a uniform velocity and temperature and are discharged into the atmosphere. Also, the exterior wall of heat exchanger is considered as insulated and no slip condition is utilized on the walls.
(a) $B=25$ mm

(b) $B=33$ mm

(c) $B=50$ mm
3. Governing equations and problem parameters

Continuity equation

\[
\frac{\partial (u_i)}{\partial x_i} = 0 \quad (1)
\]

Momentum equation

\[
\frac{\partial}{\partial x_j} (u_j \rho_{nf} u_i) = - \frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[ \mu_{nf} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right] + \frac{\partial}{\partial x_j} \left( -\rho_{nf} \bar{u}_j' \bar{u}_i' \right) \quad (2)
\]

Energy equation

\[
\frac{\partial}{\partial x_i} (\rho_{nf} T u_i) = \frac{\partial}{\partial x_i} \left[ (\mu_t/P_{rt} + \mu_{nf}/P_{rnf}) \frac{\partial T}{\partial x_i} \right] \quad (3)
\]
where $\mu_t$ and $\rho_{nf}\bar{u}_j\bar{u}_i$ are defines as follows;

$$\mu_t = \rho_{nf}C_\mu k^2/\varepsilon$$

(4)

$$\mu_t \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3} \delta_{ij} \left(\rho_{nf}k + \mu_t \frac{\partial u_k}{\partial x_k}\right) = -\rho_{nf}\bar{u}_j\bar{u}_i$$

(5)

$k$ & $\varepsilon$ parameters are defined as follows;

$$\frac{\partial}{\partial x_j} \left[\frac{\partial}{\partial x_j} \left(\mu_{nf} + \frac{\mu_t}{\sigma_k}\right) - \rho_{nf}\varepsilon + G_k = \frac{\partial}{\partial x_i} \left(u_i\rho_{nf}k\right)\right]$$

(6)

where

$$G_k = -\frac{\partial u_j}{\partial x_i} \rho_{nf}\bar{u}_j\bar{u}_i$$

(7)

$$\frac{\partial}{\partial x_i} \left(\rho_{nf} u_i \varepsilon\right) = \frac{\partial}{\partial x_j} \left[\frac{\partial}{\partial x_j} \left(\mu_{nf}+\frac{\mu_t}{\sigma_\varepsilon}\right) + \frac{\varepsilon}{k} G_k C_{1\varepsilon} - \rho_{nf} \frac{\varepsilon^2}{k} C_{2\varepsilon}\right]$$

(8)

where

$$Pr_t = 0.85; C_\mu = 0.0845; \sigma_k = 1; \sigma_\varepsilon = 1.3; C_{1\varepsilon} = 1.42; C_{2\varepsilon} = 1.68.$$  

The irreversible generation in forced convection flow of NF flow is due to two sources, namely fluid friction and heat exchange. Therefore, the total irreversibility is computed as:

$$S_{gen} = \frac{\mu_{nf}}{T_0} \left\{2 \left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2 + \left(\frac{\partial w}{\partial z}\right)^2\right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)^2 + \left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}\right)^2\right\}$$

$$+ \frac{k_{nf}}{T_0^2} \left[\left(\frac{\partial T}{\partial x}\right)^2 + \left(\frac{\partial T}{\partial y}\right)^2 + \left(\frac{\partial T}{\partial z}\right)^2\right]$$

(9)

By integrating the whole computational domain, the total irreversibility can be obtained:
\begin{align*}
S_{tot} &= \int S_{gen} \, dv \quad (10) \\
Bejan number: \\
Be &= \frac{S_{gen}}{S_{tot}} \quad (11) \\

The characteristic of a NF includes its thermo-physical properties that establish the relationship between the base fluid and the nanoparticles. These basic relationships are defined as follows [28-33]: \\
\rho_m &= \rho_f (1 - \varphi) + \rho_p \varphi \quad (12) \\
(\rho C_p)_m &= (\rho C_p)_f (1 - \varphi) + (\rho C_p)_p \varphi \quad (13) \\
k_{nf} &= k_f \frac{k_p + 2k_f + 2\varphi(k_p - k_f)}{k_p + 2k_f - \varphi(k_p - k_f)} \quad (14) \\
\mu_{nf} &= (123\varphi^2 + 7.3\varphi + 1)\mu_{bf} \quad (15) \\

The heat transfer rates for hot and cold fluids are calculated as follows: \\
q_h &= \dot{m}_h C_{p,h} (T_{h,i} - T_{h,o}) \quad (16) \\
q_c &= \dot{m}_c C_{p,c} (T_{c,i} - T_{c,o}) \quad (17) \\

Average Nusselt number: \\
Nu &= \frac{h D_h}{k_f} \quad (18) \\

The heat transfer coefficient [34-35]: \\
h &= \frac{q''}{\Delta T_{LMTD}} \quad (19) \\
where: \\
\Delta T_{LMTD} &= \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \quad (20)
\end{align*}
where;

\[ \Delta T_2 = T_{h,o} - T_{c,i} \] (21)

\[ \Delta T_1 = T_{h,i} - T_{c,o} \] (22)

\( q'' \) is the total heat flux that the fluid receives over the entire computational domain and is calculated as [28, 29]:

\[ q'' = \frac{1}{A} \int q''(x)dx \] (23)

The pressure drop (\( \Delta P \)) is defined as follows:

\[ \Delta P = f \frac{L \rho u^2}{2 \frac{D_h}{2}} \] (24)

Thermal performance can be computed as:

\[ \eta = \frac{q}{\Delta P} \] (25)

4. Numerical scheme, validation and Grid independency

In the current numerical investigation, the simulations are conducted using the ANSYS Fluent 18.1 software. The governing equations are discretized using the second-order upwind technique. Besides, the SIMPLE scheme is utilized to perform the pressure-velocity coupling. The convergence metric is set to \( 10^{-6} \).

Four different types of grids are used to make the outputs of the problem independent of the mesh. Tetrahedral mesh is used. The walls have a boundary layer mesh with a factor of 5%. The Nusselt number is used to perform the mesh study. Finally, it was found that the most appropriate mesh is the one with 650538 element number.

To ensure the accuracy and validity of the results of present work, the experimental findings of Wen and Ding [37] and the numerical outcomes of Goktepe et al. [38] for the local convection coefficient of NF flowing through a uniformly heated tube are employed. The outcomes are
reported in Fig. 2. As can be observed, there is a good consistency between the outcomes and the largest discrepancy between our results and the data from the other works (i.e. Refs. [37, 38]) is 10%.

To further validate the simulations, the average Nusselt number of NF flow inside a channel with rectangular rib (in-line) reported by Vanaki and Mohammed [39] is compared with our results. As can be seen in Fig. 3, at low velocities, a good consistency is observed, and as the flow velocity elevates, the discrepancy of the results is elevated and reaches to 7%.

![Fig. 2. Validation with experimental [37] and numerical [38] works.](image-url)
5. Results and Discussion

The main focus of this numerical investigation is to analyze the turbulent flow of water-CuO NF inside a hairpin heat exchanger with helical baffle in the annulus side. The effects of $\varphi$ (0-4%), $Re$ (5000-10000) and baffle pitch (25-100 mm) on the performance metrics are investigated.

Fig. 4 depicts the contour plots of NF velocity and temperature for different helical baffle pitches at $Re_{nf} = 5000$ and $\varphi = 4\%$. It is seen that as baffle pitch augments, the NF velocity diminishes.
$B = 25 \text{ mm}$

$B = 33 \text{ mm}$

$B = 50 \text{ mm}$
Fig. 4. Contours of NF velocity (left) and temperature (right) in terms of baffle pitch at $Re_{nf} = 5000$ and $\varphi = 4\%$.

Fig. 5 gives the consequences of elevating $Re$ on the velocity and temperature contours of NF for baffle pitch of 25 mm. As is seen, intensifying the $Re$ entails an elevation in the NF velocity and a decrement in the NF temperature.
Fig. 5. Contours of NF velocity (left) and temperature (right) in terms of $Re$ for $B = 25$ mm.

Fig. 6 gives the Nusselt number in terms of baffle pitch for various $\varphi$. The Nusselt number improves by elevating both the $Re$ and $\varphi$, while it declines by augmenting the baffle pitch. For instance, at $Re_{nf} = 5000$ and $\varphi = 0\%$, elevating the baffle pitch from 25 to 100 mm results in a 19.05% decrease in the Nusselt number of NF, while this amount for $\varphi = 4\%$ is 15.62%. In addition, at baffle pitch of 25 mm and $\varphi = 4\%$, intensification of $Re$ from 5000 to 10000 causes a 61.83% decline in the Nusselt number of NF. Elevating the $Re$ entails a rise in the thickness of the velocity and thermal boundary layer, which elevates the temperature gradient and, consequently, elevates the convective heat transfer coefficient and Nusselt number. Moreover, intensification of $\varphi$ causes an augmentation in the $k_{nf}$ and, thereby, an augmentation in the
convective heat transfer and Nusselt number. Furthermore, intensifying the baffle pitch entail a decrease in the NF velocity and, therefore, a decrease in the convective heat transfer coefficient and Nusselt number.

(a)

(b)
Fig. 6. Variations of Nusselt number versus baffle pitch in terms of $\varphi$ for (a) $Re_{nf} = 5000$, (a) $Re_{nf} = 7500$ and (a) $Re_{nf} = 10000$.

One of the important issues when choosing a working fluid is that the fluid performance should be examined from both the heat transfer and pressure drop (pumping power) aspects. Fig. 7 gives the variations of the pressure drop of NF versus $\varphi$ and baffle pitch for different $Re$. It is observed that the pressure drop intensifies by boosting both the $\varphi$ and $Re$, while it reduces by boosting the baffle pitch. For example, at $Re_{nf} = 5000$ and $\varphi = 0\%$, boosting the baffle pitch from 25 to 100 mm results in a 73.59% reduction in the pressure drop of NF, while this amount for $\varphi = 4\%$ is 73.55%. Additionally, at baffle pitch of 25 mm and $\varphi = 4\%$, augmentation of $Re$ from 5000 to 10000 results in a 232.82% decrease in the pressure drop of NF. Boosting both the $Re$ and $\varphi$ causes an elevation in the NF velocity and, therefore, according to Eq. (), the pressure drop of NF elevates. On the other hand, elevating the baffle pitch causes an increment in the annulus fluid path and, as a result, an elevation in the pressure drop, while the NF velocity declines by
boosting the baffle pitch which results in a decrease in the pressure drop of NF. According to the obtained results, it can be concluded the effect of elevated annulus flow path on the pressure drop outweighs the effect of decreases NF velocity and, therefore, the NF pressure drop reduces with an elevation in the baffle pitch.

(a)
The results presented so far have shown that intensifying the \( \varphi \) entails an elevated Nusselt number and pressure drop, which the first is a favorable effect and the latter is undesirable. For the final decision on the usefulness of using NF in the heat exchanger under investigation, the performance index should be examined. Fig. 8 demonstrates the variations of the performance index versus \( \varphi \) and baffle pitch for different \( Re \). The outcomes reveal that the hydrodynamic performance of the water-CuO NF in the considered heat exchanger is superior to that of the pure water in just the following four cases:

- \( B = 100 \text{ mm}, \varphi = 4\% \text{ and } Re_{nf} = 5000 \text{ where } \eta = 1.029. \)
- \( B = 33.3 \text{ mm}, \varphi = 2\% \text{ and } Re_{nf} = 10000 \text{ where } \eta = 1.067. \)
- \( B = 33.3 \text{ mm}, \varphi = 4\% \text{ and } Re_{nf} = 10000 \text{ where } \eta = 1.054. \)
- \( B = 100 \text{ mm}, \varphi = 2\% \) and \( Re_n = 10000 \) where \( \eta = 1.000 \).
Fig. 8. Variations of performance index versus baffle pitch in terms of $\phi$ for (a) $Re_nf = 5000$, (a) $Re_nf = 7500$ and (a) $Re_nf = 10000$.

In the remainder of this section, the flow of water-CuO NF with $\phi = 2\%$ in the considered heat exchanger is examined from the perspective of irreversibility production. Fig. 9 displays the influences of $Re$ and baffle pitch on the frictional irreversibility of NF. As can be seen, the frictional irreversibility declines and rises with increasing baffle pitch and $Re$, respectively, and the $Re$ effect on the frictional irreversibility at lower baffle pitch is greater. For example, at $Re = 5000$, increasing the baffle pitch from 25 to 100 mm results in a 67.84% decrease in the frictional irreversibility. In addition, at $B = 100$ mm, the augmentation of $Re$ from 5000 to 10000 causes a 209.45% increment in the frictional irreversibility. By boosting the $Re$ at a constant $\phi$ (i.e. constant Prandtl number) and baffle pitch, the thickness of the velocity boundary layer decreases, which results in an elevated velocity gradient and thus an elevated frictional irreversibility. On the other hand, increasing the $Re$ at a constant $\phi$ and baffle pitch, results in an
elevated NF velocity and hence a decrease in the average NF temperature which results in an elevated frictional irreversibility. Elevating the baffle pitch results in a decrease in the flow mixing, which results in a decrease in the velocity gradient and a decrease in the NF temperature, which in turn decreases and elevates the frictional irreversibility. It can be concluded that the decreasing effect of velocity gradient on the frictional irreversibility is overcome by the increasing effect of NF temperature and, therefore, the frictional irreversibility decreases with increasing baffle pitch.

![Figure 9](image_url)

**Fig. 9.** Variations of frictional irreversibility of NF with $\varphi = 2\%$ versus baffle pitch in terms of $Re$.

Fig. 10 gives the changes of thermal irreversibility of NF with $\varphi = 2\%$ versus baffle pitch in terms of $Re$. It is seen that the thermal irreversibility declines with the elevation of $Re$. For example, at $B = 100$ mm, the intensification of $Re$ from 5000 to 10000 causes a 52.57% decrease in the thermal irreversibility. As mentioned before, the NF temperature decreases with increasing $Re$ at a constant $\varphi$ and baffle pitch, which results in an elevated thermal
irreversibility. Also, the augmentation of $Re$ results in improved mixing of the flow and consequently, a decrease in the temperature gradient which ultimately results in a decrease in the thermal irreversibility. It can be said that the impact of temperature gradient on the thermal irreversibility outweighs the impact of NF temperature and therefore, the thermal irreversibility declines with the rise of $Re$. Moreover, Fig. 10 shows that with the rise of baffle pitch at a constant $Re$ and $\varphi$, the thermal irreversibility first elevates and then reduces. The highest thermal irreversibility occurs at baffle pitch of 50 mm. Boosting of baffle pitch at a constant $\varphi$ and $Re$ reduces the flow mixing which results in the decrease of both the NF temperature and temperature gradient which respectively elevates and diminishes the rate of thermal irreversibility. Fig. 10 reveals that for the baffle pitch lower than 50 mm, the increasing impact of temperature outweighs the decreasing of temperature gradient and ultimately, the thermal irreversibility declines while the opposite is true for the baffle pitch higher than 50 mm.

Fig. 10. Variations of thermal irreversibility of NF with $\varphi = 2\%$ versus baffle pitch in terms of $Re$. 
The results presented in Figs. 9 and 10 show that increasing the baffle pitch results in a decrease in both the thermal and frictional irreversibilities, and therefore, it can be easily deduced that increasing the baffle pitch entails a declined total irreversibility. However, the influence of $Re$ on the total irreversibility cannot be predicted because the thermal irreversibility elevates with decreasing $Re$ and then decreases. Fig. 11 displays the variations of total irreversibility versus baffle pitch in terms of $Re$. For the baffle pitch of lower than 50 mm, the rise of $Re$ entails an elevated total irreversibility whereas for the baffle pitch of 100 mm, the total irreversibility first elevates with the rise of $Re$ and then decreases. The lowest total irreversibility occurs at $Re = 7500$ and $B = 100$ mm.

![Fig. 11. Variations of thermal irreversibility of NF with $\varphi = 2\%$ versus baffle pitch in terms of $Re$.](image)

At the end of this section, we examine how each of the frictional and thermal entropies contribute to the total irreversibility. To investigate this problem, the variations of the Bejan number with the baffle pitch and $Re$ are assessed. As can be seen in Fig. 12, the Bejan number of NF augments with the grow of baffle pitch and decline of $Re$. It is therefore concluded that the
contribution of thermal irreversibility to total irreversibility is higher in lower $Re$ and higher baffle pitch.

**Fig. 12.** Variations of Bejan number of NF with $\varphi = 2\%$ versus baffle pitch in terms of $Re$.

### 6. Conclusion

In this study, the first-law and the-second law of thermodynamics are employed to investigate the turbulent flow of aqueous CuO NF through a hairpin heat exchanger equipped with helical baffle in the annulus side. The influence of $Re$ (5000-10000), $\varphi$ (0-4\%) and baffle pitch (25-100 mm) on the performance metrics are assessed. The following results can be deduced from this simulation:

- With the rise of baffle pitch at a constant $Re$ and $\varphi$, the thermal irreversibility first elevates and then reduces.
- Intensifying the $Re$ entails an elevated NF velocity and a declined NF temperature.
• Pressure drop intensifies by boosting both the $\varphi$ and $Re$, while it reduces by boosting the baffle pitch.

• Frictional irreversibility declines and rises with increasing baffle pitch and $Re$, respectively.

• $Re$ effect on the frictional irreversibility at lower baffle pitch is greater.

References

[1] M.S. Nazir, A. Shahsavar, M. Afrand, M. Arici, S. Nizetic, Z. Ma, H.F. Oztop, A comprehensive review of parabolic trough solar collectors equipped with turbulators and numerical evaluation of hydrothermal performance of a novel model, Sustainable Energy Technologies and Assessments 45 (2021) 101103.

[2] S. Rostami, A. Shahsavar, G.R. Kefayati, A. Shahsavar Goldanlou, Energy and exergy analysis of using turbulator in a parabolic trough solar collector filled with mesoporous silica modified with copper nanoparticles hybrid nanofluid, Energies 13 (2020) 2946.

[3] M.B. Elsheniti, M.O. Elbessomy, K. Wagdy, O.A. Elsamni, M.M. Elewa, Augmenting the distillate water flux of sweeping gas membrane distillation using turbulators: A numerical investigation, Case Studies in Thermal Engineering 26 (2021) 101180.

[4] Q. Xiong, M. Izadi, M. Shokri rad, S.A. Shehzad, H.A. Mohammed, 3D Numerical Study of Conical and Fusiform Turbulators for Heat Transfer Improvement in a Double-Pipe Heat Exchanger, International Journal of Heat and Mass Transfer 170 (2021) 120995.

[5] M.E. Nakhchi, M. Hatami, M. Rahmati, Effects of CuO nano powder on performance improvement and entropy production of double-pipe heat exchanger with innovative perforated turbulators, Advanced Powder Technology 32 (2021) 3063-3074.
[6] Y. Khetib, H. Sait, B. Habeebullah, A. Hussain, Numerical study of the effect of curved turbulators on the exergy efficiency of solar collector containing two-phase hybrid nanofluid, Sustainable Energy Technologies and Assessments 47 (2021) 101436.

[7] I. Bashtani, J.A. Esfahani, K.C. Kim, Effects of water-aluminum oxide nanofluid on double pipe heat exchanger with gear disc turbulators: A numerical investigation, Journal of Taiwan Institute of Chemical Engineers 124 (2021) 63-74.

[8] H.A. Mohammed, H.B. Vuthaluru, S. Liu, Heat transfer augmentation of parabolic trough solar collector receiver's tube using hybrid nanofluids and conical turbulators, Journal of Taiwan Institute of Chemical Engineers 125 (2021) 215-242.

[9] M. Jafaryar, M. Sheikholeslami, R. Moradi, Nanofluid turbulent flow in a pipe under the effect of twisted tape with alternate axis, Journal of Thermal Analysis and Calorimetry 135 (2019) 305-323.

[10] M. Khoshbaght-Aliabadi, M. Farsi, S.M. Hassani, N.H. Abu-Hamdeh, A. Alimoradi, Surface modification of transversely twisted-turbulator using perforations and winglets: An extended study, International Communications in Heat and Mass Transfer 120 (2021) 105020.

[11] S.U.S. Choi, Enhancing thermal conductivity of fluids with nanoparticles, ASME FED 231 (1995) 99-105.

[12] A.H. Pordanjani, S. Aghakhani, M. Afrand, B. Mahmoudi, O. Mahian, S. Wongwises, An updated review on application of nanofluids in heat exchangers for saving energy, Energy Conversion and Management 198 (2019) 111886.

[13] A. Shahsavar, M. Jamei, M. Karbasi, Experimental evaluation and development of predictive models for rheological behavior of aqueous Fe3O4 ferrofluid in the presence of an
external magnetic field by introducing a novel grid optimization based-Kernel ridge regression supported by sensitivity analysis, Powder Technology 393 (2021) 1-11.

[14] A. Shahsavar, M. Shahmohammadi, E.B. Askari, CFD simulation of the impact of tip clearance on the hydrothermal performance and entropy generation of a water-cooled pin-fin heat sink, International Communications in Heat and Mass Transfer 126 (2021) 105400.

[15] A. Shahsavar, M. Rashidi, C. Yildiz, M. Arici, Natural convection and entropy generation of Ag-water nanofluid in a finned horizontal annulus: A particular focus on the impact of fin numbers, International Communications in Heat and Mass Transfer 125 (2021) 105349.

[16] E. Bellos, C. Tzivanidis, D. Tsimpoukis, Enhancing the performance of parabolic trough collectors using nanofluids and turbulators, Renewable and Sustainable Energy Reviews 91 (2018) 358-375.

[17] M.E. Nakhchi, J.A. Esfahani, Numerical investigation of turbulent Cu-water nanofluid in heat exchanger tube equipped with perforated conical rings, Advanced Powder Technology 30 (2019) 1338-1347.

[18] E.F. Akyurek, K. Gelis, B. Sahin, E. Manay, Experimental analysis for heat transfer of nanofluid with wire coil turbulators in a concentric tube heat exchanger, Results in Physics 9 (2018) 376-389.

[19] Q. Xiong, M. Jafaryar, A. Divsalar, M. Sheikholeslami, A. Shafee, D.D. Vo, M.H. Khan, I. Tlili, Z. Li, Macroscopic simulation of nanofluid turbulent flow due to compound turbulator in a pipe, Chemical Physics 527 (2019) 110475.

[20] Q. Xiong, M. Ayani, A.A. Barzinjy, R.N. Dara, A. Shafee, T. Nguyen-Thoi, Modeling of heat transfer augmentation due to complex-shaped turbulator using nanofluid, Physica A 540 (2020) 122465.
[21] H.E. Ahmed, M.Z. Yusoff, M.N.A. Hawlader, M.I. Ahmed, B.H. Salman, A.Sh. Kerbeet, Turbulent heat transfer and nanofluid flow in a triangular duct with vortex generators, International Journal of Heat and Mass Transfer 105 (2017) 495-504.

[22] M. Sheikholeslami, M. Jafaryar, J.A. Ali, S.M. Hamad, A. Divsalar, A. Shafee, T. Nguyen-Thoi, Z. Li, Simulation of turbulent flow of nanofluid due to existence of new effective turbulator involving entropy generation, Journal of Molecular Liquids 291 (2019) 111283.

[23] Z. Li, M. Sheikholeslami, M. Jafaryar, A. Shafee, A.J. Chamkha, Investigation of nanofluid entropy generation in a heat exchanger with helical twisted tapes, Journal of Molecular Liquids 266 (2018) 797-805.

[24] S.A. Farshad, M. Sheikholeslami, Nanofluid flow inside a solar collector utilizing twisted tape considering exergy and entropy analysis, Renewable Energy 141 (2019) 246-258.

[25] A.A.A.A. Al-Rashed, R. Ranjbarzadeh, S. Aghakhani, M. Soltanimehr, M. Afrand, T.K. Nguyen, Entropy generation of boehmite alumina nanofluid flow through a minichannel heat exchanger considering nanoparticle shape effect, Physica A 521 (2019) 724-736.

[26] P. Barnoon, D. Toghraie, F. Eslami, B. Mehmandoust, Entropy generation analysis of different nanofluid flows in the space between two concentric horizontal pipes in the presence of magnetic field: Single-phase and two-phase approaches, Computers & Mathematics with Applications 77 (2019) 662-692.

[27] P. Barnoon, D. Toghraie, R.B. Dehkordi, H. Abed, MHD mixed convection and entropy generation in a lid-driven cavity with rotating cylinders filled by a nanofluid using two phase mixture model, Journal of Magnetism and Magnetic Materials 483 (2019) 224-24.
[28] Z. Li, P. Barnoon, D. Toghraie, R.B. Dehkordi, M. Afrand, Mixed convection of non-Newtonian nanofluid in an H-shaped cavity with cooler and heater cylinders filled by a porous material: Two phase approach, Advanced Powder Technology 30 (2019) 2666-2685.

[29] P. Barnoon, D. Toghraie, R.B. Dehkordi, M. Afrand, Two phase natural convection and thermal radiation of Non-Newtonian nanofluid in a porous cavity considering inclined cavity and size of inside cylinders, International Communications in Heat and Mass Transfer 108 (2019) 104285.

[30] J.C. Maxwell, Treatise on electricity and magnetism, Oxford: Clarendon Press, 1873.

[31] H. Brinkman, The viscosity of concentrated suspensions and solutions, The journal of chemical physics 20 (1952) 571-577.

[32] B.C. Pak, Y.L. Cho, Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles, Experimental Heat Transfer an International Journal 11 (1998) 141-170.

[33] Y. Xuan, W. Roetzel, Conceptions for heat transfer correlation of nanofluids, International Journal of Heat and Mass Transfer 43 (2000) 3701-3707.

[34] M. Bahiraei, A. Godini, A. Shahsavar, Thermal and hydraulic characteristics of a minichannel heat exchanger operated with a non-Newtonian hybrid nanofluid, Journal of the Taiwan Institute of Chemical Engineers 84 (2018) 149-161.

[35] J. Alsarraf, A. Moradikazerouni, A. Shahsavar, M. Afrand, M.D. Tran, Hydrothermal analysis of turbulent boehmite alumina nanofluid flow with different nanoparticle shapes in a minichannel heat exchanger using two-phase mixture model, Physica A 520 (2019) 275-288.
[36] A. Karimi, A.A.A. Al-Rashed, M. Afrand, O. Mahian, A. Shahsavar, The effects of tape insert material on the flow and heat transfer in a nanofluid-based double tube heat exchanger: Two-phase mixture model, International Journal of Mechanical Sciences 156 (2019) 397-409.

[37] D. Wen, Y. Ding, Experimental investigation into convective heat transfer of nanofluids at the entrance region under laminar flow conditions, International Journal of Heat and Mass Transfer 47 (2004) 5181–5188.

[38] S. Göktepe, K. Atalık, H. Ertürk, Comparison of single and two-phase models for nanofluid convection at the entrance uniformly heated tube, International Journal of Thermal Sciences 80 (2014) 83–92.

[39] S.M. Vanaki, H.A. Mohammed, Numerical study of nanofluid forced convection flow in channels using different shaped transverse ribs, International Communications in Heat and Mass Transfer 67 (2015) 176-188.