Numerical assessment on the hydrothermal behavior and irreversibility of MgO-Ag/water hybrid nanofluid flow through a sinusoidal hairpin heat-exchanger

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Abstract
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Re and ϕ causes an enhancement in the heat transfer, while the reverse is true about the remaining performance aspects. In addition, it was found that the irreversibility due to flow friction intensifies by boosting either Re or ϕ. Moreover, the outcomes revealed that the heat transfer is the main source of irreversibility in the flow of hybrid nanofluid (NF) inside a sinusoidal hairpin heat-exchanger. Furthermore, it was reported boosting the amplitude results in a decrease in the performance index of the heat-exchanger.

**Keywords:** Hybrid nanofluid; Sinusoidal hairpin heat-exchanger; Irreversibility;

**Nomenclature**

- $a$: Wave amplitude (m)
- $A$: Dimensionless wave amplitude (-)
- $c_p$: Inner diameter (J/kg.K)
- $d_i$: Inner diameter (m)
- $d_o$: Outer diameter (m)
- $k$: Thermal conductivity (W/m.K)
- $L$: Length of heat-exchanger (m)
- $L_s$:Greek symbols
- $m_{nf}$: Mass flow rate (kg/s)
- $p$: Pressure (Pa)
- $q$: Performance index (-)
- $v$: Volumetric flow rate (m$^3$/K)
- $V$: Volumetric flow rate (m$^3$/K)
- $V̇$: Volumetric flow rate (m$^3$/K)
- $\dot{m}_{nf}$: Mass flow rate (kg/s)
- $\dot{Q}_{max}$: Maximum possible heat transfer rate (W)
- $\dot{Q}_{nf}$: Heat transfer rate (W)
- $\mu$: Viscosity (kg/m.s)
- $\rho$: Density (kg/m$^3$)
- $\rho_i$: Inlet velocity (m/s)
- $\eta$: Concentration (%)
- $\delta$: Wavelength (m)
- $\epsilon$: Effectiveness (-)
- $\dot{Q}_{nf}$: Heat transfer rate (W)
- $Re$: Reynolds number (-)
- $U$: Overall heat transfer coefficient (W/m$^2$.K)
- $V$: Velocity (m/s)
- $V̇$: Volumetric flow rate (m$^3$/K)
- $Ẇ$: Pumping power (W)
- $\Delta p$: Pressure drop (Pa)
- $\theta$: Effectiveness (-)
- $\nu$: Viscosity (kg/m.s)
- $\gamma$: Performance index (-)
- $\mu$: Viscosity (kg/m.s)

**Subscripts**
1. Introduction

Heat-exchanger is an apparatus utilized to exchange heat between two or more fluids. One of the most widely used heat exchangers in the industry is the hairpin heat exchangers, which are also known as double-pipe or tube-in-tube heat exchangers. They are largely utilized in industry because of its simplicity, ease of manufacturing and compactness. Because of their extensive industrial applications, performance improvement of heat-exchangers is one of the researchers’ fields of interests. Using corrugated wall channels is one of the techniques suggested by researchers to improve the heat-exchangers performance [1-10]. The production of corrugated channels is cheaper and easier than other heat transfer enhancement methods including vortex generators and turbulators. Corrugated walls enhance the rate of heat exchange by boosting the flow mixing intensity.

Many researchers have evaluated fluid flow over the corrugated heat-exchangers. In an experimental investigation, Moawed et al. [11] assessed the hydrothermal performance of a double-pipe heat-exchanger with sinusoidal inner pipe at various wavelengths and amplitudes of sine waves. They reported that both the heat transfer and pressure loss go up by using the corrugated heat-exchanger instead of the plain one. Hassan Khan et al. [12] performed simulations...
to evaluate the hydrothermal performance of a wavy printed circuit heat-exchanger (PCHE). They assessed the impact of angle of bend, $Re$, and flow pattern on the hydrothermal characteristics. The results revealed that the wavy channel based PCHE gives better performance than the plain channel based PCHE. Singh et al. [13] experimentally assessed the impact of inclination angle on the working aspects of a heat-exchanger with corrugated channels. They also studied the influence of flow rate of working fluid on the heat transfer and pressure loss through the heat-exchanger. They reported that the highest heat transfer occurs at the $20^\circ$ inclination angle. Esfahani et al. [14] performed simulations to examine the irreversibility generation through a wavy channel. The outcomes depicted that the thermal irreversibility is the main part of total irreversibility near the wall, while the flow friction irreversibility dominates in the central region. Additionally, the results demonstrated that the total irreversibility slightly augments with boosting the dimensionless amplitude. Impacts of corrugated channel configurations on the working matrices of a PCHE were studied by Aneesh et al. [15]. The considered channel configurations were triangular, sinusoidal and trapezoidal. A maximum of 41%, 33% and 28% heat transfer enhancement was obtained for the trapezoidal, sinusoidal, and triangular corrugated-channel PCHEs, respectively, in comparison with the smooth channel PCHE.

Several studies have demonstrated the possibility of improving the heat-exchangers efficiency through augmenting the working fluids thermal conductivity [16-22]. Typical heat transfer fluids such as ethylene glycol, water and engine oil have low thermal conductivity. For solving this problem, some researchers have presented the idea of adding metallic/non-metallic and metal oxide nano-additives with very great thermal conductivity to mutual working fluids [23-32]. Choi [33] could make these types of fluids for the first time, called nanofluids (NFs). The influences of nanofluids on the hydrothermal and irreversibility characteristics of heat-exchangers have been
evaluated by many researchers. Naik and Vinod [34] examined the heat transfer enhancement in a shell and helical coil heat-exchanger by means of different non-Newtonian NFs containing Fe₂O₃, Al₂O₃, and CuO nano-additives in aqueous carboxymethyl cellulose (CMC) base fluid. They studied the impacts of flow rate of cold fluid, shell side fluid (NF) temperature, and stirrer speed on the total heat transfer coefficient and shell-side Nusselt number (\( \text{Nu} \)). The outcomes revealed that the \( \text{Nu} \) augments by boosting \( \phi \), NF temperature and stirrer number. Besides, the results showed that the CuO-CMC NF has greater heat transfer performance than the other NFs. Bhanvase et al. [35] experimentally determined the impact of water based PANI (polyaniline) NF on the enhancement of heat transfer in a vertical helically coiled tube heat-exchanger. The heat transfer of 0.1% and 0.5% NFs were found to be 10.52% and 69.62%, respectively higher than water. Esfe et al. [36] utilized the multi-criteria optimization algorithm and response surface method to design a heat-exchanger working with organic NFs. Optimal results illustrated that to obtain a lowest pressure loss and a highest heat transfer coefficient, \( \phi \) should be at the minimum and maximum, respectively. Khoshvaght-Aliabadi et al. [37] numerically assessed the impact of simultaneous use of turbulator and aqueous NFs with Cu, Fe and Ag nano-additives on the performance matrices of a U tube heat-exchanger. They found that the heat transfer improves by 11-67% through using the turbulator. Moreover, the outcomes indicated that the heat transfer performance of Ag-water NF is better than the other two types of NFs.

The heat transfer enhancement of conventional heat transfer fluids are also possible by dispersing hybrid nano-additives, which are combination of two or more nano-additives. The fluids prepared with these hybrid nano-additives are known as hybrid nanofluids, which is the fastest developing area in materials science and engineering. The principle preferred standpoint of utilizing hybrid
nanofluid is by choosing a proper combination of nanoparticles, positive features can be improved and inconveniences can be covered due to their synergistic effect [38].

So far, very few works have been carried out on the performance enhancement of heat-exchangers using hybrid NFs. In an experimental study, Ahammed et al. [38] evaluated the irreversibility characteristics of Al$_2$O$_3$/graphene-water hybrid NF inside a hairpin heat-exchanger incorporated with a thermoelectric cooler. They reported that the hybrid NF reduces the total irreversibility in the minichannel heat-exchanger by 19.6%. In an experimental evaluation, Hormozi et al. [39] studied the impacts of surfactant on the hydrothermal behavior of Al$_2$O$_3$/Ag-water NF in a helical coil heat-exchanger. They used various surfactants in the mass fraction range of 0.1–0.4%. The results showed enhancement in thermal performance up to 16% with $\varphi = 0.2\%$. The thermal performance of the Al$_2$O$_3$/Ag-water hybrid NF in a helical coil heat-exchanger was experimentally evaluated by Allahyar et al. [40]. They presented that using the hybrid NF with $\varphi = 0.4\%$ results in a 31.58% increase in the heat-exchanger performance. Shahsavari et al. [41] numerically evaluated the impact of $\varphi$ and $Re$ on the working matrices and irreversibility characteristics of the CNT/Fe$_3$O$_4$-water NF flow in a hairpin minichannel heat-exchanger. They concluded that the minimum total irreversibility and the maximum heat transfer is achieved by applying the NFs with large $\varphi$ and low $Re$.

From the literature review, it is understood that there are not any archival works considering the influence of hybrid NFs on the hydrothermal and irreversibility aspects of a sinusoidal hairpin heat-exchanger. This work attempts to examine the impacts of various parameters such as $\varphi$, $Re$, and amplitude on the working conditions of a sinusoidal hairpin counter-current heat-exchanger from both the first and second law points of view. The studied NF is a suspension of Ag/MgO hybrid nano-additives (50% Ag and 50% MgO by volume) in water.
2. Description of the nanofluid and heat-exchanger

The numerical modeling is carried out on a water based hybrid NF containing Ag and MgO nano-additives with the average diameter of 25 nm and 40 nm, respectively. Cetyl Trimethyl Ammonium Bromide (CTAB) is applied for ensuring better stability of the NF. The method to prepare this NF has been completely introduced in Ref. [42].

Figs. 1(a) and 1(b) illustrate the three and two dimensional schematic of the studied hairpin heat-exchanger, respectively. It consists of two concentric tubes with length $L=0.212$ m, inner diameter $d_i=0.02$ m, and outer diameter $d_o=0.04$ m. Hot water flows inside the annulus side and cold NF flows through the inner tube. It is observed that the inner tube is composed of three portions. The first and third portions have flat walls, but the middle portion consists of a wavy wall with various amplitudes and wavelength of ($L_w = 0.022$ m). The upstream flat wall is used to assure a fully developed flow, while the downstream flat wall is used to escape any backflow into the test section. The wavy wall length is equal to $132$ mm ($=6 \times 22$ mm), while the length of each flat portions is equal to $40$ mm. The profile of the wavy wall is presented by:

$$s(x) = \frac{d_i}{2} + \delta \sin \left[ \frac{2\pi(x - L_d)}{L_w} \right] \quad (1)$$
Fig. 1. (a) Three and (b) two dimensional schematic sketch of the minichannel heat-exchanger under investigation.

3. Governing equations and numerical methods

For NFs, it is often considered that the base fluid and nano-additives are in thermal equilibrium and move with the same velocity. The NF is assumed incompressible in this study and the governing equations are as follows:

Continuity equation:
\[ \nabla \cdot (\rho_{nf} \mathbf{V}) = 0 \]  \( (2) \)

Momentum equation:
\[ \nabla \cdot (\rho_{nf} \mathbf{VV}) = -\nabla p + \nabla \cdot (\mu_{nf} \nabla \mathbf{V}) \]  \( (3) \)

Energy equation:
\[ \nabla \cdot (\rho \mathbf{V} c_{p,nf} T) = \nabla \cdot (k_{nf} \nabla T) \]  \( (4) \)
where $\rho_{nf}$, $\mu_{nf}$, $k_{nf}$ and $c_{p,nf}$ respectively denote density, viscosity, thermal conductivity, and specific heat. Additionally, $V$ is velocity vector, $p$ denotes pressure and $T$ represents temperature.

3.1. NF properties

The density and specific heat of the Ag/MgO-water hybrid NF are given by Eqs. (5) and (6), respectively:

\[
\rho_{nf} = (1 - \phi_{Ag} - \phi_{MgO})\rho_w + \phi_{Ag}\rho_{Ag} + \phi_{MgO}\rho_{MgO} \tag{5}
\]

\[
c_{p,nf} = \frac{(1 - \phi_{Ag} - \phi_{MgO})\rho_w c_{p,w} + \phi_{Ag}\rho_{Ag}c_{p,Ag} + \phi_{MgO}\rho_{MgO}c_{p,MgO}}{\rho_{nf}} \tag{6}
\]

where subscripts $w$ refers to water.

Also, the empirical models proposed by Esfe et al. \[43\] are used for obtaining the thermal conductivity and viscosity of the studied hybrid NF:

\[
k_{nf} = \frac{0.1747 \times 10^5 + \phi_{hnp}}{0.1747 \times 10^5 - 0.1498 \times 10^6\phi_{hnp} + 0.1117 \times 10^7\phi_{hnp}^2 + 0.1997 \times 10^8\phi_{hnp}^3} \tag{7}
\]

\[
\frac{\mu_{nf}}{\mu_w} = 1 + 32.795\phi_{hnp} - 7214\phi_{hnp}^2 + 714600\phi_{hnp}^3 - 0.1941 \times 10^4\phi_{hnp}^4 \tag{8}
\]

where

\[
\phi_{hnp} = \phi_{Ag} + \phi_{MgO} \tag{9}
\]

Thermophysical aspects of water and Ag and MgO nano-additives are presented in Table 1.

Table 1. Thermophysical aspects of base fluid and nano-additives \[44, 45\]

<table>
<thead>
<tr>
<th>Properties</th>
<th>Pure water</th>
<th>Ag</th>
<th>MgO</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>997.1</td>
<td>10500</td>
<td>3560</td>
</tr>
<tr>
<td>-------------------------</td>
<td>-------</td>
<td>-------</td>
<td>------</td>
</tr>
<tr>
<td>Density (kg/m$^3$)</td>
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<td></td>
<td></td>
</tr>
<tr>
<td>Specific heat (J/kg.K)</td>
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<td>235</td>
<td>955</td>
</tr>
<tr>
<td>Thermal conductivity (W/m.K)</td>
<td>0.613</td>
<td>429</td>
<td>45</td>
</tr>
<tr>
<td>Dynamic viscosity (kg/m.s)</td>
<td>0.000909</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

3.2. Boundary conditions

The flow of NF and water enters the heat-exchanger with uniform velocity and uniform temperature. The wall of external tube is insulated. At the center of both tubes, due to the symmetric, axis boundary-condition is utilized. On the tube wall, no-slip boundary condition is assumed. Furthermore, zero relative pressure is used at both tube outlets.

3.3 Data processing

The $Re$ of NF ($Re_{nf}$) and water ($Re_w$) can be calculated as [19]:

$$Re_{nf} = \frac{\rho_{nf} u_{in,nf} d_i}{\mu_{nf}}$$

$$Re_w = \frac{\rho_w u_{in,w} (d_o - d_i)}{\mu_w}$$

where $u_{in,nf}$ is the NF inlet velocity and $u_{in,w}$ is the water inlet velocity.

The heat transfer rate to NF (from hot water) is obtained as [19]:

$$\dot{Q}_{nf} = \dot{m}_{nf} c_{p,nf} (T_{out} - T_{in})_{nf}$$

where $\dot{m}_{nf}$ is mass flow rate of NF.

The total heat transfer coefficient is obtained by the following equation [19]:
One method for measuring heat-exchanger performance is to calculate its effectiveness. This parameter is defined as the ratio of the real heat transfer rate to the maximum possible heat transfer rate:\[\varepsilon = \frac{\dot{Q}_{nf}}{\dot{Q}_{max}} = \frac{T_{out,nf} - T_{in,nf}}{T_{in,w} - T_{in,nf}}\] (14)

To assess the heat-exchanger performance, it is vital to compute the energy consumption rate required for pumping the NF through the heat-exchanger. Pumping power is computed as [19]:\[\dot{W} = \dot{V}\Delta p\] (15)

where \(\dot{V}\) and \(\Delta p\) are respectively the volumetric flow rate and pressure loss.

When checking the heat-exchangers performance, both the heat transfer and pressure loss should be considered. To this end, the ratio of heat transfer rate to pressure loss, as follows [19]:\[\eta = \frac{\dot{Q}}{\Delta p}\] (16)

This parameter is called performance index.

3.4. Irreversibility

The second-law of thermodynamics is utilized as a basis for assessing the irreversibility (entropy generation) associated with simple heat transfer processes [23]. It can help designers identify major source(s) of irreversibility and then, improve the performance of the device by making decisions.
such as modifying the operating conditions of the device, optimizing the device geometry or changing the working fluid [23].

The irreversibility in the flow field comprises of two main parts; (i) flow friction, and (ii) heat transfer. The local irreversibilities due to heat transfer and flow friction can be gained as [23],

\[
\dot{S}_{g,h}^{'''} = \frac{k_{nf}}{T^2} \left[ \left( \frac{\partial T}{\partial x} \right)^2 + \left( \frac{\partial T}{\partial y} \right)^2 \right]
\]

(17)

\[
\dot{S}_{g,f}^{'''} = \frac{\mu_{nf}}{T} \left\{ 2 \left[ \left( \frac{\partial v_x}{\partial x} \right)^2 + \left( \frac{\partial v_y}{\partial y} \right)^2 \right] + \left( \frac{\partial v_x}{\partial y} + \frac{\partial v_y}{\partial x} \right)^2 \right\}
\]

(18)

Moreover, the total irreversibility (\(\dot{S}_{g,t}^{'''}\)) is obtained by the following equation [23]:

\[
\dot{S}_{g,t}^{'''} = \dot{S}_{g,h}^{'''} + \dot{S}_{g,f}^{'''}
\]

(19)

The global irreversibilities are computed by the integration of the local irreversibilities over the whole domain as below [23]:

\[
\dot{S}_{g,f} = \int \dot{S}_{g,f}^{'''} dV, \dot{S}_{g,h} = \int \dot{S}_{g,h}^{'''} dV, \dot{S}_{g,t} = \int \dot{S}_{g,t}^{'''} dV
\]

(20)

4. Numerical process and verification

The governing Eqs. (2)-(4) and the related boundary conditions have been solved numerically by the control volume method (SIMPLE approach). The convective and diffusion terms are discretized using a second order upwind method. For all parameters, the convergence criterion is set to \(10^{-6}\).

Mesh quality plays a very important role in forecasting precision of the result and convergence. In the present work, a structured grid according to a rectangular mesh is used throughout the model domain. A non-uniform square grid is produced during the domain with a more concreteness close
the wall, where the maximum velocity and temperature gradients arise. The local $Nu$ of the sinusoidal wall is evaluated for five several grids at $Re = 500$, $\varphi = 0\%$, $\delta=0.002$ m and $L_w=0.022$ m (Fig. 2). Finally, the mesh consisting of 60 radial nodes for both the tube side and the annulus side of the heat-exchanger, and 700 nodes in the longitudinal direction is chosen as the best one.

![Fig. 2. Variation of local $Nu$ along the sinusoidal wall for various grid sizes at $Re = 500$, $\varphi = 0\%$, $\delta=0.002$ m and $L_w=0.022$ m.](image)

The validation of results is recognized by numerical simulation of Duangthongsuk and Wongwises [46] experimental test runs for TiO$_2$-water NF flow in a hairpin heat-exchanger. Table 2 gives the outcomes for the variations of $Nu$ with $Re$. It is seen that there is a reasonable consistency between the results because of the maximum error about 5%.

**Table 2.** Finding of this present study compared to experimental results [46].
### Results and discussion

In this study, the hydrothermal and irreversibility characteristics of Ag/MgO-water hybrid NF are evaluated in a sinusoidal hairpin heat-exchanger. The numerical modeling are performed at hybrid nano-additive concentrations of 0.5 to 2%, dimensionless amplitudes of 0 to 0.3, Reynolds numbers of 100 to 1500 for the tube side, and constant $Re$ of 1000 for the annulus side. The nanofluid inlet temperature and the water temperature incoming the annulus section are respectively assumed as 298 K and 308 K.

Fig. 3 gives the velocity vectors through the hairpin heat-exchanger for $A = 0.2$, $\varphi = 2\%$ and $Re_{nf} = 1500$. By investigation of velocity vectors, it is observed that the flow recirculation areas form in diverging area of both tube side and annulus side. Fluid rotation causes the fluid flowing near the wall to mix with the cooler fluid in the center of the pipe, thereby lowering its temperature resulting in an increase in the heat transfer rate.
Fig. 3. Velocity vectors for flow of NF inside the sinusoidal hairpin heat-exchanger at $A = 0.2$, $\varphi = 2\%$ and $Re_{nf} = 1500$.

Figs. 4 and 5 show the contours of velocity and temperature for flow of Ag/MgO-water hybrid NF inside the sinusoidal hairpin heat-exchanger at various nano-additive concentrations and Reynolds numbers. It can be seen that the velocity of both fluids increases in the convergent regions of their path and decreases in divergent regions. In addition, it is observed that boosting both the parameters of $\varphi$ and $Re$ leads to an increase in the NF velocity. The temperature contours reveal that the temperature slope between the wall and the nearby fluid augments with boosting the $\varphi$ and $Re$ leading to higher wall heat flux which is because of the higher axial velocity.
Fig. 4. Velocity contours of hybrid NF flow through the sinusoidal hairpin heat-exchanger ($A = 0.2$) at (a) $\varphi = 0\%$ & $Re_{nf} = 100$, (b) $\varphi = 2\%$ & $Re_{nf} = 100$, (c) $\varphi = 0\%$ & $Re_{nf} = 1500$ and (d) $\varphi = 2\%$ & $Re_{nf} = 1500$.

Fig. 5. Temperature contours of hybrid NF flow through the sinusoidal hairpin heat-exchanger ($A = 0.2$) for (a) $\varphi = 0\%$ & $Re_{nf} = 100$, (b) $\varphi = 2\%$ & $Re_{nf} = 100$, (c) $\varphi = 0\%$ & $Re_{nf} = 1500$ and (d) $\varphi = 2\%$ & $Re_{nf} = 1500$.

Fig. 6 illustrates the changes of heat transfer rate with $\varphi$ at various Reynolds numbers for $A = 0.2$. It is observed that the augmentation of both parameters (i.e. $\varphi$ and $Re$) results in the improvement
of heat transfer rate. For example, at $Re_{nf} = 1500$, by boosting $\varphi$ from 0 to 2%, the heat transfer rate goes up by 7.88%. Also, at $\varphi = 2\%$, the augmentation of $Re$ from 100 to 1500 results in a 124.44% increase in the rate of heat transfer. At a constant $Re$, NF velocity is a function of the ratio of NF viscosity to density. The results show that this ratio increases with the augmentation of $\varphi$; meaning that the velocity and mass flow rate of NF both increase with the augmentation of $\varphi$. Conversely, the augmentation of $\varphi$ leads to the increase of thermal conductivity coefficient and the amount of heat transferred to NF and, therefore, the rise in the NF outlet temperature. So, according to Eq. (12), the augmentation of heat transfer rate with the increase of $\varphi$ is due to the augmentation of mass flow rate and outlet temperature. The augmentation of $Re$ at constant $\varphi$ increases the velocity and mass flow rate of NF and reduces the temperature of outlet NF. The results depicted in Fig. 6 indicates that the impact of flow rate increase outweighs the impact of outlet NF temperature reduction and, so, the rate of heat transfer augments with the increase of $Re$.

![Fig. 6. Heat transfer rate at various Reynolds numbers in terms of $\varphi$ for $A = 0.2$.](image-url)
Fig. 7 gives the changes of total heat transfer coefficient with $\varphi$ at various Reynolds numbers for $A = 0.2$. It is seen that the total heat transfer coefficient goes up with the augmentation of both parameters (i.e. $\varphi$ and $Re$). For example, at $Re_{nf} = 1500$, by boosting $\varphi$ from 0 to 2%, the total heat transfer coefficient increases by 7.57%. Also, at $\varphi = 2\%$, the augmentation of $Re$ from 100 to 1500 results in a 83.87% enhancement in the total heat transfer coefficient. The augmentation of $\varphi$ and $Re$ leads to the simultaneous augmentation of heat transfer rate and logarithmic mean temperature difference (LMTD). Fig. 7 reveals that the of heat transfer enhancement is higher than the LMTD and, so, the total heat transfer coefficient goes up with the augmentation of $Re$ and $\varphi$.

![Graph](image)

**Fig. 7.** Overall heat transfer at various Reynolds numbers in terms of $\varphi$ for $A = 0.2$.

Fig. 8 illustrates the heat-exchanger effectiveness against $\varphi$ at several Reynolds numbers for $A = 0.2$. According to this figure, the augmentation of $\varphi$ and $Re$ leads to the reduction of heat-exchanger effectiveness. For example, at $Re_{nf} = 1500$, by boosting $\varphi$ from 0 to 2%, the
effectiveness reduces by 5.92%. Also, at $\phi = 2\%$, the augmentation of $Re$ from 100 to 1500 reduces the effectiveness of heat-exchanger by 85.04%.

Fig. 8. Heat-exchanger effectiveness at various Reynolds numbers in terms of $\phi$ for $A = 0.2$.

The results of Fig. 6 illustrates that the rate of heat transfer is improved by using NFs; which is desirable. In selecting a working fluid for a heat-exchanger, in addition to heat transfer capacity, it is also imperative to evaluate the parameter of pumping power. Note that a considerable increase of pressure loss should not be associated with boosting the amount of heat transfer by using a NF. Hence, the changes of pumping power with $Re$ and $\phi$ should also be investigated. These results are presented in Fig. 9. It is observed that NF pressure loss rises with the augmentation of $\phi$ and $Re$; and that the impact of $Re$ on the augmentation of pressure loss is much greater than that of $\phi$.

As it was mentioned above, the flow velocity of NF increases with intensifying $\phi$ and $Re$.

According to Darcy’s equation ($\Delta p = f \frac{L}{2r_i} \frac{\rho_{nf} u_{in, nf}^2}{2}$), where $f$ denotes the friction factor defined
as $f = \frac{64}{Re_{nf}}$ (47), pressure loss is related to the square of flow velocity; and therefore, an increase in $\phi$ and $Re$ leads to a significant increase of pressure loss.

![Graph showing pressure loss at various Reynolds numbers in terms of $\phi$ for $A = 0.2$.](image)

**Fig. 9.** Pressure loss at various Reynolds numbers in terms of $\phi$ for $A = 0.2$.

Fig. 10 illustrates the changes of pumping power with $\phi$ at various Reynolds numbers for $A = 0.2$. The results reveal a surge of NF pumping power with the increase of $\phi$ and $Re$. Based on Eq. (15), pumping power is equal to the pressure loss of NF multiplied by its volume flow rate. Boosting $\phi$ and $Re$ increases the flow velocity of NF and, thus, its pressure loss and volume flow rate; thereby boosting the pumping power of NF.
The results obtained so far have revealed that the augmentation of both the $\phi$ and $Re$ results in the augmentation of heat transfer and pumping power, the first one being desirable and the second one, undesirable. Therefore, to decide on the effectiveness of using hybrid NFs in the examined corrugated heat-exchanger, from the viewpoint of the first law of thermodynamics, the performance index should also be evaluated. Fig. 11 shows the variations of performance index with $\phi$ at various Reynolds numbers for $A = 0.2$. It is observed that the performance index diminishes considerably with the augmentation of $\phi$ and $Re$. This shows that, from the perspective of the first law of thermodynamics, water has a better performance than NF in a sinusoidal heat-exchanger. The results also show that the performance index of NF is greater at lower Reynolds numbers.
Fig. 11. Performance index at various Reynolds numbers in terms of $\phi$ for $A = 0.2$.

Fig. 12 illustrates the impacts of sinusoidal tube amplitude on the heat-exchanger performance index containing a hybrid NF with $\phi = 2\%$ at various Reynolds numbers. It is seen that the increase of amplitude results in the decrease of performance index. By boosting the amplitude, the vortex flows formed in the divergent sections of the heat-exchanger are fortified; this improves fluid mixing and thus increases the heat transfer rate; however, it also adds to the pressure loss. By looking at Fig. 12 it can be found that the pressure loss due to the increase of amplitude is much higher as compared with the augmentation of heat transfer; as a result, the performance index of heat-exchanger is severely reduced with boosting the tube amplitude.
The rest of this section is devoted to the irreversibility characteristics of the Ag-MgO/water hybrid NF inside the hairpin sinusoidal heat-exchanger. Fig. 13 demonstrates the variations of the local irreversibility due to flow friction and heat transfer for the case of $A = 0.2$, $\varphi = 2\%$ and $Re_{nf} = 1500$. It can be seen that the flow friction and heat transfer irreversibilities mainly generate near the wavy wall, where the temperature and velocity gradients are high.
Fig. 13. Local irreversibilities due to (a) fluid friction and (b) heat transfer at $A = 0.2$, $\varphi = 2\%$ and $Re_{nf} = 1500$.

Fig. 14 displays the changes of the global heat transfer irreversibility with $\varphi$ at various Reynolds numbers for $A = 0.2$. The results indicate that at Reynolds numbers of 100 and 500, the augmentation of $\varphi$ results in the enhancement of global heat transfer irreversibility; while the opposite occurs at Reynolds numbers of 1000 and 1500. According to Eq. (17), the heat transfer irreversibility is a function of NF’s thermal conductivity, average temperature and temperature gradient. At a fixed $Re$, the augmentation of $\varphi$ leads to the enhancement of NF’s thermal conductivity and, thus, the enhancement of heat transfer irreversibility. Conversely, the augmentation of thermal conductivity results in the increase of heat transfer rate, thus, the increase of NF’s average temperature; therefore, the heat transfer irreversibility is reduced. Moreover, the augmentation of thermal conductivity leads to a more uniform distribution of NF temperature and the reduction of temperature gradient and, thus, the reduction of heat transfer irreversibility. Therefore, the presented results in Fig. 14 indicate that at Reynolds numbers of 100 and 500, the impact of the augmentation of thermal conductivity coefficient overcomes the impacts of average temperature increase and temperature gradient reduction; and the heat transfer irreversibility goes up with the augmentation of $\varphi$; while the opposite occurs at Reynolds numbers of 1000 and 1500.
Fig. 14 also demonstrates that at nano-additive concentrations of 0-1%, the augmentation of Re leads to the augmentation of heat transfer irreversibility; while at higher \( \varphi \), the augmentation of Re from 100 to 1000 leads to the increase of heat transfer irreversibility, but the further increase of Re causes the heat transfer irreversibility to diminish. At a fixed \( \varphi \), the augmentation of Re leads to the reduction of average NF temperature (hence boosting the irreversibility due to heat transfer), improved fluid mixing and, thus, the reduction of NF temperature gradient; which diminishes the heat transfer irreversibility. The results also show that in all the examined cases, the heat transfer irreversibility of NF is greater than that of the base fluid.

![Graph](image)

**Fig. 14.** Global heat transfer irreversibility at various Reynolds numbers in terms of \( \varphi \) for \( A = 0.2 \).

Fig. 15 displays the changes of the global flow friction irreversibility with \( \varphi \) at various Reynolds numbers. It is observed that the flow friction irreversibility augments by boosting both parameters (i.e. \( \varphi \) and Re). According to Eq. (18), the flow friction irreversibility is a function of NF’s viscosity, average temperature and velocity gradient. At a constant Re, the rise of \( \varphi \) results in the
enhancement of NF’s viscosity, velocity gradient and average temperature; the first and the second factors increase, and the third factor reduces the flow friction irreversibility. Fig. 15 shows that the augmentation of NF viscosity and velocity gradient has a greater impact on the flow friction irreversibility than the augmentation of average NF temperature; and consequently, the flow friction irreversibility goes up with the augmentation of $\varphi$. Also, at a fixed $\varphi$, the rise of $Re$ leads to the reduction of average NF temperature and the augmentation of velocity gradient, both of which result in the augmentation of flow friction irreversibility. Fig. 15 also demonstrates that in all the examined cases, the flow friction irreversibility of hybrid NF is greater than that of the base fluid. It should be mentioned that in all the considered cases, the heat transfer irreversibilities are much larger than the flow friction irreversibilities; hence, the variation pattern of the overall global irreversibility is similar to that of the heat transfer irreversibility.

**Fig. 15.** Global flow friction irreversibility at various Reynolds numbers in terms of $\varphi$ for $A = 0.2$. 

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Fig. 16 displays the influences of sinusoidal pipe amplitude on the global irreversibility due to heat transfer for hybrid NF with $\phi = 2\%$ at various Reynolds numbers. In this figure, various patterns can be observed for various Reynolds numbers. For example, at $Re_{nf} = 500$, the global heat transfer irreversibility goes up with the augmentation of amplitude from 0 to 0.2; but it is reduced with the further increase of tube amplitude. While at $Re_{nf} = 1000$, the global heat transfer irreversibility intensifies by boosting the amplitude. With the augmentation of amplitude, and thus the reinforcement of vortex flow in the divergent region of tube, the average temperature of NF rises and temperature gradient diminishes, thereby reducing and boosting the heat transfer irreversibility, respectively.

![Fig. 16. Global heat transfer irreversibility at various dimensionless amplitudes in terms of Re.](image)

Fig. 17 gives the impacts of sinusoidal tube amplitude on the global flow friction irreversibility of hybrid NF with $\phi = 2\%$ at various Reynolds numbers. The results indicate that the variations of flow friction irreversibility with amplitude have various patterns at various Reynolds numbers. For
instance, at $Re_{nf} = 100$, boosting the tube amplitude from 0 to 0.1 and from 0.1 to 0.3 leads to the reduction and the augmentation of the global flow friction irreversibility, respectively; while at $Re_{nf} = 500$, the augmentation of sinusoidal tube amplitude causes the flow friction irreversibility to diminish. With the augmentation of sinusoidal tube amplitude, the average temperature and the velocity gradient of NF go up, thereby reducing and boosting the flow friction irreversibility, respectively.

**Fig. 17.** Global flow friction irreversibility at various dimensionless amplitudes in terms of $Re$.

6. Conclusion

The aim of this work is to assess the hydrothermal behaviour and irreversibility characteristics for forced convective flow of Ag-MgO/water hybrid NF inside a sinusoidal hairpin heat-exchanger. The impacts of $\alpha$, $Re$, and amplitude on the heat transfer rate, total heat transfer coefficient, heat-exchanger effectiveness, pressure loss, pumping power, performance index, as well as the
irreversibilities due to heat transfer and flow friction are examined. The most important results are as follows:

- Heat transfer rate and total heat transfer coefficient augment by boosting either $Re$ or $\phi$, while the opposite is true about the heat-exchanger effectiveness, pressure loss, pumping power, and performance index.
- Performance index reduces with increase in either $Re$ or $\phi$.
- The heat transfer and flow friction irreversibilities mainly occur near the wavy wall, where the temperature and velocity gradients are high.
- At Reynolds numbers of 100 and 500, the global heat transfer irreversibility boosts by augmenting $\phi$; while the opposite occurs at Reynolds numbers of 1000 and 1500.
- Flow friction irreversibility augments by boosting either $Re$ or $\phi$.
- Heat transfer is the main source of irreversibility in the flow of Ag-MgO/water hybrid NF inside a sinusoidal hairpin heat-exchanger.

**References**


