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Combined effects of nanofluid and transverse twisted-baffles on the flow structures, heat transfer and irreversibilities inside a square duct-A numerical study

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Abstract:

This paper presents a three-dimensional, numerical thermo-hydrodynamic and second law analysis of nanofluid flow inside a square duct equipped with transverse twisted-baffles. A finite volume method is employed to simulate forced convection of heat in the system with the inclusion of Brownian motion of the nanoparticles. The ultimate aim is to gain further understanding of the underlying physical processes and also to determine the optimal design and working conditions of the system. The effects of variations in the pitch intensity (γ) from 180° to 540° and volume fraction of nanoparticles (ϕ) from 0 to 0.05 on the nanofluid flow, heat convection and thermodynamic irreversibilities are investigated. The numerical results show that the baffle with $\gamma=360^\circ$ features the maximum value of heat transfer coefficient among all values of γ . Additionally, the baffle with $\gamma=540^\circ$ shows the minimum pressure drop for the entire range of γ . Finally, it is shown that the thermal entropy generation decreases by increasing the volume fraction of nanoparticles or inserting baffles inside the duct.

Keywords: Nanofluid; Transverse baffles; Thermo-hydrodynamics; Entropy generation; Finite volume; Square duct.

Nomenclature

Be	Bejan number (-)
C_p	Specific heat at constant pressure ($\text{J kg}^{-1} \text{K}^{-1}$)
D	Distance between baffles (m)
D_h	Hydraulic diameter (m)
d_f	Molecular diameter of base fluid (nm)
d_p	Nanoparticle diameter (nm)
f	Friction factor (-)
H	Side of duct (m)
h	Convective heat transfer coefficient ($\text{W m}^{-2} \text{K}^{-1}$)
k	Thermal conductivity ($\text{W m}^{-1} \text{K}^{-1}$)

K_B	Boltzmann constant ($=1.3807 \times 10^{-23}$ J/K).
L	Length of the duct (m)
l_{BF}	Mean free path of water molecules (nm)
Nu	Nusselt number (-)
P	Pressure (Pa)
Pe	Peclet number (-)
Pr	Prandtl number (-)
q''	Heat flux (W/m ²)
Re	Reynolds number (-)
S	dimensionless entropy generation rate (-)
S_{th}'''	Thermal entropy generation rate (W m ⁻³ K ⁻¹)
S_v'''	Viscous entropy generation rate (W m ⁻³ K ⁻¹)
T	Temperature (K)
t	Thickness of baffles (m)
u, v, w	Velocity component in x, y and z directions, respectively (m s ⁻¹)
U_{in}	Inlet velocity (ms ⁻¹)
V_B	Brownian velocity (ms ⁻¹)
W	Baffle width (m)
x, y, z	Rectangular coordinates components (m)

23 *Greek symbols*

Δp	Pressure drop (-)
γ	twist intensity (-)
φ	Nanoparticle volume fraction (-)
μ	Dynamic viscosity (kg m ⁻¹ s ⁻¹)
ρ	Density of the fluid (kg m ⁻³)
$\forall$	Volume of the domain (m ³)

24 *Subscripts/superscripts*

25	ave	Average
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eff	Effective
0	Clear duct
in	Inlet
out	Outlet
x	Local
w	Wall
r	Ratio

1. Introduction

Ducts are frequently used in a very wide range of thermal engineering applications. The optimal design of thermal devices is of high economic and environmental significance and has been greatly emphasized in recent years [1-3]. Consequently, a number of passive techniques have been employed to improve the efficiency of thermal devices. These techniques include employing rough walls [4], inserting porous substance with large value of thermal conductivity [5], adding nanosized particles [6], installing turbulators or swirl flow devices [7], and employing corrugated surfaces [8].

Some researchers have used baffles or ribs inside ducts to enhance the heat transfer rate by restarting the thermal boundary layer after flow reconnection between the baffles or ribs. Promvonge et al. [9] investigated numerically the effects of in-line V-shaped discrete rib on the heat transfer characteristics of a fluid flow inside a square duct. They reported that the heat transfer rate increases by about 4 times through using ribs in comparison with the smooth duct at low values of Reynolds numbers. In a numerical study, Kwankaomeng and Promvonge [10] placed a 30° angled baffle on one wall of a square duct and investigated the effects of it on laminar convection heat transfer. They found that the thermal improvement factor for the modified duct by the baffle is much higher than unity and has a maximum value of 3.1. This reflects a significantly higher thermal performance of the modified duct in comparison with its clear counterpart. Noothong et al. [11] showed experimentally and numerically that the heat transfer rate in a square duct increases by using discrete V-finned tape inserts. The discrete V-finned tape creates two longitudinal counter rotating vortices along the duct, which cause an enhancement in the turbulence intensity and heat transfer rate [11].

A number of scholars have explored the potentials of transverse twisted-baffles to improve the heat transfer rate. For example, Nanan et al. [12,13] studied experimentally and numerically the heat transfer improvements in a circular tube by installing the transverse twisted-baffles. These authors used both transverse twisted-baffles and transverse baffles in the tube and performed a comparison between them. They concluded that the transverse twisted-baffles always feature a higher Nusselt number and a superior uniformity of Nusselt number in comparison with the case of transverse baffles.

It should be noted that currently design and manufacturing of the tubes and ducts containing baffles and flow swirlers and also the use of nanofluids are highly feasible and affordable. As a result, they have been already implemented in a large number of experimental studies, see for example Refs. [11,12,13,14].

Nanofluids have been vastly utilized in different ducts to increase the heat transfer rate. The literature on heat transfer enhancement in ducts by using nanofluids is rather extensive and reviews of this subject can be found in Refs. [15, 16]. For brevity reasons, here only a few exemplary works are discussed. Santra et al. [17] investigated numerically the heat transfer of laminar copper–water nanofluid flow in the space between two parallel walls. Their results showed that the Nusselt number increases with an increase in the volume fraction of nanoparticles. This was due to the increase in the effective thermal conductivity of the nanofluid for higher values of the volume fraction of nanoparticles. Zeinali Heris et al. [18] investigated experimentally the heat transfer of Al_2O_3 -water nanofluid flow inside a square duct. They observed 27.6% enhancement in heat transfer rate by adding the nanoparticles to the water with 2.5% volume fraction. Moreover, they reported that the convection coefficient enhances as the volume fraction of nanoparticles increases [18]. They further reported a reduction in the wall temperature by employing the nanofluid [18]. In another work, Zeinali Heris et al. [19] simulated numerically Al_2O_3 -water and CuO -water nanofluids flow through the ducts with square and triangular cross sections. They observed a higher value of pressure drop inside the ducts with CuO -water nanofluid in comparison with the Al_2O_3 -water nanofluid.

In addition to heat transfer analyses, there have been studies on entropy generation of flows through ducts [20]. Bianco et al. [21] performed an entropy generation analysis on nanofluid flow inside a square duct. Their analysis indicated that the thermal entropy generation is dominant at low values of the Reynolds number. Nonetheless, the viscous entropy generation becomes more significant as the Reynolds number or volume fraction of nanoparticles increases. Recently, Darbari et al. [22] calculated the entropy generation of Al_2O_3 -water nanofluid flow in the space between two parallel walls. They found that the total entropy generation increases as the Reynolds number increases. Due to the increases in Reynolds number, the magnitude of the growth in the total entropy generation is much higher for pure water in comparison with the nanofluid.

In recent years, combinations of nanofluid and other passive techniques for enhancing heat transfer have been examined. Behnampour et al. [23] used nanofluid and different ribs containing trapezoidal, rectangular and triangular shaped ribs in microchannels under laminar regimes. They concluded that for all considered Reynolds numbers, the heat transfer rates are minimum for rectangular rib shape. Sabaghan et al. [24] used nanofluid and vortex generators in a microchannel. Their results showed that the efficiency of the microchannel is enhanced by about 14% in comparison with the case of the plain channel. Moreover, it was reported that employing nanofluid by itself enhances the efficiency by about 27% [24]. Durga Prasad and Gupta [25] used Al_2O_3 -water nanofluid and twisted tape inserted in a u-tube for enhancing the heat transfer rate. For 0.03% volume fraction of nanofluid with twisted tape insertion, they reported an increase of 31.28% in Nusselt number in comparison with the pure water. Torabi et al. [26] investigated the effect of Al_2O_3 -water nanofluid and porous material on heat transfer and entropy generation of laminar forced convection in a 2D duct. The porous media in this study included two arrangements of square pillars. They found that the Bejan number increases as the nanofluid concentration increases [26].

A survey of the literature shows that the effects of longitudinally twisted-baffle have been studied extensively while there are very limited investigations on the effects of the transversely twisted-baffle [27]. Incorporation of transverse baffles leads to the generation of fluid mixing by disrupting the flow and decreasing the thickness of thermal boundary layer, which render an increase in heat transfer. Similar to other methods of enhancing heat transfer, addition of baffles magnifies the pressure drop and subsequently increases the pumping power and cost. Nonetheless, making some modifications in the geometry of baffles can improve their heat transfer and pressure drop characteristics. Such modifications may include making a twist on the baffles. Nanan et al. [12, 13] showed that twisted baffles feature superior heat transfer behaviour in comparison with the straight ones. This is due to the combined influences of the recirculation and swirl flows, while the straight baffles induce only recirculation flows. On the pressure drop penalty, it is noted that straight baffles disturb the flow by their whole front surfaces, while twisted baffles provide smaller surface for direct protrusion and the baffles also convey fluid along twisted ducts. Hence, twisted baffles offer a lower pressure drop penalty. In general, baffles containing axial and transverse baffles can be used as heat transfer augmentation devices.

In the only two available articles on the transverse twisted-baffles, the baffles were placed inside circular tubes [12, 13]. Nonetheless, the non-circular ducts are employed in numerous applications. These include compact heat exchangers, solar air heaters, and propulsion systems; and it even has some applications in biomedical devices [28]. It has been confirmed that non-circular ducts feature low-pressure loss as a superiority and low heat transfer rate as a drawback compared to circular ducts [28]. This issue of non-circular ducts has encouraged the researchers to enhance the thermo-hydrodynamic performance of such ducts through employing different passive techniques [7, 8, 28]. Understanding the underlying physical processes is essential for improving these techniques, particularly when combinations of heat transfer enhancement methods are employed. As such, understanding the thermo-hydrodynamic behaviour of twisted-baffles with a nanofluid flow is an essential necessity for design improvements. This further aids evaluating the thermodynamic irreversibilities and hence can lead to improving the second law performance of the duct. The current study aims at providing a physical insight into the thermal, hydrodynamic, and thermodynamic processes around a transverse twisted-baffle in a square duct with the nanofluid flow.

2. Numerical methods

2.1 Problem definition and assumptions

The schematic views of the duct equipped with a number of transverse twisted baffles are shown in Fig. 1. A fluid with uniform inlet velocity U_{in} and temperature T_{in} flows through a square duct with side H and length $L=14H$. The walls of the duct are heated under an iso-flux thermal boundary condition. Transverse twisted-baffles with distance D , height H , thickness $t=0.005H$, and width $W=0.25H$ are placed inside this duct. The first and last baffles are placed at the distance of H from the inlet and outlet of the duct. Three types of transverse baffles with different pitch intensities ($\gamma=180^\circ$, $\gamma=360^\circ$, $\gamma=540^\circ$) are considered in this work (see Fig. 1b). The distance between the baffles is fixed at $D/H=1.5$. A three-dimensional view of one of these baffles is shown in Fig. 2. The following assumptions are made throughout this paper.

- The flow is steady, laminar, three-dimensional, incompressible and developing.

- All simulations are performed for a fixed value of Reynolds number ($Re=200$). Moreover, the volume fraction of nanoparticles is varied in the range of 0 to 0.05.
- Water is used as the working fluid. Al_2O_3 is considered as the nanoparticles with a fixed particle diameter of 60 nm and the resultant nanofluid is assumed to be homogenous and single phase.
- The nanoparticles are in local thermal equilibrium with the base fluid and the Brownian motion of the particles is taken into account.
- Gravitational, thermal radiative and viscous heating effects are ignored.
- The thermal conductivity and hence heat absorption by the baffles is ignored.

Insert figure 1 here

Insert figure 2 here

2.2 Governing Equations

The following governing equations are used in the numerical modelling:

- Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (1)$$

- Momentum equations:

$$\rho_{eff} \left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = -\frac{\partial p}{\partial x} + \mu_{eff} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \quad (2)$$

$$\rho_{eff} \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial p}{\partial y} + \mu_{eff} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \quad (3)$$

$$\rho_{eff} \left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = -\frac{\partial p}{\partial z} + \mu_{eff} \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \quad (4)$$

- Energy equation:

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \frac{k_{eff}}{\rho_{eff} C_p} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \quad (5)$$

where k , C_p , ρ and μ denote thermal conductivity, specific heat at constant pressure, density, and dynamic viscosity, respectively. Note that the subscript “eff” represents the effective thermophysical properties. The models used for calculating these properties are fully described in Appendix A.

2.3 Entropy Generation Analysis

The local volumetric viscous and thermal entropy generations rates can be expressed as follows [1]:

$$S_{th}''' = \frac{k_{eff}}{T^2} \left[\left(\frac{\partial T}{\partial x} \right)^2 + \left(\frac{\partial T}{\partial y} \right)^2 + \left(\frac{\partial T}{\partial z} \right)^2 \right] \quad (6)$$

$$S_v''' = \frac{\mu_{eff}}{T} \left\{ 2 \left[\left(\frac{\partial u}{\partial x} \right)^2 + \left(\frac{\partial v}{\partial y} \right)^2 + \left(\frac{\partial w}{\partial z} \right)^2 \right] + \left[\left(\frac{\partial u}{\partial y} \right) + \left(\frac{\partial v}{\partial x} \right) \right]^2 + \left[\left(\frac{\partial w}{\partial y} \right) + \left(\frac{\partial v}{\partial z} \right) \right]^2 + \left[\left(\frac{\partial u}{\partial z} \right) + \left(\frac{\partial w}{\partial x} \right) \right]^2 \right\} \quad (7)$$

149 2.4 Parameters Definitions

150 The following parameters are utilized in this paper.

- 151 • Average heat transfer coefficient:

$$h_{ave} = \frac{q''}{\left[T_{w,ave} - 0.5 \times \left(\frac{T_{in,ave} + T_{out,ave}}{2} \right) \right]} \quad (8)$$

- 152 • Average Nusselt number:

$$Nu_{ave} = \frac{h_{ave} \cdot D_h}{k_f} \quad (9)$$

- 153 • Reynolds number:

$$Re = \frac{\rho_{eff} U_{in} D_h}{\mu_{eff}} \quad (10)$$

- 154 • Bejan number [29]:

$$Be = \frac{S_{th}'''}{S_{th}''' + S_v'''} \quad (11)$$

- 155 • Pressure drop ratio:

$$\Delta P_r = \frac{\Delta P}{\Delta P_0} \quad (12)$$

- 156 • Heat transfer coefficient ratio:

$$h_r = \frac{h_{ave}}{h_{ave,0}} \quad (13)$$

- 157 • Viscous and thermal entropy generation ratios:

$$(S_v)_r = \frac{S_v'''}{S_{v,0}'''}, (S_{th})_r = \frac{S_{th}'''}{S_{th,0}'''} \quad (14)$$

- 158 • Performance evaluation criteria:

$$PEC = \frac{Nu/Nu_0}{(f/f_0)^{\frac{1}{3}}} \quad (15)$$

159 where subscript 0 represents a clear duct flow (no baffles) with a zero concentration of nanoparticles.

160 2.5 Boundary Conditions

161 The following boundary conditions are defined.

- 162 • At the input section of the duct:

$$u = u_{in}, \quad v = w = 0, \quad T = T_{in} \quad (16)$$

- Along the duct surfaces:

$$u = 0, \quad v = 0, \quad w = 0, \quad q'' = 2000 \frac{W}{m^2} \quad (17)$$

- At the output section of the duct:

$$\frac{\partial u}{\partial x} = 0, \quad \frac{\partial v}{\partial x} = 0, \quad \frac{\partial w}{\partial x} = 0, \quad \frac{\partial T}{\partial x} = 0 \quad (18)$$

For the baffles, the no-slip and adiabatic boundary conditions are used.

2.6 Numerical procedure

Equations 1-5 with boundary conditions 14-16 are solved numerically by employing a pressure-based approach, as this has been shown to be suitable for low-speed incompressible flows [30]. A staggered grid arrangement (see Fig. 2) was employed to store the pressure and velocity terms at the cell centre and cell faces respectively. In addition, the SIMPLE algorithm was employed to provide the coupling between the pressure and velocity terms [30]. All terms were discretized by employing a second order UPWIND method. Eventually, the convergence criteria were satisfied when the residual values became smaller than 10^{-6} for all equations.

2.7 Mesh independence test and validations

A sample grid employed for the current simulations is disclosed in Fig. 2. This includes an unstructured mesh, which becomes denser around the baffles wherein the velocity and temperature gradients vary rapidly. A mesh independency test was conducted by employing four grid densities at $\gamma = 540^\circ$ and $\phi=0.03$ to ensure that the outputs were not sensitive to the number of cells. The results of this test are available in Fig. 3. It can be seen that the absolute value of percentage difference between the grids with 631763 and 765238 cells is 0.71%. Accordingly, the grid number of 631763 was deemed enough to generate the results independent of grid density.

[Insert figure 2 here](#)

[Insert figure 3 here](#)

To examine the accuracy of the numerical procedure, the current numerical results are compared with the experimental data of Zeinali Heris et al. [18]. These are for a clear square duct without installing any baffles at $\phi=0$ and for different values of Peclet number. The average Nusselt numbers are presented in Table 1. This table shows a good agreement between the experimentally measured and numerically calculated Nusselt numbers with the maximum difference of 9.6% and an average error of 8.8%.

[Insert table 1 here](#)

3 Results and Discussion

3.1. Thermo-hydrodynamics

The results of numerical simulations are presented in this section. Figure 4 discloses the tangential velocity vectors and non-dimensional axial velocity at $x=9H$ and $\phi=0.03$ for different values of pitch intensity. It

should be noted that in this figure the axial velocity has been normalized by the inlet velocity ($U = \frac{u}{U_{in}}$). A secondary flow with recirculation zone can be seen within the duct at $\gamma=180^\circ$. According to this figure, the number of recirculation regions increases with an increase in γ ; furthermore, the secondary flows are formed as the entrant fluid flows over the baffles. The recirculation flows created by the baffles have larger values of pressure gradient along the radial orientation and generate eccentric forces. These forces generate a stronger tangential velocity component within the nearby flow field. This leads to the enhancement of flow mixing on the internal surfaces of the duct and sweeps thermal energy from the surface of the duct more efficiently. Figure 4(b) shows that the maximum values of the non-dimensional axial velocity enhance as the value of pitch intensity increases. This implies that the longitudinal swirl flows formed by the transverse twisted baffles is stronger for the higher values of pitch intensity.

Insert figure 4 here

Figure 5 shows the variations of pressure drop ratio and heat transfer coefficient ratio with the volume fraction of nanoparticles for different values of pitch intensity. The variations of pressure drop ratio (see Eq. 11) and heat transfer coefficient ratio for the case of plain duct without baffles are also presented in these figures. It can be seen in Fig. 5(a) that the pressure drop ratio increases by increasing the volume fraction of nanoparticles for all values of pitch intensity and also for the case of the plain duct. As an example, the pressure drop ratio at $\gamma=360^\circ$ increases by about 18% as the volume fraction of nanoparticles increases from 0 to 0.05. This can be ascribed to the fact that a nanofluid with higher concentration has a larger value of viscosity. Generally, the pressure drop increases by applying baffles inside the duct in comparison with the case of a plain duct. The flow blockage generated by installing the baffles is the main reason for increasing the pressure drop in the duct. Figure 5a indicates that there is a negligible difference in the pressure drop ratios for the cases of $\gamma=180^\circ$ and $\gamma=360^\circ$. However, the pressure drop ratio reduces considerably for the case of $\gamma=540^\circ$ in comparison with the lower values of γ .

Figure 5(b) shows that the heat transfer coefficient ratio enhances as the volume fraction of nanoparticles increases. As an example, the heat transfer coefficient at $\gamma=360^\circ$ enhances by about 8.5% when the volume fraction of nanoparticles increases from 0 to 0.05. This is due to the enhancement in the effective thermal conductivity of the fluid through adding the nanoparticles with larger values of concentration. Moreover, the Brownian movement of nanoparticles (see Appendix A) is further an important mechanism for enhancing heat transfer by nanofluids. Generally, the heat transfer rate in the duct enhances by employing the baffles. As already explained, this is due to the fact that the recirculation flows generated by the baffles carry the liquid from the core of the duct towards the surface regions and enhance liquid mixing. Figure 5(b) indicates that the twisted baffle with $\gamma=180^\circ$ has the minimum value of heat transfer coefficient among all values of γ . However, the baffle with $\gamma=360^\circ$ shows the maximum heat transfer coefficient for the investigated values of γ and for all considered volume fractions of nanoparticles. As shown in Fig. 3, only one vortex exists in the case of $\gamma=180^\circ$, which has limited influences on the heat transfer process. For the case of $\gamma=360^\circ$, two large vortices are generated in the duct, which occupy most of the duct cross section. These strong vortices carry the hot liquid from the surface of the duct towards the central region and therefore are effective in enhancing the rate of heat transfer.

Insert figure 5 here

Figure 6 shows the variations of performance evaluation criterion (PEC) with the volume fraction of nanoparticles for different values of pitch intensity at $Re=200$. In general, the calculated values of PEC for this problem indicate that the use of transverse twisted baffles is advantageous in terms of the heat transfer improvement and pressure-drop. As shown Fig. 6, the PEC for this problem is varies between 1 to 1.2, depending on the numerical values of volume fraction of nanoparticles and twist intensity. It is observed that PEC increases by using twisted baffles and also by increasing the twist intensity. For example, PEC increases by about 15.8% when baffles with $\gamma=360^\circ$ are used, in comparison with the empty duct at $\phi=0\%$. Further, this figure shows that PEC increases by using nanofluid. The value of PEC increases by boosting the volume fraction of nanoparticles within $0<\phi<0.04$. Yet, it decreases by increasing the volume fraction of nanoparticles for the range of $0.04<\phi<0.05$. For example, PEC increases by about 4% through increasing the volume fraction of nanoparticles in the range of $0<\phi<0.04$ for an empty duct. These clearly indicate that using the baffles is superior to the use of nanofluids.

Insert figure 6 here

Figure 7 depicts the three dimensional flow structure for different values of pitch intensity at $Re=200$ and $\phi=0.03$. As shown in this figure, longitudinal swirl flows are induced by baffles, while the empty duct only generates a longitudinal flow. It should be mentioned that swirl flow induces a higher pressure gradient in the transverse direction. In addition, this flow induces centrifugal forces which cause a high tangential velocity component with more mixing of the liquid.

Insert figure 7 here

The three dimensional local Nusselt number distributions inside the duct for different values of pitch intensity at $Re=200$ and $\phi=0.03$, are shown in Fig. 8. It can be seen that employing the twisted transverse baffles increases the Nusselt number in comparison with the case of empty duct. Further, the Nusselt number distributions are more uniform for the case of enhanced duct by baffles in comparison with the case of empty duct. This is due to the superior fluid mixing caused by the induced swirl flow. Among baffles with different values of pitch intensities, the baffle with $\gamma=180^\circ$ has the minimum value of Nusselt number. Therefore, here the Nusselt number contours are shown for this case. It can be seen in Fig. 8 that for $\gamma=360^\circ$ the value of Nusselt number on the side walls of the duct (parallel walls with baffles) is greater than that for baffles with $\gamma=540^\circ$. However, for the baffles with $\gamma=540^\circ$ the values of Nusselt number on the upper and lower walls of the duct (Perpendicular to the baffles) are greater than those for the baffles with $\gamma=360^\circ$. This is due to the differences in the structure of the rotational flows induced by these two baffles.

Insert figure 8 here

Figure 9 shows the temperature field for different values of pitch intensity at $Re=200$ and $\phi=0$. As shown in this figure, the flow temperature increases along the duct length due to the existence of the imposed heat flux on the walls. It is clear from Fig. 6 that the flow temperature decreases by inserting transverse twisted baffles inside the duct. As anticipated, in the absence of baffles the flow temperature increases constantly and uniformly along the channel. However, the existence of a colder stream of the fluid around the baffles is evident in Fig. 6. It means the hotter fluid flows near the sharp corners of the channel while the fluid shows a

relatively lower temperature in the most of other areas. Further, the wavy structure of the temperature field reflects the influences of the induced vortices on the thermal characteristics of the duct. This is a clear manifestation of heat transfer enhancements by the baffles and indicates that the rotating flows formed by the baffles have favourable effects on the transport of heat. According to Fig. 6, insertion of transverse twisted-baffles with $\gamma=360^\circ$ leads to the lowest temperature due to strong heat transfer improvements. Yet, twisted-baffles with $\gamma=180^\circ$ cause the largest temperature increases, which is consistent with the lowest heat transfer enhancement of this baffle, discussed with respect to Fig. 5.

Insert figure 9 here

Figure 10 discloses the effects of nanoparticles on the temperature field at $Re=200$ and $\gamma=360^\circ$. This figure also shows the variation of fluid temperature around the fourth and fifth baffles. It can be seen that addition of the nanoparticles does not change the qualitative behaviour of the temperature field in the duct. However, as expected, the flow temperature decreases noticeably after adding nanoparticles to the fluid. This is due to improving the heat conduction of fluid by the nanoparticles.

Insert figure 10 here

Table 2 summarizes the results of some the published works on using different kinds of baffles in rectangular and square ducts. It can be seen that regarding the configuration of the baffles and the range of Reynolds number, there are wide ranges of heat transfer enhancement and pressure drop penalty. For example, for air flow in a square duct with 30° angled baffles and for $200 < Re < 2000$, it can be seen that the maximum Nu/Nu_0 and f/f_0 are 9 and 45 respectively. For a rectangular duct with V-shaped perforated baffles and for Reynolds number from 3800 to 19000, these values are 2.7 and 5.8 respectively. In our study, for nanofluid flow in a square duct with transverse twisted baffles and for $Re=200$, the maximum heat transfer ratio is in about 1.6, and maximum pressure drop ratio is 2.5.

Insert table 2 here

3.2 Entropy generation

Figure 11 shows the variations of viscous and thermal entropy generations ratios (see Eq. 13) with the volume fraction of nanoparticles for different values of pitch intensity. The entropy generations for the case of plain duct without employing the baffles are also presented in this figure. As shown in Fig. 11(a), the viscous entropy generation ratio increases by increasing the volume fraction of nanoparticles for all values of pitch intensity and also for the case of the plain duct. This is due to the fact that the viscosity of the fluid increases with an increase in the nanoparticles concentration and therefore intensifies the viscous irreversibility. For example, the viscous entropy generation ratio at $\gamma=360^\circ$ increases by about 20% as the volume fraction of nanoparticles varies from 0 to 0.05. Figure 11(a) shows that the viscous entropy generation ratio increases significantly by employing baffles in the duct. The swirl flows formed by the twisted baffle disturbs the liquid flow and intensifies the velocity changes particularly in the axial direction. This leads to an increase in the flow blockage and viscous irreversibility. The baffle with $\gamma=540^\circ$ generates the minimum value of viscous entropy generation, while this is maximum for the baffle with $\gamma=360^\circ$. As mentioned earlier, there are weak, small size vortices in case of $\gamma=540^\circ$ that generate less viscous irreversibility in the duct. However, two strong vortices generated in the case of $\gamma=360^\circ$ have the main effects on the intensification of viscous irreversibility in the duct.

It can be seen from Fig. 11(b) that, generally, the thermal entropy generation ratio decreases by increasing the volume fraction of nanoparticles for all values of pitch intensity and for the case of the plain duct. For instance, the thermal entropy generation ratio at $\gamma=360^\circ$ decreases by about 3.4% as the volume fraction of nanoparticles increases from 0 to 0.05. This is due to the enhancement of thermal conductivity of the fluid phase, which enhances heat transfer rate in the duct. This enhancement makes the temperature gradients in the flow field more gradual and resulting a diminution in the thermal irreversibility. In general, the thermal entropy generation ratio decreases by employing the baffles in the duct. Installing the transverse twisted-baffles increases the rate of heat transfer between the duct surface and the nearby flow and hence reduces the temperature difference between the wall and bulk flow. This creates a flow with more uniform temperature distribution and renders a lower thermal irreversibility. The baffle with $\gamma=180^\circ$ has the highest value of thermal entropy generation among the investigated values of γ , while the case of $\gamma=360^\circ$ produces the least of all. Figure 5(b) shows that these baffles have the lowest and highest values of heat transfer rate, respectively. As mentioned earlier, a higher value of heat transfer rate makes the temperature distribution uniform and causes a lower thermal irreversibility.

Insert figure 11 here

Figures 12 and 13 show the viscous and thermal entropy generations fields for different values of pitch intensities at $Re=200$ and $\phi=0.03$. The local fields for each case are shown in different axial sections. As depicted in Figs. 12 and 13, the viscous and thermal entropy increase from near the zero at the centre of the duct to a maximum value near the duct wall for the case of the plain duct. This is due to the negligible velocity and thermal changes at the core of the duct, and rather large values of such changes near the duct surfaces. However, the viscous entropy generation increases at the core of the duct by installing the baffles. As mentioned earlier, the swirl flows formed by the twisted baffle disturb the liquid around the centreline of the duct and this intensifies the velocity gradient, especially in the axial direction. Subsequently, it increases the flow blockage and magnifies the viscous irreversibility. By moving along the duct length, the thermal entropy generation extends toward the core section of the duct due to the increment in thermal boundary layer thickness. Note that the descriptions of the local entropy generation fields in Figs. 13 and 14 are consistent with the velocity and temperature fields depicted in Figs. 4 and 9.

Insert figure 12 here

Insert figure 13 here

An important difference between Figs. 12 and 13 is the magnitude of entropy generation. The viscous irreversibilities have a far smaller magnitude compared to the thermal irreversibility. Hence, the total entropy generation in the system is dominated by the thermal irreversibility. Nonetheless, the viscous entropy generation is examined here to further understand the influences of hydrodynamics on the entropy generation. At higher flow velocities, the flow blockage and resulting viscous irreversibilities will increase. Yet, the essential hydrodynamics and the basic pattern of the viscous entropy generation remain largely unchanged.

Figure 14 shows the Bejan number field at $\gamma = 360^\circ$, $Re=200$, and $\phi=0.03$. As shown in this figure, the thermal entropy generation is dominant in the most regions of the duct. An exception to this is the near-inlet part of the duct, wherein the growth of hydrodynamic boundary layer has contributed to the relative

significance of viscous irreversibilities. It is recalled that the investigated flow is in the laminar regime and the velocity gradient is rather small for this type of flow. As a result, in general, the thermal entropy generation is the dominant irreversibility in this flow. However, the velocity gradient at the inlet section of the duct is large and this increases the viscous irreversibility at this section. Furthermore, the inlet section is placed outside of the thermal boundary layer, where the temperature gradient is low.

Insert figure 14 here

4 Conclusions

This paper presented a computational study on the thermo-hydrodynamics and entropy generation of a laminar, nanofluid flow inside a square duct equipped with transverse twisted-baffles. The duct was subject to iso-flux thermal boundary conditions. The main objectives were to determine the optimal design conditions and also gaining a physical understanding of the thermal, hydrodynamic, and entropic behaviours of the system. The effects of the volume fraction of nanoparticles and pitch intensity on the pressure drop, heat transfer, and different types of irreversibilities were investigated. The main findings of this work can be summarised as follows.

- The number of recirculation regions in the flow increases with an increase in the pitch intensity.
- There is a negligible difference in the pressure drop ratios for the cases of $\gamma=180^\circ$ and $\gamma=360^\circ$ but the pressure drop ratio decreases considerably for the case of $\gamma=540^\circ$ in comparison with the other values of γ .
- The twisted baffle with $\gamma=180^\circ$ has the minimum value of heat transfer coefficient among all values of γ . On the other hand, the baffle with $\gamma=360^\circ$ has the largest heat transfer coefficient among all values of γ and volume fraction of nanoparticles.
- The viscous entropy generation ratio increases by increasing the volume fraction of nanoparticles or employing the baffles inside the duct.
- The thermal entropy generation decreases by increasing the volume fraction of nanoparticles or employing the baffles inside the duct.
- The viscous and thermal entropy generations are stronger near the duct wall.
- The thermal entropy generation is the dominant form of irreversibility in the most regions inside the duct with an exception at the inlet section.

Appendix A.

The following effective density is used for nanofluid [33]:

$$\rho_{eff} = (1 - \phi)\rho_f + \phi\rho_p \quad (A1)$$

where ϕ is the volume fraction and subscripts f and p represent fluid and particle, respectively. Moreover, the effective specific heat is calculated by employing the equation introduced by Zhou and Ni [34]:

$$C_{eff} = \frac{(1 - \alpha)\rho_f C_f + \alpha\rho_p C_p}{\rho_{eff}} \quad (A2)$$

A theoretical model developed by Masoumi et al. [35] is used for defining the effective dynamic viscosity of the nanofluid. This reads

$$\mu_{eff} = \mu_f + \frac{\rho_p V_B d_p^2}{72 N \delta} \quad (A3)$$

The effective dynamic viscosity is a subordinate of temperature (T), nanoparticle scale (d_p), nanoparticle concentration (ϕ), nanoparticle density (ρ_p), the Brownian velocity of nanoparticles (V_B), spacing between particles (δ) and the base-fluid physical characteristics. Further, N in this equation is calculated by [34]:

$$N = \mu_f^{-1} \left[(n_1 d_p + n_2) \phi + (n_3 d_p + n_4) \right]$$

$$n_1 = -0.000001113, \quad n_2 = -0.000002771$$

$$n_3 = 0.00000009, \quad n_4 = -0.000000393 \quad (A4)$$

The Brownian velocity of nanoparticles and spacing between particles are defined by [35]:

$$V_B = \frac{1}{d_p} \sqrt{\frac{18 K_B T}{\pi \rho_p d_p}} \quad (A5)$$

$$\delta = \sqrt[3]{\frac{\pi}{6 \phi}} d_p \quad (A6)$$

where K_B is Boltzmann constant ($=1.3807 \times 10^{-23}$ J/K). The effective thermal conductivity presented by Chon et al. [36] is utilized as follows:

$$\frac{k_{eff}}{k_f} = 1 + 64.7 \times \phi^{0.7460} \left(\frac{d_f}{d_p} \right)^{0.3690} \left(\frac{k_p}{k_f} \right)^{0.7476} \times \text{Pr}^{*0.9955} \times \text{Re}^{*1.2321} \quad (A7)$$

where d_f is the molecular size of the base fluid ($=0.3\text{nm}$). The Prandtl and Reynolds numbers in Eq. A7 are defined as

$$\text{Pr}^* = \frac{\mu_f}{\rho_f \alpha_f} \quad (A8)$$

$$\text{Re}^* = \frac{\rho_f K_B T}{3 \pi \mu^2 l_{BF}} \quad (A9)$$

where l_{BF} is the mean free path of water ($=0.17\text{nm}$).

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Fig. 2. A sample of unstructured grid employed in the simulations.

Fig. 3. Mesh independency test for the case of $\gamma = 540^\circ$ at $\varphi = 3\%$.

Tangential velocity vectors

Non-dimensional axial velocity

$\gamma = 180^\circ$

$\gamma = 360^\circ$

$\gamma = 540^\circ$

Fig. 4. Tangential velocity vectors and non-dimensional axial velocity at $x=9H$, $Re=200$, and $\phi=0.03$ for different values of pitch intensity.

(a)

(b)

Fig. 5. Variations of pressure drop ratio and heat transfer coefficient ratio with volume fraction of nanoparticles for different values of pitch intensity at $Re=200$.

Fig. 6. Variations of performance evaluation criterion (PEC) with volume fraction of nanoparticles for different values of pitch intensity at $Re=200$.

Empty duct

$\gamma = 180^\circ$

$\gamma = 360^\circ$

$\gamma = 540^\circ$

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Table 1. Comparison between the present results and experimental data of Zeinali Heris et al. [17] for a clear square duct without installing baffles.

Nu_{ave}			
$Pe^{1/3}$	Present study	Experimental results [18]	Percentage difference
13.73	5.49	6.02	9.6
15.08	5.87	6.27	6.8
16.31	6.25	6.84	9.4
17.55	6.69	7.23	8.1
18.69	7.12	7.78	9.3
19.68	7.55	8.23	9.0
20.62	7.94	8.70	9.6

761 **Table 2.** Results of some published works on using different baffles in rectangular and square ducts.

Authors	Configuration	Working fluid	Range of Reynolds number	Maximum Nu/Nu_0	Maximum f/f_0
Kwankaomeng and Promvonge [10]	square duct with 30° angled baffles	air	100 to 2000	9	45
Dutta and Hossain [31]	Rectangular channel with inclined baffles	air	12000 to 41000	4.8	11
Chamoli and Thakur [32]	Rectangular duct with V-Shaped Perforated Baffles	air	3800 to 19000	2.7	5.8

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