Heat transfer characteristics of an 180° bend pipe of different cross sections using nano-enhanced ionic liquids (NEILs)

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Abstract

The fluid flow and heat transfer characteristics of ionic liquids (ILs) and nano-enhanced ionic liquids (NEILs) (i.e., alumina-based $[C_4mim][NTf_2]$ and $[C_4mpyr][NTf_2]$) through circular and non-circular (rectangular) U-bends have been numerically investigated using finite volume method of ANSYS R 16. Extensive numerical simulations have been performed to study the effects of different pertinent parameters on the heat transfer and fluid flow behaviors. Two different heat transfer correlations have been developed using the Levenberg–Marquest (L–M) method of nonlinear regression analysis by varying the influencing parameters. The Reynolds number (Re), weight fraction of nanoparticles (ϕ) and aspect ratio (AR) (only for non-circular U-bend) have been varied in the ranges of 500–2000, 0–2.5% and 0.5–3.0, respectively. It has been observed that the Nusselt number increases with Reynolds number, weight fraction of nanoparticle dispersed in the ILs and the aspect ratio of the non-circular duct. We observe a higher heat transfer rate for the NEILs as compared to the ILs. For a particular type of NEIL and Reynolds number, we also obtain an optimum aspect ratio for a non-circular duct. The fluid flow features of the U-bend revealed two counter-rotating vortices in Y–Z plane physical properties, and quite frequently used in different at a curve angle of 90°. As the aspect ratio (for non-circular U-bend) increases up to 1.5, these counter-rotating vortices move toward the inner wall of the bend to intensify flow recirculation enhancing the heat transfer rate. We found the highest heat transfer rate for a rectangular U-bend (AR = 1.5) as compared to a square as well as a circular U-bend.

Keywords Ionic liquid · NEIL · Nanoparticle · U-bend · Laminar flow

List of symbols

AR	Aspect ratio
A_c	Cross-sectional area (m ²)
Nu	Average Nusselt number
D_h	Hydraulic diameter (m)
H	Duct height (m)
W	Duct width (m)
q''	Heat flux (W/m ²)
IL	Ionic liquid
T_b	Bulk temperature (K)
S	Perimeter (m)
Pr	Prandtl number

Re	Reynolds number based on hydraulic diameter
c_p	Specific heat (J/kg·K)
T_w	Wall temperature (K)
u	X-velocity component (m/s)
v	Y-velocity component (m/s)
w	Z-velocity component (m/s)

Greek symbols

ρ	Density (kg/m ³)
ϕ	Weight fraction of nanoparticles
θ	Angle in degrees
k	Thermal conductivity (W/m·K)
μ	Dynamic viscosity (Ns/m ²)

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Subscripts

in Inlet

NEIL Nano-enhanced ionic liquid

1 Introduction

In the recent years, the nano-enhanced ionic liquids (NEILs) have drawn attention of researchers and engineers because of their thermophysical properties for the heat transfer enhancement possessing a low toxic effect for the environment. NEILs have been preferred over nanofluids owing to their superior chemical and thermo-physical properties, and quite frequently used in different industries such as petrochemical, chemical, solar and nuclear to enhance the heat transfer characteristics. NEILs are typically the colloidal solution of ionic liquids and metallic nanoparticles. These colloidal solutions form a very stable suspension of metallic particles and base solution improving its thermophysical properties [1–3]. For high heat transfer applications with low environmental pollution, these fluids (ILs) possess favorable properties such as high thermal conductivity, heat capacity, thermal stability, solvating capability, low vapor pressure, low volatility, low environmental toxic effects and corrosion resistance to plastics and steel [4–7]. Moreover, NEILs possess an excellent thermal stability at high temperature [8], and unlike the water or ethylene glycol-based nanofluids, the nanoparticles in NEILs remain suspended in base fluid for a longer period.

Basically, the NEILs are organic salts at room temperature and contain organic cations (i.e., imidazolium, pyrazolium, pyridinium, pyridazinium, pyrimidinium, pyrazinium, triazolium, thiazolium and oxazolidium) and organic or inorganic anions (halogen, fluorinated). Although the ionic liquid has a low vapor pressure, but one of the drawbacks of IL is the low thermal conductivity which stems its use for the high heat transfer applications. Thus, researchers have attempted to increase the thermal conductivity of such fluids by introducing a very small amount of nanoparticles in it. The resulting solution is commonly known as nano-enhanced ionic liquids (NEILs) or ionanofluids. Several works have been carried to study the thermophysical properties of NEILs. Bridge et al. [9] studied thermophysical properties of two different types of NEILs (i.e., $[C_4mim][NTf_2] + Al_2O_3$ and $[C_4mim][NTf_2] + CB$ (Carbon Black)) by adding either Al_2O_3 or CB up to a maximum of 2.5 wt%. They found a 40% increase in volumetric heat capacity for Al_2O_3 -based NEILs. However, a 10% increase in the density of Al_2O_3 -based NEILs has been noticed. For CB NEILs, both the volumetric heat capacity and density were decreased by 30% and 10%, respectively, as compared to the IL. Paul et al. [10] experimentally measured

the rheological properties of four different NEILs: $[C_4mim][NTf_2]$, $[C_4mim][NTf_2]$, $[C_2mpyrr][NTf_2]$ and $[N_{4111}][NTf_2]$ ILs containing Al_2O_3 nanoparticles over the range of 0–2.5% by weight. They reported that the thermal conductivity of NEILs increases by 11% at 2.5% wt of Al_2O_3 , and it weakly depends on the temperature. Their measurements show that the shear viscosity of NEILs decreases with the strain rate as well as the bulk temperature depicting its non-Newtonian and shear thinning behavior. They also reported a higher heat capacity for NEIL as compared to the base fluid. In another work, the natural convection of $[C_4mim][NTf_2]$ ionic liquid in a rectangular enclosure has been experimentally studied by Paul et al. [11] to report a low heat transfer rate of IL as compared to the de-ionized (DI) water, which they attributed to the high viscosity of IL. In the above experimental study, they considered IL as a Newtonian fluid.

Minea and Murshed [12] comprehensively reviewed the thermophysical properties of various kinds of NEILs. They executed a CFD analysis of heat transfer characteristics of NEILs in a straight pipe subjected to a constant wall flux and assuming the NEILs as the homogenous, single phase and Newtonian fluids. They showed that convective heat transfer of nanofluids was better than the ILs. Multi-walled carbon nanotubes (MWCNT) and graphene particles were used to mix with ILs such as $[Hmim][BF_4]$, $[C_4mim][NTf_2]$, and $[C_2mim][EtSO_4]$. In the future, the research communities are hoping for a possible replacement of nanofluids and ILs by the high heat transfer capacity and environmental friendly NEILs for an extensive use in solar energy applications. In this regard, the absorption of solar radiation by the NEIL and subsequent energy transfer to a heating/cooling element have been studied by a few researchers [13–15].

The thermophysical and optical properties of SiC/ionic liquid nanofluids have been studied by Chen et al. [16]. They also measured optical properties such as transmissivity, extinction coefficient and the fraction of the solar energy absorbed. It was reported that the transmissivity of ionanofluids increases with the nanoparticle concentration, suggesting the concentrated ionanofluids could capture more radiation. Also, the extinction coefficient increases with SiC concentration, which makes the ionanofluids a better light absorbing fluid. Paul et al. [17] measured the thermophysical properties of 1-butyl-3-methylimidazolium bis{(trifluoromethyl) sulphonyl}imide $[C_4mim][NTf_2]$ with 0.18–0.9 vol% of Al_2O_3 nanoparticle for concentrated solar power (CSP). They showed that the thermal conductivity and heat capacity increase up to 11% and 49% for 0.9 vol% of NEIL. Jorjani et al. [18] measured various thermophysical properties of 1-butyl-3-methylimidazolium tetrafluoroborate ($[Bmim][BF_4]$) at room temperature. An experiment was carried out by Alizadeh and

Moraveji [19] to determine thermophysical properties of 1-butyl-3-methylimidazolium hexafluoro phosphate. They found that the viscosity decreases with the temperature and particle concentration, while electrical conductivity increases with the particle loading and temperature. However, it was revealed from their study that surface tension reduces with both temperature and particle concentration. Mineal et al. [20] carried out a numerical study of the natural convection heat transfer for a square enclosure through finite volume method using Al_2O_3 nanoparticles with a maximum volume fraction of 2.5% (by weight) thoroughly suspended in IL (i.e., $[\text{C}_4\text{min}][\text{NTf}_2]$). Two different correlations for Nusselt number have been proposed by them for two different configurations of the heated wall, i.e., first, the heated wall was placed at the bottom of the enclosure, and in the second case it was placed on the left wall of the enclosure.

Recently, Chereches et al. [21] conducted a thorough investigation on the heat transfer behavior of $[\text{C}_4\text{min}][\text{NTf}_2]$ and $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ dispersed with Al_2O_3 nanoparticles flowing through a straight pipe and subjected to a constant heat flux boundary condition. Finite volume method was adopted for their numerical analysis under laminar and turbulent regime as well. Minea et al. [22] carried out a thorough analysis of TiO_2 -water nanofluid in a straight pipe using both the single- and multiphase (mixture-model) approaches to benchmark their numerical results with that of well-known open experimental results. They reported some key findings, such as (1) numerical simulation underpredicts the experimental temperature, and hence overpredicts the heat transfer, (2) the gravity may not be neglected in the nanofluid simulations, (3) the multiphase model produces more reliable results than the single-phase models. Most importantly, their research revealed that the heat transfer analysis of nanofluids by a single-phase model produces error less than 10%, which may be quite good for any engineering applications. Barik et al. [23] implemented the single-phase model to study the heat transfer characteristics of a cross-flow jet impingement on a protruded surface using Al_2O_3 -water nanofluid. Recently, Barik et al. [24] used NEILs and ILs as working fluids to cool a heated square by evolving new designs for an embedded pipe implementing constructal theory.

Based on the aforementioned literature survey, a very few (i.e., only four papers available for numerical work) studies have been carried out to investigate heat transfer characteristics of NEILs. The NEIL, being an important fluid from the heat transfer augmentation point of view, requires further numerical studies exploring their heat transfer capabilities in different engineering devices relevant to practical applications. One such practical situation is the flow through a bend duct, which frequently finds

its application in many industrial setups. In this work, our prime objective is to analyze the heat transfer characteristics of a '180-degree bend pipe carrying different types of NEILs using the finite volume technique, and to develop a general purpose heat transfer correlation which would be handy and relevant for practicing engineers.

2 Validation of numerical method

The research papers on the experimental analysis of fluid flow characteristics of NEILs in a U-bend are scarce. However, there have been an abundant work on the thermofluid characteristics of nanofluid in a straight pipe subjected to a constant heat flux. First, our numerical method has been validated with the experimental result of a straight pipe carrying nanofluid [25] and subjected to a constant heat flux. A circular pipe of length 0.97 m and diameter of 0.0045 m has been taken for the validation. The water-based alumina nanofluids ($\gamma \text{ Al}_2\text{O}_3$ + water) with volume fractions of 1 vol% and 1.6 vol% are considered as the working fluid. A three-dimensional computational domain with hexahedral-type structural grid of 347,000 cells has been taken for the present validation. The validation of results is shown in Fig. 1a. It has been observed that our computational results match well with the experimental results of Ref. [25]. The maximum error (6.78% at 1.6 vol% of Al_2O_3) during this validation lies well in the accepted limit (i.e., $\leq 10\%$) for any engineering applications. Since the present study is focused on the heat transfer characteristics of NEILs, so we decided to carry out another validation for nano-enhanced ionic liquids. Thus, we selected a straight pipe of length 1.75 m and a diameter of 0.014 m in accordance with the experimental work of Sunder et al. [26]. Since the previous authors have conducted their experiments for a hybrid nanofluids $[\text{MWCNT-Fe}_3\text{O}_4/\text{water}]$, and our present work is concerned of the ILs and NEILs, we, therefore, selected Ref. [21] to validate our results.

Since the heat transfer characteristics of ILs as well as NEILs in a straight duct has been studied in Ref. [20] with a similar numerical boundary conditions as that of our present study, so we decided to validate our present numerical methods with Ref. [21]. The velocity inlet boundary condition is imposed at duct inlet with uniform velocity and inlet temperature, $T_o = 300 \text{ K}$. The pressure outlet boundary condition has been applied to the duct outlet, where the outlet pressure and temperature are equal with the ambient conditions. The duct wall is subjected to a constant heat flux of $12,998.83 \text{ W/m}^2$, with a no-slip for the velocity at the wall. The pressure outlet boundary condition essentially implies that the gradient of pressure in the direction of outward normal is zero, and the

Fig. 1 Variation of Nu with Re for straight pipe

equality of the outlet and ambient pressures suggests a zero gauge pressure at outlet. For the above validation, the pipe is meshed with hexahedral grids with 498,000 cells. The validation of present numerical methodology with Ref. [21] has been depicted in Fig. 1b. The working fluid for the validation was [C₄mpyrr][NTf₂]. It can be seen that the present numerical results very close to the results reported in Ref. [21] showing a maximum error of 3.64% at Re=500. Thus, the present results can be pretty good for engineering applications.

3 Physical geometry, numerical grid and boundary conditions

The physical domain considered in the present investigation is shown in Fig. 2. Two different cross sections (i.e., circular and rectangular cross sections) are taken for the present study. The grid arrangements for U-bends with circular and rectangular cross sections are depicted in Fig. 2a and b, respectively. For both the U-bends, structural grid

has been constructed in the computational domain. The pave and mapping meshing schemes have been used for the circular U-bend and the rectangular U-bend. Fine grids have been incorporated in the bend portion of the pipe (see Fig. 2b) so as to increase its density relative to the straight portions of the duct. The orientation of the U-bends is horizontal with diameters of 0.014 m and 0.002 m, respectively, for the circular and the rectangular U-bends. For both the configurations, the length of straight portions before and after the U-bend is taken as 0.6 m each, and the radius of curvature is taken as 0.175 m. It is expected a fully developed flow at entry to the bend because of the extra straight length provided before the bend. The pipe wall is entirely subjected to a constant heat flux of 12,998.82 w/m² [21]. The NEIL/IL enters the pipe at its inlet, where a velocity inlet (i.e., $u = V_{in}$, $v = 0$ and $w = 0$) boundary condition has been imposed. At the outlet of the pipe, the pressure outlet boundary condition is used. This boundary condition is numerically expressed as: $\frac{\partial \phi}{\partial n} = 0$, and, where 'n' is outward normal to a surface. No-slip (i.e., $u = v = w = 0$) and no temperature jump

Fig. 2 Geometrical models for a circular; and b non-circular U-bend with grid display

conditions are imposed to all the solid walls of the pipe. The entire computational domain is meshed with the hexahedral cells. Because of the existence of the secondary flow in the bend portions, the fine grids are constructed there. And relatively coarse grids are constructed in the straight portions of the U-bend. The details of the grid arrangement for the present study are illustrated in Fig. 2b.

3.1 Governing equations

To solve the governing equations, we assume that: (1) the flow is laminar and steady, (2) both the ILs and NEILs are incompressible, (3) the viscous heating is neglected as the Brinkmann number of the laminar flow is usually below one, (4) the thermal radiation is neglected, (5) the nanoparticles are well mixed with the base ionic liquids so that the resulting NEIL is treated as a single-phase flow [21]. However, the last assumption is debatable among the research community. However, it has been reported by many researchers [27–31] that there is a little or no difference in results obtained from single-phase model as compared to the two-phase model. Using the aforementioned assumptions, the continuity, momentum and energy conservation equations in inertial (non-accelerating) frame of reference are invoked as follows:

Continuity equation:

$$\nabla \cdot (\rho_{\text{NEIL}} \vec{V}) = 0 \quad (1)$$

Momentum equation:

$$\nabla (\rho_{\text{NEIL}} \vec{V} \vec{V}) = -\nabla P + \nabla \cdot (\bar{\bar{\tau}}) \quad (2)$$

Energy equation:

$$(\rho_{\text{NEIL}} \vec{V} \cdot \nabla T) = \frac{k_{\text{NEIL}}}{c_{p,\text{NEIL}}} \nabla^2 T \quad (3)$$

In Eq. (2), P , is the modified pressure, which includes the body force term due to gravity. The stress tensor, $\bar{\bar{\tau}}$, is expressed as the gradients of the velocity vector and its transpose as follows:

$$\bar{\bar{\tau}} = \mu_{\text{NEIL}} \left[(\nabla \vec{V} + \nabla \vec{V}^T) - \frac{2}{3} \nabla \cdot \vec{V} I \right] \quad (4)$$

In Eq. (4), I , is the identity tensor and the last term arises due to the volumetric dilation, which is zero for incompressible flow.

3.2 Physical properties of ILs/NEILs and numerical solution procedure

The thermophysical properties of $[\text{C}_4\text{mim}][\text{NTf}_2]$ and $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ are shown in Table 1 [10, 21]. All these properties have been utilized throughout our numerical investigation. It has been mentioned in Ref. [21] that the Prandtl number of NEIL is considerably increased with the addition of a small percentage of alumina into it.

The governing partial differential equations along with the boundary conditions as mentioned in Sect. 3 were numerically discretized over the entire U-bend through finite volume technique so as to convert the nonlinear partial differential equations to a set of algebraic equations. The algebraic equations are attentively solved by the whole field residual method of Ansys Fluent until the solutions converge. The pressure and velocity coupling has been established through the SIMPLE method. The user-specified convergence criteria are supplied to stop the solutions. In present simulations, we used 10^{-4} and 10^{-6} as convergence criteria for momentum and energy equations.

Table 1 Thermophysical properties of ILs and NEILs [10, 21]

ILs and NEILs	Specific heat (J/kgK)	Density (kg/m ³)	Thermal conductivity (W/m K)	Dynamic viscosity (Ns/m ²)
$[\text{C}_4\text{mim}][\text{NTf}_2]$	1740	1412	0.126	0.035
$[\text{C}_4\text{mim}][\text{NTf}_2] + 0.5\% \text{ Al}_2\text{O}_3$	1956	1450	0.129	0.037
$[\text{C}_4\text{mim}][\text{NTf}_2] + 1\% \text{ Al}_2\text{O}_3$	2230	1460	0.132	0.048
$[\text{C}_4\text{mim}][\text{NTf}_2] + 2.5\% \text{ Al}_2\text{O}_3$	2460	1506	0.136	0.099
$[\text{C}_4\text{mpyrr}][\text{NTf}_2]$	1560	1385	0.122	0.052
$[\text{C}_4\text{mpyrr}][\text{NTf}_2] + 0.5\% \text{ Al}_2\text{O}_3$	1750	1395	0.126	0.065
$[\text{C}_4\text{mpyrr}][\text{NTf}_2] + 1.0\% \text{ Al}_2\text{O}_3$	1950	1417	0.129	0.094
$[\text{C}_4\text{mpyrr}][\text{NTf}_2] + 2.5\% \text{ Al}_2\text{O}_3$	2630	1427	0.133	0.227

4 Results and discussion

Before discussing the results, a grid sensitivity test has been carried out to show the physical quantities of interest, i.e., the temperature and the velocity fields are independent of the number of cells/grids in the computational domain. The grid sensitivity test is shown in Fig. 3. Initially, the coarse cells (5×10^5) have been constructed in the computational domain, and the Nusselt number is found to be 26.29. Thereafter, the number of cells are gradually increased to 6×10^5 in the computational domain to register a 3.95% increment in the Nusselt number. The Nusselt number increases marginally (i.e., 0.914%) over its preceding value on further increasing the number of cells from 6×10^5 to 8×10^5 . Thus, the computational domain with 6×10^5 cells is declared as the grid independence domain.

The Nusselt number for all the calculations has been obtained from Eq. (5) as:

$$Nu_x = (h_x D_h) / k_{NEIL} \quad (5)$$

In Eq. (5), D_h is the hydraulic diameter [$D_h = \frac{4A_c}{S}$ (for non-circular U-bend) and $D_h = D$ (for circular U-bend)]. The local heat transfer coefficient (h_x) and the average Nusselt number obtained as follows:

$$h_{avg} = q'' / (T_w - T_b) \quad \text{and} \quad \overline{Nu} = \frac{1}{A} \int Nu_x dA \quad (6)$$

In Eq. (6), T_b is the bulk temperature of NEILs. The variation of the average Nusselt number ($\overline{Nu}$) with the Reynolds number of the ionic liquid ($[C_4mim][NTf_2]$) and NEILs containing 0.5–2.5 wt% of alumina is shown in Fig. 4a. The variation in $\overline{Nu}$ with Re in the same range of weight percentage of dispersed alumina in $[C_4mpyrr][NTf_2]$ has been depicted in Fig. 4b. It has been noticed from both the figures (Fig. 4a, b) that the Nusselt number increases with the Reynolds number for a particular NEIL and at a particular concentration.

As it could be seen from both Fig. 4a and b that the Nusselt number at particular weight fraction rapidly increases with the low Reynolds number, and the rate of increase is rather diminished at a high Reynolds number. The rapid increase in the Nusselt number is attributed to the short entrance lengths (both hydrodynamic and the thermal) pertaining to the low Reynolds number. The hydrodynamic and thermal entrance lengths of a circular pipe are directly proportional to the Reynolds number and Prandtl number of the fluids, and are computed as $x/D \geq 0.05Re$ and $x/D \geq 0.05PrRe$ [32], respectively. The ILs and NEILs are usually high Prandtl number fluids, and NEILs have higher thermal developing lengths as compared to the ILs due to the suspension of nanoparticles. The higher thermal developing length of NEILs ensures a smaller boundary layer thickness at a particular location [15, pp. 123–124], and as a result, the heat transfer

Fig. 3 Grid independence test

Fig. 4 Variation of $\overline{Nu}$ with Re for alumina nanoparticle-based **a** $[C_4mim][NTf_2]$; **b** $[C_4mpyrr][NTf_2]$ NEILs

coefficient is enhanced for NEILs. The longer thermal developing length of NEILs also ensures that the core of the pipe is occupied mostly by the cold fluid which can take more heat from the heated wall. The boundary layers become thinner at high Reynolds number, and hence the heat transfer rate increases with the Reynolds number.

Moreover, the effect of the weight fraction of Al_2O_3 nanoparticles on the heat transfer characteristics of NEILs is shown in Fig. 5. It has been observed that the heat transfer rate of the $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ is higher than that of the $[\text{C}_4\text{mim}][\text{NTf}_2]$. For example, at $\text{Re} = 500$ and weight fraction of 1%, it is evident from Fig. 5 that Nu for $[\text{C}_4\text{mim}][\text{NTf}_2]$ and $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ are 24.78 and 25.33, respectively. Similar observation has been obtained by [21]. Addition of the nanoparticles to ILs alters their thermophysical properties; and in particular, the thermal conductivity of the suspension (here NEIL) is dramatically improved to exhibit a

higher heat transfer characteristics as compared to base fluid (i.e., IL).

Also, the aggregation of the nanoparticles over due course of time has been a very common phenomena for most of the nanofluids: be it a simple water-based nanofluids or be it IL-based nanofluids. The aggregation of nanoparticles essentially improves the thermal conductivity [33, 34] of nanofluids and NEILs, which further assist in the heat transfer improvement. According to Ref. [34], the thermal conductivity of nanofluid is based on the Hamilton and Crosser model for multi-constituent systems, and it depends on particle volume fraction/weight fraction. The aggregation or local clustering of the nanoparticles increases the effective volume of the cluster than the volume of a single particle so as to enhance the thermal conductivity of the nanofluid. For a U-bend of cross section, we found that heat transfer is increased with nanofluid weight fractions as well as with the Reynolds number. Also, we observed that the $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ is a better heat transfer enhancer as compared to $[\text{C}_4\text{mim}][\text{NTf}_2]$.

Figure 6a illustrates the variation of the Nusselt number with Reynolds number for a square U-bend for $[\text{C}_4\text{mim}][\text{NTf}_2]$ -based NEILs. Figure 6b shows the same variations for $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ -based NEILs. At $\text{AR} = 1.5$, the variations of Nu with Re are shown in Fig. 7a and b for two different NEILs.

It has been observed that Nu increase with Re for both the NEILs, and the Nusselt number of $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ is more than that of $[\text{C}_4\text{mim}][\text{NTf}_2]$. For example, at $\phi = 2.5\%$ and $\text{Re} = 1000$, the Nusselt number for $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ and $[\text{C}_4\text{mim}][\text{NTf}_2]$ are found to be 29.12 (See Fig. 7b) and 28.09 (See Fig. 7a), respectively. For a rectangular U-bend (aspect ratio, $\text{AR} = 1.5$) and for the same working fluid (i.e., $[\text{C}_4\text{mpyrr}][\text{NTf}_2] + 2.5\% \text{Al}_2\text{O}_3$) and Reynolds number (i.e., $\text{Re} = 1000$), Nu for a square duct is 28.09 (See Fig. 6b), whereas Nu for rectangular duct ($\text{AR} = 1.5$) is

Fig. 5 Variation of Nusselt number for a circular bend for different types and volume fraction of NEIL

Fig. 6 Variation of Nu with Re for alumina nanoparticle-based a $[\text{C}_4\text{mim}][\text{NTf}_2]$; b $[\text{C}_4\text{mpyrr}][\text{NTf}_2]$ NEILs for a square U-bend

Fig. 7 Variations of $\overline{Nu}$ with Re for alumina nanoparticle-based **a** $[C_4mim][NTf_2]$; **b** $[C_4mpyrr][NTf_2]$ NEILs for a rectangular U-bend ($AR=1.5$)

Fig. 8 $\overline{Nu}$ variations with AR for **a** $[C_4mim][NTf_2] + 2.5\% Al_2O_3$; **b** $[C_4mpyrr][NTf_2] + 2.5\% Al_2O_3$

29.12 (See Fig. 7b). For all other weight fractions of Al_2O_3 , we observe a higher $\overline{Nu}$ for a rectangular duct ($AR=1.5$) as compared to that of a square duct ($AR=1$). The effect of the aspect ratios (AR) on the heat transfer rate of two different NEILs (i.e., $[C_4mim][NTf_2]$ and $[C_4mpyrr][NTf_2]$) has been depicted in Fig. 8a and b, respectively, for the Reynolds number range of 5000–2000. It could be seen from both the figures that an optimum Nusselt number is obtained at $AR = 1.5$ for both the NEILs. It is interesting to note that this optimum Nusselt number is independent of our considered Reynolds number and the type of fluids. It is noteworthy that similar optimum Nusselt number has also been obtained at other weight fractions. These plots have not been given here for the sake of brevity. We define the aspect ratio as the duct width (W) to height (H) ratio. The cross-sectional area of the duct is kept constant while changing the aspect ratio, which means that the duct width decreases and its height increases for $AR < 1$. It

resembles to a tall enclosure for $AR < 1$. Similarly, the duct width is increased while its height is reduced to maintain $A_c = \text{constant}$ and $AR > 1$, which means that the duct laterally becomes long and vertically becomes narrow. The cross-sectional area of the duct is kept constant in order to reveal the effect of the aspect ratio on the fluid dynamics of flow at a constant mass flow rate. The velocity vector, velocity contours and pathlines in a plane ($Y-Z$ plane) at a curved angle, $\theta = 90^\circ$ for three different cases: $AR = 1$, 1.5 and 2.0 have been shown in Fig. 9a–f. Figure 9a has been obtained by superimposing the velocity vectors on the velocity contour plot.

Similarly, Fig. 9b is obtained by superimposing the pathlines of the fluid particles on the velocity vector. At $AR = 1$, two counter-rotating vortices have been formed in $Y-Z$ plane at $\theta = 90^\circ$ due to the centrifugal force that pushes toward the outer wall of the bend. The fluid velocity in the vicinity of the outer wall of the bend is relatively larger than that of the

Fig. 9 Velocity vectors superimposed on velocity contours for at $Re = 1000$ and $\theta = 90^\circ$ for $[C_4mpyrr][NTf_2] + 2.5\% Al_2O_3$ **a** $AR = 1.0$; **c** $AR = 1.5$; **e** $AR = 2$, and the velocity vectors superimposed on particle pathlines **b** $AR = 1.0$; **d** $AR = 1.5$; **f** $AR = 2$

inner wall. Thus, fluid accelerates at the outer wall and decelerates near the inner wall [35] of the bend. At $AR = 1.5$, the channel width becomes more as compared to its height. It is interesting to note at $AR = 1.5$ that the two counter-rotating vortices shifts toward the inner wall of the bend, where a low velocity is expected. It is noticed from Fig. 9c and d that one stream of fluid moves from inner wall to central region of duct and other stream of fluid moves from outer wall to the central region. Since the fluid coming from the outer wall possess a higher momentum as compared to the fluid coming from the inner wall, so the inner fluid stream is pushed

by the outer fluid so that two vortices of higher strength are formed near the inner wall. However, the outer fluid is rushed to the top and bottom walls and is subsequently drawn into these vortices. Apart from these recirculation patterns, the static pressure at outer wall is higher than at the inner wall. The higher strength of the vortices along with the strong flow rationality and transverse pressure gradient are primarily responsible for high heat transfer rate at $AR = 1.5$. When the aspect ratio becomes 2.0, then the size of recirculation bubble is larger, but its strength is not so intense to impart a quick and efficient mixing of the hot and cold fluids.

It is observed from Fig. 9e and f that the size of the recirculation bubbles is larger than its size at any other aspect ratios. However, the core fluid has not been affected much due to the recirculation. Thus, the heat transfer rate is reduced. For bend pipe with circular cross section, the velocity contours superimposed with its velocity and the velocity vectors superimposed with its pathlines are shown in Fig. 10a and b. Similar to the previous observations, two counter-rotating secondary vortices are seen which directs fluid from outer wall to the inner wall. A comparison of the heat transfer rate by different cross sections has been shown in Fig. 11. At a low Reynolds number (i.e., $Re = 500$), the Nusselt number is found to be same for both the square and rectangular cross sections ($AR = 1.5$). However, as the Reynolds number increases the Nusselt number for a rectangular cross section ($AR = 1.5$) is found more than the square cross-sectional ducts. Evidently, the Nusselt number of a rectangular cross section ($AR = 1.5$) has been increased by 2.04% and 10.46%, respectively, as compared to the square and rectangular cross sections. It is believed in the perspective of heat transfer enhancement that the rectangular cross section ($AR = 1.5$) may be recommended for practical applications instead of the circular and square cross sections.

5 Heat transfer correlations

Based on aforementioned analysis, the Nusselt number of a bend duct may be expressed as the functional relationship among its pertinent independent variables for determination of its heat transfer characteristics. Two independent functional relationships, one for circular U-bend and the other one for the non-circular U-bend, have been formulated and are given as follows:

For circular U-bend:

Fig. 11 Comparison for Nusselt number for different duct shapes

$$\overline{Nu} = f(Re, Pr, \phi) \quad (7)$$

For non-circular U-bend:

$$\overline{Nu} = f(Re, Pr, AR, \phi) \quad (8)$$

In Eq. (8), $\overline{Nu}$, Re , Pr , AR and ϕ are the Nusselt number, Reynolds number, Prandtl number, aspect ratio and the weight fraction of alumina nanoparticles, respectively. The Reynolds number is based on either diameter of the duct or on the hydraulic diameter (i.e., for non-circular duct) of the duct. Extensive and careful numerical simulations have been executed to vary each of these dependent variables to get the value of Nusselt number. A nonlinear regression analysis has been carried out using L-M method of POLYMATH software. Initially, a function has been defined in terms of the independent variables with their correlation coefficients.

Fig. 10 Superimposed **a** velocity vector; and **b** particle pathlines at $Re = 1000$ and $\theta = 90^\circ$ in the Y-Z plane for $[C_4mpyrr][NTf_2] + 2.5\% Al_2O_3$

Some arbitrary guess values for these correlation coefficients have been supplied to the program to get the better values from the iterations. Trial-and-error attempts have been used to adjust the function as well as the guess values of its correlations coefficients until getting a R^2 value of 0.97. In all our present correlations, we get $R^2 = 0.97$. New functions have been defined and tested until getting the minimum error between the predicted and CFD results. Earlier, similar non-regression methodology has been adopted by Barik et al. [36] to develop heat transfer correlation for straight rectangular duct with surface ribs. The final form of the heat transfer correlations is given as follows:

For circular U-bend:

$$\overline{Nu} = 18.81Re^{0.033}Pr^{0.014} + 2(0.8\phi^{0.96} - 0.75\phi - 0.256) \quad (9)$$

For non-circular U-bend:

$$\overline{Nu} = 5.643Re^{0.0509}Pr^{0.0241} \times (1.6612 - 0.999AR + 2.232AR^{0.5218} + 0.0126\phi^{0.0399}) \quad (10)$$

It is worth mentioned here that 32 (i.e., for Eq. 9) and 160 (i.e., for Eq. 10) data points have been used to develop these correlations by varying the Reynolds number, Prandtl number, aspect ratio and weight fraction of nanoparticles in the range of: $500 \leq Re \leq 2000$, $483 \leq Pr \leq 4489$, $0.5 \leq AR \leq 3$ and $0 \leq \phi \leq 2.5$, respectively. For a circular U-bend, a comparison of the predicted and CFD values of the Nusselt number has been illustrated in Fig. 12a. It is observed from Fig. 12a that most of the values lie in the error band $\pm 1\%$, which is quite good for engineering calculations. Similarly, 160 data points have been used to develop the correlation. For non-circular ducts, a comparison of the predicted and CFD values of Nusselt numbers has been depicted in Fig. 12b. Figure 13

illustrates the comparison of the Nusselt number obtained from correlation and CFD computations against the Reynolds number. It can be observed that Nusselt number from the correlation matches quite well with the computed values having a maximum error of 4% at low Reynolds number.

Figure 14a shows variation of Nusselt number with Reynolds number for a non-circular U-bend at two different weight fractions of the nanoparticles. For both the fluids (i.e., NEIL and IL), the Nusselt number is observed to increase with Reynolds number. The Nusselt number predicted from Eq. (10) matches well with the correlation. Figure 14b shows the variation of the Nusselt number with the aspect ratio of the rectangular duct. As it has been observed earlier, the present correlation also predicts

Fig. 13 Comparison of the variations in $\overline{Nu}$ with Re obtained from CFD and correlation

Fig. 12 Comparisons of computed and predicted Nusselt numbers from a Eq. (9); b Eq. (10)

Fig. 14 Comparison of the variations in **a** Nu with Re at $AR = 0.5$ and **b** Nu with AR obtained from CFD and predicted values

the maximum Nusselt number at $AR = 1.5$, and the Nu obtained from the correlation matches well with the CFD values.

6 Conclusions

A numerical study has been executed for circular and non-circular U-bends carrying two different types of NEILs to investigate the heat transfer and fluid flow features of the bend using the finite volume method. Some conclusions drawn from this study are summarized as follows:

1. The Nusselt number is found to increase with Reynolds number irrespective of the cross-sectional area of the duct. A rapid increase in the Nusselt number has been observed at low Reynolds number, which is attributed to the low thermal and hydrodynamic entrance length at low Reynolds number.
2. For a particular duct shape and Reynolds number, the heat transfer increases with the weight fraction of the nanoparticles. A higher heat transfer has been noticed at a high weight fraction, which is probably due to the high thermal conductivity of the nanofluid at high weight fraction.
3. Among the circular and non-circular ducts with constant cross-sectional area, we find the non-circular ducts are better than the circular ones for heat transfer enhancement purpose. The transfer rate of a rectangular duct with an aspect ratio of 1.5 is found to be more as compared to the circular and square ducts. In the considered range of parameters, we obtain an optimum aspect ratio for non-circular ducts, and in

particular, for a rectangular duct when the aspect ratio is 1.5.

4. For the same Reynolds number, aspect ratio and weight fraction of alumina nanoparticles, the $[C_4mpyrr][NTf_2]$ is found to be better heat augementer as compared to $[C_4mim][NTf_2]$.
5. Two different correlations (one for circular U-bend and the other one for non-circular U-bend) for Nusselt number have been also developed by exacting a non-linear regression analysis of the data collected from CFD simulations. The predicted and computed values of Nusselt number are found to be in good agreement with each other. The present correlations are believed to be helpful for engineering applications, in particular, solar engineering applications.

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Compliance with ethical standards

Conflict of interest There is no conflict of interest among the authors.

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