Hydrothermal performance of different alumina hybrid nanofluid types in plate heat exchanger

Experimental study

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Abstract

To enhance the energetic and exergetic performances of the counter-flow plate heat exchanger with corrugations, several experiments were conducted using different Al_2O_3 hybrid nanofluids as a coolant. The combinations studied were Al_2O_3–SiC, Al_2O_3–AlN, Al_2O_3–MgO, Al_2O_3–CuO and Al_2O_3–MWCNT in 4:1 nanoparticle volume ratio and 100% Al_2O_3 with 0.1 vol% concentration suspended in DI water. Parameters which were varied are the coolant flow rate ranging from 2.0 to 4.0 lpm and coolant inlet temperature from 10 to 25 °C. Effect of coolant flow rates and inlet temperatures on different parameters like heat transfer coefficient ratio, heat transfer coefficient to pressure drop ratio, pump work, performance index and irreversibility has been investigated. A maximum enhancement of around 31.2% has been observed in the heat transfer coefficient for Al_2O_3–MWCNT (4:1) hybrid nanofluid with the negligible enhancement of 0.08% in the pump work and 12.46% enhancement in the performance index. The maximum enhancement in the irreversibility is around 1.6% for Al_2O_3–CuO (4:1) hybrid nanofluid. Among the studied nanoparticle combinations, Al_2O_3–MWCNT (4:1) gives better performance. This study suggests that Al_2O_3–MWCNT hybrid nanofluid can be a good alternative to enhance thermal performance.

Keywords Plate heat exchanger · Hybrid nanofluids · Heat transfer coefficient · Performance index · Irreversibility

Abbreviations

Al_2O_3	Alumina nanoparticle
CTAB	Cetyl trimethyl ammonium bromide
DI	De-ionized water
HEX	Heat exchanger
MWCNT	Multiwalled carbon nanotube
PHE	Plate heat exchanger
PI	Performance index
SDS	Sodium dodecyl sulfate
SEM	Scanning electron microscopy
SiO_2	Silica nanoparticle
TiO_2	Titania nanoparticle
vol%	Percentage volume concentration
mass%	Percentage mass concentration
x	Uncertainty

List of symbols

A	Effective area of heat transfer (m^2)
b	Channel spacing (m)
c_p	Specific heat ($J\ kg^{-1}\ K^{-1}$)
D	Diameter (m)
D_h	Hydraulic diameter (m)
E	Exergy rate (W)
G	Mass velocity ($kg\ s^{-1}\ m^{-2}$)
I	Irreversibility rate (W)
k	Thermal conductivity ($W\ K^{-1}\ m^{-1}$)
L_v, L_h	Horizontal and vertical distance between port centers, respectively (m)
m	Mass (kg)
$\dot{m}$	Mass flow rate ($kg\ s^{-1}$)
N_h, N_c, N_t	Number of hot channels, cold channels, total plates (dimensionless)
Nu	Nusselt number (dimensionless)
p	Pressure (Pa)
Pr	Prandtl number (dimensionless)
Q	Heat transfer rate (W)

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Re	Reynolds number (dimensionless)
t	Thickness of the plate (m)
T	Temperature (°C)
U	Overall heat transfer coefficient (W K^{-1} m^{-2})
V	Volume (m^3)
W	Pump work (W)
w	Mass (N)
X	Uncertainty (%)

Greek symbols

α	Heat transfer coefficient (W K^{-1} m^{-2})
Δp	Pressure drop (Pa)
η	Efficiency (%)
μ	Dynamic viscosity (Pa s)
Ω	Volumetric flow rate (m^3 s^{-1})
Φ	Volume concentration (dimensionless)
ρ	Density (kg m^{-3})

Subscript

av	Average
c	Cold
e	Ambient
h	Hot
i	Inlet
nf	Nanofluid
np	Nanoparticle
o	Outlet
w	Wall

Introduction

In today's scenario, plate heat exchangers (PHE) are used in many applications like chemical industry, milk chilling application and so on. A lot of effort has been made in enhancing its performance by altering plate surface texture, i.e., by providing waviness and corrugations on the plate surface which increases its heat transfer coefficient and hence thermal performance [1]. Its heat transfer characteristics can also be improved by altering the required thermophysical properties of the working fluid. This can be achieved by injecting one or more type of nano-sized particles in the base fluid. Properly engineered hybrid nanofluids obtained by suspending more than one nanoparticle in the base fluid have a high thermal conductivity as compared to base fluid and adjustable properties to suit different applications [2]. It has the potential to reduce thermal resistance and increase the thermal conductivity, which provides the gist necessary to begin research in the field of hybrid nanofluids [3, 4].

Within the last decade, many investigations were performed on plate heat exchanger using mono nanofluids and showed that the dispersion of nano-sized particles in the base fluid augments the performance of the plate heat exchanger [5–10]. But, studies on the application of hybrid nanofluid in PHE are limited. Huang et al. [11] used a mixture of Al$_2$O$_3$–water and MWCNT–water nanofluids in plate heat exchangers. They observed a rise in heat transfer coefficient and pressure drop with hybrid nanofluids. Kumar et al. [12] performed an energetic and exergetic analysis on the plate heat exchanger with Cu–Al$_2$O$_3$/water hybrid nanofluids. They showed the best performance for 5 mm plate spacing. Kumar et al. [13] performed an exergetic analysis on the plate heat exchanger with different MWCNT–water hybrid nanofluids and observed that 0.75% volume concentration of CeO$_2$–MWCNT/water hybrid nanofluid could be an excellent coolant to reduce exergy loss by 24.75%. Kumar and Tiwari [14] performed an experimental and numerical investigation on plate heat exchanger using TiO$_2$–MWCNT/water hybrid nanofluid. They found that discrete phase model results are in good agreement with the experimental results. Bhattad et al. [15] carried a numerical investigation on the corrugated plate heat exchanger with Al$_2$O$_3$–MWCNT/water hybrid nanofluid. They observed an improvement in the heat transfer coefficient. Bhattad et al. [16, 17] performed energy, exergy and economic analyses in a plate heat exchanger using brine-based hybrid nanofluids for low-temperature applications and observed that propylene glycol-based Al$_2$O$_3$–MWCNT hybrid nanofluid gives better performance. Bhattad et al. [18] used brine-based hybrid nanofluids as a coolant in plate type milk chiller and found moderate energy and exergy performance improvement.

Table 1 shows the comparison of previous experimental studies on the plate heat exchanger using hybrid nanofluids. As shown, all the studies considered only the single–hybrid nanofluid except by Kumar et al. [13]. They have considered various MWCNT-based nanoparticle combinations for hybrid nanofluid, which is not cost-effective due to the high cost of MWCNT. However, with the best of the authors' knowledge, no study considered Al$_2$O$_3$ (easily available at low cost with chemical stability)-based particle combinations for hybrid nanofluids in PHE. So an attempt has been made to explore the effect of different alumina–water hybrid nanofluids (Al$_2$O$_3$–SiC, Al$_2$O$_3$–AlN, Al$_2$O$_3$–MgO, Al$_2$O$_3$–CuO and Al$_2$O$_3$–MWCNT in 4:1 particle volume ratio) as a coolant on the energy–exergy performance of the plate heat exchanger for sub-ambient temperature. The hybrid nanofluids prepared are of 0.1 vol% (fixed) concentration. Considered hybrid nanofluids have been compared in terms of various energy–exergy parameters (heat transfer coefficient ratio, heat transfer coefficient to pressure drop ratio, performance index, pump work and irreversibility). Effects of hybrid nanofluid flow rate and inlet temperature on the performances have been discussed as well.

Table 1 Experimental studies on plate heat exchanger using hybrid nanofluids

References	Operating variables	Hybrid nanofluid	Findings
Huang et al. [11]	Used as hot fluid; Re = 182–956, Pr = 5.5–8, discharge (0–0.16 lps), T_{hi} = 28 °C, T_{ci} = 14 °C.	MWCNT + Al_2O_3/water with ratio = 1:2.5	HTC increases with a small penalty of pumping power
Kumar et al. [12]	Used as coolant, T_{ci} = 20 °C, T_{hi} = 50 °C, Ω_c = Ω_h = 3 lpm, b = 2.5–10.0 mm	Cu + Al_2O_3/DI water, (0.5–2.0 vol%), surfactant: CTAB	Exergy destruction was lowest for 5 mm spacing at 0.75 vol%
Kumar et al. [13]	Used as coolant, T_{ci} = 20 °C, T_{hi} = 50 °C, Ω_c = Ω_h = 3 lpm	Al_2O_3 + MWCNT, TiO_2 + MWCNT, ZnO + MWCNT, CeO_2 + MWCNT/water, (0.25–2.0 vol%), surfactant: CTAB.	The highest reduction in exergy loss has been observed at around 24.75%
Kumar and Tiwari [14]	Used as coolant, T_{hi} = 348 K, T_{ci} = 293 K, m_h = 100 kg h^{-1}, m_c = 220 kg h^{-1}	TiO_2 + MWCNT/water, (0.0–1.5 vol%)	Discrete phase model results are in good agreement with experimental results

Preparation and characterization

Hybrid nanofluid preparation

Hybrid nanofluid was prepared by using the two-step method. The calculated amounts of Al_2O_3 and other nanoparticles purchased from Alfa Aesar and Otto Chemika were measured by an electronic balance (SHIMADZU, ATX224, Japan) and then mixed with DI water. The mixture was stirred for 1 h with a mechanical stirrer and ultrasonicated for 4 h at a particular temperature in an ultrasonication system (MJL Lab instruments and equipments, India) in order to have excellent stability and homogenization. Firstly, Al_2O_3 nanofluid with 0.1% volume concentration was prepared and then through the same procedure, hybrid nanofluids (10 L) were prepared containing Al_2O_3 and other nanoparticles in 4:1 particle ratio with 0.1 vol% concentration. Different surfactants like CTAB and SDS have also been added to avoid the deposition of nanoparticles and make the solution stable for a longer time. Surfactants act as surface-active agents affecting the interfacial characteristics like the surface tension of the fluid. They induce electrostatic repulsions that counterbalance Van der Waals forces. However, CTAB is cationic and SDS is anionic in nature. For some particles like SiC, AlN and MWCNT, CTAB is good surfactant and for some particles like CuO and MgO, SDS is a better option [13, 19, 20]. So both were useful for the present study. The prepared solution was kept undisturbed for a long time to check its stability through gravimetric analysis. No sedimentation was observed in the prepared hybrid nanofluid solution, which means that the mixture made is stable for a long time (for at least 10 h). It is stable for an even longer time but 10 h is sufficient time for conducting our experiments, so we focused for this time period. Equation (1) was used to calculate the volume fractions of solids in nanofluids. To prepare the hybrid nanofluid with different particle ratios, the required nanoparticle mass was calculated by

$$m_{np1} = R\phi V_{nf}\rho_{np1} \quad m_{np2} = (1-R)\phi V_{nf}\rho_{np2}$$
$$R = \phi_{np1}/\phi \tag{1}$$

where ϕ is the solid volume fraction, ρ is the density in kg m^{-3}, m is the mass in kg, and V_{nf} is the required volume of hybrid nanofluid in m^3.

Hybrid nanofluid properties

For the experimental investigation, different properties have been measured using different equipments like KD2 thermal properties analyzer, the USA for thermal conductivity and heat capacitance per unit volume measurement, LVDV-II + Pro Brookfield digital viscometer for viscosity measurement and digital weighing machine for the mass of various fluids. The density was measured by applying the expression $\rho = m/V$. After getting the value of heat capacitance per unit volume, we have divided it with measured density to obtain the value of specific heat. Various measured thermophysical properties of different fluids are listed in Table 2.

# Experimental investigation

Experimental facility

Test matrix in this investigation is a plate type HEX whose dimensions are taken from Bhattad et al. [15]. There are two flow streams, for the cold and hot fluids (hybrid nanofluid and DI water flow loops) as described in Fig. 1.

The coolant loop contains an isothermal bath, a float type flowmeter and a manometer. Here, different types of Al_2O_3–DI water hybrid nanofluid are acting as a coolant.

Table 2 Thermophysical properties of different fluids at ambient temperature (35 °C)

Different fluids	Thermal conductivity/W m^{-1} K^{-1}	Specific heat/J kg^{-1} K^{-1}	Density/kg m^{-3}	Viscosity/Pa s
DI Water	0.5964	4183.0	996.8	0.0008706
Al$_2$O$_3$ (5:0)	0.6004	4170.0	999.5	0.0008786
SiC (4:1)	0.6063	4174.0	998.2	0.0008732
AlN (4:1)	0.6052	4169.0	998.3	0.0008742
MgO (4:1)	0.6018	4172.0	998.4	0.0008752
CuO (4:1)	0.5989	4167.0	1000.1	0.0008792
MWCNT (4:1)	0.611	4168.0	997.6	0.0008806
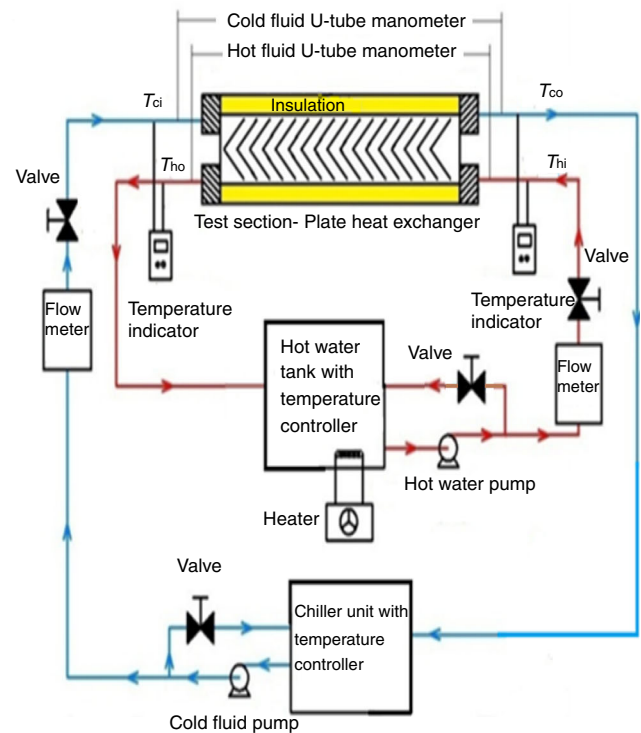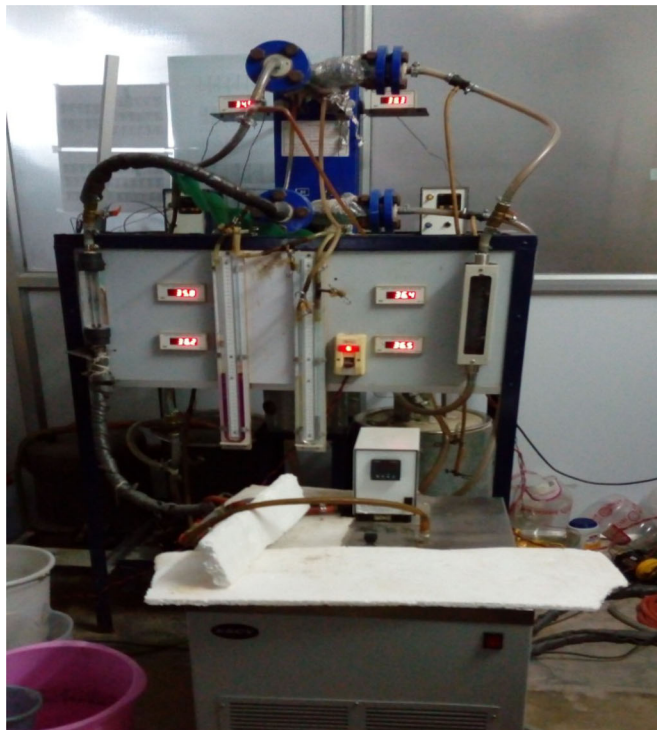
Fig. 1 Photograph and block diagram of the experimental setup

The hybrid nanofluid is stored and cooled in an isothermal bath of 10 L storage capacity to maintain the constant inlet temperature. Then, it goes to the heat exchanger via flowmeter. A control valve is fitted to vary the flow rate of the fluid and a differential manometer to measure the pressure drop of hybrid nanofluid. The hot loop contains an insulated hot water tank of 25 L storage capacity, a float type flowmeter to measure flow rate, a differential manometer to measure the pressure drop of DI water and a hot water pump to circulate the hot DI water. The desired temperature of the hot water inlet is maintained through a temperature controller. Water is stored and heated in the tank and then through hot water pump goes to the heat exchanger via flowmeter. The temperatures of the hybrid nanofluid and hot water streams are measured using thermocouples placed at the inlet and outlet of the streams.

Methodology

Data obtained from the experiments conducted were used in calculating the heat transfer and pressure drop characteristics of each fluid. Reynolds number of both hot and cold fluids can be calculated by

$$Re = \frac{GD_h}{\mu} \tag{2}$$

Channel mass velocity (G) of hot water and cold hybrid nanofluid is given by

$$G = \frac{\dot{m}}{NbL_{\mathrm{w}}} \tag{3}$$

The heat transfer from the hot fluid, Q_{h}, and that from the cold fluid, Q_{c}, are calculated by Eq. (4) using the measured temperature and mass flow rate

$$Q_{\mathrm{h}} = \dot{m}_{\mathrm{h}}c_{\mathrm{ph}}(T_{\mathrm{hi}} - T_{\mathrm{ho}}) \text{ and } Q_{\mathrm{c}} = \dot{m}_{\mathrm{nf}}c_{\mathrm{pnf}}(T_{\mathrm{co}} - T_{\mathrm{ci}}) \tag{4}$$

Due to the difference between both the heat transfer rates, the average heat transfer rate (Q) has been calculated. Based on the experimental data, the overall heat transfer coefficient (U) is calculated from Eq. (5):

$$Q = \frac{UA[(T_{\mathrm{hi}} - T_{\mathrm{co}}) - (T_{\mathrm{ho}} - T_{\mathrm{ci}})]}{\ln\left[\frac{(T_{\mathrm{hi}} - T_{\mathrm{co}})}{(T_{\mathrm{ho}} - T_{\mathrm{ci}})}\right]} \tag{5}$$

The heat transfer coefficient of cold hybrid nanofluid (α_{c}) is obtained from the overall heat transfer coefficient (U) and the heat transfer coefficient of hot DI water (α_{h}):

$$\frac{1}{U} = \frac{1}{\alpha_{\mathrm{h}}} + \frac{1}{\alpha_{\mathrm{c}}} + \frac{t}{k_{\mathrm{w}}} \tag{6}$$

where k_{w} is the thermal conductivity of the plate (W m^{-1} K^{-1}) and t is the thickness of the plate (mm).

The experimental data of overall heat transfer coefficient obtained for the DI water in both sides are compared with theoretical overall heat transfer coefficient obtained from the existing correlation of different authors [11, 21–23]; and shown in Fig. 2.

The experimental data obtained are not in good agreement with any of the correlations. Hence, the new correlation has been established for the Nusselt number of hot fluid. The heat transfer coefficient for the fluid can be calculated by predicting some profile for Nusselt number (Nu) equation that is based on the Reynolds number (Re) and Prandtl number (Pr). We assume that it follows the power law profile

Fig. 2 Comparison between theoretical and experimental U for DI water

$$Nu = aRe^{\mathrm{b}}Pr^{\mathrm{c}} \tag{7}$$

where the Nusselt and Prandtl numbers are defined as follows:

$$Nu = \frac{\alpha D_{\mathrm{h}}}{k} \tag{8}$$

$$Pr = \frac{\mu c_{\mathrm{p}}}{k} \tag{9}$$

Hence,

$$\alpha = \frac{k \cdot a \cdot Re^{\mathrm{b}}Pr^{\mathrm{c}}}{D_{\mathrm{h}}} \tag{10}$$

where α is the heat transfer coefficient (W m^{-2} K^{-1}).

On combining Eqs. (6) and (10), we get Eq. (11)

$$\frac{1}{U} = \frac{1}{\frac{k_{\mathrm{h}} \cdot a \cdot Re_{\mathrm{h}}^{\mathrm{b}}Pr_{\mathrm{h}}^{\mathrm{c}}}{D_{\mathrm{h}}}} + \frac{1}{\frac{k_{\mathrm{c}} \cdot a \cdot Re_{\mathrm{c}}^{\mathrm{b}}Pr_{\mathrm{c}}^{\mathrm{c}}}{D_{\mathrm{h}}}} + \frac{t}{k_{\mathrm{w}}} \tag{11}$$

From available literature for Nu correlations, it was found that the exponent of Re varies from 0.5 to 1.0 and that of Pr varies from 0.3 to 0.5. Taking guess value between these ranges and doing many iterations, we obtained the most suitable values of a, b and c as 0.358, 0.57 and 0.3 from the experimental data. Hence, the proposed correlation in this investigation for DI water to calculate the heat transfer coefficient is given below as Eq. (12):

$$Nu = 0.358Re^{0.57}Pr^{0.3} \quad (R^2 = 0.92) \tag{12}$$

Combining Eqs. (8), (9) and (12), we get the experimental values of heat transfer coefficients for hot fluid. Keeping the value of the hot side heat transfer coefficient in Eq. (6), we obtain the heat transfer coefficient of hybrid nanofluids. From Eq. (8), we have obtained the value of Nu for cold hybrid nanofluid by putting the value of heat transfer coefficient of hybrid nanofluid. This correlation holds good for Reynold number varying between 150 and 350 and Prandtl number varying between 4 and 5. The results obtained are in good agreement with that obtained by Tiwari et al. [24] for base fluid (DI water). The investigation has been done under the derated condition for flow rate ranging from 2 to 4 lpm. So, using a pump curve assuming 30% pump efficiency against the input flow rate [25], coolant pumping power is then calculated by

$$W_{\mathrm{pump}} = \Delta p_{\mathrm{nf}}\Omega_{\mathrm{nf}}/\eta_{\mathrm{pump}} \tag{13}$$

where Ω is volumetric flow rate (m^3 s^{-1}).

For estimating the performance of the system while using hybrid nanofluids, we calculate the non-dimensional parameter known as a performance index. The performance index is used to check the dominance of enhancement in the heat transfer rate and in the pump work and can be
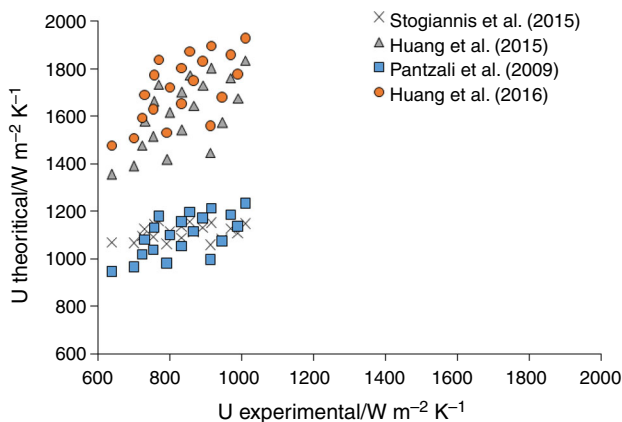
defined as the ratio of heat transfer rate to required pump work.

As the exergy represents maximum obtainable high-grade energy (work) if the system approaches the dead state (ambient) and hence the exergy increases if the temperature increase above ambient or decrease below ambient. Here, a hot fluid inlet temperature is taken as ambient temperature. In this case, cold fluid is losing the exergy and hot fluid is gaining the exergy those are calculated below:

$$\Delta E_{(loss)} = T_e \left[C_{pnf} \ln\left(\frac{T_{nfo}}{T_{nfi}}\right) + \frac{\Omega_{nf}\Delta p_{nf}}{T_{av,nf}} \right] - Q \qquad (14)$$

$$\Delta E_{(gain)} = T_e \left[C_{ph} \ln\left(\frac{T_{hi}}{T_{ho}}\right) - \frac{\Omega_h \Delta p_h}{T_{av,h}} \right] - Q \qquad (15)$$

where $T_{av} = (T_i + T_o)/2$ and T_e = ambient temperature in Kelvin.

Now, the irreversibility rate (I), also known as exergy destruction rate is given by

$$I = \Delta E_{(loss)} - \Delta E_{(gain)}, \qquad (16)$$

Uncertainty analyses

During the experiments, temperatures, flow rates and pressure loss were measured with appropriate instruments. The uncertainties occurred in the measured parameters are presented in Table 3. Considering the relative errors in the individual factors denoted by x_n. The error estimation of

Table 3 Uncertainties during the measurements of the experimental parameters

Variable	Uncertainty value/%
Nanofluid inlet temperature	± 0.2
Nanofluid outlet temperature	± 0.21
Hot outlet temperature	± 0.2
Hot inlet temperature	± 0.21
Nanofluid side mass flow rate	± 2.5
Hot side mass flow rate	± 2.7
Differential pressure	± 2.3
Thermal conductivity	± 1.0
Viscosity	± 1.0
Density	± 1.0
Specific heat	± 1.4
Overall heat transfer coefficient	5.2
Heat transfer coefficient	6.3
Pump work	5.9
Performance index	7.2
Irreversibility	6.5

the dependent parameter has been made using Eq. (17). The total uncertainties (X) found for estimated results are given in Table 3.

$$\frac{\delta X}{X} = \sqrt{\left[\left(\frac{\delta x_1}{x_1}\right)^2 + \left(\frac{\delta x_2}{x_2}\right)^2 + \cdots + \left(\frac{\delta x_n}{x_n}\right)^2 \right]} \qquad (17)$$

Results and discussion

Here investigation is done on a corrugated counter-flow plate heat exchanger with hybrid nanofluid as a coolant and DI water as the hot fluid. Hybrid nanofluid is made by dispersing alumina nanoparticles with other nanoparticles in 4:1 particle ratio (where 80% is alumina nanoparticle and 20% is another nanoparticle) in a base fluid (DI water) at 0.1 vol% concentration. Hot fluid inlet temperature and flow rate are taken as 35 °C and 3 lpm, respectively. Effect of different flow rates and inlet temperatures of different hybrid nanofluids has been investigated. The experiments were conducted many times and based on the result obtained; the error in each performance parameter has been calculated that is shown in the respective figures. Various parameters considered for performance evaluation are heat transfer coefficient ratio, heat transfer coefficient to pressure drop ratio, pump work, performance index and irreversibility. Hybrid nanofluid is represented as a nanoparticle (x: y) where x represents the volume of Al_2O_3 nanoparticle and y represents the volume of another nanoparticle.

Effect of hybrid nanofluid flow rate

Figures 3–7 show the variation of different performance parameters with a coolant flow rate at a coolant inlet temperature of 15 °C. The coolant flow rate is varied from 2 to 4 lpm. In figures, notations 2, 2.5, 3, 3.5 and 4 represent the coolant flow rates in lpm. In addition, Al_2O_3 (5:0) represents DI water-based nanofluid containing 100% alumina nanoparticles. Whereas, SiC (4:1), AlN (4:1), MgO (4:1), MWCNT (4:1) and CuO (4:1) represent DI water-based hybrid nanofluid containing 80% alumina nanoparticles and 20% silicon carbide, aluminum nitride, magnesium oxide, multiwalled carbon nanotube and copper oxide nanoparticles, respectively.

Figure 3 shows that the heat transfer coefficient ratio between hybrid nanofluid and base fluid is increasing with the volumetric flow rate. In the figure, the comparison is shown for different fluids formed by the combination of Al_2O_3 nanoparticles with other nanoparticles (SiC, AlN, MgO, CuO and MWCNT). With the increase in the mass flow rate, the Reynolds number and hence heat transfer
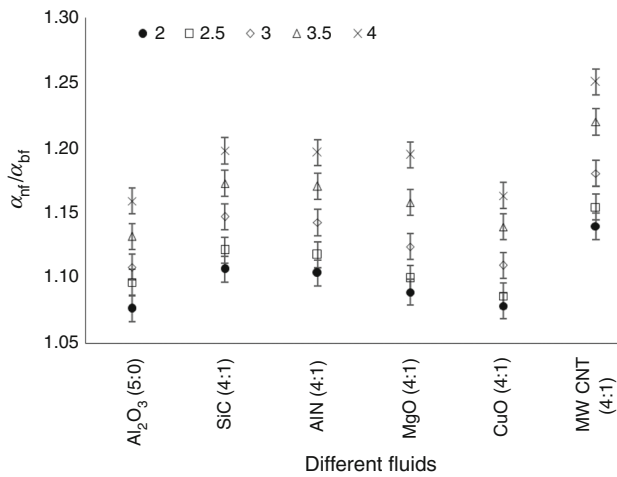
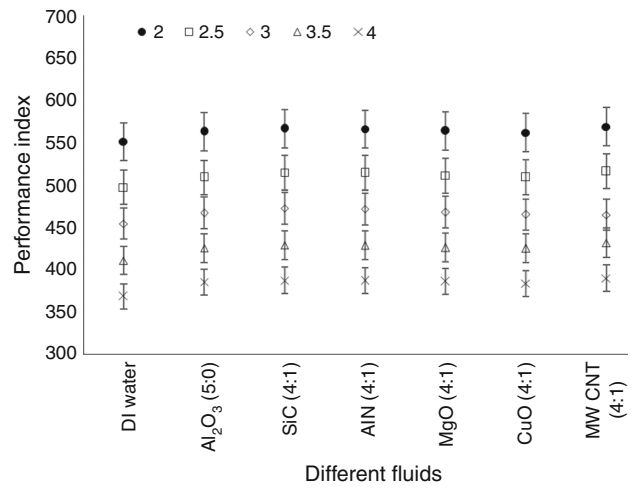**Fig. 3** Variation of heat transfer coefficient ratio with flow rate
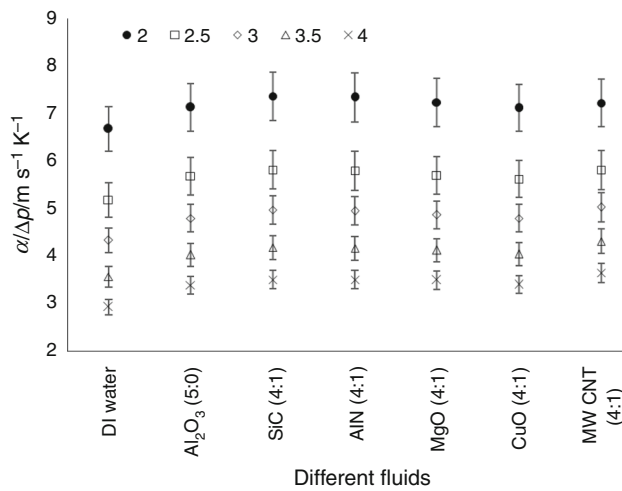

Fig. 4 Variation of heat transfer coefficient to pressure drop ratio with flow rate

Fig. 5 Variation of pump work with flow rate

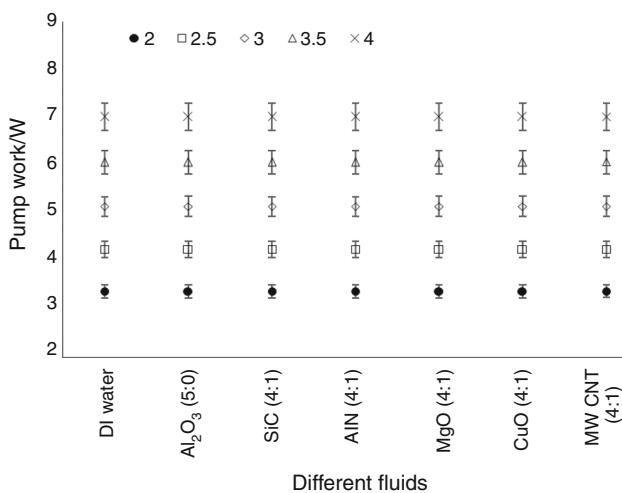**Fig. 6** Variation of performance index with flow rate

Fig. 7 Variation of irreversibility with flow rate

coefficient increase. The enhancement with the addition of nanoparticles is because of the difference in their thermophysical properties like thermal conductivity (play a key role in the heat transfer coefficient enhancement) and a combination of different nanoparticles in a hybrid nanofluid. The relative movement between the nanoparticle and base fluid is the main reason behind this improvement that leads to the circulation of the nanoparticles carrying the heat along and the micro-convection formed by the movement of the fluid around the nanoparticles [26]. The two different particles might attach to each other and form a small cluster, which reduces the thermal resistance between them and enhances the heat transfer phenomenon. The random motion of the particles also contributes to the heat transfer coefficient enhancement of the fluid. A maximum enhancement in heat transfer coefficient of around 25.1% has been observed for MWCNT (4:1) hybrid nanofluids which may be due to the highest thermal

conductivity of MWCNT nanoparticles among the studied nanosized particles. According to heat transfer coefficient enhancement, the hybrid nanofluids are arranged in ascending order as follows: CuO, alumina, magnesium oxide, aluminum nitride, silicon carbide and MWCNT hybrid nanofluids. The order of increment depends on their thermophysical properties like thermal conductivity, density, specific heat, etc.

A negligible enhancement has been observed in the pressure drop (not shown in the figure) and pump work (around 0.075%) by using hybrid nanofluids and increasing the fluid flow rate. By increasing the discharge of fluid, the friction increases, which in turn increases the pressure drop and pump work to maintain the flow in the loop. By suspending the nanoparticles in the base fluid, its viscosity and density change. The formation of nanoparticle clusters in the hybrid nanofluid mixture increases the hydrodynamic diameter of nanoparticles and thus increases the viscosity and density of the fluid. This dual effect of density and viscosity leads to an increase in the pressure drop and pumping power. Al_2O_3/MWCNT (4:1) hybrid nanofluid has a maximum, and Al_2O_3/SiC (4:1) DI water-based hybrid nanofluid shows the minimum pressure drop and pump work (Fig. 5). The reason can be the size and shape of the nanoparticles. All the studied nanoparticles except MWCNT (cylindrical shape having a length in micro-level) nanoparticles are spherical in shape having all the dimensions in nanometers. So, Al_2O_3–MWCNT (4:1)/water hybrid nanofluid has the highest pressure drop and hence highest pump work required. To depict the dominance of heat transfer coefficient over the pressure drop, a ratio of both parameters has been compared for different fluids and shown in Fig. 4. The ratio between heat transfer coefficient and pressure drop was found maximum for MWCNT (4:1) hybrid nanofluid because the relative enhancement in the heat transfer coefficient as compared to pressure drop is maximum for this combination of hybrid nanofluid. Also, with an increase in the flow rate, its value decreases because pressure drop increases more as compared to the heat transfer coefficient with the flow rate increment.

The increase in pump work with the mass flow rate is more as compared to enhancement in heat transfer rate because of which performance index (PI) decreases with the increase in flow rate as can be seen from Fig. 6. Moreover, with the addition of nanoparticles in the base fluid, its heat transfer rate increases more as compared to the pumping power. So, the performance index increases by using hybrid nanofluids. The order of enhancement in the performance index is as follows: CuO, alumina, magnesium oxide, aluminum nitride, silicon carbide and MWCNT hybrid nanofluids. The enhancement in the performance index was found maximum around 5.8% for MWCNT (4:1) hybrid nanofluid. The observed trend is

linear because the changes in the values of pump work and performance index for hybrid nanofluids are very less and the range is more.

The unavailable work during the heat transfer through the thermal systems can be presented in the form of irreversibility rate (I). Figure 7 depicts that the irreversibility of the fluid increases with the coolant flow rate as it depends on mass flow rates and terminal temperatures of fluids. At the lower flow rates, the heat transfer during the fluid flow through the PHE is ruled by the heat conduction. Hence, the value of irreversibility is less. As the flow rate increases, the irreversibility increases due to increases in the effect of irreversibility due to heat transfer and fluid friction [13]. Irreversibility increases with the nanoparticle dispersion in the base fluid. This is due to the fact that the decrease in the heat transfer irreversibility is smaller as compared to the increase in pressure drop irreversibility. Irreversibility was found highest for alumina–CuO (4:1) hybrid nanofluid (8.13%) and least for alumina–MWCNT (4:1) hybrid nanofluid.

Effect of hybrid nanofluid inlet temperature

Figures 8–12 show the variation of different performance parameters with coolant inlet temperature at a constant coolant flow rate of 3 lpm. The coolant inlet temperature is varied from 10 to 25 °C. In figures, notations 10, 15, 20 and 25 represent the coolant inlet temperatures in degree celsius (°C). An increase in the heat transfer coefficient is observed for hybrid nanofluids in comparison to the base fluid as seen from Fig. 8. A maximum enhancement of around 31.2% has been obtained, in the heat transfer coefficient, for the hybrid nanofluid with particle ratio of MWCNT (4:1). Also, the heat transfer coefficient ratio increases with an increase in inlet temperature of coolant
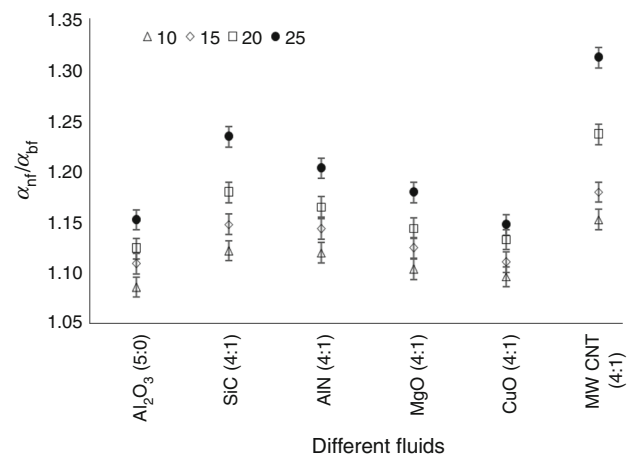
Fig. 8 Variation of heat transfer coefficient ratio with inlet temperature
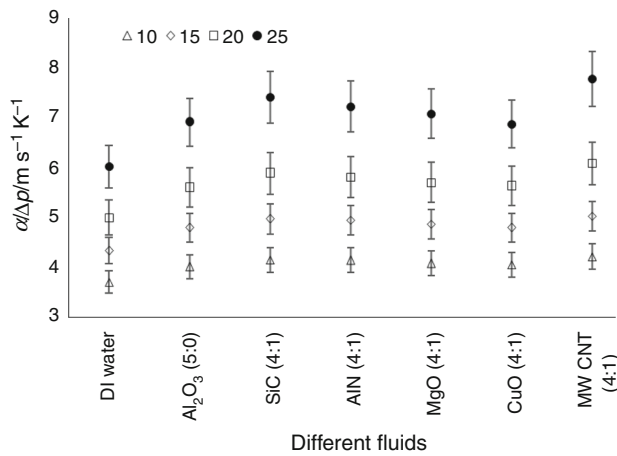
Fig. 9 Variation of heat transfer coefficient to pressure drop ratio with inlet temperature
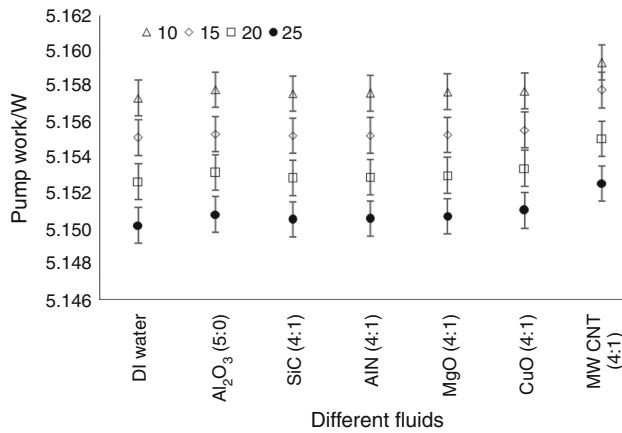

Fig. 10 Variation of pump work with inlet temperature

Fig. 11 Variation of performance index with inlet temperature

because of the rise in mean temperature, which enhances the thermophysical properties and hence gives rise to the heat transfer coefficient. The enhancement for hybrid nanofluid is more as compared to the base fluid. This may be due to an increase in the thermal conductivity of nanoparticles due to Brownian motion and thermophoresis effect [2]. As can be seen from Fig. 9, the ratio of heat transfer coefficient to pressure drop increases with an increase in coolant inlet temperature because with an increase in the temperature, the heat transfer coefficient increases and pressure drop decreases. Also, this ratio enhances with the dispersion of nanoparticles in the base fluid. It was found maximum for MWCNT (4:1) hybrid nanofluid and minimum for CuO (4:1) hybrid nanofluid because the pressure drop decreases more for MWCNT hybrid combination as compared to other studied hybrid combinations.

Figure 10 depicts a decrease in pump work with an increase in the coolant inlet temperature. The reason behind this phenomenon is that with an increase in the
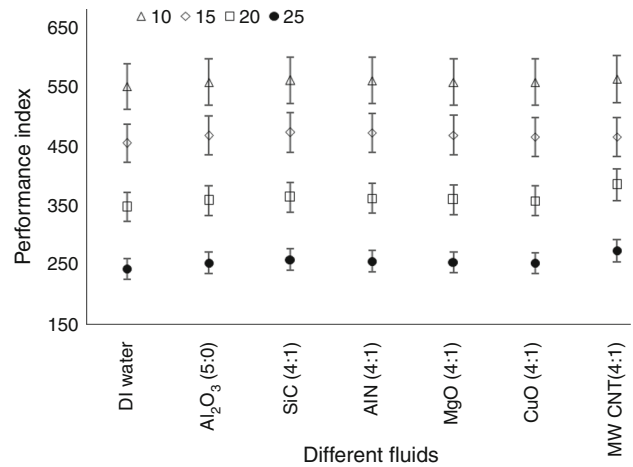
temperature, the density and viscosity of the fluid decrease due to which required pumping power decreases. But with the use of hybrid nanofluids, pump work increases. It is maximum for the fluid containing nanoparticles having a higher mass-to-volume ratio. In the present investigation, pumping power is maximum around 0.08% for hybrid nanofluid with particle ratio of MWCNT (4:1). With an increase in the inlet temperature, both the heat transfer rate and pump work decrease due to a decrease in the temperature difference and pressure drop, respectively. But the decrease in heat transfer rate is comparatively more than that in pump work due to which performance index also decreases with inlet temperature as seen from Fig. 11. Although it increases with hybrid nanofluids and its value is maximum around 12.46% for particle ratio of alumina–MWCNT (4:1) and minimum for particle ratio of alumina–CuO (4:1).

As the inlet temperature increases, the irreversibility decreases because for the constant flow rate irreversibility depends on the terminal temperatures. And with an increase in the inlet temperature of the coolant, the ratio of outlet to inlet temperature decreases, which decreases the irreversibility of fluid, as seen in Fig. 12. Its value is maximum for CuO (4:1) hybrid nanofluid (around 3.1%).

A substantial enhancement in the heat transfer coefficient is obtained by using hybrid nanofluid. A negligible enhancement in the pressure drop was observed due to more viscosity than the water, which was insignificant due to less concentration of the particles. A correlation of Nusselt number has been proposed for DI water Eq. (12). The performance index of hybrid nanofluid is comparatively more because the increase in pump work is less as compared to the rise in heat transfer rate for hybrid nanofluid. Irreversibility increases with the addition of nanoparticles. Al_2O_3–MWCNT (4:1) was found to be most effective, and Al_2O_3–CuO (4:1) was found to be least
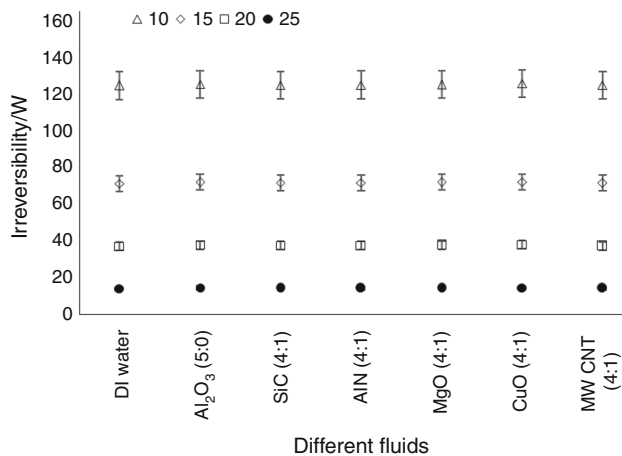
Fig. 12 Variation of irreversibility with inlet temperature

effective hybrid nanofluid among different combinations of hybrid nanofluid studied. Hybrid nanofluids improve energy performance, and hence, they are a good alternative as working fluids in the industrial applications. But because of the high cost of nanoparticles, they are not used in the industrial applications in the present scenario. It has been suggested by Singh and Sarkar [27] that the hybrid nanofluids with lower concentration give payback period earlier and hence suitable at a lower concentration. As the particle concentration increases, the payback period increases. So an effort is required in the area of innovative nanoparticle manufacturing technology to reduce its cost and make nanoparticles in use and improve the thermal performance of plate heat exchanger. The work was carried out, keeping in mind the future scope and technology because hybrid nanofluids are one among the futuristic technologies.

Conclusions

In the present study, the energy and exergy characteristics of different Al_2O_3–DI water hybrid nanofluids flowing in a counter-flow plate heat exchanger have been experimentally investigated. The experiments were conducted for nanoparticle volume concentration of 0.1% at 4:1 mixture ratio, coolant flow rate (2.0, 2.5, 3.0, 3.5 and 4.0 lpm) and the operating temperatures (10, 15, 20 and 25 °C) of the hybrid nanofluid. Based on the results and discussion, the following conclusions can be made:

- Al_2O_3–MWCNT/DI water hybrid nanofluid yields superior performance than other fluids. As the thermal conductivity of MWCNT nanoparticle is much more than other nanoparticles, so the thermal conductivity of the overall mixture increases at the cost of viscosity.

- With an increase in inlet temperature and addition of nanoparticles, the heat transfer coefficient increases due to an enhancement in transport properties. The maximum enhancement of around 31.2% has been observed in the heat transfer coefficient for MWCNT (4:1) hybrid nanofluid.
- Pumping power increases negligibly with the addition of nanoparticles.
- Performance index decreases with an increase in coolant flow rate and inlet temperature. But it increases with the application of hybrid nanofluids up to 12.46% within studied limits.
- Irreversibility increases with flow rate and decreases with inlet temperature as it is directly proportional to flow rate and inversely to inlet temperature.
- Among the different combinations, MWCNT (4:1) is the most effective and CuO (4:1) is the least effective hybrid nanofluid. The cylindrical-shaped nanoparticle gives better performance than a spherical one.
- Hybrid nanofluids can be a better coolant in plate heat exchangers for sub-ambient temperature applications.
- Hybrid nanofluids can be used in industrial applications if its cost can be decreased by improving the techniques of the nanoparticle preparation.

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