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Analysis of gasketed-plate heat exchanger performance using nanofluid

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ABSTRACT

A numerical analysis of the heat transfer and pressure drop of a water-based γ-Al2O3 nanofluid gasketed plate heat exchanger was undertaken to specify its optimum conditions. The results showed that, based on the heat exchanger's performance index, the optimal volume concentration of γ-Al2O3 is approximately 0.016. The heat transfer rate at the optimal concentration of nanofluid is approximately 12.3% higher than that of pure water (base fluid), while the pumping power increased by 1.15%. Increasing ϕ values from ϕ=0.016 to ϕ=0.028 will result in a 1% improvement in the heat transfer rate.

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1. Introduction

Since the invention of gasketed plate heat exchangers (the plate and frame) for use in the food industry in 1930, they have been widely used in various industrial fields. The high efficiency of these heat exchangers is due in part to the increased heat transfer area in proportion to the volume of the media in the heat exchanger and the corrugation of the plates, which causes turbulent flow. Gasketed plate heat exchangers are used in generating hot and cold water for schools, colleges, universities, hospitals, laboratories, offices, government buildings, banks, and leisure and sports facilities. The poor heat transfer properties of the fluids used in industry are obstacles for using different types of heat exchangers. The invention of nanofluid (fluid containing nanometer-sized particles <100 nm) has provided the possibility of overcoming this problem. Bozorgan et al. [1] numerically investigated the use of Al_2O_3/water nanofluid with volume concentrations up to 2% as a coolant in a horizontal double-tube counterflow heat exchanger under turbulent flow conditions. Their results showed that nanofluid offers higher heat performance than water and therefore can reduce the total heat transfer area and coolant flow rate to provide the same heat exchange rate. Bozorgan et al. [2] numerically investigated the use of CuO/water nanofluid as a coolant in the radiator of a Chevrolet Suburban diesel engine under turbulent flow conditions. Their results confirmed that CuO/water nanofluid offers higher overall heat transfer performance than water. Bozorgan et al. [3] have summarized the research on the applications of nanofluids in solar thermal engineering systems in recent years. Their study on theoretical and experimental data for solar systems indicated that the use of nanofluids enhances system performance. Ollivier et al. [4] numerically investigated the possible application of nanofluids in water coolant jackets in a gas spark ignition engine and reported higher thermal diffusivity of nanofluids. The thermal signal variations for knock detection increased by 15% over those predicted for the use of water alone. Khairul et al. [5] examined the effects of water and CuO/water nanofluids (as coolants) on the heat transfer coefficient in a corrugated plate heat exchanger. The heat transfer coefficient increased

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from 18.50% to 27.20% at 0.50% to 1.50% CuO/water concentrations. Zamzamian et al. [6] observed improved heat transfer by using Al_2O_3/ethylene glycol and CuO/ethylene glycol nanofluids compared to ethylene glycol in a plate heat exchanger. Pandey and Nema [7] experimentally examined Al_2O_3/water nanofluid as a coolant in a corrugated plate heat exchanger and found that the heat transfer performance of the heat exchanger decreased with increasing nanoparticle concentrations. Pantzali et al. [8] experimentally investigated the role of CuO/water nanofluid with a volume concentration of 4% as a coolant for two flow types (laminar and turbulent) in plate heat exchangers and found that the use of nanofluid is useful only in a laminar flow. Kwon et al. [9] experimentally investigated the heat transfer characteristics of water-based ZnO and Al_2O_3 nanofluids in a plate heat exchanger but failed to prove the efficiency of the nanofluids. The authors stated that this phenomenon is related to the plate structure, concluding that an increase in viscosity suppresses the convective heat transfer coefficient and that an increase in thermal conductivity cannot make up for it when analyzing the fluid's moving characteristics in the plate. Most studies to date have been limited to investigating the use of nanofluids as coolants in heat exchanging devices. Thus far, few studies have been done on the heat transfer characteristics of nanofluids as the hot stream. Haghshenasfard [10] experimentally studied ZnO/water (0.5% volume fraction) nanofluid as the hot stream at a constant mass flow rate in plate and concentric tube heat exchangers and found a greater heat transfer coefficient for nanofluid compared to water. Based on the comprehensive literature review, it can be said that the effect of using nanofluid on heat transfer performance in plate heat exchangers is not clear. In this paper, the convective heat transfer and pressure drop of γ-Al_2O_3/water nanofluid in a gasketed plate heat exchanger is numerically investigated for a wide range of particle concentrations (0%–6%). The thermo-physical properties of γ-Al2O3/water nanofluid are calculated using well-known empirical correlations.

2.Methodology

Here, we investigate the heat transfer and energy performance of a gasketed plate heat exchanger using water-based γ-Al_2O_3 nanofluid where cold water is heated by nanofluid. Fig. 1 shows the structure of the plate heat exchanger, and the detailed specifications are shown in Table 1. Cold water with a flow rate of 140 kg/s enters the gasketed plate heat exchanger at 22 °C and is heated to 42 °C. The γ-Al_2O_3/water nanofluid has the same flow rate, entering at 65 °C and leaving at 45 °C. The calculation in this analysis has been divided into three sections comprising the nanofluid; the cold water; and the heat transfer performance of the gasketed plate heat exchanger. The following assumptions have been made:

i. The flow is incompressible, steady-state, and turbulent.
ii. The effect of body force is neglected.
iii. Heat transfer with the environment is negligible.

2.1 Thermo-physical properties of nanofluid

In this study, the thermal properties of γ-Al_2O_3/water nanofluid are determined by employing well-known empirical correlations. The thermos-physical properties of γ-Al_2O_3 nanoparticles and base fluid (water) are tabulated in Table 2.

The density of the nanofluid is calculated as follows:

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_p , \tag{1}$$

where ρ_p and ρ_{bf} are the densities of the nanoparticles and the base fluid, respectively, and ϕ is the volume concentration of nanoparticles [11].

Specific heat is calculated as follows:

$$c_{p,nf} = \frac{(1-\phi)\rho_{bf}c_{p,bf} + \phi\rho_p c_{p,p}}{\rho_{nf}} , \tag{2}$$

where $c_{p,p}$ and $c_{p,bf}$ represent the specific heat of the nanoparticles and the base fluid, respectively [12].
Thermal conductivity is calculated as follows:

$$\frac{k_{nf}}{k_{bf}} = 1 + 4.4Re^{0.4}Pr_{bf}^{0.66}\left(\frac{T}{T_{fr}}\right)^{10}\left(\frac{k_p}{k_{bf}}\right)^{0.03}\phi^{0.66} , \tag{3}$$

1. where k_{bf} is the thermal conductivity of the base fluid, Re is the nanoparticle Reynolds number, Pr_{bf} is the Prandtl number of the base fluid, T is the nanofluid temperature, T_{fr} is the freezing point of the base fluid, and k_p is the thermal conductivity of the nanoparticles.
Dynamic viscosity is calculated as follows:

$$\mu_{nf} = \frac{\mu_{bf}}{1-34.87(d_p/d_{bf})^{-0.3}\phi^{1.03}} \tag{4}$$

where μ_{bf} is the dynamic viscosity of the base fluid, d_p is the diameter of the nanoparticles, and d_{bf} is the equivalent diameter of a base fluid molecule, which can be calculated as follows [13]:

$$d_{bf} = 0.1\left(\frac{6M}{N\pi\rho_{bf0}}\right)^{1/3} , \tag{5}$$

where M and N are the molecular weight of the base fluid and the Avogadro number (6.022×10^{23} mol^{-1}), respectively, and ρ_{bf0} is the mass density of the base fluid.

The Reynolds number of the suspended nanoparticles can be calculated as follows [13]:

2. $$Re = \frac{2\rho_{bf}k_bT}{\pi\mu_{bf}^2 d_p} , \tag{6}$$

where k_b=1.38066×10^{-23} J/K is the Boltzmann constant.

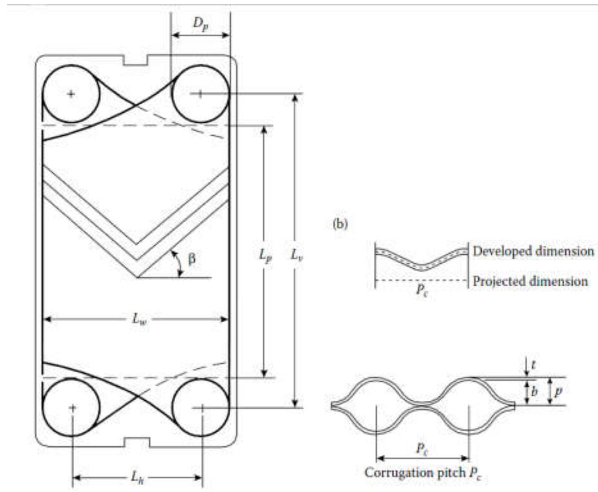

Fig. 1 Simple schematic of a plate heat exchanger [14]

Table 1. Geometric characteristics of the plate heat exchanger

Property	Water (hot stream)	Water (cold stream)	γ-Al2O3
cp [J kg-1K-1]	4183	4178	880
ρ [kg m-3]	985	995	3700
k [Wm-1K-1]	0.645	0.617	46
μ[kg m-1 s-1]	5.09×10-4	7.66×10-4	

Table 2. Thermo-physical properties of water and γ-Al_2O_3 nanoparticles

Plate thickness (*t*)	0.6 mm
Chevron angle (*β*)	45°
Total number of plates (N_t)	105
Enlargement factor (*ϕ*)	1.25
Number of passes	One pass/one pass
Total effective area (A_{real})	110 m^2
All port diameter (D_p)	200 mm
Effective channel width (L_w)	0.63m
Vertical port distance (L_v)	1.55 m
Horizontal port distance (L_h)	0.43 m
Compressed plate pack length (L)	0.38 m
Thermal conductivity of the plate material (k_w)	17.5 W/m.K

Table 3. Summary of the numerical results

ϕ	k_r	h_r	μ[kg m^{-1} s^{-1}]	h_{nf} (W/m^2K)	U (W/m^2K)	*PP* (W)	*Re*
0	1	1	5.09×10^{-4}	7907.097	5035.490	2323.29	13377.44
0.01	1.3415	1.4716	5.7×10^{-4}	11636.134	6336.310	2330.36	11892.50
0.02	1.5396	1.6894	6.6×10^{-4}	13358.954	6818.178	2368.36	10345.17
0.03	1.7052	1.8087	7.76×10^{-4}	14302.211	7056.893	2433.96	8773.37
0.04	1.8526	1.8435	9.47×10^{-4}	14576.821	7122.774	2536.87	7185.50
0.05	1.9879	1.7913	1.22×10^{-3}	14164.245	7021.085	2698.08	5585.50
0.06	2.1142	1.6375	1.7×10^{-3}	12948.02	6705.530	2966.35	3975.82

2.2 Water heat transfer

The heat transfer coefficient of the cold water under a turbulent regime can be calculated as follows [14]:

$$h_c = \frac{Ck_c}{D_h}\mathrm{Re}_c^N \, \mathrm{Pr}_c^{\frac{1}{3}}\left(\frac{\mu_{nf}}{\mu_{wnf}}\right)^{0.17}, \qquad (7)$$

where c and nf denote the relevant parameters of the cold water and of the nanofluid as the hot fluid. C and N are constants for single-phase heat transfer in gasketed plate heat exchangers and are 0.3 and 0.663 for Re_c>100 and β=45° [14]. $(\mu_{nf}/\mu_{wnf})^{0.17}$ is the viscosity correction factor and D_h is the hydraulic diameter of the channel, which is expressed in the following form:

$$D_h = \frac{4\times \text{flow area}}{\text{wetted perimeter}} = \frac{2b}{\phi_a}, \qquad (8)$$

where b and ϕ_a are the channel depth and the multiplication factor. According to Fig. 1, the channel depth (b) is equal to the thickness of the corrugated plate minus the thickness of the metal sheet (b=p-t). Because the plates are in contact with each other, the thickness of the corrugated plates or the plate pitch (p) can be obtained by dividing the length of the plate pack by the number of plates ($p=L/N_t$).

ϕ_a is a multiplication factor representing the enhancement of the heat transfer area due to the corrugations and can be obtained from:

$$\phi_a = \frac{(A_{real}/N_c)}{L_p \times L_w}, \qquad (9)$$

where A_{real} is the total effective area, N_c is the effective number of plates equal to N_t-2, and $L_p \times L_w$ is the project plate area, as can be seen in Fig. 1.

The Reynolds and Prandtl numbers in (7) are calculated considering the cold water properties as follows:

$$\mathrm{Re}_c = \left(\frac{\dot{m}_c/N_{cp}}{A_{ch}}\right)\frac{D_h}{\mu_c} \qquad (10)$$

$$\mathrm{Pr}_c = \frac{c_{p,c}\mu_c}{k_c}, \qquad (11)$$

where $\dot{m}_c/N_{cp}$ is the cold water mass flow rate per channel, and A_{ch} is the one-channel flow area equal to $b\times L_w$. The number of channels per pass can be calculated as follows:

$$N_{cp} = \frac{N_t - 1}{2N_p}, \qquad (12)$$

where N_p is the number of passes.

2.3 Nanofluid heat transfer

(a) The heat transfer coefficient of the nanofluid as the hot fluid can be calculated based on the formula in Li and Xuan [15]:

$$\frac{h_{nf}D_h}{k_{nf}} = Nu_{nf} = 0.0059\left(1.0 + 7.6286\phi^{0.6886}Pe_d^{0.001}\right)$$
$$\times \mathrm{Re}_{nf}^{0.9238}\,\mathrm{Pr}_{nf}^{0.4}\left(\frac{\mu_{nf}}{\mu_{wnf}}\right)^{0.17} \qquad (13)$$

where Pe_d is the nanofluid's Peclet number and is defined in the following form:

$$Pe_d = \frac{u_{nf}d_p}{\alpha_{nf}}, \qquad (14)$$

where dp is the diameter of the nanoparticles and α_{nf} is the nanofluid's thermal diffusivity that is defined as follows:

$$\alpha_{nf} = \frac{k_{nf}}{\rho_{nf}c_{p,nf}}. \qquad (15)$$

The Reynolds and Prandtl numbers in equation (13) are calculated considering the nanofluid's properties as follows:

$$\mathrm{Re}_{nf} = \frac{(\dot{m}_{nf}/N_{cp})D_h}{A_{ch}\mu_{nf}} \qquad (16)$$

$$\mathrm{Pr}_{nf} = \frac{c_{p,nf}\mu_{nf}}{k_{nf}}, \qquad (17)$$

where $\dot{m}_{nf}/N_{cp}$ is the nanofluid mass flow rate per channel.

(b) The friction factor of γ-Al_2O_3/water nanofluid can be calculated using the following formula [16]:

$$f_{nf} = 0.316\,\mathrm{Re}_{nf}^{-0.25}\left(\frac{\rho_{nf}}{\rho_{bf}}\right)^{0.797}\left(\frac{\mu_{nf}}{\mu_{bf}}\right)^{0.108}, \qquad (18)$$

where

$$4000 < Re_{nf} < 16{,}000$$
$$0 \le \phi \le 10\% \qquad (19)$$

(c) The pressure drop (Δp_{nf}) and pumping power (PP) for the Al_2O_3/water nanofluid used as a coolant in a double-tube heat exchanger are calculated as follows [14]:

$$\Delta p_{nf} = 2\frac{f_{nf}L_v N_p}{D_h}\frac{G_{nf}^2}{\rho_{nf}}\left(\frac{\mu_{nf}}{\mu_{wnf}}\right)^{-0.17} \text{ and} \qquad (20)$$

$$PP = (\dot{m}_{nf}/\rho_{nf})\times \Delta P_{nf}, \qquad (21)$$

where G_{nf} is the mass velocity of the nanofluid and is expressed in the following form:

$$G_{nf} = \frac{\dot{m}_{nf}}{N_{cp}bL_w}. \qquad (22)$$

2.4 Total heat transfer

(a) Knowing h_c and h_{nf}, the total heat transfer coefficient can be calculated as follows:

$$U = \left(\frac{1}{h_c} + \frac{1}{h_{nf}} + \frac{t}{k_w} \right)^{-1}, \tag{23}$$

where t and k_w are the plate thickness and the thermal conductivity of the plate material, respectively.

(b) In this work, the calculated area, A_{calc}, is computed from the following equation [17]:

$$A_{calc} = \frac{q}{U \times F \times LMTD} \tag{24}$$

$$LMTD = \frac{(T_1 - t_1) - (T_2 - t_2)}{ln \dfrac{(T_1 - t_1)}{(T_2 - t_2)}} \tag{25}$$

$$q = \varepsilon C_{min} (T_{nf,i} - T_{c,i}) \tag{26}$$

$$\varepsilon = 1 - \exp[(\frac{1}{C^*})(NTU)^{0.22} \{\exp[-C^*(NTU)^{0.78}] - 1\}] \tag{27}$$

$$C^* = \frac{C_{min} = (\dot{m}_{nf} C_{p,nf})}{C_{\max} = (\dot{m}_h C_{p,h})}; NTU = \frac{UA_{real}}{C_{\min}}, \tag{28}$$

where q is the heat transfer rate, F is the temperature correction factor that is assumed to be 0.96 (for N_t>40 and NTU<1 [14]), ε is the heat exchanger effectiveness, and NTU is the number of heat transfer units.

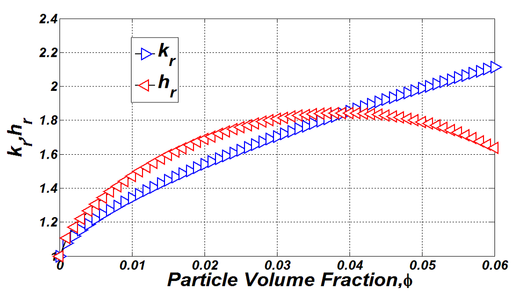

Fig. 2 Comparison between k_r and h_r for γ-Al_2O_3/water nanofluid

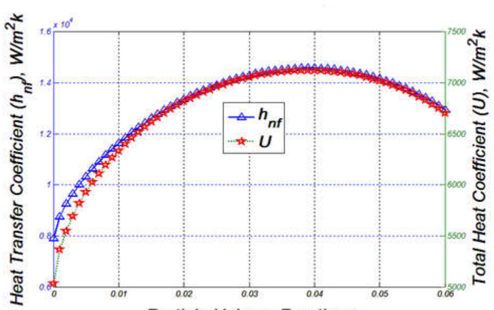

Fig. 3 Convection coefficient and total heat transfer coefficient for γ-Al_2O_3/water nanofluid at various concentrations

3. Results and discussion

The results are reported in terms of the relative conductivity k_r (k_{nf}/k_{bf}), the relative convection coefficient h_r (h_{nf}/h_{bf}), the nanofluid convection coefficient h_{nf}, the overall heat transfer coefficient U, the total heat transfer rate q, the nanofluid pressure drop ΔP_{nf}, and the pump power PP as a function of volume concentration ϕ. As mentioned previously, the Corcione model has been applied to predict the thermal conductivity of the nanofluid. In all cases, the particle size is 11 nm. Fig. 2 and Table 3 show the k_r and h_r of the γ-Al_2O_3/water nanofluid at various concentrations (0–6%). The present results are similar to those found by Esfe et al. [18] and Jwo et al. [19]. They showed the heat transfer coefficient ratio (h_r) of 1.36 for a 1.0% concentration of MgO nanoparticles in water at Re=7331. Our numerical results show that h_r=1.47 for a 1.0% concentration of γ-Al_2O_3 nanoparticles in water at Re=11892.50 (Table 3). The improvement of heat transfer by nanofluids may be the result of the following aspects: (i) nanoparticles have higher thermal conductivity, so a higher concentration of nanoparticles resulted in a more obvious heat transfer improvement. (ii) Nanoparticles collided with the base fluid molecules and the wall of the heat exchanger, thus strengthening energy transmission. (iii) The nanofluid increased friction between the fluid and the wall, improving heat exchange.

Increasing particle concentrations increase the fluid viscosity, decrease the Reynolds number, and consequently decrease the heat transfer coefficient (Table 3). As can be seen in Fig. 2, increasing particle concentrations increase the h_r ratio up to ϕ=0.04. Beyond this concentration level, the h_r ratio is less than the k_r ratio. The present results are similar to the observations of Lelea et al. [20], who reported that the Al_2O_3/water nanofluid with $\phi = 3\%$ has a lower heat transfer coefficient compared to $\phi = 1.33\%$ and 2%.

As seen in Fig. 3, the total heat transfer coefficient shows a consistent trend with the heat transfer coefficient. The present results are similar to the observations of Jwo et al. [21], who experimentally confirmed that nanofluid has a better total heat transfer performance than the base fluid.

As can be seen in Fig. 4, the heat transfer rate is calculated using Equation (26) by computing U, NTU, C^*, and ε for γ-Al_2O_3/water nanofluid at various concentrations. The results show that the best volume fraction for the maximum heat transfer rate is equal to ϕ=0.028.

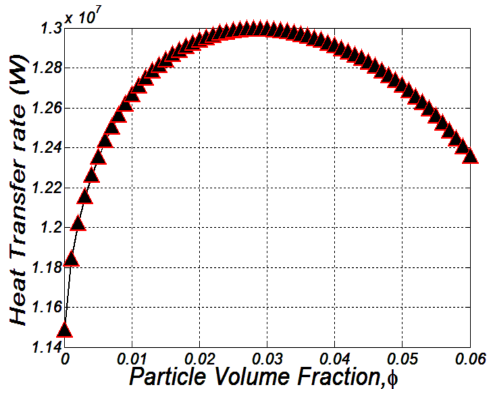

Fig. 4 Heat transfer rate for nanofluid at different particle concentrations

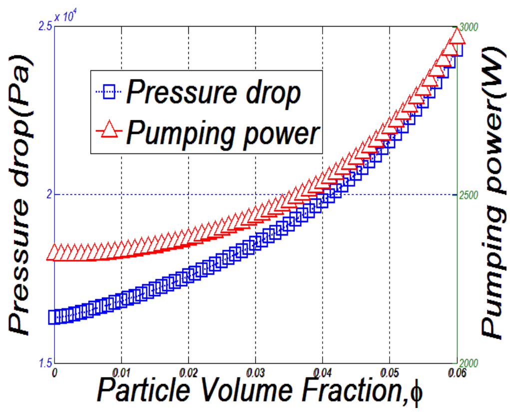

Fig. 5 Influence of γ-Al_2O_3 volume fraction on the pumping power and pressure drop

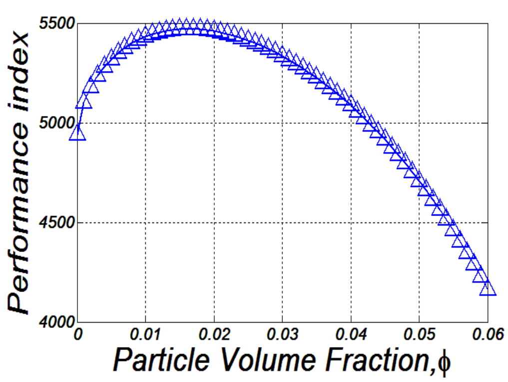

Fig. 6 Variation of the performance index with particle volume fraction

The nanofluid viscosity is an important parameter for practical applications because it directly affects the pressure drop. The pressure drop of the nanofluid in heat exchangers is one of the central parameters determining the efficiency of the application of nanofluids. The pressure drop and pumping power are closely related. Fig. 5 clearly shows that the pressure drop of γ-Al_2O_3/water nanofluid increases with increased volume concentration. This may be because the density and viscosity are the main thermo-physical parameters that could influence the pressure drop and pumping power.

In this study, the ratio of the heat transfer rate and pumping power is defined as the performance index [22]:

$$\eta = \frac{q}{PP} \tag{29}$$

Fig. 6 shows that the optimum concentration for the maximum performance index is ϕ=0.016. A further inspection of Fig. s 4 and 6 shows that the optimum concentration for the maximum performance index is lower than that for maximum heat transfer. This observation is consistent with the experimental results presented by Tiwari et al. [23].

The optimum concentration for the maximum performance index is selected to be 0.016. The heat transfer rate at 0.016 volume concentration is approximately 12.3% higher than that of pure water (base fluid), while the pumping power is increased by 1.15%.

As mentioned previously, the present results are in good agreement with the results of several other works [18-23]. To validate the numerical code, the calculated area (A_{calc}) is compared with the total effective area of the gasketed plate heat exchanger (A_{real}) for pure water as the hot fluid. The difference between A_{calc} obtained by the code and A_{real} is acceptable at approximately 8% (A_{calc}=101.52 and A_{real}=110 m^2).

4. Conclusions

Based on the analysis, the following conclusions can be drawn:

(1) Adding Al_2O_3 nanoparticles to the water increases the heat transfer coefficient up to a certain level. Therefore, there is an optimal volume concentration for the nanofluid to improve the heat transfer rate in the heat exchanger.

(2) The results show that the best volume fraction for the maximum heat transfer rate is equal to ϕ=0.028.

(3) The optimum concentration for the maximum performance index is ϕ=0.016.

(5) The heat transfer rate of the nanofluid at the optimal concentration is approximately 12.3% higher than that of pure water (base fluid), while the pumping power increased by 1.15%.

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Nomenclature

A : total heat transfer area, m^2
A_{ch} : heat transfer area of one channel, m^2
b: channel depth, m
c_p: specific heat, J/kg K
D_h: hydraulic diameter, m
D_p: port diameter, m
d_{bf}: equivalent diameter of a base fluid molecule, m
d_p: nanoparticle diameter, m
F : *LMTD* correction factor, m
f: friction factor
G: mass velocity, $kgm^{-2}s^{-1}$
h: heat transfer coefficient, W/m^2 K
k: thermal conductivity, W/m K
LMTD : logarithm mean temperature difference
L: compressed plate pack length, m
L_h : horizontal port distance, m
L_v: vertical port distance, m
L_w: effective channel width, m
M: molecular weight of the base fluid, kg mol^{-1}
$\dot{m}$: mass flow rate, kg/s
N_c : effective number of plates
N_{cp} : number of channels per pass
N_p : number of passes
N_t : total number of plates
Nu : Nusselt number
NTU: number of heat transfer units
Δp : pressure drop, Pa
Pe_d: Peclet number
Pr : Prandtl number
PP : pumping power, W
P : plate pitch, m
q: heat flow, W
Re: Reynolds number
T: temperature, °C
T_{fr}: freezing point of the base fluid, °C
T_1,T_2 : inlet and outlet temperatures of hot fluid, °C
t_1,t_2 : inlet and outlet temperatures of cold fluid, °C
t : plate thickness, m
U : total heat transfer coefficient, W/m^2K

Greek letters

B : chevron angle, deg
ρ : density, kg/m^3
ϕ: volume concentration
ϕ : multiplication factor
μ : viscosity, kg/ms
α : thermal diffusivity, m^2/s
ε : heat exchanger effectiveness

Subscripts

bf : base fluid
c: cold fluid
nf : nanofluid
p : particles
w : wall

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