PHE Heat Transfer Performance Using 29nm CuO-Water Nanofluid

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Abstract: - An experimental investigation was performed to study the forced convective heat transfer and hydraulic characteristics of a chevron-type two-channel plate heat exchanger (PHE) using a water based nanofluid. The PHE considered, composed of two fluid passages, is formed by the assembly of three corrugated plates. These plates have herringbone pattern- trapezoidal-shape corrugations. Heated water is used on the hot side and a mixture of 29nm-diameter CuO-nanoparticles in suspension in water is used on the cold side. Experimental data for the nanofluid side cover two particle volume fractions (2%, 4.65%) and the range of the Reynolds number up to ~1000. Hydraulic results have shown that, for a given value of Re, 4.65% CuO-water nanofluid clearly produces higher friction factor when compared to water and the 2% nanofluid. Heat transfer results show an enhancement of the convective coefficient when using the 2% volume fraction nanofluid, but a clear decrease of heat transfer for the 4.65% nanofluid. An earlier laminar-turbulent transition was found for the 4.65% nanofluid, while similar behavior was observed for water and the 2% nanofluid.

Key-Words: - Plate heat exchanger, chevron-type plate heat exchanger, CuO-water nanofluid, friction factor, heat transfer coefficient, laminar-turbulent transition

1 Introduction

Nanofluid, a mixture of a liquid and dispersed nanometre-size particles, seems to be an interesting alternative for the heat transfer enhancement purpose, as found for internal flow configurations [1-4] as well as for cooling of high heat output electronic components [5-7]. Plate heat exchangers (PHE), on the other hand, have widely been used in various thermal applications because of their superior features (high heat transfer surface/volume ratio, high thermal performance, reduced size and weight). The hydraulic and thermal characteristics of PHEs have received a great attention from researchers, both analytically and numerically [8-16] and experimentally [17-21] (see Kakac and Liu [22], Wang et al. [23] for a review and a complete presentation and discussion of the results/empirical correlations related to the PHEs). Relevant works studied the nanofluid use in a PHE still remain scarce. Pantzali et al. [24] studied numerically and experimentally the heat transfer performance of CuO-water nanofluid in a miniature PHE. They have found that for a given heat duty, the nanofluid volumetric flow rate required is lower than that for water, thus requiring lower pumping powers. However, in another study, they have found that a substitution of conventional fluids by CuO-water nanofluid appears inauspicious [25]. Maré et al. [26] studied two different nanofluids in a PHE for the low temperature range (from 0 to 10°C) and found a considerable heat transfer enhancement. Zamzamian et al. [27] recently found that an important heat transfer enhancement can be obtained using nanofluid turbulent flows in double pipe and plate heat exchangers.

In this work, we experimentally studied the hydraulic characteristics and heat transfer performance of a two-channel chevron-type PHE using a CuO-water nanofluid. Results are obtained for the friction factor and the surface heat transfer coefficient as function of the channel flow rate for both laminar and turbulent regimes. The laminar-turbulent transition for the nanofluid is also studied and compared to that of water. The main objective of our study is to assess the advantages, if any, of using a nanofluid in a plate heat exchanger.

2 Nomenclature

\[ A \] heat-transfer area of the plate, m\(^2\)
\[ b \] gap between two consecutive plates, m
\[ C_{\text{min}} \] minimum heat capacity rate, W K\(^{-1}\)
\[ C_{\text{Nu}} \] constant
\[ C_p \] specific heat, J kg\(^{-1}\) K\(^{-1}\)
3 Experimental setup and procedures

Fig. 1 shows the HX plate with a herringbone pattern; details of trapezoidal-shape corrugations are shown in Fig. 2. The two fluid channels are formed by three plates (two end-plates and an intermediate one). The PHE is of. The length, width and height of the channels are, respectively, aligned along the X, Z and Y axis (X=0 and Z=0 correspond to the two main symmetry axes of the plates). The inlets and exit ports for each fluid are located on the same side of the plate (‘the side-flow configuration’). The inlets are located on opposite sides at the extremities of one of the plate’s diagonals (Fig. 1). During the tests, the PHE plates are held vertical with the cold fluid flowing upwards and the hot fluid flowing downwards (the counter-flow type). The plate dimensions are, respectively, \(D_p=55.4\) mm, \(L_{h1}=170\) mm, \(L_{h2}=114\) mm, \(L_{v1}=390\) mm, \(L_{v2}=210\) mm; the herringbone angle is \(\beta=60^\circ\) or \(\beta=30^\circ\) (Fig. 1).

![Fig. 1 Views of the PHE plates used (left: view, right: main dimensions)](image)

![Fig. 2 Shape and dimensions of corrugations](image)
air-forced car radiator (CR) and an in-house counter-current double pipe heat exchanger (DPHE) in which city cold tap water is used as cooling fluid. In order to monitor the flow rates, a precision volumetric water meter VWM₁ is used for the hot side; on the cold one, an accurate magnetic flow meter VWM₂ is used. The circulation of fluids is realized using two volumetric gear-pumps, P₁ and P₂; in order to adjust the flow rates, both pumps are equipped with a three-way valve deviation-circuit. To monitor the pressure loss between the inlet and outlet of each fluid stream of the PHE, two differential pressure transducers (ΔP₁ and ΔP₂) are used. The fluid stream inlet outlet temperatures are measured with type T-thermocouples (T). In addition, 40 other thermocouples are clued on the exterior sides of each of the end-plates. All these 84 thermocouples are connected to a system of acquisition cards (AC).

Figure 4 shows the layout of 40 thermocouples that are clued onto the outer surface of each end-plate. The thermocouple positions are symmetrical with respect to both the plate main axis X and Z.

3.1 Experimental uncertainties

All thermocouples are of T-type with a good accuracy (±0.5°C at 0°C; ±0.65°C at 100°C); they were calibrated using a temperature-controlled bath. The data monitoring system consists of a 24 bit analog/digital converter NI-9213 acquisition boards (National Instruments). The differential pressure transducers (Omega) are used for measuring the pressures; they both are bidirectional and possess an accuracy of ±0.03% within the 0-2500Pa range. The volumetric flow rate meter (Actaris, Model TD8) is used for the hot fluid stream; while a magnetic flow meter (Badger Meter, Model M2000) is used for the cold stream. These meters possess a very good accuracy: for the range of the considered volumetric flow rate (4.8x10⁻⁶ to 7.1x10⁻⁵ m³/s or 0.29 to 4.31/min- the maximum error is less than 1% according to the manufacturer’s calibration data. The PHE heat balance was consistently verified for every experiment; the maximum difference in the heats exchanged between two streams was found less than 5.5% (1.9% on the average basis). Such difference, quite acceptable in conjunction with all experimental uncertainties, is believed due to heat losses from various piping parts. Using a standard methodology [28] for the error calculations, the maximum and average uncertainties are estimated to be ±6.6% and ±6.4% for the friction factor, and ±13.1% and ±7.0% for the Nusselt number.

3.2 Thermal properties of fluids

All properties of fluids (water, nanofluid) are temperature-dependent. The following formulas used for computing water thermal properties are obtained by curve-fitting of data published in [29]:

\[ \rho = 1.5286 \times 10^{-5} T^3 - 1.83074 \times 10^{-2} T^2 + 6.58954 T + 254.588 \]  

\[ C_p = -3.62543 \times 10^{-8} T^5 + 6.10278 \times 10^{-5} T^4 - 4.10544 \times 10^{-2} T^3 + 13.8049 T^2 - 2.32127 \times 10^3 T + 1.60363 \times 10^5 \]  

\[ \mu = -2.305729118 \times 10^{-9} T^3 + 2.40925548 \times 10^{-6} T^2 - 8.435600282 \times 10^{-4} T + 9.935042382 \times 10^{-2} \]  

\[ k = -9.772372107 \times 10^{-6} T^2 + 7.50623497 \times 10^{-3} T - 7.618478949 \]  

where T is fluid temperature in degrees K.

The plate material is stainless steel having properties: ρ = 8030 kg/m³, C_p = 502.48 J/kg·K and k=16.27 W/m·K.

The nanofluid used, a mixture of 29nm CuO particles in suspension in water, was purchased from a commercial source [30]. The following formulas are used for computing nanofluid density and specific heat (subscripts ‘nf’, ‘bf’ and ‘p’ respectively refer to nanofluid, base fluid (water) and nanoparticles):

\[ \rho_{nf} = (1 - \Phi)\rho_{bf} + \Phi \rho_p \]  

\[ C_{pnf} = C_{pbf} (1 - \Phi) + \Phi C_{pp} \]  

\[ \mu_{nf} = \mu_{bf} (1 - \Phi) + \Phi \mu_p \]  

\[ k_{nf} = k_{bf} (1 - \Phi) + \Phi k_p \]
(\rho \cdot C_p)^nf = (1 - \Phi) \cdot (\rho \cdot C_p)^f + \Phi(\rho \cdot C_p)_p \quad (6)

CuO particles have properties: density \( \rho = 6500 \) kg/m³ and specific heat \( C_p = 554 \) J/kg·K\(^{[30]} \).

Due to the nanofluid specificities, its thermal conductivity and viscosity are estimated using in-house measured data \([31-33]\). It is interesting to note that as surfactants are reused\([30]\), the nanofluid stability and quality of particle suspension are quite satisfactory as no particular problem is encountered so far for all particle concentrations studied. Such a point is clearly ascertained by the consistency of results from repeated test cases.

3.3 Experimental procedures, governing parameters and data processing

For every test, the same procedure is used. The fluid volumetric flow rate is first calculated by determining the time required to record a certain volume of fluid using readings of meters. Such a step is repeated several times during the test, which gives the average mass flow rate and then the errors is estimated (all less than 2%). The flow rates is adjusted using the three-way valve by-pass systems.

A typical test requires approximately two hours for the system to reach the steady-state conditions, which are ascertained from readings of monitored temperatures. Once the thermal equilibrium conditions obtained, then the fluid flow rates and temperature readings are recorded. The heat flux of the water heater is varied by adjusting the tension applied to the heating elements. Voltage and current are monitored and used for computing the applied electric power in the water heater.

The main parameters are the channel Reynolds number, the friction factor and the Nusselt number, defined as follows:

\[
Re = \frac{\mu m D_h(\rho m)}{\mu m} \quad (7)
\]

\[
f = 2 \frac{\Delta P D_h(\rho L u_m^2)}{\mu m} \quad (8)
\]

\[
Nu = \frac{\mu m}{D_h} \quad (9)
\]

All fluid properties are computed at the mean fluid temperature in the channel; \( D_h \) is the channel hydraulic diameter (double of the gap between plates \( D_h=5 \) mm); \( u_m \) is the average velocity in the channel; \( \Delta P \), the pressure drop between the channel inlet and outlet, is computed from the measured pressure loss by subtracting the port and the pipe losses corresponding to the fluid stream considered. The average velocity in the channel \( u_m \) is estimated as:

\[
u_m = \frac{\dot{V}}{b L_h} \quad (10)
\]

The channel inlet-outlet pressure loss \( \Delta P \) is calculated from the measured pressure loss, by subtracting all the pressure losses (ports and piping components) of the fluid stream considered.

In this work, the hot fluid flow rate is several times greater than that of the cold one. This results in a narrow difference \( T_h - T_c \) of the hot channel and a great difference \( T_h - T_o \) on the cold side. A small hot side \( \Delta T \) induces however substantial errors when evaluating the overall heat balance from the hot and cold channels. For such a reason, the overall heat transfer rate \( \Delta Q \) is uniquely based on \( T_i \) and \( T_o \) values of the cold side. This method has been adopted solely based on the proven fidelity of measurements from experiments. The heat transfer rate \( \Delta Q \) is then calculated using the following equation based on the cold channel values only:

\[
\Delta Q = \dot{\nu} \rho c_p \Delta T \quad (11)
\]

The value of the quantity \( \Delta Q \) is used to estimate the average surface heat transfer coefficient for both the hot and cold passage (Eqs. 12):

\[
h_{m,c} = \frac{\Delta Q}{A \cdot (T_{w,c} - T_{h,c})} \quad (12a)
\]

\[
h_{m,h} = \frac{\Delta Q}{A \cdot (T_{w,h} - T_{h,c})} \quad (12b)
\]

The average wall surface temperatures of the cold and hot sides of the central plate, \( T_{w,c} \) and \( T_{w,h} \), are also related to \( \Delta Q \) and the average convection heat transfer coefficients \( h_{m,c} \) and \( h_{m,h} \) as follows:

\[
\Delta Q = A \cdot h_{m,c}(T_{w,c} - T_{h,c}) = A \cdot h_{m,h}(T_{w,h} - T_{h,c}) = k_v \cdot \frac{A}{\delta} \cdot (T_{w,h} - T_{w,c}) \quad (13)
\]

One should note that the computing of the temperatures \( T_{w,c} \) and \( T_{w,h} \) and the convection heat transfer coefficients \( h_{m,c} \) and \( h_{m,h} \) requires an iterative process, which can be summarized as follow. At first, we assume that \( T_{w,c} = T_{w,h} \) = the average of the temperatures at the two inlets, and calculate \( h_{m,c} \) and \( h_{m,h} \) using Eqs. 12a and 12b. Then, these values are introduced into Eq. 13 to obtain the new i.e. guessed values for \( T_{w,c} \) and \( T_{w,h} \). Such a process is repeated until the difference between two successive values of each variable is less than 0.01%. The reader may consult Ref. [21] for complete details of the calculation and data reduction procedures.
4 Experimental data and discussion

In order to validate the experimental setup, we first carried out isothermal tests, using first water and then for each particle volume fraction of the CuO-water nanofluid. The fluid at ambient temperature (≈23°C) was forced into the cold channel; the channel Reynolds number varied between 50 and 1000. A total number of nearly one hundred and twenty isothermal tests were performed, including those using water for comparison purposes.

For non-isothermal tests, the water in the hot channel was maintained at a bulk temperature of ≈37°C and with a steady high flow (Reynolds number maintained at ≈4100). Such a high hot water flowrate is necessary to maintain the same thermal condition (i.e. a nearly uniform surface temperature) on the hot side of the intermediate plate. In the cold channel, the CuO-water nanofluid inlet temperature was maintained at ≈16°C while the Reynolds number varied from 50 to 1000 (note that the channel Reynolds number test range varied slightly for each particle volume fraction, 2% and 4.65%). For comparison purpose, non-isothermal tests were also performed with water in both channels. A total number of ≈60 non-isothermal tests were performed in this study.

4.1 Nanofluid hydraulic characteristics

We process first to compare the hydraulic behavior of CuO-water nanofluid and that of water, the pressure loss through the cold channel is measured over the range of volumetric flow rate. The results are shown in Fig. 5 for water and the two nanofluid particle volume fractions studied. It is observed that for a same flow rate, the measured pressure loss for 4.65% particle volume fraction is clearly higher than that of the 2% nanofluid. The latter and water produce the same pressure drop. The increase of ∆P for the 4.65% particle fraction, which clearly becomes more important with increasing flow rate, is obviously due to the combined effect of the convection forces and the higher viscosity of the CuO-water nanofluid while compared to water (see in particular [33]). It is worth mentioning that such behavior regarding the nanofluid pressure loss appears consistent with that previously found from other works - see in particular Pantzali et al. [25].

Fig. 5 Nanofluid v/s water pressure losses

Fig. 6 shows on log-log scales the results for water and nanofluids friction factor as function of the cold channel Reynolds number. For the comparison purpose, the corresponding results computed using the following Kakaç and Liu [22]’s theoretical correlation – correlation that is specific for the present PHE’s characteristics - are also displayed:

\[ f = 11.96 \Re^{-0.183} \]  

(14)

As previously observed for the nanofluid pressure loss trend (Fig. 5), we can observe here again the same behavior regarding the variation of the friction factor \( f \) versus \( \Re \) in Fig. 6. Thus, for a given value of the Reynolds number, it is first observed that \( f \) is nearly identical for water and the 2% CuO-water nanofluid. However, the 4.65%-particle-volume-fraction nanofluid clearly produces higher friction factors. Such an increase of \( f \) appears more important for the range \( \Re \leq 350 \) approximately; for \( \Re > 350 \), this increase, although still persists, appears less pronounced.

Fig. 6 Nanofluid v/s water friction factor
In the present work, we are interested to investigate the laminar-turbulent transition for nanofluids flowing in the PHE studied. Fig. 7 shows a close-up view of Fig. 6 for Re varying from 100 and 300. One can observe, at first, that Kakaç and Liu [22]'s correlation, Eq. 14, is in good accordance with water and nanofluid with 2% volume fraction for Re higher than 200 approximately (it should be noted that Eq.14 is only applicable to turbulent regime). We have attempted to graphically determine the value of $Re_{crit}$, the critical Reynolds number corresponding to the laminar-turbulent transition in the cold channel. We found that $Re_{crit} \approx 120$ for both water and the 2% particle-volume-fraction nanofluid and $Re_{crit} \approx 160$ for the 4.65% nanofluid. It is worth noting that, according to previous works, for a chevron-type PHE as the one under study, the turbulent regime may be expected for $Re \geq 400$ (see in particular [20, 34]). Such a transition may even occur for the Reynolds number as low as 200, depending upon the internal geometry of the plates and corrugations. In this work, it seems that the higher viscosity of the CuO-water nanofluid has delayed the transition to turbulence (when compared to water). Such a behavior is physically realistic from the fluid dynamics viewpoint as it is well-known that a more viscous fluid tends to be more stable to disturbances.

4.2 Nanofluid heat transfer characteristics

Fig. 8 shows the results obtained for the cold channel average heat transfer coefficient for water and the two nanofluids as function of the cold channel volumetric flow rate in l/min. Thus, it is very interesting to observe that the 2% nanofluid gives highest heat transfer coefficients over the entire range of flow rate tested. However, it is somewhat surprising to find that the 4.65% nanofluid provides nearly the same performance than that of water. For very low flowrate, less than 1.5 l/min approximately, the heat transfer coefficient given by the 4.65% is indeed the lowest. On the other hand, for sufficiently high flow rates, > 5 l/min, or minor Re > 800 approximately, all the three fluids tested provide nearly the same heat transfer coefficient. Such a behavior is believed to be due to the dominant turbulent effects. We believe that the above deceiving heat transfer performance of the CuO-water 4.65% nanofluid is due to the viscosity effects on the internal flow within the cold channel. In fact, it has been shown that for the chevron-type PHE as the one under study, the internal flow structure is rather complex and often very tortuous due to corrugations from adjacent plates [16]. The high viscosity of the 4.65% CuO-water nanofluid would induce some important modification into the internal flow field and consequently on the thermal field. In other words, the presence of nanoparticles, especially with high concentrations, may decrease or completely suppress the corrugations beneficial effects on heat transfer. It is interesting to mention that such a behavior regarding this poorer nanofluid performance seems confirmed by recent numerical results [35] for the same PHE studied. In [35], it is shown that an optimal particle volume fraction seems to exist which maximizes the nanofluid convective heat transfer coefficient.

It is worth mentioning that although only a two-channel-PHE is considered, we believe that similar hydraulic and thermal behaviors can be expected for multiple-channel PHE, as shown through our recent works in this area [21]. Also, as the data regarding the hydraulic and heat transfer characteristics of a PHE using nanofluids remains relatively scarce, the results from this work constitute, in our opinion, an interesting contribution to the nanofluid knowledge.

4.3 PHE end-plates’ isotherm structure

Fig. 8 Nanofluid v/s water heat transfer coefficient
Fig. 9 (a, b, c) shows the isotherm structure as given by the thermocouples fixed on the outer surface of the PHE end-plates. These results present three particular cases for the nanofluid with 4.65% particle volume fraction and volumetric flow rates respectively of 0.35 l/min, 3.82 l/min and 7.2 l/min approximately. We can observe at first that for a very low nanofluid flow, Case 1, isotherms on the cold side show a large zone of cold fluid in an extended bottom region cover almost the entire channel width, which indicates the dominant effect of heat conduction within the fluid. On the hot side end-plate, one can see that high temperature gradients exist along the vertical edges of the channel, while in the large area of the central region, an extended fluid volume at uniform temperature is present. In Case 2, with an intermediate nanofluid flow rate (3.82 l/min), isotherm patterns on both plates appear more regular denoting an increasing strength of the convection effects. In particular, on the hot side end-plate, the large zones of nearly uniform temperature observed in Case 1 are now no longer existed. Also, isotherms’ pattern on the cold end-plate look more regular which shows the heating of the cold fluid along the X-axis towards the top region. The same behaviour can be observed for Case 3 (flow rate $\approx 7.2$ l/min) with more pronounced convection effects.

4 Conclusion
An experimental investigation was performed to study forced convective heat transfer and hydraulic characteristics of a chevron-type two-channel PHE.

![Fig. 9 Isotherms structure on PHE end-plates for the 4.65% nanofluid (Left view: cold side, fluid enters at left lower corner and exits at left upper corner; Right view: hot side, fluid enters at upper right corner and exits at right lower corner)](image)

The fluid passages are formed by three plates with herringbone pattern and trapezoidal shape corrugations. Heated water was forced into the hot side and the 29 nm-CuO nanoparticles-water nanofluid was used in the cold side. Two particular particle volume fractions, 2% and 4.65%, were studied. Experimental data have shown that the 4.65% nanofluid produces higher pressure loss and friction factor compared to those of water and the 2% nanofluid. A clear delay of the laminar-turbulent transition was found for the 4.65% nanofluid with respect to water. The 2% nanofluid exhibits nearly the same friction factor and laminar-turbulent-transition Reynolds number as those of water. Regarding the heat transfer, results shown that only the nanofluid with 2% particle volume fraction can provide a heat transfer enhancement over the range of flow rate considered.

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