



## Impact of channel geometry on the fluid flow of newtonian and non-Newtonian

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### Abstract

In this manuscript, an experimental study was performed regarding the heat transfer performance and pressure drop specifications of a Newtonian fluid and a non-Newtonian fluid. To this end, laminar flow of the deionized (DI) water and the aqueous solution containing 0.2 wt.% carboxyl methyl cellulose (CMC) were used as the Newtonian fluid and the non-Newtonian fluid, respectively, in wavy and straight mini-channel cross-flow heat exchangers (CFHE). Each mini-channel cross flow plate heat exchanger (MCFPHE) was made of three copper plates for the hot and cold sides, which contain 27 mini channels in parallel. The mini-channel CFHE was examined for Reynolds numbers (Re) in the range of 200 to 1800 and the hot-side inlet temperature was between 40 °C and 500 while the cold-side temperature varied in the range of 10 °C to 20 °C. The thermal and rheological properties of the non-Newtonian fluid were investigated. The results showed that for both fluids, the thermal performance of the wavy mini-channel is more than that of the straight mini-channel and, although the pressure drop was intensified, its effect is low at high Re values.

**Keywords:** Heat exchanger, mini- max channel, non-Newtonian, CMC, wavy)

### Introduction

The increasing industrial development of recent decades is directly linked to the improvement and development of new technologies. Heat exchangers are devices that are used to perform the thermal exchange between two fluids at different temperatures and, considering their wide applications, they have become the object of several research studies and developments. However, even along with this increasing evolution, the improvement of this system continues to be sought<sup>[1, 2]</sup>, aiming at both the conservation of energy and the elaboration of projects applied to specific situations. Among the several existing types, one of the best known is the plate heat exchanger. This type of equipment is widely used in various industrial segments and with numerous applications. Due to the need for smaller equipment with high effectiveness, micro and mini-channel heat exchangers were developed. The higher volumetric heat transfer densities require advanced manufacturing techniques and more complex manifold designs. The manufacturing processes used in the fabrication of mini/micro-channel heat sinks and compared the different techniques related to tolerances and material compatibility<sup>[2]</sup>. The different geometric designs of mini/micro-channels, which were derived from numerical simulation and experimental works<sup>[3]</sup>. Comprehensively discussed the passive techniques for heat transfer augmentation in mini/micro-channels<sup>[4]</sup>. Even though mini- and micro-channels can enhance the heat transfer rates significantly, it is not sufficient for some applications that need large heat fluxes. Thus, the usage of wavy channels has been considered. It was reported that wavy channels with larger wave amplitudes and shorter wavelengths, and channels employing coolants with nano-particles displayed better cooling performance. In another study, water-Cu and water-Al<sub>2</sub>O<sub>3</sub>, as nanofluid was investigated in terms of heat transfer capability and fluid flow by using a rectangular micro-channel. It was shown that the Nusselt number increases with the increase in volume fraction and the decrease in particle diameter and that it is about three times

higher for a nanofluid in a wavy micro-channel as compared to water in a straight micro-channel.

The fluid flow and heat transfer capability of the TiO<sub>2</sub> nanofluid were studied<sup>[5]</sup> experimentally in the straight mini-channel heat sink for laminar and transitional flows. The nanofluid containing 15 wt.% nanoparticle was used as a coolant. It is reported that the thermal performance of the nanofluid is a function of heating power, which could be enhanced at lower heating power. However, by decreasing the heating power, the pressure drop was increased. The nanoparticles in addition to enhancing the heat transfer, do not lead to excessive pressure. Thermal performance of a microprocessor was investigated by using pure water and nanofluid of Al<sub>2</sub>O<sub>3</sub> and Cu<sup>[6]</sup>. It was observed the Al<sub>2</sub>O<sub>3</sub> water nanofluid yields significant performance enhancing the heat transfer coefficient and reducing the thermal resistance. Also, by increasing the Reynolds number, the Cu-water nanofluid provided better results for thermal enhancement. The Nanofluids were found to increase the heat transfer rate compared to water. Parametric studies are carried out for different particle concentrations of 1, 3, and 5 vol.%. The particle size effect is observed for different particle sizes of 50 nm and 150 nm. In addition to the Nanofluid, non-Newtonian fluid<sup>[12]</sup> to<sup>[18]</sup> is an alternative for working fluid. Compared with Nanofluids, non-Newtonian fluids widely exist and can easily be prepared. Many common fluids such as solutions of polymers or surfactants, shows non-Newtonian fluid behaviour. The behaviour of non-Newtonian fluids is of great interest to many applications, including collecting, distributing, detecting, mixing and separating different types of biological and chemical species in a microchip. As a working fluid of a heat exchanger, for example, the heat transfer performance of non-Newtonian fluids was studied<sup>[8]</sup> in square micro-channels with triangular ribs and in a manifold micro-channel heat sink, respectively. They have reported that by increasing the volume fraction of the nanoparticles, the heat transfer is adversely affected. Moreover, an increase in heat and mass transfer by non-

Newtonian flows and noticed that the change in the dynamic viscosity of the materials in working chemical materials is the main reason for strengthening the secondary flow in the dimpled/protruded passage with flow separation [9]. The Nusselt numbers (Mn) for the non-Newtonian fluid flows in a spiral coil were, on mean, somewhat higher than those for Newtonian fluids because the viscous element of the shear-thinning polymer shows a tendency to affect the mixing effect of the Dean vortices [10]. In the current work, the non-Newtonian characteristic of aqueous solutions containing 2 wt.% carboxyl methyl cellulose (CMC) was assessed, and calculations were also performed regarding the rheological behaviour of the non-Newtonian fluid. Furthermore, for a new cross-flow heat exchanger, pressure drop and heat transfer performance were studied with experiments with straight and wavy mini-channels.

### Materials and Methods

The CMC solutions were prepared by adding sodium carboxyl methyl cellulose with a nominal molecular weight of 900,000 [g $\cdot$ mol $^{-1}$ ] and a DS of 0.8 to 0.95 (Dac-Jung company, South Korea) to distilled water, and agitating it on an electromagnetic stirrer at low axial speed for 2 h, to achieve complete hydration and avoiding the formation of air bubbles. The studied concentration was 0.2 wt.%. The specific heat  $c_p$  [J $\cdot$ kg $^{-1}$  K $^{-1}$ ] and thermal conductivity  $k$ , were obtained for 0.2% CMC solution, at temperatures between 40 deg $^{\circ}$ C and 50  $^{\circ}$ C. By the thermal conductivity and specific heat at constant were obtained for a non-Newtonian fluid. This is a portable device that is used in thermal analyses. For evaluating the specific heat (heat capacity) and thermal conductivity, the transient line heat source method is used with a KD $_2$  device A small needle sensor (SH-1) was employed for assessing the specific heat in which the heat pulse method is used, and reliable values are obtained for specific heat capacity  $c_p$  by using nonlinear least square method during the processes. Only the SH-1 sensor can be considered for determining the specific heat, which is 30 mm in length and 1.28 mm in diameter and distance between two needles is 6 mm. By adopting this sensor, the thermal conductivity can also be determined [11]. The precision of this instrument is 5% which meets the standards of EN 55022 [12]. The density of the CMC solution was determined in triplicates, by pycnometer. Rheological measurements were determined using a Rheometer (MCR 301 by Anton Paar, Graz, Austria) equipped with a cone-and-plate geometry (angle: 0.034 rad, diameter: 40 mm). The shear rate was varied between 12.2 s $^{-1}$  and 200s $^{-1}$  and both upward and downward tests were performed in duplicate for the CMC solution, at five different temperatures between 10  $^{\circ}$ C and 50  $^{\circ}$ C.

An experimental setup has been developed to examine the heat transfer characteristics and the drop in the pressure of the wavy and straight mini-channels under different flow conditions. It consists of two parts a hot fluid and a cold fluid. For maintaining the inlet temperature and flow velocity at constant values, a thermostatic bath, a controllable magnetic gear pump and a paddle wheel flow meter were used for each part of the setup. Two filters were installed at the entrance of each flow meter to remove the external matter in the liquid and to prevent fluctuations and the rusting of the flow meter. Insulation is applied throughout the unit in an experimental setting in order to minimize the heat loss. Thermocouples, as well as absolute

and differential pressure transmitters, were installed on all inputs and outputs. Before starting each experiment, each measuring device was calibrated. Non-Newtonian fluid flow rate has been varied in a range of 0.5 Lmin to 4 Lmin whereas the inlet hot stream temperature ranges from 40  $^{\circ}$ C to 50  $^{\circ}$ C A the cold stream flow rate has been varied in range of 0.5 Lmin $^{-1}$  to 4 Lmin $^{-1}$  along with inlet temperature being in a range of 10  $^{\circ}$ C to 20  $^{\circ}$ C A typical test usually takes about 20 minutes. A time interval was needed for the system to achieve stable conditions. To verify that such conditions have reached a stable state, the temperature was monitored continuously. Once the thermal equilibrium conditions were obtained, the flow rate and temperature readings were recorded. The average values of the measurements were recorded beside the estimated errors.

The heat exchanger plate used in this study was fabricated from copper alloy. Each plate was 80 mm in width and 80 mm in length, and the thickness is 2 mm. The wavy and straight mini-channels with cross-sectional dimensions of 1 mm in width and 1 mm depth were machined with a CNC machine. Schematic patterns of the constructed plates with two keyway dimensions. As can be observed in the figure, specific geometrical parameters of wavy channels can be considered for investigating the relative waviness (2A/2L) In this regard, in the current research, the impact of such parameters on the performance of cross-flow heat exchangers was studied on two levels of relative waviness (i.w. 2A/2L = 0.2 and 0.3). The dimensions of the mini-cross-flow heat exchanger are also tabulated in Table-1.

**Table 1:** Geometrical parameters of tested plate heat exchanger

Parameter	Value
Wave amplitude [2A]	2.2,3.2
Dimensions of heat exchanger (w $\times$ l $\times$ h) [mm]	100 $\times$ 100 $\times$ 100
Dimensions of plates (w $\times$ l $\times$ t) [mm]	80 $\times$ 80 $\times$ 80
Wavelength (2L) [mm]	10.67
Thermal conductivity of plate material [Wm $^{-1}$ K $^{-1}$ ]	385
Total number of channels	27
Number of channels per plate	6

### Thermal Physical Properties of CMC Aqueous Solutions:

Specific heat capacity and thermal conductivity were determined at three different temperatures. The viscosity against the shear stress of the CMC aqueous solution at different temperatures. By increasing the shear rate, the viscosity of the CMC aqueous solutions decreases, also at the same shear rate, the viscosity of the CMC aqueous solutions increases with decreases in temperature. It is disclosed that the CMC solutions present shear-thinning behaviour. However, the CMC aqueous solutions follow the power law model, with a flow behaviour index or power law index. of less than unity ( $n < 1$ )

### Results and Discussion

Experiments were conducted for two wavy channels with different relative waviness (2A/2L) of the wavy mini-channels and in the presence of CMC to study the effects of these parameters on heat transfer capability and friction factor. Initial tests were also performed for a straight channel and base fluid to validate the experiment procedure and form baselines for the comparison. The inlet temperature for the hot liquid was 40 deg  $^{\circ}$ C and 50 deg  $^{\circ}$ C with a cold-water temperature of 10 deg  $^{\circ}$ C to 20 deg  $^{\circ}$ C the

hot and cold flow rate in Reynolds was measured from 200 to 600, The heat transfer rate in hot and cold fluids through the test section can be achieved using dot  $Q = m c_p \Delta T$ . By considering the logarithmic temperature average (LMTD), the heat transfer coefficient ( $U$ ) can be evaluated:

$$\Delta T_{LMTD} = \frac{(T_{i,hot} - T_{o,cold}) - (T_{o,hot} - T_{i,cold})}{\ln \left\{ \frac{(T_{i,hot} - T_{o,cold})}{(T_{o,hot} - T_{i,cold})} \right\}} \quad (1)$$

and mean heat transfer  $Q_m$  [W] via

$$U = \frac{Q_m}{F A_s \Delta T_{LMTD}} \quad (2)$$

That  $A_s$ [m<sup>2</sup>] is the total heat transfer area and  $Q_m = (Q_{hot} + Q_{cold})/2$ . Also, the coefficient factor  $F$  for the cross-flow heat exchanger in this equation is 0.97. The hydraulic diameter is calculated using the proposed method [14].

$$d_h = \frac{4V_s}{A_s} \quad (3)$$

where  $V_s$  [m<sup>3</sup>] is the enclosed (wetted) volume. For power law fluids, A new generalized Re. For power law fluid flow, the generalized Re for non-Newtonian fluid is presented [15]

$$Re = \frac{\rho d_h^n u^{2-n} \left( \frac{n}{a+bn} \right)^n}{k} \quad (4)$$

where  $\rho$  [kgm<sup>-3</sup>] and  $u$  [ms<sup>-1</sup>] are density and the average velocity of the fluid respectively, and  $k$  and  $n$  are the rheological parameters, for the CMC solution. The constants are  $\alpha = 0.2121$  and  $b = 0.6766$  for square cross-sectional channels. It is that the heat transfer coefficients of the hot and cold loop should be calculated independently. The Fanning friction factor was also calculated using the following equation:

$$f = \frac{1}{2} \frac{d_h \Delta P / L}{\rho u^2} \quad (5)$$

The overall uncertainty for the estimated results is given in Table -2. Repeated tests show that all the sets of the experimental data are within the uncertainty limits.

**Table 2:** Parameters and estimated uncertainty

Instrument	Model	Accuracy
Flow meter (l/min)	Fotek - KTM	±0.5% of readings
Thermocouples[T]	Type-T	±1.0 [°C] of readings
Pressure Transmitter [p]	EWD2-2.5	≤1%
Pump [24DC]	Gear pump	Head 21 meter
PID controller	Delta	+1% of readings
Bolt heater [W]	2000	±2 % of readings
Thermo-physical properties	-	-
Viscosity	Brookfield	±2.5% of readings
Density	Anton paar, MCR 301	±3% of readings
Thermal conductivity	Decagon KD2-Pro	+5% of readings
Specific heat	Decagon KD2-Pro	+5% of readings
Calculated parameters	-	-
Reynolds number of hot and cold side	Moffat	+1% to 2%
Heat transfer coefficient of hot and side	Moffat	+6% to 7%
Friction factor	Moffat	+6%

**Validation:** To validate the present experimental results and form a baseline for comparison, the initial tests are conducted for the water flow in the straight mini-channel cross-flow heat exchanger. The straight mini-channel has the same number of mini-channels and magnitude of geometrical parameters as the wavy channel. The fitted curve of the heat transfer rate is calculated using correlations of Nu proposed [16] which are used to validate the experimental data of heat transfer rate as depicted for the water flow against the Re of the cold fluid. As shown in the figure, the obtained. results in the present study are in good agreement with analytical analysis. For estimating the heat transfer coefficient, the thickness of the plate,  $K_{copper}$  [Wm<sup>-1</sup>K<sup>-1</sup>] is the thermal conductivity of copper heat exchanger has been fabricated from copper), [Wm<sup>-2</sup>k<sup>-1</sup>] and  $h_{cold}$  [Wm<sup>-2</sup>K<sup>-1</sup>] are the convective heat transfer coefficients of the hot and cold water. respectively. The correlations overestimate the obtained results. However, it can be related to the fouling resistances that are ignored during the calculations and, considering the fact that the

correlations are empirical, the obtained data would be acceptable.

**Effect of Geometrical Parameters:** There are three submissive mechanisms [7] that enhance the heat transfer rate in single-phase flow, including: (1) decrease in thickness of the thermal boundary layer, (2) increase in fluid disturbance [18]; and (3) increase in gradient velocity in vicinity of the heat transfer wall. When the fluid flows in wavy channels, disturbance of the streamlines of the flow occurs [19]. Mixing of the flow is the result of this interruption and, as a consequence, the heat transfer capability is enhanced. The regions that possess the maximum amplitude values in a wavy channel initiate the vortices and secondary flows. While flowing the liquid through curved passages, the liquid is exposed to centrifugal force resulting in secondary flow, which is commonly known as Dean Vortices or chaotic advection; as vortices are formed, the fluid undergoes a rapid shearing and stretching, which results in better fluid mixing and, consequently, higher rates of heat transfer are

acquired. In addition, the pressure near the outer concave wall is increased due to the centrifugal force in the curvature of the channel and, as a result, the pressure gradient is enhanced toward the centre of the flow. Considering this pressure gradient, the high-pressure fluid is moved from the outer region toward the core [20]. However, all these processes enhance not only the heat transfer rate but also the pressure drop. The variations of  $a$  as a function of  $Re$ . It is clear that the higher heat transfer is obtained for a wavy mini-channel cross-flow heat exchanger (MCFPHE) compared with the straight channels in which the heat transfer rate is increased with the relative waviness. The analysis of  $U$  vs.  $Re$  of the cold fluid confirms this conclusion.

**Effect of Non-Newtonian Behaviour:** The viscosity increase caused by the CMC dissolution in water can affect the flow pattern or flow regime considerably. To study the effects of the non-Newtonian fluid, ie, the water-CMC mixtures, the variations of the  $q$  as a function of the  $Re$ . According to this figure, the same conclusion regarding the wavy effect of Newtonian fluid can be drawn. However, higher heat transfer values compared to the Newtonian fluid are obtained at all configurations.

**Pressure Drop Characteristics:** Experiments showed that using the CMC/water non-Newtonian fluids can slightly enhance the pressure drop in comparison to the base fluid. In fact, the presence of CMC inside the water can enhance the viscosity and, as a result, higher values of pressure drop are obtained for the non-Newtonian fluids, compares the pressure drop of the systems with water and CMC solution as hot fluid versus  $Re$  of the hot fluid. According to this figure, the ratio is always higher than 1, which verifies the above-mentioned conclusion in comparing the non-Newtonian and Newtonian fluids. The effect of temperature is negligible and wavy 2 (with higher relative waviness) shows the least pressure drop ratio, especially at higher  $Re$ . The experimental results of the friction factor are for CMC, the friction factor of the wavy channel is always higher than that of the straight channel; Be the influence of non-Newtonian fluid in enhancing the pressure drop is less in the case of wavy MCFPHE with higher relative waviness.

**Overall Performance Factor (PF):** To investigate the performance of wavy passages, the heat transfer performance of the two wavy channel configurations is compared along with the pressure drop. The performance evaluation criterion (PEC) as a function of the Reynolds number. It was calculated from Eq. 6 [21].

$$PEC = \frac{U_{nm}/U_{ws}}{(f_{nm}/f_{ws})^{0.333}}, \quad (6)$$

where  $U$ , and, are heat transfer coefficient and friction factors of a non-Newtonian fluid flow in a wavy and straight channel with different relative waviness and the  $U$  and  $f$  are heat transfer coefficient and friction factors of a DI water in a straight, respectively. It is clear that when the PEC is higher than unity, the applied technique is more in favour of heat transfer improvement rather than increasing the pressure drop. The geometric parameters, the working fluid composition, and the Reynolds number affect the amount of

this parameter, it is clear that for each case, the PEC increases approximately as the Reynolds increases, and the overall improvement due to the wave channel in the higher flow current is significant. Also, for a given flow rate, the non-Newtonian fluid with higher weight fraction has higher values of the PEC

## Conclusion

In this research work, the thermal-hydraulic performance of a wavy mini-channel cross-flow heat exchanger was studied using the CMC/water as non-Newtonian fluid, and the effects of relative waviness ( $A/L$ ), non-Newtonian fluid, and Reynolds number were studied. The results obtained for the straight MCHS were first confirmed by considering the previous works for which a reasonable agreement was established. It was observed that for all considered cases, higher heat transfer was obtained for the wavy channel compared with a straight channel. The CMC/water non-Newtonian showed better heating performance compared with the base fluid. The pressure drop of the wavy mini-channels was higher than that of the straight mini-channels. This is due to the change of the geometrical parameter, especially with relative waviness ( $A/L$ ) change. It is suggested that applying the combined method (Le. simultaneous use of wavy mini-channels and non-Newtonian fluid) as proposed in this research, can be a good choice in practical applications to enhance the heat transfer performance of cross-flow mini-channel heat exchangers.

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