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Performance Evaluation of a New Nano Fluid Through Micro Channels Heat Exchanger

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The purpose of this study was to investigate heat transfer performance of a wavy micro channels heat exchanger with a new nanofluid. Nonporous graphene in an aqueous solution containing 0.2 wt.% carboxyl methyl cellulose (CMC) were used as base hot fluid. In preparation of non-Newtonian nanofluids by using properties of the base fluid, Gum Arabic, Tween 80, CTAB, Triton X100, and Acumer 300 were used as surfactants. The Results show that 1- as flowrate, relative waviness (2A/2L) and concentration of NPG increase the convective heat transfer coefficient could be intensified. 2- it was seen that with increasing the flowrate and relative waviness (2A/2L) the pressure drop was intensified. The impacts of the obtained results are introducing a new method to enhance heat transfer. In addition, a new definition for comprehensive performance of heat exchangers is introduced and used to evaluate the feasibility of the proposed system.

1. Introduction

With Because because of the vital importance of the equipment such as heat exchangers in heat transfer in most of engineering fields, the issue of enhancing their heat transfer capability is of great importance in increasing the thermal efficiencies (Kulkarni et al., 2018). Key geometrical parameters of plate heat exchangers were reported by (Tan et al., 2018). Using waviness and indentation over the equipment's wall has been focused as a passive heat transfer enhancement method which is based on creating vortexes, rotating movement, flow instability or increasing the turbulence. Significant improvement of heat transfers along with a little pressure drop and simplicity of the manufacturing method, have made creation of waviness more interesting than other passive methods including use of indentations or creator of vortexes, especially in compact exchangers. The effects of alternative variegation of pressure gradient and curvature of flow line, induce turbulent structures. Wavy channel enhances the heat transfer performance by removing or decreasing the boundary layer diameter and increasing the mixing of the flow. Usage of nanofluid as the base fluid is another alternative for improving the heat transfer capability in heat exchangers which has recently attracted the attention of researchers. In this respective in the past few years, it has been found that addition of nanoparticles to a base fluid such as water, oil, or ethylene glycol can increase the thermal conductivity. In the field of using nanofluids and wavy channels to increase the heat transfer, some researches have been performed (Rostami et al., 2018; Khoshvaght-Aliabadi et al., 2017). Due to various advantages of wavy microchannel over the straight microchannel, they are considered as practical choices for cooling systems of computing and electronic equipment.

(Rush et al., 1999) studied the local heat transfer and fluid behavior of the laminar and transient flows experimentally in the Reynolds number (Re) range of 100-900 in the Sinusoidal sinusoidal wave channels. Their studied geometry includes a channel which is confined by two wavy walls and also there is phase difference between upper and bottom walls. Their results showed that the Nusselt number (Nu) is increased by increment of Re. (Wang et al., 2002) analyzed the heat transfer rate for the laminar flow inside a convergent – divergent channel in the Re range of 100-700. They concluded that by increasing the Re and ratio of waviness amplitude/wavelength, Nu range and friction coefficient are increased. (Bahaidarah et al., 2005) investigated the fluid flow and permanent two-dimensional heat transfer inside a channel with alternative wavinesses on a fluid by Prandtl number (Pr) of 0.7 numerically and compared it with flow inside a relevant fluid in a straight channel. In their work, at low Re for both geometries, partial or negligible heat transfer increment was observed in comparison to channel with parallel walls; however, at high Re in some cases, the heat transfer increment up to 80% were reported. (Sui et al., 2010) performed a numerical study on laminar flow of water and heat transfer in three-dimensional wavy channel with rectangular cross section. They have reported that when the cooler liquid is flowed inside wavy microchannel, a secondary flow can be formed. (Heydari et al., 2010) investigated the heat transfer and flow field numerically by using a nanofluid in wavy channel. They have concluded that by adding nanoparticles and using wavy horizontal walls, heat transfer performance can be enhanced up to 50%. (Ahmed et al., 2011) studied the heat transfer and pressure drop of the water-copper nanofluid inside the wavy channel numerically. They found that the heat transfer performance is improved by increasing the volume fraction of nanoparticles and Re, however, a partial increase in pressure drop was also observed. (Mohammed et al., 2011) investigated the heat transfer and flow characteristics of water in thermal well of the wavy microchannel (WMCHs) numerically. They observed that wavy microchannel have better performance in heat transfer compared with the straight microchannel with the same cross section. (Raghuraman et al., 2013) performed experiments on microchannel with rectangular cross section made of copper by using titanium nanofluid as the working fluid. They found that friction coefficient and shear stress of the wall was increased proportional to increase in waviness amplitude of the microchannel. The heat transfer enhancement is related to the formation of vortices in the channel cross sections (Lin et al., 2017). In contrary to Newtonian liquids, the relation between shear stress and shear level of the non-Newtonian fluids is not linear. (Akbarinia et al., 2011) have claimed that by raising the nanoparticles volume fraction, viscosity was increased. (Barkhordari et al., 2007) showed that more slipping of the particles decreased the heat transfer rate in the tubes. Numerical investigation of the laminar fluid in a microchannel under boundary condition of lipping and no slipping by using nanowire nanofluids, was performed by (Raisi et al., 2011). They have investigated the effects of Re, volume fraction and slipping rate on amount of permeant heat transfer.

In the current research work, in addition to investigating the effects of geometrical parameters of channel, Re and volume fraction of the nanoparticles on enhancement of the heat transfer and pressure drop, justifiability of using wavy channels and Non-Newtonian nanofluid as the working fluid is investigated by using the coefficient of performance, justifiability of using wavy channels and Non-Newtonian nanofluid as the working fluid, is investigated. Also considering the coefficient of performance, it is tried to optimize the geometrical parameters of the channel such as amplitude of wave and wavelength of the wave, in order to obtain the highest increment of heat transfer along with the lowest pressure drop.

Here, at first experimental and numerical method and finally results and conclusions are explained

2. Laboratory method

The nonporous graphene (NPG) was purchased from Research Institute of Petroleum Industry (RIPI). For purifying, 18% HCl was used followed by washing with distilled water and drying at 100°C. To characterize the samples, SEM and TEM techniques were employed as shown in Figure 1.

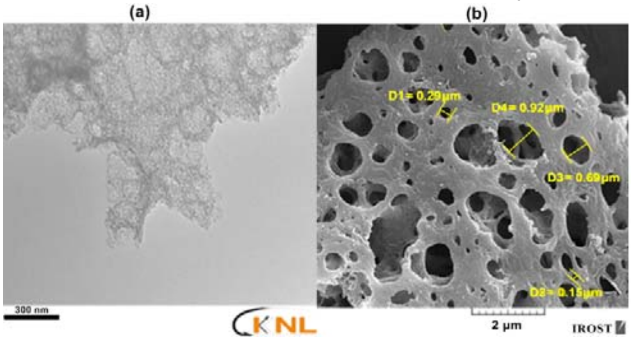

Figure 1: (a) TEM and (b) SEM images of nonporous graphene.

To prepare the non-Newtonian nanofluids, 0.2 wt.% CMC with molecular weight of 900000 [gmol-1] and DS in the range of 0.8-0.95 was added to the distilled water. To this end, an electromagnetic mixer at low axial velocity was used for 2 h to reach a homogenous fluid and avoiding the formation of bubbles. In preparation of non-Newtonian nanofluids by using properties of the base fluid, it should be noted that stabilizing the nanoparticles in this base fluid, specifically in presence of different cations and anions, is of great importance. To make the Nano porous graphene particles stable, surfactants including Gum Arabic, Tween80, CTAB, Triton X100, and Acumer 300 were used. A two-step method was used to form nano porous graphene nanofluids (Sidik et al., 2014). At first, appropriate amounts of nanoparticles and surfactants were added to the base fluid (0.2 wt.% CMC aqueous) and the resulting fluid was sonicated at 150 watts in an ultrasonic mixer (UP4005 Heilscher, Inc. made in USA). Viscosity measurement was performed by using reverse flow capillary viscometer (U-tube BS/ IP/ RF (ASTM D445-06)). As the temperature has significant effect on viscosity of the fluids, viscosity of the nanofluids was determined at different temperatures by using constant temperature bath. Thermal conductivity of the nanofluids s was obtained by using the hot heat transfer method. To determine the thermal conductivity KD2 Pro Analyzer (Decagon devices, Inc. USA) was employed. The bath with constant temperature was adopted by using a thermolator (PolyScience, Model 9712, USA) to maintain the temperature homogeneity in the range of ± 0.1°C.

3. Experiment setup

In order to investigate the heat transfer performance of the non-Newtonian nanofluids, an experimental setup composed of a hot circuit and a cold circuit in a cross-flow micro heat exchanger was used. The plates of the heat exchanger with the dimension of 8×8 cm, where three plates in hot vessel and three in cold vessel are placed in crossed condition over each other. In this regard, in the present research, the effect of this parameter on cross heat exchanger with two different geometric ratios (2A/2L = 0.2, 0.3) has been investigated. Each plate of the exchanger includes 27 wavy grooves by two different geometrical ratios. Water was used in the cold vessel and the non-Newtonian nanofluids was placed in the hot vessel. Figure 2 shows the schematic illustration of the experimental setup. Each of the loops was composed of a thermostatic bath and a gear pump, a flowmeter of paddle wheel type and two filters in inlet. In the inlet and outlet of the exchanger, Type-T thermocouple was embedded to measure the temperature and pressure difference was read by pressure difference sensors. The flow of the non-Newtonian nanofluids was adjusted in the range of 0.5-4 ml/min; however, the temperature was varied in the range of 40-50°C. A common experiment usually takes 20 min. To check whether equilibrium condition is reached, temperature is continuously adjusted by using the PID controller. When the state of thermal equilibrium is reached, flow rate, temperature, and pressure are recorded.

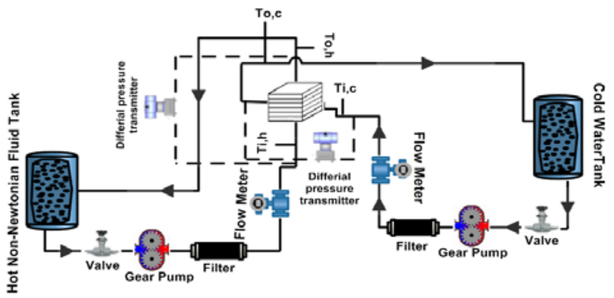

Figure 2: A schema of experimental loop

The Thermo thermo -physical properties of the fluids were calculated using the average bulk temperature. Having the plate depth (equivalent to 2 mm) and surface enhancement parameters, the hydraulic diameter of a corrugated plate heat exchanger can be estimated by

$$d_h = \frac{4A_{ch}}{P_{ch}} \tag{1}$$

For non-Newtonian power-law nanofluids Re is defined as Metzner and Reed (1955)

$$\mathrm{Re} = \frac{\rho u^{(2-n)} d_h}{K} \tag{2}$$

4. Numerical method

In the case of numerical discussion and simulation, the 3D computing fluid dynamics (CFD) is developed on the basis of a single-phase fluid system for the simulation of steady state flows in a mini channel cross flow heat exchanger. The non-Newtonian nanofluids have been used as a single, continuous phase fluid with completely different physical properties, such as density, thermal conductivity and viscosity. The numerical solutions for governing equations were obtained using the CFD tool, which performs FEA (Finite Element Analysis). The solver uses the GMRES (Generalized Minimal RESidual) method to solve the linearized equations. The discretization of both pressure and velocity components of the first order is used. The governing equations are the continuity equation, X, Y and Z-momentum equation, energy equation for fluid and diffusion equation for nanoparticles. The power law model used is for a non-Newtonian fluid. The entrance boundary condition was normal inflow velocity. At the outlet, the imposed boundary condition was zero pressure with no viscous stress. No-slip boundary conditions were applied at the walls of the microstructure. The 3D geometry was built using commercial CAD program. Due to the complicated structure of the cross-flow heat exchanger, the tetrahedral grid was elected to mesh computational domain, where different angles of view of the generated mesh are presented. The CFD tool utilized an iterative scheme to solve the governing partial differential equations. A parallel computation with 16 numbers of processes was performed on workstation with 32 Intel Xeon-core CPUs and 32 GB RAM. It takes approximately 2 h to acquire converged solutions. The average number of degrees of freedom in the studied cases was between 1,200,000 and 2,400,000. The prescribed converging tolerance was 0.0001.

5. Results and discussion

Figure 3 shows the variation of the viscosity (η) of the base fluid and nanofluids with the shear rate γ at 25 °C as well as consistency index k and power law index n. As can be seen, the viscosities of all working fluids decrease with the increasing of the shear rate, indicating that all the samples are typical non-Newtonian fluids with shear thinning behaviour (n < 1). For a given shear rate γ, the viscosity increases slightly with the increase in weight fraction of nanoparticle.

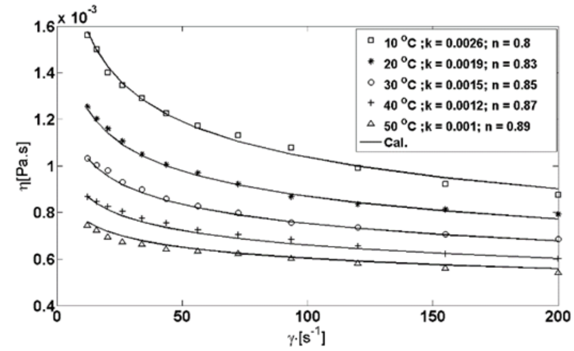

Figure 3: The relation of viscosity and shear rate for non-Newtonian nanofluid

The mean square error of data defined as are shown in Table .1. The mean of the deviations between the simulation and experimental results do not exceed 11.5 % and 15.2 % for heat transfer at relative waviness 0.2 and 0.3, respectively. Similarly, for pressure drop do not exceed 12.6 % 13.6 % and 13.6% 15.2 % at relative waviness 0.2 and 0.3, respectively that indicates a single-phase flow model is perfectly suitable for simulation of the fluid flow and heat transfer of non-Newtonian nanofluid into the mini-channel heat exchanger. It is expected that single phase flow model can impressively simulate the flow and heat transfer of non-Newtonian nanofluid at low concentrations, because, in this case, the enhancement of thermal properties is the main driver for the heat transfer improvement, while, with the increasing of nanoparticles concentration, the Brownian motion of nanoparticles and the interaction between nanoparticles and base fluid play increasingly important role in the processes of flow and heat transfer.

Table 1: MSE of U and ΔP between experimental and numerical data

Criterion	Fluid	MSE % (2A/2L = 0.2)	MSE % (2A/2L = 0.3)
U	Base fluid	5.5	6.2
	0.05 wt. %	7.7	8.3
	0.1 wt. %	11.5	15.2
	0.15 wt. %	4.02	6.5
ΔP	Base fluid	9.3	12.0
	0.05 wt. %	12.2	13.8
	0.1 wt. %	13.6	15.2
	0.15 wt. %	10.6	12.9

For evaluating the exact overall thermal performance of NPG/ CMC aqueous, pressure drop and heat transfer characteristics should simultaneously be considered. According to Fakheri (2010) if and how the second law can advantageously be used in heat exchanger design is still an open question. Here a simple way is applied to define a new performance index (PI) that can be determined directly from experimental data. For this purpose, beside the heat flow rate of test fluid, the pumping power is defined as

$$W_{pump} = \eta_{pump} \frac{\dot{m}}{\rho_h} \Delta P \tag{3}$$

and regardless of pump efficiency, the PI can be estimated by the ratio of heat transfer at hot side to its pumping power as

$$PI = \frac{Q_h}{W_{pump}} = \frac{\rho_h C_{p,h} \Delta T_h}{\Delta P} \tag{4}$$

All of the right-hand side terms are determined experimentally and PI is a dimensionless number proportional to heat transfer rate and inversely proportional to pumping power, therefore higher PI will denote better heat exchanger. In addition, the relation of PI/PI_0 where subscript 0 denotes the conditions before change, can be considered as a criterion for improvement extent. For example, Figure 4 compares PI of the fluids used in this work with relative waviness 0.3.

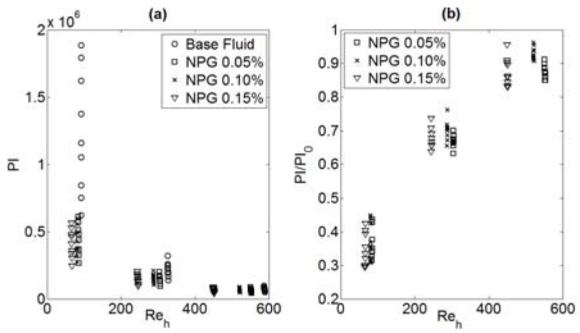

Figure 4: (a) PI of non-Newtonian fluids (the base fluid) and different concentration of non-Newtonian nanofluids (2A/2L=0.3), (b) PI/PI_0 of non-Newtonian fluids (the base fluid) and different concentration of non-Newtonian nanofluids(2A/2L=0.3)

According to the results, PI of nanofluids is always less than base non-Newtonian fluid. However, PI/PI0 increases against Re and have the highest value at 0.1 wt. % NPG. In conclusion, although the final choice will be based on other factors like relative waviness, size, weight, cost, durability, maximum pressure, etc. nevertheless, PI and PI/PI0 help the designer to have the best design.

6. Conclusions

Thermal–hydraulic characteristics of a wavy mini channels heat exchanger (WMCHS) using the NPG / CMC aqueous non-Newtonian nanofluid is investigated by an experimental and numerical approach. The effects of geometrical parameter (wave-length and wave-amplitude), nanoparticles weight fraction, and mass flow rate are examined. The main results are summarized as follows:

I. The results of the experiments have shown that the accuracy of the numerical simulation model is satisfactory. There is a good agreement between experimental and numerical data. A single-phase flow model can simulate the flow and heat transfer properties.

II. NPG/ CMC aqueous non-Newtonian nanofluids have higher forced convective heat transfer coefficient in comparison with CMC aqueous, however presence of NPG caused a penalty for pressure drop.

III. According to the introduced performance index, the results showed that there is a desirable concentration and relative waviness that needs to be carefully determined. In this work, we tested only three concentrations (0.05, 0.1 and 0.15 wt. %). Results showed that the best thermal performance of this concentration can be obtained in the 0.1 wt. % and relative waviness 0.3 also applying the compound technique studied in this investigation (corrugated mini channels + non-Newtonian nanofluid) can be a good choice in practical applications to enhance the heat transfer performance of MCHSs, However, further studies are needed to accurately determine the optimal concentration for graphene Nano porous Nano particle.

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