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Improving the rate of solar heat recovery from nanofluids by using micro heat exchanger

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ABSTRACT

This work investigates the possibility of improving the solar heat recovery from copper–water nanofluids using a micro heat exchanger. This micro heat exchanger consisted of double brass tubes with a micro annular space. The effect of nanoparticles addition to the base fluid and its flow rate on the heat transfer characteristics and heating rate for both nanofluids in solar simulator section and water in shell side of micro heat exchanger were studied. The nanofluids was forced to flow in the tube side, while the fluid in the shell side was pure water. The effect of copper (Cu) weight fraction in nanofluids and the flow rate of nanofluids on the heat transfer coefficient (h), overall heat transfer coefficient (U), thermal resistance (R), and heating rate for both nanofluids in heating section and cold water in micro heat exchanger were studied. Four different concentrations of nanofluids in the range 0.01–0.1 wt.% had been used. The flow rate of nanofluids was changed in laminar region using a control valve. The temperature of heated nanofluids varied from 22 to 73˚C. The heat transfer and overall heat transfer coefficients of the nanofluids 0.1 wt.% were found to increase by 49.9 and 42.08%, respectively, when it was compared with pure water. Also to be increased by increasing the nanofluid flow rate. Also, it was observed that the weight fraction has no significant effect on the final temperature of the nanofluid exit from heating section.

Keywords: Nanofluids; Nano copper; Micro heat exchanger; Heat recovery; Solar energy

1. Introduction

With evolvements of fluid engineering and thermal engineering, many efforts have been dedicated to heat transfer enhancement. Heat transfer fluids such as water, glycol/water mixture, hydrocarbon oil refrigerants phase change fluids, and Silicones play a sprightly role in many industrial processes. These fluids have relatively poor thermal properties when it's compared with the solid particle. As a result, an important need still exists to develop new strategies in order to improve the effective heat transfer behaviors

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of conventional fluids. There are several methods to improve the heat transfer efficiency. Some methods are exploitation of extended surfaces, presentation of vibration to the heat transfer surfaces, and procedure of microchannels. Heat transfer efficiency can also be improved by increasing the thermal conductivity of the working fluid by suspending nano/micro or larger-sized particle materials in liquids. Nanoparticle has gained popularity compared with millimeter or micrometer-sized particle suspensions as it possesses better long-term stability, avoid clogging of flow channels, and can have intensely increased thermal conductivities. Various methods are used for this purpose, such as a single-step method [1], two-step method [2], and phase transfer method [3].

Two-step method can be conceded as the most economic method to produce nanofluids in large scale. Because nanopowder synthesis techniques have already been scaled up to industrial production levels. Many researchers have focused on studying the effect of nanofluids in heat transfer coefficient. Zamzamian et al. prepared nanofluids to consist of nanoparticles of Al_2O_3 and CuO with ethylene glycol as base fluid [4] also Heris et al. used Al_2O_3/water nanofluids at constant wall temperature to investigation forced convection heat transfer coefficient [5]. The results indicate that improving the thermal properties of fluid, not the only reason that causes the heat transfer enhancement. Copper could be considered one of the most important metals, which can be used in heat transfer as it has high thermal conductivity, so it's attractive to prepare it in nanoscale to disperse it in water for preparing the nanofluids. The studies showed that the heat transfer feature of a nanofluids increases with increase in the volume fraction of nanoparticles. On the other hand, the friction factor for the dilute Cu/water nanofluids was approximately the same as water [6]. Shell and tube heat exchanger under turbulent flow condition was used to the heat transfer characteristics. Farajollahi et al. dispersed γ-Al_2O_3 and TiO_2 in water to form nanofluids [7]. For both nanofluids, the overall heat transfer coefficient at a constant Peclet number increases with nanoparticle concentration compared to the pure water. Duangthongsuket et al. investigated the effect of Reynolds number, the temperature of the nanofluids, and the temperature and flow rate of the heating fluid on the heat transfer coefficient. Flow characteristics for this purpose TiO_2/water nanofluids were used [8]. The results showed that the convective heat transfer coefficient of nanofluids were slightly higher than the convective heat transfer base liquid by about 6–11%. The heat transfer coefficient of the nanofluids increased with an increase in the mass flow rate of the hot water and nanofluids. Also, they found that the temperature of the heating fluid had no significant effect on the heat transfer coefficient of the nanofluid [9].

In this study, the effect of nanofluids Cu/water in enhancing the heat transfer coefficient, overall heat transfer, and heating rate for both the nanofluids and cold water was studied using a double-tube micro heat exchanger under a laminar flow condition.

2. Experimental apparatus

2.1. Experimental setup

An experimental setup was built to study the flow and convection heat transfer features of the nanofluids flowing in a micro heat exchanger as shown in Fig. 1. The schematic diagram of the apparatus was used in this experiment consisted of two sections: heating section and micro heat exchanger section. The heating section was designed to simulate the solar system. The isolated glass box contained the component of the system which consisted of two Tangsrium lamps (250 W), 12 glass tubes (L = 50 cm and D_{in} = 6 mm) laid on nickel chrome sheet as reflector, the tank of nanofluids, and submerged pump. The heat, which produced from lamps, was used to raise the temperature of nanofluids which would force to micro heat exchange section. The micro heat exchanger section was consisting of micro heat exchanger double tubes, four thermocouples K-type, a plunger pump, cooler tank, and a collection tank. The micro heat exchanger was made from two smooth copper tubes with 45 cm length. The inner and outer diameters of the inner and outer tube were 7.88, 9.88, 9.92 and 13.05 mm, respectively. By these dimensions annular space would be average of 20 μm. Counter flow with heating nanofluids was flowing inside the tube, while cold water was flowing in the annular space. The micro heat exchanger section was thermally isolated by asbestos rope promotion in order to minimize the heat loss along the perpendicular direction. Four K-type thermocouples were mounted at the inlet and outlet of each flow to measure its temperature. The nanofluids flow rate was controlled by control valve. The cold water flow rate was adjusted by control the plunger pump. At the beginning of the test, the flow rate of the nanofluids and cold water is measured by determining the time taken for a given volume of fluids to be discharged. During the test run, the inlet and exit temperatures of the cold water and hot nanofluids were measured.

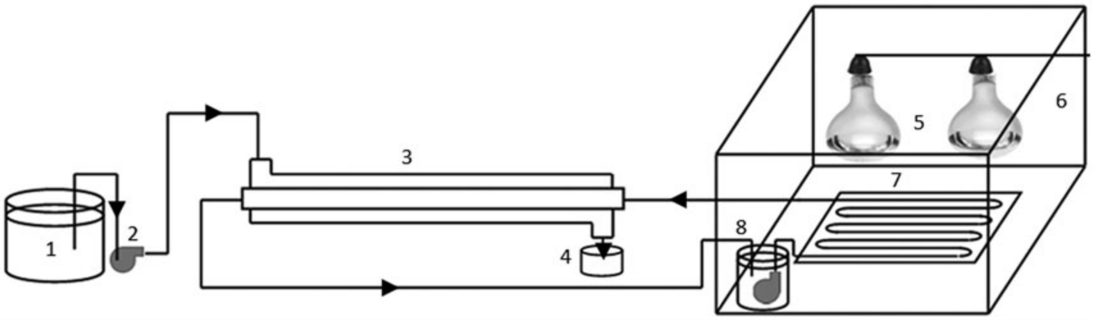

Fig. 1. Schematic diagram of the experimental apparatus.
Notes: (1) cooler tank, (2) plunger pump, (3) micro heat exchanger, (4) collector tank, (5) two tangrium lamp, (6) isolated glass box, (7) twelve glass tube and (8) centrifugal pump.

2.2. Preparation of nanofluids

When preparing nanofluids, there were some special necessary requirements, such as stable suspension for a longer time, low agglomeration of particles to keep in nanoscale, and no chemical change of the fluid and nanoparticle. In this study, nano copper powder was selected to prepare Cu–water nanofluids due its high thermal conductivity (401 W/m K) [10]. A previous study proves that dispersion of the nano copper in ethylene glycol with different concentration could enhance the thermal conductivity of a base fluid up to 40% [11]. Cu powder was produced by reduction method and the crystal structure of copper was indicated by XRD pattern as shown in Fig. 2. The nanofluids with different particle weight fractions (0.01, 0.05, 0.075, and 0.1 wt.%) were used to test the effect of the nanoparticle concentration on the enhancement of the heat convection coefficient. The ultrasonic vibration with additional surfactant method was applied to obtain the Cu–water nanofluids. Ascorbic acid with low concentration (20 wt.%) was used as a surfactant to enhance the stability of the suspension. First, ascorbic acid was mixed with water under an ultrasonic wave for 15 min then specific amount of Cu powder was added to the water base fluid. Finally, ultrasonic vibrate was applied to the above mixture continuously for 2 h to ensure complete dispersion. The physical properties such as the density, viscosity, specific heat, and thermal conductivity of the nanofluids and water flow in shell side were calculated at a bulk temperature of each fluid. A nanofluids stability was measured by using sedimentation method. At the end of heat transfer experiment, XRD test was used to predict the chemical stability of powder.

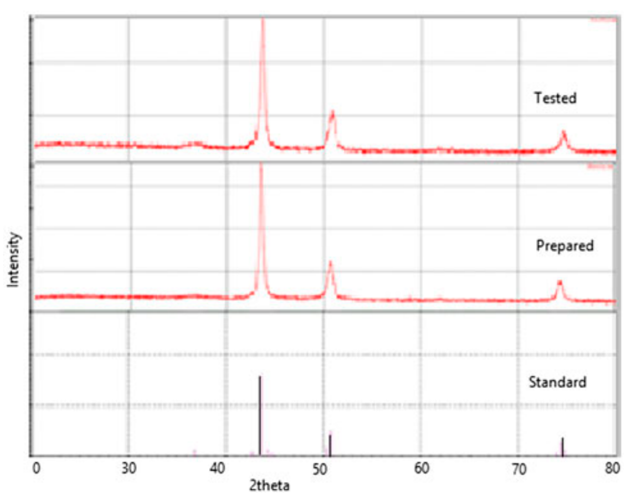

Fig. 2. XRD for copper sample.

2.3. Correlation for thermo-physical properties of nanofluids

To calculate the thermo-physical properties of nanofluids the following published correlations were used [12,13]:

$$\rho_{\mathrm{nf}} = \phi\rho_{\mathrm{p}} + (1-\phi)\rho_{\mathrm{w}} \tag{1}$$

$$\mu_{\mathrm{nf}} = (1+2.5\phi)\mu_{\mathrm{w}} \tag{2}$$

$$(\rho C_{\mathrm{p}})_{\mathrm{nf}} = \phi(\rho C_{\mathrm{p}})_{\mathrm{p}} + (1-\phi)(\rho C_{\mathrm{p}})_{\mathrm{w}} \tag{3}$$

$$K_{\mathrm{nf}} = ((k_{\mathrm{p}} + 2k_{\mathrm{w}} + 2\phi(k_{\mathrm{p}} - k_{\mathrm{w}})/(k_{\mathrm{p}} + 2k_{\mathrm{w}} - \phi(k_{\mathrm{p}}k_{\mathrm{w}}))/k_{\mathrm{w}} \tag{4}$$

where ϕ is the volume fraction of the nanoparticles, ρ_{nf}, ρ_{p} and ρ_{w} are the density of the nanofluids, nanoparticles, and base fluid (kg/m^3) respectively, μ_{nf} and μ_{w} is the viscosity of the nanofluids and base fluid (kg/m s). $C_{\mathrm{p_{nf}}}$, $C_{\mathrm{p_p}}$ and $C_{\mathrm{p_w}}$ the heat capacity of the nanofluids, nanoparticles, and base fluid (kJ/kg K), respectively, K_{nf}, k_p and k_{bf} are the thermal conductivity of nanofluids; nanoparticle and base fluid (W/m K), respectively. These correlations are applicable to spherical particles in volume fractions of less than 5.0 vol.%.

2.4. Numerical study for laminar flow

The heat transfer rate from the heated nanofluids is defined as:

$$Q_{nf} = m_{nf} C_{p_{nf}} (T_{hin} - T_{hout})_{nf} \tag{5}$$

$$Q_c = m_w C_{p_w} (T_{cout} - T_{cin})_w \tag{6}$$

$$Q_{avg} = \frac{Q_c + Q_{nf}}{2} = \mathrm{UA}\Delta T_{LM} \tag{7}$$

$$\Delta T_{LM} = \frac{(T_{hin} - T_{cout}) - (T_{hout} - T_{cin})}{\ln \frac{(T_{hin} - T_{cout})}{(T_{hout} - T_{cin})}} \tag{8}$$

where Q_{nf} and Q_c are the heat transfer rate lose by hot nanofluids and gain to cold water (W), m_{nf} and m_w is the mass flow rate of nanofluid (kg/s), T_{hin}, T_{hout}, T_{cin}, and T_{cout} are the temperature of inlet and outlet temperature of nanofluid and cold water (K), respectively, U is overall heat transfer coefficient (W/m^2 K), A is the outer area of inner tube (m^2). By assuming the temperature of wall as the average between the bulk temperature of nanofluid and cold water, the measured heat transfer coefficient, and Nusselt number of the nanofluid are calculated from the following equation:

$$h_{nf} = \frac{q_{nf}}{(T_w - T_{bh})} \tag{9}$$

$$T_w = \frac{(T_{bc} + T_{bh})}{2} \tag{10}$$

$$\mathrm{Nu}_{nf} = \frac{h_{nf} \times D}{K_{nf}} \tag{11}$$

where h is heat transfer coefficient (W/m^2 K), T_w is a wall temperature, $T_{bh} = (T_{hin} + T_{hout})/2$ and $T_{bc} = (T_{cin} + T_{cout})/2$, q_{nf} is heat flux of nanofluid (W/K m^2), Nu is Nusselt number, D is the inner diameter of inner tube (m). Before starting to determine the convective heat transfer coefficient accuracy of the experimental system was estimated by using water as the working fluid. Results of the experimental heat transfer coefficient were compared with those obtained from the following equation which is defined as follows [13]:

$$\mathrm{Nu} = 0.57\,\mathrm{Re}^{0.5}\,\mathrm{Pr}^{0.35} \tag{12}$$

$$\mathrm{Re}_{nf} = \frac{DU\rho_{nf}}{\mu_{nf}} \tag{13}$$

$$\mathrm{Pr}_{nf} = \frac{C_{p_{nf}} \mu_{nf}}{K_{nf}} \tag{14}$$

where Re is Reynolds number and Pr is Prandtl number.

3. Result and discussion

3.1. Chemical stability of copper in the nanofluids

XRD was used to check the chemical stability of the produced copper after experiment. The result was compared to the standard. Fig. 2 and Table 1 approve that the sample of freshly produced and tested copper matches very closely with the standard data of the copper. This means the capability to use the published correlation to determine the thermos-physical properties of nanofluids. Thermos-physical properties of Cu/water nanofluids were calculated and listed in Table 2 using Eqs. (1)–(4) at T_{hin}.

As listed in Table 2 thermo-physical properties values for nanofluids were calculated from previous mentioned correlation where thermos-physical properties for water could be considered constant at certain temperature (ρ_w = 982 kg/m^3, $\mu_w = 0.466 \times 10^{-3}$ kg/m s and C_p = 4.183 kJ/kg K).

To make the comparison for the results of convective heat transfer using nanofluids, a similar experiment was done using pure water as the working fluid. Fig. 3 shows the experimental results of the pure water in laminar flow regime prediction from Eq. (12). It was appeared the good coincidence between the experimental results and calculated values for water reveals that the precision of the experimental system was considerably high. The uncertainty of the experimental system was less than seven percent.

As shown in Fig. 3, the Nusselt number and heat transfer convection were increased with increase in the Reynolds number i.e. at Re = 177 the Nu and h were 10 and 847 (W/m^2 K), respectively and at Re = 1,627 the Nu and h were 38.05 and 2,503 (W/m^2 K), respectively which means there was an increase in Nu by 73.7% and h by 66%.

Table 1
Standard, prepared and tested diffraction angles of Cu specimen [8]

Standard diffraction angle (2θ°)	Prepared diffraction angle (2θ°)	Tested diffraction angle (2θ°)
43.29	43.56	43.65
50.433	50.68	50.76
74.13	74.32	74.41

Table 2
Theoretical thermos-physical properties of Cu/water nanofluids

Weight fraction (%)	Volume fraction, ϕ (%)	Density of nanofluids, ρ_{nf} (g/cm^3)	Heat capacity of nanofluids, C_{nf} (J/g K)	Thermal conductivity, K_{nf} (W/m K)	Viscosity of nanofluids, μ_{nf} (kg/m s)
0.01	0.0011	0.99	4.18	261.49	0.00049
0.05	0.0059	1.026	3.99	261.5	0.00049
0.075	0.00816	1.05	3.91	261.503	0.000498
0.1	0.01	1.07	3.83	261.506	0.0005

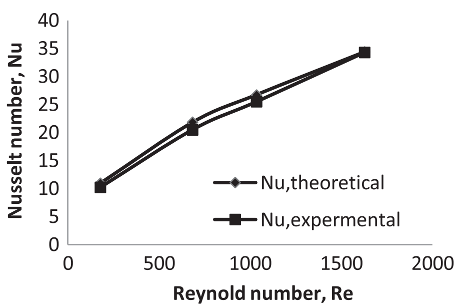

Fig. 3. Comparison between measured Nusselt number and calculated from Eq. (12).

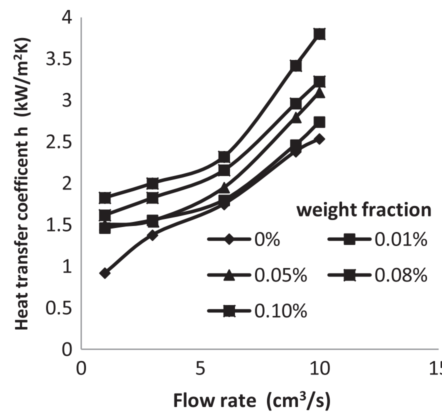

Fig. 4. Comparison between heat transfer coefficient obtain from base fluid (water) and Cu/water nanofluids vs. flow rate of nanofluids with different weight fraction (0.01, 0.05, 0.075 and 0.1 wt.%).

In the present study, Cu–water nanofluids were produced by mixing Cu nanoparticles with the water by several weight fractions. This nanofluids were used to investigate the effect of the Reynolds number and temperature of the flowing nanofluids on the heat transfer characteristics of the nanofluids. The experimental conditions that were used in this study are as follows: the temperature of the nanofluids was heated from 22 to 75℃, The mass flow rates of the hot nanofluids ranged from 1 to 10 cm^3/s, and the temperature of inlet cold water was 22℃.

Within these conditions, the Reynolds numbers ranged from 147 to 1,505 which indicated that the flow was laminar. The effect of nanofluids flow rate and the concentration of nanoparticle on the heating rate for this nanofluids and cooling water was studied as shown in Figs. 4 and 5, the heat transfer coefficient increased with a flow rate of nanofluids raising. It could be clearly seen that the convective heat transfer coefficient of the nanofluids was higher than that of the base fluid (water) at a given flow rate. The possible reason for this enhancement might be associated with the following: (1) the thermal conductivity of nanofluids was greater than base fluid due to present of nanopowder and (2) heat exchanger between the nanoparticles and base fluid due to a chaotic movement of nanoparticles [14]. As shown in Fig. 5, for 0.01 and 0.1 wt.% the ratio of the convective heat transfer coefficient of the nanofluids to pure water increased from 1.02 (2.7%) to 1.32 (32.5) under the same flow rate (6 cm^3/s). This means that the nanofluids had a higher heat transfer coefficient than pure water and this value was increased by increasing the

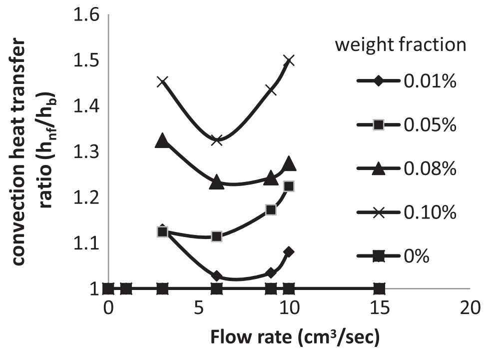

Fig. 5. Heat transfer coefficient ratio (h_{nf}/h_b) vs. flow rate of nanofluids.

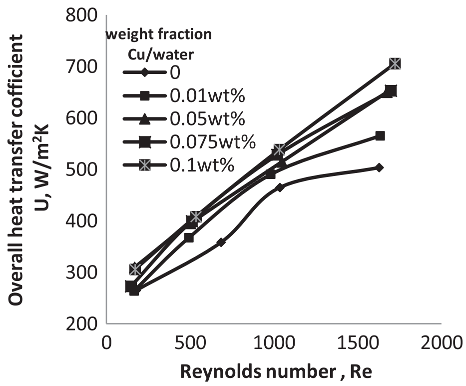

Fig. 6. Overall heat transfer coefficient of pure water and Cu-water nanofluids vs. Reynolds number at different weight fraction.

weight fraction of nanoparticles. The maximum heat transfer enhancement obtained in this experimental was 49.9% which was achieved with the maximum applied flow rate 10 cm^3/s and maximum weight fraction 0.1 wt.%.

The overall heat transfer and thermal resistance vs. the Reynolds number for pure water and nanofluids with different particle weight fraction of Cu–water were calculated from the average heat transfer rate and overall heat transfer coefficient (U) as shown in Figs. 6 and 7.

For the pure water and the Cu/water nanofluids, the overall heat transfer increased and the thermal resistance decreased with increase in the Reynolds number as shown in Fig. 6. When making a comparison from the overall heat transfer for pure water and nanofluids with 0.1 wt.%, it was found that U value was increased from 503.866 to 715.94 W/m^2 (42.08%) at the same flow rate (10 cm^3/s). This result gave a good indicator for using the micro heat exchanger for testing the ability to enhance the heat transfer by

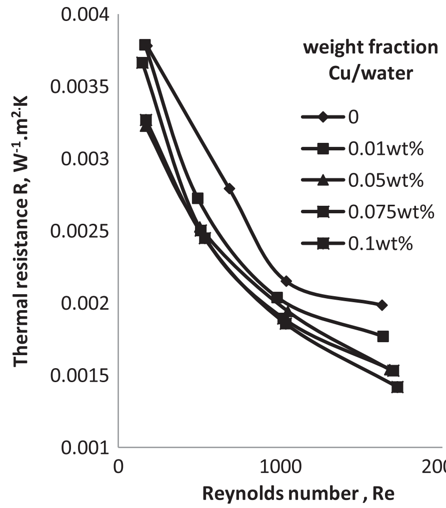

Fig. 7. Thermal resistance pure water and Cu/water nanofluids vs. Reynolds number at different weight fraction.

using nanofluids. The improvement in the performance of the nanofluids results not only from its high thermal conductivity, but also from the random movement and dispersion effect of nanoparticle which reduced the thermal boundary layer. The decreasing rate of thermal resistance with increasing Reynolds number was relatively fast at the lower Reynolds number, but became slow with an increase in the Reynolds number. The thermal resistance decreased with increase in the weight fraction of nanoparticles. As illustrated in Fig. 7 the thermal resistance decreased from 0.00198 to 0.00142 $m^2\,W^{-1}$ (28.2%) with increase in the weight fraction from 0% (pure water) to 0.1% at the same flow rate (10 cm^3/s).

The effects of the concentration and flow rate of nanofluids on the final temperature of heated fluid, heating and cooling rate were studied. As show in Fig. 8 the increase in the weight fraction of nanoparticles had inconsiderable effect on the final temperature of heated fluid. On the other hand, as seen in Figs. 9–11 increase in the weight fraction of nanoparticles and decrease in the flow rate of heated fluid, increase the final temperature of heated fluid and increase the heating rate for both nanofluids and cooling water in micro heat exchanger. For example, changing in flow rate for pure water from 1 to 10 cm^3/s led to increasing in the final temperature from 59 to 74°C. This increase in final temperature might refer to by lowing the flow rate of fluid it would provide spending enough time to gain heat emission from Tangsrium lamps in heating section which laid obtaining higher temperature at steady state and also the decreasing flow rate would withdraw sufficient

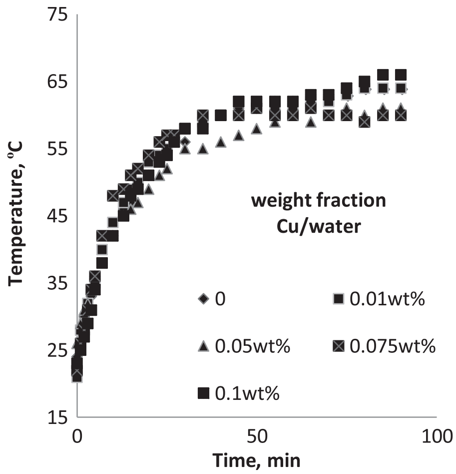

Fig. 8. Temperature of tube in nanofluids with time at different weight fraction of Cu/water nanofluids.

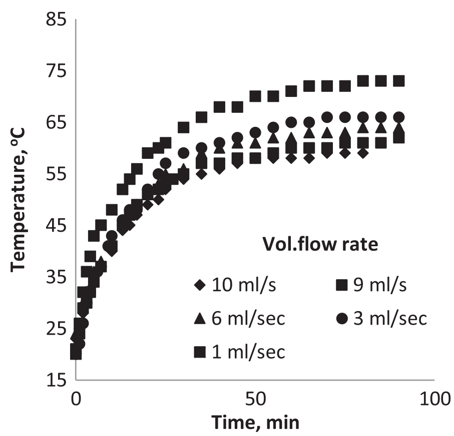

Fig. 9. Final outlet temperature of pure water in tube with time at different flow rates.

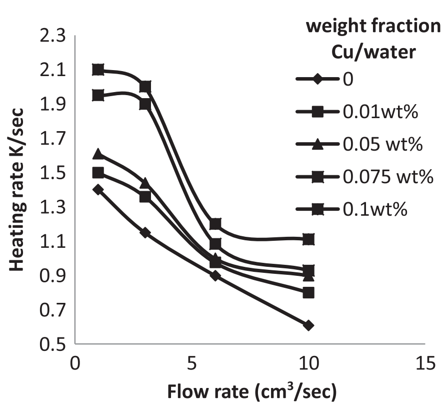

Fig. 10. Temperature of tube in fresh water with time at different flow rates (11, 9, 6, 3 and 1 cm^3/s).

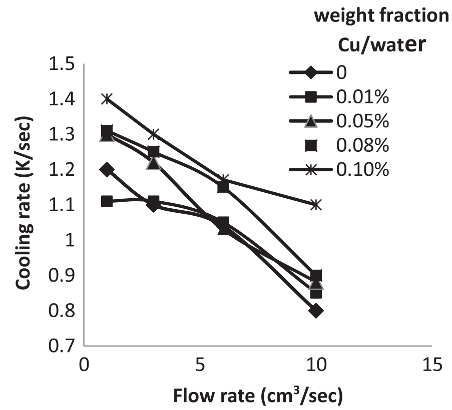

Fig. 11. Temperature of tube in nanofluids with time at different weight fractions of Cu/water nanofluids.

time to let the nanofluids transfer heat with cooling water in micro heat exchanger also increase the weight fraction of nanoparticles in base fluid enhancement its heat transfer properties which lead to reach to certain temperature at shorter time i.e. by increase the weight fraction of Cu from 0 to 0.1% the time needed to reach to 333 K is reduce from 60 to 35 s at 10 cm^3/s.

4. Conclusion

The convective heat transfer performance and flow characteristic of a Cu/water nanofluids flowing in a micro heat exchanger were experimentally investigated. Experiments were carried out under laminar flow conditions. The effects of the flow rate of nanofluids and the weight fraction of nanoparticle on the convection heat transfer coefficient, overall heat transfer coefficient, thermal resistance, and heating rate for both shell and tube fluid was investigated. The following conclusions have been obtained: (1) the heat transfer coefficient of the Cu/water nanofluids are significantly higher than those of the pure water. The maximum heat transfer coefficient enhancement was obtained in this study 49.9% which was achieved with the maximum flow rate 10 cm^3/s and maximum weight fraction 0.1 wt.%, (2) by using the micro heat exchanger the increase in weight fraction of nanoparticle and Reynolds number lead to increase in overall heat transfer coefficient and decrease in the thermal resistance. For 0.1 wt.% of nanoparticle with higher flow rate the enhancement in overall heat transfer could be reached to 42.08% and the thermal resistance will be 28.2%, when it is compared with pure water, (3) in micro heat exchanger lowering flow rate and higher weight fraction of nanoparticle improve the heating rate for hot and cold fluid.

Additional work is required to investigate the effect of different annular space in micro heat exchanger on the overall heat transfer coefficient, the thermal resistance and the heating rate of shell fluid.

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