

1 **Impact of variable fluid properties on forced convection of Fe₃O₄-CNT/water**
2 **hybrid nanofluid in a double-pipe mini-channel heat exchanger**

3
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9
10 **Abstract**

11 The objective of this study is to assess the hydrothermal performance of a non-Newtonian hybrid
12 nanofluid with temperature-dependent thermal conductivity and viscosity compared with a
13 Newtonian hybrid nanofluid with constant thermophysical properties. A counter-current double-
14 pipe mini-channel heat exchanger is studied to analyze the effects of hybrid nanofluid. The
15 nanofluid is employed as the coolant in the tube side while the hot water flows in the annulus side.
16 Two different nanoparticles including Tetra Methyl Ammonium Hydroxide (TMAH) coated Fe₃O₄
17 (magnetite) nanoparticles and Gum Arabic (GA) coated Carbon Nanotubes (CNTs) are used to
18 prepare the water based hybrid nanofluid. The results demonstrated that the non-Newtonian hybrid
19 nanofluid always has a higher heat transfer rate, overall heat transfer coefficient, effectiveness, and
20 performance index than those of the Newtonian hybrid nanofluid, while the opposite is true for
21 pressure drop and pumping power. Supposing that the Fe₃O₄-CNT/water hybrid nanofluid is a
22 Newtonian fluid with constant thermal conductivity and viscosity, leads to a large error in the

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23 computation of pressure drop (1.5-9.71%), pumping power (1.5-9.71%), and performance index
24 of heat exchanger (1.86-11.25%), whereas the errors in the computation of heat transfer rate,
25 overall heat transfer coefficient, and effectiveness aren't considerable (less than 2.91%).

26

27 **Keywords:** non-Newtonian hybrid nanofluid; Double-pipe heat exchanger; Magnetite; Carbon
28 nanotube; convective heat transfer

29

30 **1. Introduction**

31 Double-pipe heat exchangers have been widely employed in various applications to exchange heat
32 between two fluids called as heat transfer fluids [1, 2]. They are an essential part of almost all the
33 industries, including the oil and gas industry, power generation, refrigeration, and nuclear power.
34 Due to the great importance of heat exchangers, improving their efficiency is a very important
35 issue. So far, several methods have been proposed in the literature to enhance heat exchanger
36 performance such as using various fins and turbulators. However, these modifications offer several
37 disadvantages like increase in pressure drop, weight and volume of heat exchangers that limit their
38 usage.

39 Over the past decade, scientists and researchers around the world have revealed that the heat
40 exchanger performance can be considerably enhanced by improving the thermal conductivity of
41 working fluids [3, 4]. This goal can be achieved through the use of nanofluids, which are prepared
42 by suspending nanoparticles with sizes typically of 1-100 nm in conventional heat transfer fluids
43 such as water, oil, and ethylene glycol [5-8]. This term was first suggested by Choi [9] in 1995,
44 and it has since gained in popularity [10-17].

45 A great number of experimental and numerical works have been performed on the various aspects
46 of different nanofluids performance in double-pipe heat exchangers [18-22]. Maddah et al. [23]
47 experimentally evaluated the effects of Al_2O_3 -water nanofluid on the performance of a horizontal
48 double-pipe heat exchanger under turbulent flow regime and showed 52% and 12% enhancements
49 in the friction factor and heat transfer rate, respectively. Mousavi et al. [24] numerically studied
50 the effect of a variable magnetic field on the hydrothermal characteristics of Fe_3O_4 -water nanofluid
51 flowing through a sinusoidal double-pipe heat exchanger and reported the enhancement of Nusselt
52 number in the presence of magnetic field. Saeedan et al. [25] numerically examined the effect of
53 Cu-water, CuO-water and CNT-water nanofluids on the performance of a finned type heat
54 exchanger. They found that both the Nusselt number and pressure drop intensify with increasing
55 nanoparticle concentration. Sarafraz et al. [26] experimentally studied the use of CNT-water
56 nanofluid inside a double-pipe heat exchanger. They assessed the impact of different effective
57 parameters on the convective heat transfer coefficient in laminar and turbulent flow regimes and
58 found that the proposed nanofluid can enhance the heat transfer by almost 44% compared with the
59 pure water. Kumar et al. [27] experimentally surveyed the effect of Fe_3O_4 -water nanofluid on the
60 performance of a double pipe heat exchanger with a longitudinal fin with return band under
61 turbulent flow regime. They showed the enhancement of Nusselt number with increasing the
62 Reynolds number and nanoparticle concentration. Hussein [28] experimentally examined the flow
63 of Aluminum Nitride- ethylene glycol nanofluid through a double-pipe heat exchanger and showed
64 the increase of Nusselt number with increasing the flow rate and volume concentration of
65 nanofluid. Shirvan et al. [29] studied the influence of Reynolds number and nanoparticle
66 concentration on the performance of Al_2O_3 -water nanofluid inside a double-pipe heat exchanger

67 and showed the enhancement of Nusselt number with increasing the Reynolds number and
68 decreasing the nanoparticle concentration.

69 To enhance the rate of heat transfer, hybrid nanofluids has attracted lots of attention using a
70 combination of different nanoparticles in the nanofluids in order to take the advantage of them
71 [30-36]. Esfe et al. [37] experimentally studied the thermal conductivity of ethylene glycol based
72 hybrid nanofluid containing ZnO-CNT nanoparticles. They showed the improvement of thermal
73 conductivity using ZnO and CNT nanoparticles compared with the base fluid and developed a new
74 correlation for the calculation of thermal conductivity based on the experimental data using an
75 artificial neural network (ANN).

76 The combination of Fe_3O_4 with CNT nanoparticles is widely used as a promising hybrid nanofluid.
77 Baby and Sundara [38] studied the effects of nanoparticles concentration on the thermal
78 conductivity of Fe_3O_4 -CNT/water hybrid nanofluid and reported 6.5-10% improvement in the
79 thermal conductivity of nanofluid in the temperature range of 30-50 °C compared with the base
80 fluid. Felicia and Philip [39] investigated an oil-based Fe_3O_4 -CNT hybrid nanofluid in the presence
81 of a magnetic field and showed the enhancement of viscosity with increasing magnetic field
82 intensity. Sundar et al. [40] experimentally assessed the hydrothermal characteristics of Fe_3O_4 -
83 CNT/water hybrid nanofluid in a circular tube and presented 14.8% improvement in the Nusselt
84 number using nanofluid with concentration of 0.3% at Reynolds number of 3000. Shahsavar et al.
85 [41] studied the use of Fe_3O_4 -CNT/water hybrid nanofluid in a heated tube in the presence of both
86 constant and alternating magnetic fields. They showed higher improvement of heat transfer using
87 a constant magnetic field compared with an alternating one. Harandi et al. [42] conducted
88 experiments to determine the thermal conductivity of Fe_3O_4 -CNT/EG hybrid nanofluid at different

89 temperatures and found the improvement of thermal conductivity with increase in temperature and
90 nanoparticle concentration.

91 In most of the previous research works on the performance of heat exchangers containing various
92 nanofluids, the thermophysical properties of the nanofluid have been assumed as constant and the
93 nanofluid itself has been considered as Newtonian [43, 44]; while various studies have shown that
94 the thermophysical properties of nanofluids are a function of temperature, and that the majority of
95 nanofluids exhibit a non-Newtonian behavior [45-47]. In this research, we want to see if a
96 significant difference is observed in the performance parameters of a heat exchanger (i.e. pumping
97 power, effectiveness, and performance index) by assuming constant properties and a Newtonian
98 nature for nanofluids. We also want to find out: under what conditions the assumptions of constant
99 properties and Newtonian nature of nanofluid can be used in the analysis of heat exchangers? This
100 is done by comparing the performance parameters of a counter-current double-pipe heat exchanger
101 containing Newtonian Fe_3O_4 -CNT/water nanofluid of constant properties with the performance
102 parameters of a heat exchanger containing the non-Newtonian Fe_3O_4 -CNT/water hybrid nanofluid
103 with temperature dependent thermal conductivity and viscosity, at different Reynolds numbers and
104 concentrations.

105

106 **2. Physical properties of nanofluid**

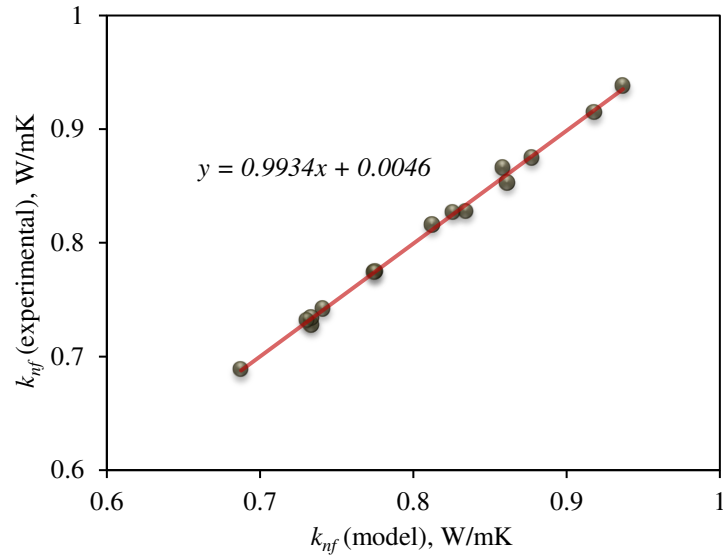
107 This investigation is conducted on a hybrid nanofluid consisting of TMAH coated magnetite
108 nanoparticles and GA coated CNTs. It was prepared by mixing different volume ratios of Fe_3O_4 -
109 water nanofluid and CNT-water nanofluid, followed by 5 min sonication [48]. The detailed
110 description of the preparation method can be found in Refs. [48-50]. The magnetite and CNT

111 nanoparticles are attached physically because of interaction between the molecules of TMAH and
112 GA.

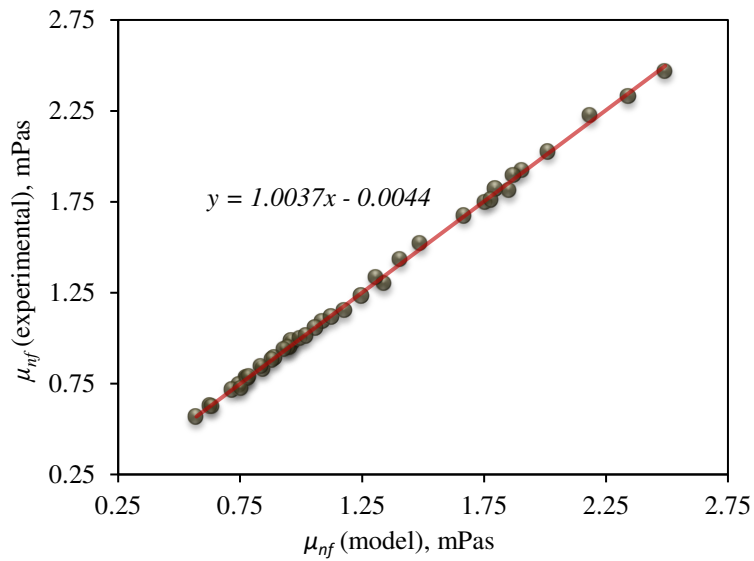
113 After careful preparation and characterization, a series of experiments were performed to evaluate
114 the thermophysical properties of the hybrid nanofluid. The hybrid nanofluid shows the non-
115 Newtonian and Newtonian behaviors at low (up to 70 s^{-1}) and high shear rates, respectively.
116 Additionally, the viscosity of the hybrid nanofluid enhances with increase in volume concentration
117 of nanoparticles, while reduces with increasing the temperature. However, the thermal
118 conductivity increases with temperature and volume concentration.

119 Based on the data obtained from experiments, the artificial neural network (ANN) was used to find
120 a correlation between the thermal conductivity and temperature and volume concentration of Fe_3O_4
121 and CNT nanoparticles [51]. For the viscosity, a correlation is developed as a function of
122 temperature, shear rate, and volume concentrations of Fe_3O_4 and CNT nanoparticles [51]. The
123 acquired neural network models illustrate a good accuracy to predict the thermal conductivity and
124 viscosity according to Fig. 1. The correlations developed are presented in appendix A, and it is
125 clear that the models are temperate dependent.

126



(a)



(b)

Fig. 1. Results obtained from the developed models in comparison with the experimental data: (a) thermal conductivity, (b) viscosity [18].

127

128 The viscosity and thermal conductivity of the considered Newtonian hybrid nanofluids are

129 reported in Table 1. The considered viscosity for the Newtonian nanofluid is equal to the viscosity

130 of the non-Newtonian nanofluid at the same concentrations of CNT and magnetite nanoparticles
 131 at shear rates higher than 70 s^{-1} . Also, the considered thermal conductivity for the Newtonian
 132 nanofluid samples is the same as the thermal conductivity of the non-Newtonian nanofluid at the
 133 inlet temperature of nanofluid.

134

Table 1. Characteristics of the studied Newtonian nanofluid samples.

	$\varphi_{CNT} (\%) = 0.1\%$					$\varphi_{CNT} (\%) = 1.35\%$				
	$\varphi_{CNT} (\%)$					$\varphi_{CNT} (\%)$				
	0.1	0.3	0.5	0.7	0.9	0.1	0.3	0.5	0.7	0.9
$\mu_{nf} \times 10^4 \text{ (kg/ms)}$	8.15	9.48	11.08	12.81	14.48	11.33	13.03	14.61	15.95	17.01
$k_{nf} \text{ (W/mK)}$	0.691	0.725	0.739	0.759	0.794	0.703	0.759	0.772	0.866	0.902

135

136 Moreover, the nanofluid bulk density (ρ_{nf}) and specific heat ($c_{p,nf}$) are computed as:

$$\rho_{nf} = \varphi_M \rho_M + \varphi_{CNT} \rho_{CNT} + (1 - \varphi_M - \varphi_{CNT}) \rho_w \quad (1)$$

$$c_{p,nf} = \varphi_M c_{p,M} + \varphi_{CNT} c_{p,CNT} + (1 - \varphi_M - \varphi_{CNT}) c_{p,w} \quad (2)$$

137 where φ is the volume concentration of nanoparticles and, subscripts M , CNT and w refer to
 138 magnetite, CNT and water, respectively.

139

140 3. Mathematical modelling

141 Due to the small size of nanofluids, they can thus be approximately evaluated as a pure fluid
 142 considering no velocity slip and local thermal equilibrium between the base fluid and
 143 nanoparticles. The governing equations for laminar, steady state forced convection flow of the
 144 studied nanofluid are given as follows:

145 Continuity:

$$\nabla \cdot (\rho_{nf} \mathbf{V}) = 0 \quad (3)$$

146 Momentum:

$$\nabla \cdot (\rho_{nf} \mathbf{V} \mathbf{V}) = -\nabla p + \nabla \cdot (\mu_{nf} \nabla \mathbf{V}) \quad (4)$$

147 Energy:

$$\nabla \cdot (\rho \mathbf{V} c_{p,nf} T) = \nabla \cdot (k_{nf} \nabla T) \quad (5)$$

148 where \mathbf{V} is the velocity, p is the pressure, and T is the temperature.

149 Reynolds number for the flow of nanofluid (Re_{nf}) and water (Re_w) through the tube side and

150 annulus side, respectively, can be calculated as:

$$Re_{nf} = \frac{\rho_{nf} u_{in,nf} (2r_i)}{\mu_{nf}} \quad (6)$$

$$Re_w = \frac{\rho_w u_{in,w} [2(r_o - r_i)]}{\mu_w} \quad (7)$$

151 where $u_{in,nf}$ and $u_{in,w}$ are the inlet velocity of the nanofluid and water, respectively.

152 Considering the fact that the outer wall of the heat exchanger is adiabatic and the problem under

153 consideration is steady state, the rate of heat transfer to the nanofluid from the hot water is equal

154 to that of the hot water according to the conservation of energy ($\dot{Q}_{nf} = \dot{Q}_w = \dot{Q}$) which are

155 obtained as:

$$\dot{Q}_{nf} = \dot{m}_{nf} c_{p,nf} (T_{out} - T_{in})_{nf} \quad (8)$$

$$\dot{Q}_w = \dot{m}_w c_{p,w} (T_{in} - T_{out})_w \quad (9)$$

156 where \dot{m}_{nf} and \dot{m}_w are mass flow rate of the cold nanofluid and the hot water, respectively.

157 The overall heat transfer coefficient is given as:

$$U = \frac{\dot{Q}}{A \Delta T_{LMTD}} \quad (10)$$

158 where A is the internal tube area, and ΔT_{LMTD} is the logarithmic mean temperature difference
 159 computed as:

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} \quad (11)$$

160 where $\Delta T_1 = T_{in,w} - T_{out,nf}$ and $\Delta T_2 = T_{out,w} - T_{in,nf}$.

161 One way of measuring the performance of a heat exchanger is to compute its effectiveness. The
 162 heat exchanger effectiveness is ratio of the actual heat transfer rate to the maximum possible one
 163 given as:

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{\dot{Q}}{C_{min}(T_{in,w} - T_{in,nf})} \quad (12)$$

164 where C_{min} represents the minimum heat capacity rate given as:

$$C_{min} = \min[C_w, C_{nf}] \quad (13)$$

165 Here, C_w and C_{nf} are respectively heat capacity rates of the water and the nanofluid defined as:

$$C_w = \dot{m}_w c_{p,w} \quad (14)$$

$$C_{nf} = \dot{m}_{nf} c_{p,nf} \quad (15)$$

166 The minimum heat capacity rate is obtained for the nanofluid and hence, the effectiveness is
 167 calculated as:

$$\varepsilon = \frac{T_{out,nf} - T_{in,nf}}{T_{in,w} - T_{in,nf}} \quad (16)$$

168 The rate of energy consumption required to pump the nanofluid in the heat exchanger is given as:

$$\dot{W} = \dot{V} \Delta p \quad (17)$$

169 where \dot{V} and Δp denote volumetric flow rate and pressure drop, respectively.

170 To evaluate the heat transfer rate and the pumping power simultaneously, a parameter called
 171 performance index is defined as the ratio of heat transfer rate to the pressure drop given as [52]:

$$\eta = \frac{\dot{Q}}{\Delta p} \quad (18)$$

172

173 **4. Numerical method and validation**

174 ANSYS-FLUENT software is used to solve the governing equations employing the SIMPLE
 175 method for pressure and velocity coupling. The second order upwind method is used to discretize
 176 the convective and diffusion terms using the finite-volume method. The convergence criteria is
 177 also set to 10^{-6} . A structured quad based mesh was used throughout the domain with a more grid
 178 density near the wall. The grid independence study was carried out by considering the numerical
 179 results of six different grid resolutions. The results of this investigation is summarized in Table 2.
 180 It should be noted that the grid resolution was reported as number of longitudinal nodes \times number
 181 of radial nodes in central tube \times number of radial nodes in annulus. So, by comparing the results,
 182 the grid with resolution of $1000 \times 35 \times 35$ was chosen. To verify the present numerical procedure,
 183 the results are compared with the experimental data of Duangthongsuk and Wongwises [53] for
 184 water-TiO₂ nanofluid in a double-pipe heat exchanger shown in Fig. 2. Good agreement between
 185 the present results and Ref. [53] is shown with the maximum error of about 5%.

186

187 **Table 2.** Grid independence study for non-Newtonian Fe₃O₄-CNT/water hybrid nanofluid at $\varphi_M = 0.9\%$, $\varphi_{CNT} =$
 188 1.35% and $Re = 2000$.

Grid	\dot{Q} (W)	Percentage difference	ΔP (Pa)	Percentage difference
800 \times 25 \times 25	33.92		120.1	
900 \times 30 \times 30	35.82	5.6	125.5	4.5
950 \times 30 \times 30	37.06	3.5	129.1	2.9

1000×30×30	37.92	2.3	132.1	2.3
1000×35×35	38.23	0.8	133.7	1.2
1100×35×35	38.51	0.7	134.6	0.67

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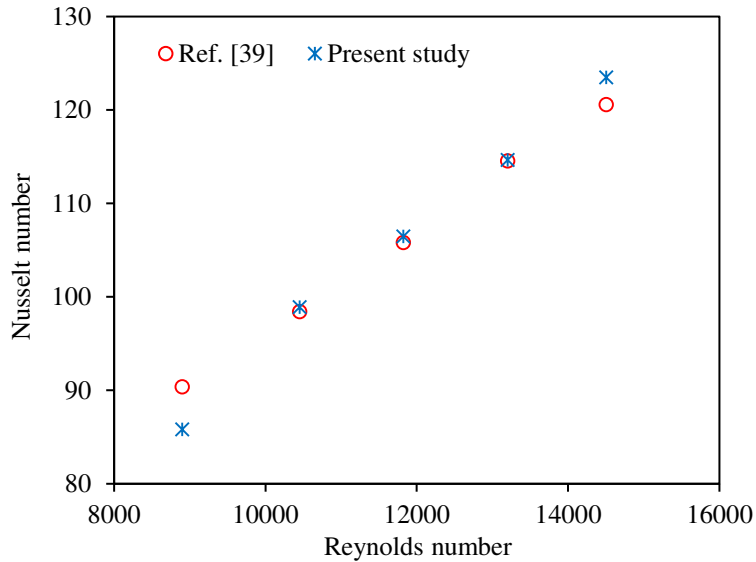


Fig. 2. Comparison between results obtained from present study and experimental results of Ref. [53].

190

191 5. Geometry and boundary conditions

192 The present investigation is conducted in a double-pipe counter-current mini-channel heat
 193 exchanger with the length of 1 m, inner diameter of 1 mm, and outer diameter of 2 mm. The
 194 thickness of the inner tube's wall is neglected. Fig. 3 illustrates the schematic of the geometry
 195 including the flow directions of both hot water and cold nanofluid. Due to the axisymmetric nature
 196 of the problem, only half of the geometry is considered as the computational 2-D domain. For the
 197 outer wall, adiabatic boundary condition is used. Uniform velocity and uniform temperature are

198 also considered at both tube and annulus entrances while zero relative pressure is utilized at the
 199 outlets. Additionally, the no-slip condition is employed on the inner and outer walls.

200

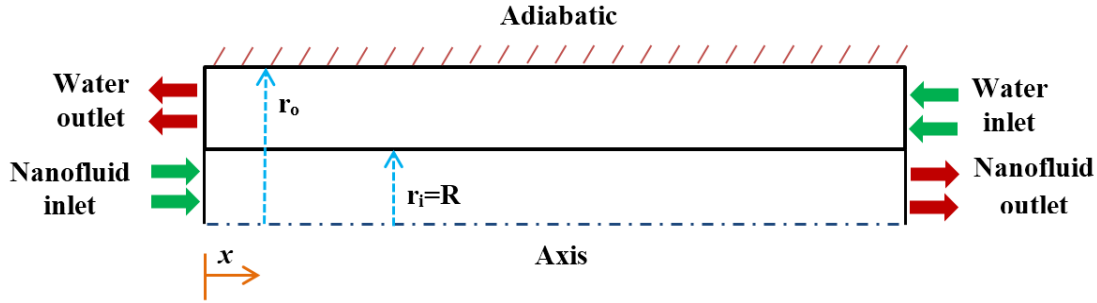


Fig. 3. The mini-channel heat exchanger under study.

201

202 6. Results and discussion

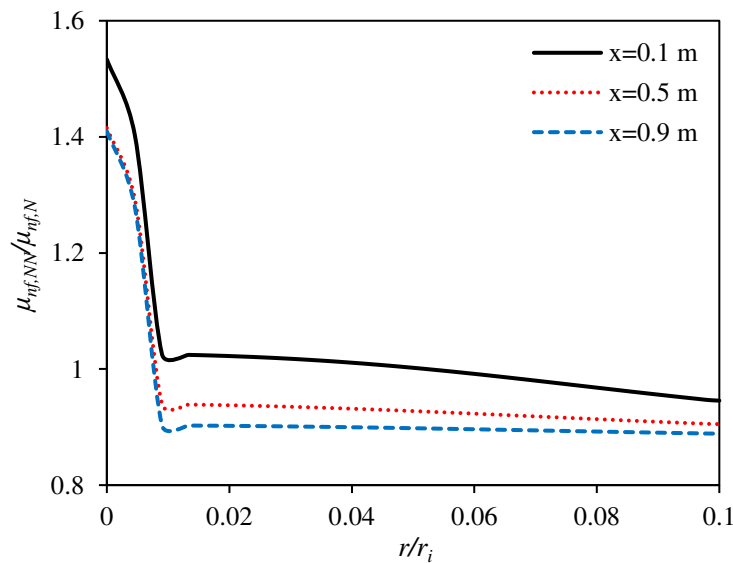
203 In this research, the influences of the shear rate and temperature dependent viscosity and the
 204 temperature-dependent thermal conductivity on the hydrothermal characteristics of Fe₃O₄-
 205 CNT/water hybrid nanofluid flowing inside a double-pipe heat exchanger are evaluated and
 206 compared with those obtained by regarding the hybrid nanofluid as a Newtonian fluid with
 207 constant thermal conductivity and viscosity. The simulations are conducted at magnetite
 208 concentration range of 0.1-0.9%, CNT concentration range of 0-1.35%, Reynolds number range
 209 of 500-2000 for the tube side, and constant Reynolds number of 1000 for the annulus side. The
 210 inlet temperature of the nanofluid and water are considered as 298 K and 308 K, respectively. Note
 211 that the results of the non-Newtonian and Newtonian Fe₃O₄-CNT/water hybrid nanofluids will be
 212 reported by letters ‘NN’ and ‘N’, respectively.

213 Fig. 4 illustrates the variations of viscosity ratio ($\mu_{nf,NN}/\mu_{nf,N}$) for $\varphi_M = 0.7\%$ and $\varphi_{CNT} = 0.7\%$
 214 at three different cross sections (i.e. $x=0.1$ m, $x=0.5$ m, and $x=0.9$ m). For $Re = 500$, by
 215 increasing the distance from the tube axis, viscosity of the non-Newtonian hybrid nanofluid

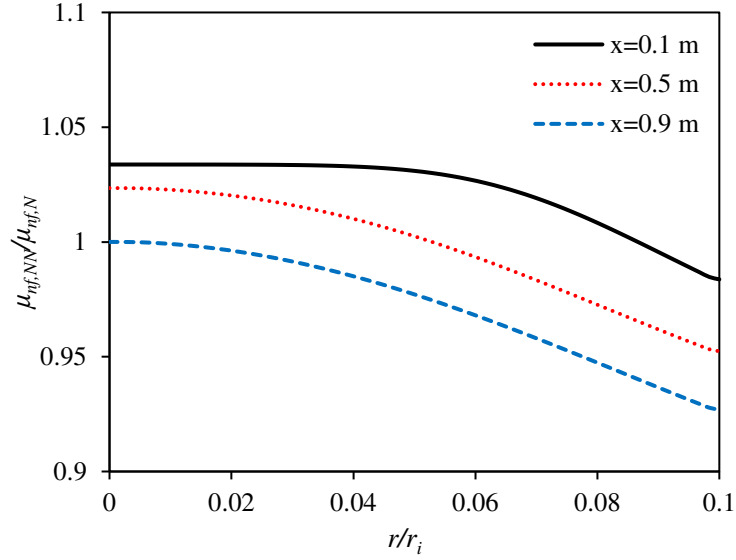
216 diminishes severely at first, and then its descending trend continues at a milder slope, and degree
217 of variations increases with increase in distance from the tube inlet. Near the tube axis, due to
218 small values of shear rate and temperature, viscosity is high. However, by moving away from the
219 tube axis toward the tube wall, both shear rate and temperature increase and consequently,
220 viscosity reduces. The results for $Re = 2000$ indicate that by moving away from the central
221 regions of tube toward the tube wall, viscosity reduces and degree of viscosity variation is lower
222 than that for $Re = 500$. By increasing the Reynolds number at a fixed concentration, the thickness
223 of velocity boundary layer reduces and therefore, the velocity gradient increases. Therefore, there
224 are two reasons for the negligible changes of viscosity in central regions of tube at cross-section
225 $x=0.1$ m. The first reason is that the shear rate is greater than 60 s^{-1} at most of points of this section,
226 and consequently, fluid viscosity is constant. The second reason is that the thickness of thermal
227 boundary layer in this area is small, which causes constant temperature of the hybrid nanofluid in
228 central regions of tube and thus, viscosity remains unchanged. By moving away from the tube
229 inlet, the thermal boundary layer grows which raises the temperature of the nanofluid in vicinity
230 of the tube wall and, thus, reduces the viscosity. Therefore, the velocity of nanofluid diminishes
231 near the tube wall and increases at the tube axis; i.e. the velocity profile becomes flatter. As a
232 result, the amount of shear rate increases near the tube wall and diminishes near the tube axis;
233 which causes viscosity to diminish near the tube wall and increase near the tube axis. Therefore, it
234 can be said that by moving away from the tube axis, viscosity of the non-Newtonian hybrid
235 nanofluid near the tube wall diminishes; however, its behavior near the tube axis depends on
236 whether the effect of viscosity decrease due to the rise of temperature is greater or the effect of
237 viscosity increase due to the reduction of velocity gradient. Therefore, it is concluded that the
238 effect of temperature increase overcomes the effect of temperature gradient reduction, and

239 viscosity of the non-Newtonian nanofluid diminishes by moving away from the tube inlet.
 240 Furthermore, Figs. 4(a) and 4(b) show that viscosity of the non-Newtonian nanofluid diminishes
 241 with the increase of Reynolds number. This can be justified based on the reduction of the velocity
 242 boundary layer thickness with increasing the Reynolds number, which leads to the increase of
 243 velocity gradient and thus the reduction of fluid viscosity. In addition, the comparison between the
 244 viscosities of the Newtonian and non-Newtonian hybrid nanofluids indicates that in central regions
 245 of the tube, viscosity of the non-Newtonian nanofluid is greater than that of the Newtonian
 246 nanofluid; however, in vicinity of the tube wall, the Newtonian fluid has a higher viscosity and by
 247 moving away from the tube inlet, the region in which viscosity of the Newtonian nanofluid is
 248 greater becomes vaster, since the viscosity of the non-Newtonian nanofluid diminishes by moving
 249 away from the tube wall. Both the temperature and shear rate are higher near the tube wall than
 250 the tube axis. Therefore, both of these factors lead to the viscosity reduction of the non-Newtonian
 251 nanofluids, while the opposite is true near the tube axis.

252



(a)



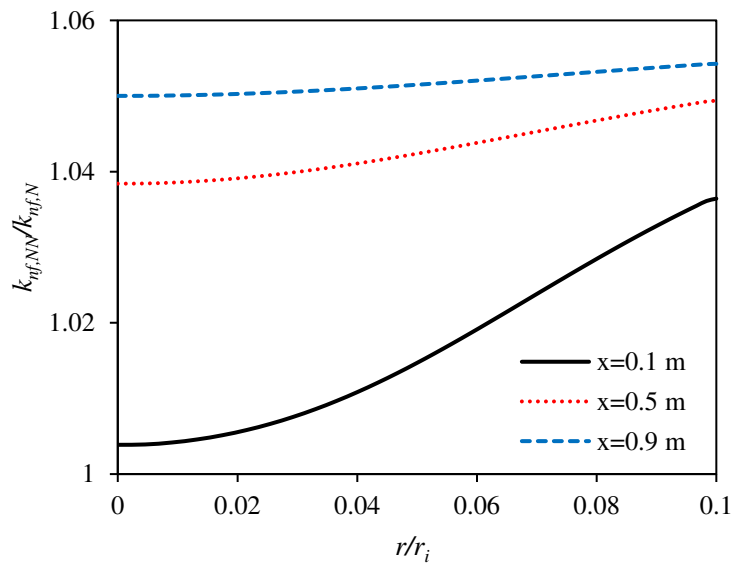
(b)

Fig. 4. Viscosity ratio for $\varphi_M = 0.7\%$ and $\varphi_{CNT} = 0.7\%$ at three different cross sections for (a) $Re = 500$ and (b) $Re = 2000$.

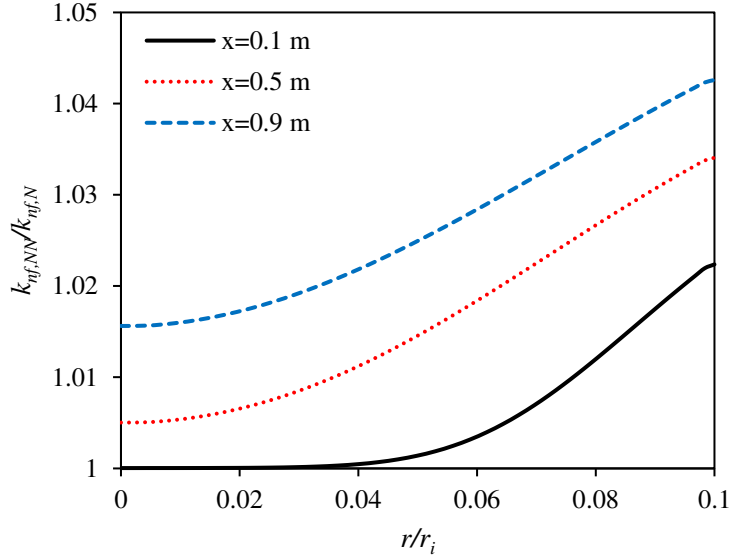
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254 Fig. 5 displays the variations of thermal conductivity ratio ($k_{nf,NN}/k_{nf,N}$) for $\varphi_M = 0.7\%$ and
 255 $\varphi_{CNT} = 0.7\%$ at three different cross sections (i.e. $x=0.1$ m, $x=0.5$ m, and $x=0.9$ m). For $Re =$
 256 500, by moving away from the tube axis toward the tube wall, thermal conductivity of the non-
 257 Newtonian nanofluid increases continually due to the higher temperature of nanofluid near the
 258 wall. The improvement of thermal conductivity with the increase of distance from the tube inlet is
 259 due to the higher nanofluid temperature resulting from the increase of heat transfer to the nanofluid.
 260 Similar observations exist for $Re = 2000$, with the difference that the slope of thermal
 261 conductivity increment near the tube wall is greater for $Re = 2000$. This is due to the rise of
 262 nanofluid temperature near the tube wall, resulting from the lower thermal boundary layer
 263 thickness that occurs because of the flow velocity enhancement. Moreover, the comparison
 264 between thermal conductivity of the Newtonian and non-Newtonian nanofluids shows that thermal

265 conductivity of the non-Newtonian nanofluid is always greater than that of the Newtonian
 266 nanofluid; however, the difference between thermal conductivities of the nanofluids reduces with
 267 the increase of Reynolds number. Considering the fact that the inlet temperature of nanofluid is 25
 268 °C, and the thermal conductivity improves with the rise of temperature, it was predictable for the
 269 thermal conductivity of non-Newtonian nanofluid to always surpass that for the Newtonian
 270 nanofluid. In addition, increasing the Reynolds number reduces the thermal boundary layer
 271 thickness and consequently, the internal layers of nanofluid are affected more slowly by wall
 272 temperature. This reduces the nanofluid temperature and thereby reduces the thermal conductivity
 273 of non-Newtonian nanofluid.
 274



(a)



(b)

Fig. 5. Thermal conductivity ratio for $\varphi_M = 0.7\%$ and $\varphi_{CNT} = 0.7\%$ at three different cross sections for (a) $Re = 500$ and (b) $Re = 2000$.

275

276 Fig. 6 demonstrates the difference between the pressure drop of the Newtonian and non-Newtonian

277 Fe_3O_4 -CNT/water hybrid nanofluids ($d_{\Delta p} = \frac{(\Delta p)_{NN} - (\Delta p)_N}{(\Delta p)_N} \times 100$) in terms of magnetite

278 concentration at different Reynolds numbers. It is seen that the pressure drop of the non-Newtonian

279 nanofluid is always less than that of the Newtonian nanofluid. The minimum pressure drop

280 difference (1.5%) is obtained at $\varphi_M = 0.9\%$, $\varphi_{CNT} = 1.35\%$ and $Re = 2000$, while the maximum

281 difference (9.71%) occurs at $\varphi_M = 0.5\%$, $\varphi_{CNT} = 0.1\%$ and $Re = 500$. Additionally, it is

282 observed that the difference between the pressure drop of the Newtonian and non-Newtonian

283 nanofluids reduces with the increase of Reynolds number. According to Fig. 4, this is caused by

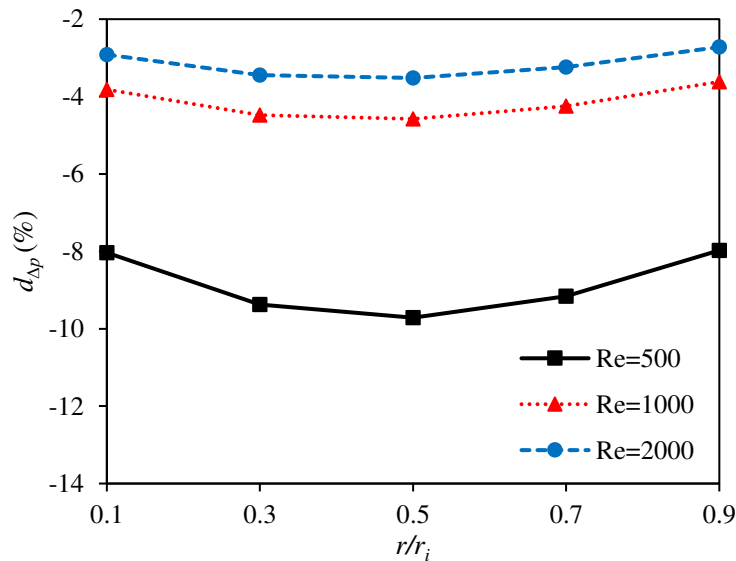
284 the reduction in the difference between the average viscosity of the Newtonian and non-Newtonian

285 nanofluids by increasing the Reynolds number. Furthermore, at $\varphi_{CNT} = 0.1\%$, the pressure drop

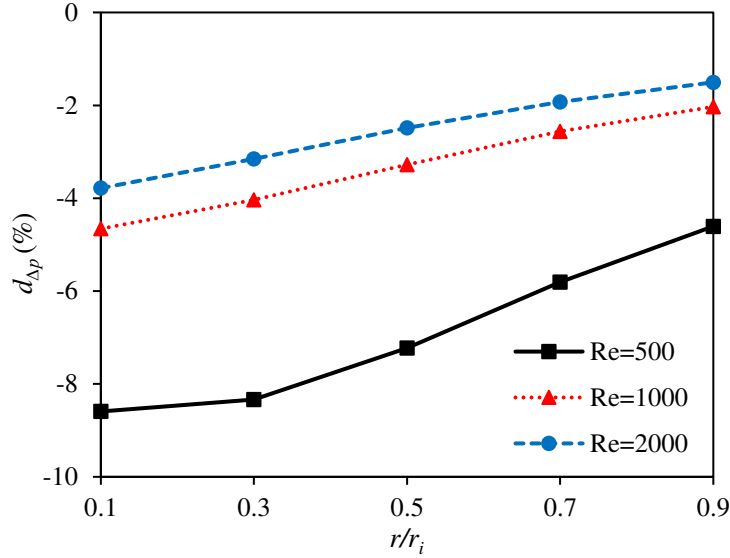
286 difference augments when the magnetite concentration increases from 0.1 to 0.3% and then

287 reduces by the further increment of magnetite concentration; while for $\varphi_{CNT} = 1.35\%$, the
 288 increase of magnetite concentration results in the reduction in the pressure drop difference.
 289 Besides, at $\varphi_M = 0.1\%$, the pressure drop difference rises with increasing the CNT concentration
 290 from 0.1 to 1.35%, while the opposite is true at higher magnetite concentrations. According to
 291 Darcy's equation ($\Delta p = f \frac{L}{2r_i} \frac{\rho u_{in}^2}{2}$, where f is the friction factor defined as $f = \frac{64}{Re}$ [54]) and by
 292 considering the fact that the non-Newtonian and Newtonian nanofluids have the same density and
 293 friction factor at an identical Reynolds number, the difference between the pressure drop of the
 294 Newtonian and non-Newtonian nanofluids is only due to the difference between their viscosities.
 295 It can be concluded from the presented results that the assumption of constant thermal conductivity
 296 and viscosity of the hybrid nanofluid, at a low Reynolds number, leads to large errors in the
 297 computation of pressure drop; however, the obtained error decreases with the increase of Reynolds
 298 number.

299



(a)



(b)

Fig. 6. Pressure drop at different Reynolds numbers in terms of magnetite concentration at (a) $\phi_{CNT} = 0.1\%$ and (b) $\phi_{CNT} = 1.35\%$.

300

301 The effects of magnetite concentration on the difference between the heat transfer rate of the

302 Newtonian and non-Newtonian $\text{Fe}_3\text{O}_4\text{-CNT/water}$ hybrid nanofluids ($d_Q = \frac{\dot{Q}_{NN} - \dot{Q}_N}{\dot{Q}_N} \times 100$) at

303 different Reynolds numbers are illustrated in Fig. 7. It is seen that the heat transfer rate of the non-

304 Newtonian hybrid nanofluid is greater than that of the Newtonian nanofluid. The minimum

305 difference (0.31%) is achieved at $\phi_M = 0.9\%$, $\phi_{CNT} = 1.35\%$ and $Re = 500$, while the maximum

306 difference (1.23%) occurs at $\phi_M = 0.1\%$, $\phi_{CNT} = 1.35\%$ and $Re = 1000$. Additionally, it is

307 observed that with increase in the Reynolds number, the difference between the heat transfer rate

308 of the non-Newtonian and Newtonian nanofluids increases first and then decreases. Increasing the

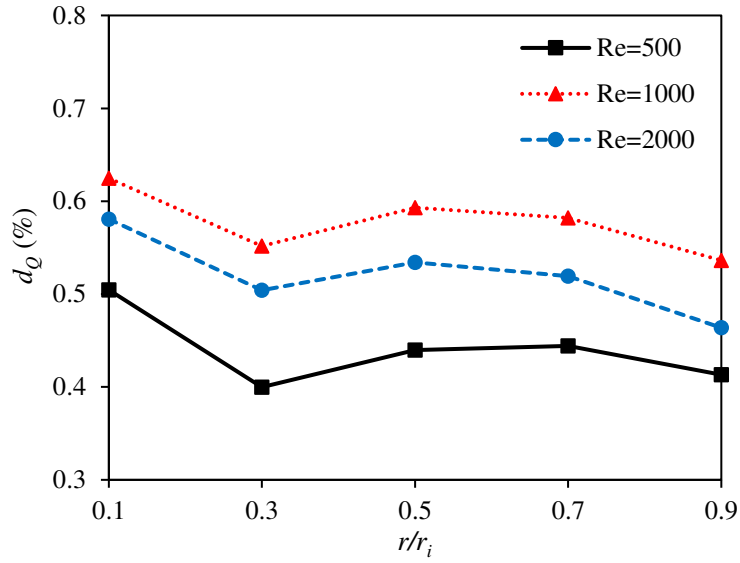
309 Reynolds number reduces the thermal conductivity and the thermal boundary layer thickness of

310 the non-Newtonian nanofluid, which respectively reduces and increases the rate of heat transfer.

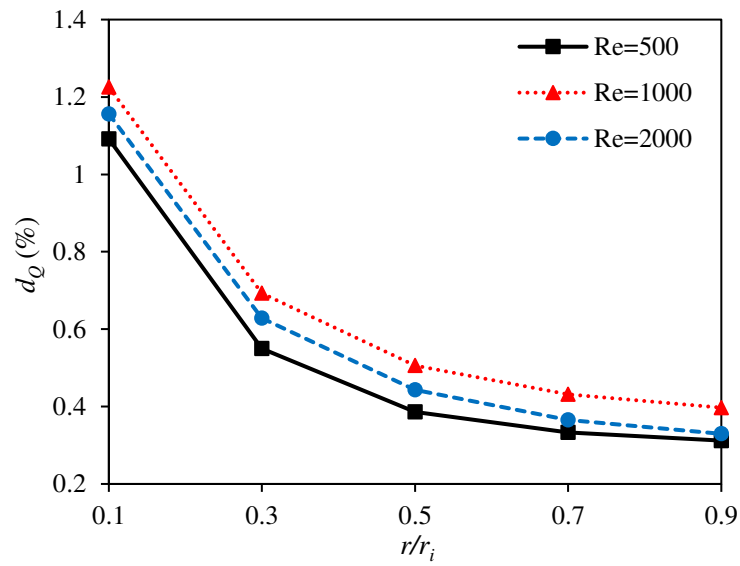
311 In view of Fig. 7, it can be realized that at $Re = 1000$, the effect of reducing the thickness of

312 thermal boundary layer is dominant in comparison with the reduction of thermal conductivity and
313 therefore, the difference between the heat transfer rate of the Newtonian and non-Newtonian
314 nanofluids increases. Meanwhile, for $Re = 2000$, the reduction of thermal conductivity is
315 dominant, which causes a decrease in the difference between the heat transfer rate of the non-
316 Newtonian and Newtonian nanofluids. Moreover, Fig. 7 reveals that at magnetite concentrations
317 of 0.1% and 0.3%, increasing the CNT concentration from 0.1% to 1.35% leads to an increase in
318 the difference between the heat transfer rate of the Newtonian and non-Newtonian nanofluids,
319 whereas the opposite is true for higher magnetite concentrations. Increasing the magnetite
320 concentration leads to the increase of thermal conductivity of the non-Newtonian nanofluid and
321 therefore, the increase of nanofluid outlet temperature, and eventually to the increase of difference
322 between the heat transfer rate of the non-Newtonian and Newtonian nanofluids. Further increase
323 in the magnetite concentration leads to the decrease of the difference between the thermal
324 conductivity of the non-Newtonian and Newtonian nanofluids and therefore, the decrease of the
325 heat transfer rate difference. The results also show that there is no specific pattern on the
326 relationship between the difference in the heat transfer rate of the Newtonian and non-Newtonian
327 nanofluids and the magnetite concentration.

328



(a)



(b)

Fig. 7. Heat transfer rate at different Reynolds numbers in terms of magnetite concentration at (a) $\phi_{CNT} = 0.1\%$ and (b) $\phi_{CNT} = 1.35\%$.

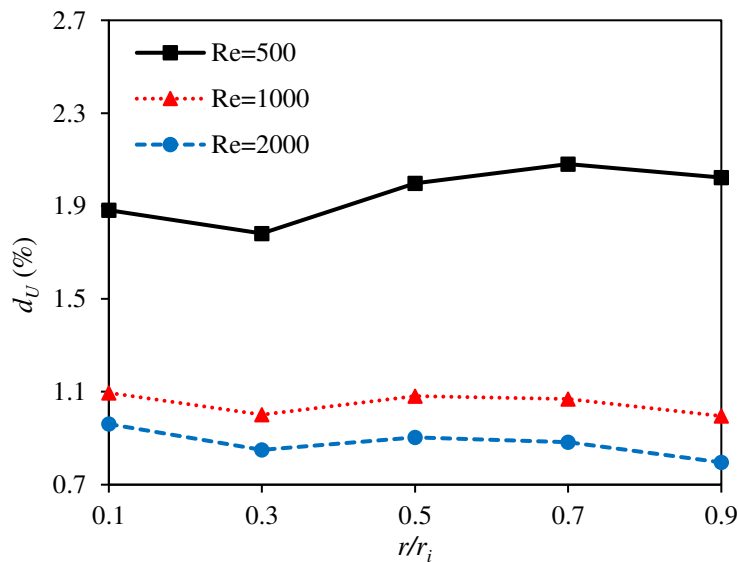
329

330 Fig. 8 shows the difference between the overall heat transfer coefficient of the Newtonian and non-

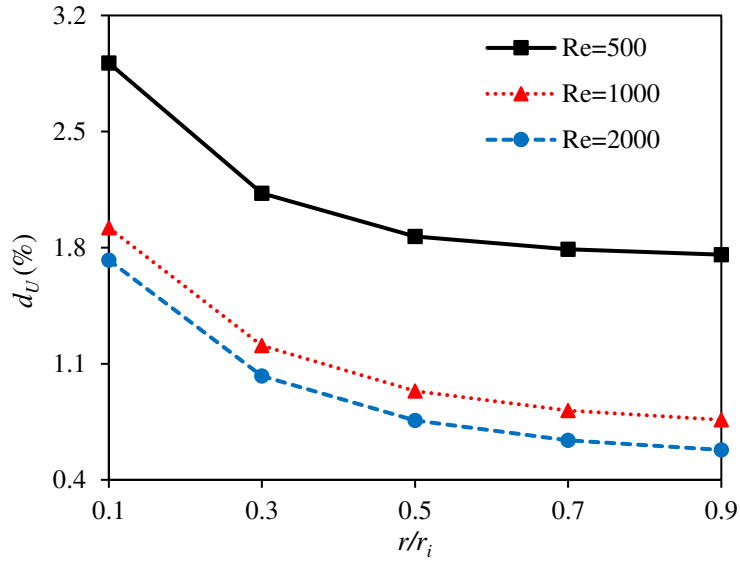
331 Newtonian $\text{Fe}_3\text{O}_4\text{-CNT/water}$ hybrid nanofluids ($d_U = \frac{U_{NN} - U_N}{U_N} \times 100$) in terms of magnetite

332 concentration at various Reynolds numbers. It is clear that the overall heat transfer coefficient of
 333 the non-Newtonian hybrid nanofluid is greater than that of the Newtonian nanofluid. The minimum
 334 difference of the overall heat transfer coefficients (0.58%) is obtained at $\varphi_M = 0.9\%$, $\varphi_{CNT} =$
 335 1.35% and $Re = 2000$, while the maximum difference (2.91%) is achieved at $\varphi_M = 0.1\%$,
 336 $\varphi_{CNT} = 1.35\%$ and $Re = 500$. Furthermore, the results depicted that the variations of difference
 337 between the overall heat transfer coefficient of the Newtonian and non-Newtonian nanofluids with
 338 the magnetite and CNT concentrations are similar to that of the difference between the heat transfer
 339 rate of these nanofluids. According to the results presented in Fig. 8, it can be concluded that the
 340 difference between the overall heat transfer coefficient of the Newtonian and non-Newtonian
 341 nanofluids is less than 3%, which is not significant.

342



(a)



(b)

Fig. 8. Overall heat transfer coefficient at different Reynolds numbers in terms of magnetite concentration at (a) $\varphi_{CNT} = 0.1\%$ and (b) $\varphi_{CNT} = 1.35\%$.

343

344 The impacts of magnetite concentration on the difference between the effectiveness of the heat
 345 exchangers containing Newtonian and non-Newtonian Fe_3O_4 -CNT/water hybrid nanofluids ($d_\varepsilon =$

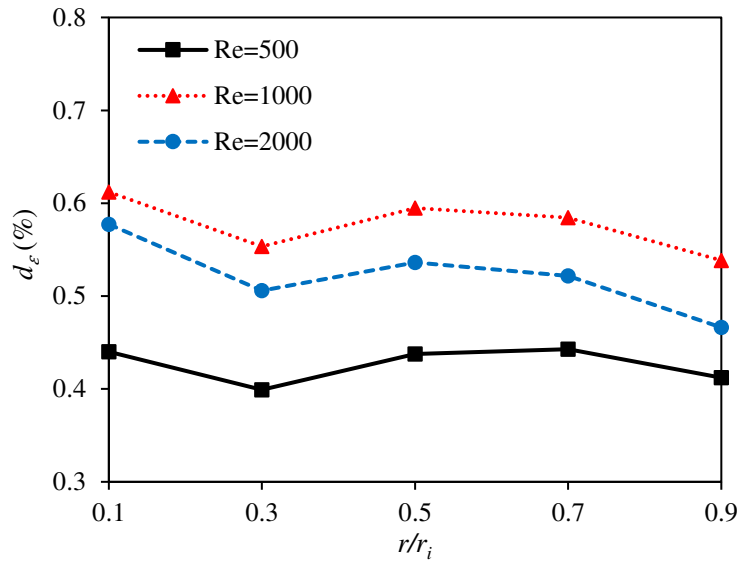
346 $\frac{\varepsilon_{NN} - \varepsilon_N}{\varepsilon_N} \times 100$) at different Reynolds numbers are illustrated in Fig. 9. In view of Eq. (12), it can

347 be realized that the trend of effectiveness variations is similar to that of the heat transfer rate

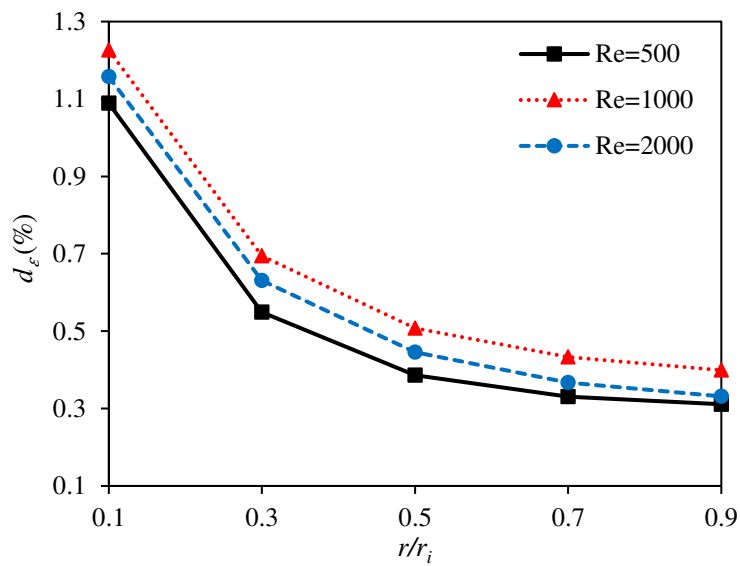
348 variations. Therefore, all the conclusions reached above regarding the heat transfer rate are also

349 true for the effectiveness.

350



(a)



(b)

Fig. 9. Effectiveness of heat exchanger at different Reynolds numbers in terms of magnetite concentration at (a)

$\varphi_{CNT} = 0.1\%$ and (b) $\varphi_{CNT} = 1.35\%$.

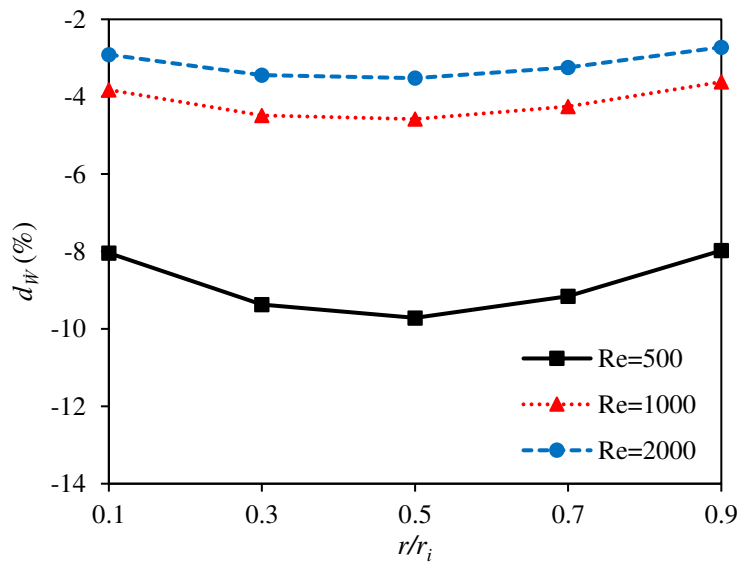
351

352 The pumping power indicates the amount of energy utilized in a heat exchanger. Fig. 10 depicts

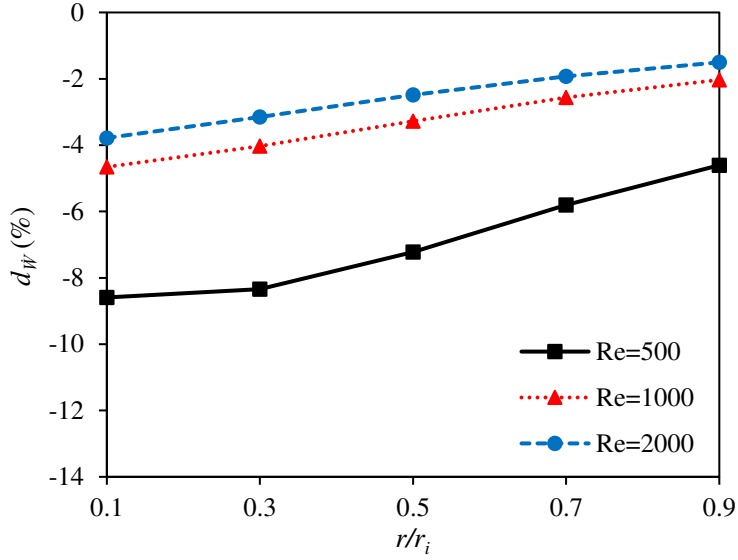
353 the difference between the pumping powers of the Newtonian and non-Newtonian Fe_3O_4 -

354 CNT/water hybrid nanofluids ($d_U = \frac{U_{NN} - U_N}{U_N} \times 100$) in terms of magnetite concentration at
 355 different Reynolds numbers. It is seen that at a constant Reynolds number, the non-Newtonian
 356 hybrid nanofluid always requires less pumping power than the Newtonian nanofluid. The
 357 minimum pumping power difference (1.5%) is obtained at $\varphi_M = 0.9\%$, $\varphi_{CNT} = 1.35\%$ and $Re =$
 358 2000, while the maximum difference (9.71%) occurs at $\varphi_M = 0.5\%$, $\varphi_{CNT} = 0.1\%$ and $Re =$
 359 500. In view of Eq. (17), and considering the same average velocity for Newtonian and non-
 360 Newtonian nanofluids at similar Reynolds numbers, the difference between the pumping power of
 361 the Newtonian and non-Newtonian nanofluids is only related to the difference between their
 362 pressure drops. Therefore, at a low Reynolds number, the assumption of constant properties leads
 363 to a considerable increase in the pumping power of heat exchangers, whereas the difference
 364 reduces with increasing the Reynolds number.

365



(a)



(b)

Fig. 10. Pumping power of heat exchanger at different Reynolds numbers in terms of magnetite concentration at

(a) $\varphi_{CNT} = 0.1\%$ and (b) $\varphi_{CNT} = 1.35\%$.

366

367 The influences of magnetite concentration on the difference between the performance index of the

368 heat exchangers containing Newtonian and non-Newtonian Fe_3O_4 -CNT/water hybrid nanofluids

369 ($d_\varepsilon = \frac{\eta_{NN} - \eta_N}{\eta_N} \times 100$) at different Reynolds numbers are displayed in Fig. 11. It is observed that

370 the heat exchanger containing non-Newtonian nanofluid has a higher performance index than that

371 containing Newtonian nanofluid. The minimum difference (1.86%) is obtained at $\varphi_M = 0.9\%$,

372 $\varphi_{CNT} = 1.35\%$ and $Re = 2000$, while the maximum difference (11.25%) is achieved at $\varphi_M =$

373 0.5% , $\varphi_{CNT} = 0.1\%$ and $Re = 500$. Moreover, it is seen that the difference between the

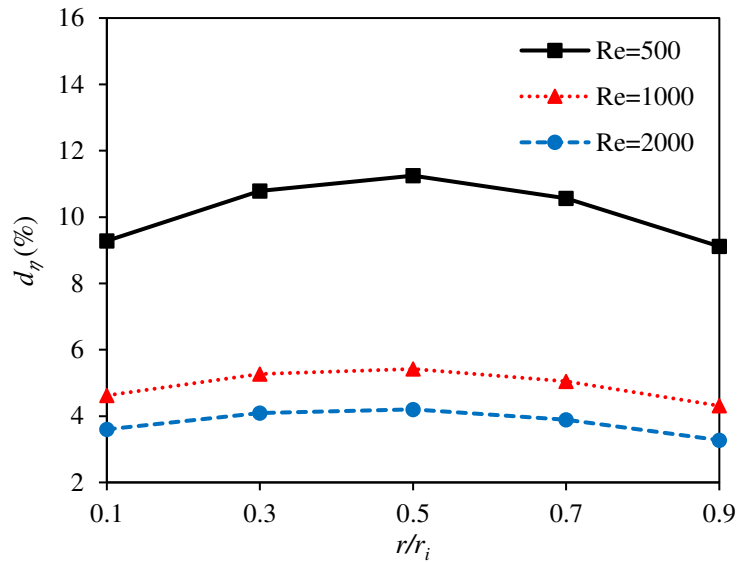
374 performance index of the heat exchangers containing Newtonian and non-Newtonian nanofluids

375 reduces with increase in the Reynolds number. In addition, at $\varphi_{CNT} = 0.1\%$, the performance

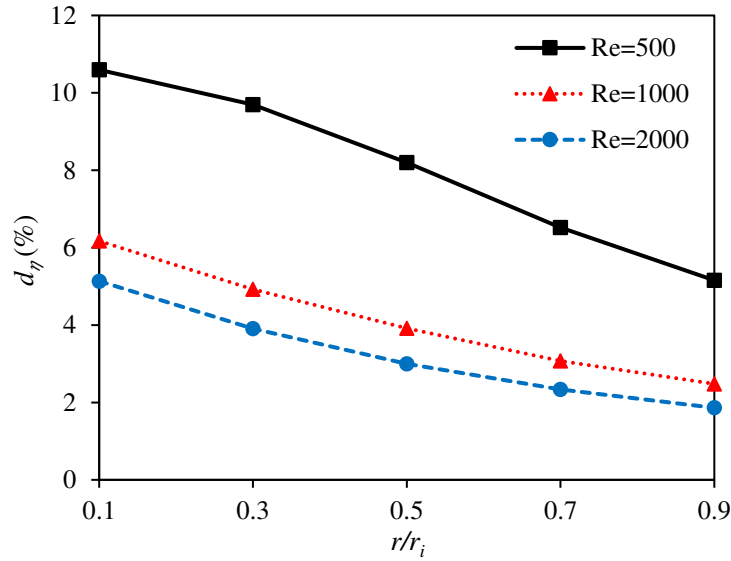
376 index difference augments when the magnetite concentration rises from 0.1 to 0.5% and then

377 decreases by the further increment of magnetite concentration; while for $\varphi_{CNT} = 1.35\%$, the

378 increase of magnetite concentration results in the reduction in the performance index difference.
379 Moreover, at $\varphi_M = 0.1\%$, the performance index difference increases with increase in CNT
380 concentration from 0.1 to 1.35%, while the opposite is happen at higher magnetite concentrations.
381 Finally, it can be said that the assumption of constant properties of the $\text{Fe}_3\text{O}_4\text{-CNT/water}$ hybrid
382 nanofluid at low Reynolds numbers and high concentrations of magnetite and CNT nanoparticles,
383 leads to large errors in the computation of performance index of heat exchanger.
384



(a)



(b)

Fig. 11. Performance index of heat exchanger at different Reynolds numbers in terms of magnetite concentration at (a) $\varphi_{CNT} = 0.1\%$ and (b) $\varphi_{CNT} = 1.35\%$.

385

386 7. Conclusion

387 In this research, the hydrothermal performance of the non-Newtonian Fe_3O_4 -CNT/water hybrid
 388 nanofluid considering temperature-dependent thermal conductivity and viscosity is numerically
 389 evaluated in a double-pipe mini-channel heat exchanger compared with Newtonian Fe_3O_4 -
 390 CNT/water nanofluid with constant thermal conductivity and viscosity. The comparison is used in
 391 order to find how the assumption of constant thermophysical properties of a hybrid nanofluid
 392 affects the hydrothermal characteristics in a double-pipe heat exchanger. The obtained results show
 393 that in central region of the tube, the non-Newtonian hybrid nanofluid has a higher viscosity
 394 compared to the Newtonian nanofluid, while the opposite is true in vicinity of the tube wall.
 395 Besides, it is found that the non-Newtonian hybrid nanofluid always has a higher thermal
 396 conductivity than the Newtonian nanofluid. In addition, it is seen that the heat transfer rate, overall

397 heat transfer coefficient, effectiveness, and performance index of the non-Newtonian hybrid
398 nanofluid are greater than those of the Newtonian hybrid nanofluid, while the opposite is true for
399 pressure drop and pumping power. The difference between heat transfer rate, overall heat transfer
400 coefficient, effectiveness, and performance index of Newtonian and non-Newtonian hybrid
401 nanofluids augments with increase in the Reynolds number, whereas the difference between the
402 pressure drop and pumping power of nanofluids reduces with increasing the Reynolds number.
403 Furthermore, increment in magnetite and CNT concentrations has no particular effect on the
404 considered parameters. Finally, it can be concluded that by supposing that the Fe₃O₄-CNT/water
405 hybrid nanofluid is a Newtonian fluid with constant thermal conductivity and viscosity, large
406 errors occur in the computation of pressure drop, pumping power, and performance index, whereas
407 the errors in the computation of heat transfer rate, overall heat transfer coefficient, and
408 effectiveness aren't considerable. The results of this study could provide guidelines to better
409 understand the real behaviors of hybrid nanofluids in heat exchangers.

410

411 **Appendix A**

412 The thermal conductivity correlation:

$$\begin{aligned}
413 \quad k_{nf} = & 0.22274 \tanh(-0.02119T + 0.09807\phi_M - 0.06975\phi_{CNT} + 0.02528) - 0.67299 \tanh(-0.00379T \\
414 \quad & - 0.69125\phi_M + 0.11290\phi_{CNT} + 0.03221) - 0.26968 \tanh(0.12778T + 0.00334\phi_M - 0.00362\phi_{CNT} + \\
415 \quad & 0.00284) - 0.22184 \tanh(0.02121T - 0.09748\phi_M + 0.06875\phi_{CNT} - 0.02471) - 1.01112 \tanh \\
416 \quad & (0.00755T - 0.99285\phi_M - 0.05887\phi_{CNT} - 0.42417) - 1.04948 \tanh(-0.00513T + 0.10775\phi_M - \\
417 \quad & 1.43226\phi_{CNT} - 0.49474) + 0.51061 \tanh(0.00157T - 1.10296\phi_M - 1.32512\phi_{CNT} + 0.43476) + \\
418 \quad & 0.23038 \tanh(-0.02104T + 0.10309\phi_M - 0.07821\phi_{CNT} + 0.03035) + 0.08974 \tanh(0.00333T + \\
419 \quad & 0.05961\phi_M + 0.02413\phi_{CNT} - 0.03048) + 0.45090 \tanh(0.02304T - 0.13733\phi_M - 0.48067\phi_{CNT} -
\end{aligned}$$

420 $0.50330) - 0.36153 \tanh (-0.06346T + 0.10622\varphi_M + 0.33903\varphi_{CNT} + 0.29052) - 0.49423 \tanh (-$
421 $0.00131T + 1.70368\varphi_M - 0.87848\varphi_{CNT} - 0.19465) - 0.21662 \tanh (0.02131T - 0.09401\varphi_M +$
422 $0.06297\varphi_{CNT} - 0.02155) - 0.57108 \tanh (-0.00374T - 0.59628\varphi_M + 0.10685\varphi_{CNT} + 0.04961) -$
423 0.27492 (A.1)

424

425 The viscosity correlation:

426 $\mu_{nf} = -0.24861 \tanh (0.04611\gamma - 0.00068T + 1.06226\varphi_M + 0.13756\varphi_{CNT} + 1.43142) + 1.03130$
427 $\tanh (0.47273\gamma + 0.00143T - 0.04534\varphi_M - 0.02812\varphi_{CNT} + 0.40817) - 0.20231 \tanh (-0.17180\gamma +$
428 $0.00067T + 1.20978\varphi_M - 0.18044\varphi_{CNT} - 0.25325) - 0.32811 \tanh (0.13316\gamma - 0.00050T -$
429 $1.21402\varphi_M + 0.14462\varphi_{CNT} + 0.53138) + 0.30415 \tanh (-0.11840\gamma - 0.00165T + 4.60293\varphi_M -$
430 $0.72515\varphi_{CNT} - 3.12857) - 0.00215 \tanh (-0.31607\gamma + 0.00027T - 0.98832\varphi_M + 0.15752\varphi_{CNT} +$
431 $4.97709) + 0.41053 \tanh (0.11589\gamma + 0.00148T - 4.37455\varphi_M + 0.62623\varphi_{CNT} + 3.03569) + 0.04707$
432 $\tanh (-0.09258\gamma - 0.04017T + 1.06859\varphi_M - 0.09049\varphi_{CNT} - 0.48490) + 0.59719 \tanh (0.04287\gamma +$
433 $0.33517T + 1.13670\varphi_M - 0.95007\varphi_{CNT} + 0.04857) + 0.03178 \tanh (-0.02358\gamma - 0.04493T +$
434 $0.45525\varphi_M - 0.01014\varphi_{CNT} + 0.45150) + 0.08139 \tanh (-0.09280\gamma + 0.00407T - 1.75844\varphi_M -$
435 $0.22328\varphi_{CNT} + 1.44124) - 0.52171 \tanh (0.14031\gamma + 0.00052T - 4.39738\varphi_M - 0.00719\varphi_{CNT} +$
436 $3.30400) - 0.04611 \tanh (-0.06569\gamma + 0.00073T + 2.48100\varphi_M + 0.00205\varphi_{CNT} - 1.03603) + 0.08759$
437 $\tanh (0.08788\gamma - 0.00410T + 1.69327\varphi_M + 0.22057\varphi_{CNT} - 1.33532) + 0.00066 \tanh (-0.00508\gamma -$
438 $0.03150T + 1.26008\varphi_M + 0.43853\varphi_{CNT} + 0.55153) + 0.01716 \tanh (0.14865\gamma - 0.00045T -$
439 $0.98651\varphi_M + 0.25307\varphi_{CNT} - 0.85218) + 2.25789 \tanh (-0.06180\gamma + 0.000003T + 2.26987\varphi_M -$
440 $0.00222\varphi_{CNT} - 2.86749) - 1.08194 \tanh (-0.11371\gamma - 0.00017T + 3.66682\varphi_M + 0.02690\varphi_{CNT} -$
441 $3.01085) + 0.49907 \tanh (-0.48296\gamma - 0.00190T + 0.00935\varphi_M + 0.05003\varphi_{CNT} + 0.09833) - 0.13648$
442 $\tanh (-0.02383\gamma - 0.03677T + 0.37656\varphi_M - 0.02106\varphi_{CNT} - 0.42817) + 0.70822$ (A.2)

443

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