The Non-Dimensional Analysis of Heat Transfer and Fluid Flow in Wavy Mini Channel Heat Exchangers

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ABSTRACT: *In this research, the heat transfer behavior of a wavy mini-channel heat exchanger was studied. Using the experimental data of heat transfer, the convective heat transfer coefficients were estimated. Among numerous trials, the Nusselt number (Nu) best correlation is a linear function of Reynolds number (Re) independent of Prandtl number (Pr), and similar correlations for hot and cold sides were obtained. The coefficient range is 0.01 to 0.03 for different fluids. The previous experimental works verify this conclusion. Also, in the case of non-Newtonian fluids and nanofluids, the definition of Re is related to its rheological behavior. However, if the velocity profile is specified, it can be used to derive the relation between the Fanning friction factor (C_f) and Re. Here, a suitable velocity profile for wavy configuration is used, and the experimental values of Re are estimated by the experimental pressure drop data. It is shown that the application of the derived relation between C_f and Re is preferred compared to the assumption of a circular pipe that is convenient for fluid mechanics studies. In addition, it is proved that if experiments with different fluids or relative waviness are done at similar flow rates, the U versus the Re plot can be used to compare heat exchanger performance.*

KEYWORDS: *Velocity profile; Heat transfer coefficient; CNT/CMC solution nanofluid; Non-Newtonian fluid; Wavy channel.*

INTRODUCTION

Compact heat exchanger as one of four areas in the process intensification [1] aims to minimize energy consumption or improve economic as well as environmental metrics [2]. The idea of using mini-channels as a way to enhance heat transfer rates was proposed in 1981 and the idea of wavy channels has been proposed to enhance heat transfer rates [3,4]. The nanoparticles' addition to the base fluid may enhance thermal conductivity [5] as well as the convective heat transfer coefficient [6] of the base fluid.

A study on a concentric-tube heat exchanger with a wavy

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inner pipe shows an increment of Nusselt number (Nu), friction factor, and thermal performance factor for the air cooling process [7]. The experimental results on microchannel heat exchangers for carbon dioxide cycles show that the application of S-shaped fins results in much less pressure drop compared to zigzag fins [8]. Experimental data on zigzag or sinusoidal microchannel pathways show that significant heat transfer enhancement is achieved in wavy channels compared with the equivalent straight channel, although an increased pressure drop penalty is also observed [9]. An experimental investigation carried out in a zigzag microchannel with water show that the flow transition from steady to unsteady flow occurs at a relatively low Reynolds number (Re about 215) in the laminar regime [10]. Experimental work on the cooling performance of sinusoidal–wavy mini-channel heat sinks shows the decreasing ratio of heat transfer rate to pumping power against Re [11]. Therefore, the application of such equipment is preferred at low Re. However, the application of nanofluid has been advised at higher flow rates [12]. An investigation of the effects of nanoparticle type and concentration, as well as base fluid type, shows that SiO_2/water is the best fluid among the studied fluids for heat transfer of the corrugated wavy channel at a constant wall temperature [13]. Also, an investigation of the effects of perforations, winglets, and nanofluids, on heat transfer of the wavy plate-fins heat exchanger shows the highest performance for the winged structure [7]. Recently, fluid flow and mass transfer characteristics of CMC solutions were studied in wavy-walled tubes for steady and pulsating flow [14].

Prediction of convection coefficients as a function of Re usually plays the main role in the heat transfer study of any type of heat exchanger. The Wilson plot method and its different modifications provide an outstanding tool for the analysis and design of convection heat transfer processes in heat exchangers. The original form of the Wilson plot method is applicable if the thermal resistance of one side of the heat exchanger is constant and the other side is in a fully developed flow condition where its convective coefficient is proportional to a known power of its reduced velocity. The modifications of the Wilson plot method used a power of the Re and Prandtl number (Pr) of the fluid in which the mass flow is varied (fluid A) instead of its reduced velocity. The later modifications of the Wilson plot method consist of formulating a functional form for the convection coefficient for fluid B, instead of constant thermal resistance. The most general form of the Wilson plot method involves three constants, two of them corresponding to the assumed functional form for one of the two fluids and the third unknown constant stands for the exponent of the flow parameter inherent to one of the fluids [15]. The validity of all Wilson plot methods is not easily assessed and depends on the accuracy of the data, the number of data points, the range of the data, the relationship between the heat flux and temperature difference on each side, relative magnitudes whose residuals are minimized and the number of constants to be found [16]. Some researchers studied heat transfer numerically [17–22]. For example, the entropy generation rate in a sinusoidal wavy-wall channel with the nanofluid flow was computed numerically [23]. Also, the flow and heat transfer of shear-thinning power-law CMC solutions in dimpled and protruded microchannels were studied numerically [24]. Recently, convective heat transfer in a semi-spherical fin was modeled and solved [25]. *Rostami et al*. studied numerically the heat transfer by nanofluids in wavy microchannels [26].

It has been shown that for the pipe flow, the velocity profile of the base fluid is similar to that of nanofluid and the enhancement of convection heat transfer is attributed to the change of the temperature profile [27] besides the other properties of the fluid. The effect of the fluid properties may be considered through dimensionless variables. If the form of the dimensionless velocity profile in a conduit is specified and it can be considered independent of the used fluid according to the above discussion, the pressure drop data can be used to estimate Re as will be discussed in the next sections. It was shown that Re is the most effective factor on Nu [28]. In the case of non-Newtonian fluids to solve the ambiguity of the viscosity term, a generalized Re is usually defined [29,30], however, the above-mentioned method can be used to evaluate Re via experimental data of pressure drop based on the prevailing velocity profile. It has been shown that the mean velocity profile of a laminar wavy fluid film is parabolic [31].

$$u = Ay - By^2 \tag{1}$$

Where *A* and *B* depend on Re in general form, however, the specified wavy channel are assumed to be constants.

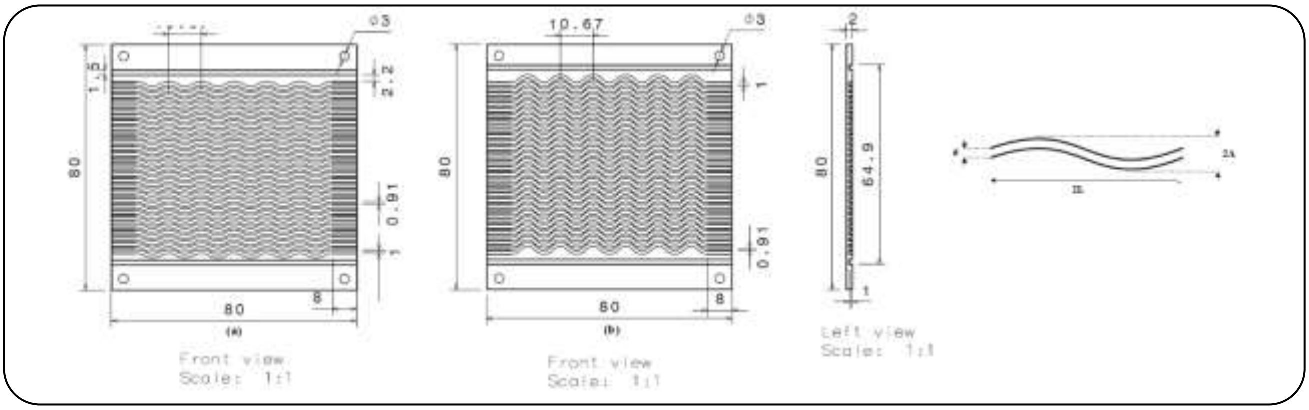

Fig. 1: The structure of the wavy mini channel, the relative waviness is 2A/2L.

In this research, the relation between the Fanning friction factor (C_f) and Re is derived based on Eq. (1). The experimental data of pressure drop together with the derived relation is used for estimating Re without the need to define dynamic viscosity. It is especially advisable for non-Newtonian fluids. According to the results, The Nu of wavy channels obeys a linear relation to Re and is independent of Pr. Also, pressure drop and heat transfer enhancement usually have trade-off effects. Here, it is proved that the defined Re can be used to have a suitable comparison of heat exchangers' performances.

EXPERIMENTAL SECTION

The CMC solutions were prepared by adding 0.2 wt.% sodium carboxymethyl cellulose (Dae-Jung company, South Korea) to distilled water and used as the base fluid. The Single-Wall Carbon Nano Tubes (SWCNTs) were provided by the Research Institute of Petroleum Industry (R.I.P.I). The non-Newtonian nanofluids were prepared at concentrations of 0.2 wt.% of SWCNTs to the non-Newtonian CMC solution. The nanofluids were stable for at least two weeks and each sample was tested for all heat exchangers at various flow rates and inlet temperature values. The specific heat capacity and thermal conductivity were obtained by KD2 Pro (Decagon Devices, Inc., USA [32]). The complete description of the test facility [33], experiment uncertainty [34], and nanofluid preparation method [35] were described in the previous works of the authors. Two relative waviness, (2A / 2L = 0.2) and (2A / 2L = 0.3), for the wavy mini channel as depicted in Fig. 1 are studied in this work and denoted by Wavy 1 and Wavy 2, respectively. The cross-section area of the channel is considered as $A_c = 105.3\times 10^{-6}$ m^2, its surrounding area as $A_s = 0.029808$ m^2 and the hydraulic diameter as $D_h = 0.904\times10^{-3}$ m for all channel configurations.

The experiments were done at various flow rates as 0.5 L/min, 1.5 L/min, and 2.5 L/min. Also, different inlet temperature values of hot fluid were set to 40 ºC, 45 ºC, and 50 ºC.

The experiments with equal flow rates of cold and hot streams are investigated. Nevertheless, since Re is estimated using Eq. (6) together with experimental data of pressure drop, the Re of hot and cold streams are different.

THEORETICAL SECTION

Estimation of Re

The first assumption for the derivation of the model is the use of Eq. (1) for the velocity profile of wavy channels with constants A and B that can be written in the dimensionless form [36]

$$u^+ = A^+ y^+ - B^+ \left(y^+\right)^2 \quad (2)$$

where $u^+ = u/u^*$, $u^+ = \sqrt{\tau_w/\rho}$, $y^+ = y/\delta^*$, $\delta^* = v/u^*$, $A^+ = A\left(v/u^{*2}\right)$, $B^+ = B\left(v^2/u^{*3}\right)$, τ_m is shear stress at the wall, $v = \mu/\rho$ is kinematic viscosity, μ is dynamic viscosity, and ρ is density. According to the definition of shear stress at the wall as $\tau_w = \mu \left.\frac{du}{dy}\right|_{y=0} = A\mu$, and dividing both sides by ρ it can be rearranged to

$$A^{+} = 1 \tag{3}$$

Based on the assumption of maximum velocity at the center point, h = $D_h/2$, where D_h is the hydraulic diameter of the conduit, $\tau_{y=h}$ = 0= A-2Bh, the second constant is obtained as

$$B^{+} = \frac{1}{2h^{+}} = \frac{1}{D_h^{+}} \tag{4}$$

The final dimensionless velocity profile is written as

$$u^{+} = y^{+} - \frac{1}{D_h^{+}}\left(y^{+}\right)^{2} \tag{5}$$

According to $u_m^{+} = \sqrt{2/C_f} = \frac{1}{h^{+}}\int_0^{h^{+}} u^{+}dy^{+}$ and $Re = \frac{u_m D_h}{v} = u_m^{+} D_h^{+}$ the Fanning friction factor C_f, is derived as

$$C_f = \frac{12}{Re} \tag{6}$$

On the other hand, τ_w and therefore C_f can be evaluated from pressure drop data via $\tau_w.A_s = \Delta P.A_c$, where As is the side area and Ac, is the cross-section of the conduit. As a result, Eq. (6) besides experimental pressure drop values can be used to estimate Re at various flow rates. This conclusion is generalized to non-Newtonian fluids in this work. Nevertheless, the dimensionless velocity profile form as Eq. (5) for non-Newtonian fluids needs the assumption of a viscous length scale similar to Newtonian fluids since δ^* relates to viscosity.

Performance comparison using the defined Re

To compare heat exchangers, their heat transfer is usually evaluated at an equivalent value of Re (Re_{eq}) that takes into account the effect of pressure drop and so different pumping power [37]. The mechanical power required for the transport of the fluid through the heat exchanger equals its pressure drop multiply the flow rate. According to the definition of the Fanning friction factor and its relation to pressure drop as well as Eq. (6), the power is [38]

$$W \propto \Delta P.Q = \frac{6}{Re}\frac{A_s}{A_c^3}\rho Q^3 \tag{7}$$

Where Q is the volumetric flow rate. The values of Re_{eq} are calculated to achieve $W_{eq} / W = 1$, where W_{eq} is the theoretical power of a hypothetic system with Re_{eq} instead of *Re* and water as the working fluid. The ratio of surfaces is approximately constant and the experiments were done at the same flow rates of different fluids. Also, the density of nanofluids is approximately similar to the base fluid. Therefore, for this research condition, the evaluated *Re* according to pressure drop data plays the role of Re_{eq}

$$Re_{eq} \approx Re \tag{8}$$

It should be noted that if Re had been evaluated through $\rho u_m D_h / \mu$, the values of Re would not be considered independent of Q and the above conclusion had not been true. However, with similar flow rates together with Re calculated according to pressure drop data, a plot of U against Re can be used to compare different working fluids and relative waviness.

Convective heat transfer coefficient

For the estimation of the convective heat transfer coefficient, a method similar to the Wilson plot is used here. However, each wavy mini channel heat exchanger was tested using three different hot fluids including water, CMC solution, and its nanofluid containing CNT, but the cold side was always water. Therefore, four Nusselt numbers (Nu = hD_h/k) should be estimated and notations of c, Water, CMC, and CNT refer to the cold side, water on the hot side, CMC solution, and CNT/CMC nanofluid. The assumed relation form is Nu = c $Re^b Pr^n$. Nevertheless, as it was said in the previous sections, the addition of particles will change the temperature profile and therefore the constants of the equation will be different for various fluids. The prevailing relation for wavy configuration is proposed in this work as

$$Nu = aRe^{b} \qquad b = 1 \tag{9}$$

with different *a* parameter for studied fluids. The value of unity for *b* together with the independency of Pr is derived based on the experimental data of this work. The relation between the overall heat transfer coefficient, *U*, and convective heat transfer coefficients, *h*, is [39]

$$\frac{1}{U} = \frac{1}{h_c} + \frac{t}{k_w} + \frac{1}{h_h} \tag{10}$$

Where $U = Q_{avg} / (A_s \Delta T_{lm})$ and ΔT_{lm} denotes log mean temperature difference, and k_w is the plate thickness and thermal conductivity respectively and subscripts *c* and *h* denote cold and hot flows. Among the terms of Eq. (10),

U is derived through experimental data of input/output temperature, mass flow rate, and heat capacity of hot and cold flows. Therefore, the unknowns of Eq. (10) are h_c and h_h which should be evaluated by Eq. (9) using non-linear minimization methods (here Nelder-Mead). Based on trails and error procedure, for each test fluid, a similar form of convective heat transfer relation to Re at hot and cold sides gives reasonable results and b in Eq. (9) is unity for all cases.

RESULTS AND DISCUSSION

The velocity profile, Eq. (5), is independent of Re if u/u_m versus y is studied. Assuming Re = 300, the derived velocity distribution is calculated and shown in Fig. 2.

Although the specification of Re is needed to have numerical values for velocity, its typical shape does not relate to Re.

The conventional definition of $Re = \rho u_m D_h / \mu$ is compared to that derived by Eq. (6) and the approximate relation as $C_f = 16/Re$ is based on the assumption of circular duct in Fig. 3 for water as working fluid.

According to this figure, the Re derived by Eq. (6) based on the experimental pressure drop and without using viscosity is nearer to the conventional definition using the known value of viscosity compared to Re calculated based on the circular pipe assumption. It is worth noting that using circular pipe relations with a hydraulic diameter or an effective diameter has been reported in the previous literature [37]. Therefore, it is used for non-Newtonian fluids in this work. In this way, the study of the rheological properties of the working fluids is omitted here. It is noteworthy that the flow of non-Newtonian in a conduit undergoes a range of shear strains and therefore assigning a constant viscosity is not the precisely correct view.

Usually improving heat exchangers to have a higher heat transfer rate is accompanied by the drawback of higher pumping power. Fig. 4 shows the overall heat transfer coefficient of the heat exchanger with different wavy channels, against Re. The comparison of U at the same Re data denotes the best condition for nanofluid and higher relative waviness. Also, the improvement due to the addition of CMC and nanoparticles is in the same order as for the application of higher relative waviness.

It is worth noting that the values of Re for non-Newtonian fluids in this research show a noticeable difference compared to Re for water, because of various pressure drops.

The estimated constants of convective heat transfer are derived by non-linear minimization Nelder-Mead

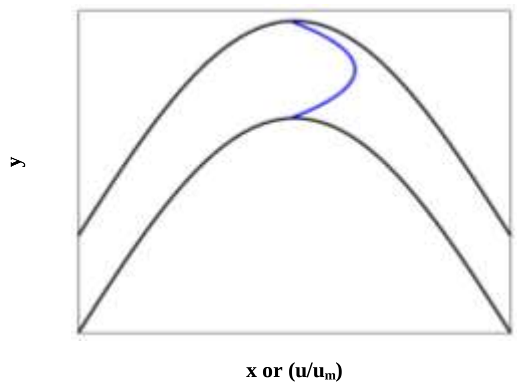

Fig. 2: ***The velocity profile in a wavy configuration according to Eq. (5). u/u_m is dimensionless velocity and the units of x and y are omitted to have a typical shape.***

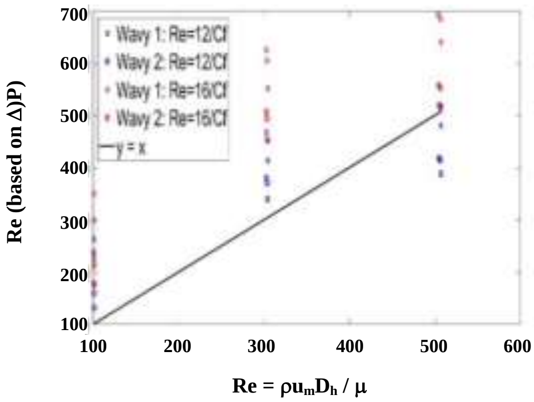

Fig. 3: ***Re derived by Eq. (6) and circular pipe assumption compared to the conventional definition for two wavy channels.***

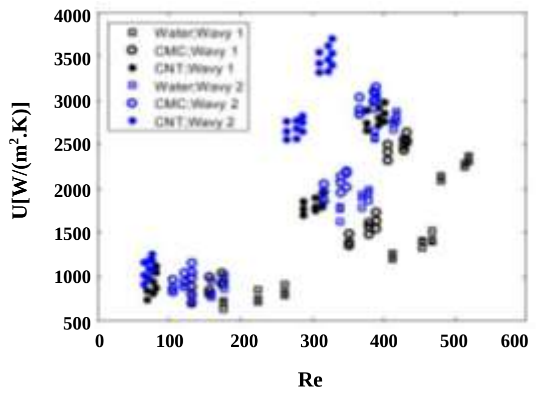

Fig. 4: ***Overall heat transfer coefficient of different heat exchangers against Re. Since Re is evaluated based on the pressure drop data, higher values of U imply higher performances.***

Table 1: The constant a of Eq. (9) based on the regression of Eq. (10) to the experimental data for different fluids and configurations.

	a_{Water}	a_{CMC}	a_{CNT}
Wavy 1	0.0104	0.0133	0.018
Wavy 2	0.0166	0.02	0.0285

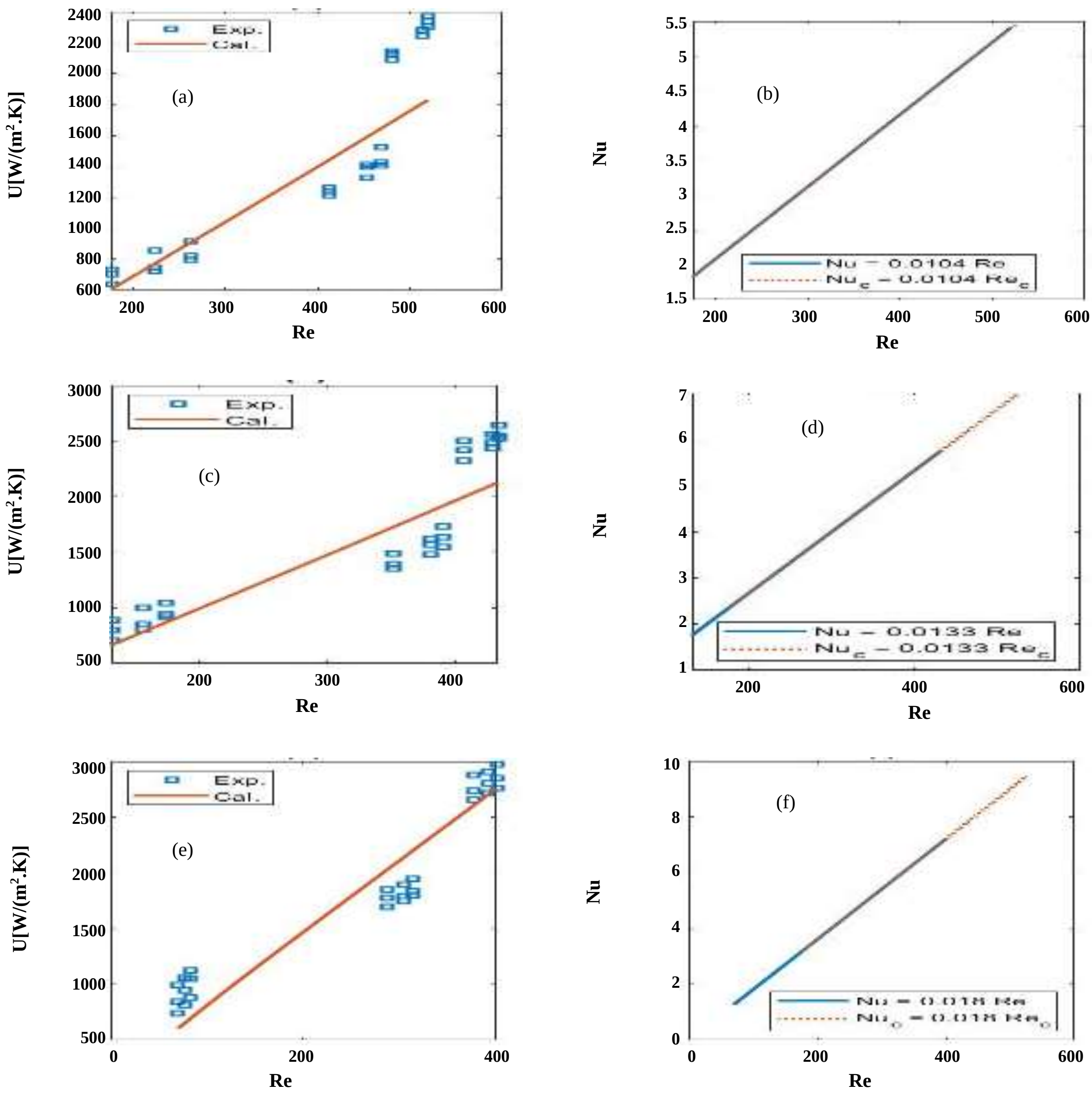

Fig. 5:* E*xperimental U compared to the calculated one for a) water, c) CMC solution, and e) CNT nanofluid and the calculated Nu with b) water, d) CMC solution and f) CNT nanofluid for the Wavy 1 configuration.

simplex method. The exponent of the Re, *b* in Eq. (9), is always unity. Table 1 shows the proposed multipliers, *a* in Eq. (9), for estimation of convective heat transfer coefficients.

Fig. 5 shows the evaluated U and Nu of Wavy 1 heat exchanger against Re of the hot side evaluated using pressure drop, using the data of Table 1. Also, Fig. 6 shows similar data for the Wavy 2 configuration.

The statistical concept may be used to study the difference

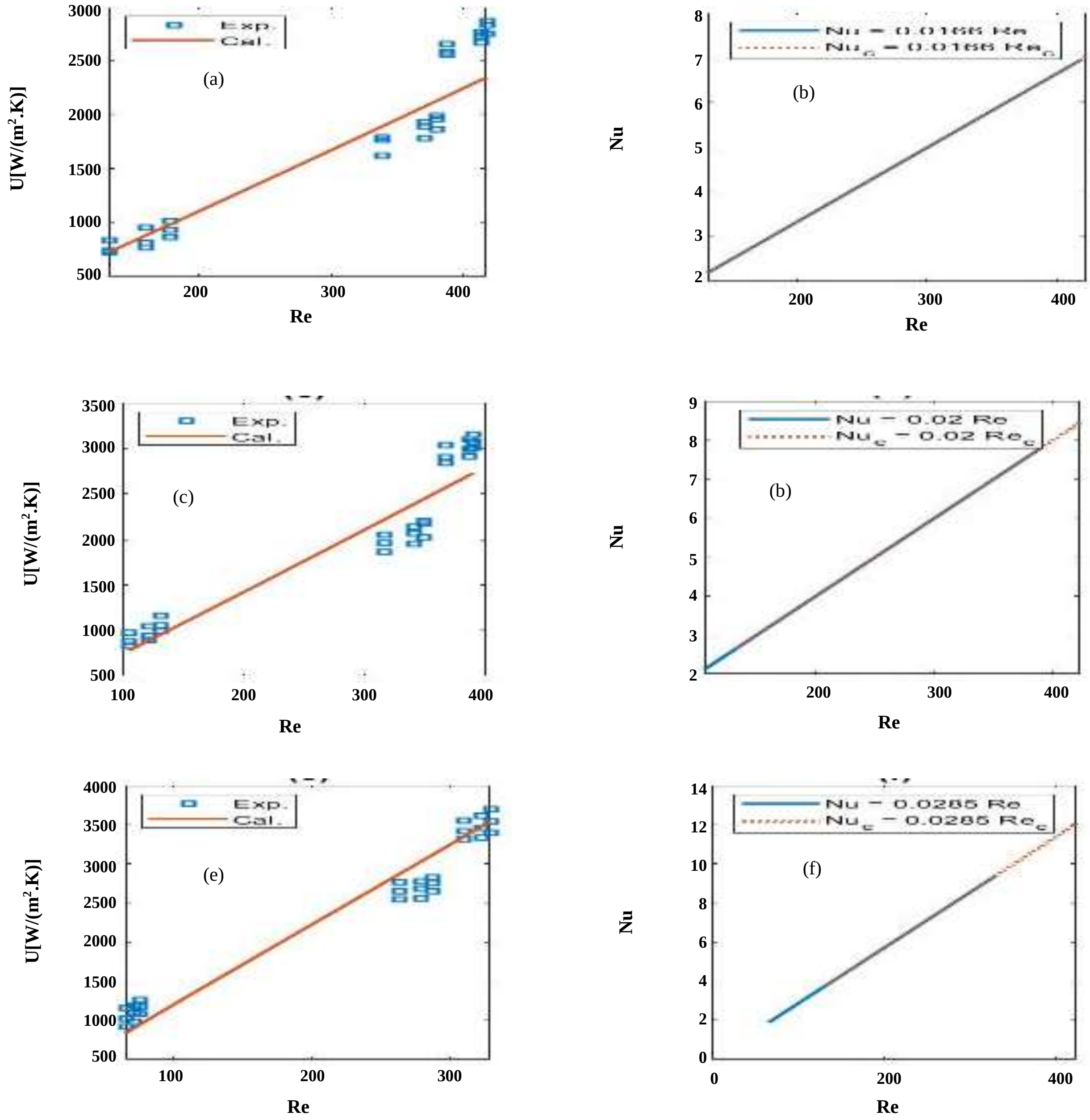

Fig. 6: Experimental U compared to the calculated one for a) water, c) CMC solution, and e) CNT nanofluid and the calculated Nu with b) water, d) CMC solution and f) CNT nanofluid for the Wavy 2 configuration.

between the experimental and calculated data. In this view, if the p-value is less than 0.05, there is a significant difference. For comparison of experimental and calculated U for Wavy 1 configuration, in the case of water p-value is 0.48, for CMC solution it is 0.16 and for CNT nanofluid the p-value is 0.15 where all of them are greater than 0.05 and calculated U is not different from experimental one.

Statistical comparison of experimental and calculated U for Wavy 2 configuration results in p-values of 0.62 for water, 0.49 for CMC solution, and 0.28 for CNT nanofluid where all of them are greater than 0.05 and the calculated U is not different from the experimental one.

The order of heat transfer coefficient of water, CMC solution, and CNT nanofluid can be attributed to their thermal diffusivities as shown in Fig. 7.

The thermal diffusivity of CNT nanofluid is the highest however, the thermal diffusivity of the CMC solution is somewhat higher than water. It may change the temperature

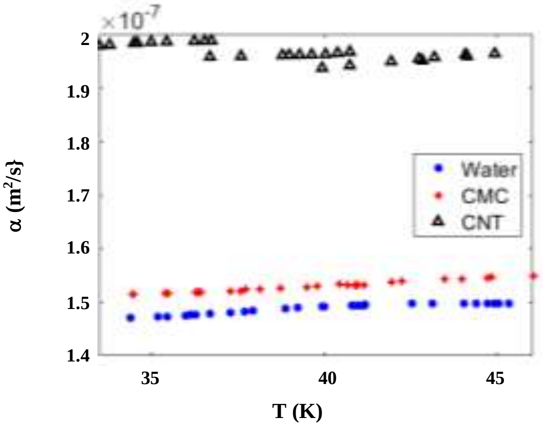

Fig. 7: The thermal diffusivity of the applied fluids in this research.

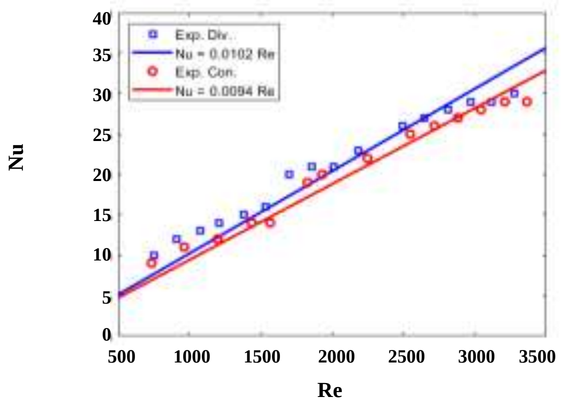

Fig. 8: The experimental data of Dominic et al. [40] compared to the linear relation proposed in this research.

profile of the fluid and consequently leads to different heat transfer coefficients.

According to these figures, the suitable prediction of U implies the acceptable used method for the evaluation of convection heat transfer. The fitted parameters are derived via numerous trials to avoid unusual Nu and/or negative exponent of Re or Pr. Also, suitable constants should be less sensitive to high-order decimals and general constants are better. Among these conditions, the effect of Pr is ignored and the derived equations for hot and cold sides are similar.

Validation of the model using other data

In addition to the experimental data of this work, Fig. 8 shows that the reported data by *Dominic et al.* [40] verify the proposed model for heat transfer of wavy heat exchangers as the linear relation between Nu and Re and independent of Pr.

The experimental convective heat transfer coefficient values in Fig. 8 (or Nu) were determined based on the wall temperature measurement and therefore *Dominic et al.* used different methods compared to this research. The reported data in Fig. 8 relate to two channels configuration including divergent (denoted by Div.) and convergent (denoted by Con.). The selected fluid is 0.5% Al_2O_3/water nanofluid but the trend is similar for other fluids that are not shown here.

CONCLUSIONS

In this research heat transfer of a wavy heat exchanger was modeled based on the theoretical basis and using experimental data. According to the results, the prediction of the model is better than the assumption of a circular pipe. Therefore, the Re was evaluated from the experimental pressure drop values without the need for viscosity especially is useful for non-Newtonian fluids and nanofluids. A trend of experimental U against the evaluated Re can be used to compare the performance of different heat exchangers. The proposed Nu relation for wavy channels is a linear function of Re that is verified using the experimental data of this work together with other researchers. An important conclusion of this study is the independency of Nu to Pr for wavy channel heat exchangers. Also, it was shown that similar relations for Nu to Re at hot and cold sides are satisfactory for wavy mini-channel heat exchangers. Therefore, the primary results are as follows

- Re can be evaluated experimentally using pressure drop values for wavy mini channels without the need for viscosity.
- The proposed relation for friction factor against Re is better than the circular pipe assumption.
- The proposed linear relation of Nu against Re is verified based on the experimental data of this research together with previous works.

For future studies, the performance of an industrial heat exchanger may be tested.

Nomenclature

A,B,a,b	Constants
A_c	Cross-section area, m^2
A_s	Side area, m^2
C_f	Fanning friction factor
D_h	Hydraulic diameter, m
h	Convective heat transfer coefficient, W/m^2K
k	Thermal conductivity, W/mK
Nu	Nusselt number

Pr	Prandtl number
Q	Volumetric flow rate, m^3/s
Re	Reynolds number
T	Temperature, K
t	Wall thickness, m
U	Overall heat transfer coefficient, W/m^2K
u	Velocity profile, m/s
W	Power, W
y	Distance from the wall, m

superscript

+	Dimensionless
*	Characteristic

Subscript

c	Cold
h	Hot
m	Mean
w	Wall

Greek letters

δ	Characteristic length scale, m
μ	Dynamic viscosity, pa.s
ν= μ /	Kinematic viscosity, m^2/s
ρ	Density, kg/m^3
τ	The time constant, space-time, stress

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