

# Thermo-Hydraulic Performance Analysis of CNT-H<sub>2</sub>O Nanofluid Flow in Turbulent region through Rectangular Plate Fin Heat Exchangers -A Numerical Study

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**ABSTRACT:** The heat exchanger performance of a system can be enhanced employing a combination of passive strategies, particularly nanofluids and varied structures of fins geometries. These methods can enhance the heat transfer coefficient and consequently reduce the weight of the system. In this paper, the effect of fin geometry and nanofluids in Plate Fin Heat Exchangers (PFHE) are studied by numerical analysis. The forced convective heat transfer performance is evaluated in terms of Nanofluid Thermal Performance Number (*NTPN*), Nusselt number and Colburn *j* factor. Pressure drop characteristics is estimated in terms of Fanning friction factor '*f*' of the CNT-H<sub>2</sub>O nanofluid in the turbulent zone of Reynolds number in the range 3000 to 7500. Thermo-hydraulic performance variations with rectangular fin geometry and volume fraction 3% to 5% of the nanoparticles in nanofluid is analyzed. Performance of the nanofluids are also compared with the performance of pure water. It has been observed that heat transfer enhancement using nanofluid is more than 20%. A correlation of Fanning friction factor '*f*' and Colburn *j* factor with geometry of fin, Reynolds number and volume fraction of nanofluid is developed. This correlation is the heart of the rating and design of PFHEs.

**KEYWORDS:** Compact PFHE, Numerical study, Fanning friction factor, Colburn *j* factor, Nanofluid.

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## 1. INTRODUCTION

Forced convective heat transfer plays a major role in heat exchange applications with fluid flow. The improvement of forced convective heat transfer can reduce the energy loss and hence scale down the size of a system. Typical liquids such as water, ethylene glycol and oil are used in the heat exchange applications include micro heat sink, solar panel, heat exchanger and electronic components as coolant. Researchers are now focused in improving the heat transfer coefficient which leads to miniaturization of thermal equipment. The convective heat transfer coefficient is strongly dependent on the surface of the solid, thermo-physical properties of the working fluid and the type of flow. The turbulence flow has higher heat transfer coefficient due to the increase in fluid velocity. Adding metal into the traditional fluid will facilitate in increasing the thermo-physical properties of the fluid. This is because of the fact that metal has higher thermal conductivity than fluid, that can help to improve the rate of heat transfer. The idea of adding metal particles into base fluids was introduced by Maxwell [1]. He added micro-sized metal particles into the base fluids to improve the heat transfer of the system. Although it increases the thermal conductivity of the system but the micro-sized particles clogged and increased pressure drop and erosion was sighted by Das et al. [2]. Due to these issues, the metal dispersed into the base fluids does not bring significant enhancements in heat transfer. With the new invention of nano-sized particles, Masuda et al. [3] dispersed the Al<sub>2</sub>O<sub>3</sub>, SiO<sub>2</sub> and TiO<sub>2</sub> nano-sized metal

oxide particles into water as the base fluid. They found that the enhancement was significant and also reduced the problems caused by the micro-sized metal particles. Later, Eastman et al. [4] introduced the nano-sized metal particles and worked with a variety of base fluids that bring better enhancements in effective thermal conductivity. Choi [5] had introduced the term "nanofluids", a reference to suspended nanoparticles smaller than 100 nm and dispersed into base fluids. The common base fluids used were typically water (W), ethylene glycol (EG), W/EG base mixture and oil shown by Redhwan et al. [6]. Earlier studies by Lee et al. [7] and Xuan and Li [8] proved that nanofluids have higher thermal conductivity compared to their base fluids. Lee et al. [7] found that the 13 nm Al<sub>2</sub>O<sub>3</sub> nanoparticles dispersed in water increased by 30% of its thermal conductivity compared to water at 4.3% volume concentration.

The compactness of heat exchanger is the index of progress in the present days scenario of industrial growth [8] especially with increasing need for developing the cryogenics field. Usually, plate fin heat exchanger is suitable for numerous types of heat exchanger application for a wide range of industries [9–12] like chemical process, power industries, aerospace, automobile and electronics. Plate fin units are normally arranged for counter flow heat exchanger but cross flow arrangements are also very common. Plate fin heat exchanger has thin corrugated fins. Surface density of compact heat exchangers are very high, that means large surface area per unit

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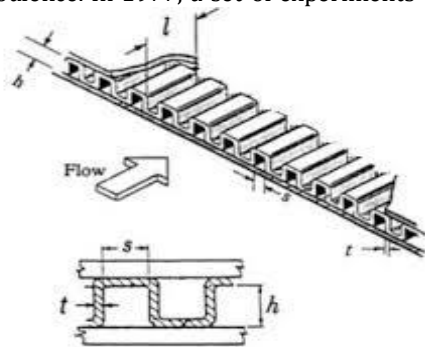
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volume (it could be as high as  $1800\text{m}^2/\text{m}^3$ ). The plate fin heat exchanger is suitable for a close approach temperature as low as  $21^\circ\text{C}$  and two or more streams can be used by changing the section. Plate fin heat exchanger is significant now a days and most widely used due to high heat transferrate.

It is investigated that compact heat exchangers such as plain fin strip, offset fin, wavy fin, perforated fin etc. the pressure drop decrease with respect to increasing the turbulence in working fluid. Onwards 1942 by Norris and Spofford [13] provide the first experimental report they draw out the effect of heat transfer coefficient on the basis of length, thickness and pitch of fins and also reduced the friction factor and Colburn j factor.

As the practical demand of plate fin heat exchanger has increased experimental studies, have been made by London and Shah [14] in 1967 and they concluded that's small offset spacing fin thickness and a large number of fins per inch gives better heat transfer. In 1975 Wieting [15] setup a relationship between the variables from earlier experimental heat transfer and fluid flow friction data for a plate-fin heat exchanger of offset fin and by using this relationship (un-tested offset fin geometries can be predicted realistically and accurately within the parameter range of the correlation). One can predict virtually and correctly within the parametric range of newest offset plate-fin heat exchanger having no previous tested data. Experimental validation of numerical simulation and also provides a comparison of experimental result in computationally obtained results from the effects of fin thickness and free stream turbulence. In 1977, a set of experiments was



**Fig 1: Geometry of Plain Rectangular Plate Fin Heat Exchanger**

**Table 1: Working Ranges of Model Parameters**

Parameters	Working range
Reynolds number	$3000 \leq Re \leq 7500$
Volume fraction of CNT-H <sub>2</sub> O nanofluid	$3\% \leq \phi \leq 5\%$
Height of the fin	$2\text{mm} \leq h \leq 6\text{mm}$
Fin spacing	$2\text{mm} \leq s \leq 6\text{mm}$
Thickness of the fin	$0.0254\text{mm}$
Length of the fin	$100\text{mm}$

performed by Sparrow [16] to observe the heat transfer for a mass flow rate with varying Reynolds number. The thickness ratio and the spacing ratio are the other factors establish that the Nusselt number varies while changing the plate thickness and also come upon after searching that it is not necessarily equal spacing and length gives optimal results.

Not much work had been found on study of PFHE with CNT nanofluids. Moreover, the results are not consistent and rather conflicting in few cases. So, more research in this area is required. A numerical simulation using Finite Volume Method is done to analyze the performance of the plain rectangular PFHE in a Reynolds number range 3000-7500 for different height(h) and fin space(s) ratio with fin thickness(t) with varying volume concentration of CNT-H<sub>2</sub>O nanofluid. A performance comparison has also been done among the nanofluids with varying volume fractions and pure water. A new nanofluid thermal performance number (NTPN) is defined as  $NTPN = Nu \frac{k_{nf}}{k_{bf}}$  to compare the heat transfer performance of the nanofluid and base fluid. A multiple regression analysis has been done on the simulation data and correlations are developed of 'f' and 'j' as function of Re and other non-dimensional geometrical terms.

## 2. NUMERICAL ANALYSIS

### 2.1 Physical description of the model

The PFHE of rectangular cross section as shown in Fig 1 is of our interest of study for different ranges as mentioned below with CNT-water nanofluid of a cold fluid to be heated in the heat exchanger.

### 2.2 Governing equations

The governing equations to describe the fluid flow and heat transfer phenomena are continuity, momentum and energy equation. The mentioned equations are solved to get the velocity, pressure and temperature field. We have considered 3-D, turbulent and steady flow model, so the governing equations are as following,

Continuity equation:

$$\nabla \cdot (\rho \mathbf{V}) = 0 \quad (1)$$

Momentum equation:

$$\nabla \cdot (\rho \mathbf{V} \mathbf{V}) = -\nabla P + \nabla \cdot (\mu \nabla \mathbf{V}) \quad (2)$$

Energy equation:

$$\nabla \cdot (\rho c_p \mathbf{V} T) = \nabla \cdot (k \nabla T) \quad (3)$$

Standard K-ε model equations:

$$\bar{u}_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \frac{v_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right] + 2v_t E_{ij} E_{ij} - \varepsilon \quad (4A)$$

$$\bar{u}_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \frac{v_t}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_j} \right] + 2v_t E_{ij} E_{ij} \varepsilon \frac{C_{\varepsilon 1}}{k} - \frac{C_{\varepsilon 2}}{k} \varepsilon^2 \quad (4B)$$

### 2.3 Thermophysical properties of nanofluid

By applying the principle of mass conservation to the two species in finite control volume of the

nanofluids, the nanofluid density was obtained from the relation:

$$\rho_{nf} = \varphi \rho_{np} + (1 - \varphi) \rho_w \quad (5)$$

By applying the principle of calorimetry in the mixture the overall specific heat of the nanofluid was calculated from the relation:

$$c_{nf} = \frac{\varphi(\rho c)_{np} + (1 - \varphi)(\rho c)_w}{\varphi(\rho)_{np} + (1 - \varphi)(\rho)_w} \quad (6)$$

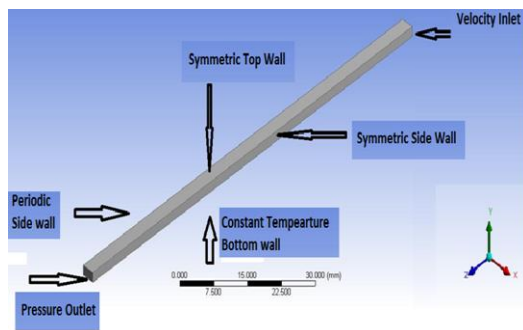
The equation of Azmi et al. [17] was used for the estimation of viscosity and thermal conductivity for water based nanofluid. The following equations are applicable for spherical shaped particles of diameter 20-150nm, temperature between 20-70°C and volume concentration less than 4.0%.

$$\mu_{nf} = \mu_w (1 + \varphi)^{11.3} \left(1 + \frac{T_{nf}}{70}\right)^{-0.038} \times \left(1 + \frac{d_{np}}{150}\right)^{-0.061} \quad (7)$$

$$K_{nf} = 0.8938 K_w (1 + \varphi)^{1.37} \left(1 + \frac{T_{nf}}{70}\right)^{0.2777} \times \left(1 + \frac{d_{np}}{150}\right)^{-0.0336} \quad (8)$$

**2.4 Boundary conditions**

The inlet, outlet and wall boundary conditions are as shown in Fig 2.



**Fig 2: Geometry of the Model and Boundary Conditions**

Top, side wall, symmetric top, symmetric side wall and bottom wall are assumed no slip boundary condition.

Sidewall is kept at periodic boundary condition and bottom wall is kept at constant temperature boundary condition. Inlet set as velocity inlet boundary condition corresponding to the Re range from 3000 to 7500 to assure the flow is turbulent.

**2.5 Meshing and model setup**

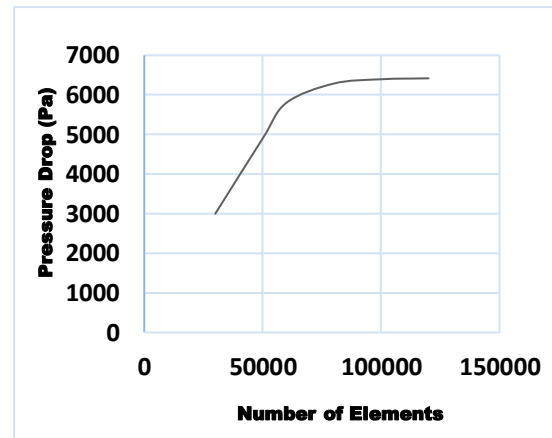
The mesh of the computational domain was generated using a triangular patch conforming method. This mesh contains hexahedral cells having rectangular faces at the boundaries. The generated mesh consisted of 119856 elements and 120552 nodes.

A 3-D, incompressible and pressure-based solver was chosen for the computational domain. A viscous turbulent k-ε model with standard wall function is used for numerical simulation. A 2<sup>nd</sup> order upwind interpolation formula was used for discretization of momentum and energy equation. The conventional SIMPLE (Semi-Implicit Method for Pressure-Linked

Equations) algorithm was used to solve the pressure velocity coupled equations, where several iterations were performed to ensure convergence of the numerical solution was assured by monitoring the scaled residuals to a constant level of 10<sup>-6</sup> for each variable.

**2.6 Grid-independency**

The solution obtained in this work is mesh independent as we have studied the mesh independency test. The result is shown in Fig 3.



**Fig 3: Grid Independence Test**

**Table 2: Validation of the model with experimental results**

Re=6000	Our Simulation Result	Experimental Result of Kays and London	Error
Friction factor(f)	0.01	0.009	10%
Colburn j factor	0.0033	0.004	17.5%

**2.7 Experimental validation of the model**

Validation is done to check the authenticity of the CFD tool. We have validated the model by comparing the results of the experiment done by Kays and London (1984), Table 10-3, FPI 6.2.

**3. RESULTS AND DISCUSSION**

This is clear from the solution of the numerical analysis as shown in Fig 4(a) and Fig 5(a) that with the increase of Re in the turbulent region from 3000 to 7500 friction factor and j factor decreases. These variation of f and j with Re is similar in both nanofluids and pure water. It is also observed in set of curves shown in Fig 4(a), (b) and (c) and 5(a), (b) and (c) that f and j factors are almost independent of the volume fraction of the nanofluid. In Fig 4(b) and 5(b) it is clear that f and j factor are increasing with the ratio of fin height and fin spacing(h/s). In Fig 4(c) and 5(c) it is clear that f and j factor are decreasing with the ratio of fin thickness and fin spacing(t/s). Friction factor is a non-dimensional term used to calculate the pressure drop in the PFHE. On the other hand,

Colburn j factor is a non-dimensional term used to estimate the heat transfer coefficient and hence the effectiveness of the PFHE.

Fig 6(a) and 6(b) are the curves used to compare the thermal performance of water and CNT-water nanofluid with varying volume fraction. It is clear from these curves that heat transfer enhancement is quite significant in nanofluid in comparison to water only in terms of Nusselt number and NTPN (for the same geometry if NTPN is more than heat transfer coefficient would be more). It is observed that NTPN is increased around 18 to 25% after using nanofluid.

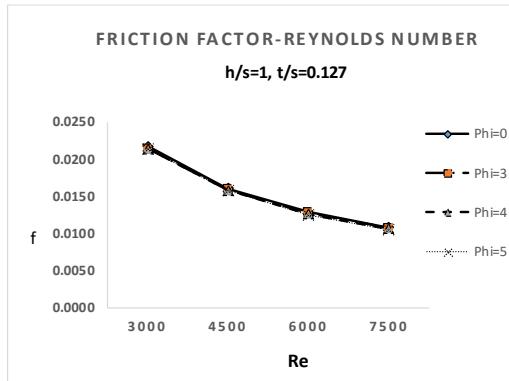


Fig 4a: Variation of friction factor with Re

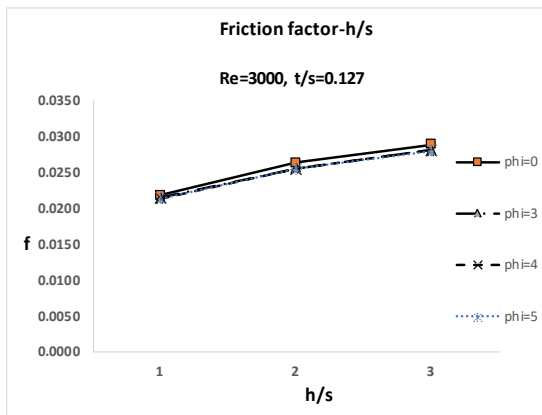


Fig 4b: Variation of friction factor with (h/s)

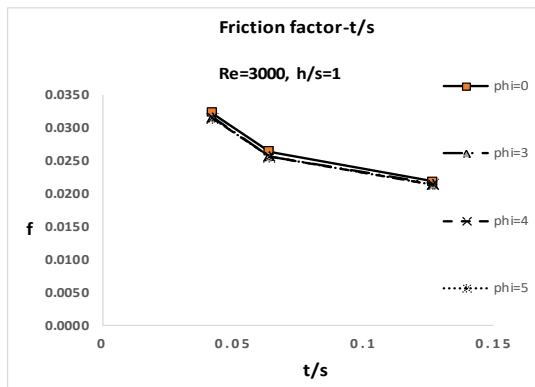


Fig 4c: Variation of friction factor with (t/s)

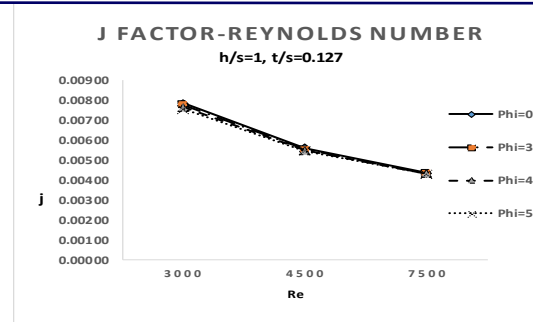


Fig 5a: Variation of Colburn j factor with Re

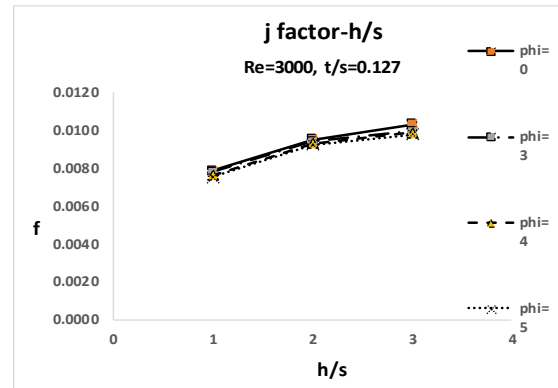


Fig 5b: Variation of Colburn j factor with (h/s)

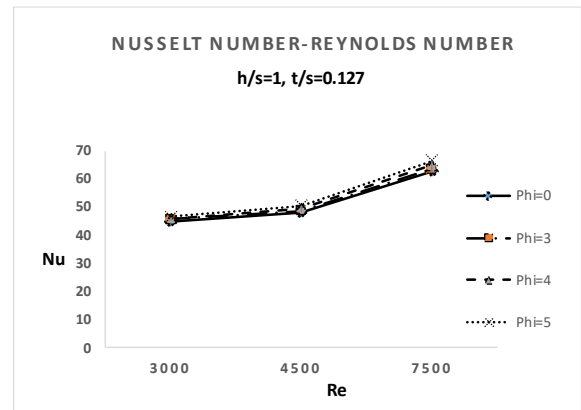


Fig 6a: Variation of Nusselt number with Re

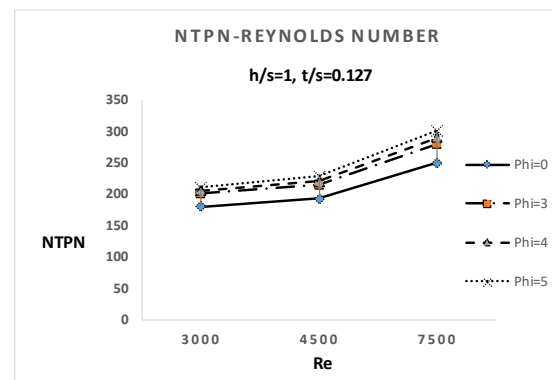


Fig 6b: Variation NTPN with Re

Multiple regression analysis has been done on the simulation results and the correlation of 'f' and 'j' are suggested as function of Re and non-dimensional geometrical parameters (h/s) and (t/s). The correlations are valid in the range of  $3000 < Re < 7500$ ,  $1 \leq h/s \leq 3$  and  $0.04 < t/s < 0.14$

$$f = 3.23Re^{-1.1} \left(\frac{h}{s}\right)^{0.26} \left(\frac{t}{s}\right)^{-0.21} \left(1 - \frac{\phi}{100}\right)^{0.009} \quad (9)$$

$$j = 0.15Re^{-1.2} \left(\frac{h}{s}\right)^{0.29} \left(\frac{t}{s}\right)^{-0.37} \left(1 - \frac{\phi}{100}\right)^{0.011} \quad (10)$$

#### 4. CONCLUSION

From the results of the present study it may be concluded that within a specified limit of fin geometry (i.e.  $1 \leq h/s \leq 3$  and  $0.04 < t/s < 0.14$ ) and turbulent region of  $3000 < Re < 7500$ , f and j factor vary in a similar way. It is observed that f and j both are decreasing with increase of Re and (t/s) but increasing with increase of h/s. Heat transfer coefficient in terms of NTPN for the same geometry found to be increased around 20% when CNT-H<sub>2</sub>O nanofluid is used as one of the working fluids in the PFHE in compare to water only. A regression analysis has been done to find a correlation of f and j factor with Re and non-dimensional geometric parameters h/s and t/s. These correlations are very useful in finding out the heat transfer coefficient and pressure drop characteristic during heat exchanger design problems.

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