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Numerical study of nanofluid heat transfer for different tube geometries – A comprehensive review on performance

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ABSTRACT

The heat transfer performance of a system can be improved using a combination of passive methods, namely nanofluids and various types of tube geometries. These methods can help enhance the heat transfer coefficient and consequently reduce the weight of the system. In this paper, the effect of tube geometry and nanofluids towards the heat transfer performance in the numerical system is reviewed. The forced convective heat transfer performance, friction factor and wall shear stress are studied for nanofluid flow in different tube geometries. The thermo-physical properties such as density, specific heat, viscosity and thermal conductivity are reviewed for the determination of nanofluid heat transfer numerically. Various researchers had measured and modelled for the determination of thermal conductivity and viscosity of nanofluids. However, the density and specific heat of nanofluids can be estimated with the mixture relations. The different tube geometries in simulation work are analyzed namely circular tube, circular tube with insert, flat tube and horizontal tube. It was observed that the circular tube with insert provides the highest heat transfer coefficient and wall shear stress. Meanwhile, the flat tube has greater heat transfer coefficient with a higher friction factor compared to the circular tube.

1. Introduction

Forced convective heat transfer plays a significant role in heat transfer between the surface of the medium and fluid flow in the application of thermal energy transfer. With the improvement of forced convective heat transfer, the energy consumption of a system can reduce. The heat transfer applications include micro heat sink, solar panel, heat exchanger and electronic component using conventional liquids such as water, ethylene glycol and oil as the coolant. However, the performance of these systems has reached the limit; hence enhancement in heat transfer coefficient leads to miniaturization of thermal equipment which has become more important. The convective heat transfer coefficient is strongly dependent on the surface of the solid, thermo-physical properties of coolant and the type of flow. The researcher studied the effects of the flow, which showed that turbulence flow has higher heat transfer coefficient due to the increase in fluid velocity.

Adding metal into the conventional fluid can help in increasing the thermo-physical properties of the fluid. This is because metal has higher thermal conductivity that can help 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 enhance the heat transfer of the system. Although it increases the thermal conductivity of the system but the micro-sized particles also cause some problems such as clogging, increased pressure drop and erosion [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_2O_3, SiO_2 and TiO_2 nano-sized metal oxide particles into water as the base fluid. They found that the enhancement was in effective thermal conductivity which had consequently 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 namely water (W), ethylene glycol (EG), W/EG base mixture and oil [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_2O_3 nanoparticles dispersed in water increased by 30% of its thermal conductivity compared to water at 4.3% volume concentration.

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Later, Xuan and Li [8] dispersed Cu nanoparticles into water for 2.0% and 7.5% volume concentrations. They found that the thermal conductivity increased from 1.24 to 1.78 times higher than the base fluid. In addition, the other researchers also found that the thermal conductivity of the nanofluids are influenced by the shape, size, type of nanoparticles and base fluids [6]. Researchers [9–12] conducted the thermal conductivity measurement for different types and shapes of nanoparticles. They dispersed nanoparticles such as multiwall carbon nanotube, CuO and SiO_2 in water and ethylene glycol. The results showed a positive trend of thermal conductivity enhancement which also increased with temperature. The increase in the thermal property of nanofluids namely, the thermal conductivity, consequently could enhance the heat transfer performance and lead to improved thermal efficiency of the system.

The shape and geometry of the piping system will also influence the heat transfer performance and subsequently altered the optimum weight of the system. Therefore, various types and geometries of the piping system have been studied by many researchers, which include circular tubes, flat tubes, triangular ducts, and rectangular ducts. Ahrend et al. [13] studied the heat transfer of flat tubes and compared the performance with circular tubes for heat exchanger system design. They reported that both heat exchangers could enhance the heat transfer. However, they found that the flat tube improved the heat transfer much better than the round tube. A similar comparison was done by Fiebig et al. [14] who had observed similar outcomes. Besides that, Bergles [15] and Ahuja [16] proposed that heat transfer can be enhanced using the passive technique that modifies the geometry of the surface or flow channels by incorporating inserts or additional devices [17]. Hence, research was conducted on heat transfer of the system by adding inserts such as swirl flow devices, twisted tapes [18] and corrugated tubes.

Nowadays, the numbers of numerical work related to nanofluids operating in various tube geometries and sometimes with inserts were increased considerably. The numerical method is the best way to analyze the heat transfer system for a wide range of Reynolds number and operating conditions. In this paper, the effect of tube geometry and nanofluids towards the heat transfer performance in the numerical system is reviewed. The different tube geometries in simulation work is analyzed namely, circular tubes, circular tubes with inserts, flat tubes and horizontal tubes. The main objective of this paper is to analyze the studies of various types of tube geometries operating with nanofluids as the working fluid in the numerical system for heat transfer augmentation. This paper concentrates on the numerical and simulation of single phase fluids under turbulent flow for various types of nanofluids. The effects of the Reynolds number, volume concentration and temperature are also thoroughly discussed in this paper.

2. Mathematical modelling

The design of tube geometry depends on the application of the cooling system. In this paper, different types of tube geometries are reviewed for numerical systems which includes the circular tubes, flat tubes, rectangular ducts, and plates. The forced convection heat transfer of single phase nanofluids under turbulent flow is studied. Some assumptions were considered for numerical analysis of nanofluids. The nanofluids are assumed to be under steady state condition, incompressible, under turbulent flow and have Newtonian behaviour. The fluid phase and nanoparticle phase are in thermal equilibrium. Further, no slip condition occurs between them and flow in the same local velocity. Finally, the compression work and the viscous dissipation are considered negligible throughout the analysis.

2.1. Governing equation

For single phase flow, the nanofluids act as a normal fluid with enhanced thermo-physical properties due to the inclusion of nanoparticles. The one dimensional steady state governing equations for turbulent flow are given by Eqs. (1) to (3) representing the continuity, momentum and thermal energy equations [19], respectively.

Continuity equation:

$$(\nabla\cdot\overline{V}) = 0 \tag{1}$$

Momentum equation:

$$\rho(\nabla\cdot\overline{V})\overline{V} = -\nabla\overline{P} + \mu(\nabla^2\overline{V}) - \rho(\nabla\cdot\overline{V'V'}) \tag{2}$$

Energy equation:

$$\rho C_P(\overline{V}\cdot\nabla)\overline{T} = k(\nabla^2\overline{T}) - \rho C_P(\nabla\cdot\overline{V'T'}) \tag{3}$$

where $\overline{V}$, $\overline{P}$ and $\overline{T}$ are the time average mean value, and $\overline{V'}$ and $\overline{T'}$ are the turbulence fluctuation in velocity and temperature, respectively. The term of $\rho\overline{V'V'}$ and $\rho C_P\overline{V'T'}$ represent the turbulent shear stress and turbulent heat flux, correspondingly. These additional time-averaged terms that appear in turbulence modelling can be solved if the Reynolds stresses and extra temperature transport term can be related to the mean flow velocity and the heat quantities.

2.2. Turbulence modelling

Several turbulence models have been developed to predict the Reynolds stresses and extra temperature transport for closure of the governing equation of the fluid flow. There are six established turbulence models. The first model is the zero equation, where the eddy viscosity is directly applied to the momentum equation. The next model is the one equation which is usually implemented by the turbulent-kinetic-energy model. The other model of the two equations is the combination of turbulent kinetic energy relation plus a model of dissipation $(K-\varepsilon)$, turbulence length scale $(K-L)$ or vorticity fluctuation$(K-\omega)$. Then, the Reynolds stress model is suitable for more complex and time-consuming solutions but with higher accuracy and wider applicability. The next model is almost model-free which is suitable for large eddy simulations in turbulence flow. The last model is the model-free with the direct numerical simulation of turbulence. The two equations with the combination of turbulent kinetic energy plus dissipation $(K-\varepsilon)$ is the most popular for turbulent simulation [20]. Most of the researchers selected the turbulent kinetic energy plus dissipation in their turbulence modelling. For example, the numerical result of Bayat and Nikseresht [21] found that the realizable and standard $K-\varepsilon$ agreed very well with the experimental result of Gnielinski [22] and Petukhov [23].

Launder and Spalding [24] had proposed the $K-\varepsilon$ model for their turbulence flow. The two equations are expressed in Eqs. (4) and (5).

$$\nabla(\rho\overline{V_k}) = \nabla\left[\frac{(\mu+\mu_t)}{\sigma_k\Delta k}\right] + G_k - \rho\varepsilon \tag{4}$$

$$\nabla(\rho\overline{V_\varepsilon}) = \nabla\left[\frac{(\mu+\mu_t)}{\sigma_\varepsilon\Delta\varepsilon}\right] + C_{1\varepsilon}\left(\frac{\varepsilon}{k}\right)G_k + C_{2\varepsilon}\rho\left(\frac{\varepsilon^2}{k}\right) \tag{5}$$

where G_k is the generation of the turbulent kinetic energy due to the mean velocity gradient, and σ_k and σ_ε are the effective Prandtl number for turbulent kinetic energy and rate of dissipation, respectively. $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are constant. μ_t is the eddy viscosity model and expressed in Eq. (6).

$$\mu_t = \frac{\rho C_\mu k^2}{\varepsilon} \tag{6}$$

where C_μ is the constant with the value of 0.09. The constant for $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are 1.44 and 1.92, respectively while the value for σ_k and σ_ε are 1.0 and 1.3, respectively.

2.3. Thermo-physical properties

The thermo-physical properties of nanofluids play an important role in the simulation because simulation results are strongly affected by