Heat Transfer and Fluid Flow Over a Bank of Circular Tubes
Heat Exchanger Using Nanofluids: CFD Simulation

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Abstract: In the present work, the influence of the basefluids and nanoparticle types on the fluid flow and the heat transfer behavior were studied. Numerical investigation has been done over a bank of tubes heat exchanger in a triangular arrangement. Turbulent forced convection of Al₂O₃ and SiO₂ nanoparticles-based water and glycerin nanofluids was predicted, spherical nanoparticles with a diameter of 30 nm and a volume fraction of 3% were assumed in this simulation. Commercial software so called Ansys fluent used as a computational fluid dynamics code to solve steady (2-D) Navier-Stokes and energy equations adopting finite volume techniques. The k-ε model was used to modelling the effect of turbulent. The obtained results demonstrated that the heat transfer for SiO₂ nanoparticles based deionized water nanofluids was higher than the other types of nanofluids, which means that this working fluid could be promising cooling liquid in many heat exchange systems. The friction coefficient for all nanofluids reduced with increasing Reynolds number for all tubes. Furthermore, the results showed that the heat transfer enhancement increased with increasing the Reynolds number in all nanofluids with constant volume concentration and nanoparticles size.

Keywords: Nanofluids, Heat Transfer, Glycerine, Al₂O₃ and SiO₂ nanoparticles.

1. Introduction

A device designed to transferring of the heat efficiently from a medium to another called heat exchanger. Heat exchanger has a wide range of applications in the area of thermal power plant, automobile, indirect fired heater, waste heat recovery system, air conditioning and refrigeration system (1,2). Crossflow tubular is one of the heat exchangers which called tube bank. This type of exchanger consists of several tubes arranged in rows. There are three kinds of tube banks based on tube arrangements, tubes arranged in a line, triangle, and staggered arrangement. One of the fluids will pass through the tubes and the other crossing it (3,4).

A lot of different applications can be conducted due to the cross- flow's phenomena in the tube banks. One of the main ways to understand the flow of the fluids and the heat transferring during the tube banks is by development the tube banks’ design (5). Two main phenomena are in the heat exchanger. First is the flowing of the fluids in the channels, and the second is between the channel
walls and the fluids in terms of heat transfer. Hence, the developing of the process happening among these two phenomena produces better heat exchanging. Decreasing the dimensions of the channel improves the coefficient of the heat transfer, because of the heat transfer rating count on the surface area to the volume ratio. Furthermore, enhancement of the properties of the nanofluids which used as heat transfer fluids produces a heat exchanger with a high heat transfer coefficient (6,7). Adding a suspension nanoparticle usually between 1-100 nm to dilute the liquid called nanofluids, or carbon nanotubes. The traditional base liquid to which the nanoparticles are added is water or mixture of water with glycol. Adding nanoparticles to the liquids is one of the important ways to improve the thermal properties resulted from improving the heat transfer in the heat exchanger. Nanofluids have qualities of heat transfer better than traditional fluids. Better heat transfer of the nanofluids is because it is very stable and it has a more efficient thermal conductivity than the conventional medium air, water, oil, glycol, and ethylene which normally have low thermal conductivity than the solids (8,9). Nowadays, due to tiny electronic parts, compact heat exchangers and significant heat transfer of cooling fluids become necessary. The main nanofluids' topic is additive of solid particles to the liquid for enhancement of the working fluid (coolant liquids) heat transfer behavior in term of thermophysical properties. The micro heat exchanger limits the heat transfer coefficients by two factors: higher pressure drops decrements the channel dimensions, and limitation of the heat transfer amounting by the used fluid because of his heat transferring (10).

Not enough studies exist to develop the heat transfer of the tube banks using nanofluids. To achieve significant cooling for huge scale integrated circuits, Tuckerman and Peace (11) proposed a microchannel heat sink concept. Based on a review published by Wang and Mujumdar (12) Generally, two approaches implemented in numerical researches to check nanofluids heat transfer properties. Firstly, the suspension of nanoparticles by assuming the continuum theory for fluids. Secondly, better characterization can be achieved for the fluid and solid using the two-phase model. Pantzali et al. (13) studied the performance of the miniature plate heat exchanger that has a modulated surface effected by nanofluids. The study carried out numerically and experimentally, they are obvious that the increment in thermal conductivity companies by a considerable decline in heat capacity also, increase in viscosity. The experimental and numerical studies notified that development in heat transfer using microchannels will be at the expense of increment of pressure drop. However, to reveal the concepts of nanofluids and better understanding, more studies are needed (6).

In this work, a numerical investigation on heat transfer and different types of nanofluids flow over a bank of tubes heat exchanger was done using commercial CFD packages. Besides, a single-phase approach for nanofluids was assumed to study the turbulent forced convection of nanofluids flow through a triangular arrangement of tubes bank with constant wall temperature boundary conditions.

2. Model description and numerical method

2.1 Physical model

Figure 1 shows the physical model of the flow concerning the bank heat exchanger that has a triangular arrange of tubes. The diameter of the tubes is 32 mm and the flow velocity at the inlet is uniform. One part of the whole domain needs to be modeled due to the geometry symmetrized of the tube banks, see Figure 2. The contract for difference CFD domain has three tubes with a circular shape. All the parameters of the geometry for the model have been taken the same as used in (14), see Table 1.
2.2 Physical model

A rectangular channel was used for analyzing. Thus, it is essential to prepare the governing equations (momentum, continuity, and energy) to run and parse the CFD. The equations are continuity steady two-dimensional form, incompressible time-averaged Navier-Stokes equation, and the energy equation to control the considered phenomenon. The mentioned equations can be formed in some systems of a tensor as (15):

Continuity equation: \[ \nabla \cdot ( \rho_{nf} \mathbf{U} ) = 0 \]  \hspace{1cm} (1)

Momentum equation: \[ \nabla \cdot ( \rho_{nf} \mathbf{UU} ) = \nabla P + \nabla (\tau + \tau^T) \]  \hspace{1cm} (2)

Energy equation: \[ \nabla \cdot (\rho_{nf} c_{nf} \mathbf{T} \mathbf{U}) = \nabla \cdot (\lambda_{nf} (\nabla \mathbf{T})) - (\mathbf{U}^T \nabla \mathbf{T}) \]  \hspace{1cm} (3)
The standard k-ε model was used based on Launder and Spalding (16):

\[
\nabla \cdot (\rho_{nf} k \mathbf{U}) = \nabla \cdot \left[ \left( \mu_{nf} + \mu_{nf} \frac{\sigma_k}{\sigma_k} \right) (\nabla k) \right] + G_k - \rho_{nf} \varepsilon,
\]

(4)

\[
G_k = -\rho_{nf} \bar{u}_{ij} \nabla \bar{u}_j \mathbf{U} , \quad \mu_t = \rho_{nf} C_{\mu} \frac{K^2}{\varepsilon} \]

(5)

Where \( C_{\mu} = 0.09, \sigma_k = 1.00, \sigma_{\varepsilon} = 1.30, C_{1s} = 1.44 \) and \( C_{2s} = 1.92 \)

2.3 Thermo-physical properties

Table 2 lists the properties of the base fluid and the nanoparticles properties used in this study. The following equation defines the effective nanofluids properties used. The density and heat capacity equations introduced by (17), while the thermal conductivity equation introduced by (18).

The density:

\[
\rho_{nf} = \varphi \rho_p + (1 - \varphi) \rho_{bf}
\]

(6)

The heat capacity:

\[
c_{p_{nf}} = \frac{\varphi \rho_p c_p + (1 - \varphi) \rho_{bf} c_{pf}}{\rho_{nf}}
\]

(7)

Thermal conductivity:

\[
\frac{\lambda_{nf}}{\lambda_{bf}} = \frac{\lambda_p + (n-1)\lambda_{bf} - (n-1)\varphi(\lambda_{bf} - \lambda_p)}{\lambda_p + (n-1)\lambda_{bf} + \varphi(\lambda_{bf} - \lambda_p)}
\]

(8)

where; (n) is a geometry factor and for used spherical nanoparticles its equal to (3).

The viscosity: The empirical correlation for the nanofluids viscosity as used in (19,20), can be achieved as follow:

\[
\mu_{nf} = \mu_{bf} \left( \frac{1}{(n-1)^{3.44} \left( \frac{d_p}{d_{bf}} \right)^{2.83} \varphi^{1.03}} \right)
\]

(9)

where: \( d_{bf} = \left( \frac{6M}{N_{A} \rho_{bf}} \right)^{1/3} \)

(10)

where, \( M \) is the molecular weight of the base fluid. \( N \) is the Avogadro number, which is equal to 6.022x10^23 mole^{-1}. \( \rho_{bf} \) is the basefluid mass density at temperature \( T_0 = 300 \) K. The used fluid and nanoparticles thermo-physical properties of nanoparticles listed in Table 2.

Table 2 The thermo-physical properties of nanoparticles and working fluids used in the present work.

| Types     | \( \rho [kg/m^3] \) | \( C_p [J/kg.K] \) | \( \mu [Pa.s] \) | \( \lambda [W/m.K] \) |
|-----------|---------------------|--------------------|----------------|---------------------|
| water     | 998.2               | 4182               | 0.001003       | 0.6                |
| Glycerin  | 1259.9              | 2427               | 0.950          | 0.285              |
| Al_{2}O_{3} | 3970               | 765                | -              | 40                 |
| SiO_{2}   | 2200               | 703                | -              | 1.2                |
2.4 Boundary Conditions and Parameter definition

Steady-state case with the turbulent flow of the fluid that considered as constant properties and incompressible. Fixed temperature $T_{in} = 300$ K was applied to the entered fluid with different adjusted velocities $V_{in}$ are implemented. Proportional average pressure equal to zero defined at the model output. Constant temperature determined for the tube wall $T_w = 350$ K. The upper and lower surface of the domain except for the tube surface is in the similarity case. Also, similar boundary condition as shown in Figure 2. The parameters used were defined as following:

Reynolds number $Re$ is defined as:

$$Re = \frac{\rho U_{in} D}{\mu}$$  \hspace{1cm} (11)

Where; $U_{in}$ is the mean velocity entered for the flow channel in the minimum flow cross-section, and $D$ is the diameter, which is equals to $0.5 \, S_f$

In (14) calculate the average Nusselt number and the friction coefficient as follow:

$$Nu_{avg} = \frac{1}{\pi D^2} \int_0^{180} Nu \frac{D}{2} d\theta$$ \hspace{1cm} (12)

$$\epsilon_f = \frac{\tau_w}{0.5 \rho U_{m}^2}$$ \hspace{1cm} (13)

where; $\theta$ is the angular displacement from the front stagnation, and $\tau_w$ is the wall shear stress.

2.5 Numerical method

In present work Fluent solver used to solve the model equations. To display the flowing of the nanofluids and the transfer of the heat around the circular tubes finite volume technique was used. Ansys workbench used to build the geometric model and mesh tool was also used to generate the mesh as shown in Figure 3. The solving equations are the conservation of mass, momentum, and energy. Quadratic elements for the grid was implemented for good prediction of the numerical result at the curved surface. SIMPLE algorithm implemented for coupling between the pressure and the flow velocity. Second-order upwind schemes used as numerical discretization methods for reducing the error. The convergence of the continuity, energy equations, and other predicted variables have been specified on scaled residuals. Generally, for such convergence should be 200 iterations and more, and the iteration was solved implicitly individually.

![Fig. 3. The model geometry and mesh generation](image)

3. Results

The aims of this study can be summarized in two parts. first is to show the numerical results of the deference of the heat transfer and the flow of the tested fluid in high Reynolds number around the staggering array of tubes has a circular shape. The second is to present the results and effect of nanofluids by using different base fluids and various nanoparticles type. Besides, turbulent flow with
a Reynolds numbers 20000, 40000, and 60000 was used to simulate the heat transfer and fluid flow over a bank of tubes in this study.

3.1 Validation of the code and grid independent test.

**Figure 4** shows the result of evaluating the effect of the grid size in the grid independence test. In the present paper, the uniform grid built with four mesh nodes implementing (11543, 20523, 34407 and 48334) at (Re = 7,000). Similar values of pressure drop noticed in the last three nodes accompanied by a pressure drop of less than 1%. Thus, 20523 mesh nodes were chosen as a domain to decrease the time of computation. The code was validated depending on the geometry and boundary conditions introduced in (14). Author studied the flow of the fluid and the heat transfer around a tubes bank has a circular shape with including the vortex generators. The difference between the work was done by Ahmed (14) and the present study in term of pressure drop along with the domain at same Reynolds number are shown in **Figure 5**. In addition, the comparison of this study in the distribution of the velocity for the baseline on the three rows of the tube banks shows good convention with (19) or Re = 7000. see **Figure 6**.

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**Fig. 4.** The grid dependency test for present study.  
**Fig. 5.** Variation of pressure drop along the domain with Reynolds number between the work of Ahmed (14) and present work  
**Fig. 6.** Velocity vectors distribution on the three rows of tube bank for baseline case (water), for Re = 700. (A) Ahmed (14), (B) present model.
3.2 The effect of Reynolds number

Figure 7 (A, B and C) illustrates the relationships between average Nusselt number and Reynolds number for different types of nanofluids over a bank of tubes heat exchanger. It can be clearly seen that the average Nusselt number increased with increasing the Reynolds number for all tubes and all the nanofluids that be used in this study. This direct proportionality between the Nusselt number with the Reynolds number due to the vortexes behind the tube created by the flow when the Reynolds number is pretty high \((Re > 5000)\), also the high Reynolds number led to flow of fluid around the surfaces which make the heat transfer faster. Finally, the fast flow of the fluid conduces increases the convection of the heat transfer. Figure 8 (A, B and C) shows the effect of Reynolds number on friction factor for different types of nanofluids over tubes bank heat exchanger. It can be noted that by increasing Reynolds number, the friction coefficient can be reduced for all tubes and all nanofluids due to the increasing the velocity of flow when increase the Reynolds number and that cause the less frictionless between the surface and boundary layer of fluid by means decreasing the wall shear stresses.

3.3 The effect of nanoparticles type

Two types of nanoparticles \(\text{Al}_2\text{O}_3\) and Si\(\text{O}_2\) and two types of base fluids pure deionized water and glycerine were used in this study. Figure 7 shows the Nusselt number of different nanofluids and different Reynolds number values. Si\(\text{O}_2\) has a thermal conductivity lower than the other nanoparticles \(\text{Al}_2\text{O}_3\) and higher than the water and glycerine. Also, Si\(\text{O}_2\) has a low density compared other nanoparticles, which makes it has the most massive value of the average velocity. Therefore, it is easy for Si\(\text{O}_2\) to pass through other fluids and finally to get the largest average of Nusselt number in deionized water. The velocity of the fluids has a very essential role on the heat transfer of forced convection mode. This functionality resulted to increase the heat transfer of the silica based water nanofluids on three tubes. Additionally, it was stated that the stability of silica based water good compared to other oxide nanoparticles based deionized water, and this could another reason for high heat transfer. The Reynolds number of different nanofluids plays an important role in all tubes; hence, increasing this number led to a decrease in the friction factor, as shown in Figure 8.

3.4 The effect of basefluids types

Water and glycerine as a conventional fluids and their effect on the heat transfer against the Reynolds number for three tubes bank illustrated in Figure 7 (A, B and C). The highest thermal conductivity and isobaric heat capacity for deionized water in nature and the proper mixing of the low density oxide metal of silica nanopowder Si\(\text{O}_2\) inside it, make this mixture enhances for increment the capacity of thermal transporting and increases the Nusselt number compared to other base fluid. The mentioned reasons result in that the Nusselt number achieved by Si\(\text{O}_2\) and \(\text{Al}_2\text{O}_3\) based deionized water is the highest value. Whereas, the lowest value got by Si\(\text{O}_2\) and \(\text{Al}_2\text{O}_3\) based glycerine.
Fig. 7. The average Nusselt number against Reynolds number for different nanofluids on (A) Tube 1, (B) Tube 2, and (C) Tube 3

Fig. 8. The Friction coefficient against Reynolds number for different nanofluids on (A) Tube 1, (B) Tube 2, and (C) Tube 3

4. Conclusions

Heat transfer and nanofluids flow over a bank of tubes heat exchanger was numerically investigated. From the results obtained, it can be concluded:

- Nusselt number increased as Reynolds number increasing for all cases and all tubes.
- Friction coefficient decreases with increasing Reynolds number for all nanofluids and all tubes.
- The SiO$_2$ nanoparticles in both water and glycerin base fluids has the highest average Nusselt number compared to Al$_2$O$_3$ nanoparticles in same basefluids.
The SiO$_2$ and Al$_2$O$_3$ based deionized water nanofluids have the highest value of heat transfer, while SiO$_2$ and Al$_2$O$_3$ based glycerine have the lowest value of heat transfer.

Nomenclature

| Roman Symbols | Description |
|---------------|-------------|
| $D$ | Circular tube diameter, [m] |
| $S_L$ | Longitudinal pitch, [-] |
| $S_T$ | Transverse pitch, [-] |
| $L$ | Length of computational domain, [m] |
| $H$ | Height of channel, [m] |
| $P$ | Pressure, [Pascal] |
| $P_R$ | Prandtl number, [-] |
| $Re$ | Reynolds number, [-] |
| $T_{in}$ | Inlet temperature, [K] |
| $T_w$ | Wall temperature, [K] |
| $d_p$ | Nanoparticle diameter, [nm] |
| $M$ | The molecular weight of fluid, [g/mol] |
| $C_f$ | Skin friction coefficient, [-] |
| $U_\infty$ | Inlet velocity of fluid, [m/s] |

| Greek symbols | Description |
|---------------|-------------|
| $C_\mu$, $C_{\varepsilon1}$, $C_{\varepsilon2}$ | Constants appeared in $k-\varepsilon$ model, [-] |
| $\varepsilon$ | Dissipation rate of turbulence kinetic energy, [m$^2$/s$^3$] |
| $\mu$ | Dynamic viscosity, [Pa.s] |
| $\lambda$ | Thermal conductivity, [W/m.K] |
| $\varphi$ | Particle volume fraction, [-] |
| $\rho$ | Density of the fluid, [kg/m$^3$] |

| Subscripts | Description |
|-------------|-------------|
| eff | Effective |
| bf | basefluid |
| w | wall |
| p | particles |

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