



Analysis study of nano fluids and longitudinal fins on the heat transfer in the counter flow double pipe heat exchanger

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Abstract

In this paper, the heat transfer enhancement in a counter-flow double pipe heat exchanger with longitudinal rectangular fins on the outer surface of the inner tube was numerically investigated by ANSYS 20.R1 software using CFD package and finite volume method. Hot water flows in the inner tube at 60°C while cold water flows in the outer tube at 25°C. The flow is laminar with five mass flow rates from 0.012kg/s to 0.02kg/s for hot side and from 0.01kg/s to 0.03kg/s for cold side. Two types of nanoparticles Al_2O_3 and SiO_2 have been added to produce nanofluids as heat transfer fluid in the external channel (cold fluid flow) with four different concentrations from 0.04% to 1%. Results showed, as the concentration of Nanofluids increased, the heat transfer rate also increased. At 1% concentration, the maximum heat transfer coefficient was belonged to Al_2O_3 /water nanofluid by 22.3% enhancement in comparison with distilled water, while the SiO_2 /water nanofluid has enhancement of 19% in heat transfer coefficient in comparison with the distilled water. Both nanofluids show higher pressure drop compared with distilled water where the SiO_2 /water nanofluid gives a higher drop of (33.2%), while the Al_2O_3 /water nanofluid has (32.1%) of the pressure drop.

Keywords: Heat exchanger, Nanofluids, Longitudinal Fins, Laminar flow, CFD, Rectangular Fins.

الخلاصة: في هذا البحث تم فحص تعزيز نقل الحرارة عددياً في المبادل الحراري ذو الأنابيب المزدوج ذو التدفق المعاكس مع زعانف مستطيلة على السطح الخارجي للأنبوب الداخلي بواسطة برنامج ANSYS 20.R1. تم إجراء المحاكاة العددية ثلاثية الأبعاد باستخدام حزمة CFD وطريقة الحجم المحدود. يتدفق الماء الساخن في الأنبوب الداخلي عند 60 درجة سيليزية بينما يتدفق الماء البارد عند 25 درجة سيليزية في الأنبوب الخارجي. يكون التدفق صفائحياً بخمس معدلات تدفق جماعية من 0.012 كجم/ثانية إلى 0.02 كجم / ثانية للجانب الساخن و 0.01 كجم/ثانية إلى 0.03 كجم / ثانية للجانب البارد. تم إضافة نوعان من السوائل النانوية المحتوية على Al_2O_3 و SiO_2 بمثابة سائل ناقل للحرارة في القناة الخارجية (تدفق السائل البارد) بأربعة تراكيزات مختلفة من 0.04% إلى 1%. وأظهرت النتائج أنه كلما زاد تركيز السوائل النانوية زاد معدل انتقال الحرارة. عند تركيز 1% ، كان الحد الأقصى لمعامل نقل الحرارة ينتمي إلى المائع النانوي Al_2O_3 / ماء بنسبة تحسين 22.3% بالمقارنة مع الماء المقطر ، في حين أن المائع النانوي SiO_2 / ماء له تعزيز بنسبة 19% في معامل انتقال الحرارة مقارنة مع الماء المقطر. يُظهر كلا من الموائع النانوية انخفاضاً أعلى في الضغط مقارنةً بالماء المقطر حيث يعطي المائع النانوي SiO_2 / ماء انخفاضاً أعلى بنسبة (33.2%) ، بينما يحتوي السائل النانوي Al_2O_3 / الماء على (32.1%) من انخفاض الضغط.

1. INTRODUCTION

The double pipe heat exchanger plays an important role in the application of industry and engineering such as air-conditioning, refrigeration, power plants and nuclear reactor etc. It is quite obviously and well documented this kind of heat exchanger making a significant contribution to reheating, preheating, pasteurizing, effluent heating process and digester heating. Double pipe heat exchangers have been used by many small industries due to their small needed for maintenance and low cost of design [1]. The effect of using longitudinal rectangular

and triangular fins, which were placed on the outer surface of the inner pipe in the double pipe heat exchanger, was investigated numerically. Moreover, the effect of using different heights (4mm, 6mm and 8mm) of rectangular fins was examined. The results showed that the rectangular fins could give the highest heat transfer as the 6mm height of fins was optimum [2]. A number of different longitudinal fins (2*3*1000mm), which were installed on the outer surface of the inner pipe in the DPHE, were used. The results showed that as the number of fins increased, they would maximize the heat transfer rate. At the same time, the gap between the fins will be minimized which in turn increased the velocity. Due to the additional velocity of the fluid, the effectiveness will be affected and the pressure drop increased and needed more pumping power. By comparison between the mechanical power losses and the thermal energy gain, it was found that the optimum design was the model with eight fins [3].

Researchers and academics have been paying close attention to the heat transfer enhancement by employing nanofluids in recent years. Because traditional fluids (clean water, ethylene-glycols, etc.) have limited thermophysical qualities and poor thermal conductivity, novel fluids have been developed to improve heat transfer [4]. In a double pipe heat exchanger using TiO_2 /water nanofluid with four mass fractions (0, 0.1, 0.3 and 0.5) in corrugated and smooth pipes. The results show that the corrugated heat exchanger with TiO_2 /water enhanced the heat transfer. The flow characteristics of all three mass fractions are better compared to distilled water which provided a higher capability of heat than smooth double pipe heat exchanger. When cold fluid is the Nanofluids, the heat capacity enhanced as the hot fluid flow increased. [5].

A CFD study in a DPHE by using Al_2O_3 with a volume concentration (1-2%) for both (Newtonian and nonnewtonian), turbulent flow in the outer cylinder and with an insulating surface from the surrounding was conducted. The inner pipe had a constant heat flux and different fin heights (1, 2, 3mm) and fin pitches (80, 160, 320mm) were used. The findings showed that as the volume concentration increased the Nusselt number also increased and decreased with the increasing of the fin pitch. When the friction factor decreased the use of circular fins enhanced the heat transfer for both Newtonian and non-Newtonian nanofluids. The Newtonians' fluid showed a high efficiency [6]. A twisted tape with a rectangular cut on its edges and various nanoscale fluids of titanium dioxide (TiO_2), beryllium oxide (BeO), zinc oxide (ZnO) and copper oxide (CuO), was inserted into the base fluid (water) using a double pipe heat exchanger (DPHE). The results showed that the thermal performance increased for the three types of nanofluids and with the insertion of twisted tape as well. The TiO_2 nanofluids had the best option to exchanged heat with a special twisted tape insertion that was inserted in the flow range.[7].

An experimental study in the counter-flow DPHE by using multiple inner longitudinal fins under turbulent conditions and different volume concentrations of Fe_3O_4 from 0% to 0.4%, was conducted. The results indicated an improvement in heat transfer in the finned tubes with a higher volumetric concentration of 0.4% of Fe_3O_4 - water nanofluids due to the fin design and presence of nanofluids compared to the normal pipe. Furthermore, the friction factor was three to four times greater than that of the ordinary tube and 3.75 times at 0.4% [8]. Two configurations (parallel and counter flow) and cooling fluids of two types of nanofluids (Al_2O_3 and TiO_2) were used in a DPHE and volume concentrations from 0.05% to 0.3%, the results presented that at small rates of flow the usage of nanofluids absorbed more heat than water by improving the thermo physical properties which enhanced the thermal performance of the heat exchanger. The thermal conductivity of (Al_2O_3) nanoparticles was greater than the nanoparticles of (TiO_2) and increased at a high concentration of 0.3% as well for all types of nanofluids the counter flow configuration was better than the parallel flow configuration [9]. An experimental study in a tube in tube heat exchanger using CeO_2 /water nanofluid with (1,2 and 3%) concentrations and various flow rates (1.5, 2, 2.5 and 3lpm). The results illustrated that as the concentration of nanoparticles increased, the viscosity and the thermal conductivity of nanofluid increased and the enhancement observed with 3% concentration. The overall heat transfer coefficient rises with the increase in volume concentration. At higher nanofluid rate of flow, the pressure drops increased and caused the pumping power required also increased. A higher thermal performance factor was found at 2% volume concentration and 2.5lpm flow rate [10]. In the current study, rectangular fins are used with a height of 6mm [2] and two types of nanofluids (Al_2O_3 /water and SiO_2 /water) are used with four concentrations of nanoparticles from (0.4% to 1%). The current study includes two parts; the first one is to investigate the effect of concentrations of nanofluids and the second one is to inspect the type of nanofluids of heat transfer rate/ heat transfer coefficient and Performance evaluation criteria.

2. THE PHYSICAL MODEL AND BOUNDARY CONDITIONS

The CFD model includes a finned tube of longitudinal fins geometry are shown in Figure (1), which simulated using Ansys.R1 program. Table (1) shows the dimensions of the heat exchanger. The flow is laminar for both sides and the hot water flow in the inner tube at 60°C with five mass flow rates (0.012, 0.014, 0.016, 0.018 and 0.02 kg/s), while the cold-water flow in the annular tube at 25°C with five mass flow rates (0.01, 0.015, 0.02, 0.025 and 0.03 kg/s). The outlet field was set as pressure outlet for the hot and cold sections and the thermal condition was coupled for both PVC and copper tubes with stationary wall motion and no-slip shear condition. The simple approach is used to solve momentum and energy equations with a second order of pressure and velocity, resulting in a stable solution to the problem. For the energy equation and other equations, the convergence requirements are set to 10^{-6} . Figure (2) exhibited iterations of the simulations work of the finned pipe of the heat exchanger at 0.016kg/s (hot water mass flow rate) and 0.03 kg/s (cold water mass flow rate). Figure (3) clearly shows the output cold water temperature that became constant at (2000) iteration of the rectangular fins.

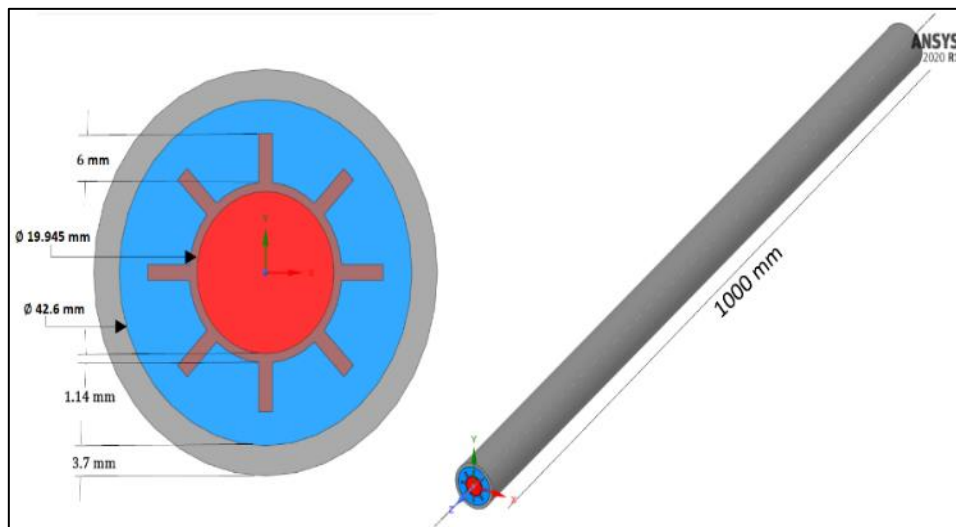


Figure 1 Configurations of the CFD models.

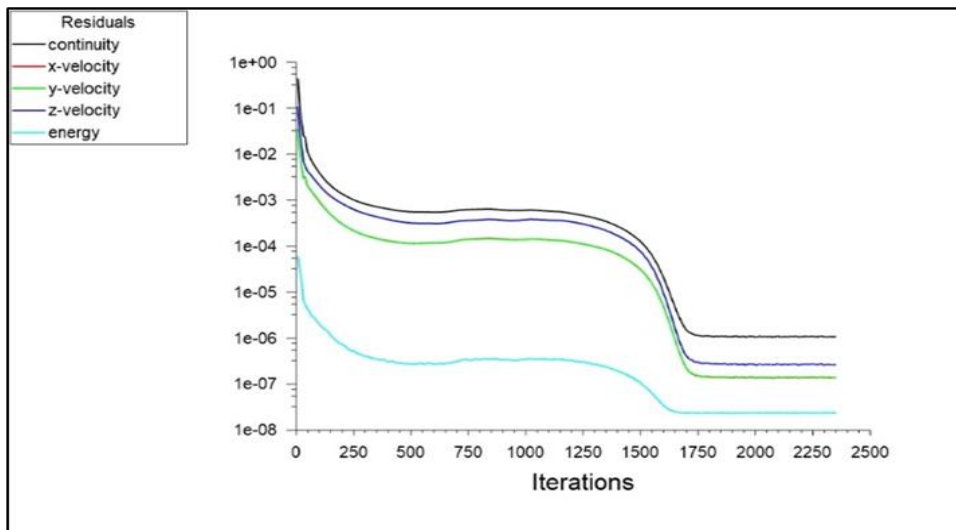


Figure 2 Residuals of the CFD simulation.

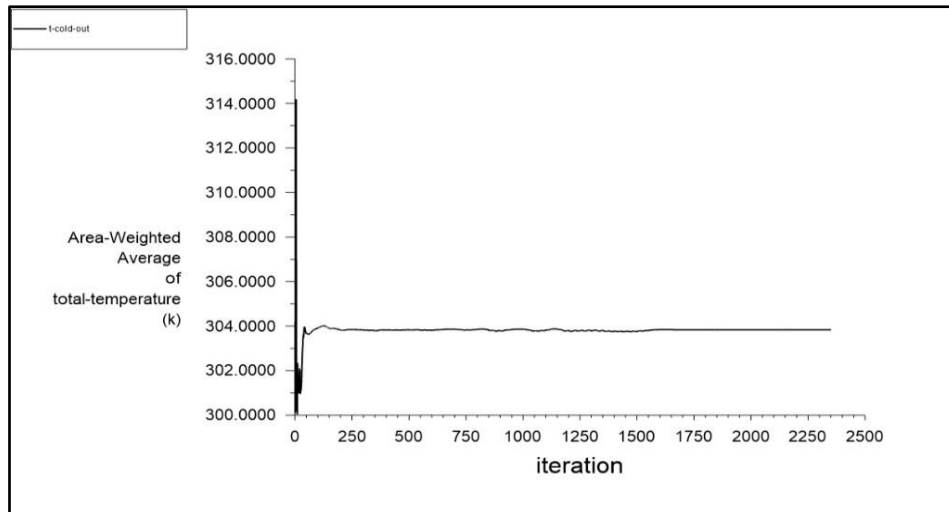


Figure 3 Relation between the outlet cold water Temperature and the number of iterations

Table 1 The dimensions of the heat exchanger

No.	Specifications	Dimensions (mm)	Materials
1	Fin height	6	Copper
2	Fin thickness	2	
3	Inner pipe diameter	19.945	
4	Thickness	1.14	PVC
5	Outer pipe diameter	42.6	
6	Thickness	3.7	
7	Length of Heat exchanger and fins	1000	-

3. GOVERNING EQUATIONS

Some assumptions should be addressed in order to complete the numerical solution and acquire the final form of the governing equations for the current investigation. The flow is assumed to be in a steady-state, laminar and three-dimensional design. The fluid flow is Newtonian in the nature and incompressible state. A single-phase model and the outer surface of the annular pipe were insulated and the Radiation heat transfer is neglected. The continuity equation, momentum conservation, and energy equations in three dimensions are the governing equations that are derived from the basic equations of fluid dynamics and utilized to solve for the flow field [11].

- Continuity equation

$$\frac{\partial \bar{u}}{\partial x} + \frac{\partial \bar{v}}{\partial y} + \frac{\partial \bar{w}}{\partial z} = 0 \quad (1)$$

- Momentum equation

x-axis

$$\left(\bar{u} \frac{\partial \bar{u}}{\partial x} + \bar{v} \frac{\partial \bar{u}}{\partial y} + \bar{w} \frac{\partial \bar{u}}{\partial z} \right) + \left(\frac{\partial}{\partial x} (\overline{u'^2}) + \frac{\partial}{\partial y} (\overline{u'v'}) + \frac{\partial}{\partial z} (\overline{u'w'}) \right) = -\frac{1}{\rho} \frac{\partial P}{\partial x} + \gamma \nabla^2 \bar{u} \quad (2)$$

y-axis

$$(\bar{u} \frac{\partial \bar{v}}{\partial x} + \bar{v} \frac{\partial \bar{v}}{\partial y} + \bar{w} \frac{\partial \bar{v}}{\partial z}) + (\frac{\partial}{\partial x} \overline{(u'v')}) + \frac{\partial}{\partial y} \overline{(v'^2)} + \frac{\partial}{\partial z} \overline{(u'w')} = \frac{1}{\rho} \frac{\partial P}{\partial y} + \gamma \nabla^2 v' \quad (3)$$

z-axis

$$(\bar{u} \frac{\partial \bar{w}}{\partial x} + \bar{v} \frac{\partial \bar{w}}{\partial y} + \bar{w} \frac{\partial \bar{w}}{\partial z}) + (\frac{\partial}{\partial x} \overline{(u'w')}) + \frac{\partial}{\partial y} \overline{(u'v')} + \frac{\partial}{\partial z} \overline{(w'^2)} = \frac{1}{\rho} \frac{\partial P}{\partial z} + \gamma \nabla^2 w' \quad (4)$$

- Energy equation

$$\bar{u} \frac{\partial \bar{T}}{\partial x} + \bar{v} \frac{\partial \bar{T}}{\partial y} + \bar{w} \frac{\partial \bar{T}}{\partial z} = \alpha \nabla^2 \bar{T} + (-\frac{\partial}{\partial x} \overline{(u'T')}) - \frac{\partial}{\partial y} \overline{(v'T')} - \frac{\partial}{\partial z} \overline{(w'T')}) \quad (5)$$

4. RESULTS AND DISCUSSIONS

4.1. Mesh generation and grid independency

In the engineering CFD simulation, to increase the accuracy of the simulation, the complicated geometries are splitting into pieces that may be used to discretize a domain because the more accurate the mesh, the more correct the 3D model. There are two types of mesh structured and unstructured. In the current work, the unstructured mesh of tetrahedron and wedges were used and take notice of wedges elements which are positioned near the walls using inflation layers [12]. Figure (4) shows the mesh of the full model while figure (5) shows the grid independence test was achieved in the reliability of mesh because the poor design of the grid may lead to a decrease in the accuracy of the simulation results. The mesh was refined in each case until it reached a point where the results were not noticeable. Figure (5) shows that the variation in the temperature is not significant with the increase in the number of elements. The number of elements (4788691) was found to be appropriate for the current study.

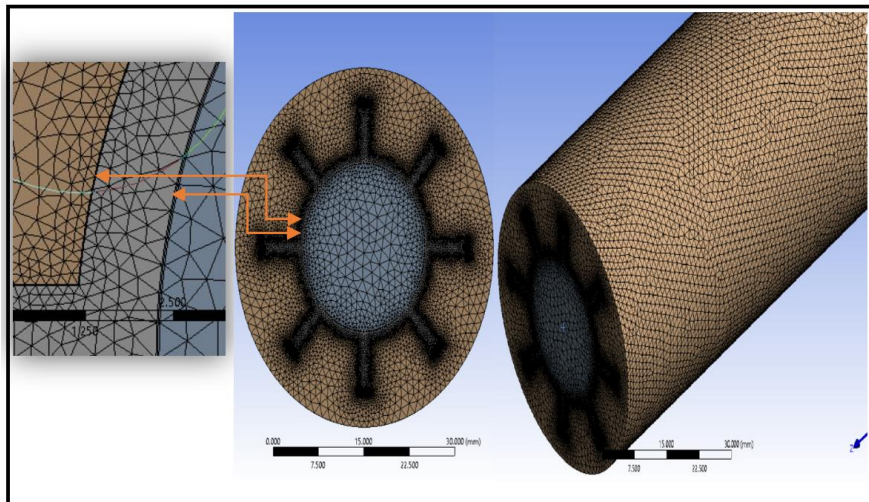


Figure 4 Mesh domain

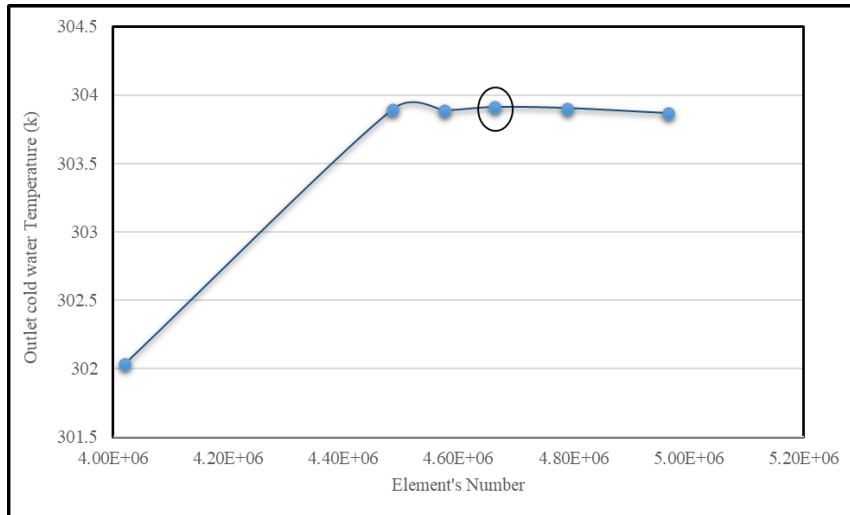


Figure 5 Grid independence test between the cold-water outlet and No. of elements

4.2. Validation work

The current work was validated with an experimental work of a counter-flow double pipe heat exchanger of [9] with a constant inlet velocity. Figure (6) shows the comparison between outlet temperatures of the hot water side and mass flow rates. The error deviation of the hot temperature difference was 9.88% which refers to a good agreement.

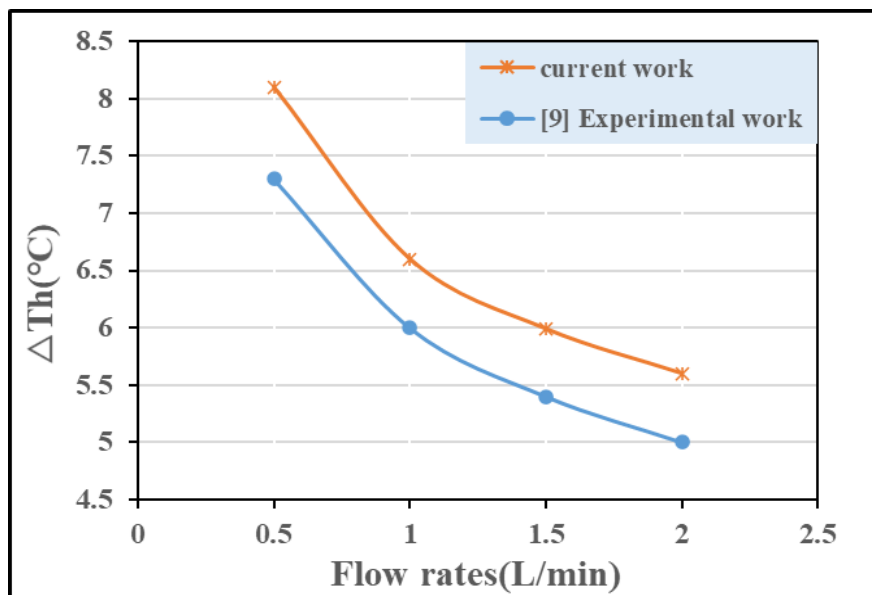


Figure 6 The difference of hot water temperature variation flowrates

4.3. The effect of nanofluids concentrations

In this part, different concentrations of nanofluids with (0.4%, 0.6%, 0.8% and 1%) volume fractions of ($\text{Al}_2\text{O}_3/\text{water}$ and $\text{SiO}_2/\text{water}$) evaluated and investigated. Figure (7) illustrates the relation between the heat dissipation and the mass flow rates of the cold side at different concentrations. It's found that as the concentration of nanoparticles increased, the heat dissipation also raised. The heat dissipation of nanofluids is found to be higher than the water with respect to increasing mass flow rates. The enhancement percentage of heat dissipation where used nanofluids compared to distilled water, The ($\text{Al}_2\text{O}_3/\text{water}$) nanofluids is found to be (5.6%, 6.2%, 6.7% and 7.2%) while ($\text{SiO}_2/\text{water}$) was (3.9%, 5%, 5.6% and 6.3%) at (0.4%, 0.6%, 0.8% and 1%) of volume fractions which both respectively, higher than water and nanoparticles of (Al_2O_3) show higher

heat dissipation than (SiO₂). The heat transfer rate increased with the increase in the concentrations and this increment is due to adding nanoparticles which have higher thermal characteristics in the water that has low thermal conductivity. So, it enhances the thermal properties of water in addition to the thermal conductivity that plays an important role in heat dissipation by fluids.

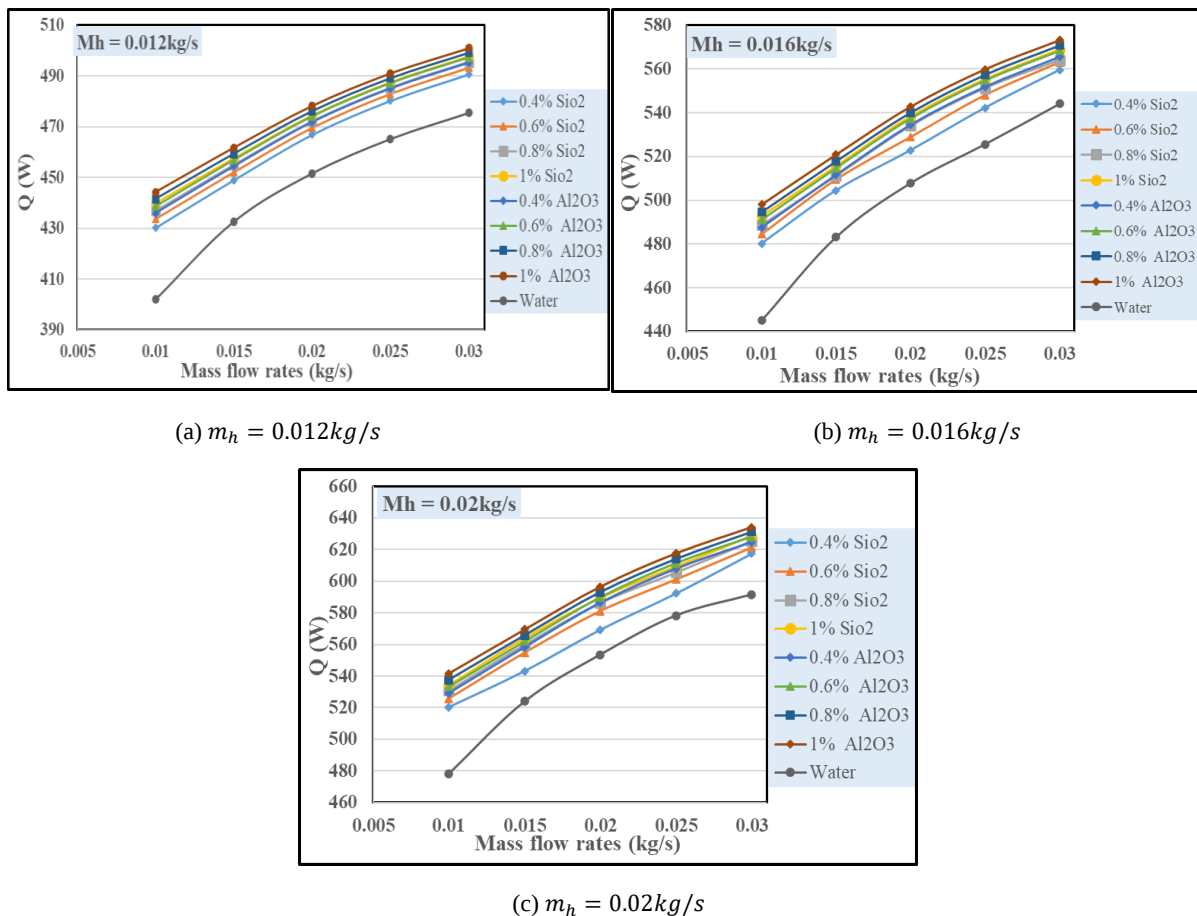


Figure 7 Relation between heat dissipation and cold-water mass flow rates using rectangular fins at *constant* m_h for water and different concentrations of SiO₂ and Al₂O₃

Figure (8) shows the variation between the heat transfer coefficients and different mass flow rates of the cold side. Obviously, it can be seen that the heat transfer coefficients were improved with the increase in the concentration of nanofluids of (Al₂O₃/water and SiO₂/water) and both nanofluids show higher heat transfer coefficients than water. The enhancement of heat transfer coefficients of (Al₂O₃/water) nanofluids is found to be (17%, 19%, 21% and 22.3%) while (SiO₂/water) was (13%, 15.7%, 18% and 19%) at (0.4%, 0.6%, 0.8% and 1%) of volume fractions. This growth in heat transfer coefficient is due to growing of nanofluids' thermal conductivity and lessening the thermal boundary layer thickness which caused by the Brownian velocity which represents the ratio between the nanoparticle diameter and the time taken by it to move a distance equal to the nanoparticle diameter which in turn the intensity of turbulence was increased and caused the thermal variance near the walls raised. The (Al₂O₃/water) nanofluids has (14%) higher heat transfer coefficient than (SiO₂/water) at concentration of (1%).

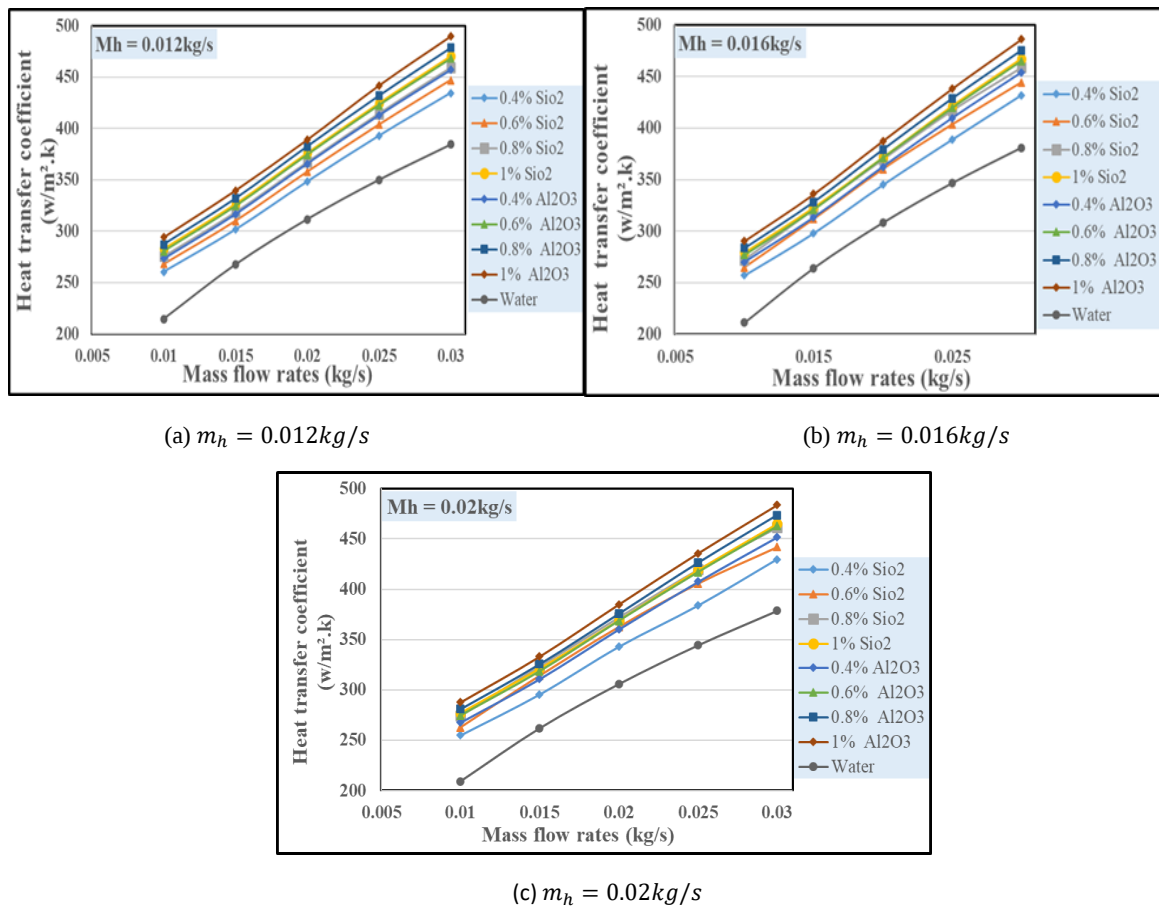


Figure 8 Relation between heat transfer coefficient and cold-water mass flow rates using rectangular fins at constant m_h for water and different concentrations of SiO₂ and Al₂O₃

4.4. The effect of nanofluid types

Two types of nanofluid (Al₂O₃/water and SiO₂/water) with (1%) volume fraction and 20 nm molecular diameter according to the gained results from (Sect. 4.3) are used to investigate their effects on heat transfer rate and heat transfer coefficients for different mass flow rates of (0.01kg/s to 0.03kg/s) for cold side and (0.012kg/s to 0.02kg/s) for hot side. Table 2 indicates the characteristics of the nanoparticles and water.

Table 2 Properties of nanoparticles and water

Fluid	Diameter (nm)	Density (Kg/m ³)	Thermal conductivity (W/m.°C)	Specific heat (J/kg.°C)
Al ₂ O ₃	20	3970	40	765
SiO ₂	20	2220	1.2	703
Water	-	997	0.607	4180

Figure (9) displays the relation between heat dissipation and different mass flow rates of cold side at various types of nanofluids. With the increase of mass flow rates, all nanofluids show a rise in heat dissipation. Highest amount of heat was noted in the (Al₂O₃/water) compared to water and (SiO₂/water) nanofluid. The density of the (Al₂O₃/water) higher than (SiO₂/water) and this is may be the reason that the turbulence increased and effected on the recycling regions with the increasing in the mass flow rates which in turns rising the velocity of the nanofluid. The thermal properties of nanofluids were higher than base water and the thermal conductivity of the (Al₂O₃/water) higher than (SiO₂/water) and for these reasons the temperature gradients near the walls increased

and raising the heat dissipation. The enhancement in heat dissipation at (1%) volume fraction for ($\text{Al}_2\text{O}_3/\text{water}$) was (7.2%), while ($\text{SiO}_2/\text{water}$) was (6.3%) over distilled water.

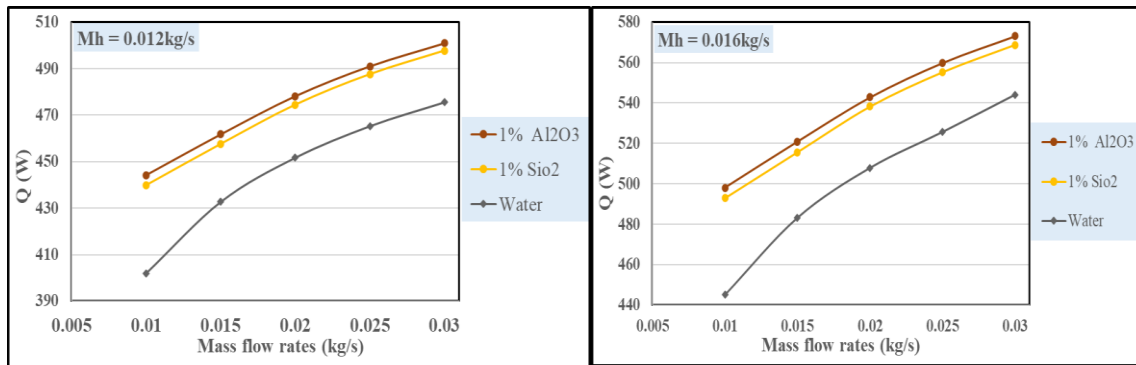
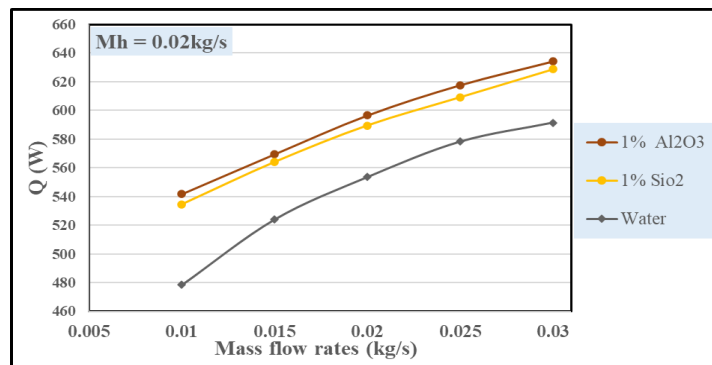
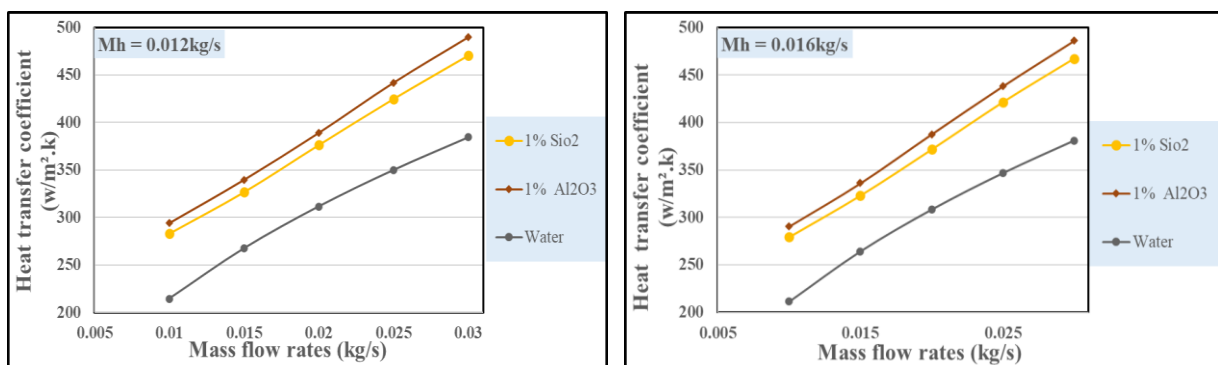
(a) $m_h = 0.012 \text{ kg/s}$ (b) $m_h = 0.016 \text{ kg/s}$ (c) $m_h = 0.02 \text{ kg/s}$

Figure 9 Relation between heat dissipation and cold-water mass flow rates using rectangular fins at constant m_h for water and (1%) concentration of SiO_2 and Al_2O_3

Figure (10) show the variation between the heat transfer coefficients at different nano concentrations for rectangular fins with different mass flow rates of cold side. At concentration of (1%) the $\text{Al}_2\text{O}_3/\text{water}$ show an enhancement of (22.3%) in heat transfer coefficients, while $\text{SiO}_2/\text{water}$ was (19%) over distilled water and the reason for this case is due to thermal conductivity of nanofluids was higher than water and the thermal conductivity of $\text{Al}_2\text{O}_3/\text{water}$ was higher than $\text{SiO}_2/\text{water}$.

(a) $m_h = 0.012 \text{ kg/s}$ (b) $m_h = 0.016 \text{ kg/s}$

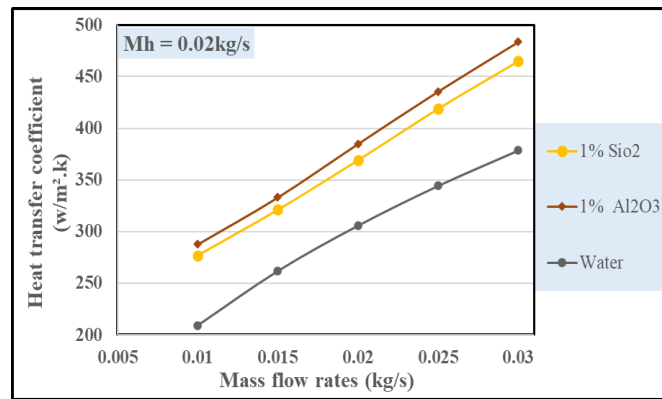
(c) $m_h = 0.02 \text{ kg/s}$

Figure 10 Relation between heat transfer coefficient and cold-water mass flow rates using rectangular fins at constant m_h for water and (1%) concentration of SiO₂ and Al₂O₃

4.5. The effect of the nanofluids on the thermal performance factor

Figure (11) depicts the effect of nanoparticles' concentrations of (Al₂O₃/water) and (SiO₂/water) on Performance evaluation criteria with the using of rectangular fins at 6 mm height. It was noted that by increasing concentrations of nanofluids from (0.4% to 1%) and increasing mass flow rates for cold side, the performance for the heat exchanger increased for both nanofluids due to increasing the heat transfer performance. At (1%) concentration the (Al₂O₃/water) has (9.4%) an enhancement of thermal performance, while (SiO₂/water) has (5.2%) over distilled water.

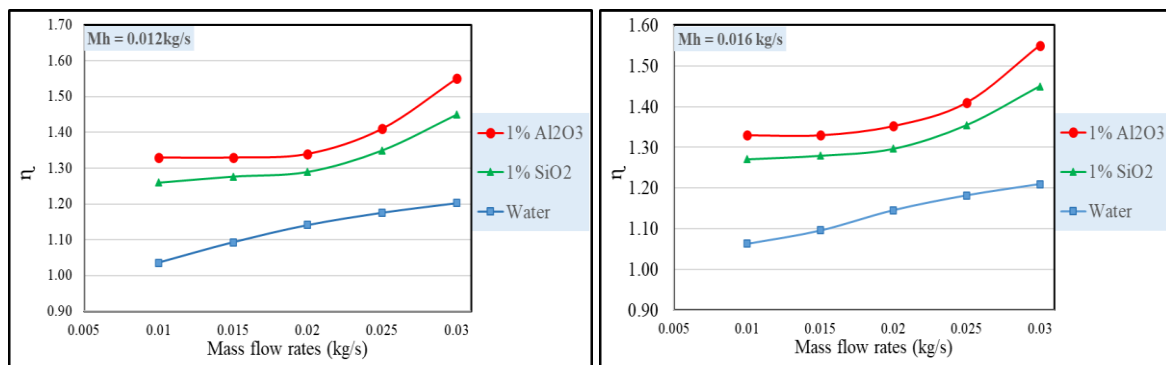
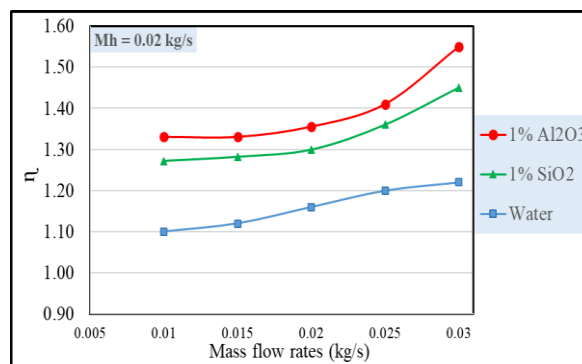
(a) $m_h = 0.012 \text{ kg/s}$ (b) $m_h = 0.016 \text{ kg/s}$ (c) $m_h = 0.02 \text{ kg/s}$

Figure 11 Relation between Performance evaluation criteria and cold-water mass flow rates using rectangular fins at constant m_h for water and (1%) concentration of SiO₂ and Al₂O₃

4.6. The effect of nanoparticles concentrations on pressure drops

In this part the effects of (Al_2O_3 and SiO_2) nanoparticles on pressure drop was investigated at a practical diameter of (20nm) and four volume fractions from (0.4% to 1%) using rectangular fins and water as a base fluid as shown in Figure (12) which displays the difference between the pressure drop and different mass flow rates of cold side for water and nanofluids at (1%) concentration. By compared to pure water. As the concentration of nanofluids increased, the pressure drop was also increased due to the higher density and viscosity of nanofluids with compared to water. The pressure drop of SiO_2 nanoparticle has a pressure drop of (33.2%), while (Al_2O_3) has (32.1%) with compared to water at (1%) compared to distilled water also it has (3.5%) pressure drop higher than (Al_2O_3) and since the ($\text{SiO}_2/\text{water}$) nanofluid has a smaller amount of density than the ($\text{Al}_2\text{O}_3/\text{water}$) nanofluid, it has a faster velocity, resulting in increased pressure drop.

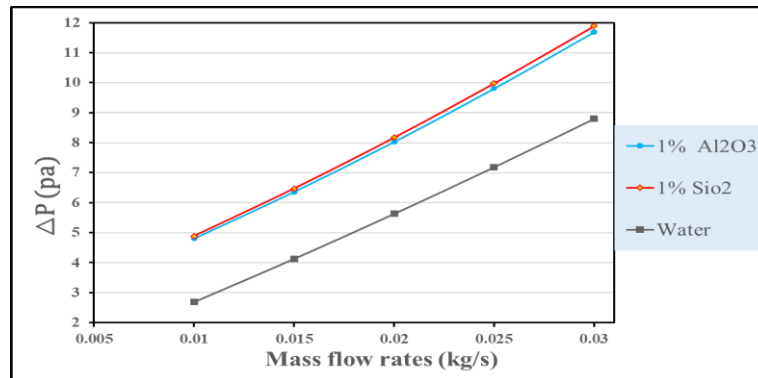


Figure 12 Relation between pressure drop and cold-water mass flow rates using rectangular fins at *constant* m_h for water and concentration (1%) of SiO_2 and Al_2O_3

4.7. Temperature Contours

In the present study, the heat transfer rate increased by using longitudinal fins and adding nanoparticles to distilled water to increase the thermal conductivity. Two types of nanofluids, (Al_2O_3 and $\text{SiO}_2/\text{water}$) nanofluid with particles diameter of 20nm and volume fraction of (0.4%, 0.6%, 0.8% and 1%) have been considered. Rectangular fins were presented over mass flow rates ranged from (0.01 to 0.03) kg/s for cold side and (0.012 to 0.02) kg/s for hot side as shown in figure (13) displays the temperature contours along finned tube of rectangular fins for different nanofluids with (1%) concentration which simulated by ANSYS FLUENT its obviously seen that the temperature of cold fluid increase from the inlet to outlet while the hot fluid decreased the temperature from inlet to outlet. The higher temperature represented by reddish colour while the lower temperature represented the bluish colour.

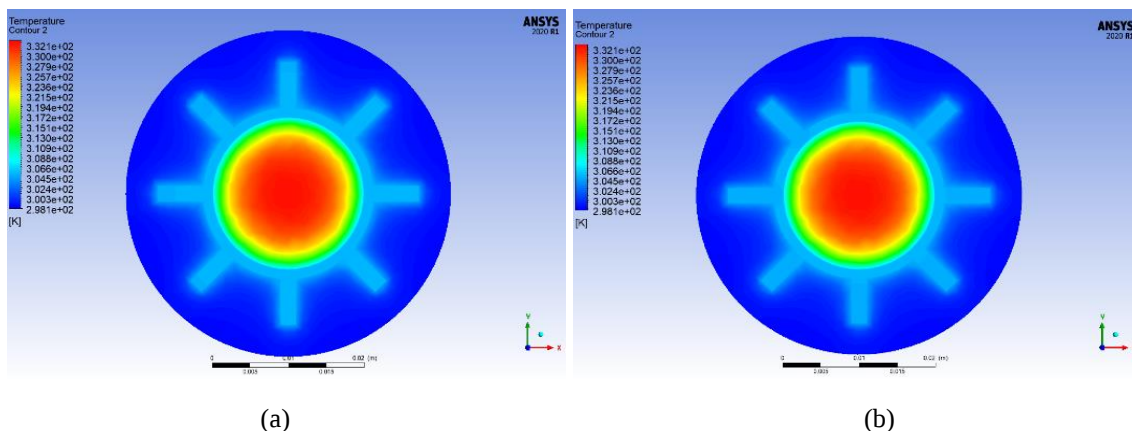


Figure 13 Displays the contours of mid-section at 1% concentration of a) SiO_2 , b) Al_2O_3 at $m_h = 0.016 \text{ kg/s}$ and $m_c = 0.03 \text{ kg/s}$

5. CONCLUSIONS

The heat exchange of various water-based nanofluids in a double pipe counter-flow heat exchanger with longitudinal fins placed on the outer surface of the inner tube is investigated in this paper. The following are the results reached.

- The design of the rectangular fins and the presence of nanofluids could improve the heat transfer for finned tubes
- By increasing the concentration of the nanofluids from 0.4% to 1%, the thermal performance increased for both ($\text{Al}_2\text{O}_3/$ and SiO_2) with compared to water and the 1% concentration gives a high heat transfer performance.
- The maximum thermal performance factor attained at 1% concentration and by use of ($\text{Al}_2\text{O}_3/\text{water}$) nanofluid which gives an enhancement of (44.7%) higher than ($\text{SiO}_2/\text{water}$) nanofluid.
- Heat transfer coefficient increases with the increase of mass flow rates and concentration of nanofluids and the ($\text{Al}_2\text{O}_3/\text{water}$) nanofluid has 14% an enhancement over $\text{SiO}_2/\text{water}$ at 1% concentration.
- As the nanofluids concentrations increased, the pressure drop was also increased and the ($\text{Al}_2\text{O}_3/\text{water}$) nanofluid gives a high rate of pressure drop by (3.5%) higher than ($\text{SiO}_2/\text{water}$) nanofluid.

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