THE HEAT TRANSFER ENHANCEMENT OF CONCURRENT FLOW AND COUNTER CURRENT FLOW CONCENTRIC TUBE HEAT EXCHANGERS BY USING HEXAGONAL BORON NITRIDE-WATER NANOFLUID

by

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Heat exchangers are used in many applications including chemical, oil, and gas power generation, refrigeration, pharmaceuticals, and food processing. Because of their widespread usage, they have various types to serve at different working conditions. Increasing the performance of heat exchangers has become a very interesting field of study since the efficiency of various industrial and domestic systems depend on them. In this study, a coaxial double tube experimental set-up was prepared, and the effect of using nanohexagonal boron nitride nanofluid as hot working fluid on the heat transfer performance increase was investigated. The experiments were carried out in concurrent flow and counter flow conditions for various hot fluid-flow rates to obtain the heat transfer coefficients. Nanohexagonal boron nitride obtained in powder form was used to prepare the nanofluid by a two-step method. The 4 kg nanofluid containing 2% nanohexagonal boron nitride with 0.5% Triton X-100 as a surfactant in terms of mass ratio was prepared for the experiments. Heat transfer experiments were carried out three times by using the prepared nanohexagonal boron nitride/water nanofluid and pure water as hot fluid to reach more precise results. In the result of this study, the total heat transfer coefficient showed an average improvement of 48.78% for the concurrent flow heat exchanger, while an average improvement of 0.36% was observed in the counter flow conditions compared to the base fluid. This study shows the potential of application of nanohexagonal boron nitride/water nanofluid in heat management applications.

Key words: *tube heat exchangers, boron nitride, nanofluid, heat transfer*

Introduction

Heat exchangers have found widespread usage area in today's technology, such as heating and cooling of buildings, in industrial production processes, in the energy production and energy recovery processes, cooling of internal combustion engines and in many applications we use in our daily life. It is thought that through the studies to improve the performance of the heat exchangers, both the size of the heat exchangers can be reduced and the energy efficiency of the heat exchangers can be increased. For this reason, increasing the performance of heat exchangers has become a very interesting field of study. The heat transfer fluid's thermal

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conductivity has been an important research area in the studies on improving the performance of heat exchangers.

The idea of improving the poor thermal conductivity of the conventional fluids by adding millimeter or micrometer sized particles is not new and can be based on Maxwell. However, the usage of large particles has caused these particles to settle rapidly into the fluid and have caused wear and blockage problems [1]. Nanofluids are promising solution to overcome these problems. The term nanofluid which is a suspension of nanoparticles (1-100 nm) in conventional liquids is coined by Choi in 1995 at the Argonne National Laboratory [2]. Thanks to advances in technology, many studies have been reported on various nanoparticles like Al_2O_3 [3], ZnO [4], TiO_2 [5], SiO_2 [6], Fe_3O_4 [7], Cu [8], carbon nanotubes [9], and Au [10]. The thermal conductivity of conventional fluids has been enhanced surprisingly by suspended nanoparticles with small particle concentrations [11]. Studies on nanofluids indicated that remarkable changes occurred on heat transfer characteristics of the suspension [12, 13].

Many studies have been conducted on the intensifying the performance of the assorted kinds of heat exchangers by using nanofluids. For example, Peyghambarzadeh *et al.* [14] studied intensifying the cooling performance of automobile radiator by using Al_2O_3-water nanofluid. They found that heat transfer efficiency can be enhanced up to 45% in comparison with pure water. Kabeel *et al.* [15] experimentally studied improving the heat transfer characteristics of the corrugated plate heat exchanger. They reached a maximum increase of 13% in heat transfer coefficient by using Al_2O_3 nanofluid. Colangelo *et al.* [16] investigated the application of Al_2O_3-water nanofluid on flat solar thermal collectors. They found that convective heat transfer coefficient can be intensified up to 25% at a concentration of 3% by volume. Sonawane *et al.* [17] investigated the heat transfer performance of Al_2O_3-water nanofluids as a coolant used in concentric tube heat exchanger. They showed that the average heat transfer rates for nanofluids are higher than those for the water as cooling media and this increases with concentration of nanofluids' composition. Sozen *et al.* [18] investigated the heat transfer improvement of ammonia/water couple with Al_2O_3 particles in nano size in diffusion absorption coolers. Their results of experiments indicated that the system with nanoparticles provided better absorption of heat from the generator and faster evaporation of the cooler from the cooling/absorption fluid. They observed that the operation time of the system was decreased due to shorter heat transfer periods. Srinivas and Vinod [19] experimentally investigated heat transfer intensification of an agitated shell and helical coil heat exchanger by using three water based nanofluids (Al_2O_3, CuO, and TiO_2). They showed that higher stirrer speed, nanofluid concentration and shell side fluid temperature increased the effectiveness of heat exchanger. They observed maximum augmentation of 30.37%, 32.7%, and 26.8% in effectiveness of heat exchanger for Al_2O_3, CuO, and TiO_2 nanofluids in order. Sozen *et al.* [20] studied enhancing the thermal performance of parallel and cross-flow concentric tube heat exchangers using fly ash nanofluid. In another study Sozen *et al.* [21] investigated comparison of improving heat transfer by using Al_2O_3 and fly ash nanofluids in parallel and cross-flow concentric tube heat exchangers. They showed that efficiencies of the parallel flow concentric tube heat exchanger and the cross-flow concentric tube heat exchanger can be improved by 31.2% and 6.9%, respectively, when using fly ash nanofluid. They reached an improvement of 5.1% and 2.8%, respectively, when using Al_2O_3 nanofluid. Ilhan *et al.* [22] investigated the preparation, stability and thermophysical properties of hexagonal boron nitride (hBN) containing water, ethylene glycol and ethylene glycol water mixture (by volume 50%) based nanofluids. It is observed that the hBN nanofluids have remarkably higher thermal conductivity values than their corresponding base fluids, depending on the volume concentration of dispersed nanoparticles. Moreover, water based hBN nanofluids with relatively dilute particle suspensions, exhibits significant increase in thermal conductivity with respect to the viscosity increase.

The hBN is a highly stable dielectric ceramic that shows useful properties including exceptionally high thermal conductivity [22]. Even though it is usually more expensive than other nanoparticles, its favorable properties render nano hBN a suitable material for heat transfer applications.

There are several reports of various nanofluids including nano hBN for heat transfer enhancement but there is no consensus among them, thus new studies are needed to verify and improve the heat transfer efficiency of nanofluids at different experimental conditions. This study aims to experimentally compare the enhancement of heat transfer in concurrent flow and counter flow conditions by using nanohBN-water nanofluid instead of pure water. For this study, a coaxial double tube experimental set-up was prepared which can be adjusted to either concurrent flow condition or counter flow condition. NanohBN-water nanofluid is prepared and the thermophysical properties of the nanohBN-water nanofluid was determined before the experimental analysis. As discussed in the introduction, nanoparticle concentration is an important factor in heat transfer enhancement. It is shown in various studies that increasing the amount of nanoparticles also increase the heat transfer efficiency. In this study, the highest concentration of nanoparticles that can be prepared with our supply of hBN nanoparticles that would give the maximum heat transfer enhancement is chosen. It is decided to keep the nanoparticle concentration fixed and change the flow rates to evaluate the performance of the nanofluid compared to base water fluid.

Materials and method

Experimental set-up

A coaxial double tube experimental set-up was prepared for this study which is capable to compare the enhancement of heat transfer in concurrent flow and counter flow conditions. The prepared nanofluid and pure water circulated in turn as hot working fluid through the inner tube of the heat exchanger. The schematic diagram of the experimental set-up is given in fig. 1.

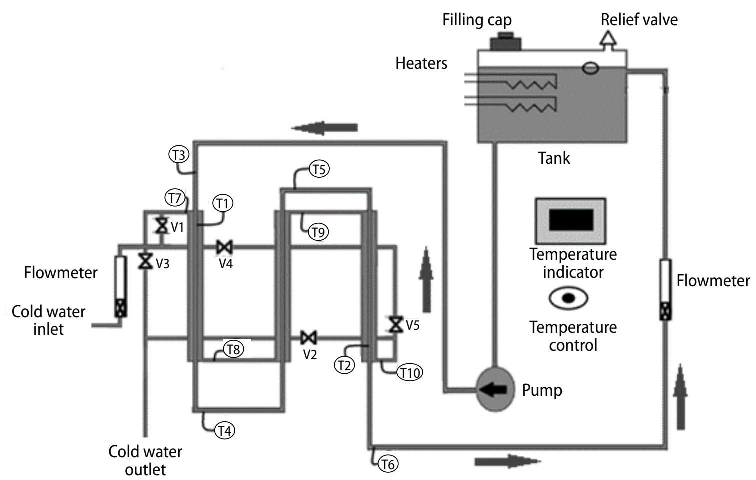

Figure 1. The schematic diagram of the experimental set-up

The heat exchanger was made of copper. Hot working fluid heated in a tank by an electrical resistance type water heater and circulated by a pump unidirectional into the inner tube of the heat exchanger. As hot working fluid circulated through the heat exchanger it got cooler and after it passed through a flowmeter and turned to the tank to get reheated. The tank of the experimental set-up was equipped with a filling cap and a drain so that the hot working fluid could be changed among nanofluid and pure water. The direction of the circulation of the cold fluid could be changed by the valves shown in the experimental set-up. Hot working fluid circulated only by one directional flow in the experimental set-up. There were 10 thermocouples represented by T1-T10 in the experimental set-up to monitor the temperature at different points. The technical specifications of the experimental set-up are given in tab. 1.

Table 1. The technical specifications of the experimental set-up

The length of the heat exchanger	3×350 mm
Heat exchanger inner tube outside diameter	9.5 mm
Heat exchanger inner tube inside diameter	7.9 mm
Heat exchanger outer tube outside diameter	12.7 mm
Heat exchanger outer tube inside diameter	11.1 mm
Internal heat transfer area	0.0261 m^2
External heat transfer area	0.031 m^2
Mean heat transfer area	0.0288 m^2
Flow area	49×10^{-6} m^2

Preparation of the nanofluid

Two-step technique was used for the preparation of the nanofluid examined in the experimental study. NanohBN was supplied in powder form. The 4 kg of nanofluid, that was containing 2% nanohBN particles by mass, was prepared to be used in the experimental set-up. The proper amount of nanohBN particles was weighed with the aid of a precision balance and transferred into the container that was used for preparing the nanofluid. By mass, 0.5% Triton X-100 was used as the surfactant. The surfactant weighed with the aid of a precision balance was transferred into the container that was containing the nanohBN particles. The remaining part, 3900 g, was filled with pure water that was weighed on a precision balance and transferred into the container. The mixing process was carried out mechanically until that there was no particles precipitated at the bottom of the container. The mechanically mixed fluid was then mixed in an ultrasonic bath. Ultrasonic bath temperature was set at 45 °C and ultrasonic mixing was applied for 60 minutes.

Measurement of the thermophysical properties of the nanofluid

The thermophysical properties of the prepared nanohBN-water nanofluid was determined before the experimental analyses. Firstly, the density of the prepared nanohBN-water nanofluid was calculated at each temperature at which the experiments were carried out. For density calculations, the value of nanofluid's mass that was 100 ml in volume was measured by a precision balance for each temperature value at which the experiments were performed. Then density was calculated using eq. (1) for each temperature value at which the experiments were conducted. In eq. (1) ρ denotes density, m denotes mass, and V denotes volume.

$$\rho = \frac{m}{V} \tag{1}$$

In order to determine the specific heat capacity of the prepared nanohBN-water nanofluid at each temperature at which the experiments were carried out, an experimental set-up, which operated with nested containers, was used.

Theoretical analysis

In eq. (2), $\dot{Q}_h$ denotes the rate of heat transferred from the hot fluid and can be written:

$$\dot{Q}_h = \dot{m}_h C_{p,h}(T_3 - T_6) \tag{2}$$

Since concurrent flow and counter flow of cold fluid changed the direction of input and output of the cold fluid, the absolute value was used instead of parentheses. In eq. (3), $\dot{Q}_c$ denotes the rate of heat transferred to the cold fluid and can be written:

$$\dot{Q}_c = \dot{m}_\mathrm{c} C_{p,\mathrm{c}} \left| T_7 - T_{10} \right| \tag{3}$$

In ideal conditions the maximum efficiency would be achieved when the rate of heat transferred from the hot fluid equals to the rate of heat withdrawn by the cold fluid, assuming that there is no heat loss because of the thermal insulation and the resistance caused by the thickness of the inner pipe as well as there is no equipment and measurement errors. So, eq. (4) can be written:

$$\dot{Q}_h = \dot{Q}_c = \dot{Q} \tag{4}$$

In eq. (5), U denotes the overall heat transfer coefficient and $\Delta T_{l,m}$ denotes the log mean temperature difference so eqs. (5) and (6) can be written:

$$\dot{Q} = \dot{Q}_h = U A_m \Delta T_{l,m} \tag{5}$$

$$\Delta T_{l,m} = \frac{\Delta T_\mathrm{inlet} - \Delta T_\mathrm{outlet}}{\ln\left(\dfrac{\Delta T_\mathrm{inlet}}{\Delta T_\mathrm{outlet}}\right)} = \frac{(T_3 - T_7) - (T_6 - T_{10})}{\ln\left(\dfrac{T_3 - T_7}{T_6 - T_{10}}\right)} \tag{6}$$

So, the overall heat transfer coefficient can be calculated by eq. (7):

$$U = \frac{\dot{Q}_h}{A_m \left[\dfrac{(T_3 - T_7) - (T_6 - T_{10})}{\ln\left(\dfrac{T_3 - T_7}{T_6 - T_{10}}\right)} \right]} \tag{7}$$

The methodology for obtaining the hot fluid's heat convection coefficients is given in eq. (8). In eq. (8), h_h denotes the hot fluid's heat convection coefficient, A_h denotes heat transfer area, and $\Delta T_{l,m_\mathrm{inner}}$ denotes the log mean temperature difference for this stream so eqs. (8)-(10) can be written:

$$h_h = \frac{\dot{Q}_h}{A_h \, \Delta T_{l,m_\mathrm{inner}}} \tag{8}$$

$$\Delta T_{l,m_{inner}} = \frac{\Delta T_\mathrm{inlet} - \Delta T_\mathrm{outlet}}{\ln\left(\dfrac{\Delta T_\mathrm{inlet}}{\Delta T_\mathrm{outlet}}\right)} = \frac{(T_3 - T_1) - (T_6 - T_2)}{\ln\left(\dfrac{T_3 - T_1}{T_6 - T_2}\right)} \tag{9}$$

$$h_h = \frac{\dot{Q}_h}{A_h \left[\dfrac{(T_3 - T_1) - (T_6 - T_2)}{\ln\left(\dfrac{T_3 - T_1}{T_6 - T_2}\right)} \right]} \tag{10}$$

Results and discussion

Uncertainty analysis

For checking the reliability of the experimental set-up, the experimental system was tested by using pure water before starting the experimental measurements. To reach more precise results the experiments were repeated three times. The temperature values were measured by using high accuracy J-type thermocouples. The experimental set-up was insulated thermally in order to minimize heat losses. All flow rate and temperature measurements were conducted at steady state during the experiments. In order to conduct the uncertainty analysis of the experimental findings, the uncertainty analysis method suggested by Kline and McClintock [23], was used. Calculations were made by using eq. (11) according to the uncertainty analysis method suggested by Kline and McClintock. In eq. (11) R denotes the magnitude to be measured and n is the number of the independent variables affecting this magnitude. Then the independent variables are shown by: $x_1, x_2,...,x_n$; so R can be written as $R = R(x_1, x_2,...,x_n)$. The error rate of each independent variable is shown by $w_1, w_2,........., w_n$, and W_R denotes the error rate of R. So, considering the sensitivity of the devices used for the experiments the uncertainty for the calculated heat transfer coefficients remained at 3%.

$$W_R = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \ldots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{1/2} \tag{11}$$

Results and discussion for the concurrent flow heat exchanger

For obtaining the values and characterizing the changes of overall heat transfer coefficients and the hot fluid's heat convection coefficients for concurrent flow condition, the experiments were conducted at a constant cold fluid-flow rate of 0.01 kg/s and varying hot fluid-flow rates of 3.2 lpm, 3.7 lpm, 4.2 lpm, 4.7 lpm, 5.2 lpm, 5.7 lpm, 6.2 lpm, 6.7 lpm, 7.2 lpm, 7.7 lpm, 8.2 lpm, 8.7 lpm and 9.2 lpm. The graph showing the variation of the overall heat transfer coefficients as a result of the measurements obtained from the experiments performed and the calculations made by using these measurements are given in fig. 2.

When the graph in fig. 2 is examined, it can be seen that the overall heat transfer coefficients of nanohBN-water nanofluid is higher than those of pure water's at all flow rates examined in the experiments for concurrent flow. The fluctuations seen in the plot are likely due to the experimental error that is also shown with the error bars for each data point. This result has shown that the use of nanohBN-water nanofluid improves heat exchanger performance at all flow rates examined in the experiments for concurrent flow. When the nanohBN-water nanofluid is used as the hot working fluid, the average value of the overall heat transfer coefficient for the flow rates investigated in the experiments was determined to be 3385.95 W/m^{2}K for concurrent flow. From the value of 6.2 lpm of the hot working fluid's flowrate, it has been observed that the value of U rises above the mean value of the overall heat transfer coefficient and

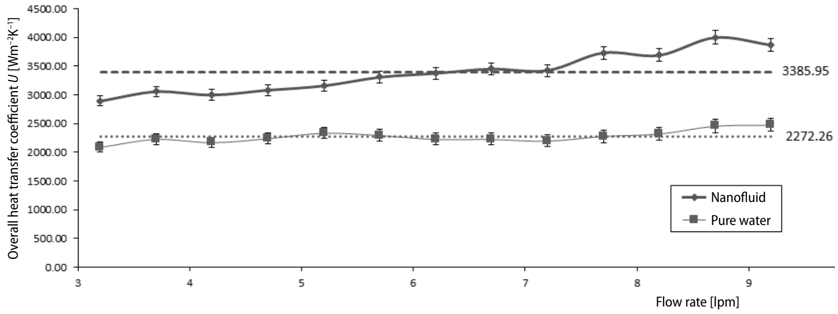

Figure 2. The graph of the distribution of overall heat transfer coefficients for concurrent flow; the error bars represent the standard deviation of repeated experiments (n = 3).

the value of U is always above the mean value of the overall heat transfer coefficient from this flowrate for nanohBN-water nanofluid. The overall heat transfer coefficient of nanohBN-water nanofluid for the concurrent flow conditions reached the highest value and it was calculated to be 3997.21 W/m^{2}K at a flow rate of 8.7 lpm. When pure water was used as hot fluid for the same flow rates, the mean value of overall heat transfer coefficient U was determined to be 2272.26 W/m^{2}K for concurrent flow. The % improvement observed in the overall heat transfer coefficient for each flowrate for concurrent flow by using nano hBN/water nanofluid as hot working fluid is given in fig. 3.

In the graph given in fig. 3, the U value of the nanohBN-water nanofluid obtained at each flow rate of the hot working fluid in experiments was compared with the U of the pure water for concurrent flow conditions. It was found that an average of 48.78% improvement was observed in the concurrent flow conditions by using nanohBN-water nanofluid instead of pure water as hot working fluid. It has been observed that from the hot fluid-flowrate of 6 lpm, the percentige of improvement value was above the average percentige of improvement value of 48.78%. In this comparison, the percentige of improvement observed in the U reached the highest value of 63.76% at 7.7 lpm flowrate of the hot working fluid.

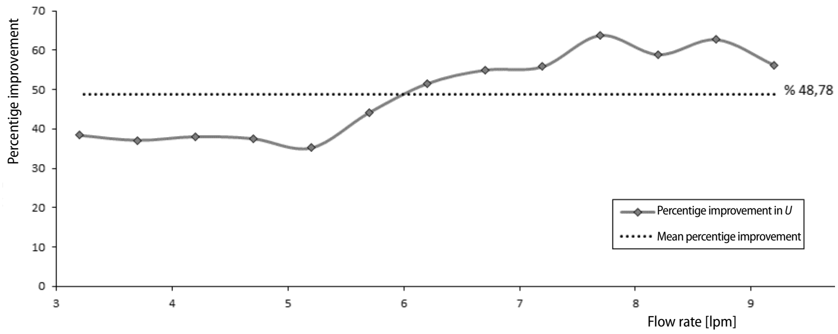

Figure 3. The graph of the distribution of percentige improvement observed in the overall heat transfer coefficient for each flowrate for concurrent flow

The graph showing the variation of hot fluid's heat convection coefficients as a result of the experimental measurements and the calculations made by using these measurements are given in fig. 4. When the graph in fig. 4 is examined, it can be seen that the hot fluid's heat convection coefficients of nanohBN-water nanofluid is higher than those of pure water's at all flow-

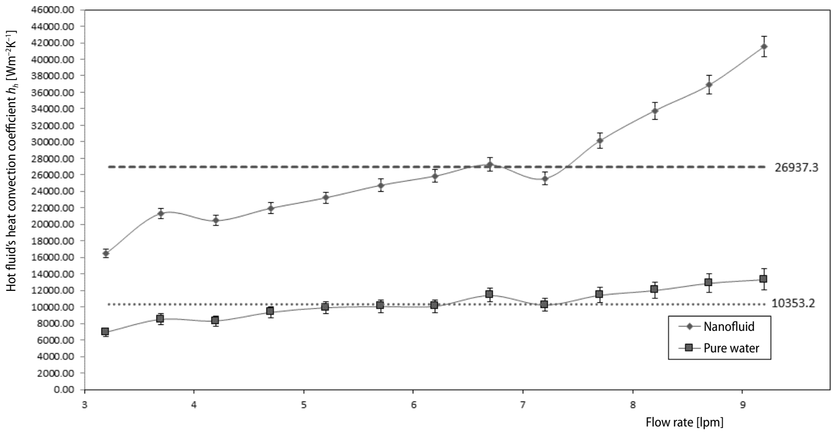

Figure 4. The graph of the distribution of the hot fluid's heat convection coefficients for concurrent flow; the error bars represent the standard deviation of repeated experiments (*n* = 3)

rates examined in the experiments for concurrent flow. When the nanohBN-water nanofluid is used as the hot working fluid, the average value of the hot fluid's heat convection coefficient h_h for the flow rates investigated in the experiment was determined to be 26937.3 W/m^{2}K for concurrent flow. From the value of 6.5 lpm of the hot working fluid's flowrate, it has been observed that the value of h_h rises above the mean value of the hot fluid's heat convection coefficient for nanohBN-water nanofluid. The hot fluid's heat convection coefficient of nanohBN-water nanofluid for the concurrent flow conditions reached the highest value and was calculated to be 41540.4 W/m^{2}K at a flow rate of 9.2 lpm. When pure water was used as hot working fluid for the same flow rates, the mean value of the hot fluid's heat convection coefficient h_h was determined to be 10353.2 W/m^{2}K for concurrent flow.

Results and discussion for the counter flow heat exchanger

The experimental set-up was prepared before starting the experiments for counter flow condition. For obtaining the values and characterizing the changes of *U* and the hot fluid's heat convection coefficients for counter flow condition, the experiments were conducted at a constant cold fluid-flow rate of 0.01 kg/s and varying hot fluid-flow rates of 3.2 lpm, 3.7 lpm, 4.2 lpm, 4.7 lpm, 5.2 lpm, 5.7 lpm, 6.2 lpm, 6.7 lpm, 7.2 lpm, 7.7 lpm, 8.2 lpm, 8.7 lpm, and 9.2 lpm. The graph showing the variation of the *U* as a result of the measurements obtained from the experiments performed and the calculations made by using these measurements are given in fig. 5.

When the graph in fig. 5 is examined, it can be seen that it has a different character than the graph given in fig. 2. The *U* of nanohBN-water nanofluid is not higher than those of pure water's at all flowrates examined in the experiments for counter flow. It was observed that the average value of the calculated *U* values of the nanofluid was slightly higher than the average value of the *U* values of pure water for counter flow. When the nanohBN-water nanofluid is used as the hot working fluid, the average value of the *U* for the flow rates investigated in the experiments was determined to be 3081.76 W/m^{2}K for counter flow. From the value of 6.2 lpm of the hot working fluid's flowrate, it has been observed that the value of *U* rises above the mean value of the *U* and the value of *U* is always above the mean value of the *U* from this flowrate for nanohBN-water nanofluid. The *U* of nanohBN-water nanofluid for the counter flow

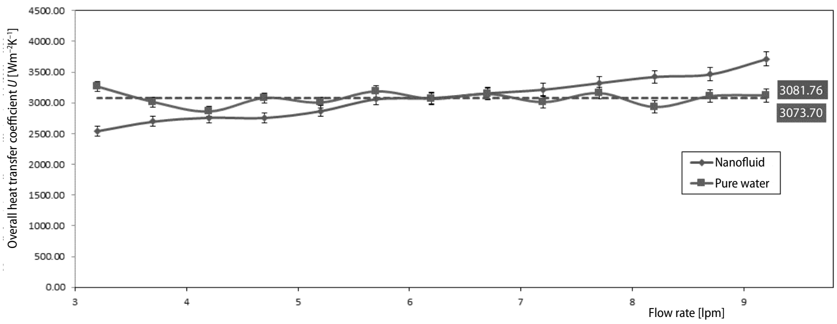

Figure 5. The graph of the distribution of overall heat transfer coefficients for counter flow; the error bars represent the standard deviation of repeated experiments (n = 3)

conditions reached the highest value and it was calculated to be 3713.66 W/m^{2}K at a flow rate of 9.2 lpm. When pure water was used as hot fluid for the same flow rates, the mean value of U was determined to be 3073.7 W/m^{2}K for counter flow. The percentige of improvement observed in the U for each flow rate for counter flow by using nanohBN-water nanofluid as hot working fluid is given in fig. 6.

In the graph given in fig. 6, the U value of the nano hBN/water nanofluid obtained at each flow rate of the hot working fluid in experiments was compared with the U of the pure water for counter flow conditions. It was found that an average of 0.36% improvement was observed in the counter flow conditions by using nano hBN/water nanofluid instead of pure water as hot working fluid. It has been observed that from the hot fluid-flow rate of 6.2 lpm, the % improvement value was above the average % improvement value of 0.36%. In this comparison, the % improvement observed in the U reached the highest value of 19.02% at 9.2 lpm flowrate of the hot working fluid for counter flow.

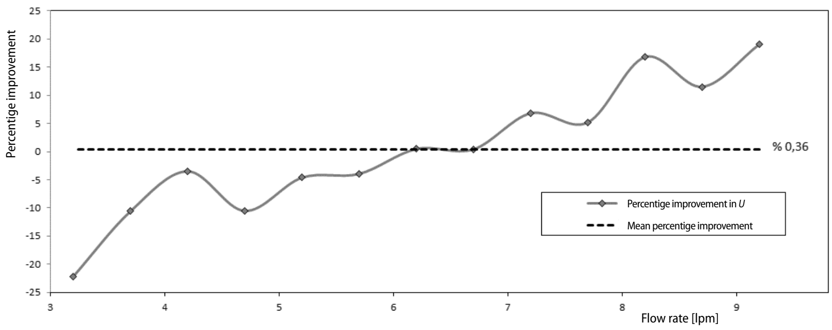

Figure 6. The graph of the distribution of percentige improvement observed in the overall heat transfer coefficient for each flowrate for counter flow

The graph showing the variation of h_h as a result of the measurements obtained from the experiments performed and the calculations made by using these measurements are given in fig. 7 for counter flow. It can be seen that the h_h of nanohBN-water nanofluid is higher than those of pure water's at all flowrates examined in the experiments for counter flow. When the nanohBN-water nanofluid is used as the hot working fluid, the average value of the h_h for the flow rates investigated in the experiment was determined to be 52700.1 W/m^{2}K for counter flow. From the value of 5.7 lpm of the hot working fluid's flowrate, it has been observed that the value of h_h rises above the mean value of the h_h for nanohBN-water nanofluid. The h_h of nanohBN-water nanofluid for counter flow conditions reached the highest value and was calculated to be 78657.84 W/m^{2}K at a flow rate of 9.2 lpm. When pure water was used as hot working fluid for the same flow rates, the mean value of the h_h was determined to be 17578.44 W/m^{2}K for counter flow.

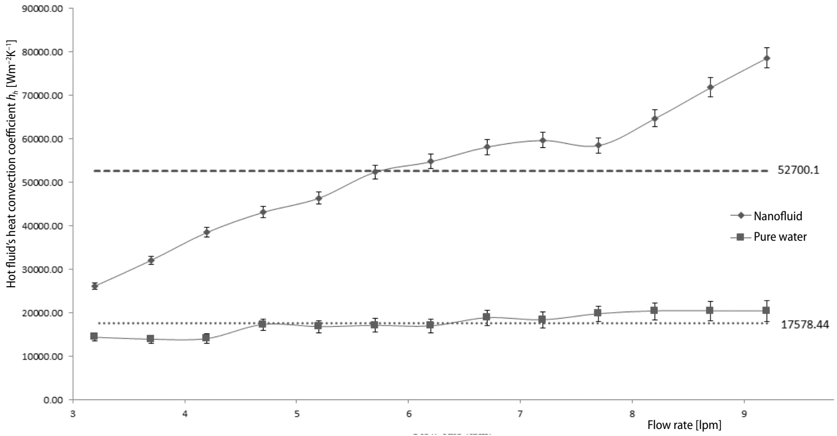

Figure 7. The graph of the distribution of the hot fluid's heat convection coefficients for counter flow; the error bars represent the standard deviation of repeated experiments (*n* = 3)

Conclusion

The heat transfer performance of the prepared nanohBN nanofluid flowing in the prepared coaxial double tube experimental set-up was characterized for concurrent flow and counter flow conditions in the present experimental study. The performance evaluation on the usage of the prepared nanohBN-water nanofluid according to pure water as hot working fluid was made with respect to the *U*. By using nanohBN-water nanofluid instead of pure water as hot working fluid, the average values of percentige improvement observed in the values of the *U* for concurrent flow and counter flow conditions were 48.78% and 0.36%, respectively. Cengel [24] stated that for counter flow conditions, the cold and hot fluids enter the heat exchanger from opposite ends and the outlet temperature of the cold fluid may exceed the outlet temperature of the hot fluid. The cold fluid can be heated to the inlet temperature of the hot fluid in the limiting case. The log mean temperature difference for a counter flow heat exchanger is always greater than that for a concurrent flow heat exchanger for specified inlet and outlet temperatures. A smaller surface area (and thus a smaller heat exchanger) is needed to achieve a specified heat transfer rate in a counter flow heat exchanger. In the present study the same experimental set-up was used for concurrent flow and counter flow conditions and the heat transfer area was the same. It is reasonable to expect a greater improvement effect for a situation where the heat transfer is not near the limiting case. Therefore, it is reasonable to observe such a difference in the improvement of the average total heat transfer coefficients for the cases of concurrent flow (48.78%) and counter current flow (0.36%) conditions by the usage of nanofluid. This study shows promising results and validate the potential of application of nanohBN-water nanofluid in heat management applications, which is a critical factor in efficiency determination of various industrial and domestic systems.

Nomenclature

A – area, [m^2]
C_p – specific heat capacity, [kJkg^{-1}K^{-1}]
h – heat convection coefficient, [Wm^{-2}K^{-1}]
m – mass, [kg]
$\dot{m}$ – mass flow rate, [kgs^{-1}]
Q – heat, [kJ]
$\dot{Q}$ – heat transfer rate, [kW]
R – the magnitude to be measured
T – temperature, [K]
ΔT – temperature difference, [K]
$\Delta T_{\mathrm{l,m}}$ – mean logarithmic temperature difference, [K]
U – overall heat transfer coefficient, [Wm^{-2}K^{-1}]
V – volume, [m^3]
x – independent variables
w – the error rate of each independent variable
W_R – the error rate of R magnitude

Greek symbol

ρ – density, [kgm^{-3}]

Subscripts

c – cold working fluid
h – hot working fluid
m – mean

Acronym

hBN – hexagonal boron nitride

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