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Experimental Analysis of Heat Transfer and Friction Factor in Plate Heat Exchanger with Different Orientations Using Al_2O_3 Nanofluids

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ABSTRACT

Experimental investigations have been done to find out the heat transfer characteristics and friction factor of water based Al_2O_3 nanofluids as a coolant in brazed plate heat exchangers. In most of the studies plate heat exchangers are used in horizontal or vertical conditions. The base plate of the plate heat exchanger was kept inclined at (0^o, 30^o, 60^o, 90^o). The experimentation has been carried out with two different concentration of the nanofluids (0.1 and 0.2 v/v%). It was observed that the heat transfer characteristics improves with an increase in Reynolds number. It has been shown that nanofluids in a plate heat exchanger have a maximum of 34% heat transfer rate over the base fluid. It has been observed that from horizontal to vertical orientation heat transfer rate decreases with increase in Reynolds number. The average heat transfer coefficient has been found to reduce by 10-15% when the angle of inclination of base plate of heat exchanger i from horizontal is 30^o.

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NOMENCLATURE

A	Cross-sectional area, (m^2)	ϕ	Volume fraction of nanoparticles
C_p	Specific heat capacity, (J/kg-k)	θ	Enlargement factor
D_H	Hydraulic diameter	β	Chevron angle.
f	Friction factor	**Subscripts**	
g	Gravity(m/s^2)	c	Cold water
h	Heat transfer coefficient	e	Equivalent
k	Thermal conductivity, (W/m-k)	h	Hot water
L_w	Plate width inside gasket, (mm)	H	Hydraulic diameter
m	Mass flow rate, (Kg/s)	i	Inlet conditions
N_{cp}	Number of channels per pass	LMTD	Logarithm mean temperature difference
Nu	Nusselt number	o	Outlet conditions
Q	Heat transfer rate, (W)	nf	Nanofluids
Re	Reynolds number	np	Nanoparticle
T	Temperature, (^{0}C)	bf	Base fluid
T_b	Bulk mean temperature, (^{0}C)	b	Bulk mean
T_w	Wall temperaure, (^{0}C)	w	Wall
U	Overall heat transfer cofficient, (W/m^2-k)	cp	Channels per pass
Greek Symbols		avg	Average
ρ	Density (Kg/m^3)		
μ	Viscosity (Ns/m^2)		

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1. INTRODUCTION

A heat exchanger is a device which is used to exchange the thermal energy between two or more fluids. In some type of heat exchangers the fluids are in direct contact, while in other types heat transfer between fluids takes place through a separating walls. Direct contact type heat exchangers are also referred to as a direct transfer type, or simply recuperators, whereas indirect type heat exchangers referred to as indirect transfer type or simply generators. The main purpose of using the heat exchanger is to reuse the waste energy for different purposes [1]. Plate heat exchanger is an indirect contact type heat exchanger. These type of heat exchangers are usually built of thin plates. The plates are either smooth or have some form of corrugation. Plate heat exchangers are widely used in dairy and food industries and in chemical plants. The contemporary use of plate heat exchanger is not limited to liquid to liquid duty, but it has been used for condensing and evaporating applications. Heat transfer in plate heat exchanger highly depends on the geometrical properties of plates and type of orientations in which the plate heat exchanger is placed (horizontal or vertical) and specifically on the physical properties of the working fluid.

In order to take care of growing demand of energy, heat transfer rate has to be increased, and this can be attained by using fluid having good thermophysical properties. Choi et al. [2] found out the nanofluids which are made by nanometer sized solid particles (0-100nm). They found out that for heat transfer enhancement the thermal conductivity of nanofluids plays a vital role. After that, many researchers did extensive work on nanofluids technology, its applications and its acceptability in the domain of heat transfer device. The thermal conductivity of nanofluids has been studied by many authors, whereby it was observed that the temperature and particle size plays an important role on the thermal conductivity of nanofluids. It has been observed that the thermal conductivity increases with increase in the particle volume concentration, particle size and temperature [3-8]. Sheng- Qi Zhou et al. [9] presented an experimental investigation of specific heat capacity of water based Al_2O_3 nanofluids. The author predicted that the specific heat of nanofluids decreases with increase in nanoparticle volume fraction. Hong et al. [10] studied experimentally the effect of clustering of nanoparticles. Ethylene glycol has been used as the base fluid for preparing the Fe_2O_3 nanofluids. The authors observed that the time of sonication increases the thermal conductivity enhancement. Effect of concentration of surfactant on viscosity and thermal conductivity of Al_2O_3 nanofluids has been studied by Lotfizadeh Dehkordi et al. [11]. The authors observed that the lower concentration of surfactant enhances the thermal conductivity and provides the better dispersion, but the higher concentration of surfactant resulted in the drop in the thermal conductivity of nanofluids and accelerates the viscosity increase of nanofluid. M. Tajik et al. [12] experimentally measured the thermal conductivity of water based Cu and Al nanofluids. The authors found out that thermal conductivity of nanofluids is higher than the base fluid and the thermal conductivity of Cu/water nanofluids is higher than that of Al/water nanofluids. MA Mehrabian et al. [13] experimentally investigated the overall heat transfer characteristics of double pipe heat exchanger and compared it with the prediction of standard co-relations. The authors concluded that in counter-flow arrangement, inner side heat transfer coefficient is almost 1.5 times larger than that of the outer side. And in the case of counter-flow arrangement, the results are in good agreement with the prediction. Thierry Mare et al. [14] experimentally compared the thermal performance of two nanofluids at low temperature in the plate heat exchangers. The two type of commercial nanofluids used are alumina (Al_2O_3) dispersed in water and the other an aqueous suspension of carbon nanotubes (CNTs). Their results show that alumina and carbon nanotubes show a better thermal hydraulic performance in terms of heat transfer enhancements in comparison with the pure water. Arun Kumar Tiwari et al. [15] experimentally investigated the heat transfer and pressure drop characteristics of nanofluids in a plate heat exchanger. The experiment has been done on the wide range of concentration and for various flow rates. It has been observed that the convective heat transfer coefficient increases with increase in nanoparticle volume concentration, volume flowrate and the decrease in nanofluids temperature.

Fatih Akturk et al. [16] experimentally investigated the performance analysis of gasket plate heat exchangers. The authors used different plate heat exchangers consisting of different number of plates (10, 15, and 21). New correlations for friction factor and Nusselt number were developed. Both the correlation for Nusselt number as well as for friction factor was found out as the function of Reynolds number. Yueh-Hung Lin et al. [17] experimentally examined the effect of flow direction for evaporation of heat transfer in plate heat exchanger. R-410A has been used as refrigerant (working fluid) in a plate heat exchanger. The author revealed that the superheated region for counter flow is much less as compared with the parallel flow. The authors also revealed that there is no effect of Reynold's number on it i.e. flow rate has a negligible effect on the superheated region. Shive Dyal Pandey et al. [18] experimentally investigated the heat transfer and friction factor of nanofluids as a coolant in corrugated plate heat exchangers; and the nanofluids so used, highest heat transfer rate has been observed for the nanofluids with 2 v/v % particle concentration. It has been found that the maximum enhancement of convective heat transfer coefficient is higher for 2 v/v % Al_2O_3/water nanofluids than that of water by 11%. Pirhayati et al. [19] carried out their study on the convective heat transfer of oil based CuO nanofluids in a circular tube. Nanofluids with different weight fraction has been used in this research. Tube inclination was considered to be one of the most important parameters. The authors concluded that with increase in nanofluids weight fraction, heat transfer co-efficient increases. It has also been concluded that heat transfer co-efficient increases with the inclination. Ajay and Kundan et al. [20] presented their work on the performance of (Al_2O_3/H_2O-$C_2H_6O_2$) nanofluids in a solar plate collector using both experimental and CFD techniques. Results show that with increase in volume flow rate of the working fluid, the overall efficiency of solar plate collector increases.

The main purpose of the present study is to find out the heat transfer and pressure drop characteristics of Al_2O_3/water nanofluids in the brazed plate heat exchanger with different orientations. Till now, plate heat exchangers are used only horizontally or vertically. No other orientation has not been used for finding the heat transfer characteristics of plate heat exchangers. The plate heat exchanger has made inclined with the help of master circle. Prior to the experiments, the thermophysical properties (thermal conductivity, viscosity, specific heat) have been measured. Two different concentrations of nanofluids have been used (0.1 and 0.2v/v %) for the experimentations. The results have been obtained and compared with the different angle of inclinations.

2. NANOFLUIDS PREPRATION

In the present research the experiments were carried out with Al_2O_3-water based nanofluids. The Al_2O_3 nanoparticles of average size below 20nm (purity 99.9%) were purchased from the Nanoshells, Derra Bassi, India. The SEM, TEM, EDX and XRD images of Al_2O_3 nanoparticles are shown in Figures 1, 2, 3 and 4 respectively.

In the present study nanofluids were prepared by Two-step method. The amount of nanoparticles for the required volume concentration of nanofluids was calculated by Equation (1):

$$\phi = \frac{\frac{m_{np}}{\rho_{np}}}{\frac{m_{np}}{\rho_{np}}+\frac{m_{bf}}{\rho_{bf}}} \tag{1}$$

Table 1 Shows the thermophysical properties of Aluminum nanoparticles and base fluid (water):

Analytical balance of least count of 1mg has been used to measure the amount of nanoparticles. To prepare the nanofluids, the required amount of nanoparticles were mixed with distilled water using a magnetic stirrer.

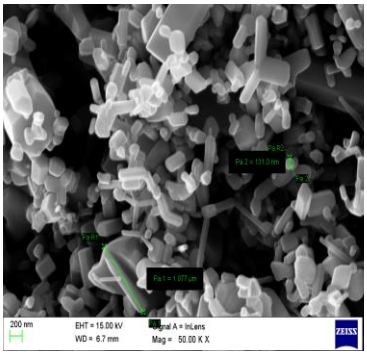

Figure 1. SEM photograph of nanoparticles

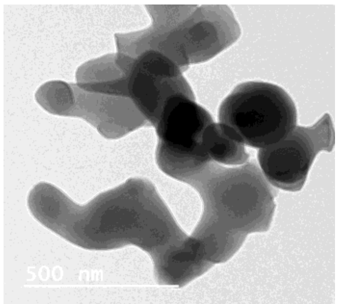

Figure 2. TEM photograph of Al_2O_3 nanoparticles

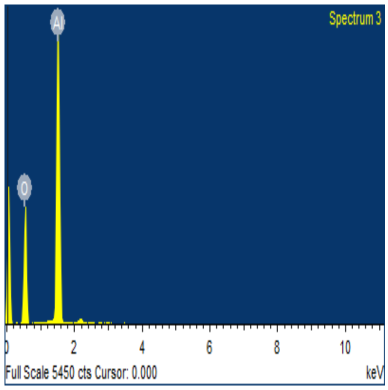

Figure 3. EDX photograph of nanoparticles

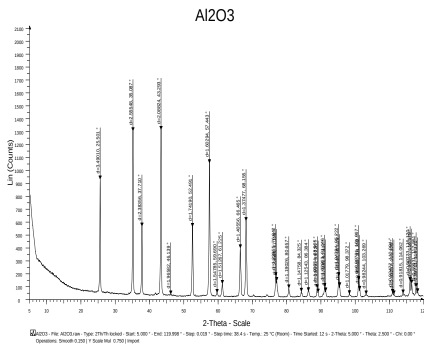

Figure 4. XRD photograph of Al_2O_3 nanoparticles

TABLE 1. Physical properties of Al_2O_3 nanoparticles and base fluid

Physical properties	Fluid phase (water)	Al_2O_3
C_p (J/Kg-k)	4179	765
ρ (Kg/m^3)	997.1	3970
k (W/m-k)	0.613	25
D_p (nm)	0.384	47

To ensure complete dispersion of the nanoparticles, nanofluids were sonicated for 2-3 hours with ultrasonic sonicator. Nanofluids were prepared without any use of surface agent; and the nanofluids so prepared were used in performing the experiments.

3. EXPERIMENTAL SETUP

An experimental setup has been designed to investigate the friction factor and heat transfer characteristics of Al_2O_3/water nanofluids in plate heat exchangers at different orientations. The schematic diagram and photograph in Figures 5 and 6 give the exact view of experimental setup, respectively. The experimental setup consists of brazed plate heat exchanger containing 21 plates. The geometric details of plate heat exchanger are provided in Table 2. The experimental setup also consists of two flowmeters to control the flowrate of water and nanofluids on both sides, two water tanks, a PID controller, two centrifugal pumps and four k-type thermocouples. Hot water stored in one tank which is allowed to pass through the plate heat exchanger from one side with the help of a centrifugal pump. A 2kW heater was used to heat the water. Flow rate of the hot side was varied from (0.5-3) lpm with the help of flowmeter. The experimentation was done on a wide range of hot water temperatures (30°C, 40°C, 50°C, and 60°C). The nanofluids were stored in the other tank. The nanofluids were circulated to the plate heat exchanger with the help of a centrifugal pump. The flow rate of the nanofluids was maintained constant at 2 lpm. Differential pressure manometers were used to find out the pressure drop between inlet and outlet ports of the plate heat exchanger. K- type thermocouples located at the inlet and outlet ports of the plate heat exchanger were used to measure the temperature. Master circle has been used for the inclination of the plate heat exchanger. There is an uncertainty of ±0.2°C in the measurements.

4. DATA PROCESSING

Experimental data was used to calculate the heat transfer characteristics of plate heat exchanger. Experimentation was conducted with 0.1 and 0.2% Al_2O_3 nanoparticles. Basic heat transfer equations were used for this purpose.

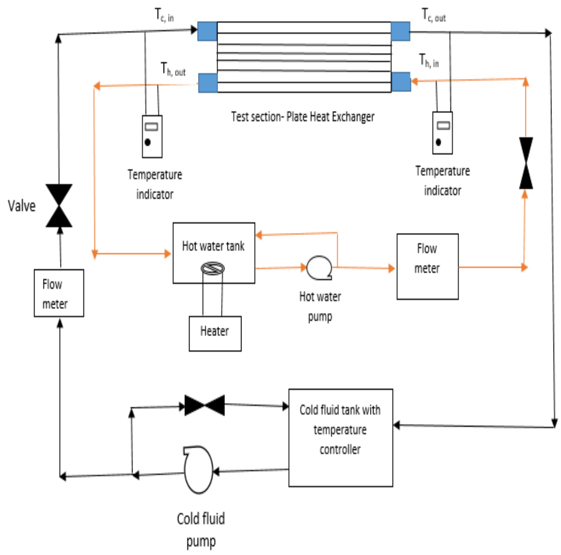

Figure 5. Schematic diagram of Experimental Setup

Figure 6. Image of the Experimental setup

TABLE 2. Geometrical parameters for tested section

Plate width inside gasket , L_w (mm)	100
Number of plates	20
Heat exchanger area, A (m^2)	0.5
Mean channel spacing, b (mm)	2.3
Chevron angle,	43^0
Gasket thickness, (mm)	0.35
Gasket width, (mm)	7.1
Gap between two plates	2.1
Corrugation pitch, P_C (mm)	14.6
Plate thickness (mm)	0.5
Vertical distance between center of ports, (mm)	340
Horizontal distance between center of ports, (mm)	62

Reynolds number of the fluid was calculated as follows:

$$Re_h = \frac{G_h D_H}{\mu_h} \tag{2}$$

where the hydraulic diameter (D_H) and channel mass velocity (G_h) are as follows:

$$G_h = \frac{m_h}{N_{cp} b L_W} \tag{3}$$

where, the hydraulic diameter calculated as:

$$D_H = \frac{2b}{\theta} \tag{4a}$$

where, Θ is defined at the enlargement factor and is calculated with the help following relation:

$$\theta = \frac{1}{6}\left[1 + \left\{1 + \left(\frac{\pi}{2cos\beta}\right)^2 \gamma^2\right\}^{0.5} + 4\left\{1 + \left(\frac{\pi}{2\sqrt{2}cos\beta}\right)^2 \gamma^2\right\}^{0.5}\right] \tag{4b}$$

where, γ is the function of mean channel spacing and corrugation pitch of the plate heat exchanger.

The heat gained by the nanofluids and heat loss by the hot fluid can be calculated with the help of Equations (5a) and (5b):

$$Q_h = m_h C_{ph}(T_{h,i} - T_{h,o}) \tag{5a}$$

$$Q_c = m_c C_{pc}(T_{C,O} - T_{C,i}) \tag{5b}$$

where, Q is defines as the rate of heat transfer.

The overall heat transfer coefficient of hot fluid is calculated by the following equation:

$$U = \frac{Q_{avg}}{A\Delta T_{LM}} \tag{6}$$

The average heat transfer rate is defined as follows:

$$Q_{avg} = \frac{Q_h + Q_{nf}}{2} \tag{7}$$

where,

$$\Delta T_{LM} = \frac{(T_{h,i} - T_{c,o}) - (T_{h,o} - T_{c.i})}{LN(\frac{T_{h,i} - T_{c,o}}{T_{h,o} - T_{c,i}})} \tag{8}$$

The heat transfer coefficient of nanofluids was evaluated with the help of wall temperature of plate heat exchanger, bulk mean temperature of nanofluids and average heat transfer rate of plate heat exchanger as follows:

$$h = \frac{Q_{avg}}{A(T_b - T_w)} \tag{9a}$$

The wall temperature of the plate heat exchanger can be calculated with the relationship given below:

$$T_w = \left[\frac{T_{c,i} + T_{C,O} + T_{h,i} + T_{h,o}}{4}\right] \tag{9b}$$

where, the Nusselt number is defined as follows:

$$Nu = \frac{hD_H}{k} \tag{10}$$

The friction factor can be calculated with the following relation:

$$f = \frac{\Delta P}{\left[\left(\frac{L_{eff}}{D_H}\right)\left(\frac{2G^2}{\rho}\right)\right]} \tag{11}$$

In the above relation, L_{eff} indicates the effective length which is equal to L_v -which is the vertical distance between centers of ports whose value is provided in Table 2- and ΔP is the pressure drop.

5. RESULTS

The experimentations were carried out at three different runs. Firstly, experimentation were carried out with the water which is base fluid. While in the second and third run nanofluids with 0.1% and 0.2% concentration were used as the working fluid. The experimentation were carried out with the specific range of Reynolds number. (100<Re<2000). The value of the cold side flow was maintained as constant (2 lpm). The experimentation was performed at temperatures ranging from (30°C - 60°C).

Figure 7 shows the variation of overall heat transfer coefficient of water and nanofluids with Reynolds number ranging from (100-2000). Reynolds number is the function of flow rate. So, at constant inlet temperature, with increase in flow rate, the overall heat transfer coefficient increases. Equation (6) was used to calculate the overall heat transfer coefficient of water and nanofluids. It has been observed that the overall heat transfer coefficient purely depends on the thermophysical properties of the working fluid. Figure 8 shows that the overall heat transfer is highest for nanofluids at concentration 0.2 v/v% and lowest for water. The total enhancement of approximately 29% has been found out in the case of nanofluids. The same enhancement has been found out at different flow rates. The heat transfer enhancement may be affected by various mechanisms such as thermal conductivity of nanoparticles and the collision of nanoparticles with the water molecules which causes transmission of energy, thus improving heat exchange. Saxena et al. [21] carried out their research on the water based Aluminum nanofluids and got the same results for nanofluids compared with water, i.e., heat transfer coefficient increases with increase in the nanoparticle concentration.

Figure 8 shows the effect of inclination of the base of the plate heat exchanger on overall heat transfer coefficient.

On the bases of experimental results, it has been noticed that heat transfer coefficient decreases from horizontal position (0º) to the vertical position (90º). Highest heat transfer characteristics has been noticed when the plate heat exchanger placed horizontally. However, when the angle of inclination increases the overall heat transfer coefficient decreases.

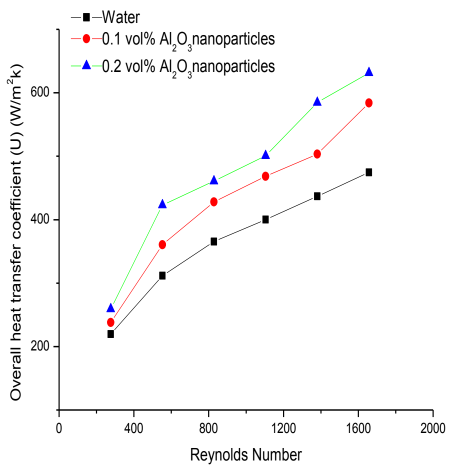

Figure 7. Effect of Reynold number on overall heat transfer coefficient at 50°C of operating temperature

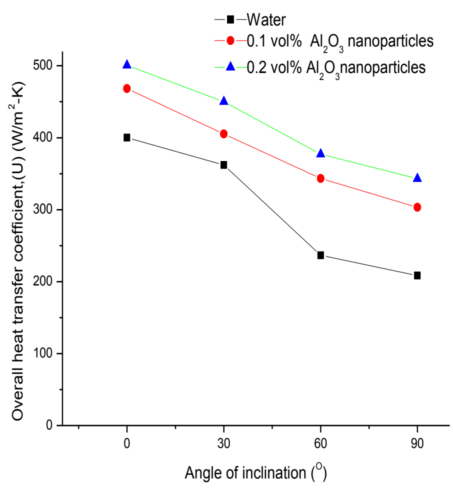

Figure 8. Effect of inclination on overall heat transfer coefficient at 50°C of operating temperature with flow rate of 2 lpm on both sides

Approximately (12-15) % decrement has been found out with increases in inclination of base of plate heat exchanger in order of 0°>30°>60°>90°. Therefore, it is clear that the plate heat exchanger gives maximum efficiency when used horizontally.

Figure 9 shows the effect of Reynolds number on heat transfer coefficient. Heat transfer coefficient basically depends upon the difference between the bulk mean temperature of nanofluids and wall temperature of the plate heat exchanger. Equation (9) has been used to calculate the heat transfer coefficient of the plate heat exchanger. Figure 9 shows that the heat transfer coefficient increases with increase in Reynolds number and it is maximum for nanofluids at concentration 0.2 v/v%. The average enhancement of approximately 28% has been found out in case of nanofluids.

Figure 10 shows the effect of inclination on heat transfer coefficient at operating temperature of 50ºC with 2 lpm flow rates on both sides. Figure 10 shows that the heat transfer coefficient increases from vertical to horizontal position. From horizontal to inclination of 30º, approximately (12-15) % of diminution has been found out and this diminution is also same for the other flow rates. Heat transfer coefficient was observed to be higher in case of horizontal heat exchanger as compared to other positions. In plate heat exchangers hot fluid and cold fluid streams either in parallel flow or counter-flow configuration and they are thermally contacted with each other. In horizontal position, both streams remain thermally in contact with each other for more time before the fluid streams approach the end part of the plate. But according to authors, in other positions, i.e, 30°, 60°, and 90° fluid streams are thermally in contact for less time as compared to horizontal position due to the gravitational force.

The variation of Nusselt number with Reynolds number is shown in Figure 11. It is clear from the figure that with increase in Reynolds number, the Nusselt number increases. It attains its maximum value at 0.2 v/v% concentration of nanofluids. Average enhancement of approximately 30% occurs in case of nanofluids. Akturk et al. [16] did their experiment with the water as a working fluids and got the same results when there are 21 plates in plate heat exchanger.

The effect of inclination on Nusselt number is shown in Figure 12. Nusselt number basically depends on the thermal conductivity of the working fluid and the heat transfer coefficient. Equation (10) has been used to calculate the Nusselt number. Figure 12 clearly shows the decrease in Nusselt number with respect to the increment in inclination.

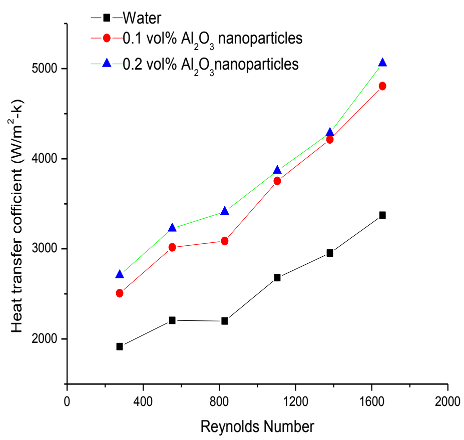

Figure 9. Effect of Reynolds number on heat transfer coefficient at 50°C of operating temperature fluid

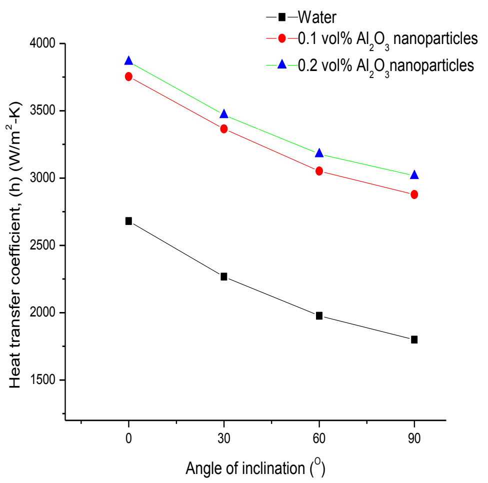

Figure 10. Effect of inclination on heat transfer coefficient at 50⁰C of operating temperature with flow rate of 2 lpm on both sides

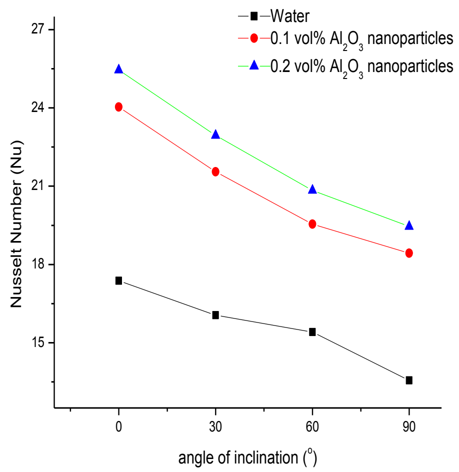

Figure 12. Effect of inclination on Nusselt number at 50⁰C of operating temperature with flow rate of 2 lpm on both sides

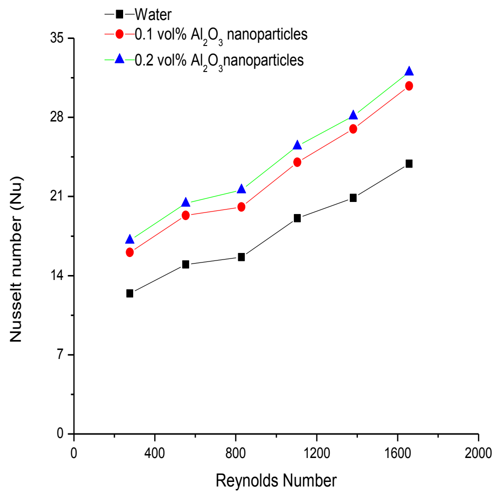

Figure 11. Effect of Reynolds number on Nusselt number at 50⁰C of operating temperature

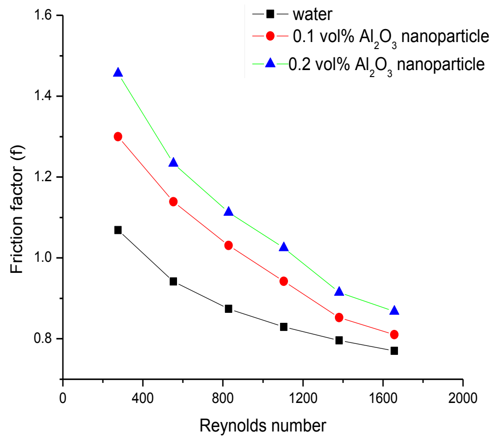

Figure 13. Effect of Reynolds number on friction factor at 50⁰C of operating temperature

The average decrement of approximately (10-15) % has been noticed.

Figure 13 shows the effect of Reynolds number on friction factor at operating temperature of 50⁰C. The flow rate has been varied from (0.5 – 3) lpm. Equation (11) has been used to calculate the friction factor for the working fluids in a plate heat exchanger.

From Figure 13 it is clear that friction factor decreases with increase in the Reynolds number and increases with increase in the nanoparticle concentration. The reason behind this is that friction factor strongly depends on the density and slightly upon the viscosity of the working fluid.

Tiwari et al. [15] carried out their research on the CeO_2/water nanofluids and got the same trends of graphs at diverse concentration of nanofluids.

6. CONCLUSION

In present experimental study heat transfer performance of plate heat exchanger with Al_2O_3/water nanofluids has been studied. The experimentation has been done with the two different concentrations of the nanofluids (0.1 and 0.2 v/v%) with wide range of flow rates (0.5, 1, 1.5, 2, 2.5,3.0) lpm at diverse operating temperatures. At 0.2 volume percent concentration, Al_2O_3/water nanofluids are more effective. At 0.2 volume percent concentration of nanofluids highest heat transfer characteristics has been obtained. For Al_2O_3/water nanofluids, a maximum heat transfer enhancement of approximately 30% has been observed. The results revealed that an increment of heat transfer rate occurs from vertical to horizontal position, which shows that the plate heat exchanger gives the maximum efficiency when used horizontally. Average decrement of (10-15)% has been found out when the plate heat exchanger is inclined from 0^o to 30^o.

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Experimental Analysis of Heat Transfer and Friction Factor in Plate Heat Exchanger with Different Orientations Using Al_2O_3 Nanofluids

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چکیده

در این پژوهش تحقیقات تجربی برای پیدا کردن ویژگی های انتقال حرارت و ضریب اصطکاک نانوسیال Al_2O_3 پایه‌ی آب به عنوان خنک کننده در مبادله‌کن‌های حرارتی صفحه‌ای لحیم سخت (زردجوش‌کاری) شده انجام شده است. در بسیاری از مطالعات پیشین مبادله‌کن‌های حرارتی صفحه‌ای، مبادله‌کن در شرایط افقی یا عمودی استفاده شده است. در این پژوهش پایه مبادله‌کن شیب دار در زاویه‌های صفر، ۳۰، ٦٠ و ۹۰ درجه نگه داشته شد. آزمایش با دو غلظت نانوسیال (۰/۱ و ۰/۲ درصد حجمی انجام شده است. مشاهده شد که افزایش عدد رینولدز ویژگی‌های انتقال حرارت را بهبود می‌بخشد. نشان داده شده است که در یک مبادله‌کن حرارتی صفحه‌ای حداکثر نرخ انتقال حرارت با نانوسیال ۳٤٪ بیش از آن با سیال پایه است. همچنین، مشاهده شده است که با تغییر زاویه‌ی استقرار از افقی به عمودی، نرخ انتقال حرارت با افزایش عدد رینولدز کاهش می‌یابد. معلوم شده است که ضریب انتقال حرارت میانگین زمانی که زاویه شیب صفحه پایه مبادله‌کن حرارتی ۳۰ درجه نسبت به افق باشد به میزان ۱۰ تا ۱۵٪ کاهش می‌یابد.

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