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Convective Heat Transfer of Oil based Nanofluid Flow inside a Circular Tube

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ABSTRACT

An experimental investigation was carried out to study convective heat transfer of nanofluid flow inside an inclined copper tube under uniform heat flux condition. Required data are acquired for laminar and hydrodynamically fully developed flow inside a round tube. The stable CuO-base oil nanofluid with different nanoparticle weight fractions of 0.5%, 1% and 2% was produced by means of ultrasonic device in two steps method. In this study, the effect of different parameters such as tube inclination, nanofluid weight fraction and Reynolds number on heat transfer coefficient was considered. Results show that the heat transfer coefficient of nanofluid with different weight fractions increases with the increasing Reynolds number inside horizontal and inclined round tubes. Also, nanofluid flow inside the inclined tube at 30 degrees exhibit the most heat transfer enhancement amongst other tube inclinations at same Reynolds number.

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1. INTRODUCTION

Thermal load removal is a great concern in many industries including power plants, chemical processes and electronics. In order to meet the ever increasing need for cooling the high heat flux surfaces, different enhanced heat transfer techniques have been suggested. Most of these methods are based on structure variation, vibration of heated surface, injection or suction of fluid and applying electrical or magnetic fields which are well documented in literature [1, 2]. However, applying these enhanced heat transfer techniques are no longer feasible for cooling requirement of future generation of microelectronic systems, since they would result in undesirable cooling system size and low efficiency of heat exchangers. To obviate this problem, nanofluids with enhanced thermo-fluidic properties have been proposed since the past decade. Nanofluid is a uniform dispersion of nanometer-sized particles inside a liquid which was first pioneered by Choi [3].

Excellent characteristics of nanofluids such as enhanced thermal conductivity, long time stability and little penalty in pressure drop increasing and tube wall abrasion have motivated many researchers to study the thermal and flow behavior of nanofluids. These studies were mainly focused on effective thermal conductivity, phase change behavior, tribological properties, flow and convective heat transfer of nanofluids.

A wide range of experimental and theoretical studies has been performed on the effect of different parameters such as particle fraction, particle size, mixture temperature and Brownian motion on thermal conductivity of nanofluids. The results showed an increase in thermal conductivity of nanofluid with the increase of nanoparticles weight fraction and mixture temperature [4-7]. Wen and Ding [8] have studied Al_2O_3/water nanofluid heat transfer in laminar flow under uniform wall heat flux and reported an increase in nanofluid heat transfer coefficient with the increase of Reynolds number and nanoparticles weight fraction particularly at the entrance region.

In addition, few works have studied convective heat transfer of nanofluids flow [9, 10]. Xuan and Li [11] investigated the flow features and convective heat transfer characteristics for Cu-water nanofluids inside a straight tube with a uniform heat flux, experimentally. Results showed substantial heat transfer enhancement by using nanofluid instead of pure water. They also

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claimed that the friction factor for the nanofluids at low volume fraction did not produce extra penalty in pumping power. In laminar flow, Nasr Esfahany et al. [12] carried out some experiments on the convective heat transfer of Al_2O_3-Water nanofluid in circular tube. Results clarify that nanofluid in compare to reference fluid posses enhanced heat transfer properties. Furthermore, the increase in heat transfer coefficient due to presence of nanoparticles is much higher than that predicted for single phase heat transfer correlation. Lie Wen Hu et al. [13] have studied convective heat transfer and pressure drop of the zirconia-water and alumina-water nanofluid flow in heated tube at the vertical case. The results show that the heat transfer coefficients are increased 17% and 27% by using aluminai-water nanofluid at 6% vol. in the entrance region and in the fully developed region, respectively. The zirconia–water nanofluid heat transfer coefficient increases by 2% in the entrance region and 3% in the fully developed region at 1.32% vol. Also, they found that generally the pressure drop of nanofluid is higher than pure water. Review of literature shows that only few articles have considered the heat transfer of nanofluid flow inside an inclined tube other than horizontal tube. Recently, Ben Mansour et al. [14] have numerically investigated Water-Al_2O_3 nanofluid inside an inclined tube. By passing laminar nanofluid flow in heated tube, they found that using the nanofluid increases the bouyancy of secondary induced flow and decreases friction at the inner wall. Ben Mansour et al. [15] have also performed an experimental investigation to study mixed convection of Al_2O_3-water nanofluid inside a vertical and horizontal copper tube submitted to a uniform wall heat flux at its outer surface. The effects of concentration of nanoparticles and power supply on the development of thermal field are studied and discussed under laminar flow conditions, but they did not compare the results of the flow in horizontal and vertical tubes. On the other hand, the effect of tube inclination was not considered in this study. Results show that the experimental heat transfer coefficient decreases slightly with an increase of particle volume concentration from 0 to 4%. Two new correlations are proposed to calculate the Nusselt number in the fully developed region for horizontal and vertical tubes.

In the present work, the simultaneous effects of adding nanoparticles to the base fluid, Reynolds number, and tube inclination on the heat transfer of fluid flow are studied. Finally, optimized nanofluid concentration has been defined in each inclination angle of the tube. A new suspension of nanofluid, namely CuO-Base oil is selected for this investigation. The main reason for choosing CuO-base oil nanofluid is that copper oxide nanoparticles are used as additives for industrial oils such as engine oil, heat transfer oil and lubricating oil in order to remove heat from high heat flux surfaces [16, 17]. Also, to study the behavior of CuO nanoparticles more effectively, a type of oil with no additives (SN-500) is used. This type of oil is the basic component of a large number of industrial oils. It is apparent that the effect of nanoparticles on heat transfer performance of the specified oil can be generalized to the mentioned industrial oils for the sake of heat transfer enhancement.

2. NANOFLUID PREPARATION

The nano-sized solid particles used in this study were CuO. These particles were made of an average particle size of 50 nm and high purity via characterization method. The SEM (scanning electron microscope) image of the CuO nanoparticles and the XRD (X-ray diffractions) pattern is shown in Figures 1 and 2, respectively. Reflections in the XRD pattern can be attributed to the CuO particles using JCPDS (Joint Committee on Powder Diffraction Standards). Also, it can be seen from the SEM image of the sample that the majority of nanoparticles are in the form of large agglomerates before dispersion.

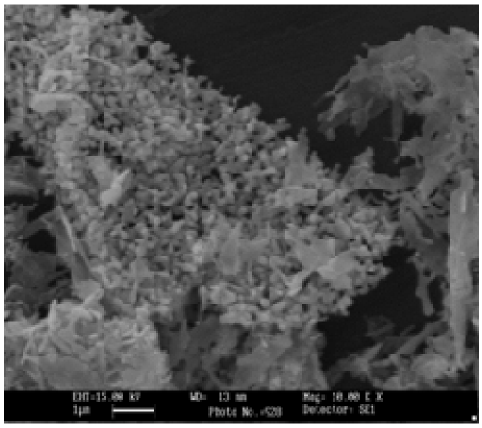


Figure 1. SEM image of CuO nanoparticles

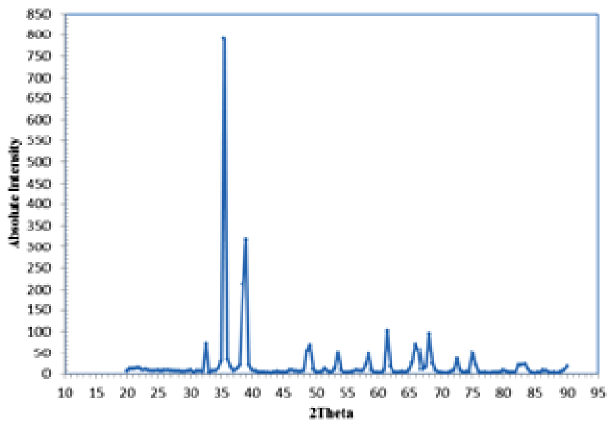
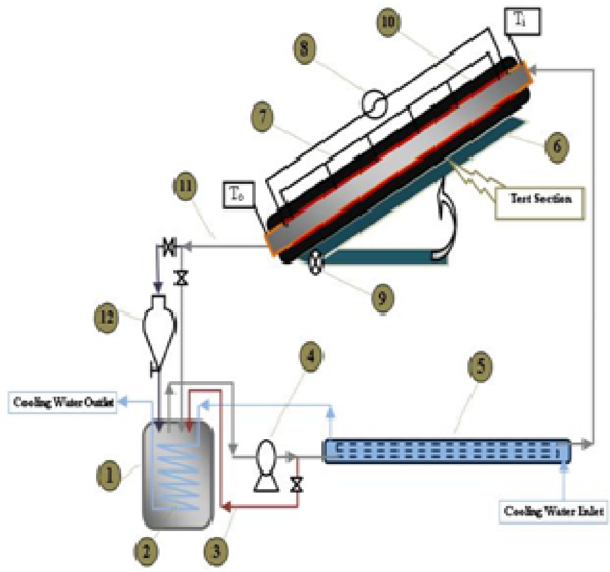


Figure 2. XRD analysis of CuO nanoparticles

Nanofluids with weight fractions of 0.5, 1 and 2% were prepared by dispersing specified amount of CuO nanoparticle in base oil using an ultrasonic processor generating ultrasonic pulses of 400 W at 24 kHz frequency. This device is used to break large agglomerate of nanoparticles in the fluid and make stable suspension. No surfactant was used as they may have some influence on the effective thermal conductivity of nanofluids. It was observed that the nanofluids were uniformly dispersed for 36 hours and the complete sedimentation occurred after a couple of weeks.

3. EXPERIMENTAL SET-UP

The schematic diagram of experimental apparatus is shown in Figure 3. The flow loop consists of a rotary test section, heat exchanger, reservoir, gear pump, flow meter and flow controlling system. The fluid leaves the test section, enters the flow meter, cools partially in the reservoir and then is pumped through a heat exchanger in which water is used as cooling fluid, and again enters the test section. The test section can be rotated from zero degrees (horizontal) to 60 degrees, counter clockwise.



1-	Reservoir	7-	Thermocouples (T_1-T_6)
2-	Pre Cooling Set	8-	AC Power Supply
3-	By Pass Line	9-	Angle control sys.
4-	Gear Pump	10-	Insulator
5-	Heat Exchanger	11-	Main Line
6-	Electrical Resistance	12-	Flow Meter

Figure 3. Schematic diagram of experimental apparatus and guide table

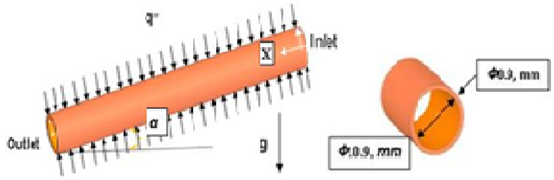


Figure 4. Cross section and inclination of rotary test section

TABLE 1. The range of operating parameters

Parameter	Range
Nanofluid	CuO-base Oil
Nanoparticles weight fraction, %	0.5,1 and 2
Net heat flux, W/m2	8600
Reynolds number	20–140
Tube length, mm	1200
Outer diameter, mm	12.7

In this study, nanofluids with different weight fractions of 0.5, 1% and 2% were used. Also, pure base oil was used for the sake of comparison. A round copper tube of 12.7 mm outer diameter, 0.9 mm wall thickness and 1200 mm length was used for test section.

Figure 4 shows the cross sectional area rotary test section. The nanofluid flowing inside the test section is heated by an electrical heating resistance wrapped around it to generate uniform heat flux. Flow measuring section is consisted of a glass vessel with a valve at its bottom. Flow rate is measured directly from the time required to fill the glass vessel. To adjust the flow rate, a valve in the bypass line is used. 6 K-type (Chromel–Alumel) thermocouples were mounted along the test section to measure the tube wall outside temperature.

The thermocouples were calibrated in a thermostat water bath and the accuracy was found to be within ±0.1 K. Provisions are also made to measure all the other necessary parameters. The ranges of operating parameters are defined in Table 1.

4. DATA COLLECTION AND REDUCTION

All the physical properties of base oil and different weight fractions of nanofluids at various temperatures were measured using accurate measuring instruments. To measure the density of the base oil and nanofluids, SVM3000 was used. The rheological behavior and viscosity of the CuO–base oil nanofluid was measured using Brookfield viscometer with a temperature controlled bath. A differential scanning calorimeter was used to measure the specific heat (Cp) of different nanofluids and the pure oil. In addition, thermal conductivity of the nanofluids with different weight concentrations up to 2% was measured using a KD2 thermal properties analyzer. The oil properties such as density, specific heat, thermal conductivity and dynamic viscosity at 30°C were 871.13[kg.m^{-3}], 4.7[kJ.kg^{-1}.K^{-1}], 0.131[W.m^{-1}.K^{-1}] and 96[cP] respectively. All the base fluid and nanofluid properties were evaluated at the arithmetic mean bulk temperature:

$$T_b = \frac{T_{f,in}+T_{f,out}}{2} \quad (1)$$

The convective heat transfer coefficient is defined as:

$$h(x) = \frac{q''}{(T_w(x)-T_f(x))} \quad (2)$$

where x represents axial distance from the entrance of the test section, q'' the heat flux base on an assumption of zero heat loss, T_w the measured wall temperature, and T_f the fluid temperature determined by the following energy balance equation:

$$T_f(x) = T_{in} + \frac{q''\times p}{\dot{m}\times c_p} x \quad (3)$$

where C_p and $\dot{m}$ are the specific heat and mass flow rate, respectively. T_{in} is the inlet temperature of the flow, P the perimeter of tube and x the distance from the tube inlet. Finally, the following expressions are used to calculate the mean heat transfer coefficient and Nusselt number.

$$\bar{h} = \frac{1}{L}\int_0^L h(x)dx \quad (4)$$

$$\overline{Nu} = \frac{\bar{h}\times D_i}{k} \quad (5)$$

5. RESULTS AND DISCUSSION

5. 1. Validation Check In order to verify the accuracy and the reliability of the experimental system, the heat transfer is experimentally measured using base oil as the working fluid before obtaining those of oil based CuO nanofluids. The experiments are conducted at the Reynolds number of 170. Due to the low Reynolds number of the flow, hydrodynamically fully developed laminar flow is assumed for theoretical calculations. Also, because of the high Prandtl number of the oil, the flow is in the thermal entrance region (x/D<0.05RePr).

In the case in which the temperature variation is small and the thermal properties of the fluid can be assumed uniform, there is an infinite-series solution for the local Nusselt number [18].

$$\mathrm{Nu(x)} = \left[\frac{1}{\mathrm{Nu}_\infty} - \frac{1}{2}\sum_{\mathrm{m}=1}^{\infty}\frac{\exp(-\gamma^2 x^+)}{\mathrm{A_m}\gamma_\mathrm{m}^4}\right]^{-1}\left(\frac{\mu_\mathrm{s}}{\mu_\mathrm{m}}\right)^{-0.14} \quad (6)$$

The variable Nu_∞ is the asymptotic Nusselt number for a laminar uniform heat flux problem and has a value of 4.364; *x*+ is a nondimensional axial location defined as [2*x*/*Do*/RePr], where Re and Pr are the Reynolds and Prandtl numbers of the flow, respectively. A_m and γ_m are defined in Table 2. This solution is used for obtaining local Nusselt number of a fluid flow with temperature varying viscosity inside round tube under uniform heat flux condition. μ_s and μ_m are the fluid viscosity at surface temperature and fluid bulk temperature, respectively. Having the local heat transfer coefficients at six axial locations shown in Figure 3, the average Nusselt numbers were obtained using Equation (5). Figure 5 shows the variation of theoretical with experimental values for average Nusselt number. As it is seen from this figure, the deviation of the experimental data from the theoretical one is within −10% and 10%. Having built-up confidence in the experimental system, systematic experiments were performed on the heat transfer characteristics of oil-based CuO nanofluids flowing inside the round tube over a Reynolds number of 20–170 under uniform heat flux.

5. 2. Uncertainty Analysis The uncertainty in calculating the major heat transfer parameters has been conducted using the values presented in Table 3, and based on the method proposed by Kline and McClintock [19]. The uncertainties of heat transfer coefficient, Reynolds and Nusselt numbers were 3.55, 4.4 and 2.98percent, respectively.

5. 3. Heat Transfer in Horizontal Tube The results are presented and discussed in this section. The effects of Reynolds number and nanoparticle weight fraction on the convective heat transfer of the pure oil and nanofluid flow in horizontal tube were examined under constant heat flux (see Figure 6).

TABLE 2. Values of A_m and γ_m

M	γ_m^2	$A_m \times 10^{-3}$
1	25.68	7.630
2	83.86	2.053
3	174.2	0.903
4	296.5	0.491
5	450.9	0.307

For higher values of m: $\gamma_m = 4m + \frac{4}{3}$, $A_m = 0.4165\gamma_m^{-7/3}$

TABLE 3. Measuring instruments range and accuracy

Description	No.	Model	Range	Accuracy
1-Temp. of the outer tube surface	4	Type K	−100 to 1370 °C	±1 °C
2-Temp. of fluid flow(inlet and outlet)	2	Type K	−100 to 1370 °C	±0.2 °C
3-Fluid vol. flow rate	1	Glass vessel	0 to 0.5 l	±50cc

As expected, the mean heat transfer coefficient enhances by increasing Reynolds number extremely. .According to this figure, as the Reynolds number goes up, heat transfer coefficient increases for both base fluid and nanofluid flows. However, Reynolds number increment has stronger effect on the heat transfer enhancement of nanofluid with 2% wt. fraction. At low Reynolds numbers, the high viscosity of the oil prevents the nanoparticles from free movements and keeps them in a limited area. Also at higher flow rates, the dispersion effects and chaotic movement of the particles are intensified which causes the temperature profile to be more flattened, similar to turbulent flow, and leads to an increase in the heat transfer coefficient.

In addition, , the result show obviously the beneficial effects due to the presence of nanoparticles. As regards to thermal conductivity is uniform along the tube length for the test fluid, Adding nanoparticle to the base fluid made the thermal conductivity of the resulting mixture improve considerably. Besides, presences of nanoparticles cause a delay in developing thermal boundary layer. Both thermal conductivity and thermal boundary layers are important factors for increasing the convective heat transfer coefficient in laminar flow.

It is interesting to note that the most enhancement of mean heat transfer coefficient of 2% wt. nanofluid with reference to base oil at Re=110 is approximately 10% .

5. 4. The Effect of Inclination angle on Heat Transfer Figures 7 and 8 illustrate the effect of increase in tube angle on heat transfer of base fluid and nanofluid with 2% wt. Increase of tube angle from 0 to 30 degrees leads to an increase of mean heat transfer coefficient, but heat transfer coefficient decreased slightly when tube inclination angle increased by 60 degrees.

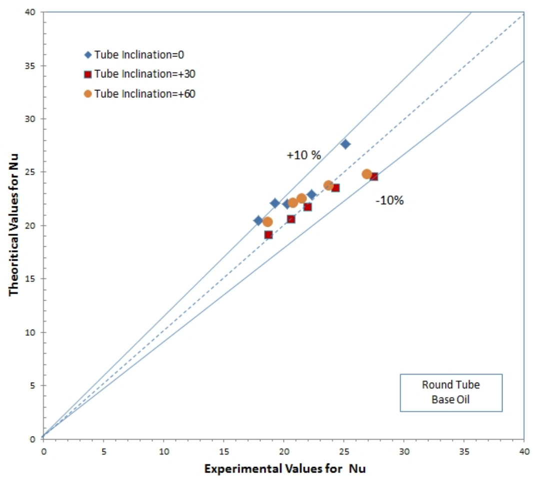


Figure 5. Comparsion between theoretical and experimental Nusselt number of base oil flow inside round tube at zero and different inclinations of tube

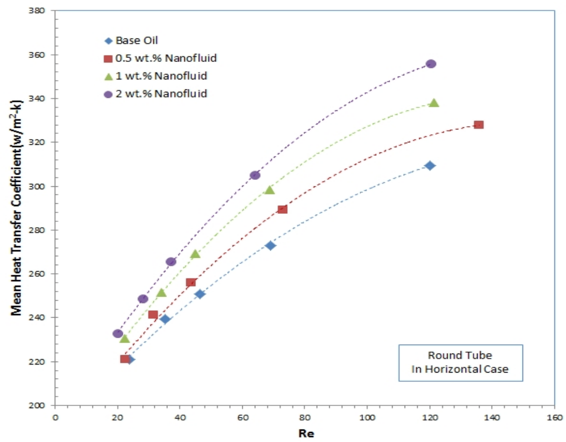


Figure 6. Variation of heat transfer with Reynolds number for base oil and nanofluids flow inside the horizontal tube at uniform heat flux

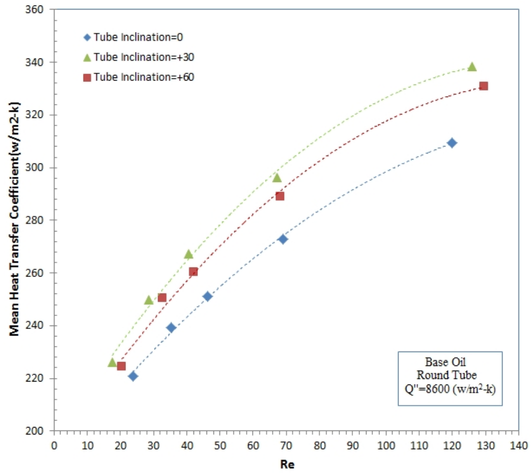


Figure 7. Variation of heat transfer with Reynolds number for base oil flow at different tube inclinations

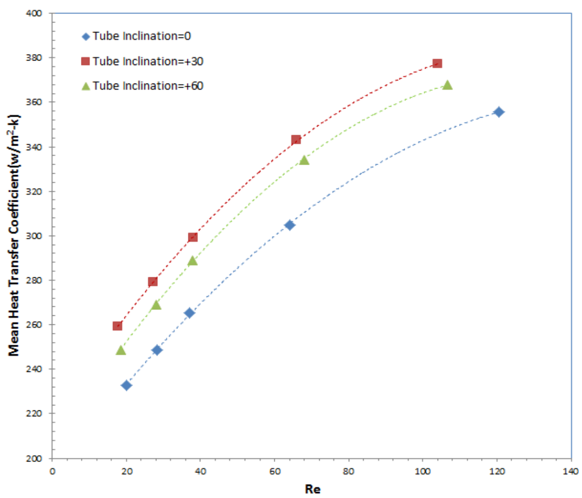


Figure 8. Comparsion of variation of mean heat transfer coefficient with Reynolds number of 2% wt. nanoflouid flow for different tube inclinations

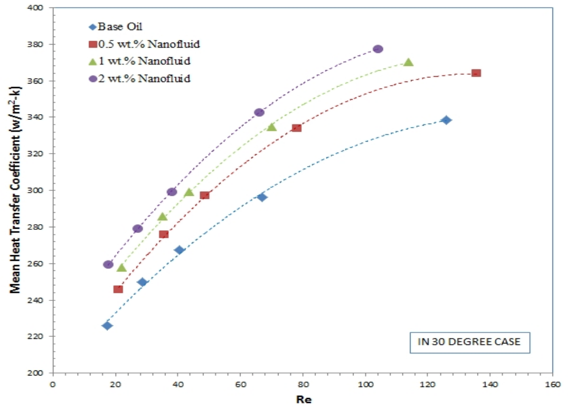


Figure 9. Variation of mean heat transfer coefficient with Reynolds number for various nanofluid concentrations at tube inclination=30°

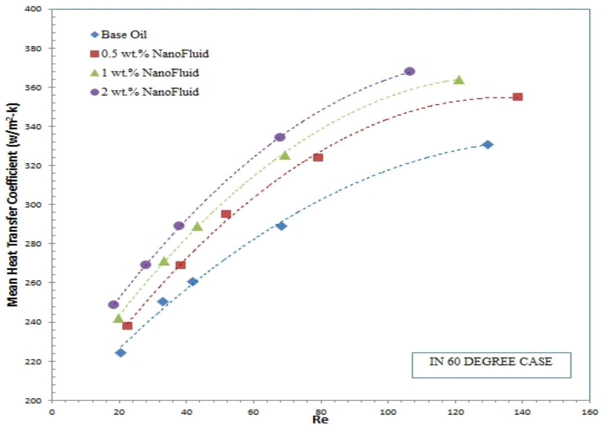


Figure 10. Variation of mean heat transfer coefficient with Reynolds number for various nanofluid concentrations at tube inclination=60°

Figures 9 and 10 show the augmentation of nanofluid weight fraction in tube inclination at 30 and 60 degrees. The results exhibit the heat transfer coefficient enhancement because of presence of nanoparticles in pure oil inside non horizontal tube.

By increasing the tube angel, axial component of buoyancy force greatly increases. Since flow direction and axial buoyancy force are opposite, so velocity of hot fluid flow near the inner wall decrease and random motion of nanoparticles in the base fluid will be more.

As a result, the heat transfer coefficient will increase. On the other hand, by increasing the axial component of buoyancy force, radial component of it that makes rotational flow decreases so the average heat transfer coefficient is reduced. Finally, all of these factors cause Nano-fluid to have most heat transfer enhancement in tube angle of 30 degrees.

We defined index of optimized nanofluid weight fraction, γ, to study more precisely the effect of tube angle changes from horizontal state to 60 degrees on the heat transfer enhancement, as below (Equation (7)):

$$\gamma = \frac{\bar{h}_{nf,\alpha}-\bar{h}_{nf,0^\circ}}{\bar{h}_{bf,\alpha}-\bar{h}_{bf,0^\circ}} \quad (7)$$

Index of optimized nanofluid weight fraction, γ, expressed in terms of increase of nanofluid heat transfer to basic fluid one. Here, α is equal to 30 or 60 degrees. Variations of γ along with Reynolds number for nanofluid with various weight fractions is shown in Figures 11 and 12.

Apparently, when the index of optimized nanofluid weight fraction is greater than 1, it implies that the effect of tube inclination on nanofluid heat transfer is more than that of the basic fluid. Therefore, the heat transfer methods are applicable for nanofluid inside inclined tube.

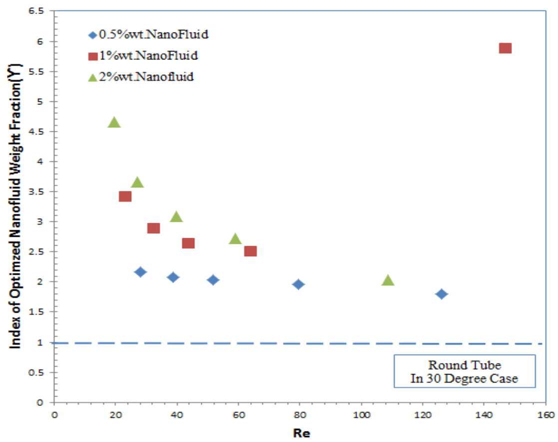


Figure 11. Variation of index of optimized nanofluid weight fraction with Reynolds number in 30° tube inclination

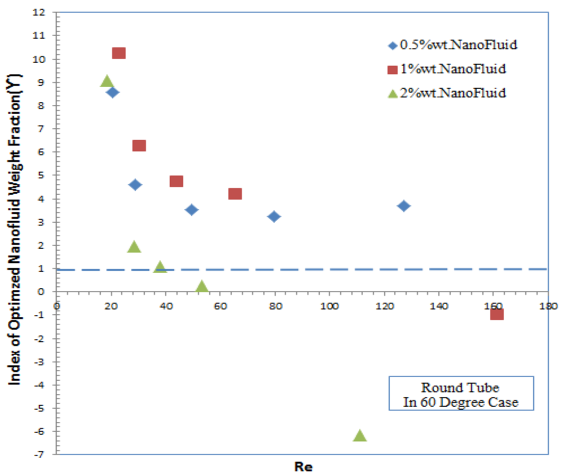


Figure 12. Variation of index of optimized nanofluid weight fraction with Reynolds number in 60° tube inclination

6. CONCLUSION

In the present study, effects of nanoparticle weight fraction, tube inclination and Reynolds number on the mean convective heat transfer coefficient of CuO-base oil nanofluids in a single 10.9 mm inner diameter, uniform heat flux copper tube was investigated for laminar flow in thermal developing and hydrodynamical fully developed region. It is concluded that addition of CuO to base oil increases the mean convective heat transfer coefficient in horizontal and inclined tube extremely. At the same flow conditions and for a given nanofluid with constant particle fraction, increasing the tube inclination angle increases the heat transfer.

In the 30 degrees tube angle, γ is greater than 1 for various nanofluid flow concentrations, while, in 60 degrees tube angle, that one is true except for 2%wt. fraction nanofluid.

The mean heat transfer coefficient greatly increased with increasing Reynolds number augmentation in horizontal and inclined tube. Most enhancement in mean heat transfer coefficient for nanofluid with 2% wt. at 30° and 60° with reference to horizontal data are 15.2 and 8.1percent in Re=40, respectively.

Buoyancy force and chaotic movement were the main reason of the average heat transfer coefficient enhancement of nanofluid against basic fluid in horizontal and inclined tubes. Nanofluid flow with 2% wt. had the best performance at tube inclination=30°, but at tube inclination=60°, the 1% wt. nanofluid had it generally.

7. ACKNOWLEDGEMENT

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Laminar

چکیده

آزمایش‌های تجربی برای مطالعه انتقال حرارت جابه‌جایی جریان نانوسیال داخل لوله مسی شیب‌دار تحت شار حرارتی یکنواخت انجام شده است. اطلاعات مورد نیاز برای جریان آرام و توسعه یافته هیدرودینامیکی داخل لوله دایروی به دست آمدند. نانوسیال پایدار اکسید مس-روغن پایه با کسرهای وزنی مختلف ۰/۵، ۱ و۲ درصد نانوذرات به وسیله دستگاه آلتراسونیک به روش دو مرحله‌ای تولید شد. در این مطالعه اثر پارامترهای مختلف مانند شیب لوله، کسر وزنی نانوسیال و عدد رینولدز روی ضریب انتقال حرارت بررسی شد. نتایج نشان می‌دهد که ضریب انتقال حرارت نانوسیال با کسرهای وزنی مختلف با افزایش عدد رینولدز داخل لوله دایروی افقی و شیب‌دار افزایش می‌یابد. همچنین، جریان نانوسیال داخل لوله شیب‌دار در ۳۰ درجه، بیشترین افزایش انتقال حرارت را در میان شیب‌های دیگر لوله در اعداد رینولدز یکسان از خود نشان می‌دهد.

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