Global Journal of Researches in Engineering: A
Electrical and Electronics Engineering
Volume 17 Issue 4 Version 1.0 Year 2017
Type: Double Blind Peer Reviewed International Research Journal
Publisher: Global Journals Inc. (USA)
Online ISSN: 2249-4596 & Print ISSN: 0975-5861

Heat Transfer and Pressure Drop Characteristics of the Buoyancy-Aided Heat Transfer Oil-Copper Oxide (HTO-Cuo) Nanofluid Flow in Vertical Tube

By Farhad Hekmatipour, Mohammad-Ali Akhavan-Behabadi & Behrang Sajadi
Science and Research Branch-Islamic Azad University

Abstract- In this paper, the mixed natural-forced convection is experimentally investigated for the heat-transfer oil-copper oxide (HTO-CuO) nanofluid flows upward in a vertical tube. The flow regime is laminar and the temperature of the tube surface is constant. The effect of the nanoparticles concentration on the heat transfer rate and the pressure dropare studied as Richardson number varies between 0.1 and 0.7. It is observed that the mixed convective heat transfer rate increases with both the nanoparticles concentration and Richardson number. New correlations are proposed to predict the Nusselt number of the nanofluid flow with the reasonable accuracy. In addition, Darcy friction factor of the nanofluid flow is investigated and a new correlation is presented to evaluate the friction factor of HTO-CuO nanofluid flow in vertical tubes.

Keywords: nanofluid; heat transfer oil; mixed convection; vertical tube; laminar flow.

GJRE-A Classification: FOR Code: 091399

HEATTRANSFERANDPRESSUREDROPCHARACTERISTICSOFTHEBUOYANCYAIDEDHEATTRANSFEROILCOPPEROXIDEHTOCUONANOFLUIDFLOWINVERTICALTUBE

Strictly as per the compliance and regulations of:

Heat Transfer and Pressure Drop Characteristics of the Buoyancy-Aided Heat Transfer Oil-Copper Oxide (HTO-Cuo) Nanofluid Flow in Vertical Tube

Farha Hekmatipour [α], Mohammad-Ali Akhavan-Behabadi [σ] & Behrang Sajadi [ρ]

Abstract- In this paper, the mixed natural-forced convection is experimentally investigated for the heat-transfer oil-copper oxide (HTO-CuO) nanofluid flows upward in a vertical tube. The flow regime is laminar and the temperature of the tube surface is constant. The effect of the nanoparticles concentration on the heat transfer rate and the pressure dropare studied as Richardson number varies between 0.1 and 0.7. It is observed that the mixed convective heat transfer rate increases with both the nanoparticles concentration and Richardson number. New correlations are proposed to predict the Nusselt number of the nanofluid flow with the reasonable accuracy. In addition, Darcy friction factor of the nanofluid flow is investigated and a new correlation is presented to evaluate the friction factor of HTO-CuO nanofluid flow in vertical tubes. As the heat transfer enhancement methods usually accompany with increment in the pressrure drop, the performance index is evaluated experimentally. As such, the maximum perfomance index of 1.27 is achieved using the 1.5% concentration of the nanoparticles at Richardson number of 0.7. This study provides a platform to design next generation of low flow rate nanofluid-based heat exchangers and may improve the accuracy of predicting the mixed convective characteristics of nanofluid flows.

Keywords: nanofluid; heat transfer oil; mixed convection; vertical tube; laminar flow.

I. Introduction

Mixed convection heat transfer is widely used in industrial applications, e.g. advanced technologies such as microelectronics cooling, air conditioning, as well as petrochemical, oil and gas industries. Enhancement of the mixed convection heat transfer has a significant role on the energy saving and on the compactness perspectives of heat exchangers. As one the first study, Sider and Tate [1] carried out an experimental investigation on the mixed convection heat transfer in vertical isothermal tubes. Although, they also proposed a correlation to predict the experimental data, the error of correlation was approximately 30%.

During the last decade, some correlations have been presented to predict the mixed convection heat transfer in vertical [2-4], horizontal [5,6], and inclined [7,8] tubes. As one of the first interesting research work, Joye [3] conducted an experimental investigation on the pressure drop of the laminar mixed convection flow in vertical tubes. In addition, he presented a new correlation to evaluate the pressure drop with the maximum error of 10%.

The inherent relatively low thermal conductivity of conventional fluids, e.g., water, glycol solution, and oil, impacts the convective heat transfer rate. It is accepted that adding nanoparticle to the base fluid is an effective method to enhance its thermal-rheological characteristics and the flow thermal performance. Accordingly, many research works have been carried out to enhance the heat transfer rate using nanoparticles. For the first time, uniform suspension of nanoparticles in a liquid was introduced by Choi and Eastman [9] to create a new type of solid. Afterwards, many research works has been conducted to investigate the effect of adding nanoparticles as a heat transfer enhancement [10-12]. As one of the first study, Lee [10] studied four nanofluids consisted of CuO and Al2O3 nanoparticles in water and glycol based fluids. The maximum volume concentration of nanoparticles was 5% leaded to 30% increase in the thermal conductivity. Through the recent years, some empirical and theoretical study have been done in order to be measured the properties of high Prandtl nanofluid and impact of high Prandtl nanofluid on forced convection heat transfer in horizontal [13-22]. Based on the [18], three type of nanoparticles such as copper dioxide, titanium dioxide, and aluminium dioxide were used to mix with turbine oil. The result of this investigation is conclude that the heat transfer and Nusselt number increased using nanoparticles. Although, the results show that the influence of using CuO is more than TiO_2, and Al_2O_3. During the recent decades, many experimental and numerical research work have focused on the influence of the nanoparticles on the mixed natural-forced convection heat transfer and pressure drop, in horizontal [23-27], vertical and inclined [27-33]

Author α: Department of Energy and Environment, Science and Research Branch, Islamic Azad University, Tehran, Iran.
e-mail: farhad.hekmatipour@srbiau.ac.ir
Author σ ρ: Center of Excellence in Design and Optimization of Energy Systems, School of Mechanical Engineering, College of Engineering, University of Tehran, Tehran, Iran.
e-mails: Akhavan@ut.ac.ir, B.Sajadi@ut.ac.ir

Global Journal of Researches in Engineering (A) Volume XVII Issue IV Version I Year 2017 1

tubes. According to the [33], mixed convection heat transfer and pressured drop are risen using the nanoparticle and changing the tube inclination. Furthermore, several correlations are proposed to be measured the effect of nanofluid in inclined tube.

Numerical studies on the fully developed laminar mixed convection heat transfer of water in horizontal and inclined [34-37] tubes have been performed during the recent years. The results demonstrated that the mixed convection heat transfer was enhanced by adding nanoparticle to base fluids or changing the inclination angle. Whereas, Ben Mansour [28,34] stated although the nanoparticles concentration has no significant effect on the hydrodynamics of the flow, it may enhance the heat transfer coefficient. Based on his results, Darcy friction factor increases monotonically with the inclination angle, while the heat transfer coefficient shows a peak at the angle of 45°.

In this paper, the mixed convective heat transfer and pressure drop characteristics of a buoyancy-aided nanofluid flow in a vertical tube is investigated experimentally. As such, this research is conducted to study the effect of using copper oxide nanoparticles on the heat transfer and pressure drop characteristics of the heat transfer oil flow. The tube wall temperature is constant and the flow rate is low enough to ensure that the flow regime is always laminar.

II. Experimental Apparatus

a) Nanofluid Properties

In this study, solid particles of copper oxide with the average size of 40 nm and the purity of 99% were used as nanoparticles. XRD (X-ray diffraction) analysis and SEM (scanning electron microscope) image of thenanoparticles are shown in Figs .1, and 2, respectively. As shown in Figs. 1 and 2, the nanoparticles are almost spherical. In order to obtain a homogeneous and a relatively stable nanofluid, an ultrasonic UPS400 apparatus with the frequency of 24 kHz and the power of 400 W was used. In this study, three samples of nanofluids were prepared including suspension of the heat transfer oil-copper oxide nanoparticles with the mass concentration of 0.5%, 1% and 1.5%. The nanofluids were stable for 216hr, after then the nanoparticles started to precipitate and settled down completely after 14 days.

The range of operations flow and Copper oxide (CuO) nanoparticles are shown in Tables 1 and 2, respectively.

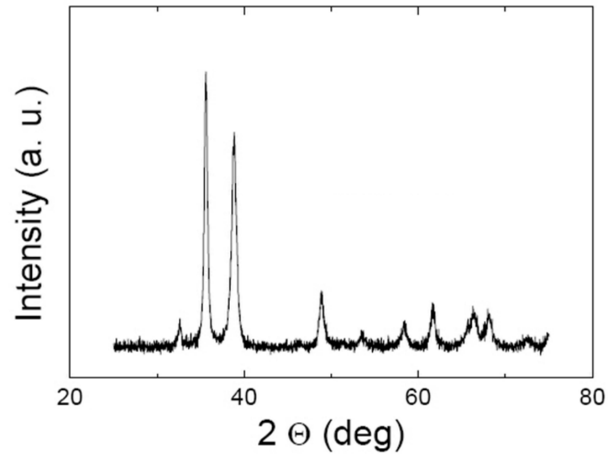

Fig. 1: XRD analysis of copper oxide nanoparticles

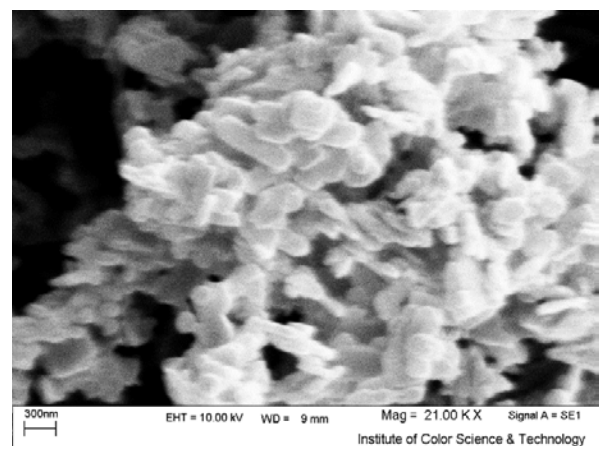

Fig. 2: SEM image of copper oxide nanoparticles

The thermal-rheological properties of copper oxide-heat transfer oil nanofluids are reported [12]. In addition, the ranges of applicability of these correlations and investigation are presented in Table 3.

b) Test Set Up

Table 1: Thermo-physical properties of the heat transfer oil

Thermo-physical property	Temperature (°C)	
	83	100
Density (kg/m^3)	855	815
Heat capacity (kJ/kg.K)	2.03	2.30
Kinematic viscosity (mm^2/s)	32	5.2
Thermal conductivity (W/m.K)	0.133	0.128
Prandtl number	395	76

Table 2: Thermo-physical properties of CuO nanoparticles

Thermo-physical property	Value
Morphology	Nearly spherical
Particle size (nm)	40
Purity	99%
Bulk density (kg/m^3)	790
True density (kg/m^3)	6400
SSA (m^2/g)	20
Thermal conductivity (W/mK)	20

Table 3: The range of applicability of correlations

Items	Value
Gr	8000 to 37400
Pr	330 to 385
Re	200 to 750
D/L	0.0178
Ri	0.1 to 0.7
Gz	1387 to 3676
a	0.0005 to 0.00083

In order to study the heat transfer and the pressure-drop of the nanofluid flow in vertical tubes, an experimental setup was designed as presented schematically in Fig. 3. The flow circuit has several parts including: test section, pre-cooler, reservoir tank, heat exchanger, gear pump, flow meter, differential-manometer, thermocouples and flow control system. In the experiments, a 500 mm smooth tube with inner and outer diameter of 8.92 mm and 9.52 mm was used. The test tube is located in a steam tank to keep the tube wall at a constant temperature of 98°C. The steam tank is insulated using fiberglass to reduce its heat losses. Due to installing the pressure transmitter and main line to tube test, steam hoses enduring the 230°C and 7.0 bar are used due to steam hoses which eliminate the effect of elbow and horizontal tube on increasing pressure drop and prevent the heat transfer between fluid and steam tank. In addition, in order to approve the results of experimental investigation, the steam hoses have insulated using fiberglass. The cooling system of the setup has two stages. In the first stage, the cooling water is used to precool the nanofluid using a copper coil embedded in the reservoir tank. In the second one, the cooling water cools down the nanofluid flow to about 15°C in a shell and tube heat exchanger. After initial cooling of the nanofluid inside the reservoir tank, it is pumped to the main line by a gear pump. As the gear pump speed is fixed, a bypass line is used to control the flow rate in the main line. Adjusting the flow rate is accomplished using a globe valve to bypass some of the flow to the reservoir tank. The main line flow rate is such that the flow is always in the laminar regime. After the RTD sensor entered into tube, the tube is attached to a steam hose connected to test tube. This test section, four thermocouples are installed at specified intervals to measure the tube wall temperature. In addition, two thermocouples are installed at the inlet and outlet of the test section to measure the inlet and outlet flow temperatures. The time required for the flow to become steady was about 15 minutes and the data were recorded after 30 min.

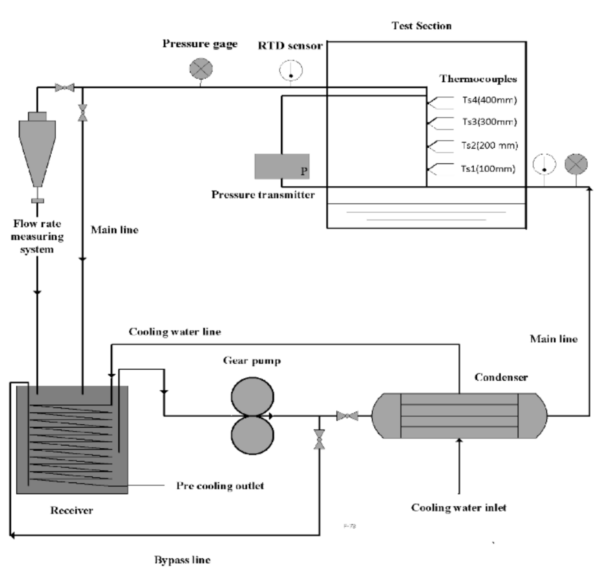

Fig. 3: Schematic of the test setup

c) Instrument

To measure the nanofluid temperature in the test section inlet and outlet, two RTD PT 100joined to thermometers are used with the accuracy of ±0.1°C.. The RTD sensors entered in tubes are sense the central temperature of fluid as the inlet and outlet temperatures. In addition ,in order to determine the tube wall temperature which is constant during the tests, four K-type thermocouples with the SU-105 KPR sensor were welded on the tube with the 100 mm interval, T_1 (100 mm), T_2 (200 mm), T_3 (300 mm), and T_4 (400 mm). Since the velocity of fluid which is near the surface tube is zero or have downward flow. Thus, the fluid temperature which is near the wall tube is approximately equal with the temperature of surface tube. In addition, the temperature of film flow calculated using the temperature of surface tube and bulk temperature is required to determine the Grashof and Richardson number. The surface temperatures which are obtained for T_1, T_2, T_3, T_4 are 98.1, 97.8, 98, 98.1, respectively. Since the velocity of fluid which is near the surface tube is near zero or have downward flow. Thus, the fluid temperature which is near the wall tube is approximately equal with the temperature of surface tube. Thus, it is possible to use the temperature of tube surface as wall fluid temperature.

A PMD-75 pressure transmitter with the accuracy of ±0.075% was implemented to measure the pressure drop. To measure the flow rate, a 1000 ml scaled separation funnel was used. In this method, the flow rate may be directly measured by means of measuring the funnel filling time using a digital timer with the accuracy of 0.01s.

Error analysis of the heat transfer and the pressure drop measurements were performed based on Kline and McClintock [39] method using the data depicted in Table 4. The specimen of computing the error analysis is mentioned in appendix 1.

Accordingly, the maximum measurement error of Darcy friction factor, Nusselt number, and the performance index were 6.8%, 4.3% and 6.5%espectively.

III. Result and Discueeion

Hydro-dynamically fully developed laminar flow relies on Reynolds number, which increases up to 730 in this work. As a consequence, the flow of pure heat transfer oil and nanofluid are assumed fully-developed (L/D > 0.05Re). Furthermore, owing to the high Prandtl number of the pure heat transfer oil and nanofluid, the flow is in the thermal entrance region (L/D < 0.05RePr). The Darcy friction factor and the Nusselt number are used to evaluate the nanofluid flow pressure drop and heat transfer coefficient, respectively. White [40], and Bergman et al [41]:

$$f = \frac{\pi^2 \rho D^5}{2L\dot{m}^2}\Delta p \qquad (1)$$

$$Nu = \frac{\dot{m} C_p}{\pi L k}\, ln\left(\frac{T_w - T_{b,i}}{T_w - T_{b,o}}\right) \qquad (2)$$

T_w is surface temeperature. $T_{b,i,}$ and $T_{b,o}$ are bulk inlet temperature and bulk outlet temperature.

To verify the accuracy of the experimental results, in Fig. 4, the experimental mixed convection Darcy friction factor and Nusselt number of the pure heat transfer oil flow in a vertical tube are compared with the results of classic Joye [3] and Eubank and Proctor [2] correlations, respectively. The Joye [3] and Eubank Proctor correlation [2] are reported as is followed:

$$\Delta P_{lam} = \left(\frac{128\,\mu L Q}{\pi D^4}\right)(\rho g)\left(\frac{Sp.G.}{\left(\frac{\mu_b}{\mu_w}\right)^{0.38}}\right) \qquad (3a)$$

$$\frac{\Delta P}{\Delta p_{lam}} = 1 + 1565\,\frac{Gr_L^{3/4} Pr^{1/2}}{(0.952+Pr)^{3/4} Re^2} \times (L/D)_b^2 \qquad (3b)$$

a)

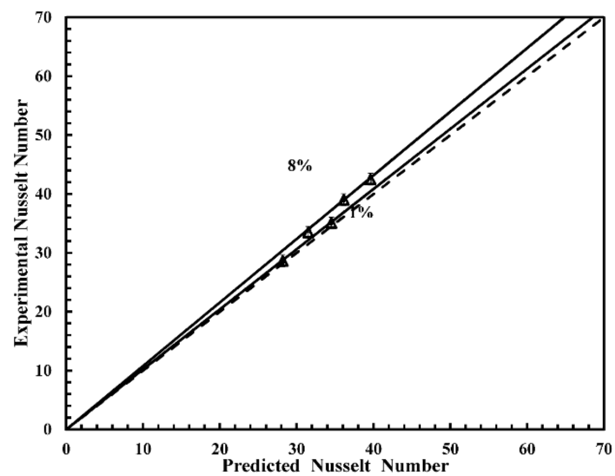

Fig. 4: Comparison of the experimental data with the classic correlations: (a) Nusselt number; (b) Darcy friction factor

The correlation of Joye is vailed for 200≪Re≪2100, Pr =0.7. The vadiation of Euband and Proctor correlation is adequate for $10^3 \ll Gr\,\mathrm{Pr}\,d/l \ll 10^9$, Pr= 0.7. The maximum error of the experimental results for Darcy friction factor and Nusselt number are 16% and 8%, respectively, which demonstrates the accuracy of the expetimental results.

a) Heat Transfer

At first, the effect of the nanoparticles concentration on the mixed thermally developing heat transfer rate in vertical tubes is investigated. Fig. 5 shows that Nusselt number increases with both Gz number and the nanoparticles mass concentraion. The maximum Nuesselt number is reached at the concentration of 1.5% leads to 16% enhancement with respect to the base fluid flow. The experimental results for Nusselt number are compared with the prediction of Eubank and Proctor [2] correlation in Fig. 6. The maximum error of Eubank and Proctor is 24%. It is evident from Fig. 6 that this equation cannot predict accuratelythe mixed convection Nusselt number of nanofluid flows. As a consequence, to predict the thermally developing Nusselt number of nanofluid flows accurately, Eubank and Proctor correlation [2] should be modified based on the obtained experimental results of this study as:

$$\mathrm{Nu} = 1.8\left(\frac{\mu_b}{\mu_w}\right)^{0.14}\left[Gz + 0.12\left(GrPr\,\frac{D}{L}\right)^{0.44}\right]^{1/3} \qquad (5)$$

Rang of applicability of correlation is mentioned in Table 3. In which the thermo-physical properties of the nanofluid is used to evaluate Graetz and Nusselt numbers.

Table 4: Specification of the instruments

Property	Instrument	Range	Accuracy
Inlet/outlet temperature	RTD PT 100	-200 to 400°C	±0.1°C
Tube surface temprature	K-type thermocouple	-200 to 999°C	±0.1°C
Flow rate	Separation funnel	0 to 11	±100 ml
Pressure drop	PMD-75	10 mbar to 40 bar	±0.075

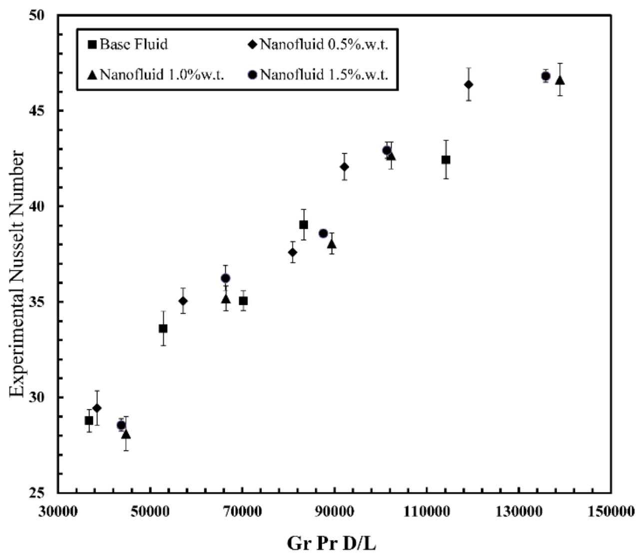

Fig. 5: The effect of using CuO nanoparticles on Nusselt number of the nanofluid flow

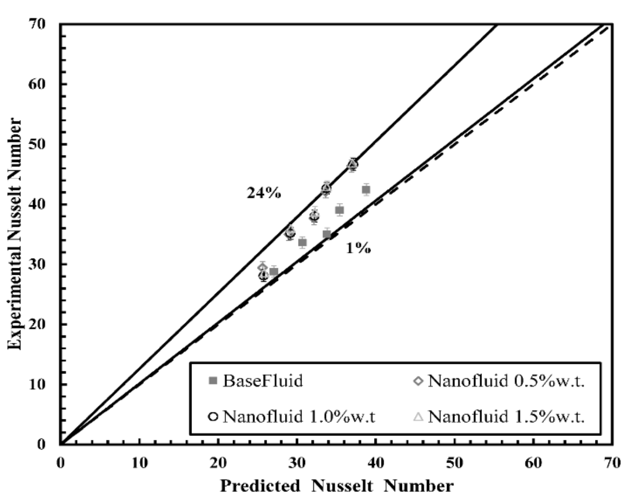

Fig. 6: Comparison of the experimental Nusselt number of the nanofluid flow with the prediction of Eubank and Proctor (1951) correlation

Fig. 7 compares the experimental Nusselt number of the HTO-CuO nanofluid flow with the prediction of Eq. (5). As shown in Fig. 7, the maximum discrepancy between the predictions of Eq. (5) and the obtained experimental results is less than 10% which iscompletely acceptable for this type of flow.

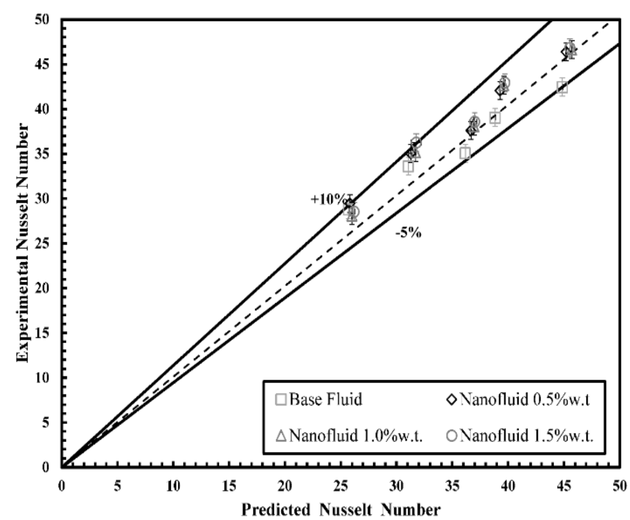

Fig. 7: Comparison of the experimental Nusselt number of the nanofluid flow with the prediction of Eq. (3)

To emphasize on the effect of natural convection on the mixed heat transfer enhancement of the nanofluid flow Nusselt, Fig. 8 illustrates the natural convection effects of the heat transfer enhancement of nanofluids in vertical tubes. As shown in the Fig., the effect of natural convection increases with Richardson number. In addition, the Nusselt number augments with the nanoparticles mass concentration so that 50% enhancement is achieved at the nanoparticles concentration of 1.5% and Richardson number of 0.7. The experimental results in Fig. 8 may be correlated based on Richardson number to normalize the effect of mixed convection by the forced one:

$$\frac{\mathrm{Nu}_{nf}}{\mathrm{Nu}_{bf}} = 1.17\left(1 + \left(\frac{Gr}{Re^2}\right)^{0.8}\right)^{0.4} \qquad (6)$$

Rang of applicability of correlation is mentioned in Table 3. Fig. 9 shows the comparison of the experimental data with the results of Eq. (6). As the maximum error of the correlation is 12%, it can be used to estimate the effects of mixed heat transfer of the nanofluid flow in vertical tubes with an acceptable accuracy.

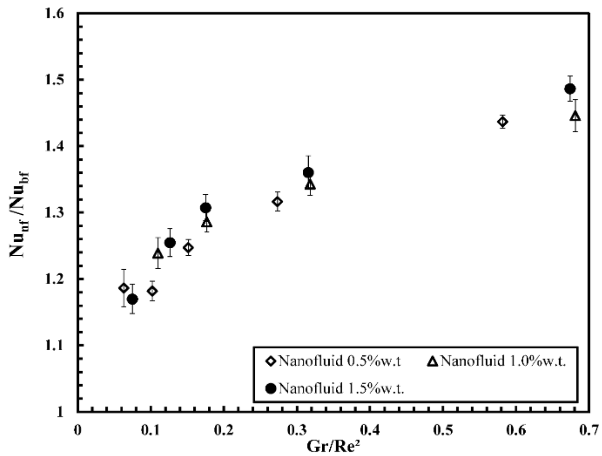

Fig. 8: The effect of natural convection on the mixed convection heat transfer of the nanofluid flow

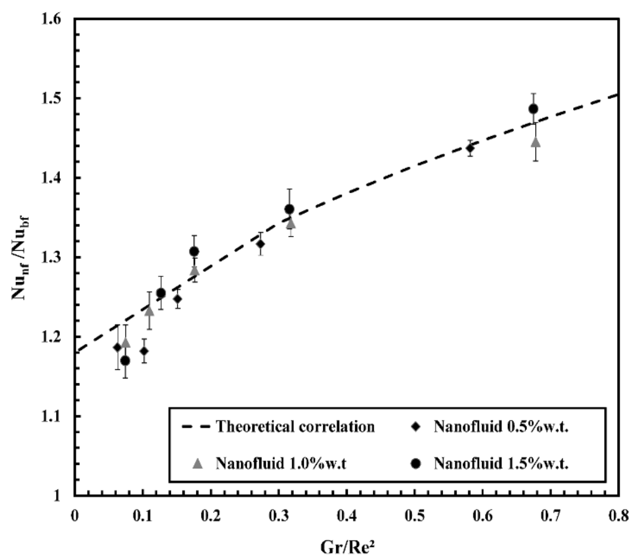

Fig. 9: Comparison of the experimental Nusselt number of the nanofluid flow with the prediction of Eq.

b) Pressure Drop

The effect of using CuO nanoparticles on the mixed convection friction factor of the nanofluid flow in vertical tubes is shown in Fig. 10. The flow friction factor increases with Richardson number and the nanoparticles mass concentration which is mainly attributed to the nanoparticles Brownian motion makes the velocity profile more uniform and increases the shear wall stress. The CuO nanofluid flow is compared with Joye [3]correlation in Fig. 11. As shown in the Fig., although Joy equation may successfully predict the friction of base fluid flow, it is not suitable to evaluate the nanofluid flow friction factor. As a result, the following correlation is presented to estimate the Darcy friction factor of the HTO- CuO nano fluid flow, using the least square

$$0.12Ri^{0.66}\left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{7}$$

Rang of applicability of correlation is mentioned in Table 3. Fig. 12 compares the experimentalDarcy friction factor data with the predictions of Eq. (5). Accordingly,

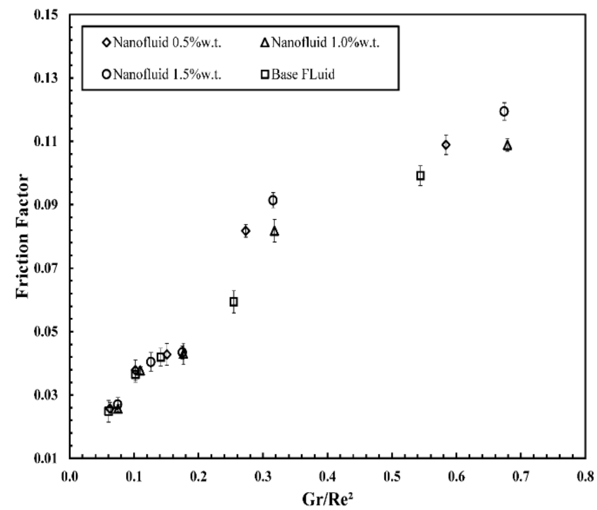

Fig. 10: The effect of using CuO nanoparticles on the Darcy friction factor of the nanofluid flow

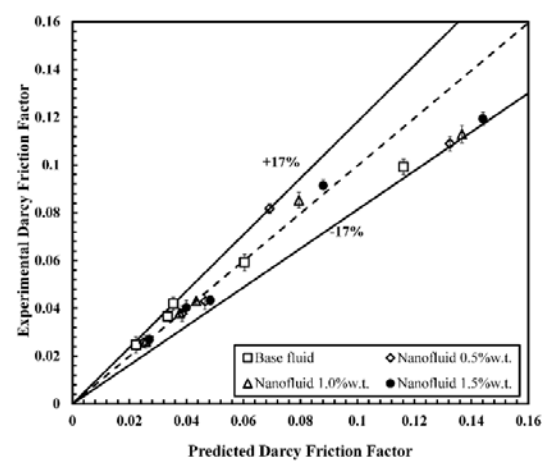

Fig. 11: Comparison of the experimental Darcy friction factor of the nanofluid flow with the prediction of Joye [3] correlation

the presented correlation computes the frcition factor of laminatr bouancy-aided nanofluid flow in vertical tubes with a good accuracy.Rang of applicability of correlation is mentioned in Table 3. Fig. 12 compares the experimental Darcy friction factor data with the predictions of Eq. (5). Accordingly, the presented correlation computes the frcition factor of laminatr bouancy-aided nanofluid flow in vertical tubes with a good accuracy.

c) Performance Index

Heat transfer enhancement methods, e.g. using nanoparticles, are usually accompanied with an increase in the pressrure drop. In order to determine the effect of simultaneous increase in the heat transfer and the pressure drop, the prefromance index may be defined as:

$$\eta = \frac{h_{nf}/h_{bf}}{\Delta P_{nf}/\Delta P_{bf}} \qquad (8)$$

The preformance index larger than one shows thatusing nanoparticles is more in favor of heat trasfer improvement rather than in pressure drop increment.

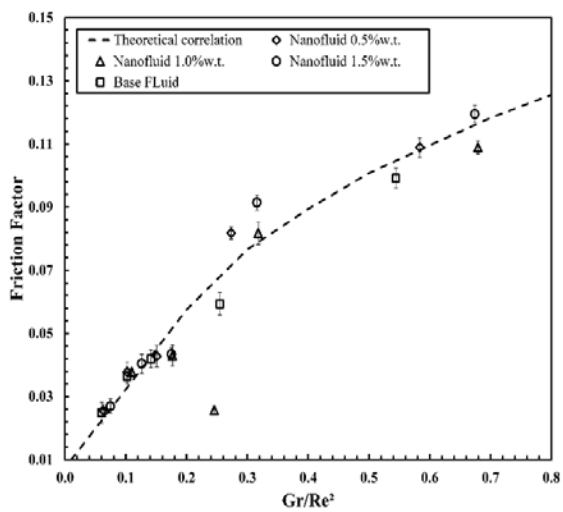

Fig. 12: Comparison of the experimental friction factor of the nanofluid flow with the prediction of Eq. (5)

The performance index of the system can be calculated based on the heat transfer rate and the pressure drop of the pure heat transfer oil and the HTO-CuO nanofluid flow. Fig. 13 indicates the effect of Richardson number and the nanoparticles mass concentraion on the perfomance index. Based on the results, it is observed that although the performance index is not always larger than unity, its maximume is aboud 1.27 which is achieved with 1.5% nanoparticles concentraion and Richrdson number of 0.7.

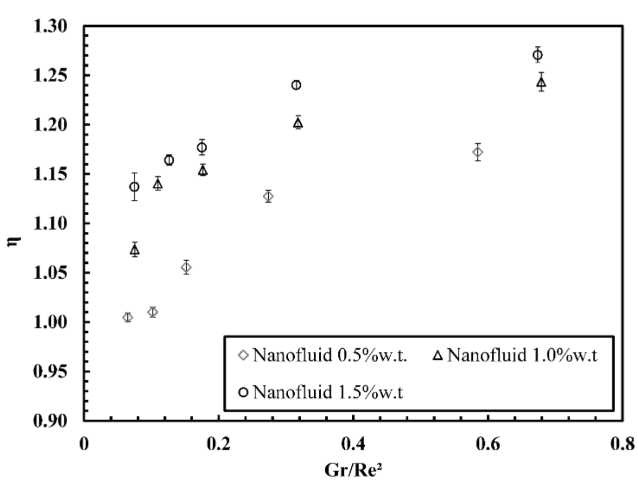

Fig. 13: The performance index of HTO-CuO mixed convection nanofluid flow in vertical tubes

IV. Conclusion

The effects of CuO nanoparticles on the mixed natural forced convection heat transfer rate and pressure drop of the buoyancy-aided heat transfer oil flow in vertical tubes were investigated experimentally. The result may be summarized as follows:

Adding nanopartilces enhanced the mixed convection heat transfer rate up to 50%.

Two new correlation was presented to predict the thermally developing mixed convection Nusselt number. As the maximum error of the correcletions is about 10%, they are reliable to estimate the heat transfer rate of the nanofluid flow with a good accuracy.

The maximum increment of the flow friction factor, due to adding nanopaticles, was about 20%. To estimate the Darcy friction factor, a new correlation was developed based on the experimental data which may predict the HTO-CuO flow behaviour in vertical tubes with the maximum error of 12%.

The system performance index was introduced to evaluate the effect of nanoparticles on the heat transfer rate and the pressure drop simultaneously. The majority of the results were larger than unity which indicates that using nanoparticles is more in favor of heat trasfer improvement rather than in pressure drop increment. The maximum performance index of 1.27 was obtained in the nanoparticle concentration of 1.5% and Richardson number of 0.7.

V. Acknowledgment

The authors would like to express their thanks to the Centre of Excellence in Design and Optimization of Energy Systems, School of Mechanical Engineering, College of Engineering, University of Tehran for the financial supports through the setup construction and research implementation.

VI. Appendix

The Kline and McClintock are defined by:

$$U_R = \left[\sum_{i=1}^{n}\left(\frac{\partial R}{\partial V_i} U_{V_i}\right)^2\right]^{1/2} \qquad (9)$$

where U_R is the overall uncertainty in the result, U_{V_i} is uncertainty in one variable, n is number of variable.

In order to compute the uncertainty of the friction factor, Nusselt number, and performance index, some independent variable error should be required to be where U_R is the overall uncertainty in the result, U_{V_i} is uncertainty in one variable, n is number of variable.

In order to compute the uncertainty of the friction factor, Nusselt number, and performance index, some independent variable error should be required to be calculated. The error of system is presented in Table 5. The error of thermos-physical and nanofluid flow characteristic used to calculate the Nusselt number are presented as it is followed:

$$U_\rho = \left[\left[\frac{1}{Q} U_m\right]^2 + \left[\frac{-m}{Q^2} U_Q\right]^2\right]^{1/2} \quad (10)$$

where U_ρ is density uncertainty in the result, U_m is weight uncertainty, U_V is volume uncertainty used for measuring density . The error of volumetric flow rate is obtain 11:

$$U_Q = \left[\left[\frac{1}{t} U_V\right]^2 + \left[\frac{-V}{V^2} U_t\right]^2\right]^{1/2} \quad (11)$$

Table 5: Uncertainty of system and equipment

Variables	items	Uncertainty
Diameter	U_D	±0.033 mm
length	U_L, U_x	±0.05 mm
Temperature of TC	U_{Ts},U_{Ti}	±0.035°C
Temperature of RTD	U_{Ts},U_{Ti}	±0.03°C
The uncertainty of flowmeter	U_{v2}	±1 ml
Specimen of weight used for density	U_m	±0.5 mgr
Specimen of Volume used for density	U_{v1}	±0.5 milt
Heat capacity	U_{cp}	±3%
Thermal conductivity	U_k	±2.5%
Dynamic viscosity	U_v	±3%
Pressure drop	$U_{\Delta p}$	±0.075%

where U_V is volumetric flow fluid uncertainty, U_{V_1} is volumetric flow fluid, and U_t is the duration of base fluid or nanofluid loading the flowmeter system. Mass flow uncertainty is calculated by:

$$U_{\dot m} = \left[\left[\rho\, U_{\dot Q}\right]^2 + \left[Q U_\rho\right]^2\right]^{1/2} \quad (12)$$

where $U_{\dot m}$ is mass flow, $U_{\dot V}$ is volumetric flow fluid, U_ρ is density uncertainty. The convection coefficient is calculated by:

$$U\overline{h_{nf}} = \pm\left[U^2_{Cp_{nf}} + U^2_{\dot m} + U^2_{(T_{b,O}-T_{b,i})} + U^2_D + U^2_L + U^2_{(T_w-T_b)}\right]^{1/2} \quad (13)$$

Where $U\overline{h_{nf}}$ is convection coefficient$U_{\dot m}$, U_D,U_L are mass flow , error of diameter, and length, respectively.

$$U_{\overline{Nu_{nf}}} = \pm\left[U\overline{h_{nf}}^2 + U^2_D + U^2_k\right] \quad (14)$$

where $U_{\overline{Nu_{nf}}}$ is average Nusselt, U_D, and U_k are diameter and conductivity error, respectively. Characteristics of nanofluid are calculated [39]. The characteristics of concentration of 0.5% which is perceived as specimen are presented in Table 6. In addition, the results of specimen uncertainties are reported in Table 7. Others uncertainties like friction factors error are calculated according the equation 9.

Table 6: The characteristics of nanofluid used for specimen

Items	Value
Volumetric flow (m^3/s)	1.15×10^{-4}
Tube surface temprature (°C)	98
Bulk temperature(°C)	49.65
Heat capacity(J/kg K)	1.71
Density(kg/m^3)	854.6
Duration of nanofluid loaded the flow rate measuring system(second)	8.7
Heat conductivity(w/m.k)	0.1497
Dynamic viscosity (Pa.s)	0.152
Pressure drop (Pa)	12220.2

Table 7: The result of specimen uncertainty

Items	Value
U_Q (uncertainty of volumetric flow rate)	1.15×10^{-4}
U_ρ (uncertainty of density)	0.023
$U_{\dot m}$ (uncertainty of mass flow)	0.0023
U_{Re} (uncertainty of Reynolds)	0.023
U_{pr} (uncertainty of Prandtle)	0.007
$U_{t_{b_i}}$ (uncertainty of inlet bulk tempearture)	0.0018
U_{t_w} (uncertainty of wall tempearture)	0.0096
$U_{t_{b_o}}$ (uncertainty of out let bulk temperature)	0.002
$U_{(t_{b_o}-t_{b_i})}$ (uncertainty of defrential between inlet and outlet temperature)	-0.00034
$U_{(T_w-T_b)}$ (uncertainty of defrential between wall and bulk temperature	-.00104
$U_{\overline{h_{nf}}}$ (uncertainty of mean convection heat transfer)	0.0293
$U_{\overline{Nu}}$ (uncertainty of Nusselt number)	0.0156
$U_{\Delta p}$ (uncertainty of pressure drop)	0.016
U_f (uncertainty of friction factor)	0.0267
U_η (uncertainty of performance index)	0.0578

VII. Nomenclature

Cp	specific heat capacity $(kJ/kg.K)$
D	tube diameter (m)
F	Darcy friction factor $(\pi^2\rho D^5\Delta p)/2L\dot m^2$
g	gravity (kg/m^2)
Gr	Grashof number $(\beta\Delta t D^3\rho^2 g/\mu^2)$
Gz	Graetz number $(Re\,Pr\,D/L)$
h	convection coefficient $(w/m^2.k)$
K	thermal conductivity $(W/m.k)$
L	tube length (m)
m	mass (kg)
$\dot m$	mass flow rate (kg/s)
Nu	Nusselt number $(\bar h/k)$
Q	volumetric flow rate (m^3/s)
Re	Reynolds number $(\rho u D/\mu)$
Ra	Rayleigh number $(GrPr)$
Ri	Richardson number (Gr/Re^2)
Sp. G.	Specific gravity-density relative to that of water at 4°C (m^3/kg)
T	temperature (K)
V_1	volume of flow rate measuring system (m^3)
ΔP	Pressure drop (Pa)
Greek symbols	

M Viscosity (Pa. s)
η Performance index
ρ density (kg/m3)

Subscripts

b_f **base fluid**
b,o bulk outlet
b,i bulk inlet
exp experimental
lam logarithmic
n_f nanofluid
w wall

References Références Referencias

1. Sider, E.N., and Tate, G.E., Heat transfer and pressure drop of liquids in tables, Industrial and Engineering Chemistry, 28(1) (1936), 1429-1435.
2. Eubank, C.C., and Proctor, W.S., Effect of natural convection on heat transfer with laminarflow in tubes, M.S. Thesis, Massachusetts Institute of Technology. Boston, 1951.
3. Joye, D.D., Pressure drop correlation for laminar, mixed convection, Aiding flow heat transfer in a vertical tube, International journal heat and Fluid Flow, Vol. 24, no. 2, pp. 260-266. 2003.
4. Huang, L., and Farrell, K.J., Mixed convection in vertical tube: high Reynolds number, Proc. 14th International Heat Transfer Conference, Washington, DC,Vol. 2, pp. 269-276, 2010.
5. Laskowski, G.M., Kearney, S.P., and Evans, G., and Greif, R., Mixed convection heat transfer to and from a horizontal cylinder in cross-flow with heating from below, International Journal of heat and Fluid Flow, Vol. 28, no. 3, pp. 454-468, 2007.
6. Hasadi, Y.M.F.E.I., Busedra, A.A., and Rusturn, I.M., Laminar mixed convection in the entrance region of horizontal semicrocular duct with the flat wall at the top, Journal of Heat Transfer, Vol. 129, no. 9, pp. 1203-1211, 2007.
7. Choudhury, D., and Pantakar, S.V., Combined forced and free laminar convection in the entrance region of an inclined isothermal tube, Applied Thermal Engineering., Vol. 26, no. 14-15, pp. 1677-1683, 2006.
8. Mare, T., Voicu, I., and Miriei, J., Experimental analysis of mixed convection in inclined tube, Applied Thermal Engineering, Vol. 26, no. 14-15., pp. 1677-1683, 2006.
9. Choi, S.U.S., and Eastman, J.A., Enhancing thermal conductivity of fluid with nanoparticles, Proc. International Mechanical Engineering Congress and Exhibition, San Francisco, California, Vol. 231, pp. 95-105, 1995.
10. Lee, S., Choi, S.U.S., Li, S., and Eastman, J.A., Measuring thermal conductivity of fluid containing oxide nanoparticles, Journal of Heat Transfer, Vol. 121, no. 2, pp. 280-289, 1999.
11. Fakoor Pakdaman, M., Akhavan-Behabadi, M.A., and Razi, P., An experimental investigation on thermos-physical properties and overall performance of MWCNT/ Heat Transfer Oil nanofluid flow inside vertical helically coiled tubes, Experimental Thermal Fluid and Science, Vol. 40, no. 2, pp. 103-111, 2012.
12. Akhavan- Behabadi, M.A. Hekmatipour, F., Mirhabibi, S.M., and Sajadi, B., Experimental investigation of thermal-rheological properties and heat transfer behavior of the heat transfer oil-copper oxide (HTO- CuO) nanofluid in smooth tubes, Experimental Thermal Fluid and Science, Vol. 68, no. 10, pp. 681-688, 2015.
13. Akhavan- Behabadi, M.A. Hekmatipour, F., Mirhabibi, S.M., and Sajadi, B., An empirical study on heat transfer and pressure drop properties of heat transfer oil-copper oxide nanofluid in microfin tube, International communication in Heat and Mass Transfer, Vol.57,pp. 150-156,2014.
14. Madhesh, D., and Parameshwaran, R., experimental studies on convective heat transfer and pressure drop characteristics of meta and metal oxide nanofluids under turbulent flow regime, Heat Transfer Engineering Vol. 37, no.5, pp. 422-434, 2016.
15. Amiri, A., Shanbedi, M., Yarmand, H., Arzani, H.K., Gharehkhani, S., Montazer, E., Sardri, R., Sarsam, W., Chew, B.T., and Kazi, S.N., laminar convective heat transfer of hexylamine-treated MWCNTs-Based turbine oil nanofluid, Energy Conversion and Management, Vol. 105, pp. 355-367, 2015.
16. Karami, M., Akhavan-Behabadi, M.A., Fakoor-Pakdaman, M., Heat transfer and pressure drop characteristics of nanofluid flows inside corrugated tubes, Heat Transfer Engineering, Vol. 37, no. 1, pp. 1-9, 2015.
17. Jafarimoghaddam, A., Aberoumand, S., Aberoumand, H., and Javaherdeh, K., Experimental study on CU/Oil nanofluids through concentric annular tube: a correlation, Heat Transfer-AsianResearch, DOI: 10.1002/htj.21210.
18. Abbaslan Arani, A.A., Aberoumand, H., Aberoumand, S., and Jafari Moghaddam, A., An empirical investigation on thermal characteristics and pressure drop of Ag-oil nanofluid in concentric annular tube, Heat Mass Transfer, Vol. 52, no.8, pp. 1693-706, 2016.
19. Zeinali Heris, S., Farzin, F., and Sardarabadi, H., Experimental comparison among thermal characteristics of three metal oxide nanoparticles/Turbine Oil –based nanofluids under laminar flow regime, International Journal of Thermophysics, Vol. 36, no.4, pp. 760-782, 2015.
20. Moghari, R.M., Talebi, F., Rafee, R., and Shariat, M., Numerical study of pressure drop and thermal chararcteristics of Al2O3-water nanofluid flow in

horizontal Alumuli, Heat Transfer Engineering, Vol. 36, no.2, pp. 166-177,2015.
21. Ghobadi, M., and Muzychka, Y.S., A review of heat transfer and pressure drop correlations for laminar flow in curved circular ducts, Heat Transfer Engineering, Vol. 37, no. 10, pp. 815-839, 2016.
22. CFD examination of convection heat transfer and pressure drop in a horizontal helically coiled Tube with CuO/oil base nanofluid, Numerical Heat Transfer, Part A, Vol. 66, no. , pp. 315-329, 2014.
23. Ghazvini, M., Akhavan-behabadi, M.A., Rasouli, E., Raisee, M., Heat transfer properties of nanodiamond-Engine oil nanofluid in laminar Flow, Heat Transfer Engineering, Vol. 33, no. 525-532, 2012.
24. Feng, Z.Z., and Li, W., Laminar mixed convection of large-Prandtl number in tube nanofluid flow. Part I: Experimental study, International Journal Heat and Mass Transfer, Vol. 65, no. 10, pp. 919-927, 2013.
25. Li, W., and Feng, Z.Z., Laminar mixed convection of large-Prandtl number in tube nanofluid flow. Part II: Experimental study, International Journal Heat and Mass Transfer, Vol. 65, no. 10, pp. 98-935, 2013.
26. Hekmatipour, F., Akhavan-Behabadi, M.A., and Sajadi, B., Combined free and forced convection heat transfer of the copper oxide- heat transfer oil (CuO-HTO) nanofluid inside horizontal tubes under constant wall temperature, Applied Thermal Engineering, Vol. 100, pp. 621-627, 2016.
27. Aberoumand, S., and Jafarimoghaddam, A., On the thermal characteristics of Ag/heat transfer oil nanoluids flow inside curved tubes, Applied Thermal Engineering, Vol. 108, pp. 967-979, 2016.
28. Ben Mansour, R., Galanis, N., and Nguyen, C.T., Experimental study of mixed convection with water-Al2O3 nanofluid in inclined tube with uniform wall heat flux, International Journal Thermal. Science, Vol. 50, no. 3, pp. 403-410, 2011.
29. Pirhayati, M., Akhavan-Behabadi, M.A., and khayat, M., 2014, Convection heat transfer of oil based nanofluid inside a circular tube, International Journal Engineering Transaction. B: Applied. Vol. 27, no. 2, pp. 341-348, 2014.
30. Derakhshan, M.M., Akhavan-Behabadi, M.A., and Mohseni, S.G., Experiments on mixed convection heat transfer and performance evaluation of MWCNT-Oil nanofluid in Horizontal and vertical microfin tubes, Experimental Thermal and Fluid Science, Vol. 61, no. 2, pp. 241-248, 2015.
31. Derakhshan, M.M., and Akhavan-Behabadi, M.A., Mixed convection of MWCNT-heat transfer oil nanofluid inclined plain and microfin tubes laminar assisted flow, International Journal of Thermal Science, Vol.99,pp. 1-8, 2016.
32. Derakhshan, M.M., Akhavan-Behabadi, M.A., and Ghazvini, M., Rheological characteristics, pressure drop, and skin friction coefficient of MWCNT-oil nanfluid flow inside an incline microfin tube, Heat Transfer Engineering, Vol. 36, no.17, pp. 1436-1446, 2015.
33. Akhavan-Behabadi, M.A., Hekmatipour, F., and Sajadi, B., An empirical study on the mixed convection transfer and pressure drop of HTO/CuO nanofluid inclined tube, Experimental Thermal and Fluid Science, Vol. 78, pp.10-17, 2016.
34. Ben Mansour, R., Galanis, N., and Nguyen, C.T., Developing laminar mixed convection of nanofluids in an inclined tube with uniform wall heat flux, International Journal Numerical Mathematics Heat and Fluid Flow, Vol. 19, no. 2, pp. 146-164, 2009.
35. Malvandi, A., and Ganji, D.D., Fully developed flow ana heat transfer of nanofluid inside a vertical annulas, Journal of the Brailian Society of Mechanical Science, Vol. 37, no.1, pp.141-147, 2015.
36. Mirmasoumi, S., and Behzadmeher, A., Effect of nanoparticle mean diameter on the particle migration and thermos-hydraulic behavior of laminar mixed convection of a nanofluid in an inclined tube, Heat Mass Transfer, Vol. 48, no. 8, pp. 1397-1308, 2012.
37. Izadi, M., Behzadmeher, A., and shahmardan, M., Effecti of inclination angle on laminar mixed convection of a nanofluid flowing through an annulus, Chemical Engineering community, Vol. 202, no. 12, pp. 1693-1702, 2015.
38. Vahidinia, F., and Miri, M., Numerical study of the effect of the Reynolds numbers on thermal and hydrodynamic parameter of turbulent flow mixed convection heat transfer in an inclined tube, Journal of Mechanical Engineering, Vol. 61, no.11, pp.669-679,2015.
39. Kline, S.J., McClintock, F.A., Describing uncertainties in single- sample experiments, Mechanical Engineering, Vol. 75, pp. 3-8, 1953.
40. White, F.M., Fluid Mechanic fourth ed. McGraw-Hill, New York. 2010.
41. Bergman, T.L., Lavine, A.S., Incorpera, F.P., and Dewitt, D.P., Fundamentals heat and mass transfer, Seventh ed. John Wiley and Scons, New Jersey. 2015.