Experimental Study of Mixed Convection Laminar Flow of Water-Al_2O_3 Nanofluid in Horizontal Tube with Uniform Wall Heat Flux

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Abstract: - In this work, an experimental investigation was carried out to study a laminar mixed convection flow and heat transfer of Al_2O_3-water nanofluid inside a horizontal tube submitted to a uniform wall heat flux at its outer surface. Measured data were collected for the following ranges of the governing parameters: the Reynolds number between 170 and 630, the Grashof number between 1.5 10^4 and 9.2 10^4 and the Prandtl number between 7 and 7.42. Results have shown that the experimental heat transfer coefficient remains nearly constant with an increase of particle volume concentration from 0 to 2%. However, we have observed a slight decrease of the Nusselt number with an increase of the particle volume fraction from 0 to 2%.

Key-Words: - Heat transfer, Laminar flow, Mixed convection, Natural and forced convection, Nanofluid, Al_2O_3-Water mixture, Alumina nanoparticles, Experimental study.

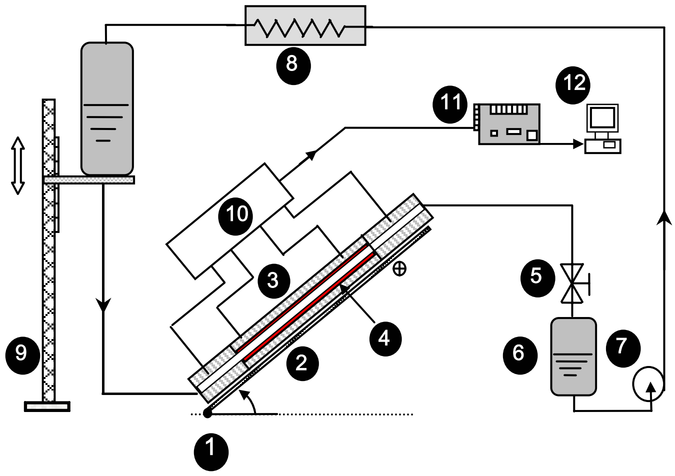
1 Introduction

A nanofluid is a two-phase mixture (solid-fluid) composed of a continue phase, generally a saturated liquid such as water, engine oil and ethylene glycol, and a dispersed phase constituted of nanometer-size particles (≤ 40nm). It has been shown that these mixtures generally possess thermal properties that are clearly improved while compared to the ones of the base 'classical' liquids. In fact, experimental works [1, 4] have indicated that a particle volume fraction as low as 1% of copper nanoparticles or carbon nanotubes dispersed within ethylene glycol can provide an increase of, respectively, 40% and 150% of the resulting mixture thermal conductivity. Hence, the nanofluides can constitute a very interesting alternative for heat transfer enhancement purposes in various thermal applications, especially for the cooling of electronic components and devices [1, 3, 7]. Several numerical works [5-7] and experimental investigations [8-10] were performed by researchers to study the hydrodynamic and thermal behaviors of nanofluids in some confined forced flow. It is worth noting that there is an important lack of data and knowledge regarding the combined forced and natural convection flow of a nanofluid; only some scarce studies [11-13] may be consulted.

In the present work, we have experimentally investigated the problem of the mixed laminar convection flow and heat transfer of a nanofluid inside a horizontal tube that is subjected to a uniform wall heat flux. The nanofluid is constituted of distilled water and Al_2O_3 nanoparticles of 36nm averaged-diameter. Some most significant data and results are presented and discussed in this paper.

2 Experimental Apparatus and Procedures

Figure 1 shows a schematic illustration of the experimental system used. The fluid, distilled water or a nanofluid, flows down from an upper constant level tank and enters in the heated section. The latter is a copper tube of inside diameter D=6.35 mm and total length of 2.225 m. The tube is submitted to a constant and uniform wall heat flux applied over a large portion of it, L2=200D. The heating was realised using a standard flat-ribbon-type electrical resistance (Omega, USA) of 2 m in length and 12 mm in width. This resistance is rolled on the tube outer surface at constant pitch. A precise potentiometer was used to adjust, if necessary, the output voltage supplied to the resistance. In order to be able to create appropriate boundary conditions for both the tube inlet and outlet, two adiabatic sections with respective lengths of L1=50D and L3=100D are installed, one preceding the heated section and the other one immediately after it. Hence, after traversing the first long adiabatic section served as a hydrodynamic developing section, the fluid would have a parabolic axial velocity profile at the entrance of the heated section. In order to reduce heat losses towards the surrounding environment, a thick layer (4 cm) of fiber-glass insulating blanket was wrapped around the tube-and-resistance system. After traversing the test tube, the fluid passes by a valve and finally reaches the second lower collecting reservoir.



1	Inclined support	2	Test section	3	Insulating material
4	Electrical Resistance	5	Valve	6	Reservoir
7	Pump	8	Exchanger	9	Manivelle support
	Thermocouples	11	Data acquisition	12	Computer

Fig. 1 Schematic of the experimental system used

Using a magnetic-driven centrifugal pump (MD-20RZ model from IWAKI), the fluid collected in the lower reservoir is then pumped upwards and traverses a tubular spiral-type heat exchanger where heat is released towards a constant temperature cold water source.

A desired flow rate of the liquid has been achieved by appropriately adjusting the height at which is fixed the upper reservoir. The fluid mass flow rate, which is measured using a classical and reliable ‘stop-watch-and-weighting’ technique, can be determined with a accuracy as good as ±2 %. The heating electrical power, indirectly deduced from the measured voltage and current supplied using a precise multi-meter, can be determined with an accuracy of ±2 %.

For the measurement of temperature at various places in the system, many thermocouples with an accuracy of ±0.1°C were installed:

- 18 surface-type thermocouples were clued on the tube outer surface using a high-temperature ciment-type compound, among them, 14 T-type thermocouples were placed along the heated section and 4 other ones of J-type were fixed in the two adiabatic sections. All these thermocouples serve to monitor wall temperatures along the tube.
- 3 other J-type thermocouples were fixed at different locations on the outer surface of the insulating layer. The measured temperatures were used to calculate the heat losses towards ambient air. Such heat losses remain very low and did not exceed, so far, 3 % of the total heat input.
- 2 other T-type thermocouples were used to measure fluid temperatures at the tube inlet and outlet. These thermocouples, made of stainless steel, have a very small diameter to minimise the obstruction effect to the flow.

All the thermocouples and sensors were connected to a data acquisition system composed of a SCXI-1102 module, a SCXI-1303 terminal and an acquisition board; all components were from *National Instrument*. The SCXI-1303 terminal possesses an internal thermistor for an automatic cold junction correction. The standard LabView software, also from *National Instrument*, was employed to perform the data acquisition and treatment.

The first set of tests was carried out using distilled water for a laminar ‘pure’ forced convection flow as well as for mixed convection flows. Figure 2 shows, in particular, a satisfactory comparison between wall and fluid bulk as obtained numerically [13] and experimental data collected for a mixed convection flow of water in an horizontal tube for a power supply of P=296.45 W ± 4 % and a mass flow rate of m’=1.8g/s±2%. The corresponding values of the Reynolds and Grashof numbers are Re= 363.38 and Gr=7.30 10^4. Results from this test as well as from other ones performed have permitted to assess that the experimental apparatus has performed quite satisfactorily.

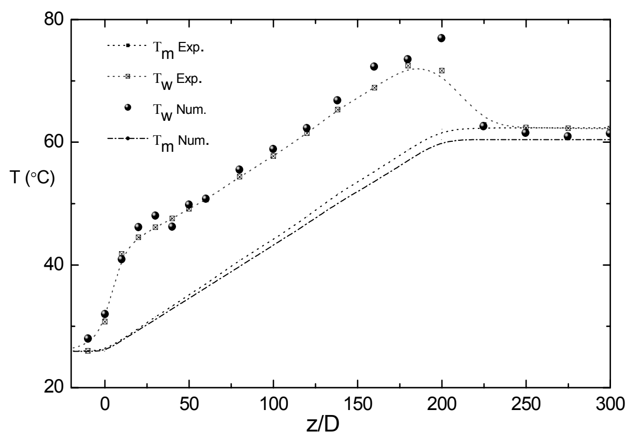


Figure 2. Axial development of wall and fluid bulk temperatures

The experiments have been carried out using Al_2O_3- - water mixture with particles of averaged diameter 36nm, and this for the following ranges of governing parameters: the Reynolds number between 170 and 630, the Grashof number varies from $1.5x10^4$ to $9.2x10^4$. In the following, some results and data are shown for the particular particle volume fraction of 2%.

3 Results and Discussion

3.1 Properties of Al_2O_3-Water Nanofluid

Using classical formulas derived for a two-phase mixture and some experimental data [5, 8, 14], the thermal and physical properties of the nanofluid under consideration can be computed using equations (1-5). It is worth noting that the subscripts 'nf', 'bf', 'p' refer, respectively, to nanofluid, base fluid and particles; ρ, C_p, β, μ and k are fluid density, specific heat, thermal expansion coefficient, dynamic viscosity and thermal conductivity, respectively; φ is the particle volume fraction; all fluid properties are computed at the fluid inlet temperature (indicated by the subscript '0').

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_p \qquad (1)$$

$$(\rho C_p)_{nf} = (1-\varphi)(\rho C_p)_{bf} + \varphi(\rho C)_p \qquad (2)$$

$$(\rho\beta)_{nf} = (1-\varphi)(\rho\beta)_{bf} + \varphi(\rho\beta)_p \qquad (3)$$

$$\mu_{nf} = \mu_{bf}(123\varphi^2 + 7.3\varphi + 1) \qquad (4)$$

$$\frac{k_{nf}}{k_{bf}} = \frac{k_p + 2k_{bf} - 2\varphi\left(k_{bf} - k_p\right)}{k_p + 2k_{bf} + \varphi\left(k_{bf} - k_p\right)} \qquad (5)$$

For the presentation of results, the following dimensionless governing parameters, namely the Reynolds, Prandtl and Grashof numbers, are introduced:

$$Re = \rho\, u_0 D/\mu \qquad (6)$$

$$Pr = C_p\mu/k \qquad (7)$$

$$Gr = \rho^2 g\beta q_{net} D^4/\mu^2 \qquad (8)$$

where u_0 is the fluid average axial velocity at the tube inlet and q_{net} is the net heat flux. The convective heat transfer coefficient and the corresponding Nusselt number are given as follows:

$$h(z) = \frac{q_{net}}{T_w(z) - T_f(z)} = \frac{q_{imput} - q_{loss}}{T_w(z) - T_f(z)} = \frac{UI - q_{loss}}{T_w(z) - T_f(z)} \qquad (9)$$

$$Nu(z) = \frac{h(z)D}{k} \qquad (10)$$

where U and I are the current voltage and intensity, q_{input} and q_{loss} are, respectively, the heat flux input and heat losses towards the ambient air, T_w and T_f are the wall and fluid temperature and z represents a local axial coordinate from the entrance of the test section. The fluid temperature profile in the test section was obtained through the energy balance:

$$T_f(z) = T_{in} + \frac{q_{net}\ \pi DL(z)}{m'\ Cp} \qquad (11)$$

It is interesting to mention that for all of the tests performed in this study, the differences between the measured temperature differences (T_{out}-T_{in}) and the corresponding values as computed by using Eq. 11 did not exceed, so far, 5%, results that is very acceptable in conjunction with the experimental uncertainties.

3.2 Presentation of Data

Figure 3 shows the axial development of the circumferentially interfacial wall temperature, T_w, and the fluid bulk temperature, T_b, for two particle volume concentrations and a net heat flux of q_{net}=11502 W/m^2 and a mass flow of m'=1.8 g/s. One can observe that the wall temperature increases rapidly up to the location z/D=127 because of the development of the thermal field. Beyond this position, the fully developed conditions are reached and all temperatures increase linearly with the axial coordinate. At a short distance from the end of the heated section, both temperatures T_w and T_b tend to the same value that is dependent on the overall heat balance. Results in Fig. 3 also show that the wall temperature remains nearly constant between the case of water and the one with 2% of particle volume fraction, in particular in the end of the heated section. However, the fluid temperature has been found to increase slightly with φ =2%. Thus, the difference between the fluid temperature and the wall temperature, T_w-T_b, decreases slightly with an increase of φ, which indicates that there is a slight increase of the heat transfer coefficient, although such an increase appears not much different from the experimental error (Fig. 4).

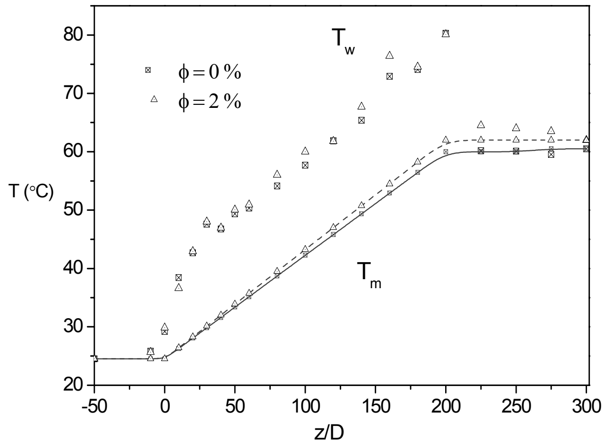


Figure 3. Axial development of temperatures for m' =1.8 g/s and q_{net}=11502 W/m^2

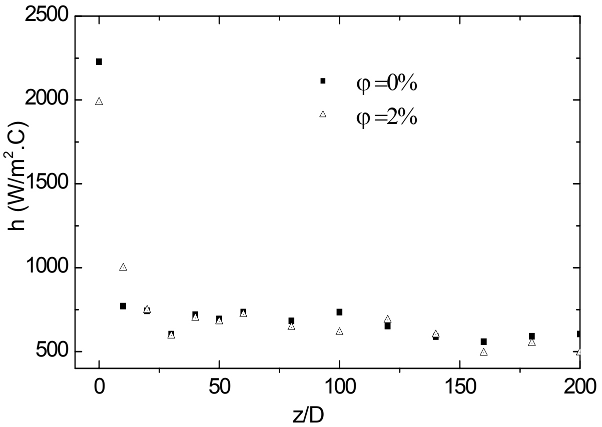


Figure 4. Axial development of local heat transfer coefficient for m' =1.8 g/s and q_{net}=11502 W/m^2

It is worth noting that the 'nanofluid/water' thermal conductivity ratio is about 1.06 for the case of 2% particle concentration, and the enhancement of the local heat transfer coefficient is estimated less than 6%. From Fig. 5, it is observed that the addition of Al_2O_3 nanoparticles into water decreases slightly the Nusselt number. Such a results may appear somewhat paradoxical, but it should be recalled that the Nusselt definition also include the fluid thermal conductivity.

Figure 6 shows the results for the asymptotic Nusselt number as function of the Grashof number, for cases with the Reynolds number fixed at Re= 440 and two different particle volume fractions (φ =0 and 2%). One can observe that the Nusselt number considerably increases with an increase of the Grashof number. Thus, for φ =2% in particular, the Nusselt number increases by nearly 11% when Gr increases from 4 10^4 to 6.4 10^4. Moreover, these results clearly show that the use of nanofluids decreases slightly natural convective heat transfer coefficient as compared to that of pure water. Thus, we can observe that the asymptotic Nusselt number has decreased from 6.96 to nearly 6.55 for φ increasing from 0% to 2% for Gr ≈ 6 10^4. It is very interesting to mention that these results are qualitatively consistent with the experimental data by Putra et al. [15] and Wen and Ding [16] both studied the natural convection of nanofluids inside various configurations.

It is worth mentioning that there are some contradictory observations regarding the heat transfer behaviour of nanofluids under the natural convection effect. There is a disagreement between analytical and experimental investigations. From some numerical studies (see for example, Khanafer et al. [11] and Ben Mansour et al. [13]), the natural convective heat transfer of nanofluids increases as the particle concentration increases, behaviour that is opposite to the experimental results [15, 16]. Several possible reasons can explain this paradoxical result. Under the laminar flow regime, say for Re less than 500 according to some experimental studies, the random movement of the particles, the one that is responsible of an increase of the energy exchange process within the fluid, remains small. The presence of nanoparticles inside the base fluid tends to increase the mixture apparent viscosity, which may suppress the natural convection effect. Also, it has been believed that the particle-surface interactions may be a major factor that can contribute to a deterioration of the natural convective heat transfer. The above contradictory results regarding the influence of natural convection within a nanofluid show obviously that the urgent need for more research works in this area is indeed present.

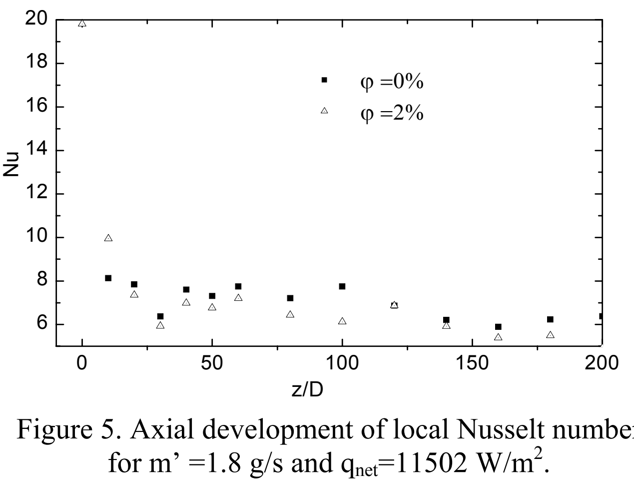


Figure 5. Axial development of local Nusselt number for m' =1.8 g/s and q_{net}=11502 W/m^2.

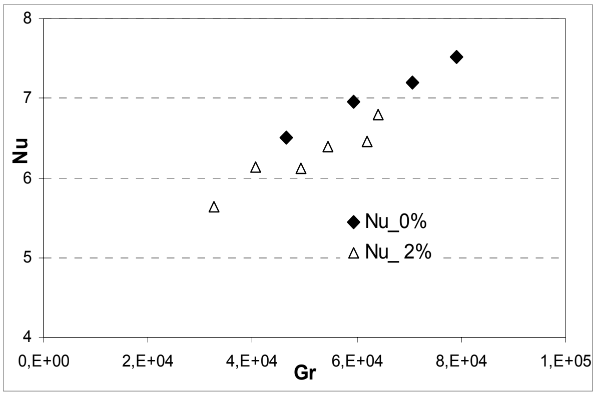


Figure 6. Variation of Nu function of Gr

4. Conclusion

In this work, the problem of the mixed convection laminar flow of Al_2O_3-water nanofluid with particle-size of 36nm flowing inside a horizontal tube has been investigated experimentally. The tube is subjected to a constant and uniform wall heat flux at its outer surface. Experimental results have shown that for a relatively low particle volume fraction, 2%, the use of the nanofluid did not appear to enhance much the heat transfer coefficient. More analytical and experimental data are needed to understand the physical phenomenon and to assess the true potential of nanofluids in thermal applications involved with buoyancy effects.

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