EFFECT OF USING VISCOSITY AND THERMAL CONDUCTIVITY MODELS ON EXPERIMENTAL CAVITY FLOW NATURAL CONVECTION OF CUO-WATER NANOFLUIDS

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ABSTRACT

Publication on cavity flow natural convection by using nanofluids has increased in recent years. On the other hand, contrary results offered by different researchers, both in experimental and numerical works. In this research it is tried to indicate that the accuracy of the viscosity and the thermal conductivity of the nanofluids is the most important reason of the contrary results. Therefore, cavity flow natural convection of CuO-water nanofluids has considered experimentally for volume fractions 0.5% and 1% in this research. The results show that the natural convection of nanofluids are more sensitive on accuracy of the viscosity than the thermal conductivity as well as it recommends to measure the viscosity and thermal conductivity in experimental natural convection works. However, for the range of the volume fractions tested in this research, CuO-water nanofluids (30-50nm) have not shown heat transfer advantage.

NOMENCLATURE

A	[m^2]	Surface area
c_p	[J/kg.K]	Specific heat capacity
E	[%]	Energy balance
g	[m/s^2]	Gravitational acceleration
h	[m]	Vertical height of the cavity
k	[W/m.K]	Thermal conductivity
L	[m]	Length of the cavity
$\dot{m}$	[kg/s]	Mass flow rate
Nu	[-]	Nusselt number
Pr	[-]	Prandtl number
$\dot{Q}$	[W]	Heat transfer rate
Ra	[-]	Rayleigh number
T	[K ,ºC]	Temperature
w	[m]	Horizontal width of the cavity

Special characters

β	[1/K]	Volume expansion coefficient
μ	[kg/m.s]	Dynamic viscosity
ρ	[kg/m^3]	Density
ϕ	[-]	Nanoparticles volume fraction

Subscripts

1,2	Channel one and two through the heat exchanger (wall)
bf	Base fluid
c	Cold fluid channel side
char	Characteristic
exp	Experimental
h	Hot fluid channel side
i	Inlet
nf	Nanofluid
nl	Nanolayer
np	Nanoparticle
o	Outlet

INTRODUCTION

Nanofluids which are new heat transfer fluids have attracted the attention of researchers in heat transfer area. They have enhanced thermophysical properties such as thermal conductivity and heat capacity to those of base fluids like oil, water or glycol [1]. Nanofluids have the potential to be involved into many applications including heat transfer, automotive, electronic cooling and biomedical applications.

Many reviews have recently been published on different features of nanofluids [2-3]. Wong and De Leon [4] presents a broad range of current and future applications that involve nanofluids, emphasizing their improved heat transfer properties that are controllable and the specific characteristics that these nanofluids possess that make them suitable for such applications. Wang and Fan [5] take heat conduction nanofluids as examples to review methodologies available to effectively tackle these key but difficult problems. They reviewed techniques include nanofluid synthesis through liquid-phase chemical reactions in continuous-flow microfluidic micro-reactors. Mohammed et al. [6] worked on convective heat transfer on internal separated flows of nanofluids. The heat transfer enhancement along with the nanofluid preparation technique, base fluids and additives, stability of the suspension, types and shape of nanoparticles, and transport mechanisms were also discussed. According to Murshed et al. [7], researchers have mostly focused on the inconsistent thermal conductivity of nanofluids. Although investigations on convective heat transfer, droplet spreading, and boiling are very important in order to exploit nanofluids as the next generation coolants, considerable less efforts have been made on these major features of nanofluids. Therefore, they reviewed these features together with an exhaustive review of research and development made in these areas of nanofluids. Sarkar [8] summarizes the correlations developed for fluid flow and heat transfer characteristics of nanofluids in forced and free convection. It is concluded that a large deviation of predicted values for proposed equations has been observed and that it may be to strong influence of particle properties and nanofluid composition on flow and heat transfer characteristics, lack of common understanding on basic mechanism of nanofluid flow and insufficient experimental data on nanofluid heat transfer.

Cavity flow [9-17] is one of the methods of benchmarking the natural convection performance of one heat transfer working fluid against another, since it tests the ratio of the buoyancy-driven currents of the fluid against the viscous friction of the fluid. Assimacopoulis et al. [9] used logarithmic wall functions to mathematically model heat transfer in a cavity for a wide range of Rayleigh numbers. They used uniform and non-uniform grids and air as the working fluid. The results were in good agreement with numerical attempts although the wall functions overestimated heat transfer with the limitations of the k-ε turbulence model cited as the cause. The mathematical modelling of natural convection heat transfer in a square cavity with nanofluids was later modelled by Khanafer et al. [10] using a particle dispersion technique for a range of Rayleigh numbers. Good agreement with experimental work in literature was found which verified the method of modelling. Heat transfer was predicted to increase as the solid volume fraction of the nanoparticles increased.

Chen et al. [11] recognised contradictory results in literature and proposed to quantify the effects on heat transfer in laminar flow by modelling nanofluids as Newtonian fluids in a square cavity, but using two different thermal conductivity and two different dynamic viscosity models. The results revealed that, depending on the combination of models used, heat transfer can be predicted to increase or decrease. This emphasises the need for rigorous experimental investigations into nanofluid viscosity and thermal conductivity.

Nanofluid cavity flow has been analysed theoretically by various authors [12-16], but there is very limited experimentally works (i.e. Ho et al. [17]), that has been conducted. Therefore, the purpose of this paper is to find the effect of using thermal conductivity and viscosity models in cavity flow natural convection using CuO-water nanofluid at two different volume concentrations experimentally.

EXPERIMENT AND RESULTS

To find the effect of viscosity and thermal conductivity models on experimental natural convection calculations, an experimental work conducted for CuO-water nanofluids. The nanofluids consist of 0.5% and 1% volume fractions. Figure 1 shows the cavity which includes two differentially heated walls and the other walls insulated. Constant hot water and cold water was used for heating and cooling of two walls (heat exchangers). The hot water was pumped from a 600 litre hot water storage tank which was heated with an electrical resistance element and thermostatically controlled to a constant temperature up to 58ºC ±1ºC. The cold water was also pumped from another 600 litre storage tank which was connected to a chiller and thermostatically controlled to 25ºC ± 1ºC. It was possible to control the pumps with frequency controllers so that the mass flow rates through the cavity walls could be controlled electronically. In each pipeline the mass flow rates were measured with a bank of three Coriolis mass flow metres. The selection of the flow metre range was dependant on the mass flow rate during experiments. Figure 2 shows the set-up schematically.

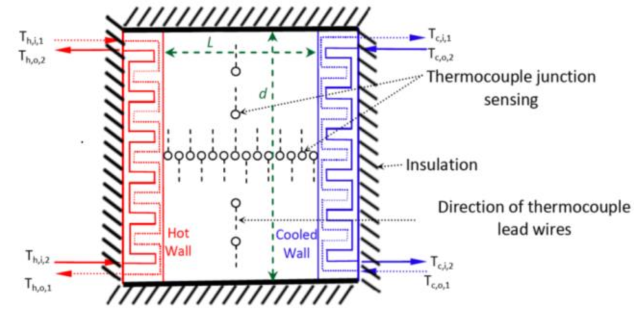

Figure 1 The cavity

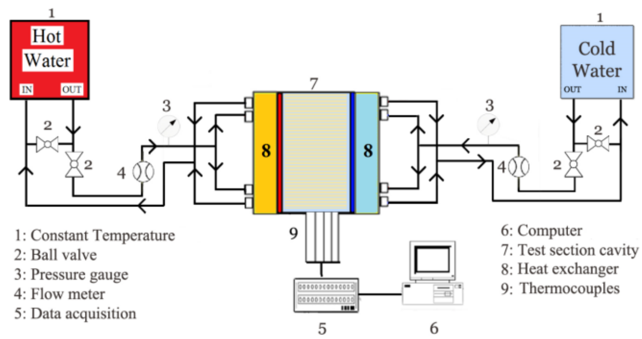

Figure 2 The schematic diagram of the set-up

To ensure uniform wall temperatures, the water was circulated in each wall (heat exchanger) through two channels in a counter flow direction. Each wall was 10 mm thick and was made from stainless steel. The counter flow and thick high thermal conductivity plates for the two walls ensured a constant wall temperature for each plate.

The cavity experimental set-up and pipelines from the storage tanks to the cavity test section were well insulated to prevent heat transfer to the environment. The most important dimensions of the cavity were height of heat exchanges, h =100 mm, the length between to heat exchanges, L =100 mm, and width of heat exchanger, w = 150 mm. The width to height of the heat exchangers which provided constant temperature walls designed to be w/h = 1.5, to eliminate sidewall boundary effects on the cavity flow at the mid-plan where the source-to-sink temperature profiles were measured, as illustrated in Fig. 1.

Temperature measurements were made with 0.1 mm, T-type thermocouples. Thermocouples were placed on the centre of each inner wall. A further 11 thermocouples were placed in the cavity at equally spaced points from the centre of the hot wall to the centre of the cold wall. Four thermocouples were inserted at mid-plain parallel to the hot and cold walls as shown in Fig 1. This is to as far as possible ensure symmetrical physical conditions.

The cavity was well insulated with 50 mm of insulation material with a low thermal conductivity which was covered with six PVC plates of 8 mm thickness. Thermocouples were connected to the centre of each of the plates and the temperatures were measured and compared to ambient temperatures. Elementary, one dimensional, heat transfer rate calculations were conducted to estimate the heat transfer rate from each one of the six sides to the environment.

The test cavity was a sealed cavity and was filled completely with water first and then nanofluids. To ensure that there was no air trapped inside the test cavity, the cavity was slightly overfilled before being sealed.

The nanoparticles used were CuO nanoparticles with average diameters from 30-50 nm. It was mixed with distilled water (the base fluid) and experiments were conducted at two different volume concentrations of 0.5% and 1.0%. To form a homogeneous suspension the nanoparticles were mixed with the water using an ultrasound mixer for nine hours [18]. The resultant nanofluids were as dark as could not be possible to measure the Zeta potential or UV-spectrophotometer, but visually stabile during experiment (in this on-going research, the investigation into the stability from other ways is in progress). For this (on-going) investigation it was assumed that the properties of the CuO nanoparticles remained constant with a density of 6 310 kg/m3, specific heat capacity at constant pressure of 540 J/kg.K, and thermal conductivity of 20 W/m.K.

The experimental heat transfer rate in the cavity was determined from the net heat transfer rates of the hot wall, $\dot{Q}_h$ and the cold wall, $\dot{Q}_c$ where

$$\dot{Q}_h = \dot{m}_{h,1}C_{ph,1}\left(T_{h,i,1} - T_{h,o,1}\right) + \dot{m}_{h,2}C_{ph,2}\left(T_{h,i,2} - T_{h,o,2}\right) \quad (1)$$

$$\dot{Q}_c = \dot{m}_{c,1}C_{pc,1}\left(T_{c,o,1} - T_{c,i,1}\right) + \dot{m}_{c,2}C_{pc,2}\left(T_{c,o,2} - T_{c,i,2}\right) \quad (2)$$

Therefore, the experimental heat transfer rate was determined as the average of the heat transfer rates of the hot and cold walls

$$\dot{Q}_{\exp} = \frac{\dot{Q}_h + \dot{Q}_c}{2} \quad (3)$$

and the energy balance error, E, for a measurement was determined as

$$E = \left|\frac{\dot{Q}_{\exp} - \dot{Q}_h}{\dot{Q}_{\exp}}\right| = \left|\frac{\dot{Q}_{\exp} - \dot{Q}_c}{\dot{Q}_{\exp}}\right| \quad (4)$$

The theoretical heat transfer rate was modelled using Eq. (5) [19] as

$$\dot{Q}_{theor} = \frac{kNuA\left(T_H - T_C\right)}{L_{char}} \quad (5)$$

The thermal conductivity values at different bulk temperatures (average between the hot wall and the cold wall) for water were obtained from the thermodynamic tables of Sonntag and Borgnakke [20]. For the nanofluids thermal conductivity at different concentrations three different theoretical models were considered and compared. The models were Maxwell [21], Xie et al. [22] and Yu and Choi [21]. With the Xie et al. model, two different nanolayer thicknesses of 0.5 nm and 1.0 nm were considered. With the Yu and Choi model, the same two nanolayer thicknesses were used as well as two different nanolayer thermal conductivities of firstly equal to that of the base fluid (knl=kbf) and then 10 times higher than the base fluid (knl=10kbf). The comparisons are given in Figure 3 for different volume fractions, nanolayer thicknesses and nanolayer thermal conductivities. The results show that for low volumetric concentrations up to 1% all three models correspond very well with all errors between 2.7% and 3%. Therefore, the Maxwell model was used for simplicity reasons.

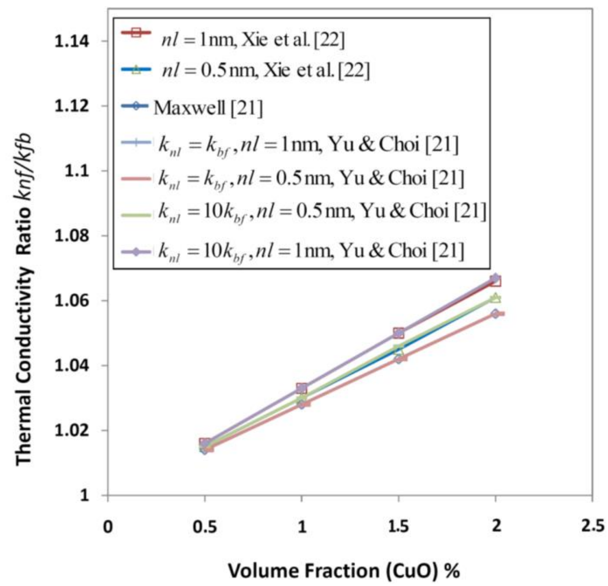

Figure 3 Comparison of effective thermal conductivity models for CuO-water nanofluid

The heat transfer area used in Eq. (5) was A = h.w, and the characteristic length used was h=L=Lchar.

The Rayleigh number based on the same characteristic length of the cavity was determined as

$$Ra = \left(\frac{g\beta\left(T_H - T_C\right)\rho^2 L_{char}^3}{\mu^2}\right)\Pr \quad (6)$$

where, the Prandtl number, Pr = cpμ/k and the properties (cp, μ and k) were obtained at the bulk fluid temperature. For water it was obtained from the thermodynamic tables of Sonntag and Borgnakke [20]. For the nanofluids as different concentrations, the specific heat and densities were obtained from the following equations:

$$c_{p,nf} = \phi c_{p,np} + (1-\phi) c_{p,bf} \quad (7)$$

and

$$\rho_{nf} = \phi \rho_{np} + (1-\phi)\rho_{bf} \quad (8)$$

The viscosities at different concentrations were obtained from two different models namely Maïga [11, 23] and Brinkman [10-12]. The thermal conductivities were obtained from the Maxwell model [21], for the reasons as previously discussed. In this paper, the viscosity and thermal conductivity has chosen from literature (however, for up to 1% volume fraction, the thermal conductivity has not a lot difference, but viscosity) to show how they can effect on the results.

A full experimental uncertainty analysis was performed on the experimental system and data analysis by the method suggested by Kline and McClintock [24]. The uncertainties for the heat transfer rate and Rayleigh numbers were maximum 6.2% and 7.6%, respectively. The energy balance errors (Eq. 4) for all experiments were less than 2%.

Fig. 4 shows the result of experimental heat transfer of Eq. (2) and theoretical heat transfer of Eq. (5) versus Rayleigh number Eq. (6). It is clear that by using different model for viscosity, there will be found different Ra by order of 5 when volume fraction of the nanoparticles is 1%. This shows that the natural convection of nanofluids are more sensitive on accuracy of the viscosity than the thermal conductivity. The experimental heat transfer for both of the 0.5% and 1% volume fraction were recorded less than the base fluid which shows there is no advantage of natural convection using CuO-water nanofluids for this rang of volume fractions. Therefore, the volume fraction less than 0.5% needs to be investigated.

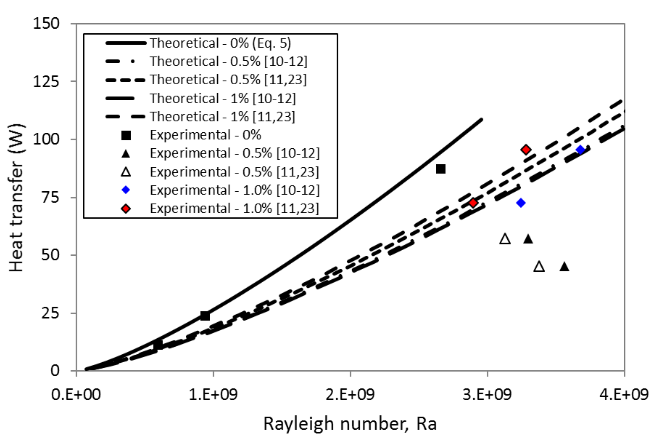

Figure 4 Heat transfer for the cavity versus Rayleigh number

On the other hand Maripia et al [25] showed that the effect of using different models for thermal conductivity and viscosity can bring the uncertainty for the results by using ANSYS-FLUENT. Moreover, Sharifpur et al. [26] presented the parametric analyses of thermal conductivity models and Meyer et al. [27] for viscosity models of nanofluids which show more work on accurate and hybrid models will be needed.

CONCLUSION

Natural convection heat transfer using CuO-water nanofluids has conducted in a squire cavity experimentally. The aspect ratio of the cavity was one and the nanofluids concentrations were 0.5 and 1%. The heat transfer rate showed the CuO-water nanofluids cannot bring advantages on heat transfer for this range of volume fraction. Using different model for viscosity and thermal conductivity of the nanofluids produced different Rayleigh number and different theoretical heat transfer rate. Therefore, it can recommend that for experimental heat transfer analyses of nanofluids the viscosity and thermal conductivity must be measured experimentally.

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