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Assessment of thermal conductivity, viscosity and specific heat of nanofluids in single phase laminar internal forced convection

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ABSTRACT

Nanofluids are considered for improving the heat exchange in forced convective flow. In literature, the benefit of nanofluids compared to the corresponding base fluid is represented by several figures-of-merit in which the heat transfer benefit and the cost of pumping the fluid are considered. These figures-of-merit do not account for the limited heat capacity of the fluid, the geometry and the flow conditions that in addition affect the temperature of the heat-transfer surface. We address these limitations by deriving a figure-of-merit that includes all the relevant thermophysical properties of the fluids, the flow conditions and the geometry of the heat-transfer device. This figure-of-merit is a useful benchmark tool to evaluate nanofluids in a laminar flow regime for heat sink applications.

1. Introduction

Nanofluids are suspensions of solid particles of nanometer and submicron sizes in a base fluid. Adding nanoparticles to a base fluid is supposed to enhance the heat-transfer properties of the fluid as reported in several reviews [1–4]. Addition of these particles increases the viscosity [5] and density and decreases the specific heat of the fluid. Specific heat and density of the nanofluid depend only on the concentration of the particles in the fluid. Thermal conductivity and viscosity in addition depend on the nanoparticle morphology (size, shape, etc) and the pH of the solution [6]. It is observed that the nanofluid heat transfer coefficient and pressure drop in laminar flow conditions are in good agreement with traditional convective models [7], provided the nanofluid thermophysical properties are utilized in the evaluation of the dimensionless numbers. Many studies on laminar heat transfer claim heat transfer enhancement in nanofluids when compared to base fluids at the same Reynolds numbers. This comparison gives an unfair and unrealistic advantage to nanofluids because at equal Reynolds number, the velocity of the nanofluid is higher compared to the base fluid to compensate for the higher nanofluid viscosity. A more realistic comparison of fluid heat transfer performance is at constant pumping power. Yu et.al. [8] measured thermal performance of silicon carbide nanofluid in a circular tube and observed augmentation of the heat transfer coefficient when measured at the same Reynolds number. However, at equal velocities, the nanofluid had a lower heat transfer coefficient. When compared at the same pumping power, the performance of the nanofluid was a factor 0.4 lower than the base fluid. So it is important to develop a reliable evaluation procedure in assessing the heat transfer enhancement of nanofluids including all relevant parameters.

Recent reviews on nanofluids in actual cooling applications [9,10] highlight the need for a holistic approach combining all relevant properties in the comparison of nanofluids and their corresponding base fluids. In a typical heat sink application, the trade-off between the increase in thermal conductivity and the viscosity of the nanofluid compared to the base fluid is measured by comparing the COP, the ratio of the heat absorbed by the fluid to the pumping power. In an earlier work Prasher et al. [11] suggested that in a fully developed laminar flow regime, the nanofluid will have a higher COP compared to the base fluid when the relative increase in viscosity of the nanofluid is less than a factor of four times the relative increase in thermal conductivity. However, Escher et al. [12] performed flow experiments with nanofluids in micro heat sinks and have not seen any performance augmentation although the viscosity increase of their nanofluids is less than four times the relative increase in thermal conductivity. They further suggest the need to include specific heat of the fluid when comparing nanofluids to their base fluids. Comparison based on the COP of a fluid does not take into account the maximum temperature of the heat sink, which is often a performance parameter in the design of the heat sinks. Garg et al. [13] proposed a figure-of-merit in which the maximum temperature of the heat sink is fixed, and at equal velocities of nanofluid and basefluid, related the required pumping

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power to the heat that is removed. In their analysis, the viscosity could increase by a factor of five times the relative increase in thermal conductivity. Unfortunately, their figure-of-merit is not correct since they consider constant heat flux and an increase of the wall temperature whereas the fluid temperature change is considered to be negligible. These are conflicting conditions.

Bergman [14] considered the effect of reduced specific heat of the nanofluids on single phase, laminar internal forced convection. For a flow in a tube with a constant heat flux boundary condition, the increased thermal conductivity of the nanofluid reduces the temperature difference between the heat transfer surface and the fluid. But the decrease in specific heat will increase the temperature of the fluid. Because of these two competing effects, the net benefit of the nanofluid to reduce the surface temperature depends on the dimensions of the flow channel, the thermal conductivity and the specific heat of the fluid. However, viscosity increase of the nanofluid is not taken into account while benchmarking the nanofluids. In a different study, Bergman [15] considered the effect of viscosity in turbulent flow conditions to evaluate nanofluids in heat sink application. This paper presents an analysis that includes the effects of viscosity, thermal conductivity and specific heat of the fluids on the nanofluid performance in a heat sink application with laminar flow conditions. In doing so, it will be shown that for particular channel dimensions and constant pumping power and heat input, nanofluids can lead to augmentation or degradation of overall thermal performance of the heat sink.

2. Analysis

In this section the thermal performance of a nanofluid compared to the base fluid is derived taking into consideration the following parameters:

- Geometry: circular tube with internal diameter D, and length l.
- Flow: mean flow velocity v.
- Thermophysical: density ρ, specific heat capacity c_p, thermal conductivity λ, viscosity μ.

Considering a fully developed laminar flow in a tube with a constant heat flux boundary condition, $\dot{Q}/\pi DL$, the axial wall temperature is determined from Newtons law of cooling, incorporating the definition of the Nusselt number, and assuming constant properties. This is

$$T_w(x) = T_f(x) + \frac{D\dot{Q}}{Nu\lambda\pi Dl}; \quad \frac{\dot{Q}}{\pi Dl} = \frac{Nu\lambda}{D}[T_w(x) - T_f(x)]. \tag{1a, b}$$

The axial mean temperature of the liquid T_f is determined from the energy balance and is given by

$$T_f(L) = T_f(0) + \frac{Q}{\rho v A c_p}; \quad \dot{Q}\frac{x}{l} = \rho v A c_p[T_f(x) - T_f(0)], \tag{2a, b}$$

where A is the tube cross-sectional area. The fluid temperature increases in the flow direction along the length of the tube. For a constant heat-flux boundary condition, the wall temperature also increases along the length of the tube and is largest at the end of the tube. In heat sink applications, the objective is to minimize the highest temperature of the sink. Hence, we consider the case when $x = l$.

A useful figure-of-merit for a heat sink is the ratio of the difference between the maximum wall temperature and the fluid temperature at the tube inlet to the heat transfer rate,

$$R = \frac{T_w(L) - T_f(0)}{\dot{Q}/\pi Dl} = \frac{4l/D}{\rho v c_p} + \frac{D}{Nu\lambda} = \frac{4l/D}{cv} + \frac{D}{Nu\lambda}. \tag{3}$$

Combining equations of the wall temperature and the fluid derived above yields the right hand side of the expression. The volumetric heat capacity c used in this expression is the product of ρc_p. This expression is also used by several authors to evaluate the thermal performance of nanofluids [12,14]. Two fluids having different thermophysical properties can be compared by evaluating R for each fluid. The fluid having lower R maintains a lower temperature of the heat sink for the same heat flow and velocity of the liquid. Fig. 1 shows the values of R for water and titanium oxide seeded nanofluid. The Nusselt number used in this case is 4.36. The thermal conductivity ratio of the nanofluid to water is 1.04, and the viscosity ratio is 1.8. The diameter and the length of the tube is 4 mm and 0.5 m respectively. For a flow velocity of 0.50 m/s, the value of R with titanium oxide nanofluid is 0.00174 K/(W/m^2) and a pumping power of 5.5 mW and with water R is 0.00177 K/(W/m^2) a pumping power of 3.1 mW. To obtain the same value of R the velocity of the flow with water should be increased to 0.58 m/s. The pumping power in this case is 4.1 mW, which is still lower than that with the nanofluid at a velocity of 0.50 m/s. Water performs better than the nanofluid in the same setup because of lower viscosity.

Considering a fixed geometry of the heat sink the number of independent variables are (i) the velocity of the flow v, (ii) the thermal conductivity of the liquid λ and (iii) the volumentric specific heat of the fluid c. To assess the affect of these variables on the thermal resistance, the difference ΔR is considered between nanofluid and the basefluid

$$\Delta R = \frac{4l/D}{c_{nf}v_{nf}} + \frac{D}{Nu\lambda_{nf}} - \frac{4l/D}{c_{bf}v_{bf}} - \frac{D}{Nu\lambda_{bf}}. \tag{4}$$

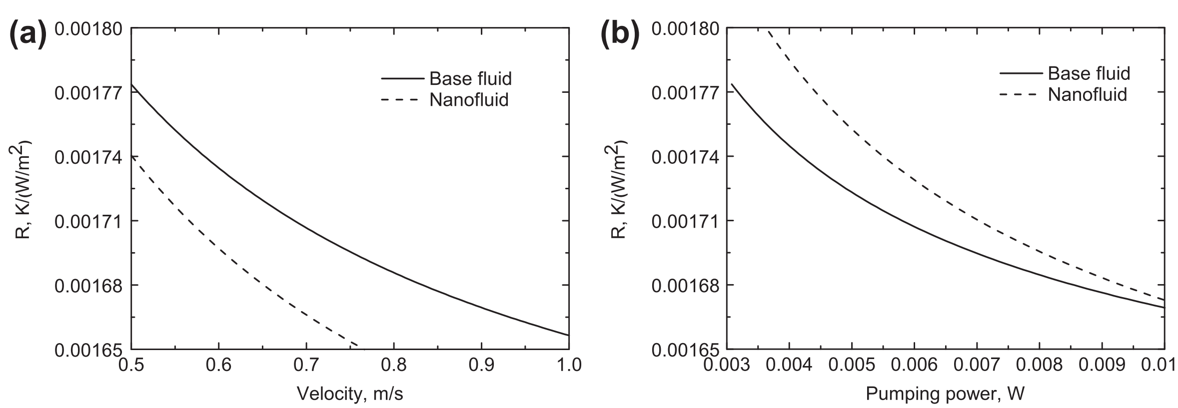

Fig. 1. Thermal resistance R with water and titanium oxide nanofluid for several values of (a) velocity (b) pumping power, in a tube of internal diameter 4.0 mm and length 0.5 m.