Heat Transfer Enhancement in Circular Tube Flow Using Graphene Oxide Nanofluid and Its Application to Multi-Heat Pipe

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Summary: The experimental study is performed to study convective heat transfer behavior of nanoparticles, i.e., graphene oxide (GO) flowing through a horizontal tube heated under constant heat flux condition. Its application to a multi-heat pipe cooling device in the condition of different filling ratio (40%, 60%, 80% and 100%) under constant heat flux conditions is experimentally investigated. It was found that (i) heat transfer enhancement is caused by suspending nanoparticles, so that maximum value of the Nusselt number is over twice than that of the pure working fluid, (ii) the thermal performance of heat pipe increases with increasing the concentration of GO nanoparticles in the base fluid, while the maximum heat transfer enhancement yields at 0.2% volume concentration, (iii) GO/water nanofluid shows lower thermal resistance compared to pure water, (iv) the optimal thermal resistance is obtained at 100% filling charge ratio with 0.2% volume concentration, and (v) heat transfer coefficient of the heat pipe significantly increases with an increase in heat flux and GO nanoparticles concentration.

Keywords: Circular tube flow, Multi-heat pipe, Graphene oxide, Filling ratio, Volume fraction, Thermal resistance.

1. INTRODUCTION

The utilization of solid particles as an additive suspended into the base fluid is a technique for heat transfer enhancement. Li and Xuan [1] studied experimentally convective heat transfer performances of nanofluids for laminar and turbulent flow inside a tube and disclosed that a remarkable increase in heat transfer performances of nanofluids causes for the same Reynolds numbers. Heat transfer enhancement is also proposed by Xuan and Roetzel [2] who used copper nanoparticles. Xuan and Li [3] measured convective heat transfer coefficient of Cu/water nanofluids in circular pipe flow under constant heat flux condition, and reported that the suspended nanoparticles remarkably enhance heat transfer process with smaller volume fraction of nanoparticles. As for the heat transfer performance of CNT nanofluids in a tube, Ding et al. [4] found that the observed enhancement of heat transfer coefficient is much higher than the increase in the effective thermal conductivity. Using nanofluids, i.e., γ-Al₂O₃ and TiO₂ water, Pak and Cho [5] studied convective heat transfer performance in tube and found that for fixed average fluid velocity, the convective heat transfer coefficient of the nanofluids is lower than that of pure water. They postulated that the suspensions have higher viscosity than that of pure water, especially at high particle volume fractions. Yang et al. [6] measured much lower increase of convective heat transfer coefficient with respect to the effective thermal conductivity using nanofluid including the disc-shape nanoparticle, and concluded that the particle shape or aspect ratio of the particle is a significant parameter to affect the thermal performance of nanofluids. Throughout the existing reports, Wang and Mujumdar [7] described that: (i) the application of nanofluids for heat transfer enhancement should not be decided only by their effective thermal conductivity and (ii) many other factors such as particle size, shape and distribution, micro-convection, pH value, and the particle–fluid interactions should have important influence on the heat transfer performance of the nanofluids.

Heat pipes have a variety of advantages, such as high heat removal rate per unit volume, a fully passive working principle, and easy applicability. Heat pipes have been used in various thermal engineering fields such as computer CPUs, solar energy collectors and micro device transmitting equipment.
Kim and Bang [8] experimentally observed the effect of the working fluid fill ratio and the cross sectional area of the vapor path on the heat removal capacity and thermal performance of an annular thermosyphon that contains a neutron absorber. They observed that increasing the fill ratio enhanced the entrainment limit by 18%. Sarafraz and Hormozi [9] studied experimentally the effect of different operating parameters such as applied heat flux to the evaporator section, fill charge ratio of working fluid, tilt angle of the heat pipe and volume concentration of nanoparticle on the thermal performance and efficiency of the thermosyphon heat pipe. Lips et al. [10] carried out experiments to observe the effect of the filling ratio and vapor space thickness on the thermal performance of a flat plate heat pipe (FPHP) using n-pentane in horizontal orientation. They concluded that a small vapor space thickness induces liquid retention in the FPHP sides and corners and thus reduces the thermal resistance of the system even for a liquid quantity greater than the optimum value. Mameli et al. [11] focused on the combined effect of the inclination angle and filling ratio at different heat input levels on the device operation stability and the thermal performances of a multi-turn closed loop pulsating heat pipe (CLPHP) made of copper. The test fluid was FC-72. Results showed that this CLPHP is very much sensitive to the inclination angle and that the vertical operation is affected by unstable operation at high input levels. Barua et al. [12] investigated experimentally the thermal performance of a closed loop pulsating heat pipe using two different working fluids (water and ethanol) at various filling ratios (100%, 82.5%, 63%, 41.3% and 28%). Qu et al. [13] investigated experimentally the performance of a stainless steel oscillating heat pipe (OHP) charged with base water and spherical Al2O3 particles of 56 nm in diameter. They showed that the maximum thermal resistance was decreased by 0.14 °C/W (or 32.5%) when the power input was 85.8 W at 70% filling ratio and 0.9% mass fraction. Lin et al. [14] studied the effect of silver nanofluid (20 nm in diameter) on copper pulsating heat pipe thermal performance. The thermal performance was studied at different concentration (100 ppm and 450 ppm), various filled ratio (20%~80% FR) and different heat power (5 W~85 W). The results showed that the best filled ratio was 60% and the better working fluid was 100 ppm of silver nanofluid. Khandekar et al [15] investigated the effect of working fluid (water, ethanol and R-123) and filling ratio on the thermal performance of closed loop pulsating heat pipe in vertical and horizontal orientation. They found that the best performance was measured at low filling ratio for all working fluids.

The purpose of this study is to disclose the thermal fluid flow transport phenomenon of nanofluid in a circular tube under constant heat flux condition. Emphasis is placed on the effects of the suspension with the particles, i.e., the volume fraction of particles and Reynolds number on heat transfer performance in the turbulent flow region. Here, graphene-oxide-nanofluid developed here is used as the working fluid and are tested under the constant heat flux boundary condition. As the application to a cooling device with the aid of graphene-oxide-nanofluid, experimental studies is conducted on the thermal performance of a copper multi-heat pipe charged with pure water and graphene oxide (GO)/water nanofluids as working fluid.

2. MATERIALS AND METHODS

Figure 1 illustrates the experimental apparatus for measuring the convective heat transfer coefficient which consists of a closed flow loop, a heating unit, a cooling part, and a measuring and control unit. A straight stainless tube with 2000 mm in length, 3.96 mm in inner diameter, and 0.17 mm in thickness is employed as the test section and electrodes for the direct electric current heating are connected at both ends. The DC power supply is employed and its voltage is adjustable using the voltmeter to control the heat flux at the pipe wall. The test tube is surrounded by a thick thermal insulation material to suppress heat loss from the test section. The twelve thermocouples are used to measure the local wall temperature along the heated surface of the tube, and the other thermocouples are inserted into the flow at the inlet (Tin) and outlet (Tout) of the test section to measure the bulk temperature of a working fluid. The working fluid in the test loop is circulated by a magnet pump and is measured by an electromagnetic flow meter. The inlet working fluid was maintained at Tin=293K and the corresponding value, Tout, of the outlet which depend on Reynolds number, volume fraction and heat flux, was less than 313K throughout all experiments.
Figure 1 Experimental Apparatus

Figure 2 represents a multi-heat pipe which consists of evaporator, adiabatic section and condenser. The external dimensions for heating and cooling sections are 45 x 45 x 8mm, and the internal dimensions are 42 x 42 x 5mm. The adiabatic section is consisted of four parallel circular tubes whose dimension is $\phi 6$ (external diameter) x $\phi 5$ (inlet diameter) x 45 mm (length). As shown in figure, twelve k-type thermocouples were installed on the test section, with five of them embedded in the evaporator section (H1, H2, H3, H4, and H5), four in the adiabatic section (a1, a2, a3, and a4), and three in the condenser (C1, C2, and C3).

Figure 3 depicts the experimental setup, which consists of a test section (multi-heat pipe). Vacuum pump was connected with vacuum gauge to generate vacuum pressure inside the heat pipe. The evaporator section was electrically heated by heater block. The condenser section was cooled by immersing it into the plastic cooling chamber and water was used as the coolant fluid which pumped from the thermostatic bath.

The test section was then charged with the working fluid. The filling charge ratios (FR) (volume ratio of the working fluid to the internal volume of the evaporator section) were varied at 40%, 60%, 80% and 100% for each working fluid. The vacuum pressure inside the test section was set to 9.5 kPa for all cases. The test section was heated gradually until a steady state was attained.

Figure 2 Multi-heat pipe with thermocouple locations (all dimensions are in mm).
3. RESULTS AND DISCUSSION

Figure 4 depicts the relationship between Nusselt number \( Nu \) and Reynolds number \( Re \) with volume fraction as the parameter. The corresponding data of CuO and Al\(_2\)O\(_3\) are superimposed for comparison [16]. Here the heat transfer coefficient at \( x/D=200 \), which corresponds to the hydrodynamically and thermally fully-developed region based on the pre-experimental result, is employed to obtain the Nusselt number.

The following Gnielinski equation [17] in the turbulent flow is superimposed as a solid line for reference,

\[
Nu = \frac{(f/8)(Re-1000)Pr}{1.07 + 12.7 \sqrt{f/8[Pr^{0.4} - 1]}}
\]

where

\[
f = [1.82 \log_{10}(Re) - 1.64]^2
\]

Figure 4  Nusselt numbers for each nanofluid.

One observes that the Nusselt numbers for each nanofluid are slightly higher than that for pure water. And heat transfer performance becomes larger in the higher Reynolds number region, as expected. Thus since the volume fraction of nanoparticle is substantially low, heat transfer enhancement is slightly attributed due to the suspension of nanoparticles. In contrast, substantial heat transfer enhancement yields for graphene-oxide nanofluid. Throughout the experimental results, heat transfer enhancement is caused by the suspension of particles and its trend becomes larger for graphene-oxide nanofluid. In general, an increase in the particle volume fraction yields the particle distribution in the center region of the pipe whose behavior leads to the increasing of the viscosity. That is, a consequence of the viscosity increases near the centerline, resulting in flat velocity profile. This trend becomes larger, because for the nanofluid including graphene-oxide nanosheets the corresponding central velocity in the pipe is suppressed due to its flat sheet in comparison with that of nanofluids for CuO and Al\(_2\)O\(_3\). At the same time, the substantial velocity gradient is induced in the vicinity of the heated wall. It is postulated, therefore, that the particle, i.e., nanosheet suspension induces the velocity gradient near the heated wall, resulting in enhancement of heat transfer performance for lower volume
fraction of grapheme-oxide nanofluid.

Next task is to study the heat transfer characteristics in multi-heat pipes. The overall heat transfer coefficient of the heat pipe calculated using the surface temperature of evaporator and condenser section using Eq. (3).

\[
h = \frac{Q}{A_e (T_H - T_C)},
\]  

(3)

where \(A_e\) is the cross section area of the evaporator section. Figure 5 shows the overall heat transfer coefficient against the heat load for all the filling charge ratios and the volume concentrations. For GO/water nanofluid, a higher heat transfer coefficient is registered for all volumetric concentrations of nanoparticles in comparison with those reported for pure water at a similar condition. The GO nanoparticles in the heat pipe not only increased the fluid thermal conductivity but also enhanced the heat transfer coefficient due to the particles migration. It is clear from Fig. 5 that the increase of the heat load intensifies the heat transfer coefficient of the heat pipe for each filling ratio.

Results demonstrate that the heat transfer coefficient of heat pipe drastically increases with increasing the filling charge ratio at the same input heat fluxes, because the temperature difference between the evaporator and condenser section decreases with increasing the filling charge ratio. The optimum heat transfer coefficient was obtained at 100% filling ratio for 0.15 vol.% and 0.2 vol.% and at 80% for 0.1% volume concentration. For pure water and 0.05 vol.%, the heat transfer coefficient increases with the rise of filling ratio till 15W input heat load. Beyond 15W, the maximum heat transfer coefficient was obtained at 80% filling ratio.

![Overall heat transfer coefficient as a function of input heat load and filling ratio.](image)

Figure 5 Overall heat transfer coefficient as a function of input heat load and filling ratio.
4. CONCLUSION

Experimental study has been performed to investigate heat transfer performance of aqueous suspensions of nanoparticles, i.e., graphene oxide. The corresponding application to the device is carried out to the thermal characterizations of a copper multi-heat pipe. The results are summarized as follows:
(a) Heat transfer performance in the circular tube flow is amplified by suspension of nanoparticles in comparison with that of the pure water.
(b) Substantial heat transfer enhancement for graphene-oxide nanofluid becomes larger than that for different nanofluids.
(c) Heat transfer performance of a multi-heat pipe is apparently improved after the addition of GO nanoparticles in the working fluid.
(d) With increasing the input heat power and volumetric concentration of nanofluid, the overall thermal resistance of the heat pipe is increased.
(e) The overall heat transfer coefficient depends greatly on the filling ratio, and the lower filling ratio (40%) yields smaller heat transfer coefficient.

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