THERMAL CHARACTERISTICS OF GROOVED HEAT PIPE WITH HYBRID NANOFLUIDS

by

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In the present study, the specially designed grooved heat pipe charged with nanofluids was investigated in terms of various parameters such as heat transfer rate (50-300 W with 50 W interval), volume concentration (0.005%, 0.05%, 0.1%, and hybrid combinations), inclination (5°, 45°, 90°), cooling water temperature (1 °C, 10 °C, 20 °C), surface state, transient state and so on. Hybrid nanofluids with different volume concentration ratios with Ag-H_2O and Al_2O_3-H_2O were used as working fluids on a grooved heat pipe. Comparing with the pure water system, nanofluidic and hybrid nanofluidic systems shows greater overall thermal resistance with increasing nano-particle concentration. Also hybrid nanofluids make the system deteriorate in terms of thermal resistance. The post nanofluid experimental data regarding grooved heat pipe show that the heat transfer performance is similar to the results of nanofluid system. The thermal performance of a grooved heat pipe with nanofluids and hybrid nanofluids were varied with driving parameters but they led to worse system performance.

Key words: *grooved heat pipe, nanofluids, heat transfer*

Introduction

Generally, a heat pipe is a heat mover or heat spreading device and it acquires heat from a source and transfers or spreads it to a sink region. The heat transfer ability is the most important factor, and it has to be designed to maximize with various influencing parameters. It needs very little drop in temperature to move this heat. Typically the heat pipe is a simply sealed and vacuumed tube with a porous wick structure to generate capillary force which is the most important factor to improve the heat transfer performance and a very small amount of working fluid is charged inside. The heat pipe has mainly three connected sections: evaporator, adiabatic section, and condenser. Various parameters to maximize heat transfer ability of a heat pipe are influenced by working fluids such as nanofluids, pumping pressure, friction loss toward length, charging ratio, inclination angle, and so on. [1-7]

If there is no structural change in the heat pipe such as porous wick, container, orientation, diameter, length and so on, the working fluid is the main solution to improve the

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heat transfer performance of heat pipe. The correct choice of working fluid is the important design factor which can lead to decrease the thermal resistance of a heat pipe. In recent days, nanofluids as the new working fluids of heat pipe have been tried to be adopted as the heat transfer fluid. Nanofluids are a new class of heat transfer fluids that 1% or less of particles, fibers, or tubes with lengths on the order of 1-50 nm are stably suspended in traditional heat transfer fluids such as water, alcohol, *etc*. [3-21].

It was first time to be called the fluids with nano sized particles as "nanofluid" by Choi in Argonne National Laboratory, USA, [13]. Nanofluid refers to multi-phase dispersed mixture constituted of "nanoparticles" *i. e.* extremely fine metallic particles of size below 50 nm or much less [13]. Main advantage of nanofluids is that it can enhance the heat transfer characteristics with improved thermal properties comparing with the base fluid. From some experimental data obtained by previous researchers, it has shown that even with a relatively low concentration of particles from 1% or less in volume, the effective thermal conductivity of the nanofluid has increased by almost 20% or more compared to that of the base fluid. Such an increase of thermal conductivity depends mainly on several factors such as the particle shape, its size and concentration of particles, the thermal properties of the host fluid and the particles [13, 14].

Few experimental or analytical investigations using nanofluids on a heat pipe have been carried out [3-7, 22-27]. A wide range of nano particles such as Ag, CuO, Diamond, Ti, Ni, and Au have been used to study the influence of nanofluid on the heat pipe's thermal performance [3-7, 22-27]. Enhanced heat transfer performance of heat pipe with nanofluids has been reported by few researchers [3-7, 22-27]. They showed improved heat transfer performance in terms of thermal resistance and temperature gradient. Kang *et al.* [4, 10, 22-24] reported heat transfer enhancement with various nannofluids such as Ag and Au-nanofluids in terms of thermal resistance. Specially, Tsai *et al.* [3] reported the effect of Au-nanofluid in heat pipes. At a same volume concentration of nanofluids, they [3, 4, 10, 25-27] showed the significant reduction in thermal resistance of heat pipe as compared with distilled water. Kang *et al.* [10] showed the thermal enhancement of grooved heat pipe performance using Ag-nanofluid as the working fluid. Ma *et al.* [3] showed the investigation of pulsating heat pipe charged with diamond nanofluid with decrease of a thermal resistance.

Liu *et al.* [25] and Park *et al.* [26] reported that the nanoparticle concentration of nanofluids has remarkable influence on the both of boiling heat transfer coefficient and the critical heat flux (CHF) of the nanofluids with the increase of the concentration. However, when the concentration is over a certain value, the CHF is basically close to a constant value, and the heat transfer deteriorates gradually. There exists an optimum mass concentration for nanofluids. Ma *et al.* mentioned the heat pipe with water after using nanofluids also showed similar trend in CHF with nanofluid system.

Heat pipe works based on capillary driven force associated with boiling heat transfer. The literature for pool boiling heat transfer performance showed a decrease in heat transfer ranging from 10-40% such as Bang *et al.* [28] (Al_2O_3 nanofluids up to 4% by volume concentration), Das *et al.* [29] (with Al_2O_3 nanofluids, surface fouling and deterioration of heat transfer), Jackson *et al.* [30] (Au-nanofluids on a Cu block), and Kim *et al.* [31] (Al_2O_3, ZrO_2, SiO_2 nanofluids). The authors observed deterioration of heat transfer performance. The above mentioned authors found that a significant amount of nanoparticle deposition on the heating surface and that the bubble contact angle was reduced in certain range depending on their experimental conditions. They reported boiling heat transfer deterioration caused by nanoparticle deposition. For boiling researches with nanofluids, few papers could be found, but they could not show clear influence to the applicable systems, whether positive or negative effects.

The rate of heat transfer falls with the increase in nanoparticle concentration and eventually becomes inferior even to pure water. Chopkar *et al.* [32] conducted tests with nucleate pool boiling of ZrO_2 based nanofluid on a Cu block. At low volume concentration, heat transfer performance was enhanced, but at higher a decrease in heat transfer was seen. They reported that addition of surfactant to the nanofluid shows a drastic deterioration in nucleate boiling heat transfer. Also, they reported nanoparticle deposition on the heated surface. Narayan *et al.* [33] tested Al_2O_3 nanofluids on a vertical tube with a variety of surface finishes. They defined a "surface interaction parameter" as called the surface roughness. They reported that when the interaction parameter is close to or less than unity, boiling heat transfer performance is deteriorated. If the parameter is higher than unity, and heat transfer performance is improved. They concluded that nucleation cavities can be basically blocked if particles are roughly the same size as the nucleation sites – causing deterioration. Most interestingly, Narayan *et al.* [33] suggests that enhancement or deterioration can be controlled by surface conditions. Also one [11] of above studies noticed particle deposition as a result of nanofluid boiling.

Some research results mentioned that the CHF could be increased comparing with that of pure water [11, 12, 33]. Nanofluid is a new frontier working fluid to be investigated in deep. It was reported that the nanofluid system is a good challenge to heat transfer applications due to its high thermal conductivity which is improved by metallic nano sized particles.

Shafahi *et al.* [34, 35] investigated flat and cylindrical heat pipes with nanofluid analytically based on thermal property correlations such as thermal conductivity, viscosity and density, *etc.* His analytical studies on heat pipes with nanofluids (Al_2O_3, CuO, and TiO_2) show the thermal enhancement. Their analytical studies did not show comparable results with experimental results.

In the present study, GHP with Ag, Al_2O_3-nanofluids and Ag + Al_2O_3 hybrid nanofluids was investigated in terms of particle volume concentrations, charged amounts, and orientation, *etc.*

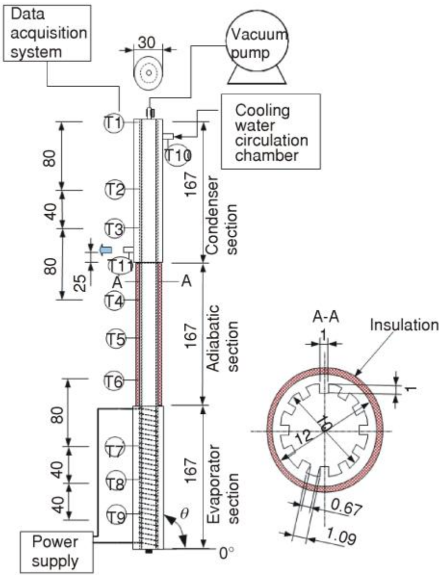


Figure 1. Experimental set-up

Experiments

The present GHP systems using nanofluids have potential applications for industrial applications. Therefore, it is important to select the right component combination for the optimum performance and reliability in terms of working fluids, charged amount, orientation, and so on.

The experimental apparatus, illustrated in fig. 1, mainly consists of the main GHP assembly, the cooling system in the condenser section for GHP, the heat generation section and the charging system, nanofluids as the working fluid. The physical characteristics of the main GHP assembly used in the study are divided into three parts: the evaporation section with the evaporator, the transporting sections, and the condenser section with water jacket.

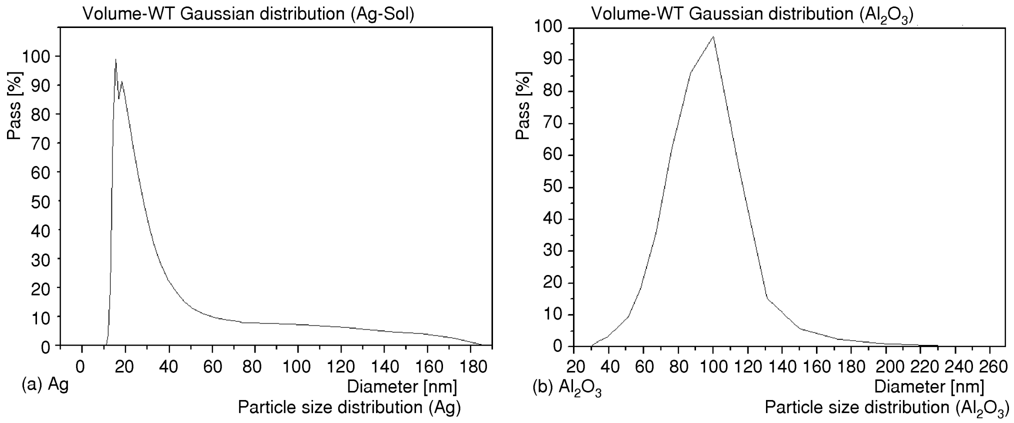
The coolant for GHP circulated through the cooling water jacket as the rate of 20 liter per minute, where heat was removed from the condenser section by forced convection, and then to a constant-temperature water circulation bath. The water bath was mainly set to the experimental temperature and held at a constant cooling temperature through the tests. The power supply and measurement system utilized an electrical resistance heater powered by a regulated DC power supply (Unicorn UP-3050). The data precision was ±0.03% for voltage and current, respectively.

As shown in fig. 1 and tab. 1, the present GHP was made from stainless steel tubes with a 12 mm O. D., and a wall thickness of 1 mm. GHP was designed as described in fig. 1. Grooves of GHP were designed and machined and extruded with 1 mm in depth, 1 mm width on top, and 1.3 mm in bottom. Each groove has taper shape from groove tip as shown in fig. 1. The total length of the test heat pipe was about 500 mm. Three working sections of heat pipe have 167 mm, respectively, in length. From the top ends of the tubes, water jacket was made from brass tube with 30 mm O. D. The overall length of the condenser was 167 mm. A temporary seal of the test heat pipe was made with vacuum pump (PJ KODIVAC). To measure the temperature distribution over the length of the heat pipe, 9 K-type thermocouples (∅ = 0.25 mm) were used. At the start of this investigation, the temperature variation of the test system was recorded on computer based data acquisition system (MX100 Yokogawa). The accuracy was specified to be a maximum of ± 0.05% for K-type thermocouple at the range of –200-1370 ^{o}C.

Table. 1 Geometric Specification of rectangular GHP

Wick	Working fluids	Specification
Groove	– Water(H_2O), – Acetone (C_8H_2O) – Ag-nanofluids (0.005%, 0.05%, 0.1%) – Al_2O_3-nanofluid (0.005%, 0.05%, 0.1%) – Ag and Al_2O_3 hybrid nanofluids; Ag (0.005%) + Al_2O_3 (0.005%) Ag (0.05%) + Al_2O_3 (0.05%) Ag (0.1%) + Al_2O_3 (0.1%) Ag (0.1%) + Al_2O_3 (0.005%) Ag (0.1%) + Al_2O_3 (0.005%)	– Total length: 500 mm – Wick structure: 12 grooves; each groove: 1 mm high, 1 mm wide, 1 mm bottom width – Evaporator: stainless steel 167 mm long, 12 mm O. D., 10 mm I. D. – Heater: 3.3 Ω resistance wire heater – Adiabatic section: Stainless Steel 167 mm long, 12 mm O. D., 10 mm I. D. – Condenser section: stainless steel – tube 167 mm long, 12 mm O. D., 10 mm I. D. – Cooler: stainless steel water jacket, 30 mm O. D., 1 mm thick, 167 mm long, – Coolant flow rate : 20 liter per minute

As shown in fig. 1, the special heater was designed and manufactured. The heater consisted of wire resistance heater (internal resistance: 3.3 Ω) and the maximum power of the present heater could supply up to 350 W. The pure distilled water, acetone, and nanofluids as the working fluids were prepared to fill into the GHP. Nanoparticles used in this work were Ag and Al_2O_3 with a size range of 27 nm (Ag), 89 nm (Al_2O_3), supplied by NanoANP Co, Korea. Pure water was used as the base liquid. Nanofluids were mixed between Ag-nanofluid and Al_2O_3-nanofluid and the mixture was sonicated continuously for 16-20 hours in an ultrasonic bath (DaeRyun Science Inc., Korea). Figure 2 shows nanoparticle distribution dispersed in Ag and Al_2O_3 nanofluid, respectively, which have been used in the present experimental study. The working fluid in a heat pipe has a significant effect on its performance, thus present experimental investigation will consider the various parameters on nanofluids in terms of particle concentrations and hybrid combinations, *etc*.



(a) Ag (b) Al_2O_3

Figure 2. Particle size distribution of nanofluid

The heat losses through the insulation of the heating, adiabatic, and evaporation section were seen negligible because of the good insulation of the heating and the evaporator sections. The errors involved in the data reduction for the heat transfer performance and thermal resistance were generally due to the inaccuracy of the temperature and the power measurements. Even if the readings of the power and the temperatures were recorded with regulated DC power supply (±0.03% for voltage and current) and MX-100 Data acquisition system (±0.05% for temperature). The accumulated error for thermal resistance was ±1.65%.

Results and discussion

In the present study, the working fluid was charged with 32% of total inner volume. An important constraint in heat pipe operation is the charged amount of working fluid. As reported with a same GHP system by Shin [5], the best heat transfer performance with a GHP was placed in 32% charged amount of pure water. Therefore, the following experiments were carried out with 32% charged amount.

Figure 3 shows the effect of various working fluids (pure water, acetone, Ag, Al_2O_3, Ag-Al_2O_3 mixed hybrid nanofluids) on dimensionless overall temperature difference, and T^* defined in eq. (1). As shown in fig. 3, pure water is a reference base liquid to be compared with others. In fig. 3, the GHP system with Ag, Al_2O_3

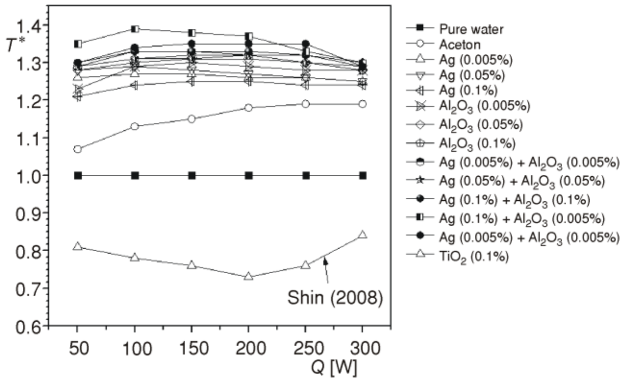


Figure 3. Effect of Q on T^* with different working fluids (θ = 90°, 50~300 W)

and hybrid nanofluids showed higher values of T^* than those of pure water. As shown in fig. 3, GHP with Ag, Al_2O_3 and hybrid nanofluids shows worse heat transfer performance than TiO_2 system. Ag-0.1% nanofluid shows better performance comparing with other working fluids. Shin [5] with TiO_2-nanofluid shows about 20% lower value of T^* comparing with water system

$$T^* = \frac{\Delta T_{\text{Working fluids}}}{\Delta T_{\text{water}}} \tag{1}$$

$$R = \frac{\Delta T_{\text{h-c}}}{Q} \tag{2}$$

Figure 4 shows the thermal resistance of GHP system *vs.* different working fluids. Thermal resistance data have been collected as eq. (2) $\Delta T_{\text{h-c}}$ between evaporator surface and condenser surface temperature and Q. The collected data are analyzed with ±1.65% uncertainty. As shown in fig. 4, Shin [5] shows that the thermal resistance of GHP system with TiO_2-nanofluids as the working fluid is lower than result with pure water. Also figure shows the effect of nanofluid on thermal resistance with various volume concentrations of nanoparticle. As shown in fig. 4, thermal resistance was 40-50% higher than results of the pure water GHP system. For GHP, thermal resistance was the lowest with pure water (0.5 W/^{o}C at 300 W). The highest thermal resistance of the present grooved heat pipe was obtained with Ag (0.1%) + Al_2O_3 (0.005%) hybrid nanofluid. As shown in figure, Ag + A_2O_3 hybrid nanofluid has led to the deterioration of the thermal performance in the present GHP system comparing with the pure water system. From previous researches [28-33, the deterioration of heat transfer performance could be predicterd due to the settling, agglomeration of the nanoparticles and/or a surface deposition during the boiling experiments, the experimental procedure will also play a key role in the outcome. Few of the existing sources investigating the pool-boiling of nanofluids in the GHP address this issue about the trend of heat transfer performance whether it is positive or negative comparing with the pure water. The reasons of the deterioration of heat transfer of GHP would be described such as inner surface deposition,

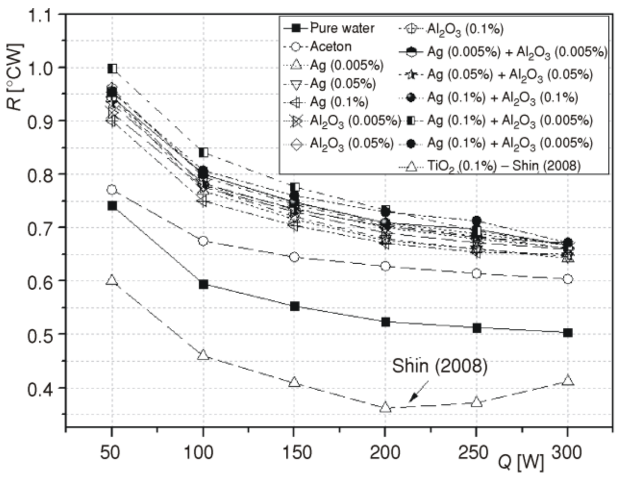
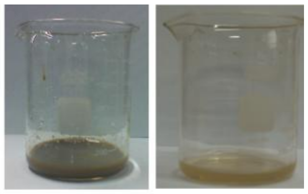
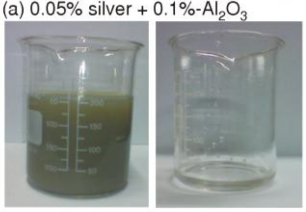


Figure 4. Effect of Q on thermal resistance with different working fluids (θ = 90°, 50~300 W)



Before experiment After experiment
(a) 0.05% silver + 0.1%-Al_2O_3



Before experiment After experiment
(b) 0.05% silver

Figure 5. Nanofluids before and after experiment

nano-particle agglomeration, and particle clustering. As shown in fig. 5, the particle deposition could be expected because nanofluid after experimental investigation shows light color with reduced particle concentration. From fig. 5, the nanofluid after experiment was much lighter than nanofluid before experiment. This can be explained as the nanoparticles were deposited in some surface inside GHP. These deposited particles can affect to the GHP in various ways for GHP operation. Effects of particle deposition were reported as in different manners. One is the heat transfer performance deterioration, another one is CHF improvement. As shown in fig. 4, the heat transfer performance was degraded compared with pure water. This heat transfer degradation due to the particle deposition can be explained with previous literatures as fig. 6. In the present study, the experimental work has been performed to find the mechanism by which nanofluids create this heat transfer deterioration.

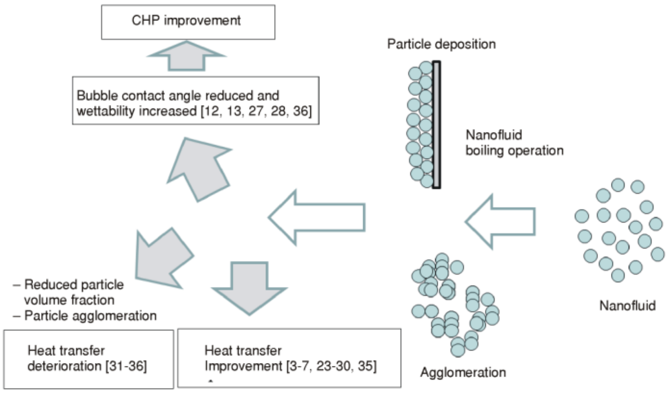


Figure 6. Thermal working mechanism of nanofluid

The results presented here investigate the pool-boiling characteristics in GHP with Ag + Al_2O_3 hybrid nanofluids. Figure 7 shows the effect of cooling medium temperature (T_c = 1, 10, and 20 °C) in the condenser section on the thermal resistance. It was obtained that the GHP system with lowest T_c (1 °C) shows high thermal resistance. As shown in fig. 7, the GHP system with nanofluids in the T_c = 10 and 20 ^{o}C has 20% higher thermal resistance than pure water. Also the GHP system with Ag-Al_2O_3 hybrid nanofluids shows much higher thermal resistance value than single Ag and Al_2O_3 nanofluids. This could be that the GHP system with pure water in the low T_c is working in inactive state because the T_c is close to freezing point. Also this would be because the thermal properties of working fluid are changed when nanoparticles are added to the base fluid.

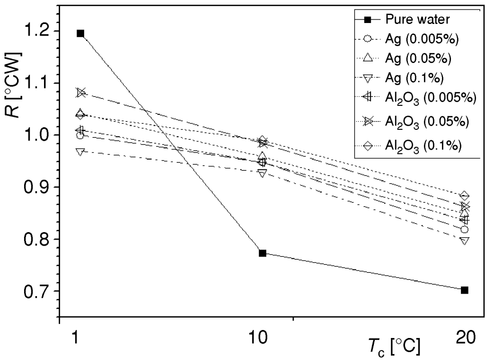


Figure 7. Effect of T_c on thermal resistance with different working fluids (θ = 90°, Q = 50 W).

Figure 8 shows the temperature profile along the heat pipe with 50 and 200 W of low and high heat flux. It was observed that the GHP system with nanofluids and hybrid nanofluids operates in higher temperature condition comparing with pure water system. As shown in figure, the system temperatures with Ag and Al_2O_3, and hybrid nanofluids in each position were higher than those of conventional pure water and acetone. Especially the GHP system with hybrid nanofluids shows high temperature profile. When the system has high temperature profile along the system, it means that the system has difficulties to transfer heat from a source.

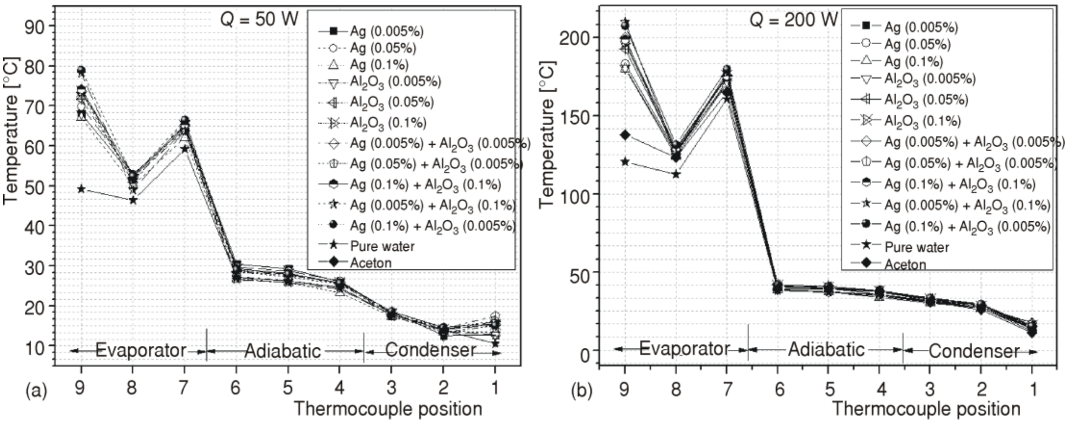


Figure 8. Temperature profile along heat pipe with different working fluids

Figure 9 shows the temperature fluctuation in terms of amplitude between low peak and high peak ($\Delta T_{\text{p-p}}$) in the evaporator and condenser section of the GHP system with different working fluids. As shown in fig. 8, the GHP system with some nanofluids of 50 W is working in unstable state. Hybrid – Ag (0.05%) + Al_2O_3 (0.005%) – nanofluid shows seriously high temperature fluctuation ($\Delta T_{\text{p-p}}$) and the system with high particle concentration shows high $\Delta T_{\text{p-p}}$. In 200 W, the GHP system shows different trend comparing with the case of 50 W. As shown in fig. 9, it was observed that the systems with 200 W and pure water, low particle concentrations are working in much higher fluctuation temperature. It was observed that the condenser section shows much higher $\Delta T_{\text{p-p}}$ than the evaporator section. This can be expected because the nanoparticle abruptly deports their heat moved from the evaporator and their active motion dramatically shirked. This makes large fluctuation in the condenser section.

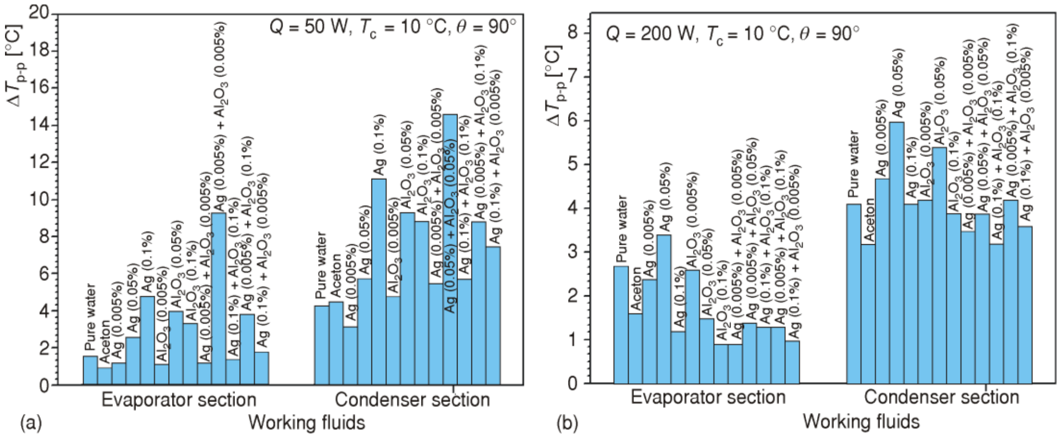


Figure 9. Effect of different working fluids on temperature fluctuation

Figure 10 shows the effect of orientation of heat pipe. Although the system with nanofluids shows 20-30% higher T^*, this means that the heat transfer performance is decreased with dispersed nanoparticles. Also, the GHP system oriented close to horizontal position with 50 W shows that T^* with nano-fluids and hybrid nanofluids are less than that of pure water. Increasing heat flux, the T^* of GHP system is higher than T^* of nanofluids.

Figure 11 shows the post nanofluid experiments. As shown in figure, experimental data with the post nanofluid with GHP show that the post nanofluid heat pipe performance is similar to that of Ag-0.05% system. The reason for heat pipe thermal enhancement for the post GHP can be explained with the particle deposition on the surface which can make higher wettability and increase surface heating surface area.

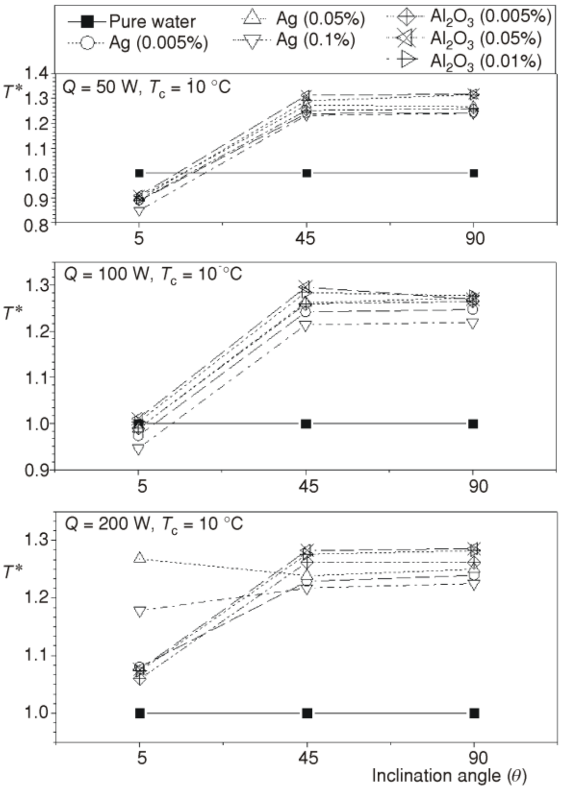


Figure 10. Effect of θ on T^* with different Q

Figure 12 shows the temperature ratio of inlet and outlet surface temperature of the evaporator and the condenser section. It can explain whether the nano-fluid is useful for the evaporation or the condensation. As shown in the figure, it was observed that Ag and Al_2O_3 nanofluids have high dimensionless temperature values. The GHP system with hybrid nanofluids shows lower value than value of the pure water system. In the present study, nanofluids with Ag and Al_2O_3 react in the condenser section. This means that the heat transfer performance with short length of the condenser can be utilized, not based on the conventional heat transfer.

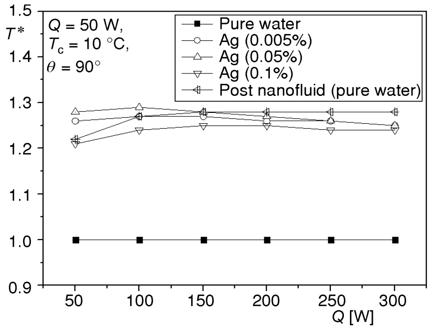


Figure 11. Effect of Q and T^* with pure water with post nanofluid system

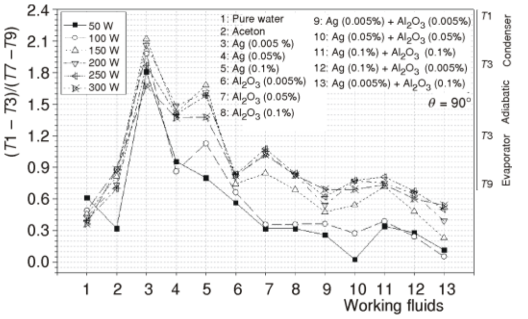
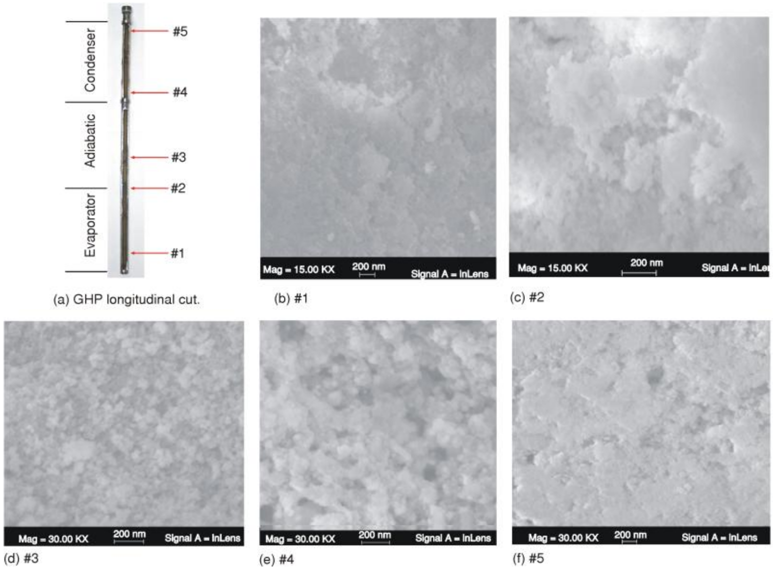


Figure 12. Effect of different working fluids on temperature ratio



(a) GHP longitudinal cut. (b) #1 (c) #2
(d) #3 (e) #4 (f) #5

Figure 13. Heat pipe longitudinal surface views

Figure 13 shows the visualization of inside heat pipe surface. The heat pipe after experimental study was axially cut along the length and SEM photos were taken to observe surface variation with particle deposition. As shown in fig. 11, inside surface of heat pipe was varied depending on the working positions. In the evaporator, the shape of deposition was planer type and large size. But moving to the condenser section, the deposition shape looks like spherical particles on the surface. This phenomenon can be observed in the 5 local photos of fig. 13.

Conclusions

The present study investigated the thermal enhancement of a GHP and heat pipe performance using various nanofluids and hybrid nanofluids as the working fluid. In this comparative study, the results of the performance test are as following concluding remarks:

- In lower angle, GHP shows the better performance than vertical position. Comparing with the pure water system, nanofluidic and hybrid nanofluidic system shows higher overall thermal resistance.
- Thermal resistance was high with increasing nano-particle concentration. Also hybrid nanofluids make the system deteriorate in terms of thermal resistance and T^*.
- The post nanofluid experimental data with GHP shows that the post nanofluid heat pipe performance is similar to that of nanofluid system.
- In this investigation, the thermal performance of a GHP with nanofluids and hybrid nanofluids was varied with driving parameters but they led to worse system performance. This means that nanofluids are not always attractive as a cooling or energy transfer fluid for devices with high energy density.
- In the present hybrid working fluid system, hybrid nanofluids were not much effective comparing with the pure nanoparticle nanofluid system. But the various trials with different combination need to be studied in future.
- The nanofluid shows the different working phenomena based on the surface deposition shapes of the inside surface in the evaporator, adiabatic and condenser section.

- In the present system involved with boiling, it was predicted that nanofluids as the working fluid of the system would not work by the conventional Brownian motion.

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Nomenclature

Q – power input, heat transfer rate, [W]
R – thermal resistance,[°C/W]
T – temperature, [°C]
T_c – cooling media temperature, [°C]
$\Delta T_\text{h-c}$ – overall temperature difference (temperature difference between source and sink), [°C]
$\Delta T_\text{p-p}$ – temperature diferenco between low peak and high peak, [°C]
T^* – dimensionless temperature ($\Delta T_\text{working fluid}/\Delta T_\text{water}$)

Greeck letter

θ – Inclination angle, [degree]

Acronyms

CHF – critical heat flux
CHP – combined heat and power
DC – direct current
GHP – grooved heat pipe
I. D. – inside diameter
O. D. – outside diameter
WT – weight

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