Enhancement of Pool Boiling Heat Transfer Coefficients Using Carbon Nanotubes

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In this study, the effect of carbon nanotubes (CNTs) on nucleate boiling heat transfer is investigated. Three refrigerants of R22, R123, R134a, and water were used as working fluids and 1.0 vol.% of CNTs was added to the working fluids to examine the effect of CNTs. Experimental apparatus was composed of a stainless steel vessel and a plain horizontal tube heated by a cartridge heater. All data were obtained at the pool temperature of 7℃ for all refrigerants and 100℃ for water in the heat flux range of 10~80 kW/m^2. Test results showed that CNTs increase nucleate boiling heat transfer coefficients for all fluids. Especially, large enhancement was observed at low heat fluxes of less than 30 kW/m^2. With increasing heat flux, however, the enhancement was suppressed due to vigorous bubble generation. Fouling on the heat transfer surface was not observed during the course of this study. Optimum quantity and type of CNTs and their dispersion should be examined for their commercial application to enhance nucleate boiling heat transfer in many applications.

Key Words : Nucleate Boiling Heat Transfer, Heat Transfer Enhancement, Nano Fluids, Carbon Nanotubes (CNTs)

Nomenclature

A	: Heat transfer area [m^2]
h	: Heat transfer coefficient [W/m^{2}K]
k	: Thermal conductivity [W/m·K]
L	: Length [m]
P	: Pressure [kPa]
q	: Heat transfer rate [W]
r	: Radius [m]
T	: Temperature [K or ℃]

Subscripts

avg	: Average
bottom	: Bottom
l	: Liquid
o	: Outside diameter
s	: Surface
sat	: Saturation
side	: Side
t	: Thermocouple
top	: Top
tube	: Tube

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1. Introduction

Recently, environmental protection and energy conservation have become important issues due to greenhouse warming caused mainly by the use of fossil fuels. One of the fundamental ways of reducing greenhouse warming is to increase the efficiency of energy conversion devices. In order to accomplish this goal, heat exchanger performance needs to be improved in power and chemical plants and refrigeration and air-conditioning equipment.

Boiling heat transfer has been employed in many thermal energy dissipation systems due to its high heat removal capacity. For the past few decades, many studies were carried out to further increase the boiling heat transfer coefficients (HTCs) and some successful passive type enhanced surfaces such as low fin tube, Thermo-Excel-E tube, Turbo-B tube, Turbo-C tube have been applied to many heat exchangers (Webb, 1994). This kind of passive technique relying upon the surface geometry, however, is almost at the point of saturation and at this time active heat transfer enhancement technique needs to be developed to overcome the present environmental problem. One of the active methods is to apply electric field to the vicinity of heat transfer surface to cause a strong convection current. But this method was proved to be good only in laboratory experiments and the commercialization of this method in actual heat exchangers seems to be very difficult and impractical (Ohadi, 1991).

Another active way of enhancing heat transfer is to use so called 'nano particles' which have been developed for the past 10 years (Eastman et al., 1997 ; Lee et al., 1999). For this purpose, various nano fluids based on copper and aluminum nano particles were developed. Theoretically, these particles with high thermal conductivity should improve the heat transfer in the laminar sublayer. Even though many nano particles were applied to the single phase heat transfer of water, actual heat transfer improvement was not yet reported. Furthermore, when these particles were applied to the boiling heat transfer, they even caused fouling on heat transfer surface and consequently HTCs were decreased (Das et al., 2003 ; Vassallo et al., 2004 ; Bang and Chang, 2004).

In this study, carbon nano tubes (CNTs) were applied instead of conventional nano particles to the boiling of water and some halocarbon refrigerants for power plant and refrigeration applications. Over the past decade, CNTs with a honeycomb carbon structure have attracted much attention due to their remarkable mechanical and electrical properties and they are known to have very high thermal conductivity (Ajayan, 1999 ; Dresselhaus et al., 2001). Therefore, it is expected that liquids containing CNTs would increase the heat transfer near the laminar sublayer. In turn, this will result in overall heat transfer improvement. The objective of this study is to examine the effect of CNTs on overall heat transfer performance in pool boiling.

2. Experiments

2.1 Experimental apparatus

Figure 1 shows a schematic diagram of the experimental apparatus for nucleate boiling heat transfer that can be used to take measurements up to 2500 kPa. The apparatus was composed of a test vessel and a refrigerant circulating loop. The test vessel was made of a stainless steel pipe of 102 mm inside diameter and 230 mm length. In order to observe the boiling phenomenon, a sight glass was installed in the front section of the vessel. The refrigerant vapor boiled off from the heat transfer tube went into the condenser and was condensed there and the liquid was fed to the bottom of the test tube by gravity. An external chiller with an accurate temperature controller was used to condense the vapor and at the same time to maintain the pool temperature to set a temperature. The entire vessel was insulated thoroughly with polyurethane insulation to prevent possible heat transfer from the surroundings.

Six stainless steel sheathed thermocouples were inserted into the liquid pool while one was attached to the vapor space to measure the liquid and vapor temperatures respectively. Both pressure gage and pressure transducer were installed at the top of the vessel to measure the system pressure. For each test, subcooling was checked

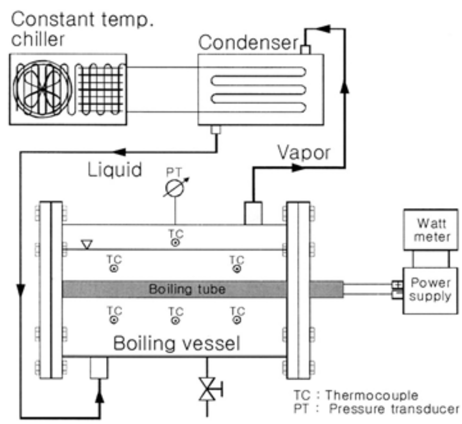


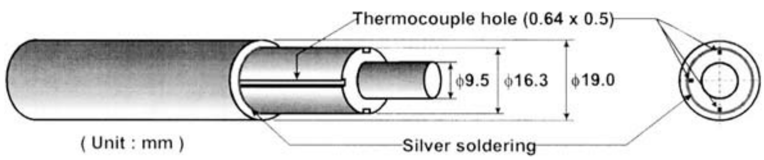
Fig. 1 Schematic diagram of the test facility

by observing the temperature difference between the calculated saturated liquid temperatures based upon the measured pressure and the measured one and was kept to less than 0.2℃ all the time.

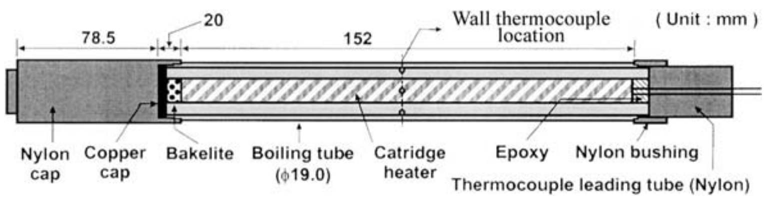
2.2 Manufacture of the tube

Surface temperatures should be measured in order to determine nucleate pool boiling HTCs. But if thermocouples were to be attached to the surface, the surface condition would be altered resulting in erroneous data. One way of resolving this difficulty would be machining a thick wall tube from a copper rod with small holes drilled 2～3 mm beneath the surface in the longitudinal direction. This method is good for plain tubes but is not easy for realization with enhanced tubes of various geometries since machining fins or other surface geometries on a short piece of thick walled tube is difficult and impractical (Jung et al., 1999). In fact, most of the centrifugal chillers employ enhanced tubes for boiling and hence boiling HTC data for these tubes are needed eventually for the successful application of new refrigerants. Therefore, it has been felt that a consistent manufacturing method needs to be developed which is good for both plain and enhanced tubes. For this purpose another method is developed and employed in this study which provides a means to measure surface temperatures of any heat transfer tube at locations roughly 1.2 mm beneath the surface as was done by Jung et al. (1999) for their external condensation heat transfer measurements for both plain and enhanced tubes.

Since this study targets the boiling tubes used in flooded evaporators of conventional centrifugal chillers, a copper tube of 19.0 mm outside diameter and 1.2 mm thickness was chosen as the heat transfer tube. The manufacturing process for the tube specimen with surface temperature measurement holes is described below. As shown in Figure 2(a), a copper rod of 16.3 mm outside diameter was prepared with a center hole of 9.5 mm which was machined by a gun drill. Later a cartridge heater was inserted into this center hole for heating. On this thick walled copper tube of 3.4 mm thickness, three slits of 0.64 mm width were machined 90° apart at the top, side, and bottom by a milling cutter in a longitudinal direction all the way from one end to the other. And then, this thick walled copper tube was inserted into the test tube with stainless steel wires of 0.6 mm diameter placed into the slits across the entire length of the tube. Afterwards these two tubes were silver soldered together with the wires pulled out at the final stage. Through this process, slits accommodating stainless steel sheathed thermocouples of 0.5 mm diameter were made at locations roughly 1.2 mm beneath the surface as illustrated in Figure 2(a). If the tubes were not well soldered together, then the temperatures would vary quite significantly in the longitudinal direction resulting in erroneous data. Hence, a consistent way of joining the two copper tubes was tried many times and thus soldered tubes were cut into many pieces



(a) Components of test tube



(b) Completed test tube

Fig. 2 Cartridge tube specifications to make sure that the silver solder flowed well into the gap in between the tubes. Finally, the best tube showing a good repeatability was selected for the tests.

In this study, a cartridge heater was used to generate uniform heat flux on the surface of the heat transfer tube. The uniformity of the heater was thoroughly checked before its use and it was inserted into the center hole of 9.5 mm diameter which was machined by a gun drill as mentioned earlier. And then a paste of high thermal conductivity was applied between the inner surface of the hole and cartridge heater and the heater was pushed through the hole tightly.

Finally, the left end of the tube was capped with a bakelite piece and epoxy for insulation and soldered with a copper cap for sealing and the other end with the heater's electrical connections was sealed with epoxy and a nylon cap as shown in Figure 2(b).

2.3 Experimental details

Heat transfer performance of boiling tubes used in shell-tube heat exchangers tends to degrade over time due to fouling (Webb, 1994). Therefore, it is important to maintain uniform surface condition for all tests to generate a reliable data set for various refrigerants. For this purpose, the surface of the tube was cleaned with #1000 emery paper for three minutes on a lathe and then cleansed with acetone whenever a refrigerant was changed. For fair comparison, HTCs should be measured under the same steady-state condition at the same pool temperature. Since typical refrigerant temperature in evaporators of commercial chillers is 5～7℃, the pool was maintained at 7℃ for three halocarbon refrigerants. As for water, the pool was maintained at 100℃.

All thermocouples used in this study were T-type (copper-constantan) and were calibrated against a temperature calibrator of 0.01℃ accuracy and wall thermocouples were inserted through the holes in the middle of the test tube longitudinally as shown in Figure 2 while liquid thermocouples were located in the pool as shown in Figure 1. On the other hand, the pressure transducer was calibrated against a pressure calibrator of 0.1 kPa accuracy. The power input to the cartridge heater was measured by an accurate digital power meter of 0.1% accuracy.

The experimental procedure for a given refrigerant was as follows:

(1) Nitrogen was charged to the refrigerant loop up to 1500 kPa with some halogenated refrigerants to check with a halogen detector if there was any leak.

(2) A vacuum pump was turned on few hours to evacuate the system thoroughly and the refrigerant was charged to the system up to 30 mm higher than the top of the heat transfer tube.

(3) The liquid was heated for 2 hours by supplying power to the cartridge heater maintaining the heat flux of 60 kW/m^2 on the heat transfer tube and the vapor was vented a few times for degassing which was especially important for low pressure refrigerants that might have trapped some air within.

(4) After one hour, power to the cartridge heater was initiated and the heat flux was increased to 80 kW/m^2 gradually. And data were taken under steady state at 7℃ from 80 kW/m^2 to 10 kW/m^2 with an interval of 10 kW/m^2 in the order of decreasing heat flux to avoid a hysteresis effect.

(5) Working fluid was changed and the same procedures of (1～4) were repeated after the surface was cleaned as described earlier.

2.4 Application of CNTs

In this study, multiwalled CNTs were mixed with working fluids. The average diameter and length were 20 nm and 1 μm respectively. There has been no report on the boiling heat transfer with CNTs and hence optimum amount of CNTs was not known apriori. Therefore, CNTs were added to the working fluids with the volume fraction of 0.01 as a first trial. Good dispersion of CNTs in the working fluids would be important. In this study, however, no special dispersion method was applied and CNTs were merely added to see their macroscopic effect.

2.5 Data reduction

A local heat transfer coefficient at every wall thermocouple location was determined by equation (1).

$$h = \frac{(q/A)}{(T_s - T_l)} \tag{1}$$

where h, q, A, T_s, T_l are the local heat transfer coefficient (W/m^2K), power input to the cartridge heater (W), heat transfer area (m^2), surface and average liquid temperatures (℃) respectively. As mentioned earlier, the actual surface of the boiling tube is 1.2 mm away from the thermocouple hole and hence the surface temperature, T_s in equation (1), might need to be modified from the measured temperature at each thermocouple location, T_t, by applying an 1-D heat conduction equation in radial direction as follows:

$$T_s = T_t - \frac{q}{2\pi L}\left[\frac{\ln\left(\frac{r_o}{r_t}\right)}{k_{tube}}\right] \tag{2}$$

where $T_t, L, r_o, r_t, k_{tube}$ are the measured temperature by a wall thermocouple (℃), length and radius of the tube (m), the radial distance from the center of the tube to the thermocouple (m), thermal conductivity of the tube (W/mK) respectively. Since the heat transfer tube was made of copper, the temperature compensation term in equation (2), $(T_t - T_s)$, was small, typically less than 0.2℃. Therefore, this term did not have any significant effect on the HTCs and the measured wall temperatures were used directly in the calculation of local HTCs.

Finally, the average HTC at a given heat flux was determined by averaging the three local HTCs as follows:

$$h_{avg} = \frac{h_{top} + 2h_{side} + h_{bottom}}{4} \tag{3}$$

In fact, some tests were conducted to determine if the local HTC measured at one side was similar to that of the other side by rotating the tube 180° circumferentially and the results confirmed that they were very similar. Therefore, using a weighting factor of two for the local HTC measured at one side, h_{side} in equation (3), was justified. Also by rotating the tube 180° the HTCs at the top and bottom of the tube could be compared and the results indicated that they were similar regardless of the rotation indicating indirectly a uniform thermal contact around the circumference of the tube.

The measurement errors were estimated by the method suggested by Kline and McClintock (1953) and turned out to be less than 7% at all heat fluxes. In pool boiling heat transfer, the repeatability is very important and hence many measurements were taken repeatedly with an interval of one week to one month for many fluids to check the repeatability. Overall the repeatability was always within 5% which was within the measurement errors.

3. Results and Discussion

In this study, nucleate boiling heat transfer coefficients of R22, R123, R134a, and water were measured on a horizontal plain tube with and without 1.0 vol.% of carbon nanotubes. For all experiments, the pool was maintained at 7℃ and 100℃ for halocarbon refrigerants and water respectively. The heat flux varied from 10 to 80 kW/m^2 with an interval of 10 kW/m^2.

3.1 Confirmation of test result with correlations

Figure 3 shows the comparison of the measured data of R22 with Stephan and Abdelsalam (1980), Cooper (1982) and Jung et al.(2003)'s

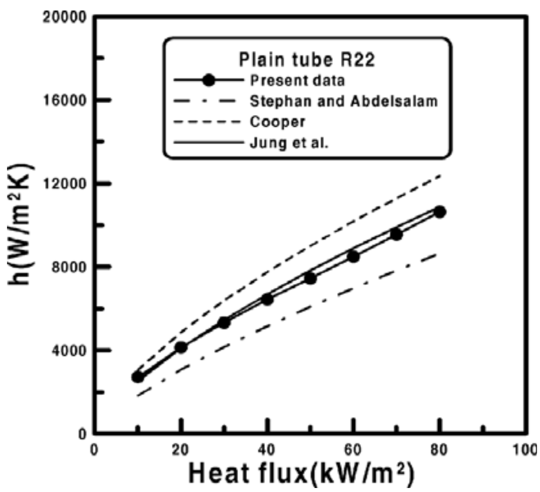


Fig. 3 Comparison of test result for R22 with some correlations without CNTs

correlations. One can easily see that R22 data agree with the well known correlations with a deviation of less than 20%. Recently, Jung et al. (2003) measured nucleate boiling heat transfer data of 8 halocarbon refrigerants and made a specific correlation for halocarbon refrigerants based upon those data. As one can see in Figures 3-6, their correlation agreed best with halocarbon refrigerants. Figure 7 shows that the measured data for water agree well with Stephan and Abdelsalam (1980)'s correlations showing 9% deviation. This comparison indirectly confirms the validity of the test data.

3.2 Effect of CNTs on horizontal plain tube

Figures 4~7 show the HTCs of R22, R123, R134a, and water with and without CNTs and Table 1 lists the heat transfer enhancement with CNTs at heat fluxes of 20 and 60 kW/m^2. Test results show that nucleate boiling HTCs of all working fluids were increased with the addition of CNTs. Especially, heat transfer was enhanced up to 37% at low heat flux. As the heat flux increased, however, the heat transfer enhancement

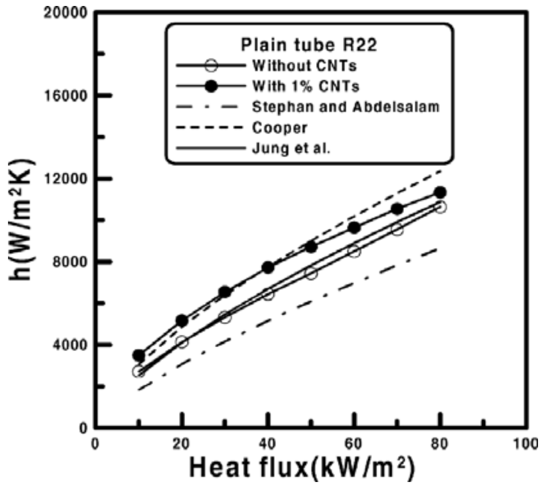


Fig. 4 Boiling heat transfer coefficients with 1.0 vol.% CNTs for R22

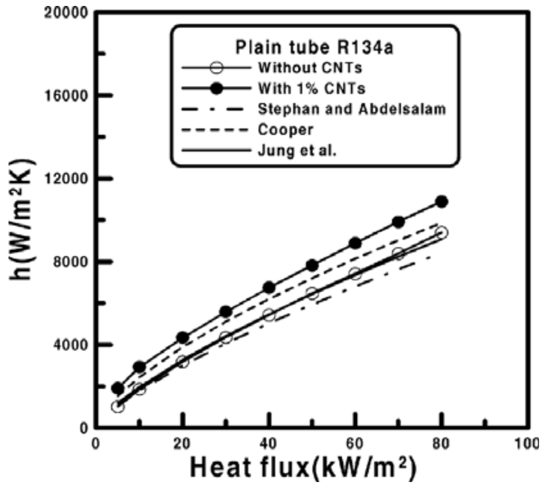


Fig. 6 Boiling heat transfer coefficients with 1.0 vol.% CNTs for R134a

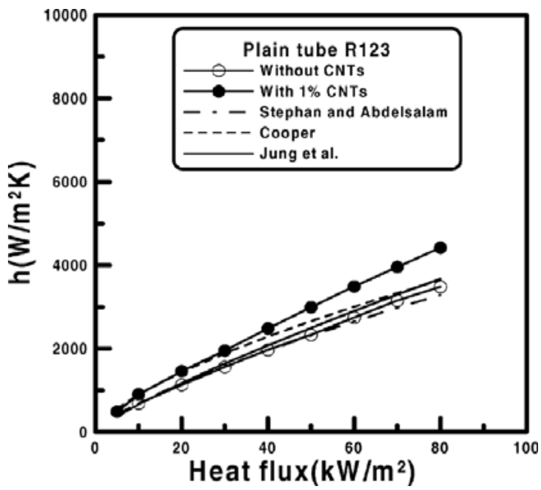


Fig. 5 Boiling heat transfer coefficients with 1.0 vol.% CNTs for R123

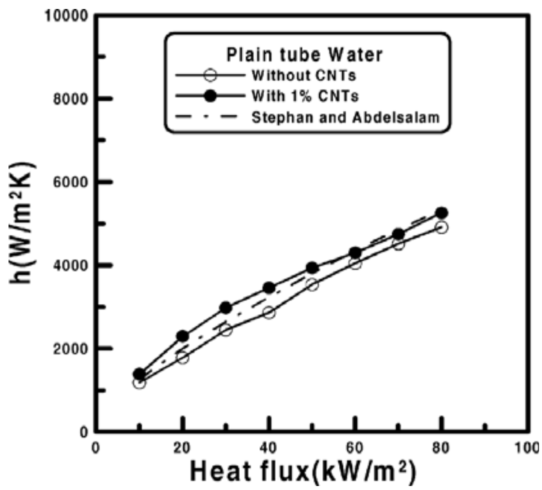


Fig. 7 Boiling heat transfer coefficients with 1.0 vol.% CNTs for water

Table 1 Enhancement in heat transfer coefficients with the use of CNTs for various fluids

Fluids	T_{sat} (℃)	P_{sat} (kPa)	Heat flux (kW/m^2)	HTCs without CNTs ($W/m^2℃$)	HTCs with 1% CNTs ($W/m^2℃$)	Enhancement (%)
R22	7	621.5	20	4145	5169	24.7
			60	8494	9639	13.5
R123	7	44.5	20	1136	1461	28.6
			60	2747	3493	27.1
R134a	7	374.6	20	3184	4349	36.6
			60	7421	8888	19.8
Water	100	101.3	20	1784	2296	28.7
			60	4049	4302	6.3

with CNTs decreased as shown in Table 1. Especially this reduction in heat transfer enhancement was noticeable with water. At 80 kW/m^2, the heat transfer enhancement of water was only 6%.

From this observation, it may be tentatively concluded that CNTs help enhance boiling heat transfer greatly at low heat fluxes. For this case, the bubble generation is not vigorous and CNTs with high thermal conductivity can penetrate into the bubble zone near the surface and touch the surface (or thermal boundary layer) to instantly generate more bubbles. As the heat flux increases, however, more bubbles are generated and the chance of penetration and touching the thermal boundary layer by CNTs becomes low. This was seen visually in an open glass flask containing R123 which was heated at the bottom by an alcohol burner. At low heat flux, addition of CNTs induces explosion of bubbles on the glass surface. As the heat flux increased, the explosion of bubbles by adding CNTs was reduced significantly. Based on this observation, it is expected that large boiling heat transfer enhancement is expected with the addition of CNTs in the boilers (or evaporators) of refrigeration system in which normal heat flux range is 20～30 kW/m^2. For power plants, however, adding CNTs may not result in a large improvement because the normal heat flux encountered in steam generators is very high. For halocarbon refrigerants, the average heat transfer enhancement at all heat fluxes was 18～35% while for water that was 15%.

In 2003, You and Kim (2003) showed that Al2O3 nano particles can increase the critical heat flux of water by 200%. Measurements could not be taken at or near critical heat flux with the present experimental apparatus and hence their finding could not be verified in this study. But present nucleate boiling heat transfer data indicate that CNTs would act as good agitators at or near critical heat flux where bubble generation is minimum. This characteristic may be useful for the safety enhancement in nucleate power plants.

Unlike conventional nano particles, CNTs did not cause fouling on the heat transfer surface. Same measurements were carried out a few times to see the fouling effect over a period of 3 weeks and the results varied little and little contamination was seen on the surface. Conventional nano particles have the affinity to the metal surface but CNTs did not show this kind of behavior. For further confirmation, however, a long term study needs to be carried out.

Finally, it is expected that there are optimum type and amount of CNTs for various working fluids. In the present study, 0.5 vol.% of CNTs was also added for 2 fluids and no change in the results was observed. In the long run, more studies are needed to obtain optimum type and amount of CNTs for boiling heat transfer enhancement of various fluids. Furthermore, dispersion of CNTs in working fluids needs to be studied in conjunction with boiling heat transfer.

4. Conclusions

In this study, nucleate boiling heat transfer coefficients of R22, R123, R134a, and water were measured on a horizontal plain tube with and without 1.0 vol.% of carbon nanotubes. For all experiments, the pool was maintained at 7℃ and 100℃ for halocarbon refrigerants and water respectively. The heat flux varied from 10 to 80 kW/m^2 with an interval of 10 kW/m^2. From the test results, following conclusions are drawn.

(1) For all working fluids, addition of CNTs resulted in heat transfer enhancement. Especially at low heat flux the enhancement was up to 37%. As the heat flux increased, the enhancement decreased due to vigorous bubble generation.

(2) Unlike conventional nano particles, no fouling was observed on the surface with CNTs. For commercial use, however, a long term fouling study is needed.

(3) A further study is needed to determine optimum type and amount of CNTs for boiling heat transfer of various working fluids.

Acknowledgments

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