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Alumina Nanofluid for Spray Cooling Heat Transfer Enhancement

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By

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Dedication

To

My fiancée Vaishali

Acknowledgments

I want to thank my Professor Dr. Frank Pyrtle, III for his guidance and support through my research work and all my friends especially John, Sheetal, Niranjan, Raghu and Saurabh their help.

Thank you

Table of Contents

List of Tables

List of Figures

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ABSTRACT

Nanofluids have been demonstrated to be promising for heat transfer enhancement in forced convection and boiling applications. The addition of carbon, copper, and other high-thermal-conductivity material nanoparticles to water, oil, ethylene glycol, and other fluids has been determined to increase the thermal conductivities of these fluids. The increased effective thermal conductivities of these fluids enhance their abilities to dissipate heat in such applications. The use of nanofluids for spray cooling is an extension of the application of nanofluids for enhancement of heat dissipation.

In this investigation, experiments were performed to determine the level of heat transfer enhancement with the addition of alumina nanoparticles to the fluid. Using mass percentages of up to 0.5% alumina nanoparticles suspended in water, heat fluxes and surface temperatures were measured and compared. Compressed nitrogen

was used to provide constant spray nozzle pressures to produce full-cone sprays in an open loop spray cooling system. The range of heat fluxes measured were for single-phase and phase-change spray cooling regimes.

Chapter One

Introduction

As the microelectronic technology is advancing towards manufacturing more advanced and powerful devices, their thermal management is becoming a growing concern. From airplane and automobile systems to the most common daily use item as cell phones and laptops, all of them generate heat that must be dissipated at specified temperatures to ensure better functionality and long service life.

Many methods of thermal management have be devised, the most ancient being the use of air/ water cooled flat-fin heat sinks. But as the heat generated increased and with the demand of smaller and compact devises with more and more functionalities, these heat sinks staring reaching their saturation limits thereby demanding the need of better and more efficient heat sinks. This in turn led to the design of various cooling systems. With the advances in manufacturing technology, flat-fin heat sinks were replaced with flat-pin fin and pin-finned heat sinks. Also apart from air and water various other cooling

fluids like ethyl glycol, FC72, FC87 and many more were experimented. But all these coolants had there own advantages and disadvantages. Air and water have limited cooling capacities, refrigerants are hazardous to the environment, cryogens are too expensive to use as they need special equipments for usage and also energy intensive production costs.

Apart from enhancing the surface area, the method of cooling was also worked, from simple air cooling to pool boiling and spray cooling. With the growing need of manufacturing more and more compact devises especially in the micro-electronic industry, the air cooled pin-finned heat sinks have long been replaced by liquid cooled heat pipes. These heat pipes are based on pool-boiling technology. Based on simple convection, these heat pipes have their limitation to the distance they can carry the heat from the source to the sink and also cannot have a very complicated structure.

With liquid cooling, some of the means of increasing the thermal conductivity and thereby enhancing the critical heat flux in pool boiling conditions were to increase the surface roughness, mainly by etching the surface or by creating micro-nano structures on it. Other included producing vibrations on either the surface or liquid or both, thereby

assisting the bubbles to move off the heated surface. Application of electric field also helped in departing the bubbles from the heated surface.

Experimentally many researchers have been observed that spray cooling is almost up to six times more efficient than pool boiling. Hence if we could apply spray cooling technology instead of pool boiling to our micro electronic devices, we would be able to keep our devises cool thereby keeping them away from the danger of being burning out and hence assuring their longer life and better functionality.

Now we know that many of the metals and non-metals like gold, silver, copper, aluminum and carbon have very high thermal conductivity as compared to the cooling fluids like water and ethyl glycol. Hence if we add small amount of these high thermally conducting material in base fluids like water, we would be able to increase the thermal conductivity of the base fluids to a great extent. But in order to get a homogenous mixture and to avoid sedimentation, increased pressure drop, erosion, fouling of common slurries, the particle size of the conducting material should be really small. With the advancement in manufacturing technology, synthesizing particles of

nano-size is possible. These mixtures of nano-size particles in the base fluid are termed as NANOFLUIDS. Research has found that nano-fluids show very high thermal conductivity as compared to base fluid. They have emerged as promising coolants for managing the ultra high cooling requirements of the present day and recent future.

The thermal conductivity of the base fluid can be increased as up to two times by adding very small amount of nano-particles of high thermal conductivity material. This is due to the high thermal conductivity of the nano-size materials, enhancing the convective process dispersion, high surface to volume ratio due to which the atoms are located almost at the surface, also it has been observed that the thermal conductivity of the nano-fluids increases with the increase in temperature.

Now if we combine two most efficient cooling technologies, i.e. spray cooling and nanotechnology, we can attain amazing results in the field of Thermal Sciences.

But Spray Cooling by itself is a very complex phenomenon. There are many factors which affect spray cooling heat transfer including the type of nozzle used as every nozzle has a unique droplet size

distribution, droplet number density, droplet impact velocity and flow rate/ nozzle pressure. Other factors affecting spray cooling heat transfer include the orientation of heater surface, the droplet impact angle, the heater material (its properties like thermal conductivity), its surface roughness, the fluid used, the distance between the nozzle and the heater surface, the type of spray produced (full cone or flat cone) and the presence of other nozzles and walls. Hence all these factors are taken into consideration while designing a spray cooling setup as that optimum performance could be achieved.

In the present research the efficiency of alumina nano-fluid as a spray cooling fluid was explored. The nano-fluid was synthesized in the laboratory by mixing 40-50 nm size Alumina particles and de-ionized water in ultrasonic bath for 24 hours. The fluid was sprayed in the form of mist with the substrate placed vertically against the nozzle. The performance of the nanofluid as compared to plain de-ionized water was explored at various pressures and flow rates varying from 25psi to 60psi and 0.001334GPM to 0.002034GPM.

It was observed the nanofluid showed better performance as compared to water at lower temperatures and Heat fluxes but at heat fluxes higher 11W/in^2 and Surface temperatures above 70°C the

performance of nanofluid deteriorated as compared to water. But on the other hand much higher CHF were achieved for nanofluids as compared to water.

Chapter Two

Literature Review

During the pool boiling experiments, Das et al. (2003) observed that the boiling performance is systematically deteriorated with the increase of particle concentration in nanofluids. This was explained basically by the change of surface characteristics during boiling due to trapped particles on the surface which were smaller in magnitude as compared to the surface roughness itself.

For pool boiling experiments on a flat surface using Alumina Nanofluids, Bang et al. (2005) observed that because of delayed boiling activity CHF was enhanced by ~32% and ~13% for horizontal and vertical boiling. They also observed that the addition of nano-particles decreased the nucleate pool boiling heat transfer which was mainly attributed to the change of surface roughness which caused a fouling effect on the surface with poor conduction in single phase heat transfer.

On the other hand the observations of Xuan et al. (1999) for hot wire experiments were completely different. They carried out experiments using Cu nanoparticles in water and observed that the ratio of thermal conductivity of the nanofluid to the base fluid varied from 1.24 to 1.78 with the increase in particle concentration from 2.5% to 7.5%.

Also Zhang et al. (2006) had similar observations when they carried out experiments with hot wire using Au/toluene, Al_2O_3/water, TiO_2/water, CuO/water and CNT/water. They observed that the effective thermal conductivity and thermal diffusivity increased with particle concentration.

For experiments with micro-channel heat sinks, Jang et al. (2006) observed that diamond nanofluids increased the enhanced the cooling performance in micro-channel heat sinks by 10%. This increase in cooling performance was basically attributed to the decrease in thermal resistance and the temperature difference between the heated micro-channel wall and the coolant due to the addition of diamond nano-particles in the base liquid water.

According to Chon et al. (2005) observations for Alumina Nanofluids, particle Browning motion was attributed as the most dominant factor

governing the high thermal conductivity of nanofluids. According to them, the Brownian motion of the particles increases at higher temperatures, which leads to the increase in thermal conductivity of nanofluids at higher temperatures.

Palm et al. (2006) observed a significant increase of nearly 25% in thermal conductivity of Alumina nanofluid by adding only 4% volume fraction of nanoparticles in the base fluid water. They also observed that the temperature dependent nanofluid showed greater heat transfer responses as compared to temperature independent nanofluids. They found that with the increase in wall heat flux, the wall shear stress decreases whereas the average heat transfer coefficient increases for temperature dependent nanofluids.

Keblinski et al. (2002), in order to better understand that why the thermal conductivity of nanofluids increases with the decrease in the particle size of the nanoparticles, explored four possible causes i.e. Brownian motion of the particles, molecular level layering of the liquid at the liquid particle interface, the nature of heat transport in nanoparticles and the clustering of nanoparticles. In the experimentation that followed, they observed that it is the ballistic nature of nanoparticles rather than diffusive combined with direct or

fluid mediated clustering, that better explains their thermal behavior. Zhou (2004) in his pool boiling experiments with Copper nanofluids along with acoustic cavitations around a heated horizontal copper tube observed that nanofluids enhanced single phase convection while the boiling heat transfer was reduced. The acoustic field helped in increasing the heat transfer.

Hwang et al. (2006) in their transient hot wire pool boiling experiments with Multi Walled Carbon Nanotube (MWCNT) in water, CuO in Water, SiO_2 in water and CuO in ethylene Glycol, observed that the thermal conductivity of MWCNT water based nanofluid increased up to 11.3% with only .01 volume fraction of MWCNT. They observed that the thermal conductivity enhancement of nanofluids depend both on the thermal conductivity of nanoparticles as well as the thermal conductivity of the base fluid.

Kang et al. (2006) in their pool boiling experiments with heat pipe observed that the addition of silver nanoparticles decreased the thermal resistance of the heat pipe from 10 – 80% as compared to plain DI-water. In their experiments the concentration of nanoparticles varied from 1mg/l to 10mg/l.

Wen et al. (2004) in their pool boiling experiments with Alumina Nanoparticles in DI water in the laminar flow regime of Copper tube observed that the use of nanoparticles enhanced the heat transfer in the laminar flow regime. Heat transfer enhancement was observed along with increase in Reynolds Number and particle concentration. Enhancement was particularly observed to be significant in the entrance region and decreased with axial distance inside the tube. The main reasons were particularly attributed to particle migration which resulted in non-uniform distribution of thermal conductivity, viscosity field and reduction in thermal boundary layer thickness.

Vadasz et al. (2005) investigated theoretically the reasons for heat transfer enhancement in nanofluids by applying hyperbolic heat conduction constitutive relationship and comparing the same with the corresponding Fourier conduction results. They proposed that the hyperbolic thermal conduction was the reason behind high heat transfer in nanofluids.

Das et al. (2003) in their investigation on pool boiling of Alumina-Water nanofluids in horizontal narrow tubes (4 and 6.5mm in Diameter), observed that in narrow tubes the deterioration in boiling performance was less as compared to larger diameter tubes which

made the narrow tubes less susceptible to local overheating in convective applications. The main reason attributed to the behavior of nanofluids was the difference in bubble sliding mechanism in the case of narrow tubes.

Jang et al. (2003) in their theoretical study on nanofluids proposed Brownian motion of nanoparticles at molecular and nano-scale levels as the key mechanism governing the thermal behavior of nanofluids.

Vassallo et al. (2004) in their pool boiling experiments with Ni-Cr wire, incorporating silica-water nano and micro fluids, observed a remarkable increase in Critical Heat Flux (CHF) for both nano and micro solutions of silica particle in water. But no appreciable difference was observed in powers lesser than CHF. Also observed were stable film boiling at temperatures close to the wire melting point which were achievable only with nanofluids and not with micro fluids of silica and water.

Hwang et al. (2006) in their investigation utilized Multi-walled carbon nano-tube (MWCNT), fullerene, copper oxide, silicon dioxide and silver, to produce nanofluids for enhancing the thermal conductivity and lubrication properties of base fluids like DI water, ethylene glycol, oil,

silicon oil and poly-a-olefin oil (PAO). They observed that the Thermal conductivity of nanofluids increases with increasing the particle volume fraction except for water-based fullerene nanofluid which has a lower thermal conductivity as compared to the base fluid, 0.4 W/mK. Also they observed that by addition of fullerene in oil, the extreme pressure of nanofluids increases up to 225% and also that the Stability of nanofluid is influenced by both the characteristics of the base fluid and the suspended nanoparticles.

Kwak et al. (2005) in their investigation on Copper Oxide – Ethylene Glycol Nanofluid, observed that the thermal conductivity measurements demonstrated that substantial enhancement in thermal conductivity with respect to particle concentration is attainable only when particle concentration is below the dilute limit.

Pasandideh-Fard et al. (2001) studied the impact of water droplet on hot stainless surface using both numerical and experimental model. They observed that increasing the droplet velocity enhanced the heat flux from the substrate by a very small amount. They observed that by increasing the droplet velocity made the droplet spread more on impact thereby increasing the wetted area for enhancing heat transfer.

Bernardin et al. (1997) carried out Spray Cooling experiments on surfaces with different roughness, viz. polished, particle blasted and rough sanded with average surface roughness values as 97, 970 and 2960 respectively. They observed that the temperature corresponding to critical heat flux remained independent of the surface conditions; on the other hand Leidenfrost Point Temperature was sensitive to surface conditions. The protruded features on the rough surface ruptured the liquid film, thereby reducing the pressure beneath the droplet hence yielding lower LFP temperatures as compared to polished surfaces. They observed that surface features influence the boiling regimes of the droplets in two major ways, which included violent breakup of the spreading liquid film at high temperatures corresponding to film boiling and upper portion of transition boiling regimes and increasing nucleation site density at lower temperatures corresponding to nucleate boiling and lower portion of transition boiling regime. According to them enhanced nucleation at lower temperatures was largely responsible for decreasing droplet lifetime on rougher surfaces.

Kim et al. (2004) performed heat transfer experiments with air and evaporative spray cooling of plain and micro-porous coated surfaces on flat and cylindrical heaters. They determined the heat transfer coefficients as a function of heat flux and studied three water flow

rates (1.25, 1.75 and 2.40 ml/min) for the flat heater and one rate (3.0 ml/min) for the cylindrical heater, maintaining the air pressure of 7 psig (48 kPa) at the inlet of the nozzle. They used Micron-size aluminum particles to build the micro-porous structures on the heated surfaces. They observed that the combination of evaporative cooling and coated micro-porous surface enhanced the heat transfer coefficient by up to 400% as compared to dry air cooling on plain surface, also they observed that the micro-porous coating extended the dry-out heat flux significantly (~21 kW/m^2) over the plain surface (15 kW/m^2). They found that when the heat flux was lesser than 10 kW/m^2, water spray amounts (1.25–2.4 ml/min) had no effect on evaporative cooling for micro-porous coating; However, for higher heat fluxes, the heat transfer increased with water flow rate for both plain and micro-porous surfaces.

Oliphant et al. (1998) in their study on heat transfer compared Liquid jet and spray impingement cooling experimentally in the non-boiling regime. They found that jet heat transfer was dependent on the number and velocity of the impinging jets, whereas, spray cooling on the other hand demonstrated a strong dependence on mass flux and to some amount on droplet velocity as well. In comparison of the two cooling techniques, spray cooling emerged as the winner as it could

provide the same heat transfer coefficient as jets at a substantially lower mass flux, reason being, the unsteady boundary layer resulting from droplet impact and evaporative cooling.

Lin et al. (2003) carried out experiments on a closed loop spray cooling system, comprising of eight miniature nozzles and a 1 ·2 cm^2 target cooling surface and using FC-87, FC-72, methanol and water as the working fluids. They observed that the spray cooling critical heat fluxes reached up to 90 W/cm^2 with fluorocarbon fluids and 490 W/cm^2 with methanol and higher than 500 W/cm^2 for water. Air purposely introduced in the spray cooling system with FC-72 fluid has a significant influence on heat transfer characteristics of the spray over the cooling surface. They observed that non-condensable gases adversely affect the overall heat transfer of the closed loop spray cooling system at heat fluxes lower than CHF because of a higher thermal resistance to condensation heat transfer.

Table 1: Nomenclature

q''	Heat Flux (W/m^2)
h	Heat Transfer Coefficient (W/m^2K)
CHF	Critical Heat Flux (W/m^2)
ΔT	Temperature Difference (°C)
T_s	Surface Temperature (°C)
T_f	Fluid Temperature (°C)

Chapter 3

Preparation of Nanofluid

Nanofluids are emerging as one of the most promising cooling reagents in the present world. The reason for using alumina as the nano-particles in the present study is that it is very stable and can be easily dispersed in water, thus forming a colloidal solution. Though silver and gold have higher thermal conductivities, they are much more expensive than alumina. Copper oxide, though not very expensive, is very unstable, and requires a dispersant to form a colloidal solution with the base fluid. Carbon Nano Tubes have very high thermal conductivity but it is difficult to disperse them in base fluid as they entangle and agglomerate to settle.

The colloidal solution of alumina nanofluid was prepared by dispersing Al_2O_3 Nano Dur® Nano-particles in the base fluid de-ionized water. The nanoparticles used were manufactured by Nanophase Technologies Corporation. Following are the properties of the nanoparticles used:

Purity = 99.5+%

APS (Average Particle Size) = 45 nm (determined from SSA)

SSA (Specific Surface Area) = 45 m^2/g (BET)

Appearance = Off-White to Gray Powder

Bulk Density = 0.26 g/cc

True Density = 3.6g/cc

Morphology = Spherical

Crystal Phase = 70:30 Delta: Gamma

An Ultrasonic Cleaner FS140 was used to disperse the nanoparticles in water. To ensure proper homogenization of the alumina nanoparticles and to obtain stable, uniform solution, the nanofluid was ultrasonically mixed for 24 hours. The ultrasonic mixing of the nanofluids also raised their temperatures much above ambient due to which we observe a negative temperature difference in many of the graphs with nanofluids.

The nanofluids tested had ratios of 0.25% w/w, 0.2525% w/w and 0.505% w/w and were tested at nozzle pressures of 30psi and 40psi. The results were compared to those of pure water and various plots were made.

Chapter 4

Experimentation

4.1 Experimental Set-up and Procedure

The heater surface was made of copper block of dimensions 25.4x25.4x3mm. A serpentine mesh of nichrome wire (42 Gauge/ 0.0635mm diameter) was used as the heating element. The nichrome wire mesh was laid in-between the copper block and the Teflon substrate which was in turn connected to a DC Power supply (Agilent Technologies, N5771A 300V/5A/1500W) to provide electric current. To ensure that the setup was electrically insulating, a high thermally conductive, electrically insulating silicon paste (OMEGATHERM® 201) was applied between the nichrome wire and the copper block. The paste also ensured a secure bonding of the copper block to the Teflon substrate.

Since the Biot Number for the copper block under the test conditions, was lesser than 0.1, hence a condition of negligible temperature gradient was assumed within the copper heater surface.

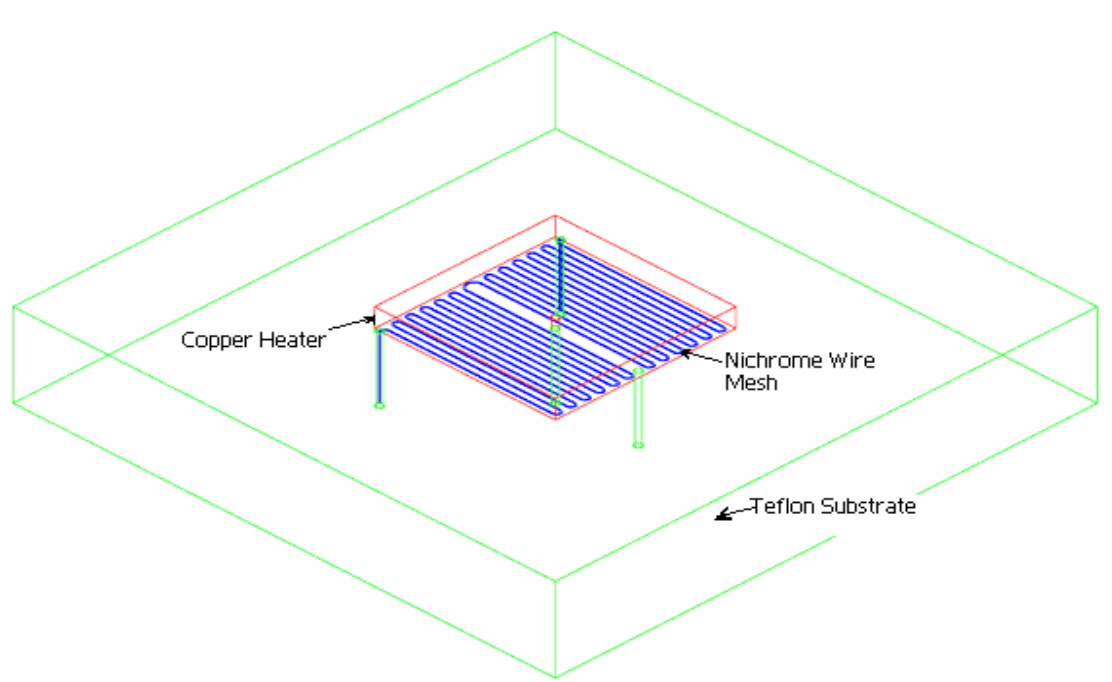

Figure 1: Teflon Substrate with Copper Block and Nichrome Resistance Heater

Four thermocouples (K-Type, 30 Gauge) were used to constantly measure the surface temperatures of the copper block, fluid temperature and ambient temperature. As indicated in Figure 2, thermocouple T1 was placed in a 2mm deep inside the copper block. Thermocouple T4 constantly measured the temperature of the fluid.

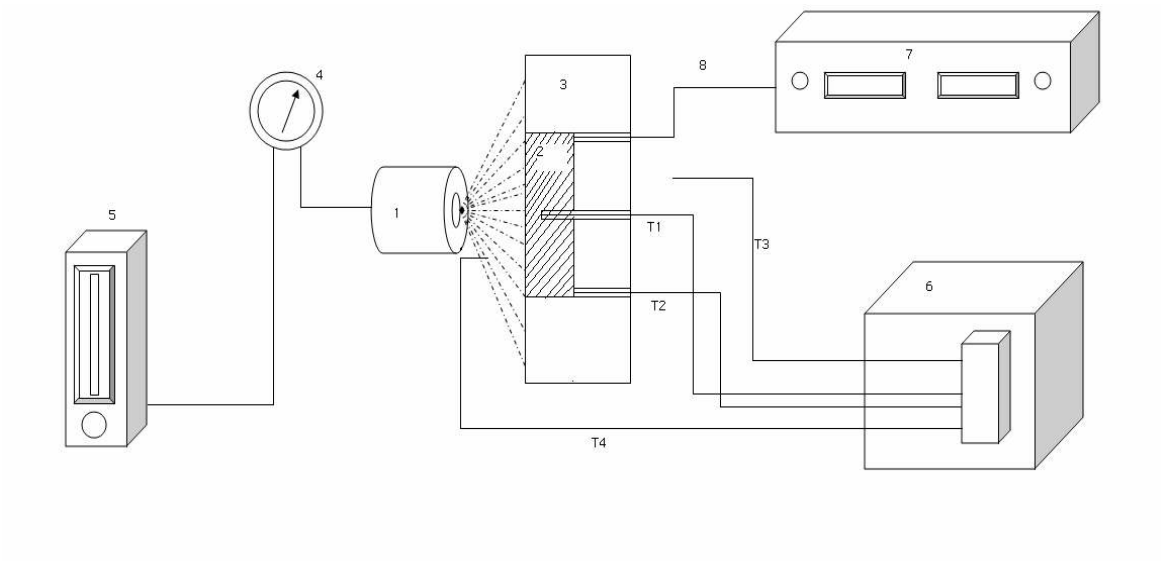

Figure 2: Experimental Setup

[T1, T2, T3, T4] Thermocouples; [1] Nozzle;

[2] Copper Block (1”x1”x3mm); [3] Teflon Substrate;

[4] Pressure Gauge; [5] Flow Meter; [6] Data Acquisition Board;

[7] DC Power Supply; [8] Nichrome Wire

The thermocouples were connected to a data acquisition board (National Instruments, NI-SCXI-1303) which was connected to a computer where LabView software was used to interpret the data graphically and acquire the thermocouple readings.

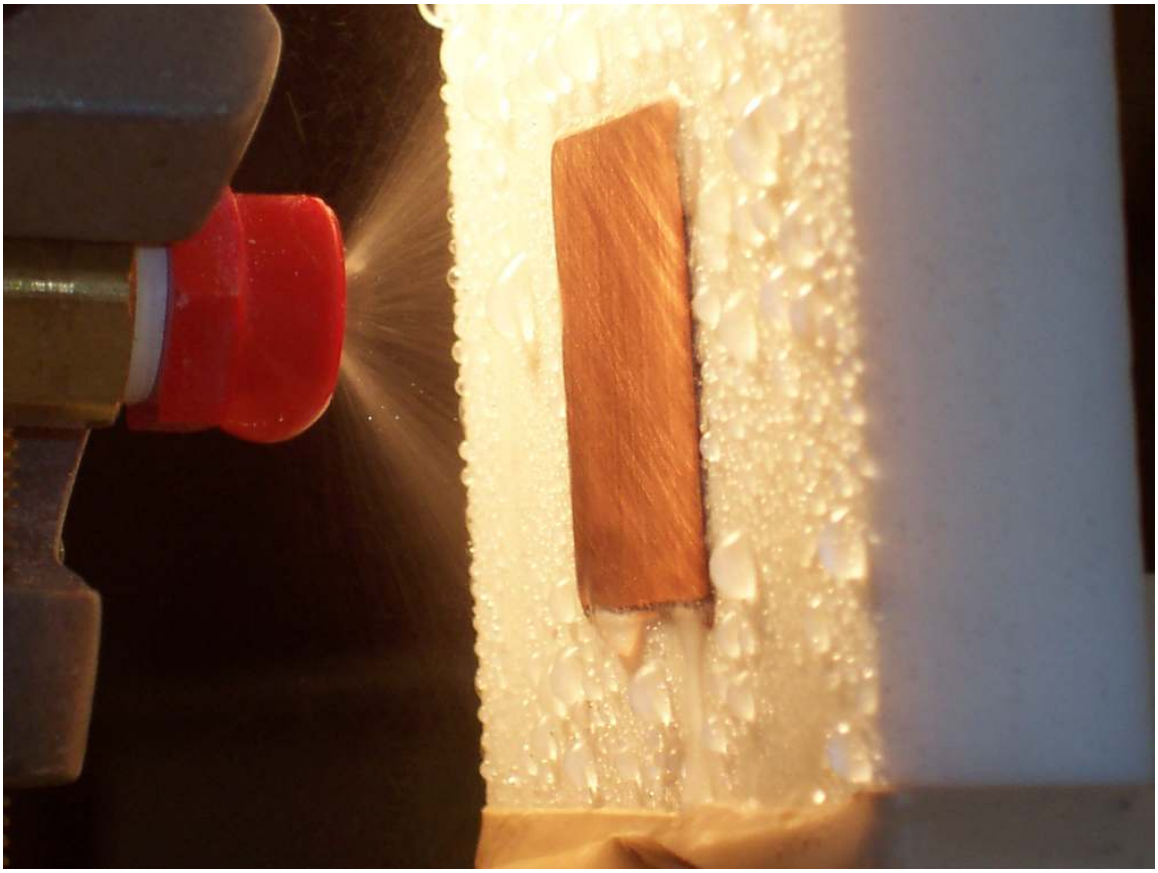

Figure 3: Nozzle and Heater Surface

Figure 4: Experimental Setup

Table 2: Flow Rates

Pressure (psi)	Flow Rate (gpm)	Mass Flux(l/m^{2}s)
30	0.001466	0.14336
40	0.001519	0.148543
50	0.001731	0.169274
60	0.002034	0.198905

A full cone misting spray nozzle was used to spray the fluids onto the copper block. The nozzle used was designed to deliver a uniform size and spatial distribution of droplets within the spray pattern. The average droplet size produced by the nozzle was 75 micron. The fluid flow rates and volume fluxes impinging the copper surface at various pressures are presented in Table 1. The nozzle was kept at a distance of 25.75mm away from the surface of the copper block.

A flow meter was used to control the flow rate and pressure of the fluid, which is expelled through the full-cone spray nozzle. Fluid reached the nozzle and departed as a high velocity spray. A compressed nitrogen cylinder provided pressure to the pressure tank filled with working fluid. Due to pressurization of the tank, the working fluid was forced out of the pressure tank through the flow

circuit. The compressed nitrogen cylinder was fitted with a two-stage valve, which provided constant pressure to the pressure tank, producing constant flow rates during the experiments.

The experiments were first carried out using de-ionized water as the cooling fluid. The experiments were run at various nozzle pressures ranging from 30psi to 60psi. The flow rates corresponding to these nozzle pressures are presented in Table 1. Graphs were plotted for various flow rates of water as heat flux vs. temperature difference (Figure 3), heat transfer coefficient vs. heat flux (Figure 4) and Heat transfer coefficient vs. temperature difference (Figure 5).

The second set of experiments was performed using nanofluids at nozzle pressures of 30 and 40psi. To avoid agglomeration or settling of the particles, the nanofluids were used immediately after synthesizing them in the ultrasonic bath. While using the nanofluids, care was taken to properly clean the surface of the copper block before and after the experiment with emery paper (Grade P320 with average particle size 36 micron). Also immediately after completing the experiment with nanofluid for a particular set of conditions, the nozzle and the complete pipe section along with flow meter was thoroughly cleaning by rinsing with de-ionized water.

4.2 Uncertainty Analysis

The uncertainties in the measurement parameters were analyzed using the error propagation method. The uncertainty in the heat flux measurements was (0.1% + 300mV)*(0.1% + 15mA) W/m^2. The uncertainties in pressure measurement, temperatures and distance between the nozzle and heater surface were found to be ±2.7psi, ±1.56°C and ±1mm respectively.

Chapter 5

Results and Discussion

The objective of the research work was to study the effectiveness of nanofluids for spray cooling. The first three graphs (Figure 5, 6 and 7) show the spray cooling curves for water at various nozzle pressures varying from 30psi to 60psi. The curves were observed to be quite linear at lower heat fluxes indicating single phase convection as the primary mode of heat transfer. It was observed that with water, as the flow rate increased, the heat flux increased and more heat was dissipated at lower temperatures than with the nanofluids.

Figures 8, 9 and 10 show a comparative behavior of nanofluids at pressures of 30psi and 40psi. Figures 11, 12 and 13 compare the behavior of nanofluids with mass concentration of 0.25% and water at nozzle pressures of 30psi. Figures 14, 15 and 16 compare the behavior of nanofluids with mass concentrations of 0.2525% and 0.505% and water at nozzle pressures of 40psi.

Figure 17 shows the hysteresis effect for Nanofluids with particle concentration of 0.5% and pressures of 50psi. It can be observed that when the heater power is increased and decreased the graph does not follow the same curve and a hysteresis is observed, one of the primary reasons being the deposition of nanoparticles and thereby change in surface characteristics.

In Figure 18 the effects of surface features on spray cooling performance is clearly evident. The graph is plotted for water at pressure of 50psi. It depicts the behavior of water in spray cooling when the experiments were run on a clean plain surface and when the surface was contaminated and roughened by nano-particles deposited during the experiments run for nanofluids at the same pressure conditions. It can be seen that the heat transfer performance is affected by the deposition of particles on the heater surface.

Figure 19 shows the heat lost by natural convection through the heater insulation. The heat loss was used to determine the actual heat dissipated by spray cooling.

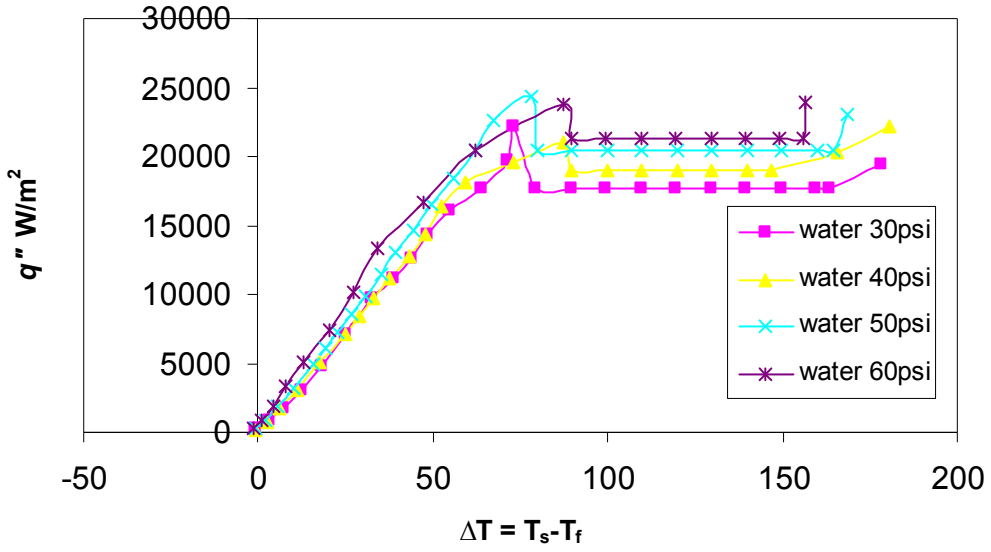

Figure 5: Heat Flux vs. Temperature Difference for Water at Different Pressures

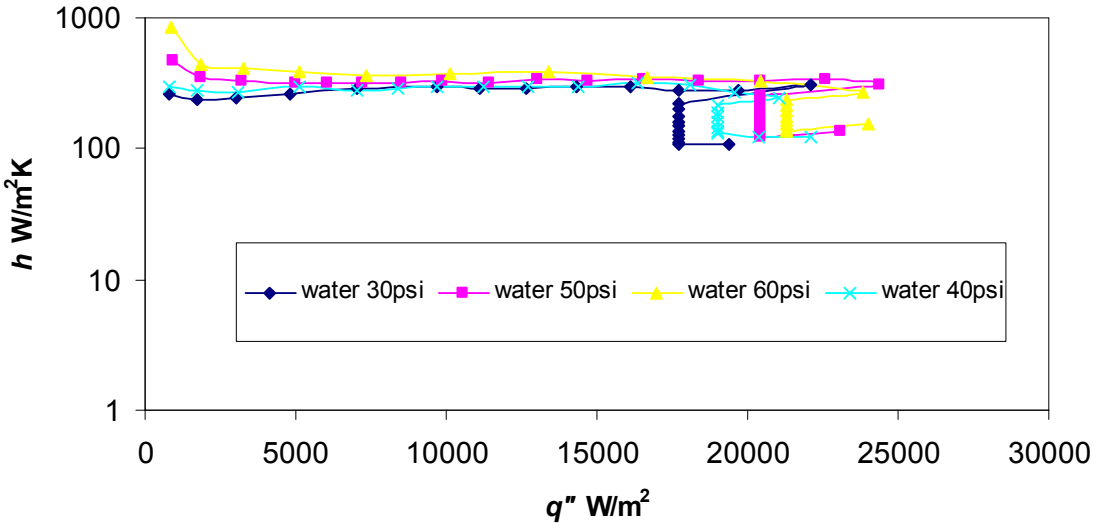

Figure 6: Heat Transfer Coefficient vs. Heat Flux for Water at Different Pressures

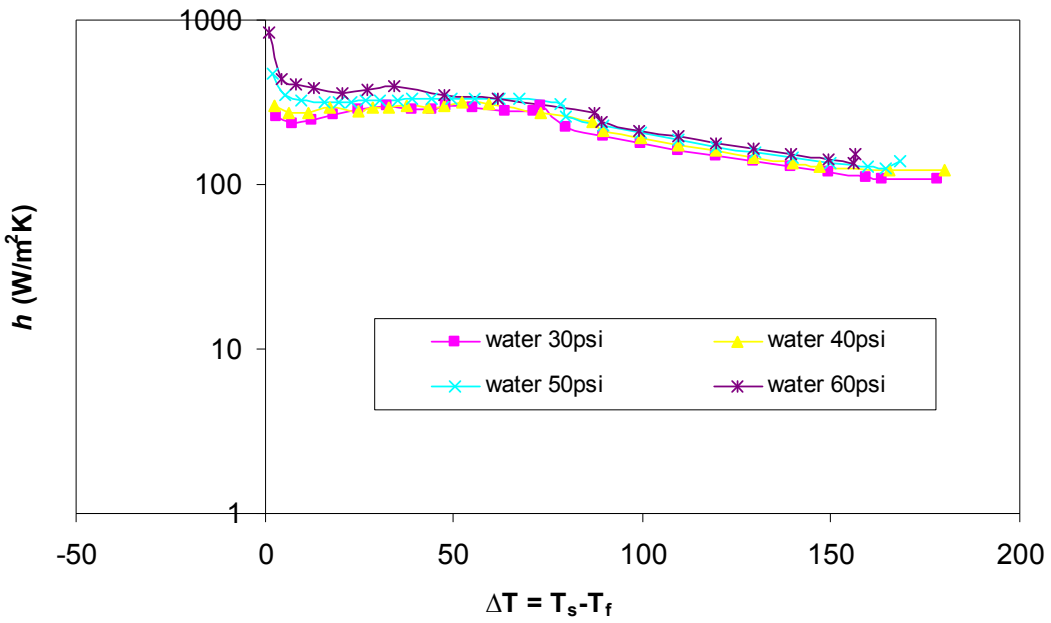

Figure 7: Heat Transfer Coefficient vs. Temperature Difference for Water at Different Pressures

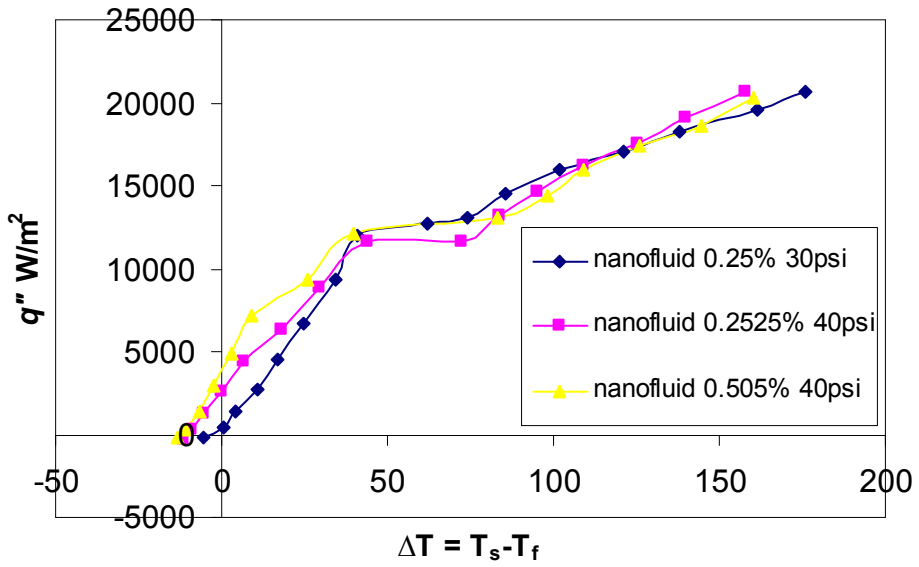

Figure 8: Heat Flux vs. Temperature Difference for Mass Concentration of Nanofluids at Pressures of 30psi and 40psi

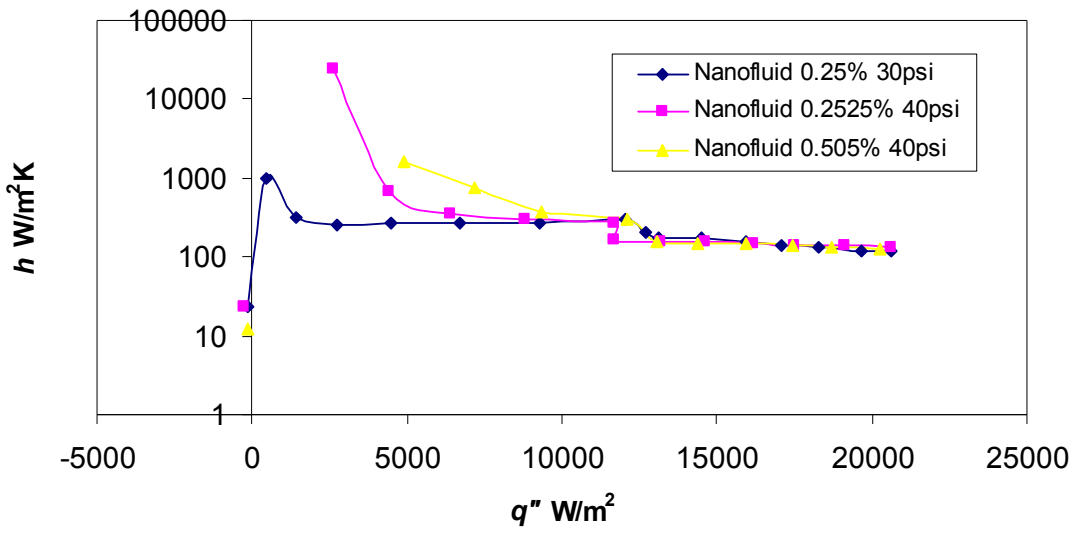

Figure 9: Heat Transfer Coefficient vs. Heat Flux for Mass Concentration of Nanofluids at Pressures of 30psi and 40psi

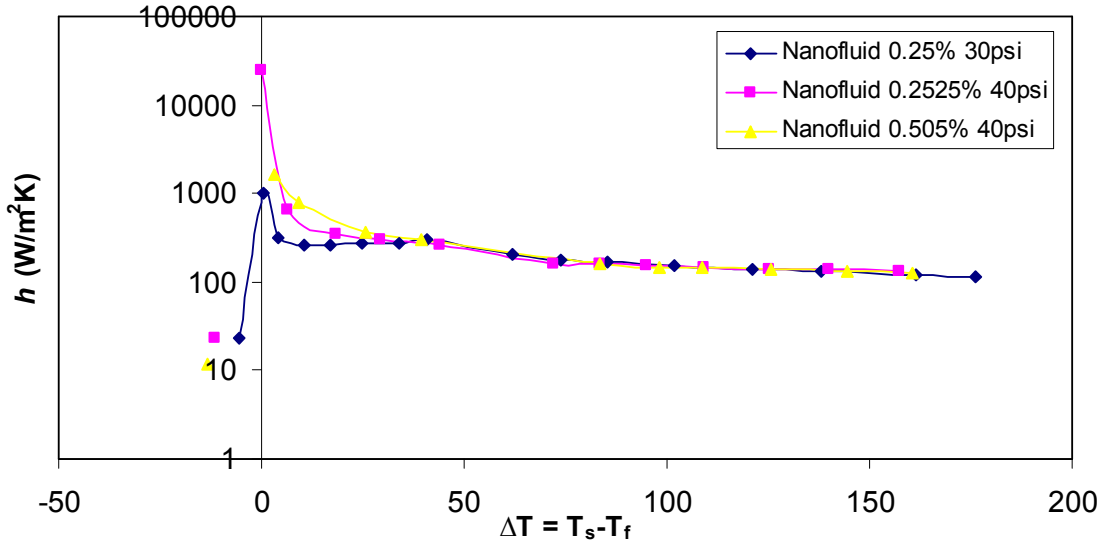

Figure 10: Heat Transfer Coefficient vs. Temperature Difference for Mass Concentration of Nanofluids at Pressures of 30psi and 40psi

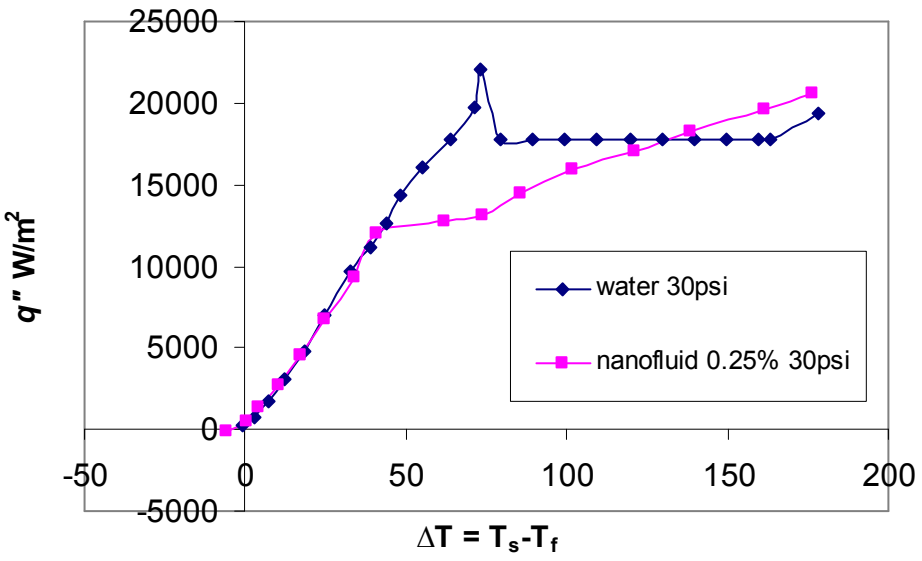

Figure 11: Heat Flux vs. Temperature Difference for Water and Nanofluid at 30psi

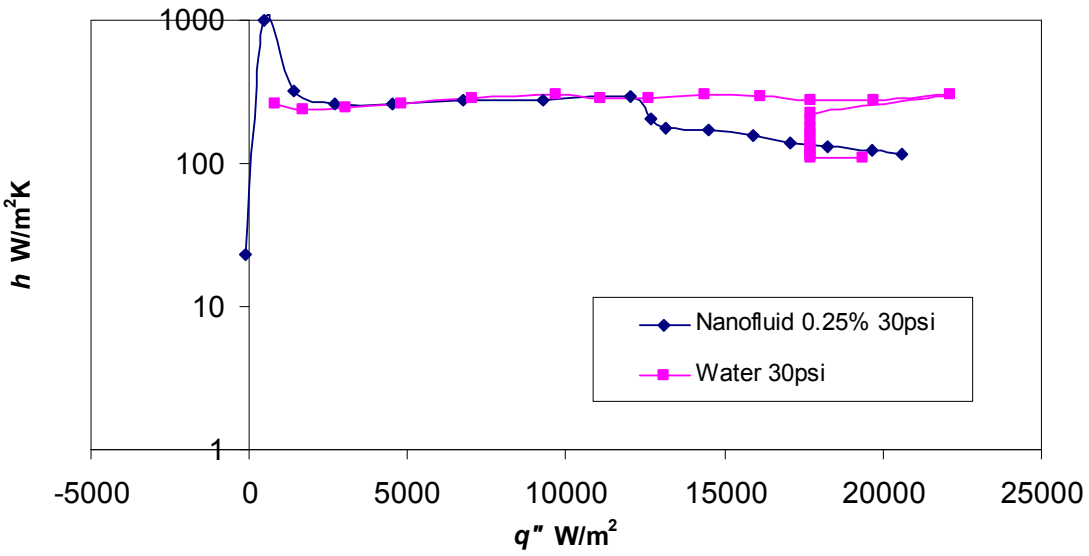

Figure 12: Heat Transfer Coefficient vs. Heat Flux for Water and Nanofluid at 30psi

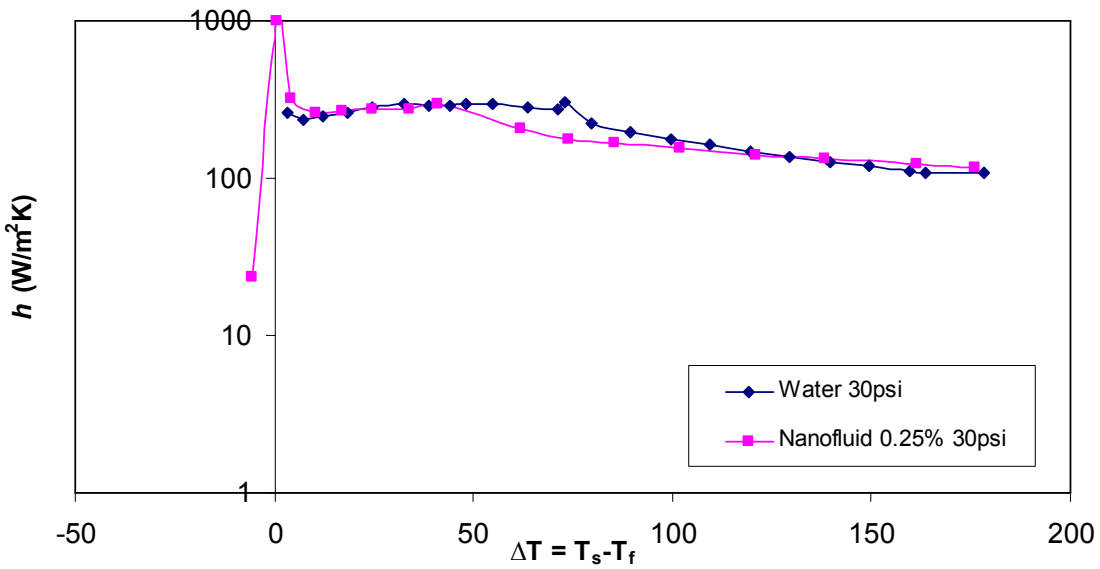

Figure 13: Heat Transfer Coefficient vs. Temperature Difference for Nanofluid and Water at 30psi

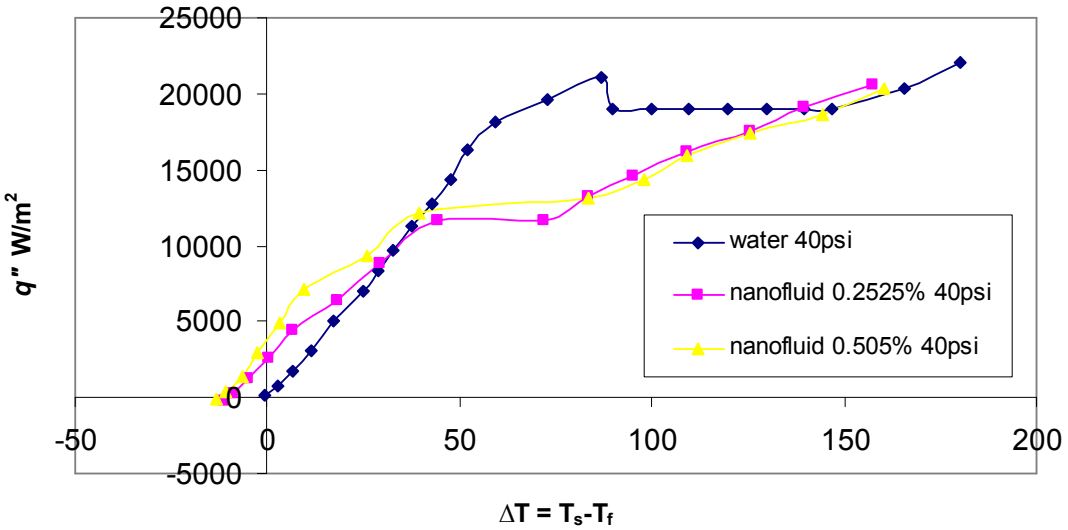

Figure 14: Heat Flux vs. Temperature Difference for Water and Nanofluid at 40psi

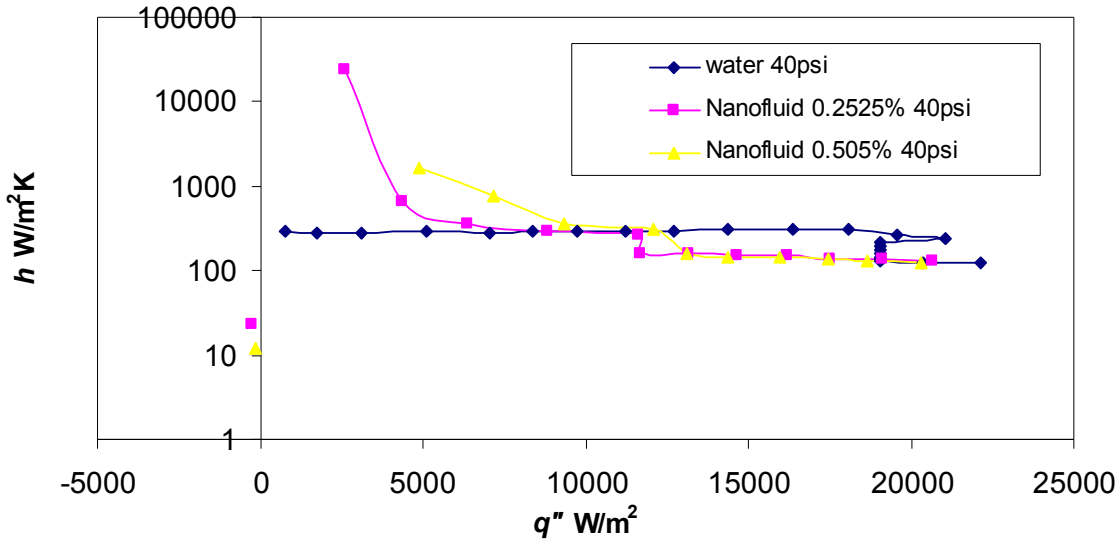

Figure 15: Heat Transfer Coefficient vs. Heat Flux for Water and Nanofluid at 40psi

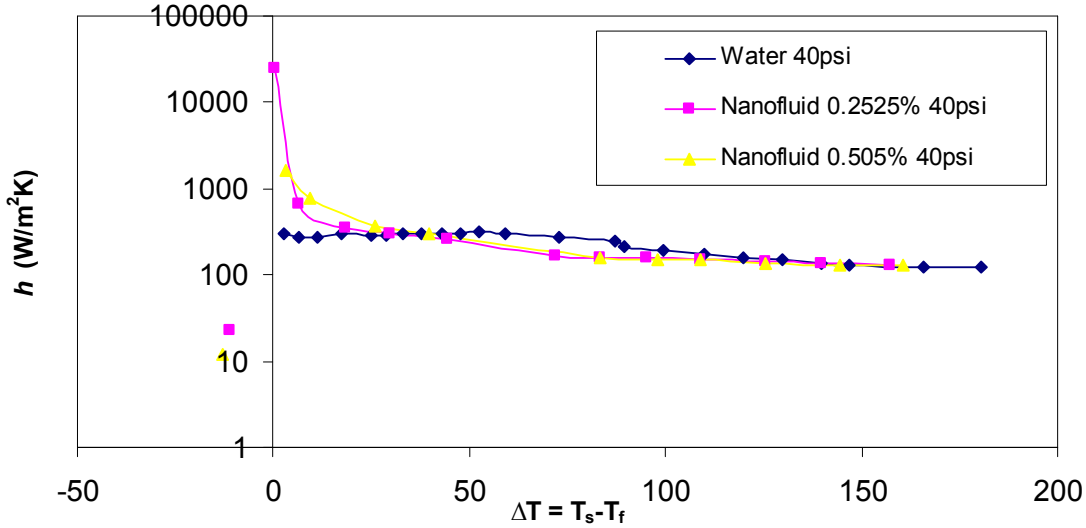

Figure 16: Heat Transfer Coefficient vs. Temperature Difference for Water and Nanofluid at 40psi

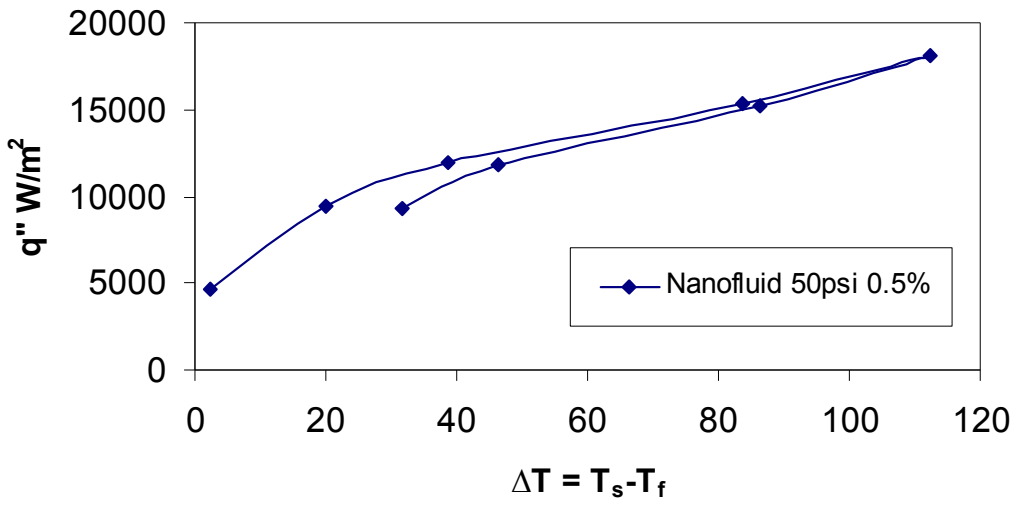

Figure 17: Heat Flux vs. Temperature Difference for Nanofluid at 50psi Showing Hysteresis

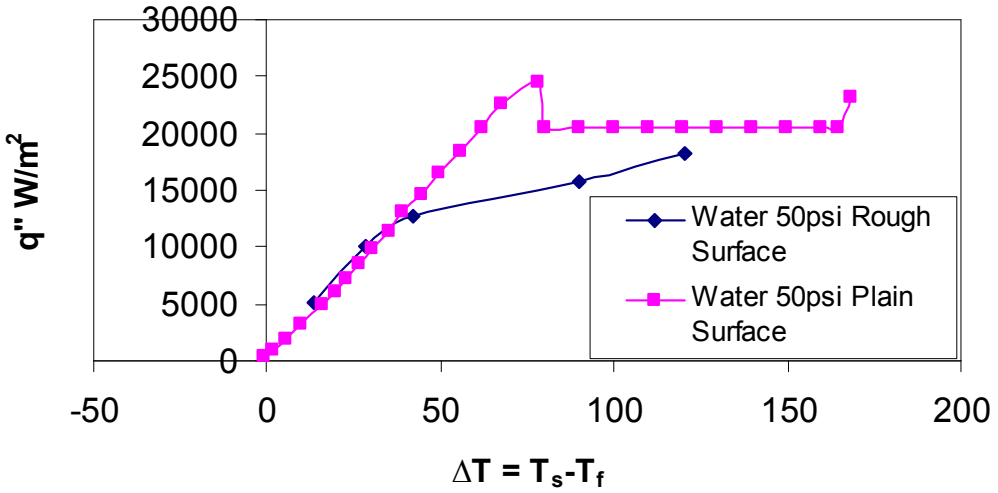

Figure 18: Heat Flux vs. Temperature Difference for Water at 50psi

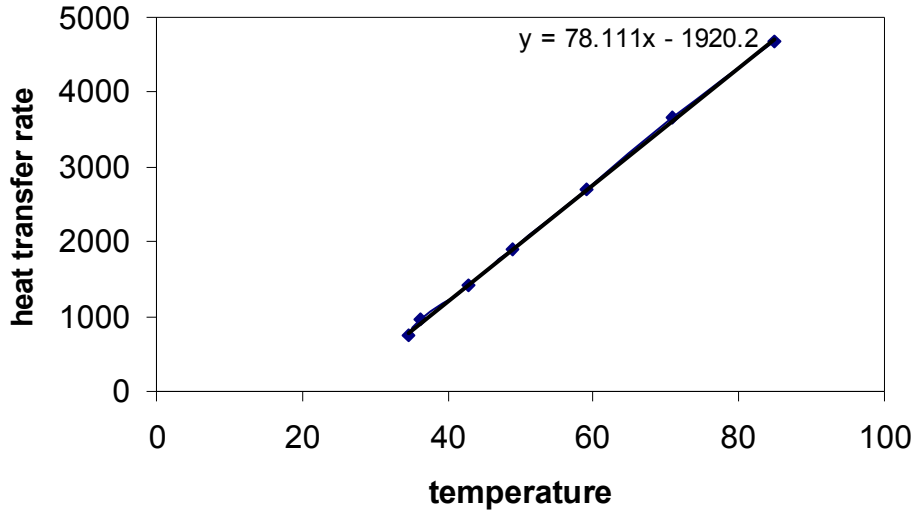

Figure 19: Heat Loss through Heater Insulation

The negative temperature difference of the nanofluids was due to the fact that that ultrasonic mixing raised the temperature of the fluid. The temperature recorded at the end of ultrasonic mixing for 24 hours, was 92°C. These nanofluids began to cool down once the ultrasonic mixing was stopped. Their temperatures were continuously monitored by the thermocouple T4 attached at the spray zone, between the nozzle and the copper heater surface.

In the case of nanofluids, as can be seen in the above figures, the heat transfer performance was different as compared to water. Initially, at lower heat fluxes, more heat was dissipated at lower temperatures. At heat fluxes above 17,000W/m^2 and surface temperatures around 90°C

the heat transfer performance of nanofluids deteriorated as compared to water. Above 17,000W/m^2 the heat transfer performance of nanofluids did not increase as particle concentration increased. This behavior may be explained by changes of the surface roughness due to the impingement of nano-particles on the heated surface. It was observed that as the nanofluids were sprayed, alumina nano-particles began to stick to the heater surface. The reason for sticking of the nano-particles to the copper heater surface may be due to the electrostatic forces. When the alumina nano-particles are sonicated in the ultrasonic bath, they become charged. The charged nano-particles repel each other and hence get evenly distributed in water. These particles impinge on the copper surface, loose their kinetic energy, and attach to the surface. The impinging spray droplets disperse some of these nano-particles from the point of direct impact on the surface but not totally off the surface. Hence a band of nano-particles was observed sticking around the region of direct impact of the spray cone on the surface of copper. As the temperature increased, the fluid started vaporizing as it came in contact with the surface and hence was not able to push the nanoparticles away from the surface, thereby increasing the density of nanoparticles sticking on the surface. These nanoparticles sticking on the surface created local heating regions which facilitated the increase in temperature of the copper block.

Hence as the concentration of nanoparticles increased, for the same flow rates, the thickness of the layer of nanoparticles sticking on to the heater surface increased which led to deteriorated performance of nanofluids at higher temperatures.

As observed in the case of pool-boiling experiments, much higher critical heat fluxes were achieved with nanofluid as compared to pure de-ionized water. Due to experimental limitations and to avoid burnout, for the nanofluids, the points of CHF were not determined as they were beyond the operating limits of the test apparatus used in the investigation. The nano-particles that were stuck to the heater surface were removed before running subsequent experiments.

Chapter 6

Conclusion

Experimental investigation of alumina nanofluids in comparison to water for spray cooling was carried out. The nanofluids were prepared by a two step method in the laboratory by mixing alumina nanoparticles in de-ionized water using ultrasonic vibration in an ultrasonic bath. Experiments were performed at various concentrations and flow rates for water and alumina nanofluids. For water it was observed that with the increase in flow rate, the heat transfer increased and we were able to reach higher heat flux at higher flow rates. In the case of nanofluids, it was observed that initially at lower temperatures and heat fluxes, the nanofluids performed better as compared to water and higher heat-fluxes were reached at lower temperatures. As the surface temperature reached the saturation temperature of water, the performance of nanofluids deteriorated as compared to water. Nanofluids with lower particle concentration demonstrated better heat transfer. The reason for this can be explained as the change in surface roughness of the copper heater as evaporation occurred, leaving nano-particles on the surface.

If nanoparticles can be prevented from sticking to the heater surface, much higher heat fluxes could be achieved with nano-particles as compared to water, and hence certainly nanofluids as the next generation of liquid coolants for cope with the increasing thermal management requirements.

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