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COOLING PERFORMANCE OF MICRO HEAT PIPE USED FOR MOBILE ELECTRONIC DEVICES

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ABSTRACT
The paper presents experimental results of cooling performance of micro heat pipe using infra-red Thermography (IRT) technique for mobile electronic devices and validates those results by using commercial software ANSYS Fluent 16.1 in which 3-D numerical model is developed and solved by Volume of Fluid (VOF) model. The computational simulation is performed at constant heat flux applied to the evaporator section and convective boundary condition at condenser end. The experimental measurements were conducted on micro heat pipe using distilled water *(DW)* and *Al_2O_3/water* Nano-fluids. The set of experiments were performed to measure the surface temperature of the micro heat pipe (MHP) by varying cooling fan voltages i.e. 2V, 4V, and 6V. The effective thermal conductivity, thermal resistance and effectiveness for both phases along the length of micro heat pipe were calculated. It was observed that the use of Nano-fluid reduces 5-10 ^oC surface temperature of MHP as compared to DW, hence use of Nano-fluid as a coolant in MHP improves the thermal performance and reliability of such devices.

KEYWORDS: *Cooling Performance, Micro heat pipe, Nanofluid, Surface temperature, Thermal Resistance.*

I. INTRODUCTION
Mobile electronic devices, such as smartphones, mini laptops, and tablets are increasing multifunctional and fast. The heat load of such modern electronic devices had increased as it becomes smaller and denser, however, high heat generated by internal components and overheating of localized parts in devices has become problematic [1]. This induces thermal stresses in such devices, leading to failure in the components and thus require thermal management to improve reliability and prevent premature failure. Hence, Effective cooling is essential to remove heat generated in such compact electronic devices. One of the most efficient methods to dissipate heat from such compact devices is MHP since they were first introduced by Cotter [2]. MHP has the high efficiency and reliability and maintains temperature uniformity on the surface of such electronic devices. Even though the space constrained in electronic devices, micro heat pipes can provide sufficient cooling effects on microelectronic chips due to their small dimensions, normally with the hydraulic diameter of the order of 100µm and the average length of few centimeters.

Commonly, water is used as the working fluid in MHP for cooling electronics, as it possesses adequate thermal and hydrodynamic properties in the required range of operating temperature. However, the thermal conductivity of water is lower than most of the metals and metal oxides. Therefore, an innovative way to elevate the thermal conductivity of fluids may be the addition of nanometre-sized metal or metal oxide particles into the base fluid, called Nanofluids, was first introduced by choi in 1995. The enhancement of thermal conductivity can be significant and improve the efficiency of fluids used in heat transfer applications [3].

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During the last many years, the use of heat pipes to cool the electronic devices has been rapidly increasing. A heat pipe for electronics application was introduced to cool the CPU of personal desktop computers [4]. Jang et al. [5] showed the comparison of the thermal performance of a flat micro heat pipe using both DW and Al_2O_3/*water* Nanofluid and also the effect of volume fraction on thermal performance. Liu et al. [6] analyzed the heat transfer through the wall of the triangular micro heat pipe with evaporator reaches 63.2% of the temperature difference between its initial temperature and final steady temperature for heat input of 6W using three different working fluids. Shahed et al. [7] introduced an ultra-thin heat pipe for various handheld mobile electronic devices such as smartphones, tablets, camera etc., also developed heat pipe cooling module to eliminate hot spot issue and conducted several experiments to characterize thermal properties. Moon et al. [8] conducted an experiment to evaluate thermal performance on micro heat pipe with a triangular and rectangular cross-section of copper material which operates at a temperature of 60-90^{o}C and dissipates heat up to a thermal load of 7W. Fadhl et al. [9] simulate the two-phase flow and heat transfer phenomena of a wickless heat pipe with water as working fluid and volume of fluid (VOF) model was used in ANSYS Fluent and compared those simulation results with experimental measurements at the same condition. Rahmat et al. [10] investigates and simulates a two-phase model of micro heat pipe using finite element method and calculate the effective thermal conductivity, pressure and velocity field along the length of the micro heat pipe and compared to the literature.

As per the author knowledge, the use of MHP to dissipate heat from mobile electronic devices was rarely reported in the literature. The present experimental study considers the processor chip with a thermal design power (TDP) of 4.5 W. The experiment was conducted taking such heat as input with 50%, 75% and 100% 0f 4.5W in the circular MHP test section and to dissipate these by using cooling fan running at different voltages inputs of 2V, 4V and 6V. Also, DW and Al_2O_3/*water* Nanofluid were used as working fluid to evaluate the cooling performance, so that the reliability of such devices increases. Also, compare those experimental results with a computational simulation using volume of fluid(VOF) model at same operating conditions in ANSYS Fluent.

II. PHYSICAL PROBLEM DESCRIPTION

In this paper, the author showed a real-life problem basically in handheld mobile electronic devices. Nowadays, the uses of these devices are increasing very fast and the heat generation also increases due to the compactness of such devices. The 7th Generation Intel Core Processor generates heat at the processor chip of 10mmX10mm, whose thermal design performance (TDP) is 4.5W. Table 1 shows the technical specification of the processor. The MHP structure is used as an integral part of processor chip and removes heat directly from the area of the chip. The schematic diagram of this physical model is shown in figure 1.

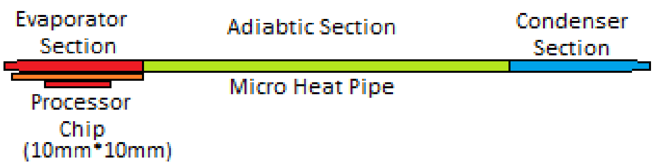

Figure 1: Schematic diagram of a physical model

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Table 1: Technical specification of 7th generation Intel core processor [11]

Specifications	Values
Processor Number	M7-6Y75
Processor Base Frequency (PBF)	1.20 GHz
TDP	4.5 W
Configurable TDP-up	7 W
Configurable TDP-down	3.5 W
Critical Temperature	50ºC
Package Size	20mm*16.5mm

III. DESCRIPTION OF EXPERIMENTAL SETUP

The experimental setup was built to carry out the thermal performance investigation on wickless MHP. The MHP in this study is made of copper (*k=385 W/mk*) with a diameter of 0.8 mm and a length of 80 mm. It mainly divided into three sections namely evaporator, adiabatic and condenser sections of lengths 20mm, 40mm and 20mm respectively. The schematic diagram and working mechanism of the test section of MHP as shown in figure 2.

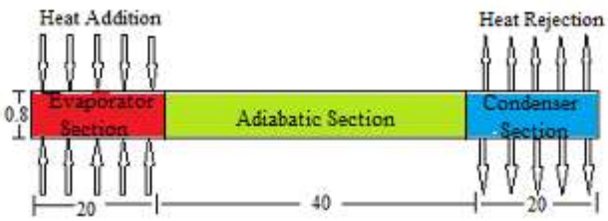

Figure 2: Schematic diagram and working mechanism of MHP

The setup was composed of MHP, a data acquisition system, thermal IR camera, cooling fan and a DC power supplying unit as shown in figure 3.

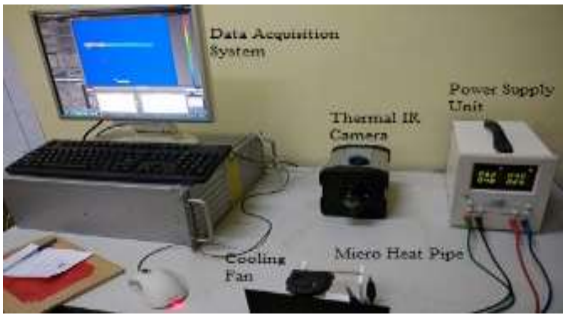

Figure 3: The Experimental Facility

The evaporator of MHP was heated using electric resistance heater and DC power supply unit. A kanthal wire was used as a resistance heater with 0.274mm wire thickness and 7 Ω/feet resistance, was wound around the outer wall of evaporator section. A DC dual tracking power supply unit with a rating of 0-30V/3A was used to provide uniform heating to resistance heater. The condenser of MHP was cooled using a cooling fan with a power rating of 5V. To measure wall temperature of MHP, Thermal IR camera was used whose spectral range is 3.0-5.0 µm with a resolution of 640*512 records temperature distribution using IRT technique.

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The specification of design parameters used to conduct the experiment is listed in Table 2.

Table 2: Experimental design conditions [12]

Specifications	Parameters	Values
Design specifications	Power input (W)	4.5
	Fan voltage (V)	2-6
	Critical temperature (°C)	50
Design constraints	Inner Diameter of MHP (μm)	800
	Outer Diameter of MHP (μm)	1000
	Length of MHP (mm)	80

Working fluids are employed as DW and Al_2O_3/water with nanoparticle concentration of 2.0% by volume Nanofluid. The filling ratio(FR) 75%, which is the ratio between the volume of liquid used and the total volume, was maintained. The heat was supplied at the evaporator section using DC power unit as input with 50%, 75% and 100% 0f 4.5W and to dissipate these heat to the surrounding by using cooling fan running at different voltages inputs of 2V, 4V, and 6V. The experiment was conducted with one heat input for these three voltages of the cooling fan and repeated the same for the other heat inputs. The temperature at the end of the condenser is maintained at 50ºC which is a critical temperature.

IV. DESCRIPTION OF COMPUTATIONAL MODEL

In this model, the commercial software ANSYS Fluent 16.1 was used to simulate the two-phase flow for heat transfer analysis in an MHP. Computational analysis is increasingly developing an approach to solve engineering problems. This approach involves the simulation of the model and applying the appropriate boundary conditions for a solution.

A straight circular closed copper tube model was developed with dimensions of 0.8mm diameter and 80mm length, which consist of three section as an evaporator, adiabatic and condenser sections of 20mm, 40mm, and 80mm lengths respectively. The computational domain as developed in ANSYS shown in figure 4.

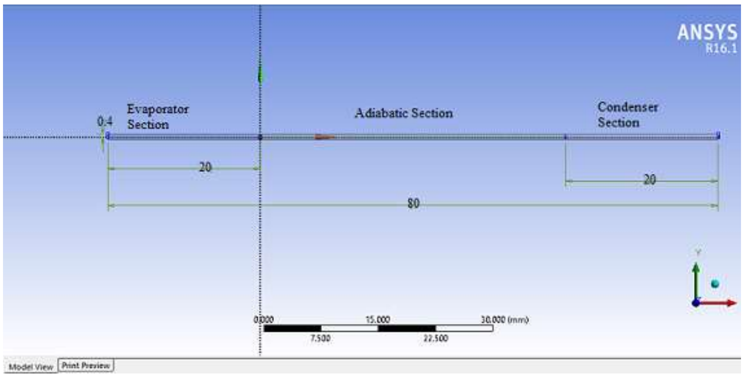

Figure 4: Computational domain of MHP

The model meshed as shown in figure 5, which is very fine mesh and discretized into 34452 nodes and 27641 elements.

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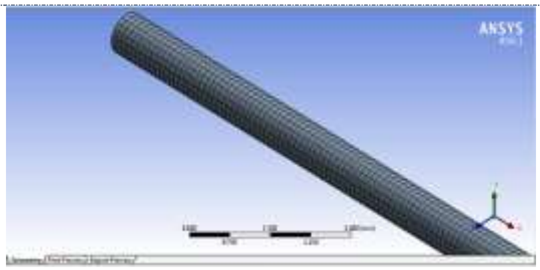

Figure 5: A small section of geometry mesh

In computational fluid dynamics, the Volume of fluid(VOF) method is used to analyze our model. Mixture model on multiphase analysis is adapted to accommodate the evaporation-condensation mechanism that exists within the micro heat pipe model with vapor as primary phase and liquid and air as a secondary phase. The motion of all phases is modeled by solving a single set of transport equations with appropriate jump boundary conditions at the interface. The VOF technique has been used to solve the model because of the difficulty arises for multiphase flow, as the interface between the phases are not stationary and physical properties changes at the interface, which require an intensive computational effort, as this problem cannot be solved by finite volume method [13]. The VOF model can model two or more immiscible fluids by solving a single set of momentum equations and tracking the volume fraction of each of the fluids throughout the domain.

Fluent solves the conservation form of unsteady energy equations. The governing equations of continuity (Eq.(i))[10], momentum (Eq.(ii))[10] and energy conservation (Eq.(iii))[10] are used to describe the motion of working fluid in a MHP.

$$\frac{\partial \rho}{\partial t} + \nabla.(\rho U) = 0 \qquad \text{(i)}$$

$$\frac{\partial \rho U}{\partial t} + \nabla.(\rho U * U) = \nabla.(-p\delta + \mu(\nabla U + (\nabla U)^T)) + S_M \qquad \text{(ii)}$$

$$\frac{\partial \rho h_{tot}}{\partial t} - \frac{\partial p}{\partial t} + \nabla.(\rho U h_{tot}) = \nabla.(\lambda \nabla t) + \nabla.(\mu \nabla U + \nabla U^T - \frac{2}{3}\nabla.U\delta U) + S_E \qquad \text{(iii)}$$

Where ρ is density, t is time, U is velocity, p is a static pressure, δ is the identity matrix, µ is dynamic viscosity, S_M is momentum source, h_{tot} is specific total enthalpy, λ is the second coefficient of viscosity, T is temperature, and S_E is an energy source. The operator ∇ indicates divergence gradient, while T superscript indicates matrix transposition.

In order to simulate the boundary conditions was applied at three sections of MHP, a constant heat flux is defined at wall boundary of evaporator section, depending on the power input. Adiabatic section assumed as insulated and no boundary condition was defined. The convective heat transfer coefficient as boundary condition was defined at condenser section. The heat transfer coefficient has been calculated using the equation (iv)[9]:

$$h = \frac{Q}{2\pi r l_c \left(T_{c,av} - T_\infty\right)} \qquad \text{(iv)}$$

Where h is condenser heat transfer coefficient, Q is heat transfer from the condenser, l_c is condenser length, $T_{c,av}$ is condenser average temperature and T_∞ is surrounding air temperature. The model considered DW and *Al_2O_3/water* Nanofluid as working fluid with filling ratio of 75%.

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Thermo-physical properties of Nano fluid are calculated by using equations of density (Eq,(v))[14], Specific heat (Eq,(vi))[14], thermal conductivity (Eq.(vii))[14] and viscosity (Eq.(viii))[14].

$$\rho_{nf} = (1-\phi)\rho_{bf} + \phi\rho_s \tag{v}$$

$$C_{p,nf} = \frac{\phi\rho_s C_{p,s} + (1+\phi)\rho_{bf} C_{p,bf}}{\rho_{nf}} \tag{vi}$$

$$k_{eff} = \frac{k_s + 2k_{bf} + 2(k_s - k_{bf})\phi}{k_s + 2k_{bf} - (k_s - k_{bf})\phi} * k_{bf} \tag{vii}$$

$$\mu_{nf} = (1+2.5\phi)\mu_{bf} \tag{viii}$$

Where ρ_{nf} is density of Nano fluid, ρ_{bf} is density of base fluid, ρ_s is density of solid particle, ϕ is volume concentration, μ_{nf} is viscosity of Nano fluid, μ_{bf} is viscosity of base fluid, $C_{p,nf}$ is the specific heat of Nano fluid, $C_{p,bf}$ is specific heat of base fluid, $C_{p,s}$ is specific heat of solid particle, k_{bf} is thermal conductivity of base fluid and k_s is solid particle thermal conductivity.

A transient simulation is conceded with a time step of 0.005 to calculate the problem and iterate the solution with 200 iterations. The computation performed the solution and converged when the residuals were lower than 1E-4.

V. RESULTS AND DISCUSSION

The result of experimental tests on the variation of the voltage cooling fan performance is presented and evaluate the results with a computational model. In addition, experimental results of using Al_2O_3-Water Nano-fluid as a working fluid are compared to the DW. In this paper, the experimental investigation of the micro heat pipe performance was conducted for the heat input of 4.5 *W* and evaluates the thermal performance using Infra-Red (IR) imaging technique to measure and visualize the temperature distribution on the evaporator and condenser section. Further, the results through experiments were performed to verify by the computational simulation behavior of MHP throughout the entire length.

Surface Temperature Distribution

Figure 6(a), 6(b) shows the experimental and computational results of the outer surface temperature distribution along the MHP for heat input of 75% of 4.5W with cooling fan voltage of 6V for DW and *Al_2O_3/water* Nanofluid.

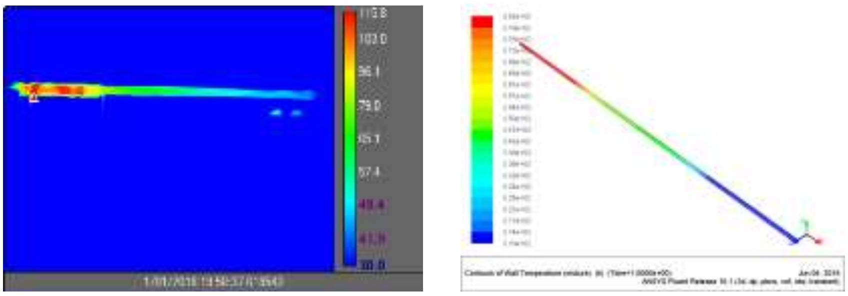

Figure 6(a): Experimental and Computational wall temperature distribution for DW

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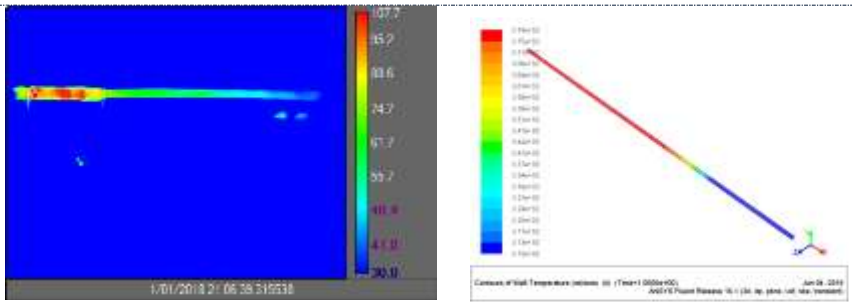


Figure 6(b): Experimental and Computational wall temperature distribution for Al_2O_3/water Nanofluid

Table 3 shows the surface average temperature in evaporator (T_e) and condenser (T_c) sections between CFD simulation and experimental results.

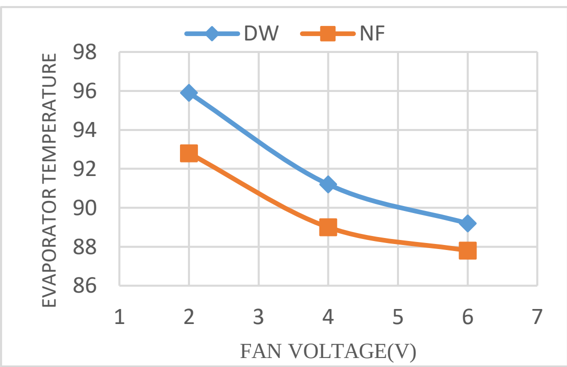
Table 3: Comparison between experimental and simulation result for 6V fan voltage

	Qin (4.5W)	Fan Voltage (V)	Evaporator (°C)		Condenser (°C)	
			Te,av Experimental	Te,av Simulation	Tc,av Experimental	Tc,av Simulation
DW	75%	6	109	110	43	37
Al_2O_3/water Nano fluid		6	104	106	42	37

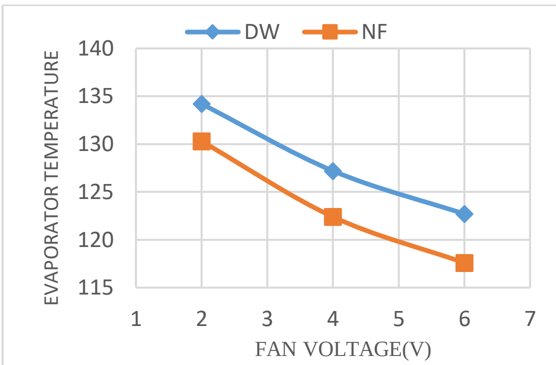
The distance between 0 to 20 mm shows the evaporator section where, in the experiment, a heat resistance wire is wrapped around it to provide uniform heating through DC power supply. The distance between 20-60 mm shows adiabatic section where there is no heat loss and the temperature raised in this section due to axial conduction heat transfer. The distance between 60-80 mm shows the condenser section where heat dissipates by air circulation around it through the cooling fan. In case of Simulation, similar boundary condition applied on a computational model where constant heat flux was provided on evaporator section, no heat loss in adiabatic section and convective heat transfer coefficient on condenser section. The Simulation evaporator average temperature has deviated from experimental results by 3-4°C.

Maximum Surface Temperature

The maximum surface temperature of MHP for different base frequencies of 4.5W heat input is shown in figure 7. The result shows that the surface temperature decreases along the length of the MHP and occur



(a) 50% PBF



(b) 75% PBF

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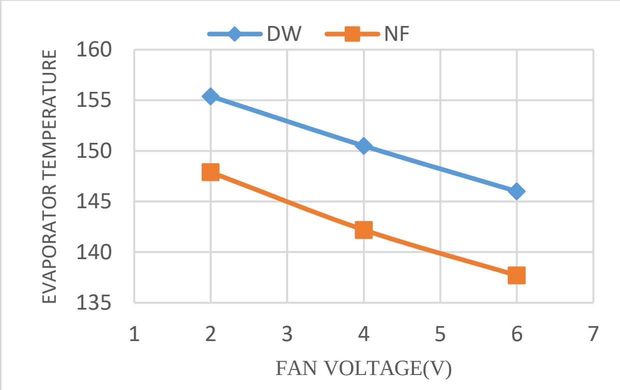


(c) 100% PBF

Figure 7: Maximum surface temperature of MHP

maximum temperature at evaporator section. It is observed that the maximum surface temperature of MHP in case of DW for all the three voltages of the cooling fan is greater than in case of Al_2O_3/*water* Nano-fluid.

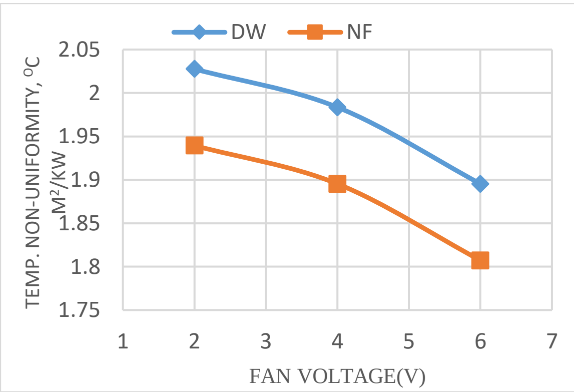
Temperature Non-Uniformity

The temperature non-uniformity of MHP is defined as the difference between the average temperature of evaporator and condenser section per unit heat flux. The temperature non-uniformity of MHP is given by equation(ix):

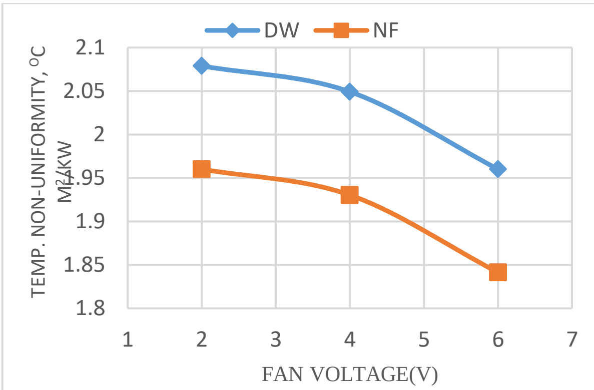
$$\theta = \frac{T_{av,evap} - T_{av,cond}}{q'} \qquad \text{(ix)}$$

Where q' is the heat flux, $T_{av,evap}$ is the average surface temperature of evaporator section and $T_{av,cond}$ is the average surface temperature of condenser section. The temperature non-uniformity of MHP for different base frequencies of 4.5W heat input is shown in figure 8.

The result shows that the temperature non-uniformity of MHP decreases with increasing the fan voltage speed. It is observed that the temperature non-uniformity of MHP in case of Al_2O_3/*water* Nano-fluid for all the three different base frequencies of heat input is lower, which is desirable to improve the reliability and performance of the device.



(a) 50% PBF



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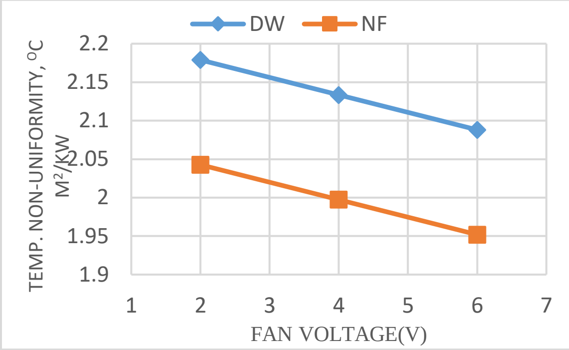

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Figure 8: Temperature non-uniformity of MHP

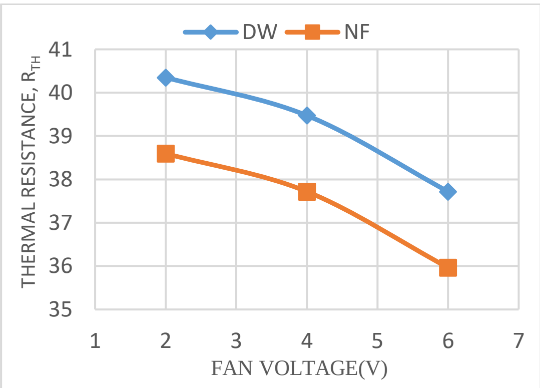
Thermal Resistance

The thermal resistance of MHP is defined as the ratio between the surface average temperature difference of evaporator and condenser to the heat input. The thermal resistance is calculated by using equation (x):

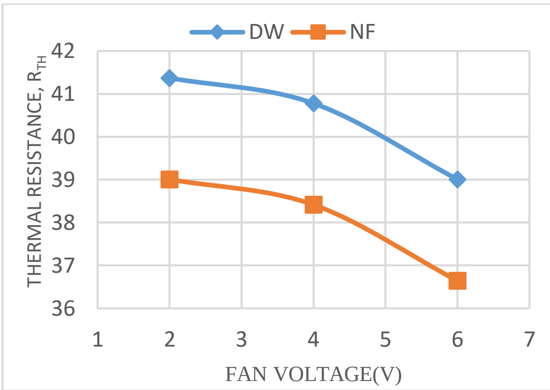
$$R_{th} = \frac{T_{av,evap} - T_{av,cond}}{Q_{in}} \qquad \text{(x)}$$

Where Q_{in} is the heat input, $T_{av,evap}$ is the average surface temperature of evaporator section and $T_{av,cond}$ is the average surface temperature of condenser section. The thermal resistance of MHP for different base frequencies of 4.5W heat input is shown in figure 9.

The result shows that the thermal resistance of MHP decreases with increasing the fan voltage speed. It is observed that the thermal resistance of MHP in case of DW for all the three different base frequencies of heat input of 4.5W is greater than in case of Al_2O_3/*water* Nano-fluid.



(a) 50% PBF



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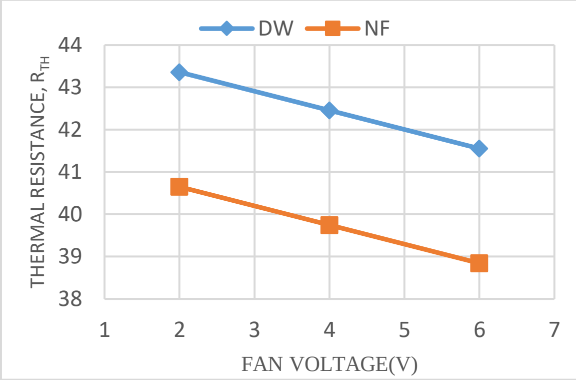

(c) 100% PBF

Figure 9: Thermal resistance of MHP

VI. CONCLUSION

The set of experiments was conducted using IR imaging camera to measure the surface temperature of the micro heat pipe for different voltages of the cooling fan with two different working fluids. The following conclusions can be drawn:

(1) The surface temperature of micro heat pipe decreases with increase in voltage of cooling fan for both working fluids.
(2) As shown in figure 7, the difference in maximum surface temperatures at evaporator section is about 5-10 ºC for DW and Nano-fluid.

Hence, Nano-fluid is more efficient to cool the device and improves cooling performance and reliability.

Also, the average surface temperature along the MHP has been compared with the computational results at the same conditions, showing that the results with the computation deviate with 2-3% from the experimental results. The thermal performance of MHP has also been characterized at a different base frequency of heat input of 4.5W by the temperature non-uniformity and effective thermal resistance, and it is found that increases the fan voltage speed increases the thermal performance parameter of MHP.

VII. ACKNOWLEDGEMENTS

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