Numerical Thermal Analysis of Nano Fluid Based Heat Pipe for Electronic Devices

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Highlights

- Numerical Analysis
- Heat Pipe
- Thermal Management

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Abstract: An analytical and numerical model is developed for heat pipe having Magnesium Oxide Nano fluids of concentration 0.5, 1.0 & 1.5%. Heat pipe with the induction of Nano fluids are becoming the preferred choice as a heat sink in electronic devices. These heat pipes are being used due to their higher thermal conductivity, compact design and enhanced life with better efficiency. This study has numerically and experimentally investigated the thermal analysis and thermal management by the induction of Magnesium Oxide (MgO / water) Nano fluids in sintered wick heat pipe. The analytical model is developed with suitable assumptions. The ANSYS FLUENT software has been used for numerical analysis. A comparison of analytical modeling determines the temperature profile result shows that the analytical modelling and numerical modeling results are in good range with the numerical modeling results. There is less than 5% error in both results.

Keywords: Thermal Analysis; Nano Fluids; Numerical Approach

1. Introduction

The Metal oxide nanoparticle technology is expanding very quickly in the universe [1]. Several industrial processes were using water and ethylene glycol previously. In this race the very first method that was in used in enhancing heat transmission was the use of these particles in millimetre and micro meter size. The issue with this new emerging new method was sedimentation of nanoparticles. Furthermore, this flaw also increased the rate of material loss, fouling and pressure loss [2].

Choi for the first time in this field of study enhanced thermal conductivity of base fluid by adding metal and non-metal oxide nanoparticles. There was impact of size, shape and kind of nanoparticle on thermal conductivity of nanofluid, further thermal properties of base fluid also has solid impact [3].

Meril Eainen et al. [4] studied about the impact of size and shape of nanoparticles like Al2O3, SiO2, MgO to improve study of pressure and heat transfer based losses in water based nanofluids. There was 40 % increment in average heat transfer coefficient when metal and non-metal based nanoparticles were introduced. They also observed that small shaped nanoparticles less than 10 nm also minimize the pressure loss issues.

Utomo et al. [5] processed both numerical and experimental research on nanofluids. Numerical simulation of heat transfer in nanofluids was also discussed in literature [6-8] from the studies it was observed that two phase model was more likely to be accurate than single phase model. They further predicted that heat transfer coefficient was increased by increasing the concentration of nanoparticles in base fluid. From their study it was also observed that for CuO / water nanofluids there was error of 16 % between experimental and CFD study for single phase model but this error was minimized to 08 % for the same study in case of two phase model.

Various studies were observed regarding impact of shape of heat pipe and its effect on thermal and physical properties of nanofluid [9-11]. Akhavan Behabadi et al. change the geometry of heat pipe from simple plain heat pipe to helically curved tube [12]. Moving forward Yarmand et al. [13] proceed to study the behaviour of rectangular shape heat pipe. H. Togun et al. [14] also contributed in this study by changing the shape of tube based heat pipe.

In further studies some researchers introduced carbon based nanoparticles and found that ability of heat transfer increased. Sadeghinezhad et al. [15] studied impact of graphene based nanoparticles on heat flux and found that there was great impact and the results were improved from previous studies graphene nanoparticles GNP were found to have the best heat transfer capability.

The current numerical study looked at the thermal performance of MgO / water nanofluid inside sintered wick heat pipe. The MgO nanoparticles were thought to be spherical and average diameter of 20 nm. Utilizing single phase, mixed, and VOF models, the thermal behaviour of nanofluids was developed. The programme was configured to utilise the k-e laminar model, which is a common model in this kind of simulation [16]. The fully formed segment of the heat pipe was where the simulation was run. The numerical method's findings were contrasted with experimental findings in pervious study.

2. Physical performance of nanofluids

The equations below can be used to compute the physical parameters of MgO Nano fluids. [17].

Nanofluids density is calculated by using mixing theory [18]

$$\rho = \varphi \rho_P + (1 - \varphi) \rho \quad (1)$$

By using thermal equilibrium model we can calculate specific heat capacity of Nano fluids as,

$$\frac{\varphi (\rho Cp)\, p+(1 \quad \varphi) (\rho Cp)\, f}{} \quad (2)$$

Similarly, dynamic viscosity can be drawn as,

$$= (1 \qquad\qquad) \quad (3)$$

Table 1. Properties of Nanofluids

MgO				
ρ(kg/m3)		Cp(j/kg*K)		K(W/mK)
3560.0		955.0		45.0
H2O				
ρ(kg/m3)	μ(kg/m*s)	Cp(j/kg*K)		K(W/mK)
998.20	0.00100	4182.0		0.60
Nano fluid Properties (Volume Fraction)				
MgO Volume Fraction %	ρ(kg/m3)	μ (kg / m*s)	Cp(j/kgK)	K(W/mK)
0.00063	998.802	0.00101	4178.8	0.64
0.00125	1001.40	0.00102	4167.7	0.65
0.00254	1004.60	0.00103	4153.4	0.67
Nano fluid Properties (Weight Percentage)				
MgO Weigh %	ρ(kg/m3)	μ (kg / m*s)	Cp(j/kgK)	K(W/mK)
0.5 %wt.	157.202	0.00016	656.42	0.11
1.0 %wt.	314.403	0.00032	1312.8	0.20
1.5 %wt.	471.604	0.00048	1969.3	0.30

3. Numerical Procedure

To understand the physical facts behind an engineering challenge, experimental investigations are crucial. Furthermore, the experimental investigation also shows excellent levels of solution correctness and dependability. It does have some drawbacks, though, including expensive costs, longer processing times, equipment costs, and the need to maintain operating parameters and steady-state temperatures.

Numerical study is useful here in such situations [19]. ANSYS FLUENT 2021-R2, a piece of commercial CFD software, was used to carry out numerical analysis for this investigation. The experimental findings supported the CFD results. Additionally, the use of CFD is growing quickly along with computing resources. The

verification is useful in locating and measuring modelling and solution flaws. The boundary condition assumptions used and flaws such as discretization, truncation, and rounding off have an impact on how accurate the numerical analysis is. Numerical analysis was done for the copper sintered HP in the current work to more easily manage the design issues of two-phase heat transfer, liquid state fluid flow via the wick structure, and the interface between the gaseous and liquid regions.

Table no. 02 displays heat pipe geometrical dimensions that were employed in this research. The heat pipe's wick structure is a copper powder-based sintered wick with a 0.5 mm thickness that was chosen to be the heat pipe's outer part and was chemically cleaned copper pipe. Considering sintered wicks with porosities of 0.63.

In this study, the mesh of HP was generated by skew method. Surface temperatures of the heat pipe were measured for this test.

Surface temperature results obtained were closest to the experimental results with approximate 02 % error.

Table 2. Heat Pipe Description

Sr. No.	Description	Units	Dimensions
1.	Total Length	mm	380
2.	Outer Diameter	mm	08
3.	Wall Thickness	mm	0.2
4.	Wick Type		Sintered Wick
5.	Wick Thickness	mm	0.5
6.	Evaporator Length	mm	120
7.	Adiabatic Length	mm	70
8.	Condenser Length	mm	190
9.	Porosity of wick	-	0.63
10.	Working fluid	m l	7.3
11.	Operative Pressure	K pa	3.448

3.1. Assumptions

Following assumptions can be made in this study [20],

- Both vapor and liquid flow are treated as being incompressible and laminar.
- Vapor is assumed as saturated at interval t – 0 (time).
- All thermophysical properties except vapor density are assumed constant.
- Body forces are negligible.
- Saturation temperature of heat pipe depending on local pressure was estimated at each location.

4. Governing Equations

Laminar flow is considered so; laminar incompressible equations are used and solved for vapor and liquid region in heat pipes and computational area compromise of vapor core, wick and walls of heat pipe.

Keeping in mind previously discussed assumptions following equations are used [21].

Vapor Flow Region Continuity Equation

$$\frac{\partial \rho_v}{\partial t} + \frac{1}{r}\frac{\partial}{\partial r}(\rho_v r V_v) + \frac{\partial}{\partial z}(\rho_v W_v) = 0 \quad (4)$$

Momentum Equation

Momentum equations can be represented as,

R – Momentum

$$\rho_v\left[\frac{\partial V_v}{\partial t} + Vv\frac{\partial V\,v}{\partial r} + Wv\frac{\partial Vv}{\partial z}\right] = -\frac{\partial \rho_v}{\partial r} + \mu_v\left[\frac{\partial^2 V_v}{\partial z^2} + \frac{4}{3r}\frac{\partial}{\partial r}\left(r\frac{\partial V_v}{\partial r}\right) + \frac{1}{3}\frac{\partial^2}{\partial z}\frac{W_v}{\partial r} - \frac{4}{3}\frac{V_v}{r^2}\right] \quad (5)$$

Z – Momentum

$$\rho_v\left[\frac{\partial W_v}{\partial t} + Vv\frac{\partial W\,v}{\partial r} + Wv\frac{\partial Wv}{\partial z}\right] = -\frac{\partial \rho_v}{\partial z} + \mu_v\left[\frac{4}{3}\frac{\partial^2 W_v}{\partial z^2} + \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial W_v}{\partial z}\right) - \frac{2}{3}\frac{\partial}{\partial z}\left[\frac{1}{r}\frac{\partial}{\partial r}(rV_v)\right]\right] \quad (6)$$

Energy Equations

Energy equations are mentioned as,

$$\rho_v\left[\frac{\partial V_v}{\partial t} + Vv\frac{\partial V\,v}{\partial r} + Wv\frac{\partial Vv}{\partial z}\right] = -\frac{\partial \rho_v}{\partial r} + \mu_v\left[\frac{\partial^2 V_v}{\partial z^2} + \frac{4}{3r}\frac{\partial}{\partial r}\left(r\frac{\partial V_v}{\partial r}\right) + \frac{1}{3}\frac{\partial^2}{\partial z}\frac{W_v}{\partial r} - \frac{4}{3}\frac{V_v}{r^2}\right] \quad (7)$$

Where, Ø can be given by,

$$\varphi = -Z\left[\frac{\partial V_v}{\partial r}+\left(\frac{V_v}{r}\right)^2+\left(\frac{\partial W_v}{\partial z}\right)^2\right]+\left[\frac{\partial V_v}{\partial z}+\left(\frac{\partial W_v}{\partial r}\right)^2\right]-\frac{2}{3}\left[\frac{1}{r}\frac{\partial}{\partial r}(rV_v)+\left(\frac{\partial W_v}{\partial r}\right)^2\right] \quad (8)$$

MgO Nanofluids and DI Water Region

The flow through porous material has the greatest impact on the working fluid going through the wick via capillary action. The main parameters are porosity and permeability, and data from the experimental setup are used in this research.

The flow of nanofluids is considered to be incompressible, and the viscosity between layers is laminar. So,

Continuity Equation

$$\frac{1}{r}\frac{\partial}{\partial r}(rV_l)+\frac{\partial W_l}{\partial z}=0 \quad (9)$$

R – Momentum

$$\frac{1}{\epsilon}\left[\frac{\partial V_l}{\partial t}+\frac{1}{\epsilon}\left(V_l\frac{\partial V\,l}{\partial r}+W_l\frac{\partial Vl}{\partial z}\right)\right]=-\frac{1}{P_l}\frac{\partial \rho_l}{\partial r}-\frac{\mu_l V_l}{P_l K_l}+\frac{\mu_l}{P_l\epsilon}\left[\frac{\partial^2 V_l}{\partial z^2}+\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial V_l}{\partial r}\right)-\frac{V_l}{r^2}\right] \quad (10)$$

Where, ε is wick porosity,

Z – Momentum

$$\frac{1}{\epsilon}\left[\frac{\partial W_l}{\partial t}+\frac{1}{\epsilon}\left(V_l\frac{\partial W\,l}{\partial r}+W_l\frac{\partial Wl}{\partial z}\right)\right]=-\frac{1}{P_l}\frac{\partial \rho_l}{\partial z}-\frac{\mu_l W_l}{P_l K_l}+\frac{\mu_l}{P_l\epsilon}\left[\frac{\partial^2 W_l}{\partial z^2}+\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial W_l}{\partial r}\right)\right] \quad (11)$$

Energy Equation

$$\rho_l C_{pl}\left[\frac{\partial T_l}{\partial t}+V_l\frac{\partial T\,l}{\partial r}+W_l\frac{\partial Tl}{\partial z}\right]=\frac{1}{r}\frac{\partial}{\partial r}\left(rK_{eff}\frac{\partial T_l}{\partial r}\right)+\frac{\partial}{\partial z}\left(rK_{eff}\frac{\partial T_l}{\partial z}\right) \quad (12)$$

Where, effective thermal conductivity can be

$$K_{eff}=\frac{K_l[(K_l+K_s)-(1-\epsilon)(K_l-K_s)]}{(K_l+K_s)-(1-\epsilon)(K_l-K_s)} \quad (13)$$

Heat Pipe Wall Region

Equations for heat pipe wall region can be represented as,

$$\rho_w C_{pw}\frac{\partial T_w}{\partial t}=K_w\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T_w}{\partial r}\right)+\frac{\partial^2 T_w}{\partial z^2}\right] \quad (14)$$

Wick Vapor Interface

The wick-vapor interaction is modelled as a flat surface in this study. As a result, it is considered that the liquid-vapor interface corresponds to the contact between the vapour core and the liquid. At the wick-vapor contact, energy balance is established between the liquid and vapour sides to allow for heat transmission. To compute the interface temperature, the energy balance equation is employed.

$$-k_{wick}\,A_i\frac{\partial T}{\partial y}+\dot{m}_i\,C_l\,T_i=-K_v\,A_i\frac{\partial T}{\partial y}+\dot{m}_i C_v T_i+\dot{m}_i h_{fg} \quad (15)$$

Operating Pressure in Vapor Core

In order to maintain the system to remain pressurize under above assumption. It can be stated as [22],

$$P=\hat{p}+P_{op} \quad (16)$$

Here $\hat{p}$ represents hydrodynamic component of pressure. The system operating pressure in above equation is calculated by using ideal gas law.

Equation for this derivation is defined as,

$$P_{op}=\frac{M_{\circ v}+\nabla t\left(m_{wick/vapors}-\dot{m}_i\right)}{\frac{1}{R}\sum_{all\ vapor\ cells}\frac{V_{cell}}{T_p}} \quad (17)$$

Calculation equation for ṁi can be defined as,

$$\dot{m}_i=\left(\frac{2\sigma}{2-\sigma}\right)\frac{A_i}{(2\pi R)^{1/2}}\left(\frac{\hat{p}_V+P_{op}}{T_V{}^{1/2}}-\frac{P_i}{(T_i)^{1/2}}\right) \quad (18)$$

5. Meshing

Meshing of heat pipe can be seen in figure 1

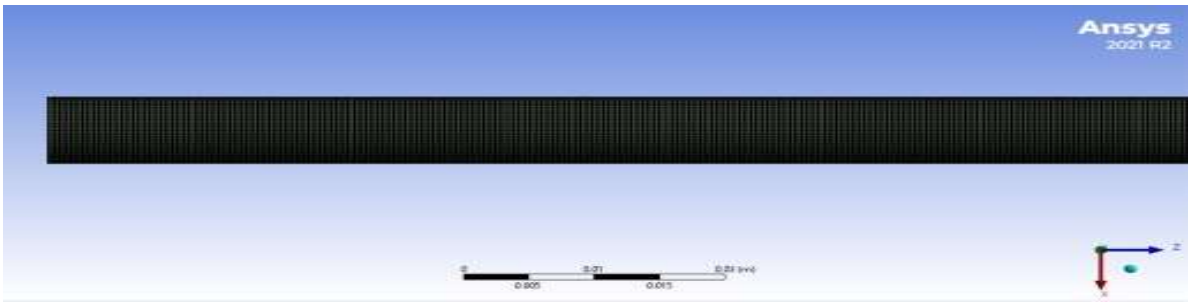

Figure 1. Meshing of Heat Pipe

6. Results and Discussion

These simulations were performed firstly by just considering the parameters of vapors and water and considering them as DI water mixture and the finding the parameters for various concentrations of MgO Nano fluids i.e. 0.5 %, 1.0 %, 1.5 % by weight and various power inputs ranging from 10 – 30 watts with an increment of 05 watt and can be seen in figure 2.

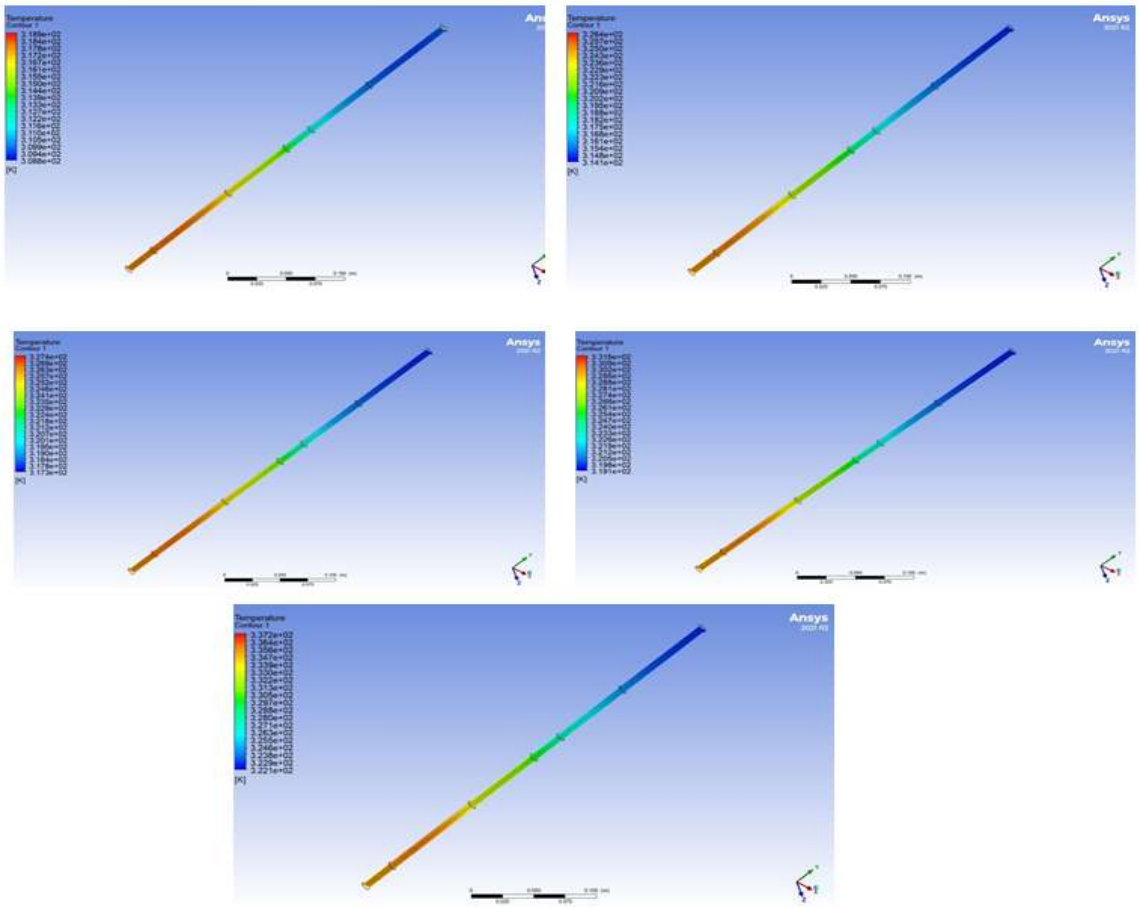

Figure 2. This Temperature Contours of Heat Pipe with 1.0 % MgO wt. % at 10,15,20,25 & 30 watt

6.1.Validation of Numerical Study

The present numerical study utilizing the heat pipe using MgO as Nano-particle and water as based fluid with three different concentration levels (i.e. 0.5 %, 1.0 % and 1.5 % wt.). The maximum reduction of temperature obtained from the heat pipe sample using 1.0 % wt. of Nano-fluid at all heating loads. The deviation between the temperatures is found to be ~ 02 - 03 °C with experimental setup readings and ~ 3.3 °C approx. with previous study. The analytical results were compared with experimental values and previous studies and one of the comparisons is shown in the graphical representation. Figure 5.29 shows the surface temperature at different sections of heat pipe.

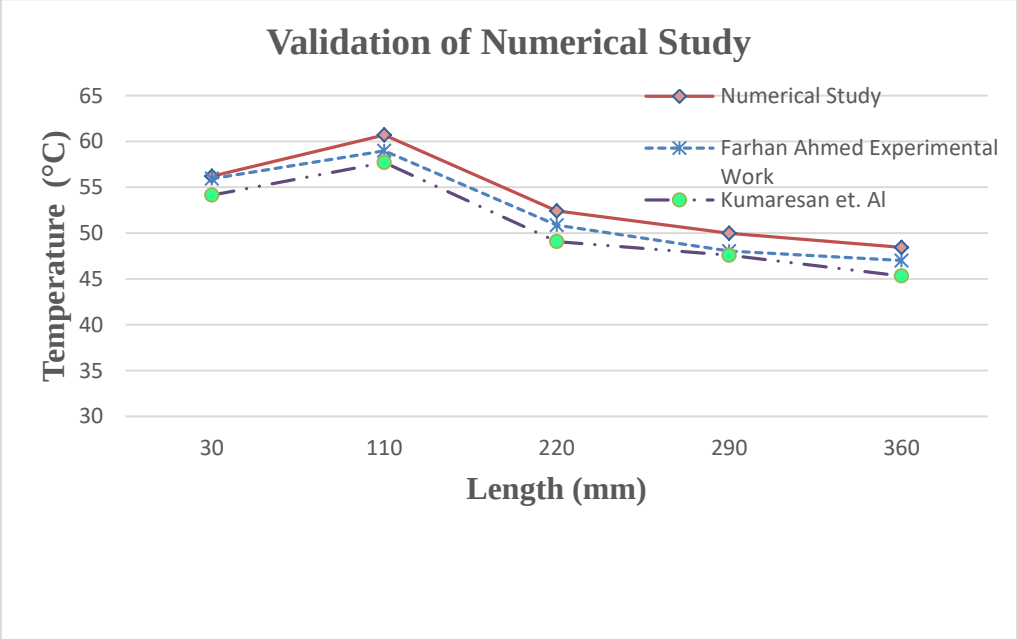

Figure 3. Validation of Numerical Study

4. Discussion

Authors should discuss the results and how they can be interpreted in perspective of previous studies and of the working hypotheses. The findings and their implications should be discussed in the broadest context possible. Future research directions may also be highlighted.

5. Conclusions

In this study CFD analysis of simple heat pipe with DI water and different concentrations of MgO Nano fluid was performed in order to validate an experimental setup. Heat pipe thermal performance was investigated at different power input levels, which on another hand can be very beneficial for various electronic devices. Results at various concentrations of MgO Nano fluid and heat pipe with simple DI water in term of temperature distribution are presented in pervious chapter.

From this study it is concluded that

- The best suitable results were found for the concentration of 0.1 % by weight of MgO Nano fluid.
- Nano fluid based heat pipe shows more temperature drop as compared to simple heat pipe with Di water.
- Maximum ranges of temperature were found at evaporator section and minimum temperature values were traced at condenser section. This is because of the fact that heat input was on evaporator end.
- Evaporator and condenser section temperature distribution graphs at different power inputs were plotted for both numerical and experimental studies and found that the results of numerical study were slightly higher as compared to experimental study. This is clearly because of convection losses which occur during experimental study.

This section is not mandatory, but can be added to the manuscript if the discussion is unusually long or complex.

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