Numerical Investigation of Loop Heat Pipe Performance with Al₂O₃-H₂O Nanofluid

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KEYWORDS

Loop heat pipe
Heat transfer
Nanofluid
Single phase
Two phase

ABSTRACT

The main aim of this study is to enhance the thermal performance of loop heat pipe (LHP) charged with nanofluid as the working fluid. Thus, numerical simulations are conducted to investigate heat transfer characteristics of using Al₂O₃-H₂O nanofluid with nanoparticle mass concentration ranging from 0% to 3% in a LHP as a working fluid for heat input range from 20 W to 60 W. The three-dimensional model, laminar flow and heat transfer governing equations are solved using the finite volume method. The simulations are carried out with three-dimensional model based on the characterization of the working fluid inside the LHP to give an insight into the heat transfer and fluid flow mechanism. The LHP performance is evaluated in terms of fluid temperature distributions, pressure drop and bubble formations across the vapour line of LHP. It is inferred that the fluid temperature increases as the nanoparticle mass concentration increases, which indicates that the LHP charged with Al₂O₃-H₂O nanofluid able to absorb more heat from the surface due to enhanced heat removal in the LHP by higher thermal conductivity of nanofluid. Only a slight increase in the pressure drop across the LHP is found compared with the pure water-charged LHP. It is also found that the presence of Al₂O₃ nanoparticles in the working fluid contributes to increase the nucleation sites necessary for bubble formation. Thus, it is highly recommended to use nanofluid in LHP to enhance the boiling heat transfer.

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1. INTRODUCTION

A number of single- and two-phase liquid cooling techniques, such as liquid cooled microchannel heat sinks, advanced heat pipes, immersion cooling involving pool boiling of a dielectric working fluid, and liquid jet impingement cooling, have been developed to handle high levels of heat dissipation in electronics at the chip or package level. Air-cooled aluminum heat pipes developed at Intel can remove a total power of 200 W with a heat flux up to 200 W/cm². Loop heat pipes or capillary pumped loops can operate at power levels in excess of 600 W/cm² without depriming the wick. A number of devices have been developed that are capable of removing heat fluxes in excess of 1000 W/cm², such as the microjet impingement cooling array [1]. For some consumer electronics applications, such as high-power-density semiconductor devices (solid-state light sources) for projection television sets, heat fluxes exceed 2000 W/cm². Therefore, conventional liquid cooling such as water appears to be inevitable.

With nanofluids, the cooling performance could be improved as reported by Lee and Choi [2]. In the realm of electronics cooling, some companies are conducting research to use nanofluids instead of water. In addition, nanofluids could effectively remove hot spots and maintain components at uniform temperatures. Considering the range of efforts under way to extend liquid cooling technologies and the superior thermal properties of nanofluids, the future seems bright for using nanofluids for high-heat-flux and hot-spot cooling systems for computer, telecom, power and defense electronics uses, among others.

Heat pipes have been the centre of attention as high heat transfer devices for a couple of decades. During the time of heat pipe development two approaches have been followed in order to enhance the heat pipe heat transfer capacity and reduce its thermal resistance. One approach is to improve heat pipes’ structure including the pipe modification—e.g. using finned pipes, annular structure—and wick modification—e.g. new wick designs or wicks combinations. Another approach is to seek better working fluids which are compatible with the heat pipe structure and have higher thermal properties. Some liquids such as water, oils, and ethylene glycol have been used as industrial heat transfer fluids for a long time, but due to higher thermal conductivities of metals compared to these liquids, suspensions of micro-sized solid particles—especially metal or metallic oxides particles—in base fluids, which were named microfluids,
have been used as heat transfer fluids with higher thermal conductivities. Disadvantages of microfluids such as instability, sedimentation and corrosion of the flow channels, and also some theoretical ideas about nanofluids-based on the Maxwell equation-led to introduce nanofluids as a new media for heat transfer.

Nanofluid is a suspension of nano-sized particles in a base fluid. In addition to disposing of microfluids disadvantages, nanofluids suggest greater effective thermal conductivities than that of microfluids because the interface between nanoparticles and the base fluid is much more than the interface between microparticles and the base fluid. Nanofluid was firstly developed by suspending metallic or nonmetallic nanometer dimension particles in traditional fluids, such as water, engine oil, and ethylene glycol [3].

The fundamental studies of nanofluids applied in heat pipes are still in its initial stage, most of research works are carried out experimentally to focus on finding out key factors affecting the reliable application of nanofluids in the heat pipes and some experimental results cannot be unified yet. The type, size of heat pipes and operating conditions of heat pipes, the kind of base fluids, the material and size of nanoparticles all varied in very wide ranges among these experiments. Therefore, it is difficult to quantitatively make the comparison among different experimental data and then the most existing research conclusions are qualitative.

The numerical studies on the thermal performance of heat pipe charged with nanofluid are very few and the available studies are only addressed on micro-grooved heat pipe and mesh wick heat pipe, which are [4], [5], [6], [7] and [8].

The observations based on the reviewed literature showed that the theoretical investigations on nanofluids in heat pipes are very few and hence validating the experimental findings is difficult. However, many issues such as the transfer of nanoparticles by the vapor phase during heat pipe operation can be investigated only with adequate experiments. As for the numerical simulation, most studies were reported on two-dimensional modeling. To give an insight into the heat transfer and nanofluid flow mechanism, the 3D model based on the characterization of the nanofluid inside heat pipes have not been explored so far. Nevertheless, most of the previous works considered on conventional heat pipes such as micro-grooved heat pipe, mesh wick heat pipe and oscillating heat pipe, and there is far less work conducted for LHPs. Due to lack of study on thermal performance of LHP charged with nanofluid has been reported in the past, more investigations are needed. Accordingly, in the present research, all the aforementioned issues on application of nanofluids in LHP are addressed numerically.

2. MODEL DESCRIPTION

The three-dimensional (3D) model of the LHP assembly based on the experimental arrangement in study of Gunnasegaran et al. [9] is built in GAMBIT and is exported to FLUENT. The LHP consists of copper base plate (heat source) at underneath of evaporator, evaporator, vapour line, condenser with 50 aluminum rectangular fins and liquid line. The physical configuration of LHP is shown in Fig. 1. The heat supplied to the bottom wall of evaporator of LHP is removed by flowing nanofluid through a fins in condenser. The specified dimensions and specifications of the LHP are listed in Table 1.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Dimension/Material</th>
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<tbody>
<tr>
<td>Evaporator</td>
<td></td>
</tr>
<tr>
<td>Dimension (mm)</td>
<td>L50 × W50 × H4</td>
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<tr>
<td>Material</td>
<td>Copper</td>
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<tr>
<td>Reservoir</td>
<td></td>
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<tr>
<td>Volume (Liter)</td>
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<tr>
<td>Dimension (mm)</td>
<td>L149 × W100 × H85</td>
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<td>Material</td>
<td>Aluminum faceplate</td>
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<tr>
<td>Sintered Nickel Wick</td>
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<td>Pore radius (µm)</td>
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<td>Permeability (m²)</td>
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<tr>
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</tr>
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<tr>
<td>Inner Diameter, Di (mm)</td>
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<tr>
<td>Length (mm)</td>
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<tr>
<td>Material</td>
<td>Copper</td>
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<tr>
<td>Condenser</td>
<td></td>
</tr>
<tr>
<td>Dimension (mm)</td>
<td>L150 × W100 × H1</td>
</tr>
<tr>
<td>Material</td>
<td>Aluminum</td>
</tr>
</tbody>
</table>

Fig. 1. Schematic diagram of the computational domain

3. THREE-DIMENSIONAL SIMULATION OF CHARACTERIZATION OF WORKING FLUID

3.1 Single-phase model

Nanofluids are generally dilute mixtures of solid-liquid that contains very fine particles, which is smaller than 100nm. Because of such extremely reduced dimension, it has been suggested that these particles may be easily fluidized and consequently, can be considered to behave more like a fluid. Furthermore, by assuming negligible motion slip between the particles and the continuous phase, and the thermal equilibrium conditions also prevail, the nanofluid may be then considered as a conventional single-phase fluid with effective physical properties being function of the properties of both constituents and their respective concentrations. An interesting result from
such an assumption resides in the fact that an extension from a conventional fluid to a nanofluid appears feasible, and one may expect that the classical theory as developed for conventional single-phase fluids can be then applied to nanofluids as well. Thus, the equation of conservation of mass, momentum and energy as well known for single-phase fluids can be directly extended and employed for nanofluids. Yu et al. [10] stated that in order to study the heat transfer behavior, nanofluid can be considered as a homogenous fluid and the theory of single-phase heat transfer can be used for computations. Bianco et al. [11] and Esfandiary et al. [12] studied forced convection heat transfer in the circular pipe by using single-phase and two-phase models. They reported that the single-phase analysis results are closer to experimental results. In the present study, in conjunction with the aforementioned argument, the single-phase approach has been adopted to study the thermal behaviours of the nanofluid as a homogeneous fluid with effective properties. To obtain best estimate results with the single-phase model, it is very important to use the most appropriate correlations for the effective nanofluid properties. Up to now, however, there are no universal correlations that can accurately predict nanofluid properties for any combination of independent variables (nature of particles, diameter of particles, etc.). Many different correlations available in the literature lead to contradictory results [13] and it is not clear which one is the best for a given situation. Nevertheless, all sources indicate that the nanofluid properties depend on the particle mass concentration as well as on the corresponding properties of the base fluid and the solid particles. Consequently, in this study all the nanofluid properties are expressed as functions of the nanoparticle mass concentration as follows [14-17]:

Density: \( \rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_p \) (1)

Heat capacity: \( \rho c_p)_{nf} = (1-\varphi)\rho c_p + \varphi\rho c_p \) (2)

Thermal conductivity: \( k_{nf} = \frac{k_p + 2k_{bf} + 2(k_p - k_{bf})\varphi}{k_p + 2k_{bf} - (k_p - k_{bf})\varphi} \) (3)

Viscosity: \( \mu_{nf} = \mu_p(1 + 2.5\varphi) \) (4)

Where \( \varphi \) is particle mass concentration, the subscript “\( nf \)” refers to nanofluid, “\( bf \)” refers to base fluid, and “\( p \)” refers to particle.

To focus on the effect of using nanofluid with different nanoparticle mass concentrations on the LHP performance, the following assumptions are made: (i) both fluid flow and heat transfer are in steady-state and three-dimensional; (ii) fluid is in single phase, incompressible and the flow is laminar; (iii) properties of both fluid and LHP material are temperature-independent; and (iv) all the surfaces of LHP exposed to the surroundings are assumed to be insulated except the bottom base of the LHP where constant heat flux boundary condition simulating the heat generation from electronic chip is specified. The differential equations for this model (conservation of mass, momentum and energy) are obtained from [18] and can be written as:

\( \frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \) (5)  

\( x \)-Momentum: \( \rho \left( \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} \right) = -\frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \) (6a)

\( y \)-Momentum: \( \rho \left( \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} \right) = -\frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) \) (6b)

\( z \)-Momentum: \( \rho \left( \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} \right) = -\frac{\partial p}{\partial z} + \mu \left( \frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \) (6c)

Energy: \( \rho c_p \left( \frac{\partial u}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = k \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \) (7)

where \( p \) is the fluid pressure, \( \rho \) is the fluid density and \( T \) is the fluid temperature. The \( c_p \) fluid specific heat, \( \mu \), dynamic viscosity and \( k \), thermal conductivity are taken to be constant. \( u, v \) and \( w \) are the velocity components in the \( x, y \) and \( z \) directions, respectively.

3.2 Boundary conditions

The boundary conditions are specified to the present computational domain. At the entrance (inlet) of the LHP assembly as illustrated in Fig. 1, two types of boundaries are encountered which are nanofluid flows through the heat pipe and removes heat to the surface of the fins. The remainder of the entrance is occupied by the copper substrate. At the inlet section, the inlet nanofluid temperature is taken as 22 °C and the inlet nanofluid velocity is obtained from the experimental study of Gunnasegaran et al. [9], which is 0.0005821 m/s corresponding to 5 ml/min. The transverse velocities at the inlet are assumed to be zero. On the copper substrate, the velocities are zero, and it is assumed to be an adiabatic surface. The heat flux applied at the bottom base of the LHP is assumed to be the heat source.

Boundary conditions at the inlet:

\( u = u_{in}, T = T_{in} \) (8)

At the outlet:

\( p = p_{out} \) (9)

At the surface of fin:

\( h = \frac{Q}{A_j(T_c - T_A)} \) (10)

At the bottom plate:

\( \dot{q}_w = -k_s \frac{\partial T}{\partial n} \) (11)
In Eqs. (8)-(11), $u_{in}$ and $T_{in}$ are the nanofluid inlet velocity and inlet temperature, respectively, $p_{out}$ is the pressure at the outlet, $h$ is the heat transfer coefficient of fin, $Q$ is the heat input, $A_f$ is the area of rectangular fin, $T_c$ is the steady state condenser temperature obtained from experiment, which is corresponding to the applied heat input, $T_a$ is the ambient temperature, $n$ is the direction normal to the wall, and $q_v$ is the heat flux applied at the bottom base of the LHP. The thermophysical properties for nanofluids used in the computation are calculated using Eqs. (1)-(4) and $k_s = 402$ W/m·K.

3.3 Two-phase model

The volume of fluid (VOF) model in the present study is used to simulate the bubble formation during the evaporation in the vapor line of LHP. Thus, the simulations are carried out in a horizontal tube based on vapor line of LHP in present experiment with an inner and outer diameter of 13.5 mm and 15 mm, respectively, and a length of 250 mm. The VOF model solves a single set of momentum equations for all the phases and tracks their volume fraction all over the domain of study by solving a continuity equation for the secondary phases. Nanofluid in a liquid form is chosen as primary phase and the vapor is selected for secondary phase. The total summation of the volume fractions for all the phases is equal to unity. Therefore, the magnitude of the primary phase volume fraction will be calculated. In this method, all of the physical properties are calculated by taking a weighted average of different phases based on their volume fraction throughout each control volume. The single set of momentum equation is solved to find the velocity components, which are shared by all the phases. In the same manner, a shared temperature is calculated from a single energy equation. Specifically, mass conservation is expressed as [19]:

$$\nabla \cdot (\varphi_q \rho_q \vec{V}_q) = 0 \quad (12)$$

where $\sum_{q=1}^{n} \varphi_q = 1$ and all properties are calculated like $N = \sum_{q=1}^{n} \varphi_q N_q$.

In this study, the fluid in the vapour line is assumed to be incompressible. In practice, compressibility is related to the capability of pressure forces to induce density variations. This can be related to the Mach (M) number of the flow. For the flow with M number less than about 0.3, the pressure forces are significantly small that the fluid density does not change and hence the flow is incompressible. This fact is good enough to prove that the flow in present study is incompressible because the M number is extremely less than 0.3 and can be justified as follow:

$$M = \frac{u}{c} = \frac{0.0005821 \text{m/s}}{1450 \text{m/s}} = 4.7 \times 10^{-7} < 0.3 \quad (13)$$

where $u$ is the velocity of fluid corresponding to coolant flow rate of 5 ml/min that used in experiment of [9] and $c$ is the speed of sound of water at room temperature of 20-25°C. Therefore, the phase changes in working fluid in vapour line is mainly due to the temperature difference despite of pressure difference and hence, it is reasonable to said the flow in present study is incompressible or compressibility is negligible. Thus, the conservation of momentum and energy equations are identical to Eqs. (6) and (7).

For all simulations, a no-slip condition is imposed at the tube wall. The influence of the gravitational force on the flow has been taken into account. At the inlet of the tube, uniform profiles for all the variables have been employed. Both liquid and vapor enter the horizontal tube perpendicular to its inlet plane. They have an inlet temperature which has been obtained from single phase simulation on fluid temperature distribution, corresponding with a given heat input and the inlet velocity of 0.0005821 m/s. The fluid pressure at the tube inlet is set to 101.325 Pa. A pressure outlet boundary is imposed to avoid difficulties with backflow at the outlet of the tube.

4. NUMERICAL IMPLEMENTATION AND GRID REFINEMENT

The differential equations are discretized with the control volume technique. For the convective and diffusive terms a second order upwind method is used while the SIMPLE procedure is employed for the velocity-pressure coupling. In order to arrive at the optimum mesh size, a mesh independency test was conducted by increasing the number of cells from $2.3 \times 10^5$ to $6.6 \times 10^5$ grids, each time checking the temperature distribution along the vapor line of LHP by comparing with experimental result. The results are stabilized when the number of cells reaches $4.7 \times 10^5$ grids, and no further improvement in accuracy is observed beyond it. Therefore, the optimum number of computational cells with $4.7 \times 10^5$ grids is employed throughout the computations in this study. In Fig. 2, the comparison of experimental result with the liquid temperature along the length of the vapor line of LHP charged with pure water for single phase model is shown using three different grid densities under heat input of 60 W. The numerical results are well matching with the experimentally observed trends and indicated the validity of the present computational methodology for the thermal analysis of the LHP.

![Fig. 2. Comparison of the experimental result with the variation of liquid temperature along the vapor line length of LHP charged with pure water for three different computational grids](image-url)
5. RESULTS AND DISCUSSION

5.1 3D Simulation results of characterization of Al$_2$O$_3$–H$_2$O nanofluid

The thermophysical properties required for the flow of Al$_2$O$_3$–H$_2$O nanofluid flow within the LHP with three different nanoparticle mass concentrations are listed in Table 2. The thermophysical properties are calculated using the Eqs. (1) - (4). For the alumina–water nanofluid, the density, thermal conductivity, and dynamic viscosity are significantly increased while the specific heat of nanofluid is decreased with the increase of its particle volume fraction compared to pure water. The nanofluid of φ=3% alumina, for instance, has a relative increase of 8.93%, 8.85%, and 7.48% in the density, thermal conductivity, and dynamic viscosity and has a relative decrease of 8.95% in specific heat, respectively, compared with pure water. In this work, the effect of particle volume fraction on heat transfer and fluid flow characteristics of LHP with specific dimensions in experimental study of Gunnasegaran et al. [9] was considered. However, more computational effort is required since denser grids are needed.

Table 2. Thermophysical properties of nanofluid at 298 K

<table>
<thead>
<tr>
<th>Properties</th>
<th>Nanoparticle (Al$_2$O$_3$)</th>
<th>Base fluid (water)</th>
<th>Nanofluid (Al$_2$O$_3$–H$_2$O)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ρ (kg/m$^3$)</td>
<td>3970</td>
<td>998.2</td>
<td>1013.06 1027.918 1087.354</td>
</tr>
<tr>
<td>C (J/kg·K)</td>
<td>765</td>
<td>4182</td>
<td>4115.05 4049.734 3807.729</td>
</tr>
<tr>
<td>k (W/m·K)</td>
<td>40</td>
<td>0.613</td>
<td>0.62183 0.63074 0.66726</td>
</tr>
<tr>
<td>μ (Ns/m$^2$)</td>
<td>0.001003</td>
<td>0.001016</td>
<td>0.001028 0.001028 0.001078</td>
</tr>
</tbody>
</table>

5.2 Liquid temperature distribution

Fig. 3 shows the temperature contour of liquid inside the LHP charged with pure water at 40 W. The maximum temperature is at the evaporator (396 K), then the temperature gradually decreases along the flow path, and reaches the minimum at the end of liquid line (330 K). The predicted liquid temperature contour inside the LHP charged with Al$_2$O$_3$–H$_2$O for nanoparticle mass concentration of 0.5%, 1% and 3% at heat input of 40 W is shown in Figs. 4, 5 and 6, respectively. As depicted in Figs. 4, 5 and 6, the liquid temperature increases with increase in nanoparticle mass concentration (varied from 0.5% to 3%), which indicates the better heat transport capability and able to absorb more heat from the wall in further increase of nanoparticle mass concentration. This is due to enhanced heat removal in the LHP by higher thermal conductivity of nanofluid. The thermal conductivity enhancement of nanofluid showing a strong sensitivity to the liquid temperature was also found by some other researchers such as Patel et al. [20], Wei et al. [21] and Eastman et al. [22]. The nanolayers of liquid molecules around nanoparticles, clustering of nanoparticles, and the Brownian motion of nanoparticles for the anomalously enhanced thermal conductivity of Al$_2$O$_3$–H$_2$O nanofluid compared to that of pure water is the most reliable explanation for the obtained result [23].
Fig. 4. Contours of liquid temperature for LHP charged with Al\textsubscript{2}O\textsubscript{3}-H\textsubscript{2}O nanofluid (0.5\%).

Fig. 5. Contours of liquid temperature for LHP charged with Al\textsubscript{2}O\textsubscript{3}-H\textsubscript{2}O nanofluid (1%).
5.3 Pressure distribution

The pressure drop between evaporator and condenser of the LHP charged with pure water (0% of nanoparticle mass concentration) and Al₂O₃-H₂O nanofluid for nanoparticle mass concentration of 0.5%, 1% and 3% for heat input range from 20 W to 60 W is presented in Fig. 7. From this figure, it can be obviously seen that the pressure drop increases as the nanoparticle mass concentration increases from 0.5% to 3% due to the particle deposition. Another reason that contributes to increase the pressure drop on further increase in nanoparticle mass concentration is the increase in viscosity of nanofluid as defined in Eq. (4), which increase the resistance to fluid flow. However, the difference in pressure drop among the nanofluids and pure water is not apparent. At the heat input of 60 W, test results showed the average increase of 2.75% to 4.69% at nanoparticle mass concentrations ranging from 0.5% to 3% in pressure drop of LHP charged with Al₂O₃-H₂O nanofluid as compared with pure water.

5.4 The effect of nanoparticle mass concentration on the flow regimes in vapor line

At present, the study on the nanofluid two-phase flow in a LHP is rather limited. However considering that phase change in the LHP occurred mainly in the vapour line during the vaporization, it is likely that the convection heat transfer mechanism dominated the two-phase flow in the LHP. The boundary conditions for modeling the bubble flow inside the Al₂O₃-H₂O nanofluid charged LHP are exactly similar to that adopted for the LHP charged with pure water. The inlet temperature of liquid entering the tube of vapor line is taken as 79.16 °C, 94.32 °C, and 102.23 °C corresponding to 0.5%, 1% and 3%, respectively. These inlet temperature values are obtained from simulation of single phase model. The simulation is performed for heat input of 20 W and the thermophysical properties selected for two phase flow in current study are summarized in Table 3. It is to be noted that the value of surface tension, for Al₂O₃-H₂O nanofluid in Table 3 is assumed to be same with pure water. This assumption is made based on the experimental results of Golubovic et al. [24] and Kim et al. [25], which have shown that the surface tensions of the Al₂O₃-H₂O nanofluids without surfactant is independent on mass concentration and has the same values as that of pure water. Since the Al₂O₃-H₂O nanofluid prepared without any surfactant added to it in current study, the value of is taken as same as pure water.

The simulation results for the bubble flow regimes across vapour line at applied heat input of 20 W can be found in Figs. 8 (a), (b), (c) and (d) at nanoparticle mass concentration of 0% (pure water), 0.5%, 1% and 3%, respectively. These figures represent the contours of mixture density (kg/m³) for LHP charged with pure water and Al₂O₃-H₂O nanofluid. The red color refers to the presence of pure liquid, while the dark blue color refers to the presence of pure air. Based on Fig. 8, it is seen that the amount of air bubbles (dark blue color) formed in the vapour line of LHP increases with the increase in nanoparticle mass concentration (varied from 0.5% to 3%).
But, the increase in amount of bubbles formation is not significant. Hence it can be deduced that nanoparticle mass concentration does not have a major effect on the surface tension of the nanofluids, which leads to very minor change in the amount of bubbles formation in the vapour line of LHP in present numerical study.

### Table 3. Thermophysical properties of air and Al₂O₃-H₂O nanofluid at 298 K and 101325 Pa

<table>
<thead>
<tr>
<th>Properties</th>
<th>Air</th>
<th>Nanofluid (Al₂O₃-H₂O)</th>
</tr>
</thead>
<tbody>
<tr>
<td>ρ (kg/m³)</td>
<td>1.225</td>
<td>1013.06 1027.918 1087.354</td>
</tr>
<tr>
<td>C_p (J/kg.K)</td>
<td>1005</td>
<td>4115.05 4049.734 3807.729</td>
</tr>
<tr>
<td>k (W/m.K)</td>
<td>0.0257</td>
<td>0.62183 0.63074 0.66726</td>
</tr>
<tr>
<td>μ (Ns/m²)</td>
<td>0.001016</td>
<td>0.001028 0.001078</td>
</tr>
<tr>
<td>σ (N/m)</td>
<td>0.07194</td>
<td>0.07194 0.07194</td>
</tr>
</tbody>
</table>

Fig. 8. Contours of mixture density (kg/m³) for liquid-air flow inside the vapour line of LHP charged with: (a) pure water, (b) 0.5% Al₂O₃-H₂O, (c) 1% Al₂O₃-H₂O, and (d) 3% Al₂O₃-H₂O.

### 6. CONCLUSIONS

The key findings from the numerical simulation of nanofluid behavior inside the LHP in order to predict the fluid temperatures, pressure drop and bubble formations across the vapor line of LHP are:

- A rise in liquid temperatures is observed for LHP charged with nanofluid as compared to LHP charged with pure water, which indicates that the LHP charged with nanofluids able to absorb more heat from the surface due to enhanced heat removal in the LHP by higher thermal conductivity of nanofluid.
- Thermal properties of nanofluid very much depend on mass concentration of nanoparticles in the base fluid. So, nanofluid has higher value of thermal conductivity and thermophysical characteristics e.g. density, viscosity etc. which enhance the heat absorption capacity.
- It can be observed that the liquid temperature increases as the nanoparticle mass concentration increases, which yields the maximum heat transfer enhancement as found in current study.
- The presence of nanoparticles substantially increases the pressure drop of LHP. The pressure drop rises with the increase of nanoparticle mass concentrations.

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### REFERENCES


