Theoretical Model to Investigate the Heat Transfer Mechanism through a Heat Pipe with Graphene Oxide/Distilled Water as Working Fluid

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Abstract. This study investigates heat transfer through a heat pipe by analysing the evaporation and condensation of working fluid (distilled water) in two parts of the heat pipe to find the phase change phenomenon. A computational methodology was developed to simulate the phase change processes and a thermal model was simulated to determine the spatial and temporal temperature profile of the heat pipe and phase change location. The velocity and location of the phase change in the evaporator and condenser sections of the pipe were calculated at each stage, as was the temperature across the domain. The thermal efficiency of the heat pipe was improved by adding graphene oxide nanoparticles at an increase of 15%. When nanoparticles were added to the distilled water, the time of the phase change phenomenon was reduced in the evaporator and condenser sections. The capillary property was enhanced when nanoparticles were deposited in the wall of the evaporator’s wick structure. The temperature distribution of the phase change phenomenon was evaluated via the evaporation and condensation processes and the location of the interface line in the two main parts of the heat pipe.

Keywords: Heat transfer mechanism; evaporation; condensation processes; graphene oxide nanoparticles.

1. Introduction
A heat pipe is a passive cooling device; high amounts of heat can be transferred by evaporation and condensation across a range of temperatures in a short period. The pipe consists of a copper container with a capillary wick structure and three main components: evaporator, adiabatic and condenser. The wick structure works as a capillary property and generates the constraint of liquid to reduce velocity and pressure when it is poured into the evaporator. The consistency, structure and size of the wick materials also contribute to controlling the performance of the heat pipe. Different wicks are used depending on the application of the heat pipe, such as dissipation or heat transfer from one heat source to another.
The passive device (heat pipe) can be used to remove heat generated by friction and drag force from a layer of air on the frame of an aircraft or from heat generated by electronic devices. The heat pipe can also be used a heating cycle; for example, pre-heating the air in a steam boiler to improve the thermal

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efficiency of the boiler. The most recent use of heat pipes to transfer heat is to move geothermal heat from underground to the inside of greenhouses without using traditional means. The shut-down systems in nuclear reactors are composed of neutron absorbers (rod assembly) where heat decay can be removed using passive safety or containment systems. A flat heat pipe that combines the functions of current control rod assembly and passive cooling systems could avoid future accidents. Solar energy technologies, such as panels and solar water heaters, could use heat pipes as cooling devices to prevent the freezing and backflow of working fluid during the night.

A numerical procedure was developed to create predictive behaviour to analyse the heat transfer mechanism coming into contact with a phase change problem in the evaporation and condensation processes. The plan was executable in a nonlinear limited component programme for mathematical assessment. The outcomes clarified reasonable information within the accessible expository arrangement. Kim et al. investigated the vaporisation and condensation processes when water was used as the working fluid in heat pipe components [1]. Zainal et al. studied the heat transfer mechanism on working fluid temperature and the effect of latent heat on vaporisation and condensation. They studied energy release and the proportion and time taken for a liquid to vaporise when it was put in the evaporator section of a hollow heat pipe. The experiment was conducted in observational conditions to determine and control the charge of the working fluid and the release of energy in a liquid-vapour change phenomenon [2]. Siriwan et al. created a numerical model of a helical oscillating heat pipe to predict the time required to get to steady-state temperature under transient conditions. The temperature of the pipe wall and the heat transfer rate were evaluated [3].

Computer codes were developed for the design and simulation of a high-temperature heat pipe. The codes applied empirical equations, such as modifying the finite element technique. Computational fluid dynamics were applied to study the heat pipe working in different steady and unsteady conditions. Panda et al. analysed the saturated temperature of liquid-vapour interface conditions using a commercial computational fluid dynamic code. A three-dimensional unsteady numerical model was proposed to study the vapour zone, the temperature of the wall, the pressure of vapour and vapour velocity in a screen-mesh wick in a sodium heat pipe [4]. Senthilkumar et al. analysed heat pipe operation characteristics by applying the Taguchi approach. The study included calculating flow properties for the working fluid when it started from laminar, and it predicted the active nodes in the domain [5].

Ding et al. investigated three types of heat sinks using a 180 W high-powered diode lamp using the finite element analysis process. Thermal performance at different working angles was investigated in the optimised heat sinks. The working angles of the lamps greatly impacted the heat sink's thermal performance. The angle of operation was recommended for a heat sink with the best thermal efficiency. The findings played a significant role in the design of high-power diode lamps [6]. Mall evaluated the maximum heat transfer rate using the C++ program for different operating temperatures of a particular heat pipe. The selected fluid was water. The dimension was constant and the temperature was varied; the heat pipe was tested in a horizontal position. The heat transferred through the pipe was approximately 15 times the heat transferred through pure conduction in the heat pipe. Pure conduction occurs when the working fluid absorbs the latent heat and is converted to vapour. This could be considered low heat transfer. In contrast, the overall heat transfer for a large amount of energy from the heat sink to other parts was a lot higher [7].

Harri et al. studied flow specifications in the wick of a flat heat pipe when the working fluid inside the wick structure was subjected to boiling and condensing. The two-phase mixture model was applied.
The laminar boundary-layer flow was steady in the wick structure and was examined in both vertical and horizontal orientations. The governing equations were the partial differential of the boundary layer simplified using transformation. The transformed equations were then solved numerically using an iterative shooting technique. The study confirmed that orientation plays a significant role in the flow and temperature fields [8]. Beutician et al. developed a three-dimensional model to simulate the heat transfer rate of a heat pipe in a transient state. They calculated the temperatures for a domain consisting of a wall, wick and vapour core using the finite element method; numeric equations were simulated. The model attempted to predict the domain's transient temperature, velocity and heat transfer rate profiles. The estimated values were then compared with the experimental results of the studies. The experiment showed that the time needed to achieve steady-state agreed with the model [9]. Rashidian developed a computer simulation using MATLAB in what could be considered a simple tool for modelling and designing heat pipe heat exchangers. The results of the programme were compared to a pilot plant and demonstrated agreement with the experimental data. The programme was capable of figuring out economic evaluations for designing a heat exchanger, such as cash flow, payback and fuel-saving [10]. Chen et al. determined the influence of wick structure parameters using the theory of capillary limitation for the porosity powder diameter and thickness of the wick structure. A miniature heat pipe with sintered dendritic copper powder wicks was fabricated and tested. The maximum heat transfer rate reached over 13 W with an outer diameter of 3 mm and an effective heat pipe length of 20 cm. A simple correlation equation was proposed for the prediction of the maximum heat transfer rate of the sintered miniature heat pipe with dendritic powder [11]. Suh and Par investigated the effects of interfacial shear stress on the thermal performance of a micro-heat pipe with an axial flat-grooved channel. Shear stress was caused by the difference in velocity of the liquid and vapour phases. The heat pipe was tested using ammonia as the working fluid. The results were obtained mainly at the 300 K operating temperature, i.e. liquid and vapour flow passed through trapezoidal grooves [12]. Torii analysed convective heat transfer, adding graphene oxide nanoparticles that were coursed through a low level of cylindrical heat under heat transition conditions. A multi-heat pipe cooling device was studied when filled in various proportions (40%, 60%, 80% and 100%) in steady heat transition conditions. The results found that heat transfer was improved when the nanoparticles were suspended, so the most extreme estimation of the Nusselt number was over twice that of the pure working fluid. The heat pipe incrementally expanded the grouping of graphene oxide nanoparticles in the base working fluid and the greatest heat transfer upgrade yielded 0.075% of the fixed weight [13]. The graphene oxide and water nanofluid demonstrated lower thermal opposition compared with pure water. The ideal thermal obstruction was acquired when filled at 100%, and the heat transfer coefficient of the heat pipe expanded with the fixation of graphene oxide nanoparticles. Rui et al. theoretically investigated fully developed turbulent heat transfer characteristics of liquid sodium inside a high-temperature heat pipe. A model was built to predict the heat transfer coefficient and was dependent on universal velocity, turbulent eddy diffusivity, linear law and the inter-thermal boundary layer [14]. In this study, the phase change phenomenon was analysed by evaporating and condensing working fluid in two parts of a heat pipe. A computational methodology was developed to simulate the spatial and temporal temperature profile through the heat pipe. The velocity and location of the phase change in the evaporator and condenser sections were calculated. Graphene oxide nanoparticles were added to the base working fluid (distilled water) to improve the thermal efficiency of the heat pipe by reducing the
phase change time and location along the r-z axes. When nanoparticles were deposited in the wall of the evaporator, the capillary property of the wick was enhanced.

2. Mathematical Model

The assumptions of this case study are:

1. The temperature of the adjacent phase is equal to the prescribed constant temperature tube, which is normally the boiling temperature of the working fluid.
2. An energy balance must be satisfied at the interface.
3. The thermophysical properties of each phase are uniform and remain constant.
4. The distinction of density of liquid and vapour does not create a considerable nearby movement of fluid.
5. Heat transfer is controlled by an energy equation.
6. The material is homogenous and isotropic.

![Figure 1 Schematic of the heat pipe](image)

A cylindrical coordinates approach has been used to predict the temperature distribution through the heat pipe and explicate the phase change phenomenon of the working fluid. The working fluid changed from liquid to vapour due to the location of the boiling point in the evaporator and vice versa from a vapour to a liquid in the condenser. This is called a moving boundary problem because of the unexplored moving liquid–vapour interface. The moving boundary problem is nonlinear and rendered more intricate.

2.1 Evaporator section

\[
\frac{1}{\alpha} \frac{\partial T}{\partial t} + \frac{v_1}{\alpha_1} \frac{\partial T}{\partial x} = \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{1}{r^2} \frac{\partial^2 T}{\partial \theta^2} + \frac{\partial^2 T}{\partial z^2} \tag{1}
\]
when $\alpha = \frac{k}{\rho c_p}$.

All of the focuses have domain-limit (a-b-c-d) dimensions, as shown in Figure 2. It has been assumed that a half-domain of the cylinder has been used to design the model because of its symmetrical shape. Numerical simulations have been performed using an explicit technique.

$$\frac{T_{i+1,j}^n-2T_{i,j}^n+T_{i-1,j}^n}{\Delta r^2} + \frac{1}{r_i}\frac{T_{i+1,j}^n-T_{i-1,j}^n}{2\Delta r} + \frac{T_{i,j+1}^n-2T_{i,j}^n+T_{i,j-1}^n}{\Delta z^2} = \frac{T_{i,j}^{n+1}-T_{i,j}^n}{\Delta t} + \frac{\nu_i}{\alpha} \frac{T_{i,j}^n-T_{i,j-1}^n}{2\Delta z}$$

(2)

The boundary conditions are at $z=0$ and $r=R$, $-k\frac{\partial T}{\partial r} = q^{-}$ (heat flux) at $z=50$, $T=T_v$.

**Figure 2** The two-dimensional computational mesh of the evaporator

### 2.1.1 Temperature distribution through the liquid phase

In the liquid phase, a two-dimensional transient energy equation with polar coordinates allows the temperature distribution in an interface region of limited space to be determined before a change of phase occurs on an interface line, as shown in Figure 2. The interface line is located 35 cm from the left side of the evaporator. The liquid temperature was increased to a saturated temperature (boiling point) at the interface line; this records the vaporisation process. At this time, the remaining fluid has stagnated and its velocity is equal to zero. The computer program, depending on the boundary conditions, has a greater capacity to measure temperature profiles through the r and z plane when the explicit scheme is applied.

### 1. Temperature at Point a

Point a represents the starting location of the liquid phase in the evaporator. At this point, r and z coordinates are 0, 0 after applying the energy balance to this boundary node, which is a corner node.

$$T_{i,j}^{n+1} = 4\lambda_1 T_{i+1,j}^n + (1 - 6\lambda_1) T_{i,j}^n + 2\lambda_1 T_{i,j+1}^n - 2 \frac{\lambda_1 q^+ \Delta r}{\kappa}$$

(3)

where $\lambda_1 = \frac{\alpha_1 \Delta t}{r_i \Delta r^2} \leq 0.5$ is the Fourier number and $\lambda_1 = \frac{\alpha_1 \Delta t}{r_i \Delta r}$. 


2. Temperature distribution through the r-axis
To find the temperature through the r-axis in the liquid phase, an energy balance on the boundary line was applied. For \( r > 0 \) and \( z = 0 \), the explicit scheme was constructed as:

\[
T_{i+1,j}^{n+1} = \lambda T_{i+1,j}^n + \lambda T_{i,j+1}^n + (1 - 4 \lambda_x) T_{i,j}^n + 2 \lambda_1 T_{i,j+1}^n - \frac{2 \lambda q^\alpha \Delta r}{K} \tag{4}
\]

3. Temperature distribution at corner point b
The energy balance was applied for this boundary node, and the temperature was estimated as:

\[
T_{i,j}^{n+1} = 2 \lambda T_{i,j}^n + (1 - 4 \lambda) T_{i,j}^n + 2 \lambda T_{i,j+1}^n - \frac{2 \lambda q^\alpha \Delta r}{K} \tag{5}
\]

4. Temperature distribution through line bc in the liquid phase
This is also a boundary line; therefore, the energy balance was needed. Temperature distribution was obtained when \( r = R \) and \( z > 0 \), i.e. \( i = R \) to \( R \) and \( j = 0 \) to \( 349 \).

\[
T_{i,j}^{n+1} = 2 \lambda T_{i-1,j}^n + (1 - 4 \lambda) T_{i,j}^n + 2 \lambda T_{i,j-1}^n - \frac{2 \lambda q^\alpha \Delta r}{K} \tag{6}
\]

5. Temperature distribution through the z-axis in the liquid phase
The temperature was calculated by applying the two dimensions of the energy equation when \( r = 0 \) and \( z > 0 \), i.e. \( i = 1 \) to \( 1 \), \( j = 2 \) to \( 349 \) and \( T_{i+1,j} = T_{i-1,j} \).

\[
T_{i,j}^{n+1} = 4 \lambda T_{i+1,j}^n + (1 - 6 \lambda) T_{i,j}^n + \lambda T_{i,j+1}^n + \lambda T_{i,j-1}^n \tag{7}
\]

6. Temperature distribution through the r-z plane in the liquid phase
The temperature was calculated by applying the two dimensions of the energy equation where \( r > 0 \), \( i = 1 \) to \( L-1 \), \( z > 0 \) and \( j = 2 \) to \( 349 \).

\[
T_{i,j}^{n+1} = (\lambda + \frac{\lambda_1}{2}) T_{i+1,j}^n + (1 - 4 \lambda) T_{i,j}^n + (\lambda - \frac{\lambda_1}{2}) T_{i,j-1}^n + \lambda T_{i,j+1}^n + \lambda T_{i,j-1}^n \tag{8}
\]

2.1.2 Temperature distribution at the interface line cd
To satisfy the fundamental relationship at an interface, the temperature of adjacent phases should be equal to the boiling temperature at the interface line. \( T_v = T_l = T_{\text{boiling}} \) at \( z = \delta(t) \), where \( T_v \) is the vapour temperature, \( T_l \) is the liquid temperature and \( T_{\text{boiling}} \) is the boiling temperature of the liquid.

The energy balance must be satisfied at the interface; therefore, the energy equation at the liquid–vapour interface should be expressed as:

\[
k_v \frac{\partial T}{\partial z} - k_v \frac{\partial T}{\partial z} = \rho L_a \frac{\delta(t)}{\partial t} \text{ at } z = \delta(t) \tag{9}
\]

\[
k_v \frac{\partial T}{\partial z} = \rho L_a \frac{\delta(t)}{\partial t} \text{ at } z = \delta(t) \tag{10}
\]
This is a moving boundary condition that supplies the energy equation to evaluate the temperature distribution through a phase change. Alwan has outlined the details of the velocity of the phase change [18].

\[
\frac{d\delta(t)}{dt} = V_a = \frac{\text{heat flux}}{[\rho L_a + \rho c_l (T_v - T_l)]} \quad (11)
\]

1. Phase change at point \(d\)
This point on the domain surface of the phase change is noted in Figure 2. The interface line between the liquid and vapour phase change is assumed to be propagated through the \(r\)-axis. The finite-difference form is written as follows:

\[
k L \frac{T_{ij+1} - T_{ij-1}}{2\Delta z} = \rho L_a V_a
\]

\[
T_{ij-1} = T_{ij+1} - \frac{\rho L_a V_a 2\Delta z}{k_L} \quad (12)
\]

\[
T_{ij}^{n+1} = 4\lambda T_{ij}^{n+1} + (1 - 6\lambda)T_{ij}^{n} + 2\lambda T_{ij+1}^{n} - \frac{2\lambda}{k_L} \rho L_a V_a \Delta z
\]

\[
\quad + (\lambda + \frac{\Delta}{2}) T_{ij+1}^{n+1} + (1 - 4\lambda) T_{ij+1}^{n} + (\lambda - \frac{\Delta}{2}) T_{ij+1}^{n} - \frac{2\lambda}{k_L} \rho L_a V_a \Delta z \quad (14)
\]

2. Phase change through the \(r\)-axis
For \(r>0\) and \(z=350\), the following equation was used to demonstrate the phase-change mechanism through the \(r\)-axis.

\[
T_{ij}^{n+1} = (\lambda + \frac{\Delta}{2}) T_{ij}^{n+1} + (1 - 4\lambda) T_{ij}^{n} + (\lambda - \frac{\Delta}{2}) T_{ij}^{n} - 2\lambda \frac{q^a \Delta z}{k} - \lambda T_{ij+1}^{n} - \frac{2\lambda}{k_L} \rho L_a V_a \Delta z
\]

\[
\quad + (\lambda + \frac{\Delta}{2}) T_{ij+1}^{n+1} + (1 - 4\lambda) T_{ij+1}^{n} + (\lambda - \frac{\Delta}{2}) T_{ij+1}^{n} + (\lambda + \frac{\Delta}{2}) T_{ij+1}^{n+1} + (1 - 4\lambda) T_{ij+1}^{n} + (\lambda - \frac{\Delta}{2}) T_{ij+1}^{n} - \frac{2\lambda}{k_L} \rho L_a V_a \Delta z \quad (15)
\]

3. Phase change at point \(c\)
This point is a boundary node; therefore, a heat balance must be performed when \(r=38\) and \(z=350\).

\[
T_{ij}^{n+1} = 2\lambda T_{ij+1}^{n} + (1 - 4\lambda) T_{ij}^{n} + 2\lambda T_{ij-1}^{n} - 2\lambda \frac{q^a \Delta z}{k} - \lambda T_{ij+1}^{n} - \frac{2\lambda}{k_L} \rho L_a V_a \Delta z
\]

\[
\quad + 2\lambda T_{ij+1}^{n+1} + (1 - 4\lambda) T_{ij+1}^{n} + (\lambda - \frac{\Delta}{2}) T_{ij+1}^{n} + (\lambda + \frac{\Delta}{2}) T_{ij+1}^{n+1} + (1 - 4\lambda) T_{ij+1}^{n} + (\lambda - \frac{\Delta}{2}) T_{ij+1}^{n} + (\lambda + \frac{\Delta}{2}) T_{ij+1}^{n+1} + (1 - 4\lambda) T_{ij+1}^{n} + (\lambda - \frac{\Delta}{2}) T_{ij+1}^{n} - \frac{2\lambda}{k_L} \rho L_a V_a \Delta z \quad (16)
\]

2.1.3 Temperature distribution through the vapour phase in the evaporator
1. Temperature distribution through the \(z\)-axis
At the \(z\)-axis where \(z>350\), \(r=0\) and \(c = \frac{\Delta t v_a}{\Delta z}\) is the courant number \(0 \leq c \leq 1\), the following energy equation should be used:

\[
T_{ij}^{n+1} = 4\lambda T_{ij+1}^{n+1} + (1 - 6\lambda) T_{ij}^{n+1} + \lambda - \frac{c}{2} T_{ij+1}^{n+1} + (\lambda + \frac{c}{2}) T_{ij-1}^{n+1} \quad (17)
\]

2. Temperature distribution in the \(r\)-\(z\) plane in the vapour phase
\[
T_{ij}^{n+1} = \lambda + \frac{\Delta}{2} T_{ij+1}^{n+1} + (1 - 4\lambda) T_{ij}^{n+1} + (\lambda - \frac{\Delta}{2}) T_{ij+1}^{n+1} + (\lambda + \frac{\Delta}{2}) T_{ij+1}^{n+1} \quad (18)
\]
2.2 Adiabatic Section
The adiabatic section was designed using suitable dimensions, materials and insulation properties so that the heat pipe would work as a cooling device. The working fluid vapour was passed through this section with no loss of temperature. The vapour temperature of the working fluid was considered the initial value of the temperature in the condenser section because there was no heat transfer between the system and its surroundings in the adiabatic section when it was insulated by a layer of glass wool. The phase change phenomenon did not occur in this section [22].

2.3 Condenser section
The working fluid entered the condenser section as a vapour for 120 min. The vapour changed to liquid at the interface line and phase change occurred at 45 cm from the z-axis. Heat was rejected by the high-fins that were specifically designed for this section. The liquid phase continued to the last node of the condenser’s dimensions. The energy equation was applied to predict the temperature distribution for the condenser section in two zones (vapour–liquid), as shown in Figure 3.

![Figure 3 The two-dimensional computational mesh of the condenser](image_url)

2.3.1 Temperature distribution through the vapour phase in the condenser
The domain area of the vapour phase and the interface line was evaluated at 45 cm from the left, as illustrated in Figure 3. The initial temperature in the condenser was the vapour temperature when it left the evaporator after two hours in the heat pipe.

1. Temperature distribution through the z-axis
Where \( r=0 \) and \( z>0 \), i.e. \( i=1 \) to 1 and \( j=0 \) to 449, then the energy equation was used. The equation was represented with an explicit technique when \( r=0 \) as:

\[
T_{i,j}^{n+1} = 4\lambda T_{i+1,j}^n + (1 - 6\lambda) T_{i,j}^n + (\lambda - \frac{c_p}{2}) T_{i+1,j+1}^n + (\lambda + \frac{c_p}{2}) T_{i,j-1}^n
\]  (19)

2. Temperature distribution through the r-z plane in the vapour phase
Temperatures of the vapour phase in the r-z plane were calculated at \( r>0 \) and \( z>0 \), i.e. \( i=2 \) to \( L-1 \) and \( j=2 \) to 449. The energy equation was used to estimate the temperatures through the r-z plane:

\[
T_{i,j}^{n+1} = (\lambda + \frac{c_p}{2}) T_{i+1,j}^n + (\lambda - \frac{c_p}{2}) T_{i-1,j}^n + (1 - 4\lambda_3) T_{i,j}^n + (\lambda - \frac{c_p}{2}) T_{i,j+1}^n + (\lambda + \frac{c_p}{2}) T_{i,j-1}^n
\]  (20)
2.3.2 Temperature distribution at the interface line fg

The interface energy at \(z=z_\delta(t)\) was represented in the equation 9, for the condensation process it was expressed as:

\[
-k_v \frac{\partial T}{\partial z} = \rho \frac{\partial T}{\partial t} + \frac{\partial T}{\partial t} = \frac{\text{heat flux}}{\rho L_a + \rho_v c_v (T_v - T_L)}
\]  

(21)

1. Temperature distribution at point g

Where \(r=0\) and \(z=450\), the phase change from vapour to liquid began.

\[
T_{ij-1} = T_{ij+1} + \frac{2}{k_v} \rho_v L_v V \Delta z
\]  

(23)

\[
T_{ij}^{n+1} = 4\lambda T_{i+1,j}^n + (1 - 6\lambda) T_{ij}^n + 2\lambda T_{ij+1}^n + \frac{2\lambda}{k_v} \rho_v L_v V \Delta z
\]  

(24)

2. Temperature distribution at line gf

Where \(r>0\) and \(z=450\), for sweep \(i=2\) to \(L-1\) and \(j=450\) to \(450\).

\[
T_{ij}^{n+1} = (\lambda + \frac{\lambda_1}{2}) T_{i+1,j}^n + (1 - 4\lambda) T_{ij}^n + \left(\lambda - \frac{\lambda_1}{2}\right) T_{i-1,j}^n + 2\lambda T_{ij+1}^n + \frac{2\lambda}{k_v} \rho_v L_v V \Delta z
\]  

(25)

2.3.3 Temperature distribution through the liquid phase in the condenser

It was assumed that the vapour would change to liquid after interface line gf. Then, the temperature distribution of the liquid domain in the condenser was simulated using the energy equation.

1. Temperature distribution through the z-axis in the liquid phase

When \(r=0\), \(z>0\), i.e. \(i=1\) to \(1\), \(j=451\) to \(m-1\) and \(c_1 = \frac{\nu \Delta t}{\Delta z}\), the energy equation was presented explicitly as:

\[
T_{ij}^{n+1} = 4\lambda T_{i+1,j}^n + (1 - 6\lambda) T_{ij}^n + \left(\lambda - \frac{\lambda_1}{2}\right) T_{i,j+1}^n + \left(\lambda + \frac{\lambda_1}{2}\right) T_{i,j-1}^n
\]  

(26)

2. Temperature distribution through the r-z plane in the liquid phase

When \(r>0\) and \(z>0\), i.e. \(i=2\) to \(L-1\) and \(j=451\) to \(m-1\), the energy equation was formed explicitly as:

\[
T_{ij}^{n+1} = (\lambda + \frac{\lambda_1}{2}) T_{i+1,j}^n + (1 - 4\lambda) T_{ij}^n + \left(\lambda - \frac{\lambda_1}{2}\right) T_{i-1,j}^n + \left(\lambda + \frac{\lambda_1}{2}\right) T_{i,j+1}^n + \left(\lambda + \frac{\lambda_1}{2}\right) T_{i,j-1}^n
\]  

(27)

3. Results and Discussion

Figure 4 shows the temperature distribution through the liquid phase in the evaporator with respect to the r-z axis. The temperature ranged from 26 °C to 58 °C during the 60 min when the heat pipe was in operation. Phase change did not occur in the evaporator at this time because of the heat flux, and the latent heat supplied was not enough to cause the liquid to change to vapour. Figure 5 shows that the temperature of the phase change was considered at 96–96.5 °C. The phase change position was displayed in the cylindrical enclosure (evaporator) at 180 min when \(z=35\) cm. The latent heat of the distilled water
was estimated at 337 KJ/kg in this model, which calculated that the phase change position shifted. The interface line of the phase change varied according to the latent heat supply and the heat current; the speed was equal to 0.0003431 m/s in the step position. The isothermal contour map in Figure 6 identifies the temperature range of the vapour phase in the r-z plane at 180 min. The working fluid acquired more latent heat and energy due to the phase change from liquid to vapour when the time reached two hours. The maximum vapour temperature was 108 °C in the domain at the interface line of the evaporator.

![Figure 4](image)

**Figure 4** Temperature distribution through the liquid phase at 60 min in the evaporator

![Figure 5](image)

**Figure 5** Temperature distribution of the phase change at z=35 cm and 180 min in the evaporator
Figure 6 The isothermal contour of the temperature distribution through the r-z plane in the vapour phase at 180 min in the evaporator

Figure 7 indicates the temporal temperature distribution in the area of the cylindrical enclosure in different regions during evaporation. When the temperature increased after three hours, the liquid–vapour interface line shifted to another position, especially in the upper portion. The shift happened because the heat flux was near the upper layer. When the working fluid reached the evaporation temperature of 62 °C at 120 min, the interface line of the liquid–vapour phase was calculated at 35 cm. When the temperature increased up to evaporation point, the working fluid was not immediately vaporising, but it was needed for long enough to convert all of the liquid to vapour by increasing the temperature of the working liquid molecules and latent heat. After the working fluid evaporated, the latent heat increased and the structure of atoms was less stable in lattice positions.

Figure 7 Temporal temperature distribution through the evaporator at r=19 mm and z=50 cm
Figure 8 displays the temperature distribution at the location $r=0$ at different evaporation times. The temperature response at a positioning layer less than 35 cm showed that the working fluid remained liquid after 60 min, which implies that low-temperature values and latent heat were not enough to cause the phase change phenomenon. This means that the phase change did not occur until 120 min, as shown by the temperature response function at 60 and 120 min. This figure demonstrates that the phase change position rises when the working fluid heated layer moves forward very slowly. The maximum temperature was 108°C after three hours and reached a steady-state of device operation.

Figure 9 explains the isothermal temperature distribution of the phase change phenomenon at 180 min into the evaporation process. It depicts the liquid phase in the first section of the heat pipe and before the interface line at 35 cm. When the working fluid reached an evaporation temperature of 60°C under a vacuumed pressure of 15 KPa inside the enclosure, the interface line of the phase change was located at 35 cm. During this investigation, the estimation of temperature at an interface location was around 94°C. The working fluid at this point was needed to acquire latent heat to change to vapour, as referenced by Muhieddine et al. [20].

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**Figure 8** Temperature distribution at the $z$-axis at $r=0$ in the evaporator at different periods
Figure 9 Isothermal contour map of the temperature distribution of evaporation at 180 min

The temperature distribution in the condenser was different from the evaporator. The condensation process started 120 min after vapour entered the condenser at 68 °C. The vapour was lost as latent heat and converted to liquid, and the phase change occurred at 45 cm. Figure 10 shows the isothermal contour map of temperature distribution through the r-z plane in the vapour phase at 120 min in the condenser. Figure 11 indicates the phase change temperature at 180 min when the interface line was at 45 cm.

Figure 10 The isothermal contour of temperature distribution through the r-z plane in the vapour phase after 120 min in the condenser
Figure 11 Temperature distribution of the phase change at z=45 cm after 120 min in the condenser

Figure 12 represents the isothermal contour of temperature distribution through the r-z plane for the liquid phase at 120 min in the condenser when the temperature of the liquid decreased to 49.5 °C. During this investigation, the phase change temperature was around 94 °C at 180 min. The condensation and conversion of vapour to liquid did not happen quickly. When it reached the phase change temperature, it continued to expend energy and needed more time for heat transition. The temperature decreased; therefore, the design of the adiabatic and condenser sections of the heat pipe with suitable dimensions and materials to work as a cooling device was effective.

Figure 12 The isothermal contour of temperature distribution through the r-z plane for the liquid phase at 120 min in the condenser
Figure 13 explains the temperature distribution through the z-axis at r=0 of the condenser at different periods. The condenser was operational for over two hours, and the vapour of the working fluid changed to liquid when the interface line for the z-axis was at 45 cm. The temperature values decreased from 68–54 °C over two hours. When the temperature increased in the evaporator, the dissipation of heat increased and operated the cooling process. Figure 14 shows the temporal temperature values through the centre of the condenser; the final nodes at r=0 and z=75 cm were satisfied throughout the working time.

![Figure 13](image1.png)

**Figure 13** Temperature distribution through the z-axis at r=0 of the condenser section at different periods

![Figure 14](image2.png)

**Figure 14** Temporal temperature differences through the centre of the condenser at z=75 cm
When graphene oxide nanoparticles were added to the base working fluid (distilled water), the thermal performance of the heat pipe improved and the phase change occurred in 60 min at 30 cm on the z-axis of the evaporator. Figure 15 shows that the evaporation temperature was less than 60 °C when graphene nanofluid was used. The weight fraction percentages of graphene nanoparticles were added to the distilled water in a range from 0.01–0.075 wt.%. Phase changes time occurrence and location of its enhanced caused by transfer high amount of heat at least period of operating time. The nanoparticles deposited on the wick near the evaporator wall improved its capillary property and reduced the pressure of the liquid entering the evaporator. The effect of nanoparticles in this analysis is in agreement with results from a study by Zhao [19].

The efficiency of the heat pipe was enhanced by increasing the absorption heat from the heat source caused by the addition of graphene oxide nanoparticles to the base working fluid. This enhanced the dissipation of heat in the condensation process when compared to the work by Alijani [21]. The heat pipe activity increased with the heat load as thermal resistance is lower at higher heat loads. The vapour movement from the evaporator to the condenser was primarily due to density. Figure 16 explains the effect of adding graphene oxide nanoparticles to the base working fluid on the condensation process. This is where heat was transferred at a high temperature from the condenser wall and the drop in temperature from the evaporator to the condenser enabled the phase change to occur in less time at 40 cm on the z-axis.

![Figure 15](image-url)

**Figure 15** Temperature distribution of the nanofluid during evaporation at different time intervals at $r=0$ mm
4. Conclusions
A mathematical model predicted the heat transfer mechanism inside a flat heat pipe by analysing the spatial and temporal temperature during the evaporation and condensation processes. The prediction was dependent on the design of the three main parts of the heat pipe and the position and amount of heat flux in the evaporator section. This model was simulated numerically using an explicit technique, which provided more accurate results and damped oscillation errors than other long-term calculation methods. The new assumption that the boundary layer moved was used to forecast distance in the cluster domain with time to learn the phenomenon of the phase change in the evaporator and condenser. The interface line of the liquid–vapour phases jumped into an adjacent position by acquiring latent heat in the evaporator and releasing energy in the condenser when vapour converted to liquid. The interface line moved with time in the z-direction. It was proven that the phase change occurred at 35 cm from the left of the evaporator, where heat flux was fixed. The maximum temperature in the evaporator was 381 K and 370 K in the condenser after 180 min from the start of operation. After 180 min, steady-state was reached, the thermodynamic cycle started and the evaporator temperature decreased. In the condenser, temperature distribution was evaluated through the z-axis and phase change location at 45 cm. This was achieved when the heat pipe had been operational for two hours. Graphene nanoparticles affected the time the phase change occurred as it happened in less than two hours. The phase change location moved to 30 cm from the z-axis of the evaporator and 40 cm from the condenser axis. Heat pipe thermal efficiency was enhanced by 15% because of the addition of the graphene oxide nanoparticles.
Nomenclature

- $c_{pL}$: Specific heat capacity of liquid at constant pressure J/kg.k
- $c_{pv}$: Specific heat capacity of vapour at constant pressure J/kg.k
- $k_L$: Thermal conductivity of liquid phase W/m.k
- $k_v$: Thermal conductivity of vapour phase W/m.k
- $k_{nf}$: Thermal conductivity of nanofluid W/m.k
- $k_{bf}$: Thermal conductivity of base fluid W/m.k
- $V_a, V^*$: Velocities of interface m/s
- $T$: Temperature k
- $t$: Time s
- $\rho$: Density kg/m$^3$
- $\alpha_L$: Thermal diffusivity of liquid phase m$^2$/s
- $\alpha_v$: Thermal diffusivity of vapour phase m$^2$/s
- $\rho_L$: Density of liquid phase kg/m$^3$
- $\rho_v$: Density of vapour phase kg/m$^3$
- $\delta(t)$: Location of liquid-vapour interface m
- $\Delta r$: Increment distance through r-axis mm
- $\Delta z$: Increment distance through z-axis cm
- $\lambda_{1,2,3,4}$: Convergent

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