แบบจำลอง 2 มิติ สถานะชั่วครู่เพื่อทำนายแนวโน้มของอุณหภูมิที่ผนังท่อความร้อน
Two-dimensional Transient Model for Prediction of Temperature Profile at Heat Pipe Wall

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บทคัดย่อ

การศึกษาในครั้งนี้มีวัตถุประสงค์เพื่อศึกษาแนวโน้มการกระจายตัวของอุณหภูมิที่สถานะชั่วครู่ของท่อความร้อนที่ทำจากท่อทองแดงในระหว่างเริ่มต้นให้อุณหภูมิแบบคงที่ที่ผนังด้านนอกของส่วนทำระเหยจนอุณหภูมิที่ผนังท่อเข้าสู่สถานะคงที่ โดยแบบจำลอง 2 มิติ ที่ศึกษาจะใช้ระเบียบวิธีเชิงตัวเลขตามระเบียบวิธีผลต่างสืบเนื่อง และผลที่ได้จากการกระจายตัวของอุณหภูมิที่สถานะชั่วครู่ของท่อความร้อนทั้ง 3 ส่วน จะนำมาใช้ในการเปรียบเทียบกับผลที่ได้จากระเบียบวิธีเชิงตัวเลขของแบบจำลอง 2 มิติ โดยจากการศึกษา พบว่า ผลจากการทำนายแนวโน้มของอุณหภูมิทางทฤษฎีเปรียบเทียบกับผลจากการทดลองจะมีแนวโน้มที่ใกล้เคียงกัน และเมื่อเปรียบเทียบกับแบบจำลองเชิงตัวเลขของ Mistry และคณะ 2010 ก็จะมีแนวโน้มที่เหมือนกัน ดังนั้น การกระจายตัวของอุณหภูมิของท่อความร้อนที่ได้จากแบบจำลองเป็นแบบจำลองที่ประสบผลสำเร็จในการจำลองแบบ 2 มิติ

Abstract

Distribution trend of transient temperature profile of copper heat pipe from the start-up of outer wall temperature of evaporator section until the temperature was steady was studied. A two-dimensional model used numerical solution which based on finite difference method. The transient temperature distributions measured in all 3 heat pipe sections were compared with results from the numerical solution of the two-dimensional model that utilized the concept of temperature distribution in the pipe wall during transient operation. Results showed that the experimental trend and the theoretically predicted temperature profiles were similar. Moreover, there was good agreement when compared with the numerical simulation of Mistry et al. (2010). Therefore, the numerical-validated heat pipe temperature distributions were successfully simulated in a two-dimensional model.

คำสำคัญ : ท่อความร้อน 2 มิติ สถานะคงที่ แนวโน้มของอุณหภูมิ

Keywords : Heat Pipe, Two-dimensional, Transient, Temperature Profiles

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1. Introduction

Heat pipes are currently used in a wide variety of heat transfer related applications such as electronic and electrical packing, heat exchangers, production tools, engines, industry, and solar systems. A heat pipe can efficiently transfer heat through the use of internal phase change. Analyses of heat pipe operations both analytical and numerical have been performed extensively by many investigators. Almost all of the analytical studies have been of the experimental type, which is difficult and requires great detail. As such, modeling is often used to predict the performance of heat pipes. Performance analysis including the concept of temperature distribution, frozen startup behavior of a heat pipe, an equation for analyzing temperature distribution of the heat pipe wall, design of the heat pipe, and variable conductance devices for spacecraft thermal control has been conducted. During the analysis, subjects considered included hydrostatics, hydrodynamics, heat transfer into and out of the pipe, fluid selection, materials compatibility and variable conductance of a heat pipe (Ates et al., 2010).

Several approaches to numerical simulation of heat pipes have been previously studied. Mahjoub and Mahtabroshan (2008) studied the effect of different parameters, such as wall thermal conductivity, wick porosity, and heat pipe dimensions on the heat pipe operation. The governing equations have been solved by using the SIMPLE algorithm with collocated grid scheme. The results show that thermal resistance of a conventional cylindrical heat pipe grows with increasing wick porosity and decreases with an increase in wall thermal conductivity and heat pipe radius. Ismail and Mirandat (1997) proposed the two-dimensional axisymmetrical model for a rotating porous wicked heat pipe. The proposed model is based upon the conservation equations that were numerically solved using the algorithm SIMPLE. Xiaohong et al. (2011) studied the two-dimensional transient thermal analysis of the PCM canister of a heat pipe receiver under microgravity. Wang and Vafai (2000) studied Analytical models for predicting the transient performance of a type of plate heat pipe. The results revealed that the thermal diffusivity, the thickness of the wall and the wick, and the heat input pattern, affect the heat pipe time constants. The wicks create the main thermal resistance resulting in the largest temperature drop in the heat pipe. Shabgard and Faghri (2011) studied the performance characteristics of cylindrical heat pipes with multiple heat sources. It was found that the exclusion of the effects of axial heat conduction in the wall can result in an overestimation of the pressure drops in a heat pipe.

Axial heat conduction is also important to obtain accurate temperature distribution along the heat pipe wall. Jumroonrut and Pitakthapanaphong (2009) studied the filling and solidification simulation of the aluminum casting process. This work investigates the casting of aluminum via finite difference simulation. Zhua and Vafai (1999) studied a two-dimensional analytical model for cylindrical heat pipes. They investigated cylindrical heat pipes incorporating the effects of liquid-vapor coupling and non-Darcian transport. A close-form solution was obtained based on an in-depth integral method. The results show that the interfacial effects are small and can be neglected. However, substantial errors can occur when using Darcy's law in calculating the liquid pressure distributions as well as the maximum heat removal capability of the heat pipe. Thuchayapong et al. (2012) studied the effect of capillary pressure on performance of a heat pipe. Their investigation of two-dimensional heat transfer and fluid flow in a heat pipe at a steady state was numerically simulated using the Finite Element Method (FEM). It found that the two-dimensional, steady-state heat transfer and fluid flow for heat pipe operations have been successfully simulated using FEM and numerical results of the vapor and wall temperature distributions were in good agreement with the experimental data obtained by Huang et al. (1993). The standard deviations of vapor and wall temperature were 0.48 °C and 1.78 °C, respectively. Ates et al. (2010) studied the transient conjugated heat transfer in thick walled pipes. Their investigation involved two-dimensional wall and axial fluid conduction. The problem is solved numerically by a finite-difference method. The results are presented by non-dimensional interfacial heat flux values, and it is observed that heat transfer characteristics are strongly dependent on the wall thickness ratio, wall-to-fluid thermal conductivity ratio, wall-to-fluid thermal diffusivity ratio and the Peclet number. Mishra et al. (2005) studied the effect of temperature dependent thermal conductivity on transient conduction and radiation heat transfer. The thermal conductivity is assumed to vary linearly with temperature. The radiative part of the energy equation was solved using the collapsed dimension method. The energy equation was solved using the alternating direction implicit scheme. Results for the effects of the variable thermal conductivity were found for temperature and heat flux distributions. Mistry et al. (2010) studied the transient behavior of a heat pipe during start-up. The governing equations and boundary conditions were discretized using the finite difference technique. Euler's explicit method was utilized to solve the set of equations and the results were used to quantify the time required to reach steady state. It was found that the experimental transient temperature profile for the adiabatic section compared well with the theoretically obtained temperature profiles. It was concluded from the above comparison that the axial conduction during the transient operation of a heat pipe is significant and, therefore, cannot be neglected. The experimentally obtained temperature profiles, for both the transient and steady state, were successfully compared with predictions from the two-dimensional model.

The present work, therefore, focuses on the temperature profile of heat pipe wall in a two-dimensional transient model and validates it with the experimental results. The model was solved numerically with the finite difference method. The concept of the temperature distribution as proposed by Faghri (1995) was utilized in the analysis to comprehensively simulate the start-up period. The experiments were conducted with a low-temperature copper mesh heat pipe, and the temperature distribution was measured as a function of time for different constant outer wall temperature in the evaporator section. The temperature of evaporator, condenser, and adiabatic sections was measured with time to analyze the transient temperature profiles of the heat pipe and the attainment to the steady state. The experimental and theoretical results, in terms of transient temperature profiles in all three regions of the heat pipe, the steady-state profile, and the theoretical and experimental times required to attain the steady state are compared in this study.

2. Materials and Methods

2.1 Conceptual Framework

2.1.1 Two-Dimensional Model

The proposed model was formulated based on the assumptions that thermophysical properties of the wall are constant throughout the operation.

The schematic of the heat pipe is shown in Figure 1 with the coordinate system used in the present analysis. The heat pipe was also divided into three sections: evaporator section, adiabatic section, and condenser section. A two-dimensional heat conduction equation was used to model the wall of the heat pipe.

The equation for the wall is given as

$$\rho_w C_{p,w} \frac{\partial T}{\partial t} = k_w \frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right) + k_w\left(\frac{\partial^2 T}{\partial y^2}\right) \quad (1)$$

Where ρ_w is the density of the pipe wall. $C_{p,w}$ is the specific heat of pipe wall. k_w is the thermal conductivity of pipe wall.

For a low temperature heat pipe, the analysis of the transient heat transfer in the wall was divided into three time zones. At time t = 0, the temperature was applied to the evaporator wall outer surface, and the pipe wall started to increase the temperature. All of the input temperature was stored within the pipe wall, causing an increase in the temperature of the wall with time. A considerable portion of the input temperature was transported to the adiabatic and condenser sections in the form of conduction. The amount of temperature transported to the condenser section increased gradually and eventually became equal to the input temperature at the steady state.

The initial condition of wall temperature is

$$t = 0, \qquad T = T_i$$

Where T_i is the uniform initial condition temperature. The corresponding boundary conditions for each of the time stages follow.

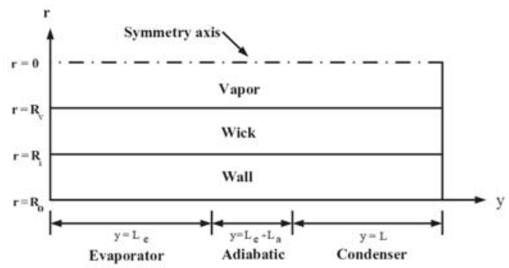

Figure 1 Coordinate System of a Heat Pipe

The time period during the growing temperature was within the wall. Three different boundary conditions were used for the outer wall of each of the three regions. In the evaporator and condenser section, constant temperature was supplied during the entire operation. The adiabatic section was insulated, and hence, the adiabatic boundary condition was used.

$$\left\{\begin{array}{ll} T_e = 60\ ^{\circ}C & 0 \le y < L_e \\ \dfrac{\partial T}{\partial r} = 0 & L_e \le y < L_e + L_a \\ T_c = 20\ ^{\circ}C & L_e + L_a \le y < L \end{array}\right\}$$

The coolant flow rate on the condenser section was in the laminar region. At the wall-wick interface, no heat transfer took place during this time. Therefore,

$$\frac{\partial T}{\partial r} = 0 \text{ at } r = R_i \text{ for all } y$$

Since both ends of heat pipe were insulated

$$\frac{\partial T}{\partial y} = 0 \text{ at } y = 0 \text{ and } y = L$$

2.1.2 Numerical Scheme

The governing equations and boundary conditions were discretized using the finite difference technique. The spatial derivatives were approximated using the five-point second-order central difference approximation for the interior points and four-point second-order approximation for the boundary nodes.

The governing equations are parabolic equations. The coefficients are constant, and thus, they become non-linear equations. The Forward Time Center Space method (FTCS) was utilized to solve the set of equations. According to this scheme, the temperature was known at every node at time t. Temperature for each node at time $t + \Delta t$ was calculated using the values of neighboring nodes at time t. The marching procedure was continued until attaining a steady state. Time step was calculated using stability criteria for the governing equations. The solution procedure, the time required for the temperature to reach steady state was determined by detecting a temperature rise at the wall in the evaporator section. When the temperature rose from the initial temperature, it was assumed that the temperature had reached to transient temperature.

The time step was taken according to stability criteria, which depends on the grid size. As the number of nodes is increased, the time step will decrease. In the present analysis, the time step was taken to be 10-5 sec. Grid independence was checked by examining the percentage change of the steady-state temperature at a fixed location, namely the middle of the wall of the evaporator, adiabatic, and condenser sections. Results were obtained for 5×12 and 10×24 nodal points (radial and axial, respectively) for outer wall temperature at the evaporator section of 60 °C and are presented in Table 1 for these two grid sizes. The 10×24 grid size takes about three times more computational time than that of the 5×12 grid size to reach steady state. Therefore, a 5×12 grid was used in this study. For this grid, the cell dimensions were 2×10^{-4} m in the radial direction for the pipe wall while 2×10^{-3} m in the axial direction.

Table 1 Grid Independence Study

Steady-state Temperature (°C)			
Heat Pipe Sections	**5x12**	**10x24**	**Change (%)**
Evaporator	77.4937	77.4498	0.057
Adiabatic	49.2191	49.2070	0.025
Condenser	20.0888	20.0901	-0.006

2.1.3 Experimental Study

The heat pipe used in this study was a copper pipe, sealed at both ends with copper end caps. Water was used as the working fluid. The heat pipe was 240 mm long, with a 8 mm outer diameter, and a 1 mm wall thickness. The details and properties of the heat pipe wall are given in Table 2 and assumed to be constant along the heat pipe wall. The experimental set-up is shown in Figure 2. The evaporator section was heat using a hot water jacket (24 mm outer diameter). The outer side of the hot water jacket at the evaporator section was covered with aeroflex insulation, and the heat pipe was placed vertically in a testing position. The adiabatic section was covered with Styrofoam to prevent heat loss to the surroundings. The condenser section was cooled using a cool water jacket (24 mm outer diameter). The water flow in the cool water jacket was controlled by a flow meter, and the outside of the cool water jacket was also insulated with aeroflex. Seven thermocouples type-K were attached on the outside wall of the heat pipe. All the thermocouples were calibrated against any variation in the room temperature. The inlet and outlet temperature of cool water were also measured carefully.

Table 2 Specifications and Properties of Heat Pipe Wall

Heat Pipe and End Cap Material	Copper
Pipe outer diameter (D_o)	8 x 10^{-3} *m*
Pipe inner diameter (D_i)	7 x 10^{-3} *m*
Pipe length (L)	0.24 *m*
Length of the evaporator section (L_e)	0.11 *m*
Length of the adiabatic section (L_o)	0.02 *m*
Length of the condenser section (L_c)	0.11 *m*
Thermal conductivity (k_w)	401 *W/m K*
Density (ρ_w)	8933 *kg/m*^{3}
Specific heat capacity ($C_{p,w}$)	385 J/ *kg K*

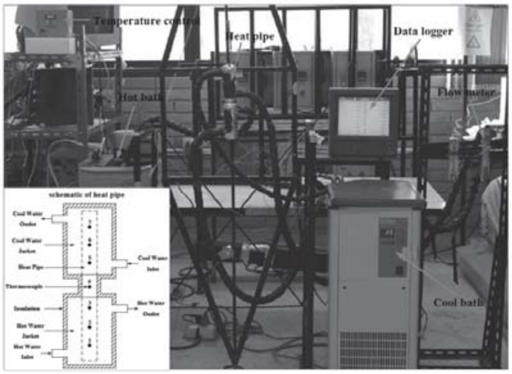

Figure 2 The Experimental Set-up

At time t = 0, the heater was turned on to input the temperature with water in hot water jacket and the water flowed to evaporator section. At the same time, the cooling water flowed to the condenser section. The outer wall temperatures and the water inlet and outlet temperatures were recorded at an interval of 2 sec. The cool water flow rate was carefully measured during each time interval. The heat recovery from the condenser was measured by means of the water inlet and outlet temperatures until the attainment of steady state. The input outer wall temperature of evaporator section of 60, 70, and 80 °C and temperature inputs at outer wall of condenser section of 20 °C were used for the experiments.

3. Results and Discussion

The results from the experimental, the numerical solutions of the two-dimensional model, and the experimental compared with the numerical solutions of the two-dimensional model are represented in Figures 3, 4, and 5 respectively for 60 °C of outer wall temperature inputs at the evaporator section, Figures 6, 7, and 8 for 70 °C of outer wall temperature inputs at the evaporator section, and Figures 9, 10, and 11 for 80 °C of outer wall temperature inputs at the evaporator section. The transient temperatures at the evaporator, adiabatic, and condenser sections from experimental and two-dimensional model are shown in the Figures 3 and 4 for the outer wall temperature inputs at the evaporator section of 60 °C. As can be seen from the figures, the transient temperature profile in the evaporator section rise during start-up from the initial temperature to the input outer wall temperature of the evaporator section, the time required for the steady state was about 780 seconds for the experimental and 800 seconds for the two-dimensional model. The transient temperature profile in the adiabatic section rose during start-up from the initial temperature to the operation temperature (40 °C), the time required for the steady state was equal to the evaporator section. In the two sections the temperatures quickly increased in the first time period, and then the temperatures were slowly increased for the experiment. In the middle time period the temperatures for the two-dimensional model were lower than the experimental model for the evaporator section while the adiabatic section was only slightly higher than the experimental model. In the final time period the temperature for the two-dimensional model was equal to the temperature input with the outer wall temperature, while for the experimental model, the temperature was slightly lower than the two-dimensional model. The transient temperature profile in the condenser section rose during start-up from the initial temperature to the input outer wall temperature. The time required for the steady state was equal to the evaporator and adiabatic sections, but the temperature in this section was slowly decreased because the outer wall temperature of the condenser section was lower than the initial temperature of condenser section.

The results from the experimental model were compared with the two-dimensional model and are represented in Figure 5 for 60 °C of outer wall temperature of the evaporator section. The transient temperature profiles at intermediate locations in the evaporator, adiabatic, and condenser sections are plotted as functions of time. The symbols correspond to experimentally measured temperatures at middle locations of each section, and the solid lines correspond to the two-dimensional model predicted response of the temperature with time. As can be seen from these figures, the comparisons between the results of the experimental and the two-dimensional model response curves are excellent for all the three sections. The difference in the evaporator temperatures at the end of run (1300 sec) for 60 °C of the outer wall temperature is only 3.43%, the difference is 3.21% for the temperature of the adiabatic section, while for the temperature of the condenser section the difference is only 2.05%.

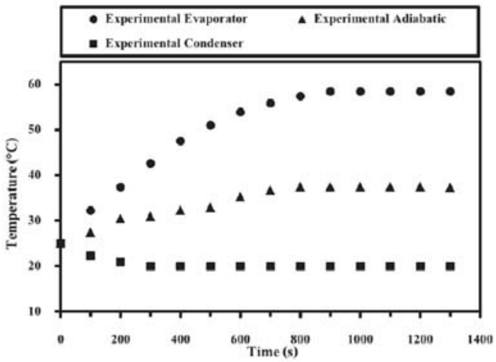

Figure 3 Temperature Profile of Heat Pipe Wall from Experimental for T_e = 60 °C

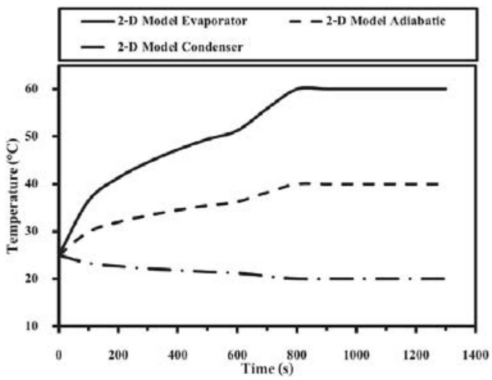

Figure 4 Temperature Profile of Heat Pipe Wall from 2-D Model for T_e = 60 °C

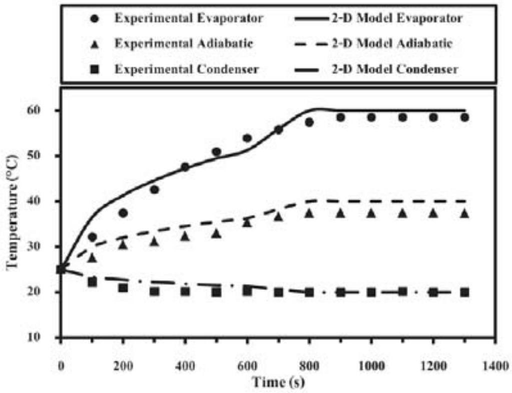

Figure 5 Comparison between Temperature Profile from Experimental and 2-D Model of Heat Pipe Wall for T_e = 60 °C

The transient temperatures at the evaporator, adiabatic, and condenser sections from experimental and two-dimensional model are shown in the Figures 6 and 7 for the outer wall temperature inputs at the evaporator section of 70 °C. As can be seen from the figures, the transient temperature profile in the evaporator section rose during start-up from the initial temperature to the input outer wall temperature of the evaporator section, the time required for the steady state is about 1264 seconds for the experimental and 1300 seconds for the two-dimensional model. The transient temperature profile in the adiabatic section rose during start-up from the initial temperature to the operation temperature (45 °C), the time required for the steady state was equal to the evaporator section. The transient temperature profile in the condenser section rose during start-up from the initial temperature to the input outer wall temperature, the time required for the steady state was equal to the evaporator and adiabatic sections, but the temperature in this section was slowly decreased because the outer wall temperature of the condenser section was lower than the initial temperature of the condenser section.

The results from the experimental model are compared with the two-dimensional model and are represented in Figure 8 for 70 °C of the outer wall temperature of the evaporator section. The transient temperature profiles at intermediate locations in the evaporator, adiabatic, and condenser sections are plotted as functions of time. The symbols correspond to experimentally measured temperatures at middle locations of each section, and the solid lines correspond to the two-dimensional model predicted response of the temperature with time. As can be seen from these figures, the comparisons between the results of the experimental and the two-dimensional model response curves are excellent for all the three sections. The difference in the evaporator temperatures at the end of run (1800 sec) for 70 °C of the outer wall temperature is only 0.10%, the difference is 1.66% for the temperature of the adiabatic section, while for the temperature of the condenser section the difference is only 0.61%.

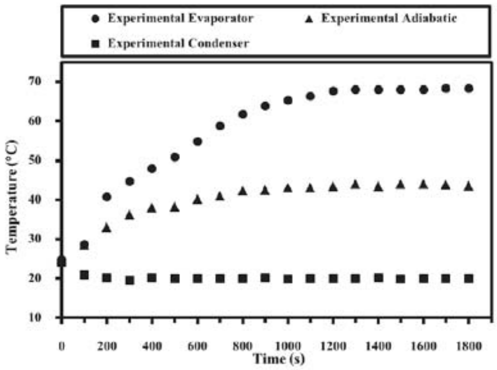

Figure 6 Temperature Profile of Heat Pipe Wall from Experimental for T_e = 70 °C

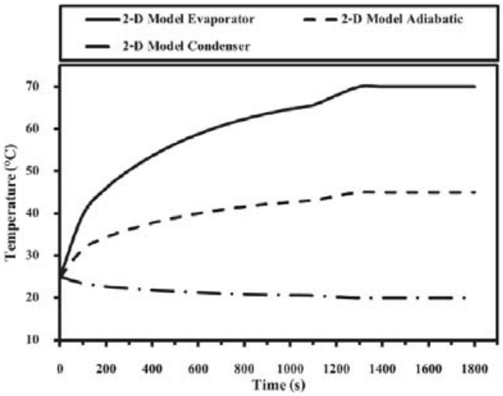

Figure 7 Temperature Profile of Heat Pipe Wall from 2-D Model for T_e = 70 °C

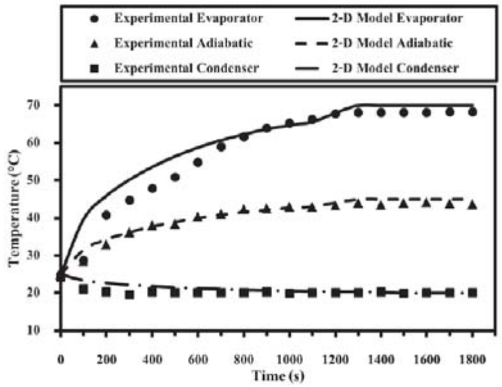

Figure 8 Comparison between Temperature Profile from Experimental and 2-D Model of Heat Pipe Wall for T_e = 70 °C

The transient temperatures at the evaporator, adiabatic, and condenser sections from experimental and two-dimensional model are shown in the Figures 9 and 10 for the outer wall temperature inputs at the evaporator section of 80 °C. As can be seen from the figures, the transient temperature profile in the evaporator section rose during start-up from the initial temperature to the input outer wall temperature of the evaporator section, the time required for the steady state was about 1622 seconds for the experimental and 1700 seconds for the two-dimensional model. The transient temperature profile in the adiabatic section rose during start-up from the initial temperature to the operation temperature (50 °C), the time required for the steady state was equal to the evaporator section. The transient temperature profile in the condenser section rose during start-up from the initial temperature to the input outer wall temperature. The time required for the steady state was equal to the evaporator and adiabatic sections, but the temperature in this section was slowly decreased because the outer wall temperature of condenser section was lower than the initial temperature of condenser section.

The results from the experimental model were compared with the two-dimensional model and are represented in Figure 11 for 80 °C of outer wall temperature of the evaporator section. The transient temperature profiles at intermediate locations in the evaporator, adiabatic, and condenser sections are plotted as functions of time. The symbols correspond to experimentally measured temperatures at middle locations of each section, and the solid lines correspond to the two-dimensional model predicted response of the temperature with time. As can be seen from these figures, the comparisons between the results of the experimental and the two-dimensional model response curves are excellent for all the three sections. The difference in the evaporator temperatures at the end of run (2300 sec) for 80 °C of the outer wall temperature is only 0.55%, the difference is 3.59% for the temperature of the adiabatic section, while for the temperature of the condenser section the difference is only 0.10%.

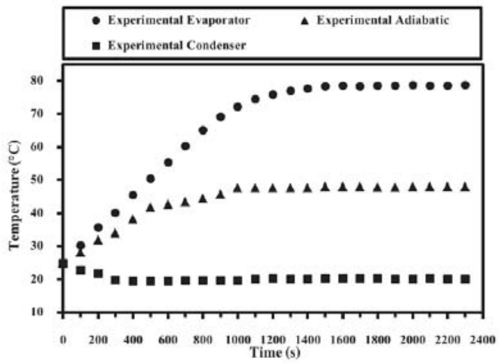

Figure 9 Temperature Profile of Heat Pipe Wall from Experimental for T_e = 80 °C

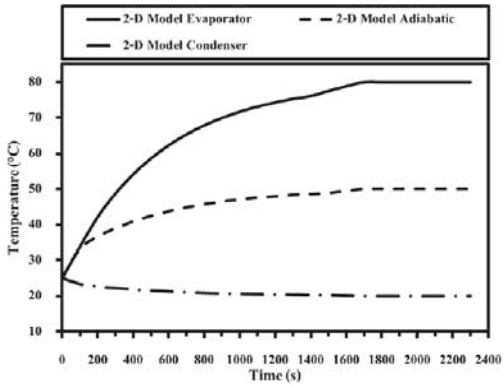

Figure 10 Temperature Profile of Heat Pipe Wall from 2-D Model for T_e = 80 °C

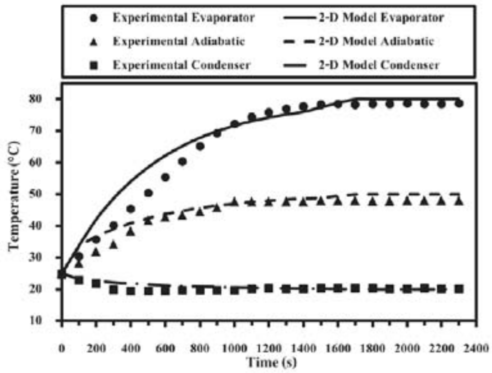

Figure 11 Comparison between Temperature Profile from Experimental and 2-D Model of Heat Pipe Wall for T_e = 80 °C

The time required to reach steady state is an important parameter for the start-up of a heat pipe. The time required to reach a steady state has been defined as the time required to steady state values. This has been calculated from the experimental results and was evaluated from the two-dimensional model as well. The results are presented in Table 3 for the different outer wall temperature inputs of evaporator section. The two values are in good agreement. The error between the experimental and the two-dimensional model shows a difference of less than 5%. The result from experimental model has the values lower than the two-dimensional model due to the fact that the experimental model had an error in the experimental set-up. When the outer wall temperature changed from 60 to 70 and 80 °C, the time required to reach steady state increased.

Table 3 Transient Response of the Heat Pipe

Temperature Input (°C)	Time Required to Attain Steady State (Sec) Experimental	Theoretical	Error (%)
60	780	800	-2.56
70	1264	1300	-2.84
80	1622	1700	-4.80

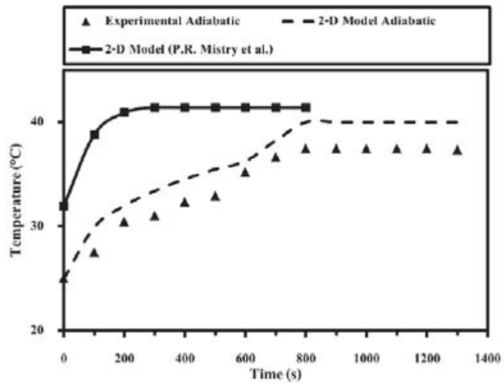

Figure 12 A Comparison of Temperature Profile in Adiabatic Section of this Study and Mistry et al., (2010)

The Figure 12 shows the experimental result and the numerical simulation in this study compared to the numerical simulation obtained by Mistry et al., (2010). It was found that the temperature profile in the heat pipe wall is in good agreement with the numerical simulation and the experimental model in this study. The time required to the steady state temperature of the experimental result and the numerical simulation in this study and the numerical simulation obtained by Mistry et al., (2010) are different due to the fact that the heat input at the evaporator section had different values.

4. Conclusion

Experiments were conducted to evaluate the transient response of a cylindrical heat pipe made of copper with water as a coolant. Constant temperature was supplied to the evaporator section, and the outer wall temperature at the condenser section was experimentally measured. The governing equations for a two-dimensional model were solved numerically using the concept of the temperature distribution, and the results were used to compare the time required to reach steady state.

The experimental model transient temperature profile for the evaporator, adiabatic, and condenser section was compared with the two-dimensional model temperature profiles with time required to reach steady state. The experimentally obtained temperature profiles, for both the transient and steady state, successfully compared with predictions from a two-dimensional model. Similar agreements were obtained in terms of the time required to reach the steady state (about 13-14 min for the outer wall temperature of 60 °C, 21-22 min for the outer wall temperature of 70 °C, and 28-29 min for the outer wall temperature of 80 °C). The experimental result and the numerical simulation in this study were compared to the numerical simulation obtained by Mistry et al., (2010) and were found to be in good agreement.

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Nomenclature

C_p	specific heat (*J/kg K*)
k	thermal conductivity (*W/m K*)
L	total length of heat pipe (*m*)
L_a	length of adiabatic section (*m*)
L_c	length of evaporator section (*m*)
L_e	length of condenser section (*m*)
r	radial coordinate (*m*)

R_o outer wall radius (m)

R_i inner wall radius (m)

R_v radius of vapor core (m)

t time (s)

T temperature (°C)

T_i initial temperature (°C)

y axial coordinate (m)

Greek Symbols

ρ density (kg/m^3)

Subscripts

w heat pipe wall

6. References

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