Journal of Applied Sciences

ISSN 1812-5654

Journal of Applied Sciences 9 (8): 1531-1537, 2009
ISSN 1812-5654

Modeling the Transient Response of the Thermosyphon Heat Pipes

[1]R. Parand, [1]B. Rashidian, [2]A. Ataei and [2]Kh. Shakiby
[1]Department of Energy Engineering, K.N. Toosi University of Technology, Tehran, Iran
[2]Department of Energy Engineering, Graduate School of the Environment and Energy, Science and Research Branch of Islamic Azad University, Tehran, Iran

Abstract: This study presents a theoretical investigation of the thermosyphon heat pipe behavior in transient regime. The advantage of the transient lumped model will be taken to simulate the response of the heat pipe. The transient thermal behavior of the heat pipe has been developed in order to obtain an analytic expression of the system response. A computer simulation program based on the lumped method was developed to estimate temperature of the heat pipe as well as the time needed to reach steady state condition. This program can be considered as a simple tool for modeling and designing heat pipe in transient regime. An influence analysis of the heat pipe response to various operating conditions has been shown. The results from this model were found to be in general agreement with the other numerical models.

Key words: Heat pipe, lumped model, MATLAB, modeling, response time

INTRODUCTION

The thermosyphon has been proved as a promising heat transfer device with very high thermal conductance. In practice, the effective thermal conductivity of thermosyphon exceeds that of copper 200-500 times. A two-phase closed thermosyphon is a high performance heat transfer device which is used to transfer a large amount of heat at a high rate with a small temperature difference (Noie, 2005).

Heat pipes are widely used for heat recovery and energy saving in various ranges of applications because of their simple structure, special flexibility, high efficiency, good compactness and excellent reversibility. The advantage of using heat pipe over other conventional methods is that large quantities of heat can be transported through small cross section area over a considerable distance with no additional power input to the system (Faghri, 1995).

The thermosyphon is a promising heat transfer device with very high thermal conductivity that employs the principle of evaporation and condensation of the working fluid, which is filled in a closed metal container. The thermosyphon has been proved to have a comparatively high heat transfer capability with a very small temperature difference between the heat source and heat sink. Moreover, it is simple and cheap, therefore more attractive to the thermal industry (Payakaruka *et al*., 2000).

A two-phase closed thermosyphon (TPCT) is passive high performance heat transfer device. It is a closed container filled with a small amount of a working fluid. In such a device, heat is supplied to the evaporator wall, which causes the liquid contained in the pool to evaporate. The generated vapor then moves upwards to the condenser. The heat transported is then rejected into the heat sink by a condensation process. The condensate forms a liquid film which flows downwards due to gravity (Farsi *et al*., 2003; Park *et al.,* 2002). Figure 1 shows a conventional heat pipe with three sections: (1) the evaporator section where heat is added to the system (Qe); (2) the condenser section where heat is removed from the system (Qc); (3) the transport section which connect the evaporator and the condenser, serving as a flow channel.

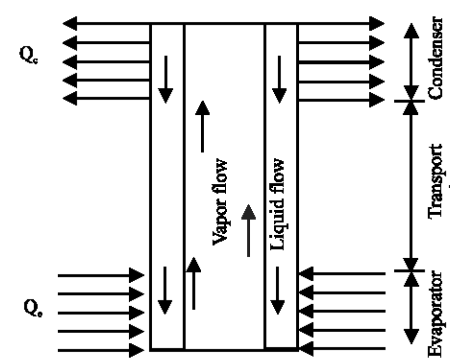

Fig. 1: A sketch of a heat pipe in operation

Corresponding Author: Dr. Abtin Ataei, Department of Energy Engineering, Graduate School of the Environment and Energy, Science and Research Branch of Islamic Azad University, P.O. Box 14515-775, Tehran, Iran
Tel: 0098-912-5980104 Fax: 0098-21-44465383

The working fluid flows inside a heat pipe can be divided into four components: (1) vapor flow in the transport section; (2) liquid flow in the transport section; (3) vapor flow in the evaporator and condenser and (4) liquid flow in the evaporator and condenser. The main advantage of TPCTs is that no mechanical pumping is needed. As a consequence, they are cheap and reliable. After a heat pipe reached a fully steady state condition, where the vapor flow is in the continuum state, it is often desirable to determine the temperature and the length of time needed to reach another steady state, for a given new operating conditions. The main investigations performed on TPCT behavior in transient regimes comprise several features. First, theoretical and numerical models describing the behavior of the whole system in response to a step of heat rate applied to the evaporator wall, have been developed by Harley and Faghri (1994), Reed and Tien (1987) and Khodabandeh (2004). It must be emphasized that these models have not been validated during the transient regime because of the lack of experimental data. A second feature deals with the difficulty of modeling unsteady pool boiling regimes occurring in the evaporator such as intermittent boiling, pulse boiling or geyser effect (Farsi *et al*., 2003; Niro and Beretta, 1990; Liu and Wang, 1992; Casarosa *et al*., 1983).

Reed and Tien (1987) and Farsi *et al*. (2003) presented a theoretical investigation of the TPCT response time with a model describing the behavior of the whole system in both transient and steady regimes. The researchers indicate that, for most TPCTs, the governing time scale was the film residence time. The researchers correlated the response time of the TPCT with the dimensionless parameters: Gr, Bo, Ja, $\rho v/\rho l$ and γ. Thus, the TPCT response time can be reduced in various ways: by lowering the thermal inertia of the working fluid, by acting on parameters which play a part in the expression of the residence film time: Gr, Bo, or by increasing the ratio $\rho v/\rho l$ which make vapor and liquid flow more intense.

In this study, an analysis of the heat pipe response time has been presented with the aim of providing design advice to minimize this quantity. The transient behavior has been investigated, when a sudden change in operating condition is applied to the heat pipe. A mathematical model based on the lumped model presented. Then, the dependence of the heat pipe response time to several parameters has been analyzed.

MATERIALS AND METHODS

Transient lumped model: After a heat pipe reached a fully steady-state condition, where the vapor flow is in the continuum state, it is often desirable to determine the length of time needed to reach another steady state for a given increase in the heat input. Although the 2-D numerical model is generally more comprehensive and accurate (Cao and Faghri, 1990), it usually requires considerable time and effort for computer coding. The lumped analytical model, on the other hand, provides a quick and convenient tool for the heat pipe designer. Faghri and Harley (1994) presented a transient lumped heat pipe model which determines the average temperature as a function of time. This formulation is derived from the general lumped capacitance analysis which is an application of an energy balance over a control volume, as shown in Fig. 2. The general lamped analysis results in the following energy equation for the case of both radiative and convective heat transfer from the condenser surface and an imposed heat input to the evaporator (Eq. 1).

Fig. 2: Control volume for lumped capacitance analysis

$$Q_e - (q_{conv} + q_{rad})S_c = C_t \frac{dT}{dt} \quad (1)$$

The total thermal capacity is defined as:

$$C_t = \rho V_t c_p \quad (2)$$

However, in heat pipe applications, the total thermal capacity is defined as the sum of the heat capacities of the solid and liquid components of the heat pipe. The heat capacity of the liquid saturated wick is modeled accounting for both the liquid in the wick and the wick structure itself (Eq. 3).

$$(\rho c_p)_{eff} = \varphi\, (\rho c_p) + (1-\varphi)(\rho c_p)_s \quad (3)$$

Typically, the thermal capacity of the vapor phase is neglected since the mass of vapor is small compared to that of the pipe wall or liquid working fluid. The heat rejection terms due to convection or radiation are found from the external heat transfer coefficients and temperature differences (Eq. 4 or 5).

$$Q = q_{conv} S_c = h S_c\, (T - T_{\infty,\, c}) \quad (4)$$

or

$$Q = q_{rad}S_c = \varepsilon\sigma S_c\,(T^4\text{-}T^4_{\infty,c}) \quad (5)$$

For a convective boundary condition, a closed-form solution can be obtained. However, for a radiative boundary, a simple first-order ordinary differential equation must be solved iteratively.

To simplify the formulation for a convective boundary condition, two external thermal resistances arc defined as:

$$R_c = \frac{1}{h_c S_c} \quad (6)$$

$$R_e = \frac{1}{h_e S_e} \quad (7)$$

In a traditional lumped capacitance analysis, is it very important to verify that the Biot number of the system is less than 0.1, where the Biot number is a measure of the temperature drop within the system as compared to the temperature drop between the system and the fluid. However, the effective thermal conductivity of a heat pipe is very high, which results in a nearly isothermal temperature profile. In most cases, the effective thermal conductivity of the heat pipe is several orders of magnitude larger than the heat transfer coefficient between the condenser and the environment and thus the Biot number criterion is satisfied (Dunn and Reay, 1994; Peterson, 1994). Several conditions are examined and closed-form solutions are derived for a convective boundary, which describe the transient lumped temperature of the heat pipe. For the case of pulsed heat input to the evaporator, constant condenser ambient temperature is as follows:

$$Q_e = \begin{cases} Q_{e1} & t \prec 0 \\ Q_{e2} = f(t) & t \geq 0 \end{cases} \quad (8)$$

and $T_{\infty,c} = T_{\infty,c1}$ = constant

To determine the initial conditions, the heat pipe is assumed to be at the steady state at (t<0), such that the general lumped capacitance equation simplifies to:

$$Q_{e1} = \frac{T - T_{\infty,c1}}{R_c} = 0 \quad (9)$$

Which gives the initial heat pipe temperature as followings:

$$T\,(0) = R_c Q_{e1} + T_{\infty,c1} \quad (10)$$

At t = 0, the heat input pulses to Qe = Qe2 and the general lumped capacitance equation is given by:

$$Q_{e2} - \frac{(T - T_{\infty,c1})}{R_c} = C_t \frac{dT}{dt} \quad (11)$$

This is a first order differential equation. The responses are shown in Fig. 3, 4 and the simulink (Karris, 2006) model is shown in Fig. 5.

The Eq. 1 can be solving for various boundary conditions, in example for the case of pulsed evaporator ambient temperature, constant condenser ambient temperature is as follows:

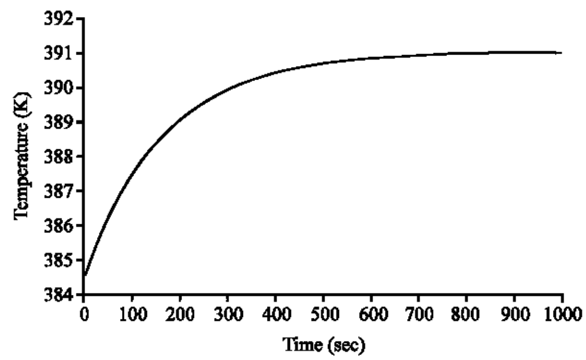

Fig. 3: Heat pipe response to pulsed heat input to the evaporator (Qe2 = const., Qe2> Qe1)

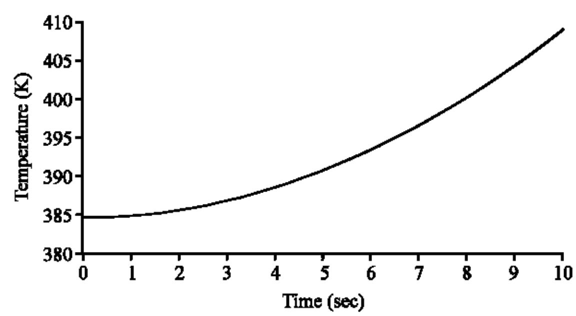

Fig. 4: Heat pipe response to Qe2 = Qe1(t+1)

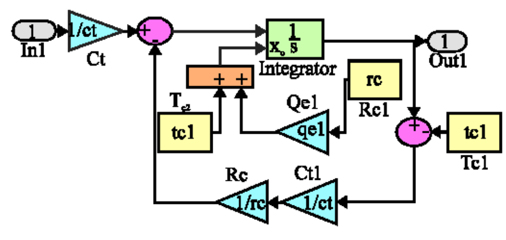

Fig. 5: Simulink model for lumped model. Pulsed heat input to the evaporator, constant condenser ambient temperature (Eq. 11)

$$T_{\infty,e} = \begin{cases} T_{\infty,e1} & t \prec 0 \\ T_{\infty,e2} = f(t) & t \geq 0 \end{cases} \quad (12)$$

and $T_{\infty,c} = T_{\infty,c1}$ = constant

In this case, the ambient temperature at the evaporator is a function of time and the transient heat input is found using the thermal resistance between the evaporator and the environment. The initial condition is:

$$T(0) = \frac{T_{\infty,e1}R_c + T_{\infty,c1}R_e}{R_c + R_e} \quad (13)$$

The energy equation is:

$$\frac{(T_{\infty,e2} - T)}{R_e} - \frac{(T - T_{\infty,c1})}{R_c} = C_t\frac{dT}{dt} \quad (14)$$

The exact solution for $T_{\infty,e2}$ = Constant is given by:

$$T(t) = \frac{T_{\infty,c1}R_e + T_{\infty,e1}R_c}{R_c + R_e} + \frac{\left(T_{\infty,e2} + T_{\infty,e1}\right)}{R_c + R_e}\left(1 - e^{\frac{-t}{\tau}}\right) \quad (15)$$

Where the time constant is given by:

$$\tau = \frac{C_t R_c R_e}{R_c + R_e} \quad (16)$$

The simulink model for this case is shown in Fig. 6 and response is shown in Fig. 7.

In the case of constant heat input to the evaporator, pulsed condenser ambient temperature is as follows:

$$T_{\infty,c} = \begin{cases} T_{\infty,c1} & t \prec 0 \\ T_{\infty,c2} = f(t) & t \geq 0 \end{cases} \quad (17)$$

and $Q_e = Q_{e1}$ = constant

The initial condition is:

$$T(0) = R_c Q_{e1} + T_{\infty,c1} \quad (18)$$

and the general lumped equation is:

$$Q_{e1} - \frac{(T - T_{\infty,c2})}{R_c} = C_t\frac{dT}{dt} \quad (19)$$

In the case of constant evaporator ambient temperature, pulsed condenser ambient temperature:

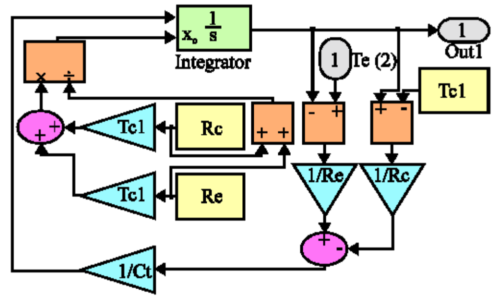

Fig. 6: Simulink model for lumped model. Pulsed evaporator ambient temperature, constant condenser ambient temperature (Eq. 14)

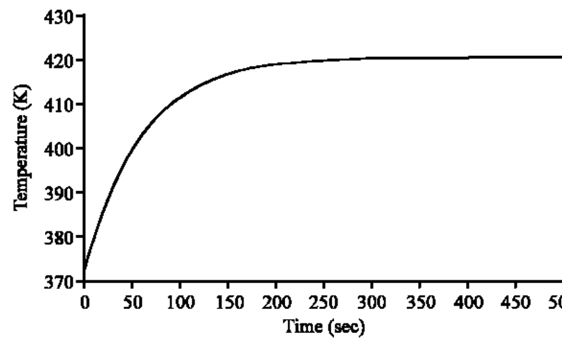

Fig. 7: Heat pipe response to pulsed evaporator ambient temperature (T_{e2} = constant, T_{e2}>T_{e1})

$$T_{\infty,c} = \begin{cases} T_{\infty,c1} & t \prec 0 \\ T_{\infty,c2} = f(t) & t \geq 0 \end{cases} \quad (20)$$

and $T_{\infty,e} = T_{\infty,e1}$ = constant

In this case, the condenser ambient temperature changes at t = 0, while the evaporator ambient temperature remains constant. The initial temperature is:

$$T(0) = \frac{(T_{\infty,e1}R_c + T_{\infty,c1}R_e)}{R_e + R_c} \quad (21)$$

The general lumped capacitance equation for t > 0 is:

$$\frac{T_{\infty,e1} - T}{R_e} - \frac{T - T_{\infty,c2}}{R_c} = C_t\frac{dT}{dt} \quad (22)$$

Figure 8, represents response of the system to pulsed condenser ambient temperature (Eq. 22).

For the case of radiative heat transfer at the condenser, with a constant ambient condenser temperature and pulsed heat input to the evaporator, the general lumped capacitance equation becomes:

$$\frac{dT}{dt} + \frac{\varepsilon\sigma S_c T^4}{C_t} = \frac{Q_{e2}}{C_t} + \frac{\varepsilon\sigma S_c T^4_{\infty,cl}}{C_t} \quad (23)$$

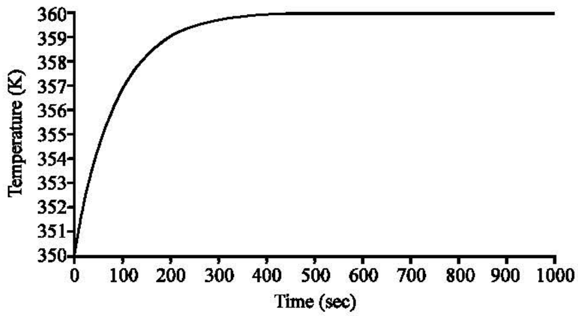

Fig. 8: Response to pulsed condenser ambient temperature

Equation 23 is a first-order, nonlinear, nonhomogeneous, ordinary differential with no closed form analytical solution. However, a numerical solution can be found using the Newton/Raphson secant technique.

Using the same procedure as in the convective cases above, the initial temperature can be found as:

$$T_0 = \left(\frac{Q_{e1}}{\varepsilon\sigma S_c} + T_{\infty,c1}^4\right)^{1/4} \qquad (24)$$

RESULTS AND DISCUSSION

Matlab programming: To analysis response of the heat pipe the program heat pipe response has been developed by Matlab. The input arguments are: physical properties of the heat pipe wall (Cp, thickness, length of the heat pipe, fill ratio, …), physical properties of the working fluid and properties of the wick.

Program estimate heat pipe response for these cases:

- Constant condenser ambient temperature, changing of the heat input to the evaporator
- Constant heat input to the evaporator, changing of the condenser ambient temperature
- Constant condenser ambient temperature, changing of the evaporator ambient temperature
- Constant evaporator ambient temperature, changing of the condenser ambient temperature
- Constant ambient condenser temperature, with radiative heat transfer at the condenser, changing of the heat input to the evaporator

The change of the operating condition are pulse, linear and exponential, in the case of pulsed input, program calculate time to reach the new steady state condition and the temperature of the new steady state condition. Some shots of the graphical user interface of the program are shown in Fig. 9-12.

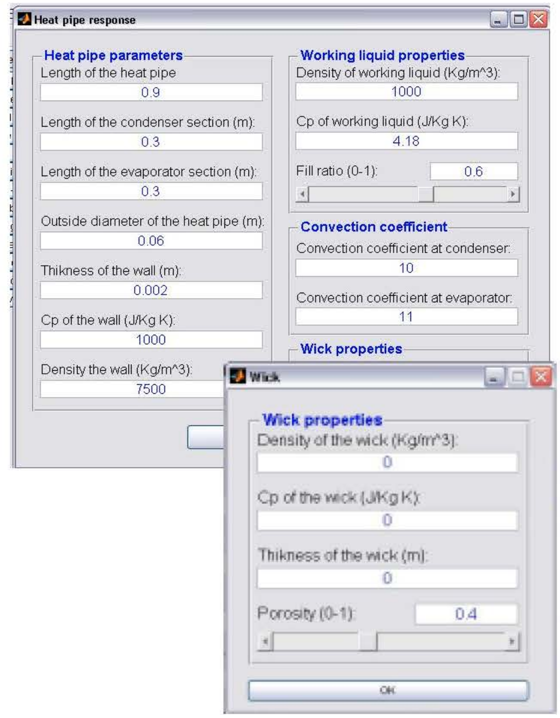

Fig. 9: First page of the heat pipe response program

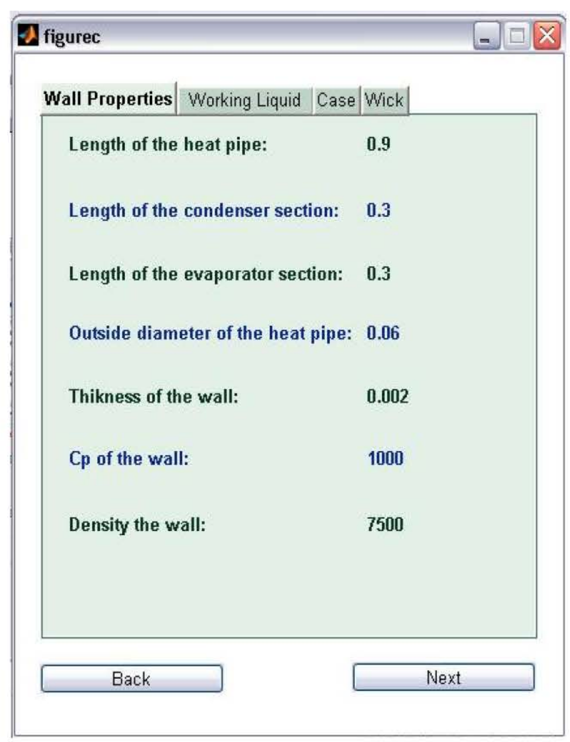

Fig. 10: Case selecting and assigning input type

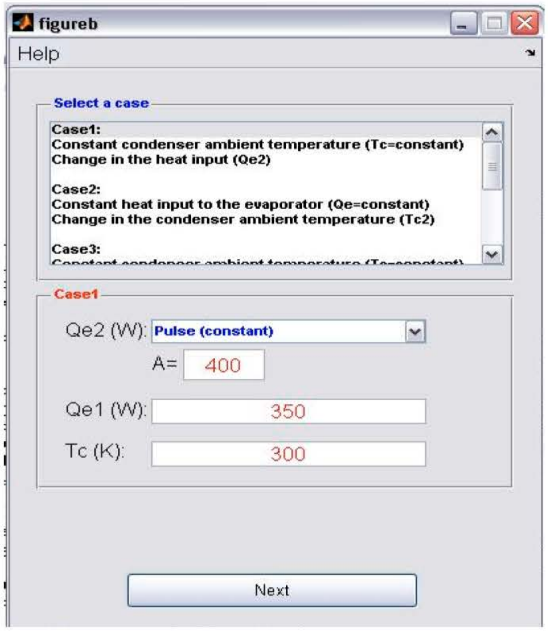

Fig. 11: Verifying data

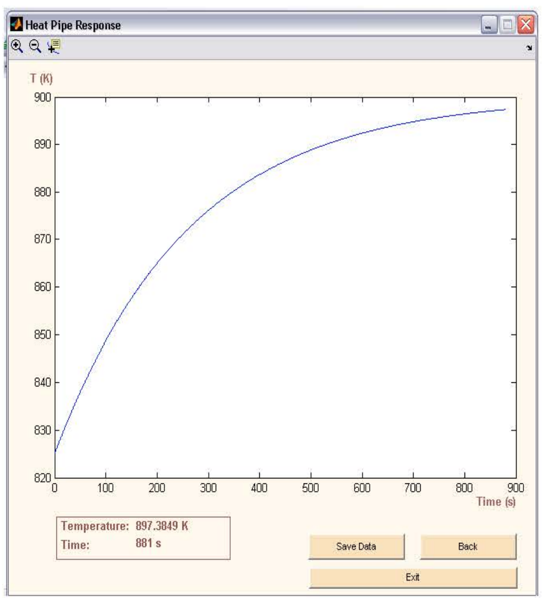

Fig. 12: Results of pipe response

Figure 13 represents validation of the lumped capacitance analysis. The results of the lumped model with a pulsed heat input and a constant condenser ambient temperature (Eq. 11) were compared to the numerical analysis for a sodium heat pipe with a stainless steel wall having a total heat capacity of $C_t = 175.4$ J k^{-1}.

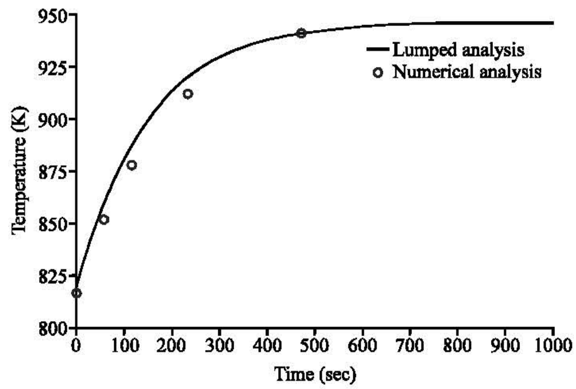

Fig. 13: Transient temperature profile for the lumped capacitance model as compared with the numerical analysis

CONCLUSION

Heat pipes are widely used for heat recovery and energy saving in various ranges of applications because of their simple structure, special flexibility, high efficiency, good compactness and excellent reversibility. The advantage of using heat pipe over other conventional methods is that large quantities of heat can be transported through small cross section area over a considerable distance with no additional power input to the system.

In this study, a mathematical model has been constructed for thermosyphon heat pipe, in order to describe thermal behavior in transient regime. An influence analysis of the heat pipe response to various operating conditions has been shown.

A computer simulation program based on the lumped method has been developed to estimate temperature of the heat pipe as well as the time needed to reach steady state condition. This program can be use for heat pipes with capillary media as well as wickless heat pipes and also, can be considered as a simple tool for modeling and designing heat pipe in transient regime. The results from this model were found to be in general agreement with the other numerical models.

NOMENCLATURE

Ct : Total thermal capacity of the system (J k^{-1})
c_p : Heat capacity (J kg^{-1} K^{-1})
h_c : External heat transfer coefficient at the condenser (W m^{-2} K^{-1})
he : External heat transfer coefficient at the evaporator (W m^{-2} K^{-1})
Q_e : Heat input (W)
q_{conv} : Output heat flux by convection (W)

q_{rad} : Radiation heat flux (W)
r_i : Inner diameter (m)
S_c : Surface area around the condenser section (m^2)
S_e : Surface area around the evaporator section (m^2)
T_{sat} : Saturation temperature (K)
T_{sur} : Surface temperature (K)
$T_{\infty,c}$: Ambient temperature surrounding the condenser (K)
$T_{\infty,h}$: Ambient temperature surrounding the evaporator (K)
V_t : Total volume (m^3)

Greek letters
ϵ : Radiative emissivity
ρ : Density (kg m^{-3})
σ : Stefan-Boltzman constant
ϕ : Porosity of the wick
γ : Filling ratio of the heat pipe (%)
ρ_l : Liquid density (kg m^{-3})
ρ_v : Vapor density (kg m^{-3})
τ : Time constant (s)
υ : Kinematic viscosity
β : Volumetric thermal expansion coefficient (equal to approximately 1/T for ideal fluid) (1/K)
Γ : Surface tension (N m^{-1})

Subscripts
c : Condenser
e : Evaporator
in : Inlet location
l : Liquid
sat : Saturation
v : Vapor
sur : Surface

Dimensionless numbers

Bond : $Bo = \frac{r_i}{\sqrt{\frac{\Gamma}{g(\rho_l - \rho_v)}}}$

Grashof : $Gr = \frac{g\beta r_i (T_{sur} - T_{sat})}{v^2}$

Jacob : $Ja = \frac{\rho_l Cp_l (T_{sur} - T_{sat})}{\rho_v L_v}$

REFERENCES

Cao, Y. and A. Faghri, 1990. Transient two-dimensional compress-ible analysis for high-temperature heat pipes with pulsed heat input. Numerical Heat Transfer Part A, 18: 483-502.

Casarosa, C., E. Latrofa and A. Shelginski, 1983. The geyser effect in a two-phase thermosyphon. Int. J. Heat Mass Transfer, 26: 933-941.

Dunn, P.D. and D.A. Reay, 1994. Heat Pipes. 4th Edn., Pergamon Press, New York, ISBN: 0080419038.

Faghri, A. and C. Harley, 1994. Transient lumped heat pipe analysis. Heat Recovery Syst. CHP, 14: 351-363.

Faghri, A., 1995. Heat Pipe Science and Technology. Taylor and Francis, USA., ISBN: 1560323833.

Farsi, H., J.L. Joly, M. Miscevic, V. Platel and N. Mazet, 2003. An experimental and theoretical investigation of the transient behavior of a two-phase closed thermosyphon. Applied Thermal Eng., 23: 1895-1912.

Harley, C. and A. Faghri, 1994. Complete transient two-dimensional analysis of two-phase closed thermosyphons including the falling condensate film. ASME J. Heat Transfer, 116: 418-426.

Karris, T.S., 2006. Introduction to Simulink with Engineering Applications. Orchard Publication, ISBN: 0974423971.

Khodabandeh, R., 2004. Thermal performance of a closed advanced two-phase thermosyphon loop for cooling of radio base stations at different operating conditions. Applied Thermal Eng., 24: 2643-2655.

Liu, J.F. and J.C.Y. Wang, 1992. On the pulse boiling frequency in thermosyphons. ASME J. Heat Transfer, 114: 290-292.

Niro, A. and G.P. Beretta, 1990. Boiling regimes in a closed two-phase thermosyphon. Int. J. Heat Mass Transfer, 33: 2099-2110.

Noie, S.H., 2005. Heat transfer characteristics of a two-phase closed thermosyphon. Applied Thermal Eng., 25: 495-506.

Park, Y.J., H.K. Kang and C.J. Kim, 2002. Heat transfer characteristics of a two-phase closed thermosyphon to the fill charge ratio. Int. J. Heat Mass Transfer, 45: 4655-4661.

Payakaruka, T., P. Terdtoonb and S. Ritthidechb, 2000. Correlations to predict heat transfer characteristics of an inclined closed two-phase thermosyphon at normal operating conditions. Applied Thermal Eng., 20: 781-790.

Peterson, G.P., 1994. An Introduction to Heat Pipes: Modeling, Testing and Applications. Wiley-Interscience, ISBN: 047130512X.

Reed, J.G. and C.L. Tien, 1987. Modeling of the two-phase closed thermosyphon. ASME J. Heat Transfer, 109: 722-730.