EFFECT OF VARIATION IN LENGTH OF THE CONVENTIONAL HEAT PIPE ON THE THERMAL PERFORMANCE

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ABSTRACT

One-dimensional numerical simulations were conducted for the performance of the conventional heat pipe(CHP). In the current investigation, the operating limitations of an CHP were theoretically studied in order to determine its maximum heat transport capability. The thermal network approach was used to calculate the performance of the CHP, based on the methodology to produce results of reasonable accuracy. In the present study, the effect of variation in length of the CHP on the performance was analyzed using the corresponding thermal networks.

INTRODUCTION

Heat pipes are passive devices with a minimal driving temperature difference through a small cross sectional area. These devices are operated by utilizing the latent heat of an internal working fluid to transfer large amounts of heat. Heat pipe applications range from the cooling consumer electronics such as laptops to thermal management of spacecraft [1-3].

For several decades a number of researchers have investigated the effects of various parameters on the performance of many heat pipe configurations as the numerical methodology was developed. Nemec et al. [4] studied the heat transport limitation standard-used construction wick heat pipes with various working fluids operating at wide temperature ranges. They reported that the limitation values depend on heat pipe parameters, wick structure parameters and the themophysical properties of the working fluid. D.Yin et al. [5] theoretically studied the operating limitation of the oscillating heat pipe based on the mass-spring system. They found that the operating limitation of an OHP depends on the working fluid, turn number, operating temperature, and filling ratio.

Therefore, there are various parameters that put limitations and constraints on the steady and transient operation of heat pipes and consequently the rate of heat transport through a heat pipe is subject to a number of operating limits. The consequents of boiling in a heat pipe depend on numerous design and performance factors, and are difficult to predict, and hence the complicated mathematical expressions and numerical schemes in previous studies have been developed. However, it is usually not desirable or necessary to get into detail effects for most practical applications. Therefore, a simple way, which can calculate the performance of heat pipe, is essential.

NOMENCLATURE

Symbols		
A	[m^2]	Surface area
C	[-]	Correlation coefficient
d	[m]	Diameter
f_v	[-]	Correlation coefficient for friction
g	[m/s^2]	Gravity acceleration
h	[W/m^{2}K]	heat transfer coefficient
h_{fg}	[kJ/kg]	
k_{eff}	[W/mK]	Effective thermal conductivity of the wick
k	[W/mK]	thermal conductivity
L	[m]	length
L_{eff}	[m]	Effective length of the pipe
P_v	[Pa]	Vapor pressure
$\Delta P_{c.m}$	[Pa]	Maximum capillary pressure differenc
Q	[W]	Heat transfer rate
Q_b	[W]	Boiling limit
Q_c	[W]	Capillary limit
Q_e	[W]	Entrainment limit
Q_s	[W]	Sonic limit
Q_v	[W]	Viscous limit
r_{ce}	[m]	Effective capillary radius of the wick
r_{hw}	[m]	Hydraulic radius of the wick
r	[m]	HP radius
r_n	[m]	Critical nucleation site radius
R	[K/W]	Thermal resistance
R_g	[J/kgK]	Specific gas constant
Re_v	[-]	Reynolds number for vapor
t	[m]	thickness
T_v	[K]	Vapor temperature
Greek symbols		
α	[-]	Accommodation coefficient
σ	[N/m]	Surface tension of the fluid
μ_l	[Pa-s]	Viscosity of the liquid
μ_v	[Pa-s]	Viscosity of the vapour
ρ_l	[kg/m^3]	Density of the liquid
ρ_v	[kg/m^3]	Density of the vapor
γ_v	[-]	Ratio of specific heat
Ψ	^{o}	Tilt angle
Sub/superscripts		
a		Adiabatic secion
c		Condenser section
$c.i$		Condenser liquid-vapor interface
$c.in$		Inner surface of the liquid film in the condenser
e		Evaporator section length
$e.i$		Evaporator liquid-vapor interface
$e.in$		Inner surface of the liquid film in the evaporator
hp		Heat pipe
i		Inner

inter	Interfacial
o	Outer
tot	Total
v	Vapor
w	wall
wk	Wick

ZUO and Faghri [6] suggested a unique view into the physics behind the heat pipe operation, which was considered a thermal network of various components. They described transient heat pipe behaviour using first order, linear, ordinary differential equation. Tardy and Sami [7] developed a mathematical model to predict the thermal behaviour of heat pipes with thermal storage during a cooling cycle. They reported that the developed numerical model is in excellent agreement with the experimental data. Sharifi et al. [8] studied externally-finned heat pipes with thermal contact resistances. They used a detailed HP model to quantify the influence of the contact resistances on the heat transfer and fluid flow within the HP, as well as on the overall heat transfer rate.

In the present study, therefore, authors consider the conventional heat pipe configuration having various heat transfer limitations, and evaluate the performance of a heat pipe model using the thermal resistance networking. Detailed heat transfer limitations were calculated using the analytical expression for predicting boiling, capillary, entrainment, sonic, and viscous heat transport limits in the convectional heat pipe. In addition, the performance of the heat pipe was analysed using the representative correlations and expressions about the thermal resistance of a single model suggested by Shabgard et al. [1], and consequently the effect of variation in length of convectional heat pipe was addressed.

NUMERICAL METHODS

In the design of heat pipes, it is important to establish that heat transport will take place at the required rate without encountering limits on heat transport capability. Therefore, the heat pipes are subject to a number of heat transfer limitation. These limitations determine the maximum heat transfer rate a particular heat pipe can achieve under certain working conditions. Mathematical methods for the prediction of heat transport limits are presented as follows:

- Boiling limit Q_b

$$Q_b = \left(\frac{2\pi L_e K_{eff} T_v}{h_{fg}\rho_v \ln(r_i / r_v)}\right)\left(\frac{2\sigma}{r_n} - \Delta P_{c.m}\right) \tag{1}$$

- Capillary limit Q_c

$$Q_c = \frac{\frac{2\sigma}{r_{ce}} - \rho_l g(d_v \cos\Psi + L_{eff} \sin\Psi)}{\left(\frac{Cf_v \operatorname{Re}_v \mu_v}{2r_{hw}{}^2 A_v \rho_v h_{fg}} + \frac{\mu l}{KA_w h_{fg}\rho_l}\right)L_{eff}} \tag{2}$$

- Entrainment limit Q_e

$$Q_e = A_v h_{fg}\left(\frac{\sigma\rho_v}{2r_{hw}}\right)^{1/2} \tag{3}$$

- Sonic limit Q_s

$$Qs = A_v \rho_v h_{fg}\left(\frac{\gamma_v R_v T_v}{2(\gamma_v + 1)}\right)^{1/2} \tag{4}$$

- Viscous limit Q_v

$$Q_v = \frac{\pi r_v{}^4 h_{fg}\rho_v P_v}{12\mu_v L_{eff}} \tag{5}$$

The heat transfer limitation can be any of the above limitations depending on the size and shape of the pipe, working fluid, wick structure, and operating temperature. The lowest limit among the constraints defines the maximum heat transport limitation of a heat pipe at a given temperature. Further details on the operating limitation of the conventional heat pipe are described in Tari [9].

The thermal network approach is a robust engineering tool that is easy to implement and program, is user friendly, straightforward, computationally efficient, and serves as a baseline methodology to produce results of reasonable accuracy. Therefore, in the present study, the thermal resistance network was applied to calculate the heat transfer performance of heat pipe, by adopting the similar method used by Shabgard et al. [1].

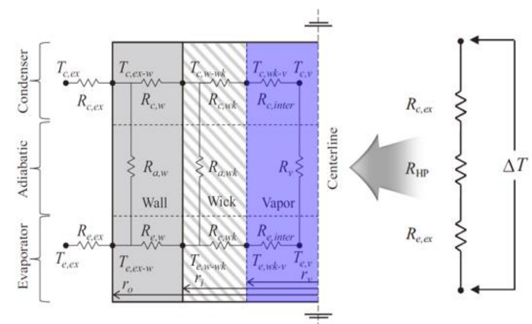

Figure 1 Thermal resistances of a heat pipe used to calculate R_{hp} and total thermal resistance to calculate R_{tot} used by Shabgard et al. [1].

Figure 1 shows the major thermal resistances of a cylindrical heat pipe at steady state. Here, $R_{e.w}$, $R_{c.w}$, R_v, $R_{e.wk}$, $R_{c.wk}$, $R_{e.inter}$, $R_{c.inter}$, $R_{a.w}$, $R_{a.wk}$, $R_{e.ex}$ and $R_{c.ex}$ denote the resistance associated with the evaporator wall, the condenser wall, the vapour pressure drop along the HP length, the evaporator wick, the condenser wick, the interfacial evaporation, the interfacial condensation, the adiabatic section wall, the adiabatic section wick, the external thermal resistance at evaporator and condenser, respectively. The individual resistances are as follows:

- Radial conduction through the evaporator section wall, $R_{e.w}$

$$R_{e.w} = \frac{\ln(\frac{r_o}{r_i})}{2\pi k_w L_e} \tag{6}$$

- Radial conduction through the condenser section wall, $R_{c.w}$

$$R_{c.w} = \frac{\ln(\frac{r_o}{r_i})}{2\pi k_w L_c} \tag{7}$$

- Thermal resistance due to vapour pressure drop, R_v

$$R_v = \frac{8R_g \mu_v T_v^2}{\pi h_{fg}^2 P_v \rho_v}\left[\frac{(L_e + L_c)/2 + L_a}{r_i^4}\right] \quad (8)$$

- Radial conduction through the evaporator wick, $R_{e.wk}$

$$R_{e.wk} = \frac{\ln(\frac{r_i}{r_v})}{2\pi k_{eff} L_e} \quad (9)$$

- Radial conduction through the condenser wick, $R_{c.wk}$

$$R_{e.wk} = \frac{\ln(\frac{r_i}{r_v})}{2\pi k_{eff} L_c} \quad (10)$$

- Evaporation at the evaporator liquid-vapor interface, $R_{e.inter}$

$$R_{e.inter} = \frac{1}{h_{e.inter} A_{e.i}} \quad (11)$$

- Condensation at the condenser liquid-vapor interface, $R_{c.inter}$

$$R_{c.inter} = \frac{1}{h_{c.inter} A_{c.i}} \quad (12)$$

where $h_{e.inter}$ and $h_{c.inter}$ are the interfacial evaporation and condensation heat transfer coefficient, and are determined as follows.

$$h_{e.inter} = \left(\frac{2\alpha}{2-\alpha}\right)\left(\frac{h_{fg}^2 \rho_v}{T_v}\right)\sqrt{\frac{1}{2\pi R_g T_v}}\left(1 - \frac{P_v}{2h_{fg}\rho_v}\right) \quad (13)$$

Hear, α is the accommodation coefficient ($0 < \alpha \le 1$) and depends on the nature of the surface, as well as on gas conditions such as composition, pressure, and other factors. Further details on the accommodation coefficient are described in Hall and Doster [10] and Goodman and Wachman [11].

- Axial conduction through the adiabatic section wall, $R_{a.w}$

$$R_{a.w} = \frac{L_a}{\pi(r_o^2 - r_i^2)k_w} \quad (14)$$

- Axial heat conduction through the adiabatic section of the HP wick, $R_{a.wk}$

$$R_{a.wk} = \frac{L_a}{\pi(r_i^2 - r_v^2)k_{eff}} \quad (15)$$

- Convection resistance between the external flow and the evaporator section, $R_{e.ex}$

$$R_{e.ex} = \frac{1}{h_e A_{e.o}} \quad (16)$$

- Convection resistance between the external flow and the condenser section, $R_{c.ex}$

$$R_{c.ex} = \frac{1}{h_c A_{c.o}} \quad (17)$$

The internal thermal resistance of a single heat pipe can be calculated using the following equation, due to the parallel connection among the components of thermal resistance.

$$R_{hp} = \frac{\left[R_{e.w} + R_{c.w} + \frac{(R_{e.wk} + R_{e.inter} + R_v + R_{c.inter} + R_{c.wk})(R_{a.wk})}{(R_{e.wk} + R_{e.inter} + R_v + R_{c.inter} + R_{c.wk}) + (R_{a.wk})}\right](R_{a.w})}{\left[R_{e.w} + R_{c.w} + \frac{(R_{e.wk} + R_{e.inter} + R_v + R_{c.inter} + R_{c.wk})(R_{a.wk})}{(R_{e.wk} + R_{e.inter} + R_v + R_{c.inter} + R_{c.wk}) + (R_{a.wk})}\right] + (R_{a.w})} \quad (18)$$

Once the thermal resistances of a single heat pipe are determined, the total thermal resistance may be calculated from the following equation:

$$R_{tot} = R_{e.ex} + R_{hp} + R_{c.ex} \quad (19)$$

The total thermal resistance, R_{tot}, is as the ratio of the driving potential (temperature) to the corresponding heat transfer as follows:

$$R_{tot} = \frac{\Delta T}{Q}, \quad Q = \frac{\Delta T}{R_{tot}} \quad (20)$$

COMPUTATIONAL CONDITIONS

Figure 2 shows the computational domain of the convectional heat pipe configuration of the system considered in the present study. The interior surface of the closed container is lined with a thin layer of porous material, usually referred to as a wick. The pores of the wick are filled with a liquid appropriate to the application, and the vapour of the liquid occupies the remaining internal volume inside the container having a cylindrical shape.

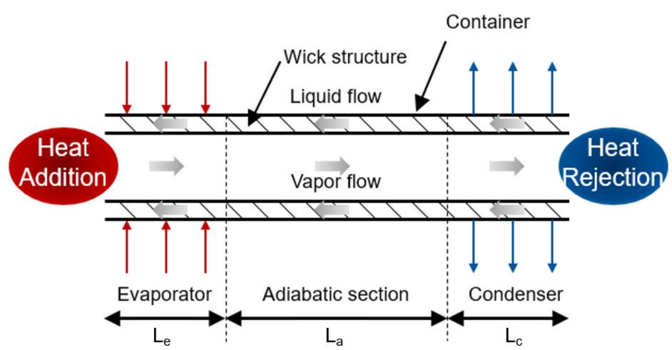

Figure 2 Schematic of conventional heat pipe

Heat applied to the evaporator section by an external source is conducted through the pipe wall and wick structure, where it vaporizes the working fluid. The resulting vapour pressure drives the vapour from the evaporator to the condenser through the adiabatic section, and consequently the vapour condenses, releasing its latent heat of vaporization to the provided heat sink. In addition, the condensed fluid moves from the condenser to the evaporator through the wick structure due to capillary pressure. This process will continue as long as there is a sufficient capillary pressure to drive the condensate back to the evaporator.

Table 2 Heat pipe configuration

Heat pipe configuration		Working fluid	Water
d [m]	0.02	Container material	Copper
L [m]	0.75	Orientation	Horizontal
L_e [m]	0.25 (0.05~0.45)	Wick structure	Wire screen
L_c [m]	0.25 (0.05~0.45)	N	100
L_a [m]	0.25 (0.05~0.45)	Wire diameter [m]	1.143X10^{-4}
t [m]	0.0025	Number of layers	3

In the present study, the heat pipe is considered using the configuration of heat pipe suggested by Tari [9]. The diameter, the total length and the thickness of heat pipe are 0.02[m], 0.75[m], and 0.0025[m] respectively. The working fluid and the container material are water and copper. The angle of heat pipe is fixed to 0^O, meaning the horizontal of orientation. The wick structure is considered to wire screen shape having the mesh number of 100, the wire diameter of $1.143X10^{-4}$, and the layer number of 3. The length of the heat pipe is divided into three parts having a length of 0.25[m], respectively: evaporator, adiabatic section, and condenser. Thus, authors focus on the variation in performance of convectional heat pipe at various length of three parts on the range of 0.05<L<0.45.

VALIDATION TEST

The preliminary numerical simulations on the conventional heat pipe considered by Sharifi et al. [8] were conducted to validate the present numerical simulation code. The results about the thermal resistance in each component of a heat pipe obtained by the present code were compared to those obtained by Sharifi et al. [8].

Table 1 shows the thermal resistances in each component of a heat pipe with channel flow cooling. Sodium-Stainless steel are used as working fluid and container material, respectively. Heat transfer coefficient and temperature at evaporator and condenser were given in Sharifi et al. [8]. As a result, the thermal resistances in each component of a heat pipe is in excellent agreement with the empirical/analytical results reported in the previous studies, except for R_v having the order of 10^{-4} due to an error induced by the difference in thermal properties. However, as shown in Table 1, the thermal resistance of R_{hp} and R_{tot} in the present study match well with those of previous study. Therefore, according to the validation test, the analysis of the thermal resistance networking about a single heat pipe can be conducted with reasonable accuracy.

Table 1 Comparison of thermal resistance obtained from the present study with those of previous empirical/analytical and numerical results.

Thermal resistance [K/W]	Sharifi et al. [8] Empirical/ Analytical	Sharifi et al. [8] Numerical	Present study	Error [%]
$R_{e.w}$	5.69e-03	5.47e-03	5.69e-03	0.03
$R_{e.wk}$	3.14e-03	3.02e-03	3.14e-03	0.09
$R_{e.inter}$	4.33e-04	1.69e-04	4.36e-04	0.70
R_v	8.37e-04	6.23e-04	5.17e-04	38.28
$R_{c.w}$	1.63e-03	1.61e-03	1.63e-04	0.24
$R_{c.wk}$	8.96e-04	8.61e-04	8.96e-04	0.04
$R_{c.inter}$	1.24e-04	2.29e-04	1.25e-04	0.47
$R_{e.ex}$	1.87e-01	1.88e-01	1.87e-01	0.13
$R_{c.ex}$	1.34	1.34	1.34	0.19
R_{hp}	1.27e-02	1.20e-02	1.24e-02	2.52
R_{tot}	1.54	1.54	1.54	0.17

RESULTS AND DISCUSSION

Figure 3 shows the results of operating limitations in heat pipe configuration. The temperature range is considered from 273.16[K] to 373.16[K], because the useful operating range is generally smaller than the whole temperature range between melting point and the boiling point.

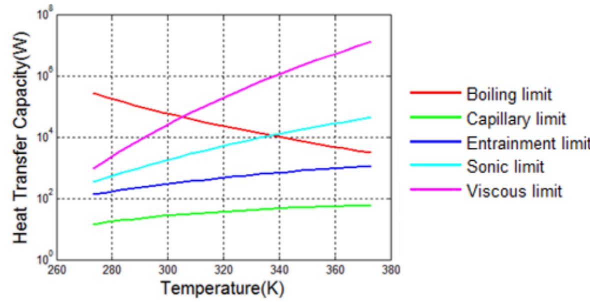

Figure 3 Operating limitations of heat pipe configuration.

As shown in Figure 3, in the present study, the capillary limit is always lower that the values of the other heat transfer limits. Therefore, the dominant limit of the heat pipe, which determines its maximum heat transport capability, is the capillary limit. In addition, according to the fundamental equations of heat transfer limits, the capillary, the entrainment, the sonic, and the viscous limit increases with increasing useful operating range of temperature, while the boiling limit decreases with increasing temperature. Therefore, the heat transfer performance of heat pipe should be smaller than the capillary limit having the smallest value among the operating limitations.

Table 2 The performance of heat pipe from the thermal resistance networking analysis

Thermal resistance of heat pipe [K/W]					
$R_{e.ex}$	0.063662	$R_{a.w}$	2.19903	$R_{c.ex}$	0.795775
$R_{e.w}$	0.00022	$R_{a.wk}$	177.352	$R_{c.w}$	0.00022
$R_{e.wk}$	4.84063	R_v	5.39e-08	$R_{c.wk}$	4.83401
$R_{e.inter}$	8.87e-06			$R_{c.inter}$	8.38e-06
R_{hp}	1.77408	R_{tot}	2.63352	Q[W]	37.972

Table 2 shows the performance of heat pipe from the thermal resistance networking analysis. The vapor temperature was calculated by the heat balance between the evaporator and the condenser. The maximum value of the thermal resistance in the component of heat pipe is 177.352 of $R_{a.wk}$. The values of R_v, $R_{e.inter}$, and $R_{c.inter}$ are very smaller than $R_{a.wk}$, and hence these values are almost insignificant on R_{hp} and R_{tot}. The value of R_{hp} is calculated to 1.77408 by the parallel connection. The performance of heat pipe is obtained to 37.972[W] by the value of R_{tot}. Therefore, this value is smaller than the capillary limit of vapor temperature, and consequently the operating limitations of heat pipe are satisfied.

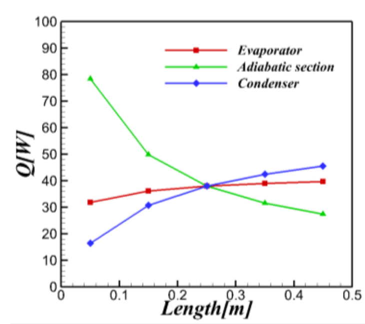

Figure 4 Comparison in the performance about length variation of heat pipe

Figure 4 shows the comparison in the performance about the length variation of heat pipe three parts (evaporator, adiabatic section, condenser) calculated by the thermal resistance networking. As shown in Figure 4, the performance of heat pipe increases with increasing length of evaporator and condenser, because the resistances in components associated with evaporator and condenser decreases. However, it decreases with increasing length of adiabatic section due to the increase of R_v, and consequently the performance of heat pipe shows the biggest change. The parameters in order of the strength of their effects are the condenser length, the evaporator length, and the adiabatic length, and hence these parameters must be considered as design variables for heat pipe configuration optimization.

CONCLUSION

One-dimensional numerical simulation were conducted for the performance of the conventional heat pipe(CHP) in this study. The operating limitations of an CHP were calculated in order to determine its maximum heat transport capability. In addition, the effect of variation in length of the CHP on the performance was analyzed using the corresponding thermal networks.

In the present study, the dominant limit of the heat pipe, which determines its maximum heat transport capability, was the capillary limit. Therefore, as a result of thermal resistance networking analysis, the performance of heat pipe obtained from R_{tot} is smaller than the capillary limit of vapor temperature, and consequently the operating limitations of heat pipe are satisfied.

In addition, the performance of heat pipe becomes increased with increasing length of evaporator and condenser, because the resistances in components associated with evaporator and condenser decreases. However, it becomes decreased with increasing length of adiabatic section due to the increase of R_v, and consequently the performance of heat pipe showed the biggest change.

ACKNOLEDGMENTS

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