Transient Lumped Parameter Analysis of Heat Pipe for a Space Nuclear Reactor

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1. Introduction

Korea Atomic Energy Research Institute (KAERI) has been developing a design concept and key technologies for a space nuclear reactor [1,2]. The space nuclear reactor adopts a heat pipe to transfer the nuclear fission heat to an electricity generating device, e.g., stirring engine. The heat pipe is a capillary-driven and two-phase flow device. It is attractive in space since it is capable of transporting large amount of heat using passive and reliable manners with small sizes.

This paper describes a lumped parameter numerical model which is able to simulate steady-state as well as transient operation of the heat pipe. Although the physical mechanisms related to transient heat pipe operation are numerous and complex, transient response of a heat pipe has been well studied and functional detailed models such as THROHPUT [3] and HPTAM [4] are already available. However, it is doubtful that such levels of details are necessary in engineering-approach simulation of a heat pipe (in particular, under a conceptual design stage).

The main objective of the present work is to build a simple, reliable, and robust numerical model to design and analyze a heat pipe for practical engineering applications.

2. Numerical Method

A lumped parameter model was adopted in this work due to its simplicity as well as popularity in heat pipe applications. The lumped parameter model was adopted by classical computer programs (e.g., HTPIPE [5] and ANL/HTP [6]) as well as recent researches (e.g., Ferrandi et al.'s work [7]).

Fig. 1 shows a thermal network diagram adopted in this work. A total of six temperature nodes are used for the pipe and wick temperatures of the heat pipe. It is assumed that liquid and solid temperatures are the same in the wick zone. Heat source (Q_{in}) imposed to the external surface of the evaporator and the temperature of heat sink fluid (T_f) are the boundary conditions.

The general transient heat conduction equation governing six nodes ($T_{pe}, T_{pa}, T_{pc}, T_{we}, T_{wa}, T_{wc}$) is:

$$C_i \frac{dT_i}{dt} = Q_{in} \pm \sum_{j=1, j\neq i}^{nearby\ nodes} \frac{|T_i - T_j|}{R_{ij}} \qquad (1)$$

where C_i = effective heat capacity, R_{ij} = thermal resistance. In the right hand side of Eq. (1), the heat source term appears only for the evaporator pipe node (= T_{pe}). In addition, '+' is used for incoming heat flow to node *i* whereas '-' is used for outgoing heat flow from node *i*.

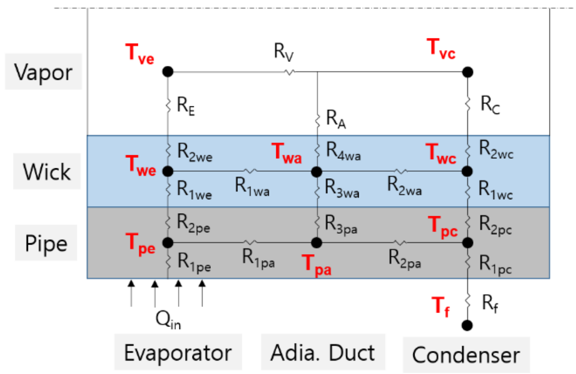

Fig. 1. Thermal network of the present heat pipe model.

For the vapor temperatures (T_{ve}, T_{vc}), quick thermal equilibrium is assumed in this work.

$$\frac{T_{we} - T_{ve}}{R_{2we} + R_E} = \frac{T_{vc} - T_{wc}}{R_{2wc} + R_C} = \frac{T_{ve} - T_{vc}}{R_V} = Q_{trans} \qquad (2)$$

where Q_{trans} = amount of heat transfer by vapor flow. Such an assumption can be justified by the fact that heat capacity of vapor is much smaller than that of wick or pipe. For example, the volumetric heat capacity of saturated sodium vapor is less than 0.02 % of stainless steel 304 at 900 ^{o}C.

The analytical expressions of thermal resistances for pipe and wick are:

$$R_{radial} = \frac{\ln(\frac{r_e}{r_i})}{2\pi kL} \qquad (3)$$

$$R_{axial} = \frac{L}{kA} \qquad (4)$$

where r_e = external radius, r_i = internal radius, k = thermal conductivity, A = cross-sectional area, and L = length of the given zone. Effective value is used for the thermal conductivity of wick. The thermal resistance at the outside surface of the condenser (R_f) can be expressed as:

$$R_f = 1/(h_f A_f) \qquad (5)$$

where h_f = heat transfer coefficient and A_f = external surface area of the condenser. The axial thermal resistance of the vapor space is calculated from the Clausius-Clayperon equation, which relates the change in saturation pressure and temperature of the working fluid [8].

$$R_V = \frac{T_{ave}\,\Delta P}{\rho_{ave}\,h_{fg}\,Q_{trans}} \quad (6)$$

where T_{ave} = average temperature, ΔP = vapor pressure drop, ρ_{ave} = average density, h_{fg} = latent heat of vaporization. The thermal resistance at the boiling and condensing surfaces are calculated using the model presented by Dunn and Reay [9]. The model uses the Clausius-Clayperon relationship for saturated vapor and the fact that the evaporating or condensing mass transport is proportional to the heat transport. In case of boiling at the evaporator, the thermal resistance at the liquid-vapor interface is derived as:

$$R_E = \frac{\sqrt{2\pi RT}\,RT^2}{P\,h_{fg}^{\,2}A_E} \quad (7)$$

where R = gas constant of vapor, P = pressure, and A_E = surface area for evaporation. A similar expression for condensing (= R_C) can be defined by replacing the surface area only.

Since the thermal resistances R_V, R_E, R_C depend on the vapor temperature, iterative calculations are necessary to obtain the vapor temperature.

Eqs. (1) ~ (7) can be solved simultaneously by using a standard numerical solver for combined ordinary differential equations.

3. Validation Results and Discussions

The feasibility of the present method was studied by using the Los Alamos National Laboratory (LANL) experiment using a sodium heat pipe [10]. Stainless steel sodium heat pipe modules were built and tested at LANL for use in a thermo-hydraulic simulation of a space nuclear reactor. A cutaway drawing of the heat pipe module with its four cartridge heater is shown in Fig. 2. The major dimensions of the heat pipe are provide in Table I. The annular wick was fabricated from 304 L stainless steel screens. The wick consists of one support layer of 100-mesh screen, three capillary pumping layers of 400-mesh screen, and two liquid flow layers of 60-mesh screen. The test for the effective pore radius verified that the pore radius of the wick was less than 47 microns.

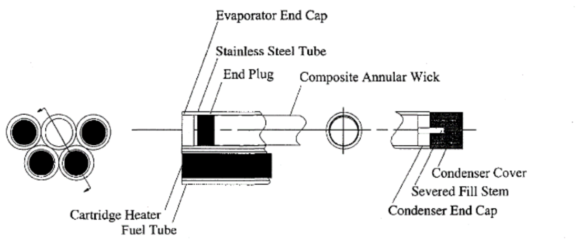

Fig. 2. Cutaway view of sodium heat pipe used for the LANL experiment [10].

Table I: Heat Pipe Module Dimensions

Item	Value
Heat pipe length (cm)	120
Evaporator length (cm)	43
Condenser length (cm)	77
Heat pipe outside diameter (cm)	2.54
Heat pipe inside diameter (cm)	2.21
Wick outside diameter (cm)	2.07
Annular gap thickness (cm)	0.07
Wick inside diameter (cm)	1.74
Wick thickness (cm)	0.17
Effective pore radius (μm)	47

There was no adiabatic zone in the heat pipe. The condenser of the heat pipe was exposed to ambient air with room temperature and totally cooled by thermal radiation. The heat pipe was heated from room temperature. The heater power was increased to the heat pipe module in 80 W increments at five minute intervals until 3 hours, kept constant during 0.5 hour, and decreased slowly during 1.5 hours. Fig. 3 shows the applied heat input to the heat pipe. It was assumed that the heat input was linearly applied. The reference [10] reported that the peak power was 660 W at the measured peak surface temperature of the heat pipe (=900 K). Using this information, the emissivity of the heat pipe was estimated by the analytic expression of radiation heat transfer in this work.

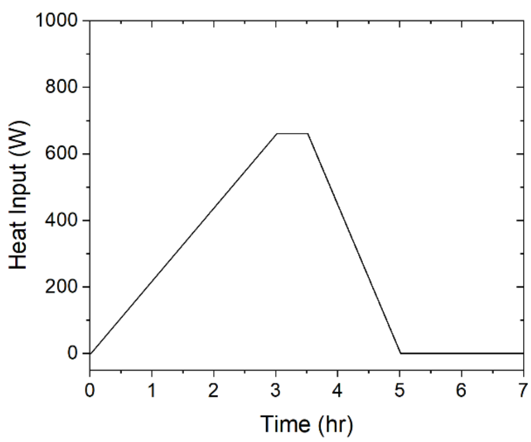

Fig. 3. Applied heat input vs time.

Fig. 4 compares the predicted and measured surface temperature of the heat pipe. A good agreement is shown in higher temperature region whereas some deviations exist in lower temperature region. It seems that the discrepancy in lower temperature region is mainly due to Eqs. (6) and (7), which are valid for small temperature difference of vapor between the evaporator and the condenser. Improved models are necessary for accurate simulation of start-up behavior of the heat pipe.

Fig. 5 shows the calculated heat removal by vapor and the operational limits of the heat pipe. During early stage of the experiment, the vapor heat transport is slightly above the viscous limit. The heat pipe is normally operated during entire duration of the experiment.

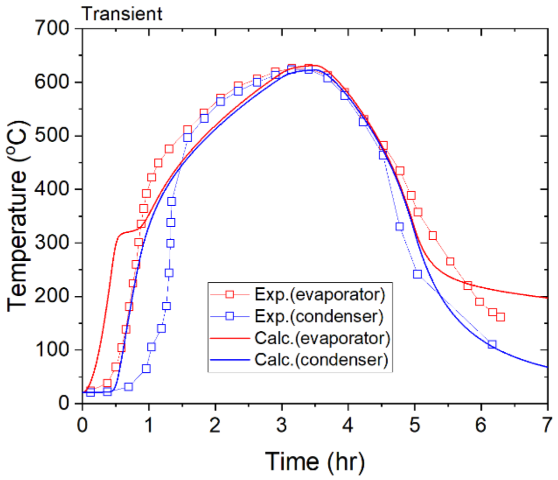

Fig. 4. Comparison of the predicted and measured surface temperatures of the heat pipe.

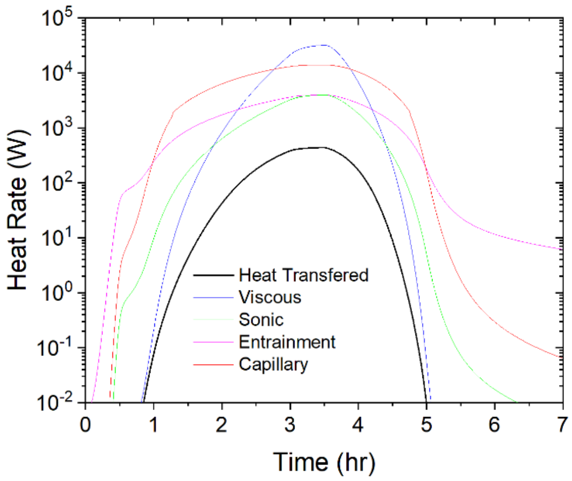

Fig. 5. Calculated vapor heat transport and operational limits of the heat pipe.

4. Conclusions

In this paper, a transient lumped parameter method was introduced for analysis and design of a heat pipe. It was found that the proposed numerical model can provide reliable and accurate results with fast computational speed when the heat pipe is operated under normal operating conditions. Further studies to improve the prediction during start-up of a heat pipe are necessary. More studies on the verification and validation of the present method are on-going.

Acknowledgements

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