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Numerical Simulation of Variable Conductance Heat Pipe with Cold Reservoir by Single Phase Flow Approach

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ABSTRACT: The heat pipes are usually simulated by using a two phase model and a model describing the phase-change process. The computational costs of the two-phase approaches are relatively high and the model generally needs small-size time steps, which leads to a long simulation run times in the order of several days. In the present study, a variable conductance heat pipe is simulated by using a set of single-phase fluid flow models. It is shown that the proposed approach needs to a simulation time in the order of minutes that considerably facilitates the parametric study process of the variable conductance heat pipe. The effect of heat rate, sink temperature, mass of non-condensable gas, vapor radius, and wick porosity on the performance of variable conductance heat pipe are investigated. For the considered variable conductance heat pipe, the obtained numerical results indicate that sink temperature has the greatest effect on distributions of average wall temperature, overall heat transfer coefficient, the active length of condenser, and its average temperature. By increasing the sink temperature of 10 K, the active length of condenser is increased about 48 mm and average wall temperature is increased about 6.4 K.

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1- Introduction

A typical heat pipe, as a device that can efficiently transfer heat, works based on the liquid-vapor phase change and the use of a porous medium, named wick, to provide capillary pressure as the driving force for the liquid phase flow. Conventional heat pipe consists of three main parts, including an evaporator, a condenser, and an adiabatic part. As a special type of heat pipes, a Variable Conductance Heat Pipe (VCHP) consists of the condenser, evaporator, adiabatic parts, as shown in Fig. 1. Compared to the conventional one, a variable conductance heat pipe has an extra part, which is a reservoir containing a Non-Condensable Gas (NCG). In some cases, the portion of one end part of the heat pipe (i.e. the condenser or evaporator) is occupied by the non-condensable gas, which plays the role of the NCG reservoir. In the present study, a cold type VCHP is numerically simulated in which the NCG reservoir is located beside the condenser part (Fig. 1). Whenever the sink temperature is reduced, the vapor pressure reduces and the non-condensable gas expands to occupy some portion of the condenser part. This avoids condensation of the working fluid in that part of the condenser occupied by the NCG. It means that this part of the condenser is deactivated, due to the presence of non-condensable gas. Whenever the vapor pressure is increased, the non-condensable gas is accumulated at the end part of the condenser. Thus, most portion of the condenser is activated and heat transfer via phase change takes place with the environment.

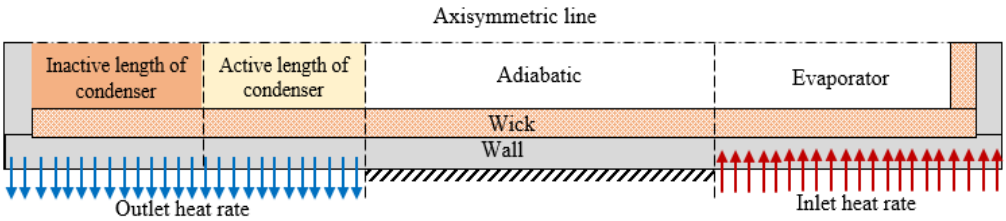

Fig. 1. Schematic of the variable conductance heat pipe

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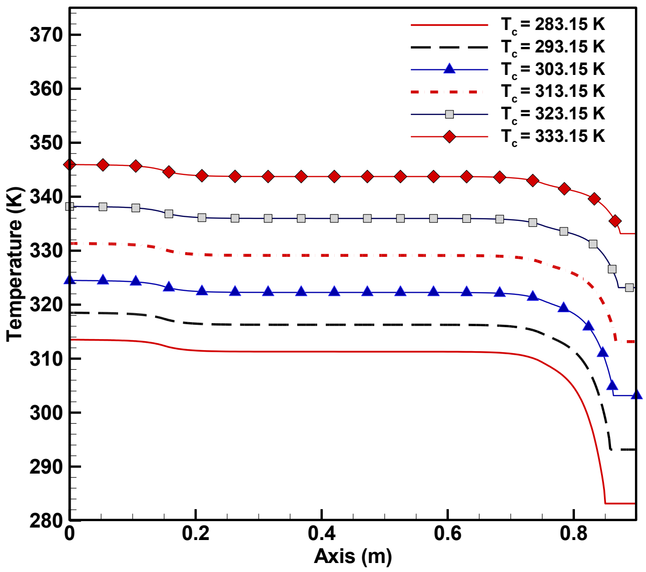

Fig. 2. Effect of sink temperature on the wall temperature profile

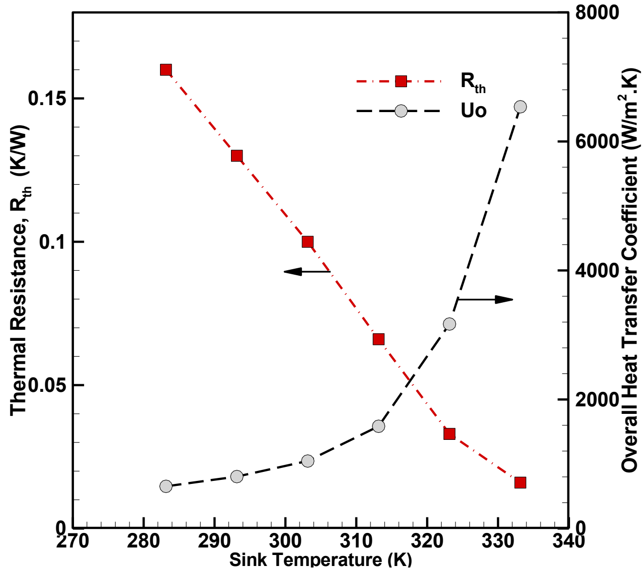

Fig. 3. Changes in thermal resistance and overall heat transfer coefficient versus the sink temperature

Due to the existence of a vacuum environment, lack of gravity, and variable working temperature, experimental studies on the heat pipes used in space applications is very expensive. In addition, laboratory data can have errors, as the real working conditions cannot be exactly provided in the experimental setup. Therefore, numerical studies, in general, have received many attentions as their cost is cheaper than that of experimental ones and also numerical simulations can produce more accurate results by considering more realistic working conditions. However, the numerical simulation of mass transfer and heat transfer phenomena of a VCHP is complicated. There exist several factors, such as working fluid type and NCG and geometric and physical characteristics of the reservoir, affecting the VCHP performance [1-4]. One of the major problems of numerical simulations of heat pipes is the high computational costs of two-phase models, which leads to long simulation times. A two-phase flow model, such as the Volume of Fluid (VOF) or level set, is used to determine the liquid-vapor interface. This needs small time steps, which leads to a long run time.

It has been shown in various references [5-8] the simulation time of heat pipe based on the multi-phase flow models such as VOF, is extremely high in some cases in such a way that it needs several days of completion. In the present study, it is shown that by using the proposed single-phase flow approach the required simulation time on a system with a Core-i7 processor (3.2 GHz) and 8Gb memory reaches to a maximum of 20 minutes, which is a very reasonable time for the parametric study unlike those of simulation based on the multi-phase models.

2- Methodology

In this section, the governing equations are written for three different sections consist of a vapor space, a wicked space, and a wall space (Fig. 1), as follows.

Vapor space:

$$\nabla\cdot(\rho_v\vec{u}_v)=0 \tag{1}$$

$$\rho_v(\overrightarrow{u_v}\cdot\nabla)\overrightarrow{u_v}=-\nabla p_v+\mu_v\nabla^2\overrightarrow{u_v} \tag{2}$$

$$\rho_v C_{pv}\overrightarrow{u_v}\cdot\nabla T=K\nabla^2 T \tag{3}$$

$$-D_{vg}\nabla^2 c_v+\overrightarrow{u_v}\cdot\nabla c_v=0 \tag{4}$$

Wick space:

$$\nabla\cdot(\rho_l\vec{u}_l)=0 \tag{5}$$

$$\vec{u}_l=-\frac{k}{\mu_l}\nabla p_l \tag{6}$$

$$\rho_l C_{pl}\vec{u}_l\cdot\nabla T-K_{eff}\nabla^2 T=Q_0 \tag{7}$$

$$-D_{cap}\nabla^2 c_l+\vec{u}_l\cdot\nabla c_l=R_l \tag{8}$$

Wall space:

$$k_w\nabla^2 T_w=Q \tag{9}$$

3- Results and Discussion

In the present study, the governing equations describing the VCHP are solved by using based on the finite element method. To discretize the governing equations, the linear first order shape functions are used for pressure and temperature variables and second order shape functions for velocity variable. Enthalpy of vaporization and vapor saturation pressure is represented as the linear functions of temperature. The variation of wall temperature associated with different sink temperatures is illustrated in Fig. 2. By an increase in sink temperature of 10 K, the wall temperature is increased by an average value of 6.4 K, while the temperature difference between evaporator and condenser is decreased. In order to achieve the heat dissipation rate of 30 W from a condenser with a high sink temperature, the average temperature of the VCHP should be high enough to provide the required temperature difference for this specified heat transfer rate. Fig. 3 shows the variation of the thermal resistance and also overall heat transfer coefficient versus the sink temperatures. It can be observed that thermal resistance decreases by an average value of 0.028 $\mathrm{K/W}$ by an increase in the sink temperature. The overall heat transfer coefficient increased by an average value of 170.6 $\mathrm{W/(m^2 \cdot K)}$ by increasing the sink temperature. In fact, the temperature difference between evaporator and condenser is decreased by increasing the sink temperature. This results in a decrease in the thermal resistance and an increase in the overall heat transfer coefficient. The variations of the active length of the condenser and its average temperature corresponding to the different sink temperature is shown in Fig. 5. As the sink temperature rises, the active length of the condenser and its temperature is increased respectively by an average value of 48 mm and 6.6 K. By an increment in the sink temperature (i.e. 10 K), the vapor pressure increases, which results in a decrease in the volume of non-condensable gas. By a reduction in the volume of NCG, the effective length of the condenser for heat dissipation is increased.

4- Conclusions

In the present study, the computational cost and simulation run time of a VCHP were significantly reduced by using the proposed single-phase approach instead of multiphase flow models. It was shown that the simulation time of VCHP reduced from the order of several days to the minutes with acceptable accuracy. This ability of the proposed approach, definitely, facilitates the procedure of parametric study on the VCHPs. Based on the numerical results for the considered VCHP, the following points can be highlighted; The sink temperature has the most effect on the condenser active length, its temperature, overall heat transfer coefficient, and also the wall temperature in such a way that the active length of condenser increases on average 48 and its temperature increases on average 6.6 Kby an increase of 10 K in sink temperature. By increasing sink temperature, the average wall temperature increases by an average value of 6.4 K, and the overall heat transfer coefficient increases by an average value of 170.6 $\mathrm{W/(m^2 \cdot K)}$.

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