Effect of the Heat Pipe Adiabatic Region

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The main motivation of conducting this work is to present a rigorous analysis and investigation of the potential effect of the heat pipe adiabatic region on the flow and heat transfer performance of a heat pipe under varying evaporator and condenser conditions. A two-dimensional steady-state model for a cylindrical heat pipe coupling, for both regions, is presented, where the flow of the fluid in the porous structure is described by Darcy–Brinkman–Forchheimer model which accounts for the boundary and inertial effects. The model is solved numerically by using the finite volumes method, and a FORTRAN code was developed to solve the system of equations obtained. The results show that a phase change can occur in the adiabatic region due to temperature gradient created in the porous structure as the heat input increases and the heat pipe boundary conditions change. A recirculation zone may be created at the condenser end section. The effect of the heat transfer rate on the vapor radial velocities and the performance of the heat pipe are discussed. [DOI: 10.1115/1.4025132]

Keywords: heat pipes, finite volume, two-dimensional, adiabatic region, high flux

Introduction

Heat pipes are often referred to as the superconductors of heat as they possess an extraordinary heat transfer capacity and rate with almost no heat loss. The transport phenomena in the heat pipe have been of continuing interest for the five decades. This interest stems from the complicated and interesting phenomena associated with energy transport within the heat pipe regions.

Their investigations range in complexity from simple state thermodynamic analyses [1,2] to transient, multidimensional computational models of the fluid flow, and heat transfer for the liquid and vapor phases [3,4]. The pioneering in steady-state heat pipe analysis was carried out by Cotter [5]. His work assumed steady-state incompressible flow in both liquid and vapor regions with constant boundary conditions in the evaporator, adiabatic, and condenser sections. His model predicted heat pipe fluid velocity and pressure profiles, as well as delineating some of the theoretical limits to heat pipe operation. As the vapor flow in the pipe is a complicated problem, different approaches have been used to simplify the problem. A detailed vapor flow analysis was performed by Busse [6], provided that the vapor flow is steady, laminar, and incompressible. HTPIPE was the first comprehensive heat pipe design and analysis code to be developed [7]. Many heat pipe designers have found that more detailed analyses of the fluid flow and heat transfer in the heat pipe are required with multiple heat sources and sinks [8] or for heat pipe behavior at start-up from its frozen state [9]. Jong et al. [10,11] elaborated a one-dimensional transient model for the wall and wick regions. The one-dimensional transient model for the vapor region was coupled to examine the effect of the boundary conditions specification at the surface of the outer wall on the successful start-up.

A two-dimensional heat and mass transfer in a vapor gas region of gas loaded circular heat pipe is made by Rajashree and Sankara Rao [12]. The analysis is restricted to the vapor space and to the qualitative behavior of the start-up transient in the vapor gas region. Issacci et al. [13] examined the vapor flow patterns in heat pipes during the start-up transient phase. The vapor core is modeled as a channel flow using a two-dimensional compressible flow model. For high heat input, a reverse flow is detected in both

adiabatic and condenser regions, suggesting that a parametric study is needed to explore the reverse flow versus design parameters like the heat pipe regions' lengths for different heat inputs.

Heat pipe transient analysis model code represents one of a notable contribution to the field of heat pipe modeling and it becomes a valuable tool when working with metal liquid heat pipes, where start-up from the frozen state is an important design consideration parameter [14,15].

Zhu and Vafai [16] analyzed a three-dimensional vapor and liquid flow in an asymmetric flat plate heat pipe. They studied the vapor flow by finite element method using FIDAP code. They also used a non-Darcian model for investigation of the liquid flow in the porous media. Vafai and Wang [17] analyzed in-depth various physical aspects of the asymmetrical flat plate heat pipe indicating that the temperature wall surface is quite uniform and the porous wick of the evaporator section creates the main thermal resistance. Xu et al. [18] conducted a parametric study focusing on the influences of various factors on flow resistance and heat transfer performance for fully developed forced convection in a tube partially filled with open-celled metallic foams. Nouri-Borujerdi and Layeghi [19] analyzed the vapor flow in a concentric annular heat pipe using SIMPLE algorithm and staggered grid scheme. They found the pressure distribution for different radial Reynolds numbers. A two-dimensional analytical model for low-temperature cylindrical heat pipes was elaborated by Zhu and Vafai [20]. The effects of liquid–vapor interfacial hydrodynamic coupling and non-Darcian transport through the porous wick on the vapor and liquid velocity and pressure distributions as well as the heat pipe capillary limit are discussed and assessed.

A sodium/stainless steel heat pipe with a simple circumferential screen wick and multiple heat sources were successfully fabricated and tested by Fahgri [21,22], both in air and under vacuum environment conditions. He observed that the low condenser heat rejection rate during vacuum operation apparently prevented supersonic vapor velocities and allowed the condenser to slowly rise in temperature to the final steady-state condition. He notes also that the capillary limits for block-heated operation were slightly higher than the circumferentially heated limits. This can be partially attributed to the shorter adiabatic transport section of the block-heated heat pipe. Fahgri and Buchko [23] showed that decreasing the adiabatic transport length can significantly increase the capillary limit. A liquid-metal heat pipes have been investigated by Chang [24] to remove the high heat flux associated with

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the missile fins being located in hot exhaust gas paths. Using steady-state and transient analyses, a solution method has been developed to identify the required length of the condenser for a design operation when the heat pipe is coupled with aerodynamic heating and cooling environments. It has been found that the proper length of the condenser is critical for heat pipe design. If the condenser length is too short, the heat pipe temperature increases dramatically. Meanwhile, if it is too long, there is a risk of freezing the vaporized sodium in the inactive condenser region, causing dry out in the evaporator due to insufficient liquid.

To our knowledge, less attention has been paid to study the heat pipe behavior where it is exposed to higher heat input at the evaporator or higher heat output in the condenser region. Also the effect of the adiabatic and the condenser lengths in these conditions was not been well investigated.

In this paper, a two-dimensional numerical steady-state model for the entire heat pipe regions has been solved in cylindrical co-ordinates with two specific boundary conditions (conduction and forced convection) at the surface of the outer wall in the condenser region. The Darcy–Brinkman–Forchheimer model has been employed for the momentum equation in the porous wick. The local average volume form using SIMPLE algorithm with the collocated grid scheme is used to solve the governing equations and a FORTRAN code was developed. Four cases are investigated, where two cases are analyzed for the two configurations.

Mathematical Formulation

The schematic of the cylindrical heat pipe and the co-ordinate system used in the present analysis is shown in Fig. 1.

Vapor and liquid flows are assumed to be steady, laminar, and incompressible. The wick is assumed to be isotropic and homogeneous, saturated with the working fluid which has constant physical properties and in local thermal equilibrium with the wick structure. Following the analysis given in Vafai and Tien [25], the governing equations for the liquid flow and heat transfer in the porous wick are

$$\frac{1}{r} \frac{\partial (rv)}{\partial r} + \frac{v}{r} + \frac{\partial u}{\partial z} = 0 \quad (1)$$

Fig. 1 Schematic of conventional cylindrical heat pipe and simulation cases investigated

$$\rho \left(v \frac{\partial u}{\partial r} + u \frac{\partial u}{\partial z} \right) = -\varepsilon \frac{\partial P}{\partial z} + \mu \left(\frac{\partial^2 u}{\partial r^2} + \frac{1}{r} \frac{\partial u}{\partial r} + \frac{\partial^2 u}{\partial z^2} \right) - \frac{\mu \varepsilon}{K} u - \frac{\rho C_F \varepsilon^2}{\sqrt{K}} |u| u + \rho g \sin \theta \quad (2)$$

$$\rho \left(v \frac{\partial v}{\partial r} - \frac{v^2}{r} + u \frac{\partial v}{\partial z} \right) = -\varepsilon \frac{\partial P}{\partial r} + \mu \left(\frac{\partial^2 v}{\partial r^2} + \frac{1}{r} \frac{\partial v}{\partial r} - \frac{v}{r^2} + \frac{\partial^2 v}{\partial z^2} \right) - \frac{\mu \varepsilon}{K} v - \frac{\rho C_F \varepsilon^2}{\sqrt{K}} |v| v \quad (3)$$

$$v \frac{\partial T}{\partial r} + u \frac{\partial T}{\partial z} = \alpha_{\text{eff}} \left(\frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} \right) \quad (4)$$

u and v are the axial and radial velocity components, respectively, where $|u| = |v| = \sqrt{u^2 + v^2}$. $\sqrt{K}$ represents the effective pore diameter, C_F is the Ergun's constant [26] which is taken to be approximately 0.55, and α_{eff} is the effective thermal diffusivity which is equal to

$$\alpha_{\text{eff}} = \frac{\lambda_{\text{eff}}}{\varepsilon \rho_l C_{pl}} \text{ in the wick structure, and } \alpha_{\text{eff}} = \frac{\lambda_v}{\rho_v C_{pv}} \text{ in the vapor region} \quad (5)$$

In the vapor region, the porosity is considered to be $\varepsilon = 1$ and the permeability $K \rightarrow \infty$. This provides a convenient formulation for the numerical solution in this region.

The heat transfer through the heat pipe wall is transferred purely by conduction. The corresponding energy equation is

$$\left[\frac{1}{r} \frac{\partial}{\partial r} \left(r \frac{\partial T_s}{\partial r} \right) + \frac{\partial^2 T_s}{\partial z^2} \right] = 0 \quad (6)$$

where subscript "s" means the solid wall of the heat pipe. The effective thermal conductivity in cylindrical co-ordinate is obtained under the assumption that the wick was composed of wrapped screen mesh [27]

$$\lambda_{\text{eff}} = \frac{\lambda_l [(\lambda_l + \lambda_s) - (1 - \varepsilon)(\lambda_l - \lambda_s)]}{[(\lambda_l + \lambda_s) + (1 - \varepsilon)(\lambda_l + \lambda_s)]} \quad (7)$$

Table 1 Terms in the generic conservation equation

Equation	S_a	S_b
$r_momentum$	$-\frac{2}{Re} \frac{1}{R^2} - \frac{2\varepsilon(1+C'_F V)}{ReDa}$	$-\frac{\varepsilon}{2} \frac{\partial P'}{\partial R}$
$z_momentum$	$-\frac{2\varepsilon(1+C'_F U)}{ReDa}$	$-\frac{\varepsilon}{2} \frac{\partial P'}{\partial Z} + g'$
Continuity and energy	0	0

Table 2 Basic specifications of the different parameters of the heat pipe

Heat pipe wall	Wick structure	Vapor region
$R = 0.022$ m	$r_c = 0.02$ m	$r_i = 0.0127$ m
$L_{evap} = 0.4$ m	$\varepsilon = 0.46$	$\rho_v = 0.599$ kg/m ³
$L_{ad} = 0-0.2$ m	$K = 0.267 \times 10^{-10}$ m ²	$\lambda_v = 0.0251$ W/mK
$L_{cond} = 0.6-0.4$ m	$\rho_l = 960.63$ kg/m ³	$\mu_v = 0.129 \times 10^{-4}$ kg/m s
$L_{tot} = 1.0$ m	$\lambda_l = 0.680$ W/mK	$C_{pv} = 1888$ J/kg K
$\lambda_s = 387.6$ W/mK	$\mu_l = 2.8243 \times 10^{-4}$ kg/m s	$h_{fg} = 225.6267 \times 10^4$ J/kg
$\rho_s = 8978$ kg/m ³	$C_{pl} = 4216$ J/kg K	$\sigma = 5.89 \times 10^{-2}$ Pa m
$T_a = 25$ °C	$\lambda_{eff} = 3.0476$ W/mK	$P_0 = 1.0197 \times 10^5$ Pa
$C_{ps} = 381$ J/kg K	$C_{p,eff} = 3.8 \times 10^6$ J/kg K	$T_{sat} = 373.15$ K
$h_{conv} = 800$ W/m ² K	Material: sintered copper	$R_v = 488$ J/kg K
Material: copper	—	—

To solve the above equations, boundary conditions should be given at end caps of the heat pipe, centerline, vapor–wick interface, wick–wall interface, and outer surface of the heat pipe in both regions.

■ In the vapor region

$$v_v = u_v = 0; \quad \frac{\partial T_v}{\partial z} = 0 \text{ at } z = 0 \text{ and } z = L \quad (8a)$$

$$v_v = 0; \quad \frac{\partial u_v}{\partial r} = 0; \quad \frac{\partial T_v}{\partial r} = 0 \text{ at } r = 0 \quad (8b)$$

$$v_v = v_i(z); \quad u_v = 0; \quad T_v = T_{sat}(P_i) \text{ at } r = r_i \quad (8c)$$

At the vapor–wick interface, the temperature is assumed to be the saturation temperature corresponding to the interface pressure. Thus, by applying Clausius–Clapeyron equation, the saturation temperature can be determined by

$$T_{sat} = \left(\frac{1}{T_0} - \frac{R_v}{h_{fg}} \ln \frac{P_v}{P_0} \right)^{-1} \quad (8d)$$

T_0 and P_0 are reference saturation temperature and the reference saturation pressure (i.e., 373 K and 1.01×10^5 Pa). R_v is the gas constant for the vapor. Since the density of the vapor is linked to saturation temperature and vapor pressure to surface tension, an

Fig. 2 Validation code with comparison at $Q = 12,639$ W/m²: (a) outer surface wall temperature distribution, (b) liquid and vapor pressure distribution, (c) maximal liquid velocity, and (d) mean vapor velocity

Fig. 3 Axial and radial vapor velocities profiles at 0.5 with $Q = 5000 \text{ W/m}^2$ (left) and $Q = 50,000 \text{ W/m}^2$ (right) with and without adiabatic section

iterative schedule is then added to calculate the vapor saturation temperature where the physical fluid properties depend on temperature. In this work, it is assumed also that the volume of the liquid within the vapor phase is neglected [28].

By using the energy balance at the interface liquid–vapor, the boundary condition for the injection velocity $v_i(z)$ which couples the vapor and the porous region of this interface is defined as

$$Q + \lambda_{\text{eff}} \frac{\partial T_1}{\partial r} = \rho_v v_i h_{fg} \quad (8e)$$

■ In the porous region

$$v_1 = u_1 = 0 \text{ and } \lambda_{\text{eff}} \frac{\partial T_1}{\partial r} = \lambda_s \frac{\partial T_s}{\partial r} \text{ at } r = r_e \quad (9a)$$

$$v_1 = u_1 = 0; \quad \frac{\partial T_1}{\partial z} = 0 \text{ at } z = 0 \text{ and } z = L \quad (9b)$$

$$v_1 = v_i(z); u = 0; T_1 = T_{\text{sat}}(P_1) \text{ at } r = r_i \quad (9c)$$

■ In the wall region

$$\lambda_{\text{eff}} \frac{\partial T_1}{\partial r} = \lambda_s \frac{\partial T_s}{\partial r} \text{ at } r = r_e \quad (10a)$$

$$\frac{\partial T_s}{\partial z} = 0 \text{ at } z = 0 \text{ and } z = L \quad (10b)$$

Two boundary conditions are considered at the outer surface condenser section ($r = r_s$)

$$\text{Constant heat flux : } -\lambda_s \frac{\partial T_s}{\partial r} = -Q \text{ (first b.c kind)} \quad (10c)$$

$$\text{Forced convection : } -\lambda_s \frac{\partial T_s}{\partial r} = h_{\text{conv}}(T_s - T_a) \text{ (second b.c kind)} \quad (10d)$$

For the numerical analysis, the governing equations (1)–(4) in the dimensionless variables are

$$V = v/v_i \quad U = u/v_i \quad P' = 2P/\rho v_i^2 \quad C'_F = \rho \varepsilon C_F \frac{\sqrt{K}}{\mu} v_i \quad (11)$$

$$R = r/r_i \quad Z = z/r_i \quad g' = 2gr_i/v_i^2$$

Reduces to the form

$$\frac{\partial V}{\partial R} + \frac{V}{R} + \frac{\partial U}{\partial Z} = 0 \quad (12)$$

$$\left(V \frac{\partial U}{\partial R} + U \frac{\partial U}{\partial Z} \right) = -\frac{\varepsilon}{2} \frac{\partial P'}{\partial Z} + \frac{2}{\text{Re}} \left(\frac{\partial^2 U}{\partial R^2} + \frac{1}{R} \frac{\partial U}{\partial R} + \frac{\partial^2 U}{\partial Z^2} \right) - \frac{2\varepsilon}{\text{ReDa}} U - \frac{2\varepsilon C'_F}{\text{ReDa}} |U|U + g' \sin \theta \quad (13)$$

Fig. 4 Axial vapor velocity streamlines for different fluxes and boundary conditions

$$\left(V \frac{\partial V}{\partial R} + U \frac{\partial V}{\partial Z} \right) = -\frac{\varepsilon}{2} \frac{\partial P'}{\partial R} + \frac{2}{\text{Re}} \left(\frac{\partial^2 V}{\partial R^2} + \frac{1}{R} \frac{\partial V}{\partial R} - \frac{V}{R^2} + \frac{\partial^2 V}{\partial Z^2} \right) - \frac{2\varepsilon}{\text{ReDa}} V - \frac{2\varepsilon C_F}{\text{ReDa}} |V|V \quad (14)$$

$$V \frac{\partial T}{\partial R} + U \frac{\partial T}{\partial Z} = \frac{2}{\text{Pe}} \left(\frac{\partial^2 T}{\partial R^2} + \frac{1}{R} \frac{\partial T}{\partial R} + \frac{\partial^2 T}{\partial Z^2} \right) \quad (15)$$

where

$$\begin{aligned} \text{Re} &= 2r_i \rho v_i / \mu & \text{Pr} &= \mu C_p / \lambda \\ \text{Pe} &= \text{RePr} & \text{Da} &= K / r_i^2 \end{aligned} \quad (16)$$

The full set of governing equations (12)–(15) is transformed in a generalized convection–diffusion equation as follows:

$$\begin{aligned} \frac{1}{R} \frac{\partial}{\partial R} (RV\varphi) + \frac{1}{R} \frac{\partial}{\partial Z} (RU\varphi) &= \frac{2}{\text{Pe}^*} \frac{1}{R} \frac{\partial}{\partial R} \left(R \frac{\partial \varphi}{\partial R} \right) \\ &+ \frac{2}{\text{Pe}^*} \frac{1}{R} \frac{\partial}{\partial Z} \left(R \frac{\partial \varphi}{\partial Z} \right) + S_\varphi \end{aligned} \quad (17)$$

where

$\varphi = V, U, T$: dependent variables

$$\text{Pe}^* \begin{cases} = \text{Re} & \text{in Momentum equation} \\ = \text{RePr} & \text{in Energy equation} \end{cases}$$

$S_\varphi = S_a \varphi + S_b$ is the dimensional source term (Table 1).

Solution Method

The conjugate heat transfer and flow problem were solved using the control volume method with the power law scheme and SIMPLE algorithm. The line-by-line scan, which is a combination of the tridiagonal matrix algorithm and the Gauss–Seidel procedure, was used and a FORTRAN code was developed to solve the resulting set of generic equations along with boundary conditions over the entire heat pipe regions.

The simulation of the heat pipe is established with heat pipe dimension and water working fluid is shown in Table 2.

The precision of the solution was evaluated through the mass transfer residual in the interface.

Since the continuity of mass fluxes in the radial direction at the liquid–vapor interface yields $\rho_v v_v(z, r = r_i) = \rho_l v_l(z, r = r_i) = Q(z)/h_{fg}(T)$ therefore, the mass flow rate can be obtained by the energy conservation equation.

The mass flux residual like velocities and pressure residuals are used as a criterion of convergence [29].

The mass flux is $\dot{m} = \rho V_i \int 2\pi R(z) dz$, and in its discrete form $\dot{m} = \sum_{n=1}^N \rho V_i R(z) \Delta z$.

In wall region, the temperature residual is used for convergence criteria. The converged solution is considered when the temperature residual decreases three to four orders of magnitude.

Fig. 5 Radial vapor velocity streamlines for different fluxes and boundary conditions

Results

To investigate the heat pipe behavior under different operating conditions, four cases are considered. In configuration (a), the heat pipe is exposed to moderate heat flux at the evaporator section (5000 W/m^2). In this case, two boundary conditions are applied in the condenser outer surface region by condition or forced convection at higher heat flux. In the configuration (b), a higher heat flux is applied to the heat pipe outer surface evaporator section ($50,000 \text{ W/m}^2$) and heat is removed in the condenser section by condition or by forced convection.

The comparison of our computation results was made with those numerical of Tounier and El-Genk [15] and analytical of Zhu and Vafai [20]. In Fig. 2, the results have a good agreement when using Darcy's law for wall temperature distribution, vapor and liquid pressure distribution, the maximum liquid velocity and the mean vapor velocity.

Figures 3(a) and 3(b) show the mean axial vapor velocity distribution for various heat fluxes and boundary conditions considered with adiabatic region ($L_a = 0.2 \text{ m}$) and without adiabatic region ($L_a = 0 \text{ m}$). According to the nonslip boundary conditions, the axial vapor velocity in the heat pipe ends is zero; it gradually increased at the evaporator section toward the end of evaporator section, while it was almost constant at the adiabatic section and gradually decreased at the condenser section. However, one can see that this velocity decreases slowly in the adiabatic section with the second boundary condition where the heat flow transmitted to the ambient environment by convection is small.

Figures 3(c) and 3(d) show the radial mean vapor velocity distribution along the pipe length with two different heat fluxes.

Figure 3(d) shows that the vapor is highly accelerated toward the condenser region, a peak is obtained where the process of condensation is unable to condense all the vapor flow reaching the condenser. This phenomenon is not observed when heat is extracted slowly (second kind b.c). Indeed, we can conclude that when the heat pipe is exposed to high heat flux, less heat removal in condenser section is needed for the heat pipe operation stability and the presence of an adiabatic region is necessary to reduce higher radial velocities in condenser region. This result was observed experimentally by Vafai et al. [30] by comparing the condensation process for a square disc and a triangular disc. They concluded that for the same heat flux input, disk-shaped heat pipe will condense the vapor mass more efficiently with small cooling surface.

The axial vapor streamlines velocity for the studied configurations are given in Fig. 4. When the heat flux received at the evaporator region is relatively weak, the generated vapor moves toward the condenser region slowly via the adiabatic zone related to the pressure drop profile by the thermodynamic equilibrium at the vapor-liquid interface (see Fig. 4(a)). With high heat flux input (Fig. 4(c)), the evaporation rate is high and the vapor is ejected with a high momentum from the evaporator to condenser region causing a significant pressure drop. This causes the recirculation of the flow in condenser end caps and the deformation of the velocity profile which start at the adiabatic region ends.

Figure 5 shows the radial vapor velocity streamlines. One can see that for the second boundary condition, a circulation flow starts in the adiabatic region and occupied a significant portion of the area. A phase change may occur in the adiabatic region due to the larger friction effect of the liquid caused by the temperature gradient in the wick structure (see Fig. 6(c)). Also with a high

Fig. 6 Liquid isotherms with and without adiabatic section and for the two kind boundary conditions ($Q = 50,000 \text{ W/m}^2$)

heat input condition, we can see the abrupt increase in the radial velocity especially in the condenser region because the length of the condenser region becomes insufficient to condense the totality of the evaporated vapor.

Figure 6 shows the temperature profile in the porous media for heat input condition of $Q = 50,000 \text{ W/m}^2$. It can be observed that in the porous region, the heat transfer process occurs mainly by heat conduction. For the first kind b.c (Figs. 6(a) and 6(b)), the presence of the adiabatic region has no significant effect on the temperature distribution within the heat pipe wick region. However, for the second kind b.c (Figs. 6(c) and 6(d)), a temperature gradient is created in the adiabatic region and a phase change can occur in this region (Fig. 6(c)) causing the change of the flow shape in the vapor region. This allows us to suppose that the formation and scattering of recirculation are dependent of the heat pipe adiabatic and condenser region lengths and the kind of boundary condition.

Figure 7 shows the streamlines in the end corner of the condenser region at high heat flux input with and without adiabatic section. For constant flows (first kind b.c), the presence of the adiabatic zone in a heat pipe does not affect, notably, the number of the formed cells. Contrary to the case of cooling by forced convection, the presence of the adiabatic zone reduced the number of forming cells (Figs. 7(b) and 7(d)). So, not only slowly cooling condition can reduce the phenomena of reverse flow but also the

presence of an adiabatic section. One can see that the difference between Figs. 7(a) and 7(c) is the number of cells. In Fig. 7(a), the number of cells is reduced because of the larger condenser length which plays the role of an adiabatic section to maintain the heat pipe operation stability.

Figures 8 and 9 show the influence of the adiabatic region on the velocity, pressure, and temperature in the porous wick structure for higher heat input. Figure 8 shows that the axial velocity will be increased without adiabatic region and the liquid will be accelerated in the heat pipe end. This can cause a reverse flow in the condenser section and negative shear force appears in the wick structure because the inertia forces dominate the viscous effects. Disregarding Forchheimer term in the momentum equation in the wick structure can be linked to the error due the neglecting of non-linear effects at high heat flux.

Figure 9 shows that the length of the adiabatic region does not affect significantly the pressure profile except a small increasing in the heat pipe end. This pressure gradient can cause unfavorable accumulation of liquid in the condenser section. So the presence of an adiabatic section improves the heat pipe operation and stability. Note that Faghri and Parvani [31] have reported that in the heat pipes with longer condenser lengths, flow reversal may occur in lower radial Reynolds numbers. Therefore, an optimal length L_c/L_c must be chosen.

Fig. 7 Axial vapor velocity streamlines at condenser corner end at 50,000 W/m² with and without adiabatic section for the two simulation case

Fig. 8 Velocity profiles in porous media with and without adiabatic region ($Q = 50,000 \text{ W/m}^2$, first kind b.c)

Fig. 9 Pressure and temperature profiles in porous media with and without adiabatic region ($Q = 50,000 \text{ W/m}^2$, first kind b.c)

Conclusion

In this paper, a proper code was developed in cylindrical coordinates which involve both heat pipe regions. The governed equation was solved using the finite volume method, and the Darcy–Brinkman–Forchheimer model was used to describe the flow in the porous media. The heat pipe behavior under normal and high heat flux conditions with two boundary conditions in the condenser region was investigated. Results show that at higher heat flux conditions, the liquid temperature within the heat pipe increases and a reverse flow is shown in the heat pipe condenser region end. As the conditions become more critical, a recirculation cell multiplies due to the radial velocity alteration. The heat pipe limit is then related to heat fluxes but also controlled by the temperature of the porous structure. The adiabatic region plays a major role in maintaining heat pipe stability especially when the heat pipe is exposed to high heat flux. Therefore, an optimal heat pipe's region lengths must be chosen to enhance the heat transfer within the heat pipe and perform the heat removal capability especially when the heat fluxes increase and the heat pipe boundary conditions change.

Nomenclature

C_p = specific heat, J/kg K
 Da = Darcy number
 g = acceleration, m/s²
 h_{conv} = film coefficient for convection heat transfer, W/m² K
 h_{fg} = latent heat, J/kg
 K = permeability, m²
 L = heat pipe length, m
 $\dot{m}$ = mass flow rate, kg/s
 P = pressure, Pa
 Pe = Peclet number
 Pr = Prandtl number
 Q = heat flux, W/m²
 R_v = gas constant for the vapor, J/kg K

r_s = radius of heat pipe container, m
 r = radial co-ordinates, m
 r_i = internal radius, m
 r_e = external radius, m
 Re = Reynolds number
 S_ϕ = source terms, dimensionless
 T = temperature, K
 u = axial velocity component, m/s
 v = radial velocity component, m/s
 v_i = radial injection velocity, m/s
 z = axial co-ordinates, m

Greek Symbols

α = thermal diffusivity, m²/s
 ε = porosity
 θ = heat pipe inclination, deg
 λ = thermal conductivity, W/mK
 μ = fluid absolute viscosity, kg/m s
 ρ = fluid density, kg/m³
 σ = surface tension, Pa m
 ν = kinematic viscosity, m²/s

Subscripts and Superscripts

a = ambient
ad = relative to adiabatic region
cond = relative to condenser region
evap = relative to evaporator
eff = effective
i = internal, injection, interface
l = liquid
s = solid
sat = saturation
v = vapor
0 = reference

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