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Thermal Hydraulic Design Features for the BNCT Application

Thomas E. Blue and Kambiz Vafai
Department of Mechanical Engineering

Department of Energy
Argonne, Illinois 60439

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INTRODUCTION

This project report is based on our investigations for thermal design of a heat pipe for removing generated heat resulting from Proton bombardments of a Lithium target for a BNCT application. Heat pipes are devices that are used for transporting heat very efficient way (Vafai and Sozen, 1990; Tien and Vafai, 1989). They have been widely used in heat transfer related applications for nearly 30 years. Theoretical and practical research aspects concerning the transport mechanism, the structure of the container, wick and materials used in the heat pipe have been well established on this subject. The fluid flow within a heat pipe dealing with the vapor flow in the core region and condensate flow in the wick is an important topic in most of these research studies. Most of the investigators have concentrated on the study of conventional heat pipes, which have a symmetrical heat source and sink. The finite difference method has been extensively employed to solve the governing differential equations in vapor phase (Tien and Rohani, 1974; Narayana, 1986; Faghri, 1986). But, the asymmetrical characteristics of a heat pipe have been mostly neglected (Vafai and Wang, 1992).

Intensive heat pipe research and development have been carried out for the past two decades, and heat pipes have been employed successfully in many applications, such as spacecraft thermal control, electronic-systems cooling and many commercial thermal devices. However in the present work because of the presence of an unconventional system it is required to make significant number of modifications in the design of the heat pipe. Therefore part of our investigation has and will concentrate on such design considerations.

A comprehensive review of the physical phenomena and applications of heat pipes is given by Tien (1975), Winter and Barsch (1971). Tien and Rohani (1974) also presented the numerical results displaying vapor pressure distributions and heat pipe performance for axisymmetrical heat pipes, and general trends for pressure drop in the evaporator section and pressure recovery in the condenser section. Further numerical results reported by Narayana (1986) including the pressure and the velocity distributions. He has pointed out that an increase in suction rate in the condenser section produces a vapor flow reversal. There have been several analytical and experimental studies of the heat pipe phenomena. For example, very simplified analytical and experimental studies for a one dimensional, steady state

problem on the heat pipe effect in porous media have been reported. In the model developed, gravity forces, capillarity and change of phase were considered. Leverett correlation relating capillary pressure to permeability, porosity and surface tension was also used in this study. Relative permeability correlations found by Fatt and Klikoff (1959) were also employed.

A full understanding of the multiphase transport through porous media (Vafai and Whitaker, 1986; Vafai and Sarkar, 1986; Vafai and Tien, 1989; Tien and Vafai, 1990; Vafai et. al., 1989; Vafai et. al. 1990) boundary and inertial effects (Ettfagh, Vafai and Kim, 1991) variable porosity effects (Vafai, 1986), local non equilibrium effects (Vafai and Sozen, 1990a; Sozen and Vafai, 1990; Vafai and Sozen, 1990b) and the porous fluid interfacial effects (Vafai and Thiyagaraja, 1987; Vafai and Kim, 1990; Vafai and Kim, 1990b) are needed for adopting the heat pipe for a particular application especially for non-conventional types of applications as is the case in the present proposal. In fact, some of these particular aspects for the non-conventional heat pipes were presented in a symposium volume on the non-conventional heat pipes at the ASME National Heat Transfer Conference in San Diego, California in August of 1992 (Vafai and Chang, 1992).

In the heat pipe research performed to-date, researchers have made some assumptions in modeling and analysis. One of the most important assumptions made is that the capillary porous wick is always saturated with liquid phase working fluid and vapor flows only in the core section during the operation of the heat pipe. Hence, evaluation of the effective thermal conductivity (or the heat removal capacity) and formulation of the capillary pressure were simplified. This assumption will be employed in the present investigation. In the present work the heat pipe will be designed such that it would be able to comfortably remove the heat generated as a result of the proton bombardments of the lithium target. Since 75 kW of energy falls upon a relatively small area of the lithium surface, we need to ensure that there will not be any flash evaporation. To satisfy the above referenced criteria all the assumptions will be based on conservative estimates with respect to the heat removal capabilities of the heat pipe.

The selection of the material which will be used in manufacturing of the heat pipe also needs some careful consideration. This is because the interference of the reflected (or outwardly diffused) neutrons (which are produced as a result of the interaction between the protons and the lithium) with certain materials produces

Gamma rays. The presence of the Gamma rays for our operation is of course unacceptable and must be avoided. For this reason heavy water (D_2O) is chosen as the operating fluid for the heat pipe and the heat pipe material itself is chosen to be aluminum

In our investigation, an integral analysis was employed to investigate the vapor and liquid flow in a flat plate heat pipe heated asymmetrically for removal of the 75 kW generated from the BNCT application. The flat plate heat pipe configuration will be used for removing the heat which is generated as a result of proton bombardments of the lithium target. The working fluid in the heat pipe occurs in two phases, namely liquid and vapor. The wick contains all the liquid phase and the vapor phase is mainly in the core region. Heat is applied by an external source at the evaporator section which vaporizes the working fluid in this section. This results in a pressure difference which drives the vapor to the condenser section where it condenses and releases latent heat of vaporization to a heat sink in the condenser section. Due to the vaporization of liquid in the evaporator, the liquid-vapor interface enters into the wick surface and hence capillary pressure is developed there. This capillary pressure causes the condensed liquid in the condenser to be pumped back to the evaporator again. Thus this cyclic mechanism takes place continuously. The flat plate heat pipe will be used as an effective heat sink as well as an additional moderator for the neutron production. It should be noted that our system performance is essentially independent of the orientation of the target assembly. That is the heat removal capabilities are basically the same regardless of whether the target assembly is in a horizontal, vertical, etc. position.

The results of our investigation so far has enabled us to correlate such diverse information as; the thickness of the wick, the diameter of the heat pipe, the wetting angle, the capillary radius, the surface tension, the latent heat of evaporation, the permeability and porosity of the chosen wick, the length of the heat pipe, and the viscosity and density of the two phases; with the *heat removal capabilities* of the heat pipe. Expressions for the pressure and velocity distributions are obtained and discussed in relation to our application to BNCT. The qualitative behavior for our results compare quite favorably with the results reported by other researchers. The present design clearly shows that it is possible to attain temperatures well below the melting temperature of the lithium in the BNCT application. This in turn prevents

any adverse pressure build up within the system. Finally, we establish the feasibility and safety of using the flat heat pipe for BNCT application.

ANALYSIS AND FORMULATION

The schematic diagram of the heat pipe for the first configuration that has been used for the present proposal is shown in Fig.1. It can be seen that the heat pipe is mounted symmetrically under the lithium target area. A cooling jacket is used around the outer periphery of the heat pipe for removing the dissipated heat. Based on our analysis the surface temperature of the heat pipe turns out to be low enough that the fluid inside the cooling jacket can pass through an appropriate cooling loop. This is quite advantageous since it will result in a closed loop system thus further reducing any possibility of an accidental contamination of the surroundings. The heat generated from the proton bombardment of the lithium is imposed on that region, and then is transferred to both sides of the heat pipe equally. The multiple independent vapor flow channels and the coordinate system for the flat plate heat pipe are shown in Figs. 2A and 2B. The generated heat causes the vaporization and subsequent pressurization of the liquid in the wick. Part of the vapor is condensed on the opposite surface of the evaporator. However, most of it is condensed downstream of the evaporator on both sides of the condenser region.

Obviously, the vapor flow in the heat pipe does not have a symmetrical velocity profile. The location of the maximum vapor velocity will be shifted towards the lower plate, as shown in Fig.2C, due to the vapor injection from the heating side of the upper plate. As the vapor flows downstream, the location of the maximum velocity will gradually shift towards the center line due to the presence of symmetrical cooling conditions. The vapor space is divided into several channels by the vertical wicks which transport liquid from the lower wick to the upper wick. Therefore, the vapor velocity along the longitudinal direction will be function of x , y and z directions. The condensate in the upper wick (section 2 of Fig. 2C), flows back directly to the evaporator along the wick by capillary force. However, in section 4 of the lower wick, the condensate will move directly along the horizontal wick towards section 3 of the lower wick, then laterally towards the vertical wicks and upwards through them towards the evaporator section. Therefore, within $0 \leq x \leq \phi l$ region (where ϕ will be established later), the liquid in the upper and lower wicks possesses two distinct velocity components, u_1 in the x direction,

Fig. 1. The Thermal System Analyzed and Designed for Phase I

(A) Dimensions of the Heat Pipe

(B) Coordinate System

Fig. 2. Details of the Heat Pipe Used in the Phase I Investigation.

Fig. 2C. Different regimes which are analyzed for the vapor and liquid phases in the x-y plane of the heat pipe

and w_l in the z direction, since, in this region, the upper wick (evaporator) gets part of the liquid supply from the lower wick. However, the liquid will have only one velocity component, u_l , along the x direction in the $\phi l \leq x \leq l$ section, since both upper and lower wicks act as condensers. That is over $\phi l \leq x \leq l$ region due to the symmetrical boundary conditions, there is no liquid exchange through the vertical wicks. In the heat pipe research performed to-date, researchers have made some assumptions in modeling and analysis. An important assumption made by various researchers is that the capillary porous wick is always saturated with liquid phase working fluid and vapor flows only in the core region during the operation of the heat pipe. Hence, evaluation of the effective thermal conductivity and formulation of the capillary pressure were simplified. This assumption will be employed in the present investigation. In order to obtain the integral solutions, we will make some common assumptions which are usually made in analyzing the heat pipes. The assumptions made in this work are:

1. Vapor and liquid flow are steady and laminar.
2. Transport properties for the vapor and liquid are taken as constants.
3. The vapor injection and suction rate are uniform in the evaporator and condenser sections.
4. The vapor velocity component in the z direction is negligible since there are no injection or suction on vertical wicks.

.c1.Vapor Phase Analysis

Once, the fluid flow characteristics within one of the channels is determined the fluid flow characteristics of the entire heat pipe can be constructed. In our analysis the position coordinates, velocities and pressure are non-dimensionalized by $\frac{h^2 U_{0v}}{v_v}$, h , b , U_{0v} , $\frac{v_v}{h}$ and $\rho_v U_{0v}^2$ in the vapor phase, and by $\frac{h_{wick}^2 U_{0l}}{v_l}$, U_d , $\rho_l U_{0l}^2$ in the liquid phase and the temperature is non-dimensionalized by h_{fg}/R factor. Where v_v and v_l are the kinematic viscosities, U_{0v} and U_d are the maximum vapor and liquid velocities, ρ_v and ρ_l are the vapor and liquid densities, h and b are the

height and half width of the vapor space of one vapor flow channel, h_w is the thickness of the wick and R is the ideal gas constant. Based on the numerical results given by Narayana (1986) and Bankston and Smith (1973), a parabolic profile will be used for vapor flow in the heat pipe. The three-dimensional vapor velocity distribution $u_v(x,y,z)$ in the x , y and z directions will be used in our analysis. The location of the maximum velocity will be shifted towards the lower plate due to the vapor injection from the heating side of the upper plate. Due to the asymmetry of velocity distribution in the x - y plane, the velocity profile will be divided into two parts in y direction, the lower part and the upper part, by the curve $y = f(x)$. The curve $y = f(x)$ is the location corresponding to zero shear stress for the velocity distribution on the x - y plane. After a lengthy analysis and the use of proper boundary conditions the velocity distribution for the upper part is derived as

$$u_v^+(x^+, y^+) = U_v^+(x^+) \left[2 \left(\frac{1-y^+}{1-f^+(x^+)} \right) - \left(\frac{1-y^+}{1-f^+(x^+)} \right)^2 \right] \quad (f^+(x^+) \leq y^+ \leq 1) \quad (1)$$

The above equation along with the integrated continuity equation

$$\int_0^1 \left(\frac{\partial u_v^+}{\partial x^+} + \frac{\partial v_v^+}{\partial y^+} \right) dy^+ = \int_0^{f^+(x^+)} \left(\frac{\partial u_v^+}{\partial x^+} + \frac{\partial v_v^+}{\partial y^+} \right) dy^+ + \int_{f^+(x^+)}^1 \left(\frac{\partial u_v^+}{\partial x^+} + \frac{\partial v_v^+}{\partial y^+} \right) dy^+ = 0 \quad (2)$$

are used to obtain $U_v^+(x^+)$ and ϕ

$$U_v^+(x^+) = \begin{cases} \frac{3(v_1^+ - v_2^+)}{2} x^+ & 0 \leq x^+ \leq \phi l^+ \\ -3v_2^+(x^+ - l^+) & \phi l^+ \leq x^+ \leq l^+ \end{cases} \quad (3)$$

$$\phi = \frac{2v_2^+}{v_1^+ + v_2^+} \quad (4)$$

The $f^+(x^+)$ will be determined by integrating the x^+ -momentum equation with respect to y^+ for the heat pipe. Since the longitudinal length is much larger than the transverse length in a heat pipe, the axial sheer stress will be neglected (Faghri, 1986). Following a cumbersome and lengthy analysis in which mass flow crossing

the interfaces as well as the vapor injection from the upper wick and suction from the lower wick are taken into account will result in the following set of equations

$$\frac{df^+(x^+)}{dx^+} = \begin{cases} \left\{ \left[\frac{3(1-\phi)Re_h}{(2-\phi)} f^+(x^+) + \frac{5}{f^+(x^+)} - \frac{\phi Re_h}{3(2-\phi)} \right] \frac{(1-\phi)Re_h}{(2-\phi)} x^+ \right. \\ \quad \left. + \frac{f^+(x^+)}{9} \frac{dp^+(x^+)}{dx^+} \right\} \left[\frac{(2-\phi)}{(1-\phi)Re_h x^+} \right]^2 & 0 \leq x^+ \leq \phi l^+ \\ \left\{ \left[-\frac{16\phi Re_h}{(2-\phi)} f^+(x^+) + \frac{10}{f^+(x^+)} \right] \frac{\phi Re_h}{(2-\phi)} (l^+ - x^+) \right. \\ \quad \left. + \frac{f^+(x^+)}{9} \frac{dp^+(x^+)}{dx^+} \right\} \left[\frac{(2-\phi)}{\phi Re_h (l^+ - x^+)} \right]^2 & \phi l^+ \leq x^+ \leq l^+ \end{cases} \quad (5)$$

The above set of equations will provide us with the complete velocity profile within the flat heat pipe assembly.

To obtain the pressure distribution in the vapor phase, the integrated x^+ -momentum equation within a channel bounded by porous wicks can be used to derive the following vapor pressure distribution after a very lengthy and detailed analysis:

$$\Delta p_v^+(x^+) = \begin{cases} -\frac{4(1-\phi)}{(2-\phi)} Re_h \left\{ \left[\frac{16(1-\phi)}{25(2-\phi)} Re_h + \frac{1}{2(h_b^+)^2} \right] (x^+)^2 + \int_0^{x^+} \frac{x^+}{f^+(x^+) [1 - f^+(x^+)]} dx^+ \right\} & 0 \leq x^+ \leq \phi l^+ \\ \Delta p(\phi l^+) - \frac{4\phi}{(2-\phi)} Re_h \left\{ \left[\frac{16}{25} \frac{\phi}{(2-\phi)} Re_h - \frac{1}{2(h_b^+)^2} (x^+ - l^+) - (\phi l^+ - l^+)^2 \right] \right. \\ \quad \left. - \int_0^{x^+} \frac{x^+ - l^+}{f^+(x^+) [1 - f^+(x^+)]} dx^+ \right\} & \phi l^+ \leq x^+ \leq l^+ \end{cases} \quad (6)$$

The solutions for $\Delta p_v^+(x^+)$ and $f^+(x^+)$, are then obtained by solving equations (5) and (6).

Liquid Phase Analysis

Within the thin porous wick, Darcy's law will be applied. The dimensionless Darcy's law is

$$\frac{\partial p_l^+}{\partial x^+} = \frac{1}{K^+} u_l^+ \quad (7)$$

where $K^+ = \frac{K}{h_w^2}$. It should be noted that based on the works of Vafai (1984) and Tien and Vafai (1989) the boundary and inertia effects will be neglected in within the porous wick. Utilizing the lower and upper wick mass conservation equations and incorporating both vapor injection and suction velocities subject to the appropriate boundary conditions will result the following liquid phase velocity distribution

$$u_l^+(x^+) = \begin{cases} \frac{h_w^+ \mu^+ (v_1^+ - v_2^+)}{2} x^+ & 0 \leq x^+ \leq \phi l^+ \\ h_w^+ \mu^+ v_2^+ (l^+ - x^+) & \phi l^+ \leq x^+ \leq l^+ \end{cases} \quad (8)$$

Using the above equation we can then derive the pressure distribution in the wick in the x^+ direction as

$$\Delta p_l^+(x^+) = \begin{cases} \Delta p_v^+(l^+) - \frac{h_w^+ \mu^+ (1 - \phi) Re_h}{2(2 - \phi) K^+} \left\{ \phi(1 - \phi)(l^+)^2 + [(\phi l^+)^2 - (x^+)^2] \right\} & 0 \leq x^+ \leq \phi l^+ \\ \Delta p_v^+(l^+) - \frac{h_w^+ \mu^+ \phi Re_h}{2(2 - \phi) K^+} (l^+ - x^+)^2 & \phi l^+ \leq x^+ \leq l^+ \end{cases} \quad (9)$$

Once the vapor pressure distribution is found, the vapor temperature distribution can be specified by using the Clausius-Clapeyron equation. In this investigation, the compressibility effects on the vapor flow were neglected. It should be noted that also radiative effects were neglected (Vafai and Ettefagh, 1988).

Operational Criterion

In general, the operational limits which need to be satisfied for a heat pipe under steady state and low temperature situations (Tien, 1975) are usually based on the boiling and capillary limit criteria. First, the capillary operational limit is

established. To satisfy the capillary limit, the maximum capillary pressure head $(\Delta p_{\text{cap}})_{\text{max}}$ must be greater than the total vapor, liquid and gravitational pressure drops (Tien, 1975). That is

$$(\Delta p_{\text{cap}})_{\text{max}} \geq \Delta p_{\text{total}} \quad (10)$$

The total pressure head is presented as

$$\Delta p_{\text{total}} = \Delta p_v + \Delta p_l = p_v(0) - p_l(0) \quad (11)$$

in which the gravitational head Δp_g has been neglected for the horizontal heat pipe.

Based on (Tien, 1975), Dunn and Reay (1982), Chisholm (1971) the wetting angle between the liquid and wick in the evaporator section is $\theta_e = 0$ whereas in the condenser section this wetting angle is $\theta_e = \pi/2$. Therefore the maximum capillary pressure head of Δp_{cap} can be written as

$$(\Delta p_{\text{cap}})_{\text{max}} = 2\sigma_l / r_w \quad (12)$$

where σ_l denotes the surface tension of the liquid in the wick, and r_w is the porous radius of the wick. It will be shown later that the operational criterion for the heat pipe is indeed established. In what follows it will be shown that the boiling limit criterion is also satisfied. For a water heat pipe near atmospheric pressure, the critical heat flux for boiling limit is of the order of 10^8 W/m^2 . However, for our case, the critical heat flux is (Tien, 1975)

$$q_{\text{boil}} = \frac{h_{fg} p_0}{\sqrt{2\pi RT}} \exp\left(-\frac{h_{fg}}{RT}\right) = 1.06 \times 10^7 \text{ W/m}^2$$

which is almost an order of magnitude higher than the maximum heat flux imposed on the heat pipe. Therefore, both the capillary and boiling limit criteria are satisfied for high heat flux situations such as the BNCT application.

RESULTS AND DISCUSSION

The results presented here are based on a flat heat pipe with heavy water as working fluid. The pipe dimensions based on our application are chosen as: $l = 0.392$ m, $b = 0.0125$ m, and $h = 0.0254$ m. The evaporator section, $l_e = 0.09$ m, is only on the top surface of the heat pipe, which based on the chosen dimensions has an area of $A_e = 0.041$ m². The rest of the heat pipe's external area acts as the condenser section with an area of $A_c = 0.316$ m². The heat flux at the evaporator section of the heat pipe can then be calculated as

$$q_e = Q/A_e = 1.8293 \times 10^6 \text{ W/m}^2 = 182.93 \text{ W/cm}^2$$

Since the light water will hamper the traverse of neutrons, heavy water (D₂O), which has almost the same thermophysical properties except having a larger density than light water, is chosen as the working fluid for the heat pipe. The material for the container and the wick is chosen to be aluminum. The heat pipe must be designed such that the lithium target will be solid at all times. Since the melting temperature for lithium is 180 °C, the target will remain solid if the vapor temperature is controlled at $T_{\text{vapor}} = 80$ °C. This can be seen through examination of the corresponding lithium temperature

$$T_{\text{Li}} = T_{\text{vapor}} + \Delta T_{\text{wick}} + \Delta T_{\text{wall}} + \Delta T_{\text{Li}} < 180 \text{ °C} \quad (13)$$

where

$$\Delta T_{\text{wick}} = q_e(h_w/k_{\text{wick}}) = 32.9 \text{ °C}$$

$$\Delta T_{\text{well}} = q_c(h_{\text{well}}/k_{\text{well}}) = 15.5 \text{ °C}$$

$$\Delta T_{\text{Li}} = q_e(h_{\text{Li}}/k_{\text{Li}}) = 42.7 \text{ °C}$$

and where $h_{\text{Li}} = 1 \times 10^{-3}$ m, $h_{\text{wall}} = 2 \times 10^{-3}$ m and $h_w = 2 \times 10^{-3}$ m are the thicknesses of lithium target, the container wall and the wick respectively, and $k_{\text{Li}} = 42.8$ W/m°C, $k_{\text{wall}} = 236$ W/m°C and $k_{\text{wick}} = 94.65$ W/m°C are thermal conductivities of the lithium, aluminium and the wick respectively. Based on the above calculations, the vapor temperature should be less than 88.9 °C to satisfy equation (13). Therefore, the operating vapor temperature is chosen to be 80 °C to ensure that there will be no phase change in the lithium layer.

The thermophysical properties of the heavy water used in our calculations (at $T_v = 80\text{ }^\circ\text{C}$) are: $h_{fg} = 2128\text{ kJ/kg}$, $\mu_l = 1.1876 \times 10^{-5}\text{ Ns/m}^2$, $\mu_v = 41.6 \times 10^{-5}\text{ Ns/m}^2$, $\rho_v = 0.3055\text{ kg/m}^3$ and $\rho_l = 1078.3\text{ kg/m}^3$. The saturated vapor pressure corresponding to $T_v = 80\text{ }^\circ\text{C}$ is $p_{\text{sat}} = 4.44 \times 10^4\text{ Pa}$. The imposed energy rate based on the BNCT application is $Q = 75\text{ kW}$, which will result an average injection velocity in the evaporator section

$$v_1 = \frac{Q}{A_e h_{fg} \rho_v} = 2.813\text{ m/s}$$

This will correspond to an injection Reynolds number of $Re_h = 1460$ for $Q = 75\text{ kW}$. As seen from equations (9), (11), (13), (18), (20), the $f^+(x^+)$ and non-dimensional velocity and pressure distributions for both the vapor and liquid phases are functions of two pertinent parameters. These are: Re_h , which represents the input power, and ϕ , which represents the geometric ratio of the length of evaporator to the entire length of the heat pipe. Equations (5) and (6) were solved by using a standard Runge-Kutta method. The numerical results for functions $U_v^+(x^+)$, $f^+(x^+)$, vapor velocity distributions $u_v^+(x^+, y^+, z^+)$ at different z^+ locations, average vapor and liquid pressure distributions and the average vapor temperature distribution are shown on Figures 3 to 7.

Figure 3 demonstrates the variation of $U_v^+(x^+)$ along the x^+ axis. It can be seen that $U_v^+(x^+)$ increases in the $0 \leq x^+ \leq \phi l^+$ region corresponding to the evaporator section and decreases in the $\phi l^+ \leq x^+ \leq l^+$ region corresponding to the condenser section due to the vapor injection (increasing mass) and suction (decreasing mass) over the corresponding regions. This variation of $U_v^+(x^+)$ is linear because of uniform vapor injection and suction velocities. This behavior is similar to the results reported by Bankston and Smith (1973).

Figure 4 illustrates the functional distribution for $f^+(x^+)$, which represents the location for the maximum value of $u_v^+(x^+, y^+)$, at different injection Reynolds numbers. It can be seen that right after $x^+ > 0$, the maximum velocity is profoundly shifted towards the cooling side. As expected for larger Re_h 's, the maximum vapor velocity will be shifted more prominently towards the cooling side, and it also takes

Fig. 3. Variations of U_v^+ vs x^+/l^+

Fig. 4. Variations of $f^+(x^+)$ vs Re_h

a longer length before $f^+(x^+)$ approaches the center line. Fig. 5(a) clearly shows that for smaller Reynolds numbers, the symmetrical transition of the velocity profile will occur earlier. Fig. 5(b) shows the $x^+ - y^+$ velocity distribution at different lateral locations for $Re_h = 1460$.

To obtain a better feel for the operational criterion, the solutions for the pressure and temperature distributions are presented dimensionally. Figure 6 presents the vapor and liquid pressure distributions along the heat pipe at different Reynolds numbers. These profiles show the effects of the pressure head loss in the evaporator region due to vapor velocity acceleration in that region, and the pressure head recovery in the condenser region due to the deceleration of the vapor velocity in that region. In the liquid phase, there is only pressure head loss, since the flow is dominated by the bulk frictional resistance from the porous matrix. As expected, the smaller the Reynolds number, the smaller the pressure head loss. The qualitative behavior of the overall pressure distribution along the heat pipe is quite close to results presented by Tien and Rohani (1974) and Winter and Barsch (1971). Figure 7 demonstrates the variations for the mean vapor temperature along the longitudinal direction at different injection Reynolds numbers. It can be seen, the temperature is quite uniform ($DT \ll 1$ °C), even for the highest input power ($Re_h = 1460$). This further demonstrates the successful operation of the heat pipe Winter and Barsch (1971).

Finally, the capillary operational criterion for the heat pipe can be contrasted from the numerical results given on Fig. 6. The total pressure drop in the heat pipe is

$$\Delta p_{\text{total}} = 133 \text{ Pa}$$

corresponding to $Re_h = 1460$ ($Q = 75 \text{ kW}$). Using equation (12) results

$$(\Delta p_{\text{cap}})_{\text{max}} = 2\sigma_l/r_w = 545 \text{ Pa}$$

Obviously, $(\Delta p_{\text{cap}})_{\text{max}} > \Delta p_{\text{total}}$. Thus the capillary operational criterion for the heat pipe under the design conditions dictated by the BNCT application are also satisfied.

Fig. 5(a). Effects of the injection Reynolds number
on the velocity distributions ($z^* = 0$)

Fig. 5(b). The velocity distributions at different lateral locations for $Re_h = 1460$

Fig. 6. Vapor and liquid pressure distributions along the heat pipe

Fig. 7. Vapor temperature profiles for different injection Reynolds numbers

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