A Two Dimensional Steady State Roll Heat Pipe Analyses For Heat Exchanger Applications

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ABSTRACT

Roll heat pipe is a novel type of heat pipe which is different from conventional or concentric annular heat pipes because heat is transferred radially from the inner pipe (evaporator region) to the outer pipe (condenser region). The condensed liquid flows back from condenser to evaporator by bridge wicks. In this paper, a two dimensional steady state analysis of the performance of the roll heat pipe is reported. The effect of boundary conditions in on the heat pipe thermal resistance is investigated. Result reveal that the RHP thermal resistance decrease with the heat input and the RHP thermal performance is controlled by the outer surface temperature. Result also revealed that a higher grade of isothermalisation can be obtained with heat input location management.

Keywords: Steady states, Thermal resistance, Boundary conditions, Localized heat

1. INTRODUCTION

Heat Pipe Heat Exchangers have been used in various thermal systems for improving their effectiveness. There are used in different technological processes due to their advantages (high effective thermal conductivity, low cost, low volume, easy to manufacture, etc.).

Since 1970 many heat pipe heat exchangers are used and manufactured for many applications such as air conditioning or cooling [1-3], heat recovery in hospital, laboratories and building to ensure the energy saving and environmental protection [4] or waste heat recovery [5-8] to reduce primary energy consumption, thus reducing carbon dioxide production and cost. A review of Heat pipes in modern heat exchangers and their recent applications are summered by Vasilev [9].

Roll heat pipe can be an effective device in various heat transfer, heat recovery, and heat exchanger applications. It can be designed and manufactured in a wide range of sizes and shapes and can have a wide temperature range of operation.

In this paper, a theoretical study has been performed to investigate the thermal performance of roll heat pipe heat exchanger in different working conditions. Based on the transient analysis [10] and the following the results carried out by Jalilvand [11]. A two dimensional steady state analysis is performed to examine the effect of different outer boundary conditions on the RHP thermal response for future heat exchanger applications. The two dimensional steady state energy equations are solved using the finite difference method.

2. MODEL DESCRIPTION

A roll heat pipe of 30 mm diameter and 0.5 m length has been investigated. We assume that the circumferential temperature is neglected, the porous media is supposed to be saturated with the working fluid (distilled water). In the vapor region, it is generally accepted in the literature [12] that the temperature "T_{sat}" is constant. We did not make this assumption in this model. Instead, we take into account the dependence of T_{sat} and the vapor temperature is constant and equal to that of T_{sat} on time and space (i.e $=T_{sat} = T_{sa}(z, r, t)$). In the Figure 1, a uniform heat "Q_{in}" is applied to the inner heat pipe surface. "Q_{out}" is the heat dissipation from the outer heat pipe surface into the atmosphere.

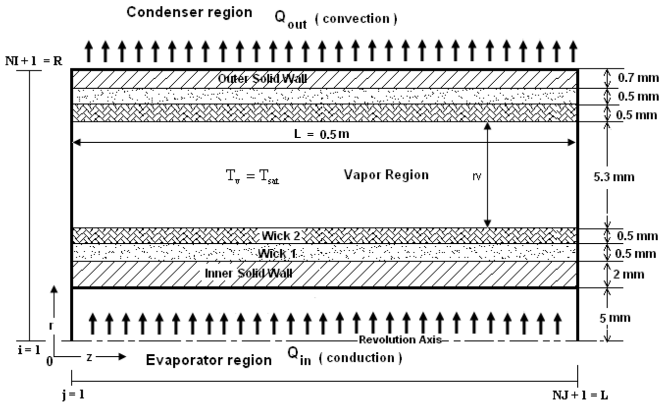

Figure 1: the cross section view of RHP

The two-dimensional steady state heat conduction equation in cylindrical coordinate within the RHP is given as follow:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\, k\frac{\partial T}{\partial r}\right) + k\frac{\partial^2 T}{\partial z^2} + S(r, z, T) = 0 \qquad (1)$$

Where $S(r, z, T)$ is the source term which can be presented as the sum of heat generation term and the fin term.

Eqn. (1) can be rewritten as:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(r\,k\frac{\partial T}{\partial r}\right)+k\frac{\partial^2 T}{\partial z^2}+S_c(r,z)-S_P(r,z,T)T=0 \tag{2}$$

.

Eqn. (2) is integrated over the area of control volume $\Delta V = rdrdz$ as shown in Fig.2.

$$r_n k_n \Delta z\frac{T_N-T_P}{\delta r_n}-r_s k_s \Delta z\frac{T_P-T_S}{\delta r_s}+\ r_p k_e \Delta r\frac{T_E-T_P}{\delta z_e}-$$

$$r_p k_w \Delta r\frac{T_P-T_W}{\delta z_w}+S_C\Delta V-S_P\Delta V T_P=0 \tag{3}$$

Since in radial direction different materials are presented with different thermal conductivities, the effective thermal conductivity at control volume between two parts (for example between N and P) can be written as follow (**Erreur ! Source du renvoi introuvable.**):

$$\delta r_n/k_{eff}=\left(SNP/k_n+\ SPS/k_p\right) \tag{4}$$

Eqn. (2) is then can be written in compact form as follow:

$$a_P T_P - a_N T_N - a_S T_S - a_E T_E - a_W T_W = b \tag{4}$$

With:

$$\begin{cases} a_N = \frac{r_n k_n \Delta z}{\delta r_n} = \ r_n \Delta z\left(SNP/k_n + SPS/k_p\right) \\ a_S = \frac{r_s k_s \Delta z}{\delta r_s} = \ r_s \Delta z\left(SNP/k_p + SPS/k_s\right) \\ a_E = \frac{r_p k_e \Delta r}{\delta z_e} \\ a_W = \frac{r_p k_w \Delta r}{\delta z_w} \\ a_P = a_N + a_s + a_E + a_W + S_P\Delta V \\ b = S_c \Delta V \end{cases} \tag{5}$$

In general, Eqn. (4) can be thought of as having the form:

$$a_P T_P - \sum a_{nb} T_{nb} = b \tag{6}$$

Where T_P represents the temperature of the central grid point, subscript "nb" denotes a neighbor, and the summation is taken over all neighboring cells.

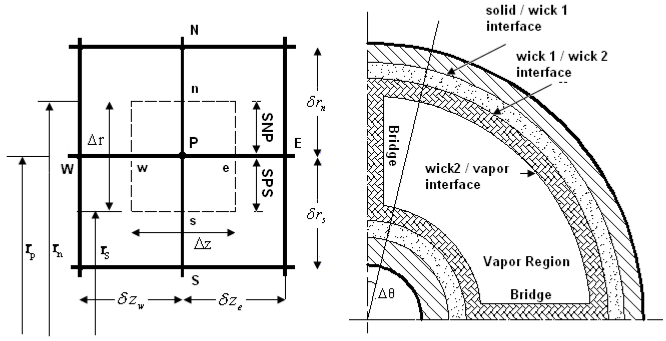

Figure 2: control volume for RHP simulation and ¼ RHP radial cross section configurations.

2.1 Boundary Conditions

The boundary conditions are summered in Table 1.

k_{w1}; k_{w2} are the effective thermal conductivity of sintered powder wick $(166.5\,m/mK)$ and mesh screen wick "50 mesh/inch"$(1.23\ W/mK)$ respectively [10].

Solid-wick1 interface: $k_{w1}\frac{\partial T_{w1}}{\partial r}=k_s\frac{\partial T_s}{\partial r}$	Wick2-vapor interface: $T_{w2}=T_{sat}$
Wick1-wick2 interface: $k_{w1}\frac{\partial T_{w1}}{\partial r}=k_{w2}\frac{\partial T_{w2}}{\partial r}$	Condenser region: 3 kind boundary conditions
Heat pipe ends: $z=0;\ z=L$ $\frac{\partial T}{\partial r}=0$	Evaporator region: $-k_s\frac{\partial T_s}{\partial r}=q_{in}$

Table 1: RHP boundary conditions

Three outer boundary conditions were tested in the condenser region: constant wall temperature (1st kind), constant heat fluxes (2nd kind) and constant convective coefficient (3rd kind) at $T_\infty = 25°C$.

$$\begin{cases} T = T_R & (1^{st} kind) \\ -k_s\frac{\partial T_s}{\partial r} = q_{out} & (2^{nd} kind) \\ -k_s\frac{\partial T_s}{\partial r} = h(T_\infty - T_s) & (3^{rd} kind) \end{cases} \tag{7}$$

Due to two phase flow characteristics, constant heat input can be applied. The formulation given in Eqn. (2) is valid in the internal volume control. The zero control volumes at the boundaries are used to incorporate the boundary conditions. For example, the boundary condition at the outer surface of the cylinder, for the specified heat flux "q_{out}" and convective conditions " h " can be written as:

$$-k\frac{\partial T}{\partial r}=q_{out}+h(T_\infty - T) \tag{8}$$

Eqn. (8) is expressed in terms of coefficients in its discretized form as (see Fig.1):

$$a_{NI+1,j}T_{NI+1,j}-a_{NI,j}T_{NI,j}=b_{NI+1,j} \tag{9}$$

Where

$$\begin{cases} a_{NI,j} = \frac{k_{NI,j}}{\frac{\Delta r_{NI,j}}{2}} \\ a_{NI+1,j} = \ a_{NI,j} + h \\ b_{NI+1,j} = q_{out} + hT_\infty \end{cases} \tag{10}$$

With the 1st kind boundary condition where a constant temperature $T = T_R$ is a specified, the Eqn. (9) can be used with the following coefficients:

$$a_{NI,j}=0;\ a_{NI+1,j}=1;\ b_{NI+1,j}=T_R \tag{11}$$

Note that, the other boundary conditions can be derived in the same manner.

The solution of the algebraic equations can be solved easily using the Tridiagonal Matrix Algorithm Method (TDMA) and a computer code using FORTRAN language that implements the finite difference method is developed.

4. RESULT AND DISCUSSION

The numerical study during the transient analysis is done with a number of grid points of 210 in radial direction after a sensibility analysis which correspond to one point each 1mm in the wick and the solid structures; and about 0.245 mm in vapor space. In the axial direction a number of grid points of 500 is used correspond to 1mm distance. Based on safety design of RHP, a maximum temperature of 200°C is allowable within the heat pipes which correspond to saturation pressure of 0.5MPa.

For verification of current model, the results of axial heat pipe temperature have been compared with the experimental results given by jilalvand [11]. The same temperature profile is obtained as seen in Figure 3 and a good agreement between the current numerical model and the experiment results is obtained.

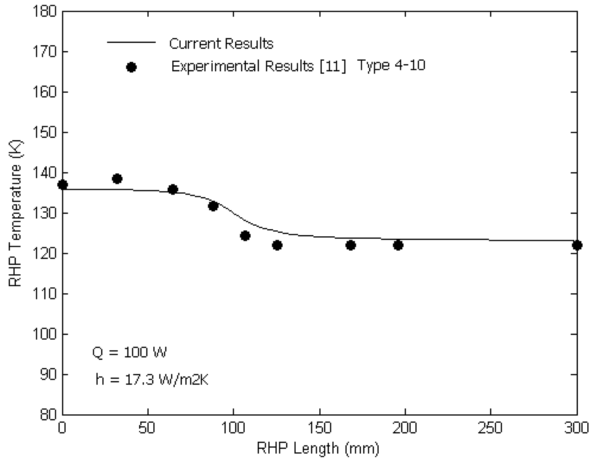

Figure 3: Axial temperature distribution on the outer surface of RHP compared with the experimental results given by [11].

Another comparison in made in Figure 4 with the experimental results on RHP type 4-20 carried out by Jalivand [11] for different heat input. A good agreement is found between the current numerical model and the experiments results. The figure shows that the heat pipe thermal resistance is reduced exponentially with the increase of heat supplied to the heat pipe evaporator section. Note that, the amount of fill charge, the heater length and the amount of heat input are important experimental parameters on temperature distribution of the outer surface of RHP as mentioned by [11].

Figure 5 shows the effect of the heat transfer coefficient on the thermal heat pipe resistance for different heat input. The thermal resistance decrease as the heat transfer increase due the decrease in surface temperature of the condenser region. For lower heat input, the thermal resistance increase due the lower surface area on the heat pipe used. A study pointing on the effect of wicks structures specific characterizations, heat pipe limits and the effect of the surface area of heat transfer, which increase the size of the heat exchanger but can be a solution to enhance the effectiveness of the sensible heat exchangers, is recommended.

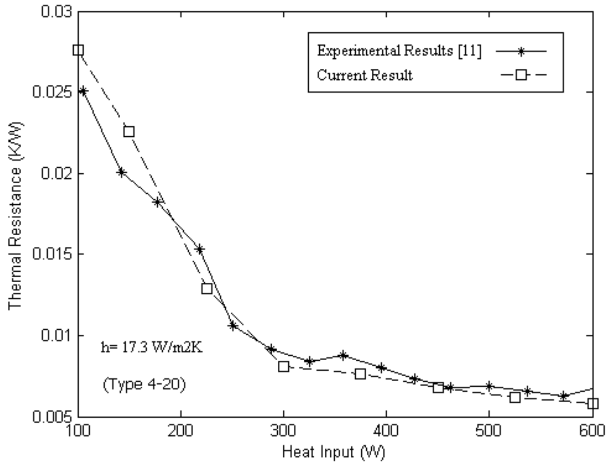

Figure 4: Variation of thermal resistance with heat input compared with the experimental results given by [11].

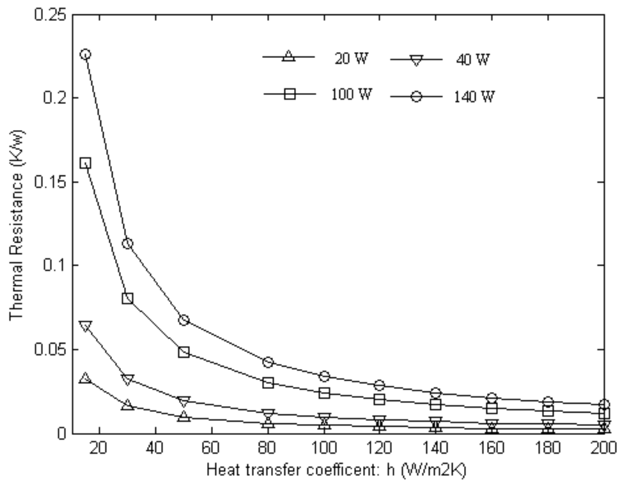

Figure 5: Effect of heat transfer coefficient on RHP thermal resistance

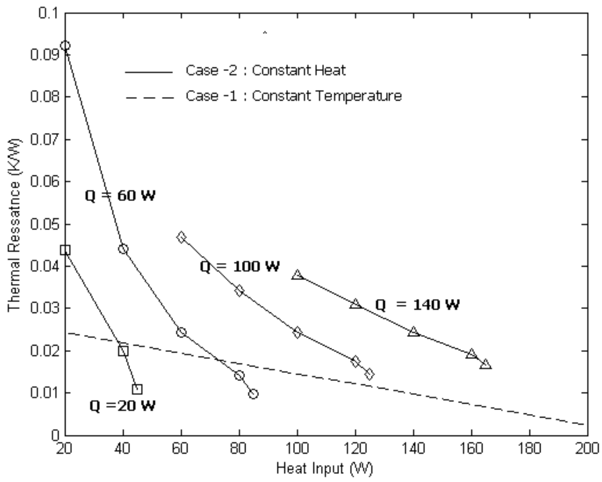

Figure 6: Variation of RHP thermal resistance with heat input for two boundary conditions type.

The Figure 6 shows the effect of 1st kind and the 2nd kind boundary conditions on the heat pipe thermal resistance. The heat pipe thermal resistance is independent of the constant temperature exposed at the condenser outer surface. Constant temperature boundary condition may be used to describe bodies with very high heat conductivity. Indeed, when heat load increase, one can see that the thermal resistance becomes very low.

For constant heat flux boundary condition, the thermal resistance decreases when the heat extracted increases and depend on the heat input at the inner RHP surface. Indeed, if not properly heat can be extracted; critical temperatures and higher thermal resistance can be reached which can affect the heat pipe normal operation.

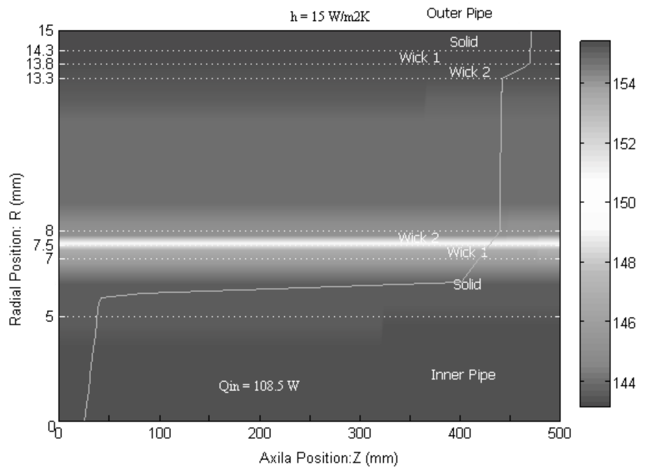

Figure 7: Temperature distribution within the RHP

Figure 7 shows the two dimensional temperature distributions within the heat pipe when exposed to a constant heat input of 108.5W in the inner surface area. Based on the transient analysis given in [10], a condenser temperature of about 142°C must be obtained at the condenser surface. We can see that the vapor temperature is nearly constant which correspond to its saturation temperature. The results illustrate well the function of the heat pipe and the circulation of the working fluid inside the heat pipe regions.

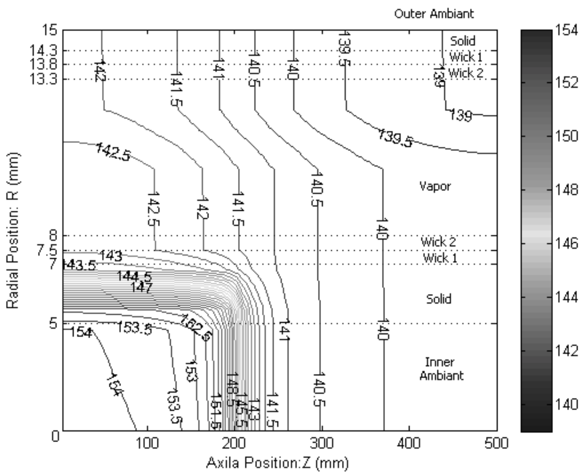

Figure 8: Temperature distribution within the RHP for localized heat input (at 0.2m).

The effect of localized heat at the evaporator region (0.2m) is shown in Figure 8. For the same external conditions as in Figure 7, the outer surface temperature is slowly decreased at the condenser end. This is due to the higher thermal conductivity of the heat pipe which can transfer rapidly all the heat load even in localized region. We can conclude that even with localized heat input a nearly constant temperature can be obtained in the outer heat pipe surface. This is the main characteristic of such heat pipe. The area of heat exchangers where RHP heat exchangers are used show a higher grade of isothermalisation in comparison to conventional tubular heat exchanger

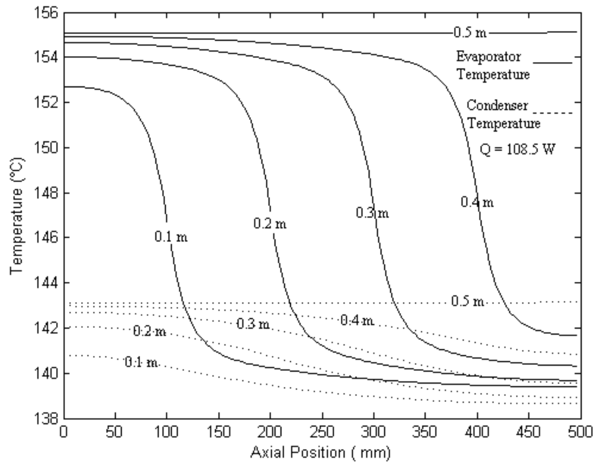

Figure 9: Effect of localized heat input on the evaporator and the condenser surface temperatures.

Figure 9 shows for different heat input locations, the axial evaporator and condenser surface temperatures for the same heat input. Decease in outer surface temperature in the condenser region can be shown especially in the end corner as mentioned before. .

Figure 10 plots also the tow dimensional heat pipe temperatures were two localized heat sources are added. The first is Q1 = 108.5 W along a distance of 0.3 m, and the second Q2 = 50W along a distance of 0.1 m. One can see that the second heat source has no significant effect on the heat pipe outer surface temperature. The localized heat input in the evaporator region can controls the outer surface temperature along the heat pipe axial position. Indeed, thermal control is a generic need of any dissipation system.

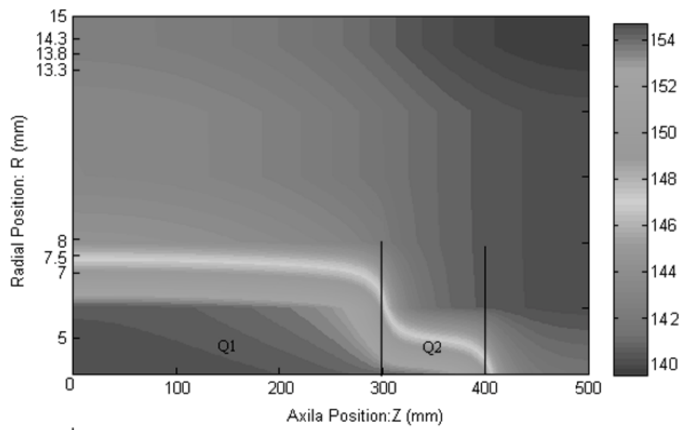

Figure 10: Temperature distribution within the RHP for two localized heat input.

5. CONCLUSION

A two dimensional steady state analysis of the roll heat pipe is performed in this paper considering the effect of three boundary conditions kind on the heat pipe thermal resistance. The model is compared first with the experimental results carried out by Jalilvand [11]. A good agreement in obtained. Result reveal that the thermal resistance in controlled by the outer surface conditions and it decreases as the heat load increases. The heat pipe operation is performed when heat is extracted faster from the condenser region. Also, the results illustrate well the function of the heat pipe for uniform and localized heat input in the evaporator region and thermal management can be controlled by the heat input location with precise manner and wall temperature can be predicted.

Due their higher thermal efficiencies thermal management, roll heat pies can be used in large scale as heat exchanger for future terrestrial or aerospace applications.

The effects of the various parameters such as wick characterizations, heat pipe diameter and length, and property variations can be considered in the future studies. Making improvements in performance of heat exchanger system is possible by experimental testing.

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