Roll Heat Pipe Transient Analysis

Taoufik BRAHIM*, Abdelmajid JEMNI*

* Université de Monastir, Ecole Nationale d'Ingénieurs de Monastir, Laboratoire d'Etudes des Systèmes Thermique et Energétique LESTE, Avenue Ibn Jazzar 5019 Monastir, TUNISIE
Email : enim_leste@yahoo.fr, taoufik.brahim@yahoo.fr*

ABSTRACT

A transient analysis has been carried out in detail using the finite difference method for the roll heat pipe wish is firstly studied by Jalilvand [1] in photocopy machines. The objective of this study is to better understand this type of heat pipes for further heat recovery applications or other designs processes that are essentially transient. The numerical results shows that the roll heat pipe perform better than the annular heat pipe of same dimension and the condenser region is essentially controlled by the heat Input and the convective heat transfer coefficient.
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1. INTRODUCTION

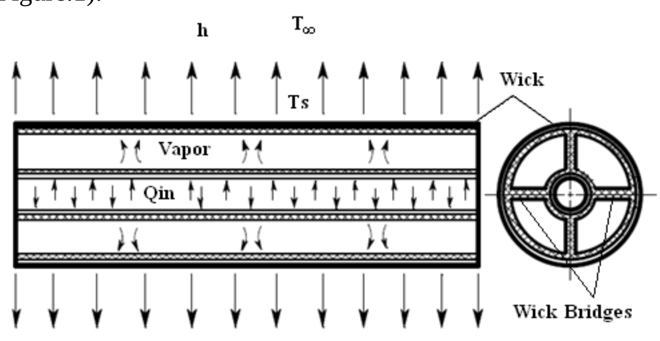
A heat pipe is a device that allows very high rates of heat transfer over medium distances with low temperature differences. It is light weight and responds quickly to changes in heat load. A heat pipe heat exchanger composed of an array of individual heat pipes installed in a metal casing are constructed to provide a simple, efficient and compact method of air-to-air heat recovery. They resemble conventional heating and cooling coils witch can be easily implemented inside sorption machines, transport for heating, cooling, air-conditioning, heat recovery and use of waste heat and many other types of heat transfer devices [2–5]. A review of Heat pipes in modern heat exchangers are summered by Vasilev [6].

The heat pipe analysed in this paper is named Roll Heat Pipe. This type was firstly fabricated and tested as application in the fusing unit in the photocopy machine by Jalilvand [1]. Roll heat pipe has the possibility to transfer a large amount of heat using a very short pipe. At 7th International Heat Pipe Symposium, Kim et al [7] presented a work entitled "Development of Heat Pipe Heating Roller in Laser Printer". This device was claimed as an innovative product of which printing quality was improved dramatically.
Roll Heat pipes resemble to concentric heat pipe but they consists of two wicks structure attached to the inner wall. Wicks at different diameters are connected to each other by bridge wicks.

Boo et al. [8], studied the thermal performance of Concentric Annular Heat Pipe (CAHP) with a heat source inserted in the inner pipe with the ability to transfer heat in radial direction.
Faghri et al. [9] observed the overall performance of the concentric annular heat pipe with an emphasis on increase in the heat transport capacity as compared to the conventional heat pipe.
A.N Borujerdi and Layeghi [10] analyzed the vapor flow in concentric annular heat pipe. They found the pressure distribution for different radial Reynolds numbers and they found that concentric heat pipe perform better than conventional heat pipe.
We can conclude, as our knowledge, little attention has been paid to study roll heat pipes
One of the main purposes of this work is to study RHP time response behavior for further heat recovery applications due to its annular shape or for future designs processes that are essentially transient. The analysis presented used the finite difference method with the discretization detail analysis.

2. MODEL DESCRIPTION

Because of its cylindrical shape, the RHP consists of two concentric annular pipes with wick structure attached to the inner wall with four bridges at the end sides of the heat pipe. The two pipes are of unequal diameter as RHP container (see Figure.1).



. Fig.1: The cross section view of RHP

The case investigated is when a constant heat flux in applied in the inner tube and a free convection in the outer tube. We supposed that the heat pipe has a initial ambient temperature T^0.

Because of its simplicity and quick predictions, the Lumped Capacitance Model has been employed. If the condenser is only cooled by convection, Newton's Law of Cooling can model the output power, and the lumped parameter model reduces to:

$$Q(t)-hA_s\left(T(t)-T_\infty\right)=\left(\rho\, C_p\, V\right)\frac{dT}{dt} \qquad (1)$$

V: Is the entire volume of the heat pipe. The total thermal capacity is defined as the sum of the heat capacities of the solid and liquid components of the heat pipe. The effective heat capacity of the liquid saturated wick is modelled accounting for both the liquid in the wick and the wick structure itself.

$$(\rho C_P)^1_{eff} = \varepsilon_1 (\rho C_P)_l + (1-\varepsilon_1)(\rho C_P)^1_{ws} \qquad (2)$$

$$(\rho C_P)^2_{eff} = \varepsilon_2 (\rho C_P)_l + (1-\varepsilon_2)(\rho C_P)^2_{ws} \qquad (2a)$$

Then

$$(\rho C_P) = (\rho C_P)_s + (\rho C_P)^1_{eff} + (\rho C_P)^2_{eff} \qquad (2b)$$

Subscripts 1 and 2 refer to wick1 and wick2 respectively and in the vapor zone we supposed that the vapor temperature is equal to saturation temperature corresponds to its saturated pressure. The wick structure is saturated with working fluid and due to its small thickness; heat conduction through the wick bridges is ignored.

The boundary conditions are summered Table 1.

Tab 1. Model Boundary Conditions
In the inner heat pipe where heat is transmitted to heat pipe
$-\lambda_s A_{in}\dfrac{\partial T_s}{\partial r} = Q_{in}$
In the outer wall surface where convection produced
$-\lambda_s \dfrac{\partial T_s}{\partial r} = h(T_s - T_\infty)$
In the interface between the wick 1 and the solid surface
$\lambda_s \dfrac{\partial T_s}{\partial r} = \lambda_{w1}\dfrac{\partial T_{w1}}{\partial r}$
In the interface between the wick 1 and the wick2
$-\lambda_{w2}\dfrac{\partial T_{w2}}{\partial r} = -\lambda_{w1}\dfrac{\partial T_{w1}}{\partial r}$

Where $A_{in} = 2\Pi\, r_{in} L_{tot}$ is the inner heat pipe's area and λ_{w1} is the effective thermal conductivity of the sintered powder wick and λ_{w2} is the effective thermal conductivity of the screen mesh wick saturated with liquid. These thermal conductivities are calculated respectively as Faghri [13]:

$$\lambda_{eff1} = \lambda_s \frac{(2+\lambda_l/\lambda_s) - 2\,\varepsilon_1 (1-\lambda_l/\lambda_s)}{(2+\lambda_l/\lambda_s) + \varepsilon_1 (1-\lambda_l/\lambda_s)} \qquad (3a)$$

$$\lambda_{eff2} = \lambda_l \frac{(\lambda_l + \lambda_s) - (1-\varepsilon_2)\,(\lambda_l - \lambda_s)}{(\lambda_l + \lambda_s) + (1-\varepsilon_2)\,(\lambda_l - \lambda_s)} \qquad (3b)$$

In the Equations (3), ε is the porosity of wick and subscript s and subscript l mean solid material and liquid working fluid. The specific characterisations of the roll heat pipe analysed in this study are summered in Table 2.

3. FINITE DIFFERENCE METHOD

A detailed view cross section of the RHP is shown in the Figure 2

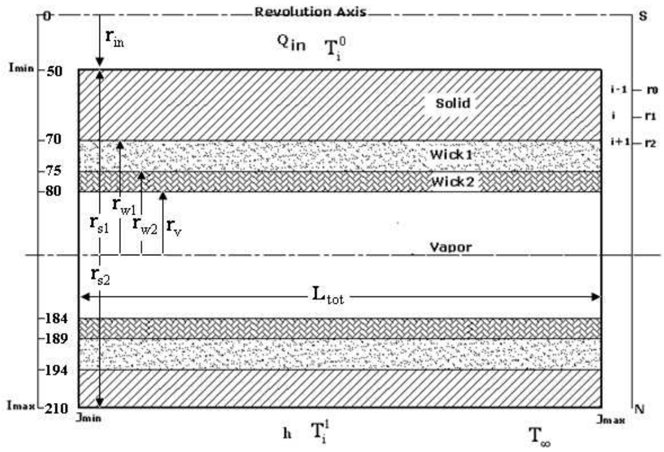


Fig.2: The Discretization form in the RHP

Le heat equation is:

$$\rho\, C_p \frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(r\,\lambda \frac{\partial T}{\partial r}\right) \qquad (4)$$

The fully implicit descritized form of the Equation (4) is as follow if we consider a volume control.

$$\rho\, C_p \Delta V \frac{T_P^1 - T_P^0}{\Delta t} = r_n \lambda_n\, \Delta\theta \frac{T_N^1 - T_P^1}{\delta\, r_n} - r_s \lambda_s \Delta\theta \frac{T_P^1 - T_S^1}{\delta\, r_s} \qquad (5)$$

Equation (10) can be written in a compact form as follow:

$$a_P\, T_P = a_N\, T_N + a_S\, T_S + b \qquad (6)$$

Where

$$a_P = \frac{\lambda_n\, r_n\, \Delta\theta}{(\delta\, r)_n} \qquad a_N = \frac{\lambda_n\, r_n\, \Delta\theta}{(\delta\, r)_n} \qquad a_S = \frac{\lambda_S\, r_S\, \Delta\theta}{(\delta\, r)_S}$$

$$a_0 = \frac{\rho\, C_P\, \Delta V}{\Delta t} \qquad (7)$$

$$b = S_c \Delta V + a_0 . T_0 \quad \text{And} \quad a_P = a_N + a_S + a_S \text{ -} S_p \Delta V \qquad (8)$$

Equation (6) is solved by TDMA method (Tri-Diagonal Matrix Algorithm) in order to determine the temperature of different parts of RHP during the transient analysis.
For the discretization, an interval time of 10^{-4} seconds has been chosen and the radial discretization form at t =0 s is:

$$r_0 = \Delta\, r\,(i-1) \qquad (9a)$$

$$r_1 = \Delta r(i) \quad (9b)$$

$$r_2 = \Delta r(i+1) \quad (9c)$$

$$D_N = \frac{r_2}{\left(\frac{0.5\Delta r}{\lambda_{i+1}^0} + \frac{0.5\Delta r}{\lambda_i^0}\right)} \quad \text{And} \quad D_S = \frac{r_0}{\left(\frac{0.5\Delta r}{\lambda_i^0} + \frac{0.5\Delta r}{\lambda_{i+1}^0}\right)} \quad (10)$$

- In : $i = imax$

$$D_N = r_2\, h \quad \text{And} \quad D_C = \frac{\rho\, C_p r_1\ \Delta r}{\Delta t} T_i^0 - r_2\, h\, T_\infty \quad (11)$$

- In : $i = imin$

$$D_S = 0 \quad \text{And} \quad D_C = \frac{\rho\, C_p r_1\, \Delta r}{\Delta t} T_i^0 + r_0\, Q_{in} \quad (12)$$

$$D_P = -(D_N + D_S) + \frac{\rho\, C_p r_1\, \Delta r}{\Delta t} \quad (13)$$

Tab 2. Roll Heat Pipe Specifications

heat pipe wall	**Wick structure**	**Vapor region**
$r_{in} = 0.0055$	$r_{w1} = 0.005$	$r_v = 0.0127$
$r_{s1} = 0.0157$	$r_{w2} = 0.005$	$\rho_v = 0.599$
$r_{s2} = 0.0153$	$\varepsilon_1 = 0.36$	$\lambda_v = 0.0251$
$L_{tot} = 0.4$	$\varepsilon_2 = 0.69$	$C_{Pv} = 1851$
$\lambda_s = 387.6$	$\rho_l = 960.63$	
$\rho_s = 8978$	$\lambda_l = 0.680$	
$T_a = 297.15$	$C_{pl} = 4216$	
$C_{ps} = 381$	$\lambda_{w1} = 166.5$	
$h = 17.3$	$\lambda_{w2} = 1.23$	
$r_i = 0.0022$	Material : powdered and sintered copper	
Material : copper	Bridge wick =4	

4. RESULT AND DISCUSSION

The numerical study during the transient analysis is done with a number of grid points of 210 in radial direction. This corresponds to one point each distance of 1 mm in the wick and the solid structures; and about 0.245 mm in vapor space. The simulation of the heat pipe is established with the specific characterisations summered in table.1. The precision of the solution was evaluated until the outer surface temperature reaches 180 °C.

First, the transient responses of the outer surface temperature for two heat input applied are compared with the numerical results given by Jalilvand [1] as shown is Figure.3 considering the same basic heat pipe conditions and dimensions. A relative error can be found because the different mesh grid size and the different liquid fill charge since we supposed that the porous structure is saturated with liquid. In Figure.4, a good agreement is obtained when compared the radial heat pipe temperature in steady state condition (reached when time is superior or equal to 5000 s).

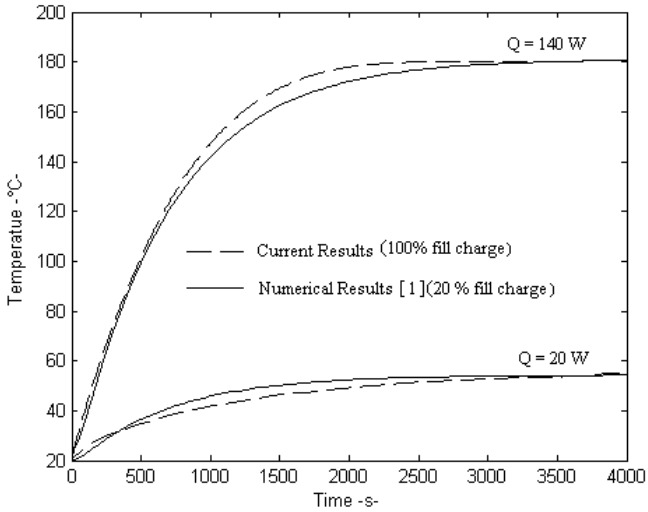


Fig.3: A comparison of the transient response of the heat pipe with thee numerical results given by Jalilvand [1].

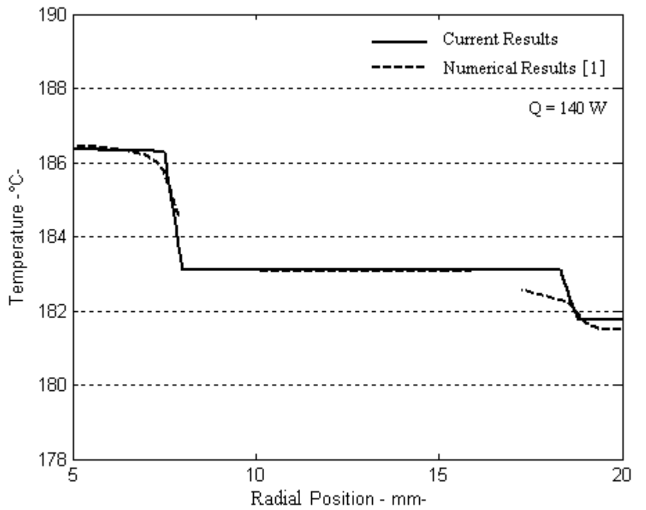


Fig.4: A comparison of the radial heat pipe temperature with the numerical results given by Jalilvand [1].

In Figure.5 the transient response of the heat pipe outer wall temperature (outer diameter of 44 mm) is shown for different heat input. The heat pipe response is faster with higher heat flux. An outer surface temperature of approximately 167 °C is obtained in steady state condition with a heat input of 140W. This value is different from that obtained with the heat pipe compared in Figure.3 because the heat pipe's specifications and inner areas are not the same in this simulation study.

Figure.6 shows the time response of the heat pipe for different heat input. The time response is obtained while the outer surface temperature reaches 180°c. These time constants can be used to determine how fast a heat pipe responds to an applied input power and depend of the heat transfer coefficient. As shown, increasing the heat transfer coefficient can greatly reduce the time constant.

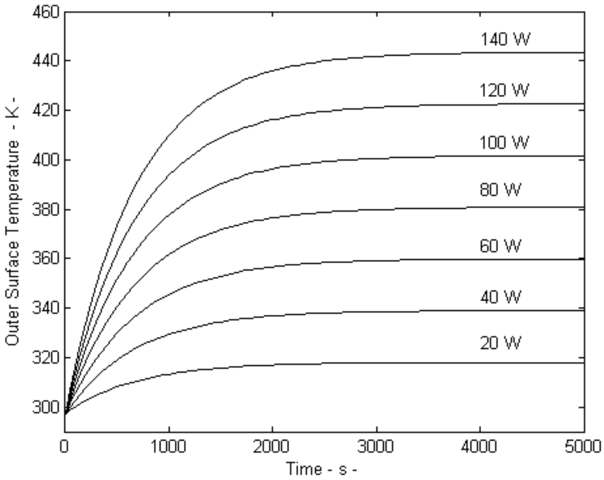


Fig.5: The outer surface temperature response for different heat input

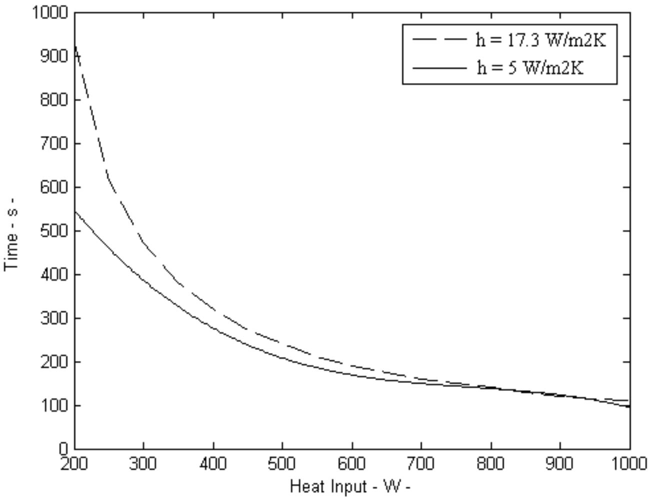


Fig.6: The RHP time response constant for different heat input

Figure.7 shows the start up of the heat pipe. The top wick acts as condenser and the bottom wick acts as evaporator. the temperature drops across the heat pipe liquid wick is much larger than that across the heat pipe wall but one can see that the temperature drops across the sintered wick (wick 1) is nearly constant since its thermal conductivity is much larger then that of the screen mesh (wick 2). Note that a deeper study is needed to investigate the liquid flow in the wick structure since the flow in the wick is pretty much governed by the hydraulic conductivity of the porous medium.

A comprehensive study was carried out by comparing the heat pipe with an annular solid pipe of pure copper of the same dimensions. This will give a better understanding of under what conditions the heat pipe operate more effectively compare with a pipe of pure conduction. Figure.8 shows that the annular solid pipe responds much more slowly than roll heat pipe due to its larger mass and thermal energy storage capacity and even after 5000 seconds. For a larger heat transfer coefficient, it can be seen that the steady state condition is reached after only 700s approximately and 3000s approximately for the annular solid pipe with lower temperature. It can be concluded that the outer surface temperature is controlled by the heat transfer coefficient (also the heat input) and the roll heat pipe has a much stronger heat transfer capability than annular heat pipe of same dimension.

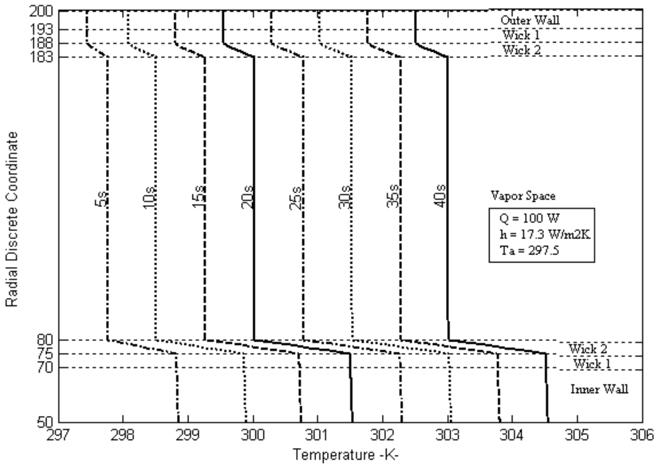


Fig.7: The radial temperature distributions of the RHP at different times during the transient startup process.

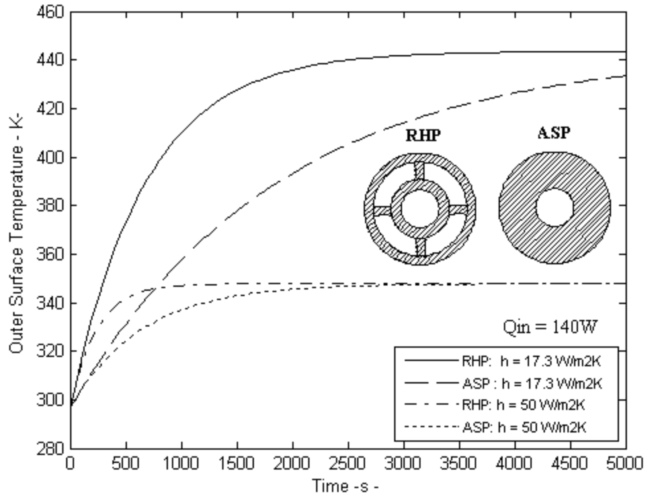


Fig.8: Comparison between a roll heat pipe and an annular solid pipe.

5. CONCLUSION

In this paper, a contribution study of the transient response of the roll heat pipe is performed. A finite difference method of the equation governing the operation of a heat pipe during transient conditions are presented and discussed in detail. The analysis presented in this study will be useful for such heat pipe design in processes that are essentially transient.

It can be concluded that the condensation temperature is controlled by the heat transfer coefficient and the heat input applied and the roll heat pipe perform better than the annular heat pipe of same dimension.

6. REFERENCES

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