Design of a heat pipe for spacecraft cooling

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ABSTRACT

Heat pipes are highly efficient thermal management devices that transfer heat utilizing the phase change of a working fluid. Their ability to achieve thermal conductivities far exceeding those of solid conductors makes them ideal for extreme environments, such as spacecraft thermal control systems. This study focuses on designing a heat pipe for spacecraft cooling, addressing challenges such as weight and space limitations. The design process incorporates thermodynamic principles, heat transfer, and structural mechanics, to ensure optimal performance under stringent conditions.

Keywords: Heat pipe, thermal management, spacecraft cooling, phase change, thermal conductivity.

1. INTRODUCTION

Heat pipes are highly efficient devices that transfer heat using the phase change of a working fluid, leveraging latent heat rather than sensible heat. This allows them to achieve thermal conductivities significantly higher than those of solid conductors, enabling the transfer of large amounts of heat across small cross-sectional areas with minimal temperature differences. The concept of the heat pipe was first proposed by Gaugler in 1942, with further developments by Trefethen in 1962. A heat pipe consists of a sealed tube with a porous wick structure saturated with a working fluid. When heat is applied at the evaporator, the fluid vaporizes, creating a pressure difference that drives the vapor to the condenser, where it releases latent heat. The main sections of a heat pipe are the evaporator, condenser, and adiabatic regions, as shown in Figure 1.

Unlike solid conduction, heat pipe performance depends not only on size, shape, and material but also on construction, working fluid, and heat transfer requirements. The three essential components of a heat pipe are:

1. The wick structure
2. The working fluid
3. The container

1. HEAT PIPE THEORY

There are many variants of heat pipe but in all cases the working fluid must circulate from the evaporator section to the condenser section, and it returns to the evaporator when a temperature difference exists between the evaporator and the condenser. In order for the heat pipe to operate, it must satisfy the following limitations:-

1- The maximum capillary pumping head $\Delta p_{c,\max}$ must be greater than the total pressure drop in the pipe.

$$(\Delta p_{c,\max} > \Delta p_l + \Delta p_v + \Delta p_g) \qquad (1)$$

2- Checking of vapor flow " sonic limitation”.

3- Tearing of liquid of the liquid–vapor interface by vapor flowing at high velocity "entrainment limitation".

4- Disruption of the liquid flow by nucleate boiling in the "boiling limitations".

If one of the above conditions is not fulfilled the wick will dry out in the evaporator region and the pipe will not operate, these limits are illustrated in figure (2).

Pressure Balance

For the required balance of pressure it is necessary that along the entire length of the heat pipe, the pressure at the liquid side of the liquid–vapor interface is different from that at the vapor side, except at the point where the difference is minimum. This pressure difference is called the capillary pressure.

The maximum capillary pressure can be calculated using Laplace and Young equation in case of cylinder pore.

$$p_{c,\max} = \frac{2\sigma_l}{r_c} \qquad (2)$$

The values of the effective capillary radius (r_c) depends on the wire diameter, the space between wires and the wick structure shape.

The pressure drop in wick structure can be calculated by integrating the liquid pressure gradient.

$$\Delta p_l = -\int_{x_1}^{x_2} \frac{dp_l}{dx} dx \qquad (3)$$

The liquid pressure drop depends on many parameters such as friction drag, Reynolds Number, hydraulic radius, axial heat flux, latent heat of vaporization, wick cross section area, inclination angle and the wick permeability. The final expression of the liquid pressure drop is

$$\frac{dp_L}{dx} = -F_L\, Q \pm \rho_L\, g \sin\phi \qquad (4)$$

The values of $(F_L\,,\,k)$ can be obtained directly using special charts. The vapor pressure drop in heat pipe is calculated by integrating the vapor pressure gradient

$$\Delta p_V = -\int_{x_1}^{x_2} \frac{dp_V}{dx}\, dv \qquad (5)$$

The principles of conservation of axial momentum can be applied to an elementary control volume, and the final relation is as following

$$\frac{dp_V}{dx} = -F_V Q - D_V\, \frac{dQ^2}{dx} \qquad (6)$$

The values of the friction coefficient F_v and the dynamic pressure coefficient D_v can be obtained using special charts. Hence the effective capillary pressure $P_{c,e}$ is calculated using equation

$$P_{c,e} = \int_{X_{min}}^{X} \left(\frac{dP_V}{dX} - \frac{dP_L}{dX} \right) dX \qquad (7)$$

The maximum effective capillary pressure $P_{c,me}$ will be smaller than the maximum capillary pressure $P_{c,max}$,

This difference is due to the effect of the gravitational force in a direction perpendicular to the heat pipe axis such that

$$P_{c,me} = P_{c,m} - \rho_L gdv \cos\varphi$$

$$i.e \quad \frac{2\sigma_L}{r_c} - \Delta P_L = \int_0^{L_t} \left(\frac{dP_X}{dX} - \frac{dP_L}{dX} \right) dX \qquad (8)$$

substitution about $\frac{dP_V}{dX}, \frac{dP_L}{dX}$ from equations ((4), (5) in equation (6), and solving for Q_L yields

$$Q_{L_{C,MAX}} = \int_0^{L_t} Q_{c,\max} dX = \frac{\frac{2\sigma_L}{r_c} - \Delta P_L - \rho_L g L_t \sin\varphi}{F_L + F_V} \qquad (9)$$

In the case where the heat pipe has a uniform heat flux distributions along its evaporator and condenser sections, the axial heat flux will has the following final form.

$$Q_{c,\max} = \frac{Q_{L,\max}}{\frac{L_c}{2} + L_a + \frac{l_e}{2}} \qquad (10)$$

The general procedures to evaluate the capillary limitation of heat pipe is described in the material that follows:

1Calculate the required capillary pressure and compare it with the maximum effective capillary pressure as follows

1. if $P_{c,r}=P_{c,max}$,the assumed heat load is the required capillary limitation .
2. if $P_{c,r}<P_{c,max}$,increase the assumed heat load ,and repeat step1.
3. If $P_{c,r}>P_{c,max}$, decrease the assumed heat load ,and repeat step1.

Limitations to heat transport in heat pipe

1- Boiling limitation

The liquid pressure at the evaporator is equal to the saturation pressure at the temperature of the liquid vapor interface minus the capillary pressure at the temperature of the liquid vapor interface. Since the difference increase with increase of the radial heat flux of the heat pipe at the evaporator, vapor bubbles may be formed in the evaporator wick, and this may cause hot spots of the liquid and abstract the circulation of the liquid. Hence, there is a heat flux limit for the evaporation at the heat pipe. This limit is known as the boiling limit which represents the limitations of the axial heat flux.

The boiling heat transport limit is calculated using the following expression

$$Q_{b,\max} = \frac{2\pi L_e K_e T_v}{L \rho_V \ln\left(\frac{r_i}{r_v}\right)} \left(\frac{2\sigma_L}{r_n} - P_c\right) \tag{11}$$

2- Sonic limitation

The sonic limit is defined as a limit at which the vapor is moving at speed of sound at the evaporator exit. The maximum mass flow rate relates directly to the maximum heat transfer rate. This takes place when the evaporator exit velocity reaches the local sonic velocity. Increasing the heat rejection rate decreases the condenser temperature, induces the supersonic vapor flow and creates very large axial temperature gradients along the pipe. The sonic limit is given by the following expression

$$Q_{s,\max} = A_v L \rho_v \left(\frac{1+\gamma_v}{2+\gamma_v}\right)\left(\gamma_v R T_v\right)^{0.5} \quad (12)$$

3- Viscous limitation

At low operating temperature, the vapor pressure difference between evaporator and condenser regions of a heat pipe may be extremely small. The viscous forces within the vapor region may dominate over the pressure gradient because of the temperature field. In this condition, the pressure gradient may not be sufficient to generate flow and the vapor may stagnate. This is called the viscous limit. This limit occurs when the heat pipe is operating at temperatures below its normal operating range, such as during startup from frozen state. The Viscous heat transport limit can be calculated using the following formula:

$$\frac{Q_{viscous}}{A_V} = \frac{r_V^2 L \rho_V P_V}{16 \mu_V L_{e,e}} \quad (13)$$

Design criteria

The design criteria of the heat pipe is based on the following general points:

1. Selection of the suitable working fluid.
2. Design and selection of the wick structure.
3. Structural design of the container.
4. Heat transfer limits must be checked to ensure the pipe will operate within all limits.

1- Selecting of working fluid:

The performance and life of a heat pipe is greatly dependent on working fluid employed. Care must be exercised in selecting a suitable fluid for the operating condition. The working fluid should satisfy the following requirements:

- The working fluid must have a melting point temperature below and critical temperature above the pipe operating boiling temperature.

- Good thermal stability to prevent the fluid breaking down into different components.

- A high enthalpy of vaporization is desirable in order to transfer a large amount of heat.

- The thermal conductivity should be high in order to minimize the radial temperature gradient and to reduce the possibility of nucleate boiling at wick wall interface.

- Compatibility with wick and wall material is necessary to increase the life of the heat pipe.

- High surface tension is desirable in order to enable the heat pipe to operate against the gravity.

- Wettability of wick and wall material is necessary so that the working fluid will meet the wick and container material with very small contact angles.

Material selection:

The selection of the heat pipe material and wick material is based on the following requirements:

- Compatibility with working fluid.

- High thermal conductivity.
- Ease of fabrication to decrease the cost of heat pipe manufacturing.
- High strength to weight ratio.

Heat pipe design procedures

The procedures of designing a heat pipe are as following

1- Pipe diameter will first be determined so that the vapor velocity is not high, such that the maximum Mach number in the vapor flow passage is not greater than 0.2 (exceed)

2- Mechanical design theory will be used to determine the container details.

3- Wick details will be designed considering the capillary limit.

4- Other heat transfer limits (entertainment, sonic and boiling) should be checked to ensure that the pipe will operate within all limiting conditions .

Design of vapor cone diameter

The vapor core diameter (d_v) at vapor Mach number Mv = 0.2 is determined by using the following equation

$$d_V = \left(\frac{20\, Q_{\max}}{\pi\, \rho_V\, \lambda\, (\gamma_V R_V T_V)^{\frac{1}{2}}} \right)^{1/2} \qquad (14)$$

Design of heat pipe container

The most widely used design technique for heat pipe container that must withstand vapor pressure is the (ASME) code. The calculations of the maximum stress depend on the geometry of the tube and the wall thickness, for rounded tube and t/d >10 %

$$f_{\max} = \frac{\Delta P\left(d^2{}_o + d^2{}_i\right)}{d^2{}_o - d^2{}_i} \qquad (15)$$

Wick design

There are many charts which are useful for quick determination of dimension of the heat pipe wick. The general procedures for designing the wick are as following :-

1- Calculate the hydro-static pressure using the following formula

$$P_{HYD} = \rho_L \, g(di\cos\phi + Lt\sin\phi) \qquad (16)$$

2- Select the mesh number such that the P_c is much smaller than twice hydrostatic pressure

3- Calculate the wick thickness

$$t_w = \frac{di - dv}{2} \qquad (17)$$

4- Calculate the maximum capillary heat transfer using equation

$$Ql_{c,\max} = \frac{P_C - P_{hyd}}{F_L + F_V} \qquad (18)$$

5- Check the maximum capillary heat transfer rate, which must be greater than ~~that~~ the required heat transport rate.

6- Check the entertainment, sonic, viscous and boiling limitation that;

$$Q_{req} < (Q_{e,\max}, Q_{b\max}, Q_{s,reg})$$

Design problem

In this project it is required to design a heat pipe which will be capable of transferring a minimum of 15W at a vapor temperature between 0^0 C to 80^0C over a distance of 1m in zero gravity (a satellite application). Restrains on the design are such that the evaporator and condenser sections are each (8 cm) long, while the adiabatic section is (84 cm) long located between the evaporator and condenser sections. Because of weight and volume limitation, the cross section area of the vapor space should not exceed $0.197cm^2$. Knowing that the average working temperature is 80^0C and the average vapor pressure is about $40.9x10^5$ N/m^2.

The properties of Ammonia?? are shown in Table(1).

Table 1. Properties of Ammonia at $T_H=80^0$ C

Property	Symbol	Magnitude	Unit
Latent heat	L	891	KJ/Ka
Liquid thermal conductivity	Kl	0.235	W/(mC)
Liquid density	Pl	505.7	Kg/m^3
Liquid viscosity	Ml	0.107*10-3	Kg/m.sec
Liquid surface tension	Σl	0.00767	N/m^2
Vapor density	Pv	34.13	Kg/m^3
Vapor viscosity	Mv	0.365*10-3	Kg/m.sec
Vapor pressure	Pv	40.9*105	N/m^2
Vapor specific heat	Γv	2.21	(KJ/Kg)C

Design of container parameter:

Assuming vapor cone Area $A_V = 0.197 cm^2$, using a specific chart?? which indicates that for $P_v = 40.11 X 10^5 N/m^2$ r_v / r_i = 0.754 ,

Let $d_i = 6.629x10^{-3}$ m, hence $d_o = 9.525x10^{-3}$ m

- **Design of the wick thickness:**

$t_w = (r_i – r_v) = (3.314x10^{-3} – 2.5*10^{-3}) = 0.814*10^{-3}$ m

$$A_w = \frac{\pi}{4}\left(d_i^2 - d_v^2\right) = 1.486*10^{-5} m^2$$

- **Check for hoop stress:**

$$f_{hoop} = \frac{\Delta P_v\left(d_0^2 + d_i^2\right)}{d_0^2 - d_i^2}$$

$$\frac{f_{ult}}{9} = 29.11\times10^6 \ N/m^2$$

$$f_{hoop} = 11.77\times10^6 \frac{N}{m^2}$$

hence $f_{hoop} < f_{ult}$

Assume that the number of wick layer is n = 20, hence the wire diameter (d_w) becomes

$d_w = t_w / 2 n = 0.814*10^{-3} / 40 = 2.03*10^{-5}$ m

- **Calculating of the maximum capillary pressure:**

$$P_{c,\max} = \frac{2\sigma_L}{r_c}, r_c = \frac{1}{2*3937} = 1.27*10^{-4} m$$

$$P_{c,\max} = \frac{2*0.00767}{1.27*10^{-4}} = 120.78 \frac{N}{m^2}$$

- **Normal hydrostatic pressure:**

$$\Delta P_L = \rho_L\, g\, d_V \cos\phi, \qquad \phi = 0^{o},$$

$$\Delta P_L = 24.8\ \mathrm{N/m^2}$$

Check limitation of the heat pipe

1- Sonic limit check

$$Q_{s,\max} = A_v \rho_v L \left(\frac{\gamma_v + 1}{\gamma_v + 2} \right) (\gamma_v R_v T_v)^{\frac{1}{2}}$$

$Q_{S,\max} = 8.914$ KW

Since $Q_{s,\max} > Q_{required}$,

Thus it satisfies the required condition.

2- Viscous limit check:

$$\frac{Q_{viscous}}{A_v} = \frac{r_v^2 L \rho_V P_V}{16 \mu_v L_{e,e}}$$

$$Q_{viscous} = 11.4 \times 10^7\ Kw$$

Satisfy?

3- Boiling limit check

$$Q_{b,\max} = \frac{2\pi L_e K_e T_V}{\rho_V L \ln\left(\frac{r_i}{r_V} \right)} \left(\frac{2\delta_L}{r_n} - P_c \right)$$

$$Q_{b,\max} = 1425.9W$$

$Q_{b,\max} > Q_{\text{required}}$.

This satisfies the required condition.

Calculation of heat pipe performance

The heat pipe performance is characterized by the overall coefficient of heat transfer which is defined by the equation

$Q = A\, U_{HP}\, (T_{P,e} - T_{P.c})$

- The thermal resistance at the evaporator ($R_{P,e}$):

$$R_{P,e} = \frac{Ln\left({r_o}/{ri} \right)}{2\pi\, L_e K_P} = 4.4\text{x}10^{-3}\ C^0/\ W$$

- The thermal resistance of the saturation wick at the evaporator:

$$R_{W,e} = \frac{Ln\left({r_i}/{r_v} \right)}{2\pi\, L_e K_{e,e}} = 0.0396\ m^2.C^0/W$$

$K_{e,e}$=14.15 W/m°C

- The thermal resistance of the vapor flow R_v

$$R_V = \frac{F_V T_V \left(\frac{L_e}{6} + L_a + \frac{L_C}{6} \right)}{\rho_V L}$$

$$R_v = 1.5\times10^{-8}\ C^{o}/w$$

- The thermal resistance of the wick at condenser (R_{wic})

$$R_{W,C} = \frac{Ln\left({r_i}/{r_v} \right)}{2\pi\, L_C K_{e,c}}$$

$$R_{w,c} = 0.0396\, C^{o}/w$$

$$U_{H,P} = \left(\frac{1}{R_{p,e} + R_{w,v} + R_{v} + R_{p,c} + R_{wic}} \right) \frac{1}{A_p}$$

$$= 162337.6 \, {}^{W}\!/_{m^2 C}$$

- Estimation of temperature variation across the pipe wall

  The vapor temperature differences

$$\Delta T_v = QR_v = 2.25 \times 10^{-7} C^{o}$$

so vapor temperature of the condenser is:

$$T_{v,c} = T_{v,e} - \Delta T_v = 80 - 2.25 \times 10^{-7} = 79.99 C^{o}$$

- Temperature difference at wick pipe interface at condenser

$$\Delta T_{w,c} = QR_{wic} = 0.594 C^{o}$$

$$T_{pw,c} = T_{v,c} - \Delta T_{w,c} = 79.99 - 0.594 = 79.396 C^{0}$$

- Temperature difference across the pipe wall at the condenser

$$\Delta T_{p,c} = Q R_{p,c} = 0.066 C^{0}$$

1. Hence the condenser surface temperature
2. $T_{p,c} = 79.396 - 0.066 = 79.33$ C^{0}

- The temperature difference across the surface wick at the evaporator

$$\Delta P_{w,e} = Q R_{w,e} = 0.594 C^{o}$$

3. Hence the temperature of pipe wick interface at the evaporator

$$T_{P,we} = T_{v,e} + \Delta T_{w,e} = 80 + 0.594 = 80.594 C^{0}$$

- The temperature difference across the pipe wall at the evaporator

$$\Delta T_{P,e} = Q\,R_{p,e} = 0.066\,C^{0}$$

4. So the pipe temperature at the evaporator is equal to

$$T_e = T_{pw,c} + \Delta T_{pe} = 80.4 + .066 = 80.66\ C^0$$

Final design results

The final results of the designed heat pipe are as following

Work fluid	Ammonia
Container material	Aluminum
Wick material	Stainless steel (SS304)
Outside diameter	$d_0 = 9.525\text{x}10^{-3}$ m
Inside diameter	$d_i = 6.629\text{x}10^{-3}$ m
Vapor cone diameter	$d_v = 5\text{x}10^{-3}$ m
Pipe wall thickness	$t_p = 1.448\text{x}10^{-3}$ m
Wick thickness	$t_w = 0.814\text{x}10^{-3}$ m
Wick area	$A_w = 1.486\text{x}0^{-5}\text{m}^2$
Wick diameter	$d_w = 2.03\text{x}10^{-5}$ m
Mesh number	N = 3937 m
Number of wick lager	n = 20
Maximum capillary pressure	$P_{c,max} = 120.78\ \text{N/m}^2$
Axial hydrostatic pressure	$P_{h,e} = 24.8\ \text{N/m}^2$
Pipe length	$L_t = 1$ m
Length of heat pipe adiabatic length.	$L_a = 0.84$ m
Length of heat pipe condenser section	$L_c = .08$ m
Length of heat pipe evaporator section	$L_e = 0.08$ m
Vapor Reynold number	$R_{ev} = 1170.642$

Vapor Mach number	$M_V = 0.0012$
Sonic limit	$Q_{s,\max} = 8.914KW$
Viscous limit	$Q_{V,\max} = 11.4x10^{7}\ Kw$
Boiling limit	$Q_{b,\max} = 1425.92W$
Vapor temperature o f the condenser	$T_{V,c} = 79.99^{o}c$
Pipe wick temperature of the condenser section	$T_{pw,c} = 79.396^{o}c$
Condenser surface temperature	$T_{P,C} = 79.33^{o}c$
Pipe wick temperature of the evaporation section	$T_{P,w,e} = 80.594^{o}c$
Surface temperature of the evaporator section	$T_{P,e} = 80.66^{o}c$

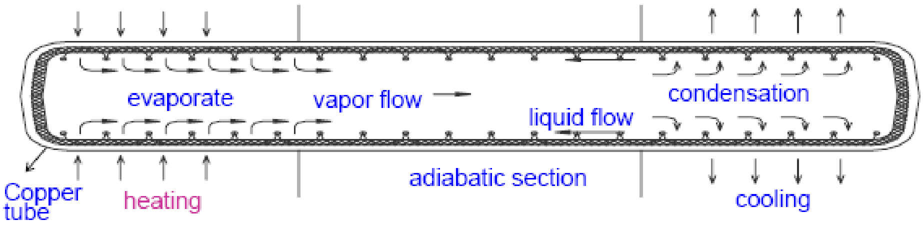

Fig. 1. Components of heat pip

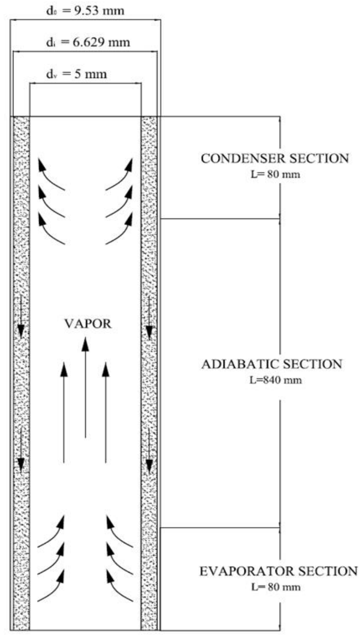

Fig. 2. The designed heat pipe

CONCLUSION :

1- An effective wick structure requires small surface pores for large capillary pressure .

2- Design of the best heat pipe type and size for any given application is, however, a complex process involving many factors such as heat pipe performance, reliability, compatibility and ease of manufacturing

3- The performance and life of a heat pipe is greatly dependent on the compatibility between the working fluid and the material of the heat pipe .

4- Copper suited most as a material for heat pipe container, to resist the stresses developed , and being compatible with variety of working fluids

5- Compatibility property of the material of heat with the working fluid is very important factor.

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NOMENCLATURE

Symbol	Definition	Unit
d_0	*Pipe outside diameter*	*m*
d_i	*Pipe inside diameter*	*m*
d_v	*Vapor core diameter*	*m*
d_w	*Screen wire diameter*	*m*
A_p	*Cross-section area based up on pipe outside diameter*	m^2
A_v	*Vapor core cross-section area*	m^2
D_v	*Dynamic pressure coefficient*	*m.sec/ka*
σ_L	*Surface tension force at liquid -wick interface*	*N*
F_v	*Frictional coefficient for vapor flow*	$(N/m^2)/W.m$
f_L	*Drag coefficient for liquid flow*	[ ]
f_v	*Drag coefficient for vapor flow*	[ ]
G	*Gravitational acceleration*	m/sec^2
$K_{e,c}$	*Effective thermal conductivity of liquid saturated wick at condenser*	$W/m.C^0$
K_l	*Thermal conductivity of liquid*	$W/m.C^0$
K_w	*Thermal conductivity of wick material*	$W/m.C^0$
K_p	*Thermal conductivity of pipe material*	$W/m.C^0$
L	*Latent heat of vaporization*	*KJ/Ka*
L_a	*Length of heat pipe adiabatic section*	*M*
L_c	*Length of pipe condenser*	*M*
L_e	*Length of pipe evaporator*	*M*
P_c	*Capillary pressure*	*Pas*
$P_{c,r}$	*Required capillary pressure*	*Pas*
$P_{c,max}$	*Maximum capillary pressure*	*Pas*
$P_{c,me}$	*Effective capillary pressure*	*Pas*

P_L	Liquid pressure	Pas
P_v	Vapor pressure	Pas
ΔP_g	Pressure drop due to the gravity	Pas
ΔP_l	Liquid pressure drop	Pas
ΔP_v	Vapor pressure drop	Pas
ΔP_L	Normal hydrostatic pressure drop	Pas
Q	Heat flow rate	W
$Q_{b,max}$	Boiling limit on heat transfer rate	W
$Q_{c,max}$	Capillary limit on heat transfer rate	W
$Q_{e,max}$	Entrainment limit on heat transfer rate	W
$Q_{s,max}$	sonic limit on heat transfer rate	W
R_v	Thermal resistance for vapor flow from evaporator to condenser	$m^2.C^0/W$
R	Radius of cylinder	m
r_c	Effective capillary radius	m
r_i	Inside radius of pipe	m
r_v	Vapor core radius	m
T_v	Vapor temperature	C^0
t_p	Pipe thickness	m
t_w	Wick thickness	m
γ_v	Vapor specific heat ratio	[ ]
ρ_L	Liquid density	Kg/m^3
V_L	Liquid velocity	m/sec
F_L	Friction coefficient for liquid flow	$(N/m^2)/W.m$
f_{max}	Maximum tensile stress	N/m^2
f_{ult}	The ultimate stress	N/m^2
$f_{ult,d}$	The ultimate design stress	N/m^2
K_ρ	Thermal conductivity of pipe material	$W/m.C^0$
N	Screen mesh number	Meshes/m
$r_{h,l}$	Hydraulic radius for liquid flow	m

$r_{h,e}$	*Hydraulic radius of wick at vapor-wick interface*	*m*
$r_{h,v}$	*Hydraulic radius for vapor flow*	*m*
r_i	*Inside raduis of pipe*	*m*
r_v	*Vapor core radius*	*m*
V_L	*Liquid velocity*	*m/sec*
V_v	*Vapor velocity*	*m/sec*
W	*Wire spacing*	*m*
ρ_v	*Vapor density*	Kg/m^3
Φ	*Heat pipe inclination measured from the horizontal position*	*degree*