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Optimization of Copper-Water Negative Inclination Heat Pipe with Internal Composite Wick Structure

I. Brandys, M. Levy, K. Harush, Y. Haim, M. Korngold

***Abstract*—**Theoretical optimization of a copper-water negative inclination heat pipe with internal composite wick structure had been performed, regarding a new introduced parameter: the ratio between the coarse mesh wraps and the fine mesh wraps of the composite wick. Since in many cases, the design of a heat pipe matches specific thermal requirements and physical limitations, this work demonstrates the optimization of a 1m length, 8mm internal diameter heat pipe without an adiabatic section, at a negative inclination angle of -10º. The optimization is based on a new introduced parameter, L_R: the ratio between the coarse mesh wraps and the fine mesh wraps.

***Keywords*—**Heat pipe, inclination, optimization, ratio.

I. Introduction

A heat pipe is an efficient phase-change device which enables to dissipate large amount of heat from a required component with a minimal temperature drop [1] through a relative small cross-sectional area [2].

This high performance device is usually used in many applications: electronics cooling, heat recovery system, waste management and different thermal management applications, such as vehicle thermal management [3] etc. It can even be used to construct a heat exchanger for heat recovery in hospitals and laboratories, as displayed in the research of Noie-Baghban and Majideian [4].

There are no unique dimensions for heat pipes. Each case requires its own solution as well as its materials: both mechanical structure and the working fluid.

It is well known that a heat pipe's efficiency is strongly dependent on shape, working fluid and physical structure. A heat pipe can be classified according to its physical structure or its working fluid, as described in [5]. Its heat capacity limitations (Fig. 1) are: capillary limit, viscous limit, sonic limit, entrainment limit and boiling limit [1].

In a positive inclination heat pipe, a case in which the evaporator is lower than the condenser, the gravitational force enables the heat pipe's proper operation. A wick structure is a known way to upraise its capillary limit since the wick enables greater capillary pressure difference [2], [6].

A negative inclination heat pipe's (Fig. 2) proper operation is more complicated due the need to overcome the gravitational force and prevent the heat pipe's dryout.

I. Brandys is with Nuclear Research Center Negev, Beer Sheva, 84190 Israel (e-mail: iradbr@gmail.com).
M. Levy and Y. Haim are with Department of Mechanical Engineering, Ben-Gurion University of the Negev, Beer-Sheva, Israel.
K. Harush and M. Korngold are with Nuclear Research Center Negev, Beer Sheva, 84190 Israel.

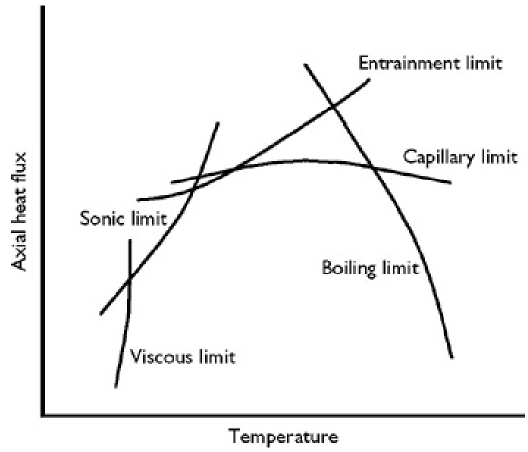

Fig. 1 Recommended working range of a heat pipe [1]

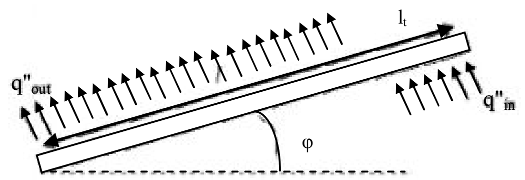

Fig. 2 Schematic view of a negative inclination heat pipe

A wick structure is one of the solutions for this kind of heat pipe: using a wick structure can create the required pumping force, which enables the return of the working fluid to the evaporator by overcoming the gravitational force. The major parameters which affect the capillary pressure in screen wicks are the effective pore radius and the permeability. The capillary pressure, which is required to overcome the gravitational force, can be calculated by:

$$\Delta P_c = 2\sigma \cos\theta / r_c \tag{1}$$

where σ is the surface tension of the working fluid and θ is the contact angle between the working fluid and the wick structure. For a single wick, the effective pore radius, r_c, can be determined based on experimental results as follows [8]:

$$r_c = 1.11w \tag{2}$$

where w is the space between the wick's wires. The permeability of the wick structure can be simply calculated according to [9]:

$$K = d^2\varepsilon^3 \big/ \left[122(1-\varepsilon)^2\right] \qquad (3)$$

where d is the wick's wire diameter and $\varepsilon = 1\text{-}1.05\pi Nd/4$ is the wick's porosity, N is the mesh number in [1/inch].

Fig. 3 displays a schematic cross section of a heat pipe with a wick structure.

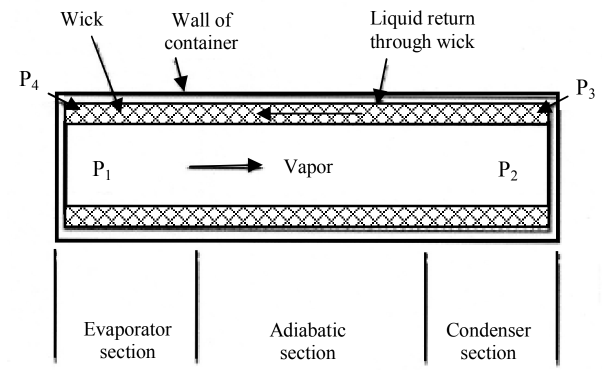

Fig. 3 Longitudinal cross section of a heat pipe with a screen wick

For a proper heat pipe operation, beside the need to overcome the gravitational (both axial and radial) pressure drop, ΔP_g, the capillary limit must overcome the liquid pressure drop from the condenser to the evaporator ($\Delta P_l = P_3 - P_4$ in Fig. 3), the vapor pressure drop ($\Delta P_v = P_2 - P_1$ in Fig. 3) as well, as described in (4)-(6) respectively:

$$\Delta P_g = \rho_l g l_t \sin\varphi + \rho_l g d_v \cos\varphi \qquad (4)$$

$g=9.81\ m\cdot s^{-2}$ is the acceleration of gravity, l_t is the total length of the heat pipe, φ is the inclination angle relative to the horizontal and d_v is the vapor space diameter.

$$\Delta P_l = \frac{\mu_l l_{eff} \dot{m}}{\rho_l K A_w} \qquad (5)$$

μ_l is the liquid viscosity, l_{eff} is the effective length of the heat pipe, $\dot{m}$ is the mass flux of the working fluid which is determined by $\dot{m}=q/h_{fg}$ where q is the heat input at the evaporator and h_{fg} is the latent heat of the working fluid, ρ_l is the liquid density and A_w is the cross sectional area of the wick, as displayed in Fig. 4.

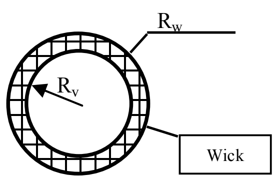

Fig. 4 Cross section of a wick heat pipe

$$\Delta P_v = \left(\frac{C(f_v \,\mathrm{Re}_v)\mu_v}{2 r_v^2 A_v \rho_v h_{fg,v}} \right) l_{eff}\, q \qquad (6)$$

Assuming laminar incompressible flow [1] yields $C(f_v \mathrm{Re}_v)=16$, μ_v is the vapor viscosity, r_v is the radius of the vapor space (Fig. 4), A_v is the vapor flow cross sectional area, ρ_v is the vapor density and $h_{fg,v}$ is the latent heat at the end of the evaporator.

II. Heat Pipe Design: Theory

In the design of a wick heat pipe, heat transport limitations for each working fluid must be considered for validating the requirement fulfillment.

In a negative inclination heat pipe, the capillary limit is given as:

$$q_{capillar} = \frac{1}{L_{eff}} \cdot \frac{\Delta P_c + \Delta P_g}{F_l + F_v} \qquad (7)$$

ΔP_c is calculated according to (1), ΔP_g is calculated according to (4). F_l and F_v are derived from (5) and (6) respectively and are written as:

$$F_l = \mu_l \big/ \left(\rho_l A_w K \cdot h_{fg}\right) \qquad (8)$$

$$F_v = C(f_v \,\mathrm{Re}_v)\mu_v \big/ \left(2 r_v^2 A_v \rho_v h_{fg}\right) \qquad (9)$$

The entrainment limit is calculated according to the following equation:

$$q_{ent} = A_v h_{fg} \left(\sigma_l \rho_v / 2 r_{h,w}\right)^{0.5} \qquad (10)$$

in case of screen wicks, $r_{h,w}=0.5w$, where w is the space between the wick's wire.

The sonic limit is calculated according to:

$$q_{sonic} = \rho_v A_v h_{fg} \sqrt{\gamma_v R_v T_v / 2(\gamma_v + 1)} \qquad (11)$$

$\gamma_v=1.33$ is the specific heat capacity of the working fluid, $R_v=R_u/M$ is the specific gas constant, where $R_u=8314\ J\cdot kg^{-1}\cdot mol^{-1}\cdot K^{-1}$ is the universal gas constant and $M=18\ gr.\cdot mol^{-1}$ is the molecular weight of the working fluid, water and T_v is the temperature at the end of the evaporator.

The final expression for calculating the viscous limit is:

$$q_{viscous} = \pi d_v^4 h_{fg} \rho_v P_v \big/ \left(256 \mu_v l_{eff}\right) \qquad (12)$$

where d_v is the vapor space diameter.

The boiling limit is calculated according to:

$$q_{boiling} = \frac{2\pi L_e k_{eff} T_v}{h_{fg} \rho_v \ln(r_i/r_v)} \left(\frac{2\sigma_l}{r_n}\right) \qquad (13)$$

L_e is the length of the evaporator, k_{eff} is the effective conductivity of the wick structure, r_i is the inner diameter of the heat pipe and r_n is the critical nucleation site radius and is taken to be 2.54×10^{-7}m [1].

III. Optimization of the Internal Structure

From the above equations it can be clearly seen that the most important parameters to consider during the design of a wick heat pipe are the effective pore radius (r_c), the permeability (K) of the wick, which determine the capillary limit, and the vapor space diameter (d_v). In a homogenous wick heat pipe, the finer the mesh (i.e. the smaller pore radius) is favorable for higher capillary pressure but it increases the liquid path through the wick, means lower permeability due to higher liquid pressure drop. Conversely, the coarser the mesh would decrease the liquid pressure drop but would also decrease the capillary pressure.

Composite wicks offer an enhanced solution in case of a negative inclination heat pipe. The coarser mesh layer yields higher permeability while the finer mesh layer provides high capillary pumping pressure, thus extending the heat pipe's capillary limit relative to homogenous wick structure [7]. There is a tradeoff between the number of coarser screen wraps and the vapor space diameter. More wraps means easier liquid return to the evaporator and less vapor space, which decreases the operating limits. Therefore an optimization has been carried out.

The vapor space can be defined based on the inner wick structure geometric parameters as follows:

$$d_v = d_{in} - 4n_f d_f - 4n_c d_c - 4d_c \tag{14}$$

n is the number of mesh wraps, d is the wick wire diameter, where "f" represents the fine mesh of the designed structure and "c" represents the coarse mesh of the structure. The term $4d_c$ is added due to one inner wrap of the coarse mesh which is used to support the entire wick structure and keep it well attached to the inner wall of the heat pipe. This is a technical limitation, due to the low mechanical strength of the fine mesh.

In this work we defined a new variable, L_R, as the ratio between the coarse mesh wraps and the fine mesh wraps of the designed wick structure:

$$L_R = n_c / n_f \tag{15}$$

The optimization of the composite wick will be based on this ratio.

IV. Optimization of the Internal Structure

An optimization has been carried out for a 1m length copper-water heat pipe, whose internal diameter is 8mm and evaporator length of 0.1m, without an adiabatic section. The required temperature range of the heat pipe's operation is from 80ºC to 120ºC.

Since the capillary limit is of prime concern, a comparison between two wick structures was performed, using the assumption of a large contact angle: θ=65º.

Figs. 5 and 6 show a comparison between the calculated capillary limit of those structures. The first is based on 2 wraps of 100 mesh wick and 2 wraps of 400 mesh wick. The second is based on 2 wraps of 50 mesh wick and 2 wraps of 400 mesh wick. The geometric parameters of these wicks are displayed in Table I. As mentioned earlier, on both cases, the composite structure would be supported by one inner wrap of 50 mesh wick.

TABLE I
Geometric Parameters of Screen Wicks

Mesh [1/inch]	Wire diameter, d [mm]	Space between wires, w [mm]	Effective pore radius, r_c [mm]	Permeability, K [m^2]
50	0.195	0.313	0.254	9.93×10^{-10}
100	0.1143	0.154	0.134	1.934×10^{-10}
400	0.03	0.033	0.031	1.1×10^{-11}

As can be seen, the optimized phosphor-bronze composite structure that yields the higher capillary pressure at a maximum negative inclination angle of -10º is based on 50 mesh wick along with 400 mesh wick.

Adding more wraps of 400 mesh wick hardly increases the capillary limit of the heat pipe. Conversely, adding more wraps of 50 mesh wick significantly increases the capillary limit but decreases the vapor space diameter. The capillary limit of the heat pipe with 2 wraps of 400 mesh wick and as a function of the 50 mesh wick wraps is displayed in Fig. 7.

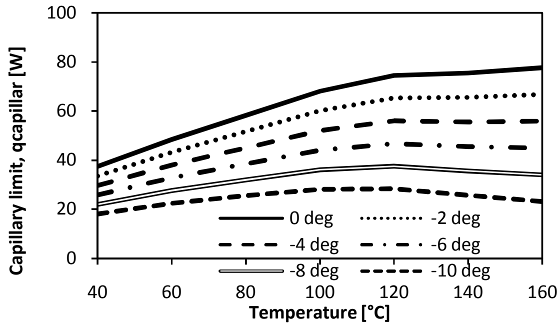

Fig. 5 Capillary limit estimation of 2 wraps of 100 mesh and 2 wraps of 400 mesh composite wick structure

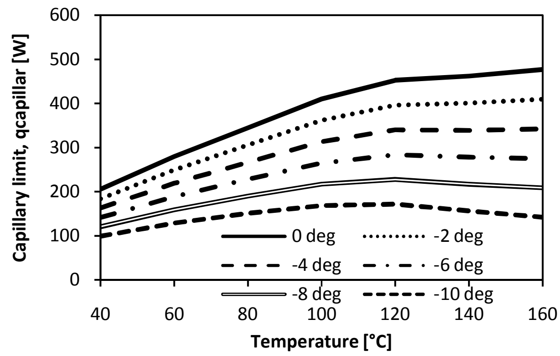

Fig. 6 Capillary limit estimation of 2 wraps of 50 mesh and 2 wraps of 400 mesh composite wick structure

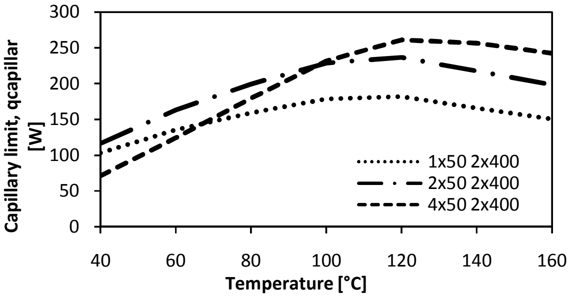

Fig. 7 Capillary limit estimation at an inclination angle of -10º as a function of the coarse wick's wraps number

As can be seen, the higher capillary pressure in the required temperature range is achieved when using 3 wraps of 50 mesh wick.

Rewriting (14) in terms of the new defined ratio, L_R yields:

$$d_v = d_{in} - 4w_c - 4n_f\left(w_f + L_R w_c\right) \tag{16}$$

The heat pipe's optimization was carried out based on the new defined ratio, L_R. Table II summarizes the array for which the heat pipe's limits were calculated.

TABLE II
L_R RATIOS FOR HEAT PIPE'S OPTIMIZATION

n_f	1	2	3	4
n_c		L_R ($=n_c/n_f$)		
2	2	1	0.66	0.5
3	3	1.5	1	0.75
4	4	2	1.33	1

V. RESULTS AND DISCUSSION

It can be pointed out from calculations that the boiling limit, entrainment limit and capillary limit are the limits those which will determine the maximum heat pipe's capacity in the required temperature range.

Results of the mentioned limits for heat pipe's with L_R=2 are displayed in Figs. 8 and 9.

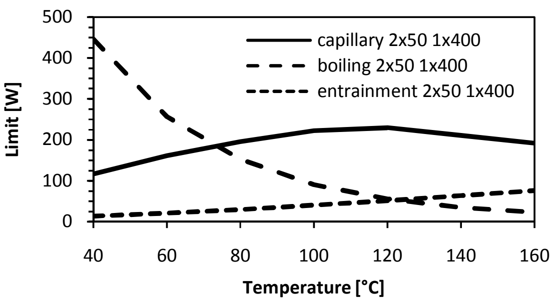

Fig. 8 Heat pipe's limits at an inclination angle of -10º, L_R=2, 2 wraps of 50 mesh and 1 wrap of 400 mesh wicks

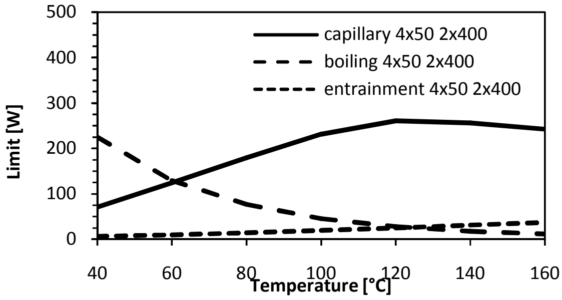

Fig. 9 Heat pipe's limits at an inclination angle of -10º, L_R=2, 4 wraps of 50 mesh and 2 wrap of 400 mesh wicks

Several values of these limits, based on the equations which were mentioned earlier in this work, are displayed in Table III. According to the numerical results it can be seen that for the same inclination angle and defined ratio, L_R, doubling the coarse mesh wraps reduces the heat pipe's performance by approximately half. It can also be seen that at an inclination angle of -10º the heat pipe's performance will be determined by the boiling limit and the entrainment limit.

TABLE III
HEAT PIPE BOILING LIMIT AND ENTRAINMENT LIMIT FOR VARIOUS L_R RATIOS

Operating limit [W]	L_R	n_f	Temperature, T [ºC]		
			80	100	120
Boiling	2	1	152.4	90.1	55.8
Entrainment			29	39.8	51.8
Boiling		2	76.9	45.4	28.1
Entrainment			14.2	19.3	24.9
Boiling	1.5	2	102.8	60.8	37.6
Entrainment			20.5	28	35.9
Boiling	1	2	143.9	85	52.6
Entrainment			28	38.1	49
Boiling		3	98.1	58	35.9
Entrainment			19.4	26.5	34.1
Boiling		4	70.6	41.8	25.8
Entrainment			12.4	17	21.9

From the results above it can be seen that the higher the value of the new defined ratio, L_R, the higher operating limits of the heat pipe at an inclination angle of -10º. For the same number of fine mesh wraps (n_f) and different L_R values it can be seen that the lower L_R the higher operating limits.

VI. SUMMARY AND CONCLUSIONS

Composite wick have been found suitable for heat pipe's proper operation at negative inclination angles, which are restricted by a system's physical dimensions, available space and path.

The heat pipe's operating limits are strongly dependent on the vapor space diameter. A new parameter, L_R, which is the ratio between the coarse mesh wraps and the fine mesh wraps of the composite wick, has been introduced. An optimization of a heat pipe's limits at a known inclination angle was performed based on the new defined parameter.

In order to meet the design requirements of a negative inclination heat pipe, its internal composite structure could be designed using the defined ratio, L_R. This is based on the initial results, which are displayed in this work.

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