Thermal performance of flat-shaped heat pipes using nanofluids

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A B S T R A C T

Analytical models are utilized to investigate the thermal performance of rectangular and disk-shaped heat pipes using nanofluids. The liquid pressure, liquid velocity profile, temperature distribution of the heat pipe wall, temperature gradient along the heat pipe, thermal resistance and maximum heat load are obtained for the flat-shaped heat pipes utilizing a nanofluid as the working fluid. The flat-shaped heat pipe's thermal performance using a nanofluid is substantially enhanced compared with one using a regular fluid. The nanoparticles presence within the working fluid results in a decrease in the thermal resistance and an increase in the maximum heat load capacity of the flat-shaped heat pipe. The existence of an optimum nanoparticle concentration level and wick thickness in maximizing the heat removal capability of the flat-shaped heat pipe was established.

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1. Introduction

Heat pipes have been utilized in heat transfer related applications for many years. Depending on their application area, they can operate over a wide range of temperatures with a high heat removal capability. Heat pipes have been found to be useful in a number of technologies such as electronic cooling, spacecraft thermal control, transportation systems, automotive industry, permafrost stabilization, bio-related applications, solar systems and manufacturing [1–6].

Mathematical and experimental models of the transport mechanisms for flat-shaped heat pipes have been developed by Vafai and co workers [2–6] and a comprehensive analytical solution for a cylindrical heat pipe incorporating the effect of liquid–vapor coupling and non-Darcian transport was presented by Zhu and Vafai [7].

The idea of utilizing nanoparticles within the working fluid of a heat pipe has become a subject of interest in recent years [8–12]. There are a number of studies on thermal conductivity of the nanofluids [13–15]. The nanoparticles within the fluid change its thermal conductivity, viscosity and density. It has been shown experimentally that, for a given concentration level, the thermal conductivity of the nanofluids increases with a decrease in particle diameter [15–17]. However, there are some investigations which do not confirm this enhancement in the thermal conductivity [18,19].

Most of the investigations on the effect of using nanofluids within a heat pipe have been carried out experimentally. The thermal performance of a nanofluid based cylindrical heat pipe was studied analytically by Shafahi et al. [8]. In the current work, analytical models established by Vafai and co workers [3,4] are utilized to study the effect of using a nanofluid within the flat-shaped heat pipes. Disk and rectangular shaped heat pipes have substantial advantages over conventional cylindrical heat pipes. A more flexible geometry, substantially higher heat removal capability, ability to incorporate asymmetrical heating and a built in secondary transport to assimilate overload and prevent or delay a dry out are some of the advantages of the flat-shaped heat pipes analyzed by Vafai and co workers [2–6]. A wide range of nano particles such as silver [20–23], CuO [10,24], diamond [11,25], titanium [9,12], nickel oxide [26] and gold [27] have been used to study the influence of nanofluid on the heat pipe performance. It has been observed experimentally that the heat pipe thermal resistance [11,12,20,22–24,27] and temperature gradient [20,23,25] decrease when using a nanofluid as the working fluid.

In this work, a comprehensive analytical solution for disk and rectangular shaped heat pipes [2,4] has been modified to incorporate the presence of nanoparticles within the working fluid. The liquid pressure and velocity profile, temperature distribution, temperature gradient along the heat pipe, thermal resistance and maximum heat removal capability are obtained for different concentration levels of nanoparticles within flat-shaped heat pipes. Three of the most common types of nanoparticles namely Al2O3, CuO, and TiO2 are considered in this work. There is substantial enhancement in thermal performance of a flat-shaped heat...
2. Analysis and formulation

A simplified schematic of a rectangular or disk-shaped heat pipe's cross section is shown in Fig. 1. It is assumed that vapor and liquid flow are steady and laminar and transport properties of the vapor and liquid are considered to be constant; the vapor injection and suction rates are considered to be uniform in the evaporator [2].

2.1. Pressure distribution

Pressure distribution is obtained by integrating the momentum equation while applying the appropriate boundary conditions [2,4]. This results in the following expressions for the vapor and liquid pressure distribution in disk and rectangular shaped heat pipes respectively.

2.1.1. Disk-shaped heat pipe's vapor pressure distribution

\[
p_{v}^{+}(r^+)=\begin{cases} 
    p_{v}^{+}(0)-\frac{2m_{v}}{\pi}(\frac{\rho_v}{\rho_{nf}})^{\frac{1}{2}}\left(\frac{\rho_{nf}}{\rho_v}\right)^{\frac{1}{2}} \left(\frac{r^+}{r_0}\right)^2 & 0 \leq r^+ \leq \lambda R^+ \\
    -2\left(3-2\ln 3-\frac{3}{4}\right) & \lambda R^+ \leq r^+ \leq R^+ 
\end{cases}
\]

Fig. 1. Simple schematic of rectangular or disk- shaped heat pipe's cross section.
where $\lambda$ is the square root of the ratio of the evaporator area to the total area: $r^* = \frac{r}{r^*}$ and $p^*_v = \frac{p}{p^*}$ for the disk-shaped heat pipe.

2.1.2. Rectangular shaped heat pipe's vapor pressure distribution

The vapor flow in the rectangular shaped heat pipe does not have a symmetrical velocity profile. The maximum vapor velocity position moves towards the lower plate due to vapor injection from the upper wall. An approximate solution for $\bar{f}$ is obtained using Bernoulli's equation [2]:

$$
\frac{df^+(x^*)}{dx^*} = \begin{cases}
-\frac{1}{2} (1 - \lambda \beta^+ (x^*) + 5 \frac{\beta^+ (x^*)}{R e_h} \frac{1}{(1 - \lambda \beta^+ (x^*)) - \frac{1}{2} \frac{1}{1 - \lambda \beta^+ (x^*)}} & 0 < x^* < \lambda R^+
\end{cases}
$$

where the dimensionless coordinate for rectangular shaped heat pipe's vapor phase is $x^* = \frac{x}{L_1}$ and $U_{o,w}$ is the maximum vapor velocity.

As such the vapor pressure distribution can be presented as:

$$
\Delta p_v^+(x^*) = \begin{cases}
\frac{\nu_v^* R e_h}{4 \nu} \left[ \left( \frac{1}{\beta^*} \right)^2 + 2 \ln \left( \frac{1}{\beta^*} \right) \right] & 0 < x^* < \lambda R^+
\end{cases}
$$

where dimensionless vapor pressure is $p^*_v = \frac{p}{p^*}$ and $\lambda$ is the ratio of the evaporator length to the heat pipe length for the rectangular shaped heat pipe.

The liquid pressure distribution for the rectangular and disk-shaped heat pipes can be presented as [2,4]:

2.1.3. Disk-shaped heat pipe's liquid pressure distribution

$$
p_v^*(0) = \nu_v^* \frac{R e_h}{2 \nu} \left( \frac{1}{\beta^*} \right)^2 + 2 \ln \left( \frac{1}{\beta^*} \right) \quad 0 < r^* < \lambda R^+
$$

2.1.4. Flat-shaped heat pipe's liquid pressure distribution

$$
\Delta p_v^+(x^*) = \begin{cases}
\mu^* \left( \lambda (1 - \lambda) x^* \right)^2 & 0 < x^* < \lambda h^+
\end{cases}
$$

where $\mu^* = \frac{\mu}{\mu^*}$; the dimensionless coordinate for the rectangular shaped heat pipe's liquid phase is $x^* = \frac{x}{D_2}$ and $U_{o,d}$ is maximum liquid phase velocity. It can be seen that liquid pressure is related to the liquid viscosity which is affected when using nanoparticles within the working fluid. The classical Brinkman model [28] is used to obtain the nanofluid viscosity:

$$
\mu_{nf} = \frac{\mu_{nf}}{(1 - \phi)^{2.5}}
$$

where $\mu_{nf}$ is the viscosity of the base fluid which is water for our investigation. Nanofluid's density is calculated based on a simple representation of the particle and fluid phases proposed by Pak and Cho [29]:

$$
\rho_{nf} = \rho_p \phi + (1 - \phi) \rho_f
$$

where $\rho_{nf}$, $\rho_p$, and $\rho_f$ are the nanofluid, particle and base fluid densities respectively.

2.2. Velocity distribution

Using Darcy's and applying the mass balance for the lower and upper wicks the liquid velocity distribution for the rectangular and disk-shaped heat pipes can be presented by [2,4]:

$$
u_r^+(r^*) = \begin{cases}
\frac{v_r^* - v_s^*}{d_{o,w} h_w} r^* & 0 < r^* < \lambda R^+
\end{cases}
$$

where $\rho_{w}^* = \rho_{nf} \rho_{p}$.

Liquid velocity distribution for the rectangular shaped heat pipe:

$$
u_r^+(x^*) = \begin{cases}
\frac{h_w^* \mu_v^* v_v^* (x^* - x^* \lambda)}{h_w^* \mu_v^* v_v^* (x - x \lambda)} & 0 < \lambda < \lambda R^+
\end{cases}
$$

2.3. Temperature distribution

The wall temperature profile is obtained based on the assumption of uniform temperature along the condenser and evaporator sections. Based on simple energy balances, the condenser and evaporator wall temperatures as well as the vapor temperature can be respectively presented as:

Condenser wall temperature:

$$
T_{wall,c} = T_b + \frac{Q}{\lambda A_f}
$$

Evaporator wall temperature:

$$
T_{wall,e} = T_v + \frac{Q}{\lambda A_f} \left( \frac{h_w}{h_{wall}} + \frac{h_w}{h_{nf}} \right)
$$

where $h_{wall}$ and $h_w$ are the wall and wick thicknesses respectively.

Vapor temperature:

$$
T_v = T_b + \frac{Q \left( \frac{1}{\lambda h_f} \frac{h_{wall}}{h_{wall}} + \frac{h_w}{h_{nf}} \right)}{\lambda h_f}
$$

The heat pipe's temperature distribution is dependent on the wick thermal conductivity which can be presented as [30]:

$$
k_{eff} = \frac{k_w (k_w + k_i) - (1 - \epsilon) (k_w - k_i)}{[k_w + k_i] + (1 - \epsilon) [k_w - k_i]}
$$

where $k_i$ and $k_w$ are solid and liquid conductivities. For this work $k_i$ is replaced with $k_{nf}$. It should be noted that $k_{eff}$ increases with an increase in thermal conductivity of nanofluid. The nanofluid conductivity is obtained from Yu and Choi [31] model that can be presented as:

$$
k_{nf} = \frac{k_{pe} + 2 k_i + 2 (k_{pe} - k_i) (1 + \beta)^3}{k_{pe} + 2 k_i - (k_{pe} - k_i) (1 + \beta)^3}
$$

To account for the effect of nanolayer, solid particle conductivity is modified as
\[ k_{pe} = \frac{2(1 - \alpha) + (1 + \beta)^3(1 + 2\alpha)\alpha}{-(1 - \alpha) + (1 + \beta)^3(1 + 2\alpha)} k_p \]

where \( k_p \) and \( k_{layer} \) are the thermal conductivities of the nanoparticle and nanolayer, respectively.

**2.4. Maximum heat transfer capability**

For a heat pipe under steady-state operation, capillary pressure of the wick establishes a stable circulation for the working fluid. The maximum heat transport capillary limit of the heat pipe is achieved when the total pressure loss reaches the maximum capillary pressure of the porous wick which can be presented as:

\[ T_0 - T_L = \frac{Q}{h} \]

Fig. 2. Liquid pressure distribution for different concentration levels of CuO nanoparticles within the heat pipe; \( Q = 9 \, \text{kW} \); (a) disk-shaped heat pipe and (b) rectangular shaped heat pipe.

Fig. 3. Liquid velocity profiles for various concentration levels of CuO nanoparticles within the flat heat pipes; (a) disk-shaped heat pipe and (b) rectangular shaped heat pipe; \( Q = 9 \, \text{kW} \).

Fig. 4. Heat pipe wall temperature distribution for different nanoparticles concentration levels; \( Q = 9 \, \text{kW} \); \( d_p = 20 \, \text{nm} \); (a) disk-shaped heat pipe, and (b) rectangular shaped heat pipe.

Fig. 5. Temperature difference between evaporator and condenser in disk-shaped heat pipe for different CuO nanoparticle concentration levels, (a) \( d_p = 10 \, \text{nm} \), (b) \( d_p = 20 \, \text{nm} \) and (c) \( d_p = 40 \, \text{nm} \).
\[ \Delta P_{\text{cap}} = \Delta P_{v,\text{total}} + \Delta P_{l,\text{total}} + \Delta P_{g,\text{total}} \]  
\[ (\Delta P_{\text{cap}})_{\text{max}} = \frac{2\sigma_l}{r_c} \]  

where \( \sigma_l \) is the liquid surface tension, and \( r_c \) is the effective pore radius of the wick. The total vapor, liquid, and gravitational pressure drops should be less than the maximum capillary pressure head. Neglecting the gravitational head, Eq. (17) can be represented as:

\[ (\Delta P_{\text{cap}})_{\text{max}} \geq \Delta P_{v,\text{total}} + \Delta P_{l,\text{total}} \]  

3. Results and discussion

The analytical models for disk and rectangular shaped heat pipes [2,4] were utilized to study the influence of using a nanofluid on the thermal performance of these heat pipes. Results are obtained for disk and rectangular shaped heat pipes which include nanoparticles within the working fluid. The diameter of the disk-shaped heat pipe is taken equal to the length of the rectangular shaped heat pipe which is 0.46 m for this work. Other nominal dimensions are taken from Vafai and Wang [2] and Vafai et al. [4] works. Three of the most common nanoparticles namely, Al₂O₃, CuO, and TiO₂ with a range of diameters (10, 20 and 40 nm) are utilized to form the nanofluid. The influence of the nanofluid on the liquid pressure, temperature distribution, liquid velocity, temperature gradient, thermal resistance, heat pipe size, and maximum heat removal capability of the flat-shaped heat pipes is investigated.

Fig. 2 displays the liquid pressure distribution for disk and rectangular heat pipes for a range of CuO based nanofluid concentration levels. Initially, the liquid pressure drop decreases by utilizing the nanoparticles within the water due to an increase in density. After reaching a critical concentration level a reversal effect occurs resulting in an increase in the pressure drop as the increase in viscosity of the nanofluid becomes more prominent. The competing effects of the density and viscosity of the nanofluid creates an optimum level of nanoparticle concentration in minimizing the pressure drop for the flat heat pipes.

The influence of nanofluid concentration level on the liquid velocity profile is shown in Fig. 3. An increase in the concentration level decreases the velocity in both disk and rectangular shaped heat pipes. It can be seen through Eqs. (8) and (9) that liquid velocity is inversely proportional to either the density or the viscosity of the liquid.

Fig. 4 displays the wall temperature distribution for flat and disk-shaped heat pipes utilizing the CuO based nanofluid as the working fluid. The input heat and the nanoparticle diameter are 9 kW and 20 nm, respectively. As seen in Fig. 4, the evaporator
temperature drops in the presence of nanoparticles. An increase in
the concentration level of nanoparticles increases the thermal con-
ductivity of the nanofluid, thus further decreasing the evaporator
temperature.

Figs. 5 and 6 show the reduction in the evaporator and con-
denser temperature difference for different nanoparticle concen-
tration levels for various heat loads. The effect of variations in
nanoparticle diameter is also displayed in Figs. 5 and 6. As can
be seen the smaller nanoparticle or the larger concentration level
the more pronounced the reduction in the temperature difference
between the condenser and the evaporator.

Thermal resistance of the flat-shaped heat pipes, defined as
\[ R_T = \frac{T_e - T_c}{Q} \]
becomes noticeably smaller when a nanofluid is utilized
as seen in Figs. 7 and 8. As seen in these figures, the thermal resis-
tance of a flat-shaped heat pipe can be reduced up to 83% of its ini-
tial resistance.

Figs. 9 and 10 demonstrate that using the nanoparticles pro-
vides the possibility of a reduction in the size of flat-shaped heat
pipe while maintaining the same heat load removal capability.
The results are shown for different concentration levels of \( \text{Al}_2\text{O}_3 \),
\( \text{CuO} \), and \( \text{TiO}_2 \) with different diameters. For these figures the
applied heat input was kept at 9 kW and the evaporator tem-
perature was kept at 90 °C. As can be seen the smallest particle size
results in the largest reduction in the size of the heat pipe. For
\( \text{CuO} \) based nanofluid a 30% and 20% reduction in size can be real-
ized for the disk and rectangular shaped heat pipes respectively.

Fig. 8. Effect of nanofluids (\( d_p = 20 \) nm) on rectangular shaped heat pipe thermal resistance for various heat inputs: (a) \( \phi = 1\% \); (b) \( \phi = 2\% \); (c) \( \phi = 3\% \); (d) \( \phi = 4\% \).

Fig. 9. Percentage reduction level of a disk-shaped heat pipe when using different size nanoparticles and different concentration levels, \( T_{\text{evaporator wall}} = 90 \) °C and \( Q = 9 \) kW.

Fig. 10. Percentage reduction level of a rectangular shaped heat pipe when using different size nanoparticles and different concentration levels, \( T_{\text{evaporator wall}} = 90 \) °C and \( Q = 9 \) kW.
The effect of changing the wick thickness and nanoparticle concentration level on the maximum heat removal capability of a flat-shaped heat pipe is shown in Fig. 11. The existence of an optimum point for both wick thickness and nanoparticle concentration level can be seen in this figure. The existence of an optimum thickness for the wick was established earlier in the work of Vafai et al. [4]. The appearance of an optimum nanoparticle concentration level based on the competing effects of density and viscosity of the nanofluid level was also established in the work of Shafahi et al. [8] for cylindrical heat pipes.

4. Conclusions

The performance of flat-shaped heat pipes utilizing nanoparticles within the working fluid was investigated in this work. Some of the more widely used nanoparticles, such as Al₂O₃, CuO and TiO₂ with a range of nanoparticle diameters were considered. The flat-shaped heat pipe's liquid pressure, liquid velocity profile, temperature distribution, temperature gradient and maximum heat removal capability were investigated. Furthermore, the possibility of reducing the flat-shaped heat pipe's dimension using a nanofluid, was explored. Results show that the presence of nanoparticles in the working fluid leads to a reduction in the speed of the liquid, smaller temperature difference along the heat pipe and the possibility of reduction in size under the same operational conditions. Using a nanofluid reduces the thermal resistance of the flat-shaped heat pipe similar to what has been observed experimentally. The maximum heat removal capability of the flat-shaped heat pipe was displayed for a range of wick thicknesses and nanoparticles concentration levels. The existence of an optimum nanoparticle concentration level and wick thickness in maximizing the heat removal capability of the flat-shaped heat pipe was established.

References