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THERMAL PERFORMANCE OF OSCILLATING HEAT PIPES WITH NANOFLUID: A THEORETICAL STUDY

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ABSTRACT

This numerical study is performed to investigate heat transfer performance and effect of nanofluids on U-shaped oscillating heat pipes. Pure water is employed as the base fluid while Al_2O_3 with two different particle sizes i.e., 47nm and 38.4 nm is used as nanoparticle. The results show that nanofluid has significant effect on heat transfer enhancement of the system and with increasing volume fraction and decreasing particles diameter the enhancement intensifies.

INTRODUCTION

Due to advances in high performance electronic packages, there is a need for better thermal management. Oscillating Heat Pipe (OHP) is one of the most efficient heat transfer devices for removal of high localizes heat fluxes which provide a necessary level of temperature uniformity across the electronic components. Due to attractive aspect of OHPs, extensive investigations have been conducted in this area which resulted in a better understanding of heat transfer and fluid flow mechanisms in these devices.

In the recent years, nanofluids have been widely used as an alternative working fluid for various applications such as electronic cooling [1-3], heat exchangers [4] as well as heat pipes. Some researchers experimentally studied the effect of nanofluids on thermal performance of OHPs [5-7]. For example, Lin et al. [5] conducted an experimental investigation in an OHP charged with aqueous silver nanofluid with particle diameter of 20 nm and showed significant reduction in thermal resisteance. Qu et al. [6] investigated the effect of spherical 56 nm alumina nanoparticle on heat transport capability of an OHP and found that nanofluid can enhance the heat transfer capability of the system. Qu and Wu [7] experimentally investigated thermal performances of two OHPs charged with Al_2O_3 and SiO_2 water nanofluids. The results indicated that alumina nanofluid enhances heat transfer of OHP while silica nanofluid instead of pure water deteriorated the thermal performance of the system. It appears that there is no theoretical model for calculating thermal performance of OHPs with nanofluid as working fluid. Therefore, the objective of this paper is to conduct a theoretical analysis predicting the heat transfer enhancement of OHPs and the effect of nanoparticles volume concentration and size on thermal performance of the system.

NOMENCLATURE

A	Area, m^2
c_p	Specific heat at constant pressure, J/kg-K
c_v	Specific heat at constant volume, J/kg-K
d	Diameter, m
h_{lsen}	Convection heat transfer coefficient, W/m^2-K
k	Thermal conductivity, W/m^2-K
L	Length, m
m	Mass of vapor plugs, kg
p	Vapor pressure, Pa
R	Gas constant of vapor, kJ/kg-K
t	Time, s
T	Temperature of liquid slug, K
x_p	Displacement of liquid slug, m
Greek Symbols	
γ	Ratio of specific heats
ρ	Density, kg/m^3
ϕ	Volume fraction
τ_p	Shear stress, N/m^2
μ	Viscosity
Subscripts	
0	Initial condition
c	condenser
e	evaporator
eff	Effective nanofluid
f	Base fluid
l	Liquid
p	Nanoparticle
v	Vapor plug
w	Wall

PHYSICAL MODEL

Figure 1 shows the physical model of a U-shaped minichannel with its two ends sealed which is considered as the building block of an OHP. The length of each evaporator section is L_e and its temperature is maintained at T_e. The condenser section is located between two evaporator sections with a length of L_c and temperature of T_c. The length of the liquid slug which depends on the filling ratio is L_p. The displacement of the liquid slug is represented by x_p. When the liquid slug is exactly in the middle of the U-shaped miniature channel, x_p is zero. When the liquid slug shifts to the right side, x_p is positive; when it moves to the left side x_p is negative. It is assumed (1) shear stress at liquid-vapor interface is negligible, (2) heat conduction in the liquid slug is assumed to be 1-D in the axial direction and exchange of heat between the liquid and wall is considered by a convective heat transfer coefficient, (3) the liquid is incompressible and the vapor is saturated and behaves as an ideal gas, (4) evaporative and condensation heat transfer coefficients are assumed to be constants, (5) the U-shaped minichannel is assumed to be a straight pipe and the effect of pressure loss in bend is considered using an empirical correlation, (7) Nanofluid is treated single phase and its properties are temperature dependent.

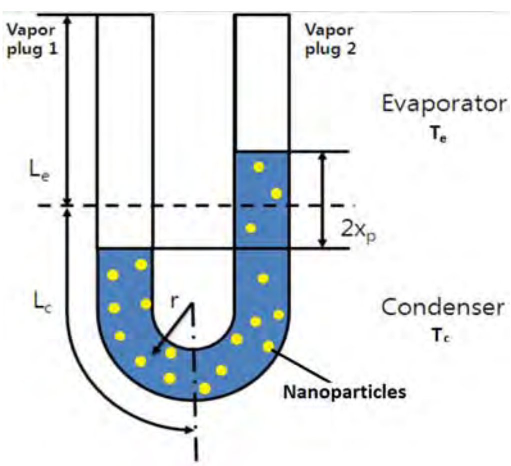


Figure 1 Configuration of oscillating heat pipe

The acceleration of the liquid slug is related with pressure, gravity and shear stress. Thus, the momentum equation is [8]:

$$AL_p\rho_{eff}\frac{d^2x_p}{dt^2}=[(p_{v1}-p_{v2})-\Delta p_b]A-2\rho_{eff}\,gAx_p-\pi dL_p\tau_p \qquad (1)$$

where Δp_b is the pressure loss at the bend [8] and ρ_{eff} is the effective density of nanofluid and can be calculated using correlation developed by Khanafer and Vafai [9]:

$$\rho_{eff}=1001.064+2738.6191\,\varphi_p-0.2095\,T \qquad 0\le\varphi_p\le 0.04,\ 5^oC\le T\le 40^oC \qquad (2)$$

where φ_p is the volume fraction of nanoparticles and τ_p is the shear stress which is function of Reynolds number, density and velocity of flow [8]. The temperature and particle size dependent viscosity of nanofluid is calculated by one of the most accurate relations developed by Khanafer and Vafai [9]:

$$\mu_{eff}=-0.4491+\frac{28.837}{T}+0.574\varphi_p-0.1634\varphi_p^2+23.053\frac{\varphi_p^2}{T^2}+0.0132\varphi_p^3$$
$$-2354.735\frac{\varphi_p}{T^3}+23.498\frac{\varphi_p^2}{d_p^2}-3.0185\frac{\varphi_p^3}{d_p^2} \qquad (3)$$

$$1\%\le\varphi_p\le 9\%,\ 20^oC\le T\le 70^oC,\ 13nm\le d_p\le 131nm$$

where d_p is the diameter of nanoparticles.

The energy equations of the two vapor plugs are [8]:

$$\frac{d(m_{v1}c_vT_{v1})}{dt}=c_pT_{v1}\frac{dm_{v1}}{dt}-p_{v1}\frac{\pi}{4}d^2\frac{dx_p}{dt} \qquad (4)$$

$$\frac{d(m_{v2}c_vT_{v2})}{dt}=c_pT_{v2}\frac{dm_{v2}}{dt}+p_{v2}\frac{\pi}{4}d^2\frac{dx_p}{dt} \qquad (5)$$

The vapor plug can be assumed to be an ideal gas. Thus, the equation of state can be presented by:

$$p_{v1}(L_e+x_p)\frac{\pi}{4}d^2=m_{v1}RT_{v1} \qquad (6)$$

$$p_{v2}(L_e-x_p)\frac{\pi}{4}d^2=m_{v2}RT_{v2} \qquad (7)$$

Considering the initial conditions, the masses and the temperatures of the two vapor plugs can be obtained:

$$m_{v1}=\frac{\pi d^2p_0}{4RT_0}\left(\frac{p_{v1}}{p_0}\right)^{\frac{1}{\gamma}}(L_e+x_p) \qquad (8)$$

$$m_{v2}=\frac{\pi d^2p_0}{4RT_0}\left(\frac{p_{v2}}{p_0}\right)^{\frac{1}{\gamma}}(L_e-x_p) \qquad (9)$$

$$T_{v1}=T_0\left(\frac{p_{v1}}{p_0}\right)^{\frac{(\gamma-1)}{\gamma}} \qquad (10)$$

$$T_{v2}=T_0\left(\frac{p_{v2}}{p_0}\right)^{\frac{(\gamma-1)}{\gamma}} \qquad (11)$$

where T_0 and P_0 are the initial temperature and the pressure, respectively [8].

The temperature distribution of the liquid slug can be obtained by solving the energy equation [8]:

$$\frac{\rho_{eff}C_{p,eff}}{k_{eff}}\frac{dT_l}{dt}=\frac{d^2T_l}{dx_p^2}-\frac{h_{lsen}\pi d}{k_{eff}A}(T_l-T_w) \qquad (12)$$

The specific heat of nanofluid can be calculated using the following equation [5]:

$$C_{p,eff}=\frac{1}{\rho_{eff}}\left[\left(1-\varphi_p\right)\rho_fC_{p,f}+\varphi_p\rho_pC_{p,p}\right] \qquad (13)$$

where c_p specific heat and indices of p and f refer to nanoparticles and base fluid. The thermal conductivity is calculated by following equation [9].

$$\frac{k_{eff}}{k_f} = \left\{ \begin{array}{c} 0.9843 + 0.398\varphi_p^{\ 0.7383} \left(\dfrac{1}{d_p(nm)} \right)^{0.2246} \left(\dfrac{\mu_{eff}(T)}{\mu_f(T)} \right)^{0.0235} \\ -3.9517\dfrac{\varphi_p}{T} + 34.034\dfrac{\varphi_p^{\ 2}}{T^3} + 32.509\dfrac{\varphi_p}{T^2} \end{array} \right\} \quad (14)$$

$1\% \le \varphi_p \le 10\%$, $20^o C \le T \le 70^o C$, $11nm \le d_p \le 150\,nm$

where k_f is thermal conductivity of base fluid. The initial and boundary conditions for Eq. (18) are:

$$T = T_0, \quad t = 0, \quad 0 < x_p < L_p \qquad (15)$$

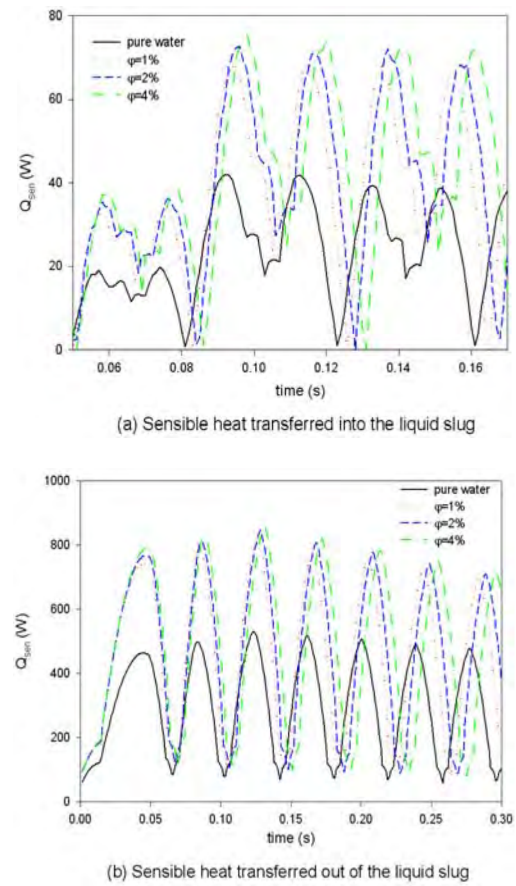
$$T = T_{v1}, \quad x_p = 0 \qquad (16)$$

$$T = T_{v2}, \quad x_p = L_p \qquad (17)$$

The wall temperature of the tube can be either T_e or T_c, depending on the displacement of the liquid slug [8]. The sensible heat transfer in and out of the liquid slug as well as heat transfer rate in the two vapors are can be obtained the method mentioned in Ref [8]. Finally, the governing equations with their associated boundary conditions were solved using implicit finite difference method and Tridiagonal Matrix Algorithm. For more information an interested may refer to Ref [8].

RESULTS AND DISCUSSIONS

The present model is used to simulate the oscillating heat pipe ith the following parameters: L_e=0.1m, L_c =0.1m, L_p =0.2m, d= 3.34 mm, T_e=396 K, T_c=293 K, and h_e=h_c = 200W/m^{2}K. Figure 2 shows the comparison of sensible heat transfer transferred into and out of the liquid slug for different volume fractions of nanoparticles as well as pure water. There is a delay of the phase and a strong enhancement of sensible heat transfer in and out of the liquid slug when using nanofluid instead of pure water as working fluid of heat pipe. The phase delay in sensible heat is due to higher viscosity and mass of nanofluid compared to pure water which cause the velocity of liquid slug decrease so we observe a phase delay in sensible heat. The enhancement in sensible heat is due to enhanced thermophysical properties of nanofluid especially thermal conductivity. With increasing volume fraction of nanoparticles, the thermal conductivity of nanofluid increases which leads to enhancement in sensible heat of the system. Due to enhanced oscillatory motion of the liquid slug, the total sensible heat transfer rate is increased from 38.985 W for pure water case to 69.415 W for nanofluid case with volume fraction of 4%. At this point it is important to mention that the influence of nanoparticles elucidates two opposing effects on the heat transfer coefficient and total thermal resistance: an undesirable effect promoted by high level of viscosity experienced at high volume fractions of nanoparticles and a favorable effect that is driven by the presence of high thermal conductivity of nanoparticles. In other words, the addition of nanoparticles to the base fluid enhances the thermal conduction and with increasing particle volume fraction for a given particle size, the enhancement increases. The enhancement of the thermal conduction should increase the convective heat transfer coefficient. However, the nanofluid viscosity increases with increasing particle volume fraction, which should result in a reduction in velocity and convection and consequently thicker boundary layer thicknesses on the heat pipe wall. The growth in thermal boundary layer thickness is responsible for the lesser temperature gradients at the wall of system which lowers convective heat transfer coefficient. As shown clearly in Fig. 2, increasing nanoparticle volume fraction cause thermal performance enhancement. This indicates that under the condition of this work, the positive effect of the thermal conduction enhancement outweighs the negative effect of viscosity enhancement. The comparison of latent heat transfer of vapor first plug for different cases is shown in Fig. 3. It is seen that the delay of the phase of the oscillation also results the delay of the phase of evaporation heat transfer but the amplitude of condensation heat transfer does not increase much.



(a) Sensible heat transferred into the liquid slug

(b) Sensible heat transferred out of the liquid slug

Fig. 2 Effect of nanofluid on sensible heat transferred into the liquid slug for d_p=38.4nm

Figure 4 shows the comparison of the sensible heat transfer for different nanoparticles diameters. With decreasing the particle diameter the phase of sensible heat advances because of lower velocity of liquid plug for smaller particle diameter. Furthermore, it can be seen that there is an increase of the sensible heat transfer when the size of nanoparticles increases.

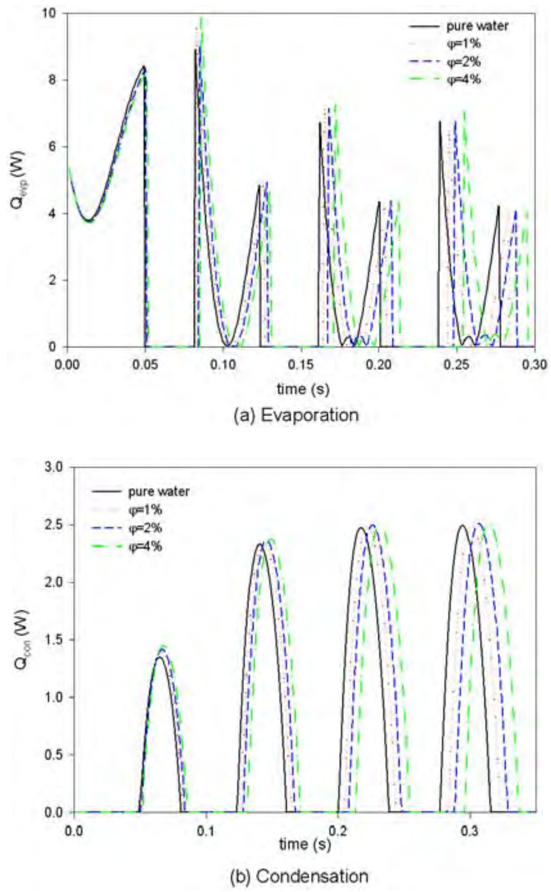


Fig. 3 Effect of latent heat of vapor plug 1 for different volume fraction at d_p=38.4

This is due to the fact that smaller size nanoparticles lead to higher effective thermal conductivity of nanofluid. This is due to aggregation of nanoparticles and stronger Brownian motion at smaller nanoparticles diameters, which lead to higher thermal conductivity of nanofluids [1]. At this point, it should be noted that the Brownian motion is one of the main mechanisms of thermal conductivity enhancement and with decreasing the particles diameter, the effect of Brownian motion increases due to improvement of micro-convection around nanoparticles. Secondly, the formation of liquid layers around nanoparticles is another mechanism of enhancement of thermal conductivity. With decreasing particles diameter, the specific area of nanoparticles increases which results in enhancement in thermal conductivity due to nanolayers. Another reason for higher thermal conductivity of Al_2O_3 water nanofluid for smaller size nanoparticles is that the thermal energy transfer is dependent on surface area and smaller particles of same volume fraction provide more surface area for the transfer of thermal energy. Understanding the effects of which mechanism or mechanisms might be stronger for the results presented in this paper will require significant future work, including molecular dynamic simulation or developing complete theoretical models taking into account the effect of all mentioned mechanisms. Therefore, for Al_2O_3 water nanofluid with decreasing the size of nanoparticles the effect of aggregation, nanolayer formations, Brownian motion and energy transfer increase which lead to enhancement of thermal conductivity and consequently heat transfer coefficient [1]. Figures 5 shows the comparison of latent heat transfer for different nanoparticle diameters. The latent heat transfer is constant for different particle diameters while there is a delay between two nanoparticle sizes. The effect of particles diameter and volume fraction on total heat transport rate and contribution of latent and sensible heat transfer of OHP is presented in Table 1. From this table, it can be found that nanofluid has a major role in increasing the heat transfer rate. As seen, with increasing the volume fraction and decreasing nanoparticles size, the sensible heat transfers increases. As mentioned earlier, these enhancements are due to enhancement in thermophysical properties of nanofluid.

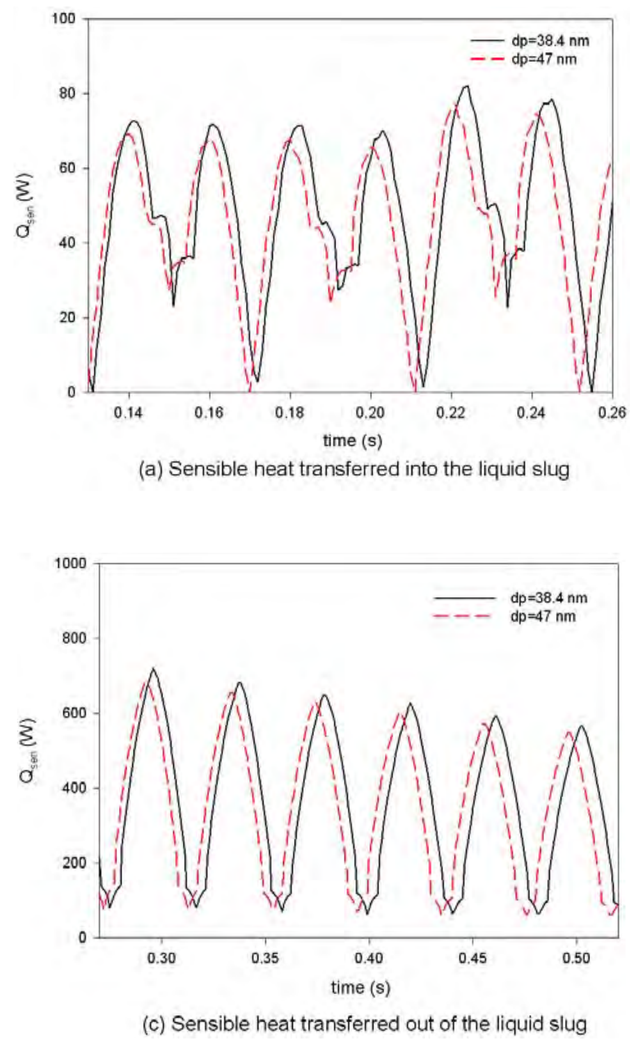


Fig. 4 Effect of nanoparticle size on sensible heat of the liquid slug

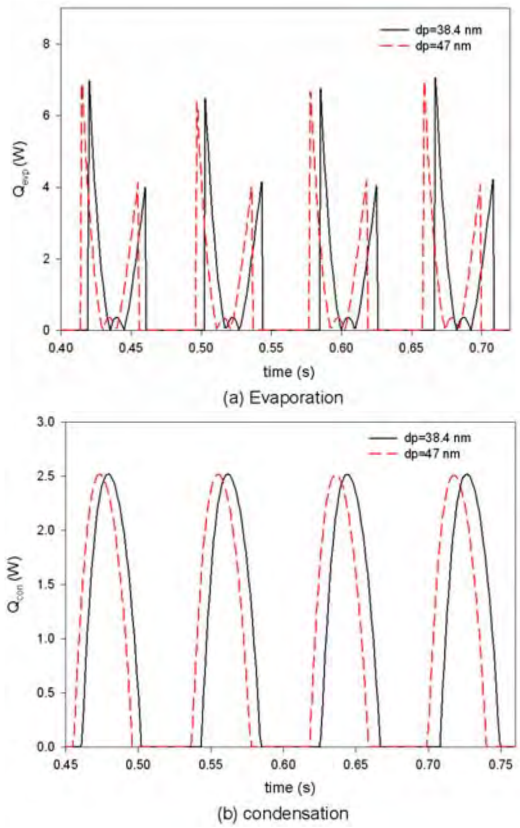


Fig. 5 Effect of nanoparticle size on latent heat vapor plug 1

Table 1. Total heat transport rate and contribution of latent and sensible heat transfer

dp (nm)	ϕ	$\overline{Q_{evp}}$	$\overline{Q_{h,s}}$	$\overline{Q_t}$	$\overline{Q_{h,s}}/\overline{Q_t}$
38.4	0%	0.917	38.985	39.902	97.701
	1%	0.927	64.203	65.130	98.577
	2%	0.929	66.897	67.827	98.630
	4%	0.940	69.415	70.355	98.662
47	0%	0.917	38.985	39.902	97.701
	1%	0.923	60.207	61.131	98.489
	2%	0.926	61.899	62.825	98.525
	4%	0.934	65.404	66.399	98.592

CONCLUSIONS

Nanofluid effects on oscillatory flow and heat transfer of a pulsating heat pipe are investigated by comparing displacement of liquid slug, temperature and pressure of vapor plugs, as well as latent and sensible heat transfer. It is concluded that nanofluid has significant effect on thermal performance of the system and with increasing the volume fraction the enhancement intensifies. Furthermore, the size of nanoparticles influences the magnitude of sensible heat and smaller nanoparticles leads to larger sensible heats. With increasing the particle size, the amplitude of oscillation is nearly constant while the frequency of the oscillation increases.

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