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USING MULTI-WALL CARBON NANOTUBE (MWCNT) BASED NANOFLUID IN THE HEAT PIPE TO GET BETTER THERMAL PERFORMANCE*

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Abstract– In this study, thermal performance of a cylindrical heat pipe is investigated numerically. Three different types of water based nanofluids, namely (Al_2O_3+Water), (Diamond+Water) and (Multi-Wall Carbon Nano tube (MWCNT) +Water) have been used. The influence of using the simple nanofluids and MWCNT nanofluid on the heat pipe characteristics such as liquid velocity, pressure profile, temperature profile, thermal resistance and heat transfer coefficient of heat pipe have been studied. A new correlation developed by Bakhshan and Saljooghi, for viscosity of nanofluids has been implemented. The results show good agreement with the available analytical and experimental data. Also, the MWCNT based nanofluid has lower thermal resistance, higher heat transfer coefficient and lower temperature difference between evaporator and condenser sections, so it has good thermal specifications as a working fluid for use in heat pipes. The prepared code has capability for parametric studies also.

Keywords– Heat Pipe, nanofluid, carbon nanotube, thermal performance

1. INTRODUCTION

Heat pipe is a device for transferring heat with high heat conduction ratio. It transfers heat energy by vaporization and condensation of a fluid with little temperature reduction. A heat pipe has three sections in general including evaporator, adiabatic and condenser sections. When the heat reaches evaporator, the fluid evaporates and it forms a different pressure in the pipe. The different fluid pressure causes the vapor to move through the pipe and reach condenser. In condenser section, the vapor condenses and its latent heat is released and then the fluid returns from inside the wick by capillary pressure to evaporator section. This study has been conducted to show the effect of using three water based nanofluids (water+ Al_2O_3), (carbon nanotube+water) and (water+diamond) on the heat operation of heat pipe. We also applied a new correlation for viscosity of nanofluids developed by Bakhshan et al. [1, 2].

A mathematical model of a cylindrical heat pipe using TiO_2, CuO, and Al_2O_3 nanofluids was solved analytically by Shafahi et al., [3]. The thermal performance of cylindrical heat pipes utilizing the above nanofluids as the working fluid has been investigated. They obtained the velocity, pressure, temperature and maximum heat transfer limit for different nano- particle concentration levels and sizes. Their results show that the thermal performance of a heat pipe is improved and temperature gradient along the heat pipe and thermal resistance across the heat pipe are reduced when nanofluids are utilized as the working fluid.

Kyu Hyung Do et al. [4] investigated a mathematical model that was developed for quantitatively evaluating the thermal performance of a water-based Al_2O_3 nanofluid heat pipe with a rectangular grooved wick. Their results show that, thin porous coating layer formed by nanoparticles suspended in nanofluids

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is a key effect of the heat transfer enhancement for the heat pipe using nanofluids. Also, the effects of the volume fraction and the size of nanoparticles on the thermal resistance of nanofluid heat pipes were studied. Furthermore, the experimental results showed that the thermal resistance of the nanofluid heat pipe tended to decrease with increasing the nanoparticle size.

Zhenhua et al. [5] carried out an experimental study on the application of nanofluids in an inclined miniature grooved heat pipe using water-based CuO nanofluid as the working fluid. The working fluid consisted of CuO nanoparticles with diameter of 50 nm and pure water. The experimental results showed that with increasing of the inclination angle, the heat transfer coefficient of the evaporator and the condenser section will increase. The evaporation heat transfer coefficient and the condensation heat transfer coefficient of the inclined heat pipes increases about 60-100% compared with that of the horizontal heat pipe using the nanofluid. Furthermore, the experimental results showed that the evaporation and the condensation heat transfer coefficient of the inclined heat pipes increase about 60-80% compared with that of the horizontal heat pipe using water. They considered that the maximum heat fluxes of the inclined heat pipes vary little compared with that of the horizontal heat pipe.

Zhenhua et al. [6] carried out an experimental study on the application of nanofluids in a heat pipe using de-ionized water-based CuO nanofluid as the working fluid. The nanofluid consisted of CuO nanoparticles with a diameter of 50 nm. The wall temperature distributions and the total heat resistance of the heat pipe utilizing a nanofluid as the working fluid and effect of the nanoparticles mass concentration on the heat transfer in the evaporator and condenser have been investigated. The experimental results showed that average wall temperatures of the heat pipe for the nanofluid are lower than those for de-ionized water and both the heat transfer coefficients of the evaporator and the condenser increases with the increase of the mass concentration. Also, they showed that the maximum heat flux increases with the increase of the mass concentration when the mass concentration is less than 1.0 wt.% and the total heat resistance of the heat pipe decreases when substituting the 1.0 wt.% CuO nanofluid for de-ionized water.

Mousa [7] investigated the effect of nanofluid concentration on the performance of circular heat pipe. The effect of filling ratio, volume fraction of nano-particle in the base fluid and heat input rate on the thermal resistance is investigated. Their results showed that by increasing the concentration of the nanofluid, the thermal performance of heat pipe can be decreased.

Nat Thacayapong et al. [8] studied the capillary pressure effect on the performance of a heat pipe numerically with finite element method. They simulated the heat transfer and fluid flow at two-dimensional and steady state, and showed the capillary pressure gradient inside the wick at the end of the evaporator section was very large due to a result of fast liquid motion, thus providing efficient heat transfer through convection.

Ji Li and Peterson [9] analyzed the heat pipe evaporator with a fully saturated wick operation. They used a practical quasi three-dimensional numerical model in a square flat evaporator of a loop heat pipe with a fully saturated wicking structure. They solved the governing equations for the heat and mass transfer (continuity, Darcy and energy) in 3D space. By comparing with the experimental test results, they confirmed their simulation results of the local heat transfer mechanism.

Hao Peng et al. [10] studied the oscillating heat pipes numerically using finite element modeling. An Euler predictor-corrector method with convergence check was used to solve the oscillation and the temperature distribution within each fluid plug. They showed that OHP is a parametrically excited nonlinear thermo-mechanical system and latent heat transfer provides the driving force for the oscillation, and sensible heat transfer induced by forced convection contributes more than 80% of the total heat transfer rate.

Yan-Jun Chen et al. [11] tested the use of nanofluids in the copper wire-bonded flat heat pipe experimentally. They used the water, ethanol and nanofluids as working fluids and studied the influences of the working fluid, the operating temperature or operating pressure, the wire diameter and the space gap between two wires on the thermal performances of the heat pipe. They showed, using nanofluid as working fluid can improve the heat transfer performance of the heat pipe and the best heat transfer performance of heat pipe is achieved at the concentration of 1.0 wt.%.

Asirvatham et al. [12] used silver-water nanofluid in the screen mesh wick heat pipes. They tested the heat pipe for heat inputs ranging from 20W to 100W in five steps, which is suitable for removing heat from power transistors in electronics and processors in computers. They studied the effects of various operational limits and test parameters such as heat inputs, volume fraction, and vapor temperature on the thermal resistance, evaporation and condensation heat transfer coefficients experimentally. They observed reduction in thermal resistance of 76.2% for 0.009Vol% concentration of silver nanoparticles and concluded the use of nanoparticles enhances the operating range of heat pipe by 21% compared with that of DI water. Sheikholeslami et al. [13] studied the effects of different governing parameters on natural convection heat transfer in an inclined L-shape enclosure filled with Cu-water nanofluid. Nabi et al. [14] studied the Brownian motion of nanoparticles and clustering of them on the thermal conductivity of nanofluids. In this research, by implementation of a new correlation for viscosity of nanofluids developed by authors and using a new nanofluid (MWCNT) as working fluid in the cylindrical heat pipes, its thermal performance has been studied numerically. The thermal performance of heat pipe parameters such as the liquid pressure, temperature distribution, heat pipe wall temperature, heat pipe temperature difference, heat pipe heat transfer coefficient and thermal resistance are investigated at different heat inputs and operating conditions.

2. MATHEMATICAL MODELING

The used geometry here is shown in Fig. 1 schematically. A conventional cylindrical heat pipe with a constant conductance with fully saturated annular porous wick has been used. It has three zones: evaporator, adiabatic section and condenser. The simulation has been donned in two-dimensional and steady state, so the governing equations are: continuity, momentum and energy for both region of vapor and liquid. Also, the fluid flow has been assumed laminar and incompressible.

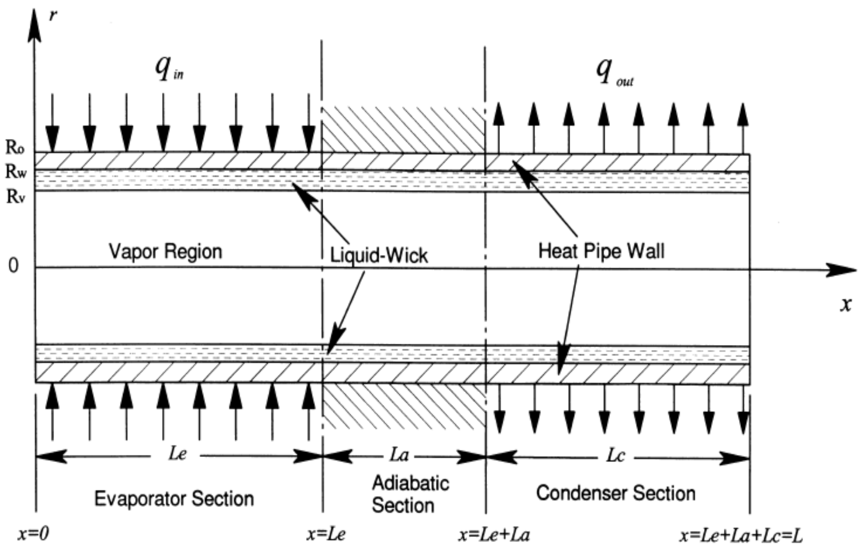

Fig. 1. Cylindrical heat pipe schematic

a) Governing equations for vapor flow

Continuity:

$$\frac{\partial u_v}{\partial z}+\frac{v_v}{r}+\frac{\partial v_v}{\partial r}=0 \tag{1}$$

R-momentum:

$$\rho_v(u_v\frac{\partial u_v}{\partial z}+v_v\frac{\partial u_v}{\partial r})=-\frac{\partial p_v}{\partial z}+\mu_v\left(+\frac{1}{r}\frac{\partial}{\partial r}\left(\frac{r\,\partial u_v}{\partial r}\right)+\frac{1}{r}\frac{\partial}{\partial r}\left(\frac{r\,\partial v_v}{\partial r}\right)-\frac{2}{3}\frac{\partial}{\partial z}\left(\frac{1}{r}\frac{\partial}{\partial r}(rv_v)\right)\right) \tag{2}$$

z-momentum:

$$\rho_v(u_v\frac{\partial v_v}{\partial z}+v_v\frac{\partial v_v}{\partial r})=-\frac{\partial p_v}{\partial r}+\mu_v\left(\frac{\partial^2 v_v}{\partial z^2}+\frac{4}{3r}\frac{\partial}{\partial r}\left(\frac{r\,\partial v_v}{\partial r}\right)-\frac{4}{3}\frac{v_v}{r^2}-\frac{1}{3}\frac{\partial^2 u_v}{\partial z\,\partial r}\right) \tag{3}$$

Energy equation:

$$\rho_v c_{p_v}(u_v\frac{\partial T_v}{\partial z}+v_v\frac{\partial T_v}{\partial r})\ =\frac{k_v}{r}\left[\frac{\partial}{\partial r}\left(r\frac{\partial T_v}{\partial r}\right)+r\frac{\partial^2 T_v}{\partial z^2}\right] \tag{4}$$

b) Governing equations for liquid flow

For porous media in the wicking material, the Darcy's law has been utilized as shown below:

R-momentum:

$$\rho_l(u_l\frac{\partial u_l}{\partial z}+v_l\frac{\partial u_l}{\partial r}=-\frac{\partial p_l}{\partial z}+\mu_l\left(\frac{1}{r}\frac{\partial}{\partial r}\left(\frac{r\,\partial u_l}{\partial r}\right)+\left(\frac{\partial^2 u_l}{\partial z^2}\right)-\frac{\mu_l\epsilon_z u_l}{k_z}\right) \tag{5}$$

Z-momentum:

$$\rho_l(u_l\frac{\partial v_l}{\partial z}+v_l\frac{\partial v_l}{\partial r})=-\frac{\partial p_l}{\partial r}+\mu_l\left(\frac{1}{r}\frac{\partial}{\partial r}\left(\frac{r\,\partial v_l}{\partial r}\right)+\left(\frac{\partial^2 v_l}{\partial z^2}\right)-\frac{\mu_l\epsilon_r v_l}{k_r}\right) \tag{6}$$

Energy equation:

$$\rho_l c_{p_l}(u_l\frac{\partial T_l}{\partial z}+v_l\frac{\partial T_l}{\partial r})=\frac{k_{eff}}{r}\left[\frac{\partial}{\partial r}\left(r\frac{\partial T_l}{\partial r}\right)+r\frac{\partial^2 T_l}{\partial z^2}\right]+S \tag{7}$$

The values of the source terms applied to evaporator and condenser sections by considering the phase change phenomenon are:

$$\begin{cases} s_e=-\frac{q_e}{l_e\pi(r_w{}^2-r_v{}^2)} \\ s_a=0 \\ s_c=+\frac{q_c}{l_c\pi(r_w{}^2-r_v{}^2)} \end{cases} \tag{8}$$

In the above equations the effective thermal conductivity is combined from conductivity of nanofluid and saturated wick and is calculated from the equation below [15]:

$$k_{eff}=\frac{k_{nf}[(k_{nf}+k_s)-(1-\epsilon)(k_{nf}-k_s)]}{[(k_{nf}+k_s)+(1-\epsilon)(k_{nf}-k_s)]} \tag{9}$$

Where k_{nf} is the conductivity of nanofluid and is calculated from:

$$k_{nf}=\frac{k_l[(k_l+k_s)-(1-\epsilon)(k_l-k_s)]}{[(k_l+k_s)+(1-\epsilon)(k_l-k_s)]} \tag{10}$$

For calculating the viscosity of conventional nanofluids the model developed by Bakhshan et al. [1] has been used and is:

$$\mu_{nf}=\frac{\mu_{bf}}{a+bt^{2.5}+ce^{-\varphi}} \tag{11}$$

Where a, b, c are constants and are a=-5.161804, b=-3.0904269e-8, c=6.1825371, respectively.
Density and heat capacity of nanofluids can be calculated from the following equations:

$$\rho_{nf} = \rho_p \varphi + (1-\varphi)\rho_{bf} \tag{12}$$

$$cp_{nf} = \frac{(1-\varphi)(\rho cp)_{bf} + \varphi(\rho cp)_p}{\rho_{nf}} \tag{13}$$

c) Boundary condition

The radial vapor and suction velocities at the sections of heat pipe are:

$$\begin{cases} v_e = +\dfrac{q_e}{2\pi r_v l_e \rho_v h_{fg}} & \text{evaporator} \\ v_a = 0 & \text{adiabatic} \\ v_c = -\dfrac{q_c}{2\pi r_v l_c \rho_v h_{fg}} & \text{condenser} \end{cases} \tag{14}$$

The vapor-liquid interface temperature for all sections is calculated using Clausius-Clapeyron formula as follows:

$$T_{int} = \frac{1}{\frac{1}{T_{v,sat}} \frac{r}{h_{fg}} \ln \frac{p_v}{p_{v,sat}}} \tag{15}$$

The value of the heat flux over each section of the pipe can be calculated using the following formulas:

$$\begin{aligned} q''_e &= +\frac{q_e}{2\pi r_v l_e \rho_v h_{fg}} && \text{evaporator} \\ q''_a &= 0 && \text{adiabatic} \\ q''_c &= -\frac{q_c}{2\pi r_v l_c \rho_v h_{fg}} && \text{condenser} \end{aligned} \tag{16}$$

3. RESULTS AND DISCUSSION

The dimensions of the considered model for heat pipe are shown in the Table 1. The heat pipe has a total length of 89 cm. The condenser has a length of 20 cm, while the evaporator and adiabatic sections are 60cm and 9 cm in length respectively. The outer radius of the heat pipe is taken as 9.55 mm, the inner radius is 9.4 mm and the vapor core radius is 8.65 mm. It should be noted that the dimensions used here are just nominal.

Table 1. Dimensions of modeled heat pipe

Length of heat pipe(mm)	890
Outer diameter of heat pipe(mm)	19.1
Length of evaporator(mm)	600
Length of condenser(mm)	200
Porosity of wick	0.9
Permeability($m^{2)}$	1.5×10^{-9}
Thickness of contain wall(mm)	0.15
Thickness of wick(mm)	0.75

In the present study, three different types of water based nanofluids, namely (Al_2O_3+Water), (Diamond+Water) and (Multi-Wall Carbon Nano tube (MWCNT) +Water) have been considered. In what follows, the influence of simple nanofluids and MWCNT nanofluid as working fluid on the thermal characteristics of heat pipe such as liquid velocity, pressure profile, temperature profile, thermal resistance and maximum heat transport capacity is investigated. For validation, the pressure and temperature results of simulation were compared with analytical results given by Shafahi [3] at Re=10. These comparisons are shown in Figs. 2 and 3. This comparison shows, our simulation results have good agreement with analytical results using the conventional nanofluids as working fluid. With this accuracy, our developed code has been modified for implementation of multi wall carbon nanotube (MWCNT) with water as working fluid. Also, it has capability for parametric studies.

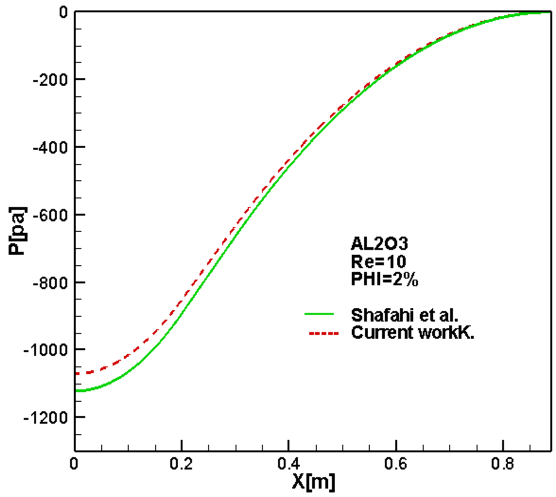

Fig. 2. Comparison of the liquid pressure distribution with analytical result by Shafahi et al. [3] at Re=10

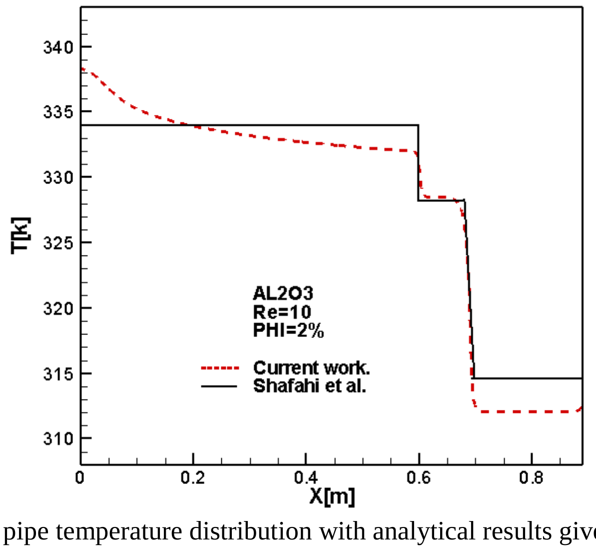

Fig. 3. Comparison of heat pipe temperature distribution with analytical results given by Shafahi [3] at Re=10

The thermo-physical properties of used nanofluids at specified volume fraction of nanoparticles are shown in Table 2. Figure 4 shows the heat pipe temperature variation through the heat pipe axis, for different working fluid at constant input heat. Temperature variation through the heat pipe axis is the same for all working fluid in global, so the temperature difference at the initial and end of heat pipe is higher for nanofluids in comparison with pure water and the MWCNT nanofluid has better performance in comparison with others at the same constant power for all. Because the MWCNT nanofluid has higher thermal conductivity, the temperature difference between evaporator and condenser is higher than other fluids and it makes MWCNT nanofluid a suitable working fluid for heat pipes.

Table 2. Properties of different materials at ϕ=0.921%

Properties	Pure water	Diamond-water	Al_2O_3-water	MWCNT-water
$\rho (kg/m^3)$	995	1018.2	1022.4	1080
c_p (j/kg k)	4181	4063.9	4058.84	4150
k(w/m k)	0.617	0.634	0.633	$= -287.6+3.022T-0.010557T^2+1.22885*10^{-5}T^3$
μ (pa s)	0.000797	0.000815	0.000815	$=k\Upsilon^{n-1}$ (k=00135, n=0.96)

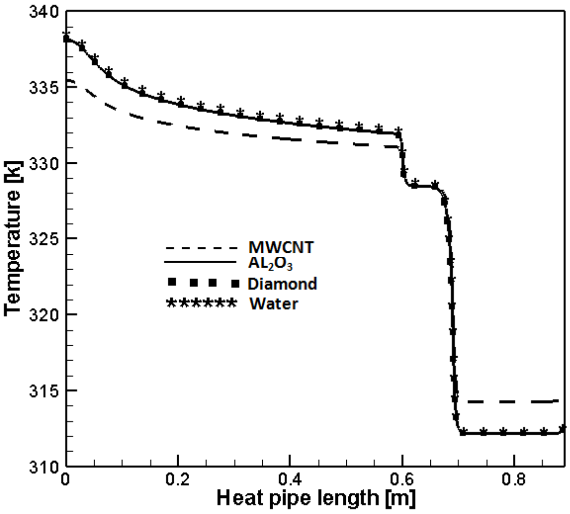

Fig. 4. Heat pipe temperature distribution; q=1180 w

Figure 5 displays the liquid pressure distribution, with respect to the liquid pressure, it can be seen that the pressure gradient decreases as the MWCNT nanofluid is used as working fluid. It is observed that pressure distribution along the heat pipe changes when the working fluid is changed to the MWCNT nanofluid. This behavior is due to the opposite roles played by density and viscosity, both growing by using the MWCNT in the water. Physically, an increase in density reduces the liquid velocity which in turn results in a lower shear stress. In contrast, an increase in viscosity increases the shear stress, and this can be seen in Fig. 6. The maximum liquid velocity decreases with adding the carbon nanotubes within the working fluid. This is due an increase in the liquid density in the presence of carbon nanotubes. As a result of this increase in the nanofluid density, a slower flow is observed.

The MWCNT based nanofluid has minimum value of velocity and pure water has its maximum value and this is consistent with temperature difference and density of both fluids.

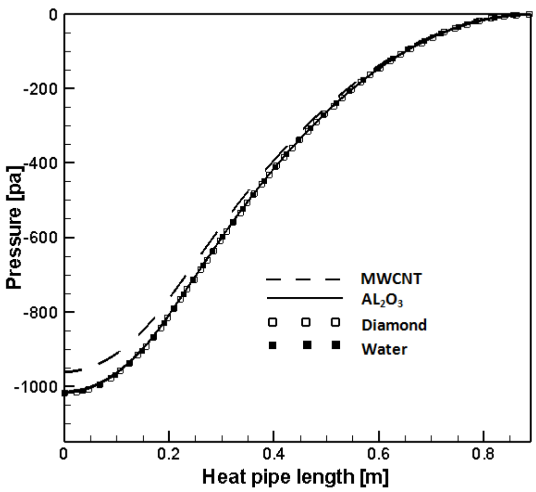

Fig. 5. The effect of different nanoparticle materials on the liquid pressure; q=1180 w

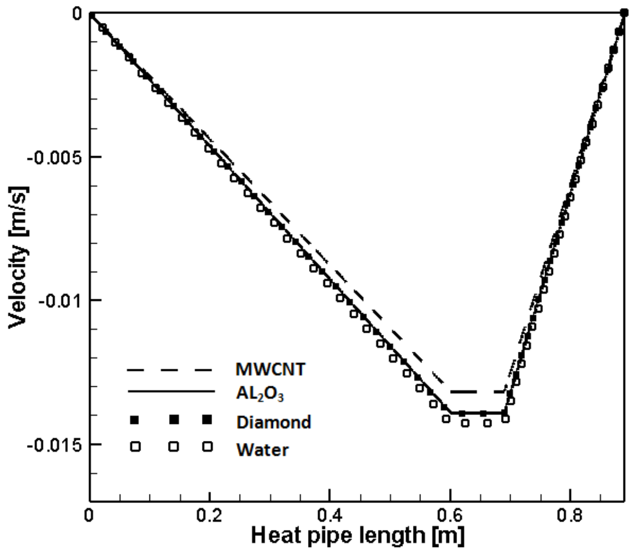

Fig. 6. The effect of different nanoparticle materials on the liquid velocity at q=1180 w

The temperature difference between evaporator and condenser sections of heat pipe is an important parameter in its design, because it shows the capability of heat pipe for removing the considered heat load. The variation of this parameter with heat loads is shown in Fig. 7. With increasing the heat loads, the temperature difference between evaporator and condenser will increase. The variation of temperature difference trend is the same for all working fluids, but the MWCNT nanofluid has minimum temperature difference in all heat loads. It can be seen that for constant temperature difference between condenser and evaporator the use of a nanpfluid, especially water based MWCNT nanofluid as a working fluid allows the heat pipe to operate under a larger heat load. A MWCNT nanofluid based heat pipe is able to dissipate more heat without experiencing increase in the wall temperature which makes it a suitable working fluid for use in heat pipe.

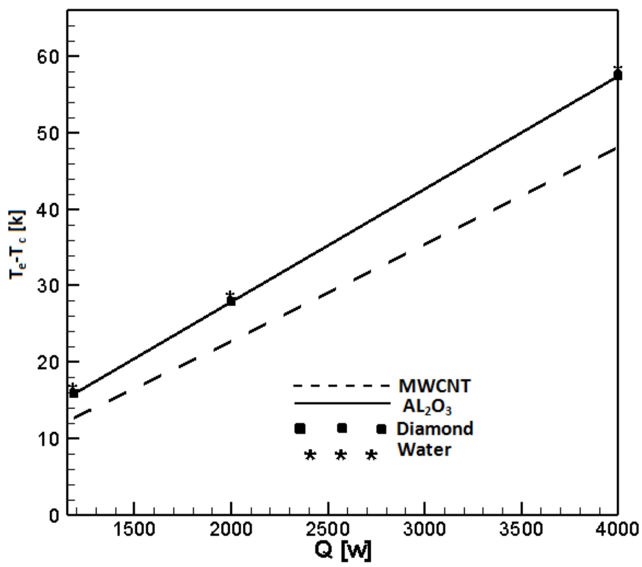

Fig. 7. The effect of heat loads on the heat pipe temperature difference

The variation of heat transfer coefficient at the condenser section of heat pipe versus heat loads is shown in Fig. 8. It can be seen that, the MWCNT based nanofluid has higher heat transfer coefficient in comparison with pure water and other conventional nanofluids and this is another reason that it is such a suitable working fluid for use in heat pipes.

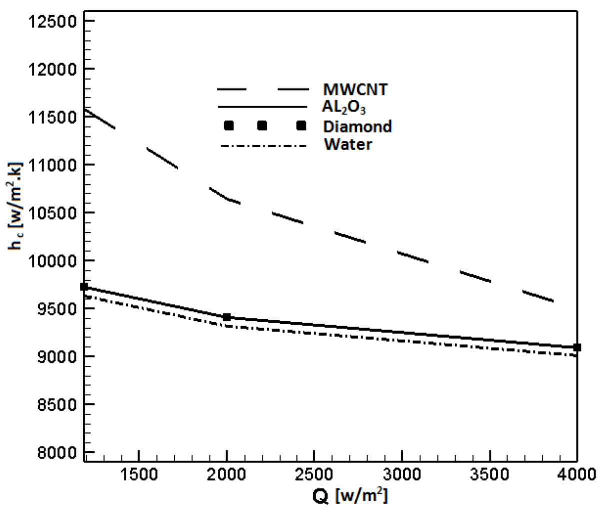

Fig. 8. The variation of heat transfer coefficient at the condenser section at different heat loads

The overall heat pipe thermal resistance is a key parameter for comparing the heat pipes performance with different working fluids. Thermal resistance of heat pipe is calculated from the following equations:

$$R = \frac{\overline{T}_E - \overline{T}_C}{Q} \tag{17}$$

Where

$$\overline{T}_e = \frac{T_{emax} + T_{emin}}{2} \tag{18a}$$

$$\overline{T}_C = \frac{T_{Cmax} + T_{Cmin}}{2} \tag{18b}$$

The thermal resistance variation of heat pipe versus heat loads is shown in Fig. 9, and the MWCNT based nanofluid has lower thermal resistance in comparison with other simple nanofluids, and this is good for having higher thermal efficiency of heat pipes that uses the MWCNT based nanofluids.

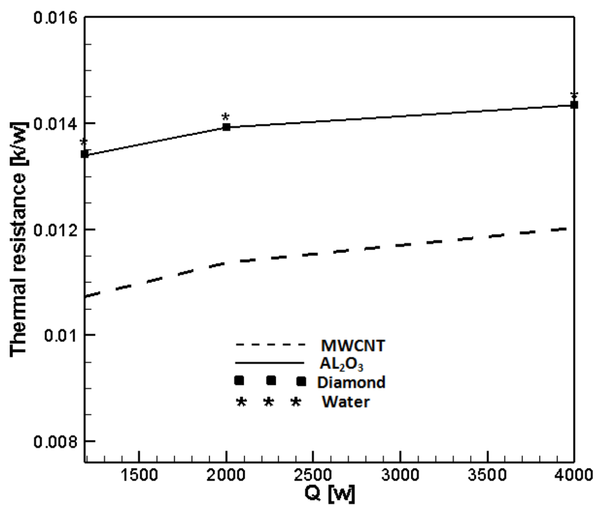

Fig. 9. The variation of thermal resistance with heat loads with different nanofluids

4. CONCLUSION

Thermal performance of a cylindrical heat pipe is investigated numerically using nanofluids. Three different types of water based nanofluids, namely (Al_2O_3+Water), (Diamond+Water) and (Multi-Wall

Carbon Nano tube (MWCNT) +Water) were considered. The influence of conventional nanofluids and MWCNT based nanofluid on the heat pipe characteristics such as liquid velocity, pressure profile, temperature profile, thermal resistance and heat transfer coefficient of heat pipe is studied. The results show, the MWCNT based nanofluid has lower thermal resistance, higher heat transfer coefficient and lower temperature difference between evaporator and condenser sections and it is a suitable working fluid for use in heat pipes. Also, we used our developed correlation for viscosity of nanofluid that was published earlier.

NOMENCLATURE

P	pressure (pa)	Greeks symbols	
T	temperature (k)	μ	dynamic viscosity (N s/ m^2)
u	horizontal velocity component (m/s)	ϵ	wick porosity
v	vertical velocity component (m/s)	ρ	density (kg/m^3)
K_{eff}	saturated wick thermal conductivity (w/m k)		
L	heat pipe length (m)	Subscripts	
Q	input heat rate (w)	e	evaporator
K	wick permeability (m^2)	c	condenser
S	source term (w/m^3)	a	adiabatic
h_{fg}	latent heat of vaporization (j/kg)	int	interface
R	heat pipe thermal resistance, gas constant (k/w), (j/mol k)	o	out
r	radius coordinate (m)	w	wall
x	length coordinate (m)	v	vapour
C_p	specific heat (j/kg k)	s	solid
K	thermal conductivity (w/m k)	i	internal
Re	Reynolds number	nf	nanofluid
		l	liquid

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