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AN EFFECT OF BINARY FLUID ON THE THERMAL PERFORMANCE OF PULSATION HEAT PIPE

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A pulsation heat pipe is an efficient heat pipe used in many engineering applications. This study aims to test the effect of working fluids on the thermal performance of pulsation heat pipe. Seven turned pulsation heat pipes were designed and manufactured from a copper pipe with a *3.5 mm* inner diameter. The lengths of an evaporation part, an adiabatic passage, and a condenser part were *300 mm*, *210 mm*, and *300 mm*, respectively. In this study, three different fluids were used as the working fluid: distilled water, methanol, and binary fluid (a mixture of water and methanol) with a *50%* filling ratio. Compared to water, the experimental results suggested that methanol had a better thermal performance when used as a working fluid in the *PHP*. On the other hand, a binary fluid enhanced the lower thermal performance of water (*29%* reduction in the thermal resistance and a *20%* increase in the effective thermal conductivity of the *PHP*).

Key words: binary fluid, oscillation heat pipe, two phase flow, thermal resistance. heat transfer.

1. Introduction

Akachi first introduced a pulsation heat pipe (*PHP*). The *PHP* is considered a suitable heat transfer device and an excellent solution for engineering applications requiring comparatively long-distance heat transportation [1]. In comparison with a conventional heat pipe, the *PHP* has many different features, such as low-pressure drop due to no wick structure, the flow of vapor and liquid phases in the same direction, and the pulsation phenomenon and the oscillating motion of working fluid inside the *PHP*, which plays an essential role in heat transfer [2]. However, the significant feature of a heat pipe is its capability to transfer a large quantity of heat across its length with a slight temperature difference. Heat is transported in such a heat pipe by sequential liquid evaporation and vapor condensation at heat source and sink regions [3]. Many parameters affect the *PHP* performance, such as the effect of tube geometry, the shape and diameter of the pipe, number of turns, length of evaporator and condenser sections, and a total length of the *PHP*, operational conditions like heat input, inclined angle, and working fluid physical properties and the volumetric filling ratio. A check valve was used to control the direction of a fluid motion [4]. Various investigations on the parameters affecting the thermal performance of the *PHP* have also been published. Kim *et al*. [5] investigated the formation and growth of bubbles and patterns of flow with and without heat supply on the *PHP*. They investigated two types of *PHP* (open and closed-loop): a rectangular cross-sectional area of $1.5\,mm^2$, and four turns for open-loop type and ten turns for closed-loop type. They used

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electrical heaters to supply heat $(0.45, 0.63, 0.8 \text{ and } 0.97)\ W/cm^2$ at the evaporator section. A condenser was cooled by water at a constant flow rate and temperature. They used ethanol and R142b as working fluids with a *FR* (filling ratio) about *40%*. The results revealed very fast formation and growth of bubbles which joined to become a vapor plug in the evaporation section. The recorded speed of the oscillation motion of liquid slugs and vapor plugs was *20 m/s*. The circulation and oscillation phenomena of the working fluid inside the *PHP* are more active when the number of turns and heat flux increase. R142b has a higher heat transfer performance than ethanol. Ma *et al*. [6] investigated using water and nanofluid as a working fluid with a *FR* of *50%* in *PHP*. The nanofluid was water containing nanoparticles of diamond with $(5-50)\ nm$ charged with a *1%* volume ratio. They used 12 turns of *PHP* manufactured from copper with an inner diameter of *1.65 mm* and the lengths of an evaporator, adiabatic and condenser sections were *60*, *101*, *60 mm*, respectively. The results proved that the nanofluid had the lowest thermal resistance in comparison with water $(0{,}03^\circ C/W \text{ at } 70^\circ C \text{ and } 336\,W)$.

Charoensawan *et al.* [7] used a horizontal closed loop *PHP* with different inner diameters (*1*, *1.5*, *2 mm*) to study the thermal performance. The *PHP* had a different number of turns (5, 11, 16, and 26), working fluids (water and ethanol) by (*30*, *50*, and *80%*) filling section ratios. The evaporator heated temperatures were $(40-90)^\circ C$, and the condenser part cooled at constant temperature and velocity $25^\circ C$ and *0.4 m/s*. Experimental results explained an effect of the number of turns on a horizontal closed loop *PHP* operation. They concluded that the shortening of the evaporator length and the increasing number of turns and pipe diameter improve the thermal performance of the *PHP*. Dadong *et al.* [8] explored a five turns *PHP* manufactured from copper with ID of *2 mm*, and an evaporator, condenser, and adiabatic section lengths were *80*, *90*, and *10* mm, respectively. The evaporator section was heated by an electrical heating plate (*20*, *40*, *60*, *80*, and *100*) *W* and cooled air was used to cool the condenser zone at $11^\circ C$ and *1.5 m/s*. They used water, methanol, ethanol, and acetone as pure working fluids and binary fluids with a mixture ratio of 1:1 by volume. They found that the mixture of water :acetone and water: methanol had a low R_{th} (thermal resistance). Mameli *et al.* [9] studied the thermal performance of a two turns *PHP* fabricated from copper with an inner diameter (*ID*) of *2 mm*. Glass was used to manufacture the adiabatic part to ensure visualization. They used ethanol and binary fluid of water (*4.5%* wt) and ethanol (*95.5%* wt) as a working fluid with a *65% FR* (filling ratio). A heat input at the evaporator section started from *40 W* using two heaters, and the heat was lost at the condenser section using the water bath at $15^\circ C$. The results revealed that the highest heat transfer coefficient was $4600\ W/m^2K$ before the dry out state. Pachgharea and Mahalle [10] used a copper heat pipe (closed-loop type) with ten turns and an inner diameter (*ID*) of *2 mm*. The lengths of the evaporator section, adiabatic section, and condenser section were *50 mm*. They used pure fluids (water, acetone, methanol, and ethanol) and mixture solutions (water/ acetone, water/ methanol, and water/ ethanol) with *FR* of *50%*. The experimental results showed the smooth decrease of thermal resistances by rising heat supply for all working fluids. Also, they found the water /methanol had a high temperature in the evaporator section.

Chien *et al.* [11] manufactured two types of eight turns *PHP* from copper and glass with square pipes of dimensions $2mm \times 2mm$ and non-uniform *PHP* eight pipes of dimensions $2mm \times 2mm$ and eight tubes of dimensions $2mm \times 1mm$. The working fluid was water with various *FR* (filling ratio) (*40 – 70*)*%* for both types. Results reveal that the non-uniform *PHP* can operate at a horizontal configuration, and the orientation angles affect the thermal resistance of non-uniform *PHPs* less than uniform *PHPs*. Nipon and Thanya [12] tested a *PHP* (closed-loop type) with a check valve (*CV*). They investigated the influences of the tilt angle, *FR*, *PHP* temperature, working fluid, and evaporator length of *PHP* with a *CV*. They fabricated the *PHP* from a copper tube of a *2.03 mm ID*, 40 turns, and two *CVs*. The working fluids were pure water, ethanol, and R123 with a *FR* of (*30%*, *50%*, and *80%*). The results showed that for R123 with a *FR* of *50%*, the heat supplied at the tilt angle 80° was higher than other tilted arrangements. Also, the maximum heat input was at 90° by using R123 with a *FR* of *50%* for evaporator length *150 mm*.

Rahman *et al.* [13] studied the impact of inserting fins in the *PHP*. They studied a finned and unfinned of *PHPs*. They used a copper *PHP* meandering in eight turns with *2.5 mm ID*. The evaporator, condenser, and adiabatic length were *50*, *80*, *120 mm*, respectively. Wires were made of copper with $D = 1mm$. Wires were inserted as fins only in the condenser part at the same distance among fins. The working fluids were methanol and ethanol with *FR* of *50%*. The heater was isolated and heat supplied at the evaporator section was about $(10 - 80)W$ at three inclination angles (*0*, *30*, and *45* degrees). The results demonstrated that methanol had lower R_{th} (thermal resistance) than ethanol, and the finned–inserted arrangement gave a more excellent performance than the unfinned arrangement. Goshayeshi *et al.* [14, 15] used Fe_2O_3/Kerosene nanofluids as working fluids in *PHP* with a *50%* filling ratio. The heights of the evaporator, adiabatic, and condenser sections were *100 mm* and *1.75 mm* in inner diameter. They studied the effects of using various nanoparticles (*10–30*) *nm* under the magnetic field. Also, the magnetic field effect on the heat transfer rate and the temperature distribution of the heat pipe was investigated. The inclination angles were from *0°* to *90°*, with the heat inputs range from *10 W* to *90 W*. The results showed that the addition of Fe_2O_3 nanoparticles and the presence of the magnetic field enhanced the thermal performance of the oscillating heat pipe. The optimum type and size of nanoparticles that gave the best heat transfer performance was *20 nm* γ-Fe_2O_3 nanoparticles. Xie *et al.* [16] used the volume of fluid (*VOF*) model to study the geometry, and multisource heat input on the thermal performance *PHP*. Two oscillating heat pipes with round and right-angled ends were used as physical models. The experimental data demonstrated that the *PHP* with right-angles at the evaporator started faster, and it had a higher mean heat transfer coefficient than *PHP* with the evaporator round end.

Barrak *et al.* [17] used a copper *PHP* with seven turns and *3.5 mm* inner diameter. The working fluid was water with a filling ratio of *50%* of the total volume. They experimentally studied the thermal performance of *PHP*. The experimental part used a CFD model based on the turbulent standard (*k* – epsilon) method with the volume of fluid (VOF) method to simulate the heat transport and mass transfer at the operations of evaporation and condensation. The results showed the maximum K_{eff} (effective thermal conductivity) of $4050 W/m \cdot {}^{\circ}C$ at $50^{\circ}C$ & *0.5 m/s* and the maximum thermal resistance was $1036\ {}^{\circ}C/W$ at $35^{\circ}C$ and *0.5 m/s*. Zufar *et al.* [18] used two-hybrid nanofluids, namely: Al_2O_3-CuO and SiO_2-CuO (by weight concentration of *0.1* wt. *%*) and water as working fluids to study the thermal performance of a four-turns oscillating heat pipe. The evaporator section was heated by heat input ranging (*10 – 100*) *W* with a *50* and *60 %* filling ratio. At the heat supplied and *FR* (filling ratio) of *80W* & *60%*, the results indicated that the R_{th} (thermal resistance) $(SiO_2\text{-CuO})$ and $(Al_2O_3\text{-CuO})$ hybrid nanofluids exhibited lower values than water by *57%* and *34%*, respectively. The (*k*) thermal conductivity and viscosity of hybrid nanofluids of $(SiO_2\text{-CuO})$ are lower than $(Al_2O_3\text{-CuO})$. Therefore, the *PHP*, using both hybrid nanofluids, can start up faster than that with water and needed lower heating input. Vo *et al.* [19] investigated the flow motion in the oscillating heat pipe experimentally and theoretically. They made the *PHP* with eight turns from Pyrex with a *1.85 mm* inner diameter. The working fluid was R123, with a *50* and *60%* filling ratio. Two water jackets were used to supply heat and cool the evaporator and condenser sections of the *PHP*. The three-dimensional simulation based on the *k*–ε turbulence model has been developed to predict the wall temperature distributions and heat transfer. The simulation results showed that the heat transfer rate had a good agreement (*5%*) with the experimental results.

Previous works have studied the thermal performance of the pulsation heat pipe (*PHP*) and studied many parameters that would affect the enhancement of overall heat transportation of *PHP*. This study aims to investigate the effects of working fluids on the thermal performance of the *PHP*. Water and methanol were selected to be pure working fluids, and the mixture of water-methanol with mixing ratio (*0.5:0.5* by volume) was investigated. The use of different working fluids in this work demonstrates the operation performance by studying the thermal resistance and the effective thermal conductivity of the *PHP*. The present study focused on using a binary fluid as a working fluid and studying the effective thermal conductivity and thermal resistance of the *PHP* under various inlet air temperatures and velocity ranges.

2. Pulsating heat pipe (*PHP*)

A *PHP* consisting of 3 parts includes of a long capillary serpentine pipe bent in many turns. Evaporator, adiabatic, and condenser parts were positioned in the turns as shown in Fig.1. The tube diameter of the *PHP* must be small enough to ensure the capillarity force to be excited. Liquid slug can be formed by surface tension. The *PHPs* an arrangement of vapor plugs and liquid slugs can be observed [20, 21]. It is important to study the design criteria of the *PHP* by investigating the forces that act on the bubbles rising in isothermal stagnant liquids. The bubbles rise in the denser fluid because of its bouncy force. These forces include the interaction between the buoyancy force with the liquid inertia, liquid viscosity, and surface tension forces. The balance between buoyancy and surface tension forces can be expressed by the non-dimensional Bond number [22].

$$Bo = \sqrt{\frac{D^2 g\left(\rho_{liq} - \rho_{vap}\right)}{\sigma}} . \tag{2.1}$$

The capillary's diameter of the pipe defined as D (critical diameter), is related to the Bond number, as below [23]:

$$D \leq 2\sqrt{\frac{\sigma}{\left(\rho_{liq} - \rho_{vap}\right) . g}} . \tag{2.2}$$

An increase in the internal diameter (in the allowable range smaller than the critical diameter) enhanced the circulation of the working fluid inside the oscillation heat pipe. Thus, a wider cross-sectional area of flow passage led to reducing the frictional force at the contact surface between the working fluid and the inside tube wall. Therefore, the working fluid transferred more heat and the thermal performance consequently enhanced [24].

The *PHP* has no additional capillary structure inside its tube. Firstly, the *PHP* is drained and then charged partially by a proper working fluid. The evaporator section (One end) of the *PHP* takes heat from the heat source. The flow of the working fluid transfers heat from the evaporator section (across the adiabatic section) to the condenser section (another end) of the *PHP* by a pulsating phenomenon and oscillation motion of the liquid slug and vapor plug [25, 26].

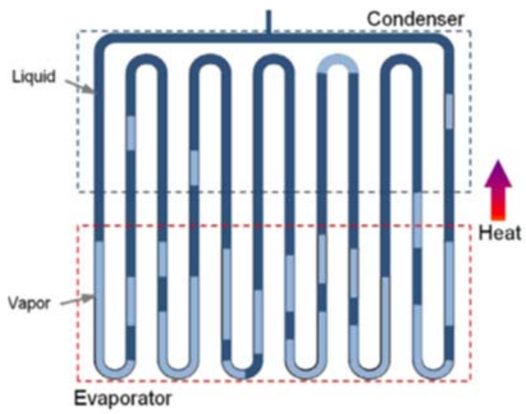

Fig.1. Oscillating heat pipe.

The choice of the working fluid depended on several properties as the working fluid can produce a high value of (dP/dT). The higher value of $(dP/dT)_{sat}$ implies the capability to create a high related saturation pressure (P_{sat}) for a small difference temperature (T_e) in the nucleation bubble. It will assist the vapor pumping phenomena and oscillation motion. In the condenser region, this fact is also valid but reversely.

Furthermore, other properties like high specific heat, low surface tension, low dynamic viscosity, and low latent heat can help active bubble generation and collapse. The convective heat transfer can be enhanced by improving the thermal conductivity of the fluid. Nanofluids and binary fluids are proposed to improve the process of heat transfer performance of the working fluid [27, 28].

3. Experimental Setup

A copper tube was used to manufacture an *PHP* because of the compatibility of copper with water, so it is very suitable for setting up the experimental apparatus. The copper long tube is bent to seven turns, with an *ID* of *3.5 mm* and *5 mm*. The length of the evaporator, adiabatic, and condenser sections were *300 mm*, *210 mm*, and *300 mm*, respectively. The evaporator section was heated by hot air, and the condenser section is cooled by forced air at $15^{\circ}C$. The adiabatic passage was separated from the evaporator section and condenser section of the *PHP* by two plates. The adiabatic areas were well-insulated with a fiberglass insulation*50 mm* thick. Figure 2 shows the experimental apparatus.

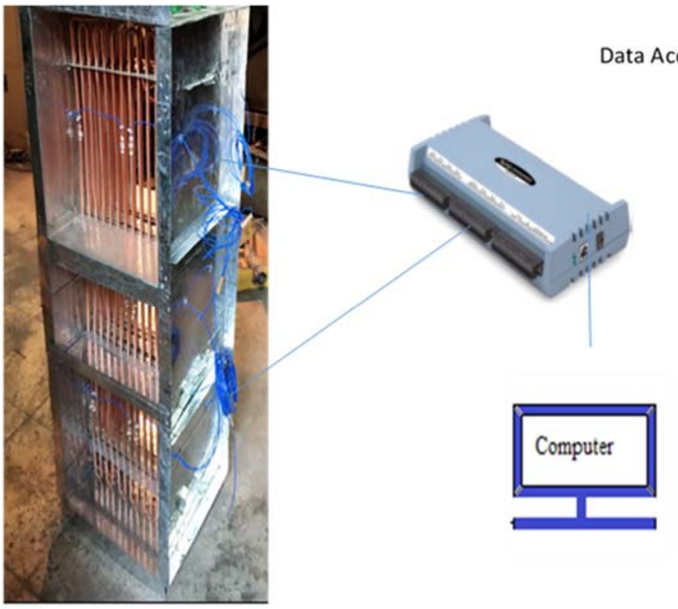

Fig.2. Photograph of 7 turns of copper *PHP*.

The temperature was measured by a data logger coupled with ten *K* type thermocouples. Four thermocouples, each attached to the tube walls of the evaporator and condenser section of the *PHP*, are shown in Fig.3. The calibrated sensors were used to decrease the experimental uncertainties (±*4.5%*), and then the sensors were placed on the *PHP* and attached to the data acquisition. A hotwire probe was used to estimate the air velocity with an uncertainty of ±*2%*. The *PHP* was first drained by a filling valve assembly connected to the condenser part and then filled with water as the working fluid by *FR* (filling ratio) *50%*. The operating temperature remains an important factor to select an acceptable working fluid. In this study, water was selected to be a reference case for the other two working fluids (methanol and binary fluid) due to the thermodynamic properties. The high surface tension allowed the fluids to produce a lot of capillary force. The high surface tension also generated additional friction force that can harm the *PHP* performance by hindering the working fluid flow oscillation in the *PHP*.

Binary fluids as working fluid mixture in the *PHP* need to be investigated in more detail due to a lack of studies in this field. A mixture of (water; methanol) as binary fluid (*0.5*:*0.5* by volume) is applied in the tests as the working fluid. Fluids were charged in the *PHP* with *FR* of *50%* and the vacuum pressure was less than -*70 cmHg*. The vacuum pump (reciprocating) was used to connect hose of the vacuum pump, the vacuum gauge, the filling valve service, and the *PHP*. Before the charging procedures, the vacuum pressure in the *PHP* was created to remove the free gases in a pulsating heat pipe. Next, a service valve was closed before disconnecting the vacuum unit from the *PHP* service valve. Then, it also attached to a scale pipe to charge a required volume of the working fluid. These procedures of charging can be repeated for all types of working fluids in all experiments.

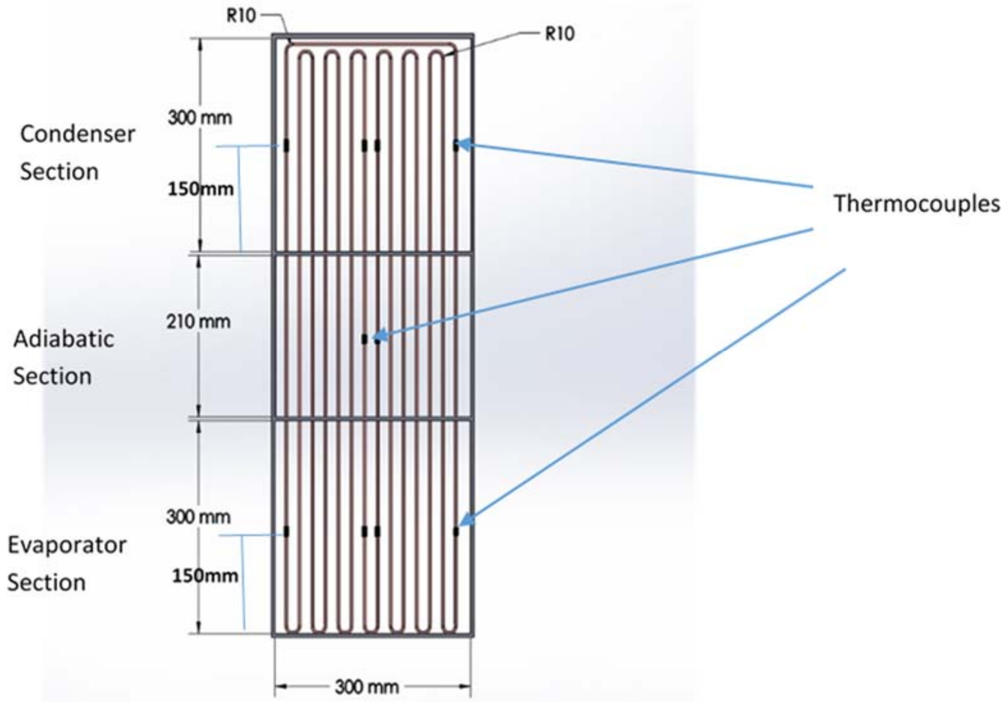

Fig.3. Schematic of *PHP* with thermocouples positions.

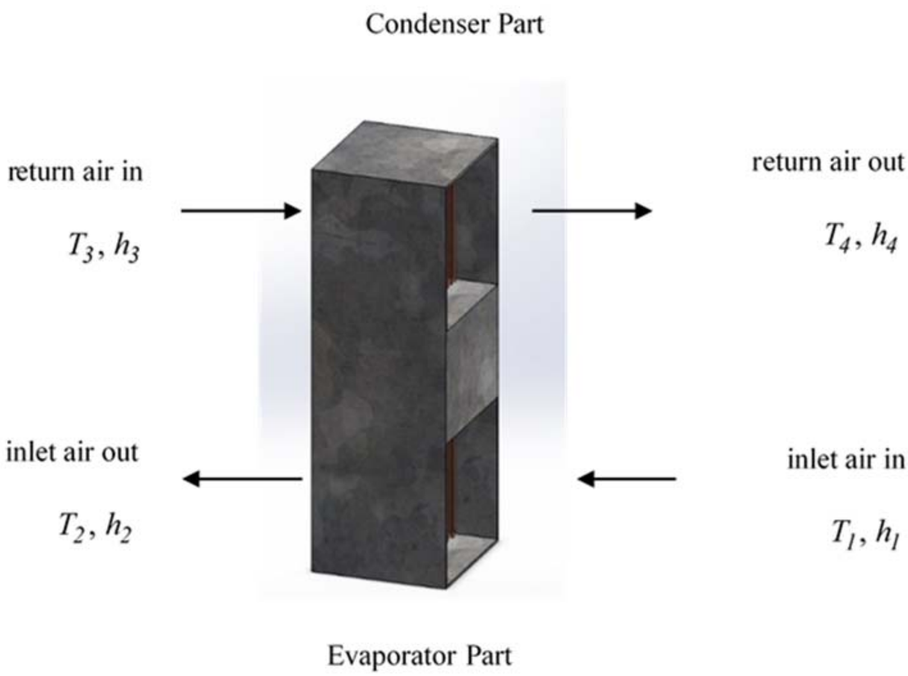

Fig.4. Air flow schematic.

Table 1. Properties of fluids.

Substance	Tsat $\left({}^{\circ}C\right)$	ρ $\left(Kg/m^3\right)$	hfg $\left(kJ/kg\right)$	C_p $\left(J/kg \cdot K\right)$	$\nu\left(m^2/s\right)$ 10-6	σ $\left(N/m\right)\cdot 10$
Water	*999.9*	*958.9*	*266.95*	*4215*	*89.2*	*0.563*
Methanol	*64.2*	*786.9*	*1100.2*	*2823*	*69.1*	*0.251*
Binary	*77.3*	*914.9*	*1890.3*	*3474*	*83.9*	*0.42*

4. Experimental calculation

The estimation of thermal performance of the *PHP* requires introduction of sensible heat and thermal resistance of *PHP*. The thermal resistance of the heat pipe is considered to be the effect of an inlet temperature and air velocity. Temperatures of the air have been recorded at the inlet and outlet of the evaporation part. The cooled return air moved over the condenser part of the *PHP*. The sensible heat that added and removed by the evaporation part and condensation part can be estimated as follows [17]

$$Q_{eva} = \dot{m}_a C_p \left(T_1 - T_2\right), \tag{4.1}$$

where T_1 is average inlet air temperature at the evaporator section, T_2 is average outlet air temperature at the evaporator section, T_3 is average inlet air temperature at the condenser section, T_4 is average outlet air temperature at the condenser section, see Fig.(4), C_p is specific heat capacity for air at constant pressure.

Air mass flow rate $\left(\dot{m}_a\right)$ is estimated by:

$$\dot{m}_a = \rho \cdot u_{air} \cdot A, \tag{4.2}$$

where ρ is air density, u_{air} is air velocity, and A is duct area (cross-sectional).

R_{th} (thermal resistance) of *PHP*s is calculated by: the difference of average temperatures between the evaporator and condenser sections /heat supplied at the evaporator part.

$$R_{th} = \frac{\overline{T}_{eva} - \overline{T}_{con}}{Q_{eva}}. \tag{4.3}$$

K_{eff} (effective thermal conductivity of *PHP*) is estimated by [17].

$$Q_{eva} = \frac{\overline{T}_{eva} - \overline{T}_{con}}{R} = \frac{\overline{T}_{eva} - \overline{T}_{con}}{\dfrac{L_{eff}}{K_{eff} \times N \times A_{cross}}}, \tag{4.4}$$

$$K_{eff} = \frac{Q_{eva} \times L_{eff}}{N \times A_{cross} \times \left(\overline{T}_{eva} - \overline{T}_{con}\right)}, \tag{4.5}$$

where $\overline{T}$ is average temperatures of *PHP* wall of evaporator and condenser, $L_{eff.}$ is effective length between centers of evaporator and condenser, N is pipes number of the evaporator part and A_{cross} is *PHP* area (cross-sectional).

5. Error analysis

The root-sum-square (RSS) method has been used to report the data analysis in many thermal engineering laboratories [29]. The results of (R_{th}) and heat input to the evaporator region are the most source of uncertainty that caused by the temperature and air velocity readings.
The propagation of uncertainties associated with calculations of the heat input values $S_{Q_{in}}$ is:

$$\frac{S_{Q_{in}}}{Q_{in}} = \sqrt{\left(\frac{S_{(T_i - T_o)}}{T_i - T_o}\right)^2 + \left(\frac{S_{(u)}}{u}\right)^2}\,, \tag{5.1}$$

$$S_{(T_i - T_o)} = \sqrt{S_{T_i}^2 + S_{T_o}^2}\,, \tag{5.2}$$

and for the thermal performance:

$$\frac{S_{R_{th}}}{R_{th}} = \sqrt{\left(\frac{S_{(T_{eva} - T_{con})}}{T_{eva} - T_{con}}\right)^2 + \left(\frac{S_{Q_{in}}}{Q_{in}}\right)^2}\,, \tag{5.3}$$

where:

$$S_{(T_{eva} - T_{con})} = \sqrt{S_{T_{eva}}^2 + S_{T_{con}}^2}\,. \tag{5.4}$$

The maximum uncertainty related to the calculation of Q_{in} and R_{th} values was around to *6.5%*.

6. Results and discussion

K_{eff} (effective thermal conductivity) of the *PHP* was plotted versus the temperature of inlet air at the evaporator for various air velocities. Figures 5a-c show that K_{eff} (effective thermal conductivity) increased with raising the temperature of inlet air because of heat supplied to the evaporator section and then transported by *PHP*. Thus, the growth of the practical thermal conductivity value of the *PHP* will have the ability to transfer heat at a much greater rate and eventually enhances the thermal performance of the *PHP*.

For example, the highest estimated K_{eff} (effective thermal conductivity) was $4168\,W/m\cdot K$, $5532\,W/m\cdot K$ and $4506\,W/m\cdot K$ at $50^\circ C$ and *2 m/s* for water, methanol, and binary fluid, respectively. At the same time, the lowest K_{eff} (effective thermal conductivity) was $829\,W/m\cdot K$, $1632\,W/m\cdot K$, $1352\,W/m\cdot K$ and $1141\,W/m\cdot K$ for water, methanol, and binary fluid at $35^\circ C$ and *0.5 m/s*, respectively.

The results show that water had a lower performance than methanol due to its lower practical K_{eff} (effective thermal conductivity) values. On the other hand, the highest values of K_{eff} (effective thermal conductivity) of methanol can improve the water behavior by around *20%* after adding methanol to water (binary fluid). A practical K_{eff} (effective thermal conductivity) values of the *PHP* which used methanol, water, and binary fluid are higher than 14, 10, and 11 times the K_{eff} of pure copper ($401\,W/m\cdot K$). The high thermal conductivity of the *PHP* produced significant heat transfer characteristics, even with a simplistic structure.

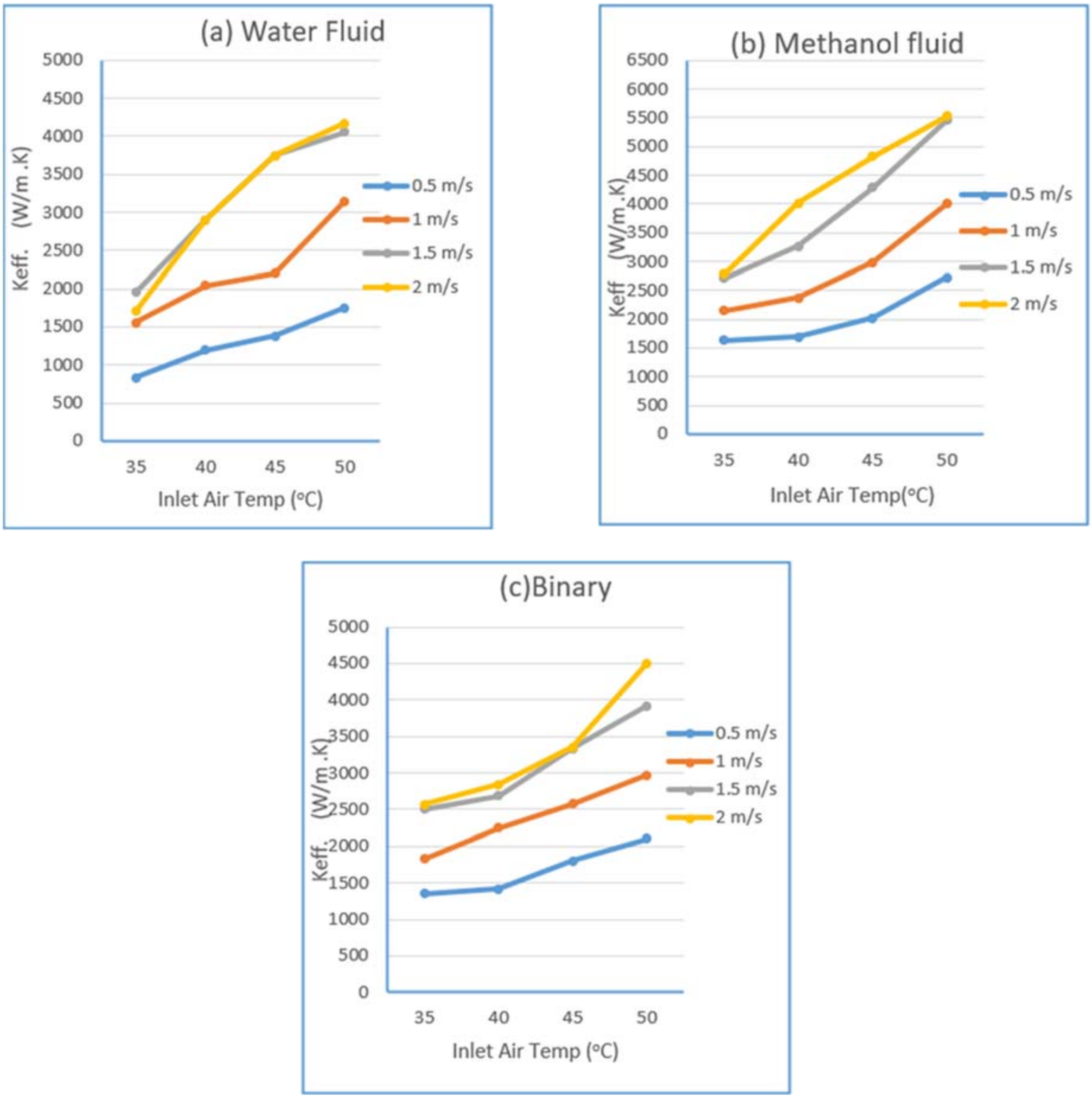

Fig.5. Comparison of effective thermal conductivity and inlet air temperature for the *PHP* evaporator with (a) water, (b) methanol and (c) binary fluids.

Figures 6a-c show the various R_{th} values (thermal resistance) of a conventionalized *PHP* for water, methanol, and binary fluid by varying inlet air temperatures and velocities. The R_{th} values (thermal resistance) can significantly decrease at higher heat supply. The curves displayed the decreasing R_{th} values (thermal resistance) with a rising inlet air temperature for all air speeds considered in the tests.

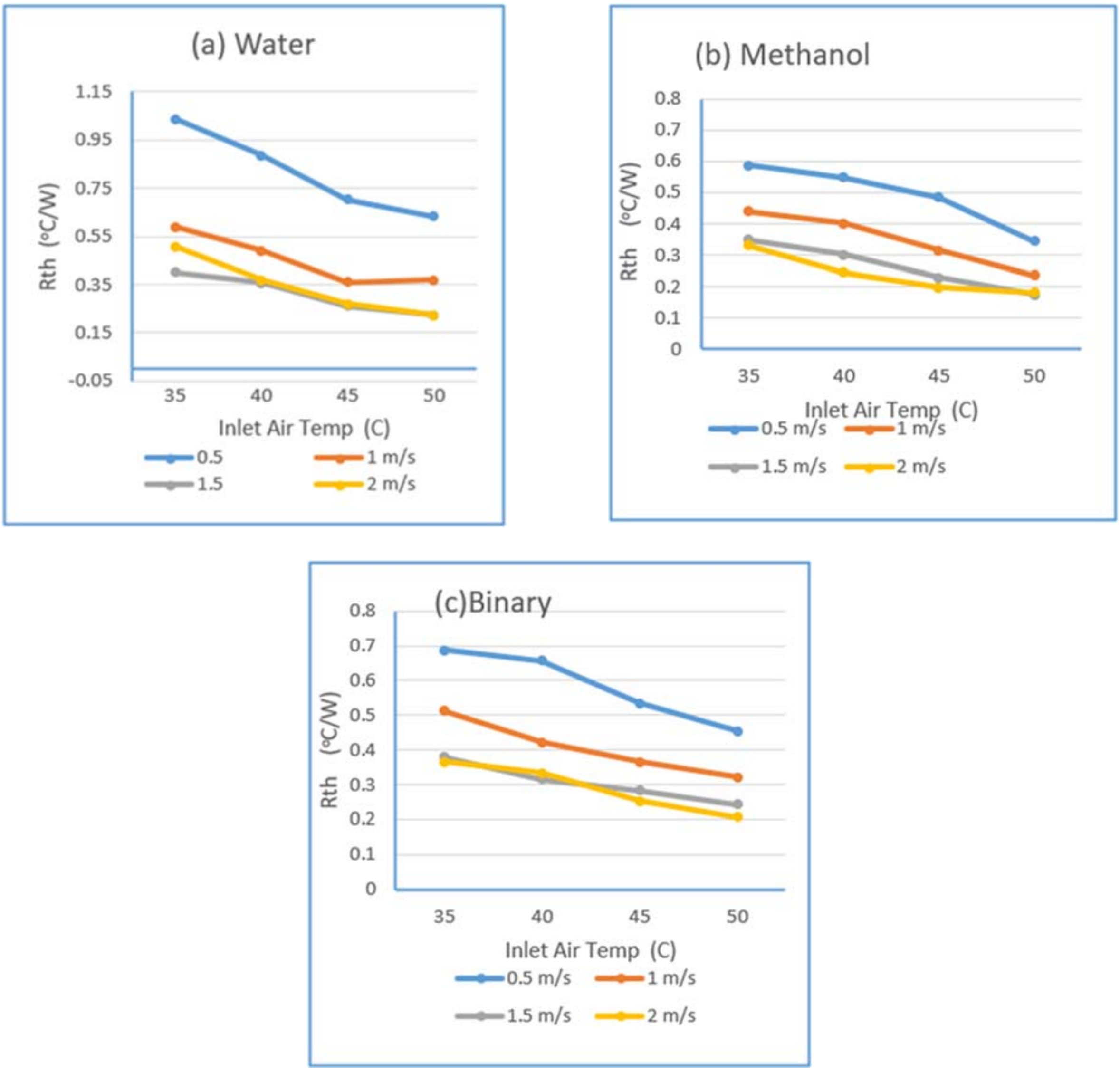

Fig.6. R_{th} vs. inlet air temperature of the *PHP* evaporator for typical pipe: (a) water, (b) methanol, (c) binary fluid.

The *PHP* depicted better thermal performance by lowering R_{th} values (thermal resistance) at a high inlet temperature of $50^{\circ}C$ for all air velocities. Results revealed also that an increase of the inlet air temperature of the evaporator part of *PHP* would increase the heat supplied at this device part. So, the temperature of the evaporator section was raised, following a high fluid density drop in the *PHP* pipes and low wall friction (low liquid viscosity). This decrease in the R_{th} values (thermal resistance) of the *PHP* at a high heat supply could also demonstrate that the temperature of the working fluid at the evaporator part will be high enough to raise the boiling rate of liquid, and that the working fluid flowed smoothly in one way. The lowest R_{th} values (thermal resistances) were *0.2249*, *0.1818* and $0.207^{\circ}C/W$, at $50^{\circ}C$ & *2 m/s* for water, methanol, and binary fluid.

The weak thermal performance of *PHP* is achieved at a low temperature $\left(35^{\circ}C\right)$ of inlet air. The low temp. of the air inlet leads to low heat supplies in the evaporator part. So, the working fluids of *PHP* behaved as a higher R_{th} values (thermal resistance) for all velocities. Because, at lower heat supply, the *PHP* can't sustain steady behavior. The max thermal resistance was $1.037^{\circ}C/W$, $0.588^{\circ}C/W$ and $0.688^{\circ}C/W$ at $35^{\circ}C$ & *0.5 m/s* for water, methanol, and binary fluid.

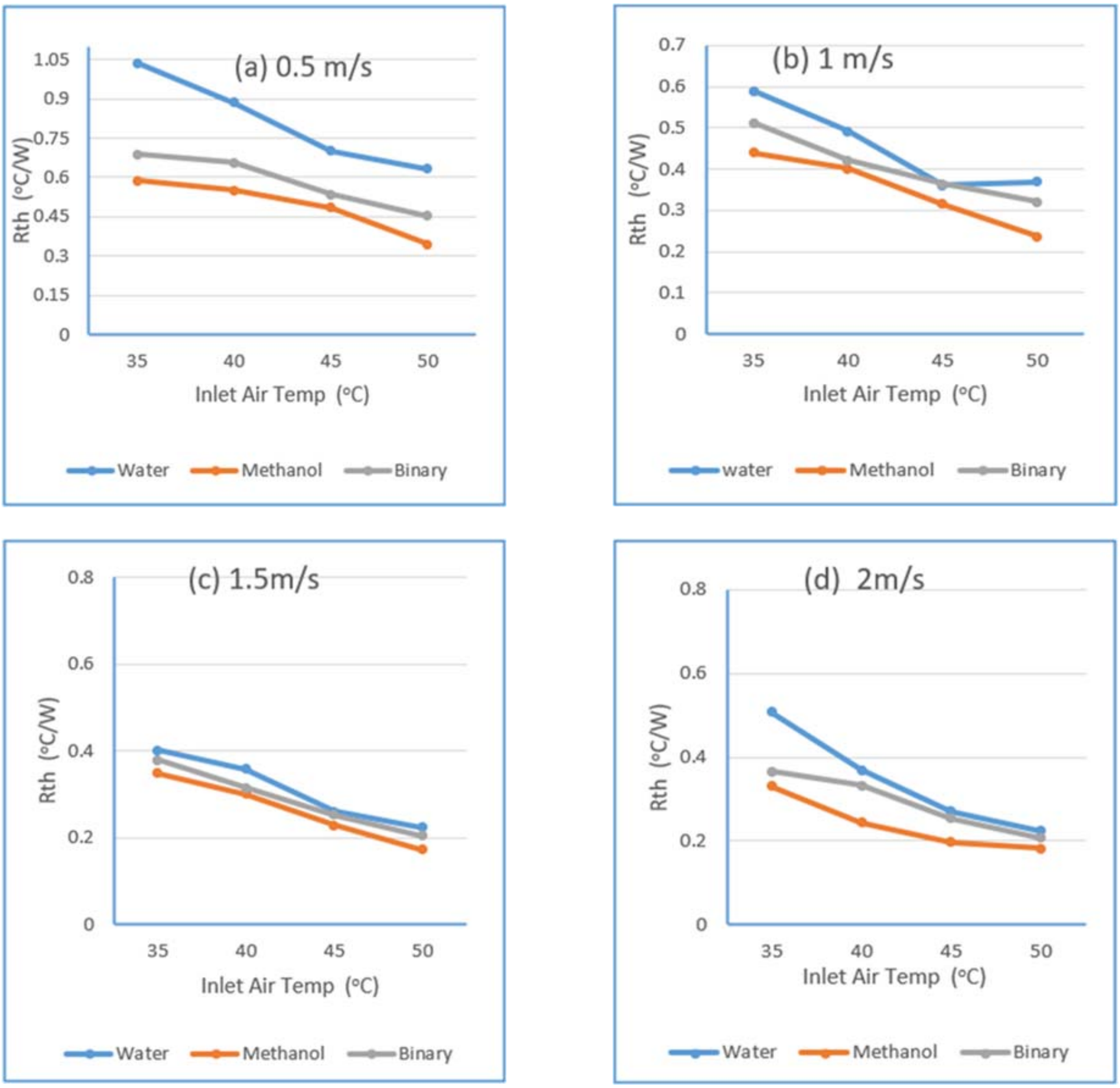

Fig.7. R_{th} vs. inlet air temperature for the typical *PHP* evaporator using water, methanol and binary fluid at (a) *0.5 m/s*, (b) *1 m/s*, (c) *1.5 m/s* and (d) *2 m/s.*

A comparison of R_{th} values (thermal resistances) of a typical *PHP* for three working fluids at varying air velocities is displayed in Figs 7a-d. The curves demonstrated that all working fluids show an identical trend R_{th} (thermal resistance) in experimental cases. Water had a higher R_{th} (thermal resistance), while methanol had a lower (thermal resistance) than the other working fluids for all air velocities. Methanol has many properties that lead to enhancing the thermal performance of the *PHP*. These properties of methanol are low boiling temperature, low latent heat of vaporization, and low surface tension, which play a significant role in methanol performance as the *PHP* working fluid. Results showed that the *PHP* thermal performance in terms of R_{th} of the water could be improved by around *29%* by using a mixture fluid instead of pure fluid.

The improvement in the thermal performance of the binary fluid was chiefly due to the reduction in thermal resistance caused by mixing methanol with water, which has greatly promoted the growth of bubbles in the *PHP*. As a result, greater heat can be absorbed in the evaporator section and more heat removed at the condenser section when more vapor plugs are generated in the *PHP*. Furthermore, the formation of vapor plugs in the working fluid promoted the pulsation phenomena and oscillation motion of the working fluid in the *PHP*. As a result, larger heat energy was transported by these vapor plugs and liquid slugs, and more heat wasted at the condenser zone.

7. Conclusion

The heat pipe is an efficient heat transfer device used in many engineering applications. Therefore, an experimental study was conducted to evaluate the thermal performance of an oscillating heat pipe. The working fluid plays a significant role in the thermal performance of the oscillating heat pipe because of its thermal properties.
From the experimental results, several conclusions could be drawn as follows:

- It was noticed that the thermal resistance and the effective thermal conductivity were related to the heat input for all working fluids.
- Using methanol as a working fluid in the *PHP* produced a higher thermal performance than water. The advantage of methanol was lower boiling temperature, lower latent heat of vaporization, and lower surface tension because it required less heat to convert the liquid into vapor. The maximum effective thermal conductivity was *4168* and $5532\ W/m \cdot K$ when using water and methanol.
- The thermal resistance and the effective thermal conductivity of the *PHP*s improved after using the binary fluid, and this enhanced the thermal performance of the *PHP*. In comparison to water, the results indicated a *29%* reduction in the thermal resistance and a *20%* increase in the effective thermal conductivity of the *PHP* when binary fluid was used.

Further investigation will be performed to explore the effect of using binary fluids and nanofluids in the thermal performance of *PHPs*. Therefore, in future work, the plan is to use another binary fluid and find the optimum percentage of the mixing ratio for use in *PHPs*.

Nomenclature

A – area diameter
C_p – specific heat capacity
D – diameter
ID – inner diameter
K_{eff} – effective thermal conductivity
m – flow rate
OD – outer diameter
PHP – pulsating heat pipe
Q – heat energy
R – thermal resistance
T – temperature
μ – dynamic viscosity
ρ – density
σ – surface tension

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