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Thermal performance of different working fluids in a dual diameter circular heat pipe

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Abstract In this paper, heat transfer performance of a 40 cm-length circular heat pipe with screen mesh wick is experimentally investigated. This heat pipe is made of copper with two diameters; larger in the evaporator and smaller in the adiabatic and condenser. Three different liquids including water, methanol, and ethanol are separately filled within the heat pipe. Low heat fluxes are applied (up to 2500 W/m^2) in the evaporator and constant temperature water bath is used at three levels including 15, 25, and 35 °C in the condenser. Results demonstrate that higher heat transfer coefficients are obtained for water and ethanol in comparison with methanol. Furthermore, increasing heat flux increases the evaporator heat transfer coefficient. For the case of methanol, some degradation in heat transfer coefficient is occurred at high heat fluxes which can be due to the surface dryout effect. Increasing the inclination angle decreases the heat pipe thermal resistance.

1. Introduction

As alternatives to the conventional heat sinks, two-phase cooling devices such as heat pipe have been emerged as promising heat transfer devices with effective thermal conductivity over 200 times higher than that of copper [1]. As a high thermal conductor, heat pipes have been used in different applications such as energy conversion, energy storage systems, and electronic cooling. Heat pipes are able to dissipate substantial amount of heat with a relatively small temperature drop while providing a self-pumping ability due to an embedded porous material in their structure. Regarding this importance, several researches have been conducted to evaluate the thermal performance of heat pipes with different geometries and different working fluids.

Reay [2] carried out experiments on a plate heat pipe with 100 cm length and 10 cm width. The orientation was horizontal in this study. Freon 11 and 113 were separately used as working fluids. It was found that Freon 11 was superior to Freon 113 from the point of view of thermal transport. Riehl and Dutra [3] presented the development of an experimental loop heat pipe (LHP) that deals with miniaturization and the

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Nomenclature

h	heat transfer coefficient (W/m^2 °C)
Q	heat transfer rate (W)
R	thermal resistance (°C/W)
T	temperature (°C)
1	notation for evaporator section in Figs. 3 and 4
2	notation for adiabatic section in Figs. 3 and 4
3	notation for condenser section in Figs. 3 and 4

Subscripts

c	condenser
e	evaporator
v	vapor
w	wall

use of an alternative working fluid. An experimental LHP was built with acetone as the working fluid, designed to manage up to 70 W and using a capillary evaporator with reduced active length. The selection of acetone was due to a tendency to substitute hazardous working fluids in two-phase capillary pumping systems.

Bai et al. [4] and Xi et al. [5] experimentally and numerically studied an integrated heat sink using heat pipes with methanol as phase change coolant. Kempers et al. [6] characterized the individual condenser and evaporator thermal resistances of a copper–water screen mesh wicked heat pipe. They examined the existence of boiling heat transfer in the heat pipe and its importance for the modeling of the heat pipe performance. Their results showed that a composite heat transfer model should be used for wicked heat pipes to take into account that either conduction or boiling can occur in the evaporator, with conduction only at the condenser.

Chang et al. [1] presented an experimental investigation of the thermal performance of a flat plate heat pipe. The results show that the porous wick in the evaporator section constitutes the main thermal resistance resulting in a larger temperature drop as compared to the other layers within the heat pipe. Naphon et al. [7] fabricated a heat pipe from the straight copper tube with the outer diameter and length of 15,600 mm, respectively. The heat pipe with the de-ionic water, alcohol, and nanofluids (alcohol and titanium nanoparticles) were tested. Effects of charge amount of working fluid, heat pipe tilt angle and nanoparticles volume concentrations on the thermal efficiency of heat pipe were considered.

Lips et al. [8] tested a flat plate heat pipe filled with methanol for different filling ratios and for different heat fluxes. Various phenomena like the occurrence of the boiling phenomenon were highlighted. It appears that boiling has a noticeable impact on the heat transfer at the evaporator. Chernysheva and Maydanik [9] generalized and presented the results of development and tests of 15 different variants of ammonia MLHPs with cylindrical evaporators 5 and 6 mm in diameter, which had an active zone length of 20 mm and were equipped with titanium and nickel wicks. Tsai et al. [10] presented a novel dynamic test method and compared it with the conventional steady-states test. Bending angles, fill ratios, and shapes of heat pipes were investigated in order to study their influences on the thermal performances of heat pipes for both steady-state and dynamic tests. Experimental results demonstrate that deformation of heat pipes would damage the thermal performances of heat pipes most significantly. Larger fill ratios would increase the operation limitations but also lead to less sensitive temperature responses of heat pipes.

Wong and Lin [11] investigated three different working fluids including water, methanol and acetone, which possess different figures of merit at the same volumetric liquid charge. Different degrees of wettability were obtained by varying the exposure times in air after the wicked plates were taken out of the sintering furnace. It was found the lowered copper surface wettability led to reduced critical heat loads for water rather than for methanol and acetone. From the view of thin-film evaporation mechanism, water has larger surface tension, polarity, viscosity, and latent heat than methanol and acetone. Attia and El-Assal [12] conducted an experimental study to evaluate thermal performance of a heat pipe with water and methyl alcohol at different charge ratios. Also, a solution of water and propylene glycol at two concentrations were tested to study the effect of using surfactant as enhancement agent for working fluid.

Wang et al. [13] performed an experimental study to investigate the thermal performance of an inclined miniature mesh heat pipe using water-based CuO nanofluid as the working fluid. The study focused mainly on the effects of the inclination angle and the operating temperature on the heat transfer performance of the heat pipe. Experimental results show that the inclination angle has a strong effect on the heat transfer performance of heat pipes using both water and the nanofluid. The inclination angle of 45° corresponds to the best thermal performance for heat pipes.

Zhang et al. [14] experimentally investigated the operating characteristics of a copper–water loop heat pipe (LHP), including start-up property, heat-transfer capacity, and heat resistance, under four different charge ratios.

Although the research work on traditional cylindrical and annular heat pipes has been well documented, there is far less work conducted for dual diameter cylindrical heat pipes. Furthermore, since heat pipes utilize the phase change of the working fluid to transport the heat, the selection of working fluid is of essential importance to promote the thermal performance of heat pipes [15,16]. It is the aim of this study to compare the thermal performance of water, methanol, and ethanol as the working fluid in a dual diameter heat pipe. This comparison had not been reported before and can be useful for understanding the working fluid selection for heat pipes.

2. Experimental

The experiments were performed using a heat pipe which was made of a smooth copper tube. Porous wicks are attached to the inner surfaces of the heat pipe wall, as shown in Fig. 1. There were three layers of aluminum meshes (mesh number 100) inside the tube. This type of mesh has a suitable flexibility

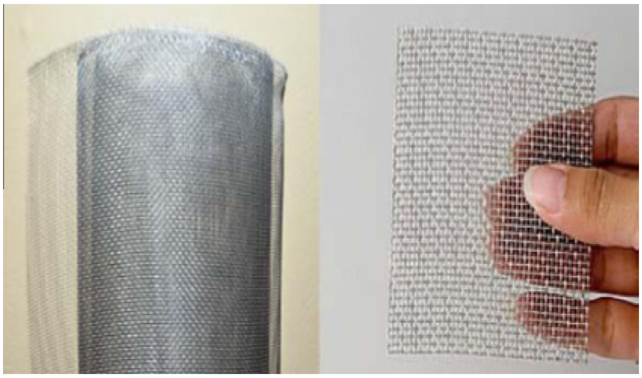

Figure 1 Screen mesh wick used in the heat pipe.

to be deformed. A close contact between the meshes and the inner wall can be guaranteed due to the internal tension of the mesh. The dimensions of evaporator, adiabatic, and condenser sections of the heat pipe were presented in Table 1. As can be seen, the evaporator section has larger diameter than the remaining sections of the heat pipe. Dual diameter circular heat pipe has not been studied previously. This configuration causes the evaporated liquid to pass the nozzle shaped entrance of the adiabatic section with higher velocity. When the vapor reaches the condenser faster, the heat transfer performance of the heat pipe would be improved.

The evaporator section was heated by an electrical heater (Watlow Company) surrounding at its circumference. The condenser section was cooled by the cooling water circulating in a constant-temperature thermal bath which was a cube with the dimensions of 0.2 m. The temperature and velocity of the cooling water were carefully controlled to keep the steam pressure in the tested heat pipe at a constant value for various heat fluxes. The evaporator and adiabatic sections were carefully insulated using glass wool.

Three E-type thermocouples were installed to measure the outside surface temperatures of the heat pipe and three others to measure the working fluid temperatures. Each group includes one thermocouple at the evaporator section, one at the condenser section and one at the adiabatic section. Very tiny grooves were machined in the heat pipe walls and a high conductivity cement was utilized to embed the thermocouples within the heat pipe wall. Distributions of the thermocouples along the axial direction are indicated in Fig. 2. The wall temperature distributions along the circumference direction were quite uniform because the mesh structure could make the liquid film uniformly filled into the mesh layers of the inclined heat pipe. A pressure transducer placed at the central location of the adiabatic section which is used to measure the saturation pressure of the steam in the heat pipe, i.e., the operating pressure.

Table 1 Dimensions of different sections of the heat pipe.

Specifications	Evaporator section	Adiabatic section	Condenser section
Length (mm)	100	200	100
Internal diameter (mm)	25.4	19	19
External diameter (mm)	26.4	20	20
Area (mm^2)	8290	12,591	6295
Volume (mm^3)	50,645	56,976	28,487

Different working fluids including water, methanol, and ethanol were filled into the heat pipe through a syringe. Some important physical properties of these working fluids are shown in Table 2. Since these working fluids have different physical properties, their implementation in the heat pipe may be useful in understanding the effect of each properties in heat transfer performance. The filling volume was fixed at 50% of the heat pipe volume. Before each test, the vacuum pumping and liquid preheating processes were performed to remove the dissolved gases in the heat pipe and the working fluid.

Almost all of the experiments were performed with the heat pipe in the horizontal orientation except for some runs which performed to analyze the effect of contact angle on the performance of the heat pipe. Tests were performed at different constant condenser temperatures of 15, 25 and 35 °C.

3. Data reduction

The thermal resistance is one of the most important parameters that reflect the performances of the heat pipe during the heat transfer tests. The evaporator thermal resistance is defined as:

$$R_e = \frac{T_{e,w} - T_{e,v}}{Q_e}\left(\frac{^\circ C}{W}\right) \tag{1}$$

where $T_{e,w}$ is the evaporator wall temperature, $T_{e,v}$ is the vapor temperature at evaporator section and Q_e is the input power.

The condenser thermal resistance is defined as:

$$R_c = \frac{T_{c,v} - T_{c,w}}{Q_e}\left(\frac{^\circ C}{W}\right) \tag{2}$$

where $T_{c,w}$ is the condenser wall temperature, $T_{c,v}$ is the vapor temperature at the condenser section and Q_e is the input power.

The total thermal resistance of the heat pipe can be defined as [18]:

$$R = \frac{T_{e,w} - T_{c,w}}{Q_e}\left(\frac{^\circ C}{W}\right) \tag{3}$$

The heat transfer coefficient at the evaporator section is calculated as follows:

$$h_e = \frac{\left(\frac{Q}{A}\right)}{T_{e,w} - T_{e,}}\left(\frac{W}{m^2\,^\circ C}\right) \tag{4}$$

An uncertainty analysis has been performed according to the method proposed by Kline and McClintock [19]. The estimated uncertainties of diameter, length and area are less than ±0.4%. The uncertainty of temperature is ±0.2 K for the thermocouples. The maximum value of uncertainty of input power is 0.8% and the maximum value of uncertainties of the evaporator thermal resistance and the heat pipe thermal resistance are ±4.5% and ±5.1%, respectively.

4. Results and discussion

4.1. Temperature measurement

The vapor core temperatures at the evaporator, adiabatic, and condenser sections of the heat pipe are shown in Fig. 3. These temperature data were obtained at constant input power

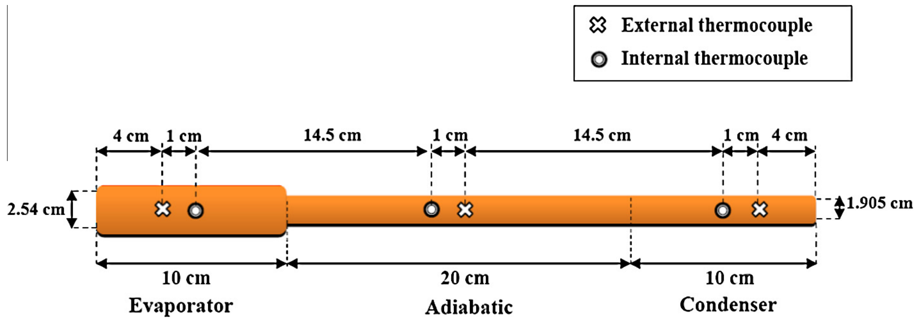

Figure 2 Locations of thermocouples in the heat pipe.

Table 2 Physical properties of water, methanol, and ethanol [17].

Fluid	NBP (°C)	ρ (kg/m^3)	P^{sat*} (kPa)	μ^{**} (kg/m s)	σ^{**} (N/m)	λ (kJ/kg)
Water	100	1000	2.33	1.79×10^{-3}	7.56×10^{-2}	2256
Ethanol	78	789	5.95	1.77×10^{-3}	2.41×10^{-2}	846
Methanol	65	792	13.02	8.17×10^{-3}	2.45×10^{-2}	1100

* The vapor pressure data are at 293 K.
** Surface tension and viscosity data are at 273 K.

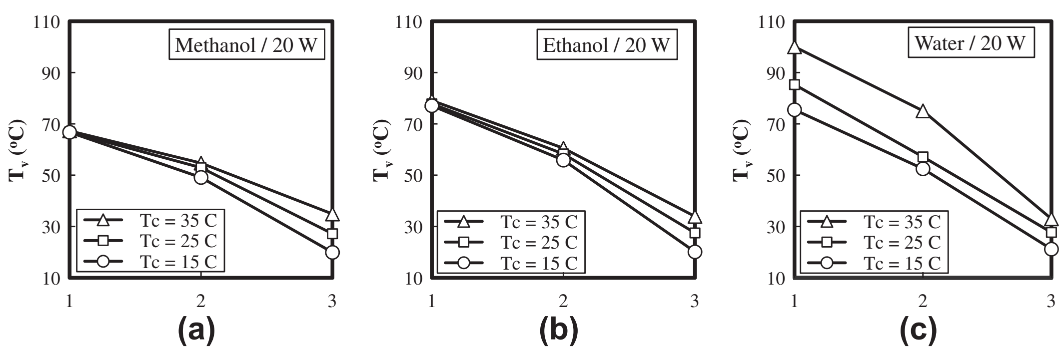

Figure 3 Variation of axial vapor core temperature at constant input power for different working fluids (a) methanol, (b) ethanol, (c) water.

(20 W) and at different condenser temperatures. Results for different working fluids including methanol, ethanol, and water are shown in Fig. 3a–c, respectively. The numbers 1, 2 and 3 at the horizontal axis of Fig. 3 demonstrate the location of evaporator, adiabatic and condenser of the heat pipe, respectively. As can be seen, higher vapor temperatures are obtained for water in comparison with methanol and ethanol at constant input power. The maximum temperature of the evaporator section of a heat pipe is related to the boiling point of the working fluid. As shown in Table 2, water has the largest boiling point among the other working fluids used in this study. Furthermore, difference in the condenser temperature has small influences on the vapor core temperature at the evaporator section for methanol and ethanol, while this effect is more pronounced for water.

Fig. 4 demonstrates the variation of heat pipe wall temperature when filled with different working fluids. These results were also obtained at 20 W input power and at different condenser temperatures. Once again, higher wall temperatures are obtained at the evaporator section and for the case of water. Higher temperature of the condenser causes the returning liquid to the evaporator to be warmer and consequently, higher wall temperature of the evaporator obtained. For methanol and ethanol the effect of condenser temperature is not strong, since the returning liquid to the evaporator section would boil when 20 W input power applies. When a liquid boils on the surface through which a constant heat flux is applied, the temperature of the surface will be constant. Conversely, Fig. 4c shows that when T_c = 15 °C, the returning liquid does not boil. So, the surface temperature differs from other condenser temperatures.

Another important point which can be seen in Figs. 3 and 4 is that the curvature of the temperature curves is different in these two figures. It means that for vapor core, larger temperature drop occurred at the distance between adiabatic and condenser while for wall temperature, larger temperature drop occurred between evaporator and adiabatic sections.

4.2. Heat transfer coefficient

The heat transfer coefficients for implementing different working fluids in the heat pipe are shown in Fig. 5 as a function of

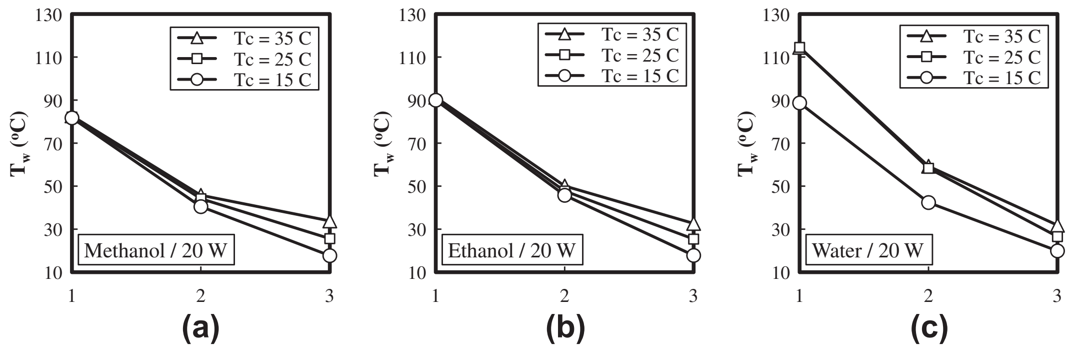

Figure 4 Variation of axial wall temperature of the heat pipe at constant input power for different working fluids (a) methanol, (b) ethanol, (c) water.

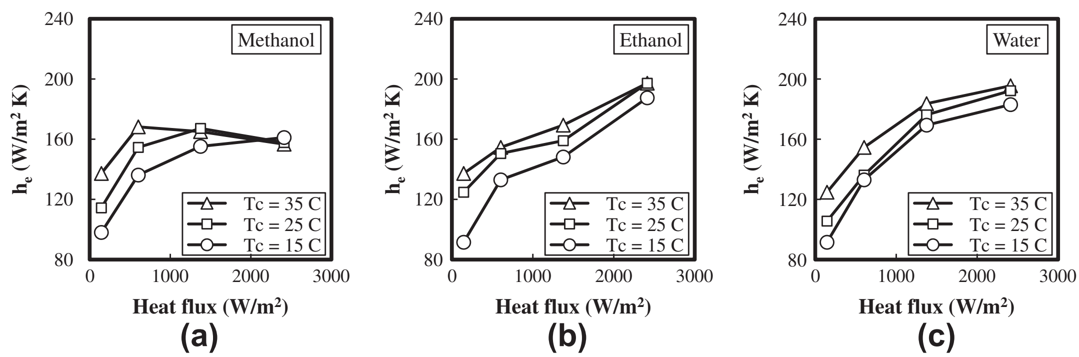

Figure 5 Variation of evaporator heat transfer coefficient as a function of input heat flux at different condenser temperatures for (a) methanol, (b) ethanol, (c) water.

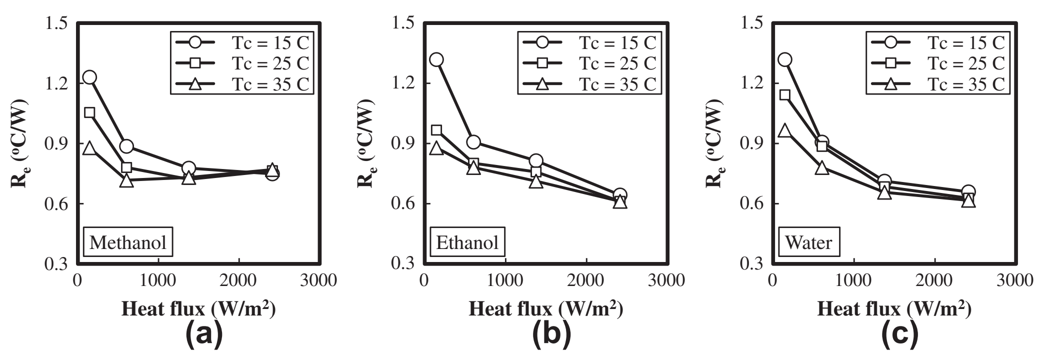

Figure 6 Variation of evaporator thermal resistance as a function of input heat flux at different condenser temperatures for (a) methanol, (b) ethanol, (c) water.

heat flux and condenser temperature. As can be seen, the heat transfer coefficient increases with increasing the applied heat flux at the evaporator. For methanol and at higher heat flux, the heat transfer coefficient decreases with increasing heat flux. It is due to dry out occurred at the evaporator surface which causes heat transfer degradation. The dry out for methanol is not observed at lower condenser temperature (T_c = 15 °C), since the returning liquid from the condenser is too cold to be completely evaporated. This phenomenon would probably be occurred for ethanol and water at higher heat fluxes than that investigated in this study.

4.3. Thermal resistance

Thermal resistance of the evaporator is shown in Fig. 6a–c for methanol, ethanol and water respectively. Increasing heat flux decreases the thermal resistance of the evaporator. The effect of dry out for methanol is again shown in Fig. 6a at high heat fluxes and high condenser temperature. Furthermore, higher condenser temperature causes the evaporator resistance to be decreased.

Fig. 7 presents the variation of condenser thermal resistance as a function of heat flux and condenser temperature. At the

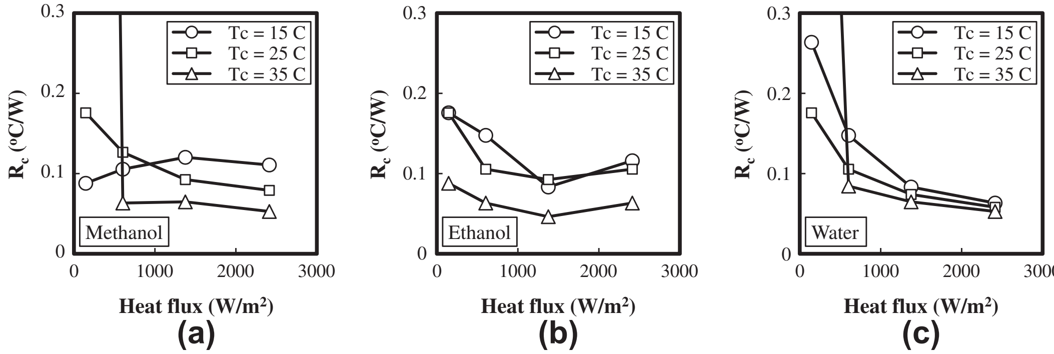

Figure 7 Variation of condenser thermal resistance as a function of input heat flux at different condenser temperatures for (a) methanol, (b) ethanol, (c) water.

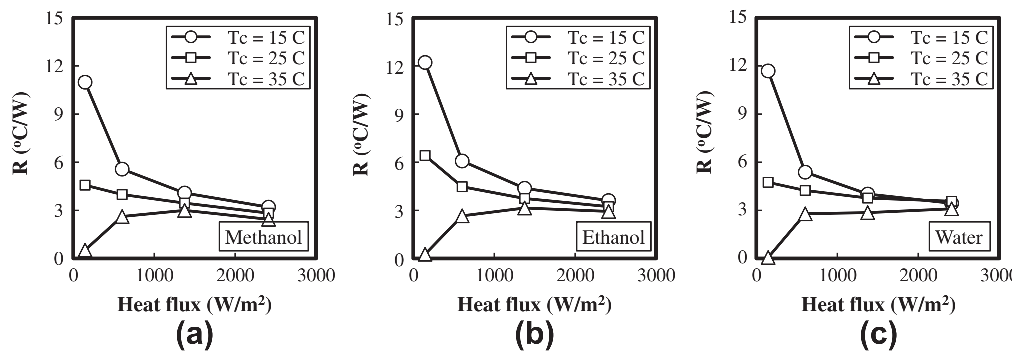

Figure 8 Variation of heat pipe thermal resistance as a function of input heat flux at different condenser temperatures for (a) methanol, (b) ethanol, (c) water.

lowest heat flux applied in this study, very large condenser thermal resistance observed specially for water and methanol when using the highest condenser temperature (T_c = 35 °C). It is not strange behavior; since the filled liquid temperature to the heat pipe is 25 °C and the applied heat flux is very low to change the liquid temperature at the evaporator significantly. Therefore, it was observed that the temperatures of all the working fluids at the evaporator would be less than that of condenser. It means that heat pipe works conversely in this condition. Liquid warms at the condenser and cooled at the evaporator. So, higher condenser thermal resistance would be expected in this condition.

Fig. 8 demonstrates the total resistance of the heat pipe for different working fluids as a function of heat flux. Although similar investigations showed that increasing heat flux causes the total resistance of the heat pipe to be decreased (see e.g. [14,15]), in this investigation and at T_c = 35 °C different trends were observed for all the working fluids. As previously discussed, high condenser temperature and low heat flux leads to the lower temperature difference between condenser and evaporator ($T_{e,w}$–$T_{c,w}$) and this reduces the overall resistance close to zero. Increasing heat flux in this condition warms the evaporator wall and increases the overall resistance.

4.4. Effect of inclination angle

The effect of inclination angle of the heat pipe on its total resistance is demonstrated in Fig. 9. As can be seen, increasing the contact angle (setting the evaporator in lower level than the condenser) causes the condensed vapor returns faster to the evaporator section by means of gravity and consequently, lower thermal resistances and higher heat transfer coefficients were obtained.

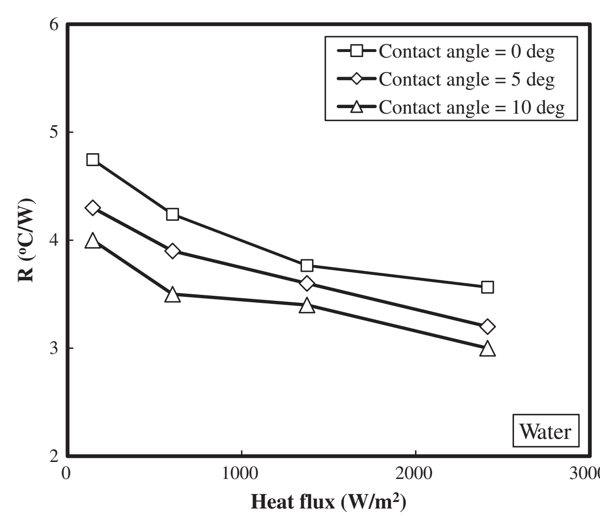

Figure 9 Effect of inclination on the overall thermal resistance of the heat pipe for water as the working fluid.

5. Conclusion

In this study, thermal performance of a dual diameter circular heat pipe was investigated using three different working fluids including water, methanol, and ethanol and the following results were reported:

(a) It was shown that water is the best working fluid among the other working fluids regarding the higher temperature and heat transfer coefficient in the evaporator section.
(b) Thermal resistance of the evaporator section was an order of magnitude higher than that of the condenser section for all the working fluids tested.
(c) At lower condenser temperatures, lower heat transfer coefficients and higher thermal resistances were obtained.
(d) Although the experiments were performed at low heat fluxes, dryout was observed for methanol at the highest condenser temperature. This phenomenon causes the heat transfer coefficient decreases with increasing heat flux, contrary to the usual behavior.
(e) The inclination angle has a great effect on the heat pipe thermal resistance using water as the working fluid.

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