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Heat Pipe Performance Characteristics with Different Operating Fluids and Fill Ratio

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Abstract

The performance of the heat pipe depends largely on the thermofluid properties of the operating fluid used and its filling ratios, which is the focus of current work. In this paper, an experimental study was conducted to evaluate the performance of the heat pipe under different operating conditions. A smooth tube heat pipe with four operating fluids of different thermofluid properties, which are water, ethanol, methanol, and acetone were studied. Investigation parameters were examined at ranging for filling ratios and power inputs between 30-60% and 80-200W, respectively. The experimental main results showed that the best operating fluid that provided the higher performance was water at a 60% filling ratio and a power input of 200W%, while the lowest performance of the heat pipe was with acetone. The increase in heat input led to an increase in the condensation and evaporation heat transfer coefficients of the heat pipe by approximately 87.5% and 45.2%, respectively.

Keywords:

Heat pipe, Power input, Fill ratio, Ethanol, Methanol, Acetone.

1. Introduction:

The heat pipe advantages are characterized by its small size, cheapness, simple construction, and ability to cool down by transferring large amounts of heat in less time with the least possible space. The features of heat pipes are a way to be used with the precision of artificial intelligent technology systems that need to eliminate the generated heat. The heat generated in most systems, if not removed in reliable and ideal ways, leads to a loss of millions of dollars, especially with systems that contain important data for highly confidential and private systems [1]. Therefore, the heat pipe is used on a large scale in most modern technical devices due to its high performance, small size, and thus the heat pipe is an effective strategic option in these applications. The performance of the heat pipe is affected by a set of factors, the most important of which is the type of operating fluid, which depends on the thermofluid properties of the fluids, the percentage of fullness, and the heat input.

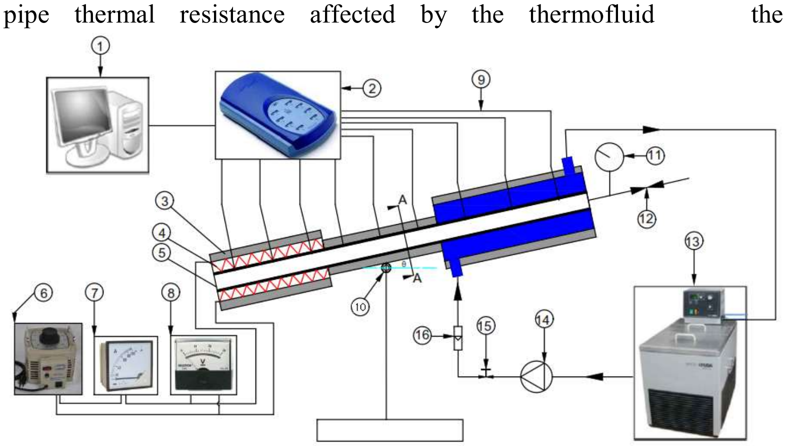
Bezrodly and Alekseenko [2] carried out a heat pipe performance at different filling ratios. The investigation result was recommended the fill ratio must be at least 50% of the evaporator volume to avoid the dry-out phenomenon. Akachi et al. [3] examined the same issue of reference [2] while they founded almost the same findings. T. Payakaruk et al. [4] studied the heat transfer heat pipe characteristics using different working fluids of R123, R134a, R22, ethanol, and water, at different evaporator diameter to length ratios of 5–40%, and different filling ratios of 50–100%. The findings were the filling ratio has no effect on the heat transfer performance, while the thermofluid properties affected the heat transfer coefficient. Y.J. Park et al. [5] experimentally examined a copper tube heat pipe with a diameter of 22 mm, filled by FC-72 with a power input range 50-600 W at different filling ratios of 10–70%. The observed results were that a dry-out phenomenon was depicted at a filling ratio of less than 20% whilst for the large filling ratio of more than 50%, the flooding phenomenon manifested. S.H. Noie [6] investigated experimentally the thermosyphon at different water operating fluid under filling ratios of 30–90% with input power rates of 100–900 W. The results targeted to find the optimum filling ratio at different evaporator lengths. Wang and Peterson [7] analyzed a heat pipe with a sintered copper screen mesh as mainly wick structure at different power input and filling ratios. The predicted results manifested the maximum obtained heat transfer could be 123 W at 50% water filling. Ong et al. [8] studied experimentally a heat pipe with an operating fluid of water and R410a under power input ranging from 100 to 830 W at different filling ratios of 25–100%. The findings depicted that the evaporator wall temperature at higher heat input and lower filling ratio which was not uniform. Robinson and Jouhara [9] experimentally studied the thermosyphon performance filled by water and dielectric heat transfer liquids. The findings were the water charged thermosyphon was outperformed the other working fluids in both the effective thermal resistance as well as maximum heat transport capabilities. P. Nemec [10] experimentally carried out the effect of gravity on the heat pipe at different filling ratios. The predicted finding depicted that the higher characteristics of heat pipe was obtained at higher filling ratios. Rhi and Han [11] investigated a heat pipe with a hybrid nanofluids of Ag/H_2O and Al_2O_3/H_2O at volume concentrations 0.005, 0.05, and 0.1% with various filling ratios and heat input. The findings manifested that a higher thermal resistance was observed at lower both heat inputs and filling ratios. Arab et al. [12] analyzed the thermal resistance of the heat pipe for different working fluids under various thermophysical properties. The finding was the heat pipe thermal resistance affected by the thermofluid properties of operating fluids. Heng Tang et al. [13] studied the heat pipe with different heat inputs and flattened thicknesses of wick structured under different filling ratios. The manifested findings proved that a greater thickness of wick structure with higher filling ratios was enhanced the heat pipe characteristics. Juanwen Chen et al. [14] analyzed experimentally the effect of different operating fluids of water and ethanol on the performance of heat pipe with different heat input range 200-1000 W. The important finding revealed that the water as operating fluid was the higher thermal performance than ethanol. By a careful review of the relevant topic, it was revealed that there are many studies that have studied the thermal performance of the heat pipe with one operating fluid without comparing more than one fluid under different thermofluid properties at multiple filling ratios.

From the above, it appears that there was a gap in the review that shows the effect of the thermofluid properties of operating fluids under different filling ratios and power input, which the point of interest for the current research work. In the present research work, different working fluids under various properties (water-ethanol-methanol-acetone) were studied to evaluate the performance of the heat pipe with different filling ratios under different input power.

2. Experimentation test rig and measuring instruments.

2.1 Test rig apparatus

The experimental test rig consisted of a heat pipe involving measuring devices and chilled water system as depicted in the schematic Fig. 1.



1-computer	6-Variac	11-Pressure gauge
2-Data logger	7-Ameter	12-Filling and evacuation line
3-Thermal insulation	8-Voltmeter	13-Chiller unit
4-Electric heater	9- K type Thermocouples	14-Electrical pump
5-Heat pipe	10-Pivot with protractor	15-Ball valve

Fig. 1 Sketch of the experimental system

The heat pipe for the experimental apparatus was chosen as a smooth copper tube, with an inner diameter of 12.87 mm and a total length of 400 mm. While it consisted of three sections namely, the condenser section with length of 150 mm, adiabatic section of 100 mm, and evaporator section of 150 mm. The two ends of the heat pipe were closed from its ends with caps of the same outer diameter with the installation of a pre-calibrated bourdon-type pressure gauge to measure the operating pressure of the heat pipe.

A refilling line with a valve is also fixed to evacuate and fill heat pipe again with the changing of operating fluids. The heat pipe is centered from its center of gravity on a pivot with a fixed protractor at a constant tilt angle of 60° which recommended by [15-17] as seen in Fig. 2.

Fig. 2 Experimental apparatus test rig

To remove the heat generated from the condenser, the condenser section was surrounded by an outer casing of PVC with an outer diameter of 52 mm. Cold water at a constant temperature was passed through the condenser jacket by a unit of chilled water. The chilled water unit was consisted of a mechanical refrigeration cycle (open compressor, air cooled condenser, TXV expansion device, shell, and helical evaporator) of 5 HP with R-22. The cooling water unit was adjusted and controlled to supply the condenser housing with water cooled at 10 ºC to dissipate the condenser heat. The cooling water was pumped through the condenser housing using a 1 HP-220 V centrifugal pump. A pre-calibrated flowmeter is installed to measure the flow rate of chilled water through the condenser while the flow rate is controlled using ball valves. On the other side, an electric heater with a heating capacity of 1 kW was tightly wrapped along the evaporator section uniformly which made of nickel chrome wire. The heater power was varied by using a voltage regulator associated by a wattmeter. To isolate the heat pipe from the surrounding, a ceramic fiber sheet was wrapped, followed by a layer of glass wool insulation and a layer of armflex insulation with 30 mm thick each, respectively. A thermometer sensor was placed on the outer and inner surfaces of the heat pipe to measure the amount of heat lost from the surface of the heat pipe through the thermal insulation layers. The temperatures are measured at the certain positions a long heat pipe test rig by a pre-calibrated data acquisition unit with K-type thermocouples.

2.2 Thermofluid characteristics of operating fluids

The current study focused on testing and evaluating the performance of the smooth heat pipe with four different types of operating fluids, water, ethanol, methanol, and acetone. Table 1 displays the thermofluid properties of the fluids selected according to the operating conditions of the boiling temperature and the application range.

Table 1 Thermophysical properties of suggested working fluids [18-20].

Fluid type	Application range (˚C)	Boiling temp. (°C)	Thermofluid properties					
			$[h_{fg}]$ (kJ/kg)	$[\rho]$ $(\mathrm{kg/m^3})$		$[\mu]\mathrm{x}10^{-7}$ $(N.s/m^2)$		$[k_L]$ $(W/m.°\mathrm{C})$
				ρ_L	ρ_V	ρ_L	ρ_V	
Water	30–200	100	2256.37	958.7	0.598	2790.2	121.1	0.681
Ethanol	0–130	78.2	962.5	758.2	1.371	4453.5	102.4	0.169
Methanol	0–130	64.6	1119.6	750.6	0.565	3791.2	109.7	0.202

Acetone	0–120	56.3	520.6	748.4	2.121	2340.7	89.3	0.169

2.3 Measuring devices and accuracies

The temperatures were measured through a data logger model (PICO/TC-08) installed with 12 thermocouples of type K and attached directly by a USP cable to a computer to record temperatures directly via Excel sheet. Type K thermocouples were distributed in a manner according to the measurements required for the calculations are used to measure the temperatures on the surface of the heat pipe. Fig. 3 shows the place where the thermocouples are distributed along the heat pipe. Eight thermocouples were tightly fixed along the heat pipe, while 2 other thermocouples were fixed through the insulation to measure the amount of heat lost. In addition, other two thermocouples were installed at the inlet and outlet of the condenser water cooling medium. A thermo-epoxy adhesive glue type was used to fix the thermocouples through selected places.

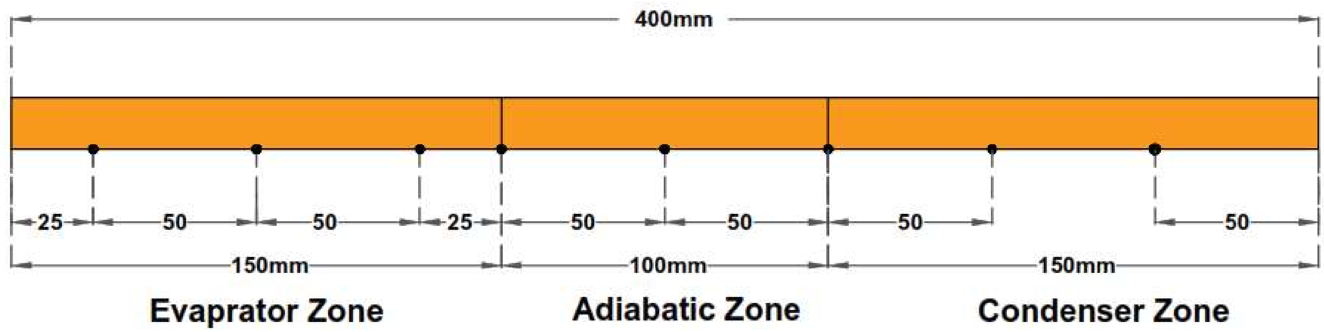


Fig. 3 Positions of thermocouple terminals through heat pipe.

A Bourdon type analog pressure gauge was installed to monitor the operating pressure of the heat pipe through testing. The power of the electric heater around the evaporator was also controlled using the Variac transformer, which was controlled using a variable resistance of the electrical windings. A wattmeter was also connected to the heater circuit to determine the consumed power. The water flow rate was measured through the condenser housing using a rotameter. All measuring instruments in use were calibrated and sufficient time was allowed for the system during operation to stabilize before any measurements were collected. Table 2 gives the accuracy of the instrumental devices used in the experimentations.

Table 2 Accuracy of the measuring devices

Instrument/parameter	Unit	Accuracy (%)	Model
K-type data logger	ºC	±0.1	PICO/TC-08
Pressure gauge	Bar	±0.01	Dura-Hoice PB254L-V30
Flowmeter	Lpm	±1.25	PLATON
Power meter	W	±0.1	A LEYBOLD 727

2.4 Measuring techniques

The present study was carried out with the intention of evaluating the performance of the heat pipe under four different types of thermofluid properties of the operating fluids of, water, methanol, ethanol, and acetone. The filling ratio was changed using a calibrated flask to a range of 30-90%, step 15% with all previous selected fluids. The temperature of the condenser cooling water was fixed at 10 °C ± 0.5 °C with the flow rate measured using a rotameter. Heating power input was varied in the range of 80-200 W ± 0.5 W, step 20 W measured with a wattmeter. The surface temperatures of the heat pipe at eight positions, inlet/outlet temperature of condenser cooling water, and two temperatures before and after the thermal insulation were measured using 12 pre-calibrated K-type thermocouples. The operating pressure of the heat pipe was monitored with a calibrated analog pressure gauge. The heat pipe tilt angle was adjusted to a fixed value of 60° using a protractor and guide arm. Table 3 shows the operation parameters of the current study cases.

Table 3 Experimantation parameters with limitations

Study parameters	Test-1	Test-2	Test-3	Test-4
Working fluid types	Water	Ethanol	Methanol	Acetone
Heat input ranges, $Q^{\cdot}_{in}$ (W)	80-200	80-200	80-200	80-200
Inclination angle ranges, θ (Degree)	60	60	60	60
Filling ratio ranges, FR (%)	30-90	30-90	30-90	30-90
Condenser cooling water temperature, T_{cw} (°C)	10	10	10	10
Pressure of heat pipe, P (kPa)			30	

2.5 Experimental procedures

To ensure reliable results, the heat pipe must be cleaned thoroughly from inside in an appropriate manner after each test of operating fluid has been completed. The heat pipe was triple cleaned continuously with alcohol, DW, then the operating fluid at 80% fill ratio and evacuated each test to a pressure of about 0.5 bar. The heat pipe was checked for leakage over a period of 6 hours under 0.5 bar pressure. The required quantities of operating fluid were charged to the heat pipe according to the research plan. The present research variables are included effect of four types of operating fluid (Water, Ethanol, Methanol, and Acetone). The effect of five filling ratios of 30, 45, 60, 75, and 90% of the ratio of filling volume to the internal volume of the evaporator section was also studied. Experiments were done at the electric heater power ranges (80-200 W). During the implementation of the research experiments, it was confirmed that constant pressure of 30 kPa was maintained.

Experiments began by adjusting the electrical energy input to the electric heater at a certain value via Variac. When the operating pressure of the heat pipe reaches a value of 30 kPa, it was kept constant by controlling the flow rate of the cooling water to approach the steady-state during each operation. The temperatures of all thermocouples were monitored until a steady state was achieved over time, reaching a time of 45 minutes.

2.6 Uncertainty of calculated parameters

The uncertainty resulted from the computations for the measured data was analyzed through experiments using a differential approach methodology of the approximation analysis [21]. Uncertainties were predicted in the parameters computed according to Kline and Mc. Clintock method [22]. The maximum uncertainties of the calculated parameters are given in Table 4.

Table 4 Uncertainty of the study parameters

Instrument/parameter	Unit	Uncertainty
Heat power input	W	±2.51
Thermal resistance	ºC/W	±7.87
Evaporation heat transfer coefficient	W/m². ºC	±6.7
Condensation heat transfer coefficient	W/m². ºC	±7.1

3. Data reduction

The heat input of heat pipe electric heater $Q^{\cdot}_{h}$ is calculated as:

$$Q^{\cdot}_{h} = I.V \qquad (1)$$

where the net input power $Q^{\cdot}_{net}$ is estimated as $Q^{\cdot}_{net} = Q^{\cdot}_{h} - Q^{\cdot}_{loss}$ (2)

The total heat loss $Q^{\cdot}_{loss}$ by convection and radiation can be calculated by using the following equations 3-5.

$$Q^{\cdot}_{loss} = Q^{\cdot}_{conv} + Q^{\cdot}_{rad} \quad (3)$$

$$Q^{\cdot}_{conv} = h_{conv}.A_e(T_{ins} - T_{sur}) \quad (4)$$

$$Q^{\cdot}_{rad} = \in.\delta.A_e(T_{ins}^4 - T_{sur}^4) \quad (5)$$

While the convection heat transfer coefficient h_{conv} can be predicted by correlation [23] as:

$$Nu = \frac{h_{conv}.\,L_{tot}}{K_{sur}} = \left[0.825 + \frac{0.387 \quad L^{\frac{1}{6}}}{\left[1+\left\{\frac{0.492}{pr}\right\}^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right]^2 \quad (6)$$

Where Ra is the Rayleigh number that can be calculated as:

$$Ra = \frac{g.\beta(T_{ins}-T_{surr})L^3}{\alpha.v} \quad (7)$$

The evaporator section wall heat flux q_e is computed according to:

$$q_e = \frac{Q^{\cdot}_{net}}{A_e} \quad (8)$$

The evaporator section heat transfer coefficient he is given as:

$$h_e = \frac{q_e}{T_e - T_a} \quad (9)$$

The condenser section heat transfer rate $Q^{\cdot}_{cond}$ is calculated as:

$$Q^{\cdot}_{cond} = \dot{m}_w.c_w(T_{wo} - T_{wi}) \quad (10)$$

The condenser section heat flux q_c is obtained as:

$$q_c = \frac{Q^{\cdot}_{cond}}{A_c} \quad (11)$$

The condenser section heat transfer coefficient h_c is computed as:

$$h_c = \frac{q_c}{T_a - T_c} \quad (12)$$

The heat pipe thermal resistance R_{th} can be given from:

$$R_{th} = \frac{T_e - T_c}{Q^{\cdot}_{net}} \quad (13)$$

4. Results and data analysis

In this section, the current research are evident to demonstrate the effect of using four different types of operating fluids at various thermofluid properties on heat pipe performance with variable filling ratios and heat inputs as follows:

4.1. Influence of working fluid type

Four different working fluid types of (water, ethanol, methanol, and acetone) at different power input ranges (80–200 W) with a constant inclination angle of 60º, condenser water cooling temperature of 10 ºC, and filling ratio of 60% are investigated through this section.

Figure 4 depicts the effect of the heat pipe thermal resistance against heat inputs with ranges of (80 to 200 W) with an interval of 20 W under different four working fluids (water-ethanol-methanol-acetone) at a fixed θ=60°, $T_{cond.}$= 10 ºC, and FR=60%. It can be seen from the figure that the heat pipe thermal performance decreases with the increase in thermal input for all types of working fluids at the same conditions. The reason for the decrease in the thermal resistance with the increase in the heat input is referred to the fact that the thermal resistance of all operating fluids at start-up is high due to the heat absorption in the evaporator section and because of the temperature difference between the condenser and the evaporator is high and thus the thermal resistance increases [24]. Further, at the same heat input of 160 W, the higher performance working fluid is found with water, followed by ethanol, followed by methanol, then acetone. The lowest value of the total thermal resistance is predicted at 0.52 °C/W with water at a heat input of 200 W. The improvement rate of thermal resistance for water as a working fluid compared to the fluid of acetone at the same heat input of 160 W is 20.2%. Moreover, it can be observed that the working fluid of water operates at a higher temperature range than any other fluids at the higher input power of 200 W. This is due to the specific heat of water, which has a greater value than any other fluids.

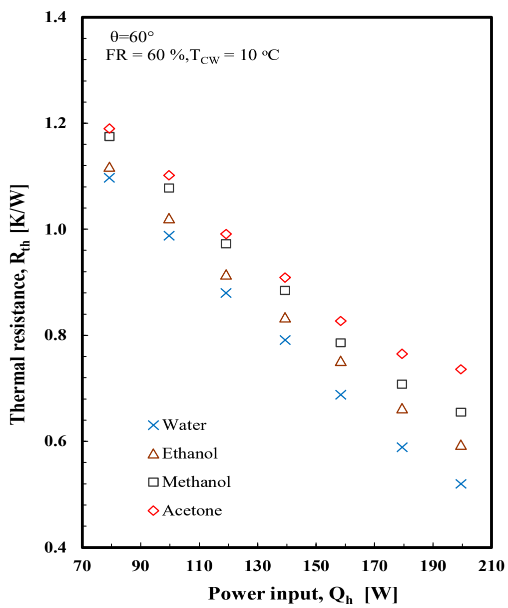


Fig. 4 Thermal resistance of the heat pipe versus the input power for different fluid types

Figure 5 displays the effect of the evaporative heat transfer coefficient with the heat input. It is evident from the figure that the working fluids greatly affect the evaporative heat transfer of the heat pipe. The evaporation heat transfer coefficient is strongly increased by the increase of the heat input for all types of working fluids. The thermal properties of the working fluids affect the performance of the heat pipe. Hence the ability of the working fluid to absorb heat depends on the amount of the working fluid and its physical properties [25]. The evaporation process increases with water as the working fluid and the evaporation decreases relatively with ethanol as the working fluid, while the evaporation heat transfer coefficient decreases strongly with methanol and acetone, respectively. The results depend on the difference in the physical properties of the working fluids (density and viscosity). The improvement ratio in the evaporative heat transfer coefficient is achieved with water as working fluid when compared to ethanol, methanol, and acetone by about 5, 10% and 15%, respectively at (Q_h=180 W, θ=60°, $T_{Cond.}$ = 10 °C, FR=60°).

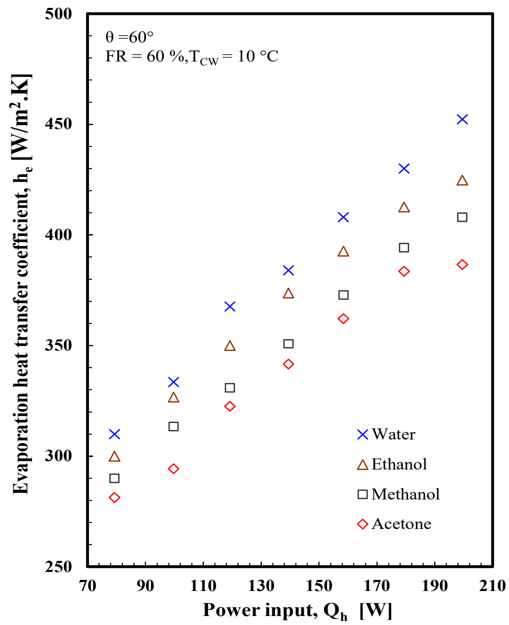


Fig. 5 Evaporation heat transfer coefficients versus power input for different fluid types

Figure 6 presents the variation of the condensation heat transfer coefficient with the power input. The results from experimentations of different working fluids found that the HTC by condensation is increased with all values of power input under all working fluids. The figure also manifests that the maximum improvement of the condenser heat transfer coefficient due to the use of water as the working fluid of the heat pipe is higher than other working fluids of ethanol, methanol, and acetone by approximately 6%, 9%, and 16%, respectively at a constant power input of 200 W. Heat transfer through heat pipe occurs in the condensation zone by means of film condensation [26] which is higher for water than other working fluids.

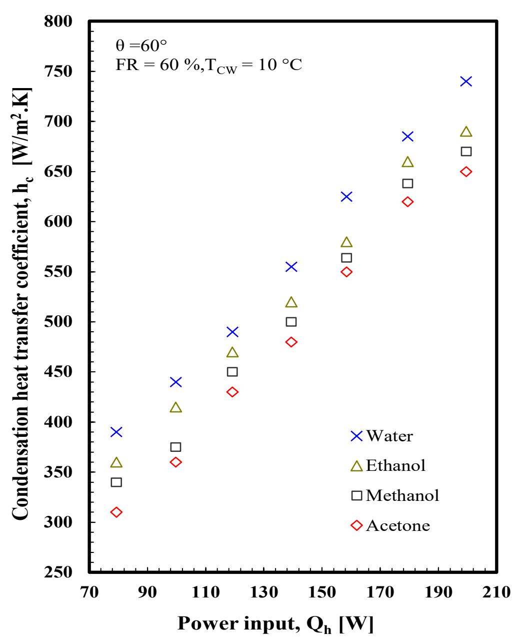


Fig. 6 Condensation heat transfer coefficients versus power input for different fluid types

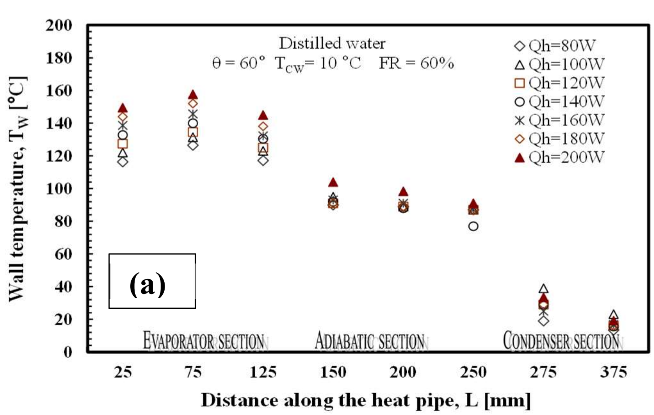
4.2. Influence of power input

The effect of the change of power input ranges (80–200 W) on the surface temperature distribution of heat pipe with a constant inclination angle of 60º, condenser water cooling temperature of 10 ºC, and filling ratio of 60% are experimentally investigated.

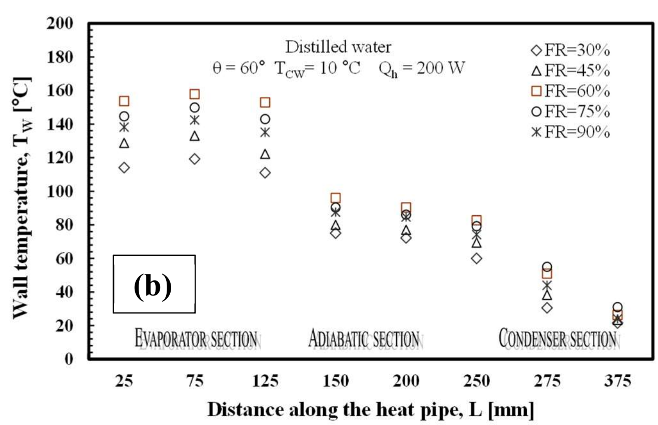
The temperature distribution through the heat pipe surface is plotted at fixed distances. The experiments are conducted using the operating fluid (distilled water). Figure (7a) shows the temperature distribution along the smooth tube heat pipe for different power inputs of a range 80 to 200 W with a constant increase rate of 20 W. The temperature distribution begins from the higher point at the evaporator zone that increased to a maximum value through the middle of the evaporator and then starts to decrease at the end of the evaporator zone. While, at the adiabatic zone the temperature is decreased and almost constant through this zone due to the thermal insulation effect. A sharp temperature drop to condenser zone is observed whereas through the condenser section the temperature remains almost constant. The difference in temperature distribution is due to the difference in the latent heat of the water as a working fluid which characterizes the heat pipe performance.

Figure (7b) depicts the distribution of heat pipe wall temperature for different filling ratios ranges 30-90% at constant θ=60º, T_{CW}=10 ºC, and Q_h=200 W. It can be seen from the figure that the temperature distribution relation is seemed the same trend with different values according to the working fluid filling ratios. Also, the temperature distribution through the evaporator section is affected by the filling ratio more than the other sections of the heat pipe (adiabatic section and condenser section). The filling ratio of 30% achieved a lower temperature distribution causing a dry out due to a lower working fluid which evaporates quickly for higher power input (200 W). While increasing the filling ratio, a remarkable increase of heat pipe wall temperature is observed and consequently increases the heat pipe performance. The higher performance is obtained with a filling ratio of 60%. This can be attributed to a lower thermal resistance with a smooth tube heat pipe at a fixed tilt angle of 60º and a power input of 200 W.

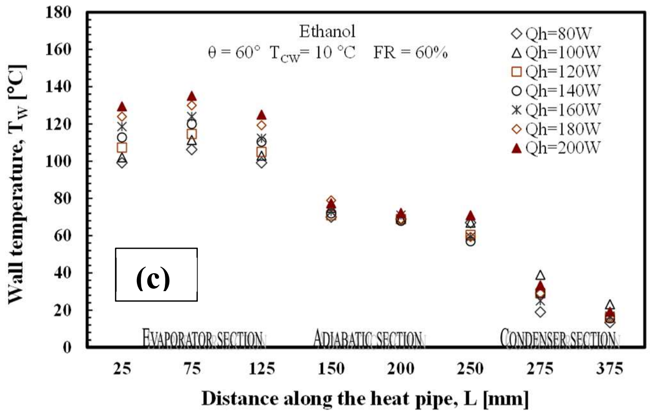
Figures (7c, 7d, 7e, 7f, 7g, and 7h) also showed the same trend of heat transfer mechanism through a temperature distribution of the heat pipe according to the different thermophysical properties of the working fluid, which played an important role in controlling the performance characteristics of the heat pipe. The results manifested that the best thermal performance provides a large operating range of temperatures by distilled water, where the maximum temperature was achieved in the middle of the evaporator of T_e=157.7 °C at Q_h=200 W. Ethanol was followed by its thermophysical properties to achieve the maximum temperature in the middle of the evaporator of T_e=136.5 °C at Q_h=200 W. Methanol was followed, achieving the maximum temperature in the middle of the evaporator of T_e=125.6 °C at Q_h=200 W. While acetone came in the rear, it achieved the maximum temperature in the middle of the evaporator of T_e=120.4 °C at Q_h=200 W. This explanation is due to the fact that the operating fluid is a factor affecting the range of operating temperatures of the heat pipe due to the difference in viscosity and the physical properties that are represented in the small diameter of the three other operating fluids droplets from the water droplet. The difference in temperature distribution is also due to the difference in the latent heat of evaporation. Acetone provides the lowest latent heat for evaporation while water has the highest latent temperature of evaporation. The previous figures confirm that water provides higher thermal performance than other operating fluids with a constant filling ratio of 60%.



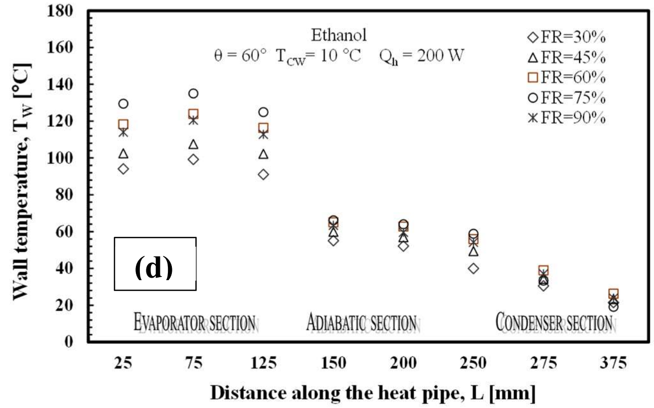
(a) Distribution of wall temperature under different power inputs for distilled water as operating fluid



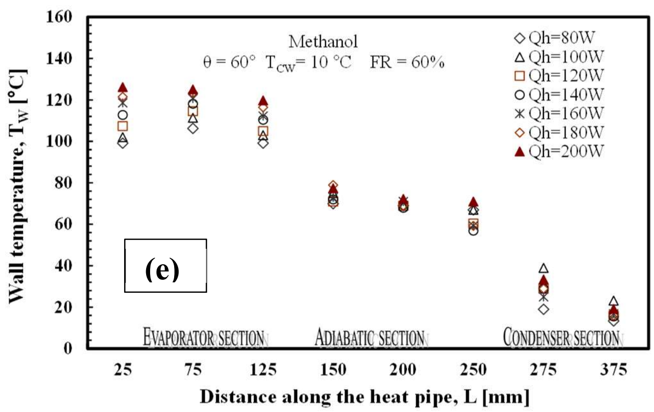
(b) Distribution of wall temperature under different filling ratios for distilled water as operating fluid



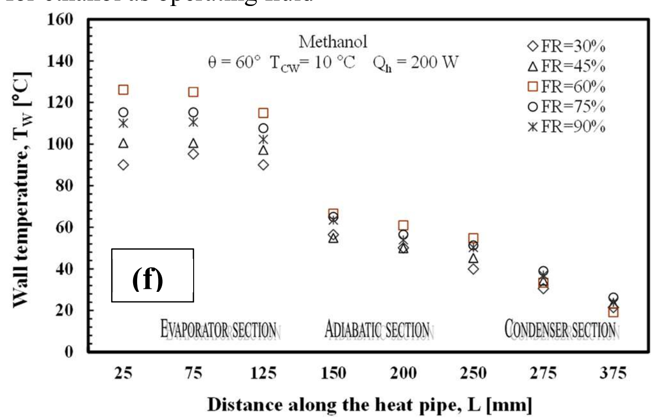
(c) Distribution of wall temperature under different power inputs for ethanol as operating fluid



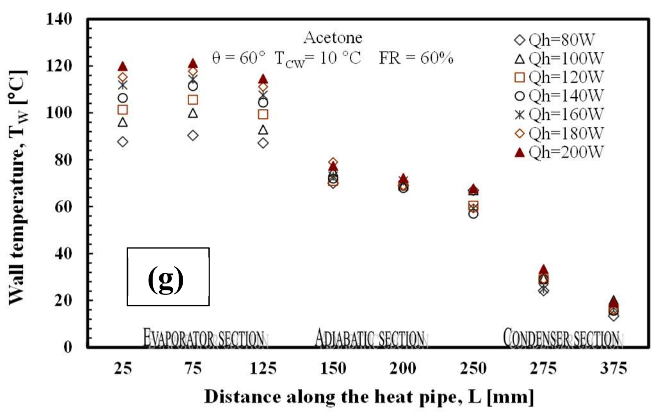
(d) Distribution of wall temperature under different filling ratios for ethanol as operating fluid



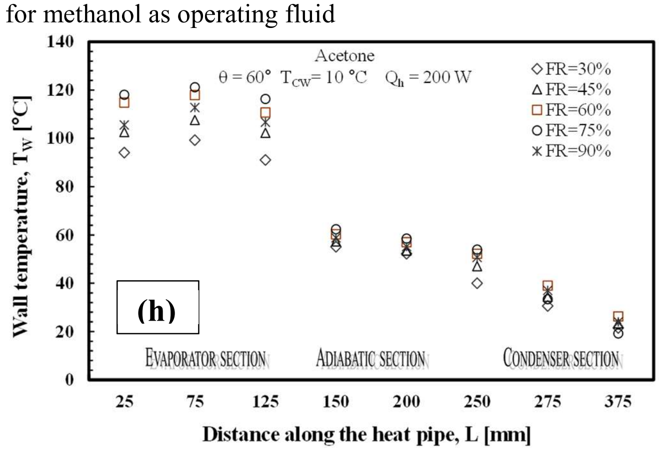
(e) Distribution of wall temperature under different power inputs for methanol as operating fluid



(f) Distribution of wall temperature under different filling ratios for methanol as operating fluid



(g) Distribution of wall temperature under different power inputs for acetone as operating fluid



(h) Distribution of wall temperature under different filling ratios for acetone as operating fluid

Fig. 7 Distribution of wall temperature a long heat pipe under different power inputs and filling ratios for various operating fluids

4.3. Influence of filling ratio

Under this sub-section, the performance of the smooth tube heat pipe with different filling ratios (30-90%) of water as a working fluid with the constant inclination angle of 60% and the temperature of the condenser cooling fluid is T_{CW}=10 ºC at various values of the power input of (80-200 W) is studied.

The effect of total thermal resistance versus changing the filling ratio of the evaporator charged with DW with various power input values from (80 to 200 W) is shown in the Fig. 8. It can be illustrated the relationship between the thermal resistance and the filling ratio that with an increase in the filling ratio, the thermal resistance decreases with a non-linear decrease, and consequently, the performance of the heat pipe increases to a specific filling ratio value at 60%, and then the thermal resistance rises with any increase in filling ratio for all power input values. The thermal resistance is high at the start of operation due to the large absorption of liquid in the evaporator with low filling ratios of the operating fluid and transfers to vapor, which provides rapid evaporation and creates a large temperature difference between the condenser and evaporator zones, and the thermal resistance increases accordingly [24]. Also, it can be noted from the figure that the higher power input leads to a decrease in the thermal resistance due to the higher evaporation of the working fluid and thus the increase in performance compared to the lower power input.

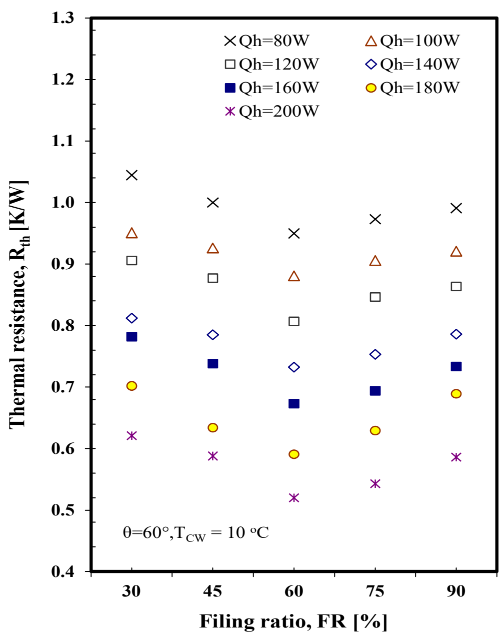


Fig. 8 Thermal resistance versus filling ratios for different power inputs

Figure 9 shows the effect of the evaporation heat transfer coefficient versus filling ratio under different power input ranges (80 to 200 W) with DW as operating fluid. It is clear from the figure that the filling ratio of the heat pipe greatly affects the evaporation heat transfer coefficient for all values of power inputs. The evaporation heat transfer coefficient of the heat pipe increases and strongly affected by the increase in the power input through the evaporator. The evaporation process maximizes and the difference between the power input is observed from 80 W when the power input is increased to 140 W. A good agreement for all values of power inputs has been reported that the best filling ratio is achieved with the highest heat transfer by about 60%, especially at low power input. In addition, the evaporation heat transfer coefficient increases with the power input reaching 200 W. The result may be due to the boiling point of working fluid which affected with the higher power input [27]. The highest value of evaporation heat transfer coefficient reaches 433 W/m^2 at 60% fullness which achieved the maximum improvement rate compared to 80 W at the same filling ratio by about 48%.

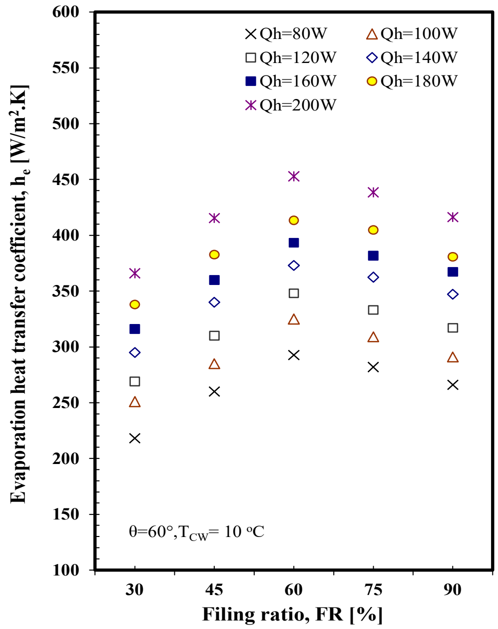


Fig. 9 Evaporation heat transfer coefficient versus filling ratio for different power inputs

Figure 10 manifests the effect of condensation heat transfer coefficient against filling ratios under various power input at constant operating conditions of (θ=60°, T_{CW}=10 °C, and DW). The experimental results depicted that the HTC has increased rapidly in condenser region of heat pipe. Moreover, the figure shows an increase in the condensation heat transfer coefficient gradually with the power input. The maximum limit to improve the condensation heat transfer coefficient at 60% fullness which decreased the thermal resistance. The improvement in condensation heat transfer coefficient at 200 W compared to 80 W is found 74% under the same filling ratio.

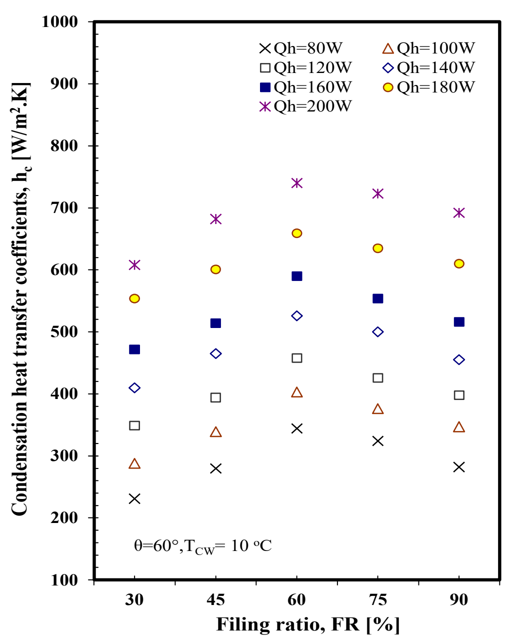


Fig. 10 Condensation heat transfer coefficient versus filling ratio for different power inputs

5. Conclusions

Due to its versatility in various applications and its ability to transfer high rates of heat in less time and space, heat pipe needs more experimental details about its performance with multiple operational conditions. The current research work experimentally studied the thermal performance characteristics of a heat pipe with four operating fluids under different thermophysical properties, namely (water-

methanol-ethanol-acetone) with various filling ratios and power inputs with a range of (30-90%) and (80-200 W), respectively. Experiments were carried out under constant conditions in terms of the inclination angle of the heat pipe at 60° and the temperature of the condenser cooling water at 10 °C. The most important results showed that the best heat pipe filling ratio provides the highest performance, which is 60% of the internal evaporator volume, for its ability to transfer the required amount of heat with no dryness or excessive flooding. In addition, the increase in the input power rates, which reaches 200 W, led to an improvement in the heat pipe condensation and evaporation convective heat transfer coefficients by approximate ratios of 87.5% and 45.2%, respectively. Using water as a heat transfer media achieves the lower thermal resistance and consequently a higher thermal performance for the heat pipe compared to other operating fluids of ethanol, methanol, and acetone by about 25%, 41.7%, and 50%, respectively.

Nomenclature

Symbol	Description	Symbol	Description
A_C	Condenser outside surface area, m^2	R_{th}	Total thermal resistance of heat pipe, °C/W
A_e	Evaporator outside surface area, m^2	T_a	Adiabatic section average temperature, °C
c_w	Specific heat of water, J/kg.°C	T_c	Condenser section average temperature, °C
g	Local acceleration due to gravity, m/s^2	T_{CW}	Condenser water cooling inlet temperature, °C
h_c	Condenser heat transfer coefficient, W/m^2.°C	T_e	Evaporator section average temperature, °C
h_{conv}	Convective heat transfer coefficient, W/m^2.°C	T_{ins}	External surface temperature of insulation, °C
h_e	Evaporator heat transfer coefficient, W/m^2.°C	T_{sur}	Surrounding temperature, °C
h_{fg}	Latent energy of vaporization, J/kg	T_{wi}	Inlet temperature of cooling water, °C
I	Electric current, A	T_{wo}	Outlet temperature of cooling water, °C
k_{sur}	Thermal conductivity of air, W/m.°C	V	Voltage drop, V
L	Total length of heat pipe, m	**Greek Letters**	
m_w	Cooling water mass rate, kg/s		
Pr	Prandtl number W	ϵ	Emissivity factor
$\dot{Q}_h$	Heat input power W	δ	Boltzmann constant, W/m^2.°C^4
$\dot{Q}_{net}$	Net input power	β	Volumetric thermal expansion coefficient,
$\dot{Q}_{rad}$	Radiation heat rate W	ρ	Volumetric mass density, kg/m^3
$\dot{Q}_{loss}$	Heat loss, W	ν	Kinematic viscosity of the air, m^2/s
$\dot{Q}_{cond}$	Outlet heat by condensation, W	**Abbreviation**	
$\dot{Q}_{conv}$	Convection heat transfer rate, W	FR	Filling ratio of evaporator volume %
q_c	Condenser heat flux, W/m^2	HTC	Heat transfer coefficient W/m^2.K
q_e	Evaporator heat flux, W/m^2	FHP	Flat heat pipe

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