Experimental Study on Thermal Performance of a Loop Heat Pipe with Different Working Wick Materials

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Abstract: In loop heat pipes (LHPs), wick materials and their structures are important in achieving continuous heat transfer with a favorable distribution of the working fluid. This article introduces the characteristics of loop heat pipes with different wicks: (i) sintered stainless steel and (ii) ceramic. The evaporator has a flat-rectangular assembly under gravity-assisted conditions. Water was used as a working fluid, and the performance of the LHP was analyzed in terms of temperatures at different locations of the LHP and thermal resistance. As to the results, a stable operation can be maintained in the range of 50 to 520 W for the LHP with the stainless-steel wick, matching the desired limited temperature for electronics of 85 °C at the heater surface at 350 W (129.6 kW·m⁻²). Results using the ceramic wick showed that a heater surface temperature of below 85 °C could be obtained when operating at 54 W (20 kW·m⁻²).

Keywords: electronics cooling; loop heat pipe; wick materials; thermal conductivity

1. Introduction

Loop heat pipes use a passive two-phase heat transport device in which advantageous characteristics are offered, such as operation against the gravity through the capillary effect and flexible characteristics. They have been widely used in energy applications, spacecraft thermal control, electronic device cooling, and commercial radiators [1]. Zhou [2] fabricated a plate-type loop heat pipe (LHP) and investigated the effect on the heat transfer performance of the LHP with multilayer metal foams as a wick structure. Their flat evaporator’s experimental results show that multilayer copper foams have better performance than nickel foam because of higher thermal conductivity and smaller pore size. Siedel [3] presented a numerical simulation, comparing it with the experimental data of a flat disk-shaped evaporator. The results showed that the larger effective thermal conductivity of the wick has an influence on the overall thermal performance of the LHP. However, other researchers, Maydanik [4] and Hoang [5], suggested the wick material, with its lower effective thermal conductivity, to handle the problem of heat leakage. In addition, the hydrophilic effect on the ceramic wick structure is higher than that of stainless steel [6]. Therefore, the performances of stainless-steel wicks and ceramic wicks in LHPs are compared in this study, of which the ceramic wick has much lower thermal conductivity and a higher hydrophilic effect [7,8]. In an LHP system, the porous wicks’ structure is essential because it provides the capillary force for working fluid circulations, a liquid flow path, and a place for phase change heat transfer. Moreover, the porosity effect of the wick materials inside the evaporator is necessarily considered for improving the overall performance of the system.
LHPs [9]. Therefore, the porosity, pore diameter, and thermal conductivity of the wicks were measured. The influence of the different wicks on the performance of LHPs is tested with the designed evaporator. Although this is a simple case study of an arrangement with different wicks, the appropriate engineering approach of the calculation procedure related to applying the designed evaporator is presented in this paper. It may improve the understanding of the problems that exist in the LHP evaporator and future LHP systems.

The LHP has some similar operation principles with conventional HPs. The phase changing process is happened in the LHP system and the capillary force is used as the motivation for the operation. Figure 1 demonstrates the scheme of the analytical LHP and an operation cycle diagram.

At first, the system is not supplied with heat load. In this case, the liquid stays at Level A–A as an assumption. The liquid found at the evaporator zone (Point 1) and the compensation chamber evaporates from the wick when heat is supplied to the evaporator. The vapor from the evaporator flows. Then, it approaches to the heating wall. As a result, the vapor pressure reduces. Simultaneously, the temperature rises a little (Point 2) and is higher than the vapor located at the compensation chamber. In this situation, the wick behaves as a thermal barrier. The vapor which is superheated condition in the evaporator zone cannot pass the compensation chamber through the wick which is saturated due to the force of capillary that keeps the liquid inside. Then, the hydraulic lock is happened in the wick. Afterwards, the vapor continuously flow to the condenser’s inlet (Point 3). When vapor flows from Point 2 to Point 3, both temperature and pressure decrease. The de-superheat, condensation, and subcooled processes happen from Point 3 to Point 5. In this case, there is no pressure loss from Point 3 to Point 5 as an assumption. Because of the pressure loss and the hydrostatic resistance caused by friction, the pressure difference \( \Delta P_{56} \) includes the pressure loss. Afterward, the liquid located at Stage 6 flows into the chamber for compensation. Moreover, some parts of the heat load are provided to the evaporator at the expense of the working fluid. This condition is heated to temperature \( T_7 \). The progress from Point 7 and Point 8 are relative to the filtration of the liquid through the wick into the evaporation place. In this way, the liquid may prove to be superheated. However, its boiling-up does not happen because of its short duration in such a state. The state of the working fluid in the vicinity of the evaporating menisci is represented by Point 8.

![Figure 1. Principles of LHP: (a) Scheme of analytical LHP. (b) LHP working cycle diagram. Reproduced from [1], Yu.F.Maydanik: 2005.](image-url)
Pressure loss \((dP_{1-8})\) corresponds to the total value of pressure losses in all the working-fluid circulation sections. From the above analysis, the function of the LHP is considered in three conditions. At first, the capillary condition is the condition for conventional HPs for an operation.

\[
\Delta P_c \geq \Delta P_v + \Delta P_l + \Delta P_g \tag{1}
\]

where

- \(\Delta P_v\) is the working fluid’s pressure loss during the vapor state’s motion.
- \(\Delta P_l\) is the working fluid’s pressure loss during the liquid state’s motion.
- \(\Delta P_g\) is the pressure loss because of the hydrostatic of the liquid column.
- \(\Delta P_c\) is the wick capillary pressure.

The second condition is just for the LHP. At the LHP’s startup, it ensures that the liquid is exhibited from the evaporating zone to the compensation chamber.

\[
\frac{\partial P}{\partial T}_{T_v} \Delta T_{1-7} = \Delta P_{EX} \tag{2}
\]

where

- \(\frac{\partial P/\partial T}{\partial T}\) is the derivative of the saturated line’s slope.
- \(T_v\) is the mean temperature between \(T_1\) and \(T_7\).
- \(\Delta P_{EX}\) is the total pressure loss in all the sections of the working fluid’s circulation except the wick.

At the third condition, the liquid is prevented from steaming in the liquid line because of the ambient pressure loss and heating.

\[
\frac{\partial P}{\partial T}_{T_v} \Delta T_{4-5} = \Delta P_{5-6} \tag{3}
\]

where

- \(\frac{\partial P/\partial T}{\partial T}\) is the derivative of the saturation line’s slope.
- \(T_v\) is the mean temperature between \(T_4\) and \(T_5\).
- \(\Delta P_{5-6}\) is the pressure loss in total from State 5 to State 6.

The purpose of the tested LHP in this study is the cooling of electronics such as processors located at Data Centers (DCs). Therefore, the evaporator surface which is heated needs to be flat to improve the evaporator and electronics’ contact quality. Moreover, it will make the heat flux \(q\) and distribution of temperature on the active surfaces uniform and eliminate the occurrence of mounting blocks at the evaporator. The investigated evaporator design belongs to the evaporator group with opposite replenishment (EOA). In this design, liquid flows from the top to the bottom surface of the wick structure, as shown in Figure 2a. Figure 2b shows the evaporator with longitudinal replenishment (ELR). The liquid from ELR is provided from the compensation chamber which is located behind the wick. Otherwise, the liquid flows perpendicular to the heat flow rate. The evaporator with opposite replenishment own a simple structure and the liquid absorption surface is large. However, the evaporator can become thicker because the compensation chamber is located above the capillary system. Moreover, the heat leak from the evaporator to the compensation chamber through the wick will be more severe because of the large cross-section of the wick.
The investigated evaporator has two parts: the vapor collector and the wick’s space and compensation chamber. The copper plate is used for the separation between two elements. The brazing method is used to fix them. Additionally, to install the thermocouples, a hole with a 1-mm diameter and 22.5 mm for the length was fabricated at the evaporator’s base. The heating block’s top surface and the evaporator’s bottom surface have the same area: 27 cm$^2$ (45 × 60 mm). This dimension was considered based upon the specifications of some processors, as presented in Table 1. Different LHP technologies focussing on various designs of the LHP’s evaporator and other types of LHPs such as miniature LHPs, micro LHPs, evaporator with longitudinal replenishment (ELR) LHPs, and LHPs with parallel condensers are compared and summarized in Table 2.

Table 1. Processor specifications.

| No. | Modern          | Thermal Power Design (W) | Case Dimensions (mm × mm) | Heat Flux (W/cm$^2$) |
|-----|-----------------|--------------------------|---------------------------|----------------------|
| 1   | Core i7 5960X   | 140                      | 52.5 × 45                 | 5.9                  |
| 2   | Core i7 5930K   | 140                      | 52.5 × 45                 | 5.9                  |
| 3   | Core i7 4960X   | 130                      | 52.5 × 45                 | 5.48                 |
| 4   | Core i7 4930X   | 130                      | 52.5 × 45                 | 5.48                 |
| 6   | Core i7 3790X   | 150                      | 52.5 × 45                 | 6.3                  |
| 7   | Xeon E7 8891 v3| 165                      | 52 × 45                   | 7.05                 |
| 8   | Xeon E7 8880 v3| 150                      | 52 × 45                   | 6.41                 |
| 9   | Xeon E7 8890 v2| 155                      | 52 × 45                   | 6.62                 |
| 10  | Itanium 9300    | 185/155/130              | 48.5 × 40.25              | 9.47/7.94/6.66       |
| 11  | Itanium 9500    | 170/130                  | 48.5 × 40.25              | 8.71/6.66            |
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Table 2. Different LHP technologies from the literature.

| No. | References | Specifications of LHPs from the Literature | Objects and Results |
|-----|------------|------------------------------------------|---------------------|
| 1.  | K. Fukushima—2017 [11] | Porous polytetrafluorethylene wick $17 \times 9 \times 2$ mm (PTFE) (50%, 2.2 $\mu$m, $6.48 \times 10^{-14}$ m$^2$, 0.25 W/m·K) | Objects: A wick with a liquid core is proposed. An experimental and computational investigation was conducted. The temperature distribution inside the evaporator and the heat load’s break down from the mathematical model are obtained. Results: Minimum $R_{LHP}$ is 1.2 K/W and maximum $Q = 11$ W, from the experiment. |
| 2.  | M. Nishikawara—2017 [12] | PTFE wick (bulk thermal conductivity = 0.25 W/m·K). Stainless steel case ($k = 16$ W/m·K) Machined widths (minimum): 0.3 (circumferential groove) and 0.4 mm (axial groove) Increasing axial grooves will reduce the number of circumferential grooves. Three wicks were fabricated for experimental examination: $L_{tr} = 3150$ /m (4 axial × 71 circumferential) $L_{tr} = 2630$ /m (16 axial × 56 circumferential) Classical wick with only 16 axial grooves and 1 mm width of the groove | Objects: It presents a method for the wick shape optimization via calculation and experiment. Using only the three-phase contact line’s length, the $h_{tr}$ is maximized ($q = 2$ W/cm$^2$). Effects of the case, wick material, and the working fluid, are discussed. Results: The effect of the three-phase contact line (TPCL) on the groove pressure loss (by calculation) is shown. Comparison between evaporator heat-transfer coefficient (HTC) obtained using Equation (9) and that obtained through the experiments (ethanol) is presented. The heat transport’s contribution at TPCL was estimated at 0.87 when fitting to experiment results and 0.63 in simulation. Comparison of various working fluids and wick materials, $h_{tr}$ increased with wick’s thermal conductivity. The value of $h_{tr}$ was higher for ammonia due to the changes in interfacial HTC. |
### Table 2. Cont.

| No. | References | Specifications of LHPs from the Literature | Objects and Results |
|-----|------------|-------------------------------------------|---------------------|
| 3.  | Jinliang Xu—2014 [13] | LHP, evaporator, heat transfer, modulated porous wick | It is to enhance heat transfer of pool boiling using the modulated porous wick sintered on the heater wall. Three types of evaporators: MWE (microchannel/wick evaporator), MME (modulated monoporous wick evaporator), and MBE (modulated biporous wick evaporator) were fabricated. |
|     | Keywords: | LHP, heat transfer, evaporation | |
|     | (2 mm–149 μm), and third absorbent wool layer | Evaporator: ϕ80 × 10 (without CC thickness); | Results: MBE LHP reduces the startup time and achieves more stable operation than MWE. At heat flux of 40 W/cm² (heater heat flux), the MBE LHP can operate when \( T_c \) is around 63 °C; \( R_t = 0.12 \) K/W) Optimum CR = 51.3%. Operation antigravity condition is better than others with the MBE LHP's proper design. Best fin's geometric parameters: \( h = 1.5 \) mm; \( p = 1.5 \) mm; \( w = 3 \) mm; best particle size: 88 μm |
|     | (ϕ80 × 10 (without CC thickness); | Vapor line: ID6/OD8 × 550 mm | |
|     | | Liquid line: ID6/OD8 × 300 mm | |
|     | | Condenser: 130 × 130 × 25 | |
|     | | CR = 38.5%, 51.3%; 64.1%, 64.1%, 76.9% | |
|     | | Sintering process: oven temperature 900 °C for 4 h. | |
| 4.  | S.C. Wu—2014 [14] | Wick structure, LHP, evaporator area, grooves | The effects of increasing the number of grooves on a wick’s surface on LHP performance were investigated. |
|     | Keywords: | | |
|     | | LHP with cylindrical stainless-steel evaporator (ϕ16 × 65) | |
|     | | Water cooling | |
|     | | Wick material: nickel (ID/OD = 9/12.5); largest \( r_{pore} = 1.9 – 2.5 \) μm; \( K = 1.3 – 3.25 \times 10^{-13} \) m²/°C, porosity: 63–67%. | |
|     | | Vapor line: ID5/OD6 × 470 | |
|     | | Liquid line: ID4.5/OD6 × 585 | |
|     | | Condenser: ID5/OD6.4 × 800 | |
|     | | Ammonia as working fluid. | |
|     | | | |
|     | | | Results: The wick with 16-groove was quickly damaged. The other wicks have similar properties, such as porosity, pore radius, \( K \). Sintering condition: 45 min at 600 °C. Increasing the groove number increases the LHP’s performance \( (Q = 500 \) W, \( R_t = 0.14 \) K/W). An optimal number of grooves fabricated on the wick surface is presented. |
Table 2. Cont.

| No. | References | Specifications of LHPs from the Literature | Objects and Results |
|-----|------------|------------------------------------------|--------------------|
| 5.  | Jeehoon Choi—2013 [15] | Miniature LHP, evaporator, sintering, contact conductance, thermal resistance | Objects: For fabricating the LHP’s evaporator, a low-cost sintering method is explored. The porous material partially fills the vapor collection channel embedded in the evaporator’s base; two evaporators were fabricated. |
|     |            | Flat-disk shape evaporator. Wick material = Nickel (ϕ42 × 3), particles size: 3 μm, \( P_{\text{capillary}} = 401 \text{ kPa} \), \( K = 0.99 \times 10^{-11} \text{ m}^2 \), porosity = 64%, \( k_{\text{eff}} = 9 \text{ W/K m} \) | |
|     |            | CC: stainless-steel (ϕ46 × 7) | |
|     |            | Vapor line: ID4.95/OD6.35 × 250 (made by copper) | |
|     |            | Liquid line: ID4.95/OD6.35 × 300 (made by stainless-steel) | |
|     |            | Forced convective air cooling | |
|     |            | Working fluid: water | |
|     |            | Horizontal orientation | |
|     |            | Heater surface area: 30 × 30 mm² | |
| 6.  | Randeep Singh—2008 [16] | Miniature LHP using flat disk-shaped copper evaporator (ϕ30 × 10 mm) | Objects: For thermal control of small electronic equipment, it addresses the thermal characteristics of mLHP using a flat disk-shaped evaporator. |
|     |            | Working fluid: water Nickel wick (thickness = 3 mm, \( r_{\text{pore}} = 3–5 \mu \text{m} \); porosity: 75%) | |
|     |            | Condenser using Air forced cooling \( (T_a = 22 ± 2 \text{ °C}) \) | |
|     |            | Copper Vapor line: \( \phi 2 \times 150 \text{ mm} \) | |
|     |            | Copper Liquid line: \( \phi 2 \times 290 \text{ mm} \) | |
|     |            | Heater surface area: 25 × 25 mm | |
|     |            | Condenser: \( \phi 2 \times 50 \text{ mm} \) | |
|     |            | | |
|     |            | Microevaporator: 2000 μm; porosity: 75% | |
|     |            | Liquid line: ID4.95/OD6.35 × 300 | |
|     |            | Condenser: 130 × 130 × 25 | |
|     |            | CR = 38.5%, 51.3%; 64.1%, 64.1%, 76.9% | |
|     |            | Best fin’s geometric parameters: | |
|     |            | Oscillating behavior was found when \( Q \) was between 10 and 20 W. This oscillation occurred due to the fluctuation of heat loss from the evaporator to the CC and the subcooled liquid temperature. |
### Table 2. Cont.

| No. | References | Specifications of LHPs from the Literature | Objects and Results |
|-----|------------|------------------------------------------|---------------------|
| 7.  | M.A. Chernysheva—2014 [17]  
Keywords: LHP, supercomputer, cooling system, operating temperature, thermal resistance | Flat-oval evaporator (longitudinal replenishment evaporator) $80 \times 42 \times 7 \text{ mm}$, $A_{active} = 32 \times 42 \text{ mm}^2$  
12 Vapor grooves $\phi 1.8 \times 33 \text{ mm}$  
Copper wick (porosity 43%, $t_{por} = 27 \mu \text{ m}$)  
Vapor line: (1) $ID/OD = 305 \text{ mm}$, (2) $ID/OD = 305 \text{ mm}$  
Liquid line: $ID/OD = 810 \text{ mm}$  
Condenser: $ID/OD = 160 \text{ mm}$ | Objects: A cooling system with an LHP for a supercomputer’s thermal control is presented.  
Copper loop heat pipes with different vapor pipe IDs (4 and 3 mm each) were fabricated.  
The test was carried out with a heat load from 20 to 600 W during the cooling water’s temperature was changed from 20 to 80 °C.  
Results: LHP’s operating temperature varied slightly when the condenser cooling temperature changed in the range below 40 °C (called as variable conductance mode).  
It is more applicable to use copper-water LHPs when cooling temperatures of condenser is above 50 °C. |
| 8.  | Guohui Zhou—2016 [18]  
Keywords: Miniature LHP, ultrathin, thermal resistance, mobile electronics | Flat evaporator (thickness, $\delta = 1.2 \text{ mm}$), vapor line, liquid line, and condenser line ($\delta = 1 \text{ mm}$)  
Evaporator: $60 \times 23 \times 1.2 \text{ mm}$  
$A_{active} = 15 \times 9 \text{ mm}$  
Primary porous material (inside evaporator): sintered from 10 layers of 500 mesh copper wire mesh $(50 \times 21 \times 0.8 \text{ mm})$ (porosity: 65.2%)  
Secondary wick (in liquid line) sintered from 4 layers of 150 mesh copper wire mesh $(\delta = 0.43 \text{ mm})$  
Liquid line: $105 \text{ mm}$, vapor line: $105 \text{ mm}$  
Condenser: $125 \text{ mm}$ (Natural cooling)  
LHP’s inclination: horizontal, anti-, and assisted gravity.  
Water as working fluid | Objects: mLHP for mobile electronics.  
Results: Startup of LHP happened at 2 W with evaporator temperature 43.9 °C.  
When $Q$ is at 11 W, $R_{LHP} = 0.11 \text{ K/W}$  
There is no noticeably different performance with different orientations.  
For cooling mobile electronics: a tablet or smartphone, this mLHP achieves a promising thermal management solution. |
Table 2. Cont.

| No. | References | Specifications of LHPs from the Literature | Objects and Results |
|-----|------------|--------------------------------------------|---------------------|
| 9.  | Takeshi Shioga—2015 [19] | Keywords: Micro loop heat pipe Chemical-etching and diffusion bonding process for fabrication Evaporator: 20 × 17 × 0.6 mm Vapor line: (1) 5.6 × 0.4 and (2) 1 × 75 mm in length Liquid line 4 × 0.4 × 120 mm Condenser (1) 5.6 × 0.4 and (2) 1 × 110 mm in length Working fluid: water | Objects: This micro LHP was fabricated for mobile electronic devices. The effect of vapor and condenser thickness on μLHP performance was investigated. |
| 10. | Ji Li—2013 [20] | Keywords: LED cooling, LHP, parallel condensers, thermal resistance Evaporator: 30 × 30 × 15 (in mm). Gravity assisted LHP Connecting line: ID 5 mm Wick material is copper (porosity 50%, \( \tau_{pore} = 65 \) μm, \( K = 6 \times 10^{-11} \) m²) \( A_{Wick} = 25 \times 25 \) mm² Condenser size: 120 × 80 × 50 mm | Objects: The investigation of copper-water LHP using dual parallel condensers was conducted primarily for LED illumination applications with high-power. |

2. Experimental Setups and Data Reduction

2.1. Description of Test LHP

The elevation difference between the evaporator and condenser is 350 mm in this experiment. Cartridge heaters were used and inserted into the copper heating block at the evaporator base. The heat is rejected at the water-cooled condenser. The mass flow rate and inlet temperature of cooling water were around 7.5 × 10⁻³ kg s⁻¹ and 27.5 °C, respectively. The power supplied to the evaporator was adjusted and measured by a volt slider and a digital power meter, respectively. The thermal contact resistance between the heating block and the evaporator is minimized by thermal grease, which was filled in the interface and fixed with screws. Pressure transducers and four thermocouples were inserted directly into the path of the working fluid at different positions of the LHP, such as at evaporator outlet \( T_{evaporator} \), condenser inlet \( T_{ci} \), condenser outlet \( T_{co} \), and the inlet of compensation chamber \( T_{ci} \). These temperature transducers detect the temperature distribution inside the LHP. Therefore, the characteristics of circulation, as well as phase distribution, can be evaluated. Pressure transducer \( P_e \) was installed at the outlet of the evaporator. Figures 3 and 4 and Table 3 show the schematic diagram, photo of the real experimental setup, and the main specifications of the LHP.
Based on the operating orientation, the LHP’s performance was slightly changed.

**Keywords:** LED cooling, LHP, parallel condensers, thermal resistance

Evaporator: 30 × 30 × 15 (in mm).

**Gravity assisted LHP**

Connecting line: ID 5 mm

Wick material is copper (porosity 50%, \( r_{pore} = 65 \mu m \), \( K = 6 \times 10^{-11} m^2 \))

Heater = 25 × 25 mm²

Condenser size: 120 × 80 × 50 mm

**Objects:** The investigation of copper-water LHP using dual parallel condensers was conducted primarily for LED illumination applications with high-power.

**Results:** At \( Q = 300 \) W, the value of \( R \) is 0.4 °C/W; with \( T_{air} = 15 \) °C, \( Q = 0–100 \) W, \( T_{junction} < 75 \) °C.

At low heat loads, the condenser’s unpredictable nonuniform performance caused the unstable behavior of the LHP.

**2. Experimental Setups and Data Reduction**

**2.1. Description of Test LHP**

The elevation difference between the evaporator and condenser is 350 mm in this experiment. Cartridge heaters were used and inserted into the copper heating block at the evaporator base. The heat is rejected at the water-cooled condenser. The mass flow rate and inlet temperature of cooling water were around 7.5 × 10^{-3} kg s^{-1} and 27.5 °C, respectively. The power supplied to the evaporator was adjusted and measured by a volt slider and a digital power meter, respectively. The thermal contact resistance between the heating block and the evaporator is minimized by thermal grease, which was filled in the interface and fixed with screws. Pressure transducers and four thermocouples were inserted directly into the path of the working fluid at different positions of the LHP, such as at evaporator outlet \( T_eo \), condenser inlet \( T_{ci} \), condenser outlet \( T_{co} \), and the inlet of compensation chamber \( T_{cci} \). These temperature transducers detect the temperature distribution inside the LHP. Therefore, the characteristics of circulation, as well as phase distribution, can be evaluated. Pressure transducer \( P_e \) was installed at the outlet of the evaporator. Figures 3 and 4 and Table 3 show the schematic diagram, photo of the real experimental setup, and the main specifications of the LHP.

(a) (b)

**Figure 3.** Test LHP: (a) Schematic diagram of the LHP and temperature measurement points. (b) The locations of thermocouples on heating block and evaporator.

**Figure 4.** Photograph of the real experimental setup under gravity-assisted conditions.

| Evaporator Body Values |
|------------------------|
| Material               | Copper |
| Length (mm)            | 80     |
| Width (mm)             | 70     |
| Height (mm)            | 24.5   |
| Active area (mm²)      | 60 × 45|
| Fin geometry           |
| Cross area (mm²)       | 2 × 2  |
| Height (mm)            | 1.5    |
| Fin pitch (mm)         | 4      |
| Wick structure         |
| Material               | Stainless steel |
| Bulk volume (mm³)      | 50 × 41 × 5 |
| Material               | Ceramic  |
| Bulk volume (mm³)      | 50 × 41 × 5 |
| Vapor line             |
| OD/ID (mm)             | 6.35/4.35 |
| Length (mm)            | 800    |
| Condenser line         |
| OD/ID (mm)             | 6.35/4.35 |
| Length (mm)            | 600    |
| Liquid line            |
| OD/ID (mm)             | 3.2/1.7 |
| Length (mm)            | 1300   |
| Working fluid          | Water  |

**Figure 4.** Photograph of the real experimental setup under gravity-assisted conditions.
Table 3. Main parameters of the LHP.

| Evaporator Body | Values |
|-----------------|--------|
| Material        | Copper |
| Length (mm)     | 80     |
| Width (mm)      | 70     |
| Height (mm)     | 24.5   |
| Active area (mm\(^2\)) | 60 \times 45 |
| Fin geometry    |        |
| Cross area (mm\(^2\)) | 2 \times 2 |
| Height (mm)     | 1.5    |
| Fin pitch (mm)  | 4      |
| Wick structure  |        |
| Material        | Stainless steel |
| Bulk volume (mm\(^3\)) | 50 \times 41 \times 5 |
| Material        | Ceramic |
| Bulk volume (mm\(^3\)) | 50 \times 41 \times 5 |
| Vapor line      |        |
| OD/ID (mm)      | 6.35/4.35 |
| Length (mm)     | 800    |
| Condenser line  |        |
| OD/ID (mm)      | 6.35/4.35 |
| Length (mm)     | 600    |
| Liquid line     |        |
| OD/ID (mm)      | 3.2/1.7 |
| Length (mm)     | 1300   |
| Working fluid   |        |
| Water           |        |
| Amount (mL)     | 33     |

Thermocouples \(T_1, T_2, T_3\) were installed, as shown in Figure 3b, to consider the temperature gradient caused by the heat flux and the temperature on the top surface of heating block \(T_{s1}\), and the correct values of heating power and heat flux supplied to the evaporator were accessed. In the evaporator base, thermocouple \(T_4\) was inserted to estimate the temperature at the evaporator’s bottom surface; \(T_{s2}\) measures the temperature at the base of fin \(T_{bf}\). The heat released from the condenser, \(Q_c\), is obtained from the temperature difference of cooling water \(T_{\text{wa-o}} - T_{\text{wa-i}}\), and the mass flow rate of cooling water. All measured data were collected and recorded by the KEITHLEY 2701 data acquisition system. Moreover, the level of liquid inside the compensation chamber can be observed by the polycarbonate evaporator cover.

Table 4 describes the uncertainties of the mass flow meter and thermocouples (obtained from the calibration process in which a Pt100 thermometer (Chino Co. Model—R900-F25AT) was used as the standard source). Figures 5 and 6 demonstrate the structure of the evaporator and the geometry of the evaporator’s inner surface.
Table 4. Uncertainty values.

| Parameters               | Uncertainty |
|--------------------------|-------------|
| $T_1$, $T_2$, $T_3$      | 0.06 °C     |
| $T_4$                    | 0.07 °C     |
| $T_{co}$                 | 0.06 °C     |
| $T_{ci}$                 | 0.06 °C     |
| $T_{co}$, $T_{aci}$      | 0.1 °C      |
| $T_{wo-i}$               | 0.1 °C      |
| $T_{wa-o}$               | 0.06 °C     |
| Pressure transducer      | 1.5 kPa     |
| Mass flow meter          | 0.18% of reading |

Figure 5. Schematic design of the evaporator. 1: vapor collector; 2: compensation chamber; 3: polycarbonate lid; 4: O-ring; 5: charging pipe; 6: copper evaporator body; 7: wick; 8: vapor grooves; 9: copperplate; 10: vapor pipe.

Figure 6. The inner surface geometry of the evaporator (a) before and (b) after installing the wick.

Furthermore, the experimental uncertainty analysis of the computed heat transfer coefficients is given by the following equation (Equations (4)–(13)).

The experimental result $R$ is assumed to be calculated from a set of measurements using a data interpretation program presented by [21].

$$R = R(X_1, X_2, X_3, \ldots, X_N)$$

(4)

Each measurement uncertainty’s effect on the calculated result if only that one measurement were in error would be

$$\delta R_{X_i} = \frac{\partial R}{\partial X_i} \delta X_i$$

(5)
The partial derivative of $R$ with respect to $X_i$ is the sensitivity coefficient for result $R$ with respect to measurement $X_i$. The single terms are combined by a root-sum-square method when respective independent variables are used in function $R$.

$$\delta R = \left\{ \sum_{i=1}^{N} \left( \frac{\partial R}{\partial X_i} \delta X_i \right)^2 \right\}^{\frac{1}{2}}$$

(6)

Then, the parameter $\Delta T_{12}$ can be defined as,

$$\Delta T_{12} = T_1 - T_2$$

(7)

The uncertainty for $\Delta T_{12}$ is shown in Equation (8).

$$\delta(\Delta T_{12}) = \left( \delta T_1^2 + \delta T_2^2 \right)^{\frac{1}{2}}$$

(8)

where $\Delta T_{12}$ is the temperature difference between $T_1$ and $T_2$ at the heating block.

The parameter $\Delta T_{23}$ can be defined as,

$$\Delta T_{23} = T_2 - T_3$$

(9)

The uncertainty for $\Delta T_{23}$ is shown in Equation (10).

$$\delta(\Delta T_{23}) = \left( \delta T_2^2 + \delta T_3^2 \right)^{\frac{1}{2}}$$

(10)

where $\Delta T_{23}$ is the temperature difference between $T_2$ and $T_3$ at the heating block.

The parameter $\Delta T_{13}$ can be defined as,

$$\Delta T_{13} = T_1 - T_3$$

(11)

The uncertainty for $\Delta T_{13}$ is shown in Equation (12).

$$\delta(\Delta T_{13}) = \left( \delta T_1^2 + \delta T_3^2 \right)^{\frac{1}{2}}$$

(12)

where $\Delta T_{13}$ is the temperature difference between $T_1$ and $T_3$ at the heating block.

The parameter $q$ can be defined as,

$$q = k \frac{\Delta T_{12}}{\delta_1}$$

(13)

The uncertainty for $q$ is shown in Equation (14).

$$\delta q = \frac{k}{\delta_1} \delta(\Delta T_{12})$$

(14)

where $q$ is the heat flux; $k$ is the thermal conductivity of copper.

$\delta_1$ is the distance between the heating block’s thermocouples.

The parameter $T_{bf}$ can be defined as,

$$T_{bf} = T_4 - \frac{q \delta_2}{k}$$

(15)
The uncertainty for $T_{bf}$ is shown in Equation (16).

$$\delta(T_{bf}) = \left((\delta T_4)^2 + \left(\frac{\delta q}{k}\right)^2\right)^{\frac{1}{2}}$$

(16)

where

- $T_{bf}$ is the evaporator fin' base temperature.
- $T_4$ is the evaporator’s base temperature.

The parameter $h_e$ can be defined as,

$$h_e = \frac{q}{T_{bf} - T_{eo}}$$

(17)

The uncertainty for $h_e$ is shown in Equation (18).

$$\delta(h_e) = \left(\left(\frac{\delta q}{T_{bf} - T_{eo}}\right)^2 + \left(\frac{q\delta T_{eo}}{(T_{bf} - T_{eo})^2}\right)^2 + \left(\frac{q\delta T_{bf}}{(T_{bf} - T_{eo})^2}\right)^2\right)^{\frac{1}{2}}$$

(18)

where

- $h_e$ is the heat transfer coefficient of the evaporator.
- $T_{eo}$ is the evaporator outlet temperature.

The parameter $h_{esat}$ can be defined as,

$$h_{esat} = \frac{q}{T_{bf} - T_{esat}}$$

(19)

The uncertainty for $h_{esat}$ is shown in Equation (20).

$$\delta(h_{esat}) = \left(\left(\frac{\delta q}{T_{bf} - T_{esat}}\right)^2 + \left(\frac{q\delta T_{esat}}{(T_{bf} - T_{esat})^2}\right)^2 + \left(\frac{q\delta T_{bf}}{(T_{bf} - T_{esat})^2}\right)^2\right)^{\frac{1}{2}}$$

(20)

where

- $h_{esat}$ is the heat transfer coefficient of the evaporator, calculated from the saturation temperature.
- $T_{esat}$ is the saturation temperature accessed from vapor pressure.

2.2. Description of the Wick’s Thermal Conductivity Measuring System

The wick materials’ thermal conductivity was measured by the transient hot-wire method. Figure 7 shows a schematic diagram of the thermal conductivity measuring device and sensor unit. It consists of a power supply device, a measuring device, a sensor unit, and a recording device (Personal Computer, PC). At the time of measurement, a constant current is applied from the power supply to the sensor. In addition, the voltage drop of the sensor unit is measured by a measuring device at regular intervals and recorded by a recording device.
The basic equation for thermal conductivity measurement is expressed by the following equations (Equations (21)–(24)).

The thermal conductivity of the wicks (Equation (21)) [22],

\[ \lambda_2 = \frac{q^*}{2\pi} \frac{d \ln t}{d \Delta T} - \lambda_1 \]  

(21)

where

\( \lambda_2 \) is the thermal conductivity of the wick sample.
\( \lambda_1 \) is the thermal conductivity of the insulator.
\( q^* \) is the amount of heating per wire length.
\( t \) is the time taken.
\( \Delta T \) is the temperature change from \( t = 0 \) s.

Resistance of thin wire at a certain temperature,

\[ R = R_o(1 + \alpha \Delta T) \]  

(22)

where

\( R_o \) is the initial electrical resistance.
\( \alpha \) is the temperature coefficient of resistance.

From Equation (22) and Ohm’s Law,

\[ \Delta T = \frac{1}{\alpha} \left( \frac{V(t)}{IR_o} - 1 \right) \]  

(23)

where

\( V \) is the voltage drop on the thin wire.
\( I \) is the current.

By substituting \( \Delta T \) into Equation (21),

\[ \frac{d V(t)}{d \ln t} = \frac{\alpha I^2 R_o^2}{2\pi(\lambda_1 + \lambda_2)} = \text{Constant} \]  

(24)

Then, the wick materials’ thermal conductivity can be obtained from the slope of the relationship between the voltage drop of the thin wire and the logarithm of the heating time. The measured thermal conductivity of the tested wicks and the comparison with literature data [7,8] are presented in Table 5.
Table 5. Thermal conductivity of wicks.

| Wick                | \( \lambda_2 \) [W (m K\(^{-1}\)] | Wick                | \( \lambda_2 \) [W (m K\(^{-1}\)] |
|---------------------|-------------------------------|---------------------|-------------------------------|
| Stainless Steel     | 6.4                           | SP-Sintered         | 11.87                         |
| Ceramic             | 3.5                           | Ceramic             | 4                             |

2.3. Description of Condenser

In this study, the condenser is a double-pipe heat exchanger with a counter flow arrangement. Cooling water from the constant temperature pump flows in the annular area while the vapor condenses inside the copper tube. The condenser structure is described in Figure 8 and Table 6.

Figure 8. Photograph of the condenser.

Table 6. Specification of the condenser.

| Parameters | Inner Tube                      | Outer Tube                     |
|------------|---------------------------------|---------------------------------|
| Material   | Smooth Copper tube              | Poly-carbonated resin           |
| Length, mm | 600 mm                          | 600 mm                          |
| OD/ID, mm  | 6.35/4.35                       | 13/9                            |

Furthermore, the same copper smooth tube is used for the vapor pipe and liquid pipe of the LHP, whose OD/ID is 6.35/4.35 and 3.2/1.5 mm, respectively.

2.4. Data Reduction

From Figure 3b, the values of heat flux \( q \), heat transfer coefficient \( h_e \), and thermal resistances \( R_e, R_c, \) and \( R_{ct} \) can be estimated by the following equations (Equations (25)–(36)): \( A = 27 \text{ cm}^2, \delta_1 = 5 \text{ mm}, \delta_2 = 2.5 \text{ mm} \).

Heat flux flowing from the heating block to the evaporator’s active area is calculated by Fourier’s Law of heat conduction as

\[
q = k \frac{T_1 - T_2}{\delta_1} = k \frac{T_2 - T_3}{\delta_1} = k \frac{T_1 - T_3}{2\delta_1}\]

(25)

where
- \( q \) is the heat transfer rate per unit heating block’s surface area.
- \( k \) is the thermal conductivity of the copper heating block.
- \( T_1 \) to \( T_3 \) is the heater temperature.
- \( \delta_1 \) is the distance between the thermocouples inside the heating block.
Then, the mean of \( q \) can be expressed by Equation (26).

\[
q = \frac{1}{3} \left( k \frac{\Delta T_{12}}{\delta_1} + k \frac{\Delta T_{23}}{\delta_1} + k \frac{\Delta T_{13}}{2\delta_1} \right)
\]

(26)
\[ Q = q * A \]  \tag{27}

Additionally, the temperature of the heating block’s top surface, \( T_{s1} \), and the evaporator bottom’s temperature, \( T_{s2} \), can be determined by Equations \( 28 \) and \( 29 \).

\[
T_{s1} = \frac{1}{3} \left[ \left( T_1 - \frac{3(q \delta_1)}{k} \right) + \left( T_2 - \frac{2(q \delta_1)}{k} \right) + \left( T_3 - \frac{q \delta_1}{k} \right) \right] \tag{28}
\]

\[
T_{s2} = T_4 + \frac{q \delta_2}{k} \tag{29}
\]

\[
R_e = \frac{T_{s2} - T_{eo}}{Q} \tag{30}
\]

\[
R_c = \frac{T_{ci} - T_{wa-i}}{Q_c} \tag{31}
\]

\[
Q_c = m_{wa} C_p (T_{wa-o} - T_{wa-i}) \tag{32}
\]

\[
R_{cl} = \frac{T_{s1} - T_{s2}}{Q} \tag{33}
\]

\[
h_e = \frac{q}{T_{bf} - T_{eo}} \tag{34}
\]

\[
h_{esat} = \frac{q}{T_{bf} - T_{esat}} \tag{35}
\]

\[
T_{bf} = T_4 - \frac{q \delta_2}{k} \tag{36}
\]

3. Results and Discussion

Figure 9 displays the values of temperature at various locations in the experiment. The results show that the stainless-steel wick exhibits higher heat flux when compared to the ceramic wick. During the heat load range of 50 to 520 W in Figure 9a and 53 to 155 W in Figure 9b, the values of \( T_{eo} \) and \( T_{ci} \) are almost similar, and \( T_{co} \) and \( T_{cci} \) are nearly equal. We affirm that the working fluid circulates stably inside the loop heat pipe. However, in the LHP with a ceramic wick, some fluctuations were found in the range of heat power, from 53 to 80 W, where the values of \( T_{cci} \) were higher than \( T_{co} \). This phenomenon happened because there was the intermittent appearance of a vapor–liquid interface near the position of thermocouple \( T_{cci} \) in the liquid line. In the experiment of the LHP with a stainless steel wick, the supplying liquid for the compensation chamber was more stable than the LHP with the ceramic wick. Moreover, in this experimental research, temperature \( T_{s1} \) on the heating block’s top surface can be viewed as the electronics temperature, which is normally recommended to be lower than 85 °C for reliable and effective operation [23]. Therefore, this present LHP, with stainless steel and ceramic wicks, can satisfy the recommendation until the heat load reaches 350 and 54 W, respectively. The measured values and frequency of pore distribution by the mercury injection method [24] are given in Table 7 and Figure 10.
Figure 9. LHP’s temperatures distribution at different heat loads when installed with (a) a stainless-steel wick and (b) a ceramic wick.

Table 7. Test results of pore distribution measurement by the mercury injection method.

| Wick Materials     | Ceramic | Stainless Steel |
|--------------------|---------|-----------------|
| Pore volume in total (cm³·g⁻¹) | 0.20    | 0.07            |
| Central pore diameter (µm)    | 1.29    | 16.40           |
| Total pore specific surface area (m²·g⁻¹) | 0.66    | 0.02            |
| Average pore size (µm)       | 1.18    | 11.85           |
| Bulk density (g·cm⁻³)        | 2.01    | 4.34            |
| Porosity (%)                | 39.3    | 31.5            |

Figure 10. Pore frequency distribution graphs (a) Ceramic wick (b) Stainless steel wick. Reproduced from [25], Saga Ceramics Research Laboratory: 2019.

3.1. Performance Evaluation

The thermal performance can be evaluated in terms of heat flux $q$, heat transfer coefficient $h$, and thermal resistances, which are evaporator $R_e$, condenser $R_c$, and contact surface $R_{ct}$, as shown in Figures 11 and 12. In this study, the evaporator heat transfer coefficient was calculated from vapor temperature $T_{vap}$ at the outlet of the evaporator and saturation temperature $T_{sat}$, accessed from the vapor pressure measured at the outlet of the evaporator.
In both experiments of stainless steel and ceramic LHPs, the evaporator heat transfer coefficient obtained from $T_{eo}$ was higher than the values calculated from saturation temperature $T_{esat}$. Observing the difference in the results, the vapor might be superheated before leaving the evaporator. This superheating process could happen when vapor flows in the crossing grooves, and the heat of this process comes from the surrounding area of the fins and the grooves' surface. The possible operating heat flux for the stainless-steel wick was higher than the ceramic wick. Comparing the evaporator heat transfer coefficient of the LHPs, the evaporator operating with the ceramic wick had the higher evaporator heat transfer coefficient when the heat transfer coefficient, $h_e$, linearly increased with the heat flux in the LHP with the stainless steel wick. However, the ceramic LHP did not operate better than the stainless steel one. The significant mass flow rate of vapor requires large pore sizes in the ceramic wick to reduce resistance. Otherwise, the performance of the
condenser decreases. The numerical values obtained from the experimental uncertainty analysis of the computed heat transfer coefficients are presented in Tables 8 and 9.

Table 8. Uncertainty values of the computed heat transfer coefficients for LHPs with stainless steel wicks.

| $Q$ (W) | $q$ (W/m²) | $\delta(q)$ | $h_e$ (W/m² K) | $\delta(h_e)$ | $h_{esat}$ (W/m² K) | $\delta(h_{esat})$ |
|--------|-------------|--------------|--------------|--------------|----------------|-----------------|
| 50.22  | 18,598.44   | 30.42        | 4592.67      | 30.46        | 1345.93        | 62.64           |
| 100.29 | 37,299.05   | 15.17        | 5588.10      | 15.25        | 2150.30        | 37.71           |
| 147.91 | 54,451.96   | 10.39        | 8458.43      | 10.39        | 3984.10        | 32.14           |
| 188.17 | 69,832.60   | 8.10         | 8323.83      | 8.12         | 4804.16        | 25.30           |
| 237.16 | 87,723.91   | 6.45         | 8368.00      | 6.43         | 5676.28        | 17.73           |
| 290.03 | 107,645.92  | 5.26         | 8427.34      | 5.14         | 6428.54        | 12.52           |
| 350.72 | 129,896.27  | 4.35         | 9388.90      | 4.31         | 7633.87        | 9.80            |
| 401.67 | 148,767.07  | 3.80         | 10,023.39    | 3.77         | 8228.43        | 7.74            |
| 445.87 | 165,137.27  | 3.43         | 10,651.51    | 3.46         | 9177.51        | 6.74            |
| 503.47 | 186,465.90  | 3.03         | 11,785.25    | 3.08         | 10,499.12      | 5.67            |
| 520.70 | 192,826.71  | 2.93         | 11,900.59    | 2.99         | 10,742.33      | 5.26            |

Table 9. Uncertainty values of the computed heat transfer coefficients for LHPs with ceramic wicks.

| $Q$ (W) | $q$ (W/m²) | $\delta(q)$ | $h_e$ (W/m² K) | $\delta(h_e)$ | $h_{esat}$ (W/m² K) | $\delta(h_{esat})$ |
|--------|-------------|--------------|--------------|--------------|----------------|-----------------|
| 53.41  | 19,782.71   | 28.59        | 13,195.93    | 26.67        | 13,932.62      | 48.68           |
| 79.59  | 29,477.46   | 19.19        | 11,352.25    | 16.61        | 9709.18        | 21.12           |
| 105.72 | 39,154.18   | 14.45        | 10,783.11    | 14.27        | 8046.37        | 16.92           |
| 132.38 | 49,028.99   | 11.54        | 11,805.62    | 11.73        | 8359.43        | 14.03           |
| 155.45 | 57,574.38   | 9.83         | 10,737.34    | 9.87         | 7810.08        | 11.71           |

Performance of Evaporator and Condenser

Figure 12a,b displays the evaporator and condenser thermal resistances, $R_e$ and $R_c$, of the stainless steel and ceramics LHPs. In the LHP with the stainless steel wick, the values of $R_e$ became smaller with the increase in heating power. However, thermal resistance $R_e$ for the ceramic one gradually increased in the same operating conditions, although the performance of the evaporator with the ceramic wick was more effective than the one with the stainless steel wick. For the thermal resistances of the condenser at the stainless steel LHP, it was reduced to the minimum value of 0.09 K.W⁻¹, then raised up slightly, as shown in Figure 12a. The higher the heat power supplied to the evaporator, the less liquid exists inside the compensation chamber. As a result, with more liquid present in the condenser with the increasing heat load, the performance of the condenser was slightly reduced.

For the LHP with the ceramic wick, $R_c$’s values were significantly high, i.e., 1.8 K.W⁻¹ at 53 W, due to the lower performance of the condenser. Additionally, particle differences between the stainless steel wick and ceramic wick are apparent in Figure 11a,b; the interfaces between the particles in Figure 11b are not as straightforward as those in Figure 11a. In the loop heat pipe with the stainless-steel wick, the working fluid was smoothly circulating inside the LHP because the flow resistance through the stainless-steel wick was lower than that of the ceramic wick. Therefore, the heat supplied made the working fluid evaporate, and the heat leak through the stainless-steel wick became small. In the ceramic wick’s case, the working fluid was not able to circulate correctly inside the LHP due to the high hydraulic resistance in the ceramic wick. As a result, heat flow, called a heat leak, became dominant in the evaporator with the ceramic wick.
4. Conclusions

The influence of two different wicks was tested in this study and used to describe the LHPs’ performance with a flat rectangular evaporator. The experimental results demonstrate that the LHP with the stainless-steel wick has better cooling performance than the LHP with the ceramic wick. The heater surface temperature of the LHP with the stainless steel wick increased from 40 to 105 °C in the range of a heat load from 50 to 520 W. In the ceramic wick LHP, this surface temperature reached 106 °C at a heat load of 155 W. Under gravity-assisted conditions, the LHP with the stainless steel wick could keep the temperature on the heater surface at 85 °C for a heat load of 350 W. However, the LHP with the ceramic wick could operate only at lower than 118 W for the same LHP, and the heater’s surface temperature approached 85 °C when the LHP was conducted at 54 W (20 kW·m⁻²). The ceramic wick cannot handle a high heat flux because of the conflict between vapor release and liquid suction. Large pore sizes are required to reduce resistance for the significant mass flow rate of vapor. It has been explained by Meléndez and Reyes [26].

\[ m_v = \frac{\pi}{128} \left( \frac{\rho_v \sigma}{\mu_v} \right) \left( \frac{\epsilon d_e^3}{\delta} \right) \]  

(37)

where \( m_v \) is the vapor mass flow rate; \( \rho_v, \sigma, \mu_v, \) and \( \epsilon \) are vapor density, surface tension, viscosity, and porosity, respectively; \( \delta \) is the wick thickness, and \( d_e \) is the effective pore diameter.

On the other hand, large capillary pressure is achieved by tiny pores for liquid suction, according to the Laplace-Young equation.

\[ \Delta P = \frac{4\sigma \cos \alpha}{d_e}, \ \alpha \ is \ the \ contact \ angle \]  

(38)

As a result, the vapor release and liquid suction of the ceramic wick require different pore sizes for better heat performance in LHPs.

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Nomenclature

\( T_{13} \) | temperature at heater surface (°C)
---|---
\( ID,OD \) | pipe inner,outer diameter (mm)
\( k \) | thermal conductivity of copper heating block [W-(m K)\(^{-1}\)]
\( \dot{q} \) | heat flux (kW m\(^{-2}\))
\( Q \) | heat load (W)
\( R_e \) | thermal resistance of evaporator (K W\(^{-1}\))
\( R_c \) | thermal resistance of condenser (K W\(^{-1}\))
\( R_d \) | thermal contact resistance (K W\(^{-1}\))
\( T_1 \) to \( T_3 \) | heater temperature (°C)
\( T_4 \) | evaporator base temperature (°C)
\( Q_c \) | heat released from condenser (W)
\( m_{sat} \) | mass flow rate of cooling water (kg s\(^{-1}\))
\( h_{e} \) | transfer coefficient of evaporator (kW m\(^{-2}\) K\(^{-1}\))
\( \Delta T \) | temperature change from t = 0 (°C)
\( \dot{q}^* \) | amount of heating per wire length (W m\(^{-1}\))
\( R_e \) | initial electrical resistance (Ω)
\( V \) | voltage drop on thin wire (V)
\( A \) | area of heating block surface (m\(^2\))
\( I \) | electric current (I)
\( \eta \) | heat transfer rate per unit heating block’s surface area (kW m\(^{-2}\))

\( T_{T1} \) | fin base temperature (°C)
\( T_{ci} \) | condenser inlet temperature (°C)
\( T_{co} \) | condenser outlet temperature (°C)
\( T_{ci} \) | compensation chamber inlet temperature (°C)
\( T_{co} \) | evaporator outlet temperature (°C)
\( T_{co} \) | evaporator bottom surface temperature (°C)
\( T_{cool} \) | cooling water temperature at inlet position (°C)
\( T_{cool} \) | cooling water temperature at outlet position (°C)
\( \delta_1 \) | distance between the thermocouples inside heating block (m)
\( \delta_2 \) | distance between the thermocouple \( T_4 \) and the bottom surface of evaporator (m)
\( C_p \) | specific heat of cooling water [J (kg K\(^{-1}\)]
\( T_{sat} \) | saturation temperature accessed from vapor pressure
\( T_{heat} \) | evaporator heat transfer coefficient calculated from saturation temperature (kW m\(^{-2}\) K\(^{-1}\))
\( t \) | time (s)
\( T_{eq} \) | initial temperature (°C)
\( a \) | temperature coefficient of resistance (°C\(^{-1}\))
\( L \) | thin wire length (mm)
\( \lambda_2 \) | thermal conductivity of wick sample [W (m K\(^{-1}\)]
\( \lambda_1 \) | thermal conductivity of insulator [W (m K\(^{-1}\)]

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