An accurate calorimeter-based method for the thermal characterization of heat pipes

Joseph P. Mooney, Jeff Punch, Nick Jeffers, Vanessa Egan

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ABSTRACT

This study presents a method that can be used to accurately determine the thermal performance of a cylindrical heat pipe. In the method, the heat pipe is placed between two stainless steel 304 cylindrical blocks, configured as radial calorimeters that achieve thermal contact with the evaporator and condenser sections of the pipe. A flexible isothermal electrical heater mat surrounds the evaporator block, and a liquid-cooled copper pipe wrapped around the condenser block is used to remove heat. High precision thermistors (±0.01 K) positioned at fixed radial locations within the calorimeters are used to measure the heat supplied to the evaporator and the heat extracted from the condenser. One-dimensional radial conduction is assumed to occur within each calorimeter, and this enables the quantification of heat flows from the temperature readings. This assumption is verified by a steady-state analysis of the radial, axial and circumferential temperature differences within the evaporator calorimeter, based on data recorded for the lowest and highest heat inputs. Furthermore, a numerical model is used to confirm that end effects have a negligible influence on radial conduction within each calorimeter. This study concludes that the most commonly used characterization techniques for heat pipes can greatly overestimate thermal performance (15–32% for input powers of 7.5–25 W respectively) due to inaccurate quantification of heat flows into the evaporator and from the condenser. The calorimetric technique reported here achieves uncertainties in thermal resistance of <7.5% for low thermal loads (<12.5 W) and <6% for higher loads (>12.5 W). Moreover, the method achieves a significant improvement in the experimental thermal efficiency, with values of >75% recorded for all heat inputs in this study. The use of radial calorimeters in the current study obviates the requirement for calculating the losses from the heater to ambient, hence achieving low uncertainties in thermal resistance and effective thermal conductivity for a range of heat inputs.

1. Introduction

Heat pipes are passive heat transfer devices that utilise the latent heat of vaporisation of a working fluid in order to transport heat from a high temperature source to a colder sink [1]. A conventional heat pipe consists of a hollow metal capsule lined with an internal porous structure, known as a wick, which is saturated with a working liquid. The remaining internal volume of the heat pipe is occupied by the vapor of the liquid. Due to the stable coexistence of liquid and vapor, the internal pressure is equal to the vapor pressure that corresponds to saturation conditions. This ensures effective use of the latent heat of the liquid for quick evaporation and heat transfer. When heat is applied to one end of the heat pipe, the liquid evaporates from the wick into the hollow vapor channel. A temperature driven pressure between the evaporator and condenser ends of the heat pipe causes the vapor to flow in the direction of the heat sink [2]. At the condenser end, heat is removed, and the vapor condenses into the wick, normally saturating it. Due to the vacated pore space at the evaporator and the saturated pores in the condenser, capillary pressures force the liquid back to the evaporator.

Heat pipes feature effective thermal conductivities of order 10^2–10^4 times superior to those of good heat-conducting materials (copper, silver, etc.) [3]. Heat pipes come in various types, depending on their application, including: concentric, micro heat pipes, vapour chambers, flat heat pipes, and rotating heat pipes, amongst others. They are used in a multitude of applications such as: electronics cooling, energy conversion systems, nuclear and isotope reactor cooling, high-performance space infrastructure, and cooling devices for the leading edges of nose
cones of re-entry vehicles [1,4,5]. One of the more commonly used types of heat pipe is a concentric tube, sintered copper powder heat pipe with water as a working fluid. This heat pipe is known as a homogeneous wicked heat pipe because the wick comprises a single material with an axially-invariant cross sectional structure [5].

Thermal resistance ($R_{th}$), effective thermal conductivity ($k_{eff}$), and average temperatures along the axis of the pipe ($T_x$) are the most commonly used thermal characteristics for heat pipes [1,6–9]. Effective thermal conductivity is inversely proportional to thermal resistance, hence the better performing heat pipes feature superior effective thermal conductivity values. In the literature, there is an abundance of experimental and numerical analyses on heat pipes of varying geometries, lengths and materials with reported values for effective conductivity and thermal resistance [10].

Across a wide range of applications, greater thermal performance is demanded of heat pipes in order to meet increasing stringent design requirements. Fig. 1 illustrates a frequently-used technique for experimentally characterizing the performance of a heat pipe. A heat source is thermally attached to one end of the heat pipe (the evaporator end), and the other end of the pipe thermally connects to a heat sink (the condenser). Measures are implemented to minimize the thermal contact resistance between the pipe and the heat source and sink. Fig. 1 also illustrates some of the most common means of supplying heat to the evaporator section which include: electrical heater mats, a heating block with cartridge heaters, a copper coil, or direct contact cartridge heaters. Heat is typically supplied to the pipe via the power inputted by the power supply units (i.e. Joule heating), and Ohm's law ($Q_s = P = IV$, where $Q_s \geq Q_{in}$) is typically used to quantify this heat [1,7–9,11–15]. At the condenser end, a range of techniques can be used to remove heat: a direct water jacket, a liquid cooled block, forced air-cooling, or natural convection [1,7–9,11–15]. A popular method of heat removal is a water jacket, as shown in Fig. 1 [1,7–9,12–15]. In this case, the heat rejected is calculated using the temperature difference of the coolant entering and leaving the cooling jacket (i.e. a thermal energy balance, $Q_{out} = \dot{m}c_p\Delta T$). The heat supplied ($Q_s$) is often compared to the heat rejected ($Q_{out}$) in order establish the experimental quality or amount of heat lost from the experiment to ambient, i.e experimental thermal efficiency ($\eta_{th} = Q_{out}/Q_s$) [7–9,14–16]. However, most of these studies [8,9,14,15] use thermal efficiency as a measure of the performance of a heat pipe and assume that all the environmental losses are from the heat pipe to ambient. This assumption neglects any losses from the experimental apparatus, and as shown in Table 1, the resulting $Q_{in}/Q_{out}$ values imply that heat pipes are highly inefficient heat transfer devices which Conventionally is not the case. Table 1 also details the methods used for heat pipe characterization from recent literature and the range of measured $R_{th}$ values for the heat pipe studies. It can be seen that for all referenced studies, heat pipes tend to have low thermal resistances of ~2.0–0.01 K/W. However, inconsistencies arise when evaluating $R_{th}$

### Nomenclature

| Roman Letters | Greek Symbols | Subscripts |
|---------------|--------------|------------|
| $A$           | $\Delta$     | $\text{average}$ |
| $c_p$         | $\eta$       | $\text{Condenser}$ |
| $d$           | $\theta$     | $\text{Evaporator}$ |
| $I$           | $U$          | $\text{Inner}$ |
| $k$           | $V$          | $\text{Outer}$ |
| $L$           | $x$          | $\text{supplied}$ |
| $P$           | $T$          | $\text{Condenser}$ |
| $Q$           | $U$          | $\text{Evaporator}$ |
| $r$           | $V$          | $\text{inner}$ |
| $R_{th}$      | $T$          | $\text{outer}$ |
| $T$           | $x$          | $\text{Thermal}$ |
| $U$           | $\text{Temperature, } {^\circ}C$ | $\text{supply}$ |

### Greek Symbols

- $\Delta$: Uncertainty
- $\eta$: Efficiency, %
- $\theta$: Angle (°)

### Subscripts

- average: Average
- $c$: Condenser
- $e$: Evaporator
- $i$: Inner
- $in$: in
- $o$: Outer
- $out$: out
- $th$: Thermal
- $s$: supplied

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**Fig. 1.** Conventional experimental set up for heat pipe characterization illustrating the methods of supplying and removing heat [17].
and $\eta_{th}$ which can be explained with reference to the equation which defines the thermal resistance of a heat pipe:

$$R_{th} = \frac{T_e - T_c}{Q_{in}}$$  \hspace{1cm} (1)$$

where, $T_e$ and $T_c$ are the surface temperatures of the evaporator and condenser respectively, and $Q_{in}$ is the heat flow into the evaporator. By definition, thermal resistance is dependent on $Q_{in}$ [6]. For the studies presented in Table 1, $Q_{in}$ is assumed to be $Q_S$ from the power supply. However, the values of $\eta_{th}$ reveal that only a fraction (~20–83%) of $Q_{in}$ is being rejected by the condenser. This implies that although low thermal resistance values were recorded for the pipes, significant amounts of heat were lost from the pipe to ambient. It is evident that these studies generally did not account for the heat losses from the experimental apparatus to ambient. As a consequence, the reported thermal resistances can be considered lower-bound values, and the $\eta_{th}$ values (if given) are unreliable. Therefore, it is impossible to accurately evaluate $R_{th}$ on the basis of $Q_{in}$ if there are significant uncertainties in the heat flowing through the pipe or lost from the apparatus.

Two studies accounted for the losses, $Q_{loss}$, from the evaporator to ambient [18,19]. In both studies, the losses are used to adjust the heater power to obtain the heat input to the evaporator (i.e. $Q_{in} = Q_S - Q_{loss}$), and any losses between the evaporator and condenser end of the heat pipe are calculated using the thermal energy balance of the condenser. These studies achieved improved uncertainty in the measured thermal characteristics of a heat pipe. The uncertainties associated with $R_{th}$ and $k_{th}$ of these methods were up to ±20% for lower thermal loads (below ~10 W) and ±10% for loads between 10–30 W. Precise knowledge of the experimental environment is required, however, and extra computational effort is involved. Other methods can be used to improve the thermal characterization of a test sample by calculating $Q_{loss}$ from the experimental apparatus. These include, but are not limited to, thermal guarding [20,21], the insertion of thermocouples into the insulation to measure $Q_{loss}$, or the use of the thermal energy balance for evaporator and condenser water jackets [22]. These techniques are all limited by the uncertainties associated with the quantification of heat loss, specifically: the accuracy and spatial location of the thermocouples used to quantify $Q_{loss}$; the thermal conductivity of the insulation; ambient conditions; the geometric and thermal properties of the heaters; and the losses due to Joule heating.

Similarly, there is also uncertainty when considering the thermal energy balance for the direct water jackets. As previously stated (see Table 1), water is a commonly used condenser coolant for these experiments [1,7–9,11–16,18,19]. For water, its high specific heat capacity ($C_p \approx 4,180 \text{ J/kg.K}$) results in low temperature differences between the inlet and outlet flows of the cooling jacket, since most of the experiments have investigated power inputs of 20–100 W [1,7,9,11–16,18,19]. Small temperature differences in the condenser fluid require precision thermometry in order to reduce the experimental uncertainties in $Q_{out}$, which were ~26–73.2% for one study [7]. These uncertainties could be improved if very small mass flow rates were used, but this would result in increased uncertainties in flow rate. Additionally, at low flow rates, the flow around the pipe can feature vortex shedding that can induce fluctuations in the rate at which the pipe is cooled [23,24], and transient variations in the surface temperature of heat pipe.

In this study, a novel method is presented to accurately characterize the thermal performance of a heat pipe. The method uses radial calorimeters to quantify the heat flow into the evaporator and the heat removed from the condenser. The use of radial calorimeters obviates the requirement for calculating the losses to ambient from the heater and

### Table 1

| $Q_{in}$ (P - LV) Heating Technique | $Q_{out}$ Cooling Technique | Thermal Resistance $R_{th}$ (K/W) | $\eta_{th}$ ($Q_{loss}/Q_{in}$) % |
|-----------------------------------|----------------------------|-----------------------------------|----------------------------------|
| Copper block with six electrical cartridge heaters ($Q_{in} = 13–100 \text{ W}$) | $mc_p(T_s - T_{in})$ Chilled water jacket with flowmeter (125 ml/m @ 25 $^\circ$C) | 0.3–0.15 | 45–79% |
| Electrical heater ($Q_{in} = 200–400 \text{ W}$) | $mc_p(T_s - T_{in})$ Cooling jacket | ~0.1 | 83–86% |
| Block heating units ($Q_{in} = 20–120 \text{ W}$) | $mc_p(T_s - T_{in})$ Refrigerated bath circulator cooling box | 2–0.3 | 20–40% |
| Electrical heater | $mc_p(T_s - T_{in})$ Cooling jacket Water | ~ | 40–83% |
| Circumferential ceramic heater ($Q_{in} = 2.5–38 \text{ W}$) | $mc_p(T_s - T_{in})$ Cooling jacket Water | 1.3–0.38 | 40–70% |
| Circumferential ceramic heater ($Q_{in} = 10–150 \text{ W}$) | $mc_p(T_s - T_{in})$ Cooling jacket Water | 1.4–0.4 | 39–74% |
| Flexible resistance heater ($Q_{in} = 5–15 \text{ W}$) | Cooling block cooled by a water cooling bath | 1.6–0.18 | ~ |
| Two copper blocks heated by a DC power supply ($Q_{in} = 10–60 \text{ W}$) | | | |
| Electrical resistance heater powered by a DC power supply ($Q_{in} = 30–70 \text{ W}$) | $mc_p(T_s - T_{in})$ Constant temperature thermal bath and a cooling chamber (40 $^\circ$C) | 0.215–0.17 | ~ |
| Heating block ($Q_{in} = 10–20 \text{ W}$) | Isothermal water cooled block | 0.24–0.01 | ~ |
| Aluminum block with electric band heaters $Q_{in} = Q_S - Q_{loss}$ ($20–180 \text{ W}$) | Acrylic water jacket $mc_p(T_s - T_{in})$ | ~0.6–0.17 | ~ |
| Brass annular block with electrical band heater $Q_{in} = Q_S - Q_{loss}$ ($10–300 \text{ W}$) | Acrylic water jacket $mc_p(T_s - T_{in})$ | ~2–0.3 | ~ |
the use of the thermal energy balance for a cooling water jacket. The method achieves low uncertainties in $R_h$ (≤7.5%), $K_{eff}$ (≤8%) and $\eta_h$ (≤7%) for all thermal inputs. Section 2 details the operation of the calorimeters which are the key elements of the method. In Section 3, an experimental apparatus that implements the method is described, with data reduction and results presented in Sections 4 and 5 respectively.

2. Calorimetry

Methods to measure the thermal conductivity of a homogeneous material of unknown thermal properties, such as solid flat plate homogeneous materials, are well established in the literature. The well-known implementation of the ASTM D5470 standard [25], depicted in Fig. 2(a), is a popular approach for characterizing such materials. Although ASTM D5470 is conventionally used for solid homogeneous materials, some studies have applied this standard to achieve higher precision thermal calorimetry measurements of non-solid materials. For example, one study [26] used ASTM D5470 to measure the thermal properties of thermal interface materials (TIMs) with high precision and sensitivity. This study managed to minimize heat leakage through the TIMs, and maintain minimal temperature gradients through the meter bars while following ASTM D5470, in particular for lower heat powers. For the TIMs, a minimum thermal resistance of ~2.93E-3 K/W was measured with an uncertainty of 2.7% using a heat transfer rate of ~16.8 W. The study achieved calorimetric measurements with an order of magnitude improvement in precision and sensitivity over any previous investigation in its field.

The objective of the current study is to establish an accurate method of characterizing the thermal performance of a heat pipe with similar attributes to those shown by previous analyses of solid materials [25] and TIMs [26]. The key element of the method is the implementation of a variation of the ASTM D5470 standard via the use of radial calorimeters to obtain accurate values of the heat supplied to the evaporator section of the pipe, and the heat removed from the condenser section. Fig. 2 (b) illustrates the configuration of the calorimeters located at the evaporator and condenser sections of a sample heat pipe. Each calorimeter is formed from a cylindrical section of a solid material of known thermal conductivity. The heat pipe is tightly fit within a hole drilled axially through the center of each calorimeter, as shown in Fig. 2. An isoflux heat source is wrapped around the outer radius of the calorimeter at the evaporator end of the pipe, and some form of cooling technique is used to achieve isoflux heat removal from the calorimeter at the condenser end. If it is assumed that heat losses from the end caps of the calorimeters are negligible (as shown in Sections 4.2 and 4.3), then one-dimensional radial conduction will occur within each calorimeter, similar to ASTM D5470. Hence, the radial heat flows through each calorimeter can be quantified from point temperature readings at the embedded locations shown in the figure. In this manner, the calorimeters yield accurate values of the heat flows $Q_{in}$ and $Q_{out}$. Eqs. (2) and (3) quantify one-dimensional radial conduction, in the cylindrical coordinate system, into the evaporator ($Q_{in}$) and out of the condenser ($Q_{out}$), respectively:

$$Q_{in} = \frac{T_1 - T_2}{2\pi k l \ln \frac{r_2}{r_1}}$$

(2)

$$Q_{out} = \frac{T_3 - T_4}{2\pi k l \ln \frac{r_3}{r_4}}$$

(3)

where, $k$ is the known conductivity of the calorimeter blocks, $l$ is the length of the blocks, $r_1$, $r_2$, $r_3$ and $r_4$ are the radial locations of the measured temperatures $T_1$, $T_2$, $T_3$ and $T_4$ respectively, as illustrated in Fig. 2(b). It should be noted that these radial calorimeters behave in an analogous manner to calorimeters based on one-dimensional flow in Cartesian geometries, ASTM D5470 [25], as illustrated in Fig. 2(a).

Finally, the average surface temperatures of the evaporator and condenser sections of the pipe, shown as $T_e$ and $T_c$ in the figure, can either be recorded directly using sensors, or by extrapolation from the sensors embedded within the calorimeters. An experimental implementation of this method will be detailed in the following section.

![1-D Heat Pipe Diagram](image.png)

Fig. 2. A schematic of the calorimeter-based method for the thermal characterization of a flat plate test sample (a: ASTM D5470) and a heat pipe (b: current study), illustrating the radial calorimeters, and their embedded temperature sensors, at the evaporator and condenser ends of a sample heat pipe.
3. Experimentation

An experimental apparatus was realized to demonstrate the calorimeter-based thermal characterization method, and to record the characteristics for a sample heat pipe. The apparatus, illustrated in Fig. 3, featured radial calorimeters to quantify heat flow from an electrical heat source into the evaporator section of the heat pipe, and from the condenser section of the pipe to a liquid-based cooler. This section presents details of the experimentation apparatus (Section 3.1) and procedure (Section 3.2). Further information relating to data reduction, calibration and uncertainty analysis is presented in Section 4.
3.1. Apparatus

The experimental apparatus included (Fig. 3(a), (b) and (c)):

- A cylindrical heat pipe of 300 mm length × 6 mm diameter from Wakefield-Vette (WKV part number: 124656) with sintered Cu powder wicks and water as the working fluid.
- Thermal paste (heat sink compound, Dowsil/Dow Corning, thermal conductivity 0.59 W/m.K as per the supplier’s data sheet).
- Two 100 mm diameter × 100 mm long stainless steel 304 cylinders, configured as calorimeters. The cylinders featured fully drilled through 6.1 mm diameter holes, axially along their centerlines, for snug fitting of the heat pipe. The thermal paste was used to reduce the thermal contact resistance at the interfaces between the calorimeters and the pipe. The calorimeter at the evaporator end was wrapped within a 315 mm × 100 mm × 2 mm Keenova silicon flexible heater mat which was powered by an EL302RT Aim-TTi bench power supply (Fig. 3(a) and (b)). The calorimeter at the condenser end was surrounded by a 6 mm copper coil which carried a coolant, silicone oil, at a controlled temperature (25 °C) that was regulated using a Huber Ministat 125 water bath. An M500 TTS Micro Series pump was used to circulate the coolant from the isothermal bath through plastic tubing to the copper coil.
- Two split blocks made from poured fit polyurethane expandable insulation surrounded the calorimeters (Fig. 3(b)). Slots were machined for snug fitting of insulation around the copper coil and heater mat. This ensured that the evaporator and condenser calorimeters were surrounded with insulation (Note: the polyurethane material had a quoted thermal conductivity of <0.05 W/m.K).
- Eight calibrated negative temperature coefficient (NTC) precision thermistors (Amphenol Advanced Sensors, 0.768 mm diameter probe, Manufacture No:MA100BF103A) were embedded into two 1 mm diameter × 15 mm deep holes on both faces of each of the calorimeter blocks, see Fig. 3(c). The thermistors were used to accurately characterize the heat flow through the calorimeter blocks, into and out of the heat pipe. Thermal paste was used to minimize potential uncertainties associated with interfacial thermal resistance between the thermistors and the calorimeter blocks.
- A total of 32 K-type thermocouples (Omega PFA-insulated K type thermocouples, Stock No. 5TC-TT-KI-40) were used to record the following: the average surface temperature at the evaporator, adiabatic and condenser sections of the heat pipe (at axial distances of 5 mm, 95 mm, 115 mm, 185 mm, 205 mm and 295 mm from the outer end of the evaporator, see Fig. 3(c)); the internal temperatures within the SS304 calorimeter blocks; the inlet and outlet temperatures of the coolant; and the ambient air temperature; see Fig. 3 (a), (b) and (c). Thermal paste was used, in a similar manner as the thermistors, for the thermocouples that were attached to the SS304 blocks and the heat pipe.
- Two National Instruments 4 channel thermistor NI-9219 modules with USB Hi-Speed USB-9162 carrier chassis and a National Instruments 32 channel thermocouple NI TB-4353 module were used to wire the thermistors and thermocouples to a National Instruments NI PXIe-1082 data recorder respectively, for measurement via LabView software (see Section 4.5 for thermistor calibration).

3.2. Procedure

The experimental apparatus was assembled and connected to all ancillary equipment as shown in Fig. 3(a–c). For further information relating to the details of calibration see Section 4.5. To achieve steady-state conditions, the Huber Ministate 125 was used to circulate the coolant through the copper coils. When a steady-state temperature of ~25 °C was achieved on the thermocouples at the outer radius of the condenser calorimeter block (i.e. to ensure constant thermal conditions), the data recorder was started and the power supplied to the heater mat switched on. Input power levels (Qs) were tested over a range of 7.5–25 W. For each power input setting, the system was allowed to reach steady-state, defined as a change of <0.5 K over a 30 min period. Furthermore, the thermocouples at the same radial locations as the thermistors had to be within 0.5 K so that steady-state radial conduction could be assumed (Fig. 3(c)). Once each test run was complete, the data logger was stopped and the data stored. Steady-state temperature readings were averaged over the final 15 min of each power level. As the insulation used had a 96 °C transition to burn temperature the experiment was stopped if any temperatures exceeded 90 °C. At temperatures above this value, there was potential for insulation deformation.

4. Data reduction

This section presents the relations applied to achieve data reduction for the calorimeter and heat pipe characterization [1,7–9,11–13,29,30]. The full set of relations is used to obtain thermal characteristics of a sample heat pipe (i.e. Rthm and keff) from the temperature data recorded in the experiment. To this end, the calorimeters were used to quantify the heat flow going into the evaporator section (Qin) and out of the condenser section (Qout) of the heat pipe. Steady state temperature measurements of the calorimeter blocks, data reduction and an uncertainty analysis are also presented.

4.1. Thermal conductivity

The thermal conductivity of the stainless steel 304 used to form the calorimeters is shown as a function of temperature in Eq. (4) [29], which is quoted to less than ±2% uncertainty [8]:

\[
\log_{10} k = -1.4087 + 1.3982 \log_{10}(T) + 0.2543 (\log_{10}(T))^2 \\
-0.6260 (\log_{10}(T))^3 + 0.2334 (\log_{10}(T))^4 \\
+ 0.4256 (\log_{10}(T))^5 - 0.4658 (\log_{10}(T))^6 \\
+ 0.1650 (\log_{10}(T))^7 - 0.0199 (\log_{10}(T))^8
\]  

\[(4)\]

where, k is the thermal conductivity of the material, and T is the average absolute temperature of the object. The k value was calculated at a fixed, radially-averaged temperature within each of the evaporator and condenser calorimeters, for each power setting. This value ranged from 15.2 to 16.3 W/m.K over a temperature range of 25–60 °C.

4.2. Quantification of heat flow

In designing the experiment, it was intended that the temperature field in the calorimeter blocks would only vary radially and therefore be invariant in the circumferential and axial directions. It can be seen in Fig. 3(c) that the temperature measurements for calorimetry were recorded at known radial locations and axial depths. Axially, the thermistor temperatures were averaged, in the evaporator and condenser, to give \( T_{c\text{inner}} \), \( T_{c\text{inner}} \), \( T_{c\text{inner}} \) and \( T_{c\text{outer}} \) respectively, inner representing the thermistors located closest to the heat pipe. These temperatures were used in Eqs. (5) and (6) to evaluate the average flow of heat in (Qin) and out (Qout) of the heat pipe. It is important to note that the power supplied (Qs) was also recorded for all experiments, in order to quantify the difference between Qin and Qin:

\[
Q_{in} = \frac{T_{c\text{inner}} - T_{c\text{outer}}}{2\pi L_e \ln \left( \frac{L_e}{L_i} \right)}
\]

\[(5)\]

\[
Q_{out} = \frac{T_{c\text{inner}} - T_{c\text{outer}}}{2\pi L_e \ln \left( \frac{L_e}{L_i} \right)}
\]

\[(6)\]

where, \( L_e \) and \( L_i \) are the evaporator and condenser lengths, \( r_i \) and \( r_e \) are the radial locations of the thermocouples, and k is the thermal conductivity of the calorimeter blocks which was calculated using Eq. (4).
In order to confirm the validity of the assumption that end effects due to losses from the faces of the calorimeters are negligible (i.e. that the temperature field within each calorimeter varies radially, and is largely invariant in the circumferential and axial directions), a heat transfer analysis was carried out on the experimental apparatus using SOLIDWORKS® 2018 (x64 SP3.0 Flow Simulation, version 26.3.0.0063), a thermal simulation package based on a finite element solver. The boundary conditions and material selection were set to represent the experimental test rig as follows:

- the calorimeters were modelled as stainless steel AISI 304 cylinders of conductivity dependent on Eq. (4);
- an isoflux condition totaling to 20 W heat was applied around the circumference of the evaporator calorimeter;
- a fixed temperature boundary condition of 25 °C was applied around the circumference of the condenser calorimeter;
- the heat pipe was modelled as a 300 mm long hollow copper pipe (wall thickness 0.8 mm, k = 395 W/m.K) with a highly conductive core (k = 55,000-60,000 W/m.K, to ensure that the thermal resistance of the heat pipe ranged from 0.15 to 0.2 K/W);
- 25 mm thick thermal insulation of 0.05 W/m.K conductivity surrounded the calorimeters and adiabatic section of the heat pipe, and to account for any radiative losses an emissivity value of 0.7 was applied to the insulation;
- external cooling by natural convection and radiation was applied from all exposed surfaces into an ambient set to 23 °C.

Fig. 4 below illustrates the temperature gradients, axially and radially, for a cross section of the simulation. It was found that the heat lost, from 1-D conduction through the insulation, at the evaporator and condenser end caps was <2% of the overall power inputted to the evaporator. This served to confirm the assumption that end effects due to losses from the faces of the calorimeters are negligible. It was also found that for 15 mm deep temperature readings, in both faces of the blocks (i.e. similar to the experimental conditions), the axial temperature differences were small (~0.1–0.4 K for 20 W) and considered to be negligible; and, moreover, temperatures were invariant circumferentially. This confirmed that uniform heating and cooling of the heat pipe occurred in the calorimeter blocks. To put this in context, the temperature difference was ~2 K in the radial direction, between the inner and outer radii in both faces of the calorimeters. Furthermore, the simulation showed that for a supplied heat of 20 W (i.e. Q supplying to heat pipe), the heat input to the evaporator (Qin) was ≈16 W, and the heat transferred from the condenser (Qout) was ≈15 W. Thus, the simulation was consistent with the findings of the experimentation (see Section 5) that a significant proportion of the supplied heat (Qin) is lost to the surroundings, and is not transported into the condenser (as Qin).

4.3. Experimental verification of 1-D radial calorimetry for low Qin

In order to confirm 1-D radial conduction, an analysis of calorimeter temperatures at the lowest and highest thermal loads (6 W and 22 W) was performed. These configurations relate to the highest experimental uncertainty and temperature gradient respectively. This test measured the difference in the circumferential and axial point temperatures, with ±0.01 K uncertainty in the thermistors and ±0.1 K in the thermocouples, on both faces of the evaporator and condenser calorimeter blocks as presented in Fig. 5(a). The steady state temperature measurements for the 6 W and 22 W configurations, used to verify that end losses were negligible, are presented in Fig. 5(b) and (c) respectively. To confirm that the circumferential temperature variation was negligible and the radial temperature gradients followed the same slope (i.e. constant circumferential Qin), four circumferential temperature measurements were recorded at both the inner and outer radii in the calorimeter blocks (with maximum positional uncertainty of ±0.4%). For verification of 1-D axial conduction, a comparison was made between both faces of the calorimeter blocks. Fig. 5(b) and (c) illustrate that, for the 6 W and 22 W configuration, the differences in the radial and axial temperatures were ≤0.5 K and the temperature differences (i.e. slope of the curves in Fig. 5(b) and (c)) range from ~0.0001 K/°C to ~0.0003 K/°C for the evaporator and from 0.0003 K/°C to 0.0005 K/°C for the condenser) that define Qin and Qout were constant/parallel. This verifies the assumption that conduction within the calorimeter blocks is predominantly radial in direction.

4.4. Thermal characteristics

Using the values of Qin and Qout obtained from the calorimeters, the thermal performance of the heat pipe was analyzed in a similar manner to that presented in previous studies [1,7–9,11–13] in terms of thermal resistance (Rth), effective thermal conductivity (kth) [6], and thermal efficiency (ηth) [7–9]. Thermal resistance is calculated using Eq. (7), and it is used in Eq. (8) to evaluate the effective thermal conductivity:

\[
R_{th} = \frac{T_c - T_e}{Q_{in}}
\]

\[
k_{th} = \frac{L}{R_{th}A}
\]

where, \(T_c\) and \(T_e\) are the averages of the point surface temperatures of the heat pipe in the evaporator and condenser sections respectively (see Fig. 3(c)), L is the length of the heat pipe, A is the cross sectional area of the heat pipe, and Qin is the heat supplied to the evaporator section of the heat pipe (Eq. (5)), as quantified using the evaporator calorimeter.

Thermal efficiency, \(\eta_{th}\), is a measure of the quality of the

![Fig. 4. Cross-sectional temperature fields from a numerical thermal simulation of the experimental apparatus: heat supplied was 20 W.](image-url)
experimental apparatus. It describes the proportion of the heat received by the evaporator of the heat pipe that is captured by the condenser block:

\[ \eta_{th} = \frac{Q_{out}}{Q_{in}} \]  

(9)

4.5. Calibration and uncertainty analysis

A rigorous uncertainty analysis was employed in order to quantify how the uncertainties of each measured quantity propagate to the overall uncertainties in the thermal resistance and effective thermal conductivity of the heat pipes under test. Each measured quantity and its associated uncertainty are listed in Table 2.

The maximum uncertainties in each quantity calculated in Eqs. (10)–(16) were obtained using the method of Kline and McClintock (1953) for constant odds results, using the root-sum-square contribution of the individual contributions of the variables (Eq. (10)). Where \( \Delta Z \) was the uncertainty \((\pm)\) in the derived quantity \( Z = f (x_1, x_2, x_3 \ldots x_N) \), and \( \Delta x_i \) was the uncertainty \((\pm)\) in the primary variable \( x_i \). The uncertainty \( \Delta Z \) was then calculated as:

\[ \Delta Z = \sqrt{ \sum_{i=1}^{N} \left( \frac{\delta Z}{\delta x_i} \Delta x_i \right)^2 } \]  

(10)

Further relations capturing the uncertainties associated with the radial calorimeter apparatus are represented in Eqs. (11)–(15) below.

Although \( Q_s \) was not used in the analysis of thermal resistance and effective conductivity in this paper, it is assessed here to provide a comparison to reported techniques that utilize \( Q_s \). In this context, the voltages and currents supplied to the heat source were calibrated using a Fluke 175 True-RMS digital multimeter against a EL302RT Aim-TTi

Fig. 5. (a) Thermistor and thermocouple location and steady state temperature data for the (b) lowest thermal load (\( Q_{in} = 6 \text{ W} \)) and (c) highest thermal load (\( Q_{in} = 22 \text{ W} \)) on the inner and outer faces of the calorimeter blocks at the evaporator and condenser. Where, the data points at 0° locations are the thermistors and the points at 90°, 180° and 270° locations are the thermocouples, inner face 1 represents the side of the calorimeter closest to the adiabatic section of the heat pipe and outer face 2 is the corresponding outer face.
bench power supply to minimize any potential uncertainty in $Q_{th}$, see Table 2.

$$\Delta R_{th} \over R_{th} = \left[ \left( \Delta Q_{in} / Q_{in} \right)^2 + \left( \sqrt{2} (\Delta T) / \delta T \right)^2 \right]^{1/2} \tag{13}$$

The highest uncertainty associated with the effective thermal conductivity of the heat pipe was thermal resistance. There were, however, additional uncertainties associated with the length and diameter of the heat pipe. The manufacturer specification and Vernier precision of the length and diameter are presented in Table 2:

$$\Delta k_{eff} \over k_{eff} = \left[ \left( \Delta Q_{in} / Q_{in} \right)^2 + \left( \Delta T / T \right)^2 + \left( \Delta d / d \right)^2 \right]^{1/2} \tag{14}$$

Uncertainties in $Q_{out}$ do not influence the uncertainties in $R_{th}$ and $k_{eff}$, however, $Q_{out}$ was used to evaluate the losses from the apparatus to ambient and to quantify the experimental thermal efficiency ($\eta_{th}$), Eq. (15). Hence, a similar calorimeter configuration was used for the condenser as the evaporator, and Eq. (12) applies. However, the temperature gradients were slightly smaller in the condenser (due to ambient losses, $Q_{in} > Q_{out}$), leading to marginally larger percentage uncertainties in $Q_{out}$. An isothermal Huber Ministat 125 water bath was used to hold the condenser coolant that circulated around the calorimeter bar at a constant temperature, $\Delta T = \pm 0.1 \, K$. However, this uncertainty does not contribute to the uncertainty in $Q_{out}$ because the temperature measurements were read directly inside the calorimeter block.

$$\Delta \eta_{th} \over \eta_{th} = \left[ \left( \Delta Q_{in} / Q_{in} \right)^2 + \left( \Delta Q_{out} / Q_{out} \right)^2 \right]^{1/2} \tag{15}$$

The maximum uncertainties for $Q_{in}, Q_{th}, R_{th}, k_{eff}$ and $\eta_{th}$ at the lowest and highest thermal loads (5–22.5 W) are presented in Table 2. Where, the maximum uncertainties are associated with the lowest thermal load. It is evident that the calorimeter method produced thermal characterization data with low uncertainties compared to the leading experimental techniques reported in the literature ($\pm 20\%$ for $\leq 10 \, W$ and $\pm 10\%$ for loads between 10–30 W [18,19]). Moreover, note that in this study the uncertainty for all measured variables was within $\pm 6.5\%$ for $Q_{in} \geq 12.5 \, W$.

### 5. Results and discussion

This section presents the measured thermal characteristics, $R_{th}, k_{eff}$ (Section 5.1) and $\eta_{th}$ (Section 5.2) of a heat pipe tested over a power input range of 7.5–25 W. Specific focus is placed on the values of the characteristics that are calculated when $Q_{in}$ is quantified using radial calorimetry instead of $Q_{th}$, as per other reported techniques [1,7–9,11–13].

#### 5.1. Thermal resistance & thermal conductivity

Fig. 6(a) and (b) below respectively present the thermal resistance ($R_{th}$) and effective thermal conductivity ($k_{eff}$) of the heat pipe as a function of nominal power supplied, when the electrical heat input, $Q_{in}$, and the calorimeter, $Q_{th}$, are used to evaluate the heat supplied to the evaporator (Eq. (7)). From Fig. 6(a) and (b), it is evident that the heat pipe behaved in a manner that is consistent with previous studies, since an increase in the nominal heat input led to a decrease in $R_{th}$ ($\sim 0.3–0.19 \, K/W$), with a corresponding increase in $k_{eff}$ ($\sim 35–54 \, kW/m.K$). It is also evident that the thermal resistance value of the heat pipe is higher when calorimetry was used (corresponding to effective thermal conductivity values 13–30%, lower). Arguably, due to the calorimeter blocks employed in this method, it is likely that a lower

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### Table 2

| Measured Variable | Uncertainty | Derived Quantities | Uncertainty (%) |
|-------------------|-------------|--------------------|-----------------|
| Temperature for $Q_{in}$, $T$ | $\pm 0.01 \, K$ | Power, $Q_{in}$ | $\pm 4.0$ |
| Heat Pipe, $T_r$, $T_e$ | $\pm 0.1 \, K$ | Heat In, $Q_{th}$ | $\pm 4.6$–$4.9$ |
| Length, $l$ | $\pm 0.05 \, mm$ | Heat Out, $Q_{out}$ | $\pm 4.7$–$5.1$ |
| Voltage, $V$ | $\pm 100 \, mV$ | Thermal Resistance, $R_{th}$ | $\pm 5.8$–$7.5$ |
| Current, $I$ | $\pm 10 \, mA$ | Effective Thermal Conductivity, $k_{eff}$ | $\pm 6.0$–$8.0$ |
| Radius, $r$ | $\pm 0.05 \, mm$ | Thermal Efficiency, $\eta_{th}$ | $\pm 6.5$–$7.0$ |
| Diameter, $d$ | $\pm 0.1 \, mm$ | $\pm 0.32 \, W/m. K$ | |

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[1,7–9,11–13]
Experimental thermal efficiency, $\eta_{th}$ as a function of nominal heat input, $Q_{\text{nominal}}$. Both conditions, $Q_{\text{in}}$ and $Q_{\text{s}}$, used the same value of $Q_{\text{out}}$ to illustrate the relatively low $\eta_{th}$ values associated with previously published characterization techniques [7,9,14,15]. Note: the ‘Q, data’ and literature are presented for illustrative purposes only, so uncertainty bars are not relevant except for [7] who presented uncertainty in $\eta_{th}$.

Assume that all of the (electrical) heat supplied to the apparatus, $Q_{\text{th}}$, is transferred to the evaporator section of the heat pipe (i.e., $Q_{\text{in}} = Q_{\text{th}}$). In this context, the calorimeter block method aims to illustrate the discrepancy between $Q_{\text{th}}$, and $Q_{\text{in}}$ and, in addition, the quality of the experimental test rig ($Q_{\text{th}} = \frac{Q_{\text{out}}}{Q_{\text{in}}}$).

Fig. 7 illustrates the discrepancies in assuming that all of the heat supplied to the apparatus, $Q_{\text{th}}$, is extracted from the condenser section (i.e., $Q_{\text{out}} = Q_{\text{th}} = Q_{\text{s}}$). It can be seen that over the power input range of 7.5–25 W, $Q_{\text{in}}$ was only 70–90% of $Q_{\text{th}}$ with the largest difference occurring at lower power inputs (7.5–15 W). This illustrates that heat is lost from the source to ambient and $Q_{\text{in}} \neq Q_{\text{th}}$. The figure also includes the relevant data for the published studies that recorded $\eta_{th}$ as a function of $Q_{\text{in}}$ [7,9,14,15]. Only one of these studies [7] features credible uncertainties in $\eta_{th}$ due to quantification of the environmental losses from the heat source. Hence, data from [7] are presented in the figure. For the referenced studies in Fig. 7 there is a large discrepancy between $Q_{\text{in}}$ and $Q_{\text{out}}$ with $\eta_{th}$ varying between 20% [9] to 79% [7]. For the calorimeter-based method $\eta_{th}$ is much higher giving values of 70–80% for the lower thermal loads (6–12.5 W), rising to 85–91% for the higher loads (i.e., 16.5–22 W). Hence, in the current study the difference between $Q_{\text{in}}$ and $Q_{\text{out}}$ or $\eta_{th}$ should predominantly represent losses from the ‘adiabatic’ section of the pipe if calorimeter end cap losses are considered negligible, Section 4.2. If the electrical heat input, $Q_{\text{th}}$, is substituted for the calorimeter-based value of $Q_{\text{in}}$, the $\eta_{th}$ value of the experimental apparatus dropped to 54–80% for all thermal loads – an indication of the magnitude of the errors incurred in previously published characterization techniques [1,7–9,11–16]. It is evident that the calorimeter-based thermal characterization method presented in this paper yields greatly improved quantification of heat transferred through the pipe.

5.2. Experimental thermal efficiency

The primary focus of this study was to examine the limitations of previously published characterization techniques [7,9,14–16] which present for illustrative purposes only, so uncertainty bars are not relevant except for [7] who presented uncertainty in $\eta_{th}$.

Assume that all of the (electrical) heat supplied to the apparatus, $Q_{\text{th}}$, is transferred to the evaporator section of the heat pipe (i.e., $Q_{\text{in}} = Q_{\text{th}}$). In this context, the calorimeter block method aims to illustrate the discrepancy between $Q_{\text{th}}$, and $Q_{\text{in}}$ and, in addition, the quality of the experimental test rig ($Q_{\text{th}} = \frac{Q_{\text{out}}}{Q_{\text{in}}}$).

Fig. 7 illustrates the discrepancies in assuming that all of the heat supplied to the apparatus, $Q_{\text{th}}$, is extracted from the condenser section (i.e., $Q_{\text{out}} = Q_{\text{th}} = Q_{\text{s}}$). It can be seen that over the power input range of 7.5–25 W, $Q_{\text{in}}$ was only 70–90% of $Q_{\text{th}}$ with the largest difference occurring at lower power inputs (7.5–15 W). This illustrates that heat is lost from the source to ambient and $Q_{\text{in}} \neq Q_{\text{th}}$. The figure also includes the relevant data for the published studies that recorded $\eta_{th}$ as a function of $Q_{\text{in}}$ [7,9,14,15]. Only one of these studies [7] features credible uncertainties in $\eta_{th}$ due to quantification of the environmental losses from the heat source. Hence, data from [7] are presented in the figure. For the referenced studies in Fig. 7 there is a large discrepancy between $Q_{\text{in}}$ and $Q_{\text{out}}$ with $\eta_{th}$ varying between 20% [9] to 79% [7]. For the calorimeter-based method $\eta_{th}$ is much higher giving values of 70–80% for the lower thermal loads (6–12.5 W), rising to 85–91% for the higher loads (i.e., 16.5–22 W). Hence, in the current study the difference between $Q_{\text{in}}$ and $Q_{\text{out}}$ or $\eta_{th}$ should predominantly represent losses from the ‘adiabatic’ section of the pipe if calorimeter end cap losses are considered negligible, Section 4.2. If the electrical heat input, $Q_{\text{th}}$, is substituted for the calorimeter-based value of $Q_{\text{in}}$, the $\eta_{th}$ value of the experimental apparatus dropped to 54–80% for all thermal loads – an indication of the magnitude of the errors incurred in previously published characterization techniques [1,7–9,11–16]. It is evident that the calorimeter-based thermal characterization method presented in this paper yields greatly improved quantification of heat transferred through the pipe.

6. Conclusions

In this study, an accurate calorimeter-based method for the thermal characterization of a heat pipe has been demonstrated. The key element of the method is the use of radial calorimeters to quantify the heat flow into the evaporator section, and from the condenser section, of the heat pipe, with low levels of uncertainty, $\pm 4.9\%$ for thermal loads $\simeq 6–12.5$ W and $\pm 4.6\%$ for loads $\simeq 12.5$ W. In this regard, the method is greatly superior to other characterization techniques, reported in the literature, that do not quantify heat losses. To demonstrate the method, a sintered copper heat pipe sample was characterized in terms of thermal resistance, effective thermal conductivity and thermal efficiency, for an input power range of $\simeq 7.5–25$ W and a maximum experimental uncertainty of 8%. The conclusions are as follows:
● Thermal resistances in the range of 0.3–0.19 K/W were recorded for heat inputs of approximately 6–22.5 W, with corresponding effective thermal conductivity values ranging from ~35–54 kW/m.K. These resistance values are 15–32% higher than those obtained using the techniques reported in the literature, because the method features a more accurate means of quantifying the heat flow into the evaporator section of the heat pipe.

● Experimental thermal efficiency values of ~70–80% and 85–91% were recorded for the lower (6–12.5 W) and higher (16.5–22 W) ranges of input power, respectively. These values of experimental thermal efficiency are far superior to those reported in the literature, which range from 45 to 79% due to inaccuracies in the methods used to quantify heat flows into and out of heat pipes.

The calorimeter-based thermal characterization method demonstrated in this paper resolves two critical limitations in previously-reported techniques: the assumption that the heat flow into the evaporator section of the heat pipe can be quantified from the electrical heat supplied to the test apparatus (in fact, it can be significantly lower depending on experimental setup); and the very high uncertainties (up to ~80%) in the measured values of the heat flow from the condenser section of the pipe. In this regard, the calorimeter-based method provides accurate thermal characteristics for heat pipes and, as such, represents a potential basis for a standard thermal characterization technique for heat pipes.

CRediT authorship contribution statement

Joseph P. Mooney: Conceptualization, Methodology, Data curation, Formal analysis, Software, Investigation, Writing - original draft, Writing - review & editing. Jeff Punch: Conceptualization, Methodology, Validation, Formal analysis, Writing - review & editing, Supervision, Project administration, Funding acquisition. Nick Jeffers: Conceptualization, Methodology, Validation, Formal analysis, Writing - review & editing, Supervision.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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