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Theoretical and experimental analysis of a horizontal Planar Liquid-Vapour Thermal Diode (PLVTD)

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Keywords

Thermal diode; one-way heat flow; phase change; solar collector; heat rejection

Highlights

- Horizontal planar unidirectional heat transfer device was realised (0.15m² PLVTD)
- Applications in solar collectors and climate control building envelopes
- One-dimensional lumped parameter heat transfer model developed & interrogated
- Forward mode heat transfer $150 < U_f < 500 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ depends on temperature & heat flux
- Reverse mode insulation $U_r = 2 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ achievable using 70mm cavity for 97% diodicity

Abstract

Thermal diodes are unidirectional heat transfer devices, analogous to electrical diodes, which offer low resistance (thermal conductance) in one direction and high resistance (thermal insulation) in the other. Thermal diodicity has significant potential to improve the efficacy of a wide variety of heating and cooling devices. This paper presents pioneering work to experimentally measure the heat transfer characteristics of a 0.15m² passive Planar Liquid-Vapour Thermal Diode (PLVTD). Prior work has typically examined much smaller devices aimed at micro-electronics cooling applications whereas the present work aims to improve understanding of larger scale devices for incorporation in solar collectors and multi-function climate control building envelopes. Such applications can facilitate local renewable energy generation (solar and ambient heat collection) and improve cooling system energy efficiency (passive heat rejection) in order to address the climate crisis by decarbonising built environment energy use. Experimental work involved a horizontally oriented PLVTD formed of two parallel isothermal plates with integral serpentine heat exchangers and external insulation. Plate, fluid, and ambient temperatures were controlled and measured to determine heat transfer coefficients under various temperature difference and heat flux operating scenarios in order to validate a simple one-dimensional lumped parameter model.

Measured forward mode heat transfer $150 < U_f < 500 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ combined with reverse mode insulation $U_r = 10 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ corresponds $\zeta \approx 88\%$ diodicity at low condenser temperatures and fluxes ($T_2 \approx 15^\circ\text{C}$ and $q/A \approx 120 \text{ W} \cdot \text{m}^{-2}$) and $\zeta \approx 96\%$ at high condenser temperatures and heat fluxes ($T_2 \approx 60^\circ\text{C}$ and $q/A \approx 2800 \text{ W} \cdot \text{m}^{-2}$). Forward mode performance increases with increasing heat flux and (to a lesser extent) with increasing operating temperature but is largely independent of PLVTD dimensions. Reverse mode performance is largely independent of heat flux and temperature but reduces (improves thermal insulation) with increasing cavity depth. The model has been used to show that a stainless steel PLVTD with $x = 70 \text{ mm}$ cavity can achieve $U_r = 2 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ with similar forward mode performance to the experimental prototype. Such a device would achieve diodicity of $\zeta > 97\%$ and be suitable for application in solar collectors and climate control building envelopes.

1 Introduction

A thermal diode is a unidirectional heat transfer device that operates in a manner analogous to an electrical semiconductor diode by offering low resistance (thermal conductance) in one direction and high resistance (thermal insulation) in the other. The broad concept of a “Thermal diode” and the ideas of “Thermic diodicity” and “Thermal Rectification” have been examined by a number of authors (Go & Sen, 2010; Roberts & Walker, 2011; Boreyko et al., 2011; Ben-Abdallah & Biehs, 2013; Dos Santos Bernardes, 2014; Bairi et al., 2014; Pei et al., 2017). Many of the aforementioned studies relate to thermal rectification using asymmetric solid material conduction, surface radiation, or natural convective thermosyphon arrangements rather than liquid-vapour phase change, but the definitions are equally applicable irrespective of the heat transfer mechanisms involved. A key parameter describing the effectiveness of a thermal diode is its diodicity (ζ), for which several definitions are found in the literature. The definition preferred by the present authors is the “diodicity coefficient” which conveniently scales the degree of rectification between zero and unity. Diodicity coefficient (Equation 1) is defined in terms of the apparent thermal conductivity (k) of the diode in forward (f) and reverse (r) modes but can alternatively be written in terms of thermal power (q), heat flux (q/A), thermal conductance (U), or thermal resistance (R) where the dimensional quantities x and A relate to the depth and surface area of the diode with $A = yz$ applying in the case of a rectangular flat plate diode of the form shown in Figure 1.

$$\zeta = \frac{k_f - k_r}{k_f + k_r} \quad (0 \leq \zeta \leq 1)$$

Equation 1

Figure 1 – Schematic diagram of a horizontal PLVTD

As indicated in Figure 1, a Planar Liquid-Vapour Thermal Diode (PLVTD) realises thermal diodicity by enabling latent transfer (evaporation, vapour mass flow, and condensation) in forward mode and by preventing latent heat transfer in reverse mode. The two operating modes are illustrated in Figure 2 and explained in more detail below. High rates of heat transfer occur during steady state forward mode operation when latent heat transfer processes are enabled because:

- Plate 1 is wetted by the liquid working fluid, and;
- Plate 1 is hotter than Plate 2 and hotter than the working fluid saturation temperature so that evaporation occurs, and;
- Plate 2 is cooler than Plate 1 and cooler than the working fluid saturation temperature so that condensation occurs.
- Working fluid vapour mass flows across the cavity from Plate 1 to Plate 2, primarily driven by the saturation pressure differential, but also in part by natural convection resulting from the density differential and gravity (buoyancy effects).
- Working fluid liquid returns from Plate 2 to Plate 1. In the simple horizontally oriented PLVTD featured in this study, condensate returns as drips driven by gravity. In PLVTD concepts which operate in vertical or tilted orientations, condensate flow returning to the evaporator must overcome gravity by utilising a capillary wick (Smyth, 2017), mechanical pump (Pugsley et al., 2016), or other driver such as surface wettability gradient (Boreyko and Chen, 2013) or applied electrostatic force (Traipattanakul et al., 2019).

Much lower rates of heat transfer occur during steady state reverse mode operation because Plate 2 is not wetted, meaning that no evaporation can occur, and thus no

latent heat flows between the plates. Residual sensible heat transfer between the plates occurs in reverse mode due to:

- Natural buoyant convection and gaseous conduction associated with working fluid vapour (and non-condensable gases, if present) in the cavity.
- Conductive heat transfer through the diode sidewalls and supporting structure.
- Radiative heat transfer between the two plate surfaces.

The PLVTD's diodicity relies upon ensuring that the working fluid circulation arrangement continually wets Plate 1 in forward mode and enables Plate 2 to remain dry in reverse mode. Inadvertent wetting of Plate 2 during reverse mode operation will facilitate latent heat transfer that will undermine the insulation performance and significantly diminish the diodicity.

Key to annotations

- A) Evaporation of the working fluid due to heat input at temperature T_1
- B) Working fluid vapour mass transfer across cavity due to pressure difference $\Delta P_{12} = P_1 - P_2$. The working fluid exists in a condition close to the liquid-vapour saturation point. The average vapour pressure (P_{Lv}) corresponds to the saturation temperature (T_{Lv}) which is approximately equal to the average operating temperature $T_{12} = (T_1 + T_2) / 2$
- C) Condensation of the working fluid produces heat output at temperature T_2
- D) Buoyancy induced working fluid vapour convection due to temperature difference $\Delta T_{12} = T_1 - T_2$
- E) Buoyancy induced working fluid liquid convection
- F) Residual heat transfer via radiation between plates and conduction through sidewalls and structural elements

Figure 2 – Heat transfer mechanisms in a Planar Liquid-Vapour Thermal Diode

113 Thermal diodes are well suited to electronics cooling applications (Boreyko and Chen,
114 2011 & 2013; Chen et al., 2012; Blet et al., 2017; Avanessian and Hwang, 2018;
115 Traipattanakul et al., 2019; Wong et al., 2019), solar collector applications (De Beijer,
116 1998; Quinlan, 2010; Souliotis et al., 2011&2017; Smyth et al., 2015a&b, 2017, 2018,
117 2019; Pugsley et al., 2016; Pugsley, 2017; Muhumuza et al., 2019a & b), climate
118 control building envelopes (Chen et al., 1998; Varga, 2002; Reay et al., 2014;
119 Villeneuve et al., 2017), and permafrost stabilisation applications (Pan and Wu, 2002;
120 Pei et al., 2017). Applications in spacecraft (Gaddam and Huxtable, 2017) and caloric
121 refrigeration systems (Hess et al., 2019), and thermal computing (Tso and Chao, 2016;
122 Shen et al., 2018) have also been proposed. The majority of extant published work on
123 liquid-vapour phase change thermal diodes relates to either axial devices (eg heat
124 pipes), annular devices (eg cylindrical or elliptical tanks), or small planar devices (eg
125 jumping drop diodes).

126 Wong et al. (2019) examined an axial liquid-vapour thermal diode with small circular
127 heat transfer plates (equal evaporator and condenser areas $A_e=A_c=0.01\text{m}^2$) oriented
128 horizontally and separated by a relatively large cavity ($x=76\text{mm}$). The device is
129 superficially similar to a wickless heat pipe, and essentially similar to the PLVTD
130 described in Figures 1 and 2, relying upon gravity to induce the return flow of
131 condensate from the condenser to the evaporator pool. Tests demonstrated that the
132 device achieved forward and reverse mode thermal conductances of $U_f \approx 40 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$
133 and $U_r \approx 16 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ corresponding to diodicity of $\zeta \approx 43\%$. Reverse mode heat transfer
134 appears to be mainly governed by conduction whereas forward mode heat transfer is
135 governed by buoyant natural convection (thermosyphonic action of air and water) and
136 boosted by liquid-vapour-liquid phase change (latent heat transfer).

137 Smyth et al. (2017 & 2018) and Muhumuza et al. (2019a & b) examined various
138 cylindrical solar water heaters featuring annular liquid-vapour thermal diodes. Early
139 prototypes were oriented vertically whilst more recent evolutions have been horizontal
140 such that the thermal diode operates broadly similarly to Figure 3a with condensate
141 return via gravity. According to Quinlan (2010), Smyth et al. (2017) and Pugsley
142 (2017), the "Mark IV" was formed of a 300mm diameter, 1000mm long, cylindrical
143 outer vessel and a 200mm diameter, 900mm long, inner vessel arranged concentrically
144 to form an annulus cavity of $x=50\text{mm}$. During heat collection (forward mode) wetted
145 pockets on the inner surface of the outer vessel act as evaporators ($A_e \approx 0.5\text{m}^2$)
146 transferring latent heat in the form of vapour which condenses on the outer surface of
147 the inner vessel ($A_c \approx 0.6\text{m}^2$). During heat retention (reverse mode) the annulus acts
148 as an insulator because the outer surface of the inner vessel is kept dry. Test results
149 showed that the annular thermal diode achieved forward and reverse mode heat
150 transfer $U_f A \approx 13 \text{ W}\cdot\text{K}^{-1}$ and $U_r A \approx 1.3 \text{ W}\cdot\text{K}^{-1}$ corresponding to diodicity of $\zeta \approx 82\%$.

151 Comprehensive work on planar liquid-vapour thermal diodes was undertaken by
152 Boreyko and Chen (2011 & 2013) who developed, tested and modelled a PLVTD of
153 similar form to that illustrated in Figures 1 and 2. A key innovation in their device was
154 the use of a superhydrophobic condenser and a superhydrophilic wicked evaporator to
155 enable the working fluid to return from the condenser to the evaporator via the
156 “jumping drop” phenomenon. This feature gave the device gravity-independent
157 operation enabling it to function in any orientation (horizontal, vertical or tilted) which
158 is important for its primary application in mobile electronic devices, but also relevant
159 to solar collectors and building envelopes. Experimental work utilised a small prototype
160 consisting of two square copper plates ($A_e=A_c=0.006\text{m}^2$) separated by an EPDM gasket
161 (various thicknesses up to $x=4\text{mm}$) and achieved forward and reverse mode thermal
162 conductances of $U_f=40500\text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ and $U_r=165\text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ corresponding to diodicity
163 of $\zeta=99\%$. Using a similar but smaller device ($A_e=A_c=0.0012\text{m}^2$), Traipattanakul et al.
164 (2019) explored the possibility of using an electric field to enable jumping drop action
165 across a wider separation of the diode plates ($x=7\text{mm}$) to enable lower reverse mode
166 performances. The work successfully proved the concept, but the relatively small plate-
167 to-plate cavity width (x) limited the overall thermal insulation values to only
168 $\sim 10\text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ which is an order of magnitude greater than typically required for solar
169 collector and building envelope applications.

170 All of the abovementioned experimental studies used water as the working fluid to
171 facilitate the latent heat transfer, and with the exception of the work by Wong et al.
172 (2019), all utilised cavity evacuation to remove non-condensable gases and facilitate
173 low temperature evaporation of water at sub-atmospheric pressures. Boreyko and
174 Chen (2013) demonstrated clearly that forward mode heat transfer is maximized when
175 non-condensable gases are removed such that the cavity pressure approaches the
176 saturation pressure of the working fluid.

177 The most basic realization of a PLVTD is the horizontally oriented form shown in
178 Figure 2 whereby Plate 1 is permanently wetted by a pool of liquid working fluid which
179 covers the evaporator surface. Condensate forms on Plate 2 when forward mode heat
180 transfer occurs and this liquid flows back to Plate 1 due to gravity so that operation
181 can continue indefinitely. This form of PLVTD could feasibly be used as a heat transfer
182 element in a concentrating solar water heater (see Figure 3a) or as a flat roofing panel
183 for buildings in hot climates where daytime solar gain control and night-time free-
184 cooling are required (see Figure 3b).

185 The study presented in this paper aims to build upon the aforementioned studies by
 186 improving understanding of large scale PLVTDs. Prior work on PLVTDs has generally
 187 been focussed on small devices with evaporator and condenser plate areas of
 188 $A \leq 0.01 \text{m}^2$ and cavity depths of $x \leq 7 \text{mm}$ which, whilst relevant to micro-electronics
 189 cooling applications, are of limited direct relevance to solar collector and multi-function
 190 climate control building envelope applications. Prior work on larger scale thermal
 191 diodes has generally focussed on devices with inherently axial or annular forms which
 192 differ significantly from the flat plate form of a true PLVTD. This study describes
 193 pioneering work to experimentally measure the heat transfer characteristics of a
 194 0.15m^2 PLVTD under various realistic operating conditions and presents a
 195 corresponding validated one-dimensional lumped parameter heat transfer model.
 196

197 **Figure 3 – Practical applications for a horizontal PLVTD**

199 **2 Theoretical model**

200 **2.1 Heat flux, temperature and pressure**

201 Heat transfer is driven by the difference in temperature between the two plates (ΔT_{12}).
202 The transferred thermal power (q_{12}), transferred heat flux (q_{12}/A_p), overall thermal
203 resistance (R_{fr}), and overall thermal conductance (U_{fr}) through the PLVTD are related
204 according to Equation 2. Figure 4 shows the main heat transfer paths and
205 corresponding thermal resistances through a PLVTD in both forward and reverse mode.

$$206 \quad U_{fr} = \frac{q_{12}}{yz(T_1 - T_2)} = \frac{q_{12}}{A_p \Delta T_{12}} = \frac{1}{A_p R_{fr}} \quad \text{Equation 2}$$

207 In forward mode, the dominant thermal transmission mechanism is latent heat transfer
208 associated with working fluid liquid-vapour-liquid phase changes and the net transfer
209 of working fluid vapour mass across the cavity between the two plates. In reverse
210 mode, thermal transmission occurs via several different mechanisms including working
211 fluid natural convection (vapour and liquid without phase change), radiation across the
212 cavity, and conduction through the envelope and supporting structural elements, but
213 no latent heat transfer occurs. In order to transfer heat in forward mode the working
214 fluid must exist in vapour phase at the evaporator temperature (T_1) and in liquid phase
215 at the condenser temperature (T_2). If all non-condensable gases are removed, then
216 the resulting pressure inside the PLVTD will be approximately equal to the saturation
217 pressure (P_{Lv}) of the working fluid at the average operating temperature
218 $T_{12} = (T_1 + T_2)/2$. This internal pressure will usually be different from the atmospheric
219 pressure outside necessitating internal structures to ensure that the PLVTD envelope
220 can withstand the pressure differential (hence need for supporting structure).

221 **2.2 Thermal resistance lumped parameter model**

222 The total thermal resistance associated with latent heat transport (R_{LvL}) through the
223 working fluid can be defined as the sum of individual evaporation (R_e), vapour mass
224 flow (R_{ec}), and condensation (R_c) components acting in series. Thermal resistance R_{LvL}
225 is inherently much lower than the sum of the series thermal resistances associated
226 with sensible heat transport through the working fluid in its liquid (R_L) and vapour (R_v)
227 forms (ie natural convection without phase change). Thermal resistances associated
228 with conduction through the two plates ($2R_p$) act in series with the aforementioned
229 components, whereas thermal resistances associated with radiation between the plates
230 (R_R), conduction through the sidewalls (R_w), and conduction through structural support
231 elements (R_s) act in parallel. Inspection of the resulting thermal resistance network in
232 Figure 4c suggests that the overall thermal resistance through the PLVTD during
233 forward mode operation (R_f) can be approximated by Equation 3.

a) Forward mode thermal resistance

b) Reverse mode thermal resistances

c) Thermal resistance network describing heat transfer behaviour

d) Descriptions of thermal resistances used in the model

R_{pe}	Conductive thermal resistance through the plate which acts as the evaporator in forward mode	R_w	Conductive thermal resistance through four sidewalls forming the PLVTD envelope
R_{pc}	Conductive thermal resistance through the plate which acts as the condenser in forward mode	R_s	Conductive thermal resistance through the structural members which support the PLVTD envelope
R_e	Resistance associated with working fluid evaporation occurring at the evaporator surface	R_v	Resistance associated with vapour phase natural convection of the working fluid
R_c	Resistance associated with working fluid condensation occurring at the condenser surface	R_L	Resistance associated with Liquid phase natural convection of the working fluid
R_{ec}	Mass transfer resistance associated with the flow of vapour from the evaporator to the condenser	R_R	Resistance associated with heat transfer via long wave electromagnetic radiation between the two plates
R_{LvL}	Overall Liquid-vapour-Liquid phase change heat transfer resistance		

Figure 4 – Forward and reverse mode heat transfer paths with corresponding thermal resistance network

238 The evaporator plate is dry in reverse mode so that the latent heat transport thermal
 239 resistances tend towards infinity and $1/(R_e+R_{ec}+R_c) \rightarrow 0$. Inspection of the thermal
 240 resistance network then yields Equation 4 to describe the overall thermal resistance
 241 through the PLVTD during reverse mode operation (R_r).

$$242 \quad R_f = 1/\left(\frac{1}{R_e+R_{ec}+R_c} + \frac{1}{R_w} + \frac{1}{R_s} + \frac{1}{R_R}\right) + 2R_p \quad \text{Equation 3}$$

$$243 \quad R_r = 1/\left(\frac{1}{R_L+R_v} + \frac{1}{R_w} + \frac{1}{R_s} + \frac{1}{R_R}\right) + 2R_p \quad \text{Equation 4}$$

244 **2.3 Envelope, structure, and radiation thermal resistances**

245 The envelope and structure thermal resistances can be evaluated using conventional
 246 Fourier methods. Assuming that the PLVTD envelope is of a basic rectangular box form
 247 (as per Figure 1) and the internal supporting structure is an array of cylindrical struts
 248 similar to that used in vacuum glazing panels (Collins and Fischer-Cripps, 1991; Fang
 249 et al., 2014) the thermal conductances and corresponding resistances of the
 250 evaporator and condenser plates ($U_{pe} \approx U_{pc}$), the envelope sidewalls (U_w), and the
 251 supporting structure (U_s), can be evaluated according to Equations 5 to 9. The
 252 nomenclature used for the various dimensional terms are defined in full in Section 9.
 253 The symbols U , R , k , A and d refer to thermal conductance, thermal resistance, thermal
 254 conductivity, area and distance respectively. The symbols and subscripts x , y , z refer
 255 to dimensions in the respective axes defined on Figure 1. The subscripts p , w , and s
 256 relate to the plate, wall and struts respectively. The term N_s refers to the number of
 257 struts in the internal supporting structural array where each strut is assumed to be a
 258 cylindrical tube with diameter d_s , wall thickness d_{sw} , and approximate length x . The
 259 term d_{ss} refers to the centre-to-centre spacing between adjacent struts and the terms
 260 d_{sxy} and d_{sxz} refer to spacings between struts and adjacent sidewalls.

$$261 \quad U_p = \frac{1}{A_p R_p} = \frac{k_p}{x_p} \quad \text{Equation 5}$$

$$262 \quad U_w = \frac{1}{A_p R_w} = \frac{2z_w(y+z-2z_w)k_w}{xyz} \quad \text{Equation 6}$$

$$263 \quad U_s = \frac{1}{A_p R_s} = \frac{N_s A_s k_s}{xyz} \quad \text{Equation 7}$$

$$264 \quad A_s = \pi \left(\frac{d_s}{2}\right)^2 - \pi \left(\frac{d_s - 2d_{sw}}{2}\right)^2 \quad \text{Equation 8}$$

$$265 \quad N_s = \left(\frac{y-2d_{sxy}}{d_{ss}} + 1\right) \left(\frac{z-2d_{sxz}}{d_{ss}} + 1\right) = \left(\frac{y-x}{d_{ss}} + 1\right) \left(\frac{z-x}{d_{ss}} + 1\right) \quad \text{Equation 9}$$

266 Radiative heat transfer resistance between the two parallel plates can be determined
 267 using Equation 10 (Twidell & Weir, 2006) based on the PLVTD absolute operating
 268 temperatures in Kelvin degrees (T_{12}), emissivities of the wet or dry plates (ε_1 and ε_2),
 269 the Stefan-Boltzmann constant ($\chi = 5.67 \times 10^{-8} \text{ W}\cdot\text{m}^{-2}\text{K}^{-4}$) and the area over which
 270 radiative heat exchange occurs ($A \approx A_p$, assuming negligible wall thicknesses).

$$271 \quad R_R = \frac{1/\varepsilon_1 + 1/\varepsilon_2 - 1}{4 \chi A (T_{12})^3} \quad \text{Equation 10}$$

272 **2.4 Evaporation, condensation and working fluid mass flow**

273 Heat transfer through the PLVTD via the working fluid is dependent upon a number of
 274 different mechanisms:

- 275 • Sensible heat transfer via buoyancy driven natural convection of the working
 276 fluid, in both vapour (R_v) and liquid (R_L) states, in reverse mode, due to plate-
 277 to-plate temperature differential ΔT_{12} .
- 278 • Working fluid evaporation (R_e) in forward mode.
- 279 • Latent heat transfer due to net vapour mass flow between the plates in forward
 280 mode (R_{ec}). Fluid motion is primarily driven by the vapour pressure differential
 281 (ΔP_{12} arising from ΔT_{12}) and to a lesser extent by corresponding buoyancy forces.
- 282 • Working fluid condensation (R_c) in forward mode.

283 Working fluid thermal resistances and corresponding conductances can be evaluated
 284 using Equation 11 from appropriate Nusselt number correlations (Nu), characteristic
 285 dimensions (L) and working fluid thermal conductivities (k).

$$286 \quad R = \frac{1}{A_p U} = \frac{L}{A_p k Nu} \quad \text{Equation 11}$$

287 For sensible convective heat transfer through vapour within a horizontal PLVTD of the
 288 form shown in Figure 2 (relevant to evaluation of R_v) the Nusselt number is
 289 approximately $Nu \approx 1$ during reverse mode operation because the hot surface is at the
 290 top of the cavity. The characteristic dimension is approximately equal to the cavity
 291 depth ($L \approx x$).

292 For sensible convective heat transfer through liquid within a horizontal PLVTD (relevant
 293 to evaluation of R_e and R_L) the Nusselt number can be evaluated using Equations 12
 294 to 14 depending upon Rayleigh number and operating mode. Rayleigh numbers are
 295 typically evaluated using Equation 15 which accounts for buoyancy and frictional

effects based upon gravitational acceleration ($g=9.81 \text{ m/s}^2$ at sea level), plate-to-liquid temperature difference ($\Delta T_{L1}=T_1-T_{12}$) and temperature dependent working fluid thermodynamic properties including kinematic viscosity (ν), Prandtl number (Pr) and volumetric expansion coefficient (β) which should be evaluated at the liquid temperature $T_{L1}=(T_1+T_{12})/2$. The characteristic dimension is defined as the ratio of the plate area divided by the plate perimeter length according to Equation 16. In solar collector and climate control building envelope applications, the vaporisation process during forward mode occurs primarily as a result of free-surface evaporation because heat fluxes are low and resulting plate-to-liquid temperature differences are not large enough to facilitate nucleate boiling. Equations suited to modelling high heat flux operating scenarios and alternative PLVTD orientations (vertical, tilted, and inverted) are given by Pugsley (2017) but have been omitted here for brevity.

$$Nu = 0.54Ra^{0.25} \quad (\text{Forward mode for } 10^4 \leq Ra < 10^7) \quad \text{Equation 12}$$

$$Nu = 0.15Ra^{1/3} \quad (\text{Forward mode for } 10^7 \leq Ra < 10^{11}) \quad \text{Equation 13}$$

$$Nu = 0.27Ra^{0.25} \quad (\text{Reverse mode for } 10^5 \leq Ra < 10^{11}) \quad \text{Equation 14}$$

$$Ra = \frac{g \beta \Delta T L^3}{\nu^2} Pr \quad \text{Equation 15}$$

$$L = \frac{yz}{2(y+z)} \quad \text{Equation 16}$$

For latent heat transfer caused by vapour mass flow and vapour convection during forward mode operation (relevant to evaluation of Re_c) the Nusselt number can be evaluated using Equation 17 (Cengel, 2006) which is appropriate to rectangular enclosures where the hot surface is at the bottom of the cavity and is valid for Prandtl numbers $0.1 < Pr < 10$ and Rayleigh numbers $Ra < 10^8$. Whilst the Rayleigh number for sensible heat transfer is determined by considering buoyancy as the driving force (Equation 17), for latent heat transfer in forward mode, the Rayleigh number is additionally dependent upon the vapour pressure differential $\Delta P_{12}=|P_1-P_2|$ associated with the plate-to-plate temperature difference $\Delta T_{12}=|T_1-T_2|$. This phase change pressure differential provides the dominant driving force which, in addition to buoyancy force, overcomes the viscous and frictional forces associated with vapour convection. Methods for determining modified Rayleigh (Ra^*) numbers appropriate for evaluating vapour mass transfer thermal resistance in phase change scenarios (both with and without non-condensable gases present) were proposed by Pugsley (2017) drawing on the work of Stein et al. (1985) and Peterson (1996). Equations 18 to 20 can be used

in cases where all non-condensable gases have been removed such that the PLVTD cavity exists at the working fluid saturation pressure (P_{LV}) corresponding to the average temperature $T_{12} = (T_1 + T_2)/2$. The terms h_{LV} , ϑ , $\mathcal{M}$, and $\mathcal{R}$ are the latent heat of phase change, thermal diffusivity, working fluid molar mass, and universal gas constant respectively. All thermodynamic properties in Equations 18 to 20 (k , ν , ϑ , β , ρ) relate to the vapour state which exists at a temperature close to that of the average of the two plates ($T_v \approx T_{12}$). In the case of vapour flowing due to phase changes, the effective thermal conductivity (k of Equation 11) has both sensible vapour (k_v) and latent (k_{Lv}) components which can be quantified using Equation 19 and 20.

$$Nu = 1 + 1.44 \left[1 - \frac{1708}{Ra} \right]^+ + \left[\frac{Ra^{1/3}}{18} - 1 \right]^+ \quad \text{Equation 17}$$

$$Ra^* = \frac{\left[\frac{P_{Lv}}{\rho_L} + g \right] \beta \Delta T_{12} L^3}{\nu^2} Pr \quad \text{Equation 18}$$

$$k = k_v + k_{Lv} \quad \text{Equation 19}$$

$$k_{Lv} = \frac{h_{Lv}^2 P_{Lv} \mathcal{M}^2 \vartheta}{\mathcal{R}^2 T_{12}^3} \quad \text{Equation 20}$$

Heat transfer due to condensation on the underside of the top plate during forward mode operation (relevant to evaluation of R_c) can be evaluated using Equations 21 and 22 which are based on expressions given by Gerstmann & Griffith (1967) and by Stein et al. (1985). The difference in the liquid and vapour densities (ρ_L and ρ_v) plays an important role in both expressions alongside the liquid kinematic viscosity (ν_L), surface tension (σ_L), and thermal conductivity (k_L). Vapour thermodynamic properties and saturation condition thermodynamic properties can be evaluated at the estimated saturation temperature ($T_{Lv} \approx T_{12}$). Liquid condensate thermodynamic properties can be evaluated at the estimated condensate film temperature $T_{L2} = (T_{12} + T_2)/2$.

$$Ra = \frac{g (\rho_L - \rho_v) h_{Lv} \left(\frac{\sigma_L}{g (\rho_L - \rho_v)} \right)^{1.5}}{k_L \nu_L (T_{Lv} - T_2)} \quad \text{Equation 21}$$

$$R_c = \frac{\left(\frac{\sigma_L}{g (\rho_L - \rho_v)} \right)^{0.5}}{A_p k_L 0.69 Ra^{0.2}} \quad \text{Equation 22}$$

3 Experimental apparatus and methodology

The experimental methodology largely followed the precedent set by the work of Boreyko & Chen (2013) but using apparatus with larger plate area dimensions (to better reflect the sizes of relevance to solar collector and climate control building envelope applications) and without special treatments to the evaporator or condenser surfaces (aluminium plates with standard mill finish). Steady state temperatures and heat fluxes were achieved by circulating heated and chilled water through channels located within the body of the evaporator and condenser plates. The specific objectives of the experimental investigation were to quantify the effects of varying the:

- Reverse mode plate-to-plate temperature differential in the range $5 < \Delta T_{12} < 70^{\circ}\text{C}$ corresponding to nominal plate temperatures of $20 < T_2 < 75^{\circ}\text{C}$ and $5 < T_1 < 15^{\circ}\text{C}$.
- Forward mode heat fluxes in the range $100 < q/A < 6000 \text{ W}\cdot\text{m}^{-2}$ corresponding approximately to plate-to-plate temperature differences of $1 < \Delta T_{12} < 10^{\circ}\text{C}$.

Photographs of the PLVTD used for the experimental investigation are shown on Figures 5 and 6. The key components included:

- Spacer frame (A) acted as the sidewalls forming the PLVTD cavity. Several different spacer frames were fabricated using a CNC controlled water-jet cutting machine. Test results presented in this paper were undertaken using an 11mm thick nylon spacer which was found to provide superior reverse mode (insulation) performance owing to its low thermal conductivity.
- Two heat transfer plates (B1 & B2) which acted as the evaporator and condenser, each consisting of a 300x500 mm piece of 12 mm thick aluminium plate which was chosen for its relatively high thermal conductivity at relatively low cost. Each plate was CNC machined to have a parallel flow serpentine water flow channel (4 x 4 mm x 7m long) on one side and a perimeter groove on the other side to accommodate an O-ring vacuum seal. The chosen plate size was primarily determined by the constraints of the CNC machine. The plate surfaces forming the evaporator and condenser were not machined (standard smooth, dull, mill finish) but were cleaned using isopropyl alcohol to remove any grease.
- Two water-side cover plates (B3 & B4) each consisting of a 300x500 mm piece of 6mm thick aluminium plate. Holes were drilled to accommodate vacuum fittings, water supply connections, and clamping bolts.

385 Figure 7 shows a schematic of the PLVTD test apparatus, consisting of the insulated
386 PLVTD (components A, B & C), the measurement datalogger (D), the hot and cold
387 water supply system (E & F) and the vacuum pump system (G). The datalogger
388 monitored flow rates (M), temperatures (T) and cavity pressure (P). The horizontal
389 orientation of the PLVTD assembly (A, B, C) was verified using a digital inclinometer
390 prior to tests.

391 Non-condensable gases were removed from the PLVTD cavity using the vacuum pump
392 before and after injection of the working fluid through the fill point. The working fluid
393 consisted of 0.28L de-ionised water corresponding to approximately 2.5kg/m^2 of
394 evaporator surface area. Cavity pressures were monitored throughout the tests and
395 maintained at <25% above the expected saturation pressure by repeating evacuation
396 (prior to, but not during, each measurement) to remove the small amounts of non-
397 condensable gas which arose from air infiltration and outgassing. Slight loss of working
398 fluid due to repeated evacuations was observed (0.26L remaining after several days
399 of testing) but is not considered significant.

400 During each test, heat was supplied to one of the PLVTD plates from the water heating
401 circuit and removed from the other plate by the chilled water circuit. The two circuits
402 each featured insulated flexible pipework, one oval gear flow meter (M) and two
403 variable speed pumps which were arranged in series in order to overcome the high
404 pressure drop imposed by the water flow channels within each plate. An arrangement
405 of gate valves was used to regulate flow rates and to determine operating mode such
406 that, in forward mode, the evaporator was connected to the heating circuit and the
407 condenser was connected to the chilling circuit. Valve settings were inverted during
408 reverse mode operation to enable the plates to be connected the opposite way around.
409 Additional valves allowed the system to be filled or flushed with water from the mains
410 supply or drained down. Temperatures, flow rates and pressures were continuously
411 recorded throughout the experiments using the datalogger set to have a 1 minute
412 sampling time. Calibration checks indicated that the accuracy of temperature readings
413 provided by the thermocouples was $\pm 1.0^\circ\text{C}$ for temperature readings indicated by any
414 one thermocouple and $\pm 0.3^\circ\text{C}$ for temperature differences derived from readings
415 indicated by a pair of thermocouples. The uncertainty associated with the flow rate
416 readings was found to be $\pm 9\%$ for volume flow rates of water in the range 100 to 1000
417 mL/min at temperatures in the range 5 to 75°C .

418

419

Figure 5 – Annotated photograph of the assembled prototype PLVTD

Figure 6 – Annotated photographs showing main component parts of the prototype PLVTD before assembly

425 The heating and chilling circuit supply temperatures and flow rates determined the
 426 amount of thermal power available for transfer through the PLVTD prototype and hence
 427 also controlled the evaporator and condenser plate temperatures. The amount of heat
 428 supplied (q_{HC}) or removed (q_{CH}) by each plate was determined from Equations 23 and
 429 24 respectively accounting for the inlet-to-outlet temperature differences ($\Delta T_{HC}=T_H-T_C$
 430 when supplying heat or $\Delta T_{CH}=T_C-T_H$ when removing it), the fluid mass flow rates fed
 431 through the plates (M_{HC} and M_{CH} respectively) and the specific heat capacity of that
 432 fluid (c_p) at its average temperature $T_{HC}=(T_H+T_C)/2$. Heat supplied by one plate (q_{HC})
 433 should always be equal to the amount of heat removed by the other plate ($-q_{CH}$) if
 434 ambient losses are properly accounted for. The amount of heat lost (q_l) from a hot
 435 plate to the ambient environment (or gained from ambient by a cold plate) can be
 436 determined according to Equation 25 based on the back-of-plate temperature (T_B), the
 437 edge-of-plate average temperature ($T_D=0.5T_{Dy}+0.5T_{Dz}$) the ambient temperature (T_a)
 438 and the heat loss thermal resistance (R_l). In order to obtain reliable measurement
 439 data, the inlet-to-outlet temperature differences (ΔT_{HC} and ΔT_{CH}) and plate-to-plate
 440 temperature difference (ΔT_{12}) need to be higher than the $\pm 0.3^\circ\text{C}$ uncertainty
 441 associated with such measurements. Temperature differences of 1.2°C would
 442 theoretically have an uncertainty of $\pm 25\%$. The magnitude of the plate-to-plate
 443 temperature difference is dependent upon the heat flux flowing through the PLVTD
 444 ($q_{HC}/A_p \approx -q_{CH}/A_p$) and the overall thermal conductance (U_{fr}) of the PLVTD as described
 445 by Equation 26.

$$446 \quad q_{HC} = M_{HC} c_{p,HC} \Delta T_{HC} - q_{lHC} \quad \text{Equation 23}$$

$$447 \quad q_{CH} = M_{CH} c_{p,CH} \Delta T_{CH} - q_{lCH} \quad \text{Equation 24}$$

$$448 \quad q_l = \frac{0.5 (T_B + T_D) - T_a}{R_l} \quad \text{Equation 25}$$

$$449 \quad \Delta T_{12} = \frac{0.5 (q_{HC} - q_{CH})}{U_{fr} A_p} \quad \text{Equation 26}$$

450 **4 Measurement results and model validation**

452 Figure 8 shows an example temperature, pressure and flow rate time history measured
 453 during one set of forward mode tests. Numbered labels show five condenser plate
 454 temperature setpoints corresponding to approximately $T_2=15, 25, 35, 45$ and 50°C .

455 Steady state temperatures were typically achieved about half an hour after each
 456 setpoint change. Flow rates were maintained broadly constant at
 457 $M_{HC}=M_{CH}=0.65\pm0.1$ L/min throughout the tests. Measured pressures (P_{meas}) increased
 458 with increasing temperature and corresponded broadly to saturation pressure
 459 estimates ($P_{Lv(HCCH)}$) which were evaluated from the average flow and return
 460 temperatures using REFPROP software (NIST, 2013). Figure 9 shows the corresponding
 461 derived thermal powers ($q_{HC} = -q_{CH} = 400\pm30$ W throughout the featured tests) and
 462 thermal conductances which rose from $U_f=300$ W·m⁻²K⁻¹ at the lowest operating
 463 temperature up to $U_f=430$ W·m⁻²K⁻¹ at the highest operating temperature. Figures 10
 464 and 11 present results comparing measured and modelled forward mode performances
 465 at varying condenser temperature and plate-to-plate temperature difference set-
 466 points, including several repeat tests to check repeatability. Measured thermal
 467 conductances varied from $U_f=150$ to 500 W·m⁻²K⁻¹. These have been compared to
 468 corresponding values predicted by the theoretical model which was implemented as a
 469 Microsoft Excel spreadsheet with supporting Visual Basic macros and lookup tables to
 470 evaluate working fluid thermodynamic properties based on REFPROP software outputs.

471
 472 **Figure 8 – Time history of measured temperatures, pressures & flows during forward mode tests with 11mm nylon spacer**
 473
 474

Figure 9 – Time history of derived thermal powers and conductance during forward mode tests with 11mm nylon spacer

Figure 10 – Forward mode performance of PLVTD with 11 mm nylon spacer (varying condenser plate temperature)

Figure 11 – Forward mode performance of PLVTD with 11 mm nylon spacer (varying plate-to-plate temperature difference)

The measurement and modelling results show clearly that forward mode thermal conductance increases with increasing condenser temperature (T_2) and with increasing plate-to-plate temperature difference (ΔT_{12}). Detailed examination of the model suggests that this is primarily due to decreases in evaporation thermal resistance (R_e) and vapour mass flow resistance (R_{ec}) caused by reductions in working fluid viscosity as temperature increases and by increased driving forces as plate-to-plate temperature differences (and hence saturation pressure differences) increase.

Error bars on the graphs indicate estimated combined measurement uncertainties (95% confidence limits) which are primarily associated with temperature and temperature difference measurement uncertainties. Results for cases where plate-to-plate temperature differences exceed $\Delta T_{12} > 4^\circ\text{C}$ have measurement uncertainties of less than $\pm 40\%$ but uncertainty increases at lower plate-to-plate temperature differences. Uncertainties propagate into the modelled results because measured plate temperatures are used as a calculation input parameter. Notwithstanding the declared measurement uncertainties (which have been estimated using a very conservative approach) it is clear that the model predicts the measured forward mode heat transfer behaviour reasonably well. An inherent bias is apparent in the data causing the model to consistently overpredict the heat transfer coefficients by +20% relative to the measured data. There are a number of possible reasons for this bias but the most likely explanation is the presence of non-condensable gasses (air) in the PLVTD which

increases thermal resistances associated with vapour mass transfer (R_{ec}) and condensation (R_c) as described by Stein et al. (1985) and Peterson (1996). This explanation is consistent with the fact that measured pressures (P_{meas}) were in some cases up to 25% higher than the estimated saturation pressures ($P_{Lv(HCCH)}$) as apparent in results towards the right of Figure 8.

Figures 12 and 13 present results comparing measured and modelled reverse mode performances at varying condenser plate temperatures and varying plate-to-plate temperature difference set-points. The measured thermal conductance was consistently around $U_r=10 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ as predicted by the model and largely unaffected by varying plate temperatures or temperature differences. Detailed examination of the model suggests that conduction through PLVTD side walls (the 11mm nylon spacer frame, vacuum seals, and perimeter bolts) is the dominant heat transfer path amounting to $U_w=6.7 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$. Gaseous conduction through water vapour in the cavity amounts to $U_v=1.8 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ and radiant heat transfer amounts to $U_R=1.1 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$.

Figure 12 – Reverse mode performance of PLVTD with 11 mm nylon spacer (varying condenser plate temperature)

Figure 13 – Reverse mode performance of PLVTD with 11 mm nylon spacer (varying plate-to-plate temperature difference)

6 Discussion

The measurement results provide a reasonable validation for the theoretical results despite significant uncertainty associated with measurements of inherently small plate-to-plate and inlet-outlet water circuit temperature differences. The forward mode heat transfer performances in the range $150 < U_f < 500 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ combined with the reverse mode insulation of $U_r = 10 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ corresponds to diodicity of $\zeta \approx 88\%$ at low condenser temperatures and low heat fluxes ($T_2 \approx 15^\circ\text{C}$ and $q/A \approx 120 \text{ W} \cdot \text{m}^{-2}$) and $\zeta \approx 96\%$ at high condenser temperatures and high heat fluxes ($T_2 \approx 60^\circ\text{C}$ and $q/A \approx 2800 \text{ W} \cdot \text{m}^{-2}$). Forward mode performance increases with increasing heat flux and (to a lesser extent) with increasing operating temperature. Reverse mode performance is largely independent of heat flux and temperature within the ranges investigated.

Comparing with the work of Boreyko & Chen (2011 & 2013) and Traipattanakul et al. (2019) suggests that the PLVTD examined in this study achieves similar or lower reverse mode (better thermal insulation) performances but somewhat lower forward mode (worse heat transfer) performances. The theoretical model indicates that the reverse mode performance is largely governed by cavity depth and the influence of thermal bridging elements (sidewalls and supporting structure) hence the findings are unsurprising given that cavity depths are of broadly similar magnitudes. The lower forward mode performance achieved in the present study is partly attributable to heat

548 flux (authors cited above examined behavior at higher heat fluxes) but more
549 importantly due to the simple design of the evaporator and condenser (authors cited
550 above used hydrophobic condenser coatings and hydrophilic evaporator wick coatings
551 to boost performance). Compared to the simple thermosyphonic thermal diode
552 investigated by Wong et al. (2019) the evacuated PLVTD examined in this study
553 achieves better performance in both forward and reverse modes, despite having a
554 smaller cavity depth. Evacuation reduces natural convection in reverse mode due to
555 removal of non-condensable gases and increases latent heat transfer in forward mode
556 because the cavity pressure is similar to the working fluid saturation pressure.

557 A key consideration for solar collector and climate control building envelope
558 applications is the thermal insulation performance (reverse mode) which typically
559 needs to be in the range offered by conventional insulation materials. For reference,
560 high quality double glazing typically achieves $U \approx 1.2 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ and fibre or foam-based
561 insulation products (assuming 50mm thick) achieve $U \approx 0.7 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$. These values are
562 considerably lower (better) than those achieved by the PLVTD examined in this study
563 or by most other authors. Heat transfer performance of around $1000 \text{ W} \cdot \text{m}^{-2} \text{K}^{-1}$ is a
564 reasonable target for PLVTD forward mode performance when integrated in solar
565 thermal collectors (Dupeyrat et al., 2011). Analysis suggests that the overall collection
566 efficiency of a solar water heater would not be significantly affected by a 5-fold increase
567 or decrease to this target (Pugsley, 2017).

568 Using the theoretical model it is possible to examine a variety of PLVTD design options
569 in order to optimise performance for solar collector and climate control building envelope
570 applications. Figure 14 shows how reverse mode performance (U_v) via natural
571 convection of water vapour is dependent upon the cavity depth ($L \approx x$) for nominal
572 assumed operating conditions ($T_1 = 10^\circ\text{C}$ and $T_2 = 60^\circ\text{C}$). Calculations consider plate-to-
573 plate cavity depths in the range $10 \leq x \leq 200 \text{ mm}$. It is very clear that increasing the
574 cavity depth improves the insulation performance. Figures 15 and 16 show the
575 calculated variation of reverse and forward mode performances with temperature for
576 a PLVTD of dimensions $x = 70 \text{ mm}$, $y = 0.7 \text{ m}$ and $z = 1.4 \text{ m}$ constructed of $x_p = z_w = 1 \text{ mm}$
577 thick stainless plates and sidewalls with internal supporting structure formed of
578 stainless steel tubular struts ($d_s = 8 \text{ mm}$ diameter and $d_{sw} = 1 \text{ mm}$ wall thickness) spaced
579 at $d_{ss} = 2d_{sxy} = 2d_{sxz} = 0.07 \text{ m}$ centres. The results demonstrate that such a design should
580 be capable of achieving insulation performances which are close to those desired and
581 diodicity $\zeta > 97\%$. It is interesting to note that the performances shown on Figures 15
582 and 16 are very similar to those shown on Figures 10 and 11, highlighting that forward
583 mode performance of a horizontal PLVTD is largely independent of dimensions.

584 Figure 17 shows that forward mode heat transfer is linearly dependent upon the plate-
 585 to-plate temperature difference below $\Delta T_{12}=3^{\circ}\text{C}$ but follows an exponential
 586 relationship at higher plate temperature differences. The knee point corresponds to
 587 heat fluxes $>1500\text{ W}\cdot\text{m}^{-2}$ which may be of relevance to concentrating solar applications.

588 Pugsley (2016) constructed and tested a hybrid photovoltaic thermal integrated
 589 collector-storage solar water heater incorporating a PLVTD which had dimensions
 590 similar to those examined in Figures 15 to 17. The PLVTD achieved reverse mode
 591 performance of $U_r=1.7\text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ which enabled a satisfactory overall loss coefficient
 592 ($U_{\text{sys}} = 0.85\text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$) to be achieved when the solar absorber was fitted with a
 593 rudimentary transparent plastic cover. The forward mode performance
 594 ($U_f=45\text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$, similar to that achieved by Wong et al., 2019) was lower than
 595 expected owing to problems with dry patches on the evaporator surface. Overall
 596 diodicity ($\zeta=93\%$) was marginally lower than expected.

597

598
 599 **Figure 14 – Dependence of reverse mode water vapour thermal conductance on cavity depth in a horizontal PLVTD**

600

601

Figure 15 – Reverse mode temperature dependence ($\Delta T_{12} = 50^{\circ}C$) for a horizontal $x = 0.07$, $y = 0.7$, $z = 1.4m$ stainless steel PLVTD

Figure 16 – Forward mode temperature dependence ($\Delta T_{12} = 1.5^{\circ}C$) for a horizontal $x = 0.07$, $y = 0.7$, $z = 1.4m$ stainless steel PLVTD

Figure 17 – Forward mode heat-flux dependence ($T_2 = 45^\circ\text{C}$) for a horizontal $x = 0.07$, $y = 0.7$, $z = 1.4\text{m}$ stainless steel PLVTD

7 Conclusions

The heat transfer behaviour of a horizontal planar liquid-vapour thermal diode (PLVTD) has been examined with the aim of supporting engineering for applications in solar thermal collectors and climate control building envelopes. An experimental PLVTD was fabricated and its forward mode heat transfer and reverse mode thermal insulation characteristics were quantified. The experimental device ($A_p = 0.15\text{m}^2$ and $x = 11\text{mm}$) examined in this study had a much larger area and wider cavity than PLVTDs reported elsewhere in the literature ($A_p < 0.01\text{m}^2$ and $1 < x < 7\text{mm}$). The experimental work involved a horizontally oriented PLVTD formed of two parallel isothermal plates with integral serpentine heat exchangers and external insulation. Temperature difference between the plates was controlled by connecting each serpentine heat exchanger to a separate heating-cooling fluid circuit. Plate, fluid, and ambient temperatures were measured to determine heat transfer coefficients under various temperature difference and heat flux operating scenarios. Forward mode heat transfer performances were found to be in the range $150 < U_f < 500 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ and reverse mode insulation was $U_r = 10 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$. Forward mode performance was found to increase with increasing heat flux (in the range $120 < q/A_p < 2800 \text{ W}\cdot\text{m}^{-2}$) and to a lesser extent with increasing operating temperature (condenser plate temperatures in the range $15 < T_2 < 60^\circ\text{C}$). Reverse mode

630 performance appears to be largely independent of heat flux and temperature within
631 the ranges investigated. The measured results correspond to diodicity in the range
632 $88 < \zeta < 96\%$.

633 A one-dimensional lumped parameter heat transfer model has been proposed and
634 successfully validated using the experimental results. Examination of the model shows
635 that behaviour in forward mode is largely governed by the evaporation thermal
636 resistance (R_e) and vapour mass flow resistance (R_{ec}) which are sensitive to working
637 fluid viscosity and plate-to-plate temperature differences. The experimental work was
638 undertaken with non-condensable gases removed as far as possible, although a degree
639 of outgassing appears to have taken place during measurements. The vapour mass
640 flow resistance (R_{ec}) and the condensation thermal resistance (R_c) are sensitive to non-
641 condensable gases resulting the model generally overestimating the measured forward
642 mode performance by about 20%. Reverse mode performance is largely governed by
643 cavity depth which determines the thermal resistance of the sidewalls (R_w), the
644 supporting structural elements (R_s), and the gaseous conduction and convection across
645 the cavity (R_v). The radiative resistance between the two plates (R_r) had only a small
646 effect on the overall behavior of the device tested.

647 Whilst maximisation of forward mode heat transfer has been a priority for researchers
648 examining PLVTDs for electronics cooling applications, minimisation of reverse mode
649 heat transfer (ie improving the insulation performance) is the primary focus for PLVTDs
650 in solar collector and climate control building envelope applications. The theoretical
651 model has been used to calculate the forward and reverse mode performance of a
652 large area ($A_p = 1\text{m}^2$) wide cavity ($x = 70\text{mm}$) PLVTD constructed of 1mm stainless plates
653 and sidewalls with internal supporting structure formed of stainless-steel tubular struts
654 spaces at 70mm centres. The results demonstrate that such a design should be capable
655 of achieving forward mode heat transfer of $150 < U_f < 500 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$, reverse mode heat
656 transfer of $U_r = 2 \text{ W}\cdot\text{m}^{-2}\text{K}^{-1}$ and diodicity of $\zeta > 97\%$, which would be suitable for solar
657 collector and climate control building envelope applications.

658 The work presented in this paper is limited to simple horizontally oriented PLVTDs
659 where evaporator wetting and condensate return is driven by gravity. The authors
660 have subsequently examined the behaviour of modified PLVTDs in vertical and tilted
661 orientations. Experimental results for vertically oriented PLVTDs together with a
662 modified version of the lumped parameter model (to account for orientation) will be
663 presented in a future publication.

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672 **9 Nomenclature**

673 *Latin symbols*

674	A	Surface area [m^2]
675	c_p	Specific heat capacity at constant pressure [$\text{J}\cdot\text{kg}^{-1}\text{K}^{-1}$]
676	d	Distance [m]
677	g	Acceleration due to gravity ($g=9.81\text{m}\cdot\text{s}^{-2}$ on earth at sea level)
678	h_{LV}	Latent heat of liquid-vapour phase change [$\text{J}\cdot\text{kg}^{-1}$]
679	k	Thermal conductivity [$\text{W}\cdot\text{m}^{-1}\text{K}^{-1}$]
680	L	Characteristic dimension of Nusselt number [m]
681	M	Mass flow rate [$\text{kg}\cdot\text{s}^{-1}$]
682	N	Number of struts in structural support array
683	Nu	Nusselt number
684	P	Pressure [Pa]
685	Pr	Prandtl number
686	q	Thermal power [W]
687	Ra	Rayleigh number
688	T	Temperature [$^{\circ}\text{C}$] except for Equation 10 which uses absolute [K]
689	U	Thermal conductance or heat transfer coefficient [$\text{W}\cdot\text{m}^{-2}\text{K}^{-1}$]
690	ν	Kinematic viscosity [$\text{m}^2\cdot\text{s}^{-1}$]
691	x	Distance along an axis which is parallel to the PLVTD depth [m]
692	y	Distance along horizontal axis perpendicular to PLVTD depth [m]
693	z	Distance along an axis which is perpendicular to x and y axes [m]

694 *Greek and other symbols*

695	$\mathcal{M}$	Molecular weight [$\text{g}\cdot\text{mol}^{-1}$]
696	$\mathcal{R}$	Universal gas constant ($8.314\text{J}\cdot\text{mol}^{-1}\text{K}^{-1}$)
697	ϑ	Thermal diffusivity [$\text{m}^2\cdot\text{s}^{-1}$]

698	ΔP	Pressure difference [Pa]
699	ΔT	Temperature difference [$^{\circ}\text{C}$]
700	ς	Diodicity [%]
701	σ	Surface tension [$\text{N}\cdot\text{m}^{-1}$]
702	β	Coefficient of volumetric expansion [K^{-1}]
703	ρ	Density [$\text{kg}\cdot\text{m}^{-3}$]
704	χ	Stefan Boltzmann constant ($5.67 \times 10^{-8} \text{ W}\cdot\text{m}^{-2}\text{K}^{-4}$)
705	Subscripts	
706	1	Plate 1 which is the evaporator in forward mode
707	2	Plate 2 which is the condenser on forward mode
708	12	Between (or average of) the two plates
709	a	Ambient environment
710	B	Back of plate
711	c	Condenser, condensate or condensation
712	C	Cold water feed or cooled return
713	CH	Cooling plate or cooling water circuit
714	D	Edge of plate
715	e	Evaporator, evaporation or evaporator liquid film
716	ec	Between (or average of) the evaporator and condenser surfaces
717	f	Forward mode
718	H	Hot water feed or heated return
719	HC	Heating plate of heated water circuit
720	L	Working fluid liquid state property
721	l	Losses to ambient environment
722	L1	Liquid on the evaporator
723	L2	Liquid on the condenser
724	Lv	Latent property of working fluid at the liquid-vapour saturation point
725	LvL	Liquid-vapour-liquid transition
726	p	Plate
727	r	Reverse mode
728	R	Radiative component
729	s	Structural element(s) such as internal support struts
730	ss	Between two adjacent struts (centre-to-centre distance)
731	sw	Strut tube wall (thickness)
732	sxy	Between the xy-sidewall and the closest strut (distance along the z-axis)

733	szz	Between the xz-sidewall and the closest strut (distance along the y-axis)
734	v	Working fluid vapour state property
735	w	Sidewalls of the PLVTD
736	x	In the direction parallel to the PLVTD depth
737	y	In the direction of the horizontal axis perpendicular to PLVTD depth
738	z	In the direction of the axis which is perpendicular to x and y axes
739		

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