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### **Experimental investigation of a novel vertical loop-heat-pipe PV/T** heat and power system under different height differences 3 2 a contract to the contract of the contract o 4

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 $\frac{31}{32}$  **Abstract:** For a novel vertical solar loop-heat-pipe photovoltaic/thermal system, the <sup>33</sup> height difference between evaporator and condenser plays an important role in the heat <sup>35</sup> transport capacity, which has significant impact on the solar thermal efficiency and parametrical optimization of this system. Therefore, based on the results derived from 37 a prototype of this novel system was designed, constructed, and an experimental 40 <sup>42</sup> investigation under different height difference was undertaken to study the impact of  $\frac{44}{15}$  height difference on the system performance. It was found that the relationship between the solar thermal efficiency of this vertical system and the height difference is nonlinear. 46 In present study, the optimal height difference is around 1.1m, which was selected as 48 an optimal value for the following experimental investigations, and below 1.1m, the PV module surface temperature decreased with the increase of the height difference. 51 Furthermore, the transient solar thermal and electrical performance of this system with<br>
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<sup>58</sup> Corresponding authors. 32 **ADSTRICT** FOR a HOVER VER 34 hotgit difference between  $36 \t\t 1 \t\t 1 \t\t 3$ 38 39 the authors' previous ana 41 **prototype of this novel** s 43 mycsugation under unicicl  $45$ 47 49 50 an optimal value for the foll 52 module surface temperature 54 **Humermore**, the transient

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the selected optimal height difference was investigated under outdoor real weather  $\frac{1}{2}$  condition. These results of this experimentation can help optimize the system  $\frac{3}{4}$  construction and thus help to develop the high thermal performance and low-cost solar  $PV/T$  system for space heating and power generation. 2 condition. These results of  $4 \qquad \qquad \text{constant}$  and thus help to 

Keywords: Micro-channel; Loop heat pipe; PV/T; Height difference; Efficiency **Keywords:** Micro-channel;

#### $\frac{12}{13}$  Nomenclature  $\frac{14}{15}$  Parameters 13 Nomenciature  $\blacksquare$   $\blacksquare$   $\blacksquare$   $\blacksquare$





#### 43 **1. Introduction** 44

It is reported by the International Energy Agency (IEA) that during 2017-2023, 46 renewables will meet more than 70% of global electricity generation growth. The growth of the solar photovoltaic (PV) is larger than all other renewables in combination, 49  $\frac{51}{52}$  while China remains the absolute solar PV leader by far, holding almost 40% of global  $^{53}$  installed PV capacity in 2023[1]. However, it has been proved that the increment by 1<sup>o</sup>C solar electrical efficiency [2][3] , which is in the range of 4%-20%[4] . In order to 57 47 48 renewables will meet mor 50 growth of the solar photovo 52 WHILE CHINA TEMPLE BUT AV  $54$  mstanca r v capacity in  $202$ 55 of the PV cells' temperature would result in around 0.25% to 0.5% linear drop of the 56 58 59 control the PV cells' temperature, methods were utilized to take away the accumulated 60

heat of the solar PV modules and then to make full use of the removed heat, which is  $\frac{1}{2}$  known as solar photovoltaic/thermal (PV/T) technology[5], making advanced  $\frac{3}{4}$  utilisation of absorbed solar energy as both electricity and heat by one module.  $\frac{5}{6}$  According to different heat transfer mediums, the PV/T systems can be divided into air-<sup>7</sup> based PV/T systems [6][7] where the PV cells were cooled by naturally or mechanically ventilated air; water-based PV/T systems [8][9] cooled by circulating water across the 9 back-side coils or tubes; and heat pipe / loop heat pipe-based PV/T systems cooled by  $\frac{12}{13}$  applying (loop) heat pipe through the working fluid evaporation and condensation to  $\frac{14}{15}$  achieve heat removal and recovery of the PV/T system [10]. 2 known as solar photovol  $4 \quad 4 \quad 4$ 6 Recording to different from the 8 and 2 <br>8 and 2 10 11 back-side coils or tubes; an 13 applying (loop) heat pipe t 15 **a**chieve heat reflioval and r

 $\frac{17}{18}$  During recent decades, the application of Loop Heat Pipe (LHP)/Heat Pipe (HP) for PV/T systems has obtained wide attention owing to its high heat transfer capability, no 19 working fluid leakage risk, and no frozen problems in winter. Zhang et al. [11] 21 introduced a novel solar photovoltaic/loop-heat-pipe (PV/LHP) heat pump system for 23  $\frac{24}{25}$  hot water generation, and experimentally investigated the performance of the system,  $\frac{26}{27}$  indicating that the electrical, thermal and overall efficiency of the PV/LHP module were around 10%, 40% and 50% respectively under the given experimental conditions. 28 Zhang et al.[12]also investigated the dynamic performance of the novel PV/LHP heat 30 pump system for potential use in space heating or hot water generation, applying 32 theoretical computer simulation and experimental validation to analyse and make comparison. He et al. [13] proposed the novel heat pump assisted solar façade loop-35  $\frac{37}{38}$  heat-pipe water heating system and studied its operational performance by applying both theoretical and experimental methods, indicating that the average thermal 39 18 **During recent decides**, the 20 22 25 hot water generation, and e 27 multaling that the credit real 29 arcticle 10%, 10% and 50  $31$  c  $\sim$  c  $\sim$  1 33 34 **theoretical computer simu**  $36$  comparison. He et al. [13] 38 mat-pipe water heating sy 40 commencement and express efficiency of the LHP module was around 71% with the heat pump's assistance.<br>42 42

However, almost all existing loop heat pipes integrated into the PV/T system were 44 designed and constructed with round tubes and plate fins (RTPF) with linear contact  $\frac{47}{48}$  area in touching on the PV model. This results in relative higher heat transfer resistance. Therefore, micro channel loop heat pipe (MCLHP) applying flat Al-flat tube, which  $\frac{51}{20}$  can touch well with the PV panel with surface contact is utilized. This MCLHP is one 53 kind of two phase (evaporation/condensation) loop heat pipe heat transport device, comprising the micro-channel array integrated evaporator, condenser, separate vapour 55 and liquid transportation lines, and compensation chamber[14], was recently applied to cool PV cells and make use of the solar thermal energy [15][16].This kind of micro- 58 45 46 designed and constructed v 48 area in touching on the P v i 50 THULOUL, HILLOURING  $52$ 54 56 57 and liquid transportation lin 59 **COOL PV CELLS and make us** 

channel loop heat pipe can greatly enhance the heat transfer ability with smaller cross- $\frac{1}{2}$  section area, consume no power for refrigerant compressors through longer distance,  $\frac{3}{4}$  and has more compact structure compared to heat pipe with round tubes and plate  $\frac{5}{6}$  fins[17]. Furthermore, owing to its flat-plate surface, the micro-channel array has an <sup>7</sup> advantage of easy binding with the PV model. Within these limited researches/studies, the authors have previously [15][16][18] developed the novel solar micro-channel loop 9 heat pipe PV/T system filled with refrigerant R134a, which has been investigated  $\frac{12}{13}$  through computer modelling and theoretical analysis from different perspectives. Both the separate multiple-micro-channel evaporator and the condenser of this LHP-PV/T <sup>16</sup> system are laid vertically, which make the refrigerant gravity inside the loop to be the driving force to the downward fluid flow. In that case, a higher HD will cause bigger 18 thermal resistance, while a lower HD could result in lower gravity. The height 20 difference between the evaporator and the condenser will therefore play an important  $\frac{23}{24}$  role in the performance of this system. Zhao et al. [19] designed and theoretically <sup>25</sup> studied a novel LHP solar heating system, indicating that variation in the height  $\frac{27}{20}$  difference between the evaporator and the condenser would cause different system performance. In regards with the LHP solar system and the weather condition in Beijing, 29 they found when height difference is larger than 0.4m, the system would have enough 31 heat transfer capacity for the absorbed heat. At an early stage, Zhang et al. [20]design 32  $\frac{34}{35}$  a LHP based solar thermal façade heat pump system mentioning that HD results in the 36<br>37 separative effect to control the liquid feeding speed and thus to balance the liquid evaporation and delivery. At the same time, they [21] also investigated a solar 38 photovoltaic/loop-heat-pipe (PV/LHP) heat pump system with different height 40 difference from the evaporator to the condenser and found that the capillary limit, 42 governing the heat transfer capacity when the heat difference was lower than 1.1m, 43  $\frac{45}{46}$  raised with the height difference, and a linear relationship between the height difference  $\frac{47}{48}$  and the capillary limit was also proved to be existing because of the effect of gravity, which is directly influenced by height difference. However, some researchers ignored 49 51 the impact of height difference and supposed that assuming the evaporator outlet temperature equals to the condenser inlet temperature is enough[22][23]. 2 section area, consume no po  $4 \qquad \qquad \text{and has more complex sum}$  $6 \text{ m}$   $\text{m}$ <sup>1</sup>,  $\text{m}$   $\text{m}$   $\text{m}$  8 10 11 heat pipe PV/T system fil 13 **Infough computer modellin** 15 and separate mump content  $17$  by seem are rare vertically, 19 21 and the contract of the con 22 difference between the eva 24 role in the performance of 26 **Studied a Hover ETH** Sola 28 **Experience of the contract of the second se** 30 33 heat transfer capacity for the 35 a LHP based solar inermal 37 giavity criter to control  $39$ 41 44 governing the heat transfer 46 raised with the height drift. 48 and the capital y mint was 50 52 53 **temperature equals to the c** 

In terms of this key point, this paper experimentally investigated the impact of the 55  $\frac{57}{58}$  height difference between the evaporator and the condenser on the solar thermal 59<br>
so **performance of this novel vertical LHP-PV/T system, aiming at finding an optimal** 56 In terms of this key point 58 height difference between 60 Performance of this nover

value of the height difference to optimize the system construction and thus helping to  $\frac{1}{2}$  develop an impact, easy installed, high thermal performance and low-cost solar PV/T  $\frac{3}{4}$  system for space heating and power generation. 2 develop an impact, easy inst 4 System for space freating and

#### $\frac{6}{7}$  2. Description of this novel vertical LHP-PV/T system 7

 $\frac{8}{3}$  Based on the results derived from the analytical investigation and computation studies 10 presented in the authors' previous papers[15][16], this novel vertical LHP-PV/T system was designed as schematically shown in **Figure 1**. This system comprises: (1) multiple micro-channel array integrated into PV/evaporator; (2) co-axial triple-pipe heat 13 <sup>15</sup> exchanger applying PCM to be the condenser; (3) separate vapour and liquid  $\frac{17}{18}$  transportation lines; (4) PV module covered with glass without air gap, which is pressed on the aluminium (Al) plate, and as a result, reducing the thermal resistance between 19 the PV cells and the Al plate compared with the conventional PV panel with thermal 21 grease. Meanwhile, this novel system has three crucial innovations: (1) The upper liquid 23  $\frac{24}{25}$  feeding pipe working as the chamber of a LHP and also in parallel, a vapour collecting  $\frac{26}{27}$  pipe integrated on the top of the micro-channel evaporator, which can effectively gather <sup>28</sup> the vapour from the micro-channels as shown in **Figure 1(a)**, and importantly, can separate the vapour and liquid to reduce the resistance from the rising vapour to the 30 falling liquid; (2) Inside the upper liquid feeding tube, four tiny holes are opened on the 32 side wall of each micro-channel as shown in **Figure 1(c)(d)**, which can distribute the condensing liquid across the inner micro-channel walls and achieve continuous liquid 35  $\frac{37}{38}$  film on the rough/wicked side walls of micro-channels, thus improving the heat transfer ability which has been validated by the investigation on heat transfer limits; (3) Triple-39 pipe heat exchanger with Phase Change Materials (PCM) as shown in **Figure 1(b)**,<br>
43 which is designed to condense the evaporated fluid coming from the micro-channel evaporator and store the excess heat by melting the PCM inside the outer tube. 9 11 12 was designed as schematical 14 micro-channel array integ 16 **Exchanger** applying FCIVI  $18$  and  $18$  $20 \times 71$ 22 25 **Teeding pipe working as the** 27 pipe integrated on the top of 29 and the rapport from the finere  $\frac{1}{1}$ 33 34 side wall of each micro-ch 36 **condensing liquid across traction** 38 http://www.min.org/witner.org/  $40$ <sup>41</sup> pipe heat exchanger with Phase Change Materials (PCM) as shown in Figure 1(b), <sup>42</sup> 42 44 45 evaporator and store the ex



 $\frac{29}{30}$  Figure 1. The novel solar LHP-PV/T heating and power system rigure 1. The no

When the PV/evaporator receives the solar radiation striking on its upper surface, most of the solar radiation is absorbed by the PV/T panel, part of which is converted into electricity by PV cells to supply power and the rest is finally transformed into thermal energy. The thermal energy is absorbed by the Al plate and then transferred to the <sup>40</sup> micro-channels, thus evaporating the working fluid (Refrigerant R-134a) inside the <sup>42</sup> channels. The evaporated fluid floating from the micro-channel array is collected in the upper vapour header and then transported towards into the heat exchanger (condenser) through the single vapour transportation line. In the heat exchanger, the latent heat from the vapour condensing in the central pipe is transferred to the water flowing inside the  $\frac{49}{50}$  middle annulus tube, then the hot water will be circulated for space heating etc. At the **Figure 1. The novel solar LHP-PV/T heating and power system**<br>When the PV/evaporator receives the solar radiation striking on its upper surface, most<br>of the solar radiation is absorbed by the PV/T panel, part of which is c <sup>53</sup> tube to storage excess thermal energy and also to keep the water temperature within a certain temperature range. As a result, the condensed fluid from heat exchanger, driven by gravity, flows directly into the upper liquid feeding header through the single liquid transportation line, and then penetrates the micro-channel wall via the tiny holes to 37 electricity by PV cells to su 39 energy. The thermal energy 41 micro-channels, thus evap 43 Channels: The evaporated H 48 the vapour condensing in the 50 middle annuius tube, then t 51 same time the heated wate 52 saint time, the heated water 54 and two states of the s **transportation line, and the** 

formulate the continuous downward liquid streams (films) among the porous wick to  $\frac{1}{2}$  keep the side wall of channels at the wet condition, which can thus offer timely  $\frac{3}{4}$  replenishment of the evaporating fluid for absorbing heat from the solar energy. All these processes formulate the cycle loop as illustrated in Figure 2. 2 Reep the side wall of chan 4 reprefirming to the evapor 6 and the processes remained in



Figure 2. Cycle loop of the novel solar LHP-PV/T heating system

#### $\frac{34}{35}$  2. Experimental setup and procedure 2. Experimental setup at

#### $\frac{36}{27}$  3.1 Experimental set up

<sup>39</sup> The integrated prototype of this vertical LHP-PV/T heating and power system was built in an indoor space (Lab. in Guangdong University of Technology, China) as presented in Figure 3, and equipped with various measurement instruments and sensors, including pyranometer, rotameter, thermocouples/temperature probes, and pressure <sup>46</sup><br><sup>47</sup> sensors, which were installed at appropriate positions to measure solar radiation, flow <sup>48</sup> rate, temperature and pressure, respectively. In addition, the Solar Power Controller (MPPT) was used to be both the electrical recorder and converter. All the testing outputs were transmitted into relevant data logger and temperature recorder, respectively, and then gathered to the computer. The parametrical specifications of this system are listed in Table 1. **Including pyranometer**, ro 47 sensors, which were install 49 race, temperature and press 51 (Alexandre Press, 1988)  $\qquad \qquad$   $\qquad \qquad$   $\qquad$   $\qquad$ 

<b>Parameters</b>	Value
Micro-channel port width	$0.0017 \text{ m}$
Micro-channel port height	0.001m
Evaporator length	1.9 <sub>m</sub>
Number of micro-channel flat heat pipes	20
Number of micro-channel ports	10
Operating temperature range	$20-60$ °C
Transportation line outer diameter	$0.0174$ m
Transportation line inner diameter	$0.015 \text{ m}$
Liquid head length	1 <sub>m</sub>
Liquid Head diameter	$0.022 \text{ m}$
Vapour header length	1 <sub>m</sub>
Hole diameter	$0.00075$ m
Transportation line length	$1.5/1.5$ m
Heat exchanger central tube total length	5 <sub>m</sub>
Heat exchanger central tube diameters	$0.016/0.017$ m
Heat exchanger middle tube diameters	$0.019/0.021$ m
PCM tube diameters	$0.027/0.029$ m
PCM melting temperature	44 °C
PCM density	$800 \text{ kg/m}^3$
PCM Latent Heat	$242$ kJ/kg
PCM thermal conductivity	$0.18 W/(m \cdot K)$
PCM Maximum operating temperature	300 °C

Table 1. Parametrical specifications of the vertical LHP-PV/T system

 



<sup>24</sup> Figure 3. Construction of the novel solar LHP-PV/T heating system<br><sup>25</sup> **a b b c c c c c c c c** 

## 28 3.2 Experimental process

 $\frac{29}{30}$  In order to study the impact of height difference between the PV/evaporator and the  $\frac{31}{32}$  condenser on the performance of this novel solar LHP-PV/T system, based on the 33 authors' previous analytical computation investigation of heat transfer limits under different height differences[15], which illustrated that from 0.1m to 1m, a larger height difference between the evaporator and the condenser leads to a higher capillary limit and thus a higher heat transfer capacity. It is found that the heat transfer limit of HD at <sup>40</sup> 1.0m is larger than that of 0.6, the difference between 0.6 and 1.0 is larger than that  $\frac{42}{43}$  between 0.3 and 0.6, therefore, for present experimental study, 0.7 is set as start-up, and **Figure 3. Construction of the novel solar LHP-PV/T heating system**<br>3.2 Experimental process<br>In order to study the impact of height difference between the PV/cvaporator and the<br>condenser on the performance of this novel s m) were carried out under relatively steady conditions to determine the optimal value. The measurement positions are presented in Figure 4, where the average solar radiation is maintained as  $565\pm10$  W and the ambient temperature is around 30 °C. The steady state defined in this experiment is that the thermal properties of the MCLHP stay the condenser on the performance of this novel solar LHP-PV/T system, based on the authors' previous analytical computation investigation of heat transfer limits under different height differences [15], which illustrated that **In order to study the impart CONGERISCE ON THE PETROLLE** 34 additional provision and given 36 and the contract of the con 38 and the contract of the con 39 and thus a higher heat trans 41 1.0m is larger than that of Detween 0.9 and 0.0, there is four different height differe **Four different neight different**  is maintained as  $565\pm10$  W 52 state defined in this experi-53 same against the operating 54 same against the operating

Each lab test started at 15:30 pm and run for sufficient hours to obtain the relative steady state data. After the optimal height difference confirmed, this experimental system was 

moved to outdoor to test under the real weather from 9:30 am to 16:30 pm. During all  $\frac{1}{2}$  the tests, the measurement data was recorded at 30s interval and logged into the  $\frac{3}{4}$  computer to enable the follow-on analysis to be completed. 2 the tests, the measurement  $4 \qquad \qquad \text{conjugate the ratio}$ 

The analyses of the experimental data need to consider that the potential equipment uncertainties, e.g., the tolerance of the solar radiation (±10W) and the temperature measurement discrepancy  $(\pm 0.1^{\circ}C)$ . Additionally, there are some potential random <sup>12</sup> errors, which may be caused by personal fluctuation, random electronic data logger  $\frac{14}{15}$  fluctuation and influences of friction inside cooling pipes due to impure cooling water, etc. 11 measurement discrepancy 13 errors, which may be caus 15 maxwallon and implements etc **and contract the contract of the contrac** 



P1 - Pressure of the LHP; T5, T6, T7 - Surface temperature of PV panel; T8, T9, T10 -<br>Surface temperature of Al plate; T11, T12, T13 -Temperature of micro-channel array 54 P1 - Pressure of the LHP; T5, T6, T7 - Surface temperature of PV panel; T8, T9, T10 - **Surface temperature** 

Figure 4. Experimental setup

## 3.3 Error analysis

Based on the error propagation theory, the experimental error of the independent  $\frac{4}{5}$  variables, such as temperature, output power and solar radiation, is determined by the  $\frac{6}{7}$  accuracy of the specific measurement instrument; while the error of dependent variables,  $\frac{8}{3}$  such as solar thermal and electrical efficiencies, can be calculated based on the error of the independent variables. Thus, for a given dependent variable y: 3 Based on the error propage variables, such as temperature 7 accuracy of the specific filed. 

13 
$$
y = f(x_1, x_2, \dots, x_n)
$$
 (1)

16 The relative error (RE) can be calculated by [24][25]:

$$
RE = \frac{dy}{y} = \frac{\partial f}{\partial x_1} \frac{dx_1}{y} + \frac{\partial f}{\partial x_2} \frac{dx_2}{y} + \dots + \frac{\partial f}{\partial x_n} \frac{dx_n}{y} , \qquad (2)
$$

where  $x_i$  ( $i = 1, 2, ..., n$ ) is the variable of the dependent variable y; and  $\partial f / \partial x_i$  is the  $\frac{24}{25}$  error transfer coefficient of the variables. 23 where  $x_l$   $(\ell - 1, 2, ..., n)$  is 25 CHO CHANGE COMMENT OF

<sup>27</sup> The experimental relative mean error (RME) during the test period can be expressed as: 28 and  $\frac{1}{2}$ 

$$
RME = \frac{\sum_{1}^{N} |RE|}{N}
$$
 (3)

Based on the Eqs. (1) to (3), the REMs of all the variables were calculated and the 34  $\frac{35}{36}$  results were listed in **Table 2.** All the experimental relative mean errors are in the <sup>37</sup> acceptable range. 34 Based on the Eqs. (1) to ( 36 results were listed in **Tab** 38 acceptable range.

42						
43 44	Variable	T (%)	G(%)	$\eta_e$ (%)	$\eta_{th}$ (%)	
45 46	<b>RME</b>	0.067	2.0	4.2	21.4	

Table 2. The experimental RMEs of the variables

#### 4 Theoretical equations

#### 4.1 Thermal resistance

Based on the theoretical model established by the authors in previous works [26,27], it is assumed that there would be no existing temperature gradient along the length direction of the Micro-channel heat pipe evaporation section, owing to the even heat **Is assumed that there would airection** of the Micro-char

input. Then the total useful solar heat transferred into the water in the middle annulus tube of the co-axil triple-tube heat exchanger from absorbed Al plate can be calculated  $\frac{3}{4}$  according to the Hottel-Whillier model[26][28][29]: 2 tube of the co-axil triple-tube  $4 \quad \text{acoding to the notice-PWIII.}$ 

$$
Q_u = LWF_{th}[q_{abs} - K_L(T_w - T_a) - q_e]
$$
 (4)

<sup>10</sup> Where, *L* and *W* are the length and width of the Al plate (m);  $q_{abs}$  is the absorbed heat per unit (W/m<sup>2</sup>);  $q_e$  is the solar energy converted into electricity;  $T_w$  is the mean 13<br>14 **temperature of water in the middle annulus tube of the co-axil triple-pipe heat** <sup>15</sup> exchanger;  $F_{th}$  the thermal efficiency factor of the PV/T system which, representing  $\frac{17}{18}$  the ratio of the actual useful heat gain by the system to the overall converted solar heat at a certain working fluid temperature, can be expressed as: 19 11 12 heat per unit  $(W/m^2)$ ;  $q_e$  is 14 temperature of water in the  $16$  exenanger,  $r_{th}$  we well all 18 and the ratio of the actual useful 20 main a communication of the contract of the

23  
\n
$$
F_{th} = \frac{K_L^{-1}}{N_{MCT}} \left( \frac{1}{1 + K_L \left[ \left( \frac{W}{N_{MCT}} - D_{MCT} \right) F_f + \frac{D_{MCT}}{1 + T_{p-41} K_L} \right]} + \sum_{i=1}^{6} R_i \right)}
$$
\n(5)

Where,  $N_{MCT}$  is the number of the micro-channel flat tube;  $\sum R_i$  is the overall thermal resistance of from the absorbed Al plate to the working fluid. 31 29 30 Where,  $N_{MCT}$  is the number 32 resistance of from the absor

<sup>35</sup> This thermal efficiency factor of this system which, is a constant figure under the fixed physical and operating condition, takes consideration of all the thermal resistances from 37 the absorbed Al plate to the water in the middle annulus tube of the co-axil triple-pipe 39 heat exchanger, including: (1) thermal resistances between the PV cells and absorbed 40 <sup>42</sup><br>Al plate  $(R_1 = R_{p-Al})$ ; (2) thermal resistance of the micro-channel heat pipe wall  $(R_2 =$  $R_{MC}$  (3) equivalent radial thermal resistance of the flow  $(R_3 = R_{ea,f})$ , which is <sup>46</sup> composed of the resistance of the liquid film from holes on the side of the channel  $R_{lf,c}$  $\frac{48}{49}$  assumed parallel to the radial thermal resistance of the two phase flow on the other side <sup>50</sup> of the channel  $R_{ea,v}$ ; (4) the resistance of the axial vapour flow  $(R_4 = R_{v,a})$ ; (5) the resistance of the condensation two phase flow  $(R_5 = R_{tn,c})$ ; (6) the thermal resistance <sup>54</sup> of the central tube wall of the co-axial triple-pipe heat exchanger  $(R_6 = R_{HE1})$ . These resistances will be analysed in the following sections. And before analysing, it should 56 be noticed that the resistance of the silicon sealant and the heat capacity of the adhesive 58 36 million continues the control of the set of  $38 \qquad \qquad 1 \qquad \qquad 1 \qquad \qquad 2$ 41 heat exchanger, including: 43 Al plate  $(K_1 = K_{p-Al})$ ; (2) t  $A_5$   $R_{MC}$ ,  $B_5$  equivalent radia 47 **Composed of the resistance** 49 assumed parametro the radi  $51$  of the enannel  $\mathfrak{g}_{q,v}$ ,  $(\neg)$  u 53 resistance of the condensat 55 of the central table want of 57

65

layer are neglected, resulting from their significantly smaller values compared to other  $\overline{2}$  factors [22]. 

 $\frac{5}{6}$  Among these thermal resistances, the equivalent radial thermal resistance of the flow is composed of the resistance of the liquid film from the holes on one side of the micro-channel wall, which is closely related to the height difference between the evaporator and the condenser, and the resistance of the two-phase flow flowing along the other two sides of the channel port, where these two type of thermal resistance are  $\frac{14}{15}$  assumed to be parallel. Thus the equivalent radial thermal resistance of the flow can 16 be expressed as: **Annual** Superintendent Processes 11 evaporator and the condens 13 and the other two sides of the c. assumed to be parameter. 

$$
R_{eq,f} = \left(\frac{1}{R_{lf}} + \frac{1}{R_{tp,f}}\right) \tag{6}
$$

<sup>24</sup> Where,  $R_{1f}$  is the total thermal resistance of the condensate liquid film with thickness 26<br>27 of  $\delta_{lf}$  at the two opposite micro-channel walls with holes, which can be calculated by: where,  $\pi_{lf}$  is the total then or  $o_{lf}$  at the two opposite i  $\mathbf{U}$ y.

$$
R_{lf} = \frac{\delta_{lf}}{\lambda_{lf}(2a \times L_{MC})N_{MCT}N_{ch}} \tag{7}
$$

 $\frac{36}{37}$  And  $R_{tn,f}$  is the thermal resistance of the two-phase flow flowing on the other two sides wall of the micro-channel expressed as: And  $R_{tp,f}$  is the thermal res 39 sides wall of the micro-cha

$$
R_{tp,f} = \frac{1}{2b\varepsilon_{of}L_{MC}h_{tp}N_{MCT}N_{ch}}
$$
(8)

47 Where,  $h_{tp}$  calculated by considering the overall fin efficiency  $\varepsilon_{of}$  in the microchannel heat pipe is the overall heat transfer coefficient of the two-phase flow in the  $\epsilon_{51}$  micro-channel. The  $h_{tp}$  and  $\varepsilon_{of}$  can be obtained from the authors' previous paper[26].  $\delta_{15}$   $\delta_{16}$  is the liquid film thickness which, assumed the same along the adjacent wall, is <sup>54</sup><br>55<br>55  $^{56}$  where the height difference is denoted by  $H_{ce}$ . Based on this height difference, the condensed liquid penetrate the tiny holes on the side wall of the micro-channels to formulate this liquid film, which can be approximated as **channel heat pipe is the ove**  micro-channel. The  $n_{tp}$  and  $\sigma_{lf}$  is the figure film thickn 55 Imked to the pressure different 57 Where the height difference **Concernsive Institute** Personality 

$$
\text{When } Re \le 400, \quad \delta_{lf} = \left(\frac{3\mu_l^2}{\rho_1^2 g}\right)^{1/3} Re^{1/3} \text{ ;}
$$

$$
\delta_{6}
$$
 When  $Re > 400$ ,  $\delta_{lf} = 0.369 \left( \frac{3\mu_l^2}{\rho_l^2 g} \right)^{1/2} Re^{1/2}$ 

When  $Re > 400$ ,  $\delta_{lf} = 0.369 \left( \frac{3\mu_l}{\rho_l^2 g} \right)$   $Re^{1/2}$ <br>
<sup>9</sup><br>
<sup>9</sup><br>
Where  $\mu_l$  is the liquid dynamic viscosity  $(Pa \cdot s)$ ;  $\rho_l$  is the liquid density (kg/m<sup>3</sup>);  $Re$  $\frac{11}{12}$  is the Reynolds number given by Where  $\mu_l$  is the liquid dyna **IS THE REVIOLAS HUILDEL BIV** 

$$
Re = \frac{\rho u_{lf}}{\mu_1} \delta_{lf} \tag{9}
$$

 $u_{1f}$  is the superficial velocity of the liquid film flow expressed as following, which  $\frac{21}{22}$  depends on the pressure difference of the loop and is assumed to be the same for each  $\frac{23}{24}$  tiny hole opened on the side wall of the micro-channel.  $u_{\text{if}}$  is the superficial veloci 22 acpends on the pressure and 24 <sup>any non-epence on the state</sup>

$$
u_1 = A_h \frac{(2g_{ev} - c_0)^{1/2}}{A_{lf}(1 - C_d^2(d_h/d_{lh}))}
$$
(10)

 $W_{32}$  Where,  $A_h$  is the cross-sectional area of the tiny hole;  $H_{ev-co}$  is the height difference  $\frac{33}{34}$  between the top of the PV/micro-channel evaporator and the condenser working as the  $\frac{35}{26}$  driving pressure head;  $d_h$  and  $d_{lh}$  are the diameter of the tiny hole and the liquid  $\frac{37}{10}$  header respectively;  $A_{1f}$  is the liquid film cross-sectional area expressed as: where,  $A_h$  is the cross-sect 34 between the top of the 1 y/i  $\frac{1}{26}$   $\frac{1}{26}$ 

$$
A_{lf} = \delta_{lf} a \tag{11}
$$

 $\frac{44}{45}$  And  $C_d$  is the discharge coefficient of the flow from the upper liquid header to the  $\frac{46}{47}$  tiny holes, which can be calculated by [30][31]: And  $C_d$  is the discharge cor 47 they holds, which can be ca

$$
C_d = 0.611 \left[ 87 \left( \frac{4.5\mu_l}{\rho_l d_h \sqrt{g H_{ev} - c_0}} \right)^{1.43} + \left( \frac{4.5\mu_l}{\rho_l d_h \sqrt{g H_{ev} - c_0}} \right)^{-1.26} \right]^{-0.7} \tag{12}
$$

 $4.2$  Key indicators  $4.2$  Ney mulcalors

A number of key indicators, including the output power, solar electrical and thermal  $\frac{1}{2}$  efficiencies, and overall solar efficiency, were applied to evaluate the performance of  $\frac{3}{4}$  the LHP-PV/T system. These are defined as below: 2 erriciencies, and overall sola  $4 \quad \text{III}$   $\text{III}$  -r  $\text{V}$  i system. Thes

 $\frac{6}{7}$  Output power of PV/T ( $P_{PVT}$ ) can be expressed as:  $7 \qquad \qquad$  Output power of  $1 \vee 1 \vee \neg$  (*PVT* 

$$
P_{PVT} = U_{PVT} I_{PVT} \t\t(13)
$$

<sup>13</sup> where,  $U_{PVT}$  is the output voltage of the PV/T panel and  $I_{PVT}$  is the current of the PV/T panel. 15 14

16<br>17 Solar electrical efficiency  $(\eta_e)$  can be expressed as: 17 Solar electrical efficiency (

$$
\eta_e = \frac{P_{PVT}}{G \cdot A_{PV}} = \frac{U_{PVT} I_{PVT}}{G \cdot A_{PV}} \quad , \tag{14}
$$

24 where,  $\eta_e$  is the electrical efficiency of PV/T panel, G is the solar radiation, and  $A_{PV}$  is 26 the PV area of base panel. Alternatively, Electrical efficiency  $(\eta_e)$  can be written as: 25 and 26 an

$$
\eta_e = \eta_{rc} [1 - \beta_{PV} (T_{PV} - T_{rc})], \qquad (15)
$$

33 where,  $\eta_{rc}$  is the initial electrical efficiency at reference temperature,  $\beta_{PV}$  is <sup>35</sup> temperature coefficient,  $T_{PV}$  is PV cells temperature at operation, and  $T_{rc}$  is reference temperature. 37  $34$ 36

40<br>
Solar thermal efficiency  $(\eta_{th})$  can be expressed as: 41 **Solar mermal efficiency** (*n*)

$$
\eta_{th} = \frac{q_{th}}{G A_{PV}} = \frac{c_W m_W (T_{wout} - T_{win})}{G A_{PV}},
$$
\n(16)

where,  $q_{th}$  is the useful thermal energy (heat) output of the system (W);  $c_w$  is the specific heat at constant pressure of liquid (water, 4186 W/kg/ $\degree$ C),  $m_w$  is the mass flow <sup>50</sup> rate of water flow in co-axial triple-pipe heat exchanger (kg/s),  $T_{wout}$  and  $T_{win}$  is the  $\frac{52}{53}$  outlet and inlet water temperature of the triple-pipe heat exchanger (°C). 48 49 specific heat at constant pro 51 rate of water flow in co-ax 53 **OUTEL AND THE WATER TEMP** 

 $56$  Overall solar efficiency  $(\eta_o)$  can be expressed as:  $57$ 

$$
\eta_o = \eta_e + \eta_{th} \tag{17}
$$

## 5 Results and discussion

#### <sup>2</sup> 5.1 Determination of optimal height difference 3 **5.1 Determination of optima**

 $\frac{5}{6}$  Height difference between the PV/evaporator and the condenser in this novel solar  $\mu_{\rm g}^7$  LHP-PV/T system plays a significant role. If the height difference is too small, the  $\frac{9}{20}$  gravity is not enough to drive the liquid to feed back the micro-channels leading to dry <sup>11</sup> out in the evaporator section, and then the lower section of the micro-channels could have no liquid to saturate the wick, which would result in increasing wall temperature 13 and the solar thermal performance degradation. While if the height difference is too 14  $\frac{16}{17}$  high, the vapour transportation would be too long and the total volume of the loop will <sup>18</sup> significantly increase which means that the loop needs more working fluid filling mass.  $\frac{20}{21}$  In addition, higher height difference leads to higher driven force to accelerate the liquid feeding back velocity, which will cause excess liquid and result in the micro-channels 22 full of liquid when bubbles form due to boiling, and finally decrease the evaporation 24 heat transfer ability. Furthermore, too large height difference could consequently 25  $\frac{27}{28}$  enlarge the size of the system thus make it not compact to install and apply. Therefore,  $\frac{29}{20}$  an optimal value of height difference of this system exerts a pivotal role to optimize the system size and make sure that the system operates safely and efficiently. 31  $6 \qquad \qquad$  Height difference between 8  $10$   $8\pi$   $\ldots$   $9\pi$   $\ldots$   $\ldots$   $\ldots$ 12 15 and the solar thermal perto <sub>17</sub> mgn, me vapour transporta 19 **Significantly increase winer** 21 and the contract of the con 23 and 24 and 25 and 26 an 26 **heat transfer ability. Furtl** 28 Chiarge the size of the syste 30 an optimal value of height  $32$ 

<sup>34</sup> In this process, the PV module surface temperature and solar thermal efficiency were applied to evaluate the performance of this novel solar LHP-PV/T heating system under 36 various height difference. The height difference is defined to be the distance from the 37 refrigerant inlet of the PV/evaporator to the middle position of the condenser as shown 39  $\frac{41}{42}$  in Figure 1. 35 38 various height difference. 40 reingerant inter of the P V/C  $42 \quad \text{m}$   $\text{H}$ 

For the given testing condition (i.e. ambient temperature at  $30\pm0.5^{\circ}$ C, cooling water <sup>46</sup> flow rate at 0.17, solar radiation at  $565\pm10$  W, different inlet water temperature), the solar thermal efficiencies of the system and the outlet water temperature under different 48 height difference were presented in Figure 5. It was found that increasing inlet water  $t_{52}$  temperature led to decreasing in solar thermal efficiency of the vertical LHP-PV/T  $\frac{53}{54}$  system when keeping the height difference constant. However, for the different inlet 55 water temperature groups, the parabola-like trend of the solar thermal efficiency variation with the different height difference is obvious, where there are maximum 57 thermal efficiencies occurring in the height difference of 1.1m, i.e. the optimization 59  $45$  Terms given testing come 47 49 50 height difference were pres 52 remperature led to decreas 54 system when keeping the 1 56 **Matter Components** Security 58

height difference is not at the highest HD of 1.3m in this testing condition, which is a height difference is not at the highest HD of 1.3m in this testing condition, which is a<br>little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this height difference is not at the highest HD of 1.3m in this testing condition, which is a<br>little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this height difference is not at the highest HD of 1.3m in this testing condition, which is a<br>little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this height difference is not at the highest HD of 1.3m in this testing condition, which is a<br>little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this from the evaporator to condenser, thus the heat transfer capacity of the LHP increased. 9 height difference is not at the highest HD of 1.3m in this testing condition, which is a<br>little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this theight difference is not at the highest HD of 1.3m in this testing condition, which is a<br>little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this height difference is not at the highest HD of 1.3m in this testing condition, which is a<br>little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this <sup>16</sup> vapour/liquid transportation line and the total volume of the loop will significantly height difference is not at the highest HD of 1.3m in this testing condition, which is a little different from the previous single testing result in reference[26] with larger refrigerate filling ratio. The reason for this be experimentally investigated in next step in details. In addition, height difference of 20 1.1 m can make the system more compact leading lower cost than that of 1.3 m. little different from the previous single testing result in reference[26] with larger<br>refrigerate filling ratio. The reason for this may be that (1) when the height difference<br>increased, the thermo-siphon effect caused by refrigerate filling ratio. The reason for this may be that (1) when the height difference<br>increased, the thermo-siphon effect caused by the density difference between the<br>refrigerant vapour and liquid was enhanced, leading increased, the thermo-siphon effect caused by the density difference between the refrigerant vapour and liquid was enhanced, leading to the increased heat transfer rate from the evaporator to condenser, thus the heat trans experimental investigations. 29 1  $\frac{1}{1}$   $\frac{1}{1}$ 2 little different from the pro- $\frac{3}{2}$  as for contract  $\frac{1}{2}$  $4 \quad \text{Iemgerate minus ratio.}$  $\frac{5}{2}$  increased the thermo-sinh 6 more more than spine 7 refrigerant vapour and liquid was enhanced, leading to the increased heat transfer rate  $8$  c  $1$  i 10 11 As a result, the solar therm  $12$   $1 \cdot 1 \cdot 1$   $1 \cdot 1$ 13 line neight difference is abo  $14$  decreesing which mey 15 accreasing which may 17 reportinguir transportant 18 increase which means that the loop needs more working fluid filling mass which will 19 21 22 1.1 m can make the syste 23 TH C · 1 1 24 **Interviews**, in present study, 25 the construction from and 26 and construction name and value of height difference between evaporator and heat exchanger for the following 28 and the compact of the c



 $\frac{53}{54}$  Figure 5. Solar thermal efficiency changing with height difference  $54$  rigure 3. Solar the

The PV module surface temperature against the height difference at the same given <sup>58</sup> testing condition is shown in Figure 6. It is found that at the first operation hour, PV module surface temperature shows lowest value at the height difference of 1.1 m, and 60 57 hours invaded surface to 59 61

then PV module surface temperature shows lowest value at the height difference of 1.3  $\frac{1}{2}$  m and that of 1.1 m become the second lowest value. When the system becomes steady,  $\frac{3}{4}$  the PV module surface temperature almost decreases with the rise of the height  $\frac{5}{6}$  difference. This may be because higher height difference causes larger driven force <sup>7</sup> which is superior to the heat transfer ability to efficiently cool the PV module thus leading lower PV module surface temperature. 2 m and that of 1.1 m become  $4 \quad \text{III}$   $4 \quad \text{III}$  $6 \,$   $\ldots$   $\ldots$   $\ldots$   $\ldots$   $\ldots$   $\ldots$ 



 $\frac{34}{35}$  Figure 6. PV module surface temperature against height difference **Figure 6. PV mo** 

#### $\frac{37}{38}$  5.2 Temperature distribution for optimal height difference under steady solar radiation **38** *S.2 Lemperature alstributic*

 $\frac{40}{41}$  Keeping the parameters unchanged (i.e., height difference at optimal value of 1.1 m, ambient air temperature at  $30\pm0.3^{\circ}$ C, initial inlet cooling water temperature at  $19\pm1^{\circ}$ C,  $^{44}$  cooling water flow rate at 0.17 m<sup>3</sup>/h), the temperature distribution of different <sup>46</sup> components of the PV/evaporator was experimentally investigated. Figure 7 shows the variation of the PV/T evaporator components surface temperature against operation  $\frac{49}{50}$  time. It is found that (1) there is a start-up process at the test beginning, when the  $\frac{51}{52}$  temperature of different components is rising with operating time, and then becomes steady state; (2) the highest temperature of the PV model surface is still lower than 55<br>  $\degree$ C, which is lower than existing PV/T system at the same solar radiation condition, indicating that the micro-channel loop heat pipe shows an efficient heat transfer ability to cool the PV model; and (3) the mid-level of panel has higher temperature than the 41 Recepting the parameters un 43 amorem un remperador al *c*  48 variation of the PV/T evaluation time. It is found that (1) t 52 competant of university to 53 steady state:  $(2)$  the highest Steady state,  $(2)$  are ingress 59 to cool the PV model; and

lower and higher levels of the panel, indicating that the simulated solar radiation  $\frac{1}{2}$  distribution across the panel is non-uniform and mid-level of the panel receives higher  $\frac{3}{4}$  radiation compared to the lower and upper parts. **alstribution across the panel** 4 radiation compared to the io



(a)



(b)



 $\frac{23}{24}$  Figure 7. Temperature distribution of different components for the PV/T panel right  $\lambda$  remperature to

26 In view of different layers of the panel, as shown in Figure 8, it is surprising to see that the aluminium plate in the mid-layer achieved a little higher temperature compared to  $\frac{29}{30}$  the PV surface and micro-channel evaporator which were above and below the  $\frac{31}{22}$  aluminium plate respectively. This indicates that the heat transfer across the panel was <sup>33</sup> two-directional: one part of the heat is transferred from the aluminium plate to ambient and the other part is directed into the refrigerant within the LHP evaporator. 28 the aluminium plate in the and  $\frac{1}{2}$  and  $\$ **attribution** place respective 34 and the second contract of  $\mathbf{r}$  and  $\mathbf{r}$ 



 $\frac{59}{60}$  Figure 8. Temperature of different components of PV/T panel vs Operating time **Figure** of **Emperature** of

# 5.3 Effect of different cooling water flow rate for optimal height difference under steady<br>solar radiation<br>With the red points and the text figures showing the means of solar thermal efficiency  $\frac{4}{5}$  solar radiation 2 3 5 Solar radiation

 $\frac{7}{8}$  With the red points and the text figures showing the means of solar thermal efficiency  $\frac{9}{20}$  of different flow rate groups, Figure 9 shows variation of the solar thermal efficiency 5.3 *Effect of different cooling water flow rate for optimal height difference under steady solar radiation*<br>With the red points and the text figures showing the means of solar thermal efficiency<br>of different flow rate gro 5.3 Effect of different cooling water flow rate for optimal height difference under steady<br>solar radiation<br>With the red points and the text figures showing the means of solar thermal efficiency<br>of different flow rate group 5.3 *Effect of different cooling water flow rate for optimal height difference under steady solar radiation*<br>With the red points and the text figures showing the means of solar thermal efficiency<br>of different flow rate gro  $\frac{16}{17}$  flow rate is increasing, so the heat transfer rate between the refrigerant within the  $\frac{18}{10}$  interior channel of the co-axial tubular exchanger and the cooling water within the mid-5.3 *Effect of different cooling water flow rate for optimal height difference under steady solar radiation*<br>With the red points and the text figures showing the means of solar thermal efficiency<br>of different flow rate gr 5.3 Effect of different cooling water flow rate for optimal height difference under steady solar radiation<br>
With the red points and the text figures showing the means of solar thermal efficiency<br>
of different flow rate gr 5.3 Effect of different cooling water flow rate for optimal height difference under steady<br>solar radiation<br>With the red points and the text figures showing the means of solar thermal efficiency<br>of different flow rate grou system of the reduced. The text of the text of the specific operation is specific operation. With the red points and the text figures showing the means of solar thermal efficiency of different flow rate groups, **Figure 9** Solar ratuation<br>With the red points and the text figures showing the means of solar thermal efficiency<br>of different flow rate groups, **Figure 9** shows variation of the solar thermal efficiency<br>of the system against the co  $\frac{29}{20}$  flow rate increased from 0.05m3/h to 0.17 m3/h. 8 WILLI LOT POINTS and the  $10$ <sup>11</sup> of the system against the cooling water flow rate. It is clear to see that the thermal 12 13 efficiency of this vertical system increased obviously with the increase of the cooling 14 15 water flow rate firstly and t 17 110w rate is increasing, so 19 micro channel of the co a 20 channel of the exchanger is 21 22 MCLHP, reflected in increasing refrigerant flow rate. As a result, the heat loss of the 23 24 vertical LHP-PV/T panel to ambient decreased and solar thermal efficiency of the<br>25 25 and 26 an 26 system increased. With the 27  $\mathbf{c}$   $\$ 28 CHICLERY OF THIS VEHICAL SY 30 how rate increased from 0.



Figure 9. Solar thermal efficiency against cooling water flow rate 58

## 5.3 Effect of different air temperature for optimal height difference under steady solar  $\overline{2}$  radiation 1 and  $\mathbf{r}$  and  $\mathbf{r}$

 $\frac{4}{5}$  The effect on solar thermal efficiency of different ambient air temperature was  $\frac{6}{7}$  experimentally compared in groups and discussed. The results are shown in Figure 10  $\frac{8}{3}$  with the red points and the text figures showing the means of solar thermal efficiency 10 of different groups. When keeping solar radiation at 560±5W/m2, cooling water flow rate at 0.17m<sup>3</sup>/h at optimal height difference unchanged. The different ambient air temperature is corresponding to different solar thermal efficiency, which can be observed that with the increase of the air temperature from  $24.1^{\circ}$ C to 30  $^{\circ}$ C, the solar <sup>18</sup> thermal efficiency has an obvious increase tendency from 46.72% to 57.48%. The reason for this phenomenon is that the increasing ambient air temperature would decrease the heat transfer performance between the PV/evaporator panel surface and the surrounding and thus the heat loss from the vertical LHP-PV/T panel surface to the  $\frac{25}{26}$  ambient became less. Therefore, the average system solar thermal efficiency is  $\frac{27}{28}$  improved.  $5 \qquad \qquad$  The effect on solar therma 7 Capernmentary compared in 9 and 1 12 rate at 0.17m<sup>3</sup>/h at optima **temperature** is correspond 16 bserved that with the incr 19 months of the contract of the same set 24 the surrounding and thus the 26 amolent became less. In 28 mpoved.



Figure 10. Solar thermal efficiency vs ambient air temperature. 

## 5.3 Transient state response for optimal height difference under real weather

2<br>
In this section, the transient solar thermal and electrical performance of this novel LHP-5.3 Transient state response for optimal height difference under real weather<br>In this section, the transient solar thermal and electrical performance of this novel LHP-<br>PV/T system with an optimal height difference of 1.1 weather condition (experiment was carried out on  $22<sup>nd</sup>$  Dec. 2018 in GDUT, China).  $\frac{8}{3}$  Figure 9 presents the PV module temperature, ambient air temperature and solar radiation under real weather during the outdoor testing day. The solar radiation was in the range of 250 to 610 W/m<sup>2</sup>, and the ambient temperature is around 22 to 29 °C. It is 13<br>
14 **Solume 60 found that the PV module temperature is sensitively changing with the solar radiation,** <sup>15</sup> and it is interesting to observe that the temperatures from bottom to top along the PV  $\frac{17}{18}$  module surface were lower than 60 °C and were almost in equivalence at the same <sup>19</sup> moment, respectively. This is owing to the fact that the solar radiation to different position of the PV module surface are uniform at the same time under the real weather, which is superior to the performance of this system. 3 In this section, the transient s 4  $\mathbf{D} \mathbf{V} / \mathbf{T}$  (1)  $5 \tF V/T$  system with an optim 7 weather condition (experiment the range of 250 to 610 W/ 14 found that the PV module t 16 and it is interesting to obse 18 module surface were fower 20 months, respectively. The 



 $^{49}_{50}$  Figure 9 PV module temperature, ambient air temperature and solar radiation under real weather rigure 9 PV module tem 

with an optimal height difference of 1.1 m are illustrated in Figure 10. Under these specific operational conditions and the real weather solar radiation, the solar thermal efficiency of the system was in the range from 34.8% to 71.4%, the solar electrical <sup>54</sup> The transient solar thermal and electrical performance of this novel LHP-PV/T system 55 The during bound increase **Figure 1**  60 efficiency of the system w

efficiency was in range from 9.2% to 18.8%, and overall efficiency was in range from  $\frac{1}{2}$  51.2% to 88.4% respectively, which are higher than most of existing conventional solar  $\frac{3}{4}$  heat pipe PV/T system. It is obvious that the solar thermal efficiency is not sensitive to  $\frac{5}{6}$  the solar radiation, where there is always lagging response and even opposite trend, <sup>7</sup> reflecting that the transient heat transfer process is not sensitive to the solar radiation; and the solar electrical efficiency almost keeps the same value during the testing time and has the opposite trend with the PV module temperature, which complied well with the common knowledge of the solar PV/T system that PV panels normally have linear  $\frac{14}{15}$  drop-off in efficiency as the surface temperature rises, which typically lose efficiency <sup>16</sup> of up to 0.4% per degree centigrade rise in the PV-2003; Chow, 2010).  $2$   $2$   $2\%$  to 88.4% respectively lieat pipe  $\mathbf{r}$  v/1 system. It is 6 and solut furthermone, where if 8 and 2 <br>8 and 2 11 and has the opposite trend 13 line common knowledge of  $\frac{15}{15}$  and  $\frac{15}{15}$  and  $\frac{1}{15}$   $\sigma$   $\alpha$   $\mu$   $\sigma$   $\sigma$   $\sigma$   $\mu$   $\sigma$   $\alpha$   $\sigma$  $\mu$ 



Figure 10. Solar thermal and electrical performance under real weather

# 6 Conclusion

50<br>
In this paper, the impact of the height difference (HD) between the evaporator and the  $\frac{52}{53}$  condenser on the solar thermal performance of the novel LHP-PV/T system was <sup>54</sup> experimentally investigated. Various tests were carried out to find out the optimal value 56 of the height difference between the PV/evaporator and the condenser to optimize the system construction and thus helping to develop a high thermal performance and low cost solar PV/T system for domestic use. The conclusions were summarized as follows: 51 In this paper, the impact of **CONDITION** CONTINUES ON THE SUITE OF 55 Capermentary investigated **C** 

(1) Under the specific operational and testing condition, the parabola-like trend of the  $\frac{1}{2}$  solar thermal efficiency variation with the different height difference is obvious, where  $\frac{3}{4}$  the maximum thermal efficiencies takes place at the height difference of 1.1m, which  $\frac{5}{6}$  is considered as the optimal value of height difference between heat exchanger and evaporator. In addition the height difference of 1.1 m can make the system more compact leading to a lower cost than that of 1.3m. 2 Solar thermal efficiency varia  $4 \quad \text{III}$  and  $4 \quad \text{III}$ **B** considered as the optimal 

(2) When the system operates at the steady state condition, the PV module surface 13<br>14 **temperature is decreased with the increase of the height difference.** 12 (2) When the system oper **temperature is decreased w** 

<sup>16</sup> (3) The aluminium plate in the mid-layer achieved a little higher temperature compared  $\frac{18}{10}$  to the PV surface and micro-channel evaporator which were above and below the aluminium plate respectively, indicating that the heat transfer across the panel was two directional. (3) The aluminium plate in to the 1 v surface and  $\text{m}$ **Expanding Property** Property 21 

(4) Under the specific operational conditions and the real weather solar radiation, the solar thermal efficiency of the system was in the range of 34.8% to 71.4%, the solar <sup>28</sup> electrical efficiency was in range of 9.2% to 18.8%, and overall efficiency was in range <sup>30</sup> of 51.2% to 88.4% respectively, which are higher than most of the existing conventional  $\frac{32}{22}$  solar heat pipe PV/T system. In addition, the solar thermal efficiency is not sensitive to the solar radiation, where there is always lagging response and even opposite trend. This indicated that the transient heat transfer process is not sensitive to the solar radiation. 27 solar thermal efficiency of 29 electrical efficiency was in 01.270 to 00.770 respectively 

 $\frac{40}{41}$  Acknowledgements: The authors would acknowledge our appreciation to the financial <sup>42</sup> supports from the EPSRC (EP/R004684/1) and Innovate-UK (TSB 70507-481546) for <sup>44</sup> the Newton Fund – China-UK Research and Innovation Bridges Competition 2016 46 Project 'A High Efficiency, Low Cost and Building Integrate-able Solar from the UK BEIS for the project titled 'A low carbon heating system for existing public buildings employing a highly innovative multiple-throughout-flowing micro channel solar-panels from Department of Science and Technology of Guangdong Province, China (2018A050501002 & 2019A050509008), European Commission H2020-MSCA- RISE-2016 Programme (734340-DEW-COOL-4-CDC), European Commission **ACKHOWIEGGEMENTS:** The a **Supports from the ET SIXC**  48 Photovoltaic/Thermal (PV/T) System for Space Heating, Hot Water and Power Supply', 49 50 from the UK BEIS for the 52 public buildings employing 54 Channel Solar-panels-array **Figure 1.5 April 1** 

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#### <sup>8</sup> Reference 9

- [1] International Energy Agency (IEA). Market Report Series Renewables 2O18 11 Analysis and Forecast to 2023. 2018. 13 12 and 12 an
- [2] Luque A, Hegedus S. Handbook of Photovoltaic Science and Engineering.  $\frac{17}{18}$  2003. https://doi.org/10.1002/0470014008.  $15$ 16 [2] Luque A, Hegedus S 18 2003. https://doi.org
- <sup>20</sup> [3] Skoplaki E, Palyvos JA. On the temperature dependence of photovoltaic module electrical performance: A review of efficiency/power correlations. 22 24<br>
Solar Energy 2009;83:614–24. https://doi.org/https://doi.org/10.1016/j.solener.2008.10.008. 26 21 [2]  $\delta$ KOPIAKI E, raiyvos 23 model of the control point  $25$
- [4] Chow TT. A review on photovoltaic/thermal hybrid solar technology. Applied 29 Energy 2010;87:365 79. https://doi.org/10.1016/j.apenergy.2009.06.037. 30 31 Energy 2010;87:365
- $\frac{33}{34}$  [5] Zhou J, Zhao X, Yuan Y, Li J, Yu M, Fan Y. Operational performance of a  $\frac{35}{36}$  novel heat pump coupled with mini-channel PV/T and thermal panel in low 37 solar radiation. Energy and Built Environment 2020;1:50–9. https://doi.org/https://doi.org/10.1016/j.enbenv.2019.08.001. 39  $[34]$  [3]  $2 \text{ mod } 3$ ,  $2 \text{ mod } 3$ ,  $1 \text{ mod } 3$ 36 hover hear pump contract to the same state of the same st 38 **Sam Taunahon** Energy 40  $\sim$  1
- [6] Hussain F, Othman MYH, Sopian K, Yatim B, Ruslan H, Othman H. Design 42 development and performance evaluation of photovoltaic/thermal (PV/T) air base solar collector. Renewable and Sustainable Energy Reviews 2013;25:431 45 47<br>41. https://doi.org/https://doi.org/10.1016/j.rser.2013.04.014. 43 44 development and per 46 base solar collector. 48 **11.** Https://**u**01.01g/11t
- $[7]$  Delisle V, Kummert M. A novel approach to compare building-integrated photovoltaics/thermal air collectors to side-by-side PV modules and solar 52 54 thermal collectors. Solar Energy 2014;100:50–65. https://doi.org/https://doi.org/10.1016/j.solener.2013.09.040. 56  $51$   $\lfloor y \rfloor$  Density, Rummer 53 55
- [8] Fraisse G, Ménézo C, Johannes K. Energy performance of water hybrid PV/T 58 59 [8] Fraisse G, Menezo C

collectors applied to combisystems of Direct Solar Floor type. Solar Energy  $\frac{1}{2}$  2007;81:1426–38. https://doi.org/https://doi.org/10.1016/j.solener.2006.11.017. 2  $200$  /;81:1420-38. http://

- $\frac{4}{5}$  [9] Herrando M, Markides CN, Hellgardt K. A UK-based assessment of hybrid PV  $\frac{6}{7}$  and solar-thermal systems for domestic heating and power: System  $\frac{8}{9}$  performance. Applied Energy 2014;122:288–309. https://doi.org/https://doi.org/10.1016/j.apenergy.2014.01.061. 10  $[5]$  [9] Herrando M, Markide 7 and solar thermal syst 9 1 1
- [10] Long H, Chow TT, Ji J. Building-integrated heat pipe photovoltaic/thermal 13 <sup>14</sup> system for use in Hong Kong. Solar Energy 2017;155:1084–91. https://doi.org/10.1016/j.solener.2017.07.055. 16 15 system for use in Ho 17 mups://doi.org/10.10
- <sup>19</sup> [11] Zhang X, Zhao X, Xu J, Yu X. Characterization of a solar photovoltaic/loop-21 heat-pipe heat pump water heating system. Applied Energy 2013;102:1229–45. https://doi.org/https://doi.org/10.1016/j.apenergy.2012.06.039. 23 20 [11]  $\angle$ mang  $\angle$ x,  $\angle$ mao  $\angle$ x,  $\angle$ x 22 may pipe near pamp 24 **1 2 1**
- [12] Zhang X, Zhao X, Shen J, Xu J, Yu X. Dynamic performance of a novel solar 26 photovoltaic/loop-heat-pipe heat pump system. Applied Energy 2014;114:335 28 52. https://doi.org/10.1016/j.apenergy.2013.09.063. 29 27 30 **52. https://doi.org/10**
- $\frac{32}{33}$  [13] He W, Hong X, Zhao X, Zhang X, Shen J, Ji J. Operational performance of a novel heat pump assisted solar façade loop-heat-pipe water heating system. 34 36 Applied Energy 2015;146:371-82. https://doi.org/10.1016/j.apenergy.2015.01.096. 38  $33$  [15] He w, Holly  $\Lambda$ , Zha 35 november pump ass  $37 \t\t 11 \t\t 37$
- [14] Reay DA, Kew PA, McGlen RJ. Chapter 6 Special types of heat pipe. In: 41 Reay DA, Kew PA, McGlen RJ, editors. Heat Pipes (Sixth Edition). Sixth Edit, 42 <sup>44</sup><br>
0xford: Butterworth-Heinemann; 2014, p. 135–73. 46<br>https://doi.org/https://doi.org/10.1016/B978-0-08-098266-3.00006-6. 43 Reay DA, Kew PA, 45 **OXIOIG:** Butterworth 47 https://wor.org/mups.
- <sup>49</sup> [15] Yu M, Diallo TMO, Zhao X, Zhou J, Du Z, Ji J, et al. Analytical study of 51 impact of the wick's fractal parameters of 53 micro-channel loop heat pipe. Energy 2018;158:746–59. https://doi.org/10.1016/j.energy.2018.06.075. 55  $\begin{bmatrix} 1 & 0 \\ 0 & 0 \end{bmatrix}$   $\begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix}$ 52 54
- [16] Diallo TMO, Yu M, Zhou J, Zhao X, Shittu S, Li G, et al. Energy performance 57 analysis of a novel solar PVT loop heat pipe employing a microchannel heat 59 58 [10] Dialo I MO, Y U M, 60 analysis of a hover see

pipe evaporator and a PCM triple heat exchanger. Energy 2019;167.  $\frac{1}{2}$  https://doi.org/10.1016/j.energy.2018.10.192. 2 https://doi.org/10.101

- $\frac{4}{5}$  [17] Ling L, Zhang Q, Yu Y, Liao S, Sha Z. Experimental study on the thermal  $\frac{6}{7}$  characteristics of micro channel separate heat pipe respect to different filling  $\frac{8}{9}$  ratio. Applied Thermal Engineering 2016;102:375–82. https://doi.org/10.1016/j.applthermaleng.2016.03.016. 10  $\begin{bmatrix} 1/ \end{bmatrix}$  Ling L, Znang Q, Tu 7 Characterístics of fine  $9 \hspace{1.5cm} \overline{\hspace{1.5cm}}$ 11
- [18] Diallo TMO, Yu M, Zhou J, Zhao X, Ji J, Hardy D. Analytical investigation of 13 the heat-transfer limits of a novel solar loop-heat pipe employing a mini-14 channel evaporator. Energies 2018;11. https://doi.org/10.3390/en11010148. 16 15 the heat-transfer lim 17 Channel evaporator.
- $\frac{19}{20}$  [19] Zhao X, Wang Z, Tang Q. Theoretical investigation of the performance of a novel loop heat pipe solar water heating system for use in Beijing, China. 21 23 Applied Thermal Engineering 2010;30:2526–36. 20  $\left[17\right]$   $\left[2\right]$   $\left[2\right]$   $\left[2\right]$   $\left[1\right]$   $\left[2\right]$   $\left[2\right$ 22 november 22 24 and  $\overline{11}$
- [20] Zhang X, Shen J, Adkins D, Yang T, Tang L, Zhao X, et al. The early design 26 stage for building renovation with a novel loop-heat-pipe based solar thermal 28 facade (LHP-STF) heat pump water heating system: Techno-economic analysis 29 in three European climates. Energy Conversion and Management 31 2015;106:964 86. https://doi.org/10.1016/j.enconman.2015.10.034. 33 27  $30$  racade (LHP-SIF) n 32 m unce European en 34 **2010**,100.501 **00.** In
- [21] Zhang X. Investigation of a Novel Solar Photovoltaic/Loop-Heat-Pipe Heat 36 Pump System. University of Hull, 2014. 38 https://hydra.hull.ac.uk/resources/hull:8422. 37 L J *G G* 39 40 https://hydra.hull.ac.
- [22] Ren X, Yu M, Zhao X, Li J, Zheng S, Chen F, et al. Assessment of the cost 42 reduction potential of a novel loop-heat-pipe solar photovoltaic/thermal system 44 <sup>46</sup> by employing the distributed parameter model. Energy 2020;190:116338. https://doi.org/10.1016/j.energy.2019.116338. 48 43 [22] Ren X, Yu M, Zhao 45 reduction potential of 47 by comproying the dist 49
- [23] Su Q, Chang S, Yang C. Loop heat pipe-based solar thermal façade water 51 heating system: A review of performance evaluation and enhancement. Solar 53 Energy 2021;226:319 47. https://doi.org/10.1016/j.solener.2021.08.019.  $52$ 54 55 Energy 2021;226:31
- $\begin{bmatrix} 57 \\ 58 \end{bmatrix}$  [24] Li G, Pei G, Ji J, Yang M, Su Y, Xu N. Numerical and experimental study on a 59<br>
PV/T system with static miniature solar concentrator. Solar Energy 58 [ $24$ ] L1 G, Pel G, J1 J, Yal  $\mathbf{r}$   $\mathbf{v}$  is system with st

2015;120:565 74. https://doi.org/https://doi.org/10.1016/j.solener.2015.07.046.

- <sup>2</sup> [25] Li G, Diallo TMO, Akhlaghi YG, Shittu S, Zhao X, Ma X, et al. Simulation  $\frac{4}{5}$  and experiment on thermal performance of a micro-channel heat pipe under  $\frac{6}{7}$  different evaporator temperatures and tilt angles. Energy 2019;179:549–57. 8 https://doi.org/https://doi.org/10.1016/j.energy.2019.05.040. 3  $[25]$  Li G, Diallo I MO, Al 7 anticipal components with the components of the components  $9 \t 1 \t 5 \t 1$
- [26] Diallo TMO, Yu M, Zhou J, Zhao X, Shittu S, Li G, et al. Energy performance analysis of a novel solar PVT loop heat pipe employing a microchannel heat <sup>14</sup> pipe evaporator and a PCM triple heat exchanger. Energy 2019;167:866–88. https://doi.org/10.1016/j.energy.2018.10.192. **pipe evaporator and** 17 mups://doi.org/10.10
- $\frac{19}{20}$  [27] Yu M, Chen F, Zheng S, Zhou J, Zhao X, Wang Z, et al. Experimental 21 Investigation of a Novel Solar Micro-Channel Loop-Heat-Pipe Photovoltaic/Thermal (MC-LHP-PV/T) System for Heat and Power Generation. Applied Energy 2019;256:113929. https://doi.org/10.1016/j.apenergy.2019.113929. 20  $[27]$  1 a M, Chen 1, Zhen 27 https://doi.org/10.10
- [28] Zhang X, Zhao X, Xu J, Yu X. Characterization of a solar photovoltaic/loop- <sup>31</sup> heat-pipe heat pump water heating system. Applied Energy 2013;102:1229–45. https://doi.org/10.1016/j.apenergy.2012.06.039. 30  $[\angle \delta]$  Znang X, Znao X, X 32 mea-pipe near pump 34 mps. assets to 101
- [29] Chen CQ, Diao YH, Zhao YH, Wang ZY, Zhu TT, Wang TY, et al. Numerical evaluation of the thermal performance of different types of double glazing flat plate solar air collectors. Energy 2021;233:121087. https://doi.org/https://doi.org/10.1016/j.energy.2021.121087.  $37 \qquad \qquad 1 \qquad \qquad 37$  **https://doi.org/https:**
- <sup>44</sup> [30] Swamee PK, Swamee N. Discharge equation of a circular sharp-crested orifice.  $\frac{46}{47}$  Journal of Hydraulic Research 2010;48:106-7. [30] Swamee PK, Swame **Soutial Strategy**
- [31] Shabanlou, Saeid. Improvement of Extreme Learning Machine Using Self- Adaptive Evolutionary Algorithm for Estimating Discharge Capacity of Sharp- Crested Weirs Located on the end of Circular Channels. Flow Measurement and Instrumentation 2017:S0955598617301942.  $[50$  [31] Shabamoa, Sacra: m 55 and Instrumentation