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Title: Performance analysis of solar assisted heat pump coupled with build-in PCM heat storage based on PV/T panel

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Abstract: PV/T (photovoltaic/thermal) technology is a combination of PV module (photovoltaic utilization) and collector (photothermal utilization), which can improve the comprehensive utilization efficiency of solar energy and has a broad application prospect. In this paper, PV/T module is coupled with heat pump evaporator to form a direct-expansion solar PV/T heat pump which is suitable for heat application in high latitude area. To achieve stable residential heating, a solar PV/T heat pump system coupled with build-in PCM (phase change material) heat storage is therefore proposed and simulated. Meanwhile, the mathematical model of solar PV/T heat pump coupled with build-in PCM heat storage system is established and verified. The simulation results show that the temperature of underfloor heating which using build-in PCM heat storage can reach 22 °C to 31 °C after 39 hours when the circulating water is 40 °C. Moreover, the heating COP (Coefficient of Performance) increases with the increase of solar radiation, ambient temperature and area of PV/T collector, and decrease of wind speed, respectively. A 20 m² PV/T panel module can output 21.4% of the electricity to power grid when the solar radiation intensity is 600 W/m² and meet the heat demand of a 100 m² room while maintain the operation of the system. Meanwhile, the heating COP can reach 5.79 which is 70% higher than the conventional air conditioning system and the electrical, thermal, overall efficiencies are 17.77%, 55.76% and 75.49%, respectively.

Highlights

- A solar assisted heating/power generation system based on PV/T panel is proposed for residential heating in high latitude area.
- Design of build-in PCM heat storage unit is proposed and simulated.
- The comprehensive energy utilization efficiency of this solar PV/T heat pump system can reach above 75%.

Dear editor and reviewers:

Thanks for your responsible comments on our manuscript. It is very valuable and helpful to improve our paper. We have revised the manuscript carefully according to your suggestions. The revised portions are marked in red in the revised manuscript.

Reviewer 1

#1. The validation process are wrong. Firstly, the system reported in the manuscript is totally different from the validation case, which comes from the last Reference, Zhou et al. 2019. In Zhou's study, there is no any pcm thermal energy storage (PCM-TES) unit, while the PCM-TES unit in this study is the novelty in the system. By using different system to validate this work, it is hardly possible to get reasonable results, even the given data seems agree well.

Response: Thanks for your comments. As shown in Fig. 2, the solar PV/T heat pump cycle is the main sub cycle of the proposed system. In Zhou's paper, they proposed a solar PV/T heat pump system which has the same components (PV/T collector/evaporator, compressor, condenser, expansion valve). The only difference between Zhou's paper and our proposed system is that in our proposed system, the thermal energy will be stored in build-in PCM heat storage unit, but in Zhou's system, it will be stored directly in hot water tank. In other words, the difference is just the type of heat storage, in PCM-TES or in water tank. Therefore, the added PCM-TES unit in our proposed system will not influence the performance of solar PV/T heat pump system (For example, thermodynamic state points of each components, thermal energy gain through PV/T panel, heat of condensation, etc.). It's reasonable to refer Zhou's experiments results for validation due to the same basic thermodynamic cycle and operating conditions.

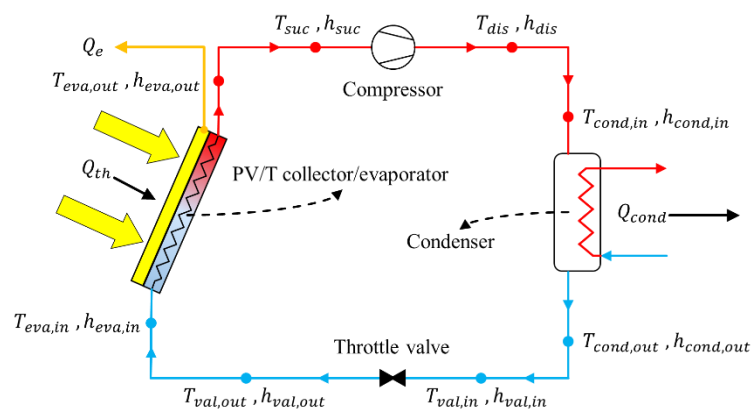


Fig. 2. The thermodynamic state points for each component.

#2. In Fig. 12, there is simulated photovoltaic efficiency. How to simulate it? It is totally influenced by the material and structure of the PV panel. It seems hardly possible to simulate its performance. Furthermore, even there is some semi-empirical calculated data, it can not be used to validate the coupled system reported in this study. Only by using the same system, the model could be validated. It seems common sense.

Response: Thank you for your comments. According to Huide's paper, the photovoltaic efficiency can be calculated by Eq. (2). η_{rc} is the reference photovoltaic efficiency value of the PV cells at $T_{rc}=298$ K, $I=1000$ W/m², $\eta_{rc}=0.18$; β_{pv} is the temperature coefficient (1/K) of PV cell efficiency, $\beta_{pv}=0.0045$ (Huide et al., 2017).

$$\eta_e = \eta_{rc} \cdot \left[1 - \beta_{pv} \cdot (T_p - T_{rc}) \right] \quad (2)$$

As shown in Eq. (2), the reference photovoltaic efficiency (η_{rc}) is determined by material and structure of PV panel. However, the photovoltaic efficiency (η_e) is only influenced by PV cells' temperature (T_p) which could be simulated when the material and structure is fixed.

The simulation process of PV cells' temperature is as follows: firstly, input all the environmental, system design, operation parameters and assume the temperature of PV cells (T_p). Secondly, calculate the overall heat loss rate (Q_L), PV electrical output power (Q_e), thermal energy gain rate through PV/T collector/evaporator (Q_{th}) and the useful heat transfer rate by refrigerant (Q_u). Then the program starts iteration and moves to next step when $|(Q_{th} - Q_u)/Q_{th}| < 0.1\%$ which means the system achieves energy balance. If $|(Q_{th} - Q_u)/Q_{th}| \geq 0.1\%$, the PV cells' temperature (T_p) would be modified and the program would start another iteration using the modified T_p . Thus, the PV cells' temperature could be calculated through the program as well as the photovoltaic efficiency. The algorithm has been presented in section 3.5.

The proposed system has the same thermodynamic cycle and operating conditions with Zhou's study which consist of PV/T collector/evaporator, compressor, condenser, expansion valve. Compared to Zhou's system, the only difference is that the proposed system has added a thermal storage part which will not influence the heat pump system's performance. Thus, the proposed solar PV/T heat pump system could be validated by Zhou's system.

#3. There are too many figures and tables are useless in this manuscript, which makes this manuscript long and even dull. For example, Fig. 3c is not mentioned at all.

I do know why authors use the solar radiation intensity data in Figs. 11 and 12, but not mentioned them at all. This makes the figures much more complex.

Response: Thank you for your comments. The authors have revised the manuscript, several figures have been deleted and Fig. 11, Fig. 12 have been modified. The revised figures are listed as follows:

1. Fig. 3 (c)(d) have been deleted.
2. Fig. 8 (c~l) have been deleted.
3. Fig. 26 has been deleted.
4. Fig. 11 and Fig. 12 have been revised to Fig. 11. (a)(b) according to your comments. The curves of solar radiation intensity have been deleted in revised Fig. 11. (a)(b) and these two figures are simpler and easier to understand.

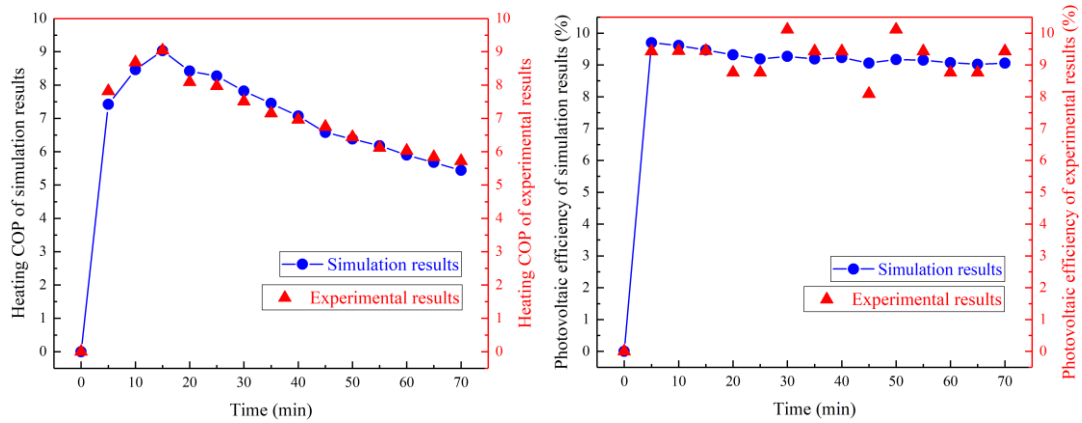


Fig. 11. (a) Comparison results of experimental and simulated heating COP. **(b)** Comparison results of experimental and simulated photovoltaic efficiency.

#4. Many unimportant results are given in the conclusions, such as the following one.

(2) The heating COP increases with the increase of solar radiation, ambient temperature, area of PV/T collector, and decrease of wind speed, respectively. Solar radiation intensity and area of PV/T collector are two major factors affecting system performance compared to wind speed and ambient temperature.

As far as I know, this conclusion could be given even this work is not conducted. It could be given easily. Clearly, the novelty of this work is lack.

Response: Thank you for your comments. The second conclusion in Line 625-628 has been deleted according to your comments. The novelty of this work has been added in section Introduction in Line 102-104.

The revised manuscript of Line 102-104 is as follows:

The objective of this paper is to provide a promising method to realize stable, high efficiency, environmental friendly residential heating in high latitude area with no energy consumption from power grid.

#5. Even authors revised the manuscript many times, there are still many problems. For example, in Fig.8, the color bar starts at different temperatures, 290 K, 300 K, 287 K, 297 K, etc. Why?

Response: Thank you for your comments. According to your Comment #3, Fig. 8 (c~l) have been deleted. The color bar starts at the same temperature in revised Fig. 8 (a)(b)(c).

The revised manuscript of Fig. 8 is as follows:

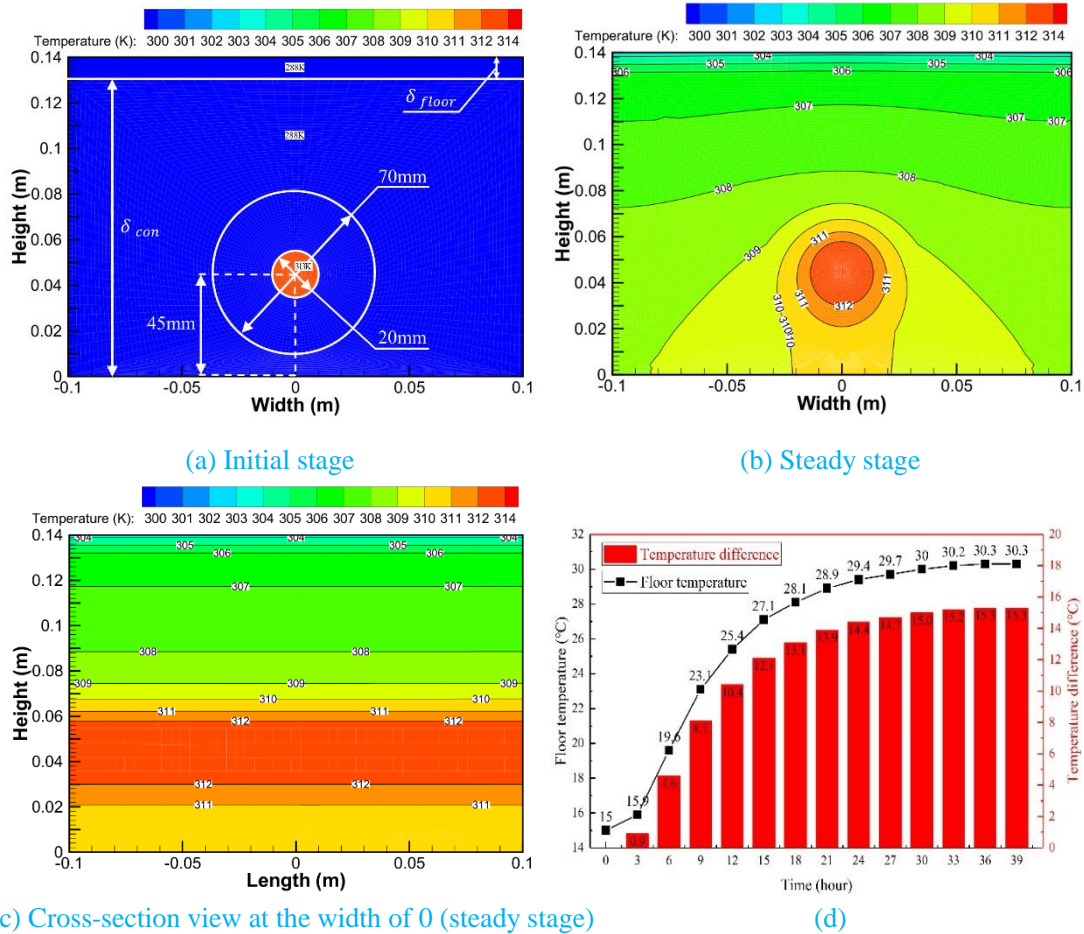


Fig. 8. (a~c) Cross-section temperature contour of the build-in PCM heat storage unit at initial and steady stage. (d) Variation curve of floor temperature and temperature difference between floor and ambient.

#6. The manuscript has a poor structure, and the length could be divided into 2-3 articles. It is hard to make reader understand the main goal of this work from this manuscript.

Response: Thank you for your comments.

Our structure for writing this article is as follows: we firstly propose the solar PV/T heat pump system coupled with build-in PCM heat storage. Then, we design and evaluate the build-in PCM heat storage. Next, we therefore investigate the performance of solar PV/T heat pump system coupled with PCM-TES. Finally, we conduct the feasibility analysis and merits of the proposed system.

The main goal of this paper which has been added in Line 102-104 is to provide a promising method to realize stable, high efficiency, environmental friendly residential heating in high latitude area with no energy consumption from power grid.

According to your comments, several paragraphs have been deleted. The revised parts in manuscript are as follows:

1. Line 104-106.
2. Line 160-161.
3. Line 164-165.

4. Line 248-255.
5. Line 303-308.
6. Line 606-611.
7. Line 625-628.

Reviewer 5

The authors have made clear responses. Now the present manuscript is acceptable for publication.

Response: Thank you for your comments.

Performance analysis of solar assisted heat pump coupled with build-in PCM heat storage based on PV/T panel

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Abstract

PV/T (photovoltaic/thermal) technology is a combination of PV module (photovoltaic utilization) and collector (photothermal utilization), which can improve the comprehensive utilization efficiency of solar energy and has a broad application prospect. In this paper, PV/T module is coupled with heat pump evaporator to form a direct-expansion solar PV/T heat pump which is suitable for heat application in high latitude area. To achieve stable residential heating, a solar PV/T heat pump system coupled with build-in PCM (phase change material) heat storage is therefore proposed and simulated. Meanwhile, the mathematical model of solar PV/T heat pump coupled with build-in PCM heat storage system is established and verified. The simulation results show that the temperature of underfloor heating which using build-in PCM heat storage can reach 22 °C to 31 °C after 39 hours when the circulating water is 40 °C. Moreover, the heating COP (Coefficient of Performance) increases with the increase of solar radiation, ambient temperature and area of PV/T collector, and decrease of wind speed, respectively. A 20 m² PV/T panel module can output 21.4% of the electricity to power grid when the solar radiation intensity is 600 W/m² and meet the heat demand of a 100 m² room while maintain the operation of the system. Meanwhile, the heating COP can reach 5.79 which is 70% higher than the conventional air conditioning system and the electrical, thermal, overall efficiencies are 17.77%, 55.76% and 75.49%, respectively.

1 Introduction

The total amount of energy consumption in the world is constantly climbing (2019; Caetano et al., 2017). The consumption of fossil energy has brought about energy crisis and environmental crisis (Pietrosemoli and Rodríguez-Monroy, 2019). Without action, CO₂ emissions from burning fossil fuels will be doubled by 2050 (Paolo Frankl, 2010). Therefore, the development and utilization of renewable energy has become one of the effective solutions (Keček et al., 2019). Solar energy has become the first choice due to its characteristics of ubiquity, abundance and sustainability (Kuik et al., 2019; Tsai, 2015), which is mainly used in two ways: photothermal and photovoltaic. 11% of global electricity will be provided by PV by 2050 (Paolo Frankl, 2010). However, the electrical efficiency of PV cells decreases with the increase of the temperature of PV cells (Huide et al., 2017). A cooling system can be added to reduce the temperature of PV cells while the remaining heat of PV panel are absorbed by working fluid which can be employed as a useful thermal energy for heat applications in buildings.

42 The PV/T technology coupled PV modules with thermal collectors was first proposed by
43 Wolf et al (Wolf, 1976) to reduce PV cells temperature and improve electrical efficiency. The
44 PV/T system can recover waste heat from the PV panel to improve comprehensive energy
45 utilization efficiency. PV/T design optimizations are carried out to improve the system efficiency
46 in recent years. Nahar et al. (Nahar et al., 2017) designed a novel pancake-shaped flow channel for
47 PV/T system, and integrated the flow channel with the PV baseboard. They found that the
48 temperature of the PV panel is reduced by 42 °C, and the electrical efficiency is increased by 2%.
49 Othman et al. (Othman et al., 2016) proposed a parallel, double pass flat plate collector which was
50 adopted in a two fluids PV/T system. Their results showed that the electrical efficiency and
51 thermal efficiency are 17% and 76%, respectively.

52 The combination of PCM and PV/T panel is an effective way to stabilize the operating
53 temperature of PV cells and improves the overall efficiency. Hosseinzadeh et al. (Hosseinzadeh et
54 al., 2018) investigated the effect of simultaneous use of nanofluid as coolant as well as an organic
55 paraffin as the phase change material on the electrical and thermal efficiencies. They demonstrated
56 that the use of PCM in nanofluid based PVT/PCM system enhances the thermal output power of
57 conventional PV/T system by 29.6%. Kazemian et al. (Kazemian et al., 2019) developed and
58 simulated a comprehensive three-dimensional model of PV/T system integrated with PCM. Their
59 simulation results presented that the PV/T-PCM system have lower surface temperature compared
60 to PV/T system, and as the thermal conductivity of PCM enhances, both electrical and thermal
61 efficiencies increase. Fayaz et al. (Fayaz et al., 2019) investigated the PCM based PV/T system,
62 and the experimental validation was carried out to verify the numerical model. They found that the
63 electrical efficiency is achieved as 13.98% and 13.87% numerically and experimentally
64 respectively, and the electrical performance is improved as 6.2% and 4.8% for PV/T-PCM system
65 based on the numerical and experimental results respectively.

66 Different working fluids like water, air, nanofluid and refrigerant are also used to cool the PV
67 module. Huang and Lee (Huang and Lee, 2004) conducted long-term tests on the direct-expansion
68 solar heat pump which adopted refrigerant as working fluid to verify the stability of the work. The
69 total running time of their prototype is over 20000 hours, and the measured energy consumption is
70 0.019 kWh/l of hot water at 57 °C which is much less than traditional solar water heater.
71 Stojanović and Akander (Stojanović and Akander, 2010) used direct-expansion heat pump for
72 independent buildings heating and domestic hot water supply. In their system, the collector area is
73 42.5 m² and heat pump power is 8.4 kW, they measured that the actual indoor temperature is no
74 less than 20 °C during the testing period. Alejandro Del Amo et al. (Del Amo et al., 2019) verified
75 the feasibility of solar PV/T heat pump through experiments. They obtained that the highest COP
76 of the system can reach 4.62. Meanwhile, the PV module provides 67.6% of the power demand,
77 and the payback period is 6 years.

78 In addition to optimize the PV/T panel, the adoption of PCM as heat storage is also a good
79 way to stabilize the system. Kuznik et al. (Kuznik et al., 2008) adopted PCM wallboard heat
80 storage and conducted comparative experiments. In their study, the system can effectively reduce
81 heat loss, keep the room warm and improve indoor thermal comfort. Fiorentini et al. (Fiorentini et
82 al., 2015) combined PCM storage with PV/T system, and the roof was used as PV/T layout
83 location. The PCM storage adopted in their system can keep indoor comfort within a certain and
84 potentially variable thermal comfort range. Diallo et al. (Diallo et al., 2019) proposed the
85 PVT-LHP (PVT Loop Heat Pipe) technology employing PCM triple heat exchanger, the total

86 energy efficiency of the presented system is improved by 28%, and the heating COP is 2.2 times
87 than that of a traditional PV/T system.

88 Owing to the instability of solar energy, traditional solar PV/T system cannot continuously
89 and stably supply heat or power generation when solar irradiation is weak such as rainy day or
90 winter. Consequently, the market of PV/T technology compared with PV or PT system is still very
91 low. PV/T can adapt to the characteristics of low intensity, instability and intermittency of solar
92 energy better if it can be combined with accumulator and heat storage. However, additional space
93 is required to install heat storage tank, which is not suitable for use in urban areas where land
94 resources are scarce. Therefore, in this paper, a coupling design of solar PV/T heat pump and
95 build-in PCM heat storage is proposed and the parallel air source heat exchanger is also adopted to
96 enhance the stability of the system. The build-in PCM heat storage used for underfloor heating is a
97 combination of PCM and building materials, which can save more space compared to
98 conventional PCM storage tank system. Firstly, the composition and operation modes of the
99 system are introduced. According to the system principle, the mathematical model is established
100 and verified, and the build-in PCM heat storage sub-system which using for residential heating is
101 also proposed and simulated. Then the influences of different parameters on system performance
102 are analyzed. Finally, the feasibility analysis of the system is conducted. The objective of this
103 paper is to provide a promising method to realize stable, high efficiency, environmental friendly
104 residential heating in high latitude area with no energy consumption from power grid.~~The~~
105 ~~proposed and calculated analysis of the scheme is conducive to assisting in the design and~~
106 ~~optimization of the solar PV/T heat pump system coupled with build-in PCM heat storage.~~

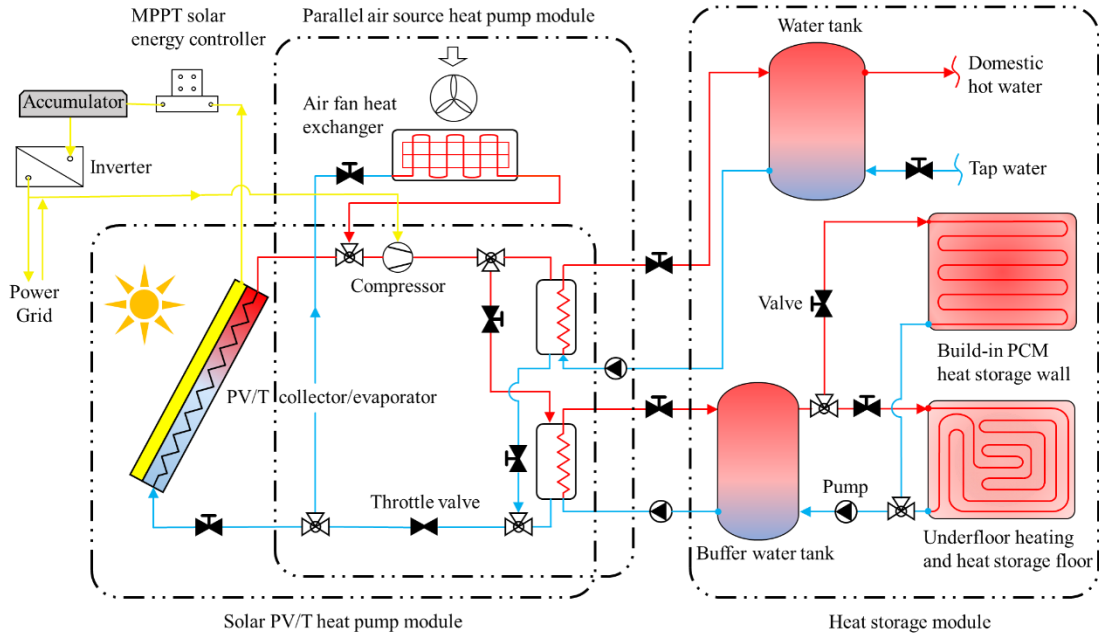
107 2 System description

108 Fig. 1 shows the schematic diagram of the system based on solar PV/T heat pump, which is
109 consisted of four main parts: solar PV/T heat pump module, parallel air source heat pump module,
110 heat storage module and electrical module. The blue lines represent low temperature working fluid,
111 and in the opposite, red lines represent high temperature. The yellow lines represent the electricity
112 flow direction. The arrows show the working fluid direction. The system can be divided into two
113 operating modes, which are listed as follows.

114 (1) Sunny day operating mode: The electricity generated by PV panel is used to drive the
115 pumps and compressor, and the excess electricity will be recovered to the power grid or drive the
116 air conditioning. However, the PV panel heated by solar radiation will be resulted in an increase of
117 the temperature of PV cells. Meanwhile, the heat transferred by the PV/T collector can be
118 absorbed by the refrigerant. Then, the superheated refrigerant vapor with low pressure is
119 compressed by the compressor to the high temperature and pressure refrigerant vapor. The
120 condensation heat will be absorbed and stored in the build-in PCM heat storage. The heat
121 transferred by condensation can also be used for producing domestic hot water. The liquid
122 refrigerant will expand through the throttle valve after condensation process, and flow into the
123 PV/T collector/evaporator. The heat released from the heat storage module will be used to keep
124 the indoor temperature constant during the night.

125 (2) Rainy day operating mode: The system will switch to the air source heat pump mode
126 when the solar radiation is insufficient to maintain system operation. The pumps and compressor
127 driven by power grid are used to keep the system work. The air fan heat exchanger is adopted to
128 absorb heat from the ambient air. The refrigerant will be heated by the air fan heat exchanger and

129 compressed by compressor into high temperature and pressure vapor. The heat released by the
 130 refrigerant vapor will be transferred to the PCM or water. This mode can make full use of the
 131 valley electricity to store heat at night, and maintain the indoor temperature through the heat
 132 storage module during the day.

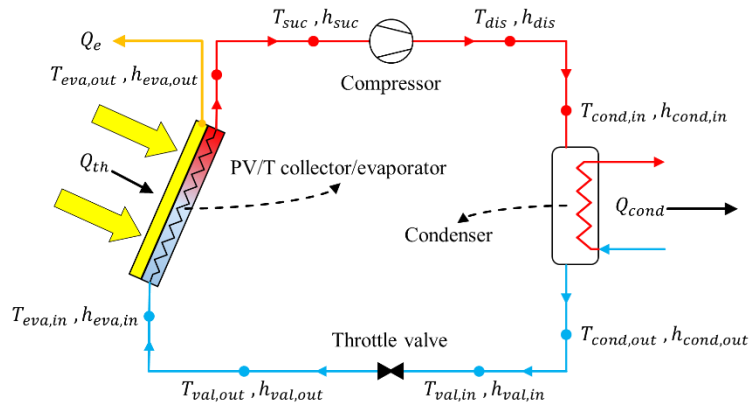


133
 134
 135

Fig. 1. Schematic of the system based on solar PV/T heat pump.

136 3 Mathematical model

137 The thermodynamic state points for each process are shown in Fig. 2. The solar PV/T heat
 138 pump cycle could be simplified to four components: PV/T collector/evaporator, compressor, PCM
 139 heat exchanger and throttle valve. Different temperature (T) and enthalpy (h) at each state point
 140 are shown in Fig. 2. Q_{th} (W) is the heat transfer rate between refrigerant and the PV/T panel, Q_e
 141 (W) is the electrical power provided by PV panel, and Q_{PCM} (W) is the heat transfer rate between
 142 refrigerant and phase change materials.



143
 144
 145

Fig. 2. The thermodynamic state points for each component.

146 The design parameters of the system and characteristics of different PV/T layers are listed in
 147 Table. 1.

Table. 1. Design parameters of the system and characteristics of different PV/T layers.

Parameters	Nomenclature	Value	Unit
Thickness of PV-glazing cover	$\delta_{g,pv}$	1	mm
Emissivity of PV-glazing cover	ε_c	0.84	[-]
Transmissivity of PV-glazing cover	$\tau_{g,pv}$	0.9	[-]
Thickness of PV cells	δ_{pv}	0.3	mm
Emissivity of PV cells	ε_p	0.96	[-]
Absorptance of PV cells	a_p	0.85	[-]
Thermal conductivity of PV cells	κ_p	203	W/m·K
Absorptance of PV baseboard	a_b	0.8	[-]
Thickness of EVA grease	δ_{EVA}	0.5	mm
Thermal conductivity of EVA grease	κ_{EVA}	0.311	W/m·K
Thickness of electrical insulation	δ_{ei}	0.5	mm
Thermal conductivity of electrical insulation	κ_{ei}	0.15	W/m·K
Electrical insulation material	[-]	Tedlar	[-]
Packing factor	β_p	1	[-]
Length of PV/T collector/evaporator	L	2.0	m
Width of PV/T collector/evaporator	W	1.0	m
Area of the PV/T collector/evaporator	A	2.0	m ²
Thermal conductivity of roll-bond panel	κ_{rb}	151	W/m·K
Thickness of roll-bond panel pipe	δ_{rb}	1	mm
Refrigerant type	ref	R134A	[-]

149

150 **3.1. Model of PV/T collector/evaporator**

151 The heat absorbed by the PV/T panel is expressed as follows:

$$152 \quad Q_{abs} = (1 - \eta_e) \cdot A \cdot I \cdot \tau_{g,pv} \cdot [\alpha_p \cdot \beta_p + \alpha_b \cdot (1 - \beta_p)] \quad (1)$$

153 where A is the collector area of the PV/T panel (m²); I is the solar radiation intensity (W/m²); $\tau_{g,pv}$
154 is the transmittances of the PV-glazing cover; a_p and a_b are the absorption ratios of the PV cells
155 and its baseboard, respectively; β_p is the packing factor of PV cells; η_e is the PV cells' efficiency,
156 calculated by (Huide et al., 2017):

$$157 \quad \eta_e = \eta_{rc} \cdot [1 - \beta_{pv} \cdot (T_p - T_{rc})] \quad (2)$$

158 η_{rc} is the reference photovoltaic efficiency value of PV cells at $T_{rc}=298$ K, $\eta_{rc}=0.18$; β_{pv} is the
159 temperature coefficient (1/K) of PV cell efficiency, $\beta_{pv}=0.0045$ (Huide et al., 2017).

160 ~~The heat loss, physical model of the PV/T collector/evaporator, the front view of PV/T panel~~
161 ~~on the roof and the roll bond panel which encapsulated in the PV/T panel are shown in Fig. 3. Fig.~~
162 ~~3 shows the heat loss and physical model of PV/T panel which has a multi-layer structure, the heat~~
163 ~~loss of PV/T panel consists of two parts: (1) heat transfer from PV cells to PV-glazing cover; (2)~~
164 ~~heat transfer from PV-glazing cover to ambient air. The thermal resistances of the PV/T module are~~
165 ~~shown in Fig. 3(a), and all consist of convection and radiation thermal resistance in different part.~~

166 The total heat loss rate of PV/T module is given as:

167

$$Q_L = U_L \cdot A \cdot (T_p - T_a) \quad (3)$$

168

where T_p and T_a are the temperature of PV cells and ambient air, respectively. U_L is the overall

169

heat loss coefficient which can be written as:

170

$$U_L = \left[1 / (h_{cv,p-c} + h_{rd,p-c}) + 1 / (h_{cv,c-a} + h_{rd,c-a}) \right]^{-1} \quad (4)$$

171

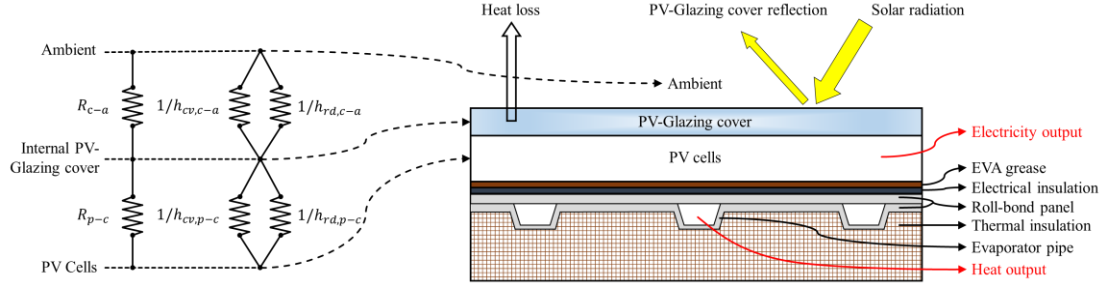
$h_{cv,p-c}$ and $h_{rd,p-c}$ are the convective and radiative heat-transfer coefficients between PV cells and

172

glass cover; $h_{cv,c-a}$ and $h_{rd,c-a}$ are the convective and radiative heat-transfer coefficients between glass

173

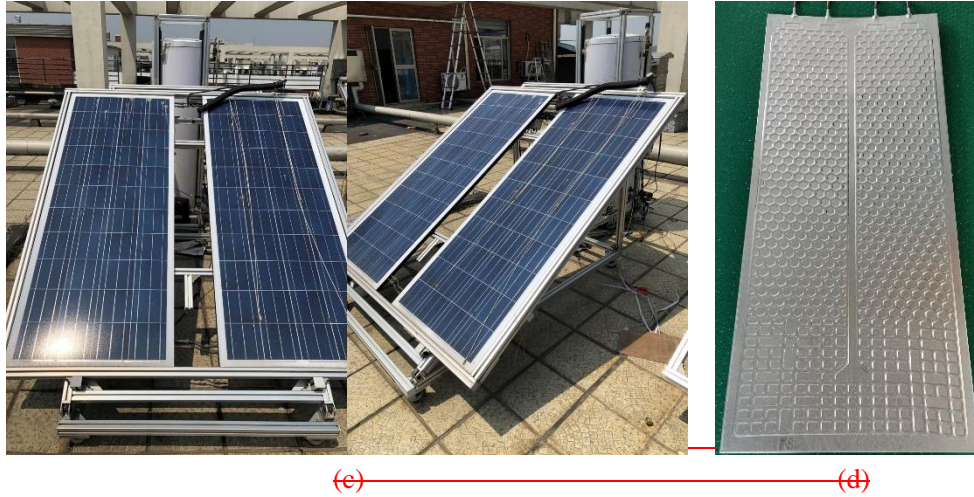
cover and ambient.



174

175

Fig. 3. Heat loss and physical model of PV/T collector/evaporator.



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179

180

Fig. 3. (a) Heat loss of the PV/T collector/evaporator. (b) Physical model of the PV/T collector/evaporator. (c) Front view of PV/T panel on the roof. (d) Roll bond panel.

181

The overall electricity output power of PV cells is given as:

182

$$Q_e = A \cdot I \cdot \tau_{g,pv} \cdot \alpha_p \cdot \beta_p \cdot \eta_e \quad (5)$$

183

Under the steady-state condition, the heat transfer rate delivered by the module equals the rate of the absorbed heat minus the overall heat loss, expressed as:

185

$$Q_{th} = Q_{abs} - Q_L \quad (6)$$

186

The total useful solar heat received by the PV/T collector/evaporator is expressed as:

187

$$Q_u = F_R \cdot A \cdot I \cdot \tau_{g,pv} \cdot \left[\alpha_p \cdot \beta_p + \alpha_b \cdot (1 - \beta_p) \right] \quad (7)$$

188

where F_R is the PV/T collector thermal efficiency factor, can be defined as (Diallo et al., 2019):

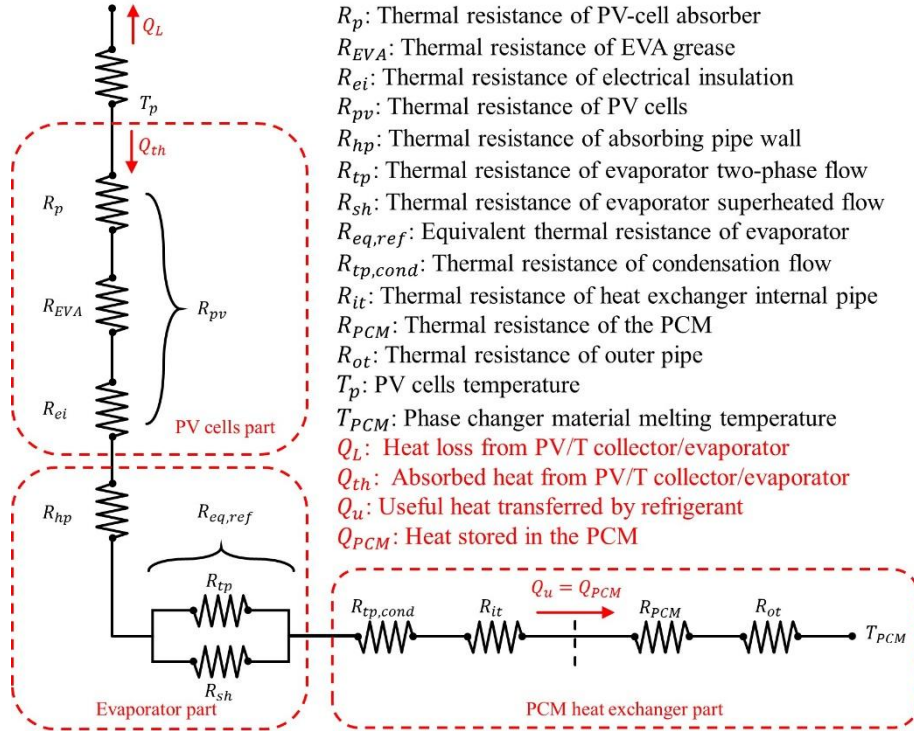
$$F_R = (1 - \eta_e) \cdot \frac{1/U_L}{L \cdot W / N_{rb} \cdot \left\{ 1 / \left(L \cdot U_L - \left[(W / N_{rb} - L_{rb}) F_{rb} + L_{rb} / (1 + R_{pv} \cdot U_L) \right] \right) + \sum_1^5 R_i \right\}} \quad (8)$$

190 N_{rb} is the equivalent number of roll-bond panel pipe; L_{rb} is the equivalent length of roll-bond panel
 191 pipe; $\sum R_i$ is the overall thermal resistance from the PV cells to the PCM; F_{rb} is the efficiency of the
 192 roll-bond panel which encapsulated in the backside of the PV/T panel which can be defined as
 193 (Diallo et al., 2019):

$$F_{rb} = \tanh \left[\frac{\sqrt{\frac{U_L}{k_{rb} \delta_{rb} (1 + R_{pv} U_L)}} (W / N_{rb} - L_{rb}) / 2}{\sqrt{\frac{U_L}{k_{rb} \delta_{rb} (1 + R_{pv} U_L)}} (W / N_{rb} - L_{rb}) / 2} \right] \quad (9)$$

195 k_{rb} is the thermal conductivity of roll-bond panel; δ_{rb} is the thickness of the roll-bond panel pipe;
 196 R_{pv} is the thermal resistance of PV cells.

197 The thermal resistances of the PV/T system and heat transfer along the system are shown in
 198 Fig. 4.



199
 200 **Fig. 4.** Thermal resistances of the PV/T system and heat transfer along the system.
 201

202 Heat transfer between PV module and heat pipe is a conventional one dimensional heat
 203 conduction process and its associated thermal resistance is:

$$R_{pv} = \delta_p / k_p + \delta_{EVA} / k_{EVA} + \delta_{ei} / k_{ei} \quad (10)$$

205 Superheated region and two-phase region existed in the refrigerant side of the PV/T
 206 collector/evaporator. The equivalent thermal resistance of the two different regions can be
 207 calculated as (P. Hartnett and M. Rohsenow, 1973):

$$R_{eq,ref} = \left(1 / R_{tp} + 1 / R_{sh} \right)^{-1} \quad (11)$$

209

210 **3.2. Model of build-in PCM heat storage**

211 All the heat gained by the build-in PCM heat storage is transferred by the PCM heat
 212 exchanger. The heat store rate in the PCM and concrete is expressed as follows:

$$213 \quad Q_{PCM} + Q_{conc} = m_{ref} \cdot (h_{dis} - h_{in}) \quad (12)$$

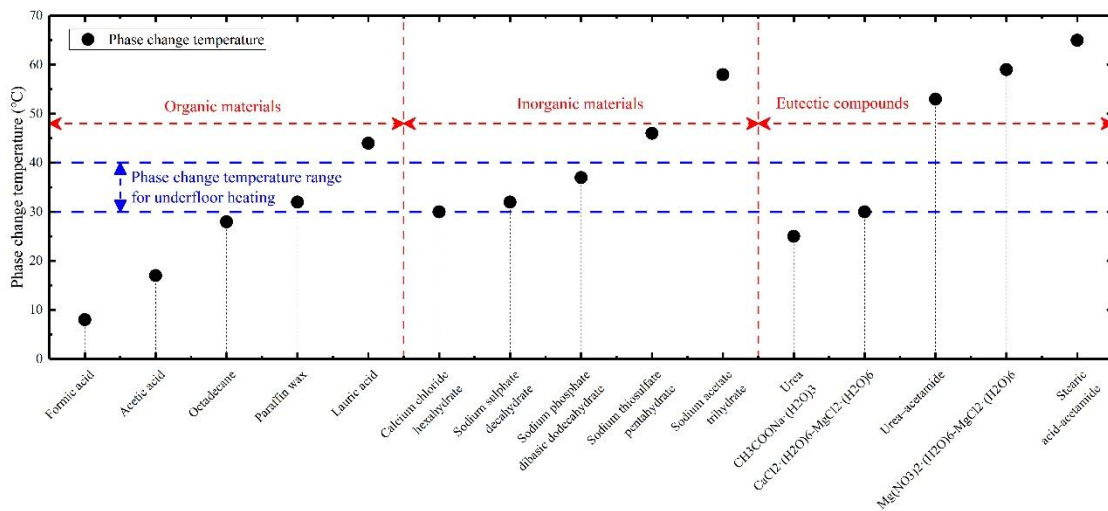
214 The PCM melting rate is calculated by:

$$215 \quad \dot{m} = Q_{PCM} / \Delta H_{PCM} \quad (13)$$

216 where ΔH_{PCM} is the latent heat of the PCM (kJ/kg).

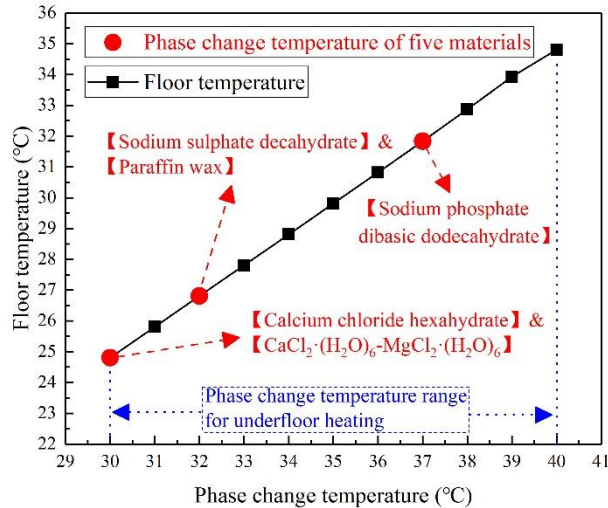
217 Fig. 5 shows the phase change temperature of 15 kinds of phase change materials including
 218 organic/inorganic/eutectic compounds materials. According to L.F. Cabeza et al. (Cabeza et al.,
 219 2011), the recommendation phase change temperature range for underfloor heating is between
 220 30 °C to 40 °C. There are five kinds of PCM that have phase change temperature in this range:

221 Paraffin wax, $\text{CaCl}_2 \cdot (\text{H}_2\text{O})_6$ - $\text{MgCl}_2 \cdot (\text{H}_2\text{O})_6$, Calcium chloride hexahydrate, Sodium sulphate
 222 decahydrate, Sodium phosphate dibasic dodecahydrate.



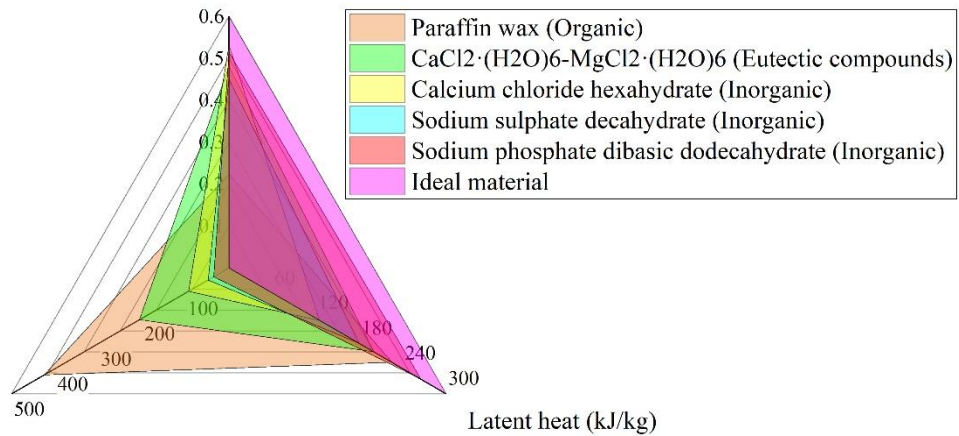
223
 224 **Fig. 5.** Phase change temperature of 15 kinds of PCM including Organic/Inorganic/Eutectic
 225 compounds materials.
 226

227 The variation curve of floor temperature with phase change temperature and the comparison
 228 of three indices including thermal conductivity, latent heat and price (Pereira da Cunha and Eames,
 229 2016) of above five kinds of PCM are shown in Fig. 6. These five materials have different phase
 230 change temperature which all in the recommendation temperature range for underfloor heating.
 231 The ideal material should have a higher thermal conductivity and latent heat while the price is low.
 232 In order to evaluate these five materials, the graph of ideal material is also plotted in Fig. 6. The
 233 organic and eutectic compounds phase change materials have a higher price than inorganic
 234 materials while the thermal conductivity and latent heat are lower. Moreover, the larger of the
 235 overlap area in Fig. 6 between each material and ideal material, the better of the material
 236 performance. As shown in Fig. 6, Sodium phosphate dibasic dodecahydrate has the largest overlap
 237 area, thus, it is used in the simulation of build-in PCM heat storage unit.



(a)

Thermal conductivity (W/m·K)

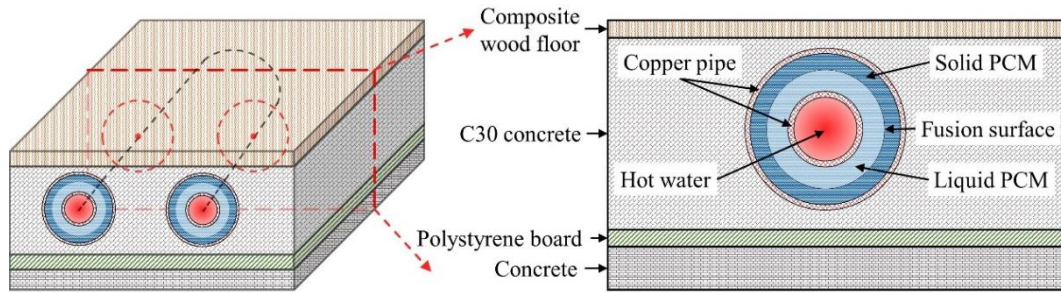


(b)

Fig. 6. (a) Variation curve of floor temperature with phase change temperature. **(b)** Thermal conductivity/Latent heat/Price comparison of six kinds of materials including ideal material.

Fig. 7 illustrates the structure and cross-section view of a build-in PCM heat storage unit which is the component of underfloor heating module. The phase change materials (Sodium phosphate dibasic dodecahydrate) are placed in the outer tube of a double-wall tube which can store heat during the day and release latent heat at night. ~~The floor temperature range of the underfloor heating is 20 °C to 35 °C for residential applications. The phase change materials should have a melting temperature lying in the practical range of operation, melt congruently with minimum subcooling and be chemically stable, low in cost, nontoxic and non-corrosive. Moreover, as shown in Fig. 8(b), the outer tube of the build-in PCM heat storage is around 37 °C when the circulating water temperature is 40 °C. Thus, it is reasonable to choose a phase change material which melting point is around 37 °C. The cross-section view of the build-in heat storage unit is shown in Fig. 7(b).~~ Polystyrene board and foam concrete are used as thermal insulation and building materials, respectively. The hot water which produced by solar PV/T heat pump module is pumped into the pipe of underfloor heating to keep the indoor temperature steady. The properties of PCM (Jankowski and McCluskey, 2014) and concrete as well as underfloor heating

259 working conditions are listed in Table. 2.



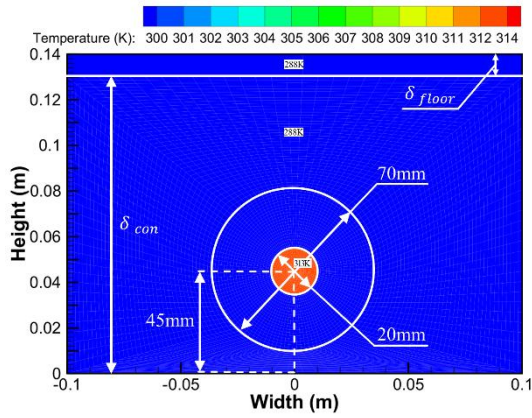
260
261 **Fig. 7.** Structure and cross-section view of the build-in PCM heat storage unit.

262
263 **Table. 2.** Properties of PCM and concrete as well as underfloor heating working conditions.

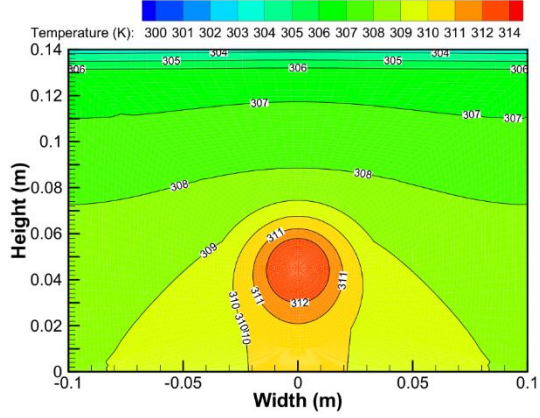
Parameters	Nomenclature	Value	Unit
Type of PCM	[-]	Na ₂ HPO ₄ ·12H ₂ O	[-]
Latent heat of PCM	ΔH_{PCM}	265	kJ/kg
Density of PCM	ρ_{PCM}	1507	kg/m ³
Temperature of transition of PCM	T_{mel}	37	°C
Specific heat at constant pressure of PCM	C_{p-PCM}	1.69	kJ/kg·K
Thermal conductivity of PCM	κ_{PCM}	0.514	W/m·K
Thermal conductivity of copper coil	κ_C	397	W/m·K
Type of concrete	[-]	C30	[-]
Specific heat at constant pressure of concrete	C_{p-conc}	0.97	kJ/kg·K
Thermal conductivity of concrete	κ_{con}	1.6	W/m·K
Density of concrete	ρ_{conc}	2300	kg/m ³
Thickness of the concrete	δ_{con}	0.13	m
Thickness of the wood floor	δ_{floor}	0.01	m
Volume flow rate of circulating water	V_{cw}	2.5	m ³ /h

264

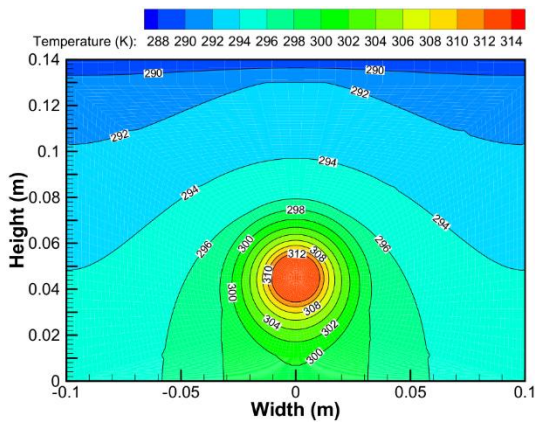
265 Fig. 8 (a-~~lc~~) presents the simulation results of heat transfer in build-in PCM heat storage unit
 266 at different time which carried out by software of Ansys Fluent 17.0. The thermal conductivity of
 267 the heat transfer between the coil and PCM has been considered in the setup of the boundary
 268 conditions. Thus, the boundary conditions of inner and outer tube have been set as “coupled” in
 269 Ansys Fluent 17.0 which means the solution of heat transfer process would be carried out through
 270 the coupled method in the program. The initial temperature of the heat storage unit is 15 °C, and
 271 the underfloor heating can reach above 30 °C after 39 hours when the circulating water is 40 °C. As
 272 shown in Fig. 8(~~md~~), the floor temperature increases with time and reaches 25 °C in the first 12
 273 hours which has 10 °C difference with the ambient. The changing curve mountains rapidly in the
 274 initial stage and becomes steady after 39 hours, the build-in PCM heat storage would supply heat
 275 to indoor area and maintain the room temperature.



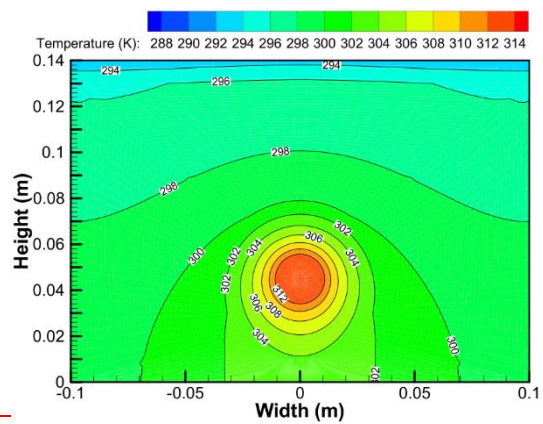
(a) Initial stage



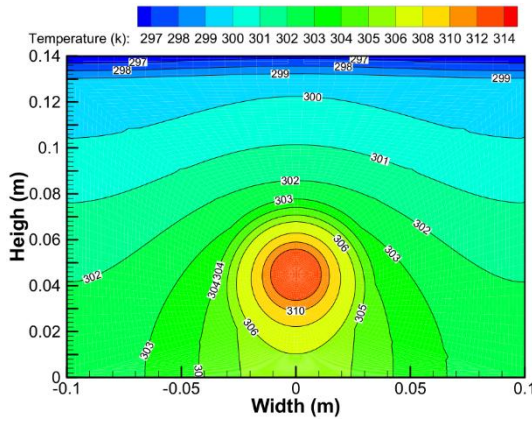
(b) Steady stage



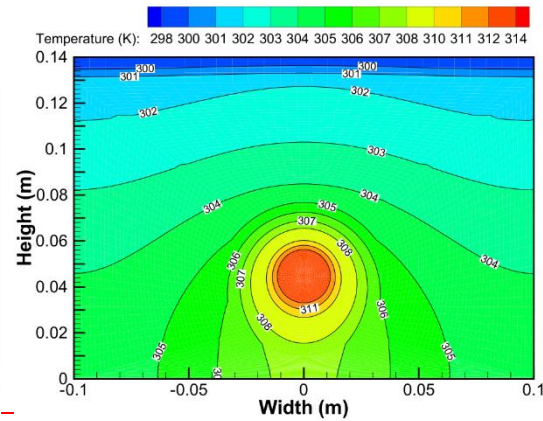
(c) 3 hours



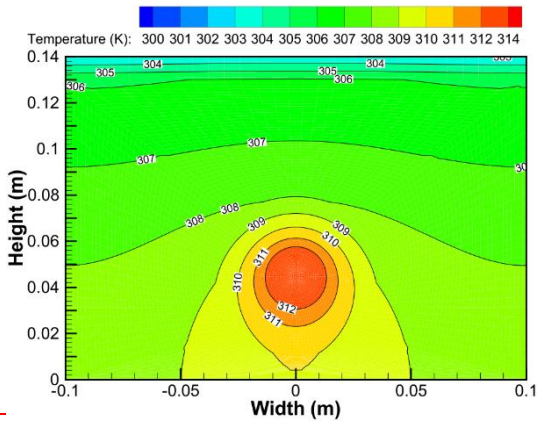
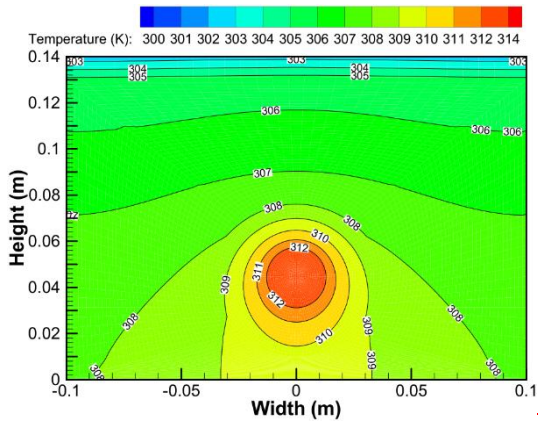
(d) 6 hours



(e) 9 hours

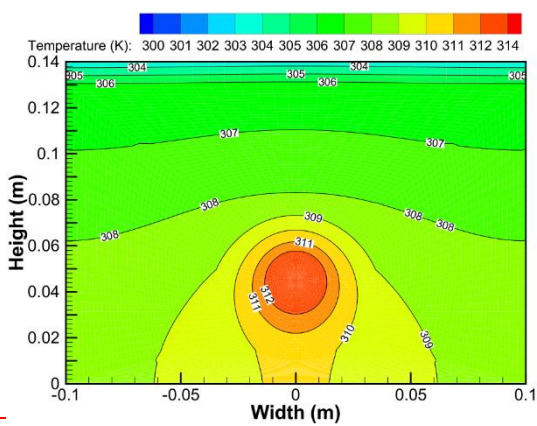
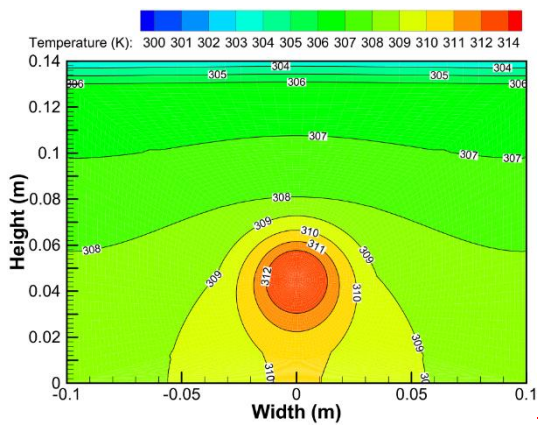


(f) 12 hours



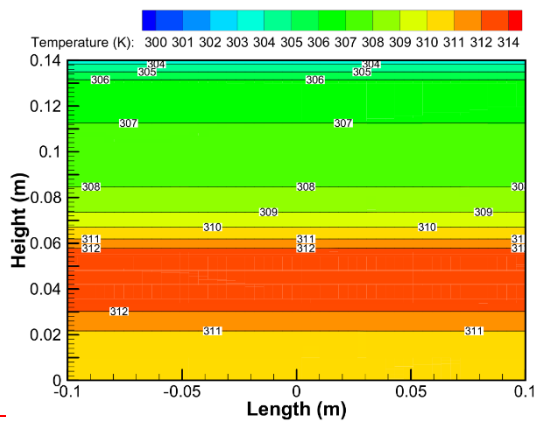
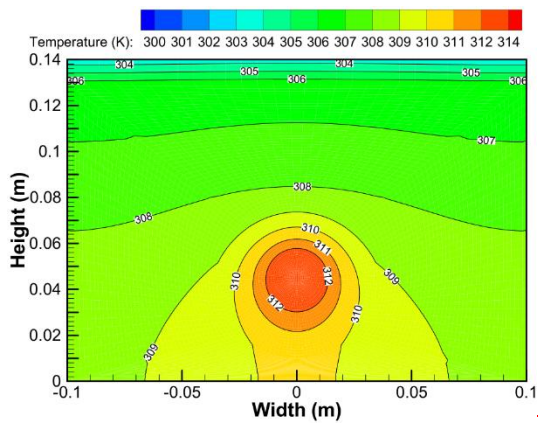
(g) 24 hours

(h) 30 hours



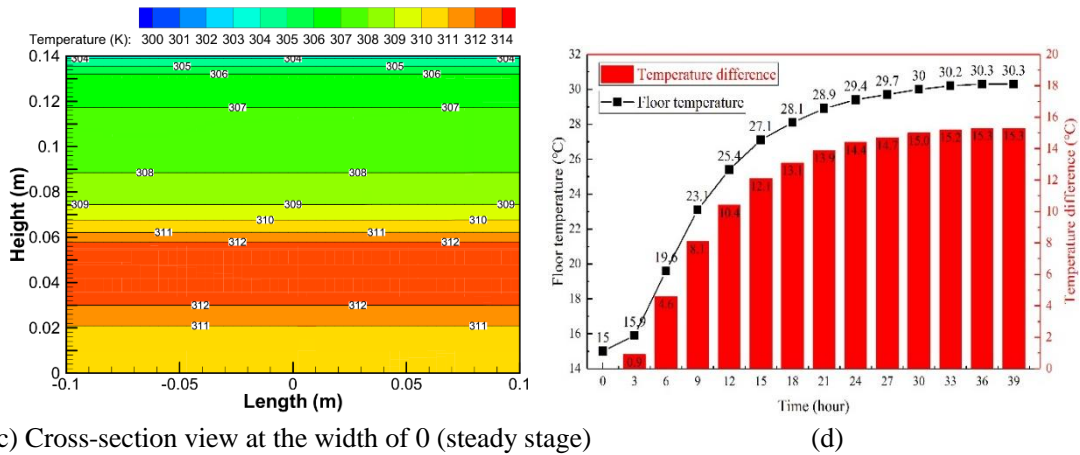
(i) 33 hours

(j) 36 hours



(k) 39 hours

(l) Cross-section view at the width of 0 (39 hours)



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(c) Cross-section view at the width of 0 (steady stage)

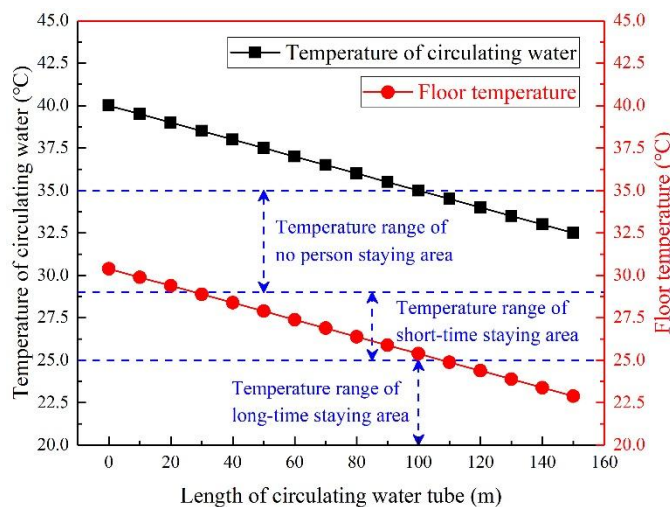
(d)

Fig. 8. (a-d) Cross-section temperature contour of the build-in PCM heat storage unit at different time. (b) Variation curve of floor temperature and temperature difference between floor and ambient.

Fig. 8. (a-c) Cross-section temperature contour of the build-in PCM heat storage unit at initial and steady stage. (d) Variation curve of floor temperature and temperature difference between floor and ambient.

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Fig. 9 shows the variation curves of circulating water temperature and floor temperature with the length of circulating water tube. The temperature of circulating water would decrease from 40 °C (inlet) to 32.5 °C (outlet) while the floor temperature varies from 30.4 °C to 22.9 °C. This figure also presents the three temperature range of underfloor heating which can be divided into three categories: Temperature range of no person staying area (29 °C-35 °C), Temperature range of short-time staying area (25 °C-29 °C) and Temperature range of long-time staying area (20 °C-25 °C). For a typical family, the heating area is about 100 m² which needs about 150 meters of underfloor heating tube (circulating water tube). As shown in Fig. 9, the first 30 meters tube can be arranged in the no person staying area (storage room etc.) while 30 to 110 meters tube can be arranged in the short time staying area (living room, kitchen etc.), and 110 to 150 meters tube can be arranged in the long time staying area (bedroom etc.). Therefore, when using build-in PCM heat storage as underfloor heating, the system could meet the heat demand of users and keep indoor temperature steady.



311

312 **Fig. 9.** Variation curves of circulating water temperature and floor temperature with the length of
 313 circulating water tube.
 314

315 3.3. Model of compressor

316 The refrigerant mass flow rate \dot{m}_{ref} could be calculated by (Ma, 2013)

$$317 \quad \dot{m}_{ref} = \lambda \cdot V_{th} / v_{suc} \quad (14)$$

318 where the λ is the compressor volumetric efficiency, V_{th} is the theoretical displacement volume of
 319 compressor (m^3), v_{suc} is the specific volume of the refrigerant in the suction period (m^3/kg).

320 The power consumption of compressor is written as:

$$321 \quad P_{com} = \dot{m}_{ref} \cdot (h_{dis} - h_{suc}) / \eta_{ele} \quad (15)$$

322 where η_{ele} is the efficiency of the compressor which can be expressed by (Ma, 2013):

$$323 \quad \eta_{ele} = -0.17938 + 0.87501 \frac{P_{dis}}{P_{suc}} - 0.30014 \left(\frac{P_{dis}}{P_{suc}} \right)^2 + 0.04135 \left(\frac{P_{dis}}{P_{suc}} \right)^3 - 0.00206 \left(\frac{P_{dis}}{P_{suc}} \right)^4 \quad (16)$$

324 The refrigerant vapor through compressor is isentropic, thus the equation can be expressed
 325 by:

$$326 \quad T_{dis} / T_{suc} = (P_{dis} / P_{suc})^{(\chi-1)/\chi} \quad (17)$$

327 where the χ is the polytropic index of refrigerant. h_{dis} is the enthalpy of the vapor after compressed
 328 (kJ/kg), h_{suc} is the enthalpy of the vapor before compressed (kJ/kg).

329 The heat of refrigerant absorbed from the PV/T collector/evaporator equal to Q_u :

$$330 \quad \dot{m}_{ref} \cdot (h_{eva,out} - h_{eva,in}) = Q_u = Q_{th} \quad (18)$$

331 3.4 Definition of the system performance

332 The COP (Coefficient of Performance) of the system can be defined as the ratio of overall
 333 heat output of system and power consumption of the compressor as following (R. Turns, 2006)

$$334 \quad COP = \dot{m}_{ref} \cdot (h_{cond,in} - h_{cond,out}) / P_{com} \quad (19)$$

335 The η_{th} is the PV/T collector/evaporator's thermal efficiency, which can be defined as:

$$336 \quad \eta_{th} = Q_{th} / (A \cdot I) \quad (20)$$

337 The η_e is the PV cells electrical efficiency which can be defined as:

$$338 \quad \eta_e = Q_e / (A \cdot I) \quad (21)$$

339 The η_{ove} is the overall efficiency which can be defined as:

$$340 \quad \eta_{ove} = (Q_{cond} + Q_e) / (A \cdot I) \quad (22)$$

341 3.5 Presentation of the algorithm by flow chart

342 The numerical simulation procedure of the system is shown in Fig. 10 to predict the system
343 performance using the software of MATLAB. The solution steps are as flows:

- 344 (1) Input all the environmental parameters, such as solar radiation intensity, wind speed,
345 ambient temperature, etc.
- 346 (2) Input the system design parameters and operation parameters, such as collector area,
347 packing factor, collector slop, transmissivity of external glass cover, PV-glazing cover,
348 thickness of each layer, etc.
- 349 (3) Assume the temperature of PV cells T_p .
- 350 (4) Calculate the overall heat loss rate Q_L , and the PV electrical output power Q_e .
- 351 (5) Calculate the thermal energy gain rate through PV/T collector/evaporator Q_{th} , and the
352 useful heat transfer rate by refrigerant Q_u .
- 353 (6) Calculate $(Q_{th} - Q_u)/Q_{th}$. If $|(Q_{th} - Q_u)/Q_{th}| < 0.1\%$, the system achieves the heat balance
354 and move to next step.
- 355 (7) Input the superheat degree T_{sh} .
- 356 (8) Assume the compressor discharge pressure P_{dis} .
- 357 (9) Calculate the PV/T collector/evaporator inlet enthalpy $h_{eva,in}$ and the PCM heat exchanger
358 outlet enthalpy $h_{cond,out}$. If $|(h_{eva,in} - h_{cond,out})/h_{cond,out}| < 0.1\%$, the system achieves the
359 pressure balance and move to next step.
- 360 (10) Calculate the COP, PCM melting rate \dot{m} , thermal efficiency η_{th} , electrical efficiency η_e ,
361 and overall efficiency η_{ove} .
- 362 (11) Results output, and stop the program.

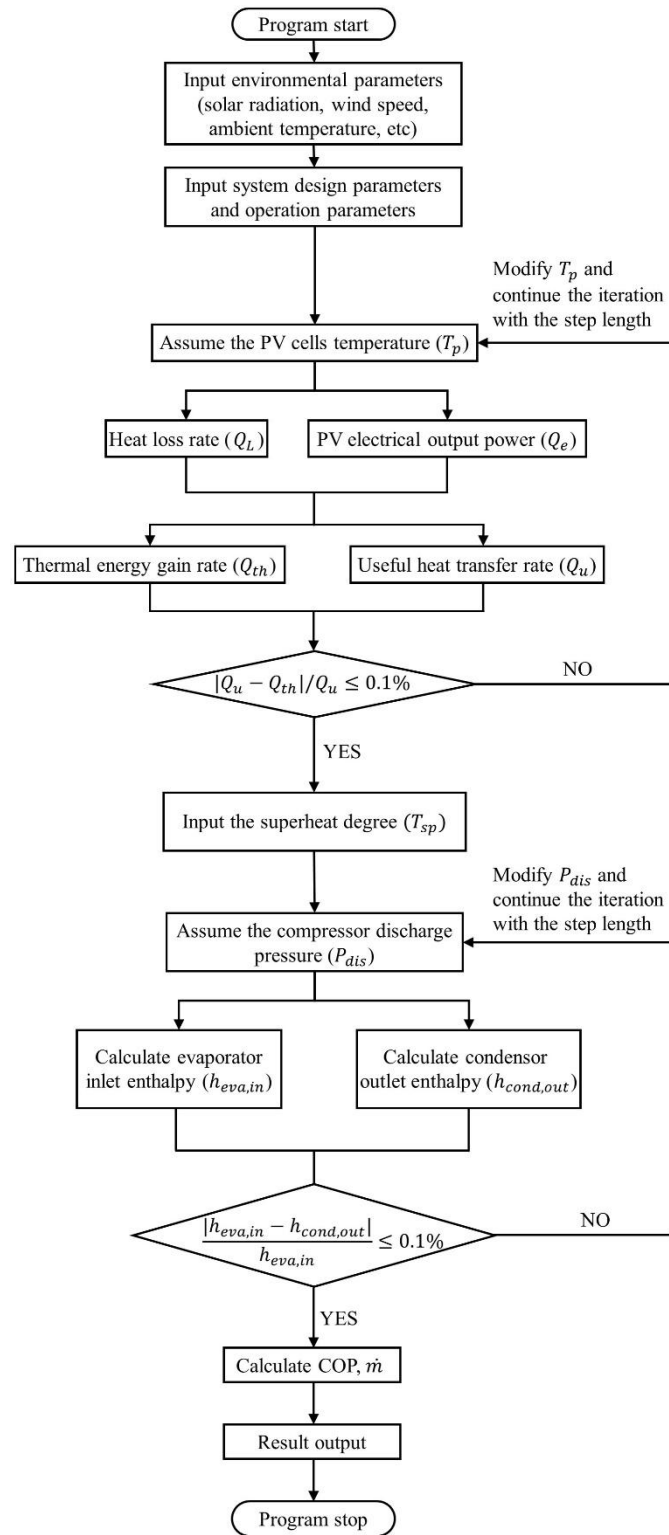


Fig. 10. Numerical solution procedure of the system.

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366 4 Validation of the model

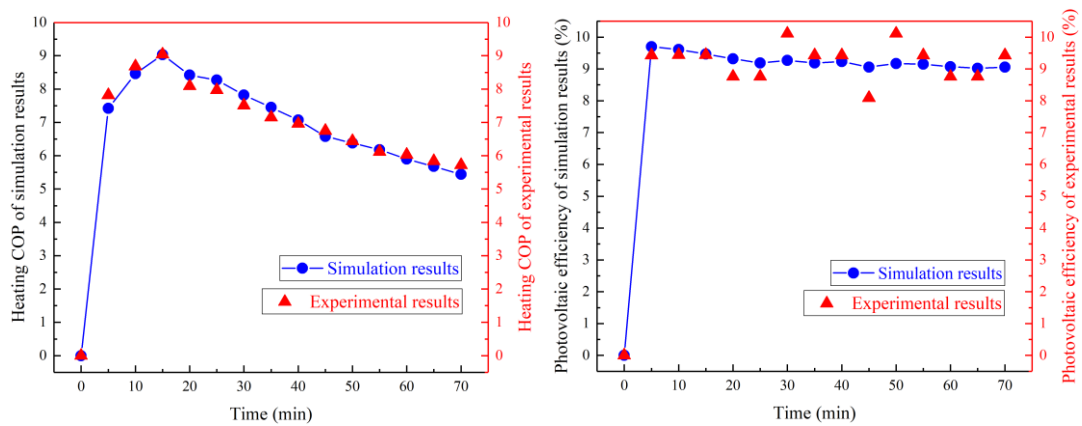
367 To ensure the reliability of the mathematic model, the simulation results should be compared
368 with the experimental results. The experimental parameters used in the simulation are listed in
369 Table. 3.

Table 3. Experimental parameters (Zhou et al., 2019).

Parameters	Nomenclature	Value	Unit
Thickness of glass cover	δ_g	3.2	mm
Thickness of air gap	δ_{air}	35	mm
Thickness of PV-glazing cover	$\delta_{g,pv}$	1	mm
Thickness of PV cells	δ_{pv}	0.2	mm
Thickness of EVA adhesive film	δ_{EVA}	0.4	mm
Thickness of electrical insulation	δ_{ei}	0.5	mm
Length of PV/T	L	3.0	m
Width of PV/T	W	1.6	m
Wind speed	v_{wind}	1.5	m/s
Ambient temperature	T_a	298.5	K
Refrigerant type	ref	R22	[-]
Packing factor	β_p	0.64	[-]

371

372 The comparison results of heating COP are presented in Fig. 11(a), the operating conditions
373 are refer from Zhou et al.(Zhou et al., 2019). Under the same system components (PV/T
374 collector/evaporator, compressor, condenser, expansion valve, water tank), the simulation results
375 are in good agreement with the experimental results. Heating COP of the PV/T system increases in
376 the first 15 minutes because the water in the tank still in the low temperature range. Thus, the
377 temperature differences between the refrigerant fluid and water remains large in the condenser
378 leading to a high heat transfer efficiency. However, the heat transfer efficiency of the condenser
379 would be decreased when the temperature of the inlet water rises up during the operation of the
380 whole system. That is the reason of the reduction of heating COP after 15 minutes. The average
381 error of heating COP is 2.84% while the maximum error is 5.12%. Fig. 12–11(b) presents the
382 experimental and simulation results of the photovoltaic efficiency. The maximum photovoltaic
383 efficiency is 10.11% while the minimum is 8.09%, but all the experimental photovoltaic
384 efficiencies fluctuate around 9.24%. The simulation photovoltaic efficiencies remain around 9.25%
385 but fluctuate from 9.01% to 9.7% due to the influence of solar radiation intensity. The average
386 error of photovoltaic efficiency is 4.48% while the maximum error is 9.30%.



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(a) (b)
Fig. 11. (a) Comparison results of experimental and simulated heating COP. (b) Comparison results of experimental and simulated photovoltaic efficiency.

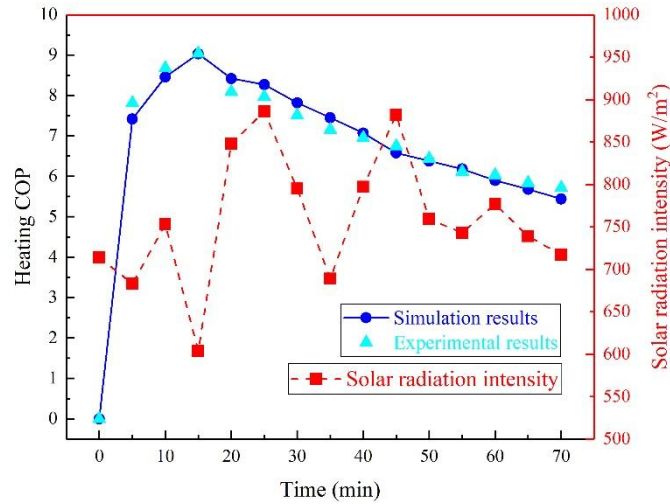


Fig. 11. Comparison results of experimental and simulated heating COP.

Fig. 12 presents the experimental and simulation results of the photovoltaic efficiency. The maximum photovoltaic efficiency is 10.11% while the minimum is 8.09%, but all the experimental photovoltaic efficiencies fluctuate around 9.24%. The simulation photovoltaic efficiencies remain around 9.25% but fluctuate from 9.01% to 9.7% due to the influence of solar radiation intensity. The average error of photovoltaic efficiency is 4.48% while the maximum error is 9.30%.

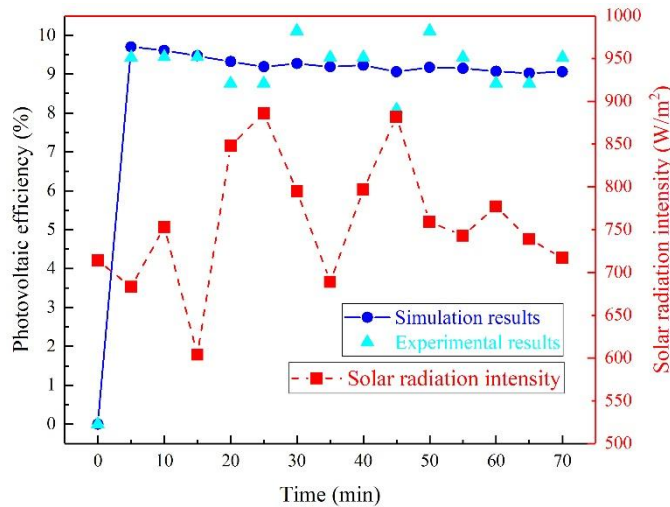


Fig. 12. Comparison results of experimental and simulated photovoltaic efficiency.

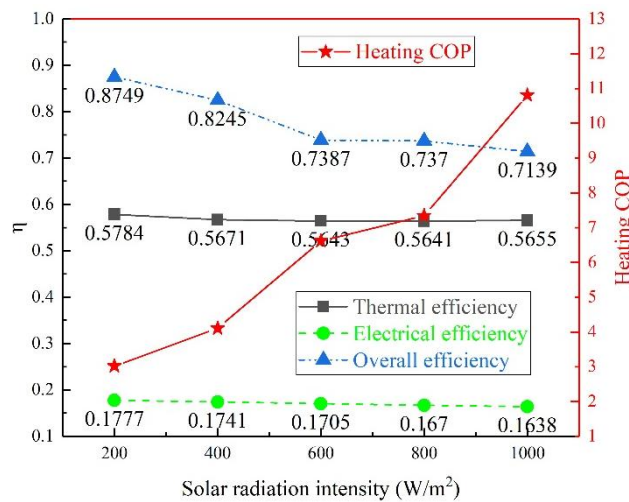
5 Parameter analysis

In this section, the influences of different parameters (solar radiation intensity, ambient temperature, wind speed, area of PV/T collector) on this system are investigated, and the performance indices of the system under typical working conditions are also given. It should be noted that when one parameter is varied, others keep constant. Pressure ratio of the compressor refers the ratio of pressure of discharged refrigerant vapor and charged refrigerant vapor.

5.1 Solar radiation intensity

409 The influences of solar radiation intensity which varying from 200 W/m² to 1000 W/m² are
 410 shown as follows at the working conditions are: ambient temperature is 25 °C, wind speed is 1.5
 411 m/s and area of PV/T collector is 2 m².

412 Fig. 12 presents the rising curve of the heating COP and declining curves of thermal,
 413 electrical and overall efficiencies. The heating COP is 3.0 for a solar radiation of 200 W/m², and it
 414 can reach up to 10.8 when the solar radiation is 1000 W/m². The refrigerant evaporation
 415 temperature and pressure will be increased due to a higher temperature of PV cells. Thus, the
 416 compressor consumes less electricity to compress the refrigerant vapor leading to this upward
 417 trend. Meanwhile, the heat loss will mount due to a higher temperature difference between PV
 418 cells and ambient. Thereby, the thermal and electrical efficiencies of the PV/T panel reduce
 419 resulting in a reduction of the overall efficiency.

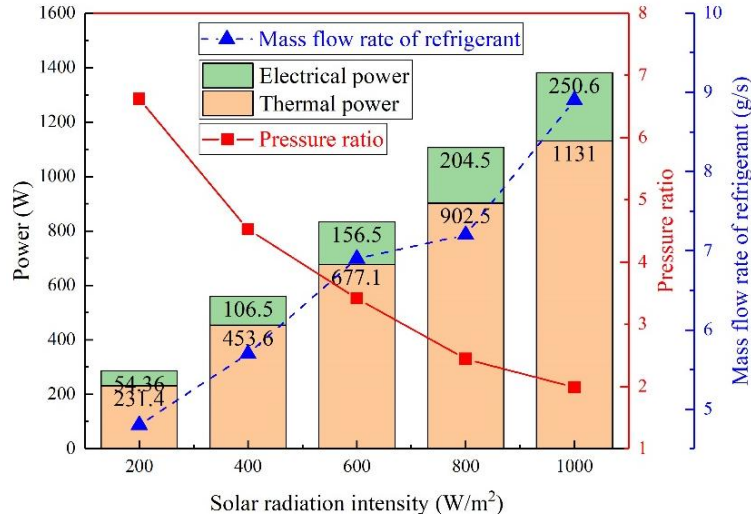


420

421 **Fig. 12.** Influence of solar radiation intensity on heating COP and thermal, electrical, overall
 422 efficiencies.

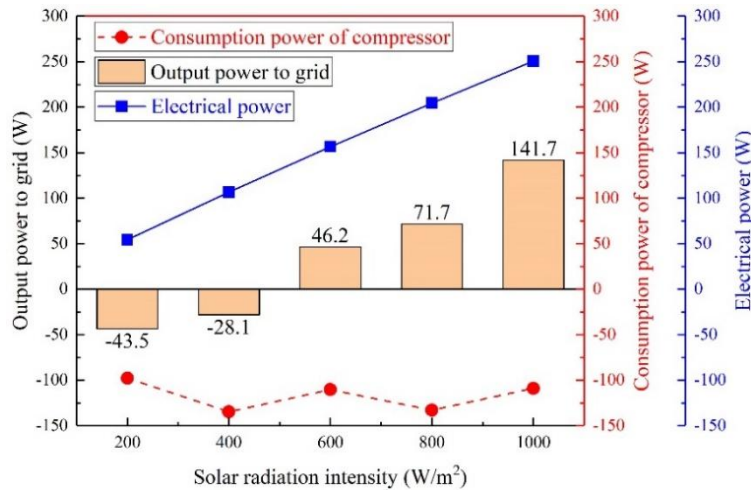
423

424 Fig. 13 shows the effect of solar radiation intensity on thermal and electrical output power,
 425 pressure ratio and mass flow rate of refrigerant. The thermal and electrical power keep mounting
 426 with the increase of the solar radiation intensity. Meanwhile, the pressure ratio of the compressor
 427 decreases and the mass flow rate of the refrigerant increases. That is because the increase of the
 428 evaporation temperature causes a higher evaporation pressure which equals to the suction pressure
 429 of compressor leading to a lower pressure ratio. Furthermore, a larger amount of refrigerant will
 430 be needed to transfer extra heat from PV/T collector/evaporator to PCM heat exchanger
 431 (condenser) when the PV/T panel absorbs more heat from solar radiation.



432
433 **Fig. 13.** Influence of solar radiation intensity on electrical and thermal power, pressure ratio and
434 mass flow rate of refrigerant.
435

436 As shown in Fig. 14, there is a positive linear correlation between PV cells' electrical power
437 generation and solar radiation intensity while the consumption power of compressor fluctuates
438 around 120 W. When the output power to grid is less than zero, it means the system consumes
439 electricity from the power grid. The electrical power generated by PV panels could meet the
440 demand of the compressor and the system could output electricity to power grid when the solar
441 radiation intensity exceeds 500 W/m².



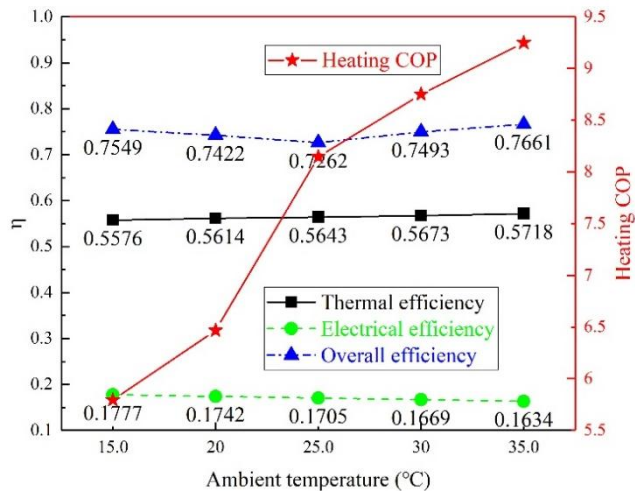
442
443 **Fig. 14.** Influence of solar radiation intensity on electrical power, consumption power of
444 compressor and output power to grid.
445

446 5.2 Ambient temperature

447 The influences of ambient temperature which varying from 15 °C to 35 °C are shown as
448 follows at the working conditions are: solar radiation intensity is 600 W/m², wind speed is 1.5 m/s
449 and area of PV/T collector is 2 m².

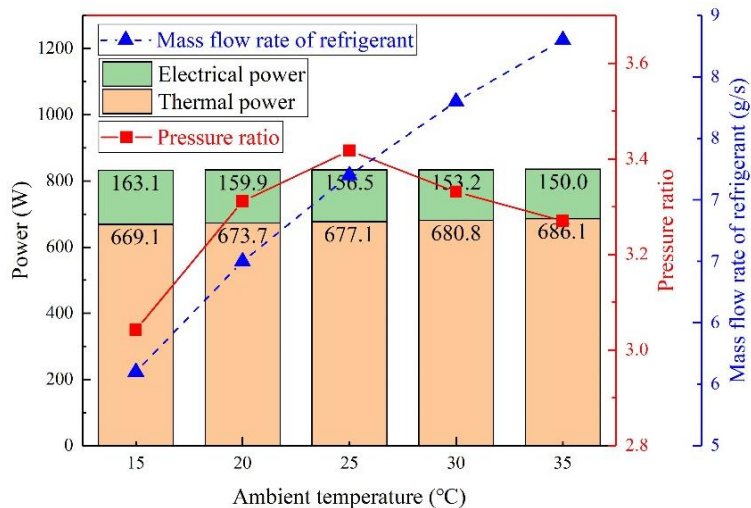
450 Fig. 15 shows that increasing the ambient temperature will increase the heating COP and
451 thermal efficiency but decrease the electrical and overall efficiencies. The temperature difference
452 between PV cells and ambient will reduce when the ambient temperature rises leading to a less

453 heat loss from PV/T panel to the surrounding. Thus, the heating COP increases and the electrical
 454 efficiency decreases due to a higher temperature of the PV cells while the thermal efficiency
 455 increases. However, the electrical efficiency outweighs the thermal efficiency resulting in a
 456 reduction of the overall efficiency when the ambient temperature is below 25°C. The heating COP
 457 (9.25) at 35 °C is higher than COP (5.79) at 15 °C by 59.8%, thus a higher ambient temperature is
 458 better for the system performance.



459
 460 **Fig. 15.** Influence of ambient temperature on heating COP and thermal, electrical, overall
 461 efficiencies.
 462

463 The variation curves of thermal and electrical power, pressure ratio and mass flow rate of
 464 refrigerant with the increase of ambient temperature are shown in Fig. 16. The changing curves of
 465 the electrical and thermal output power are the same as the electrical and thermal efficiencies. The
 466 pressure ratio of the compressor will increase when the ambient temperature is below 25 °C and
 467 decrease when the ambient temperature is over 25 °C. That is because a lower ambient temperature
 468 leads to a lower superheat degree of refrigerant which cause a lower pressure ratio, and in the
 469 opposite, a higher ambient temperature leads to a higher superheat degree. Moreover, when the
 470 ambient temperature exceeds 25 °C, it will influence the thermal efficiency of PV/T collector
 471 causing the reduction of pressure ratio. The mass flow rate of the refrigerant will keep climbing
 472 when the ambient temperature rises because less heat will lose in the ambient while more heat will
 473 be absorbed by refrigerant.

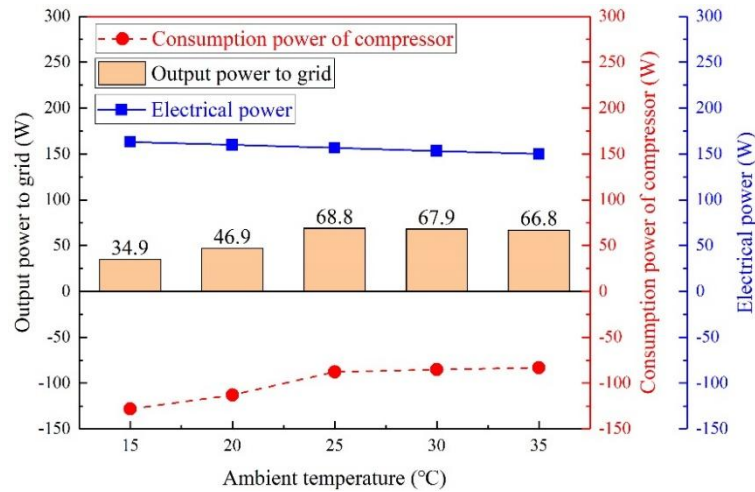


474

475 **Fig. 16.** Influence of ambient temperature on electrical and thermal power, pressure ratio and mass
 476 flow rate of refrigerant.

477

478 As shown in Fig. 17, the electrical power of PV decreases linearly with the ambient
 479 temperature while the consumption power of compressor decreases when the ambient temperature
 480 is below 25 °C and increases when the ambient temperature exceeds 25 °C. The output power to
 481 grid reaches its maximum at 25 °C. Because when the ambient temperature is below 25 °C, the
 482 effect of environmental heat loss is greater than that of heat-collecting efficiency due to a large
 483 temperature difference. However, the effect of a higher ambient temperature on heat-collecting
 484 efficiency is greater than that of environmental heat loss when the ambient temperature exceeds
 485 25 °C.



486

487 **Fig. 17.** Influence of ambient temperature on electrical power, consumption power of compressor
 488 and output power to grid.

489

490 5.3 Wind speed

491 The influences of wind speed which varying from 0.5 m/s to 2.5 m/s are shown as follows at
 492 the working conditions are: solar radiation intensity is 600 W/m², ambient temperature is 25 °C and
 493 area of PV/T collector is 2 m².

494 Fig. 18 shows the variation curve of heating COP, thermal, electrical and overall efficiencies
 495 with wind speed varying from 0.5 m/s to 2.5 m/s. The heating COP will decrease rapidly when the
 496 wind speed is low and steadily when the wind speed increases. More heat will be transferred to the
 497 environment and less heat is absorbed by the PV/T panel under a higher wind speed. Meanwhile,
 498 evaporation temperature of refrigerant will be decreased and the consumption power of
 499 compressor will be increased when the temperature of PV cells rises. Thus, the heating COP drops
 500 from 9.4 to 6.8, which means better wind protection measures should be taken to improve the
 501 system performance.

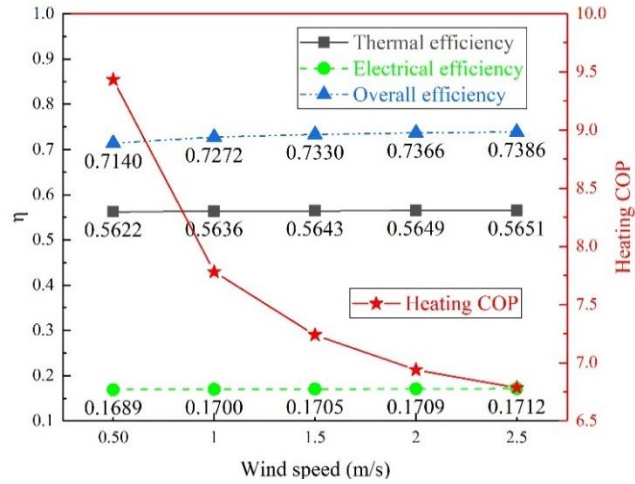


Fig. 18. Influence of wind speed on heating COP and thermal, electrical, overall efficiencies.

The variation curves of thermal and electrical power, pressure ratio and mass flow rate of the refrigerant with the increase of wind speed are shown in Fig. 19. The changing curves of the thermal and electrical power have the same trend as the thermal and electrical efficiencies. Both the pressure ratio and mass flow rate of refrigerant reduce with the increase of the wind speed. That is because more heat will be absorbed by the ambient air while less heat will be transferred by the refrigerant.

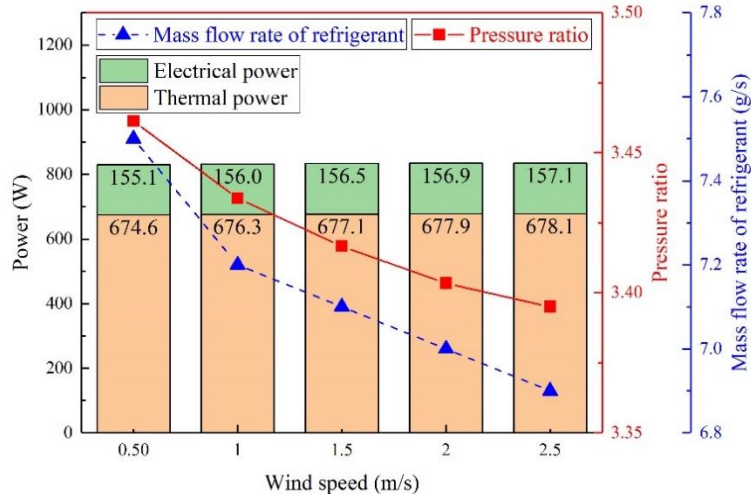
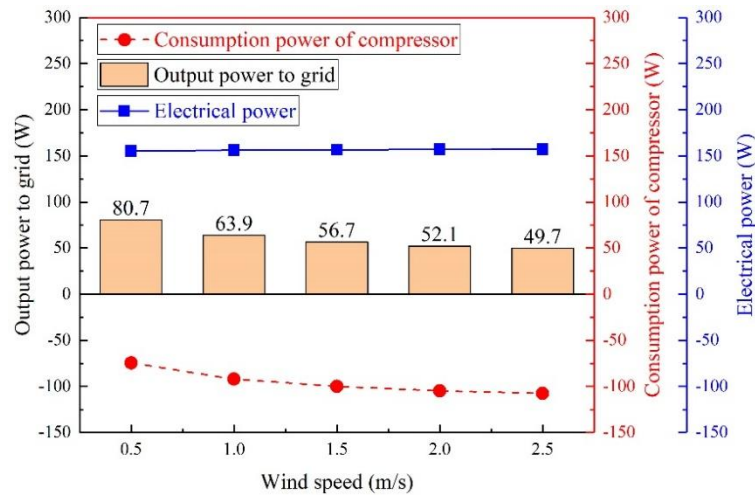


Fig. 19. Influence of wind speed on electrical and thermal power, pressure ratio and mass flow rate of refrigerant.

Fig. 20 presents the influence of wind speed on electrical power, consumption power of compressor and output power to grid. The output power to grid decreases rapidly when the wind speed increases from 0.5 m/s to 1.5 m/s and steadily when the wind speed exceeds 1.5 m/s. More heat will loss in the environment due to a higher wind speed, and the thermal efficiency will decrease causing a higher consumption power of compressor.



520

521

Fig. 20. Influence of wind speed on electrical power, consumption power of compressor and output power to grid.

522

523

524 5.4 Area of PV/T collector

525

The influences of the area of PV/T collector which varying from 1 m² to 3 m² are shown as follows at the working conditions are: solar radiation intensity is 600 W/m², ambient temperature is 25 °C and wind speed is 1.5 m/s.

528

Fig. 21 presents the influence of the area of PV/T collector on pressure ratio, heating COP and overall efficiency. Pressure ratio and overall efficiency reduce with the increase of the area, while the heating COP mounts. That is because a larger area can absorb more heat from the solar radiation, and the extra heat will be transferred by PV/T panel to refrigerant. Meanwhile, the evaporation temperature and pressure will increase leading to a lower pressure ratio and a higher heating COP. The consumption power of the compressor will decrease when the pressure ratio reduces, thus the heating COP increases more rapidly under large area conditions. A higher temperature of the PV cells causes more heat dissipates in the ambient resulting a decrease of the overall efficiency.

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530

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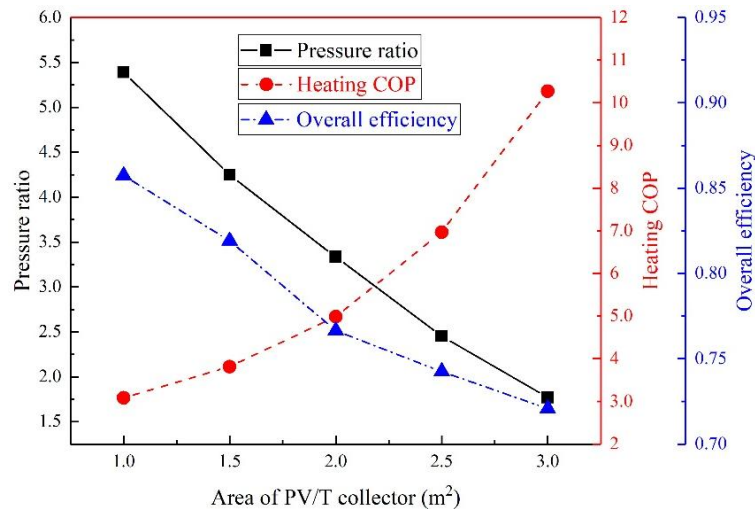
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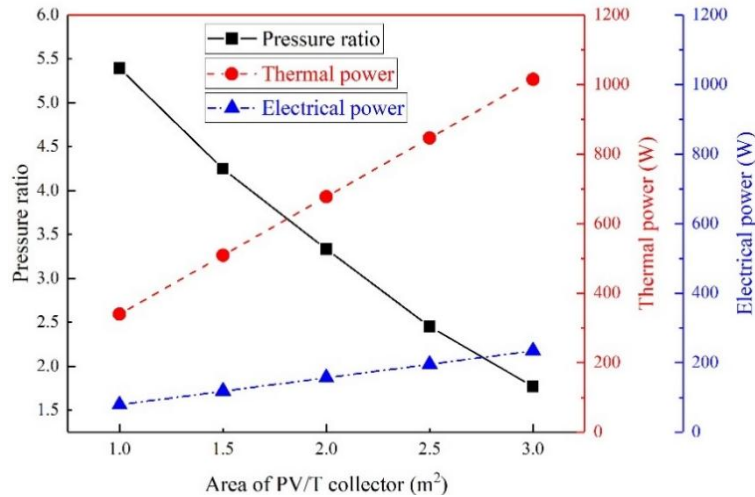
538

Fig. 21. Influence of the area of PV/T collector on pressure ratio, heating COP and overall efficiency.

539

540

541 Fig. 22 shows the declining trends of the pressure ratio and increasing trends of the thermal
542 and electrical power influenced by the area of PV/T collector. The thermal and electrical output
543 power are almost linearly and positively correlated with the area of PV/T collector. That is
544 because more heat will gain from the solar radiation and be transferred to the refrigerant when the
545 area increases.

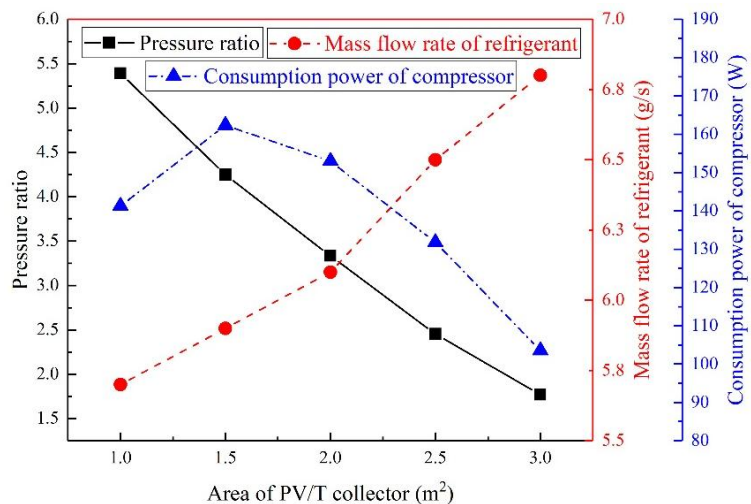


546

547 **Fig. 22.** Influence of the area of PV/T collector on pressure ratio, thermal and electrical power.

548

549 As shown in Fig. 23, the mass flow rate of refrigerant will increase when the area increases,
550 while the consumption power of compressor will increase when the area is below 1.5 m² and
551 decrease when the area is over 1.5 m². The mass flow rate of refrigerant will increase because the
552 latent heat and heat capacity of refrigerant are the same when more heat is transferred to the
553 refrigerant. The heat absorbed by the PV/T panel and mass flow rate of refrigerant are low, thus
554 the compressor consumes less electricity to compress the refrigerant vapor. With the increase of
555 the area, the mass flow rate mounts resulting in an increase of consumption power. However, the
556 pressure ratio of the compressor will reduce when the area increases over 1.5 m² leading to a
557 lower consumption power.

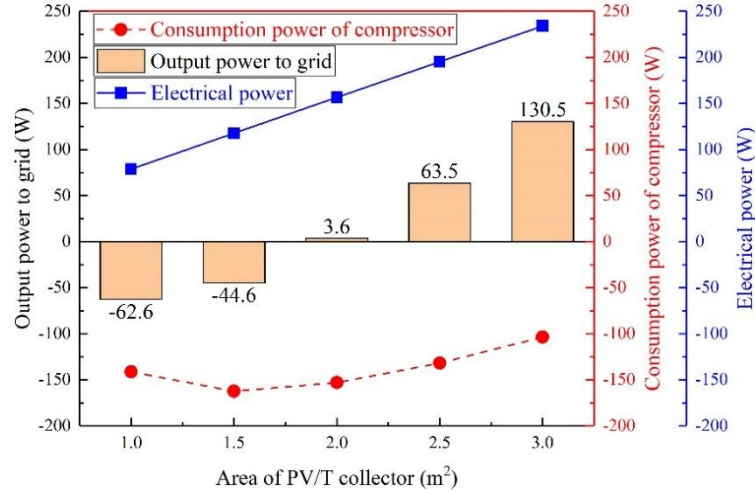


558

559 **Fig. 23.** Influence of the area of PV/T collector on pressure ratio, mass flow rate of refrigerant and
560 consumption power of compressor.

561

562 Fig. 24 illustrates the variation curve of output power to grid with the area of PV/T collector.
 563 The electrical power generated by PV panels will meet the demand of compressor when the area
 564 of PV/T collector is 2 m². Moreover, the PV cells start to produce electricity to power grid
 565 when the area is over 2 m² which means a larger PV/T panel is better for the system performance under
 566 the same system conditions.



567

568 **Fig. 24.** Influence of area of PV/T collector on electrical power, consumption power of
 569 compressor and output power to grid.
 570

571 **5.5 Feasibility analysis of the system**

572 Nowadays, the Chinese government promotes the policy of using electricity for residential
 573 heating instead of burning coal for heating in northern China to reduce carbon dioxide emissions.
 574 The solar PV/T heat pump coupled with build-in PCM heat storage system is suitable for
 575 residential heating due to its advantages: (1) high efficiency; (2) low energy consumption; (3)
 576 stable residential heating supply; (4) zero carbon emissions. Table. 4 presents the typical operating
 577 conditions and parameters of the system. It is a typical spring/autumn day in northern China.

578

Table. 4. Typical operating conditions and parameters of the system.

Parameters	Nomenclature	Value	Unit
Solar radiation intensity	I	600	W/m ²
Sunshine duration	t_s	8	hour
Ambient temperature	T_a	15	°C
Wind speed	v_{wind}	1.5	m/s
Area of the collector	A	20	m ²
Packing factor	β_p	1	[-]
Heating area	A_{ha}	100	m ²
Heat loss per square meter	P_L	50	W/m ²
Filling volume of the PCM	V_{PCM}	0.57	m ³
External diameter of the inside pipe	D_1	0.012	m
Internal diameter of the inside pipe	d_1	0.010	m
External diameter of the outside pipe	D_2	0.074	m
Internal diameter of the outside pipe	d_2	0.072	m

579

580 Table. 5 presents the simulation performance indices of the system under typical day
 581 conditions. The PV/T collector can transfer 213.9 MJ heat from the ambient to the PCM heat
 582 exchanger, and the heat will be stored in phase change materials and concrete. The build-in PCM
 583 heat storage can release the heat for 10 hours at 5.94 kW during the night while the heat loss
 584 power of 100 m² area is 5 kW. During autumn, winter and early spring in northern China, the
 585 heating system is necessary to keep indoor temperature above 20°C. Thus, the system can achieve
 586 the heating needs of the users and keep the indoor temperature steady. Meanwhile, the system can
 587 output 21.4% (2.79 kWh) of the power generated by PV panels to the grid while 78.6% of it
 588 consumed by the compressor. The heating COP of the system is 5.79, and the overall efficiency is
 589 75.49%.

590

Table. 5. Simulation performance indices of the system under typical day conditions.

Parameters	Value	Unit
Total heat storage	213.9	MJ
Photovoltaic power	1.63	kW
Photovoltaic efficiency	17.77	%
PV/T thermal efficiency	55.76	%
Heating COP	5.79	[-]
Overall efficiency	75.49	%
Cumulative power generation	13.05	kWh
Consumption power of compressor	10.26	kWh
Output power to grid	2.79	kWh
Temperature range of underfloor heating	22-31	°C
Heating power at night	5.94	kW
Heat loss power	5.00	kW
Heating hour	10.00	hour

591

592 Table. 6 illustrates the comparison of cost between the proposed system and conventional air
 593 conditioning system. The operating cost of the proposed system is under zero because users could
 594 sell spare electricity to power grid and get profit. Fig. 25 shows the cost variation curves of these
 595 two systems. The initial cost of the proposed system is much higher than conventional air
 596 conditioning system due to the underfloor heating equipment and PV/T panels, etc. However, the
 597 air conditioning system consumes a lot of electricity during the night for heating supply. Thus, the
 598 cost of these two systems will be the same after about 4 years, and the cost of proposed system
 599 keeps reduce while the cost of air conditioning system still climbs. Moreover, underfloor heating
 600 system which using radiative heating is more comfort and silence for users than air conditioner.

601

Table. 6. Cost comparison between the proposed system and conventional air conditioning system.

602

Heating system	Initial cost (¥)	Operating cost (¥/year)	Maintenance cost (¥/year)
Proposed system	22000	-764	550
Air conditioning system	4500	4024	225

603

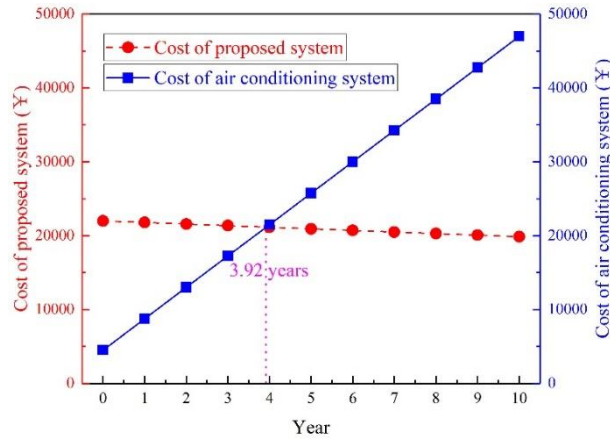


Fig. 25. Cost variation curves of proposed system and air conditioning system.

Moreover, Fig. 26 illustrates the comparison of thermal and electrical efficiencies of different PV/T systems. The electrical and thermal efficiencies of solar PV/T heat pump which proposed in this paper are higher than that of most air, water or PCM based PV/T systems. Thus, the solar PV/T heat pump coupled with build-in PCM heat storage system has a higher comprehensive energy utilization efficiency than most PV/T systems. Meanwhile, the adoption of PCM heat storage can also enhance the stability of solar PV/T heat pump system.

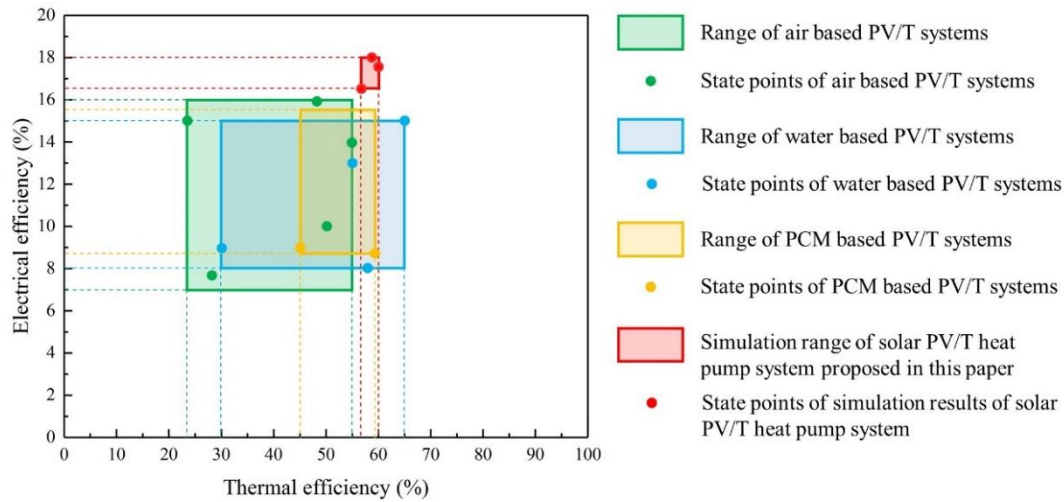


Fig. 26. Comparison of thermal and electrical efficiencies of different PV/T systems (Ahn et al., 2015; Al-Waeli et al., 2017; Hu et al., 2016; Li et al., 2015; Mojumder et al., 2016; Qiu et al., 2015)

6 Conclusion

A building-coupled cogeneration system using solar PV/T heat pump and build-in PCM heat storage is proposed in this paper. The mathematical model of the system is established and verified to analyze the system performance under different conditions. The main conclusions can be drawn as follows:

(1) The temperature of underfloor heating which using build-in PCM heat storage can reach 22 °C to 31 °C after 39 hours when the circulating water is 40 °C which is stable and suitable for residential heating.

(2) The heating COP increases with the increase of solar radiation, ambient temperature, area

626 | ~~of PV/T collector, and decrease of wind speed, respectively. Solar radiation intensity and area of~~
627 | ~~PV/T collector are two major factors affecting system performance compared to wind speed and~~
628 | ~~ambient temperature.~~

629 | (32) The heating COP can reach 6.6 which is 94% higher than conventional air conditioning
630 | system when solar radiation intensity is 600 W/m^2 , ambient temperature is $25 \text{ }^\circ\text{C}$, wind speed is 1.5
631 | m/s and area of PV/T collector is 2 m^2 while the electrical, thermal and overall efficiencies are
632 | 17.05% , 56.43% and 73.87% , respectively.

633 | (43) A 2 m^2 PV/T panel can meet the power demand of the system and heating demand of a
634 | 10 m^2 room when the solar radiation intensity is 500 W/m^2 . Moreover, the PV/T panel can output
635 | electricity to power grid if the panel area is bigger than 2 m^2 or solar radiation intensity is higher
636 | than 500 W/m^2 .

637 | The mathematical model established in this paper can also be used to analyze and optimize
638 | the solar PV/T heat pump system. However, the mathematical model of this system is for stable
639 | working conditions instead of transient. The establishment of dynamic model is needed for further
640 | predict and analyze accurately of solar assisted heat pump system under dynamic working
641 | conditions.

642 | **Acknowledgements**

643 | This research work is funded by the International Research Cooperation Program of Shanghai
644 | (Grant No. 18160710500).

645 | **Nomenclature:**

646 | Symbols

A	area (m^2)
W	width of the PV/T collector/evaporator (m)
L	length of the PV/T collector/evaporator (m)
ΔH	latent heat (kJ/kg)
h	heat transfer coefficient ($\text{W/m}^2 \cdot \text{K}$)
U	heat loss coefficient ($\text{W/m}^2 \cdot \text{K}$)
C_p	specific heat at constant pressure (kJ/kg·K)
d	inner diameter (m)
D	external diameter (m)
t/T	temperature (K)
I	solar radiation intensity (W/m^2)
Q	heat transfer rate (W)
F	collector efficiency (-)
R	thermal resistance ($\text{m}^2 \cdot \text{K/W}$)
Re	Reynolds number (-)
Ra	Rayleigh number (-)
Pr	Prandtl number (-)

	<i>v</i>	wind speed (m/s)
	<i>m</i>	mass flowrate (kg/s)
	<i>g</i>	gravitational acceleration (m/s ²)
	<i>p</i>	pressure (kPa)
	<i>P</i>	power (W)
	<i>V</i>	volume flow rate (m ³ /h)
647		
648	Greek symbols	
	δ	thickness (m)
	τ	transmittance (-)
	<i>a</i>	absorption ratios (-)
	β	packing factor (-)
	ε	emissivity (-)
	κ	thermal conductivity (W/m·K)
	$\bar{\sigma}$	Stefan-Boltzmann constant (-)
	ρ	density (kg/m ³)
	λ	compressor volumetric efficiency (-)
	η	efficiency (-)
	χ	polytropic index (-)
649		
650	Subscripts	
	<i>p,pv</i>	PV cells
	<i>g,c1</i>	external glass cover
	<i>c/c2</i>	PV-glazing cover
	<i>conc</i>	concrete
	<i>EVA</i>	EVA grease
	<i>PCM</i>	phase change material
	<i>mel</i>	melting point
	<i>ref</i>	refrigerant
	<i>b</i>	baseboard
	<i>cv</i>	convection
	<i>rd</i>	radiation
	<i>rb</i>	roll-bond panel pipe
	<i>a</i>	ambient
	<i>l</i>	liquid
	<i>L</i>	lost
	<i>rc</i>	reference
	<i>e</i>	electrical
	<i>ei</i>	electrical insulation
	<i>u</i>	uesful
	<i>th/R</i>	thermal
	<i>tp</i>	two-phase flow
	<i>ove</i>	oveall
	<i>abs</i>	absorb

<i>hp</i>	heat pipe
<i>sh</i>	superheated
<i>v</i>	vapor
<i>cond</i>	condensation
<i>eva</i>	evaporation
<i>ot</i>	outer pipe
<i>eq</i>	equivalent
<i>hx</i>	heat exchanger
<i>dis</i>	discharge
<i>suc</i>	suction
<i>in</i>	inlet
<i>out</i>	outlet
<i>cw</i>	circulating water

651

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Performance analysis of solar assisted heat pump coupled with build-in PCM heat storage based on PV/T panel

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Abstract

PV/T (photovoltaic/thermal) technology is a combination of PV module (photovoltaic utilization) and collector (photothermal utilization), which can improve the comprehensive utilization efficiency of solar energy and has a broad application prospect. In this paper, PV/T module is coupled with heat pump evaporator to form a direct-expansion solar PV/T heat pump which is suitable for heat application in high latitude area. To achieve stable residential heating, a solar PV/T heat pump system coupled with build-in PCM (phase change material) heat storage is therefore proposed and simulated. Meanwhile, the mathematical model of solar PV/T heat pump coupled with build-in PCM heat storage system is established and verified. The simulation results show that the temperature of underfloor heating which using build-in PCM heat storage can reach 22 °C to 31 °C after 39 hours when the circulating water is 40 °C. Moreover, the heating COP (Coefficient of Performance) increases with the increase of solar radiation, ambient temperature and area of PV/T collector, and decrease of wind speed, respectively. A 20 m² PV/T panel module can output 21.4% of the electricity to power grid when the solar radiation intensity is 600 W/m² and meet the heat demand of a 100 m² room while maintain the operation of the system. Meanwhile, the heating COP can reach 5.79 which is 70% higher than the conventional air conditioning system and the electrical, thermal, overall efficiencies are 17.77%, 55.76% and 75.49%, respectively.

1 Introduction

The total amount of energy consumption in the world is constantly climbing (2019; Caetano et al., 2017). The consumption of fossil energy has brought about energy crisis and environmental crisis (Pietrosemoli and Rodríguez-Monroy, 2019). Without action, CO₂ emissions from burning fossil fuels will be doubled by 2050 (Paolo Frankl, 2010). Therefore, the development and utilization of renewable energy has become one of the effective solutions (Keček et al., 2019). Solar energy has become the first choice due to its characteristics of ubiquity, abundance and sustainability (Kuik et al., 2019; Tsai, 2015), which is mainly used in two ways: photothermal and photovoltaic. 11% of global electricity will be provided by PV by 2050 (Paolo Frankl, 2010). However, the electrical efficiency of PV cells decreases with the increase of the temperature of PV cells (Huide et al., 2017). A cooling system can be added to reduce the temperature of PV cells while the remaining heat of PV panel are absorbed by working fluid which can be employed as a useful thermal energy for heat applications in buildings.

42 The PV/T technology coupled PV modules with thermal collectors was first proposed by
43 Wolf et al (Wolf, 1976) to reduce PV cells temperature and improve electrical efficiency. The
44 PV/T system can recover waste heat from the PV panel to improve comprehensive energy
45 utilization efficiency. PV/T design optimizations are carried out to improve the system efficiency
46 in recent years. Nahar et al. (Nahar et al., 2017) designed a novel pancake-shaped flow channel for
47 PV/T system, and integrated the flow channel with the PV baseboard. They found that the
48 temperature of the PV panel is reduced by 42 °C, and the electrical efficiency is increased by 2%.
49 Othman et al. (Othman et al., 2016) proposed a parallel, double pass flat plate collector which was
50 adopted in a two fluids PV/T system. Their results showed that the electrical efficiency and
51 thermal efficiency are 17% and 76%, respectively.

52 The combination of PCM and PV/T panel is an effective way to stabilize the operating
53 temperature of PV cells and improves the overall efficiency. Hosseinzadeh et al. (Hosseinzadeh et
54 al., 2018) investigated the effect of simultaneous use of nanofluid as coolant as well as an organic
55 paraffin as the phase change material on the electrical and thermal efficiencies. They demonstrated
56 that the use of PCM in nanofluid based PVT/PCM system enhances the thermal output power of
57 conventional PV/T system by 29.6%. Kazemian et al. (Kazemian et al., 2019) developed and
58 simulated a comprehensive three-dimensional model of PV/T system integrated with PCM. Their
59 simulation results presented that the PV/T-PCM system have lower surface temperature compared
60 to PV/T system, and as the thermal conductivity of PCM enhances, both electrical and thermal
61 efficiencies increase. Fayaz et al. (Fayaz et al., 2019) investigated the PCM based PV/T system,
62 and the experimental validation was carried out to verify the numerical model. They found that the
63 electrical efficiency is achieved as 13.98% and 13.87% numerically and experimentally
64 respectively, and the electrical performance is improved as 6.2% and 4.8% for PV/T-PCM system
65 based on the numerical and experimental results respectively.

66 Different working fluids like water, air, nanofluid and refrigerant are also used to cool the PV
67 module. Huang and Lee (Huang and Lee, 2004) conducted long-term tests on the direct-expansion
68 solar heat pump which adopted refrigerant as working fluid to verify the stability of the work. The
69 total running time of their prototype is over 20000 hours, and the measured energy consumption is
70 0.019 kWh/l of hot water at 57 °C which is much less than traditional solar water heater.
71 Stojanović and Akander (Stojanović and Akander, 2010) used direct-expansion heat pump for
72 independent buildings heating and domestic hot water supply. In their system, the collector area is
73 42.5 m² and heat pump power is 8.4 kW, they measured that the actual indoor temperature is no
74 less than 20 °C during the testing period. Alejandro Del Amo et al. (Del Amo et al., 2019) verified
75 the feasibility of solar PV/T heat pump through experiments. They obtained that the highest COP
76 of the system can reach 4.62. Meanwhile, the PV module provides 67.6% of the power demand,
77 and the payback period is 6 years.

78 In addition to optimize the PV/T panel, the adoption of PCM as heat storage is also a good
79 way to stabilize the system. Kuznik et al. (Kuznik et al., 2008) adopted PCM wallboard heat
80 storage and conducted comparative experiments. In their study, the system can effectively reduce
81 heat loss, keep the room warm and improve indoor thermal comfort. Fiorentini et al. (Fiorentini et
82 al., 2015) combined PCM storage with PV/T system, and the roof was used as PV/T layout
83 location. The PCM storage adopted in their system can keep indoor comfort within a certain and
84 potentially variable thermal comfort range. Diallo et al. (Diallo et al., 2019) proposed the
85 PVT-LHP (PVT Loop Heat Pipe) technology employing PCM triple heat exchanger, the total

86 energy efficiency of the presented system is improved by 28%, and the heating COP is 2.2 times
87 than that of a traditional PV/T system.

88 Owing to the instability of solar energy, traditional solar PV/T system cannot continuously
89 and stably supply heat or power generation when solar irradiation is weak such as rainy day or
90 winter. Consequently, the market of PV/T technology compared with PV or PT system is still very
91 low. PV/T can adapt to the characteristics of low intensity, instability and intermittency of solar
92 energy better if it can be combined with accumulator and heat storage. However, additional space
93 is required to install heat storage tank, which is not suitable for use in urban areas where land
94 resources are scarce. Therefore, in this paper, a coupling design of solar PV/T heat pump and
95 build-in PCM heat storage is proposed and the parallel air source heat exchanger is also adopted to
96 enhance the stability of the system. The build-in PCM heat storage used for underfloor heating is a
97 combination of PCM and building materials, which can save more space compared to
98 conventional PCM storage tank system. Firstly, the composition and operation modes of the
99 system are introduced. According to the system principle, the mathematical model is established
100 and verified, and the build-in PCM heat storage sub-system which using for residential heating is
101 also proposed and simulated. Then the influences of different parameters on system performance
102 are analyzed. Finally, the feasibility analysis of the system is conducted. The objective of this
103 paper is to provide a promising method to realize stable, high efficiency, environmental friendly
104 residential heating in high latitude area with no energy consumption from power grid.

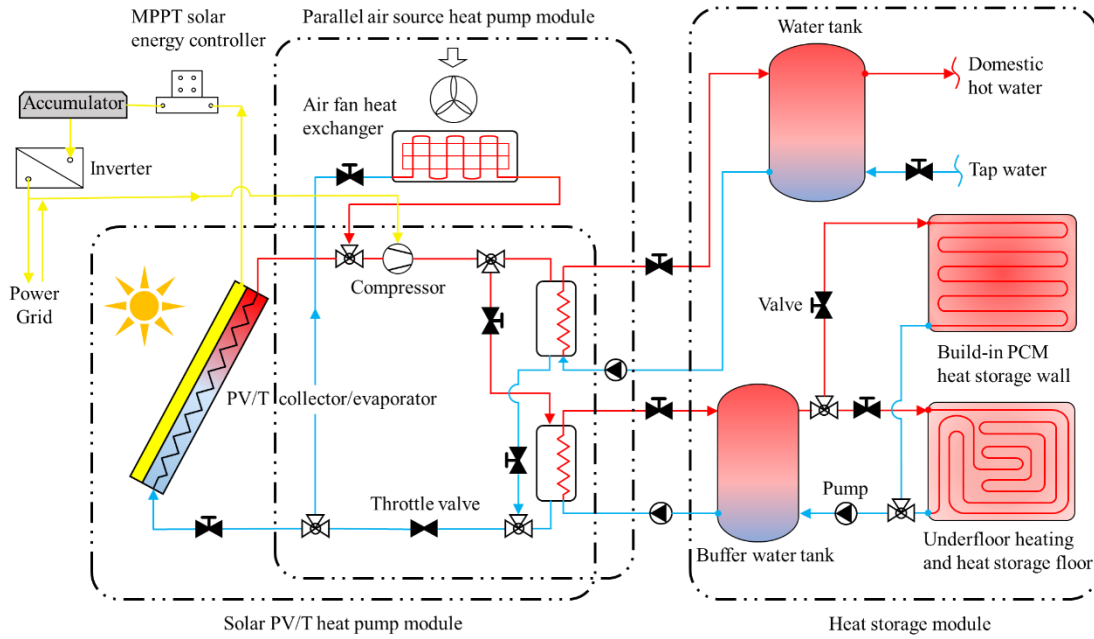
105 **2 System description**

106 Fig. 1 shows the schematic diagram of the system based on solar PV/T heat pump, which is
107 consisted of four main parts: solar PV/T heat pump module, parallel air source heat pump module,
108 heat storage module and electrical module. The blue lines represent low temperature working fluid,
109 and in the opposite, red lines represent high temperature. The yellow lines represent the electricity
110 flow direction. The arrows show the working fluid direction. The system can be divided into two
111 operating modes, which are listed as follows.

112 (1) Sunny day operating mode: The electricity generated by PV panel is used to drive the
113 pumps and compressor, and the excess electricity will be recovered to the power grid or drive the
114 air conditioning. However, the PV panel heated by solar radiation will be resulted in an increase of
115 the temperature of PV cells. Meanwhile, the heat transferred by the PV/T collector can be
116 absorbed by the refrigerant. Then, the superheated refrigerant vapor with low pressure is
117 compressed by the compressor to the high temperature and pressure refrigerant vapor. The
118 condensation heat will be absorbed and stored in the build-in PCM heat storage. The heat
119 transferred by condensation can also be used for producing domestic hot water. The liquid
120 refrigerant will expand through the throttle valve after condensation process, and flow into the
121 PV/T collector/evaporator. The heat released from the heat storage module will be used to keep
122 the indoor temperature constant during the night.

123 (2) Rainy day operating mode: The system will switch to the air source heat pump mode
124 when the solar radiation is insufficient to maintain system operation. The pumps and compressor
125 driven by power grid are used to keep the system work. The air fan heat exchanger is adopted to
126 absorb heat from the ambient air. The refrigerant will be heated by the air fan heat exchanger and
127 compressed by compressor into high temperature and pressure vapor. The heat released by the
128 refrigerant vapor will be transferred to the PCM or water. This mode can make full use of the

129 valley electricity to store heat at night, and maintain the indoor temperature through the heat
 130 storage module during the day.

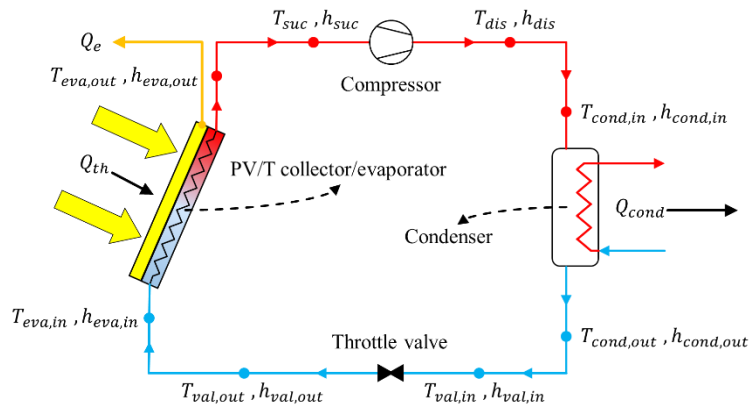


131
 132
 133

Fig. 1. Schematic of the system based on solar PV/T heat pump.

134 **3 Mathematical model**

135 The thermodynamic state points for each process are shown in Fig. 2. The solar PV/T heat
 136 pump cycle could be simplified to four components: PV/T collector/evaporator, compressor, PCM
 137 heat exchanger and throttle valve. Different temperature (T) and enthalpy (h) at each state point
 138 are shown in Fig. 2. Q_{th} (W) is the heat transfer rate between refrigerant and the PV/T panel, Q_e
 139 (W) is the electrical power provided by PV panel, and Q_{PCM} (W) is the heat transfer rate between
 140 refrigerant and phase change materials.



141
 142
 143

Fig. 2. The thermodynamic state points for each component.

144 The design parameters of the system and characteristics of different PV/T layers are listed in
 145 Table. 1.

146 **Table. 1.** Design parameters of the system and characteristics of different PV/T layers.

Parameters	Nomenclature	Value	Unit
------------	--------------	-------	------

Thickness of PV-glazing cover	$\delta_{g,pv}$	1	mm
Emissivity of PV-glazing cover	ε_c	0.84	[-]
Transmissivity of PV-glazing cover	$\tau_{g,pv}$	0.9	[-]
Thickness of PV cells	δ_{pv}	0.3	mm
Emissivity of PV cells	ε_p	0.96	[-]
Absorptance of PV cells	a_p	0.85	[-]
Thermal conductivity of PV cells	κ_p	203	W/m·K
Absorptance of PV baseboard	a_b	0.8	[-]
Thickness of EVA grease	δ_{EVA}	0.5	mm
Thermal conductivity of EVA grease	κ_{EVA}	0.311	W/m·K
Thickness of electrical insulation	δ_{ei}	0.5	mm
Thermal conductivity of electrical insulation	κ_{ei}	0.15	W/m·K
Electrical insulation material	[-]	Tedlar	[-]
Packing factor	β_p	1	[-]
Length of PV/T collector/evaporator	L	2.0	m
Width of PV/T collector/evaporator	W	1.0	m
Area of the PV/T collector/evaporator	A	2.0	m ²
Thermal conductivity of roll-bond panel	κ_{rb}	151	W/m·K
Thickness of roll-bond panel pipe	δ_{rb}	1	mm
Refrigerant type	ref	R134A	[-]

147

148 3.1. Model of PV/T collector/evaporator

149 The heat absorbed by the PV/T panel is expressed as follows:

$$150 \quad Q_{abs} = (1 - \eta_e) \cdot A \cdot I \cdot \tau_{g,pv} \cdot [\alpha_p \cdot \beta_p + \alpha_b \cdot (1 - \beta_p)] \quad (1)$$

151 where A is the collector area of the PV/T panel (m²); I is the solar radiation intensity (W/m²); $\tau_{g,pv}$
152 is the transmittances of the PV-glazing cover; α_p and α_b are the absorption ratios of the PV cells
153 and its baseboard, respectively; β_p is the packing factor of PV cells; η_e is the PV cells' efficiency,
154 calculated by (Huide et al., 2017):

$$155 \quad \eta_e = \eta_{rc} \cdot [1 - \beta_{pv} \cdot (T_p - T_{rc})] \quad (2)$$

156 η_{rc} is the reference photovoltaic efficiency value of PV cells at $T_{rc}=298$ K, $\eta_{rc}=0.18$; β_{pv} is the
157 temperature coefficient (1/K) of PV cell efficiency, $\beta_{pv}=0.0045$ (Huide et al., 2017).

158 Fig. 3 shows the heat loss and physical model of PV/T panel which has a multi-layer
159 structure, the heat loss of PV/T panel consists of two parts: (1) heat transfer from PV cells to
160 PV-glazing cover; (2) heat transfer from PV-glazing cover to ambient air.

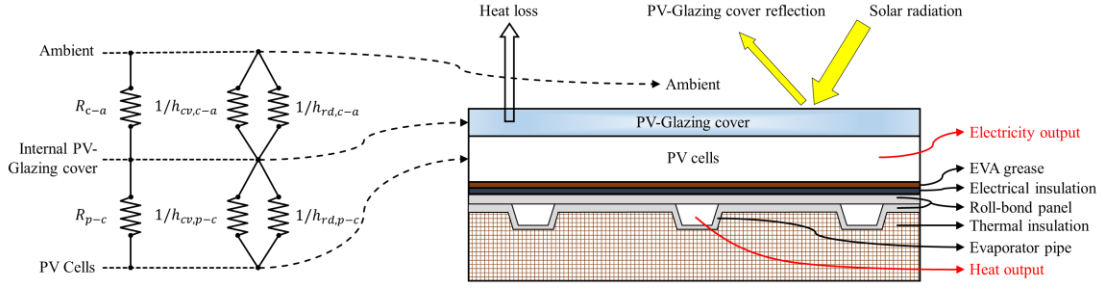
161 The total heat loss rate of PV/T module is given as:

$$162 \quad Q_L = U_L \cdot A \cdot (T_p - T_a) \quad (3)$$

163 where T_p and T_a are the temperature of PV cells and ambient air, respectively. U_L is the overall
164 heat loss coefficient which can be written as:

$$165 \quad U_L = \left[1 / (h_{cv,p-c} + h_{rd,p-c}) + 1 / (h_{cv,c-a} + h_{rd,c-a}) \right]^{-1} \quad (4)$$

166 $h_{cv,p-c}$ and $h_{rd,p-c}$ are the convective and radiative heat-transfer coefficients between PV cells and
 167 glass cover; $h_{cv,c-a}$ and $h_{rd,c-a}$ are the convective and radiative heat-transfer coefficients between glass
 168 cover and ambient.



169
 170 **Fig. 3.** Heat loss and physical model of PV/T collector/evaporator.

171
 172 The overall electricity output power of PV cells is given as:

$$173 \quad Q_e = A \cdot I \cdot \tau_{g,pv} \cdot \alpha_p \cdot \beta_p \cdot \eta_e \quad (5)$$

174 Under the steady-state condition, the heat transfer rate delivered by the module equals the
 175 rate of the absorbed heat minus the overall heat loss, expressed as:

$$176 \quad Q_{th} = Q_{abs} - Q_L \quad (6)$$

177 The total useful solar heat received by the PV/T collector/evaporator is expressed as:

$$178 \quad Q_u = F_R \cdot A \cdot I \cdot \tau_{g,pv} \cdot \left[\alpha_p \cdot \beta_p + \alpha_b \cdot (1 - \beta_p) \right] \quad (7)$$

179 where F_R is the PV/T collector thermal efficiency factor, can be defined as (Diallo et al., 2019):

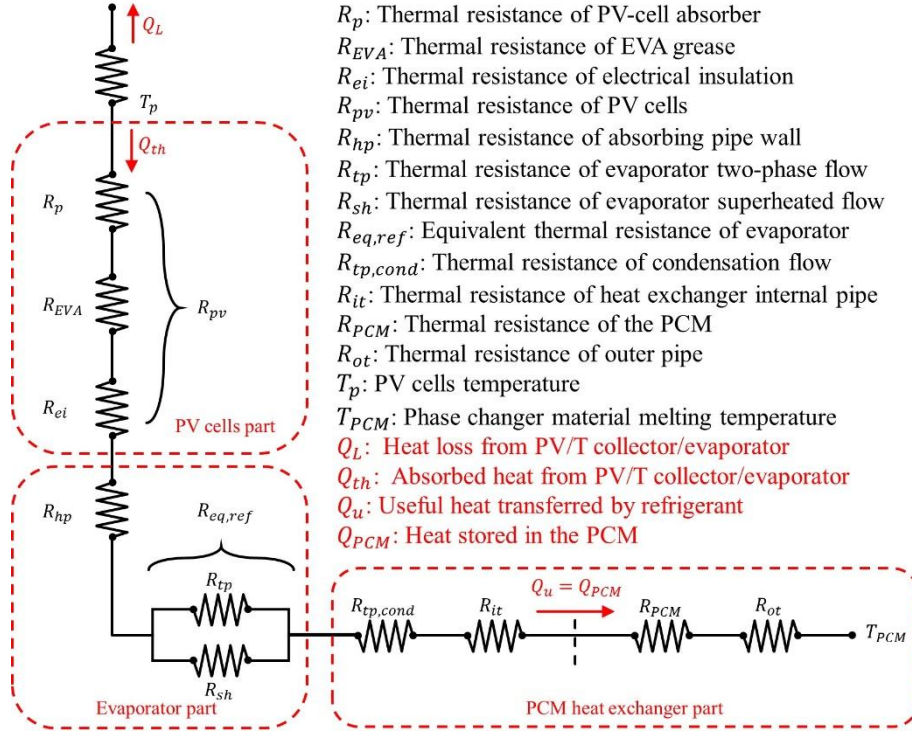
$$180 \quad F_R = (1 - \eta_e) \cdot \frac{1/U_L}{L \cdot W / N_{rb} \cdot \left\{ 1 / \left(L \cdot U_L - \left[(W / N_{rb} - L_{rb}) F_{rb} + L_{rb} / (1 + R_{pv} \cdot U_L) \right] \right) + \sum_1^5 R_i \right\}} \quad (8)$$

181 N_{rb} is the equivalent number of roll-bond panel pipe; L_{rb} is the equivalent length of roll-bond panel
 182 pipe; $\sum R_i$ is the overall thermal resistance from the PV cells to the PCM; F_{rb} is the efficiency of the
 183 roll-bond panel which encapsulated in the backside of the PV/T panel which can be defined as
 184 (Diallo et al., 2019):

$$185 \quad F_{rb} = \tanh \left[\frac{\sqrt{\frac{U_L}{(k_{rb} \delta_{rb} (1 + R_{pv} U_L))}} (W / N_{rb} - L_{rb}) / 2}{\sqrt{\frac{U_L}{(k_{rb} \delta_{rb} (1 + R_{pv} U_L))}} (W / N_{rb} - L_{rb}) / 2} \right] \quad (9)$$

186 κ_{rb} is the thermal conductivity of roll-bond panel; δ_{rb} is the thickness of the roll-bond panel pipe;
 187 R_{pv} is the thermal resistance of PV cells.

188 The thermal resistances of the PV/T system and heat transfer along the system are shown in
 189 Fig. 4.



190
191 **Fig. 4.** Thermal resistances of the PV/T system and heat transfer along the system.
192

193 Heat transfer between PV module and heat pipe is a conventional one dimensional heat
194 conduction process and its associated thermal resistance is:

195
$$R_{pv} = \delta_p / k_p + \delta_{EVA} / k_{EVA} + \delta_{ei} / k_{ei} \quad (10)$$

196 Superheated region and two-phase region existed in the refrigerant side of the PV/T
197 collector/evaporator. The equivalent thermal resistance of the two different regions can be
198 calculated as (P. Hartnett and M. Rohsenow, 1973):

199
$$R_{eq.ref} = \left(1/R_{tp} + 1/R_{sh} \right)^{-1} \quad (11)$$

200

201 3.2. Model of build-in PCM heat storage

202 All the heat gained by the build-in PCM heat storage is transferred by the PCM heat
203 exchanger. The heat store rate in the PCM and concrete is expressed as follows:

204
$$Q_{PCM} + Q_{conc} = \dot{m}_{ref} \cdot (h_{dis} - h_{in}) \quad (12)$$

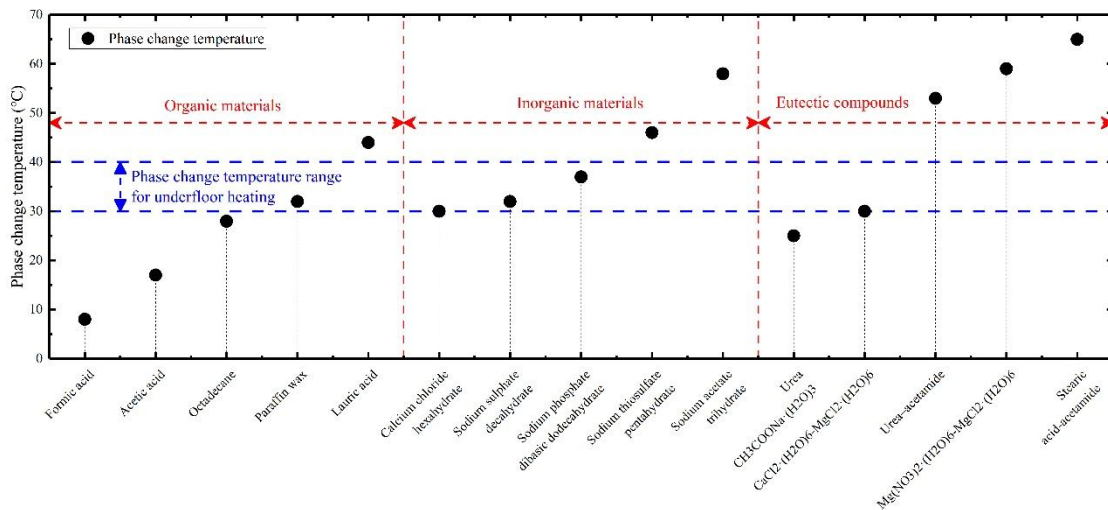
205 The PCM melting rate is calculated by:

206
$$\dot{m} = Q_{PCM} / \Delta H_{PCM} \quad (13)$$

207 where ΔH_{PCM} is the latent heat of the PCM (kJ/kg).

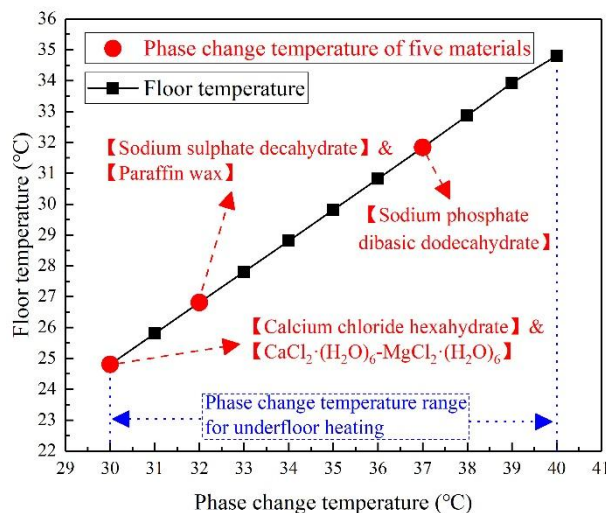
208 Fig. 5 shows the phase change temperature of 15 kinds of phase change materials including
209 organic/inorganic/eutectic compounds materials. According to L.F. Cabeza et al. (Cabeza et al.,
210 2011), the recommendation phase change temperature range for underfloor heating is between
211 30 °C to 40 °C. There are five kinds of PCM that have phase change temperature in this range:

212 Paraffin wax, $\text{CaCl}_2 \cdot (\text{H}_2\text{O})_6$ - $\text{MgCl}_2 \cdot (\text{H}_2\text{O})_6$, Calcium chloride hexahydrate, Sodium sulphate
 213 decahydrate, Sodium phosphate dibasic dodecahydrate.



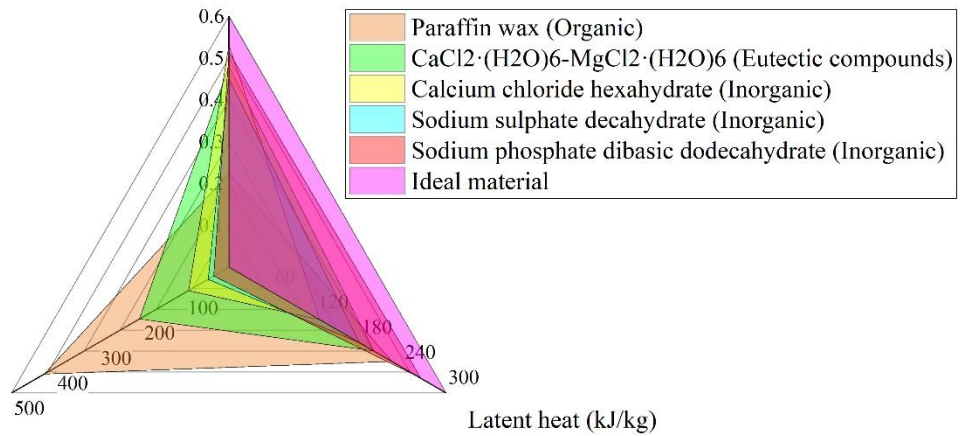
214
 215 **Fig. 5.** Phase change temperature of 15 kinds of PCM including Organic/Inorganic/Eutectic
 216 compounds materials.
 217

218 The variation curve of floor temperature with phase change temperature and the comparison
 219 of three indices including thermal conductivity, latent heat and price (Pereira da Cunha and Eames,
 220 2016) of above five kinds of PCM are shown in Fig. 6. These five materials have different phase
 221 change temperature which all in the recommendation temperature range for underfloor heating.
 222 The ideal material should have a higher thermal conductivity and latent heat while the price is low.
 223 In order to evaluate these five materials, the graph of ideal material is also plotted in Fig. 6. The
 224 organic and eutectic compounds phase change materials have a higher price than inorganic
 225 materials while the thermal conductivity and latent heat are lower. Moreover, the larger of the
 226 overlap area in Fig. 6 between each material and ideal material, the better of the material
 227 performance. As shown in Fig. 6, Sodium phosphate dibasic dodecahydrate has the largest overlap
 228 area, thus, it is used in the simulation of build-in PCM heat storage unit.



229
 230 (a)

Thermal conductivity (W/m·K)



(b)

Fig. 6. (a) Variation curve of floor temperature with phase change temperature. **(b)** Thermal conductivity/Latent heat/Price comparison of six kinds of materials including ideal material.

Fig. 7 illustrates the structure and cross-section view of a build-in PCM heat storage unit which is the component of underfloor heating module. The phase change materials (Sodium phosphate dibasic dodecahydrate) are placed in the outer tube of a double-wall tube which can store heat during the day and release latent heat at night. Polystyrene board and foam concrete are used as thermal insulation and building materials, respectively. The hot water which produced by solar PV/T heat pump module is pumped into the pipe of underfloor heating to keep the indoor temperature steady. The properties of PCM (Jankowski and McCluskey, 2014) and concrete as well as underfloor heating working conditions are listed in Table. 2.

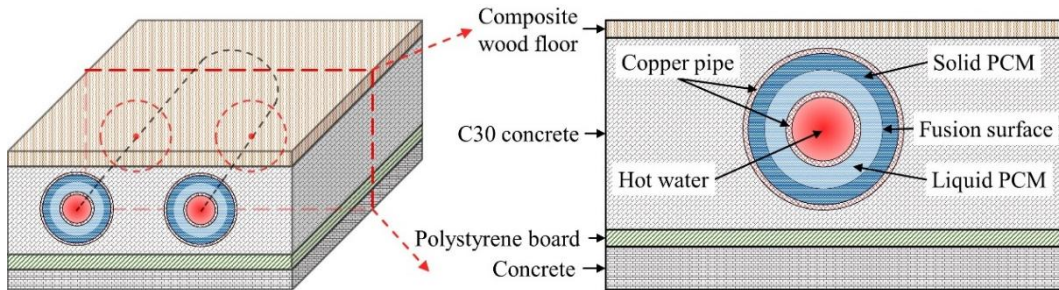


Fig. 7. Structure and cross-section view of the build-in PCM heat storage unit.

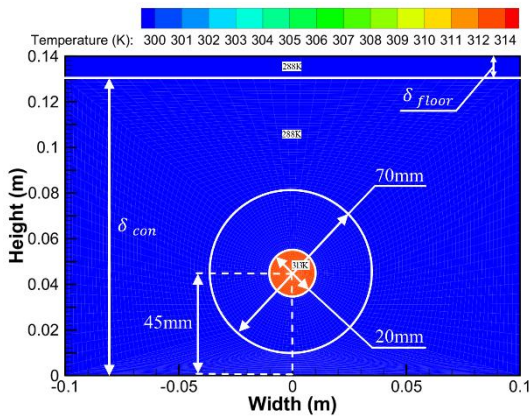
Table. 2. Properties of PCM and concrete as well as underfloor heating working conditions.

Parameters	Nomenclature	Value	Unit
Type of PCM	$[-]$	$\text{Na}_2\text{HPO}_4 \cdot 12\text{H}_2\text{O}$	$[-]$
Latent heat of PCM	ΔH_{PCM}	265	kJ/kg
Density of PCM	ρ_{PCM}	1507	kg/m ³
Temperature of transition of PCM	T_{mel}	37	°C
Specific heat at constant pressure of PCM	C_{p-PCM}	1.69	kJ/kg·K
Thermal conductivity of PCM	κ_{PCM}	0.514	W/m·K
Thermal conductivity of copper coil	κ_C	397	W/m·K

Type of concrete	[-]	C30	[-]
Specific heat at constant pressure of concrete	C_{p-conc}	0.97	kJ/kg·K
Thermal conductivity of concrete	κ_{con}	1.6	W/m·K
Density of concrete	ρ_{conc}	2300	kg/m ³
Thickness of the concrete	δ_{con}	0.13	m
Thickness of the wood floor	δ_{floor}	0.01	m
Volume flow rate of circulating water	V_{cw}	2.5	m ³ /h

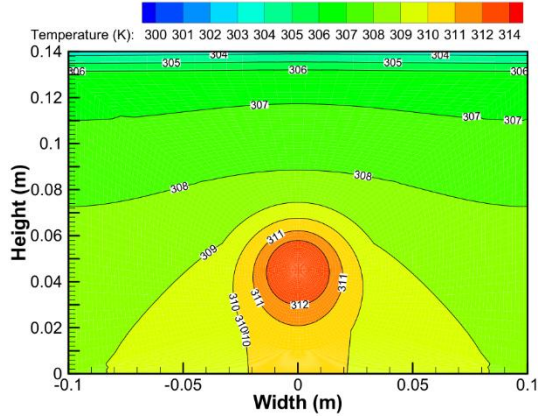
248

249 Fig. 8 (a~c) presents the simulation results of heat transfer in build-in PCM heat storage unit
 250 at different time which carried out by software of Ansys Fluent 17.0. The thermal conductivity of
 251 the heat transfer between the coil and PCM has been considered in the setup of the boundary
 252 conditions. Thus, the boundary conditions of inner and outer tube have been set as “coupled” in
 253 Ansys Fluent 17.0 which means the solution of heat transfer process would be carried out through
 254 the coupled method in the program. The initial temperature of the heat storage unit is 15 °C, and
 255 the underfloor heating can reach above 30 °C after 39 hours when the circulating water is 40 °C. As
 256 shown in Fig. 8(d), the floor temperature increases with time and reaches 25 °C in the first 12
 257 hours which has 10 °C difference with the ambient. The changing curve mountains rapidly in the
 258 initial stage and becomes steady after 39 hours, the build-in PCM heat storage would supply heat
 259 to indoor area and maintain the room temperature.

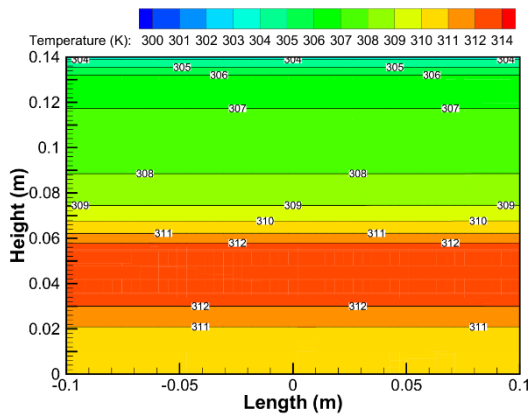


260

(a) Initial stage

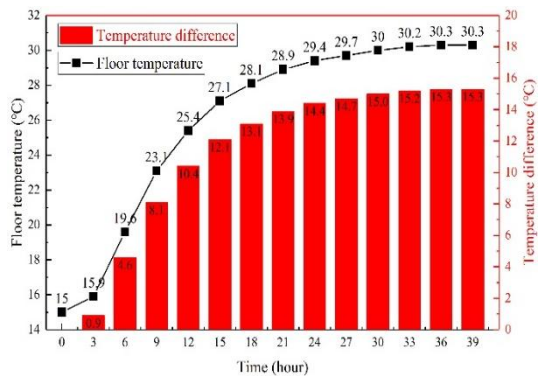


(b) Steady stage



263

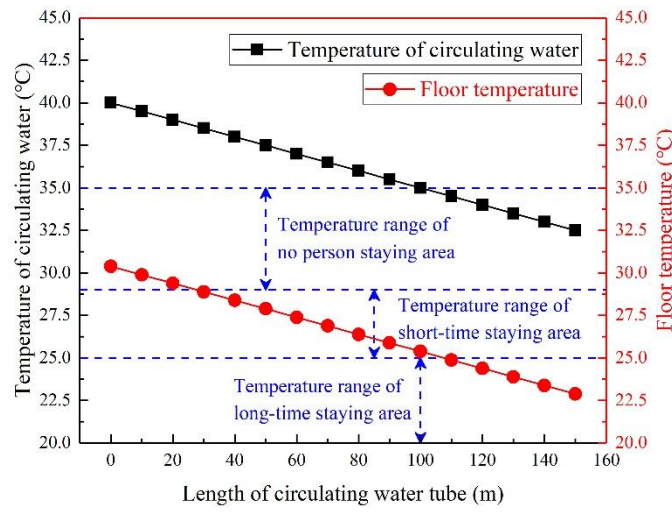
264 (c) Cross-section view at the width of 0 (steady stage)



(d)

265 **Fig. 8. (a-c)** Cross-section temperature contour of the build-in PCM heat storage unit at initial and
 266 steady stage. **(d)** Variation curve of floor temperature and temperature difference between floor
 267 and ambient.
 268

269 Fig. 9 shows the variation curves of circulating water temperature and floor temperature with
 270 the length of circulating water tube. The temperature of circulating water would decrease from 40 °C
 271 (inlet) to 32.5 °C (outlet) while the floor temperature varies from 30.4 °C to 22.9 °C. This figure also
 272 presents the three temperature range of underfloor heating which can be divided into three categories:
 273 Temperature range of no person staying area (29 °C-35 °C), Temperature range of short-time
 274 staying area (25 °C-29 °C) and Temperature range of long-time staying area (20 °C-25 °C). Therefore,
 275 when using build-in PCM heat storage as underfloor heating, the system could meet the heat
 276 demand of users and keep indoor temperature steady.



277
 278 **Fig. 9.** Variation curves of circulating water temperature and floor temperature with the length of
 279 circulating water tube.
 280

281 3.3. Model of compressor

282 The refrigerant mass flow rate \dot{m}_{ref} could be calculated by (Ma, 2013)

$$283 \quad \dot{m}_{ref} = \lambda \cdot V_{th} / v_{suc} \quad (14)$$

284 where the λ is the compressor volumetric efficiency, V_{th} is the theoretical displacement volume of
 285 compressor (m^3), v_{suc} is the specific volume of the refrigerant in the suction period (m^3/kg).

286 The power consumption of compressor is written as:

$$287 \quad P_{com} = \dot{m}_{ref} \cdot (h_{dis} - h_{suc}) / \eta_{ele} \quad (15)$$

288 where η_{ele} is the efficiency of the compressor which can be expressed by (Ma, 2013):

$$289 \quad \eta_{ele} = -0.17938 + 0.87501 \frac{P_{dis}}{P_{suc}} - 0.30014 \left(\frac{P_{dis}}{P_{suc}} \right)^2 + 0.04135 \left(\frac{P_{dis}}{P_{suc}} \right)^3 - 0.00206 \left(\frac{P_{dis}}{P_{suc}} \right)^4 \quad (16)$$

290 The refrigerant vapor through compressor is isentropic, thus the equation can be expressed
 291 by:

$$292 \quad T_{dis} / T_{suc} = (p_{dis} / p_{suc})^{(\chi-1)/\chi} \quad (17)$$

293 where the χ is the polytropic index of refrigerant. h_{dis} is the enthalpy of the vapor after compressed
 294 (kJ/kg), h_{suc} is the enthalpy of the vapor before compressed (kJ/kg).

295 The heat of refrigerant absorbed from the PV/T collector/evaporator equal to Q_u :

$$296 \quad \dot{m}_{ref} \cdot (h_{eva,out} - h_{eva,in}) = Q_u = Q_{th} \quad (18)$$

297 3.4 Definition of the system performance

298 The COP (Coefficient of Performance) of the system can be defined as the ratio of overall
 299 heat output of system and power consumption of the compressor as following (R. Turns, 2006)

$$300 \quad COP = \dot{m}_{ref} \cdot (h_{cond,in} - h_{cond,out}) / P_{com} \quad (19)$$

301 The η_{th} is the PV/T collector/evaporator's thermal efficiency, which can be defined as:

$$302 \quad \eta_{th} = Q_{th} / (A \cdot I) \quad (20)$$

303 The η_e is the PV cells electrical efficiency which can be defined as:

$$304 \quad \eta_e = Q_e / (A \cdot I) \quad (21)$$

305 The η_{ove} is the overall efficiency which can be defined as:

$$306 \quad \eta_{ove} = (Q_{cond} + Q_e) / (A \cdot I) \quad (22)$$

307 3.5 Presentation of the algorithm by flow chart

308 The numerical simulation procedure of the system is shown in Fig. 10 to predict the system
 309 performance using the software of MATLAB. The solution steps are as flows:

- 310 (1) Input all the environmental parameters, such as solar radiation intensity, wind speed,
 311 ambient temperature, etc.
- 312 (2) Input the system design parameters and operation parameters, such as collector area,
 313 packing factor, collector slop, transmissivity of external glass cover, PV-glazing cover,
 314 thickness of each layer, etc.
- 315 (3) Assume the temperature of PV cells T_p .
- 316 (4) Calculate the overall heat loss rate Q_L , and the PV electrical output power Q_e .
- 317 (5) Calculate the thermal energy gain rate through PV/T collector/evaporator Q_{th} , and the
 318 useful heat transfer rate by refrigerant Q_u .
- 319 (6) Calculate $(Q_{th} - Q_u)/Q_{th}$. If $|(Q_{th} - Q_u)/Q_{th}| < 0.1\%$, the system achieves the heat balance
 320 and move to next step.
- 321 (7) Input the superheat degree T_{sh} .
- 322 (8) Assume the compressor discharge pressure P_{dis} .
- 323 (9) Calculate the PV/T collector/evaporator inlet enthalpy $h_{eva,in}$ and the PCM heat exchanger
 324 outlet enthalpy $h_{cond,out}$. If $|(h_{eva,in} - h_{cond,out})/h_{cond,out}| < 0.1\%$, the system achieves the

- 325 pressure balance and move to next step.
 326 (10) Calculate the COP, PCM melting rate \dot{m} , thermal efficiency η_{th} , electrical efficiency η_e ,
 327 and overall efficiency η_{ove} .
 328 (11) Results output, and stop the program.

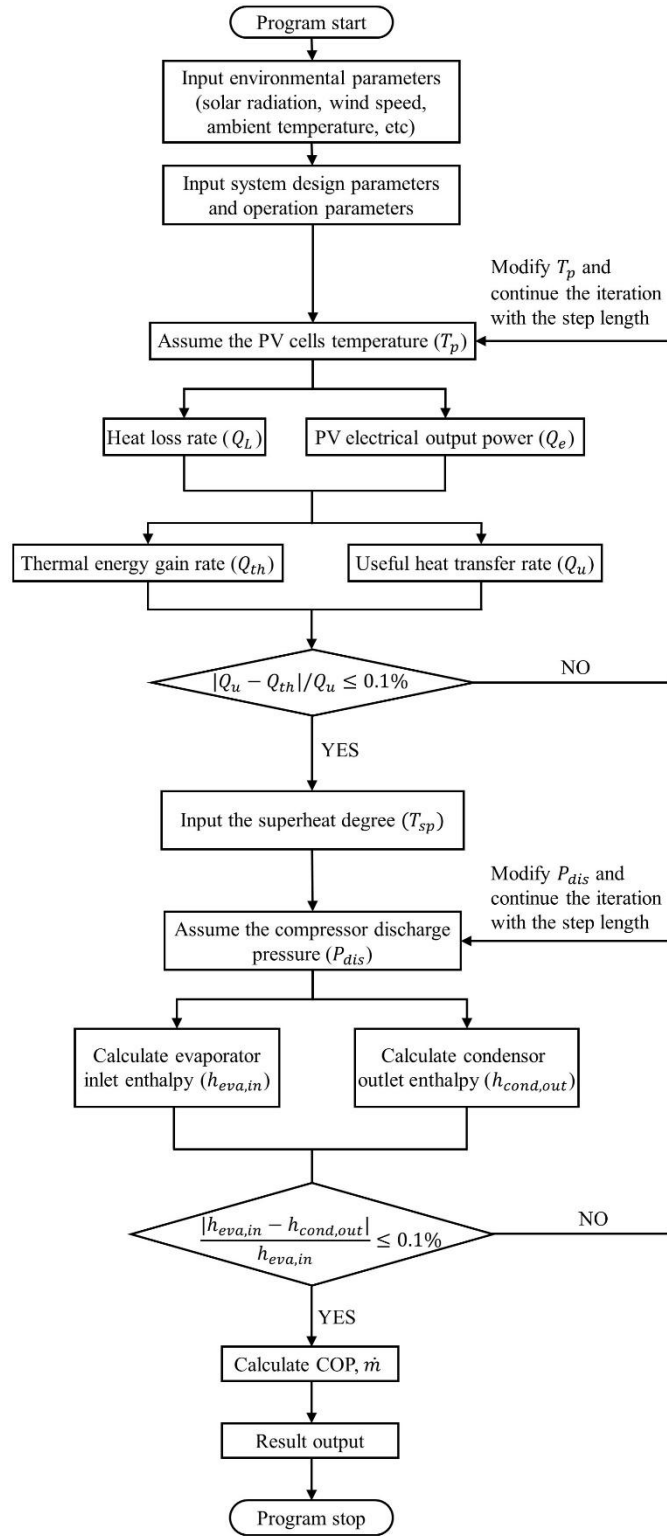


Fig. 10. Numerical solution procedure of the system.

329
 330
 331

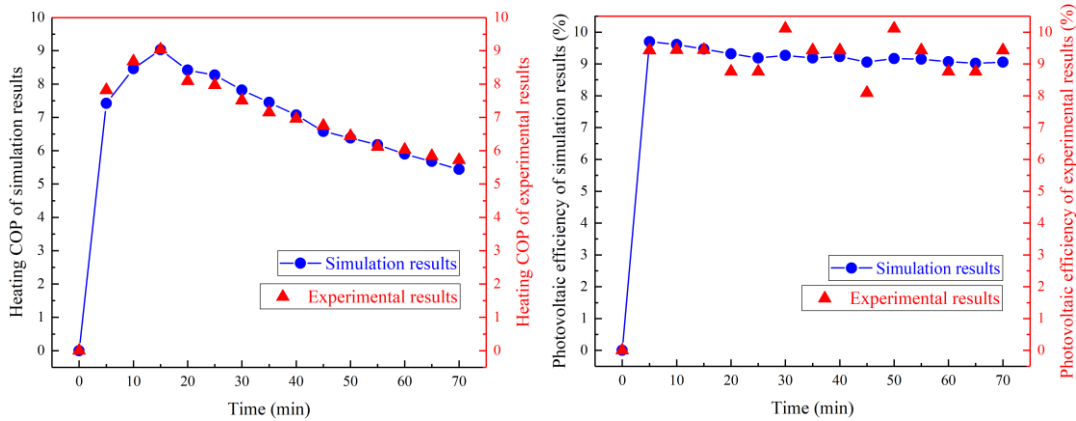
332 **4 Validation of the model**

333 To ensure the reliability of the mathematic model, the simulation results should be compared
 334 with the experimental results. The experimental parameters used in the simulation are listed in
 335 Table. 3.

336 **Table. 3.** Experimental parameters (Zhou et al., 2019).

Parameters	Nomenclature	Value	Unit
Thickness of glass cover	δ_g	3.2	mm
Thickness of air gap	δ_{air}	35	mm
Thickness of PV-glazing cover	$\delta_{g,pv}$	1	mm
Thickness of PV cells	δ_{pv}	0.2	mm
Thickness of EVA adhesive film	δ_{EVA}	0.4	mm
Thickness of electrical insulation	δ_{ei}	0.5	mm
Length of PV/T	L	3.0	m
Width of PV/T	W	1.6	m
Wind speed	v_{wind}	1.5	m/s
Ambient temperature	T_a	298.5	K
Refrigerant type	ref	R22	[-]
Packing factor	β_p	0.64	[-]

337
 338 The comparison results of heating COP are presented in Fig. 11(a), the operating conditions
 339 are refer from Zhou et al.(Zhou et al., 2019). Under the same system components (PV/T
 340 collector/evaporator, compressor, condenser, expansion valve, water tank), the simulation results
 341 are in good agreement with the experimental results. Heating COP of the PV/T system increases in
 342 the first 15 minutes because the water in the tank still in the low temperature range. Thus, the
 343 temperature differences between the refrigerant fluid and water remains large in the condenser
 344 leading to a high heat transfer efficiency. However, the heat transfer efficiency of the condenser
 345 would be decreased when the temperature of the inlet water rises up during the operation of the
 346 whole system. That is the reason of the reduction of heating COP after 15 minutes. The average
 347 error of heating COP is 2.84% while the maximum error is 5.12%. Fig. 11(b) presents the
 348 experimental and simulation results of the photovoltaic efficiency. The maximum photovoltaic
 349 efficiency is 10.11% while the minimum is 8.09%, but all the experimental photovoltaic
 350 efficiencies fluctuate around 9.24%. The simulation photovoltaic efficiencies remain around 9.25%
 351 but fluctuate from 9.01% to 9.7% due to the influence of solar radiation intensity. The average
 352 error of photovoltaic efficiency is 4.48% while the maximum error is 9.30%.



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(a) (b)
Fig. 11. (a) Comparison results of experimental and simulated heating COP. **(b)** Comparison results of experimental and simulated photovoltaic efficiency.

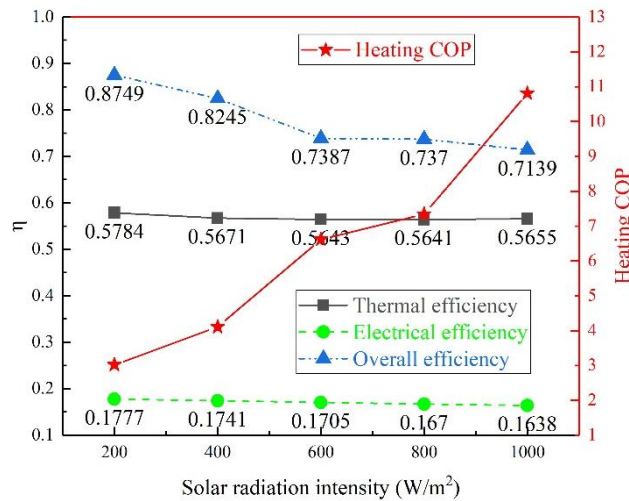
358 **5 Parameter analysis**

359 In this section, the influences of different parameters (solar radiation intensity, ambient
360 temperature, wind speed, area of PV/T collector) on this system are investigated, and the
361 performance indices of the system under typical working conditions are also given. It should be
362 noted that when one parameter is varied, others keep constant. Pressure ratio of the compressor
363 refers the ratio of pressure of discharged refrigerant vapor and charged refrigerant vapor.

364 **5.1 Solar radiation intensity**

365 The influences of solar radiation intensity which varying from 200 W/m² to 1000 W/m² are
366 shown as follows at the working conditions are: ambient temperature is 25 °C, wind speed is 1.5
367 m/s and area of PV/T collector is 2 m².

368 Fig. 12 presents the rising curve of the heating COP and declining curves of thermal,
369 electrical and overall efficiencies. The heating COP is 3.0 for a solar radiation of 200 W/m², and it
370 can reach up to 10.8 when the solar radiation is 1000 W/m². The refrigerant evaporation
371 temperature and pressure will be increased due to a higher temperature of PV cells. Thus, the
372 compressor consumes less electricity to compress the refrigerant vapor leading to this upward
373 trend. Meanwhile, the heat loss will mount due to a higher temperature difference between PV
374 cells and ambient. Thereby, the thermal and electrical efficiencies of the PV/T panel reduce
375 resulting in a reduction of the overall efficiency.

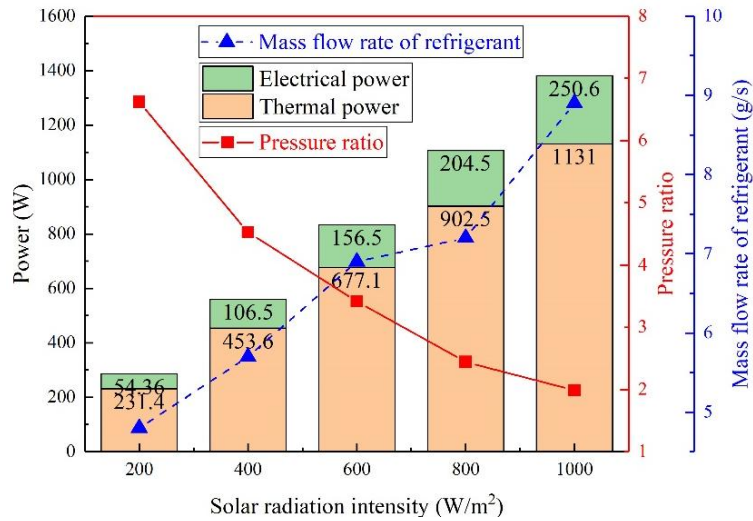


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Fig. 12. Influence of solar radiation intensity on heating COP and thermal, electrical, overall efficiencies.

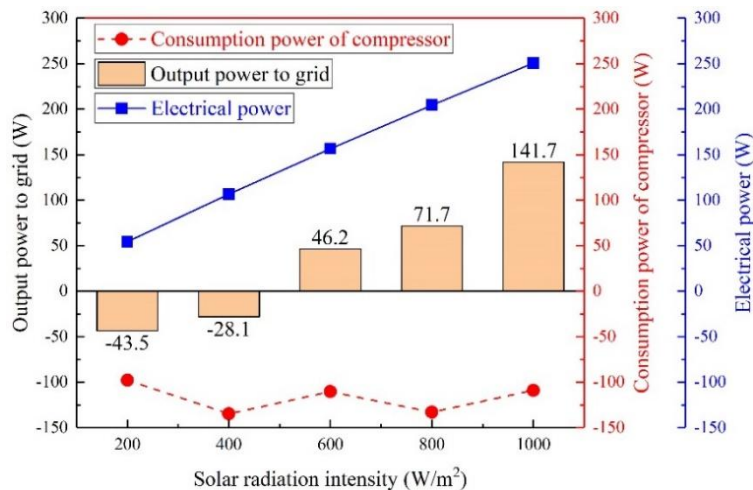
380 Fig. 13 shows the effect of solar radiation intensity on thermal and electrical output power,
381 pressure ratio and mass flow rate of refrigerant. The thermal and electrical power keep mounting
382 with the increase of the solar radiation intensity. Meanwhile, the pressure ratio of the compressor
383 decreases and the mass flow rate of the refrigerant increases. That is because the increase of the

384 evaporation temperature causes a higher evaporation pressure which equals to the suction pressure
 385 of compressor leading to a lower pressure ratio. Furthermore, a larger amount of refrigerant will
 386 be needed to transfer extra heat from PV/T collector/evaporator to PCM heat exchanger
 387 (condenser) when the PV/T panel absorbs more heat from solar radiation.



388
 389 **Fig. 13.** Influence of solar radiation intensity on electrical and thermal power, pressure ratio and
 390 mass flow rate of refrigerant.
 391

392 As shown in Fig. 14, there is a positive linear correlation between PV cells' electrical power
 393 generation and solar radiation intensity while the consumption power of compressor fluctuates
 394 around 120 W. When the output power to grid is less than zero, it means the system consumes
 395 electricity from the power grid. The electrical power generated by PV panels could meet the
 396 demand of the compressor and the system could output electricity to power grid when the solar
 397 radiation intensity exceeds 500 W/m².



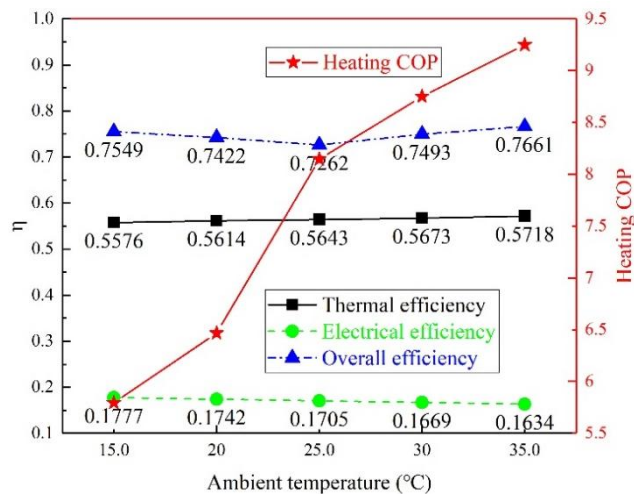
398
 399 **Fig. 14.** Influence of solar radiation intensity on electrical power, consumption power of
 400 compressor and output power to grid.
 401

402 5.2 Ambient temperature

403 The influences of ambient temperature which varying from 15 °C to 35 °C are shown as
 404 follows at the working conditions are: solar radiation intensity is 600 W/m², wind speed is 1.5 m/s

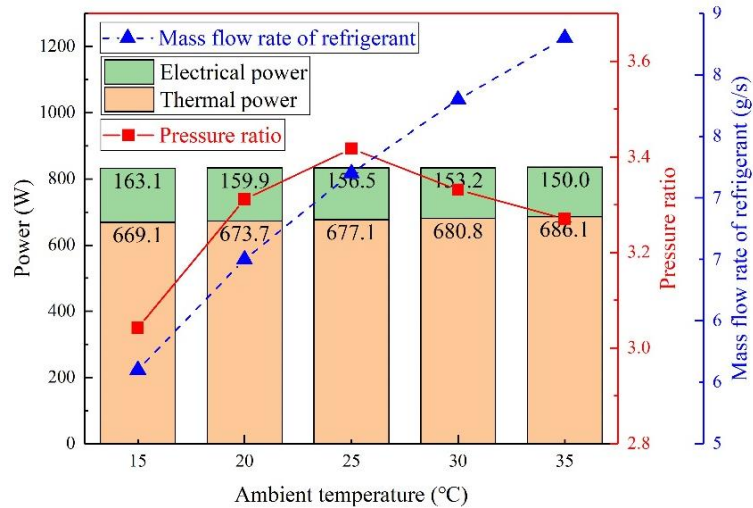
405 and area of PV/T collector is 2 m^2 .

406 Fig. 15 shows that increasing the ambient temperature will increase the heating COP and
407 thermal efficiency but decrease the electrical and overall efficiencies. The temperature difference
408 between PV cells and ambient will reduce when the ambient temperature rises leading to a less
409 heat loss from PV/T panel to the surrounding. Thus, the heating COP increases and the electrical
410 efficiency decreases due to a higher temperature of the PV cells while the thermal efficiency
411 increases. However, the electrical efficiency outweighs the thermal efficiency resulting in a
412 reduction of the overall efficiency when the ambient temperature is below 25°C . The heating COP
413 (9.25) at 35°C is higher than COP (5.79) at 15°C by 59.8%, thus a higher ambient temperature is
414 better for the system performance.



415
416 **Fig. 15.** Influence of ambient temperature on heating COP and thermal, electrical, overall
417 efficiencies.
418

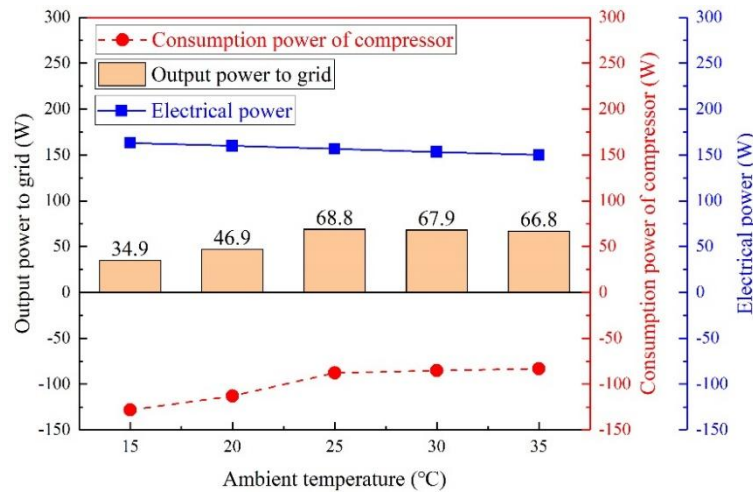
419 The variation curves of thermal and electrical power, pressure ratio and mass flow rate of
420 refrigerant with the increase of ambient temperature are shown in Fig. 16. The changing curves of
421 the electrical and thermal output power are the same as the electrical and thermal efficiencies. The
422 pressure ratio of the compressor will increase when the ambient temperature is below 25°C and
423 decrease when the ambient temperature is over 25°C . That is because a lower ambient temperature
424 leads to a lower superheat degree of refrigerant which cause a lower pressure ratio, and in the
425 opposite, a higher ambient temperature leads to a higher superheat degree. Moreover, when the
426 ambient temperature exceeds 25°C , it will influence the thermal efficiency of PV/T collector
427 causing the reduction of pressure ratio. The mass flow rate of the refrigerant will keep climbing
428 when the ambient temperature rises because less heat will lose in the ambient while more heat will
429 be absorbed by refrigerant.



430

431 **Fig. 16.** Influence of ambient temperature on electrical and thermal power, pressure ratio and mass
 432 flow rate of refrigerant.
 433

434 As shown in Fig. 17, the electrical power of PV decreases linearly with the ambient
 435 temperature while the consumption power of compressor decreases when the ambient temperature
 436 is below 25 °C and increases when the ambient temperature exceeds 25 °C. The output power to
 437 grid reaches its maximum at 25 °C. Because when the ambient temperature is below 25 °C, the
 438 effect of environmental heat loss is greater than that of heat-collecting efficiency due to a large
 439 temperature difference. However, the effect of a higher ambient temperature on heat-collecting
 440 efficiency is greater than that of environmental heat loss when the ambient temperature exceeds
 441 25 °C.



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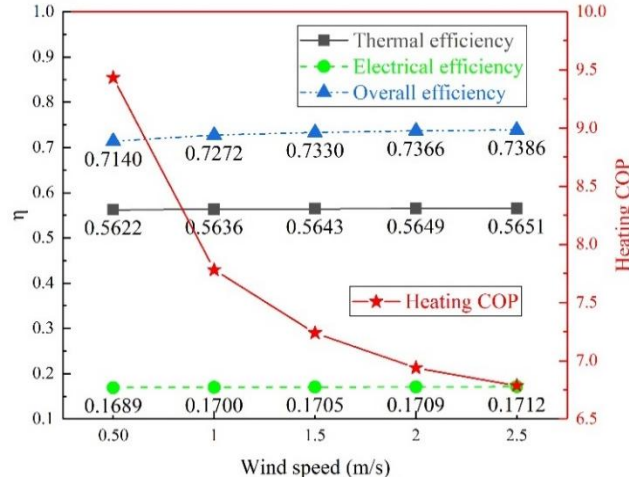
443 **Fig. 17.** Influence of ambient temperature on electrical power, consumption power of compressor
 444 and output power to grid.
 445

446 5.3 Wind speed

447 The influences of wind speed which varying from 0.5 m/s to 2.5 m/s are shown as follows at
 448 the working conditions are: solar radiation intensity is 600 W/m², ambient temperature is 25 °C and
 449 area of PV/T collector is 2 m².

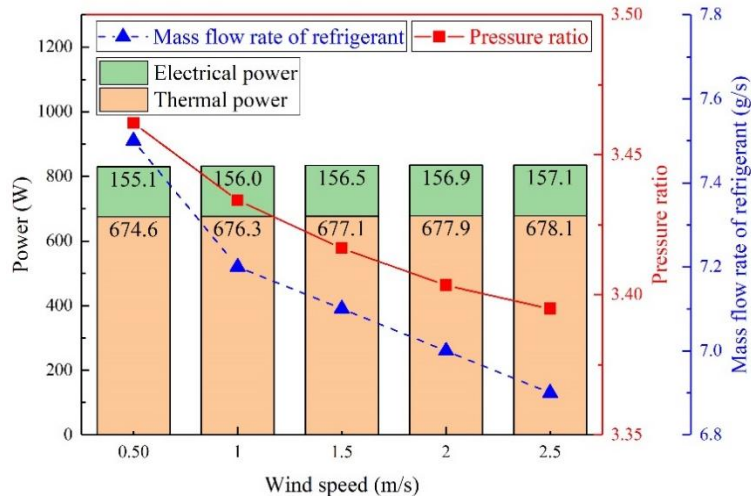
450 Fig. 18 shows the variation curve of heating COP, thermal, electrical and overall efficiencies

451 with wind speed varying from 0.5 m/s to 2.5 m/s. The heating COP will decrease rapidly when the
 452 wind speed is low and steadily when the wind speed increases. More heat will be transferred to the
 453 environment and less heat is absorbed by the PV/T panel under a higher wind speed. Meanwhile,
 454 evaporation temperature of refrigerant will be decreased and the consumption power of
 455 compressor will be increased when the temperature of PV cells rises. Thus, the heating COP drops
 456 from 9.4 to 6.8, which means better wind protection measures should be taken to improve the
 457 system performance.



458
 459 **Fig. 18.** Influence of wind speed on heating COP and thermal, electrical, overall efficiencies.
 460

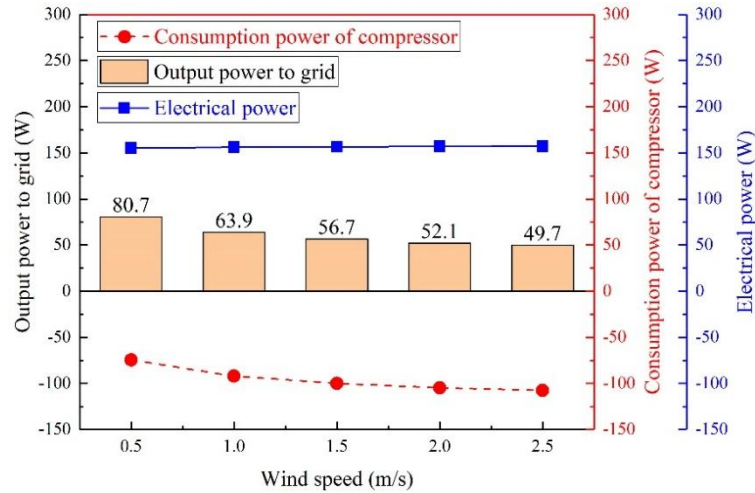
461 The variation curves of thermal and electrical power, pressure ratio and mass flow rate of the
 462 refrigerant with the increase of wind speed are shown in Fig. 19. The changing curves of the
 463 thermal and electrical power have the same trend as the thermal and electrical efficiencies. Both
 464 the pressure ratio and mass flow rate of refrigerant reduce with the increase of the wind speed.
 465 That is because more heat will be absorbed by the ambient air while less heat will be transferred
 466 by the refrigerant.



467
 468 **Fig. 19.** Influence of wind speed on electrical and thermal power, pressure ratio and mass flow
 469 rate of refrigerant.
 470

471 Fig. 20 presents the influence of wind speed on electrical power, consumption power of
 472 compressor and output power to grid. The output power to grid decreases rapidly when the wind

473 speed increases from 0.5 m/s to 1.5 m/s and steadily when the wind speed exceeds 1.5 m/s. More
 474 heat will loss in the environment due to a higher wind speed, and the thermal efficiency will
 475 decrease causing a higher consumption power of compressor.

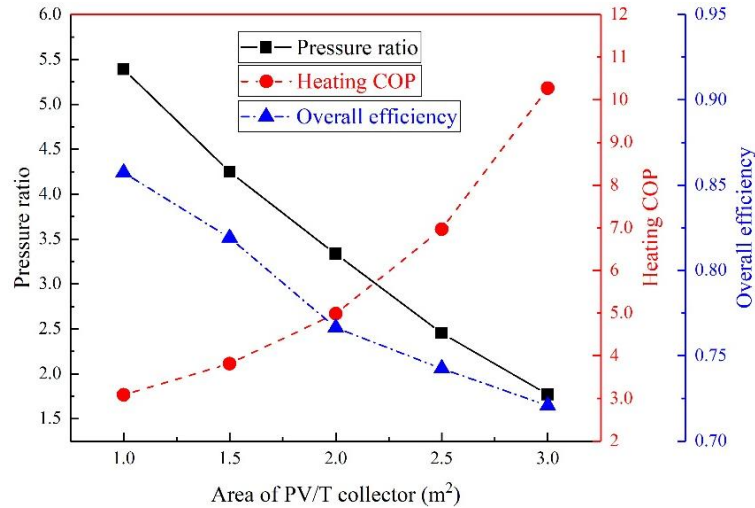


476
 477 **Fig. 20.** Influence of wind speed on electrical power, consumption power of compressor and
 478 output power to grid.
 479

480 5.4 Area of PV/T collector

481 The influences of the area of PV/T collector which varying from 1 m² to 3 m² are shown as
 482 follows at the working conditions are: solar radiation intensity is 600 W/m², ambient temperature
 483 is 25 °C and wind speed is 1.5 m/s.

484 Fig. 21 presents the influence of the area of PV/T collector on pressure ratio, heating COP
 485 and overall efficiency. Pressure ratio and overall efficiency reduce with the increase of the area,
 486 while the heating COP mounts. That is because a larger area can absorb more heat from the solar
 487 radiation, and the extra heat will be transferred by PV/T panel to refrigerant. Meanwhile, the
 488 evaporation temperature and pressure will increase leading to a lower pressure ratio and a higher
 489 heating COP. The consumption power of the compressor will decrease when the pressure ratio
 490 reduces, thus the heating COP increases more rapidly under large area conditions. A higher
 491 temperature of the PV cells causes more heat dissipates in the ambient resulting a decrease of the
 492 overall efficiency.

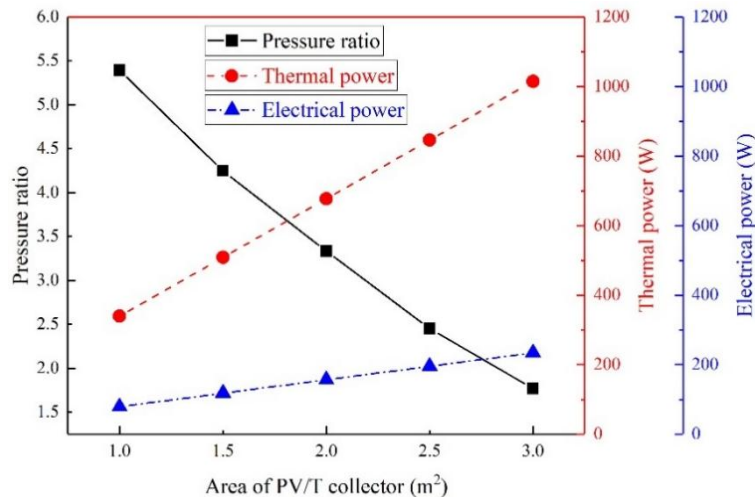


493

494 **Fig. 21.** Influence of the area of PV/T collector on pressure ratio, heating COP and overall
 495 efficiency.
 496

497

498 Fig. 22 shows the declining trends of the pressure ratio and increasing trends of the thermal
 499 and electrical power influenced by the area of PV/T collector. The thermal and electrical output
 500 power are almost linearly and positively correlated with the area of PV/T collector. That is
 501 because more heat will gain from the solar radiation and be transferred to the refrigerant when the
 area increases.

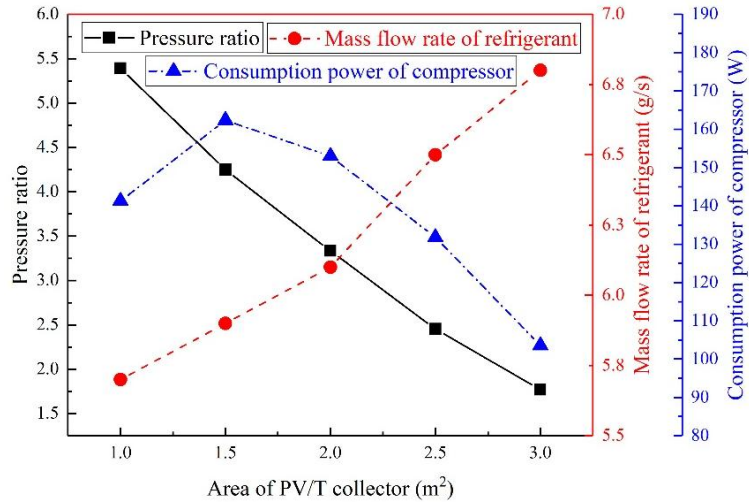


502

503 **Fig. 22.** Influence of the area of PV/T collector on pressure ratio, thermal and electrical power.
 504

505

506 As shown in Fig. 23, the mass flow rate of refrigerant will increase when the area increases,
 507 while the consumption power of compressor will increase when the area is below 1.5 m² and
 508 decrease when the area is over 1.5 m². The mass flow rate of refrigerant will increase because the
 509 latent heat and heat capacity of refrigerant are the same when more heat is transferred to the
 510 refrigerant. The heat absorbed by the PV/T panel and mass flow rate of refrigerant are low, thus
 511 the compressor consumes less electricity to compress the refrigerant vapor. With the increase of
 512 the area, the mass flow rate mounts resulting in an increase of consumption power. However, the
 513 pressure ratio of the compressor will reduce when the area increases over 1.5 m² leading to a
 lower consumption power.

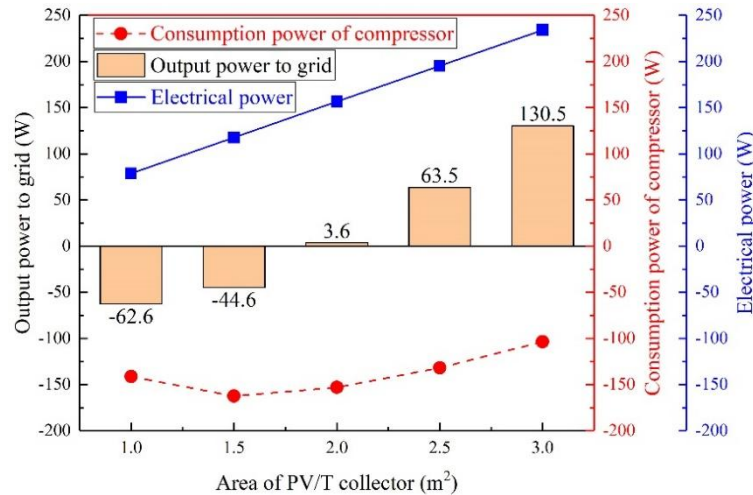


514

515 **Fig. 23.** Influence of the area of PV/T collector on pressure ratio, mass flow rate of refrigerant and
 516 consumption power of compressor.

517

518 Fig. 24 illustrates the variation curve of output power to grid with the area of PV/T collector.
 519 The electrical power generated by PV panels will meet the demand of compressor when the area of
 520 PV/T collector is 2 m². Moreover, the PV cells start to produce electricity to power grid when
 521 the area is over 2 m² which means a larger PV/T panel is better for the system performance under
 522 the same system conditions.



523

524 **Fig. 24.** Influence of area of PV/T collector on electrical power, consumption power of
 525 compressor and output power to grid.

526

527 5.5 Feasibility analysis of the system

528 Nowadays, the Chinese government promotes the policy of using electricity for residential
 529 heating instead of burning coal for heating in northern China to reduce carbon dioxide emissions.
 530 The solar PV/T heat pump coupled with build-in PCM heat storage system is suitable for
 531 residential heating due to its advantages: (1) high efficiency; (2) low energy consumption; (3)
 532 stable residential heating supply; (4) zero carbon emissions. Table. 4 presents the typical operating
 533 conditions and parameters of the system. It is a typical spring/autumn day in northern China.

534

Table. 4. Typical operating conditions and parameters of the system.

Parameters	Nomenclature	Value	Unit
Solar radiation intensity	I	600	W/m ²
Sunshine duration	t_s	8	hour
Ambient temperature	T_a	15	°C
Wind speed	v_{wind}	1.5	m/s
Area of the collector	A	20	m ²
Packing factor	β_p	1	[-]
Heating area	A_{ha}	100	m ²
Heat loss per square meter	P_L	50	W/m ²
Filling volume of the PCM	V_{PCM}	0.57	m ³
External diameter of the inside pipe	D_1	0.012	m
Internal diameter of the inside pipe	d_1	0.010	m
External diameter of the outside pipe	D_2	0.074	m
Internal diameter of the outside pipe	d_2	0.072	m
Length of the PCM heat exchanger	L_{hx}	164	m

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Table. 5 presents the simulation performance indices of the system under typical day conditions. The PV/T collector can transfer 213.9 MJ heat from the ambient to the PCM heat exchanger, and the heat will be stored in phase change materials and concrete. The build-in PCM heat storage can release the heat for 10 hours at 5.94 kW during the night while the heat loss power of 100 m² area is 5 kW. During autumn, winter and early spring in northern China, the heating system is necessary to keep indoor temperature above 20°C. Thus, the system can achieve the heating needs of the users and keep the indoor temperature steady. Meanwhile, the system can output 21.4% (2.79 kWh) of the power generated by PV panels to the grid while 78.6% of it consumed by the compressor. The heating COP of the system is 5.79, and the overall efficiency is 75.49%.

Table. 5. Simulation performance indices of the system under typical day conditions.

Parameters	Value	Unit
Total heat storage	213.9	MJ
Photovoltaic power	1.63	kW
Photovoltaic efficiency	17.77	%
PV/T thermal efficiency	55.76	%
Heating COP	5.79	[-]
Overall efficiency	75.49	%
Cumulative power generation	13.05	kWh
Consumption power of compressor	10.26	kWh
Output power to grid	2.79	kWh
Temperature range of underfloor heating	22-31	°C
Heating power at night	5.94	kW
Heat loss power	5.00	kW
Heating hour	10.00	hour

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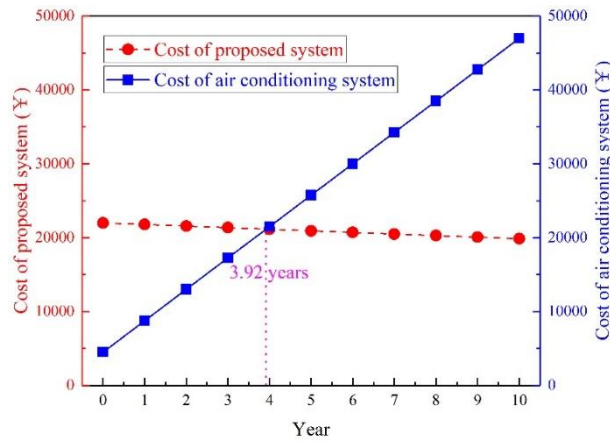
Table. 6 illustrates the comparison of cost between the proposed system and conventional air

549 conditioning system. The operating cost of the proposed system is under zero because users could
 550 sell spare electricity to power grid and get profit. Fig. 25 shows the cost variation curves of these
 551 two systems. The initial cost of the proposed system is much higher than conventional air
 552 conditioning system due to the underfloor heating equipment and PV/T panels, etc. However, the
 553 air conditioning system consumes a lot of electricity during the night for heating supply. Thus, the
 554 cost of these two systems will be the same after about 4 years, and the cost of proposed system
 555 keeps reduce while the cost of air conditioning system still climbs. Moreover, underfloor heating
 556 system which using radiative heating is more comfort and silence for users than air conditioner.

557 **Table. 6.** Cost comparison between the proposed system and conventional air conditioning
 558 system.

Heating system	Initial cost (¥)	Operating cost (¥/year)	Maintenance cost (¥/year)
Proposed system	22000	-764	550
Air conditioning system	4500	4024	225

559



560

561 **Fig. 25.** Cost variation curves of proposed system and air conditioning system.
 562

563 6 Conclusion

564 A building-coupled cogeneration system using solar PV/T heat pump and build-in PCM heat
 565 storage is proposed in this paper. The mathematical model of the system is established and verified
 566 to analyze the system performance under different conditions. The main conclusions can be drawn
 567 as follows:

568 (1) The temperature of underfloor heating which using build-in PCM heat storage can reach
 569 22 °C to 31 °C after 39 hours when the circulating water is 40 °C which is stable and suitable for
 570 residential heating.

571 (2) The heating COP can reach 6.6 which is 94% higher than conventional air conditioning
 572 system when solar radiation intensity is 600 W/m², ambient temperature is 25 °C, wind speed is 1.5
 573 m/s and area of PV/T collector is 2 m² while the electrical, thermal and overall efficiencies are
 574 17.05%, 56.43% and 73.87%, respectively.

575 (3) A 2 m² PV/T panel can meet the power demand of the system and heating demand of a 10
 576 m² room when the solar radiation intensity is 500 W/m². Moreover, the PV/T panel can output
 577 electricity to power grid if the panel area is bigger than 2 m² or solar radiation intensity is higher

578 than 500 W/m^2 .

579 The mathematical model established in this paper can also be used to analyze and optimize
580 the solar PV/T heat pump system. However, the mathematical model of this system is for stable
581 working conditions instead of transient. The establishment of dynamic model is needed for further
582 predict and analyze accurately of solar assisted heat pump system under dynamic working
583 conditions.

584 **Acknowledgements**

585 This research work is funded by the International Research Cooperation Program of Shanghai
586 (Grant No. 18160710500).

587 **Nomenclature:**

588 Symbols

A	area (m^2)
W	width of the PV/T collector/evaporator (m)
L	length of the PV/T collector/evaporator (m)
ΔH	latent heat (kJ/kg)
h	heat transfer coefficient ($\text{W/m}^2\cdot\text{K}$)
U	heat loss coefficient ($\text{W/m}^2\cdot\text{K}$)
C_p	specific heat at constant pressure (kJ/kg·K)
d	inner diameter (m)
D	external diameter (m)
t/T	temperature (K)
I	solar radiation intensity (W/m^2)
Q	heat transfer rate (W)
F	collector efficiency (-)
R	thermal resistance ($\text{m}^2\cdot\text{K/W}$)
Re	Reynolds number (-)
Ra	Rayleigh number (-)
Pr	Prandtl number (-)
v	wind speed (m/s)
m	mass flowrate (kg/s)
g	gravitational acceleration (m/s^2)
p	pressure (kPa)
P	power (W)
V	volume flow rate (m^3/h)

589

590 Greek symbols

δ	thickness (m)
τ	transmittance (-)

a	absorption ratios (-)
β	packing factor (-)
ε	emissivity (-)
κ	thermal conductivity (W/m·K)
$\bar{\sigma}$	Stefan-Boltzmann constant (-)
ρ	density (kg/m ³)
λ	compressor volumetric efficiency (-)
η	efficiency (-)
χ	polytropic index (-)

591

592 Subscripts

p,pv	PV cells
g,c_1	external glass cover
c/c_2	PV-glazing cover
$conc$	concrete
EVA	EVA grease
PCM	phase change material
mel	melting point
ref	refrigerant
b	baseboard
cv	convection
rd	radiation
rb	roll-bond panel pipe
a	ambient
l	liquid
L	lost
rc	reference
e	electrical
ei	electrical insulation
u	uesful
th/R	thermal
tp	two-phase flow
ove	oveall
abs	absorb
hp	heat pipe
sh	superheated
v	vapor
$cond$	condensation
eva	evaporation
ot	outer pipe
eq	equivalent
hx	heat exchanger
dis	discharge
suc	suction

in inlet
out outlet
cw circulating water

593

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Conflict of Interest

We declare that there are no conflicts of interest in the work we submitted.



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Dear Editor,

Enclosed please find our manuscript entitled: “**Performance analysis of solar assisted heat pump coupled with build-in PCM heat storage based on PV/T panel**” (written by J. Yao, H. Xu, Y. J. Dai *, M. J. Huang), for possible publication in **SOLAR ENERGY**. The paper has been prepared according to the guidelines listed in the “Guide for Authors-Contents List”. The paper has been received by ‘*1st International Conference for Global Chinese Academia on Energy and Built Environment*’, and the number of **Abstract** is **2019-601**.

We appreciate your consideration of our manuscript, and look forward to receiving comments from the reviewers. Please acknowledge receipt of this manuscript at your convenience, and let us know if you need any further information.

Thanks for your consideration.

Sincerely,

A handwritten signature in black ink that reads "Yanjun Dai".

Prof. Yanjun Dai (Y.J. Dai)

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