

Ultra-low temperature heating system based on dual-source solar assisted heat pump using compound parabolic concentrator-capillary tube solar collector

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ARTICLE INFO

Keywords:

Low temperature heating
Compound parabolic concentrator-capillary tube solar collector
Solar assisted air source heat pump
Domestic heating
Seasonal performance factor

ABSTRACT

The fifth-generation heating – ultra-low temperature heating benefits to reduce electricity consumption and achieve the net zero goal. The dual-source solar assisted heat pump based heating system has been demonstrated to be an attractive green heating technology for the domestic sector. However, the slower response speed of the low temperature heating to the variation of heating load in responding to the variations of weather conditions limits its thermal comfort performance. The enhancement in solar collector performance brings valuable improvements in response speed of the heating system. In the present work, a heating system based on solar assisted air source heat pump using a compound parabolic concentrator-capillary tube solar collector (CPC-CSC) is investigated with the set heating temperatures of 40, 45, 50, and 55 °C. This heating system works for both space heating and hot water under the weather conditions in London. The results suggest that using a concentrated solar collector improves the response speed of the heating system at low set heating temperatures. For such a heating system, the ultra-low heating temperature increases the application of renewable energy and passive heating (by 6.4%). Compared with the dual-source indirect expansion solar assisted heat pump using flat plate collector, the heating system using CPC-CSC can reduce TEWI by 4.6% with a slightly longer (1.9%) payback period. As the set heating temperature decreases from 55 to 40 °C, the seasonal and yearly system seasonal performance factors significantly increase by 17.1% and 20.5%, respectively.

1. Introduction

Low temperature heating system refers to heating system providing hot water at a temperature within 45 °C [1]. The concept of low temperature heating has been widely adopted for district heating [2] and is strongly urged for decarbonisation in the heating sector [3]. For distributed heating, Kilis [4] highlighted the benefits of low temperature heating in technology, environment and economic terms. Heat pump (HP) is a promising distributed heating technology to introduce low temperature heating to the domestic sector [5]. It can be combined with solar energy [6], i.e., solar assisted air source heat pump (SAASHP), for better system efficiency [7]. SAASHP can be divided into direct expansion type (DX-SAASHP) and indirect expansion type (IX-SAASHP) by expansion type; and can also be divided into serial, parallel and dual

source types according to the arrangement of heat sources. According to a review from Yang et al. [8], IX-SAASHP has more application potential for both space heating (SH) and hot water (HW). Simulation results from Yang et al. [9] suggested the possibility of applying parallel and dual-source IX-SAASHP for UK weather conditions.

Current research on low temperature heating mainly focuses on district heating while attention to distributed heating, especially SAASHP heating systems, is insufficient. Kaygusuz [10] conducted experiments and theoretical analyses on a serial-parallel IX-SAASHP with condensing temperatures of 40–55 °C. The system obtained coefficient of performance (COP) of 4 in serial mode and 3 in parallel mode. Yerdash et al. [11] studied a solar assisted cascade HP using different refrigerant pairs. The system has a condensing temperature of 40–60 °C and COP of 1.8–3. Qiu et al. [12] analysed a cascade serial IX-SAASHP and two kinds of two-stage dual-source DX-SAASHP. The condensing

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<https://doi.org/10.1016/j.enbuild.2024.114903>

Received 24 June 2024; Received in revised form 23 September 2024; Accepted 8 October 2024

Available online 10 October 2024

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Nomenclature		v_a	wind speed
A	collector area	W_{HP}	electricity consumed by a heat pump
COP	coefficient of performance	W_{tot}	total electricity consumed
Q_{HP}	thermal energy obtained at the condenser of a heat pump	Abbreviation	
Q_{HW}	thermal energy for hot water	ASHP	air source heat pump
$Q_{loss, SC}$	heat loss from solar collector	CPC-CSC	compound parabolic concentrator-capillary tube solar collector
Q_{loss}	linear heat loss from solar collector	DX-SAASHP	direct expansion solar assisted air source heat pump
Q_{SC}	thermal energy obtained by solar collector	DSSAHP	dual-source solar assisted heat pump
Q_{SH}	thermal energy for space heating	FPC	flat plate collector
Q_{su}	solar energy used	HP	heat pump
Q_{sup}	thermal energy supply	HW	hot water
Q_{TES}	thermal energy storage	IX-SAASHP	indirect expansion solar-assisted air source heat pump
I	local solar irradiance for the tilted surface	SAASHP	solar-assisted air source heat pump
SF	solar fraction	SFH	single family house
SPF_{HP}	seasonal performance factor of the heat pump	SH	space heating
SPF_{sys}	seasonal performance factor of the system	SHW	solar hot water
T_{amb}	ambient air temperature	SWHP	heat pump used hot water from solar collector as heat source
T_{indoor}	indoor air temperature	TES	thermal energy storage
T_{SC}	temperature of solar collector	TRNSYS	TRaNsient SYstem Simulation program
T_{HWS}	actual heating temperature (hot water temperature at the outlet of TES tank 2)		
T_{HWS}^*	set heating temperature		

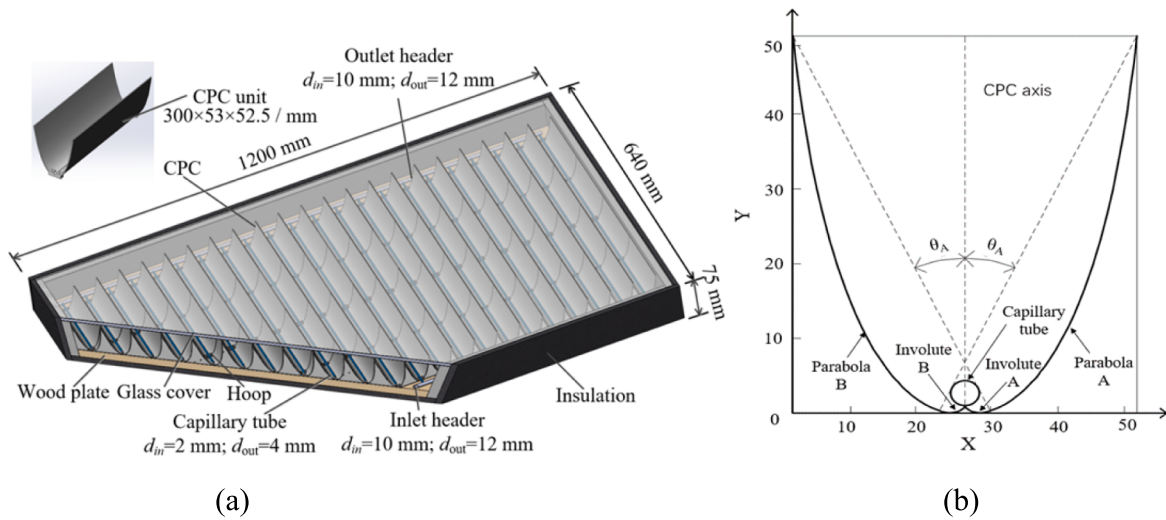


Fig. 1. Schematic of CPC-CSC: (a) 3D model (b) 2D model [19].

Table 1
Summary of experimental conditions.

	Solar irradiance, W/ m ²	Ambient air temperature, K	Air velocity, m/s	Date of experiments
1	290–1000	288–291	1–2	10 Oct. 2016—20 Nov. 2016
2	920–1000	289–293	1–2	5 Apr. 2017—10 May 2017

temperature of the systems was 50 °C and the COP is 2–2.9. Yang et al. [13] numerically simulated a dual-source IX-SAASHP with hot water supply temperatures of 40–55 °C for UK weather conditions where seasonal performance factor (SPF) can reach 5.0. Though low-temperature heating shows potential to achieve a higher system

efficiency, its low response speed to heating load limits its thermal comfort performance.

Innovation in system components, such as solar collectors, is a promising method to improve system operation performance. In addition to the commonly used flat plate solar collector (FPC), many kinds of advanced solar collectors have been designed and adopted for SAASHP, such as evacuated tube solar collectors [14], heat pipe solar collectors [15], solar roof [16], air-type solar collector [17] and compound parabolic concentrator (CPC) solar collector [18]. However, few studies have been conducted to use advanced collectors to improve low temperature heating. Among the advanced solar collectors, CPC collectors help to overcome the limit of solar availability and obtain higher water temperature as it can concentrate indirect incidence to the absorber, performing the possibility to make up the thermal comfort performance of ultra-low heating. Our research group [19] has proposed a CPC-capillary tube solar collector (CSC) whose collection efficiency is superior to those

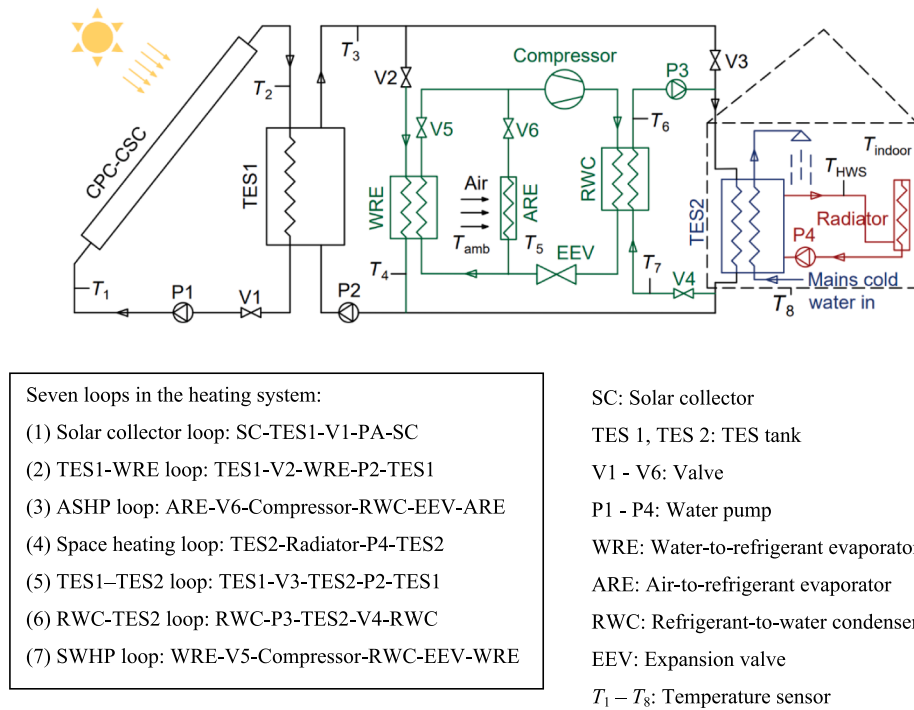


Fig. 2. Schematic of the DSSAHP system [21].

Table 2
modules used in the DSSAHP model in TRNSYS.

Component	Module	Parameter	Value
CPC-CSC	Type 219 (self-established)	Collector area	8 m ²
		Inclination angle	51.5°
FPC	Type 1b	Collector area	8 m ²
		Inclination angle	51.5°
		Tested flow rate	30 kg/hm2
		Intercept	0.8
		Efficiency	
		Efficiency slope	13 kJ/hm2k
		Efficiency curvature	0 kJ/hm2k2
TES tank 1	Type 4a	Heat loss coefficient	0.2 W/(m ² K)
		Volume	500 L
		Height	1.175 m
TES tank 2	Type 4a	Heat loss coefficient	0.2 W/(m ² K)
		Volume	300 L
		Height	1 m
ASHP	Type 941	Blower power	0.15 kW
		Total air flow rate	1500 l/s
		User defined file	YVAS012, York, Jonson Control
SWHP	Type 668	User defined file	30HXC-HP2, Carrier United Technologies

of other CPC collectors. In this research, the CPC-CSC is further applied in SAASHP heating system for low temperature heating.

This paper aims to investigate the operation performance for low temperature heating of a dual-source indirect expansion solar assisted heat pump (DSSAHP) using CPC-CSC. This system provides space heating and hot water for a single family house (SFH) 45 building under the weather conditions in London. The set heating temperature (T_{HWS}^*) is set to be 40 °C, 45 °C, 50 °C and 55 °C to represent the applications of low temperature heating (40 °C and 45 °C cases) and normal heating

(50 °C and 55 °C cases). To evaluate the advantage of using CPC-CSC on low temperature heating, the operation performance of the system using FPC under the same working conditions is also analysed for comparison.

2. Compound parabolic concentrator-capillary solar collector

CPC-CSC is composed of CPC concentrators, absorbers (capillary tubes), headers, cover and insulation, as Fig. 1(a) shows [19]. The schematic of the CPC concentrators is shown in Fig. 1(b). The geometry of the CPC concentrators is determined by the principle of edge optics to concentrate solar incidence onto the receiver. The groove depth of the CPC is 52.5 mm and the aperture is 53 mm. The outer diameter of the absorber is 4 mm, equal to the diameter of the involute circle for CPC. This can ensure all the reflection of the solar radiation to be concentrated on the capillary tube. The inner diameter of the capillary tube is 2 mm. The concentrating ratio of CPC is 4.22 and the acceptance angle is 13.7°.

2.1. Numerical simulation and verification

The CPC-CSC is numerically simulated by ANSYS Fluent using a 3-D model. The computation domain is selected to be one CPC unit to represent the whole collector. The length of the CPC unit is 600 mm, the width is 53 mm, the height is 75 mm and the area is 0.0318 m². Water is adopted to be the fluid inside the capillary tube. The CPC-CSC is assumed to be in an open environment and thus convective heat transfers on cover and bottom are considered. The solar collection process combines conduction, convection, and radiation heat transfer. Convective heat transfer occurs between the outer wall of the capillary tubes and the air layer. The water flow inside the tubes is regarded as a 3D, steady, constant-property, laminar flow. Since the diameter of the copper capillary tubes is small compared with the size of CPCs, they are regarded as a homogeneous body heat source. All the surface temperatures of solid components are obtained from the coupled numerical simulations of air convection inside the collector, water flowing inside the capillary tubes and heat conduction in the solids.

Corresponding experiments were undertaken in Beijing, China. The

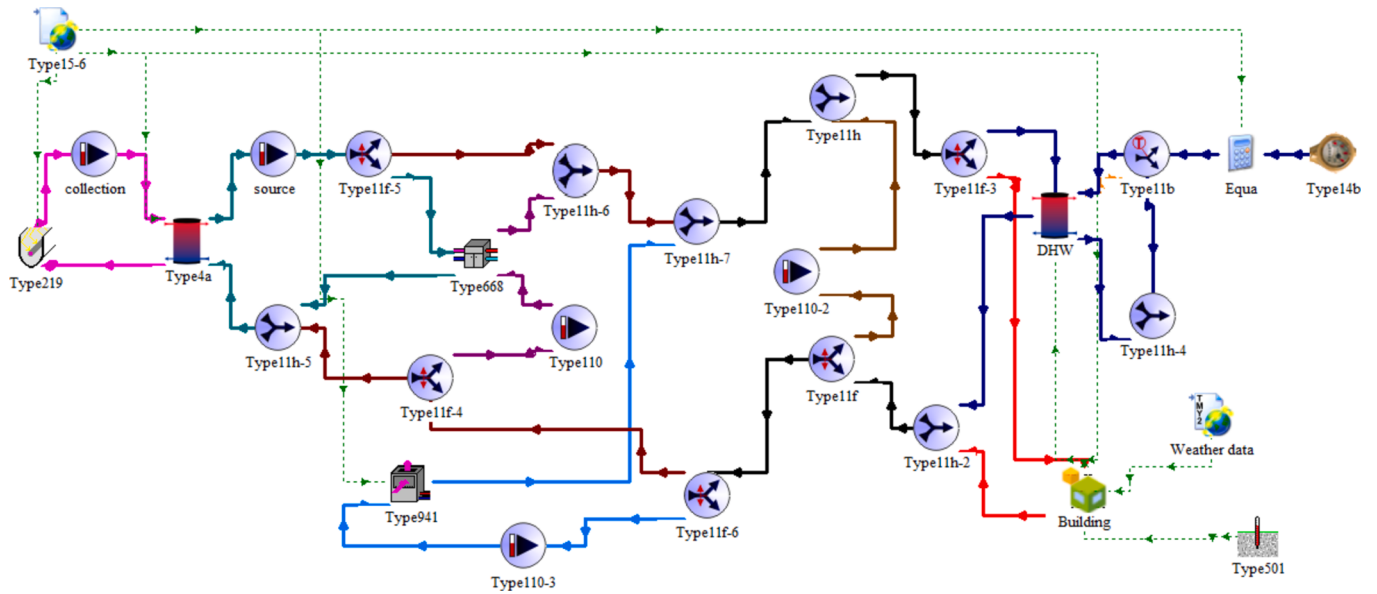


Fig. 3. TRNSYS model of the DSSAHP using CPC-CSC [21].

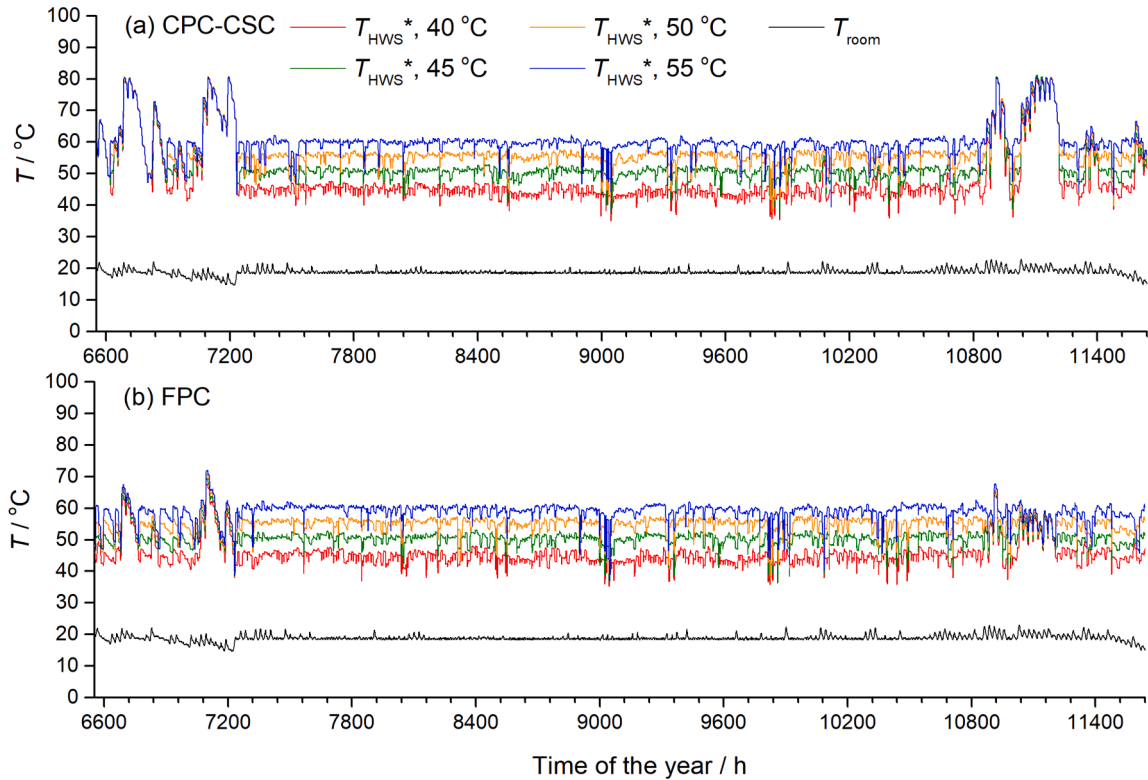


Fig. 4. Variations of indoor air temperature and hot water temperature at the outlet of TES tank 2 over a heating season for different T_{HWS}^* .

test rig consists of CPC-CSC in the east–west direction, a hot water circulation loop and a data acquisition system. Two T-type thermocouples with an accuracy of ± 0.1 K are inserted in the two mixers to measure the inlet and outlet bulk temperatures, respectively. 44 thermocouples are arranged at the inlet and outlet of each capillary tube. The turbine flow meter with an accuracy of 1 % measures the mass flow rate of water. The solar irradiation intensity is measured by TES1333R solar radiation recorder (with an accuracy of ± 10 W/m²).

Two experiments were conducted to test the influence of solar irradiance (290 W/m² to 1000 W/m²) and inlet water temperature (290 K to

345 K). The experimental conditions are summarised in Table 1. The errors between the numerical and experimental results are 8.5 % and 15 % for case 1 and case 2, respectively. The good agreements between experimental and simulated results suggest the validity of the model.

2.2. Component module and system model in TRNSYS

The CPC-CSC module in TRNSYS is self-established based on the CFD simulation results from the model by ANSYS Fluent. The useful thermal energy absorbed by the capillary tube, Q_{SC} , is calculated by Eq. (1):

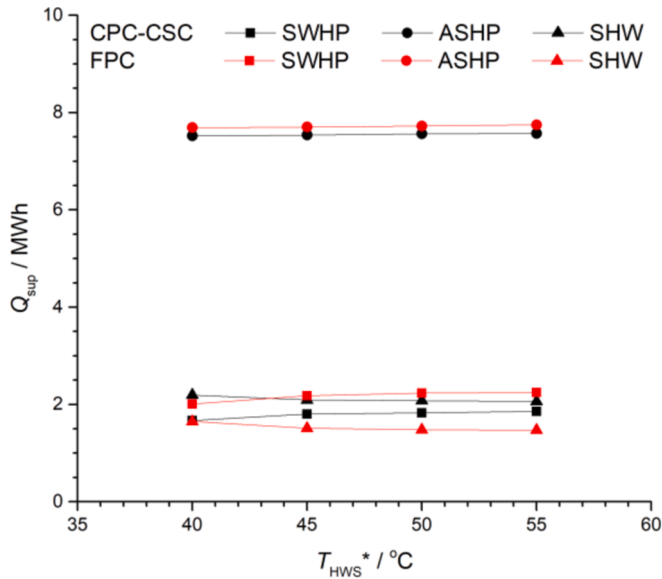


Fig. 5. Variations of heat for space heating and hot water by SWHP, ASHP and direct SHW against T_{HWS}^* for heating systems using CPC-CSC and FPC.

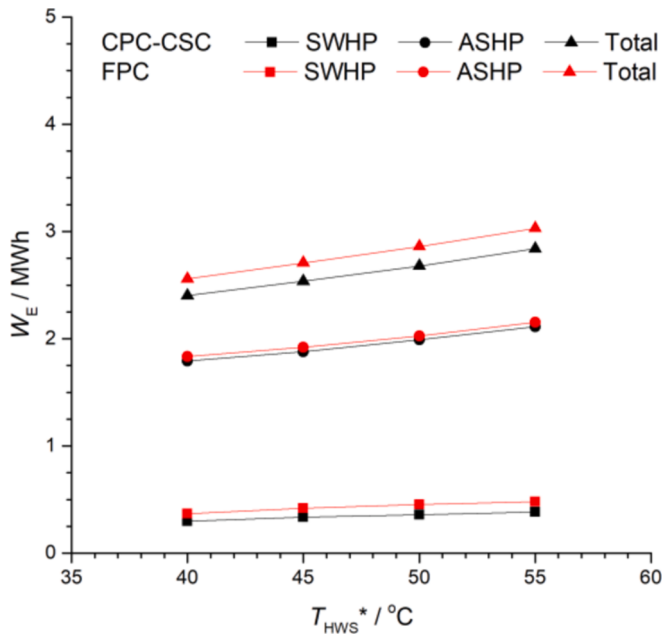


Fig. 6. Variations of electricity consumed by SWHP and ASHP and the total electricity consumed by the DSSAHP against T_{HWS}^* for DSSAHPs using CPC-CSC and FPC.

$$Q_{SC} = IA - Q_{loss,sc} = cm(T_{out} - T_{in}) \quad (1)$$

where I is the solar irradiance for the tilted collector and A is the area of the CPC-CSC, m is the mass flow rate of water, c is the specific heat of water, T_{in} and T_{out} are the temperatures of water at the inlet and outlet of the collector. $Q_{loss,sc}$ is the heat loss from the CPC-CSC. Dickes et al.'s [20] model is used to calculate the heat loss from the solar collector per meter (in length direction), Q_{loss} :

$$Q_{loss} = c_0 + c_1(T_{sc} - T_{amb}) + c_2(T_{sc} - T_{amb})^2 + c_3T_{sc}^3 + I(c_4\sqrt{v_a} + c_5T_{sc}^2) + v_a[c_6 + c_7(T_{sc} - T_{amb})] + \sqrt{v_a}[c_8 + c_9(T_{sc} - T_{amb})] \quad (2)$$

where T_{sc} and T_{amb} are the collector temperature and ambient air

temperature, respectively, and v_a is wind velocity.

According to the operation parameters of the DSSAHP system, the water temperature at the inlet of the CPC-CSC varies from -5°C to 80°C . The operation performances of the CPC-CSC under the working conditions are numerically simulated by ANSYS Fluent, covering the weather conditions in the UK. In the CFD simulations, the water flow rate of the CPC-CSC is $7.23 \text{ kg/h}\cdot\text{m}^2$.

The CFD simulation results are set to be training group and test group (3):1 to obtain the following empirical formula of the linear heat loss:

$$Q_{loss} = 0.1458(T_{sc} - T_{amb}) + 2.384310^{-4}(T_{sc} - T_{amb})^2 - 5.830310^{-6}T_{sc}^3 + I(0.0013\sqrt{v_a} + 8.130210^{-7}T_{sc}^2) + v_a[-0.088 + 5.237710^{-4}(T_{sc} - T_{amb})] + \sqrt{v_a}[0.422 - 0.0104(T_{sc} - T_{amb})] \quad (3)$$

The goodness of fit is 0.9903 and 0.9915 for the training and test groups. This means that the obtained empirical formula has good fitness to the origin data. The CPC-CSC module in TRNSYS is built based on Eqs. (1) and (3).

3. Dual-source solar assisted heat pump

A heating system based on DSSAHP using CPC-CSC is shown in Fig. 2 [21]. The DSSAHP has a solar collection loop, a thermal energy storage (TES) 1 to water-refrigerant-evaporator loop, an air source heat pump (ASHP) loop, a space heating loop, a TES1-TES2 loop, a refrigerant-water-condenser to TES2 loop and a solar water heat pump (SWHP) loop. At sufficient solar irradiation, solar energy is converted into thermal energy and stored in TES tank 1. When the water temperature in TES tank 1 is higher than that in TES tank 2, TES tank 1 provides direct solar hot water for domestic heating. When the water temperature in TES tank 2 is below T_{HWS}^* , if the water temperature in TES tank 1 is between the ambient temperature and that in TES tank 2, SWHP works; if the temperature of water in TES tank 1 is lower than the ambient temperature, ASHP works.

3.1. Working conditions

The system is aimed to serve an SFH 45 building in London for space heating and hot water. The hot water consumption is assumed to be 300 L/day. The reference SFH 45 building is established following the instructions in [22]. The space heating period is determined to be October 29th to April 24th in London to maintain indoor air temperature above 15°C . For convenience, the heating season is set from 1st October to 30th April to compare the operation performance of the heating system. In the heating season, the designed indoor temperature is around 20°C . This corresponds to the peak heating load of 3.53 kW and the average heating load of 1.76 kW in London.

To evaluate the operation performance of the system with different T_{HWS}^* , T_{HWS}^* is set to be 40°C , 45°C , 50°C and 55°C . The highest limit for the water temperature in the TES tanks is set to be 80°C for safety. Hot water temperature at the outlet of the water draw is set to be 40°C to avoid scalding [23]. The water draws are set to be four 15-minute water draws of 300 L/h per day.

3.2. Numerical simulation model in TRNSYS

Operation performances of the DSSAHP system using CPC-CSC and FPC are simulated over a whole year. The numerical simulation starts in summer with a time step of 1 min. The CPC-CSC is modeled based on analyses in section 2 with a collector area of 8 m^2 and the FPC is set at the same collector area. In the TRNSYS model, a SWHP module and an ASHP module are used to represent the SWHP and ASHP modes of the dual-source HP unit with a heating capacity of 8 kW. The parameters for

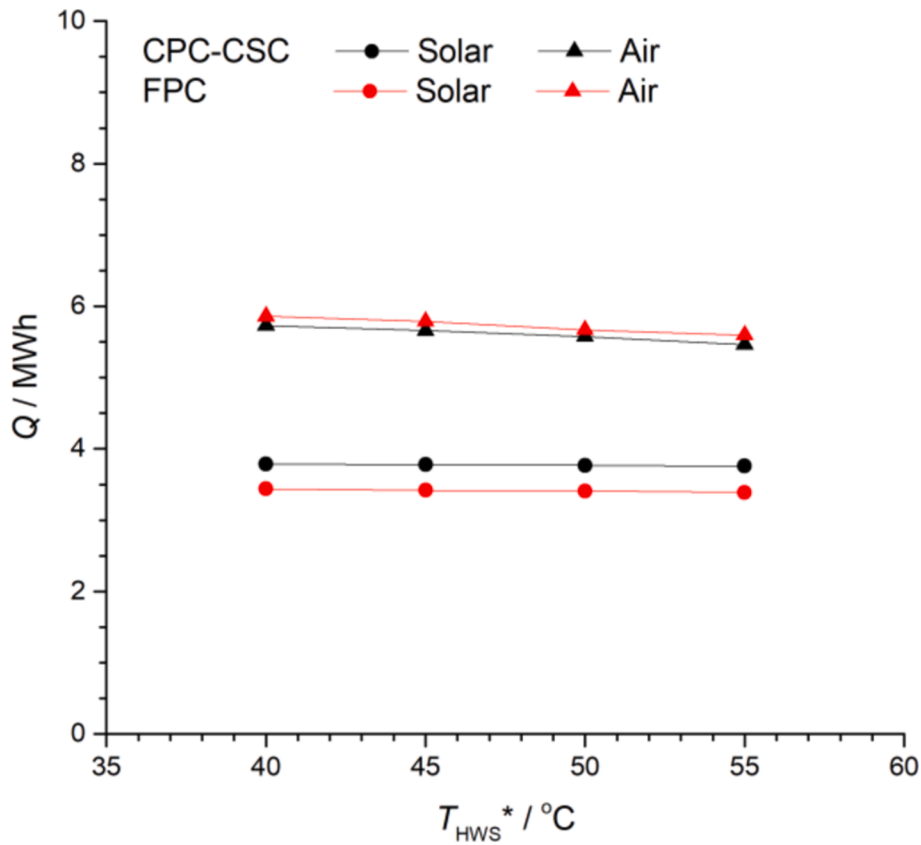


Fig. 7. Variation of thermal energy (Q) extracted from solar energy and ambient air against T_{HWS}^* for DSSAHPs using CPC-CSC and FPC.

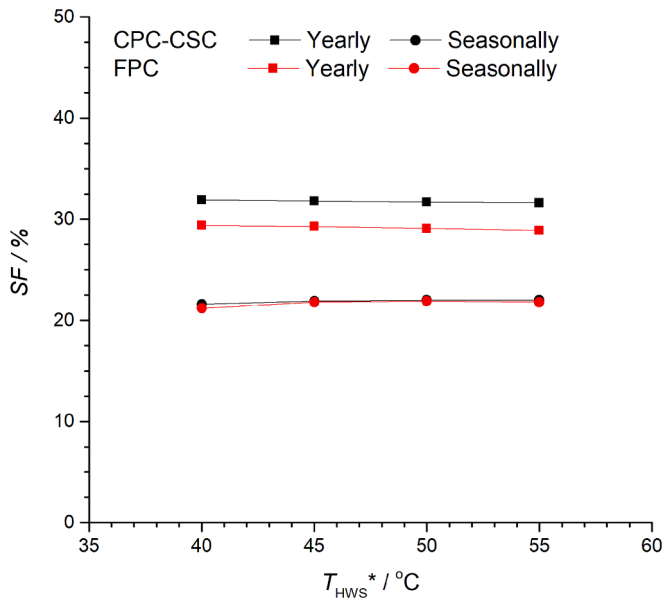


Fig. 8. Variations of year and seasonal SF with T_{HWS}^* for DSSAHPs using CPC-CSC and FPC.

the HP modules are user-defined according to the sample files of real HP units. Parameter selection for the modules and the schematic of the system model are displayed in Table 2 and Fig. 3 [21].

3.3. System performance

Operation performances of the DSSAHP system are mainly evaluated

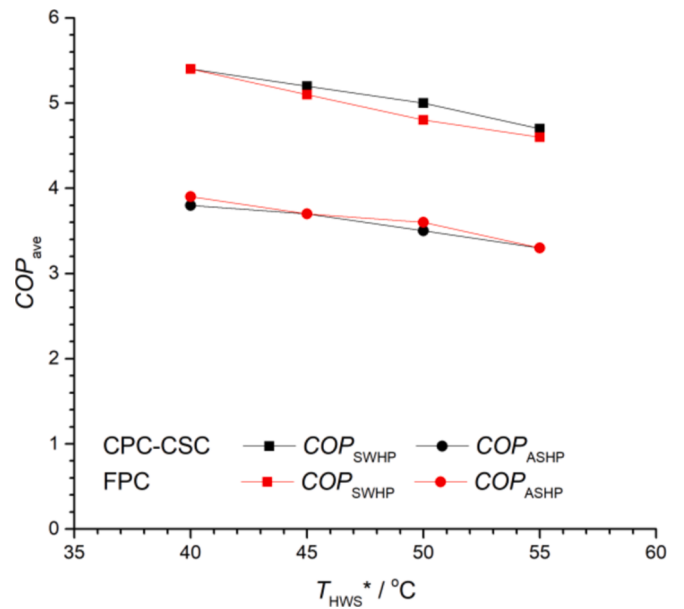


Fig. 9. Averaged COP of SWHP and ASHP with T_{HWS}^* for DSSAHPs using CPC-CSC and FPC.

by COP , solar fraction (SF) and SPF . COP represents the proportion of the heat provision from an HP unit, Q_{HP} , to the corresponding amount of electricity consumed, W_{HP} , given by Eq. (4):

$$COP = Q_{HP} / W_{HP} \quad (4)$$

SF represents the proportion of utilised solar energy to the total heat

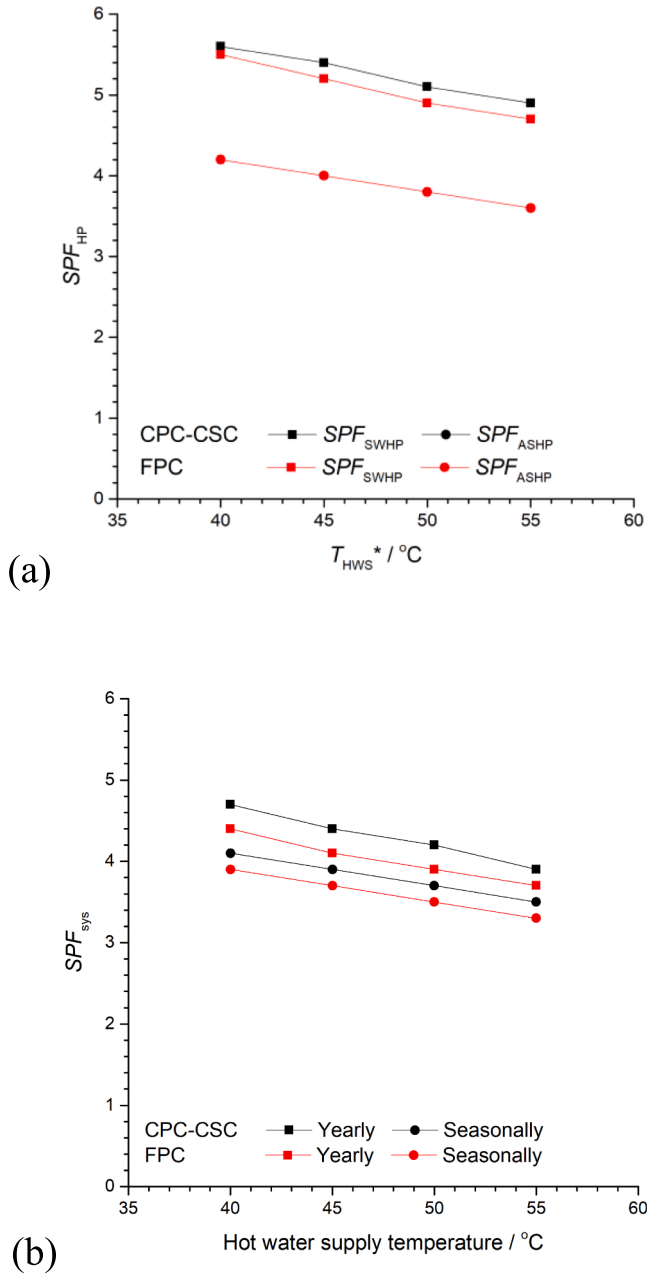


Fig. 10. Variations of year and seasonal SPF_{HP} and SPF_{sys} with T_{HWS}^* for DSSAHPs using CPC-CSC and FPC.

provision of the DSSAHP, given by Eq. (5):

$$SF = \frac{\int Q_{su} dt}{\int (Q_{SH} + Q_{HW}) dt} \quad (5)$$

where Q_{su} is the utilised solar energy, Q_{SH} plus Q_{HW} is the total heat provision of the DSSAHP. SPF represents the proportion of heat provision to the amount of electricity consumed during a given period, given by Eq. (6):

$$SPF = \frac{\int Q dt}{\int W dt} \quad (6)$$

where Q is the heat provision and W is the electricity consumed by the heating system or the HP accordingly.

4. Results and discussion

The heating performances of the DSSAHP using CPC-CSC and FPC with different T_{HWS}^* are investigated in TRNSYS 17. The daily operation performance for low temperature heating of DSSAHP using CPC-CSC is analysed in detail.

4.1. Comparison of low temperature heating operation performance

Fig. 4 shows the variations of indoor air temperature and hot water supply temperature for different designed T_{HWS}^* during the heating season. It can be seen that in heating periods, at all T_{HWS}^* , both the DSSAHP can generally provide sufficient thermal energy and maintain the hot water temperature in TES tank 2 (T_{HWS}) around 5 $^\circ C$ above the T_{HWS}^* . The variations of T_{HWS} have two main reasons: heat provision from the DSSAHP and heat supply to the load. When hot water is drawn for supply, cold mains water and cold water after space heating enters the TES tank 2 to compensate for the water draw. T_{HWS} varies in a larger range for higher T_{HWS}^* due to the larger temperature difference. For higher T_{HWS}^* , since T_{HWS} varies in a larger range and is easier to drop below T_{HWS}^* , more heat provision is required from the DSSAHP. The response to T_{HWS} drop of the DSSAHP using CPC-CSC is faster, especially for lower T_{HWS}^* , showing benefits to the thermal comfort of low temperature heating. In non-heating periods, where direct solar hot water (SHW) works to provide thermal energy, DSSAHP using CPC-CSC can more often have higher T_{HWS} than a system using FPC.

Fig. 5 shows the variations of heat for space heating and hot water by SWHP, ASHP and direct SHW against T_{HWS}^* for heating systems using CPC-CSC and FPC over a year. Generally, for the system using CPC-CSC, the heat provision from ASHP and SWHP is less than that for the system using FPC, by around 2.2 % and 16.7 %; but that from direct SHW is obviously more than that from the system using FPC, by around 32.9 %. As T_{HWS}^* decreases from 55 $^\circ C$ to 40 $^\circ C$, for the system using CPC-CSC, the heat provision by SWHP and ASHP decreases from 1.86 MWh to 1.67 MWh and from 7.57 MWh to 7.52 MWh, respectively, and that for the system using FPC decreases from 2.25 MWh to 2.00 MWh and from 7.75 MWh to 7.69 MWh, respectively. At the same time, heat provision by direct SHW increases from 2.06 MWh to 2.19 MWh for DSSAHP using CPC-CSC and from 1.47 MWh to 1.65 MWh for DSSAHP using FPC. For DSSAHP using CPC-CSC, the decrease in T_{HWS}^* has more influence on the heat provision from SWHP (−10 %) and direct SHW (6.4 %) than on that from ASHP (−0.6 %).

Fig. 6 shows variations of electricity consumed by SWHP and ASHP and the total electricity consumed by the DSSAHP against T_{HWS}^* over a year. Overall, DSSAHP using CPC-CSC can reduce electricity consumption by 6.1 % compared with DSSAHP using FPC. Electricity savings from ASHP, SWHP and pumps are 2.1 %, 19.0 % and 13.5 %, respectively. As T_{HWS}^* decreases from 55 $^\circ C$ to 40 $^\circ C$, for the system using CPC-CSC, electricity consumption by ASHP, SWHP and pumps decreases from 2.11 MWh to 1.79 MWh, from 0.38 MWh to 0.30 MWh and from 0.35 MWh to 0.31 MWh, respectively; those for DSSAHP using FPC decrease from 2.16 MWh to 1.83 MWh, from 0.48 MWh to 0.37 MWh and from 0.40 MWh to 0.36 MWh, respectively. For DSSAHP using CPC-CSC, the decrease in T_{HWS}^* apparently reduces the electricity consumption from ASHP (15.0 %), SWHP (22.2 %) and pumps (10.3 %).

Fig. 7 shows the variation of thermal energy (Q) extracted from solar energy and ambient air against T_{HWS}^* for DSSAHP over a year. Compared with DSSAHP using FPC, DSSAHP using CPC-CSC collects more thermal energy from solar energy (10.2 %) and less thermal energy from ambient air (2.2 %). As T_{HWS}^* decreases from 55 $^\circ C$ to 40 $^\circ C$, for DSSAHP using CPC-CSC, thermal energy collections from solar energy and ambient air increase from 3.76 MWh to 3.79 MWh and from 5.46 MWh to 5.73 MWh; those for DSSAHP using FPC increase from 3.39 MWh to 3.44 MWh and from 5.59 MWh to 5.86 MWh. For DSSAHP using CPC-CSC, the decrease in T_{HWS}^* slightly increases thermal energy obtained from solar energy (0.8 %) and ambient air (4.9 %).

Table 3

Overall performance of the DSSAHPs using CPC-CSC and FPC operating in London.

System	Period		CPC-CSC				FPC			
			40	45	50	55	40	45	50	55
Heat provision (kWh)	HW	Heating-	2235.4	2237.6	2238.2	2237.8	2235.1	2237.8	2238.6	2238.2
		Non-heating-	1427.1	1427.4	1427.5	1427.4	1427.4	1427.8	1428.3	1428.0
	SH		7520.1	7526.7	7532.9	7525.0	7520.3	7519.7	7528.1	7522.9
		Total	11182.6	11191.7	11198.6	11190.2	11182.7	11185.4	11195.0	11189.1
Heat provision(kWh)	SWHP		1672.2	1802.6	1829.8	1859.4	2007.7	2180.5	2234.8	2248.2
		ASHP	7522.7	7538.0	7564.9	7571.7	7692.0	7703.0	7723.4	7749.5
	Solar	Heating-	734.8	675.5	676.0	675.0	424.8	366.9	355.7	360.2
		Non-heating-	1459.8	1417.3	1401.5	1387.8	1226.8	1142.9	1123.8	1110.2
Electricity consumption (kWh)	SWHP		297.7	336.6	357.6	382.6	367.4	418.5	455.6	479.9
		ASHP	1794.1	1879.4	1989.3	2110.4	1832.5	1919.8	2024.5	2155.0
	Water pumps	Heating-	274.0	283.6	295.9	310.1	304.6	314.5	326.4	341.3
		Non-heating-	36.4	36.0	35.9	35.8	54.4	53.5	53.1	53.2
SPF _{HP}	Total		2402.2	2535.6	2678.8	2838.8	2558.9	2706.1	2859.6	3029.5
			5.6	5.4	5.1	4.9	5.5	5.2	4.9	4.7
COP _{ave}	ASHP		4.2	4.0	3.8	3.6	4.2	4.0	3.8	3.6
			5.4	5.2	5.0	4.7	5.4	5.1	4.8	4.6
Solar thermal energy (kWh)	ASHP		3.8	3.7	3.5	3.3	3.9	3.7	3.6	3.3
Solar thermal energy (kWh)	To SWHP		1374.5	1466.0	1472.2	1476.8	1640.2	1762.0	1779.2	1768.3
		To end use	734.8	675.5	676.0	675.0	424.8	366.9	355.7	360.2
	Total	Heating-	1459.8	1417.3	1401.5	1387.8	1226.8	1142.9	1123.8	1110.2
		Non-heating-	3789.0	3777.1	3768.8	3757.8	3439.4	3419.5	3407.8	3387.4
Thermal energy from ambient air (kWh)			5728.6	5658.6	5575.5	5461.3	5859.5	5789.2	5699.0	5594.5
SF	Heating season		21.6 %	21.9 %	22.0 %	22.0 %	21.2 %	21.8 %	21.9 %	21.8 %
		Yearly	31.9 %	31.8 %	31.7 %	31.6 %	29.4 %	29.3 %	29.1 %	28.9 %
SPF _{sys}	Heating season		4.1	3.9	3.7	3.5	3.9	3.7	3.5	3.3
		Yearly	4.7	4.4	4.2	3.9	4.4	4.1	3.9	3.7

Note: Heating-: Heating season; Non-heating-: Non-heating season.

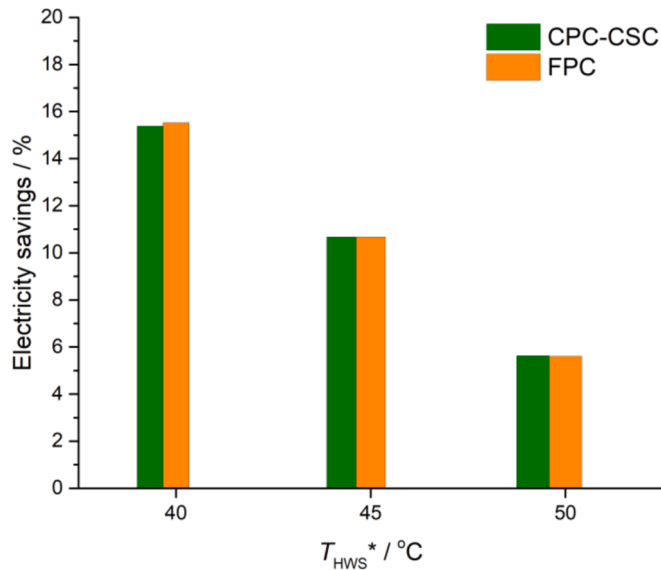
**Fig. 11.** Electricity savings at T_{HWS}^* of 40 °C, 45 °C and 50 °C compared with electricity consumption at T_{HWS}^* of 55 °C for DSSAHP using CPC-CSC and FPC.

Fig. 8 shows the variations of year and seasonal SF with T_{HWS}^* for DSSAHP over a year. The year SF of DSSAHP using CPC-CSC is around 8.5 % higher than that of DSSAHP using FPC while the seasonal SF values for DSSAHP using both CPC-CSC and FPC are generally the same. As T_{HWS}^* decreases from 55 °C to 40 °C, year SF values for systems using CPC-CSC and FPC slightly increase from 31.6 % to 31.9 % and from 28.9 % to 29.4 %. The seasonal SF values for systems using CPC-CSC and FPC slightly decrease from 22.0 % to 21.6 % and from 21.8 % to 21.2 %, respectively.

Fig. 9 shows the averaged COP of SWHP and ASHP with T_{HWS}^* for DSSAHP over a year. The difference between systems using CPC-CSC

and FPC is complex since using different solar collectors affects both the evaporating and condensing temperatures and thus the influence on heat pump operation performance is complex. As T_{HWS}^* decreases from 55 °C to 40 °C, for systems using CPC-CSC, COP_{ASHP} and COP_{SWHP} increase from 3.3 to 3.8 and from 4.7 to 5.4; those for the system using FPC increase from 3.3 to 3.9 and from 4.6 to 5.4. For the system using CPC-CSC, the decrease in T_{HWS}^* significantly increases COP_{ASHP} and COP_{SWHP} by 15.2 % and 14.9 %.

Fig. 10 shows the variations of year and seasonal SPF of ASHP (SPF_{ASHP}), SWHP (SPF_{SWHP}) and system (SPF_{sys}) with T_{HWS}^* over a year. DSSAHP using CPC-CSC shows higher SPF_{SWHP} than DSSAHP using FPC, by around 40 %; SPF_{ASHP} for both systems is almost the same. At the system level, DSSAHP using CPC-CSC has higher both seasonal (5.5 %) and year SPF_{sys} (7.0 %) compared with the system using FPC. As T_{HWS}^* decreases from 55 °C to 40 °C, SPF_{SWHP} values for systems using CPC-CSC and FPC increase from 4.9 to 5.6 and from 4.7 to 5.5; SPF_{ASHP} for both systems increase from 3.6 to 4.2. At the same time, for the system using CPC-CSC, seasonal and year SPF_{sys} increase from 3.5 to 4.1 and from 3.9 to 4.7; those values for the system using FPC increase from 3.3 to 3.9 and from 3.7 to 4.4. For DSSAHP using CPC-CSC, the decrease in T_{HWS}^* significantly increases SPF_{ASHP} and SPF_{SWHP} by 16.7 % and 14.3 %, as well as seasonal and year SPF_{sys} , by 17.1 % and 20.5 %. The details for the operation performance of the heating systems using CPC-CSC and FPC with different T_{HWS}^* are given in Table 3

Fig. 11 shows the yearly electricity savings at T_{HWS}^* of 40 °C, 45 °C and 50 °C compared with electricity consumption at T_{HWS}^* of 55 °C. Systems using both CPC-CSC and FPC have similar electricity saving rate at different T_{HWS}^* under the weather conditions in London. As T_{HWS}^* decreases from 55 °C to 50, 45 and 40 °C, electricity consumption is saved by 5.6 %, 10.7 % and 15.4 %.

4.2. Low temperature heating operation performance on the coldest day

Fig. 12 shows the daily variations of thermal energy charged and discharged in the TES tank 2 for different T_{HWS}^* . For different T_{HWS}^* , the daily variations on thermal energy storage show a similar trend though

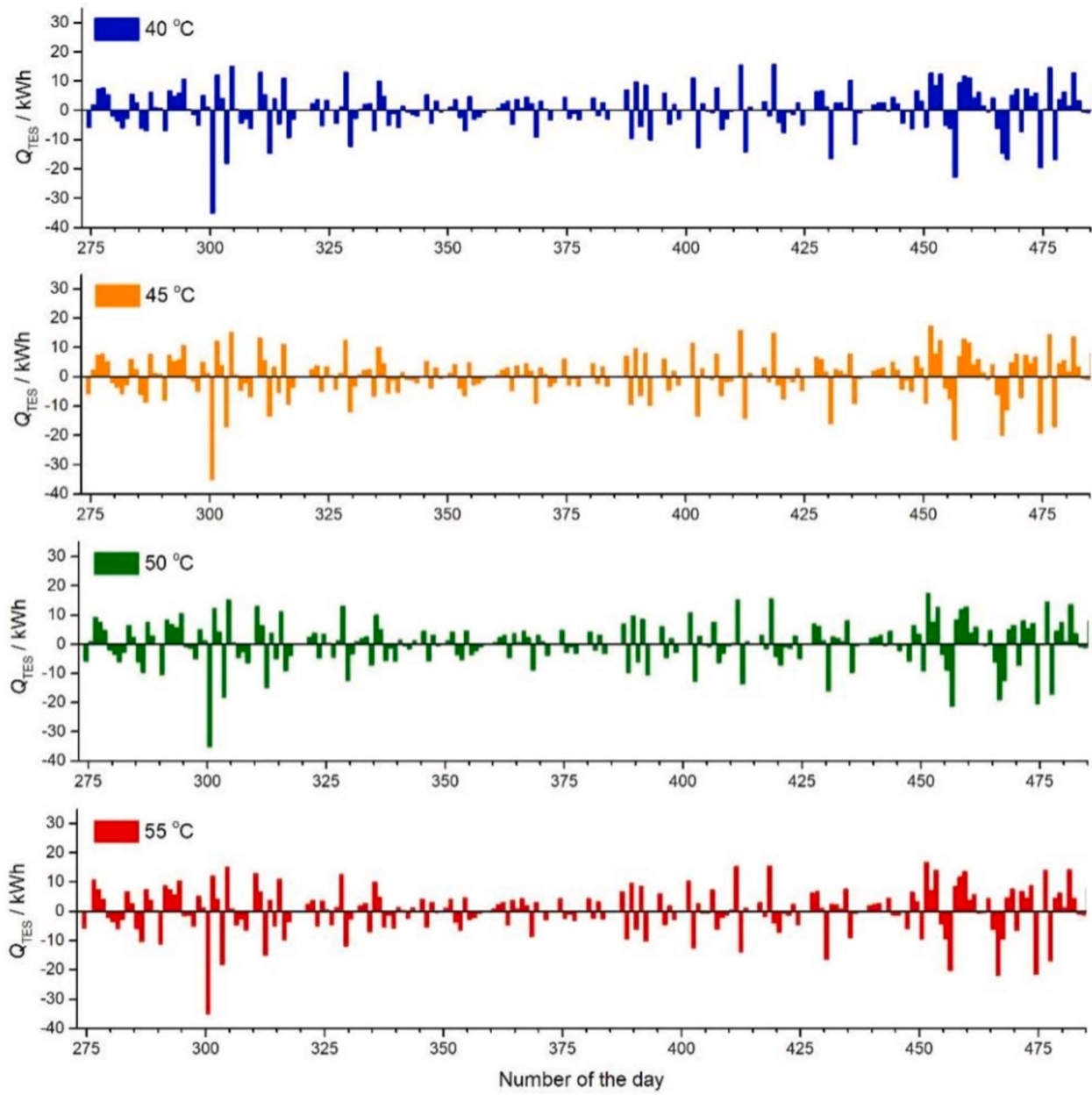


Fig. 12. Daily variations of Q_{TES} charged (positive) and discharged (negative) over a heating season for different T_{HWS}^* .

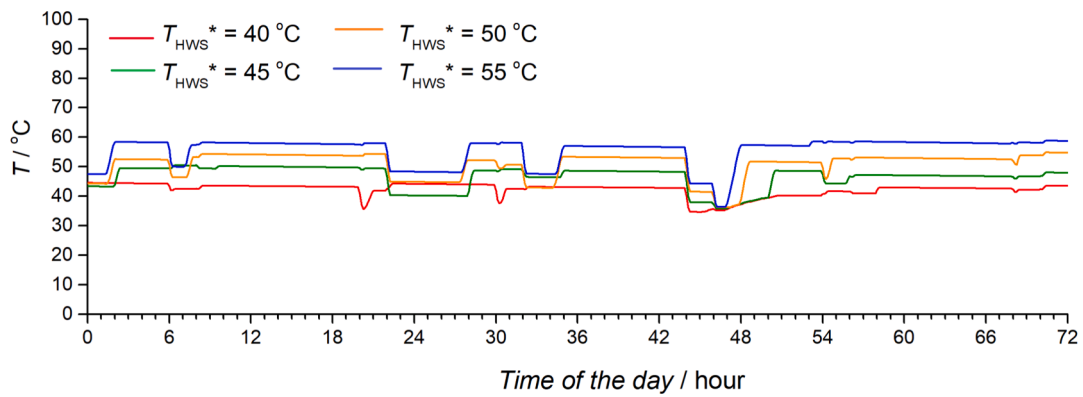


Fig. 13. Variations of hot water temperature at the outlet of TES tank 2 on the 11-13th day for different T_{HWS}^* .

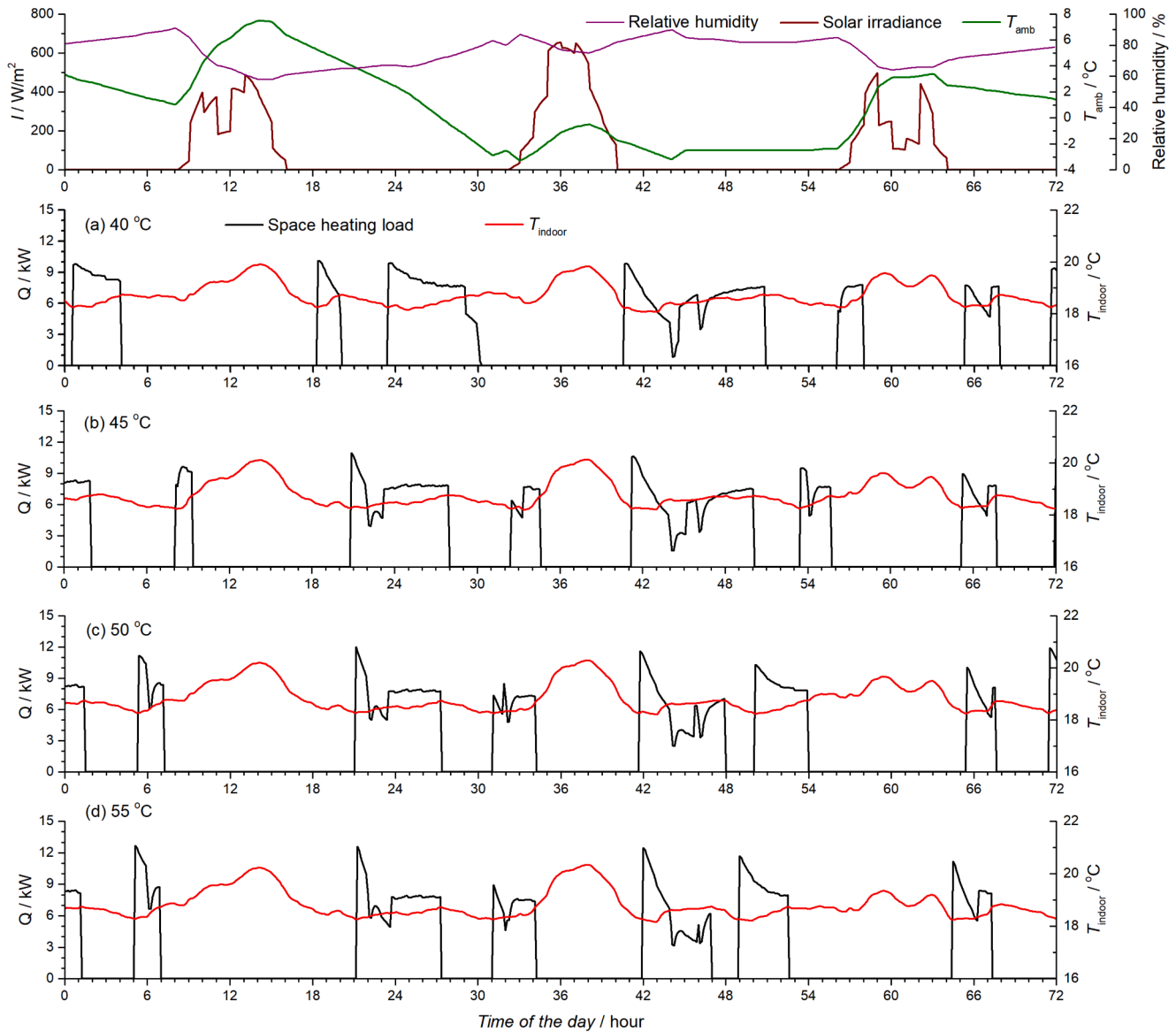


Fig. 14. Variations of space heating load and indoor air temperature on the 11-13th day for different T_{HWS}^* .

T_{HWS}^* affects the storage capacity in TES tank 2. However, for the heating system at the same T_{HWS}^* but different collector area of CPC-CSC, the thermal energy in TES tank 2 (Q_{TES}) charged and discharged varies more apparently [21]. This suggests that the collector, rather than the T_{HWS}^* , has more influence on the TES performance of the DSSAHP. Thus, to modify the operation performance of low temperature heating, optimisation in solar collector can be a useful approach.

Operation performance for low temperature heating of the system using CPC-CSC on the 11-13th day (including the day where the lowest ambient air temperature occurs, the 12th day) is selected for the case study. Fig. 13 shows the variations of hot water temperature at the outlet of TES tank 2 for different T_{HWS}^* . In most time, T_{HWS} are around 3 °C higher than T_{HWS}^* for all T_{HWS}^* . The hot water supply is set at 6.00 am, 8.00 am, 8.00 pm and 10.00 pm. At that time, water in TES tank 2 transfers heat with mains water and its temperature would drop. It is seen that the minimum variation of actual heating temperature (T_{HWS}) occurs at T_{HWS}^* of 40 °C, followed by T_{HWS}^* of 45 °C.

Fig. 14 shows the variations of space heating load and indoor air temperature for different T_{HWS}^* . Due to the different temperature differences between T_{HWS} and indoor temperature, space heating load shows different characteristics for different T_{HWS}^* . As T_{HWS}^* increases,

higher temperature difference brings higher heat transfer efficiency. As T_{HWS}^* increases from 40 to 45, 50 and 55 °C, the peak SH load increases from 10.11 kW to 10.95, 12.00 and 12.67 kW. The heat transfer rate is increased and thus the SH period is reduced. The SH periods are 27.13, 26.38, 25.75 and 24 h for T_{HWS}^* of 40, 45, 50 and 55 °C.

Fig. 15 shows the variations of heating supplied by ASHP and SWHP for different T_{HWS}^* . On the 11-13th day, direct SHW cannot provide thermal energy to the end use. When both ASHP and SWHP do not operate, it refers to the periods that no thermal energy is required by end-use or by TES tank 2 to maintain the T_{HWS}^* . For reference, water temperature in TES tank 1 (T_{TES1}) and TES tank 2 (T_{HWS}) are added. Though higher T_{HWS}^* brings less period of heating demand for end use, it increases the requirement for TES tank 2. Overall, for T_{HWS}^* of 40, 45, 50 and 55 °C, the operation periods of ASHP are 21.88, 21.5, 22.25 and 22 h; those of SWHP are 9.88, 10.13, 10.25 and 9.75 h; the heating periods for TES tank 2 solely are 4.62, 5.00, 6.75 and 7.75 h. On these days, the total heating provision is 1.84, 1.82, 1.87 and 1.84 MWh for T_{HWS}^* of 40, 45, 50 and 55 °C, where 23.60 %, 23.95 %, 23.44 % and 23.44 % is provided by SWHP and 76.40 %, 76.05 %, 76.56 % and 76.56 % is provided by ASHP.

Fig. 16 shows the variations of electricity consumed rate of ASHP,

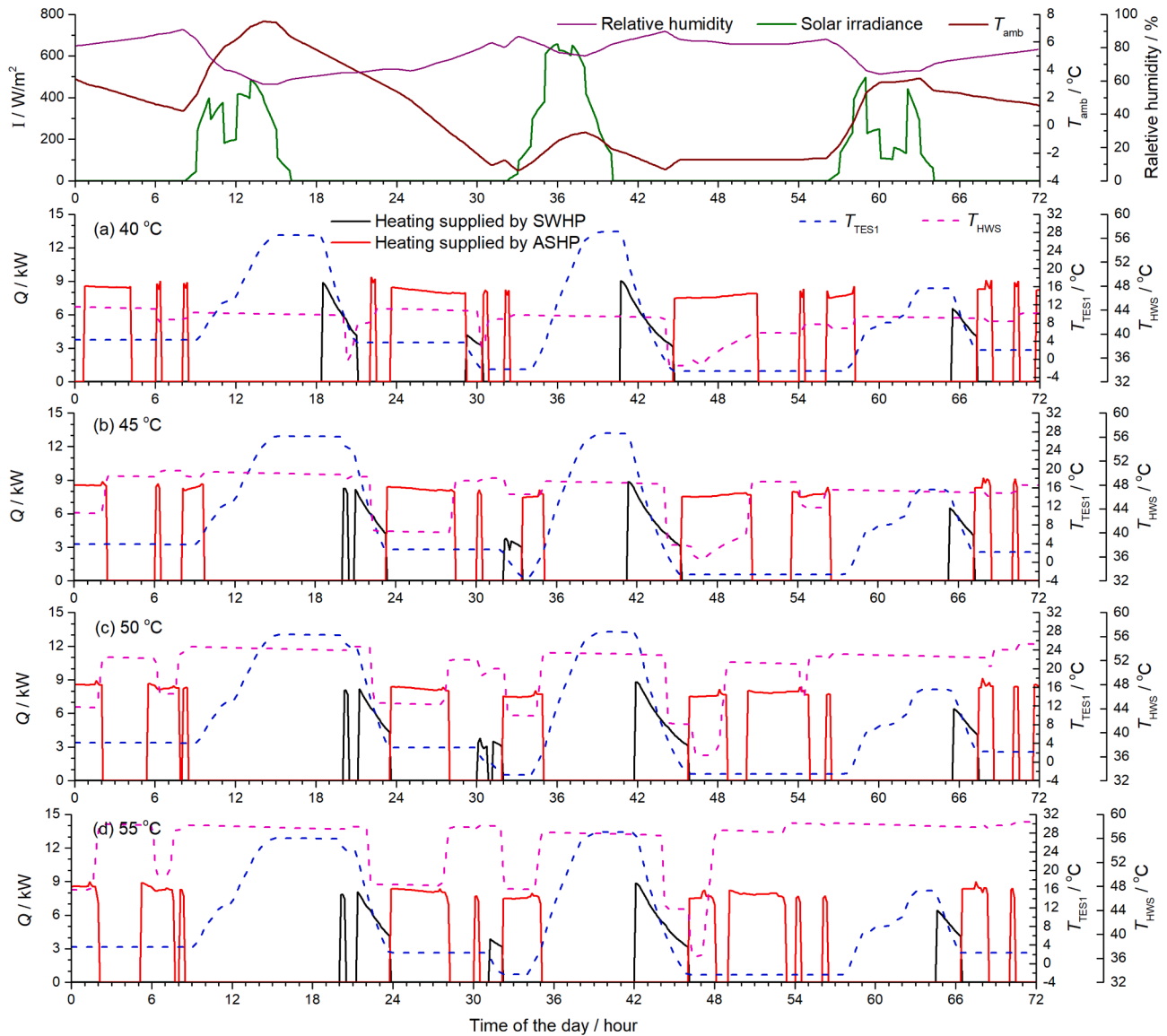


Fig. 15. Variations of heating supplied by ASHP and SWHP on the 11-13th day for different T_{HWS}^* .

SWHP and pumps for different T_{HWS}^* . Though heating power of ASHP is generally the same, for example, from 24.00 to 28.00 h, the electricity consume rate of ASHP increases as condensing temperature increases. This situation does not occur for SWHP in both systems since SWHP mainly works for the primary periods of heating where the condensing temperature is not apparently increased. According to the solar irradiance, SWHP mainly works after solar collection for a day. Then as stored water temperature in the TES tank 1 decreases below ambient temperature, ASHP works. At early morning where ambient temperature drops below store water temperature in TES tank 1, SWHP also works for a while, such as at around 30.00 h. For T_{HWS}^* of 40, 45, 50 and 55 °C, the average electricity consumption rates of ASHP are 2.16, 2.22, 2.31 and 2.42 kW; those of SWHP are 1.12, 1.13, 1.16 and 1.16 kW. Therefore, even though system at lower T_{HWS}^* may require longer operation periods, the total electricity consumption is reduced due to the lower average electricity consumption rate. On these days, the total electricity consumption is 520.19, 528.21, 564.58 and 576.13 kWh for T_{HWS}^* of 40, 45, 50 and 55 °C, where 16.73 %, 17.10 %, 16.66 % and 15.68 % is consumed by SWHP; 72.24 %, 71.94 %, 72.46 % and 73.45 % is consumed by ASHP; 11.02 %, 10.96 %, 10.88 % and 10.87 % is consumed by pumps.

Fig. 17 shows the variations of COP of ASHP (COP_{ASHP}), SWHP (COP_{SWHP}) and system (COP_{sys}) for different T_{HWS}^* . COP_{SWHP} is mainly in the range of 2.0–7.0 and the COP_{ASHP} is mainly in the range of 2.0–4.0. As T_{HWS}^* increases, COP_{SWHP} increases while COP_{ASHP} decreases. The variation trend of COP_{SWHP} is caused by the TES capacity in TES tank 2. Heat provision is urgently needed for DSSAHP at lower T_{HWS}^* , as shown in around 40th – 42nd hour. In the coldest days, solar irradiance is insufficient to compensate the usage of TES in TES tank 1. When the DSSAHP for higher T_{HWS}^* operates, its evaporating temperature is higher and the operation performance can be better. The COP_{ASHP} shows a common trend that, since the ambient temperature is generally the same, higher condensing temperature brings low system efficiency. It is interesting to find that, the COP_{sys} can be in the range of 1.0–9.0, broader than the ranges of COP_{SWHP} and COP_{ASHP} . This is due to the time glide of demand and supply caused by TES in TES tank 2.

4.3. Low temperature heating performance on the day with direct solar hot water heating

Operation performance for low temperature heating of system using CPC-CSC on the 64-66th day is selected for case study where direct SHW

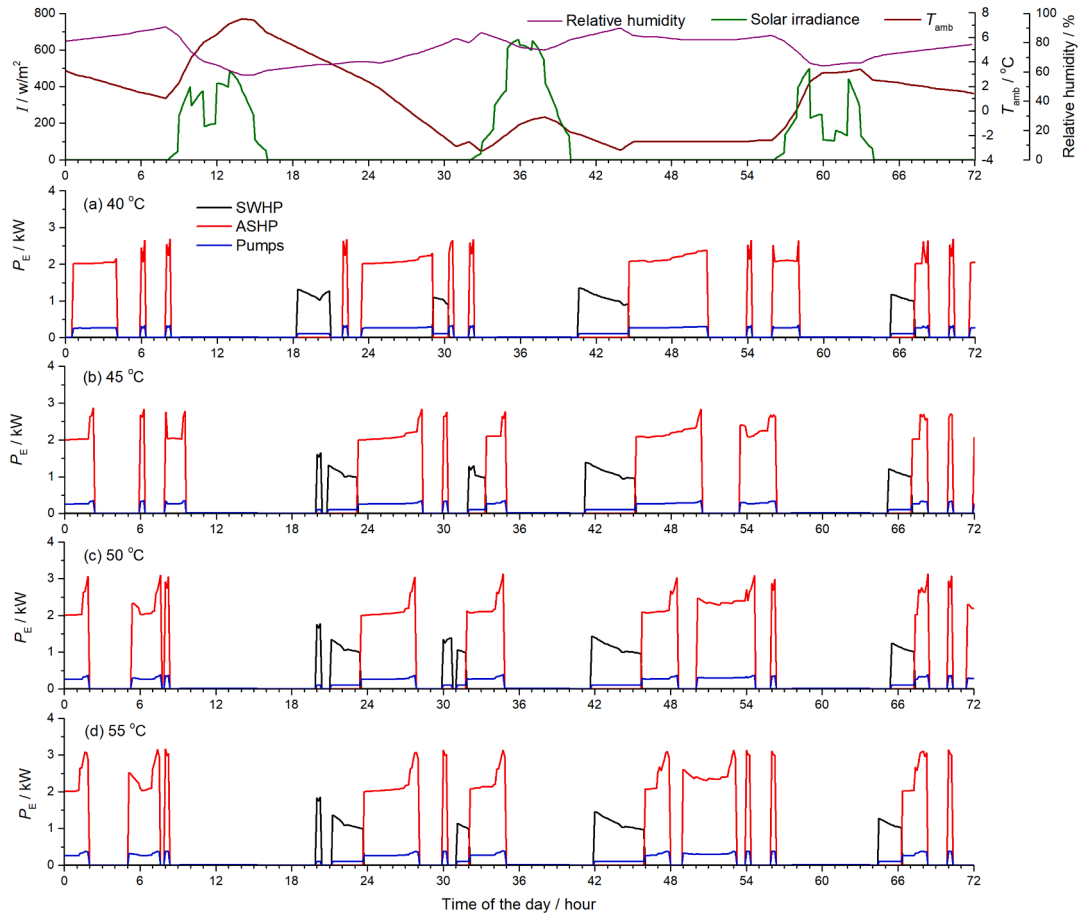


Fig. 16. Variations of electricity consumed rate of ASHP, SWHP and pumps on the 11-13th day for different T_{HWS}^* .

can provide heating. Fig. 18 shows the variations of hot water temperature at the outlet of TES tank 2 for different T_{HWS}^* . In most time, T_{HWS} are around 5 °C higher than T_{HWS}^* for all T_{HWS}^* . System using CPC-CSC has frequent variations in T_{HWS} because it uses more thermal energy stored in TES tank 1 via direct SHW and stored thermal energy left for SWHP is insufficient. Therefore, the system cannot respond quickly to increase the T_{HWS} when SWHP operates.

Fig. 19 shows the variations of space heating load and indoor air temperature for different T_{HWS}^* . The SH periods are 13.00, 12.25, 11.88 and 11.25 h for T_{HWS}^* of 40, 45, 50 and 55 °C. As T_{HWS}^* increases from 40 to 45, 50 and 55 °C, the peak SH load increases from 11.95 kW to 14.50, 15.88 and 17.16 kW.

Fig. 20 shows the variations of heating supplied by ASHP, SWHP and direct SHW for different T_{HWS}^* . For reference, water temperature in TES tank 1 and TES tank 2 are added. Direct SHW can provide thermal energy to the end use among all three days in some cases. However, in some cases, though stored solar energy is sufficient to be used, the T_{HWS} is maintained higher than T_{HWS}^* and the operation of direct SHW is reduced. Thus, heat provision by direct SHW only contributes to a small portion of total heat provision. For T_{HWS}^* of 40, 45, 50 and 55 °C, for system using CPC-CSC, the operation periods of ASHP are 8.00, 7.88, 8.0 and 7.5 h; those of SWHP are 7.75, 8.0, 7.75 and 7.88 h; the heating periods for TES tank 2 solely are 2.75, 3.63, 3.87 and 4.13 h. On these days, for system using CPC-CSC, the total heating provision is 1.13, 1.12, 1.09 and 1.07 MWh for T_{HWS}^* of 40, 45, 50 and 55 °C, where 38.8 %, 40.0 %, 40.1 % and 38.9 % is provided by SWHP; 48.8 %, 48.2 %, 50.5 % and 47.3 % is provided by ASHP; 12.5 %, 11.8 %, 9.4 % and 13.8 % is provided by direct SHW.

Fig. 21 shows the variations in electricity consumed rate of ASHP, SWHP and pumps for different T_{HWS}^* . For T_{HWS}^* of 40, 45, 50 and 55 °C,

the average electricity consumption rates of ASHP are 2.16, 2.22, 2.37 and 2.44 kW; those of SWHP are 1.19, 1.25, 1.26 and 1.29 kW. On these days, for the system using CPC-CSC, the total electricity consumption is 240.60, 248.28, 259.18 and 256.14 kWh for T_{HWS}^* of 40, 45, 50 and 55 °C, where 30.7 %, 32.2 %, 30.2 % and 31.6 % is consumed by SWHP; 57.5 %, 56.4 %, 58.6 % and 57.2 % is consumed by ASHP; 11.8 %, 11.4 %, 11.2 % and 11.2 % is consumed by pumps.

Fig. 22 shows the variations of COP of ASHP, SWHP and system for different T_{HWS}^* . These on the 64-66th days are apparently different from those on 11-13th days. COP_{SWHP} is mainly in the range of 2.0–8.0 and the COP_{ASHP} is mainly in the range of 2.0–4.0. Both COP_{SWHP} and COP_{ASHP} increase as T_{HWS}^* decreases. The COP_{sys} is mainly in the range of 3.0–9.0.

5. Environmental and economic analyses

5.1. Life-span environmental analysis

For the DSSAHP system, total equivalent warming impact (TEWI) is used to describe both direct and indirect impacts on CO₂ emission caused by direct refrigerant loss and the electricity consumption and the corresponding generation, defined as Eq. (7) [24]:

$$TEWI = (GWP \times L \times n) + [GWP \times m \times (1 - a_{\text{recovery}})] + (n \times W_E \times \beta) \quad (7)$$

where GWP is the global warming potential of the refrigerant, 1430 for R134a; L is the refrigerant leakage rate per year (kg), 10 % of total refrigerant charge; n is the life span, 15 years; m is the refrigerant charge, 1.3 kg; a_{recovery} is the recycling factor of refrigerant, 0.7; W_E is the yearly

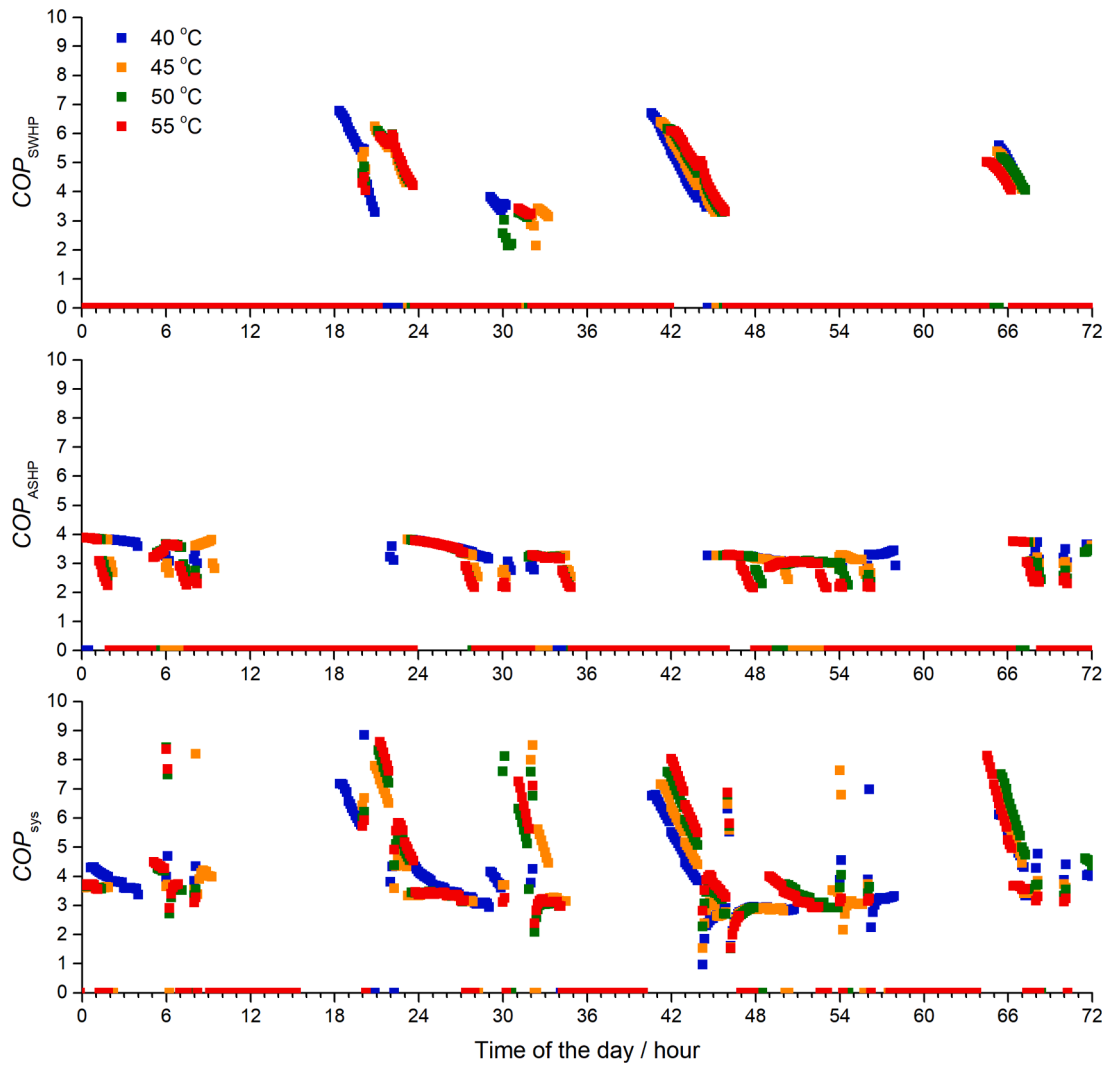


Fig. 17. Variations of COP of ASHP, SWHP and system on the 11-13th day for different T_{HWS}^* .

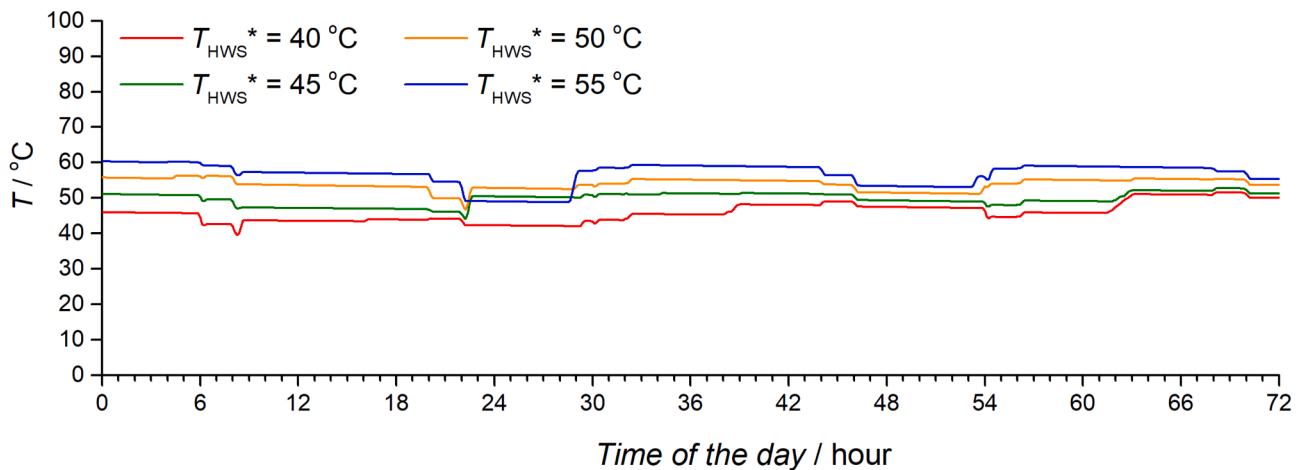


Fig. 18. Variations of hot water temperature at the outlet of TES tank 2 on the 64-66th day for different T_{HWS}^* .

electricity consumption; and β is the CO₂ emission ratio for electricity generation. In the UK, according to the Intergovernmental Panel on Climate Change, the CO₂ emission ratio for the electricity generation sector is calculated to be 0.254 kg/kWh [25].

For comparison, CO₂ emissions of heating systems using fossil fuel

boilers are calculated with the same life span. According to [26], the ratios of CO₂ emission to heat provision of coal, oil, liquefied petroleum gas and natural gas are 0.517, 0.335, 0.262 and 0.267 kg/kWh. According to the yearly electricity consumption and heat provision in Table 3, the life-span CO₂ emission of DSSAHP and the equivalent fossil

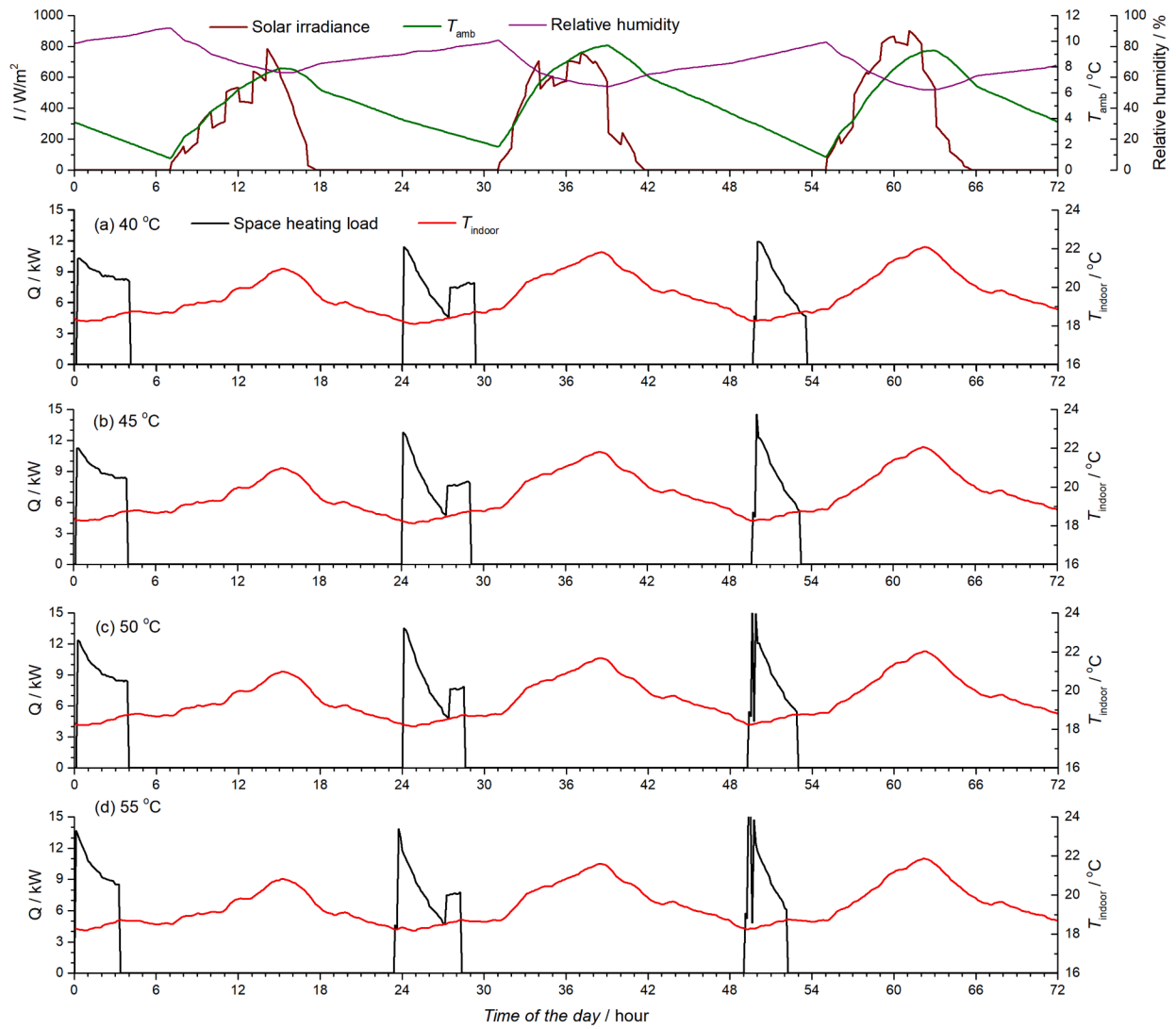


Fig. 19. Variations of space heating load and indoor air temperature on the 64-66th day for different T_{HWS}^* .

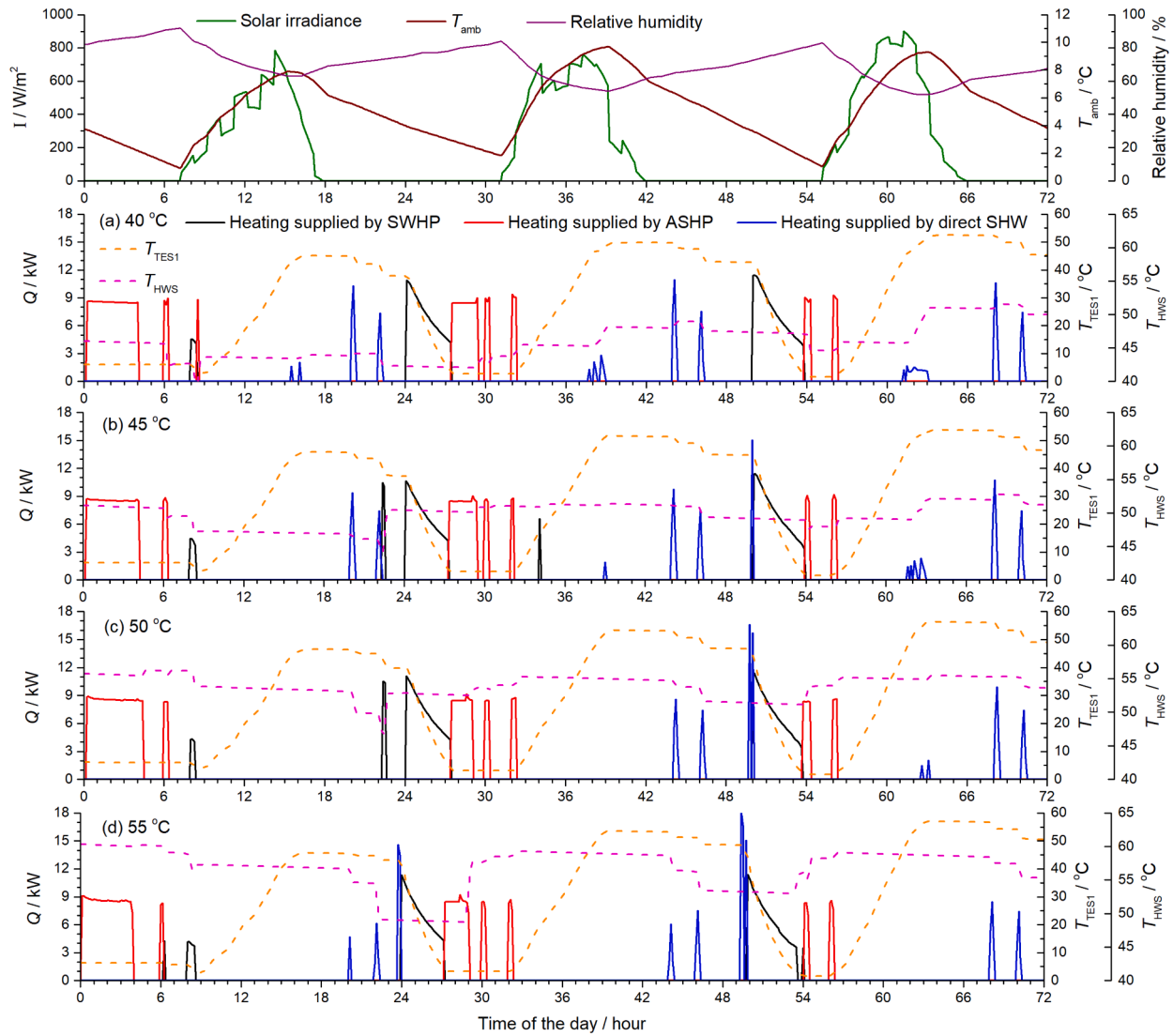


Fig. 20. Variations of heating supplied by ASHP, SWHP and direct SHW on the 64-66th day for different T_{HWS}^* .

fuel heating systems are listed in Table 4.

It can be seen that, using DSSAHP can significantly reduce CO₂ emission compare with using fossil fuel heating system. Generally, the TEWI of DSSAHP using R134a is only around 14.4 %, 22.2 %, 28.4 % and 27.9 % of CO₂ emissions of heating system using coal, oil, liquefied petroleum gas and natural gas. With the development of eco-friendly refrigerants, further reduction in CO₂ emission can be achieved from the direct impact of DSSAHP.

Using ultra-low temperature heating can decrease TEWI by 11.7 % and 12.0 % for DSSAHP using CPC-CSC and FPC. Compared with DSSAHP using FPC, DSSAHP using CPC-CSC has a reduction of around 4.6 % in TEWI. This illustrates a green potential for HP optimisation via using well-designed components and operation modes.

5.2. Economic analysis

Economic analyses of the DSSAHP heating system using CPC-CSC and FPC for low temperature heating are performed according to the electricity price in the UK. An electric heater is used as the benchmark for comparison. W_{tot} is the total electricity consumption of the heating systems, given by Eq. (8):

$$W_{tot} = (Q_{SH} + Q_{HW})/\eta \quad (8)$$

where η is the efficiency of electric heater.

P_{pb} is the payback period of the heating systems against the electric heater, calculated by Eq. (9):

$$P_{pb} = C_i/C_{spy} \quad (9)$$

where C_i is the difference of the initial cost and C_{spy} is the cost saving per year, obtained by Eqs. (10) and (11).

$$C_i = C_{i0} - C_{ieh} \quad (10)$$

$$C_{spy} = C_{o0} - C_{oeh} \quad (11)$$

where C_{i0} and C_{ieh} are the initial costs of the heating system and the electric heater, C_{o0} and C_{oeh} are the operation costs of the heating system and the electric heater.

The efficiency of the electric heater is 0.95 [27]. The heat provision of the electric heater is set to be the average heat provision of the DSSAHP heating system, 11.19 MWh. The electricity price is obtained from statistics to be £72.34 per MWh (price in May 2024) [28]. Currently, the price of CPC collector is around twice of that of ETC [18]. Since CPC-CSC is a new collector in research and development status, price of ETC is adopted to assume CPC-CSC price after commercialisation. The prices of electric heater of 8 kW (£3000) [29], TES tank of 300

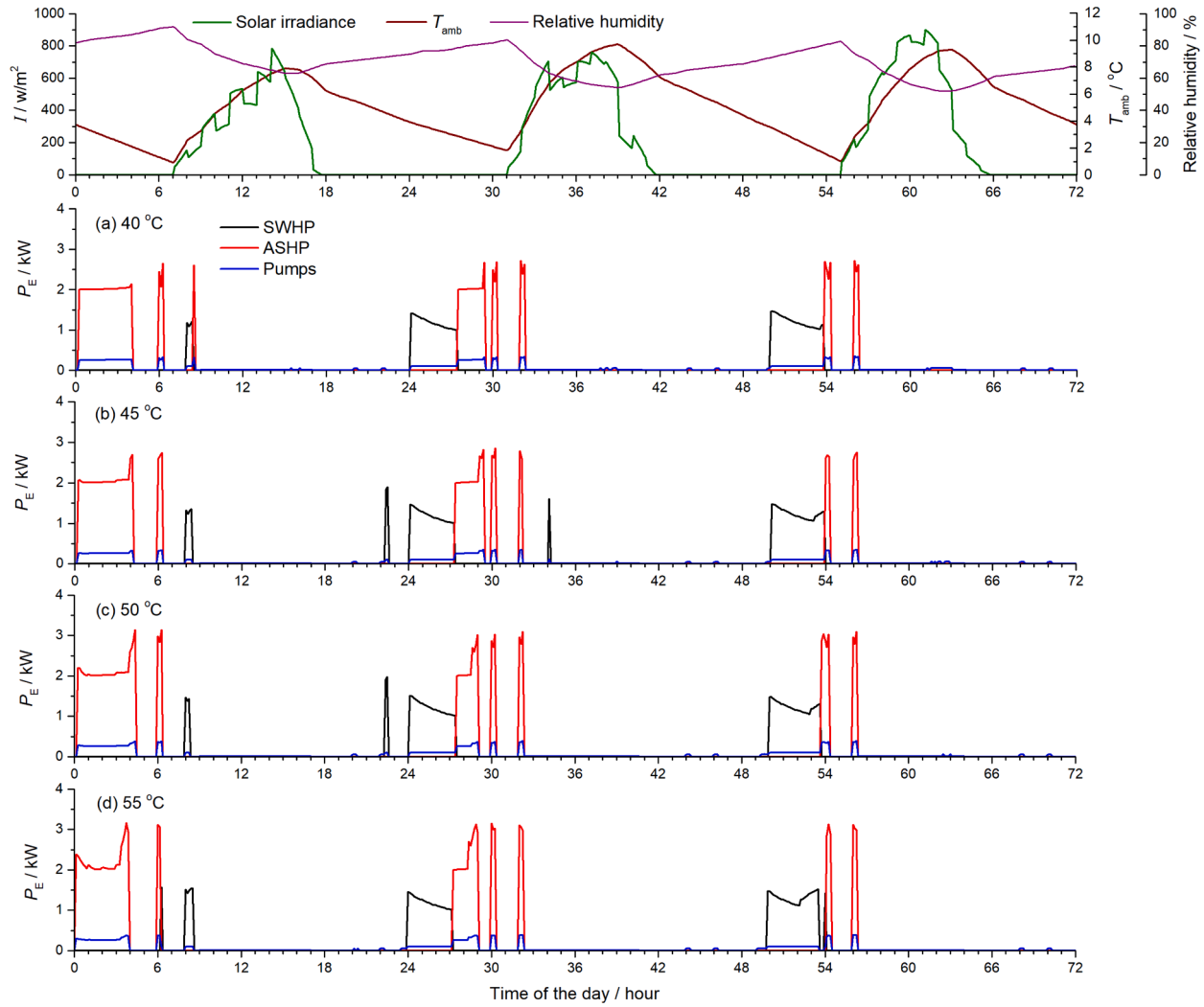


Fig. 21. Variations of electricity consumed rate of ASHP, SWHP and pumps on the 64-66th day for different T_{HWS}^* .

L (£815) and 500 L (£1650) [30], water pump with a head of 10 m (£360) [31], flat plate collector [32] and evacuated tube collector [33] are obtained from UK domestic and European sellers. The price of heat pump is assumed based on UK government report [34]. Installation for DSSAHP is assumed to be 6 h. The engineer fee is taken to be £80 per hour [35]. The economic analysis for DSSAHP heating system integrating CPC-CSC and FPC for low temperature heating is displayed in Table 5.

According to the economic analysis, it can be seen that the initial cost of the DSSAHP system using CPC-CSC is slightly higher than that of the system using FPC, by around 12.9 %. The operation cost per year of the DSSAHP system using CPC-CSC is reduced by 6.8 % compared with that of the system using FPC. Under the current electricity price, DSSAHP systems using CPC-CSC and FPC share a similar payback period of 11 ~ 12 years for different set heating temperatures. However, in 2022, electricity price arrived around £400 per MWh dramatically. In that case, the payback period can be reduced to around 2 years for both systems. The DSSAHP system using CPC-CSC shows comparable economic performance to that using FPC, with a mere extend in a payback period of around 1.9 %. Considering thermal comfort and environmental benefits, CPC-CSC can be acceptable for wide application to improve low temperature heating performance.

6. Conclusions

This paper investigates the ultra-low temperature heating performances of a DSSAHP heating system using CPC-CSC. The advantages of the system providing heating for the SFH 45 building in London have been analysed. The conclusions can be drawn below:

1. Using CPC-CSC benefits the response speed of DSSAHP as low T_{HWS}^* , improving the thermal comfort performance of ultra-low temperature heating. The collector, rather than the T_{HWS}^* , has more influence on the TES performance of the DSSAHP. To further develop ultra-low temperature heating, innovation in solar collectors plays an important role.
2. Low temperature heating increases the application of renewable energies. For DSSAHP using CPC-CSC, ultra-low temperature heating increases thermal energy obtained from solar energy (0.8 %) and ambient air (4.9 %).
3. Low temperature heating contributes to passive heating. For DSSAHP using CPC-CSC, low temperature heating has more influence on the heat provision from SWHP (−10 %) and direct SHW (6.4 %) than on that from ASHP (−0.6 %).
4. Compared with DSSAHP using FPC, the heating system using CPC-CSC can reduce TEWI by 4.6 % with a slight extension in payback period by 1.9 %.

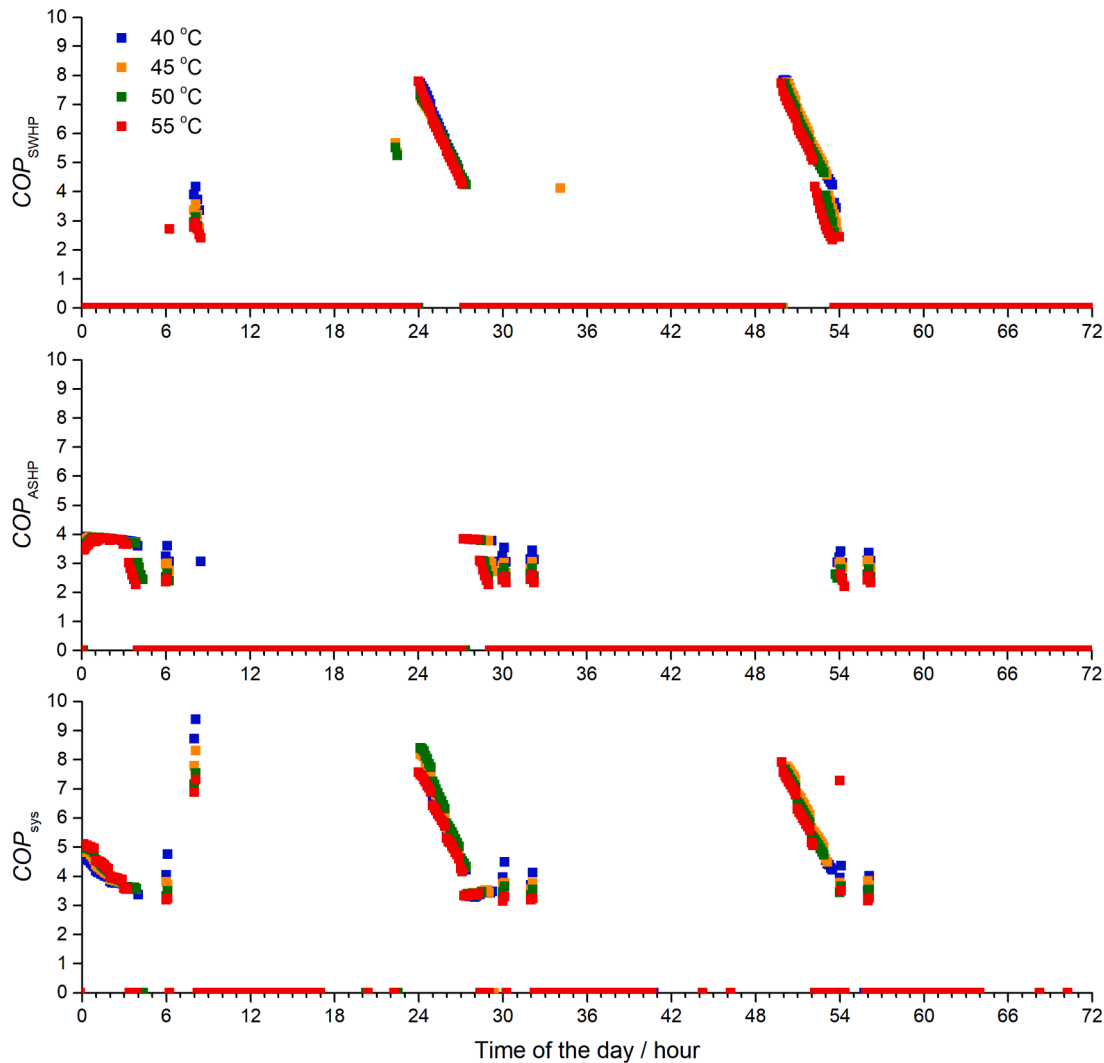


Fig. 22. Variations of COP of ASHP, SWHP and system on the 64–66th day for different T_{HWS}^* .

Table 4
Life-span CO₂ emission of DSSAHP and equivalent fossil fuel heat systems (ton).

		CPC-CSC				FPC			
		40	45	50	55	40	45	50	55
Coal		86.72	86.79	86.85	86.78	86.72	86.74	86.82	86.77
Oil		56.19	56.24	56.27	56.23	56.19	56.20	56.25	56.23
Liquefied petroleum gas		43.95	43.98	44.01	43.98	43.95	43.96	44.00	43.97
Natural gas		44.79	44.82	44.85	44.81	44.78	44.80	44.84	44.81
DSSAHP	Direct impact	3.35							
	Indirect impact	9.15	9.66	10.21	10.82	9.75	1.31	10.90	11.54
	TEWI	12.50	13.01	13.55	14.16	13.10	13.66	14.24	14.89

5. For DSSAHP using CPC-CSC, as T_{HWS}^* decreases from 55 to 40 °C, SPF_{ASHP} and SPF_{SWHP} significantly increase by 16.7 % and 14.3 %, and seasonal and yearly SPF_{sys} increase by even 17.1 % and 20.5 %. This results from both passive heating and HP heating sectors. On the one hand, low temperature heating allows more heat provision from direct SHW and thus apparently increases system efficiency. On the other hand, generally, the low heating temperature benefits for low condensing temperature and thus high HP efficiency.

CRediT authorship contribution statement

Li Wei Yang: Writing – original draft, Visualization, Methodology,

Investigation, Formal analysis, Data curation, Conceptualization. **Jin Huan Pu**: Writing – review & editing, Supervision, Project administration, Methodology. **Rong Ji Xu**: Writing – review & editing, Validation, Resources, Investigation. **Tong Yang**: Writing – review & editing, Resources, Methodology. **Hua Sheng Wang**: Writing – review & editing, Supervision, Software, Resources, Project administration, Funding acquisition.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence

Table 5
Economic analysis for DSSAHP using CPC-CSC and FPC based on electricity prices in May 2024.

	Electric water heater	DSSAHP using CPC-CSC				DSSAHP using FPC			
Set heating temperature	—	40	45	50	55	40	45	50	55
Heat provision per year. MWh	11.19	11.18	11.19	11.2	11.19	11.18	11.19	11.2	11.19
Efficiency / SPF	0.95	4.7	4.4	4.2	3.9	4.4	4.1	3.9	3.7
Electricity consumption, MWh/year	11.78	2.38	2.54	2.67	2.87	2.54	2.73	2.87	3.02
Initial cost, GBP	collector	0	2160	2160	2160	2160	1880	1880	1880
	tanks	815	2465	2465	2465	2465	2465	2465	2465
	heater/HPs	3000	6000	6000	6000	6000	6000	6000	6000
	pumps	0	360	360	360	360	360	360	360
	installation	0	480	480	480	480	480	480	480
	total	3815	11,465	11,465	11,465	11,185	11,185	11,185	11,185
Operation cost, GBP	852.09	172.08	183.97	192.91	207.56	183.81	197.44	207.75	218.78
Cost saving, GBP/year	—	680.01	668.12	659.18	644.53	668.28	654.65	644.34	633.31
Payback period, year	—	11.25	11.45	11.61	11.87	11.03	11.26	11.44	11.64

the work reported in this paper.

Acknowledgment

The authors gratefully acknowledge the financial support from the Joint PhD Studentship of China Scholarship Council (CSC) and Queen Mary University of London (201808060459).

Data availability

The authors do not have permission to share data.

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