Experimental and theoretical study on a solar assisted CO₂ heat pump for space heating

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ABSTRACT

In this paper, a solar combi-system which consists of solar collector and a CO₂ heat pump is investigated experimentally and theoretically. Two experiments are primarily conducted to show the performance of this solar combi-system under different operation conditions. A system model is developed and validated in TRNSYS to analyze the influence of main components parameters. Subsequently, a multi-parameter optimization is carried out in GENOPT to obtain a final optimal result. The simulated results show that the optimized system can save 14.2% electricity and improve the solar fraction by 8%. The solar fraction of the optimized system can reach 71.1%. Finally, the optimized system performance is studied with the weather and load characteristics in Shanghai. Compared with the CO₂ HP heating system alone, the solar assisted system can save 1790.8kWh electricity on the basis of year round operation.

Keywords: solar heating; CO₂ heat pump; solar energy; optimization

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Nomenclature**Greek symbols**

A	area of solar collector [m^2]	τ	time
COP	coefficient of performance[-]	Subscripts	
c	specific heat of water [$\text{kJ}/(\text{kg}\cdot^\circ\text{C})$]	HP	heat pump
η	solar collector efficiency[-]	FCU	fan coil unit
G	total incident radiation [W/m^2]	i	inlet
P	power of heat pump [kW]	o	outlet
Q	heat gain [kJ]	OT	operation tank
ρ	density of water [kg/m^3]	ST	storage tank
SF	solar fraction[-]	SP	supply pump
T	temperature [$^\circ\text{C}$]	DP	delivery pump
V	volume [m^3]	SCP	SC pump
W	electricity consumption [kJ]	S	whole system
E	annual electricity consumption [kWh]	SC	solar collector
F	flow rate [m^3/s]		

1. Introduction

Climate change has become a worldwide concern in recent years. In heating, ventilation, air conditioning and refrigeration (HVAC&R) industry, carbon dioxide (CO_2) shows an environmentally friendly advantage over the widely-used hydro chlorofluorocarbons (HCFCs) refrigerants. Due to the characteristic of transcritical CO_2 cycle, CO_2 heat pumps (HPs) can meet the demand of heating systems very well, especially in hot water heater applications. Neksa et al[1] presented an experimental study of a 50 kW CO_2 HP water heater in 1998. The coefficient of performance (COP) reached 3.6 with the hot water supply temperature of 80°C . Furthermore, Fernandez et al[2] conducted a performance evaluation on the overall COP during full tank heating tests in three scenarios typical of residential water heating and results showed that the overall COP was 30% higher for heating a full tank of cold water than reheating a warm tank water after standby losses. Yokoyama et al[3] investigated the performance of a CO_2 water heating system based on numerical simulation. The daily change in a standardized hot water demand, and some features of the temperature distribution in the storage tank and the system performance criteria are investigated. Petersen et al[4] presented

43 a high-efficiency R744 heat pump water heater of approximately 35kW heating capacity
44 for commercial applications with effective utilization of the cooling capability for air conditioning
45 and/or refrigeration, targeted at commercial use in restaurants, hotels, nursing homes, and hospitals,
46 where some cooling load is typically needed year round. Recent studies have proven that HP water
47 heater is one of the most promising applications of the transcritical CO₂ cycle. The cumulative number
48 of the installed units of the commercial case in Japan, well-known as “Eco-Cute”, surpassed 3 million
49 by 2011[5].

50 Besides the HP water heater, another application of transcritical CO₂ cycle is a combined heat
51 supply system for space heating (SH) and domestic hot water (DHW). Stene[6] presented an
52 experiment of a 6.5 kW CO₂ HP integrated in a combined heat supply system under different operation
53 conditions. The HP unit gave off heat to a floor heating system at supply/return temperatures of 33/28,
54 35/30 or 40/35 °C, and the set-point temperature for the DHW was 60, 70 or 80 °C. The results showed
55 that at DHW heating demand ratios above approximately 50%, the CO₂ HP system outperformed the
56 HFC heat pump system. Based on the analyses of results, the author recommended the use of such
57 integrated CO₂ heat pump in low-energy and passive houses[7]. Due to the low space heating demand
58 in such buildings, the heat demand for DHW typically constitutes 50-85% of annual heating demand.
59 These kinds of load characteristics are suitable for CO₂ HP.

60 In addition to the HP water heaters and combined heat supply systems, some studies paid attention
61 on solar-assisted CO₂ HPs to further reduce the electricity consumption. Chaichana et al[8] presented a
62 comparative study of natural working fluids with R-22 for solar-boosted heat pumps. The results
63 showed that CO₂ may be unsuitable for solar-boosted heat pumps because of its low critical
64 temperature and high operational pressures. Kim et al[9] conducted a simulation study of a

solar-assisted CO₂ HP system, in which a CO₂ HP is used as an auxiliary thermal source. The impact of various operation conditions, especially expansion valve (EEV) opening, is analyzed. Kim et al[10] did a simulation study of hybrid solar-geothermal heat pump system for residential applications. The influence of different operation conditions on the system parameters is conducted. Deng et al[11] proposed a study on optimization of a solar combi-system with CO₂ HP via TRNSYS[12] simulation model. The system was optimized and analyzed by using both single and multi-parameter optimization. However, only a few studies, particularly experiment studies, have been conducted to analyze the characteristics of a solar combi-system with a CO₂ HP.

The main purposes of this study are: (1) to conduct an experiment study on the performance of a pre-existing solar combi-system with a CO₂ HP; (2) to do an optimization analysis based on the pre-existing system. The influences of main components in the system are studied and the performance of the final optimized system is analyzed.

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2. System description

Fig. 1 shows a schematic diagram of the proposed solar combi-system in the residential building and Fig. 2 shows the main components. The 40 m² solar collector (SC) array (vacuum tube solar collector with micro CPC reflector at the back) is installed on the roof with an installation angle of 30°. The captured solar energy is delivered to the 500L storage tank by the SC pump. The delivery pump is used to bring the hot water in the storage tank into the 300L operation tank. The DHW (set point: 40 °C) is heated by an immersed heat exchanger in the operation tank with a mixing valve to adjust the temperature. A 6 kW CO₂ HP is employed as an auxiliary heater, which is connected with the operation tank. The CO₂ HP consists of a fin-tube evaporator with fan, a smooth tube in tube gas cooler, a fixed speed swing compressor and an electronic expansion valve. The discharge pressure of

the compressor is controlled by adjusting the effective flow area of the expansion valve. Two switching valves are installed to switch the heat source between storage tank and CO₂ HP. 5 fan coil units (FCUs) act as the indoor terminal units for SH, with the hot water delivered by supply pump from operation tank. The main parameters of the system components are listed in Table 1. This system separates the operation tank from the storage tank. The storage tank acts as the main storage of the captured solar energy and delivers energy to the operation tank only when the temperature level is achieved. In this way, the storage volume is separated and the operation tank volume can be much smaller. The CO₂ HP will consume less electricity to maintain the temperature in the small volume operation tank when the solar radiation is poor. The operation tank is also a buffer of heat for the whole system since the temperature of the solar system changes obviously with the solar radiation. It can weaken the temperature fluctuation of the heat source.

The temperatures of the system are measured by a group of PT100 sensors (A class, error: 0.1 °C). The water flow rate is measured by an integrating flow meter with an accuracy of $\pm 5\%$ (of full scale 10 m³/h) in the range 0 – 10 m³/h. The ambient condition is monitored by a micro weather station (Davis 6152c) and Kipp & Zone CM22 pyranometer , and the accuracies of the radiation and temperature are $\pm 5\%$ (of reading) and ± 0.5 °C, respectively. The electricity consumption and transient power are measured by one single-phase meter (QINGZHI ZW3415) and one three-phase meter (QINGZHI ZW3432). The accuracies of measurements are $\pm 1\%$ (of reading) . All the data is captured and handled by a Graphtec GL800 acquisition system.

The control strategy is as follows: (1) CO₂ heat pump heating mode: as long as the water temperature at the middle position of operation tank is lower than 45 °C, the CO₂ HP starts to operate until the temperature reaches 50 °C; (2) Solar heating mode: the switching valves switch to storage tank

110 and the delivery pump operates when the temperature at the top of storage tank is higher than the
 111 temperature at the bottom of the operation tank and the middle position of operation tank is higher than
 112 45 °C ; (3)Room temperature for the heat supply is set at 18 °C for starting and 22 °C for stopping; (4)
 113 The SC loop is controlled by an on/off differential controller with the set value of 8/4 °C.

114 Table 1 Parameters of main components

Parameters	Initial values
SC area	40m ²
Inclination angle of SC array	30°
CO ₂ HP heating capacity	6kW
CO ₂ HP power	2.3kW
Storage tank volume	500L
Operation tank volume	300L

115 Test condition of CO₂ HP capacity and power: dry/wet bubble temperature: 7/6 °C, inlet/outlet water
 116 temperature: 45/50°C

117

1. Solar Collector(SC) Array 2. SC pump 3. Storage Tank 4. SC Delivery Pump 5. CO2 HP 6. HP Delivery Pump
7,8. Switching Valve 9. Operation Tank 10. Civil Water Inlet 11. Mixing Valve 12. Supply Pump 13. Fan Coil Units
14. Testing Room

Fig. 1 Diagram of proposed solar combi-system with CO₂ HP

Fig. 2 main components of the solar assisted CO₂ heat pump

Fig. 3 and Fig. 4 are the floor plan and the inside view of the test apartment. The total area of the building is 90 m², consisting of a 19.2 m² bedroom A, a 16m² bedroom B, a 46.8 m² living room with open kitchen, a 5 m² bathroom and a 3 m² storage room. The north, south and east building envelopes are exterior walls and the west building envelop is adjacent to another apartment. The heat transfer coefficients of the building envelop are listed in Table 2.

Fig. 3 Floor plan of test apartment

Fig. 4 inside view of test room

Table 2 Building envelop information

Test Building	U value (W/(m ² ·K))	G value (-)
Roof	0.34	-
Exterior wall	0.57	-
Interior wall	0.92	-
Floor	1.08	-
Window	2.83	0.62

G value=total solar heat gain/incident solar radiation

133

134 3. Performance index

135 The system performance is tested in two modes: solar heating mode and CO₂ HP heating mode.

136 The COP of CO₂ HP (COP_{HP}) and the COP of the system (COP_s) are calculated by the equation (1-4).

137 The coefficient of performance of the CO₂ heat pump is given below.

$$COP_{HP} = \frac{c\rho F_{DP}(T_{HPo} - T_{HPi})}{P_{HP}} \quad (1)$$

138

139 Equation (2) is the calculation of the total heat gain of the system, including the heat storage in the

140 two tanks and heat consumed by the FCUs and DHW.

$$\begin{aligned} Q_s &= Q_{ST} + Q_{OT} + Q_{FCU} + Q_{DHW} \\ &= c\rho V_{ST}(T_{ST\tau} - T_{ST0}) + c\rho V_{OT}(T_{OT\tau} - T_{OT0}) \\ &\quad + \Delta\tau \sum_{i=1}^N c\rho F_{SP}(T_{FCUi} - T_{FCUo}) + \Delta\tau \sum_{i=1}^N c\rho F_{DHW}(T_{DHWo} - T_{DHWi}) \end{aligned} \quad (2)$$

141 As is shown in equation (3), the total electricity consumption consists of the electricity consumed

142 by CO₂ HP, water pumps and FCUs.

$$W_s = W_{HP} + W_{SCP} + W_{DP} + W_{SP} + W_{FCU} \quad (3)$$

143 The coefficient of performance regarding the whole solar assisted CO₂ heat pump system is given

144 below.

$$COP_s = \frac{Q_s}{W_s} \quad (4)$$

145 Solar fraction:

$$SF = \frac{Q_{SC}}{Q_s} \quad (5)$$

146 SC efficiency:

$$\eta = \frac{Q_{SC}}{\int_0^\tau A_{SC} G d\tau} \quad (6)$$

147

148 4. Experiments and discussion

149 4.1 Solar heating

150 Fig. 5 shows the ambient conditions when the tests were performed and the test results under solar

151 heating mode. The solar heating mode starts at 9:00 and ends at 17:00. The ambient temperature ranges
 152 from 7.1 °C to 11.4 °C and the average value is 10.1 °C. The solar radiation at 9:00 is 493W/m² and
 153 reaches the maximum value 816W/m² at 12:00. Initial water temperature in the storage tank is 37.2 °C
 154 and increases to the maximum value 61.8 °C at 13:10. Then, it decreases to 45.4 °C at the end of the test.
 155 The indoor room temperature rises from 13.5 °C to the maximum value 23.1 °C at 14:10 and stops
 156 at 22.1 °C in the end. As is shown in Fig. 5, the variation of the storage tank temperature shows the
 157 same trend as that of solar radiation, but the former is smoother and lags behind the latter. The same
 158 phenomenon happens between the indoor room temperature and the storage temperature. This is on
 159 account of the thermal storage capacity of the two tanks and the test building. Therefore, it leads to a
 160 smaller temperature fluctuation of indoor air when the solar radiation changes greatly during the whole
 161 process. Total heat gain and electricity consumption of the system are respectively 344.41 MJ and 7.1
 162 kWh (equal to 25.56 MJ). The COP of the system (COP_s) is calculated by means of equation (4). After
 163 8 hours operation, the COP_s reaches 13.5 under the solar heating mode.

Fig. 5 Test results under solar heating mode

4.2 CO₂ heat pump heating

During rainy days or at night, the water temperature in the storage tank can not meet the demand for space heating. Then the switching valves change to CO₂ heat pump heating mode. Fig.6 presents the variation of ambient temperature and indoor room temperature, as well as the inlet and outlet temperature of CO₂ HP. The ambient temperature ranges from 4.5 °C to 5.7 °C and the average value is 4.8 °C. Because the changing of the ambient temperature is relatively small, the indoor room temperature fluctuates slightly around 20 °C. The average temperature of the CO₂ HP inlet and outlet is 37.9 °C and 48.6 °C, respectively.

Fig. 6 test condition and results of CO₂ HP heating mode

Fig. 7 Heat capacity and electricity consumption of CO₂ HP

Fig. 7 shows the real time heat capacity and electricity consumption of the system, including the electricity consumption of the CO₂ HP, water pumps and FCUs. The average heat capacity of CO₂ HP is 6.1 kW and the average electricity consumption is 2.8 kW. As a result, the average COPs is 2.18 and the maximum and minimum value during the experiment is 2.24 and 2.11. During the test, the average high pressure in gas cooler is 10.3 Mpa and the evaporation pressure is 3.75 Mpa, which results in a compression ratio of 2.75. Under this condition, the average electricity consumption of CO₂ HP is 2.37 kW and by means of equation (1), the average COP_{HP} reaches 2.57 and the real time value is between 2.54 to 2.63. Due to the influence of auxiliaries such as FCU and pumps, the system COP_s is 15.2% lower than the heat pump COP_{HP}. The COP of CO₂ HP in this system is lower than a CO₂ HP water heater [13, 14] because of its high inlet water temperature when operating for residential heating[10].

5. Optimization and analysis

A system model based on TRNSYS is developed to analyze the influence of different parameters of the main components. The main purpose is to minimize the total annual electricity consumption of the whole system.

5.1 System model

Fig. 8 shows the schematic diagram of the TRNSYS model. The main components used in the simulation model are: weather data read and process module, solar collector module, controller module, tank module, pump module, FCU module, building module and CO₂ HP module. The type number of each module is listed in the figure. The parameters of main components are introduced in details in the following paragraphs.

Fig. 8 Diagram of TRNSYS model

5.1.1 Solar collector

Type 71 (evacuated tube solar collector) is used for the simulation of the SC array. The efficiency of the solar collector is calculated using the Hottel-Whiller[15] equation based on the data from the SC manufacturer, which is listed in the following equation:

$$\eta_{SC} = 0.71 - 0.50 \times \frac{(T_{SCo} + T_{SCi})/2 - T_{amb}}{G} - 0.0035 \times \frac{[(T_{SCo} + T_{SCi})/2 - T_{amb}]^2}{G} \quad (7)$$

5.1.2 CO₂ HP

Type 841[11] is a module developed in FORTRAN to simulate the performance of the 6 kW CO₂ HP, based on a series of real test data from the manufacturer. Fig. 9 shows the 2-D performance contour of the CO₂ HP with different supply water temperature and ambient temperature.

Fig. 9 COP versus ambient temperature and supply water temperature

5.1.3 Tanks

Type 4 and type 534 are chosen to simulate the storage tank and operation tank, respectively. The tank loss efficient is set to 0.8 W/(m²·K), which is tested by the manufacturer in the lab. One immersed coiled tube is set in type 534 to generate the domestic hot water. The domestic hot water distribution is shown in Fig.10, which is a real test data of a family of three people in Shanghai[16].

Fig. 10 Distribution of domestic hot water consumption

5.2 Model validation

In order to check the system model, a group of data under both two operation modes is chosen to validate the simulation. The weather data is shown in Fig.11, and average temperature and solar radiation intensity are 11°C and $410\text{W}/\text{m}^2$, respectively.

Fig. 11 Weather condition of validation test

As is shown in Fig.12, the system started under heat pump heating mode until 8:23 and switched to solar heating mode, when the temperature in the storage tank reached the level for space heating. Then the solar heating mode last until 20:00 and the system changed back to heat pump heating mode.

229 Because the heat pump model is compiled from the manufacturer real test data, the results of
 230 simulation and experiment are similar when operating under heat pump heating mode. During the solar
 231 heating mode, the simulated temperature in operation tank shows a good consistence with the tested
 232 results. The simulated temperature is higher and smoother than the tested results because the pipe heat
 233 loss and the ambient air convection influence on the system (especially on the tanks) are neglected in
 234 the model. The maximum error of temperature difference between simulation and experiment turns to
 235 be 6.9 °C at 14:30. The fluctuation of system power under solar heating mode is due to the control
 236 signal switch of the pumps at different time step. The integral values of the electricity consumptions for
 237 simulation and experiment are 43.9 kWh and 39.1 kWh (relative deviation 12.3%), respectively. In
 238 addition, the heat gain in the system are 760.1 MJ and 670.5 MJ (relative deviation 13.5%) for
 239 simulation and experiment during the integration period. By comparing the simulation and experiment
 240 results under hybrid mode operation condition, the system model is validated to do the further
 241 performance prediction and optimization of the system.

Fig. 12 Comparison of simulation and experiment results

5.3 Parametric analysis

245 In this session, the influences of two important parameters, the volume of the storage tank and the
246 operation tank, will be studied. The configuration and the operation parameters of CO₂ HP will not be
247 considered separately as the unit has been well designed as a product. However, the solar assisted CO₂
248 heat pump system as a whole will be optimized by means of multi-parameters optimization.

249 **5.3.1 Storage tank volume**

250 The storage tank volume has influence on the efficiency of the SC array as well as the heat
251 delivered to the operation tank. As is shown in Fig.13 (a), the heat from SC array rises rapidly at first
252 and then reaches a plateau with only a slight increase with increasing tank volume. The increase of
253 storage tank volume enlarges the total storage capacity and reduces the temperature of the SC array,
254 which leads to a high efficiency of the SC. However, as the area of the SC array is limited, with the
255 continuous growth of storage tank volume, the water temperature will be more difficult to reach the
256 temperature level for the operation tank demand. This results in that the HP needs more electricity to
257 generate more heat for the system in the end. The total electricity consists of two main parts: the pumps
258 electricity consumption and the HP electricity consumption. Actually, the pumps electricity
259 consumption increases directly with the growth of storage tank volume. The decrease of total electricity
260 consumption at first is due to the reduction of HP electricity consumption, and then the pumps and HP
261 electricity consumption comes to a balance. The variation of solar fraction is presented in Fig.13 (b),
262 which is consistent with the trend of heat from SC array.

(a)

(b)

Fig. 13 Influence of the storage tank volume

5.3.2 Operation tank volume

The operation tank volume also has a great influence on the heat from SC array, because it determines the storage capacity and temperature level of the supply side. However, the increase of the operation tank volume leads to a higher heat loss at the same time. The HP serves as an auxiliary heat source; therefore it will consume more electricity to maintain the temperature level for a bigger size of

the operation tank.

It's shown in Fig.14 (a) that the heat from SC array grows in step with the operation tank volume. This is on account of the increased total storage capacity. With a higher operation tank volume, the temperature fluctuation in the operation tank will be smoother and the temperature difference between storage tank and operation tank is formed easily. As a result, more heat will be delivered from the storage tank and the load of CO₂ HP decreases. However, when the operation tank volume reaches a critical value, the heat loss of the whole tank becomes an important factor, especially when the CO₂ HP operates alone. In this case, the growth of total heat demand leads to the increase of electricity consumption of the CO₂ HP. Therefore, the total electricity consumption decreases at the beginning and starts to increase when the operation tank volume reaches 1.7 m³. As is shown in Fig.14 (b), even though the heat from SC array has a continuous increase trend, the solar fraction becomes flat gradually at the end, which is different from the influence of storage tank. The increase of operation tank volume enlarges the total storage volume of the whole solar system, as a result, it improves the SC efficiency indirectly.

(a)

(b)

Fig. 14 Influence of operation tank volume

5.4 Multi-parameters optimization

As is shown in parametric analysis, both storage tank volume and operation tank volume have a great influence on the system performance. In practical design, the sizes of the two tanks need optimization simultaneously in a real operation mode. The objective of the optimization is to find reasonable values of the tank sizes to reach the minimum electricity consumption.

In addition to the model mentioned in section 4.1, TRNOPT (type583[17]) is used to call the external program Genopt[18] for iterative calculations. Considering the balance of time cost and accuracy of calculation, Hook-Jeeves algorithm (a direct search algorithm) is applied to perform the multi-parameters optimization. The objective function, shown in equation (8), is the total annual electricity consumption of the whole system, including the electricity consumption of CO₂ HP, FCUs and all the pumps.

$$E_s = E_{HP} + E_{SCP} + E_{DP} + E_{SP} + E_{FCU} \quad (8)$$

The optimization process and results are shown in Fig.15. The optimization starts from the initial

condition of the real system and reaches the final optimized value after 66 iterations. Furthermore, the total electricity consumption converges to 1550.4 kWh, while the storage tank volume is 2.21 m³ and the operation tank volume is 0.3 m³.

Fig.15 History plot of optimization process

The comparison of performance of the original and optimized system is shown in Fig.16. During the whole heating season, the optimized system can get 12.3% more heat from SC array, which result in an improvement of 8% of solar fraction. The electricity consumption comparison is shown in Fig.16 (b). Compared with the original system, the optimized system can save 14.2% percent of electricity. The comparison of COPs as well as the solar fraction is presented in Fig.16 (c) and (d), and the optimized system shows a significant improvement except in January. Due to the load characteristic, this system consumes 51% electricity of the whole heating season in January. As a result, the solar fraction in January reaches the lowest value of 44%, while the mean value of the heating season is 71.1%.

(a)

(b)

(c)

(d)

Fig.16 Performance comparison of original and optimized system

By integrating hourly results, the thermal balance between supply and consumption is obtained in

Fig.17. Total thermal energy supplied from CO₂ HP and SC array is 29454.4 MJ. The CO₂ HP accounts

for 28.9% of the total thermal supply with a mean COP of 2.55, so the simplified solar fraction is 71.1% for the entire heating season. The thermal output for the SH accounts for 86.7% of the total consumption, while the DHW accounts for 8.3%. The low percentage of DHW thermal is owing to the set temperature of DHW. Because the main purpose of this system is for space heating, the set temperature of DHW is limited to the water temperature in the operation tank. For the thermal loss of tank surfaces, the percentage is 5%.

Fig.17 Thermal balance of whole heating season

Subsequently, by setting the SC array to zero, the performance of a CO₂ HP space heating system without SC array can be obtained. The annual electricity consumption of the system is 3341.2 kWh, while value of the solar assisted CO₂ HP system is 1550.4 kWh. In this case, the addition of SC array can save 53.6% of the electricity consumption, which is 1790.8kWh every year.

6. Conclusion

In this study, a solar assisted CO₂ HP system was tested and optimized for space heating. With combination of CO₂ HP and SC, renewable energy was used to assist HP for heating function. Different operation mode was tested and analyzed. A simulation model based on TRNSYS was built to illustrate the influence of components' parameters on the system performance. Furthermore, a multi-parameters optimization was presented with regard to the storage tank volume and operation

349 tank volume to minimize the system electricity consumption. The following conclusion can be drawn:

350 (1) The system COPs of the solar heating and CO₂ HP heating modes can reach 13.5 and 2.18
351 respectively under typical test conditions.

352 (2) Based on the simulation model built in TRNSYS, the influence of single parameter was
353 analyzed. Two high weight parameters are chosen: the storage tank volume and operation
354 tank volume. The influence of these parameters on the system performance was illustrated.

355 (3) The multi-parameters optimization was presented with two interactional parameters in terms
356 of the results of single parameter analysis. The optimal values of storage tank volume and
357 operation tank are 2.21m³ and 0.3m³ to minimize the electricity consumption. The electricity
358 consumption of the optimized system is 1550.4 kWh and the solar fraction is 71.1%. The
359 optimized system can save 14.2% electricity and improve the solar fraction by 8%.

360 (4) Due to the weather and load characteristics in Shanghai, the system consumes about 51%
361 electricity in January, which results in the lowest solar fraction of 44%.

362 (5) Compared with the CO₂ HP heating system, the studied solar assisted system can save 53.6%
363 of the electricity consumption, which is 1790.8kWh every year.

364 In conclusion, the proposed system can meet the heating demand with less electricity consumption
365 by utilizing the feature of both CO₂ HP and solar thermal systems for heating. The solar thermal part
366 can also provide DHW with a higher temperature in summer. The solar energy can also be used to
367 drive absorption or adsorption chillers in summer in order to make full use of solar thermal system.

368

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370

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