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Performance enhancement of a solar-assisted air source heat pump system by phase change material using

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Abstract

In the previous studies, a solar-assisted air source heat pump (SAHP) system which consists of a solar thermal collector, an air source heat pump (ASHP) and a phase change material (PCM) unit has been proposed. The system is expected to contribute to energy saving and efficiency improving. In this paper, the design philosophy and procedure of the system is detailed first. Then a prototype with the corresponding experimental platform is set up, and a series of tests are designed and conducted. Through a performance analysis, it is clarified how system performance values are enhanced, and particularly, how the external environment influences the operation strategies of the system. To be specific, the results indicate that when the outdoor air temperature is higher than 38.0 °C, the energy efficiency ratio (EER) of the system for space cooling rises by 17%, assisted with the PCM unit, compared with that of ASHP under the same conditions. At a severely low ambient temperature (-10.0 °C below), the EER of the system rises by 65%, assisted with solar hot water (20.0 °C above), comparing to that of ASHP in the same environment. Briefly, new system performs well in both cooling and heating operations, even at the severely high / low ambient temperatures.

Keywords: Solar-assisted air source heat pump; Phase change material; Energy efficiency ratio; Performance enhancement

1. Introduction

Over the past two decades, heat pump technology has been increasingly mature. Employing heat pumps for residential heating and cooling can slash a half or more of CO2 emissions per annum [1]. Currently, the advanced cycle designs, improved cycle components and expanded cycle applications have been the main directions in this field [2]. Meanwhile, the latent heat thermal energy storage (LHTES) technology has been widely used in the composite heat pump systems for energy efficiency improving.

Active and-or passive heat storage for air conditioning (AC) do not only reduce energy consumption but balance the thermal energy inside building as well [3]. The application of PCM for heating energy discharge to defrost shows an evident enhancement of the system efficiency [4]. Free cooling networks reflect a nice potential in energy saving, efficiency thriving and environment protecting [5]. Ice storage combined with an AC system needs to improve energy efficiency further, while phase change material (PCM), which has a higher phase transition temperature, coupled with an AC / HP system is more promising [6].This paper presents a PCM based solar-assisted air source heat pump system (PCM-SAHP). Firstly, we introduce the design concept, configuration, and operation principle of the system. Then, we detail the testing methods and materials. Finally, we present the performance of the system and analyze it.

2. Methods and materials

A PCM-SAHP system consists of three parts: an air source heat pump (ASHP), a solar thermal collector, and a PCM unit (for cooling and heating energy storage). In this paper, we construct the PCM unit as a triplex tube heat exchange (TRTHE) unit, which is made of three concentric copper tubes. Details about the TRTHE unit and the system have been presented in literatures [7, 8]. Niu and his colleagues primarily proposed the SAHP system. Then, they validated the feasibility of the system used for cooling and heating operations. Schematic diagram of the system and structure of the TRTHE unit are shown in Fig. 1.In this study, the TRTHE units and a prototype of the system are redesigned. Nine operating modes and corresponding tests in the cooling / heating operation are projected and implemented.

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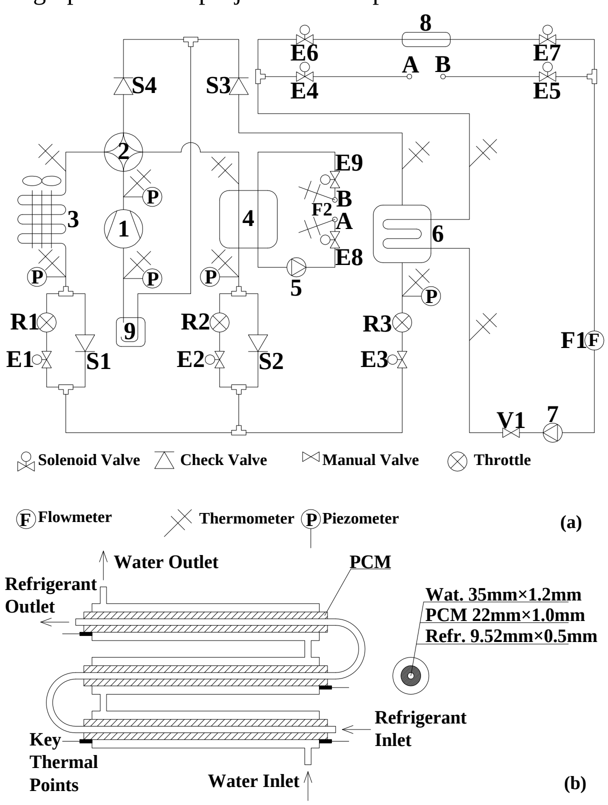

Fig. 1 Schematic of the SAHP system and structure of the TRTHE unit. (a) Schematic of the hybrid system, (b) structure of TRTHE unit. 1—Compressor, 2—Four-way reversing valve, 3—Outdoor finned tube heat exchanger (FTHE), 4—Indoor plate heat exchanger (PHE), 5—Water pump I, 6—Triplex tube heat exchanger (TRTHE), 7—Water pump II, 8—Solar thermal collector, 9—Gas-liquid separator, A—Supply water, B—Return water.

The TRTHE units in the SAHP system can realize nine operation modes, including (1) ASHP for space cooling mode (M1), (2) cooling energy storage mode (M2), (3) TRTHE units for space cooling mode (M3), (4) TRTHE units assisting ASHP for space cooling mode (M4), (5) ASHP for space heating mode (M5), (6) heating energy storage mode (M6), (7) TRTHE units for space heating mode (M7), (8) TRTHE units assisting ASHP for space heating mode (M8) and (9) solar hot water assisting TRTHE units for space heating mode (M9). All of those running modes are listed in Table 1.

For a TRTHE unit, the capacity of thermal energy storage is the prime parameter, which depends on the operation strategy of the system and the characteristic of air conditioning load for building. In this paper, a virtual building situated in Shanghai (China) is selected as the reference of air conditioning load and operation strategy. Because in this city, both the cooling load and the heating load have a similar value over a whole year. Considering the initial cost of equipments and the special discount tariff in Shanghai, the strategy of partial thermal energy storage is appropriate, meaning that the thermal energy discharged from the PCM has a responsibility to eliminate part of the thermal energy load of building. At the beginning and the end of space cooling season (less than 90 days), the percentage of cooling load to maximum value is about 30%, the TRTHE units can maintain thermal comfort level in chamber without any other chilled energy source operation from 10:00 to 18:00. Yet, at the highest ambient temperatures (from June 1st to September 30th, about 100 ~ 120 days), the TRTHE units assisting ASHP provide cooling energy from 10:00 to 18:00 to control the temperature and humidity in indoor space.

As mentioned afore, the mass consumption in the TRTHE units will determine the operation strategy of the system. As Table 2 shown, when the TRTHE units undertake 32% of the total cooling energy in building, the power consumption of compressor will be lower, and the "cheap" cooling energy stored by TRTHE units will be more. Finally, we choose this program for cooling operation. Thereafter, the structure design of the TRTHE unit is the issue.

The core of structure design is to ascertain the thickness of PCM interlayer in the TRTHE unit. In this paper, the diameter of refrigerant tube is given, so that to determine the diameter of medium tube is the core. Combining the heat transfer equations with the thermodynamic equations can achieve a relation between the thickness of PCM interlayer and the duration of cooling energy charge (or the duration of heating energy discharge). To store the cooling energy fast, and to release the heating energy mildly and steadily, the thickness of the PCM in the TRTHE unit should be controlled discreetly. In this paper, the geometric characteristics of the TRTHE unit are given in Table 3.

In this paper, the experimental platform is composed of a prototype of the SAHP system and a large enthalpy difference laboratory, which is set up according to the National Standard of Unitary Air Conditioners in China (GB/T 17758 - 2010). Three groups of TRTHE units are prepared. Pressure-points, temperature-points and volume-flow-points are plotted in Fig. 1. Electric parameters and time are observed and recorded likewise. All test parameters except electric parameters are measured and recorded every 30 s, and electric parameters are observed and recorded every 6 s. We have to confess that the solar hot water is prepared by the hot water tank instead. Space cooling and heating tests are designed and implemented, as Table 4 shown.

The COPs, which are calculated by Eq. (1) to Eq. (4), are used to analyze the system performance. Those paraphrases for each symbol are listed in Nomenclatures.

$$COP_{e\cdot ASHP} = Q_e \big/ W_{comp} \tag{1}$$

$$COP_{c\cdot ASHP} = Q_c \big/ W_{comp} \tag{2}$$

$$COP_{peri\cdot TTHE} = Q_{accu} \big/ W_{comp\cdot accu} \tag{3}$$

$$COP_{peri\cdot comb} = Q_{accu} \Big/ \left(W_{comp\cdot accu} + W_{TTHE\cdot accu} \right) \tag{4}$$

Table 1 Operation modes of the prototype

Mode	Manual Valve V1		Solenoid Valve E1 to E9		Pump 5		Pump 7	
	On	Off	On	Off	On	Off	On	Off
M1		●	2, 8, 9	1, 3~7	●			●
M2		●	3	1, 2, 4~9		●		●
M3	●		4, 5	1~3, 6~9		●	●	
M4	●		2, 4, 5, 8, 9	1, 3, 6, 7	●		●	
M5		●	1, 8, 9	2~7	●			●
M6	●		6, 7	1~5, 8, 9		●	●	
M7		●	3, 8, 9	1, 2, 4~7	●			●
M8		●	1, 3, 8, 9	2, 4~7	●			●
M9	●		3, 6~9	1, 2, 4 ,5	●		●	

Table 2 Capacity design for three groups of TRTHE units

Capacity design	The quotient about cooling capacity for TRTHE to total capacity		
	24%	28%	32%
Mass consumption of PCM (kg)	6.37	8.12	10.17
Chilled energy storage (MJ)	1.56	1.99	2.49
Duration of combined space cooling ($h \cdot d^{-1}$)	9	9	9
The number of combined space cooling days (d)	20	42	42
The number of space cooling by TRTHE days (d)	20	44	60

Table 3 Geometry details of the TRTHE unit

Items	Unit	Number
Diameter of the inner tube (outer diameter / wall thickness)	mm	9.52 / 0.5
Diameter of the medium tube (outer diameter / wall thickness)	mm	22 / 1.0
Diameter of the outer tube (outer diameter / wall thickness)	mm	35 / 1.2
Length of the tube (available heat transfer section)	m	0.8
Total length of the tube per unit (available heat transfer section)	m	64
Total number of unit	—	3
Total mass of the PCM per group	kg	3.5
Amount of thermal energy storage per group	kJ	857.5

Table 4 Experiments under various operation modes

Mode	Experimental conditions
M1	The ambient temperature is 30.0, 32.0, 35.0, 38.0, 40.0 °C respectively; the flow rate of water for space cooling is 800 $L \cdot h^{-1}$; the temperature of return water for space cooling is 12.0 ± 0.5 °C.
M2	The ambient temperature is 15.0, 20.0, 25.0, 30.0, 35.0 °C respectively; the activation condition is the PCM mean temperature of 15.0 ± 0.5 °C; the termination criterion is the PCM minimum temperature of 0 °C below.
M3	The temperature of return water for space cooling is 18.0 ± 0.5 °C; the flow rate of water through the TRTHE units is 40 $L \cdot h^{-1}$; the termination criterion is the supply chilling water temperature of 17.0 °C above.
M4	The ambient temperature is 35.0, 38.0, 40.0, 43.0 °C respectively; the flow rate of water through the plate heat exchanger is 800 $L \cdot h^{-1}$; the flow rate of water through the TRTHE units is 40 $L \cdot h^{-1}$; the temperature of return water for space cooling is 12.0 ± 0.5 °C.
M5	The ambient temperature is 7.0, -10.0, -12.0, -15.0, -17.0 °C respectively; the flow rate of water for space heating is 1260 $L \cdot h^{-1}$; the temperature of return water for space heating is 31.0 ± 0.5 °C.
M6	The temperature of solar hot water is 28.0 °C, the flow rate of solar hot water is 300 $L \cdot h^{-1}$, the termination criterion is the PCM temperature at the outlet of the TRTHE units of 20.0 °C above.
M7	The flow rate of water for space heating is 1260 $L \cdot h^{-1}$; the temperature of return water for space heating is 31.0 ± 0.5 °C.
M8	The ambient temperature is -17.0 °C; the flow rate of water for space heating is 1260 $L \cdot h^{-1}$; the temperature of return water for space heating is 31 ± 0.5 °C.
M9	The temperature of solar hot water is 28 °C; the flow rate of solar hot water is 300 $L \cdot h^{-1}$; the flow rate of water for space heating is 1260 $L \cdot h^{-1}$; the temperature of return water for space heating is 31.0 ± 0.5 °C.

3. Results and discussion

3.1 Tests under different cooling modes

At the outset, we test the performance of the system under M1 (ASHP for space cooling). At the typical condition, in which the ambient temperature is 35.0 °C, as the Standard of Unitary Air Conditioners in China (GB/T 17758 - 2010) suggested, the compressor power consumption and the cooling capacity achieve 2.08 kW and 5.41 kW respectively, which reaches the designing objective.

Under M2 (cooling energy storage), the charging duration of cooling energy achieves 11 min, when the ambient temperature at night is 25.0 °C. This value also realized the designing expectation. The total stored energy in the TRTHE units is 2.62 MJ, which is slightly higher than the designing demand.

From Fig. 2, we notice that every COP under different cooling strategies declines with the rising ambient temperature. When the ambient temperature rises from 30.0 to 40.0 °C, the COP under M1 falls off from 2.9 to 2.3. With the outdoor air temperature at night rises from 15.0 to 25.0 °C, the COP in a cooling cycle that M3 followed M2 decreases from 3.7 to 3.0. Yet, as this ambient temperature (at night) achieves 30.0 °C, the COP drops to 1.8 drastically. While the ambient temperature goes up from 35.0 to 43.0 °C, the COP in a cooling cycle that M4 followed M2 declines from 3.0 to 2.3. In the mean time, the percentage of cooling capacity of the TRTHE units to the total cooling capacity is about 31%, very close to the designed value of 32%.

Table 5 details the effect of ambient temperature on the compressor power consumption, the charging duration of cooling energy, and the temperature character of PCM in the TRTHE units. Notice that, both the power consumption and the charging duration increase with the rise of ambient temperature at night. Thus, the total energy consumption during the charging period of cooling energy jumps with the increase of outdoor air temperature at night. Meanwhile, with the ambient temperature at night rising from 15.0 to 35.0 °C, the mean value of PCM temperature rises from 1.3 to 2.4 °C, and the mean square error of that goes up from 3.8 to 5.5 °C. Considering the relatively high energy consumption at a high ambient temperature, the operation of M3 followed M2 is more appropriate at a relatively low ambient temperature at night.

3.2 Heating tests during diverse heating modes

To begin with, we test the performance of the system under M5 (ASHP for space heating). At a typical outdoor air temperature of 7.0 °C, the compressor power consumption and the heating capacity achieve 1.52 kW and 4.26 kW respectively, which realizes the intended design.

Yet, as Table 6 shown, with the ambient temperature decreasing from -10.0 to -17.0 °C, the heating capacity reduces from 2.98 to 2.33 kW. In the mean time, the power consumption of compressor falls off from 1.42 to 1.37 kW. At the same condition, we test the performance of the system under M7, M8 and M9 respectively, and the results are shown in Table 7. Notice that, under M9, heating capacity of the system achieves the maximum value, and the distribution of PCM temperature exhibits a great uniformity. The operation under M8 displays the highest power consumption but the lowest heating capacity. Although the operation under M7 shows a higher PCM temperature (both the mean value and the mean square error), the system heating capacity is not bad (despite there is a limit of the discharging duration of heating energy). Thus, alternating the TRTHE units for heating operation is a bright choice, so as to avert a degradation of heating performance at low ambient temperatures. Moreover, if the solar hot water is sustainable, to assist the TRTHE units, using solar hot water to supply heating energy (with a relatively low temperature, 20.0 °C, for instance) is preferable.

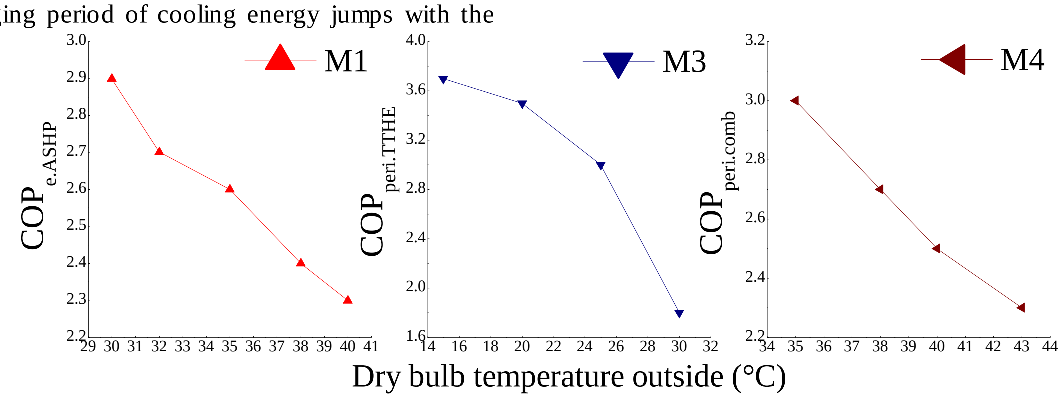

Fig.2 The average coefficient of performance during M1, M3 and M4 respectively

Table 5 Operating parameters in M2

Operating parameters	Dry bulb temperature outside at night (°C)				
	15.0	20.0	25.0	30.0	35.0
Compressor power consumption (kW)	1.22	1.27	1.41	1.56	1.69
Cooling charge duration (min)	16	16	17	26	33

Amount of compressor power consumption (MJ)	1.17	1.22	1.44	2.43	3.35
Mean temperature of PCM (°C)	1.8	1.3	2.4	2.5	2.4
Mean square error of PCM temperature (°C)	4.5	3.8	4.7	5.1	5.5

Table 6 Operating parameters in M5 during low ambient temperature condition

Operating parameters	Dry bulb temperature outside (°C)			
	-10.0	-12.0	-15.0	-17.0
Heating capacity (kW)	2.98	2.66	2.36	2.33
Compressor power consumption (kW)	1.42	1.40	1.39	1.37

Table 7 Operating parameters in comparison among diverse space heating mode

Operating parameters	Space heating mode			
	M5	M7	M8	M9
Compressor power consumption (kW)	1.37	1.40	1.56	1.30
Heating capacity (kW)	2.33	5.30	4.20	6.00
Mean temperature of PCM (°C)	—	4.8	5.5	4.8
Mean square error of PCM temperature (°C)	—	4.8	6.9	2.0

6. Conclusions

A phase change material (PCM) based solar-assisted air source heat pump (SAHP) system has been developed. A prototype of the system is set up. Nine groups of tests are conducted. The performance of the system under different operation modes or strategies is discussed. From the present study, the following conclusions might be drawn.

(1) New system can be operated flexibly and efficiently, with an operation of thermal energy charging and discharging.

(2) The ambient temperature has a strong impact on the cooling performance of the system. It is found to be helpful to operate the TRTHE units for space cooling.

(3) An ASHP often suffers from a degradation of heating performance at low ambient temperatures. Using the TRTHE units instead of the outdoor heat exchanger is helpful to avoid that suffering, especially with the solar hot water (a relatively low temperature is enough, 20.0 °C for example) assisting.

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Nomenclatures

AC	Air conditioning
ASHP	Air source heat pump
COP	Coefficient of performance
$COP_{c \cdot ASHP}$	Heating COP of ASHP
$COP_{e \cdot ASHP}$	Cooling COP of ASHP
$COP_{peri \cdot TRTHE}$	Cooling COP of TRTHE during a cooling cycle
$COP_{peri \cdot comb}$	Combined cooling COP during a cooling cycle
FTHE	Finned tube heat exchanger
HP	Heat pump
HTF	Heat transfer fluid
PHE	Plate heat exchanger
Q_{accu}	Amount of cooling capacity, MJ
Q_c	Heating capacity, kW
Q_e	Cooling capacity, kW
$Q_{e \cdot ASHP}$	Cooling capacity of ASHP, kW
$Q_{e \cdot TRTHE}$	Cooling capacity of TRTHE, kW
TRTHE	Triplex tube heat exchanger
W_{comp}	Compressor power consumption, kW
$W_{comp \cdot accu}$	Amount of power consumption for compressor, MJ
$W_{TRTHE \cdot accu}$	Amount of compressor power consumption for cooling charge, MJ

The 1st International Conference on Energy, Environment and Economics (ICEEE 2016) was held at Heriot-Watt University, Edinburgh, EH14 4AS, UK, 16-18 August 2016. ICEEE2016 focused on energy, environment and economics of energy systems and their applications. More than fifty eight delegates from 31 countries with diverse expertise ranging from energy economics, solar thermal, water engineering, automotive, energy, economics and policy, sustainable development, bio fuels, Nano technologies, climate change, life cycle analysis etc. made conference true to its name and completely international. During conference total 51 oral presentations and six posters were shared between delegates. The presentations showed the depth and breadth of research across different research areas ranging from diverse background. ICEEE2016 aimed:

- To identify and share experiences, challenges and technical expertise on how to tackle growing energy use and greenhouse gas emissions and how to promote sustainability and economical, cost effective energy efficiency measures.

In total 11 technical sessions and two invited talks both from academia and industry provided insight into the recent development on the proposed theme of the conference. Preparation, organisation and delivery of the conference started from July 2015 and further co-ordinated by vibrant team of Conference Centre, Heriot Watt University. Conference organisers would like to acknowledge support from the sponsors particularly World Scientific Publication ltd and its team members for the delivery of the conference. Organisers are also thankful to all reviewers who contributed during peer review process and their contributions are well appreciated. At the end and during vote of thanks following awards have been announced and we would like to congratulate all well deserving delegates.

- Best Paper –Academia: Amela Ajanovic, EEG, TU Vienna, Austria
- Best Paper – Student : Christian Jenne,University of Duisburg-Essen, Germany
- Best Poster – Student: Yoann Guinard, University of New South Wales, Sydney, Australia
- Best Poster – Academia: E. Salleh, Universiti Kebangsaan Malaysia, Malaysia
- Active Participation Award - Yoann Guinard, University of New South Wales, Sydney, Australia

At the end we would like to extend our gratitude to all of you for your participation and hopefully welcome you again during ICEEE2017.

Editors:
Dr. Singh is Senior Scientist at Indian Agricultural Research Institute, New Delhi, India. Her area of expertise are bio energy and bio fuels, environmental engineering, carbon accounting and renewable energy integration for rural development.
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