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Performance analysis of a multi-functional heat pump system in cooling mode

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HIGHLIGHTS

- Performance of a multi-functional heat pump system was investigated in cooling mode.
- Better performance than conventional air-source heat pump system.
- Series heat sink combination can provide better system performance.
- Gray water temperature has limited impact on parallel and series.
- Supplying hot water can improve the *COP* and cooling capacity for space cooling.

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ABSTRACT

A multi-functional heat pump system is proposed to effectively utilize waste heat and heat capacity in gray water for heating or cooling of residential buildings. Heat is also reclaimed from a plate heat exchanger installed at the discharge outlet of the compressor to provide sufficient hot water for residential use. To study the performance of this innovative system, laboratory testing has been performed with a prototype consisting of an outdoor heat pump, an indoor air handler, a gray water tank and a hot-water tank. This system is set in two environmental chambers that they mimic the outdoor and indoor environments, respectively. In this paper, the investigation of the proposed system is focused on the performance in cooling mode. The multi-functional heat pump system has been run under (i) space cooling mode and (ii) space cooling plus hot-water supply mode, with the same temperature conditions. The system performances in these two modes are compared and analyzed. The system is designed to allow four combinations of heat sinks with a water sink condenser and an air sink condenser. The four combinations are (1) air sink only, (2) water sink only, (3) air sink and water sink in parallel and (4) air sink and water sink in series, at the refrigerant cycle. Performance of the four combinations of heat sinks is experimentally investigated at a typical indoor air temperature of 26.7 °C and various outdoor air temperatures at 29.4 °C, 35 °C, and 40.5 °C. The results show that the heat sink combinations influence the cooling capacity and coefficient of performance (*COP*) of the system. The system performance and the optimal heat sink combination depend on the outdoor temperature. The impacts of outdoor temperature and gray water temperature on the performance of the system are discussed. The dynamic performance of the system for heating hot water from 30 °C to 48.9 °C is also studied. The proposed system has been shown providing significant energy savings in space cooling and hot-water supply. Moreover, the optimal source combination is critical in pursuing the maximum energy savings.

1. Introduction

Water heating accounts for an average 18% of all residential energy use in the United States, which makes it to be the second largest use of energy in residential buildings [1]. In some states (e.g., California), the portion of energy use can reach as high as 25% of the total energy consumption. In a conventional building system, water heating usually is done by electricity or gas unit. However, this method consumes much energy compared with using heat pump systems. Therefore, heat pumps are becoming more popular for heating and cooling applications in residential buildings for energy saving. Ground-source and air-source heat pumps, as well

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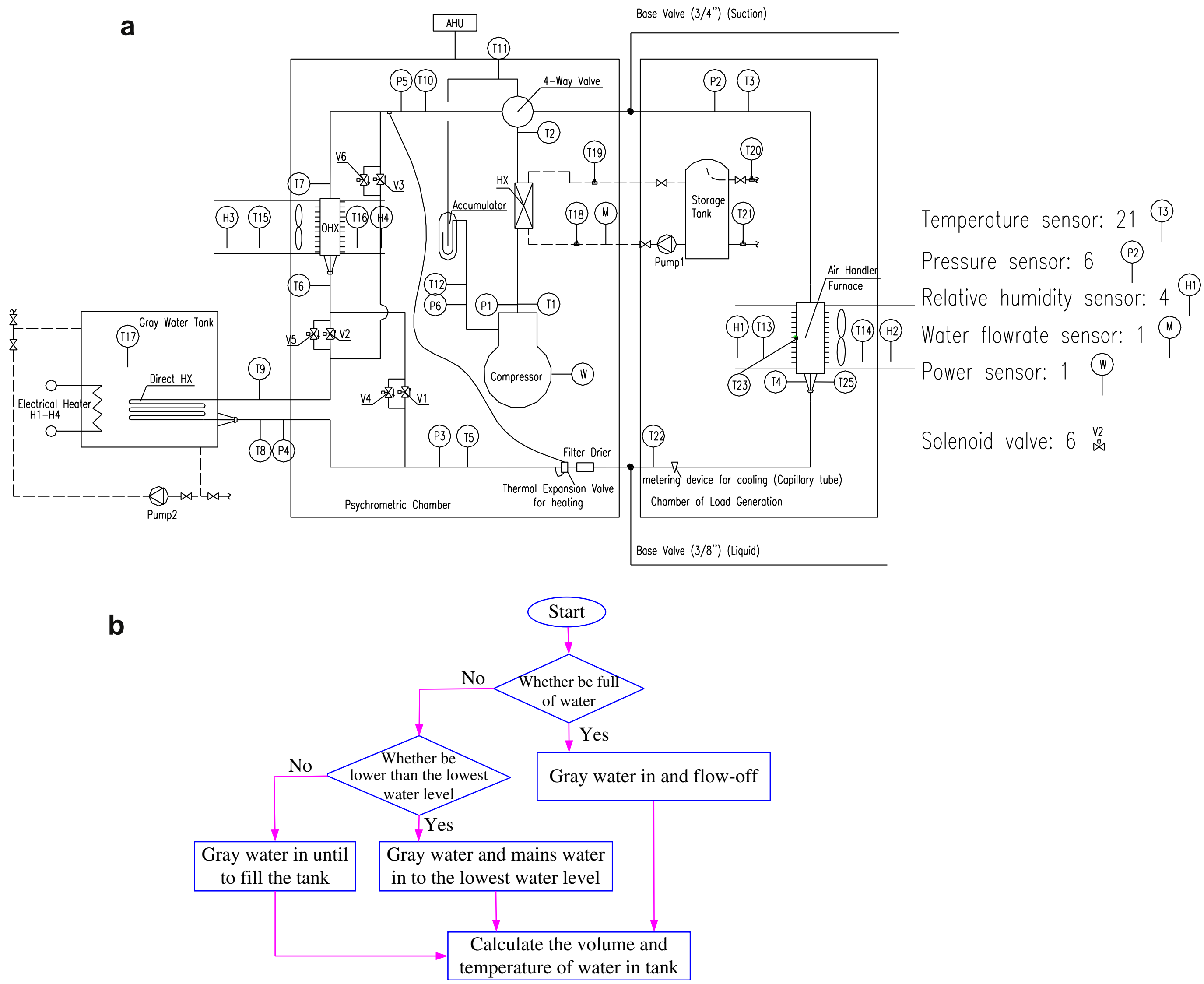

Fig. 1. (a) Schematic diagram of the prototype (multi-functional heat pump system). (b) The operation strategies of gray water tank.

as combining solar energy and geothermal heat pump, were proposed by many researchers [2]. Since the 1950s, research has been performed on heat pump water heaters [3] for energy saving. The potential energy sources (for instance, air and water) have been considered. Ito and Miura [4] have investigated the mechanisms of heat pumps for hot-water supply using combined air and water sources. The system can switch to either one or both heat sources. Direct expansion solar-assisted heat pump system that combined solar and air heat sources has been proposed to generate hot water [5–10]. Arif et al. [11,12] have been investigated exegetic modeling and performance evaluation of a solar-assisted domestic hot-water tank integrated geothermal heat pump systems for residential buildings has been performed. However, their research works are focused on saving energy in supplying hot water, and did not consider the potential energy saving with integration of air conditioning and hot-water supply systems. Heat pump water heaters for service water heating have hot-water production rate only 40–100% of that of the electric heating devices and 30–50% of that of the gas heating devices [13]. To provide quick recovery with this type of water heater, a household must have a large heat pump, an unusually large storage tank, and an electric backup heater. However, this electric backup heater will increase peak electrical demand and reduces overall energy efficiency [13].

Water heating is just a part of total energy consumption in buildings. In fact, the space heating and cooling consume a significant amount of energy. To further improve the energy efficiency of heat pumps in various applications, numerous researchers have investigated multi-functional heat pump system that not only provides hot water but also space heating and cooling. In residential buildings, the load of hot water can be satisfied by the multi-functional heat pump systems, meanwhile space cooling and heating can be provided. Ni et al. [14] investigated this type of system numerically, and showed the mean of daily hot-water load at a typical residential house in New York is about 33.6 MJ. The study is based on a calculation using the methods provided by Building America Research Benchmark [15]. Considering the usage profile [15,16], the mean hourly load of hot water is about 1.4 MJ. Through the numerical simulation, Ni et al. [14] concluded that the total source energy savings have a range of 17–57.9% among 15 cities in different climate zones in the U.S. using a combined heat pump system for hot-water heating and space heating and cooling. Reclamation of heat from the gray water was considered in their system. Hot-water heating has the most significant energy savings with over 60% reduction.

Ji et al. [13,17] developed a prototype system and simulation program for an integrated domestic air-conditioner and water heater. This prototype only utilized the air source for heat source/sink. Kara et al. [18,19] applied the direct expansion solar-assisted heat pump for space heating, space cooling and hot-water supply. Ozgener and Hepbasli [11,20–22] developed a multi-function heat pump system by utilizing solar energy and geothermal heat. However, those previous research works did not investigate the impaction of combinations of heat sources/sinks on system performance.

Ni et al. [14] have discussed the performance of a prototype of multi-function heat pump system with utilizing air and waste-water heat sinks. They have given the numerical study of the heat pump system and presented the system operation strategy and energy savings. Liu et al. [23] have experimentally studied the performance of the proposed system in heating mode. With a support from the U.S. Environmental Protection Agency, Hu [24] has developed a gray water treating system consisted of a simple screen, a bio-filter filled with shredded tire chips and a membrane bioreactor, which was intended to be applied to the proposed combined heat pump system with gray water source. In the proposed system, there are four types of combinations of air sink and waste-water sink for space cooling and hot-water heating, consisting of "air sink only," "water sink only," "air and water sinks in parallel" and "air and water sinks in series." The "air sink only" solely uses the air-to-refrigerant heat exchanger in outdoor chamber for heat rejection, while the "water sink only" only uses the water-to-refrigerant heat exchanger located in the gray water tank. The "air and water sinks in parallel" represents the air-to-refrigerant and water-to-refrigerant heat exchangers, mentioned above, are configured in parallel. The "air and water sinks in series" represents the heat exchangers are configured in series. Five basic functions can be realized by this prototype system, which are space heating, space heating with hot-water supply, space cooling, space cooling with hot-water supply and hot-water supply only. In this paper, the system performance in different functions with different types of heat sinks combinations will be discussed.

Nowadays, heat pumps become much widely used in residential buildings, especially using air-source heat pumps. The prototype system was modified from an existing air-source heat pump system. This paper will show the possibility and performance of modifying an existing air-source heat pump to a multi-function heat pump system. A significant energy saving on retrofitting existing air-source heat pumps with the multi-functional heat pump systems will be expected.

2. Prototype setup

To study the performance of this multi-functional heat pump system, a prototype system is set up in a laboratory at the University of Nebraska-Lincoln, which consists of a modified heat pump system and a hot-water supply system. The system is installed in two separated laboratory rooms that they mimic outdoor and indoor environments, respectively. This heat pump system was modified from a conventional air-source heat pump system. It consists of a compressor, an accumulator, a heat exchanger with a fan in the outdoor chamber, an indoor air handler including a heat exchanger and a fan, and a gray water tank with an immersed direct expansion (DX) coil. The hot-water supply system consists of a water pump for circulating water, a 30-gallon hot-water tank for storage of hot water, and a plate heat exchanger for heating hot water. The refrigerant filled is R410A.

In this prototype system, one four-way valve is installed at the outlet of the compressor to switch between heating mode and cooling mode. As shown in Fig. 1(a), six solenoid valves are used to guide the refrigerant bypassing different heat exchangers. There are two throttling valves being used for heating mode and cooling mode, which are a thermal expansion valve and a metering device (capillary), respectively. The pressure and temperature of refrigerant were measured at the locations shown in Fig. 1(a). Six pressure sensors are installed at different positions to measure the pressure distributions of the refrigerant flow. Temperatures are measured by copper-constantan thermocouples and platinum resistance thermometers. The air volume rate and temperature of airflow through the air-to-refrigerant heat exchanger were measured by installing hot wire airflow meter and thermometers at the inlet and outlet of the air handler. An in-line water flow meter (Piston flow meter) is used to measure the hot-water flow rate. A digital power meter is used to measure the overall power consumptions of the compressor and the fans. All the above measuring processes are monitored and controlled by a National Instruments data acquisition system which includes NI PXIe-1073, NI PXI-6224 and NI PXIe-4353. The data is recorded at one-second interval. Table 1 shows the range and accuracy of the sensors installed in the system.

This system is designed to provide (1) space heating only, (2) space heating plus hot-water supply, (3) space cooling, (4) space cooling plus hot-water supply, and (5) hot-water supply only. The first two heating modes were discussed by the authors in Liu et al. [23]. In this paper, the study will be focused on the performance of the multi-functional system in cooling mode that those are the modes of (1) space cooling only and (2) space cooling plus hot-water supply.

In this system, four types of heat–sink combinations for space cooling and hot-water heating can be implemented as showed as follows:

(a) Air sink only

Outdoor atmosphere is a good place for heat ejection. In this mode, the air-to-refrigerant (or air source) heat exchanger in the outdoor chamber serves as a condenser, while the air-to-refrigerant heat exchanger in the indoor chamber works as an evaporator. The

Table 1
Range and accuracy of sensors.

Sensors	Range	Output	Input	Accuracy
Thermocouple wire (T type)	<200 °C	N/A	N/A	0.75% above 0 °C 1.5% below 0 °C
Humidity/Temperature transmitters	(Humidity): 0 to 100% RH @ 0 to 50 °C (Temperature): 0 to 100 °C	4–20 mA	24 Vdc	±2.5% RH, ±0.3 °C
Gauge pressure transducer	0–1000 psi	4–20 mA	24 Vdc	±0.25%
	0–500 psi	4–20 mA	24 Vdc	±0.25%
	0–200 psi	4–20 mA	24 Vdc	±0.25%
In-line water flow meter	0–15 L/min, 0–121 °C	N/A	N/A	±5%
Power meter	208/240 V, 100 A	4–20 mA	9–30 Vdc	±1%

high temperature and high pressure refrigerant directly enters the outdoor heat exchanger, without passing through the immersed water-to-refrigerant heat exchanger in the water tank. The heat exchanger in the outdoor chamber is shown in Fig. 1(a).

(b) Water sink only

In this mode, water in the gray water tank serves as a heat sink to absorb ejected heat from the refrigerant. The water-to-refrigerant (or water source) heat exchanger immersed in the gray water tank works as a condenser and the air-to-refrigerant heat exchanger in the indoor chamber operates as an evaporator. The high temperature and high pressure refrigerant passes through the water-to-refrigerant heat exchanger without going through the air-to-refrigerant heat exchanger.

(c) Air and water sinks in parallel

The air-to-refrigerant heat exchanger and water-to-refrigerant heat exchanger are arranged in parallel in the refrigeration circuit. These heat exchangers in the outdoor chamber and water tank both serve as condensers in the cooling mode. The heat exchanger in the indoor chamber acts as an evaporator. The refrigerant passes through two condensers simultaneously. The refrigerant flow passing through different heat exchangers will be adjusted to enhance the condensing effect.

(d) Air and water sinks in series

It is the same as the above (c), but the air-to-refrigerant and water-to-refrigerant heat exchangers are arranged in series in the refrigeration circuit. The refrigerant, firstly, passes through the air-to-refrigerant heat exchanger and then goes through the water-to-refrigerant heat exchanger. This arrangement will make the refrigerant flow passing through each heat exchanger to eject as much heat as possible. In this prototype, air-to-refrigerant heat exchanger is ahead of water-to-refrigerant heat exchanger, because the gray water temperature is normally lower than that of ambient air. This arrangement tries to maximize the reduction of condenser temperature and thus maximize the heat rejection from refrigerant condense to gray water.

Table 2 shows the valves schedule to switch among different heat sink combinations. The locations of the valves are shown in Fig. 1(a) and all the valves are closed by default. When the air source is the sole heat sink, only solenoid valve V4 in Fig. 1(a) is opened. When the water source is the sole heat sink, only solenoid valve V6 in Fig. 1(a) is opened. When the air source and water source work in parallel, V4 and V6 valves are opened. While the air source and water source work in series, only V2 valve is opened. Pump 1 presents the circulating pump in gray water tank and pump 2 presents the circulating pump in hot-water supply system as shown in Fig. 1(a).

Table 2
Heat sink combinations and pump schedules in cooling mode.

Heat sink combinations	Opened valves	Space cooling only	Space cooling plus hot-water supply
Air sink only	V4	Pump 1 off, Pump 2 off	Pump 1 on, Pump 2 off
Water sink only	V6	Pump 1 off, Pump 2 on	Pump 1 on, Pump 2 on
Air and water sinks in parallel	V4 and V6	Pump 1 off, Pump 2 on	Pump 1 on, Pump 2 on
Air and water sinks in series	V2	Pump 1 off, Pump 2 on	Pump 1 on, Pump 2 on

In cooling mode, this system can realize two functions, which are (i) space cooling only and (ii) space cooling plus hot-water supply. In space cooling plus hot-water supply mode, two subsidiary modes will also be discussed in this section later.

(a) Space cooling only mode

An air handler is a device used to delivery cold air in cooling mode and hot air in heating mode into the room. In this mode, the circulation pump is turned off and thus there is no water circulation for hot-water heating. The heat loss can therefore be neglected, when the high temperature and high pressure refrigerant vapor from the outlet of compressor passes through the heat exchanger for hot-water heating. According to the standard ANSI/AHRI Standard 210/240 and ANSI/AHRI Standard 320 [25,26] for standard test conditions, the room temperature is maintained at dry bulb 26.7 ± 1 °C and wet bulb 16.67 ± 1 °C. The gray water temperature is set as 29.7 ± 1 °C [26]. The temperature of the outdoor chamber is controlled as required. The temperatures in the outdoor chamber and the indoor chamber are individually controlled by two roof top units (RTUs).

At the space cooling mode, there is no hot-water supply and the system *COP* at any time instant (t) can be determined by:

$$COP_{c,t} = \frac{Q_c(t)}{W(t)}, \tag{1}$$

where $Q_c(t)$ is the heat exchange rate in the evaporator. $W(t)$ denotes the total power of the compressor and fans. Within an operating period of duration, the average *COP* is defined as

$$COP_{c,t} = \frac{\int_0^\tau Q_c(t)\mathrm{d}t}{\int_0^\tau W(t)\mathrm{d}t}, \tag{2}$$

and

$$Q_c(t) = \dot{m}_a \times (h_i - h_o) - \dot{m}_a \times (w_i - w_o) \times h_w, \tag{3}$$

where $\dot{m}_a$ is the mass flow rate of dry air; h_i is the enthalpy of the air at the inlet of the evaporator; h_o is the enthalpy of the air at the outlet of the evaporator; w_i is the humidity ratio of the inlet of the evaporator; w_o is the humidity ratio of the outlet of the evaporator; h_w is the enthalpy of the condensation water. (b) Space cooling plus hot-water supply mode.

In this mode, hot-water heating and space cooling are provided simultaneously. There are two subsidiary models: (i) sub-mode I, the plate heat exchanger for hot-water heating and air-to-refrigerant heat exchanger both work as condensers while air-to-refrigerant heat exchanger is in forced convection condition, which means the fan is on. (ii) Sub-mode II, the plate heat exchanger and air-to-refrigerant heat exchanger work as condensers while air-to-refrigerant heat exchanger is in natural convection condition, which means the fan is off. The high temperature and high pressure refrigerant vapor first passes through the plate heat exchanger for heating hot water and then goes into the air-to-refrigerant heat exchanger in the outdoor chamber. The pump in the hot-water system is turned on to maintain a constant flow rate (10 L/min in the sub-mode I and 15 L/min in the sub-mode II) passing through the plate heat exchanger. The space cooling plus hot-water supply mode tries to improve the performance of the system and maximize utilization of the energy by reclaiming

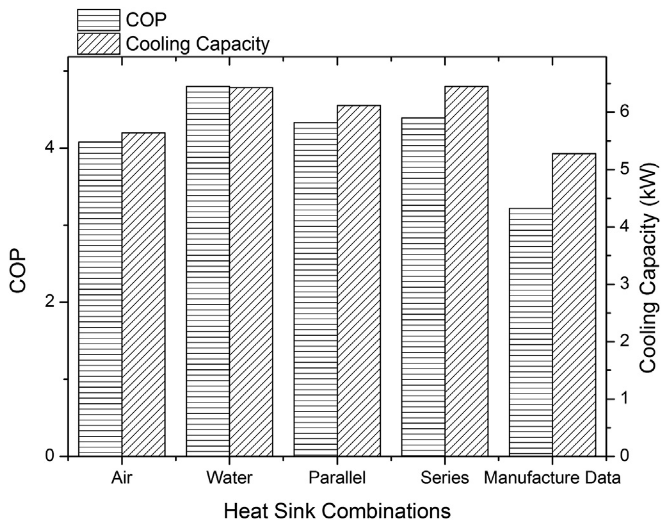

Fig. 2. *COP* and cooling capacity of the proposed and original systems at outdoor air temperature of 35 °C.

condensing heat. In the present study, the hot-water temperature is maintained at 48.9 ± 1 °C. When the space cooling mode and hot-water supply are required simultaneously, the system *COP* at any time instant (t) is given as

$$COP_{c,t} = \frac{Q_c(t) + Q_{h,w}(t)}{W(t)}, \tag{4}$$

where $Q_{h,w}(t)$ is the heat exchange rate in the plate heat exchanger for heating hot water. The pump used for circulation of hot water is excluded in the calculation due to its small power at 30 W. Within an operating period of duration τ, the average *COP* is defined as

$$COP_{c,t} = \frac{\int_0^{\tau} (Q_c(t) + Q_{h,w}(t))\mathrm{d}t}{\int_0^{\tau} W(t)\mathrm{d}t} \tag{5}$$

3. Experimental results analysis

3.1. Comparison of system performance with the manufacturer's data of original system

This multi-functional heat pump system is developed based on a market-available air-source heat pump system. To investigate the general improvement of the modified system, the performance of the multi-functional heat pump system is compared with the specification in a rated operation condition of the original (before modification) system shown in the manufacture technical guide. The purpose is to investigate the effect of system modification on the performance, compared with that of the original design. The technical guide gives the system performance rating data of the air-source heat pump on typical operating conditions in cooling mode. In this comparison, the indoor temperatures are chosen at dry bulb 26.67 ± 1 °C and wet bulb 16.67 ± 1 °C, and outdoor temperatures are chosen at dry bulb 35 ± 1 °C and wet bulb 23.9 ± 1 °C, referring to the rated operating condition. Other parameters are based on the standard ANSI/AHRI Standard 210/240 and ANSI/AHRI Standard 320 [25,26]. The gray water temperature is set as 29.4 ± 1 °C [26].

Table 3
Experiment conditions for space cooling mode.

Outside air temperature (°C)			Indoor air temperature (°C)	Gray water tank temperature (°C)
40.6	35	29.4	26.7	29.4

The system performance is obtained from the tests of the multi-functional heat pump system with all heat sink combinations. The manufacture data was obtained from the performance data in the technical guide.

Fig. 2 compares the *COP*s and cooling capacities of the multi-functional and the original air-source heat pump systems. The multi-functional heat pump system has a higher *COP* than the rated *COP* of the original system, especially when the heat sink is water only. The improvement of *COP*s of the proposed system is from 35.55% to 59.47% compared with the manufacture data of the original air-source heat-pump system. Moreover, cooling capacity of the system is improved from 14.63% to 31.1%. When heat sink is water only, the *COP* is the highest and cooling capacity is the second. When heat sinks are in series, the cooling capacity is the highest and *COP* is the second. Compared with conventional air-source heat pump system, this modification significantly improves the system performance.

3.2. Space cooling only mode

Table 3 shows the operating conditions of the multi-functional heat pump system in space cooling mode with all combinations. The experiments of four heat sink combinations are implemented at three different outdoor temperatures. The temperatures of the indoor chamber and gray water tank maintain at 26.7 °C and 29.4 °C, respectively. A total of twelve experiments (i.e., three conditions and four combinations) have been made.

Fig. 3 shows the *COP* and cooling capacity of the heat pump system with the four heat sink combinations at the outdoor temperature is 40.6 °C. The *COP* of the system is the highest when the heat sink is the water sink only. The *COP* of the system with air as a heat sink becomes the worst among those four combinations. The cooling capacities of the system with the heat sinks in series and water sink only combination are almost the same. The cooling capacity of the system with air sink only is the lowest. When the outdoor air temperature drops to 35 °C, shown in Fig. 2, the *COP* and

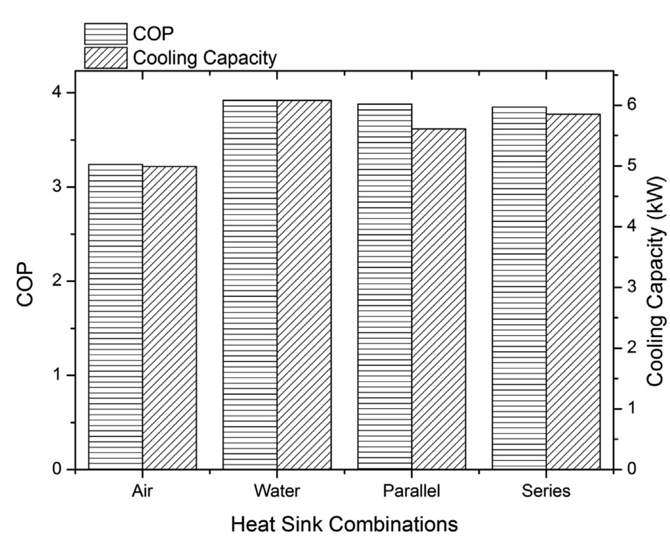

Fig. 3. *COP* and cooling capacity at outdoor temperature 40.6 °C.

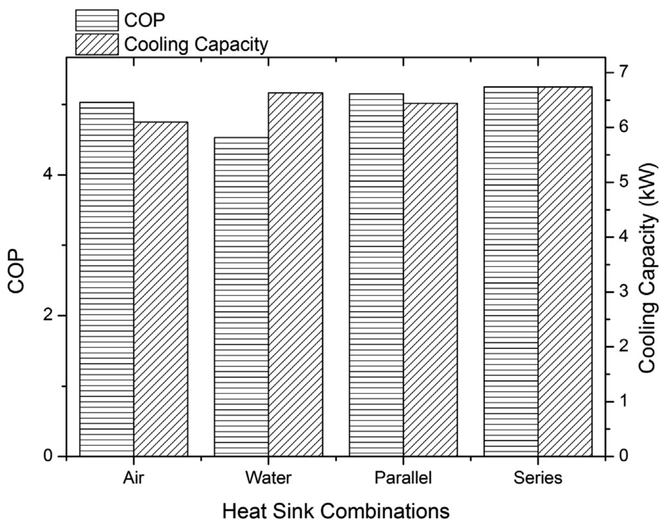

Fig. 4. *COP* and cooling capacity at outdoor temperature 29.4 °C.

cooling capacity of the air sink only are still the lowest among the four combinations. As the outdoor temperature further drops to 29.4 °C, the *COP* of the system with the water sink only becomes the worst among the four types of combinations and that of series combination is the best, as shown in Fig. 4. However, the cooling capacity of the system with water sink only is second highest. The cooling capacity of the system with series combination is the highest and that of the system with air sink only is almost the lowest. Compared of Figs. 2 and 3, when the outdoor temperature rises from 35 °C to 40.5 °C, the *COP* and cooling capacity of the system with water sinks only drop 18.33% and 5.44%, respectively. Those of the system with air sink only drop 20.59% and 11.52%, respectively. The *COP* and cooling capacity of parallel and series heat sink combinations drop less than those of air sink only or water sink only. When outdoor air temperature becomes 29.4 °C, compared of Figs. 2 and 4, the *COP* and cooling capacity of the air sink only increases 23.28% and 8.16%, respectively. The increases are larger than those of other heat sink combinations. The performance of the system with water sink only has limited improvement as the outdoor temperature changes.

From Figs. 2–4, the outdoor temperature changes affects the changes of the performance of the system with water sink only. As the similar phenomenon discussed in previous paper [23], although the air-to-refrigerant heat exchanger is bypassed by solenoid valve 6, only one end of the air-to-refrigerant heat exchanger is closed by V4 and another end is connected with the inlet of the water sink. Because there is no refrigerant receiver in this system, the heat exchanger plays a role in storage of refrigerant. Due to the change of outdoor temperature, the pressure in air-to-refrigerant heat exchanger will fluctuate which will further conduct the fluctuation of the refrigerant amount passing through the water-to-refrigerant heat exchanger. The change of the refrigerant flow rate will cause the fluctuation of cooling capacity.

3.3. Impaction of water temperature on system performance for space cooling

In controlling and maintaining water volume in gray water tank, mains water plays a role of a supplemental water sink. In some cases, due to the irrigation need, the water volume may fall below a minimum limit of gray water tank. The mains water will be filled in to maintain a minimum water volume in gray water tank. According to the control algorithm given in Ni et al. (2012), different operation modes will be carried out corresponding to the change of gray water temperature, shown in Fig. 1(b). In this sub-section, the impact of gray water temperature on the performance of the heat pump system is discussed.

The mains-water temperature in a day for a typical residential building varies significantly depending on location and the time of year. The equation below can be used to determine daily mains-water temperature, which is based on typical meteorological year data for the location of the prototype [27]:

$$T_{mains} = \left(T_{amb,avg} + offset\right) + ratio \times \left(\Delta T_{amb,\max}/2\right) \times \sin\left[0.986(day\# - 15 - lag) - 90\right], \quad (6)$$

where: T_{mains} = mains (supply) temperature to domestic hot-water tank, °F; $T_{amb,avg}$ = annual average ambient air temperature, °F; $\Delta T_{amb,\max}$ = maximum difference between monthly average ambient temperatures, °F; 0.986 = degrees/day (360/365); $day\#$ = Julian day of the year, 1–365 d; $offset$ = 6 °F; $ratio = 0.4 + 0.01(T_{amb,avg} - 44)$; $lag = 35 - 1.0(T_{amb,avg} - 44)$, °F.

Attention should be paid on the unit of the term $[0.986(day\# - 15 - lag) - 90]$, which is in degree of °F.

For the application of Eq. (6), a lower limit of 0 °C (32 °F) should be enforced for T_{mains} regardless of the local weather conditions for calculation (such lower limit will be higher in reality). The ratio and lag factors in Eq. (6) reflect the practice of burying water pipes deeper in colder climates.

Fig. 5 shows the estimated daily mains temperature in New York, NY. The maximum temperature is 22.6 °C in July. The annual mean mains-water temperature is 15.8 °C.

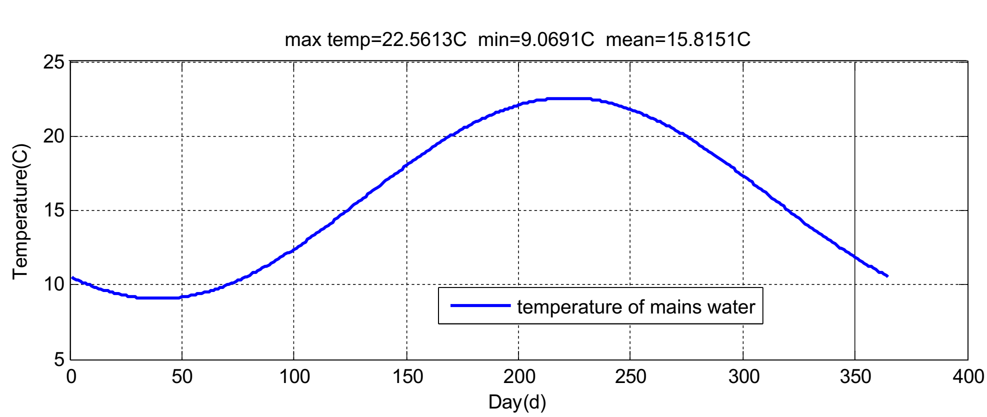

Fig. 5. Daily mains-water temperatures in New York, NY.

In New York, the daily mains-water temperature is ranging from 9 °C to 22.6 °C. The mixture of gray water temperature should vary from mains-water temperature to the gray water temperature from indoor. In cooling mode, the mains-water temperature normally is the highest in the year. To study the impact of gray water temperature, an interval 10 °F (5.56 °C) is chosen. Lower bound energy saving has been estimated based on an assumption of using 23.89 °C, 29.4 °C and 35 °C, respectively, as the gray water tank temperature. Heat pump system with all heat sink combinations are tested at outdoor temperature 35 °C in space cooling only mode as shown in Table 4.

Table 4 shows various operating conditions of the multi-functional heat pump system in space cooling only mode with different gray water tank temperatures. The experiments of three heat sink combinations are implemented at the three different outdoor temperatures. The temperatures of the indoor room and gray water tank maintain the same, which is 26.7 °C and 35 °C respectively. A total of nine experiments have been made (i.e., three conditions and three combinations).

As shown in Fig. 6, the water temperature of gray water tank has limited impaction on the cooling capacity for parallel heat sink combination. However, the *COP* of the system at gray water temperature 23.9 °C is higher than that at 35 °C and 29.4 °C. When gray water temperature is 29.4 °C, the *COP* increase 1.64% and cooling capacity decrease 0.33% compared with the *COP* and cooling capacity at gray water temperature 35 °C. When the gray water temperature is 23.9 °C, the *COP* increase 9.39% and cooling capacity decrease 0.16% compared with the *COP* and cooling capacity at gray water temperature 35 °C. Fig. 7 shows increase of the cooling capacity and *COP* with series heat sink combination as the gray water temperature decreases. When gray water temperature is 29.4 °C, the *COP* slightly decrease 0.45% and cooling capacity increase 2.54% compared with those at gray water temperature 35 °C. When the gray water temperature is 23.9 °C, the *COP* increase 6.58% and cooling capacity increase 7.15% compared with those at gray water temperature 35 °C. Fig. 8 shows the cooling capacity and *COP* with water sink only increase with the gray water temperature decline. When gray water temperature is 29.4 °C, the *COP* increase 27.66% and cooling capacity increase 6.46% compared with those at gray water temperature 35 °C. When the gray water temperature is 23.9 °C, the *COP* increase 23.4% and cooling capacity increase 10.76% compared with those at gray water temperature 35 °C. The change of *COP* and cooling capacity is notable when gray water temperature falls below 35 °C. The change of gray water temperature has the significant impaction on the system performance of series water heat sink and water sink only; however, it has limited impaction on parallel heat sink combination.

Table 4
Impaction of water temperature experiment conditions.

Water tank temperature (°C)			Indoor air temperature (°C)	Outdoor air temperature (°C)
35	29.4	23.9	26.7	35

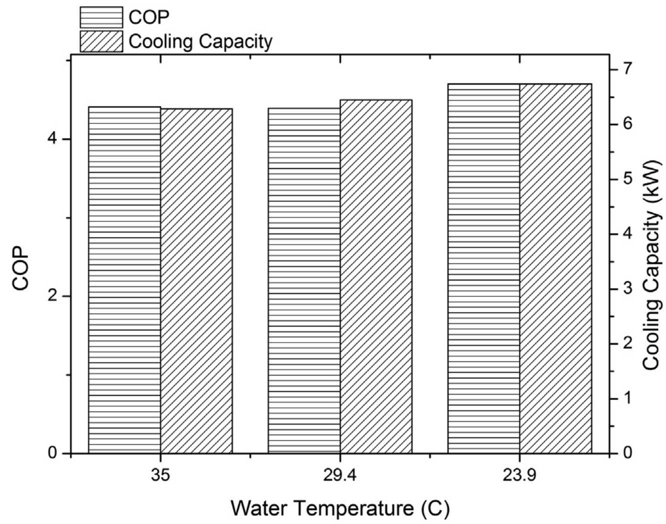

Fig. 7. *COP* and cooling capacity with the series heat sink combination.

In Fig. 8, when the gray water temperature decreases from 29.4 °C to 23.9 °C, the cooling capacity increases 4.04%, which is not significant. When the water temperature decreases, the differential pressure of evaporator and condenser become smaller, as shown in Table 5 below. The low water temperature will be beneficial for heat rejection but limits the refrigerant flow passing through the capillary tube, which may limit the improvement of the cooling capacity.

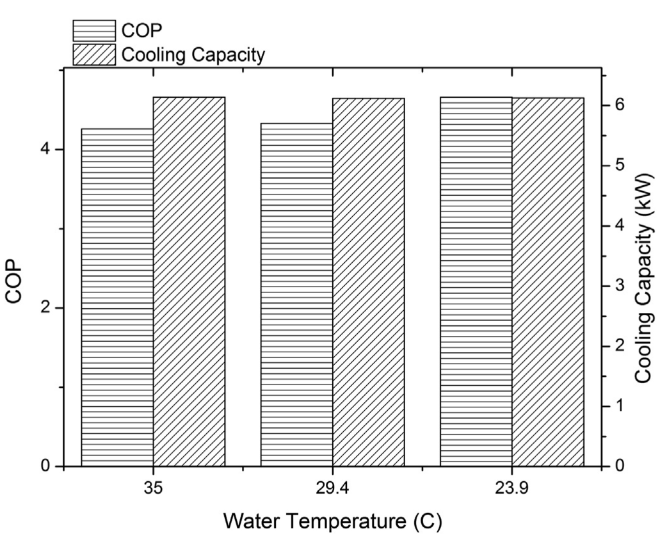

Fig. 6. *COP* and cooling capacity with the parallel heat sink combination.

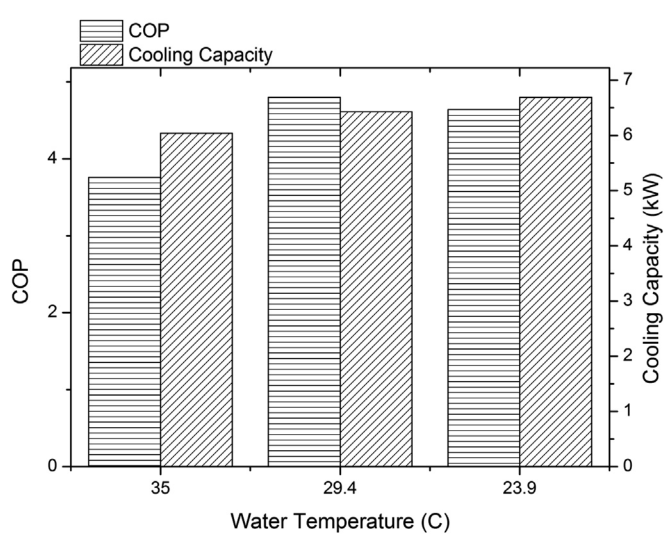

Fig. 8. *COP* and cooling capacity with the water sink only.

Table 5
Evaporator pressure and condenser pressure at water sink only.

Gray water temperature (°C)	Evaporator pressure (bar)	Condenser pressure (bar)
35	10.58	24.01
29.4	9.98	21.07
23.9	10.02	19.86

3.4. *Space cooling with hot-water supply mode*

Because hot water is heated by utilizing the heat of the refrigerant discharged from the compressor, so low discharge temperature will lower the heating capacity of supplying hot water. To satisfy the need for supplying hot water at 49 °C, the discharge temperature should higher than 49 °C. Figs. 9–11 show the fluctuation of the discharge temperature of compressor at each heat sink combinations at different outdoor temperatures in selected two minutes period. When outdoor temperature is 40.6 °C and 35 °C, only the discharge temperature of compressor with air heat sink only is higher than 49 °C. When outdoor temperature is 29.4 °C, the discharge temperature of compressor at all heat sink combinations is below 49 °C. Based on the discharge temperature distribution, there are two operation strategies for supplying hot water while provide space cooling. One is that air-to-refrigerant heat exchanger and the plate heat exchanger supplying hot water work as condensers and air-to-refrigerant heat exchanger is in force convection. Another is that air-to-refrigerant heat exchanger and the plate heat exchanger supplying hot water work as condensers

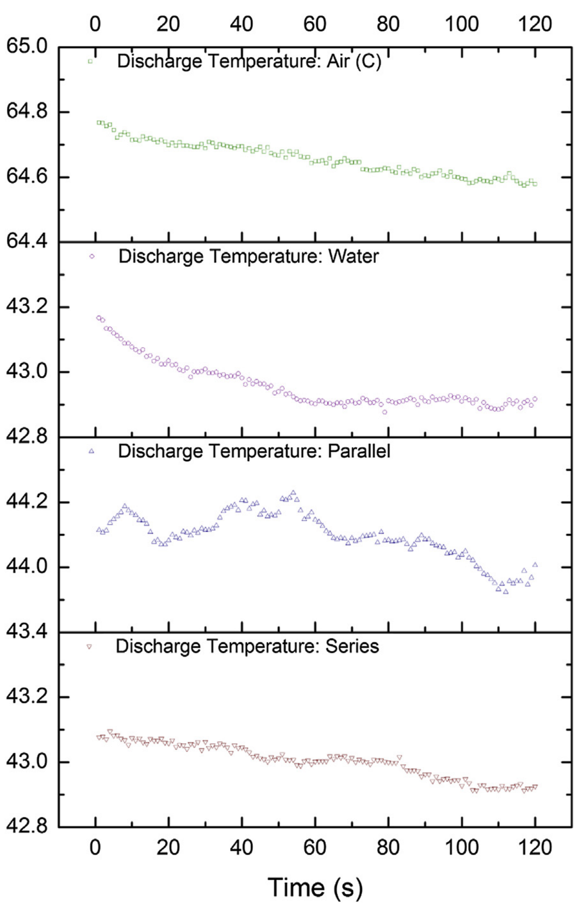

Fig. 9. Discharge temperature at outdoor temperature 40.6 °C.

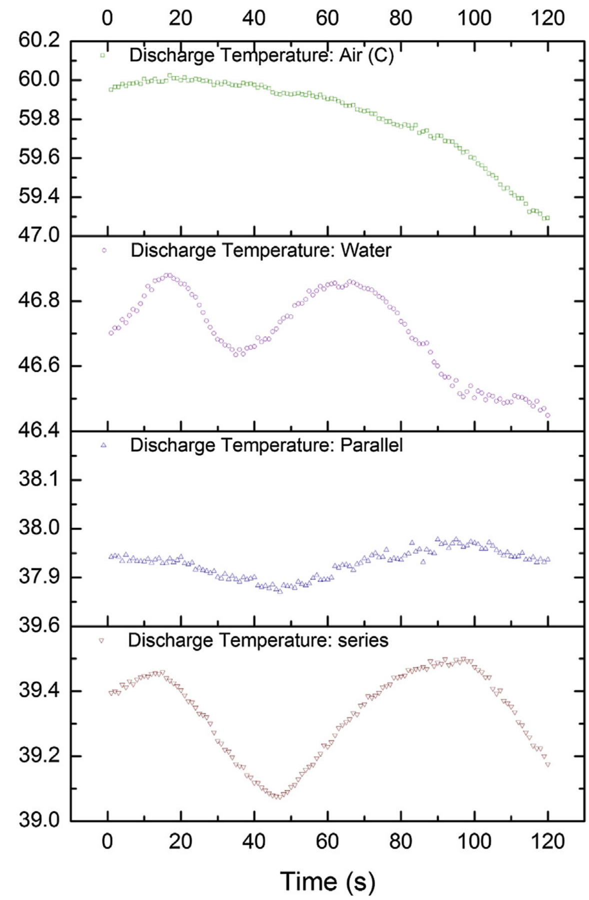

Fig. 10. Discharge temperature at outdoor temperature 35 °C.

and air-to-refrigerant heat exchanger is in nature convection. This sub-section displays the system performance at maintaining hot water at 48.9 °C.

(a) Sub-mode I

In this mode, the plate heat exchanger and air-to-refrigerant heat exchanger work as condensers, while air-to-refrigerant heat exchanger is in force convection condition. According to Figs. 9–11, this model only can supply space cooling with hot water when outdoor air temperature is higher than 29.4 °C. Based on this situation, only two experiments are implemented. Table 6 shows the test operation conditions.

Fig. 12 illustrates *COP*s of the heat pump system at sub-mode I at two outdoor temperatures. The *COP* for space cooling calculated based on the cooling capacity for only space cooling part while the total *COP* is calculated based on the cooling capacity for only space cooling part and that for hot-water supply part. As outdoor temperature decreases from 40.6 °C to 35 °C, the *COP* for space cooling increases 25.89% and total *COP* increases 15.38%. Compared with space cooling only mode shown in Figs. 2 and 3, the total *COP* of space cooling plus hot-water supply mode is greatly improved by 24.38% and 13.97% at outdoor air temperature 40.6 °C and 35 °C, respectively, while the *COP* for space cooling is also improved by 3.7% and 3.68% respectively. Fig. 13 shows the cooling capacity and heating capacity for hot water of the heat pump system at sub-mode I at two outdoor temperatures. The cooling capacity for space cooling increases 17.68% and total cooling capacity increases 7.8% as outdoor air temperature decreases from 40.6 °C to 35 °C. Compared with Figs. 2 and 3, the total cooling capacity of space cooling plus

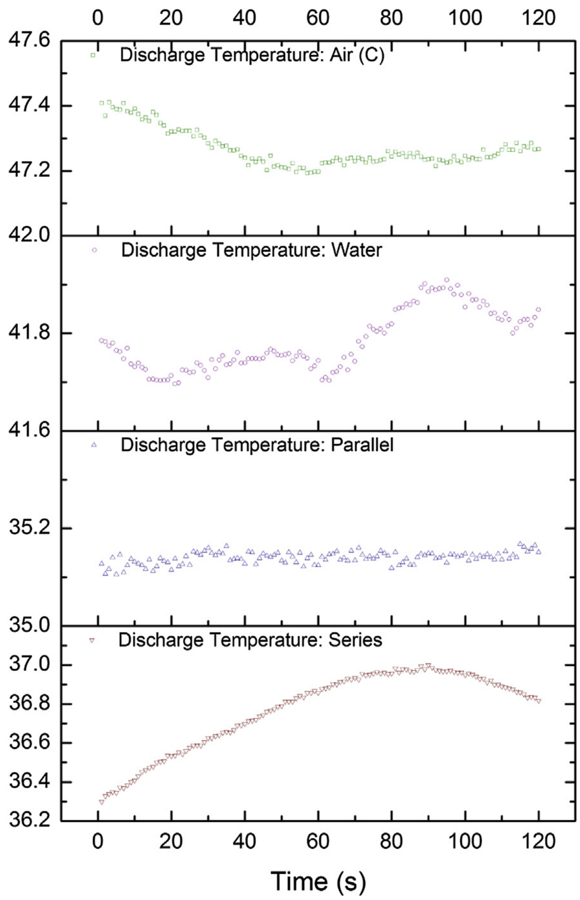

Fig. 11. Discharge temperature at outdoor temperature 29.4 °C.

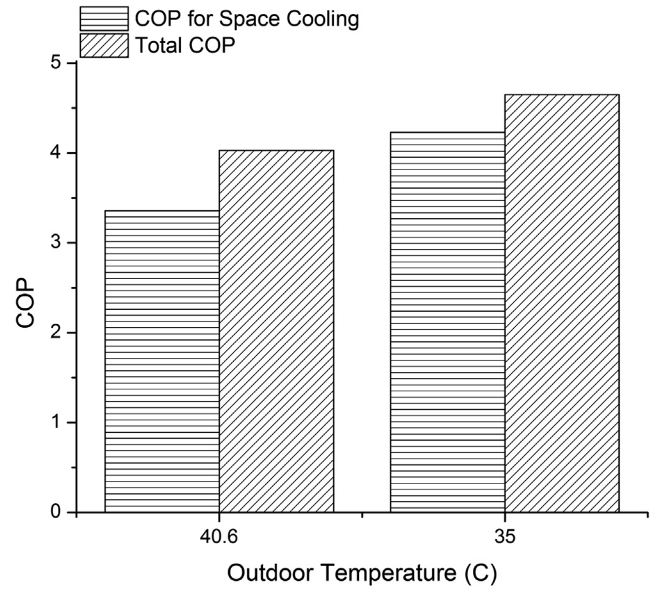

Fig. 12. *COP* of sub-mode I.

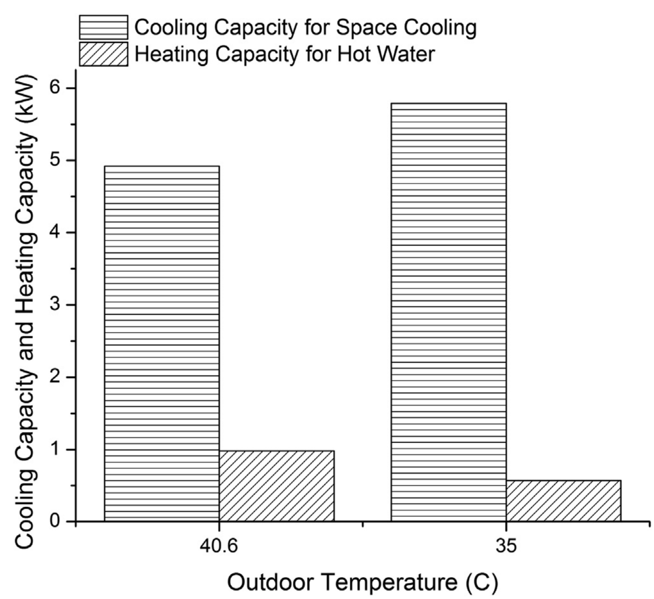

Fig. 13. Cooling capacity and heating capacity for hot water with sub-mode I.

hot-water supply mode is greatly improved by 18.24% and 12.77% at outdoor air temperature 40.6 °C and 35 °C, respectively, while the cooling capacity for space cooling is also improved by 8.61% and 2.66%, respectively. From the comparison, supplying hot water while provide space cooling will improve the system performance especially when outdoor air temperature is high.

(b) Sub-mode II

In this mode, the plate heat exchanger and air-to-refrigerant heat exchanger work as condensers, while air-to-refrigerant heat exchanger is in natural convection condition. Providing space cooling and hot water simultaneously can be realized across all outdoor air temperatures. Table 7 shows three operation conditions in the experiments.

Fig. 14 illustrates the *COP*s of the heat pump system at sub-mode II at three outdoor temperatures. As outdoor temperature decreases from 40.6 °C to 35 °C, the *COP* for space cooling increases 13.55% and total *COP* increases 26.65%. The *COP* for space cooling and total *COP* increase 37% and 41.07% respectively, as outdoor air temperature drops from 40.6 °C to 29.4 °C. Compared with space cooling only mode shown in Figs. 2 and 3, the total *COP* of space cooling plus hot-water supply mode declines by 1.54%, 0.98% and 10.54% at outdoor air temperature 40.6 °C, 35 °C, and 29.4 °C, respectively, while the *COP* for space cooling also decline by 15.74%, 24.02% and 25.65%, respectively. Fig. 15 shows the cooling capacity and heating capacity for hot water of the heat pump system with sub-mode II at three outdoor temperatures. The cooling capacity for space cooling increases 15.48% and total cooling capacity increases 28.85% as outdoor air temperature decreases from 40.6 °C to 35 °C. When outdoor air

Table 6
Experiment conditions for sub-mode I.

Heat sink combinations	Outdoor air temperature (°C)	Indoor air temperature (°C)	Hot-water tank temperature (°C)
Air	40.6	26.7	48.9
	35	26.7	48.9

Table 7
Experiment conditions for sub-mode II.

Heat sink combinations	Outdoor air temperature (°C)			Indoor air temperature (°C)
Plate heat exchanger	40.6	35	29.4	26.7

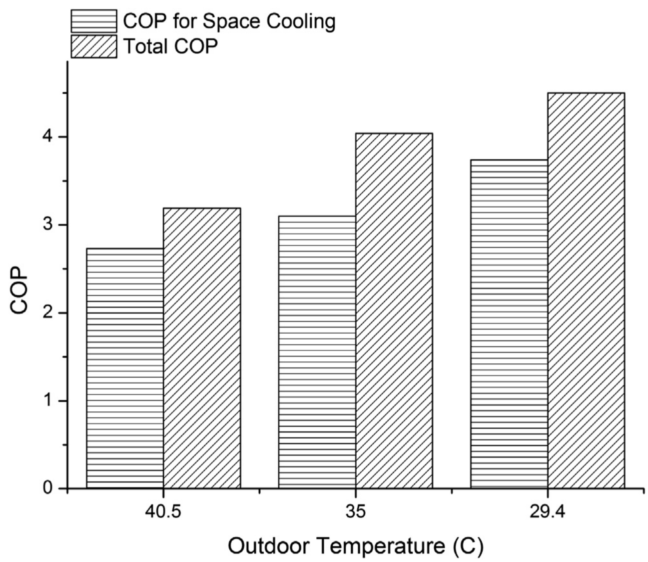

Fig. 14. *COP* of sub-mode II.

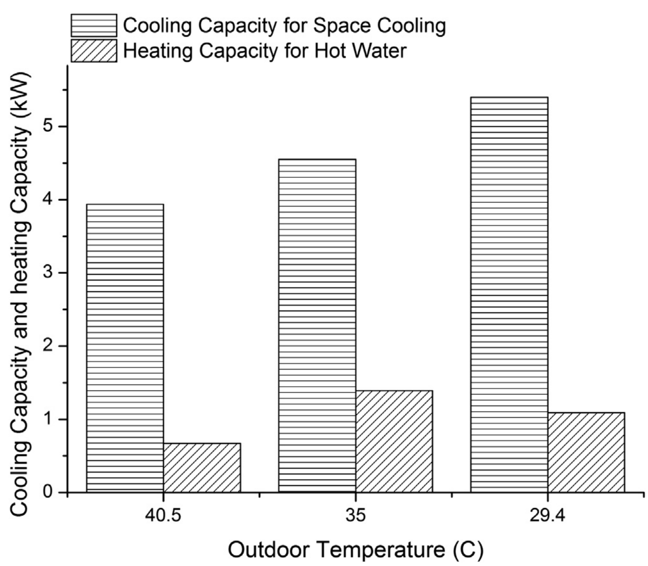

Fig. 15. Cooling capacity and heating capacity for hot water with sub-mode II.

temperature drops from 40.6 °C to 29.4 °C, the cooling capacity for space cooling and total cooling capacity increase 37.06% and 40.78%, respectively. Compared with space cooling only mode shown in Figs. 2 and 3, the total cooling capacity of space cooling plus hot-water supply mode decreases by 7.62% and the cooling capacity for space cooling decreases by 21.04% at outdoor air temperature 40.6 °C. When outdoor air temperature changes to 35 °C and 29.4 °C, the cooling capacity for space cooling decreases 19.33% and 11.48% respectively. But the total cooling capacity increases by 5.32% and 6.39% respectively.

3.5. Dynamic COP and cooling capacity in space cooling plus hot-water supply mode

(a) Sub-mode I

In space cooling plus hot-water supply mode, two experiments are made as shown in Table 8 for an investigation on the impact of water heating dynamically from low temperature to high

Table 8
Dynamic *COP* and cooling capacity experiment conditions for sub-mode I.

Heat sink combinations	Outdoor air temperature (°C)		Indoor air temperature (°C)	Hot-water tank temperature (°C)
Air	40.6	35	26.7	30–48.9

Table 9
Dynamic *COP* and cooling capacity experiment conditions for sub-mode II.

Heat sink combinations	Outdoor air temperature (°C)			Indoor air temperature (°C)	Hot-water tank temperature (°C)
Plate heat exchanger	40.6	35	29.4	26.7	30–48.9

temperature on the performance of heat pump system. This subsection focuses on the dynamic system performance in heating hot water. The standard rated condition is 35 °C according to the AHRI 240. The observations from the results under the outdoor temperature at 40.6 °C are similar. Only the cases at outdoor air temperature 35 °C were shown.

Fig. 16 shows the *COP* for space cooling, total *COP* and cooling capacity for space cooling only and heating capacity for supplying hot water. The hot-water temperature is heated from 30 °C to 49 °C. *COP* for space cooling only slightly decrease with the rise of hot-water temperature (see second bottom figure in Fig. 16). The total *COP* decreases larger (see bottom figure in Fig. 16). The heating capacity for supplying hot water decreases with the rise of hot-water temperature (see second top figure in Fig. 16). When water temperature is approaching 49 °C, the heating capacity for

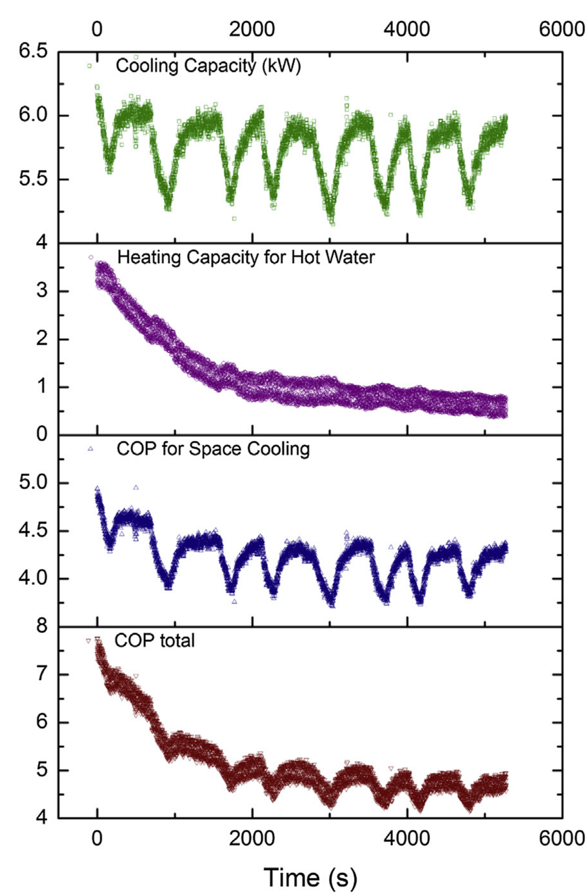

Fig. 16. Dynamic *COP* and cooling capacity at outdoor temperature 35 °C.

Table 10
Evaporation pressure and condense pressure (gauge pressure).

	Air	Water	Parallel	Series
Evaporation pressure (bar)	10.33	9.98	10.05	10.06
Condense pressure (bar)	22.81	21.07	21.46	20.08
Power	1.14	1.09	1.17	1.22

supplying hot water falls below 1 kW. The cooling capacity for space cooling only was fluctuation as shown in the first top figure in Fig. 16 (Table 9).

(b) Sub-mode II

Table 10 shows the experiment conditions for sub-mode II. Total three experiments are implemented. Only the case at outdoor temperature 35 °C was discussed. The observations from the results under the outdoor temperatures at 40.6 °C and 29 °C are similar; therefore the results are not shown here.

Fig. 17 presents the total *COP*, *COP* for space cooling only, cooling capacity for space cooling only and heating capacity for supplying hot water. The *COP* for space cooling only and total *COP* decrease with the increase of hot-water temperature from 30 °C to 49 °C. The change of cooling capacity for space cooling only is small, but the heating capacity for supplying hot water decreases over 80 percent with the increase of hot-water temperature. Compared with Fig. 16, the *COP*s and the cooling capacities for space cooling only are lower than those of sub-mode I. However, the heating capacity for supplying hot water is much larger than that of sub-mode I. The time used to heat hot water to 49 °C is also much less as shown in Figs. 16 and 17.

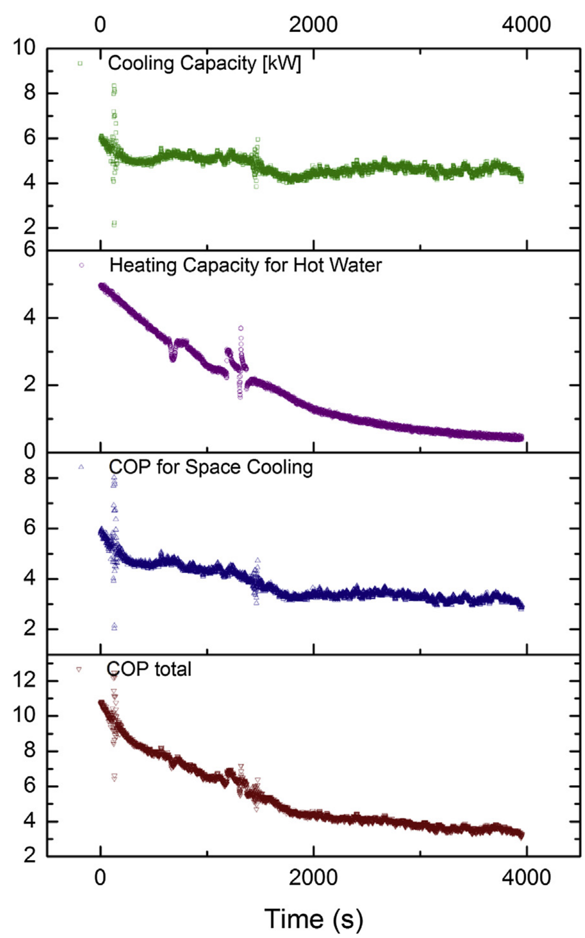

Fig. 17. Dynamic *COP* and cooling capacity at outdoor temperature 35 °C.

4. Discussion

4.1. Low discharge temperature

Figs. 9 and 10 provide the discharge temperatures of the compressor in space cooling only mode. According to these figures, the discharge temperatures of the heat pump system with water sink only, parallel and series heat sink combinations are below 49 °C. The discharge temperature of the heat pump system with air sink only is higher than 49 °C when outdoor air temperature is 40.6 °C and 35 °C. However, when outdoor air temperature is 29.4 °C, the discharge temperature of the heat pump system with air sink only becomes lower than 49 °C.

Figs. 18–20 show the superheat of the evaporator of the heat pump system with all kinds of heat sink combinations at outdoor air temperature 40.6 °C, 35 °C, 29.4 °C, respectively, in the space cooling only mode. The superheats of evaporator of all kinds of heat sink combinations are very close at all outdoor air temperatures being tested. However, compared with conventional superheat in general practice, these values are comparably lower. The superheat

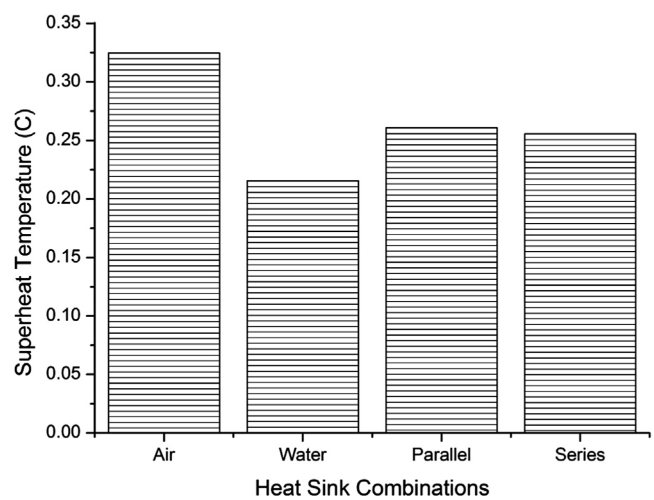

Fig. 18. Superheat of an evaporator at outdoor air temperature 40.6 °C.

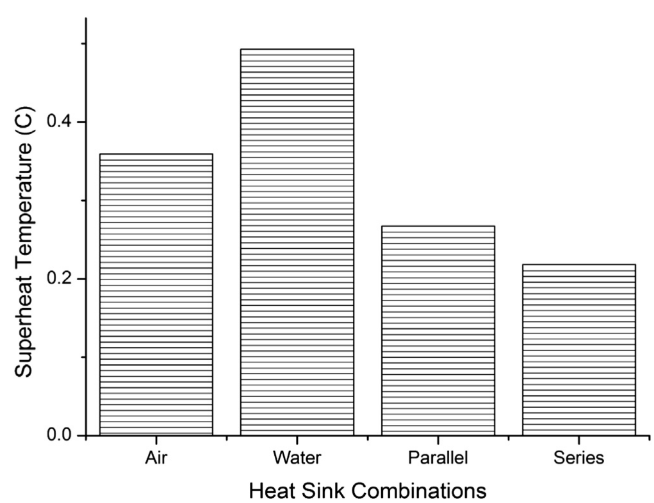

Fig. 19. Superheat of an evaporator at outdoor air temperature 35 °C.

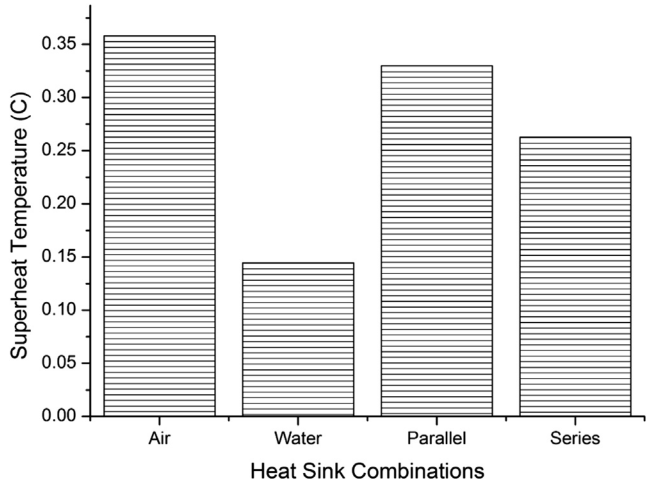

Fig. 20. Superheat of an evaporator at outdoor air temperature 29.4 °C.

controlled by thermal expansion valve is usually around 5 °C. Because the prototype is developed at the conventional heat pump, the superheat of evaporator is controlled by capillary by default. The capillary used for throttling has some merits, such as cheap and stable. However, it has a limitation to control the flow of refrigerant passing through the throttle. In this prototype, due to some modifications, such as adding a heat exchanger and a plate heat exchanger, the amount of refrigerant charge in the system is larger than that in the conventional air-source heat pump, which conducts the flow rate of refrigerant passing through throttle is larger. Therefore, the superheat of evaporator will become small. Small superheat may lower down the inlet temperature of compressor and will reduce discharge temperature of compressor.

Figs. 21–23 show the superheat of compressor inlet of the heat pump system with various heat sink combinations at outdoor air temperature 40.6 °C, 35 °C, 29.4 °C, respectively, in the space cooling only mode. The superheat of the compressor inlet of air sink is much larger than that of other heat sink combinations. In Figs. 21 and 22, the superheat of the compressor inlet with air sink is about 1.25 °C when outdoor air temperature is 40.6 °C and 35 °C. When the outdoor air temperature becomes 29.4 °C, the superheat decreases to around 0.8 °C, although it is still higher than the superheat of other heat sink combinations. The superheat of the compressor inlet with air sink at outdoor air temperature 29.4 °C is

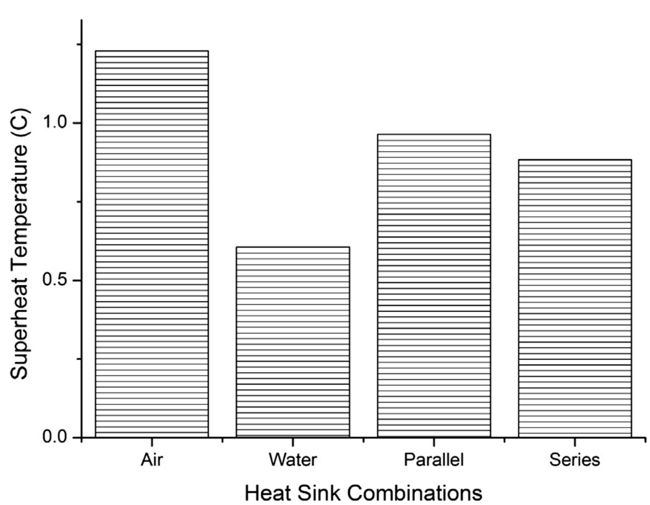

Fig. 21. Superheat of a compressor inlet at outdoor air temperature 40.6 °C.

almost the same as that of other heat sink combinations at outdoor air temperature 40.6 °C and 35 °C. Considering Figs. 9–11, the discharge temperature of compressor has close relationship with the superheat of compressor inlet. Higher superheat and higher inlet temperature will bring higher discharge temperature.

The reason caused the different superheat of compressor inlet of air sink with that of other heat sink combinations is that the heat exchanger in the water tank holds a lot of refrigerant. This amount of refrigerant will not be used when only air-to-refrigerant heat exchanger operates as a condenser. However, when air-to-refrigerant and water-to-refrigerant heat exchangers or only water-to-refrigerant heat exchanger work as condensers, this amount will be fully utilized. When only air-to-refrigerant heat exchanger works as a condenser and outdoor air temperature falls to 29.4 °C, with the decrease of condenser temperature, more refrigerant is utilized and the superheat of compressor inlet decreases, so the discharge temperature of compressor falls below 49 °C.

4.2. The impaction of heat sink combinations on the system performance and the comparison of the system performance of all heat sink combinations with conventional air-source heat pump

In cooling mode, air-source and water-source heat exchangers work as condensers. They make four heat sink combinations as mentioned above. According to Figs. 2–4, the system performance varies with different heat sink combinations. There is only one metering device, like capillary, for throttling and controlling the total refrigerant flowing through the evaporator. The amount refrigerant passing each evaporator is decided by the condensation pressure. The experiments at the outdoor temperature 35 °C in space heating mode are taken as an example, which the system performance is shown in Fig. 2. Series combination has the highest cooling capacity. Water sink only has the highest *COP* and similar cooling capacity as that of series combination, while air sink only has the lowest *COP* and cooling capacity. Table 10 shows the evaporation and condense pressures, as well as their power. The system with combinations of air sink and gray water sink consumes more power than that with single heat source. Also, the power of the system with water sink is less than that of the system with air sink. Water sink only has the lowest average evaporation pressure. Air sink only, parallel and series have similar evaporation pressure. The condenser pressure of air sink only is the highest. The evaporation pressure of water sink only and parallel is similar around 22 bar, while that of series is less than 21 bar. Plot the refrigeration cycle of the system with different heat source combinations in log (P)–h diagram, the series clearly has longer process line in evaporation section, which means more heat is absorbed by refrigerant in unit mass flow rate. The air sink only has the shortest.

The conventional heat pump normally is the air-source heat pump used in residential buildings. Based on the discussion above, the system performance of the multi-functional heat pump with air sink only is the worst. As shown in Fig. 2, the system performances of all heat sink combinations are better than that of the original air-source heat pump tested at the same condition. There is one reason is because the refrigerant filled in the proposed heat pump system is larger than the amount filled in the original air-source heat pump. This is because that more heat exchangers are added and the system becomes more complicated which means the refrigerant amount filled in the system also need to be adjusted. When air is only heat sink, the refrigerant amount passing through evaporator in the multi-functional heat pump system is more than that in conventional air-source heat pump. Moreover, the superheat of the evaporator in multi-functional heat pump system with air sink only is smaller than normal condition, which results in more refrigerant amount passing through the evaporator.

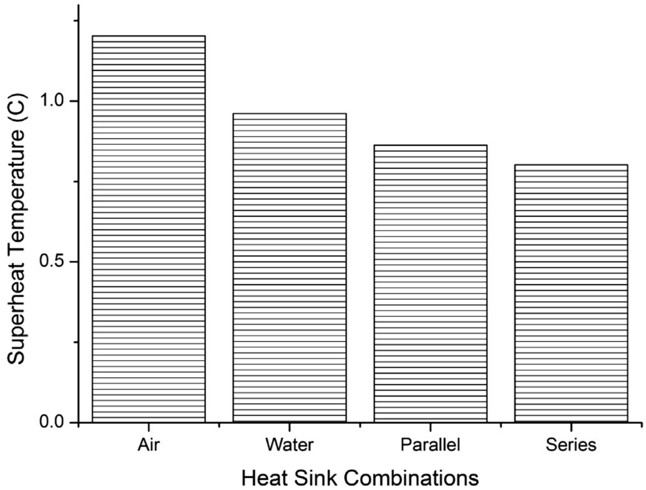

Fig. 22. Superheat of a compressor inlet at outdoor air temperature 35 °C.

4.3. The cooling capacity comparison in two sub-modes

Comparing Figs. 2 and 13, the cooling capacity for space cooling and total cooling capacity in sub-mode I is larger than the cooling capacity with air sink in space cooling only mode. This is because the plate heat exchanger increases the condense area and the lower hot-water temperature reduces the condense temperature. Taking the outdoor air temperature 35 °C as an example, Fig. 24 gives the sub-cooling of condenser in sub-mode I, sub-mode II and space cooling mode. The sub-cooling of condenser in sub-mode I is higher than that in space cooling only mode. Because the temperature of condenser in sub-mode I is much lower than that in sub-mode II, the sub-cooling of condenser in sub-mode I is less than 1 °C lower than that in sub-mode II. The refrigerant in sub-mode I has more capacity for absorbing heat than that in space cooling only mode.

Comparing Figs. 2 and 15, the cooling capacity for space cooling and total cooling capacity in sub-mode II are smaller than the cooling capacity of air sink in space cooling only mode. Although the plate heat exchanger increase the condense area, but the condenser temperature is much higher than space cooling mode which is about 48 °C. When the hot water becomes 30 °C, the cooling capacity for space cooling and total cooling capacity in sub-mode II are larger than those in space cooling only mode.

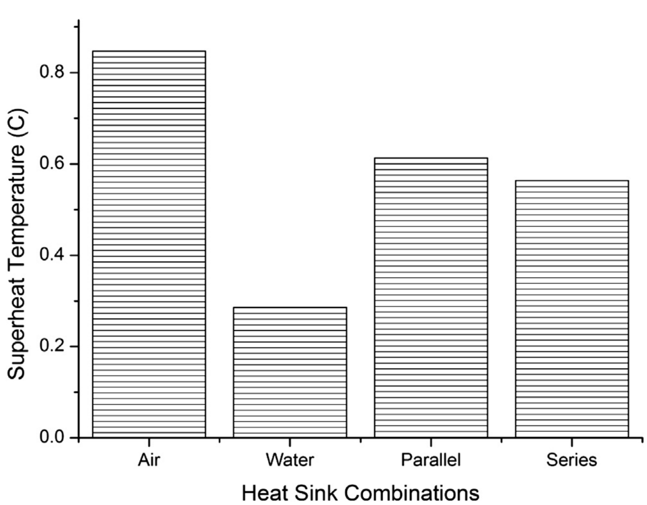

Fig. 23. Superheat of a compressor inlet at outdoor air temperature 29.4 °C.

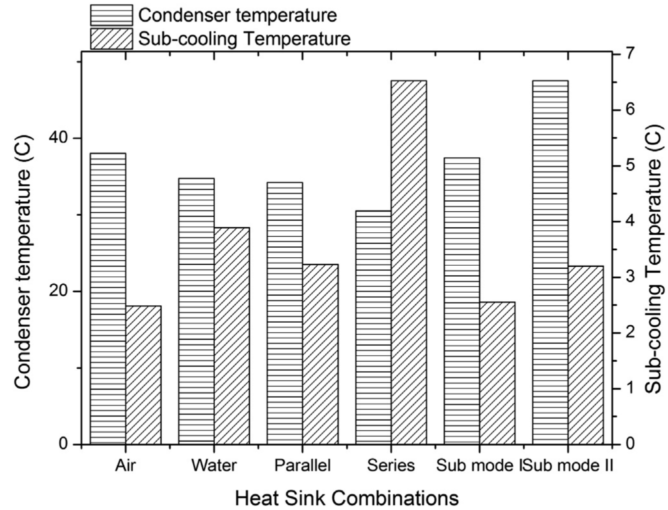

Fig. 24. Condenser temperature and sub-cooling at outdoor air temperature 35 °C.

5. Conclusion

In this study, the multi-functional heat pump system shows superior performance in cooling mode compared with the conventional air-source heat pump system. The system with air and water heat sinks in series has the best performance in space cooling mode at outdoor air temperature of 35 °C and 29.4 °C and the second best performance at outdoor air temperature of 40.6 °C, compared with the system with other heat sink combinations. The performance of the innovative system with all types of heat sink combinations decreases as the outdoor temperature increases in space cooling mode and space cooling plus hot-water supply mode. Due to the discharge temperature of compressor distribution, space cooling plus hot-water supply mode is divided into two sub-modes, which are the heat exchanger located in outdoor environment is in force convection and natural convection, respectively. When outdoor air temperature is 35 °C, sub-mode I provides 2.66% and 12.77% more cooling capacity for space cooling only and total cooling capacity, respectively, compared with the cooling capacity of air sink only in the space cooling only mode. Also the *COP* for space cooling only and total space cooling of the system in sub-mode I are 3.68% and 13.97%, respectively, higher than the *COP* in space cooling mode. When the outdoor air temperature increases, sub-mode I has better *COP*s and cooling capacities than those of space cooling only mode. However, sub-mode I only has benefit at outdoor air temperature 40.6 °C and 35 °C. The sub-mode II is not restrained by the outdoor air temperature and can provide more capacity for supplying hot water than that of sub-mode I. In sub-modes I and II, the *COP* for space cooling and total *COP* decrease as the hot-water temperature increases. The capacity for supplying hot water decreases with the increase of hot-water temperature. The process of heating hot water from 30 °C to 48.9 °C usually takes less time for sub-mode II than that for sub-mode I. Gray water temperature has limited impacts on parallel heat sink combinations and has significant impacts on series heat sink combination and water sink only. The lower bound estimated performance of the heat pump system is better than the conventional system.

Acknowledgements

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